Selected Topics in Oil & Gas Process Design for Graduate Chemical Engineers

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First Edition

Foreword

Much has changed in the last three decades with the advent of engineering softwares, causing a paradigm shift in the way Chemical Process Design calculations are performed. The 'Art of Hand Calculations' is becoming a dying art with computers & engineering software providing solutions. But this can often turn into a crippling pitfall for Chemical Engineers who lack any basic understanding of Engineering Fundamentals or Industrial Design Standards when operating an engineering software. Therefore, it is imperative that chemical engineers recognize the necessity to focus and gain proficiency in applying process design fundamentals prior to embarking on any engineering undertaking.

While the academia can sometimes lack focus on meeting Industry standards, engineering graduates are often left behind from picking up the requisite skills as demanded by Process Design Industry. With an aim to bridge some of the gaps between academia and the industry, the following free tutorial book aims to provide a practical approach and guide young chemical engineers in areas of Process Engineering Design through a series of learning modules with an emphasis on Design Standards.

The author encourages readers to explore the various topics presented and work them out. No complex engineering software is required for most of the tutorials, except for MS-Excel or even a paper, pen and a calculator to work out by hand will do just fine. With each topic progressed with dedication, engineering practioners can expect to improve their proficiency levels in some of the areas of Upstream Oil & Gas.

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With Best Wishes
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Contents

Module 1

Chemical Process Calculations

Often engineers are tasked with communicating equipment specifications with suppliers, where process data needs to be exchanged for engineering quotations & orders. Any dearth of data would need to be computed for which process related queries are sometimes sent back to the process engineer's desk for the requested data.

The following module is a refresher for non-process engineers such as project engineers, Piping, Instrumentation, Static & Rotating Equipment engineers to conduct basic process calculations related to estimation of mass %, volume %, mass flow, actual & standard volumetric flow, gas density, parts per million (ppm) by weight & by volume.

Problem Statement

A vendor requests the project engineer to provide certain natural gas process data for evaluation. The gas composition is as follows,

Table 1. Natural Gas Composition & Properties

Component	MW	Mol%
-	kg/kmol	%
Methane	16.04	76.23
Ethane	30.07	10.00
Propane	44.01	5.00
i-Butane	58.12	1.00
n-Butane	58.12	1.00
i-Pentane	72.15	0.30
n-Pentane	72.15	0.10
n-Hexane	86.18	0.05
H_2O	18.02	0.25
CO_2	44.01	3.00
H_2S	34.08	0.07
N_2	28.01	3.00

The process conditions are 40 bara, 50^{0}C & 1,000 kmol/h of natural gas. The process data requested by the vendor is as follows,

1. Natural Gas Molecular Weight & Density
2. Component & Total Mass flow
3. Component & Total Actual Volume flow
4. Component & Total Standard Volume flow
5. Component mass %
6. Component Volume %
7. Component Parts per million (ppm) by weight.
8. Component Parts per million (ppm) by volume.

Component Molar Flow [M]

To estimate the component molar flow, the mixture molecular weight [MW] is evaluated first by using Kay's mixing rule as follows,

$$MW = \sum y_i MW_i, \text{ Where, } i = 1 \text{ to } n \quad (1)$$

Where,

y_i = Mole fraction of each component, -

MW_i = Component MW, kg/kmol

The component molar flow rate is calculated as,

$$M_i = y_i \times M, \text{ Where, } i = 1 \text{ to } n \quad (2)$$

Where,

M_i = Component Molar Flow, kmol/h

M = Total Molar Flow, kmol/h

Component & Total Mass Flow

To estimate the component mass flow [m_i] & total mass flow [m], the relationships are,

$$m_i = M_i \times MW_i, \text{ Where, } i = 1 \text{ to } n \quad (3)$$

$$m = \sum m_i, \text{ Where, } i = 1 \text{ to } n \quad (4)$$

Where,

m_i = Component Mass Flow, kg/h

m = Total Mass Flow, kg/h

Component & Total Volume Flow

To estimate the component volume flow & total volume flow, the relationship is based on the principle that 1 kmol of ideal gas occupies 22.414 m^3 at 0^0C [273.15 K]. In order to estimate the volume flow of each component, the volume occupied by a gas at standard pressure & temperature [STP], i.e., 1 atm & 15^0C, the relationship is corrected to,

$$Q_i = M_i \times \frac{22.414}{273.15} \times [T[°C] + 273.15] \qquad (5)$$

The total volume flow rate is,

$$Q_{std} = \sum Q_i, \textit{ Where, i = 1 to n} \qquad (6)$$

Where,

Q_i = Component Volume Flow [Sm^3/h]

Q = Total Volume Flow [Sm^3/h]

Component Mass %

The component mass % is calculated as,

$$m_i\% = \frac{m_i}{m}, \textit{ Where, i = 1 to n} \qquad (7)$$

Component Volume %

The component volume % is calculated as,

$$Q_i\% = \frac{Q_i}{Q_{std}}, \textit{ Where, i = 1 to n} \qquad (8)$$

Actual Volumetric Flow Rate [Q_{act}]

The actual volumetric flow is computed as,

$$Q_{act} = \frac{m}{\rho_{NG}} \qquad (9)$$

Where, the density of the natural gas [ρ_{NG}] is computed from the expression that takes into account the gas compressibility factor, Z as,

$$\rho_{NG} = \frac{P_{act} \times MW}{Z_{act} \times R \times T_{act}}\ kg/m^3 \qquad (10)$$

R = 0.0831447 m^3.bar/kmol.K

The gas compressibility factor, Z of natural gas can be computed based on the DAK Equation of State [EOS] as described in Appendix A. Alternatively the Standard Volumetric Flow rate can be computed as,

$$Q_{std} = \sum Q_i = \left[\frac{P_{act} \times Q_{act}}{Z_{act} \times T_{act}}\right] \times \left[\frac{Z_{std} \times T_{std}}{P_{std}}\right] Sm^3/h \qquad (11)$$

Where, Z_{std} is taken to be 1.0

Component PPM by Weight, *ppm(w)*

The component PPM by weight, *ppm(w)* is computed as,

$$[wt\ \%]_i \times 10{,}000 = [ppm(w)]_i \qquad (12)$$

Component PPM by Volume, *ppm(v)*

The component PPM by volume, *ppm(v)* is computed as,

$$[vol\ \%]_i \times 10{,}000 = [ppm(v)]_i \qquad (13)$$

Results

Based on the steps provided, the estimated results of mass %, volume %, mass flow, actual & standard volumetric flow rates, parts per million (ppm) by weight & by volume is shown in Appendix B & Appendix C.

Appendix A: Gas Compressibility Factor, Z for Natural Gas Estimation

To assess the properties of natural gas, calculations can be begun by estimating the properties using Kay's Mixing Rule as follows,

Mixture molecular weight [MW], kg/kmol

$$MW = \sum y_i\, MW_i \qquad (14)$$

Mixture Pseudo Critical Pressure [P_c], psia

$$P_c = \sum y_i\, P_{c,i} \qquad (15)$$

Mixture Pseudo Critical Temperature [T_c], ^0R

$$T_c = \sum y_i\, T_{c,i} \qquad (16)$$

Gas Specific Gravity [γ_g], [-]

$$\gamma_g = \frac{MW_g}{MW_{air}};\ MW_{air} = 28.96\ kg/kmol \qquad (17)$$

From the above, Kay's Mixing Rule does not give accurate pseudocritical properties for higher molecular weight mixtures (particularly C_{7+} mixtures) of hydrocarbon gases when estimating gas compressibility factors [Z] and deviations can be as high as 15%. Therefore, to account for these differences, Sutton's correlations based on gas specific gravity can be utilized as follows,

$$P_{pc} = 756.8 - 131.07\gamma_g - 3.6\gamma_g^2 \qquad (18)$$

$$T_{pc} = 169.2 - 349.5\gamma_g - 74.0\gamma_g^2 \qquad (19)$$

The above equations are valid for the gas specific gravities range of $0.57 < \gamma_g < 1.68$. Using the Sutton correlations, the reduced properties are calculated as,

$$P_r = \frac{P}{P_{pc}} \quad (20)$$

$$T_r = \frac{T}{T_{pc}} \quad (21)$$

However the pseudocritical properties are not the actual mixture critical temperature and pressure but represent the values that must be used for the purpose of comparing corresponding states of different gases on the Z-chart, as shown below in the Standing & Katz, 1959 chart for natural gases.

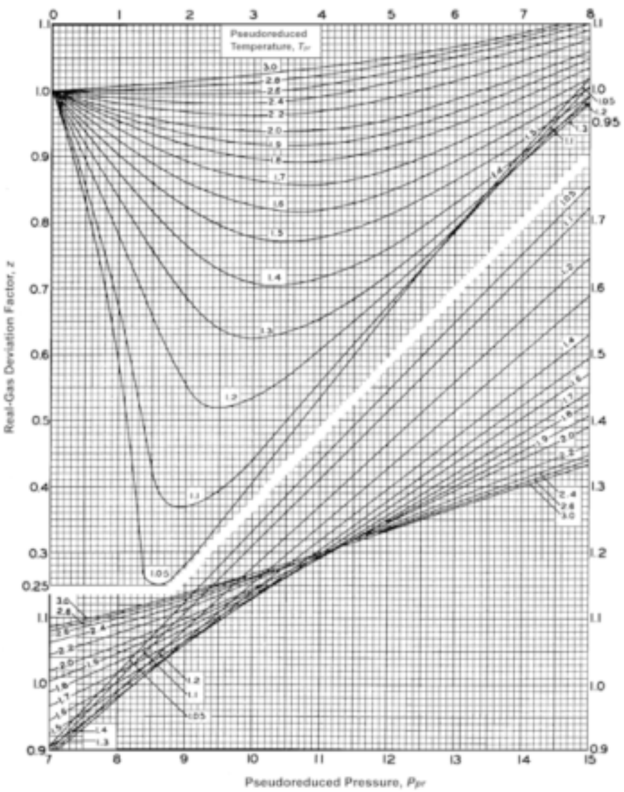

Figure 1. Natural Gas deviation factor chart (Standing & Katz, 1959)

Due to the graphical method of Standing & Katz chart, the Z factor can be estimated using Dranchuk and Abou-Kassem Equation of State [DAK-EoS] which is based on the data of Standing & Katz, 1959 and is expressed as,

$$Z = 1 + \left[A_1 + \frac{A_2}{T_r} + \frac{A_3}{T_r^3} + \frac{A_4}{T_r^4} + \frac{A_5}{T_r^5}\right]\rho_r + \left[A_6 + \frac{A_7}{T_r} + \frac{A_8}{T_r^2}\right]\rho_r^2 - A_9\left[\frac{A_7}{T_r} + \frac{A_8}{T_r^2}\right]\rho_r^5 + +A_{10}(1 + A_{11}\rho_r^2)\left(\frac{\rho_r^2}{T_r^3}\right)e^{-A_{11}\rho_r^2} \quad (22)$$

Where,

$$\rho_r = \frac{0.27P_r}{ZT_r} \quad (23)$$

ρ_r = Pseudo-Reduced Density [-]

T_r = Pseudo-Reduced Temperature [-]

The constants A_1 to A_{11}, are as follows,

Table 2. DAK EoS A_1 to A_{11} Constants

A_1	0.3265	A_7	−0.7361
A_2	−1.0700	A_8	0.1844
A_3	−0.5339	A_9	0.1056
A_4	0.01569	A_{10}	0.6134
A_5	−0.05165	A_{11}	0.7210
A_6	0.5475		

DAK-EoS has an average absolute error of 0.486% in its equation, with a standard deviation of 0.00747 over ranges of pseudo-reduced pressure and temperature of $0.2 < P_{pr} < 30$; $1.0 < T_{pr} < 3.0$ and for $P_{pr} < 1.0$ with $0.7 < T_{pr} < 1.0$. However DAK-EoS gives unacceptable results near the critical temperature for $T_{pr} = 1.0$ and $P_{pr} > 1.0$, and DAK EoS is not recommended in this range.

DAK EoS for NG Mixtures with Acid Gases

Natural Gas is expected to contain acid gas fractions, such as CO_2 and H_2S, & applying the Standing & Katz Z-factor chart & Sutton's pseudocritical properties calculation methods would yield inaccuracies, since they are only valid for hydrocarbon mixtures. To account for these inaccuracies, the Wichert & Aziz correlations can be applied to mixtures containing CO_2 < 54.4 mol% & H_2S < 73.8 mol% by estimating a deviation parameter [ε], which is used to modify the pseudocritical pressure & temperatures. The deviation parameter [ε] whose units are in ^{0}R, and psia,

$$\varepsilon = 120[A^{0.9} - A^{1.6}] + 15[B^{0.5} - B^4] \quad (24)$$

Where,

$A = Y_{CO2} + Y_{H2S}$ in Gas mix [Y = mole fraction]

$B = Y_{H2S}$ in Gas mixture [Y = mole fraction]

Applying [ε], the modified pseudocritical pressure & temperature is,

$$T'_{pc} = T_{pc} - \varepsilon \quad (25)$$

$$P'_{pc} = \frac{P_{pc}\,T'_{pc}}{T_{pc} - B[1-B]\varepsilon} \quad (26)$$

Where, T'_{pc} & P'_{pc} are valid only in ^{0}R and psia.

Based on the calculated modified pseudocritical pressure [P'_{pc}] and temperature [T'_{pc}], the pseudo-reduced pressure [P_r] & temperature [T_r] is,

$$P_{pr} = \frac{P\,[psia]}{P'_{pc}[psia]} \quad (27)$$

$$T_{pr} = \frac{T\,[°R]}{T'_{pc}[°R]} \quad (28)$$

$$\rho_{pr} = \frac{0.27P_{pr}}{ZT_{pr}} \quad (29)$$

Using the calculated values of Pp_r Tp_r & ρ_{pr}, compressibility factor, Z is determined by using DAK EoS. Owing to the value of 'Z' being an implicit parameter in calculating ρ_{pr} as well as in DAK-EoS, an iterative approach, whereby Z value is guessed & iteratively solved to satisfy both modified pseudo-reduced density [ρ_{pr}] & DAK EoS.

References & Further Reading

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7. https://petrowiki.org/Real_gases

Appendix B: Natural Gas Composition Results

Natural Gas Composition at 1.01325 bara, 15 deg.C								
Component	Mol%	MW [MW_i]	y_iMW_i	Molar Flow	Mass Flow	Vol. Flow	Mass %	Vol %
	[%]	[kg/kmol]	[kg/kmol]	[kmol/h]	[kg/h]	[Sm^3/h]	[%]	[%]
Methane	76.23	16.04	12.23	762.3	12,227.3	18,024.5	57.7891	76.2300
Ethane	10.00	30.07	3.01	100.0	3,007.0	2,364.5	14.2118	10.0000
Propane	5.00	44.01	2.20	50.0	2,200.5	1,182.2	10.4001	5.0000
i-Butane	1.00	58.12	0.58	10.0	581.2	236.4	2.7469	1.0000
n-Butane	1.00	58.12	0.58	10.0	581.2	236.4	2.7469	1.0000
i-Pentane	0.30	72.15	0.22	3.0	216.5	70.9	1.0230	0.3000
n-Pentane	0.10	72.15	0.07	1.0	72.2	23.6	0.3410	0.1000
n-Hexane	0.05	86.18	0.04	0.5	43.1	11.8	0.2037	0.0500
C_7+	0.00	119.00	0.00	0.0	0.0	0.0	0.0000	0.0000
H_2O	0.25	18.02	0.05	2.5	45.0	59.1	0.2129	0.2500
CO_2	3.00	44.01	1.32	30.0	1,320.3	709.3	6.2401	3.0000
H_2S	0.07	34.08	0.02	0.7	23.9	16.6	0.1127	0.0700
N_2	3.00	28.01	0.84	30.0	840.4	709.3	3.9719	3.0000
Total	**100.00**	**MW**	**21.16**		**21,158**	**23,645**	**100.0**	**100.0**

Appendix C: Natural Gas Process Data

Natural Gas Conditions		
Parameter	**Value**	**Units**
Standard Pressure	1.01325	bara
Standard Temperature	15.0	^{0}C
Flowing Pressure	40.0	bara
Flowing Temperature	50.0	^{0}C
Molar Flow Rate	1,000	kmol/h
Mass Flow Rate	21,158	kg/h
Actual Volumetric Flow Rate	617.0	Am^3/h
Standard Volumetric Flow Rate	23,645	Sm^3/h
	20.04	MMscfd
Parts Per Million [PPM]		
Component	H2S	-
Wt % of H2S	0.1127	%
ppm (W) of H2S	1,127	ppm (W)
Vol % of H2S	0.0700	%
ppm (V) of H2S	700	ppm (V)

Critical Properties - Sutton Correlation with Wichert & Aziz Correction		
Parameter	**Value**	**Unit**
Flowing Pressure [P]	40.0	bara
Flowing Temperature [T]	50.0	^{0}C
Gas Specific Gravity [γ_g]	0.7305	-
Pseudocritical Pressure [P_{pc}]	659.1	psia
Pseudocritical Temperature [T_{pc}]	385.0	^{0}R
Deviation Factor [ε]	5.1605	^{0}R
Modified Pseudocrotical Pressure [P'_{pc}]	650.3	psia
Modified Pseudocrotical Temperature [T'_{pc}]	379.9	^{0}R
Modified Reduced Pressure [P_{pr}]	0.9042	-
Modified Reduced Temperature [T_{pr}]	1.5312	-
Modified Reduced Density [ρ_{pr}]	0.1736	-
DAK EOS Convergence	*0.0000*	Calculate Z
Compressibility Factor [Z]	**0.9186**	**-**
Natural Gas Density [ρ]	**34.29**	**kg/m^3**
Natural Gas Viscosity [μ]	**0.0123**	**cP**

Module 2

OPERATING ENVELOPES FOR CENTRIFUGAL PUMPS

In today's global economy, a necessity exists in process facilities to reduce costs and upgrading component specifications do not necessarily provide fail proof solutions. Of all the equipment used for both commercial and industrial applications, centrifugal pumps are a common feature. They are characterized by their high efficiency with low power consumption.

To cite a few applications, centrifugal pumps are used in buildings for water supply, as a booster and for domestic water supplies, pumping of sewage and slurries. They are also used in fire protection systems and for heating and cooling applications. In addition, they are popular in the beverage, dairy, food and oil & gas, petrochemical & chemical industries. Improper operation of centrifugal pumps, often result in mechanical integrity failures such as, high temperatures, low flow cavitation, low bearing and seal life, reduced impeller life, suction and discharge recirculation.

To circumvent such operational failures, Pump Operating Envelopes are made for a given pump specification to enable engineers and operators to make decisions regarding its operability. The below picture shows the various possible failures for the range of operation.

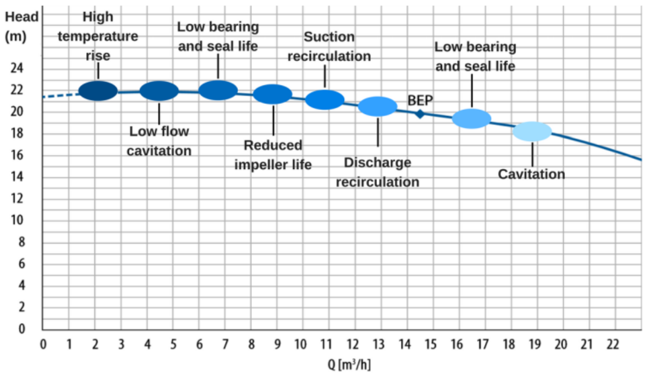


Figure 1. Pump Operating Failures [2]

The following focuses on predicting the allowable operating range or operating envelope for a pump's range of operation.

Problem Statement

A Centrifugal pump is used to transfer water from a horizontal vessel to a storage tank as shown in Figure 2.

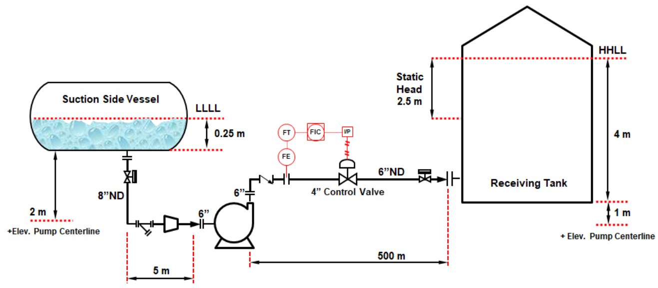


Figure 2. Pump System Schematic

The centrifugal pump has the following Head (H) vs. Flow (Q) characteristics as shown in Figure 2 and operates with a flow control valve at its discharge line. The flow rate required to be maintained is 189.2 m^3/h with the receiving tank pressure at 1.02 bara. The pump suction conditions are as follows,

Table 1. Pump Suction Conditions

Parameter	Value	Unit
Liquid Temperature [T]	20.0	^{0}C
Liquid Density [ρ]	997.8	kg/m^3
Dynamic Viscosity [μ]	1.00	cP
	0.001	kg/m.s
Vapour Pressure [P_v]	0.023	bara
Critical Pressure [P_c]	217.7	bara
Pump Operating Flow [Q]	189.2	m^3/h
	0.05256	m^3/s
Pump NPSH$_R$	0.69	bara
	7.05	m
Discharge Head [H]	6.29	bar
	64.24	m

The piping details of the pump system piping connections are as follows,

Table 2. Pump System Piping Details

Parameter	Value	Unit
Suction Pipe Size [ND]	8.625	in
Suction Pipe [WT]	8.18	mm
Suction Pipe ID [ID]	0.2027	m
Suction Pipe CS Area [A]	0.03227	m^2
Suction Pipe Roughness [ε]	45.2	µm
Suction ε/D	0.00022	-
Suction Pipe Length	5	m
Discharge Pipe Size [ND]	6.625	in
Discharge Pipe [WT]	7.11	mm
Discharge Pipe ID [ID]	0.1541	m
Discharge Pipe CS Area [A]	0.01864	m^2
Discharge Pipe Roughness [ε]	45.2	µm
Discharge ε/D	0.00029	-
Discharge Pipe Length	500	m

The pump system has piping components/fittings like block valves, check valves, flow elements, Y-strainers, bends, elbows, concentric reducers, etc. which add a dynamic pressure loss to the pump flow. These minor pressure/head losses can be estimated by using the relationship,

$$h_L = \frac{KV^2}{2g} \qquad (1)$$

Where, K is the Loss Coefficient & can be estimated from sources such as Crane's Handbook. In this module, the total minor head loss is assumed to be 0.5 bar. Additionally, as the liquid exits the discharge piping, exit losses is taken to be 5 m (~0.04 bar) based on Eq. 1

Table 3. Minor Losses

Parameter	Value	Unit
$\Sigma\Delta P_{\text{Minor Losses}}$	0.5	bar
$\Delta P_{\text{Exit Losses}}$	0.04	bar

The pump curves of the centrifugal pump are,

Table 4. Pump Curves

Flow [m^3/h]	Head [bar]	Head [m]
0.0	9.51	97.1
93	8.76	89.5
100	8.68	88.7
120	8.41	85.9
140	8.01	81.8
160	7.45	76.1
172	7.03	71.8
180	6.70	68.5
200	5.73	58.5
220	4.49	45.8
232	3.61	36.9

The equipment operating pressures and elevation from pump centerline details are as follows,

Table 5. Pump Suction Conditions

Parameter	Value	Unit
Suction Vessel Gauge Pressure	0.02	barg
	1.04	bara
Suction Vessel Elevation	2.00	m
Suction Vessel Low Low LL	0.25	m
Discharge Vessel Pressure	1.02	bara
Discharge Vessel Elevation	1.00	m
Discharge Vessel High High LL	4.00	m

The control valve details to be checked for is,

Table 6. Control Valve Details

Control Valve	C_V@100% Opening	F_L
4" Size	236	0.82
6" Size	433	0.84

Pump System Pressure Losses

The pump operating point is the point where the system resistance curve intersects the pump performance curve. This can be understood as, for a given flow rate, the resultant pump head arrived at, after accounting for all the static & dynamic losses (represented as system resistance) from the generated Total Dynamic Head (TDH). The factors contributing to system resistance is,

1. Suction Piping & Fittings Frictional Loss
2. Discharge Piping & Fittings Frictional Loss
3. Control Valve Pressure drop
4. Total Static Head

Suction & Discharge Frictional Losses

To estimate the total static head, the worst case scenario of Low Low Liquid Level (LLLL) in the suction vessel & High High Liquid Level (HHLL) in the receiving tank is considered. This is taken so to ensure that during LLLL, the pump does not suffer from cavitation due to Net Positive Suction Head (NPSH) deficiency, i.e., even during LLLL, $NPSH_A > NPSH_R$. To estimate $NPSH_A$, the suction line frictional pressure drop is calculated using the Darcy-Weisbach equation (Appendix B). The Suction Velocity (V_S) is estimated as,

$$V_s = \frac{Q}{A_s} = \frac{0.05256}{0.03227} \approx 1.63\ m/s \qquad (2)$$

The Suction Side Reynolds Number is,

$$Re = \frac{0.202715 \times 1.63 \times 997.8}{0.001} = 329{,}372 \qquad (3)$$

As the calculated suction side Reynolds number is much higher than 4000, the flow is well into the turbulent region. The friction factor can now be calculated using Colebrook equation but owing to its implicit nature, the friction factor is calculated using Swamee-Jain correlation (Appendix B),

$$f = \frac{0.25}{\left(\log_{10}\left[\frac{0.00022}{3.7} + \frac{5.74}{329372^{0.9}}\right]\right)^2} = 0.0163 \qquad (4)$$

Therefore, the frictional pressure drop is,

$$\Delta P = \frac{0.0163 \times 5 \times 997.8 \times 1.63^2}{2 \times 0.202715 \times 10^5} = 0.0053\ bar \qquad (5)$$

The results for suction side pressure drop can be summarized as,

Table 7. Suction Side Frictional Losses

Parameter	Value	Unit
Reynolds Number [Re]	329,699	-
Flow Behaviour	Turbulent	-
Friction Factor Equation	Swamee-Jain	-
Friction Factor	0.0163	-
Suction ε/D	0.00022	-
Pressure Drop [ΔP]	0.0053	bar

The calculations are similarly performed for discharge side with a discharge side velocity of 2.82 m/s & the results are summarized below,

Table 8. Discharge Side Frictional Losses

Parameter	Value	Unit
Reynolds Number [Re]	433,408	-
Flow Behaviour	Turbulent	-
Friction Factor Equation	Swamee-Jain	-
Friction Factor	0.0165	-
Suction ε/D	0.00029	-
Pressure Drop [ΔP]	2.123	bar

Total Static Head

The suction side static head is computed as,

$$\Delta P_S = \frac{(h_{LLLL} + h_{elev})_{Vessel} \times \rho \times g}{10^5} \qquad (6)$$

$$\Delta P_S = \frac{[0.25+2] \times 997.8 \times 9.81}{10^5} \approx 0.22\ bar \qquad (7)$$

Therefore the pump's $NPSH_A$ is calculated as,

$$NPSH_A = \Delta P_S + P_{surface} - P_v - \Delta P_{Fric} \qquad (8)$$

$$NPSH_A = 0.22 + 1.04 - 0.023 - 0.0053 \qquad (9)$$

$$NPSH_A = 1.23\ bara \qquad (10)$$

The calculated $NPSH_A$ is 1.23 bara and is higher than $NPSH_R$ of 0.69 bara.

The pump suction pressure is calculated as,

$$P_s = P_{vessel} + \frac{(h_{LLLL}+h_{elev})_{Vessel}\times\rho\times g}{10^5} - \varDelta P_F \quad (11)$$

$$P_s = 1.04 + \frac{(0.25+2)\times 997.8\times 9.81}{10^5} - 0.0053 \quad (12)$$

$$P_s \approx 1.25\ bara\ (12.78\ m) \quad (13)$$

The Pump Discharge Pressure for the flow rate of 189.2 m^3/h & corresponding pump head of 6.29 bara is calculated as,

$$P_d = \varDelta P + P_s = 6.29 + 1.25 \quad (14)$$

$$P_d = 7.54\ bara \quad (15)$$

Control Valve Pressure Drop

The control valve pressure drop becomes,

$$\Delta P_{CV} = P_d - \varDelta P_{F+s+min+ent} - P_{vessel} \quad (16)$$

$$\varDelta P_{F+s+min+ent} = \varDelta P_F + \varDelta P_s + \Delta P_m + \Delta P_e \quad (17)$$

$$\Delta P_{CV} = 7.54 - 2.123 - 0.27 - 0.5 - 0.04 - 1.02 \approx 3.59\ bara \quad (18)$$

Total Dynamic Losses

The total dynamic losses is computed as,

$$TDH = \varDelta P_F + \varDelta P_s + \Delta P_m + \Delta P_e + \Delta P_{CV} \quad (19)$$

$$TDH = 0.005 + 2.12 + 0.27 + 0.5 + 0.04 + 3.59 \approx 6.52\ bara \quad (20)$$

Control Valve C_v Required

The required C_v of the control valve is calculated by estimating the valve coefficients first followed by checking if choked flow exists (Appendix A),

$$K_{B1} = 1 - \left(\frac{d}{D_1}\right)^4 = 1 - \left[\frac{4}{7.981}\right]^4 = 0.937 \quad (21)$$

$$K_{B2} = 1 - \left(\frac{d}{D_2}\right)^4 1 - \left[\frac{4}{6.065}\right]^4 = 0.811 \quad (22)$$

$$K_1 = 0.5 \times \left[1 - \left(\frac{4^2}{7.981^2}\right)\right]^2 = 0.28 \quad (23)$$

$$K_2 = 1.0 \times \left[1 - \left(\frac{4^2}{6.065^2}\right)\right]^2 = 0.319 \quad (24)$$

$$\sum K = 0.28 + 0.319 + 0.937 - 0.811 = 0.726 \quad (25)$$

The selected control valve is 4" valve with C_v of 236 and F_L of 0.82 from Table 6.

$$F_P = \left[1 + \frac{0.726}{890}\left(\frac{236}{4^2}\right)^2\right]^{-1/2} = 0.9216 \quad (26)$$

$$F_{LP} = \left[\frac{0.28+0.937}{890}\left(\frac{236}{4^2}\right)^2 + \frac{1}{0.82^2}\right]^{-1/2} \approx 0.75 \quad (27)$$

The inlet pressure in psig for choke flow equation is calculated as by considering pressure drop between Pump Discharge and Control valve Inlet is very small as,

$$P_1 = ([6.29 + 1.25] \times 14.7) - 14.7 = 96.1\ psig \quad (28)$$

$$P_2 = 96.1 - (3.59 \times 14.7) = 43.4\ psig \quad (29)$$

$$\Delta P_{sizing} = 96.1 - 43.4 = 53\ psi \quad (30)$$

$$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}} = 0.96 - 0.28\sqrt{\frac{0.34}{3200}} \quad (31)$$

$$F_F = 0.957 \quad (32)$$

$$\Delta P_{choked} = \left[\frac{0.82}{0.9216}\right]^2 [(96.1 + 14.7) - 0.957 \times 0.34] \quad (33)$$

$$\Delta P_{choked} = 87\ psi \quad (34)$$

Since $\Delta P_{sizing} \leq \Delta P_{Choked}$, then $\Delta P_{sizing} = 53$ psi

$$C_v = \frac{189.2\times 4.4028675}{1\times 0.9216\sqrt{\frac{53}{0.9978}}} = 124.3\ gpm/\sqrt{psi} \quad (35)$$

Reinserting the calculated C_v value of 124.3, the value of F_p, F_{LP} and new C_v is re-computed iteratively,

Table 9. F_P, F_{LP} & C_v Iterations

Iteration	F_P	F_{LP}	C_v
1	0.92	0.75	124.3
2	0.98	0.82	117.4
3	0.98	0.82	117.1
4	0.98	0.82	117.1
5	0.98	0.82	117.1
6	0.98	0.82	117.1
7	0.98	0.82	117.1
8	0.98	0.82	117.1
9	0.98	0.82	117.1
10	**0.9789**	**0.8198**	**117.1**

Therefore, for a flow of 189.2 m^3/h, the C_v required is 117.1 gpm√psi. For the 4" valve selected, the % C_v becomes

$$\% \, C_v = \frac{117.1}{236} = 49.6 \,\% \tag{36}$$

Pump Operating Envelope

To generate the pump operating envelope, the above set of calculations is performed for various % C_v between 20% to 80% to estimate the total dynamic head at various flows. Below is the pump performance curve that includes the system resistance curves for various % C_v & flow rates.

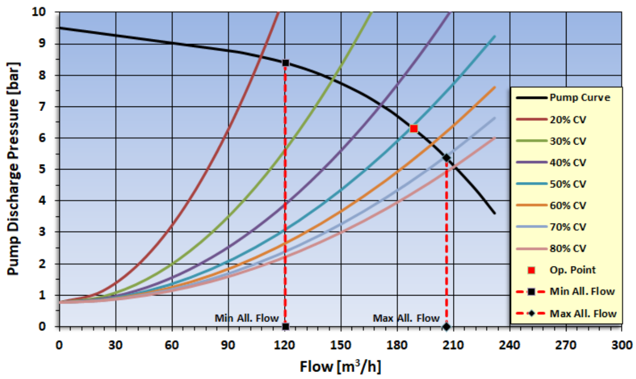


Figure 3. Pump Operating Envelope

Appendix A–Liquid Control Valve Sizing [1]

Based on ANSI/ISA S75.01.01 standards, to size the liquid control valve, the following set of equations can be used,

Step 1: Calculate Piping Geometry (F_p) & Liquid Pressure Recovery Factor (F_{LP})

$$F_P = \left[1 + \frac{\sum K}{N_2}\left(\frac{C_V}{d^2}\right)^2\right]^{-1/2} \tag{37}$$

Where, F_p = Piping geometric Factor [-]

N_1 = Constant [Value = 1.0]

N_2 = Constant [Value = 890]

C_v = Valve Coefficient [GPM/√psi]

d = Control Valve Size [inch]

The value of F_p is dependent on the fittings such as reducers, elbows or tees that are directly attached to the inlet & outlet connections of the control valve. If there are no fittings, F_p is taken to be 1.0. The term ΣK is the algebraic sum of the velocity head loss coefficients of all the fittings that are attached to the control valve & is estimated as,

$$\sum K = K_1 + K_2 + K_{B1} - K_{B2} \tag{38}$$

Where,

K_1 = Upstream fitting resistance coefficient [-]

K_2=Downstream fitting resistance coefficient [-]

K_{B1} = Inlet Bernoulli Coefficient [-]

K_{B2} = Outlet Bernoulli Coefficient [-]

Where,

$$K_{B1} = 1 - \left(\frac{d}{D_1}\right)^4 \tag{39}$$

$$K_{B2} = 1 - \left(\frac{d}{D_2}\right)^4 \tag{40}$$

Where,

D_1 = Pipe Inlet Diameter [in]

D_2 = Pipe Outlet Diameter [in]

If the upstream and downstream piping are of equal size, then, $K_{B1} = K_{B2}$, and therefore, are dropped from the ΣK equation. If the downstream pipe size is similar to upstream pipe size, i.e., $D_1 = D_2$, then $K_{B1} = K_{B2}$. The most commonly used fitting in control valve installations is the short-length concentric reducer. The equations for these fittings are,

$$K_1 = 0.5 \times \left[1 - \left(\frac{d^2}{D_1^2}\right)\right]^2, \text{ for inlet reducer.} \tag{41}$$

$$K_2 = 1.0 \times \left[1 - \left(\frac{d^2}{D_2^2}\right)\right]^2, \text{ for outlet reducer} \tag{42}$$

If the concentric reducers installed on either side of the control valve are identical, then

$$\sum K = K_1 + K_2 = 1.5 \times \left[1 - \left(\frac{d^2}{D^2}\right)\right]^2 \tag{43}$$

If the concentric reducers installed on either side of are identical, then, $\Sigma K = K_1 + K_2$. The liquid Pressure Recovery Factor (F_{LP}) is calculated as,

$$F_{LP} = \left[\frac{K_1 + K_{B1}}{N_2}\left(\frac{C_v}{d^2}\right)^2 + \frac{1}{F_L^2}\right]^{-1/2} \tag{44}$$

Step 2: Calculate Pressure Drop Required for Sizing (ΔP_{Sizing})

To estimate the ΔP required for sizing, ΔP_{sizing}, first the liquid critical pressure ratio (F_F) is calculated. Therefore,

$$F_F = 0.96 - 0.28\sqrt{\frac{P_v}{P_c}} \quad (45)$$

Where,

F_F = Liquid Critical Pressure Ratio [-]

Pv = Vapour Pressure [psia]

Pc = Critical Pressure [psia]

Using the value of F_F, ΔP_{choked} is calculated as,

$$\Delta P_{choked} = \left[\frac{F_{LP}}{F_P}\right]^2 [P_1 - F_F P_v] \quad (46)$$

If $\Delta P_{Valve} \leq \Delta P_{Choked}$, then $\Delta P = \Delta P_{Sizing}$

Else, Repeat calculations for next size.

Step 3: Calculate Required Control Valve C_v

The required control valve C_v is calculated as,

$$C_v = \frac{Q}{N_1 F_p \sqrt{\frac{\Delta P_{Sizing}}{[\rho_1/\rho_0]}}} \quad (47)$$

Or, estimating in terms of ΔP gives us,

$$\Delta P_{sizing} = \left[\frac{Q}{C_v N_1 F_p}\right]^2 \left[\frac{\rho_1}{\rho_0}\right] \quad (48)$$

Where, Q = Flowrate [gpm]

ρ_1/ρ_0 = Specific Gravity of Fluid [-]

ΔP_{sizing} = pressure drop [psig]

Upon calculating the required C_v, it is required to check if the calculated C_v is within the C_v limit of the selected control valve. If not, the next size of control valve is chosen and the calculations are repeated. To arrive at accurate predictions for C_v of the selected size, the calculations are repeated by re-inserting the calculated C_v & control valve size (d) value into the F_p equation, i.e., Eq. 1 to calculate the new value of F_p & further continued to estimate the final value of C_v. If the F_L value were to change between iterations, these values would need to be updated, and C_v re-calculated.

Appendix B – Line Sizing [3]

Pressure loss in piping without any size changes or fittings occurs due to friction between the fluid and the pipe walls. To estimate the piping pressure loss, the Darcy-Weisbach correlation is used as follows,

$$\Delta P = \frac{f \times L \times \rho \times V^2}{2D} \quad (49)$$

Where, ΔP = Pressure drop [bar]

f =Darcy Friction Factor [-]

L = Pipe Length [m]

ρ = Fluid Density [kg/m^3]

V = Fluid Velocity [m/s]

D = Pipe Inner Diameter, ID [m]

The Darcy friction factor may be determined by either using the appropriate friction factor correlation, or from a Moody Chart which is a function of the Reynolds number (Re).

$$Re = \frac{DV\rho}{\mu} \quad (50)$$

Where, μ = Dynamic Viscosity [kg.m/s]

The Darcy Friction Factor [f] depends on the Reynolds number follows the criteria,

If Re <= 2100 - Laminar Flow Equation

If Re <= 4000 - Churchill Equation

If Re > 4000 - Colebrook White Equation

The Laminar Flow equation also referred to as the Hagen Poiseuille's equation is,

$$f = \frac{64}{\mathrm{Re}} \quad (51)$$

The Churchill equation combines both laminar and turbulent flow regime friction factor expressions. It is accurate to within the error of the data used to construct the Moody diagram. This model also provides an estimate for the intermediate (transition) region; however this should be used with caution.

The Churchill equation shows very good agreement with the Darcy equation for laminar flow, accuracy through the transitional flow regime is unknown, in the turbulent regime a difference of around 0.5-2% is observed between the Churchill equation and the Colebrook equation. For Reynolds number up to ~4000,

$$f = 8\left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{(A+B)^{1.5}}\right]^{1/12} \quad (52)$$

$$A = \left[2.457 ln\left(\frac{1}{\left(\frac{7}{Re}\right)^{0.9} + 0.27\frac{\varepsilon}{D}}\right)\right]^{16} \quad (53)$$

$$B = \left[\left(\frac{37{,}530}{Re}\right)\right]^{16} \quad (54)$$

The Colebrook equation was developed taking into account experimental results for the flow through both smooth and rough pipe. It is valid only in the turbulent regime for fluid filled pipes. Due to the implicit nature of this equation it must be solved iteratively. A result of suitable accuracy for almost all industrial applications will be achieved in less than 10 iterations. For Reynolds number up to ~4000,

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{\varepsilon/D_H}{3.7} + \frac{2.51}{Re\sqrt{f}}\right] \quad (55)$$

Owing to the implicit nature of the Colebrook equation which requires iterations, an alternate correlation may be used to estimate the Darcy friction factor, i.e., Swamee-Jain Correlation which is calculated for Re > ~4000 as,

$$f = \frac{0.25}{\left(\log_{10}\left[\frac{\varepsilon/D}{3.7} + \frac{5.74}{Re^{0.9}}\right]\right)^2} \quad (56)$$

Notes on Operating Curves

1. For a given set of hydraulic conditions, a pump is designed to operate for one set of flow and head. Deviation from this operating point is allowed only to some degree.
2. Pump selection closer to the Best Efficiency Point (BEP) yields a more efficient pump with the least amount of vibration & radial forces acting on the shaft. Pump system resistance curve when calculated accurately ensures the pump operates where the performance curve intersects the system curve.
3. It is not always possible to operate the pump at BEP for the conditions required and hence a flow variation of ±10% of BEP is allowed.
4. Minimum stable continuous flow (MSCF) is the minimum flow below which the pump is not allowed to operate. Although API 610 recommends that the rated region is located between 80% to 110% of BEP, the preferred region of flow is between 70% to 120% of BEP.
5. Clause 6.1.12 of API 610 11th edition states "Setting limits for preferred operating region and the location of rated flow is not intended to lead to the development of additional sizes of small pumps or preclude the use of high-specific-speed pumps. Small pumps that are known to operate satisfactorily at flows outside of the specified limits and high specific speed pumps that may have a narrower preferred operating region than specified should be offered…" Therefore the Allowable Operating Region is set by the manufacturer as the allowable region to operate with stability whilst conforming to predefined API 610 vibration limits.
6. Pumps that are expected to operate less frequently can be chosen such that they operate at lower speeds at the cost of efficiency. Since the pump is selected to operate intermittently, a slightly lower efficiency pump is acceptable compared to a higher speed pump. This will also ensure a longer operating life cycle.

References & Further Reading

1. Control Valve Handbook, 5th Edition, Emerson
2. https://www.tapflopumps.co.uk/understanding-centrifugal-pumps
3. https://neutrium.net/fluid_flow/pressure-loss-in-pipe/

Appendix C – Pump Performance Curves Estimates

Pump Curves

Flow [m³/h]	Head [bar]	Head [m]
0.0	9.51	97.1
93	8.76	89.5
100	8.68	88.7
120	8.41	85.9
140	8.01	81.8
160	7.45	76.1
172	7.03	71.8
180	6.70	68.5
200	5.73	58.5
220	4.49	45.8
232	3.61	36.9

Suction Side Friction Factor [f]

Parameter	Value	Unit
Reynold's Number [N_{Re}]	329,372	-
Flow Behaviour	Turbulent	-
Friction Factor Equation	Swamee-Jain	-
Friction Factor	0.0163	-
Suction ε/D	0.00022	-
Pressure Drop [ΔP]	**0.0053**	**bar**

Discharge Side Friction Factor [f]

Parameter	Value	Unit
Reynold's Number [N_{Re}]	433,408	-
Flow Behaviour	Turbulent	-
Friction Factor Equation	Swamee-Jain	-
Friction Factor	0.0165	-
Discharge ε/D	0.00029	-
Pressure Drop [ΔP]	**2.123**	**bar**

Pump Suction Conditions

Parameter	Value	Unit
Service	Water	
Liquid Temperature [T]	20.0	°C
Liquid Density [ρ]	997.8	kg/m³
Dynamic Viscosity [μ]	1.00	cP
	0.001	kg/m.s
Vapour Pressure [P_v]	0.023	bara
Critical Pressure [P_c]	217.7	bara
Pump Operating Flow [Q]	189.2	m³/h
	0.053	m³/s
	52.44	kg/s
Pump $NPSH_R$	0.69	bara
	7.05	m
Discharge Head [H]	6.29	bar
	64.24	m
Total Piping Minor Losses		
$\Sigma\Delta P_{Minor\ Losses}$	0.5	bar
$\Delta P_{Exit\ Losses}$	0.04	bar
Suction Vessel Elevation - Pump Centreline		
Suction Vessel P_{Gauge}	0.02	barg
Suction Vessel Pressure	1.04	bara
Suction Vessel Elevation	2.00	m
Suction Vessel Low Low LL	0.25	m
Pump $NPSH_A$	1.23	bara
Is $NPSH_A$ Sufficient	Yes	-
Pump Suction Pressure [P_1]	1.25	bara
Static Head -Suction Side	12.78	m

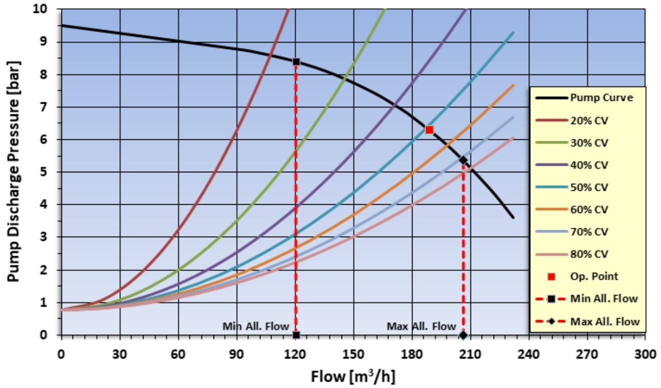
PIPING DETAILS

Parameter	Value	Unit
Suction Pipe Size [ND]	8.625	in
Suction Pipe [WT]	8.18	mm
Suction Pipe ID [ID]	0.2027	m
Suction Pipe CS Area [A]	0.03227	m²
Suction Pipe Roughness [ε]	45.2	μm
Suction Fluid Velocity [V]	1.63	m/s
Suction Pipe Length	5	m
Discharge Pipe Size [ND]	6.625	in
Discharge Pipe [WT]	7.11	mm
Discharge Pipe ID [OD]	0.1541	m
Discharge Pipe CS Area [A]	0.01864	m²
Discharge Pipe Roughness [ε]	45.2	μm
Discharge Fluid Velocity [V]	2.82	m/s
Discharge Pipe Length	500	m
Control Valve Parameters - Operating Flow		
Control Valve C_v - Op. Flow	117	GPM/√psi
Control Valve C_v %	49.6	%
Discharge Vessel Elevation - Pump Centreline		
Discharge Vessel Pressure	1.02	bara
Discharge Vessel Elevation	1.00	m
Discharge Vessel High High LL	4.00	m
Total Static Head	0.27	bara
Pump Discharge Pressure [P_2]	7.54	bara
Control Valve Pressure Drop	3.59	bara
Total Dynamic Losses	6.52	bar
Pipe Outlet Pressure	1.02	bara

CENTRIFUGAL PUMP OPERATING ENVELOPE

% Flow	Flow [Q]	Suction N_{Re}	Discharge N_{Re}	$f_{suction}$	$f_{Discharge}$	$\Delta P_{Suction}$	$\Delta P_{Discharge}$	Total ΔP_F	Total ΔP_S	$\Delta P_{Minor\ Losses}$	$\Delta P_{Exit\ Losses}$	ΔP_{CV} [bar]	ΔP_{CV} [bar]	ΔP_{CV} [bar]	ΔP_{CV} [bar]	ΔP_{CV} [bar]	ΔP_{CV} [bar]	ΔP_{CV} [bar]
[%]	[m³/h]	[-]	[-]	[-]	[-]	[bar]	[bar]	[bar]	[bar]	[bar]	[bar]	20	30	40	50	60	70	80
0	0	0.0	0.0	0.0	0.0	0.00	0.00	0.00	0.27	0.50	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
10	23	40,388	53145.1	0.0226	0.0217	0.00	0.04	0.04	0.27	0.50	0.00	0.33	0.15	0.08	0.05	0.04	0.03	0.02
20	46	80,776	106290.2	0.0198	0.0193	0.00	0.15	0.15	0.27	0.50	0.00	1.33	0.59	0.33	0.21	0.15	0.11	0.08
30	70	121,164	159435.3	0.0185	0.0182	0.00	0.32	0.32	0.27	0.50	0.01	2.99	1.33	0.75	0.48	0.33	0.24	0.19
40	93	161,552	212580.4	0.0178	0.0176	0.00	0.55	0.55	0.27	0.50	0.01	5.31	2.36	1.33	0.85	0.59	0.43	0.33
50	116	201,940	265725.5	0.0173	0.0172	0.00	0.83	0.84	0.27	0.50	0.01	8.29	3.69	2.07	1.33	0.92	0.68	0.52
60	139	242,328	318870.7	0.0169	0.0169	0.00	1.18	1.18	0.27	0.50	0.02	11.94	5.31	2.99	1.91	1.33	0.98	0.75
70	162	282,717	372015.8	0.0166	0.0167	0.00	1.58	1.59	0.27	0.50	0.03	16.26	7.23	4.06	2.60	1.81	1.33	1.02
80	186	323,105	425160.9	0.0164	0.0165	0.01	2.05	2.05	0.27	0.50	0.04	21.23	9.44	5.31	3.40	2.36	1.73	1.33
90	209	363,493	478306.0	0.0162	0.0164	0.01	2.57	2.57	0.27	0.50	0.05	26.87	11.94	6.72	4.30	2.99	2.19	1.68
100	232	403,881	531451.1	0.0160	0.0163	0.01	3.15	3.15	0.27	0.50	0.06	33.18	14.75	8.29	5.31	3.69	2.71	2.07

Flow [Q]	$\Delta P_{Head\ Losses}$	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]	ΔP_{TDH} [bar]
[m³/h]	[bar]	20	30	40	50	60	70	80
0	0.77	0.8	0.8	0.8	0.8	0.8	0.8	0.8
23	0.81	1.1	1.0	0.9	0.9	0.8	0.8	0.8
46	0.92	2.2	1.5	1.3	1.1	1.1	1.0	1.0
70	1.09	4.1	2.4	1.8	1.6	1.4	1.3	1.3
93	1.33	6.6	3.7	2.7	2.2	1.9	1.8	1.7
116	1.62	9.9	5.3	3.7	2.9	2.5	2.3	2.1
139	1.97	13.9	7.3	5.0	3.9	3.3	2.9	2.7
162	2.39	18.6	9.6	6.4	5.0	4.2	3.7	3.4
186	2.86	24.1	12.3	8.2	6.3	5.2	4.6	4.2
209	3.39	30.3	15.3	10.1	7.7	6.4	5.6	5.1
232	3.98	37.2	18.7	12.3	9.3	7.7	6.7	6.1



Appendix D – Liquid Control Valve Sizing

LIQUID CONTROL VALVE SIZING - ANSI/ISA S75.01.01 PROCEDURE [Ref: Emerson Control Valve Handbook, 5th Edition, Page 100]

Process Input Data		
Parameter	**Value**	**Units**
Service	**Water**	
PumpFlow Rate [Q]	**833**	GPM
Inlet Pressure [P_1]	**96.1**	psig
Oulet Pressure [P_2]	**43.4**	psig
Presure Drop [ΔP]	53	psi
Inlet Temperature [T_1]	68	^{0}F
ρ_1/ρ_0 [SG at Flowing T]	0.9978	-
Vapour Pressure [P_v]	0.34	psia
Critical Pressure [P_c]	3200	psia
Control Valve Details		
Valve Characteristics	**Linear**	
Size, *d* [inch]	**C_v @100%**	**F_L**
4	**236**	**0.82**
6	**433**	**0.84**
8	**846**	**0.87**

Piping Resistances [ΣK]		
U/S Pipe Size [ND]	8.625	in
U/S Pipe Wall Thickness [WT]	0.322	in
U/S Pipe Inner Diameter [ID]	7.981	in
D/S Pipe Size [ND]	6.625	in
D/S Pipe Wall Thickness [WT]	0.280	in
D/S Pipe Inner Diameter [ID]	6.065	in
N_1 [Constant]	1	-
N_2 [Constant]	890	-
Selected Control Valve Size	**4**	in
C_v of Selected Valve	236	GPM/√psi
F_L of Selected Valve	0.82	-
K_1 [U/S Fitting Resistance Coefficient]	0.280	-
K_2 [D/S Fitting Resistance Coefficient]	0.319	-
K_{B1} [Inlet Bernoulli Coefficient]	0.937	-
K_{B2} [Outlet Bernoulli Coefficient]	0.811	-
ΣK [Sum of Resitances]	0.726	-

Piping Geometry [F_p]		
Piping Geometry [F_p]	0.9216	-
Liquid Recovery Factor [F_{LP}]		
Liquid Recovery Factor [F_{LP}]	0.7485	-
Choke Conditions Check		
Liquid critical pressure ratio [F_F]	0.957	-
Choked ΔP [ΔP_{Choke}]	73	psi
Flow Condition Check	Subcritical	-
Sizing ΔP [ΔP_{Sizing}]	53	psi
Required Control Valve Size		
Required Control Valve C_v	**117.1**	GPM/√psi
Valve Characteristics	**Linear**	-
% Opening of Selected Valve	**49.6**	%
Note on Valve Characteristics:		
Quick Opening => C_v% = [Valve Open %]$^{0.5}$		
Linear => C_v% = Valve Open %		
Equal % => C_v% = [Valve Open %]^3		

F_P & F_{LP} Calculations			
Iteration	**F_P**	**F_{LP}**	**C_v**
1	0.92	0.75	124.3
2	0.98	0.82	117.4
3	0.98	0.82	117.1
4	0.98	0.82	117.1
5	0.98	0.82	117.1
6	0.98	0.82	117.1
7	0.98	0.82	117.1
8	0.98	0.82	117.1
9	0.98	0.82	117.1
10	**0.98**	**0.82**	**117.1**

Module 3

Predicting Performance Curves of Centrifugal Pumps in the Absence of OEM Data

Chemical and Mechanical Engineers in the oil & gas industry often carry out the task of conducting technical studies to evaluate piping and pipeline systems during events such as pump trips and block valve failures that can lead to pipes cracking at the welded joints, pump impellers rotating in the reverse direction and damaged pipe supports due to excessive vibrations to name a few. Although much literature is available to mitigate such disturbances, a key set of data to conduct transient studies are pump performance curves, a plot between pump head and flow. The present module is aimed at applying engineering research in industrial applications for practicing engineers. It provides a methodology called from available literature from past researchers, allowing engineers to predict performance curves for a Volute Casing End Suction Single Stage Radial Pump. In the current undertaking, the pump in question is not specific to any one industry but the principles are the same for a Volute Casing End suction radial pump.

1. Introduction

Traditionally performance curves are provided by the pump original equipment manufacturers (OEM) based on their customized/proprietary models of pump impellers which are designed using methods such as computational fluid dynamics (CFD) and also field tested to provide guarantee in meeting the requirements of the customer. With wear and tear in pumps systems in ageing facilities that causes deviation from the manufactured OEM pump curves, it becomes difficult to accurately predict if the pump can deliver the required head for the new application.

In Brownfield projects, when a plant undergoes revamp for new process conditions, often existing pumps are re-evaluated and reused for different applications with or without impeller trimming. In case where impeller trimming cannot be applied, but instead a larger impeller is required, the pump is refurbished accordingly based on head required for a given set of pump constraints. In Greenfield projects, when no pump vendor data is available, a necessity arises to use performance curves to conduct hydraulic studies, such as pipeline/piping studies for surge analysis and design pressure.

The working principle of a centrifugal pump involves using centrifugal force of a rotating impeller enclosed in a casing to impart energy to a fluid. In doing so, a portion of the energy is lost in the form of mechanical losses with the remaining being transferred to the fluid that raises the fluid's pressure when discharging from the pump casing. A pump impeller consists of vanes that are positioned on a disc to hold fluid and transfer energy as the impeller rotates. Impeller vane geometry is mainly of three types, namely, forward positioned, straight positioned and backward positioned. Backward positioned vanes are popularly used for the reason that with increase in volumetric flow, power consumption decreases.

Impellers are also characterized as open impellers, semi-open impellers and closed impellers. Open impellers consist of vanes mounted on central ring to which a rotating shaft is connected. In the case of semi-open impellers, the vane series is held on a circular disc only on one side while closed/shrouded impellers consist of the vane series encapsulated between two discs. The main disadvantage of open/semi-open impellers is that the vanes are sensitive to wear and tear but offer the advantage of maintaining the clearance of the wear ring. Closed/Shrouded Impellers on the other hand, are less susceptible to wear and tear and can also deal with volatile and explosive fluids. The disadvantage of closed/shrouded impellers though efficient initially, suffer efficiency losses due to an increase in clearance of the wear ring. A representation of the power consumption trends between the three vane geometries is shown in Fig. 1.

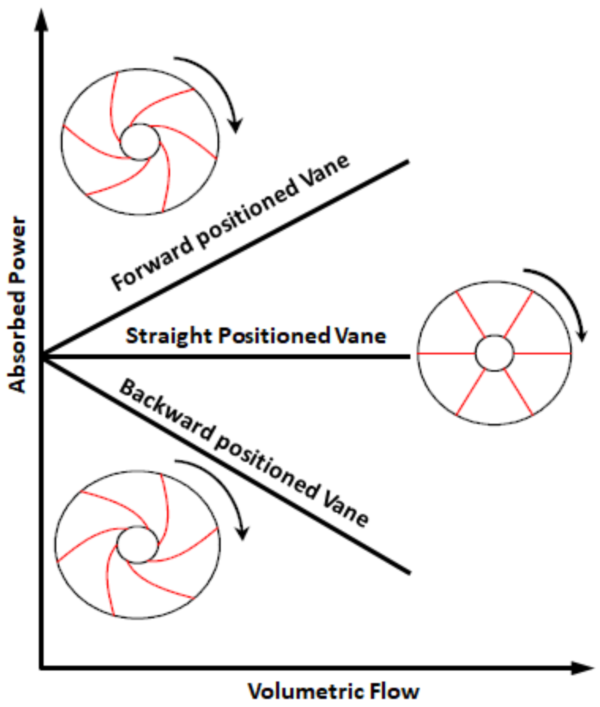


Fig 1. Vane Geometry and Power Consumption

In the current undertaking, a backward vane positioned, shrouded impeller is chosen considering lower power consumption at higher flow rates for an end suction single stage radial flow pump.

2. Principle of Performance Curves

Based on the impeller geometry, performance curves are derived from an aerodynamic analysis of the pump impeller. The basic equation that governs fluid behaviour at the pump's impeller is the Euler's Turbomachine equation relating pump head and fluid velocity. To apply Euler's Equation, the fluid's velocity components are expressed as shown in Fig. 2.

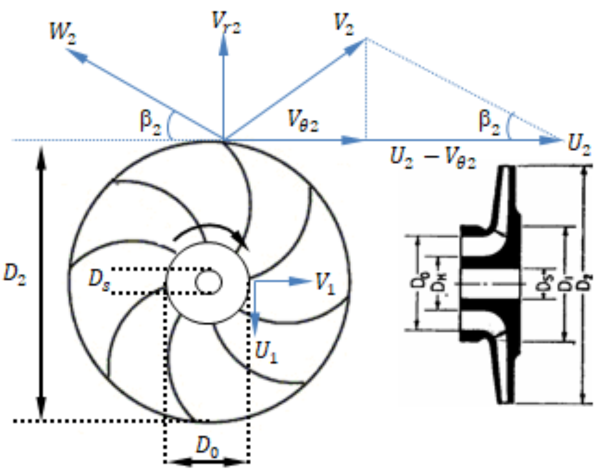


Fig 2. Velocity Triangle of Pump Impeller

2.1. Net Theoretical Head Relationship

From the velocity triangle shown in Fig. 1, the net theoretical head is the head developed based on a finite number of vanes in the impeller. The aerodynamic relationship between the net theoretical head ($H_{Net\ Theoretical}$) developed by the fluid for a given impeller speed and its respective velocity components at the impeller inner diameter (ID) represented by subscript '1' and outer diameter (OD) represented by subscript '2' is written as,

$$H_{Theoretical} = \frac{1}{g}\left[U_2 V_{\theta 2} - U_1 V_{\theta 1}\right] \qquad (1)$$

$$H_{Net\ Theoretical} = \frac{1}{g}\left[U_2 V_{\theta 2}^{'} - U_1 V_{\theta 1}^{'}\right] \qquad (2)$$

$$\text{Volumetric Flow, } Q = \pi D_2 b_2 V_{r2} \varepsilon_2 \qquad (3)$$

From the above relationships, a contraction factor (ε_2) is applied to estimate the flow that takes into account the decrease in inlet area of the impeller due to vane thickness. The impeller outlet diameter passage width (b_2) is considered to estimate the flow rate (Q) into the impeller. The chief parameter based on which other impeller parameters such as vane angle, passage width, number of vanes, etc. are calculated is the impeller inner diameter (ID), D_1 and outer diameter (OD), D_2 for a given impeller speed (N).

2.2. Pump Specific Speed (N_s)

Pump specific speed is a measure to determine what kind of pumps can be selected for a given service. Based on the pump specific speed value, the choice of pumps can vary from radial, Francis Vane, mixed flow or axial flow. The pump specific speed [3] is calculated in metric terms with the below described equation,

$$\text{Specific Speed, } N_s = \frac{N\sqrt{Q}}{H^{3/4}} \left(\frac{\text{rpm.}\frac{m^3}{\min}}{m} \right) \tag{4}$$

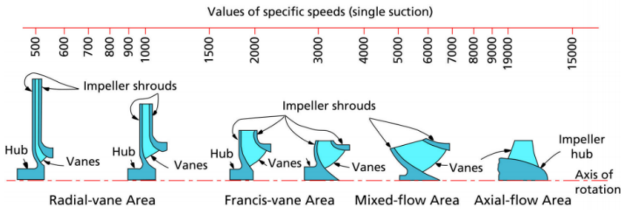


Fig 3. Pump Specific Speed Chart [4]

The above graph shows a distribution of the pump specific speeds based on which the type of pump is selected. It is to be noted that the values of specific speed can be different based on the units of measurements used and Fig 4 is only for illustrational purposes. Based on the pump speed the volumetric efficiency can be calculated [3] using the relationship,

$$\text{Volumetric Efficiency, } \eta_v = \frac{1}{1 + \frac{1.124}{N_s^{2/3}}} \tag{5}$$

Volumetric efficiency is used to estimate the total flow rate entering into the impeller eye which in turn is used to calculate the impeller eye diameter. Therefore (Q_s') is computed as,

$$\text{Total Flow rate, } Q_s' = \frac{Q}{\eta_v} \tag{6}$$

Q_s' represents the flow that is required to enter the impeller to meet the discharge flow conditions indicated by 'Q' since a portion of the incoming fluid is expected to accumulate in the pump. Hence all impeller design and performance curves calculations are made with Q_s' to meet conditions of Q.

2.3. Speed and Angular Velocity

The impeller diameters are calculated by relating the impeller dimensions to the impeller speed (N). The impeller speed is converted to velocity terms, i.e., angular velocity (ω). The relationship between impeller speed and angular velocity is,

$$\text{Angular Velocity, } \omega(m/s) = \frac{2 \times \pi \times N_{rpm}}{60} \tag{7}$$

2.4. Impeller Vane Angle (β_1, β_2)

When a fluid is rotated by a surface, a certain amount of slippage occurs between the impeller diameter tip and the fluid making contact with the impeller tip. This causes the actual fluid velocity leaving the impeller diameter to be slightly lower than the impeller tip speed with slippage expressed as a slip factor (σ). This is incorporated into the velocity triangle relationship to estimate the tangential velocity terms $V_{\theta 1}$ and $V_{\theta 2}$, radial velocity terms V_{r1} and V_{r2} as,

$$V_{\theta 1} = U_1 \sigma - \frac{V_{r1}}{Tan\beta_1} \tag{8}$$

$$V_{\theta 2} = U_2 \sigma - \frac{V_{r2}}{Tan\beta_2} \tag{9}$$

The slippage factor (σ) is computed by relating to the number of vanes (Z) and inlet and outlet diameter vane angle, β_1 and β_2 as,

$$\sigma = 1 - \frac{\sqrt{Sin\beta_2}}{\beta_1 Z^{0.7}}, \text{ For, } \frac{R_1}{R_2} \leq \varepsilon_{\lim it} \tag{10}$$

$$\text{And, } \sigma = 1 - \frac{\sqrt{Sin\beta_2}}{\beta_1 Z^{0.7}} \left[1 - \left(\frac{\left(\frac{R_1}{R_2} \right) - \varepsilon_{\lim it}}{1 - \varepsilon_{\lim it}} \right)^{1/3} \right]$$

$$\text{For, } \frac{R_1}{R_2} > \varepsilon_{\lim it} \tag{11}$$

$$\text{And, } \varepsilon_{\lim it} = e^{\left(-\frac{8.16 Sin\beta_2}{Z} \right)} \tag{12}$$

The number of vanes (Z) required is calculated as,

$$Z = 6.5 \times \left(\frac{D_2 + D_1}{D_2 - D_1} \right) \times Sin\left(\frac{\beta_1 + \beta_2}{2} \right) \tag{13}$$

The vane angle at the inner diameter (ID) is computed from the velocity triangle relationship by relating it to the radial component and impeller tip speed as follows,

Impeller ID Vane Angle, $\beta_1 = Tan^{-1}\left(\frac{V_{r1}}{U_1}\right)$ (14)

2.5. Impeller Dimensions Relationship

The main parameters required to be estimated are, End of Main Shaft Diameter (D_s), Hub Diameter (D_H), Hub Length (L_H), impeller inlet passage width (b_1), impeller outlet passage width (b_2) impeller eye diameter (D_0), impeller inner diameter (D_1), and impeller outlet diameter (D_2). The impeller outer diameter (D_2) can be calculated using Stepanoff Chart [2]. To calculate the above mentioned parameters, the following equations can be used.

Shaft Dia, $D_{sh} = \left[\frac{P(HP)\times 321000}{N(rpm)\times S_s(psi)}\right]^{1/3}$ (15)

Hub Diameter, $D_H = (1.5\,to\,2.0)\times D_{sh}$ (16)

Hub Length, $L_H = (1.0\,to\,2.0)\times D_H$ (17)

The fluid velocity at Impeller Eye (V_{eye}) is calculated as,

$V_{eye} = [(0.07\,to\,0.11) + 0.00023N_s]\times\sqrt{2gH}$ (18)

The impeller eye diameter (D_0) is taken to be,

Impeller Eye Dia, $D_0 = \sqrt{\frac{4\times Q_s^{'}}{\pi\times V_{eye}} + D_H^2}$ (19)

The various impeller speeds are as follows,

Impeller OD Tip Speed, $U_2 = K_u\sqrt{2gH}$ (20)

OD Radial Velocity, $V_{r2} = K_{m2}\sqrt{2gH}$ (21)

ID Radial Velocity, $V_{r1} = K_{m1}\sqrt{2gH}$ (22)

The impeller diameters is calculated as,

Impeller Outer Diameter, $D_2 = \frac{60\times U_2}{\pi\times N}$ (23)

Impeller Outer Diameter, $D_1 = D_2\left(\frac{D_2}{D_1}\right)$ (24)

Impeller ID Tip Speed, $U_1 = \frac{\pi\times D_1\times N}{60}$ (25)

Inlet Passage Width, $b_1 = \frac{Q_s^{'}}{\pi D_1 V_{r1}\varepsilon_1}$ (26)

Outlet Passage Width, $b_2 = \frac{Q_s^{'}}{\pi D_2 V_{r2}\varepsilon_2}$ (27)

The contraction factor (ε) for the inner and outer diameters can be estimated by using the thickness of the impeller passage (t) at the inlet and outlet diameters as,

Contraction factor, $\varepsilon = 1 - \frac{Zt}{\pi D Sin\beta}$ (28)

The values of K_u, K_{m1}, K_{m2} and D_2/D_1 can be computed from Stepanoff Chart [3],

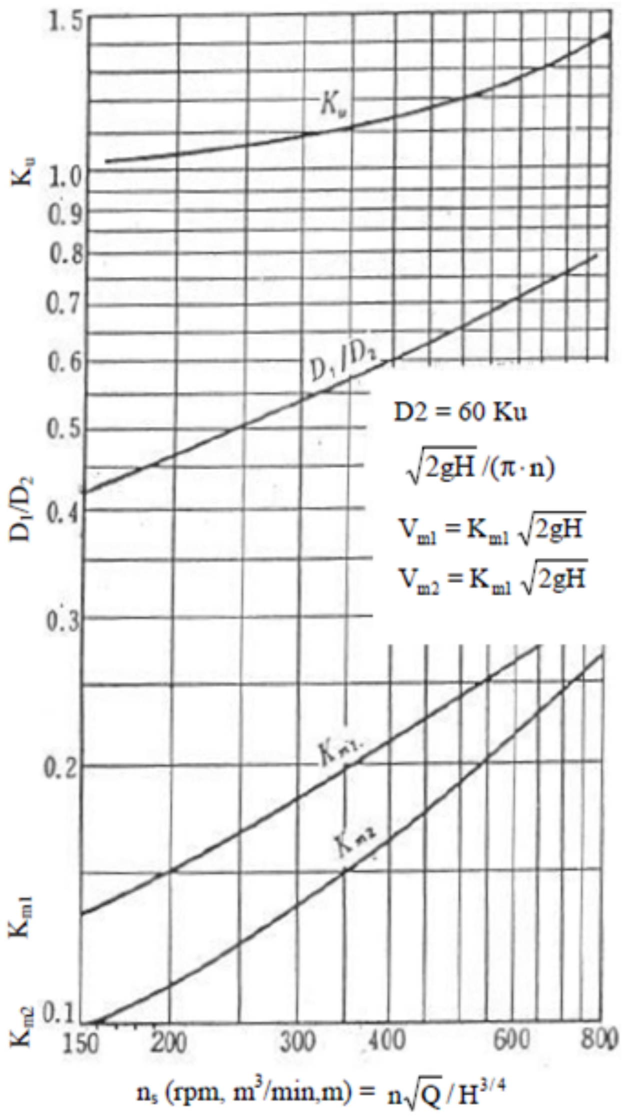


Fig 4. Stepanoff Chart for K_u, K_{m1}, K_{m2}, D_2/D_1

From the equations presented, design procedures can be commenced by assuming 'β2' & iteratively calculating until the actual head calculated matches with the required pump head. Followed by calculating the net theoretical head, the actual head is calculated by subtracting the pump losses for a range of flow rates.

3. Pump Losses

In a realistic scenario, centrifugal pumps experience different forms of mechanical losses. The different types of losses expected during pump operation are (i) Circulation losses, (ii) Inlet Incidence losses, (iii) Surface Friction losses, (iv) Volute Friction losses and (v) Diffusion losses. In addition, parasitic losses are also considered such as (vi) Disc Friction losses and (vii) Recirculation losses. When these losses are subtracted from the theoretical head, the actual head developed by the pump is arrived at. The below figure shows the difference between the net theoretical head and actual pump head.

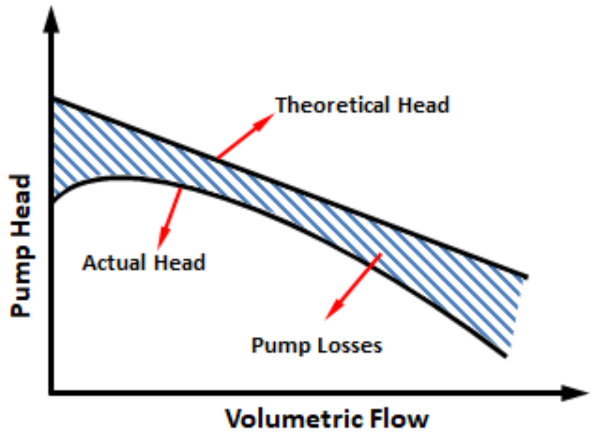


Fig 5. Theoretical Head vs. Actual Head

3.1. Circulation Losses

Circulation flow losses are characterized by circulatory flow that exists within a closed impeller channel when the impeller is rotating. At this point, there would be a mismatch of relative velocity (W) between the inlet side and outlet side of the impeller vane. The circulation head is calculated as,

$$H_{circ} = H_{Theoretical} - H_{Net\,Theoretical} \quad (29)$$

$$H_{circ} = \frac{(U_2 V_{\theta 2} - U_1 V_{\theta 1}) - (U_2 V'_{\theta 2} - U_1 V'_{\theta 1})}{g} \quad (30)$$

$$H_{circ} = \frac{U_2 (V_{\theta 2} - V'_{\theta 2}) + U_1 (V'_{\theta 1} - V_{\theta 1})}{g} \quad (31)$$

$$H_{circ} = \frac{(U_2 \times V_{s2}) + (U_1 \times V_{s1})}{g} \quad (32)$$

The slip velocity is normalized by the impeller tangential velocity as [6],

$$\sigma_s = 1 - \frac{V_s}{U} \quad (33)$$

Therefore the slip velocities at the inlet diameter (ID) and outlet diameter (OD) are,

$$V_{s1} = (1 - \sigma_{s1}) \times U_1 = V'_{\theta 1} - V_{\theta 1} \quad (34)$$

$$V_{s2} = (1 - \sigma_{s2}) \times U_2 = V_{\theta 2} - V'_{\theta 2} \quad (35)$$

With the assumption that the slip factor is nearly equal at both the impeller ID and OD, the whirl velocities are written as,

$$V'_{\theta 1} = U_1 (2 - \sigma_{s1}) - V_{r1} Cot\beta_1 \quad (36)$$

$$V'_{\theta 2} = U_2 \sigma_{s2} - V_{r2} Cot\beta_2 \quad (37)$$

3.2. Inlet Incidence Losses

Incidence flow losses are characterized by losses resulting from a forced change of velocity when fluid enters the pump impeller. When fluid enters the impeller eye in a normal direction, it is followed by a radial change in the direction of fluid flow. Additionally due to difference between the vane inlet angle and angle at which the fluid enters the vane cascade, a loss of head occurs due to forced change in velocity. The incidence losses are calculated as [6],

$$h_{in} = f_{in} \times \frac{(U_1 - V_{\theta 1})^2}{2g} \quad (38)$$

Where, $f_{in} = 0.5 - 0.7$

$$V_{\theta 1} = U_1 - \frac{Q'_s}{Tan\beta_1 \times \left[\pi D_1 b_1 - \frac{Z \times t \times b_1}{Sin(\beta_1)} \right]} \quad (39)$$

3.3. Surface Friction Losses

No pump system has perfectly smooth surfaces but instead has some amount of roughness. As a result when the fluid enters the impeller eye, friction is caused between the fluid and the disc surface. Taking into account the losses at the solid boundaries such as stationary vanes, diffuser and the rest of the impeller surfaces, the surface frictional head loss is calculated as,

$$h_{sf} = \frac{b_2(D_2 - D_1)\times(W_1 + W_2)^2}{2\times Sin\beta_2 \times H_r \times 4g} \quad (40)$$

Where, $$H_R = \frac{b_2\left(\frac{\pi D_2}{Z}\right)\times Sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z}\right)\times Sin\beta_2} \quad (41)$$

Where, $$W_1 = \frac{V_{r1}}{Sin\beta_1} \quad (42)$$

Where, $$W_2 = \frac{V_{r2}}{Sin\beta_2} \quad (43)$$

3.4. Diffusion Losses

Diffusion Losses are characterized by a loss of head when the relative velocity at the inlet near the impeller eye exceeds the outer impeller's relative velocity by a certain factor, causing loss of a portion of the velocity head difference. The diffusion head loss is [6],

$$h_{DL} = 0.25\times\left[\left(\frac{W_1}{W_2}\right)^2 - 2\right]\frac{W_2^2}{2g} \quad (44)$$

If, $W_1/W_2 > 1.4$

3.5. Volute Friction Losses

The pump volute receives the fluid pumped by the impeller. Due to its curved shape and changing area, pressure head is lost as the fluid moves towards the discharge flange. Modifying Ref [1] with respect to volute throat area, the volute friction loss is,

$$h_{vf} = 0.8\times\frac{\left[V_{\theta 2}\times\left(\frac{D_2}{D_3}\right)\right]^2 - \left[\frac{Q_s^{'}}{A_3}\right]^2}{2g} \quad (45)$$

Assuming that, $D_3 = 1.3\times D_2$ (46)

Taking Volute Width, $b_3 = 2\times b_2$ (47)

Volute Throat Area, $A_3 = \pi\times D_3\times b_3$ (48)

3.6. Disc Friction Losses

Disc friction losses are the result of a viscous friction between the outside portion of the impeller Disc and the surface of the pump casing. Hence in the case of open impellers, the Disc friction is lower than the case where closed impellers are used. The Disc friction losses can be calculated as [6],

$$P_{df} = C_M\times\rho\times\omega^3\times\left(\frac{D_2 - D_1}{2}\right)^5 \quad (49)$$

Rearranging with $\rho = Q_s'/v$,

$$h_{df} = \frac{P_{df}}{m} = \frac{C_M\times\omega^3\times\left(\frac{D_2 - D_1}{2}\right)^5}{Q_s^{'}}(J/kg) \quad (50)$$

$$h_{df} = \frac{C_M\times\omega^3\times\left(\frac{D_2 - D_1}{2}\right)^5}{Q_s^{'}}\left(\frac{kJ}{kg}\times\frac{1}{1000}\right) \quad (51)$$

$$h_{df} = \frac{C_M\times\omega^3\times\left(\frac{D_2 - D_1}{2}\right)^5}{Q_s^{'}}\left(\frac{102.04}{1000}\right)m \quad (52)$$

$$h_{df} = \frac{0.10204\times C_M\times\omega^3\times\left(\frac{D_2 - D_1}{2}\right)^5}{Q_s^{'}}m \quad (53)$$

Where,

$$C_M = \left(\frac{k_s}{0.5\times D_2}\right)^{0.25}\times\left(\frac{s}{0.5\times D_2}\right)^{0.1}\times \mathrm{Re}^{-0.2} \quad (54)$$

Where, b_4 is the volute width

$$\mathrm{Re} = \frac{U_2\times {D_2}/{2}\times\rho}{\mu} \quad (55)$$

The value of Disc friction loss coefficient (C_m) depends on the Disc surface roughness (k_s) and also the axial gap width (s).

3.7. Recirculation Losses

Recirculation losses are caused due to eddies formed in the pump impeller. The recirculation losses also depend on the size of the impeller in addition to the flow rates into the pump that decide the flow pattern. Hence with larger diameter impellers the recirculation losses increase. Pumps with high specific speeds also tend to exhibit a higher chance of recirculation. The head loss due to recirculation is estimated as [5],

$$h_{RL} = 0.005 \times \frac{\omega^3 \times D_1^2}{\rho Q}\left(1 - \frac{Q}{Q_0}\right)^{2.5} \quad (56)$$

Where, Q_0 = Design Flow rate

The value of 0.005 for the loss coefficient is described as the default value as per Ref [5]. Using the default value of 0.005, it is observed by the Author to be very high and yields recirculation losses with negative numbers. The recirculation loss coefficient depends on the piping configuration upstream of the pump in addition to the geometrical details of the inlet. The current module does not account for the upstream piping and the Author iteratively estimates that the recirculation losses coefficient is to be taken in the order of 1×10^{-3} to 1×10^{-2} in order to compensate for the piping losses and arrive at non-negative recirculation loss coefficients.

3.8. Pump Leakage Losses

Pump leakage losses cause a loss of head and subsequently efficiency due to leakages through the Disc and wearing ring. These volumetric losses can be modelled as loss of flow through an orifice. From Ref [8] and Ref [9], the leakage loss can be worked out as,

$$Q_L = C_L \times A_L \times \sqrt{2 \times g \times H_L} \quad (57)$$

From Ref [5], leakage Area is estimated as,

$$A_L = \pi \times D_1 \times b_{cl} \quad (58)$$

And Leakage Head Loss, from Ref [8] as,

$$H_L = \frac{3}{4}\left[\frac{\left(U_2^2 - U_1^2\right)}{2 \times g}\right] \quad (59)$$

Ref [9] provides an approximated value of 0.6 and this has been incorporated into the present undertaking.

As per Ref [8], a wearing ring clearance of 0.01 inch for rings up to 6 inch diameter or less is a good practice. For rings greater than 6 inches and up to 12 inch, the clearance is increased by 0.001 inch for every inch of ring diameter. For over 12 inch, increase by 0.0005 inches per inch of ring diameter over 12 inches. Therefore the clearance width taking into consideration the above criteria,

$$b_{cl} = 0.01in + 0.001 \times \left[\left(\frac{D_2 + D_1}{2}\right) - 6in\right] \quad (60)$$

3.9. Actual Pump Head

The Actual Pump Head is calculated by subtracting all the different head losses calculated from the theoretical pump head. Therefore the actual head (H_{Act}) is,

$$H_{Act} = H_{NT} - \left(h_{circ} + h_{in} + h_{sf} + h_{dL} + h_{vf} + h_{df} + h_{RL}\right) \quad (61)$$

4. Case Study

To understand and validate the described methodology, procedures are applied to estimate the performance curves for a certain model of an industrial water pump with a chosen set of process data. The pump model used for validation is a Grundfos Model No. NB 200-400/392, 4 Pole, 50 Hz, End Suction single stage centrifugal pump, Ref [10]. Table 4.1 below gives a summary of the input data used to predict the performance curves.

Table 4.1. Input Process and Mechanical Data

Service	Industrial Water
Flow Rate [Q]	364 m^3/h
Rotational Speed [N]	1493 rpm
Operating Temperature	25^0C
Fluid Density [ρ]	973.6 kg/m^3
Suction Pressure [P_1]	5.0 bara
Discharge Pressure [P_2]	10.0 bara

Required Head [H]	52.4 m
OEM Pump Efficiency [η_p]	73.1 %
Motor Rated Capacity	110 kW
OEM Impeller ID [D_2]	392 mm

5. Results

With the data presented in Table 4.1, calculations were performed and repeated as shown in Tables 7.1, 7.2, 7.3 for various range of pump flow rates to arrive at the pump performance curves as shown in below (H vs. Q, η vs. Q).

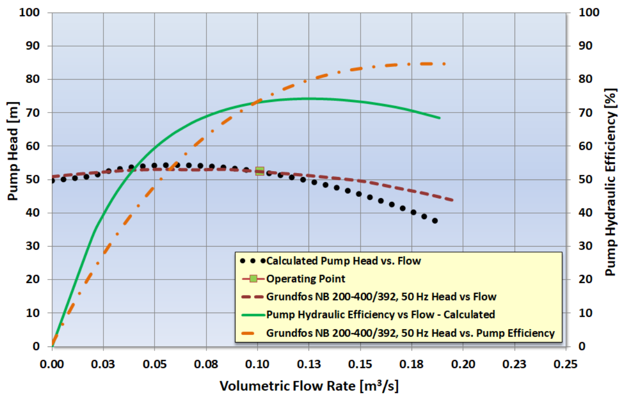


Fig 6. Calculated Pump Performance Curve

In deriving the performance curves, the min/max operable region is assigned for a range of 80% to 110% of the best efficiency point (BEP) while the preferred region of operation is 70% to 120% of BEP to minimize failure due to seal and bearing failure. A plot is made between manufacturer's data and predicted pump performance curves to assess the deviation as shown below.

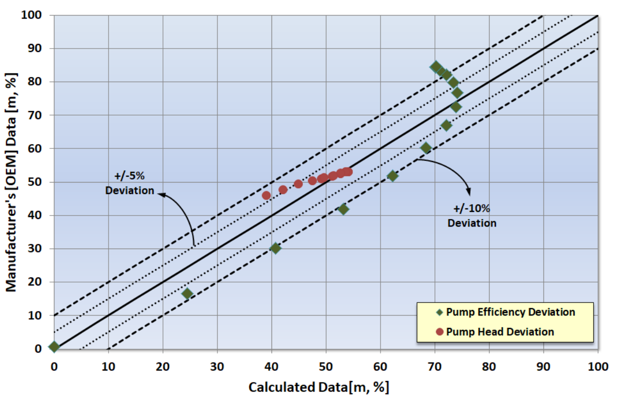


Fig 7. Deviation between Predicted and OEM Values

From the deviation calculated between the predicted pump performance data and manufacturer's data, the percentage deviation for predicted pump head is largely within ±5% for most data points. The pump hydraulic efficiency calculated however shows a deviation of most points in the range of ±10% with respect to manufacturer's data. The pump curve upon approaching shut-off head droops towards y-axis indicating a fall in head as the pump approaches zero flow. This is a characteristic of end suction centrifugal pumps where the volute friction losses begin to increase at lower flow rates contributing to a decrease in pump head as shut-off conditions approach. The key impeller geometry parameters calculated is shown in Table 5.1 as follows,

Table 5.1. Calculated Impeller Parameters

Diameter of Main Shaft End	51 mm
Hub Diameter	89 mm
Hub Length	133 mm
Diameter of Impeller Eye	197 mm
Impeller Outer Diameter	392 mm
Impeller Inner Diameter	177 mm
Vane Angle at Inlet	26.6^0
Vane Angle at Outlet	19^0
Number of Impeller Vanes	7
Blade Thickness	3.175 mm
Inlet Impeller Passage Width	57 mm
Outlet Impeller Passage Width	26 mm

5.1. Pump Losses

In capturing the pump losses experienced, which causes a departure of performance from theoretical head to actual head developed, it is seen that with an increase of inlet flow, the losses also increase with the exception of volute frictional losses. A decrease in the volute frictional losses at higher pump flow can be attributed to the decrease in tangential component of outlet velocity ($V_{\theta 2}$) when pump inlet flow (Q_s') increases.

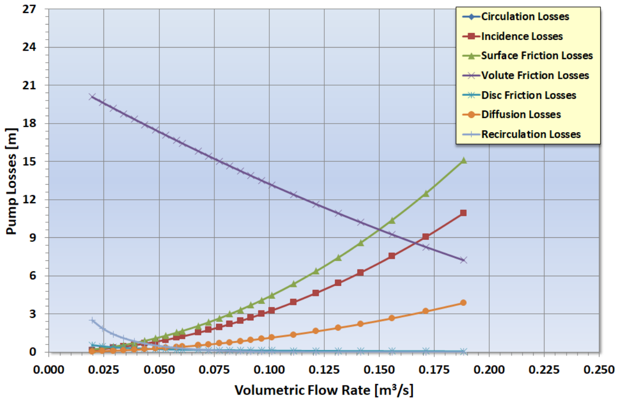


Fig 8. Pump Losses

Between the remaining pump losses the surface friction losses and Incidence losses contribute the most at higher flow. Diffusion losses also follows a similar trend, though in magnitude is smaller compared to surface friction losses and incidence losses when estimating pump head vs. flow curve [Fig. 6].

5.2. Affinity Laws

Pump flow can be treated as incompressible flow since liquids are largely incompressible that follow Fan laws. Fan laws can be used to derive performance curves for various speeds based on the following relationships.

$$Q \propto N \tag{62}$$

$$H \propto N^2 \tag{63}$$

Constants 'k_1' and 'k_2' can be estimated for the base speed of 1493 rpm by re-writing as,

$$Q = k_1 N \rightarrow k_1 = \left(\frac{Q}{N}\right)_{1493\ rpm} \tag{64}$$

$$H = k_2 N^2 \rightarrow k_2 = \left(\frac{H}{N^2}\right)_{1493\ rpm} \tag{65}$$

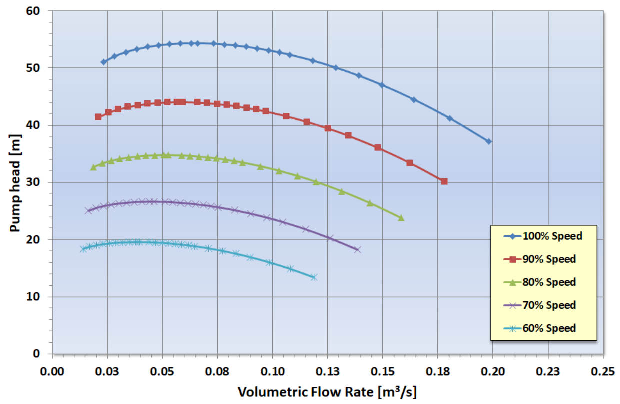


Fig 9. Pump Performance Curves – Various Speeds

With values of 'k_1'and 'k_2, H vs. Q curves for various speeds of 60%, 70%, 80% and 90%, can be computed as shown in Fig. 9.

6. Technical Notes

1. For a given set of hydraulic conditions, a centrifugal pump is designed to operate for one set of flow and head. Deviation from this operating point is allowed only to some degree.

2. Pump selection closer to the BEP will yield a more efficient pump with the least amount of vibration and radial forces acting on the shaft. Pump system resistance curve should be calculated accurately because the pump operates where the performance curve intersects the system curve.

3. In the case of single volute pumps, operating away from the BEP will cause the shaft to deflect with bearings and seals rubbing against the casing components. The fluid flow angle into the impeller will also not align to match impeller speeds and vane angles causing suction recirculation, fluid to stall and cavitation.

4. It is not always possible to operate the pump at the BEP for the conditions required and hence a flow variation of ±10% of BEP is allowed.

5. Minimum stable continuous flow (MSCF) is the minimum flow below which the pump is not allowed to operate. Although API 610 recommends that the rated region is located between 80% and 110% of BEP the preferred region of flow is between 70% and 120% of BEP.

6. Clause 6.1.12 of API 610 11th edition states *"Setting limits for preferred operating region and the location of rated flow is not intended to lead to the development of additional sizes of small pumps or preclude the use of high-specific-*

speed pumps. Small pumps that are known to operate satisfactorily at flows outside of the specified limits and high specific speed pumps that may have a narrower preferred operating region than specified should be offered..." Therefore the Allowable Operating Region is set by the manufacturer as the allowable region to operate with stability whilst conforming to predefined API 610 vibration limits.

7. The Net Positive Suction Head Available ($NPSH_A$) should always be higher than the ($NPSH_R$) required.
8. Pumps that are expected to operate less frequent can be chosen such that they operate at lower speeds at the cost of efficiency. Since the pump is selected to operate intermittently, a slightly lower efficiency pump is acceptable compared to a higher speed pump. This will ensure a longer operating life cycle.

6.1. Simplification To Estimate $V_{\theta 1}$ & $V_{\theta 2}$

To calculate the tangential component of inlet and outlet velocity ($V_{\theta 1}$, $V_{\theta 2}$) based on Impeller dimensions directly, the following simplification is made. Omitting subscripts in Eq. 9, Eq. 27, Eq. 28 and arriving at an expression for $V_{\theta 1}$ and $V_{\theta 2}$, the expression for V_θ becomes,

$$Q_s' = \pi D b V_r \varepsilon \tag{66}$$

$$V_\theta = U\sigma - \frac{V_r}{Tan\ \beta} \tag{67}$$

$$\varepsilon = 1 - \frac{Zt}{\pi D Sin\ \beta} \tag{68}$$

$$V_\theta = U\sigma - \frac{Q_s'}{Tan\ \beta \times \left[\pi D b - \frac{Ztb}{Sin\ \beta}\right]} \tag{69}$$

Using Eq. 69 with subscripts '1' and '2' for impeller inner and outer diameter respectively, the tangential velocities are calculated.

References & Further Reading

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4. *https://www.introtopumps.com*
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7. API 610, 11th Edition
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9. '*A One-Dimensional Flow Analysis for the Prediction of Centrifugal Pump Performance Characteristics',* Mohammed Ahmed El-Naggar, International Journal of Rotating Machinery Volume 2013
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Nomenclature

A_3	Volute Throat Area [m^2]
A_L	Leakage Area [m^2]
b_1	Impeller Passage Width at Inlet [m]
b_2	Impeller Passage Width at Outlet [m]
b_3	Volute Width [m]
C_L	Leakage Loss Coefficient [-]
C_m	Disc Coefficient Friction [-]
C_v	Volute Flow Coefficient [-]
d_s	Diameter of Main Shaft End [m]
D_0	Diameter of Impeller Eye [m]
D_1	Impeller Inner Diameter [m]
D_2	Impeller Outer Diameter [m]
D_2/D_1	Stepanoff Coefficient
D_3	Volute Mean Diameter [m]
D_H	Hub Diameter [m]
f	Leakage Loss Coefficient
H	Pump Head [m]
H_{Circ}	Circulation Head Loss [m]
H_{In}	Incidence Head Loss [m]
H_{sf}	Surface Friction Head Loss [m]
H_{vf}	Volute Friction Head Loss [m]
H_{df}	Disc Friction head Loss [m]
H_{dL}	Diffusion Head Loss [m]
H_L	Leakage Head Loss [m]
H_{RL}	Recirculation Head Loss [m]
H_{Actual}	Actual head Loss [m]
H_R	Hydraulic Radius [m]
k_s	Disc Surface Roughness [m]
K_u	Stepanoff Coefficient
k_{m1}	Stepanoff Coefficient
k_{m2}	Stepanoff Coefficient
L	Shaft Power [kW]
L_{AH}	Available Hydraulic Power [kW]
L_H	Hydraulic Power [kW]
m	Mass flow rate [kg/s]
N	Rotational Speed [rpm]
N_s	Pump Specific Speed [rpm]
P_1	Suction Flange Pressure [bara]
P_2	Discharge Flange Pressure [bara]
$P_{Required}$	Power Required [kW]
Q_s	Flow Rate [m^3/s]
Q_s'	Total Flow Rate [m^3/s]
Q_N	Maximum Flow Rate [m^3/s]
R_1/R_2	Radius Ratio [-]
Re	Reynolds Number [-]
s	Axial gap [m]
S_s	Shaft permissible Shear Stress [psi]
t	Blade Thickness [m]
t_1	Thickness of Impeller Passage at Inlet [m]
t_2	Thickness of Impeller Passage - Outlet [m]
U_2	Impeller OD Tip Speed [m]
U_1	Impeller ID Tip Speed [m]
V_{eye}	Velocity of Impeller Eye [m/s]
V_{r1}	Radial Velocity of Flow at Inlet [m/s]
V_{r2}	Radial Velocity of Flow at Outlet [m/s]
$V_{\theta 1}$	Tangential Velocity of Flow at Inlet [m/s]
$V_{\theta 2}$	Tangential Velocity of Flow - Outlet [m/s]
$V_{\theta 1}'$	Actual Whirl Velocity Flow at Inlet [m/s]
$V_{\theta 2}'$	Actual Whirl Velocity Flow at Outlet [m/s]
V_{s1}	Slip Velocity at Inlet [m/s]
V_{s2}	Slip velocity at Outlet [m/s]
W_1	Relative Velocity at Inlet [m/s]
W_2	Relative Velocity at Outlet [m/s]
Z	Number of Impeller Vanes [-]
β_1	Vane Angle at Inlet [degrees]
β_2	Vane Angle at Outlet [degrees]
ε_{limit1}	Limiting Radius Ratio at Inlet [-]
ε_{limit2}	Limiting Radius Ratio at Outlet [-]
ε_1	Contraction Factor at Inlet [-]
ε_2	Contraction Factor at Outlet [-]
ρ	Liquid Density [kg/m^3]
η_p	Pump Efficiency [%]
η_v	Volumetric Efficiency [%]
σ_{s1}	Slip Value at Inlet [-]
σ_{s2}	Slip Value at Outlet [-]
μ	Liquid Viscosity [kg/m.s]
ω	Angular Velocity [m/s]

Appendix A

A Set of calculations is presented to demonstrate the methodology for pump data presented.

Table 7.1. Impeller Dimensions Calculations

Shaft Dimensions	
Shaft Power [L]	$L = \frac{m \times H}{\eta_0} = \frac{0.1011 \, m^3/_s \times 973.6 \, kg/_{m^3} \times \left(\frac{52.4}{102.04}\right) kJ/_{kg}}{0.722} = 70 \, kW$
Pump Specific Speed [N_s]	$N_s = \frac{N \times \sqrt{Q}}{H^{3/4}} \left(\frac{rpm.\frac{m^3}{min}}{m}\right) = \frac{1493 \times \sqrt{6.067}}{52.4^{3/4}} = 189$
Main Shaft End Diameter [d_{sh}]	$D_{sh} = \left[\frac{P(HP) \times 321000}{N(rpm) \times S_s(psi)}\right]^{1/3} = \left[\frac{147 \times 321000}{1493 \times 4000}\right]^{1/3} = 1.99 \, inch \, (\sim 51 \, mm)$
Volumetric Efficiency [η_v]	$\eta_v = \frac{1}{1 + \frac{1.124}{N_s^{2/3}}} = \frac{1}{1 + \frac{1.124}{189^{2/3}}} = 0.967 \, or \, 96.7\%$
Hub Dimensions	
Hub Diameter [D_H]	$D_H = (1.5 \, to \, 2.0) \times D_{sh}, Taking \, 1.75, D_H = 1.75 \times 51 = 89 \, mm$
Hub Length [L_H]	$L_H = (1.0 \, to \, 2.0) \times D_H, Taking \, 1.5, L_H = 1.5 \times 89 = 133 \, mm$
Impeller Dimensions	
Total Flow Rate [Q_s']	$Q_s' = \frac{Q}{\eta_v} = \frac{0.1011}{0.967} = 0.1045 \, m^3/_s$
Velocity of Liquid at Impeller Eye [V_{eye}]	$V_{eye} = [(0.07 \, to \, 0.11) + 0.00023 N_s] \times \sqrt{2gH}, Taking \, 0.09$ $V_{eye} = [0.09 + (0.00023 \times 189)] \times \sqrt{2 \times 9.81 \times 52.4} = 4.3 \, m/s$
Diameter of Impeller Eye [D_0]	$D_0 = \sqrt{\frac{4 \times Q_s'}{\pi \times V_{eye}} + D_H^2} = \sqrt{\frac{4 \times 0.1045}{\pi \times 4.3} + \left(\frac{89}{1000}\right)^2} = 0.197 m \, or \, 197 mm$
Coefficients 'K_U'	1.043 (From Stepanoff Charts, Fig. 4)
Coefficients 'K_{m1}'	0.149 (From Stepanoff Charts, Fig. 4)
Coefficients 'K_{m2}'	0.113 (From Stepanoff Charts, Fig. 4)
Coefficients 'D_2/D_1'	0.452 (From Stepanoff Charts, Fig. 4)
Angular Velocity [ω]	$\omega = \frac{2 \times \pi \times N_{rpm}}{60} = \frac{2 \times \pi \times 1493}{60} = 156 \, m/s$
Impeller Outer Diameter [D_2]	Selecting D_2 = 0.392m (392 mm) from Table 4.1

Impeller OD Tip Speed [U_2]	$U_2 = \frac{\omega \times D_2}{2} = \frac{156 \times 0.392}{2} = 30.6\ m/s$
Impeller Inner Diameter [D_1]	$D_1 = D_2 \times \left(\frac{D_2}{D_1}\right) = 0.392 \times 0.452 = 0.177 m\ or\ 177\ mm$
Impeller ID Tip Speed [U_1]	$U_1 = \frac{\pi \times N \times D_1}{60} = \frac{\pi \times 1493 \times 0.177}{60} = 13.9\ m/s$
Inlet Flow Radial Velocity [V_{r1}]	$V_{r1} = K_{m1}\sqrt{2gH} = 0.149 \times \sqrt{2 \times 9.81 \times 52.4} = 4.8\ m/s$
Outlet Flow Radial Velocity [V_{r2}]	$V_{r2} = K_{m2}\sqrt{2gH} = 0.113 \times \sqrt{2 \times 9.81 \times 52.4} = 3.6\ m/s$
Vane Dimensions	
Vane Angle at Outlet [β_2]	Assume β_2=26.6^0 (Note: To be solved iteratively till H_{Actual}=$H_{Required}$)
Vane Angle at Inlet [β_1]	$\beta_1 = \tan^{-1}\left(\frac{V_{r1}}{U_1}\right) = \tan^{-1}\left(\frac{4.8}{13.9}\right) \approx 19^0$
Number of Impeller Vanes [Z]	$Z = 6.5 \times \left(\frac{D_2+D_1}{D_2-D_1}\right) \times \sin\left(\frac{\beta_1+\beta_2}{2}\right) = 6.5 \times \left(\frac{0.392+0.177}{0.392-0.177}\right) \times \sin\left(\frac{26.6+19}{26.6-19}\right) \approx 7$
Check $\frac{R_1}{R_2} \le \varepsilon_{\lim it}$	$\varepsilon_{limit2} = e^{\left(-\frac{8.16 \sin \beta_2}{Z}\right)} = e^{\left(-\frac{8.16 \times \sin(26.6)}{7}\right)} = 0.594$ $\varepsilon_{limit1} = e^{\left(-\frac{8.16 \sin \beta_1}{Z}\right)} = e^{\left(-\frac{8.16 \times \sin(19)}{7}\right)} = 0.685$ $\frac{R_1}{R_2} = \frac{0.177}{0.392} = 0.452 < \varepsilon_{\lim it1} < \varepsilon_{\lim it2}$, Therefore, $\sigma = 1 - \frac{\sqrt{Sin\beta_2}}{\beta_1 Z^{0.7}}$
Slippage Factor [σ_{s1}]	$\sigma_{s1} = 1 - \frac{\sqrt{Sin\beta_2}}{\beta_1 Z^{0.7}} = 1 - \frac{\sqrt{Sin(26.6)}}{19 \times 7^{0.7}} = 0.991$
Slippage Factor [σ_{s2}]	$\sigma_{s2} = 1 - \frac{\sqrt{Sin\beta_2}}{\beta_1 Z^{0.7}} = 1 - \frac{\sqrt{Sin(26.6)}}{19 \times 7^{0.7}} = 0.991$
Blade Thickness [t]	Taking, 0.125 inches (0.0032m)
Thickness of Inlet Impeller Passage [t_1]	Taking, 0.3175 inches (0.0081m)
Thickness of Outlet Impeller Passage [t_2]	Taking, 0.3175 inches (0.0081m)
Inlet Contraction Factor [ε_1]	$\varepsilon_1 = 1 - \frac{Zt_1}{\pi \times D_1 Sin\beta_1} = 1 - \frac{7 \times 0.0081}{\pi \times 0.177 \times Sin(19)} = 0.688$
Outlet Contraction Factor [ε_2]	$\varepsilon_2 = 1 - \frac{Zt_2}{\pi \times D_2 Sin\beta_2} = 1 - \frac{7 \times 0.0081}{\pi \times 0.392 \times Sin(26.6)} = 0.898$
Inlet Impeller Passage Width [b_1]	$b_1 = \frac{Q_S'}{\pi \times D_1 \times V_{r1} \times \varepsilon_1} = \frac{0.1045}{\pi \times 0.177 \times 4.8 \times 0.688} \approx 0.057\ m\ (57mm)$
Outlet Impeller Passage Width [b_1]	$b_2 = \frac{Q_S'}{\pi \times D_2 \times V_{r2} \times \varepsilon_2} = \frac{0.1045}{\pi \times 0.392 \times 3.6 \times 0.898} \approx 0.026\ m\ (26mm)$

Clearance Width (b_{cl})	$b_{cl} = 0.01in + 0.001 \times \left[\left(\frac{D_2 + D_1}{2} \right) - 6in \right]$ $b_{cl} = 0.01 + 0.001 \times \left[\left(\frac{0.392 + 0.177}{2} \times \frac{1000}{25.4} \right) - 6 \right] = 0.0152\,in(\sim 0.4\,mm)$
Leakage Area (A_L)	$A_L = \pi \times D_1 \times b_{cl} = \pi \times 0.177 \times \frac{0.4}{1000} = 0.000223\ m^2$
Leakage Head Loss [H_L]	$H_L = \frac{3}{4} \left[\frac{\left(U_2^2 - U_1^2\right)}{2 \times g} \right] = \frac{3}{4} \times \left[\frac{\left(30.6^2 - 13.9^2\right)}{2 \times 9.81} \right] = 28.5\,m$
Leakage Head Loss [Q_L]	$Q_L = 0.6 \times 0.000223 \times \sqrt{2 \times 9.81 \times 28.5} = 0.00317\,m^3/s$
Outlet Tangential Velocity [$V_{\theta1}$]	Adding Leakage Loss, $Q_s' = Q_s' + Q_L$, $V_{\theta 1} = U_1 \sigma_1 - \frac{Q_s'}{Tan\beta_1 \times \left[\pi D_1 b_1 - \left(\frac{Z \times t \times b_1}{Sin\beta_1}\right)\right]}$ $V_{\theta 1} = (13.9 \times 0.991) - \frac{0.1045 + 0.00317}{Tan(19^0) \times \left[\pi \times 0.177 \times 0.057 - \left(\frac{7 \times 0.0032 \times 0.057}{Sin(19^0)}\right)\right]} \approx 2.7\ m/s$
Outlet Tangential Velocity [$V_{\theta2}$]	$V_{\theta 2} = U_2 \sigma_2 - \frac{Q_s'}{Tan\beta_2 \times \left[\pi D_2 b_2 - \left(\frac{Z \times t \times b_2}{Sin\beta_2}\right)\right]}$ $V_{\theta 2} = (30.6 \times 0.991) - \frac{0.1045 + 0.00317}{Tan(26.6^0) \times \left[\pi \times 0.392 \times 0.026 - \left(\frac{7 \times 0.0032 \times 0.026}{Sin(26.6^0)}\right)\right]} \approx 23.4\ m/s$
Actual Whirl Velocity at Inlet [$V_{\theta1}'$]	$V_{\theta 1}' = U_1\left(2 - \sigma_{s1}\right) - V_{r1} Cot\beta_1 = \left[13.9 \times \left(2 - 0.991\right)\right] - \frac{4.8}{Tan(19)} = 0.09\,m/s$
Actual Whirl Velocity at Inlet [$V_{\theta2}'$]	$V_{\theta 2}' = U_2 \sigma_{s2} - V_{r2} Cot\beta_2 = (30.6 \times 0.991) - \frac{3.6}{Tan(26.6)} \approx 22.9\ m/s$
Slip Velocity at Inlet [V_{s1}]	$V_{s1} = V_{\theta 1}' - V_{\theta 1} = 0.09 - 2.7 = -2.6\ m/s$
Slip Velocity at Inlet [V_{s2}]	$V_{s2} = V_{\theta 2} - V_{\theta 2}' = 23.4 - 22.9 = 0.5\ m/s$
Net Theoretical Head [$H_{NetTheoretical}$]	$H_{Net\ Theoretical} = \frac{1}{g}\left[U_2 V_{\theta 2}' - U_1 V_{\theta 1}'\right] = \frac{[(30.6 \times 22.9) - (13.9 \times 0.09)]}{9.81} = 72\ m$
Theoretical Power Absorbed [$P_{Theoretical}$]	$P_{Net\ Theoretical} = \frac{Q_s' \times \rho \times H}{102.04} = \frac{(0.1045 + 0.00317) \times 973.6 \times 72}{102.04} = 74.1\ kW$
Theoretical Shut-off Head [$H_{shut\text{-}off}$]	$H_{Shut-off} = \frac{U_2^2 - U_1^2}{g} = \frac{30.6^2 - 13.9^2}{9.81} \approx 75.6\,m$

Table 7.2. Pump Losses Calculations

Circulation Losses	
Circulation Head Loss	$H_{Circ} = \frac{(U_2 V_{s2} + U_1 V_{s1})}{g} = \frac{(30.6 \times 0.5) + \left(13.9 \times (-2.6)\right)}{9.81} \approx -2.5\ m/s$

Inlet Incidence Losses	
Inlet Incidence Head Loss	$h_{in} = f_{in} \times \frac{(U_1 - V_{\theta 1})^2}{2g}$, Taking, f_{in}=0.5, $h_{in} = 0.5 \times \frac{(13.9 - 2.7)^2}{2 \times 9.81} = 3.2\,m$
Surface Friction Losses	
Inlet Relative Velocity [W_1]	$W_1 = \frac{V_{r1}}{Sin\beta_1} = \frac{4.8}{Sin(19)} \approx 15\,m/s$
Inlet Relative Velocity [W_2]	$W_2 = \frac{V_{r2}}{Sin\beta_2} = \frac{3.6}{Sin(26.6)} \approx 8\,m/s$
Hydraulic Diameter [D_H]	$H_R = \frac{b_2\left(\frac{\pi D_2}{Z}\right) \times Sin\beta_2}{b_2 + \left(\frac{\pi D_2}{Z}\right) \times Sin\beta_2} = \frac{0.026 \times \left(\frac{\pi \times 0.392}{7}\right) \times Sin(26.6)}{0.026 + \left(\frac{\pi \times 0.392}{7}\right) \times Sin(26.6)} \approx 0.02m$
Surface Friction Losses	$h_{sf} = \frac{b_2 \times (D_2 - D_1) \times (W_1 + W_2)^2}{2 \times Sin\beta_2 \times H_r \times 4g}$; $h_{sf} = \frac{0.026 \times (0.392 - 0.177) \times (8+15)^2}{2 \times Sin(26.6) \times 0.02 \times 4 \times 9.81} \approx 4.5\,m$
Volute Friction Losses	
Volute Throat Diameter [D_3]	Assuming, $D_3 = 1.3 \times D_2 = 1.3 \times 0.392 = 0.51\,m$
Volute Width [b_3]	$b_3 = 2 \times b_2 = 2 \times 0.026 = 0.052\,m$
Volute Throat Area [A_3]	$A_3 = \pi \times D_3 \times b_3 = \pi \times 0.5096 \times 0.052 = 0.084\,m^2$
Volute Friction Loss Head	$h_{vf} = 0.8 \times \frac{\left[V_{\theta 2} \times \left(\frac{D_2}{D_3}\right)\right]^2 - \left[\frac{Q_s^{'}}{A_3}\right]^2}{2g}$ $h_{vf} = 0.8 \times \frac{\left[23.4 \times \left(\frac{0.392}{0.51}\right)\right]^2 - \left(\frac{0.1034 + 0.00317}{0.084}\right)^2}{2 \times 9.81} \approx 13.2\,m$
Disc Friction Losses	
Disc Friction Loss Head	Taking Disc Surface Roughness [k_s] = 5 microns (5 x 10^{-6}) Axial Gap [s] = 12.7 mm (1.27 x 10^{-2}) Viscosity of Water at 25^0C = 0.00091 kg/m.s
Reynolds Number	$\mathrm{Re} = \frac{U_2 \times D_2/2}{\mu} = \frac{30.6 \times 0.392/2 \times 973.6}{0.00091} = 6{,}416{,}773$
Disc Coefficient Friction [C_m]	$C_M = \left(\frac{k_s}{0.5 \times D_2}\right)^{0.25} \times \left(\frac{s}{0.5 \times D_2}\right)^{0.1} \times \mathrm{Re}^{-0.2}$ $C_M = \left(\frac{5 \times 10^{-6}}{0.5 \times 0.392}\right)^{0.25} \times \left(\frac{0.0127}{0.5 \times 0.392}\right)^{0.1} \times 6416773^{-0.2} = 2.35 \times 10^{-3}$

Disc Friction Loss Head	$$h_{df} = \frac{0.10204 \times C_M \times \omega^3 \times \left(\frac{D_2 - D_1}{2}\right)^5}{Q_s^{'}}$$ $$h_{df} = \frac{0.10204 \times 2.35 \times 10^{-3} \times 156^3 \times \left(\frac{0.392 - 0.177}{2}\right)^5}{0.1045 + 0.00317} \approx 0.1m$$

Recirculation Losses

Recirculation Loss Head	$h_{RL} = 0.005 \times \frac{\omega^3 \times D_1^2}{\rho Q}\left(1 - \frac{Q}{Q_0}\right)^{2.5}$, Here 0.005 is replaced with 0.00075

Taking Max Flow Rate, Q_0= 110% of Rated Flow = 1.1 x 1.1 =0.11 m^3/s

$$h_{RL} = 0.00075 \times \frac{\omega^3 \times D_1^2}{\rho Q_s^{'}}\left(1 - \frac{Q_s^{'}}{Q_0}\right)^{2.5}$$

$$h_{RL} = 0.00075 \times \frac{156^3 \times 0.177^2}{973.6 \times (0.1045 + 0.00317)} \times \left(1 - \frac{0.1045 + 0.00317}{0.11}\right)^{2.5} \approx 0m$$

Diffusion Losses

Diffusion Loss Head	$W_1 = \left(\frac{V_{r1}}{Sin\beta_1}\right) = \left[\frac{4.8}{Sin(19)}\right] \approx 15m/s$; $W_2 = \left(\frac{V_{r2}}{Sin\beta_2}\right) = \left[\frac{3.6}{Sin(26.6)}\right] \approx 8\ m/s$ $\frac{W_1}{W_2} = \frac{15}{8} = 1.875 > 1.4$ $h_{DL} = 0.25 \times \left[\left(\frac{W_1}{W_2}\right)^2 - 2\right]\frac{W_2^2}{2g} = 0.25 \times \left[\left(\frac{15}{8}\right)^2 - 2\right] \times \frac{8^2}{2 \times 9.81} = 1.24m$

Table 7.3. Total Losses Calculations

Actual Head [H_{Actual}]	$H_{Actual} = H_{Net\,Theoretical} - (h_{circ} + h_{in} + h_{sf} + h_{vf} + h_{df} + h_{RL} + h_{DL})$ $\boldsymbol{H_{Actual} = 72 - (-2.5 + 3.2 + 4.5 + 13.2 + 0.1 + 0 + 1.24) = 52.2\ m}$
Required Power [$P_{Required}$]	$P_{Required} = \frac{Q_s^{'} \times \rho \times H_{Act}}{102.04} + \sum Losses = \frac{(0.1045 + 0.00317) \times 973.6 \times 52.2}{102.04} + 20.7 = 74.3kW$
Pump Efficiency [η_P]	$\eta_P = \frac{H_{Required}}{H_{Required} + H_{Losses}} = \frac{52.2}{52.2+19.8} = 72.5 \approx 73\%$

Note: The above set of calculations shown is made in MS-EXCEL which performs detailed calculations. In the above Tables 7.1/7.2/7.3, calculations are shown for rounded-off numbers.

Module 4

Affinity Laws for Variable Speed Centrifugal Pumps

The world is becoming more conscious of greenhouse gases (GHG) & it's estimated that nearly 10% of the electricity usage is from operating pumps ranging from domestic pumps, sewerage pumps, air conditioning and in every other industrial application. The mechanical aspect of centrifugal pumps have changed fairly little in the last 5 decades, but what has brought a vast change in pump performance is the control system based on a variable frequency drive (VFD).

A centrifugal pump consists of an impeller in a casing that raises the fluid's head for a given speed & discharges liquid at a desired pressure. For applications that require attending to a variable flow scenario, a flow control valve is installed at the pump discharge that throttles fluid pressure to achieve the desired flow. But such methods cause a loss of energy that was initially used to raise the fluid's pressure in the pump.

With the advent of variable speed drives consisting of a pressure sensor & piece of circuitry that alters the frequency of the electric current, the pump's speed parameter can be altered to achieve the required flow & also avoid throttling using a flow control valve. VFD's although tend to cause a temperature rise in the circuitry, sometimes require ventilation systems being incorporated for cooling purposes.

Advantages of Variable Frequency Drives

1. VFD's when newly fitted or retrofitted to rotating machinery such as pumps are referred to as variable speed drives (VSD). For applications where the duty is expected to be constant without much variation in process conditions, a fixed speed drive (FSD) should be more cost effective. But VSD's are mostly suitable for pumping applications where the pump duty is not expected to be constant.
2. Noise & Vibrations are reduced when running at lower speeds.
3. Consumers that use only a small portion of rated flow during varying loads would have the pump running at full load speed corresponding to full load power. In such situations, VFD's help alter the speed to consume less power during operation.
4. VFD's reduce the risk of a motor burnout during start-up from excessive in-rush current & increases the longevity of the equipment.

When VFD's Are Not Advantageous

1. VFD's are not to compensate for an improperly selected pump.
2. In systems with high resistance like boiler feed water (BFW) pumps where the pump has to generate high starting torque to overcome the static head, VFD's do not offer much in terms of providing high start-up torque.
3. Performance curves that cause the operating point to fall off the Q vs. H curves cause the operating point to operate closer to the stall region at lower speeds which can cause cavitation. In such situations, a new pump with a discharge throttle valve is required to push the operating point into the operating envelope which defeats the purpose of retrofitting with a VSD.
4. At lower speeds, though noise & vibrations are reduced, chances exist for structural resonance that can compromise the integrity of the bearing house and support structures.

Selection Process - New Pumps

1. For new pumps its common to oversize the pump but this is not recommended as it adds higher initial cost & higher life cycle costs.
2. When selecting a rotodynamic pump in combination with a VSD for a system with some static head, a pump should be chosen such that the maximum flow rate is slightly to the right-hand side of the best efficiency point (BEP). The exception is for a constant flow regulated system, in which case the recommendation is to select a pump that operates to the left hand side of BEP at maximum pressure. This approach optimizes pump operating efficiency.
3. Some operating profiles may be satisfied best by installing multiple pumps, which could be fixed or variable speed. On/off control can be used to vary flow rate for systems in which intermittent flow is acceptable.

Selection Process - Retrofit Pumps

1. Often a contingency of 20% - 25% on the required system head is added. Therefore retrofitting with VSD's could match pump systems to actual system requirements more accurately to save considerable amounts of energy.
2. When adding a VSD to an existing electric AC motor, the electrical characteristics of the motor & the frequency converter must match. Variable frequency drives work on the principle of altering the frequency of the incoming current using a frequency converter that produces a change in the synchronous speed of the motor for a given number of poles. Therefore frequency converters that give smaller levels of harmonic current distortion is to be chosen to avoid over heating the motor windings and avoid the risk of premature failure.

Performance Curves for VSD Retrofit

Centrifugal Pumps that run on a fixed speed are characterized by a single Q vs. H curve. In the event of a retrofit with a VFD, the pump can operate at various other speeds & correspondingly would have their respective Q vs. H curves.

To estimate the Q vs. H curves at other speeds, Fan Laws a.k.a Affinity Laws can be used. Affinity Laws are used under the premise that

1. Liquids are largely incompressible and their density [ρ] remains fairly constant.
2. Frictional losses due to impeller & casing construction as well as bearing losses exist at lower speeds but are considered to be lower than the losses experienced at 100% speed.

As per Affinity Laws, Pump speed [N] is related to the Pump Flow [Q], Pump Head [H] & Pump Hydraulic Power [P] as,

$$Q = k_1 N \tag{1}$$

$$H = k_2 N^2 \tag{2}$$

$$P = k_3 N^3 \tag{3}$$

Where k_1, k_2, k_3 are constants.

Once the constants k_1, k_2, k_3 are estimated for the rated curve [100% speed], the Q vs. H curves can be estimated for other speeds, i.e., 90%, 80%, 70%, 60%, 50%, 40%, 30%. It is to be noted that the pump efficiency for a given flow range and for various speeds would be fairly constant and these are referred to as Constant Efficiency [η] lines.

To demonstrate the use of Affinity Laws to derive Q vs. H curves for various speeds, a case study is shown. A motor driven centrifugal pump operating at 50 Hz, delivers water from 5 bara suction pressure to 10 bara discharge pressure. The process parameters & performance curves are as follows,

Table 1. Pump Process Parameters

Parameter	Value
Service	Water
Operating Capacity	372 m³/h
Pump Head	52.4 m
Rotational Speed	1493 rpm
Suction Flange Pressure	5.0 bara
Discharge Flange Pressure	10.0 bara
Liquid Density	973.6 kg/m³

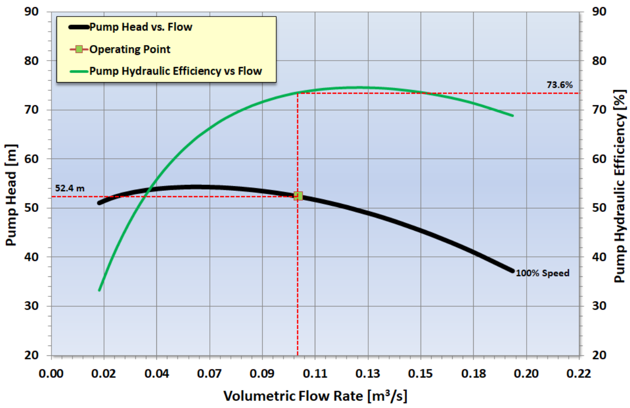


Figure 1. Pump Performance Curves

Pump speeds chosen for retrofitting are,

Table 2. Electric Motor Speeds

Speed [%]	Speed [rpm]
100	1493
90	1344
80	1194
70	1045
60	896

Design Methodology

As described previously, fan constants k_1, k_2, k_3 are estimated for each volumetric flow rate (Q) & corresponding head (H) for 100% speed. Using these k values, Q vs. H curves is calculated for various speeds [Table 2].

$$k_{1,100\%} = \frac{Q}{1493} \quad (4)$$

$$k_{2,100\%} = \frac{H}{1493^2} \quad (5)$$

$$k_{3,100\%} = \frac{P}{1493^3} \quad (6)$$

The hydraulic efficiency is estimated as,

$$\eta = \frac{Q \times \rho \times H}{P} \quad (7)$$

Tabulating the Q vs. H values for the 100% speed case, the values of k_1, k_2 & k_3 are,

Table 3. Fan Law Constants for 100% Speed

Q	H	P	$K_{1,100\%}$	$K_{2,100\%}$	$K_{3,100\%}$
[m³/s]	[m]	[kW]	[m³/s/rpm]	[m/rpm²]	[kW/rpm³]
0.024	51.1	34.9	1.60E-05	2.29E-05	1.05E-08
0.039	53.3	40.5	2.61E-05	2.39E-05	1.22E-08
0.049	54.0	45.0	3.29E-05	2.42E-05	1.35E-08
0.067	54.3	53.7	4.50E-05	2.44E-05	1.61E-08
0.085	53.9	62.4	5.68E-05	2.42E-05	1.88E-08
0.090	53.7	64.9	6.02E-05	2.41E-05	1.95E-08
0.095	53.4	67.4	6.36E-05	2.40E-05	2.03E-08
0.100	53.1	69.9	6.70E-05	2.38E-05	2.10E-08
0.105	52.7	72.3	7.04E-05	2.37E-05	2.17E-08
0.110	52.4	74.6	7.36E-05	2.35E-05	2.24E-08
0.120	51.3	79.5	8.07E-05	2.30E-05	2.39E-08
0.131	50.1	84.1	8.79E-05	2.25E-05	2.53E-08
0.152	47.1	92.4	1.02E-04	2.11E-05	2.78E-08
0.202	37.2	104.1	1.35E-04	1.67E-05	3.13E-08

Using 100% speed k_1, k_2, k_3 values, Q, H, P, η for other speeds shown in Table 2, are,

$$Q_{90\%} = k_{1,100\%} \times N_{90\%} \quad (8)$$

$$H_{90\%} = k_{2,100\%} \times N_{90\%}^2 \quad (9)$$

$$P_{90\%} = k_{3,100\%} \times N_{90\%}^3 \quad (10)$$

$$\eta_{90\%} = \frac{Q_{90\%} \times \rho \times [H_{90\%}/102.04]}{P_{90\%}} \quad (11)$$

$$Q_{80\%} = k_{1,100\%} \times N_{80\%} \quad (12)$$

$$H_{80\%} = k_{2,100\%} \times N_{80\%}^2 \quad (13)$$

$$P_{80\%} = k_{3,100\%} \times N_{80\%}^3 \quad (14)$$

$$\eta_{80\%} = \frac{Q_{80\%} \times \rho \times [H_{80\%}/102.04]}{P_{80\%}} \quad (15)$$

$$Q_{70\%} = k_{1,100\%} \times N_{70\%} \quad (16)$$

$$H_{70\%} = k_{2,100\%} \times N_{70\%}^2 \quad (17)$$

$$P_{70\%} = k_{3,100\%} \times N_{70\%}^3 \quad (18)$$

$$\eta_{70\%} = \frac{Q_{70\%} \times \rho \times [H_{70\%}/102.04]}{P_{70\%}} \quad (19)$$

$$Q_{60\%} = k_{1,100\%} \times N_{60\%} \quad (20)$$

$$H_{60\%} = k_{2,100\%} \times N_{60\%}^2 \quad (21)$$

$$P_{60\%} = k_{3,100\%} \times N_{60\%}^3 \quad (22)$$

$$\eta_{60\%} = \frac{Q_{60\%} \times \rho \times [H_{60\%}/102.04]}{P_{60\%}} \quad (23)$$

Note: 1 kJ/kg = 102.04 m.

With the above set of calculations made, for each calculated values of Q, H, P, η for speeds of 90%, 80%, 70%, 60%, the pump performance curves at constant efficiency is,

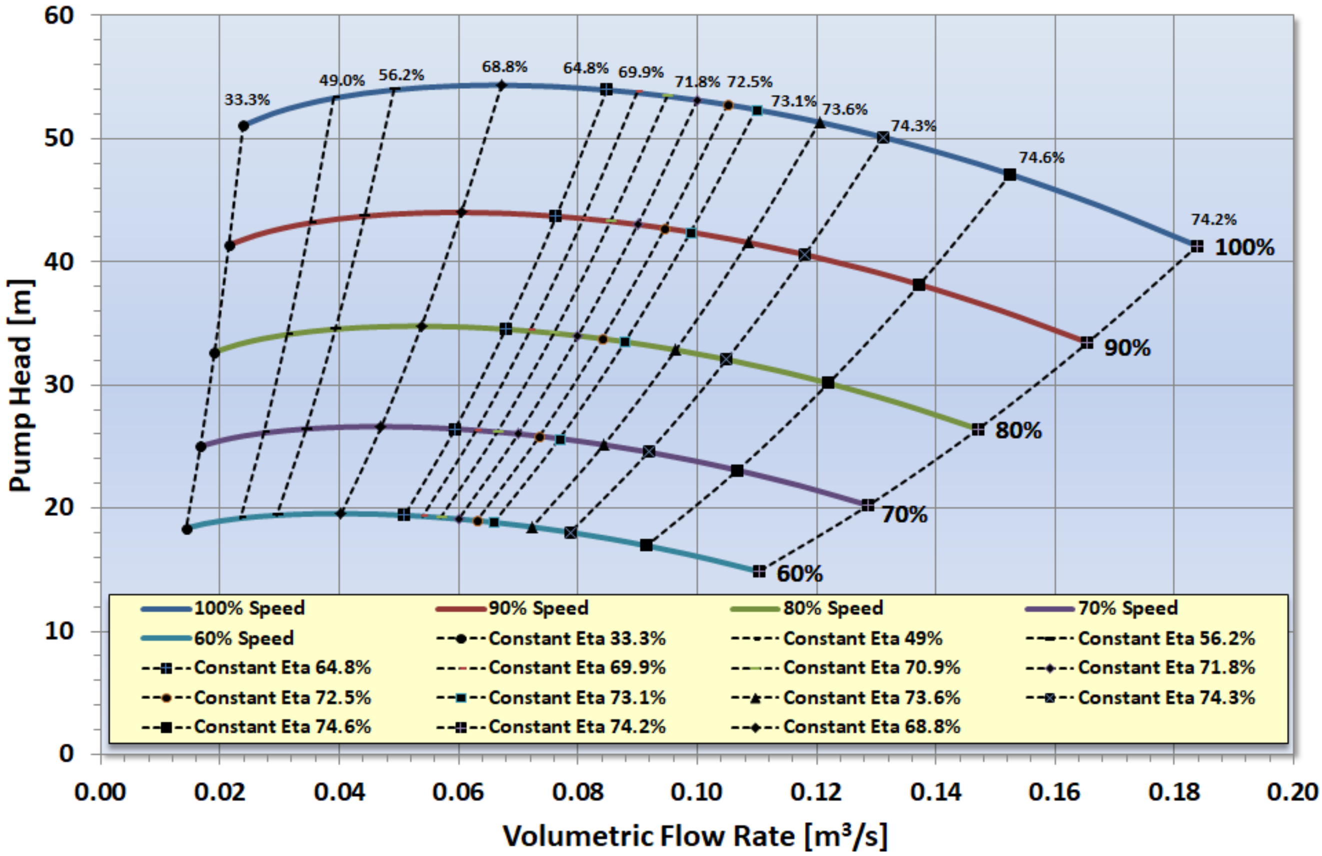


Figure 2. Performance Curves at Constant Efficiency for Various Speeds

References & Further Reading

1. "*Variable Speed Pumping - A Guide to Successful Application*", US Department of Energy, Energy Efficiency and Renewable Energy
2. https://southernswater.com.au/variable-speed-pumps/
3. http://jmpcoblog.com/hvac-blog/modern-pump-selection-part-4-why-you-should-never-over-head-a-variable-speed-pump
4. https://product-selection.grundfos.com

Appendix A

Fluid Head Conversion - 'metres' to 'kJ/kg'

$1\,J = 1.N.m$

Multiplying and Dividing by 1000 and kg_f,

$$1000\frac{J}{kg_f} = 1000\frac{N.m}{kg_f}$$

Taking 1 kg_f = 9.8 N and substituting in above,

$$1000\frac{J}{kg_f} = 1000\frac{N.m}{9.8\,N} \rightarrow 1000\frac{J}{kg_f} = 102.04\,m$$

$$\frac{1\,kJ}{kg_f} = 102.04\,m$$

Module 5

Understanding Centrifugal Compressor Surge and Control

Ask a chemical or mechanical engineer, what does a compressor surge do, and he would shudder merely thinking of the consequences. The centrifugal compressor is the heart of any oil & gas facility and since the last 100 years has been subjected to scrutiny as to what is the perfect control mechanism. Surge in a centrifugal compressor can be simply defined as a situation where a flow reversal from the discharge side back into the compressor casing occurs causing mechanical damage.

The reasons are multitude ranging from driver failure, power failure, upset process conditions, start up, shutdown, failure of anti-surge mechanisms, check valve failure to operator error to name a few. The consequences of a surge are more mechanical in nature whereby ball bearings, seals, thrust bearing, collar shafts, impellers wear out and sometimes depending on the how powerful are the surge forces, cause fractures to the machinery parts due to excessive vibrations.

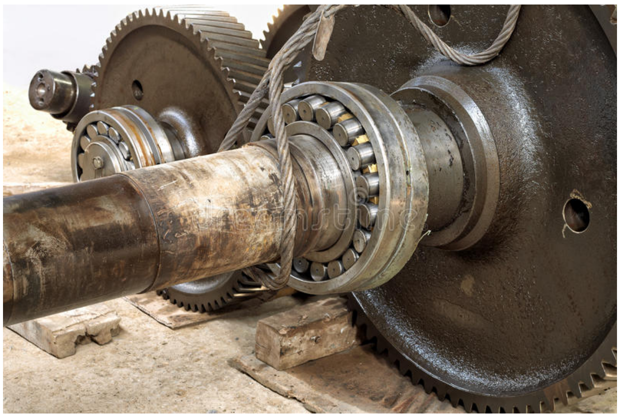


Figure 1. Bearings dislodged from containment

Here is an image that shows the bearings being dislodged from its containment. The effects of surge are also contagious and due to excessive shaft vibrations, the gearbox connected between the compressor and the driver is also not spared at the bearings and gear teeth. The power of a surge is also proportional to the capacity (flow, power, pressure ratio) and even in the case of small turbo compressors, the gear teeth wear out when the impeller rotates in the opposite direction during a surge. The bottom line is - *Always Avoid a Surge in Rotating Equipment.*

Typical Single Stage Compression System

A typical single stage compressor system shown in Figure 1, consists of,

1. A centrifugal compressor driven by a gas turbine, steam turbine or electric motor.
2. A suction scrubber to disengage any carryover liquids that can potentially wear out impellers that run at high velocities of the order of 200 m/s to 500 m/s.
3. A discharge cooler to cool the compressed vapours to the required export temperature.
4. Check valves at compressor discharge to prevent any backflow of vapours into the compressor in the event of a surge.
5. An anti-surge valve (ASV) that recycles cold gas from the discharge cooler to the suction to keep the operating point away from the surge line.
6. A hot gas recycle is included, if the ASV is inadequate.

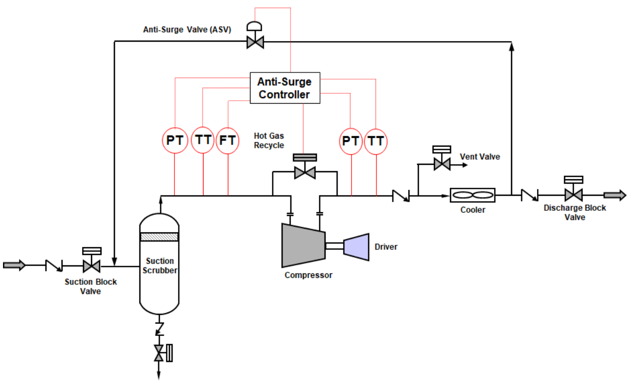


Figure 2. Typical Centrifugal Compression System

Compressor Surge Protection Agents

Anti-surge Valve (ASV)

The chief protecting agent in a centrifugal compressor is the anti-surge system that consists of a control valve with the associated piping. The ASV recycles cold gas from the discharge side cooler back to the compressor via the suction scrubber to keep the operating point away from the surge line.

Hot Gas Recycle Valve (HGV)

Although the anti-surge valve is the chief protector, in brownfield projects, often the ASV becomes inadequate to deal with a compressor surge due to addition of new compressors in parallel or series (e.g., booster compressors), change of plant piping or change of vapour composition. In such situations, a necessity arises to recycle more flow for which an additional ASV with quick opening characteristics is installed in parallel to the first ASV. When such solutions still fail to stop a surge event from occurring, a hot gas recycle (HGV) is used as a last resort. The HGV is always to be used in tandem with the ASV and only during an emergency shutdown (ESD). Excessive hot gas recycle also shortens the efficacy of the lube oil that is used for lubrication purposes. Figure 3 shows an example of ASV inadequacy leading to recycling insufficient vapour to the suction during a sudden trip caused by power failure.

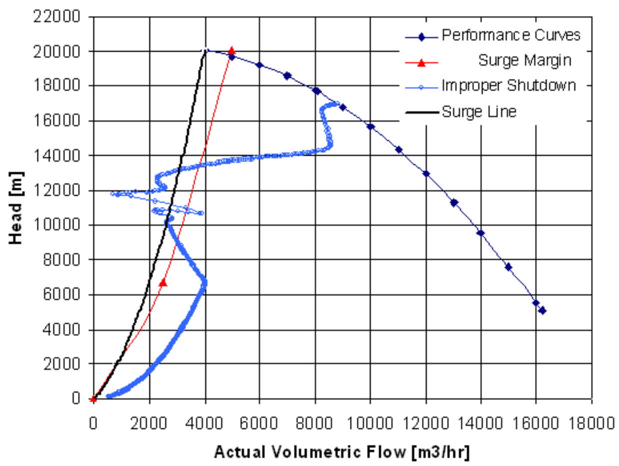


Figure 3. Surge during an Emergency Shutdown

Figure 4 shows a hot gas recycle installation that compensates for the ASV's deficiency, thereby keeping the operating point away from the surge line during an ESD.

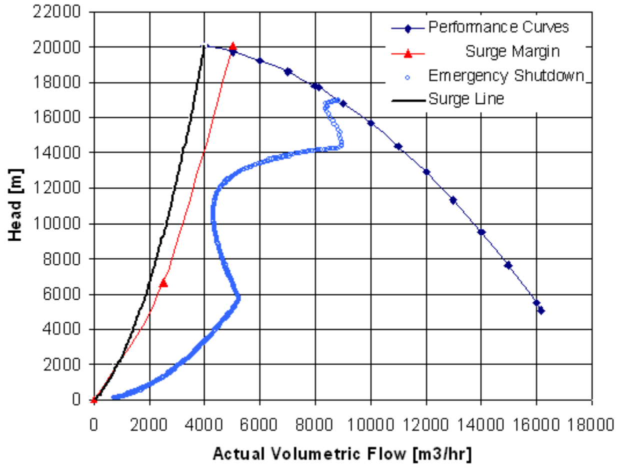


Figure 4. Surge Avoided with Hot Gas Recycle

In recent decades, with tools such as dynamic simulation, the quantity of hot gas to be recycled can be determined without recycling immoderate amounts of hot gas that can overheat the gas compressor with bearings and seals failing.

Requirements of an Anti-surge Valve (ASV) and Hot Gas Recycle Valve

A hot gas recycle/bypass system consists of piping with an On-Off Valve that is motor or pneumatic operated and should have a full opening time of < 1 sec (for valves between 4 inch to 16 inch). For larger On-Off Valves (above 16 inch), the time is taken to be < 2 sec. In the case of an electric motor driven compressor, the power source for the motor operated HGV must be independent lest during a power failure the motor operated HGV becomes futile. The hot gas valve and ASV should be fail open type and is sized for twice the flow required to keep the operating point away from surge. During operation, fluids velocities must be kept less than 0.3 Mach which otherwise causes damage to the valve and piping due to erosion. A noise limit of 110 dB is also placed and operating at around 85 dB is acceptable. For good pressure throttling, the ASV is equipped with linear opening characteristics or a combination of equal percentage characteristics up to ~40% opening with the remaining travel substituted with linear characteristics. The hot gas piping should be laid as short as possible between the discharge and suction to have a fast response.

During an ESD scenario (e.g., power loss), taking a conservative approach for design purposes, the control output signal from the compressor driver after a trip, takes ~300 msec to reach the Distributed Control System (DCS) and another ~300 msec from the DCS to reach the HGV to open. However with advances in technology, these timings can be considered at ~100 msec. *In simple terms, a lower response time increases the chances of responding faster to a compressor surge.*

Deviations from Design Criteria

As a thumb rule, the hot gas system is sized for 50% (max) during FEED stage. However this needs to be checked with a dynamic simulation study since over-sizing the hot gas system recycles excessive flow that causes the bearings and seals to overheat.

As per API 617 (7^{th} Edition, 2002), Clause 2.7.1.3, it states, '*As a design criteria, bearing metal temperatures shall not exceed 100°C (212°F) at specified operating conditions with a maximum inlet oil temperature of 50°C (120°F). Vendors shall provide bearing temperature alarm and shutdown limits on the datasheets.'* However clause No. 2.7.1.3.1 of the said document also says, '*In the event that the above design criteria cannot be met, purchaser and vendor shall mutually agree on acceptable bearing metal temperatures.'*

In the Author's experience, this deviation was seen up to ~135^{0}C depending on the manufacturer and believes that this is due to a variation of operating conditions between string test conditions and actual conditions. Nevertheless, compressor operating temperatures must never exceed the stipulated or mutually agreed values in order to protect the compressor's internals.

Compressor Control Systems

In today's world no piece of machinery can be said to be protected by modern methods without implementing a control system. A surge can occur in a matter of seconds or sometimes even milliseconds giving almost no time for operators to intervene. Hence a control system becomes a part and parcel of the compressor package.

Although the good old Proportional-Integral-Derivative (PID) control was enough to avoid a surge by minimizing the compressor recycle flow, it did not aid much in reducing / optimizing the power requirements. With a steady rise in the oil consumption since the 1970s, the necessity of energy efficiency, safety and environmental friendliness became a priority and demanded better control systems.

To respond quickly to any process upsets, high computational speeds in controllers also became a necessity. This led to the rise of specialized control equipment known as 'Black Boxes' that was the alternate to panel mounted instruments. Black boxes though addressed response times, suffered from frequent hardware and software revisions. Black box technology was proprietary with its own coding languages and often experienced compatibility issues when interfacing between different manufacturer's models. This also meant having to sometimes shutdown the machinery causing monetary implications and increased downtime if not made part of plant maintenance.

Advent of Programmable Logic Controller

With the limitations of black box technology being recognized, industry honchos realized the necessity of standardizing and generalizing control systems and their respective programming languages. These standardization efforts led to documenting the IEC 61131 (International Electrotechnical Commission Standard for Programmable Controllers) in 1993 and subsequently revised in 2003.

Programmable Logic Controllers (PLCs) provided not only computational power but also were easily integrate-able to the compressor controls. PLC's offered the advantage of scalability where new I/O's could be added during any form of plant modification/expansion depending on the type of PLC used (e.g., modular or stacked). PLCs also offer diagnostics capabilities, for example, to trace through the logs of controller output during a fault analysis.

In earlier systems, as shown in Figure 5, a primary PLC is supplemented with an auxiliary PLC that controlled systems like lube oil, seal oil / dry gas seals, start up sequencing, interlocks, etc. This also required interfacing them properly to allow operators to diagnose and do a root cause analysis in the event of, for example, a compressor trip.

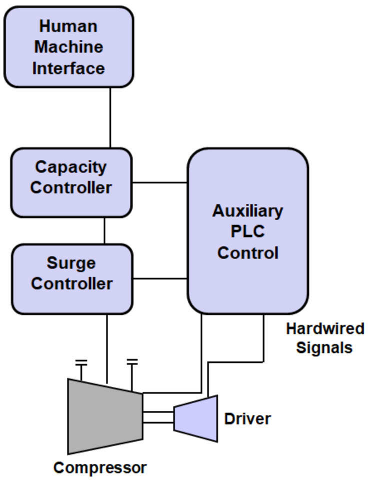


Figure 5. Compressor System with Interfaces

However with integrated systems as shown in Figure 6, that used a dedicated control PLC with a backup PLC and the necessary hard wiring, the cost of implementation also comes down, offering better efficiency, diagnostics, generic parts and scalability.

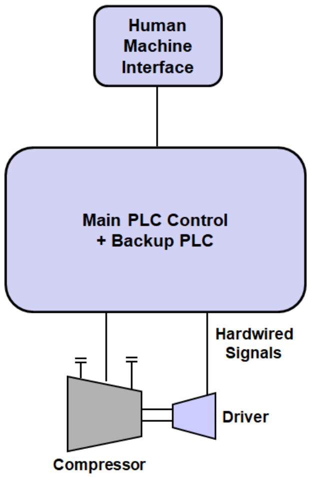


Figure 6. Compressor System with Integrated Interfaces

References & Further Reading

1. '*Compressor Control: Moving Beyond "Black Boxes" to Integrated, Automated Platforms in Oil and Gas Production*', Rick McLin, Turbomachinery Controls Business Manager, Rockwell Automation
2. '*Axial and Centrifugal Compressors and Expander-compressors for Petroleum, Chemical and Gas Industry Services*', API Standard 617, 7th Edition, July 2002

Module 6

Variable Speed Drives For Gas Compressor Operations

Historically the Oil & Gas industry has been dominated by mechanical prime movers like gas turbines or steam turbines to drive centrifugal gas compressors for large industrial applications. With time, these prime mover operations have become difficult to maintain, considering the Environmental Protection Agency (EPA) regulations that mandate strict emission compliance. A good alternative to combat emission concerns is to use electric motors as a prime mover in modern times with regards to their high rates of efficiencies of the order of 80% to 90% in addition to the absence of hydrocarbon emissions from electric motors. Electric Motors can be operated in two modes, namely, Fixed Speed Drive (FSD) Mode where the prime mover's speed cannot be altered during operation & Variable Speed Drive (VSD) Mode where the prime mover's speed can be altered by altering the frequency of the current fed to the electric motor. The following module is written to explore the power requirement implications for gas compressor operation when employing these modes of operation.

Introduction

The standard procedure employed by engineers to perform gas compressor design can be found quoted in literature such as GPSA, JM Campbell to name a few with Industry Standards, for example, API 617, to conveniently customize centrifugal gas compressors as per customer's requirements. However Chemical Engineers need to work beyond the standard practice of estimating steady state process parameters to avoid under-estimating power requirements for an effective compressor start-up & restart. This is so, since steady state calculations only provide information on the 'Absorbed Power' which can be defined as the power required to sustain the gas compressor at the required operating conditions during continuous operation. The power required to start a gas compressor will always be higher than the steady state absorbed power since, the electric motor (EM) has to overcome the inertia of the entire gas compressor system to bring it to normal operating conditions.

To understand by how much, excess power required during start-up varies, requires a transient state set of calculations including steady state calculations. With the advent of engineering software, such studies can be made and for this module Aspentech's HYSYS 2006.5 is used to investigate the power requirements for a compressor start-up based on a case study.

Case Study

To understand the effects of VSD and FSD mode of operation on gas compressor start-up, a case study is made. Based on a certain gas compressor performance curves, the following process conditions are employed.

Table 1. Gas Compressor Process Parameters

Parameter	Value
Operating Capacity	**8,100 m^3/h**
Suction Flange Pressure	**0.2 barg**
Discharge Flange Temperature	**25 ^0C**
Discharge Flange Pressure	**2.7 barg**
Gas MW	**18.38**
Compressor Speed [100% Curve]	**3,000 rpm**

The Performance Curves (Polytropic Head vs. Actual Flow & Polytropic Efficiency vs. Actual Flow) used for the study is shown below.

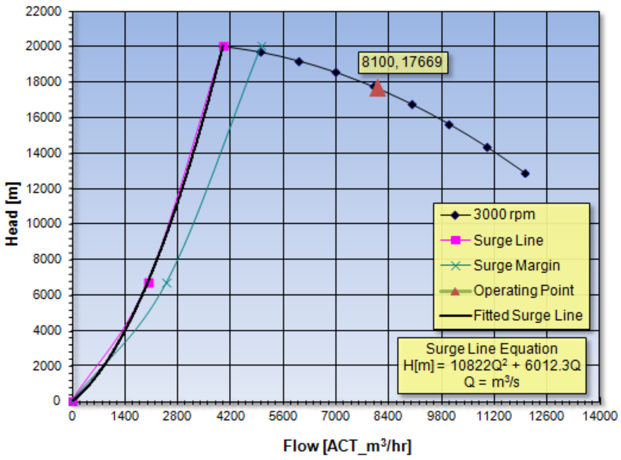


Figure 1. Polytropic Head vs. Actual Volume Flow

Since no information is available on the actual location of the surge line, the surge margin is assumed as 10% on the actual volume flow. The polytropic efficiency vs. flow rate curves is shown below as,

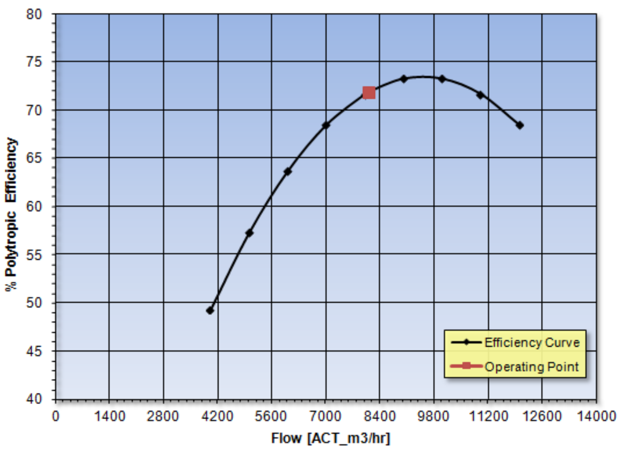


Figure 2. Polytropic Efficiency vs. Actual Volume Flow

To drive the compressor, an asynchronous electric motor is chosen. The slippage in the electric motor can vary even up to 5% during start-up depending on its design. However for this module, a value of 1% is assumed. Note that, the EM is rated at 600 kW & the start-up power required would be higher than 600 kW. The power absorbed during operating conditions is 496 kW. In addition to the VSD and FSD cases, the case of suction throttling with an FSD is also investigated.

Table 2. Electric Motor Configuration

Parameter	Value
Electric Motor Type	**Asynchronous Induction**
Motor Rating	**600 kW**
Configuration	**4 Pole, 50 Hz**
Motor Slip	**1%**
Rated Speed	**1500 rpm [With Gearbox]**

The EM is also characterized by a Speed vs. Torque curve enabling to compute the power and torque required to be generated.

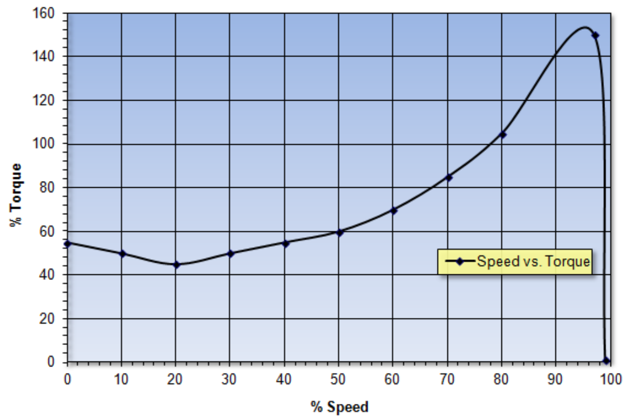


Figure 3. EM Speed vs. Torque Characteristics

Design Methodology & Approach

Prior to performing a transient study for power requirements, the gas compressor's remaining piping & equipment system details needs to be available. The general design approach consists of performing steady state calculations, i.e., heat & mass balance as well as sizing the equipment, valves & lines based on customer's needs and layout. In the current module, since no layout information is available, an approach is proposed for preliminary volumes to avoid surge during start-up & shutdown. This is followed by shaping these volumes into detailed piping & equipment estimates. A general schematic of the compressor loop envisaged is shown,

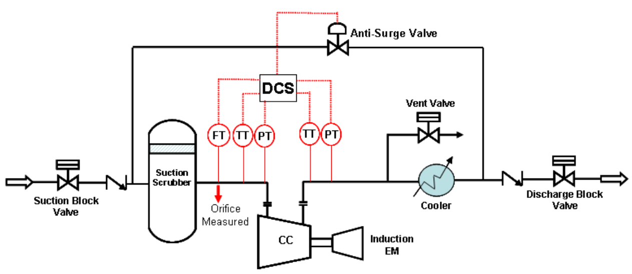


Figure 4. Gas Compressor Proposed Layout

Equipment Sizing

Equipment Volumes

The main equipment in addition to piping that contributes to the compressor loop volumes are suction scrubber and discharge cooler. In situations where gas condensation occurs after the discharge side air cooler, an additional discharge scrubber is installed to knock out any liquids. The discharge side volumes particularly affect the response time of anti-surge system. Excessive discharge side volumes result in a delay in recycling discharge gas. Hence discharge side volumes are to be kept as minimum as possible.

It must be ensured that the anti-surge take-off point before the discharge side check valve and air cooler is chosen, such that the setup is not too close to the compressor discharge flange (which can cause the anti-surge valve to rattle) inducing noise related issues. If the anti-surge line tap off point is too far, then it increases the surge response time. The rate at which the compressor coast down, i.e., speed decay occurs, also determines the size of the anti-surge valve that regulates the amount of the recycle flow. Based on the above considerations, the equipment volumes are calculated as described in the next section.

Suction Side Volume

The suction side volume can be calculated initially for twice the rated volume flow. Therefore,

$$V_{Suction,Prelim} = Q_{Suction\ Scruuber} \times Margin \quad (1)$$

$$Or,\ V_{Suction,Prelim} = \frac{8100}{3600} \times 2 = 4.5\ m^3 \quad (2)$$

Therefore on the suction side, a preliminary volume of 4.5 m^3 is taken. This is a preliminary estimate that is subjected to change depending on heat & material balance, equipment sizing & transient study results for start-up & shutdown scenarios.

Discharge Side Volume

The discharge side volume is predicted for the worst case of surge & this can happen during an emergency shutdown (ESD) which is dependent on the decay rate of the compressor speed as well as the recycle flow rate through the anti-surge valve (ASV). The discharge side volume can be taken as approximately, 1/3rd of the suction volume.

$$V_{Discharge,Prelim} = V_{Suction\ Scruuber} \times \frac{1}{3} \quad (3)$$

$$Or,\ V_{Discharge,Prelim} = \frac{1}{3} \times 4.5 = 1.5\ m^3 \quad (4)$$

Therefore, the discharge volume is 1.5 m^3.

Anti-surge Valve (ASV) Size

To estimate the anti-surge valve (ASV) C_v, the ASV inlet & outlet pressure is required. The compressor vendor would provide performance curves as a plot of discharge pressure vs. flow rate from which the discharge line losses is added to the compressor discharge pressure to arrive at the ASV's upstream pressure & the ASV's downstream pressure is the sum of suction side line losses and compressor suction pressure. Considering the maximum possible flow through the compressor is at the stonewall region, the ASV can be sized for this flow. However, to avoid equipment operation at its limits, a margin of 10% to 15% on the stonewall flow at 3000 rpm is taken to ensure that the ASV does not recycle excess fluid back to the suction side. Based on a maximum allowable compressor suction flow of 10,440 m^3/h, the ASV C_v size is taken as 980 as per ANSI/ISA 75.01-1985 standard estimation.

Start-up Operations

The compressor loop is checked for stability in terms of surge and power adequacy for different start-up modes. This is applied to

1. Fixed Speed Gearbox coupled Electric Motor (EM) + Centrifugal Compressor (CC) configuration

2. Variable Frequency Drive (VFD/VSD) + Electric Motor (EM) configuration.
3. In addition to VFD and FSD configurations, the case of using a suction throttle valve at 100% ASV opening is also studied.

Suction throttling methods involve the use of a globe valve or a butterfly valve at the suction line of the CC to cool the gas thereby increasing gas density. This operation reduces the power required to compress the incoming gas i.e., compressor start-up power is also decreased. However, a limitation exists on the suction throttling operation while reducing the suction flow rate, because it results in the operating point moving closer to the surge line during start-up. Hence the throttling valve operation must be regulated.

Table 3. Gas Compressor Start-up Cases

Start-up Case No.	Electric Motor Configuration		Anti-surge Valve (ASV) Position		Fixed Speed & Fixed ASV opening with / without Suction Throttling
	Fixed Speed	Variable Speed	Fixed ASV	Variable ASV	
1	√		√		
2	√			√	
3		√	√		
4		√		√	
5	√				√
6		√			√

Results & Discussions

Case 1: Fixed Speed with Fixed ASV at Start up

For Case 1, based on the simulations, the following plot shows the operating point migration. For the volumes & ASV size used, no surging occurs & the compressor reaches the rated point of 8,100 m³/h & 2.7 barg.

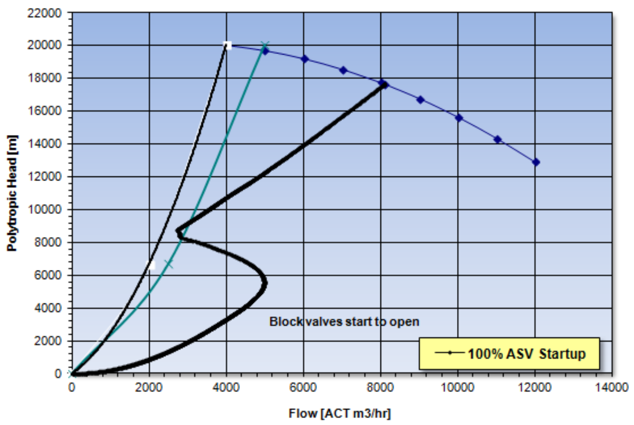


Figure 5. Case 1: Operating Point Migration

Case 2: Fixed Speed with Variable ASV Start up

For Case 2, based on the simulations, the following plot shows the operating point migration. For the volumes & ASV size used, no surging occurs & the compressor reaches the rated point of 8,100 m³/h and 2.7 barg.

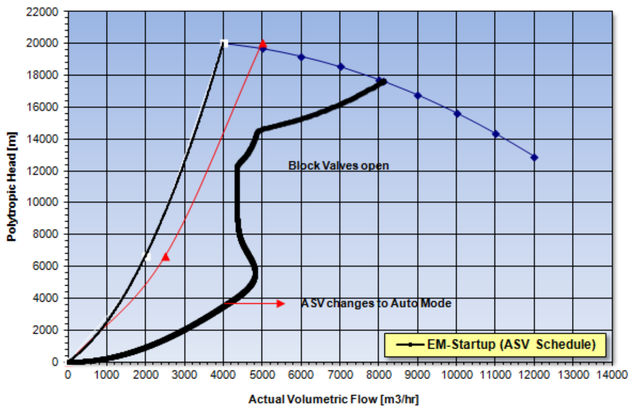


Figure 6. Case 2: Operating Point Migration

Case 3: Variable Speed with Fixed ASV Start up

Variable frequency drives (VFD) are particularly useful when a variation in speed is required, to control gas throughput during production changes. VFD's also offer the advantage of lowering compressor speed during turndown conditions thereby avoiding gas recycling that causes energy wastage.

During start-up, the EM speed ramp-up rate is achieved by altering the frequency of the current passing through the EM which thereby raises the CC speed gradually. In Case 3, for the volumes & ASV size used, no surging occurs & the compressor reaches the rated point of 8,100 m^3/h and 2.7 barg discharge pressure.

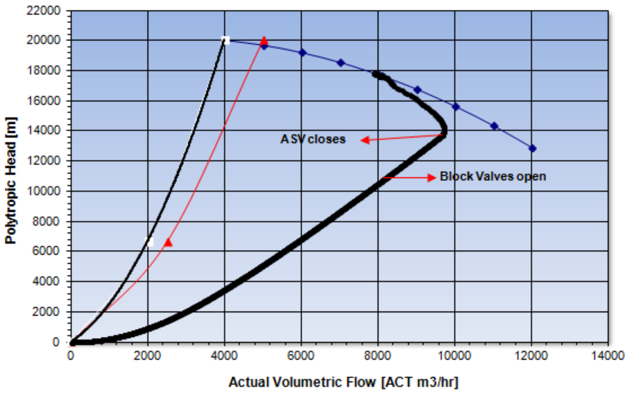


Figure 7. Case 3: Operating Point Migration

Case 4: Variable Speed with Variable ASV Start

The variable ASV position is achieved by using an Anti-surge controller (ASC). In Case 4, no surging occurs & the compressor reaches the rated point of 8,100 m^3/h and 2.7 barg discharge pressure.

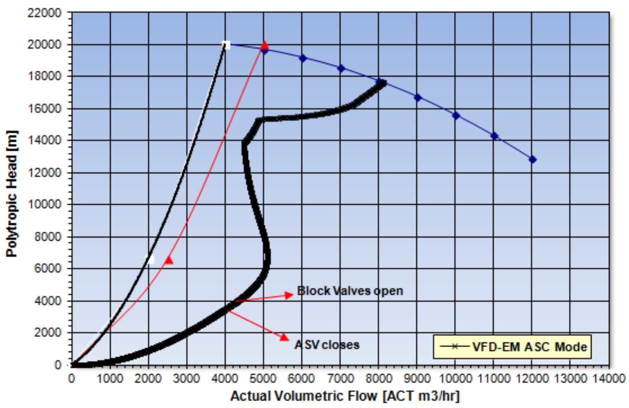


Figure 8. Case 4: Operating Point Migration

Case 5: Fixed Speed with Suction Throttling

For Case 5, with fixed speed drive, 100% ASV opening and suction throttling, suction throttling causes the operating point to closer to the surge line but does not cross. Based on the simulations, the following plot shows the migration of operating point. For the volumes & ASV size used, no surging occurs & the compressor reaches the rated point of 8,100 m^3/h and 2.7 barg discharge pressure.

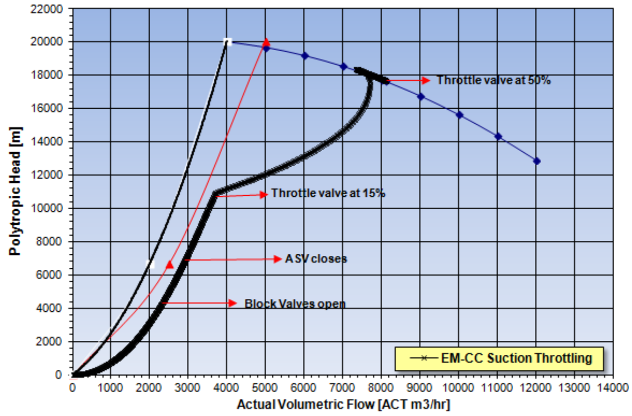


Figure 9. Case 5: Operating Point Migration

Case 6: Variable Speed with Suction Throttling

For Case 6, with variable speed drive, 100% ASV opening and suction throttling, based on the simulations, the following plot shows the migration of operating point. For the volumes & ASV size used, no surging occurs & the compressor reaches the rated point of 8,100 m^3/h and 2.7 barg. Between cases 5 & 6, the peak absorbed power is 865 kW and 666 kW respectively which can be attributed to the use of variable ASV position during start-up.

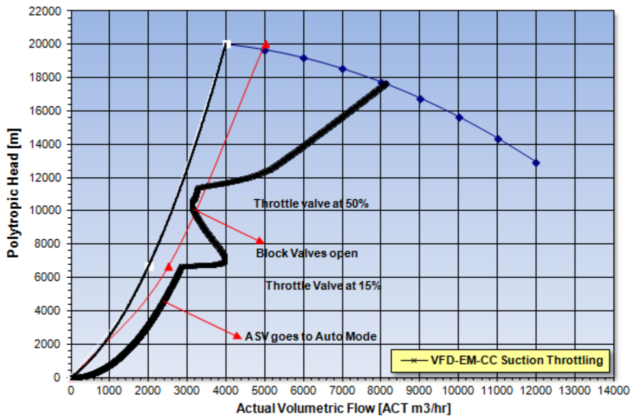


Figure 10. Case 6: Operating Point Migration

Start-Up Power Results

From the 6 cases simulated for start-up power requirements, a plot of the power absorbed vs. Time for a start-up time of 180 sec (~3 min) for cases that have VFD provision and is shown below.

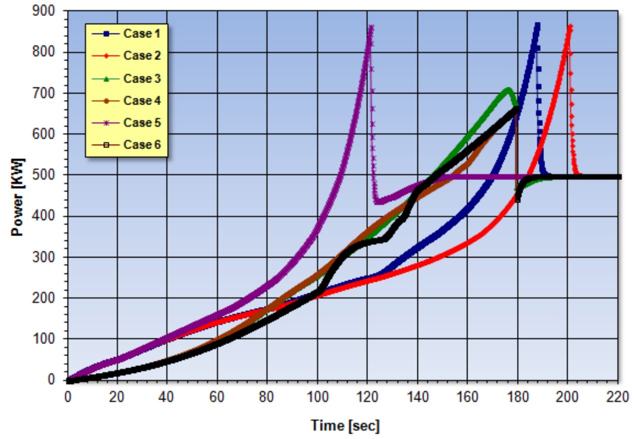


Figure 11. Compressor Start-up Power – All Cases

The compressor speed variation during start-up is also plotted and shown below.

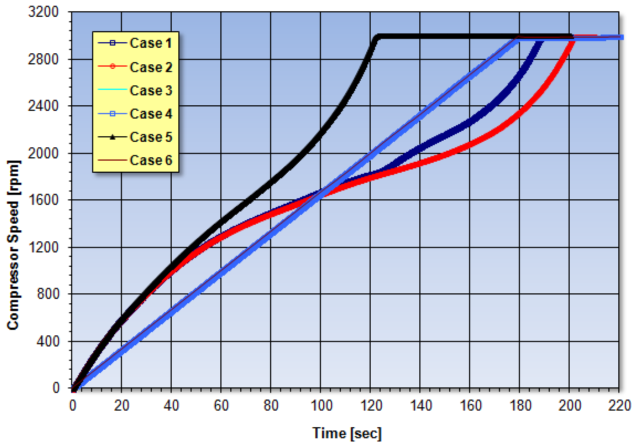


Figure 12. Compressor Speed vs. Time – All Cases

A comparison of the peak absorbed power for all the start-up cases is shown below.

Table 4. Start-up Power Comparison

Case	Peak Absorbed Power [kW]	% Power Savings [w.r.t Case 1]
1 (Base Case)	869	0
2	866	0.3
3	711	18.2
4	666	23.4
5	865	0.5
6	666	23.4

From the above table, it is seen that for cases 1, 2 & 5 which use an fixed speed induction motor coupled CC with Gearbox arrangement, the start-up power is higher while for cases 3, 4 & 6 which use a VFD, the start-up power is lower by ~23% w.r.t case 1. Cases 1, 2 & 5 are fixed speed EM operation cases and show that the peak power absorbed is higher since the EM has to reach breakdown torque threshold of ~150%. In case 2, using a variable ASV position for a fixed speed operation during start-up, does not help in energy reduction.

Conclusions

From the study made, it can be inferred that,

1. VFD's show a significant reduction in start-up peak power when compared to FSD operation. In oil & gas applications, with varying production rates, VFD's are better equipped to alter the compressor speed to match production demands & give energy savings & operational savings.
2. Electric motor's speed vs. Torque curves are designed for a higher break-down torque. This means during start-up with an FSD, there are chances that the EM would reach the break down torque but can still fail to bring the compressor online to match the required process conditions. In such cases, VFD's are an alternative.
3. With FSD's, there is no provision for speed control. Hence during a stat-up if the ramp up rate is too fast, a sluggish anti-surge controller would struggle to recycle sufficient flow & prevent surge. Use of VSD's help slowing down the start-up ramp up rate of the electric motor, enough to allow the anti-surge controller to respond & ensure sufficient recycle gas flows to the suction to prevent a compressor surge.

References & Further Reading

Aspentech HYSYS 2006.5 Documentation – Dynamic Modelling

Module 7

Load Sharing for Parallel Operation of Gas Compressors

The art of load sharing between centrifugal compressors consists of maintaining equal throughput through multiple parallel compressors. These compressors consist of a common suction and discharge header. Programmable logic controllers (PLCs) can be incorporated with load sharing functions or can be incorporated as standalone controllers also. Control signals from shared process parameters such as suction header pressure or discharge header pressure can be then fed to individual controllers such as compressor speed controllers (SC) or anti-surge controllers (UIC) to ensure the overall load is distributed efficiently between the compressors.

The following module covers load sharing schemes for parallel centrifugal compressor operation.

Load Sharing Options

The load sharing options covered are as,

1. Base Load Method
2. Suction Header - Speed Control Method
3. Equal Flow Balance Method
4. Equidistant to Surge Line Method

Base Load Method

In Base Load method of operation, one compressor is allowed to run on manual mode while the other is controlled through speed manipulation based on the discharge header pressure. The pressure controller on the discharge header is termed as the Master Pressure Controller (MPC) that alters the second compressor's speed a.k.a "Swings" the compressor speed to cater to varying throughputs. In Fig 1, the speed of compressor A is manually set (HIC) for a maximum throughput, i.e. Base Load.

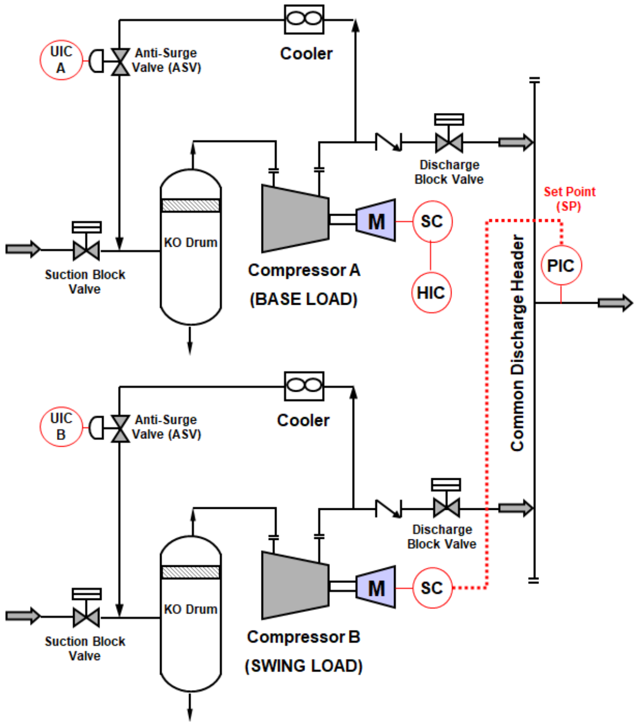


Figure 1. Base Load Operation Method

The speed of compressor B is altered based on the master pressure controller (PIC) set point (SP) to attend to the swing in flow throughputs.

During periods of low process demand, Compressor B (swing machine) can be recycling & sometimes even close enough to the Surge Control Line (SCL) causing the swing machine to trip. Additionally, due to differences in piping layouts & pressure loss, the compressor operation would not be symmetrical, causing operators to frequently intervene. With these limitations, the base load method is least preferred.

Suction Header - Speed Control Method

In the suction header - speed control method, no base load exists. Instead the master pressure controller (PIC) is shifted to the suction header. The advantage offered is, both compressors operate independently despite a common set point provided by PIC to the speed controllers (SC) of both compressors.

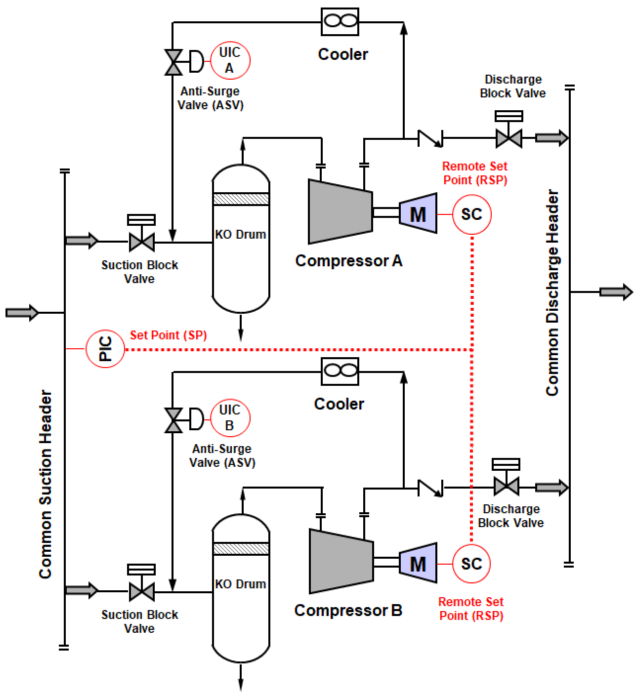


Figure 2. Suction Header Speed Control Method

It may be noted that both compressors would not necessarily be running at the same speed or flow due to differences in the piping layout as well as during a compressor recycle since both anti-surge controllers (UIC A/B) also act independently of each other.

To ensure no production losses, the configuration consists of standby compressor along with working compressors. During the failure of one of the compressor, say compressor A, the PIC issues a signal to increase the speed of compressor B, until the standby compressor can be brought online to maintain throughput. In case of layouts that have no standby compressors, a 2 × 50% configuration, with no recycle during regular operation must be chosen. This enables the remaining working compressor to cater to 100% of the throughput/load at higher speeds during failure of the one of the compressors.

Equal Flow Balance Method

In the equal flow balance method, the Master Pressure Controller (PIC) on the common discharge header determines the total load demand and alters the speeds of Compressors A & B via SC. The individual flow control signal to each speed controller is achieved by scaling the total load demand (BIAS A & BIAS B) to the individual flow controller (FC) on each compressor. Both Compressor operations are independent of the Anti-surge valve (ASV) operation.

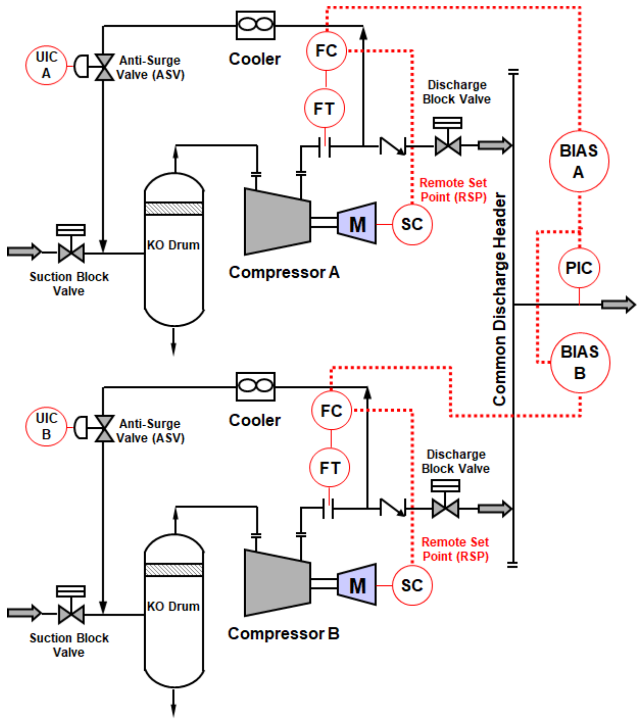


Figure 3. Suction Header Speed Control Method

However certain limitations exist with the flow balancing method. Due to additional control elements, CAPEX cost increases. Furthermore since the flow element & transmitter (FT) is installed on the compressor discharge, additional pressure drop occurs which represents energy losses.

For the cascaded control used, PIC ⇒ FC ⇒ SC, the inner loop (FC) must respond faster than the PIC outer loop. This causes the master pressure control, PIC to be sluggish. A faster FC loop also means, the compressor speed would increase rapidly than required often reaching maximum speed. Hence this does not offer the best control strategy.

Equidistant to Surge Line Method

In the equidistant method, the aim is to ensure, the deviation/distance between the operating point and the surge control line (SCL) in both trains is equidistant.

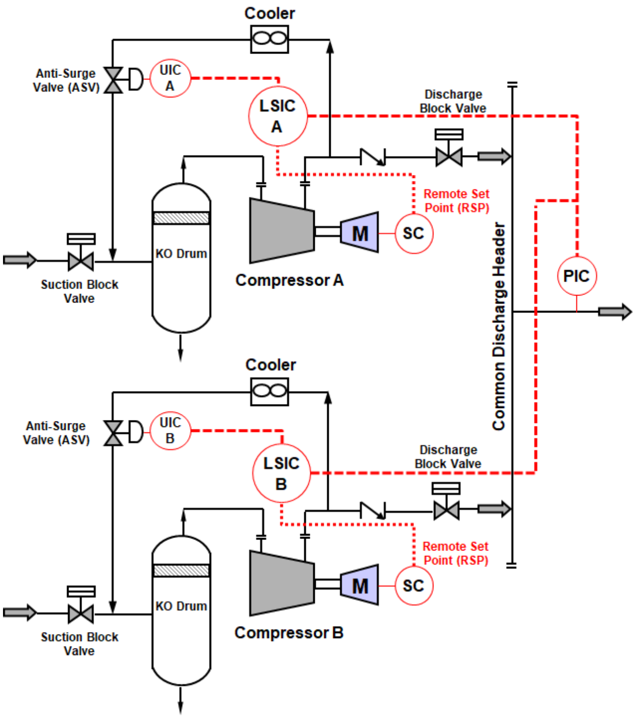


Figure 4. Equidistant to Surge Line Method

In this configuration, neither the throughputs through each compressor or the operating compressor speed is the same, but only the deviation between the operating point & SCL. It may also be noted that the load sharing function (LSIC A/B) that alters the compressor speed, is not fed with the signal from the suction flow transmitter (FT), but instead the anti-surge controller (UIC A/B) and the master pressure controller (PIC) installed on the common discharge header. This would mean, both UIC A/B and LSIC A/B have to coordinate in real time.

A significant advantage of the equidistant to surge line method is the configuration's ability to cater to asymmetrical performance curves, i.e., dissimilar compressors. In brownfield modifications, any addition of new compressors can offer synchronicity issues including variation in throughputs & pressures due to differences in performance curves & piping layouts. Therefore the equidistant method becomes an effective configuration for varying loads ensuring both compressors independently adjust their respective operations and avoid surge.

Some Design Considerations

1. The Master pressure controller which provides shared information across all compressors can often be subjected to harsh field conditions. To circumvent these issues, redundancy with multiple transmitters can be provided. This ensures not only maximum availability but also hardwiring the transmitters prevents any loss of signals to the Load sharing system.
2. Depending on the reliability of the control systems, controllers need to be replaced sometimes with third party OEM vendors, each with their own proprietary control systems. Hence load sharing systems must be able to integrate different vendors.
3. Real Time optimization (RTO) techniques based on regression models of steady state data have gained sufficient footing in recent years. Short Time RTO of the order of a few minutes & Long term RTO of the order of a few days can be employed to determine the best load distributions between compressors.

References & Further Reading

1. "*Advanced Load Sharing Controls for Compressor Networks*", Alex Benim, Brian Eldridge, Woodward Inc.

MODULE 8
CENTRIFUGAL COMPRESSOR SETTLE OUT CONDITIONS

Centrifugal Compressors are a preferred choice in gas transportation industry, mainly due to their ability to cater to varying loads. In the event of a compressor shutdown as a planned event, i.e., normal shutdown (NSD), the anti-surge valve is opened to recycle gas from the discharge back to the suction (thereby moving the operating point away from the surge line) and the compressor is tripped via the driver (electric motor or Gas turbine / Steam Turbine).

In the case of an unplanned event, i.e., emergency shutdown such as power failure, the compressor trips first followed by the anti-surge valve opening. In doing so, the gas content in the suction side & discharge side mix.

Therefore, settle out conditions is explained as the equilibrium pressure and temperature reached in the compressor piping and equipment volume following a compressor shutdown

Importance of Settle Out Conditions

The necessity to estimate settle out conditions,

1. Settle Out Pressure (SOP) & Settle Out temperature (SOT) determines the design pressure of the suction scrubber & piping.
2. The suction scrubber pressure safety valve's (PSV) set pressure as well as the dry gas sealing pressures are decided by the settle out pressure.
3. When the compressor reaches settle out conditions, process gas is locked inside the piping and equipment and grips the compressor rotor from rotating effectively when restarted. Hence depressurizing is done by routing the locked gas to a flare, via the vent valve to reduce the pressure and achieve effective re-start.

Estimating Settle Out Conditions

Although there are many process simulations tools that can be used to conduct a transient study to determine settle out conditions, hand calculations based on first principles of thermodynamics can also be easily employed. In order to do so, the gas compressor system can be reduced with the assumptions as follows, with the philosophy of using a lumped parameter model, in which an energy balance is made across the total volume of the compressor loop taking into account, the compressor deceleration rate.

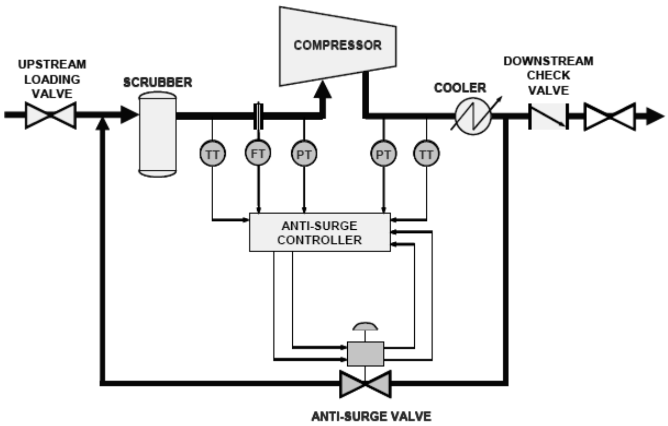


Figure 1. Schematic of Compression System

The assumptions made for this module are,

1. The compressor loop system is a closed loop & no gas has escaped the system.
2. The rate of closure of the suction & discharge block valve in addition to the check valve on the discharge side is neglected.
3. The air cooler is assumed to be running at constant duty before and after the compressor is shut down. If the cooler failure occurs due a power trip, then heat rejection ($Q_{Cooler} = 0$) is considered to stop instantaneously.

4. The piping is considered to be adiabatic & no heat escapes from the equipment & piping.
5. The suction scrubber, if considered to have accumulated liquids, then this volume is subtracted from the equipment volumes.
6. The time delay between the fully closed position & fully open position of the Antisurge valve (ASV) and check valve is not considered.
7. When the driver coasts down after a trip, some amount of residual work is done on the gas.
8. Compressor shutdown times are also influenced by the fluid resistance, dynamic imbalance, misalignment between shafts, leakage and improper lubrication, skewed bearings, radial or axial rubbing, temperature effects, transfer of system stresses, resonance effect to name a few and therefore in reality, shutdown times can be lower than estimated by the above method.

Calculation Methodology

The lumped parameter methodology applied to the compressor loop can be depicted as,

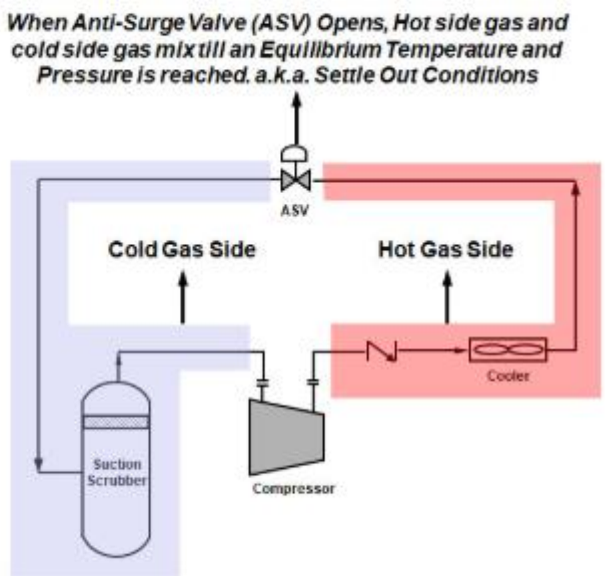


Figure 2. Gas Compressor Loop

Based on the assumptions made, the Settle Out Temperature (SOT) can be estimated as,

$$T = SOT = \frac{(m_S c_{p,S} T_S + m_D c_{p,D} T_D) - Q_{Cooler} + (m_S + m_D)(H_P(t))}{m_S c_{p,S} + m_D c_{p,D}}$$

(1)

Where,

$$H_p(t) = A(Q)^2 + B(QN) + CN^2 \quad (2)$$

$$N = N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t - t_0)}{(2\pi)^2 J}} \quad (3)$$

Where,

$H_P(t)$ = Rate of change of polytropic head as the compressor coasts down [kJ/kg/s]

$N(t)$ = Rate of compressor speed decay [rpm/s]

m_s = Suction side gas mass [kg]

m_D = Discharge side gas mass [kg]

T_s = Suction temperature before shutdown [K]

T_D = Discharge temp before shutdown [K]

$C_{p,s}$ = Suction Side Heat Capacity [kJ/kg.K]

$C_{p,D}$ = Discharge Side Heat Capacity [kJ/kg.K]

Q_{Cooler} = Cooler Duty [kJ/s]

k = Fan Power Law Constant

J = Total Inertia of Compressor System [kg.m^2]

The Settle Out Pressure (SOP) can be estimated

$$SOP = \frac{m \times Z_{avg} \times R \times SOT}{MW \times (V_1 + V_2)} \quad (4)$$

Where,

m = Total gas mass [kg]

Z_{avg} = Average Compressibility Factor [-]

R = Gas Constant [m^3.bar/kmol.K]

MW = Gas Molecular weight [kg/kmol]

SOT = Settle Out Temperature [K]

V_1 = Suction side volume [m^3]

V_2 = Discharge Side Volume [m^3]

Case Study

A validation case study is made for a Tank Vapour compressor in a Gas Compression Plant. Suction pressure exists at 1.05 bara, 54^0C with a discharge pressure of 5.5 bara, 128^0C. The coast down period is calculated initially followed by performing settle out calculations. An assumption is made, that the air cooler continues to operate after shutdown. The compressor maps used is

Table 1. Compressor Performance Curves

H_p	Q	Q/N	Hp/N^2
[kJ/kg]	[Am³/s]	[(Am³/h)/rpm]	[kJ/(rpm²)]
136.2	3.0778	0.000322	1.493E-06
133.9	3.4278	0.000359	1.468E-06
130.5	3.6806	0.000385	1.431E-06
126.6	3.8472	0.000403	1.388E-06
123.6	3.9583	0.000414	1.355E-06
115.8	4.1111	0.000430	1.269E-06
109.6	4.1806	0.000438	1.201E-06
100.0	4.2500	0.000445	1.096E-06

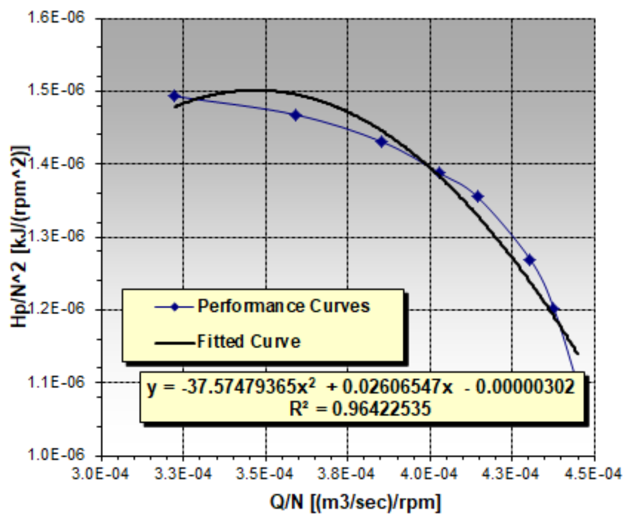


Figure 3. Compressor Performance Curves

Performing calculations as shown in previous sections in MS-Excel based on Table 2 & 3,

Table 2. Compressor Coast down Input Data

Compressor Design Details		
Compressor Inertia	376	kg.m²
Gear Box Inertia	38	Kg.m²
EM Inertia	150.6	kg.m²
Total Inertia (J)	380.6	kg.m²
EM /GT Speed	1493	rpm
Operating Speed	9551	rpm
Gear Ratio (GR)	6.40	-
Fan law constant (k)	8.38E-05	N.m.min²

Fan law Constants (*k*)		
% Speed	**Speed [rpm]**	**k [N.m.min²]**
105	10029	7.57E-05
100	9551	7.00E-05
95	9073	6.68E-05
90	8596	6.42E-05
80	7641	6.03E-05
70	4776	1.66E-04
Avg. Fan Law constant (k)		8.38E-05

It is to be noted, with the Q vs. H_p curve at 9551 rpm, Fan laws were used to derive the compressor curves for other speeds, from 70% to 105%.

Table 3. Settle Out Conditions Calculations

Suction Piping Data		
Piping Volume	74.55	m³
Gas Mass Density	1.66	kg/m³
Mass Specific Heat	*1.83*	kJ/kg.K
Gas Temperature	54.1	⁰C
Comp. Factor (Z_1)	0.9875	-
Suction KO Drum %Vol. Liq	20.0	%
Gas Mass- Suction Side	*98.82*	kg
Discharge Piping Data		
Piping Volume	7.87	m³
Gas Mass Density	*7.53*	kg/m³
Mass Specific Heat	*2.16*	kJ/kg.K
Gas Temperature	128.3	⁰C
Comp. Factor (Z_2)	0.9622	-
Discharge KO Drum %Vol. Liq	10.0	%
Gas Mass – Discharge Side	*11.22*	kg
Cooler Data		
Cooler Duty	1432	kW
Cooler Outlet Specific Heat	2.03	kJ/kg.K

Using the estimated coast down time value of 115 sec for the case studied, the settle out pressure (SOP) & Settle Out Temperature (SOT) is calculated as 0.81 bara, 55.7^{0}C & a Settle Out Time of 175 sec. The transient plots of the SOP & SOT based on HYSYS simulations of the case study is as follows,

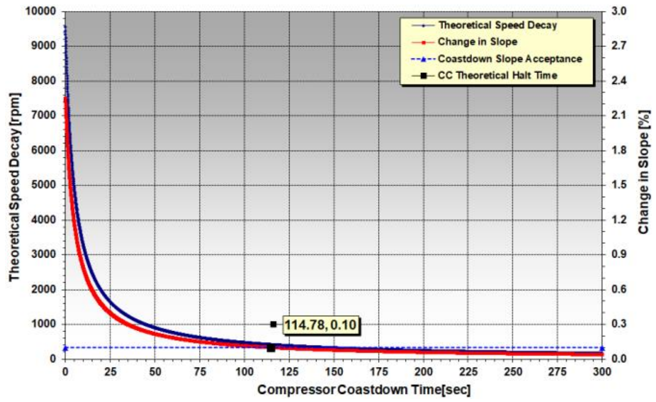


Figure 4. Compressor Coast down Time

The calculated Settle out temperature (SOT) Trend compared with HYSYS 2006.5 is shown as follows,

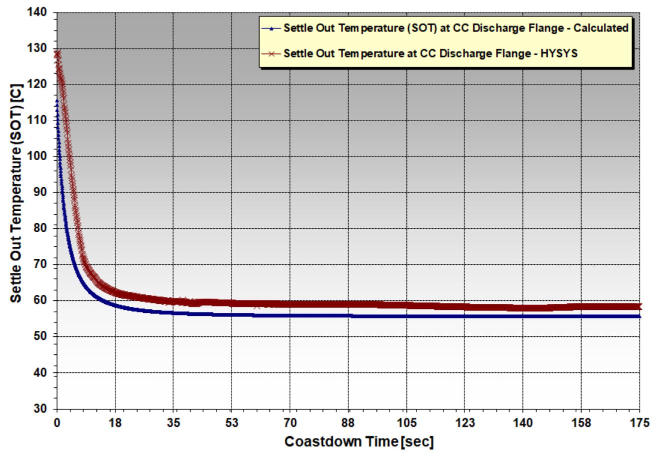


Figure 5. Settle Out Temperature Trend

A comparison made between HYSYS Simulations & the methodology presented shows,

Table 4. HYSYS vs. Calculated Results

Parameter	HYSYS	Calculated	%Error
SOT [^{0}C]	58.4	55.7	-4.8
SOP [bara]	0.53	0.81	+34.6
Settle Out Time [s]	167	175	+4.8

The SOT & Settle Out Time shows an error margin of < ±5%. Whereas for SOP, between the HYSYS predicted value of 0.34 bara and calculated value of 0.81 bara, represents ~35% error. The author attributes the error in SOP partly to the suction & discharge valve closure time in HYSYS when some vapours were discharged & the remaining for the reasons explained in the next section.

Effect of Assumptions on Results

1. **Approximation of compressor curves to Fan Laws –** Fan laws are more applicable to fluids with low compressibility, smaller pressure ratios & constant density. Use of these laws would distort the Compressor manufacturer's data thereby causing a difference in calculations. Since the overlap area is significant, the performance curve used in the calculations is assumed to be same through out the period of coast down. Figure 6 shows the shift in the compressor performance curves.

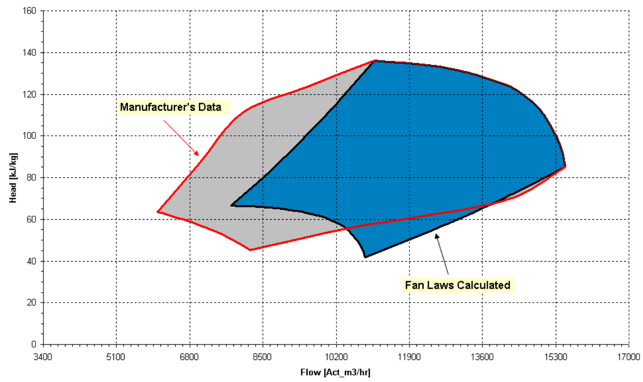


Figure 6. Comparison of Performance Curves between Fan Laws Generated & Vendor Data

2. **Equilibrium conditions during Settle Out –** During coast down, equilibrium conditions are not reached in the compressor plant piping since the system is dynamic with the gas moving & this is tracked in HYSYS 2006.5. However the calculations methodology considers complete equilibrium being reached at every time step. This causes a difference in the final settle out temperature (SOT) & settle out pressure (SOP).

3. **Average Mass Specific Heat Capacity –** The calculations methodology considers a constant averaged mass specific heat in the

suction & discharge as well as cooler volumes. However, in commercial solvers such as HYSYS 2006.5, the mass heat capacity is computed at every time step which affects the final SOP & SOT.

4. **Density & Z Variations** – In the calculations made, density and compressibility factor (Z) was assumed to be constant, whereas HYSYS provides density & *'Z'* corrections with change in temperature & pressure at every time step.

Design Standards (API 521/NORSOK)

1. In designing suction side of compressor piping & equipment, providing a design margin between settle out pressure and design pressure prevents unnecessary flaring. As per API 521, "Pressure relieving and Depressuring Systems", 5th Edition, Jan 2007, "Design Pressure shall be a minimum of 1.05 times the settle out pressure at maximum pressure drop, calculated assuming the suction side is operated at normal operating pressure and compressor discharge pressure is set to the maximum achievable".

2. As per NORSOK P-001, "The maximum operating pressure should be determined as the settle out pressure occurring at coincident PAHH" (High-High Pressure Alarm) "on both suction side and discharge side, adding a 10% margin for determining design pressure or PSV set pressure". Therefore, NORSOK P-001 standard provides a more conservative estimate of settle out pressure since it takes into account the highest possible suction & discharge pressures.

ANNEXURE A: SETTLE OUT CONDITIONS DERVIATION

The settle out conditions is calculated by considering the suction & discharge volumes as, Suction side gas mass

$$m_S = \left[\left(V_{Suction\,Side} - \%V_{Suction\,Scrubber\,Liquid\,Volume}\right) \times \rho_S\right]$$
(1)

Discharge side gas mass

$$m_D = \left[\left(V_{Discharge\,Side} - \%V_{Discharge\,Scrubber\,Liquid\,Volume}\right) \times \rho_D\right]$$
(2)

Performing heat balance over the closed loop system,

$$E_{In} = E_{Out} \tag{3}$$

Or, $$Q_{Suction} + Q_{Discharge} + Q_{CC} = Q_{Cooler} \tag{4}$$

Taking that the energy reaching the gas through the compressor is acting only on the mass of gas enclosed & calculating on a per second basis,

$$m_S c_{p,S}(T - T_S) + m_D c_{p,D}(T - T_D) + mH_P = Q_{Cooler} \tag{5}$$

Taking $m = (m_S + m_D)$ & rearranging Eq. (5)

$$T = SOT = \frac{(m_S c_{p,S} T_S + m_D c_{p,D} T_D) - Q_{Cooler} + (m_S + m_D)(H_P(t))}{m_S c_{p,S} + m_D c_{p,D}} \tag{6}$$

The mass specific heat for the cooler in Eq. (6) is taken to be an average value between the upstream & downstream flow. The polytropic head, $H_P(t)$ is treated as a function of time & is calculated by fitting the performance curves (Q vs. H_p).

$$\frac{H_p}{N^2} = A\left(\frac{Q}{N}\right)^2 + B\left(\frac{Q}{N}\right) + C \tag{7}$$

A graph is plotted between $\left(\frac{Q}{N}\right)$ (along x-axis) & $\frac{H_p}{N^2}$ (along y-axis) to obtain the constants A, B & C, followed by rewriting Eq. (E.7) as,

$$H_p(t) = A(Q)^2 + B(QN) + CN^2 \tag{8}$$

In Eq. (8), the compressor speed (N) is calculated as shown in Eq. (9)

$$N = N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t - t_0)}{(2\pi)^2 J}} \tag{9}$$

The volumetric flow calculated using Fan Laws assuming $k_1 = k_2$ during coast down is,

$$\frac{Q_t}{Q_{t+1}} = \frac{N_t}{N_{t+1}} \tag{10}$$

Or, $Q = Q_{t+1} = \frac{N_{t+1} \times Q_t}{N_t}$ (11)

It is to be noted that, the value of *'Q'* flowing into the compressor is approximated to value of *'m'* in Eq. (5) (which is constant) since the density lies between suction & discharge density. The settle out pressure is calculated using Ideal Gas equation as,

$$P = SOP = \frac{n \times \left[\frac{(Z_1 + Z_2)}{2}\right] \times R \times SOT}{V_{Total}}$$

(12)

Or, $SOP = \frac{m \times Z_{avg} \times R \times SOT}{MW \times (V_1 + V_2)}$ (13)

ANNEXURE B: COMPRESSOR COAST DOWN DERVIATION

The decay rate of driver speed is governed by the inertia of the system consisting of the compressor, coupling, gearbox & driver, which are counteracted by the torque transferred to the fluid. Neglecting the mechanical losses,

$$T = -(2\pi)J\left(\frac{dN}{dt}\right) \text{ [N-m]} \tag{1}$$

Where,
J = System Inertia (Compressor + gearbox + driver) [kg-m^2],
where,

$$J = J_C + \left[J_M \big/ (Gear\ Ratio)^2\right]$$

N = Compressor Rotor speed [rpm] or [min^{-1}]

The speed decay rate as well as the system inertia determines the compressor torque. Therefore, the power transferred to the gas, is

$$P = (2\pi NT)\ N \cdot m \big/ \min \tag{2}$$

Substituting Eq. (1) in Eq. (2), the power transferred during (ESD),

$$P = 2\pi N \times \left[-(2\pi)J\left(\frac{dN}{dt}\right)\right] \tag{3}$$

Applying fan power law as an approximation in which 'k' is relatively unvarying for a given curve,

$$P \propto N^3 \Rightarrow P = kN^3; k = \frac{60P}{N^3}\left(N \cdot m \cdot \min^2\right) \Rightarrow P = \frac{kN^3}{60}$$

(4)

Substituting Eq. (4) in Eq. (3),

$$\frac{kN^3}{60} = 2\pi N \times \left[-(2\pi)J\left(\frac{dN}{dt}\right)\right] \tag{5}$$

Rearranging,

$$\frac{dN}{dt} = \frac{kN^2}{-(2\pi)^2 J \times 60}\left(\frac{N}{kg \cdot m}\right) \Rightarrow \frac{dN}{dt} = \frac{kN^2}{-(2\pi)^2 J \times 60}\left(\frac{kg \cdot m}{kg \cdot m \cdot \sec^2}\right) \tag{6}$$

Integrating Eq. (6), and also multiplying by (60^2) to convert sec^2 (rev/s) to min^2 (rev/min)

$$\int_{N=N_0}^{N=N(t)} \frac{dN}{N^2} = \frac{k \times 60}{-(2\pi)^2 J}\int_{t_0}^{t=t} dt \tag{7}$$

$$\left[\frac{N^{-2+1}}{-2+1}\right]_{N_0}^{N(t)} = \frac{k \times 60}{-(2\pi)^2 J} \times (t - t_0) \Rightarrow \left[\frac{1}{N}\right]_{N_0}^{N(t)} = \frac{60k(t - t_0)}{(2\pi)^2 J}$$

(8)

$$\frac{1}{N(t)} - \frac{1}{N_0} = \frac{60k(t-t_0)}{(2\pi)^2 J} \Rightarrow \frac{1}{N(t)} = \frac{1}{N_0} + \frac{60k(t-t_0)}{(2\pi)^2 J} \Rightarrow N(t) = \frac{1}{\frac{1}{N_0} + \frac{60k(t-t_0)}{(2\pi)^2 J}}$$

(9)

Where, N_0 is the compressor speed before ESD. The 2nd denominator term exists with units N.m.min/kg.m^2 & is converted to min^{-1} which gives,

$$N(t) = \frac{1}{\frac{1}{N_0} + \frac{216000k(t - t_0)}{(2\pi)^2 J}} \tag{10}$$

References & Further Reading

1. www.ogj.com, Volume 113, Issue, 3, Feb 2015

Module 9

Gas Compression Stages – Process Design & Optimization

The following demonstrates how to estimate the required number of compression stages and optimize the individual pressure ratio in a multistage centrifugal compression system. A schematic of a 2-Stage compressor unit is,

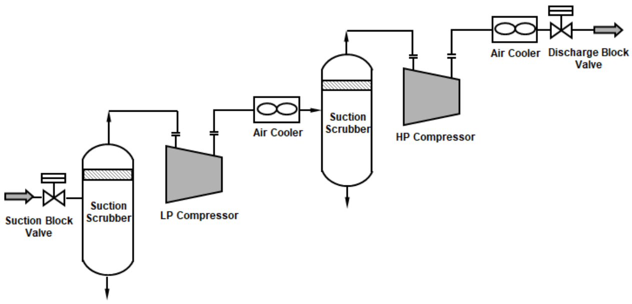


Fig 1. Two Stage Compressor Unit

A schematic of a 3-Stage Compressor Unit is,

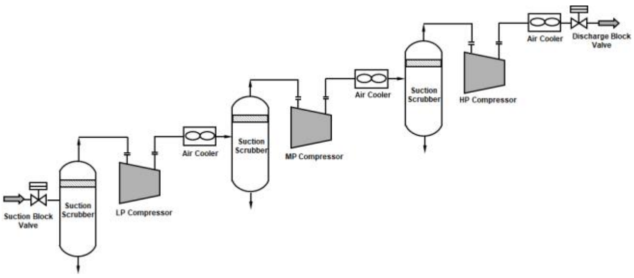


Fig 2. Three Stage Compressor Unit

General Notes

1. When vapours are compressed, its temperature increases & therefore requires provisions for gas cooling.
2. High gas temperatures can affect lube oil characteristics causing them to carbonize and turn in sludge. This results in fouling causing the bearing pads and seals to wear out and performance degradation.
3. As per API 617 (7th Edition, 2002), Clause 2.7.1.3, it states, As a design criteria, bearing metal temperatures shall not exceed 100°C (212°F) at specified operating conditions with a maximum inlet oil temperature of 50°C (120°F). Vendors shall provide bearing temperature alarm and shutdown limits on the datasheets. However clause No. 2.7.1.3.1 of the said document also says, In the event that the above design criteria cannot be met, purchaser and vendor shall mutually agree on acceptable bearing metal temperatures.
4. During gas recycling, (either by cold recycling or hot recycling), the compressor discharge temperature rises above the temperature pertaining to normal running conditions. Quantitatively, the rise in temperature depends on the pressure ratio of each stage. The maximum discharge temperature is typically limited to, in the range of 150^0C to 160^0C to avoid damage to the bearings and seals. To ensure these limits are not crossed, the compressor discharge temperature at normal running conditions must be operated at lower temperatures with a margin of 20^0C to 25^0C. This means typical compressor discharge temperatures (under normal running conditions) should be limited to the range of 120^0C to 135^0C.
5. Individual compressor pressure ratios must also be optimized to obtain the lowest amount of power required to meet the final discharge pressure. This also enables to reduce the suction scrubber volumes and air cooler duties to save on material and operating costs.

Case Study

A multistage compression system receives 30 MMScfd of hydrocarbon vapours at 2 bara, 30^0C and is required to be raised to 15 bara. The Polytropic efficiency [η] for all LP compressors is assumed to be 82%. An optimization study is performed for a 2-Stage and 3-Stage centrifugal compression system. The vapour composition is as follows,

Table 1. Gas Composition

Components	Mole Fraction [-]
Methane [C_1]	0.5232
Ethane [C_2]	0.3001
Propane [C_3]	0.1096
iso-Butane [iC_4]	0.0106
n-Butane [nC_4]	0.0346
Iso-Pentane [iC_5]	0.0076
n-Pentane [nC_5]	0.0092
n-Hexane [C_6]	0.0052
Total	**1.0000**
MW [kg/kmol] [PR EoS]	26.53
Density [1 atm, 15.6^{0}C] [kg/m^3]	1.128

Methodology

The number of compressors can be chosen by first estimating preliminary discharge pressures based on equal pressure ratio as,

$$X^n = \left[\frac{P_{Last}}{P_{First}}\right] \tag{1}$$

Where,

P_{First} = First Compressor Pressure [bara]

P_{Last} = Last Compressor Pressure [bara]

n = Number of stages [-]

X = Maximum number of Stages [-]

Rewriting the expression,

$$n \times lnX = ln\left[\frac{P_{Last}}{P_{First}}\right] \tag{2}$$

Or, $n = \dfrac{ln\left[\frac{P_{Last}}{P_{First}}\right]}{lnX}$ (3)

The separation ratio is computed as,

$$R = \left[\frac{P_{Last}}{P_{First}}\right]^{1/n} \tag{4}$$

The intermediate pressure is computed as,

$$P_i = P_{first} \times R^i \tag{5}$$

Where,

P_i = Intermediate Pressure at Stage '*i*'

Therefore considering a maximum number of stages of 3, for a two stage compressor unit, the first compressor discharge pressure [P_1] and Pressure ratio [R] is,

$$n = \frac{ln\left[\frac{15}{2}\right]}{ln[3]} = 1.83 \sim 2\ Stages \tag{6}$$

$$R = \left[\frac{15}{2}\right]^{1/2} = 2.7386 \tag{7}$$

$$P_1 = 2 \times 2.7386^1 = 5.48\ bara \tag{8}$$

For a three stage compressor unit, the LP compressor discharge pressure [P_1] and MP compressor discharge pressure [P_2] is,

$$R = \left[\frac{15}{2}\right]^{1/3} = 1.9574 \tag{9}$$

$$P_1 = 2 \times 1.9574^1 = 3.91\ bara \tag{10}$$

$$P_2 = 2 \times 1.9574^2 = 7.66\ bara \tag{11}$$

Using these preliminary values, to arrive at optimized discharge pressures, the following iterative procedure is adopted.

1. Keeping all preliminary estimated discharge pressures fixed, the LP compressor discharge pressure is varied for a range to obtain total absorbed power & total cooler duty of all compressors, and sizing each suction scrubber. Making a plot of the above values, the discharge pressure corresponding to the lowest duty is chosen [1st Iteration of LP Compressor].
2. The LP compressor initial estimated discharge pressure is now replaced with the 1st Iteration's optimized pressure.
3. Following further, the MP compressor discharge pressure is also varied for a given range to similarly obtain an optimized discharge pressure corresponding to the lowest total compressor duty and cooler duty. [1st Iteration of 2nd stage].
4. The MP compressor initial estimate pressure is now replaced with the optimized value, [1st Iteration of 2nd stage].

5. With the 1st iteration optimized pressures, calculations are repeated similar to Step 2 Step 3 & Step 4, i.e., 2nd Iteration and so forth, until a converged solution is reached.

Results

With the procedure applied for the calculated initial estimates, the optimized results of 2-Stage & 3-stage system [LP η_p = 82%] is,

Table 2. Optimized Compressor Stage Pressures

Stages-Pressure	Discharge Pressure	Pressure Ratio
-	[bara]	[-]
2 Stage LP [2S-LP]	8.12	4.060
2 Stage HP [2S-HP]	15.00	1.847
3 Stage LP [3S-LP]	6.15	3.075
3 Stage MP [3S-MP]	8.25	1.341
3 Stage HP [3S-HP]	15.00	1.818

The plots of total compressor absorbed power, total cooler duty for two stage design and three stage design is as follows,

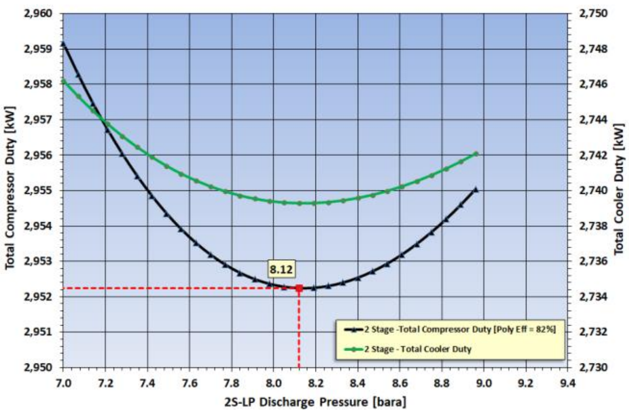


Fig 3. Two Stages –Total Compressor & Cooler Duty

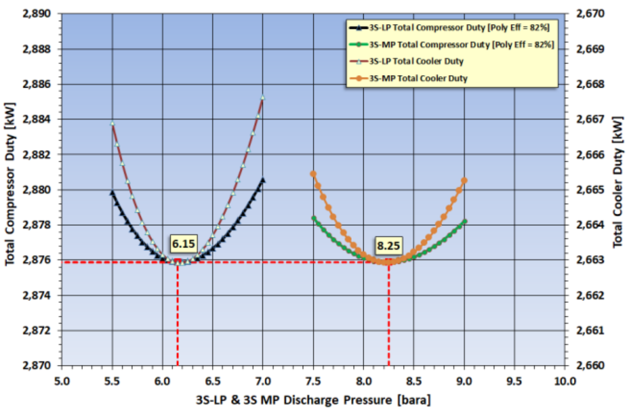


Fig 4. Two Stages – Total Compressor & Cooler Duty

Based on the optimized compression ratios, the savings on the total compressor duty and total air cooler duty is 1.59% and 1.68% for 2 stages respectively. For 3 stages, the respective savings is 1.86% and 2.03%.

Table 3. Savings on Compressor & Air Cooler Duty

Parameter	2 Stage	3 Stage
Before Optimization		
Total Comp. Duty [kW]	3,000	2,930
Total Cooler Duty [kW]	2,786	2,717
After Optimization		
Total Comp. Duty [kW]	2,952	2,876
Total Cooler Duty[kW]	2,739	2,663
% Savings [Compressor]	1.59%	1.86%
% Savings [Air Cooler]	1.68%	2.03%

Based on the optimized compression ratios, the suction scrubber sizes for both cases are,

Table 4. Suction Scrubber Sizes

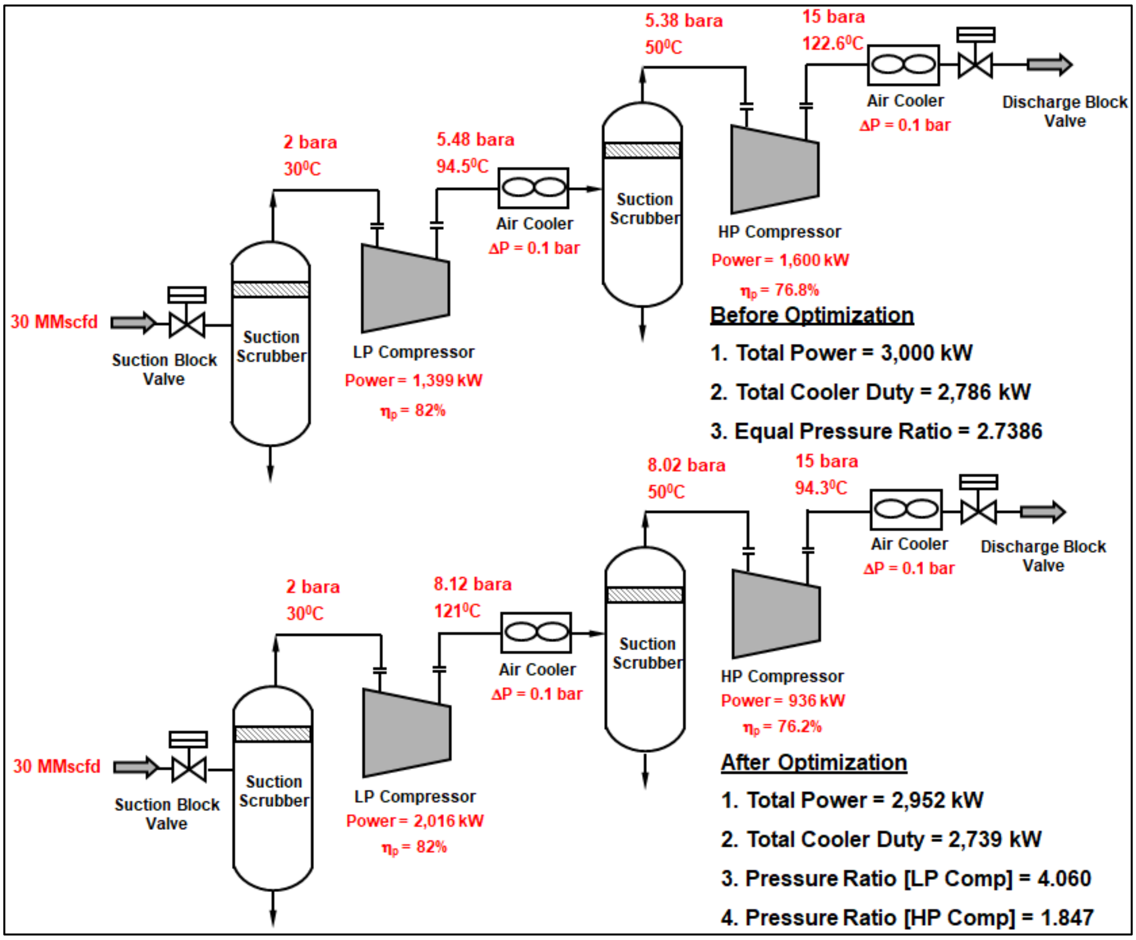
Suction Scrubber [H/D = 3.0]	Head Design [2:1 Elliptical]		
	D [mm]	H [mm]	Vessel Volume [m³]
2S-LP/3S-LP	2,400	7,200	34.08
Before Optimization			
2S-HP	1,900	5,700	17.11
3S-MP	2,100	6,300	22.98
3S-HP	1,800	5,400	14.59
After Optimization			
2S-HP	1,800	5,400	14.59
3S-MP	1,900	5,700	17.11
3S-HP	1,800	5,400	14.59

For 2S-HP & 3S-MP cases, the vessel volume decreases by 14.7% and 25.5% respectively.

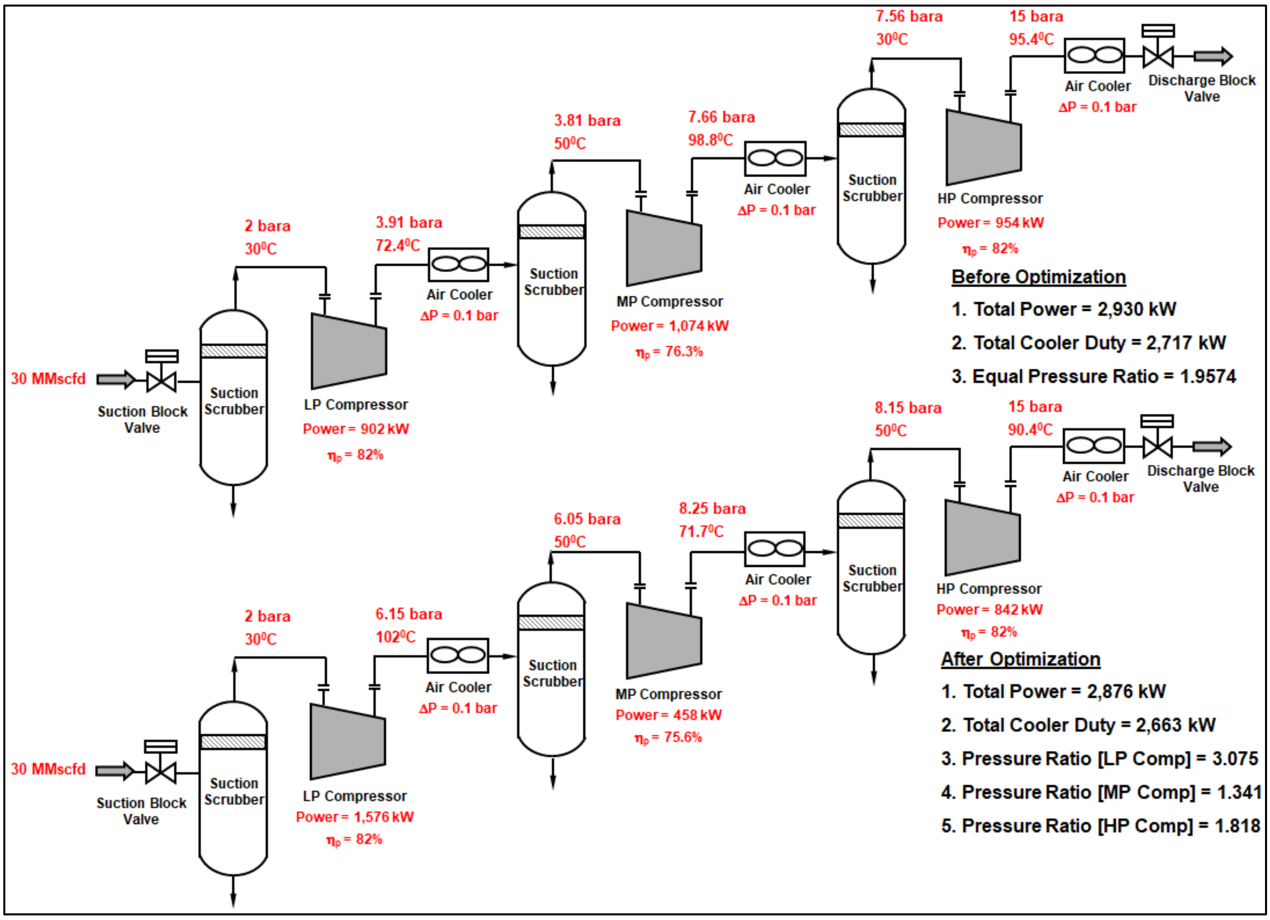
References & Further Reading

1. "*Example problems for the calculation and selection of compressors*", Intech GMBH, (intech-gmbh.com/compr_calc_and_selec_examples/)
2. www.checalc.com

Appendix A



Appendix B



Module 10

Design Considerations for Compressor Antisurge Valve Sizing

Centrifugal Compressors experience a phenomenon called "Surge" which can be defined as a situation where a flow reversal from the discharge side back into the compressor casing causing mechanical damage.

The reasons are multitude ranging from driver failure, power failure, upset process conditions, start up, shutdown, failure of anti-surge mechanisms, check valve failure to operator error to name a few. The consequences of surge are more mechanical in nature whereby ball bearings, seals, thrust bearing, collar shafts, impellers wear out and sometimes depending on the how powerful are the surge forces, cause fractures to the machinery parts due to excessive vibrations.

The following explains how to size an anti-surge valve for a single stage VSD system for Concept/Basic Engineering purposes.

General Notes & Assumptions

1. Centrifugal compressors are characterized by "Performance curves" which are a plot of Actual Inlet Volumetric Flow rate [Q] vs. Polytropic head [H_p] for various operating speeds. The operating limits for performance curves are the surge line and the choke flow line, beyond which any compressor operation can cause severe mechanical damage.

2. Below is an image of performance curves characteristics which indicates the surge flow line and choked flow line, both of which extend from the minimum speed Q vs. H_p curve to the maximum speed Q vs. H_p curve. The surge curve is defined as the Surge Limit Line [SLL] and an operating margin is provided [e.g., 10% on flow rate] which is called the surge control line [SCL].

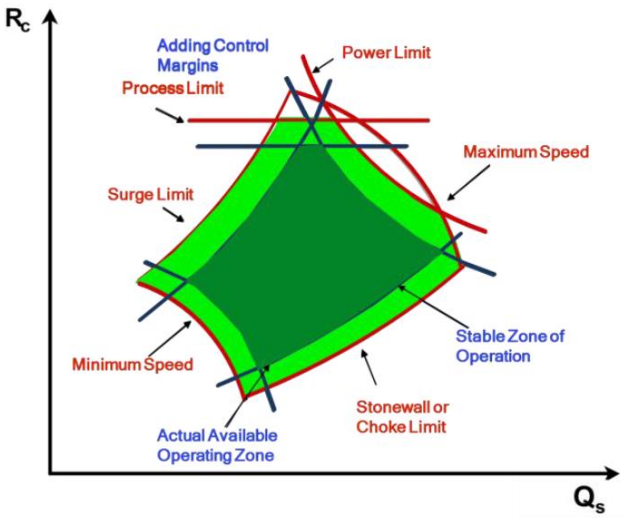


Figure 1. Performance Curves Operating Limits [1]

3. To ensure process safety & avoid mechanical damage, the anti-surge valve (ASV) must be large enough to recycle flow sufficiently. An undersized valve would fail to provide enough recycle flow to keep the compressor operating point away from SCL and SLL. Whereas over sizing the ASV leads to excess gas recycling that can drive the compressor into the choke flow region. Oversized valves also create difficulties in tuning the controllers due to large controller gain values and limited stroke.

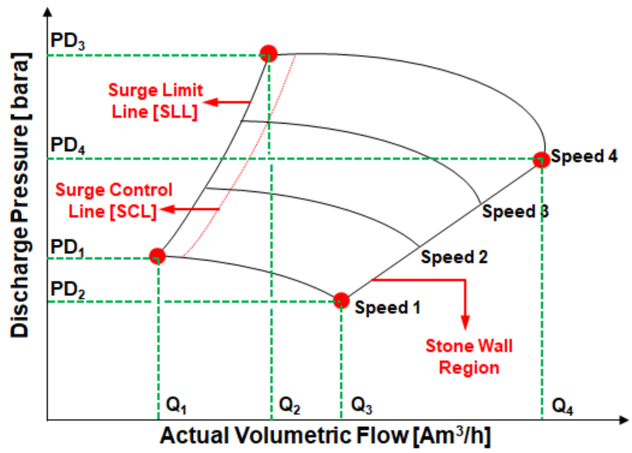


Figure 2. Sizing Criteria for Anti-surge Valve

4. To size the anti-surge valve (ASV), the philosophy employed should consider, operating the compressor on the right hand side of the SCL while also ensuring the operating point does not cross the

choke flow line. Towards this, the recycle flow rates across the ASV can be taken to be 1.8 to 2.2 times the surge flow rate.

5. Traditionally ASVs have linear opening characteristics, though sometimes equal percentage characteristics can be incorporated into the linear trend. Quick opening characteristics are not preferred due to poor throttling characteristics while Equal percentage valves suffer from slow opening during the early travel period.
6. The stroking time of the valve should be ideally less than 2 sec with less than 0.4 sec time delay and no overshoot. The actuator response time must be less than 100 msec and the noise limit is ~85 dBA. The maximum noise level allowed is 110 dBA.
7. Anti-surge valves are Fail-open [FO] type and should provide stable throttling. Fluid velocities should be less than 0.3 Mach to avoid piping damage and valve rattling.
8. The anti-surge valve can be operated pneumatically or by solenoid action. For valve sizes greater than 16", a motor operated valve can be used to effectuate the fast opening requirements.
9. Although the current module provides a methodology to size an ASV which is suitable during Concept/Basic Engineering stage, a compressor dynamic simulation shall be performed with the actual plant layout based on detailed design to verify if the ASV can cater to preventing a surge during start-up & shutdown scenarios.
10. The final ASV size must be verified and arrived in concurrence with the turbomachinery vendor, valve manufacturer, if the ASV can cater to the surge control philosophy employed, slope of the performance curves and polytropic efficiency maps at the choke points.

Anti-Surge Valve Sizing Methodology

To size the anti-surge valve, the ANSI/ISA S75.01 compressible fluid sizing expression is chosen for this exercise and the flow rates are taken for at least 1.8 to 2.2 times the surge flow rate.

Step 1: Piping Geometry Factor (F_p)

$$F_P = \left[1 + \frac{\sum K}{890}\left(\frac{C_V}{d^2}\right)^2\right]^{-1/2} \quad (1)$$

Where,

F_p = Piping geometric factor [-]

C_v = Valve coefficient [-]

d = Control valve size [inch]

ΣK = Sum of pipe resistance coefficients [-]

The value of F_p is dependent on the fittings such as reducers, elbows or tees that are directly attached to the inlet & outlet connections of the control valve. If there are no fittings, F_p is taken to be 1.0. The term ΣK is the algebraic sum of the velocity head loss coefficients of all the fittings that are attached to the control valve & is estimated as,

$$\sum K = K_1 + K_2 + K_{B1} - K_{B2} \quad (2)$$

Where,

K_1 = Upstream fitting resistance coefficient [-]

K_2 = Downstream resistance coefficient [-]

K_{B1} = Inlet Bernoulli Coefficient [-]

K_{B2} = Outlet Bernoulli Coefficient [-]

Where,

$$K_{B1} = 1 - \left(\frac{d}{D_1}\right)^4 \quad (3)$$

$$K_{B2} = 1 - \left(\frac{d}{D_2}\right)^4 \quad (4)$$

Where,

D_1 = Inlet Pipe Inner Diameter [in]

D_2 = Outlet Pipe Inner Diameter [in]

The most commonly used fitting in control valve installations is the short-length concentric reducer. The expressions are as follows,

$$K_1 = 0.5 \times \left[1 - \left(\frac{d^2}{D_1^2}\right)\right]^2 \text{, for inlet reducer} \quad (5)$$

$$K_2 = 1.0 \times \left[1 - \left(\frac{d^2}{D_2^2}\right)\right]^2 \text{, for outlet reducer} \quad (6)$$

Step 2: Calculate Valve Coefficient (C_v)

To calculate the valve C_v, the following ANSI/ISA expression is used.

$$C_v = \frac{M}{N_8 F_p P_1 Y \sqrt{\frac{X \times MW}{T_1 \times Z}}} \quad (7)$$

$$X = \frac{\Delta P}{P_1} \quad (8)$$

$$Y = 1 - \frac{X}{3 \times F_k \times X_T} \quad (9)$$

$$F_k = \frac{k_1}{1.4} \quad (10)$$

If $X > F_k \times X_T$, then flow is Critical.

If $X < F_k \times X_T$, then flow is Subcritical.

For Critical flow, the value of 'X' is replaced with $F_k \times X_T$ and the gas expansion Factor [Y] and valve coefficient [C_v] is to be computed as,

$$Y = 1 - \frac{F_k \times X_T}{3 \times F_k \times X_T} = 0.667 \quad (11)$$

$$C_v = \frac{M}{0.667 \times N_8 F_p P_1 \sqrt{\frac{F_k \times X_T \times MW}{T_1 \times Z}}} \quad (12)$$

If the control valve inlet and outlet piping is provided with reducers and expanders, then the value of X_T is replaced with X_{TP} as follows,

$$X_{TP} = \frac{X_T}{F_p^2} \times \left[1 + \frac{X_T(K_1 + K_{B1})}{1000}\left(\frac{C_v}{D_1^2}\right)^2\right]^{-1} \quad (13)$$

Where,

C_v = C_v value at Valve 100% Open [-]

M = Mass Flow Rate [kg/h]

N_8 = Constant [Value = 94.8]

F_p = Piping Geometry Factor [-]

ΔP = Pressure drop across ASV [bar]

P_1 = Inlet Pressure [bara]

Y = Gas Expansion Factor [-]

X = Pressure Drop Ratio [-]

Z = Gas compressibility Factor [-]

T_1 = Inlet Temperature [°K]

F_k = Gas specific heat to air specific heat ratio

k_1 = Gas specific heat ratio at valve inlet [-]

X_{TP} *and* X_T = Pressure drop ratio factor [-]

MW = Molecular Weight of gas [kg/kmol]

To estimate the compressor mass flow rate from the suction density [ρ_s] and compressor actual inlet flow rate, it can be estimated as,

$$\rho_s = \frac{P \times MW}{Z \times R \times T} \quad (14)$$

$$M = Q_s \times \rho_s \quad (15)$$

Where,

R = Gas Constant [0.0831447 m^3.bar/kmol.K]

Q_s = Compressor Suction Vol flow rate [m^3/h]

To arrive at a converged value of F_p, the valve C_v at each iteration, can be computed iteratively by replacing the F_p value in each iteration of the C_v equation. Applying the Sizing method, to the four points shown in Figure 2, the various sizing scenarios are,

a. Minimum Speed - Surge Flow [Q_1]

b. Minimum Speed - Surge Flow [$Q_1 \times 1.8$]

c. Minimum Speed - Surge Flow [$Q_1 \times 2.2$]

d. Maximum Speed - Surge Flow [Q_2]

e. Maximum Speed - Surge Flow [$Q_2 \times 1.8$]

f. Maximum Speed - Surge Flow [$Q_2 \times 2.2$]

g. Minimum Speed - Choke Flow [Q_3]

h. Maximum Speed - Choke Flow [Q_4]

The ASV C_v computed for the surge points would be closer to each other in most cases. Similarly, the ASV C_v at the choke points would also be closer to each other. Therefore, to arrive at conservative results, the higher of the C_v values at the surge points & the lower of the C_v values at the choke points are to be considered to determine a suitable ASV size.

Case Study

68.1 MMscfd of hydrocarbon gas at 11.61 bara [suction flange conditions] and 47.47⁰C is to be compressed to 30.13 bara pressure [discharge flange conditions]. The compressed gas is cooled to 50⁰C via an air cooler. The centrifugal compressor used is a variable speed configuration. The gas composition is as follows,

Table 1. Gas Composition

Parameter	Mol %
Methane [CH_4]	94.09
Ethane [C_2H_6]	0.03
Propane [C_3H_8]	0.02
Nitrogen [N_2]	3.93
Carbon Dioxide [CO_2]	0.96
Water [H_2O]	0.97
Total	**100**

The compressor performance curves for various operating speeds are as follows,

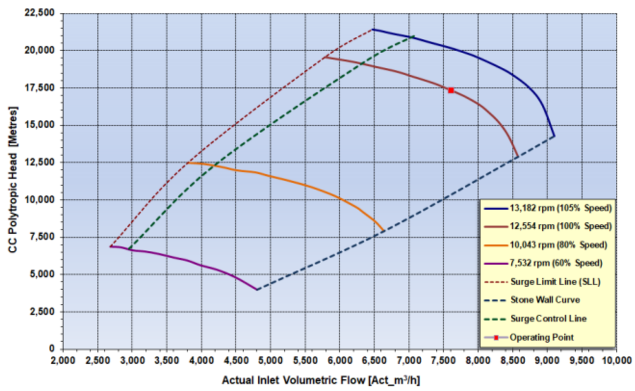


Figure 3. Compressor Performance Curves

The upstream and downstream piping for the anti-surge line is taken as NPS 4”, Ref [2] with a thickness of 0.237 inches for this exercise. The anti-surge valve chosen to be checked is a NPS 4” valve [OD 4.5”] [Single ported, Cage Guided, Globe Style Valve body] with a C_v of 236 and corresponding X_T value of 0.69.

The surge control line [SCL] chosen for this exercise is taken as 10% on the surge flow rate at each speed and is as follows,

Table 2. Surge Control Line [SCL] Parameters

Speed	Surge Flow × 10%	H_P
[rpm]	[Act_m³/h]	[m]
7,532	2,952	6,721
10,043	4,184	12,297
12,544	6,363	19,263
13,182	7,118	21,077

The Gas Properties are as follows for the suction and discharge flange conditions,

Table 3. Gas Properties at Flange Conditions

Parameter	Value	Units
Gas MW	16.81	kg/kmol
Suction Pressure	11.61	bara
Suction Temperature	47.5	⁰C
Discharge Pressure	30.13	bara
Discharge Temperature	143.0	°C
Inlet Z [Z_1]	0.9810	-
Outlet Z [Z_2]	0.9848	-
Specific Heat of Gas - Inlet	1.3229	-
Suction Density	7.464	kg/m³
Discharge Density	14.868	kg/m³
Actual Volumetric Flow	7,611	Am³/h
Inlet Mass Flow	56,809	kg/h

The compressor parameters are as follows,

Table 4. Compressor Parameters

Parameter	Value	Units
Adiabatic Head	16,887	m
Polytropic Head	17,333	m
Adiabatic Efficiency	77.61	%
Polytropic Efficiency	79.71	%
Power Consumed	3,365	kW
Polytropic Head Factor	1.0009	-
Polytropic Exponent	1.3839	-
Isentropic Exponent	1.2881	-

ASV Sizing Solution

Proceeding with the C_v calculation for the case of Minimum Speed - Surge Flow [Q_1],

$$K_{B1} = 1 - \left(\frac{4}{4.026}\right)^4 = 0.0256 \quad (16)$$

$$K_{B2} = 1 - \left(\frac{4}{4.026}\right)^4 = 0.0256 \quad (17)$$

$$K_1 = 0.5 \times \left[1 - \left(\frac{4^2}{4.026^2}\right)\right]^2 = 0.000083 \quad (18)$$

$$K_2 = 1.0 \times \left[1 - \left(\frac{4^2}{4.026^2}\right)\right]^2 = 0.00017 \quad (19)$$

$$\sum K = 0.000083 + 0.00017 + 0.26 - 0.26 \quad (20)$$

$$\sum K = 0.00025 \quad (21)$$

$$F_P = \left[1 + \frac{0.00025}{890}\left(\frac{236}{4^2}\right)^2\right]^{-1/2} = 1 \quad (22)$$

The flow rate for the minimum speed - surge flow is 2,683 Am^3/h and gas density at compressor inlet is,

$$\rho_s = \frac{11.61 \times 16.81}{0.981 \times 0.0831447 \times 320.62} \approx 7.464\ \frac{kg}{m^3} \quad (23)$$

$$M = 2{,}683 \times 7.464 \approx 20{,}028\frac{kg}{h} \quad (24)$$

The compressor discharge flange pressure is 17.44 bara at minimum speed surge flow of 2,683 Am^3/h and a discharge air cooler which offers a pressure drop for the flowing gas. Taking a max ΔP of 0.35 bar across the discharge side, the ASV inlet pressure becomes, 17.44 – 0.35 = 17.09 bara. The cooler discharge temperature is 50^0C, therefore neglecting heat losses; the ASV inlet temperature also is at 50^0C.

Making an approximation that the ASV discharge side piping and compressor suction side ΔP is negligible; the ASV outlet pressure is nearly equal to the compressor inlet pressure. Therefore the ASV outlet pressure becomes 11.61 bara. The ASV ΔP is,

$$\Delta P = 17.09 - 11.61 = 5.48\ bar \quad (25)$$

The ASV Inlet Z & k_1 value [C_p/C_v] at 17.09 bara and 50^0C is 0.9732 and 1.3348.

The gas specific heat ratio to air specific heat ratio is calculated as,

$$F_k = \frac{1.3348}{1.4} = 0.9534 \quad (26)$$

The pressure drop ratio factor [X_T] is,

$$X = \frac{5.48}{17.09} = 0.321 \quad (27)$$

Since the valve construction details are available, X_{TP} is used instead of X_T.

$$X_{TP} = \frac{0.69}{1^2} \times \left[1 + \frac{0.69(0+0.0256)}{1000}\left(\frac{236}{4^2}\right)^2\right]^{-1} \quad (28)$$

$$X_{TP} \approx 0.69 \quad (29)$$

Checking for flow condition,

$$F_k \times X_{TP} = 0.9534 \times 0.69 = 0.6579 \quad (30)$$

Since $X < F_k \times X_{TP}$, flow is Subcritical.

The gas expansion factor is estimated as,

$$Y = 1 - \frac{0.321}{3 \times 0.9534 \times 0.69} = 0.8374 \quad (31)$$

Therefore the ASV C_v is computed as,

$$C_v = \frac{20{,}028}{94.8 \times 1 \times 17.09 \times 0.8374\sqrt{\frac{0.321 \times 16.81}{323.15 \times 0.9732}}} \quad (32)$$

$$C_v = 112.7 \quad (33)$$

Re-inserting the value of C_v = 112.72 into the F_p expression to iterate, the value of C_v becomes,

$$F_P = \left[1 + \frac{0.00025}{890}\left(\frac{112.7}{4^2}\right)^2\right]^{-1/2} = 0.9999 \quad (34)$$

$$C_v = \frac{20{,}028}{94.8 \times 0.9999 \times 17.09 \times 0.8375\sqrt{\frac{0.321 \times 16.81}{323.15 \times 0.9732}}} \quad (35)$$

$$C_v = 112.719 \sim 113 \quad (36)$$

Therefore with another iteration the C_v value remains nearly the same at 112.72 ~ 113.

The ASV C_v can now be estimated for the case of Q × 1.8 at $C_{v,min}$ and Q × 2.2 at $C_{v,max}$.

$$C_{v,min} = 1.8 \times 113 = 203 \quad (37)$$

$$C_{v,max} = 2.2 \times 113 = 248 \quad (38)$$

Performing similar calculations for all cases,

Table 5. ASV Sizing Cases – Surge Points

Parameter	Min Surge	Max Surge	Units
Q_s	2,683	6,471	Am^3/h
ρ_s	7.46	7.46	kg/m^3
M	20,028	48,295	kg/h
P_D	17.44	36	bara
Discharge ΔP	0.35	0.35	bar
ASV Inlet P_1	17.09	35.99	bara
ASV Outlet P_2	11.61	11.61	bara
ASV ΔP	5.48	24.38	bar
Cooler Outlet T	323.15	323.15	^0K
ASV Inlet Z	0.9732	0.9465	-
C_p/C_v-ASV Inlet	1.3348	1.3781	-
X_T	0.69	0.69	-
F_k - ASV Outlet	0.9534	0.9843	-
X	0.321	0.677	-
X_{TP}	0.690	0.690	-
Flow Condition	Subcritical	Subcritical	-
$C_{v,\,Min}$	**113**	**110**	**-**
$C_{v,\,Min}$ [Q x 1.8]	**203**	**198**	**-**
$C_{v,\,Max}$ [Q x 2.2]	**248**	**242**	**-**

Table 6. ASV Sizing Cases – Choke Points

Parameter	Min Choke	Max Choke	Units
Q_s	4,805	9,102	Am^3/h
ρ_s	7.46	7.46	kg/m^3
M	35,860	67,932	kg/h
P_D	14.77	25.45	bara
Discharge ΔP	0.35	0.35	bar
ASV P_1	14.42	25.10	bara
ASV P_2	11.61	11.61	bara
ASV ΔP	2.81	13.49	bar
Cooler T	323.15	323.15	^0K
ASV Inlet Z	0.9769	0.9615	-
C_p/C_v	1.3279	1.3511	-
X_T	0.69	0.69	-
F_k	0.9485	0.9651	-
X	0.195	0.537	-
X_{TP}	0.690	0.690	-
Flow Condition	Subcritical	Subcritical	-
$C_{v,\,Choke}$	**286**	**229**	**-**

From the C_v values calculated, the governing case becomes the Min Speed surge point case.

$$C_{v,min} = 203 \leq C_v = 236 \leq C_v = 248 \quad (39)$$

Hence the selected 4" control valve with a C_v of 236 and X_T of 0.69 is adequately sized to provide anti-surge control.

Transient Study to Verify ASV Sizing

With the ASV size selected, a transient study is performed to check for ASV adequacy. Centrifugal compressors during shutdown experience surging & the ASV must be able to provide sufficient cold recycle flow to keep the operating point away from the SLL as the compressor coasts down.

Normal shutdown [NSD] refers to a planned event where the anti-surge valve is opened first by 100%, prior to a compressor trip.

An emergency shutdown [ESD] is an unplanned event, where for example, upon loss of driver power, the ASV opens quickly to recycle flow and prevent the operating point from crossing the SLL during coast down. For this module, the ESD case considered is "Driver trip" where the compressor driver experiences a sudden loss of power.

To simulate the transient case, the air cooler and suction scrubber can be sized with preliminary estimates to cater to maximum speed choke flow case.

Suction Scrubber Volume

Using GPSA K-Value method for suction scrubber sizing, Ref [3], for a flow rate of 67,932 kg/h and 11.61 bara operating pressure, the suction scrubber size is H × D of 6.9m × 2.3m with an ellipsoidal head and inside dish depth of 0.25m. The total scrubber volume is 30.1 m^3.

Air Cooler Volume

Similarly, the air cooler is sized for maximum speed choke flow case, Ref [4], for a flow rate of 67,932 kg/h & duty of 4,351 kW. The overall heat transfer coefficient [U] is assumed to be 25 W/m^2.K. The inlet temperature is 142^{0}C which is cooled to 50^{0}C with an air side temperature of 35^{0}C. The air cooler geometry chosen for this exercise is a single tube pass with 3 tube rows & each tube is 9.144m in length. The fan & motor efficiencies are taken as 75% and 95% respectively. With this data, the air cooler has a tube OD of 1" [0.0254m] & total number of tubes of 307 [Tube volume of 1.423 m^3].

Compressor Coast down

Coast down time is influenced by a number of factors including fluid resistance, dynamic imbalance, misalignment between shafts, leakage and improper lubrication, skewed bearings, radial or axial rubbing, temperature effects, transfer of system stresses, resonance effect to name a few and therefore in reality, shutdown times can be lower than estimated by the method shown below.

The decay rate of driver speed is governed by the inertia of the system consisting of the compressor, coupling, gearbox & driver, which are counteracted by the torque transferred to the fluid. Neglecting the mechanical losses, the compressor speed decay rate can be estimated as,

$$N[t] = \frac{1}{\frac{1}{N_O} + \frac{216{,}000 \times k \times [t - t_O]}{[2\pi]^2 \times J}} \qquad (42)$$

Where, 'N_0' is the compressor speed before ESD, 'J' is the total system inertia & 't_0' is time at which the ESD is initiated. For this exercise the total system inertia is taken as 108 kg.m^2. The coast down speed calculated is,

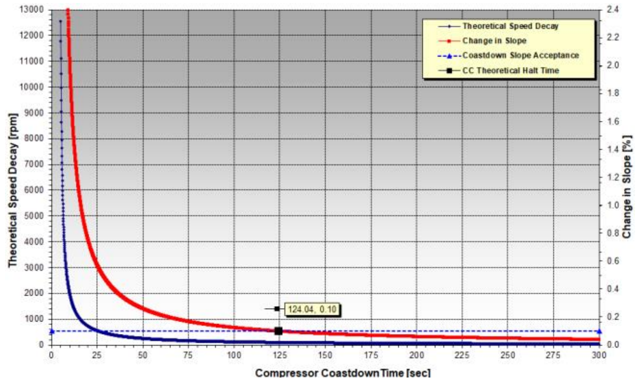


Figure 4. Compressor Coast down Time

From the curve, the compressor is expected to reach a standstill in ~124 sec.

ESD and NSD Analysis

With the equipment volumes, ASV C_v chosen and compressor speed decay rate imposed, an ESD and NSD analyses is performed to track operating point during coast down.

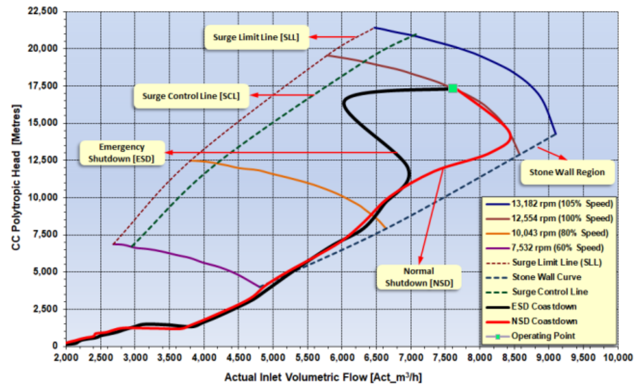


Figure 5. ESD/NSD Operating Point Migration

From the analysis made, it is seen that the selected ASV size of 4" [C_v 236] is sufficient to prevent a surge during ESD and NSD.

References & Further Reading

1. "*Development and Design of Antisurge and Performance Control Systems for Centrifugal Compressors*", Mirsky S., McWhirter J., Jacobson W., Zaghloul M., Tiscornia D., 3rd M.E Turbomachinery Symposium, Feb 2015
2. Control Valve Handbook, Emerson, 5th Ed.
3. https://checalc.com/calc/vertsep.html
4. https://checalc.com/calc/AirExch.html
5. https://www.slideshare.net/VijaySarathy7/centrifugal-compressor-settle-out-conditions-tutorial

Appendix A

Gas Composition		
Component	Mole Fraction	Mole %
Methane	0.9408	94.081
Ethane	0.0003	0.034
Propane	0.0002	0.022
Nitrogen	0.0393	3.929
CO_2	0.0096	0.964
H_2O	0.0097	0.970

Gas Properties [PR EoS] & CC Parameters		
Gas MW	16.81	kg/kmol
Suction Pressure	1,161	kPa
Suction Temperature	47.5	^0C
Discharge Pressure	3,013	kPa
Discharge Temperature	143.0	^0C
Inlet Z [Z_1]	0.9810	-
Outlet Z [Z_2]	0.9848	-
Specific Heat of Gas - Inlet	1.3229	-
Suction Density	7.464	kg/m^3
Discharge Density	14.868	kg/m^3
Actual Vol Flow	7,611	Am^3/h
Inlet Mass Flow	56,809	kg/h

Compressor Duty Details		
Adiabatic Head	16,877	m
Polytropic Head	17,333	m
Adiabatic Fluid Head	165.51	kJ/kg
Polytropic Fluid Head	169.98	kJ/kg
Adiabatic Efficiency	77.61	%
Polytropic Efficiency	79.71	%
Power Consumed	3,365	kW
Polytropic Head Factor	1.0009	-
Polytropic Exponent	1.3839	-
Isentropic Exponent	1.2881	-

Equipment Volumes		
Suction Scrubber Volume	30.10	m^3
Cooler Volume	1.423	m^3

Piping Details		
U/S Pipe Size [OD]	4.5	in
U/S Pipe Wall Thickness [WT]	0.237	in
U/S Pipe Inner Diameter [ID]	4.026	in
D/S Pipe Size [OD]	4.5	in
D/S Pipe Wall Thickness [WT]	0.237	in
D/S Pipe Outer Diameter [OD]	4.026	in

Control Valve Cv		
Control Valve [Cv% = Open%]		Linear
Size, *d* [inch]	C_v @100%	X_T
2	72.9	0.64
3	148	0.62
4	236	0.69
6	433	0.74

Surge Limit Line [SLL]		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
2,683	6,866	75.4
3,804	12,485	76.0
5,785	19,558	78.1
6,471	21,419	78.6

Stone Wall Curve		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
4,805	3,988	60.9
6,640	7,908	66.6
8,578	12,892	67.3
9,102	14,271	66.0

60% Speed - 7,532 rpm		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
2,683	6,866	75.4
2,835	6,822	75.9
3,005	6,623	76.0
3,186	6,542	75.8
3,384	6,380	76.1
3,600	6,137	76.1
3,799	5,937	75.3
3,992	5,614	73.8
4,220	5,329	72.1
4,483	4,834	68.4
4,805	3,988	60.9

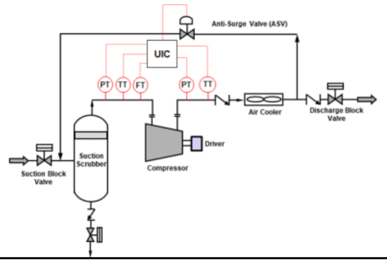
80% Speed - 10,043 rpm		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
3,804	12,485	76.0
4,037	12,407	76.6
4,300	12,195	77.6
4,511	11,981	78.7
4,779	11,843	78.7
4,966	11,620	78.7
5,288	11,255	78.7
5,592	10,856	78.5
5,896	10,328	77.1
6,077	9,918	74.9
6,258	9,454	74.3
6,335	9,185	72.7
6,510	8,557	69.5
6,640	7,908	66.6

100% Speed - 12,554 rpm		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
5,785	19,558	78.1
6,071	19,347	78.8
6,276	19,167	78.9
6,515	18,913	79.5
6,825	18,593	79.8
7,047	18,271	80.1
7,345	17,834	79.9
7,556	17,438	79.8
7,790	16,968	79.4
8,019	16,371	78.3
8,154	15,860	77.6
8,300	15,188	76.0
8,442	14,253	71.9
8,578	12,892	67.3

105% Speed - 13,182 rpm		
Q	Hp	η_p
[Act_m^3/h]	[m]	[%]
6,471	21,419	78.6
6,582	21,327	79.0
6,763	21,135	79.0
7,049	20,870	79.5
7,307	20,544	79.8
7,587	20,194	80.0
7,850	19,801	80.0
7,985	19,561	79.8
8,278	18,939	79.6
8,483	18,423	79.0
8,741	17,506	77.4
8,929	16,363	73.7
9,102	14,271	66.0

Piping Resistances [ΣK]		
N_2 [Constant]	890	-
Selected Control Valve Size	4	in
C_v of Selected Valve	236	-
K_1 [U/S Fitting Resistance Coefficient]	0.000	-
K_2 [D/S Fitting Resistance Coefficient]	0.000	-
K_{B1} [Inlet Bernoulli Coefficient]	0.026	-
K_{B2} [Outlet Bernoulli Coefficient]	0.026	-
ΣK [Sum of Resitances]	0.000	-

Selected Antisurge Valve (ASV) - Governing Case		
$C_{v,Min}$	203	Is Cv,Min <= Cv,Chosen <= Cv,Max
$C_{v,Chosen}$	236	ASV Adequate
$C_{v,Max}$	248	ASV Not Oversized



COMPRESSIBLE FLUID CONTROL VALVE SIZING - ANSI/ISA S75.01 [Ref: Emerson Control Valve Handbook, 5th Edition]

Process Input Data - ASV $C_{v,Min\ surge}$ - $Q_{Min,\ surge}$		
Parameter	Value	Units
Service	$C_{v,\ Min\ Surge}$ [Q]	
Flow Rate [Q] At Compressor Inlet	2,683	Am^3/h
Gas Density at Compressor Inlet	7.46	kg/m^3
Flow Rate [m] at Compressor Inlet	20,028	kg/h
Compressor Discharge Pressure	17.44	bara
Compressor Discharge Side ΔP	0.35	bar
ASV Inlet Pressure [P_1]	17.09	bara
ASV Oulet Pressure [P_2]	11.61	bara
ASV Presure Drop [ΔP]	5.48	bar
Inlet Temperature [T_1] [Cooler Discharge]	323.15	^0K
ASV Inlet Compressibility Factor [Z]	0.9732	-
C_p/C_v [ASV Inlet]	1.3348	-
Pressure Drop Ratio Factor [x_T]	0.69	-
F_k at ASV Outlet	0.9534	-
Pressure Drop Ratio [X]	0.321	-
Pressure Drop Ratio Factor [x_{TP}]	0.690	-
Flow Condition Check	Subcritical	-
Required Control Valve Size		
$C_{v,\ Min}$	113	-
$C_{v,\ Min}$ [Q x 1.8]	203	-
$C_{v,\ Max}$ [Q x 2.2]	248	-

Process Input Data - ASV $C_{v,Max\ surge}$ - $Q_{Max,\ surge}$		
Parameter	Value	Units
Service	$C_{v,\ Max\ Surge}$ [Q]	
Flow Rate [Q] At Compressor Inlet	6,471	Am^3/h
Gas Density at Compressor Inlet	7.46	kg/m^3
Flow Rate [m] at Compressor Inlet	48,295	kg/h
Compressor Discharge Pressure	36.34	bara
Compressor Discharge Side ΔP	0.35	bar
ASV Inlet Pressure [P_1]	35.99	bara
ASV Oulet Pressure [P_2]	11.61	bara
ASV Presure Drop [ΔP]	24.38	bar
Inlet Temperature [T_1] [Cooler Discharge]	323.15	^0K
ASV Inlet Compressibility Factor [Z]	0.9465	-
C_p/C_v [ASV Inlet]	1.3781	-
Pressure Drop Ratio Factor [x_T]	0.69	-
F_k at ASV Outlet	0.9843	-
Pressure Drop Ratio [X]	0.677	-
Pressure Drop Ratio Factor [x_{TP}]	0.690	-
Flow Condition Check	Subcritical	-
Required Control Valve Size		
$C_{v,\ Max}$	110	-
$C_{v,\ Min}$ [Q x 1.8]	198	-
$C_{v,\ Max}$ [Q x 2.2]	242	-

F_p & X_{TP} Calculations				
Iteration	F_p	X_{TP}	Y	C_v
1	1.00	0.690	0.84	113
2	1.00	0.69	0.84	113
3	1.00	0.69	0.84	113
4	1.00	0.69	0.84	113
5	1.00	0.69	0.84	113

F_p & X_{TP} Calculations				
Iteration	F_p	X_{TP}	Y	C_v
1	1.00	0.69	0.67	110
2	1.00	0.69	0.67	110
3	1.00	0.69	0.67	110
4	1.00	0.69	0.67	110
5	1.00	0.69	0.67	110

Process Input Data - ASV $C_{v,Min\ choke}$ - $Q_{Min,\ choke}$		
Parameter	Value	Units
Service	$C_{v,\ Min\ Choke}$ [Q]	
Flow Rate [Q] At Compressor Inlet	4,805	Am^3/h
Gas Density at Compressor Inlet	7.46	kg/m^3
Flow Rate [m] at Compressor Inlet	35,860	kg/h
Compressor Discharge Pressure	14.77	bara
Compressor Discharge Side ΔP	0.35	bar
ASV Inlet Pressure [P_1]	14.42	bara
ASV Oulet Pressure [P_2]	11.61	bara
ASV Presure Drop [ΔP]	2.81	bar
Inlet Temperature [T_1] [Cooler Discharge]	323.15	^0K
ASV Inlet Compressibility Factor [Z]	0.9769	-
C_p/C_v [ASV Inlet]	1.3279	-
Pressure Drop Ratio Factor [x_T]	0.69	-
F_k at ASV Outlet	0.9485	-
Pressure Drop Ratio [X]	0.195	-
Pressure Drop Ratio Factor [x_{TP}]	0.690	-
Flow Condition Check	Subcritical	-
Required Control Valve Size		
$C_{v,\ Min\ Choke}$	286	-

Process Input Data - ASV $C_{v,Max\ choke}$ - $Q_{Max,\ choke}$		
Parameter	Value	Units
Service	$C_{v,\ Max\ Choke}$ [Q]	
Flow Rate [Q] At Compressor Inlet	9,102	Am^3/h
Gas Density at Compressor Inlet	7.46	kg/m^3
Flow Rate [m] at Compressor Inlet	67,932	kg/h
Compressor Discharge Pressure	25.45	bara
Compressor Discharge Side ΔP	0.35	bar
ASV Inlet Pressure [P_1]	25.10	bara
ASV Oulet Pressure [P_2]	11.61	bara
ASV Presure Drop [ΔP]	13.49	bar
Inlet Temperature [T_1] [Cooler Discharge]	323.15	^0K
ASV Inlet Compressibility Factor [Z]	0.9615	-
C_p/C_v [ASV Inlet]	1.3511	-
Pressure Drop Ratio Factor [x_T]	0.69	-
F_k at ASV Outlet	0.9651	-
Pressure Drop Ratio [X]	0.537	-
Pressure Drop Ratio Factor [x_{TP}]	0.690	-
Flow Condition Check	Subcritical	-
Required Control Valve Size		
$C_{v,\ Max\ Choke}$	229	-

F_p & X_{TP} Calculations				
Iteration	F_p	X_{TP}	Y	C_v
1	1.0000	0.69	0.90	286
2	1.00	0.69	0.90	286
3	1.00	0.69	0.90	286
4	1.00	0.69	0.90	286
5	1.00	0.69	0.90	286

F_p & X_{TP} Calculations				
Iteration	F_p	X_{TP}	Y	C_v
1	1.00	0.69	0.73	229
2	1.00	0.69	0.73	229
3	1.00	0.69	0.73	229
4	1.00	0.69	0.73	229
5	1.00	0.69	0.73	229

Module 11

Process Design for Natural Gas Transmission

Compressor stations form a key part of the natural gas pipeline network that moves natural gas from individual producing well sites to end users. As natural gas moves through a pipeline, distance, friction, and elevation differences slow the movement of the gas, and reduce pressure. Compressor stations are placed strategically within the gathering and transportation pipeline network to help maintain the pressure and flow of gas to market. The following is a module to perform process design of a natural gas transmission system.

Problem Statement

80 MMscfd of Natural Gas at 36 bara & 48^{0}C is to be transmitted from a gas plant in a desert region to a city power station located 50 km away. The gas composition & critical properties are as follows,

Table 1. Natural Gas Composition & Properties

Component	MW	Mol%	$P_{c,i}$	$T_{c,i}$
-	kg/kmol	%	psia	^{0}R
Methane	16.04	76.23	667.8	343.0
Ethane	30.07	10.00	707.8	549.8
Propane	44.01	5.00	616.3	665.7
i-Butane	58.12	1.00	550.7	765.3
n-Butane	58.12	1.00	529.1	734.6
i-Pentane	72.15	0.30	490.4	828.7
n-Pentane	72.15	0.10	488.6	845.3
n-Hexane	86.18	0.05	436.9	913.3
C_7+	119.00	0.05	453.0	1116.0
H_2O	18.02	0.25	3206.2	1164.9
CO_2	44.01	3.00	1071.0	547.5
H_2S	34.08	0.02	1306.0	672.3
N_2	28.01	3.00	493.0	226.97

During transmission, gas pressure drops for which a booster station is installed en-route. The minimum pressure required at the booster station is 10 bara & the ambient pressure and temperature is 1.01325 bara & 30^{0}C. The design pressure & design temperature of the pipeline-compressor transmission system is 40 bara & 200^{0}C.

The requirement to be met for pipeline wall stresses is ASME B31.8. As per ASME B31.8, the Design factor [F], Temperature De-rating [T], Longitudinal Joint Factor [E] for the chosen pipeline joining methods is as follows,

Table 2. Reference Mechanical Design Parameters

Design Factors [F] - Gas Pipeline Location		
Class	**Description**	**F**
Class 1, Div 1	Deserted	0.80
Class 1, Div 2	Deserted	0.72
Class 2	Village	0.60
Class 3	City	0.50
Class 4	Densely Populated	0.40
Temperature De-rating [T] for Gas Pipelines		
T [^{0}F]	**T [^{0}C]**	**T**
≤ 250	≤ 120	1.00
300	150	0.97
350	175	0.93
400	200	0.91
450	230	0.87
Abbreviation	**Joining Method**	**E**
SMLS	Seamless	1.0
ERW	Electric Resistance Weld	1.0
EFW	Electric Flash Weld	1.0
SAW	Submerged Arc Weld	1.0
BW	Furnace Butt Weld	0.6
EFAW	Electric Fusion Arc Weld	0.8

The pipeline specification requirement is API 5L plain end line pipe specifications ranging from 6" ND to 80" ND. The product pipeline specification (PSL) with its respective Specified Minimum Yield Strength (SYMS) to be used as per API 5L are PSL 1 and PSL 2. The pipeline grades are as follows,

Table 3. Product Specification Level (PSL)

Grade	SMYS	Grade	SMYS
	MPa		MPa
PSL 1 Gr A25	172	PSL 2 Gr B	241
PSL 1 Gr A	207	PSL 2 X42	290
PSL 1 Gr B	241	PSL 2 X46	317
PSL 1 X42	290	PSL 2 X52	359
PSL 1 X46	317	PSL 2 X56	386
PSL 1 X52	359	PSL 2 X60	414
PSL 1 X56	386	PSL 2 X65	448
PSL 1 X60	414	PSL 2 X70	483
PSL 1 X65	448	PSL 2 X80	552
PSL 1 X70	483	-	-

In the current module, the API 5L pipeline grades chosen for both desert location [Class 1, F = 0.72] & city location [Class 3, F = 0.50] is PSL 1 X65 [SMYS = 448 Mpa]. The pipeline joining method chosen is Electric Resistance Weld [ERW] with a longitudinal joining factor [E] of 1.0. The temperature de-rating factor [T] of the pipelines before & after the booster station for a design temperature of 200^0C as per Table 2 is 0.91. The design capacity of the pipeline is taken as 100 MMscfd. For the module, the pipelines before & after the booster station is laid below ground that has a constant soil temperature of 30^0C & soil overall heat transfer coefficient [U] of 35 W/m^2.K. The pipeline corrosion allowance before & after the booster station is taken as 3 mm considering a corrosion rate of 5 mils/year [1 mil = 1/1000th of an inch] over a 25 year operating period.

The pipeline would include fittings and elevational differences which offer a pressure drop & hence can be expressed as an equivalent length. For this module, the fittings & elevation losses are taken as 2% of the total pipeline length. The booster compressor station is placed at a distance of 20km from the gas plant located in the desert area & the downstream pipeline travels another 30 km to reach the power station in the city. This is shown as follows,

Table 4. Pipeline Lengths

Pipeline	Length	Eq. Length		Total Eq. Length
-	[m]	[%]	[m]	[m]
Upstream	20,000	2	400	20,400
Downstream	30,000	2	600	30,600

It is assumed that the condensation in the pipeline is minimal and hence the pipeline efficiency [E_p] is taken as 0.92. To evaluate the maximum hydrostatic test pressure, the difference in elevation of the pipeline between the pipeline high point elevation and elevation at test point for upstream and down stream pipelines is taken as 100m and 70m & 100m and 90m respectively.

The pipeline booster station consists of a centrifugal compressor operating with polytropic efficiency [η_p] of 80% at 80 MMscfd. The minimum pressure required at the compressor inlet is 12 bara & a minimum pressure of 16 bara at the city power station.

Design Methodology

To perform a process design of the pipeline & booster compressor station, the design methodology consists of 3 parts – Process design of upstream pipeline from the gas plant from the desert, process design of the gas compressor at the booster station and downstream pipeline to the city power station. The process design steps for the upstream/downstream pipelines are,

1. Estimation of Pipeline wall thickness based on design pressure [DP], design temperature [DT], design flowrate [Q_d], location class, design factor [F], pipeline specification [API 5L], specified minimum yield strength [SMYS], derating factor [T].
2. Estimation of Mixture Pseudocritical Properties – Pseudo critical Pressure [T_c], Pseudo critical Temperature [T_c], Reduced Pressure [P_{pr}], Reduced Temperature [T_{pr}], Reduced Density [ρ_r], Deviation Parameter [ε], Modified Reduced Pressure [P'_{pc}], Modified Reduced Temperature [T'_{pc}], Specific Heat Capacity [C_p], Gas Compressibility Factor [Z].
3. Estimation of gas mixture density [ρ], mixture molecular weight [MW], mass flow [m] and actual volumetric flow rate [Q].
4. Estimation of upstream/downstream pipeline Fluid Velocity [V], pipeline exit temperature [P_e] based on soil/ambient temperature, overall heat transfer co-efficient [U], pipeline pressure drop [ΔP], Pipeline Exit Pressure & pressure drop per km [ΔP/L].
5. Estimation of Maximum Allowable Operating Pressure [MAOP], Test Pressure at 110% of SMYS, Maximum Hydrostatic Test Pressure & Leak Test Pressure.

For the Booster Compressor Station, the design steps are,

1. Estimation of Mixture Critical Properties – Pseudocritical Pressure [T_c], Pseudocritical Temperature [T_c], Reduced Pressure [P_{pr}], Reduced Temperature [T_{pr}], Reduced Density [ρ_r], Deviation Parameter [ε], Modified Reduced Pressure [P'_{pc}], Modified Reduced Temperature [T'_{pc}], Specific Heat Capacity [C_p], Gas Compressibility [Z].
2. Estimation of Adiabatic Exponent [k], compressor inlet gas mixture density [ρ], average polytropic exponent [n] based on polytropic efficiency [η_p], adiabatic efficiency [η_a], molar density [[ρ_m], molar volume [V_m], mass flow [m], Molar flow [M], Polytropic Head [H_p] & Absorbed Power [P].
3. Check if compressor energy balance condition, $P_1V_1^n-P_2V_2^n = 0$ is satisfied.
4. Repeat above steps for compressor discharge side for process parameters.

Property Estimation Methodology

To assess the properties of natural gas, calculations can be begun by estimating the properties using Kay's Mixing Rule as follows,

Mixture molecular weight [MW], kg/kmol

$$MW = \sum y_i \, MW_i \tag{1}$$

Mixture Pseudo Critical Pressure [P_c], psia

$$P_c = \sum y_i \, P_{c,i} \tag{2}$$

Mixture Pseudo Critical Temperature [T_c], ^{0}R

$$T_c = \sum y_i \, T_{c,i} \tag{3}$$

Gas Specific Gravity [γ_g], [-]

$$\gamma_g = \frac{MW_g}{MW_{air}}; \; MW_{air} = 28.96 \text{ kg/kmol} \tag{4}$$

From the above, Kay's Mixing Rule does not give accurate pseudocritical properties for higher molecular weight mixtures (particularly C_{7+} mixtures) of hydrocarbon gases when estimating gas compressibility factors [Z] and deviations can be as high as 15%. Therefore, to account for these differences, Sutton's correlations based on gas specific gravity can be utilized as follows,

$$P_{pc} = 756.8 - 131.07\gamma_g - 3.6\gamma_g^2 \tag{5}$$

$$T_{pc} = 169.2 + 349.5\gamma_g - 74.0\gamma_g^2 \tag{6}$$

The above equations are valid for the gas specific gravities range of $0.57 < \gamma_g < 1.68$. Using the Sutton correlations, the reduced properties are calculated as,

$$P_r = \frac{P}{P_{pc}} \tag{7}$$

$$T_r = \frac{T}{T_{pc}} \tag{8}$$

However the pseudocritical properties are not the actual mixture critical temperature and pressure but represent the values that must be used for the purpose of comparing corresponding states of different gases on the compressibility factor, Z-chart/Gas Deviation Factor, as shown below in the Standing & Katz, 1959 chart for natural gases.

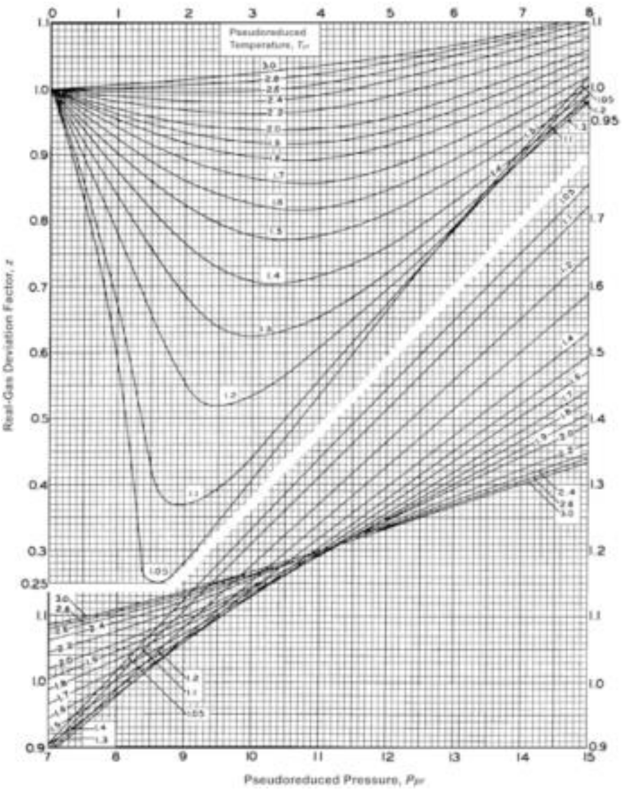


Figure 1. Natural Gas deviation factor chart (Standing & Katz, 1959)

Due to the graphical method of Standing & Katz chart, the Z factor can be estimated using Dranchuk and Abou-Kassem Equation of State [DAK-EoS] which is based on the data of Standing & Katz, 1959 and is expressed as,

$$Z = 1 + \left[A_1 + \frac{A_2}{T_r} + \frac{A_3}{T_r^3} + \frac{A_4}{T_r^4} + \frac{A_5}{T_r^5}\right]\rho_r + \left[A_6 + \frac{A_7}{T_r} + \frac{A_8}{T_r^2}\right]\rho_r^2 - A_9\left[\frac{A_7}{T_r} + \frac{A_8}{T_r^2}\right]\rho_r^5 + +A_{10}(1 + A_{11}\rho_r^2)\left(\frac{\rho_r^2}{T_r^3}\right)e^{-A_{11}\rho_r^2} \qquad (9)$$

Where,

$$\rho_r = \frac{0.27P_r}{ZT_r} \qquad (10)$$

ρ_r = Pseudo-Reduced Density [-]

T_r = Pseudo-Reduced Temperature [-]

And the constants A_1 to A_{11}, are as follows,

Table 5. DAK EoS A_1 to A_{11} Constants

A_1	0.3265	A_7	−0.7361
A_2	−1.0700	A_8	0.1844
A_3	−0.5339	A_9	0.1056
A_4	0.01569	A_{10}	0.6134
A_5	−0.05165	A_{11}	0.7210
A_6	0.5475		

DAK-EoS has an average absolute error of 0.486% in its equation, with a standard deviation of 0.00747 over ranges of pseudo-reduced pressure and temperature of $0.2 < P_{pr} < 30$; $1.0 < T_{pr} < 3.0$ and for $P_{pr} < 1.0$ with $0.7 < T_{pr} < 1.0$. However DAK-EoS gives unacceptable results near the critical temperature for $T_{pr} = 1.0$ and $P_{pr} > 1.0$, and DAK EoS is not recommended in this range.

DAK EoS for NG Mixtures with Acid Gases

Natural Gas is expected to contain acid gas fractions, such as CO_2 and H_2S, & applying the Standing & Katz Z-factor chart & Sutton's pseudocritical properties calculation methods would yield inaccuracies, since they are only valid for hydrocarbon mixtures. To account for these inaccuracies, the Wichert & Aziz correlations can be applied to mixtures containing CO_2 < 54.4 mol% & H_2S < 73.8 mol% by estimating a deviation parameter [ε], which is used to modify the pseudocritical pressure & temperatures. The deviation parameter [ε] whose units are in ^{0}R, is,

$$\varepsilon = 120[A^{0.9} - A^{1.6}] + 15[B^{0.5} - B^4] \qquad (11)$$

Where,

$A = Y_{CO2} + Y_{H2S}$ in Gas mix [Y = mole fraction]

$B = Y_{H2S}$ in Gas mixture [Y = mole fraction]

Applying [ε], the modified pseudocritical pressure & temperature is,

$$T'_{pc} = T_{pc} - \varepsilon \qquad (12)$$

$$P'_{pc} = \frac{P_{pc}\,T'_{pc}}{T_{pc} - B[1-B]\varepsilon} \qquad (13)$$

Where, T'_{pc} & P'_{pc} are valid only in ^{0}R and psia.

Based on the calculated modified pseudocritical pressure [P'_{pc}] and temperature [T'_{pc}], the pseudo-reduced pressure [P_r] & temperature [T_r] is,

$$P_{pr} = \frac{P\,[psia]}{P'_{pc}[psia]} \quad (14)$$

$$T_{pr} = \frac{T\,[^\circ R]}{T'_{pc}[^\circ R]} \quad (15)$$

$$\rho_{pr} = \frac{0.27P_{pr}}{ZT_{pr}} \quad (16)$$

Using the calculated values of Pp_r Tp_r & ρ_{pr}, compressibility factor, Z is determined by using DAK EoS. Owing to the value of 'Z' being an implicit parameter in calculating ρ_{pr} as well as in DAK-EoS, an iterative approach, whereby Z value is guessed & iteratively solved to satisfy both modified pseudo-reduced density [ρ_{pr}] & DAK EoS. Upon calculating the value of Z_{in} at the pipeline inlet, the actual volumetric flow rate [Q_{in}], Gas density [ρ_{in}], gas mass flow [m] is calculated as follows,

Actual Volumetric Flow Rate, Am^3/h

$$Q_1 = \frac{P_{std}Q_{std}Z_1T_1}{Z_{std}T_{std}P_1} \quad (17)$$

The value of Z_{std} is taken to be close to 1.0.

Gas Density, kg/m^3

$$\rho_1 = \frac{P_1MW}{Z_1RT_1} \quad (18)$$

Gas Mass Flow, kg/h

$$m_g = Q_1 \times \rho_1 \quad (19)$$

Pipeline Process & Mechanical Design

To perform Gas Pipeline design, dimensions are chosen based on the following factors,

Location of the Gas Pipelines

1. **Class 1 location** - A Class 1 location is any 1-mile pipeline section that has 10 or fewer buildings intended for human occupancy including areas such as, wastelands, deserts, rugged mountains, grazing land, farmland, sparse populations.
2. **Class 1, division 1 Location** – A Class 1 location where the design factor, F, of the pipeline is greater than 0.72 but equal to, or less than 0.80 and which has been hydrostatically tested to 1.25 times the maximum operating pressure.
3. **Class 1, division 2 Location** - This is a Class 1 location where the design factor, F, of the pipeline is equal to or less than 0.72, and which has been tested to 1.1 times the maximum operating pressure.
4. **Class 2 Location** - This is any 1-mile section of pipeline that has more than 10 but fewer than 46 buildings intended for human occupancy including fringe areas around cities and towns, industrial areas, and ranch or country estates.
5. **Class 3 Location** - This is any 1-mile section of pipeline that has 46 or more buildings intended for human occupancy except when a Class 4 Location prevails, including suburban housing developments, shopping centres, residential areas, industrial areas & other populated areas not meeting Class 4 Location requirements
6. **Class 4 Location** - This is any 1-mile section of pipeline where multi-storey buildings are prevalent, traffic is heavy or dense, and where there may be numerous other utilities underground. Multi-storey means four or more floors above ground including the first, or ground, floor. The depth of basements or number of basement floors is immaterial.

Line Specification of Gas Pipelines – API 5L

1. PSL1 pipes are available through size 2/5" to 80" whereas the smallest diameter pipe available in PSL2 is 4.5" & the largest diameter is 80". PSL1 pipelines are available in different types of ends, such as Plain end, Threaded end, Bevelled end,

special coupling pipes whereas PSL2 pipelines are available in only Plain End.

2. For PSL2 welded pipes, except continuous welding and laser welding, all other welding methods are acceptable. For electric weld welder frequency for PSL2 pipeline is minimum 100 kHz whereas there is no such limitation on PSL1 pipelines.
3. Heat treatment of electric welds is required for all Grades of PSL2 pipes whereas for PSL1 pipelines, grades above X42 require it.
4. All kinds of welding method are acceptable to manufacture PSL1; however, continuous welding is limited to Grade A25.

Gas Pipeline Wall Thickness Estimation

The B31.8 code is often used as the standard of design for natural gas piping systems in facilities, such as compressor stations, gas treatment facilities, measurement & regulation stations & tank farms. The B31.8 wall-thickness formula is stated as,

$$t = \frac{DP \times OD}{2 \times F \times E \times T \times SMYS} \qquad (20)$$

Where,

t = Minimum design wall thickness [in]

DP = Pipeline Design Pressure [psi]

OD = Pipeline Outer Diameter [in]

SMYS = Specific Minimum Yield Stress [psi]

F = Design Factor [-]

E = Longitudinal Weld Joint Factor [E]

T = Temperature De-rating Factor [-]

A min. corrosion allowance of 1 mm is taken for stainless steel & 3 mm is taken for carbon steel pipelines respectively.

Gas Pipeline Pressure Drop

To evaluate the pressure drop across the gas pipeline, the following assumptions are made,

1. Flow is steady state.
2. No work is performed by the gas.
3. Friction factor [f] is a constant function of pipeline length.

Based on these assumptions, since natural gas pipelines operate at high Reynolds numbers that are well in turbulent flow regime & Moody's friction factor becomes merely a function of relative roughness, the Weymouth equation can be applied. The Weymouth equation is expressed as,

$$Q = 433.49\, E \left[ID^{8/3}\right] \left[\frac{T_b}{P_b}\right] \sqrt{\frac{P_1^2 - e^s P_2^2}{\gamma_g T_f L_e Z}} \qquad (21)$$

$$L_e = \frac{L[e^s - 1]}{s} \qquad (22)$$

$$s = 0.0375\, \frac{\gamma_g \times \Delta H}{T_f \times Z} \qquad (23)$$

Where, T_b = Base Temperature [^{0}R] [519.7 ^{0}R]

P_b = Base pressure [psia] [14.7 psia]

P_1 = Pipeline Inlet Pressure [psia]

P_2 = Pipeline Inlet Pressure [psia]

ID = Pipeline Inner Diameter [in]

γ_g = Gas Specific Gravity [-]

T_f = Gas Flowing Temperature [^{0}R]

L_e = Pipeline Equivalent Length [ft]

s = Static head due to elevation change [ft/^{0}R]

ΔH = Elevation Difference [ft]

E = Pipeline Efficiency [-]

Z = Gas Compressibility Factor [-]

Weymouth Equation is also recommended for shorter lengths of pipeline segments (<32 kms) within production batteries and for branch gathering lines, medium to high pressure (+/–100 psig [6.9 barg] to > 1,000 psig [70 barg]) applications, and a high Reynolds number.

Typically, pipeline efficiency factors [E_p] may vary between 0.6 & 0.92 depending on the pipeline's liquid content. As the amount of gas phase liquid content increases, the pipeline efficiency factor can no longer account for the

2-phase flow behaviour and 2-phase flow equations must be used.

Pipeline Exit Temperature

For long gas transmission pipelines with moderate pressure drop, the temperature expansion due to pressure drop is considered to be small and hence the pipeline exit temperature [T_e] can be calculated as,

$$T_e = T_s + \left[T_1 - T_{s/a}\right]e^{-\frac{\pi\times OD\times U\times L}{m_g\times C_p}} \quad (24)$$

Where, T_e = Pipeline Exit Temperature [K]

$T_{s/a}$ = Soil/Ambient Temperature [K]

T_1 = Pipeline Inlet Temperature [K]

U = Overall HTC [W/m^2.K]

C_p = Mass Specific Heat [J/kg.K]

m_g = Mass Flow rate of Gas [kg/s]

OD = Pipeline Outer Diameter [m]

The ideal mass specific heat [C_p], kJ/kg.K, of natural gas can be computed as,

$$C_p = [(-10.9602\gamma_g + 25.9033) + (0.21517\gamma_g - 0.068687)T + (-0.00013337\gamma_g + 0.000086387)T^2 + (0.000000031474\gamma_g - 0.000000028396)T^3)]/MW \quad (25)$$

Max Hydro Test & Leak Pressure Test

The maximum allowable operating pressure [MAOP] is taken as 90% of the design pressure & for an 8 hour minimum test pressure, the hydro test pressure is based on the location class and maximum test pressure becomes the lower value of 8 hour minimum test pressure & test pressure at low point. The leak test pressure is taken as 80% of the design pressure.

Pipeline Exit Pressure

The pipeline exit pressure can be computed by re-arranging Weymouth's equation as,

$$P_2 = \sqrt{\frac{P_1^2 - \gamma_g T_f L_e Z\left[\frac{Q}{433.49\, E\left[ID^{8/3}\right]\left[\frac{T_b}{P_b}\right]}\right]^2}{e^S}} \quad (26)$$

Considering the static head contribution of any condensation of liquids in the pipeline, it is accounted for as an equivalent length of pressure drop. Hence ΔH = 0 & L = L_e. Therefore the pipeline exit pressure [$P_2 = P_e$] using Weymouth equation is calculated as,

$$P_e = \sqrt{P_1^2 - \gamma_g T_f L_e Z\left[\frac{Q}{433.49\, E\left[ID^{8/3}\right]\left[\frac{T_b}{P_b}\right]}\right]^2} \quad (27)$$

Since the gas flow through the pipeline is compressible, the compressibility factor is expected to vary along with gas velocity, gas pressure & gas temperature due to heat losses. Though due to temperature expansion, exit temperature is expected to change; its contribution is considered to be small. The compressibility factor [Z] would be used to compute the gas exit pressure [P_e] based on pipeline flowing gas temperature [$T_{f,a}$]. The value of Z_a is an implicit parameter & hence has to be solved for iteratively as follows,

Iteration 1: Guess Initial value of $Z_{a,1}$, pipeline exit pressure $P_{e,1}$, and calculate

$$T_{f,a} = T_{s/a} + \left[\frac{T_1 - T_e}{ln\left(\frac{T_1 - T_{s/a}}{T_e - T_{s/a}}\right)}\right] \quad (30)$$

$$T_{pr} = \frac{T_{f,a}\,[°R]}{T'_{pc}[°R]} \quad (31)$$

$$P_{pr,1} = \frac{P_{e,1}\,[psia]}{P'_{pc}[psia]} \quad (32)$$

$$P_{e,1} = \sqrt{P_1^2 - \gamma_g T_{f,a} L_e Z_{a,1}\left[\frac{Q}{433.49\, E\left[ID^{8/3}\right]\left[\frac{T_b}{P_b}\right]}\right]^2} \quad (33)$$

$$\rho_{pr,1} = \frac{0.27\times P_{pr,1}}{Z_{a,1}\times T_{pr}} \quad (34)$$

Iteration 2: Assigning $P_{e,1}$ = $P_{e,2}$,

$$P_{pr,2} = \frac{P_{e,2}\,[psia]}{P'_{pc}[psia]} \quad (35)$$

$$Z_{a,2} = 1 + \left[A_1 + \frac{A_2}{T_{pr}} + \frac{A_3}{T_{pr}^3} + \frac{A_4}{T_{pr}^4} + \frac{A_5}{T_{pr}^5}\right]\rho_{pr,1} + \left[A_6 + \frac{A_7}{T_{pr}} + \frac{A_8}{T_{pr}^2}\right]\rho_{pr,1}^2 - A_9\left[\frac{A_7}{T_{pr}} + \frac{A_8}{T_{pr}^2}\right]\rho_{pr,1}^5 + A_{10}\left(1 + A_{11}\rho_{pr,1}^2\right)\left(\frac{\rho_{r,1}^2}{T_{pr}^3}\right)e^{-A_{11}\rho_{pr,1}^2} \quad (36)$$

$$P_{e,2} = \sqrt{P_1^2 - \gamma_g T_{f,a} L_e Z_{a,2}\left[\frac{Q}{433.49\, E\left[ID^{8/3}\right]\left[\frac{T_b}{P_b}\right]}\right]^2} \quad (37)$$

$$\rho_{pr,2} = \frac{0.27 \times P_{pr,2}}{Z_{a,2} \times T_{pr}} \quad (38)$$

Iteration 3: Assigning $P_{e,2} = P_{e,3}$,

$$P_{pr,3} = \frac{P_{e,3}\,[psia]}{P'_{pc}[psia]} \quad (39)$$

$$Z_{a,3} = 1 + \left[A_1 + \frac{A_2}{T_{pr}} + \frac{A_3}{T_{pr}^3} + \frac{A_4}{T_{pr}^4} + \frac{A_5}{T_{pr}^5}\right]\rho_{pr,2} + \left[A_6 + \frac{A_7}{T_{pr}} + \frac{A_8}{T_{pr}^2}\right]\rho_{pr,2}^2 - A_9\left[\frac{A_7}{T_{pr}} + \frac{A_8}{T_{pr}^2}\right]\rho_{pr,2}^5 + A_{10}\left(1 + A_{11}\rho_{pr,2}^2\right)\left(\frac{\rho_{pr,2}^2}{T_{pr}^3}\right)e^{-A_{11}\rho_{pr,2}^2} \quad (40)$$

$$P_{e,3} = \sqrt{P_1^2 - \gamma_g T_{f,a} L_e Z_{a,3}\left[\frac{Q}{433.49\, E\left[ID^{8/3}\right]\left[\frac{T_b}{P_b}\right]}\right]^2} \quad (41)$$

$$\rho_{pr,3} = \frac{0.27 \times P_{pr,3}}{Z_{a,3} \times T_{pr}} \quad (42)$$

The above set of iterations is to be continued until the convergence criteria of $P_{e,n} - P_{e,n-1} <$ *1e-6 (say)* is met.

Gas Pipeline Inlet/Outlet Velocity

The pipeline inlet velocity is computed as,

$$V_g = \frac{60 \times Q_g \times T \times Z}{OD^2 \times P} \quad (43)$$

Where, V_g = Gas Velocity [ft/s]

Q_g = Gas Flow rate [MMscfd]

T = Inlet/Outlet Temperature [^{0}R]

P = Inlet/Outlet Pressure [psia]

Z = Compressibility Factor [Z]

OD = Pipeline Outer Diameter [in]

The velocity in gas lines should be less than 60 to 80 ft/s [18m/s to 25m/s] to minimize noise and allow for corrosion inhibition. A lower velocity of 50 ft/s [15 m/s] should be used in the presence of known corrosives such as CO_2. The minimum gas velocity should be between 10 and 15 ft/s [3 m/s to 4.5 m/s], which minimizes liquid fallout.

Pressure Drop & Pressure Drop/km

The total pressure drop across the Gas pipeline is computed as,

$$\Delta P = P_1 - P_{e,n} \quad (44)$$

The pressure drop/km is computed as,

$$\frac{\Delta P}{L_e} = \frac{P_1 - P_{e,n}}{L_e} \quad (45)$$

The optimum pressure drop for natural gas pipelines can be taken to be between 3.5 psi/mile & 5.83 psi/mile (0.15 - 0.25 bar/km)

Booster Compressor Process Design

Natural Gas Properties Estimation

The properties of natural gas at the booster station inlet can be computed based on the upstream pipeline exit process conditions which become the compressor inlet input.

1. As the gas enters into the compressor station, a loss of pressure until the compressor suction flange is expected. This is assumed to be 1 bar for the module. The temperature drop up till the compressor suction flange is assumed to be negligible and hence would be nearly the same as the upstream pipeline outlet temperature.
2. The gas upon being compressed, a pressure drop is expected to exist between the compressor discharge flange & the downstream pipeline inlet. For this module, a drop in pressure of 1 bar is taken across the discharge cooler & downstream piping. To ensure the discharge piping design temperature of 200^{0}C is not exceeded even during gas recycle across the anti-surge valve, an air cooler is installed to cool the compressed gas to 50^{0}C. The temperature drop between the air cooler discharge & downstream pipeline inlet is taken as 1^{0}C.

3. With the gas compressor inlet pressure temperature, mass flow rate, actual volumetric flow rate, gas density, the compressor suction & discharge gas properties can be calculated.
4. To estimate the discharge conditions, the compressor discharge pressure [P_2] & downstream pipeline size is chosen such that, sufficient discharge pressure exists to deliver natural gas through the downstream pipeline while respecting the pipeline velocity, pressure drop criteria and meets the city power station's required pressure criteria.
5. With these compressor inlet process conditions, the DAK EoS with Wichert & Aziz correction is used as shown in the previous section to calculate, compressor inlet's Modified properties of Reduced Pressure [P_{pr}], Reduced Temperature [T_{pr}], Reduced density [ρ_{pr}], Specific heat [C_p] & gas compressibility factor [Z].

Booster Compressor Process Conditions

To evaluate the booster compressor's process conditions, the average adiabatic exponent [k_a] & polytropic exponent [n] is required to be calculated for the compressor discharge.

This can be performed iteratively based on an initial guess of the compressor discharge temperature [T_2], discharge side specific heat [$C_{p,2}$], with which discharge side adiabatic exponent [k_2] can be calculated as follows,

Iteration n=1: Guess Initial value of $Z_{2,n}$, compressor discharge temperature, $T_{2,n}$,

$$k_1 = \frac{C_{p,1}}{C_{p,1}-R} \tag{46}$$

$$k_{2,n} = \frac{C_{p,2,n}}{C_{p2,n}-R} \tag{47}$$

The average adiabatic exponent [k_a] becomes

$$k_{a,n} = \frac{k_1+k_{2,n}}{2} \tag{48}$$

The polytropic exponent [n] for iteration n=1 [n_n] is calculated as,

$$n_n = \frac{1}{\left[1-\left[1-\frac{1}{k_{a,n}}\right]\eta_p\right]} \tag{52}$$

With a selected compressor discharge pressure [P_2], the modified properties of reduced temperature [$T_{pr,n}$], reduced pressure [$P_{pr,n}$] & reduced density [$\rho_{pr,n}$] is,

$$P_{pr} = \frac{P_2}{P'_{pc}} \tag{49}$$

$$T_{pr,n} = \frac{T_{2,n}}{T'_{pc}} \tag{50}$$

$$\rho_{pr,n} = \frac{0.27\times P_{pr}}{Z_{2,n}\times T_{pr}} \tag{51}$$

The discharge temperature [$T_{2,n}$] is,

$$T_{2,n} = T_1\left[\frac{P_2}{P_1}\right]^{\frac{n_n-1}{n_n}}\left[\frac{Z_1}{Z_{2,n}}\right] \tag{52}$$

Iteration n=2: Guess Initial value of $Z_{2,n+1}$ = $Z_{2,n}$,

$$k_{2,n+1} = \frac{C_{p,2,n+1}}{C_{p,2,n+1}-R} \tag{53}$$

The average adiabatic exponent becomes

$$k_{a,n+1} = \frac{k_1+k_{2,n+1}}{2} \tag{54}$$

The polytropic exponent [n] for iteration n=2 [n_{n+1}] is calculated as,

$$n_{n+1} = \frac{1}{\left[1-\left[1-\frac{1}{k_{a,n+1}}\right]\eta_p\right]} \tag{55}$$

The reduced temperature [$T_{pr,n}$] becomes,

$$T_{pr,n+1} = \frac{T_{2,n}}{T'_{pc}} \tag{56}$$

The discharge side gas compressibility factor [$Z_{2,n+1}$] can be calculated as,

$$Z_{2,n+1} = 1 + \left[A_1 + \frac{A_2}{T_{pr,n+1}} + \frac{A_3}{T^3_{pr,n+1}} + \frac{A_4}{T^4_{pr,n+1}} + \frac{A_5}{T^5_{pr,n+1}}\right]\rho_{pr,n} + \left[A_6 + \frac{A_7}{T_{pr,n+1}} + \frac{A_8}{T^2_{pr,n+1}}\right]\rho^2_{pr,n} - A_9\left[\frac{A_7}{T_{pr,n+1}} + \frac{A_8}{T^2_{pr,n+1}}\right]\rho^5_{pr,n} + A_{10}\left(1 + A_{11}\rho^2_{pr,n}\right)\left(\frac{\rho^2_{pr,n}}{T^3_{pr,n+1}}\right)e^{-A_{11}\rho^2_{pr,n}} \tag{57}$$

$$\rho_{pr,n+1} = \frac{0.27\times P_{pr}}{Z_{2,n+1}\times T_{pr,n+1}} \tag{58}$$

The discharge temperature [$T_{2,n+1}$] is,

$$T_{2,n+1} = T_1 \left[\frac{P_2}{P_1}\right]^{\frac{n_{n+1}-1}{n_{n+1}}} \left[\frac{Z_1}{Z_{2,n+1}}\right] \quad (59)$$

The above set of iterations are to be continued until the convergence criteria of $T_{2,n+1} - T_{2,n} < 1e\text{-}6$ *(say)* is met.

The gas density [kg/m^3] is calculated as,

$$\rho = \frac{P \times MW}{Z \times R \times T} \quad (60)$$

Where,

P = Inlet/Outlet Pressure [bara]

MW = Molecular Weight [kg/kmol]

Z = Inlet/Outlet Compressibility Factor [-]

R = 0.0831447 m^3.bar/kmol.K

T = Inlet/Outlet Temperature [K]

The molar density [ρ_m] is calculated as,

$$\rho_m = \frac{\rho}{MW} \quad (61)$$

The molar volume [V_m], m^3/kg is,

$$V_m = \frac{1}{\rho_m} \quad (62)$$

With calculated molar volume, the following balance based on polytropic exponent [n] & molar volume [V_m] is checked as,

$$P_1 V_{m,1}^n - P_2 V_{m,2}^n = 0 \quad (63)$$

The adiabatic efficiency [η_a] is calculated as,

$$\eta_a = \frac{\frac{n}{n-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]}{\frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1\right]} \times \frac{1}{\eta_p} \quad (64)$$

The polytropic head H_p [kJ/kg] is,

$$H_p = \frac{Z_{avg} R T}{MW}\left[\frac{n}{n-1}\right]\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad (65)$$

Where, H_p = Polytropic Head [kJ/kg]

Z_a = [Z_1+Z_2]/2; Avg. compressibility factor [-]

R = 8.31447 kJ/kg.K

T = Gas Inlet Temperature [K]

MW = Molecular Weight [kg/kmol]

n = Polytropic exponent [-]

P_1 = Compressor Suction Pressure [bara]

P_2 = Compressor Discharge Pressure [bara]

The absorbed power, P [kW] is calculated as,

$$P = \frac{H_p \times m}{\eta_p} \quad (66)$$

Where,

m = Mass Flow rate [kg/s]

η_p = Polytropic efficiency [-]

Results

With the above procedures executed, the pseudocritical properties of natural gas are,

Table 6. Sutton Correlation & Wichert & Aziz Correction

Parameter	Value
Gas Molecular Weight [MW] [kg/kmol]	21.20
Gas Specific gravity [γ_g] [-]	0.7320
Pseudocritical Pressure [P_{pc}] [psia]	658.93
Pseudocritical Temperature [T_{pc}] [^{0}R]	385.39
Deviation parameter [ε] [^{0}R]	4.9110
Modified Pseudo Critical Pressure [T'_{pc}] [^{0}R]	380.48
Modified Pseudo Critical Temperature [P'_{pc}] [psia]	650.53

The upstream & downstream pipeline mechanical design data is estimated as follows,

Table 7. Pipeline Mechanical Design Results

Parameter	Upstream Pipeline	Downstream Pipeline
Gas Pipeline Location	Deserted	City
Location Class	Class 1, Div 2	Class 3
Design Factor [F] [-]	0.72	0.50
Pipeline Joining Method	ERW	ERW
Longitudinal Joint Factor [E] [-]	1.0	1.0
API 5L Spec	PSL1 X65	PSL1 X65
SMYS [S] [Mpa]	448.0	448.0
Design Pressure [DP] [bara]	40	40

Parameter	Upstream Pipeline	Downstream Pipeline
Design Temperature [DT] [^{0}C]	200	200
De-Rating Factor [T] [-]	0.91	0.91
$T_{soil}/T_{ambient}$ [Above Ground/Buried] [^{0}C]	30	30
Overall HTC [U] [W/m^2.K]	35	35
Chosen Pipeline Diameter [in]	14.00	14.00
Calculated Wall Thickness [t] [mm]	2.42	3.49
Corrosion Allowance [mm]	3	3
Total Wall Thickness [mm]	5.42	6.49
Selected Wall Thickness [t][mm]	5.56	7.14
Pipeline Inside Diameter [ID] [mm]	344.47	341.33
Pipeline Length incl. Fittings [L_e] [m]	20,400	30,600

The upstream & downstream pipeline process design results are,

Table 8. Pipeline Process Design Details

Parameter	Upstream Pipeline	Downstream Pipeline
Op. Gas Flow Rate [Q] [MMscfd]	80.0	80.0
Op. Inlet Pressure [P] [bara]	36.0	39.0
Op. Inlet Temperature [T] [^{0}C]	48.0	49.0
Pipeline Inlet C_p [kJ/kg.K]	2.1258	2.1295
Gas Compressibility Factor [Z]		
Reduced Pressure [P_{pr}] [-]	0.8135	0.8813
Reduced Temperature [T_{pr}] [-]	1.5193	1.5241
Reduced Density [ρ_r] [-]	0.1564	0.1699
Gas Compressibility [Z] [-]	0.9244	0.9192
Gas Flow [Q] [Am^3/h]	2,731	2,515
Gas Density [ρ] [kg/m^3]	30.92	33.58
Mass Flow [m] [kg/h]	84,458	84,458
Pipeline Pressure Drop - Weymouth Method		
Pipeline Inlet Velocity [V] [m/s]	8.0	7.4
Pipeline Exit Velocity [V_e] [m/s]	10.9	9.4
Pipeline Exit Temperature [T_e] [^{0}C]	30.00	30.00
Pipeline Efficiency [E_p] [-]	0.92	0.92
Pipeline Exit Pressure [P_e] [bara]	25.47	29.14
Pressure Drop [ΔP] [bar]	10.53	9.86
Pressure Drop per km [ΔP/L] [bar/km]	0.52	0.32

The booster compressor design results are,

Table 9. Booster Compressor Design Details

Parameter	Suction	Discharge
Pressure [P] [bara]	24.5	40.0
Temperature [T] [^{0}C]	30.0	67.3
Gas Compressibility Factor [Z] [DAK EoS Method]		
Reduced Pressure [P_{pr}] [-]	0.5530	0.9039
Reduced Temperature [T_{pr}] [-]	1.4342	1.6136
Reduced Density [ρ_r] [-]	0.1111	0.1622
Compressibility Factor [Z]	0.9368	0.9325
Mass Specific Heat [C_p]	2.0576	2.2006
Adiabatic Exponent [k]	1.235	1.217
Gas Density [ρ]	21.97	32.07
Molar Density [ρ_m]	1.036	1.513
Molar Volume [V_m]	0.046	0.031
Actual Vol Flow [Q]	3,844	2,634
Mass Flow [m]	84,458	84,458
Molar Flow [M]	3,984	3,984
Natural Gas Compressor Parameters		
Polytropic Efficiency [η_p] [%]	80.00	
Polytropic Exponent [n] [-]	1.2997	
Is Condition: $P_1V_1^n - P_2V_2^n = 0$ Satisfied	0.0000	
Average Polytropic Exponent [k] [-]	1.2262	
Adiabatic Efficiency [η_a] [%]	79.08	
Polytropic Head [H_p] [kJ/kg]	57.80	
Absorbed Power [P] [kW]	1,695	

The Max Hydrostatic & Leak Test Pressure is,

Table 10. Hydrostatic & Leak Test Pressure

Parameter	Upstream Pipeline	Downstream Pipeline
Pipeline High Point Elevation [m]	100	100
Elevation at Test Point [m]	70	90
Static Pressure [P_{static}] [bar]	2.94	0.98
Max Allow. Op. Pressure [MAOP]	36.0	36.0
Test Pressure at 110% of SMYS [bara]	44.0	44.0
8-Hour Minimum Test Pressure [bara]	42.5	40.6
Max Hydrostatic Test Pressure [bara]	42.5	40.6
Leak Test Pressure [bara]	32.0	32.0

For the iterative steps made for compressor discharge temperature [T_2], Upstream & Downstream Pipeline Exit Pressure [P_e] calculations, the error percentage in the iterative steps is plotted below.

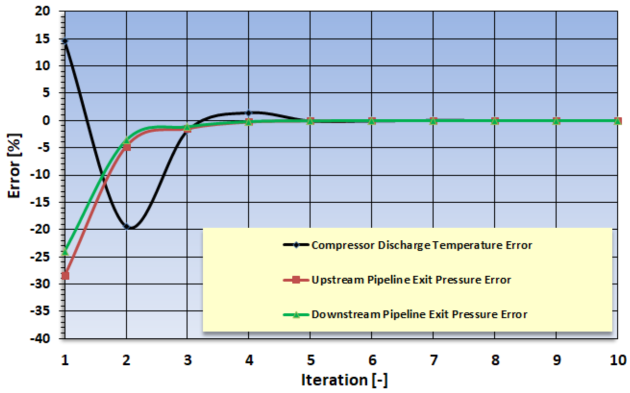


Figure 2. Error Percentage in Iterative Steps

Appendix A shows a summary of the MS-Excel Steps & Appendix B shows the iterations made to calculate the compressor discharge temperature [T_2], Upstream & Downstream Pipeline Exit Pressures [P_e].

References & Further Reading

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Appendix A

Upstream Pipeline Mechanical Design Data

Parameter	Value	Unit
Location of Gas Pipeline	Deserted	[-]
Location Class	Class 1, Div 2	[-]
Design Factor [F]	0.72	[-]
Pipeline Joining Method	ERW	[-]
Longitudinal Joint Factor [E]	1.0	[-]
API 5L Spec	PSL 1 X65	[-]
SMYS [S]	448.0	[MPa]
Design Pressure [DP]	40	[bara]
Design Temperature [DT]	200	[^oC]
De-Rating Factor [T]	0.91	[-]
$T_{soil}/T_{ambient}$ [Above Ground/Buried]	30	[^oC]
Overall HTC [U]	35	[$W/m^2.K$]
Chosen Pipeline Diameter [DN]	14.00	[in]
Calculated Wall Thickness [t]	2.42	[mm]
Corrosion Allowance	3	[mm]
Total Wall Thickness [t]	5.42	[mm]
Selected Wall Thickness	5.56	[mm]
Pipeline Inside Diameter [ID]	344.47	[mm]
Pipeline Length incl. Fittings [L_e]	20,400	[m]

Reference Table for Gas Pipeline Mechanical Design

Design Factors [F] Based on Location of Gas Pipelines		
Class	**Description**	**F**
Class 1, Div 2	Deserted	0.72
Class 2	Village	0.60
Class 3	City	0.50
Class 4	Densely Populated	0.40
Temperature De-rating [T] for Gas Pipelines		
T [^oF]	**T [^oC]**	**T**
≤ 250	≤ 120	1.00
300	150	0.97
350	175	0.93
400	200	0.91
450	230	0.87
Abbreviations	**Joining Method**	**E**
SMLS	Seamless	1.0
ERW	Electric Resistance Weld	1.0
EFW	Electric Flash Weld	1.0
SAW	Submerged Arc Weld	1.0
BW	Furnace Butt Weld	0.6
EFAW	Electric Fusion Arc Weld	0.8

Downstream Pipeline Mechanical Design Data

Parameter	Value	Unit
Location of Gas Pipeline	City	[-]
Location Class	Class 3	[-]
Design Factor [F]	0.50	[-]
Pipeline Joining Method	ERW	[-]
Longitudinal Joint Factor [E]	1.0	[-]
API 5L Spec	PSL 1 X65	[-]
SMYS [S]	448.0	[MPa]
Design Pressure [DP]	40	[bara]
Design Temperature [DT]	200	[^oC]
De-Rating Factor [T]	0.91	[-]
$T_{soil}/T_{ambient}$ [Above Ground/Buried]	30	[^oC]
Overall HTC [U]	35	[$W/m^2.K$]
Chosen Pipeline Diameter [DN]	14.00	[in]
Calculated Wall Thickness [t]	3.49	[mm]
Corrosion Allowance	3	[mm]
Total Wall Thickness [t]	6.49	[mm]
Selected Wall Thickness	7.14	[mm]
Pipeline Inside Diameter [ID]	341.33	[mm]
Pipeline Length incl. Fittings [L_e]	30,600	[m]

Natural Gas Booster Station & Pipeline Process Design Results

Upstream Pipeline to Booster Station

Parameter	Value	Unit
Operating Gas Flow Rate [Q]	80.0	MMSCFD
Operating Inlet Pressure [P_1]	36.0	[bara]
Operating Inlet Temperature [T_1]	48.0	[^oC]
Upstream Pipeline Inlet C_p	2.1258	[kJ/kg.K]
Upstream Pipeline Inlet Gas Compressibility Factor [Z_{in}]		
Reduced Pressure [P_r]	0.8135	[-]
Reduced Temperature [T_{pr}]	1.5193	[-]
Reduced Density [ρ_r]	0.1564	[-]
DAK EOS Convergence	*0.0000*	Calculate Z
Compressibility Factor [Z_{in}]	**0.9244**	**[-]**
Gas Flow [Act_m^3/h]	2,731	[m^3/h]
Gas Density [ρ]	30.92	[kg/m^3]
Mass Flow [m]	84,458	[kg/h]
Upstream Pipeline Pressure Drop - Weymouth Method		
Pipeline Inlet Fluid Velocity [V]	8.0	[m/s]
Pipeline Exit Velocity [V_e]	10.9	[m/s]
Pipeline Exit Temperature [T_e]	**30.00**	**[^oC]**
Pipeline Efficiency [E_p]	0.92	[-]
Pipeline Exit Pressure [P_e]	**25.47**	**[bara]**
Pressure Drop [ΔP]	10.53	[bar]
Pressure Drop per km [ΔP/L]	0.52	[bar/km]
Maximum Hydrostatic Test Pressure & Leak Test Pressure		
Pipeline High Point Elevation	100	[m]
Elevation at Test Point	70	[m]
Static Pressure [P_{static}]	2.94	[bar]
Max Allow. Op. Pressure [MAOP]	36.0	[bara]
Test Pressure at 110% of SMYS	44.0	[bara]
8-Hour Minimum Test Pressure	42.5	[bara]
Max Hydrostatic Test Pressure	**42.5**	**[bara]**
Leak Test Pressure	**32.0**	**[bara]**

Centrifugal Compressor Process Conditions

Parameter	Suction	Discharge	Units
Pressure [P]	*24.5*	*40.0*	**[bara]**
Temperature [T]	*30.0*	*67.9*	**[^oC]**
Compressibility Factor [Z] Estimation [DAK EoS Method]			
Reduced Pressure [P_{pr}]	0.5530	0.9039	[-]
Reduced Temperature [T_{pr}]	1.4342	1.6136	[-]
Reduced Density [ρ_r]	0.1111	0.1622	[-]
DAK EOS Convergence	*0.0000*	Calculate $Z_{1,Compressor}$	
Compressibility Factor [Z]	**0.9368**	**0.9325**	**[-]**
Mass Specific Heat [C_p]	2.0576	2.2006	[kJ/kg.K]
Adiabatic Exponent [k]	1.235	1.217	[-]
Gas Density [ρ]	**21.97**	**32.07**	**[kg/m^3]**
Molar Density [ρ_m]	1.036	1.513	[$kmol/m^3$]
Molar Volume [V_m]	0.046	0.031	[m^3/kg]
Actual Vol Flow [Q]	**3,844**	**2,634**	**[Am^3/h]**
Mass Flow [m]	**84,458**	**84,458**	[kg/h]
Molar Flow [M]	3,984	3,984	[kmol/h]
Natural Gas Compressor Parameters			
Polytropic Efficiency [η_P]		80.00	**[%]**
Polytropic Exponent [n]		1.2997	[-]
Is Condition: $P_1V_1^n - P_2V_2^n = 0$ Satisfied		0.0000	[-]
Average Polytropic Exponent [k]		1.2262	[-]
Adiabatic Efficiency [η_a]		**79.08**	**[%]**
Polytropic Head [H_p]		57.80	[kJ/kg]
Absorbed Power [P]		**1,695**	**[kW]**

Downstream Pipeline to Booster Station

Parameter	Value	Unit
Operating Gas Flow Rate [Q]	**80.0**	**MMSCFD**
Operating Inlet Pressure [P_1]	**39.0**	**[bara]**
Operating Inlet Temperature [T_1]	**49.0**	**[^oC]**
Downstream Pipeline Inlet C_p	2.1295	[kJ/kg.K]
Downstream Pipeline Inlet Gas Compressibility Factor [Z_{out}]		
Reduced Pressure [P_r]	0.8813	[-]
Reduced Temperature [T_{pr}]	1.5241	[-]
Reduced Density [ρ_r]	0.1699	[-]
DAK EOS Convergence	*0.0000*	Calculate Z
Compressibility Factor [Z_{in}]	**0.9192**	**[-]**
Gas Flow [Act_m^3/h]	2,515	[m^3/h]
Gas Density [ρ]	33.58	[kg/m^3]
Mass Flow [m]	84,458	[kg/h]
Downstream Pipeline Pressure Drop - Weymouth Method		
Pipeline Inlet Fluid Velocity [V]	7.4	[m/s]
Pipeline Exit Velocity [V_e]	9.4	[m/s]
Pipeline Exit Temperature [T_e]	**30.00**	**[^oC]**
Pipeline Efficiency [E_p]	0.92	[-]
Pipeline Exit Pressure [P_e]	**29.14**	**[bara]**
Pressure Drop [ΔP]	9.86	[bar]
Pressure Drop per km [ΔP/L]	0.32	[bar/km]
Maximum Hydrostatic Test Pressure & Leak Test Pressure		
Pipeline High Point Elevation	100	[m]
Elevation at Test Point	90	[m]
Static Pressure [P_{static}]	0.98	[bar]
Max Allow. Op. Pressure [MAOP]	36.0	[bara]
Test Pressure at 110% of SMYS	44.0	[bara]
8-Hour Minimum Test Pressure	40.6	[bara]
Max Hydrostatic Test Pressure	**40.6**	**[bara]**
Leak Test Pressure	**32.0**	**[bara]**

Appendix B

Compressor Discharge Side Estimation - Polytropic Conditions												
Iteration	$T_{2,initial}$	T_2	T_2	$C_{p,2}$	k_2	k_a	n	T_{Pr}	Z_2	ρ_{pr}	T_2 [Calc]	
[-]	[^{0}C]	[K]	[^{0}R]	[kJ/kg.K]	[-]	[-]	[-]	[-]	[-]	[-]	[K]	[^{0}C]
1	30	303.2	545.67	2.058	1.235	1.2355	1.3128	1.4342	0.9000	0.1891	354.7	81.6
2	81.6	354.7	638.52	2.252	1.211	1.2232	1.2955	1.6782	0.9303	0.1563	341.5	68.3
3	68.3	341.5	614.63	2.202	1.217	1.2261	1.2995	1.6154	0.9349	0.1616	340.2	67.1
4	67.1	340.2	612.38	2.197	1.217	1.2264	1.2999	1.6095	0.9323	0.1626	341.2	68.0
5	68.0	341.2	614.13	2.201	1.217	1.2261	1.2996	1.6141	0.9324	0.1622	341.1	68.0
6	68.0	341.1	614.01	2.201	1.217	1.2262	1.2996	1.6138	0.9325	0.1622	341.1	67.9
7	67.9	341.1	613.92	2.201	1.217	1.2262	1.2997	1.6136	0.9325	0.1622	341.1	67.9
8	67.9	341.1	613.94	2.201	1.217	1.2262	1.2997	1.6136	0.9325	0.1622	341.1	67.9
9	67.9	341.1	613.94	2.201	1.217	1.2262	1.2997	1.6136	0.9325	0.1622	341.1	67.9
10	67.9	341.1	613.94	2.201	1.217	1.2262	1.2997	1.6136	0.9325	0.1622	341.1	67.9

Upstream Pipeline - Exit Pressure Estimation							Downstream Pipeline - Exit Pressure Estimation					
Iteration	$T_{f,a}$	$P_{e,initial}$	Z_a	ρ_{pr}	P_e [Calc]		$T_{f,a}$	$P_{e,initial}$	Z_a	ρ_{pr}	P_e [Calc]	
[-]	[^{0}R]	[psia]	[-]	[-]	[psia	[bara]	[^{0}R]	[psia]	[-]	[-]	[psia	[bara]
1	547.7	514.5	0.8000	0.1854	400.5	27.2	547.1	558.6	0.8000	0.2015	450.7	30.7
2	547.7	400.5	0.9000	0.1283	381.4	25.9	547.1	450.7	0.8921	0.1458	434.4	29.6
3	547.7	381.4	0.9284	0.1184	375.8	25.6	547.1	434.4	0.9193	0.1364	429.5	29.2
4	547.7	375.8	0.9335	0.1161	374.8	25.5	547.1	429.5	0.9241	0.1342	428.6	29.2
5	547.7	374.8	0.9348	0.1156	374.5	25.5	547.1	428.6	0.9252	0.1337	428.4	29.1
6	547.7	374.5	0.9350	0.1155	374.5	25.5	547.1	428.4	0.9254	0.1336	428.4	29.1
7	547.7	374.5	0.9351	0.1155	374.5	25.5	547.1	428.4	0.9255	0.1336	428.4	29.1
8	547.7	374.5	0.9351	0.1155	374.5	25.5	547.1	428.4	0.9255	0.1336	428.4	29.1
9	547.7	374.5	0.9351	0.1155	374.5	25.5	547.1	428.4	0.9255	0.1336	428.4	29.1
10	**547.7**	**374.5**	**0.9351**	**0.1155**	**374.5**	**25.5**	**547.1**	**428.4**	**0.9255**	**0.1336**	**428.4**	**29.1**

Module 12

Vapour Compression for Propane-Propylene Splitters

Propane-Propylene (C_3) splitters are used in Petrochemical industries to distil propylene from propane prior to export. C_3 Splitter columns are traditionally operated with low pressure steam (LPS) passed to the column's reboiler to heat up the C_3 mixture thereby causing the splitting operation. However the use of Low Pressure steam is energy-demanding. A C_3 splitter is equipped with a cooler and flash tank at the column top that condenses the column top vapours but in doing so, loses precious heat of condensation. In recent years, the concept of using a heat pump, such as a compressor has become a standard practice that eliminates the necessity of using low pressure steam.

C_3 Splitter with Vapour Compression

Vapour compression methods for C_3 stripping operations are ideal for compounds that have low relative volatilities. Below is a schematic of a C_3 splitter with vapour compression of product (propylene). Low pressure steam that was used to vaporize the bottoms propylene product is replaced by installing a column top compressor and routing a portion of the discharge to vaporize the bottoms propane via the reboiler.

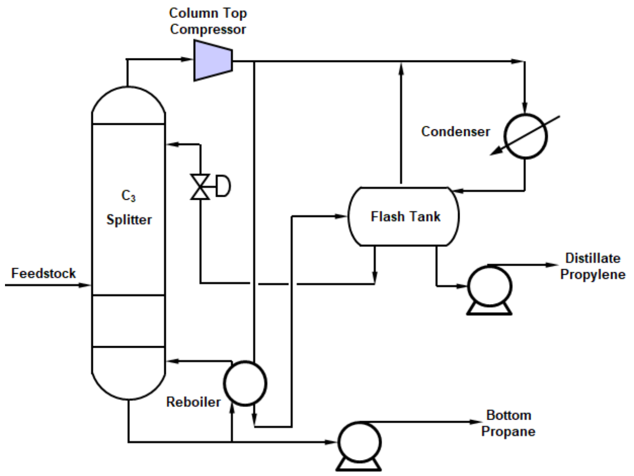

Figure 1. C_3 Splitter with Compression

As per Ref [2], for a C_3 splitter to operate effectively, the temperature difference between the column top and bottom should not be more than 25^0F (13.89^0C). Additionally, the bottom heat of vaporization should be close to overhead vapour's heat of condensation with a pressure drop less than 15 psi (1 bar) across the column internals. The heat of vaporization for propylene and propane are nearly close at 157.6 Btu/lb and 151.7 Btu/lb respectively. The excess energy required to be supplied by the compressor is around 11% - 12% of reboiler duty [Ref 2] which represents a high energy savings. Low pressure splitters also offer the advantage of fewer trays, shorter column height and lower column wall thickness that represents a capital savings with higher relative volatilities to effectuate product separation.

Low pressure C_3 splitter columns operate between 90 psig (6.2 barg) to 110 psig (7.6 barg) depending on the Technology Licensor. For the heat pump, the choice of compressor used can be centrifugal type with a typical pressure ratio of 1.8 [Ref 2] and effectively regulates the column top pressures when the throughput varies. The operation of the centrifugal compressor can be fixed type or variable speed type with the latter representing greater control but with higher installation costs.

C_3 Column Top Pressure Control Methods

Suction Throttling Method

The Suction throttling method involves using a control valve (e.g. butterfly valve) placed at suction side of the centrifugal compressor. But these are suitable only for fixed speed drives like Asynchronous Induction Motors where the driver speed cannot be manipulated. Below is a process schematic of a C_3 column top compressor that works on the principle of suction throttling. A cooler on the anti-surge line cuts down the inline pressure loss in the compressor discharge and also reduces the compressor discharge side equipment and piping volumes, contributing to the fast response of the anti-surge system. The PIC on the compressor discharge receives discharge pressures from discharge side pressure transmitter (PT) to alter the suction throttle valve opening.

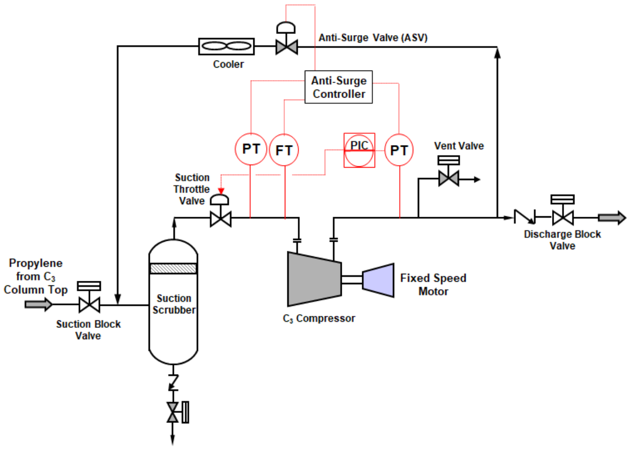

Figure 2. Suction Throttling for C_3 Column Top Compressor

For C_3 column operation, when the operating pressure at the column top increases, the suction throttle valve is altered based on the compressor discharge pressure. The operational advantage would be as follows,

1. For the case where the column top pressure increases, it results in more propylene flow into the compressor. This causes the motor to draw in more amperage to sustain the required higher torque but can trip the electric motor and subsequently the compressor. Therefore, the suction throttle valve in such an event closes accordingly to ensure the discharge pressure required is maintained without tripping the compressor.
2. In the event, the C_3 column pressure falls, and for a fixed speed of the motor and reduced throughput from the column top, the compressor discharge pressure would rise causing the column top flash tank pressure and the bottom reboiler pressure to exceed design limits. Therefore as an abatement measure, the suction throttle valve would open further to maintain the required compressor discharge pressure.
3. For both cases of C_3 column top pressure increasing/decreasing, the anti surge valve (ASV) also acts in tandem with the suction throttle valve via the Anti-surge controller. This ensures that the compressor operating point does not cross the surge control line (SCL).
4. From an energy savings perspective, the pinching operation of the suction throttle valve would make the compressor operate closer to the surge line where the power absorbed is lower.

Variable Speed Motor (VSM) Method

As an alternative, variable speed motor eliminates the need for a suction throttle valve and can cater to the tower top propylene vapours during column fluctuations by altering the motor speed based on the discharge pressure of the compressor. The discharge side pressure controller PIC cascades its output (OP) to assign a set point to the speed controller (SC) and controls the motor speed. In doing so, both output flow and pressure are regulated.

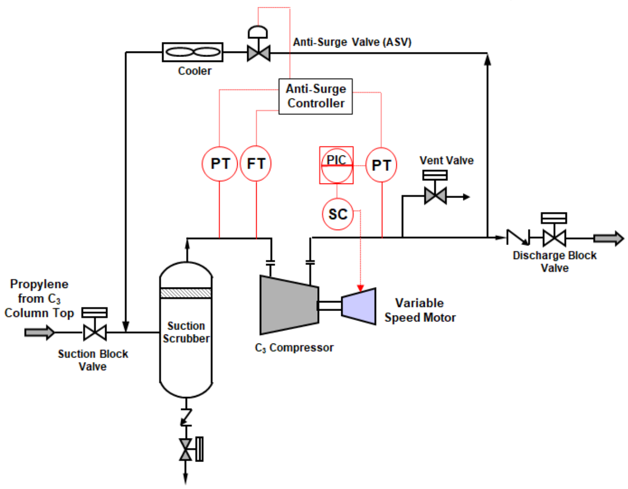

Figure 3. Variable Speed Motor for C_3 Column Top Compressor

In real situations, C_3 splitter columns can also experience fluctuations in operating pressures due to any changes in the upstream side of the C_3 splitter. This would also mean the C_3 splitter column would take a while to again attain equilibrium across trays and the vapour compressor at the tower top would also need to synchronize itself with the column operating pressures. Comparing suction throttling methods over Variable speed methods, the following points are described below.

1. Suction throttling methods though can be used but suffer from an operability range of the throttle valve. This often requires the anti surge valve (ASV) to open frequently to avoid the compressor operating point from crossing the surge control line (SCL). A disadvantage of the suction throttling method is, during a reduction in the column top pressure, when the compressor operates closer to the surge line though saving power, the risk of the operating point crossing the surge line is high. Any further fluctuation in the column top pressure is detrimental in creating a Level 1 emergency equipment shutdown (ESD-1) scenario and subsequent upsetting of the column pressure profile. Additionally due to excessive pressure drop across small throttle valve openings, the risk of Joule-Thompson cooling can produce condensate particles which can erode the compressor's impellers.

2. Variable speed compressors also cater well during turndown conditions, as much as 30%. However this must be considered during the process design basis stage when setting the design and operating limits of the C_3 splitter column. For such low turndown conditions, the compressor vendor must also be consulted prior to equipment selection to understand how much vapour recycling can occur via the anti-surge valve during such low turndown conditions.

3. In vapour compression methods, when the use of steam is eliminated to vaporize the bottoms product, the tower top vapours from the compressor are used to effectuate vaporization of the bottoms product in the reboiler. In the event of the column operating pressure exceeding the stipulated values, variable speed drives reduce the compressor speed and hence discharge pressure, thereby supplying a reduced throughput to the reboiler. This consequently reduces the reboiler temperatures, reboiler vaporization rates, avoids product degradation and aids in bringing the operating pressure of the C_3 splitter columns within the allowable limits.

References & Further Reading

1. '*Design Guidelines for Propylene Splitters Efficiencies'*, Karl Kolmetz, KLM Technology Group
2. '*Be Smart About Column Design*', Mark Pilling, P.E, Daniel R. Summers, P.E, Sulzer Chemtech, USA, AIChE November 2012

Module 13

BOIL OFF GAS ANALYSIS OF LNG AT RECEIVING TERMINALS

Liquefied Natural Gas (LNG) is a cryogenic mixture of low molecular weight (MW) hydrocarbons with its chief component being methane. Its uses cover a gamut of applications from domestic & industrial use, power generation, to transportation fuel in its liquid form. LNG is transported in double-hulled ships specifically designed to handle low temperatures of the order of -162^{0}C. As of 2012, there were 360 ships transporting more than 220 million metric tons of LNG every year. [1]

When LNG is received at most terminals, it is transferred to insulated storage tanks that are built to specifically hold LNG. These tanks can be above or below ground & keep the liquid at a low temperature to minimize evaporation & compositional changes due to heat ingress from the ambient. The temperature within the tank will remain constant if the pressure is kept constant by allowing the boil off gas (BOG) to escape from the tank. This is known as auto-refrigeration. BOG is collected & used as a fuel source in the facility or on the tanker transporting it. When natural gas is needed, LNG is warmed enough using heat exchangers to vaporize it called re-gasification process, prior to transferring it to the pipeline grid to various users.

Boil-off gas (BOG) management & assessment of LNG's thermodynamic properties are key issues in the technical assessment of LNG storage. Increased vaporization process may negatively affect the stability and safety of the stored LNG. For these reasons the rate of vaporization (boil off rate) should be precisely determined in storage terminal energy systems. [2].

Typical Design Considerations [3]

1. Single containment tank is either a single tank or a tank comprising an inner tank and outer container, designed and constructed so that only the inner tank is required to meet the low temperature ductility requirements for storage of the product. A double containment tank is a tank designed and constructed so that both the inner tank and the outer tank are capable of independently containing the refrigerated liquid stored.
2. The difference between the double containment and full containment is that the outer tank of a full containment tank is intended to be capable of both containing the refrigerated liquid and of controlled venting of the vapour resulting from product leakage after a credible event. Among these three types of LNG tanks, the full containment type is regarded as the most advanced type.
3. The inner tank is manufactured with 9% nickel steel and the outer tank is composed of reinforced concrete and pre-stressed concrete. The 9% nickel steel is widely used as a material for the inner tank since it has the strength and toughness enough for cryogenic uses. The inner tank also has a function of preventing LNG from leakage. The concrete outer tank is designed to resist all the external loads including seismic load. Insulating materials are placed between the inner and outer tank to preserve the stored LNG.
4. The design boil off rate is typically about 0.05 vol%/day & the design vacuum pressure in the dome is about -0.05 kPag.
5. The roof has a suspended ceiling deck and a steel lined concrete dome. The steel liner

installed on the inside surface of the outer concrete tank provides the gas tightness. The boil-off rate is determined by the insulation system.

6. In case of an LNG leakage, liquid may impact the outer tank. Accordingly, liquid tightness must be guaranteed by the corner protection system as well as the polyurethane foam coating installed on the inside surface of the concrete wall. The concrete outer tank protects the inner tank in case of emergency from the outside.
7. The base of the tank has a bottom heating system (BHS) using ethylene glycol as brine fluid. The bottom heating system is installed in order to avoid frost heave.
8. The roof liner consisting of a 9% Ni steel membrane stiffened with rafters in radial and tangential directions acts as formwork for the concrete sphere. The steel structure is fabricated on the bottom slab and lifted by air pressure to its final position. Rafters and roof liner plates are connected with a steel compression ring anchored in the concrete roof ring-beam by welding.

Case Study & Assumptions

To demonstrate the BOG Rate calculations, the following case study is made based on a single containment LNG Tank dimensions & composition given in Ref [2]. In addition, certain assumptions are made for this example case study.

1. The ambient temperature is taken to be 26.85^{0}C & wind velocity is taken to be 36 km/h (10 m/s). The soil temperature is taken as 16.85^{0}C.
2. For heat transfer computations, the average temperature on the LNG Tank surface is taken to be an average of LNG temperature inside the tank and the ambient air temperature.
3. Radiation plays a role & the emissivity of the outer concrete is taken to be 0.9.
4. The LNG tank's inner heat transfer coefficient is taken as 35 W/m^2.K & the soil's heat transfer coefficient is taken as 2 W/m^2.K.
5. Since no information is available from Ref [2] regarding the maximum operating liquid level in the LNG tank, the max operating liquid level (H) is taken to be 2 m less than the height of the inner containment. This would mean a vapour space is available & the heat transfer through this vapour space is neglected.
6. Certain insulation material related to the roof top & tank bottom is not presented in Ref [2], hence the insulation material and their concerned thickness is assumed as shown in the next section.
7. Heat transfer due to free convection is expected. Considering the diameter of the LNG tank to be very large, any section of the concrete wall, is assumed to be similar to flow over a vertical flat plate.
8. The LNG tank outer temperature at the concrete wall is expected to be less than the ambient temperature. In calculating the free convection, Grashof number is expected to yield a negative value. However as per Ref [4], Sec. 9.4, Page 568 indicates, that irrespective of $T_s > T_\infty$ or $T_s < T_\infty$, where, T_∞ is the ambient temperature & T_s is the surface temperature, the foregoing results to estimate Nusselt number still apply. When $T_s < T_\infty$, free convection still develops, except that the convection boundary flows downward. Therefore, in the event of obtaining a negative Grashof Number based on $T_s < T_\infty$, the absolute value is taken.
9. LNG is stored at cryogenic temperatures for significant durations and inevitable heat ingress from the surroundings into the storage tank will lead to vaporization. The more volatile components (methane

and nitrogen) will vaporize preferentially, resulting in weathering of LNG. If left unchecked, weathering can render the remaining LNG unsellable, because of regulatory requirements. Furthermore, weathering increases the overall tank pressure and in order to avoid over-pressurization some of the generated vapour is removed, as boil-off gas (BOG).

10. The LNG industry is specifically concerned in minimizing BOG rates and ensuring that weathering does not greatly impact the LNG quality. In particular weathering prediction is used in planning operations, thus ensuring appropriate allocation of LNG cargoes, its compatibility with stored LNG and avoiding catastrophic events involving stratification, sudden vapour release and rollover, Ref [5]. In the current module, the effects of stratification & roll over are neglected.

LNG Tank Design Data

For BOG estimation, the following process data for heavy LNG is used [2],

Table 1. Heavy LNG Composition

Component	MW	Mol%
-	[kg/kmol]	[%]
Methane [CH_4]	16.04	89.52
Ethane [C_2H_6]	30.07	6.89
Propane [C_3H_8]	44.01	2.42
i-Butane [$i\text{-}C_4H_{10}$]	58.12	0.62
n-Butane [$n\text{-}C_4H_{10}$]	58.12	0.47
i-Pentane [$i\text{-}C_5H_{12}$]	72.15	0.00
n-Pentane [$n\text{-}C_5H_{12}$]	72.15	0.00
Nitrogen [N_2]	28.01	0.08
Total		**100.0**

The LNG molecular weight (MW) and latent heats of vaporization are estimated using Kay's Rule of Mixing for the composition selected and are as follows,

$$MW_{LNG} = \sum_{i=1}^{i=n} x_i MW_i \tag{1}$$

$$\lambda_{LNG} = \sum_{i=1}^{i=n} x_i \lambda_i \tag{2}$$

Where,

MW = Molecular Weight [kg/kmol]

λ = Latent Heat of Vaporization [kJ/kg]

Table 2. LNG Latent Heat of Vaporization & MW

Component	λ_{LNG}	$x_i.MW_i$	$x_i.\lambda_i$
-	[kJ/kg]	[kg/kmol]	[kJ/kg]
Methane	510.8	14.36	457.3
Ethane	489.3	2.07	33.7
Propane	425.6	1.07	10.3
i-Butane	365.1	0.36	2.3
n-Butane	385.7	0.27	1.8
i-Pentane	349.3	0.00	0.0
n-Pentane	367.3	0.00	0.0
Nitrogen	199.2	0.02	0.2
MW_{LNG} & λ_{LNG}		**18.1518**	**505.5**

To estimate the density of LNG, the specific molar volumes as per Ref [6] page 145, 146, (N.B.S. - Technical note 1030 December 1980) need to be computed as,

$$\rho_{LNG} = \frac{M_{LNG}}{\upsilon_{LNG}} \tag{3}$$

$$\upsilon_{LNG} = \sum x_i \upsilon_i - \left[K_1 + (K_1 - K_1)\frac{x_{N_2}}{0.0425}\right] x_{CH_4} \tag{4}$$

Where,

ρ_{LNG} = Density of LNG [kg/m³]

M_{LNG} = Molar Mass of LNG [kg/kmol]

υ_{LNG} = LNG molar volume [L/mol]

M_i = Molar Mass of Component 'i' [kg/kmol]

x_i =Molar fraction of component *'i'* [-]

υ_i = Component molar volumes of *'i'* at LNG Temperature [L/mol]

K_1, K_2 = Correction Factors [-]

The values of K_1, K_2 can be obtained from the below graphs,

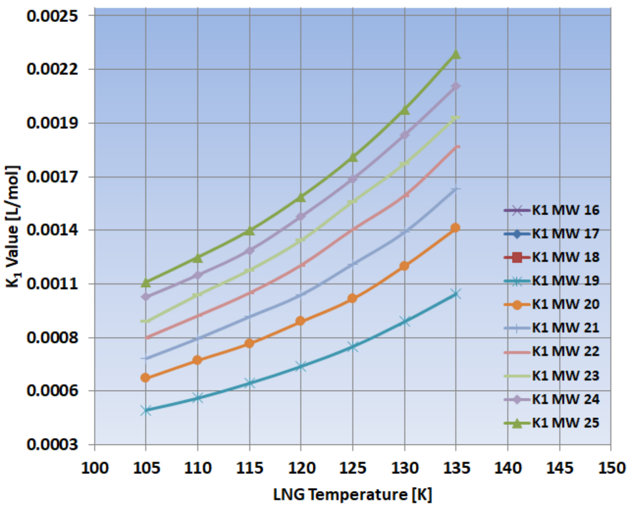

Figure 1. ***K_1*** **Correction Factor**

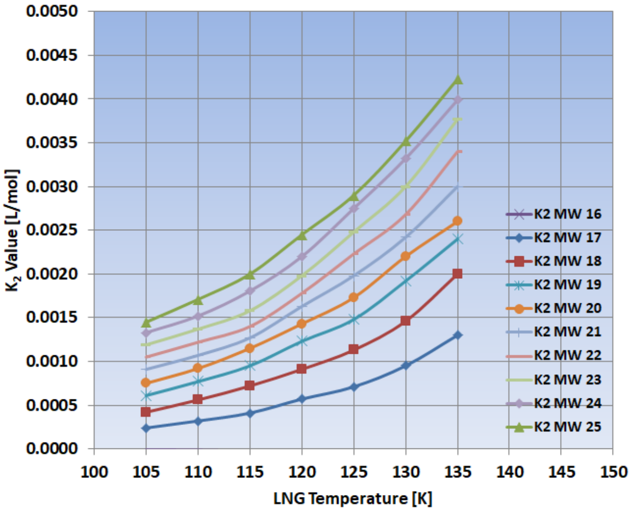

Figure 2. ***K_2*** **Correction Factor**

The component molar volumes of each component *'i'* at LNG Temperature is,

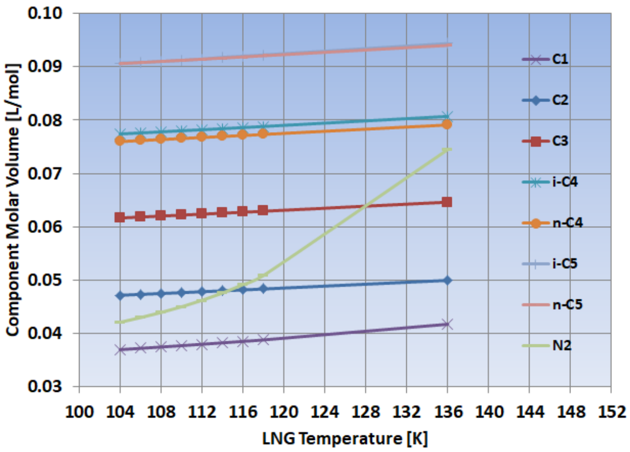

Figure 3. Component Molar Volume [υ_i]

Based on the values of K_1, K_2 & υ_i, the specific molar volumes of each of the components are estimated for an LNG Tank storage temperature of -160.65⁰C (112.5K). The LNG molar volume based on K_1, K_2 is computed as,

Table 3. LNG Molar Volume

Component	υ_i	$\upsilon_{LNG} = x_i.\upsilon_i$
-	[L/mol]	[lit/mol]
Methane	0.0381	0.034070
Ethane	0.0479	0.003299
Propane	0.0624	0.001511
i-Butane	0.0783	0.000485
n-Butane	0.0768	0.000361
i-Pentane	0.0916	0.000000
n-Pentane	0.0915	0.000000
Nitrogen	0.0465	0.000037
Total		**0.03976**

Therefore the LNG density at 112.5K is,

$$\rho_{LNG} = \frac{18.1518}{0.03976} = 456.5\ kg/m^3 \tag{5}$$

The ambient & soil conditions for the BOG rate calculations are as follows,

Table 4. Ambient & Soil Conditions

Parameter	Value	Unit
Ambient Temperature	300	K
Wind Velocity	10	m/s
Soil Temperature	290	K
Soil HTC [h_{soil}]	2	W/m².K

The air properties between -25⁰C & 50⁰C are computed using fitted equations as follows,

Air Density [kg/m³] is computed as,

$$\rho_{Air} = 0.0000158T^2 - 0.0133989T + 3.7622 \tag{6}$$

Specific Heat of Air [kJ/kg.K] is computed as,

$$C_{p,air} = 1.006 \tag{7}$$

The thermal conductivity [W/m.K] of air is,

$$k_{Air} = -2.69 \times 10^{-8}T^2 + 9.04 \times 10^{-5}T + 9.56 \times 10^{-4} \tag{8}$$

The thermal diffusivity [m²/s] of air is,

$$\alpha_{Air} = 1.99 \times 10^{-10}T^2 + 1.5 \times 10^{-8}T - 7.96 \times 10^{-7} \tag{9}$$

The dynamic viscosity [kg/m.s] of air is,

$$\mu_{Air} = -4.22 \times 10^{-11}T^2 + 7.19 \times 10^{-8}T + 8 \times 10^{-7} \tag{10}$$

The kinematic viscosity [m²/s] of air is,

$$\gamma_{Air} = 1.02 \times 10^{-10}T^2 + 3.1 \times 10^{-8}T - 2.69 \times 10^{-6} \tag{11}$$

The Prandtl Number of air is computed as,

$$Pr = -5.12 \times 10^{-7} T^2 + 3.7 \times 10^{-5} T + 0.7642 \qquad (12)$$

Based on the above correlations, for an ambient temperature of 300 K, the air properties are as follows,

Table 5. Air Properties at Ambient Conditions

Parameter	Value	Unit
Density [ρ_{air}]	1.176	kg/m³
Specific Heat [$C_{p,air}$]	1.006	kJ/kg.K
Thermal Conductivity [k_{air}]	0.0256	W/m.K
Thermal Diffusivity [α_{air}]	0.000022	m²/s
Dynamic Viscosity [μ_{air}]	0.000019	kg/m.s
Thermal Exp. Coeff [β_{air}]	0.0033	1/K
Kinematic Viscosity [γ_{air}]	0.00001578	m²/s

LNG Tank Construction Details

The construction details (Ref [2]) in addition to the assumptions made for missing data is,

Table 6. LNG Tank Construction

Parameter	Value	Unit
Tank Height	40	m
Max. Op. Level	38	m
Inner Tank ID	74	m
Inner HTC [Input]	35	W/m².K
LNG Tank Volume	160,000	m³

A schematic of the LNG Tank is shown below,

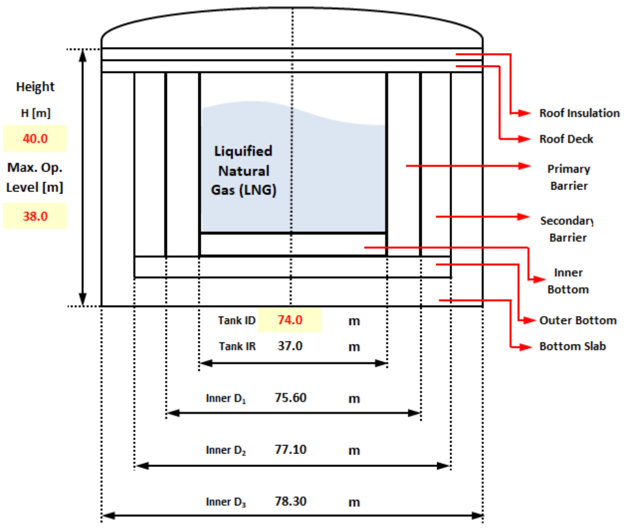

Figure 4. Single Containment LNG Tank Schematic

The various tank insulations are depicted as,

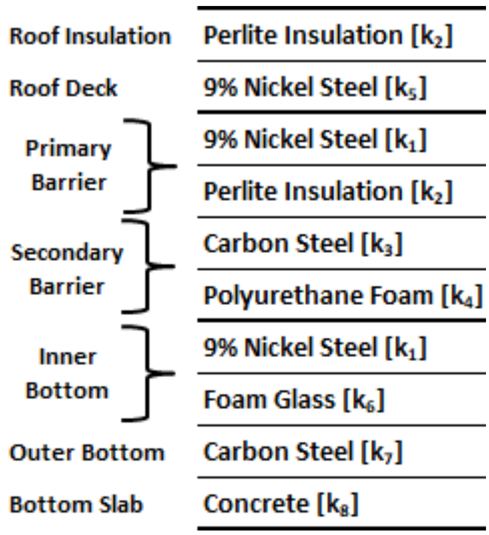

Figure 5. Containment Representation

The thermal conductivity & wall thickness of each layer of insulation is,

Table 7. LNG Tank Insulation

Material	Thermal Cond. [k]	Wall Thickness
-	[W/m.K]	[m]
Perlite Insulation [k_2]	0.038	0.30
9% Ni Steel [k_5]	90.9	0.005
9% Ni Steel [k_1]	90.9	0.20
Perlite Insulation [k_2]	0.038	0.60
Carbon Steel [k_3]	42.6	0.15
Polyurethane [k_4]	0.029	0.60
9% Ni Steel [k_1]	90.9	0.20
Foam Glass [k_6]	0.045	0.60
Carbon Steel [k_7]	42.6	0.15
Concrete [k_8]	1.80	0.60

The various tank Radii/wall thicknesses for calculating resistances in cylindrical coordinates & linear coordinates are,

Table 8. LNG Tank Insulation

Material		Tank Radii/Wall Thickness
-	-	[m]
Perlite Insulation [k_2]	r_9	0.30
9% Nickel Steel [k_5]	r_{10}	0.01
9% Nickel Steel [k_1]	r_1	37.20
Perlite Insulation [k_2]	r_2	37.80

Carbon Steel [k_3]	r_3	37.95
Polyurethane Foam [k_4]	r_4	38.55
9% Nickel Steel [k_1]	r_5	0.20
Foam Glass [k_6]	r_6	0.60
Carbon Steel [k_7]	r_7	0.15
Concrete [k_8]	r_8	0.60

Design Methodology

To estimate the heat load, thermal conductivity and heat transfer coefficients determine the amount of heat transferred to the cryogenic LNG through the walls of the tank. The modes of heat transfer driven by temperature differences are conduction through the tank wall & its various insulation layers, free convection, forced convection & radiation from the ambient.

Forced Convection

To calculate the external heat transfer coefficient ($h_{air,overall}$), taking the assumptions made, the average surface temperature (T_{avg}) of the concrete insulation is taken as an average of LNG boiling temperature ($T_{b,LNG}$) & ambient temperature (T_a),

$$T_{avg} = \frac{T_{b,LNG}+T_a}{2} = \frac{112.5+300}{2} = 206.25\ K \qquad (13)$$

Prandtl Number of ambient air is,

$$Pr = \frac{C_{P,air}\mu_{air}}{k_{air}} = \frac{1.006\times0.000019}{0.0256} \approx 0.72926 \qquad (14)$$

Based on the assumption that the tank diameter is large; the flow of air over the tank is approximated to flow over a flat plate. Therefore Reynolds number (Re) becomes,

$$Re = \frac{V_{air}L_{max\ op\ Level}}{\gamma_{air}} = \frac{10\times38}{0.00001578} = 2.41\times10^7 \quad (15)$$

Nusselt number for forced convection over circular cylinder with cross flow can be estimated using Churchill and Bernstein correlation [4]. This equation is valid for all $Re.Pr \geq 2$ and the correlation is expressed as,

$$Nu = 0.3 + \frac{0.62\,Re^{1/2}\,Pr^{1/3}}{\left[1+\left(\frac{0.4}{Pr}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5} \quad (16)$$

The above correlation is valid for all ranges of Reynolds number (Re) and $Pr \geq 0.2$, where all properties are evaluated at film temperature. It is to be noted that as per [4], Churchill & Bernstein correlation is reasonable over a certain range of conditions but for most engineering calculations, the accuracy is not expected to be much better than 20% because these are based on more recent results encompassing a wide range of conditions [4].

$$Nu = 0.3 + \frac{0.62\times2.41\times10^{7^{1/2}}\,0.72926^{1/3}}{\left[1+\left(\frac{0.4}{0.72926}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{2.41\times10^7}{282000}\right)^{5/8}\right]^{4/5}$$

(17)

$$Nu_{force} \approx 23{,}357 \qquad (18)$$

The external forced convection heat transfer coefficient can be calculated from Nusselt number as,

$$h_{force} = \frac{k_{air}Nu_{force}}{L_{MaxOp.Level}} = \frac{0.0256\times23357}{38} \approx 15.8\ W/m^2K \quad (19)$$

Radiation Heat Transfer

To estimate the radiation heat transfer between the ambient & concrete insulation on the tank, the expression is written as [4],

$$h_r = \varepsilon\sigma(T_{con}+T_a)(T_{con}^2+T_a^2) \qquad (20)$$

The radiation mode expressed above is written in a manner similar to convection, i.e., the radiation rate equation is linearized making the heat rate proportional to the temperature difference rather than to the difference between two temperatures to the fourth power [2]. Therefore the radiation heat transfer coefficient (h_r) for emissivity of concrete (ε) taken as 0.9 & Stefan Boltzmann Constant (σ) of 5.67×10^{-8} W/m^2/K is,

$$h_r = 0.9\times5.67\times10^{-8}(206.25+112.5)\times(206.25^2+112.5^2) \quad (21)$$

Or, $h_r \approx 0.9\ W/m^2K$ (22)

Natural/Free Convection

To estimate the heat transfer due to natural convection, the correlation by Churchill & Chu [4] can be used and is of the form,

$$Nu = \left\{0.825 + \frac{0.387\,Ra^{1/6}}{\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 \quad (23)$$

Where, Rayleigh number (Ra) is the product of Grashof number (Gr) and Reynolds number (Re) computed as,

$$Ra = Gr \times Pr \quad (24)$$

Grashof number is computed as,

$$Gr = \frac{g \times \beta_{air} \times [T_S - T_\infty] L^3}{\gamma_{air}^2} \quad (25)$$

Where 'β' is the thermal expansion coefficient of air approximated as $1/T_a$.

$$Gr = \frac{9.812 \times 0.00333 \times |206.25 - 300| \times 38^3}{0.00001578^2} \approx 6.75 \times 10^{14} \quad (26)$$

The Rayleigh number and Nusselt number is,

$$Ra = 6.75 \times 10^{14} \times 0.7293 = 4.93 \times 10^{14} \quad (27)$$

$$Nu = \left\{0.825 + \frac{0.387 \times \left(4.93 \times 10^{14}\right)^{1/6}}{\left[1+\left(\frac{0.492}{0.7293}\right)^{9/16}\right]^{8/27}}\right\}^2 = 8497 \quad (28)$$

Therefore the external free convection heat transfer coefficient is calculated as,

$$h_{free} = \frac{k_{air} Nu_{free}}{L_{MaxOp.Level}} = \frac{0.0256 \times 8497}{38} \approx 5.73\, W/m^2K \quad (29)$$

To determine whether the heat transfer mode is dominated by free/natural convection or forced convection, the condition that is required to be satisfied is as follows [4],

$$\frac{Gr}{Re^2} \ll 1 \text{ ; Free convection is negligible} \quad (30)$$

$$\frac{Gr}{Re^2} \gg 1 \text{ ; Forced convection is negligible} \quad (31)$$

$$\frac{Gr}{Re^2} \approx 1 \text{ ; Mixed Convection Exists} \quad (32)$$

Therefore, checking for mode of convection,

$$\frac{Gr}{Re^2} = \frac{6.75 \times 10^{14}}{(2.41 \times 10^7)^2} = 1.1652 \approx 1 \quad (33)$$

Therefore the mode of heat transfer is mixed convection and the combined heat transfer coefficient is computed as,

$$Nu_{combined} = \left[Nu_{free}^4 + Nu_{forced}^4\right]^{1/4} \quad (34)$$

$$Nu_{combined} = [23357^4 + 8497^4]^{1/4} \quad (35)$$

$$Nu_{combined} = 23{,}458 \quad (36)$$

Therefore the combined external heat transfer coefficient, h_{comb}, is computed as,

$$h_{comb} = \frac{k_{air} Nu_{comb}}{L_{MaxOp.Level}} = \frac{0.0256 \times 23458}{38} \approx 15.8\, W/m^2K \quad (37)$$

Therefore the external heat transfer coefficient, $h_{air,\,overall}$, is computed as,

$$h_{air,overall} = h_{comb} + h_r \quad (38)$$

$$h_{air,overall} = 15.83 + 0.9 = 16.73\, W/m^2K \quad (39)$$

LNG Heat Load Estimation

To estimate the heat load ingress into the LNG tank, Fourier's Law of conduction is applied along the tank wall, tank roof & tank bottom. For tank wall, the heat load is computed as,

$$Q_W = \frac{-2\pi H_{max}[T_{LNG} - T_S]}{\sum R_1} \quad (40)$$

Where R_1 is the resistances through the tank wall.

$$A_1 = \frac{1}{r_{out} h_{air,overall}} = \frac{1}{37 \times 16.73} = 0.0016\, m.K/W \quad (41)$$

$$A_2 = \frac{1}{k_{9\%\,Ni}} ln\left[\frac{r_1}{r_{in}}\right] = \frac{1}{90.9} ln\left[\frac{37.2}{37}\right] = 6 \times 10^{-5} m.K/W \quad (42)$$

$$A_3 = \frac{1}{k_{perli}} ln\left[\frac{r_2}{r_1}\right] = \frac{1}{0.038} ln\left[\frac{37.8}{37.2}\right] = 0.4211\, m.K/W \quad (43)$$

$$A_4 = \frac{1}{k_{CS}} ln\left[\frac{r_3}{r_2}\right] = \frac{1}{42.6} ln\left[\frac{37.95}{37.8}\right] = 9 \times 10^{-5} m.K/W \quad (44)$$

$$A_5 = \frac{1}{k_{poly}} ln\left[\frac{r_4}{r_3}\right] = \frac{1}{0.029} ln\left[\frac{38.55}{37.95}\right] = 0.541\, m.K/W \quad (45)$$

$$A_6 = \frac{1}{k_{conc}} ln\left[\frac{r_4 + r_8}{r_4}\right] = \frac{1}{1.8} ln\left[\frac{39.15}{38.55}\right] = 0.0086\, m.K/W \quad (46)$$

$$A_7 = \frac{1}{r_{in} h_{LNG}} = \frac{1}{39.15 \times 35} = 0.00073\, m.K/W \quad (47)$$

$$\sum R_1 = A_1 + A_2 + A_3 + A_4 + A_5 + A_6 + A_7 \quad (48)$$

$$\sum R_1 = 0.973\, m.K/W \quad (49)$$

The heat load through the tank wall is,

$$Q_W = \frac{-2\pi \times 38 \times [112.5 - 206.25]}{0.973} = 23{,}005\, W \quad (50)$$

The heat load through the Tank bottom is computed in linear coordinates as,

$$Q_B = \frac{-\pi r^2 [T_{LNG} - T_{soil}]}{\sum R_2} \quad (51)$$

Where R_2 is the tank bottom resistance,

$$B_1 = \frac{1}{h_{LNG}} = \frac{1}{35} = 0.0286\ m^2K/W \quad (52)$$

$$B_2 = \frac{r_5}{k_{9\%\ Ni}} = \frac{0.2}{90.9} = 0.0022\ m^2K/W \quad (53)$$

$$B_3 = \frac{r_6}{k_{foam}} = \frac{0.6}{0.045} = 13.34\ m^2K/W \quad (54)$$

$$B_4 = \frac{r_7}{k_{CS}} = \frac{0.15}{42.6} = 0.0035\ m^2K/W \quad (55)$$

$$B_5 = \frac{r_8}{k_{conc}} = \frac{0.6}{1.8} = 0.3334\ m^2K/W \quad (56)$$

$$B_6 = \frac{1}{h_{soil}} = \frac{1}{2} = 0.5\ m^2K/W \quad (57)$$

$$\sum R_2 = B_1 + B_2 + B_3 + B_4 + B_5 + B_6 \quad (58)$$

$$\sum R_2 = 14.2\ m^2K/W \quad (59)$$

The heat load through the tank Bottom is,

$$Q_B = \frac{-\pi \times 39.15^2 [112.5 - 290]}{14.2} = 60{,}186\ W \quad (60)$$

Similarly, the heat load through the Tank Roof is computed in linear coordinates as,

$$Q_R = \frac{-\pi r^2 [T_{LNG} - T_a]}{\sum R_3} \quad (61)$$

Where R_3 is the tank roof resistance,

$$C_1 = \frac{1}{h_{LNG}} = \frac{1}{35} = 0.0286\ m^2K/W \quad (62)$$

$$C_2 = \frac{r_{10}}{k_{9\%\ Ni}} = \frac{0.0005}{90.9} = 0.00006\ m^2K/W \quad (63)$$

$$C_3 = \frac{r_6}{k_{perlite}} = \frac{0.3}{0.038} = 7.8947\ m^2K/W \quad (64)$$

$$C_4 = \frac{r_8}{k_{Conc}} = \frac{0.6}{1.8} = 0.0033\ m^2K/W \quad (65)$$

$$C_5 = \frac{1}{h_{air,overall}} = \frac{1}{16.73} = 0.0598\ m^2K/W \quad (66)$$

$$\sum R_3 = C_1 + C_2 + C_3 + C_4 + C_5 \quad (67)$$

$$\sum R_3 = 8.32\ m^2K/W \quad (68)$$

The heat load through the tank roof is,

$$Q_R = \frac{-\pi \times 39.15^2 [112.5 - 206.25]}{8.32} \approx 54{,}281\ W \quad (69)$$

The total heat load transferred to the LNG is,

$$Q_{LNG} = Q_W + Q_B + Q_R \quad (70)$$

$$Q_{LNG} = 23 + 60.2 + 54.3 \approx 137.5\ kW \quad (71)$$

With the heat load estimated, the BOG is,

$$BOG = \frac{Q_{LNG} \times 24 \times 3600}{\lambda_{LNG}} = \frac{137.5 \times 24 \times 3600}{505.5} \approx 23{,}494 kg/d \quad (72)$$

For the LNG Tank volume of 160,000 m^3,

$$BOG\ Rate = \frac{BOG \times 100}{V_{LNG} \times \rho_{LNG}} = \frac{23494 \times 100}{160000 \times 456.5} \approx 0.032\% \quad (73)$$

From the estimated BOG Rate of 0.032 vol%, it is lower than the allowable 0.05 vol%/day.

Note: The above steps are repeated (under the assumption $T_{wall} = T_{b,LNG}$) until the average wall temperature converges, i.e., Iteration 2, $T_{b,LNG,2}$ becomes,

$$T_{b,LNG,2}[K] = T_{b,LNG,1}[K] - R_{ins}\left[\frac{m^2K}{W}\right] \times \frac{Q}{A}\left[\frac{W}{m^2}\right]$$

$$\frac{Q}{A}\left[\frac{W}{m^2}\right] = \frac{T_{wall}[K] - T_{amb}[K]}{R_{total}\left[\frac{m^2K}{W}\right]}$$

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Module 14
Gas Condensate Separation Stages – Design & Optimization

The life cycle of an oil & gas venture begins at the wellhead where subsurface engineers work their way through surveying, drilling, laying production tubing and well completions. Once a well is completed, gathering lines from each well is laid to gather hydrocarbons and transported via a main trunk line to a gas oil separation unit (GOSP) to be processed further to enhance their product value for sales. Gas condensate wells consist of natural gas which is rich in heavier hydrocarbons that are recovered as liquids in separators in field facilities or gas-oil separation plants (GOSP).

The following is aimed at demonstrating how to optimize and provide the required number of separation stages to process a gas condensate mixture and separate them into their respective vapour phase and liquid phase – termed as "Stage Separation". Stage separation consists of laying a series of separators which operate at consecutive lower pressures to strip out vapours from the well liquids & resulting in a stabilized liquid. Prior to any hydrocarbon processing in a gas processing plant or a refinery, it is imperative to maximize the liquid recovery as well as provide a stabilized liquid hydrocarbon. A schematic of a 2-Stage Separation Unit is as follows,

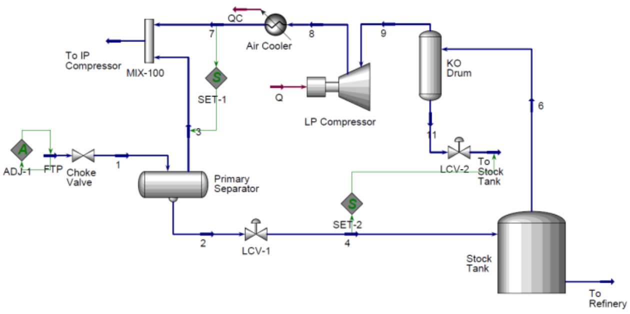

Figure 1. Two Stage Separation

A schematic of a 3-Stage Separation Unit is as follows,

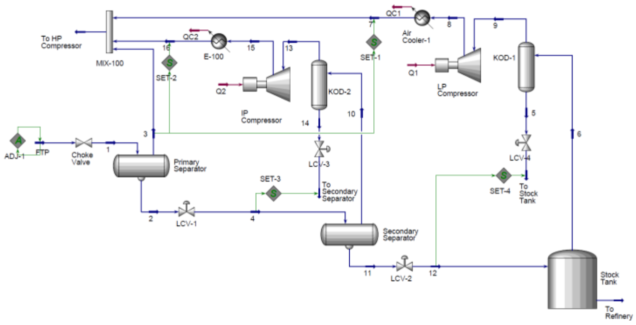

Figure 2. Three Stage Separation

General Notes

1. Stage Separation consists of series of separators which flash the incoming well fluids consisting of gas, oil and water into their respective constituents. The oil/condensates exiting the separation vessel is eventually routed to a storage tank, a.k.a, Stock Tank which operates at atmospheric conditions. The liquid in the stock tank are in turn termed as stabilized crude. To increase vaporization, sometimes a heater is installed in the liquid side of the separation stages.
2. The art of flashing well fluids in successive separation stages to increase liquid yield is not linear but inversely proportional to the number of stages. Therefore the liquid recovery with an extra stage to a single stage system can be substantial. However adding an extra stage to more than three stages does not produce a significant yield in liquid quantities. In theory, as the number of separation stages increase, so will the liquid yield increase and the gas and liquid reach equilibrium. In practice, a three stage separation process is very efficient and cost effective to arrive at a stabilized stock.
3. In terms of terminology, a 2-Stage separation system means one separator and an atmospheric storage tank (stock tank) because the storage tank also acts as

a separator. Similarly, a 3-Stage separator consists of a two separators & a stock tank.

4. The number of stages required to provide a gas and liquid at equilibrium conditions, referred to as flashing can be estimated by empirical correlations or using Equation of state (EoS). In this module, the method of EoS is used.
5. Separator calculations primarily consist of optimum separator pressure and temperature, Gas-Oil Ratio (GOR), API Gravity of the Stock Tank Oil, Oil formation factor (a.k.a Formation Volume Factor, FVF/B_o) and the respective compositions.
6. The key to selecting the optimum number of stages is based on, Minimum GOR, Maximum API gravity & Minimum Oil FVF.
7. Minimum GOR implies maximum liquid yield. GOR is termed as the cumulative gas flow from all the separators including the stock tank divided by the amount of oil/condensate exiting the stock tank. Higher API gravity implies, higher is the commercial value of the oil/condensate. Oil FVF can be defined as the volume of reservoir fluid required to produce a barrel of stock tank oil. Therefore lower Oil FVF implies more stock tank oil for a given volume of reservoir fluids. Oil FVF values typically range between 1 bbl/STB to 3 bbl/STB.
8. Since the process conditions of the stock tank are already fixed to be at atmospheric conditions while the primary separator is determined by the operating conditions required to avoid inhibiting well production, the only control had, are the operating pressures in the intermediate stages. Therefore the intermediate pressure is controlled to optimize and obtain the highest amount of liquid yield.
9. 2-Stage separation is applicable for low GOR, low API gravity oils & low flowing tubing pressure (FTP) wells. 3-Stage Separation is for intermediate GOR, intermediate API gravity oils & intermediate FTP wells. 4-Stage Separation is for high GOR, high API gravity oils & high FTP wells.
10. Separation vessels also consist of gas control valve on the gas side which also serve the purpose of controlling the wellhead back pressure.
11. For effective operation, minimizing energy & costs, the primary separator pressure should be lower than the Wellhead FTP and higher than the gas pipeline export pressure. Failure to keep the primary separator pressure higher than the pipeline pressure would require the installation of gas compressors to boost gas pressure to export requirements.

Case Study

A Gas-Condensate well with an FTP of 14.48 bara [210 psia], 45^{0}C and liquid fraction of 1.0, flows to a gas oil separation unit at a production rate of 78,295 STBPD [533.3 Am^3/h].

Assumptions

1. The stock tank pressure & temperature is constant at atmospheric conditions and weather remains unchanged.
2. The composition of the well fluids is taken to be constant for a given FTP.
3. Both Gas and liquid are assumed to be at complete equilibrium upon flashing.
4. The well fluids composition is chosen to be free from H_2S, CO_2, N_2 and produced water. Presence of water would require a 3 phase separator. For this module, a 2-Phase separator is used considering only gas and condensate.

The gas condensate composition of the well fluids is as follows,

Table 1. Gas Condensate Composition

Components	Mole Fraction	Units
Hydrogen Sulphide [H_2S]	0.0000	-
Carbon Dioxide [CO_2]	0.0000	-
Nitrogen [N_2]	0.0000	-
Methane [C_1]	0.0385	-
Ethane [C_2]	0.0391	-
Propane [C_3]	0.0516	-
iso-Butane [iC_4]	0.0145	-
n-Butane [nC_4]	0.0575	-
Iso-Pentane [iC_5]	0.0231	-
n-Pentane [nC_5]	0.0346	-
n-Hexane [C_6]	0.0491	-
Heptane Plus [C_{7+}]	0.6920	-
Total	**1.0000**	**-**
Heptane Plus [C_{7+}] SG	0.8576	
Heptane Plus [C_{7+}] MW	227	lb/lbmole

Methodology

The number of separators can be chosen by first arriving at a preliminary pressure estimate based on equal pressure ratio as,

$$X^n = \left[\frac{P_{primary}}{P_{Stock}}\right] \tag{1}$$

Where,

$P_{primary}$ = Primary Separator Pressure [bara]

P_{Stock} = Stock Tank Pressure [bara]

n = Number of stages [-]

X = Maximum number of Stages [-]

Rewriting the expression,

$$n \times lnX = ln\left[\frac{P_{primary}}{P_{Stock}}\right] \tag{2}$$

$$\text{Or, } n = \frac{ln\left[\frac{P_{primary}}{P_{Stock}}\right]}{lnX} \tag{3}$$

The separation ratio is computed as,

$$R = \left[\frac{P_{primary}}{P_{Stock}}\right]^{1/n} \tag{4}$$

The intermediate pressure is computed as,

$$P_i = \frac{P_{primary}}{R^i} \tag{5}$$

Where,

P_i = Intermediate Pressure at Stage 'i'

Therefore considering a maximum number of stages of 3, with the well fluids data, for a two stage separation unit, the primary separator pressure [P_1] and Separation ratio [R] is,

$$n = \frac{ln\left[\frac{14.48}{1.01325}\right]}{ln[3]} = 2.42 \sim 2\ Stages \tag{6}$$

$$R = \left[\frac{14.48}{1.01325}\right]^{1/2} = 3.7801 \tag{7}$$

$$P_1 = \frac{14.48}{3.7801^1} = 3.831\ bara \tag{8}$$

The stock tank becomes the second stage and the operating pressure is P_2 = 1.01325 bara.

For a three stage separation unit, the primary separator pressure [P_1] and secondary separator pressure [P_2] is,

$$R = \left[\frac{14.48}{1.01325}\right]^{1/3} = 2.4266 \tag{9}$$

$$P_1 = \frac{14.48}{2.4266^1} = 5.968\ bara \tag{10}$$

$$P_2 = \frac{14.48}{2.4266^2} = 2.459\ bara \tag{11}$$

The stock tank becomes the third stage and the operating pressure is P_3 = 1.01325 bara.

Using these preliminary stage pressures, a vapour-liquid equilibrium (VLE) flash calculation can be performed to estimate GOR, API gravity and Oil FVF. For this module, the Peng Robinson EoS is chosen. To arrive at the final separator pressures, the following iterative procedure is adopted.

1. Keeping all preliminary estimated pressures fixed, the primary separator pressure is varied for a pressure range to obtain GOR, Oil FVF & API gravity. Making a plot of the above values, the separator pressure corresponding to Min GOR, Max API gravity & Min FVF is chosen [1st Iteration of 1st Stage].

2. The primary separator initial estimate pressure is now replaced with the 1st Iteration's optimized pressure. Following further, the secondary separator pressure is also varied for a given range to similarly obtain an optimized pressure corresponding to Min GOR, Max API gravity & Min Oil FVF [1st Iteration of 2nd stage].
3. The secondary separator initial estimate pressure is now replaced with the optimized value, [1st Iteration of 2nd stage].
4. With the 1st iteration optimized pressures, flash calculations are repeated similar to Step 2 and Step 3, i.e., 2nd Iteration and so forth, until a converged solution is reached.

Results

With the flash procedure applied for the calculated initial estimates, the results of 2-Stage Separation and 3-Stage Separation for a compressor polytropic efficiency of 82% are,

Table 2. Separation Stage Pressures

Stages	Operating Pressure	Compressor Power
-	[bara]	[kW]
2-Stage Separation [Liquid Rate = 75,235 STBD]		
FTP	14.48	
Primary Separator	2.60	83.04
Stock Tank	1.014	
3-Stage Separation [Liquid Rate = 75,430 STBD]		
FTP	14.48	
Primary Separator	3.90	71.47
Secondary Separator	1.70	65.32
Stock Tank	1.014	

From the results between the 2-Stage and 3-Stage separation, the liquid yield increased by 195 STBD, i.e., 0.26%. The GOR, Oil FVF and API gravity for the separation stages are also tabulated below. Between both cases, the GOR decreases from 90.77 scf/STB to 86.68 scf/STB. API gravity increases from 37.93 to 38.08. The Oil FVF decreases from 1.0466 m^3/m^3 to 1.0435 m^3/m^3.

Table 3. GOR, Oil FVF and API Gravity

Stages	GOR	Oil FVF	API Gravity
-	[scf/STB]	[m^3/m^3]	[^{0}API]
2	90.77	1.0466	37.93
3	86.68	1.0435	38.08

The plots of Total GOR, API Gravity & Oil FVF for two stage design and three stage design are as follows,

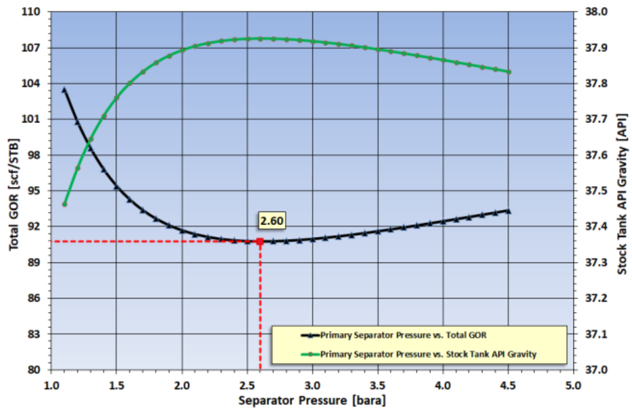

Figure 3. Two Stages – GOR & API Gravity

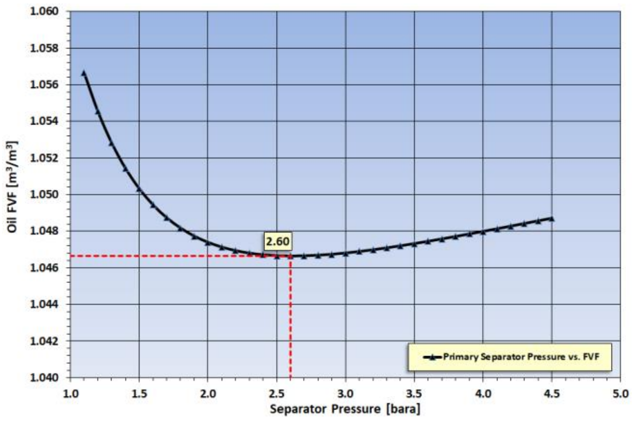

Figure 4. Two Stages – Oil FVF

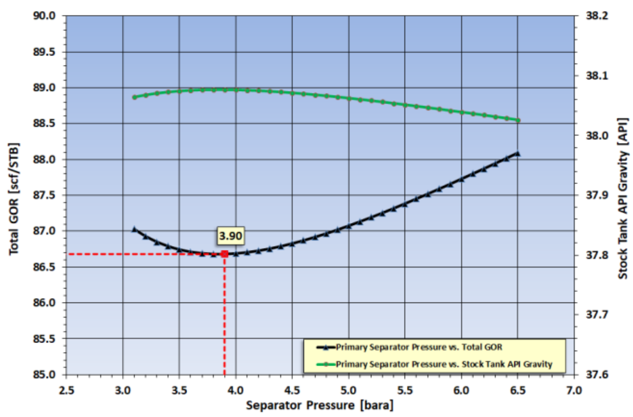

Figure 5. Three Stages – Primary GOR & API Gravity

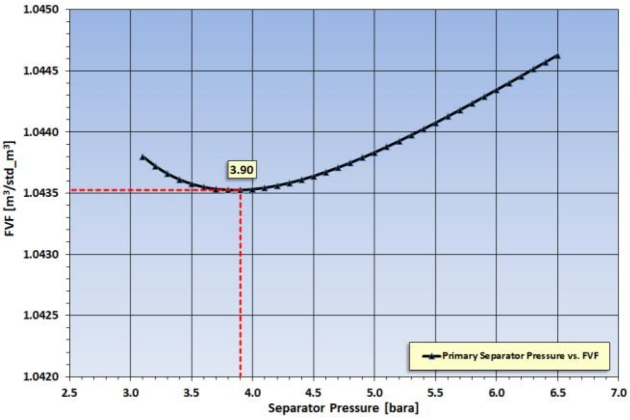

Figure 6. Three Stages – Primary Oil FVF

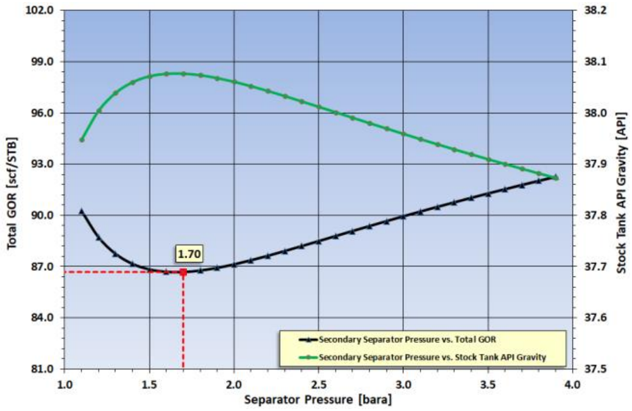

Figure 7. Three Stages – Sec. GOR & API Gravity

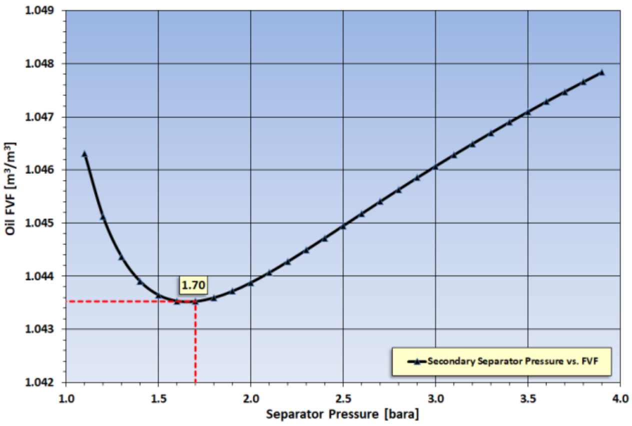

Figure 8. Three Stages – Secondary Oil FVF

Conclusions

From the calculations made, a three stage separator offers the advantage of higher liquid yield. The Gas-Oil Ratio (GOR) can also be decreased which allows to recover more liquids and thereby offering higher commercial value. Although a 4th stage can be added to increase liquid yield, since well pressures are low at ~15 bara, 3 stages would suffice. It is also seen that with increase in number of stages, the LP compressor power requirements also decreases.

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Appendix A

TWO STAGE GAS-CONDENSATE SEPARATION

Gas-Condensate Composition	
Component	**Mole fraction**
Hydrogen Sulphide [H_2S]	0.0000
Carbon Dioxide [CO_2]	0.0000
Nitrogen [N_2]	0.0000
Methane [C_1]	0.0385
Ethane [C_2]	0.0391
Propane [C_3]	0.0516
iso-Butane [iC_4]	0.0145
n-Butane [nC_4]	0.0575
Iso-Pentane [iC_5]	0.0231
n-Pentane [nC5]	0.0346
n-Hexane [C_6]	0.0491
Heptane Plus [C_{7+}]	0.6920
Total	**1.0000**

No. of Gas-Condensate Separation Stages	
Flow Tubing Pressure [bara]	14.48
Flow Tubing Temperature [bara]	45.0
Stock Tank Pressure [bara]	1.01
Max Number of Stages [X]	3
No. of Stages [Calc] [n]	2.42
No. of Stages [Rounded]	2.00
Separation Ratio [R]	3.78

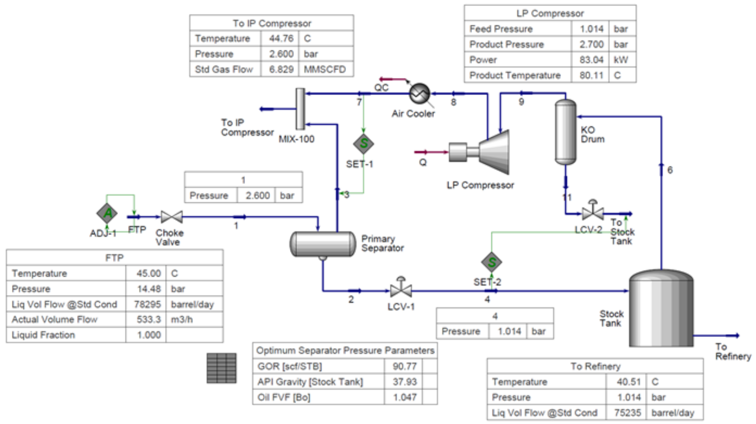
Initial Estimate of Equipment Pressure	
Stage No.	**Pressure [bara]**
1	3.831
2	1.014

RESULTS	
Min GOR [scf/STB]	90.77
Max Stock Tank API Gravity [API]	37.93
Min FVF [m^3/m^3]	1.0466
Liquid Produced [Std_BDP]	75,235
Compressor Absorbed Power [kW]	83.04
Selected Separator Pressure	**2.60**

FLASH CALCULATIONS - PENG ROBINSON EoS

Primary Separator		Stock Tank		Total GOR	Stock Tank API Gravity	Formation Volume Factor of Condensate [FVF] / B_o
Pressure	Temperature	Pressure	Temperature			
[bara]	[°C]	[bara]	[°C]	[scf/STB]	[°API]	[m^3/m^3]
1.1	39.4	1.013	39.3	103.52	37.46	1.05666
1.2	39.7	1.013	39.6	100.80	37.56	1.05455
1.3	40.0	1.013	39.8	98.59	37.65	1.05283
1.4	40.3	1.013	40.0	96.82	37.71	1.05145
1.5	40.5	1.013	40.1	95.41	37.76	1.05033
1.6	40.7	1.013	40.2	94.28	37.80	1.04945
1.7	40.9	1.013	40.3	93.39	37.83	1.04874
1.8	41.1	1.013	40.3	92.68	37.86	1.04818
1.9	41.3	1.013	40.4	92.13	37.88	1.04774
2	41.5	1.013	40.4	91.70	37.89	1.04740
2.1	41.6	1.013	40.5	91.37	37.90	1.04714
2.2	41.8	1.013	40.5	91.13	37.91	1.04694
2.3	41.9	1.013	40.5	90.96	37.92	1.04681
2.4	42.0	1.013	40.5	90.85	37.92	1.04672
2.5	42.2	1.013	40.5	90.79	37.93	1.04667
2.6	**42.3**	**1.013**	**40.5**	**90.77**	**37.93**	**1.04665**
2.7	42.4	1.013	40.5	90.78	37.93	1.04666
2.8	42.5	1.013	40.5	90.82	37.92	1.04669
2.9	42.6	1.013	40.5	90.89	37.92	1.04674
3	42.7	1.013	40.5	90.97	37.92	1.04681
3.1	42.8	1.013	40.5	91.08	37.91	1.04690
3.2	42.8	1.013	40.5	91.20	37.91	1.04699
3.3	42.9	1.013	40.5	91.33	37.91	1.04709
3.4	43.0	1.013	40.4	91.47	37.90	1.04721
3.5	43.1	1.013	40.4	91.62	37.90	1.04733
3.6	43.1	1.013	40.4	91.78	37.89	1.04745
3.7	43.2	1.013	40.4	91.94	37.88	1.04758
3.8	43.3	1.013	40.4	92.11	37.88	1.04772
3.9	43.3	1.013	40.4	92.28	37.87	1.04785
4	43.4	1.013	40.4	92.46	37.87	1.04799
4.1	43.4	1.013	40.3	92.63	37.86	1.04814
4.2	43.5	1.013	40.3	92.82	37.85	1.04828
4.3	43.5	1.013	40.3	93.00	37.85	1.04842
4.4	43.6	1.013	40.3	93.18	37.84	1.04857
4.5	43.6	1.013	40.3	93.37	37.83	1.04871

HYSYS Model



Appendix B

THREE STAGE GAS-CONDENSATE SEPARATION

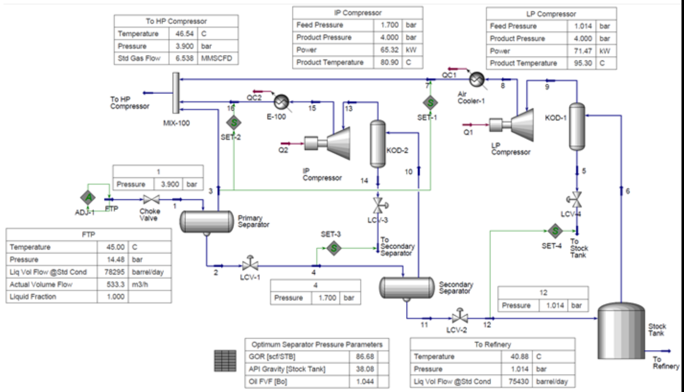
Gas-Condensate Composition

Component	Mole fraction
Hydrogen Sulphide [H_2S]	0.0000
Carbon Dioxide [CO_2]	0.0000
Nitrogen [N_2]	0.0000
Methane [C_1]	0.0385
Ethane [C_2]	0.0391
Propane [C_3]	0.0516
iso-Butane [iC_4]	0.0145
n-Butane [nC_4]	0.0575
Iso-Pentane [iC_5]	0.0231
n-Pentane [nC5]	0.0346
n-Hexane [C_6]	0.0491
Heptane Plus [C_{7+}]	0.6920
Total	1.0000
No. of Gas-Condensate Separation Stages	
Flow Tubing Pressure [bara]	14.48
Flow Tubing Temperature [bara]	45.0
Stock Tank Pressure [bara]	1.01
Max Number of Stages [X]	3
No. of Stages [Calc] [n]	2.42
No. of Stages [Rounded]	3.00
Separation Ratio [R]	2.43
Initial Estimate of Equipment Pressure	
Stage No.	Pressure [bara]
1	5.968
2	2.459
3	1.014
RESULTS	
Liquid Produced [Std_BDP]	75,430
Primary Min GOR [scf/STB]	86.68
Max Stock Tank API Gravity [API]	38.08
Min FVF [m^3/m^3]	1.0435
Compressor Absorbed Power [kW]	71.47
Primary Separator Pressure	**3.90**
Secondary Min GOR [scf/STB]	86.68
Max Stock Tank API Gravity [API]	38.08
Min FVF [m^3/m^3]	1.0435
Compressor Absorbed Power [kW]	65.32
Secondary Separator Pressure	**1.70**

Flash Calculations - Peng Robinson EoS

Primary Separator Pressure	Primary Separator Temperature	Total GOR	Stock Tank API Gravity	Formation Volume Factor of	Secondary Separator Pressure	Secondary Separator Temperature	Total GOR	Stock Tank API Gravity	Formation Volume Factor of Condensate [FVF] / B_o
[bara]	[°C]	[scf/STB]	[°API]	[m^3/m^3]	[bara]	[°C]	[scf/STB]	[°API]	[m^3/m^3]
3.1	42.8	87.03	38.06	1.04380	1.1	40.7	90.26	37.95	1.04631
3.2	42.8	86.93	38.07	1.04372	1.2	41.0	88.72	38.00	1.04512
3.3	42.9	86.85	38.07	1.04366	1.3	41.2	87.75	38.04	1.04437
3.4	43.0	86.79	38.07	1.04361	1.4	41.4	87.16	38.06	1.04390
3.5	43.1	86.74	38.07	1.04358	1.5	41.7	86.83	38.07	1.04365
3.6	43.1	86.71	38.08	1.04355	1.6	41.8	86.69	38.08	1.04353
3.7	43.2	86.69	38.08	1.04353	1.7	42.0	86.68	38.08	1.04353
3.8	43.3	86.68	38.08	1.04353	1.8	42.2	86.77	38.07	1.04359
3.9	43.3	86.68	38.08	1.04353	1.9	42.3	86.92	38.07	1.04372
4	43.4	86.69	38.08	1.04353	2	42.4	87.13	38.06	1.04388
4.1	43.4	86.71	38.08	1.04354	2.1	42.5	87.37	38.05	1.04407
4.2	43.5	86.73	38.08	1.04356	2.2	42.6	87.63	38.04	1.04428
4.3	43.5	86.76	38.07	1.04358	2.3	42.7	87.91	38.03	1.04449
4.4	43.6	86.79	38.07	1.04361	2.4	42.8	88.20	38.02	1.04472
4.5	43.6	86.83	38.07	1.04364	2.5	42.8	88.49	38.01	1.04495
4.6	43.7	86.87	38.07	1.04367	2.6	42.9	88.79	38.00	1.04518
4.7	43.7	86.92	38.07	1.04371	2.7	42.9	89.08	37.99	1.04541
4.8	43.8	86.97	38.07	1.04375	2.8	43.0	89.37	37.98	1.04563
4.9	43.8	87.02	38.06	1.04379	2.9	43.0	89.66	37.97	1.04585
5	43.8	87.08	38.06	1.04383	3	43.1	89.95	37.96	1.04607
5.1	43.9	87.13	38.06	1.04388	3.1	43.1	90.22	37.95	1.04629
5.2	43.9	87.19	38.06	1.04393	3.2	43.2	90.50	37.94	1.04650
5.3	43.9	87.26	38.06	1.04397	3.3	43.2	90.76	37.93	1.04670
5.4	44.0	87.32	38.05	1.04402	3.4	43.2	91.03	37.92	1.04690
5.5	44.0	87.38	38.05	1.04408	3.5	43.2	91.28	37.91	1.04710
5.6	44.0	87.45	38.05	1.04413	3.6	43.3	91.53	37.90	1.04729
5.7	44.1	87.52	38.05	1.04418	3.7	43.3	91.78	37.89	1.04748
5.8	44.1	87.59	38.04	1.04424	3.8	43.3	92.02	37.88	1.04766
5.9	44.1	87.66	38.04	1.04429	3.9	43.3	92.26	37.87	1.04784
6	44.2	87.73	38.04	1.04435					
6.1	44.2	87.80	38.04	1.04440					
6.2	44.2	87.87	38.03	1.04446					
6.3	44.2	87.95	38.03	1.04451					
6.4	44.3	88.02	38.03	1.04457					
6.5	44.3	88.09	38.03	1.04463					

HYSYS Model



MODULE 15

PROCESS DESIGN FOR INSTRUMENT AIR SYSTEMS

Industrial process facilities consist of a wide variety of pneumatically operated equipment which needs to be provided with a motive force for operation. Towards this, ambient air is one of the commonly used motive fluids to operate. In an Oil & Gas project, the primary step is to assess the number of elements that need instrument air (IA) and capacities of each element (e.g., Control valve) to determine the required instrument air system capacity. The following module focuses on sizing an IA air receiver vessel as well as some of the design considerations to made for an IA system.

Typical Layout & Operation of IA System

A typical layout of an Instrument Air system is shown below,

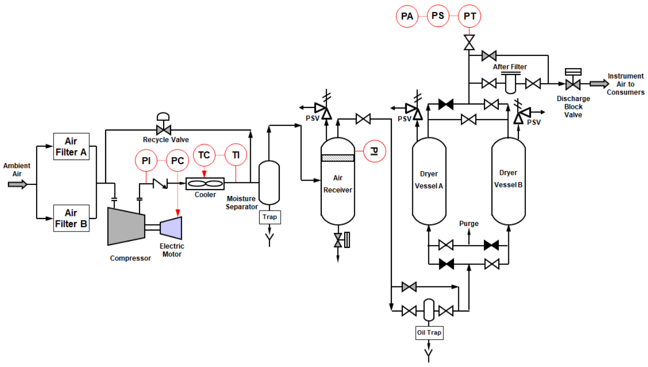


Figure 1. Schematic of Compression System

The main components of an instrument air system consist of an inlet air filter to decontaminate the atmospheric air of dust and debris, an air compressor to produce instrument air at the required pressure, a cooler to cool the hot air from the compressor discharge, a moisture separator to remove any condensates from the compressed air, an air receiver that stores the compressed air, a set of molecular sieve air dryers that act as a desiccant to dry the instrument air to the required dew point. Air dryers are operated in cycles whereby when one dryer is under operation, the other dryer is regenerated by removing water vapour using a pressure swing adsorption (PSA) process.

IA Receiver Design Considerations

A key component of the Instrument Air System is the Air Receiving Vessel. The IA Receiver when designed must consider the following factors,

1. ***Minimize Pressure Fluctuations*** – To meet IA consumers' demands and during emergency shutdown scenarios, it becomes a necessity to reduce pressure fluctuations. This also means that sufficient pressure at a steady rate must always be available for processes that use IA and is measured in units of time (minutes).
2. ***Short Term Air Demand*** – In process facilities often the demand for instrument air (IA) can fluctuate sometimes reaching a peak. This needs to be accounted for in the air compressor capacity estimates along with sufficient storage volume in the associated IA receiver to accommodate the peak demand IA flow rates.
3. ***Energy Savings*** – Instrument Air Systems run frequently consuming power and becomes imperative to achieve power savings by operating (loading/unloading) the compressor only as and when required. When the pressure in the IA Receiver drops below a threshold, the IA compressor is loaded to achieve the required pressure in the IA Receiver. Sizing the IA Receiver for longer cycles enables to cut own on power consumption while providing a steady flow of IA to the end users.

Instrument Air Quality Standards

A commonly used industry standard to set instrument air quality standards is the ANSI/ISA –S7.0.01-1996. As per the standard,

1. ***Pressure Dew Point*** – Pressure dew point is defined as the temperature at which free moisture condenses out from the instrument air into liquid water for a specific pressure. Pressure dew point at the air dryer outlet should be at least 10^0C (18^0F) below the minimum temperature to which any part of the IA system is exposed and also shall not exceed 4^0C (39^0F) at the line pressure.
2. ***Particle Size*** – The IA supplied to consumers is expected to contain particulate matter and for most pneumatically operated devices, a particulate size of 40μm is acceptable. In cases where, particulate size of < 40μm is required, additional air filtration modules can be installed to achieve < 40μm sized particulate matter.
3. ***Lubricant Content*** – In cases where the installed Air compressor in the IA system is lubricated with lube oil, there is always a risk of oil carry over along with the compressed air. This poses a threat to the pneumatic devices that receive the instrument air and affects their operation. Hence the lubricant content should be close to 0 ppm but cannot exceed 1 ppm w/w.
4. ***IA System Location*** – The location of the Instrument Air system is also important to prevent contaminants, hazardous and flammable gases from being drawn into the Inlet air filters.

Instrument Air System Design

To evaluate the process data of an instrument air unit, an example case study is used to explain. An Instrument air package is to be designed to deliver 600 ACFM of dry air at 8 barg to pneumatic device users. The ambient location is 20^0C, with a relative humidity (RH) of 60%. The barometric pressure is 1.01325 bara. The IA delivered to the IA receiver is required to be ~30^0C. The IA receiver is charged/discharged through a 10 sec cycle & the operating pressure band between lower and upper pressure of the IA receiver is 10 psi. From Steam Tables, moisture content in free air is as follows,

Table 1. Mass of H_2O in Air *(kg.H_2O/m^3 Free Sat. Air)*

Temperature	Pressure	
[^{o}C]	**0 barg**	**8 barg**
0	0.0045	0.00051
20	0.018	0.0019
40	0.059	0.0062
60	0.18	0.017
80	0.65	0.041

Water Condensation in IA Receiver

To estimate the amount of water condensing in the wet air IA Receiver, the mass of water in air at 100% RH is taken from Table 1.

1. Water Content in saturated air [100% RH] entering Comp. [0 barg] = 0.018 kg H_2O/m^3
2. Water Content in IA compressor suction at 60% RH = RH x Water Content at 100% RH = 0.6 x 0.018 = 0.0108 kg H_2O/m^3 Air
3. Water Content at IA Comp. Discharge [100% RH] at 8 barg, 30^0C = 0.00341 kg H_2O/m^3 Air [Table 1].
4. Water extracted from compressed air discharge & drained via IA receiver liquid outlet, 0.0108–0.00341=0.00739 kg H_2O/m^3
5. Water drain rate in IA Wet Receiver = 0.00739 × (600 x 0.0283168) × 60 × 3600 = 7.5 kg.H_2O/h (Note: 0.0283168 is conversion factor for ACFM to Am^3/s, i.e., [600× 0.0283168] / 60 = 0. 283168 Am^3/s.
6. Relative Humidity [RH] of Air Leaving the IA Receiver to Air Dryer = [0.00341/0.018] × 100 = 19%

Pressure Dew Point in IA Inlet & Receiver

The pressure dew point of the instrument air processed at the IA compressor inlet and IA receiver exit can be calculated using Arden-Buck equation as follows,

$$\gamma_m[T,RH] = \ln\left[\frac{RH}{100} e^{\left(b-\frac{T}{d}\right)\left(\frac{T}{c+T}\right)}\right] \quad (1)$$

$$T_{Dew\ Point} = \left[\frac{c \times \gamma_m[T,RH]}{b-\gamma_m[T,RH]}\right] \quad (2)$$

Where,

Constant 'b' = 18.678; Constant 'c' = 257.14 ^{0}C;

Constant 'd' = 234.50 ^{0}C;

RH = Relative Humidity

Therefore, the pressure dew point at IA compressor Inlet is,

$$\gamma_m[20^0C, 60\%] = \ln\left[\frac{60}{100} e^{\left(18.678-\frac{20}{234.5}\right)\left(\frac{20}{257.14+20}\right)}\right] \quad (3)$$

$$\gamma_m[20^0C, 60\%] = 0.831 \quad (4)$$

$$T_{Dew\ Point} = \left[\frac{257.14\times0.831}{18.678-0.831}\right] \approx 12^0C \quad (5)$$

The pressure dew point at the air leaving the IA Receiver is computed as,

$$\gamma_m[30^0C, 19\%] = \ln\left[\frac{19}{100} e^{\left(18.678-\frac{30}{234.5}\right)\left(\frac{30}{257.14+30}\right)}\right]$$
(6)

$$\gamma_m[30^0C, 19\%] = 0.274 \quad (7)$$

$$T_{Dew\ Point} = \left[\frac{257.14\times0.274}{18.678-0.274}\right] \approx 3.8^0C \quad (8)$$

Instrument Air Receiver Size

The Instrument Air Receiver which collects the compressed gas is sized based on the principle of excess pressure in the IA receiver volume in which the quantity of stored compressed air is above the facility's requirements. Using the actual volume flow rate flowing into the IA Receiver, the storage volume, taking into account the time cycle for charging/discharging, pressure band & barometric pressure, can be computed as,

$$V_{IA\ Receiver} = \left[\frac{Q_c\times f\times P_a}{(P_U-P_L)}\right] \quad (9)$$

Where,

Q_c = Instrument Air Capacity [ACFM]

f = Charge/discharge per IA receiver Cycle [s]

P_U–P_L = Pressure band of IA Receiver [psia]

P_a = Barometric Pressure at Location [psia]

The volume of the IA receiver is computed as,

$$V_{IA\ Receiver} = \left[\frac{600\times10\times14.7}{60\times10}\right] = 147\ ft^3$$
(10)

Or, $V_{IA\ Receiver} \approx 4.2\ m^3$ (11)

IA Dew Point at Atmospheric Pressure

The performance guarantee parameter for most industrial IA systems is based on a typical dew point requirement of -40^{0}C at atmospheric pressure at the outlet of the air dryer. In this module, no calculations are shown for air dryer unit, however taking the IA receiver process conditions, it can be estimated, what should be the air dryer's pressure dew point (i.e., at 8 barg) to achieve a performance guarantee dew point of -40^{0}C at 1 atm at the air dryer outlet.

To estimate the dew point at atmospheric pressure, the following dew point graph between atmospheric pressure & indicated pressure is used.

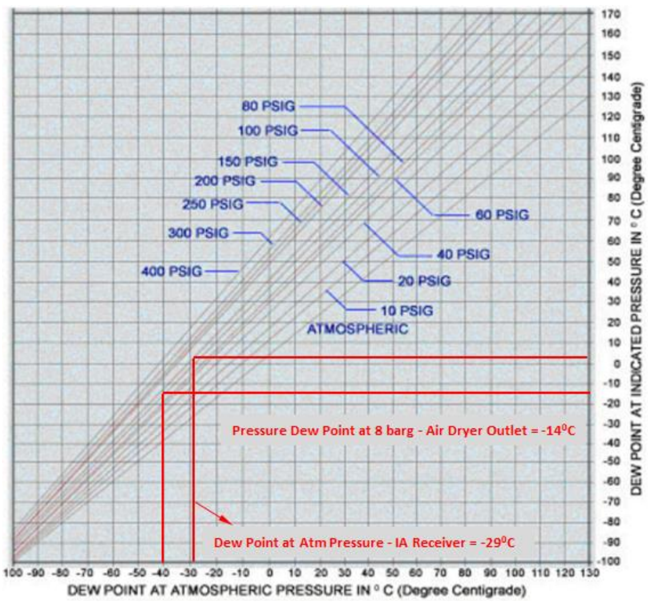


Figure 2. Air Dew Point Conversion Chart

From Fig. 2, for a pressure dew point of 3.8^{0}C at 8 barg in IA Receiver, the dew point at 1atm is -29^{0}C. Therefore, to achieve an atmospheric dew point of -40^{0}C at the air dryer outlet, the pressure dew point temperature should be -14^{0}C. To estimate the relative humidity [RH] at air dryer outlet for an atmospheric dew point of -40^{0}C, Eq. (1) and Eq. (2) can be re-arranged,

$$\gamma_m[T, RH] = \left[\frac{b \times T_{DP}}{c + T_{DP}}\right] \quad (12)$$

$$\mathrm{RH} = \left[\frac{e^{\gamma_m[T,RH]}}{e^{\left(b-\frac{T}{d}\right)\left(\frac{T}{c+T}\right)}}\right] \times 100 \quad (13)$$

Assuming the temperature rise in the air dryer is 40^{0}C, the RH of the IA exiting the air dryer is,

$$\gamma_m[T, RH] = \left[\frac{18.678 \times (-14)}{257.14 + (-14)}\right] = -1.075 \quad (14)$$

$$RH = \left[\frac{e^{-1.075}}{e^{\left(18.678-\frac{40}{234.5}\right)\left(\frac{40}{257.14+40}\right)}}\right] \times 100 = 2.8\% \quad (15)$$

Table 2. IA System Results Summary

Parameter	Value	Units
H_2O Extracted in IA Receiver	**0.0074**	kg.H_2O/m^3
H_2O Condensate in IA Receiver	**7.5**	kg.H_2O/h
Air RH leaving IA Receiver	**19**	%
$T_{Pressure\ Dew\ Point}$ - IA Comp. Inlet	**12.0**	^{0}C
$T_{Pressure\ Dew\ Point}$ - IA Receiver	**3.8**	^{0}C
$T_{Pressure\ Dew\ Point}$ – Air Dryer Exit	**-14.0**	^{0}C
Instrument Air Receiver Size	**4.16**	m^3

IA System Design Considerations

1. In process facilities, it is prudent to install IA systems with a 1W + 1S configuration. The standby can be diesel driven or steam driven, subjected to the utility available.
2. For most utility applications, nominal instrument air line pressure for the utility industry should be ~690 kPa (100 psi).
3. Since most industrial facilities operate with IA air at about 7 barg to 8 barg, the set pressure of the relief valve (RV) must be set higher accordingly but must not exceed the vessel design pressure.
4. IA distribution lines must be sized with line ΔP < 1 bar between the air dryer outlet and the farthest user of IA. Typically, a user can be taken to use 0.015 m^3 (0.5 Scf) of air/min.
5. Air dryer regeneration methods are of two types – Air purge regeneration & Heater regeneration. Air purge regeneration is a commonly used method where the packed column of molecular sieves is dried by diverting a fraction of the dry air from the active air dryer vessel enabling adsorption of the moisture and expelling via a purge line. Whereas in heater regeneration methods, a heater-blower setup is installed in the regeneration line that heats ambient air & routes the heated air through the regenerating dryer. The hot air heats the regenerated dryer till the moisture reaches boiling point and is subsequently expelled through the purge line.
6. The regeneration time of each air dryer shall not be more than 6 hours as per IPS–G-IN-200(2). The recommended cycle time between regeneration cycles for normal operation is 6 hours for regeneration and 2 hours of standby. The maximum allowable cycle time between regeneration cycles is 6 hours for regeneration and 4 hours standby. Hence the air dryers must be designed to be capable of drying for at least 10 hours without increase in dew point.
7. The air dryer adsorption operation is exothermic & causes the dried air to reach as high as 60^{0}C. If its temperature is not expected to cool to ~40^{0}C, additional after-coolers would be required at the air dryer outlet.
8. After coolers can be air-cooled type or water-cooled type. Water-cooled aftercoolers are usually sized to cool outlet air to within ~5°C to 8°C of the inlet cooling water temperature. Whereas Air-

cooled aftercoolers are usually sized to cool outlet air to within 14°C to 17°C of the ambient air temperature

9. Compression increases the partial pressure of the water vapour present. If the water vapour partial pressure is increased to the saturation water vapour pressure, condensation occurs. If the saturation water vapour pressure is reduced to the partial pressure of the water vapour present, water or ice will result. Therefore, moisture removal is a major consideration of instrument air treatment systems. Water droplets entrained in the air can initiate the formation of rust or other corrosion products which block internal passageways of electric to pneumatic converters resulting in sticking and/or binding of moving parts. Water droplets can also obstruct the discharge ports on solenoid air pilot valves thus reducing their ability to function properly. Therefore, an automatic drain (e.g., timer drain, float drain or an electronic drain) with a manual bypass should be located near the bottom of the air receiver to dispose of the condensate.

10. In cold climates, water extracted from the atmospheric air accumulates at the low points in the IA system. Hence, in such cold climates, insulation and steam tracing should be provided to both piping as well as up to a sufficient height from the bottom portion of the air receiver.

11. For the design and construction of the vessels, ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1 or any other approved standard of equivalent authority is acceptable. For design, fabrication, erection and testing of piping ASME B16.5 and ASME B31.3 are acceptable.

12. In IA distribution systems, commonly used types of valves are globe valves, gate valves & ball valves. Globe valves provide the advantage of regulating system flow rates & provide tight shut-off. On the down side, they cause reduced flow rates, increased pressure loss, and allow places for particulates to collect causing valve leakages. Gate valves & Globe valves are used for on/Off isolation & provide full, line-size port for air flow with minimal pressure drop and are conducive to internal cleaning. The disadvantages of gate valves are that they allow particulates to collect in disc guides, and valve discs can separate from their stems. The disadvantage of ball valves is that they are more expensive than comparably-sized globe or gate valves, and their sealing surfaces are susceptible to leakage from particulate scoring.

References & Further Reading

1. ANSI/ISA-7.0.01-1996, "*Quality Standard for Instrument Air*", 12th Nov 1996

2. IPS-G-IN-200(2), "*General Standard for Instrument Air Systems*", 2nd Ed, Feb 2016

3. Engineering ToolBox, (2008), Compressed Air and Water Content

4. https://www.omnicalculator.com/physics/air-density

5. http://www.nationalcompressedair.com/dat/files/600.pdf

ANNEXURE A: Saturation Vapour Pressure at Dew Point & Actual Vapour Pressure

The saturation vapour pressure of air at its dew point can be calculated as,

$$P_1 = 6.1078 \times 10^{\left[\frac{7.5 \times T_{Dew\ Point} \times 14.7}{T_{Dew\ Point} + 237.3}\right]} \quad (16)$$

Actual vapour pressure is, $P_v = P_1 \times RH$ (17)

For air leaving IA receiver at T_{DP} 3.8^{0}C, 19%RH

$$P_1 = 6.1078 \times 10^{\left[\frac{7.5 \times 3.8 \times 14.7}{3.8 + 237.3}\right]} = 8\ hPa \quad (18)$$

$$P_v = 8 \times 0.19 = 1.59\ hPa \quad (19)$$

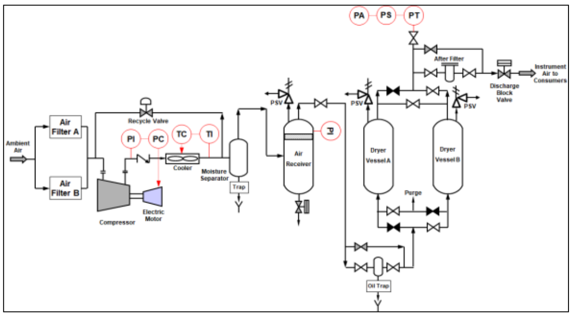
ANNEXURE B: MS Excel Calculation Sheet

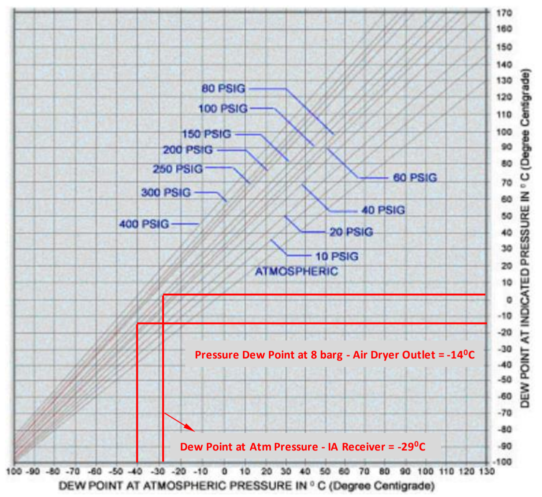
Plant Instrument Air Receiver - Process Design Tool

Standard Pressure [P_S]	1.01325	[bara]
Standard Temperature [T_S]	273.15	[K]
Ambient Pressure [P_1]	0	[barg]
IA Compressor Suction Temperature [T_1]	20	[^{0}C]
Ambient Air Relative Humidity [RH]	60	[%]
IA Compressor Discharge Pressure [Saturated] [P_2]	8	[barg]
IA Compressor Cooler Temperature [Saturated] [T_2]	30	[^{0}C]
Compressor Flow Capacity [Q_C]	4973	[SCFM]
	600	[ACFM]
	0.2832	[Am^3/s]
Water Content at 100% RH [Atmospheric Conditions]	0.0180	[kg.H_2O/m^3 free air]
Water Content in IA Compressor Suction [60% RH]	0.0108	[kg.H_2O/m^3 free air]
Water Content in IA Compressor Discharge [100% RH]	0.0034	[kg.H_2O/m^3 free air]
Water Extracted inIA Receiver & Sent to Drain	**0.0074**	[kg.H_2O/m^3 free air]
	0.690	[g.H_2O/kg free air]
Water in Air Leaving the IA Receiver	**0.318**	[g.H_2O/kg free air]
Water Drain Rate in Wet Air IA Receiver	**7.5**	[kg.H_2O/h]
Air Relative Humidity [RH] - IA Receiver Exit to Air Dryer	**19**	[%]
Dew Point Calculation [Arden-Buck Method]		
Constant 'a'	6.1121	[mbar]
Constant 'b'	18.678	[-]
Constant 'c'	257.14	[^{0}C]
Constant 'd'	234.5	[^{0}C]
γ(T, RH) at IA Compressor Inlet	0.8309	[-]
Pressure Dew Point [$T_{Pressure\ Dew\ point}$] at IA Compressor Inlet	**12.0**	[^{0}C]
γ(T, RH) at IA Receiver Exit	0.2741	[-]
Pressure Dew Point [$T_{Pressure\ Dew\ point}$] at IA Receiver Exit	**3.8**	[^{0}C]
Instrument Air Receiver Size		
IA Compressor Capacity [Q_C]	**600**	[ACFM]
Charge/Discharge per IA Receiver Cycle [f]	10	[sec]
Pressure Band of IA Receiver [P_U-P_L]	10	[psi]
Instrument Air (IA) Receiver Size [$V_{Receiver}$]	**147**	[ft^3]
	4.16	[m^3]
Air Dryer Process Design		
Required Dew Point at 1 atm Pressure at Air Dryer Outlet	-40.0	^{0}C
Temperature Rise in Air Dryer	40.0	[^{0}C]
Pressure DP required at Air Dryer Outlet [8 barg] for -40C Atm DP [DP Conv. Graph]	-14.0	[^{0}C]
γ(T, RH) at IA Air Dryer Outlet	-1.075	[-]
Relative Humidity [RH] Required at Air Dryer Outlet for -40C Atm DP	**2.82**	[%]
SUMMARY		
Water Extracted & Sent to IA Receiver Drain	**0.00739**	[kg H_2O/m^3]
Water Drain Rate in Wet Air IA Receiver	**7.5**	[kg H_2O/h]
$T_{Pressure\ Dew\ point}$ at IA Comp. Inlet	**12.0**	[^{0}C]
$T_{Pressure\ Dew\ point}$ at IA Receiver Exit	**3.8**	[^{0}C]
$T_{Pressure\ Dew\ point}$ Required at Air Dryer Outlet	**-14.0**	[^{0}C]
Instrument Air (IA) Receiver Size [$V_{Receiver}$]	**4.16**	[m^3]

Mass of Water in Air (kg.H_2O/m^3 free saturated air)

Temperature	Pressure (barg)				
(^{o}C)	0	2	4	6	8
0	0.0045	0.0015	0.00091	0.00065	0.00051
20	0.018	0.0058	0.0035	0.0025	0.0019
40	0.059	0.019	0.011	0.0079	0.0062
60	0.18	0.053	0.031	0.022	0.017
80	0.65	0.14	0.078	0.054	0.041
100	-	0.38	0.19	0.13	0.094
120	-	-	0.49	0.29	0.21





Density of Air [Based on RH]

Saturation Vapour Pressure at Dew point [p_1]	8.0	[hPa]
Actual Vapour Pressure [p_v]	1.52	[hPa]
Pressure of Dry Air [p_d] at IA Receiver Exit	9012	[hPa]
Density of Air [ρ_{Air}] at IA Receiver Exit	**10.71**	[kg/m^3]

Module 16

Understanding High Integrity Pressure Protection Systems

No chemical process facility is immune to the risk of overpressure to avoid dictating the necessity for overpressure protection. For every situation that demands safe containment of process gas, it becomes an obligation for engineers to equally provide pressure relieving and flaring provisions wherever necessary. The levels of protection are hierarchical, starting with designing an inherently safe process to avoid overpressure followed by providing alarms for operators to intervene and Emergency Shutdown provisions through ESD and SIL rated instrumentation. Beyond these design and instrument based protection measures, the philosophy of containment and abatement steps such as pressure relieving devices, flares, physical dikes and Emergency Response Services is employed.

High Integrity Pressure Protection Systems (HIPPS) are related to the third layer of protection whereby process shutdown can be initiated by shutdown valves that receive instructions from a logic solver which in turn are fed by pressure transmitters.

In the oil and gas industry, process facilities are often subjected to erratic fluctuations in wellhead pressure and flow trends. Such process systems in recent years are tended to for overpressure protection with the installation of HIPPS. HIPPS aid in shutting down instead of having to flare sour gas through pressure relieving devices that are subsequently routed to a flare system.

The following covers key guidelines and requirements for HIPPS from industry experience and, standards.

With world gas demand increasing steadily over the years, High Pressure High Temperature (HPHT) environments are also increasingly becoming common. Standard design methods involve designing the entire well head to export systems to fully rated conditions (1500#, 900#, etc.) depending on the operating pressures and temperatures. However such methods would unnecessarily increase project costs and affect installation foot print depending on how flammable or toxic is the process fluid, sometimes to the point of not giving any viable cost benefits.

To attend to such unviable scenarios, the concept of de-rating Non-HPHT equipment in downstream operations with overpressure protection can be employed. For these purposes, HIPPS is treated as a Safety Instrumented System (SIS) that is based on a Safety Integrity Level (SIL). From an SIL perspective, HIPPS follows a minimum of SIL 3 rating where the Average Probability of Failure on demand is of the order between $\geq 10^{-4}$ to $< 10^{-3}$. It must be noted that HIPPS is an SIS that aids more as risk reduction for prevention measure rather than a risk mitigation measure. The typical architecture of HIPPS is shown in the Figure below.

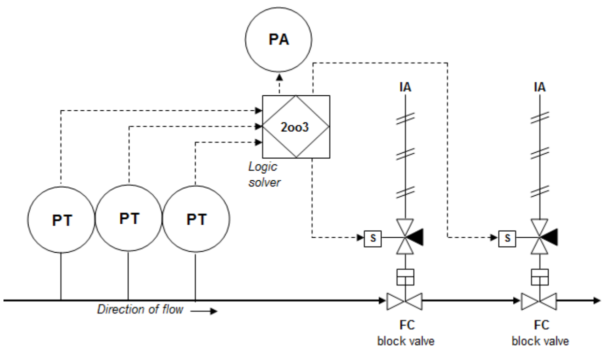


Figure 1. Example of HIPPS Architecture

HIPPS Operating Philosophy

A typical HIPPS architecture consists of three (3) pressure transmitters (PT) that constantly record the line pressure which are fed to a logic solver. In the event of an overpressure, the logic solver initiates a shutdown operation of two (2) consecutive Fail-close (FC) valves which are installed on the same line thereby shutting down fluid flow. A pressure alarm (PA) serves the purpose of informing the operation personnel. The purpose of installing the said number of transmitters and valves are as follows,

1. To avoid compromising the HIPPS functionality due to failure of any one shutdown valve (SDV), a second valve is added to provide higher redundancy. Both valves are operated on a 1oo2 voting philosophy that decides which Fail-Close (FC) valve closes.
2. To avoid receiving a premature or false signal from the pressure transmitter, a 2oo3 voting philosophy is employed as against a 1oo3 voting philosophy. This means that unless 2 pressure transmitters concur that there is an overpressure scenario, HIPPS is not activated.

HIPPS Valve Selection

HIPPS Valves can be operated hydraulically or by solenoid methods. The two (2) types of valves used are either ball type or butterfly type. Ball valves provide the best shutoff conditions and can range from 2 inch to 56 inch depending on the manufacturer. Whereas butterfly valves can be provided from 2 inch to 100 inch, again depending on the manufacturer. For HPHT applications, the piping class can vary from as high as 2500# which can be provided in the ball class range with material ranging from carbon steel, stainless steel, duplex as well as special alloys. The typical stroke time for HIPPS valves should be of the order of <2 sec. Valve selection must also consider that HPHT applications can witness temperatures as high as 500^0C. HIPPS valves must also be able to cater to Partial Stroke Testing capability (PST), Tight Shut-off (TSO) (e.g., Class V or Class VI of ANSI FCI 70-2), Fast acting, Fire Safety tested to for example, API 607. Environmental constraints must also be met for fugitive emissions such as ISO 15848-1 standards.

HIPPS Engineering Standards

HIPPS can cater to many applications such as offshore/onshore well heads, flare headers and chemical process industries. ASME Section VIII, UG-140 (Overpressure Protection Systems) provides a range of applications for which HIPPS can be used, such as,

1. High Propagation Chemical reactions resulting in loss of containment prior to the relief device opening or processes that yield impractical large vent areas
2. Runaway Polymerization, Exothermic or Reactive reactions that produce large vapour rates rendering relief devices insufficient to cater to over-pressurization scenarios.

To keep the module brief, the focus is made on Oil and Gas applications. HIPPS for the Oil and Gas industry are based on two aspects – prescriptive and performance based. Standards such as API, ASME, ANSI to suggest a few are for design, manufacture and implementation and examples are API 14C (Recommended Practice for Analysis, Design, Installation, and Testing of Basic Surface Safety Systems for Offshore Production Platforms), API 6A (Specification for Wellhead and Christmas Tree Equipment) for offshore applications, API 520/521, API 17O (Subsea High Integrity Pressure Protection Systems – HIPPS) to name a few. The other aspect is the IEC standards, chiefly IEC 61508 supplemented by IEC 61511 which are more of performance-based standards that describe how to arrive at a solution rather than prescribing a solution. This would leave room for elucidation between different operators, contractors and suppliers thereby resulting in lack of commonly accepted industry specifications. The IEC 61508, for example focuses much on the functioning of the logic solver and touching minimally on the final control element. These gaps left in IEC 61508 regarding final control elements such as valves and solenoids are covered in IEC 61511.

IEC provides SIL ratings with Probability of Failure on Demand (PFD) and respective architecture not for individual components, but for the system as a whole which must include the actuators, initiators, final control elements and logic solvers. When different manufacturers assume certain architecture for HIPPS components provided, the individual components Probability of Failure on Demand (PFD) would not necessarily represent the overall system's PFD which is used to define the SIL rating. Therefore the PFD for a SIL assessment needs to always be investigated on a case to case basis prior to understanding the limitation on the SIL rating arrived at.

Pressure Relieving Devices vs. HIPPS

A point of contention arises when one asks, if when a piece of equipment is equipped with multiple relieving devices to deal with overpressure scenarios, wherefrom arises the necessity to install a HIPPS. To suggest so, means a justification is required to install HIPPS. For any successful implementation of HIPPS, an examination of applicable regulations, standards, local codes and insurer's requirements that may mandate the need for relieving devices is required. This is to be followed up by a Hazard Analysis (HAZAN) by a multi-disciplinary team. The process risk needs to be evaluated based on frequency and consequence such that the HIPPS proposed can demonstrate that the mitigated risk is lower than the risk tolerance criteria, to allow for the removal of associated relief devices from flare load calculations.

Traditionally, pressure vessels are equipped with pressure relieving devices that are routed to an industrial flare. However when the flare load capacity is insufficient to deal with excess capacities, HIPPS offers the advantage of risk reduction, for example, shutting down well heads in oil and gas applications. But this also obliges the engineer to ask, if when a HIPPS system is installed, does it always necessitate pressure relieving devices to be installed as well.

Pressure vessels that operate above atmospheric conditions of 15 psig are designed as per Code ASME Section VIII Division 1 and to cover matters of overpressure protection, UG-125 to UG 140 of the said code provides basic requirements. The ASME UG-140 requirements and procedures are commonly known as Code Case 2211.

The requirements of a relief device covered by UG-125 to UG-138 are to be designed as per API 521. For cases where the requirement of a relief device can be overcome is based on UG-140(a) and UG-140 (b) of the said code which pertains to Inherently Safe Design and HIPPS based design under specific cases respectively. Industrial use of HIPPS certainly provides the option of installing a smaller sized relieving device but cannot eliminate the necessity of relieving devices, although in certain specific cases, the need for PRV's can be eliminated.

As per API 521 and Code Case 2211 of ASME Section VIII, Division 1 and 2, HIPPS is allowed in lieu of a Pressure relieving device provided HIPPS meets or exceeds the protection that would have been provided by the PRV. However as per, ASME Section VIII, Division 1, para UG-125(a) Section VIII, Division 2, para, AR-100, it is required to install a pressure relieving device on all pressure vessels.

Therefore, the question of whether a PRV is necessary in tandem with HIPPS depends on identifying credible overpressure scenarios in the operating system prior to installing relieving devices. HIPPS typically can be found in applications where hazardous gases are part of critical operations. Any addition of a relieving device acts more like insurance to the safety of the process.

HIPPS Procurement Life Cycle

HIPPS system which consists of various components such as logic solvers, actuators, valves, pressure transmitters can be supplied by various manufacturers. A Request for Quotation (RFQ) is placed with different manufacturers by the procurement division of the EPC contractor which in turn is provided to the engineering teams such as process, piping, Instrumentation and Safety departments.

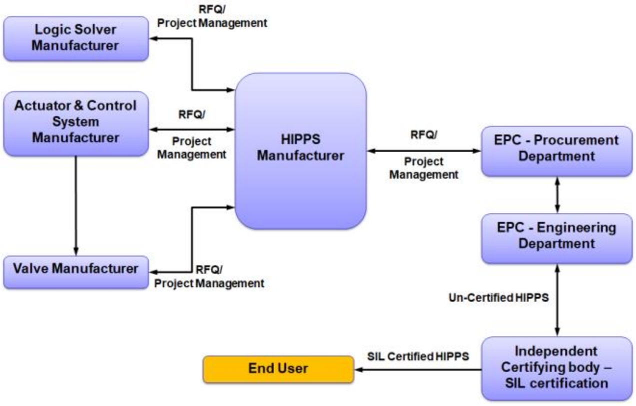


Figure 2. HIPPS Procurement and Certification Life Cycle

The EPC procurement team has to perform an additional task of project management to ensure all associated items in the bill of quantities (BOQ) are received and handed over to the engineering team. The integrated HIPPS components would then require a SIL certification by an independent certifying body for SIL 3 requirements before being implemented at the End User's facility.

The disadvantage of employing multiple suppliers causes increased lead time as well as procurement costs. An alternative would be to source HIPPS from a single manufacturer who can provide all individual components and have it certified by an independent SIL certifying body. This reduces the lead time required for procurement as well as costs associated.

References & Further Reading

1. '*Introduction to Process Unit Flares*', Kolmetz. K, KLM Technology Group
2. '*Application of UG-140 for Overpressure Protection*', Sushant G Labhasetwar, Chemical Engineering World, September 2013
3. '*Consideration in Designing HIPPS*', Willem-Jan Nuis, Rens Wolters, www.safan.com
4. *'High Integrity Protection System (HIPS) for Flare Load Mitigation'*, Angela E. Summers, SIS-TECH Solutions, LP

Module 17

Pressure Safety Valve (PSV) Sizing - API 520/521/526

No chemical process facility is immune to the risk of overpressure to avoid dictating the necessity for overpressure protection. For every situation that demands safe containment of process gas, it becomes an obligation for engineers to equally provide pressure relieving and flaring provisions wherever necessary. The levels of protection are hierarchical, starting with designing an inherently safe process to avoid overpressure followed by providing alarms for operators to intervene and Emergency Shutdown provisions through ESD and SIL rated instrumentation. Beyond this design and instrument based protection measures, the philosophy of containment and abatement steps such as pressure relieving devices, flares, physical dikes and Emergency Response Services is employed

A pressure safety valve (PSV) is a safety device used to protect equipment from over pressure conditions. Over pressure refers to any condition which would cause a system to increase beyond the specified design pressure or maximum allowable working pressure (MAWP). PSVs must open at a predetermined set pressure, flow a rated capacity at a specified overpressure, and close when the system pressure has returned to a safe level.

Types of Pressure Safety Valves

Pressure safety valves can be chiefly classified into the following types,

1. ***Conventional Safety Relief Valve*** – In a conventional safety relief valve it has a spring housing that vents fluids to the discharge side of the PSV. The operational characteristics (opening pressure, closing pressure, & relieving capacity) are directly affected by changes in PSV back pressure.
2. ***Balanced Bellow Safety Relief Valve*** - A balanced safety relief valve provides a set of bellows to reduce the effect of back pressure on the operational characteristics.
3. ***Pilot Operated Safety Relief Valve*** - In a pilot operated safety relief valve the major relieving device has a self-actuated auxiliary pressure relief valve to control the relieving conditions.
4. ***Power Actuated Safety Relief Valve*** - In a power actuated safety relief valve, the major relieving device is controlled with an external source of energy.
5. ***Temperature Actuated Safety Relief Valve***-
   In a temperature-actuated safety relief valve, the actuation takes place by external or internal temperature or by inlet side pressure.
6. **Pressure Vacuum Safety Relief Valve** - A vacuum relief valve is designed to allow fluid to prevent excessive internal vacuum & close to prevent further fluid flow after normal conditions have been restored.

Pressure Safety Valves Terminology

The terminology associated with pressure safety relief Valve can be inferred from API Recommended Practice 520 as,

1. ***Set Pressure*** - The inlet gauge pressure at which the pressure relief device is set to open under service conditions.
2. ***Back Pressure*** - The pressure that exists at the pressure relief device outlet as a result of the pressure in the discharge system. It is the sum of the superimposed and built-up back pressures.
3. ***Built up Back Pressure*** – This is the increase in pressure at the outlet of a pressure relief device that develops as a

result of flow after the pressure relief device opens.

4. ***Superimposed Back Pressure*** **–** The static pressure that exists at the outlet of a pressure relief device at the time the device is required to operate. It is the result of pressure in the discharge system coming from other sources and may be constant or variable.
5. ***Opening Pressure*** – This is the value of increasing inlet static pressure at which there is a measurable lift of the disc or at which discharge of the fluid becomes continuous, as determined by seeing, feeling or hearing.
6. ***Closing Pressure*** **–** The value of decreasing inlet static pressure at which the valve disc re-establishes contact with the seat or at which lift becomes zero as determined by seeing, feeling or hearing.
7. ***Actual Discharge Area*** **–** The minimum net area determining the flow through a valve
8. ***Effective Discharge Area*** – This is the nominal or computed area used with an effective discharge coefficient to calculate the minimum required relieving capacity for a pressure relief valve per the preliminary sizing equations contained in API 520. API 526 provides effective discharge areas for a range of sizes in terms of letter designations, D through T.
9. ***Inlet Size*** – The nominal pipe size (NPS) of the valve at inlet connection, unless otherwise designated.
10. ***Outlet Size*** – The nominal pipe size (NPS) of the valve at discharge connection, unless otherwise designated.
11. ***Effective Coefficient of Discharge*** – The effective coefficient of discharge is a nominal value used with an effective discharge area to calculate the minimum required relieving capacity of a pressure relief valve per the preliminary sizing equations of API 520.
12. ***Rated Coefficient of Discharge*** **–** The rated coefficient of discharge is determined in accordance with the applicable code or regulation and is used with the actual discharge area to calculate the rated flow capacity of a pressure relief valve.

PERMITTED SIZES OF PSV –API 526

As per API 526, the size of pressure safety valves are designated by their alphabetical designation. API 526 gives a list of permissible sizes of PSVs which are described below,

Table 1. Standard Orifices Sizes [API 526]

Sr. No.	Orifice Designation	in^2
1	D	0.110
2	E	0.196
3	F	0.307
4	G	0.503
5	H	0.785
6	J	1.287
7	K	1.838
8	L	2.853
9	M	3.600
10	N	4.340
11	P	6.380
12	Q	11.05
13	R	16.00
14	T	26.00

OVER PRESSURE SCENARIOS

Sizing a pressure relief valve begins with identifying the applicable credible scenario which determines the relieving capacity. Based on the relieving capacity, API provides procedures to estimate the required relieving area followed by choosing standardized sizes of relief devices from API 526. The following module covers the below described scenarios which are not comprehensive but represent a set of commonly encountered scenarios.

1. ***Blocked Liquid Discharge Case*** - This refers to closure of a valve on the outlet of equipment. With continuing liquid flow into the equipment and no provision to drain the liquid, fluid accumulates to building pressure to as high as the design pressure of the upstream equipment. In addition to this, static head of the liquid in the upstream equipment also contribute to the build-up of pressure. Therefore, the minimum relieving rate to be considered is the normal operating inlet flow.
2. ***Blocked Gas Outlet [Non-Fire Case]*** – Similar to the above case of liquid filled, gas accumulation in the vessel also contributes to the rise in pressure when the gas side valve fails to function by staying closed. With pressure continuing to rise, a relief device is required to relieve the equipment of the excess pressure.
3. ***Gas Control Valve Fail Open*** – This case refers to a scenario where when a control valve placed between equipment fails open, [whereby the upstream equipment has a higher design pressure and the downstream equipment is at a lower design pressure] causes over pressurization.
4. ***Thermal Expansion*** – This case refers to scenarios where liquid locked inside liquid lines. With exposure from sunlight, heat ingress occurs through the piping causing a temperature rise to vaporize the liquid resulting in over pressure. The quantity of fluid required to be relieved in temperature safety valves may be very small & therefore for thermal expansion cases the safety valve size of NPS ¾" x NPS 1" (DN 20 x DN25) should be sufficient as per API 521, [Ref 4].
5. ***Fire Case [Liquid Filled Vessel]*** – All equipment in a process facility is prone to exposure to fire due to equipment failure or man-made errors. This can result in the contents of the equipment fluid, expanding and vaporizing to create an over pressure scenario. Fire cases are of two types – Gas Filled Vessel and Liquid Filled Vessel. In the case of liquid filled vessels, the vessel is expected to contain a certain amount of liquid that wets the lower part surface of the vessel through which the liquid transfers latent heat causing liquid expansion and vaporization.

CASE STUDIES

To demonstrate the sizing of pressure safety valves for the described scenarios as per API recommended procedures, the following examples are shown.

Blocked Liquid Discharge Case

Consider a vessel relieving hydrocarbon at 300,000 kg/h which has a relief valve pressure set at 18 barg. Considering a non-fire case, the over pressure is taken to be 10%. For preliminary sizing, the back pressure at the relief valve discharge is considered to vary between 0 barg to 4 barg. A rupture disc exists and the back pressure is considered to be a variable for which a balanced bellow type of relief valve is recommended followed by the pressure relief valve requiring capacity certification as per ASME Sec VIII, Division I. The hydrocarbon fluid properties are as follows,

Table 2. Fluid Properties – Blocked Liquid Outlet

Parameter	Value	Units
Liquid Specific Gravity	0.85	[-]
Liquid Viscosity	450	cP

As per Ref [1], Sec 3.8.1.2, the initial orifice size of the PSV is sized as,

$$A_R = \left(\frac{11.78 \times Q}{K_d \times K_w \times K_c \times K_v}\right) \times \sqrt{\left(\frac{SG}{P_1 - P_2}\right)} \quad (1)$$

Where, Q = flow rate, lit/min

A_R = Required effective discharge area, mm^2

K_d = rated coefficient of discharge that should be obtained from the valve manufacturer. For a preliminary sizing, an effective discharge coefficient can be used as follows: 0.65 when PSV is installed with or without a rupture disk in combination & 0.62 when PSV is not installed and sizing is for a rupture disk in accordance with 3.11.1.2 Ref [1].

K_w = correction factor due to back pressure. If the back pressure is atmospheric, use a value for K_w of 1.0. Balanced bellows valves in back pressure service will require the correction factor determined from Figure 31 of Ref [1]. Conventional and pilot operated valves require no special correction.

K_c = combination correction factor for installations with a rupture disk upstream of the pressure relief valve. K_c value = 1.0 when a rupture disk is not installed & 0.9 when a rupture disk is installed in combination with a pressure relief valve and the combination does not have a published value.

K_v = correction factor due to viscosity. For conventional or pilot operated relief valve K_v can be taken as 1.0. For balanced bellows as determined from Figure 36 of Ref [1] or from the following equation:

$$K_V = \left(0.9935 + \frac{2.878}{R^{0.5}} + \frac{342.75}{R^{1.5}}\right)^{-1} \quad (2)$$

SG = Liquid specific gravity at flowing temperature referred to water at standard conditions. R = Reynolds Number as,

$$R = \frac{Q \times 18800 \times SG}{\eta \times \sqrt{A_R}} \quad (3)$$

P_1 = upstream relieving pressure, kPag (Set pressure plus allowable overpressure)

P_2 = back pressure, kPag

The orifice area based on corrected viscosity is then calculated as,

$$A_{corr} = \left(\frac{A_R}{K_V}\right) \quad (4)$$

Applying the above to estimate the pressure relief device, the relief flow rate is,

$$Q = \left(\frac{m}{\rho}\right) \times \left(\frac{1000}{60}\right) lit/min \quad (5)$$

$$Q = \left(\frac{300{,}000}{850 \times 60}\right) = 5{,}882\ lit/min \quad (6)$$

$$P_1 = P_{set} \times 1.1 = 19.8\ barg = 1980\ kPag \quad (7)$$

$$P_2 = 4\ barg = 400\ kPag \quad (8)$$

The percent of gauge back pressure is,

$$\%P_{Back} = \left(\frac{P_B}{P_S}\right) \times 100 \quad (9)$$

$$\%P_{Back} = \left(\frac{4 \times 14.7}{18 \times 14.7}\right) \times 100 = 22.2\% \quad (10)$$

Since a rupture disc exists and back pressure is considered to be a variable, a balanced bellow type of relief valve is recommended. Hence the coefficient of discharge [K_d] is 0.65.

The combination correction factor for use of rupture disc [K_c] in combination with a relief valve and in the absence of any published value is 0.9. The correction factor due to back pressure [K_w] for balanced bellows is determined from Fig 31 of Ref [1] as 0.955.

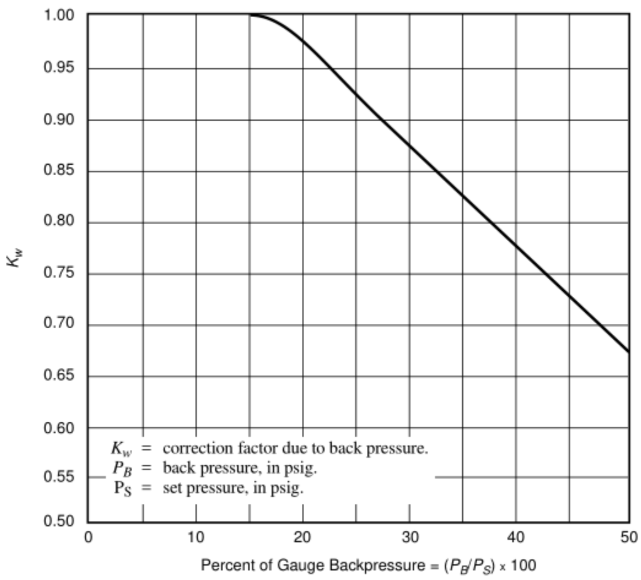


Figure 1. Capacity Correction Factor, K_w, due to Back Pressure on Balanced-Bellows Pressure Relief Valves in Liquid Service

It is to be noted that, as per Ref [1], Sec 3.3.4.4, the curve above represents values recommended by various manufacturers. This curve may be used when the manufacturer is not known. Otherwise, the manufacturer should be consulted for the applicable correction factor. Therefore, the initial orifice area sizing with no viscosity correction [K_v], i.e., $K_v = 1.0$ is,

$$A_R = \left(\frac{11.78 \times 5882}{0.65 \times 0.955 \times 0.9 \times 1}\right) \times \sqrt{\frac{0.85}{1980-400}} \quad (11)$$

$$A_R = 2877\ mm^2 \approx 4.46\ in^2 \quad (12)$$

The Reynolds number of the relieving fluid is,

$$R = \frac{5882 \times 18800 \times 0.85}{450 \times \sqrt{2877}} = 389 \quad (13)$$

The correction factor due to viscosity [K_v] is,

$$K_V = \left(0.9935 + \left(\frac{2.878}{3895^{0.5}}\right) + \left(\frac{342.75}{3895^{1.5}}\right)\right)^{-1} \quad (14)$$

$$K_V = 0.961 \quad (15)$$

The orifice area based on corrected viscosity,

$$A_{corr} = \left(\frac{4.46}{0.961}\right) = 4.64\ in^2 \quad (16)$$

From Table 1, the PSV chosen for a calculated orifice area of 4.64 in^2 is a '**P**' designated orifice which has an orifice area of 6.38 in^2.

Blocked Gas Outlet – Non-Fire Case

Consider a vessel relieving hydrocarbon vapours at 10,000 kg/h which has a relief valve pressure set at 8 barg relieving at a temperature of 42^{0}C. Considering a non-fire case, the over pressure is taken to be 10%. For preliminary sizing, the back pressure at the relief valve discharge is considered to be fairly constant at 4 barg. A rupture disc is installed upstream of the relief valve and conventional type of relief valve is recommended.

The pressure relief valve requires capacity certification as per ASME Sec VIII, Division I. The fluid properties of the hydrocarbon is shown below,

Table 3. Fluid Properties – Blocked Gas Outlet

Parameter	Value	Units
Specific Heat Ratio [k]	1.2	[-]
Compressibility Factor [Z]	0.9428	[-]
Gas Molecular Weight [MW]	33	[lb_m/lb_{mol}]

As per Ref [1], Sec 3.6.2.1 and Sec 3.6.3.1, required effective discharge area of a conventional PSV for critical flow is,

$$A_{calc} = \frac{W}{C \times K_d \times P_1 \times K_b \times K_c}\sqrt{\frac{TZ}{MW}} \quad (17)$$

Where,

A_R = Required effective discharge area, mm^2

Q = flow rate, lit/min

C = Coefficient determined from an expression of the ratio of the specific heats ($k = C_p / C_v$) of the gas or vapour at inlet relieving conditions. Where k cannot be determined, it is suggested that a value of C equal to 315 be used. The units for C are

$$C = \frac{\sqrt{lb_m \times lb_{mol} \times Rankine}}{lb_f \times hr} \quad (18)$$

K_d = Effective coefficient of discharge. For preliminary sizing, the following values can be used as follows: 0.975 when PSV is installed with or without a rupture disk in combination & 0.62 when PSV is not installed and sizing is for a rupture disk in accordance with 3.11.1.2 Ref [1].

P_1 = upstream relieving pressure, kPaa (Set pressure plus allowable overpressure plus atmospheric pressure)

K_b = Capacity correction factor due to back pressure. This can be obtained from the manufacturer's literature or estimated for preliminary sizing from Figure 30 of Ref [1]. The back pressure correction factor applies to balanced bellows valves only. For conventional and pilot operated valves, use a value for K_b equal to 1.0.

K_c = combination correction factor for installations with a rupture disk upstream of the pressure relief valve. K_c value = 1.0 when a rupture disk is not installed & 0.9 when a rupture disk is installed in combination with a pressure relief valve and the combination does not have a published value.

T = Relieving temperature of the inlet gas or vapour, R (°F + 460) [K (°C + 273)]

Z = Compressibility factor for the deviation of the actual gas from a perfect gas, a ratio evaluated at inlet relieving conditions.

MW = Molecular weight of the gas or vapour at inlet relieving conditions

V = Required flow through device, scfm at 14.7 psia, 60°F [Nm^3/min at 0°C, 101.325 kPaa]

G = Gas Specific gravity at standard conditions referred to air at standard conditions [normal conditions]. In other words, G = 1.00 for air at 14.7 psia and 60°F [101.325 kPaa and 0°C]

The critical flow pressure ratio is,

$$P_{cfr} = \left[\frac{2}{k+1}\right]^{\frac{k}{k-1}} \quad (19)$$

Where,

K = Ratio of Specific heat [C_p/C_v]

The critical flow nozzle pressure is,

$$P_{cf} = P_{cfr} \times P_1 \quad (20)$$

$$P_1 = \left([P_{set} \times 14.5] \times \left[1 + \left(\frac{P_{op}}{100}\right)\right]\right) + 14.7$$
(21)

Where,

P_{set} = Set Pressure [psia]

P_{op} = Over Pressure [%]

When the value of the ratio of specific heat is known, Coefficient, C is,

$$C = 520\sqrt{k \times \left[\frac{2}{k+1}\right]^{\frac{k+1}{k-1}}} \quad (22)$$

For subcritical flow, required effective discharge area of a conventional PSV for critical flow is,

$$A_{calc} = \frac{W}{735 \times F_2 \times K_d \times K_c}\sqrt{\frac{Z \times T}{MW \times P_1 \times (P_1 - P_2)}} \quad (23)$$

The coefficient of subcritical flow [F_2] is,

$$F_2 = \sqrt{\left(\frac{k}{k-1}\right) \times r^{\frac{2}{k}} \times \left[\frac{1-r^{\frac{k-1}{k}}}{1-r}\right]} \quad (24)$$

Where, r is the ratio of back pressure to upstream relieving pressure, P_2/P_1

$$r = \frac{K_d}{P_1} \quad (25)$$

The condition to check whether flow is critical or subcritical is, if the back pressure at relief valve discharge is less than or equal to critical flow nozzle pressure, then flow is critical, else subcritical.

Applying the above to estimate the pressure relief device, the critical flow pressure ratio is,

$$P_{cfr} = \left[\frac{2}{1.2+1}\right]^{\frac{1.2}{1.2-1}} = 0.5645 \quad (26)$$

The upstream relieving pressure is,

$$P_1 = \left([8 \times 14.5] \times \left[1 + \left(\frac{10}{100}\right)\right]\right) + 14.7 \quad (27)$$

$$P_1 = 142.3\, psia \quad (28)$$

The critical flow nozzle pressure [P_{cf}] is,

$$P_{cf} = 142.3 \times 0.5645 \quad (29)$$

$$P_{cf} = 80.3\, psia \approx 4.5 barg \quad (30)$$

From the above calculation it is seen that the back pressure at relief valve discharge is 4 barg which is less than the calculated critical flow nozzle pressure value of 4.5 barg. Hence the flow through the relief valve is critical.

Therefore, applying the relevant formulae for critical flow through the relief device,

$$W = 10{,}000 \times 2.20462 = 22{,}046\, lb/h$$
(31)

$$C = 520\sqrt{1.2 \times \left[\frac{2}{1.2+1}\right]^{\frac{1.2+1}{1.2-1}}} \approx 337 \quad (32)$$

$$T = [(1.8 \times 42) + 32] + 459.67 = 567°R \quad (33)$$

The pressure relief device is installed with a rupture disc and K_d is 0.975. The correction factor due to back pressure K_b is 1.0 for conventional relief valve. The combination correction factor for use of rupture disc [K_c] when rupture disc is installed is 0.9. Therefore, for the critical flow behaviour the required effective orifice area is estimated as,

$$A_{calc} = \frac{22046}{337\times0.975\times142.3\times1\times0.9}\sqrt{\frac{567\times0.9428}{33}} \quad (34)$$

$$A_{calc} = 2.11\ in^2 \quad (35)$$

From Table 1, the PSV chosen for a calculated orifice area of 2.11 in^2 is an '**L**' designated orifice which has an orifice area of 2.85 in^2.

Control Valve Fail Open Case

For the case of a control valve fail open case, the PSV relief rate is determined by the control valve's flow capacity. For example, in a gas oil separator, the control valve on the gas side that feeds to the downstream equipment, when it fail opens, the downstream PSV's relieving rate determines the PSV size. In the current module, procedures provided by Fisher's Control Valve Handbook, 5th Edition is used to demonstrate an example case. It is to be noted that Fisher's Handbook, for the case of sizing control valves for compressible fluids employs ISA's standardized procedure, namely the ANSI/ISA S75.01 for calculating the required valve flow coefficient, C_v. Flow is a dependent variable. Based on the ISA procedure, control valves can be sized depending on the fluid properties available i.e.,

1. *Mass Flow Rate & Fluid Density*
2. *Mass Flow Rate & Gas Compressibility Factor*
3. *Standard Volumetric Flow Rate & Gas Compressibility Factor*

The sizing equations for a control valve are also influenced by the piping geometry & attached fittings. In this module, it is assumed that the piping geometry is similar to the line size valve and no fittings are present. The data related to the control are taken from Fisher's Handbook based on "*Representative Sizing Coefficients for Single-Ported, Globe-Style Valve Bodies*" (Table 5.10.1) [Ref 2].

With the above described, taking a single ported, globe style valve, cage guided valve plug, equal percentage flow characteristic with a flow coefficient of C_v of 224 with an inlet conditions of 6.5 barg and 35^{0}C and outlet conditions of 5.8 barg and 32.7^{0}C (551^{0}R) at fail open position.

Considering a non-fire case, the RV set pressure is 4.5 barg & over pressure/maximum accumulated pressure is taken to be 10%. For preliminary sizing, the back pressure at the RV discharge is considered to be fairly constant at 1 barg.

A rupture disc is not installed upstream of the relief valve considering no corrosive fluid exists and a conventional type of relief valve is recommended. The pressure relief valve requires capacity certification as per ASME Sec VIII, Division I. The fluid properties of the hydrocarbon are shown below,

Table 4. Fluid Properties – Control Valve Fail Open

Parameter	Value	Units
Specific Heat Ratio [k]	1.25	[-]
Compressibility Factor [Z]	0.9791	[-]
Gas Molecular Weight [MW]	19.37	[kg/kmol]

The first step in calculating the PSV size for a control valve fail open case begins with estimating the fluid flow rate through the control valve when the valve fails open (say, due to failure of instrument air). This can be calculated using the expression,

$$C_v = \frac{m}{94.8\times F_P\times P_1\times Y}\sqrt{\frac{T_1\,Z_1}{x\times MW}} \quad (36)$$

Rearranging to calculate mass flow rate 'm',

$$m = C_v \times 94.8 \times F_P \times P_1 \times\ Y\sqrt{\frac{x\times MW}{T_1\,Z_1}} \quad (37)$$

Where,

$$Y = 1 - \frac{x}{3\,F_K\ X_T} \quad (38)$$

$$F_K = \frac{k}{1.4} \tag{39}$$

$$x = \Delta P / P_1 \tag{40}$$

m = Mass flowrate [kg/h]

C_V = Flow rate coefficient at rated capacity [-]

P_1 = Valve upstream absolute pressure [bara]

MW = Gas molecular weight [kg/kmol]

T_1 = Control valve inlet temperature [K]

k = specific heats factor [C_p/C_v]

Z = gas compressibility factor [-]

ΔP = Pressure drop at rated flow [bar]

F_p = Piping geometry factor (F_p =1) for line sized valve and no attached fittings [-]

F_K = Ratio of specific heats factor [-]

x = Pressure drop ratio [-]

X_T = Choked flow pressure drop factor (refer to Vendor´s catalogue) [-]

Y = Gas expansion factor [-]

Solving for the problem at hand,

$$\Delta P = P_1 - P_2 \tag{41}$$

$$P_1 = 6.5\ barg \approx 109\ psia \tag{42}$$

$$P_2 = 5.8\ barg \approx 99\ psia \tag{43}$$

$$\Delta P = 109 - 99 = 10\ psi \tag{44}$$

The ratio of specific heat factor is estimated as,

$$F_K = \frac{k}{1.4} = \frac{1.247}{1.4} = 0.891 \tag{45}$$

The rated pressure drop factor [x_T] from Table 5.10.1 for the chosen valve is 0.72 and the piping geometry factor [F_p] is taken to be 1.0 for this example. A condition that has to be satisfied to calculate the pressure drop ratio and whether the flow behaviour is critical or subcritical is,

$$If\ \frac{\Delta P}{P_1} > F_k x_T\ then\ x = F_k x_T \tag{46}$$

$$If\ \frac{\Delta P}{P_1} < F_k x_T\ then\ x = \frac{\Delta P}{P_1} \tag{47}$$

$$If\ \frac{\Delta P}{P_1} > F_k x_T\ then\ Crictical\ Flow \tag{48}$$

$$If\ \frac{\Delta P}{P_1} < F_k x_T\ then\ Sub\ crictical\ Flow \tag{49}$$

$$\frac{\Delta P}{P_1} = \frac{10}{109} = 0.0932 \tag{50}$$

$$F_k x_T = 0.891 \times 0.72 \approx 0.6415 \tag{51}$$

Therefore, based on the condition for calculating the pressure drop ratio, x is 0.0917. The gas expansion Factor [Y] is calculated as,

$$Y = 1 - \frac{x}{3\ F_K\ X_T} = 1 - \frac{0.0932}{3 \times 0.891 \times 0.72} = 0.952 \tag{52}$$

Therefore, the mass flow through the control valve is estimated as,

$$m = 224 \times 94.8 \times 1 \times 7.5 \times\ 0.952 \times \sqrt{\frac{0.0932 \times 19.37}{(35+273.15) \times 0.9791}} \tag{53}$$

$$m = 11{,}742\ kg/h \tag{54}$$

Based on the calculated mass flow rate, the PSV sizing commences & is similar to blocked gas outlet case. Hence proceeding on similar lines,

$$P_{cfr}\ = \left[\frac{2}{1.25+1}\right]^{\frac{1.25}{1.25-1}} = 0.5555 \tag{55}$$

$$P_1 = \left([4.5 \times 14.5] \times \left[1 + \left(\frac{10}{100}\right)\right]\right) + 14.7 \tag{56}$$

$$P_1 = \ 86.5\ psia \tag{57}$$

$$P_{cf}\ =\ 86.5\ \times 0.5555 \tag{58}$$

$$P_{cf}\ =\ 48\ psia \approx 2.3\ barg \tag{59}$$

Therefore, since the PSV back pressure of 1 barg is lower than the critical flow nozzle pressure [P_{cf}] of 48 psia (2.3 barg), the flow behaviour across the PSV is critical.

$$W\ =\ 11{,}742\ \times 2.20462 = 25{,}886\ lb/h \tag{60}$$

$$C\ =\ 520\sqrt{1.25 \times \left[\frac{2}{1.25+1}\right]^{\frac{1.25+1}{1.25-1}}} \approx 342$$

(61)

$$P_1 = \left([4.5 \times 14.5] \times \left[1 + \left(\frac{10}{100}\right)\right]\right) + 14.7 \tag{62}$$

$$P_1 = \ 86.5\ psia \tag{63}$$

$$T\ = [(1.8 \times 32.7) + 32] + 459.67 = 551°R \tag{64}$$

The coefficient of discharge for the pressure relief device when installed with or without a rupture disc in combination, K_d is 0.975. The correction factor due to back pressure K_b is

1.0 for conventional relief valve. The combination correction factor for use of rupture disc [K_c] when no rupture disc is installed is 1.0. Therefore, for the critical flow existing the required effective orifice area is estimated as,

$$A_{calc} = \frac{25886}{342 \times 0.975 \times 86.5 \times 1 \times 1}\sqrt{\frac{551 \times 0.9791}{19.37}} \quad (65)$$

$$A_{calc} = 4.74\ in^2 \quad (66)$$

From Table 1, the PSV chosen for a calculated orifice area of 4.74 in^2 is a '**P**' designated orifice which has an orifice area of 6.38 in^2.

Relief Valve Sizing – Thermal Expansion

For the case of a relief valve sizing – thermal expansion, consider a vessel with a liquid density of 850 kg/m^3 (SG 0.85 at 15.6^0C). The total heat transfer rate to the vessel is 1000 kJ/s (3,412,140 BTU/h). The specific heat of the trapped fluid is 3.9 kJ/kg.K (0.9315 $Btu/lb.^0F$). The cubic expansion of the hydrocarbon is 0.0005 $1/^0F$. The relief valve flow rate is calculated as,

$$q = \frac{\alpha_v \times \emptyset}{500 \times d \times c} \quad (67)$$

Where, q = Volumetric flow rate at relieving conditions, [USGPM]

ϕ = cubic expansion coefficient for the liquid at relieving conditions [$1/^0F$]

d = Relative Density referred to water [-]

c = Trapped fluid's specific heat capacity [$Btu/lb.^0F$]

Therefore, the relieving rate is,

$$q = \frac{0.0005 \times 3412140}{500 \times 0.85 \times 0.9315} \quad (68)$$

$$q = 4.3\ USGPM \approx 1\ m^3/h \quad (69)$$

Since the flowrates are very small for thermal expansion cases the safety valve size of NPS ¾" × NPS 1" (DN 20 x DN25) should be sufficient as per Ref 4. This calculation method provides only short-term protection in some cases. If the blocked-in liquid has a vapour pressure higher than the relief design pressure, then the pressure relieving device should be capable of handling the vapour generation rate. If discovery and correction before liquid boiling is expected, then it is not necessary to account for vaporization in sizing the PRD [Ref 4]. Two general applications for which thermal-relieving devices larger than above described are above ground long uninsulated pipelines of large diameter and large vessels or exchangers operating liquid full [Ref 4].

Relief Valve Sizing – External Fire Case Liquid Filled Vessel

For the case of a relief valve sizing – External Fire Case Liquid Filled Vessels that are exposed to fire begins with estimating the vapour flow rate in the event of a fire.

When vessels are exposed to heat, hydraulic expansion or thermal expansion is expected to occur wherein there is an increase in liquid volumes due to increase in temperature. In the case of external fire, API 521 distinguishes between wetted vessel & un-wetted vessels. A wetted vessel contains a liquid in equilibrium with its vapours. During an external fire, partial evaporation of liquid occurs, such that the portion of the vessel in contact with the liquid within a distance of 25 feet (7.62 m as in ISO 23251) receives heat transfer from the exposed fire and must be considered for sizing. If there is thermal cracking of the vessel occurs leading to vapour generation, other alternate sizing methods need to be considered.

An un-wetted vessel is either thermally insulated on the thermal walls or filled with gases, vapours or super critical fluids. In comparison to wetted vessels, the thermal flow from the walls to the interior are low in un-wetted vessels due to the large thermal resistance. If the vessel is subjected to exposure to the fire for prolonged periods, the vessel temperature would be high enough to cause thermal rupture of the vessel.

The total heat of absorption, Q for the wetted surface can be estimated with adequate drainage and prompt firefighting facilities as,

$$Q = 21000 F A_{ws}^{0.82} \quad (70)$$

Without adequate drainage and prompt firefighting facilities,

$$Q = 34500 F A_{ws}^{0.82} \quad (71)$$

Q = Heat load/total heat of absorption [Btu/h]

F_{env} = Environmental Factor [-]

A_{ws} = Wetted Surface Area [m^2]

The value of 'F_{env}' for a bare vessel, water application facilities on bare vessels, depressurizing and empty facilities is taken to be 1.0. For insulated vessels, the value of 'F_{env}' is taken as per Ref [5], Table T7-6

Table 5. Environmental Factor – Insulated Vessel

Insulation Thermal Conductivity	[F_{env}]
[W/m.K]	[-]
22.71	0.3
11.36	0.15
5.68	0.075
3.8	0.05
2.84	0.0376
2.27	0.03
1.87	0.026

Plotting [F_{env}] for the insulated vessel vs. Insulation Thermal Conductivity [K], a linear relationship can be arrived at as,

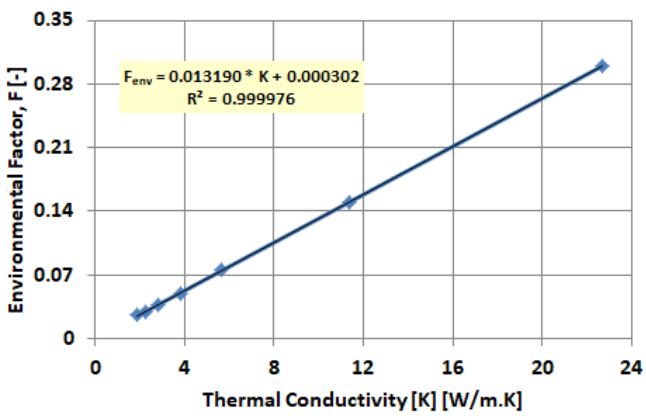


Figure 2. Environmental Factor Vs. Insulation Thermal Conductivity

i.e, $F_{env} = (0.01319 \times K) + 0.000302 \quad (72)$

The possible mounting position of the partially filled wetted vessels with liquids can be horizontal or vertical with hemispherical or elliptical heads. The figure below gives a description of the described vessel positions,

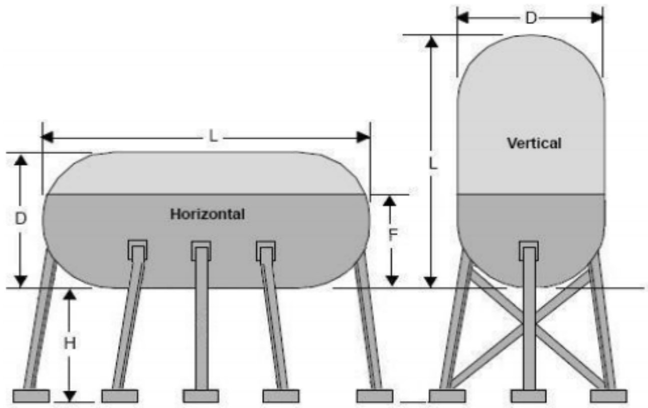


Figure 3. Possible mounting positions of the Partially Filled Wetted vessels with liquids

From the mounting positions described, considering a vessel without a boot, the effective total height of liquid surface [K_1] is,

$$K_1 = H + F \quad (73)$$

If a boot exists at the bottom of the vessel, the vessel elevation, H is computed by subtracting the boot dimensions from the vessel elevation to compute K_1 as,

For Hemispherical head,

$$H_{with\ boot} = H - \left[h_{boot} + \frac{d_{boot}}{2}\right] \quad (74)$$

For Elliptical Head,

$$H_{with\ boot} = H - \left[h_{boot} + \frac{d_{boot}}{4}\right] \quad (75)$$

K_1 = Effective total height of liquid surface [m]

H = Vessel Elevation without boot [m]

$H_{with\ boot}$ = Vessel Elevation with boot [m]

F = Liquid Level in Vessel [HLL/NLL] [m]

h_{boot} = Height of Boot [m]

d_{boot} = Diameter of Boot [m]

Considering that the portion of vessel exposed to fire, only that portion in contact with the liquid within a distance of 25 feet (7.62 m) above the fire source must be considered for sizing. Therefore, the effective total height of the liquid surface, $K_{1,eff}$ is,

$$If\ K_1 < 7.62\ m\ then\ K_{1,eff} = K_1 \quad (76)$$

$$If\ K_1 > 7.62\ m\ then\ K_{1,eff} = 7.62\ m \quad (77)$$

Where, $K_{1,eff}$ = Effective total height of liquid surface[m]

The initial liquid level, E_1 for vessel without a boot is calculated as,

$$E_1 = K_{1,eff} - H \quad (78)$$

With boot, the initial liquid level, E_1 becomes,

$$E_1 = K_{1,eff} - H_{with\ boot} \quad (79)$$

Therefore the effective liquid level, $E_{1,eff}$ that would be used in the calculations are,

$$If\ E_1 \leq 0\ then\ E_{1,eff} = 0 \quad (80)$$

$$If\ E_1 > 0\ then\ E_{1,eff} = E_1 \quad (81)$$

The effective liquid level angle, β becomes,

$$\beta = Cos^{-1}\left[1 - \frac{2E_{1,eff}}{D}\right] \quad (82)$$

β = Effective Liquid Level Angle [degrees]

D = Diameter of the Vessel [m]

It is also considered that a portion of the associated piping is also subjected to external heat & is assumed that it is completely filled with liquid. Additionally, the wetted area computed with hemispherical ends is considered approximately equal to wetted area with elliptical heads since a margin is added to account for piping. The difference is expected to be accommodated in the margin. Taking that the boot in the horizontal vessel is always liquid filled; the boot wetted area is added to the vessel wetted area. Therefore, the wetted surface area, A_{ws} for a horizontal vessel without boot is calculated as,

$$A_{ws} = \left(\pi D \times \left[E_{1,eff} + \frac{(L-D)\beta}{180}\right]\right) \times \left[1 + \frac{\%_{piping}}{100}\right] \quad (83)$$

For Horizontal Vessel with boot, A_{ws} becomes,

$$A_{ws} = \left(\pi D \times \left[E_{1,eff} + \frac{(L-D)\beta}{180}\right] + \frac{\pi}{2} d_{boot}^2 \left[\frac{h_{boot}}{2} + 1\right]\right) \times \left[1 + \frac{\%_{piping}}{100}\right] \quad (84)$$

For a vertical vessel, A_{ws} becomes,

$$A_{ws} = \pi \times E_{1,eff} \times D \times \left[1 + \frac{\%_{piping}}{100}\right] \quad (85)$$

The vapour flow rate generated due to the external fire is now calculated as,

$$W = Q/\lambda \quad (86)$$

Where, W = Vapour Flow Rate [lb/h]

Q = Heat load [Btu/h]

λ = latent Heat of Vapourization [Btu/lb]

The vapour rate estimated now becomes the relieving rate for the PSV based on which the PSV orifice size is estimated. For this module, consider a horizontally mounted vessel with boot with hemispherical ends for both boot and vessel. The vessel is insulated and adequate drainage & firefighting measures are also available. The design parameters are,

Table 6. Liquid Filled Fire Case – Vessel Data

Design Parameters	Value	Units
Tank Diameter [D]	4.50	m
Tank Length [T/T] [L]	21.0	m
Boot Diameter [d_{boot}]	1.20	m
Boot Height [h_{boot}]	2.00	m
Tank Elevation w/o Boot [H]	5.50	m
Operating Liquid Level [F]	4.00	m
Insulation Thermal Conductivity [K]	3.00	W/m.K
% Piping Exposed to Fire	20.0	%

The fluid properties are as follows,

Table 7. Liquid Filled Fire Case – Fluid Properties

Design Parameters	Value	Units
Ratio of specific heats [$k = C_p/C_v$]	1.39	-
Gas Compressibility Factor [Z]	0.53	-
Gas Molecular Weight [MW]	79.3	lb/lbmol
Relieving Temperature [T]	277	^{0}C
Latent Heat of Vaporization [λ]	150	kJ/kg

The RV set pressure is 100 barg & over pressure/maximum accumulated pressure for the fire case is taken to be 21%. For preliminary sizing, the back pressure at the RV discharge is considered to be fairly constant at 3.5 barg. A rupture disc is not installed upstream of the relief valve considering no corrosive fluid exists and a conventional type of relief valve is recommended. The pressure relief valve requires capacity certification as per ASME Sec VIII, Division I. Therefore, to estimate the relief rate due to the external fire, the tank elevation with boot, H for a hemispherical head is calculated as,

$$H_{with\ boot} = 5.5 - \left[2 + \frac{1.2}{2}\right] = 2.9\ m \tag{87}$$

$$K_1 = 2.9 + 4 = 6.9\ m \tag{88}$$

$$Since\ K_1 < 7.62\ m \ \rightarrow\ K_{1,eff} = 6.9\ m \tag{89}$$

$$E_1 = 6.9 - 2.9 = 4\ m \tag{90}$$

$$Since\ E_1 > 0\ then\ \ E_{1,eff} = 4\ m \tag{91}$$

The effective liquid level angle, β is,

$$\beta = Cos^{-1}\left[1 - \frac{2\times4}{4.5}\right] \approx 141° \tag{92}$$

Therefore, the wetted surface area, A_{ws} is,

$$A_{ws} = \left[\pi \times 4.5 \times \left[4 + \frac{(21-4.5)\times141}{180}\right] + \left(\frac{\pi\times1.2^2}{2} \times \left[\frac{2}{2}+1\right]\right)\right] \times \left[1 + \frac{20}{100}\right] \tag{93}$$

$$A_{ws} = 292.64\ m^2 \approx 3{,}150\ ft^2 \tag{94}$$

The F_{env} factor is computed as,

$$F_{env} = (0.01319 \times 3) + 0.000302 = 0.04 \tag{95}$$

Therefore, the total heat of absorption or heat load for the insulated vessel with adequate drainage and firefighting measures is,

$$Q = 21000 \times 0.04 \times 3150^{0.82} = 618{,}730\ Btu/h \tag{96}$$

The vapour flow rate produced due to the external fire for λ of 150 kJ/kg (64.5 Btu/lb),

$$W = \frac{618{,}730}{64.5} = 9{,}594\ lb/h\ or\ 4{,}352\ kg/h \tag{97}$$

Based on calculated mass flow rate, the PSV sizing commences & is similar to blocked gas outlet case. Hence proceeding on similar lines,

$$P_{cfr} = \left[\frac{2}{1.39+1}\right]^{\frac{1.39}{1.39-1}} = 0.5305 \tag{98}$$

$$P_1 = \left([100 \times 14.5] \times \left[1 + \left(\frac{21}{100}\right)\right]\right) + 14.7 \tag{99}$$

$$P_1 = 1769\ psia \tag{100}$$

$$P_{cf} = 1769 \times 0.5305 \tag{101}$$

$$P_{cf} = 938\ psia \approx 62.8\ barg \tag{102}$$

Therefore, since the PSV back pressure of 3.5 barg is lower than the critical flow nozzle pressure [P_{cf}] of 938 psia (62.8 barg), the flow behaviour across the PSV is critical.

$$C = 520\sqrt{1.39 \times \left[\frac{2}{1.39+1}\right]^{\frac{1.39+1}{1.39-1}}} \approx 355 \tag{103}$$

$$P_1 = \left([100 \times 14.5] \times \left[1 + \left(\frac{21}{100}\right)\right]\right) + 14.7 \tag{104}$$

$$P_1 = 1769\ psia \tag{105}$$

$$T = [(1.8 \times 277) + 32] + 459.67 = 990°R \tag{106}$$

The coefficient of discharge for the pressure relief device when installed with or without a rupture disc in combination, K_d is 0.975. The correction factor due to back pressure K_b is 1.0 for conventional relief valve. The combination correction factor for use of rupture disc [K_c] when no rupture disc is installed is 1.0. Therefore, for the critical flow behaviour the required effective orifice area is estimated as,

$$A_{calc} = \frac{9{,}594}{355\times0.975\times1769\times1\times1}\sqrt{\frac{990\times0.5305}{79.3}} \tag{107}$$

$$A_{calc} = 0.04\ in^2 \tag{108}$$

From Table 1, the PSV chosen for a calculated orifice area of 0.04 in^2 is a '**D**' designated orifice which has an orifice area of 0.11 in^2.

References & Further Reading

1. "*Sizing Selection and Installation of Pressure Relieving Devices in Refineries*", API Recommended Practice 520, 7th Edition, January 2000, Part -1
2. "*Control Valve Handbook*", Emerson's Fisher's Handbook, 5th Edition
3. "*Pressure Relieving and Depressuring Systems*", API Standard 521, 6th Edition, Jan 2014
4. "*API 521 7th Ed Ballot Item 6.4 Work Item 30 – Thermal Expansion Equation Definitions*"
5. *"Crosby Engineering Handbook"*, Technical Doc No. TP-V300, May 1997, Crosby Valve Inc.

Module 18

Key Process Considerations for Pipeline Design Basis

Prior to venturing into an oil & gas pipeline project, the project team would require a design basis, based on which the project is to proceed. Oil & Gas Pipeline design begins with a route survey including engineering & environmental assessments. The following document provides a few key considerations for process engineers to keep in mind, the factors that matter when preparing a pipeline design basis from a process standpoint.

1. Well Production Data/Profile

Well production profiles are required as this determines the size of the pipeline required to transport volume/time of fluid. Gases are highly compressible and cannot be treated the same as liquids such as, crude oils & petroleum distillates. The operating pressures & temperatures are required to be known as they determine the design conditions of the pipeline.

2. Fluid Physical Properties

The physical properties of the materials being transported dictate the design and operating parameters of the pipeline. Specific gravity, compressibility, kinematic & dynamic viscosity, pour point, and vapour pressure of the material are the primary considerations. The pour point of a liquid is the temperature at which it ceases to pour. The pour point for oil can be determined under protocols set forth in the ASTM Standard D-97.

In general, crude oils have high pour points. When transported hydrocarbons operate below their pour point, auxiliary measures such as heating, diluting with lighter hydrocarbons that are miscible & allows lowering the viscosity & pour point temperature, mixing with water to allow the waxes to slide through the pipe walls, or modifying the chemical composition of the hydrocarbon. It is to be noted that, in the case of finished products, (e.g., gas oil or Jet A1 fuel), many of the auxiliary measures like addition of water or mixing with lighter hydrocarbons becomes infeasible, since they affect the product specification.

Vapour Pressure of a liquid is its capacity to vaporize/evaporate into its gaseous phase. In pipeline operations, slack flow is a situation where due to the elevational & pipeline pressure drops, a portion of the hydrocarbon experiences pressure below its vapour pressure. As a result, a portion of the liquid vaporizes & reaches the high points in the pipeline. Upon restarting the pipeline, the vapour pockets experience a compressive rise in pressure due to the upstream & downstream liquid pockets, only to collapse & release energy that can rupture pipelines.

Reid vapour pressures are critical to liquid petroleum pipeline design, since the pipeline must maintain pressures greater than the Reid vapour pressure of the material in order to keep the material in a liquid state. Pipelines that handle finished products are preferred to be operated with single phase flow regime & fully filled pipes. This ensures there is no scope for volatilization that reduces the scope for fire hazards.

3. Pumping Costs

Viscous fluids require more power to deliver required motive force to the hydrocarbons to transport them across the pipeline. Waxy crudes can be pumped below their pour point However if the flow is stopped, for e.g., after a pipeline shutdown, the energy required to restart the pipeline would be much higher than what was required to keep it flowing. Pipelines also suffer from the formation of hydrates & asphaltenes. Waxes can form crystalline structures that tend to agglomerate & is referred to as gelling.

Gelling is also a phenomenon that is found in storage tanks in production facilities where the fluid sits motionless for hours or even days, resulting in operational difficulties. Hence to attend to these limitations, pour point estimation becomes vital to determine if external heating is required. In some cases, if the waxy crude does not gel enough, it can get transported to the pump where shear forces & rise in temperature allow the waxy crudes to stay above the pour point.

4. Thermal Stresses

Petroleum pipelines are normally buried unless local regulation prevents them. To do so, trenches are dug & are laid below grade/frost line level. Such measures also provide the advantage of maintaining relatively constant temperature in line with the ambient/season soil temperature, thereby ensuring the pipeline expansion does not occur to the point of deflection. Expansion joints as well as in some cases, trenches are dug extra wide to accommodate any lateral movement. In case of river crossings, the pipeline is to be laid above ground. In locations that are prone to landslides, buried pipelines option is preferred to avoid direct impact of rock structures. But this does not necessarily mean buried pipelines are free from structural damage, since the weight of the soil/rock structures deposited above the pipeline can also crush the buried pipelines.

5. Pipeline Pressure Drop

Pipelines are designed keeping in mind, the material & construction costs as well operating costs. Material costs are determined by the pipeline weight, whereas operational costs are largely impacted by the pressure drop experienced which is a function of the flow regime. The two key forces dictating the pipeline total pressure losses are – Hydrostatic pressure drop due to the pipeline elevation & frictional pressure drop which depends on the flow rate. In multiphase pipelines across hilly terrains, hydrostatic pressure drop decreases while frictional pressure drop increases with flow. The sum of both these pressure losses gives the total pressure loss. The pipeline size chosen should be preferably, the point at which the total pressure loss is the least.

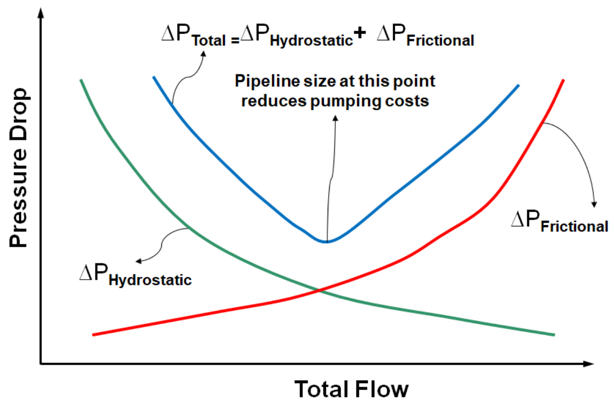


Fig 1. Pipeline Total Pressure Loss – Hilly Terrain

From the above figure it can be seen that operating a multiphase line at a lower flowrate can actually cost more to pump.

6. Max Hydro Test – ASME B31.8

For a class1 div 1, the test pressure is limited to 125% of the design pressure. For class 3 or 4, test pressure is up to 1.4 × MAOP. Therefore, for an 8 hour min. test pressure, with a design factor of 0.72 for class 1, the test would cause the hoop reaching 72% of Pipeline's SMYS. Testing at 125% of MAOP will result in 1.25 × 0.72 = 0.9 or 90% of SMYS. Hence by hydro testing the pipeline at 1.25 times the operating pressure, the pipeline is stressed out to 90% of its SMYS. The hydro test pressure is based on the location class and maximum test pressure, & becomes the lower value of 8 hour minimum test pressure & test pressure at low point.

7. Valve Spacing

Pipelines need valves to placed & spaced taking into consideration – Rapid Isolation/Shutdown of pipeline sections to minimize inventory breach, maintain pipeline design integrity, and facilitate maintenance, repairs & hot tapping operations. Pipelines would also be subjected to pigging & hence the valve placement must enable recovery of stuck pigs.

8. Hydrocarbon Flares

Pipelines would sometimes have to be blown down of any hydrocarbons (liquid or liquid mixed vapours) during events of over pressure. Burn pit lines serve this purpose. It is important to monitor pilot flames and provide pilot flame failure alarms. Since burn pit lines are a source of open flame, they are to be located at least 150 m away from roadways, process & storage facilities. In cases of pipeline in remote locations requiring maintenance or repair, mobile flare units can be used.

However not all occasions would allow open flaring, as a result of which, enclosed ground flares can be used. These conform to the requirements of flaring & disposal in populated areas or process facilities that are in close proximity to the flare system. The flaring is smokeless with no visible flame & noiseless due to insulation of the combustion chamber. To attend to the flare capacities required, a flare study report is to be made part of the design basis.

9. Pipeline Standards/Codes[1]

ASME has been a pioneer in developing industry codes & standards for oil & gas pipelines. The scope of the first draft of the ASME Code for Pressure Piping, which was approved by the American Standards Association in 1935, included the design, manufacture, installation, and testing of oil and gas pipelines (ASME B31.4). As the needs of the industry evolved over the years, rules for new construction have been enhanced, and rules for operation, inspection, corrosion control, and maintenance have been added. In addition to ASME, several other organizations, including the API and NACE International, also developed standards used by the pipeline industry. Some of the ASME/API/ANSI standards are,

1. "*Gas Transmission and Distribution Piping Systems*," ASME B31.8, 1999.
2. "*Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids,*" ASME B31.4, 1998.
3. "*Power Piping*", ASME B31.1, 1998; Addenda B31.1A, 1999; Addenda B31.1B, 2000
4. "*Process Piping*" ASME B31.1, 1999; Addenda B31.3A, 1999
5. "*Slurry Transportation Piping Systems*" ASME B31.11, '89; Addenda B31.11A, 1991
6. "*Pipeline Transportation Systems for Liquid Hydrocarbons and Other Liquids*" ASME B31.4, 2002
7. "*Gas Transmission and Distribution Systems*," ASME 31.8, 2003
8. "*Specification for Line Pipe*", API 5L, Mar 2004 / Errata 1, Jan 2005
9. "*Steel Pipelines Crossing Railroads and Highways*" API 1102 (1993)
10. "*Specification for Pipeline Valves (Gate, Plug, and Check Valves)*", 21st edition, API 6D1, June 1998 Supplement 2
11. Pipeline wall thickness (API B31.G)

Velocity Considerations[3]

Gas line velocities should be less than 60 to 80 ft/s to minimize noise & allow for corrosion inhibition. A lower velocity of 50 ft/s should be used in the presence of known corrosives such as CO_2. The minimum gas velocity should be between 10 and 15 ft/s, which minimize liquid fallout. The minimum fluid velocity in multiphase systems must be relatively high to keep the liquids moving in order to prevent/minimize slugging. The recommended minimum velocity is 10 to 15 ft/s. The maximum recommended velocity is 60 ft/s to inhibit noise & 50 ft/s for CO_2 corrosion inhibition. In two-phase flow, it is possible that the flow stream's liquid droplets can impact the pipe wall causing erosion of the corrosion products. Erosion of the pipe wall itself could occur if solid particles, particularly sand, are entrained in the flow stream.

Pipeline Mechanical Design

As an example to perform Gas Pipeline mechanical design, ASME B31.8 is used. The requirement to be met for pipeline wall stresses as per ASME B31.8 is Design factor [F], Temperature De-rating [T], Longitudinal Joint Factor [E] for the chosen pipeline joining methods. This is shown below as follows,

Table 1. Reference Mechanical Design Parameters

Design Factors [F] - Gas Pipeline Location		
Class	**Description**	**F**
Class 1, Div 1	Deserted	0.80
Class 1, Div 2	Deserted	0.72
Class 2	Village	0.60
Class 3	City	0.50
Class 4	Densely Populated	0.40
Temperature De-rating [T] for Gas Pipelines		
T [^{0}F]	**T [^{0}C]**	**T**
≤ 250	≤ 120	1.00
300	150	0.97
350	175	0.93
400	200	0.91
450	230	0.87
Abbreviation	**Joining Method**	**E**
SMLS	Seamless	1.0
ERW	Electric Resistance Weld	1.0
EFW	Electric Flash Weld	1.0
SAW	Submerged Arc Weld	1.0
BW	Furnace Butt Weld	0.6
EFAW	Electric Fusion Arc Weld	0.8

The pipeline specification requirement as per API 5L plain end line pipe specifications, ranges from 6" ND to 80" ND. The product pipeline specification (PSL) with its respective Specified Minimum Yield Strength (SYMS) to be used as per API 5L are PSL 1 and PSL 2. The pipeline grades are as follows,

Table 2. Product Specification Level (PSL)

Grade	SMYS MPa	Grade	SMYS MPa
PSL 1 Gr A25	172	PSL 2 Gr B	241
PSL 1 Gr A	207	PSL 2 X42	290
PSL 1 Gr B	241	PSL 2 X46	317
PSL 1 X42	290	PSL 2 X52	359
PSL 1 X46	317	PSL 2 X56	386
PSL 1 X52	359	PSL 2 X60	414
PSL 1 X56	386	PSL 2 X65	448
PSL 1 X60	414	PSL 2 X70	483
PSL 1 X65	448	PSL 2 X80	552
PSL 1 X70	483	-	-

Location of the Gas Pipelines

1. **Class 1 location** - A Class 1 location is any 1-mile pipeline section that has 10 or fewer buildings intended for human occupancy including areas such as, wastelands, deserts, rugged mountains, grazing land, farmland, sparse populations.
2. **Class 1, division 1 Location** – A Class 1 location where the design factor, F, of the pipeline is greater than 0.72 but equal to, or less than 0.80 and which has been hydrostatically tested to 1.25 times the maximum operating pressure.
3. **Class 1, division 2 Location** - This is a Class 1 location where the design factor, F, of the pipeline is equal to or less than 0.72, and which has been tested to 1.1 times the maximum operating pressure.
4. **Class 2 Location** - This is any 1-mile section of pipeline that has more than 10

but fewer than 46 buildings intended for human occupancy including fringe areas around cities and towns, industrial areas, and ranch or country estates.

5. **Class 3 Location** - This is any 1-mile section of pipeline that has 46 or more buildings intended for human occupancy except when a Class 4 Location prevails, including suburban housing developments, shopping centres, residential areas, industrial areas & other populated areas not meeting Class 4 Location requirements
6. **Class 4 Location** - This is any 1-mile section of pipeline where multi-storey buildings are prevalent, traffic is heavy or dense, and where there may be numerous other utilities underground. Multi-storey means four or more floors above ground including the first, or ground, floor. The depth of basements or number of basement floors is immaterial.

Line Specification of Gas Pipelines – API 5L

1. PSL1 pipes are available through size 2/5" to 80" whereas the smallest diameter pipe available in PSL2 is 4.5" & the largest diameter is 80". PSL1 pipelines are available in different types of ends, such as Plain end, Threaded end, Bevelled end, special coupling pipes whereas PSL2 pipelines are available in only Plain End.
2. For PSL2 welded pipes, except continuous welding & laser welding, all other welding methods are acceptable. For electric weld welder frequency for PSL2 pipeline is minimum 100 kHz whereas there is no such limitation on PSL1 pipelines.
3. Heat treatment of electric welds is required for all Grades of PSL2 pipes whereas for PSL1 pipelines, grades above X42 require it. All kinds of welding method are acceptable to manufacture PSL1; however, continuous welding is limited to Grade A25.

Gas Pipeline Wall Thickness Estimation

The B31.8 code is often used as the standard of design for natural gas piping systems in facilities, such as compressor stations, gas treatment facilities, measurement & regulation stations & tank farms. The B31.8 wall-thickness formula is stated as,

$$t = \frac{DP \times OD}{2 \times F \times E \times T \times SMYS} \tag{1}$$

Where,

t = Minimum design wall thickness [in]

DP = Pipeline Design Pressure [psi]

OD = Pipeline Outer Diameter [in]

$SMYS$ = Specific Minimum Yield Stress [psi]

F = Design Factor [-]

E = Longitudinal Weld Joint Factor [E]

T = Temperature De-rating Factor [-]

References & Further Reading

1. *"Standard for Gas Transmission and Distribution Piping Systems",* ANSI/ASME Standard B31.8, 1999
2. *"Overview of the Design, Construction, and Operation of Interstate Liquid Petroleum Pipelines"*, ANL/EVS/TM/08-1, Argonne National Laboratory
3. *https://petrowiki.org*

Module 19

Natural Gas Pipeline Transmission Cost and Economics

In any pipeline project, an economic analysis has to be performed to ensure the project is a viable investment. The major capital components of a pipeline system consists of the pipeline, Booster station, ancillary machinery such as mainline valve stations, meter stations, pressure regulation stations, SCADA & Telecommunications. The project costs would additionally consist of environmental costs & permits, Right of Way (ROW) acquisitions, Engineering & Construction management to name a few.The following module is aimed at conducting a pipeline economic analysis using the method of Weight Average Capital Cost (WACC) to estimate gas tariffs, project worth in terms of Net Present Value (NPV), Internal Rate of Return (IRR), Profit to Investment Ratio (PIR) and payback period. The cost of equity is estimated using the Capital Asset Pricing Model (CAPM).

Problem Statement

A pipeline project is proposed to be built to transport natural gas from a gas processing facility to a city power station. The scope of gas supply is expected for a period of 25 years & availability of 8,040 hours/year. Prior calculations indicate a 16" pipeline & a booster station will be required. The undertaking requires a CAPEX & OPEX estimation to explore project viability & profitability. Project funding is through a 60:40 debt [D] to equity [E] ratio. The cost of debt is at a lender's average prime lending rate [*r*] of 17%. The cost of equity is taken at an average risk free interest rate [r_f] of 17% and expected market portfolio return [r_m] of 24%. Considering inflation, Annual O&M costs is taken as 2%. Beta ratio [β] & linear depreciation rate [d_r] is taken as 0.9 and 8%.

Table 1. Pipeline & Booster Station Details

Description	Value	Units
Upstream Pipeline	PSL 1 X 65	
Size	16	Inch
Length incl. fittings [L]	20.4	km
Outer Diameter [OD]	406.4	mm
Inner Diameter [ID]	393.7	mm
Wall Thickness [WT]	6.35	mm
Construction Material (MOC)	Carbon Steel	
MOC Density [ρ]	7,850	kg/m^3
Downstream Pipeline	PSL 1 X 65	
Size	16	Inch
Length incl. fittings [L]	30.6	Km
Outer Diameter [OD]	406.4	mm
Inner Diameter [ID]	392.1	mm
Wall Thickness [WT]	7.14	mm
Construction Material (MOC)	Carbon Steel	
MOC Density [ρ]	7,850	kg/m^3
Booster Compressor		
Absorbed Power	2,097	kW
Margin on Absorbed Power	20	%
Installed Capacity [P]	2,517	kW

The production profile is as follows,

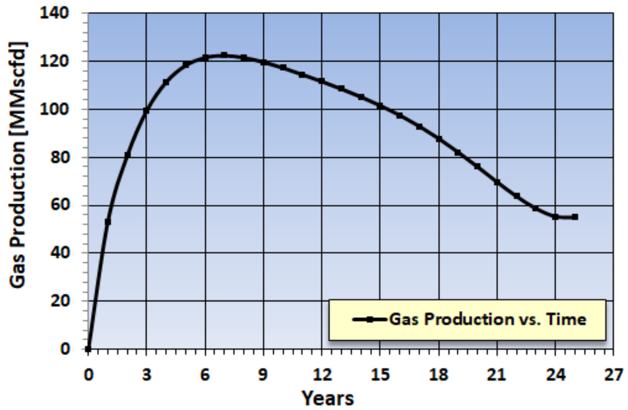


Figure 1. Production Profile at Processing Facility

The associated costs are as follows,

Table 2. Associated Pipeline Construction Costs

Parameter	Description
Total Material Costs [$Cost_{PMC}$]	\$700/ tonne
Coating & Wrapping [$Cost_{PCW}$]	\$15/m
Labour Costs [$Cost_{LC}$]	\$50/m
Install. Cost per inch-km [$Cost_{PIC}$]	\$11,926/in.km

The pipeline installation cost per inch-dia-km is taken by fitting the typical installation cost data from Table 10.1, page 332 of Ref [1] as,

$$PIC\ [\$] = -2.103D_2^3 + 153.39D_2^2 - 2883.42D_2 + 26606.36 \quad (1)$$

Table 3. Associated Station Construction Costs [2]

Parameter	Description
Station Material Cost [SMC]	\$2,877/hP
Station Labour Cost [SLC]	\$916/hP
Station Misc Cost [SMiC]	\$367/hP
Station Land Cost [SLaC]	\$13/hP

The station material costs is taken as, Ref [2],

$$SMC\ [\$/hP] = 13035 \times P^{-0.186} \quad (2)$$

The station labour costs is taken as, Ref [2],

$$SLC\ [\$/hP] = 2274.4 \times P^{-0.112} \quad (3)$$

The sales gas tariff allowed for the gas pipeline project is \$2/Mscf. The sales tax rate [T_{sale}] on the sold gas is 35% and the effective corporate tax [T_C] is 30%. The Booster station consists of a gas turbine driven centrifugal compressor (Single GT/CC). In case of other configurations, the installed cost can be taken as a factor of Single GT/CC configuration as,

Table 4. Associated Station Construction Costs

Parameter	Comparison Factor
Single GT/CC	1.00
Multiple GT/CC	1.29
EM/High Speed RC	1.30
High Speed Engine/RC	1.32
Slow Speed Engine/RC	1.54

Cost Estimation Aspects & Assumptions

1. The weight average capital cost (WACC) comprises of a firm's cost of capital in which each category of capital is proportionately weighed. All sources of capital, including common stock & preferred stock, bonds & any other long-term debt, are included in a WACC calculation. For the following module, the cost of equity [k_e] & cost of debt [k_d], (Sum of which is the cost of capital), expressed as a percentage for the pipeline proposal are included to estimate the WACC.
2. The revenue generated comes from the gas tariffs delivered to the power station. Capital is necessary to initiate a project for which some of the capital can come from private equity (where investors lend for the project) and debt, where the money is borrowed from an institution (e.g., bank).
3. In order to set the minimum gas tariff & pay for the equity, lenders/investors or financial institutions impose a hurdle rate or minimum acceptable rate of return (MARR) (i.e., rate at which NPV = 0 at IRR = WACC) to offset the cost of the investment (i.e., break-even). Generally the hurdle rate is equal to the company's cost of capital. In the current module cost of capital is taken as a weighted average, the WACC becomes the hurdle rate i.e., at IRR = WACC.
4. The internal rate of return (IRR) is the expected annual cash flows, (expressed as a percentage) that the investment can be expected to produce over and above the hurdle rate. Therefore for a project's acceptability, IRR > MARR.
5. The prime lending rate [r] is the rate imposed by the bank to service the project. The average risk free interest rate [r_f] represents the rate of return where there is no risk of defaulting/loss. This can be the interest offered as government/treasury

bonds. In actual practice, inflation is deducted from the interest rate to get a real risk free interest rate.

6. The expected market portfolio return [r_m] is a measure of the expected returns on the investments made. This is however correlated to market risks that is quantified by the beta [β] ratio and is useful in determining the volatility when arriving at the equity costs using CAPM. β<1 indicates that the security issued by the lender providing the equity, is less volatile (i.e., less risky) than the systematic market risks. Whereas a β>1 indicates that the security issued by the lender is more volatile (i.e., more risky) than the systematic market risks. For this module, β of 0.9 is taken assuming the venture is by an established firm with a history of low volatility. In actual practice, low β values need not necessarily show lower volatility if the security's value changes minimally in the short run but indicates a downward trend in the long run. Therefore the estimation of β requires historical data to arrive at a suitable value.
7. The term [$r_m - r_f$] is expressed as the equity risk premium which indicates the additional compensation for the risk taken by the investors/institutions towards providing equity. When multiplied by the β value, it accounts for the responsiveness of the venture to market changes. With higher β value, the gains/return on investment risks is higher. Similarly, with lower β value, the return on the investment risks is lower.
8. Assets over time depreciate in value due to wear and tear. Two methods used to depreciate an asset are the straight line depreciation method and declining balance depreciation method. In the straight line method, the depreciation rate is considered to be annually uniform, where the total life of the asset is divided by the operational years. Whereas in declining balance method, the asset is considered to lose more value in the early years and less in later years of operation. For this module, a straight line depreciation method is applied for the stipulated period of 25 years. It must be noted that the salvage value at the end of the operational period becomes zero. However the asset is considered to have a salvage value which is estimated at a depreciation rate of 10% for this exercise.
9. Depreciation also relates to taxable income, since the annual depreciation cost is neglected from the annual gross tax on revenue. Annual Cash flow is estimated by addition of the annual tax benefit from depreciation [ATBD] to the net annual revenue [NAR] before subtracting from the annual gross tax on revenue [AGTR] (i.e., Net Annual revenue – Annual Gross Tax on Revenue + Annual Tax Benefit from Depreciation). It may be noted that when companies tend to depreciate the asset value more, the tax payable decreases and increases the annual savings from depreciation.
10. The profit to Investment ratio (PIR) is a measure of investment efficiency. It is estimated as the ratio of NPV to the discounted capital costs (Disc. CAPEX). For a project to be viable, the PIR > 1.
11. The key differences between using NPV, IRR & PIR is that NPV and PIR measure a project value, while PIR allows screening a project against a company benchmark. For project to be viable, NPV>0, PIR>1 and IRR>WACC. The IRR value indicates the returns possible above which break even cannot be achieved and NPV<0.
12.

CAPEX Estimation

To estimate CAPEX, the pipeline and booster station tonnage and installed power is required to be estimated. The pipeline tonnage is estimated as,

$$Pipe\ Tonnage\ [tonnes] = \frac{\pi \times L \times \rho}{4 \times 1000}[OD^2 - ID^2] \quad (4)$$

$$PMC\ [\$] = Cost_{PMC} \times Pipe\ Tonnage \quad (5)$$

$$PCW\ [\$] = Cost_{PCW} \times Pipe\ Tonnage \quad (6)$$

$$LC\ [\$] = Cost_{LC} \times Pipe\ Tonnage \quad (7)$$

$$PIC\ [\$] = Cost_{PIC} \times Pipe\ Tonnage \quad (8)$$

The total pipeline [PC] cost becomes,

$$PC\ [\$] = PMC + PCW + LC + PIC \quad (9)$$

The Station Compressor cost is estimated as,

$$SC[\$] = SMC + SLC + SMiC + SLaC \quad (10)$$

Total Cost of Booster Station based on Comparison factor [C_f] for various GT/CC,

$$Total\ SC[\$] = SC \times C_f \quad (11)$$

The other machinery costs [OMC] including mainline valve stations [MVS], meter stations [MS] & pressure regulator stations [PRS] can be computed by accounting them as a percentage of the pipeline costs as,

$$MVS[\$] = PC \times \%factor \quad (12)$$

$$MS[\$] = PC \times \%factor \quad (13)$$

$$PRS[\$] = PC \times \%factor \quad (14)$$

For SCADA & Telecommunications, the cost is taken as a percentage of pipeline & station,

$$S\&T[\$] = [PC + SC] \times \%factor \quad (15)$$

The total machinery costs are calculated as,

$$OMC[\$] = MVS + MS + PRS + S\&T \quad (16)$$

The project costs involving environmental costs & permits [ECP], Right of Way Acquisition [ROWA], Engineering & Construction Management [ECM], Contingency [Con] is estimated similarly as,

$$\text{ECP}[\$] = [PC + SC] \times \%factor \quad (17)$$

$$\text{ROWA}[\$] = [PC + SC] \times \%factor \quad (18)$$

$$\text{ECM}[\$] = [PC + SC] \times \%factor \quad (19)$$

$$\text{Con}[\$] = [PC + SC] \times \%factor \quad (20)$$

The total project costs are calculated as,

$$Project[\$] = ECP + ROWA + ECM + Con \quad (21)$$

In addition to the pipeline costs, total station costs, other machinery costs & project costs, a working capital [WC] & AFUDC is included. AFUDC which stands for "*Allowance for funds used during construction*" represents costs associated with financing the project during various stages of construction.

Total Capex is now calculated as,

$$CAPEX[\$] = PC + Total\ SC + OMC + Project + WC + AFUDC \quad (22)$$

OPEX Estimation

To estimate annual OPEX, the basis is made on the annual salaries payable [S] and multiplying the annual salary by a %factor in this module.

$$\text{Salary}[\$] = S \quad (23)$$

$$\text{Payroll}[\$] = S \times \%factor \quad (24)$$

$$\text{Administrative Expenses}[\$] = S \times \%factor \quad (25)$$

$$\text{Vehicle Expenses}[\$] = S \times \%factor \quad (26)$$

$$\text{Office Expenses}[\$] = S \times \%factor \quad (27)$$

$$\text{Misc Materials \& Tools}[\$] = S \times \%factor \quad (28)$$

Similarly the related costs for compressor station maintenance is estimated as,

$$\text{Consumables}[\$] = S \times \%factor \quad (29)$$

$$\text{Periodic Maintenance}[\$] = S \times \%factor \quad (30)$$

$$\text{ROW Payments}[\$] = S \times \%factor \quad (31)$$

$$\text{Utilities}[\$] = S \times \%factor \quad (32)$$

$$\text{Gas Control}[\$] = S \times \%factor \quad (33)$$

$$\text{SCADA Maintenance}[\$] = S \times \%factor \quad (34)$$

$$\text{Corrosion Inspection}[\$] = S \times \%factor \quad (35)$$

$$\text{Cathodic Protect. Survey}[\$] = S \times \%factor \quad (36)$$

The Total O&M [OPEX] is computed as,

$$OPEX[\$] = S + Eq.23 + \cdots + Eq.36 \quad (37)$$

CAPEX & OPEX Cost Factors

Mainline Valve Stations

Block Valves on mainlines are installed to isolate pipeline sections for safety & maintenance such as blow down. In case of any pipeline rupture, block valves close to isolate the pipeline section and shut-off flow. For mainline valve stations a lump sum figure may be obtained from a construction contractor based on the size.

Meter & Pressure Regulating Stations

Meter Stations are installed for the purpose of measuring gas flow rates. Similar to mainline valve stations, a lump sum can be quoted for a given size that includes meters, valves, fittings, instrumentation & controls. Pressure regulation stations are installed for the purpose of reducing gas delivery pressure prior to delivering to the buyer. The price can be quoted as a lump sum figure.

SCADA & Telecommunications

SCADA allows pipeline process conditions data to be transmitted by electronic signals from remote control units installed on valves & meters. The signal transmission can be either by telephone lines, microwave or satellite communications. The pricing of these provisions can be expressed as a percentage of the total project cost, typically 2% to 5%.

Environmental Costs & Permits

Costs associated with environment and permits pertain to the modification of pipeline and booster station to prevent pollution. The costs also include compensation for acquisition of land to compensate for the areas disturbed due to pipeline construction. Permitting costs also include environmental study, environmental impact report and permits for rail road, stream & river crossings. These costs for a gas pipeline project can vary from 10% to 15% of the total project costs.

Right of Way (ROW) Acquisitions

Right of Way Acquisitions can be applied from private and government entities. The fee may be lump sum based at the time of acquisition, with additional fees paid on an annual basis. The initial cost for acquiring the ROW will be included in the capital cost, whereas annual ROW lease would be included as an O&M cost. For most gas pipelines, the initial ROW would be typically in the range of 6% to 10%.

Engineering & Construction Management

E&C management costs are associated with preparing design documents & drawings at both front end & detailed design phases. Details would also include specifications, operating manuals, purchase documents & equipment acquisition. Construction management costs would also include costs for field personal, transportation, rentals & all associated costs to manage the pipeline construction efforts. Engineering & Construction costs can typically vary from 15% to 20% of total project costs.

Other Project Costs

As the pipeline project progresses, it is expected that there could always be some unforeseen issues during project execution. To account for these uncertainties, contingencies & allowances for funds used during construction (AFDU) are also allotted. These costs can vary from 15% to 20% of the total project costs.

Salvage & Depreciation Costs

Pipeline infrastructures experience wear and tear thereby rendering depreciation to the project value. The salvage value [SV] based on the depreciation rate over the operational life of the project is computed as,

$$SV[\$] = [PC + SC + OMC + ROWA] \times [1 - d_r]^n \tag{38}$$

Where, n = Project Operational Life [Years]

The pipeline salvage deduction [SD] is computed as the difference between the Salvage value [SV] & the initial investment cost i.e., [PC+SC+OMC+ROWA]. Therefore the salvage costs is as follows,

$$\mathrm{SD}[\$] = [PC + SC + OMC + ROWA] - SV \quad (39)$$

Annual depreciation of the Asset [ADA]

$$\mathrm{ADA}[\$] = SD/n \quad (40)$$

Annual Tax Benefit from Depreciation [$ATBD$]

$$\mathrm{ATBD}[\$] = ADA \times T_{sale} \quad (41)$$

Cost of Capital

Based on the cost of capital structure of the project (that involves both debt & equity), the appraisal is made using the Weight average Cost of Capital [WACC] from which all the cash flows generated will be discounted to arrive at the Net Present Value [NPV]. WACC can be expressed as

$$\mathrm{WACC}[\%] = \frac{E}{C} k_e + \frac{D}{C} k_d \quad (42)$$

Where, $C = E + D$

The cost of debt [k_d] is taken as after-tax cost of debt and the cost of equity [k_e] is based on CAPM. The cost of debt is estimated as

$$k_d[\%] = r \times [1 - T_C] \quad (43)$$

The cost of equity is estimated as,

$$k_e[\%] = r_f + \left[\beta \times \left[r_m - r_f\right]\right] \quad (44)$$

The loan annual amortization cost [AAC] is computed as,

$$\mathrm{AAC}[\$] = \frac{[D \times CAPEX] \times r}{1 - \left[\frac{1}{1+r}\right]^n} \quad (45)$$

where, n=Loan period during project life [Yrs]

The Annual Revenue on Equity Capital [AREC] adjusted to effective corporate tax rate [T_C] is,

$$\mathrm{AREC}[\$] = \frac{E \times k_e}{1 - T_c} \quad (46)$$

Gas Tariff & Economic Viability

The minimum gas tariff required for economic viability is estimated based on the total cost of capital for an $NPV = 0$ at $IRR = WACC$ and yearly gas production. The pipeline operational availability is 8,040 hours/year. Therefore, the yearly gas production is,

$$\text{Gas Rate}[Mscf] = \frac{1000 \times Q_{MMscfd}}{24} \times Availability[hr] \quad (47)$$

The Annual cost of service [ACOS] is computed as,

$$\mathrm{ACOS}[\$] = OPEX + AAC + AREC + ADA \quad (48)$$

The Gas Transportation Tariff is estimated as,

$$\mathrm{Tariff}_{Trans}[\$/Mscf] = \frac{ACOS[\$]}{Gas\ Rate[MScf]} \quad (49)$$

Setting an initial guess sales tariff [Min_{Tariff}], the Net Annual Cash Flow [CF] is calculated for the condition NPV=0 and IRR=WACC. The Annual Revenue [AR] without tax is found as,

$$\mathrm{AR}[\$] = Min_{Tariff} \times Gas\ Rate[Mscf] \quad (50)$$

The Net Annual Revenue [NAR] becomes,

$$\mathrm{NAR}[\$] = AR - ACOS \quad (51)$$

The Annual Gross Tax on Revenue [AGTR] is,

$$\mathrm{AGTR}[\$] = NAR \times T_{sales} \quad (52)$$

The Tax Payable is calculated as,

$$\text{Tax Payable}[\$] = \mathrm{AGTR} - \mathrm{ADA} \quad (53)$$

The Annual Cash Flow [ACF] is calculated as,

$$\mathrm{CF}[\$] = \mathrm{NAR} - \mathrm{AGTR} + \mathrm{ATBD} \quad (54)$$

The Net Present Value (NPV) is calculated as,

$$\mathrm{NPV}[\$] = \left\{\sum_{n=1}^{n=n} [CF]_n [1 + i]^{-n}\right\} - CAPEX \quad (55)$$

Where,
n = Operational Life [years]
i = Internal Rate of Return (IRR) at WACC

In MS-Excel, the NPV can be calculated as,

$$= \left(NPV(WACC, CF_1 : CF_n)\right) - CAPEX \quad (56)$$

Similarly in MS-Excel, the Internal Rate of Return (IRR) can be calculated as,

$$= IRR(CF_1 : CF_n) \tag{57}$$

To calculate the "Min_{Tariff}" at NPV=0 and IRR=WACC, an MS-Excel Macro is written as,

Sub salestariff()

Range("NPV").GoalSeek Goal:=0.000001, ChangingCell:=Range("MinTariff")

End Sub

The Profit to Investment Ratio (PIR) is then calculated based on the discounted cash flow [DCF] as follows,

$$\text{PIR}[-] = \frac{Cash\ Outflow}{Cash\ Inflow} = \frac{[CF]_n[1+i]^{-n}}{NPV} \tag{58}$$

In the above PIR expression, after the discount factor $[1+i]^{-n}$ is applied to each year, the negative cash flow, i.e., cash outflow of each year is summed up and divided by the NPV value. At NPV=0 and WACC=IRR, PIR=0.

Calculations & Results

Based on the estimation methods, the following results are arrived at,

Table 5. Pipeline & Booster Station Costs

Description	%	Cost
Upstream Pipeline		
Total Material Costs	15%	$894,614
Coating & Wrapping	5%	$306,000
Labour Costs	17%	$1,020,000
Install. Cost per inch-km	64%	$3,892,719
Sub-Total	-	**$6,113,333**
Downstream Pipeline		
Total Material Costs	16%	$1,505,351
Coating & Wrapping	5%	$459,000
Labour Costs	16%	$1,530,000
Install. Cost per inch-km	63%	$5,839,078
Sub-Total	-	**$9,333,429**
Total Pipeline Costs	-	**$15,446,761**
Booster Compressor Station		
Station Material Cost	69%	$9,705,175
Station Labour Cost	22%	$3,089,159
Station Misc Cost	9%	$1,238,196
Station Land Cost	0.3%	$43,860
Sub-Total	-	**$14,076,390**
Total [Incl. Installed Cost Comparison]	**1.00**	**$14,076,390**
Other Machinery Costs		
Mainline Valve Stations	0.5%	$77,234
Meter Stations	0.5%	$77,234
Pressure Regulator Stations	0.1%	$15,447
SCADA & Telecomm	5.0%	$1,476,158
Total	-	**$1,646,072**
Project Costs		
Environmental Costs & Permits	15.0%	$4,428,473
Right of Way (ROW) Acquisition	10.0%	$2,952,315
Engineering & Construction Management	15.0%	$4,428,473
Contingency	10.0%	$2,952,315
Total	-	**$14,761,576**
Working Capital [WC]	-	$5,000,000
AFUDC	20.0%	$5,904,630
TOTAL CAPEX	-	**$56,835,429**

The Total O&M [OPEX] Costs estimated are as follows,

Table 6. Total O&M Costs

Description	%	Cost
Salaries	-	$1,000,000
Payroll Overhead	20.0%	$200,000
Administrative Expenses	50.0%	$500,000
Vehicle Expenses	8.0%	$80,000
Office Expenses	10.0%	$100,000
Misc Materials & Tools	10.0%	$100,000

Compressor Station Maintenance		
Consumable Materials	6.0%	\$60,000
Periodic Maintenance	18.0%	\$180,000
ROW Payments	40.0%	\$400,000
Utilities	18.0%	\$180,000
Gas Control	12.0%	\$120,000
SCADA Maintenance	25.0%	\$250,000
Internal Corrosion Inspection	30.0%	\$300,000
Cathodic Protection Survey	12.0%	\$120,000
TOTAL O&M COSTS	**-**	**\$3,590,000**

Salvage Costs

The Salvage Costs estimated are as follows,

$$SV[\$] = [15{,}446{,}761 + 14{,}076{,}390 + 1{,}646{,}072 + 2{,}952{,}315] \times [1 - 0.08]^{25} = \$4{,}243{,}501 \quad (59)$$

The pipeline salvage deduction is,

$$SD[\$] = [15{,}446{,}761 + 14{,}076{,}390 + 1{,}646{,}072 + 2{,}952{,}315] - 4{,}243{,}501 = \$29{,}878{,}037 \quad (60)$$

The Annual depreciation of the asset is,

$$ADA[\$] = \$29{,}878{,}037/25 = \$1{,}195{,}121 \quad (61)$$

The Annual Tax benefit from depreciation is,

$$\mathrm{ATBD}[\$] = 1{,}195{,}121 \times 0.35 = \$418{,}293 \quad (62)$$

Cost of Capital

The cost of debt is first estimated as,

$$k_d[\%] = 0.17 \times [1 - 0.3] = 11.90\% \quad (63)$$

The cost of equity is estimated as,

$$k_e[\%] = 0.17 + \left[0.9 \times [0.24 - 0.17]\right] = 23.3\% \quad (64)$$

The weighted average capital cost [WACC] is,

$$\mathrm{WACC}[\%] = 0.4 \times 23.3 + 0.6 \times 11.9 = 16.46\% \quad (65)$$

Loan Amortization Costs

The loan annual amortization cost [AAC] is computed as,

$$\mathrm{AAC} = \frac{[0.6 \times 56{,}835{,}429] \times 0.17}{1 - \left[\frac{1}{1+0.17}\right]^{25}} = \$5{,}913{,}957 \quad (66)$$

The Annual Revenue on Equity Capital is,

$$\mathrm{AREC} = \frac{0.4 \times 56{,}835{,}429 \times 0.233}{1 - 0.3} = \$7{,}567{,}231 \quad (67)$$

Gas Tariff, NPV, IRR & PIR Estimates

Taking the annual availability of 8,040 hours, the yearly gas production is as follows for Year 7. It is to be noted calculations are shown for a set of sample data.

$$\text{Gas Rate} = \frac{1000 \times 122.3}{24} \times 8{,}040 = 40{,}981{,}711\,MScf \quad (68)$$

The Annual O&M Costs considering inflation is,

$$\text{Annual O\&M} = \$3{,}590{,}000 \times 1.02^{7-1} = \$4{,}042{,}923 \quad (69)$$

The Annual Cost of Service [ACOS] is,

$$\mathrm{ACOS}[\$] = 4{,}042{,}923 + 5{,}913{,}957 + 7{,}567{,}231 + 1{,}195{,}121 = \$18{,}719{,}233 \quad (70)$$

The Gas Transportation Tariff is estimated as,

$$\mathrm{Tariff}_{Trans} = \frac{18{,}719{,}233}{40{,}981{,}711} = \$0.4568/Mscf \quad (71)$$

With a guess value of \$0.998/Mscf (MS Excel Rounded-off value) as minimum gas tariff, the Annual Revenue without tax rate included is,

$$\mathrm{AR} = 0.998 \times 40{,}981{,}711 \approx \$40{,}894{,}302 \quad (72)$$

The Net Annual Revenue [NAR] becomes,

$$\mathrm{NAR} = 40{,}894{,}302 - 18{,}719{,}233 = \$22{,}175{,}069 \quad (73)$$

The Annual Gross Tax on Revenue [AGTR] is,

$$\mathrm{AGTR} = 22{,}175{,}069 \times 0.35 = \$7{,}761{,}274 \quad (74)$$

The Tax Payable is calculated as,

$$\text{Tax Payable} = 7{,}761{,}274 - 1{,}195{,}121 = \$6{,}566{,}153 \quad (75)$$

The Annual Cash Flow [ACF] for Year 7 is calculated as,

$$\mathrm{CF}[\$] = 22{,}175{,}069 - 7{,}761{,}274 + 418{,}293 = \$14{,}832{,}088 \quad (76)$$

Upon calculating CF for all years, using MS-Excel for NPV=0, IRR is calculated as 16.46%. The Discounted Cash Flow [DCF] at IRR=WACC is,

$$\mathrm{DCF} = 14{,}832{,}087 \times [1 + 0.1646]^{-7} = \$5{,}104{,}636 \quad (77)$$

With the calculations performed for NPV=0 at IRR=WACC, the minimum sales gas tariff comes to $0.998/Mscf or, ~ $1/Mscf [Refer to Appendix A] and PIR is 0.

For end of operational life [Year 25], the salvage value of the asset is treated as part of the Annual Revenue [AR]; therefore the cost estimation for Year 25 alone changes to,

$$AR_{Year\ 25}[\$] = \left[Min_{Tariff} \times Gas\ Rate[Mscf]\right] + SV \quad (78)$$

The calculations are similarly repeated for Sales Tariff rate of $2/Mscf to check for NPV, IRR, PIR and estimate payback period. The results are as follows,

Table 7. Economic Analysis for $2/Mscf

Description	Cost
Total Annual Cash Flow	$664,457,040
Net Present Value [NPV]	$127,053,012
IRR	45.80%
PIR	2.24

Payback Period

With detailed NPV calculations [Refer to Appendix A], the payback period for $2/Mscf can be computed as,

$$\text{Payback Period} = Year\ [Cumulative\ Annual\ CF = Total\ CAPEX] \quad (79)$$

The payback period can be estimated graphically as follows,

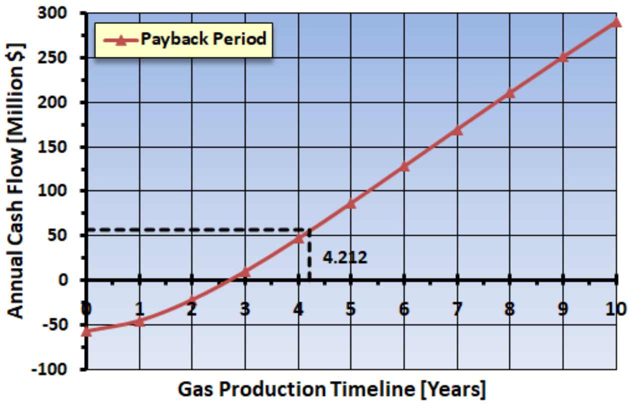


Figure 2. Payback Period – Gas Tariff at $2/Mscf

$$\text{Payback Period} = 4.212\ Yrs \quad (80)$$

Note: Estimates shown were computed on an MS-Excel sheet which takes decimal places until the 10th decimal, while hand calculations were shown by rounding off decimal places.

References & Further Reading

1. "*Gas Pipeline Hydraulics*", E. Sashi Menon, 2005, Taylor & Francis
2. *"Pipeline Compressor Station Construction Cost Analysis",* Y. Zhao, Z. Rui, Int J. Oil, Gas and Technology, Vol. 8No. 1, 2014 (https://www.researchgate.net/publication/275590240)
3. *"Economic Analysis of Gas Pipeline Project in Nigeria"*, Ahmed Adamu, Journal of Economics and Sustainable Development, Vol.8, No.2, 2017

Appendix A

Natural Gas Pipeline & Booster Station Economics

Pipeline Construction Details & Material Costs

Sr. No.	Pipeline Description	Pipeline Length		Pipeline Wall Thickness [WT]		Pipeline Outer Diameter [OD]		Pipeline ID	Pipeline Material	Pipeline Density	Pipeline Volume	Pipeline Tonnage	Pipeline Cost/tonne	Total Material Cost [PMC]	Coating & Wrapping Cost [PCW]		Labour Cost [LC]		Pipeline Installation Cost per inch-dia-km [PIC]		Unit Cost of Installation
[-]	[-]	[m]	[km]	[mm]	[m]	[in]	[m]	[m]	[-]	[kg/m³]	[m³]	[kg]	[$/tonne]	[$]	[$/m]	[$]	[$/m]	[$]	[$/in/km]	[$]	[$/m]
1	Upstream	20,400	20.4	6.35	0.0064	16.00	0.4064	0.3937	CS	7,850	163	1,278	$700	$894,614	$15	$306,000	$50	$1,020,000	$11,926	$3,892,719	$191
2	Downstream	30,600	30.6	7.14	0.0071	16.00	0.4064	0.3921	CS	7,850	274	2,151	$700	$1,505,351	$15	$459,000	$50	$1,530,000	$11,926	$5,839,078	$191
Total		51,000	51.00								437	3,429		$2,399,965		$765,000		$2,550,000		$9,731,797	

Booster Station Details

Sr. No.	Station Description	Absorbed Power	Margin on Absorbed Power & Installed Capacity			Station Material Cost [SMC]		Station Labour Cost [SLC]		Station Misc Cost [SMiC]		Station Land Cost [SLaC]	
[-]	[-]	[kW]	[%]	[kW]	[hP]	[$/hP]	[$]	[$/hP]	[$]	[$/hP]	[$]	[$/hP]	[$]
1	Station 1	2,097	20.00	2,517	3,374	$2,877	$9,705,175	$916	$3,089,159	$367	$1,238,196	$13	$43,860
Total		2,097		2,517	3,374		$9,705,175		$3,089,159		$1,238,196		$43,860

Total CAPEX Cost - Natural Gas Pipeline Project

Sr. No.	Description	% Factor	Cost
1	**Pipeline**		
1.1	***Upstream***	-	-
1.1.1	*Total Material Cost [PMC]*	15%	$894,614
1.1.2	*Coating & Wrapping Cost [PCW]*	5%	$306,000
1.1.3	*Labour Cost [LC]*	17%	$1,020,000
1.1.4	*Pipeline Installation Cost per inch-dia-km [PIC]*	64%	$3,892,719
1.1.5	***Sub-Total***		**$6,113,333**
1.2	***Downstream***		
1.2.1	*Total Material Cost [PMC]*	16%	$1,505,351
1.2.2	*Coating & Wrapping Cost [PCW]*	5%	$459,000
1.2.3	*Labour Cost [LC]*	16%	$1,530,000
1.2.4	*Pipeline Installation Cost per inch-dia-km [PIC]*	63%	$5,839,078
1.2.5	***Sub-Total***	-	**$9,333,429**
1.2.6	***Total Pipeline Costs***	-	**$15,446,761**
2	**Booster Station Compressor**	Single GT/CC	
2.1	*Station Material Cost [SMC]*	69%	$9,705,175
2.2	*Station Labour Cost [SLC]*	22%	$3,089,159
2.3	*Station Misc Cost [SMiC]*	9%	$1,238,196
2.4	*Station Land Cost [SLaC]*	0.3%	$43,860
2.5	***Sub-Total***	-	**$14,076,390**
2.6	***Total [Incl. Installed Cost Comparison]***	1.00	**$14,076,390**
3	**Other Machinery Costs**		
3.1	*Mainline Valve Stations [MVS]*	0.5%	$77,234
3.2	*Meter Stations [MS]*	0.5%	$77,234
3.3	*Pressure Regulator Stations [PRS]*	0.1%	$15,447
3.4	*SCADA & Telecommunications [S&T]*	5.0%	$1,476,158
3.5	***Total***		**$1,646,072**
4	**Project Costs**		
4.1	*Environmental Costs & Permits [ECP]*	15.0%	$4,428,473
4.2	*Right of Way (ROW) Acquisition[ROWA]*	10.0%	$2,952,315
4.3	*Engineering & Construction Management [ECM]*	15.0%	$4,428,473
4.4	*Contingency [Con]*	10.0%	$2,952,315
4.5	***Total***		**$14,761,576**
5	**Working Capital [WC]**	-	$5,000,000
6	**AFUDC**	20.0%	$5,904,630
7	**TOTAL CAPEX**		**$56,835,429**

Total O&M Cost - Natural Gas Pipeline Project

Sr. No.	Description	% Add	Cost
1	*Salaries*	-	$1,000,000
2	*Payroll Overhead*	20.0%	$200,000
3	*Administrative Expenses*	50.0%	$500,000
4	*Vehicle Expenses*	8.0%	$80,000
5	*Office Expenses*	10.0%	$100,000
6	*Misc Materials & Tools*	10.0%	$100,000
7	**Compressor Station Maintenance**		
7.1	*Consumable Materials*	6.0%	$60,000
7.2	*Periodic Maintenance*	18.0%	$180,000
7.3	*ROW Payments*	40.0%	$400,000
7.4	*Utilities*	18.0%	$180,000
7.5	*Gas Control*	12.0%	$120,000
7.6	*SCADA Maintenance*	25.0%	$250,000
7.6	*Internal Corrosion Inspection*	30.0%	$300,000
7.7	*Cathodic Protection Survey*	12.0%	$120,000
8	**TOTAL O&M Costs**		**$3,590,000**

Project Net Value After Salvage

Initial Investment Cost [IIC] (Pipeline+Compressor+Other)	$34,121,538
Operating Lifetime [Years]	25
Depreciation Rate [dr]	8.00%
Salvage Value [SV]	$4,243,501
Pipeline Salvage Deductions [SD]	$29,878,037
Annual Depreciation of Asset [ADA]	$1,195,121
Sales Tax Rate on Sold Gas [T_{sale}]	35.00%
Annual Tax Benefit from Depreciation [ATBD]	$418,293

Cost of Capital

Capital Structure - Total Value of Debt [D]	60.00%
Capital Structure - Total Value of Equity [E]	40.00%
Effective Corporate Tax Rate[T_c]	30.00%
Cost of Debt [After Taxes]	
Lender's Avg. Prime Lending Rate [r]	17.00%
Cost of Debt [k_d] [After Tax]	11.90%
Cost of Equity	
Avg. Risk Free Interest Rate [r_f]	17.00%
Expected Market Portfolio Return [r_m]	24.00%
Equity Risk Premium (ERP) [$r_m - r_f$]	7.00%
Beta [β]	0.90
Cost of Equity [k_e] [Based on CAPM]	23.30%
Weighted Average Cost of Capital [WACC]	**16.46%**

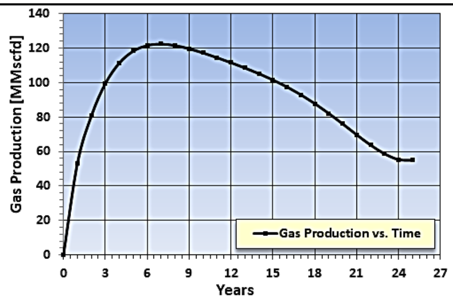
Cost of Service [O&M Excluded]

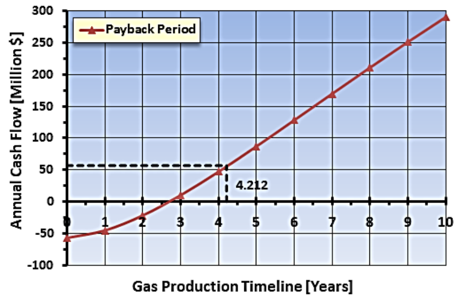
Loan Amortization on Debt Capital	
Total Capital Cost	$56,835,429
Debt Capital	$34,101,257
Loan Period [Years]	25
Annual Loan Amortization Cost [AAC]	**$5,913,957**
Annual Revenue on Equity Capital	
Equity Capital	$22,734,172
Annual Revenue on Equity Capital	$5,297,062
Adjusted Annual Revenue [with Tax Rate]	**$7,567,231**

Annual O&M Inflation

Annual O&M Inflation	2.00%

Appendix B





Gas Tariff & Economic Viability

									Cash Flow [CF] - Estimation of Minimum Gas Sales Tariff								*Cash Flow [CF] - At Sales Tariff*				
Year	Operational Availability	Gas Production		Cumulative Gas Production	Annual O&M	Annual Cost of Service [ACOS]	Transportation Tariff	Minimum Sales Tariff	Annual Revenue [AR]	Net Annual Revenue [NAR]	Annual Gross Tax on Revenue [AGTR]	Tax Payable	Annual Cash Flow [CF]	Cumulative Cash Flow	Discounted Cash Flow [DCF]	PIR Estimate	Sales Tariff	Annual Cash Flow [CF]	Cumulative Cash Flow	Discounted Cash Flow [DCF]	PIR Estimate
[-]	[hrs]	[MMscfd]	[Mscf]	[Mscf]	[$]	[$]	[$/Mscf]	[$/Mscf]	[$]	[$]	[$]	[$]	[$]	[$]	[$]	[$]	[$/Mscf]	[$]	[$]	[$]	[$]
0	0	0	0	0	$0	$0	Calculate Min Sales Tariff		-	-	-	-	-$56,835,429	-$56,835,429	-$56,835,429	-$56,835,429	-	-$56,835,429	-$56,835,429	-$56,835,429	-$56,835,429
1	8040	53.1	17,801,022	17,801,022	$3,590,000	$18,266,310	$1.03	$1.00	$17,763,055	-$503,255	-$176,139	-$1,371,261	$91,177	-$56,744,252	$78,290	$0	$2.00	$11,686,520	-$45,148,909	$8,015,227	$0
2	8040	80.7	27,030,293	44,831,315	$3,661,800	$18,338,110	$0.68	$1.00	$26,972,640	$8,634,530	$3,022,086	$1,826,964	$6,030,737	-$50,713,515	$4,446,485	$0	$2.00	$23,637,902	-$21,511,007	$11,119,112	$0
3	8040	99.4	33,296,561	78,127,876	$3,735,036	$18,411,346	$0.55	$1.00	$33,225,543	$14,814,197	$5,184,969	$3,989,847	$10,047,521	-$40,665,994	$6,361,046	$0	$2.00	$31,736,446	$10,225,439	$10,238,824	$0
4	8040	111.3	37,297,553	115,425,429	$3,809,737	$18,486,047	$0.50	$1.00	$37,218,002	$18,731,955	$6,556,184	$5,361,063	$12,594,063	-$28,071,931	$6,846,344	$0	$2.00	$36,889,181	$47,114,621	$8,162,466	$0
5	8040	118.2	39,613,268	155,038,696	$3,885,931	$18,562,241	$0.47	$1.00	$39,528,777	$20,966,536	$7,338,288	$6,143,166	$14,046,541	-$14,025,390	$6,556,702	$0	$2.00	$39,850,084	$86,964,704	$6,047,588	$0
6	8040	121.5	40,716,223	195,754,919	$3,963,650	$18,639,960	$0.46	$1.00	$40,629,380	$21,989,420	$7,696,297	$6,501,176	$14,711,416	$686,026	$5,896,493	$0	$2.00	$41,233,409	$128,198,113	$4,291,734	$0
7	8040	122.3	40,981,711	236,736,631	$4,042,923	$18,719,233	$0.46	$1.00	$40,894,302	$22,175,069	$7,761,274	$6,566,153	$14,832,087	$15,518,113	$5,104,636	$0	$2.00	$41,527,016	$169,725,128	$2,964,455	$0
8	8040	121.5	40,698,046	277,434,676	$4,123,782	$18,800,091	$0.46	$1.00	$40,611,241	$21,811,150	$7,633,902	$6,438,781	$14,595,540	$30,113,653	$4,313,262	$0	$2.00	$41,105,692	$210,830,821	$2,012,550	$0
9	8040	119.6	40,076,815	317,511,491	$4,206,257	$18,882,567	$0.47	$1.00	$39,991,336	$21,108,769	$7,388,069	$6,192,948	$14,138,992	$44,252,645	$3,587,793	$0	$2.00	$40,244,484	$251,075,304	$1,351,393	$0
10	8040	117.2	39,263,134	356,774,625	$4,290,382	$18,966,692	$0.48	$1.00	$39,179,390	$20,212,698	$7,074,444	$5,879,323	$13,556,546	$57,809,191	$2,953,801	$0	$2.00	$39,132,017	$290,207,321	$901,235	$0
11	8040	114.5	38,345,892	395,120,517	$4,376,190	$19,052,500	$0.50	$1.00	$38,264,105	$19,211,605	$6,724,062	$5,528,940	$12,905,836	$70,715,027	$2,414,579	$0	$2.00	$37,883,828	$328,091,149	$598,398	$0
12	8040	111.5	37,368,007	432,488,525	$4,463,714	$19,140,024	$0.51	$1.00	$37,288,306	$18,148,282	$6,351,899	$5,156,777	$12,214,676	$82,929,703	$1,962,278	$0	$2.00	$36,555,687	$364,646,836	$396,024	$0
13	8040	108.5	36,336,675	468,825,200	$4,552,988	$19,229,298	$0.53	$1.00	$36,259,174	$17,029,876	$5,960,456	$4,765,335	$11,487,712	$94,417,415	$1,584,657	$0	$2.00	$35,156,927	$399,803,763	$261,221	$0
14	8040	105.2	35,233,621	504,058,822	$4,644,048	$19,320,358	$0.55	$1.00	$35,158,472	$15,838,114	$5,543,340	$4,348,219	$10,713,067	$105,130,482	$1,268,933	$0	$2.00	$33,663,768	$433,467,531	$171,550	$0
15	8040	101.6	34,025,350	538,084,172	$4,736,929	$19,413,239	$0.57	$1.00	$33,952,778	$14,539,540	$5,088,839	$3,893,717	$9,868,993	$114,999,475	$1,003,740	$0	$2.00	$32,032,643	$465,500,173	$111,957	$0
16	8040	97.5	32,673,399	570,757,571	$4,831,667	$19,507,977	$0.60	$1.00	$32,603,711	$13,095,733	$4,583,507	$3,388,385	$8,930,519	$123,929,994	$779,917	$0	$2.00	$30,213,526	$495,713,700	$72,425	$0
17	8040	93.0	31,144,587	601,902,158	$4,928,301	$19,604,611	$0.63	$1.00	$31,078,159	$11,473,549	$4,015,742	$2,820,620	$7,876,099	$131,806,094	$590,617	$0	$2.00	$28,163,259	$523,876,958	$46,302	$0
18	8040	87.8	29,421,265	631,323,424	$5,026,867	$19,703,177	$0.67	$1.00	$29,358,513	$9,655,337	$3,379,368	$2,184,246	$6,694,261	$138,500,355	$431,043	$0	$2.00	$25,858,873	$549,735,831	$29,158	$0
19	8040	82.1	27,511,571	658,834,995	$5,127,404	$19,803,714	$0.72	$1.00	$27,452,892	$7,649,179	$2,677,213	$1,482,091	$5,390,259	$143,890,614	$298,024	$0	$2.00	$23,310,921	$573,046,753	$18,028	$0
20	8040	76.0	25,459,677	684,294,672	$5,229,952	$19,906,262	$0.78	$1.00	$25,405,374	$5,499,112	$1,924,689	$729,568	$3,992,715	$147,883,329	$189,554	$0	$2.00	$20,576,802	$593,623,555	$10,914	$0
21	8040	69.7	23,356,039	707,650,711	$5,334,551	$20,010,861	$0.86	$1.00	$23,306,223	$3,295,362	$1,153,377	-$41,745	$2,560,278	$150,443,607	$104,370	$0	$2.00	$17,774,084	$611,397,639	$6,466	$0
22	8040	63.7	21,347,654	728,998,365	$5,441,242	$20,117,552	$0.94	$1.00	$21,302,122	$1,184,570	$414,600	-$780,522	$1,188,263	$151,631,870	$41,593	$0	$2.00	$15,093,834	$626,491,473	$3,766	$0
23	8040	58.7	19,648,305	748,646,671	$5,550,067	$20,226,377	$1.03	$1.00	$19,606,398	-$619,979	-$216,993	-$1,412,114	$15,306	$151,647,177	$460	$0	$2.00	$12,813,945	$639,305,417	$2,193	$0
24	8040	55.4	18,548,816	767,195,486	$5,661,068	$20,337,378	$1.10	$1.00	$18,509,253	-$1,828,125	-$639,844	-$1,834,965	-$769,989	$150,877,188	-$19,872	-$19,872	$2.00	$11,312,457	$650,617,875	$1,328	$0
25	8040	55.0	18,427,298	785,622,784	$5,774,290	$20,450,600	$1.11	$1.00	$22,631,496	$2,180,896	$763,314	-$431,808	$1,835,875	$152,713,063	$40,684	$0	$2.00	$13,839,166	$664,457,040	$1,114	$0
Total				785,622,784	$114,988,776	$481,896,522				$306,294,121	$107,202,942	$77,324,905	$152,713,063		PIR	0.00		$664,457,040		PIR	2.24
				22.25	BSm^3							NPV	$0				NPV	$127,053,012	Payback Period	Pay Back Period	
				1 MMBtu =	1.1364	Msct	1 Mscf =	0.8800	MMBtu			IRR	16.46%				IRR	45.80%		4.212	Years

Module 20

Evaluating Pipeline Operational Integrity - Sand Production

Piping systems associated with production, transporting oil & gas, water/gas injection into reservoirs, experience wear & tear with time & operations. There would be metal loss due to erosion, erosion-corrosion and cavitation to name a few. The presence of corrosion defects provides a means for localized fractures to propagate causing pipe ruptures & leakages. This also reduces the pipe/pipeline maximum allowable operating pressure [MAOP].

The following document covers methods by DNV standards to quantitatively estimate the erosion rate for ductile pipes and bends due to the presence of sand. It is to be noted that corrosion can occur in many other scenarios such as pipe dimensioning, flow rate limitations, pipe performance such as pressure drop, vibrations, noise, insulation, hydrate formation and removal, severe slug flow, terrain slugging and also upheaval buckling. However these aspects are not covered in this document.

Based on the erosional rates of pipes and bends, the Maximum Safe Pressure/Revised MAOP is evaluated based on a Level 1 Assessment procedure for the remaining strength of the pipeline. The Level 1 procedures taken up in this module are RSTRENG 085dL method, DNVGL RP F-101 (Part-B) and PETROBRAS's PB Equation.

General Notes & Assumptions

1. In evaluating corrosion defects, the generally accepted or traditional approach is the ASME B31G code which gives overly conservative results in terms of lower burst pressures with which operators repair/replace the corroded pipe/pipeline segments. This represents higher maintenance costs necessitating the need to follow a procedure that meets pipeline integrity requirements while also lowering maintenance, repair & replacement costs.
2. To assess pipeline integrity, standard corrosion assessment procedures are classified on three levels – Level 1, Level 2 and Level 3. Level 1 procedure represents longitudinal area of metal loss based on the maximum defect depth and overall defect length. The ASME B31G, RSTRENG 085dL, and DNVGL RP F-101 method for single defect can be classified as Level 1 methods. Level 2 procedure represents longitudinal area of metal loss based on the defect depth profile. The RSTRENG Effective Area method and DNVGL RP F-101 method for complex shaped defects can be classified as Level 2 methods. Level 3 assessment methods involve using Finite element methods (FEM) provided the FEM model is validated against experimental results.
3. Corrosion failures are caused by two main mechanisms – Leakage resulting in a relatively small loss of product and Rupture causing a sudden release of pressure which propagates in isolation.
4. To understand corrosion assessment procedures, two terms come into play – Folias Bulging factor [M_T] and flow stress [σ_f]. Folais factor represents the bulging effect of a shell surface that is thinner in wall thickness [*WT*] than the surrounding shell. It takes into account the work-hardening effect, i.e., the increase in the stress concentration levels as the corrosion defect begins to bulge before eventually causing a failure. The flow stress is the stress at which the corrosion defect is predicted to cause a failure.

5. In pipeline assessment literature, SMYS and yield strength are used differently. Specific Minimum Yield Strength (SMYS) is the absolute minimum yield strength for a particular material grade specified by ASTM standards. Whereas, yield strength is obtained from mill conducted tensile tests. Wherever possible yield strength should be used. In cases, where the yield strength value is not available, SMYS can be used instead.
6. When a corrosion defect occurs inside a pipe/pipeline, the defect tends to propagate longitudinally. ASME B31G mandates a maximum allowable longitudinal length [L_M] for a given defect depth [d]. As per Modified ASME B31G method i.e., 085dL method, defects are classified as Long defect and short defect based on the condition, $L_S^2/Dt = 50$, Where D = Pipeline Outer diameter (OD) and t = pipeline nominal wall thickness. When field measured defect's longitudinal length, $L < L_S$, the defect is termed as short defect. When $L > L_S$, the defect is termed as long defect. The DNVGL RP F-101 method does not classify defects in relation to their longitudinal length. The pressure strength of long defects is a function of the longitudinal defect length [L]. The Longer the defect, lower is the failure pressure However a limit exists in the value of L, beyond which any large increase in the longitudinal defect length, L produces very little reduction in the failure pressure.
7. Long Internal defects are one of the various causes for geometry corrosion induced damage that occur in oil & gas pipelines. These occur on the pipe/pipeline bottom due to accumulation of liquids including water. Whereas long external defects are caused on the pipeline's outer surface due to loss of protective coatings.
8. ASME B31G assumes a parabolic profile across the area of the defect, i.e., Area of defect = 2/3×d×L, where, d = Defect depth and L = Defect longitudinal length. Whereas with the RSTRENG 085dL method, the defect area is approximated as 85% of the peak depth, i.e., by using a factor of 0.85, i.e., Defect Area = 0.85×d×L.

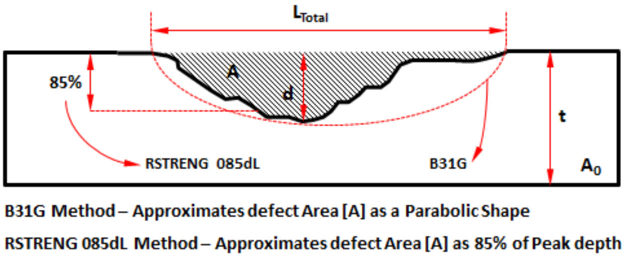


Figure 1. Corrosion Shape Approximation

9. The potential for sand particles to get carried from the formation to well bore in oil & gas wells is subjected to the reservoir geology. With the onset of water formation or rapid change in well conditions, there is sand formation. Employing a zero rate of sand production would be economically infeasible. Therefore sand management programmes are put in place whereby upstream facilities are equipped with sand traps with necessary safeguards that aid in achieving an acceptable sand rate. The standard used for this module is DNVGL RP O501 which provides empirical models that cover plain erosion & not the combined effects of corrosion-erosion, droplet erosion & cavitation. The module therefore considers plain erosion which leads to corrosion pits in the pipeline & the associated MAOP is computed using the standard corrosion assessment methods.
10. When applying the original ASME B31G method in simplified form (Appendix L of ASME B31.8), the Safe Operating Pressure given as P' must first be calculated using the pressure corresponding to a hoop stress equal to 100% of SMYS for the operating pressure, P. The resulting P' is the estimated

failure pressure, which must then be divided by the design factor/desired factor of safety to obtain the correct "Safe Operating Pressure".

Case Study: Problem Statement

30 MMscfd of well fluids at 40 bara and 40^{0}C is transported through an 8" DN carbon steel flowline from the well head to a trunk line. The process & mechanical details are,

Table 1. Process & Mechanical Details

Parameter	Value	Unit
Operational Life of Pipeline	25	Years
Location of Gas Pipeline	Deserted	-
Location Class	Class 1, Div 2	-
Design Factor [F]	0.72	-
Pipeline Joining Method	ERW	-
Longitudinal Joint Factor [E]	1.0	-
API 5L Spec	PSL1 X65	-
Ultimate Tensile Strength [σ_u]	530	MPa
SMYS [S]	448	MPa
Design Pressure [DP]	44	bara
Design Temperature [DT]	100	^{0}C
De-Rating Factor [T]	1.00	-
Pipeline Diameter [DN]	8.625	in
Corrosion Allowance [CA]	1.0	mm
Gas Flow Rate [m_g]	31,657	kg/h
Liquid Flow Rate [m_l]	14,928	kg/h
Gas Density [ρ_g]	42.0	kg/m^3
Liquid Density [ρ_l]	713.2	kg/m^3
Gas Viscosity [μ_g]	1.34E-05	kg/m.s
Liquid Viscosity [μ_l]	4.72E-04	kg/m.s
Mixture Viscosity [μ_m]	2.58E-05	kg/m.s
Sand Content [ppmW]	50.0	ppmW
Average Sand Particle Size	300	µm
No. of Pipe Diameter [90^0 Long Elbow]	1.5	[-]
Inclined Pipe Impact angle [α]	30^0	degrees

Pipeline Wall Thickness [WT] Estimation

Based on the ASME B31.8 code the wall-thickness formula is stated as,

$$t = \frac{DP \times OD}{2 \times F \times E \times T \times SMYS} \quad (1)$$

Where,

t = Minimum design wall thickness [in]

DP = Pipeline Design Pressure [psi]

OD = Pipeline Outer Diameter [in]

$SMYS$ = Specific Minimum Yield Stress [psi]

F = Design Factor [-]

E = Longitudinal Weld Joint Factor [E]

T = Temperature De-rating Factor [-]

Applying the ASME B31.8 correlation, the calculated wall thickness becomes,

$$t = \frac{[44 \times 14.5] \times 8.625 \times 25.4}{2 \times 0.72 \times 1 \times 1 \times [448 \times 145.038]} = 1.49\ mm \quad (2)$$

The total WT including CA of 1.0 mm is,

$$t_{total} = 1.49\ mm + 1.0\ mm = 2.49\ mm \quad (3)$$

The Selected WT based on API5L is,

$$t_{selected} = 3.175\ mm\ [0.125\ in] \quad (4)$$

The revised design pressure based on the selected Wall Thickness [WT] is,

$$DP = \frac{2 \times F \times E \times T \times SMYS \times t}{OD} \quad (5)$$

$$DP = \frac{2 \times 0.72 \times 1 \times 1 \times [448 \times 145.038] \times 3.175}{8.625 \times 14.5 \times 25.4} = 93.5\ bara \quad (6)$$

The pipeline inside diameter [ID] becomes,

$$ID = OD - [2 \times WT] \quad (7)$$

$$ID = [8.625 \times 25.4] - [2 \times 3.175] \approx 212.73\ mm \quad (8)$$

The pipeline cross section area [A_t] becomes,

$$A_p = \frac{\pi D^2}{4} = \frac{\pi}{4} \times \left[\frac{212.73}{1000}\right]^2 = 0.0355\ m^2 \quad (9)$$

Therefore the gas velocity is estimated as,

$$V_g = \frac{Q_g}{A_p} = \frac{31{,}657}{3600 \times 42 \times 0.0355} = 5.887\ m/s \quad (10)$$

The liquid Velocity is estimated as,

$$V_l = \frac{Q_L}{A_p} = \frac{14{,}928}{3600 \times 713.2 \times 0.0355} = 0.164\ m/s \quad (11)$$

Well Fluids Mixture Properties

The mixture density and mixture viscosity of the well fluids can be determined as follows.

$$\rho_m = \frac{\rho_g V_g + \rho_l V_l}{V_g + V_l} \quad (12)$$

$$\mu_m = \frac{\mu_g V_g + \mu_l V_l}{V_g + V_l} \quad (13)$$

Therefore applying the above correlations,

$$\rho_m = \frac{[42 \times 5.887] + [713.2 \times 0.164]}{5.887 + 0.164} = 60.2 \frac{kg}{m^3} \quad (14)$$

$$\mu_m = \frac{[0.0000134 \times 5.887] + [0.000472 \times 0.164]}{5.887 + 0.164} = 0.0000258 \frac{kg}{m.s} \quad (15)$$

Inclined Pipe Erosion Rate – DNV RP O501

The flowline profile over the terrain would have inclined sections. With the onset of water production from the wells, quartz sand particles from wells [50 ppmW, 300 μm, 2,650 kg/m^3] impinge at an impact angle of 30^0. As per DNVGL RP O501 [Rev. 2015], for ductile materials, the maximum erosion occurs for impact angles in the range of 15^0 to 30^0, whereas brittle materials experience maximum erosion at normal impact angle. The erosive wear can be estimated as,

$$E = \frac{m_p \times K \times U_p^n \times F(\alpha)}{\rho_t \times A_t} \times [3.15 \times 10^{10}] \quad (16)$$

Where,

m_p = Sand Flow rate [kg/s]

K=Material Constant (2×10^{-9} for Steel Grades)

n =Material Constant (2.6 for Steel Grades)

U_p = Particle Velocity [m/s] ($V_g + V_l$)

ρ_t = Pipeline density [kg/m^3]

A_t = Pipeline Area exposed to Erosion [m^2]

$F(\alpha)$ = Function characteristic of ductility [-]

The value of F(α) is calculated as,

$$F(\alpha) = 0.6 \times \left[Sin(\alpha) + 7.2\left(Sin(\alpha) - Sin^2(\alpha)\right)\right]^{0.6} \times [1 - e^{-20\alpha}] \quad (17)$$

For the condition, F(α)∈ [0, 1] for α ∈ [0, π/2]

Note: 1 mil = 1/1000th of an inch

The sand flow rate based on ppmW is calculated as,

$$m_p = \frac{[m_g + m_l] \times ppmW}{10^6} \quad (18)$$

The erosion rate can be calculated beginning with estimating the function characterizing pipeline ductility, F(α) as follows,

$$\text{For } 30^0, \ \frac{\alpha\pi}{180} = \frac{30\pi}{180} = 0.5236 \quad (19)$$

$$F(\alpha) = 0.6 \times \left[Sin(0.5236) + 7.2\left(Sin(0.5236) - Sin^2(0.5236)\right)\right]^{0.6} \times [1 - e^{-20 \times 0.5236}] = 0.9890 \quad (20)$$

The sand flow rate based on ppmW is,

$$m_p = \frac{[31{,}657 + 14{,}928] \times 50}{10^6} = 2.33\ kg/h \quad (21)$$

The pipeline area exposed to erosion is,

$$A_t = \frac{A_p}{Sin(\alpha)} = \frac{0.0355}{Sin(30)} = 0.0711\ m^2 \quad (22)$$

The erosion rate is therefore calculated as,

$$E = \frac{2.33 \times [2 \times 10^{-9}] \times [5.887 + 0.16]^{2.6} \times 0.989 \times [3.15 \times 10^{10}]}{3600 \times 7{,}800 \times 0.0711} \quad (23)$$

$$E = 0.0078\ mm/y\ (or)\ 0.31\ mils/y \quad (24)$$

Pipe Bend Erosion Rate – DNVGL RP O501

Pipeline bends are prone to erosional wear. When the flow direction in the bend changes, sand particles crash against the bend wall, instead of following the flow direction. Assuming a straight length [10D] before the bend, the erosion rate is estimated as,

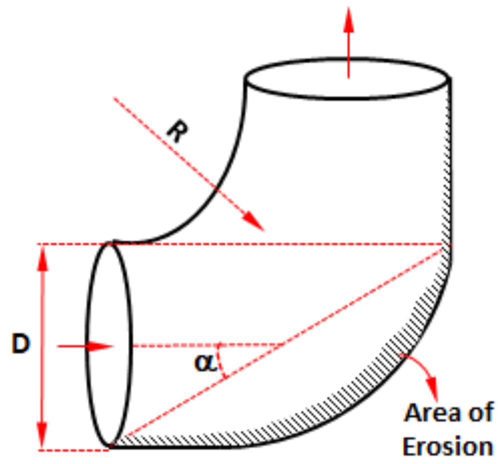


Figure 2. Impact Angle [α] in Pipeline Bends

The characteristic impact angle, α for the pipe bend geometry is calculated from the radius of curvature. The radius of curvature, R_c [i.e., bend radius] for a bend is expressed as number of pipe diameters. Considering a 90^0

long elbow, the bend radius in terms of number of pipe diameters is 1.5, i.e., $R_c = 1.5$.

$$\alpha = Tan^{-1}\left[\frac{1}{\sqrt{2R_c}}\right] \tag{23}$$

$$\alpha = Tan^{-1}\left[\frac{1}{\sqrt{2\times 1.5}}\right] = 0.5236\, rad\ or\ 30° \tag{24}$$

The length of the 90^0 bend is estimated as,

$$L_{Bend} = \frac{\theta}{360} \times 2\pi R_c, Where\ \theta = 90° \tag{25}$$

$$L_{Bend} = \frac{90}{360} \times 2\pi \times 1.5 \times 8.625 = 20.3in \tag{26}$$

As per DNVGL RP O0501, the erosional rate [E] for pipe bends is computed as,

$$E = \frac{m_p \times K \times U_p^n \times F(\alpha) \times G \times C_1 \times GF}{\rho_t \times A_t} \times [3.15 \times 10^{10}] \tag{27}$$

Where,

E = Erosion Rate [mm/year]

m_p = Sand Flow rate [kg/s]

K=Material Constant (2×10^{-9} for Steel Grades)

n =Material Constant (2.6 for Steel Grades)

U_p = Particle Velocity [m/s] ($V_g + V_l$)

ρ_t = Pipeline density [kg/m^3]

A_t = Bend Area exposed to erosion [m^2]

$F(\alpha)$ = Function characteristic of ductility [-]

G = Particle Size correction function [-]

C_1 = Constant accounting for multiple impact of sand particles at the bend's outer end [2.5]

GF = Geometry Factor

The geometry factor [GF], is taken to be 1.0 based on the assumption that the straight line section, upstream of the bend is greater than 10D. For straight line section less than 10D, the GF increases to 2 or 3. To arrive at the particle size correction term, G, the dimensionless parameter A is calculated first.

$$A = \frac{\rho_m^2 \times Tan(\alpha) \times U_p \times D}{\rho_p \times \mu_m} \tag{28}$$

The diameter relation [γ] and critical diameter relation [γ_g] is calculated as,

$$\gamma = \frac{d_p}{ID} \tag{29}$$

Where, d_p = Average Particle diameter

$$\frac{d_{p,c}}{ID} = \gamma_c = \begin{cases} \frac{\rho_m}{\rho_p[1.88ln(A)-6.04]} if\ 0 < \gamma_c < 1 \\ 0.1 \qquad if\ \gamma_c \le 0\ or\ \gamma_c \ge 0.1 \end{cases} \tag{30}$$

The particle size correction function [G] is,

$$G = \begin{cases} \frac{\gamma}{\gamma_c}\ \ if\ \gamma < \gamma_c \\ 1\ if\ \gamma \ge \gamma_c \end{cases} \tag{31}$$

The pipeline Bend Area exposed to erosion is,

$$A_t = \frac{A_t}{Sin(\alpha)} \tag{32}$$

Applying the expressions to the case study,

$$A = \frac{60.2^2 \times Tan(30) \times 6.1 \times 0.2127}{2{,}650 \times 0.0000258} \approx 39{,}315 \tag{33}$$

$$\gamma = \frac{300}{0.2127\times 10^6} = 0.00141027 \tag{34}$$

$$\gamma_g = \frac{60.2}{2{,}650 \times [1.88ln(39315)-6.04]} = 0.00163961 \tag{35}$$

Therefore since $\gamma < \gamma_g$, the particle size correction function is,

$$G = \frac{\gamma}{\gamma_g} = \frac{0.00141027}{0.00163961} = 0.8601 \tag{36}$$

The critical particle diameter [$d_{p,c}$] is calculated as,

$$d_{p,c} = ID \times \gamma_c = \frac{0.2127 \times 0.00163961}{10^{-6}} \approx 349\mu m \tag{37}$$

The pipeline area exposed to erosion is,

$$A_t = \frac{A_p}{Sin(\alpha)} = \frac{0.0355}{Sin(30)} = 0.0711\ m^2 \tag{38}$$

The function characterizing pipeline ductility, F(α) as follows,

$$For\ 30^0,\ \frac{\alpha\pi}{180} = \frac{30\pi}{180} = 0.5236 \tag{39}$$

$$F(\alpha) = 0.6 \times \left[Sin(0.5236) + 7.2\big(Sin(0.5236) - Sin^2(0.5236)\big)\right]^{0.6} \times [1 - e^{-20\times 0.5236}] = 0.9890 \tag{40}$$

Therefore the erosion rate is computed as,

$$E = \frac{2.33 \times [2\times10^{-9}] \times [6.1]^{2.6} \times 0.989 \times 1 \times 2.5 \times 0.86 \times [3.15 \times 10^{10}]}{3600 \times 7{,}800 \times 0.0711} \tag{41}$$

$$E = 0.0496\ mm/y\ (or)\ 1.95\ mils/y \tag{42}$$

Max Safe Pressure in Corroded Area

With sand erosion occurring due to sand flow, defects begin to form on the pipeline inner surface. These defects have a certain depth of penetration [d] for a given wall thickness of the pipe [t]. The following section provides calculations for the maximum safe pressure for operation based on RSTRENG 085dL method, DNVGL RP F101 Single defect method and PETROBRAS PB method.

As per ASME B31G, for a pit depth of up to 10%, the pipeline can be continued to be operated with the existing MAOP. For a pit depth between 10% and 80%, the pipeline needs to be operated at the revised/reduced MAOP based on the corroded wall thickness. For a pit depth greater than 80%, the pipeline would have to repaired or replaced.

As per ASME B31G, for a contiguous corroded area having a maximum depth of more than 10% but less than 80% of the nominal pipe wall thickness, L_m should not extend along the longitudinal axis of the pipe for a distance greater than calculated from the expression,

$$L_m = 1.12B\sqrt{Dt} \quad (43)$$

Where,

L_m = Maximum Allowable Longitudinal length of corroded area [in]

D = Pipeline OD [in]

T = Pipeline selected Wall thickness [in]

The constant B is estimated as,

$$B = \sqrt{\left[\frac{\left(\frac{d}{t}\right)}{1.1\left(\frac{d}{t}\right)-0.15}\right]^2 - 1} \quad (44)$$

As per ASME B31G, B cannot be > 4.0. For corrosion depth [d/t] between 10% and 17.5%, the value of B is to be limited to 4.0. For e.g., with d/t = 0.32, the value of B & L_m is,

$$B = \sqrt{\left[\frac{0.32}{1.1(0.32)-0.15}\right]^2 - 1} = 1.23 \quad (44)$$

$$L_m = 1.12 \times 1.23\sqrt{8.625 \times 0.125} = 1.43\ in \quad (45)$$

RSTRENG 085dL Method

The max safe pressure with RSTRENG 085dL method is determined as follows,

$$P_f = \frac{2t[SMYS+69]\times 145.04\times F\times E\times T}{D\times 14.5}\left[\frac{1-0.85\left(\frac{d}{t}\right)}{1-\left[0.85\left(\frac{d}{t}\right)M^{-1}\right]}\right] \quad (46)$$

Where,

SMYS = Specific Min Yield Strength [MPa]

D = Pipeline OD [in]

M = Folias Bulging Factor [-]

For the condition, $L^2/Dt \leq 50$, M is,

$$M = \sqrt{1 + 0.6275\left[\frac{L^2}{Dt}\right] - 0.003375\left[\frac{L^2}{Dt}\right]^2} \quad (47)$$

For the condition, $L^2/Dt > 50$, M is,

$$M = 3.3 + 0.032\left[\frac{L^2}{Dt}\right] \quad (48)$$

For this module, the measured max corroded area depth [d] and measured longitudinal length [L] in the inclined pipe is 0.04" and 3" respectively. For the selected wall thickness of 3.18 mm, d/t is 0.32, i.e., 32% pit depth.

Similarly for the pipe bend, the measured max corroded area depth [d] and measured longitudinal length [L] is 0.06" and 1.3" respectively. For the selected wall thickness of 3.18 mm, d/t is 0.48, i.e., 48% pit depth. Therefore, for the inclined pipeline,

$$\frac{L^2}{Dt} = \frac{3^2}{8.625\times 0.125} = 8.35 < 50 \quad (49)$$

Since $L^2/Dt < 50$, the Folias bulging factor is,

$$M = \sqrt{1 + [0.6275 \times 8.35] - 0.003375[8.35]^2} \quad (50)$$

$$M = 2.45 \quad (51)$$

Therefore the max safe pressure is,

$$P_f = \frac{2\times 0.125[448+69]\times 145.04\times 0.72\times 1\times 1}{8.625\times 14.5}\left[\frac{1-(0.85\times 0.32)}{1-[0.85\times 0.32\times 2.45^{-1}]}\right] \quad (52)$$

$$P_f = 88.4\ bara \quad (53)$$

Performing similar calculations for Pipe Bend with d = 0.06" and L=1.3", the Max safe pressure is 90.0 bara.

DNV RP F101 Single Defect Method

The max safe pressure with DNVGL RP F101 single defect method is determined as,

$$P_f = \frac{2\times t\times \sigma_u\times 145.04\times F\times E\times T}{[D-t]\times 14.5}\left[\frac{1-\left(\frac{d}{t}\right)}{1-\left[\left(\frac{d}{t}\right)M^{-1}\right]}\right] \quad (54)$$

Where,

σ_u = Ultimate Tensile Strength [MPa]

D = Pipeline OD [in]

M = Folias Bulging Factor [-]

$$M = \sqrt{1 + 0.31 \left[\frac{L^2}{Dt}\right]} \quad (55)$$

Applying the DNVGL RP F101 Single defect method to the same inclined pipe and pipe bend data for an ultimate tensile strength of 530 MPa, the max safe pressure is,

$$M = \sqrt{1 + [0.31 \times 8.35]} = 1.89 \quad (56)$$

$$P_f = \frac{2\times0.125\times530\times145.04\times0.72\times1\times1}{[8.625-0.125]\times14.5}\left[\frac{1-0.32}{1-[0.32\times1.89^{-1}]}\right] \quad (57)$$

$$P_f = 91.9\ bara \quad (58)$$

Performing similar calculations for pipe bend with d=0.06" and L=1.3", the Max safe pressure is 96.3 bara.

PETROBRAS PB Method

The max safe pressure with PETROBRAS PB method is determined as,

$$P_f = \frac{2 t \times \sigma_u \times 145.04 \times F \times E \times T}{[D-t]\times14.5}\left[\frac{1-\left(\frac{d}{t}\right)}{1-\left[\left(\frac{d}{t}\right)M^{-1}\right]}\right] \quad (59)$$

Where,

σ_u = Ultimate Tensile Strength [MPa]

M = Folias Bulging Factor [-]

$$M = \sqrt{1 + 0.217 \left[\frac{L^2}{Dt}\right] - \frac{1}{1.15\times10^6}\left[\frac{L^2}{Dt}\right]^4} \quad (60)$$

Applying the PETROBRAS PB method to the same inclined pipe and pipe bend data for an ultimate tensile strength of 530 MPa, the max safe pressure is,

$$M = \sqrt{1 + [0.217 \times 0.835] - \frac{0.835^4}{1.15\times10^6}} = 1.68 \quad (61)$$

$$P_f = \frac{2\times0.125\times530\times145.04\times0.72\times1\times1}{[8.625-0.125]\times14.5}\left[\frac{1-0.32}{1-[0.32\times1.68^{-1}]}\right] \quad (62)$$

$$P_f = 94.3\ bara \quad (63)$$

Performing similar calculations for Pipe Bend with d = 0.06" and L=1.3", the Max safe pressure is 99.7 bara.

Results

Summarizing, the max safe pressure is 88.4 bara for pipeline and 90 bara for pipe bend,

Table 2. Max Safe Pressures

Method	Max Safe Pressure, P_f [bara]	
-	Inclined Pipe	Pipeline Bend
RSTRENG 085dL	88.4	90.0
DNV RP F101 Single Defect	91.9	96.3
Petrobras PB	94.3	99.7
Design Pressure [DP]	93.5	93.5
Max Safe Pressure, P_f	**88.4**	**90.0**

Based on the erosion rate for an operating period of 25 years, the pipeline WT lost is,

Table 3. Pipeline WT Lost

Parameter	Inclined Pipe	Pipeline Bend
Erosion Rate [mm/y]	0.0078	0.0496
WT Lost in 25 Years [mm]	0.20	1.24

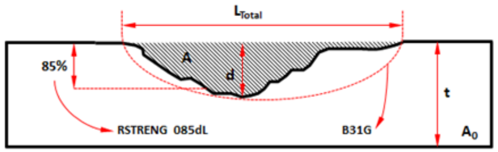
References & Further Reading

1. "*Managing Sand Production and Erosion*", DNVGL-RP-O501, Aug 2015 Edition.
2. "*Manual for Determining Remaining Strength of Corroded Pipelines*", ASME B31G-1991
3. "*Folias Factor*", Science Direct, https://www.sciencedirect.com/topics/engineering/folias-factor
4. "*Modified Equation for the Assessment of Long Corrosion Defects*", Adilson C. Benjamin, Ronaldo D Vieria, Jose Luiz F. Friere, Jaime T.P. de Castro, https://www.researchgate.net/publication/249657141

Appendix A

Pipeline Mechanical Design Data

Parameter	Value	Unit
Operational Life of Pipeline	25	[Years]
Location of Gas Pipeline	Deserted	[-]
Location Class	Class 1, Div 2	[-]
Design Factor [F]	0.72	[-]
Pipeline Joining Method	ERW	[-]
Longitudinal Joint Factor [E]	1.0	[-]
API 5L Spec	PSL 1 X65	[-]
Ultimate Tensile Strength [σ_u]	530	[MPa]
SMYS [S]	448.0	[MPa]
Design Pressure [DP]	44.0	[bara]
Design Temperature [DT]	100	[^{0}C]
De-Rating Factor [T]	1.00	[-]
Chosen Pipeline Diameter [DN]	8.63	[in]
Calculated Wall Thickness [t_{Calc}]	1.49	[mm]
Corrosion Allowance [CA]	1.00	[mm]
Total Wall Thickness [WT]	2.49	[mm]
	0.098	[in]
Selected Wall Thickness [t]	3.18	[mm]
	0.125	[in]
DP [Based on selected WT]	93.5	[bara]
Pipeline Inside Diameter [ID]	212.73	[mm]



B31G Method – Approximates defect Area [A] as a Parabolic Shape
RSTRENG 085dL Method – Approximates defect Area [A] as 85% of Peak depth

RESULTS

Parameter	Value	Unit
Pipeline Size Chosen [DN]	**8.63**	[in]
Selected Pipeline WT	**3.18**	[mm]
DP [Based on Selected WT]	**93.5**	[bara]
Max Safe Pressure [Pipeline], d = 0.04 in	**88.4**	[bara]
Max Safe Pressure [Bend], d = 0.06 in	**90.0**	[bara]
Pipeline WT Lost in 25 Years	**0.20**	[mm]
Is Pipeline Erosion Rate, E [25 Yrs] < CA	**Yes**	
Bend WT Lost in 25 Years	**1.24**	[mm]
Add. Outer CA for Bend [E = 5 mils] + t mm	**1.24**	[mm]

Inclined Pipeline - Wall Thickness Erosion Rate [Sand Erosion]

Parameter	Value	Unit
Gas Flow Rate [M_g]	31,657	[kg/h]
Liquid Flow Rate [M_l]	14,928	[kg/h]
Total Flow Rate [M_m]	46,585	[kg/h]
Gas Density [ρ_g]	42.0	[kg/m^3]
Liquid Density [ρ_l]	713.2	[kg/m^3]
Pipeline ID [ID]	0.2127	[m]
Pipeline Material of Construction [MOC]	Carbon Steel	[-]
Pipeline Material Density [ρ_p]	7,800	[kg/m^3]
Gas Velocity [V_g]	5.9	[m/s]
Liquid Velocity [V_l]	0.16	[m/s]
Mixture Density [ρ_m]	60.2	[kg/m^3]
Particle Impact Velocity [U_p]	6.1	[m/s]
Sand Content [ppmW]	50.0	[ppmW]
Sand Mass Flow [m_p]	2.33	[kg/h]
Angle of Impegment [α]	30.0	[degrees]
Material Ductility Charatersitic [F(α)]	0.9890	[-]
Pipeline Cross Sectional Area exposed to Erosion [A_t]	0.0711	[m^2]
Material Constant 'K' [Steel Grades]	2.0E-09	[-]
Material Constant 'n' [Steel Grades]	2.6	[-]
Erosion Rate referred to Depth [E]	**0.0078**	**[mm/y]**
	0.31	**[mils/Y]**

Max Safe Pressure in Corroded Areas - Inclined Pipeline

Parameter	Value	Unit
Measured Max corroded area depth [d]	0.04	[in]
Measured Longitudinal Length of Defect [L]	3.00	[in]
d/t [Where, t = Selected WT]	0.32	[-]
% Pit Depth	32.0	[%]
Action Suggested	Check Max Safe Pressure	
Constant "B"	1.23	[-]
Max Allow. $L_{Longitudinal}$ of Corroded Area [L_m] [ASME B31G]	1.43	[in]
L^2/Dt	8.35	[-]
Folias Bulging Factor 'M' [RSTRENG 085dL]	2.45	[-]
Folias Bulging Factor 'M' [DNV RP-F101 - Single Defect]	1.89	[-]
Folias Bulging Factor 'M' [Petrobras PB Method]	1.68	[-]
Max Safe Pressure [P_f] [RSTRENG 085dL Method]	88.4	[bara]
Max Safe Pressure [P_f] [DNV RP-F101 - Single Defect]	91.9	[bara]
Max Safe Pressure [P_f] [Petrobras Method]	94.3	[bara]
Max Safe Pressure after Corrosion Depth Applied	**88.4**	**[bara]**

Pipeline Bend - Wall Thickness Erosion Rate [Sand Erosion]

Parameter	Value	Unit
Gas Viscosity [μ_g]	1.34E-05	[kg/m.s]
Liquid Viscosity [μ_l]	4.72E-04	[kg/m.s]
Mixture Viscosity [μ_m]	2.58E-05	[kg/m.s]
Sand Type	Quartz Sand	[-]
Sand Density [ρ_p]	2,650	[kg/m^3]
Average Sand Particle Size	300	[µm]
Number of Pipe Diameters [90^0 Long Elbow]	1.5	[-]
Angle of Impegment [α]	0.5236	[rad]
	30.0	[Degrees]
Dimensionless Parameter 'A'	39,315	[-]
Diameter Relation [γ]	0.0014	[-]
Critical Diameter Relation [γ_c]	0.0016	[m]
Critical Particle Diamter [$d_{p,c}$]	349	[µm]
Particle Size Correction Factor [G]	0.8601	[-]
Characteristic Bend Area Exposed to Erosion [A_t]	0.0711	[m^2]
Material Ductility Charatersitic [F(α)]	0.9890	[-]
Model Geometry Factor [C_1]	2.5	[-]
Geometry Factor [GF]	Single Component	
GF	1	[-]
Erosion Rate referred to Depth [E]	**0.0496**	**[mm/y]**
	1.95	**[mils/Y]**

Max Safe Pressure in Corroded Areas - Pipeline Bend

Parameter	Value	Unit
Measured Max corroded area depth [d]	0.06	[in]
Measured Longitudinal Length of Defect [L]	1.30	[in]
d/t [Where, t = Selected WT]	0.48	[-]
% Pit Depth	48.0	[%]
Action Suggested	Check Max Safe Pressure	
Constant "B"	0.78	[-]
Max Allow. $L_{Longitudinal}$ of Corroded Area [L_m] [ASME B31G]	0.91	**[in]**
L^2/Dt	1.57	[-]
Folias Bulging Factor 'M' [RSTRENG 085dL]	1.41	[-]
Folias Bulging Factor 'M' [DNV RP-F101 - Single Defect]	1.22	[-]
Folias Bulging Factor 'M' [Petrobras PB Method]	1.16	[-]
Failure Pressure [P_f] [RSTRENG 085dL Method]	90.0	[bara]
Max Safe Pressure [P_f] [DNV RP-F101 - Single Defect]	96.3	[bara]
Failure Pressure [P_f] [Petrobras Method]	99.7	[bara]
Pipeline MAOP after Corrosion Depth Applied	**90.0**	**[bara]**

Module 21

ECONOMIC INUSLATION FOR INDUSTRIAL PIPING

Thermal Insulation for Industrial Piping is a common method to reduce energy costs in production facilities while meeting process requirements. Insulation represents a capital expenditure & follows the law of diminishing returns. Hence the thermal effectiveness of insulation needs to be justified by an economic limit, beyond which insulation ceases to effectuate energy recovery. To determine the effectiveness of an applied insulation, the insulation cost is compared with the associated energy losses & by choosing the thickness that gives the lowest total cost, termed as 'Economic Thickness'.

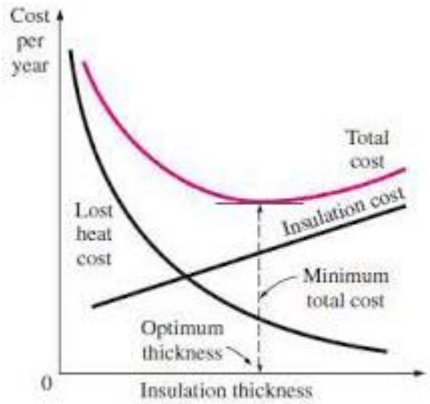


Figure 1. Economic Insulation Thickness Selection

The following module provides guidance to estimate the economic thickness for natural gas piping in winter conditions as an example case study.

Design Considerations

To estimate the economic insulation thickness, the following factors are to be given attention – Energy costs (steam/electricity), annual hours of operation, operating surface temperature, pipe dimensions, estimated cost of insulation, and average exposure to the ambient. These are critical to predict the thermal resistances and heat transfer coefficients and the total heat loss or gain, from or to the system.

Lower heat transfer coefficients & thermal conductivity offer a lower rate of heat loss/gain. It is for this reason; materials that provide low thermal conductivity are chosen to provide insulation. To provide effective insulation, the conductive heat transfer from the metal has to be kept lower than the convective heat transfer on the insulation's external side to prevent the outer insulation temperature from increasing drastically.

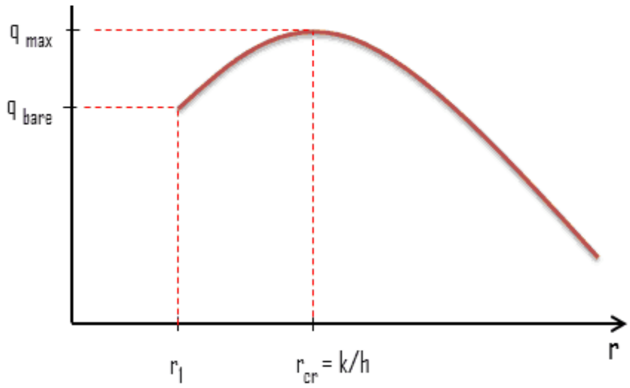


Figure 2. Critical Insulation Thickness

From the above, when insulation is applied on a bare pipe of a given nominal diameter, the heat transfer rate increases as the insulation radius/thickness increases. As the insulation thickness increases, until reaching the critical radius [R_c], there is a progressive fall in the convective resistances causing higher heat losses from the pipe.

Therefore for insulation to be properly effective in restricting heat transmission, the outer pipe radius R_2 must be greater than or equal to the critical radius [R_c] of the insulation. If this condition is not satisfied, no useful purpose will be served with the chosen material of insulation.

Case Study & Assumptions

To demonstrate the economic insulation calculations, a case study is made based on a natural gas piping operating in cold winter conditions. In addition, certain assumptions are made for this example case study.

1. Ambient temperature [T_a] is taken as 0^0C & wind velocity is taken as 18 km/h (5 m/s).
2. The pipe inside heat transfer [HT] coefficient [h_i] is neglected since it is small compared to outer/ambient HT coefficient.
3. Radiation is accounted for with the emissivity of the outer bare pipe taken as 0.9 while insulation emissivity is 0.13.
4. Heat transfer through pipe & insulation material is assumed to be perfectly radial & critical thickness is estimated at steady state conditions, i.e., at equilibrium.

Design Data

For estimation of heat transfer coefficients, the process data used is as follows,

Table 1. Natural Gas Composition

Component	MW	Mol%
-	**[kg/kmol]**	**[%]**
Methane [CH_4]	16.04	76.23
Ethane [C_2H_6]	30.07	10.00
Propane [C_3H_8]	44.01	5.00
i-Butane [i-C_4H_{10}]	58.12	1.00
n-Butane [n-C_4H_{10}]	58.12	1.00
i-Pentane [i-C_5H_{12}]	72.15	0.30
n-Pentane [n-C_5H_{12}]	72.15	0.10
Water [H_2O]	18.02	0.25
Carbon dioxide [CO_2]	44.01	3.00
Hydrogen Sulphide [H_2S]	34.08	0.07
Nitrogen [N_2]	28.01	3.00
Total		**100.0**

The air properties between -25^0C & 50^0C are computed using fitted equations as follows,

1. Air Density [kg/m^3] is computed as,

$$\rho_{Air} = 0.0000158T^2 - 0.0134T + 3.7622 \quad (1)$$

2. Air Specific Heat [kJ/kg.K] is computed as,

$$C_{p,air} = 1.006 \quad (2)$$

3. The thermal conductivity [W/m.K] of air is,

$$k_{Air} = -2.69 \times 10^{-8}T^2 + 9.04 \times 10^{-5}T + 9.56 \times 10^{-4} \quad (3)$$

4. The thermal diffusivity [m^2/s] of air is,

$$\alpha_{Air} = 1.99 \times 10^{-10}T^2 + 1.5 \times 10^{-8}T - 7.96 \times 10^{-7} \quad (4)$$

5. The dynamic viscosity [kg/m.s] of air is,

$$\mu_{Air} = -4.22 \times 10^{-11}T^2 + 7.19 \times 10^{-8}T + 8 \times 10^{-7} \quad (5)$$

6. The kinematic viscosity [m^2/s] of air is,

$$\gamma_{Air} = 1.02 \times 10^{-10}T^2 + 3.1 \times 10^{-8}T - 2.69 \times 10^{-6} \quad (6)$$

7. The Prandtl Number of air is computed as,

$$Pr = -5.12 \times 10^{-7}T^2 + 3.7 \times 10^{-5}T + 0.7642 \quad (7)$$

Based on the above correlations, for an ambient temperature of 273.15 K, the air properties are as follows,

Table 2. Air Properties at Ambient Conditions

Parameter	Value	Unit
Density [ρ_{air}]	1.293	kg/m^3
Specific Heat [$C_{p,air}$]	1.006	kJ/kg.K
Thermal Conductivity [k_{air}]	0.0236	W/m.K
Thermal Diffusivity [α_{air}]	0.000018	m^2/s
Dynamic Viscosity [μ_{air}]	0.000017	kg/m.s
Thermal Exp. Coefficient [β_{air}]	0.0037	1/K
Kinematic Viscosity [γ_{air}]	0.000013	m^2/s

Natural Gas Pipe Construction Details

The construction details of the natural gas pipe is as follows,

Table 3. Pipe Construction Details

Parameter	Value	Units
Pipe Material	Carbon Steel	
Design Pressure	11.0	bara
Design Temperature	100	^{0}C
Pipeline DN	6.625	in
Pipe WT	3.58	mm
Pipe ID	161.1	mm
Pipe Length incl. Fittings [L_e]	1,000	m
Pipe Total OD [D_3]	269.875	mm

Pipe Thermal Cond. [k_{pipe}]	45	W/m.K
Pipe Surface Emissivity[ε]	0.90	-
Ambient Temperature [T_{amb}]	0	^{0}C
Wind Velocity [V_a]	18	km/h
Insulation Material	Urethane Foam	
Insulation Thermal Cond. [k_{ins}]	0.018	W/m.K
Insula. Surface Emissivity [ε]	0.13	-

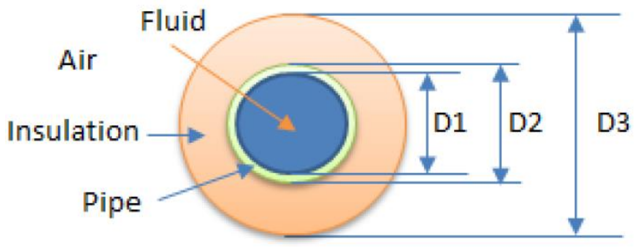


Figure 3. Pipe Construction

The Process data used for the case study is,

Table 4. Pipe Inlet Process Data

Parameter	Value	Units
Pipe Gas Flow Rate [Q]	12.0	MMSCFD
Pipe Inlet Pressure [P_1]	20.0	bara
Pipe Temperature [T_1]	40.0	^{0}C
Gas MW	21.16	kg/kmol
Pipe Inlet Cp	2.0967	kJ/kg.K
Compressibility Factor [Z_1]	0.9539	-
Gas Flow [Act_m3/h]	742	m^3/h
Gas Density [r]	17.04	kg/m^3
Mass Flow [m]	12,643	kg/h

The gas compressibility factor, Z is predicted using DAK EoS. Gas line pressure drop is estimated using Weymouth equation. Due to the presence of water in the natural gas stream, ice & hydrate formation tendencies exist. For a flow pressure of 16.14 bara, the hydrate temperature is 9.52^{0}C.

Therefore insulation is to be provided to ensure hydrate & ice formation does not take place. A Hydrate P-T plot is therefore presented as follows,

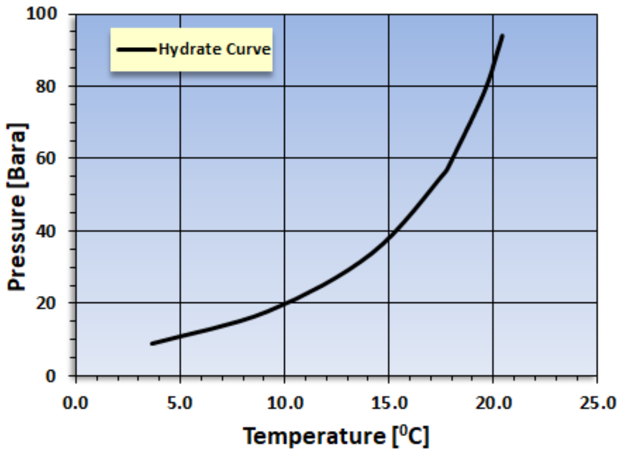


Figure 4. Hydrate P-T Curve

Results

With the methodology employed, the pipe process results computed with Weymouth & DAK-EoS are as follows,

Table 5. Pipe Process Results

Parameter	Value	Units
Pipe Inlet Velocity [V]	10.0	[m/s]
Pipe Exit Velocity [V_e]	10.5	[m/s]
Pipe Exit Temperature [T_e]	39.2	[0C]
Pipe Exit Pressure [P_e]	19.07	[bara]
Pressure Drop [ΔP]	0.93	[bar]
ΔP per km [ΔP/L]	0.93	[bar/km]

The dQ vs. Insulation radius plot shows a decreasing trend between heat loss from a bare pipe [Q_{bare}] and heat loss from an insulated pipe [Q_{Ins}] with increase in insulation thickness.

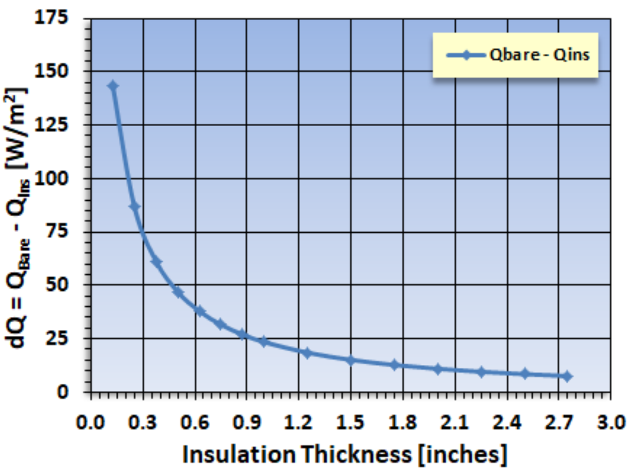


Figure 5. dQ vs Insulation Thickness

A plot between the total annual costs & insulation thickness shows that the annual total cost of the energy losses is the least at 2" insulation thickness representing 53,694€.

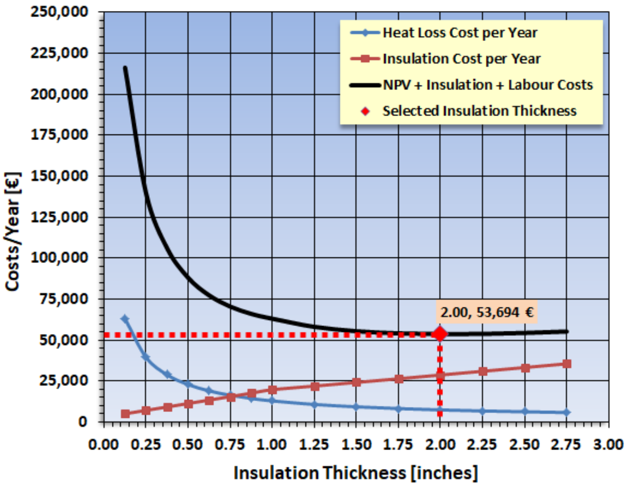


Figure 6. Costs per Year vs. Insulation Thickness

Appendix A: Design Methodology

To estimate the thermal insulation required, the heat losses & heat transfer coefficients are accounted based on 3 modes of heat transfer driven by temperature differences – namely, pipe wall conduction, free convection, forced convection & ambient radiation. For the bare pipe & insulation cases, air flows over the pipe surface thereby forming a film with a certain temperature. This film temperature determines the rate of heat losses through the pipe surface/insulation. The air film temperature [$T_{airfilm}$] on the insulation surface is estimated iteratively. Therefore for the first iteration,

$$T_{airfilm,ins,1} = T_{amb}[°C] + 1°C \qquad (8)$$

Radiation Heat Transfer

To estimate the radiation heat transfer between the ambient & concrete insulation on the tank, the expression is written as, [1],

$$h_r = \varepsilon \times \sigma\left(T_{airfilm,} + T_a\right)\left(T^2_{airfilm} + T^2_a\right) \qquad (9)$$

Where, $\sigma = 5.67\times10^{-8}$ W/m²/K

ε = Surface emissivity

The radiation mode expressed above is written in a manner similar to convection, i.e., the radiation rate equation is linearized making the heat rate proportional to the temperature difference rather than to the difference between two temperatures to the fourth power.

Forced Convection

To calculate the external heat transfer coefficient [h_o], Nusselt number for forced convection over circular cylinder with cross flow can be estimated using Churchill and Bernstein correlation [1]. This equation is valid for all $Re.Pr \geq 2$ and the correlation is expressed as,

$$Nu = 0.3 + \frac{0.62\, Re^{1/2}\, Pr^{1/3}}{\left[1+\left(\frac{0.4}{Pr}\right)^{2/3}\right]^{1/4}}\left[1 + \left(\frac{Re}{282000}\right)^{5/8}\right]^{4/5} \qquad (10)$$

Prandtl Number [Pr] of ambient air is,

$$Pr = \frac{c_{P,air}\mu_{air}}{k_{air}} \qquad (11)$$

Reynolds number [Re] becomes,

$$Re = \frac{V_{air}D_3\rho_{air}}{\mu_{air}} \qquad (12)$$

The above correlation is valid for all ranges of Reynolds number (Re) and $Pr \geq 0.2$, where all properties are evaluated at film temperature. It is to be noted that as per [1], Churchill & Bernstein correlation is reasonable over a certain range of conditions but for most engineering calculations, the accuracy is not expected to be much better than 20% because these are based on more recent results encompassing a wide range of conditions.

Natural/Free Convection

To estimate the heat transfer due to natural convection, the correlation by Churchill & Chu [1] can be used and is of the form,

$$Nu = \left\{0.6 + \frac{0.387\, Ra^{1/6}}{\left[1+\left(\frac{0.559}{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 \qquad (13)$$

Where, Rayleigh number (Ra) is computed as,

$$Ra = \frac{g\times\beta_{air}\times\left[T_{airfilm}-T_a\right]D_3^3}{\alpha_{air}\gamma_{air}} \qquad (14)$$

Where, β = Thermal expansion coefficient

α_{air} = Thermal diffusivity

Therefore the combined heat transfer coefficient is computed as,

$$Nu_{combined} = \left[Nu_{free}^4 + Nu_{forced}^4\right]^{1/4} \quad (15)$$

$$h_{comb} = \frac{k_{air} Nu_{comb}}{D_3} \quad (16)$$

Therefore the external heat transfer coefficient, $h_{air,\,overall}$, is computed as,

$$h_0 = h_{comb} + h_r \quad (17)$$

Bare Pipe & Insulation Resistance

The resistance offered by the bare pipe & insulation is estimated as follows,

$$R_{bare} = \frac{D_2 ln\left[\frac{D_2}{D_1}\right]}{2k_{pipe}} \quad (18)$$

$$R_{Ins} = \frac{D_3 ln\left[\frac{D_3}{D_2}\right]}{2k_{ins}} \quad (19)$$

Total Resistance – Bare Pipe & Insulation

For bare pipe, the total resistance is calculated as,

$$R_{Total} = R_{bare} + \frac{1}{h_0} \quad (20)$$

For Insulated Pipe, the total resistance is calculated as,

$$R_{Total} = R_{Bare} + R_{Ins} + \frac{1}{h_0} \quad (21)$$

Piping Heat Losses

$$U = {}^{1}/_{R_{Total}} \quad (22)$$

$$\left(\frac{Q}{A}\right)_{Loss} [W/m^2] = U \times (T_1 - T_a) \quad (23)$$

$$\left(\frac{Q}{A}\right)_{Loss} [W/m] = \left(\frac{Q}{A}\right)_{Loss} [W/m^2] \times \pi D_3 \quad (24)$$

$$Q_{Loss}[kW] = \left(\frac{Q}{A}\right)_{Loss} [W/m] \times L_e \quad (25)$$

$$T_{airfilm,ins,2} = T_1 - \left[\left(\frac{Q}{A}\right)_{Loss} [W/m] \times R_{ins}\right] \quad (26)$$

The above set of heat transfer calculations are performed first for a bare pipe & then performed for various insulation thicknesses to estimate the heat losses, Q_{Loss} [W/m^2] and Q_{Loss} [kW], Q_{Loss} [kWh/year] which is computed by multiplying Q_{Loss} [kW] with the annual working hours.

Insulation Economics

The economic thickness of insulation depends on the insulating & maintenance costs and also the annual value of heat loss. This would depend on the cost of producing energy & thermal conductivity of the lagging. Generally thicker insulation will represent higher investment costs and lower heat loss costs. The annual heat losses are computed as,

$$C_{Loss} = \left(\frac{Q}{A}\right)_{Loss} [kW] \times n \times Q_{cost}/kWh \quad (27)$$

Where, n = number of annual hours

Insulation Costs is the product of insulation volume and insulation cost per m^3.

$$V_{Ins}[m^3] = \frac{\pi}{4}[D_3^2 - D_2^2] \times L_e \quad (28)$$

$$C_{Ins} = V_{ins} \times [C_{ins}/m^3] \quad (29)$$

Labour Costs is the product of cost per unit metre & length of pipe

$$C_{Labour} = L_e \times [C_{lab}/m] \quad (30)$$

The cost of energy losses is quantified by the Net Present Value (NPV) of the future energy costs during an insulation life of typically 5 years. For this module, a discount rate [i] of 15% is used. The number of annual working hours is taken as 8,000 hours, cost of energy (electricity to run the gas compressor) is taken as 0.10€/kWh and the insulation cost is taken as 50€/m^3. The annual value of the energy losses for 5 years is calculated as,

$$NPV = R \times \frac{1-[1+i]^{-n}}{i} \quad (31)$$

Where, R is the cost of energy losses

The annual total cost is computed as,

$$C_{Total} = NPV + C_{Ins} + C_{Labour} \quad (32)$$

The Insulation thickness corresponding to the lowest total cost will be the economic thickness of insulation.

References & Further Reading

1. "*Fundamentals of Heat and Mass Transfer*", Incropera, DeWitt, Bergman, Lavine, 6th Edition.

Appendix A: Heat Transfer Coefficients

Heat Transfer Calculations - Iterative Procedure						
Parameter	**Units**	**1**	**2**	**3**	**4**	**5**
$T_{air\ film,\ ins}$	[^{0}C]	1.0	0.6	0.6	0.6	**0.6**
Radiation Heat Transfer						
h_{Rad}	[W/m^2.K]	0.60	0.60	0.60	0.60	**0.60**
Forced Convection Heat Transfer						
Pr	-	0.7358	0.7359	0.7359	0.7359	**0.7359**
Re	-	100,249	100,524	100,524	100,524	**100,524**
Nu_{forced}	-	218.8	219.2	219.2	219.2	**219.2**
h_{forced}	[W/m^2.K]	19.22	19.23	19.23	19.23	**19.23**
Natural Convection Heat Transfer						
Ra	-	2.85E+06	1.66E+06	1.66E+06	1.66E+06	**1.66E+06**
Nu_{Free}	-	19.69	16.86	16.86	16.86	**16.86**
h_{Free}	[W/m^2.K]	1.73	1.48	1.48	1.48	**1.48**
Combined Convection Heat Transfer						
$Nu_{Combined}$	-	218.8	219.2	219.2	219.2	**219.2**
$h_{Combined}$	[W/m^2.K]	19.22	19.24	19.24	19.24	**19.24**
Total External Heat Transfer						
h_o	[W/m^2.K]	19.83	19.84	19.84	19.84	**19.84**
Pipe Wall Resistance						
R_{pipe}	[m^2.K/W]	1.55E-03	1.55E-03	1.55E-03	1.55E-03	**1.55E-03**
Insulation Resistance						
R_{ins}	[m^2.K/W]	3.54	3.54	3.54	3.54	**3.54**
Total Resistance						
R_{Total}	[m^2.K/W]	**3.59**	**3.59**	**3.59**	**3.59**	**3.59**
U	[W/m^2.K]	0.28	0.28	0.28	0.28	**0.28**
Q/A	[W/m^2]	11.1	11.1	11.1	11.1	**11.1**
Q_{Loss}	[W/m]	9.4	9.4	9.4	9.4	**9.4**
$Q_{Total\ Loss}$	[kW]	9.4	9.4	9.4	9.4	**9.4**

Appendix B: Economic Analysis

Pipe Insulation Economics		
Bare Pipe Q_{Loss} [W/m^2]	769.9	*Copy-Paste in Cell Z31 only for Ins. Thick of 0 mm*
With Insulation [W/m^2]	11.0	*Copy-Paste in Cell Z32 Onwards & Below*
$Q_{total\ Loss}$ With Ins [kW]	9.3	*Copy-Paste in Cell AA32 Onwards & Below*
Annual Working Hours	8000	
Cost of Electricity/kWh	0.10 €	
Insulation Cost /m^3	50.00 €	
Life of Insulation [Yrs]	5	

Selected Insulation Thickness	
Thickness Chosen [in]	2.00
Thickness Chosen [mm]	50.80
Critical Radius [mm]	0.91
Insulation Effective	Yes

Insulation Thickness		Q_{Loss}	$Q_{Total\ loss}$	$Q_{Lost/Yr}$		Labour Cost		Ins. Cost	Ins.+Lab Cost	NPV Discount	NPV [Yrly Q_{loss}]	NPV+Ins+Lab Cost
[mm]	[inches]	[W/m^2]	[kW]	[kWh/yr]	[€]	[€/m]	[€]	[€]	[€]	[%]	[€]	[€]
0.0	0.00	899.9	-	-	-	-	-	-	-	-	-	-
3.175	0.13	143.6	78.8	630,263	63,026 €	5 €	5,000 €	86 €	5,086 €	15.0	211,274 €	216,359 €
6.350	0.25	86.9	49.4	395,317	39,532 €	7 €	7,000 €	174 €	7,174 €	15.0	132,516 €	139,691 €
9.525	0.38	61.4	36.1	288,904	28,890 €	9 €	9,000 €	266 €	9,266 €	15.0	96,845 €	106,111 €
12.700	0.50	47.0	28.6	228,968	22,897 €	11 €	11,000 €	361 €	11,361 €	15.0	76,753 €	88,115 €
15.875	0.63	37.9	23.8	190,604	19,060 €	13 €	13,000 €	459 €	13,459 €	15.0	63,893 €	77,353 €
19.050	0.75	31.6	20.5	163,955	16,395 €	15 €	15,000 €	561 €	15,561 €	15.0	54,960 €	70,521 €
22.225	0.88	27.0	18.0	144,365	14,437 €	17 €	17,000 €	665 €	17,665 €	15.0	48,393 €	66,058 €
25.400	1.00	23.5	16.2	129,355	12,936 €	19 €	19,000 €	773 €	19,773 €	15.0	43,362 €	63,135 €
31.750	1.25	18.5	13.5	107,856	10,786 €	21 €	21,000 €	998 €	21,998 €	15.0	36,155 €	58,153 €
38.100	1.50	15.2	11.6	93,186	9,319 €	23 €	23,000 €	1,235 €	24,235 €	15.0	31,237 €	55,473 €
44.450	1.75	12.8	10.3	82,525	8,252 €	25 €	25,000 €	1,485 €	26,485 €	15.0	27,664 €	54,149 €
50.800	2.00	11.0	9.3	74,418	7,442 €	27 €	27,000 €	1,748 €	28,748 €	15.0	24,946 €	53,694 €
57.150	2.25	9.6	8.5	68,040	6,804 €	29 €	29,000 €	2,024 €	31,024 €	15.0	22,808 €	53,832 €
63.500	2.50	8.5	7.9	62,885	6,289 €	31 €	31,000 €	2,312 €	33,312 €	15.0	21,080 €	54,392 €
69.850	2.75	7.6	7.3	58,629	5,863 €	33 €	33,000 €	2,613 €	35,613 €	15.0	19,653 €	55,266 €

Module 22

FRONT END LOADING FOR PIPELINE PROJECT MANAGEMENT

Abstract: With growing clamour for clean energy globally, the midstream industry becomes crucial for any hydrocarbon exporting country. To have an effective midstream network, would mean construction & production costs also have a role to play in ensuring globally competitive & affordable prices of oil & gas products. One could argue that short term and long term barrel prices apart from supply and demand is a motivating factor for operators to invest in hydrocarbon projects, but it can also be equally said, that despite what the global price or supply & demand is, effective project management & execution also determines the economic success for all project stakeholders.

A key stage in midstream project management is Front End Loading (FEL) where strategic information that addresses internal and external risks, resource availability, allocation and commitment is made before sanctioning or making a final investment decision (FID) on the project. Simply put, the more homework you do in the early stages of a project decides how much success can be achieved. Although project management is a vast subject, the following module focuses on the some of the repercussions of poorly executed front end loading (FEL) steps in midstream activities.

Front End Engineering (FEED)

1. ***Choosing Pipeline Sizes***: The starting point to design any oil & gas pipelines is the well production, pressure & temperature profile in addition to the composition of the contents that the pipeline will carry. Production profiles are needed to estimate the peak flow rates which the pipeline experiences and in turn determine the pipeline size, whereas the pressure and temperature profiles determine the pipeline wall thickness. When your reservoir engineer and production technologist are indecisive about the Stock Tank Oil in place (STOIP), how much & at what rate the recoverable volumes from the wells are going to be extracted, chances are that you are going to underestimate/overestimate the pipeline sizes. In case of multiphase flow, whether 2 or 3 phase, the pipeline sizes significantly affect your flow regimes and carry the risk of slug formation. The slug volumes decide the size of your slug catcher & underestimating its size can cause equipment failure. Hence work it out with your subsurface team to arrive at a conclusive and accurate production profile prior to performing pipeline FEED.
2. ***Material Costs Overrun***: Nothing can be more disastrous than realizing as the project progresses that your pipeline actually costs more because of underestimating the pipeline's wall thickness. Wall thickness is a key value that depends on the design pressure & eventually determines the pipeline weight. Since pipeline weight is proportional to the square of the outer diameter (OD), for every millimetre increase, so does the weight increase. When the parabolic increase of per unit pipeline weight is multiplied with the total kilometres of pipeline length, the pipeline material costs are going to probably overrun the project budget. Therefore, it is not just the engineering standard chosen, but following this crucial step as part of a check list prior to finalizing the pipeline sizes is a must.

3. ***Pipeline Corrosion***: Various Engineering design practices offer solutions as to what should be the corrosion allowance for a given pipeline application. Produced water and Hydrogen sulphide (H_2S) are the popular enemies that contribute to pipeline corrosion. In addition to pipeline contents, sand from well fluids that escape sand traps, hydrate particles and fluid flow rates exacerbate metal erosion.

   However, engineers sometimes fail to account for the effect of external forces. In offshore pipelines, sea waves and sand underneath the soil act upon the pipeline components such as risers thereby inducing stress. Human error also needs to be taken into account where, when ships collide with platforms, can cause dents from where corrosion propagates. Pigging operations and depressurization to dislodge hydrates can also contribute to pipeline corrosion simply because pigs can cause dents when their velocities are not regulated properly resulting in the pig getting stuck. When hydrates get dislodged during a depressurization, there are good chances that a high velocity column of hydrate can collide with pipeline bends thereby cracking and exposing metal to corrosion effects.

   Therefore, a key step during FEED is for engineering teams to take time out and allocate resources to do a pipeline stress analysis, on-bottom stability analysis, a basic corrosion management plan covering pipeline coatings & cathodic protection, a risk assessment report and Pipeline Integrity Management (PIM) report to ensure that the wall thicknesses & supporting structures chosen is adequate to meet all internal and external risks that the pipeline can experience. If one argues that this is a far fetched vision during early FEED, wait till you see blame game that starts during detailed engineering stage because of material cost overruns.

Contract Management

1. ***Vendor Contracts***: As much as the top management works on the terms and conditions of a production sharing contract (PSC), taxation, governmental regulations, etc. that shows its effects on the company's balance sheets, so must the procurement department spend time due diligently to ensure that the right vendors are available to deliver material and equipment which affects project schedule and costs. A classic case of project cost overrun is when the procurement department realizes that there is only one particular vendor to meet your project requirements after the project has been sanctioned. If a procurement strategy and supply chain is not in place, it can leave the project to the mercy of the sole vendor. Hence ITT (Invitation to Tender) documents must be prepared at the earliest to receive competing offers from various bidders which in turn allow project managers to prepare realistic schedules and costs incurred.
2. ***Interface Management***: Project management also includes interface management. In midstream projects when facilities such as booster compressors, sectionalizing valves & burn pit lines are vendor items, it is important for contractors to keep a constant open line of communication with vendors to ensure that the engineering & hook-up drawings and datasheets have been followed to meet project specifications prior to execution. There is nothing more upsetting for project managers to watch their tables pile up with Change Orders (CO). When equipment that is already manufactured & delivered to the site but do not conform to the project specifications because of poor communication with the vendor, it is the

homework of developing a contracting strategy, regular project review by engineering teams at the early stages which can minimize the damages to the project's cost & schedule. Otherwise even your legal department might be left out in the open to dry under the sun.

3. ***Man-hours Billing & LSTK***: Between Projects awarded on man-hour billing vs. Lump sum Turn key (LSTK) contracts, in reality, it depends on factors such as - how well the project owner defines the scope & shares the project vision. During FEL stage of long term projects, if project charters that have requirements changing dynamically and frequently, chances are that the engineering contractor would hesitate to engage with the project owner on LSTK basis (unless the contractor is desperate for the money to keep his company afloat). Basically, if the project requirements are not expected to change much during the course of the project, the project owner can negotiate to put the budget and schedule risks on the contractor on LSTK terms.

   But to be practical, there is no project, where complete clarity is always available prior to Sanctioning/Final Investment Decision (FID). Hence it is prudent for project owners to keep their options open to enter into a mixed contract where both man-hour billing and LSTK methods provide flexibility, transparency, accountability and ease of management to the project. Typically FEED follows a man-hour billing cycle & Detailing work follows LSTK terms. For such mixed contracts, the onus is on the engineering contractor to prove transparency and accountability during and after the man-hour based FEED by maintaining clear open book records on the project progress & work delivered. This is important for the project owner to assess, if the scope of work (SoW), quality and execution schedule has been met to satisfaction and avoid feeling like cab passengers who constantly suspects if the cab meter is functioning correctly. This also allows project owners to assess if re-work through change orders will become a habit during the detailing phase of engineering.

4. ***Under/Over Quoting***: When contractors lack experience with similar projects in the past to determine what it takes to execute a project, it either ends up working for free on all that 'extra' SoW or losing the E&C contract to a brighter guy. Therefore if you are a small contractor, start small.

5. ***Take Benchmarking Seriously***: High FEL projects or projects that have clear vision, clarity and scope are expected to have shorter schedules, predictable costs, and completion in all respects. Benchmarking with similar projects that had sound contracts with reputed suppliers gives a good idea where your project is heading towards, quicker and confident final investment decisions (FID) and also aids in eliminating uncertainties that warrant excess contingencies.

6. ***Project Economics & Standardization***: When projects run on low profit margins, instead of cutting corners and getting into trouble, it is more sensible to first understand how project economic factors such as direct & indirect costs, revenue, margin, overheads, taxation affect the project's profitability. Contracts made with vendors who employ Product Standardization, maintain sound balance sheets, ready availability of credit better bet to ensure your project's vendor items are delivered on time to meet project schedules and quality. Therefore, it is preferable during FEL for procurement teams to refer back to previously approved

contractors who meet project owner's business objectives because they better understand the Project Owner's requirements.

Local Laws & Regulations

1. ***Pipeline Location Markers***: It is no doubt a momentous joy in meeting project requirements, executing, completing, running a guarantee test, handing over the keys of the facilities to the project owner & closing the business deal. But if a buried natural gas pipeline that runs through large localities of human occupation without any pipeline location markers & the local government body in-charge of laying roads & electrical cables hit the gas pipeline while digging up, in all likelihood the incident will hit the tabloids when there is an explosion. Hence always have an emergency response plan of action as part of the project plan with constant communication with local civic authorities.
2. ***Right of Way (ROW)***: Project Owners, Project Managers, project engineers and all relevant stake holders have the duty to follow all local laws and meet environmental regulations. When project owners skip such an early FEL step & Engineers are busy proving their calibre laying an above-ground gas pipeline laden with high H_2S content through a forest area with no cognizance of the local habitat or environmental regulations, one wouldn't want to see an elephant stepping on it. When the pipeline ruptures, with all that hydrogen sulphide laden gas spewing out killing the surrounding habitat due to poisoning & explosions, the project owner can be sure to become the next subject of a Hollywood movie or a Greenpeace Activist's Documentary.

   In protected habitats, the ROW of local flora & fauna gains first priority over Project Owner's ROW. Therefore, project managers have the mandatory task of keeping track of Local environmental regulations from the earliest stage of Front End Loading.
3. ***Planning ROW Path***: Not all projects are expected to receive the kind of budgets to build oil & gas facilities in one go and hence projects are implemented in phases. Sometimes, though budgets are sanctioned, projects are not implemented considering unforeseeable poor market demand. In the event where the project is expected to go through a later stage expansion or when an underestimation of market demand causes downsizing the infrastructure but market demand increases at a later stage, the existing pipeline capacity becomes insufficient to transport. In such cases, it is prudent to plan early during the FEL stage to acquire and accommodate additional ROW for future pipeline expansions. However to do so, local landowners and governmental authorities must be consulted early to acquire the requisite land and approvals for gaining ROW rights.

References & Further Reading

1. Lessons Learned from UKCS Oil and Gas Projects 2011-2016, Oil & Gas Authority

Module 23
Flash Steam and Steam Condensates in Return Lines

In power plants, boiler feed water is subjected to heat thereby producing steam which acts as a motive force for a steam turbine. The steam upon doing work loses energy to form condensate and is recycled/returned back to reduce the required make up boiler feed water (BFW).

Recycling steam condensate poses its own challenges. Flash Steam is defined as steam generated from steam condensate due to a drop in pressure. When high pressure and temperature condensate passes through process elements such as steam traps or pressure reducing valves to lose pressure, the condensate flashes to form steam. Greater the drop in pressure, greater is the flash steam generated. This results in a two phase flow in the condensate return lines

General Notes

1. To size condensate return lines, the primary input data required to be estimated is A. Fraction of Flash Steam and condensate, B. Flow Rates of Flash Steam & condensate, C. Specific volume of flash steam & condensates, D. Velocity limits across the condensate return lines.
2. Sizing condensate return lines also require lower velocity limits for wet steam since liquid droplets at higher velocities cause internal erosion in pipes and excessive piping vibration. A rule of thumb, for saturated wet steam is 25 – 40 m/s for short lines of the order of a few tens of metres and 15 - 20 m/s for longer lines of the order of a few hundred metres.
3. Condensate return lines work on the principle of gravity draining. To effectuate this, drain lines are to be sloped downward at a ratio of atleast 1:100.
4. Proper sizing of stem condensate return lines requires consideration of all operating scenarios, chiefly start up, shutdown and during normal running conditions. During plant start up, steam is not generated instantly. As a result, the condensate lines would be filled with liquids which gradually turn two-phase until reaching normal running conditions. During shutdown conditions, with time, flash steam in the lines condense leaving behind condensates due to natural cooling.
5. Condensate return line design must also consider the effects of water hammering. When multiple steam return lines are connected to a header pipe that is routed to a flash drum, flash steam in the presence of cooler liquid from other streams would condense rapidly to cause a water hammer.

Fraction of Flash Steam

Taking an example case, condensate flows across a control valve from an upstream pressure of 5 bara to 2 bara downstream. The saturation temperature at 5 bara is 151.84 ^{0}C & 120.2^{0}C at 2 bara. The specific volume of water at 5 bara is 0.001093 m^3/kg & 0.00106 m^3/kg at 2 bara. The latent heat of saturated steam upon reaching 2 bara is 2201.56 kJ/kg. The % flash steam generated is estimated as,

$$h_{f,1} = h_{f,2} + \left[\frac{\% \, Flash}{100} \times h_{fg}\right] \qquad (1)$$

Where,

$h_{f,1}$ = Upstream specific enthalpy [kJ/kg]

$h_{f,2}$ = Downstream specific enthalpy [kJ/kg]

$h_{f,g}$ = Latent Heat of Saturated Steam [kJ/kg]

The upstream specific enthalpy, h_{f1} of saturated water at 5 bara is 640.185 kJ/kg and h_{f2} of 504.684 kJ/kg at 2 bara. The steam specific volume at 2 bara is 0.8858 m^3/kg.

The fraction of flash steam is calculated as,

$$\% \ Flash = \frac{[640.185-504.684]}{2201.56} \times 100 = 6.15\% \quad (2)$$

Therefore the condensate fraction is,

$$\% \ Cond = 100 - 6.15 = 93.85\% \quad (3)$$

The steam volume is calculated as,

$$V_{Steam} = 0.8858 \times 0.0615 = 0.05448 \ \frac{m^3}{kg} \quad (4)$$

The condensate volume is calculated as,

$$V_{Cond} = 0.00106 \times 0.9385 = 0.000995 \frac{m^3}{kg} \quad (5)$$

Condensate Return Pipe Sizing

To size the condensate return line, the bulk properties and mixture properties can be used to estimate the pipe size. It must be remembered that as the two-phase mixture travels through the pipe, there is a pressure profile that causes the flash % to change along the pipe length. Additionally due to the pipe inclination, a certain amount of static head is added to the total pressure drop.

To estimate the pipe pressure drop across the pipe length, a homogenous model for modelling the two phase pressure drop can be adopted. The homogenous mixture acts as a pseudo-fluid, that obeys conventional design based on single phase fluids characterized by the fluid's average properties.

The mixture properties can be estimated as,

$$\rho_h = \rho_L[1-\varepsilon_h] + \rho_v \varepsilon_h \quad (6)$$

Where,

ρ_L = Condensate Density [kg/m^3]

ρ_v = Steam Density [kg/m^3]

ε_h = Homogenous void fraction for a given steam quality [x] [-]

The homogenous void fraction [ε_h] for a given steam quality [x] can be estimated as,

$$\varepsilon_h = \frac{1}{1+\left[\frac{u_v}{u_L} \times \frac{1-x}{x} \times \frac{\rho_v}{\rho_L}\right]} \quad (7)$$

The dynamic viscosity for calculating the Reynolds number can be chosen as the viscosity of the liquid phase or a quality averaged viscosity, μ_h.

$$\mu_h = x\mu_v + [1-x]\mu_L \quad (8)$$

The homogenous model for gravitational pressure drop is applicable for large drop in pressures and mass velocities < 2000 kg/m^2.s, such that sufficient turbulence exists to cause both phases to mix properly and ensure the slip ratio (u_v/u_L) between the vapour and liquid phase is ~1.0. For more precise estimates capturing slip ratios and varying void fraction, correlations such as Friedal (1979), Chisholm (1973) or Muller-Steinhagen & Heck (1986) can be used.

The total pressure drop is the sum of the static head, frictional pressure drop & pressure drop due to momentum pressure gradient.

$$\Delta P_T = \Delta P_{static} + \Delta P_{mom} + \Delta P_{fric} \quad (9)$$

The Static Head [ΔP_{static}] is computed as,

$$\Delta P_{static}[bar] = \frac{H \times \rho_h \times g \times Sin\theta}{10^5} \quad (10)$$

Where,

H = Pipe Elevation [m]

θ = Pipe inclination w.r.t horizontal [degrees]

The pressure drop due to momentum pressure gradient [ΔP_{mom}] is,

$$\frac{dP}{dZ} = \frac{d\left(m/\rho_h\right)}{dZ} \quad (11)$$

If the vapour fraction remains constant across the piping, the pressure drop due to momentum pressure gradient is negligible.

The frictional pressure drop is calculated as,

$$\Delta P_f = \frac{f \times L \times \rho_h \times V^2}{2D} \quad (12)$$

Where, ΔP = Pressure drop [bar]

f =Darcy Friction Factor [-]

L = Pipe Length [m]

ρ_h = Mixture Density [kg/m³]

V = Bulk fluid Velocity [m/s]

D = Pipe Inner Diameter, ID [m]

$$\mathrm{Re} = \frac{\mathrm{DV}\rho_\mathrm{h}}{\mu_\mathrm{h}} \quad (13)$$

Where, μ_h = Dynamic Viscosity [kg.m/s]

ρ_h = Homogenous Density [kg/m³]

The Darcy Friction Factor [*f*] depends on the Reynolds number follows the following criteria,

If *Re* <= 2100 ; Hagen Poiseuille's Equation

If *Re* <= 4000 ; Churchill Equation

If Re > 4000 ; Colebrook Equation

The Laminar Flow equation also referred to as the Hagen Poiseuille's equation is,

$$\mathrm{f} = \frac{64}{\mathrm{Re}} \quad (14)$$

The Churchill equation combines both the expressions for friction factor in both laminar & turbulent flow regimes. It is accurate to within the error of the data used to construct the Moody diagram. This model also provides an estimate for the intermediate (transition) region; however this should be used with caution.

The Churchill equation shows very good agreement with the Darcy equation for laminar flow, accuracy through the transitional flow regime is unknown & in the turbulent regime a difference of around 0.5-2% is observed between the Churchill equation and the Colebrook equation. For Reynolds number up to ~4000,

$$\mathrm{f} = 8\left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + \frac{1}{(\mathrm{A}+\mathrm{B})^{1.5}}\right]^{1/12} \quad (15)$$

$$\mathrm{A} = \left[2.457\ln\left(\frac{1}{\left(\frac{7}{\mathrm{Re}}\right)^{0.9} + 0.27\frac{\varepsilon}{\mathrm{D}}}\right)\right]^{16} \quad (16)$$

$$\mathrm{B} = \left[\left(\frac{37{,}530}{\mathrm{Re}}\right)\right]^{16} \quad (17)$$

The Colebrook equation was developed taking into account experimental results for the flow through both smooth and rough pipe. It is valid only in the turbulent regime for fluid filled pipes. Due to the implicit nature of this equation it must be solved iteratively. A result of suitable accuracy for almost all industrial applications will be achieved in less than 10 iterations. For Reynolds number up greater than ~4000,

$$\frac{1}{\sqrt{\mathrm{f}}} = -2\log_{10}\left[\frac{\varepsilon/\mathrm{D_H}}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{\mathrm{f}}}\right] \quad (18)$$

Homogenous Property Calculations

The two phase mixture flows through the condensate return line. The associated density and viscosity of flash steam and condensate at 2 bara and 120.2⁰C is,

$$\rho_v = \frac{1}{0.8858} = 1.129\ \frac{kg}{m^3} \quad (19)$$

$$\rho_L = \frac{1}{0.00106} = 943.4\ \frac{kg}{m^3} \quad (20)$$

$$\mu_v = 0.000229\ \frac{kg}{m.s} \quad (21)$$

$$\mu_L = 0.0000128\ \frac{kg}{m.s} \quad (22)$$

The homogenous void fraction [ε_h] for a slip ratio (u_v/u_L) of 1.0, i.e., $u_v = u_L$, and a steam quality [x] of 6.15% is,

$$\varepsilon_h = \frac{1}{1+\left[1\times\frac{1-0.0615}{0.0615}\times\frac{1.129}{943.4}\right]} = 0.9821 \quad (23)$$

The two phase homogeneous density is,

$$\rho_h = 943.4 \times [1 - 0.9821] + [1.129 \times 0.9821] \quad (24)$$

$$\rho_h = 18.014\ \frac{kg}{m^3} \quad (25)$$

The two phase homogeneous viscosity is,

$$\mu_h = \frac{0.0615\times1.28}{10^5} + \frac{[1-0.0615]\times2.29}{10^4} \quad (26)$$

$$\mu_h = 0.000216\frac{kg}{m.s} \quad (27)$$

Pressure Drop Calculations

The return condensate line from the control valve discharge is sloped at a ratio of 1:100 for gravity drain. The layout of the return condensate line is,

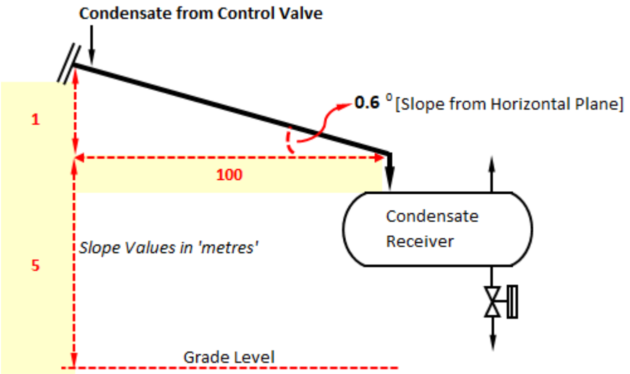


Figure 1. Condensate Return Line to Receiver

The condensate receiver operates at 1.1 bara pressure. The mechanical details of the piping for a flow rate of 1,000 kg/h, pipe size of 1.5", 100m length & pipe roughness of 45.2 μm is,

Table 1. Condensate Return Line Details

Parameter	Value	Unit
Mass Flow rate [m]	1000.0	kg/h
Volumetric Flow [Q]	55.51	m^3/h
Pipe Length [L]	100	m
Pipe Roughness [ε]	45.2	μm
Pipe Outer Diameter [OD]	48.3	mm
Pipe SMYS [Carbon Steel]	30,000	psi
Pipe Design Pressure [DP]	7	bara
Pipe Wall Thickness [WT]	0.08	mm
Corrosion Allowance [CA]	1.0	mm
Calculated WT	1.08	mm
Selected WT	3.68	mm
Pipe Inner Diameter [ID]	40.94	mm

The pipe wall thickness chosen is based on ASME/ANSI B36.10M and is calculated based on the hoop stress created by internal pressure in a thin wall cylindrical vessel as,

$$\mathrm{WT} = \frac{\mathrm{DP}\times\mathrm{OD}}{2\times\mathrm{SMYS}} = \frac{[7\times14.5]\times\left[\frac{48.3}{25.4}\right]}{2\times30{,}000}\times25.4 \quad (28)$$

$$\mathrm{WT} = 0.08mm \quad (29)$$

Adding CA of 1 mm, the WT becomes 1.08 mm. Based on ASME/ANSI B36.10M, the selected WT is 3.68mm. The inner diameter calculated for the selected WT is 40.94 mm.

The condensate return line mixture fluid velocity is calculated as,

$$\mathrm{V} = \frac{\mathrm{Q}}{\mathrm{A}} = \frac{4\times\left[\frac{1000}{18.01}\right]\times\frac{1}{3{,}600}}{\pi\times[0.04094]^2} = 11.7142\ m/s \quad (30)$$

The Reynolds number is estimated as,

$$\mathrm{Re} = \frac{\mathrm{ID}\times\mathrm{V}\times\rho_h}{\mu_h} = \frac{0.04094\times11.7\times18.01}{0.000216} \quad (31)$$

$$\mathrm{Re} \approx 39{,}971 \text{ (MS-Excel computed)} \quad (32)$$

Since the Reynolds number is much higher than 4,000, the flow is fully turbulent and the friction factor is calculated based on Colebrook equation. The friction factor is estimated as,

$$f = f_{Colebrook} = 0.0251 \quad (33)$$

The frictional pressure drop is now calculated using the Darcy-Weisbach expression as,

$$\Delta P_f = \frac{0.0251\times100\times18.01\times11.7^2}{2\times0.04094\times10^5} \quad (34)$$

$$\Delta P_f = 0.757\ bar \quad (35)$$

The slope angle is calculated as,

$$\theta = \left[Tan^{-1}\left(\frac{1}{100}\right)\right]\times\frac{180}{\pi} = 0.6° \quad (36)$$

The static pressure drop [ΔP_{static}] becomes

$$\Delta P_{static} = \frac{18.01\times9.81\times[(1+5)\times sin(0.6°)]}{10^5} \quad (37)$$

$$\Delta P_{static} = 0.000106\ bar \quad (38)$$

Therefore the total ΔP with negligible ΔP due to momentum pressure gradient [ΔP_{mom}].

$$\Delta P_{total} = \Delta P_{static} + \Delta P_f \quad (39)$$

$$\Delta P_{total} = 0.757 + 0.000106 = 0.757\ bar \quad (40)$$

The condensate exit pressure is 2 – 0.757 = 1.243 bara which is higher than the receiver's operating pressure of 1.1 bara.

References & Further Reading

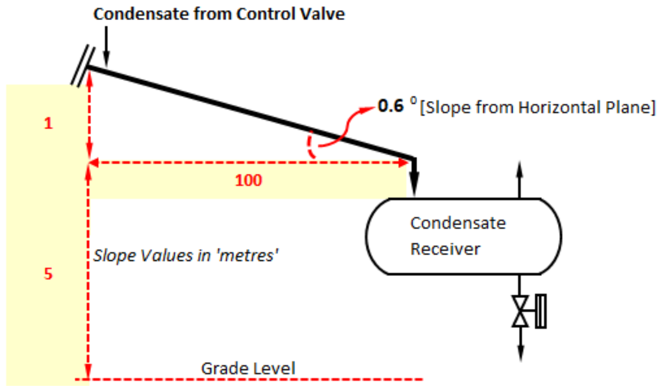
1. "*Engineering Data Book III*", Ch 13, Two Phase Pressure Drop, Wolverine Tube, Inc.
2. *"Steam Handbook"*, Dr. Ian Roberts, Philip Stoor, Michael Carr, Dr. Rainer Hocker, Oliver Seifert, Endress+Hauser

Appendix A

Steam Data		
Parameter	**Value**	**Unit**
Control Valve Upstream		
Upstream Pressure	5.0	bara
Upstream Saturation Temperature at 5 bara	151.8	^oC
Specific Volume of Saturated Water at 5 bara	0.001093	m^3/kg
Specific Enthalpy of Saturated Water at 5 bara [hf1]	640.2	kJ/kg
Control Valve Downstream		
Downstream Pressure	2.0	bara
Downstream Saturation Temperature	120.2	^oC
Specific Volume of Saturated Water at 2 bara	0.001060	m^3/kg
Specific Enthalpy of Saturated Water at 2 bara [hf2]	504.7	kJ/kg
Latent Heat of Saturated Steam at 2 bara [hfg]	2,201.6	kJ/kg
Steam Specific Volume at 2 bara	0.8858	m^3/kg
Condensate Viscosity [μ_L]	2.29E-04	kg/m.s
Vapour Viscosity [μ_V]	1.28E-05	kg/m.s
Flash Steam & Condensate Calc [ΔP Based on Homogeneous Model]		
% Flash	6.15	%
% Condensate	93.85	%
Steam Volume [V_{steam}]	0.054519	m^3/kg
Condensate Volume [V_{cond}]	0.000995	m^3/kg
Vapour Quality [x]	0.0615	-
Condensate Density [ρ_L]	943.40	kg/m^3
Vapour Density [ρ_V]	1.129	kg/m^3
Slip Ratio [U_V/U_L]	1.0	-
Homogeneous Void Fraction [ε_h]	0.9821	-
Homogeneous Density [ρ_h]	18.01	kg/m^3
Homogeneous Viscosity [μ_h]	2.16E-04	kg/m.s

Legend	
Legend	Input
	Output

Condensate Return Line Sizing Input		
Parameter	**Value**	**Unit**
Mass Flow rate [m]	1000.0	kg/h
Volumetric Flow [Q]	55.51	m^3/h
Pipe Length [L]	100	m
Pipe Roughness [ε]	45.2	μm
Pipe Outer Diameter [OD]	48.3	mm
Pipe SMYS [Carbon Steel]	30,000	psi
Pipe Design Pressure [DP]	7	bara
Pipe Wall Thickness	0.08	mm
Corrosion Allowance	1.0	mm
Calculated Wall Thickness	1.08	mm
Selected Wall Thickness [WT]	3.68	mm
Pipe Inner Diameter [ID]	40.94	mm
Return Line Fluid Velocity	11.7	m/s
Friction Factor [*f*] Calculations		
Laminar Flow - Friction Factor [f]		
Laminar Flow [f]	0.0016	-
Transition Flow - Churchill Equation Friction Factor [f]		
Term A	1.02E+20	-
Term B	3.65E-01	-
Churchill Equation - Friction Factor [f]	0.0252	-
Turbulent Flow - Colebrook Equation Friction Factor [f]		
Colebrook Equation - Friction Factor [f]	0.0251	-
Reynold's Number [N_{Re}]	39,971	-
Flow Behaviour	**Turbulent**	-
Pipe Roughness /Pipe ID [ε/D]	0.00110	-
Friction Factor Equation	**Colebrook**	
Friction Factor [f]	0.0251	-
Frictional Pressure Drop [ΔP_f]	0.757	bar
Pressure Drop per 100m [ΔP/L]	0.757	bar/100 m



Static Pressure Drop [ΔP_{Static}]	0.000106	bar
Frictional Pressure Drop [ΔP_f]	0.7568	bar
Total Pressure Drop [ΔP_{Total}]	0.7569	bar

SIZING RESULTS		
Selected Pipe Size	**48.3**	mm
	1.50	inch
Total Pressure Drop [ΔP_{Total}]	**0.7569**	bar
Return Line Fluid Velocity	**11.7**	m/s
Flow Behaviour	**Turbulent**	-
Friction Factor Equation	**Colebrook**	-

Notes

1. If N_{Re} <= 2100 ; Use Laminar Flow Friction Factor
2. If N_{Re} <= 4000 ; Use Churchill Equation
3. If N_{Re} > 4000 ; Use Colebrook Equation
4. Pressure Drop is estimated using Darcy-Weisbach Equation

Module 24

Single Phase Liquid Vessel Sizing for HYSYS Dynamics

Process Facilities often have intermediate storage facilities that store liquids prior to transporting to downstream equipment. The period of storage is short, i.e., of the order of minutes to hours & is defined as Holdup time. The Holdup time can also be explained as the reserve volume required to ensure safe & controlled operation of downstream equipment. The intermediate vessel also acts as a buffer vessel to accommodate any surge/spikes in flow rates, and is termed as surge time. Vessel volume is an input data required in process dynamic simulation and the following covers estimation of volume required for single phase liquid flow into an intermediate vertical/horizontal/flat bottomed vessel.

Problem Statement

Water at 1,341 m^3/h, flows into a vessel & held for a holdup time of 1 min before discharging into downstream equipment. The vessel's liquid percent level is desired to be held at 50% (half full) for a certain drain rate. Estimate the size of the vessel required for an L/D ratio of 1.

Design Methodology & Results

Based on the above data, the vessel volume [V] for a flow rate of 1,341 m^3/h is,

$$\boldsymbol{V = Q[m^3/h] \times \frac{T_{holdup}[min]}{60} \times \frac{100}{\%\ Liquid\ Level}} \quad (1)$$

Based on the vessel volume estimated for the holdup time, the dimensions of the vessel are,

$$\boldsymbol{D = \sqrt{\frac{4V}{\pi L}};\ Where\ L = n \times D} \quad (2)$$

Substituting the values for vessel dimensions,

$$\boldsymbol{V = 1341 \times \frac{1}{60} \times \frac{100}{50} = 44.7 m^3} \quad (3)$$

Taking L = D, i.e., L/D ratio of 1.0

$$\boldsymbol{D = \sqrt[3]{\frac{4 \times 44.7}{\pi}} \approx 3.9\ m}, \text{For L = D; } \mathbf{L = 3.9\ m} \quad (4)$$

Surge Study

From the estimates, the vessel chosen is flat bottomed on concrete foundation & is subjected to a peak flowrate rise of 1,474 m^3/h in a 2 min interval. The liquid level rises to ~57% from 50%.

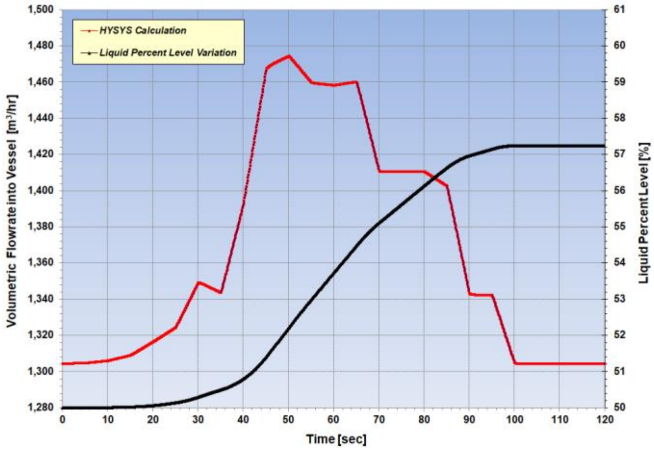


Figure 1. Surge Flow rate increase on % Liquid Level

Thumb Rules

The different arrangement types are as follows,

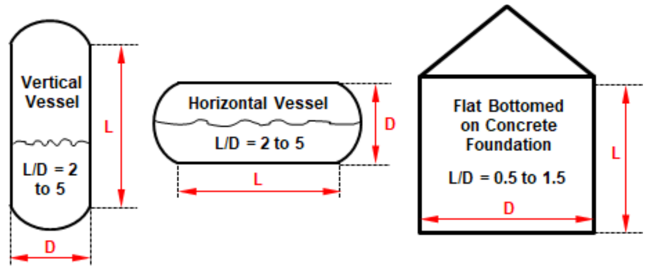


Figure 2. Intermediate Storage Vessel Types & L/D Ratio

1. Holding time for most intermediate tanks is 10 min (half full tanks)
2. Holding time for feed tanks to furnace is 30 min (half full).
3. The optimum ratio is 3 for commonly used L/D ratios of 2 to 5.
4. Vessels < 4 m^3, are vertically mounted with L/D ratio of 2 to 5 on leg supports/ brackets.
5. Vessels 4 m^3 < V < 40 m^3 is horizontal & saddle supported with L/D ratio of 2 to 5.
6. Vessels V > 40 m^3 is flat bottom tank on concrete foundation with L/D ratio of 0.5-1.5.

References & Further Reading

Perry's Chemical Engineers Handbook, 7th Edition

Module 25

Key Thermo-Physical Properties of Light Crude Oils

Process facilities are equipped with protection measures, such as pressure safety valves (PSV) & as a minimum, PSVs are sized for a fire case. To do so for a pressure vessel containing crude oil a key parameter is the Latent heat of Vaporization [H_v].

For pure components, the Joback's Method can be employed which uses basic structural information of the chemical molecule to estimate thermo-physical data. However it can be complex for equipment that contains crude oil because the plus fractions [C_{7+}] can contain thousands of straight chain, cyclic & functional groups. Therefore by splitting and lumping the crude fractions, a smaller number of components are arrived at, to characterize and be able to apply Equation of State (EoS) correlations to estimate the fraction's thermo-physical properties.

To estimate properties such as MW, Specific gravity [γ], Critical Pressure [P_c], Critical Temperature [T_c] and Latent heat of Vaporization [H_v], the following module provides few correlations applicable for light crudes with boiling points < 455^{0}C based on D-86 Distillation curves.

General Notes

1. Latent heat of Vaporization [H_v], can be estimated using critical properties of the plus fractions in the hydrocarbon mixtures.
2. Oil fractions tend to decompose at ~650^{0}F (344^{0}C) at 1 atm. As a result, it becomes necessary to lower the pressure to as low as 40mm Hg to obtain the True Boiling Point (TBP) distillation curves. ASTM methods can be used to convert the resulting boiling point curve into TBP curves using correlations from API Technical Data Book – Petroleum Refining.
3. The ASTM D-86 distillation of an oil fraction is conducted in laboratory room conditions at 1 atm and the D-86 distillation curve ends at ~650^{0}F (344^{0}C).
4. The ASTM D-1160 distillation of an oil fraction is conducted at much lower pressures, typically 10 mmHg for heavier oils with high boiling points to prevent decomposition of the oil sample. With this method, oil fractions can be distilled upto ~950^{0}F to ~1000^{0}F (510^{0}C to 538^{0}C), reported on a 760mm Hg basis.
5. The boiling point of all compounds in a crude mixture can be represented by a single characteristic boiling point called Volume Average Boiling Point [VABP]. Since the individual mole fractions of the petroleum stream is not known, VABP is calculated from standard distillation data [ASTM D-86] followed by calculating the mean average boiling point [MeABP].
6. The Molecular weight [MW], Specific gravity [γ] & boiling point [T_b] are taken as the key properties to define the makeup of a petroleum fraction. In this module, the Katz-Firoozabadi [1978], Riazi-Daubert [1980, 1987] & Ahmed [1985] correlations are shown to predict MW, specific gravity [γ], Critical Pressure [P_c] & Critical Temperature [T_c]. To estimate H_v, Riedel correlation is employed to estimate the Latent Heat of Vaporization [$H_{v,NBP}$] at Normal Boiling Point [MeABP/NBP/T_b]. Watson relation is used to estimate $H_{V,T}$ at desired temperature.

Selected Correlations

The below table gives a summary of the two generalized correlations to estimate *MW*, P_c, T_c and H_v of the petroleum fraction.

Table 1. Generalized Correlations for P_c, T_c and MW

Katz-Firozabaadi Correlation [1978]
$\mathbf{MW}\left[\frac{\mathbf{kg}}{\mathbf{kmol}}\right] = \left[\frac{6.97996-\ln[1080-T_b]}{0.01964}\right]^{3/2}$ $\gamma[-] = 1.07 - e^{[3.56073-(2.93886\times MW^{0.1})]}$ $P_c[bara] = e^{[6.34492-(0.7239\times MW^{0.299})]}$ $T_{br}[-] = 1.2 - e^{[-0.34742-(0.02327\times MW^{0.55})]}$ $T_c[°K] = \frac{T_b}{T_{br}}$ $N_c[-] = \left[\frac{[6.9955-\ln(1090-T_b)]}{0.11193}\right]^{3/2}$
$T_b = °K$
Riazi-Daubert Extended Correlation [1980]
$\mathbf{MW}\left[\frac{\mathbf{kg}}{\mathbf{kmol}}\right] = \left[42.965 \times T_b^{1.26007} \times \gamma^{4.98308}\right] \times e^{[(0.0002097\times T_b)-(7.78712\times\gamma)+(0.00208476\times T_b\times\gamma)]}$
$T_b = °K$; $\gamma = \text{Unitless}$
Riazi-Daubert Correlation [1987]
$P_c[psia] = \left[45203 \times MW^{-0.8063} \times \gamma^{1.6015}\right] \times e^{[(-0.0018078\times MW)+(-0.3084\times\gamma)]}$ $T_c[°R] = \left[544.4 \times MW^{0.2998} \times \gamma^{1.0555}\right] \times e^{[(-0.00013478\times MW)+(-0.61641\times\gamma)]}$

The Latent Heat of Vaporization [H_v] is calculated as,

Table 2. Riedel Correlation and Watson Relation

Riedel Correlation
$H_{v,NBP}[kJ/mol] = \frac{1.092\times8.3145\times T_b\times[\ln P_c-1.013]}{\left[0.93-\frac{T_b}{T_c}\right]\times1000}$
$P_c = bara$; $T_c = °K$; $T_b = °K$
Watson Relation
$H_v[kJ/mol] = H_{v,NBP} \times \left[\frac{T_c-T}{T_c-T_b}\right]^{0.38}$
$T = °K$; $T_c = °K$; $T_c = °K$; $H_v = kJ/mol$

It is to be noted that, the Katz-Firoozabadi [1978] correlation was originally based on Kreglewski and Zwolinski [1961] generalized expression which is of the form,

$$\theta = \theta_\infty - e^{[a-(b\times MW^c)]} \qquad (1)$$

Where, θ represents the properties such as T_b, γ, P_c and T_{br}.

The value of the constants, *a*, *b*, *c* in the above expression is based on a tabulated set generated from the physical properties of 26 condensates and crude oil systems. The value of γ based on MW in the Katz-Firoozabadi correlation predicts within 0.4% for straight chain numbers [SCN] groups from C_6 to C_{50}. Similarly, the absolute average deviation (AAD%) of the Kreglewski and Zwolinksi [1961] correlation gives an AAD% of 0.4%, 0.07%, 0.15% and 1% in the properties of T_b, γ, T_{br}, P_c respectively between correlations and physical properties of the 26 condensates and crude oil systems.

VABP and MeABP Calculation

For petroleum fractions usually, there would be no information available about the weight, mole or volume fractions considering the large number of compounds present. In such cases, the ASTM based D-86 distillation data for light oils (API Gravity > 31^0API and D-86 Temperatures < 455^0C) can be used to estimate the Volume Average Boiling Point (VABP) and Mean Average Boiling Point (MeABP) which can be calculated as follows,

$$VABP[°C] = \frac{T_{10\%}+T_{30\%}+T_{50\%}+T_{70\%}+T_{90\%}}{5} \qquad (2)$$

It is to be noted that when the average boiling point (ABP) of a crude sample is estimated based on weight (W), moles (M) and volume (V) basis, there would exist a difference in each of these average boiling points. To relate the different types of ABPs, the VABP value is corrected with a slope line and correction factor line to find other ABPs. The Slope Line (S) is estimated as,

$$S[°C/\%\ Recovered] = \frac{T_{90\%}-T_{10\%}}{80} \qquad (3)$$

With the S value, the correction factor, ΔT_{MeA} is estimated using the empirical expression,

$$\Delta T_{MeA} = -1.53181 - [0.0128 \times VABP^{0.6667}] + [3.646064 \times S^{0.333}] \quad (4)$$

Where,

VABP = Volume Average Boiling Point [^{0}C]

With the correction factor, ΔT_{MeA}, the MeABP is estimated as,

$$MeABP[°C] = VABP - \Delta T_{MeA} \quad (5)$$

To estimate the critical properties, MW and latent heat of vaporization [H_v], MeABP becomes the normal boiling point, T_b.

Case Study

Light Crude Oil is present in a process vessel at 325^{0}K [51.85^{0}C]. To size a PSV for fire case, the latent heat of vaporization [H_v] value is required to be computed. The D-86 distillation curves are as follows,

Table 3. ASTM D86 Vol% vs Temperature

Vol% [ASTM D86]	D86 Temperature [^{0}C]
0 [IBP]	155.1
10	179.1
30	222.4
50	260.3
70	289.0
90	315.7
100 [FBP]	352.9

With the available data, VABP is estimated as,

$$VABP[°C] = \frac{179.1+222.4+260.3+289+315.7}{5} \quad (6)$$

$$VABP[°C] = 253.3°C \quad (7)$$

The slope, S is estimated as,

$$S\ [°C/\%\ Recovered] = \frac{315.7-179.1}{80} = 1.7075 \quad (8)$$

The correction factor ΔT_{MeA} becomes,

$$\Delta T_{MeA} = -1.53181 - [0.0128 \times 253.3^{0.6667}] + [3.646064 \times 1.7079^{0.333}] = 10.1°C \quad (9)$$

The MeABP is estimated as,

$$MeABP[°C] = T_b = 253 - 10.1 \cong 243°C \quad (10)$$

$$\text{Or, } T_b \cong 243°C \cong 516°K \cong 929°R \quad (11)$$

Applying the MeABP/T_b value, the critical properties, γ and MW is estimated as follows,

$$MW\left[\frac{kg}{kmol}\right] = \left[\frac{6.97996-\ln[1080-516]}{0.01964}\right]^{3/2} = 188.4 \quad (12)$$

$$\gamma = 1.07 - e^{[3.56073-(2.93886\times188.4^{0.1})]} = 0.8238 \quad (13)$$

$$\text{API Gravity} = \frac{141.5}{0.8238} - 131.5 = 40.26°API \quad (14)$$

$$P_c = e^{[6.34492-(0.7239\times188.4^{0.299})]} = 17.8\ bara \quad (15)$$

$$T_{br} = 1.2 - e^{[-0.34742-(0.02327\times188.4^{0.55})]} = 0.733 \quad (16)$$

$$T_c = \frac{516}{0.7335} = 704°K \quad (17)$$

$$N_c = \left[\frac{[6.9955-\ln(1090-516)]}{0.11193}\right]^{3/2} = 13.78 \quad (18)$$

Similarly applying Riazi-Daubert correlations from Table 1 with γ=0.8238 and T_b = 516^{0}K.

$$MW\left[\frac{kg}{kmol}\right] = 189.8 \quad (19)$$

$$P_c = 266\text{ psia} = 18.3\ bara \quad (20)$$

$$T_c = 1{,}254°R = 697°K \quad (21)$$

The Latent Heat of Vaporization [$H_{v,NBP}$] based on Katz-Firoozabadi P_c, T_c, T_b data is,

$$H_{v,NBP} = \frac{9.079434\times516\times[\ln17.8-1.013]}{\left[0.93-\frac{516}{704}\right]\times1000} \cong 44.5\frac{kJ}{mol} \quad (22)$$

At 325^{0}K, $H_{v,T}$ is,

$$H_{v,T} = 44.49 \times \left[\frac{704-325}{704-516}\right]^{0.38} = 58.07\frac{kJ}{mol} \quad (23)$$

Similarly, using Riazi-Daubert P_c, T_c, T_b data,

$$H_{v,NBP} = 46.96\frac{kJ}{mol} \quad (24)$$

$$\text{At } 325^0K,\ H_{v,T} = 61.8\frac{kJ}{mol} \quad (25)$$

Additional Correlation – Ahmed [1985]

Based on Ahmed [1985] correlation of the Katz-Firoozabadi [1978], physical properties are tabulated with the number of carbon atoms using a regression model of the form,

$$\theta = a_1 + a_2 n + a_3 n^2 + a_4 n^3 + \frac{a_5}{n} \quad (26)$$

Where,

θ = T_c, P_c

n = number of carbon atoms

a_1, a_2, a_3, a_4, a_5 =coefficients

Table 4. Ahmed [1985] Constants - γ and MW

Property	γ	MW
Coefficients	**[-]**	**[kg/kmol]**
a_1	0.86714949	-131.11375
a_2	0.00341434	24.96156
a_3	-0.00002840	-0.34079022
a_4	2.4943308×10^{8}	0.00249412
a_5	-1.16279840	468.32575

Table 5. Ahmed [1985] Constants - P_c and T_c

Property	P_c	T_c
Coefficients	**[psia]**	**[^{0}R]**
a_1	275.56275	915.53747
a_2	-12.522269	41.421337
a_3	0.29926384	-0.7586859
a_4	-0.00284521	0.00586754
a_5	1711.7226	-1302.8779

Based on Ahmed [1985] correlation,

$$\mathrm{MW}[kg/kmol] = 188.7 \tag{27}$$

$$P_c = 266 \text{ psia} = 19.1\ bara \tag{28}$$

$$\mathrm{T_c} = 1{,}254°\mathrm{R} = 702°\mathrm{K} \tag{29}$$

$$\mathrm{H_{v,NBP}} = 46.71\,\frac{kJ}{mol} \tag{30}$$

$$\text{At } 325^0\text{K, } \mathrm{H_{v,T}} = 61.15\,\frac{kJ}{mol} \tag{31}$$

Results

Summarizing the results,

Table 6. Results Summary

Property	Katz-Firoozabadi	Riazi-Daubert	Ahmed [1985]
MW [kg/kmol]	188.4	189.8	188.7
γ [-]	0.8238	-	0.8245
P_c [bara]	17.8	18.3	19.1
T_c [^{0}K]	704	697	702
H_v [kJ/mol]	44.49	46.96	46.71
$H_{v,T}$ [kJ/mol]	58.07	61.80	61.15

Taking an average of the estimates made, the critical properties, MW and H_v is estimated as,

Table 7. Average of Estimates

Property	Average Properties
MW [kg/kmol]	189.0
γ [-]	0.8242
P_c [bara]	18.4
T_c [^{0}K]	701
$H_{v,NBP}$ [kJ/mol]	46.05
$H_{v,325^0K}$ [kJ/mol]	60.36

References & Further Reading

1. “*Physical Properties of Heavy Petroleum Fractions and Crude Oils*”, Mohammad. R. Riazi, Taher A. Al-Sahhaf, Fluid Phase Equilibria, 117 (1996) 217-224
2. “*Equation of State and PVT Analysis*”, Tarek Ahmed, Gulf Publishing Company
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4. “*Evaluation of Different Correlation Performance for the calculation of Critical Properties and Acentric Factor of Petroleum Heavy Fractions*”, Dacid B. L, Rafel B. S, Andre P.C.M.V, Adolfo P. P, Viatcheslav I. P, http://dx.doi.org/10.5772/intechopen.71166

Appendix: MS-Excel Calculations

Crude Assay Data [Light Oils < 455^{o}C, $N_c \leq$ C45]

Vol% [D-86]	D-86 Temperature	Katz-Firoozabadi [1978]		Ahmed [1985]	
[%]	[^{o}C]	MW based on MeABP/T_b	188.4	MW	188.7
0	155.1	Specific Gravity [γ]	0.8238	Specific Gravity [γ]	0.8245
10	179.1	API Gravity [API]	40.26	API Gravity [API]	40.12
30	222.4	Critical Pressure [P_c] [bara]	17.8	P_c [bara]	19.1
50	260.3	Reduced Temperature [T_{br}] [-]	0.733	P_c [psia]	277
70	289.0	Critical Temperature [T_c] [^{o}K]	704	T_c [^{o}K]	702
90	315.7	Carbon Number [N_c]	13.78	Carbon Number [N_c]	13.78
100	352.9	**Riazi-Daubert Correlation [1987]**			
VABP	253	R-D 1980 Extended [MW]	189.8		
Slope [^{o}C / %Recovered]	1.71	P_c [psia]	266		
Correction Factor [ΔT] [K]	10.1	P_c [bara]	18.3		
T_b / NBP/ MeABP [^{o}C]	243	T_c [^{o}R]	1,254		
T_b / NBP/ MeABP [^{o}K]	516	T_c [^{o}K]	697		

Latent Heat of Vapourization

Correlation	Ahmed [1985]	Req. Temperature [T] [^{o}K]	325.0
$\Delta H_{v,NBP}$ [kJ/mol]	**46.71**	**$\Delta H_{v,T}$ [kJ/mol] at T**	**61.15**

Module 26

Evaporation Pond Process Design in Oil & Gas Industry

In the upstream oil & gas industry, produced water is a by-product of well production. Hydrocarbon wells initially produce less water but in late field life, the water content increases. Produced water can contain oil carryover and a host of salts with TDS ranging anywhere from 2,000 mg/L to 40,000 mg/L for which evaporation ponds are used to concentrate by evaporating the associated water.

The energy requirement consists of pumping concentrate to the pond and in some cases aeration is provided to enhance the rate of evaporation. The ponds are lined with synthetic liner material to prevent seepage of water into the soil. In case of any corrosive compounds in the water, the number of layers is increased. Landscape and topography play a role in setting up evaporation ponds and it is necessary to have a flat terrain to avoid any overflow of the contents.

Evaporation ponds must also ensure that the amount of water entering is minimized and avoid any flooding. As part of waste disposal, the ponds maybe designed to accumulate sludge over the life time of the operating wells or can be periodically removed. The below figure depicts an evaporation pond.

Figure 1. Evaporation Pond [2]

The following module focuses on estimating the rate of evaporation, water surface temperature and rate of heat transfer to the water in an evaporation pond.

Methodology

Similarities exist between mass, momentum & heat transfer phenomenon. Therefore, the empirical correlations for heat transfer are also applicable for mass transfer. Schmidt number plays a similar role to Prandtl number in convection heat transfer. The heat transfer to the water from the air supplies the energy required to evaporate the water,

$$q = mh_{fg} = hA[t_\infty - t_s] = h_m A h_{fg}[\rho_s - \rho_\infty] \quad (1)$$

Where,

h = Convective heat transfer coefficient [W/m^2.k]

h_m = Convective Mass Transfer Coefficient [m^2/s]

A = Surface Area [m^2]

m = Evaporation Rate [kg/s]

t_s,ρ_s = Surface temperature & vapour density [K, kg/m^3]

t_∞,ρ_∞ = Air Temperature & vapour density [K, kg/m^3]

The energy balance can be arranged as,

$$\rho_s - \rho_\infty = \frac{h}{h_m}\left[\frac{t_\infty - t_s}{h_{fg}}\right] \quad (2)$$

The heat transfer coefficient, h can be estimated based on Nusselt Number (Nu) as,

$$Nu = 0.664 Re^{1/2} Pr^{1/3}\text{, For Laminar Flow} \quad (3)$$

$$Nu = 0.037 Re^{4/5} Pr^{1/3}\text{, For Turbulent Flow} \quad (4)$$

The mass transfer coefficient, h_m can be calculated using Sherwood Number (Sh),

$$Sh = 0.664 Re^{1/2} Sc^{1/3}\text{, For Laminar Flow} \quad (5)$$

$$Sh = 0.037 Re^{4/5} Sc^{1/3}\text{, For Turbulent Flow} \quad (6)$$

Where,

$$h_m = \frac{Sh \times D_v}{L} \quad (7)$$

$$h = \frac{Nu \times k}{L} \quad (8)$$

Dividing both heat and mass transfer coefficients and substituting in Eq. (2) yields,

$$\frac{h}{h_m} = \frac{Nu \times k}{Sh \times D_v} = \left[\frac{Pr}{Sc}\right]^{1/3} \frac{k}{D_v} \quad (9)$$

$$\rho_s - \rho_\infty = \left[\frac{Pr}{Sc}\right]^{1/3} \frac{k}{D_v} \left[\frac{t_\infty - t_s}{h_{fg}}\right] \quad (10)$$

The above expression is solved for ρ_s and h_{fg} is evaluated at surface temperature [t_s]. Air properties, k, D_v, Sc, Pr are evaluated at film temperature [t_f], as an average of t_∞ and t_s.

$$t_f = \frac{t_\infty + t_s}{2} \quad (11)$$

The solution is arrived beginning with a guess value of surface temperature, t_s in Eq. (10) & iteratively solved until convergence. Relating saturated vapour pressure [P_s] with moist air temperature [T_∞, ^{0}C] using Arden Buck equation,

$$P_s[T_\infty > 0°C] = 6.1121 e^{\left[\left(18.678 - \frac{T_\infty}{234.5}\right) \times \left(\frac{T_\infty}{257.14 + T_\infty}\right)\right]} \times 100 \quad (12)$$

Where,

P_s is saturated vapour pressure [Pa]

The vapour density [ρ_∞] for a given relative humidity [RH] is calculated as,

$$\rho_\infty = \left[\frac{P_s \times MW_{water}}{8314.447 \times T_\infty}\right] \times \left[\frac{RH\%}{100}\right] \quad (13)$$

The mass diffusivity of moisture in air [D_v] is estimated using Sherwood and Pigford, 1952 expression, valid for mass diffusivity of water vapour in air up to 1,100^{0}C

$$D_v = \left[\frac{0.926}{P_{amb}}\right] \times \left[\frac{T^{2.5}}{T + 245}\right] \times \frac{1}{10^6} \quad (14)$$

Where,

D_v = Mass diffusivity of moisture in air [m^2/s]

P_{amb} = Atmospheric pressure [kPa]

T = Ambient Temperature [K]

The Schmidt Number (Sc) is estimated as,

$$Sc = \frac{\mu}{\rho_{air} \times D_v} \quad (15)$$

Where, μ = Dynamic Viscosity [kg/m.s]

ρ_{air} = Air density [kg/m^3]

The Reynolds Number (Re) is estimated as,

$$Re = \frac{u_\infty \rho_{air} L}{\mu} \quad (16)$$

Where,

L = Pond Length along direction of air [m]

For the range 0^{0}C to 80^{0}C, the surface temperature [T_s] from curve fit data is,

$$T_s[K] = [(19.45777 \times ln[\rho_s]) + 100.4106] + 273.15 \quad (17)$$

Case Study

Air at 25^{0}C & 101.325 kPa flows at 10 m/s along the length of an evaporation pond of L × W of 10m × 2m. The relative humidity is 60%. The rate of heat transfer to water, rate of evaporation & the water surface temperature is to be estimated. Evaluating the saturated vapour pressure,

$$P_s = 6.1121 e^{\left[\left(18.678 - \frac{25}{234.5}\right) \times \left(\frac{25}{257.14 + 25}\right)\right]} \times 100 \quad (18)$$

$$P_s = 3{,}169\ Pa \quad (19)$$

The vapour density at 25^{0}C is estimated as,

$$\rho_\infty = \left[\frac{3{,}169 \times 18.02}{8314.447 \times 298.15}\right] \times \left[\frac{60}{100}\right] = 0.013822 \frac{kg}{m^3} \quad (20)$$

Taking an initial guess of 15^{0}C, the t_f is,

$$t_f = \frac{15 + 25}{2} = 20°C = 293.15\ K \quad (21)$$

Evaluating air properties at t_f = 20^{0}C, ρ_{air} =1.1975 kg/m^3, μ=0.0000181 kg/m.s, k = 0.0257 W/m^{0}C, h_{fg} = 2,465 kJ/kg, Pr = 0.7094,

$$D_v = \left[\frac{0.926}{101.325}\right] \times \left[\frac{293.15^{2.5}}{293.15 + 245}\right] \times \frac{1}{10^6} = 2.5 \times 10^{-5} \frac{m^2}{s} \quad (22)$$

$$Sc = \frac{0.0000181}{1.1975 \times 2.5 \times 10^{-5}} = \sim 0.6063 \quad (23)$$

$$\rho_s = \left[\frac{0.7094}{0.6063}\right]^{1/3} \times \frac{0.0257}{2.5 \times 10^{-5}} \left[\frac{298.15 - 288.15}{2{,}465 \times 1000}\right] + 0.01382 \quad (24)$$

$$\rho_s = 0.01822\ kg/m^3 \quad (25)$$

Estimating the surface water temperature [T_s] for ρ_s =0.0182 kg/m^3,

$$T_s = [(19.45777 \times ln 0.01822) + 100.4106] + 273.15 \quad (26)$$

$$T_s = 295.62\ K = \sim 22.5°C \quad (27)$$

Recalculating the air properties & iterating the calculations, T_s = 20^{0}C, ρ_s = 0.01601 kg/m³, ρ_{air} = 1.1865 kg/m³, μ = 0.00001826 kg/m.s, h_{fg} = 2,454 kJ/kg and D_v = 2.54×10^{-5} m²/s. The Reynolds number & Sherwood number is estimated as,

$$Re = \frac{10\times1.1865\times10}{0.00001826} = \sim6{,}497{,}679 \quad (28)$$

$$Sh = 0.037 \times 6{,}497{,}679^{4/5} \times 0.6058^{1/3} = 8{,}828 \quad (29)$$

The convective mass transfer coefficient is,

$$h_m = \frac{8{,}828\times2.54\times10^{-5}}{10} = 0.0224\, m/s \quad (30)$$

The rate of evaporation [m] is,

$$m = 0.022426 \times 20 \times [0.01601 - 0.013822] \quad (31)$$

$$m = \sim0.000987\frac{kg}{s} = 85.3\frac{kg}{day} \quad (32)$$

The Rate of heat transfer [q] is,

$$q = mh_{fg} = 0.000987\frac{kg}{s} \times 2{,}454\frac{kJ}{kg} = 2{,}422W \quad (33)$$

References & Further Reading

1. "*Heat Transfer*", 10th Ed, Holman JP.
2. https://www.fws.gov/ecological-services/energy-development/oil-gas.html
3. 2005 ASHRAE Handbook - Fundamentals (SI), Chapter 5

Appendix A

Properties of Air at Atmospheric pressure [From Natl. Bur. Stand. (U.S.) Circ. 564, 1955]					
Temperature		Air Density [ρ]	Dynamic Viscosity [μ]	Thermal Conductivity [*k*]	Prandtl Number [Pr]
[K]	[C]	[kg/m³]	[kg/m.s]	[W/m.°C]	[-]
200	-73.15	1.7684	0.0000133	0.01809	0.7391
250	-23.15	1.4128	0.0000160	0.02227	0.7218
300	26.85	1.1774	0.0000185	0.02624	0.7076
350	76.85	0.9980	0.0000208	0.03003	0.6972
Air-Water Film Properties					
Property = Ax²+Bx+C	**Units**	**A**	**B**	**C**	
Air Density [ρ_{air}]	kg/m³	0.00001762	-0.0147842	4.01723	
Dynamic Viscosity [μ]	kg/m.s	-0.00000000004	0.00000007	0.00000046	
Thermal Conductivity [k]	W/m°C	-0.000000039	0.00010103	-0.0005545	
Prandtl Number [Pr]	-	0.00000069	-0.000657958	0.843212374	
Latent Heat of Vapourization [h_{fg}]	kJ/kg	-0.001370769	-1.542913678	3023.853775	

Evaporation Pond Calculations		
Air Properties		
Ambient Pressure [P]	101.325	kPa
Mean Wind Velocity [u_∞]	10	m/s
Dry Bulb/Air Temperature [T_∞]	25	°C
	298.15	K
Relative Humidity [RH]	60	%
Saturated Vapour Pressure [P_s]	3169	Pa
Vapour Density [rho_inf] at 60% RH	0.01382	kg/m³
Evaporation Pond Dimensions		
Pond Length [L]	10	m
Pond Width [W]	2	m
Direction of Wind	Along Length	m²
Air Density [ρ]	1.1865	kg/m³
Dynamic Viscosity [μ]	0.0000183	kg/m.s
Reynolds Number [Re]	6,497,679	-
Sherwood Number [Sh]	8,828	-
Convective Mass Transfer Coefficient [h_m]	0.0224	m/s
Rate of Evaporation [m]	**85.3**	kg/d
Rate of Heat Transfer to Water [q]	**2,422**	W

Appendix B

Iterations	Surface Temperature of Water [T_s]		Film Temperature [T_f]		Air Density [ρ]	Dynamic Viscosity [μ]	Thermal Conductivity [k]	Latent Heat of Vapourization [h_{fg}]	Prandtl Number [Pr]	Mass diffusivity of Mositure in Air [D_v]	Schimdt Number [Sc]	Vapour Density of Water Surface [ρ_s]
[-]	°C	K	K	°C	kg/m^3	kg/m.s	W/m.°C	kJ/kg	-	m^2/s	-	kg/m^3
1	15	288.2	293.2	20.0	1.1975	0.0000181	0.0257	2,465	0.7094	2.499E-05	0.6063	0.0182
2	22.5	295.6	296.9	23.7	1.1811	0.0000183	0.0260	2,448	0.7085	2.561E-05	0.6056	0.0149
3	18.6	291.7	294.9	21.8	1.1895	0.0000182	0.0259	2,457	0.7090	2.529E-05	0.6060	0.0166
4	20.7	293.8	296.0	22.8	1.1849	0.0000183	0.0259	2,452	0.7087	2.546E-05	0.6058	0.0157
5	19.6	292.7	295.4	22.3	1.1873	0.0000183	0.0259	2,455	0.7088	2.537E-05	0.6059	0.0162
6	20.2	293.3	295.7	22.6	1.1860	0.0000183	0.0259	2,453	0.7088	2.542E-05	0.6058	0.0159
7	19.9	293.0	295.6	22.4	1.1867	0.0000183	0.0259	2,454	0.7088	2.539E-05	0.6058	0.0161
8	20.0	293.2	295.7	22.5	1.1864	0.0000183	0.0259	2,454	0.7088	2.541E-05	0.6058	0.0160
9	19.9	293.1	295.6	22.5	1.1866	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
10	20.0	293.1	295.6	22.5	1.1864	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
11	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
12	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
13	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
14	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
15	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
16	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
17	20.0	293.1	295.6	22.5	1.1865	0.0000183	0.0259	2,454	0.7088	2.540E-05	0.6058	0.0160
18	**20.0**	**293.1**	**295.6**	**22.5**	**1.1865**	**0.0000183**	**0.0259**	**2,454**	**0.7088**	**2.540E-05**	**0.6058**	**0.0160**

Module 27

Exploring LPG Cylinders for Medical Oxygen – A Preliminary Study

The following module is a study to explore the usage of LPG cylinders for medical oxygen in times of medical emergencies. The study aims at understanding how long medical oxygen can be supplied to cater to patients requiring supply between 0.5 lit/min to 2 lit/min.

General Notes & Assumptions

1. Medical Oxygen composition is taken to contain 90% O_2, 5% N_2 and 5% Ar.
2. The LPG cylinder considered has a 33.3 litre water capacity, storing 14.2 kg of LPG.
3. The analysis is performed as a vessel with an orifice discharging the fluid to the downstream and considering patients requiring oxygen in the range of 0.5 lit/min and 2 lit/min. The orifice discharge coefficient [C_d] is taken as 0.62.
4. Considering a cylinder pressure cap of 16.9 kg/cm^2, the pressure cap for the study is taken as 16.0 bara at 25^{0}C. The pressure at which medical oxygen is delivered is taken as 1.01325 bara.
5. For the analysis, an isothermal blowdown condition is taken assuming the breathing process from the medical oxygen cylinder takes sufficiently long time and the gas temperature also does not change with time. Hence heat is absorbed through the walls such that the cylinder temperature is close to ambient temperature.

Governing Relationships

To estimate the blow down time, a transient study is performed. To check if choked flow exists, the following condition is applied,

$$\frac{P_{cyl}}{P_{atm}} \geq \left[\frac{k+1}{2}\right]^{\frac{k}{k-1}} \qquad \text{(A)}$$

Where,

P_{cyl} = Cylinder Pressure [bara]

P_{atm} = Atmospheric Pressure [bara]

k = ratio of specific heats [C_p/C_v] [-]

The blowdown time can be estimated as,

$$P_{cyl} = P_0 exp\left[-\frac{t}{\tau}\right] \qquad \text{(B)}$$

$$\tau = \left[\frac{V_{cyl}}{C_d \times A \times \left[\frac{2}{k+1}\right]^{\frac{k+1}{2(k-1)}} \times \left[\frac{k \times R \times T_{cyl}}{MW}\right]^{1/2}}\right] \qquad \text{(C)}$$

Where,

τ = Discharge Time Constant [sec]

P_{cyl} = Cylinder Pressure [bara]

P_0 = Cylinder Initial Pressure [bara]

Design Data & Results

The input data and results for 0.5 lit/min is as follows,

Table 1. Input Data and Results for 0.5 lit/min

Parameter	Value	Unit
Effective Cylinder Volume [V]	0.0333	m^3
Medical Oxygen MW	32.2	kg/kmol
Initial Pressure [P_0]	16	bara
Initial Temperature [T_0]	25	^{0}C
Oxygen k [C_p/C_v]	1.395	-
Choked Flow Exists or Not	Choked Flow	
Compressibility Factor [Z]	0.9902	-
Oxygen Density [r]	20.99	kg/m^3
Mass of O2 in Cylinder [m]	0.699	kg
Orifice Throat Diameter [d]	0.30	mm
Orifice Throat CSA [A]	7.0926E-08	m^2
Discharge Coefficient [C_d]	0.62	-
Speed of Sound [C]	327.7	m/s
Discharge Time Constant [t]	3,991	sec
Mass Flow Rate [m_g]	0.0002	kg/s
Volumetric Flow Rate	0.000008	m^3/s
Required Vol. Flow Rate	0.50	lit/min

The input data and results for 2 lit/min is,

Table 2. Input Data and Results for 2 lit/min

Parameter	Value	Unit
Orifice Throat Diameter [d]	0.60	mm
Orifice Throat CSA [A]	2.8334E-07	m^2
Discharge Time Constant [t]	999	sec
Mass Flow Rate [m_g]	0.0007	kg/s
Volumetric Flow Rate	0.000033	m^3/s
Required Vol. Flow Rate	2.00	lit/min

Plotting a graph between cylinder pressure and Time for both cases of 0.5 lit/min and 2 lit/min,

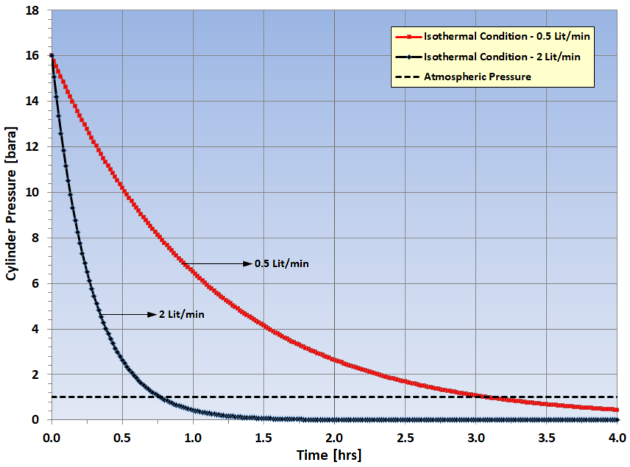

Figure 1. Cylinder Pressure vs. Time

From the above figure, the cylinder pressure beginning from an initial pressure of 16 bara reaches 1 atm in about 3 hours for a discharge rate of 0.5 lit/min and about 45 min for the case of 2 lit/min.

References & Further Reading

1. "https://www.gasmartindia.com/news/what-is-the-composition-of-medical-oxygen/#:~:text=Medical%20oxygen%20comprises%20of%20minimum,removed%20leaving%20behind%20only%20oxygen.
2. Emergency Options for Medical Oxygen Storage & Alternate Mode of Oxygen Generation, Preliminary Assessment Report, Tata Consulting Engineers Limited, April 2021 (https://www.tce.co.in/wp-content/uploads/2021/04/Meeting-Oxygen-Demand-Tata-Consulting-Engineers-Response.pdf)
3. "*Tank Blowdown Math*", Dean Wheeler, Brigham Young University, March 13, 2019 (https://www.et.byu.edu/~wheeler/Tank_Blowdown_Math.pdf)

Appendix A: Derivations of Expressions

$$\frac{dm}{dt} = -m_g \tag{1}$$

'm' = mass of gas in the cylinder, expressed as,

$$m = \rho_{cyl} \times V_{cyl} \tag{2}$$

Where,

ρ_{cyl} = Density of gas in cylinder [kg/m^3]

V_{cyl} = Volume of gas in cylinder [m^3]

Whereas, m_g is mass flow rate at choked flow conditions is expressed as,

$$m_g = C_d \times A \times \rho_c \times v_c \tag{3}$$

Where,

C_d = Orifice Discharge Coefficient [-]

A = Orifice Cross-sectional Area [m^2]

ρ_c = Density at choked flow at throat [kg/m^3]

v_c = Speed of Sound [m/s]

The speed of sound can be estimated as,

$$v_c = \left[\frac{k \times R \times T_c}{MW}\right]^{1/2} \tag{4}$$

Where,

MW = Fluid Molecular Weight [kg/kmol]

T_c = Temperature at choked conditions [K]

For a reversible adiabatic expansion, the fluid density at the orifice throat can be related to the fluid density in the cylinder as,

$$\frac{\rho_c}{\rho_{cyl}} = \left[\frac{2}{k+1}\right]^{\frac{1}{k-1}} \tag{5}$$

$$\frac{T_c}{T_{cyl}} = \left[\frac{\rho_c}{\rho_{cyl}}\right]^{k-1} \tag{6}$$

$$\text{Or, } T_c = T_{cyl} \times \left[\frac{2}{k+1}\right] \tag{7}$$

Therefore the speed of sound at cylinder conditions can be expressed as,

$$v_c = \left[\frac{2\times k\times R\times T_{cyl}}{MW\times(k+1)}\right]^{1/2} \quad (8)$$

Therefore the mass flow rate at choked flow related to cylinder process conditions becomes,

$$m_g = C_d \times A \times \rho_{cyl}\left[\frac{2}{k+1}\right]^{\frac{1}{k-1}} \times \left[\frac{2\times k\times R\times T_{cyl}}{MW\times(k+1)}\right]^{1/2} \quad (9)$$

Or

$$m_g = C_d \times A \times \rho_{cyl}\left[\frac{2}{k+1}\right]^{\frac{k+1}{2(k-1)}} \times \left[\frac{k\times R\times T_{cyl}}{MW}\right]^{1/2} \quad (10)$$

Where,

R = Gas Constant [8.314 m^3.bar/kmol.K]

The cross-sectional area of the orifice is,

$$A = \frac{\pi}{4} \times d_{orifice}^2 \quad (11)$$

Where,

$d_{orifice}$ = Orifice diameter [m]

Therefore solving for blowdown time,

$$V_{cyl} \times \frac{d\rho_{cyl}}{dt} = C_d \times A \times \rho_{cyl}\left[\frac{2}{k+1}\right]^{\frac{k+1}{2(k-1)}} \times \left[\frac{k\times R\times T_{cyl}}{MW}\right]^{1/2} \quad (12)$$

Rearranging the above,

$$-\left[\frac{V_{cyl}}{C_d\times A\times\left[\frac{2}{k+1}\right]^{\frac{k+1}{2(k-1)}}\times\left[\frac{k\times R\times T_{cyl}}{MW}\right]^{1/2}}\right]\frac{d\rho_{cyl}}{dt} = \rho_{cyl} \quad (13)$$

Simplifying the expression by taking a discharge time constant [τ],

$$\tau = \left[\frac{V_{cyl}}{C_d\times A\times\left[\frac{2}{k+1}\right]^{\frac{k+1}{2(k-1)}}\times\left[\frac{k\times R\times T_{cyl}}{MW}\right]^{1/2}}\right] \quad (14)$$

Substituting and solving for the blowdown time,

$$\frac{d\rho_{cyl}}{dt} = -\frac{\rho_{cyl}}{\tau} \quad (15)$$

$$\int_{\rho_0}^{\rho_{cyl}}\frac{d\rho_{cyl}}{\rho_{cyl}} = -\int_{t=0}^{t=t}\frac{dt}{\tau} \quad (16)$$

$$ln\left[\frac{\rho_{cyl}}{\rho_0}\right] = -\frac{t}{\tau} \quad (17)$$

$$\rho_{cyl} = \rho_0 exp\left[-\frac{t}{\tau}\right] \quad (18)$$

Applying ideal gas law to convert densities to pressures,

$$P_{cyl} = P_0 exp\left[-\frac{t}{\tau}\right] \quad (19)$$

Appendix B: MS Excel Calculations

Cylinder Blow Down Time - 0.5 Lit/min		
Domestic 14.2 kg LPG Cylinder		
Effective Cylinder Volume [V]	**0.0333**	m^3
Oxygen [O_2] MW	32.2	kg/kmol
Initial Pressure [P_0]	**16**	bara
Initial Temperature [T_0]	**25**	^0C
	298.15	K
Oxygen k [Cp/Cv]	**1.395**	-
Choked Flow Exists or Not	Choked Flow	
Compressibility Factor [Z]	**0.9902**	-
O_2 Std. Density [1 atm, 20^0C]	1.316	kg/m^3
Oxygen Density [r]	20.99	kg/m^3
Mass of O_2 in Cylinder [m]	0.699	kg
Orifice Throat Diameter [d]	**0.30**	mm
Orifice Throat CSA [A]	7.0926E-08	m^2
Discharge Coefficient [C_d]	**0.62**	-
Speed of Sound [C]	327.7	m/s
Discharge Time Constant [τ]	3991	sec
Mass Flow Rate [m_rate]	0.0002	kg/s
Volumetric Flow Rate	0.000008	m^3/s
Required Vol. Flow Rate	**0.50**	lit/min
Cylinder Blow Down Time - 2 Lit/min		
Orifice Throat Diameter [d]	**0.60**	mm
Orifice Throat CSA [A]	2.8334E-07	m^2
Discharge Time Constant [τ]	999	sec
Mass Flow Rate [m_rate]	0.0007	kg/s
Volumetric Flow Rate	0.000033	m^3/s
Required Vol. Flow Rate	**2.00**	lit/min

Module 28

Heating Value Estimation for Natural Gas Applications

For natural gas custody transfer applications, the gross calorific or gross heating value is necessary for both the buyer and seller to estimate the sales price of natural gas. In case of fuel suppliers, heat content is expressed in terms of Higher Heating value (HHV) to estimate fuel charges in kWh. Whereas Lower Heating Value (LHV) is employed to estimate fuel requirements since the total energy input for a specific power output is already fixed. To understand how fuel heating values are affected, LHV and HHV is explained as,

1. The lower heating value (LHV) or net calorific value (NCV) of a fuel is defined as the amount of heat released by combusting a specified quantity at 25°C and returning the temperature of combustion products to 150°C, with the assumption that latent heat of vaporization of water in the reaction products is not recovered, i.e., when water in the combustion product, is in its vapour form, it is called LHV/NCV. LHV is a better indication of a fuel's useful heat since the combustion products are above the boiling point of water.
2. The higher heating value or gross calorific value (GCV) of a fuel is defined as the amount of heat released by combusting a specified quantity at 25°C and the products have returned to a temperature of 25°C, taking into account the latent heat of vaporization of water in the products. i.e., when water in the combustion product, is in its liquid form, it is called HHV/GCV.

Presence of water is detrimental to a fuel's heating value, since with high combustion temperatures, water turns into steam & eats away a portion of the energy released as latent heat of vaporization [ΔH_V], i.e., HHV includes latent heat of vaporization of water.

Estimating Fuel Calorific Values

The Gross calorific value (GCV) in mass terms can be computed as per ISO 6976:1995 at 1.01325 bara & 15 ^{0}C (referred to as Standard Conditions as per ISO 6976), as follows,

$$GCV_{mass} = \frac{\sum X_i \times GCV_{i[mol]}}{\sum X_i \times M_i} \quad (1)$$

Where,

X_i = Molar fraction of component 'i'

$GCV_{i[mol]}$ = Molar Gross calorific value of component 'i', [kJ/mol]

M_i = Molecular mass of component 'i' [g/mol]

However in this module, the standard temperature is taken as 25^{0}C and the heating values are estimated based on a stoichiometric balance instead of ISO 6976:1995 method. Taking the following natural gas composition,

Table 1. Natural Gas Composition

Component	MW	Mol%
-	kg/kmol	%
Methane [CH_4]	16.043	85.0
Ethane [C_2H_6]	30.070	5.0
Propane [C_3H_8]	44.097	3.0
n-Butane [n-C_4H_{10}]	58.123	1.0
i-Butane [i-C_4H_{10}]	58.123	1.0
n-Pentane [n-C_5H_{12}]	72.15	0.5
i-Pentane [i-C_5H_{12}]	72.15	0.3
Hydrogen [H_2]	2.016	0.1
Carbon monoxide [CO]	28.011	0.1
Carbon dioxide [CO_2]	44.011	0.2
Nitrogen [N_2]	28.0135	3.8
Mixture MW [kg/kmol]		19.385

To estimate the heat of reaction, the combustion of hydrocarbons follow the below stoichiometric balance as,

$$C_nH_{2n+2} + \left[\frac{3n+1}{2}\right]O_2 \rightarrow nCO_2 + [n+1]H_2O$$

To estimate the natural gas calorific value, the calculations require,

1. Natural gas Composition
2. Heat of formation [ΔH^0_f] at Ref. Conditions
3. Heat of Reaction per mole computed as,

$$\Delta H^0_{Rxn,25°C} = \sum X_{i,P}\Delta H^0_{f,P} - \sum X_{i,R}\Delta H^0_{f,R} \quad (1)$$

4. Based on per mole heat of reaction for each combustible species/component, the individual mole fraction of each component is multiplied with the respective heat of reaction and summed up to arrive at the Net Calorific Value (NCV) or Lower Heating Value (LHV). A negative sign in the LHV/NCV value indicates, heat is released due to the combustion process.
5. To estimate the HHV, firstly, the hydrogen content in each component of the natural gas mixture (which forms water) is multiplied by its respective component's mole fraction, summed up, divided by 2 and multiplied with the heat of reaction from the conversion of $H_2O(g) \rightarrow H_2O(l)$, i.e., [$\Delta H_{Rxn,25^0C} = \Delta H^0_{f,H2O(l)} - \Delta H^0_{f,H2O(g)}$]. Followed by, subtracting the above estimate from the modulus (positive) value of LHV/NCV.

The standard heat of formation [ΔH^0_f] at 25^{0}C for the reactants and products are as follows,

Table 2. Standard Heat of Formation [25^{0}C]

Component	$\Delta H^0_{f,25^0C}$
-	kJ/mol
Methane [CH_4]	-74.84
Ethane [C_2H_6]	-84.67
Propane [C_3H_8]	-103.85
n-Butane [n-C_4H_{10}]	-124.73
i-Butane [i-C_4H_{10}]	-134.50
n-Pentane [n-C_5H_{12}]	-146.40
i-Pentane [i-C_5H_{12}]	-154.40
Carbon monoxide [CO]	-110.52
Hydrogen [H_2]	0.00
Oxygen [O_2]	0.00
Nitrogen [N_2]	0.00
Carbon dioxide [CO_2]	-393.51
Water Vapour [H_2O (g)]	-241.83
Water [H_2O (l)]	-285.84

Performing a stoichiometric balance of the combustion reactions,

Table 3. Combustion Reaction Set

CH_4	+	2	O_2	→	1	CO_2	+	2	$H_2O(g)$
C_2H_6	+	3½	O_2	→	2	CO_2	+	3	$H_2O(g)$
C_3H_8	+	5	O_2	→	3	CO_2	+	4	$H_2O(g)$
nC_4H_{10}	+	6½	O_2	→	4	CO_2	+	5	$H_2O(g)$
iC_4H_{10}	+	6½	O_2	→	4	CO_2	+	5	$H_2O(g)$
nC_5H_{12}	+	8	O_2	→	5	CO_2	+	6	$H_2O(g)$
iC_5H_{12}	+	8	O_2	→	5	CO_2	+	6	$H_2O(g)$
H_2	+	½	O_2	→				1	$H_2O(g)$
CO	+	½	O_2	→	1	CO_2			
$H_2O(g)$				→				1	$H_2O(l)$

Quoting a sample case for methane at 25^{0}C,

$$\sum X_{i,R}\Delta H^0_{f,R} = [(1 \times -74.84) + (2 \times 0)] \quad (2)$$

$$\sum X_{i,P}\Delta H^0_{f,P} = [(2 \times -241.83) + (1 \times -393.51)] \quad (3)$$

$$\Delta H^0_{Rxn} = [-877.162] - [-74.84] = -802.32\frac{kJ}{mol} \quad (4)$$

For the case of H_2 at 25^{0}C,

$$\sum X_{i,R}\Delta H^0_{f,R} = [(1 \times 0) + (0.5 \times 0)] \quad (5)$$

$$\sum X_{i,P}\Delta H^0_{f,P} = [1 \times -241.83] \quad (6)$$

$$\Delta H^0_{Rxn} = [-241.83 - 0] = -241.83\frac{kJ}{mol} \quad (7)$$

For the case of $H_2O(g)$ to $H_2O(l)$ at 25^{0}C,

$$\sum X_{i,R}\Delta H^0_{f,R} = -241.83 \quad (8)$$

$$\sum X_{i,P}\Delta H^0_{f,P} = -285.84 \quad (9)$$

$$\Delta H^0_{Rxn} = [-285.84 - (-241.83)] = -44.01\frac{kJ}{mol} \quad (10)$$

Similarly, performing calculations for other components to yields,

Table 4. Heat of Reaction Summary

Component	$\Delta H^0_{Rxn,\,25^0C}$ [kJ/mol]
CH_4	-802.32
C_2H_6	-1,427.83
C_3H_8	-2,043.98
nC_4H_{10}	-2,658.44
iC_4H_{10}	-2,648.67
nC_5H_{12}	-3,272.11
iC_5H_{12}	-3,264.11
H_2	-241.83
CO	-282.99
$H_2O(g)$	-44.014

Therefore the LHV/NCV is computed as,

Table 5. LHV/NCV Estimation

Component	$X_i \Delta H_R^0$	
-	[kJ/mol]	
Methane [CH_4]	0.85 × -802.32	-681.974
Ethane [C_2H_6]	0.050 × -1,427.83	-71.392
Propane [C_3H_8]	0.030 × -2,043.98	-61.320
n-Butane [n-C_4H_{10}]	0.010 × -2,658.44	-26.584
i-Butane [i-C_4H_{10}]	0.010 × -2,648.67	-26.487
n-Pentane [n-C_5H_{12}]	0.005 × -3,272.11	-16.361
i-Pentane [i-C_5H_{12}]	0.003 × -3,264.11	-9.792
Hydrogen [H_2]	0.001 × -241.83	-0.242
Carbon monoxide	0.001 × -282.99	-0.283
Carbon dioxide	0.000	0.000
Nitrogen [N_2]	0.000	0.000
Lower Heating Value ($\boldsymbol{LHV}$) $= \sum X_i \Delta H_R^0$		**-894.43**

The higher heating value (HHV) is computed by initially making a hydrogen balance as,

Component	Hydrogen Balance	
Methane [CH_4]	0.850 × 4	3.4
Ethane [C_2H_6]	0.050 × 6	0.3
Propane [C_3H_8]	0.030 × 8	0.24
n-Butane [n-C_4H_{10}]	0.010 × 10	0.1
i-Butane [i-C_4H_{10}]	0.010 × 10	0.1
n-Pentane [n-C_5H_{12}]	0.005 × 12	0.06
i-Pentane [i-C_5H_{12}]	0.003 × 12	0.036
Hydrogen [H_2]	0.0010 × 2	0.002
Total		**4.238**

The HHV/GCV is now calculated as,

$$HHV = 894.43 - \left[\frac{4.238 \times -44.014}{2}\right] = 987.70\,\frac{kJ}{mol} \quad (11)$$

Expressing in mass terms [kJ/kg], the heating values are as follows,

$$LHV/NCV = \frac{894.43 \times 1000}{19.385} = 46{,}140\ kJ/kg \quad (12)$$

$$HHV/GCV = \frac{987.7 \times 1000}{19.385} = 50{,}951 kJ/kg \quad (13)$$

References & Further Reading

1. "*Principles of Chemical Engineering Processes*", Nayef Ghasem, Redhouane Henda, 2nd Edition, Taylor & Francis Group
2. ISO 6976 (1995) Natural gas - Calculation of calorific values, density, relative density and Wobbe index from composition
3. https://www.clarke-energy.com/heating-value/
4. https://www.industrialheating.com/articles/90561-calculating-the-heat-of-combustion-for-natural-gas

Appendix A

Natural Gas Calorific Value Calculations

Enthalpy of Formation [25^{0}C]	
Component	**ΔH^0_f [kJ/mol]**
Methane [CH_4]	-74.84
Ethane [C_2H_6]	-84.67
Propane [C_3H_8]	-103.85
n-Butane [n-C_4H_{10}]	-124.73
i-Butane [i-C_4H_{10}]	-134.50
n-Pentane [n-C_5H_{12}]	-146.40
i-Pentane [i-C_5H_{12}]	-154.40
Carbon monoxide [CO]	-110.52
Hydrogen [H_2]	0.00
Oxygen [O_2]	0.00
Nitrogen [N_2]	0.00
Carbon dioxide [CO_2]	-393.51
Water Vapour [H_2O (g)]	-241.83
Water [H_2O (l)]	-285.84

Enthaply of Reaxtion ΔH^0_{Rxn}, 25^{0}C

$$\Delta H^0_{Rxn,25^\circ C} = \sum X_{i,P}\Delta H^0_{f,P} - \sum X_{i,R}\Delta H^0_{f,R}$$

Reaction Set	$\Delta H^0_{Rxn,\,25^0C}$ [kJ/mol]
1 CH_4 + 2 O_2 → 1 CO_2 + 2 H_2O (g)	**-802.32**
1 C_2H_6 + 3.5 O_2 → 2 CO_2 + 3 H_2O (g)	**-1,427.83**
1 C_3H_8 + 5 O_2 → 3 CO_2 + 4 H_2O (g)	**-2,043.98**
1 n-C_4H_{10} + 6.5 O_2 → 4 CO_2 + 5 H_2O (g)	**-2,658.44**
1 i-C_4H_{10} + 6.5 O_2 → 4 CO_2 + 5 H_2O (g)	**-2,648.67**
1 n-C_5H_{12} + 8 O_2 → 5 CO_2 + 6 H_2O (g)	**-3,272.11**
1 i-C_5H_{12} + 8 O_2 → 5 CO_2 + 6 H_2O (g)	**-3,264.11**
1 H_2 + 0.5 O_2 → 1 H_2O (g)	**-241.83**
1 CO + 0.5 O_2 → 1 CO_2	**-282.99**
1 H_2O (g) → 1 H_2O (l)	**-44.01**

Legend	
Input	Input Data
GCV/HHV	Gross Calorific Value / Higher Heating Value
NCV/LHV	Net Calorific Value / Lower Heating Value

Notes

Enthalpy of Formation is taken at 1.01325 bara/25^{0}C

Components	MW [kg/kmol]	Mole % [%]	Mole Fraction [-]	Xi × ΔH^0_R [kJ/mol]
Methane [CH_4]	16.043	85.0	0.850	-681.974
Ethane [C_2H_6]	30.07	5.0	0.050	-71.392
Propane [C_3H_8]	44.097	3.0	0.030	-61.320
n-Butane [n-C_4H_{10}]	58.123	1.0	0.010	-26.584
i-Butane [i-C_4H_{10}]	58.123	1.0	0.010	-26.487
n-Pentane [C_5H_{12}]	72.15	0.5	0.005	-16.361
i-Pentane [i-C_5H_{12}]	72.15	0.3	0.003	-9.792
Hydrogen [H_2]	2.016	0.1	0.001	-0.242
Carbon Monoxide [CO]	28.011	0.1	0.001	-0.283
Carbon dioxide [CO_2]	44.011	0.2	0.002	0.000
Nitrogen [N_2]	28.014	3.8	0.038	0.000
		100	1.000	**-894.43**

Results				
NG Mixture MW	**19.385**	kg/kmol		
NCV/LHV	**894.43**	kJ/mol	**46,140**	kJ/kg
GCV/HHV	**987.70**	kJ/mol	**50,951**	kJ/kg

Module 29

Empirical Approach to Hydrate Formation in Natural Gas Pipelines

Natural Gas Pipelines often suffer from production losses due to hydrate plugging. For an effective hydrate plug to form, factors can vary from pipeline operating pressure and temperature, presence of water below its dew point, extreme winter conditions & Joule Thomson cooling. In the event hydrates form in the pipeline section, their consequence depends on how well the hydrates agglomerate to grow and form a column. If the pipeline section temperature is only at par with the hydrate formation temperature, the particles do no agglomerate; instead they have to cross the metastable region which is of the order of 5^0C to 6^0C, before hydrate formation accelerates to block the pipeline.

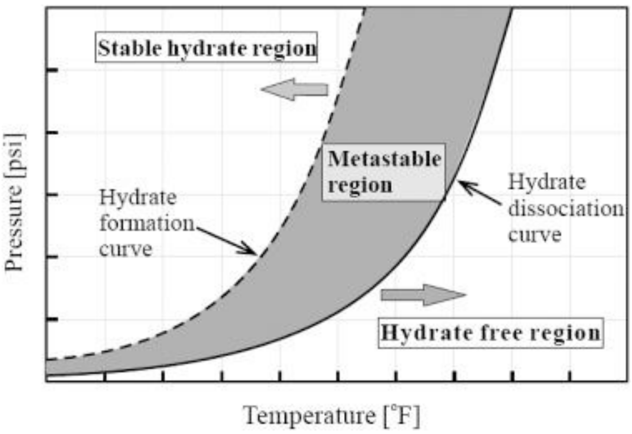

Figure 1. P-T Hydrate Curve [1]

Although engineering softwares exist to estimate pipeline process conditions and also generate a P-T hydrate curve, the following tutorial provides a guidance summary to estimate the expected pipeline temperature profile and the associated hydrate formation temperatures.

Problem Statement

A DN 14", 20 km hydrocarbon line carrying natural gas at the rate of 85,000 kg/h, 40 bara and 25^0C is fed to a receiving station. The total pipeline pressure drop per km [ΔP/km] is taken to be 1 bar/km. The overall heat transfer coefficient is taken to be 25 W/m^2.K. The ambient temperature is 12^0C. The hydrate formation temperature for the composition is experimentally estimated to be 50^0F at 325 psia. It is required to estimate the pipeline exit temperature & the hydrate formation temperature along the pipeline. For the estimates, the Joule-Thomson coefficient is assumed to be an average of 5.6^0C/bar throughout the pipeline. The natural gas composition is as follows,

Table 1. Gas Mixture [GPSA, Sec 20, Page 20-15]

Component	Mol%	MW [M_i]	y_iM_i
	[%]	[kg/kmol]	[-]
Methane	78.40	16.04	12.58
Ethane	6.00	30.07	1.80
Propane	3.60	44.01	1.58
i-Butane	0.50	58.12	0.29
n-Butane	1.90	58.12	1.10
CO_2	0.20	44.01	0.09
N_2	9.40	28.01	2.63
Total	100.00	**MW [kg/kmol]**	**20.08**

Methodology

The pipeline temperature profile can be estimated based on Coulter & Bardon (1979) correlation [4]. The steady state temperature profile is calculated from the momentum equation, while omitting the potential & kinetic energy terms in the enthalpy equation.

$$\frac{dh}{dL} + \frac{dQ}{dL} = 0 \quad (1)$$

Where,

$$Q = \frac{\pi \times OD \times U \times \Delta L}{m}[T_0 - T_s] \quad (2)$$

$$dh = c_p dT - \mu c_p dP \quad (3)$$

Where,

U = Overall HTC [W/m^2.K]

ID = Pipeline OD [m]

m = mass flow rate [kg/s]

ΔL = Pipeline length [m]

T_0 = Fluid Temperature [K]

T_s = Surrounding Temperature [K]

μ = Joule-Thompson Coefficient [^{0}C/bar]

C_p = Specific heat capacity [J/kg.K]

γ_g = Gas Specific Gravity, MW/28.9625 [-]

Solving for pipeline temperature profile,

$$T[L] = \left[T_0 - T_s - \left(\frac{\mu}{a}\right)\left(\frac{dP}{dL}\right)\right] e^{-aL} + T_s + \left(\frac{\mu}{a}\right)\left(\frac{dP}{dL}\right) \quad (4)$$

Where,

$$a = \frac{\pi \times OD \times U}{m \times C_p}$$

It is to be noted that the specific heat [C_p] and Joule-Thompson [J-T] co-efficient [μ] varies with the pipeline pressure & temperature. But for computational purposes, is assumed to be constant. The purpose of including the J-T coefficient is to account for cooling during gas expansion along the pipeline. The ideal mass specific heat [C_p], kJ/kg.K, of natural gas can be computed as,

$$C_p = [(-10.9602\gamma_g + 25.9033) + (0.21517\gamma_g - 0.068687)T + (-0.00013337\gamma_g) + 0.000086387)T^2 + (0.000000031474\gamma_g) - 0.000000028396)T^3]/\,MW \quad (5)$$

Where, T = Temperature [K]

Hydrate Formation Temperature

To estimate the hydrate formation temperature [T_h], Towler & Mokhatab (2005) [3], proposed the following correlation,

$$T_h[°\mathrm{F}] = [13.47 \times ln(P)] + [34.27 \times ln(\gamma)] - [1.675 \times ln(P) \times ln(\gamma)] - 20.35 \quad (6)$$

Where,

P = Pressure [psia]

The validity of the above expression is for the

1. Temperature Range: 260 K to 298 K
2. Pressure Range: 1200 kPa to 40,000 kPa
3. MW: 16 g/mol to 29 g/mol (0.55 < γ_g < 1.0)

Results

Substituting the values to arrive at the pipeline temperature profile, the gas specific gravity is estimated as,

$$\gamma_g = \frac{20.08}{28.9625} = 0.6933 \quad (7)$$

$$a = \frac{\pi \times \left[\frac{14 \times 25.4}{1000}\right] \times 25}{\left[\frac{85,000}{3600}\right] \times 2.071 \times 1000} = 0.0005711 \quad (8)$$

$$T[L] = 12.0195 \times e^{-0.0005711 \times L} + 286.1305 \quad (9)$$

The hydrate formation temperature [T_h] is,

$$T_h[°\mathrm{F}] = [14.0835 \times ln(P, psi)] - 32.9023 \quad (10)$$

Plotting the above expressions, we get,

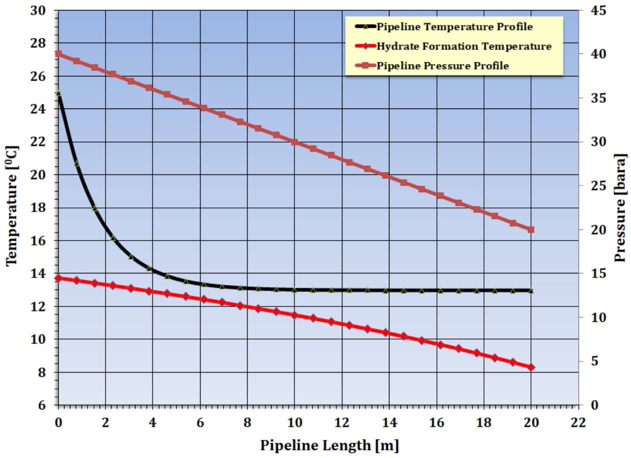

Figure 2. Hydrate Formation Temperature

From the plot, the pipeline temperature stays above the hydrate formation temperature. In practice, to increase the difference, the inlet gas can be either heated or hydrate inhibitors such as MeOH, MEG or TEG can be added.

References & Further Reading

1. https://www.sciencedirect.com/topics/engineering/hydrate-formation-curve
2. "*Handbook of Natural gas Transmission and Processing*", Saied Mokhatab, William A. Poe, John Y. Mak, 3rd Edition.
3. "*Hydrate Formation* Calculation in the Natural Gas Purification Unit", J A Prajaka et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 543 012084
4. Predicting Compositional Two Phase Flow Behaviour in Pipelines, H. Furukawa, O. Shoham, J.P. Brill, Transaction of the ASME, Vol 108, September 1986.

Annexure: MS-Excel Spreadsheet

Natural Gas Inlet Composition

Component	Mol% [%]	MW [M_i] [kg/kmol]	y_iM_i [-]
Methane	78.40	16.04	12.58
Ethane	6.00	30.07	1.80
Propane	3.60	44.01	1.58
i-Butane	0.50	58.12	0.29
n-Butane	1.90	58.12	1.10
i-Pentane	0.00	72.15	0.00
n-Pentane	0.00	72.15	0.00
C_6+	0.00	86.18	0.00
H_2O	0.00	18.02	0.00
CO_2	0.20	44.01	0.09
H_2S	0.00	34.08	0.00
N_2	9.40	28.01	2.63
Total	100.00	MW [kg/kmol]	20.08

Critical Properties - Sutton Correlation with Wichert & Aziz Correction

Parameter	Value	Unit
Inlet Pressure [P]	40.0	bara
Inlet Temperature [T]	25.0	^{0}C
Gas Specific Gravity [γ_g]	0.6933	-
Pseudocritical Pressure [P_{pc}]	664.2	psia
Pseudocritical Temperature [T_{pc}]	375.9	^{0}R
Deviation Factor [ε]	0.4410	^{0}R
Modified Pseudocrotical Pressure [P'_{pc}]	663.4	psia
Modified Pseudocrotical Temperature [T'_{pc}]	375.5	^{0}R
Modified Reduced Pressure [P_{pr}]	0.8863	-
Modified Reduced Temperature [T_{pr}]	1.4292	-
Modified Reduced Density [ρ_r]	0.1865	-
DAK EOS Convergence	*0.0000*	Calculate
Compressibility Factor [Z]	**0.8978**	**-**
Gas Density [ρ]	**36.09**	**kg/m^3**
Gas Viscosity [μ]	**0.0117**	**cP**

Pipeline Temperature Profile

Parameter	Value	Unit
Pipeline Grid Size	769.2	m
a = [$\pi\times$OD$\times$U]/[m$\times$Cp]	0.0005711	[-]
((μ/a)*(dP/dL))	0.9805	[-]

Hydrate Formation Temperature [At Outlet]

Towler & Mokhatab (2005)	8.31	^{0}C

Parameter	Value	Unit
Joule Thompson Coefficient [μ]	0.56	^{0}C/bar
Total Mass Flow Rate [m_T]	85,000	kg/h
Inlet Specific Heat [C_p]	2.071	kJ/kg.K
Pipeline Diameter [DN]	14.00	in
Pipeline Length incl. Fittings [L_e]	20,000	m
$T_{soil}/T_{water}/T_{ambient}$ [Above Ground/Buried]	12.0	^{0}C
Overall Heat Transfer Coefficient [U]	25	W/m^2.K
Pressure drop per Unit Length [dP/dL]	1.00	bar/km
Pipeline Exit Pressure	20.00	bara
Pipeline Exit Temperature	12.98	^{0}C

Pipeline Temperature Profile & Hydrate Formation Temperature

Length [m]	Length [km]	T[L] [^{0}K]	T[L] [^{0}C]	Pressure [bara]	Towler & Mokhatab [^{0}C]
0	0.00	298.2	25.00	40.00	13.73
769	0.77	293.9	20.73	39.23	13.58
1,538	1.54	291.1	17.97	38.46	13.42
2,308	2.31	289.3	16.20	37.69	13.26
3,077	3.08	288.2	15.05	36.92	13.10
3,846	3.85	287.5	14.32	36.15	12.94
4,615	4.62	287.0	13.84	35.38	12.77
5,385	5.38	286.7	13.54	34.62	12.60
6,154	6.15	286.5	13.34	33.85	12.42
6,923	6.92	286.4	13.21	33.08	12.24
7,692	7.69	286.3	13.13	32.31	12.06
8,462	8.46	286.2	13.08	31.54	11.87
9,231	9.23	286.2	13.04	30.77	11.68
10,000	10.00	286.2	13.02	30.00	11.48
10,769	10.77	286.2	13.01	29.23	11.27
11,538	11.54	286.1	13.00	28.46	11.07
12,308	12.31	286.1	12.99	27.69	10.85
13,077	13.08	286.1	12.99	26.92	10.63
13,846	13.85	286.1	12.98	26.15	10.40
14,615	14.62	286.1	12.98	25.38	10.17
15,385	15.38	286.1	12.98	24.62	9.93
16,154	16.15	286.1	12.98	23.85	9.68
16,923	16.92	286.1	12.98	23.08	9.42
17,692	17.69	286.1	12.98	22.31	9.16
18,462	18.46	286.1	12.98	21.54	8.89
19,231	19.23	286.1	12.98	20.77	8.60
20,000	20.00	286.1	12.98	20.00	8.31

MODULE 30

METHODOLOGY FOR SLUG CATCHER SIZING

Oil & Gas Pipelines are often subjected to an operation called 'Pigging' for maintenance purposes (For e.g., cleaning the pipeline of accumulated liquids or waxes). A pig is launched from a pig launcher that scrapes out the remnant contents of the pipeline into a vessel known as a 'Slug catcher'. The term slug catcher is used since pigging operations produces a Slug flow regime characterized by the alternating columns of liquids & gases. Slug catcher's are popularly of two types – Horizontal Vessel Type & Finger Type Slug catcher. However irrespective of the type used, the determination of the slug catcher volume becomes the primary step before choosing the slug catcher type. In order to estimate the pigging volume, engineers use various Flow Assurance (FA) tools to estimate & plot a graph between '*Pipeline Volumetric Flow vs Time'*. Here pipeline volumetric flow rate refers to the point at which liquid exits prior to entering into the slug catcher.

METHODLOGY

The methodology that could be adopted to estimate the excess space required in the slug catcher volume is based on measuring the *volumetric flow rate* of the fluid that exits the pipeline from the time a 'pig' is launched into the pipeline and measuring the same parameter until the pig exits the pipeline. The point at which the pipeline is routed to the slug catcher is where using an FA tool; a transient study is made to plot a graph between *Volumetric flow rate vs. Time*. Since the slug catcher would have a provision to drain the accumulated liquids, it is taken as Drain Rate (Q_D) which also decides the effective volume of the slug catcher.

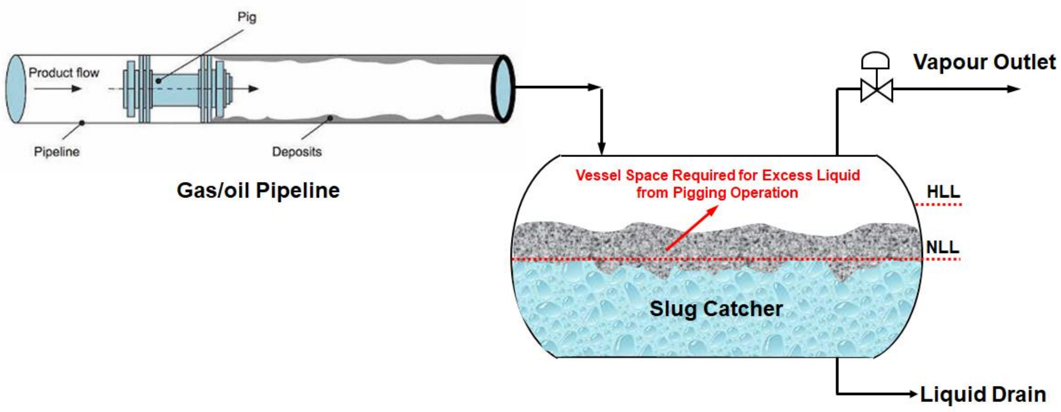

Figure 1. Pig Traversing in a Pipeline

Using the *Volumetric Flow Rate vs Time* graph, the vessel space required to accommodate the excess liquid from the pigging operation for a drain rate (Q_D) can be computed as,

$$V_{Slug\ Catcher} = \sum_{i=1}^{i=n} [\{Q_i \times Margin\} - Q_D] \times (t_{i+1} - t_i)$$

Where,

$$\{If\ [(Q_i \times Margin) - Q_D] \leq 0 \rightarrow [(Q_i \times Margin) - Q_D] = 0$$

The purpose of introducing a factor of 'margin' in estimating the Slug catcher volume is due to uncertainties that Flow Assurance Tools can produce as well as the users' convergence criteria in arriving at a well converged *volumetric flow rate vs time* graph. This can be anywhere from ±2% to ±30% and is incumbent upon the Flow Assurance Engineer to thoroughly understanding the principles of FA & the respective FA solver prior to carrying out slug catcher volume calculations.

Following on the equations provided, a tabulation of the volumetric flow rate vs time can be made in MS-Excel as,

Time [sec]	**Volumetric Flow Rate [m³/s]**	**Pigging Volume [m³]**
t_i	Q_i	$[\{Q_i \times Margin\} - Q_D] \times (t_{i+1} - t_i)$
t_{i+1}	Q_{i+1}	$[\{Q_{i+1} \times Margin\} - Q_D] \times (t_{i+2} - t_{i+1})$
t_{i+2}	Q_{i+2}	$[\{Q_{i+2} \times Margin\} - Q_D] \times (t_{i+3} - t_{i+2})$
t_{i+3}	Q_{i+3}	$[\{Q_{i+3} \times Margin\} - Q_D] \times (t_{i+4} - t_{i+3})$
..	..	....
Cumulative Slug/Pigging Volume		$V_{Slug\ Catcher} = \sum_{i=1}^{i=n} [\{Q_i \times Margin\} - Q_D] \times (t_{i+1} - t_i)$

Below is an example graph of pigging volume estimated using the above method for a given drain rate (Q_D). The cumulative slug/pig volume computed is the area represented under the total liquid volume flow into the slug catcher curve ('blue' line). The 'red' line represents the total liquid volumetric flow that enters the slug catcher while the vessel is constantly drained.

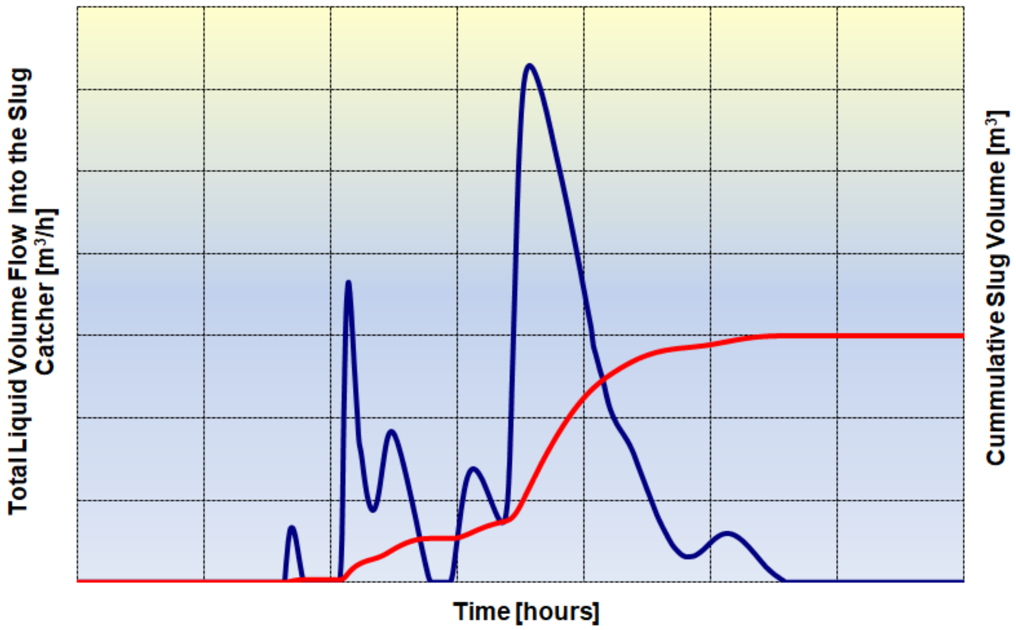

Figure 2. Example of Slug Catcher Volume Required (Illustration Purposes Only)

CHOOSING A SLUG CATCHER SIZE

The drain rate of the slug catcher is determined by the downstream liquid processing capacity in an Oil & gas facility. Therefore in Brownfield projects, the drain rate cannot be inadvertently increased beyond its design limit in the event when the pigged volume exceeds the existing slug catcher's capacity (For e.g., due to change of production). In such instances, it is prudent to install an additional slug catcher to cater to the additional pigging volume. In Greenfield projects, the Basis of Design (BOD) becomes the primary document that determines the drain rate of the slug catcher. Therefore the excess volume required in the slug catcher above the normal operating liquid level (NLL) can be determined by plotting a *Slug Catcher Volume Vs. Drain Rate* Graph. Below is a representative plot for illustration purposes.

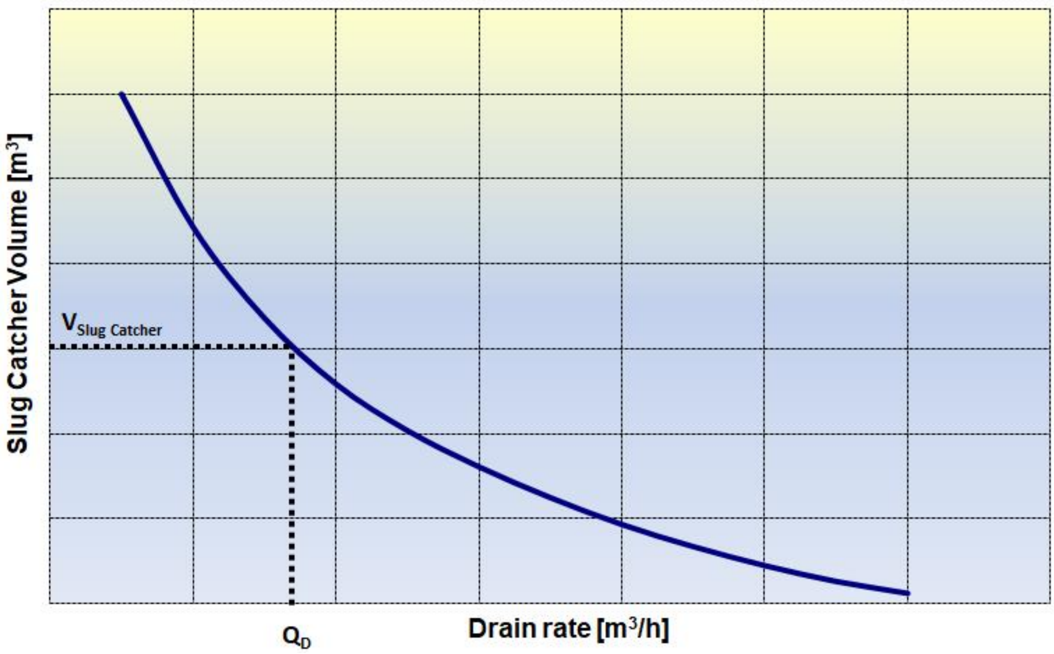

Figure 3. Example of Slug Catcher Volume Selected Based on Drain Rate (Illustration Purposes Only)

SUMMARY

The intent of the slug catcher operation is to accommodate the excess liquids generated by the pigging operation but not to alter the drain rate that can affect the downstream liquid handling equipment. This is accomplished by providing space for the excess pigging volume capacity over and above the Normal Liquid Level (NLL) but below High Liquid Level (HLL) at which liquid levels are maintained in the slug catcher.

About the Author

Vijay Sarathy is a Process Engineer with professional expertise spanning over 14 years of in the Upstream Oil & Gas industry. He started his career with GE performing CFD analysis for Gas Turbine Ventilation systems for GE-Nuovo Pignone's PGT series for ATEX certification and subsequently in areas of Front End Engineering Design, Process Dynamic Simulation including Turbomachinery, Onshore Pipeline Flow Assurance, Optimization & Cost benefit studies for Gas-Oil Separation Units.

Vijay holds a Master's Degree in Chemical Engineering from Birla Institute of Technology & Science (BITS), Pilani, India and is a Chartered Chemical Engineer from the Institution of Chemical Engineers (IChemE), UK. He is also an Industry Advisory Board Member for the International Association of Certified Practicing Engineers (IACPE, Texas, USA) as well as a Contributing Author to the *Engineering Practice* Magazine. Vijay has worked as an Upstream Process Engineer with major conglomerates of General Electric, ENI Saipem and Shell.

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