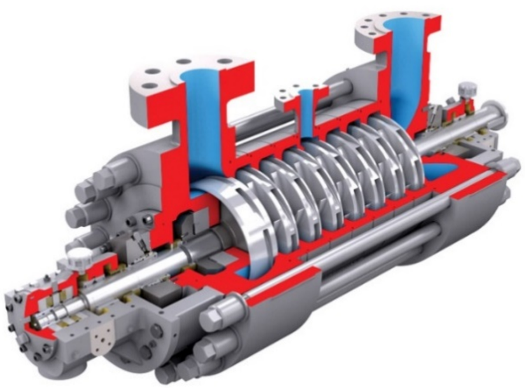
# UNUSUAL CHALLENGES WITH MINIMUM FLOW IN CENTRIFUGAL PUMPS


- **Michael Schleppe – Fluor**
- **Brian Yarnot – Goulds Pumps (ITT)**

# Presenters

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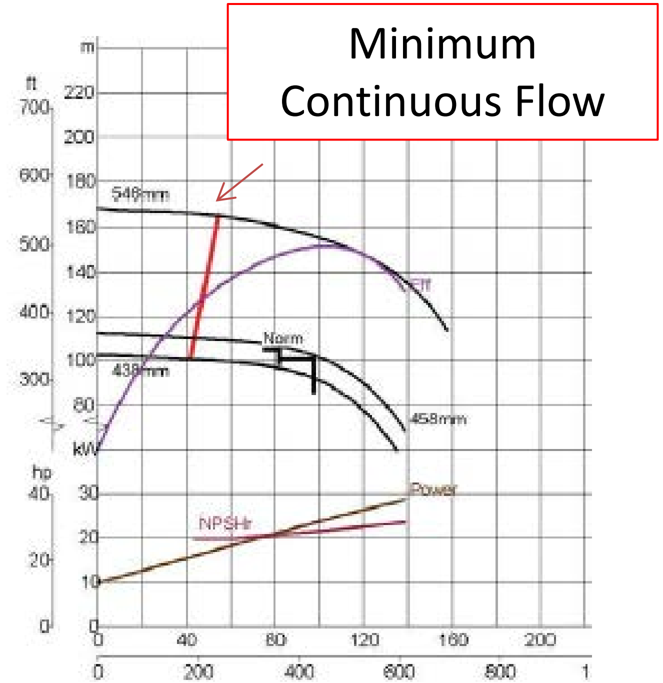
| **Michael Schleppe**<br>**- Fluor Canada Ltd -**<br> | **Rotating Equipment Engineer with Fluor. Michael has eight years of experience in specification, selection, detail design, and installation of rotating equipment. In addition, he has attained a Master of Science in Mechanical Engineering from the University of Alberta, specializing in thermodynamic modeling of combustion processes.** |
| **Brian Yarnot**<br>**- Goulds Pumps (ITT) -**<br> | **Chief Engineer for Between-Bearings API pumps for ITT Goulds Pumps. He is a licensed Professional Engineer in the state of New York, USA. Yarnot has 20 years of experience in pump design, manufacturing, testing, and troubleshooting. In addition, he possesses a Bachelor of Science in Mechanical Engineering from the Pennsylvania State University. Brian is also a taskforce member for API 610 12th Edition.** |

# Agenda

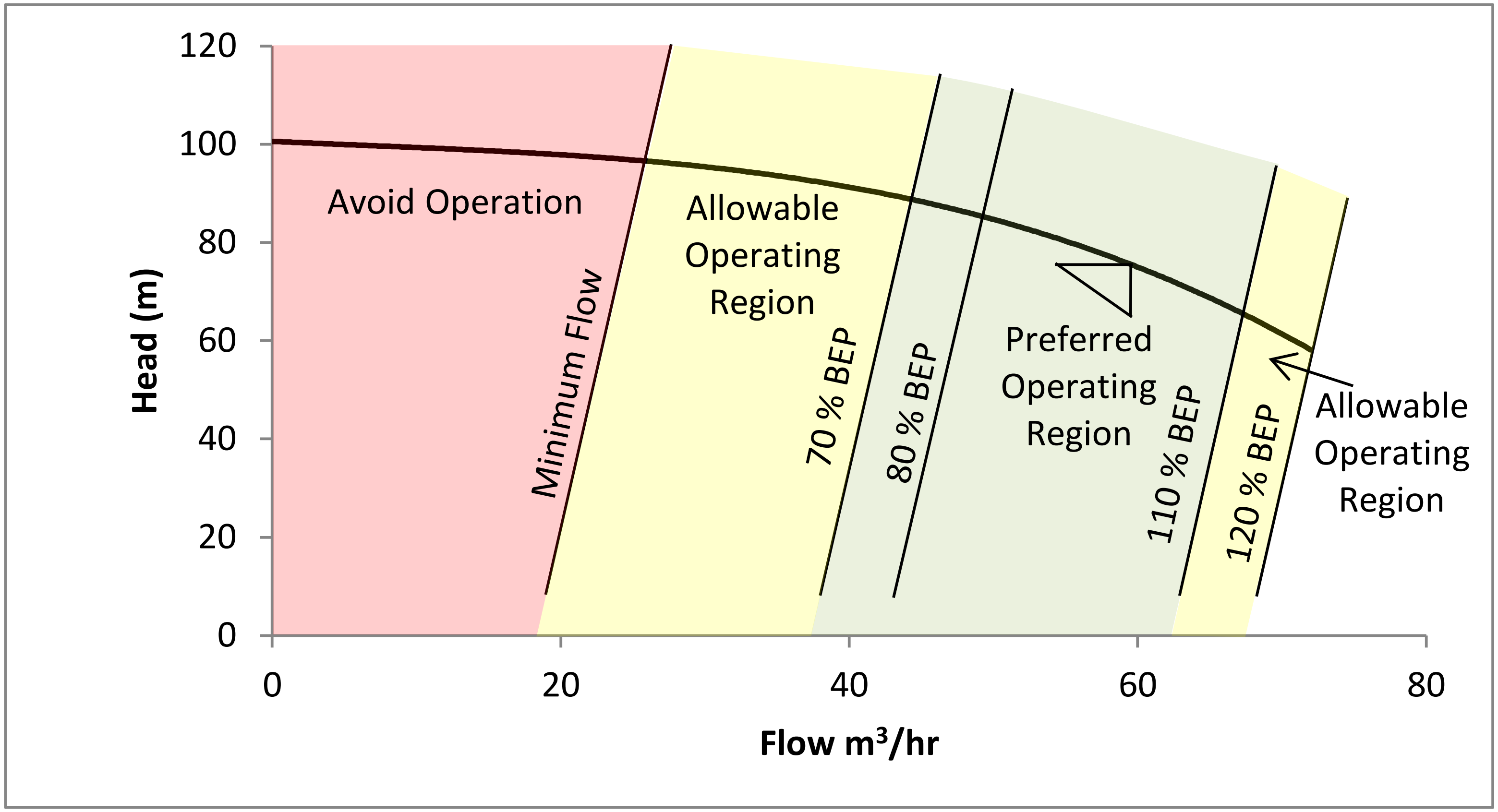
1. Introductions
2. What is Minimum Continuous Flow?
3. When and Why is Minimum Continuous Flow an Issue?
4. Case Study
5. Pump Design and Selection Considerations
6. Pump Piping Layout Considerations
7. Questions

# What is Minimum Continuous Flow?

- Definitions from API 610, 11$^{th}$ Ed:
  - Minimum Continuous Thermal Flow – “Lowest flow at which the pump can operate without its operation being impaired by the temperature rise of the pumped liquid”
  - Minimum Continuous Stable Flow – “Lowest flow at which the pump can operate without exceeding the vibration limits imposed by this International Standard (API 610)”



# What is Minimum Continuous Flow?




Pump Operating Regions

# What is Minimum Continuous Flow?

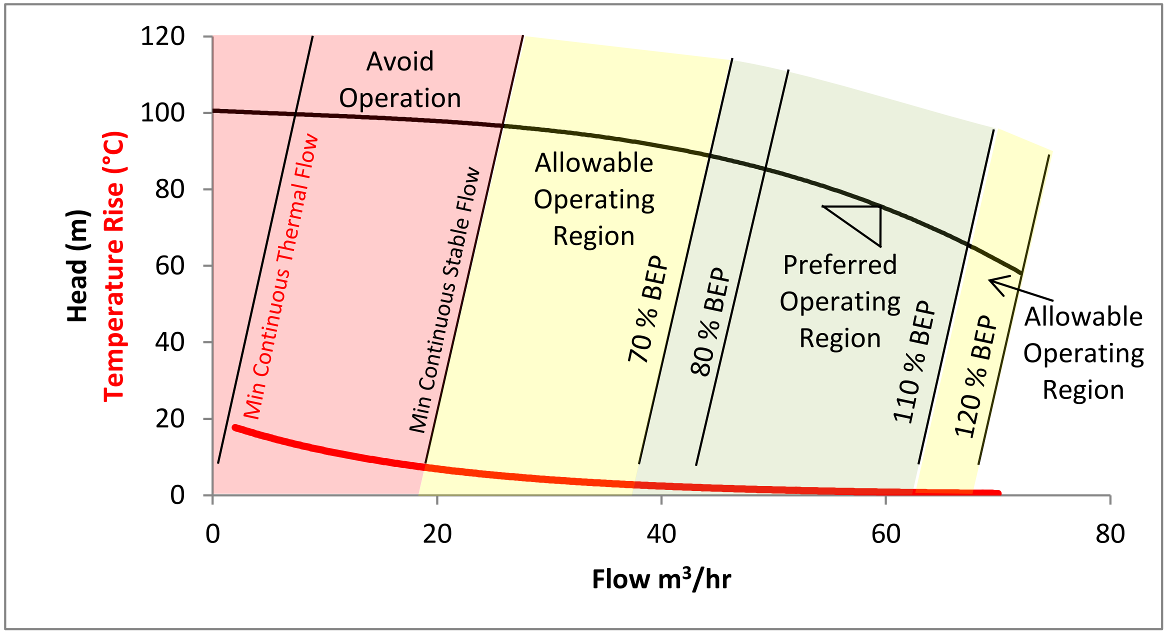
- Depending on the system, centrifugal pumps may operate at a range of flows above and below the rated flow
- Mechanical damage due to vibration and other issues can arise when that operating point approaches or passes below the minimum continuous flow line. For this reason low flow operation is often limited to prevent these damaging flow rates from occurring
- In this presentation we show cases where the fluid properties and operating conditions lead to unexpectedly high minimum continuous flow values due to excessive temperature rise

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Stable Flow (MCSF)

- Minimum continuous stable flow (MCSF) limits the operating envelope of all centrifugal pumps. MCSF is approximately:
  - 30% of Best Efficiency Point (BEP) flow for typical single suction pumps
  - Up to as much as 80% of BEP for large double suction, vertical turbine, and axial flow pumps
- Situations where MCSF is an issue:
  - Systems without a minimum flow bypass valve
  - Normal operation to the left of BEP
  - Systems operating with unexpected turndown (off design cases)

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Thermal Flow (MCTF)

- Minimum continuous thermal flow (MCTF) is only an issue with certain fluid properties or operating conditions
- Most literature suggests that MCTF occurs at flows that are much less than MCSF and is therefore not a concern
- In most cases MCTF is calculated to be much lower than MCSF



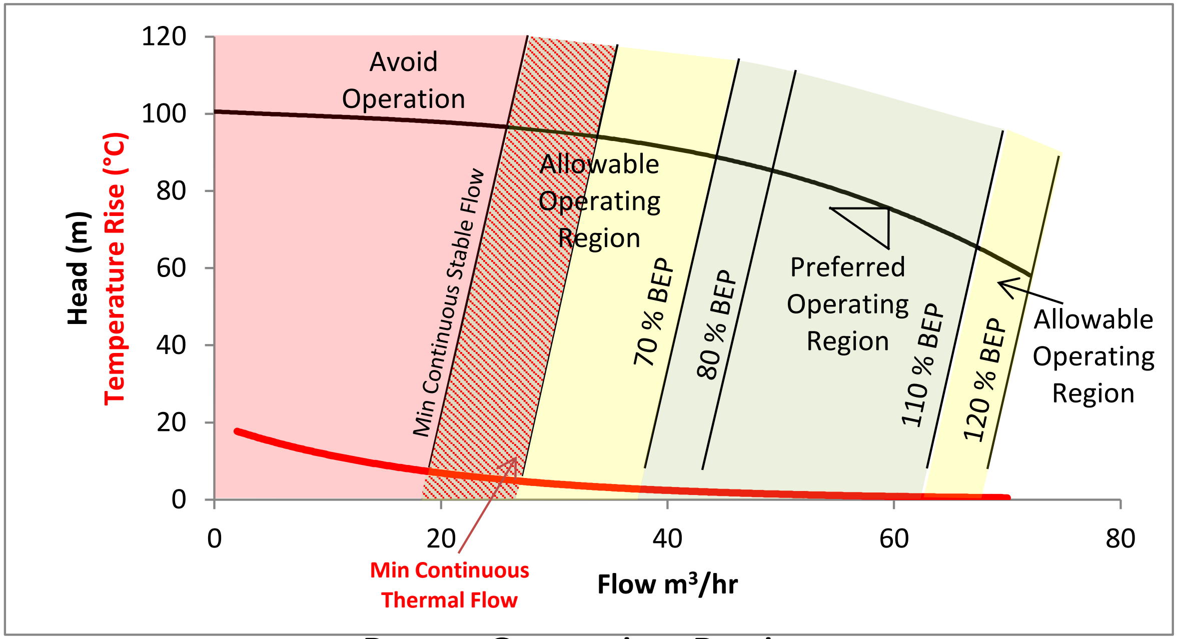

Pump Operating Regions

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Thermal Flow (MCTF)

- Situations where MCTF may be greater than MCSF and therefore low flow operation is limited:
  - Suction pressure within 15 psi (100 kPag) of vapor pressure
  - Fluids that are volatile at room temperature
  - Systems where the fluid in the suction vessel is boiling (condensate, deaerator recirculation, boiler feed water)
  - Fluids with low specific gravity (< 0.7 SG)
  - Cryogenic fluids (sensitive to heat input)
  - Pumps with low efficiency (less efficiency means more heat input to the fluid)

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Thermal Flow (MCTF)

- Mechanisms for damage at MCTF
  - Rapid temperature rise above the boiling point or auto-ignition temperature of the fluid

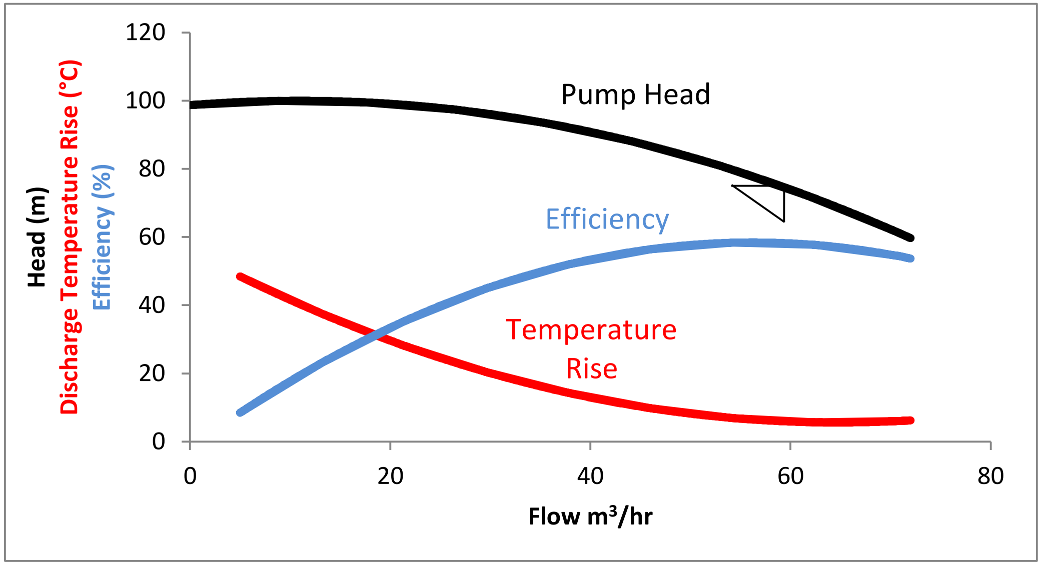



Pump Operating Regions

  - Flashing of fluid as it recirculates over the first stage impeller wear rings
  - Flashing of fluid in balance line for multi-stage pumps

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Thermal Flow (MCTF)

- Rapid temperature rise above boiling point or auto-ignition temperature



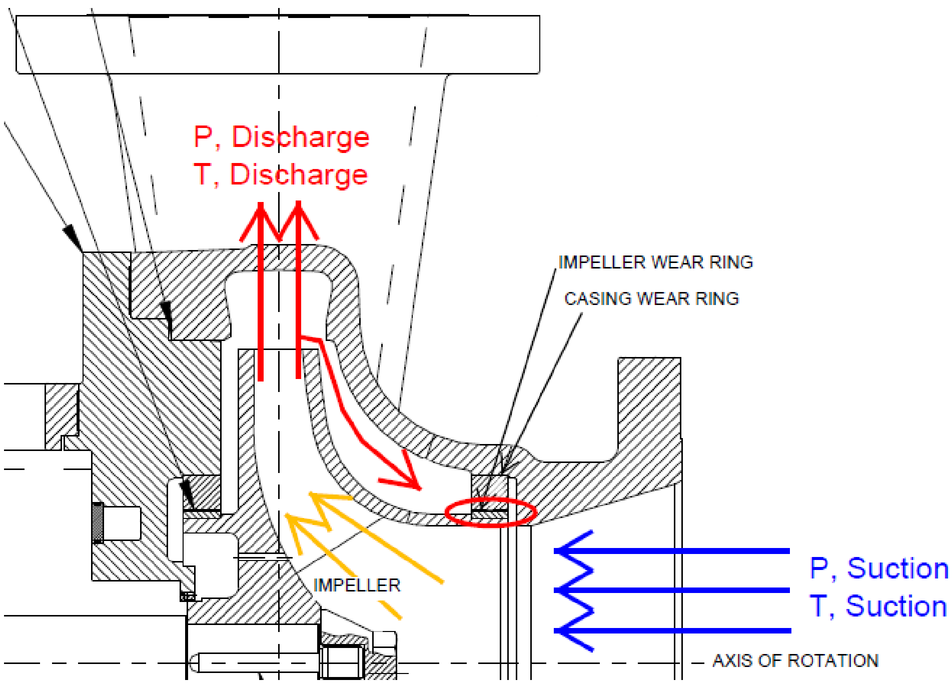

Pump Temperature Rise Curve

At lower flow rates the following conditions contribute to temperature rise:

- Decreasing efficiency
- Increased head
- Increased recirculation
- Less flow to carry away the heat

# When and Why is Minimum Continuous Flow an Issue? Minimum Continuous Thermal Flow (MCTF)

- Flashing of fluid as it recirculates over the first stage impeller wear rings

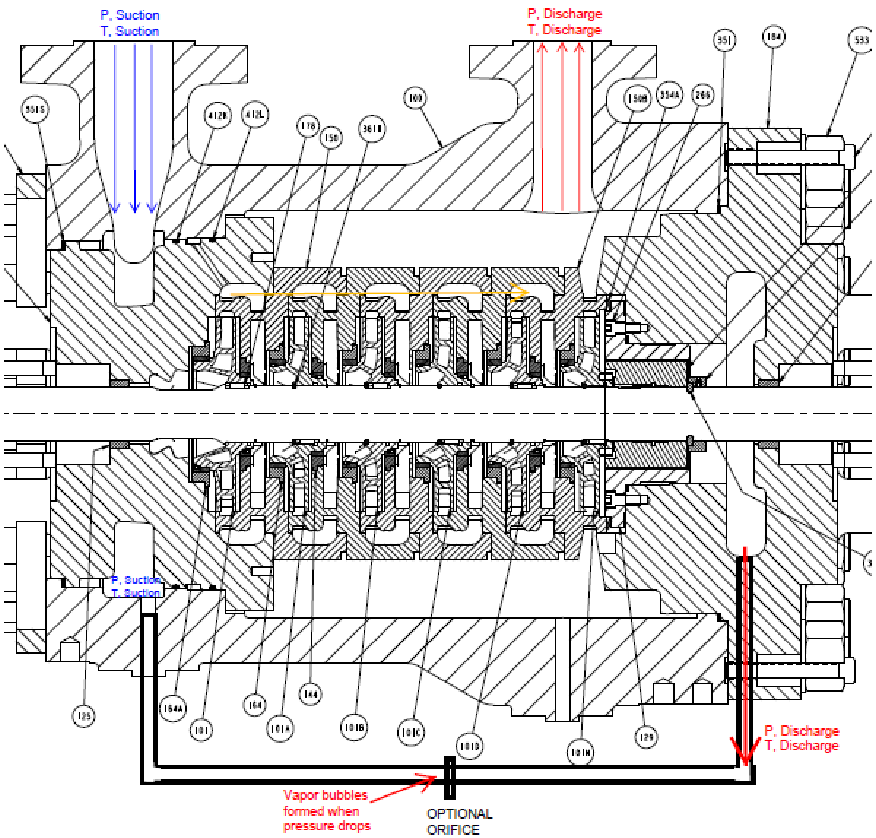



OH2 Pump Cross Section

- Fluid is heated over first stage due to friction and flow separation
- Heating increases the temperature and vapor pressure of the fluid at the first stage discharge
- The hot fluid recirculates over the impeller and flashes when it drops back to suction pressure

# When and Why is Minimum Continuous Flow an Issue?
# Minimum Continuous Thermal Flow (MCTF)

- Flashing of fluid in balance line for multi-stage pumps



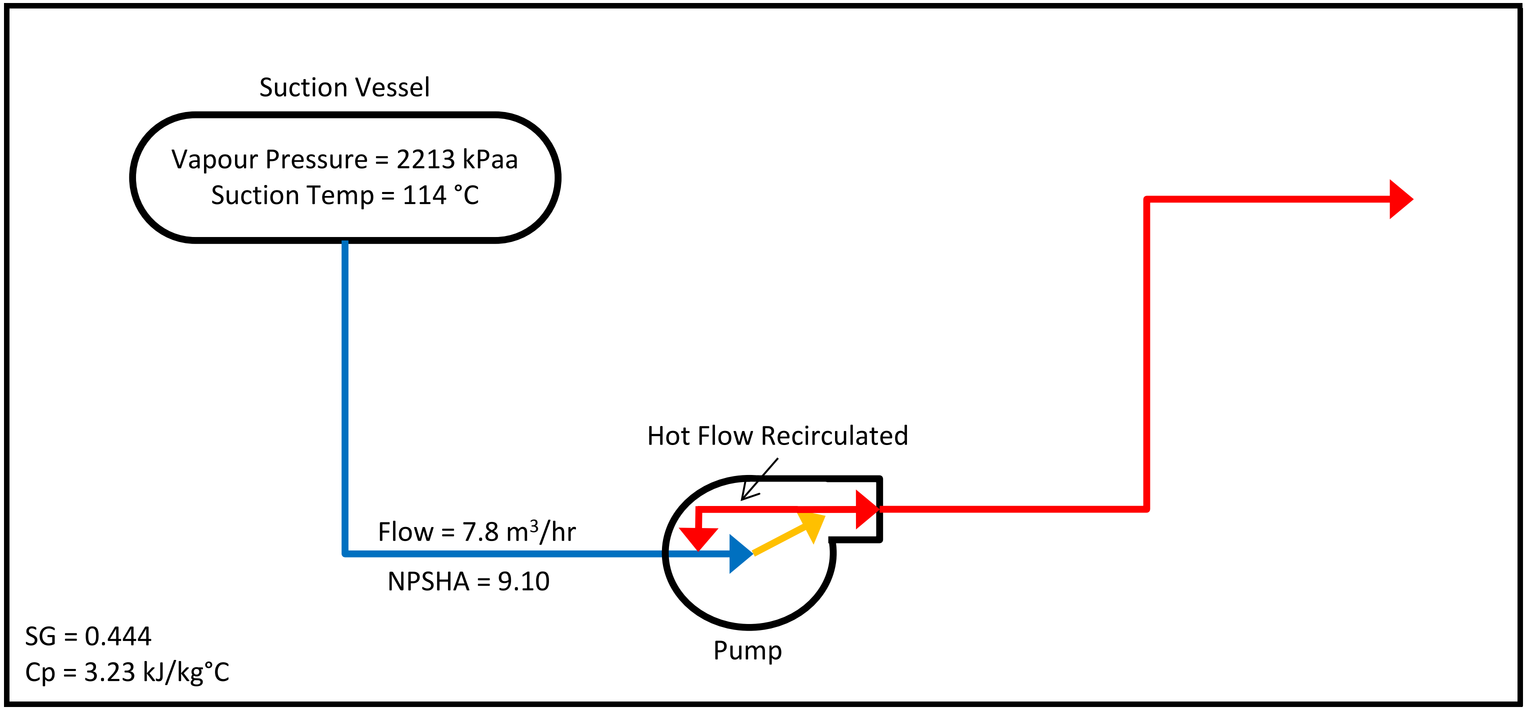

BB5 Pump Cross Section

- Fluid is heated over each stage due to friction and flow separation
- Heating increases the temperature and vapor pressure of the fluid at the balance drum
- The hot fluid is recirculated through the balance line and flashes when it drops back to suction pressure

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump

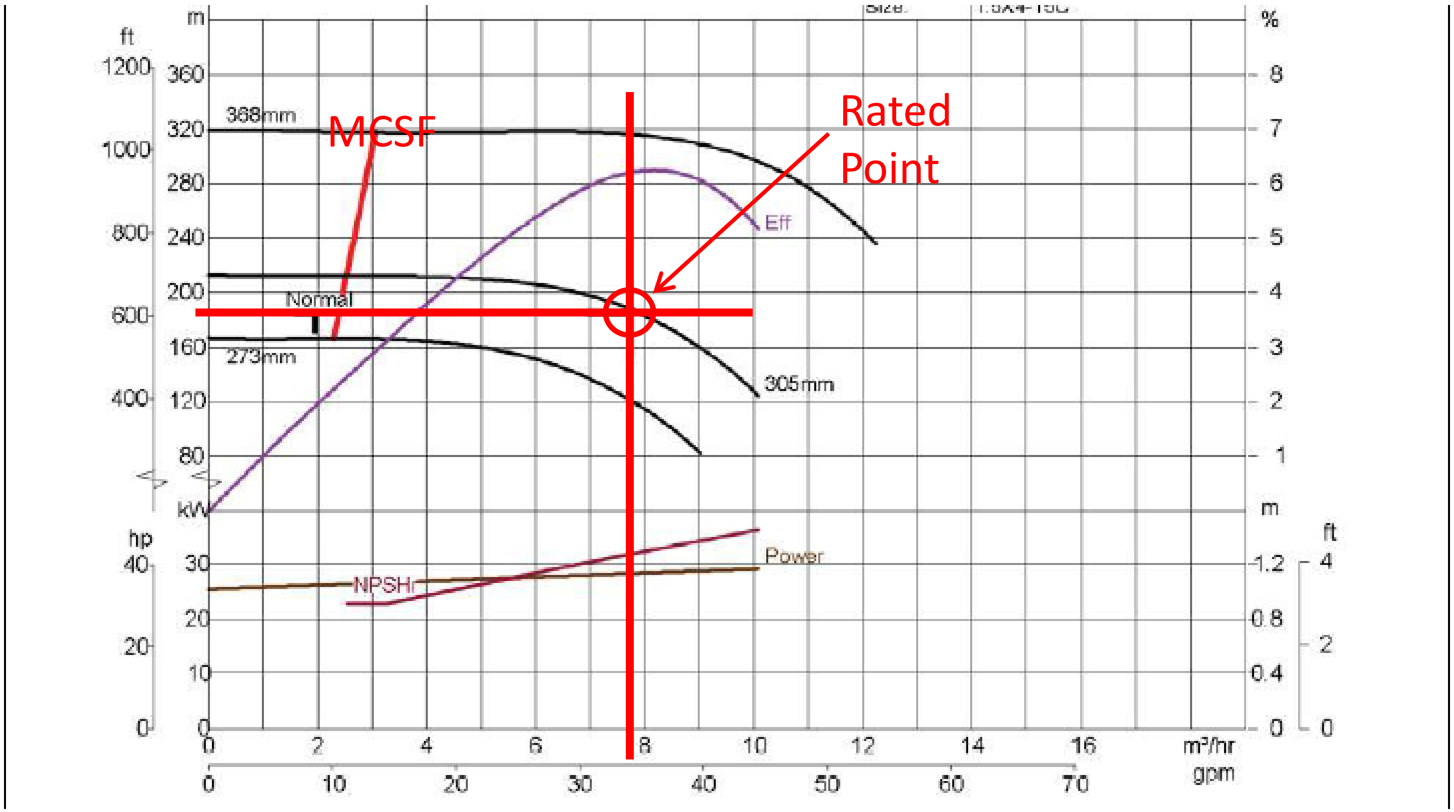
- Service: Depropanizer Net Bottoms Pump
- Process Conditions:
  - Fluid = Light Hydrocarbon
  - SG = 0.444
  - Pvap = 2213 kPaa
  - Suction Temperature = 114 °C
  - Flow = 7.8 $m^3/hr$
  - Cp = 3.23 kJ/kg°C
  - Head = 187 m
  - NPSHa = 9.10 m
  - Preliminary Selection – OH2, 1.5x4-15C, 3600 rpm

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump




System Sketch – OH2 Pump

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump




NPSHa – NPSHr = 7.81 m

Rated Flow @ 95% BEP

Efficiency at Rated = 6.0 % (Low Flow)

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump

- MCTF Considerations – Temperature Rise 8 °C
  - Temperature Rise at MCSF (per HI 1.3 – 2013) should be < 8 °C

(Metric Units) $\Delta t = \frac{H}{102 \times C_P}\left(\frac{1}{\eta} - 1\right)$

(US Customary Units) $\Delta t = \frac{H}{778 \times C_P}\left(\frac{1}{\eta} - 1\right)$

Where:

$\Delta t$ = temperature rise through the pump, in °C (°F)

$H$ = total developed heat at flow being considered, in m (ft)

778 = Constant

102 = Constant

$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)

η = efficiency of the pump at flow being considered, expressed as a decimal

* See back-up slides for derivation

  - Temperature Rise = 23 °C @ MCSF
  - Excessive temperature rise in a pump can result in vaporization of the fluid. This can cause catastrophic failure of the pump.

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump

- MCTF Considerations – Temperature Rise 8 °C(Con't)
  - Min flow for temperature rise of 8 °C (HI 1.3 – 2013)

(Metric Units) $$Q = \frac{433 P_P}{C_P \times \rho}$$

(US Customary Units) $$Q = \frac{P_P}{2.95 \times C_P \times s}$$

Where:

$Q$ = minimum flow rate, in $m^3/hr$ (gpm)

$P_P$ = input power at the minimum flow, in kW (hp)

2.95 = Constant

433 = Constant

$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)

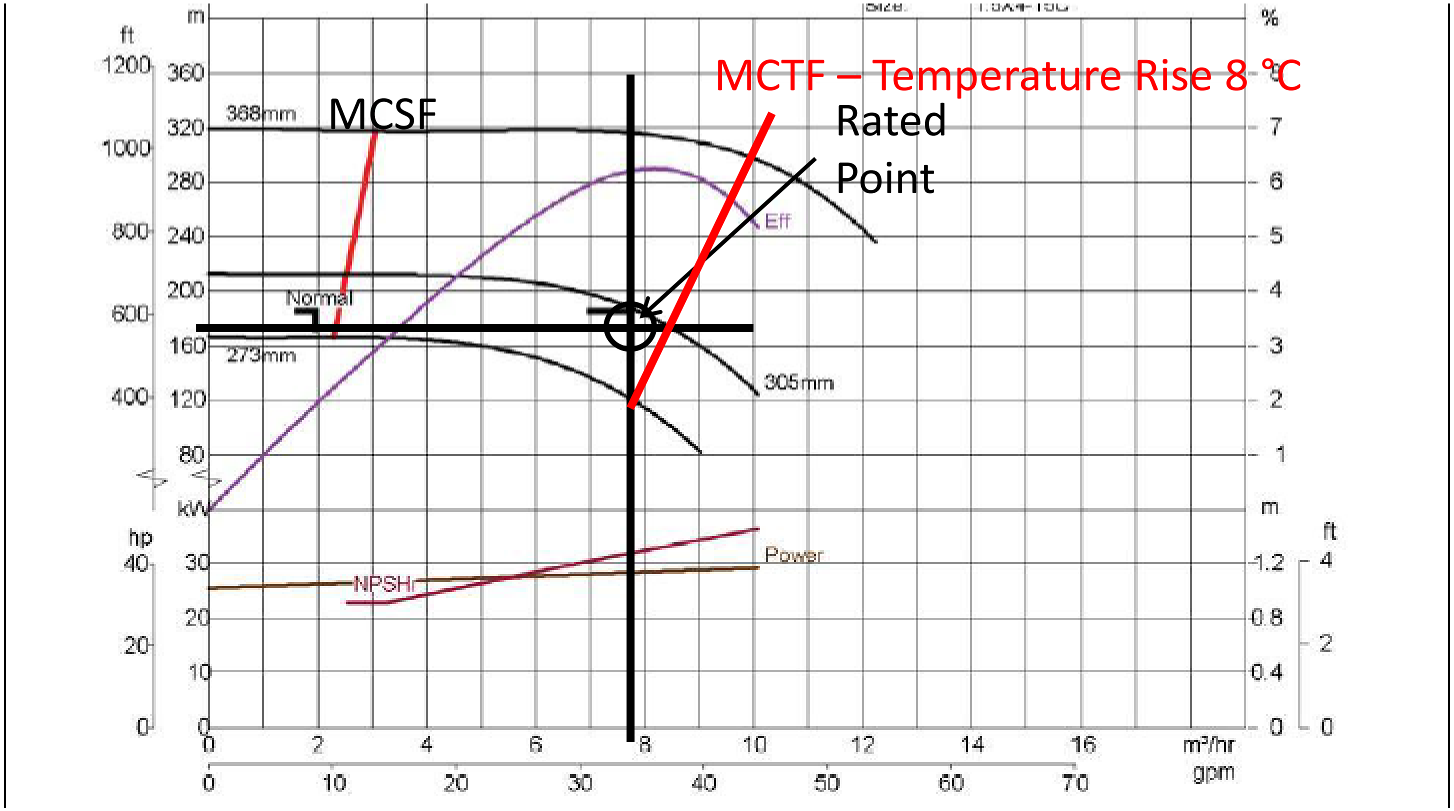
$\rho$ = density of liquid, in $kg/m^3$

$s$ = specific gravity of liquid

* See back-up slides for derivation

  - Minimum Flow (temp rise 8 °C) = ~ 8.5 $m^3/hr$

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump




NPSHr = 1.29 m

Rated Flow @ 95% BEP

Efficiency = 6.0 % (Low Flow)

MCTF = 8.5 $m^3$/hr (8 °C temp rise)

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump

- MCTF Considerations – Max Allowable Temperature Rise
  - Recirculation at first stage impeller wear rings
  - Calculate pressure available at pump suction above that required by the pump, P = 34 kPa

(Metric Units) $$P = \frac{(NPSHA - NPSHR) \times \rho \times g}{102}$$

(US Customary Units) $$P = \frac{(NPSHA - NPSHR) \times s}{2.31}$$

Where:

$P$ = Additional absolute pressure available at the pump suction nozzle, above that required by the pump, in kPa (psi)

$NPSHA$ = Net positive suction head available, in m (ft)

$NPSHR$ = Net positive suction head required, in m (ft)

2.31 = Constant

102 = Constant

g = gravitation constant, in m/s$^2$ (ft/s$^2$)

$\rho$ = density of liquid, in kg/m$^3$

$s$ = specific gravity of liquid

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump

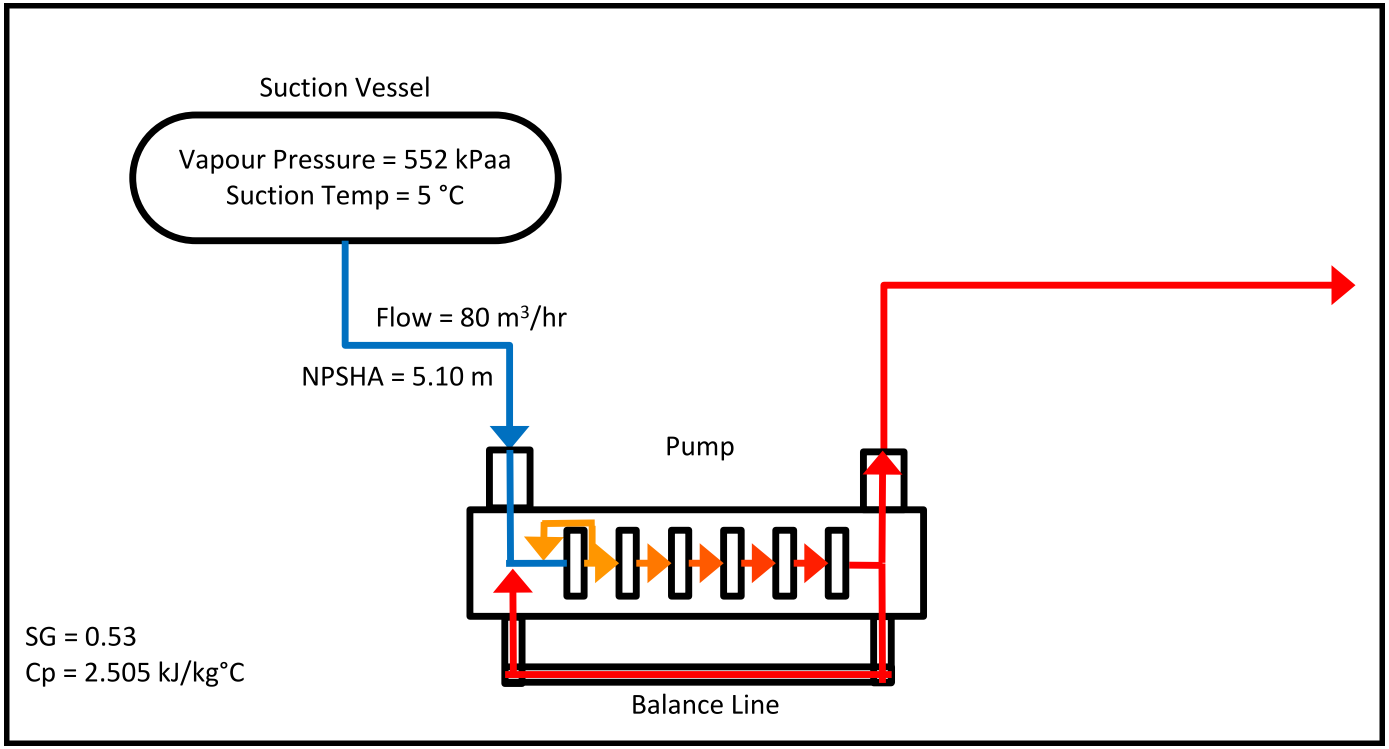
- MCTF Considerations - Max Temperature Rise (Con't)
  – Absolute pressure in suction vessel = 2213 kPaa
  – Sum of Suction Vessel Pressure and P = 2247 kPaa
  – The maximum allowable temperature rise is the difference between the saturation temperature corresponding to the pressure determined above and the temperature of the suction source
  – Saturation temperature at 2247 kPaa = 114.9°C
  – Suction temperature = 114 °C
  – Maximum Allowable Temperature Rise = <u>0.9 °C</u>
  – Minimum Flow (0.9 °C Temp Rise) = <u>No Flow for this Selection</u>

<u>Pump does not work!!</u>

# Minimum Continuous Thermal Flow Case Study BB5 Pump

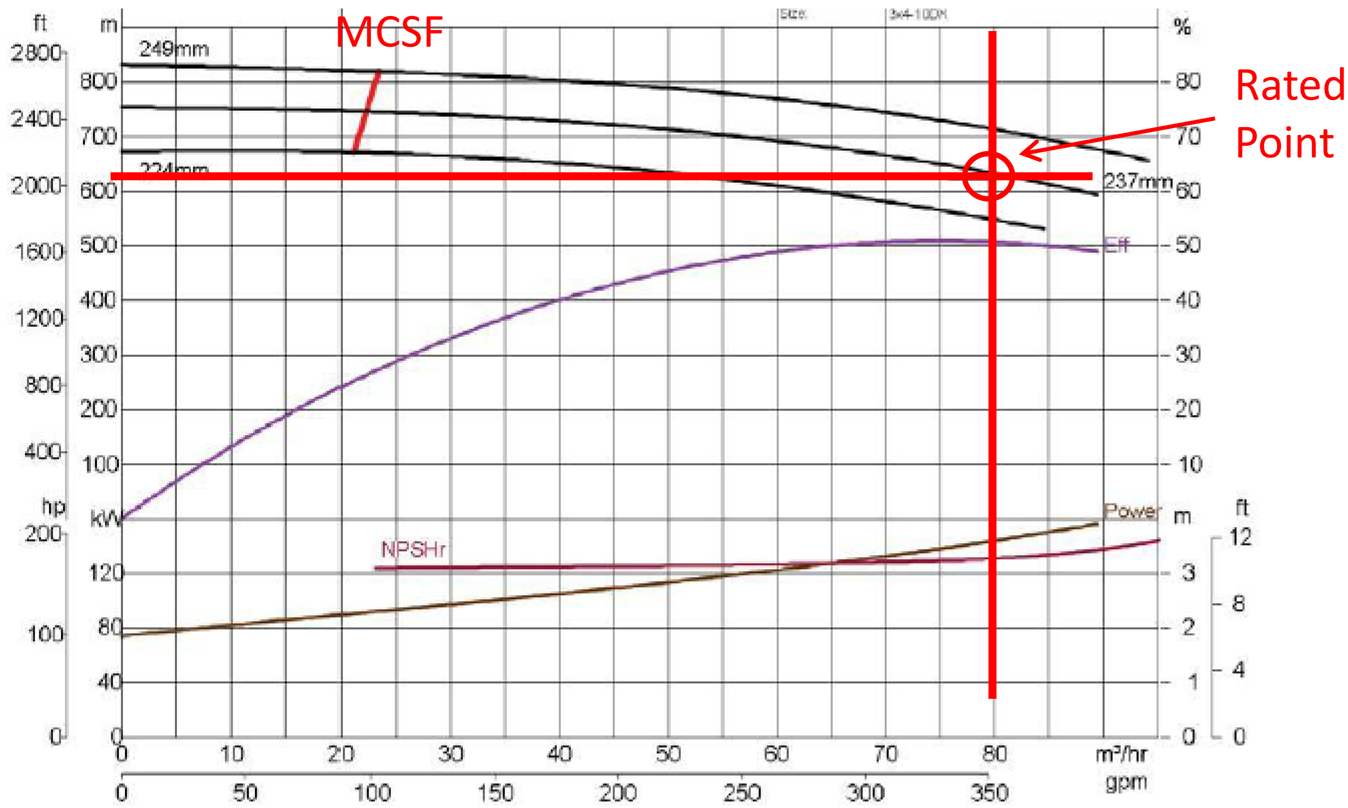
- Service: Off Spec LPG Pump
- Process Conditions:
  - Fluid = Hydrocarbon
  - SG = 0.53
  - Pvap = 552 kPaa
  - Suction Temperature = 5 °C
  - Flow = 80 $m^3$/hr
  - Cp = 2.505 kJ/kg°C
  - Head = 631 m
  - NPSHa = 5.10 m
  - Preliminary Selection – BB5, 3x4-10DX, 6 Stages, 3600 rpm

# Minimum Continuous Thermal Flow Case Study
# OH2 Pump




System Sketch – BB5 Pump

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump




NPSHa – NPSHr = 1.24 m

Rated Flow @ 104% BEP

Efficiency at Rated = 50.5%

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump

- MCTF Considerations – Temperature Rise 8 °C at Discharge
  - Temperature Rise at MCSF (per HI 1.3 – 2013) should be < 8 °C

(Metric Units) $$\Delta t = \frac{H}{102 \times C_P}\left(\frac{1}{\eta} - 1\right)$$

(US Customary Units) $$\Delta t = \frac{H}{778 \times C_P}\left(\frac{1}{\eta} - 1\right)$$

Where:

$\Delta t$ = temperature rise through the pump, in °C (°F)

$H$ = total developed heat at flow being considered, in m (ft)

778 = Constant

102 = Constant

$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)

η = efficiency of the pump at flow being considered, expressed as a decimal

* See back-up slides for derivation

  - Temperature Rise = 8.8 °C @ Discharge, MCSF Flow

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump

- MCTF Considerations – Temperature Rise 8 °C(Con't)
  - Min flow for temperature rise of 8 °C (HI 1.3 – 2013) at pump discharge

(Metric Units) $Q = \frac{433 P_P}{C_P \times \rho}$

(US Customary Units) $Q = \frac{P_P}{2.95 \times C_P \times s}$

Where:

$Q$ = minimum flow rate, in $m^3$/hr (gpm)

$P_P$ = input power at the minimum flow, in kW (hp)

2.95 = Constant

433 = Constant

$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)

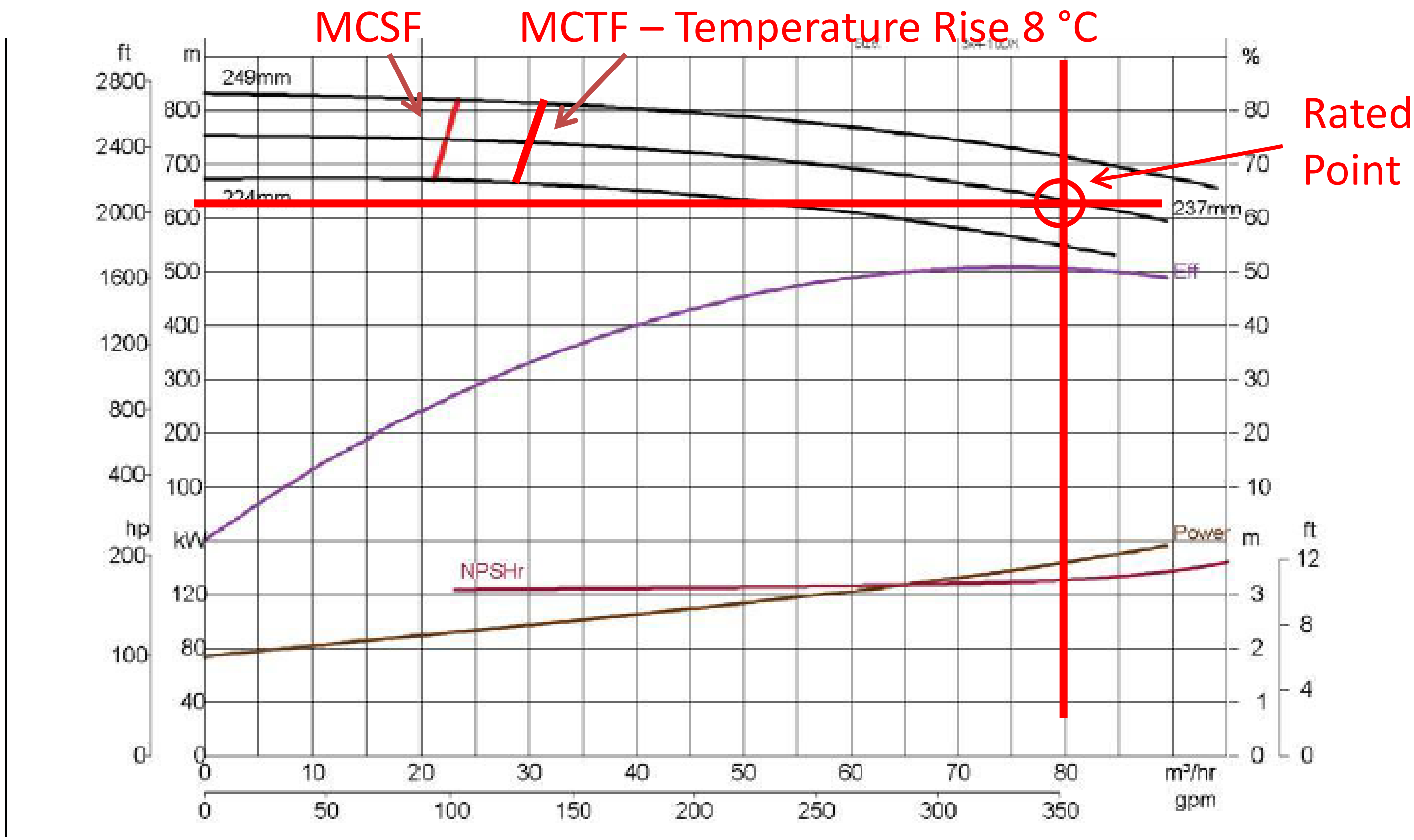
$\rho$ = density of liquid, in kg/$m^3$

$s$ = specific gravity of liquid

* See back-up slides for derivation

  - Minimum Flow (temp rise 8 °C) = ~ 29.4 $m^3$/hr

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump




NPSHa – NPSHr = 1.24 m

Rated Flow @ 104% BEP

Efficiency at Rated = 50.5%

MCTF = 29.4 $m^3$/hr (8 °C temp rise)

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump

- MCTF Considerations – Max Temperature Rise (1st Stage)
  - Recirculation at first stage impeller wear rings
  - Calculate pressure available at pump suction above that required by the pump, P = 63 kPa

(Metric Units) $$P = \frac{(NPSHA - NPSHR) \times \rho \times g}{102}$$

(US Customary Units) $$P = \frac{(NPSHA - NPSHR) \times s}{2.31}$$

Where:

$P$ = Additional absolute pressure available at the pump suction nozzle, above that required by the pump, in kPa (psi)

$NPSHA$ = Net positive suction head available, in m (ft)

$NPSHR$ = Net positive suction head required, in m (ft)

2.31 = Constant

102 = Constant

g = gravitation constant, in $m/s^2$ ($ft/s^2$)

$\rho$ = density of liquid, in $kg/m^3$

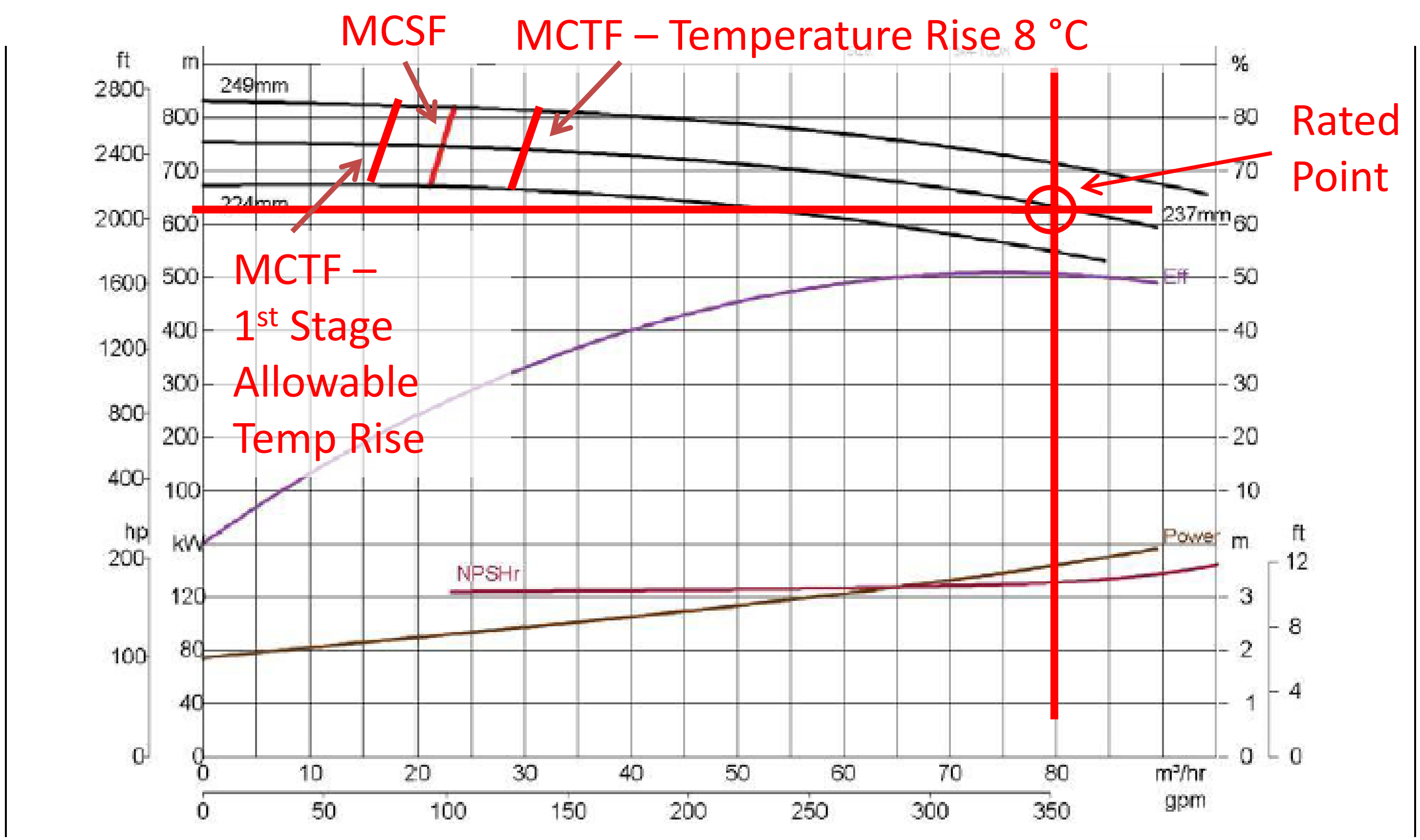
$s$ = specific gravity of liquid

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump

- MCTF Considerations - Max Temperature Rise (Con't)
  - Absolute pressure in suction vessel = 552 kPaa
  - Sum of Suction Vessel Pressure and P = 615 kPaa
  - The maximum allowable temperature rise is the difference between the saturation temperature corresponding to the pressure determined above and the temperature of the suction source
  - Saturation temperature at 615 kPaa = 6.7°C
  - Suction temperature = 5 °C
  - Maximum Allowable Temperature Rise (1st Stage) = 1.7 °C
  - Minimum Flow (1.7 °C Temp Rise, 1st Stage) = 17 $m^3$/hr

MCTF < MCSF

# Minimum Continuous Thermal Flow Case Study BB5 Pump




NPSHa – NPSHr = 1.24 m

Rated Flow @ 104% BEP

Efficiency at Rated = 50.5%

MCTF = 29.4 $m^3$/hr (8 °C Temp Rise)

MCTF = 17 $m^3$/hr (1$^{st}$ Stage Allowable Temp Rise)

# Minimum Continuous Thermal Flow Case Study
# BB5 Pump

- MCTF Considerations – Max Temp Rise (Balance Line)
  - Balance Line (B/L) flow returns to pump suction pressure
  - The temperature rise limitation of 1.7 °C for the first stage impeller is also applicable for the balance line flow, because it is based on suction conditions
  - Maximum allowable temperature rise (B/L) = 1.7 °C
  - This temperature needs to be looked at across all 6 stages of the pump, when considering for the balance line
  - Temperature rise is 8.8 °C at min flow (calculated earlier)
  - Temperature rise at rated flow is 2.4 °C
  - There is no flow for this pump selection at which the 1.7 °C temperature rise for the balance line can be met

Balance Line Must Be Returned to Suction Vessel

# Minimum Continuous Thermal Flow - Assumptions

- The calculations shown in this presentation are conservative, as they include some assumptions:
  - All heat generated is transferred into the fluid. In reality heat will be lost into the pump material, surrounding air, bearings, and oil.
  - No credit is given to quenching of the fluid recirculated over the first stage wear rings. In reality this small volume of hot fluid will be quenched somewhat by the much larger volume of cold suction fluid.

# Minimum Continuous Thermal Flow - Summary

- Both MCSF and MCTF need to be considered when evaluating minimum flow in a centrifugal pump
- When considering MCTF evaluate:
  - Temperature rise through the pump to discharge
  - Allowable temperature rise for first stage recirculation
  - Temperature rise in balance line fluid for multi-stage
- Allowable temperature rise can be calculated by comparing suction temperature to saturation temperature at suction pressure less NPSHr
- Follow best practices and design guidelines to help mitigate or avoid issues with MCTF during selection

# Pump Design and Selection Considerations to Avoid Temperature Rise Issues

- Target a vapor pressure margin of 15 psig (100 kPag) minimum for low specific gravity fluids (0.7 and lower) in all operating conditions for reliable pump operation and mechanical seal life.
- Pumps with low vapor pressure margin (<100 kPag), high vapor pressures, and low specific gravity (< 0.7) require more scrutiny during evaluation and selection
- EPC / End User should provide pump supplier with vapor pressure vs temperature curves to improve evaluation
- Avoid low efficiency selections for services with low vapor pressure margin. Consider alternative selections and pump types

# System Design and Selection Considerations to Avoid Temperature Rise Issues

- Consider raising the suction vessel
- Add a suction line cooler
- Increase flow recirculation to suction vessels to avoid low flow operation
- Return balance lines on multi-stage pumps to a suction vessel
- Discuss with supplier the possibility of routing balance lines to second stage suction instead of first stage suction
- Maximize suction pressure and NPSHa, by carefully designing suction piping (more in following slides)

# Pump Piping Layout Considerations

- Maximize suction pressure and NPSH margin by considering the following:
  - Minimize fittings and elbows upstream of the pump
  - Select suction piping to minimize friction losses
  - Consider using a temporary strainer instead of a permanent strainer. Monitor suction pressure while temporary strainer is in.
  - Maintain 5 diameters of straight run upstream of the pump, this will help maximize vapor pressure margin
  - Use eccentric reducers on horizontal end suction pump suction lines, following the guidelines in HI 9.6.6, to avoid forming vapor pockets

# Questions?

- Michael Schleppe – Fluor Canada Ltd
  - Michael.schleppe@fluor.com

- Brian Yarnot – Goulds Pumps (ITT)
  - Brian.yarnot@itt.com

# References

- Figures are used with permission from Goulds Pumps (ITT)
- Equations are based on the engineering guidelines contained within:

American National Standards Institute Inc. / Hydraulic Institute. (2013)._American National Standard for Rotodynamic Centrifugal Pumps for Design and Application_(ANSI/HI 1.3-2013).Retrieved from https://ewb.ihs.com/

# Back-Up Slides

- Derivation of temperature rise formula
  - Target Formula:

(Metric Units) $$\Delta t = \frac{H}{102 \times C_P}\left(\frac{1}{\eta} - 1\right)$$

(US Customary Units) $$\Delta t = \frac{H}{778 \times C_P}\left(\frac{1}{\eta} - 1\right)$$

Where:

$\Delta t$ = temperature rise through the pump, in °C (°F)

$H$ = total developed heat at flow being considered, in m (ft)

778 = Constant

102 = Constant

$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)

η = efficiency of the pump at flow being considered, expressed as a decimal

# Back-Up Slides

– Metric Unit Derivation:

$$P_f = \dot{m} C_P \Delta t$$

$$P_B(1-\eta) = \rho Q C_P \Delta t$$

$$\frac{P_H}{\eta}(1-\eta) = \rho Q C_P \Delta t$$

$$\frac{Q\rho g H}{\eta}(1-\eta) = \rho Q C_P \Delta t$$

$$gH\left(\frac{1}{\eta} - 1\right) = C_P \Delta t$$

$$\Delta t = \frac{gH}{C_P}\left(\frac{1}{\eta} - 1\right)$$

$$\Delta t = \frac{H}{\frac{1}{g} C_P}\left(\frac{1}{\eta} - 1\right)$$

# Back-Up Slides

– Metric Unit Derivation Continued:

$$\Delta t = \frac{H}{\frac{1}{g} C_P} \left( \frac{1}{\eta} - 1 \right)$$

$$\Delta t = \frac{H}{102 C_P} \left( \frac{1}{\eta} - 1 \right)$$

Where:

$P_f$ = power input to fluid due to friction, in kW

$P_B$ = break power, in kW

$P_H$ = hydraulic power, in kW

102 = Constant (based on 1000 (conversion from kJ/kgK to J/kgK) / 9.81 $m^2/s$)

$\dot{m}$ = mass flow, in kg/s

$Q$ = flow rate, in $m^3/s$

$\rho$ = density, in $kg/m^3$

# Back-Up Slides

- Derivation of minimum flow formula
  - Target Formula:

(Metric Units) $Q = \frac{433 P_P}{C_P \times \rho}$

(US Customary Units) $Q = \frac{P_P}{2.95 \times C_P \times s}$

Where:
$Q$ = minimum flow rate, in $m^3$/hr (gpm)
$P_P$ = input power at the minimum flow, in kW (hp)
2.95 = Constant
433 = Constant
$C_P$ = specific heat of the liquid at pumping temperature, in kJ/kg K (Btu/lbm °F)
$\rho$ = density of liquid, in kg/$m^3$
$s$ = specific gravity of liquid

# Back-Up Slides

– Metric Unit Derivation:

$$P_f = \dot{m} C_P \Delta t$$

$$P_f = \rho Q C_P \Delta t$$

Assume:

$$P_f \sim P_P \text{ at min Flow}$$

$\Delta t$ = 8.3 °C (Based on 15 °F)

$$\Longrightarrow P_P = \rho Q C_P (8.3°\text{C})$$

$$P_P = \rho Q \left(\frac{1}{3.6 \times 10^3} \frac{\frac{m^3}{s}}{\frac{m^3}{hr}}\right) C_P (8.3°\text{C})$$

$$Q = \frac{(3.6 \times 10^3) P_P}{(8.3) C_P \times \rho}$$

$$Q = \frac{433 P_P}{C_P \times \rho}$$

Where:

$P_f$ = power input to fluid due to friction, in kW