Taco
comfort made easy

Comprehensive Pump Series - Best Practices of Centrifugal Pumps

from

CAT

Central Air conditioning and Trading and Services

Presentation -1

- MAJOR PUMP COMPONENTS
- FUNCTIONS OF COMPONENTS
- DESIGN CALCULATIONS ON SHAFT DEFELCTIONS
- Losses in pumps
- Pump selection Basic
- Trouble shooting

PART I

Criteria for Selection of Suitable Type of Pump

Design Objectives

- Achieve design flow rate and total head.
- Attain optimum efficiency.
- Obtain stable head-capacity characteristics.
- Minimize NPSH required.
- Ensure wide operating range.
- Optimize pump size.
- Attain non-overloading power characteristics.
- Minimize vibration and noise.
- Minimize hydraulic axial and radial thrust loads.
- Design for ease of production.
- Ensure maximum interchangeability.
- Minimize cost.

MAJOR COMPONENTS OF A CENTRIFUGAL PUMP

STATIONARY PARTS

- CASING(TOP & BOTTOM)
- BEARING HOUSING
- BEARING BRACKET
- CASING WEAR RINGS
- BEARING END COVER

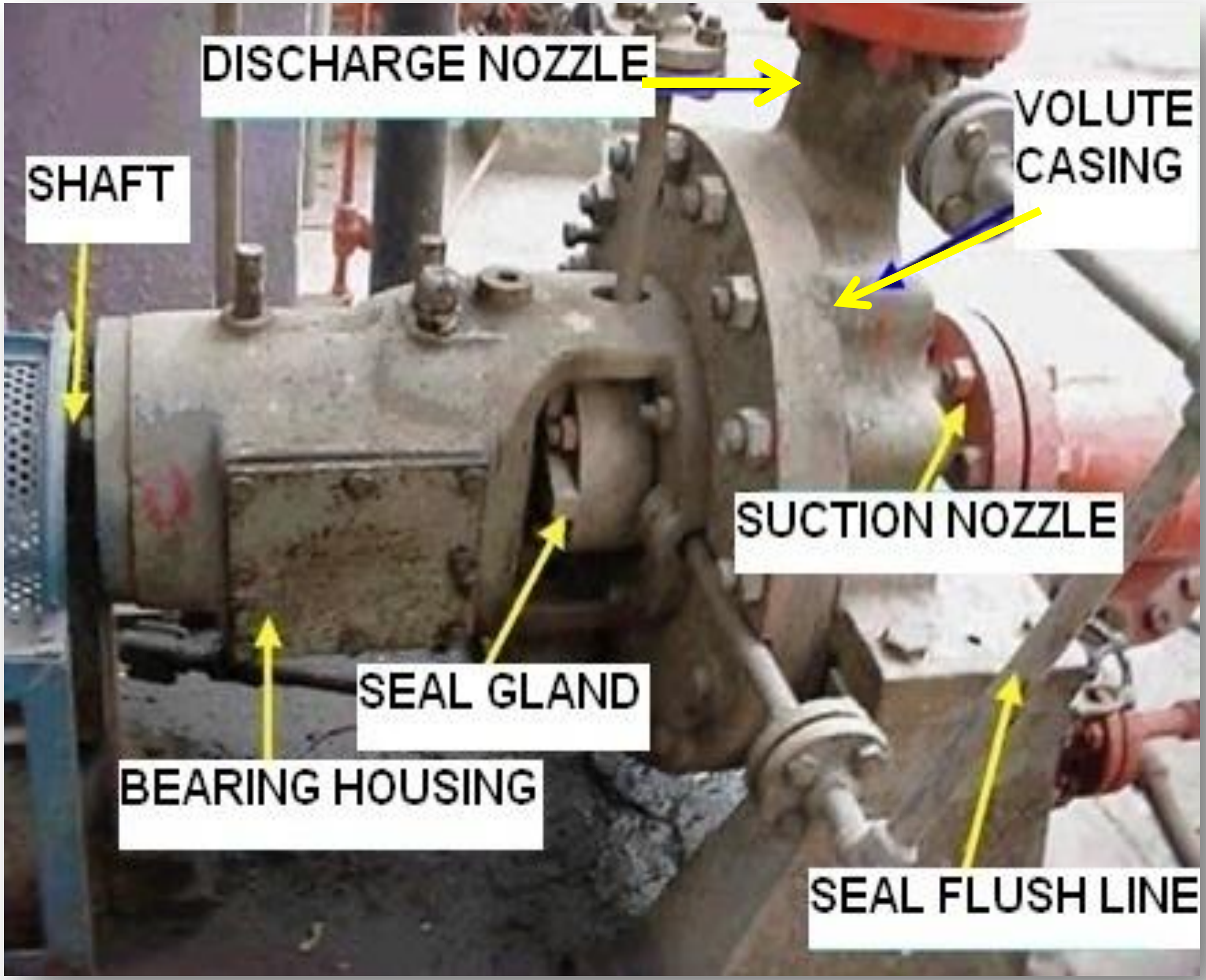
ROTATING ELEMENT

- IMPELLER
- IMPELLER WEAR RING
- SHAFT
- SHAFT SLEEVE
- SHAFT SLEEVE NUTS
- MECHANICAL SEAL (ROTATING ELEMENT)
- BEARING (INNER RACE)
- BEARING LOCK NUT

ACCESSORIES

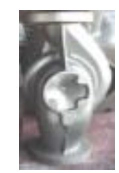
- STUFFING BOX SEAL
- SEAL FLUSHING LINE
- LUBRICATING / COOLING ARRANGEMENTS

MAJOR COMPONENTS OF AN END SUCTION PUMP

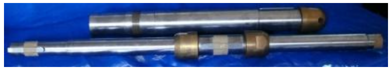
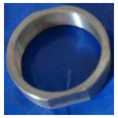


MAJOR COMPONENTS OF A SPLIT CASE PUMP

FUNCTIONS OF CENTRIFUGAL PUMP COMPONENTS

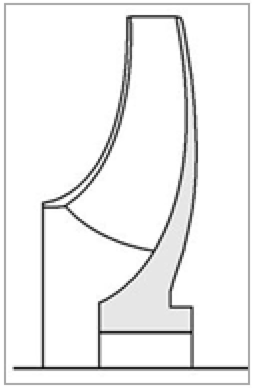
COMPONENTS	FUNCTIONS
IMPELLER	➢ TO DEVELOP DYNAMIC HEAD  SINGLE SUCTION  DOUBLE SUCTION
CASING	➢ TO CONVERT KINETIC ENERGY INTO PRESSURE ENERGY WITH MINIMUM HYDRAULIC LOSSES BY MEANS OF VOLUTE, DIFFUSERS OR GUIDE VANES ➢ INCORPORATES NOZZLES TO CONNECT SUCTION & DISCHARGE PIPING ➢ DIRECTS FLOW INTO & OUT OF THE IMPELLER. ➢ PROVIDES SUPPORT TO THE BEARING BRACKET  SPLIT-CASE  END SUCTION  IN-LINE
WEAR RING	➢ TO PROTECT THE ROTATING IMPELLER FROM RUBBING WITH THE STATIONARY CASING. TO PROVIDE A REPLACEABLE WEAR JOINT ➢ TO CONTROL THE LEAKAGE LOSSES ACROSS THE ANNULAR PATH BETWEEN IMPELLER AND WEAR RING  WEAR RING
IMPELLER NUT	➢ TO LOCK THE IMPELLER IN ITS PROPER AXIAL POSITION ➢ TO PREVENT AXIAL MOVEMENT DUE TO HYDRAULIC THRUST  IMPELLER NUT

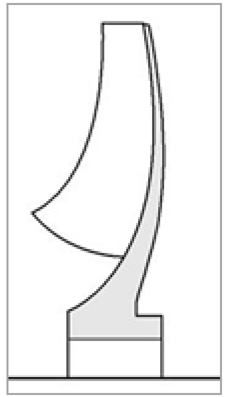
FUNCTIONS OF CENTRIFUGAL PUMP COMPONENTS

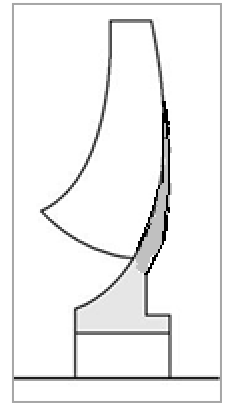
COMPONENTS	FUNCTIONS
SHAFT	➢ TRNSMITS TORQUE TO THE IMPELLER FROM THE DRIVER ➢ SUPPORTS IMPELLER AND OTHER ROTATING ELEMENTS  SHAFT
SLEEVE	➢ TO ENHANCE THE STIFFNESS OF THE ROTATING ELEMENT ➢ TO PROTECT THE SHAFT FROM ABRASION WEAR AT PACKED STUFFING BOX OR AT LEAKAGE JOINTS ➢ TO PROTECT THE SHAFT FROM EROSION & CORROSION  SLEEVE
SLEEVE NUT	➢ TO FASTEN THE SLEEVES TO THE SHAFT ➢ TO PREVENT MOVEMENT OF THE SLEEVE  SLEEVE NUT
THROTTLE BUSH	➢ CAUSES PRESSURE BREAKDOWN AS THE LIQUID THROTTLES ACROSS IT THUS BOOSTING THE SERVICE LIFE OF PACKING ➢ SERVES AS A LANDING FOR THE LOWEST RING OF THE PACKING ➢ RESTRICTS SOLID PARTICLE IN THE PUMPED LIQUID FROM GETTING EMBEDDED INTO THE PACKING AREA AND THUS PROTECTING SHAFT OR SLEEVE FROM WEAR ➢ ACTS AS ADDITIONAL HYDRODYNAMIC BEARING SUPPORT FOR THE ROTATING ELEMENT AND REDUCES SHAFT DEFLECTION ➢ PREVENTS SOME AMOUNT OF LIQUID LEAKING OUT FROM THE STUFFING BOX TO FLOW BACK  THROTTLE BUSH

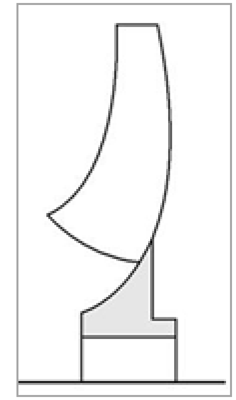
TYPES OF IMPELLERS BASED ON MECHANICAL CONSTRUCTION

- USED DEPENDING ON THE NATURE OF THE LIQUID PUMPED

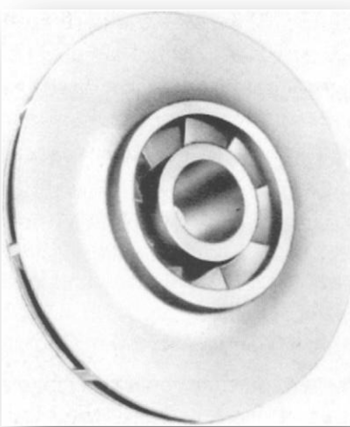








SECTIONAL VIEWS OF IMPELLERS SHOWN BELOW



CLOSED IMPELLER

SEMI-OPEN IMPELLER

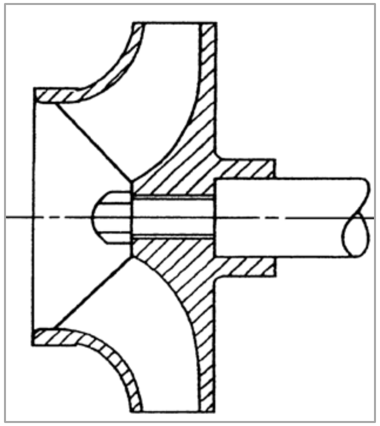
PARTIAL SHROUDED IMPELLER

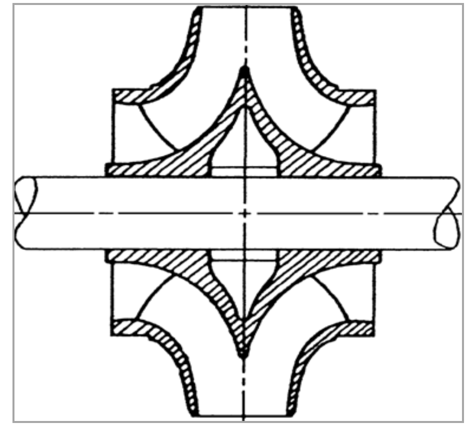
OPEN IMPELLER

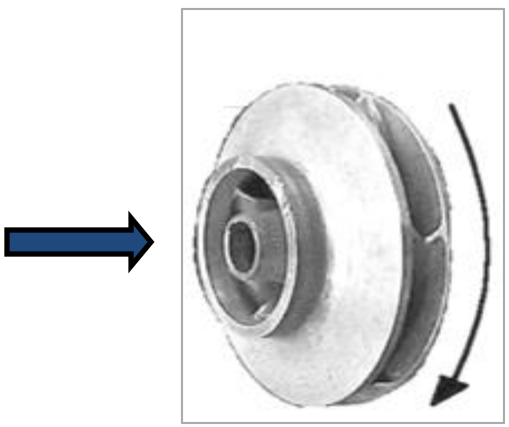
Type of impellers

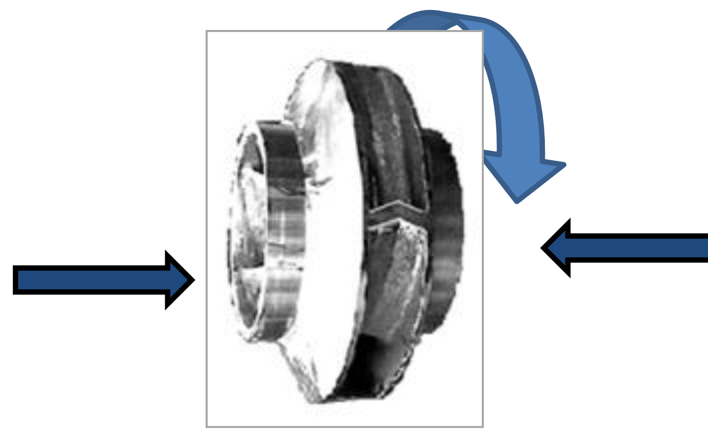
Closed Impeller	Semi Open	Open
This type of impeller works best with clear water.	*Semi open impellers are used for fibers or potentially clogging materials in the pumping liquid*	*Open impellers are used for high speed pumps of over 10,000 rpm.*
After extensive operation and wear, pump efficiency can normally be restored to original levels by replacing the inlet wearing ring (originally clearance) to the adjacent casing wearing ring.	*The impeller requires tight clearance be maintained between the open face and its mating stationary surface (clearance between 0.25 and 0.38 mm*	

TYPES OF IMPELLERS BASED ON NUMBER OF SUCTION EYES





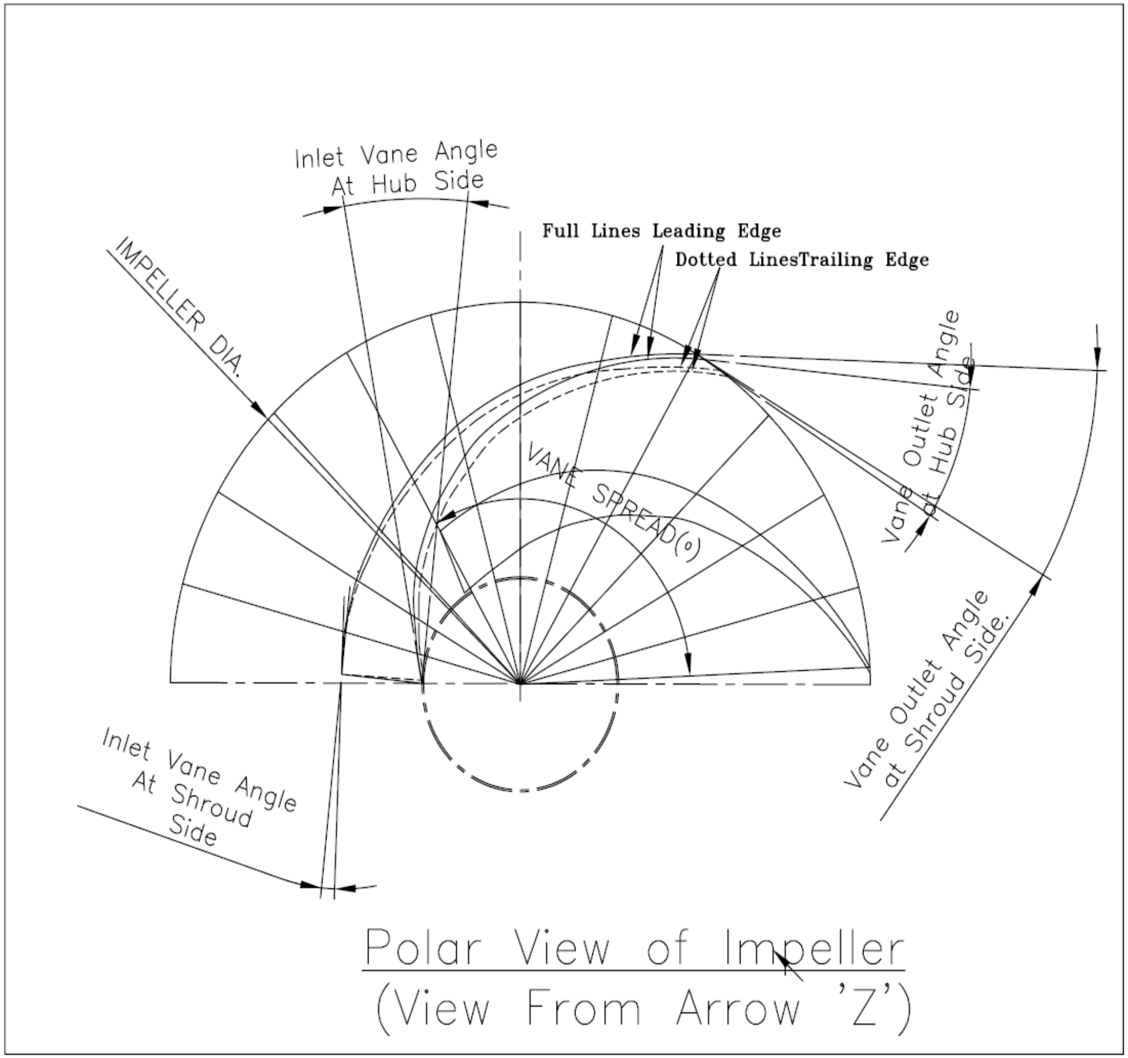




SINGLE SUCTION IMPELLER

DOUBLE SUCTION IMPELLER

MAIN DESIGN PARAMETERS OF AN IMPELLER



NUMBER OF VANES

IMPELLER DIA.

IMPELLER WIDTH

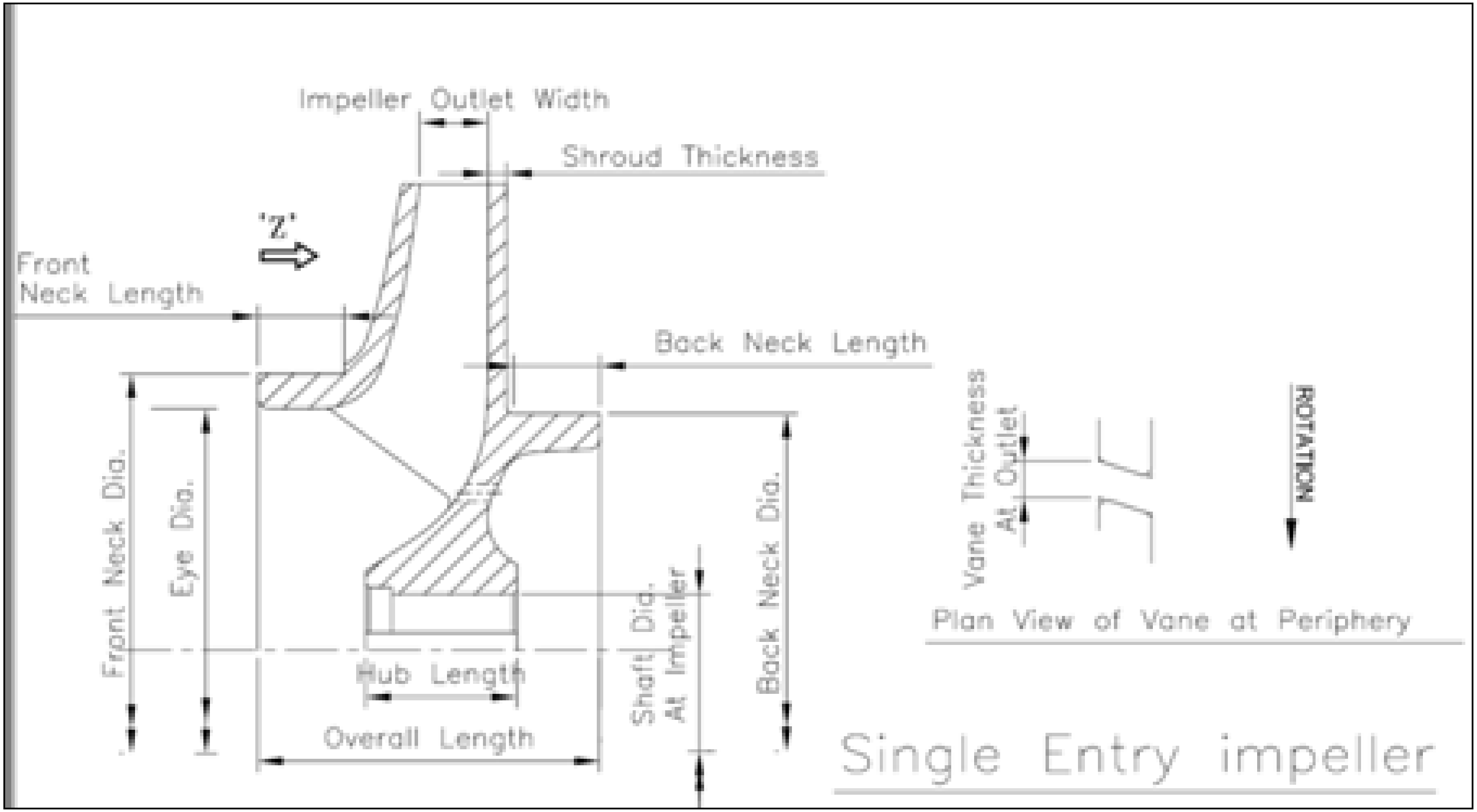
VANE OUTLET ANGLE

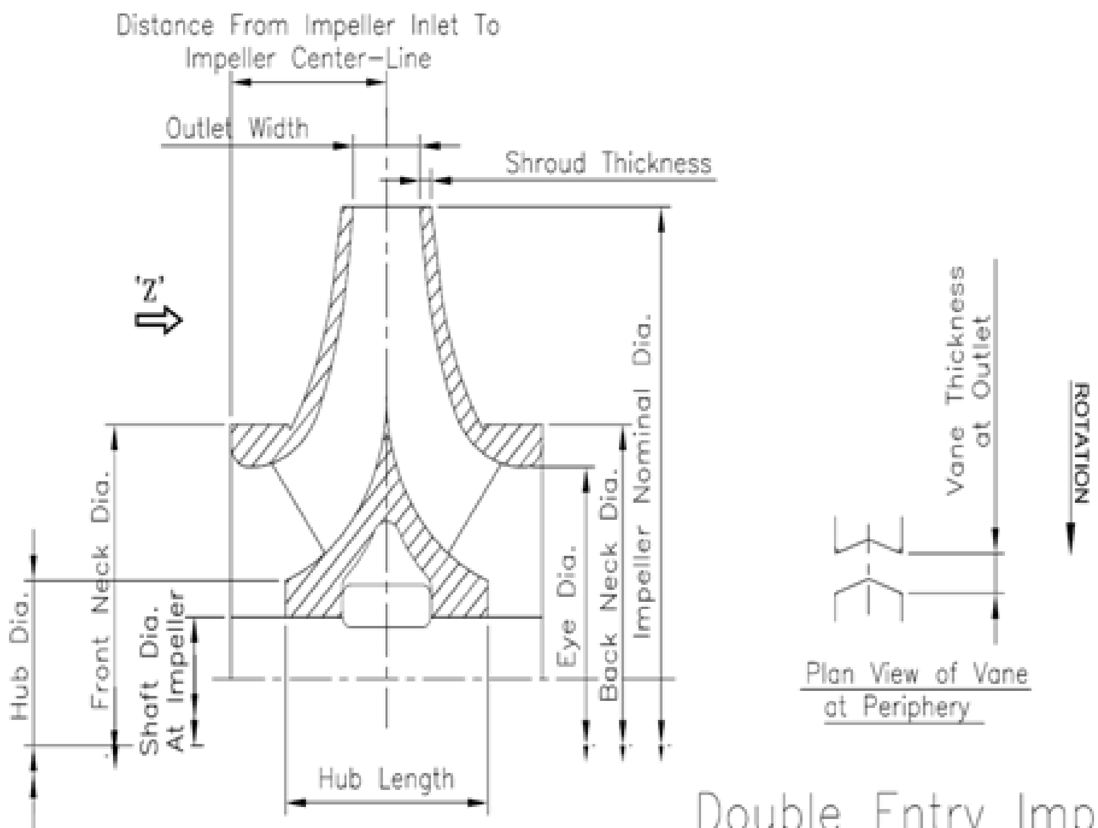
VANE SPREAD

EYE DIAMETER

VANE INLET ANGLE

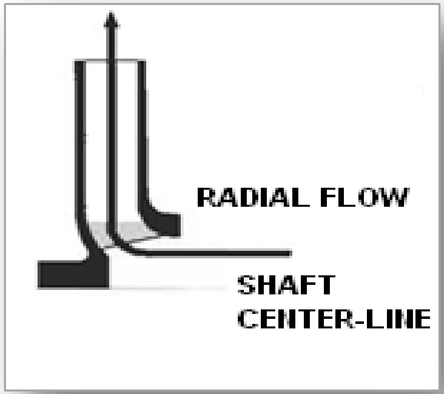
MAJOR DIMENSIONS OF AN IMPELLER

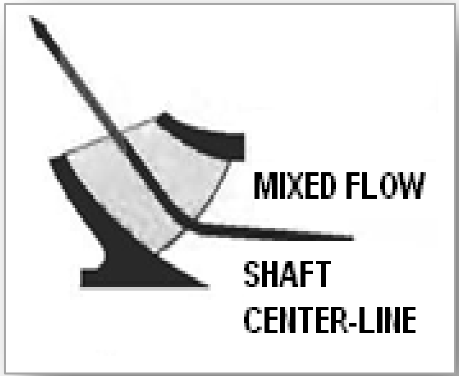


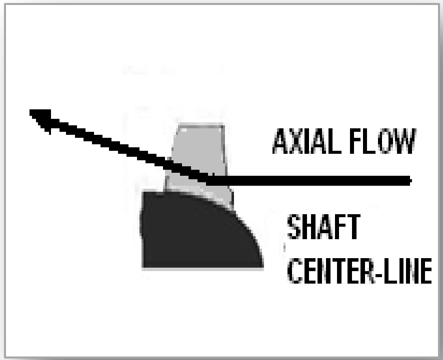


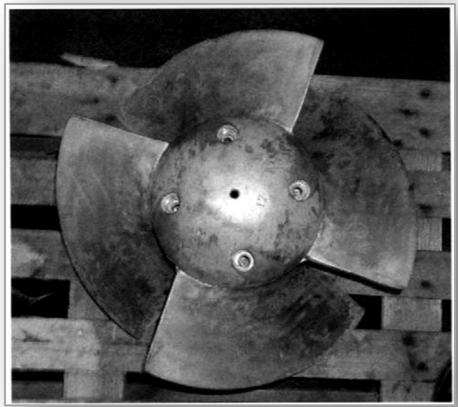
Double Entry Impeller

TYPES OF IMPELLERS BASED ON THE MAJOR DIRECTION OF FLOW
WITH RESPECT TO THE AXIS OF ROTATION









RADIAL VANE IMPELLER

SUITABLE FOR DISCHARGING RELATIVELY SMALL QUANTITY OF FLOW AGAINST HIGH HEAD

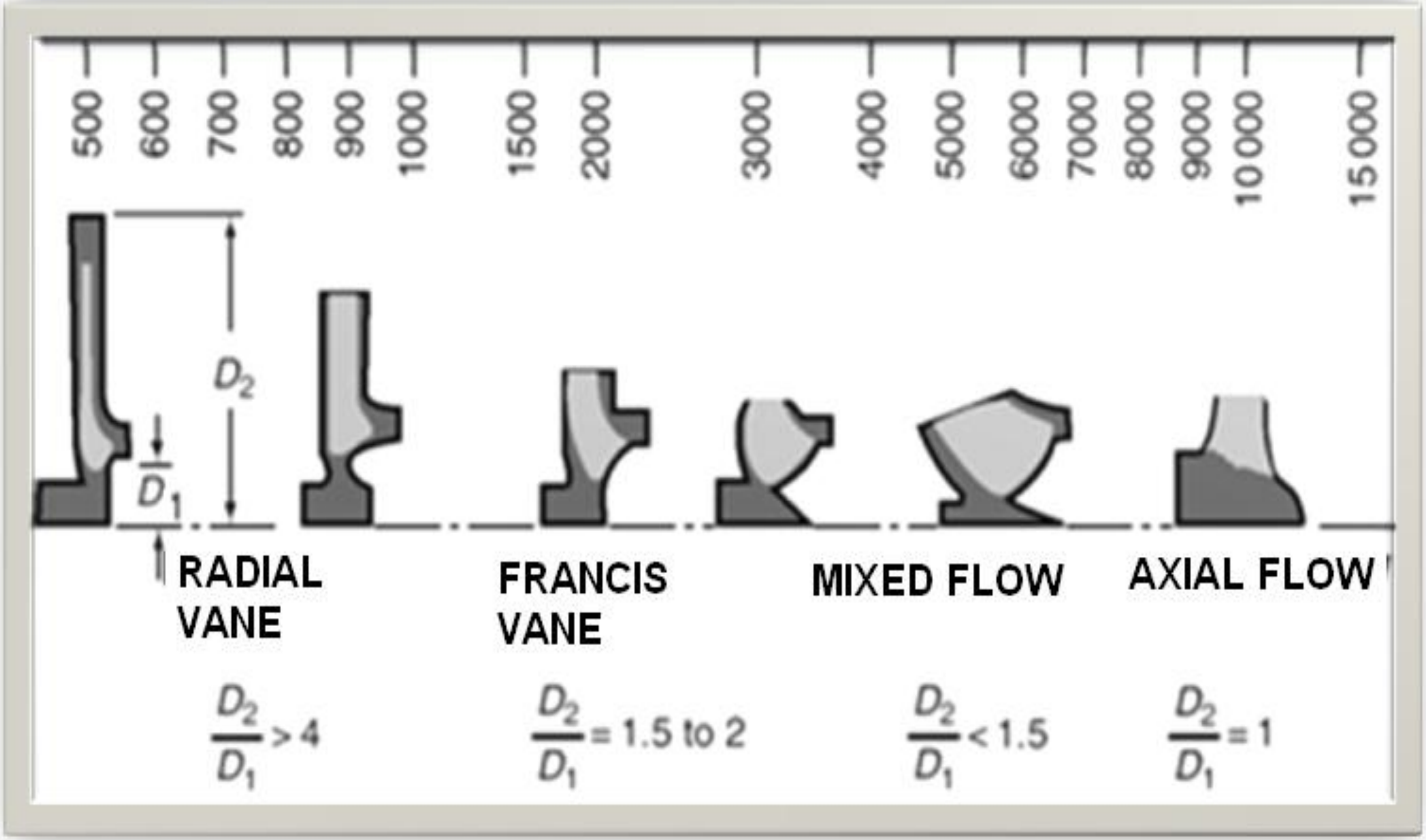
MIXED FLOW IMPELLER

SUITABLE FOR DISCHARGING LARGE QUANTITY OF FLOW AGAINST MEDIUM HEAD

PROPELLER

SUITABLE FOR DISCHARGING LARGE QUANTITY OF FLOW AGAINST SMALL HEAD

CHANGE OF IMPELLER SHAPE WITH SPECIFIC SPEED



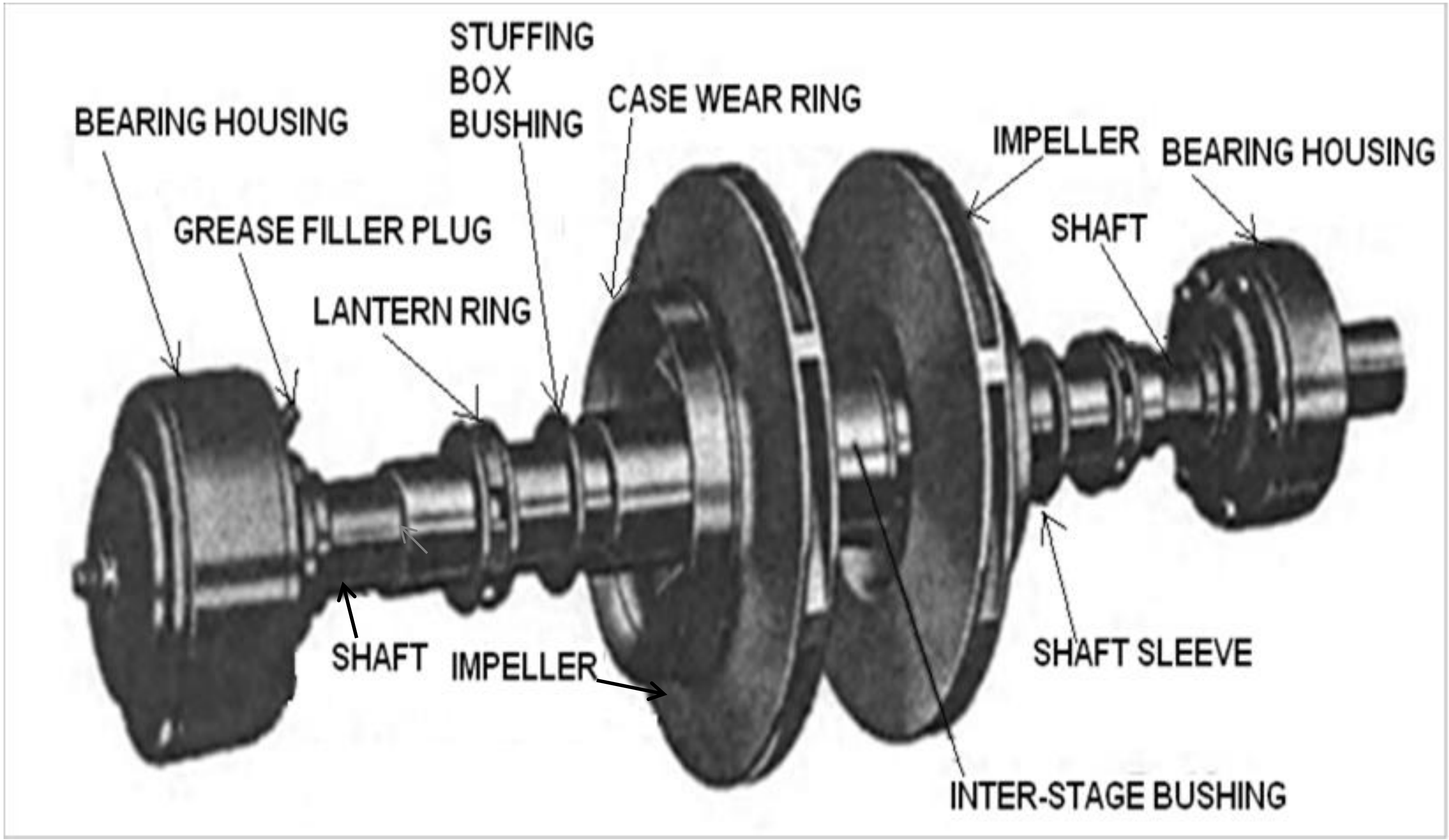
SPECIFIC SPEED
&
EFFICIENCY

TYPES OF IMPELLERS BASED ON THEIR RELATIVE POSITIONS ON THE SHAFT

OVER-HUNG IMPELLER

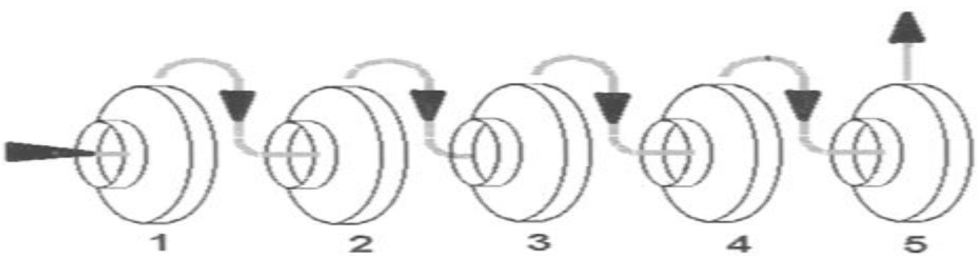
IMPELLER BETWEEN BEARINGS

TWO-STAGE PUMP ROTATING ELEMENT ASSEMBLY

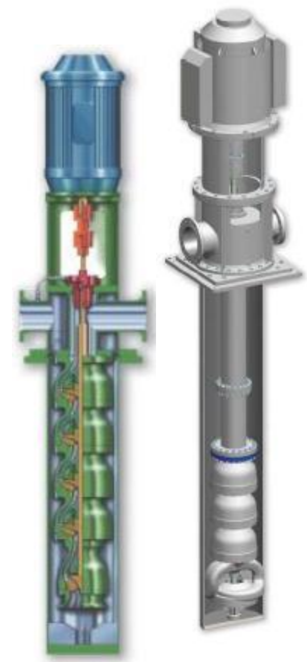


Multistage Pumps

Essentially a High Head Pump having two or more Impellers Mounted on a Common Shaft in Series

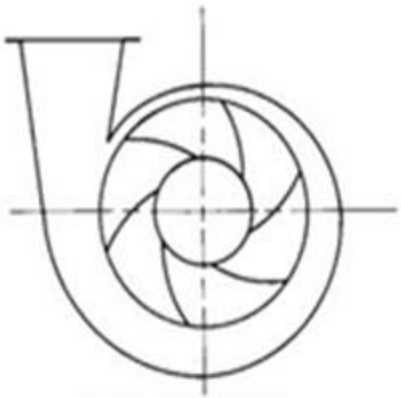


Multi-stage Pump

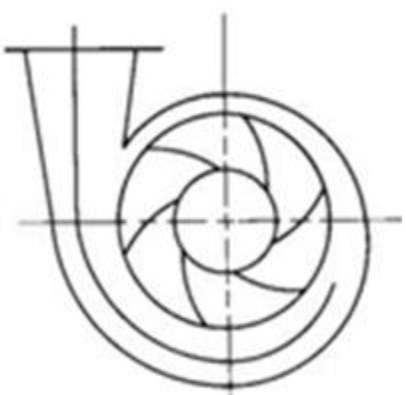


Vertical Multistage Can Pumps

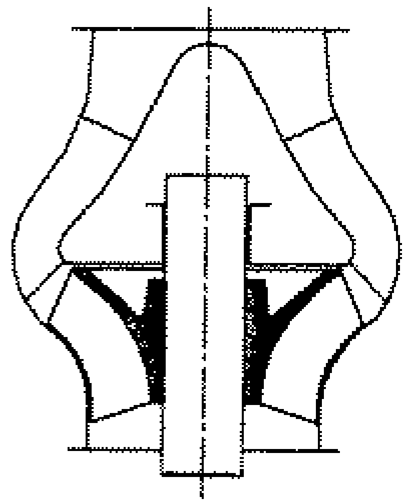
VARIOUS TYPES OF COLLECTORS & THEIR ADVANTAGES & DISADVANTAGES



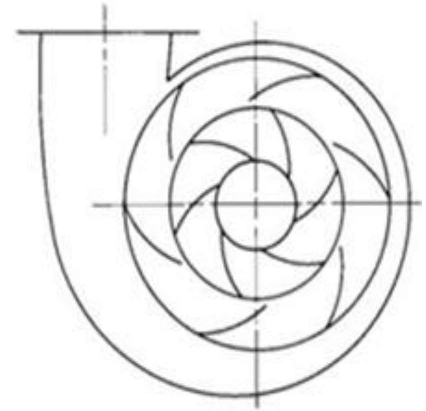
SINGLE VOLUTE



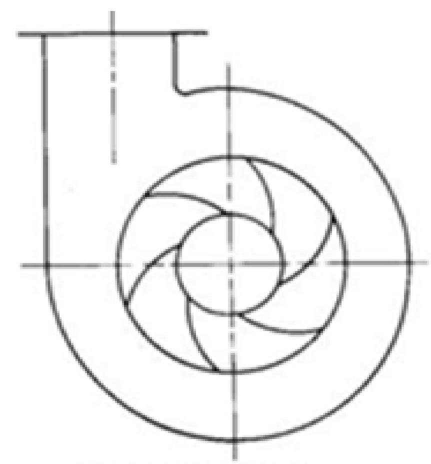
DOUBLE VOLUTE



VERTICAL PUMP WITH DIFFUSER



VOLUTE WITH DIFFUSER VANES



CIRCULAR VOLUTE

FUNCTIONS OF A PUMP VOLUTE

1. **TO CONVERT KINETIC ENERGY** IMPARTED BY THE IMPELLER INTO PRESSURE ENERGY

2. **TO MINIMIZE LOSSES** DURING THIS ENERGY CONVERSION PROCESS

3. THE PUMP CASING DOES NOT TAKE ANY **PART IN DYNAMIC HEAD GENERATION**

4. THE BEST VOLUTES ARE OF **CONSTANT VELOCITY DESIGN**

5. **KINETIC ENERGY IS CONVERTED INTO PRESSURE ENERGY** ONLY IN THE DIFFUSION NOZZLE IMMEDIATELY AFTER THE VOLUTE THROAT

TYPES OF VOLUTES – SINGLE VOLUTE CASINGS

1. SINGLE VOLUTE CASINGS ARE OF CONSTANT VELOCITY DESIGN.
2. THEY ARE EASY TO CAST AND ECONOMICAL TO MANUFACTURE.
3. THEY PRODUCE THE BEST EFFICIENCIES AT DESIGN POINT COMPARED TO OTHER COLLECTOR SHAPES.
4. PRESSURE DISTRIBUTION AROUND THE IMPELLER IS UNIFORM ONLY AT THE DESIGN POINT.
5. RESULTANT RADIAL THRUST DUE TO NON-UNIFORM PRESSURE DISTRIBUTION IS GIVEN BY:

$$\underline{P = K \times H \times D_2 \times b_2 \times S.G / 2.31}$$

P = RADIAL THRUST IN POUNDS

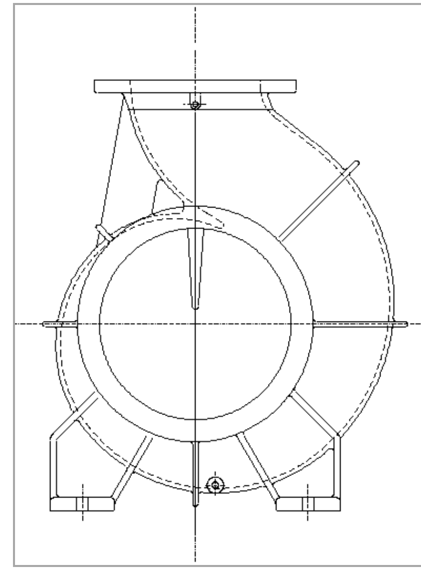
K = THRUST FACTOR

H = DEVELOPED HEAD/STAGE IN FT

D_2 = IMPELLER DIA, IN INCHES

b_2 = IMPELLER WIDTH IN INCHES (INCLUDING SHROUDS)

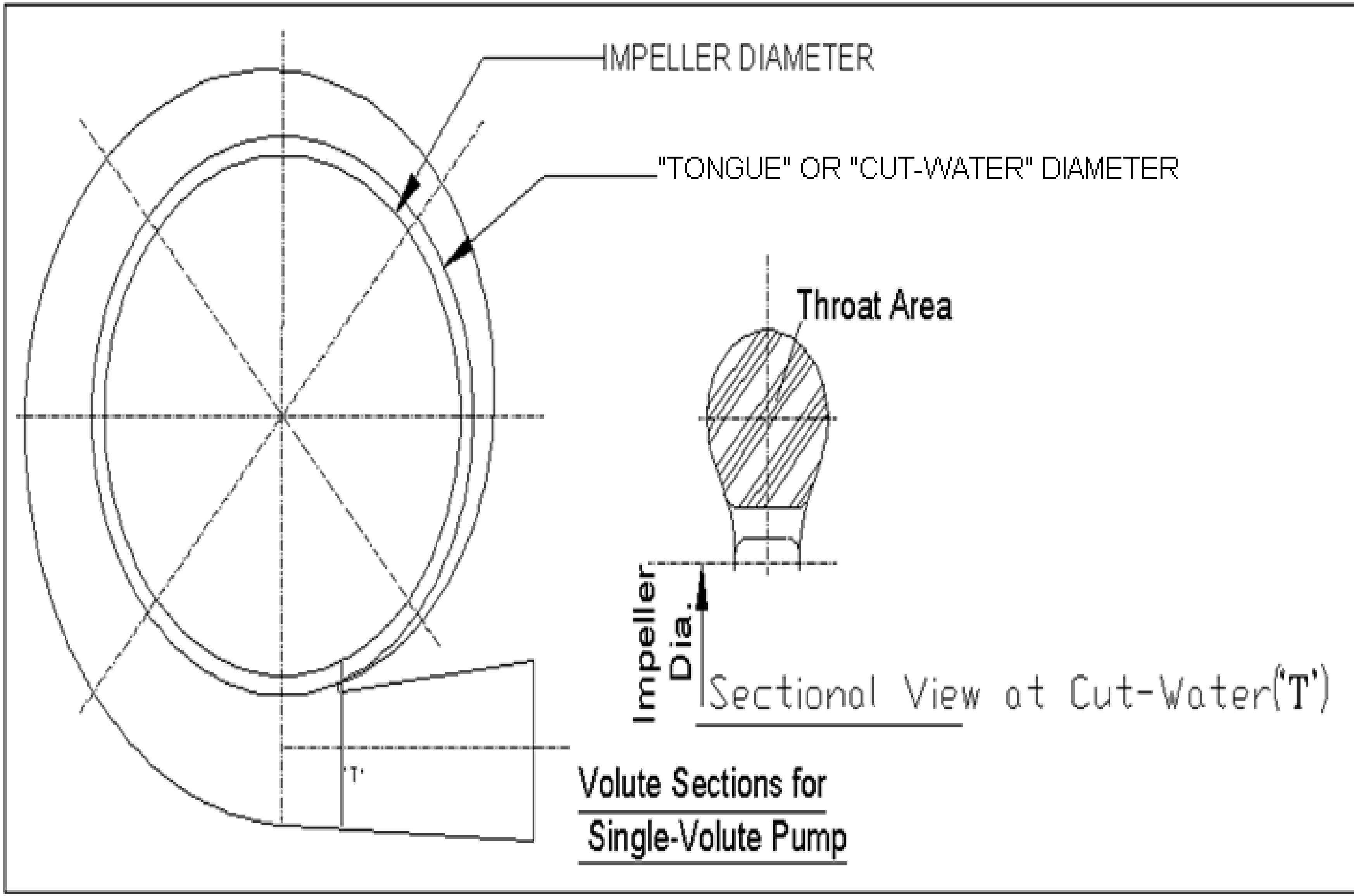
6. SINGLE VOLUTE PUMPS ARE USED MAINLY ON LOW CAPACITY, LOW SPECIFIC SPEED PUMPS.
7. THEY ARE ALSO USED FOR SPECIAL APPLICATIONS SUCH AS SLURRIES OR SOLID HANDLING PUMPS.



SINGLE VOLUTE

VOLUTE AS-CAST

VOLUTE SECTIONS FOR SINGLE VOLUTE PUMP



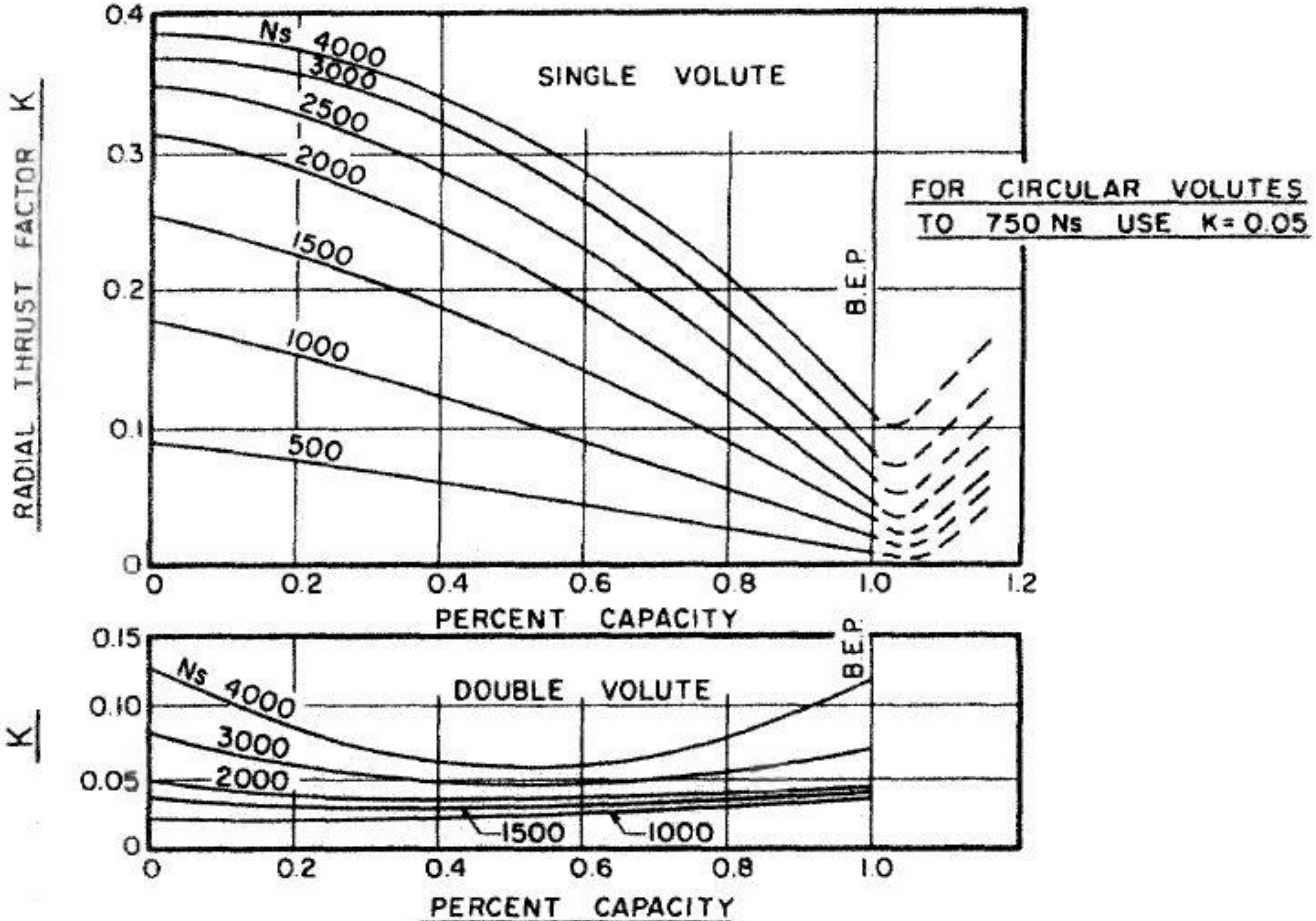
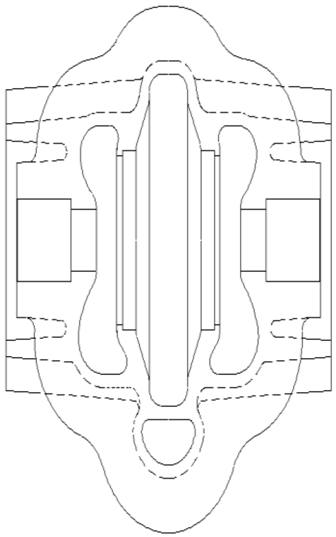


Figure 5-2. Radial thrust factor.

TYPES OF VOLUTES – DOUBLE VOLUTE CASINGS

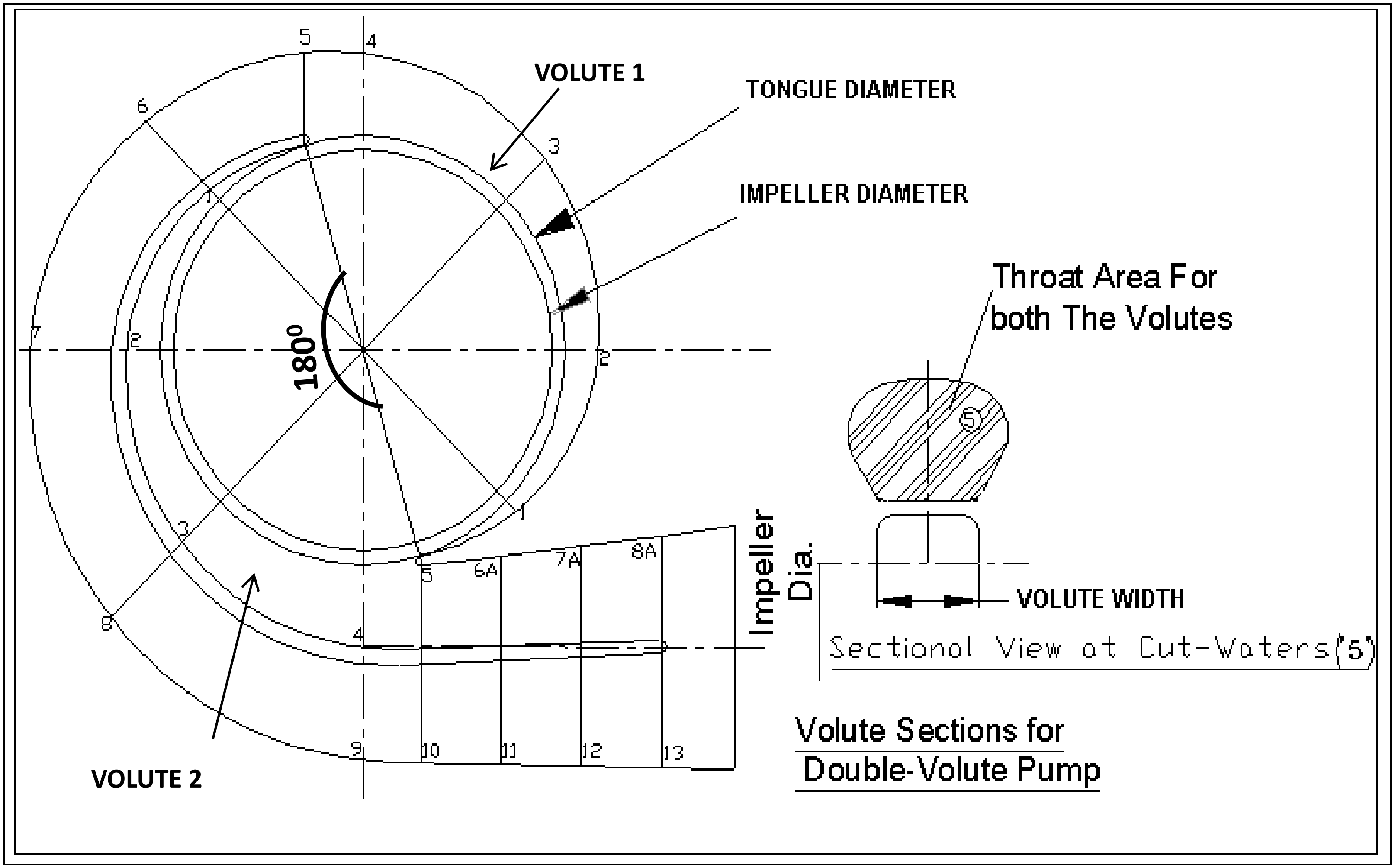
1. A DOUBLE VOLUTE CASING HAS TWO SINGLE VOLUTES 180° APART.
2. TOTAL THROAT AREA OF TWO VOLUTE IS SAME AS THE THROAT AREA OF A COMPARABLE SINGLE VOLUTE PUMP.
3. DOUBLE VOLUTE SIGNIFICANTLY REDUCES THE RADIAL LOAD PROBLEM OF THE SINGLE VOLUTE PUMP.
4. HYDRAULIC PERFORMANCE OF A DOUBLE VOLUTE PUMP IS NEARLY THE SAME AS THAT OF A SINGLE VOLUTE PUMP.
5. DOUBLE VOLUTE PUMP IS AROUND 1 - 1.5 POINT LESS EFFICIENT AT B.E.P BUT ABOUT TWO POINTS MORE EFFICIENT ON EITHER SIDE OF B.E.P .
6. DOUBLE VOLUTES ARE NOT USED FOR FLOWS BELOW 90 M^3/HR.(400 US GPM).



DOUBLE VOLUTE

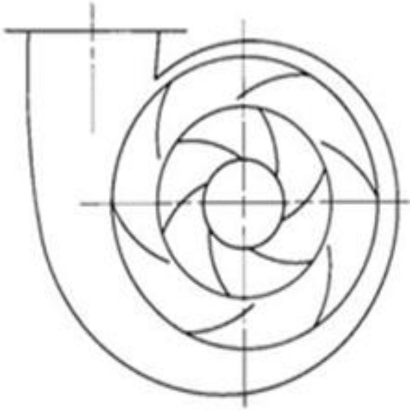
VOLUTE AS-CAST

VOLUTE SECTIONS FOR DOUBLE VOLUTE PUMP



SAME THROAT AREA FOR BOTH THE VOLUTES

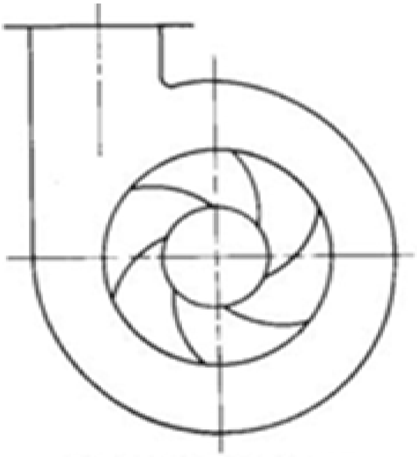
Volute with Diffuser vanes



An Impeller discharges into multiple divergent passes (normally two or more) with the outer casing functioning as a collector ,directing fluid in to the pump discharge or the next pump stage

Circular (concentric) casing

An Impeller discharges in to a circulator collector with a single discharge port.
A circular casing is often used where efficiency is not a concern
Casings are commonly fabricated and it may improve the efficiency of very low specific speed.
Application : Slurry pumps.



CASING THICKNESS CALCULATION USING ASME STANDARD

$$t = \frac{p*r}{f - .6p} + 3$$

Where, p= max working pressure= 12 bar= 174psi (for CSC range)

r = casing internal radius

f= permissible stress= .25× UTS ×.8

For, Grade 14(FG220) = .25 × 14 ×.8 × 2240= 6272 psi

Grade 17(FG 260)= 7616 psi

GGG50 = 13600 psi

Shaft deflection

Shaft deflection is the designed criterion that greatly influences pump performance due to its effect on the mechanical seal, internal clearance and bearings .

The radial loads acting on the rotating impellers are transmitted directly to the pump shaft. This forces will deflect the shaft where it is applied ,irrespective of the bearing configuration.

The shaft must be designed to accommodated this hydraulic radial load in conjunction with the additional radial load imposed due to the mass of the impellers and other rotating components . Under these condition the rotor must be stiff enough to limit the resulting deflection to within limits .

Standards

Overhung Impeller Pumps as per ASME B73.1 → Dynamic shaft deflection at the impeller centerline shall not exceed 0.125 mm in any where within the design region.

Overhung Impeller Pumps as per API standard 610 (10 th edition) → Maximum shaft deflection at primary seal faces is 0.05 mm.

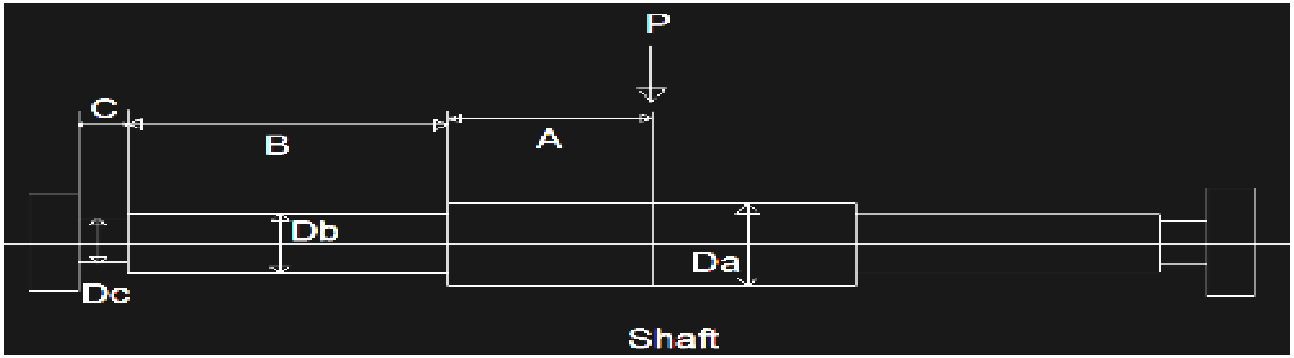
Overhung Impeller Pumps as per ISO 5199 → 0.05 MM at the face of the seal chamber

SHAFT DEFLECTION ACCORDING TO HYDRAULIC INSTITUTE

➢ ***Deflection of shaft :Between the bearing***

Deflection (Δ) = P/(6EIc)(C^3+ {(C+B)^3-C^3}/K2 + {(A+B+C)^3-(B+C)^3}/K3)

Δ is in mm.



P= *load at the impeller: (P)*

E(Modulus of elasticity) = 200 GPa =200×10^6 Kpa

A= length of shaft at section A

B= length of shaft at section B

C= length of shaft at section C

➢ ***<u>K2, K3 can be calculated as below,</u>***

K_2= Ib/Ic
K_3=Ia/Ic

➢ ***<u>Determination of moment of inertia of each section:</u>***

Ia= moment of inertia = $\pi\times Da^4/64$

Ib= moment of inertia = $\pi\times Db^4/64$

Ic= moment of inertia = $\pi\times Dc^4/64$

Da= Dia of the shaft at section A

Db= Dia of the shaft at section B

Dc= Dia of the shaft at section C

➢ ***<u>Determination of load at the impeller: (P)</u>***

Wi = Wgt of the impeller

Ws = Wgt of the shaft

W= static load of rotor = Wi + Ws

P (load at the impeller) = W+R

- ***Calculate Radial Thrust:***

$$R = Kr \times H \times \rho \times g \times B_2 \times D_2$$

Where,

Kr = thrust factor which varies with rate of flow and specific speed

H = Developed head per stage in m

ρ= density of the liquid in kg/m^3

Gravitational constant = 9.81 m/s2

D2= Impeller dia in m

B2= Impeller width at discharge including shrouds in m

Note: = *H* ×S/ 2.31 in US units

- ***<u>Calculate specific speed Ns:</u>***

Ns (US units) = N × (Q×4.403)^.5 / (H×3.28)^.75

where, N in rpm,

Q in m^3/hr

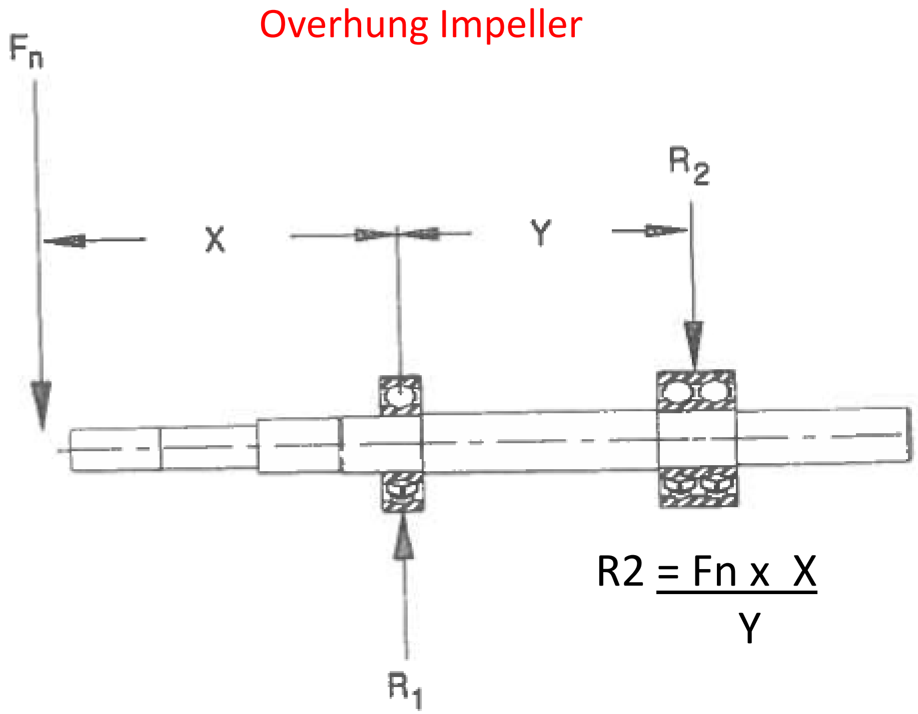
H in m

➢ ***Determine Kr:***

The variation of Kr with specific speed is given in the table:

Duty 50%	Ns(Us)	Ns(metric)	Kr
Double volute	500	600	0.02
	1000	1200	0.025
	1500	1250	0.035
	2000	2300	0.048
	3500	4000	0.05

Bearing load calculation – Ref- Hydraulic Institute

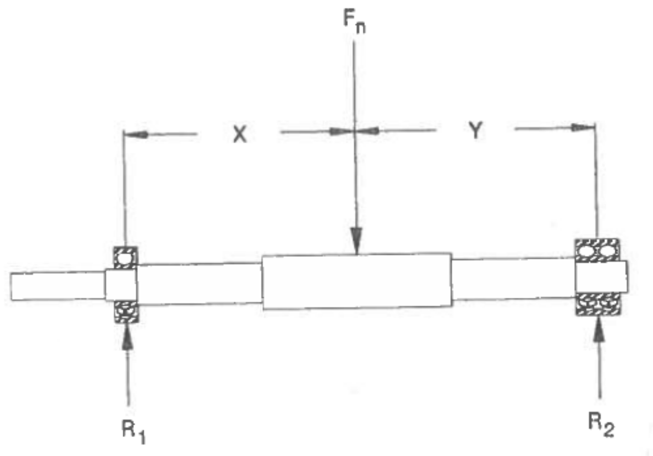


$$R1 = \frac{Fn\ X\ (X + Y)}{Y}$$

$$R2 = \frac{Fn\ x\ \ X}{Y}$$

Where,
R1 – Inboard reaction load , R2 – Outboard bearing reaction load
Fn = Fr + W ; Where
Fn – Net force typically applied at the impeller center line ,
Fr –Radial hydraulic thrust applied at impeller center line.
W-Impeller weight
X- Distance from applied load to the center of the inboard bearing
Y= Distance between in board and out board bearing

Bearing load – Between the bearing



$$R1 = \frac{Fn \times Y}{X+Y}$$

$$R2 = \frac{Fn \times X}{X + Y}$$

Where,
R1 – Inboard reaction load , R2 – Outboard bearing reaction load
Fn = Fr + W ; Where
Fn – Net force typically applied at the impeller center line ,
Fr –Radial hydraulic thrust applied at impeller center line.
W-Impeller weight
X- Distance from applied load to the center of the inboard bearing
Y= Distance between in board and out board bearing

HYDRAULIC THURSTS GENERATED IN A CENTRIFUGAL PUMP

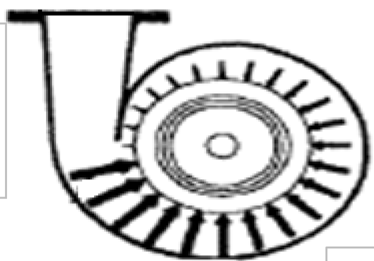
- RADIAL THRUST
- AXIAL THRUST

RADIAL THRUST

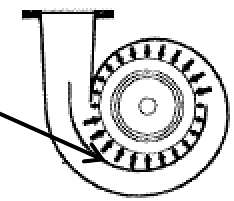
- SUMMATION OF UNBALANCED HYDRAULIC FORCES ACTING RADIALLY. DUE TO UNEQUAL VELOCITY OF THE FLUID FLOWING THROUGH THE CASING AT PART FLOW, A NON-UNIFORM PRESSURE DISTRIBUTION EXISTS OVER THE CIRCUMFERENCE OF THE IMPELLER.

RADIAL THRUST IS AN IMPORTANT PARAMETER WHEN DESIGNING PUMP'S MECHANICAL ELEMENTS LIKE SHAFT AND BEARINGS.

SINGLE VOLUTE CASING



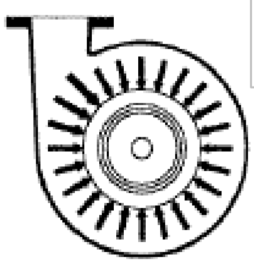
SECOND VOLUTE

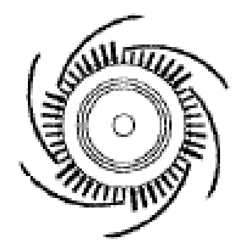


DOUBLE VOLUTE CASING

STATIC PRESSURE DISTRIBUTION OVER THE IMPELLER OUTLET FOR DIFFERENT CASING GEOMETRIES AT PART FLOW REGIME

CONCENTRIC CASING





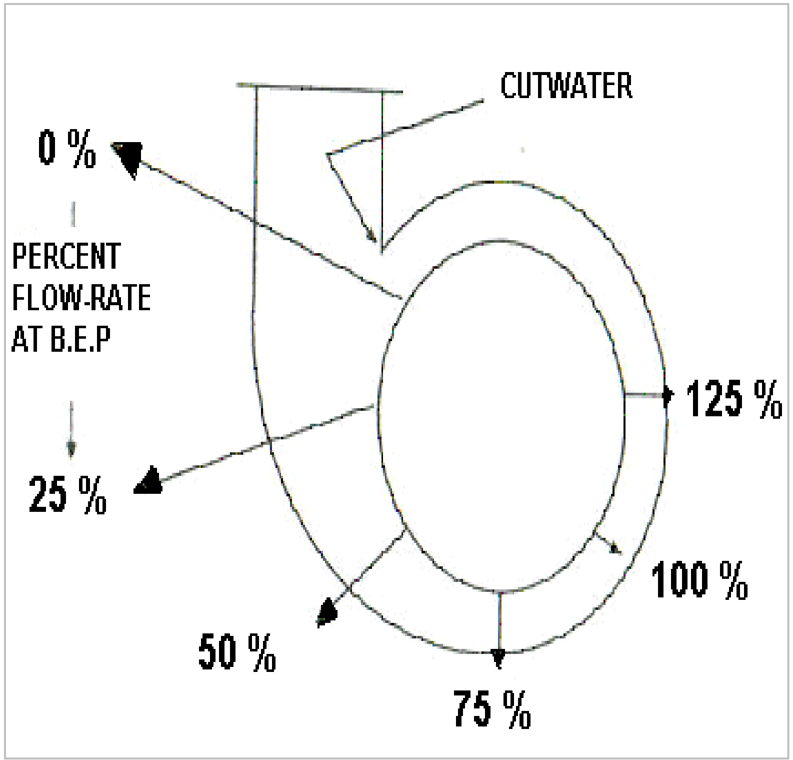
VANED DIFFUSER

RADIAL THRUST IS A FUNCTION OF TOTAL HEAD OF THE PUMP & WIDTH & DIAMETER OF THE IMPELLER

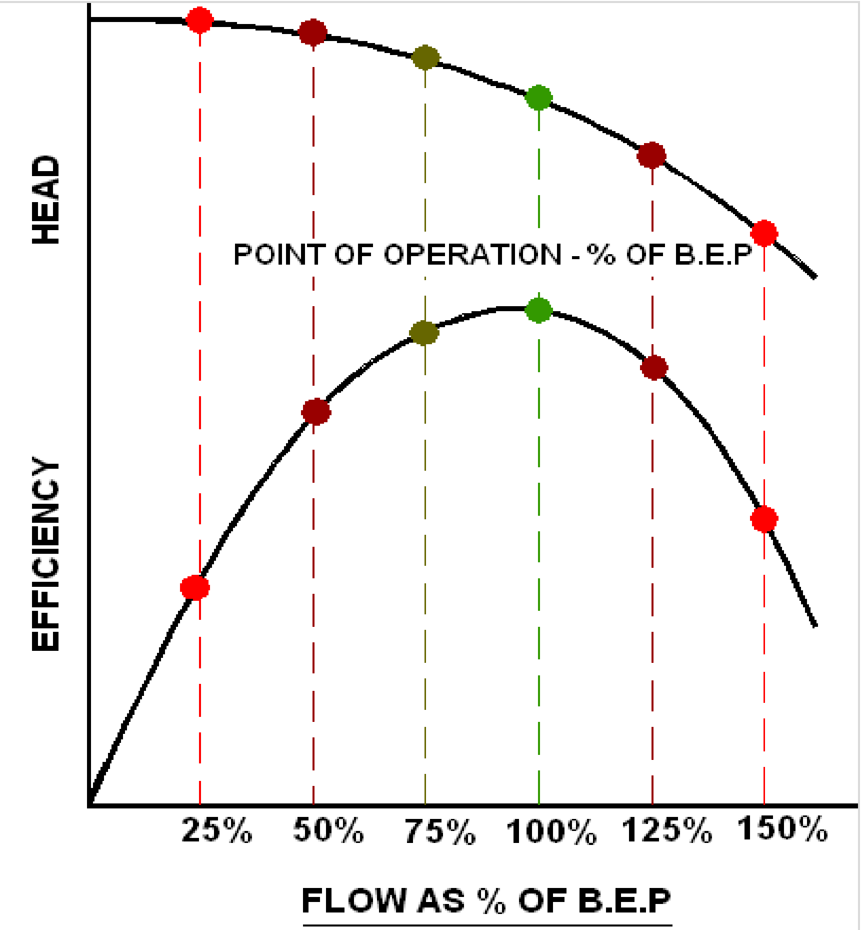
RADIAL THRUST

$$P = K \times H \times D_2 \times b_2 \times S.G / 2.31$$

P = RADIAL THRUST IN POUNDS
K = THRUST FACTOR
H = DEVELOPED HEAD/STAGE IN FT
D_2 = IMPELLER DIA, IN INCHES
b_2 = IMPELLER WIDTH IN INCHES(INCLUDING SHROUDS)



RADIAL REACTION VECTOR REPRESENTED BY THE ARROWS AT DIFFERENT FLOW CONDITIONS FOR A SINGLE VOLUTE CASING



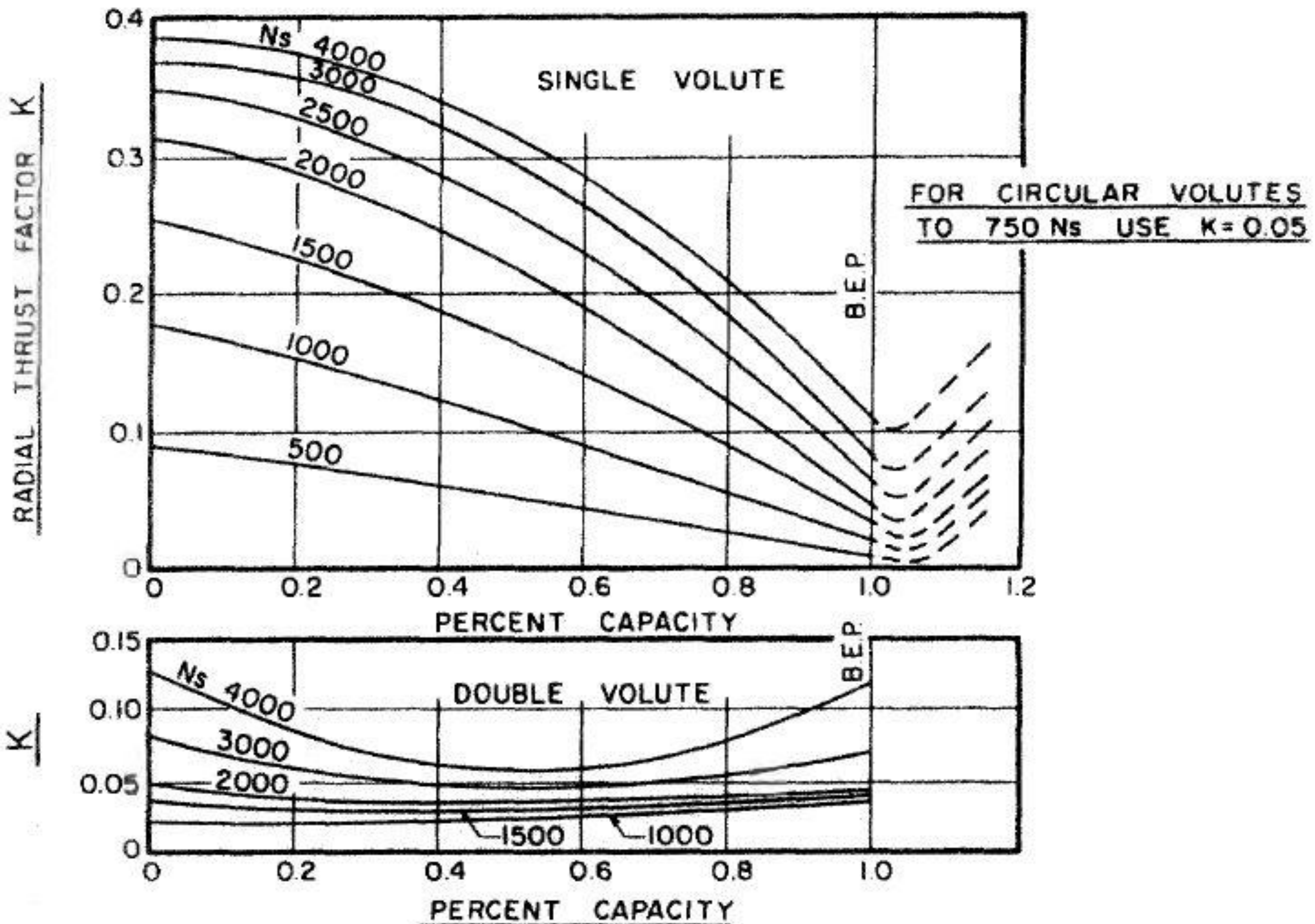
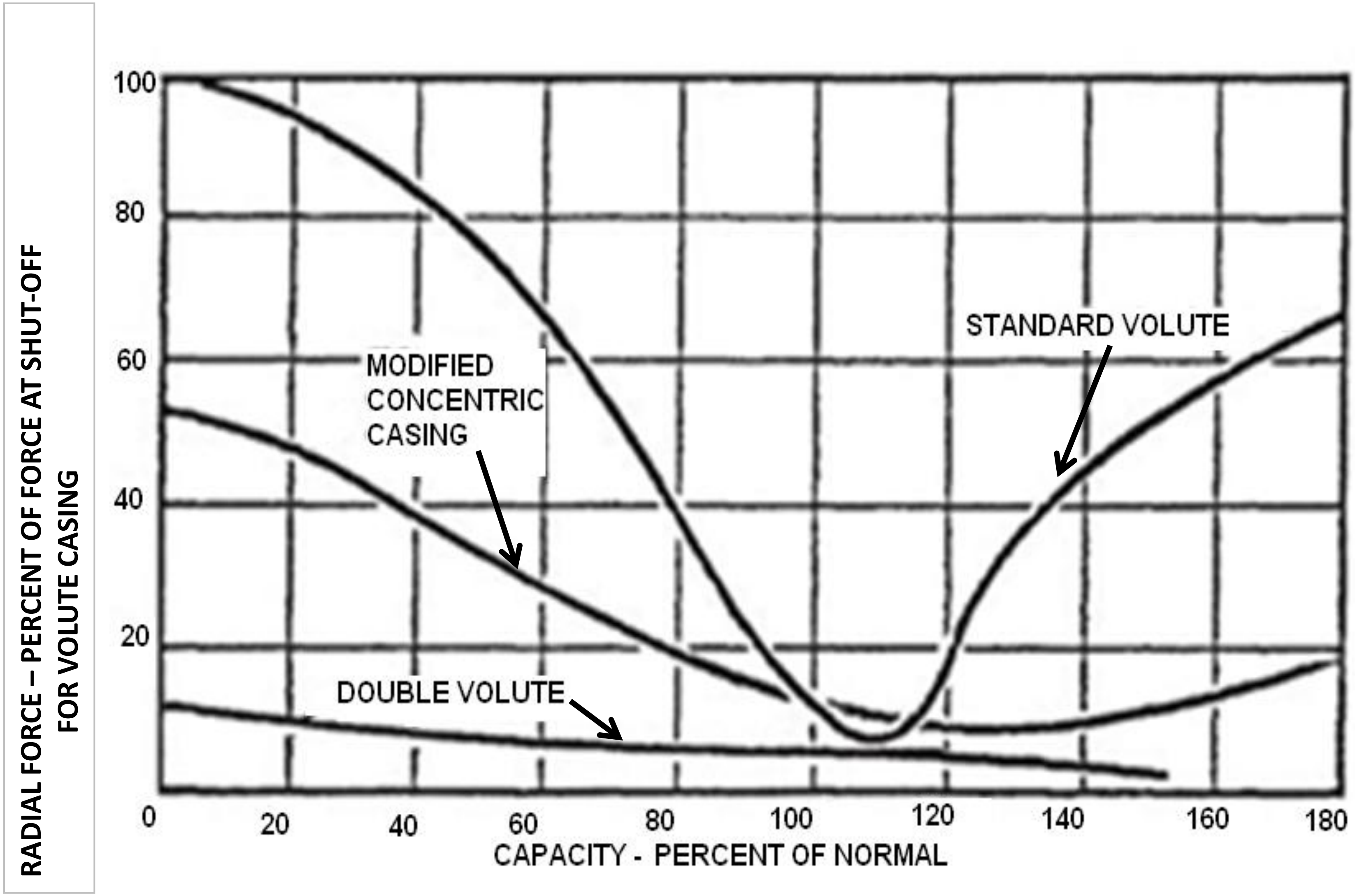


Figure 5-2. Radial thrust factor.

RADIAL FORCE FOR VARIOUS TYPES OF VOLUTES

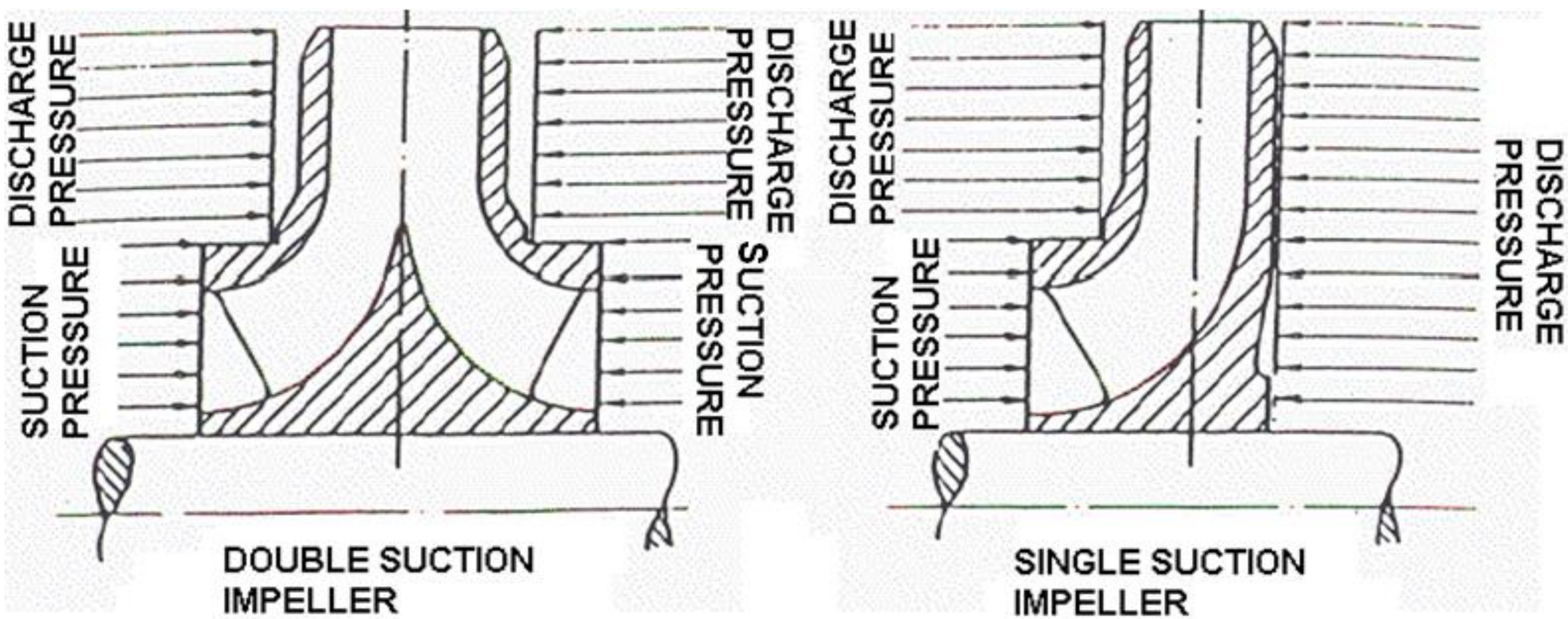


AXIAL THRUST

- SUMMATION OF UNBALANCED HYDRAULIC FORCES ACTING AXIALLY ON THE IMPELLER.

SEVERITY OF AXIAL THURST DEPENDS ON THE TOTAL HEAD, SUCTION PRESSURE & MECHANICAL CONFIGURATION OF IMPELLER.

AXIAL PRESSURE ACTING ON THE IMPELLER SHROUDS TO PRODUCE AXIAL THRUST

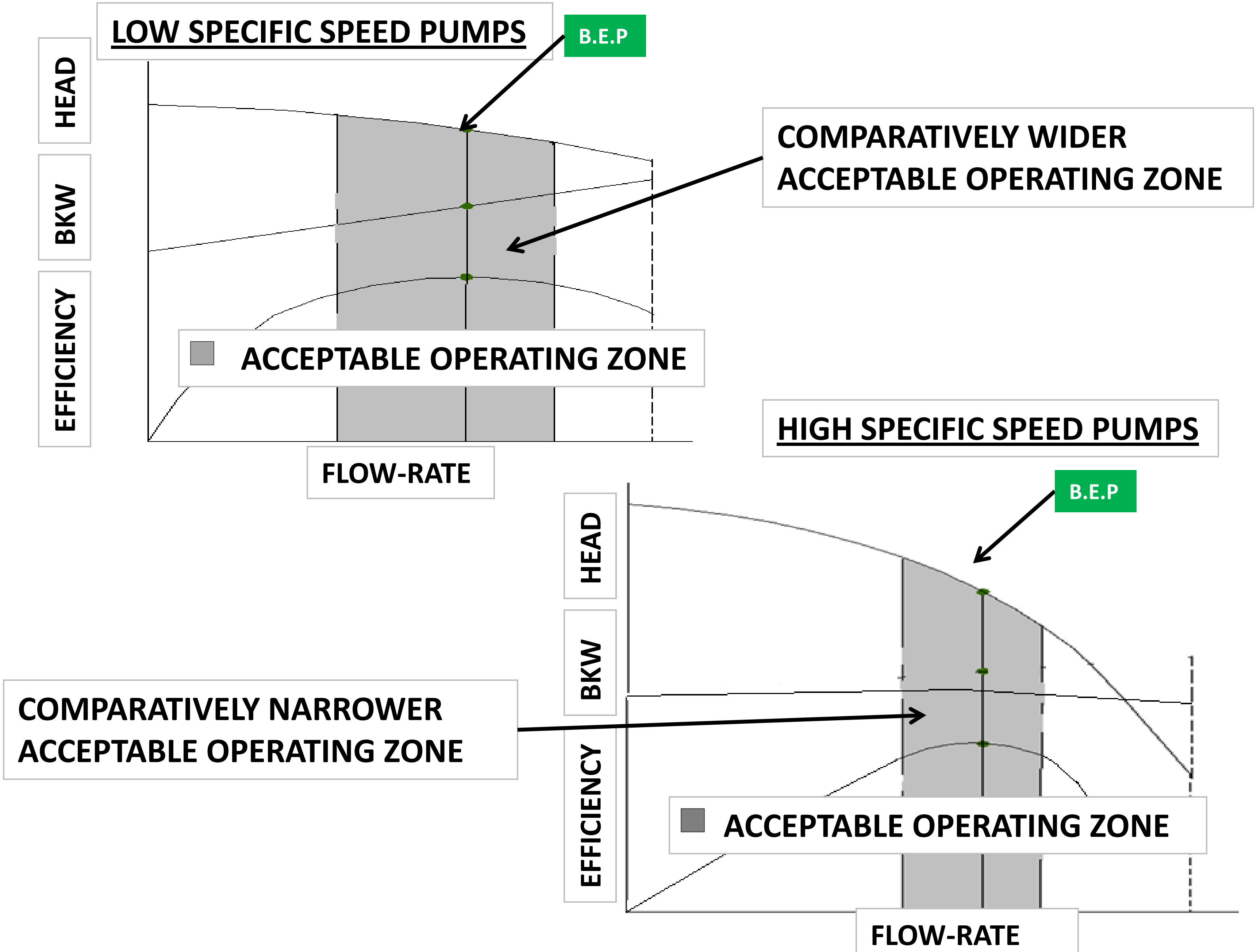


SPECIFIC SPEED OF A CENTRIFUGAL PUMP

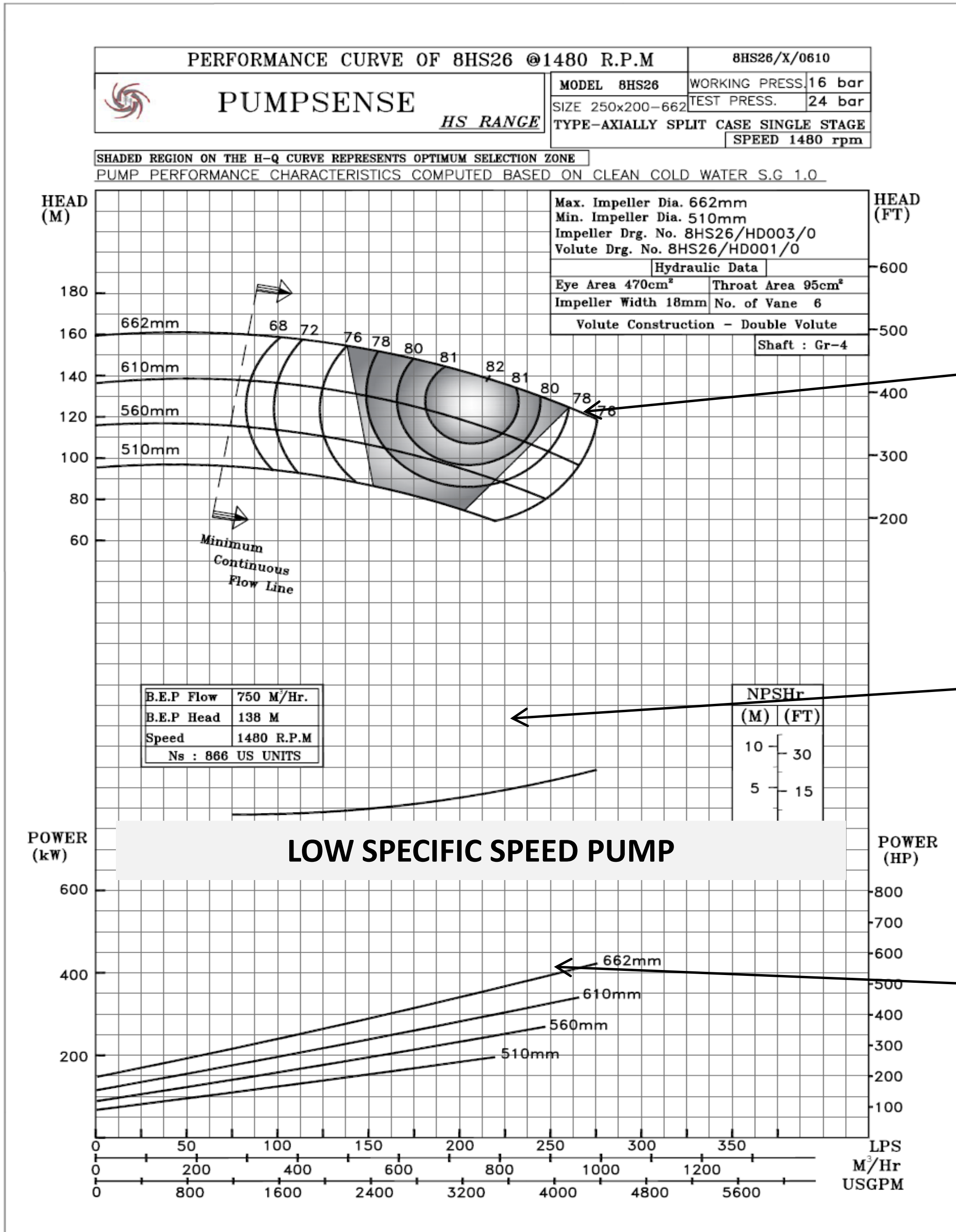
IT'S A **DESIGN INDEX** THAT DETERMINES THE IMPELLER TYPE AND GEOMETRIC SIMILARITY OF PUMPS.

$Ns = N \times \sqrt{Q} \; / \; (H)^{0.75}$ WHERE , ➢ Ns = SPECIFIC SPEED IN METRIC UNITS. ➢ Q = FLOW IN M^3/HR. AT B.E.P. ➢ N = ROTATIVE SPEED IN R.P.M . ➢ H = HEAD DEVELOPED IN M. AT B.E.P.	**$Ns = N \times \sqrt{Q} \; / \; (H)^{0.75}$** WHERE , ➢ Ns = SPECIFIC SPEED IN US CUSTOMARY UNITS. ➢ Q = FLOW IN US GPM AT B.E.P. ➢ N = ROTATIVE SPEED IN R.P.M . ➢ H = HEAD DEVELOPED IN FT. AT B.E.P.

EFFECT OF SPECIFIC SPEED ON ACCECTABLE OPERATING ZONE OF A PUMP



SPECIFIC SPEED & ITS EFFECT ON PUMP PERFORMANCE CHARACTERISTICS



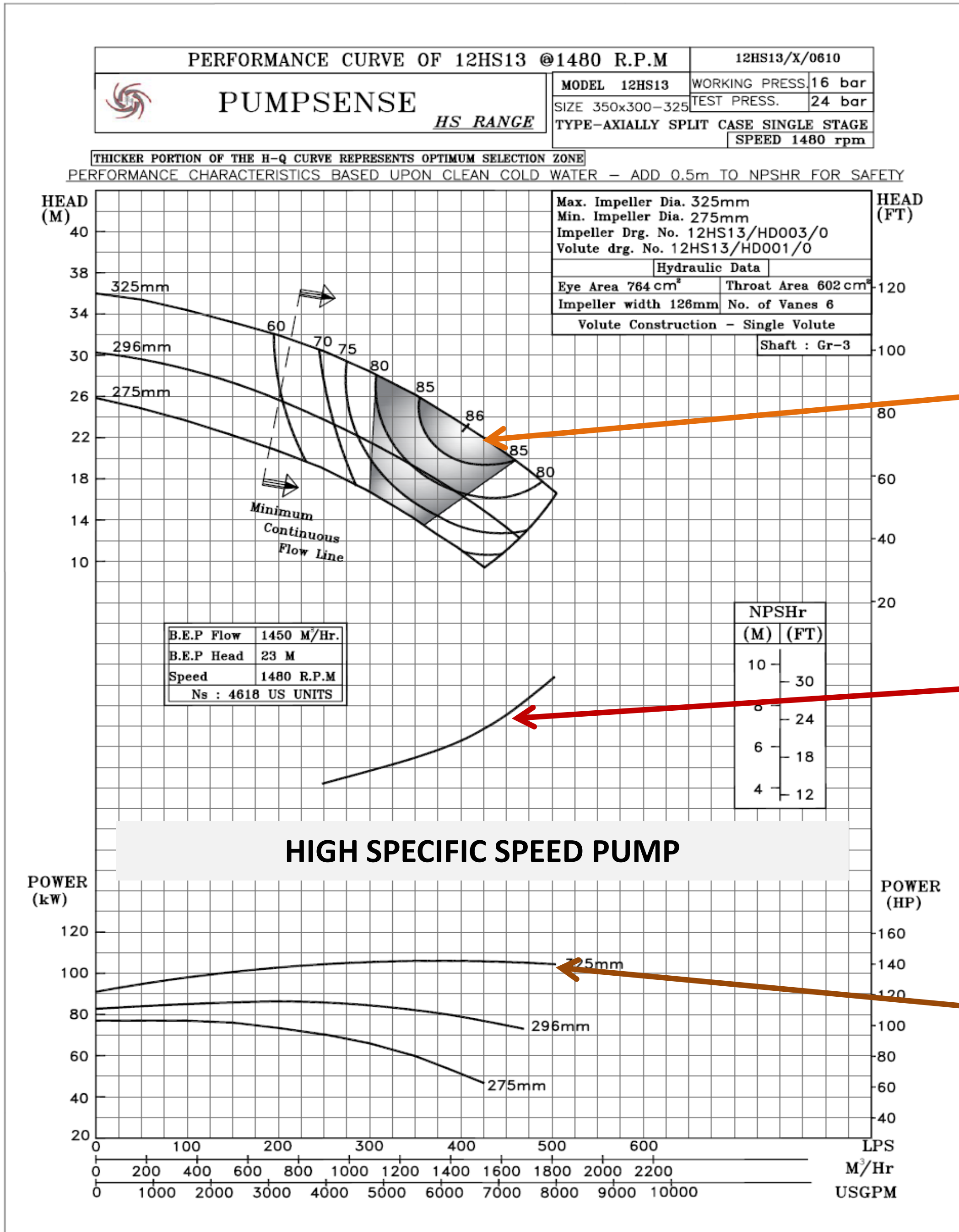
LOW SPECIFIC SPEED PUMP

COMPARATIVELY FLAT H-Q CURVE

COMPARATIVELY LOW NPSH REQUIREMENT

RISING POWER CURVE

SPECIFIC SPEED & ITS EFFECT ON PUMP PERFORMANCE CHARACTERISTICS



COMPARATIVELY STEEP H-Q CURVE

COMPARATIVELY HIGH NPSH REQUIREMENT

NON-OVERLOADING POWER CURVE

Allowable Operating Region & Preferred Operating Region

Reference Source ANSI HI 9.6.3-1997

Preferred Operating Region - POR

The flow remains well controlled within a range of rates of flow designated as the Preferred Operating Region (POR). Within this region the service life of the pump will not be significantly affected by hydraulic loads, vibration or flow separation.

Specific Speed		**POR**
Metric	US Units	
< 5200	< 4500	Between 70% & 120% of BEP
> 5200	> 4500	Between 80% & 115% of BEP

SUCTION SPECIFIC SPEED

Suction specific speed is an indicator of the net positive suction head require for 3% drop in head at a given flow rate and rotation speed.

$N_{ss} = N \times \sqrt{Q} \,/\, (NPSHr)^{0.75}$ WHERE , ➢ **Nss** = Suction specific speed in metric units ➢ **Q** = Flow in m^3/hr at b.e.p (use half of the total flow for double suction pumps) ➢ **N** = rotative speed in r.p.m ➢ **NPSHr** = Net +ve suction head required in m (established by 3% head drop test)	**$N_{ss} = N \times \sqrt{Q} \,/\, (NPSHr)^{0.75}$** where, ➢ **Nss** = Suction specific speed in us customary units. ➢ **Q** = Flow in us gpm at b.e.p (use half of the total flow for double suction pumps) ➢ **N** = Rotative speed in r.p.m ➢ **NPSHr** = Net +ve suction head required in ft (established by 3% head drop test)

SPEED LIMITATION AND SUCTION SPECIFIC SPEED

Increased pump speed without proper suction conditions

can lead to →

- Abnormal pump wear
- Failure due to excessive vibration
- Noise
- Cavitations damage

Suction specific speed has been found to be a valuable criterion in determining the maximum speed

Hydraulic institute uses a value of 10,000 metric units (8500 us units) as a practical value for determining the maximum operating speed.

In metric units,

$$n = 10{,}000 \times npsha^{0.75} / q^{0.5}$$

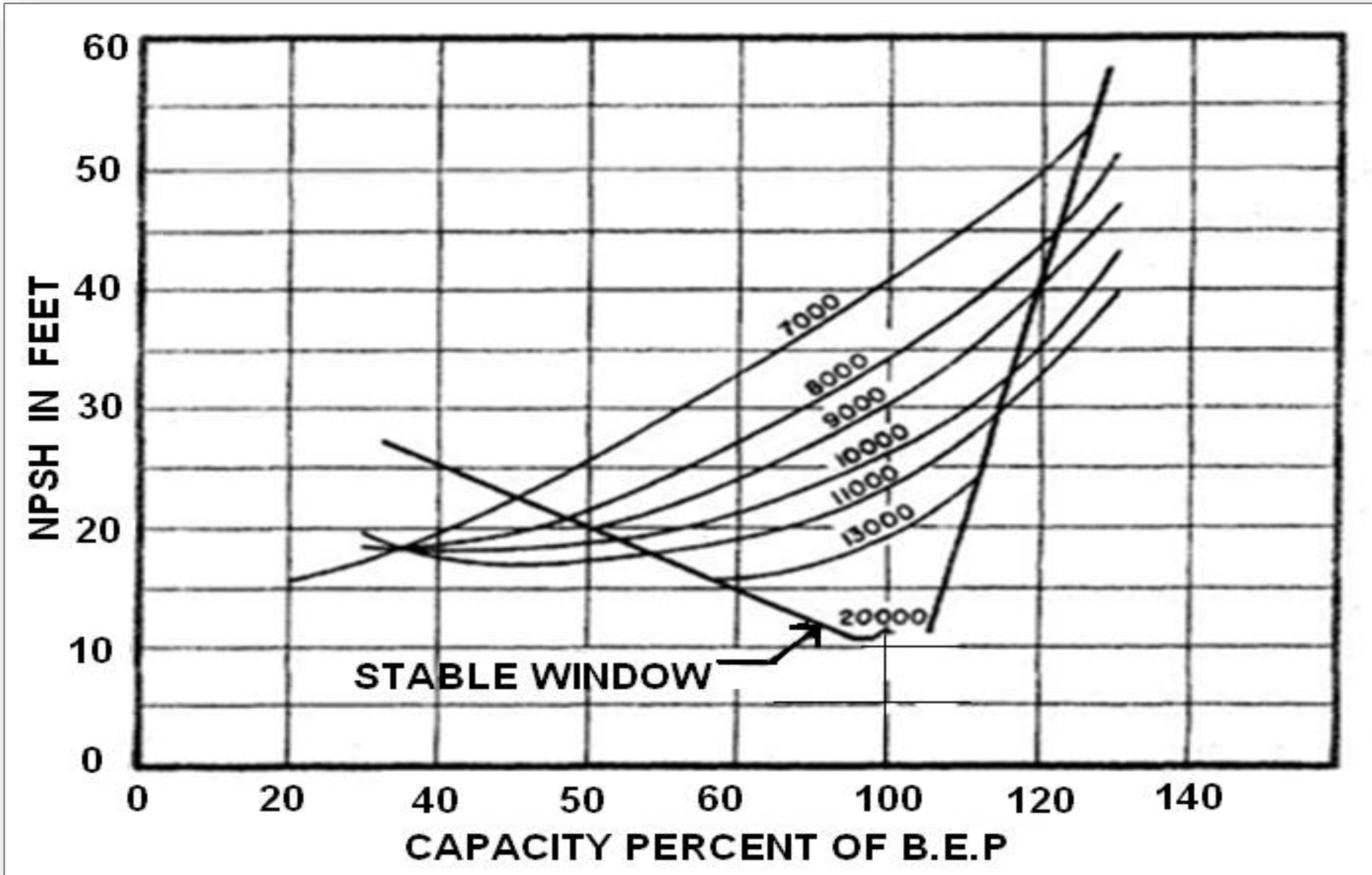
where , n = max. speed(r.p.m).
npsha = npsh available in m.
q = flow in m^3/hr. (take half of the flow for double suction pump).

In us customary units,

$$n = 8{,}500 \times npsha^{0.75} / q^{0.5}$$

where , n = max. speed(r.p.m).
npsha = npsh available in ft.
q = flow in us g.p.m. (take half of the flow for double suction pump).

SAFE OPERATING WINDOW Vs SUCTION SPECIFIC SPEED



Ttest of a 4 inch pump with different Nss impellers.
bep & impeller profiles are identical,only eye geometry is different for each Nss

NPSHr & NPSHa

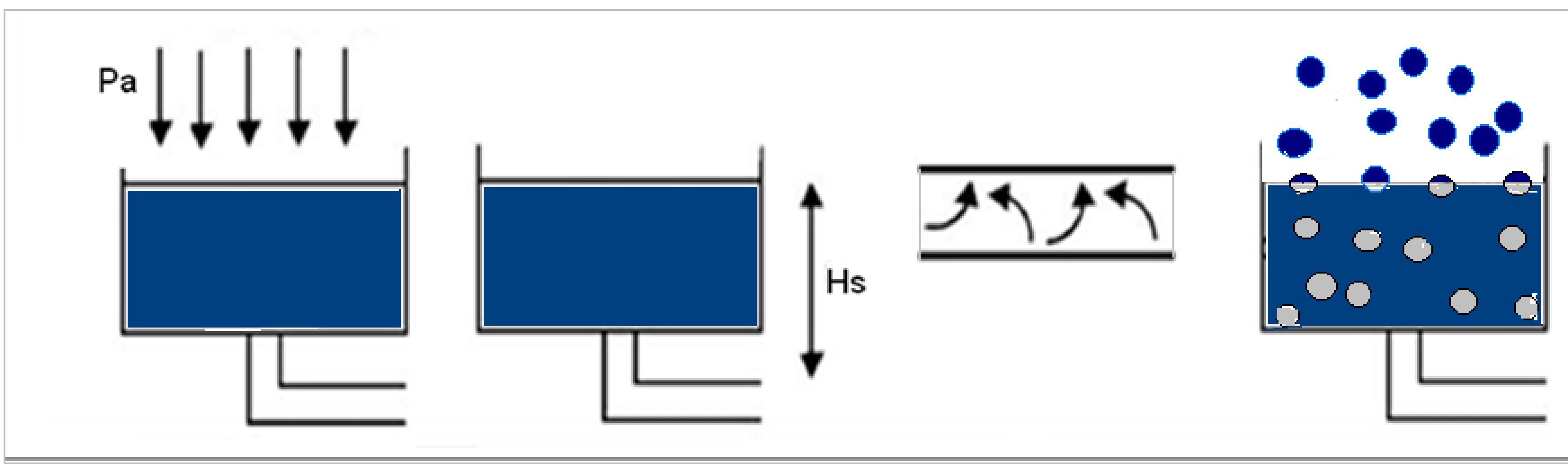
NPSHr IS A CHARACTERISTIC OF YOUR PUMP.	**NPSHa IS A CHARACTERISTIC OF YOUR SYSTEM**
THIS IS A **FUNCTION OF PUMP SUCTION DESIGN**. IT VARIES WITH THE SPEED & CAPACITY FOR A PARTICULAR PUMP.	THIS IS A **FUNCTION OF SYSTEM CONFIGURATION** ON THE SUCTION SIDE OF THE PUMP.
THIS IS THE +VE HEAD IN M ABSOLUTE REQUIRED AT PUMP SUCTION **TO OVERCOME PUMP INTERNAL LOSSES** — LOSSES DUE TO TURBULANCE (GENERATED AS THE LIQUID STRIKES THE IMPELLER AT IMPELLER INLET), LOSSES IN THE SUCTION PASSAGE & VANE INLET PASSAGES TO MAITAIN THE PUMPING FLUID IN LIQUID STATE.	IT IS THE **AVAILABLE TOTAL SUCTION HEAD** IN METRES ABSOLUTE DETERMINED AT THE INLET NOZZLE OF THE PUMP & CORRECTED TO THE PUMP DATUM LESS THE VAPOUR PRESSURE HEAD OF THE LIQUID IN METRES ABSOLUTE AT THE PUMPING TEMPERATURE.

FOR CAVITATION-FREE SAFE OPERATION, YOU'VE TO KEEP, NPSHa > NPSHr

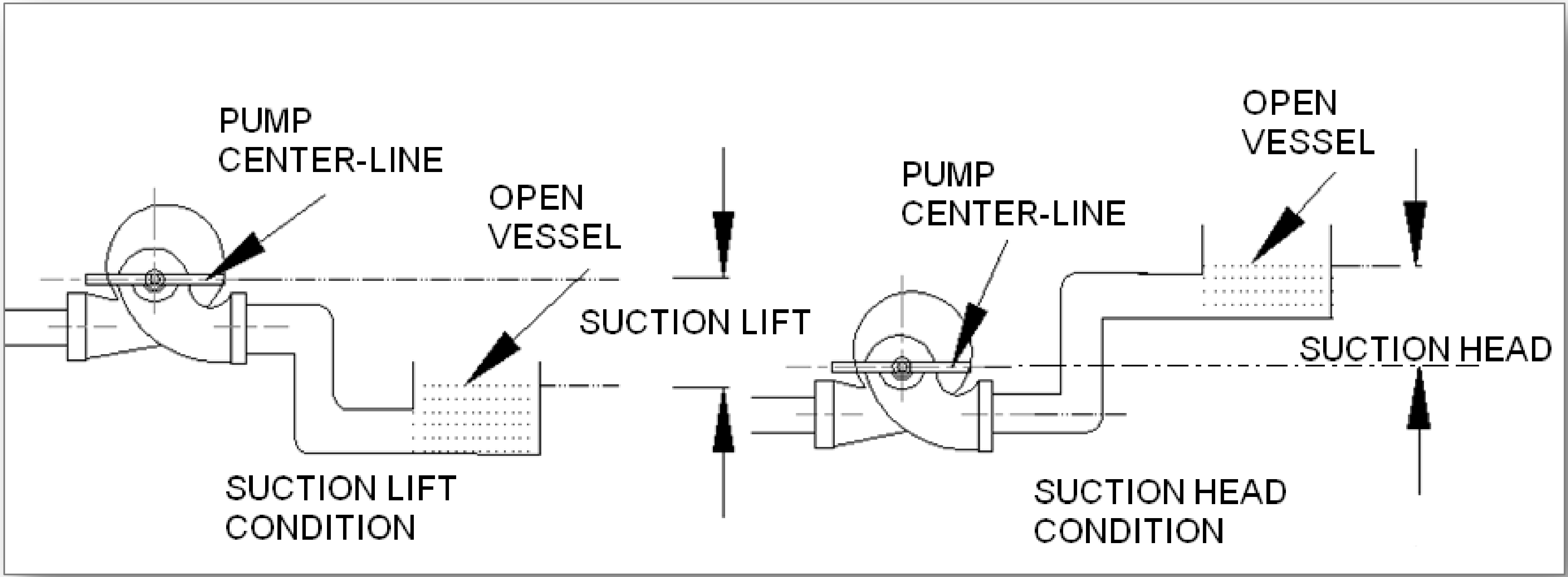
CALCULATION OF AVAILABLE NPSH (NPSHa)

- CHARACTERISTIC OF THE PROCESS SUCTION SYSTEM.
- AN ANALYSIS OF TOTAL ENERGY ON THE SUCTION SIDE OF A PUMP TO DETERMINE WHETHER THE LIQUID WILL VAPOURIZE AT A LOW PRESSURE POINT IN THE PUMP.

NPSHA = PRESSURE ACTING ON SURFACE(Pa) + STATIC SUCTION HEAD (Hs) — PRESSURE DROP(FRICTIONAL HEAD LOSS)(Hf) — VAPOUR PRESSURE(Hvp)



CALCULATION OF $NPSH_A$ FOR SYSTEMS WITH SUCTION HEAD & SUCTION LIFT



NPSHa (M) = ATMOSPHERIC PRESSURE(M) – SUCTION LIFT(M) – FRICTIONAL HEAD LOSS(M) - V.P (M)	**NPSHa (M) = ATMOSPHERIC PRESSURE(M) + SUCTION HEAD(M) – FRICTIONAL HEAD LOSS(M) - V. P (M)**

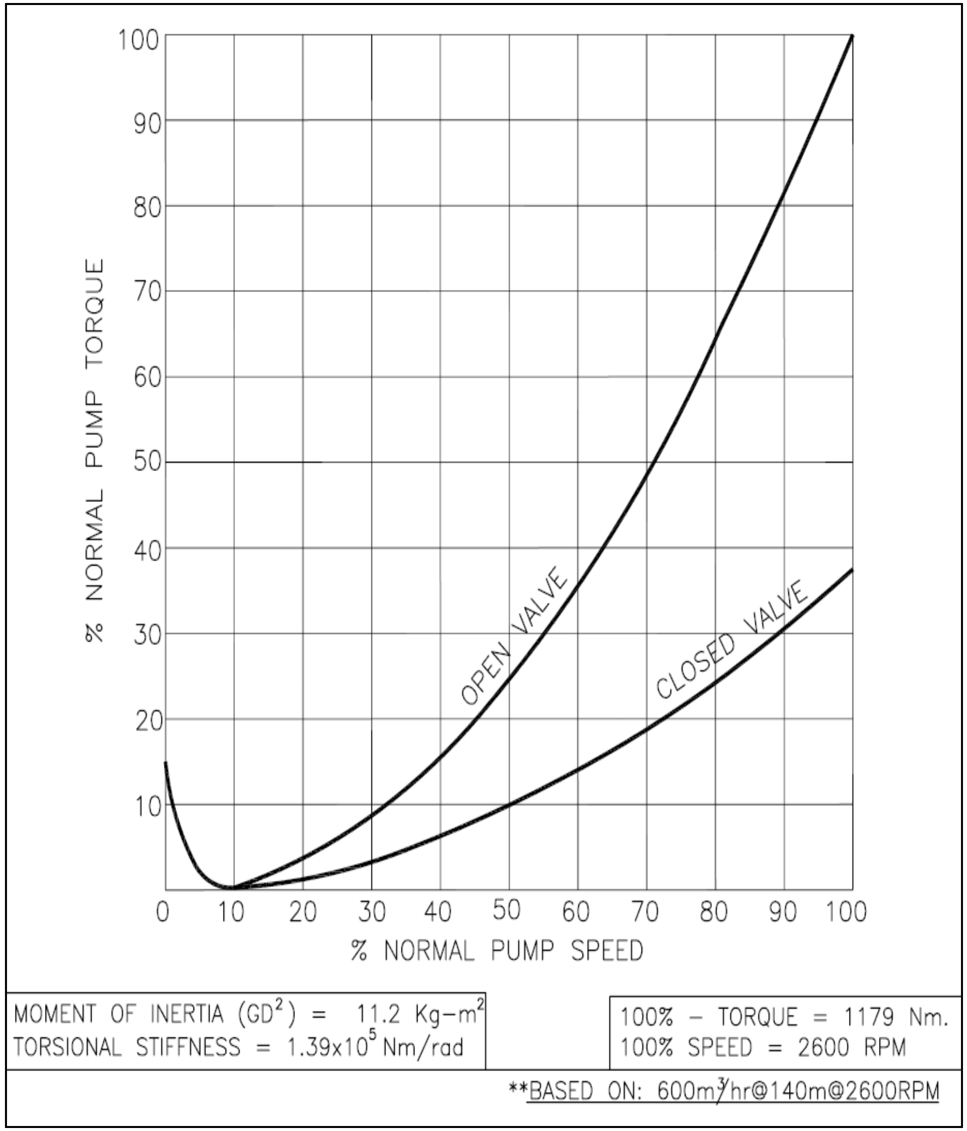
REASONS FORSAFETY MARGIN FOR MOTOR SELECTION

- **IF A MOTOR IS CONSTANTLY OVERLOADED PRODUCING A POWER MORE THAN THE RATED, IT WILL DRAW MORE CURRENT AND THIS WILL INCREASE THE POWER LOSS (I^2R). HIGHER THE LOSS, HIGHER WOULD BE THE HEAT GENERATED LEADING TO RAPID DAMAGE TO THE MOTOR INSULATION.**

$$\text{BKW (for pump)} = \frac{\text{(Capacity in cum/hr X Head in meters X sp. Gravity)}}{\text{367 x eff. of the pump at duty}}$$

A GUIDE FOR SELECTING SAFETY MARGIN – ISO 5199

MOTOR RATING	MARGIN OF SAFETY (% OF MOTOR RATING)
1 kW TO 100 kW	135% TO 110%
ABOVE 100 kW	110%



TYPICAL SPEED TOQUE CURVE WITH TVA DATA

CLASSIFICATION OF LOSSES

LOSSES IN A CENTRIFUGAL PUMP ARE CLASSIFIED INTO FIVE TYPES:

- **MECHANICAL LOSSES**
- **IMPELLER LOSSES**
- **DISK FRICTION LOSSES**
- **LEAKAGE LOSSES**
- **CASING HYDRAULIC LOSSES**

There are number of mechanical ,hydraulics losses in impeller and pump casing, this will affect the pump performance is lower than predicted by the Euler pump equation.

MECHANICAL LOSSES

EXCLUSIVELY POWER LOSS TAKES PLACE PRIMARILY IN **BEARINGS**, **MECHANICAL SEALS** OR **GLAND PACKING**

LOSSES IN CENTRIFUGAL PUMP

CONTD..

Pump Losses

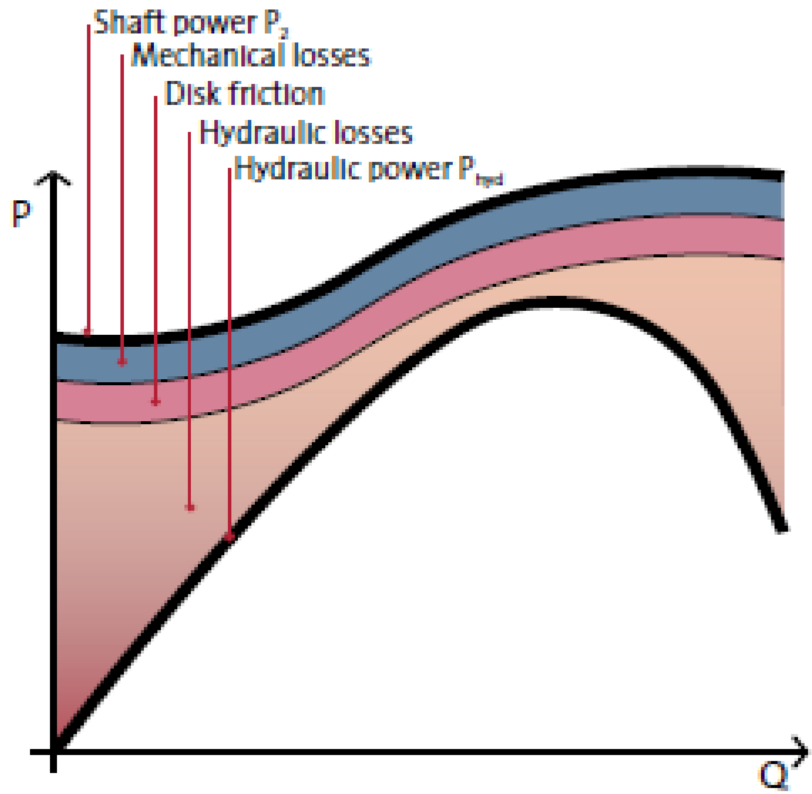
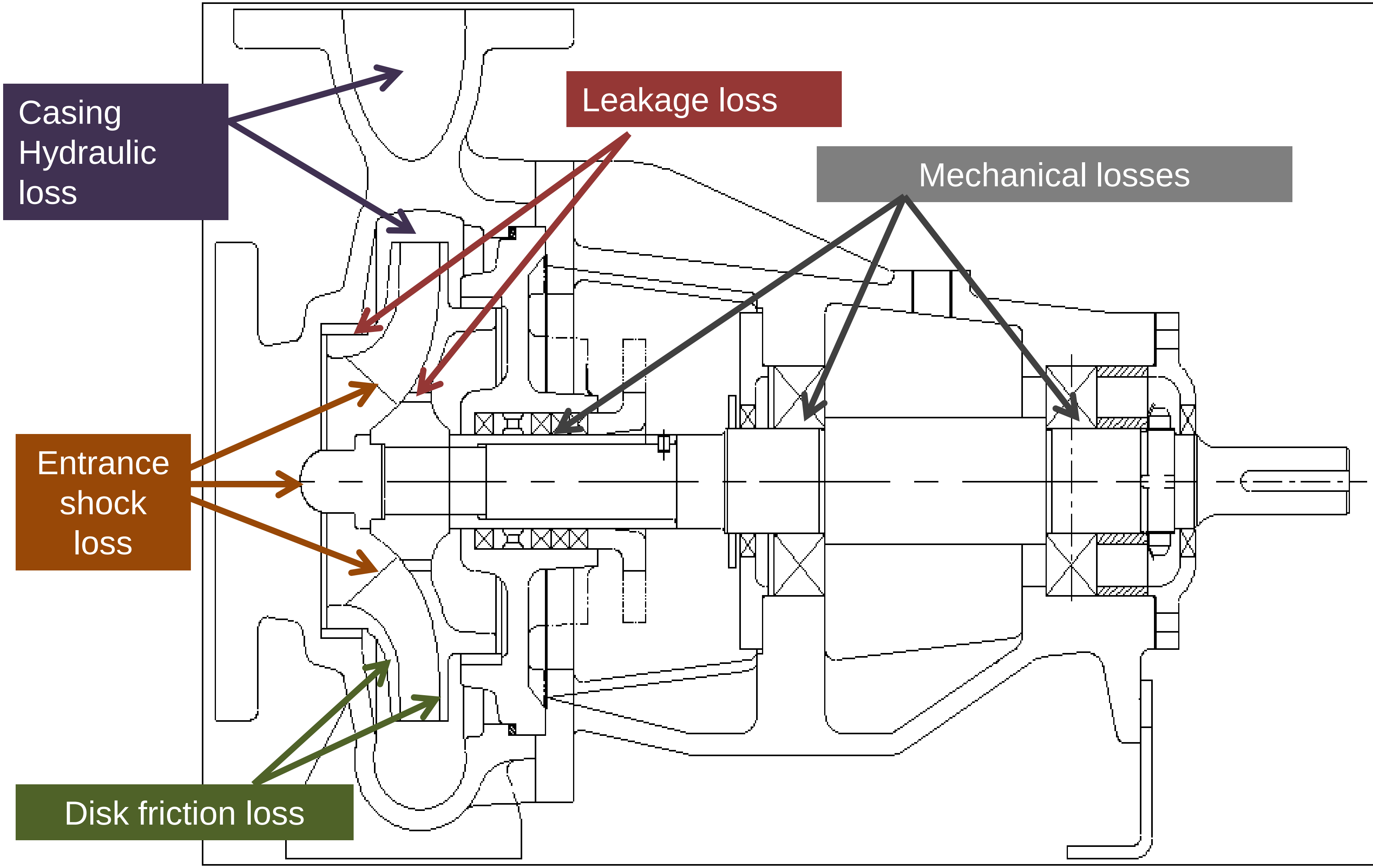
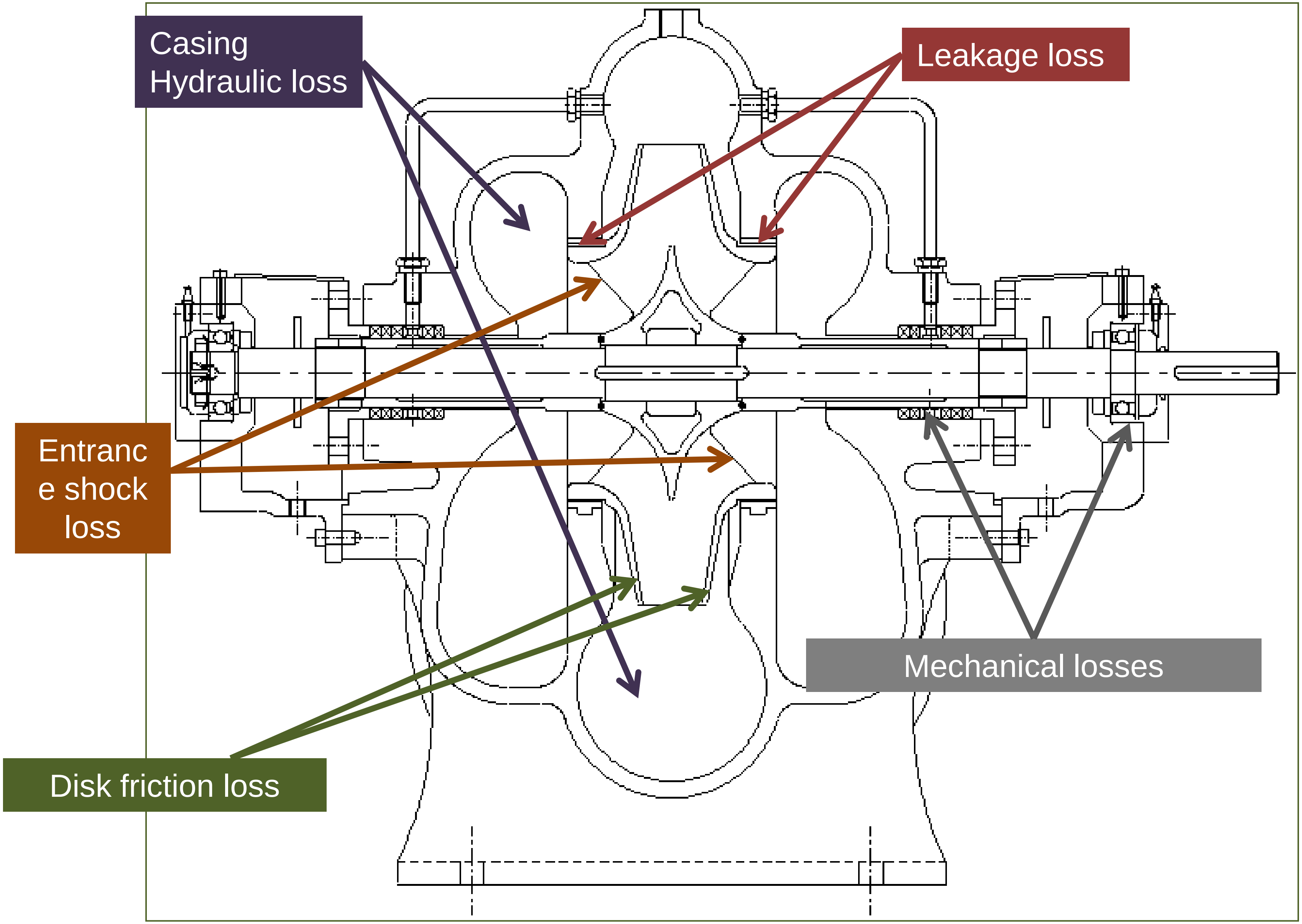


Figure 5.2: Increase in power consumption due to losses.

LOCATIONS OF VARIOUS LOSSES IN A CENTRIFUGAL PUMP



LOCATIONS OF VARIOUS LOSSES IN A CENTRIFUGAL PUMP



Mechanical losses

The pump shaft consists of bearings, shaft seals ,gear depending on pump type. These components all causes mechanical friction losses.

$$P_{\text{losses, mechanical}} = P_{\text{losses, bearing}} + P_{\text{loss, shaft seal}}$$

$P_{\text{losses, bearing}}$ - Power loss in bearing (W)

$P_{\text{loss, shaft seal}}$ - Power loss in shaft (W)

Hydraulic Losses

Hydraulic losses arise on the fluid path through the pump. The losses occur because of friction or because the fluid must change direction and velocity on its path through the pump

IMPELLER HYDRAULIC LOSSES

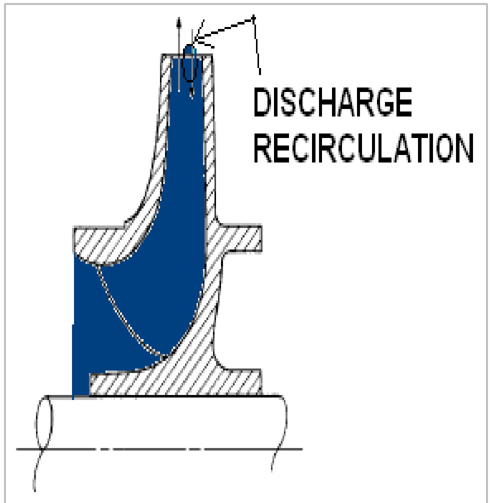
I) **Shock losses** at **inlet to the impeller**

ii) **Shock losses** **leaving the impeller**

iii) **Losses** during **conversion of mechanical energy to kinetic energy**

CASING HYDRAULIC LOSSES

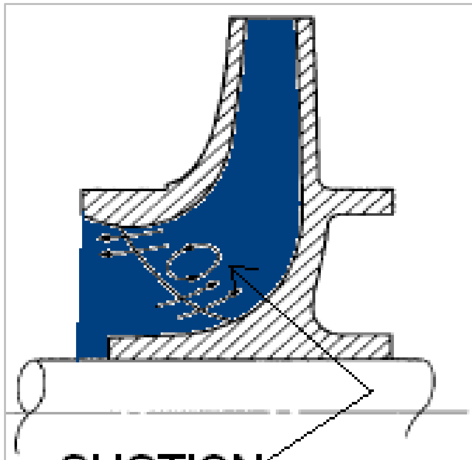
I) **RECIRCULATION** LOSSES
II) LOSSES DURING **CONVERSION OF K.E TO P.E**
III) LOSSES DUE TO **SKIN FRICTION IN CASING**

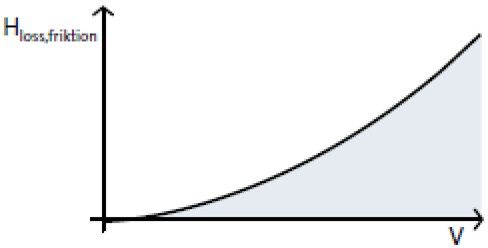


GENERAL EXPRESSION FOR DEFINING FRICTIONAL LOSSES:

$$H_f = f \times v^2/2g \times L/D_H$$

WHERE, H_f = FRICTIONAL HEAD LOSS
f = FRICTION CO-EFFICIENT
v = FLUID VELOCITY
L = PASSAGE LENGTH
D_H = HYDRAULIC DIA. OF THE PASSAGE





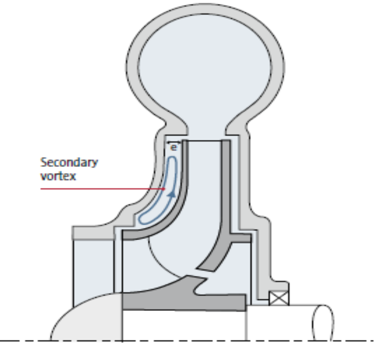
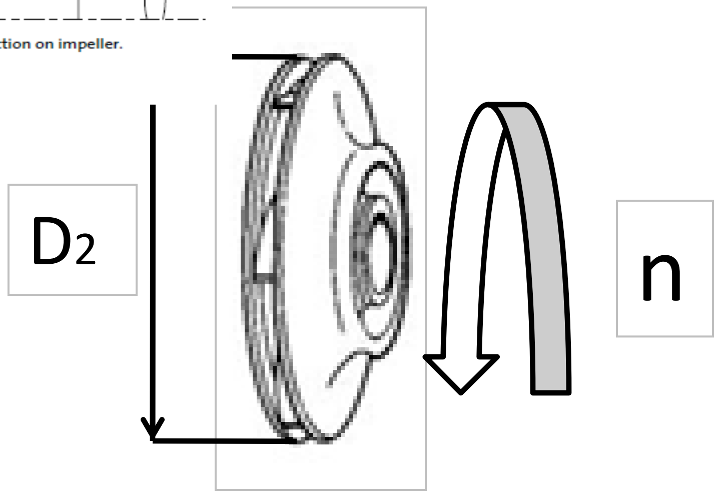


Figure 5.14: Disk friction on impeller.

DISK FRICTION LOSSES

FRICTIONAL LOSSES AT THE IMPELLER SHROUDS



General expression for disk friction power consumption:

$$P_D = k \times n^3 \times D_2^5$$

WHERE, P_D = POWER ABSORBED BY DISK FRICTION

k = CONSTANT

n = SPEED (R.P.M)

D_2 = IMPELLER OUTER DIA.

The disk friction is the increased power consumption which occurs on the shroud and hub of the impeller because it rotates in a fluid filled pump casing. The size of the disc friction depends primarily on the speed the impeller diameter as well as the dimensions of the pump housing in particular the distance between impeller and pump casing.

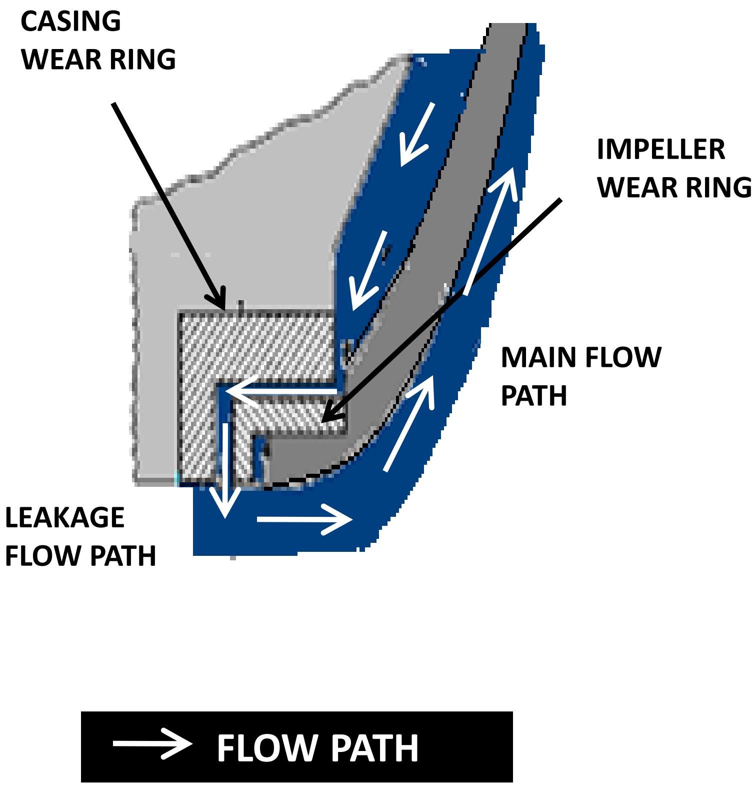
$$k = 7.3 \cdot 10^{-4}\left(\frac{2\nu \cdot 10^{6}}{U_2 D_2}\right)^{m}$$

M= exponent equals 1/6 for smooth surface , 1/7 to 1/9 for rough surface,
v- kinematic viscosity –m/sec , U2 = Peripheral velocity [m/s], D2 = Impeller diameter [m]

LOSSES IN CENTRIFUGAL PUMP

LEAKAGE LOSSES

Caused by liquid flowing past **wear rings, inter-stage bushes, mechanical seals, glands & balancing devices, Leakage losses results in a loss in efficiency because the flow in a impeller is increased compared to the flow through the entire pump.**



The general expression for determining the amount of leakage across an annular clearance is

$$Q_L = \mu \times A_{cl} \times \sqrt{(2.g.\Delta H_{cl})}$$

HERE, Q_L = LEAKAGE FLOW

μ = LEAKAGE GAP LOSS CO-EFFICIENT

g = GRAVITATIONAL ACCELERATION

ΔH_{cl} = HEAD LOSS ACROSS ANNULAR PATH

A_{cl} = AREA AT CLEARANCE ZONE

Pump selection method

Pump selection method

Available data's,

Type - double suction split casing pump

Application - Water

Capacity - 750 cum/hr

Head - 35 Meters
Suction lift - 3 Meters
(Can be estimated
NPSH A,NPASHR
Pump speed,
motor rating

Calculate Available NPSH A in the system

NPSH for open system = ATM head - (suction head + friction losses + vapor losses)

Assumed ,

Friction losses = 0.5 M

Vapour losses = 0.6 M

NPSH A = 10.3 - (3 +0.5+0.6) = 6.2 Meters

Maximum permissible speed and actual speed of the motor

$$\boxed{NSS = \frac{Ns \times Q^{0.5}}{NPSHr^{0.75}}}$$

$$Speed\ (NS) = \frac{Nss \times NPSH\ A^{0.75}}{Q^{0.5}}$$

NSS = 7500 TO 10000

Assumed - 8500 in US units

$$Ns = \frac{8500 \times (20)^{0.75}}{(1651)^{0.5}}$$

Speed = 2004 RPM

The recommended motor speed is - 1450 RPM /4 Pole

3. Motor Power

$$\boxed{BKW = \frac{\text{Capacity in cum/hr} \times \text{Head M} \times \text{sp.gravity}}{367 \times \text{Eff.at duty}}}$$

$$BKW = \frac{750 \times 35 \times 1}{367 \times 0.86}$$

85.54 KW

ADD 15% Margin = 85.54 x 1.15 = 98 KW ,So recommendable is 110 KW/1450 RPM/50 Hz

(Efficiency is taken from the HI chart)

4. NPSH Required for the pump

$$NSS = \frac{NS \times Q^{0.5}}{NPSHr^{0.75}}$$

$$NPSH^{0.75} = \frac{Ns \times Q^{0.5}}{Nss}$$

$$NPSH\ R = \frac{1500 \times (1651)^{0.5}}{8500}$$

NPSH R = 7.17 FT

NPSH R = 2.1 Meters

NPSH A is greater than NPSH R

5. Minimum shaft diameter at coupling area.

Shear stress formula = $HP = S\ N(D)^3 / 321000$

S- Permissible shear stress in shaft- PSI = 8500 (SS410)

$$D^3 = \frac{150 \times 321000}{8500 \times 1500}$$

D = 1.577 Inches = 39.55 mm at coupling area

6.Suction and delivery nozzle size :

Velocity at inlet (assume) - 4 m/sec.

$\boxed{\text{Capacity = Area X velocity}}$

Area ($\frac{3.147}{4}$/d^2)) = capacity / velocity

d^2 = (750 x 4)/4 x3600

d= 0.456 meters = 456 mm ,So =450 mm.(wetted area)

For discharge this can be one size lower - 400 mm say 16 inches.

7. Selecting Impeller Diameter:

$\boxed{\text{U2 = KU (2 X g X H) ^0.5}}$

KU = Co efficient related to specific speed -refer to the chart
for 2004 specific speed the KU value is 1.1

U2 = 1.1 (2x 9.81 x 35) ^0.5
U2 = 28.82 m/sec

Impeller diameter can be calculated by using the below formula,

$\boxed{\text{U2 = 3.147 X D2 X N /60}}$

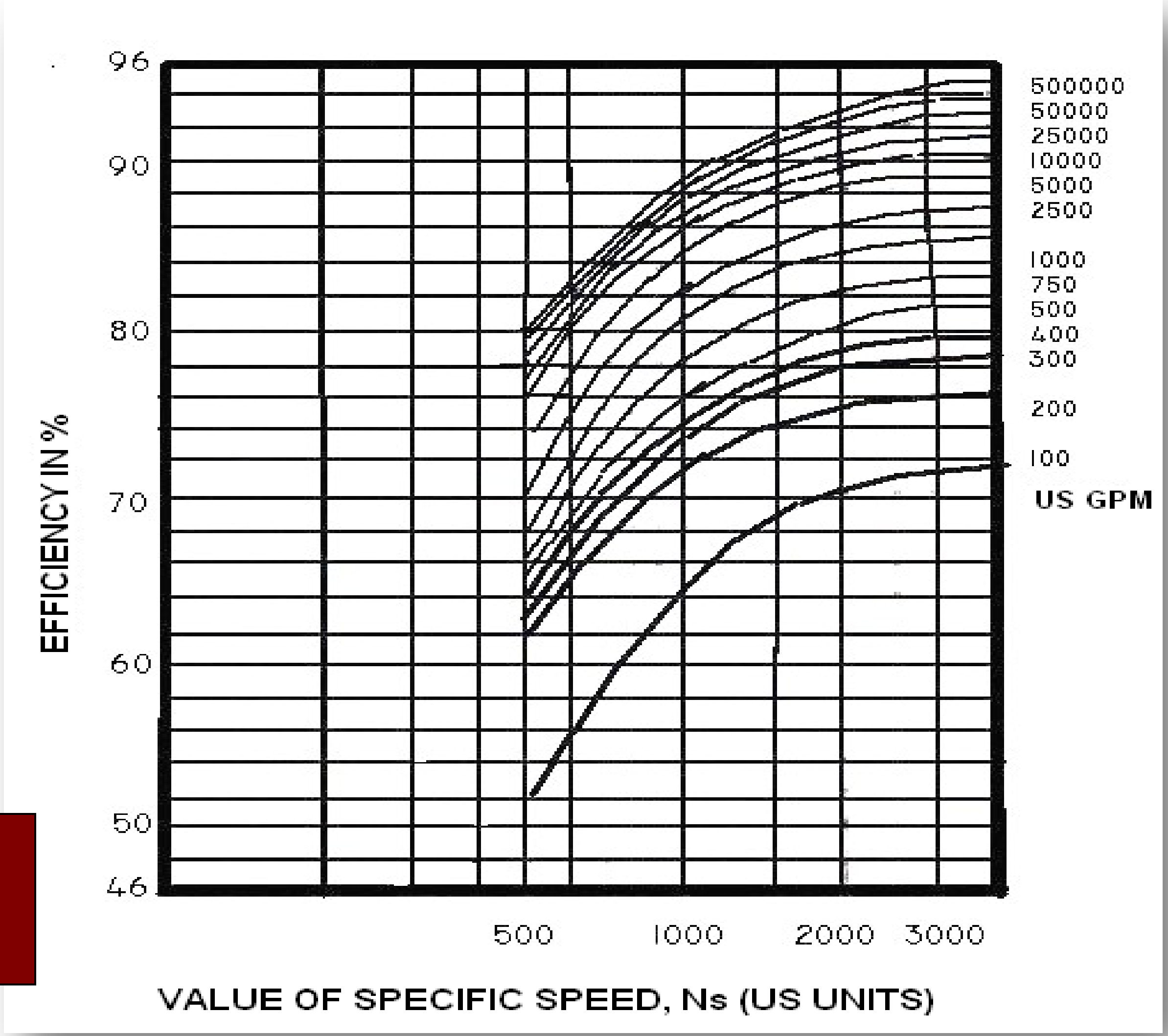
D2 =(28.82 X 60)/ 3.147 X 1500

D2= 0.366 MTRS = 366 MM.

Efficiency Chart - II

Optimum Efficiency as a Function of Specific Speed & Flow-rate

Fraser-Sabini Chart



Specific Speed & Efficiency

Specific Speed	K_u	K_{m2}	D1/D2	K_3
400	0.965	0.040	0.380	0.555
800	1.000	0.073	0.430	0.490
1200	1.035	0.100	0.470	0.425
1600	1.065	0.120	0.510	0.375
2000	1.100	0.140	0.550	0.335
2400	1.135	0.160	0.590	0.300
2800	1.165	0.175	0.620	0.275
3200	1.200	0.193	0.640	0.260
3600	1.235	0.205	0.650	0.265

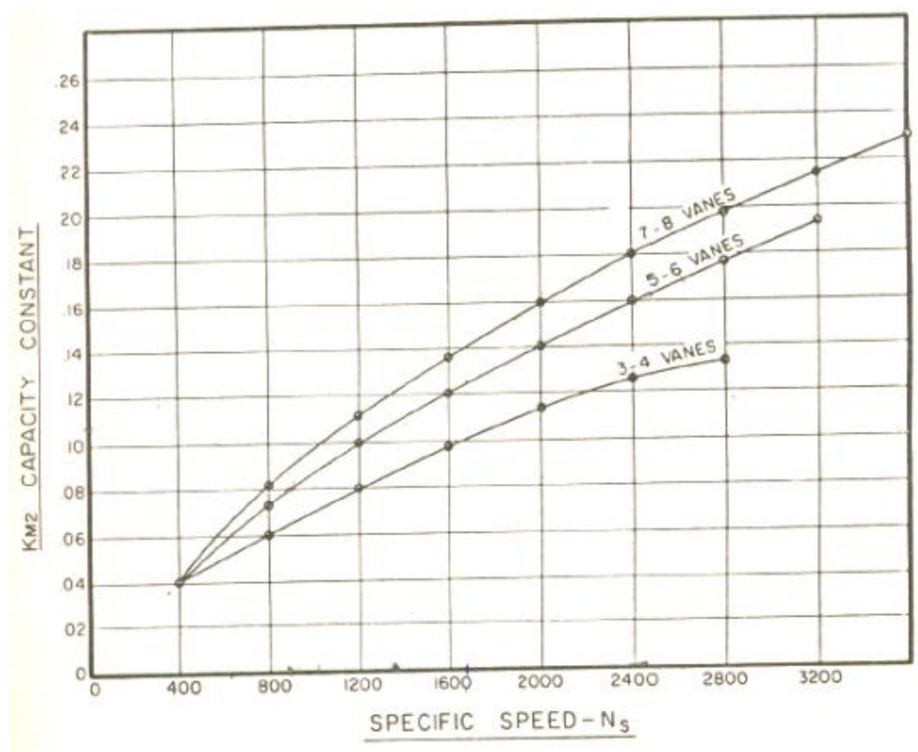
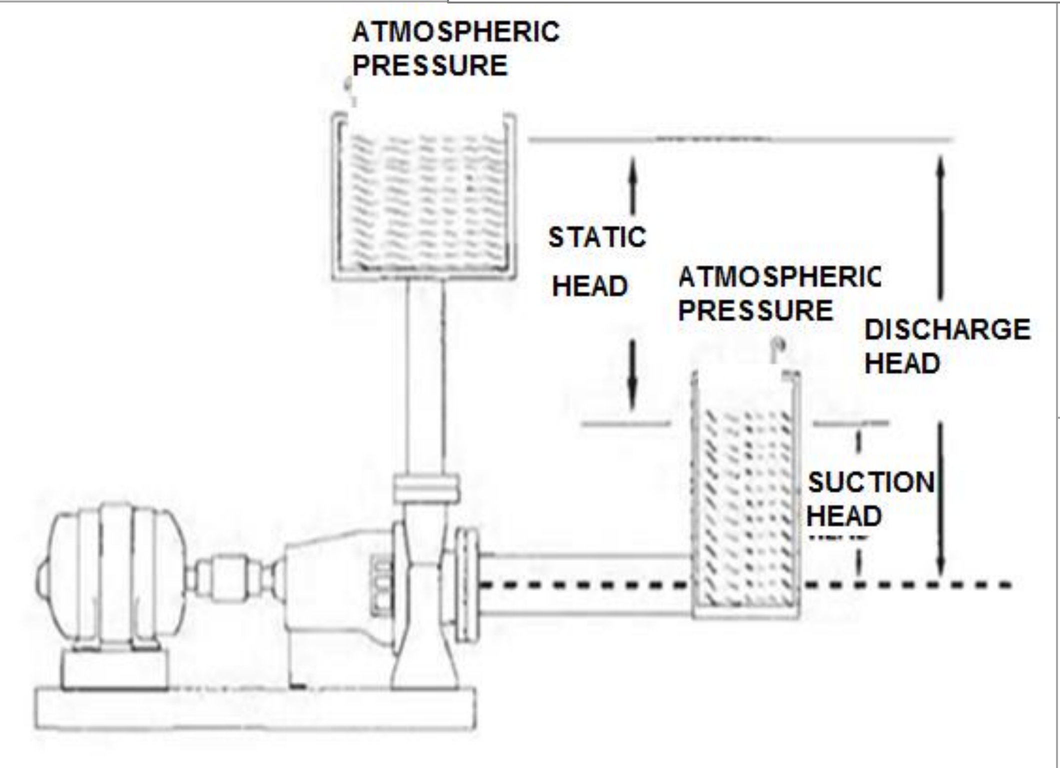
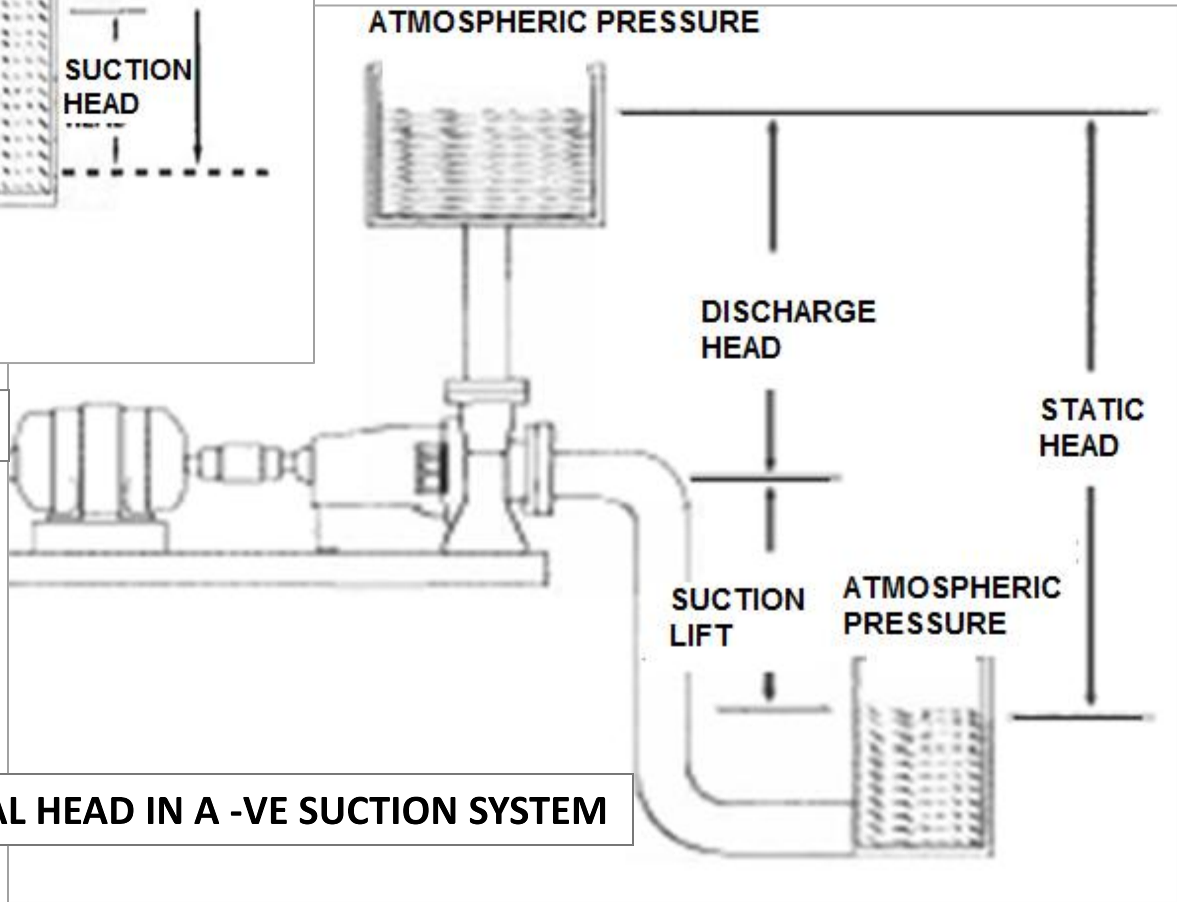


Figure 3-4. Capacity constant.

DIFFERENT HEAD TERMS IN A PUMPING SYSTEM

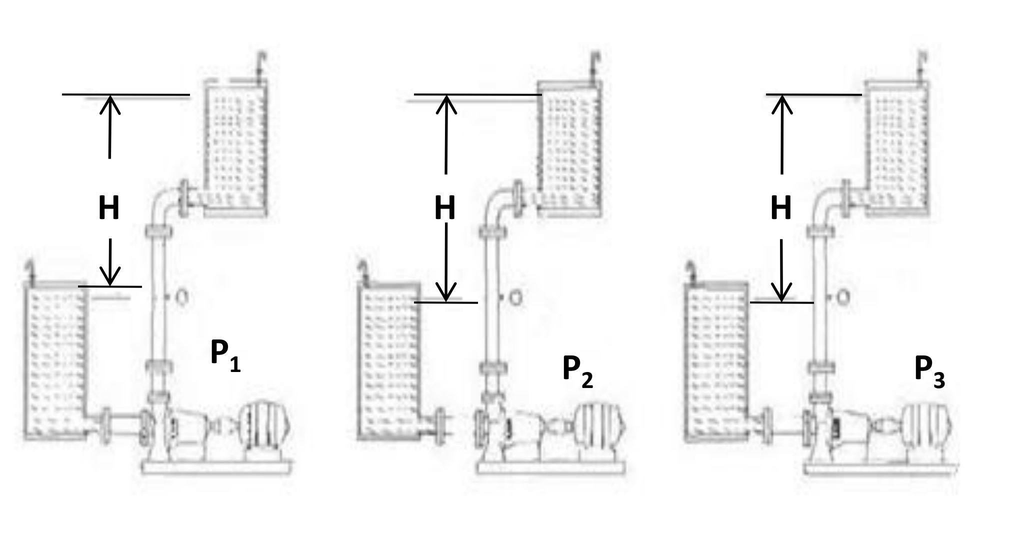


TOTAL HEAD IN A +VE SUCTION SYSTEM



TOTAL HEAD IN A -VE SUCTION SYSTEM

UNDERSTANDING PRESSURE & HEAD IN A PUMPING UNIT



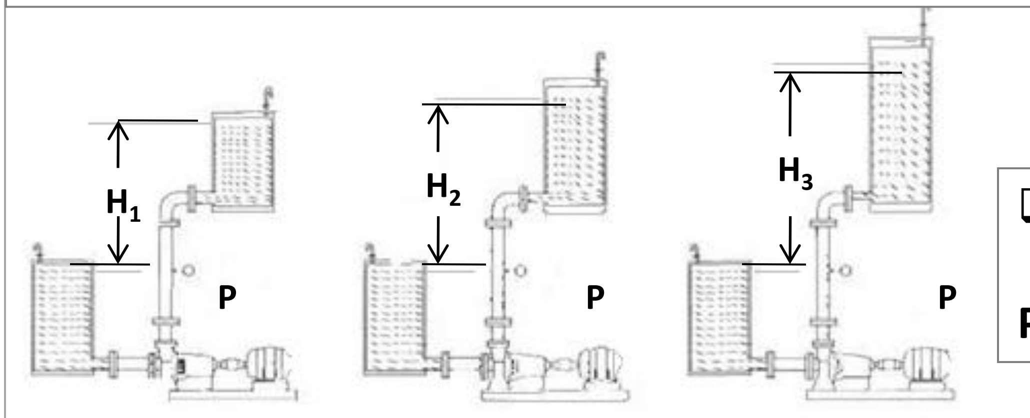
- ALL PRESSURES CAN BE VISUALIZED AS BEING CAUSED BY THE WEIGHT OF A COLUMN OF LIQUID AT ITS BASE.

$$\text{PRESSURE(PSI)} = \frac{\text{HEAD(FT)} \times \text{S.P. GRAVITY}}{2.31}$$

OR

$$\text{PRESSURE(Kgf/CM}^2\text{)} = \frac{\text{HEAD(M)} \times \text{S.P. GRAVITY}}{10}$$

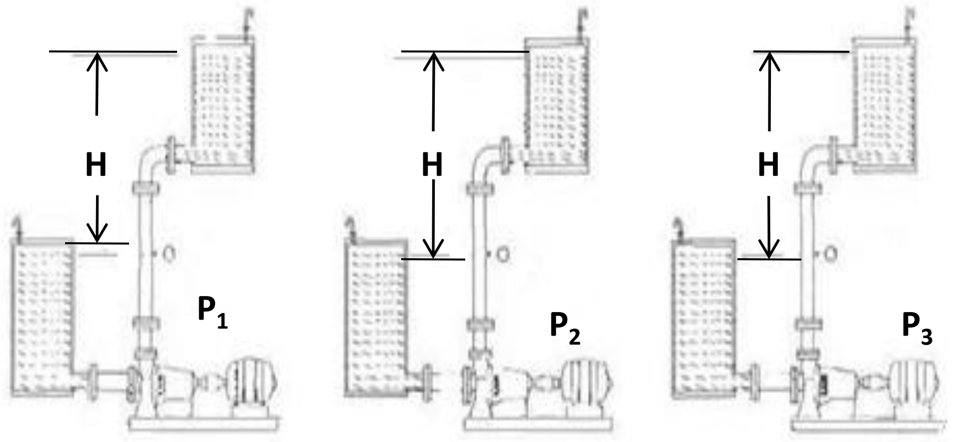
- THREE IDENTICAL PUMPS DESIGNED TO DEVELOP SAME HEAD PRODUCES DIFFERENT PRESSURES WHICH VARY IN PROPORTION TO THEIR SPECIFIC GRAVITY.



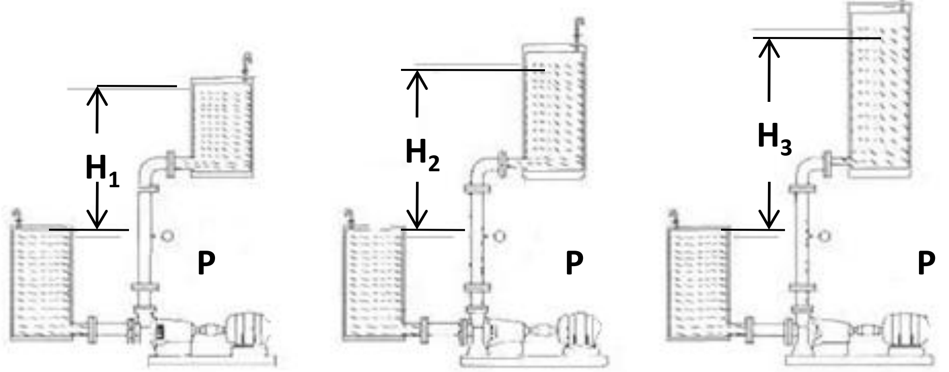
- PUMPS SHOULD BE SPECIFIED IN TERMS OF HEAD AND NOT IN TERMS OF PRESSURE TO AVOID AMBIGUITY.

- THREE PUMPS PRODUCE SAME DISCHARGE PRESSURE BUT DEVELOP HEADS INVERSELY PROPORTIONAL TO THEIR SPECIFIC GRAVITY.

UNDERSTANDING PRESSURE & HEAD IN A PUMPING UNIT



- THREE IDENTICAL PUMPS DESIGNED TO DEVELOP SAME HEAD PRODUCES DIFFERENT PRESSURES WHICH VARY IN PROPORTION TO THEIR SPECIFIC GRAVITY.



- THREE PUMPS PRODUCE SAME DISCHARGE PRESSURE BUT DEVELOP HEADS INVERSELY PROPORTIONAL TO THEIR SPECIFIC GRAVITY.

- ❑ ALL PRESSURES CAN BE VISUALIZED AS BEING CAUSED BY THE WEIGHT OF A COLUMN OF LIQUID AT ITS BASE.

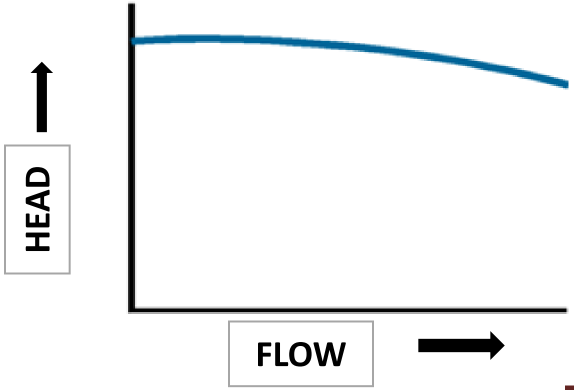
$$\text{PRESSURE(PSI)} = \frac{\text{HEAD(FT)} \times \text{S.P. GRAVITY}}{2.31}$$

OR

$$\text{PRESSURE(Kgf/CM}^2\text{)} = \frac{\text{HEAD(M)} \times \text{S.P. GRAVITY}}{10}$$

- ❑ PUMPS SHOULD BE SPECIFIED IN TERMS OF HEAD AND NOT IN TERMS OF PRESSURE TO AVOID AMBIGUITY.

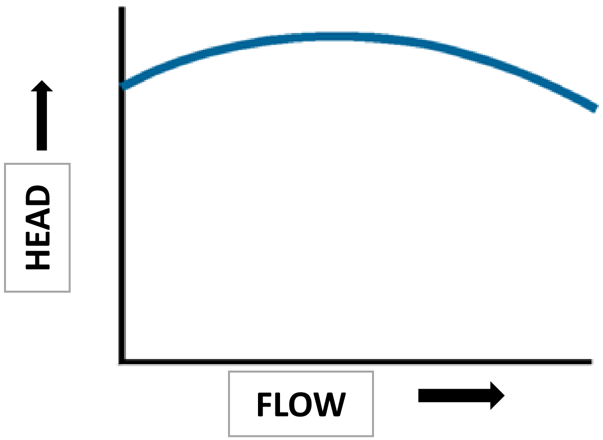
VARIOUS TYPES OF H-Q CURVE SHAPES



FLAT CURVES

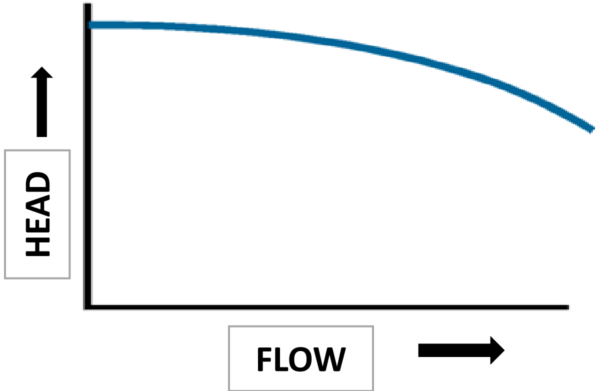
SHOWS LITTLE VARIATION OF HEAD AT ALL FLOWS BETWEEN DESIGN POINT & SHUT-OFF

DROOPING CURVE



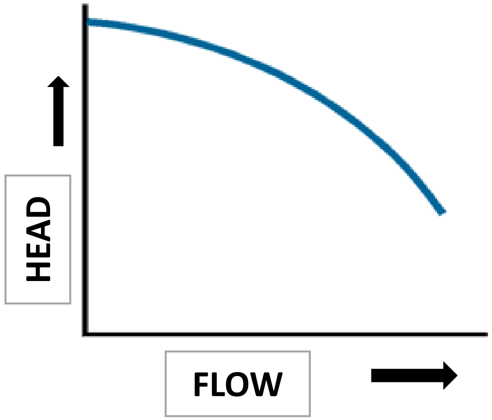
HEAD DEVELOPED AT SHUT-OFF IS LESS THAN HEAD DEVELOPED AT SOME FLOW BETWEEN B.E.P & SHUT-OFF

STEADILY RISING CURVE



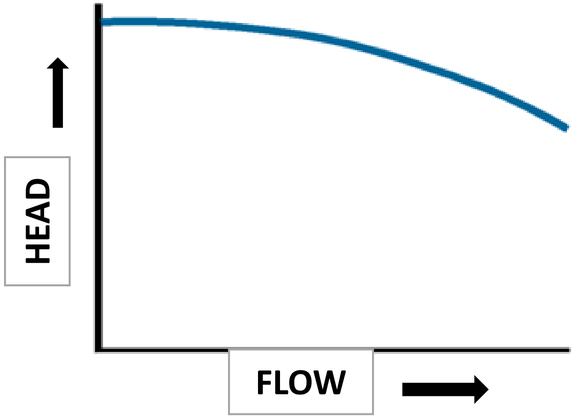
HEAD RISES CONTINUOUSLY FROM DESIGN POINT TO SHUT-OFF

VARIOUS TYPES OF H-Q CURVE SHAPES



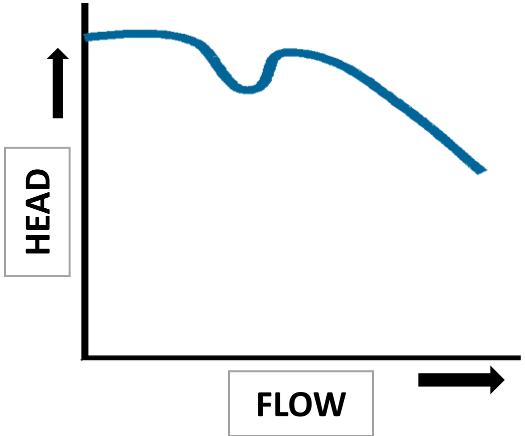
STEEP CURVE

LARGE INCREASE IN HEAD DEVELOPED AT SHUT-OFF FROM B.E.P.



STABLE CURVE

ONLY ONE FLOW RATE AT ANY ONE HEAD



UNSTABLE CURVE

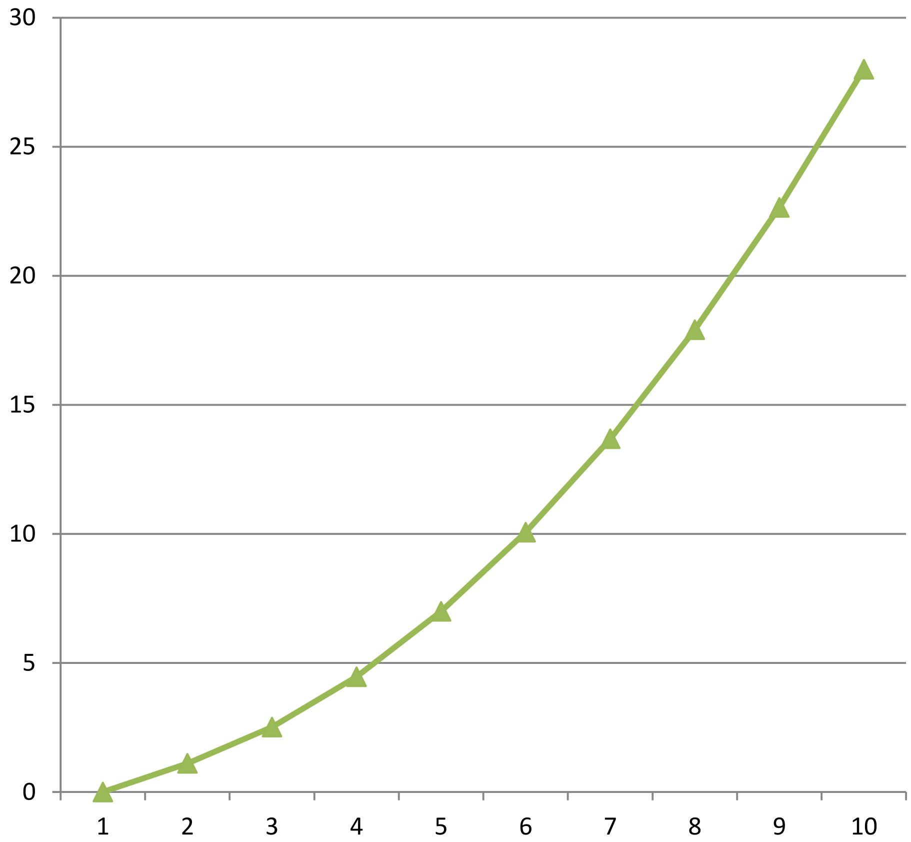
THE SAME HEAD IS DEVELOPED AT MORE THAN ONE FLOW RATES

System Curve

- A *system curve describes the relationship between the flow in a pipeline and the head* loss produced.

- The essential elements of a system curve include:
- A) The static head of the system,
- B) The friction or head loss in the piping system.
- C) Pressure head

Calculation method



System curve

944@28M

L/s	M	RL/S		
944	28	0	0	0%
944	28	188	1.110529	10%
944	28	283	2.516442	20%
944	28	377	4.465774	30%
944	28	472	7	50%
944	28	566	10.06577	60%
944	28	660	13.6868	70%
944	28	755	17.91051	80%
944	28	849	22.64798	90%
944	28	944	28	100%

Designed flow – 944 L/s

Designed Head – 28 Meters

Head at 50 % flow (472 l/s) -- ????

H2 =(Q2 ^2)x HI

Q1^2

H2= (472)^2 x 28 = 7 Meters

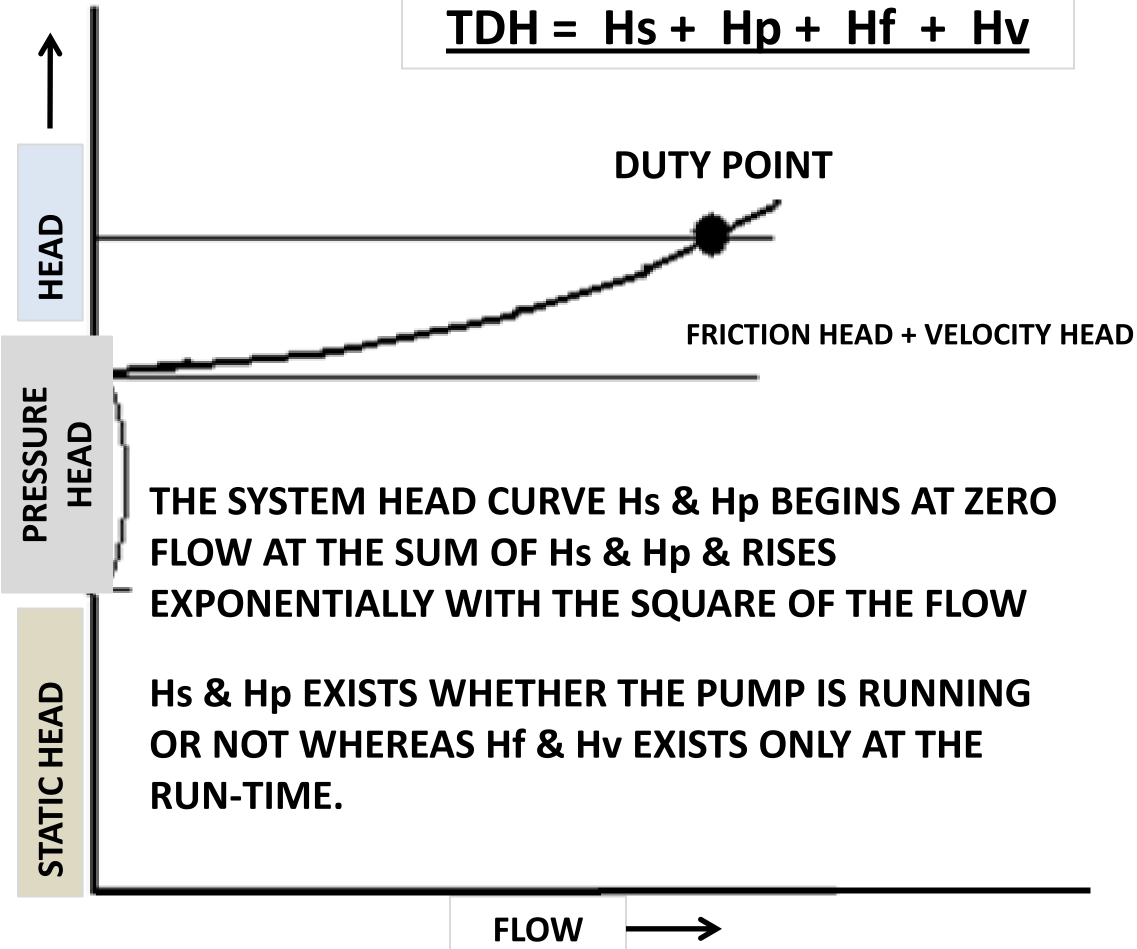
= (944) ^2

SYSTEM HEAD CURVE

IT'S YOUR SYSTEM THAT CONTROLS YOUR PUMP.

ALL PUMPS MUST BE DESIGNED TO COMPLY WITH OR MEET THE NEEDS OF THE SYSTEM & THE NEED OF THE SYSTEM IS RECOGNIZED USING THE TERM 'TDH'

TDH = Hs + Hp + Hf + Hv



THE SYSTEM HEAD CURVE Hs & Hp BEGINS AT ZERO FLOW AT THE SUM OF Hs & Hp & RISES EXPONENTIALLY WITH THE SQUARE OF THE FLOW

Hs & Hp EXISTS WHETHER THE PUMP IS RUNNING OR NOT WHEREAS Hf & Hv EXISTS ONLY AT THE RUN-TIME.

HERE, TDH = TOTAL DYNAMIC HEAD

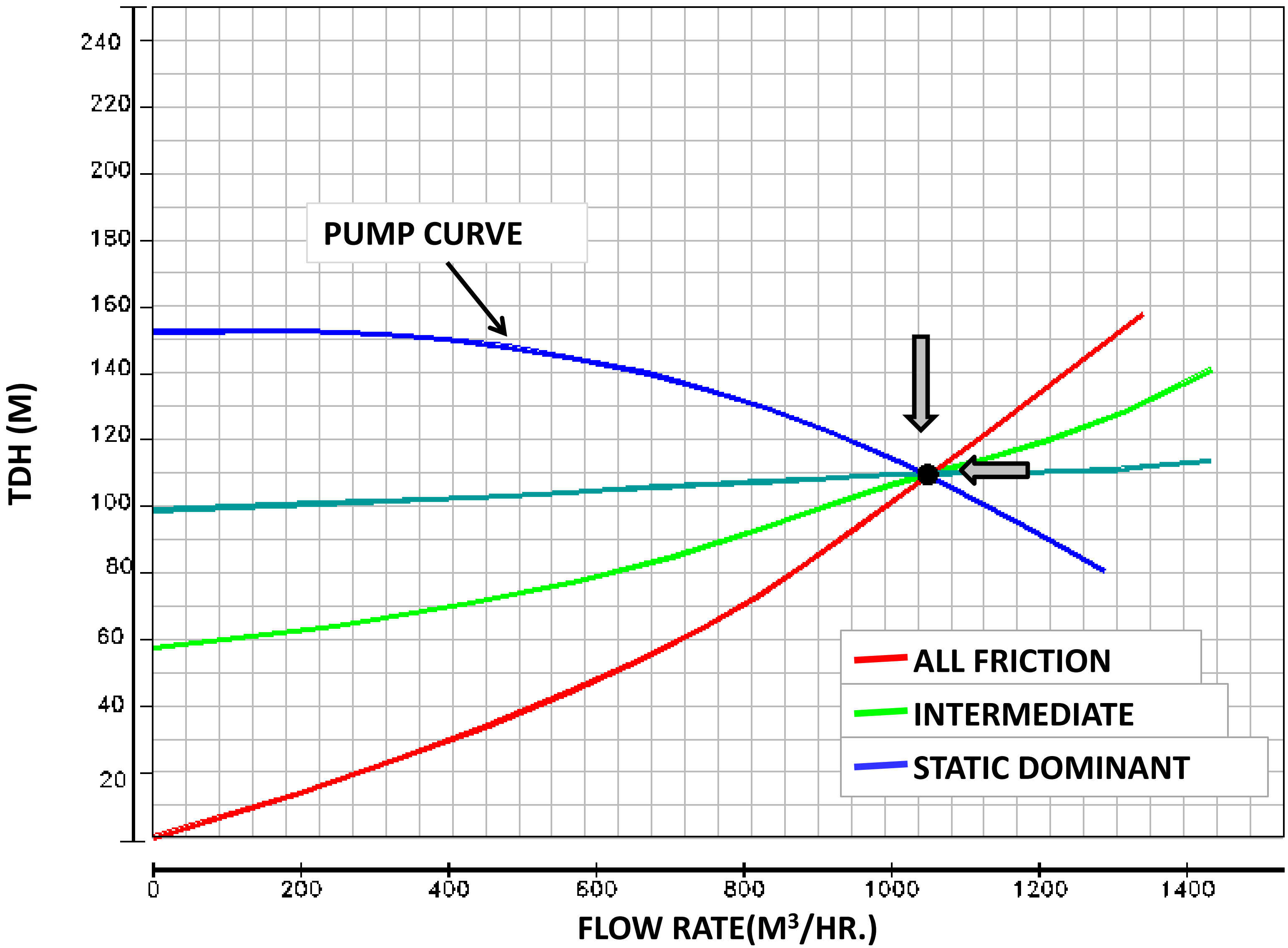
Hs = STATIC HEAD (DIFFERENCE IN THE LIQUID SURFACE LEVELS AT SUCTION SOURCE & DISCHARGE TANK

Hp = PRESSURE HEAD (CHANGE IN PRESSURE AT SUCTION & DELIVERY TANK)

Hv = VELOCITY HEAD

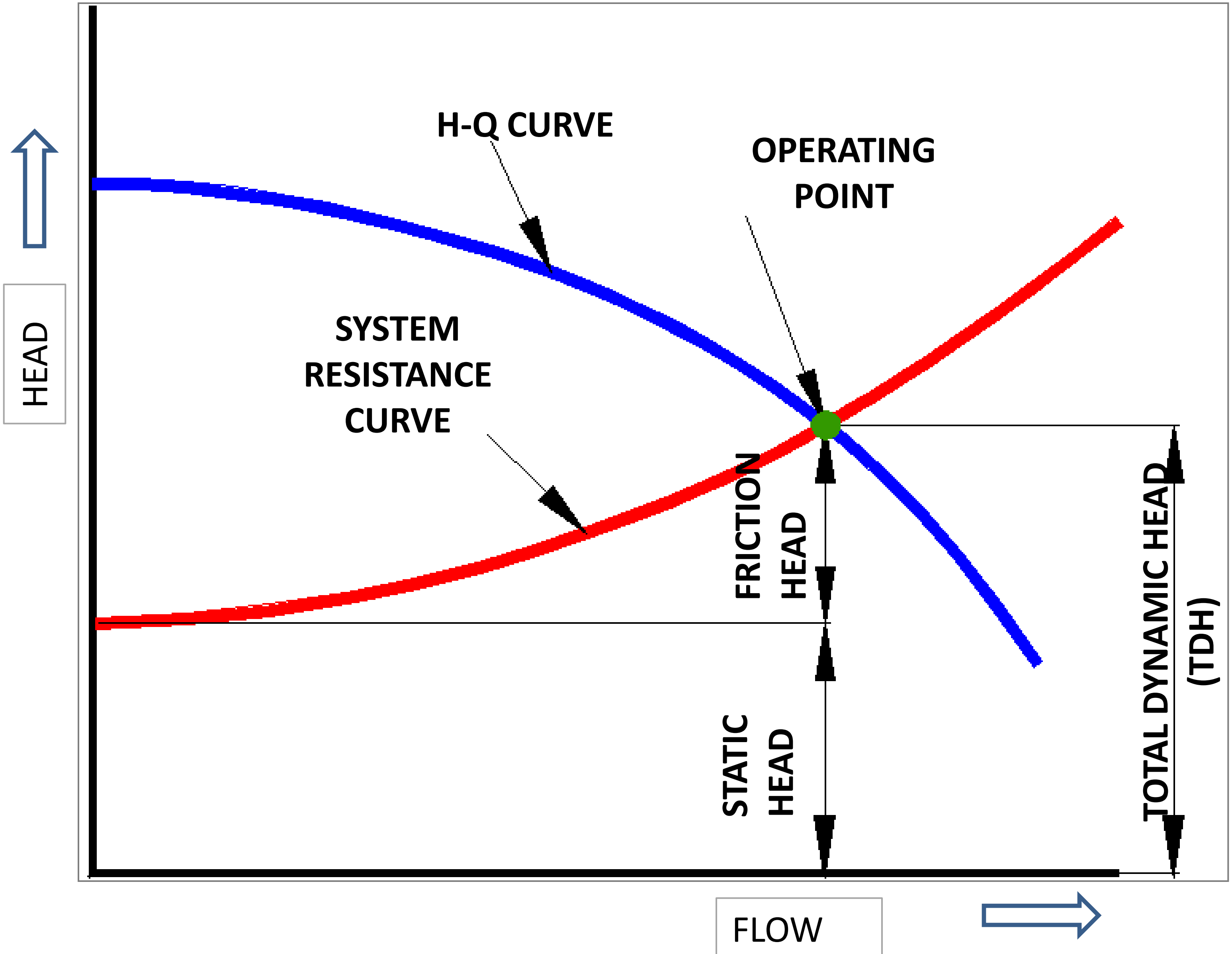
Hf = FRICTION HEAD (HEAD LOSS DUE TO FRICTION ACROSS PIPES, VALVES, CONNECTIONS & SUCTION & DELIVERY ACESSORIES).

NATURE OF SYSTEM CURVES

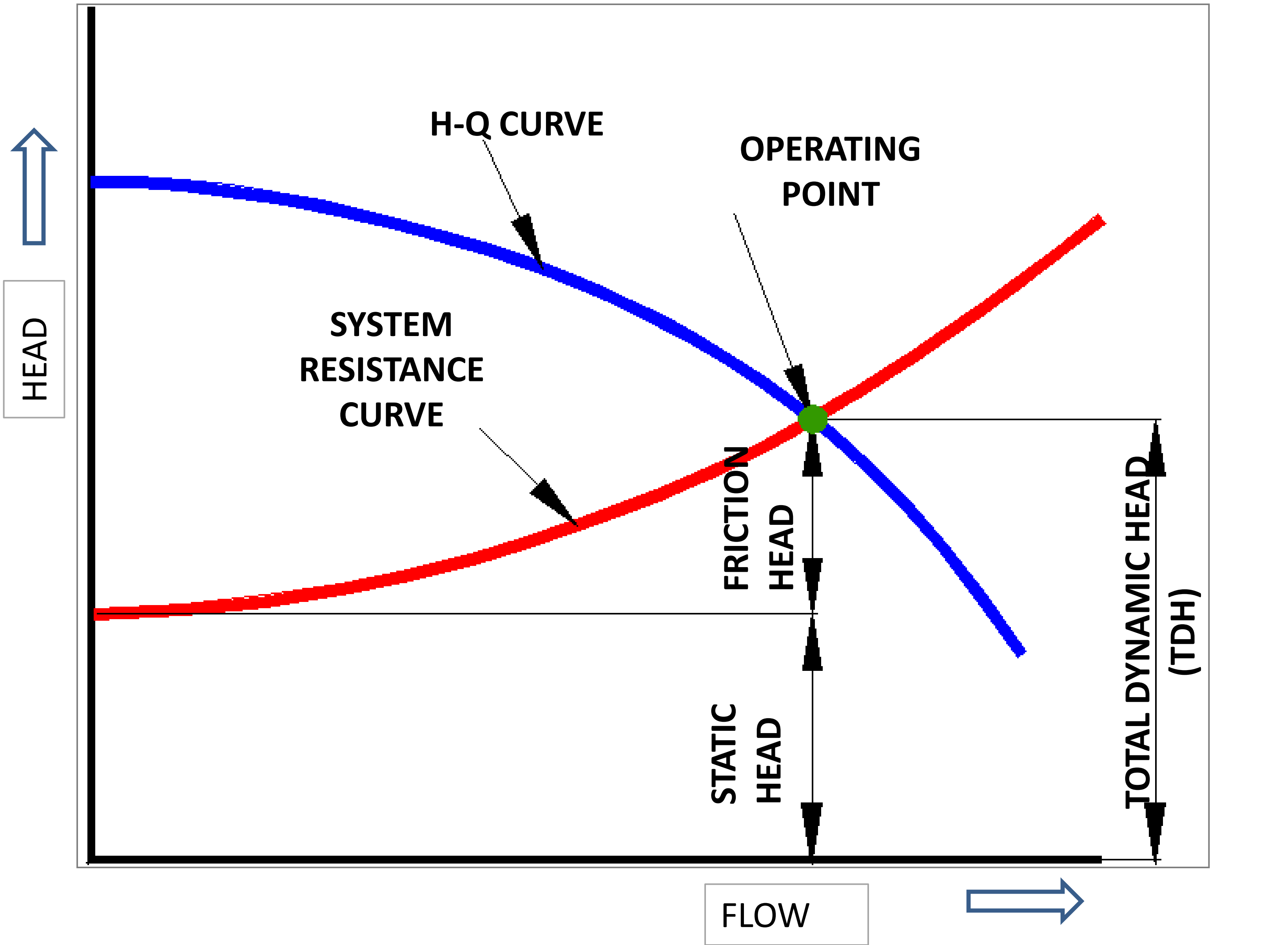


THREE DIFFERENT TYPES OF SYSTEM CURVES WITH A SINGLE COMMON FLOW RATE/HEAD POINT

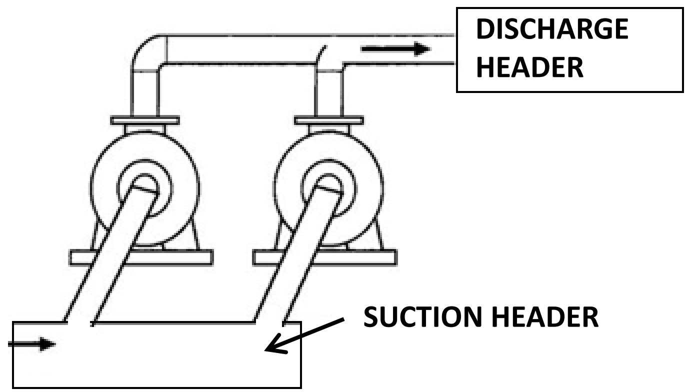
SYSTEM HEAD CURVE SUPERIMPOSED ON H-Q CURVE OF THE PUMP

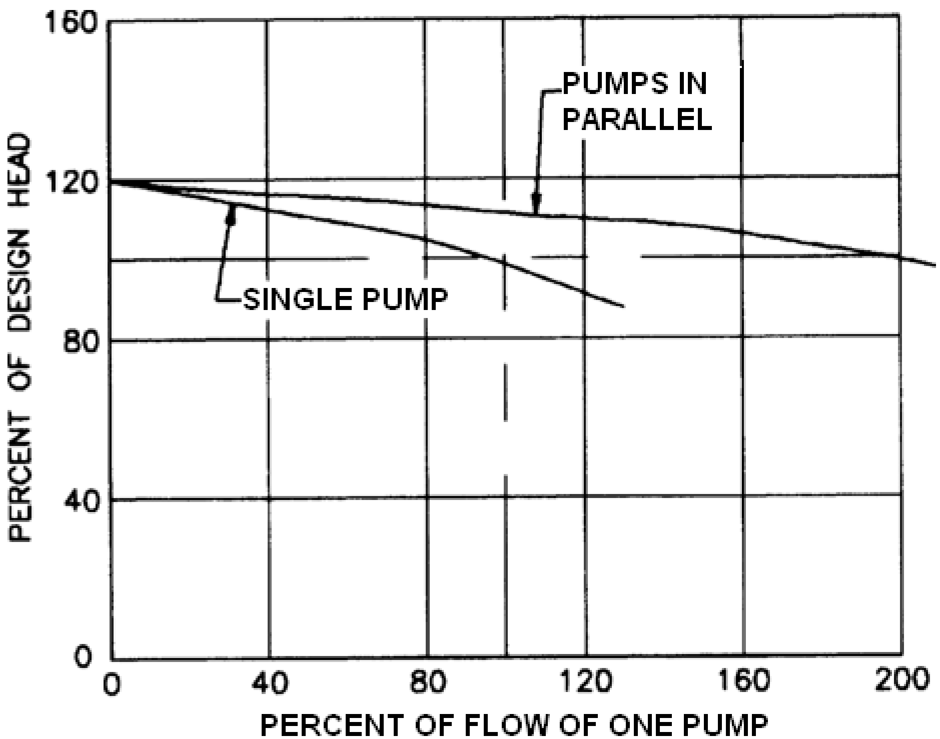


SYSTEM HEAD CURVE SUPERIMPOSED ON H-Q CURVE OF THE PUMP



PARALLEL OPERATION

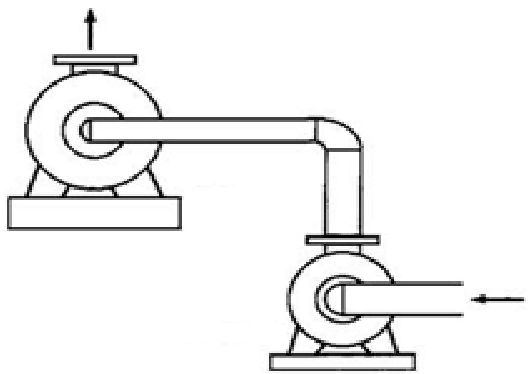


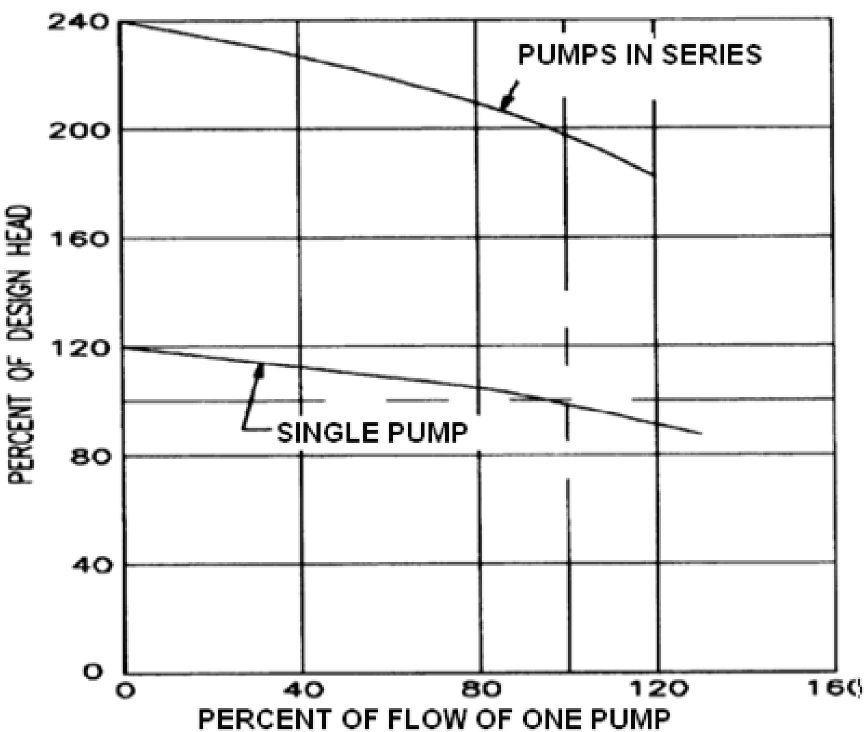


WHEN THE SYSTEM FLOW DEMAND VARIES OVER A WIDE RANGE, PARALLEL OPERATON OF SEVERAL SMALL PUMPS INSTEAD OF A SINGLE LARGE ONE MAY BE EMPLOYED.

COMBINED H-Q CURVE IS OBTAINED BY ADDING THE DISCHARGES GENERATED BY INDIVIDUAL PUMPS AT THE SAME HEADS.

SERIES OPERATION OF PUMPS





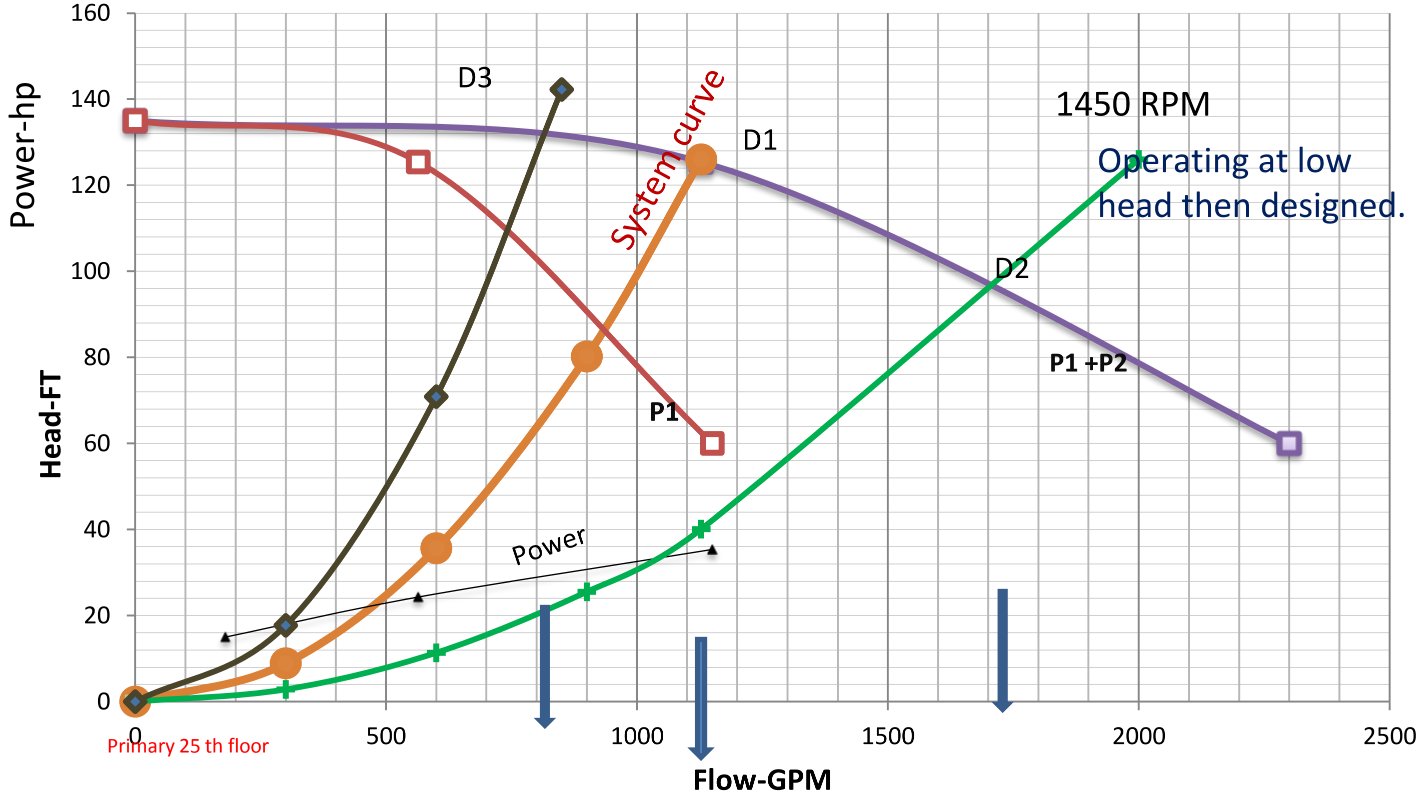
SERIES OPERATION FOR SYSTEMS WITH HIGH HEAD REQUIREMENT

- ❖ COMBINED CURVE OBTAINED BY ADDING THE HEADS DEVELOPED BY INDIVIDUAL PUMPS AT THE SAME FLOW RATES.

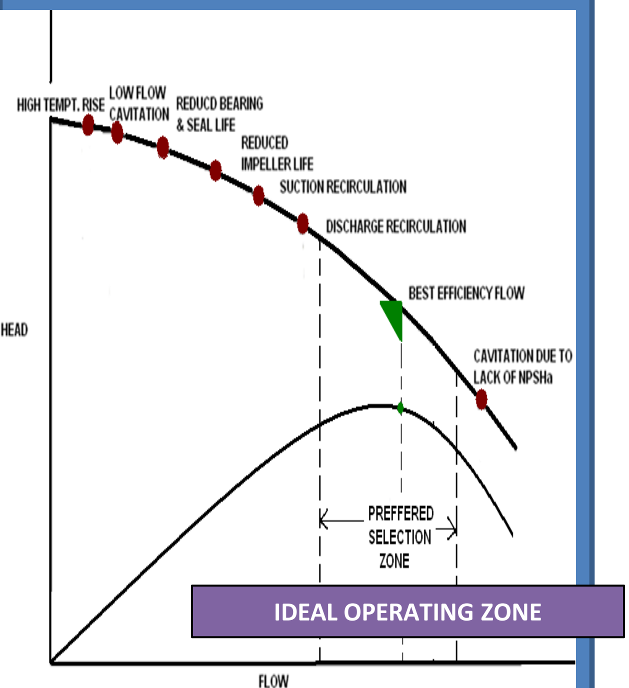
Parallel operation two pumps systems (constant speed)

- (P1 +P2)

- ***Dynamic losses dominated system curve .***



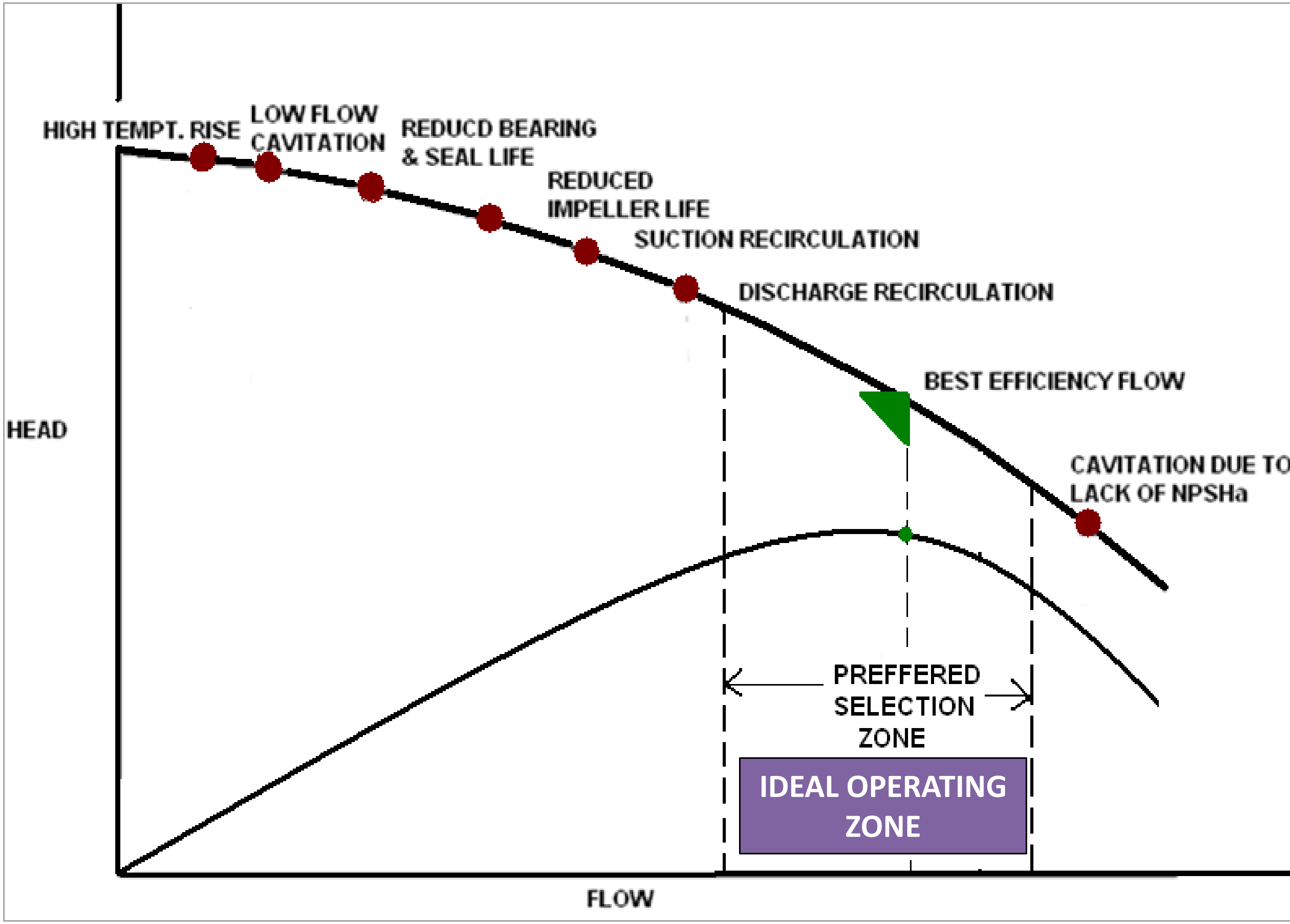
OPERATION FAR TO THE LEFT OF B.E.P — POSSIBLE PROBLEMS



OPERATION AT LOW FLOW MAY RESULT IN

- CASES OF HEAVY LEAKAGE FROM THE CASING, SEAL OR STUFFING BOX.
- DEFLECTION & SHEARING OF SHAFT.
- SEIZURE OF PUMP INTERNALS.
- CLOSE CLEARENCE EROSION.
- SEPERATION / LOW-FLOW CAVITATION.
- PRODUCT QUALITY DEGRADATION.
- EXCESSIVE HYDRAULIC THRUST.
- PREMATURE BEARING FAILURE.
- VIBRATION & NOISE
- HEATING OF LIQUID PUMPED.

ONSET OF ADVERSE EFFECTS WHEN OPERATING AWAY FROM B.E.P



OPERATIONTO TO THE RIGHT OF B.E.P
— PROBABLE PROBLEMS

SHAFT STRESS – TORSION & BENDING

COMBINED TORSIONAL & BENDING STRESSES OR SHAFT DEFLECTION IN SINGLE VOLUTE PUMPS MAY EXCEED PERMISSIBLE LIMITS.

SHAFT DEFLECTION

DUE TO HIGH THRUST VALUES SHAFT DEFLECTION IN SINGLE VOLUTE PUMPS MAY EXCEED PERMISSIBLE LIMITS.

NPSHr > NPSHa

NPSH REQUIRED MAY BE IN EXCESS OF NPSH AVAILABLE FOR THE SYSTEM.

EROSION, NOISE & VIBRATION

EROSION DAMAGE, NOISE & VIBRATION MAY OCCUR DUE TO HIGH LIQUID VELOCITIES.

CONTROL POSSIBILITIES FOR CENTRIFUGAL PUMPS

PUMP OUTPUT CAN BE CONTROLLED BY THE FOLLOWING METHODS

- THROTTLING
- CONNECTION OR DISCONNECTION OF PUMPS
  - RUNNING IN PARALLEL
  - RUNNING IN SERIES
- BYPASS REGULATION
- AFFINITY LAW – IMPELLER TRIM,SPEED REGULATION
- IMPELLER VANE & WIDTH ADJUSTMENTS
- PREROTATION CONTROL
- CAVITATION CONTROL

FIVE VANE IMPELLER

SIX VANE IMPELLER

AFFINITY LAWS

FOR A PARTICULAR PUMP THE HEAD DEVELOPED & THE DISCHARGE CAN BE CONTROLLED, WITHIN CERTAIN LIMITS, ACCORDING TO THE AFFINITY LAWS:

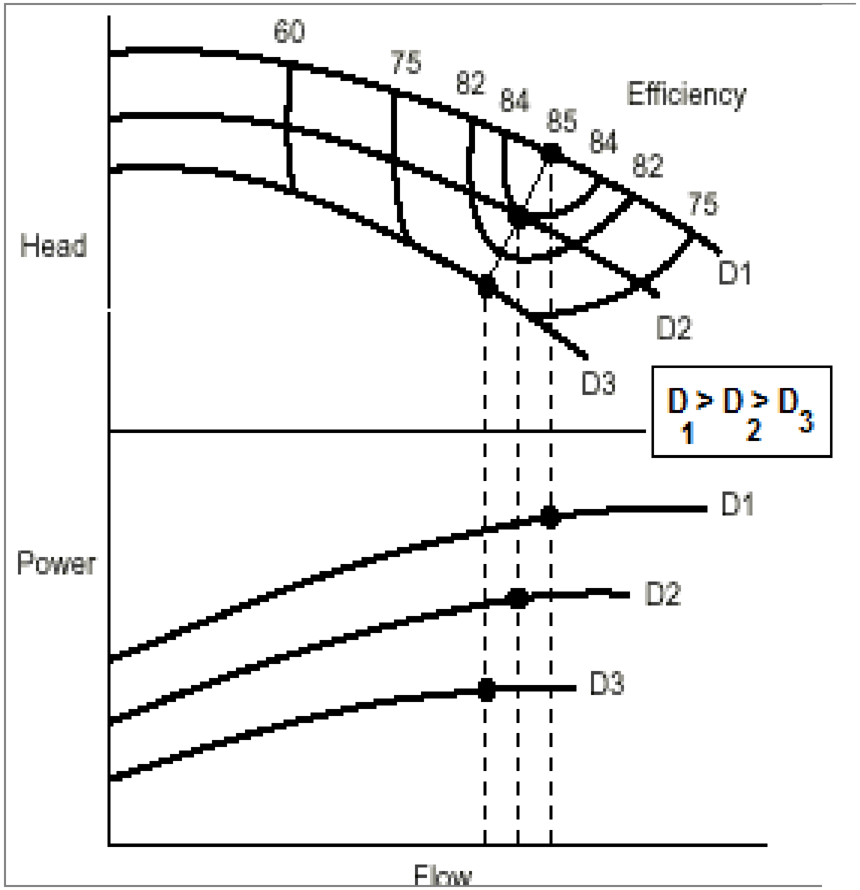
WHEN ONLY IMPELLER DIA. CHANGES & SPEED REMAINS THE SAME	**WHEN ONLY SPEED CHANGES & IMPELLER DIA. REMAINS THE SAME**	**WHEN BOTH DIA & SPEED CHANGE**
$Q_2 = Q_1 \times (D_2/D_1)$	$Q_2 = Q_1 \times (N_2/N_1)$	$Q_2 = Q_1 \times (D_2/D_1) \times (N_2/N_1)$
$H_2 = H_1 \times (D_2/D_1)^2$	$H_2 = H_1 \times (N_2/N_1)^2$	$H_2 = H_1 \times \{ (D_2/D_1) \times (N_2/N_1) \}^2$
$BKW_2 = BKW_1 \times (D_2/D_1)^3$	$BKW_2 = BKW_1 \times (N_2/N_1)^3$	$BKW_2 = BKW_1 \times \{ (D_2/D_1) \times (N_2/N_1)\}^3$

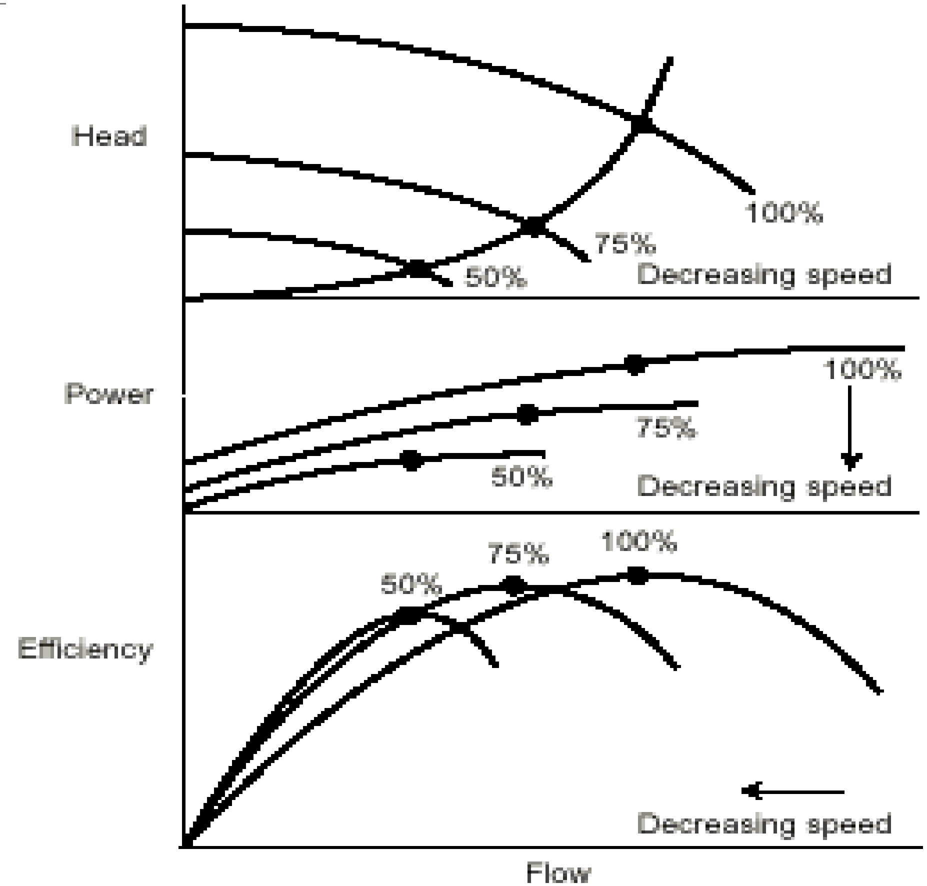
- **Q1, H1, BKW1, D1 & N1 ARE CAPACITY, HEAD, INPUT POWER IN KW, IMPELLER DIA. & SPEED AT INITIAL CONDITION.**
- **Q2, H2, BKW2, D2 & N2 ARE CAPACITY, HEAD, INPUT POWER IN KW, IMPELLER DIA. & SPEED AT CHANGED CONDITION.**

APPLICATION OF AFFINITY LAWS

ONE PUMP IS USED TO SERVICE DIFFERENT DUTIES

REDUCING THE DIAMETER OF THE IMPELLER MAKES AN EXISTING PUMP RUN MORE EFFICIENTLY AT LOWER FLOWS WITHOUT THE NEED FOR THROTTLING.





SAME PUMP WITH A RANGE OF IMPELLER DIAMETERS TO MEET DIFFERENT DUTY H & Q.

SAME PUMP WITH DIFFERENT MOTOR SPEEDS THROUGH VSD TO ALLOW ONE PUMP TO BE USED OVER A MUCH WIDER RANGE OF DUTIES.

VIBRATION IN A CENTRIFUGAL PUMP

TYPICAL PUMP **OR PUMP ELEMENT VIBRATIONS**

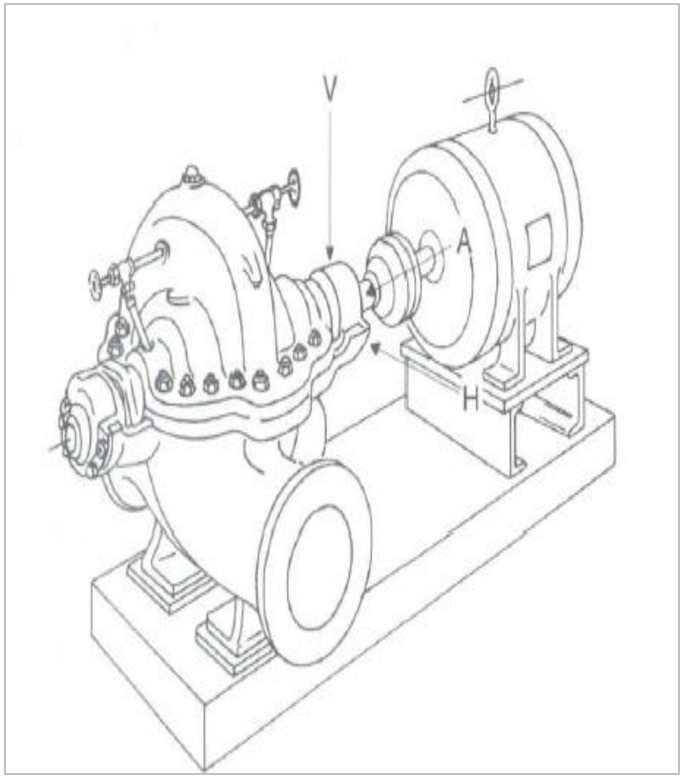
TYPES OF VIBRATION

- **LATERAL SHAFT VIBRATION**
- **VIBRATION IN THE SYSTEM - PUMP BASE PLATE**
- **BEARING HOUSING VIBRATION**

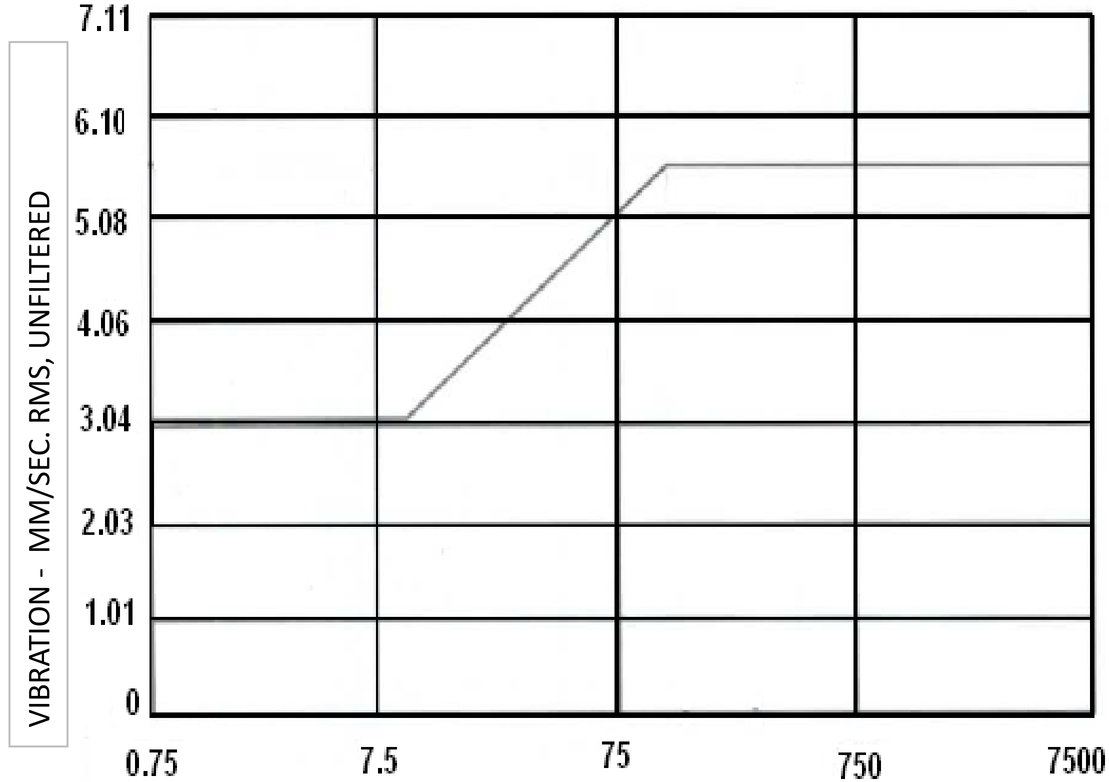
PROBLEM RELATED TO SYSTEM

- MISALIGNMENT BETWEEN PUMP & DRIVE
- EXCITATION FROM THE DRIVE
- EXCITATION FROM COUPLING
- EXCITATION FROM THE COMPONENTS OF PIPING SYSTEM
- EXCESSIVE PIPING LOAD ON THE CASING (DISCHARGE PIPE-STRESS)
- INADEQUATE LEVELLING OF THE PUMP FOUDATION BOARD & PUMP-BASEPLATE
- LOOSE FOUNDATION
- POOR FLOW QUALITY IN THE SUMP/ UNFAVOURABLE PUMP INLET CONDITIONS (NPSH, INLET VORTICES, ETC.)
- WATER HAMMER

TYPICAL VIBRATION CHART FOR SPLIT-CASE PUMP

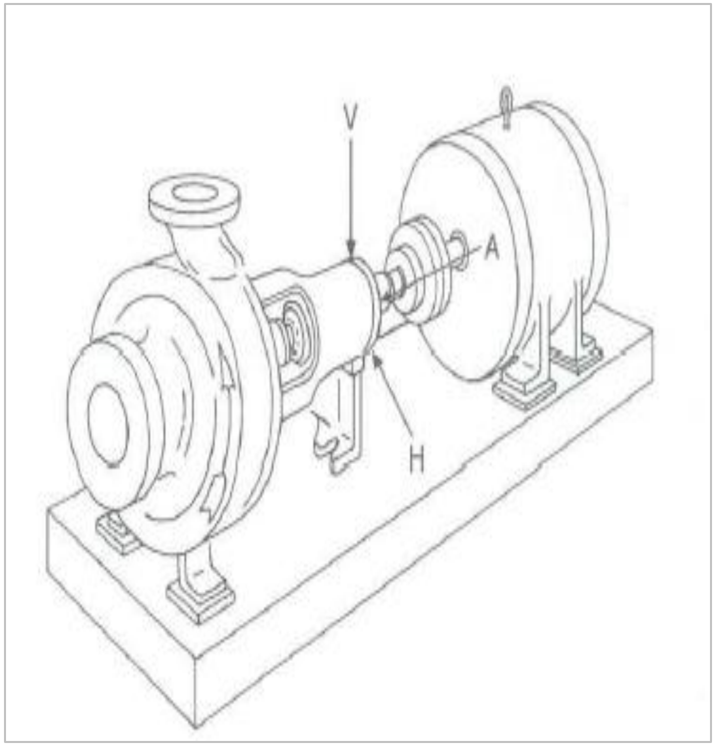


BETWEEN BEARING
SINGLE
OR
MULTI-STAGE



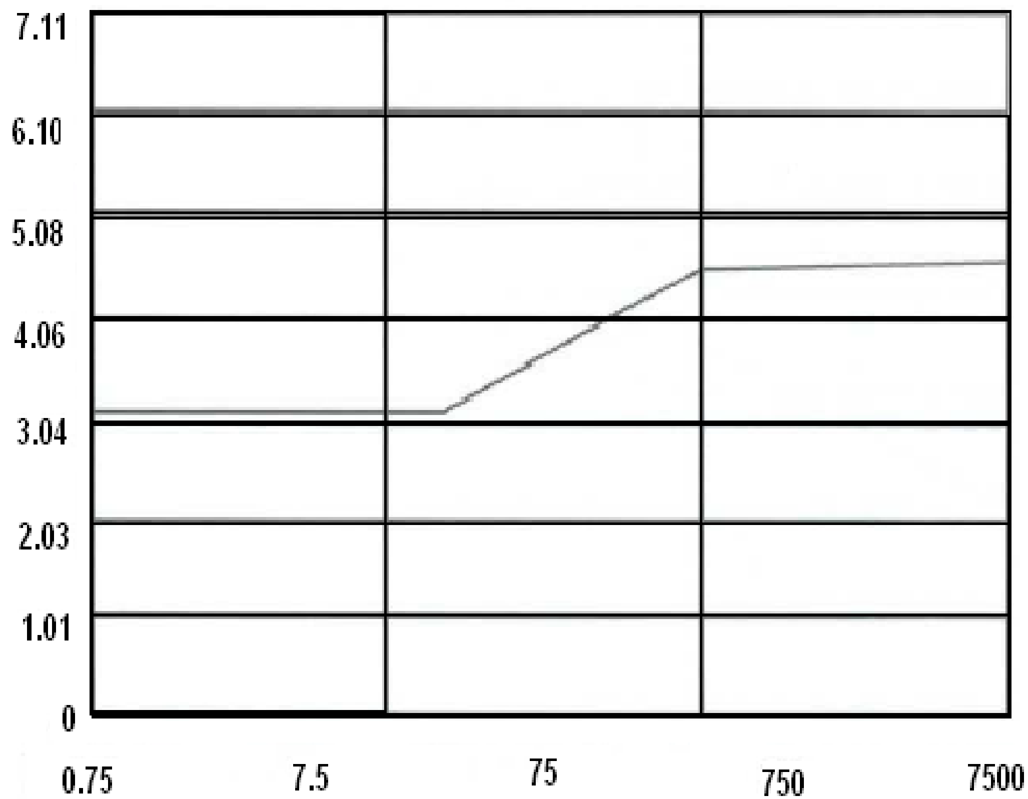
INPUT POWER AT TEST CONDITIONS - KW

TYPICAL VIBRATION CHART FOR SPLIT-CASE PUMP



END-SUCTION
FOOT-MOUNTED

VIBRATION - MM/SEC. RMS, UNFILTERED



INPUT POWER AT TEST CONDITIONS - KW

PUMP DOES NOT DELIVER RATED QUANTITY

PROBABLE FAULT	REMEDY
Air vapour lock in suction line	Stop pump and re-prime
Inlet of suction pipe insufficiently submerged	Ensure adequate supply of liquid
Pump not up to rated speed	Increase speed
Air leaks in suction line or gland arrangement	Make good any leaks or repack gland
Foot valve or suction strainer choked	Clean foot valve or strainer
Restriction in delivery pipe-work or pipe-work incorrect	Clear obstruction or rectify error in pipe-work
Head underestimated	Check head losses in delivery pipes, bends and valves, reduce losses as required
Unobserved leak in delivery	Examine pipe-work and repair leak
Blockage in impeller casing	Remove half casing and clear obstruction
Excessive wear at neck rings or wearing plates	Dismantle pump and restore clearances to original dimensions
Impeller damaged	Dismantle pump and renew impeller
Pump gaskets leaking	Renew defective gasket

PUMP DOES NOT DELIVER LIQUID	PROBABLE FAULT	REMEDY
	Impeller rotating in wrong direction	Reverse direction of rotation
	Pump not properly primed – air or vapour lock in suction line	Stop pump and re prime
	Inlet of suction pipe insufficiently submerged	Ensure adequate supply of liquid
	Air leaks in suction line or gland arrangement	Make good any leaks or repack gland
	Pump not up to rated speed	Increase speed

PUMP DOES NOT GENERATE ITS RATED DELIVERY PRESSURE	PROBABLE FAULT	REMEDY
	Impeller rotating in wrong direction	Reverse direction of rotation
	Pump not up to rated speed	Increase speed
	Impeller neck rings worn excessively	Dismantle pump and restore clearances to original dimensions
	Impeller damaged or chocked	Dismantle pump and renew impeller or clear blockage
	Pump gaskets leaking	Renew defective gaskets

	PROBABLE FAULT	REMEDY
PUMP LOSES LIQUID AFTER STARTING	Suction line not fully primed – air or vapour lock in suction line	Stop pump and reprime
	Inlet of suction pipe insufficiently submerged	Ensure adequate supply of liquid at suction pipe inlet
	Air leaks in suction line or gland arrangement	Make good any leaks or renew gland packing
	Liquid seal to gland arrangement logging ring (if fitted) chocked	Clean out liquid seal supply
	Logging ring not properly located	Unpack gland and locate logging ring under supply orifice

	PROBABLE FAULT	REMEDY
IRREGULAR DELIVERY	Air or vapour lock in suction	Stop pump and reprime
	Fault in driving unit	Examine driving unit and make good any defects
	Air leaks in suction line or gland arrangement	Make good any leaks or repack gland
	Inlet of suction pipe insufficiently immersed in liquid	Ensure adequate supply of liquid at suction pipe inlet

	PROBABLE FAULT	REMEDY
EXCESSIVE NOISE LEVEL	Air or vapour lock in suction line	Stop pump and reprime
	Inlet of suction pipe insufficiently submerged	Ensure adequate supply of liquid at suction pipe inlet
	Air leaks in suction line or gland arrangement	Make good any leaks or repack gland
	Worn or loose bearings	Disconnect coupling and realign pump and driving unit
	Rotating element shaft bent	Dismantle pump, straighten or renew shaft
	Foundation not rigid	Dismantle pump and driving unit, strengthen foundation

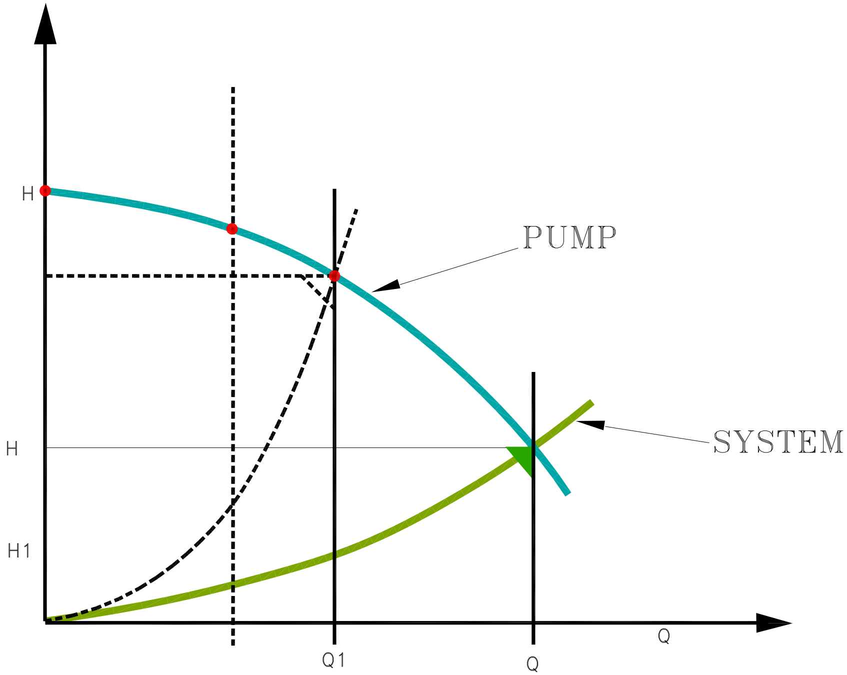
PROBABLE FAULT		REMEDY
PUMP OVERLOADS DRIVING UNIT	Pump gaskets leaking	Renew defective gasket
	Serious leak in delivery line, pump delivering more than its rated quantity	Repair leak
	Speed too high	Reduce Speed
	Impeller neck rings worn excessively	Dismantle pump and restore clearances to original dimensions
	Gland packing too tight	Stop pump, close delivery valve to relieve internal pressure on packing, slacken back the gland nuts and retighten to finger tightness
	Impeller damaged	Dismantle pump and renew impeller
	Mechanical tightness of pump internal components	Dismantle pump, check internal clearances and adjust as necessary
	Pipe work exerting strain on pump	Disconnect pipe work and realign to pump

	PROBABLE FAULT	REMEDY
BEARING OVERHEATING	Pump and driving unit out of alignment	Disconnect coupling and realign pump and driving unit
	Oil level too low or too high	Replenish with correct grade of oil or drain down to correct level
	Wrong grade of oil	Drain out bearing, flush through bearings; refill with correct grade of oil
	Dirt in bearing	Dismantle, clean out and flush through bearings; refill with correct grade of oil
	Moisture in oil	Drain out bearing, flush through and refill with correct grade of oil. Determine cause of contamination and rectify
	Bearings too tight	Ensure that bearings are correctly bedded to their journals with the correct amount of oil clearance. Renew bearings if necessary
	Too much grease in bearing	Clean out old grease and repack with correct grade and qty of grease
	Pipe work exerting strain on pump	Disconnect pipe work and realign to pump

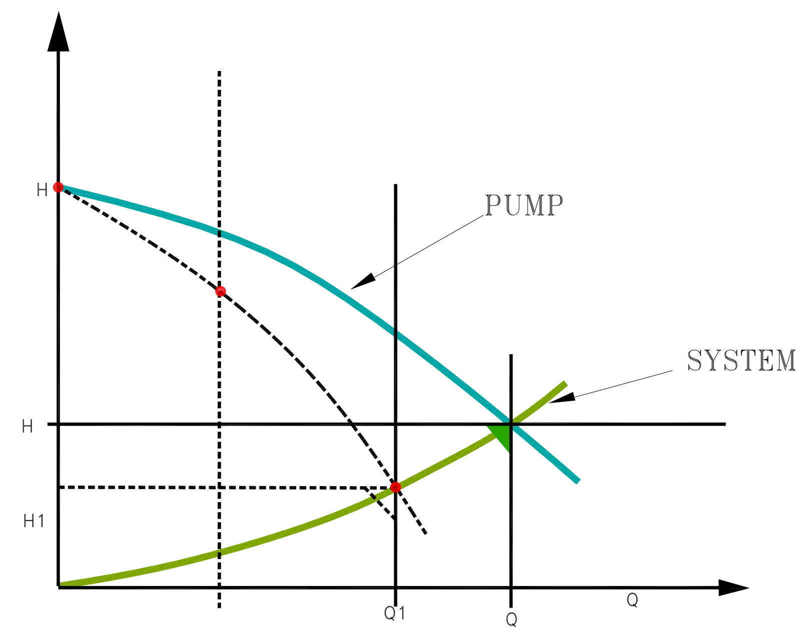
EXCESSIVE VIBRATION	PROBABLE FAULT	REMEDY
	Air or vapour lock in suction	Stop pump and reprime
	Inlet of suction pipe insufficiently submerged	Ensure adequate supply of liquid at suction pipe inlet
	Pump and driving unit incorrectly aligned	Decouple pump and driver, realign & check alignment after coupling.
	Worn or loose bearings	Dismantle pump and renew bearings
	Impeller chocked or damaged	Dismantle pump and clear or renew impeller
	Rotating element shaft bent	Dismantle pump, straighten or renew shaft
	Foundation not rigid	Remove pump, strengthen the foundation and reinstall pump
	Coupling damaged	Renew coupling
	Pipe work exerting strain on pump	Disconnect pipe work and realign to pump

BEARING WEAR

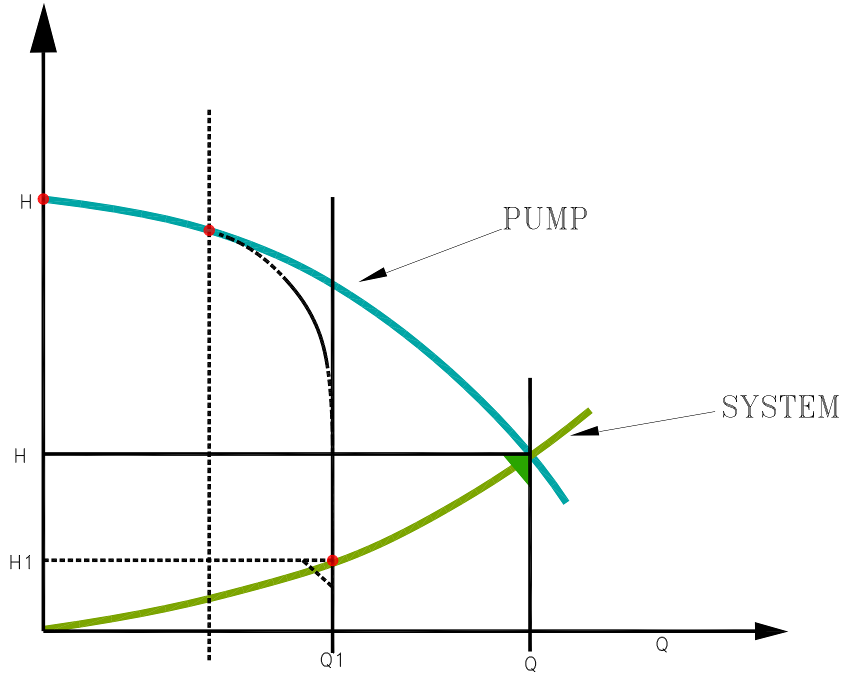
PROBABLE FAULT	REMEDY
Pump and driving unit out of alignment	Disconnect coupling and realign pump and driving unit. Renew bearings if necessary
Rotating element shaft bent	Dismantle pump, straighten or renew shaft. Renew bearings if necessary
Dirt in bearings	Ensure that only clean oil is used to lubricate bearings. Renew bearings if necessary. Refill with clean oil
Lack of lubrication	Ensure that oil is maintained at its correct level or that oil system is functioning correctly. Renew bearings if necessary
Bearing badly installed	Ensure that bearings are correctly bedded to their journals with the correct amount of oil clearance. Renew bearings if necessary
Pipe work exerting strain on pump	Ensure that pipe work is correctly aligned to pump. Renew bearings if necessary
Excessive vibration	Refer excessive vibration



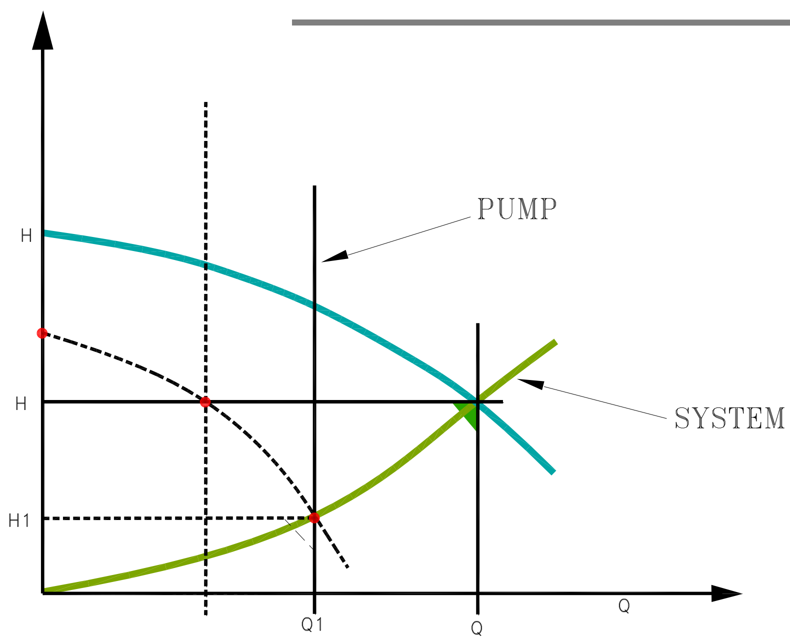
Symptom	Diagnosis
CV As per pump curve Open valve Q1 <Q, H1> H Q1, H1 on pump curve	Changed system condition – blockage pipe friction, filters, strainers, etc



Symptom	Diagnosis
CV As per pump curve Open valve Q1 <Q, H1> H	Pump fault – Blockage in impeller, increased leakage loss



Symptom	Diagnosis
CV As per pump curve Open valve Head lower in vicinity of the system curve. Sudden break down of H-Q	Insufficient NPSH leading to cavitation break-down



Symptom	Diagnosis
CV Lower than pump curve Lower Q, Lower H	Incorrect speed Incorrect diameter of impeller Wrong direction of rotation

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
NO DELIVERY OR DELIVERY NOT UPTO THE EXPECTATION	➢ PUMP NOT PRIMED	➢ FILL THE PUMP & SUCTION LINE COMPLETELY WITH LIQUID ➢ REMOVE AIR/GAS ➢ ELIMINATE HIGH POINTS IN SUCTION PIPING ➢ CHECK FOR FAULY FOOT VALVE, CHECK VALVE INSTALLATION
	➢ AIR-POCKET IN SUCTION LINE	➢ CHECK FOR GAS, AIR IN SYSTEM/SUCTION LINE ➢ INSTALL GAS SEPERATION CHAMBER ➢ CHECK FOR AIR-LEAKAGE ➢ OPEN AIR-VENT VALVE IF ANY
	➢ INSUFFICIENT IMMERSION OF SUCTION PIPE, VORTEXING	➢ LOWER SUCTION PIPE OR RAISE SUMP WATER LEVEL
	➢ SPEED OF PUMP TOO LOW OR WRONG DIRECTION OF ROTATION	➢ CORRECT SPPED, CHECK RECORDS FOR PROPER SPEED ➢ CHECK ROTATION WITH ARROW ON CASING - REVERSE POLARITY ON MOTOR ➢ CHECK IMPELLER
	➢ SYSTEM HEAD HIGHER THAN PUMP DESIGN HEAD	➢ DECREASE SYSTEM RESISTANCE ➢ CHECK DESIGN PARAMETERS ➢ INCREASE PUMP SPEED ➢ INSTALL PROPER SIZE PUMP

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
INSUFFICIENT DISCHARGE PRESSURE	➢WRONG IMPELLER SELECTION	➢ VERIFY PROPER IMPELLER SIZE
	➢ WRONG IMPELLER INSTALLATION	➢ CHECK IF THE IMPELLER IS INSTALLED BACKWARD(DOUBLE SUCTION PUMP)
	➢ AIR-GAS ENTRAINMENT IN LIQUID	➢CHECK FOR GAS, AIR IN SYSTEM/SUCTION LINE ➢ INSTALL GAS SEPERATION CHAMBER ➢ CHECK FOR AIR-LEAKAGE ➢ OPEN AIR-VENT VALVE IF ANY
	➢ SPEED OF PUMP TOO LOW OR WRONG DIRECTION OF ROTATION	➢ CORRECT SPPED, CHECK RECORDS FOR PROPER SPEED ➢ CHECK ROTATION WITH ARROW ON CASING - REVERSE POLARITY ON MOTOR ➢ CHECK IMPELLER
	➢IMPELLER CLOGGED	➢ CHECK FOR DAMAGE & CLEAN
	➢ IMPROPER PUMP SELECTION	➢ DECREASE SYSTEM RESISTANCE ➢ CHECK DESIGN PARAMETERS ➢ INCREASE PUMP SPEED ➢ INSTALL PROPER SIZE PUMP

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
SHORT SEAL LIFE	➢ MISALIGNMENT	➢ CHECK ANGULAR & PRALLEL ALIGNMENT BETWEEN PUMP & DRIVER ➢ ELIMINATE STILT-MOUNTED BASE-PLATE ➢ CHECK FOR LOOSE MOUNTING ➢ CHECK FOR UNUNIFORM THERMAL EXPANSION OF PUMP PARTS
	➢ BENT SHAFT	➢ CHECK TIR AT IMPELLER END (SHOULD NOT EXCEED 0.002”) ➢ REPLACE SHAFT OR BEARING IF NECESSARY
	➢ CASING DISTORSION DUE TO PIPE STRAIN	➢ CHECK ORIENTATION OF BEARING ADAPTER ➢ CHECK FOR PIPE ALIGNMENT & ANALYZE PIPE LOADS & SUPPORTS
	➢ PUMP CAVITATING	➢CHECK FOR NPSHa/NPSHr MARGIN & TAKE NECESSARY STEPS ➢ CHECK FOR FLASH POINT MARGIN ➢ CHECK FOR GAS ENTRAINMENT
	➢ IMPROPER OPERATING CONDITION	➢ INSTALL PROPER SEAL THAT SUITS PUMP OPERATING CONDITIONS
	➢ UNBALANCE DRIVER	➢ RUN DRIVER DISCONNECTED FROM PUMP UNIT - PERFORM VIBRATION ANALYSIS

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
SHORT BEARING LIFE	➢ BEARING FAILURES	➢ CHECK FOR PROPER LUBRICATION & CONTAMINATION OF LUBRICANT ➢ CHECK FOR PROPER BEARING INSTALLATION ➢ CHECK FOR THE SUITABILITY OF BEARING SELECTED
	➢ MISALIGNMENT	➢ CHECK ANGULAR & PRALLEL ALIGNMENT BETWEEN PUMP & DRIVER ➢ ELIMINATE STILT-MOUNTED BASE-PLATE ➢ CHECK FOR LOOSE MOUNTING ➢ CHECK FOR UNUNIFORM THERMAL EXPANSION OF PUMP PARTS
	➢ BENT SHAFT	➢ CHECK TIR AT IMPELLER END (SHOULD NOT EXCEED 0.002”) ➢ REPLACE SHAFT OR BEARING IF NECESSARY
	➢ CASING DISTORSION DUE TO PIPE STRAIN	➢ CHECK ORIENTATION OF BEARING ADAPTER ➢ CHECK FOR PIPE ALIGNMENT & ANALYZE PIPE LOADS & SUPPORTS
	➢ PUMP CAVITATING	➢CHECK FOR NPSHa/NPSHr MARGIN & TAKE NECESSARY STEPS ➢ CHECK FOR FLASH POINT MARGIN ➢ CHECK FOR GAS ENTRAINMENT
	➢ UNBALANCE DRIVER	➢ RUN DRIVER DISCONNECTED FROM PUMP UNIT - PERFORM VIBRATION ANALYSIS

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
EXCESSIVE POWER DEMAND	➢ MOTOR TRIPPING-OFF	➢CHECK STARTER ➢ CHECK RELAY SETTING ➢ CHECK FOR THE SUITABILITY OF MOTOR SELECTED FOR CURRENT OPERAING CONDITION
	➢SPEED TOO HIGH	➢ CHECK FOR SPEED OR PREVIOUS RECORDS FOR PROPER SPEED ➢ ELIMINATE STILT-MOUNTED BASE-PLATE ➢ CHECK FOR LOOSE MOUNTING ➢ CHECK FOR UNUNIFORM THERMAL EXPANSION OF PUMP PARTS
	➢ ROTOR IMPELLER RUBBING ON CASING	➢ LOOSE IMPELLER FIT ➢ WRONG ROTATION ➢ REPLCE IF SHAFT IS BENT ➢ HIGH NOZZLE LOADS ➢ VERY SMALL INTERNAL RUNNING CLEARANCES – CHECK FOR NECK RING DIMENSIONS
	➢ PUMP NOT DESIGNED FOR LIQUID DENSITY & VISCOSITY BEING PUMPED	➢ CHECK DESIGN SP. GRAVITY ➢CHECK MOTOR SIZE – USE LARGER DRIVER OR CHANGE PUMP TYPE ➢ HEAT UP THE LIQUID TO REDUCE VISCOSITY
	➢ BEARING FAILURES	➢ CHECK FOR PROPER LUBRICATION & CONTAMINATION OF LUBRICANT ➢ CHECK FOR PROPER BEARING INSTALLATION ➢ CHECK FOR THE SUITABILITY OF BEARING SELECTED
	➢ IMPROPER COUPLING SELECTION	➢ CHECK COUPLING SIZE

TROUBLE SHOOTING

SYMTOMPS	COMMON CAUSES	REMEDY
NOISE & VIBRATION	➢ PUMP IS CAVITATING	➢ CHECK FOR NPSHa/NPSHr MARGIN & TAKE NECESSARY STEPS ➢ CHECK FOR FLASH POINT MARGIN ➢ CHECK FOR GAS ENTRAINMENT
	➢ SUCTION OR DISCHARGE VALVE CLOSED OR PARTIALLY CLOSED	➢ CHECK FOR VALVE CONDITION ➢ OPEN THE VALVES
	➢ MISALIGNMENT	➢ CHECK ANGULAR & PRALLEL ALIGNMENT BETWEEN PUMP & DRIVER ➢ ELIMINATE STILT-MOUNTED BASE-PLATE ➢ CHECK FOR LOOSE MOUNTING ➢ CHECK FOR UNUNIFORM THERMAL EXPANSION OF PUMP PARTS
	➢ INADEQUATE GROUTING OF BASE PLATE	➢ CHECK GROUTING, CONSULT PROCESS INDUSTRY PRACTICE RE-IE-686 ➢ IF STILT MOUNTED, GROUT BASEPLATE
	➢ BEARING FAILURES	➢ CHECK FOR PROPER LUBRICATION & CONTAMINATION OF LUBRICANT ➢ CHECK FOR PROPER BEARING INSTALLATION ➢ CHECK FOR THE SUITABILITY OF BEARING SELECTED
	➢ IMPROPER COUPLING SELECTION	➢ CHECK COUPLING SIZE, GRAESING , ALIGNMENT