Comprehensive Pump Series - Best Practices of Centrifugal Pumps

from

Central Air conditioning and Trading and Services
• MAJOR PUMP COMPONENTS
• FUNCTIONS OF COMPONENTS
• DESIGN CALCULATIONS ON SHAFT DEFLECTIONS
  • Losses in pumps
  • Pump selection Basic
  • Trouble shooting
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
- IMPPELLER 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

- Discharge Nozzle
- Volute Casing
- Shaft
- Suction Nozzle
- Seal Gland
- Bearing Housing
- Seal Flush Line
MAJOR COMPONENTS OF A SPLIT CASE PUMP

- CASING
- IMPELLER
- WEAR RING
- STUFFING BOX UNIT
- BEARING BRACKET
- PACKING
- SLEEVE
- LANTERN RING
# FUNCTIONS OF CENTRIFUGAL PUMP COMPONENTS

<table>
<thead>
<tr>
<th>COMPONENTS</th>
<th>FUNCTIONS</th>
</tr>
</thead>
<tbody>
<tr>
<td>IMPELLER</td>
<td>➢ TO DEVELOP DYNAMIC HEAD ➢ TO CONVERT KINETIC ENERGY INTO PRESSURE ENERGY WITH MINIMUM HYDRAULIC LOSSES BY MEANS OF VOLUTE, DIFFUSERS OR GUIDE VANES ➢ INCORPORATES NOZZLES TO CONNECT SUCTION &amp; DISCHARGE PIPING ➢ DIRECTS FLOW INTO &amp; OUT OF THE IMPELLER. ➢ PROVIDES SUPPORT TO THE BEARING BRACKET</td>
</tr>
<tr>
<td>WEAR RING</td>
<td>➢ 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</td>
</tr>
<tr>
<td>IMPELLER NUT</td>
<td>➢ TO LOCK THE IMPELLER IN ITS PROPER AXIAL POSITION ➢ TO PREVENT AXIAL MOVEMENT DUE TO HYDRAULIC THRUST</td>
</tr>
<tr>
<td>COMPONENTS</td>
<td>FUNCTIONS</td>
</tr>
<tr>
<td>------------------</td>
<td>-----------------------------------------------------------------------------------------------------</td>
</tr>
</tbody>
</table>
| **SHAFT**        | ➢ TRNSMITS TORQUE TO THE IMPELLER FROM THE DRIVER  
➢ SUPPORTS IMPELLER AND OTHER ROTATING ELEMENTS                                                   |
| **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 NUT**   | ➢ TO FASTEN THE SLEEVES TO THE SHAFT  
➢ TO PREVENT MOVEMENT OF THE SLEEVE                                                                  |
| **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                   |
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

<table>
<thead>
<tr>
<th>Closed Impeller</th>
<th>Semi Open</th>
<th>Open</th>
</tr>
</thead>
<tbody>
<tr>
<td><em>This type of impeller works best with clear water.</em></td>
<td><em>Semi open impellers are used for fibers or potentially clogging materials in the pumping liquid</em></td>
<td><em>Open impellers are used for high speed pumps of over 10,000 rpm.</em></td>
</tr>
<tr>
<td><em>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.</em></td>
<td><em>The impeller requires tight clearance be maintained between the open face and its mating stationary surface (clearance between 0.25 and 0.38 mm)</em></td>
<td></td>
</tr>
</tbody>
</table>
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

Polar View of Impeller
(View From Arrow 'Z')
MAJOR DIMENSIONS OF AN IMPPELLER
Double Entry Impeller

Distance From Impeller Inlet To Impeller Center-Line
Outlet Width
Shroud Thickness
Z

Hub Dia.
Front Neck Dia.
Shaft Dia. At Impeller
Eye Dia.
Back Neck Dia.
Impeller Nominal Dia.

Plan View of Vane at Periphery

Vane Thickness at Outlet

Rotation

Hub Length
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

RADIAL VANE
FRANCIS VANE
MIXED FLOW
AXIAL FLOW

\[
\frac{D_2}{D_1} > 4
\]

\[
\frac{D_2}{D_1} = 1.5 \text{ to } 2
\]

\[
\frac{D_2}{D_1} < 1.5
\]

\[
\frac{D_2}{D_1} = 1
\]
TYPES OF IMPELLERS BASED ON THEIR RELATIVE POSITIONS ON THE SHAFT

OVER-HUNG IMPELLER

IMPELLER BETWEEN BEARINGS
TWO-STAGE PUMP ROTATING ELEMENT ASSEMBLY

BEARING HOUSING

STUFFING BOX BUSHING

CASE WEAR RING

IMPELLER

BEARING HOUSING

GREASE FILLER PLUG

LANTERN RING

SHAFT

SHAFT SLEEVE

INTER-STAGE BUSHING
Multistage Pumps

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

\[ P = K \times H \times D_2 \times b_2 \times \text{S.G} / 2.31 \]

- **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.
VOLUTE SECTIONS FOR SINGLE VOLUTE PUMP

- Impeller Diameter
- "Tongue" or "Cut-water" Diameter
- Throat Area
- Sectional View at Cut-Water (T')

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 M³/HR.(400 US GPM).
SAME THROAT AREA FOR BOTH THE VOLUMES
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 into the pump discharge or the next pump stage.
Circular (concentric) casing

An Impeller discharges into 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

\[ t = \frac{p \cdot r}{f - 0.6p} + 3 \]

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

\( r = \) casing internal radius

\( f = \) permissible stress = 0.25 × UTS × 0.8

For, Grade 14 (FG220) = 0.25 × 14 × 0.8 × 2240 = 6272 psi

Grade 17 (FG 260) = 7616 psi

GGG50 = 13600 psi
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 accommodate 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 (10th 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 \( \Delta \) = \( \frac{P}{6EI_c} \left( C^3 + \frac{(C+B)^3 - C^3}{K_2} + \frac{(A+B+C)^3 - (B+C)^3}{K_3} \right) \)

\( \Delta \) is in mm.

- **P** = load at the impeller: \( (P) \)
- **E** (Modulus of elasticity) = 200 GPa = \( 200 \times 10^6 \) Kpa
- **A** = length of shaft at section A
- **B** = length of shaft at section B
- **C** = length of shaft at section C
- K2, K3 can be calculated as below,

  \[ K_2 = \frac{I_b}{I_c} \]
  \[ K_3 = \frac{I_a}{I_c} \]

- Determination of moment of inertia of each section:

  \[ I_a = \text{moment of inertia} = \pi \times D_a^4/64 \]
  \[ I_b = \text{moment of inertia} = \pi \times D_b^4/64 \]
  \[ I_c = \text{moment of inertia} = \pi \times D_c^4/64 \]

  Da= Dia of the shaft at section A
  Db= Dia of the shaft at section B
  Dc= Dia of the shaft at section C
Determination of load at the impeller: (P)

- \( Wi = \) Wgt of the impeller
- \( Ws = \) Wgt of the shaft
- \( W = \) static load of rotor = \( Wi + Ws \)

\[ P (\text{load at the impeller}) = W + R \]
Calculate Radial Thrust:

\[ R = K_r \times H \times \rho \times g \times B_2 \times D_2 \]

Where,

\( K_r \) = thrust factor which varies with rate of flow and specific speed

\( H \) = Developed head per stage in m

\( \rho \) = density of the liquid in kg/m\(^3\)

Gravitational constant = 9.81 m/s\(^2\)

\( D_2 \) = Impeller dia in m

\( B_2 \) = Impeller width at discharge including shrouds in m

Note: \( H \times S / 2.31 \) in US units
Calculate specific speed $Ns$:

$$Ns \text{ (US units)} = \frac{N \times (Q \times 4.403)^{.5}}{(H \times 3.28)^{.75}}$$

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:

<table>
<thead>
<tr>
<th>Duty 50%</th>
<th>$Ns(Us)$</th>
<th>$Ns$(metric)</th>
<th>$Kr$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Double volute</td>
<td>500</td>
<td>600</td>
<td>0.02</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>1200</td>
<td>0.025</td>
</tr>
<tr>
<td></td>
<td>1500</td>
<td>1250</td>
<td>0.035</td>
</tr>
<tr>
<td></td>
<td>2000</td>
<td>2300</td>
<td>0.048</td>
</tr>
<tr>
<td></td>
<td>3500</td>
<td>4000</td>
<td>0.05</td>
</tr>
</tbody>
</table>
Bearing load calculation – Ref- Hydraulic Institute

\[ R_1 = \frac{F_n \times (X + Y)}{Y} \]
\[ R_2 = \frac{F_n \times X}{Y} \]

Where,
- \( R_1 \) – Inboard reaction load, \( R_2 \) – Outboard bearing reaction load
- \( F_n = F_r + W \); Where
- \( F_n \) – Net force typically applied at the impeller center line,
- \( F_r \) – 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

\[ R1 = \frac{F_n \times Y}{X+Y} \]

\[ R2 = \frac{F_n \times X}{X+Y} \]

Where,

- **R1** – Inboard reaction load, **R2** – Outboard bearing reaction load
- \( F_n = Fr + W \); Where
- \( F_n \) – 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 inboard and outboard bearing
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.
Radial thrust is a function of total head of the pump and width & diameter of the impeller.

P = K x H x D₂ x b₂ x S.G / 2.31

P = radial thrust in pounds
K = thrust factor
H = developed head/stage in ft
D₂ = impeller dia, in inches
b₂ = impeller width in inches (including shrouds)
Figure 5-2. Radial thrust factor.
RADIAL FORCE FOR VARIOUS TYPES OF VOLUTES

RADIAL FORCE – PERCENT OF FORCE AT SHUT-OFF FOR VOLUTE CASING

CAPACITY - PERCENT OF NORMAL

- STANDARD VOLUTE
- DOUBLE VOLUTE
- MODIFIED CONCENTRIC CASING
AXIAL THRUST

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

SEVERITY OF AXIAL THRUST 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.

\[
Ns = N \times \frac{\sqrt[0.75]{Q}}{H}
\]

**WHERE**,

- **Ns** = SPECIFIC SPEED IN METRIC UNITS.
- **Q** = FLOW IN M³/HR. AT B.E.P.
- **N** = ROTATIVE SPEED IN R.P.M.
- **H** = HEAD DEVELOPED IN M. AT B.E.P.

**US Customary Units**

\[
Ns = N \times \frac{\sqrt[0.75]{Q}}{H}
\]

**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 ACCEPTABLE OPERATING ZONE OF A PUMP

LOW SPECIFIC SPEED PUMPS

- Comparatively narrower acceptable operating zone
- B.E.P

HIGH SPECIFIC SPEED PUMPS

- Comparatively wider acceptable operating zone
- B.E.P
SPECIFIC SPEED & ITS EFFECT ON PUMP PERFORMANCE CHARACTERISTICS

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.

<table>
<thead>
<tr>
<th>Specific Speed</th>
<th>POR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metric &lt; 5200</td>
<td>US Units &lt; 4500</td>
</tr>
<tr>
<td>Metric &gt; 5200</td>
<td>US Units &gt; 4500</td>
</tr>
</tbody>
</table>
**SUCTION SPECIFIC SPEED**

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

\[
Nss = N \times \sqrt{Q} / (NPSHr)^{0.75}
\]

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)

\[
Nss = N \times \sqrt{Q} / (NPSHr)^{0.75}
\]

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,**

\[ n = 10,000 \times \text{npsa}^{0.75} / \text{q}^{0.5} \]

where, \( n \) = max. speed (r.p.m).
\( \text{npsa} \) = npsh available in m.
\( \text{q} \) = flow in m³/hr. (take half of the flow for double suction pump).

**In us customary units,**

\[ n = 8,500 \times \text{npsa}^{0.75} / \text{q}^{0.5} \]

where, \( n \) = max. speed (r.p.m).
\( \text{npsa} \) = npsh available in ft.
\( \text{q} \) = 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.
<table>
<thead>
<tr>
<th>NPSHr</th>
<th>NPSHa</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>NPSHr</strong> IS A CHARACTERISTIC OF YOUR PUMP.</td>
<td><strong>NPSHa</strong> IS A CHARACTERISTIC OF YOUR SYSTEM.</td>
</tr>
<tr>
<td>THIS IS A FUNCTION OF PUMP SUCTION DESIGN. IT VARIES WITH THE SPEED &amp; CAPACITY FOR A PARTICULAR PUMP.</td>
<td>THIS IS A FUNCTION OF SYSTEM CONFIGURATION ON THE SUCTION SIDE OF THE PUMP.</td>
</tr>
<tr>
<td>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 &amp; VANE INLET PASSAGES TO MAINTAIN THE PUMPING FLUID IN LIQUID STATE.</td>
<td>IT IS THE AVAILABLE TOTAL SUCTION HEAD IN METRES ABSOLUTE DETERMINED AT THE INLET NOZZLE OF THE PUMP &amp; CORRECTED TO THE PUMP DATUM LESS THE VAPOUR PRESSURE HEAD OF THE LIQUID IN METRES ABSOLUTE AT THE PUMPING TEMPERATURE.</td>
</tr>
</tbody>
</table>

FOR CAVITATION-FREE SAFE OPERATION, YOU’VE TO KEEP, $\textbf{NPSHa} > \textbf{NPSHr}$. 
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 vaporize at a low pressure point in the pump.

\[ \text{NPSHA} = \text{Pressure acting on surface (Pa)} \pm \text{Static suction head (Hs)} - \text{Pressure drop (frictional head loss) (Hf)} - \text{Vapour pressure (Hvp)} \]
**CALCULATION OF NPSH\textsubscript{A} FOR SYSTEMS WITH SUCTION HEAD & SUCTION LIFT**

\[
\text{NPSH}_a (M) = \text{ATMOSPHERIC PRESSURE}(M) - \text{SUCTION LIFT}(M) - \text{FRICTIONAL HEAD LOSS}(M) - \text{V. P} (M)
\]

\[
\text{NPSH}_a (M) = \text{ATMOSPHERIC PRESSURE}(M) + \text{SUCTION HEAD}(M) - \text{FRICTIONAL HEAD LOSS}(M) - \text{V. P} (M)
\]
**REASONS FOR SAFETY 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.**

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

**A GUIDE FOR SELECTING SAFETY MARGIN – ISO 5199**

<table>
<thead>
<tr>
<th>MOTOR RATING</th>
<th>MARGIN OF SAFETY (% OF MOTOR RATING)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 kW TO 100 kW</td>
<td>135% TO 110%</td>
</tr>
<tr>
<td>ABOVE 100 kW</td>
<td>110%</td>
</tr>
</tbody>
</table>
TYPICAL SPEED TORQUE CURVE WITH TVA DATA

MOMENT OF INERTIA \((Gd^2) = 11.2 \text{ Kg}\cdot\text{m}^2\)
TORSIONAL STIFFNESS = \(1.39 \times 10^5 \text{ Nm/rad}\)

100\% - TORQUE = 1179 \text{ Nm.}
100\% SPEED = 2600 RPM

**BASED ON: 600m/hr@140m@2600RPM**
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.
Figure 5.2: Increase in power consumption due to losses.
LOCATIONS OF VARIOUS LOSSES IN A CENTRIFUGAL PUMP

- Entrance shock loss
- Leakage loss
- Casing Hydraulic loss
- Disk friction loss
- Mechanical losses
LOCATIONS OF VARIOUS LOSSES IN A CENTRIFUGAL PUMP

- Casing
- Hydraulic loss
- Leakage loss
- Entrance shock loss
- Mechanical losses
- Disk friction loss
The pump shaft consists of bearings, shaft seals, and gear depending on pump type. These components all cause 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 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**

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

2) **Shock losses** *leaving the impeller*

3) **Losses** during *conversion of mechanical energy to kinetic energy*
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:**

\[ H_f = f \times \frac{v^2}{2g} \times \frac{L}{D_H} \]

**WHERE,**

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

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.

\[ k = 7.3 \times 10^{-4} \left( \frac{2v \times 10^6}{U_2 D_2} \right)^m \]

\( M \) = exponent equals 1/6 for smooth surface, 1/7 to 1/9 for rough surface,
- \( v \) = kinematic viscosity -m/sec
- \( U_2 \) = Peripheral velocity [m/s], \( D_2 \) = Impeller diameter [m]
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**

\[ Q_L = \mu \times A_{cl} \times \sqrt{2g \cdot \Delta H_{cl}} \]

**HERE,**
- \( Q_L \) = LEAKAGE FLOW
- \( \mu \) = LEAKAGE GAP LOSS CO-EFFICIENT
- \( g \) = GRAVITATIONAL ACCELERATION
- \( \Delta H_{cl} \) = HEAD LOSS ACROSS ANNULAR PATH
- \( A_{cl} \) = AREA AT CLEARANCE ZONE
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

NSS = \( \frac{Ns \times Q^{0.5}}{NPSHr^{0.75}} \)

\[
\text{Speed (NS)} = \frac{\text{NSS} \times \text{NPSH A}^{0.75}}{Q^{0.5}}
\]

NSS = 7500 TO 10000

Assumed - 8500 in US units

\[\text{Ns=} \quad 8500 \times (20)^{0.75} \]
\[\quad (1651)^{0.5} \]

2004

Speed = RPM

The recommended motor speed is - 1450 RPM /4 Pole

3. Motor Power

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

\[\text{BKW} = \frac{750 \times 35 \times 1}{367 \times 0.86} \]
\[85.54 \text{ KW} \]

ADD 15% Margin = 85.54 \times 1.15 = 98 \text{ KW} , So recommendable is 110 KW/1450 RPM/50 Hz

(Efficiency is taken from the HI chart)
4. NPSH Required for the pump

\[ NSS = \frac{NS \times Q^{0.5}}{\text{NPSHr}^{0.75}} \]

\[ \text{NPSH}^{0.75} = \frac{Ns \times Q^{0.5}}{\text{Nss}} \]

\[ \text{NPSH R} = \frac{1500 \times (1651)^{0.5}}{8500} \]

\[ \text{NPSH R} = 7.17 \text{ FT} \]

\[ \text{NPSH R} = 2.1 \text{ Meters} \]

NPSH A is greater than NPSH R

5. Minimum shaft diameter at coupling area.

Shear stress formula = \[ HP = \frac{S \times N(D)^3}{321000} \]

\( S \)- Permissible shear stress in shaft- \( \text{PSI} = 8500 \) (SS410)

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

\[ D = 1.577 \text{ Inches} = 39.55 \text{ mm at coupling area} \]
6. Suction and delivery nozzle size:

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

**Capacity = Area X velocity**

Area \( \left( \frac{3.147}{d^2} \right) \) = capacity / velocity

\[
d^2 = \frac{(750 \times 4)/4 \times 3600}{4}
\]

\( d = 0.456 \) 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:

\[ U_2 = KU \left( 2 \times g \times H \right)^{0.5} \]

\( KU = \) Coefficient related to specific speed - refer to the chart

for 2004 specific speed the \( KU \) value is 1.1

\[
U_2 = 1.1 \left( 2 \times 9.81 \times 35 \right)^{0.5}
\]

\( U_2 = 28.82 \) m/sec

Impeller diameter can be calculated by using the below formula,

\[ U_2 = \frac{3.147 \times D_2 \times N}{60} \]

\[
D_2 = \frac{(28.82 \times 60)}{3.147 \times 1500}
\]

\( D_2 = 0.366 \) MTRS = 366 MM.
Efficiency Chart - II
Optimum Efficiency as a Function of Specific Speed & Flow-rate

Fraser-Sabini Chart

Specific Speed & Efficiency

VALUE OF SPECIFIC SPEED, Ns (US UNITS)

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

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

- Three identical pumps designed to develop same head produces different pressures which vary in proportion to their specific gravity.

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

- Three pumps produce same discharge pressure but develop heads inversely proportional to their specific gravity.

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

- Pumps should be specified in terms of head and not in terms of pressure to avoid ambiguity.

- All pressures can be visualized as being caused by the weight of a column of liquid at its base.

\[ \text{PRESSURE} = \text{H x S.P. GRAVITY} \]
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

\[
\text{Pressure (Kgf/CM}^2) = \frac{\text{Head (M)}}{10} \times \text{S.P. Gravity}
\]

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.

Three identical pumps designed to develop same head produces different pressures which vary in proportion to their specific gravity.
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*

<table>
<thead>
<tr>
<th>L/s</th>
<th>M</th>
<th>RL/S</th>
</tr>
</thead>
<tbody>
<tr>
<td>944</td>
<td>28</td>
<td>0</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>188</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>283</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>377</td>
</tr>
<tr>
<td><strong>944</strong></td>
<td><strong>28</strong></td>
<td><strong>472</strong></td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>566</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>660</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>755</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>849</td>
</tr>
<tr>
<td>944</td>
<td>28</td>
<td>944</td>
</tr>
</tbody>
</table>

**Designed flow – 944 L/s**

**Designed Head – 28 Meters**

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

\[ H_2 = (Q_2^2) \times H_1 \]

\[ Q_1^2 \]

\[ H_2 = (472)^2 \times 28 \]

\[ = 7 \text{ Meters} \]

\[ = (944)^2 \]
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’

$$TDH = H_s + H_p + H_f + H_v$$

HERE, TDH = TOTAL DYNAMIC HEAD

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

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

$H_v$ = VELOCITY HEAD

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

THE SYSTEM HEAD CURVE $H_s$ & $H_p$ BEGINS AT ZERO FLOW AT THE SUM OF $H_s$ & $H_p$ & RISES EXPONENTIALLY WITH THE SQUARE OF THE FLOW

$H_s$ & $H_p$ EXISTS WHETHER THE PUMP IS RUNNING OR NOT WHEREAS $H_f$ & $H_v$ EXISTS ONLY AT THE RUN-TIME.
NATURE OF SYSTEM CURVES

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

- PUMP CURVE
- ALL FRICTION
- INTERMEDIATE
- STATIC DOMINANT
SYSTEM HEAD CURVE SUPERIMPOSED ON H-Q CURVE OF THE PUMP

H-Q CURVE
OPERATING POINT
SYSTEM RESISTANCE CURVE
FRICITION HEAD
STATIC HEAD
TOTAL DYNAMIC HEAD (TDH)
FLOW
SYSTEM HEAD CURVE SUPERIMPOSED ON H-Q CURVE OF THE PUMP

- H-Q Curve
- Operating Point
- System Resistance Curve
- Friction Head
- Static Head
- Total Dynamic Head (TDH)
- Flow
Paralleling the System Flow Demand

When the system flow demand varies over a wide range, parallel operation 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 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 clearance erosion.
- Separation / low-flow cavitation.
- Product quality degradation.
- Excessive hydraulic thrust.
- Premature bearing failure.
- Vibration & noise
- Heating of liquid pumped.

OPERATION FAR TO THE LEFT OF B.E.P — POSSIBLE PROBLEMS
ONSET OF ADVERSE EFFECTS WHEN OPERATING AWAY FROM B.E.P.

Diagram showing the onset of adverse effects when operating away from the ideal operating zone:
- High temperature rise
- Low flow cavitation
- Reduced bearing & seal life
- Reduced impeller life
- Suction recirculation
- Discharge recirculation

The diagram illustrates the preferred selection zone and the ideal operating zone.

CAVITATION DUE TO LACK OF NPSHr
OPERATION 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:

<table>
<thead>
<tr>
<th>WHEN ONLY IMPELLER DIA. CHANGES &amp; SPEED REMAINS THE SAME</th>
<th>WHEN ONLY SPEED CHANGES &amp; IMPELLER DIA. REMAINS THE SAME</th>
<th>WHEN BOTH DIA &amp; SPEED CHANGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Q_2 = Q_1 \times \left( \frac{D_2}{D_1} \right) )</td>
<td>( Q_2 = Q_1 \times \left( \frac{N_2}{N_1} \right) )</td>
<td>( Q_2 = Q_1 \times \left( \frac{D_2}{D_1} \right) \times \left( \frac{N_2}{N_1} \right) )</td>
</tr>
<tr>
<td>( H_2 = H_1 \times \left( \frac{D_2}{D_1} \right)^2 )</td>
<td>( H_2 = H_1 \times \left( \frac{N_2}{N_1} \right)^2 )</td>
<td>( H_2 = H_1 \times { \left( \frac{D_2}{D_1} \right) \times \left( \frac{N_2}{N_1} \right) }^2 )</td>
</tr>
<tr>
<td>( BKW_2 = BKW_1 \times \left( \frac{D_2}{D_1} \right)^3 )</td>
<td>( BKW_2 = BKW_1 \times \left( \frac{N_2}{N_1} \right)^3 )</td>
<td>( BKW_2 = BKW_1 \times { \left( \frac{D_2}{D_1} \right) \times \left( \frac{N_2}{N_1} \right) }^3 )</td>
</tr>
</tbody>
</table>

- \( Q_1, H_1, BKW_1, D_1 & N_1 \) ARE CAPACITY, HEAD, INPUT POWER IN KW, IMPELLER DIA. & SPEED AT INITIAL CONDITION.
- \( Q_2, H_2, BKW_2, D_2 & N_2 \) 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

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 FOUNDATION BOARD & PUMP-BASEPLATE
- LOOSE FOUNDATION
- POOR FLOW QUALITY IN THE SUMP/ UNFAVOURABLE PUMP INLET CONDITIONS (NPSH, INLET VORTICES, ETC.)
- WATER HAMMER

TYPES OF VIBRATION

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

INPUT POWER AT TEST CONDITIONS - KW

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

<table>
<thead>
<tr>
<th>PROBABLE FAULT</th>
<th>REMEDY</th>
</tr>
</thead>
<tbody>
<tr>
<td>Impeller rotating in wrong direction</td>
<td>Reverse direction of rotation</td>
</tr>
<tr>
<td>Pump not up to rated speed</td>
<td>Increase speed</td>
</tr>
<tr>
<td>Impeller neck rings worn excessively</td>
<td>Dismantle pump and restore clearances to original dimensions</td>
</tr>
<tr>
<td>Impeller damaged or chocked</td>
<td>Dismantle pump and renew impeller or clear blockage</td>
</tr>
<tr>
<td>Pump gaskets leaking</td>
<td>Renew defective gaskets</td>
</tr>
<tr>
<td>PROBABLE FAULT</td>
<td>REMEDY</td>
</tr>
<tr>
<td>-------------------------------------------------------------------------------</td>
<td>------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Suction line not fully primed – air or vapour lock in suction line</td>
<td>Stop pump and reprime</td>
</tr>
<tr>
<td>Inlet of suction pipe insufficiently submerged</td>
<td>Ensure adequate supply of liquid at suction pipe inlet</td>
</tr>
<tr>
<td>Air leaks in suction line or gland arrangement</td>
<td>Make good any leaks or renew gland packing</td>
</tr>
<tr>
<td>Liquid seal to gland arrangement logging ring (if fitted) chocked</td>
<td>Clean out liquid seal supply</td>
</tr>
<tr>
<td>Logging ring not properly located</td>
<td>Unpack gland and locate logging ring under supply orifice</td>
</tr>
<tr>
<td>Air or vapour lock in suction</td>
<td>Stop pump and reprime</td>
</tr>
<tr>
<td>Fault in driving unit</td>
<td>Examine driving unit and make good any defects</td>
</tr>
<tr>
<td>Air leaks in suction line or gland arrangement</td>
<td>Make good any leaks or repack gland</td>
</tr>
<tr>
<td>Inlet of suction pipe insufficiently immersed in liquid</td>
<td>Ensure adequate supply of liquid at suction pipe inlet</td>
</tr>
<tr>
<td>PROBABLE FAULT</td>
<td>REMEDY</td>
</tr>
<tr>
<td>---------------------------------------------------------</td>
<td>---------------------------------------------</td>
</tr>
<tr>
<td>Air or vapour lock in suction line</td>
<td>Stop pump and reprime</td>
</tr>
<tr>
<td>Inlet of suction pipe insufficiently submerged</td>
<td>Ensure adequate supply of liquid at suction pipe inlet</td>
</tr>
<tr>
<td>Air leaks in suction line or gland arrangement</td>
<td>Make good any leaks or repack gland</td>
</tr>
<tr>
<td>Worn or loose bearings</td>
<td>Disconnect coupling and realign pump and driving unit</td>
</tr>
<tr>
<td>Rotating element shaft bent</td>
<td>Dismantle pump, straighten or renew shaft</td>
</tr>
<tr>
<td>Foundation not rigid</td>
<td>Dismantle pump and driving unit, strengthen foundation</td>
</tr>
<tr>
<td>PROBABLE FAULT</td>
<td>REMEDY</td>
</tr>
<tr>
<td>------------------------------------------------</td>
<td>----------------------------------------------------------------</td>
</tr>
<tr>
<td>Pump gaskets leaking</td>
<td>Renew defective gasket</td>
</tr>
<tr>
<td>Serious leak in delivery line, pump delivering more than its rated quantity</td>
<td>Repair leak</td>
</tr>
<tr>
<td>Speed too high</td>
<td>Reduce Speed</td>
</tr>
<tr>
<td>Impeller neck rings worn excessively</td>
<td>Dismantle pump and restore clearances to original dimensions</td>
</tr>
<tr>
<td>Gland packing too tight</td>
<td>Stop pump, close delivery valve to relieve internal pressure on packing, slacken back the gland nuts and retighten to finger tightness</td>
</tr>
<tr>
<td>Impeller damaged</td>
<td>Dismantle pump and renew impeller</td>
</tr>
<tr>
<td>Mechanical tightness of pump internal components</td>
<td>Dismantle pump, check internal clearances and adjust as necessary</td>
</tr>
<tr>
<td>Pipe work exerting strain on pump</td>
<td>Disconnect pipe work and realign to pump</td>
</tr>
<tr>
<td>PROBABLE FAULT</td>
<td>REMEDY</td>
</tr>
<tr>
<td>--------------------------------------------------</td>
<td>------------------------------------------------------------------------</td>
</tr>
<tr>
<td>Pump and driving unit out of alignment</td>
<td>Disconnect coupling and realign pump and driving unit</td>
</tr>
<tr>
<td>Oil level too low or too high</td>
<td>Replenish with correct grade of oil or drain down to correct level</td>
</tr>
<tr>
<td>Wrong grade of oil</td>
<td>Drain out bearing, flush through bearings; refill with correct grade of oil</td>
</tr>
<tr>
<td>Dirt in bearing</td>
<td>Dismantle, clean out and flush through bearings; refill with correct grade of oil</td>
</tr>
<tr>
<td>Moisture in oil</td>
<td>Drain out bearing, flush through and refill with correct grade of oil. Determine cause of contamination and rectify</td>
</tr>
<tr>
<td>Bearings too tight</td>
<td>Ensure that bearings are correctly bedded to their journals with the correct amount of oil clearance. Renew bearings if necessary</td>
</tr>
<tr>
<td>Too much grease in bearing</td>
<td>Clean out old grease and repack with correct grade and qty of grease</td>
</tr>
<tr>
<td>Pipe work exerting strain on pump</td>
<td>Disconnect pipe work and realign to pump</td>
</tr>
<tr>
<td>PROBABLE FAULT</td>
<td>REMEDY</td>
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<td>Air or vapour lock in suction</td>
<td>Stop pump and reprime</td>
</tr>
<tr>
<td>Inlet of suction pipe insufficiently submerged</td>
<td>Ensure adequate supply of liquid at suction pipe inlet</td>
</tr>
<tr>
<td>Pump and driving unit incorrectly aligned</td>
<td>Decouple pump and driver, realign &amp; check alignment after coupling.</td>
</tr>
<tr>
<td>Worn or loose bearings</td>
<td>Dismantle pump and renew bearings</td>
</tr>
<tr>
<td>Impeller chocked or damaged</td>
<td>Dismantle pump and clear or renew impeller</td>
</tr>
<tr>
<td>Rotating element shaft bent</td>
<td>Dismantle pump, straighten or renew shaft</td>
</tr>
<tr>
<td>Foundation not rigid</td>
<td>Remove pump, strengthen the foundation and reinstall pump</td>
</tr>
<tr>
<td>Coupling damaged</td>
<td>Renew coupling</td>
</tr>
<tr>
<td>Pipe work exerting strain on pump</td>
<td>Disconnect pipe work and realign to pump</td>
</tr>
<tr>
<td>Probable Fault</td>
<td>Remedy</td>
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<tr>
<td>Pump and driving unit out of alignment</td>
<td>Disconnect coupling and realign pump and driving unit. Renew bearings if necessary</td>
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<tr>
<td>Rotating element shaft bent</td>
<td>Dismantle pump, straighten or renew shaft. Renew bearings if necessary</td>
</tr>
<tr>
<td>Dirt in bearings</td>
<td>Ensure that only clean oil is used to lubricate bearings. Renew bearings if necessary. Refill with clean oil</td>
</tr>
<tr>
<td>Lack of lubrication</td>
<td>Ensure that oil is maintained at its correct level or that oil system is functioning correctly. Renew bearings if necessary</td>
</tr>
<tr>
<td>Bearing badly installed</td>
<td>Ensure that bearings are correctly bedded to their journals with the correct amount of oil clearance. Renew bearings if necessary</td>
</tr>
<tr>
<td>Pipe work exerting strain on pump</td>
<td>Ensure that pipe work is correctly aligned to pump. Renew bearings if necessary</td>
</tr>
<tr>
<td>Excessive vibration</td>
<td>Refer excessive vibration</td>
</tr>
<tr>
<td>Symptom</td>
<td>Diagnosis</td>
</tr>
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<td>------------------</td>
<td>---------------------------------------------------------------------------</td>
</tr>
<tr>
<td>CV   As per pump curve</td>
<td>Changed system condition – blockage pipe friction, filters, strainers, etc</td>
</tr>
<tr>
<td>Open valve</td>
<td>Q1 &lt;Q, H1&gt; H</td>
</tr>
<tr>
<td>Q1, H1 on pump curve</td>
<td>Q1, H1 on pump curve</td>
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</tbody>
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<table>
<thead>
<tr>
<th>Symptom</th>
<th>Diagnosis</th>
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<tbody>
<tr>
<td>CV   As per pump curve</td>
<td>Pump fault – Blockage in impeller, increased leakage loss</td>
</tr>
<tr>
<td>Open valve</td>
<td>Q1 &lt;Q, H1&gt; H</td>
</tr>
<tr>
<td>Q1, H1 on pump curve</td>
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<td><strong>Diagnosis</strong></td>
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<td>------------------------------------------------</td>
<td>-------------------------------------------------------------------------------</td>
</tr>
<tr>
<td>CV As per pump curve</td>
<td>Insufficient NPSH leading to cavitation breakdown</td>
</tr>
<tr>
<td>Open valve Head lower in vicinity of the system curve. Sudden break down of H-Q</td>
<td></td>
</tr>
<tr>
<td>CV Lower than pump curve</td>
<td>Incorrect speed Incorrect diameter of impeller Wrong direction of rotation</td>
</tr>
<tr>
<td>Lower Q, Lower H</td>
<td></td>
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<tr>
<td>SYMPTOMS</td>
<td>COMMON CAUSES</td>
</tr>
<tr>
<td>----------</td>
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</tr>
<tr>
<td>NO DELIVERY OR DELIVERY NOT UPTO THE EXPECTATION</td>
<td>➢ PUMP NOT PRIMED</td>
</tr>
<tr>
<td></td>
<td>➢ AIR-POCKET IN SUCTION LINE</td>
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<tr>
<td></td>
<td>➢ INSUFFICIENT IMMERSION OF SUCTION PIPE, VORTEXING</td>
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<td>➢ SPEED OF PUMP TOO LOW OR WRONG DIRECTION OF ROTATION</td>
</tr>
<tr>
<td></td>
<td>➢ SYSTEM HEAD HIGHER THAN PUMP DESIGN HEAD</td>
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<tr>
<td>SYMPTOMS</td>
<td>COMMON CAUSES</td>
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<tr>
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<tr>
<td>INSUFFICIENT DISCHARGE PRESSURE</td>
<td> WRONG IMPELLER SELECTION</td>
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<td> WRONG IMPELLER INSTALLATION</td>
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<tr>
<td></td>
<td> AIR-GAS ENTRAINMENT IN LIQUID</td>
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<td></td>
<td> IMPELLER CLOGGED</td>
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<td></td>
<td> IMPROPER PUMP SELECTION</td>
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<td>------------------</td>
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<tr>
<td>SHORT SEAL LIFE</td>
<td>MISALIGNMENT</td>
</tr>
<tr>
<td></td>
<td>BENT SHAFT</td>
</tr>
<tr>
<td></td>
<td>CASING DISTORSION DUE TO PIPE STRAIN</td>
</tr>
<tr>
<td></td>
<td>PUMP CAVITATING</td>
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<td></td>
<td>IMPROPER OPERATING CONDITION</td>
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<td></td>
<td>UNBALANCE DRIVER</td>
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<tr>
<td>SYMPTOMS</td>
<td>COMMON CAUSES</td>
</tr>
<tr>
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</tr>
</tbody>
</table>
| 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 & PARALLEL ALIGNMENT BETWEEN PUMP & DRIVER  
➢ ELIMINATE STILT-MOUNTED BASE-PLATE  
➢ CHECK FOR LOOSE MOUNTING  
➢ CHECK FOR UNIFORM 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 |
<p>| | ➢ UNBALANCE DRIVER | ➢ RUN DRIVER DISCONNECTED FROM PUMP UNIT - PERFORM VIBRATION ANALYSIS |</p>
<table>
<thead>
<tr>
<th>SYMTOMPS</th>
<th>COMMON CAUSES</th>
<th>REMEDY</th>
</tr>
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<tbody>
<tr>
<td><strong>EXCESSIVE POWER DEMAND</strong></td>
<td>➢ MOTOR TRIPPING-OFF</td>
<td>➢ CHECK STARTER&lt;br&gt; ➢ CHECK RELAY SETTING&lt;br&gt; ➢ CHECK FOR THE SUITABILITY OF MOTOR SELECTED FOR CURRENT OPERATING CONDITION</td>
</tr>
<tr>
<td>➢ SPEED TOO HIGH</td>
<td>➢ CHECK FOR SPEED OR PREVIOUS RECORDS FOR PROPER SPEED&lt;br&gt; ➢ ELIMINATE STILT-MOUNTED BASE-PLATE&lt;br&gt; ➢ CHECK FOR LOOSE MOUNTING&lt;br&gt; ➢ CHECK FOR UNIFORM THERMAL EXPANSION OF PUMP PARTS</td>
<td></td>
</tr>
<tr>
<td>➢ ROTOR IMPELLER RUBBING ON CASING</td>
<td>➢ LOOSE IMPELLER FIT&lt;br&gt; ➢ WRONG ROTATION&lt;br&gt; ➢ REPLACE IF SHAFT IS BENT&lt;br&gt; ➢ HIGH NOZZLE LOADS&lt;br&gt; ➢ VERY SMALL INTERNAL RUNNING CLEARANCES – CHECK FOR NECK RING DIMENSIONS</td>
<td></td>
</tr>
<tr>
<td>➢ PUMP NOT DESIGNED FOR LIQUID DENSITY &amp; VISCOSITY BEING PUMPED</td>
<td>➢ CHECK DESIGN SP. GRAVITY&lt;br&gt; ➢ CHECK MOTOR SIZE – USE LARGER DRIVER OR CHANGE PUMP TYPE&lt;br&gt; ➢ HEAT UP THE LIQUID TO REDUCE VISCOSITY</td>
<td></td>
</tr>
<tr>
<td>➢ BEARING FAILURES</td>
<td>➢ CHECK FOR PROPER LUBRICATION &amp; CONTAMINATION OF LUBRICANT&lt;br&gt; ➢ CHECK FOR PROPER BEARING INSTALLATION&lt;br&gt; ➢ CHECK FOR THE SUITABILITY OF BEARING SELECTED</td>
<td></td>
</tr>
<tr>
<td>➢ IMPROPER COUPLING SELECTION</td>
<td>➢ CHECK COUPLING SIZE</td>
<td></td>
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<tr>
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</tr>
<tr>
<td>NOISE &amp;</td>
<td>PUMP IS CAVITATING</td>
<td>➢ CHECK FOR NPSHa/NPSHr MARGIN &amp; TAKE NECESSARY STEPS</td>
</tr>
<tr>
<td>VIBRATION</td>
<td></td>
<td>➢ CHECK FOR FLASH POINT MARGIN</td>
</tr>
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<td></td>
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<td>➢ CHECK FOR GAS ENTRAINMENT</td>
</tr>
<tr>
<td></td>
<td>SUCTION OR DISCHARGE VALVE CLOSED OR PARTIALLY CLOSED</td>
<td>➢ CHECK FOR VALVE CONDITION</td>
</tr>
<tr>
<td></td>
<td></td>
<td>➢ OPEN THE VALVES</td>
</tr>
<tr>
<td></td>
<td>MISALIGNMENT</td>
<td>➢ CHECK ANGULAR &amp; PARALLEL ALIGNMENT BETWEEN PUMP &amp; DRIVER</td>
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<td>➢ ELIMINATE STILT-MOUNTED BASE-PLATE</td>
</tr>
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<td>➢ CHECK FOR LOOSE MOUNTING</td>
</tr>
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<td></td>
<td></td>
<td>➢ CHECK FOR UNIFORM THERMAL EXPANSION OF PUMP PARTS</td>
</tr>
<tr>
<td></td>
<td>INADEQUATE GROUTING OF BASE PLATE</td>
<td>➢ CHECK GROUTING, CONSULT PROCESS INDUSTRY PRACTICE RE-IE-686</td>
</tr>
<tr>
<td></td>
<td></td>
<td>➢ IF STILT MOUNTED, GROUT BASEPLATE</td>
</tr>
<tr>
<td></td>
<td>BEARING FAILURES</td>
<td>➢ CHECK FOR PROPER LUBRICATION &amp; CONTAMINATION OF LUBRICANT</td>
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<tr>
<td></td>
<td>IMPROPER COUPLING SELECTION</td>
<td>➢ CHECK COUPLING SIZE, GRAESING, ALIGNMENT</td>
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</table>