

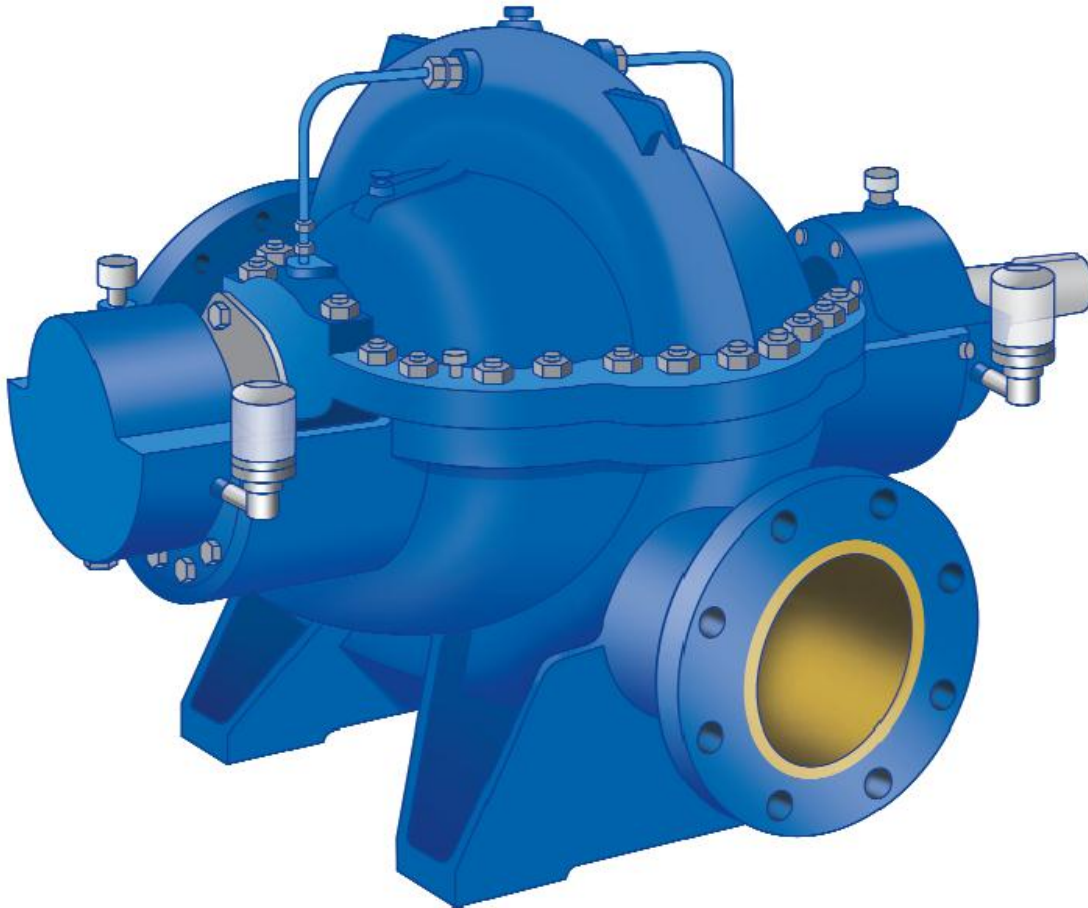


CENTRIFUGAL PUMPS TRAINING

**PUMP INSTALLATION
PUMP MAINTENANCE
PUMP TROUBLESHOOTING
FLUID MECHANIC**

PRESENTED BY :
ARIF BACHRUDDIN ADJI, ST
OPERATION DIRECTOR OF
PT. DURAQUIPT CEMERLANG

PUMP INSTALLATION



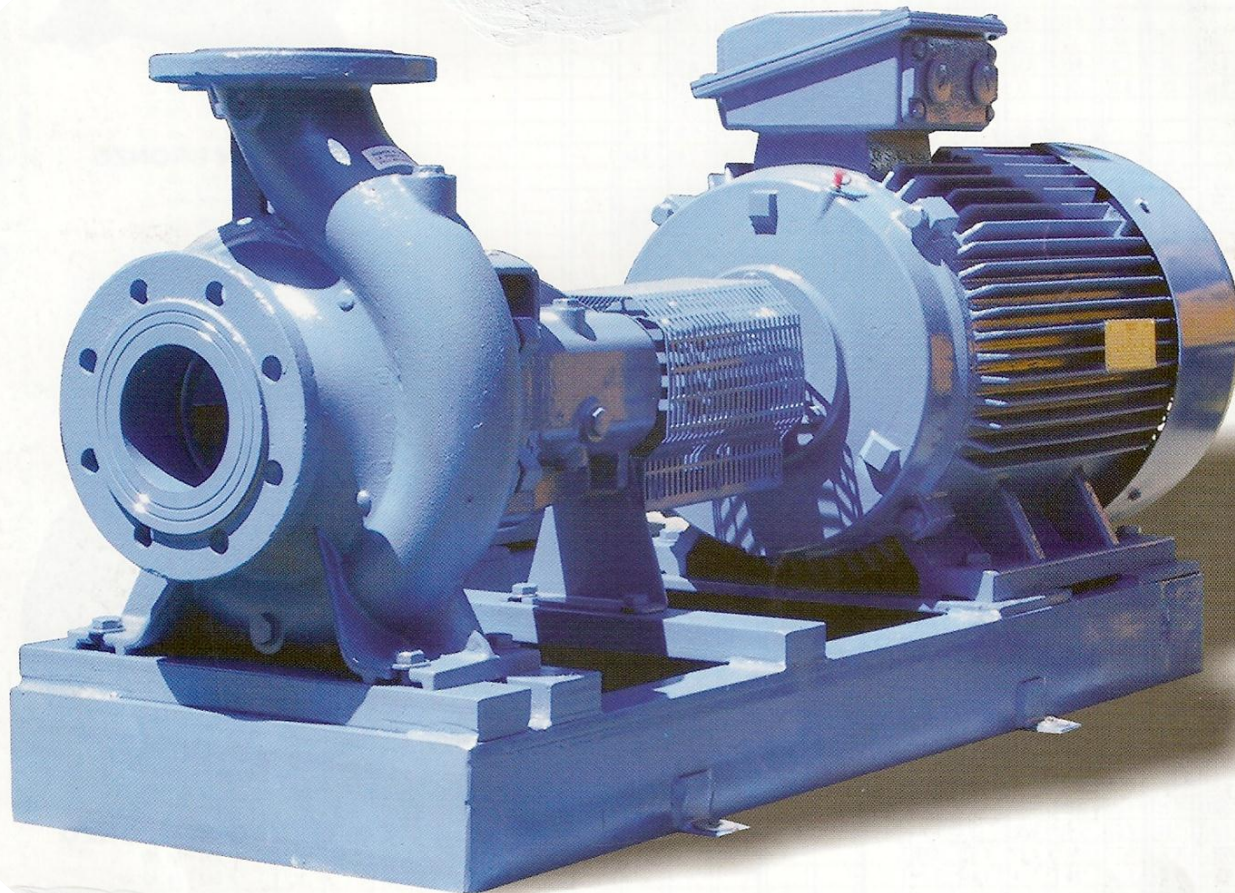
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ARIF BACHRUDDIN ADJI, ST
OPERATION DIRECTOR OF
PT. DURAQUIPT CEMERLANG

CENTRIFUGAL PUMP CLASIFICATIOIN

REFEER TO ANSI/API STANDARD 610, 10th EDITION, 2004

PUMP TYPE			ORIENTATION		TYPE CODE
CENTRIFUGLA PUMPS	OVERHUNG	FLEXIBLE COUPLED	HORIZONTAL	FOOT MOUNTED	OH-1
				CENTERLINE – MOUNTED	OH-2
		REGIDLY COUPLED	VERTICAL IN-LINE WITH BEARING BRACKET		OH-3
			VERTICAL IN-LINE		OH-4
			VERTICAL IN-LINE		OH-5
			HI-SPEED INTEGRALLY GEARED		OH-6
	BETWEEN BEARINGS	1- AND 2- STAGE	AXIALLY SPLIT		BB-1
			RADIALLY SPLIT		BB-2
		MULTISTAGE	AXIALLY SPLIT		BB-3
			RADIALLY SPLIT	SINGLE CASING	BB-4
				DOUBLE CASING	BB-5
	VERTICALLY SUSPENDED	SINGLE CASING	DISCHARGE THROUGH COLUMN	DIFFUSER	VS-1
				VOLUTE	VS-2
				AXIAL FLOW	VS-3
			SEPARATE DISCHARGE	LINE SHAFT	VS-4
				CANTILEVER	VS-5
		DOUBLE CASING	DIFFUSER		VS-6
			VOLUTE		VS-7

CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, OVERHUNG DESIGN
FLEXIBLE COUPLED
HORIZONTAL, FOOT MOUNTED
TYPE CODE = OH-1

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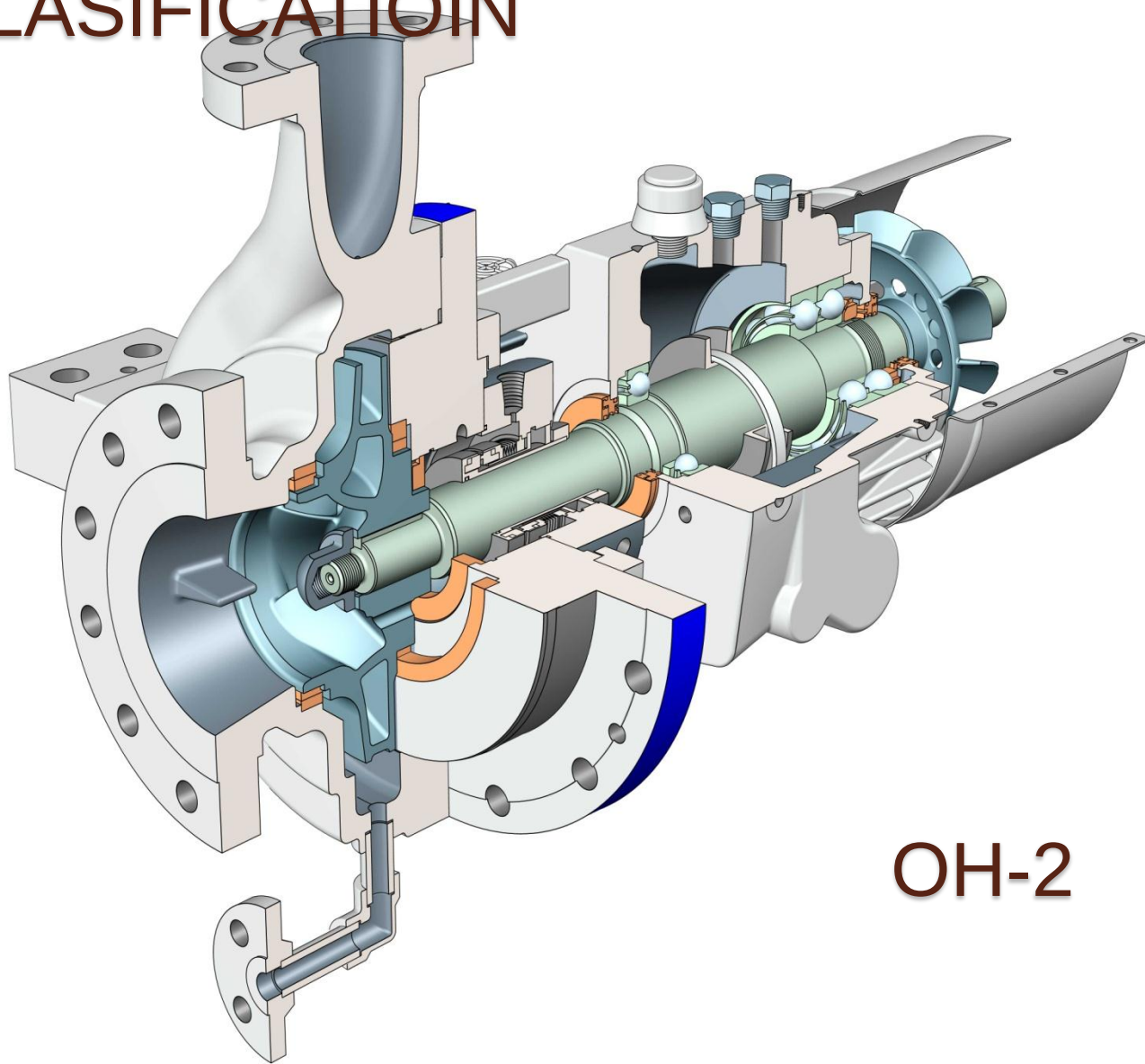
CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, OVERHUNG DESIGN
FLEXIBLE COUPLED
HORIZONTAL, CENTERLINE MOUNTED
TYPE CODE = OH-2

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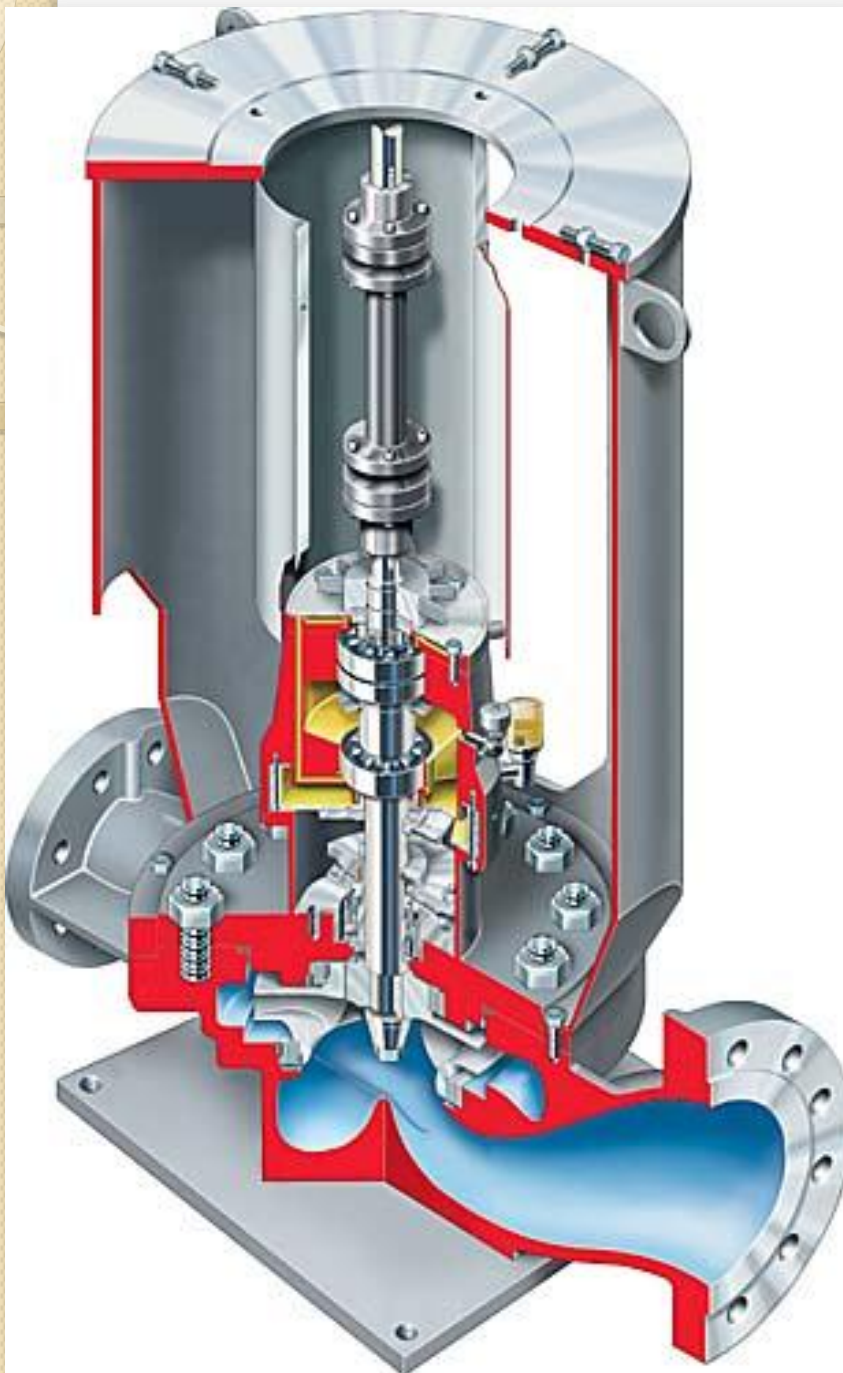
CENTRIFUGAL PUMP CLASIFICATIOIN



OH-2

CENTRIFUGAL PUMP CLASIFICATIOIN

CENTRIFUGAL PUMP,
OVERHUNG DESIGN
VERTICAL IN-LINE WITH
BEARING BRACKET
TYPE CODE = OH-3



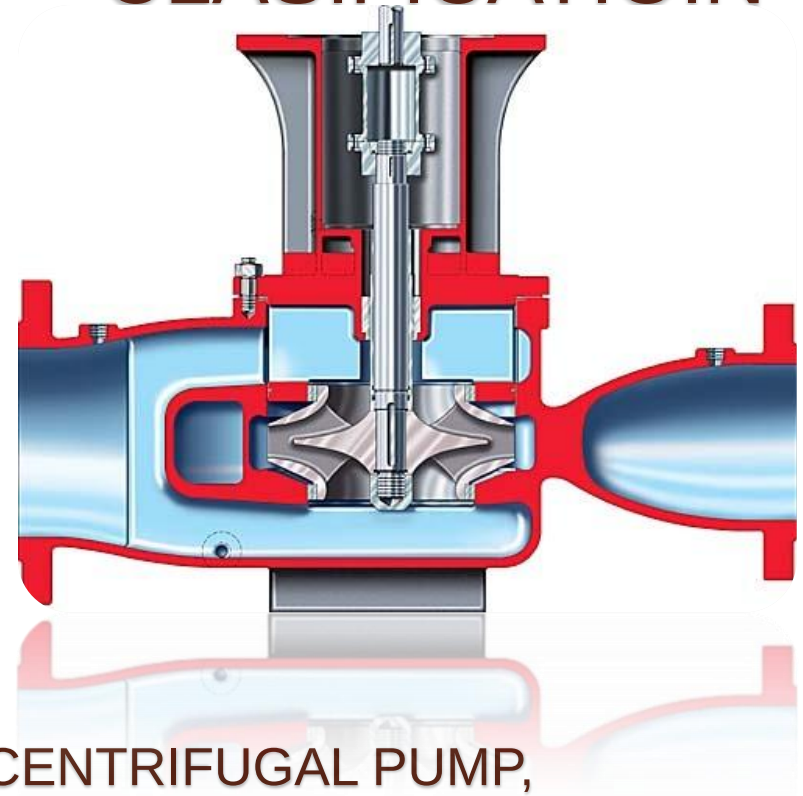
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CENTRIFUGAL PUMP CLASIFICATIOIN

OH-3

CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP,
OVERHUNG DESIGN
RIGIDLY COUPLED,
VERTICAL IN-LINE
TYPE CODE = OH-4

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CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, OVERHUNG DESIGN
CLOSE COUPLED, VERTICAL IN-LINE
TYPE CODE = OH-5

[MENU](#)

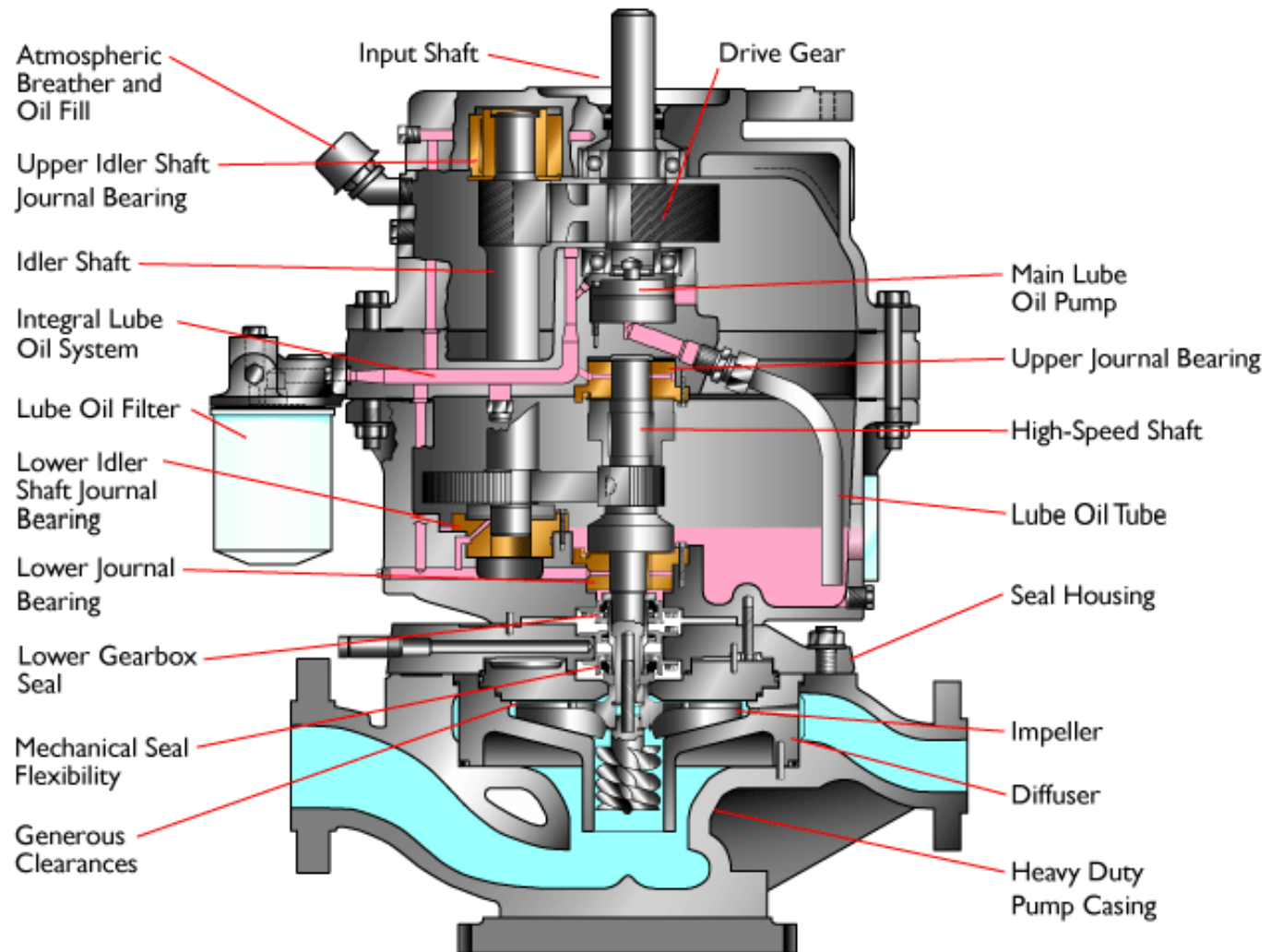
CENTRIFUGAL PUMP CLASIFICACIÓN



CENTRIFUGAL PUMP, OVERHUNG DESIGN
CLOSE COUPLED, HI-SPEED INTEGRALLY GEARED
TYPE CODE = OH-6

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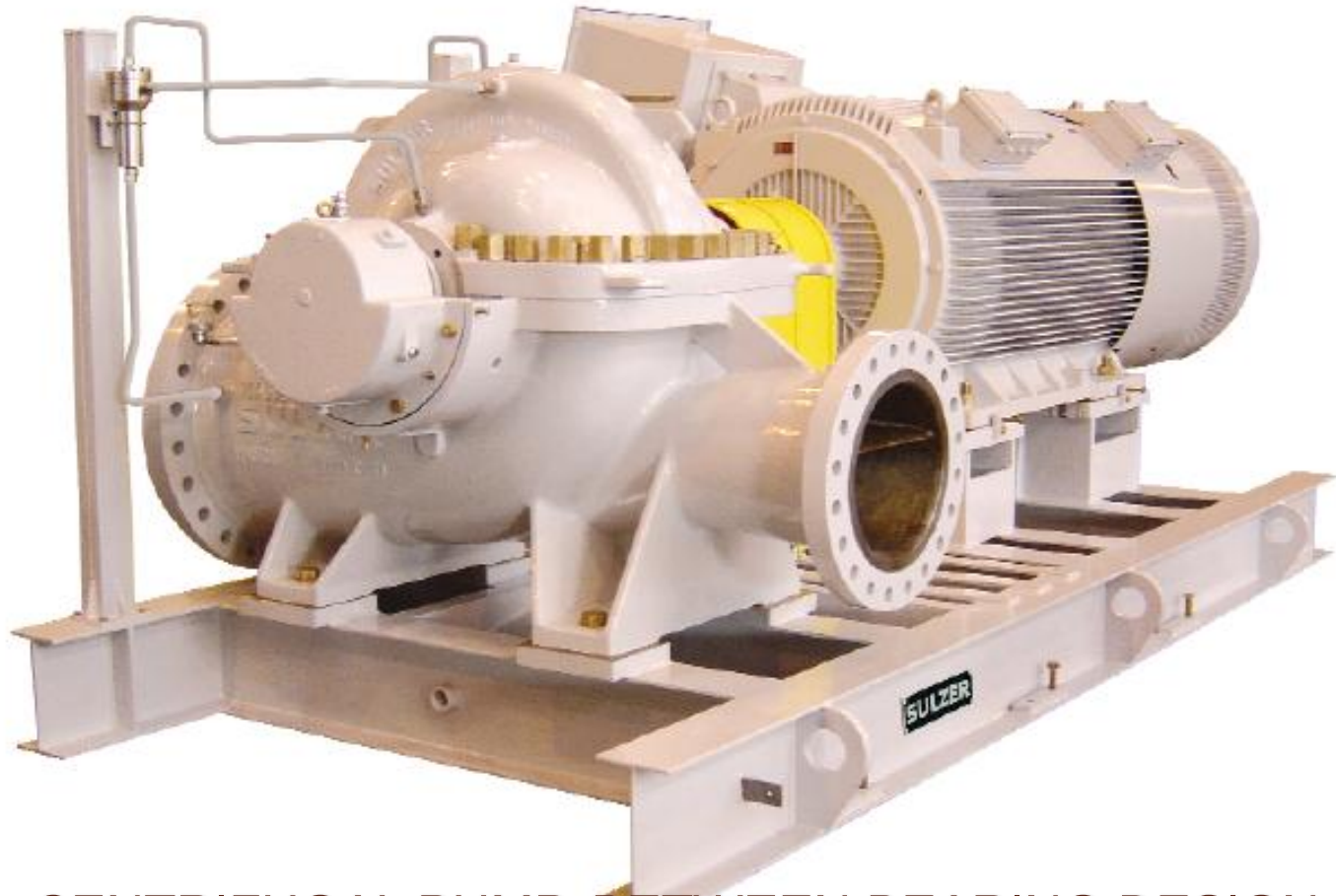
CENTRIFUGAL PUMP CLASIFICATIOIN



OH-6

Sundyne LMV/BMP-331 Pump

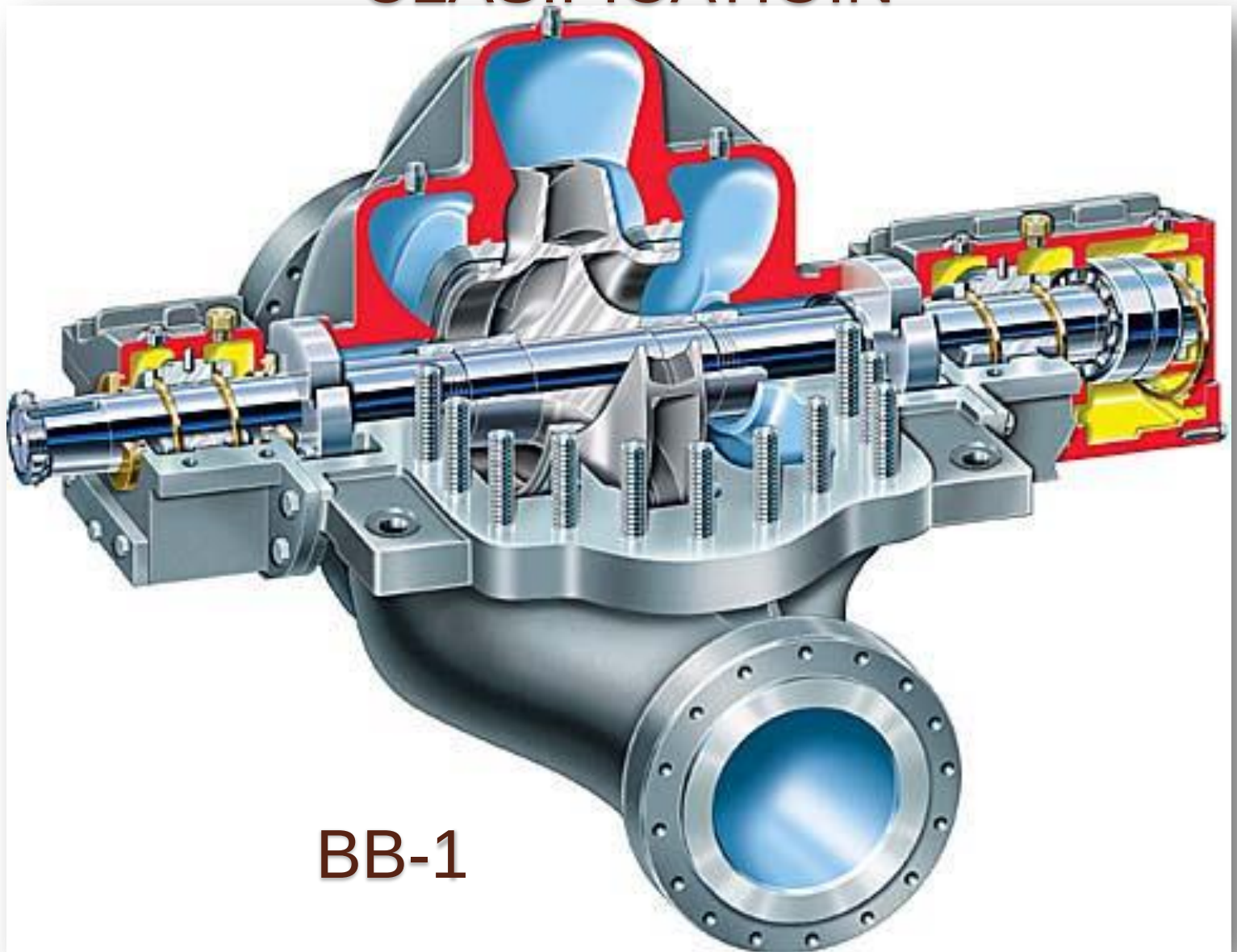
CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
SINGLE STAGE, AXIAL SPLIT CASE
TYPE CODE = BB-1

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CENTRIFUGAL PUMP CLASIFICATIOIN



BB-1

CENTRIFUGAL PUMP CLASIFICATIOIN

Axially Split Casing

- Suction and discharge branches in lower half.
- Provides easy maintenance.
- Complete rotor can be inspected and refurbished without disturbing the main pipework or the driver.

Double Entry Impeller

- Principal axial thrusts are inherently hydraulically balanced.
- Optimum vane geometry provides high efficiency, low NPSH requirement and quiet running over a wide operating range.

Heavy Duty Rotor Bearings

- Designed for long bearing life with assured lubrication.
- Back-to-back angular contact ball thrust bearing.
- Grease or oil lubrication.

Shaft Sealing

- Packed stuffing boxes.
- Mechanical seals are optional.

Hydraulic Geometry Optimized

- Resulting from intensive research and development provides ideal flow patterns with low velocities for high efficiency and quiet running.

Heavy Duty Shaft

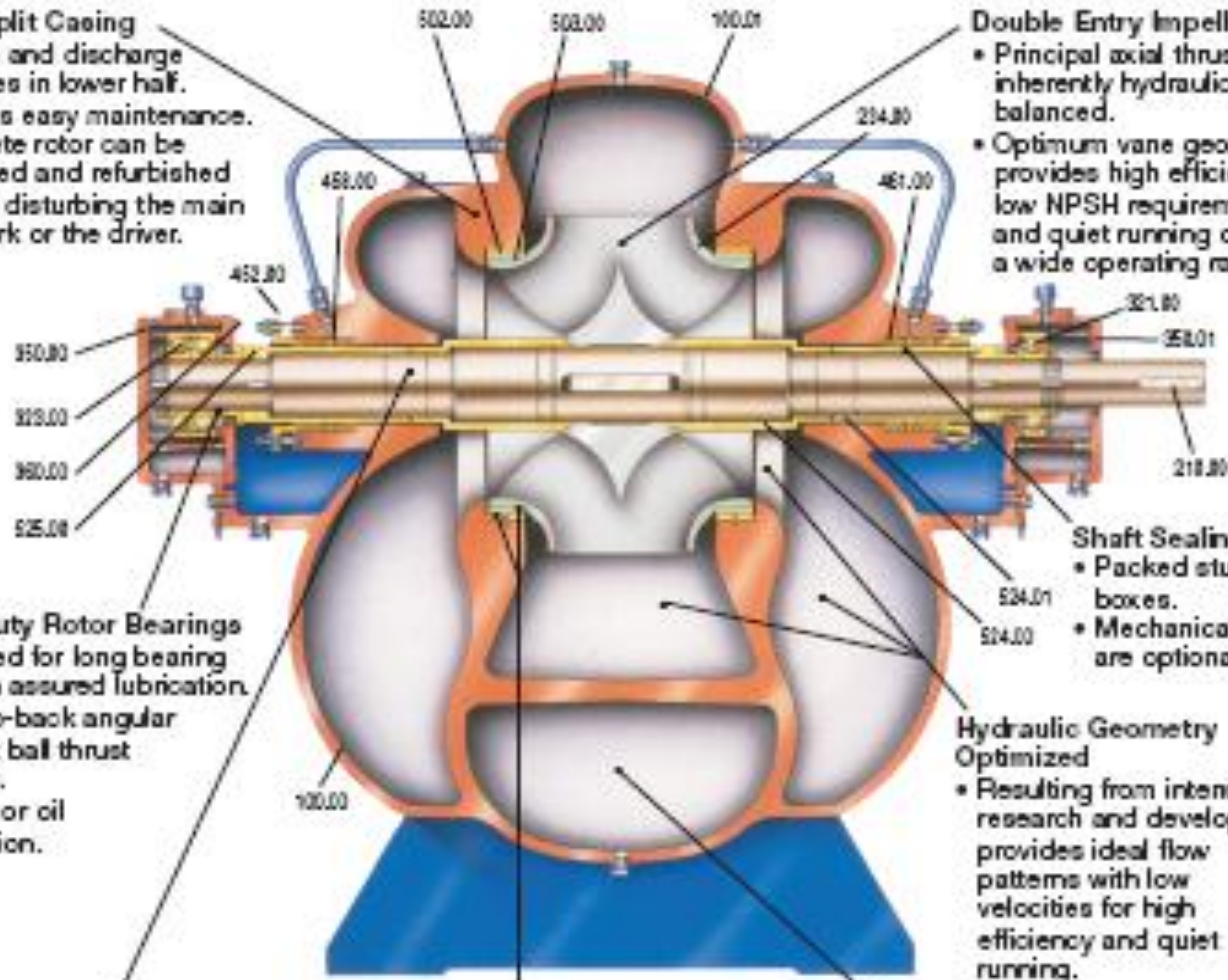
- Stiff design for hard service conditions.
- Replaceable protection sleeves throughout the waterways
- Dry shaft design.

Replaceable Wearing Rings

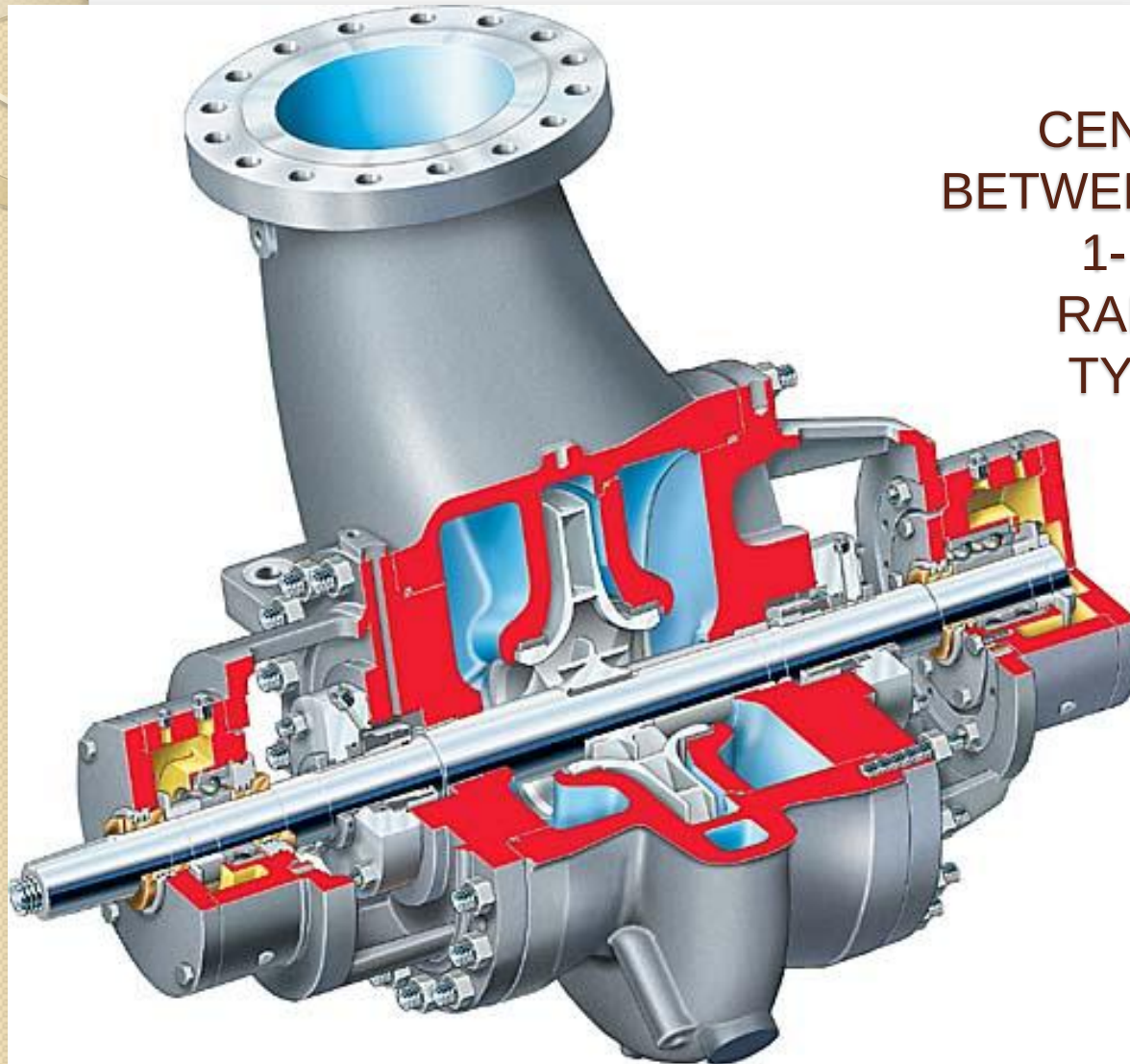
- Casing wear rings are normally provided and optionally impeller wear rings.
- Designed to optimize the hydraulic performance.
- Lower maintenance costs.

Volute

- Single or double volute to all sizes, reducing this way radial thrusts.



CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP,
BETWEEN BEARING DESIGN,
1- AND 2-STAGES,
RADIAL SPLIT CASE
TYPE CODE = BB-2

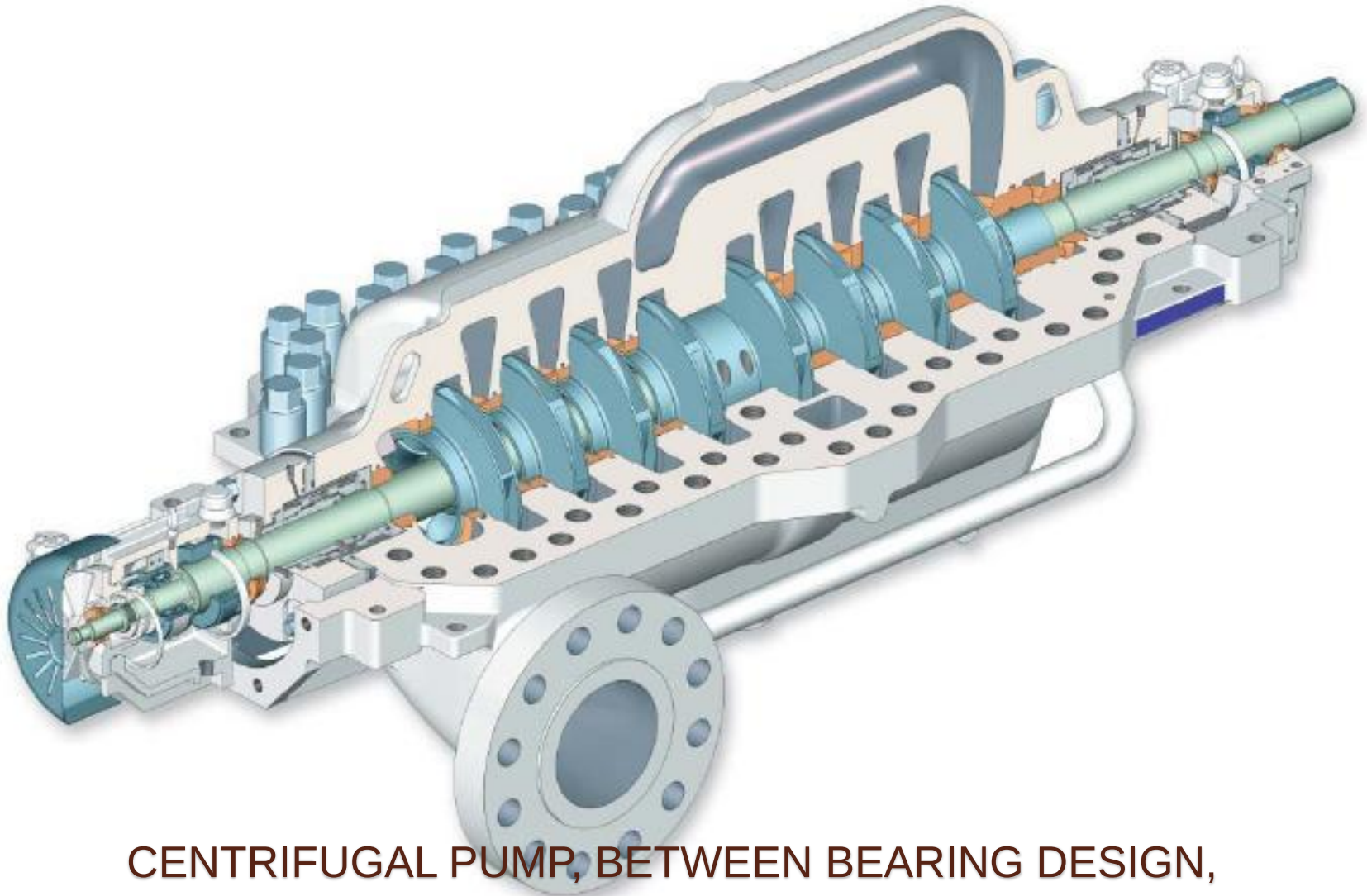
CENTRIFUGAL PUMP CLASSIFICATION



CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, AXIAL SPLIT CASE
TYPE CODE = BB-3

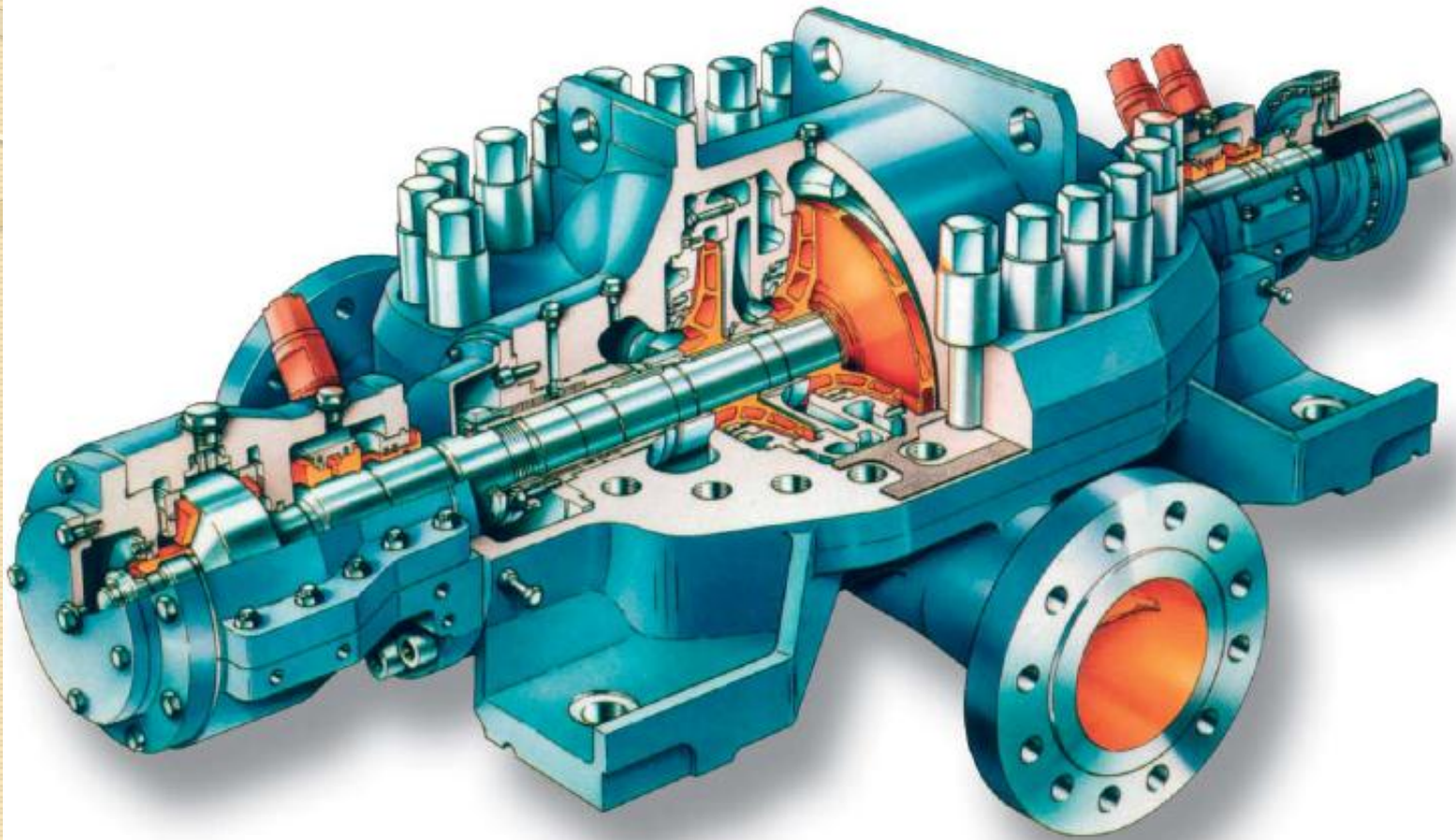
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CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, AXIAL SPLIT CASE
TYPE CODE = BB-3

CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, AXIAL SPLIT CASE
TYPE CODE = BB-3

CENTRIFUGAL PUMP CLASIFICATIOIN

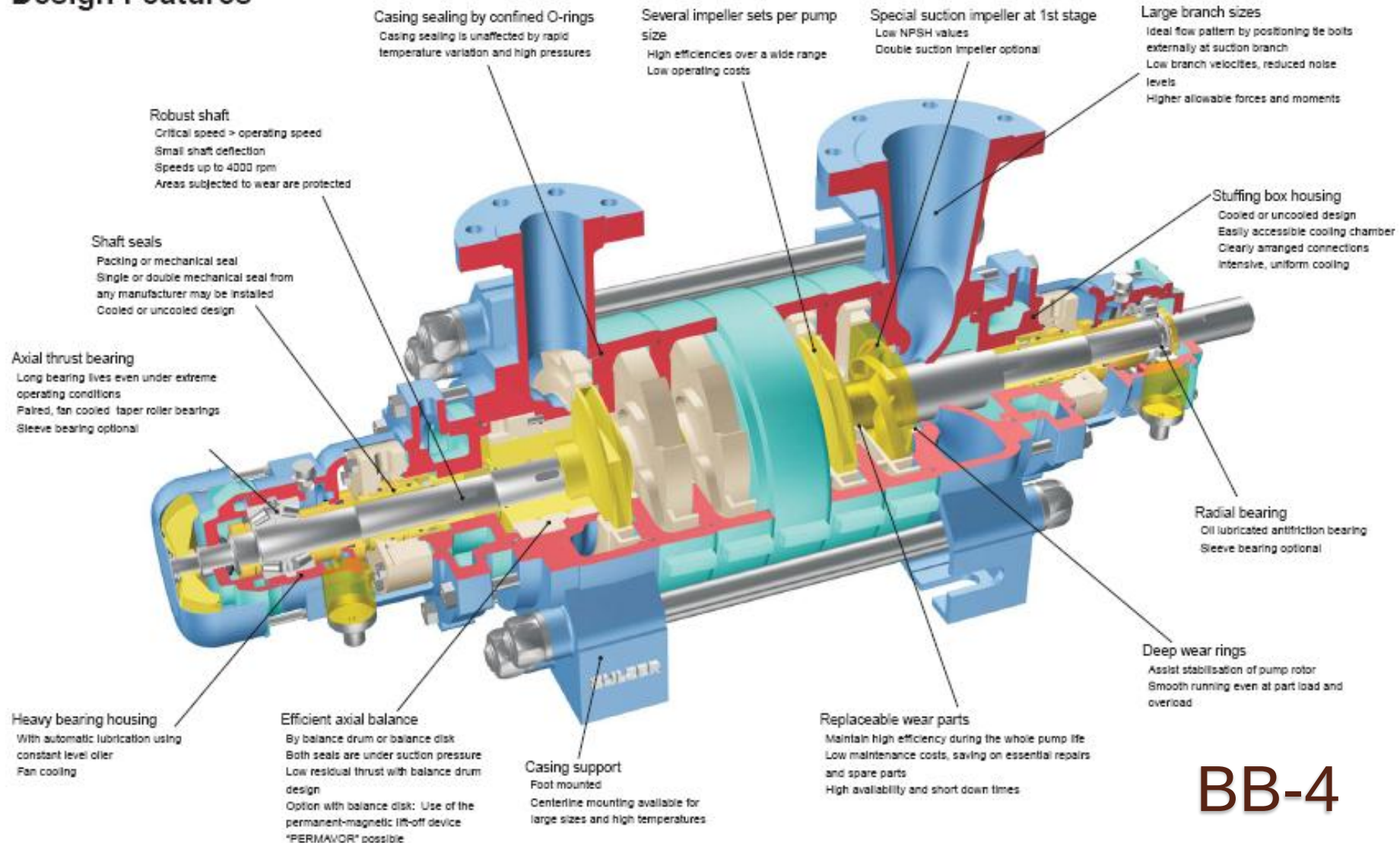


CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, RADIAL SPLIT CASE, SINGLE CASING
TYPE CODE = BB-4

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CENTRIFUGAL PUMP CLASIFICATIOIN

Design Features



BB-4

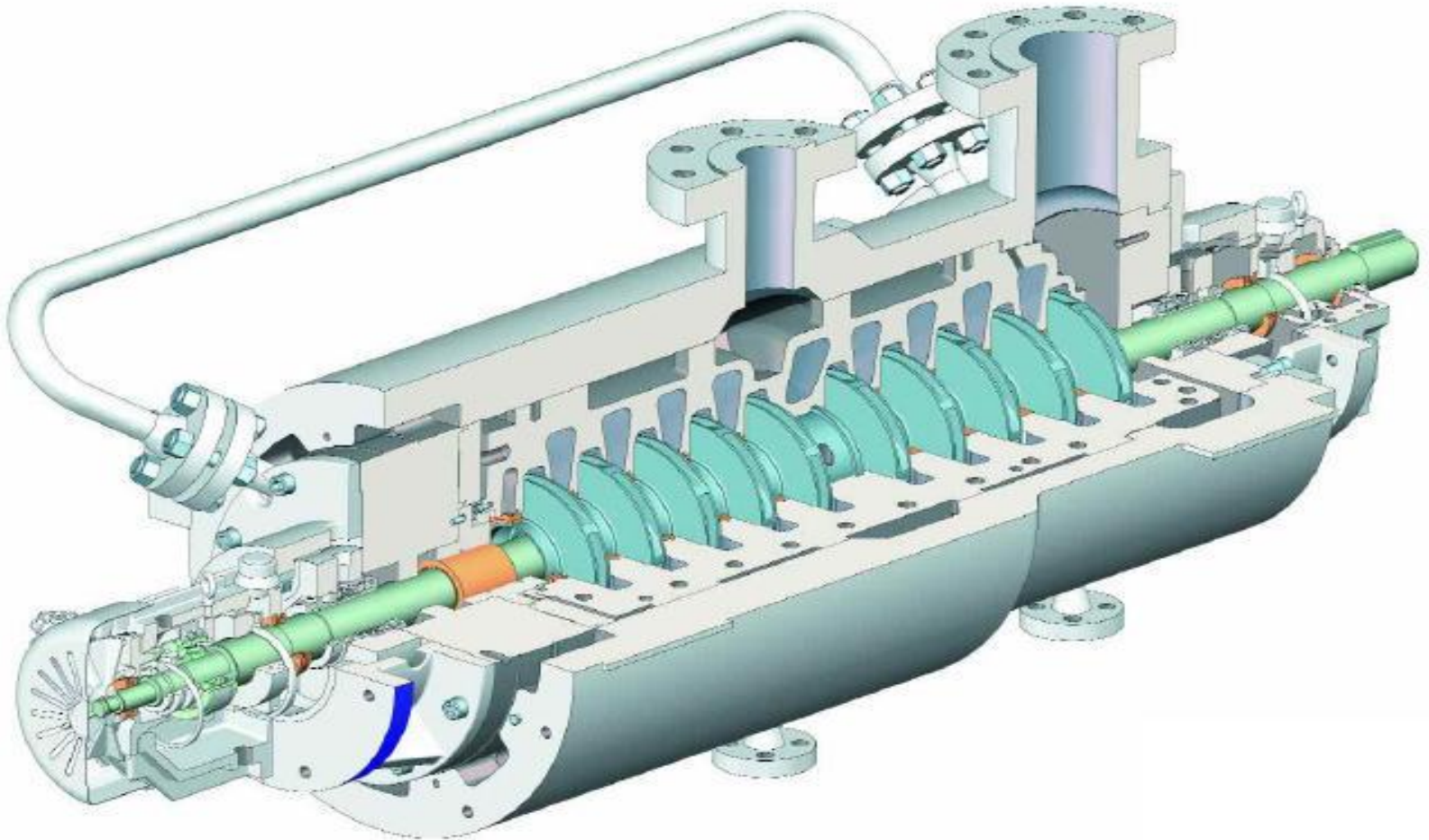
CENTRIFUGAL PUMP CLASIFICATIOIN



CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, RADIAL SPLIT CASE, DOUBLE CASING
TYPE CODE = BB-5

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CENTRIFUGAL PUMP CLASSIFICATION



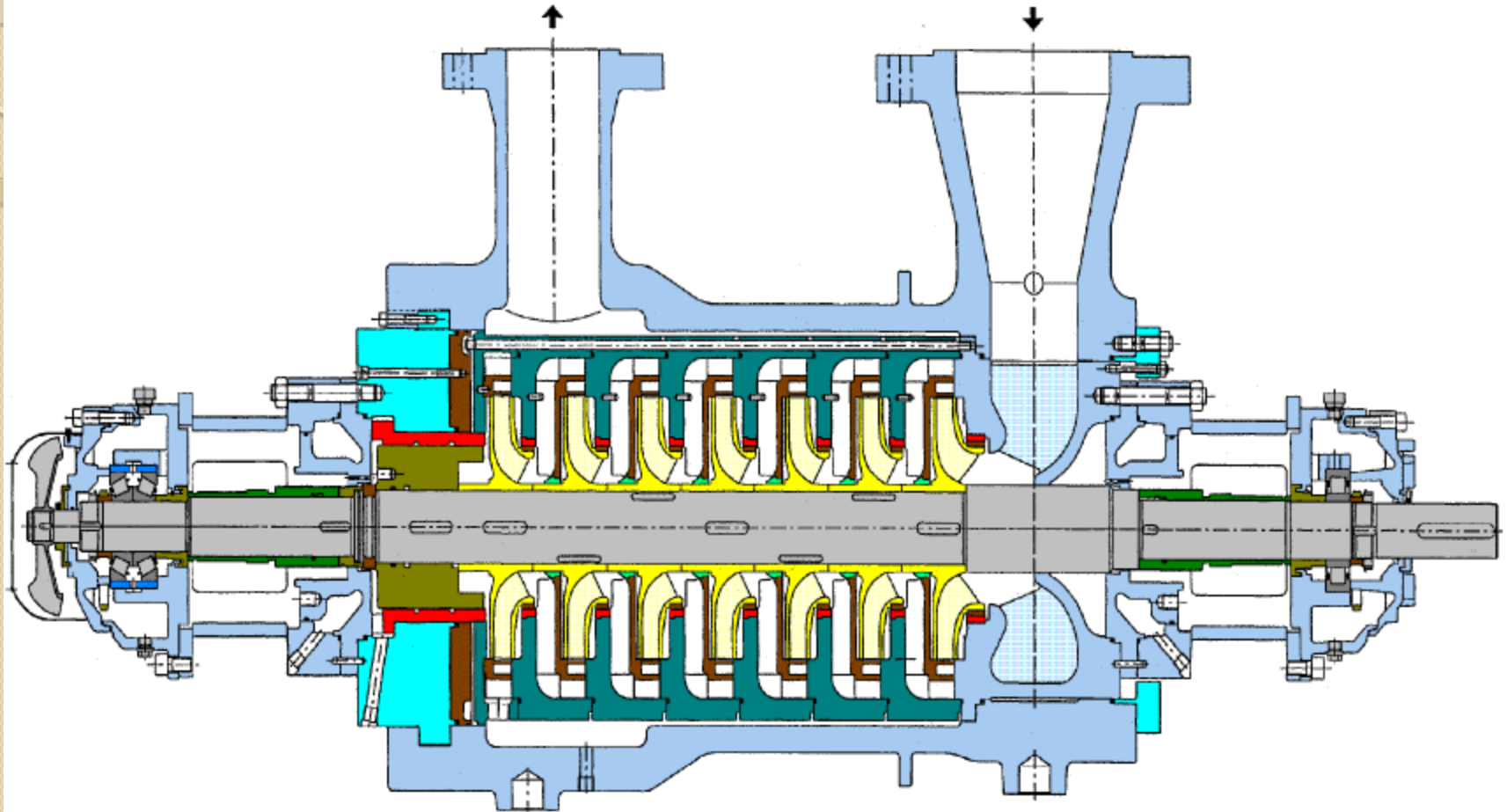
VIDEO
ANIMATION

CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, RADIAL SPLIT CASE, DOUBLE CASING
TYPE CODE = BB-5

CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, RADIAL SPLIT CASE, DOUBLE CASING
TYPE CODE = BB-5



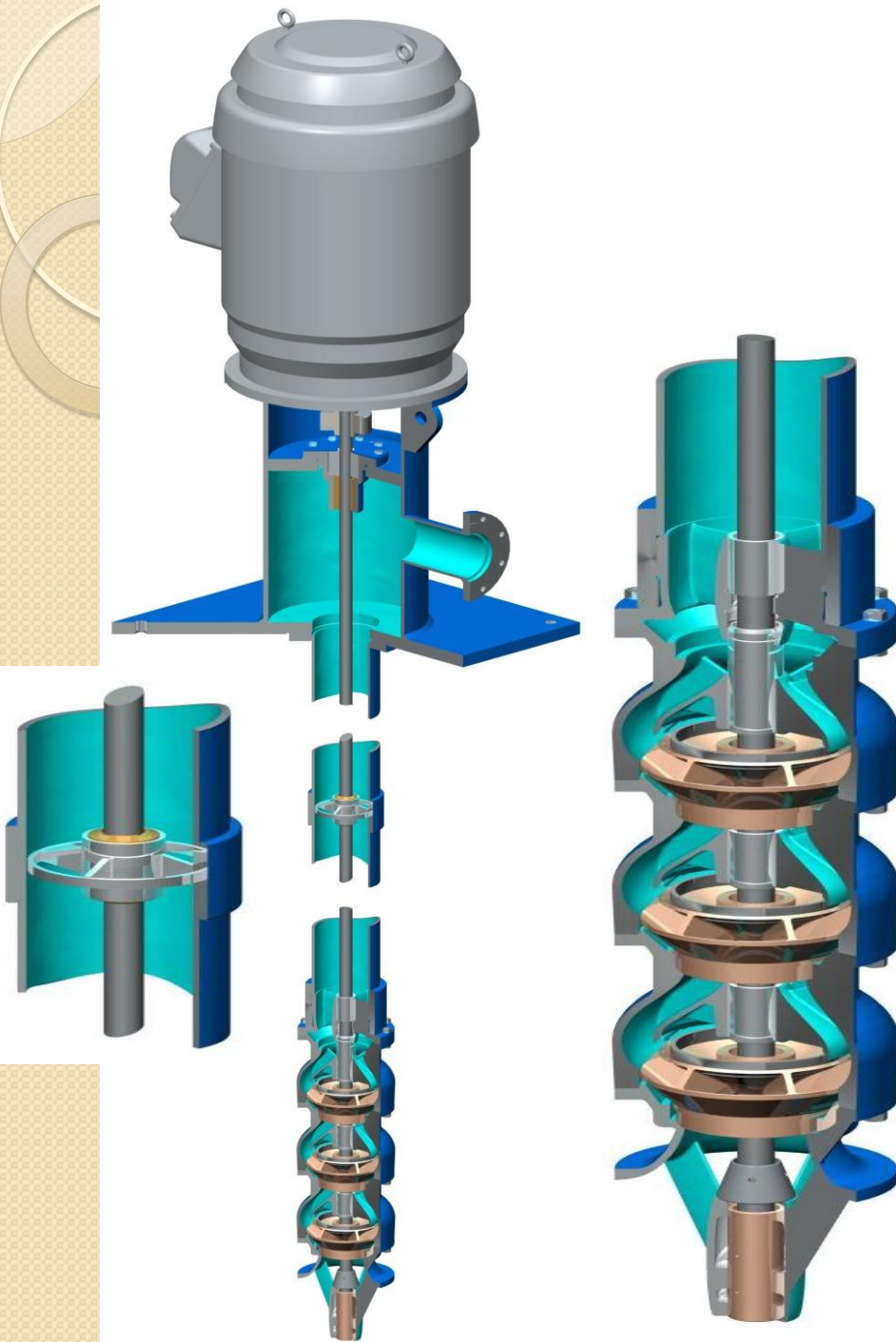
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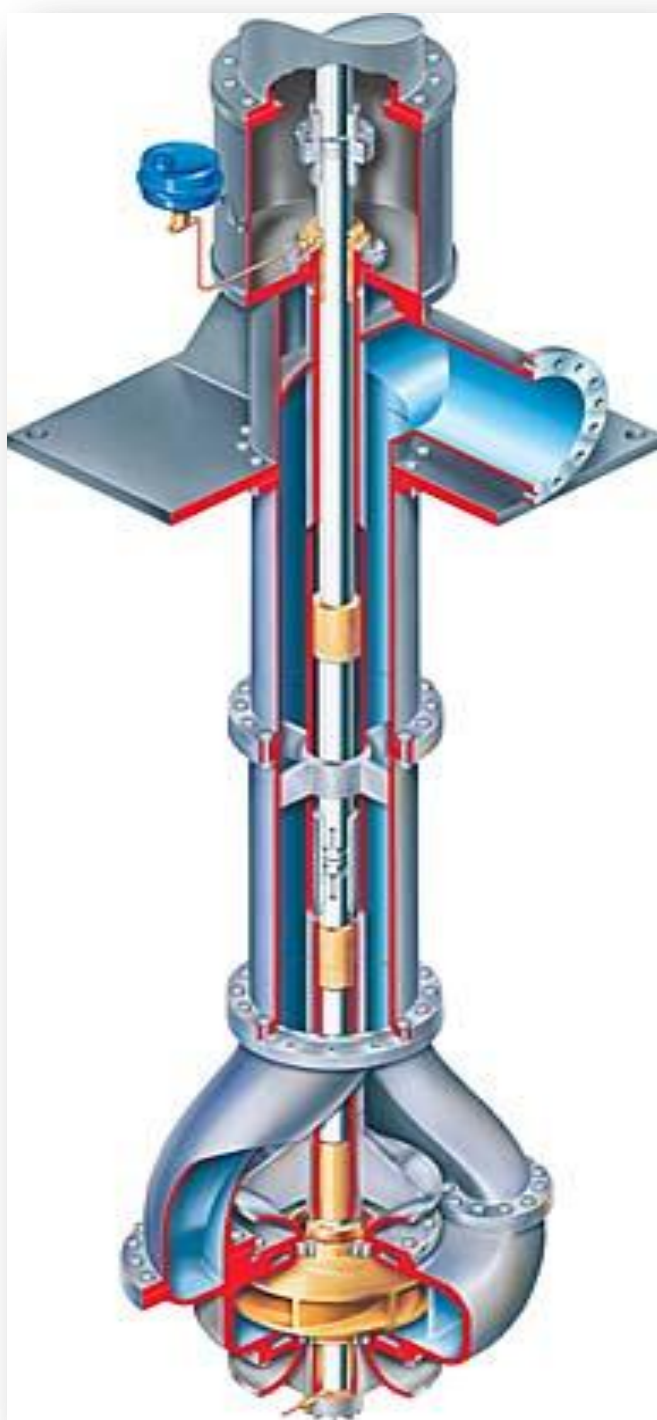
CENTRIFUGAL PUMP, BETWEEN BEARING DESIGN,
MULTISTAGES, RADIAL SPLIT CASE, DOUBLE CASING
TYPE CODE = BB-5

CENTRIFUGAL PUMP CLASIFICATIOIN

CENTRIFUGAL PUMP, VERTICAL
SUSPENDED,
SINGLE CASING, DISCHARGE
THROUGH COLUMN, DIFFUSER
TYPE
TYPE CODE = VS-1



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CENTRIFUGAL PUMP CLASIFICATIOIN

CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
SINGLE CASING,
DISCHARGE THROUGH
COLUMN,
VOLUTE TYPE
TYPE CODE = VS-2

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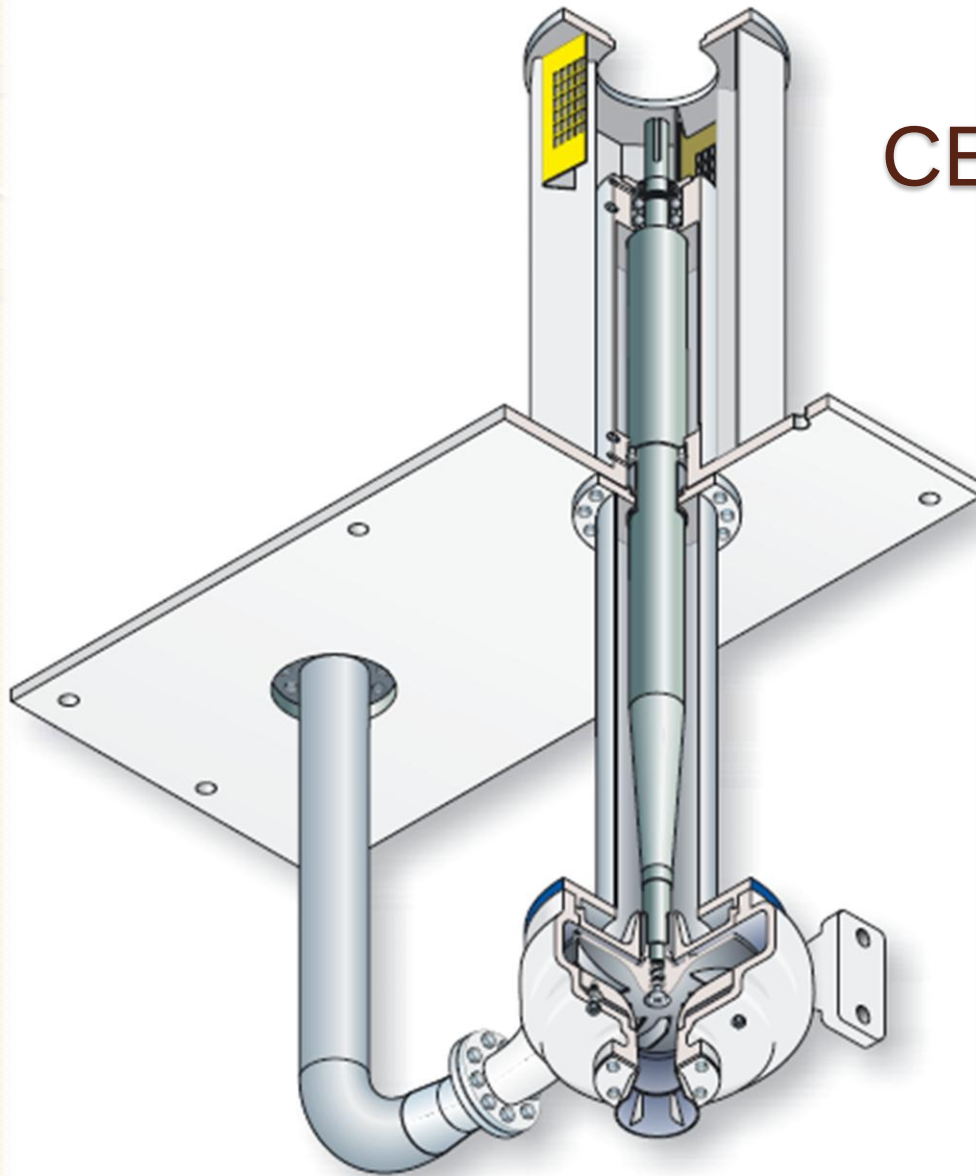
CENTRIFUGAL PUMP CLASIFICATIOIN

CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
SINGLE CASING,
DISCHARGE THROUGH COLUMN,
AXIAL FLOW TYPE
TYPE CODE = VS-3



CENTRIFUGAL PUMP CLASIFICATIOIN

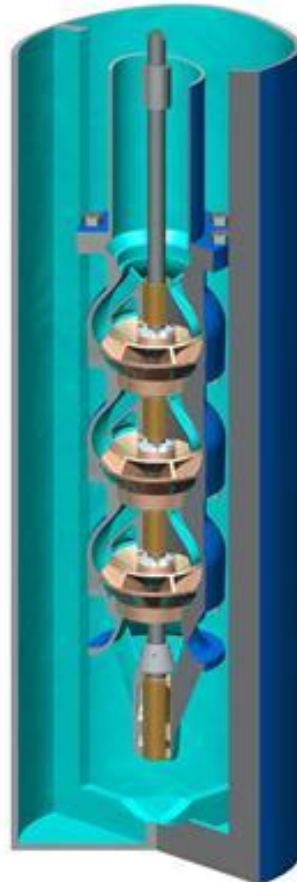
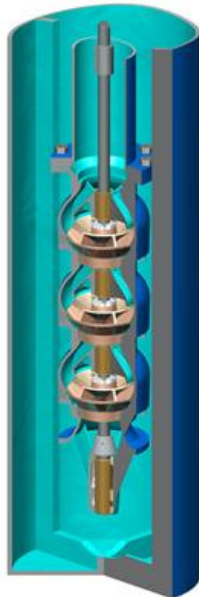
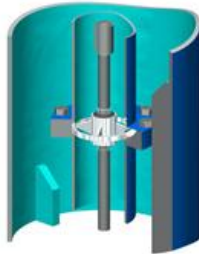
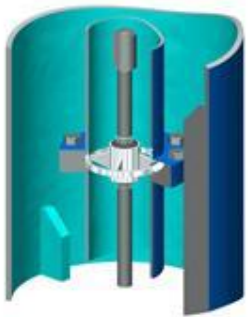
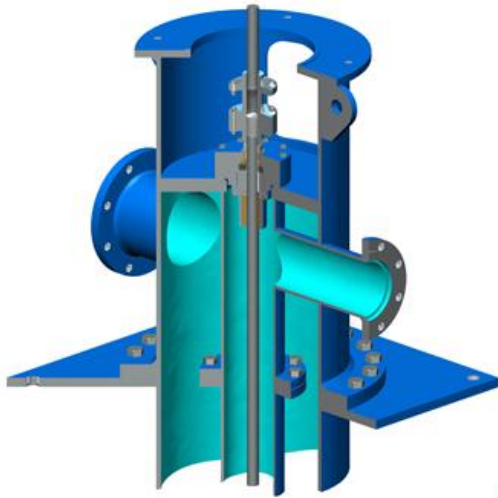
CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
SINGLE CASING,
SEPARATE DISCHARGE,
LINE SHAFT TYPE
TYPE CODE = VS-4



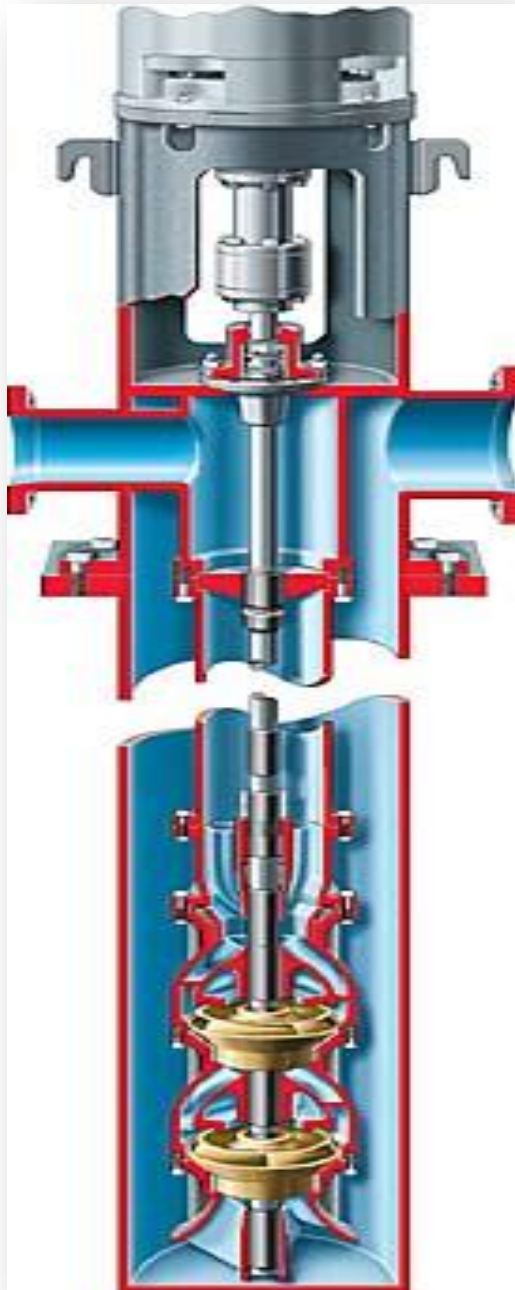
CENTRIFUGAL PUMP CLASIFICATIOIN

CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
SINGLE CASING,
SEPARATE DISCHARGE,
CANTILEVER TYPE
TYPE CODE = VS-5

CENTRIFUGAL PUMP CLASIFICATIOIN



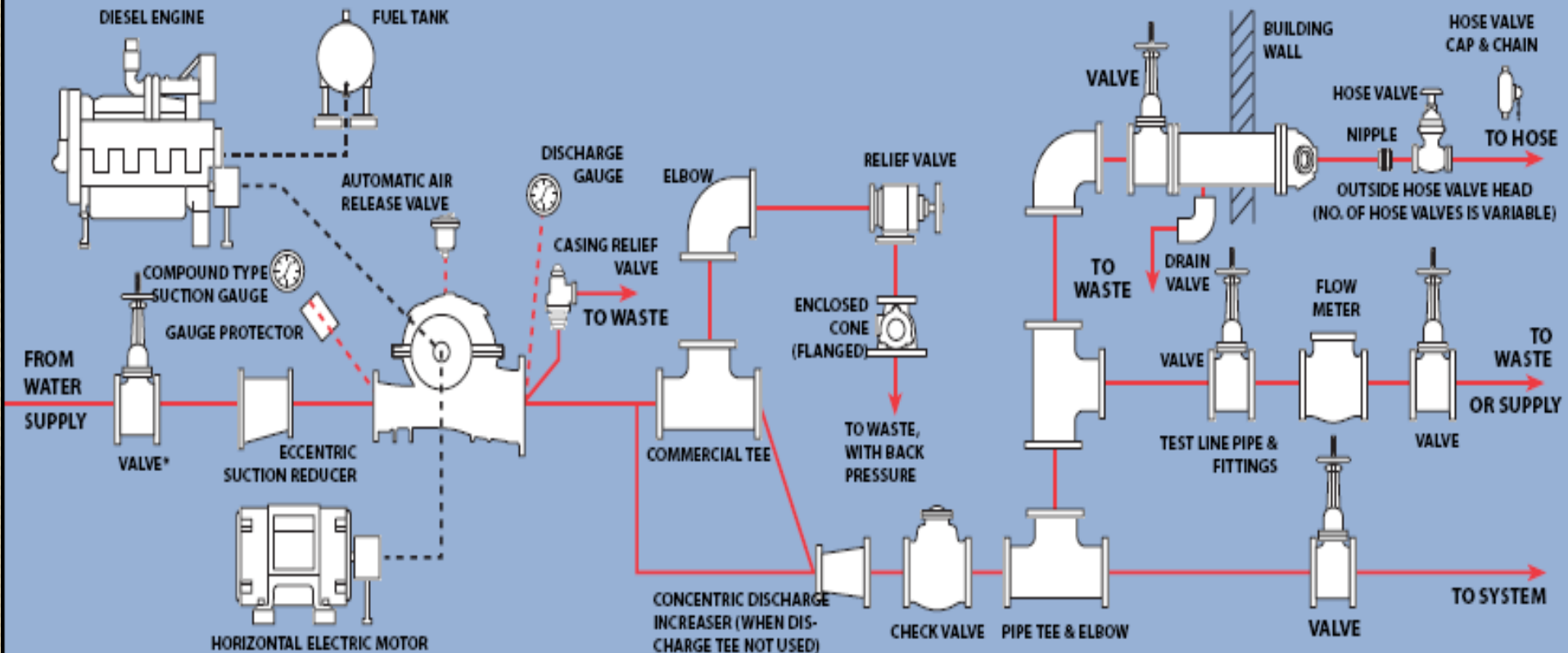
CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
DOUBLE CASING,
DIFFUSER TYPE,
TYPE CODE = VS-6



CENTRIFUGAL PUMP CLASIFICATIOIN

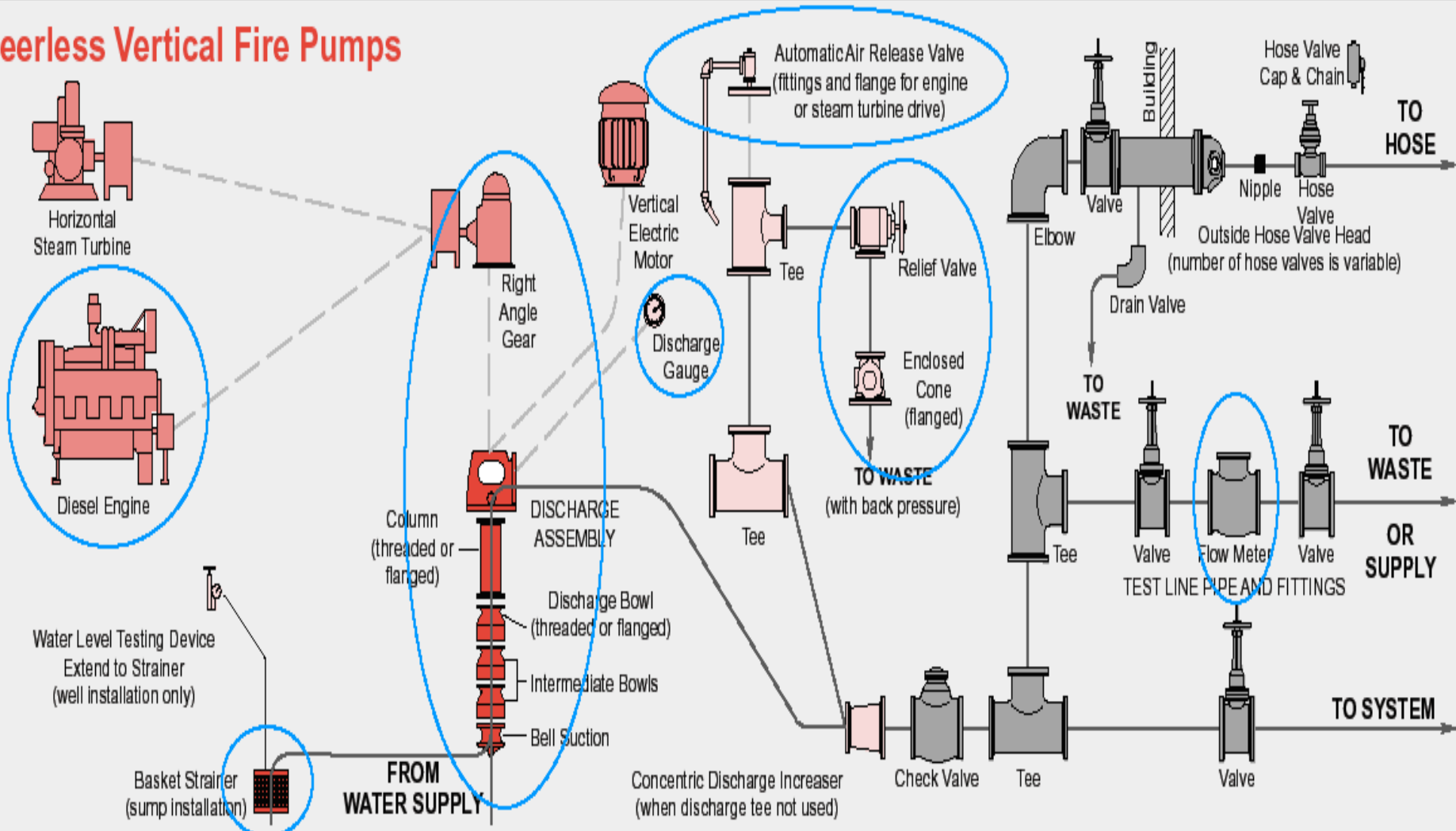
CENTRIFUGAL PUMP,
VERTICAL SUSPENDED,
DOUBLE CASING, VOLUTE
TYPE,
TYPE CODE = VS-7

CENTRIFUGAL PUMP INSTALLATION

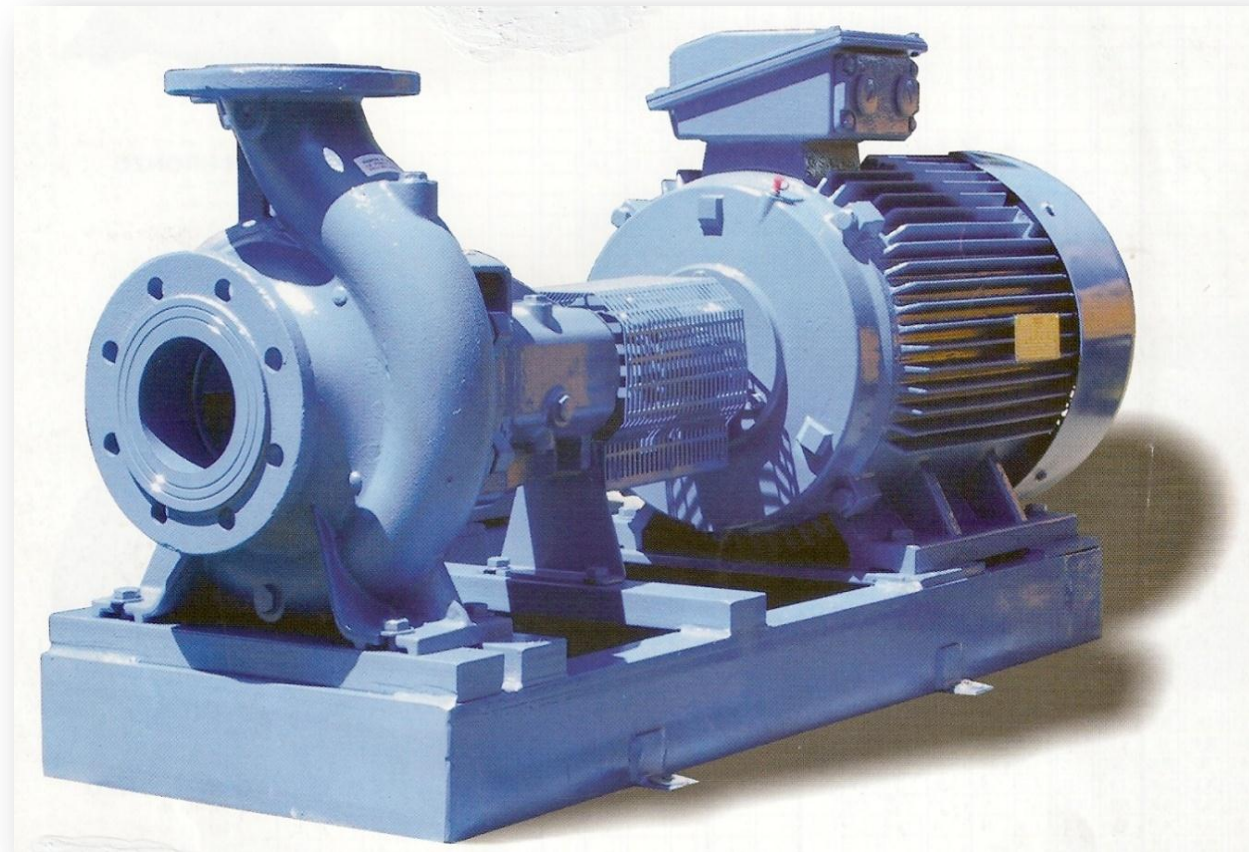


CENTRIFUGAL PUMP INSTALLATION

Peerless Vertical Fire Pumps



PUMP MAINTENANCE



PRESENTED BY :
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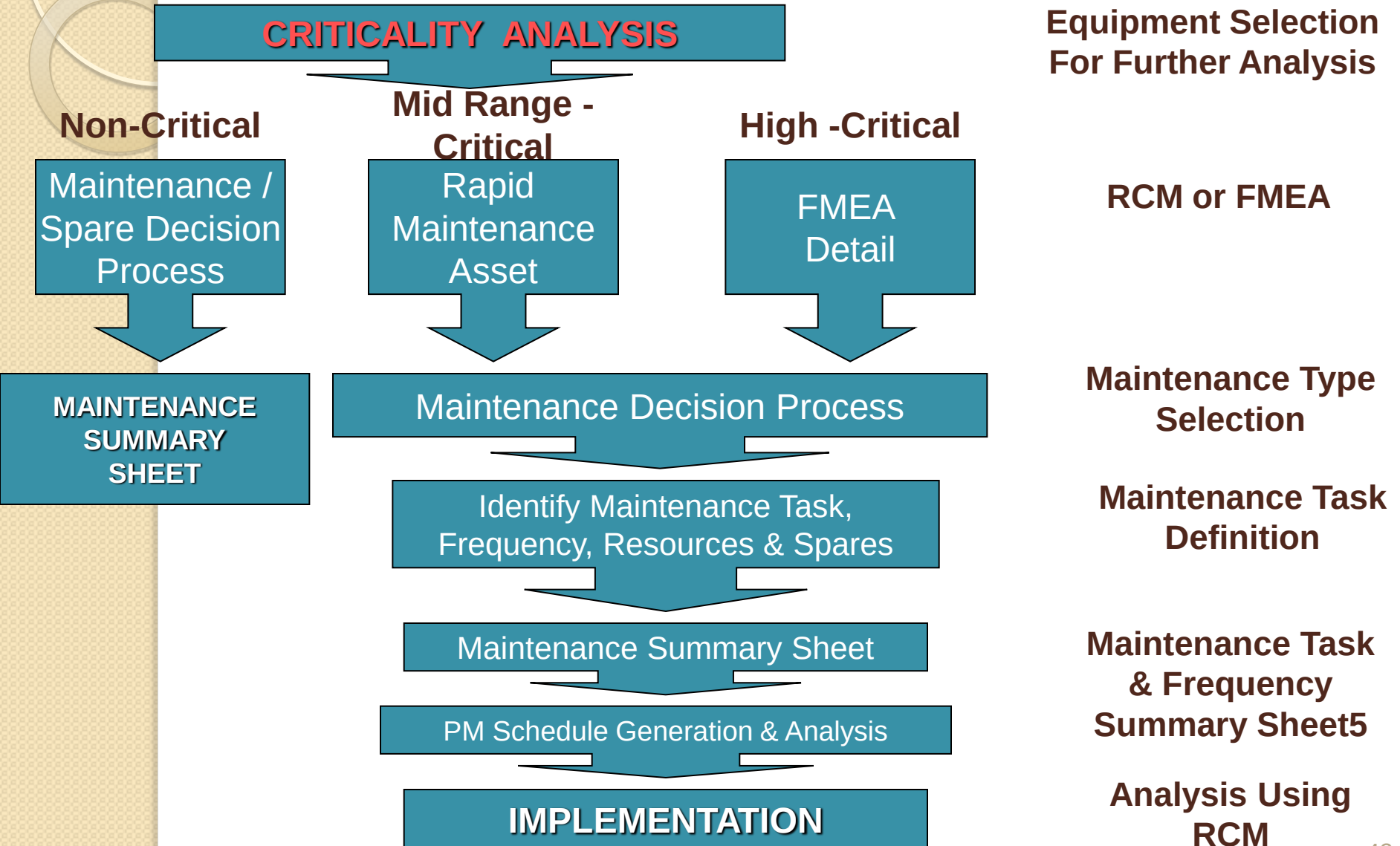
Definition of Maintenance

- Maintenance Pump is function which has an objective :
 - To optimize the overall Pump effectiveness and perform required to ensure availability.
 - Minimize maintenance cost or cost avoidance.
 - Under respect of the necessary conditions for Production

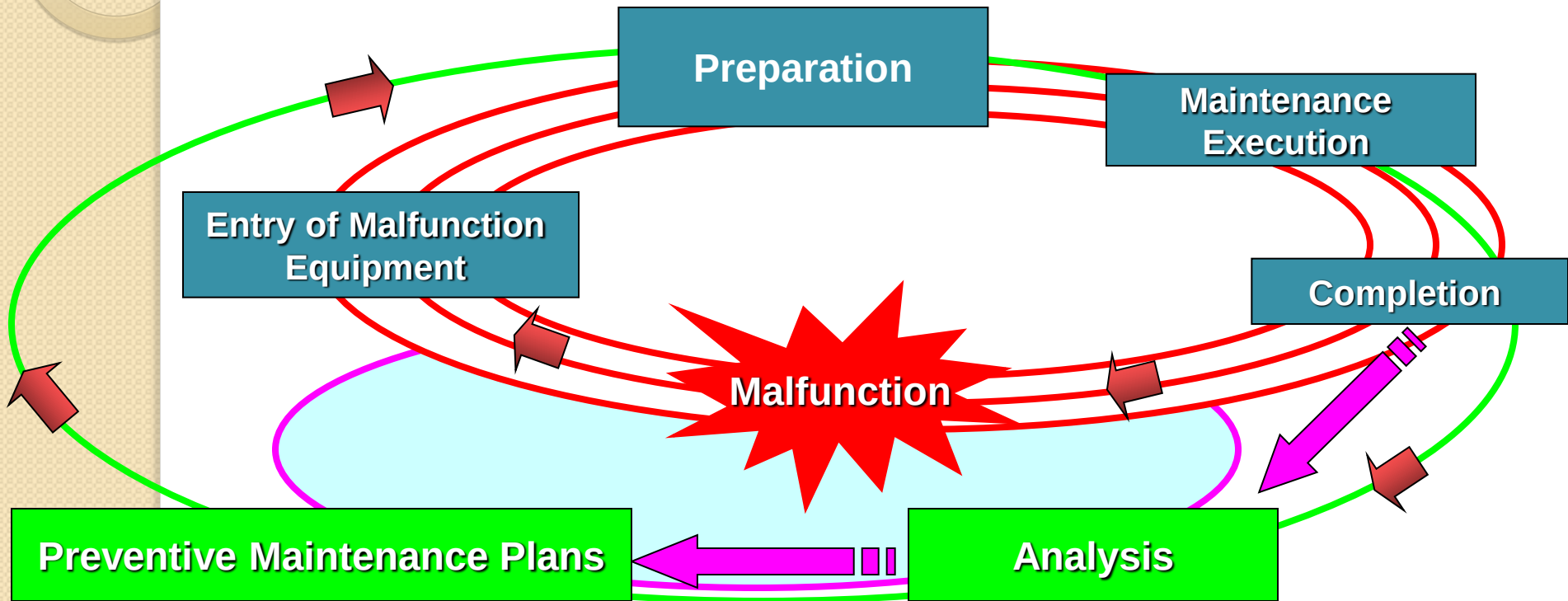
Maintenance Improvement

- TPM (Total Productive Maintenance)
 - Autonomy of operator for maintenance task
 - Improving equipment utilization
 - Better relation between Maintenance - Production
- RCM (Reliability Centered Maintenance)
 - Maintenance model base on reliability equipment
 - Failure Mode Effect Analysis (FMEA)
 - Maximize of technician experiences
- Asset Care and Life Cycle Costing (LCC)
 - Systematic approach for reducing the total maintenance cost of equipment during the whole life time of equipment. (Start from purchase until retirement)

Maintenance Process



Maintenance Process



Maintenance Implementation

- **Design review**
 - **Centrifugal Pump (ANSI/API/DIN/ISO)**
 - **Driver**
 - **Electric Motor – (NEMA, IEC)**
 - **Engine**
 - **Power Supplies**
 - Available indicator power supply on the panel (light on controller)
 - Phase reversal (or normal phase rotation)
 - Both sources of power
 - Loss of phase

Maintenance Implementation

■ Design review

■ Fuel Supply & Arrangement for Diesel Engine driven

- Fuel tank capacity shall equal 1 gal/hp plus 5% for expansion and 5% for sump.
- Fuel tank shall be located above ground, never buried.
- Fuel piping for connection at the Engine shall be flexible hose listed for this application.
- Engine cooling must be part of the Engine assembly can be either a heat exchanger or radiator.
- Heat exchanger water supply shall be taken from the pump discharge.

■ Controllers & Instrumentation

- Low oil pressure
- High Engine coolant temperature
- Failure to start
- Over speed shut down
- Battery failure
- Battery charger failure
- Low air pressure (for air starter Engine)
- Low hydraulic pressure (for hydraulic starter Engine)

Maintenance Implementation

■ Design

- **Pipe & Fittings**
- **Valve (Control valve & Relief Valve)**
 - Automatically air relief valve must be installed for all automatically controlled fire pump to release air from the pump
- **Jockey Pump (Pressure maintenance pump)**
 - Maintain constant line pressure
 - Prevent frequent operation of fire pump in non-emergency situation
 - Check capacity and pressure
 - Check setting pressure (start at 5-10 psi above start pressure of fire pump)
 - Listed pump is not required
 - Check or re-calibrated of pressure switch setting (when necessary)
- **Gauge**
- **Battery Starting**
 - Two battery units must be provided
 - Starting must be alternated between battery
 - Attempt to start – 6 crank period of 15 second each with 5 rest period of 15 second each

Maintenance Implementation

- Perform maintenance activity

Item	Activity	Frequency
Pump house, heating ventilating louvers	Inspection	Weekly
Fire pump system	Inspection	weekly
Pump Operation		
- Flow condition	Test	Weekly
Hydraulic system	Maintenance	Annually
Mechanical Transmission	Maintenance	Annually
Electrical System	Maintenance	Annually
Controller and various component	Maintenance	Annually
Motor	Maintenance	Annually
Diesel Engine system and various component	Maintenance	Annually

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequencny
A	Pump System						
	1. Lubricated Pump Bearing			X			Annually
	2. Check Pump Shaft End-play		X				Annually
	3. Check accuracy of pressure gauge and sensors		X	X			Annually (Change or recalibrated when 5% out of calibration)
	4. Check pump coupling		X				Annually
	5. Wet pit suction screen		X		X		After each pump operation.
B	Mechanical Transmission						
	1. Lubricating coupling			X			Annually
	2. Lubricant Right-Angle Gear Drive			X			Annually

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequencny
C	Electrical System						
	1. Check isolation switch and circuit breaker.					X	Monthly
	2. Trip circuit breaker (if mechanism provided)					X	Annually
	3. Operate manual starting means (electrical)					X	Semiannually
	4. Inspect and operate emergency manual starting means (without power)	X				X	Annually
	5. Tighten electrical connection as necessary.		X				Annually
	6. Lubricate mechanical moving parts (excluding starter and relays)		X				Annually
	7. Calibrated pressure switch setting		X				Annually
	8. Greasing motor bearings			X			Annually

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequencny
D	Diesel Engine System						
1	Fuel Tank						
	a. Tank level	X	X				Monthly
	b. Tank float switch	X				X	Annually
	c. Solenoid valve operation	X				X	Semiannually
	d. Strainer, filter or dirt leg or combination thereof.				X		Annually
	e. Water and foreign material in tank				X		Annually
	f. Water in system		X		X		Annually
	g. Flexible hose and connector	X					Annually
	h. Tank vents and overflow piping unobstructed		X			X	Annually
	i. Piping	X					Annually

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequency
D	Diesel Engine System						
2	Lubrication System						
	a. Oil level	X	X				Weekly
	b. Oil change			X			50 hours or annually
	c. Oil Filter			X			50 hours or annually
	d. Lube oil heater		X				Weekly
	e. Crankcase breather	X		X	X		Quarterly

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequencny
D	Diesel Engine System						
3	Cooling System						
	a). Level	X	X				Weekly
	b). Antifreeze protection level					X	Semiannually
	c). Antifreeze			X			Annually
	d). Adequate cooling water to heat exchanger.		X				Weekly
	e). Rod out heat exchanger				X		Annually
	f). Water pump	X			X		Weekly
	g). Condition of flexible hose & connection	X	X				Weekly
	h). Jacket water heater		X				Weekly
	i). Inspect duck work, clean louvers (combustion air)	X	X	X			Annually
	j). Water strainer				X		Quarterly

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequency
D	Diesel Engine System						
4.	Exhaust System						
	a). Leakage	X	X				Weekly
	b). Drain condensate trap		X				Weekly
	c). Insulation and fire hazard	X					Quarterly
	d). Excessive back pressure					X	Annually
	e). Exhaust system hangers and support	X					Annually
	f). Flexible exhaust section	X					Semiannually

Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequency
D	Diesel Engine System						
5	Battery System						
	a). Electrolyte level		X				Weekly
	b). Terminal clean and tight	X	X				Quarterly
	c). Remove corrosion, case exterior clean and dry	X		X	X		Monthly
	d). Specific gravity or state of charge					X	Monthly
	e). Charger and charge rate	X					Monthly
	f). Equalize charge		X				Monthly

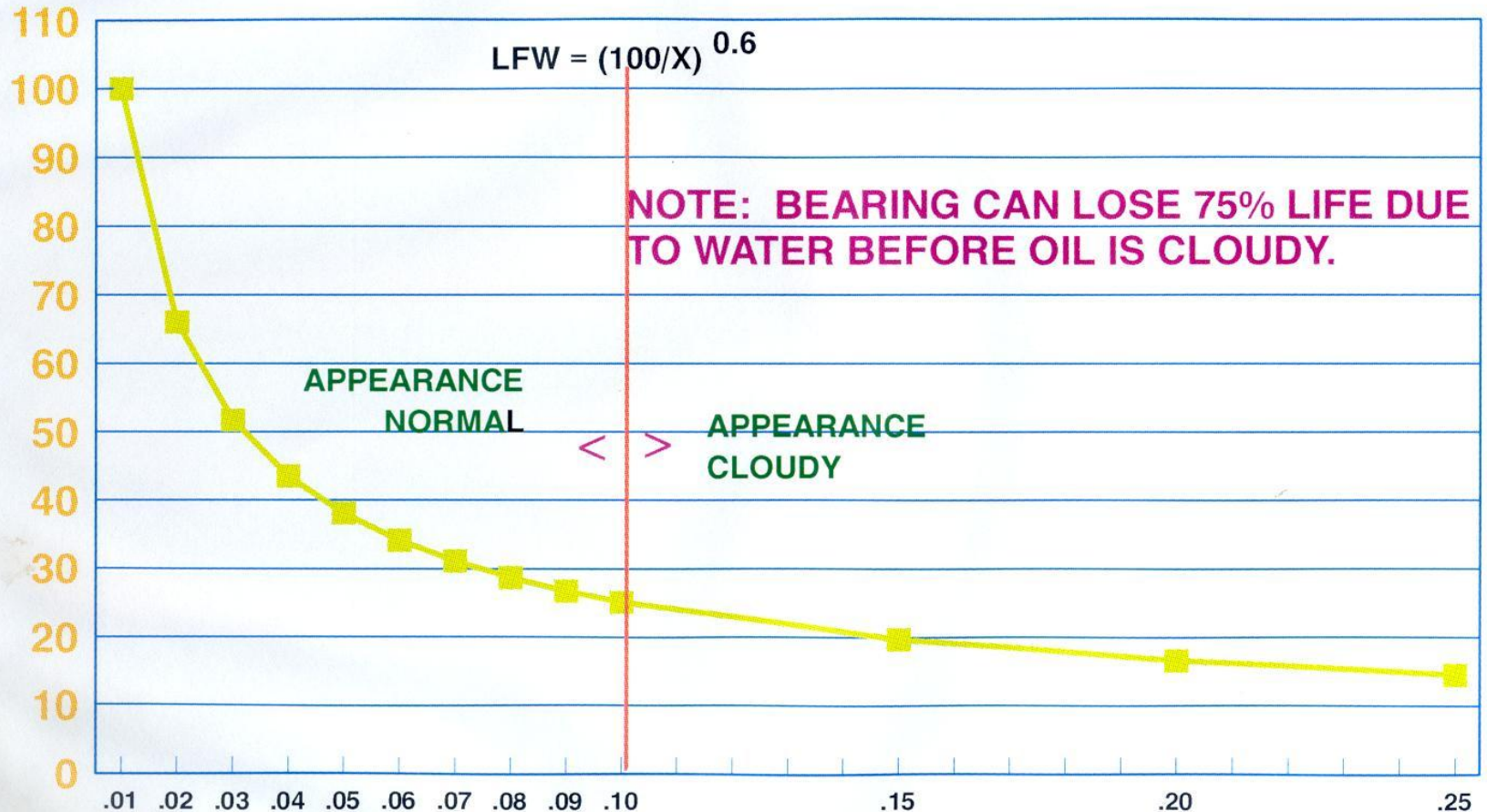
Maintenance Implementation

- Perform maintenance activity

Item	Complete as Applicable	Visual	Check	Change	Clean	Test	Frequency
D	Diesel Engine System						
6	Electrical System						
	a). General inspection	X					Weekly
	b). Tighten control and power wiring connection		X				Annually
	c). Wire chafing where subject to movement.	X	X				Quarterly
	d). Operation of safeties and alarm		X			X	Semiannually
	e). Boxes, panel and cabinets				X		Semiannually
	f). Circuit breaker and fuses	X	X				Monthly
	g). Circuit breaker and fuses			X			Biennially

EFFECT OF WATER IN OIL ON BEARING LIFE

% BRG LIFE REMAINING



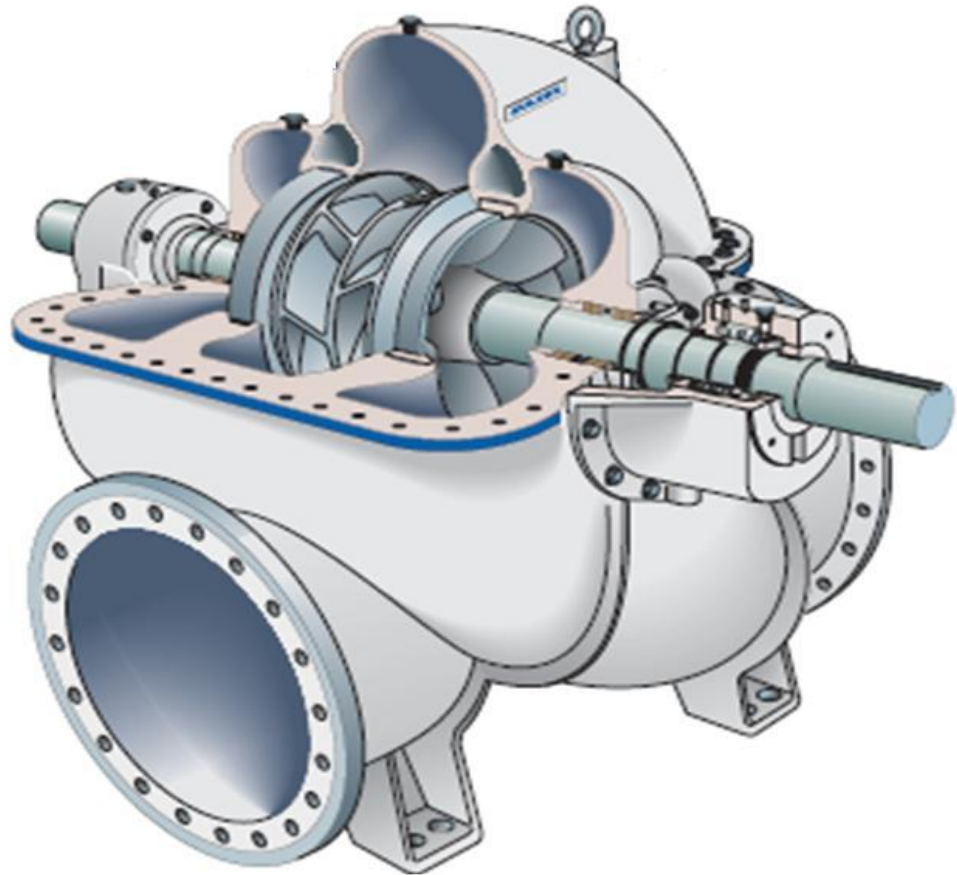
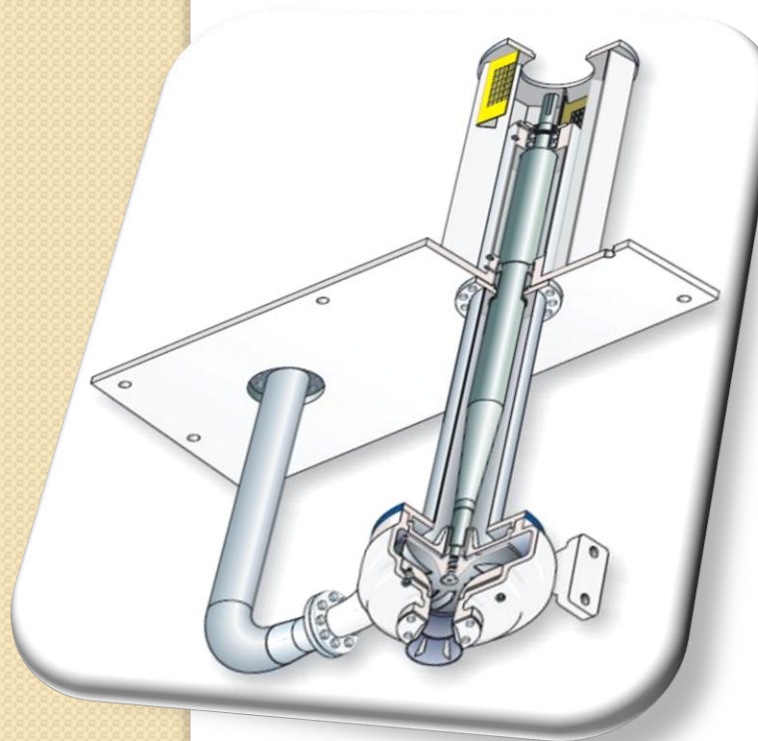
% Water in Oil

NOTE: LESS THAN 1000 PPM (.1%) WATER IN OIL IS NOT DETECTABLE BY APPEARANCE-PAST THIS POINT OIL APPEARS CLOUDY. REFERENCE: "MACHINE DESIGN" JULY '86, "HOW DIRT AND WATER EFFECT BEARING LIFE" BY TIMKEN BEARING CO.

Consequences of Bearing Failure

- **Lost production**
- **Increased vibration effects equipment performance**
- **Shortened seal life**
- **High heat generation (risk of fire)**
- **Coupling failure due to high vibration**
- **High maintenance costs**

Centrifugal PUMP TROUBLESHOOTING



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Centrifugal PUMP PROBLEMS :

LITTLE OR NO DISCHARGE FLOW

POSSIBLE CAUSE :

1. PUMP NOT PRIMED
2. SPEED TOO LOW
3. SYSTEM HEAD TOO HIGH
4. SUCTION LIFT HIGHER THAN THAT FOR WHICH PUMP IS DESIGNED.
5. IMPELLER COMPLETELY PLUGGER
6. IMPELLER INSTALLED BACKWARD
7. WRONG DIRECTION OF ROTATION
8. AIR LEAK THROUGH STUFFING BOX
9. WELL DRAW-DOWN BELOW MINIMUM SUBMERGENCE
10. PUMP DAMAGE DURING INSTALLATION
11. BROKEN LINE SHAFT OR COUPLING
12. IMPELLER LOOSE ON SHAFT
13. CLOSED SUCTION OR DISCHARGE VALE

Centrifugal PUMP PROBLEMS :

INSUFFICIENT DISCHARGE FLOW OR PRESSURE
POSSIBLE CAUSE :

1. AIR LEAKS IN SUCTION AND STUFFING BOXES
2. SPEED TOO LOW
3. SYSTEM HEAD HIGHER THAN ANTICIPATED
4. INSUFFICIENT NPSHA
5. FOOT VALVE TOO SMALL
6. WEAR RING WORN
7. IMPELLER DAMAGE
8. IMPELLER(S) LOOSE ON SHAFT
9. VORTEX AT SUCTION SUPPLY
10. SUCTION OR DISCHARGE VALVE PARTIALLY CLOSED
11. IMPELLER INSTALLED BACKWARDS
12. WRONG DIRECTION ROTATION

Centrifugal PUMP PROBLEMS :

LOSS OF SUCTION

POSSIBLE CAUSE :

1. LEAKY SUCTION LINE
2. WATER LINE TO SEAL PLUGGED
3. SUCTION LIFT TOO HIGH OR INSUFFICIENT NPSHA
4. AIR OR GAS IN LIQUID
5. SUCTION FLANGE GASKET DEFECTIVE
6. CLOGGED STRAINER
7. EXCESSIVE WELL DRAW-DOWN

Centrifugal PUMP PROBLEMS :

EXCESSIVE POWER CONSUMPTION

POSSIBLE CAUSE :

1. SPEED TOO HIGH
2. SYSTEM HEAD LOWER THAN RATING, PUMPS TOO MUCH LIQUID (RADIAL & MIXED FLOW PUMPS)
3. SYSTEM HEAD HIGHER THAN RATING, PUMP TOO LITTLE LIQUID (AXIAL FLOW PUMPS)
4. SPECIFIC GRAVITY OR VISCOSITY OF LIQUID PUMPED IS TOO HIGH
5. SHAFT BENT
6. ROTATING ELEMENT BINDS
7. STUFFING BOXES TOO TIGHT
8. WEARING RING WORN
9. UNDERSIZE MOTOR CABLE
10. INCORRECT LUBRICATION
11. MECHANICAL SEAL POWER CONSUMPTION
12. PUMP AND MOTOR OPERATING IN REVERSE DIRECTION
13. IMPELLER MOUNTED ON SHAFT WITH INVERTED ORIENTATION.

Centrifugal PUMP PROBLEMS :

PUMP VIBRATION

POSSIBLE CAUSE :

1. MISALIGNMENT SHAFT
2. UNBALANCE ROTOR ELEMENT
 1. IMPELLER ERRODED
 2. SHAFT RUBBING
 3. BEARING FAILURE
3. RESONANCE
4. PUMP CAVITATION
5. JOURNAL BEARING CLEARANCE OVERSIZE
6. SHAFT WORN-OUT
7. ROTATING ELEMENT CRITICAL SPEED
8. PUMP RUNNING LESS THAN MINIMUM CONTINUOUS FLOW.
9. PUMP TOO MUCH FLOW

VIBRATION LIMITS FOR OVERHUNG AND BETWEEN BEARING PUMP

ANSI/API Standard 610 / ISO 13709, 10th Edition, October 2004

CRITERIA	LOCATION OF VIBRATION MEASUREMENT	
	BEARING HOUSING	PUMP SHAFT
	PUMP BEARING TYPE	
	ALL	HYDRODINAMIC JOURNAL BEARING
	VIBRATION AT ANY FLOWRATE WITHIN THE PUMP'S PREFERRED OPERATING REGION	
OVERALL	FOR PUMP RUNNING AT UP TO 3600 r/min AND ABSORBING UP TO 300kW (400hp) PER STAGE : $V_u < 3,0 \text{ mm/s RMS}$ (0,12 in/s RMS) FOP PUMP RUNNING ABOVE 3600 r/min OR ABSORBING MORE THAN 300 Kw (400hp) PER STAGE	$A_u < (5,2 \times 10^6 / n)^{0.5} \mu\text{m PEAK TO PEAK}$ $\{(8000 / n)^{0.5} \text{ mils PEAK TO PEAK}\}$ NOTE TO EXCEED: $A_u < 50 \mu\text{m PEAK TO PEAK}$ (2,0 mils PEAK TO PEAK)
DISCRETE FREQUENCIES	$V_f < 0,67 v_u$	FOR $f < n$: $A_f < 0,33 A_u$
ALLOWABLE ICREASE IN VIBRATION AT FLOWS OUTSIDE THE PREFERRED OPERATING REGION BUT WITHIN THE ALLOWABLE OPERATING REGION	30 %	30 %

POWER CALCULATED FOR BEP OF RATED IMPELLER WITH LIQUID RELATIVE DENSITY (SPECIFIC GARVITY) = 1.0

WHERE :

V_u = IS UNFILTERED VELOCITY, AS ,MEASURED

V_f = IS FILTERED VELOCITY

A_u = IS THE AMPLITUDE OF UNFILTERED DISPLACEMENT, AS MEASURED

A_f = IS AMPLITUDE OF FILTERED DISPLACEMENT

f = IS THE FREQUANCY

N = IS THE ROTAIONAL SPEED, EXPRESSED IN REVOLUTION PER MINUTE

VIBRATION VELOCITY AND AMPLITUDE VALUES CALCULATED FROM THE BASIC LIMITS SHALL BE ROUNDED OFF TO TWO SIGNIFICANT FIGURE.

VIBRATION LIMITS FOR VERTICAL SUSPENDED PUMP

ANSI/API Standard 610 / ISO 13709, 10th Edition, October 2004

CRITERIA	LOCATION OF VIBRATION MEASUREMENT	
	PUMP THRUST BEARING HOUSING OR MOTOR MOUNTING FLANGE	PUMP SHAFT (ADJACENT TO BEARING)
	PUMP BEARING TYPE	
	ALL	HYDRODINAMIC GUIDE BEARING ADJACENT TO ACCESSIBLE REGION OF SHAFT
	VIBRATION AT ANY FLOWRATE WITHIN THE PUMP'S PREFERRED OPERATING REGION	
OVERALL	$V_u < 3,0 \text{ mm/s RMS}$ (0,12 in/s RMS)	$A_u < (6,2 \times 10^6 / n)^{0.5} \mu\text{m PEAK TO PEAK}$ {(10000 /n) ^{0.5} mils PEAK TO PEAK} NOTE TO EXCEED: $A_u < 100 \mu\text{m PEAK TO PEAK}$ (4,0 mils PEAK TO PEAK)
DISCRETE FREQUENCIES	$V_f < 0,67 v_u$	$A_f < 0,75 A_u$
ALLOWABLE ICREASE IN VIBRATION AT FLOWS OUTSIDE THE PREFERRED OPERATING REGION BUT WITHIN THE ALLOWABLE OPERATING REGION	30 %	30 %

VIBRATION VELOCITY AND AMPLITUDE VALUES CALCULATED FROM THE BASIC LIMITS SHALL BE ROUNDED OFF TO TWO SIGNIFICANT FIGURES

WHERE :

V_u = IS UNFILTERED VELOCITY, AS ,MEASURED

V_f = IS FILTERED VELOCITY

A_u = IS THE AMPLITUDE OF UNFILTERED DISPLACEMENT, AS MEASURED

A_f = IS AMPLITUDE OF FILTERED DISPLACEMENT

N = IS THE ROTAIONAL SPEED, EXPRESSED IN REVOLUTION PER MINUTE

NET POSITIVE SUCTION HEAD AVAILABLE (NPSHa)

NET POSITIVE SUCTION HEAD AVAILABLE (NPSHa) IS THE TOTAL SUCTION HEAD OF LIQUID ABSOLUTE DETERMINED AT THE FIRST STAGE IMPELLER DATUM, LESS THE ABSOLUTE VAPOR PRESSURE OF THE LIQUID IN HEAD OF LIQUID PUMPED:

$$\text{NPSHa} = h_{sa} - h_{vp}$$

WHERE :

$$\begin{aligned} h_{sa} &= \text{TOTAL SUCTION HEAD ABSOLUTE} \\ &= h_{atm} + h_s \end{aligned}$$

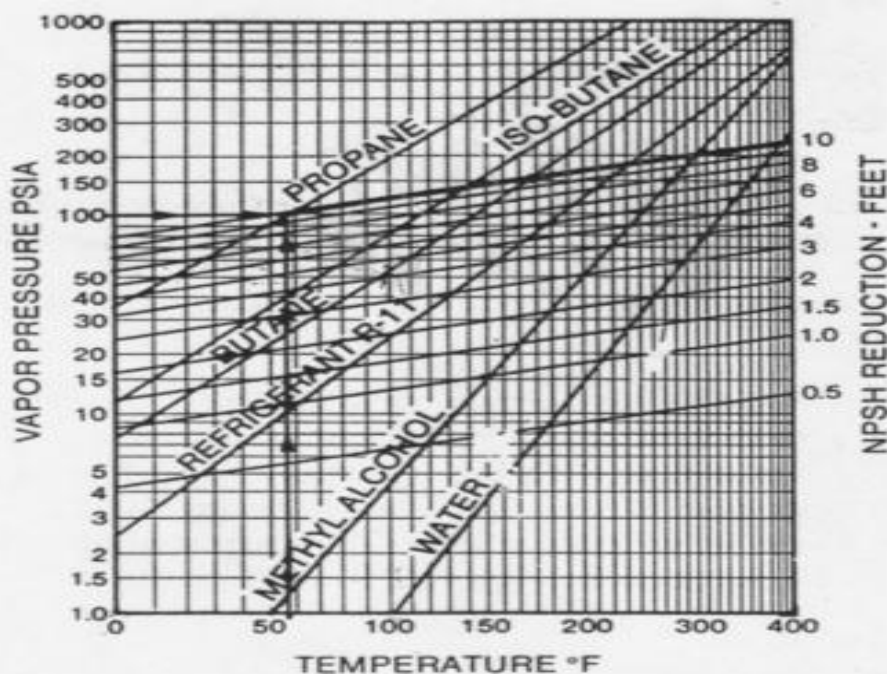
OR :

$$\text{NPSHa} = h_{atm} + h_s - h_{vp}$$

OR :

$$\begin{aligned} \text{(METRIC)} \quad \text{NPSHa} &= ((P_{atm} - P_{vp}) / 9.8 \text{ s}) + \\ &h_s \end{aligned}$$

$$\begin{aligned} \text{(US UNITS)} \quad \text{NPSAa} &= (2.31/s (P_{atm} - P_v)) + \\ &h_s \end{aligned}$$



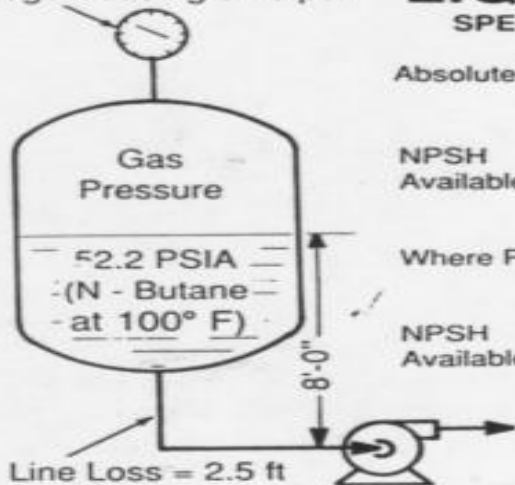
**FIGURE 12B
NPSH REDUCTION
FOR PUMPS
HANDLING
HYDROCARBON
LIQUIDS
AND HIGH
TEMPERATURE
WATER**

**FIGURE 13
NPSH CALCULATION FOR
LIQUID AT BOILING POINT**

SPECIFIC GRAVITY OF N-BUTANE AT 100°F = 0.56

$$\begin{aligned}\text{Absolute Pressure} &= \text{Gauge Pressure} + \text{Atmospheric Pressure} \\ &= \text{Gauge Pressure} + 14.7\end{aligned}$$

Gauge Reading 37.5 psi



$$\text{NPSH Available} = \text{Absolute Press., Ft.} - \text{Vapor Press., Ft.} - \text{Line Loss., Ft.} + \text{Difference In Elev., Ft.}$$

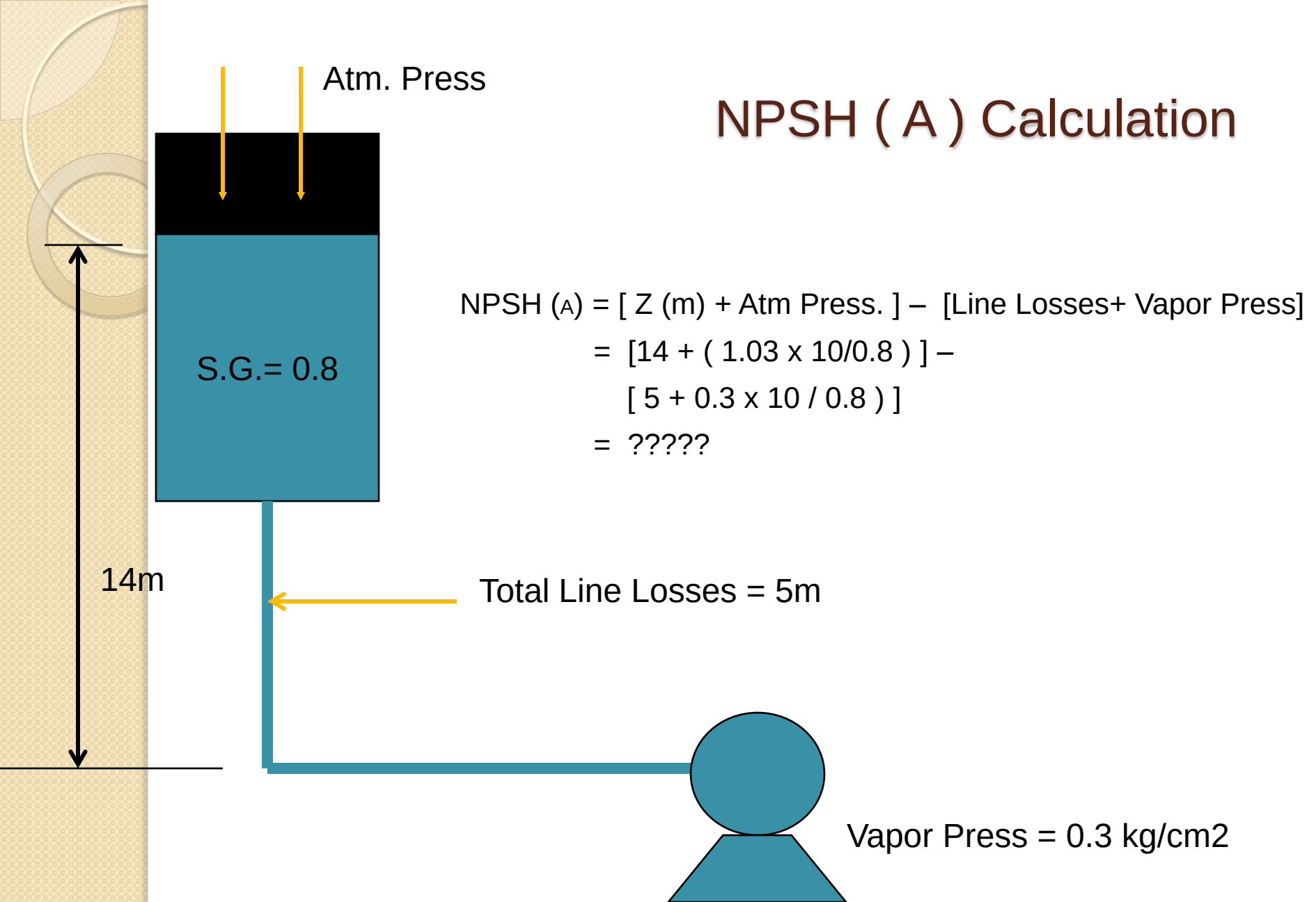
$$\text{Where Pressure in Feet} = \frac{(\text{Pressure, PSIA}) (2.31)}{(\text{Specific Gravity})}$$

$$\begin{aligned}\text{NPSH Available} &= \frac{(37.5 + 14.7) (2.31)}{(0.56)} - \frac{(52.2) (2.31)}{(0.56)} - 2.5 + 8 \\ &= 215.3 - 215.3 - 2.5 + 8 \\ &= 5.5 \text{ Feet}\end{aligned}$$

**NPSH AVAILABLE MUST
BE GREATER THAN
NPSH REQUIRED BY THE PUMP**

NPSH (A) Calculation

$$\begin{aligned}\text{NPSH (A)} &= [Z \text{ (m)} + \text{Atm Press.}] - [\text{Line Losses} + \text{Vapor Press}] \\ &= [14 + (1.03 \times 10 / 0.8)] - \\ &\quad [5 + 0.3 \times 10 / 0.8] \\ &= \text{?????}\end{aligned}$$

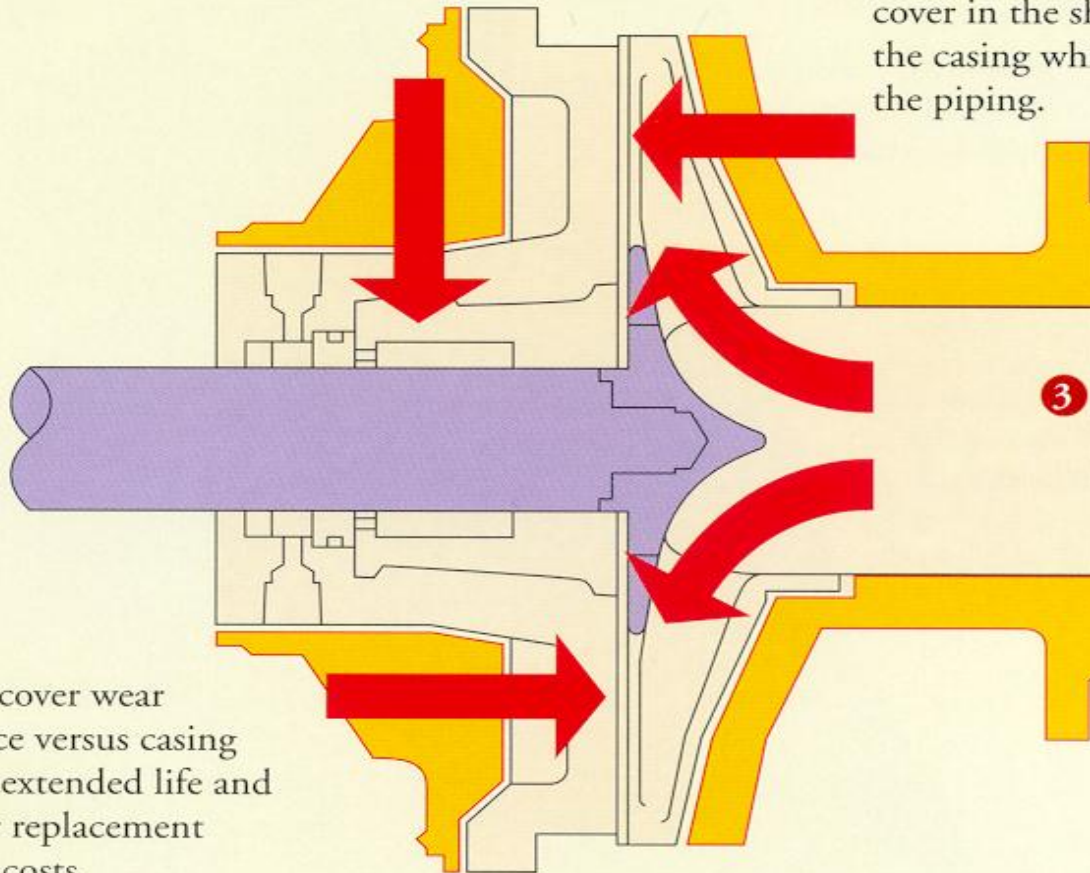


1 Low predictable seal chamber pressure gives longer seal life.

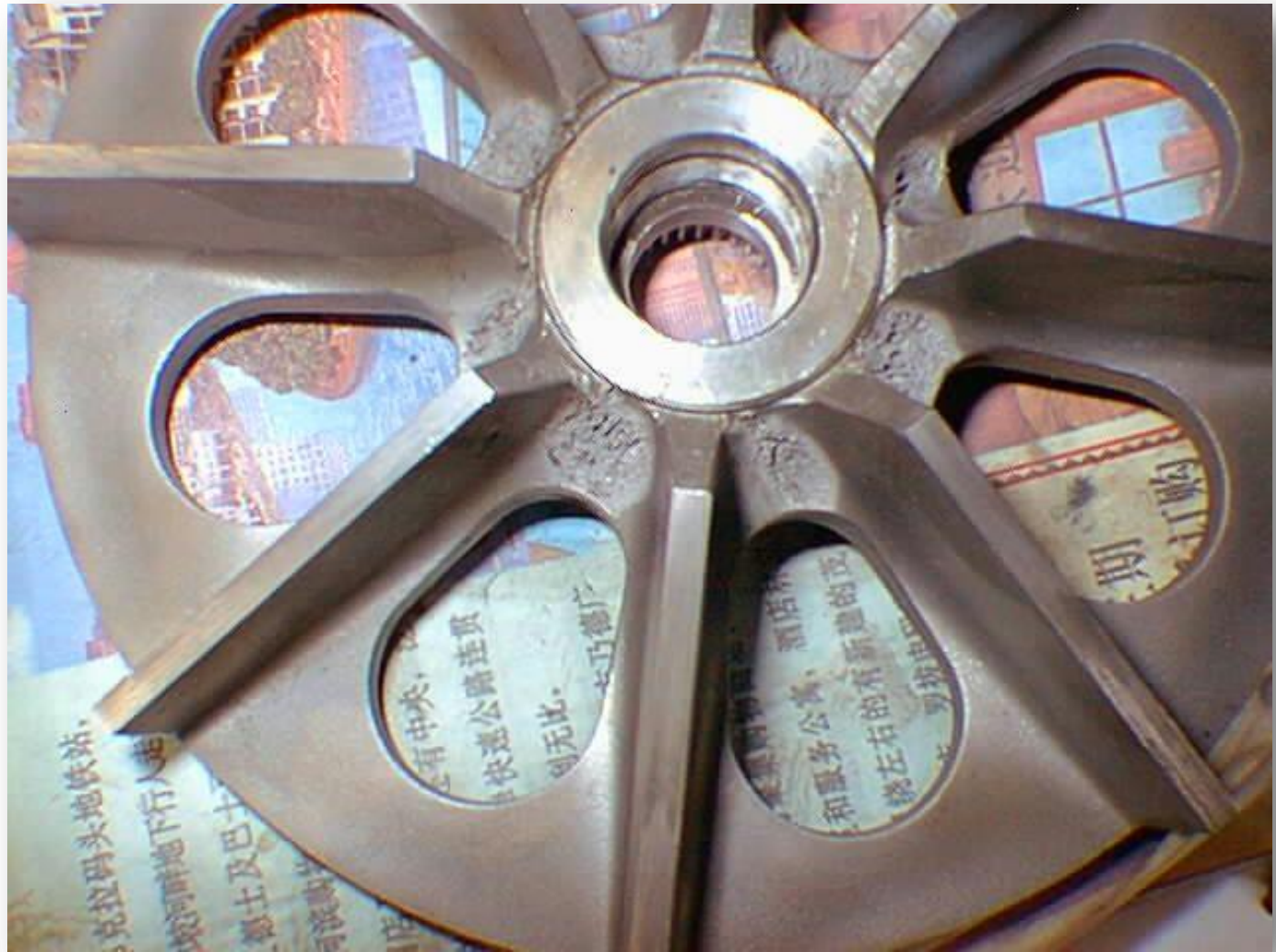
2 Clearance is set to the rear cover in the shop - not to the casing which is left in the piping.

3 Lowest overall required NPSH of any standard chemical process pump.

4 Rear cover wear surface versus casing gives extended life and lower replacement parts costs.



CAVITATION PROBLEM



CAVITATION PROBLEM



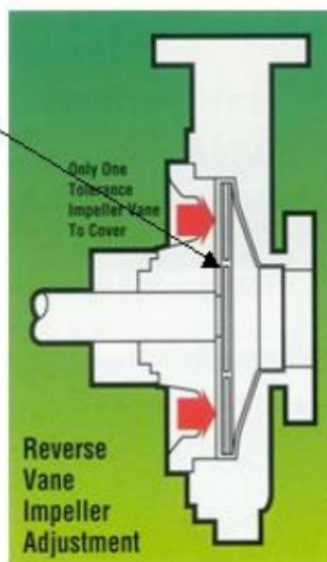
CAVITATION PROBLEM



Reverse Vane Impeller

Set at .018"

Affects:
Performance
Box Pressure
Efficiency
Thrust Loads



Total Available Adjustment .090"

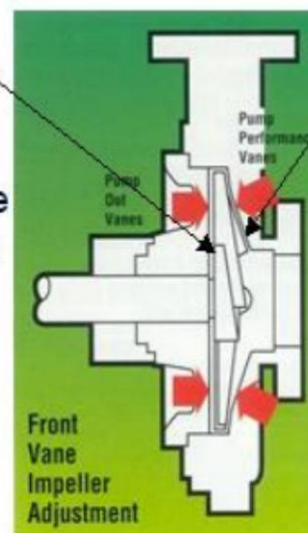
All Critical Settings to Same Location



Open Style Impeller

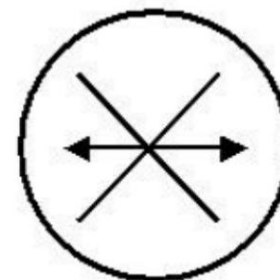
Gap +/- .027"

Affects:
Box pressure
Thrust loads



Total Available Adjustment .040"-.050"

You Can't Adjust Both Ways



???

HYDRAULIC HORSEPOWER

THE POWER IMPARTED TO THE LIQUID :

METRIC (Kw)

$$H \text{ hp} = \frac{Q \times H \times \text{Sp.Gr}}{366}$$

Where :

Q = Capacity (M3/Hr)

H = Head (Meter)

US Unit (HP)

$$H \text{ hp} = \frac{Q \times H \times \text{Sp.Gr}}{3960}$$

Where :

Q = Capacity (GPM)

H = Head (FEET)

PUMP EFFICIENCY

THE RATIO OF THE PUMP OUTPUT POWER (Pw) TO THE PUMP INPUT POWER (Pp); THAT IS THE RATIO OF THE HYDRAULIC HORSEPOWER TO THE BRAKE HORSEPOWER EXPRESSED AS A PERCENT :

$$\eta_p = \frac{\text{Hydraulic Horsepower}}{\text{Brake Horsepower}} \times 100\%$$

MOTOR HORSEPOWER

THE POWER MEASURED BASE ON MOTOR's AMPERE AND VOLTAGE WHEN THE PUMP RUNNING :

$$\text{Motor (kW)} = \text{Volt} \times \text{Ampere} \times 1.73 \times \text{Cos } \Phi / 1000$$

$$\text{Motor (HP)} = \text{Volt} \times \text{Ampere} \times 1.73 \times \text{Cos } \Phi \times 1.341 / 1000$$



PUMP PERFORMANCE CURVE

(EXAMPLE)



Johnston Pump Company

PT. GUNA PERTIWI CEMERLANG

Customer: JR. McDermott-ConocoPhillips

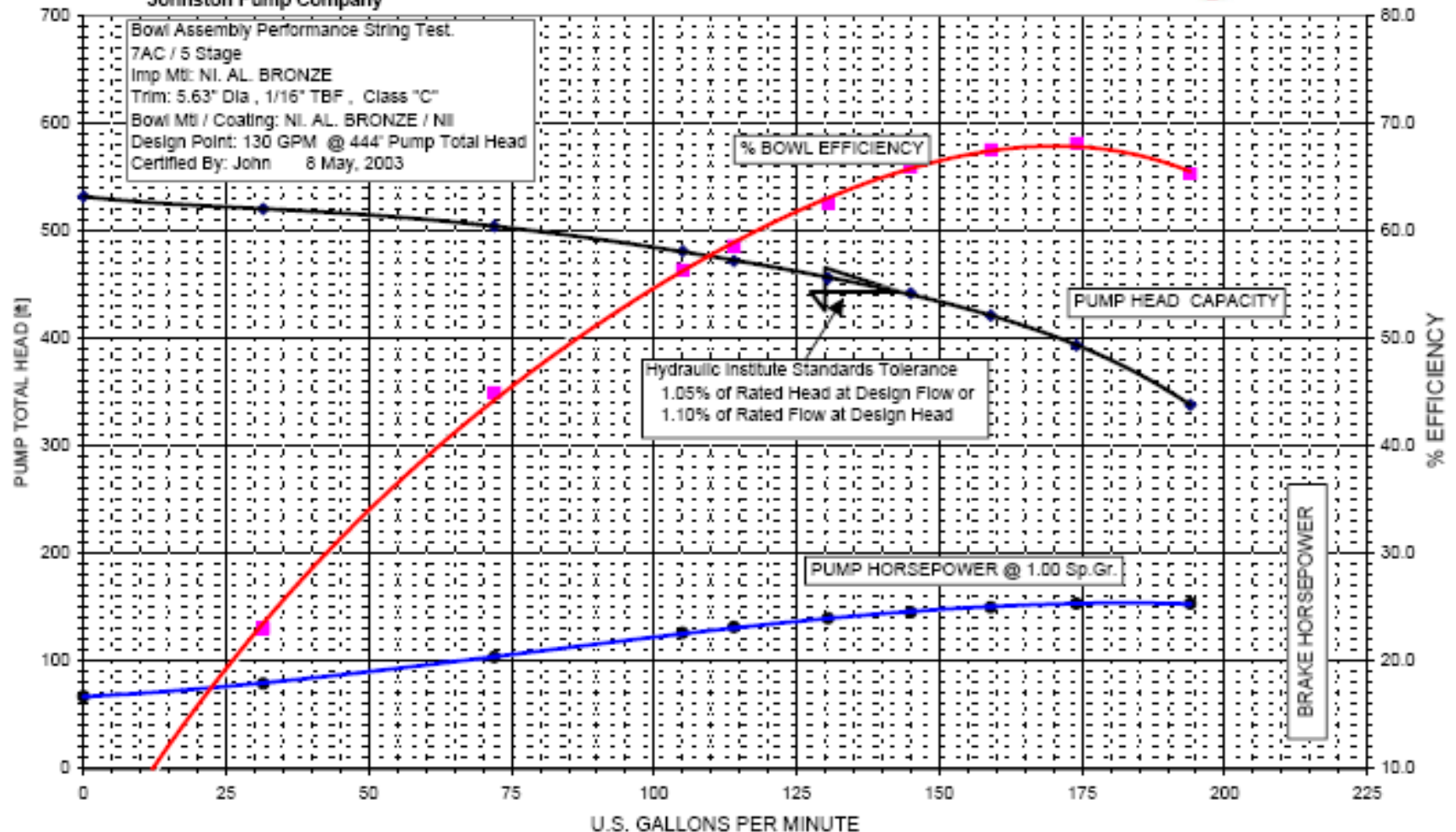
P.O. 0088-JJ004211 TC No.: JKT 03100 Rev #: 0

Pump S/N: NR-2002-434B

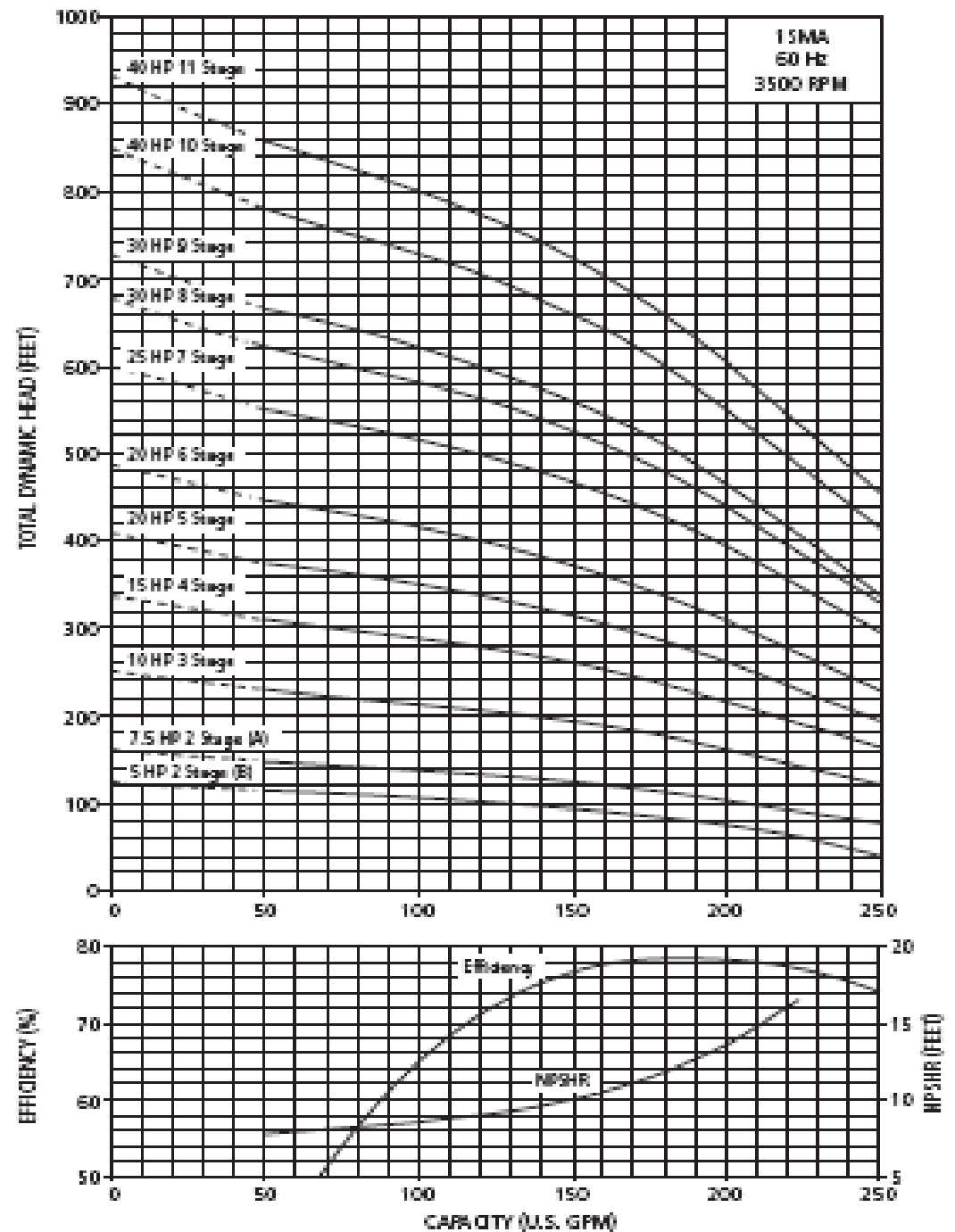
Tag No.: 61PBE-101B

Fluid: Water

3500 RPM



PUMP PERFORMANCE CURVE (EXAMPLE)



CENTRIFUGAL PUMP PERFORMANCE TOLERANCE

ANSI/API Standard 610 / ISO 13709, 10th Edition, October 2004

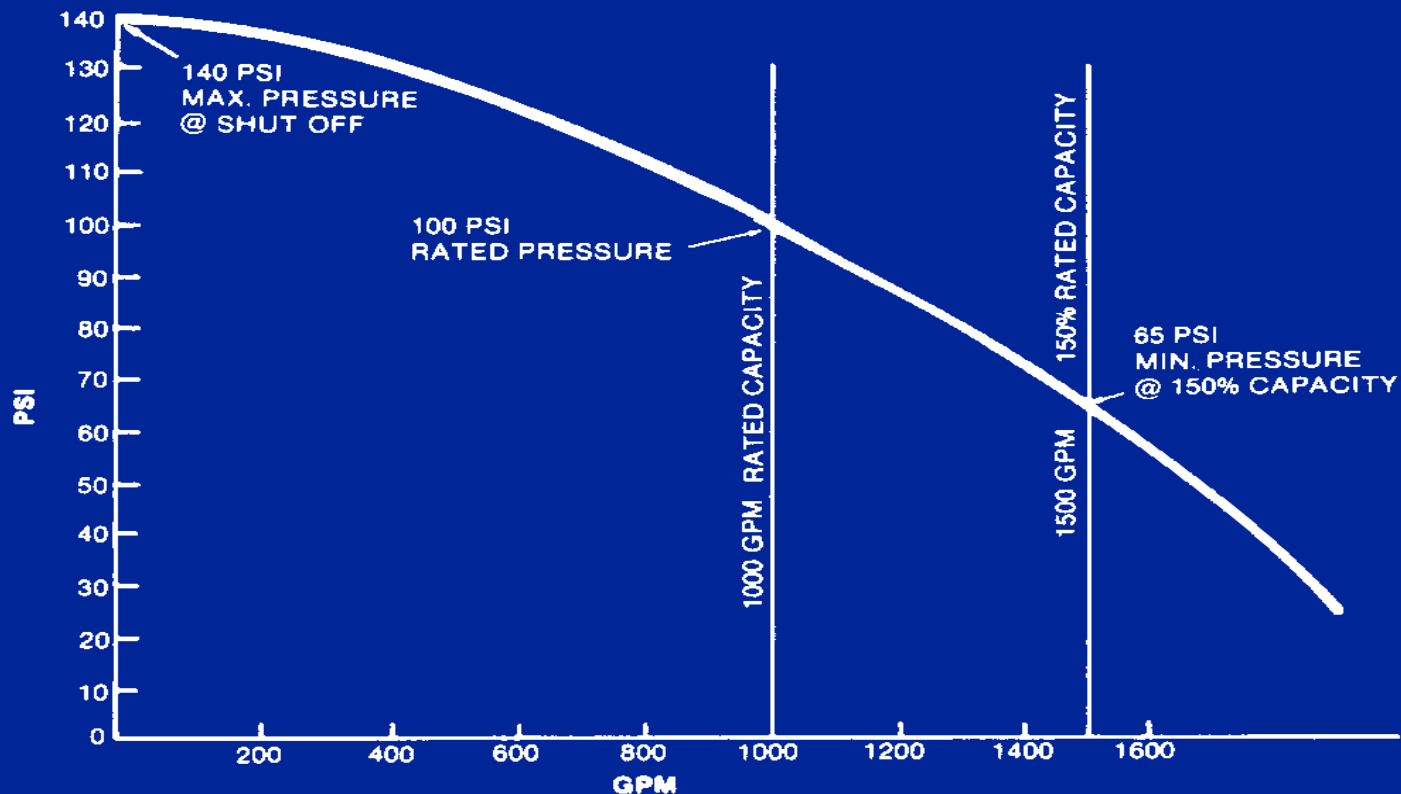
Condition	Rated Point (%)	Shutoff(%)
Rated Differential Head :		
- 0 m to 150 m (0 Ft to 500 Ft)	-2 +5	+10 -10 a
- 151 m to 300 m (501 Ft to 1000 Ft)	-2 +3	+8 -8 a
- > 300 m (1000 Ft)	-2 +2	+5 -5 a
Rated Power	+4 b	-
Rated NPSH	0	-

Note : Efficiency is not rating value

- a. If a rising head flow curve is specified (see 5.1.13) the negative tolerance specified here shall be allowed only if the test curve still shows a rising characteristic.
- b. Under any combination of the above (cumulative tolerances are not acceptable)

CENTRIFUGAL PUMP PERFORMANCE CURVE
FOR FIREWATER PUMP APPLICATION
Refer to Standard NFPA-20, 2007 Edition

2. Pumps shall furnish not less than 150% of rated capacity at not less than 65% of the total rated head and the shut off head shall not exceed 140% of the rated head. See example:

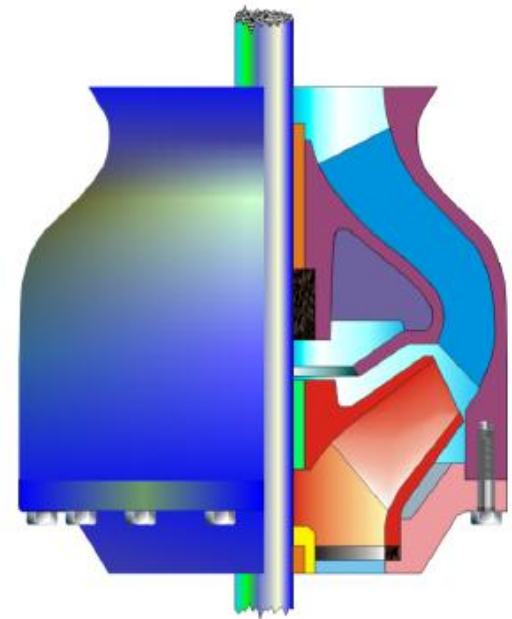


WEAR RING AND RUNNING CLEARANCE

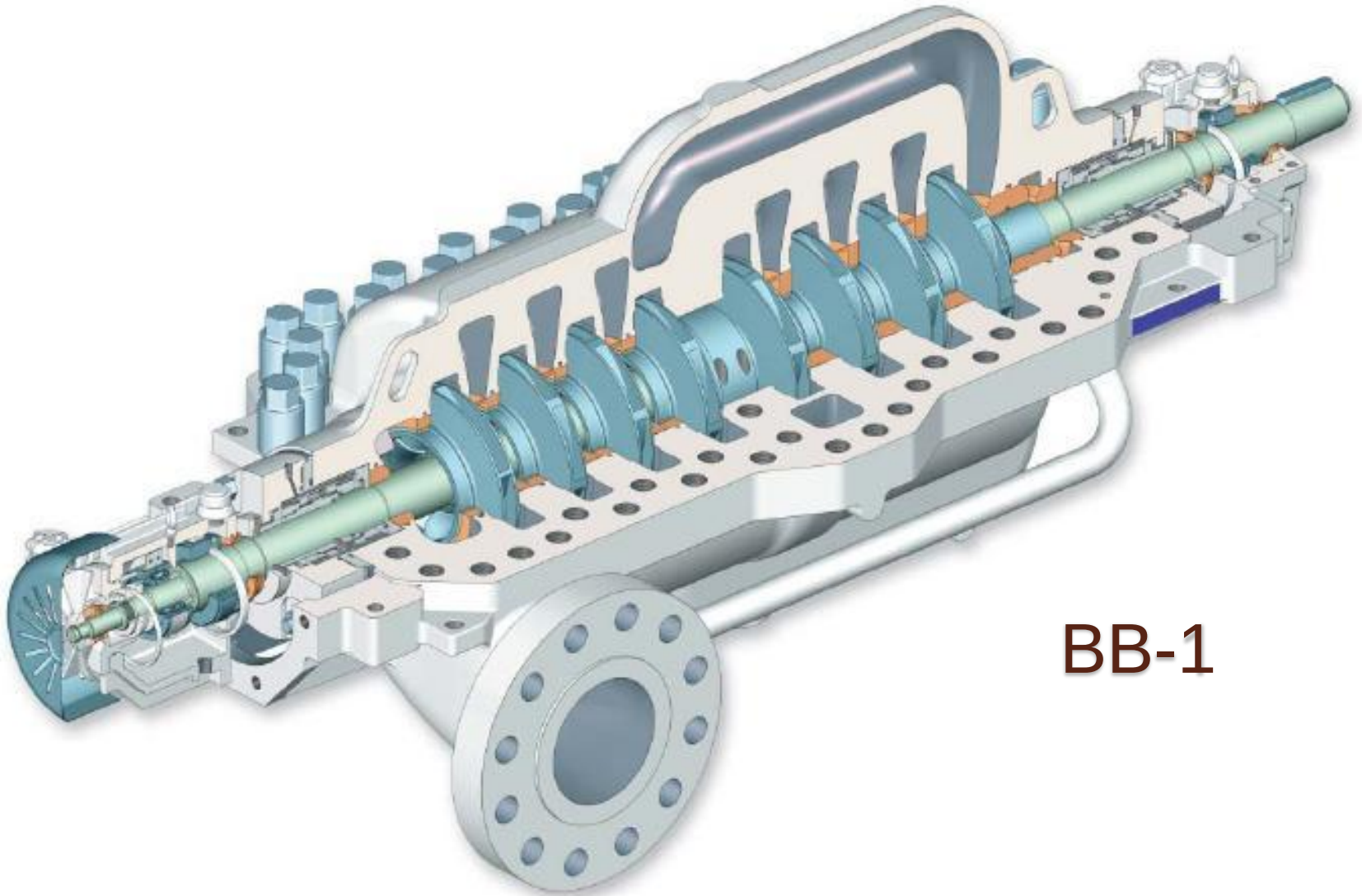
RADIAL RUNNING CLEARANCE SHALL BE USED TO LIMIT INTERNAL LEAKAGE AND, WHERE NECESSARY , BALANCE AXIAL THRUST.

RUNNING CLEARANCE SHALL MEET THE REQUIREMENT :

- CONSIDERATION SHALL BE GIVEN TO PUMPING TEMPERATURE, SUCTION CONDITION, THE LIQUID PROPERTIES, THERMAL EXPANSION AND GALLING CHARACTERISTIC OF THE MATERIALS AND PUMP EFFICIENCY.
- FOR CAST IRON, BRONZE, HARDENED MARTENSITIC STAINLESS STEEL AND MATERIAL WITH SIMILARLY LOW GALLING TENDENCIES, THE MINIMUM CLEARANCE GIVEN IN THE TABLE.
- FOR MATERIALS WITH HIGHER GALLING TENDENCIES AND FOR ALL MATERIALS OPERATING AT TEMPERATURE ABOVE 260°C (500 °F), 125 µm (0.005 Inch) SHALL BE ADDED TO THESE DIAMETRAL CLEARANCE.
- FOR NON-METALLIC WEAR RING MATERIALS WITH VERY LOW OR NO GALLING TENDENCIES CLEARANCES LESS THAN THOSE GIVEN IN TABLE.



WEAR RING CLEARANCE



WEAR RING CLEARANCE

Design Features and Benefits

Bearings

- Heavy duty specification for extended life
- Back to back angular contact thrust and single ball bearing journal bearing

Shaft Seal

- Generously dimensioned seal chambers allow unrestricted seal flushing/cooling
- Cartridge single or double mechanical seals may be fitted

Casing

- Axial split for ease of maintenance
- Branches in the lower half only
- The complete rotor may be inspected/replaced without disconnecting the main pipework

Double Entry Impeller

- Principal axial thrusts are inherently balanced
- Optimum geometry provides high efficiency, low NPSH and quiet running over a wide operating range

Bearing Housings

- Manufactured from carbon steel
- Cooling not required up to 150 °C
- Constant level oiler maintains oil level
- Labyrinth seals prevent ingress of contaminants to the housing
- Pure mist lubrication is available

Optimum Hydraulic Geometry

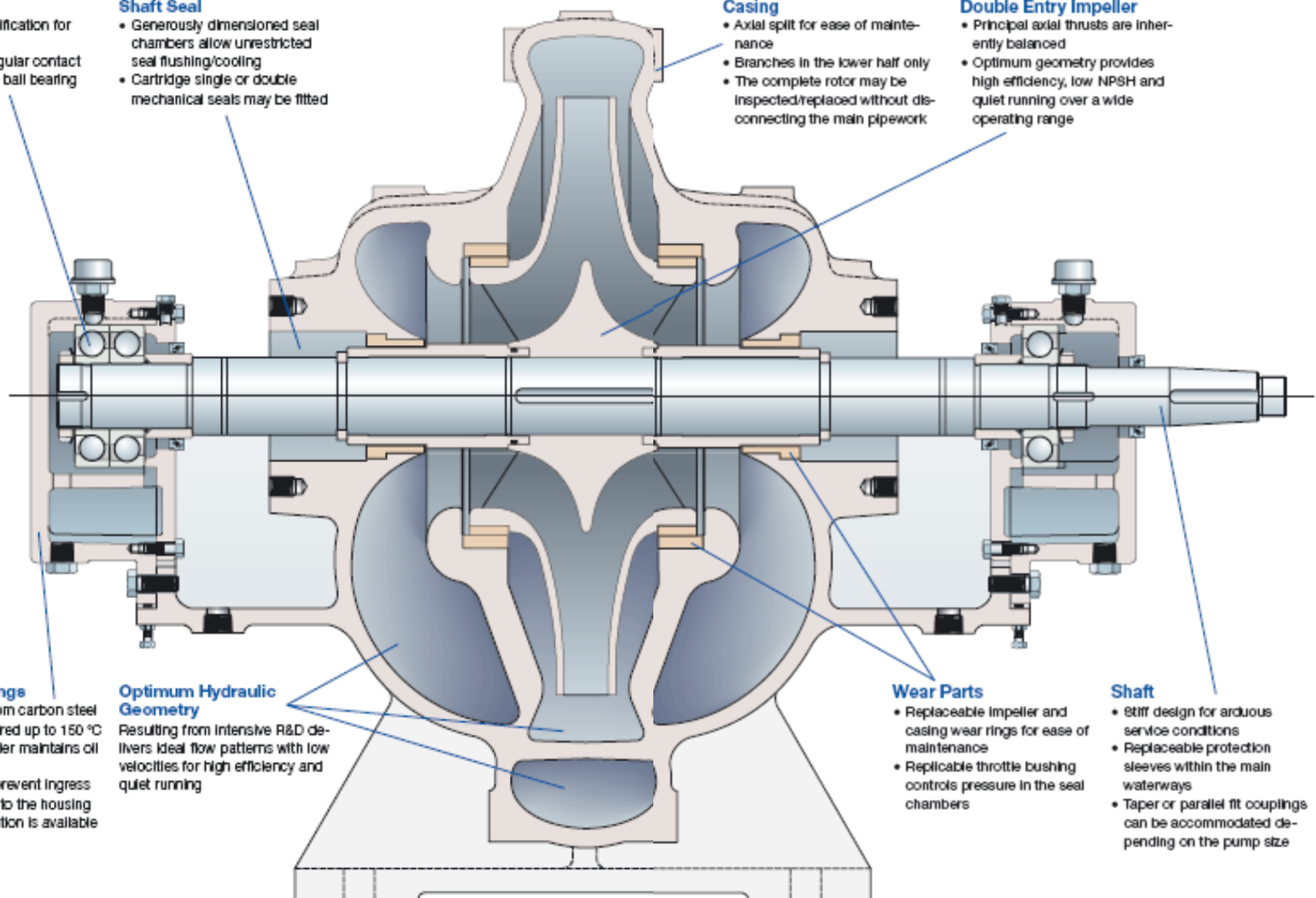
Resulting from intensive R&D delivers ideal flow patterns with low velocities for high efficiency and quiet running

Wear Parts

- Replaceable impeller and casing wear rings for ease of maintenance
- Replicable throttle bushing controls pressure in the seal chambers

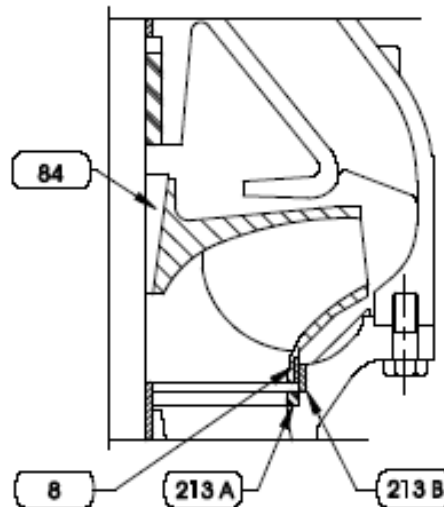
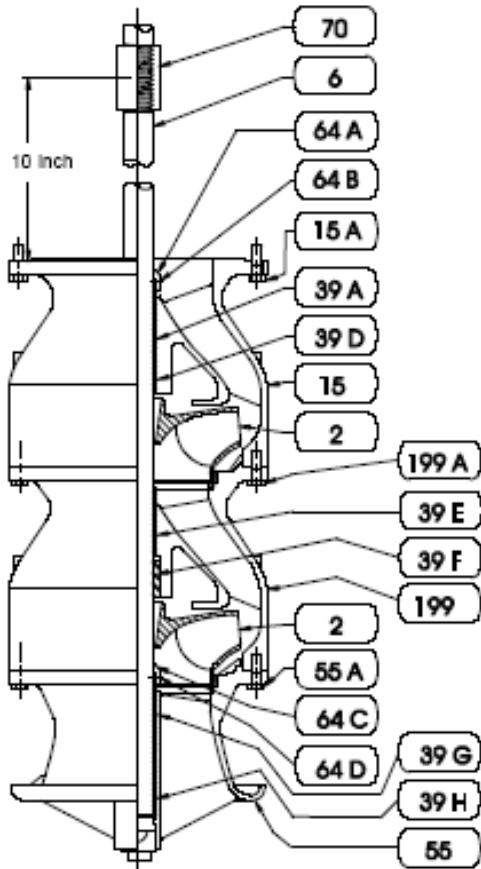
Shaft

- Stiff design for arduous service conditions
- Replaceable protection sleeves within the main waterways
- Taper or parallel fit couplings can be accommodated depending on the pump size



WEAR RING CLEARANCE

VERTICAL TURBINE PUMP SECTIONAL DRAWING



BOWL ASSEMBLY

This is a typical cross sectional drawing and may not include exactly what is supplied.

Item No.	Part Description	Material
2	Impeller	304ss
6	Shaft, Pump	ss 17-4ph
8	Ring, Wear, Impeller (optional)	Bronze
15	Bowl, Discharge, Flanged	Ni-resist
15A	Screw, Hex. Head Cap	S. Steel
39A	Sleeve, Bearing	Bronze
39D	Sleeve, Bearing	Bronze
39E	Sleeve, Bearing	Bronze
39F	Sleeve, Bearing	Bronze
39G	Sleeve, Bearing	Bronze
39H	Sleeve, Bearing	Bronze
55	Bell, Suction	Ni-resist
55A	Screw, Hex. Head Cap	S. Steel
64A	Collar, Protecting	Steel
64B	Screw, Set	None
64C	Collar, Protecting	None
64D	Screw, Set	None
70	Coupling, Shaft	ss 17-4ph
84	Key, Impeller	303ss
199	Bowl, Intermediate	Ni-resist
199A	Screw, Hex. Head Cap	S. Steel
213A	Ring, Wear, Bowl (optional)	None
213B	Ring, Wear, Bowl (optional)	Bronze

WEAR RING & MINIMUM RUNNING CLEARANCE

ANSI/API Standard 610 / ISO 13709, 10th Edition, October 2004

Diameter of rotating members at clearance (mm)	Minimum diametral clearance (mm)	Diameter of rotating members at clearance (inch)	Minimum diametral clearance (inch)
<50	0.25	< 2.00	0.010
50 to 64.99	0.28	2.000 to 2.499	0.011
65 to 79.99	0.30	2.500 to 2.999	0.012
80 to 89.99	0.33	3.000 to 3.499	0.013
90 to 99.99	0.35	3.500 to 3.999	0.014
100 to 114.99	0.38	4.000 to 4.499	0.015
115 to 124.99	0.40	4.500 to 4.999	0.016
125 to 149.99	0.43	5.000 to 5.999	0.017
150 to 174.99	0.45	6.000 to 6.999	0.018
175 to 199.99	0.48	7.000 to 7.999	0.019
200 to 224.99	0.50	8.000 to 8.999	0.020
225 to 249.99	0.53	9.000 to 9.999	0.021
250 to 274.99	0.55	10.000 to 10.999	0.022
275 to 299.99	0.58	11.000 to 11.999	0.023

WEAR RING & MINIMUM RUNNING CLEARANCE

ANSI/API Standard 610 / ISO 13709, 10th Edition, October 2004

Diameter of rotating members at clearance (mm)	Minimum diametral clearance (mm)	Diameter of rotating members at clearance (inch)	Minimum diametral clearance (inch)
300 to 324.99	0.60	12.000 to 12.999	0.024
325 to 349.99	0.63	13.000 to 13.999	0.025
350 to 374.99	0.65	14.000 to 14.999	0.026
375 to 399.99	0.68	15.000 to 15.999	0.027
400 to 424.99	0.70	16.000 to 16.999	0.028
425 to 449.99	0.73	17.000 to 17.999	0.029
450 to 474.99	0.75	18.000 to 18.999	0.030
475 to 499.99	0.78	19.000 to 19.999	0.031
500 to 524.99	0.80	20.000 to 20.999	0.032
525 to 549.99	0.83	21.000 to 21.999	0.033
550 to 574.99	0.85	22.000 to 22.999	0.034
575 to 599.99	0.88	23.000 to 23.999	0.035
600 to 624.99	0.90	24.000 to 24.999	0.036
625 to 649.99	0.95	25.000 to 25.999	0.037

Wear Ring and Bearing Clearances

Bowl size		Standard wear ring clearance		Special wear ring clearance		Bearing clearance		Standard shaft size
mm	inch	mm	inch	mm	inch	mm	inch	inch
100	4"				0.006	0.15	3/4	3/4
150	6"	0.31-0.36	0.012-0.014	0.41 -0.46	0.016-0.018	0.18	0.007	1 1/4
200	8"	0.31	0.012	0.41	0.016	0.2	0.008	1 1/4
255	10"	0.33	0.013	0.41 -0.43	0.016-0.017	0.2	0.008	1 1/2
305	12"	0.33-0.38	0.013-0.015	0.43 -0.46	0.017-0.018	0.2	0.008	1 11/16
355	14"	0.38	0.015	0.43 -0.46	0.017-0.018	0.23	0.009	1 15/16
405	16"	0.38	0.015	0.46 -0.56	0.018-0.022	0.25	0.010	1 1/4
460	18"	0.46	0.018	0.51 -0.56	0.020-0.022	0.28	0.011	2 1/4
510	20"	0.46	0.018	0.56	0.022	0.28	0.011	2 7/16
560	22"	0.46	0.018	0.56 -0.61	0.022-0.024	0.31	0.012	2 11/16
610	24"	0.66	0.026	0.56	0.022	0.31	0.012	2 7/16
635	25"	0.66	0.026	0.76	0.030	0.31	0.012	2 15/16
685	27"	0.66	0.026	0.76	0.030	0.31	0.012	3 1/4
710	28"	0.66	0.026	0.76	0.030	0.31	0.012	3 1/4
760	30"	0.81	0.032	0.91	0.036	0.31	0.012	3 1/4
810	32"	0.81	0.032	0.76	0.030	0.31	0.012	3 7/16
840	33"	0.81	0.032	0.91	0.036	0.31	0.012	3 7/16
915	36"	0.81	0.032	0.91	0.036	0.31	0.012	4
1065	42"	0.86	0.034	0.99	0.039	0.31	0.012	4 1/2
1220	48"	0.86	0.034	0.99	0.039	0.31	0.012	4 1/2
1420	56"	0.91	0.036	1.04	0.041	0.38	0.015	5
1625	64"	0.91	0.036	1.04	0.041	0.38	0.015	5 1/2

SULZER

Centrifugal PUMP PROBLEMS :

PUMP TYPE :

- Vertical Suspended Pump

PROBLEM :

- Pump jammed

POSSIBLE CAUSE :

- Shaft misalignment
- Bearing clearance oversize
- Shaft bend during install



Centrifugal PUMP PROBLEMS :

PUMP TYPE :

- Vertical Suspended Pump

EQUIPMENT :

- Firewater Pump

PROBLEM :

- Insufficient Capacity
- Insufficient Pressure

FACT FINDING :

- Casing too much scalling
- Case wear ring oversize



Centrifugal PUMP PROBLEMS :

PUMP TYPE :

- Vertical Suspended Pump

EQUIPMENT :

- Seawater Lift Pump

PROBLEM :

- Insufficient Capacity
- Insufficient Pressure
- Vibration during running

FACT FINDING :

- Impeller wear ring clearance oversize
- Impeller wear ring corroded



Checkpoints for Initial Start-up of a Vertical Pump

Initial start-up means starting a pump for very first time after it is installed, connected and wired at its location. There are some steps required to ensure correct rotation of the pump impellers BEFORE the pump is coupled with the motor and adjusted for lift. The following describes a typical procedure for hollow-shaft or solid-shaft electric motors after the pump is set and grouted in place, motor drive installed and wired but not yet connected with pump.

1. Check the hold-down bolts on the motor, baseplate and discharge flange. Check all lubrication systems. Adjust the seal or packing box gland; gland nuts should be finger-tight at start-up.
2. Make sure the pump has sufficient fluid in the sump or supply lines. Make sure all suction valves are fully open on barrel-type pumps. Do not run any pump without fluid.
3. Verify correct wiring and rotation of motor shaft:
 - a. For vertical hollow-shaft motor driven pumps, ensure the top shaft nut (adjusting nut) and gib key are removed and the motor is not coupled to the pump. The pump shaft can turn freely from the pump motor.
 - b. For vertical solid-shaft motors, ensure the pump to motor pump coupling is not connected and the pump shaft can turn freely from the pump motor.
 - c. Ensure all tools, equipment and personnel are clear and away from all rotating components.
- d. Verify rotation and correct wiring of motor by flick starting the motor. It is not necessary or desirable to energize the motor for longer than $\frac{1}{4}$ of a second or permit the motor to run to full speed.
- e. As the motor shaft begins to slow from the very brief connection with its power source, observe the motor shaft rotation. The rotation should match the direction arrow of the pump nameplate. If this does not match, a qualified electrician will need to change the leads on the motor connection and the process above repeated.
4. When rotation is verified correct, reconnect the pump and motor, and adjust the impellers (see instruction on adjustment).
5. Start the pump and check amperage on the motor. Run the pump long enough to determine that no unusual noise or vibration is present and that the mechanical seal or packing box is functioning properly.



Checkpoints for Initial Start-up of a Vertical Pump

Vertical Pump Impeller Adjustment

Improper impeller adjustment will cause unnecessary wear, reduction of capacity and pressure, and motor overload problems. Impellers set too low will drag on the pump bowls and wear both the impeller skirts and bowl castings, eventually destroying them. Impellers set too high can drag on the upper bowl case. Both situations cause high horsepower loading that frequently trips the motor overload relays. If drag is severe, shafts can snap before overloads trip.

Improper impeller settings can also create vibration and cause premature bearing wear and failure. The following instructions must be followed to avoid these problems.

Vertical pumps are provided with either a hollow-shaft (VHS) or a solid-shaft (VSS) drive. On the VHS drive, the impeller adjusting nut is situated above the motor drive coupling. On the VSS drive, the adjusting nut is a component of the flanged motor/pump coupling.

JD Pump Adjustment

The JD “Dynaline” range was designed as a high pressure can pump. Because of this the settings are very short and operators should use the dimensions in the separate table.

Always refer to settings on the pump nameplate and/or installation, operation and maintenance manual which are specific to the application and serial number.

Mechanical seals must be adjusted per the manufacturer's instructions after impeller adjustments have been completed.

Closed impellers				
Bowl size		Impeller lift		
mm	inch	mm	inch	
Up to 230	Up to 9	3	0.125	
255 - 355	10 - 14	6	0.250	
380 - 610	15 - 24	9.5	0.375	
635 - 915	25 - 36	12.5	0.500	
Over 1015	Over 40	16	0.625	
Mixed flow and semi-open impellers				
Bowl size		Impeller lift		
mm	inch	mm	inch	
150 - 330	6 - 13	0.38 - 0.51	0.015 - 0.020	
355 - 610	14 - 24	0.51 - 0.76	0.020 - 0.030	
635 - 915	25 - 36	0.76 - 1.30	0.030 - 0.050	
Over 1016	Over 40	1.30 - 1.90	0.050 - 0.075	
JD Dynaline pumps				
Bowl size		Impeller lift		
mm	inch	mm	inch	
200 - 230	8 - 9	0.40	0.016	
280 - 330	11 - 13	4.75	0.187	
400 - 760	16 - 30	6.35	0.250	
915 - 1015	36 - 40	7.87	0.310	
1220	48	9.40	0.370	

Trouble Indicators and Possible Causes of a Vertical Pump

Insufficient Pressure

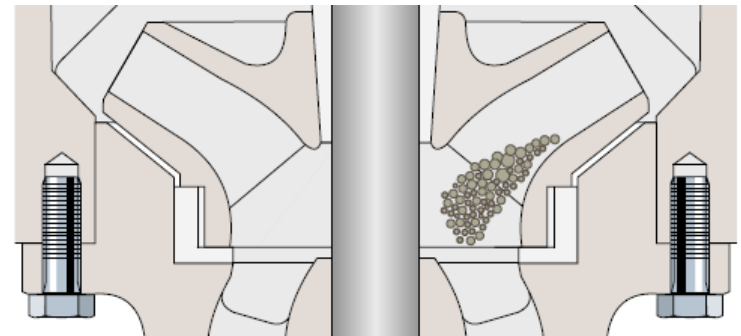
1. Speed too slow (check voltage)
2. Improper impeller adjustment
3. Impeller loose
4. Impeller plugged
5. Wear rings worn
6. Entrained air in pump
7. Leaking column joints or bowl castings
8. Wrong rotation

No Liquid Delivered

1. Pump suction broken (water level below bell inlet)
2. Suction valve closed
3. Impeller plugged
4. Strainer clogged
5. Wrong rotation
6. Shaft broken or unscrewed
7. Impeller loose

Vibration

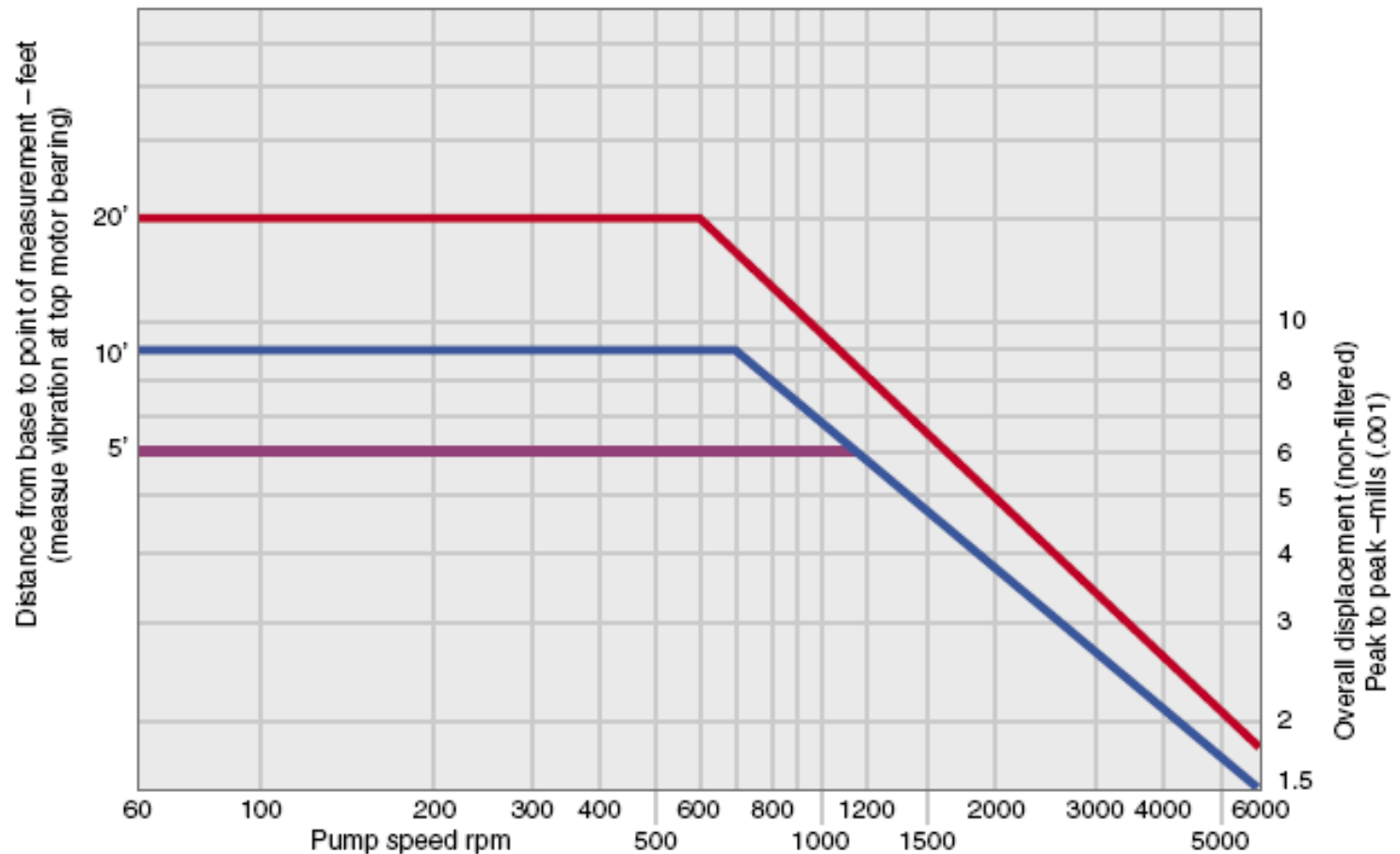
1. Motor imbalance - electrical
2. Motor bearings not properly seated
3. Motor drive coupling out of balance
4. Misalignment of pump, castings, discharge head, column or bowls
5. Discharge head misaligned by improper mounting or pipe strain
6. Bent shafting
7. Worn pump bearings
8. Clogged impeller or foreign material in pump
9. Improper impeller adjustment
10. Vortex problems in sump
11. Resonance - system frequency at or near pump speed



Vibration

Vertical Turbine Pumps

Acceptable field vibration limits for vertical pumps – clear liquids



Trouble Indicators and Possible Causes of a Vertical Pump

Using Too Much Power

1. Speed too high
2. Improper impeller adjustment
3. Improper impeller trim
4. Pump out of alignment or shaft bent
5. Lubricating oil too heavy
6. Pumping sand, silt or foreign material

Abnormal Noise

1. Motor noise
2. Pump bearings running dry
3. Broken column bearing retainers
4. Broken shaft or shaft enclosing tube
5. Impellers dragging on bowl case
6. Cavitation due to low submergence or operation beyond maximum capacity rating
7. Foreign material in pump

Insufficient Capacity

1. Speed too slow
2. Improper impeller trim
3. Impeller loose
4. Impeller or bowl partially plugged
5. Leaking joints
6. Strainer partially clogged
7. Suction valve throttled
8. Low water level
9. Wrong rotation

Wear Analysis: Vertical Turbine Pumps

Trouble Source	Probable Cause	Remedy
Uneven wear on bearings, uniform wear on shafts.	Pump non-rotating parts misaligned.	Check mounting and discharge pipe connection and check for dirt between column joints. Correct misalignment, replace bearings, and repair or replace shaft.
Uniform wear on bearings and shafts.	Abrasive action.	Replace parts. Consider changing materials or means of lubrication.
Uniform wear on bearings, uneven wear on shafts.	Shaft run-out caused by bent shafts, shafts not butted in couplings, dirt or grease between shafts.	1. Straighten shaft or replace, clean and assemble correctly. 2. Face parallel and concentric.
Wear on impeller skirts and/or bowl seal ring.	1. Abrasive action or excess bearing wear allowing impeller skirts to function as bearing journal. 2. Impellers set too high.	1. Install new bearings and wear rings. Upgrade material if abrasion occurring. 2. Re-ring and adjust impellers correctly.
Impeller seal ring end wear.	Improper impeller adjustment. Impeller running on bottom.	Install "L"-shaped bowl wear rings. Adjust impeller setting per manufacturer's recommendations.
Wear on bowl vanes.	Abrasive action.	Coat bowls, upgrade material, or rubber line.
Wear on suction bell vanes.	Cavitation due to recirculation.	Correct condition or upgrade material to extend life.

Wear Analysis: Vertical Turbine Pumps

Bearings Failures

Bearing wear.	Abrasive action.	Convert to fresh water flushing on bearings; or use pressure-grease or oil lubrication; or use bearings made of harder material.
Bearing seized or galling on shaft.	Running dry without lubrication.	Check lubrication, look for plugged suction or evidence of flashing.
Bearing failure or bearing seized.	High temperature failure.	Check pump manufacturer for bearing temperature limits. Generally: Bronze - 175° F / 80° C maximum in water Synthetics - 125° F / 50° C Carbon - 300° F / 150° C Rubber - 125° F / 50° C
Excessive shaft wear.	Rubber bearings will swell in hydrocarbon, H ₂ S and high temperature.	Change bearing material.

Impeller Wear

Exit vanes and shrouds.	Abrasive action.	Replace impeller if excessive. Consider coating or upgrading material.
Pitting on entrance vanes of impeller.	Cavitation.	Correct condition or upgrade material to extend life. See section on Cavitation.
Pitting on impellers and bowl casing.	Corrosion, erosion, or recirculation.	Investigate cost of different materials versus frequency of replacements. See section on Corrosion.

Wear Analysis: Vertical Turbine Pumps

Shaft and Couplings

Bent shaft.	Mishandling in transit or assembly.	Check straightness. Correct to 0.005 in./ft (0.13 mm/300 mm) total runout or replace.
Shaft coupling unscrewed.	Pump started in reverse rotation.	Shafts may be bent. Check shafts and couplings. Correct rotation.
Shaft coupling elongated (necked down).	1. Motor started while pump is running in reverse. 2. Corrosion. 3. Pipe wrench fatigue on reused couplings. 4. Power being applied to shafts that are not butted in coupling.	1. Look for faulty check valve. Could also be momentary power failure or improper starting timers. 2. Replace couplings. 3. Replace couplings. 4. Check for galling on shaft ends.
Broken shaft.	1. Can be caused by same reasons listed for coupling elongation. 2. Can also be caused by bearings seized due to lack of lubrication. 3. Foreign material locking impellers or galling wear rings. 4. Metal fatigue due to vibration. 5. Improper impeller adjustment or continuous upthrust conditions, causing impeller to drag.	1. Look for faulty check valve, momentary power failure or improper starting timers. 2. Same as above for bearing seizure. 3. Add strainers or screens. 4. Check alignment of pump components to eliminate vibration. 5. See sections on Impeller Adjustment and Upthrusting.
Impeller loose on shaft (rarely occurs).	1. Repeated shock load by surge in discharge line (could knock top impeller loose). 2. Foreign material jamming impeller. 3. Differential expansion due to temperature. 4. Improper parts machining and assembly. 5. Torsional loading on submersible pumps.	1. Refit impeller. 2. Usually will break shaft or trip overloads before impeller comes loose. 3. Change to material with the same expansion factor. 4. Repair and refit. 5. Overcome by adding keyway to collet mounting.

Mechanical Seal Maintenance and Troubleshooting

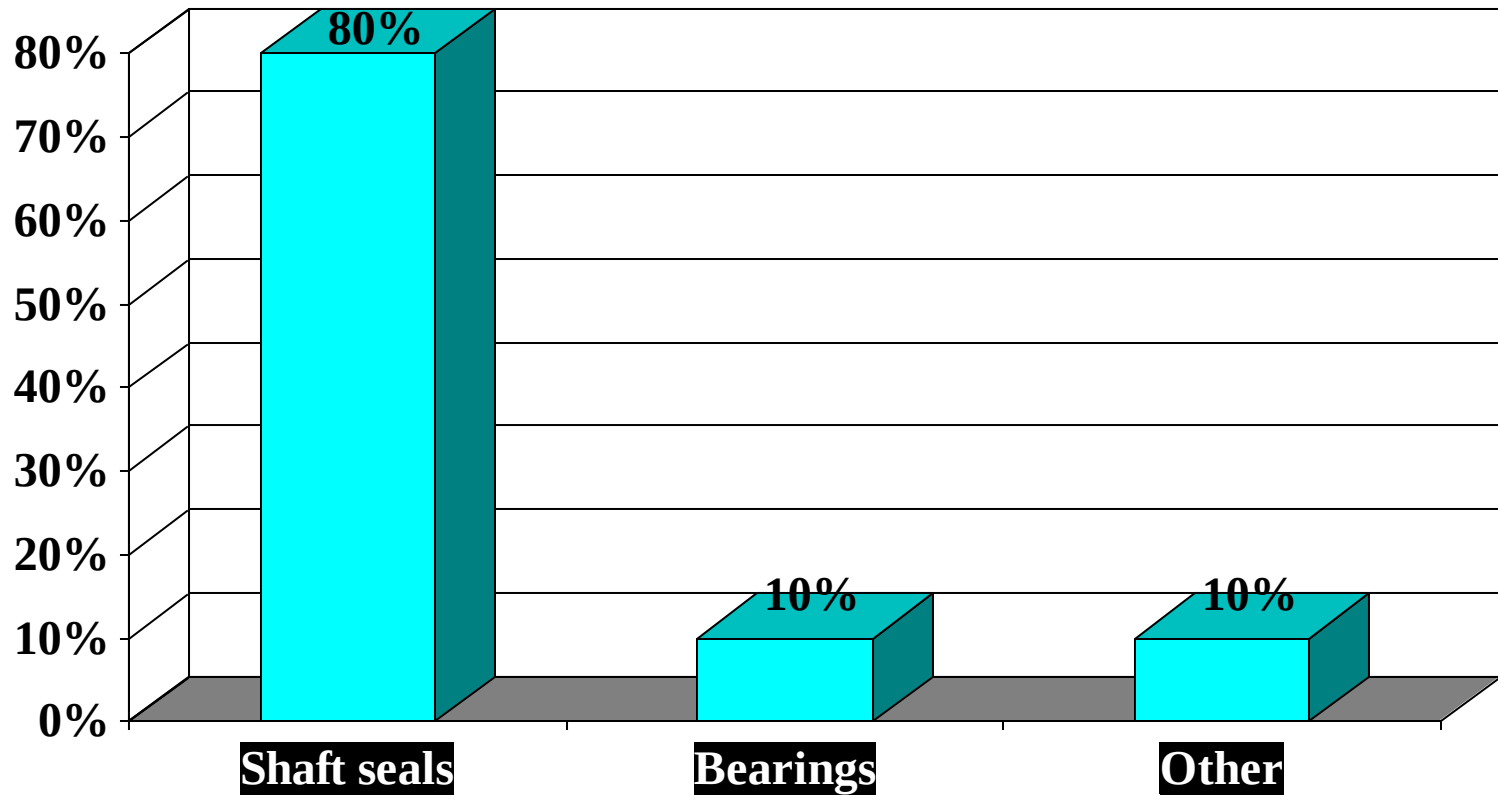
Symptom	Probable Cause	Remedy
Seal leaks steadily.	Faces not flat.	Check for incorrect dimensions.
	Blistered carbon graphite seal faces.	1. Check for gland plate distortion due to over-torquing of gland bolts. 2. Improve cooling flush line, if overheated. 3. Check gland gasket for proper compression. 4. Clean out any foreign particles between seal faces. Re-lap faces, if necessary. 5. Check for cracks and chips at seal faces during installation. 6. Replace primary and mating rings, if damaged.
	Secondary seals nicked or scratched during installation.	Replace secondary seals.
	Worn out or damaged 'O'-rings.	Check for proper seals with seal manufacturer.
	Compression set of secondary seals (hard and brittle).	Check for proper lead-in on chamfers, burrs, etc.
	Chemical attack (soft and sticky).	Check seal manufacturer for alternative materials.
	Spring failure.	Replace parts.
	Erosion damage of hardware and/or corrosion of drive mechanism.	Check seal manufacturer for alternative materials.
Carbon dust accumulating on outside of gland ring.	Inadequate amount of liquid to lubricate seal faces.	1. Flush line may be needed (if not in use). 2. Enlarge flush line and/or orifices in gland plate.
	Liquid film evaporating between seal faces.	Check for proper seal design with seal manufacturer if pressure in mechanical seal box is excessively high.

Mechanical Seal Maintenance and Troubleshooting

Symptom	Probable Cause	Remedy
Seal squeals during operation.	Inadequate amount of liquid to lubricate seal faces.	1. Flush line may be needed (if not in use). 2. Enlarge flush line and/or orifices in gland plate.
Seal leaks intermittently.	See causes listed under "Seal leaks steadily".	1. Refer to list under "Seal leaks steadily". 2. Check for squareness of mechanical seal box to shaft. 3. Align shaft, impeller and bearing to prevent shaft vibration and/or distortion of gland plate and/or mating ring.
Short seal life.	Abrasive particles in fluid.	1. Prevent abrasives from accumulating at seal faces. 2. Flush line may be needed (if not in use). Use abrasive separator or filter.
	Seal running too hot.	1. Increase cooling of seal faces (for example, by increasing flush line flow). 2. Check for obstructed flow in cooling lines.
	Equipment mechanically misaligned.	Align properly. Check for rubbing of seal on shaft.

Mechanical Seal Maintenance and Troubleshooting

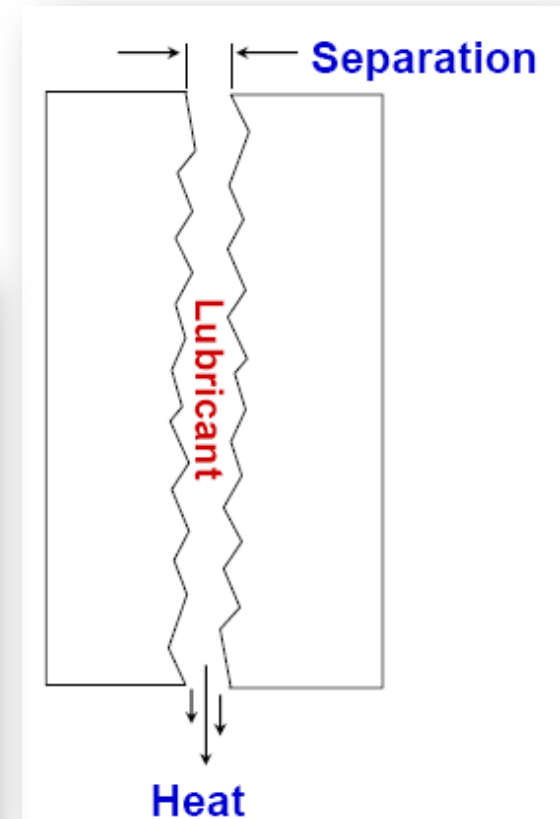
CAUSED OF FAILURES OF ANSI PUMPS USED IN THE CHEMICAL PROCESS INDUSTRY

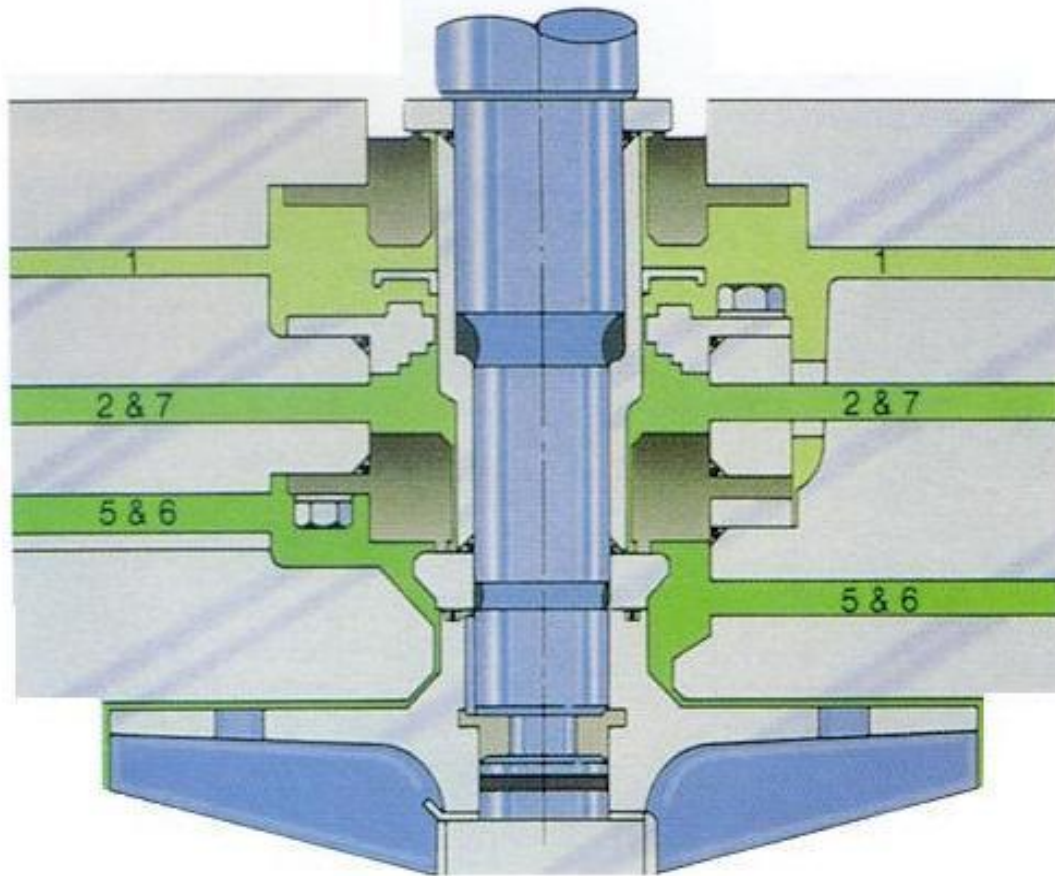




▪ **Lubrication Purpose:**

- Separate surfaces
- Prevent contact of high surface points
- Reduce friction/heat generation





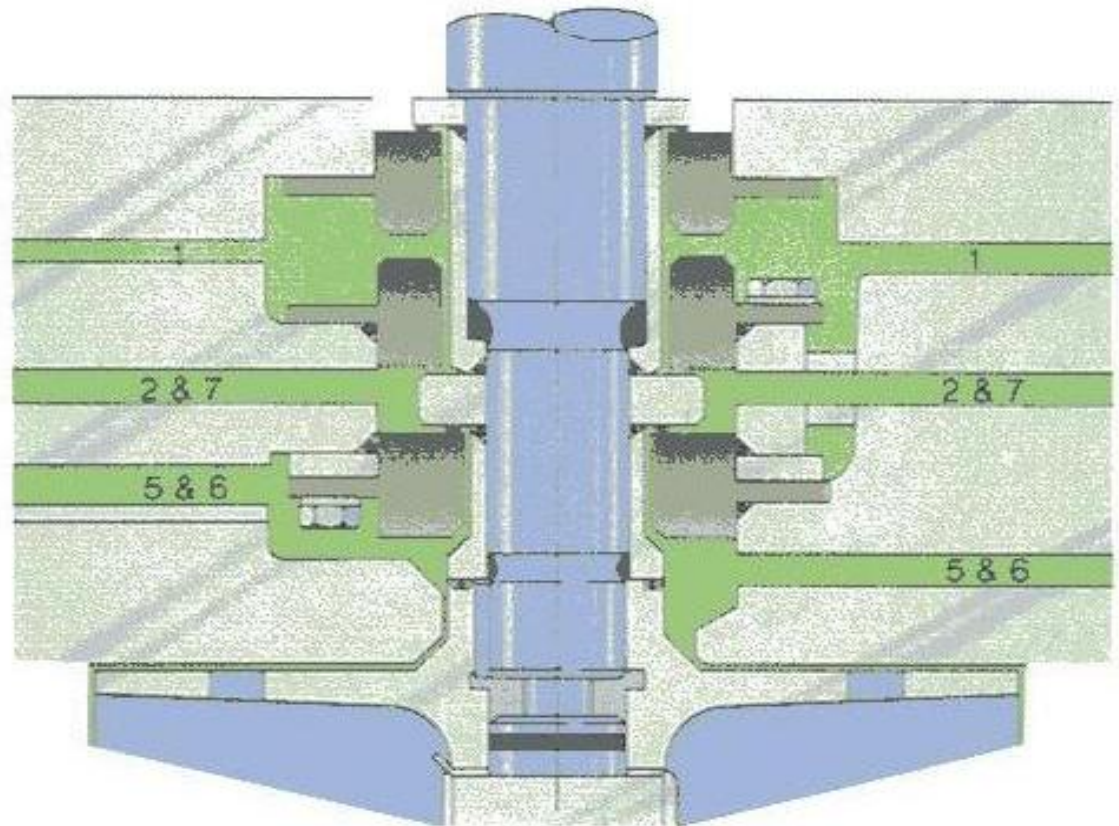
SINGLE SEAL

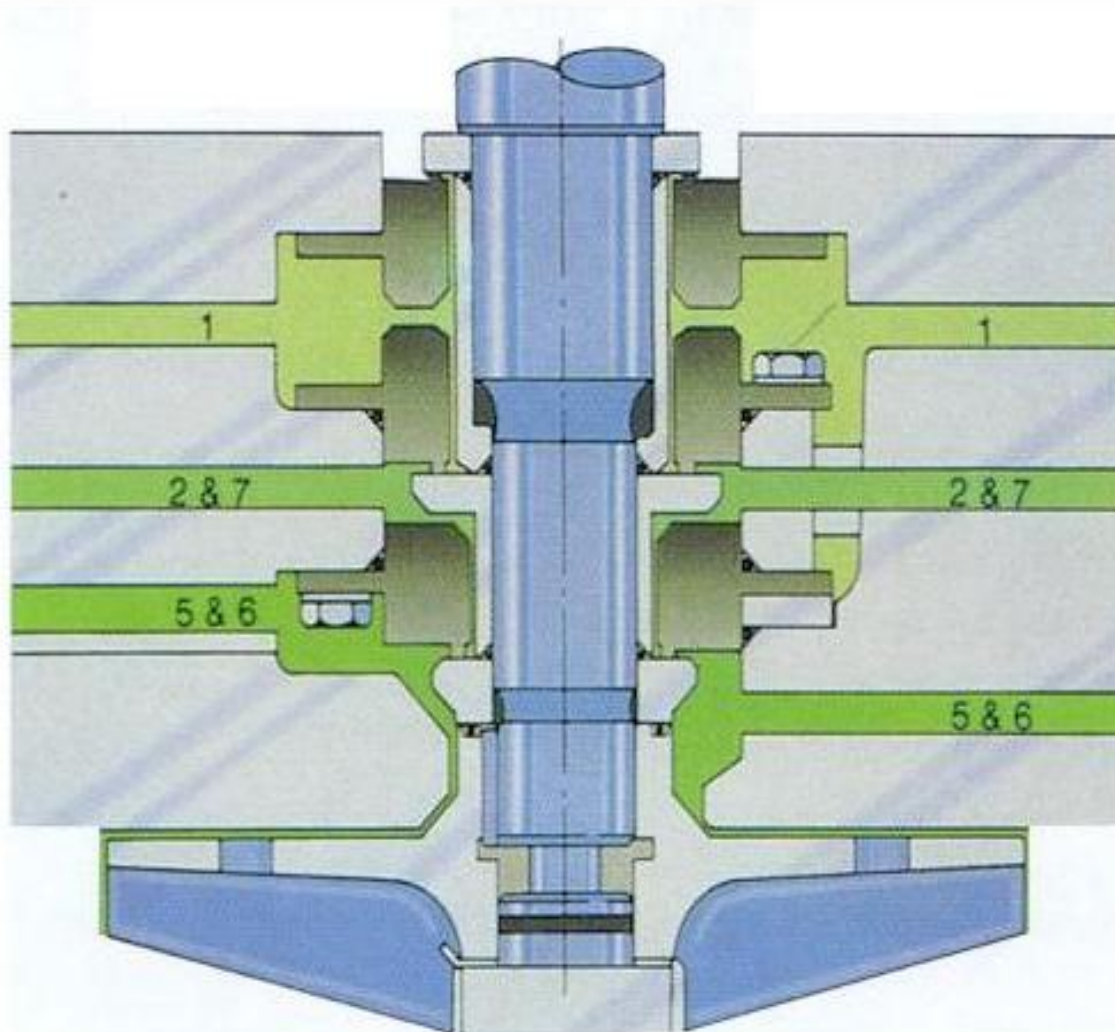
Standard seal used in most applications - for non-abrasive or non-hazardous liquids. Bellows seals are also available for higher temperatures and abrasive liquids.



DOUBLE SEAL

Recommended when the pumped liquid contains abrasives, leakage could be hazardous, or when the pump is likely to run dry.





TANDEM SEAL

Used to accommodate quenching, automatic shutdown systems, and high pressure services. With no requirement for a buffer liquid, a film-riding gas seal may be placed in the upper position, thereby providing a secondary seal backup in event of main seal failure.

Thrust in Vertical Turbine Pumps

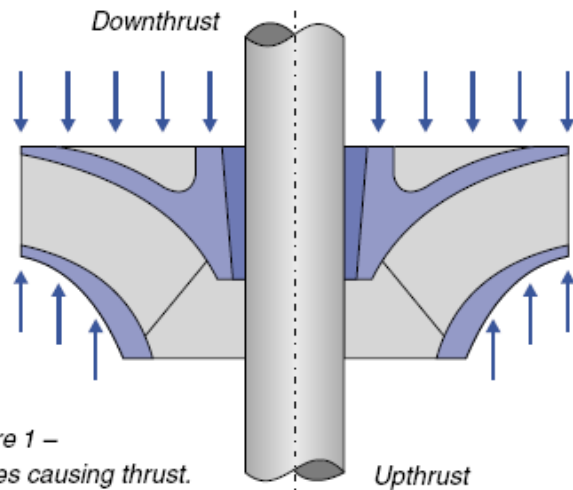
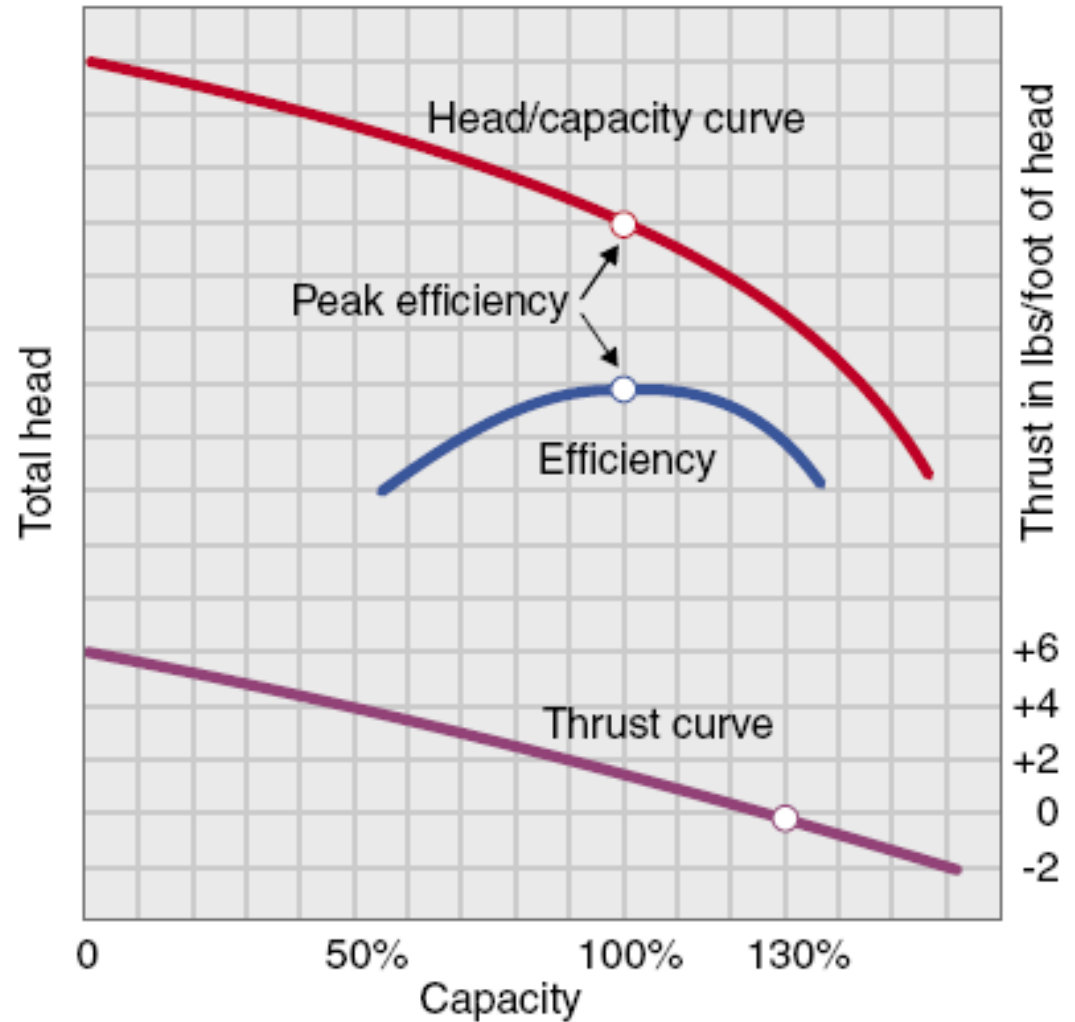


Figure 1 –
Forces causing thrust.

Figure 2 – Typical thrust curve.



Cavitation and Vortexing

Cavitation

Cavitation occurs when the absolute pressure of a moving liquid is reduced to a value equal to (or below) the vapor pressure of the liquid. Small vacuum pockets or bubbles form, then collapse in the area where pressure increases in the impeller. The collapse of these vapor pockets is so rapid that it makes a rumbling or cracking noise - like rocks passing through the pump. The forces in the collapse are generally high enough to cause minute pockets of fatigue on metal surfaces adjacent to bubbles. This action may be progressive and under severe conditions can cause serious pitting damage on the metal subject to cavitation attack.

Cavitation takes place along the impeller vane tips and vane surfaces, as shown in the cross-section. Cavitation can cause the following problems:

1. Reduced pump capacity
2. Erratic power consumption
3. Noisy operation
4. Damage to impeller
5. Pitted suction inlet vanes and impaired casting strength

Note: The same type of damage can result from recirculation caused by operating the pump away from the best efficiency point (BEP).

How to Prevent Cavitation in Existing Installations

Cavitation can be avoided by providing sufficient net positive suction head (NPSH) for the pump. However, this may be an expensive correction in the field. An alternate solution is to reduce the NPSH requirement of the pump by one of the following methods:

1. Evaluate system head conditions, NPSH available, and, if possible, reduce pump capacity.
2. Change pump impellers to obtain low NPSH design.
3. Replace the pump bowl assembly with a different model capable of operating with the system NPSH available.

Contact a Sulzer Pumps Service Center for assistance in evaluating your system NPSH and recommended solutions. Frequently, Sulzer Pumps low NPSH impeller assemblies can be furnished to fit your existing vertical pumps.

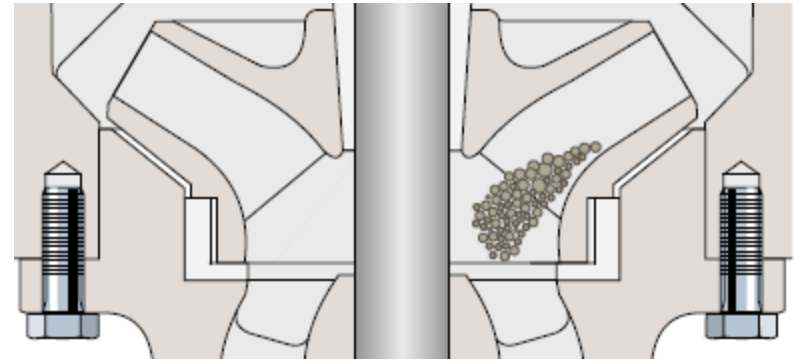
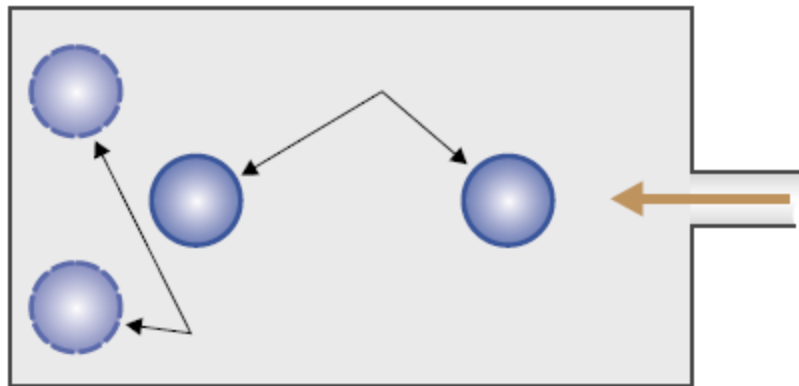
Cavitation and Vortexing

Vortexing

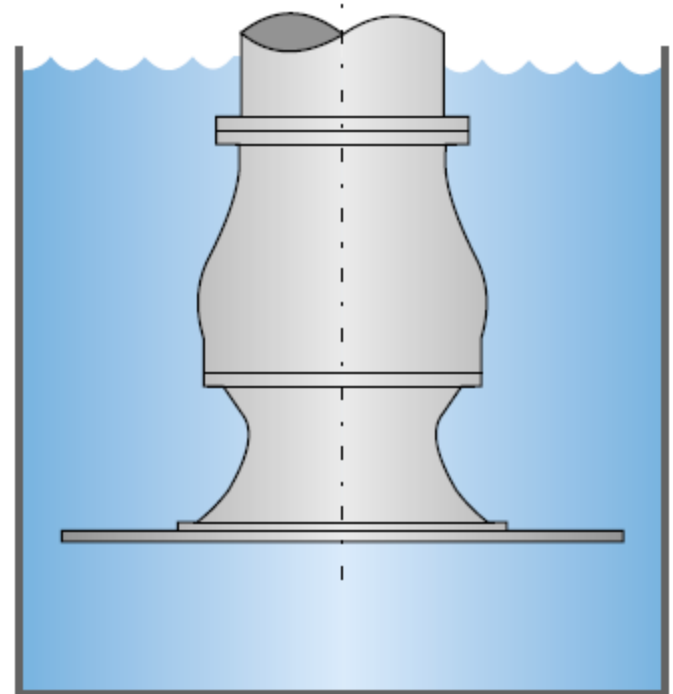
"Something is wrong with the pump! It's sucking in slugs of air". This remark is frequently made when vortices form in flow patterns, causing loud rumbling noises.

A vortex is a whirlpool caused by a combination of factors such as sump design, inlet velocity, direction and flow, submergence, and the position of the bowl assembly in the sump. Air entering the pump through these vortices causes noise and vibration, but not cavitation. Various methods can be used to prevent vortices. These include using suction umbrellas, lowering the inlet velocities in the sump, increasing submergence and relocating pumps.

Relocate pumps at back wall, as indicated by dashed lines.



Suction umbrella added.



Vertical Pump Shafting

Proper selection of shaft materials, shaft finish under bearings, machining and straightening are vital functions of vertical pump manufacturing.

Vertical pump shafting materials are carefully selected for physical properties and micro-finish to operate under sleeve bearings.

Shaft threads must be machined parallel and concentric, and shaft ends must be machined and faced perfectly square. The shaft end centers must also be properly machined to remove any raised area that would prevent proper face-to-face contact between mating shafts.

Shafts must be straightened to 0.13 mm per 305 mm or 0.0005 in/ft in total runout. Example: A ten-foot shaft cannot exceed 0.005 in total runout.

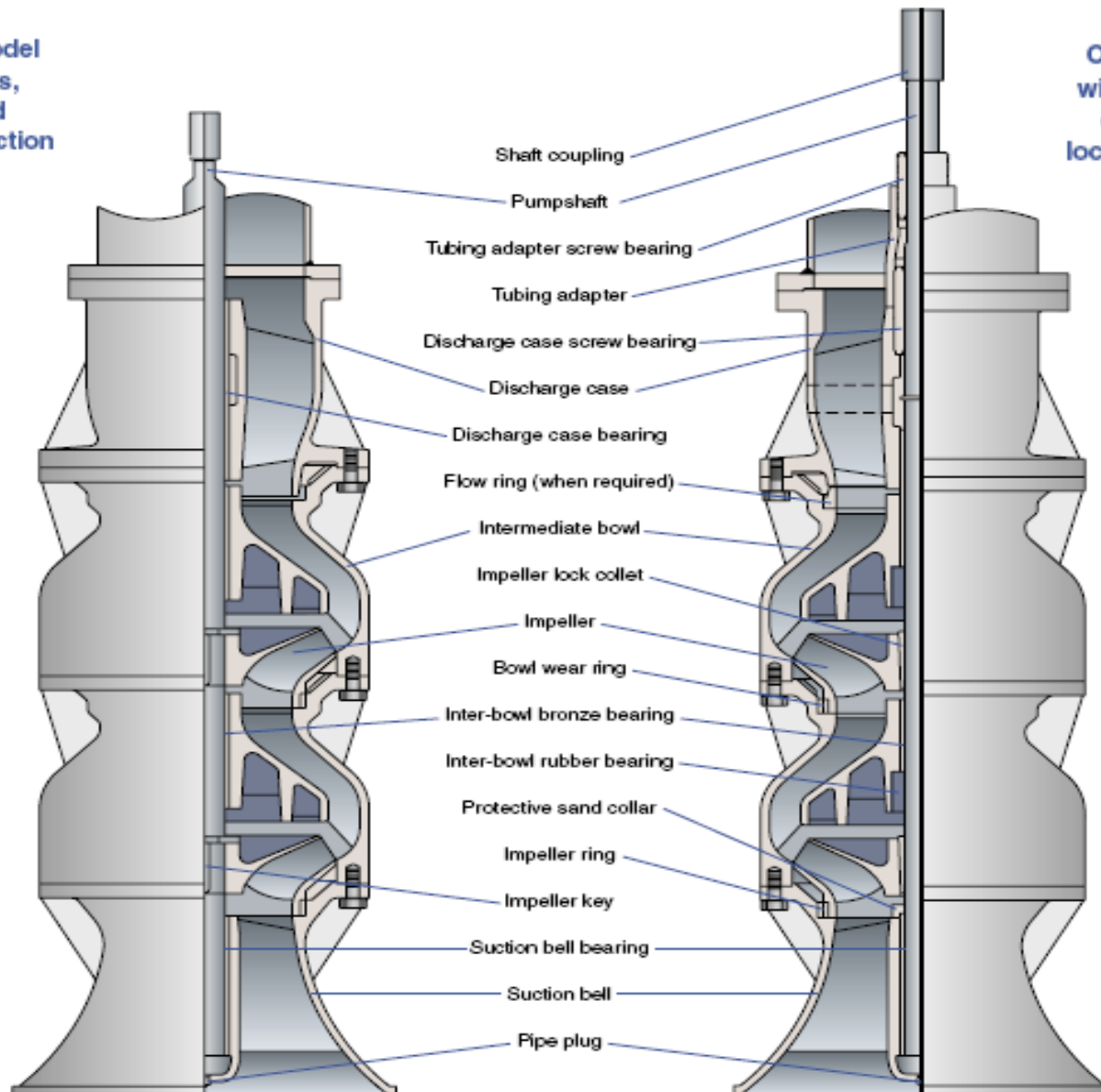
Finally, careful handling of all shafting prior to and during assembly and installation is necessary to avoid bent shafting which will cause premature pump failure.



Typical Vertical Turbine Pump Bowl Assemblies

Product lubricated model
with closed impellers,
single bearings and
keyed impeller construction

Oil lubricated model
with closed impellers,
dual bearings and
lock collet construction





EXTRACTION PROBLEMS

HYDRAULIC SYSTEMMECHANICAL
SYSTEM

Table 4-6 (cont.)

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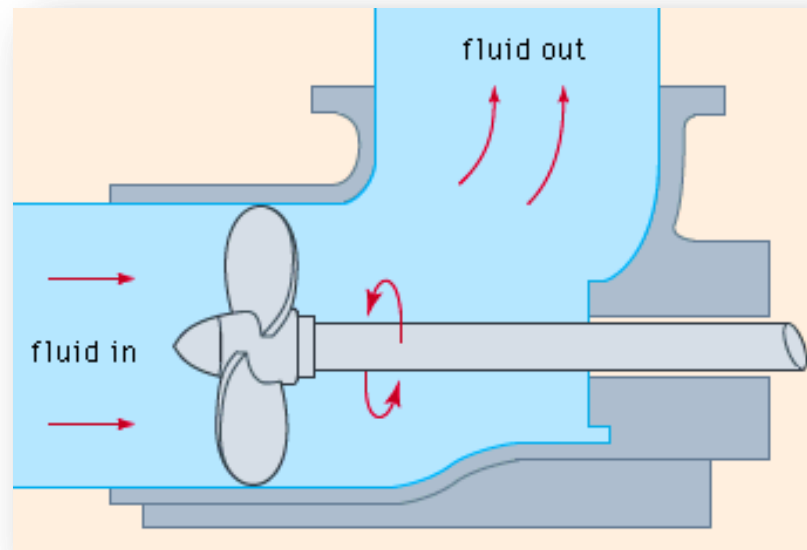
MECHANICAL SYSTEM

Table 4-6 (cont.)

Symptoms											Symptoms														
D					Insufficient Disch. Pressure					Short Bearing Life					E										
C					Intermittent Operation					Short Mech. Seal Life					F										
B					Insufficient Capacity					Vibration & Noise					G										
A					No Liquid Delivery					Power Demand Excessive					H										
Possible Causes											#	A	B	C	D	E	F	G	H	#	Possible Remedies				
MECHANICAL SYSTEM	Mechanical Defects Worn, Rusted, Defective Bearings										25						5	8	10	25	* Inspect Parts For Defects - Repair Or Replace. Use Bearing Failure Analysis Guide * Check Lubrication Procedures				
	Unbalance - Driver										26					5	7	9		26	* Run Driver Disconnected From Pump Unit - Perform Vibration Analysis				
	Unbalance - Pump										27					4	6	3		27	* Investigate Natural Frequency				
	Motor Troubles										28					6	8	10	11	28	* Consult Motor Troubleshooting Guide				
Possible Causes											#	A	B	C	D	E	F	G	H	#	Possible Remedies				

MECHANICAL SYSTEM

INTRODUCTION TO FLUID MECHANIC



PRESENTED BY :
ARIF BACHRUDDIN ADJI, ST
OPERATION DIRECTOR OF
PT. DURAQUIPT CEMERLANG

1. INTRODUCTION

- CONCEPT OF FLUID
- VOLUME FLOW, MASS FLOW & CONTINUITY EQUATION

CONCEPT OF FLUID

A FLUID IS A SUBSTANCE IN WHICH THE CONSTITUENT MOLECULES ARE FREE TO MOVE RELATIVE TO EACH OTHER.

CONVERSELY, IN A SOLID, THE RELATIVE POSITION OF MOLECULES REMAIN ESSENTIALLY FIXED UNDER NON-DESTRUCTIVE CONDITION OF TEMPERATURE AND PRESSURE. WHILE THESE DEFINITIONS CLASSIFY MATTER INTO FLUIDS AND SOLIDS, THE FLUID SUB-DIVIDE FURTHER INTO LIQUID AND GASES.

MOLECULES OF ANY SUBSTANCE EXHIBIT AT LEAST TWO TYPES OF FORCES; AN ATTRACTIVE FORCE THAT DIMINISHES WITH THE SQUARE OF THE DISTANCE BETWEEN MOLECULES, AND A FORCE OF REPULSION THAT BECOMES STRONG WHEN MOLECULES COME VERY CLOSE TOGETHER.

IN SOLIDS, THE FORCE OF ATTRACTION IS SO DOMINANT THAT THE MOLECULES REMAIN ESSENTIALLY FIXED IN POSITION WHILE THE RESISTING FORCE OF REPULSION PREVENTS THEM FROM COLLAPSING INTO EACH OTHER. HOWEVER, IF HEAT IS SUPPLIED TO THE SOLID, THE ENERGY IS ABSORBED INTERNALLY CAUSING THE MOLECULES TO VIBRATE WITH INCREASING AMPLITUDE. IF THAT VIBRATION BECOMES SUFFICIENTLY VIOLENT, THEN THE BONDS OF ATTACHMENT WILL BE BROKEN.

MOLECULES WILL THEN BE FREE TO MOVE IN RELATION TO EACH OTHER – THE SOLID MELTS TO BECOME A LIQUID.

VOLUME FLOW, MASS FLOW & CONTINUITY EQUATION

MOST MEASUREMENT OF FLUID FLOW IN PIPING SYSTEM ARE BASED ON THE VOLUME OF FLUID (M³) THAT PASSES THROUGH A GIVEN CROSS SECTION OF PIPE OR FLUID WAY IN UNIT TIME (1 SECOND). THE UNITS OF VOLUME FLOW, Q , ARE, THEREFORE, M³/S. HOWEVER, FOR ACCURATE ANALYSES WHEN DENSITY VARIATIONS ARE TO BE TAKEN INTO ACCOUNT, IT IS PREFERABLE TO WORK IN TERMS OF MASS FLOW – THAT IS, THE MASS OF AIR (Kg) PASSING THROUGH THE CROSS SECTION IN 1 SECOND. THE UNITS OF MASS FLOW, M , ARE THEN Kg/S

$$\rho = \frac{\text{mass}}{\text{volume}} \quad \frac{\text{kg}}{\text{m}^3}$$

$$\rho = \frac{\text{mass flow}}{\text{volume flow}} = \frac{M}{Q} \quad \frac{\text{kg}}{\text{s}} \frac{\text{s}}{\text{m}^3}$$

$$M = Q \rho \quad \text{kg/s}$$

IN ANY CONTINUOUS PIPE OR FLUID WAY, THE MASS FLOW PASSING THROUGH ALL CROSS SECTIONS ALONG ITS LENGTH ARE EQUAL, PROVIDED THAT THE SYSTEM IS AT STEADY STATE AND THERE ARE NO INFLOWS OR OUTFLOWS OF FLUID BETWEEN THE TWO ENDS. IF THESE CONDITIONS ARE MET THEN,

$$M = Q \rho = \text{constant} \quad \text{kg/s}$$

VOLUME FLOW, MASS FLOW & CONTINUITY EQUATION

THIS IS THE SIMPLEST FORM OF THE CONTINUITY EQUATION. A COMMON METHOD OF MEASURING VOLUME FLOW IS TO DETERMINE THE MEAN VELOCITY OF AIR, u , OVER A GIVEN CROSS SECTION, THEN MULTIPLY BY THE AREA OF THAT CROSS-SECTION, A .

$$Q = u A \quad \frac{\text{m}}{\text{s}} \text{ m}^2 \quad \text{or} \quad \frac{\text{m}^3}{\text{s}}$$

THEN THE CONTINUITY EQUATION BECOMES :

$$M = \rho u A = \text{constant kg/s}$$

AS INDICATED IN THE PRECEDING SUBSECTION, WE CAN ACHIEVE ACCEPTABLE ACCURACY IN MOST SITUATIONS WITHIN VENTILATION SYSTEMS BY ASSUMING A CONSTANT DENSITY. THE CONTINUITY EQUATION THEN SIMPLIFIES BACK TO

$$Q = u A = \text{constant} \quad \text{m}^3/\text{s}$$

2. FLUID PRESSURE

- THE CAUSE OF FLUID PRESSURE
- PRESSURE HEAD
- HEAD
- GAUGE HEAD
- ATMOSPHERIC PRESSURE

THE CAUSE OF FLUID PRESSURE

WHEN A MOLECULE REBOUNDS FROM ANY CONFINING BOUNDARY, A FORCE EQUAL TO THE RATE OF CHANGE OF MOMENTUM OF THAT MOLECULE IS EXERTED UPON THE BOUNDARY. IF THE AREA OF THE SOLID/FLUID BOUNDARY IS LARGE COMPARED TO THE AVERAGE DISTANCE BETWEEN MOLECULAR COLLISIONS THEN THE STATISCAL EFFECT WILL BE TO GIVE A UNIFORM FORCE DISTRIBUTED OVER THAT BOUNDARY. THIS IS THE CASE IN MOST SITUATION OF IMPORTANCE IN SUBSURFACE VENTILATION ENGINEERING.

TWO FURTHER CONSEQUENCES ARISE FROM THE BOMBARDMENT OF A VERY LARGE NUMBER OF MOLECULES ON A SURFACE, EACH MOLECULE BEHAVING ESSENTIALLY AS A PERFECTLY ELASTIC SPERE. FIRS, THE FORCE EXERTED BY A STATIC FLUID WILL ALWAYS BE NORMAL TO THE SURFACE. SECONDLY, AT ANY POINT WITHIN A STATIC FLUID, THE PRESSURE IS THE SAME IN ALL DIRECTIONS.

THE QUANTITATIVE DEFINITION OF PRESSURE, P , IS CLEARLY AND SIMPLE

$$P = \frac{\text{Force}}{\text{Area}} \quad \frac{\text{N}}{\text{m}^2}$$

PRESSURE HEAD

IF A LIQUID OF DENSITY, ρ IS POURED INTO A VERTICAL TUBE OF CROSS SECTIONAL AREA, A , UNTIL THE LEVEL REACHES A HEIGHT, h , THE VOLUME OF LIQUID IS

$$\text{volume} = h A \quad \text{m}^3$$

THEN FROM THE DEFINITION OF DENSITY (MASS/VOLUME), THE MASS OF THE LIQUID IS :

$$\text{MASS} = \text{VOLUME} \times \text{DENSITY}$$

$$= h A \rho \quad \text{kg}$$

THE WEIGHT OF THE LIQUID WILL EXERT A FORCE, F , ON THE BASE OF TUBE EQUAL TO MASS X GRAVITATIONAL ACCELERATION (g).

$$F = h A \rho g \quad \text{N}$$

BUT AS A PRESSURE = FORCE / AREA, THE PRESSURE ON THE BASE OF THE TUBE IS

$$P = \frac{F}{A} = \rho g h \quad \frac{\text{N}}{\text{m}^2} \quad \text{or} \quad \text{Pa}$$

HEAD (h)

HEAD IS EXPRESSED OF THE ENERGY CONTENT OF THE LIQUID REFERRED TO ANY ARBITRARY DATUM. IT IS EXPRESSED IN UNITS OF ENERGY PER UNIT WEIGHT OF LIQUID. THE MEASURING UNIT FOR HEAD IS METERS (FEET) OF LIQUID.

GAUGE HEAD (hg)

THE ENERGY OF THE LIQUID DUE TO ITS PRESSURE ABOVE ATMOSPHERIC AS DTERMINED BY A PRESSURE GAUGE OR OTHER PRESSURE MEASURING DEVICE.

METRIC
(Meter)

$$hg = \frac{\text{Pressure Gauge (Kg/cm}^2\text{)}}{(\text{Gravity} \times \text{Specific Gravity of the Liquid})}$$

US unitsn
(Feet)

$$hg = \frac{(\text{Pressure Gauge (PSI)} \times 2.31)}{\text{Specific Gravity of the Liquid}}$$

ATMOSPHERIC PRESSURE

THE BLANKET OF AIR THAT SHROUDS THE EARTH EXTENDS TO APPROXIMATELY 40Km ABOVE THE SURFACE. AT THAT HEIGHT, ITS PRESSURE AND DENSITY TEND TOWARDS ZERO. AS WE DESCEND TOWARDS THE EARTH, THE NUMBER OF MOLECULES PER UNIT VOLUME INCREASES, COMPRESSED BY THE WEIGHT OF THE AIR ABOVE. HENCE, THE PRESSURE OF THE ATMOSPHERE ALSO INCREASES. HOWEVER, THE PRESSURE AT ANY POINT IN THE LOWER ATMOSPHERE IS INFLUENCED NOT ONLY BY THE COLUMN OF AIR ABOVE IT BUT ALSO BY THE ACTION OF CONVECTION, WIND CURRENTS AND VARIATIONS IN TEMPERATURE AND WATER VAPOUR CONTENT.

ATMOSPHERIC PRESSURE NEAR THE SURFACE, THEREFORE, VARIES WITH BOTH PLACE AND TIME. AT THE SURFACE OF THE EARTH, ATMOSPHERIC PRESSURE IS OF THE ORDER OF 100,000 Pa. FOR PRACTICAL REFERENCE THIS IS OFTEN TRANSLATED INTO 100kPa ALTHOUGH THE BASIC SI UNITS SHOULD ALWAYS BE USED IN CALCULATIONS. OLDER UNITS USED IN METEOROLOGY FOR ATMOSPHERIC PRESSURE ARE THE BAR (10^5 Pa) AND THE MILIBAR (100 Pa)

FOR COMPARATIVE PURPOSE, REFERENCE IS OFTEN MADE TO STANDARD ATMOSPHERIC PRESSURE. THIS IS THE PRESSURE THAT WILL SUPPORT A 0.760M COLUMN OF MERCURY HAVING A DENSITY OF 13.5951×10^3 (Kg/m³) IN A STANDARD EARTH GRAVITATION FIELD OF 9.8066 (m/s²)

$$\text{One Standard Atmosphere} = \rho \times g \times h$$

$$= 13.5951 \times 10^3 \times 9.8066 \times 0.760$$

$$= 101.324 \times 10^3 \quad \text{Pa}$$

$$\text{or} \quad 101.324 \quad \text{kPa.}$$

ATMOSPHERIC PRESSURE

FOR MANY PURPOSES, IT IS NECESSARY TO MEASURE DIFFERENCES IN PRESSURE. ONE COMMON EXAMPLE IS THE *DIFFERENCE* BETWEEN THE PRESSURE WITHIN A SYSTEM SUCH AS A DUCT AND THE EXTERIOR ATMOSPHERE PRESSURE. THIS IS REFERRED TO AS *GAUGE PRESSURE*.

$$\text{Absolute pressure} = \text{Atmospheric pressure} + \text{gauge pressure}$$

IF THE PRESSURE WITHIN THE SYSTEM IS BELOW THAT THE LOCAL AMBIENT ATMOSPHERIC PRESSURE, THEN THE NEGATIVE GAUGE PRESSURE IS OFTEN TERMED THE *SUCTION PRESSURE OR VACUUM* AND THE SIGN IGNORED.

THE ABSOLUTE PRESSURE IS ALWAYS POSITIVE. ALTHOUGH MANY QUOTED MEASUREMENTS ARE PRESSURE DIFFERENCES, IT IS THE ABSOLUTE PRESSURE THAT ARE USED IN THERMODYNAMIC CALCULATIONS. WE MUST NOT FORGET TO CONVERT WHEN NECESSARY.



3. FLUIDS IN MOTION

BERNOULLI's EQUATION for IDEAL FLUID

BERNOULLI's EQUATION for IDEAL FLUID

AS A FLUID STREAM PASSES THROUGH A PIPE, THERE WILL BE CHANGE IN ITS VELOCITY, ELEVATION AND PRESSURE. WE WILL CONSIDER THAT THE FLUID IS IDEAL; THAT IS, IT HAS NO VISCOSITY AND PROCEEDS ALONG THE PIPE WITH NO SHEAR FORCES AND NO FRICTIONAL LOSSES. AND WILL IGNORE ANY THERMAL EFFECTS AND CONSIDER MECHANICAL ENERGY ONLY

SUPPOSE WE HAVE A MASS, m , OF FLUID MOVING AT VELOCITY, u , AT AN ELEVATION, Z , AND BAROMETRIC PRESSURE, P . THERE ARE THREE FORMS OF MECHANICAL ENERGY THAT WE NEED TO CONSIDER. ENERGY QUANTITY FROM ZERO TO ITS ACTUAL VALUE IN THE PIPE..

KINETIC ENERGY

IF WE COMMENCE WITH THE MASS, m , AT REST AND ACCELERATE IT TO VELOCITY u IN t SECONDS BY APPLYING A CONSTANT FORCE F , THEN THE ACCELERATION WILL BE UNIFORM AND THE MEAN VELOCITY IS...

$$\frac{0 + u}{2} = \frac{u}{2} \quad \frac{m}{s}$$

THEN, DISTANCE TRAVELLED = MEAN VELOCITY X TIME

$$= \frac{u}{2} t \quad m$$

BERNOULLI's EQUATION for IDEAL FLUID

FURTHERMORE, THE ACCELERATION IS DEFINED AS

$$\frac{\text{increase in velocity}}{\text{time}} = \frac{u}{t} \quad \text{m / s}^2$$

THE FORCE IS GIVEN BY :

$$F = \text{mass} \times \text{acceleration}$$

$$= m \frac{u}{t} \quad \text{N}$$

AND THE WORK DONE TO ACCELERATE FROM REST TO VELOCITY u IS

$$\text{WD} = \text{force} \times \text{distance} \quad \text{Nm}$$

$$= m \frac{u}{t} \times \frac{u}{2} t$$

$$= m \frac{u^2}{2} \quad \text{Nm or J}$$

THE KINETIC ENERGY OF THE MASS m IS , THEREFORE, $m \cdot u^2/2$ (Joules)

BERNOULLI's EQUATION for IDEAL FLUID

POTENTIAL ENERGY

ANY BASE ELEVATION MAY BE USED AS THE DATUM FOR POTENTIAL ENERGY. IF OUR MASS m IS LOCATED ON THE BASE DATUM THEN IT WILL HAVE A POTENTIAL ENERGY OF ZERO RELATIVE TO THAT DATUM. WE THEN EXERT AN UPWARD FORCE, F , SUFFICIENT TO COUNTERACT THE EFFECT OF GRAVITY.

$$\begin{aligned} F &= \text{mass} \times \text{acceleration} \\ &= m g \quad \text{N} \end{aligned}$$

WHERE, g IS THE GRAVITATIONAL ACCELERATION.

IN MOVING UPWARD TO THE FINAL ELEVATION OF Z METERS ABOVE THE DATUM, THE WORK DONE IS..

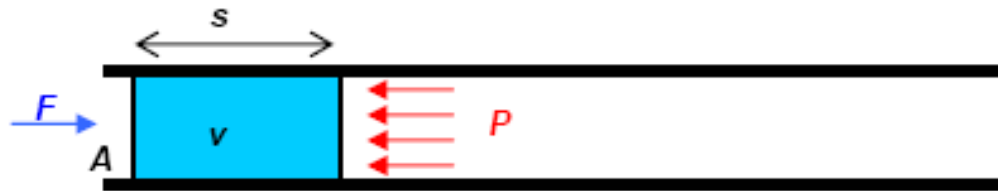
$$\begin{aligned} WD &= \text{Force} \times \text{distance} \\ &= m g Z \quad \text{Joules} \end{aligned}$$

THIS GIVES THE POTENTIAL ENERGY OF THE MASS AT ELEVATION Z .

BERNOULLI's EQUATION for IDEAL FLUID

FLOW WORK

SUPPOSE WE HAVE A HORIZONTAL PIPE, OPEN AT BOTH ENDS AND OF CROSS SECTIONAL AREA A AS SHOWN BELOW. WE WISH TO INSERT A PLUG OF FLUID, VOLUME v AND MASS m INTO THE PIPE. EVEN IN THE ABSENCE OF FRICTION, THERE IS A RESISTANCE DUE TO THE PRESSURE OF FLUID, P , THAT ALREADY EXISTS IN THE PIPE. HENCE, WE MUST EXERT A FORCE, F , ON THE PLUG OF FLUID TO OVERCOME THAT RESISTING PRESSURE. OUR INTENT IS TO FIND THE WORK DONE ON THE PLUG OF FLUID IN ORDER TO MOVE IT A DISTANCE s INTO THE PIPE.



THE FORCE, F , MUST BALANCE THE PRESSURE, P , WHICH IS DISTRIBUTED OVER THE AREA, A

$$F = P A \quad \text{N}$$

Work done = force x distance

$$= P A s \quad \text{J or Joules}$$

HOWEVER, THE PRODUCT AS IS THE SWEEPED VOLUME v , GIVING..

$$WD = P v$$

BERNOULLI's EQUATION for IDEAL FLUID

FLOW WORK

NOW, BY DEFINITION, THE DENSITY IS..

$$\rho = \frac{m}{v} \quad \frac{\text{kg}}{\text{m}^3}$$

or

$$v = \frac{m}{\rho}$$

HENCE, THE WORK DONE IN MOVING THE PLUG OF FLUID INTO THE PIPE IS :

$$\text{WD} = \frac{P m}{\rho} \quad \text{J}$$

or P/ρ Joules per kilogram.

NOW, WE ARE IN A POSITION TO QUANTITY THE TOTAL MECHANICAL ENERGY OF OUR MASS OF FLUID, m

TOTAL MECHANICAL ENERGY	=	KINETIC ENERGY	+	POTENTIAL ENERGY	+	FLOW WORK
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BERNOULLI's EQUATION for IDEAL FLUID

TOTAL
MECHANICAL
ENERGY

$$= \frac{mu^2}{2} + mZg + m\frac{P}{\rho} \quad \text{J}$$

$$\frac{u_1^2 - u_2^2}{2} + (Z_1 - Z_2)g + \frac{P_1 - P_2}{\rho} = 0 \quad \frac{\text{J}}{\text{kg}}$$

OR

HYDRAULIC HORSEPOWER

THE POWER IMPARTED TO THE LIQUID :

METRIC (Kw)

$$H \text{ hp} = \frac{Q \times H \times \text{Sp.Gr}}{366}$$

Where :

Q = Capacity (M3/Hr)
H = Head (Meter)

US Unit (HP)

$$H \text{ hp} = \frac{Q \times H \times \text{Sp.Gr}}{3960}$$

Where :

Q = Capacity (GPM)
H = Head (FEET)



END OF TRAINING
THANK YOU