Centrifugal Pump Training

Presented by:

InService Testing Owners' Group

Orlando, Florida- January 24

Dr. Lev Nelik, P.E., APICS

Dr. Nelik has 30 years experience with pumps and pumping equipment. He is a Registered Professional Engineer, who has published over fifty documents on pumps and related equipment worldwide, including a "Pumps" section for the *Encyclopedia of Chemical Technology* (John Wiley), a section for the *Handbook of Fluids Dynamics* (CRC Press), a book *"Centrifugal and Rotary Pumps: Fundamentals with Applications"*, by the CRC Press, and a book *"Progressing Cavity Pumps"*, by Gulf Publishing.

He is a President of Pumping Machinery, LLC company, specializing in pump consulting, training, and equipment troubleshooting. His experience in engineering, manufacturing, sales, field and management includes Liquiflo Equipment (President), Roper Pump (Vice President of Engineering, and Repair/Overhaul), Ingersoll-Rand (Engineering), and Goulds Pumps (Technology).

Dr. Nelik is an Advisory Committee Member for the Texas A&M International Pump Users Symposium, an Advisory Board Member of *Pumps & Systems* Magazine, Editorial Advisory Board Member of *Water and Wastewater Digest Magazine*, and a former Associate Technical Editor of the *Journal of Fluids Engineering*. He is a Full Member of the ASME, and a Certified APICS. He is a graduate of Lehigh University with Ph.D. in Mechanical Engineering and a Masters in Manufacturing Systems.

He teaches pump training courses in the US and worldwide, and consults on pumps operations and troubleshooting, engineering aspects of centrifugal and positive displacement pumps, maintenance methods to improve reliability, efficiency and energy savings, and optimize pump-to-system performance.

Michael C. Mancini

Michael Mancini is a graduate from Lehigh University with a BSME who has over 30 years experience in pump design, engineering, and repair.

He started work for Ingersoll-Rand in 1974 designing pumps for the SSN 688 and Trident submarines. He worked side-by-side with many renowned pump designers during his tenure with Ingersoll including: Dr. Paul Cooper, Igor Karassik, Val Lobonoff, and Fred Antunes.

As VP of Worldwide Aftermarket for IDP, he had profit responsibility for a $370 million business and control over 22 repair centers.

As President of a large independent pump service company, he worked closely with Dr. Elemer Makay and helped pioneer processes for inspection and repair to reduce total life-cycle costs.

As President of his consulting company, he has provided training to over 500 mechanics and engineers. He has completed work for over 25 separate customer organizations in various markets: performing root cause analysis, developing specifications, and implementing strategic pump programs.

Purpose

Significantly increase pump-related operating profits by understanding pump fundamentals, failure modes and their detection; and applying state-of-the-art design and best-in-class repair and manufacturing processes to improve or solve deficiencies for improved pump performance and life.

Course Agenda

- Introductions
- Expectations
- Pump Type Configurations
- Pump Performance
- System Curve
- Suction Conditions
- Generic Failure Mechanisms
- Question & Answer Session

Objectives

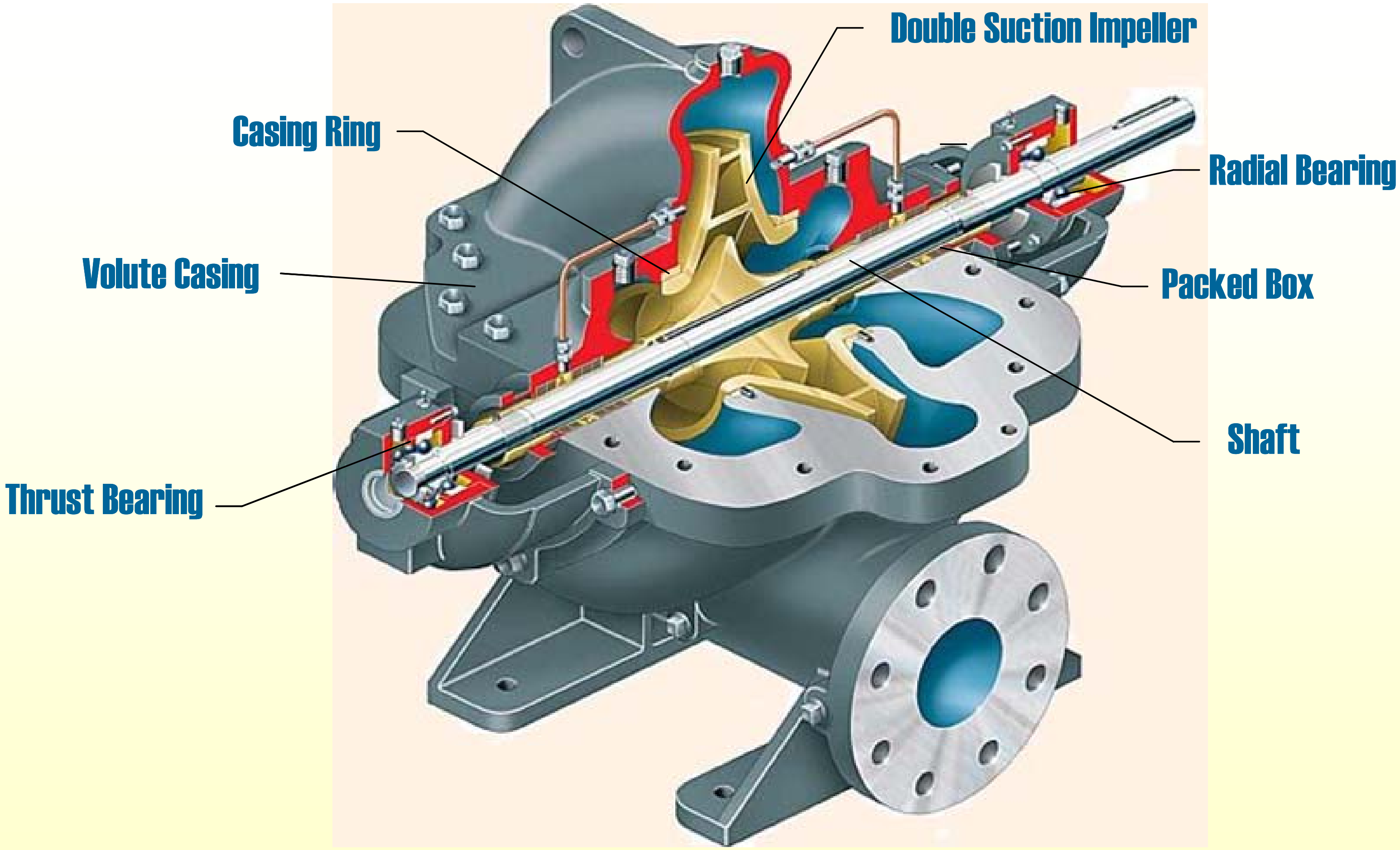
- Understand pump fundamentals
- Understand the probable root causes of degradation or failure associated with various pump problems
- Understand the state-of-the-art technologies to upgrade existing designs to achieve improved operation and life
- Learn how to determine where a pump is operating and how to modify its performance to achieve optimum performance

Suction Recirculation Video

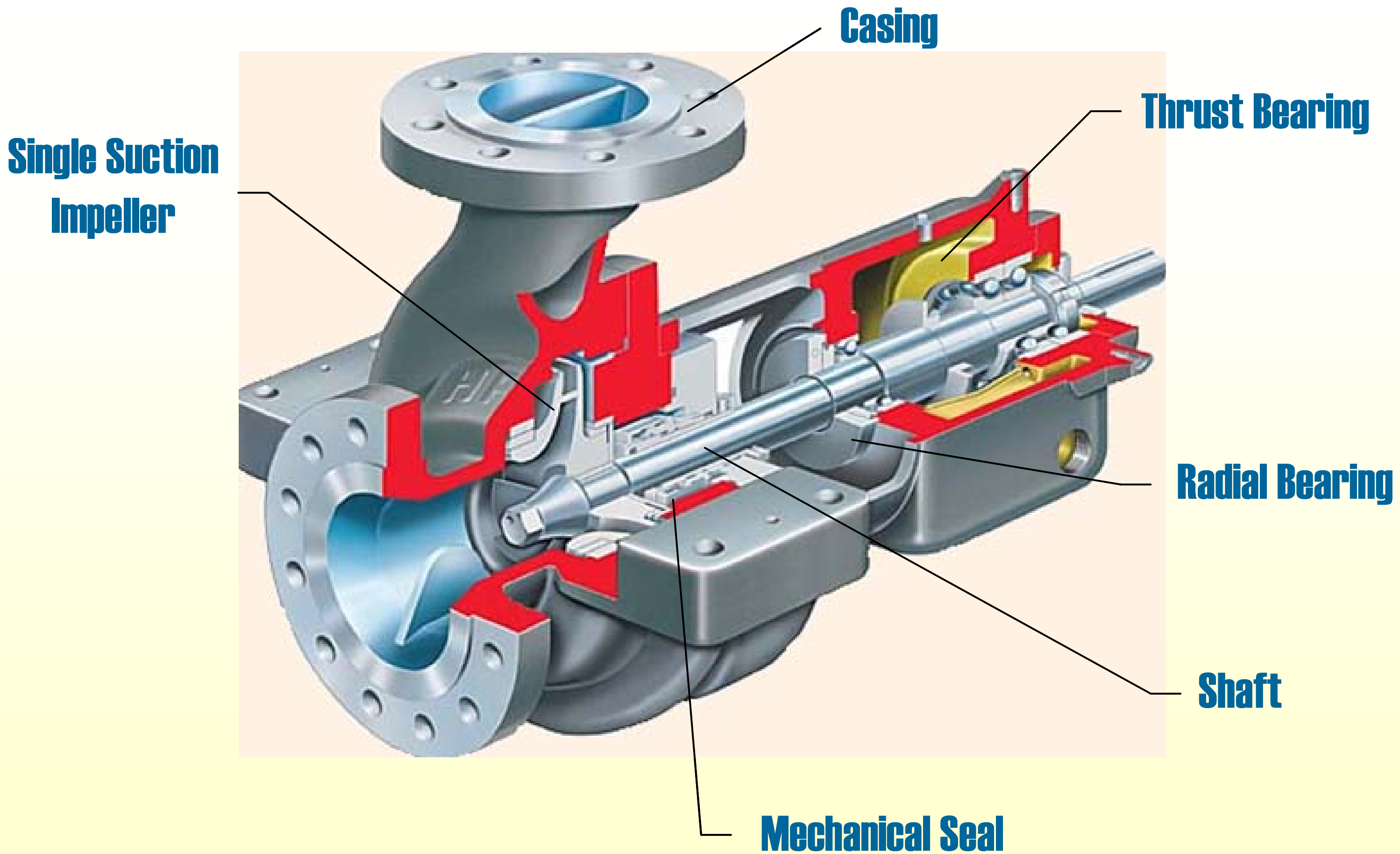
Basic Centrifugal Pump Types

- Single Stage, Double Suction (SSDS)
- End Suction
- Horizontal Multi-stage, In-line Impellers
- Horizontal Multi-stage, Opposed Impellers
- Vertical Wet-Pit
- Vertical Can

Single Stage, Double Suction

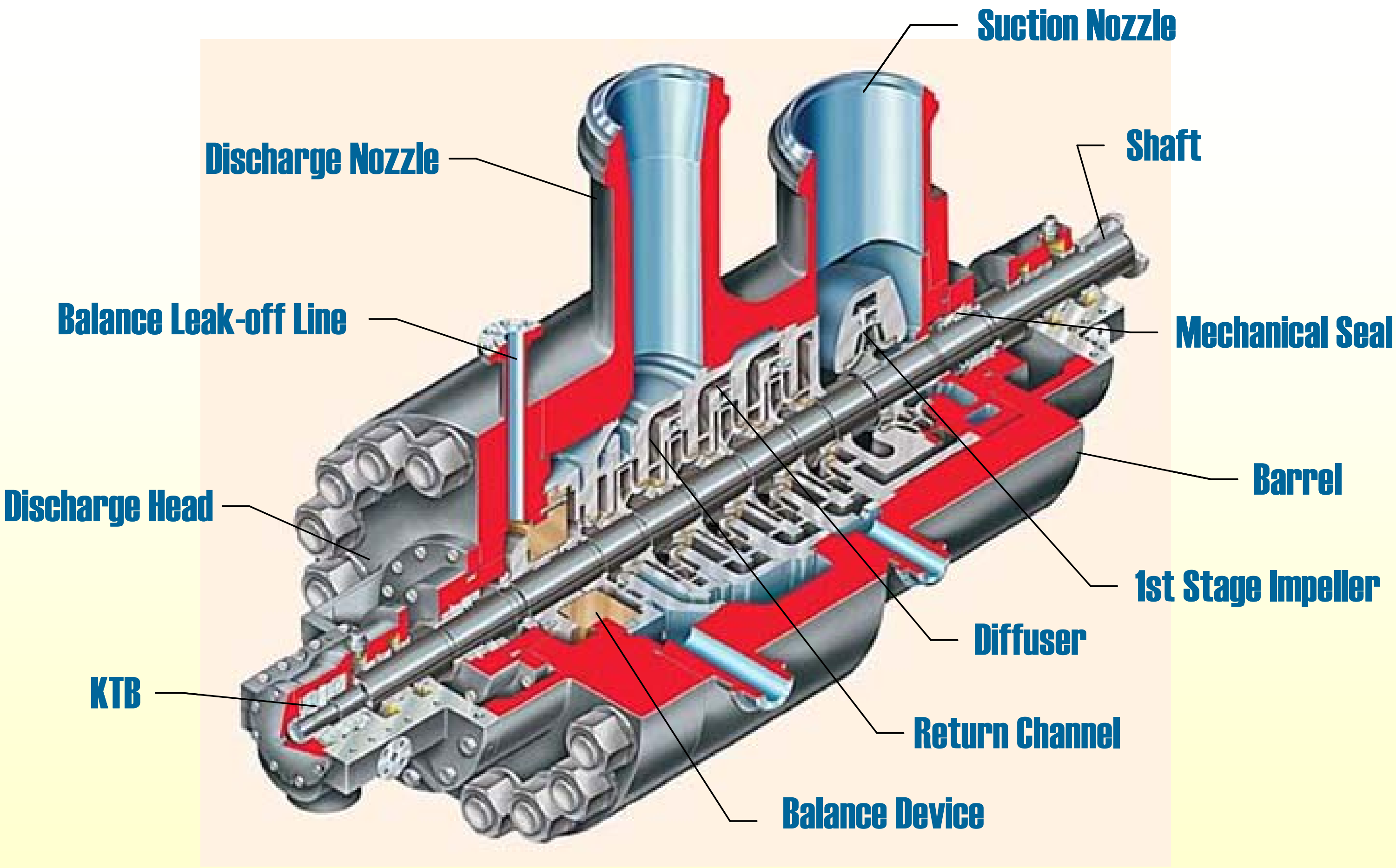


End Suction



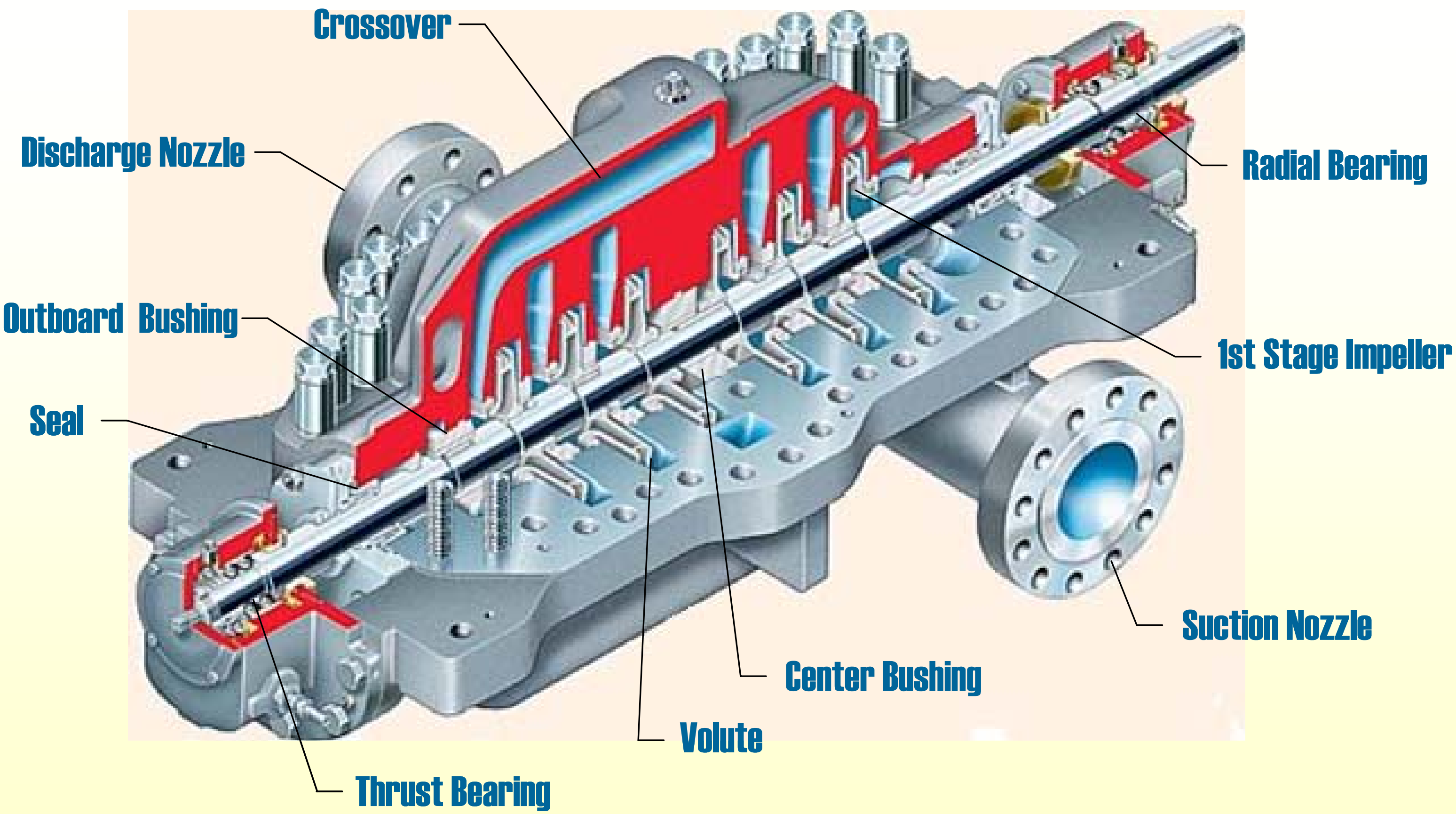
Casing
Thrust Bearing
Single Suction
Impeller
Radial Bearing
Shaft
Mechanical Seal

Multi-stage In-line Impellers

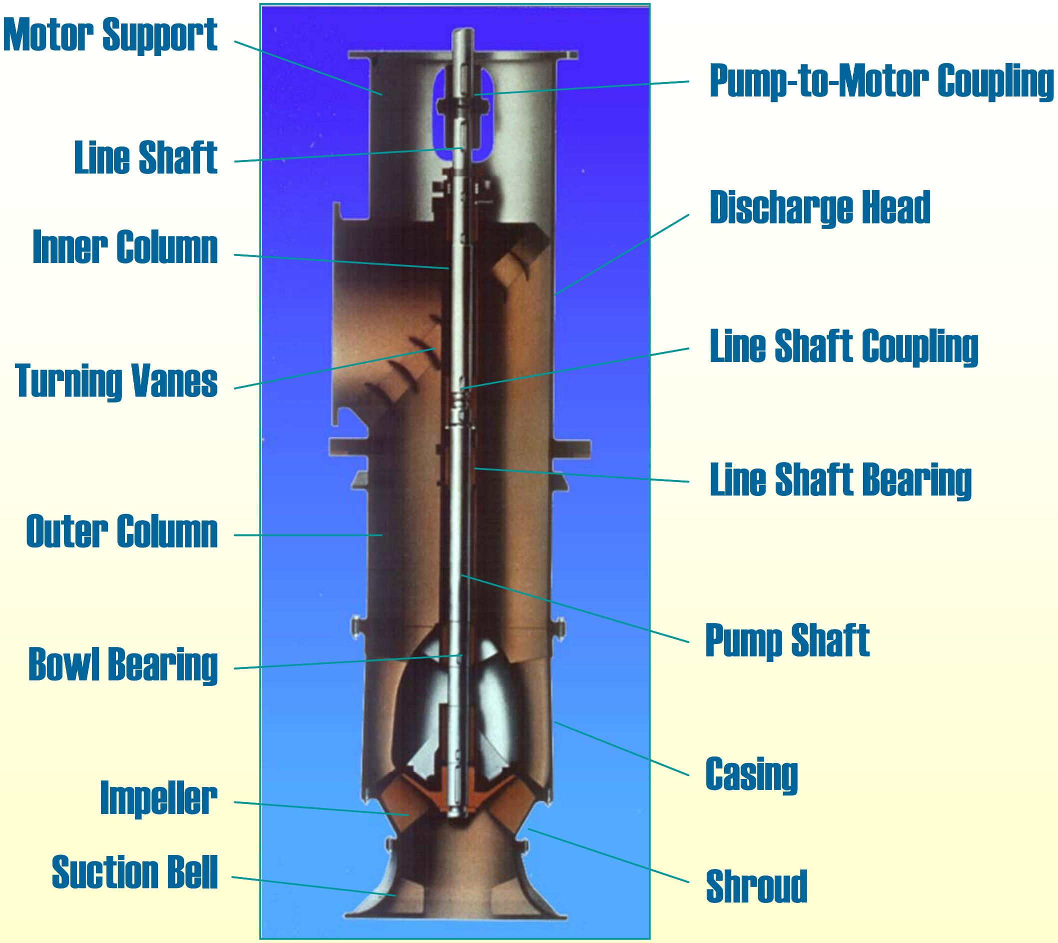


Diffuser

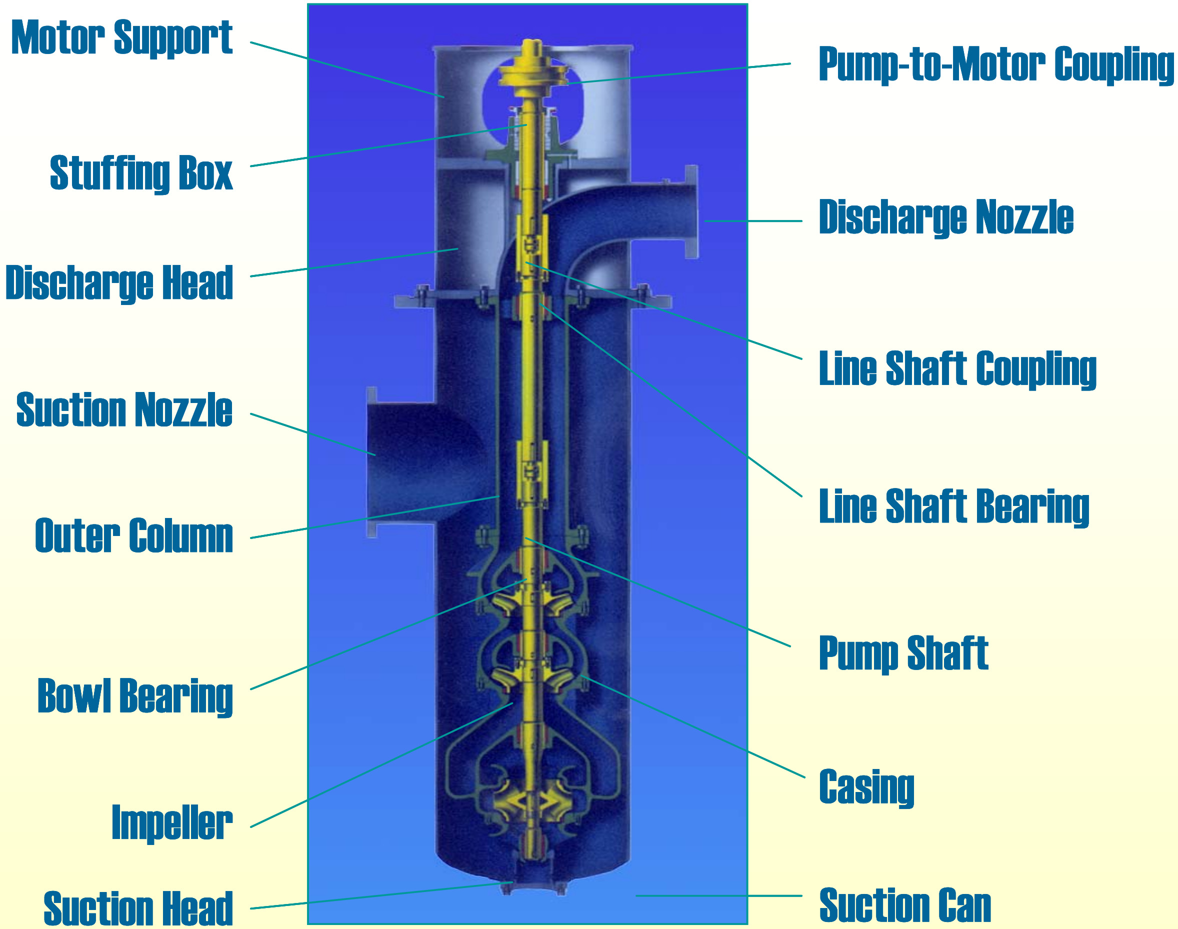
Multi-stage, Opposed Impellers



Vertical Wet-Pit



Vertical Can



Basic Conversion Rules

- FLOW:
- GPM / 4.403 = M3/HR
- GPM / 15.9 = liters/sec

- VELOCITY:
- * FT/SEC = gpm x 0.321 / (π x in2 / 4)
- * M/SEC = m3/hr x 277.8 / (π x mm2 / 4) = m3/hr x 0.43 / (π x in2 / 4)

- PRESSURE:
- * PSI / 14.7 = atm
- PSI / 14.2 = kg/cm2
- kg/cm2 = atm / 1.033
- PSI / 14.5 = Bars
- PSI / 145 = MPa

- HEAD:
- FEET = psi x 2.31 / SG
- METERS = atm x 10.3 / SG = kg/cm2 x 10.0 /SG

- POWER:
- BHP = gpm x ft x SG / 3960 / EFF
- KW = m3/hr x m x SG / (367.5 x EFF)
- HP x 0.746 = KW

Fundamentals Theory

- As pan rotates, the fluid becomes dished and overflows
- Due to the centrifugal force, the fluid is lifted or *pumped* a height “H”

Pan Partially Filled | Pan Rotating on Shaft

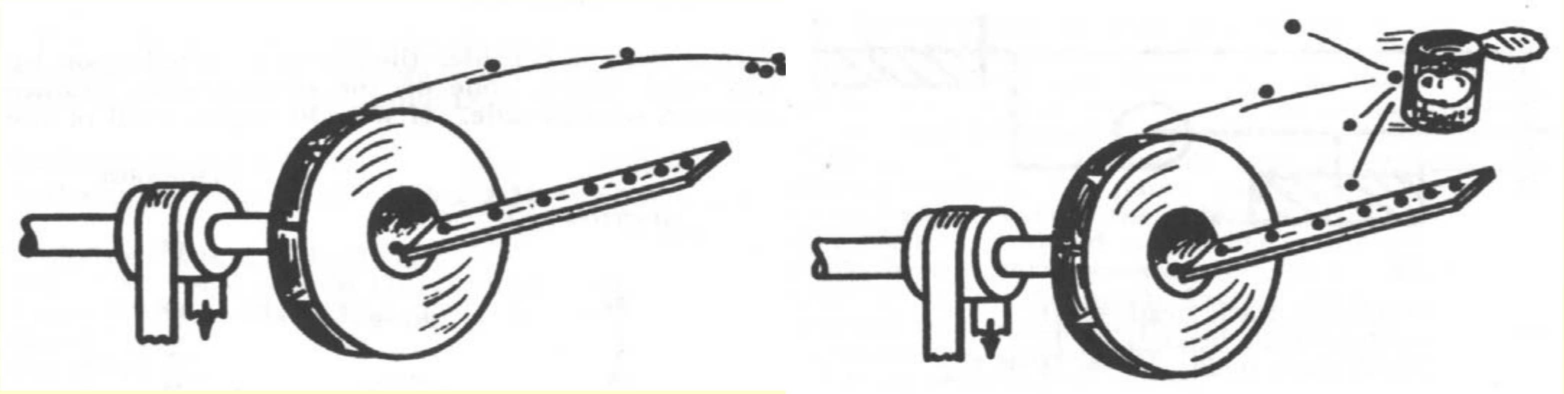
Fundamentals Theory

- Water is thrown considerable distance
- The faster you whirl, the sooner the bucket will empty, and the further the water will be thrown (greater head)

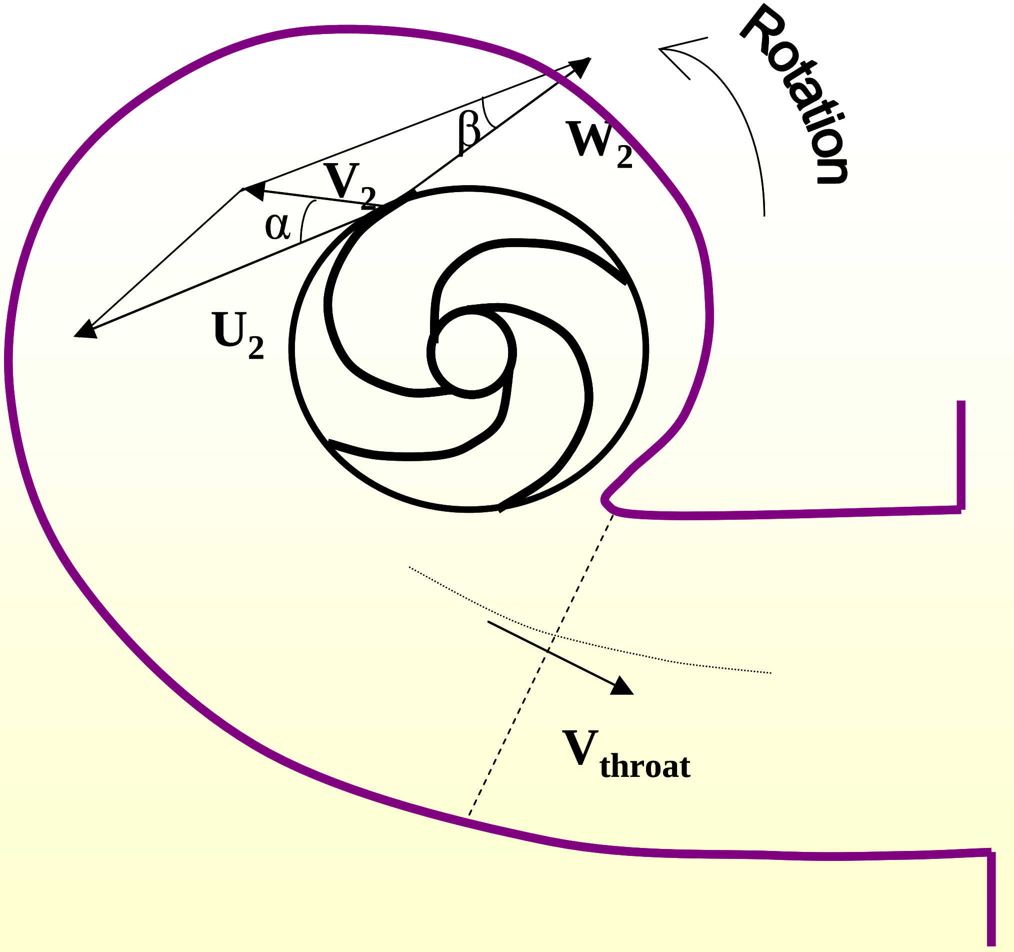
Bucket of Water with Hole in Bottom

Fundamentals Theory

- Exit velocity of BB Shot much greater than entrance velocity
- If BB's are allowed to go free thru the air, no useful work is done
- If a tin can is placed in line with the shot, the tin can will move, and the shot will exert pressure as it loses its velocity



Fundamentals Theory



Fundamentals Theory

Summary

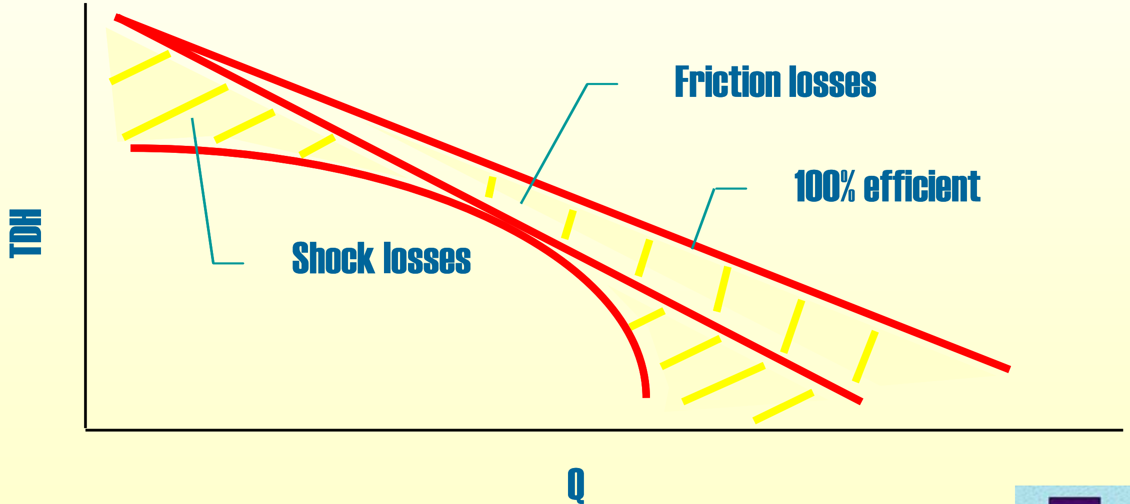
- Fluid is led to the eye or center of the impeller and is set into rotation by the impeller vanes
- Via centrifugal force, fluid is thrown from the periphery of the impeller with considerable velocity and pressure
- The casing, which surrounds the impeller, has a volute or diffuser shaped passage of increasing area
- The casing collects the fluid leaving the impeller and converts a portion of its velocity energy into additional pressure energy.
- The casing passage leads to the discharge nozzle of the pump where piping conducts the fluid to its place of use

Pump Curve

- Curve Shape
- Pump Capacity
- Total Developed Head
  - Suction Head
  - Discharge Head
- Parallel Pump Operation
- Series Pump Operation
- Brake Horsepower
- Affinity Laws
- Specific Speed

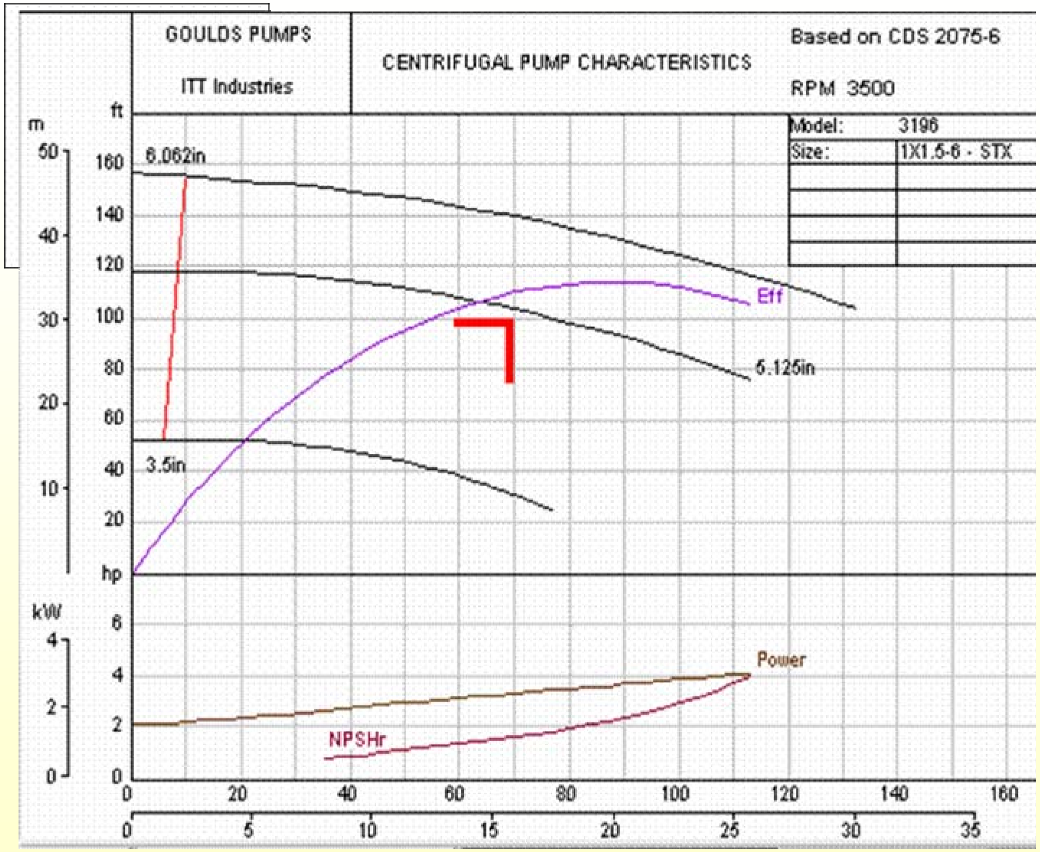
Curve Shape

- Total developed head (TDH) is inversely proportional to capacity (Q)
- Totally efficient pump would produce straight line curve.
- Inefficiencies caused by shock losses and friction losses make the H-Q curve parabolic.
- Pump's best efficiency point (BEP) is where the parabolic curve is closest to the ideal curve.



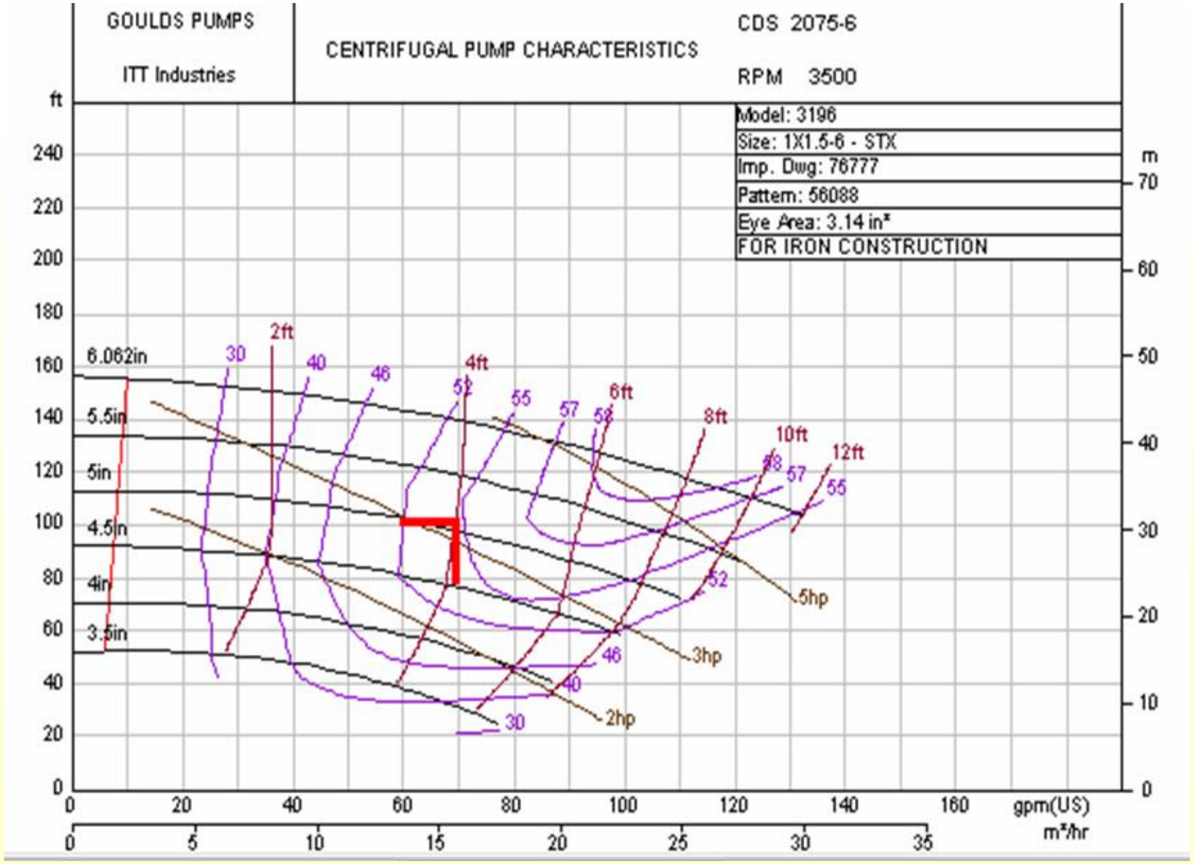
Curve Shape

Individual Curve



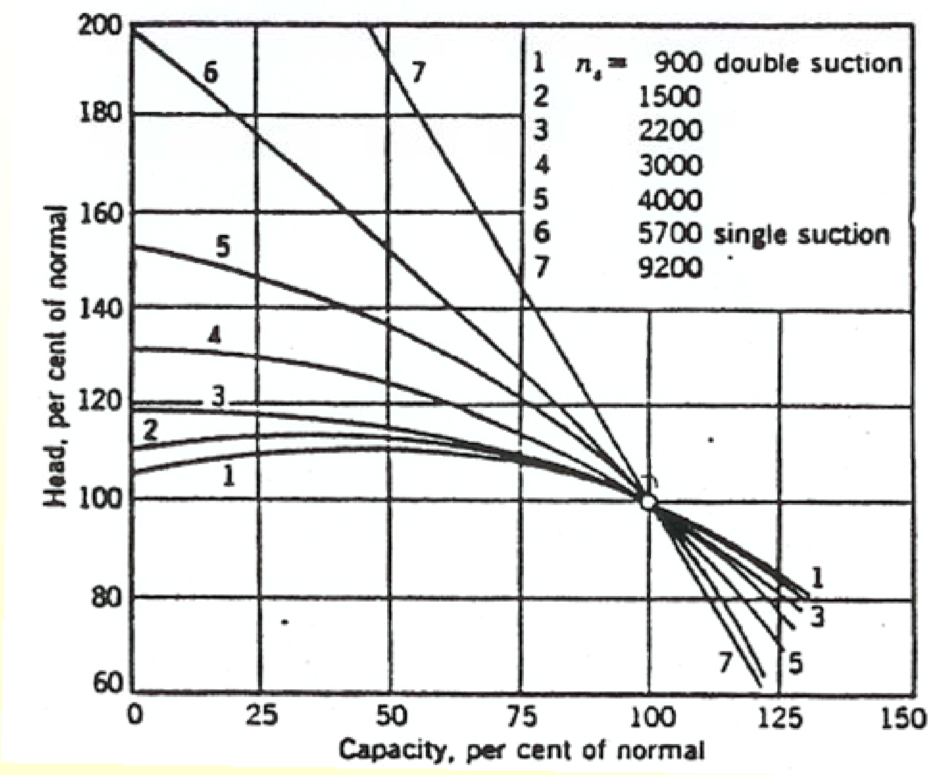
Curve Shape

Family of Curves

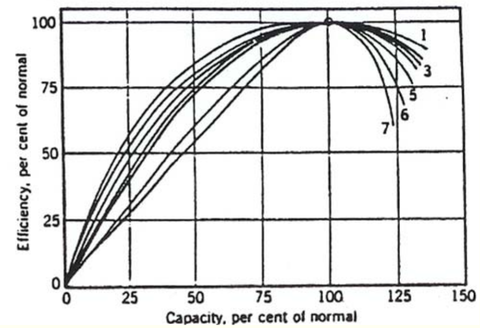


Curve Shape

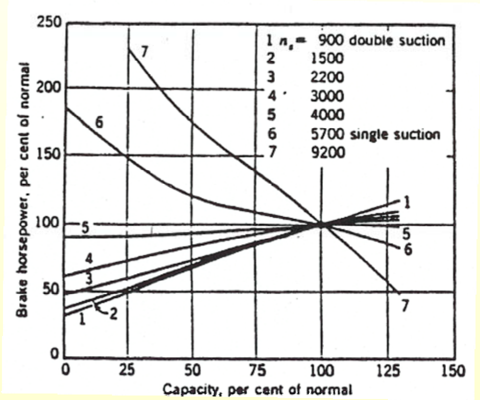
Function of Specific Speed, N_s



Total Developed Head



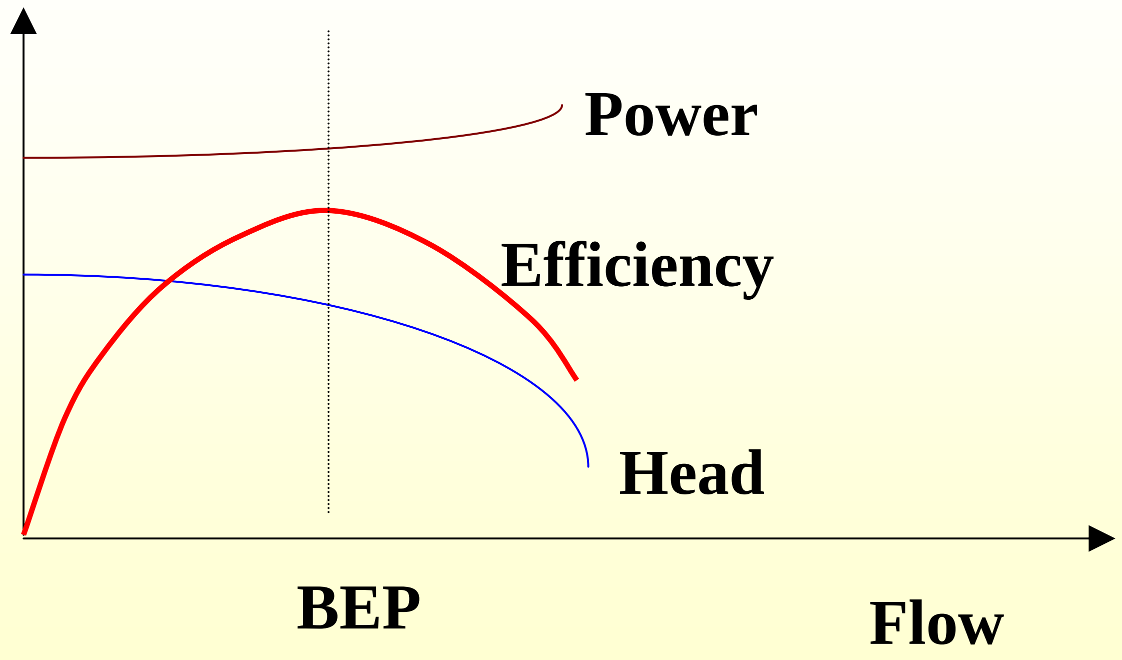
Efficiency



BHP

Best Efficiency Point

BEP is defined as flow at which the sum of all losses is the lowest. Overall efficiency is less to *the right* and *to the left* of BEP.



Pump Capacity

- Pump capacity refers to a rate of flow typically expressed in either gallons per minute (gpm), barrels per day, or pounds per hour (lb/hr).
- GPM is independent of the fluid pumped.
- LB/HR is dependent on the fluid specific gravity

$$GPM = \frac{LB/HR}{500 \times sp.gr.}$$

$$500 = 60\, min/hr \times 8.33\, lb/gal$$

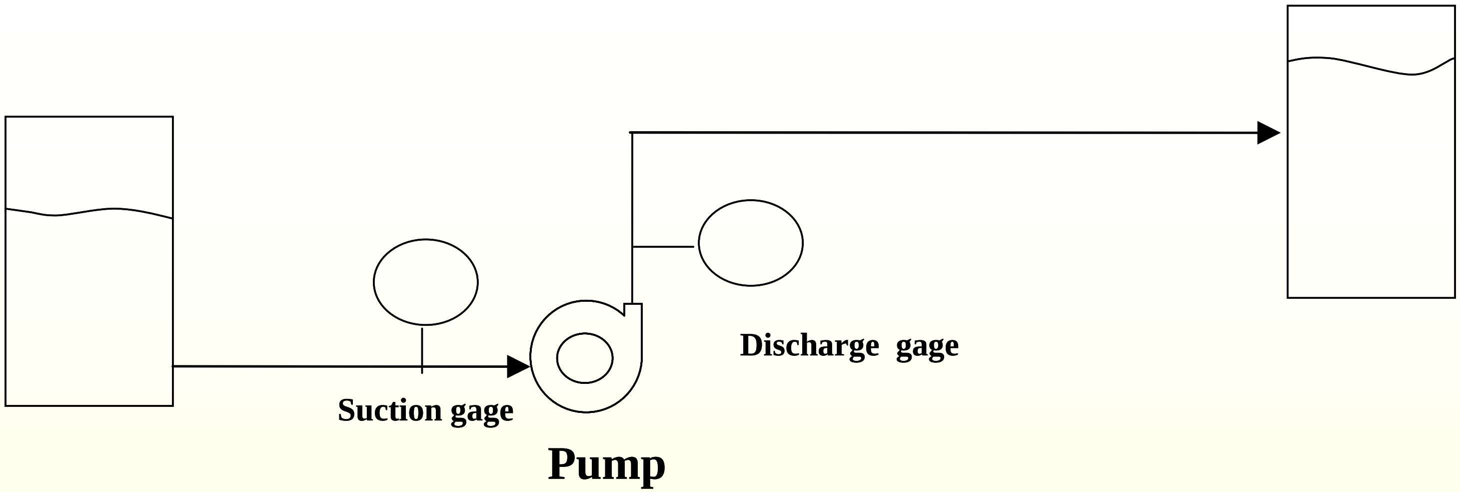
Total Developed Head

- The total developed head is equal to the discharge head minus the suction head, ($TDH = h_d - h_s$), and is typically expressed in either "feet" or "psi".
- Feet is independent of the liquid pumped
- PSI is dependent on the liquid specific gravity.

$$\text{Feet} = \frac{2.31 \times \text{psi}}{\text{sp.gr.}}$$

$$2.31 = 144\ \text{in}^2/\text{ft}^2\ /\ 62.4\ \text{lb}/\text{ft}^3$$

Pump Head



SUCTION HEAD = Total Static *plus* Dynamic, measured at pump inlet

DISCHARGE HEAD = Total Static *plus* Dynamic, measured at pump exit

PUMP HEAD = DISCHARGE HEAD *minus* SUCTION HEAD *plus* correction for the difference in gage elevations

Static Head is what the (absolute) gage reads, converted to feet of water

Dynamic Head is the same as Velocity Head

Suction Head (h_s)

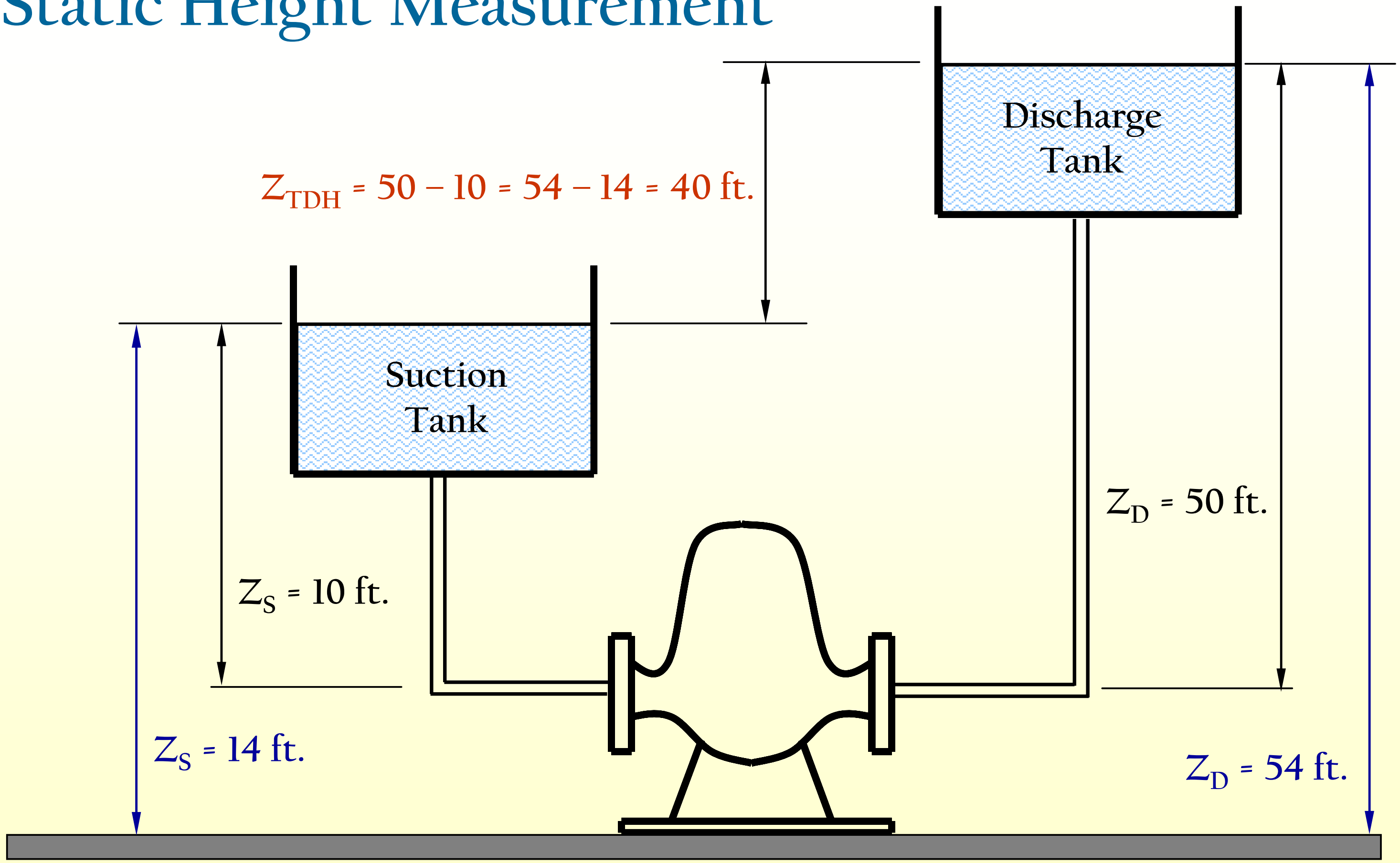
The suction head is equal to the static height that the liquid is above the 1st stage impeller eye[1] less all suction line losses (including entrance loss) plus any gage pressure existing at the suction supply source.

$$h_s = z_s - f_s + p_{s,G}$$

[1]or any other datum plane consistent with measuring total discharge head

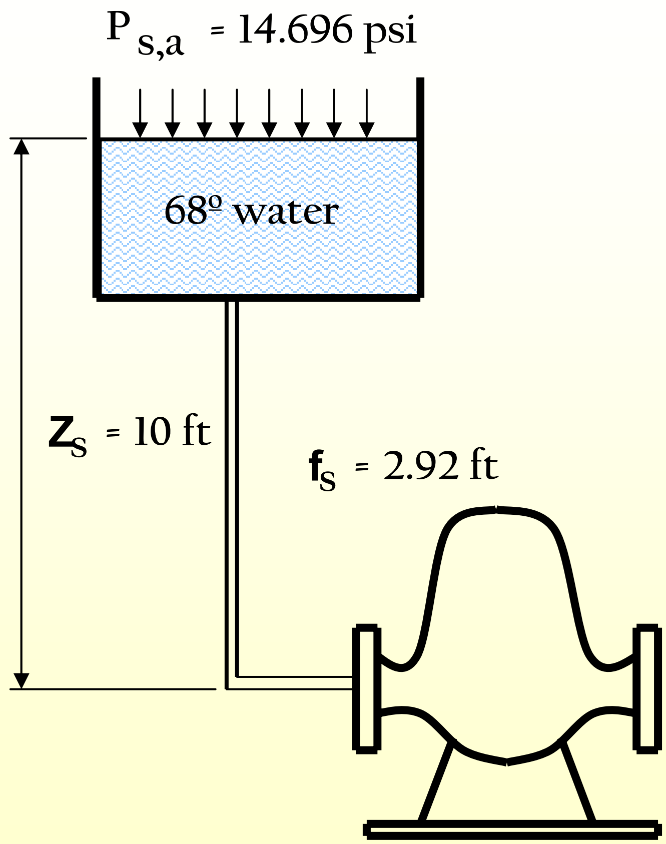
Suction Head (h_s)

Static Height Measurement



Suction Head (h_s)

Horizontal Configuration; Open Tank



$$h_s = z_s - f_s + p_{s,G}$$

Z_s = 10.00 ft.

f_s = 2.92 ft.

P_g = 14.696 psia = 0 psig

h_s = 0 + 10.00 – 2.92 = 7.08 ft.

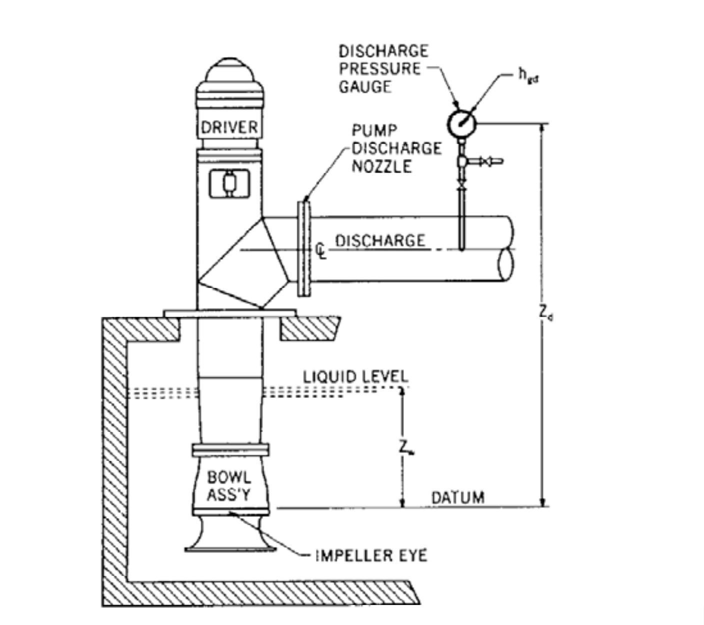
Suction Head (h_s)

Vertical pumps, open pit:

$$H_S = Z_w$$

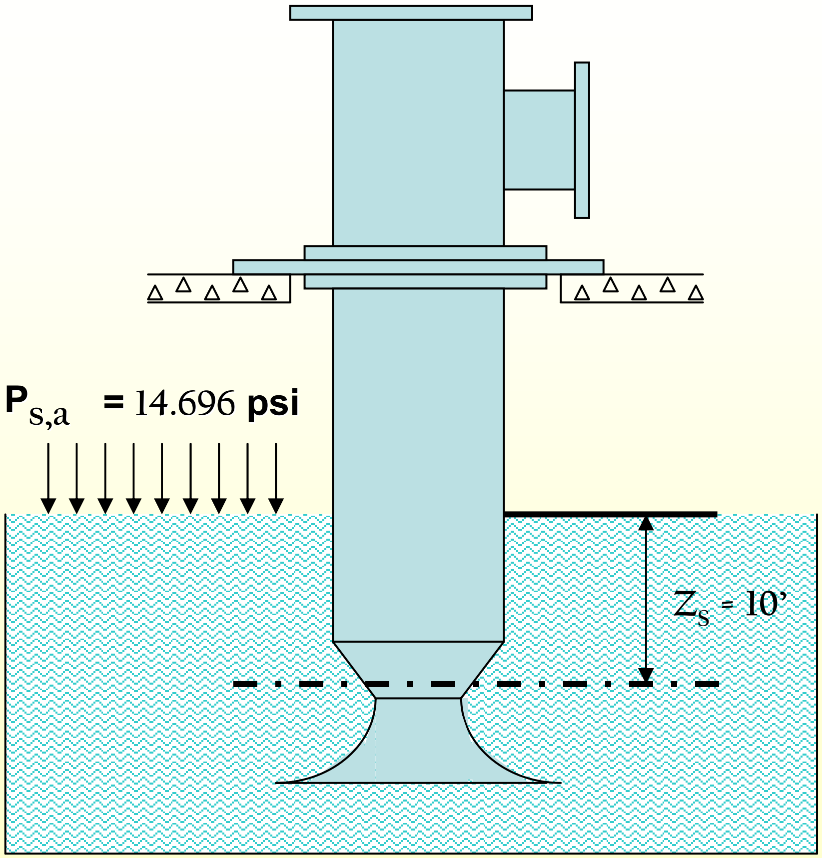
$$H_D = h_{gd} + h_{vd} + Z_d$$

$$H = H_D - H_S \quad \sim \quad h_{gd} + (Z_d - Z_w)$$



Suction Head (h_s)

Vertical Wet-Pit; Open Sump



$$h_s = z_s - f_s + p_{s,G}$$

Z_s = 10.00 ft.

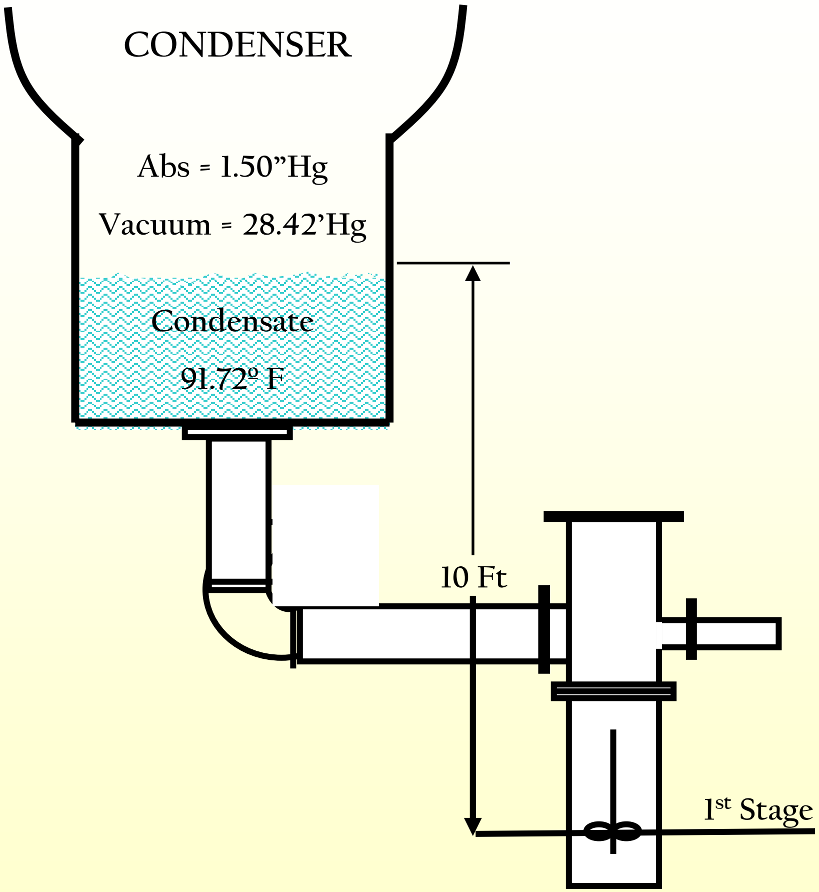
f_s = 0 ft.

P_g = 14.696 psia = 0 psig

h_s = 0 + 10.00 – 0 = 10 ft.

Suction Head (h_s)

Vertical Can Pump; Closed Tank



$$h_s = z_s - f_s + p_{s,G}$$

Z_s = 10.00 ft

f_s = 2.92 ft

Vacuum = 28.42" Hg = - 32.37 ft

h_s = 10.00 - 2.92 - 32.37 = - 25.29 ft

Suction Head (h_s)

On an existing installation, suction head would be the reading of a gage at the suction flange converted to feet of liquid and corrected to the pump centerline elevation plus the velocity head (in feet of liquid) at the point of gage attachment.

Velocity Head = $V^2/2g$ = $0.00259\ (gpm)^2/d^4$

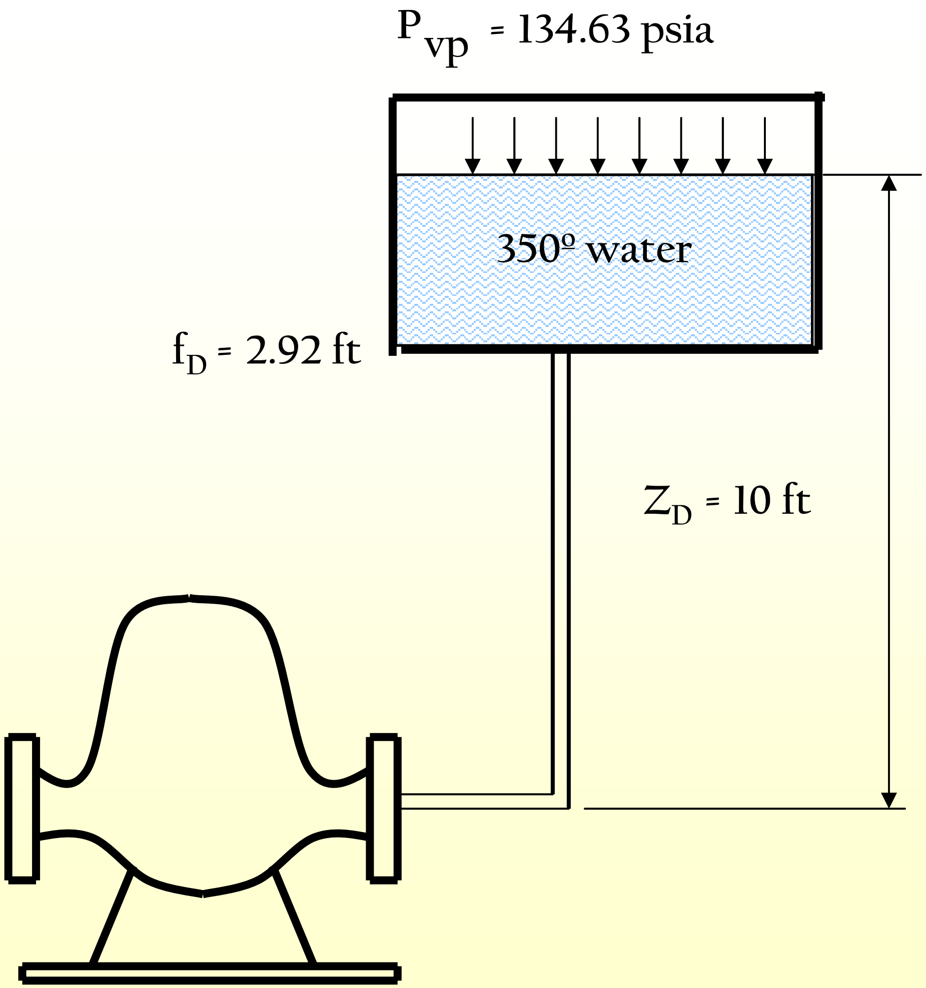
Discharge Head (h_D)

The discharge head is equal to the static height that the liquid is being pumped to above the 1st stage impeller eye[1], plus all discharge line losses (including exit loss), plus any gage pressure in discharge chamber.

$$h_D = z_D + f_D + p_{D,G}$$

[1]or any other datum plane consistent with measuring total discharge head

Discharge Head (h_D)



$$h_D = z_D + f_D + p_{D,G}$$

Z_D = 10.00 ft.

f_D = 2.92 ft.

P_D = 310.69 ft.

h_D = 310.69 + 10 + 2.92 = 323.61

$$310.69 = \frac{(134.63 - 14.696) \times 2.31}{0.892}$$

Discharge Head (h_D)

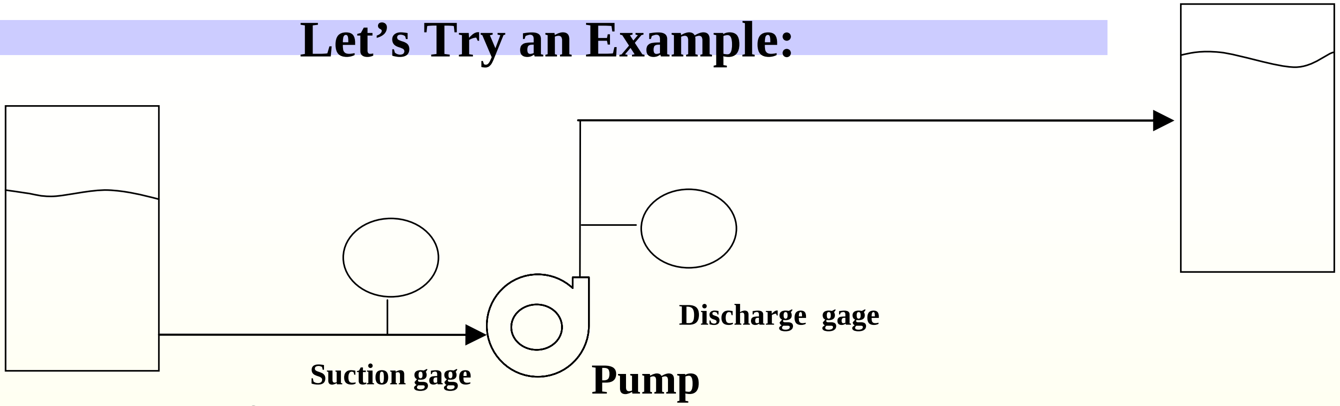
On an existing installation, discharge head would be the reading of a pressure gage at the discharge flange converted to feet of liquid and corrected to the 1st stage impeller eye[1] plus the velocity head (in feet of liquid) at the point of gage attachment.

[1]or any other datum plane consistent with measuring total suction head

Velocity Head = $V^2/2g$ = $0.00259\ (gpm)^2/d^4$

Pump Head

Let's Try an Example:



Flow = 70 gpm of water

Suction Gage reads 5 psig

Discharge Gage reads 80 psig

Suction pipe is 1.5"

Discharge pipe is 1"

Suction Gage is 1' above pump centerline

Discharge Gage is 6' above pump centerline

What is a Pump Head in this case?

Pump Head

Discharge line Velocity Head, as calculated earlier is 12.5 feet

Suction pipe area is $3.14 / 4 \times 1.5^2 = 1.8$ in^2

Suction line Velocity is $70 \times 0.321 / 1.8 = 12.4$ ft/sec, and

Velocity Head at the suction pipe is $12.4^2 / 64.4 = 2.5$ feet

In absolute units, Suction Pressure is 5+14.7 = 19.7 psiA (19.7 x 2.31 /1.0=45.5 ft)

In absolute units, Discharge Pressure is 80 + 14.7 = 94.7 psiA (94.7 x 2.31 /1.0 = 218.8 ft)

Suction Head = 45.5 + 2.5 = 48 ft

Discharge Head = 218.8 + 12.5 = 231ft

Gage Elevation difference = 6 – 1 = 5 ft

Pump Head = 231 – 48 + 5 = 188 ft

Note that in many instances the velocity head contribution is relatively small, and can be neglected for rough estimates. Same goes for gage elevation correction.

Brake Horsepower

$$\mathrm{BHP} = \frac{T \times \mathrm{rpm}}{5250}$$

where:

T = Torque, ft-lb

5250 = 33,000 ft-lb/min/bhp / 2∏ radians/rev

Brake Horsepower

$$BHP = \frac{\sqrt{phases} \times I \times E \times pf \times e_{Motor}}{746}$$

where:

I	=	Amperes
E	=	Volts
e	=	Motor efficiency
pf	=	Motor power factor

Brake Horsepower

Example

1. Using motor data, calculate BHP
2. Using horsepower equation[1], calculate flow

I	E	e_{motor}	pf	BHP	TDH	e_{pump}	sp.gr.	Q
(amps)	(volts)	(%)	(%)	(bhp)	(ft)	(%)		(gpm)
79.0	7,065	91.5	0.880	1,043	325	39.2	0.9986	4,989
81.0	7,065	91.5	0.880	1,083	272	68.1	0.9986	10,737
91.0	7,065	92.2	0.880	1,224	253	84.7	0.9986	16,239
94.0	7,065	92.2	0.880	1,258	215	86.6	0.9986	20,080

[1] $BHP = \frac{Q \times TDH \times sp.gr.}{3960 \times e}$ $\quad Q = \frac{BHP \times 3960 \times e}{TDH \times sp.gr.}$

Affinity Laws

What happens to Flow, Head and Power with Speed?

Flow changes DIRECTLY (linear) with RPM…

Head changes as a SQUARE of RPM…

Power is proportional to Flow times Head – it changes as CUBE of RPM…

$$Q \sim RPM$$

$$H \sim RPM^2$$

$$BHP \sim RPM^3$$

Affinity Laws

Example (speed change):

$$N_1/N_2 = D_1/D_2 = Q_1/Q_2 = \left(TDH_1/TDH_2\right)^{1/2} = \left(BHP_1/BHP_2\right)^{1/3}$$

1800 rpm				1200 rpm			
Q	H	e	bhp	Q	H	e	bhp
0	1000	0.00		0	444	0.00	
100	950	0.20	120	67	422	0.20	36
200	850	0.45	95	133	378	0.45	28
300	700	0.68	78	200	311	0.68	23
400	500	0.75	67	267	222	0.75	20

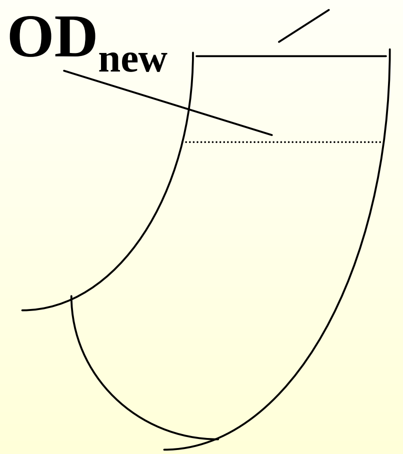
Affinity Laws

When impeller OD is trimmed – Flow, Head and Power follow the Affinity Laws very similar to the case of speed change:

$$Q \sim x$$

$$H \sim x^2$$

$$BHP \sim x^3$$



where x is a cut ratio: $x = OD_{new} / OD_{old}$

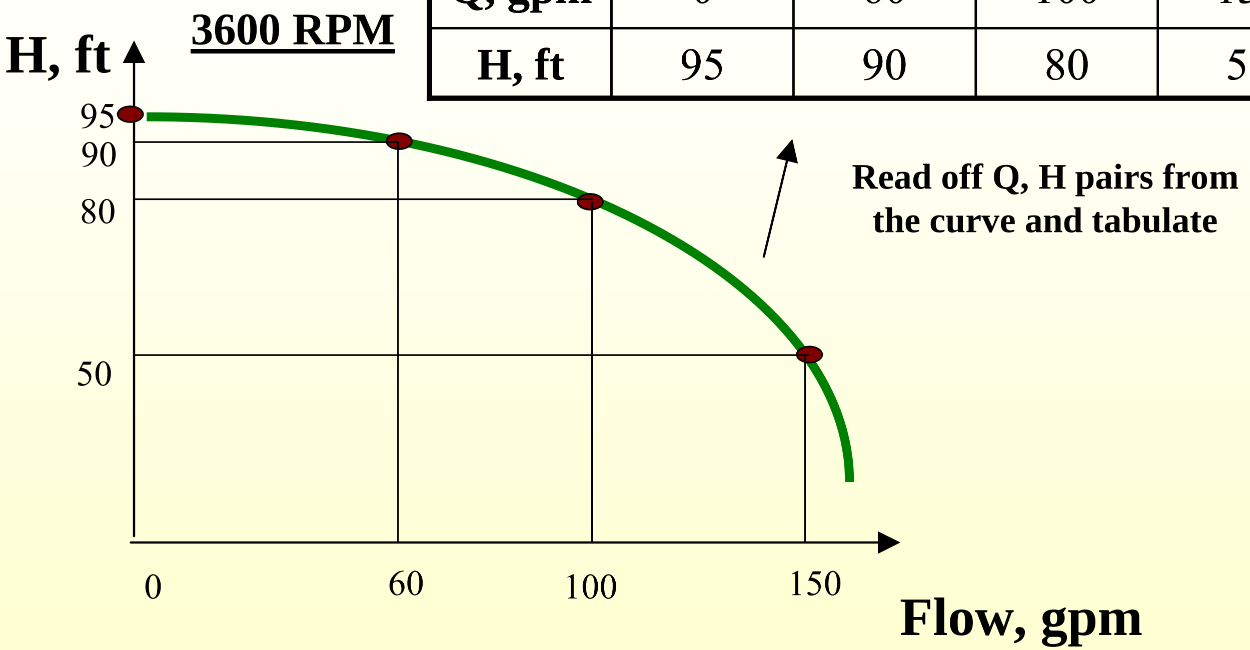
(<u>Note</u>: additional correction applies for cuts over 10%)

Affinity Laws

Example: Construct a New Curve at 70% Pump Speed

(2520/3600 = 0.70)

Q, gpm	0	60	100	150
H, ft	95	90	80	50



Affinity Laws

This is how we do it:

New speed is 0.70 x 3600 = 2520 RPM

Speed ratio is 0.70 (70% slowdown)

Q ~ 0.70

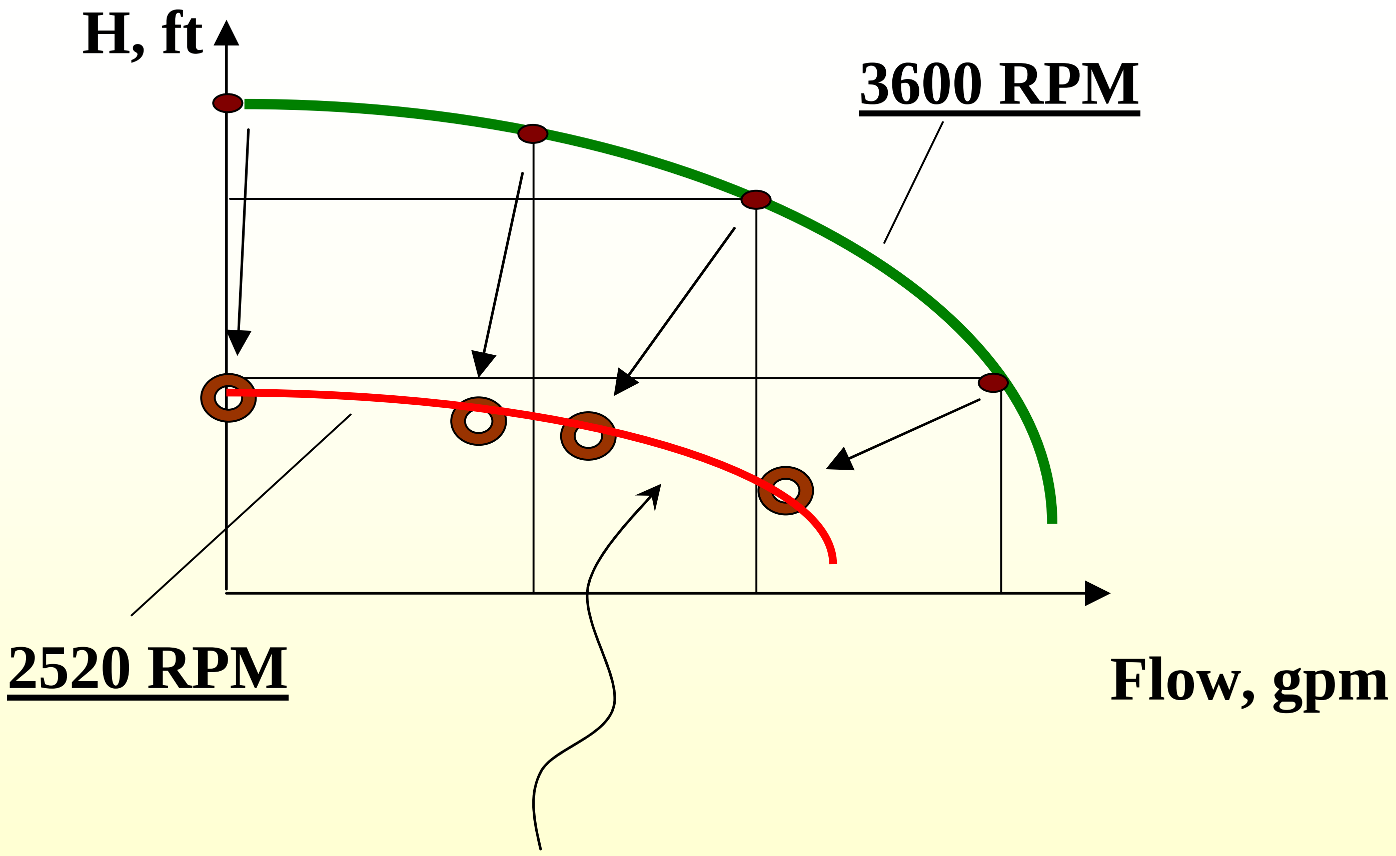
H ~ 0.70^2 = 0.49

Multiply each value of Flow in the original table by 0.70, and Head by 0.49:

Q, gpm	0 x 0.70 = 0	60 x 0.70 = 42	100 x 0.70 = 70	150x0.70 = 105
H, ft	95 x 0.49 = 46.6	90 x 0.49 = 44.1	80 x 0.49 = 39.2	50 x 0.49 = 24.5

Plot these new values:

Affinity Laws

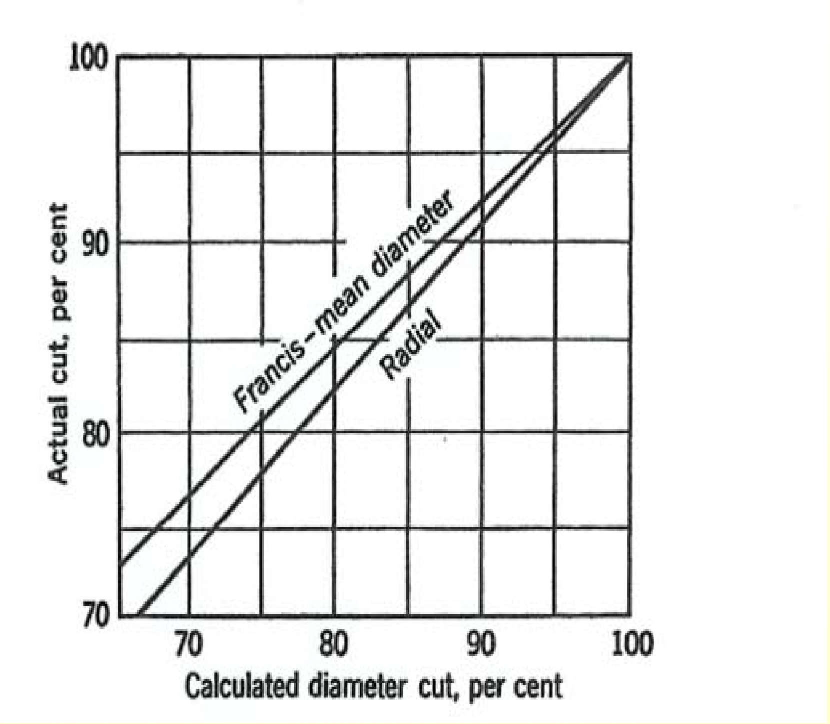


…and draw a curve through new points…

Affinity Laws

Comments:

- The actual impeller diameter ratio should be increased somewhat to compensate for inaccuracies due to other losses
- The accuracy of applying the affinity laws decreases with increasing specific speed



System Curve

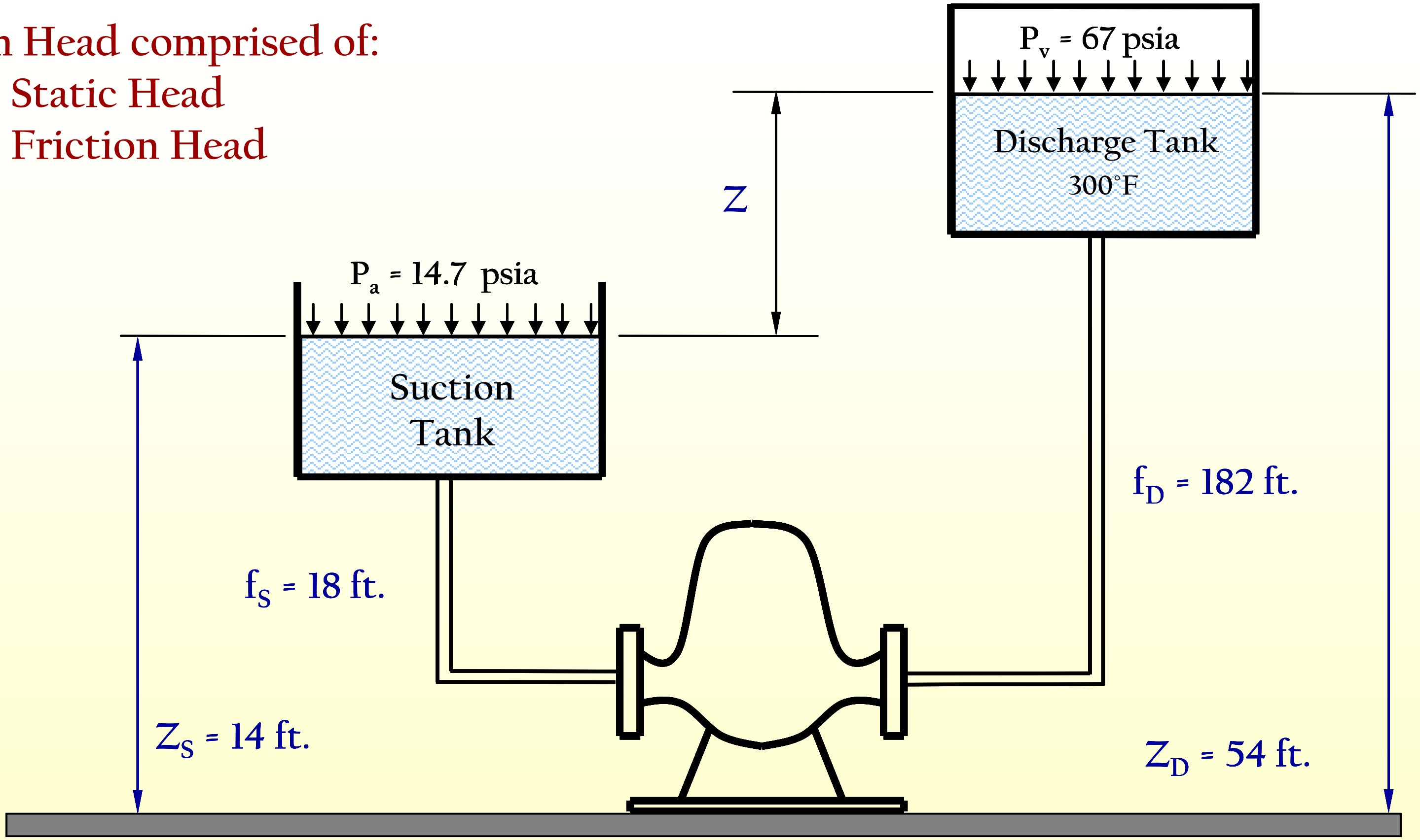
- Static Head
- Friction Head
- Total System Head
- Pump Operation
- Application Examples

System Curve

Example

System Head comprised of:

- Static Head
- Friction Head



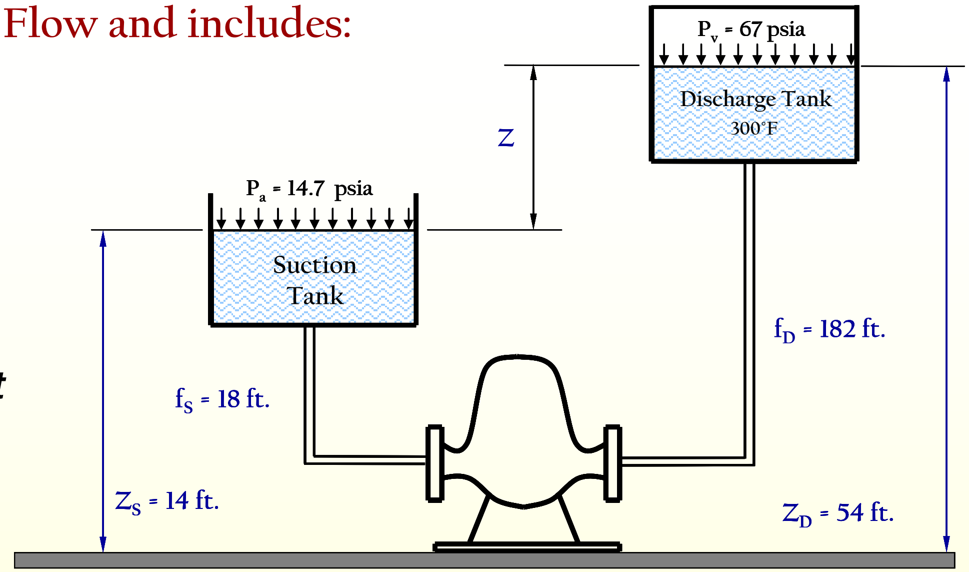
Static Head

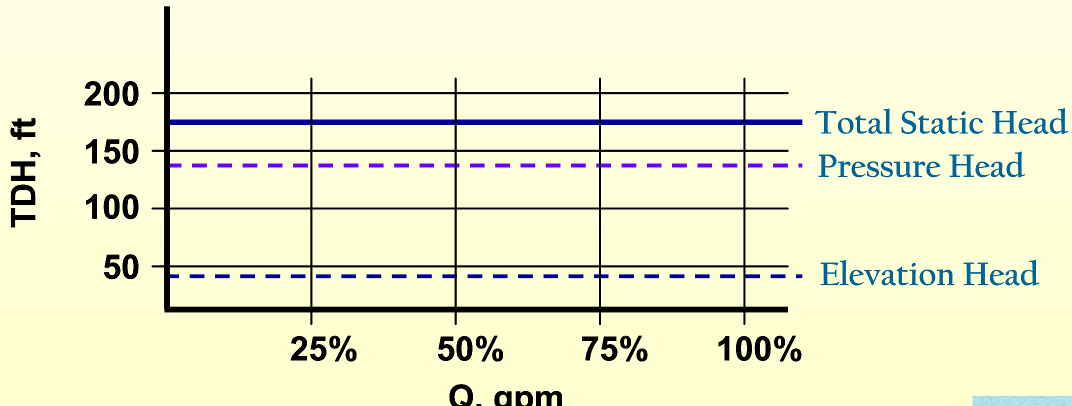
Static Head is constant with Flow and includes:

- Elevation Head, Z
- Pressure Head, P

$$Z = 54 - 14 = 40\ ft$$

$$P = \frac{67 \times 2.31}{0.918} - \frac{14.7 \times 2.31}{1.0} = 135\ ft$$





Friction Head

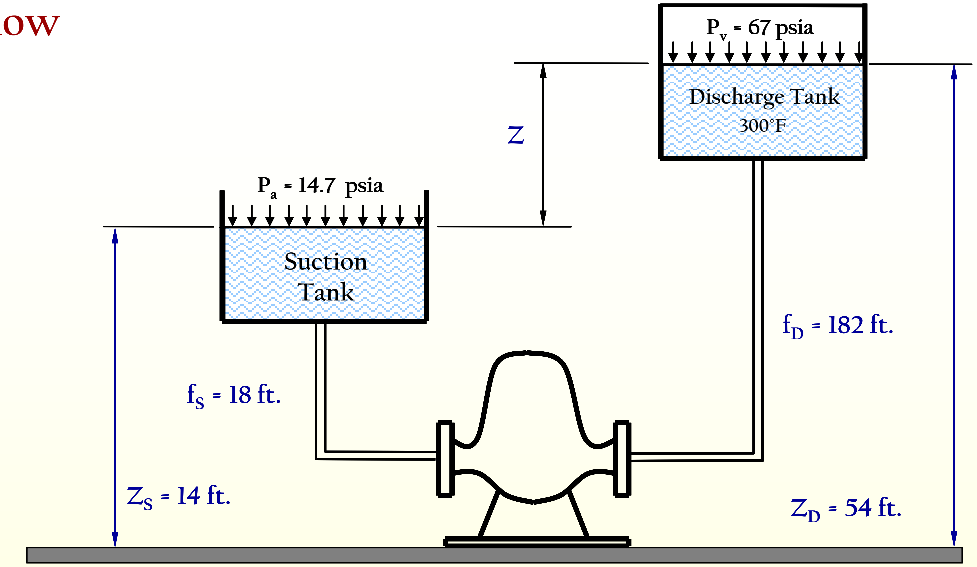
Friction Head varies with Flow

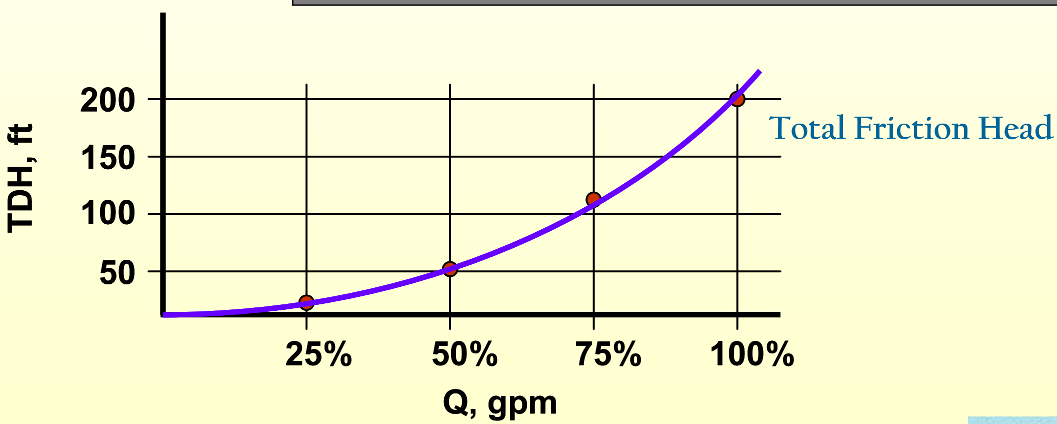
$$h_f = f \times \frac{L}{d} \times \frac{V^2}{2g}$$

$$h_f \propto V^2$$

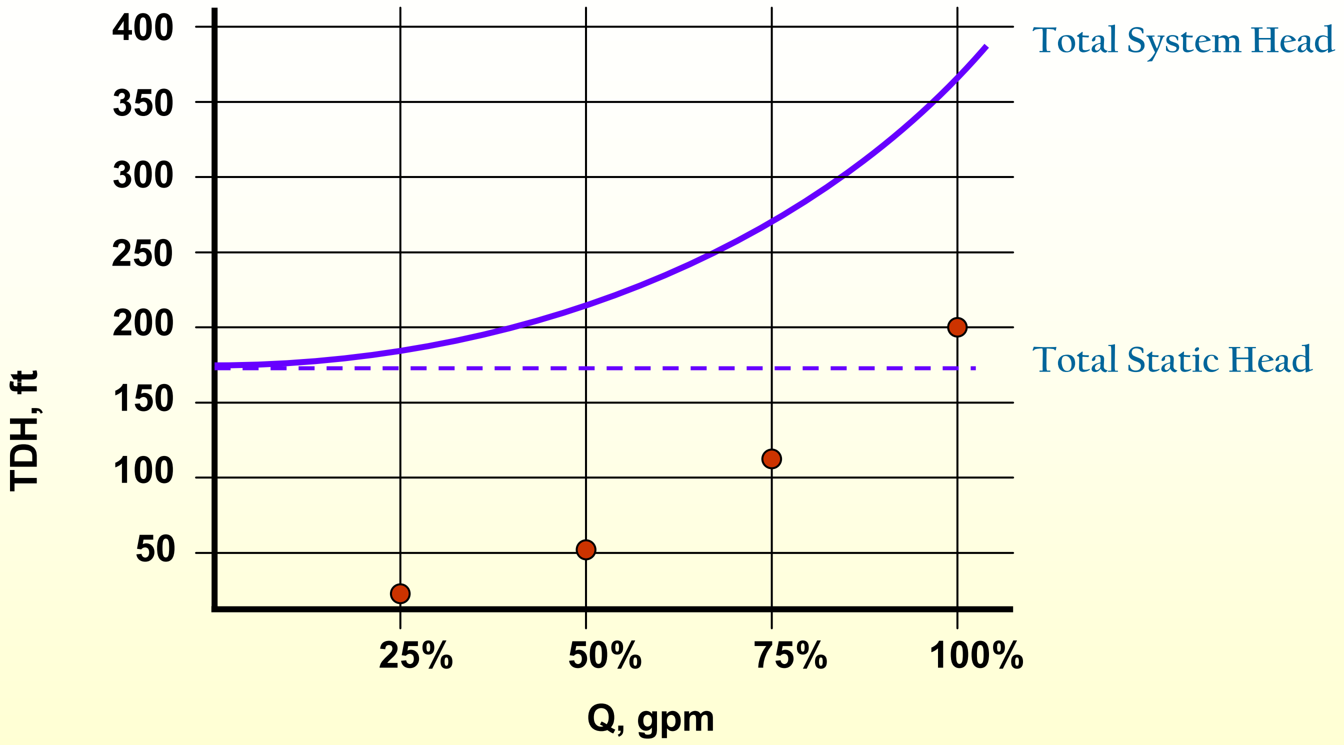
$$V = \frac{Q}{A}$$

$$h_f \propto Q^2$$





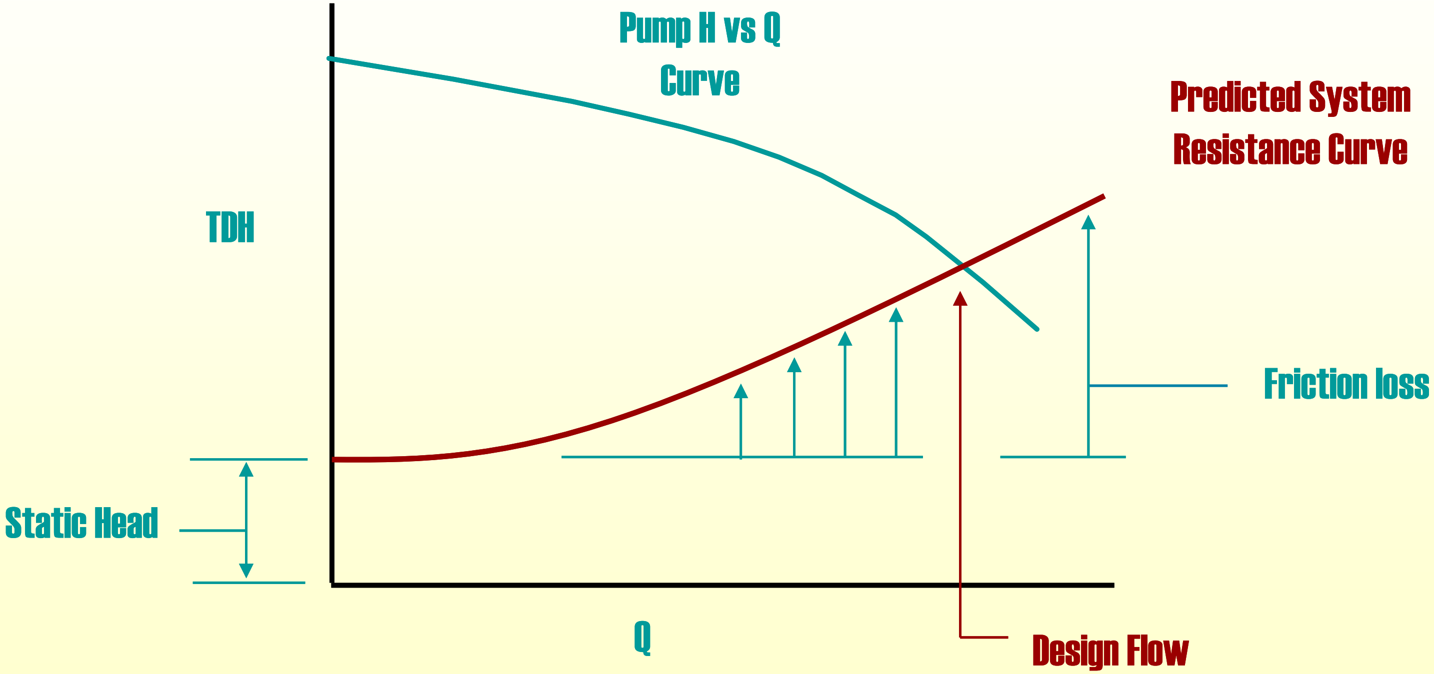
Total System Head



Pump Operation

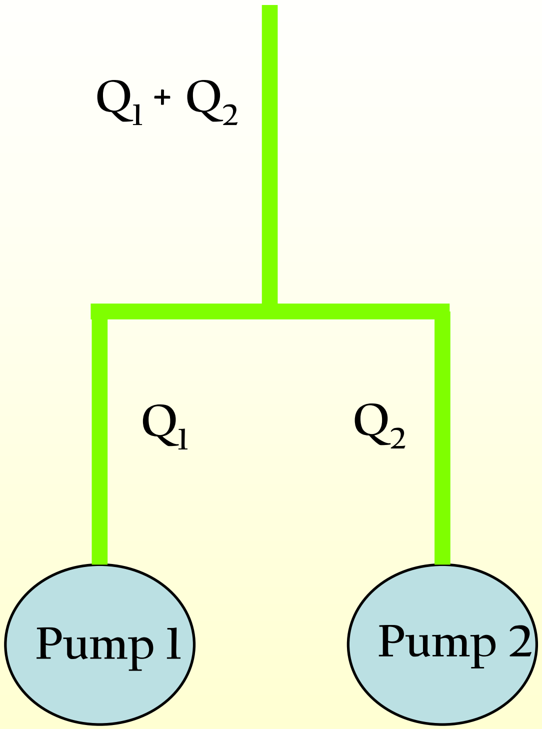
Matching the Pump to the System

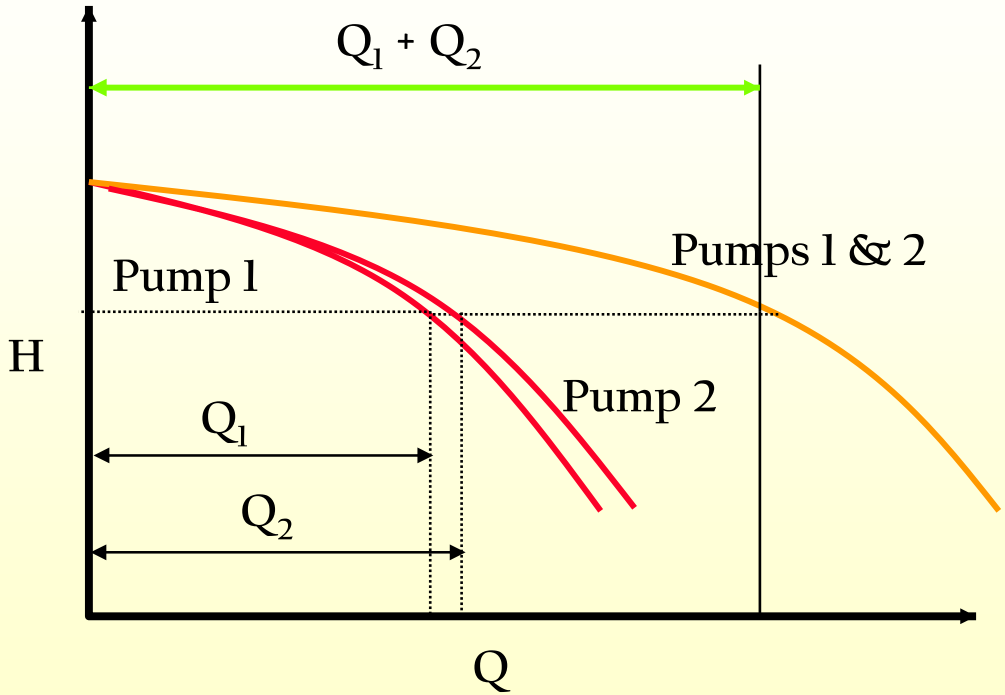
A pump only operates at the intersection of the pump and system curves.



Parallel Pump Operation

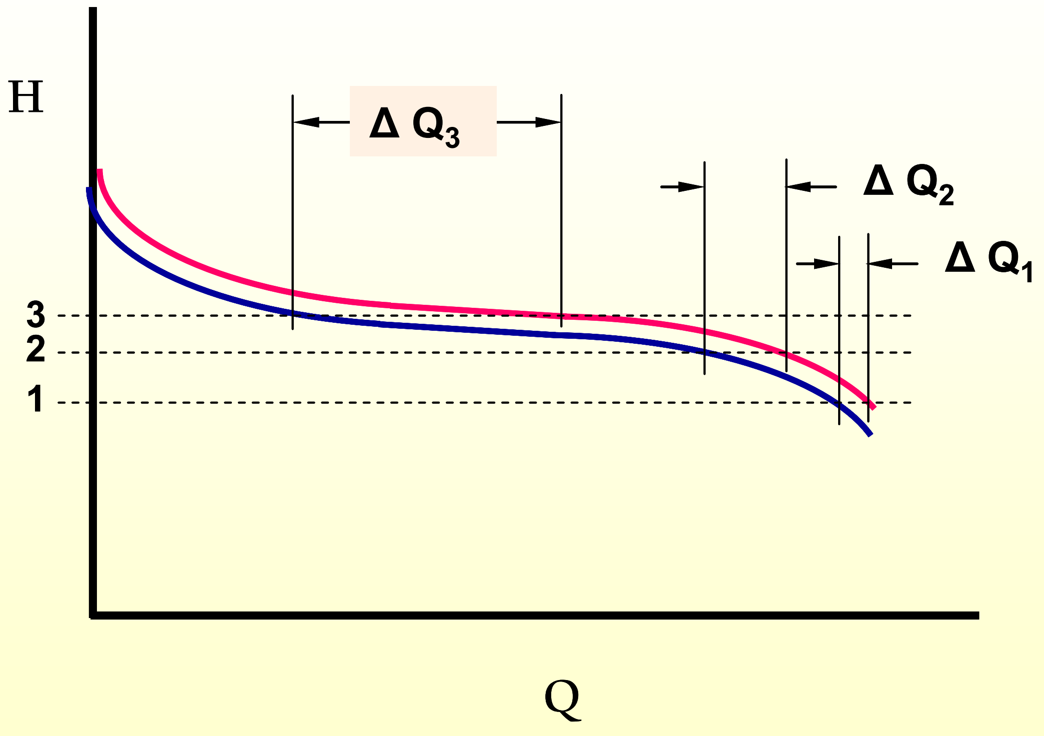
Add Individual Pump Flows at Constant Heads





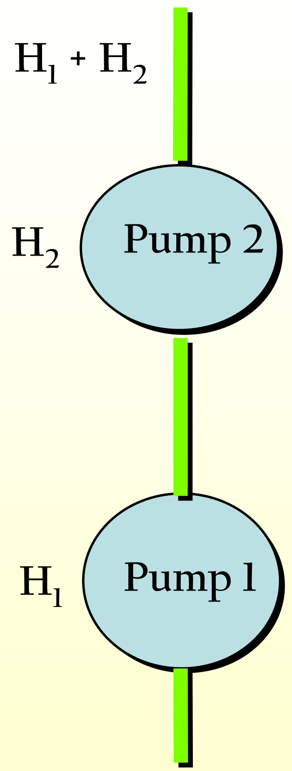
Parallel Pump Operation

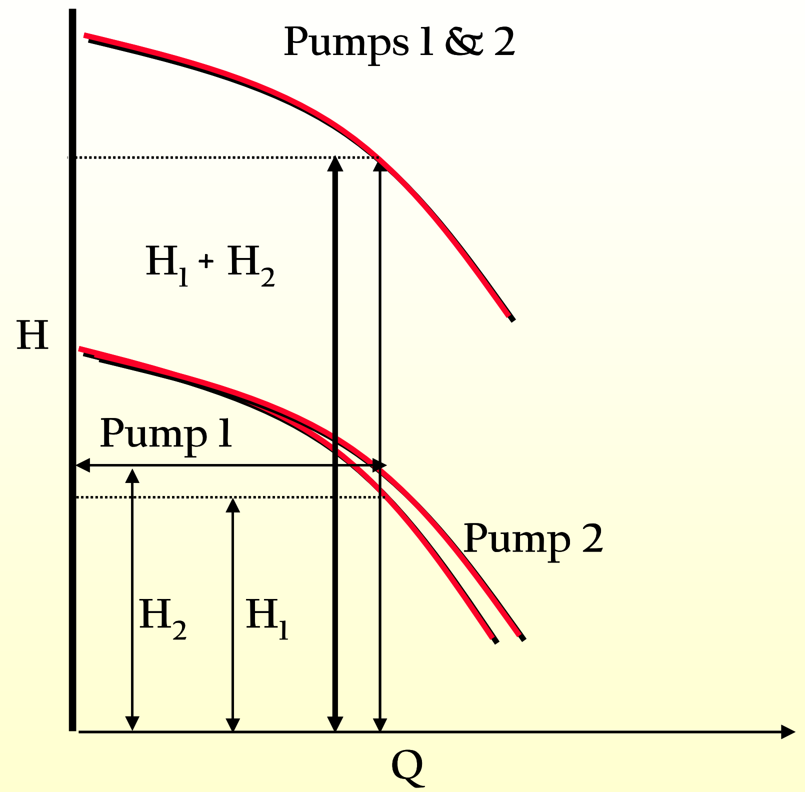
Minimum Flow Operation



Series Pump Operation

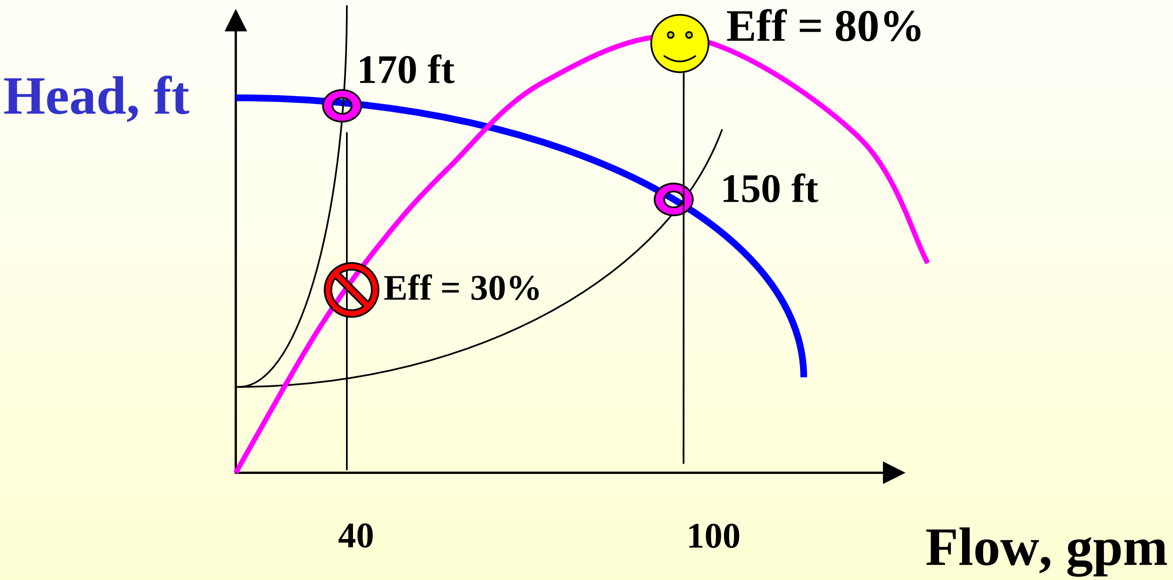
Add Individual Pump Heads at Constant Flows





Pump Operation

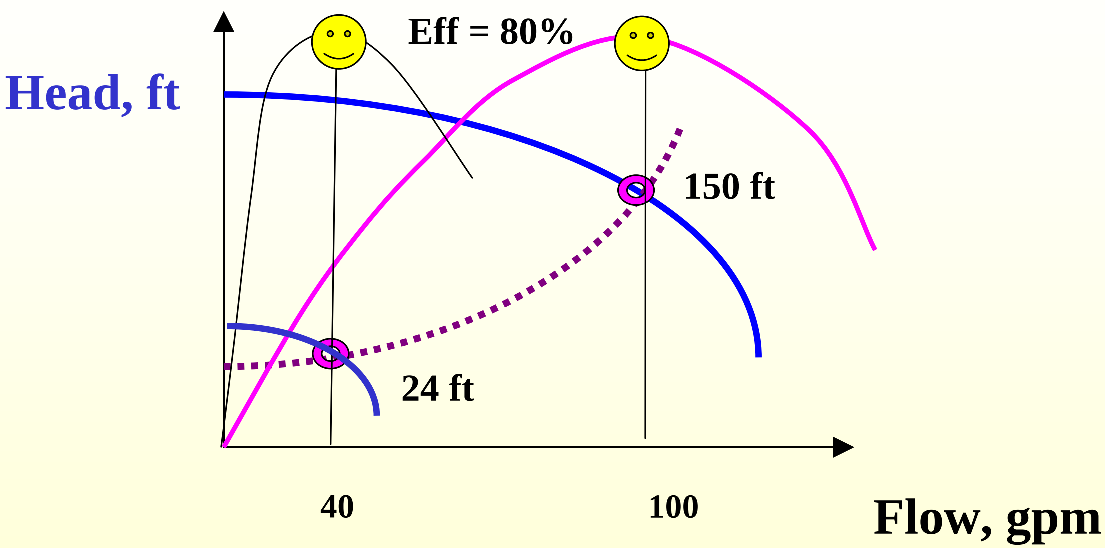
Valve Throttle Flow Control



Throttling is easy, but not efficient

Pump Operation

Varying Speed Flow Control



Speed control is efficient

Note: efficiency actually drops slightly at lower flows (80% would probably become about 77%), but not nearly as significantly as when throttling.

Pump Operation

Let's compare the difference in energy costs between the valve throttling method and speed control, using the earlier example (using SG=1):

a) Throttling:

BHP = Q x H x SG / 3960 / EFF = 40 x 170 x 1.0 / 3960 / 0.30 = 5.7 HP = 4.3 kW

Assuming 24-hour/7-day/52-week operation:

4.3 x 24 x 7 x 52 = 37,303 kW-hr

Let's assume a $0.08/kW energy cost:

37,303 x 0.08 = $2984 per year

Pump Operation

b) Speed Control:

BHP = Q x H x SG / 3960 / EFF = 40 x 24 x 1.0 / 3960 / 0.80= 0.3 HP = 0.2 kW

Assuming 24-hour/7-day/52-week operation:

0.2 x 24 x 7 x 52 = 1,747 kW-hr

Assume the same $0.08/kW energy cost:

1,747 x 0.08 = $140 per year

Pump Operation

The cost difference, i.e. savings:

$2984 – $140 = <u>$2844</u> per year

Just to get a feel for the numbers, a typical 5 hP VFD lists under $1000.

This means that the investment into a VFD would pay for itself within 1000 / 2844 x 12 = 4 months.

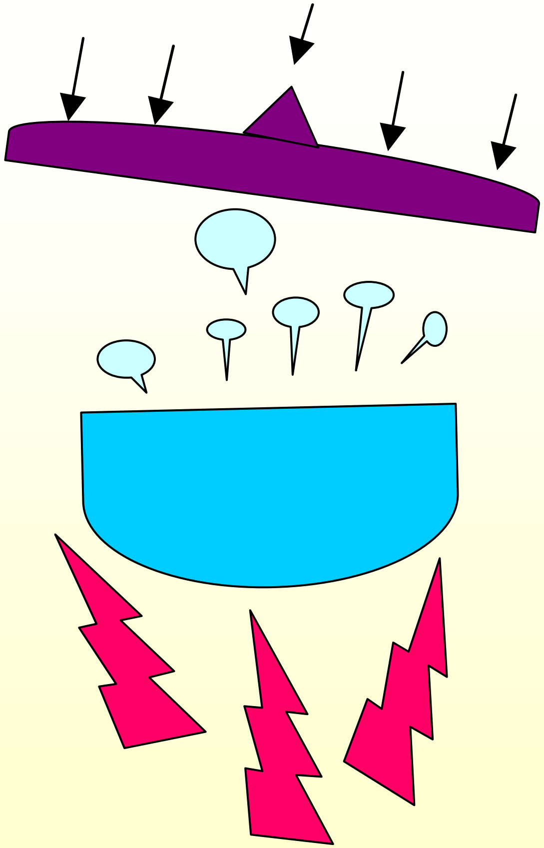
Considering also a possible elimination of the throttling valve, the savings could be even better.

Suction Conditions

- NPSHA
  - Open System (fluid above pump)
  - Open System (fluid below pump)
  - Closed System
  - Closed System (under vacuum)
- NPSHR
- Suction Specific Speed
- Damage Intensity
- NPSH, Condensate Pumps
- Submergence

NPSH

Basically, if there is not enough pressure – liquid boils!



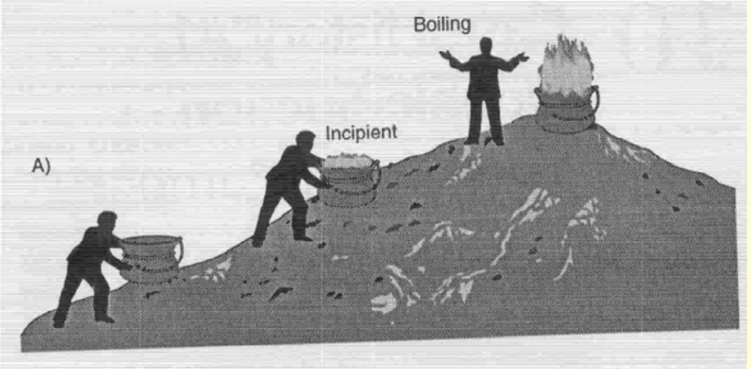
NPSH

In your kitchen, the water boils at 100 ^oC (212 ^{o} F)

– and that is at atmospheric pressure

If pressure drops, the water will boil at lower pressure

On top of high mountain water boils at perhaps 95 ^oC ?



NPSH

If low enough vacuum is achieved - water will boil at room temperature

BOILING IS <u>VAPORIZATION</u> OF LIQUID…

Inside a pump, if pressure gets low enough (below vapor pressure), liquid will boil. The lowest pressure zone is usually at the *suction* area. That is where the first bubbles begin to form..

This initial formation is called *incipient* cavitation…

If pressure drops more – more bubbles emerge…

But the pump keeps pumping…

If too many bubbles – suction gets blocked by them, and no more pumping…

That is where “pump losses its head”…

NPSHA

Net Positive Suction Head Available (NPSHA) is the total suction head (in feet of liquid absolute) at the 1st stage impeller eye less the absolute vapor pressure of the liquid (in feet) being pumped.

$$NPSHA = P_{S,A} - vp + Z_S - f_S$$

where:

P_s = Absolute pressure acting on liquid (in feet)

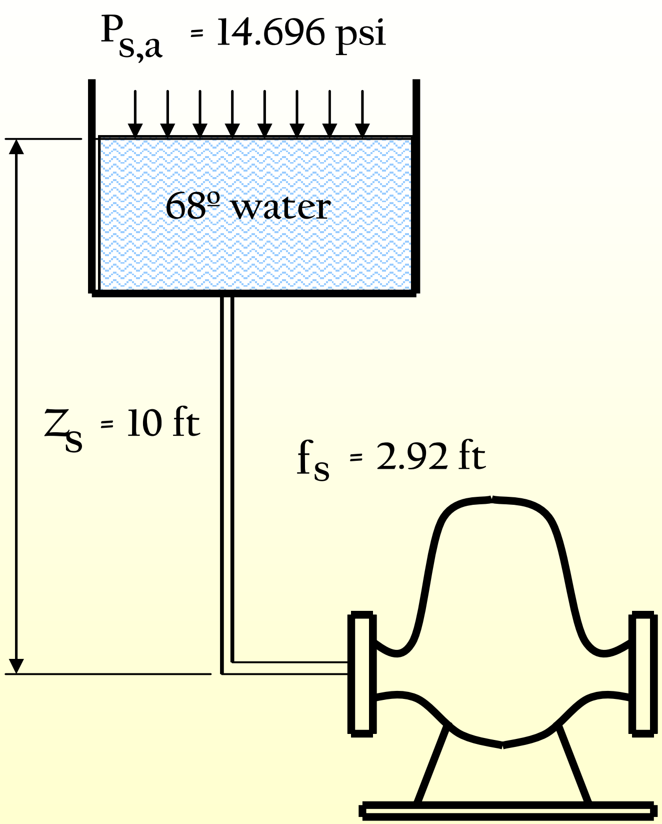
vp = Vapor pressure of the liquid (in feet)

Z_s = Static height from the suction source to the 1st stage impeller eye (in feet).

f_s = All friction losses in suction piping (in feet).

NPSHA

Open System (Fluid above Pump)



$$NPSHA = P_{s,A} - vp + Z_s - f_s$$

Z_s = 10.00 ft.

f_s = 2.92 ft.

P_g = 0

h_s = 10.00 – 2.92 = 7.08 ft.

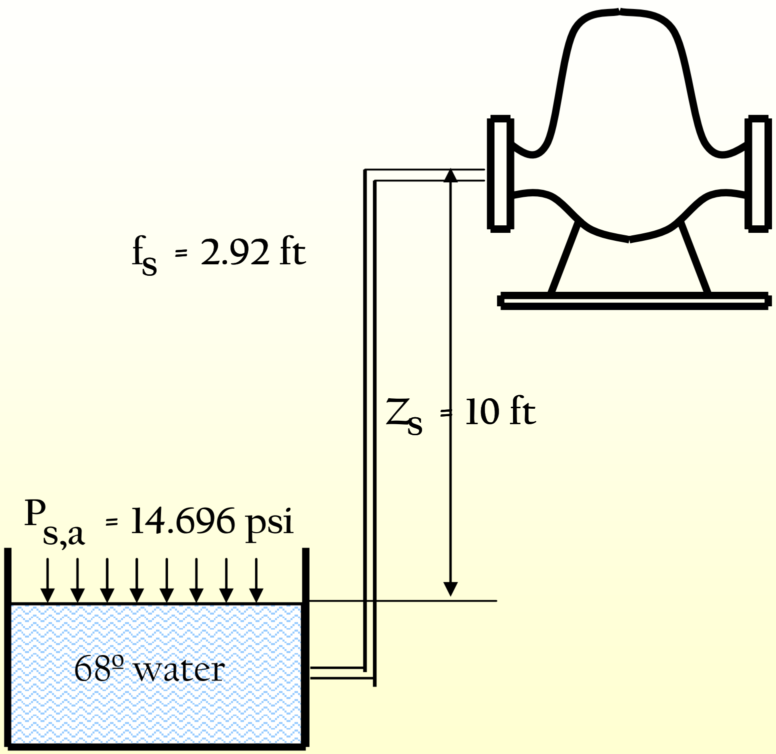
P_a = 14.696 psia = 33.96 ft abs

vp = .339 psia = .783 ft abs

NPSHA = 33.96 - .783 + 10.00 - 2.92
= 40.26 ft.

NPSHA

Open System (Fluid below Pump)



$$NPSHA = P_{s,A} - vp + Z_s - f_s$$

Z_s = 10.00 ft.

f_s = 2.92 ft.

P_g = 0

h_s = -10.00 – 2.92 = -12.92 ft.

P_a = 14.696 psia = 33.96 ft abs

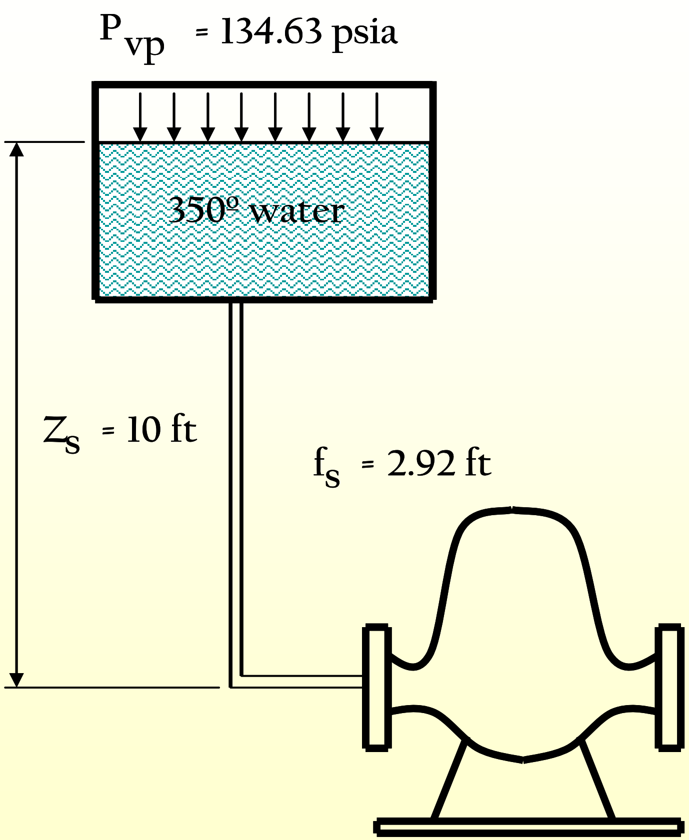
vp = .339 psia = .783 ft abs

NPSHA = 33.96 - .783 - 10.00 - 2.92

= 20.26 ft.

NPSHA

Closed System



$$NPSHA = P_{S,A} - vp + Z_S - f_S$$

Z_s = 10.00 ft

f_s = 2.92 ft

P_g = 119.91 psig = 310.69 ft

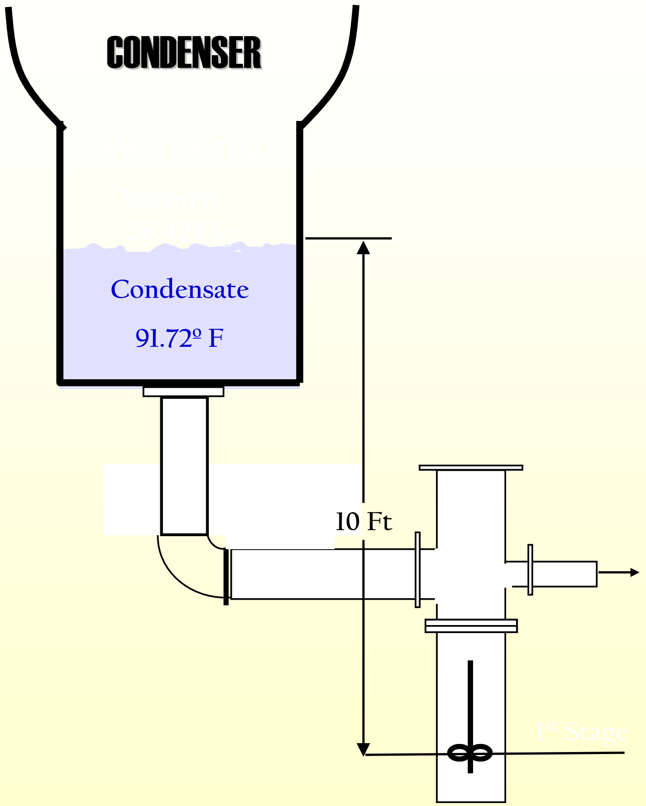
h_s = 310.69 + 10.00 – 2.92 = 317.77 ft

P_a = vp

NPSHA = 10.00 - 2.92 = 7.08 ft

NPSHA

Closed System (under vacuum)



$$NPSHA = P_{s,A} - vp + Z_s - f_s$$

Vacuum = 28.42" Hg = -32.37 ft

Z_s = 10.00 ft

f_s = 2.92 ft

h_s = -32.37 - (10.00 - 2.92) = -25.29 ft

vp = 1.50" Hg = 1.71 ft

NPSHA = 10.00 - 2.92 = 7.08 ft

NPSHA

Calculating, or estimating, suction losses is often a big controversy…

It shouldn't be – but it is

This is because it works very well in theory…but not so well in practice…

Because nobody knows if a dead mouse isn't stuck in the suction pipe.

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Pumping Machinery

NPSHA

This is why it is best to have suction *gage* and read it

Then there is no guesswork

$$H_{suction} = (H_g + Z_g + H_{atm}) + H_{vel}$$

H_g = gage pressure, psig

Z_g = correction for a gage elevation

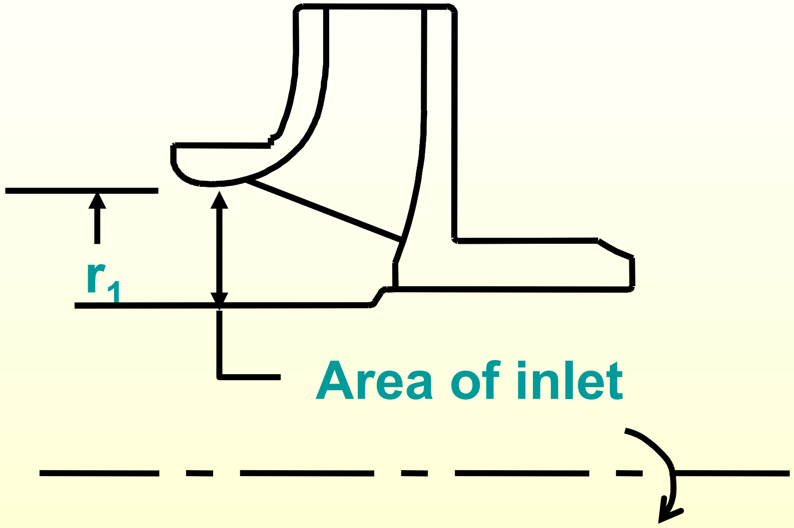
H_{atm} = atmospheric pressure (34 ft)
Together, $H_g + H_{atm}$ give us total static head in absolute.
(For example 5 psig is 5+14.7 = 19.7 psia)

H_{vel} = velocity (dynamic) head $V^2/2g$

NPSHR

The Net Positive Suction Head Required (NPSHR) is the suction capability of an impeller and is determined by:

- Inlet Diameter
- Rotating Speed
- Inlet Blade Angle
- Suction Inlet Approach



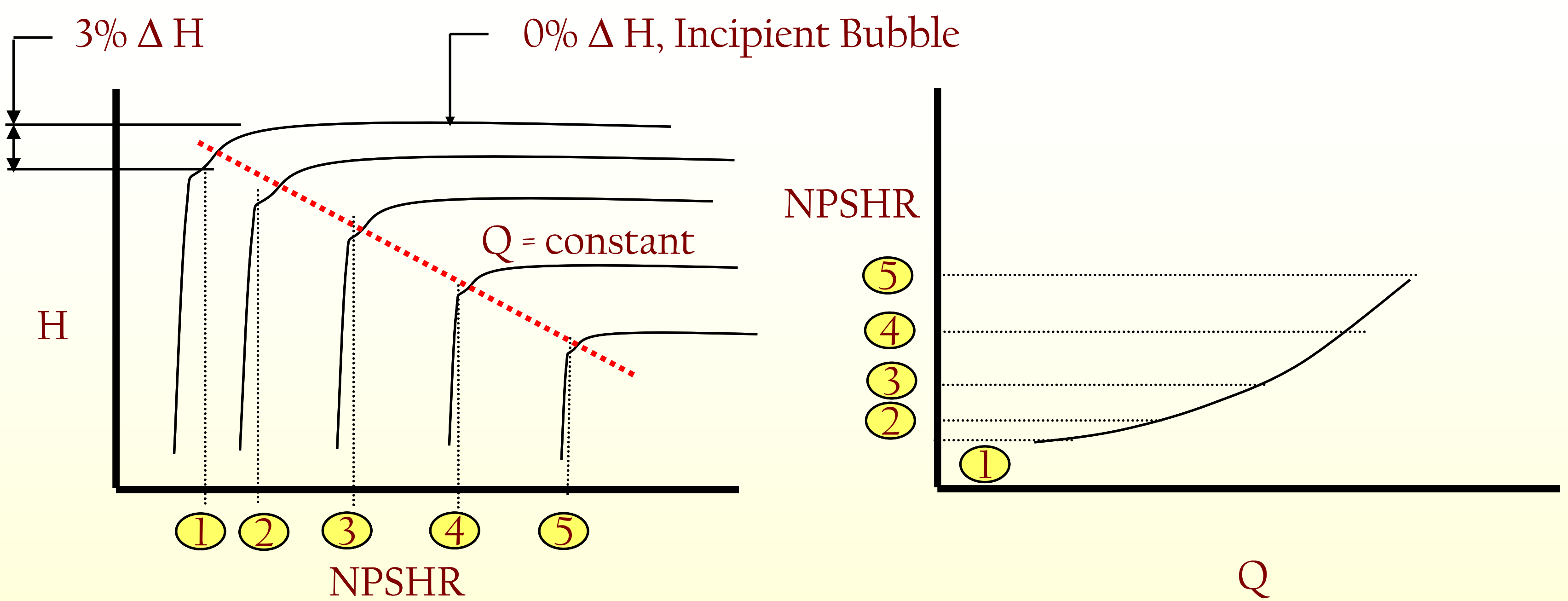
$$NPSHR = \left(U_1^2 / 2g\right) \times \left(1.485\Phi^2 + .085\right)$$

where:

$U_1 = r_1\omega$

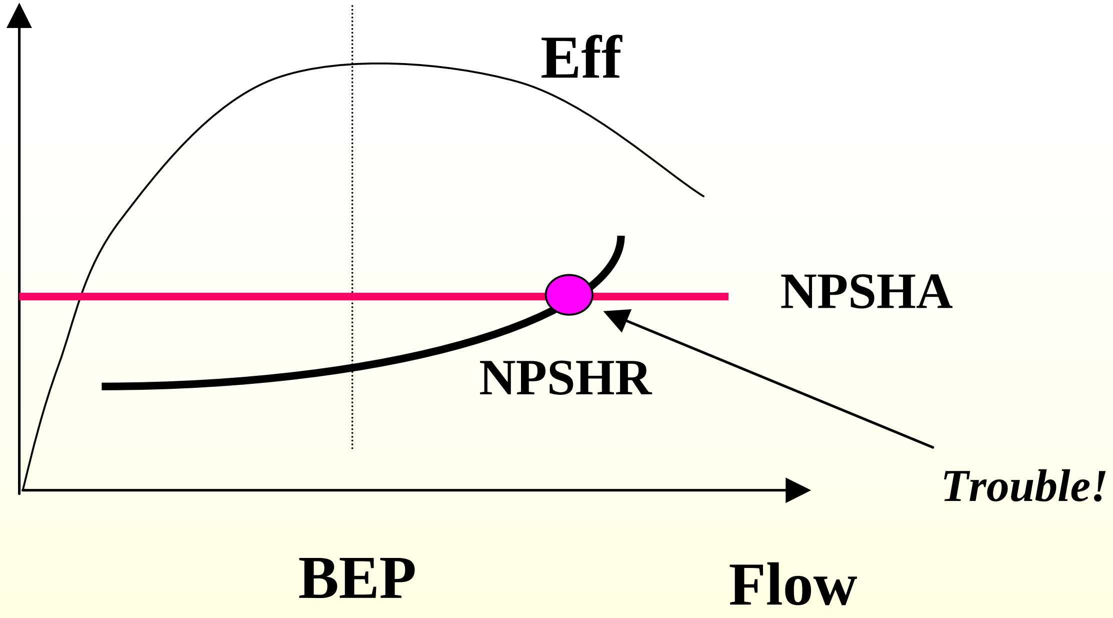
$\varnothing = (Q/AREA)/U_1$

NPSHR



$$NPSHR_{0\%} \approx 1.5 \times NPSHR_{3\%}$$

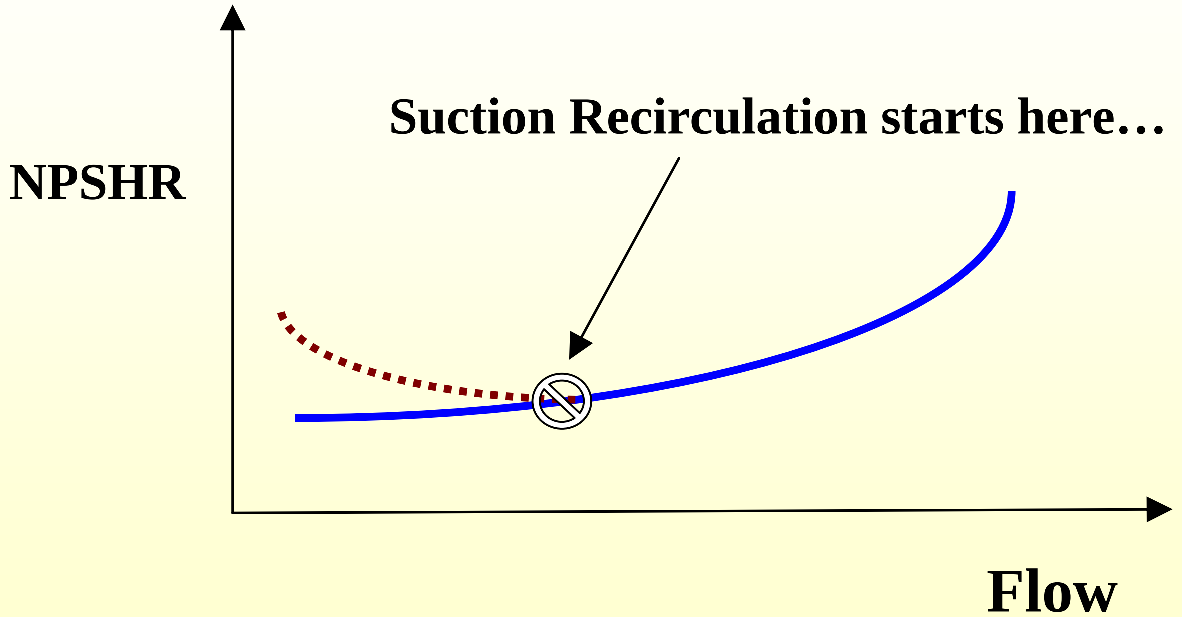
NPSHR



Velocities are higher at higher flow – this lowers static pressure, requiring more pressure to counteract that As a result, NPSHR rises at higher flow..

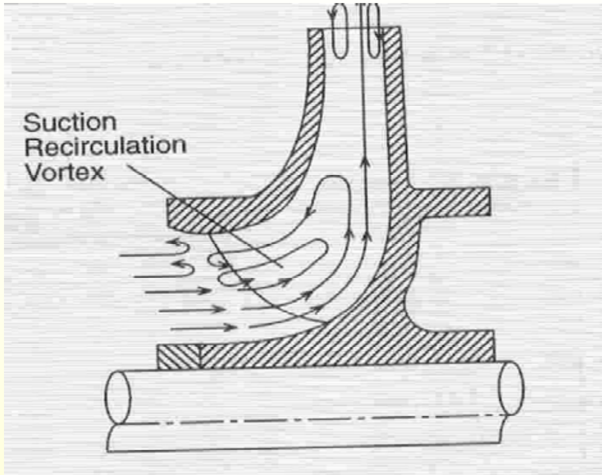
NPSHR

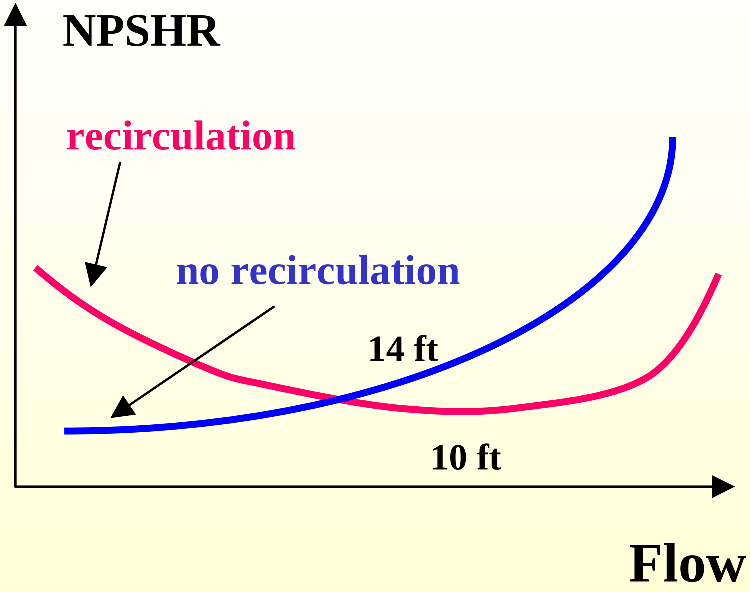
Actually, at low flow bad things begin to happen…



NPSHR

IMPELLER EYE SIZE EFFECT





Smaller eye helps suppress suction recirculation, although with some sacrifice of NPSHR at BEP

Suction Specific Speed

Suction specific speed (S_s) is a dimensionless parameter that pump engineers use to define impeller suction inlet geometry. The higher the suction specific speed, the larger the impeller eye, and the higher susceptibility to fluid separation at off-peak operation.

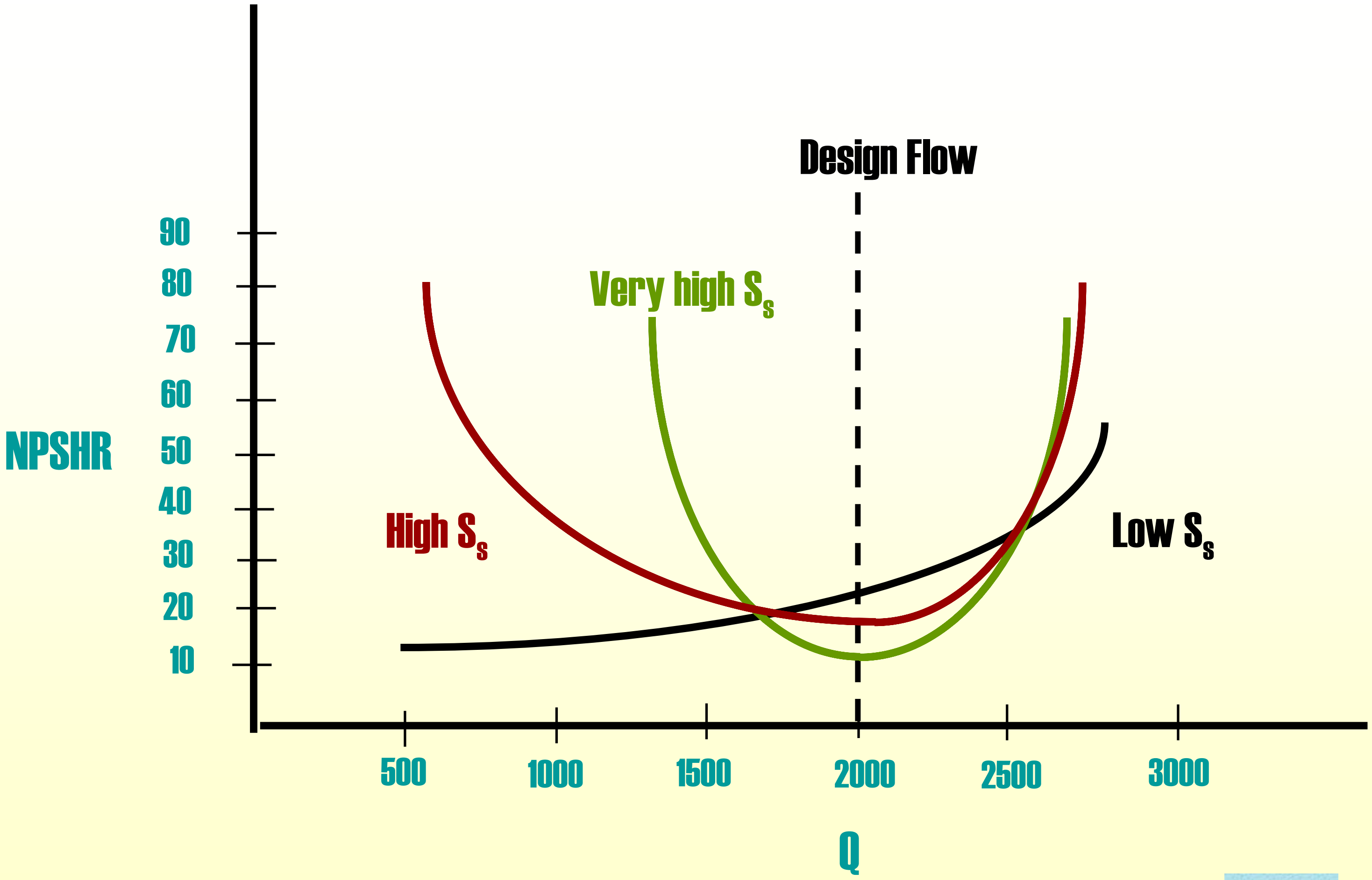
$$N_{ss} = S_s = \frac{rpm \times \sqrt{Q_{eye}}}{NPSHR^{0.75}}$$

where:

Q = Suction flow per eye of the 1st stage impeller, @ BEP in gpm

(for double suction impellers, Q = 1/2 the total suction flow)

Suction Specific Speed



Suction Specific Speed

Suggested Limits

Hydrocarbon Applications

- $S_s \leq$ 11,000 based on $NPSHR_{3\%\ \Delta H}$
- $S_s \leq$ 9,100 based on $NPSHR_{1\%\ \Delta H}$

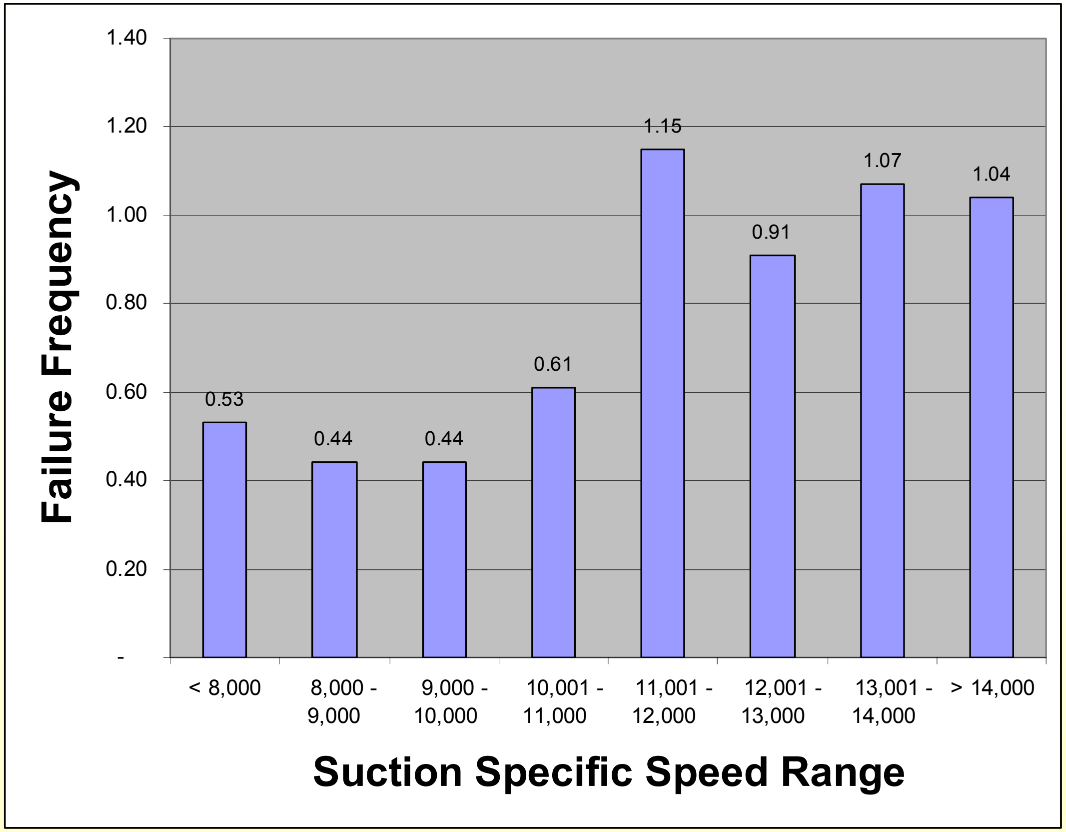
Water Applications

- $S_s \leq$ 9,500 based on $NPSHR_{3\%\ \Delta H}$
- $S_s \leq$ 7,800 based on $NPSHR_{1\%\ \Delta H}$

$$S_s = \frac{rpm \times \sqrt{Q_{eye}}}{NPSHR^{0.75}} \qquad NPSHR = \left\{ \frac{rpm \times \sqrt{Q_{eye}}}{S_s} \right\}^{4/3}$$

Suction Specific Speed

Refinery Experience (J. L. Hallam)

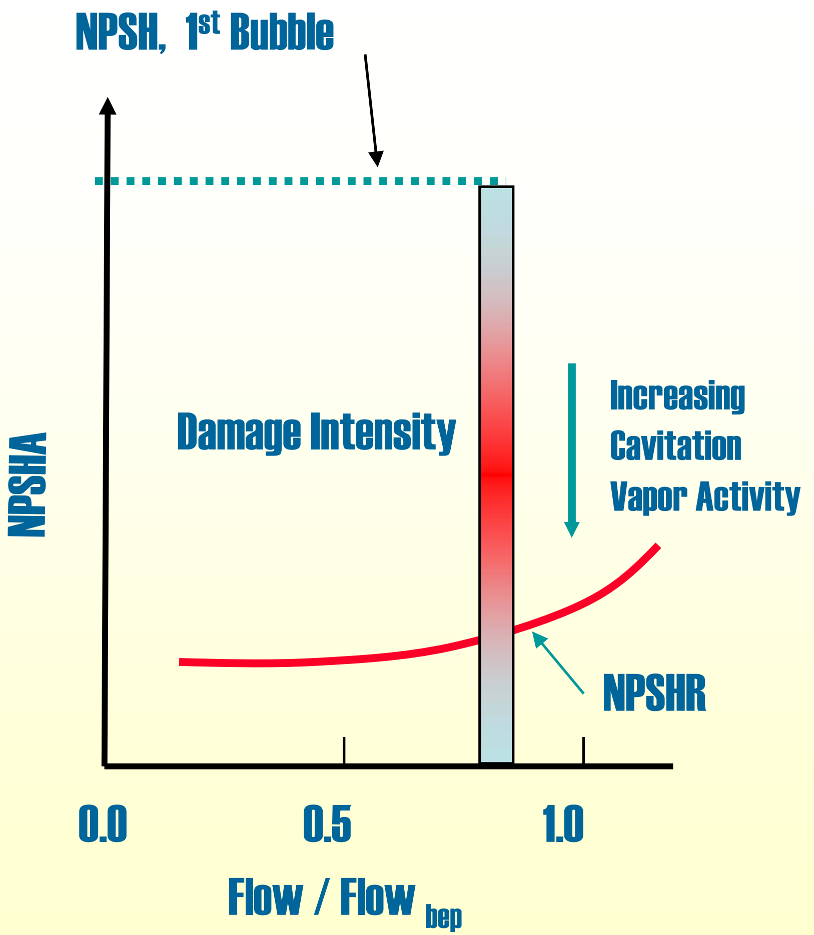


Suction Specific Speed

- **<u>Good ways to fight suction problems:</u>**
  - **Make sure a pump is as close to suction source as possible**
  - **For double-suction pumps, suction valve stem should be oriented for symmetrical feed flow**
  - **Smaller eye impeller (lower suction specific speed) can help**
  - **Inlet sump design – critical: vortexing and air entrapment could be a nightmare**
  - **Metallurgy – 316ss stainless work-hardens, while iron does not very long**

Damage Intensity

- The performance NPSHR of a pump is determined by traditional 3% head reduction testing
- Significant amounts of vapor present at NPSH higher than NPSHR
- Maximum cavitation damage can occur at NPSHA = 2 x NPSHR



Cavitation Damage

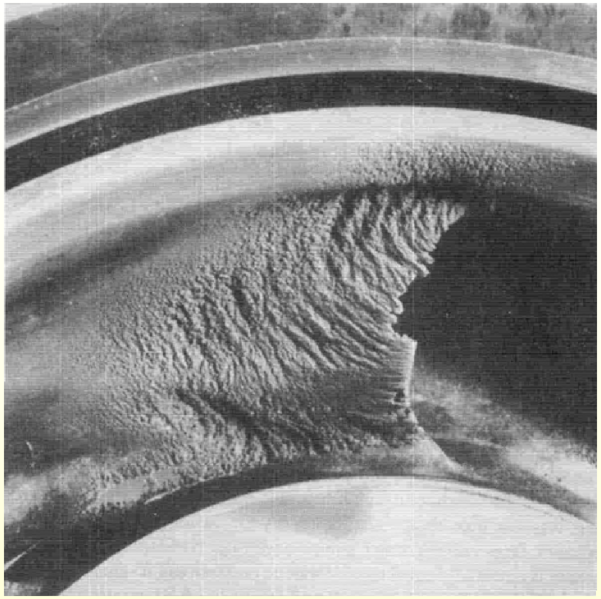
CAVITATION DAMAGE

Too many candies makes kids hyper.

But it also ruins their teeth.

Cavitation Damage

Impeller inlet – blades cavitation on a suction side



As bubbles flow from low pressure to higher, they implode against metal surfaces. These micro-hammer-like impacts erode the material, creating cavities – thus "cavitation"

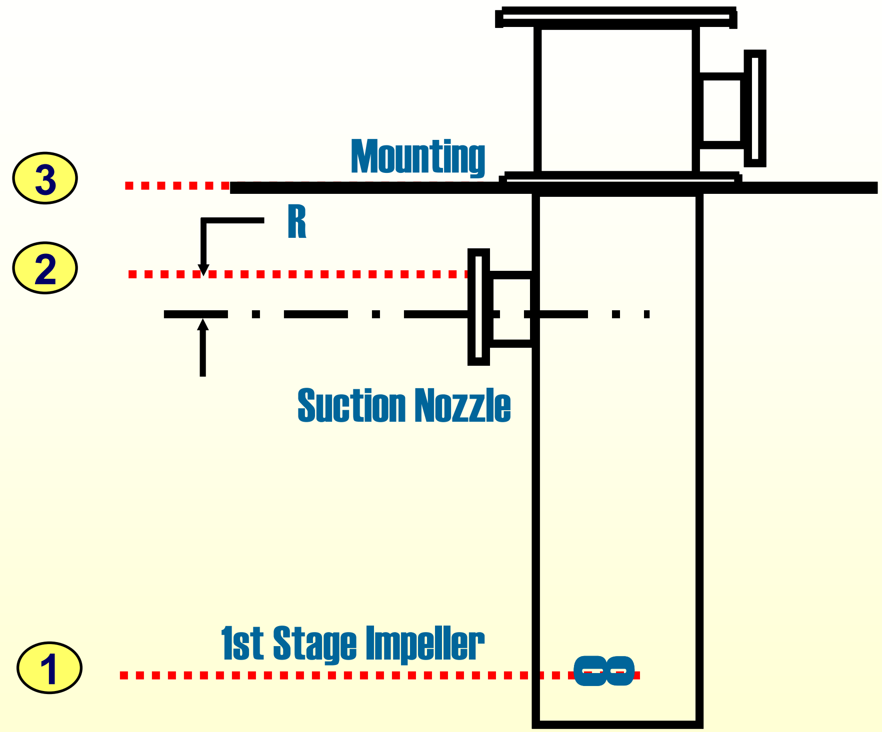
NPSH, Condensate Pumps

1 NPSHA (@ 1st stage) › NPHSR at 1 pump runout

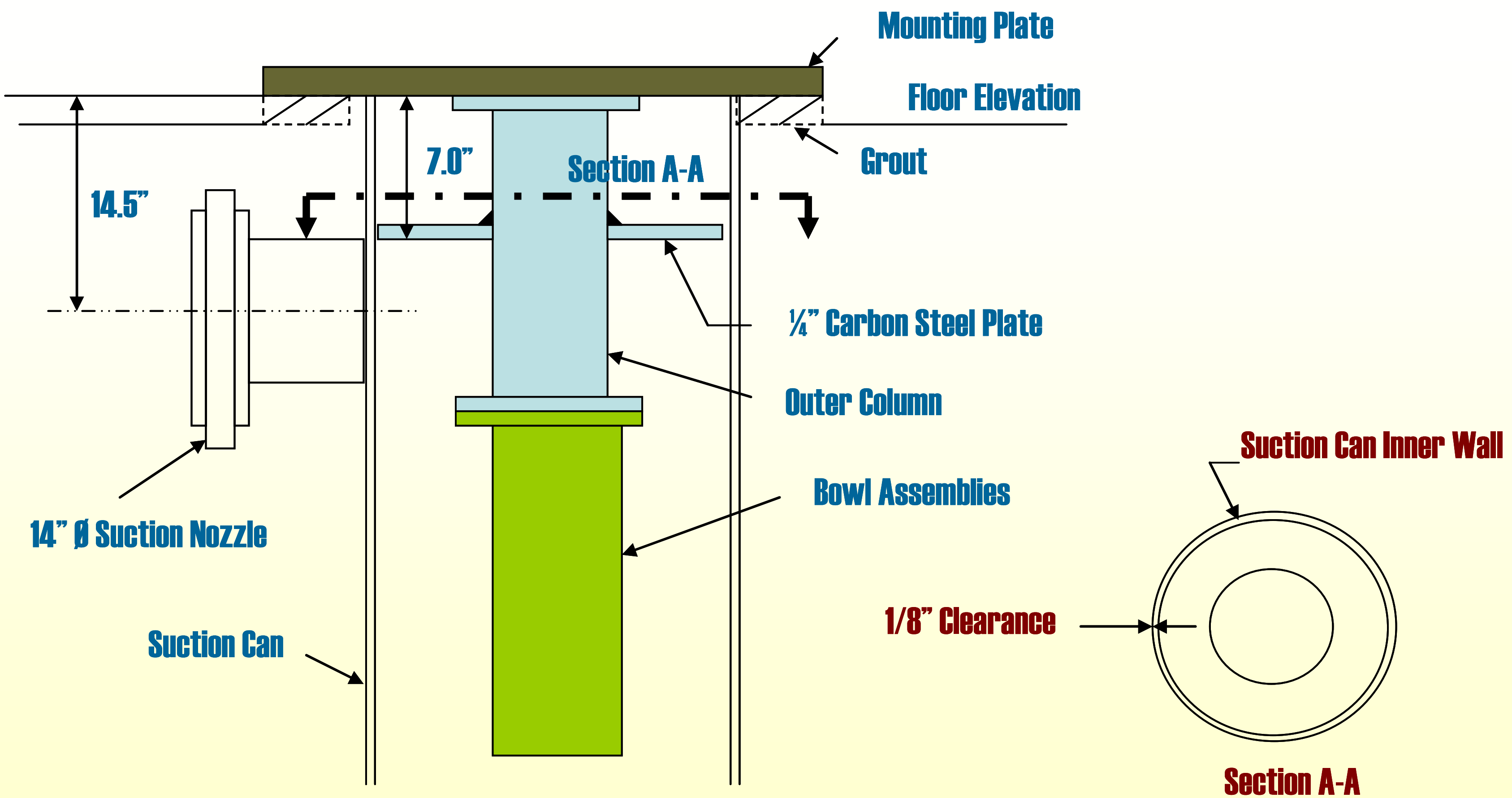
2 NPSHA (@ ℄ Suction) › R + 2.0 $V^2/2g$

3 NPSHA (@ Mounting) › 0

(distance from the minimum hotwell level to the floor must be greater than all friction losses in suction pipe)

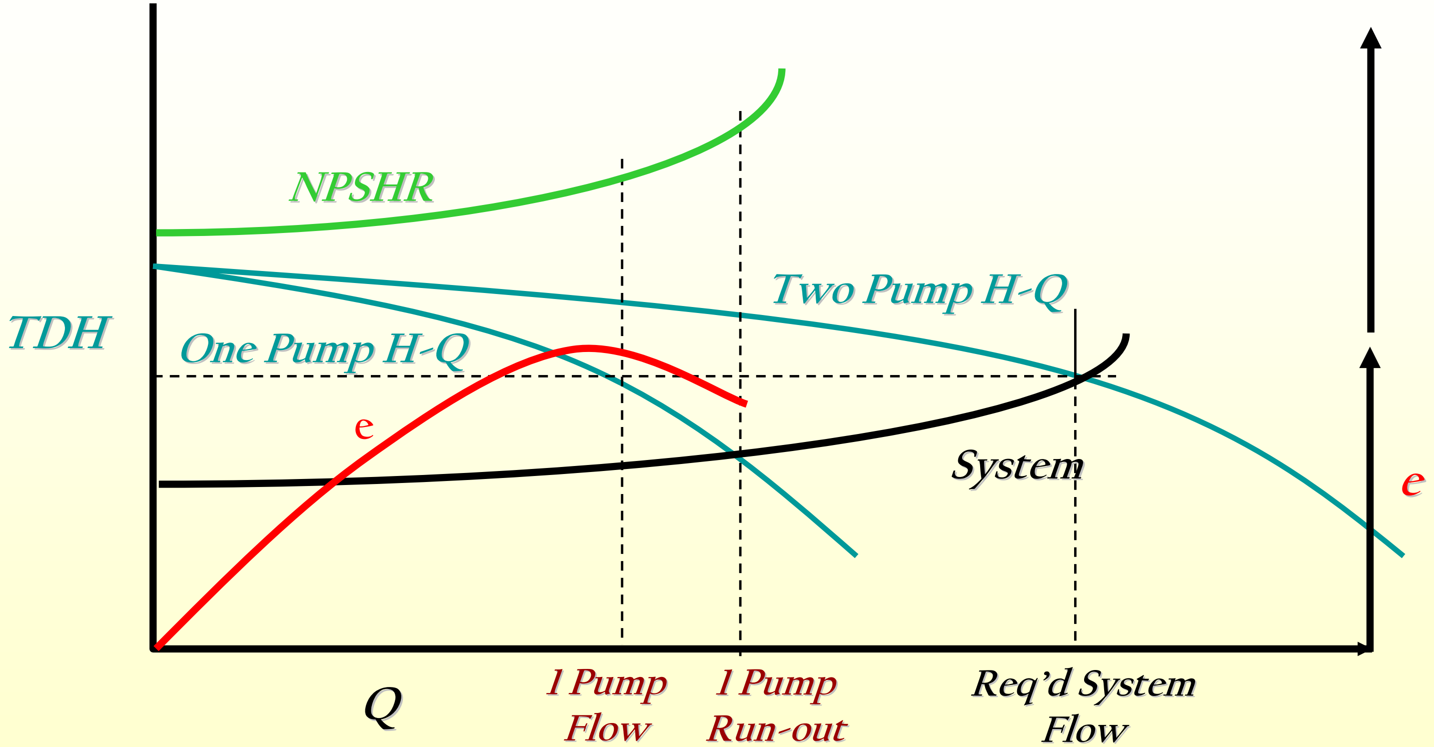


NPSH, Condensate Pumps



NPSH, Condensate Pumps

Example



NPSH, Condensate Pumps

Example

Given the data shown:

- What is the design flow per pump for 100% operation?
- What is the required pump setting?
- What is the required motor design hp?
- Should this be an above or below ground suction design?

Hotwell Elevation (ft)	
Normal	502
Minimum	500
Mounting Elevation (ft)	492
Friction Losses (ft)	
Without Strainer	5
With Strainer	9
System Flow Requirements (gpm)	
VWO	9500
100% Load	9000
80% Load	7200
One Pump Runout Flow (gpm)	6000

Flow (gpm)	TDH (ft)	Eff. (%)	BHP bhp	npshr (ft)
2000	920	60	774	11
3000	875	73	908	12
4000	780	81	972	14
5000	650	82	972	14
6000	480	71	1024	24

NPSH, Condensate Pumps

Example

- What is the design flow per pump for 100% operation?

  4500 gpm (= 9000/2)

- What is the required pump setting?

  467 ft (= 500 - 9 - 24)

- What is the required motor design hp?

  1250 bhp

- Should this be an above or below ground suction design?

  No (npsha = -1' @ mtg.)

Hotwell Elevation (ft)	
Normal	502
Minimum	500
Mounting Elevation (ft)	492
Friction Losses (ft)	
Without Strainer	5
With Strainer	9
System Flow Requirements (gpm)	
VWO	9500
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6000	480	71	1024	24

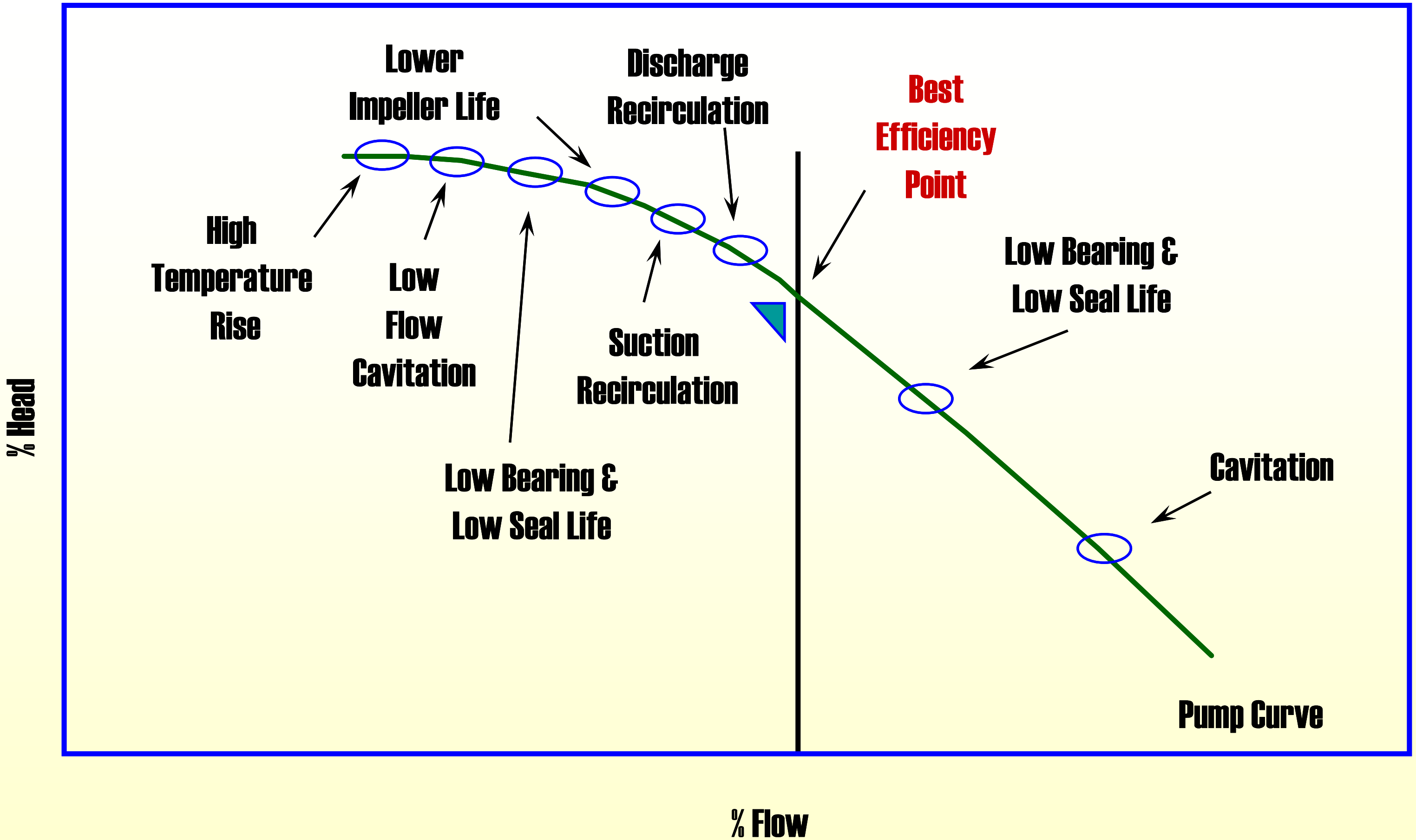
Failure Mechanisms

- Generic Issues
- Hydraulic Instability
  - Inlet Separation
  - Discharge Recirculation
- High Impact Loading
- Acoustic Resonance
- Premature Opening of Ring Clearances

Failure Mechanisms

- Poor Pump Performance
- Hot Radial Bearings
- Hot Thrust Bearings
- Black Oil
- Casing Leakage
- Seal Leakage

Generic Issues



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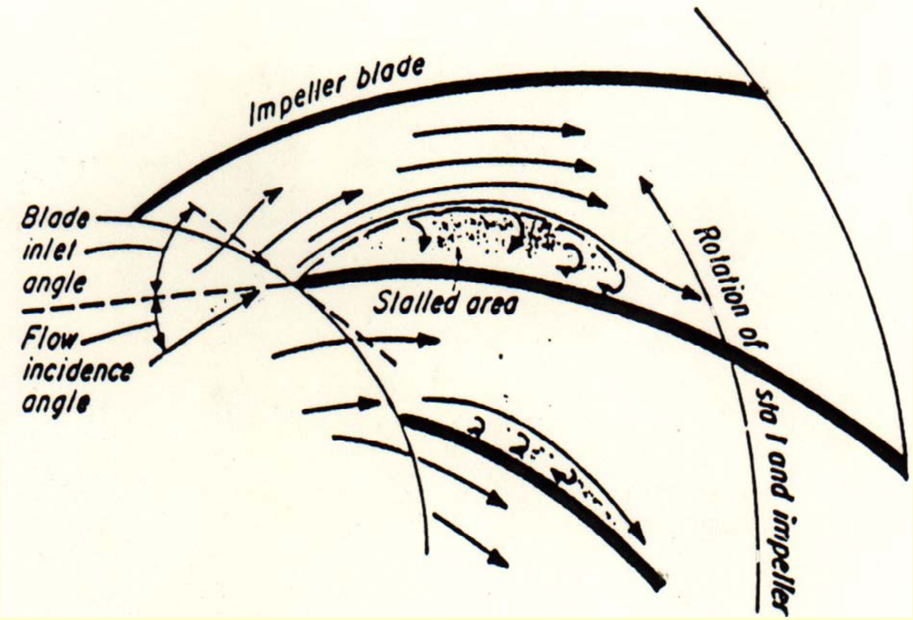
Inlet Separation

Causes

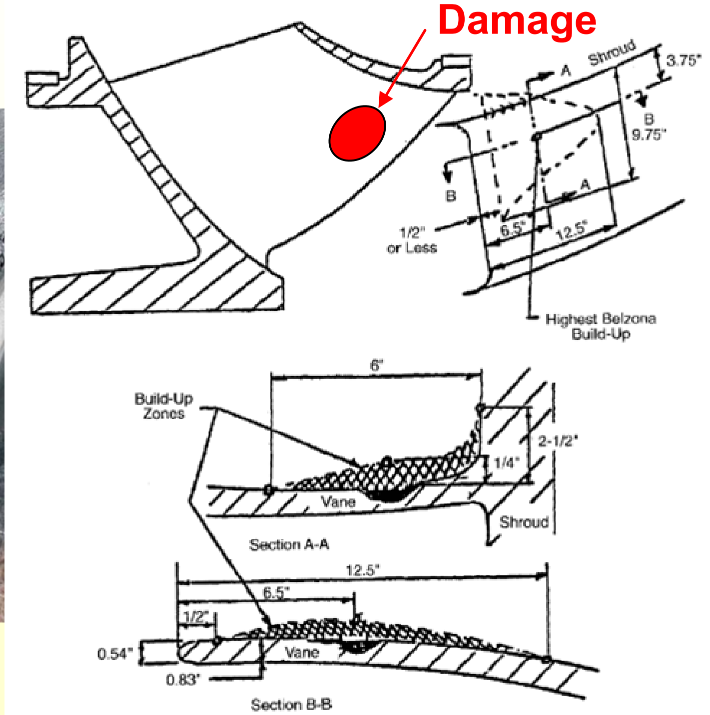
- Fluid stall occurs when the incidence angle – difference between flow angle and impeller inlet angle—increases above a specific critical value. Stalled area, which eventually washes out, reforms as rotation continues.
- Backflow” interferes with inlet flow, creating localized pressure drops, that can drop below fluid vapor pressure causing separation/cavitation
- Large areas “promote” fluid swirl



Inlet Separation

Damaging Effects

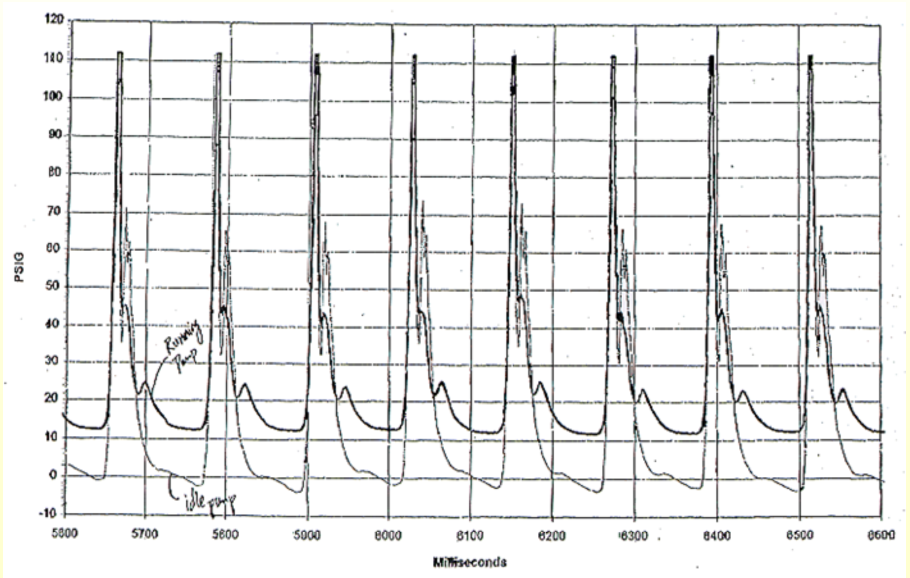
- Impeller inlet vane erosion



Inlet Separation

Damaging Effects

- Pump surging (4-10 Hz)
- Excitation of suction piping



Inlet Separation

Solutions

- Vane Damage:

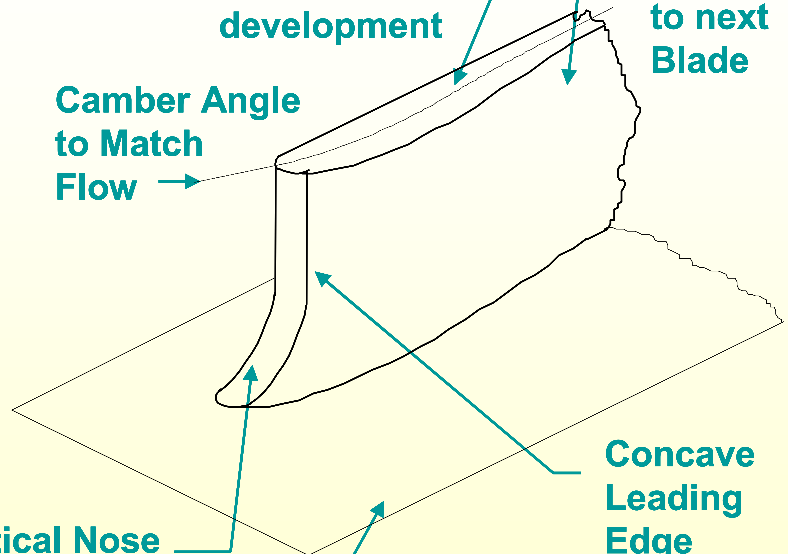
  Bias-wedge (anti-stall hump) inlet vane re-contouring

- Fillet Damage:

  Front hook

- Fluid Swirl:

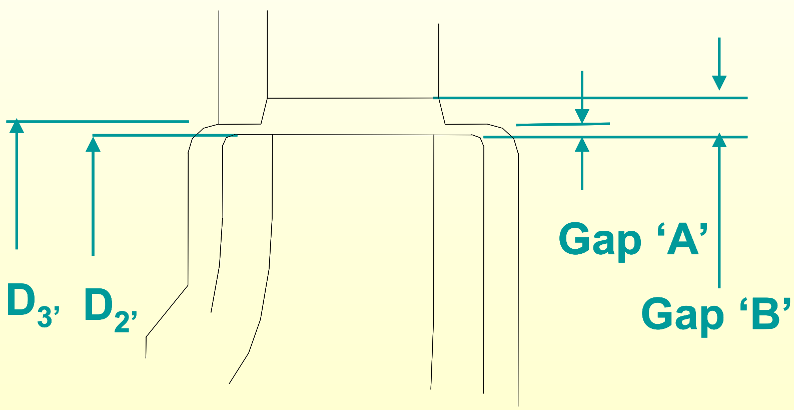
  Backflow catcher

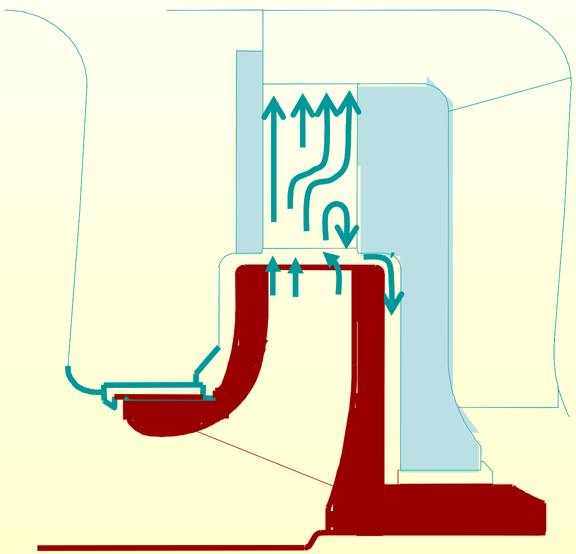


Discharge Recirculation

Causes

- Fluid stall occurs when the incidence angle – difference between flow angle and diffuser or volute inlet angle—increases above a specific critical value. Stalled area, which eventually washes out, reforms as rotation continues.
- Large areas "promote" fluid swirl





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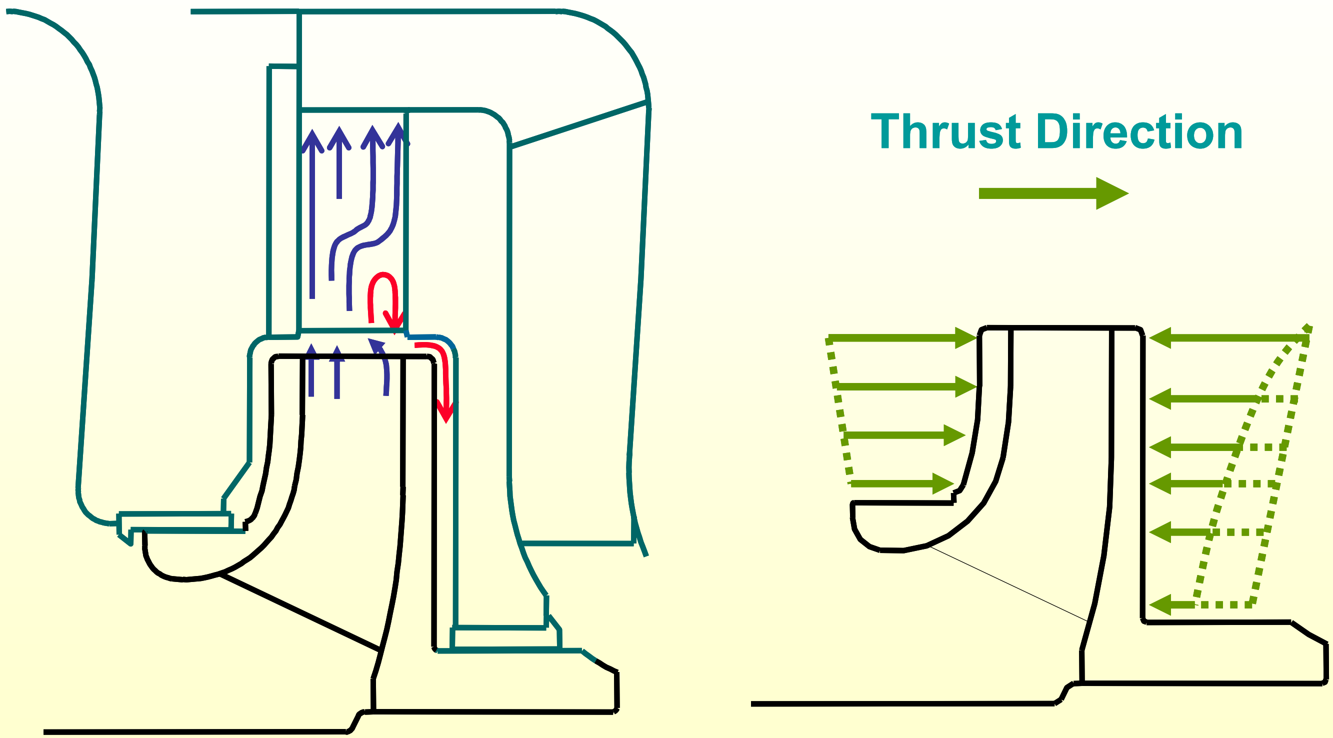
Pumping Machinery

Discharge Recirculation

Damaging Effects

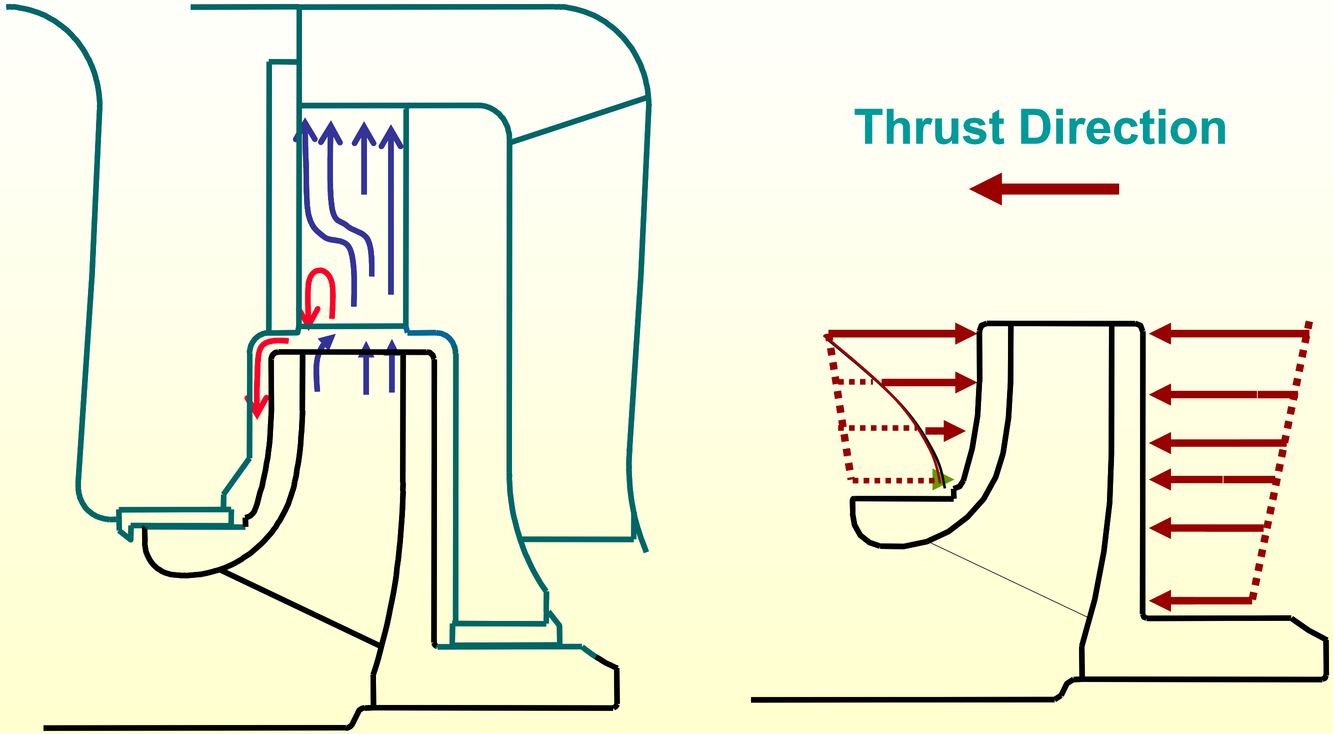
- Axial shuttling



Discharge Recirculation

Damaging Effects

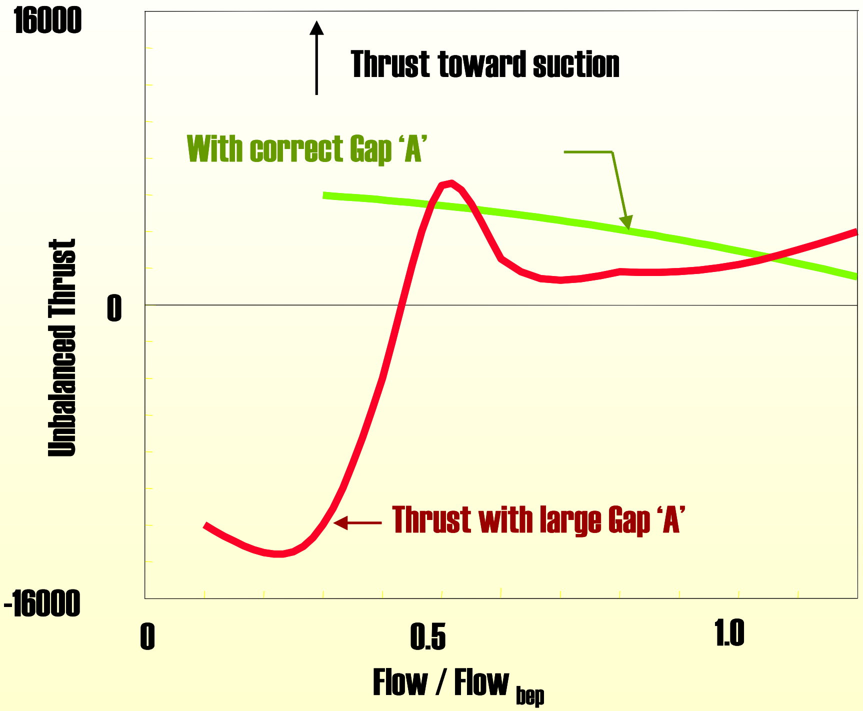
- Axial shuttling



Discharge Recirculation

Damaging Effects

- Axial shuttling



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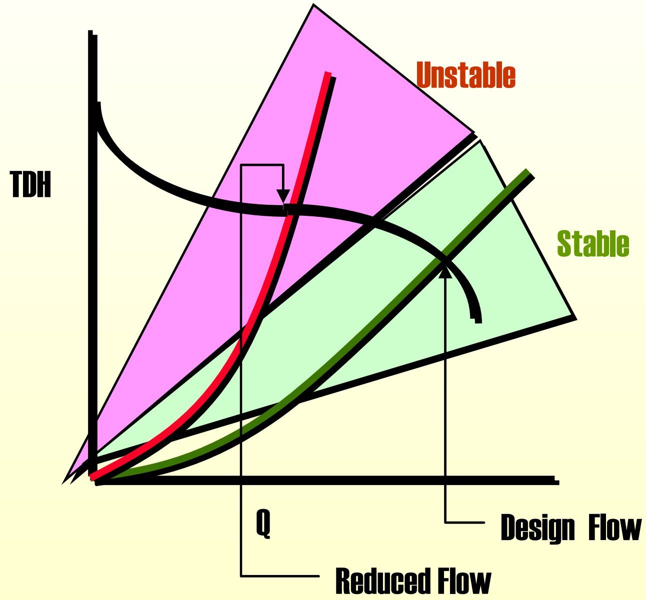
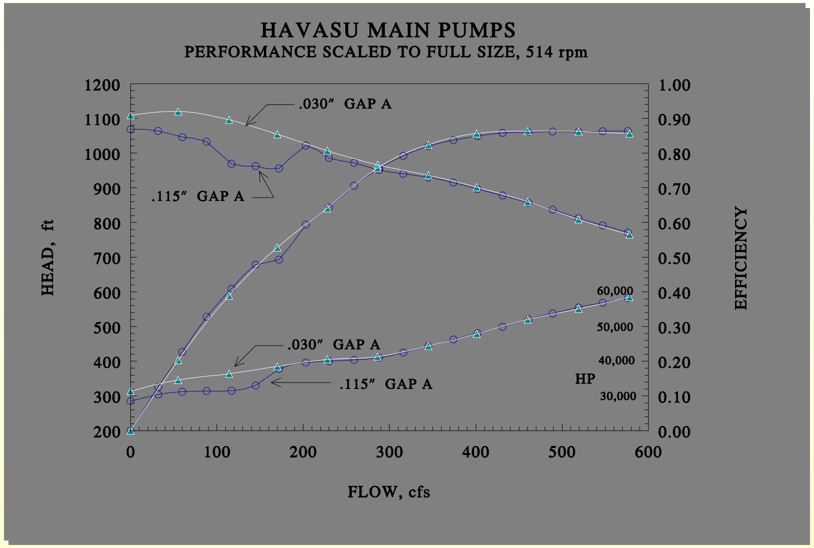
Pumping Machinery

Discharge Recirculation

Damaging Effects

- Poor pump paralleling capability or rotor hunting due to a flattening of the head-capacity curve at off-peak operation

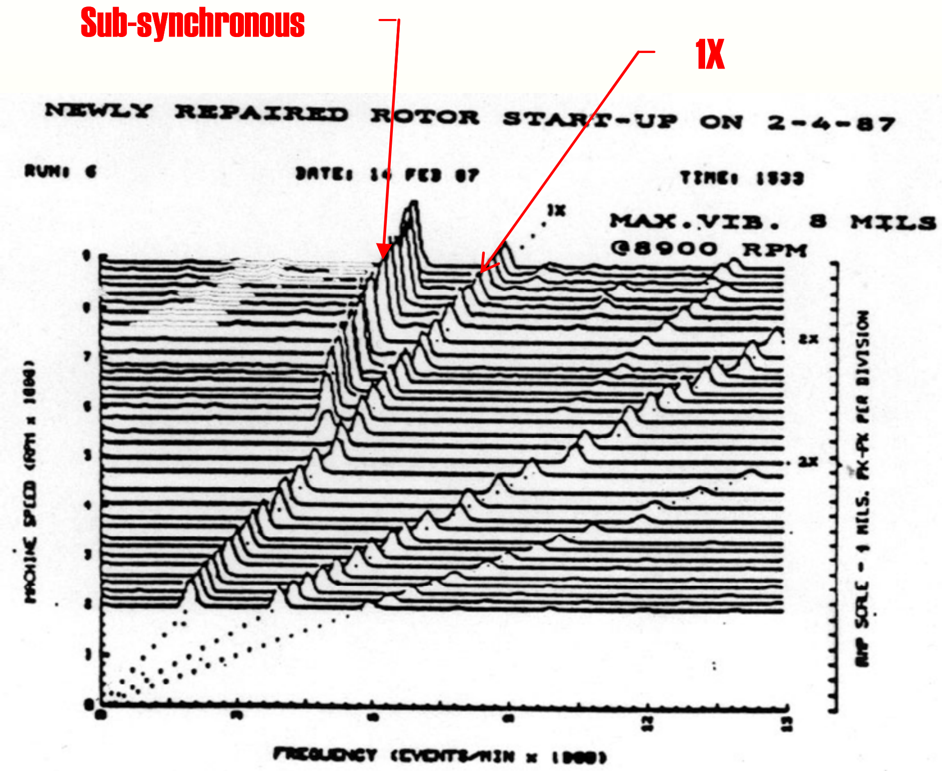




Discharge Recirculation

Damaging Effects

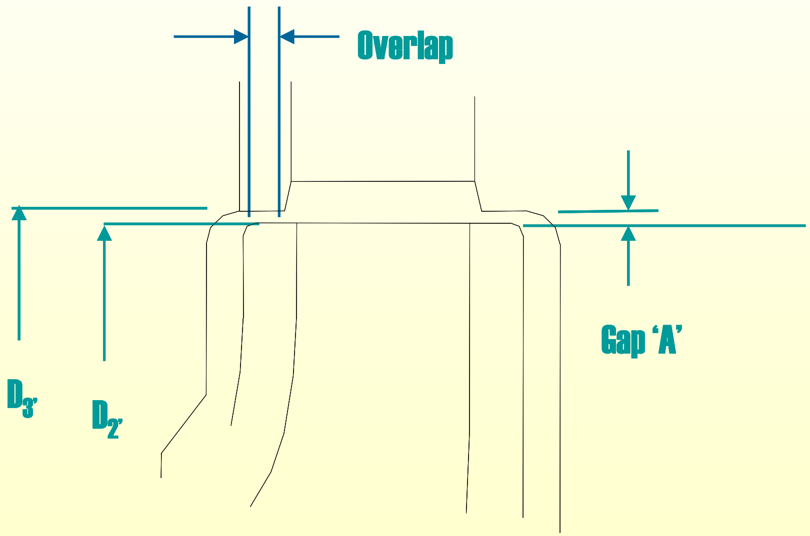
- Reduced rotor damping
- Elevated pump vibration (0.6 – 0.9 X; 1X)
- Shaft failure
- Seal failure
- Bearing failure
- Diffuser vane tip breakage
- Impeller shroud erosion



Discharge Recirculation

Solutions

- Operation at BEP
- Provide a "filter" to straighten the flow – an orifice comprised of a tight Gap A, and sufficient Overlap (Gap C)

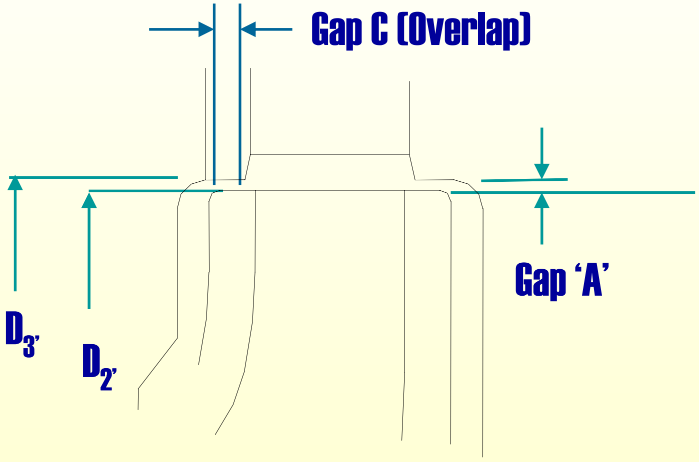


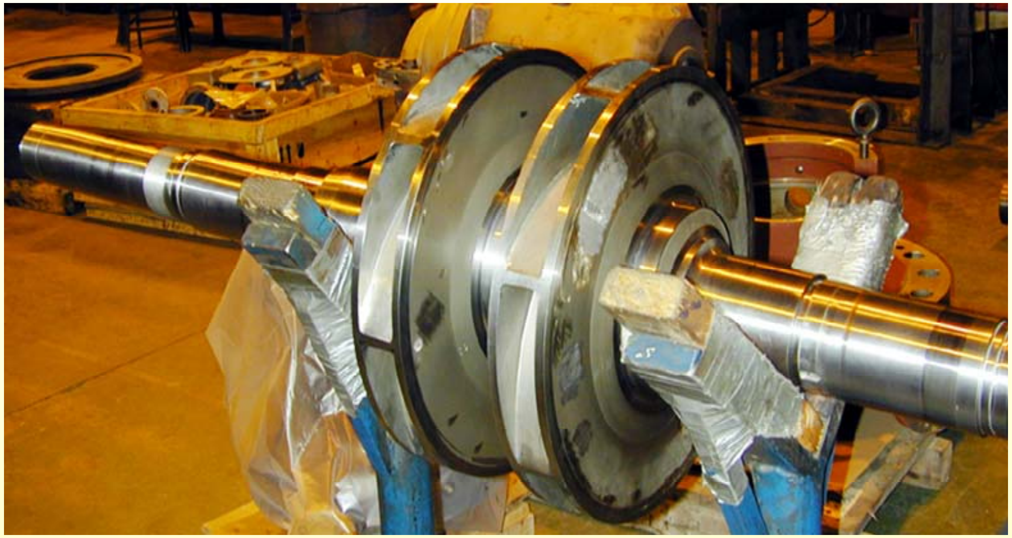
$$\text{Gap A} = \frac{D_3^{'} - D_2^{'}}{2}$$

$$\text{Gap C}\left(\text{Overlap}\right) \approx 4 \text{ to } 6 \times \text{Gap A}$$

Discharge Recirculation

Gap A and
Gap C

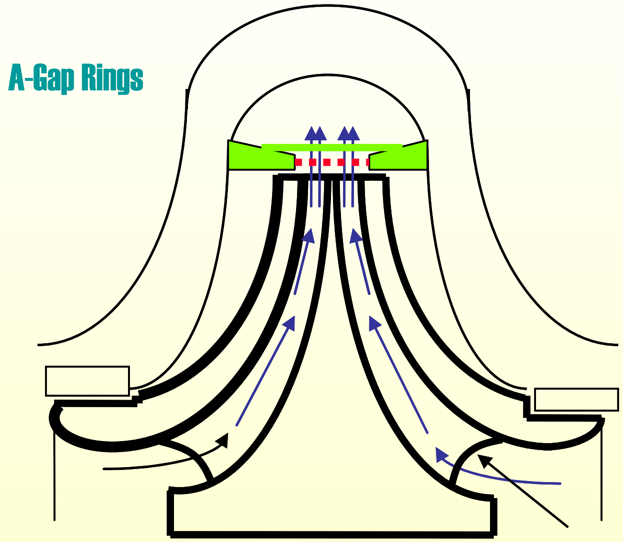


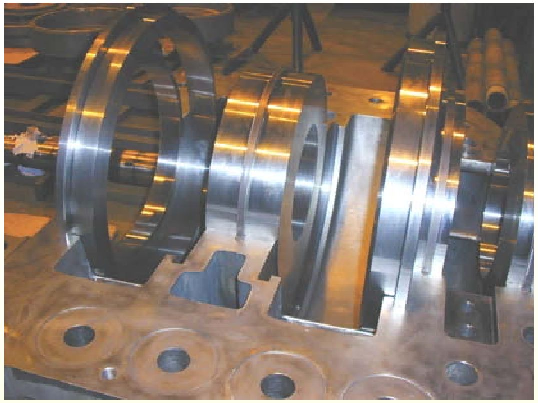


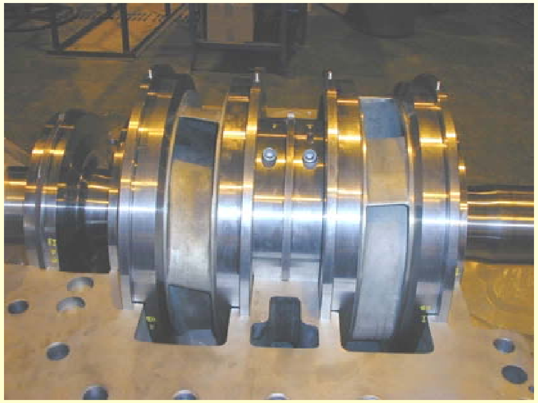
Discharge Recirculation

Gap A and Gap C

Addition of Gap A/Overlap rings required in cast iron volute designs



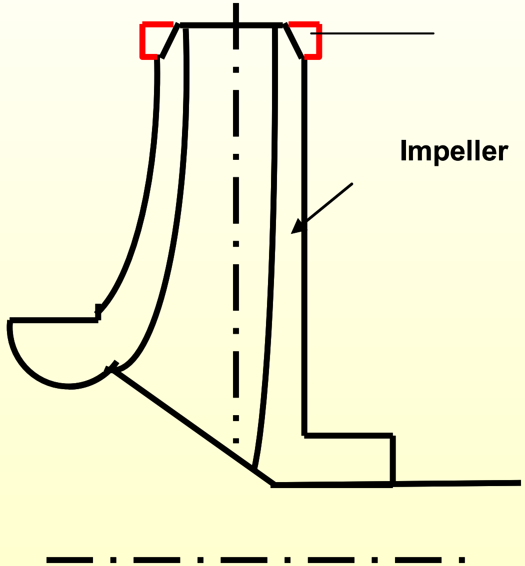




Discharge Recirculation

Gap A and Gap C

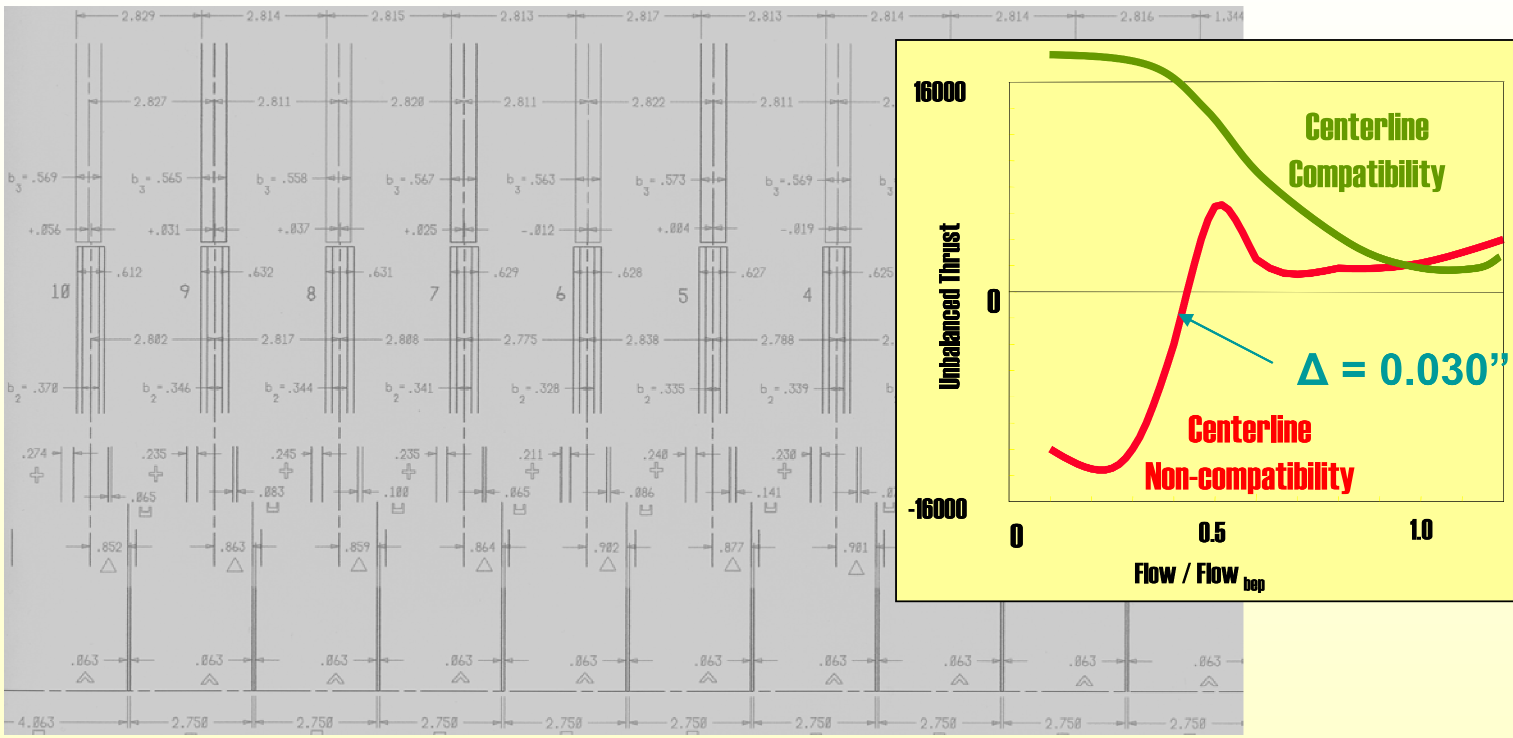
Tight Gap A increases local pressures requiring stronger impeller exit shroud designs



Discharge Recirculation

Solutions

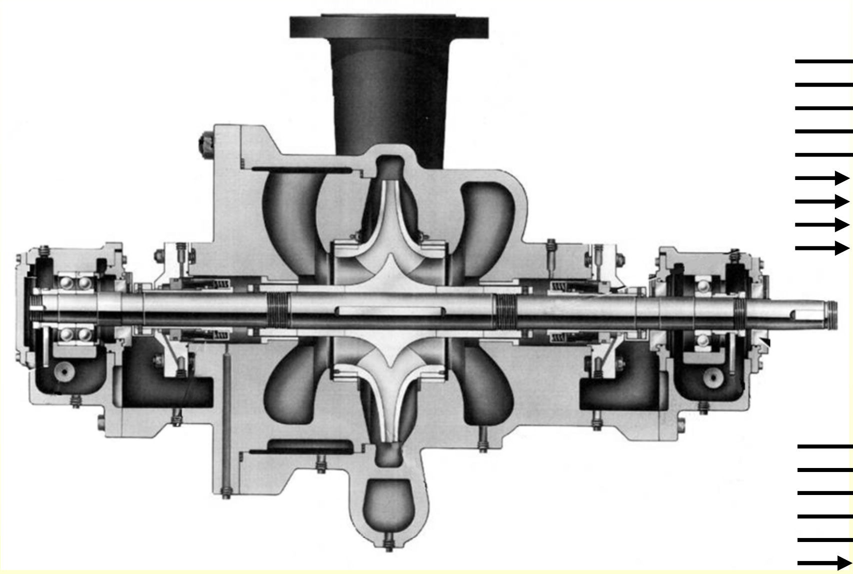
- Centerline compatibility

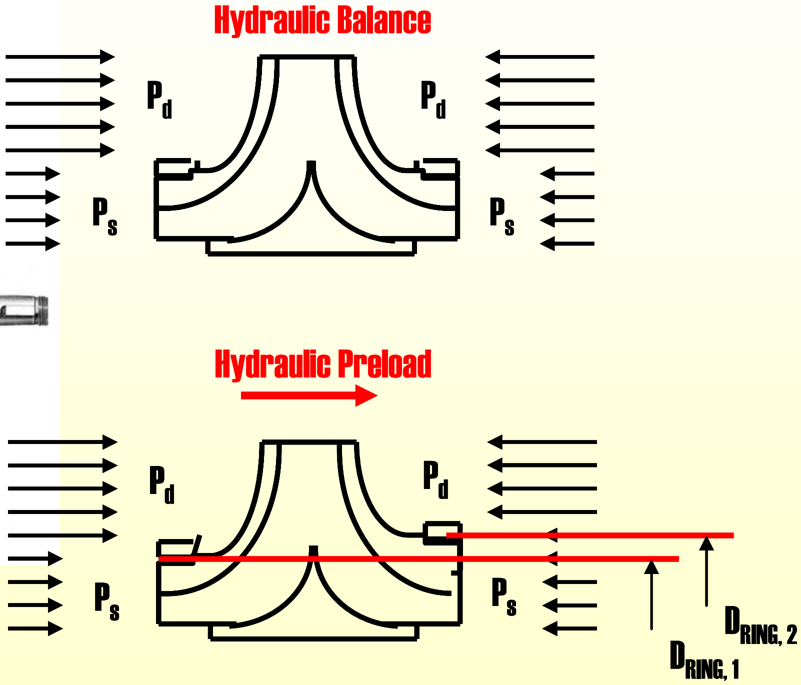


Discharge Recirculation

Solutions

- Bias casing ring design for horizontal, single-stage pumps

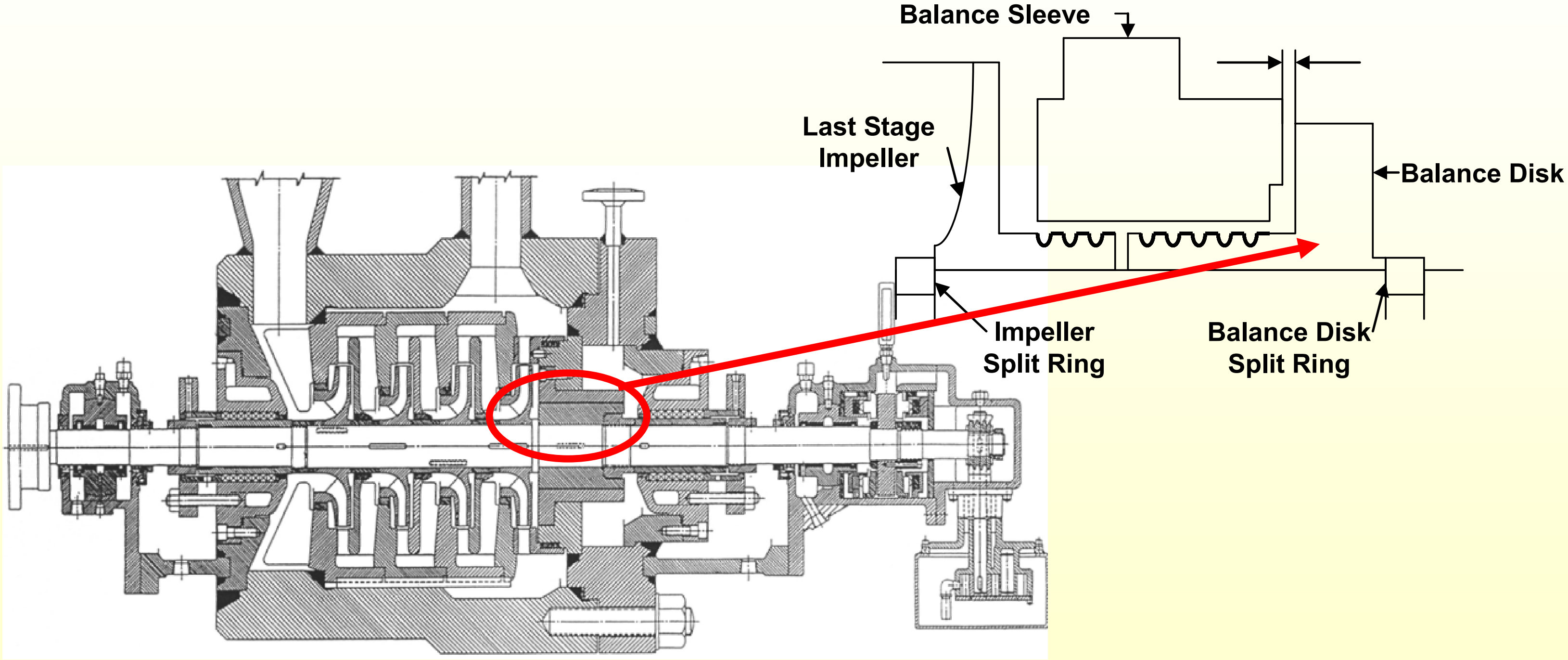




Discharge Recirculation

Solutions

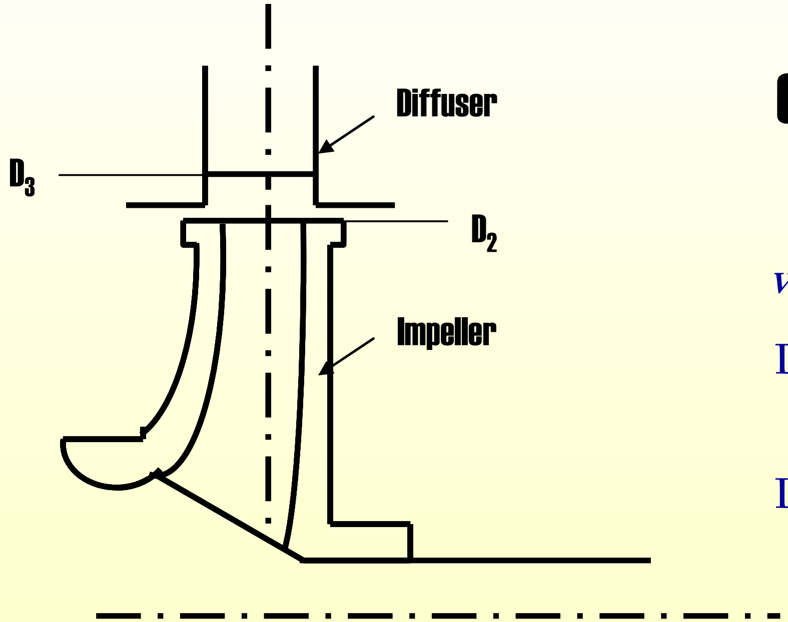
- Large flange balance disc design on horizontal, multi-stage pumps



High Impact Loading

Causes

- Off-peak operation
- Too-tight Gap B



$$\text{Gap B} = \frac{D_3 - D_2}{D_2}$$

where:

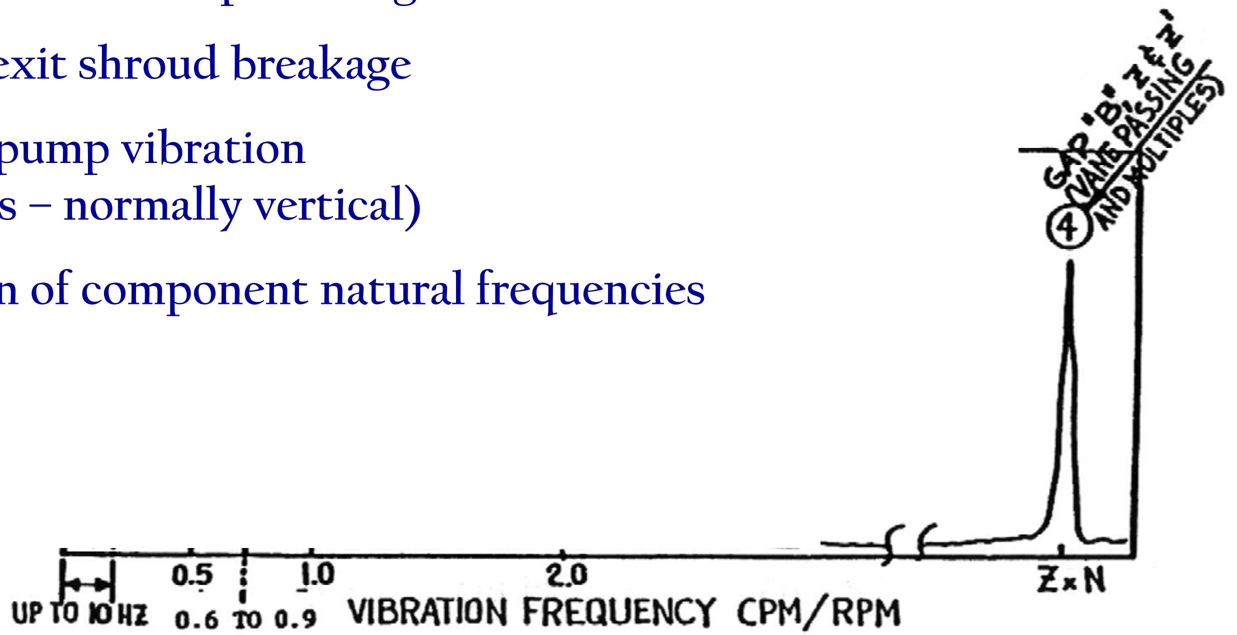
D_3 = Diffuser or volute inlet vane diameter

D_2 = Impeller exit vane diameter

High Impact Loading

Damaging Effects

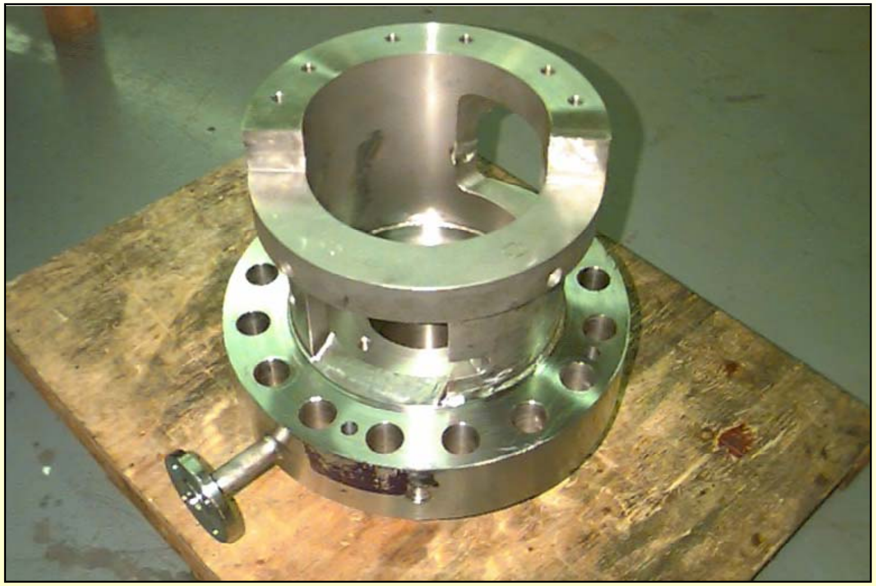
- Diffuser inlet vane tip breakage
- Impeller exit shroud breakage
- Elevated pump vibration
  (vane pass – normally vertical)
- Excitation of component natural frequencies



High Impact Loading

Solutions

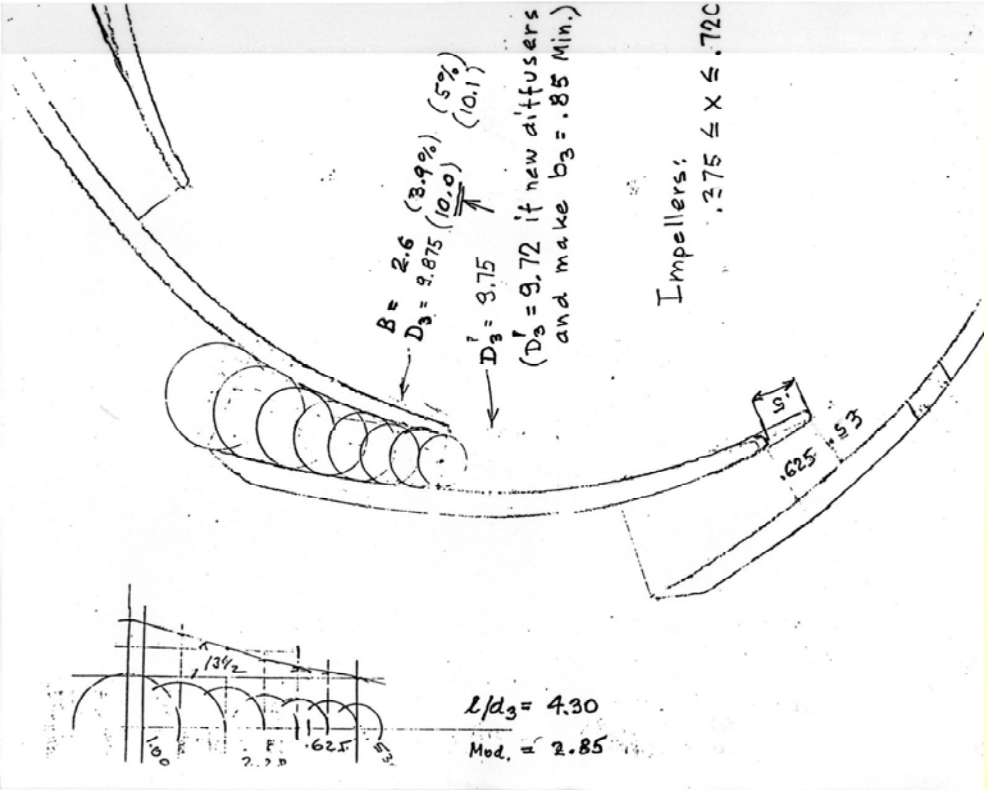
- Operation at BEP
- Provide the proper Gap B
- 360º bearing housings



High Impact Loading

Gap B

- Diffuser Pumps: ~ 6%
- Volute Pumps: ~ 10%
- L/d_3 ratio: 4:1

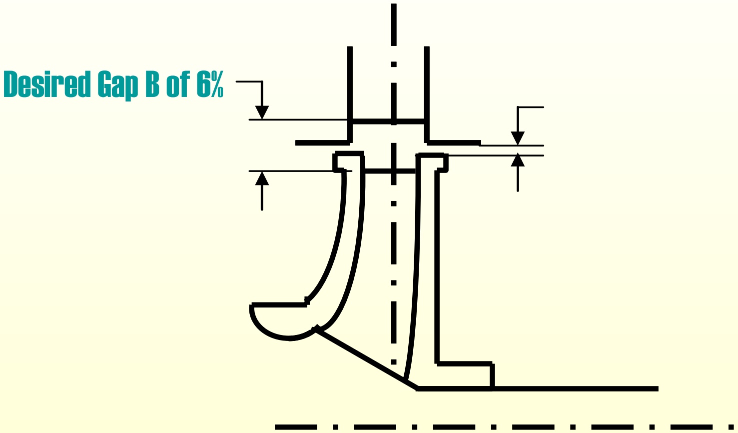


High Impact Loading

Gap B Modification

For turbine-driven pumps:

- Cut impeller *vane*, thereby not jeopardizing all important L/d_3 ratio, and not affecting Gap A to Gap C ratio
- Turbine will speed-up by the same ratio as the impeller cut *(affinity laws)* to achieve required hydraulics
- Must ensure not to exceed turbine over-speed limit

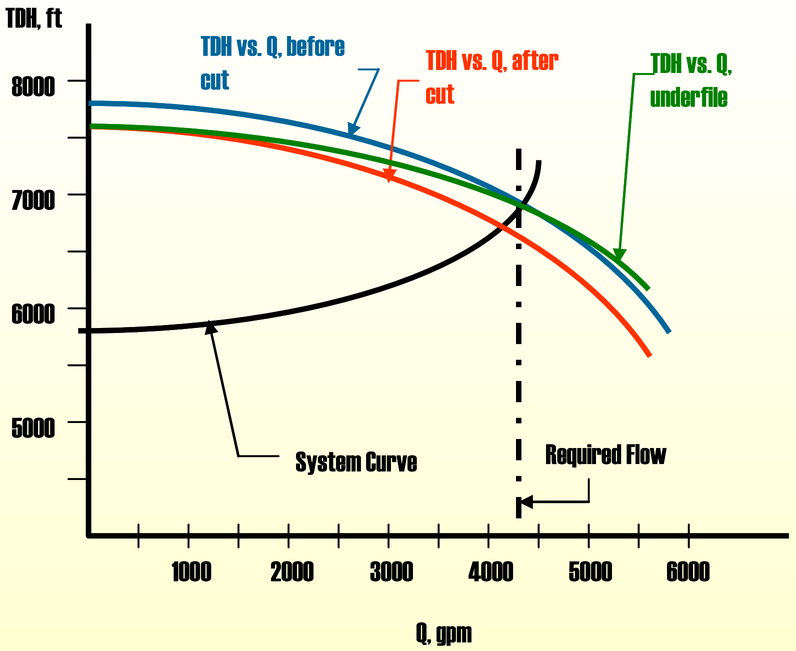


High Impact Loading

Gap B Modification

For motor-driven pumps:

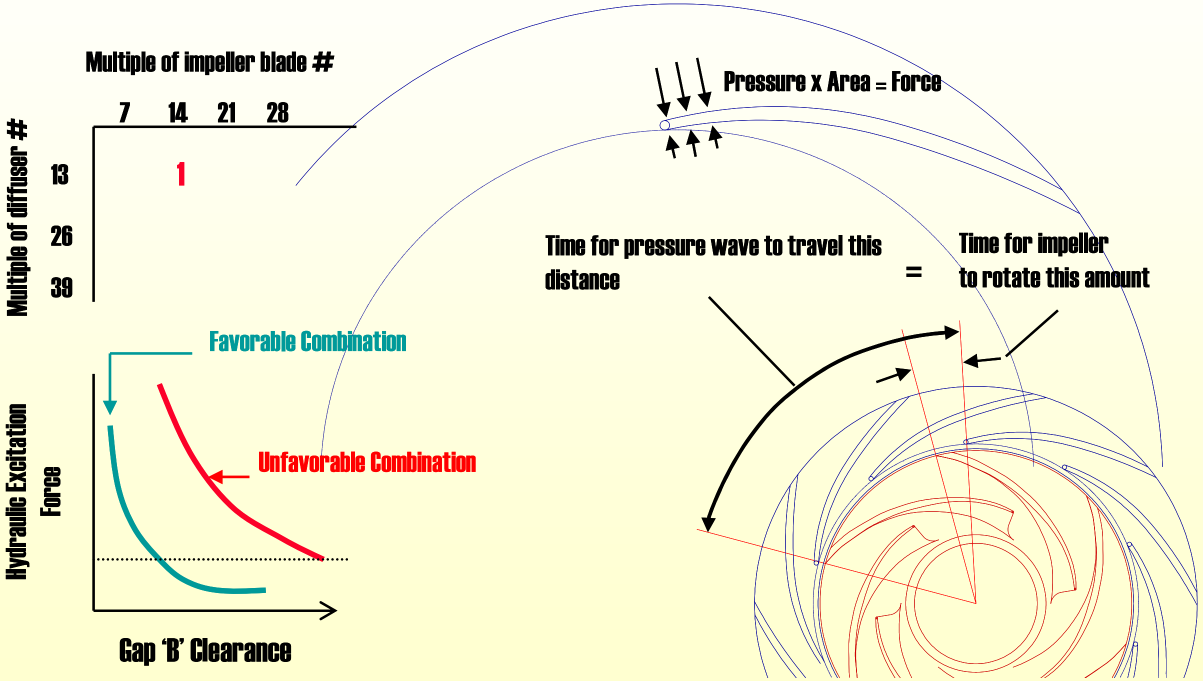
- Machine back diffuser inlet vanes to maximum allowable considering L/d_3 limitations
- Cut impeller D_2 to eliminate "excess performance"
- Cut and underfile impeller exit vanes to achieve balance of what is required for proper Gap B



Acoustic Resonance

Causes

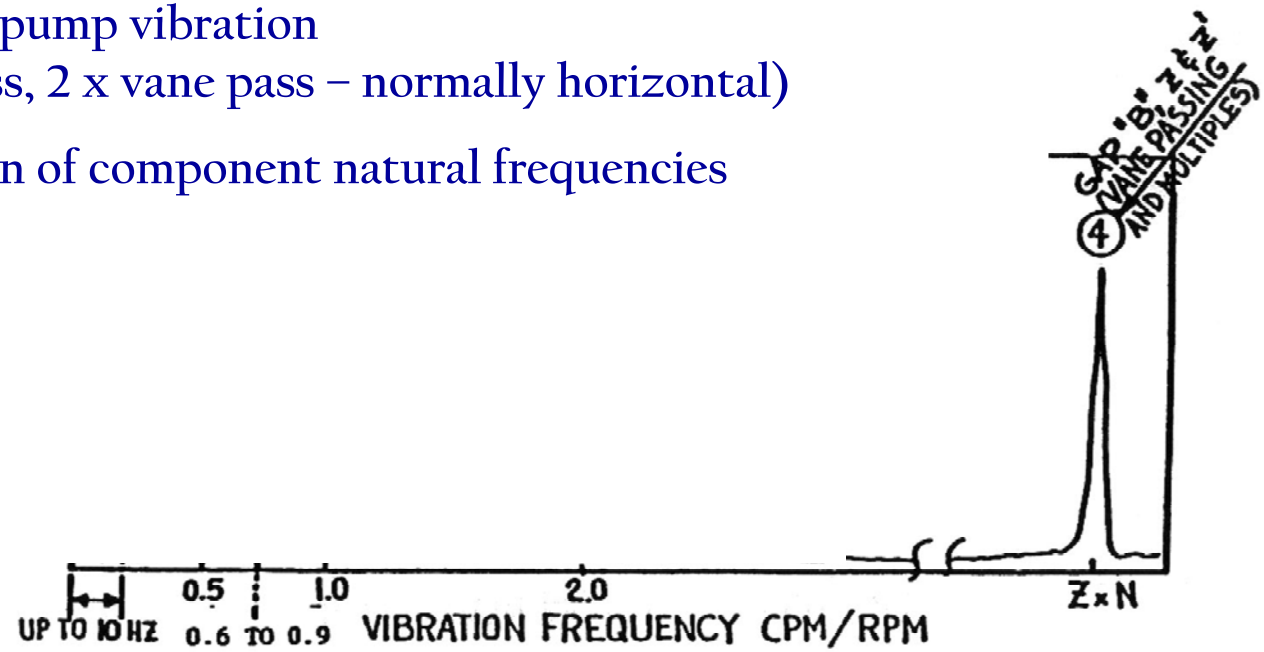
- Poor vane combination – no "0" or "1" in the absolute value of the impeller vane (and multiples) versus diffuser or volute vane (and multiples)



Acoustic Resonance

Damaging Effects

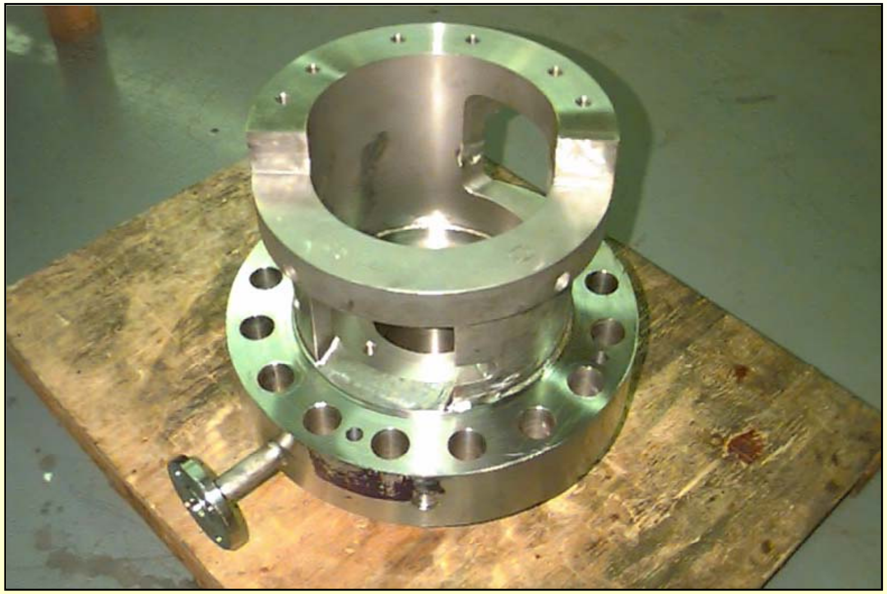
- Elevated pump vibration
  (vane pass, 2 x vane pass – normally horizontal)
- Excitation of component natural frequencies



Acoustic Resonance

Solutions

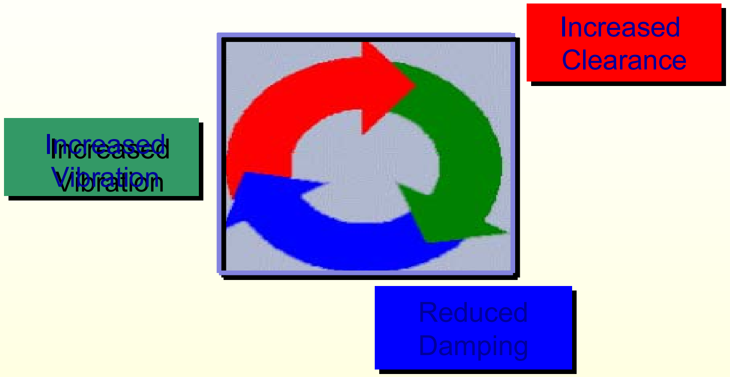
- Modification of impeller or diffuser/volute vane numbers
- Stiffening of volute/crossover passage
- 360º bearing housings



Premature Opening of Clearances

Causes

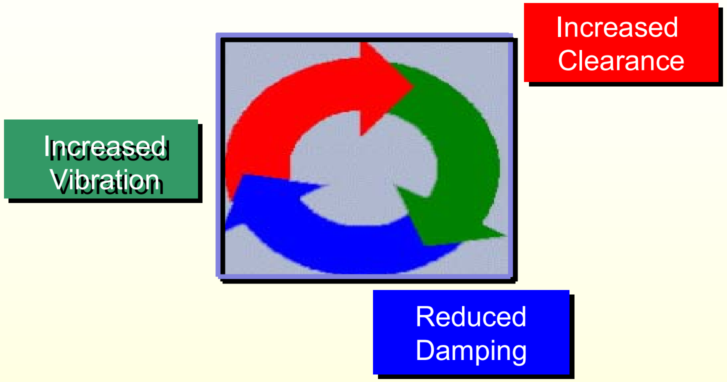
- Relative motion between rotor and stator
- Imbalance
- Misalignment
- Improper lift/setting
- Loose fit-ups
- Poor concentricity, parallelism, perpendicularity tolerances
- Discharge recirculation



Premature Opening of Clearances

Causes

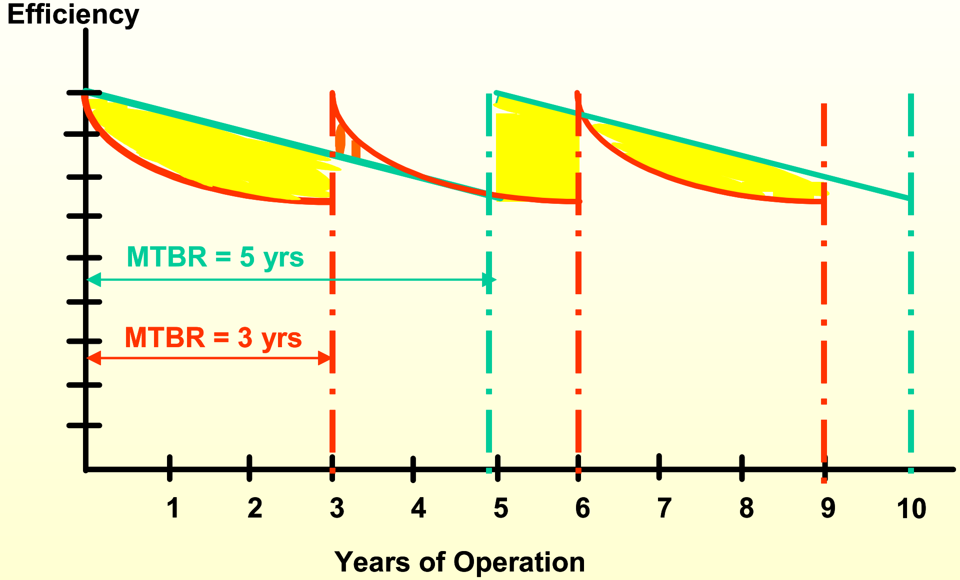
- Excessive shaft deflection
- Galling materials
- Operation at rotor wet critical
- Improper bearing design
- Non-rigid base



Premature Opening of Clearances

Damaging Effects

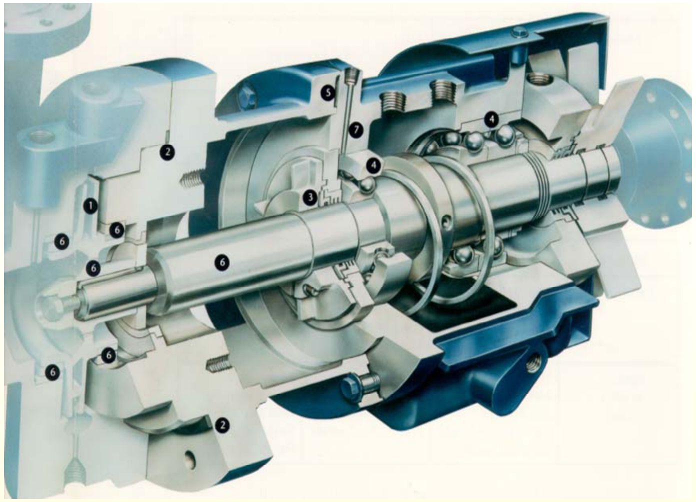
- Increased internal recirculation (loss of performance)
- Rotor seizure
- Catastrophic failure

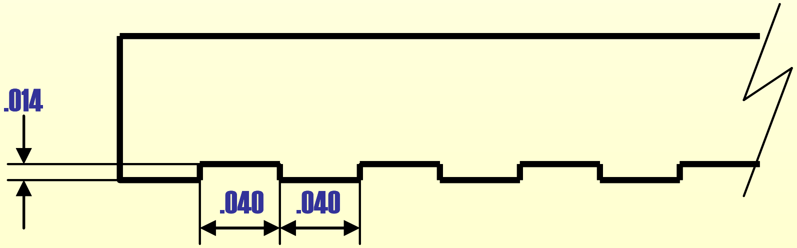


Premature Opening of Clearances

Solutions

- Gap A/Overlap
- Centerline compatibility
- Lower L^3/d^4
    - 9th edition back pull-out upgrade
    - Increased shaft diameter
- Lomakin grooving

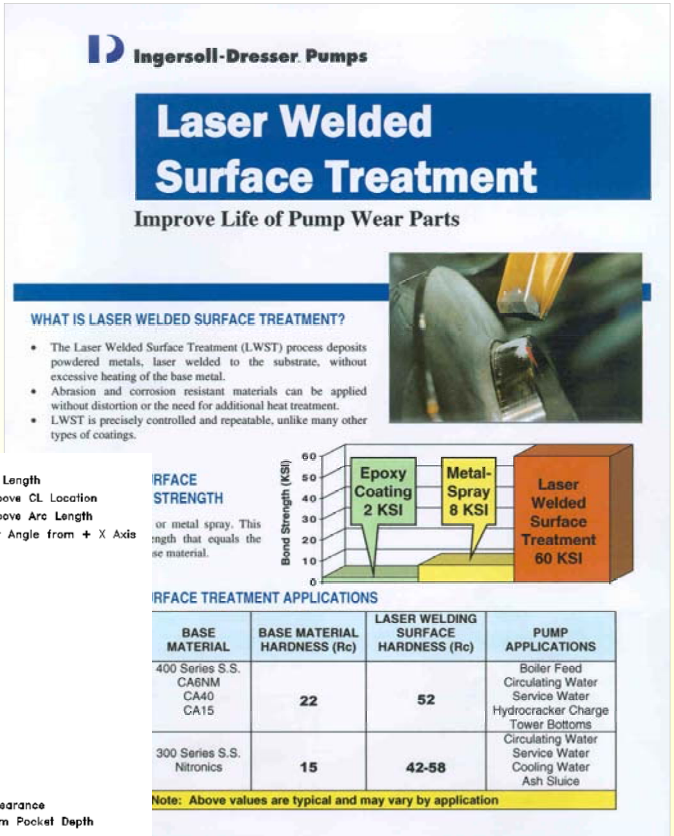




Premature Opening of Clearances

Solutions

- Non galling materials
  - Laser hardened
  - Peek
  - ARHT
  - 420F (high sulfur)
- Proper bearing design
  - Pressure dam
  - Tri-land


Ingersoll-Dresser Pumps

Laser Welded Surface Treatment

Improve Life of Pump Wear Parts

WHAT IS LASER WELDED SURFACE TREATMENT?

- The Laser Welded Surface Treatment (LWST) process deposits powdered metals, laser welded to the substrate, without excessive heating of the base metal.
- Abrasion and corrosion resistant materials can be applied without distortion or the need for additional heat treatment.
- LWST is precisely controlled and repeatable, unlike many other types of coatings.

RFACE STRENGTH

or metal spray. This ngth that equals the se material.

Bond Strength (KSI): 0, 10, 20, 30, 40, 50, 60

Epoxy Coating 2 KSI

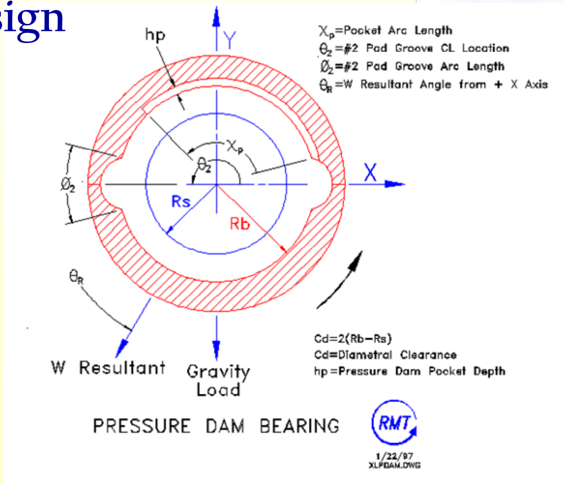
Metal-Spray 8 KSI

Laser Welded Surface Treatment 60 KSI

RFACE TREATMENT APPLICATIONS

BASE MATERIAL	BASE MATERIAL HARDNESS (Rc)	LASER WELDING SURFACE HARDNESS (Rc)	PUMP APPLICATIONS
400 Series S.S. CA6NM CA40 CA15	22	52	Boiler Feed Circulating Water Service Water Hydrocracker Charge Tower Bottoms
300 Series S.S. Nitronics	15	42-58	Circulating Water Service Water Cooling Water Ash Sluice

Note: Above values are typical and may vary by application



Premature Opening of Clearances

Solutions

- Interference fits
  - Rotor
  - Stator
- Proper bearing clearances
- More stringent tolerances

Premature Opening of Clearances

Solutions

- Removal of soft foot
    - Proper grouting
    - Polyshield baseplates
- Proper field alignment

Premature Opening of Clearances

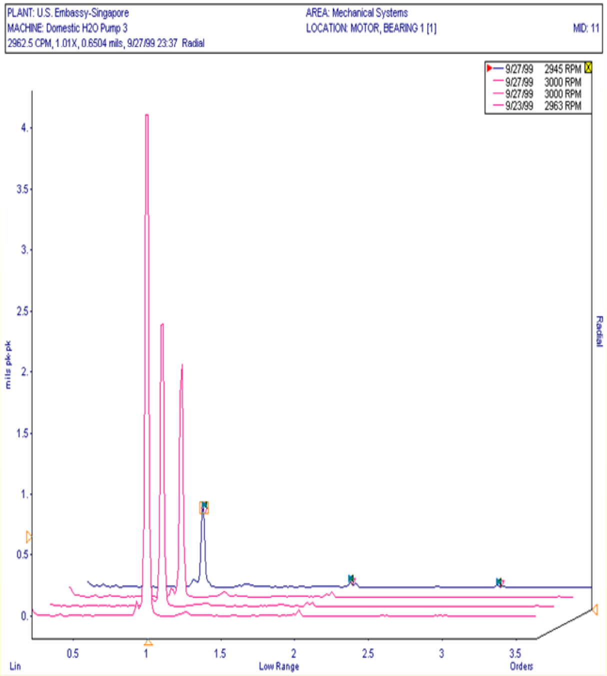
Solutions

- More stringent balance requirements
  - A hard bearing balance machine must be utilized
  - The impellers must be individually balanced on arbors
  - All keys must be fitted to keyways with no excessive stock or unfilled areas (as would occur utilizing square in lieu of full-radius keys)
  - Impellers must have a minimum interference fit-up to shaft between .000 - .0015”
  - Shaft T.I.R. cannot exceed 0.001”
  - Impeller hub turn T.I.R. cannot exceed 0.002”

1W/N

Premature Opening of Clearances

Pump Vibration, Before & After Balancing



Pump Vibration

Domestic H_2O Pumps

Premature Opening of Clearances

Vibration vs Bearing Life

- Reducing the forces caused by unbalance, looseness and misalignment will result in lower vibration levels.
- Reducing excessive belt tension will reduce machine forces but will not produce an appreciable reduction in vibration levels.

Impact of Vibration Reduction on Bearing Life

(Assuming Dynamic Load is the Major Force Component)

	% Increase in Bearing Life	
% Reduction in Vibration	Ball Bearings	Other Rolling Element Bearings
5	17	19
10	37	42
15	63	72
20	95	110
25	137	161
30	192	228
40	363	449
50	700	908

Source:L. Douglas Berry, Vibration Versus Bearing Life, Reliability, Vol. 2, Issue 4,November 1995

Poor Pump Performance

Causes

- Excessive running clearances
- Low voltage, low speed
- Improper impeller diameter(s)
- Broken hydraulic components
- Impeller inlet obstruction
- Double suction impeller installed backwards
- Insufficient l/d3
- Wrong design/poorly replicated impellers

Poor Pump Performance

Causes

- Hydraulic instability (localized flattening)
- Erosion of the volute cutwater/diffuser inlet vanes
- Inlet cavitation
  - Insufficient NPSHA
  - Separation at off-peak conditions
  - Inlet flow disturbances
    Partially closed suction valve
    Balance line leakoff too close to suction
    Clogged strainer
  - Air or steam vortices

Poor Pump Performance

Damaging Effects

- Insufficient flow and/or pressure
- Hydraulic imbalance
- Poor paralleling operation
- Separation/cavitation

Poor Pump Performance

Solutions

- Operation at BEP
- Maintaining design clearances
- CMM technology
- Hard metal overlays/coatings
- Investment castings
- In-depth inspections to drawings

Hot Radial Bearings

Causes

- Parallel misalignment
- Insufficient lubrication flow/pressure
- High oil level (ball bearings)
- Oil contamination
- Too tight journal-to-shaft clearance (sleeve bearings)
- Low flow operation
- Too tight inner race-to-shaft fit (ball bearings)
- Too tight outer race-to-bearing housing fit (ball bearings)
- Defective ball and/or cages (ball bearings)

Hot Radial Bearings

Damaging Effects

- Loss of bearing L-10 life
- High vibration
- Bearing failure
- Catastrophic pump failure

Hot Radial Bearings

Solutions

- Proper pump component tolerances
- Operation at pump BEP

Hot Thrust Bearings

Causes

- Angular misalignment
- Insufficient lubrication flow/pressure
- High oil level (ball bearings)
- Oil contamination
- Improperly sized balance device
- Excessive axial clearance (disks)

Hot Thrust Bearings

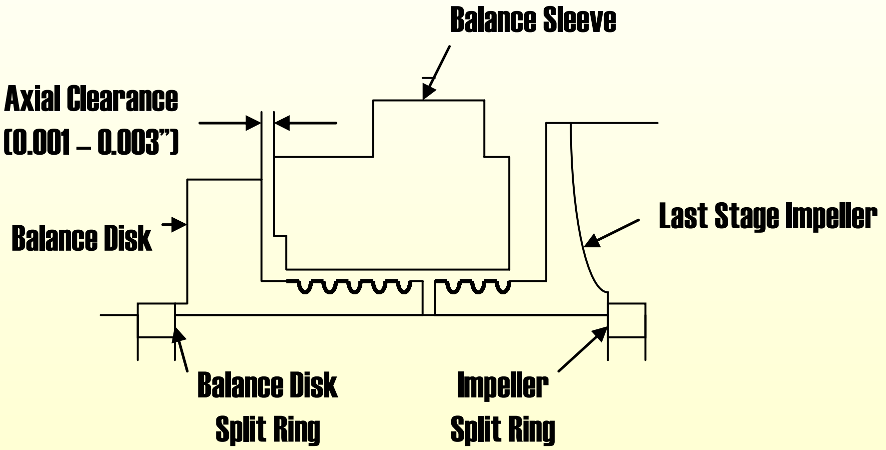
Damaging Effects

- Loss of bearing L-10 life
- High vibration
- Bearing failure
- Balance device failure
- Catastrophic pump failure

Hot Thrust Bearings

Solutions

- Proper component tolerances
- Rotor/stator dimensional analysis
- Upgraded balance device
- Proper bearing design
- Gap A, B, Overlap/C
- Operation at pump BEP



Black Oil

Causes

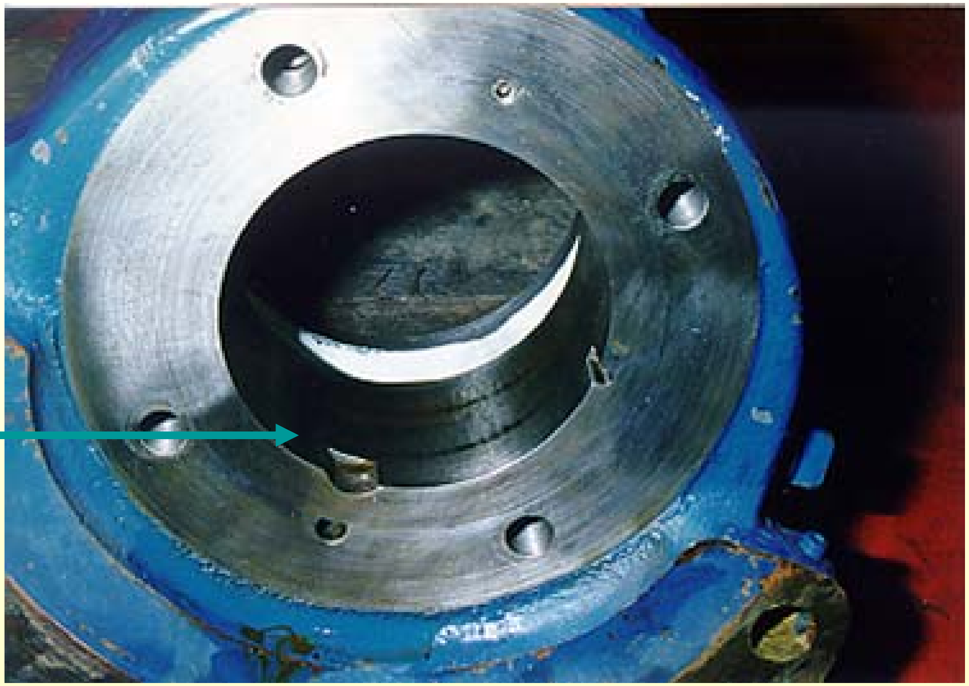
- Oversized bearing
- Insufficient pre-load
- Improper oil level
- Improper oil viscosity
- Axial shuttling
- Excessive thrust loads
- Poor fit-up to shaft shoulder
- Poor fit-up to shaft/bearing housing
- Stray motor currents

Black Oil

Damaging Effects

- Reduced bearing life
- Elevated pump vibration

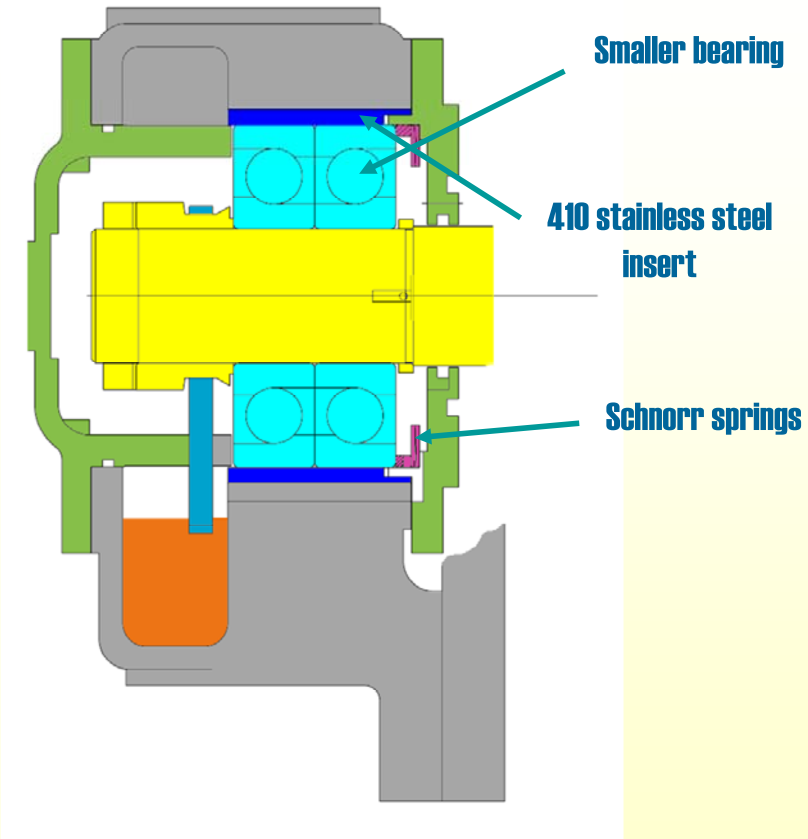
Fretting in the bore



Black Oil

Solutions

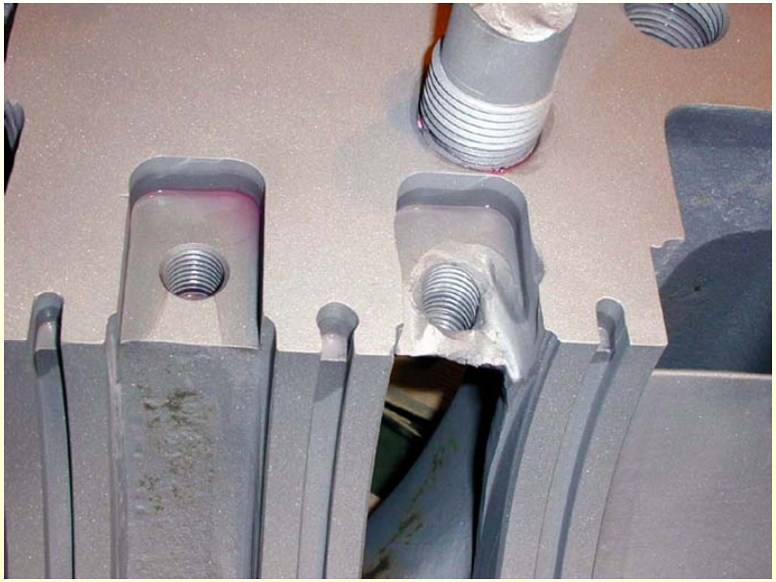
- Schnorr springs
- Proper bearing size
- Proper fit-ups
- Proper oil selection



Casing Leakage

Causes

- Excessive pipe strain
- Non-parallel parting flanges
- Deteriorated gaskets/o-rings
- System upsets
- Improper warming
- Casting defects
- Local fluid velocity exceeding material limits (erosion)



Casing Leakage

Damaging Effects

- Accelerated component erosion (wire drawing)
- Environment contamination

Casing Leakage

Solutions

- Dimensional analysis
- Proper machining operations
- Metal-to-metal fits

Seal Leakage

Causes

- Improper tightening of packing
- Improper packing material
- Improper seal setting (mechanical)
- Improper seal design
- Insufficient space
- Lack of seal chamber venting
- Improper seal flush or seal flush cleanliness

Seal Leakage

Causes

- Flashing in chamber
- Excessive bushing clearance
- Misalignment
- Axial shuttling
- Excessive shaft deflection
- Pipe strain
- High vibration

Seal Leakage

Damaging Effects

- Reduced seal life
- Environmental contamination

Seal Leakage

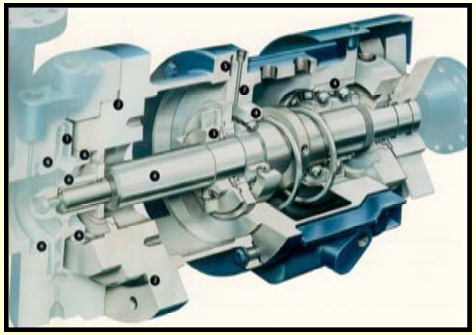
Solutions

- Removal of hydraulic instability
- Improved rotor stiffness
- Improved alignment techniques
- Upgraded coupling designs
- Upgraded seal designs and metallurgy
- Proper seal chamber space
- Improved seal flush system and cleanliness

Seal Leakage

Improved Rotor Stiffness

- Radial load increases to the left of BEP
- $L^3/d_4 < 100$



DEFLECTION CALCULATIONS, API 610, 8TH EDITION, Par. 2.5.7. PUMP TYPES OH1, OH2

PUMP TYPE: 4UB-Orig SHOP ORDER #

PUMP SIZE: PeCO

INPUT	(See Figure 1 for illustration & dimensions!)	
SPECIFIC GRAVITY OF FLUID	1.000	
IMPELLER SPECIFIC SPEED (Ns)	1450	(U.S. units)
IMPELLER WEIGHT	35.0	lbs.
IMPELLER DIAMETER (O.D.)	9.500	in.
IMPELLER EXIT WIDTH + SHROUDS	1.375	in.
DIFF. HEAD @ SHUT-OFF (Hso)	370	ft.
DIFF. HEAD @ MIN. FLOW (Hmin)	370	ft.
DIFF. HEAD @ DUTY POINT (Hdp)	325	ft.
GPM @ MIN. FLOW (Qmin)	200	gpm
GPM @ DUTY POINT (Qdp)	1000	gpm
GPM @ BEP (Qbep)	1200	gpm
LENGTH BETWEEN BRGS (L2)	10.61	in.
LENGTH RAD BRG TO IMP CL (L1)	16.16	in.
LENGTH SEAL FACE TO IMP CL (X)	10.60	in.
SHAFT DIA BETWEEN BRGS (d2)	2.820	in.
SHAFT DIAMETER AT SLEEVE (d1)	1.875	in.
SHAFT MODULUS OF ELASTICITY	2.70E+07	(See TABLE) psi
VOLUTE? (S)INGLE/ (D)OUBLE	D	
INTERMEDIATE RESULTS		
$F_{hydraulic\ (so)}$	75	lbf.
$F_{hydraulic\ (min)}$	64	lbf.
$F_{hydraulic\ (dp)}$	65	lbf.
MOMENT OF INERTIA @ d1	0.607	in.^4
MOMENT OF INERTIA @ d2	3.104	in.^4
RADIAL THRUST FACTOR @ SHUT-OFF (Kso)*	0.036	
RADIAL THRUST FACTOR @ MIN FLOW (Kmin)*	0.030	
RADIAL THRUST FACTOR @ DUTY PT (Kdp)*	0.036	

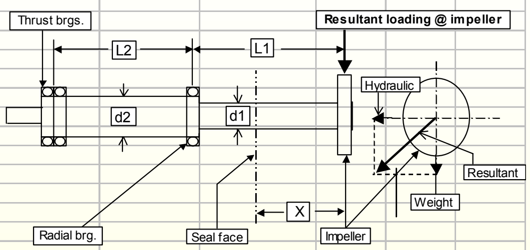


FIGURE 1

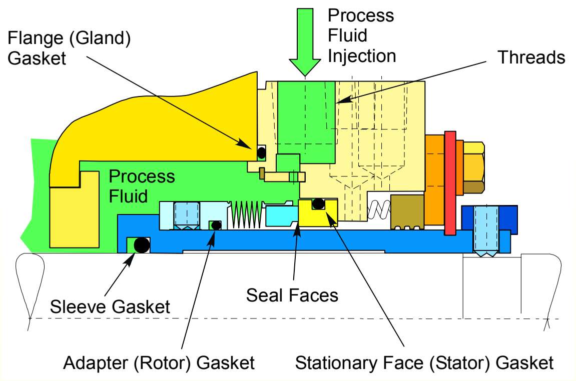
OUTPUT	RESULTANT LOADING @ IMPELLER (lbf.)	DEFLECTION @ IMPELLER (in.)	DEFLECTION @ SEAL FACE (in.)
SHUT-OFF	83	0.0081	0.0014
MIN. FLOW	73	0.0070	0.0013
DUTY POINT	74	0.0072	0.0013
L3/D4	341		

Seal Leakage

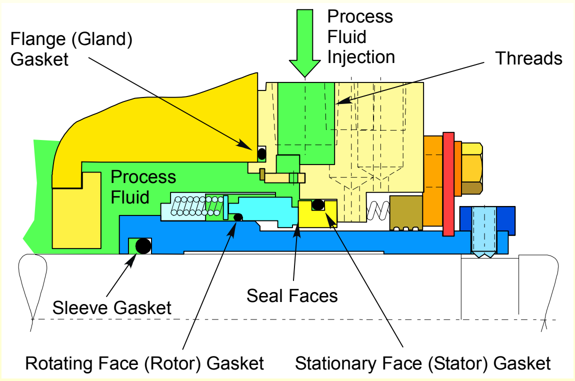
Upgraded Seal Designs

- Seal Faces
  - Different carbon resin or filler
  - Switch Tungsten Carbide to Silicon Carbide
  - Direct Sintered Silicon Carbide instead of Reaction Bonded Silicon Carbide
- Secondary Seals
  - Various O-ring options such as perfluoroelastomers
  - PTFE-based seals
- Metal
  - High alloys such as Alloy C-276 or Alloy 600
  - Bellows material selection is limited

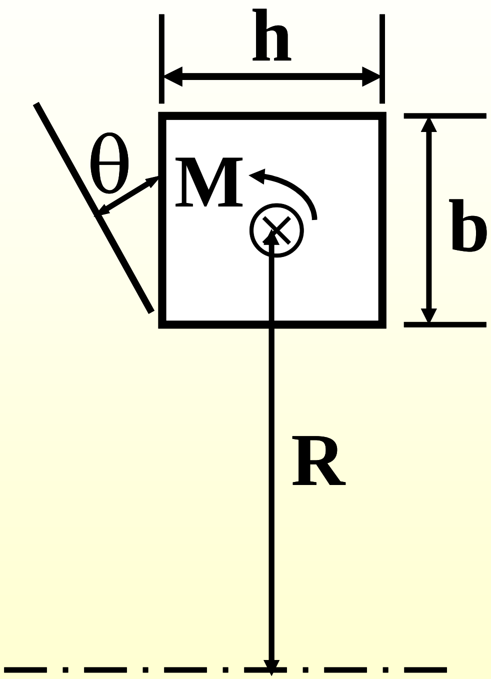
Bellows Seal



Pusher Seal



Ring Deflection

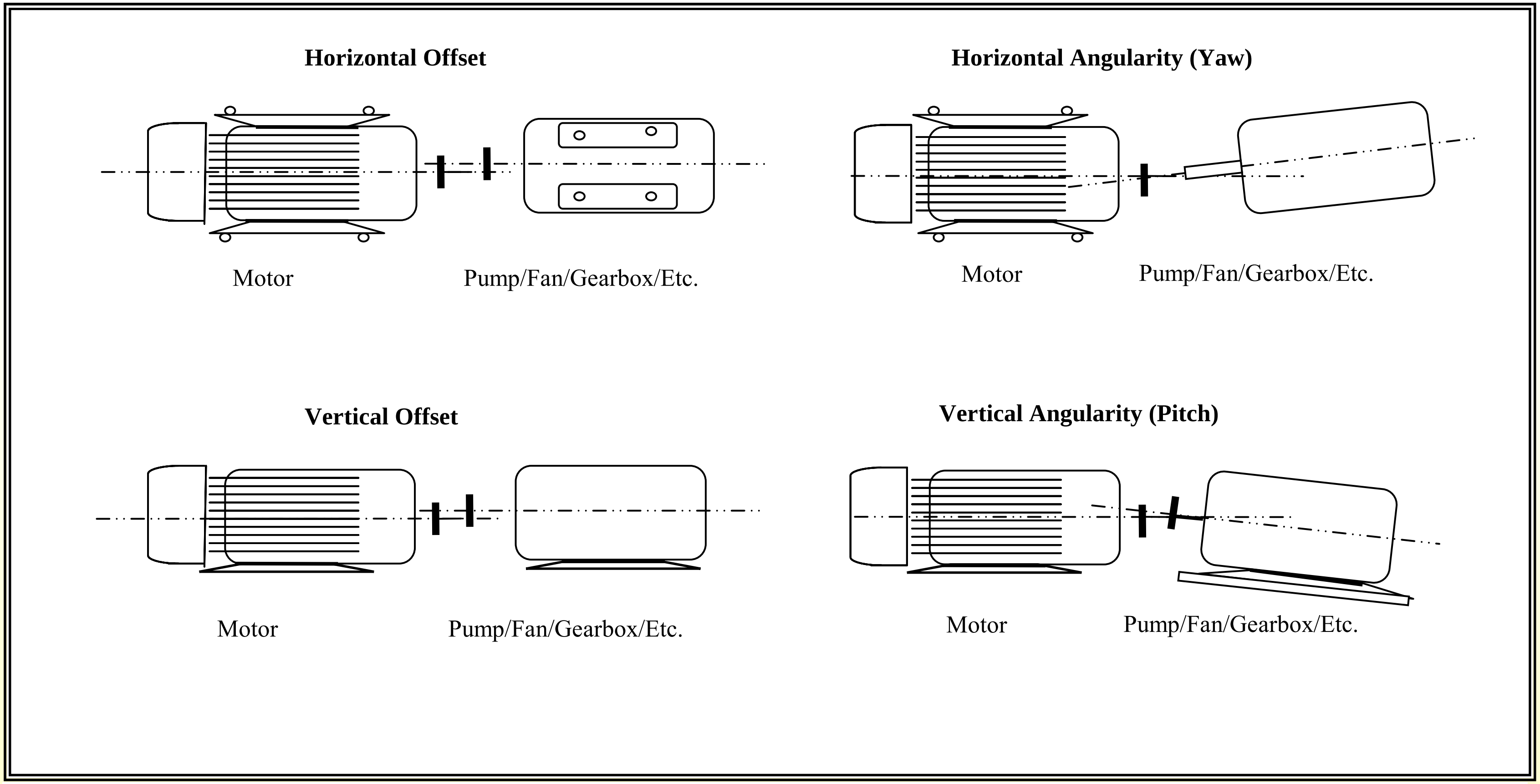


Twist angle due to moment, M:

$$\theta = \frac{12MR^2}{Ebh^3}$$

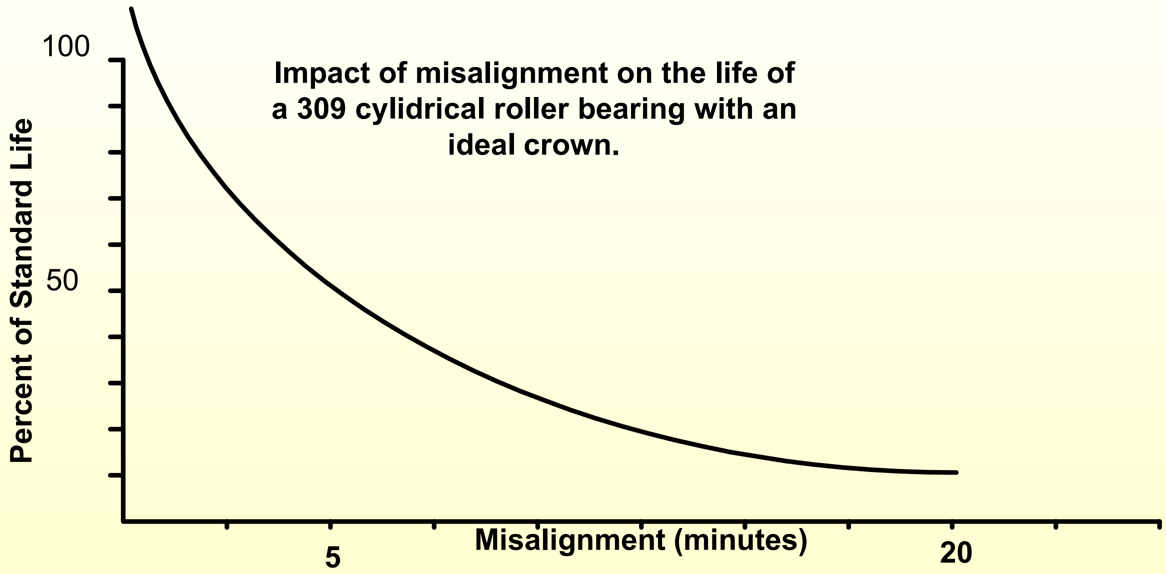
θ = Twist angle
M = Moment per unit length
E = Elastic modulus
R = Radius, axis to ring center
b = Radial width
h = Axial length

Alignment


Horizontal Offset
Motor
Pump/Fan/Gearbox/Etc.
Horizontal Angularity (Yaw)
Motor
Pump/Fan/Gearbox/Etc.
Vertical Offset
Motor
Pump/Fan/Gearbox/Etc.
Vertical Angularity (Pitch)
Motor
Pump/Fan/Gearbox/Etc.

Alignment

Effects of Misalignment



Vertical Alignment Process

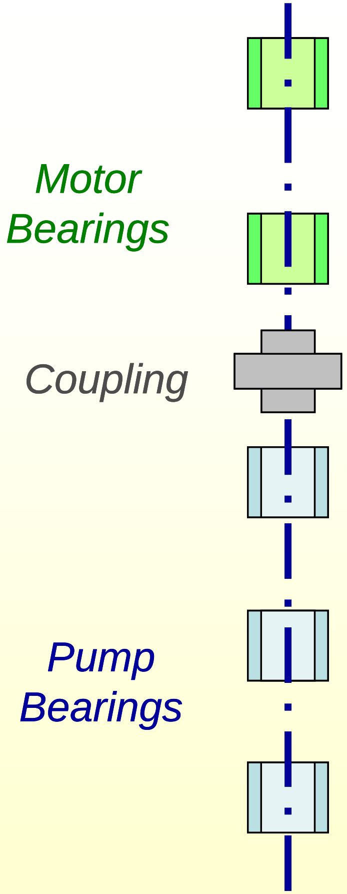
Goals

Primary:

- Align pump and motor bearing centerlines (i.e., stator alignment)

Secondary:

- Check pump and motor rotors for straightness



Vertical Alignment Process

Issues

Discharge Head/Motor Support

- Lack of adjustability
  - Rabbet fits bind
  - Body bound bolts
- Loss of concentricity/parallelism
  - Corrosion of fits
  - Stress relief of weldments
  - Incorrect from original manufacture
  - Incorrectly repaired
  - Not designed to replace parts
- Add register / truth bands

Vertical Alignment Process

Issues

Motor:

- Mounting face not perpendicular to bearing centerline
- Rabbet fit not designed to hold concentricity [generally]
- Add four jack screws

Vertical Alignment Process

Issues

Vertical Pumps depend on the stack up of tolerances for internal alignment

Vertical Alignment Process

Alignment Process

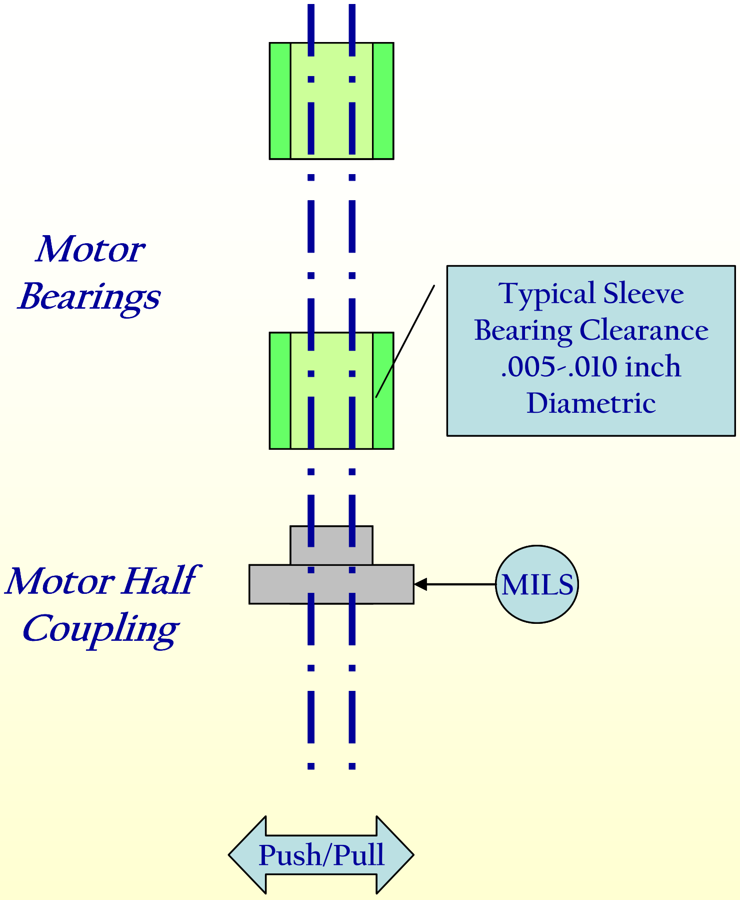
- Check motor runout
- Use motor as “alignment tool”
- Move motor to minimize TIR to pump
- Couple and check shafting

Vertical Alignment Process

Alignment Process

Check Motor Lateral Side play:

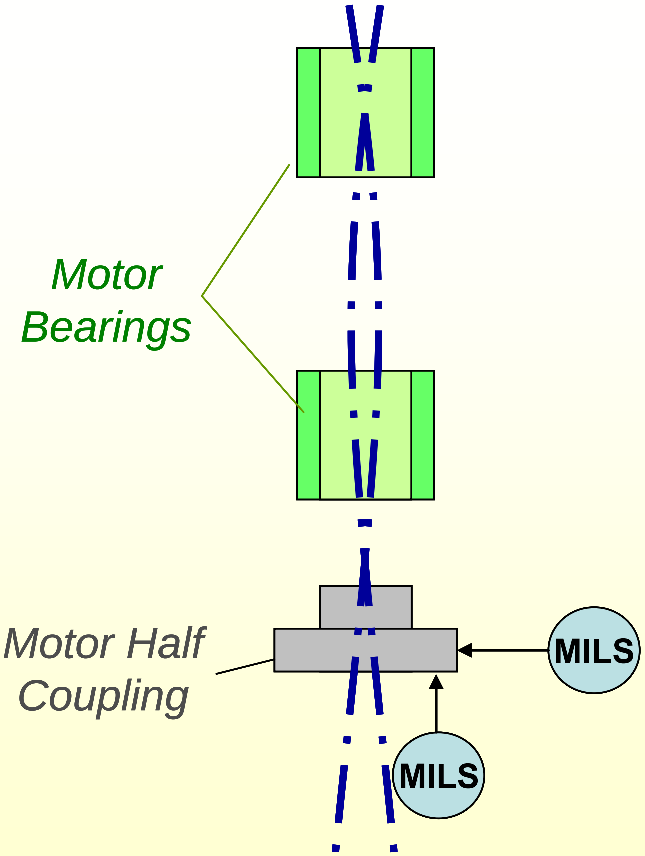
- Sleeve bearing motors: Four centering screws to hold shaft in center of bearing
- Not required for ball bearings



Vertical Alignment Process

Alignment Process

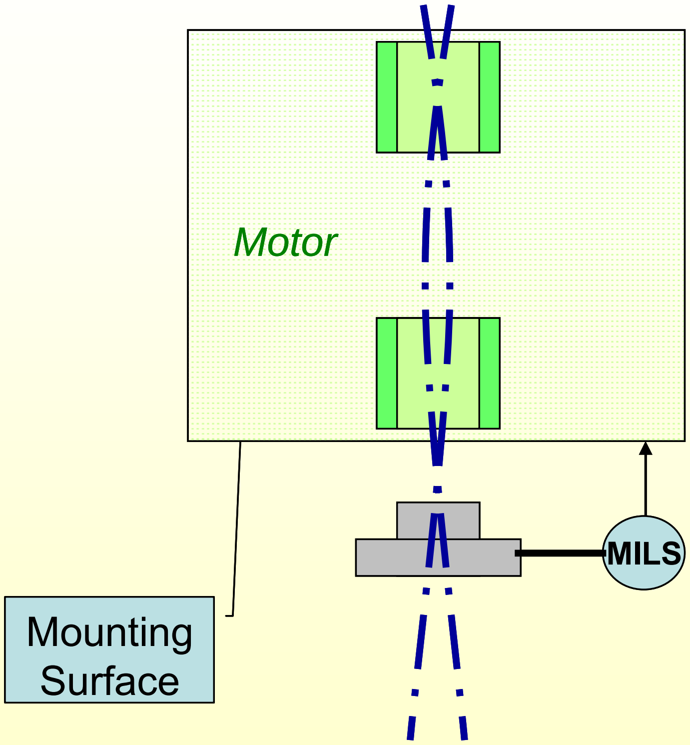
Check Motor Shaft Runout:



Vertical Alignment Process

Alignment Process

Check Motor Mounting Face Runout:



Vertical Alignment Process

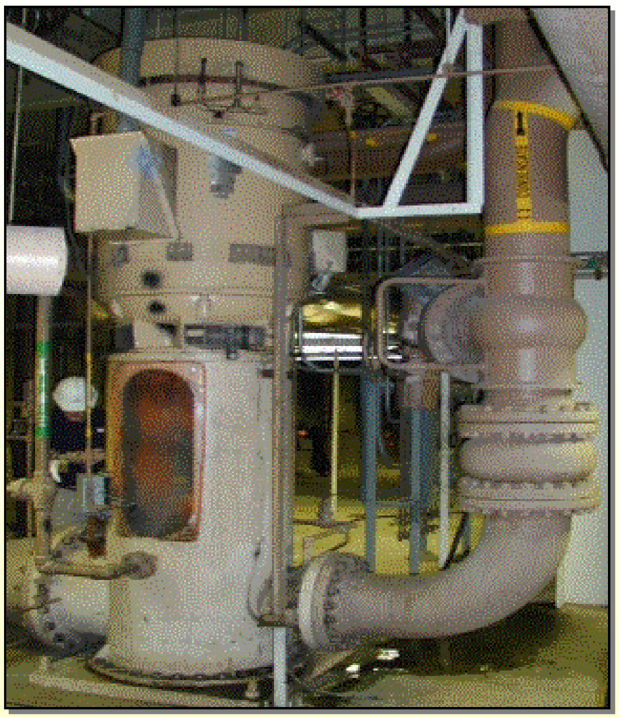
Alignment Process

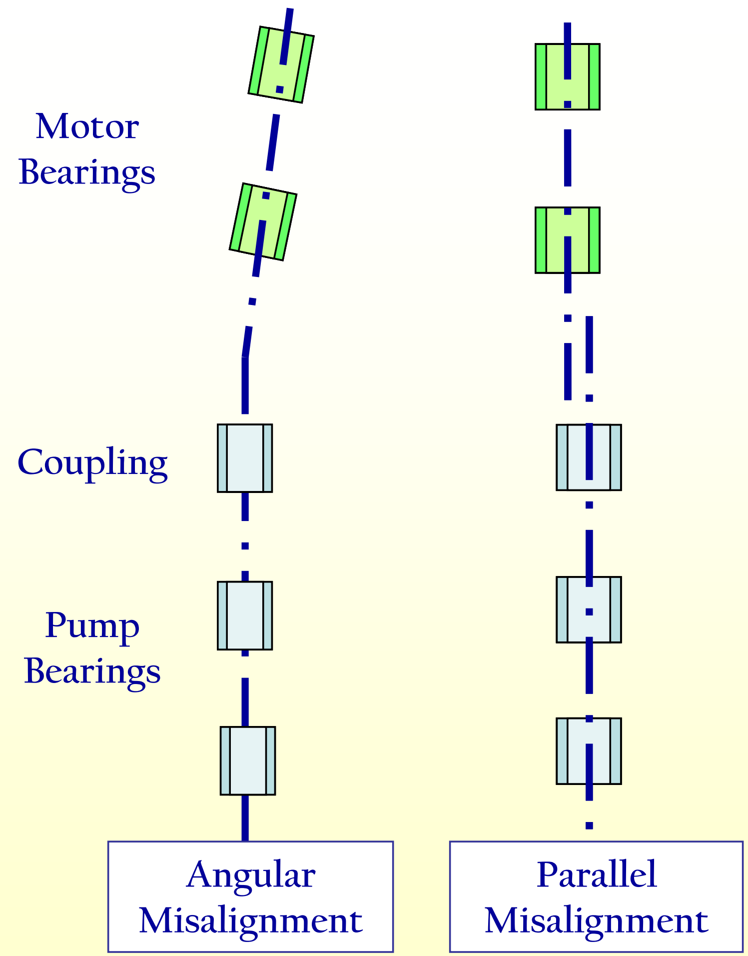
If The Motor Is Good…

- The motor base is flat and perpendicular to the bearing centerline
- Use the motor shaft as an indicator of where the motor bearing centerline is [i.e. use the motor as an alignment tool]

Vertical Alignment Process

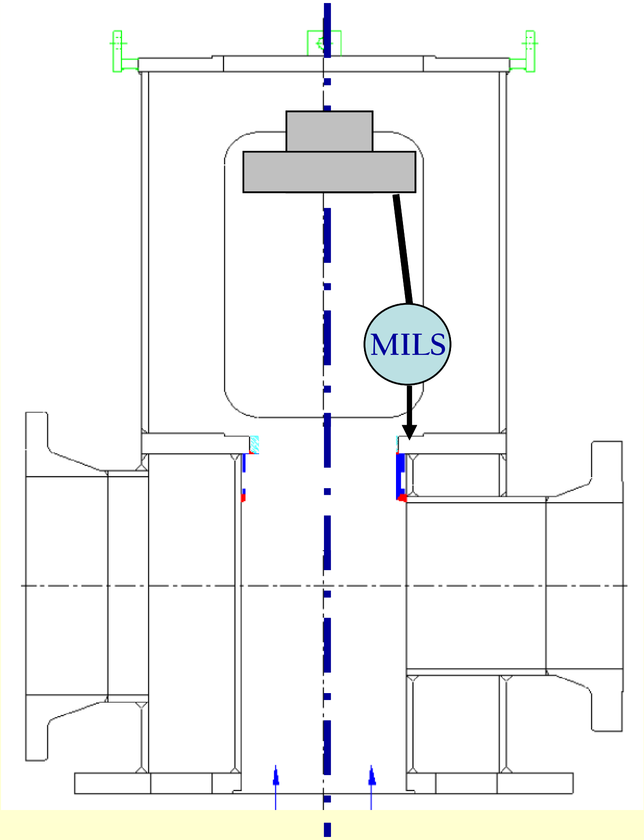
Types of Misalignment





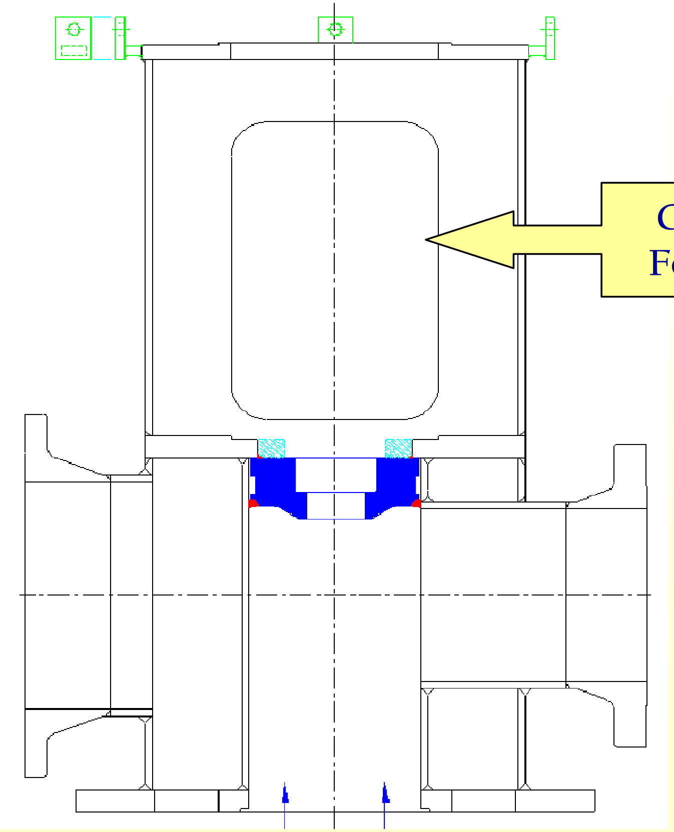
Vertical Alignment Process

Angular Alignment Check



Vertical Alignment Process

Alternate Angular Alignment Check



Check Gap at Four Locations

Vertical Alignment Process

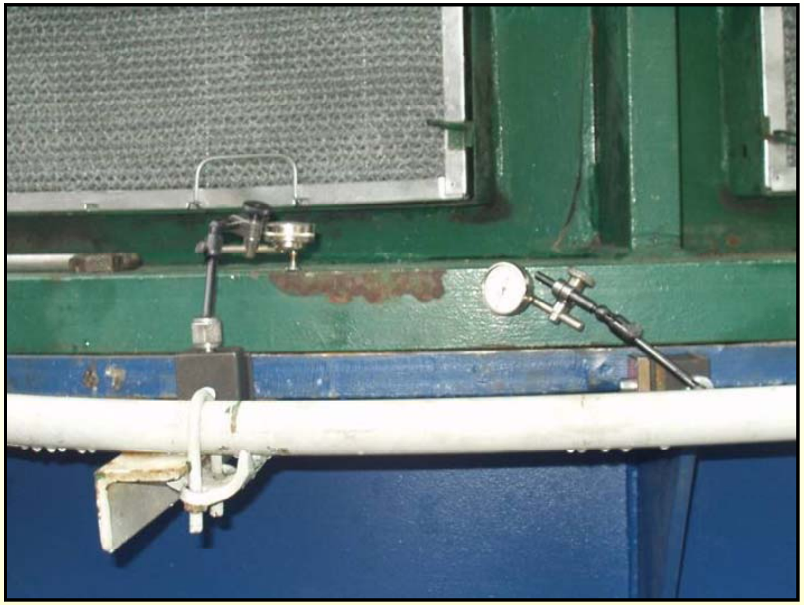
Angular Alignment Check

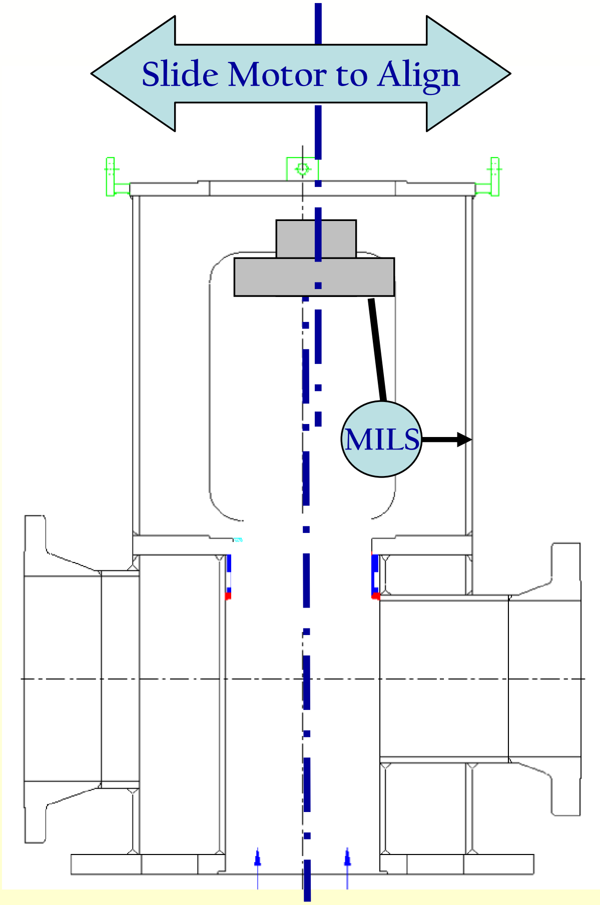
Options to correct:

- Re-machine discharge head
- Re-machine motor base
- Shim between motor and base

Vertical Alignment Process

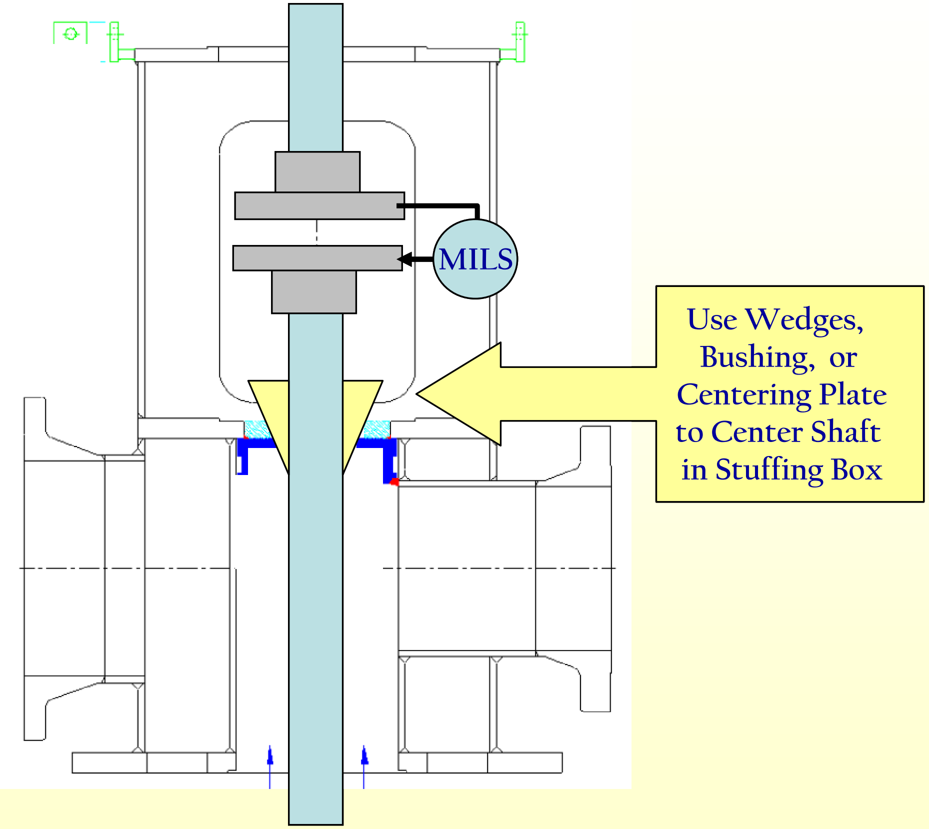
Parallel Alignment Check





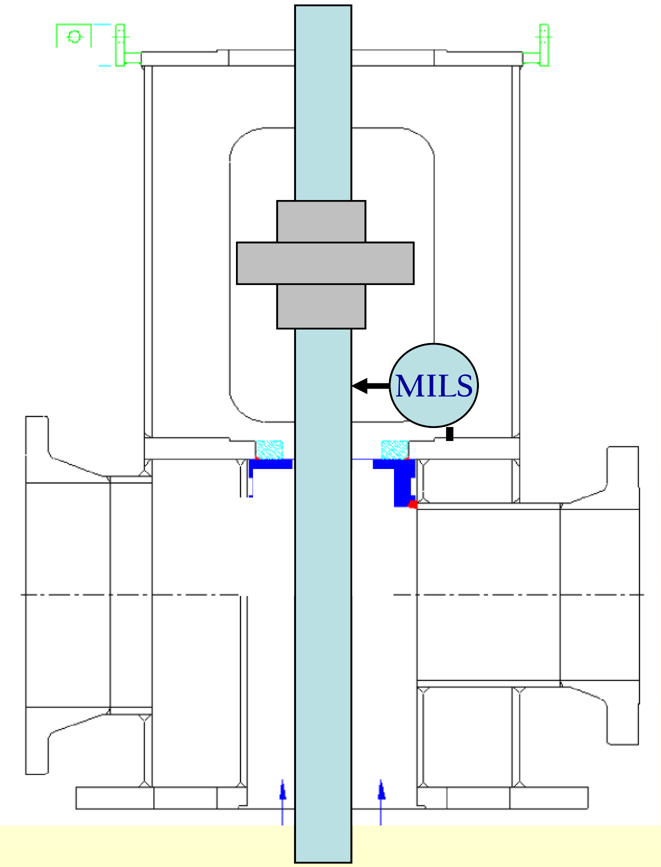
Vertical Alignment Process

Alternate Parallel Alignment Check



Vertical Alignment Process

Shaft Runout Check



Vertical Alignment Process

Conclusions

Need to check alignment of vertical pumps and motors:

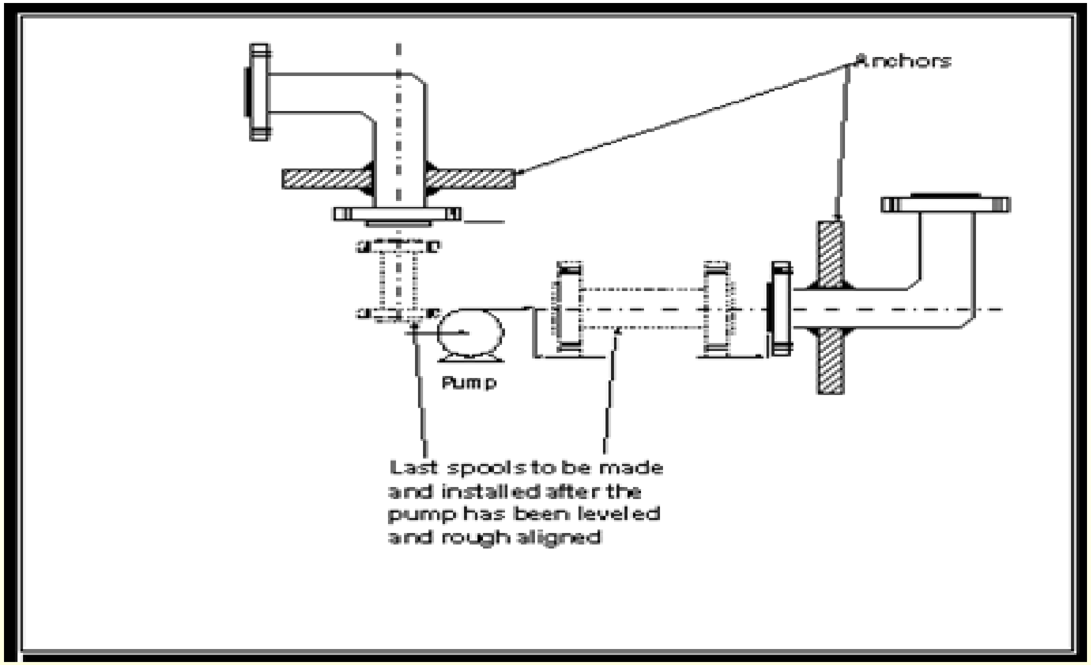
- Angular [many skip]
- Parallel [some skip]

Consider adding register fits/truth bands when doing repairs

Check vibration as final check

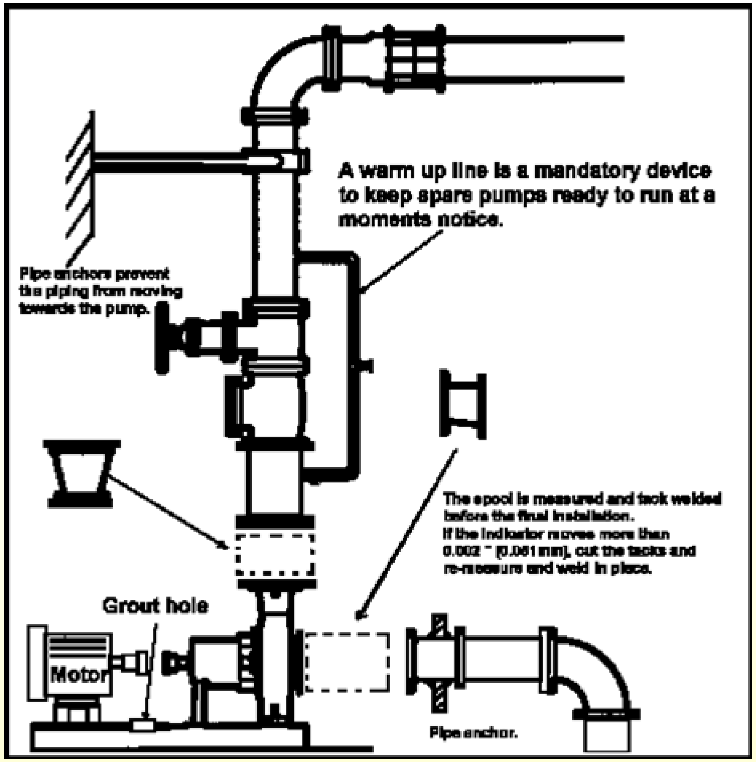
- Top of motor two directions
- Shaft above and below coupling
  - Phase across coupling
- Other points as applicable

Pump-to-Piping Alignment



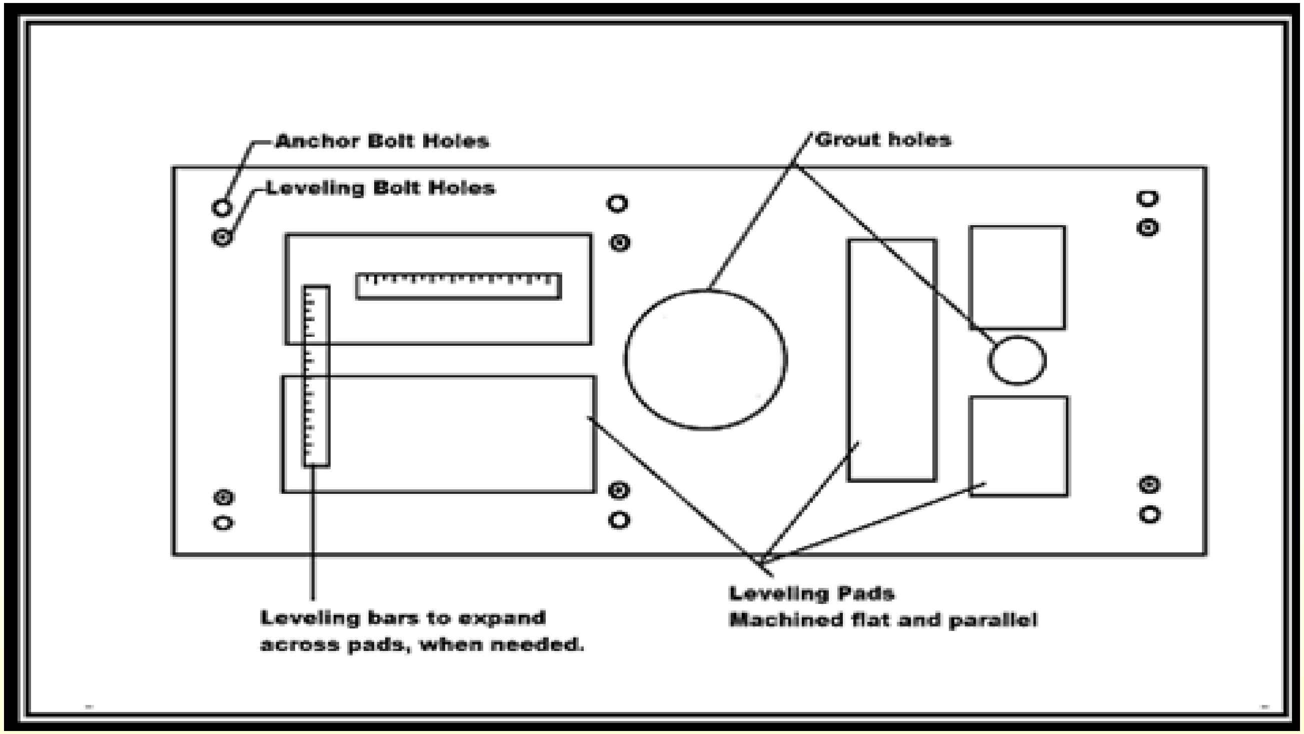
Step 1: *Pre-installation* stage (pump may not have even arrived to site yet) – anchor the main piping properly. Leave room for the final spool pieces (to be made later) by the pump.

Pump-to-Piping Alignment



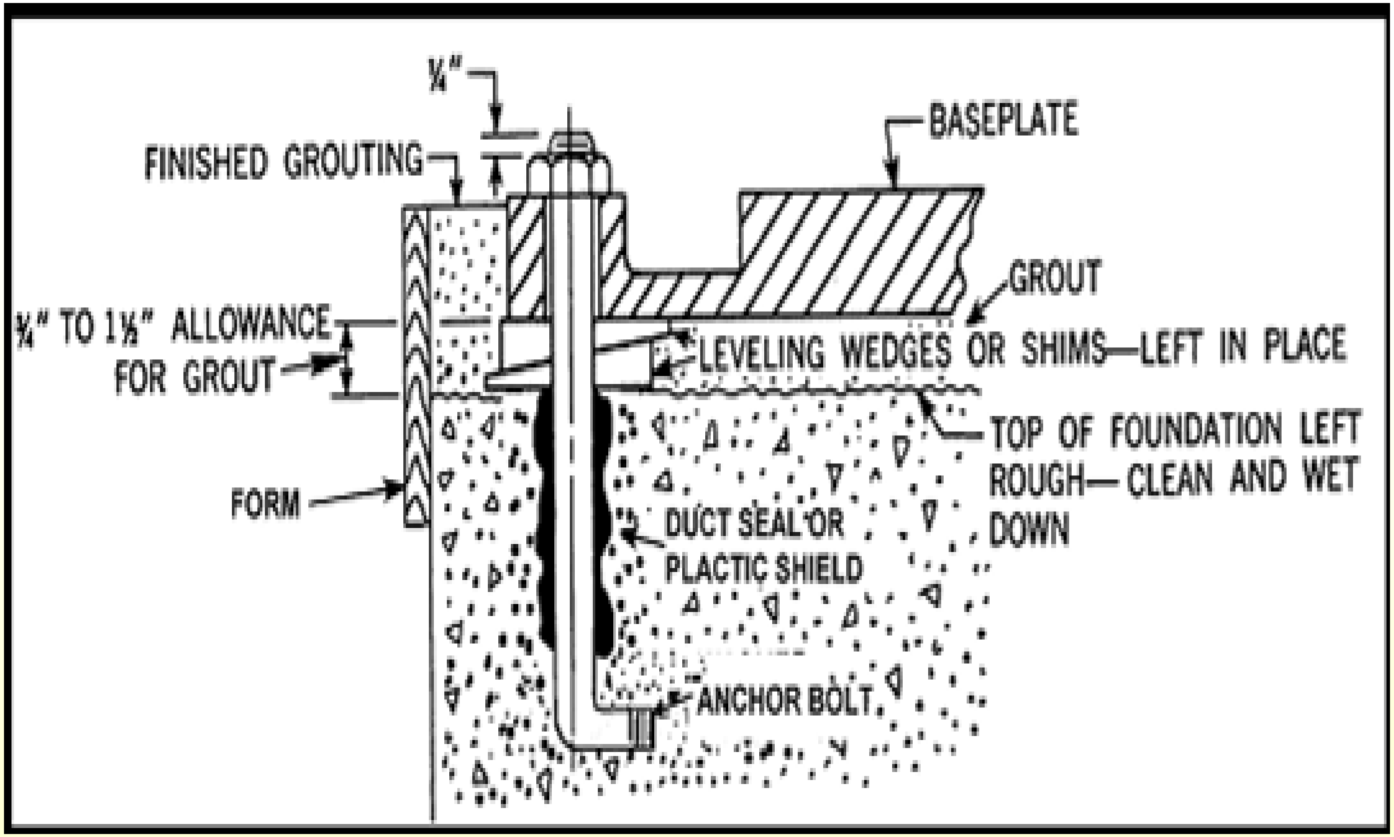
<u>Step 2</u>: Rough alignment phase. Pump has arrived. Position it and make spool pieces.

Pump-to-Piping Alignment



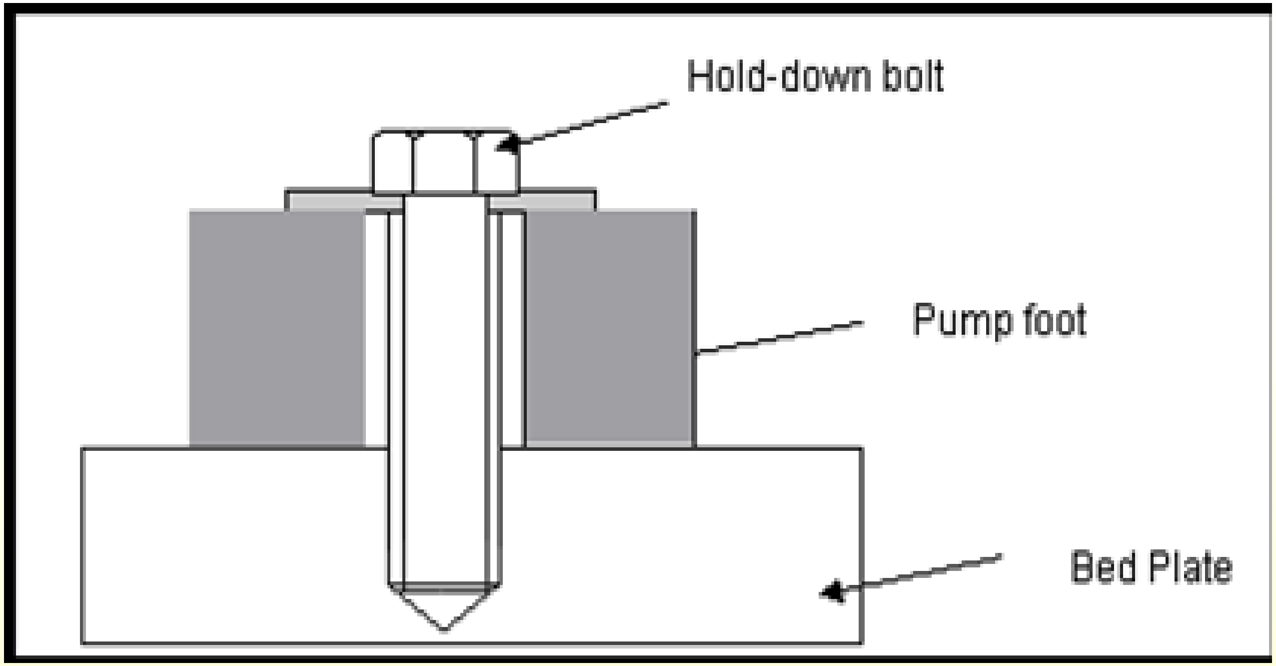
<u>Step 3</u>: Remove all equipment. Level the baseplate to 0.025” from end to end. Clean up and get ready for the grouting.

Pump-to-Piping Alignment



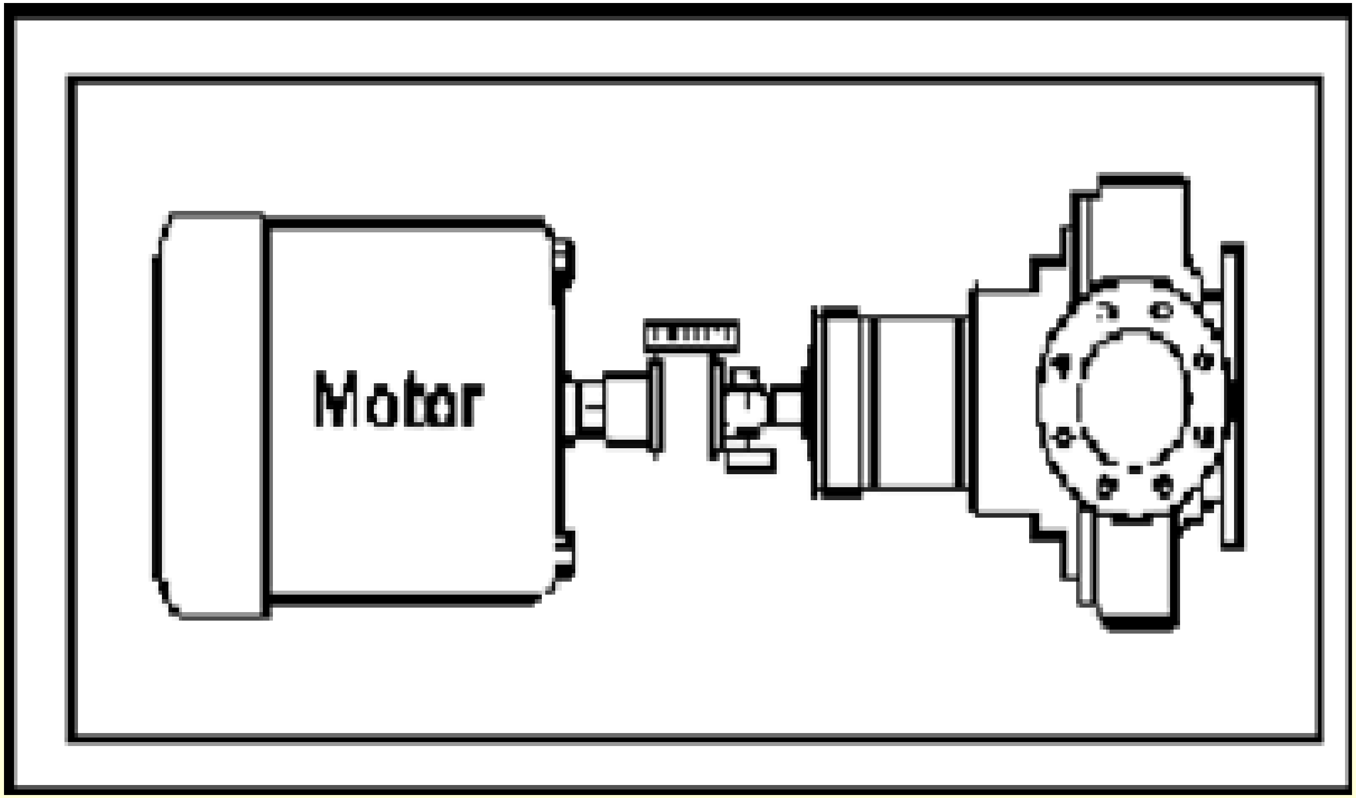
Step 4: Grouting phase.

Pump-to-Piping Alignment



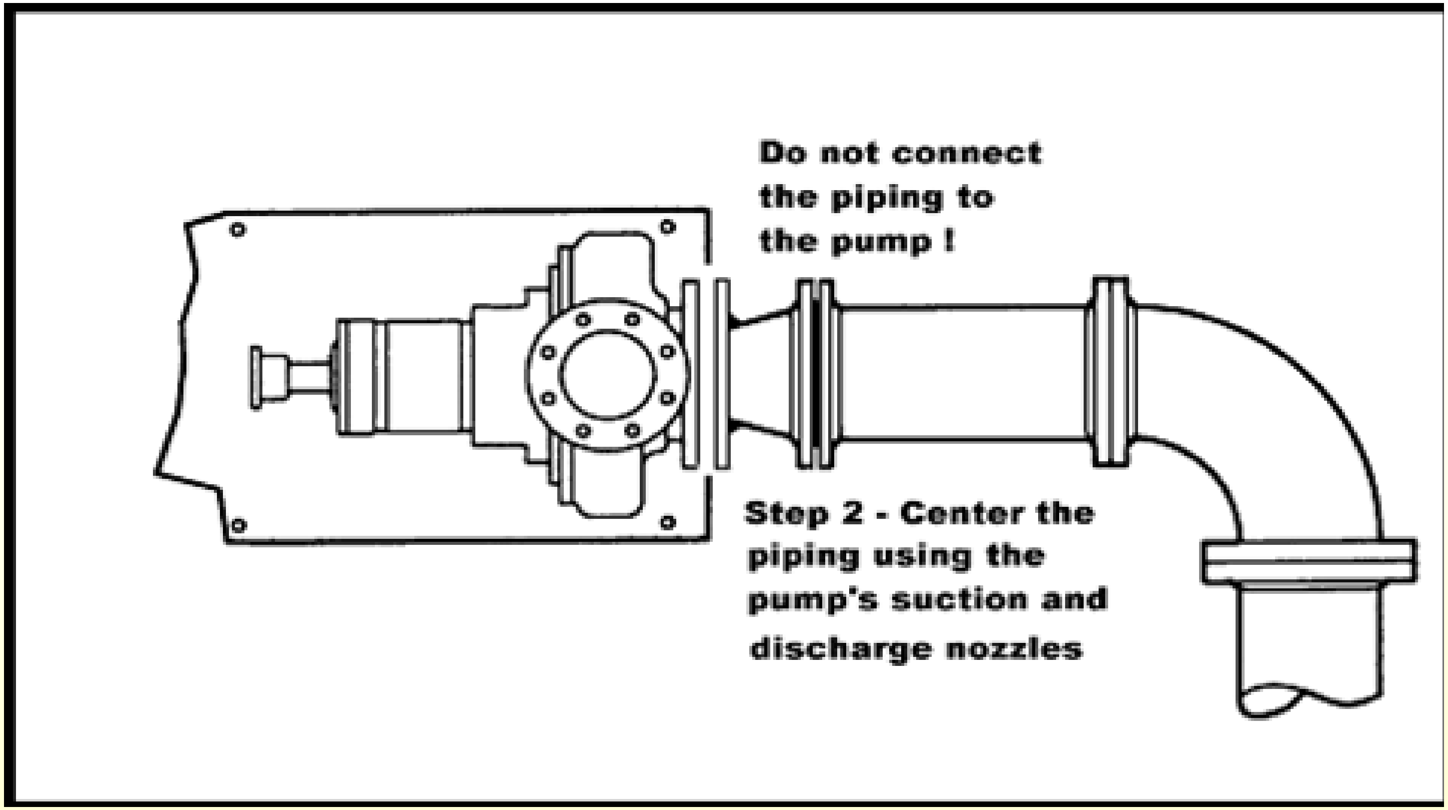
Step 5: Reinstall the pump and a motor on a baseplate. Inspect and make sure nothing is binding up.

Pump-to-Piping Alignment



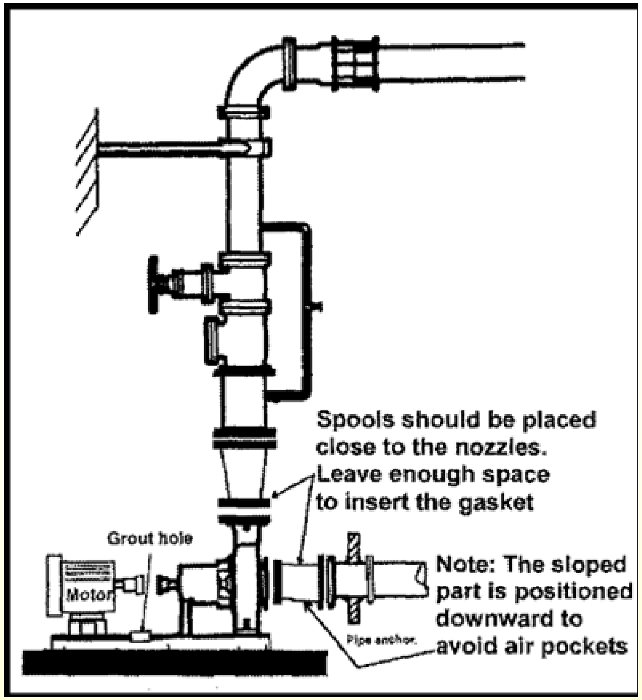
Step 6: Rough align pump to a motor.

Pump-to-Piping Alignment



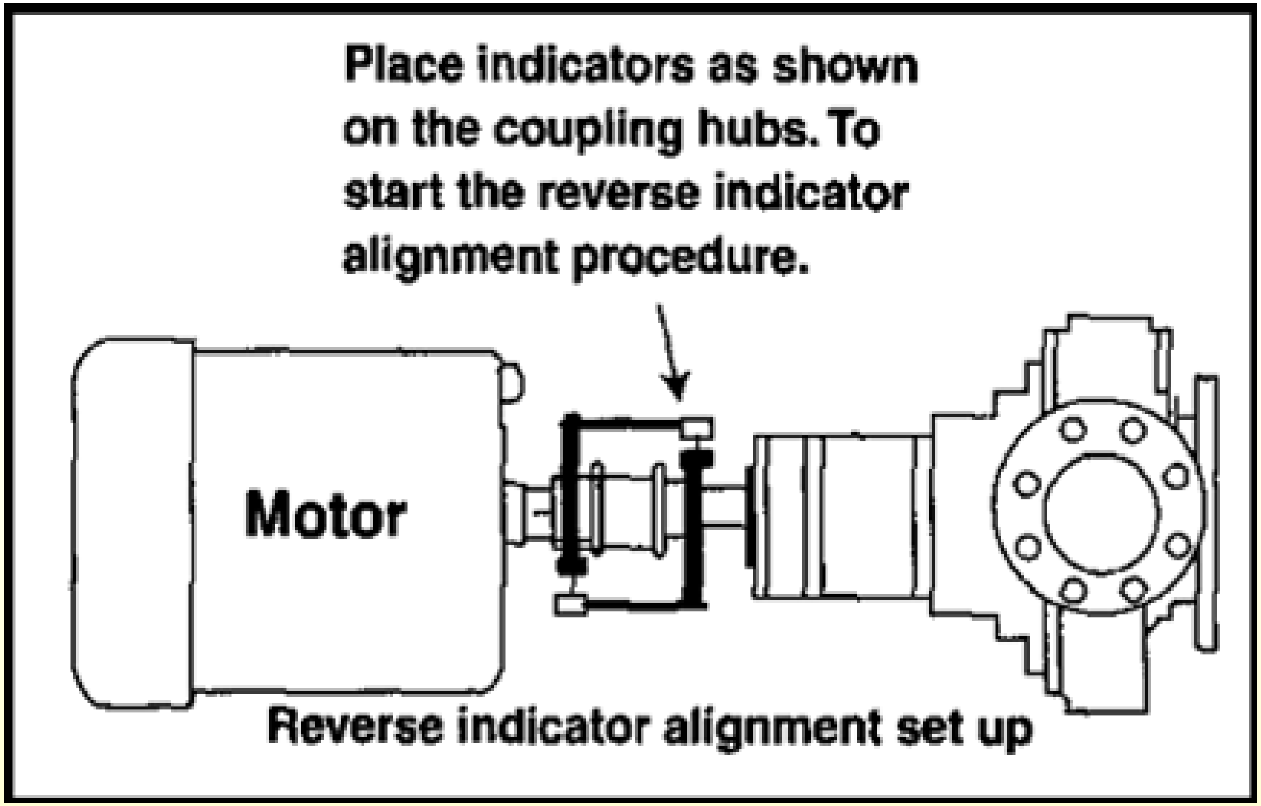
Step 7: Make up final spool pieces.

Pump-to-Piping Alignment



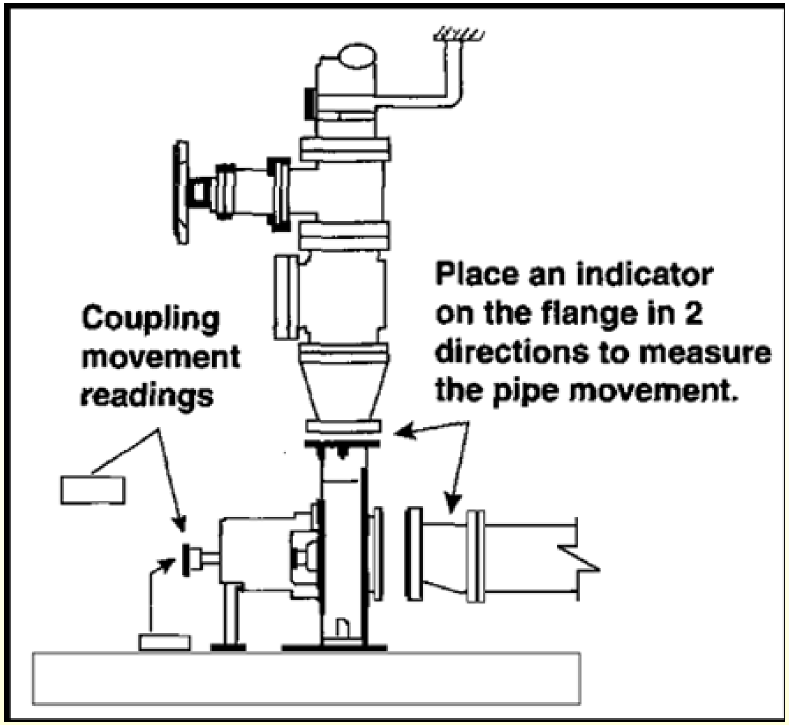
Step 8: Install the spool pieces between piping and a pump. Leave gaps (1/16" – 1/8") for the gaskets. This gap is the only distance the piping will be pulled during final bolting, and stresses will be minimal. (The pump will thank you for that).

Pump-to-Piping Alignment



Step 9: Final alignment of a motor to pump.

Pump-to-Piping Alignment



Step 10: Final piping verification. Unbolt the pump from the driver. Loosen up piping bolts and retighten. Indicator should not move more then 0.002”. Otherwise modify, adjust or remake spool pieces.

Thank You!

Pumping Machinery (guy with the accent)

Phone: (770) 310-0866

Web: www.PumpingMachinery.com

Mancini Consulting Services (tall Italian – no accent)

Phone: (215) 348-8580

e-mail: michaelcmancini@msn.com