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U.S. DEPARTMENT OF ENERGY

Pumping System Assessment

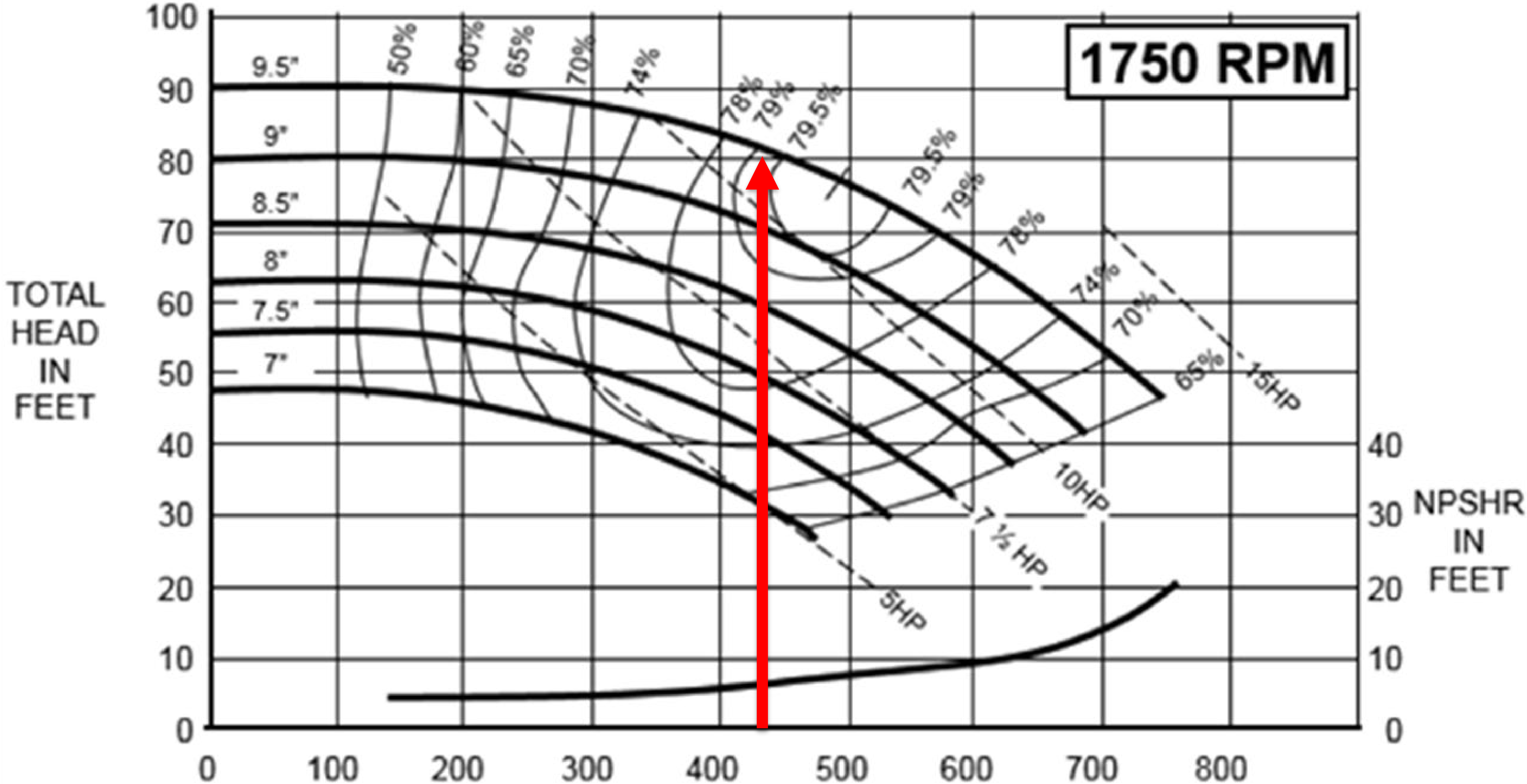
Week 5: Expanded Discussions

Finding efficiency for rpm changes

- Pump curve is for 1750 rpm operation
- Pump speed measured at 1783 rpm
- Measured flow is 442 gpm at 1783 rpm
- What is the pump efficiency at 1783 rpm?
- Pump curve is the only efficiency information you have
- Flow at 1750 rpm = 442 x (1750/1783) = 434 gpm
- Refer to the pump curve

Typical Single Stage Pump Curve

Efficiency = 79.1%



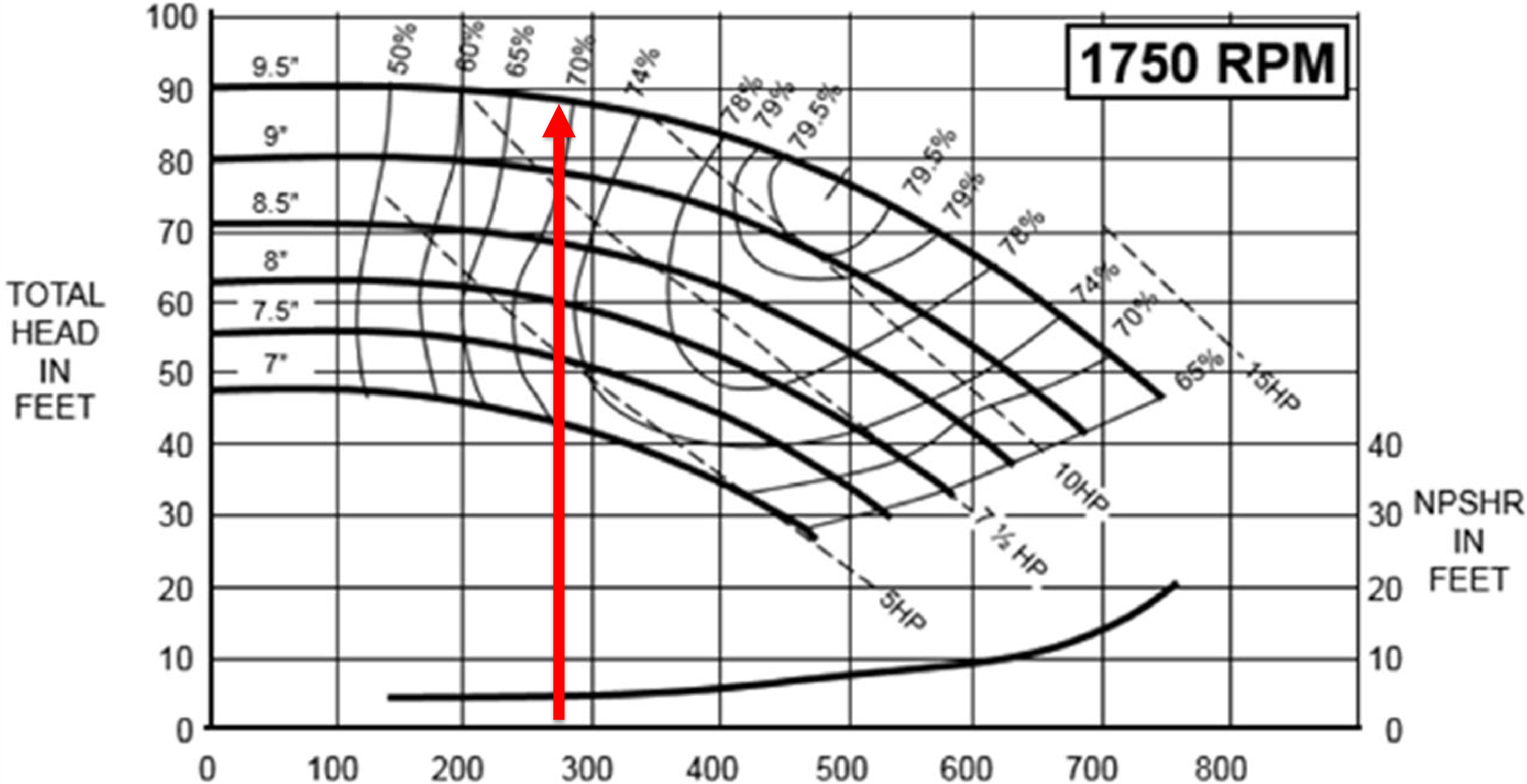
Flow = 434 gpm

Finding efficiency for rpm changes

- Pump curve is for 1750 rpm operation
- Pump speed measured at 1220 rpm
- Measured flow is 195 gpm
- What is the pump efficiency at 1220 rpm?
- Flow at 1750 rpm = 195 x (1750/1220) = 280 gpm
- Refer to the pump curve

Typical Single Stage Pump Curve

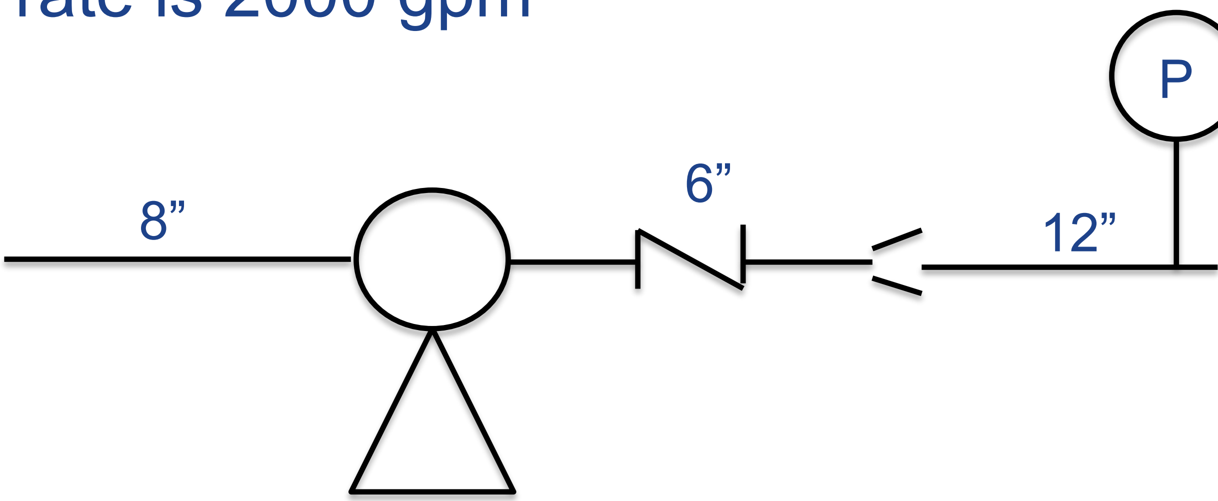
Efficiency = 69.0%



Flow = 280 gpm

Scaling Loss Coefficients

- Centrifugal pump with 8” suction and 6” discharge pipe
- On discharge there is a 6” check valve (k = 2)
- Downstream of the check valve the pipe expands to 12” diameter
- Discharge pressure gauge is located in the 12” diameter piping
- Flow rate is 2000 gpm



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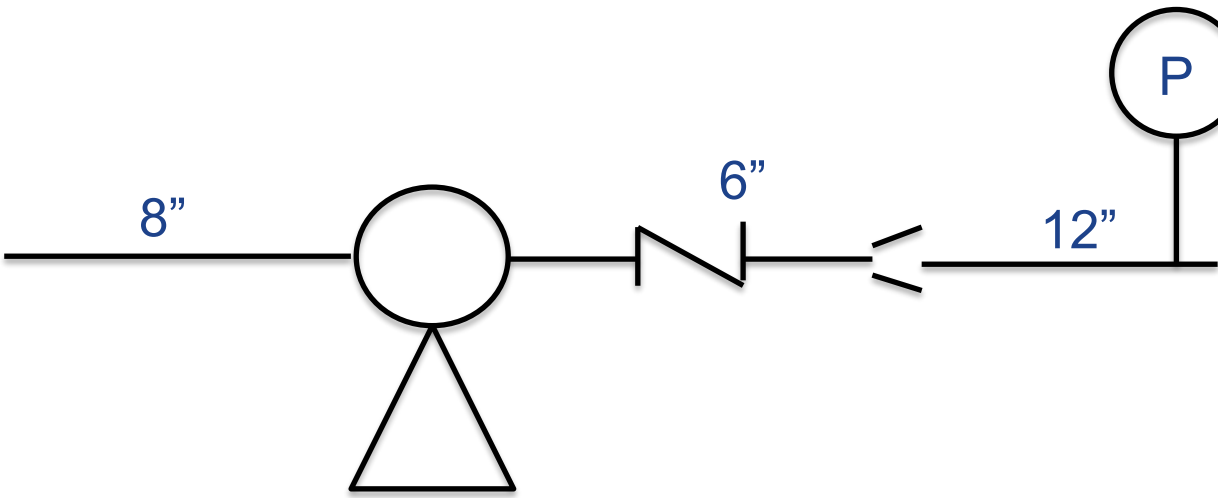
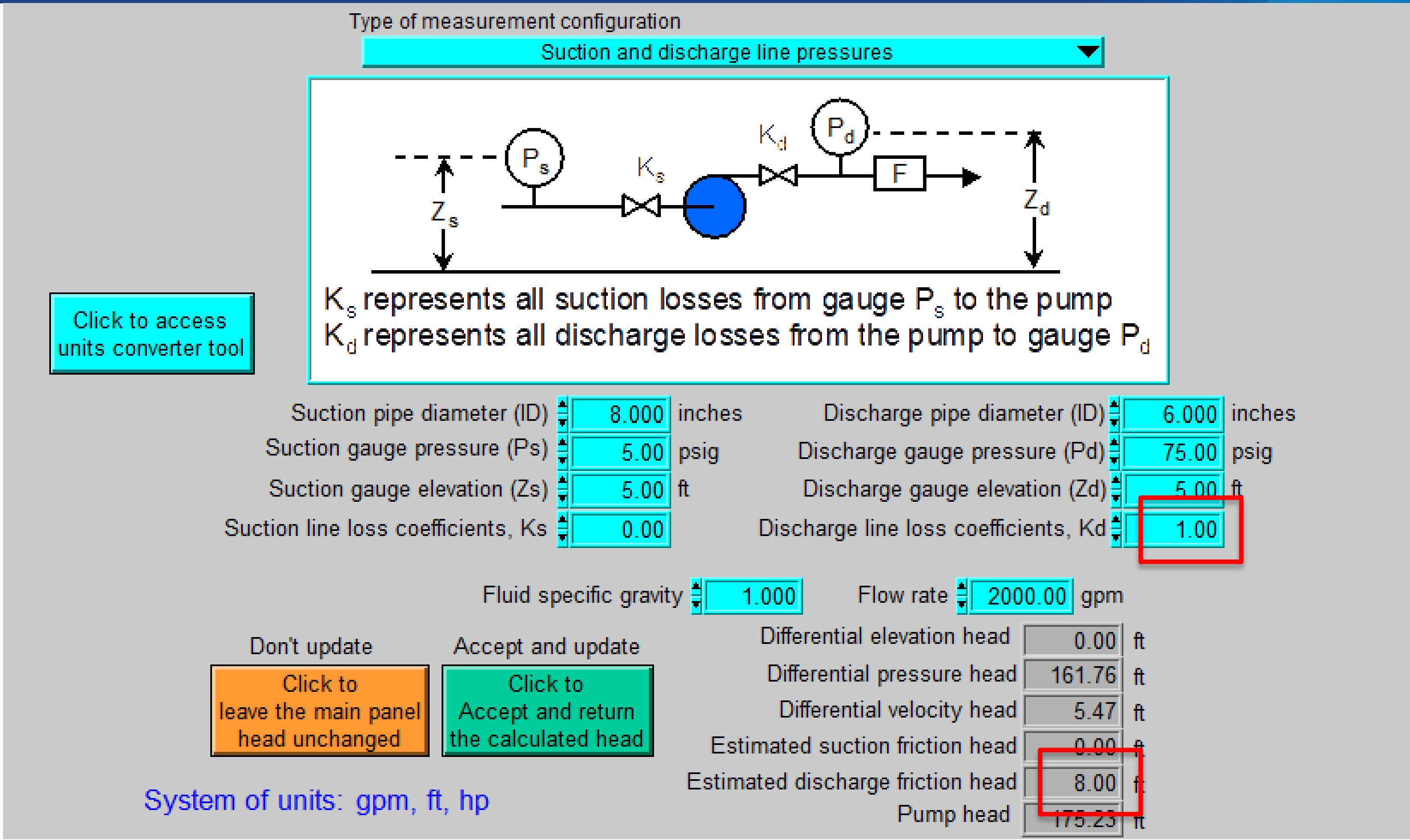
Scaling Loss Coefficients

- Friction loss for the check valve is = $k(V^2/2g)$ where $k = 2$ and V is the average velocity in the 6" diameter pipe
- Use the pump head calculator to calculate the velocity head (I'm lazy!)



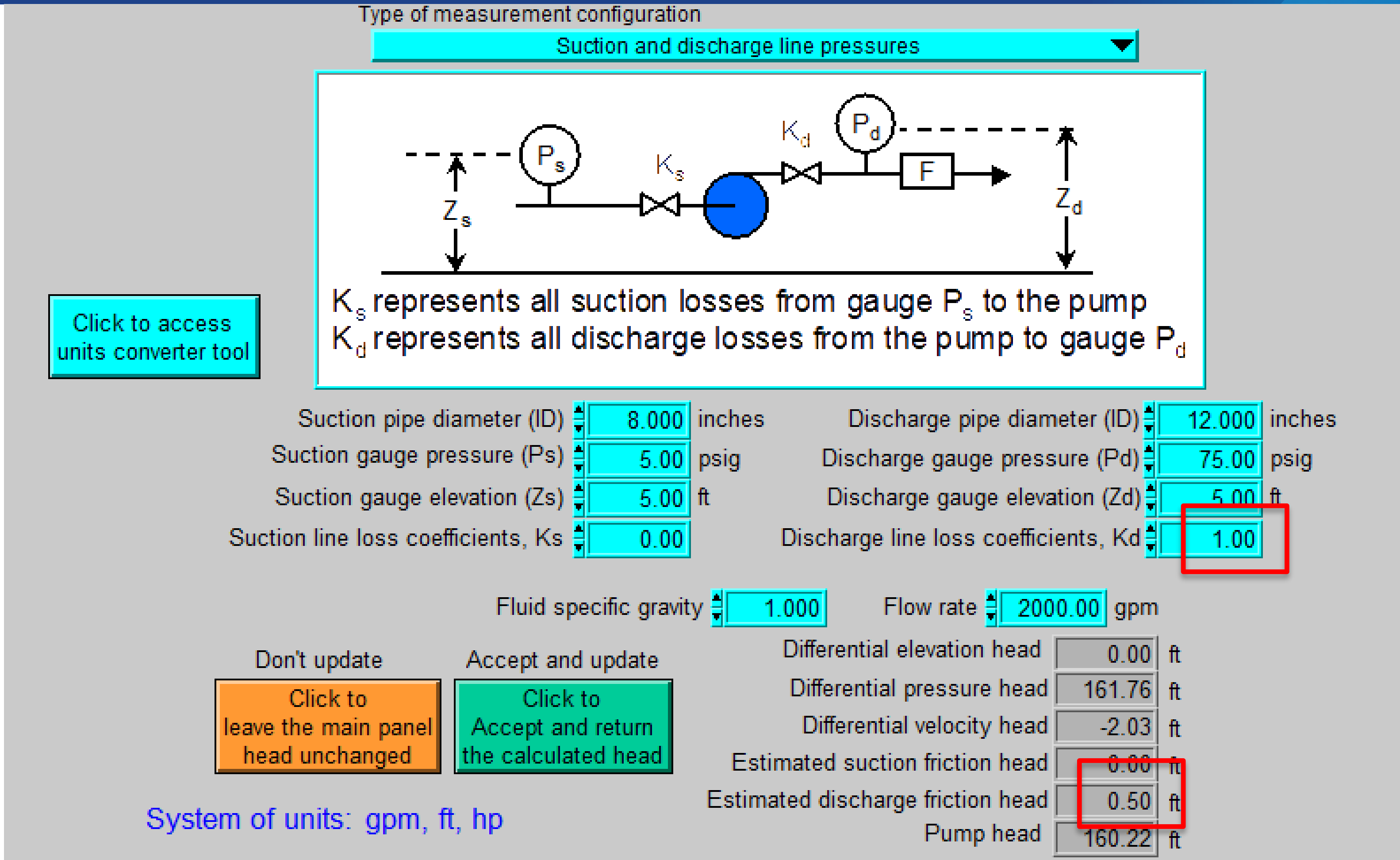
Scaling Loss Coefficients



2000 gpm in a 6” pipe: $V^2/2g$ = 8.00 ft

Check valve k = 2: valve friction loss = 2 x 8.00 = 16.00 ft

Scaling Loss Coefficients

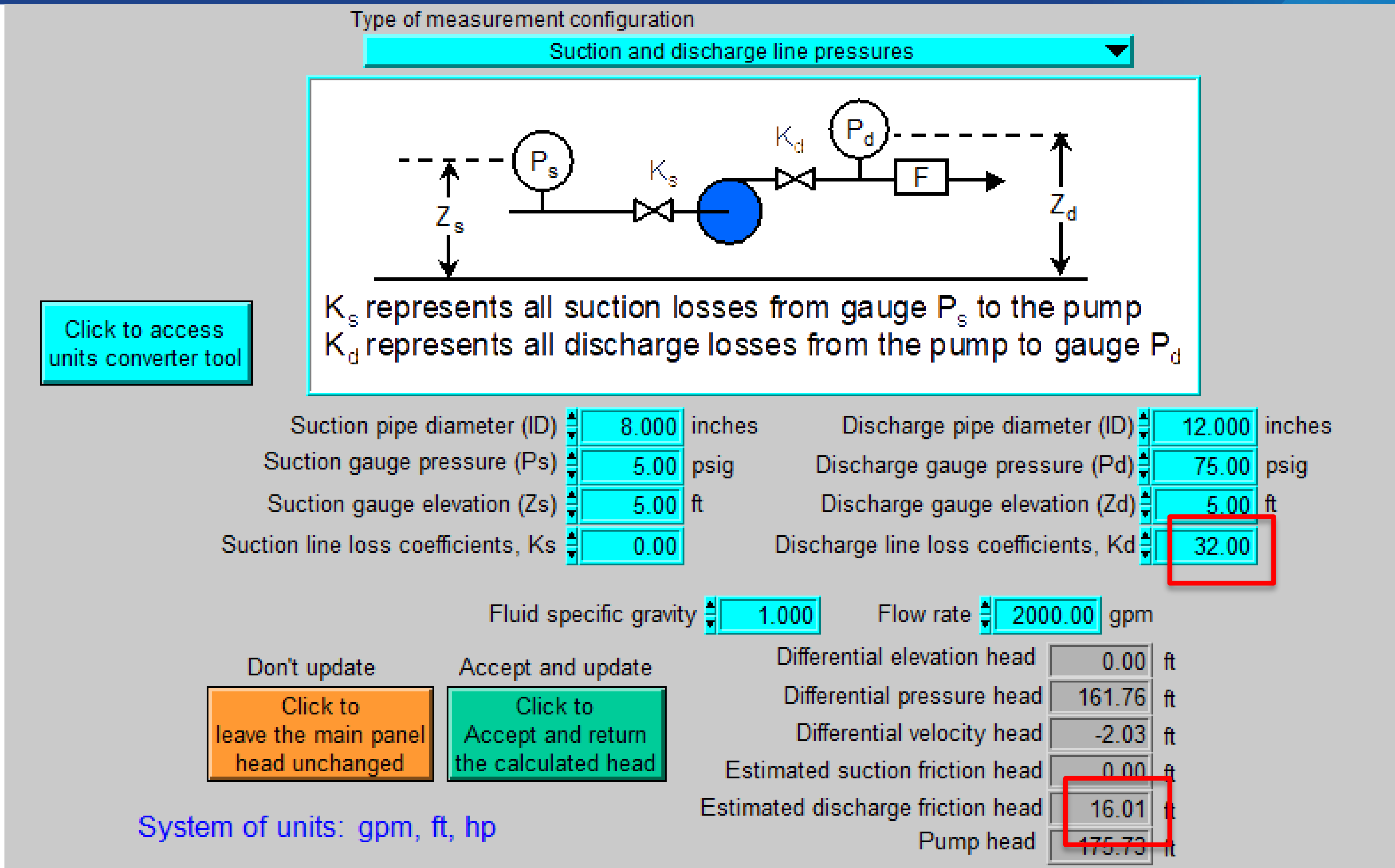


2000 gpm in a 12” pipe: $V^2/2g = 0.50$ ft

Check valve k = 2 (in 6” pipe)

valve friction loss = $2 \times (12/6)^4 \times 0.50 = 16.00$ ft

Scaling Loss Coefficients



2000 gpm in a 12” pipe: $V^2/2g = 0.50$ ft

Check valve k = 2 (in 6” pipe)

Scaled loss coefficient = $2 \times (12/6)^4 = 32$

Net Positive Suction Head

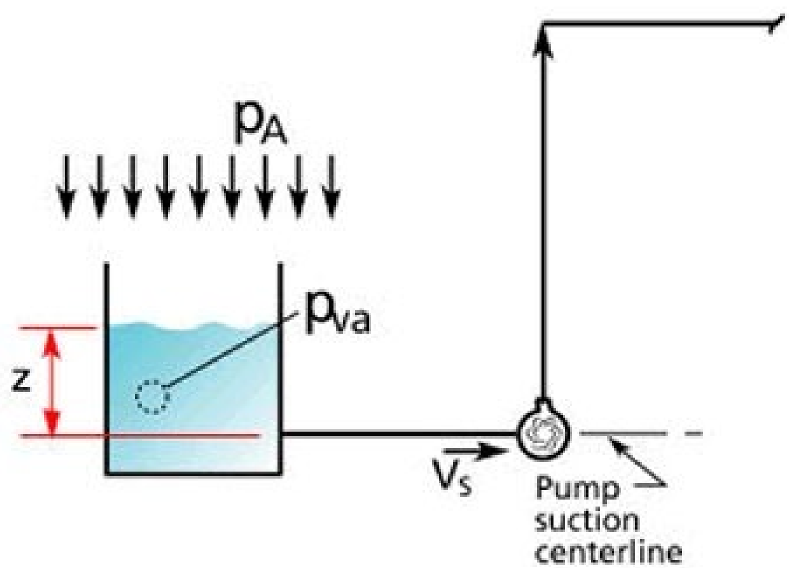
NPSHA:
Net Positive Suction Head Available

NPSHA = Total suction head (absolute) - fluid vapor pressure (absolute)

$$NPSHA = \frac{V_s^2}{2g} + \frac{2.31(P_s + P_a)}{s.g.} + Z_s - \frac{2.31P_v}{s.g.}$$

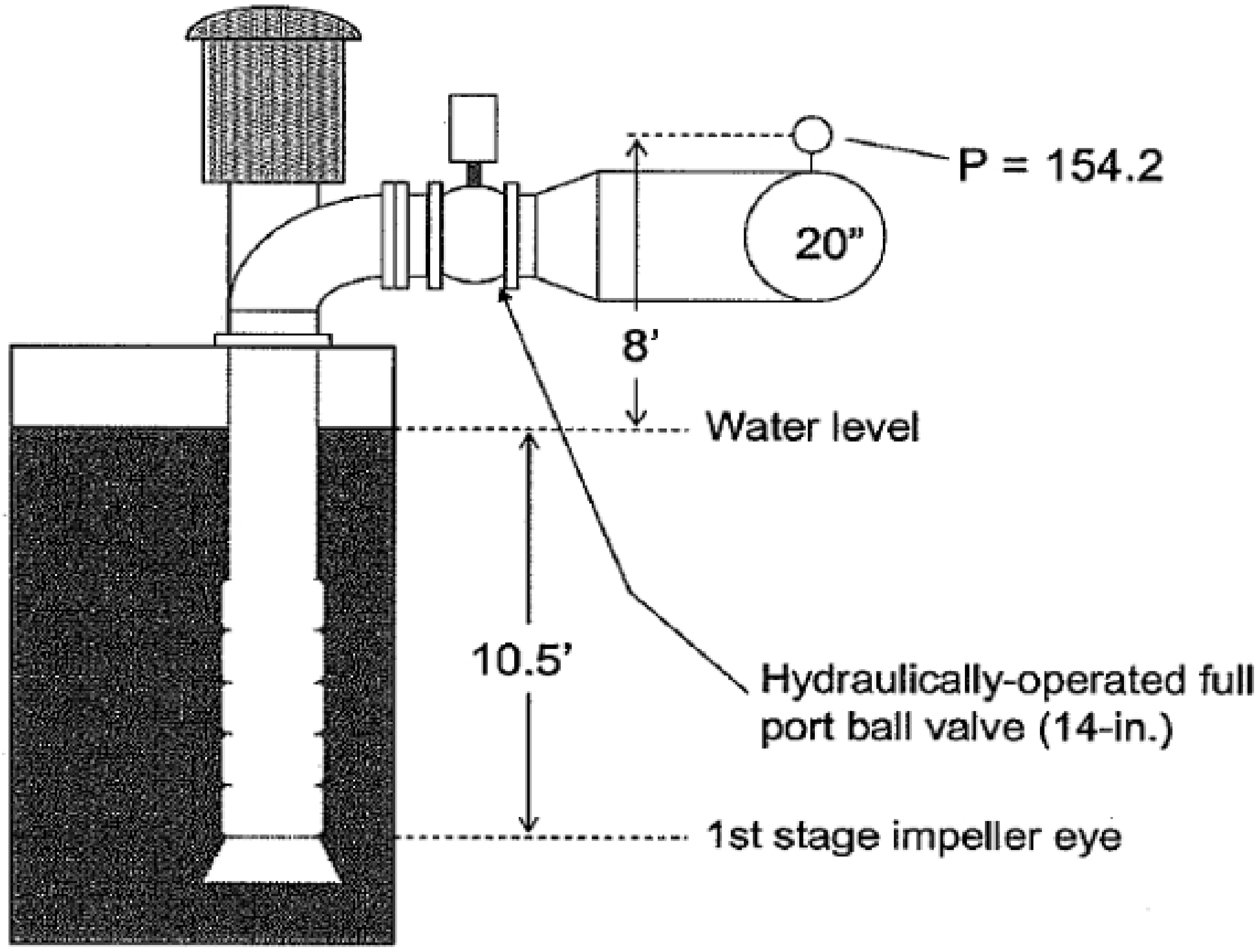
$$= \frac{V_s^2}{2g} + \frac{2.31(P_s + P_a - P_v)}{s.g.} + Z_s$$

V_s = velocity in suction (ft/sec)
P_s = suction gauge pressure (psig)
P_a = atmospheric pressure (psia)
P_v = fluid vapor pressure (psia)
Z_s = suction gauge elevation above pump datum (ft)
g = gravitational constant (32.174 ft/sec^2)
s.g. = fluid specific gravity (dimensionless)



2

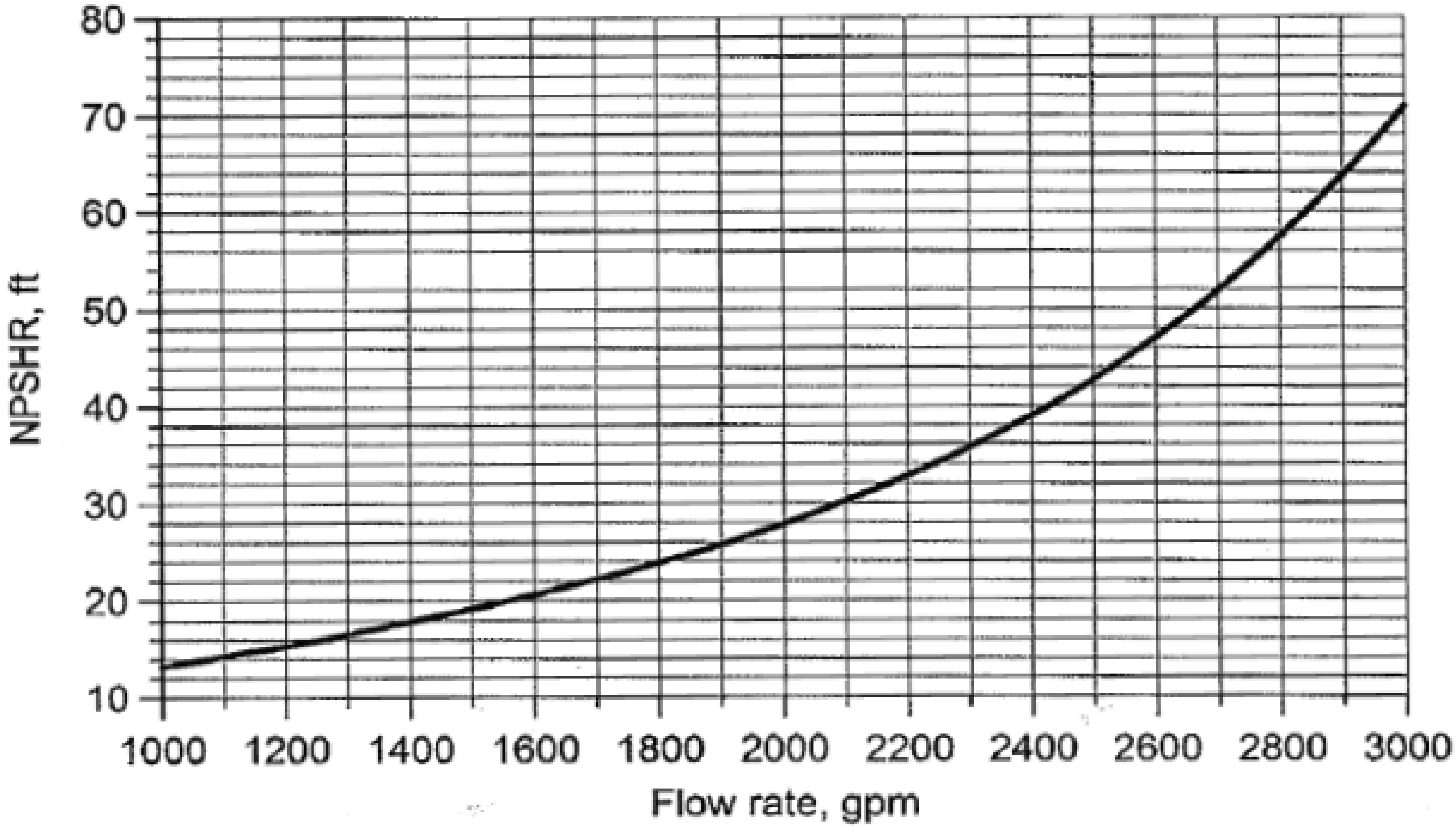
Finish water pump layout



NPSHR Curve for pump on previous slide

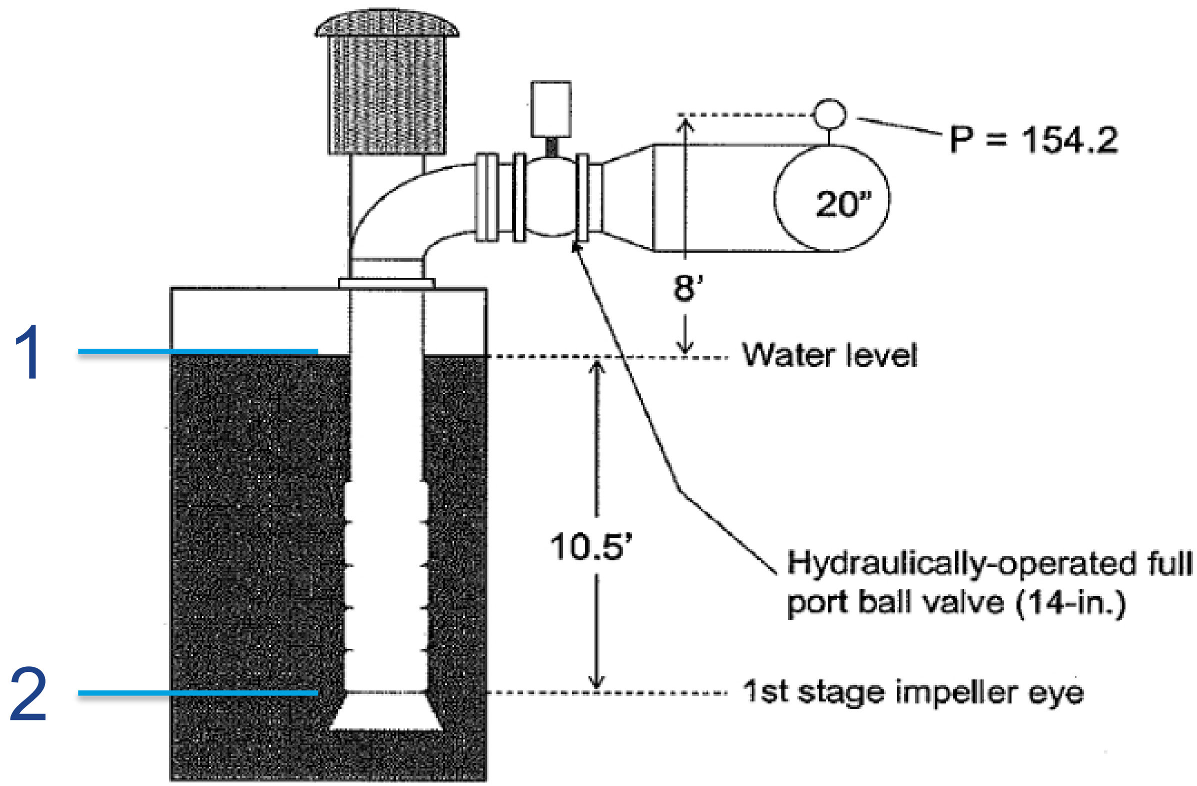
At what flow rate would NPSHR exceed NPSHA?

(Assume P_s = 14.7 psia and water temperature = 60 degrees F)



Finish water pump layout - NPSHA

- Want the total head at location 2
- Don't know P_2 and V_2, typically
- Easier to start at location 1, know P_1 = atmospheric pressure and $V_1 = 0$
- Then, $P_2 = P_1 + 10.5$ feet



Calculate NPSHA

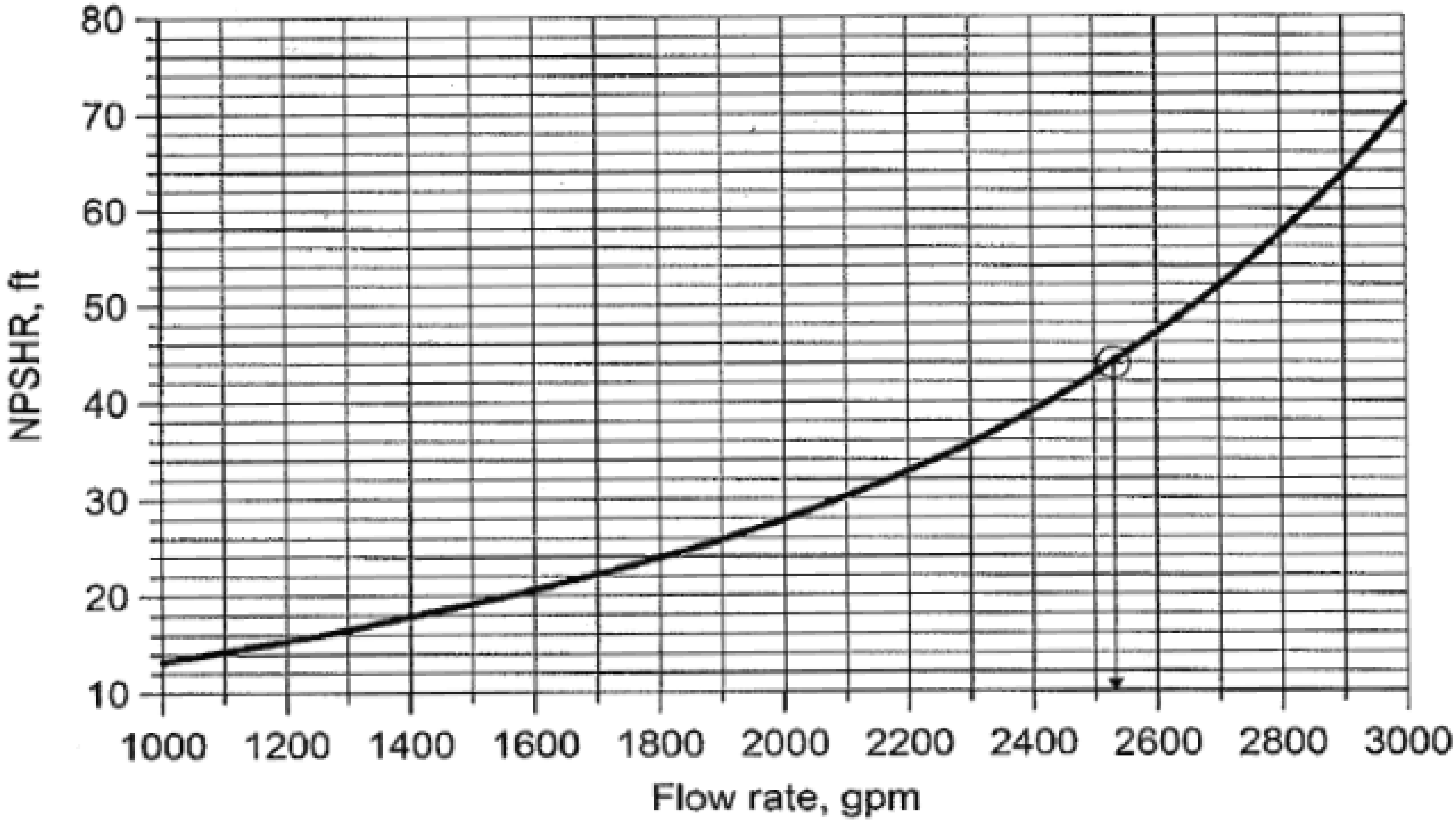
Water saturation vapor pressure at 60 F= 0.26 psia

Reference location for suction head determination is the water surface

$$\text{NPSHA} = \frac{V_s^2}{2g} + \frac{2.31\,(P_s + P_a - P_v)}{\text{s.g.}} + Z_s$$

$$\text{NPSHA} = \frac{0^2}{64.352} + \frac{2.31\,(0 + 14.7 - 0.26)}{1.00} + 10.5 = \boxed{43.9\text{ ft}}$$

Answer: NPSHR would exceed NPSHA at just over 2500 gpm



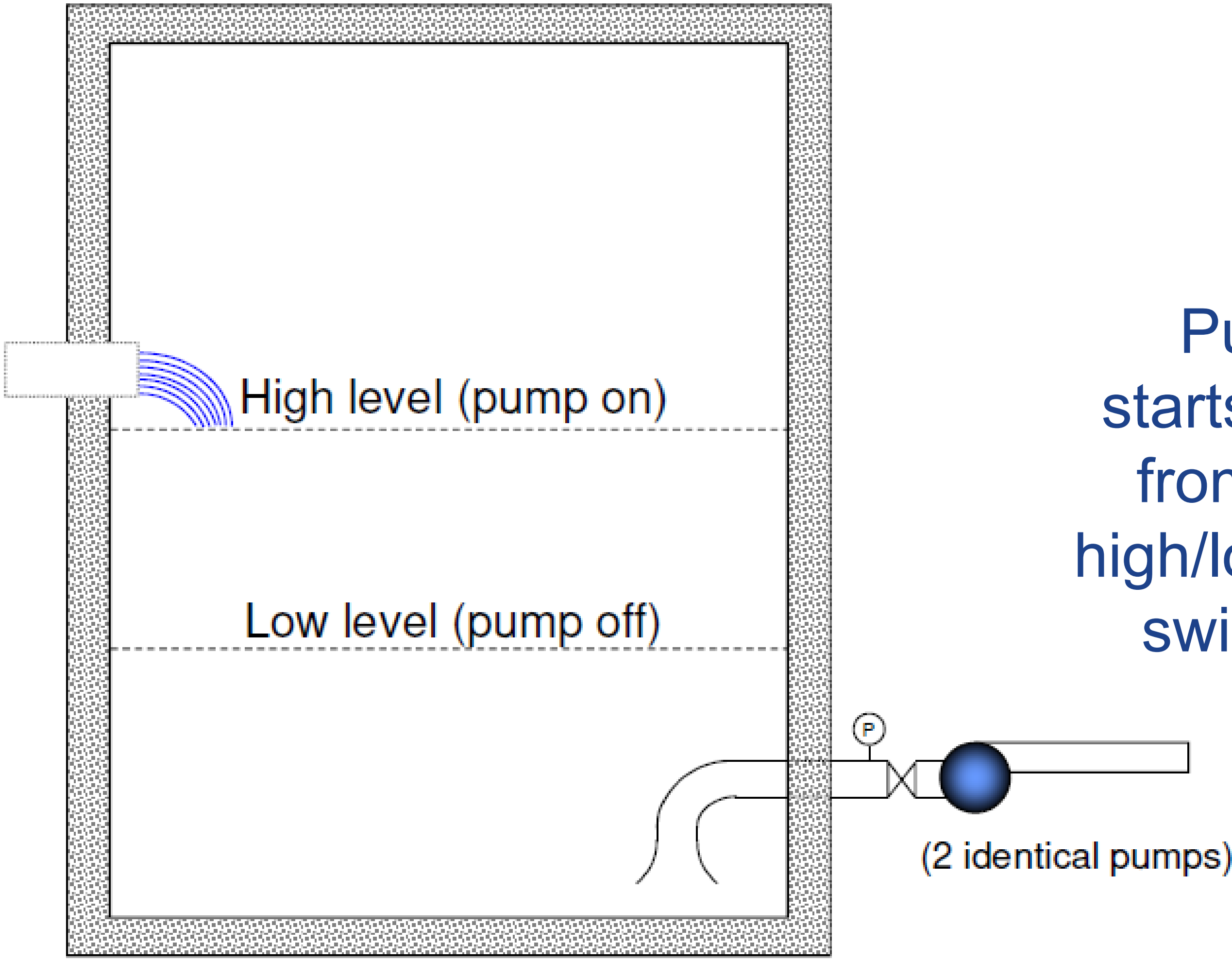
Flow estimation from suction pressure measurements

Using velocity and friction head components - two approaches

1. On-off transition
2. Comparison between running and non-running pump

A standard wastewater lift station

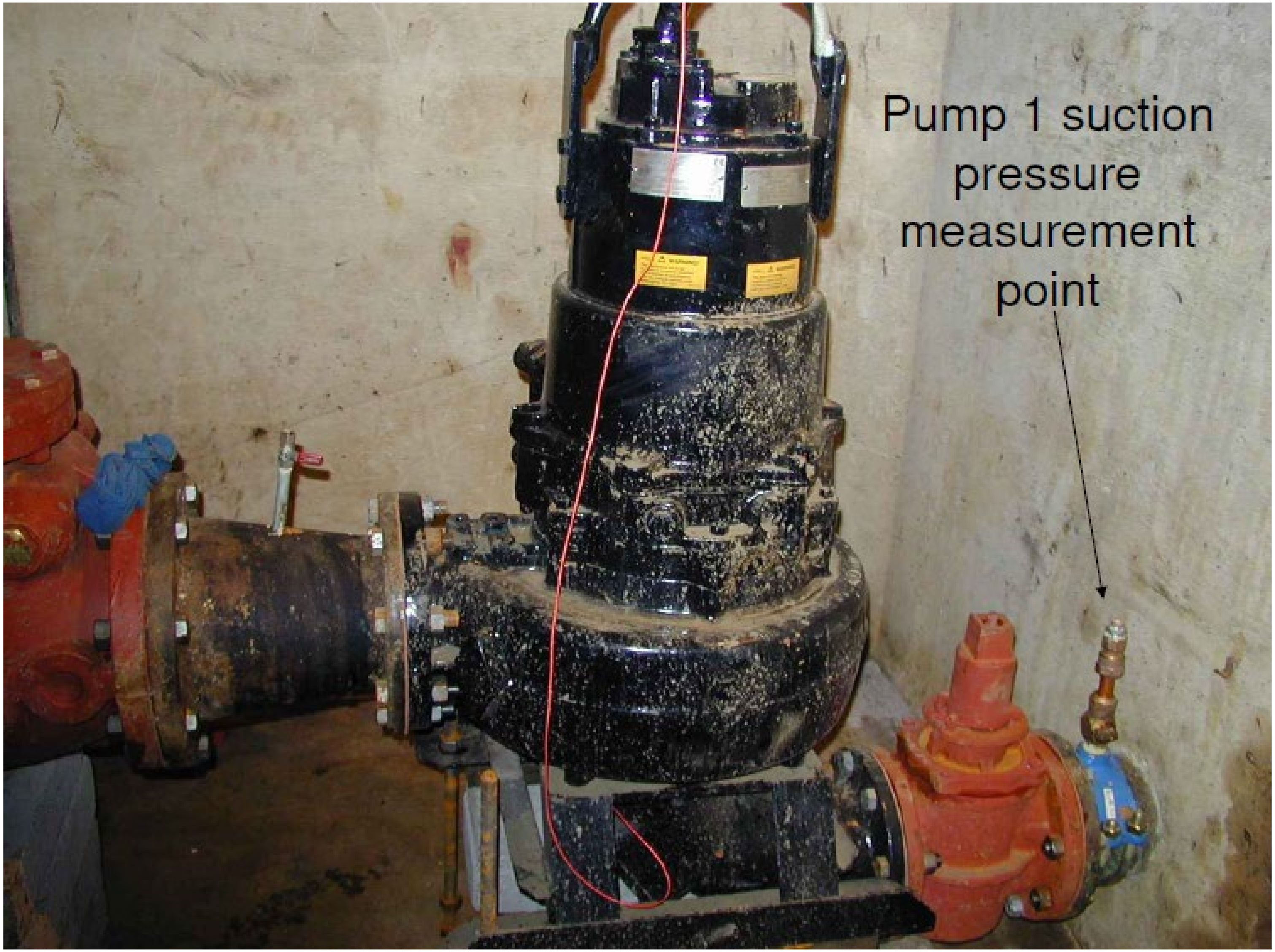
Continuous flow entering the lift station



Pump starts/stops from well high/low level switches

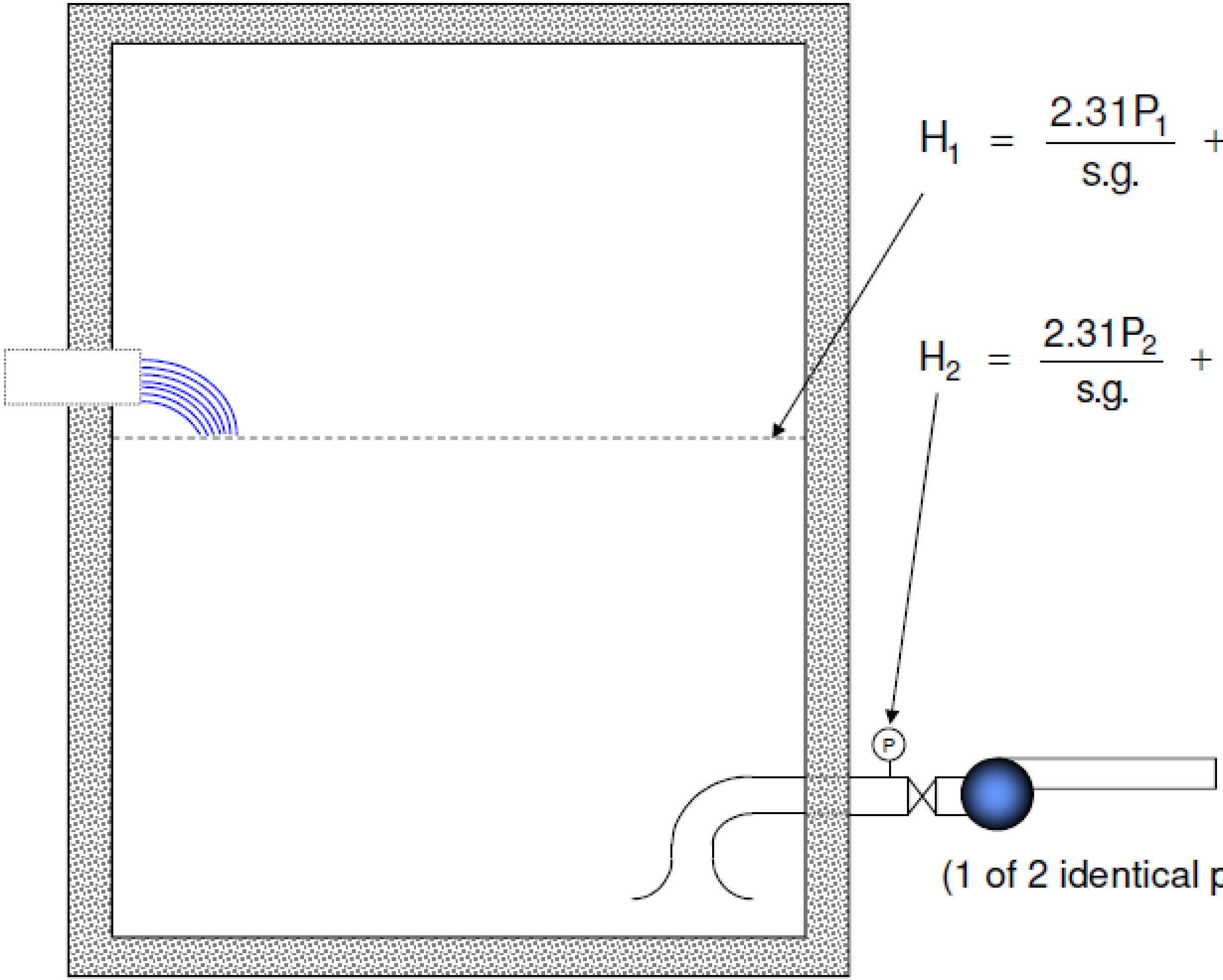
Incoming fluid spilling into the wet well

Pump 1 suction pressure measurement point



Total head (including friction) is the same at two points

(i.e., $H_1 = H_2$)

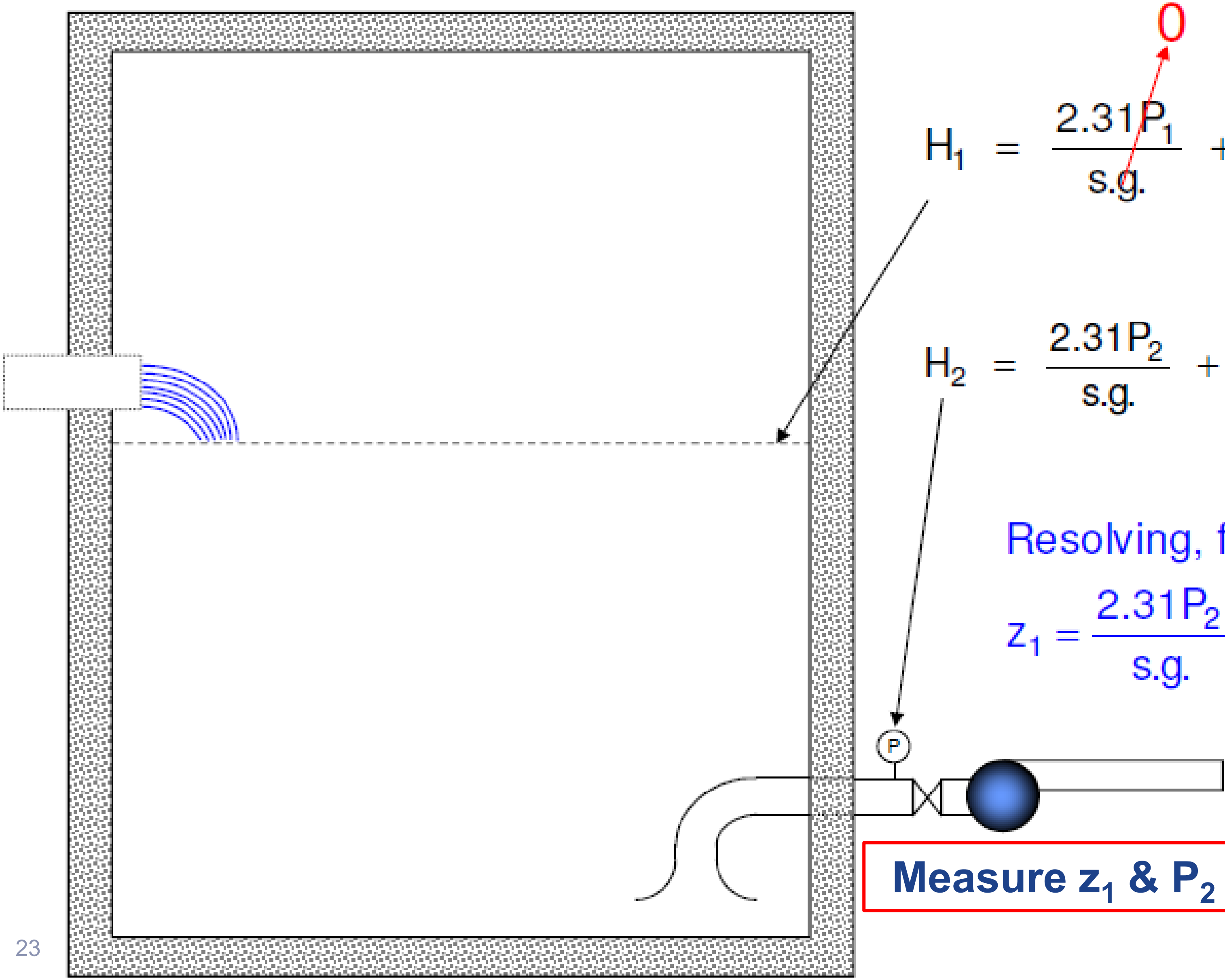


$$H_1 = \frac{2.31P_1}{s.g.} + z_1 + \frac{V_1^2}{2g}$$

$$H_2 = \frac{2.31P_2}{s.g.} + z_2 + \frac{V_2^2}{2g} + K\left(\frac{V_2^2}{2g}\right)$$

Frictional losses

We can relate terms and arrive at: Two knowns, one guess, and one unknown



$$H_1 = \cancelto{0}{\frac{2.31P_1}{s.g.}} + z_1 + \cancelto{0}{\frac{V_1^2}{2g}}$$

$$H_2 = \frac{2.31P_2}{s.g.} + \cancelto{0}{z_2} + \frac{V_2^2}{2g} + K\left(\frac{V_2^2}{2g}\right)$$

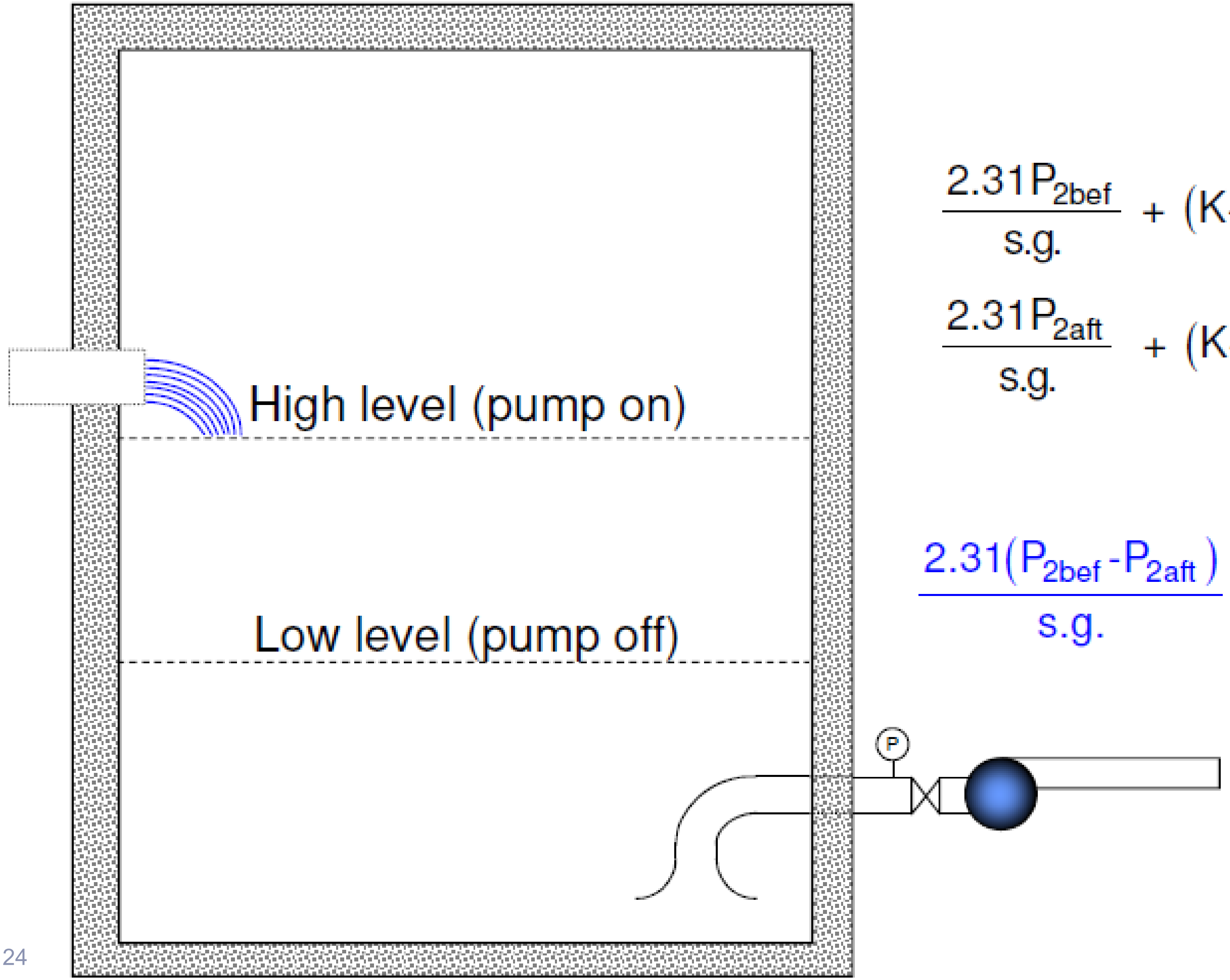
Resolving, for $H_1 = H_2$:

$$z_1 = \frac{2.31P_2}{s.g.} + (K+1)\left(\frac{V_2^2}{2g}\right)$$

Measure z_1 & P_2 and Calculate V_2

∴ if we can measure elevation and pressure, we can estimate velocity

An alternative perspective:
If level doesn't change instantaneously on pump start/stop……

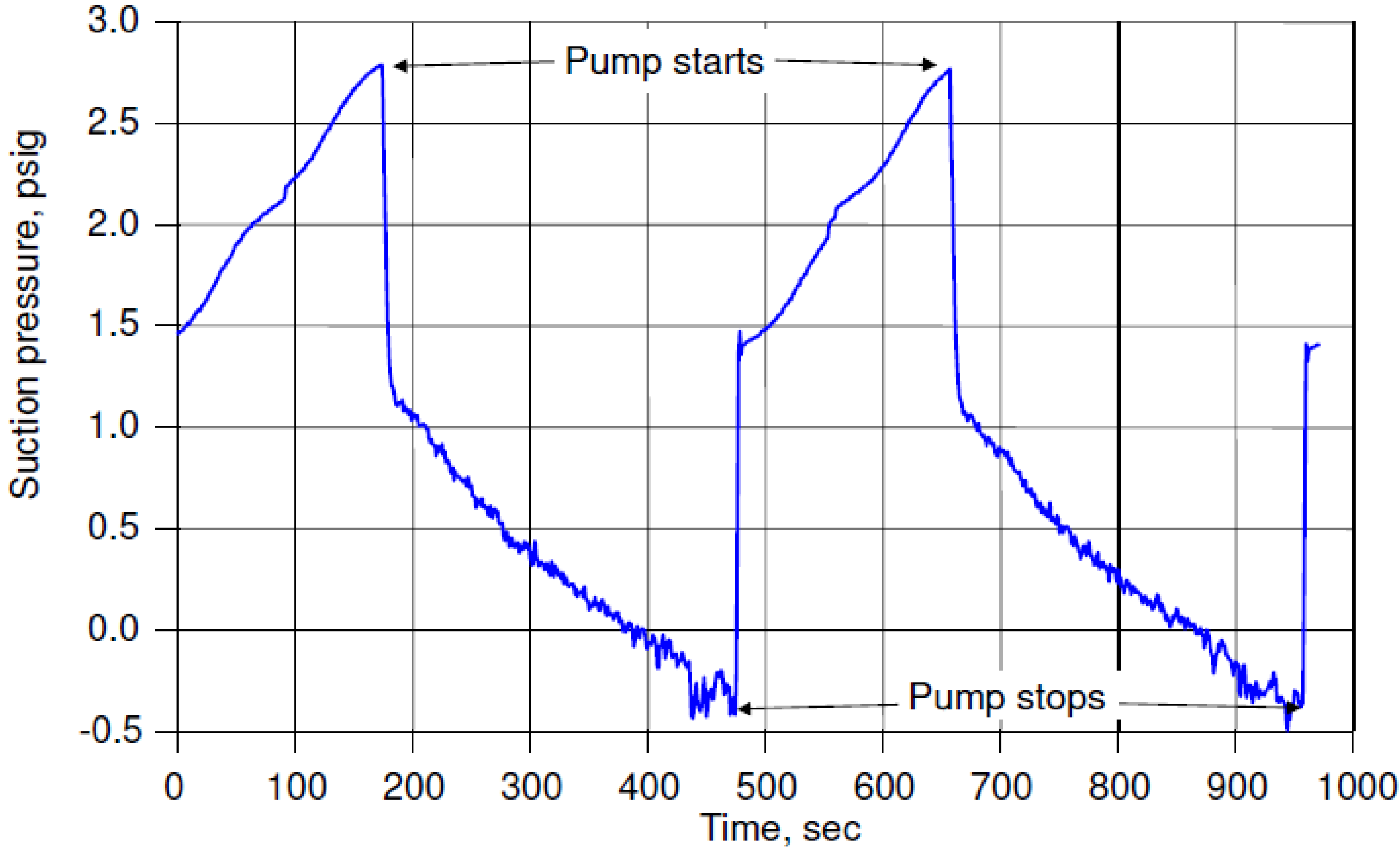


$$\frac{2.31P_{2bef}}{s.g.} + (K+1)\left(\frac{\cancel{V_{2bef}^2}^{\,0}}{2g}\right) = \frac{2.31P_{2aft}}{s.g.} + (K+1)\left(\frac{V_{2aft}^2}{2g}\right)$$

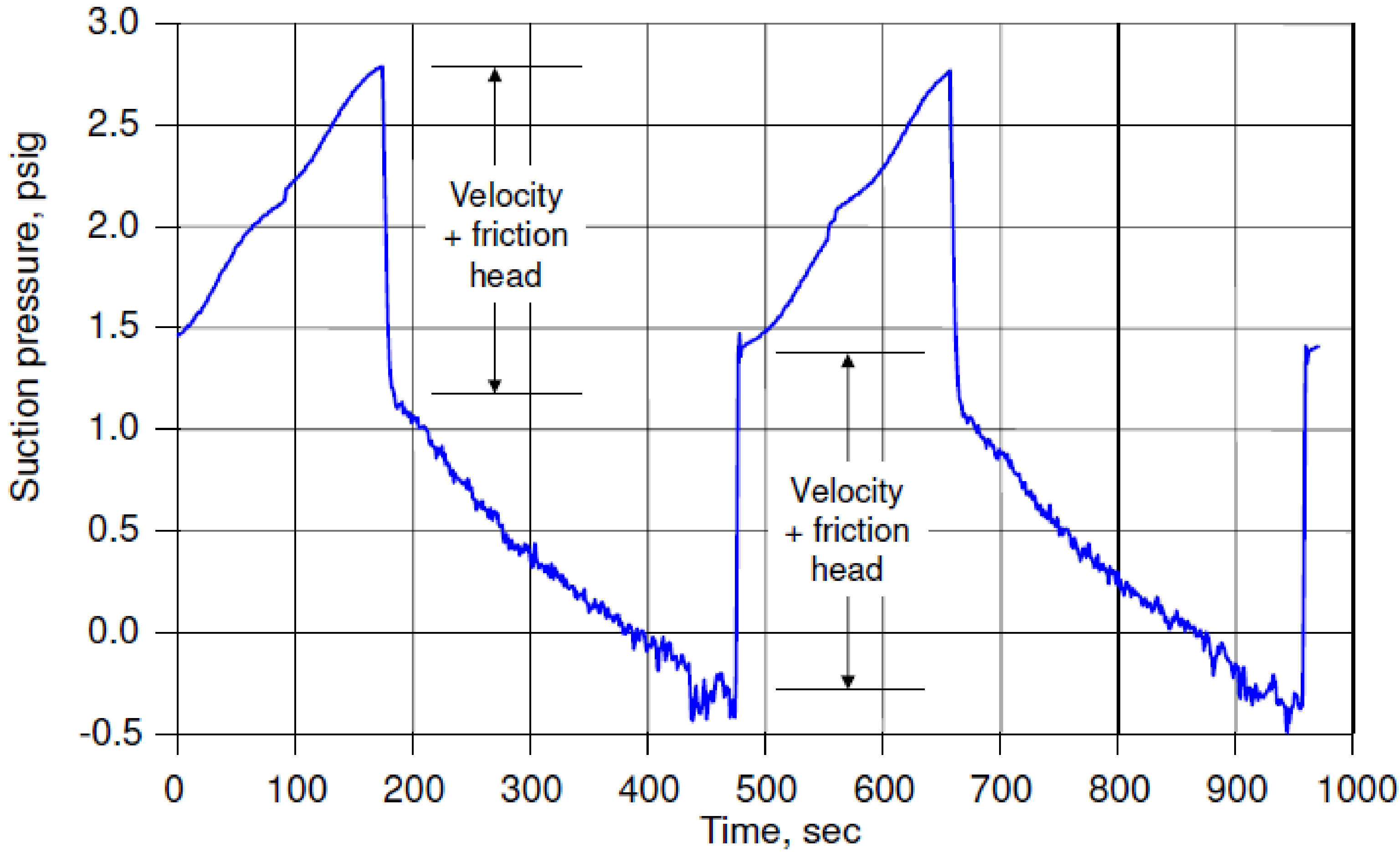
$$\frac{2.31(P_{2bef} - P_{2aft})}{s.g.} = (K+1)\left(\frac{V_{2aft}^2}{2g}\right)$$

∴ if we observe the step pressure change, we can estimate velocity

Pump suction pressure over two fill and drain cycles



Pump suction pressure over two fill and drain cycles



Average change in pressure (combining on-off and off-on transitions) was 1.67 psig

$$\frac{2.31(P_{2bef} - P_{2aft})}{s.g.} = \frac{2.31(1.67)}{1.00} = 3.86 \text{ ft}$$

Pressure depression is the combination of velocity head and frictional head losses between the wet well and pump suction

$$3.86\ \text{ft} = \frac{V_2^2}{2g} + K\left(\frac{V_2^2}{2g}\right) = \frac{V_2^2}{2g}(1+K)$$

For bell-mouth reducer:	$K = 0.05$
For long radius 6-in. elbow:	$K = 0.18$
Overall loss coefficient:	$K = 0.23$

$$3.86\ \text{ft} = \frac{V_2^2}{2g}(1+K)$$

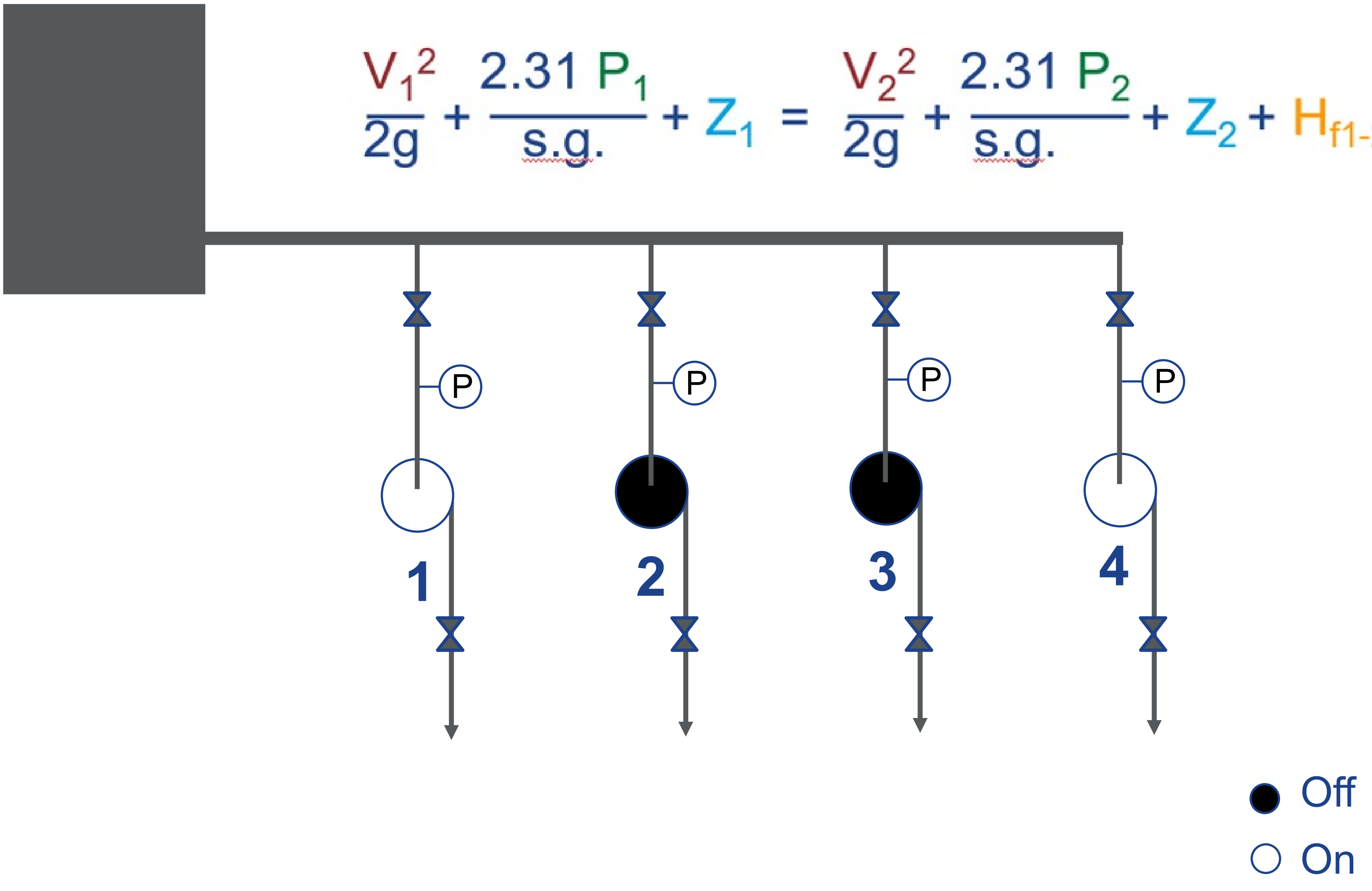
$$\Rightarrow \frac{V_2^2}{2g} = 3.14\ \text{ft}$$

$$\Rightarrow V = 14.2\ \text{ft/s}$$

$\Rightarrow$ calculated flow rate = 1209 gpm

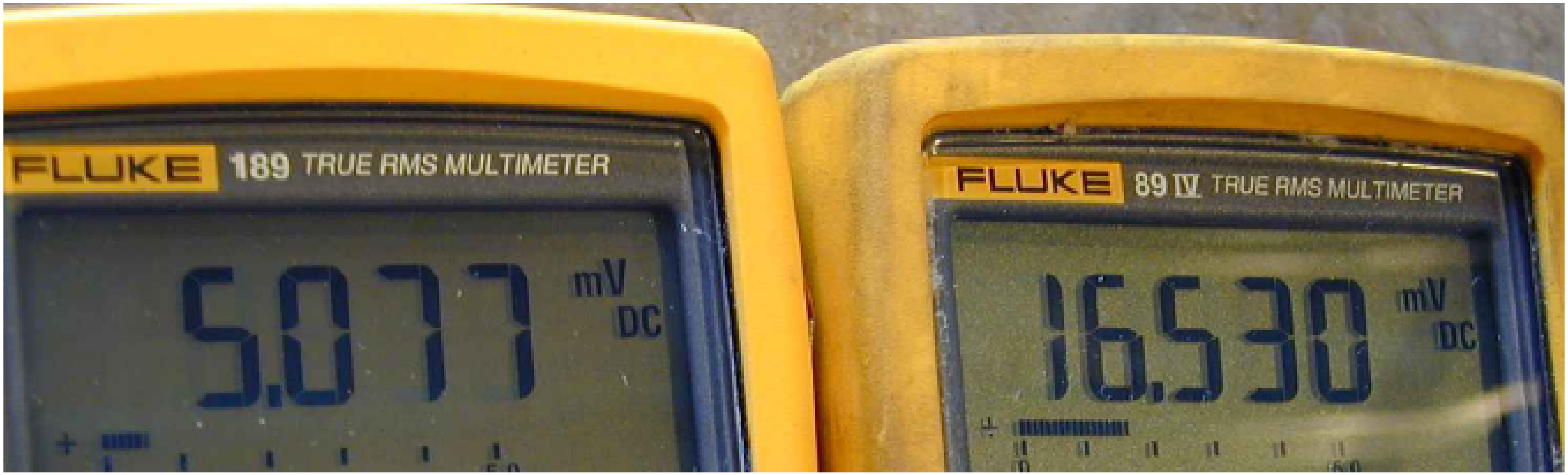
An effective way to measure flow rate in parallel pumping applications: use Bernoulli

$$\frac{V_1^2}{2g} + \frac{2.31\ P_1}{s.g.} + Z_1 = \frac{V_2^2}{2g} + \frac{2.31\ P_2}{s.g.} + Z_2 + H_{f1\text{-}2}$$



An alternative method: compare suction pressures for the pump that is on with a parallel one that is off

Suction pressures on both pumps were monitored during drawdown
(Instrument scaling: 1 mV = 1 kPa)



Pump 1 suction pressure
(Pump on)

Pump 2 suction pressure
(Pump off)

Differential = 11.453 kPa = 1.66 psig = 3.84 ft
(again, this is combined velocity head and friction loss)

Flow rate was calculated from differential pressure at several points during wet well drawdown

pipe diameter	5.895	(6.02" pipe with nominal 1/16" cement-mortar liner)						
Area, sq ft	0.190							
estimated K	0.23							
	Event	P1 (kPa)	P2 (kPa)	delta kPa	dH, ft	Vhead = dH/(1+K)	Vel, ft/s	gpm
	1	6.69	17.5	10.80	3.62	2.94	13.76	1170
	2	5.98	16.8	10.86	3.64	2.96	13.80	1174
	3	5.08	16.5	11.45	3.84	3.12	14.17	1205
	4	1.82	13.4	11.60	3.89	3.16	14.26	1213
	5	1.48	12.9	11.44	3.83	3.12	14.16	1205
	6	0.72	11.8	11.04	3.70	3.01	13.91	1183
	7	-0.1	11.2	11.32	3.79	3.08	14.09	1198
	8	-0.7	10.6	11.26	3.77	3.07	14.05	1195
				Average	**3.76**	**3.06**	**14.02**	**1193**

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	2	5.98	16.8	10.86	3.64	2.96	13.80	1174
	3	5.08	16.5	11.45	3.84	3.12	14.17	1205
	4	1.82	13.4	11.60	3.89	3.16	14.26	1213
	5	1.48	12.9	11.44	3.83	3.12	14.16	1205
	6	0.72	11.8	11.04	3.70	3.01	13.91	1183
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	8	-0.7	10.6	11.26	3.77	3.07	14.05	1195
				Average	**3.76**	**3.06**	**14.02**	**1193**

An independent method of estimating flow rate in level-controlled operations

- Measure time between on and off events
- Calculate the volume between level switches
- Incoming flow rate with pump off = volume/time
- Assume that the incoming flow rate when the pump is running is equal to that calculated from before and after pump off periods

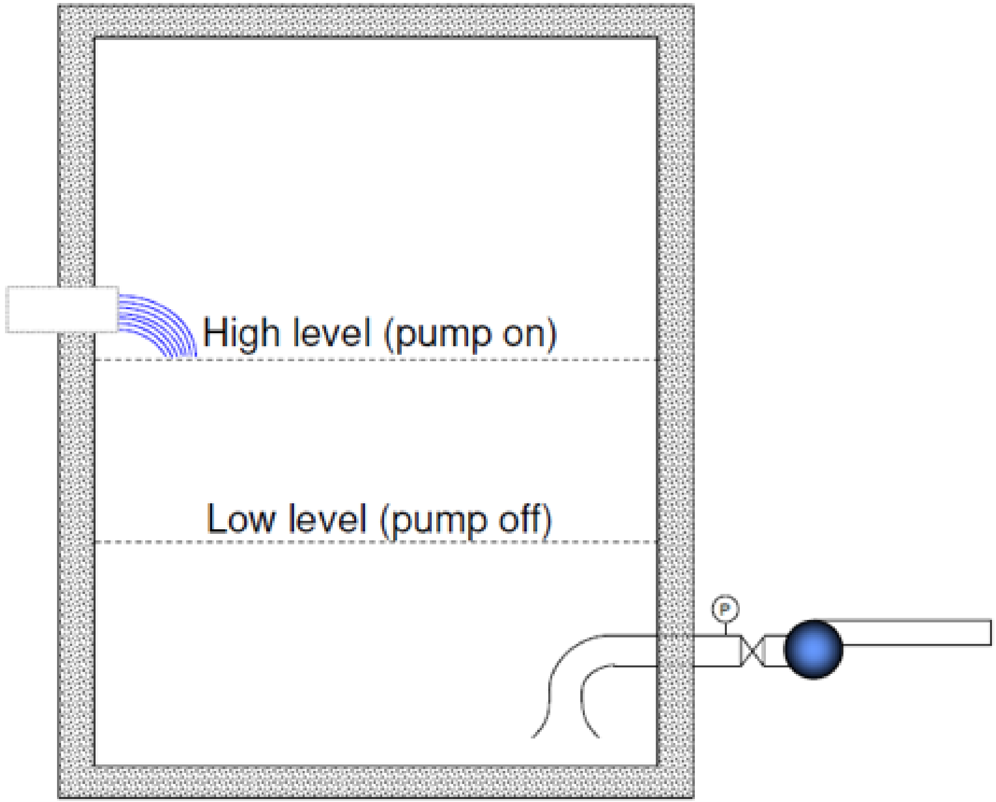
$$Q = \frac{t_{on}\left(\frac{V_w}{t_{off}}\right) + V_w}{t_{on}} = \left[\left(\frac{V_w}{t_{off}}\right) + \left(\frac{V_w}{t_{on}}\right)\right] = V_w\left(\frac{t_{off} + t_{on}}{t_{off} \times t_{on}}\right)$$

Q = Pump flow rate, gpm
t_{on} = Pump run time, min
t_{off} = Pump off time, min
V_w = Well volume, gallons

Estimated flow rate at the lift station using this method was 1161 gpm

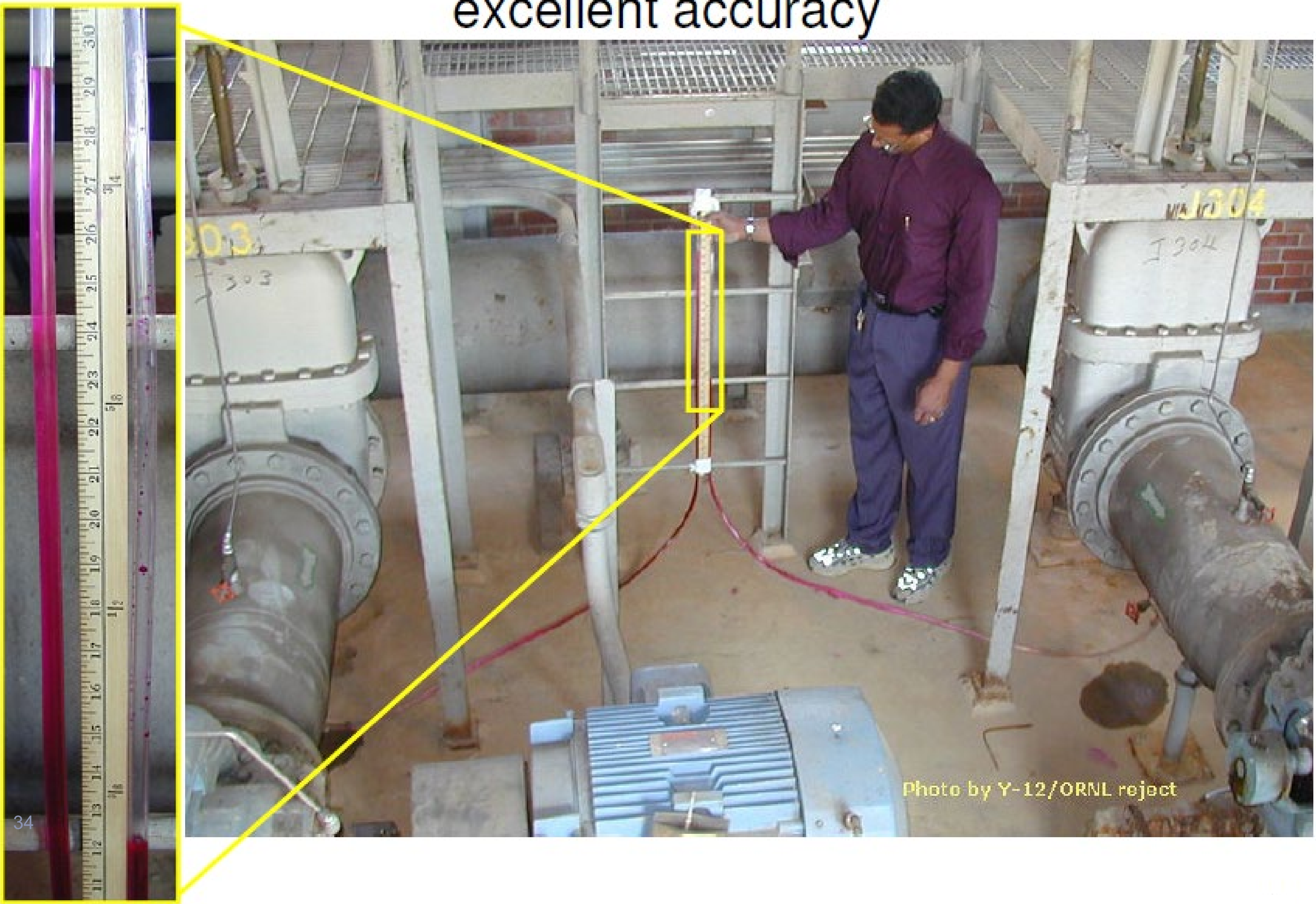
Example

- Tank Volume between level switches: 1000 gal
- Time to fill between level switches with incoming flow: 4 min (pump off)
- Incoming flow: 1000 gal/4 min = 250 gpm
- Incoming flow is continuous
- Time to empty volume: 1 min
- Total volume pumped out during 1 min pump run: 1250 gal
- Pump flow rate: 1250 gpm



$$Q = V_w(t_{on} + t_{off})/(t_{on} x\ t_{off}) = 1{,}000 \text{ x } (5/4) = 1250 \text{ gpm}$$

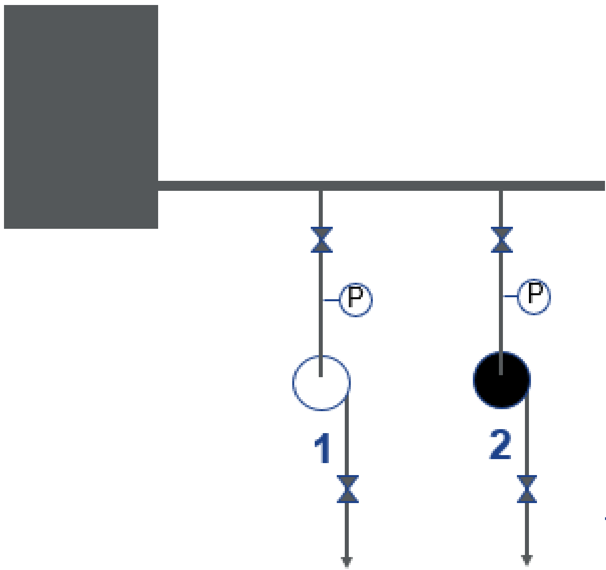
Manometers – even the home made kind – provide excellent accuracy



Equations…….

$$\frac{V_1^2}{2g} + \frac{2.31\ P_1}{s.g.} + Z_1 = \frac{V_2^2}{2g} + \frac{2.31\ P_2}{s.g.} + Z_2 + H_{f1\text{-}2}$$

- Parallel Pumps
- Pump 1 – Off
- $V_1 = 0$
- Pump 2 – Running
- Elevations $Z_1 = Z_2$



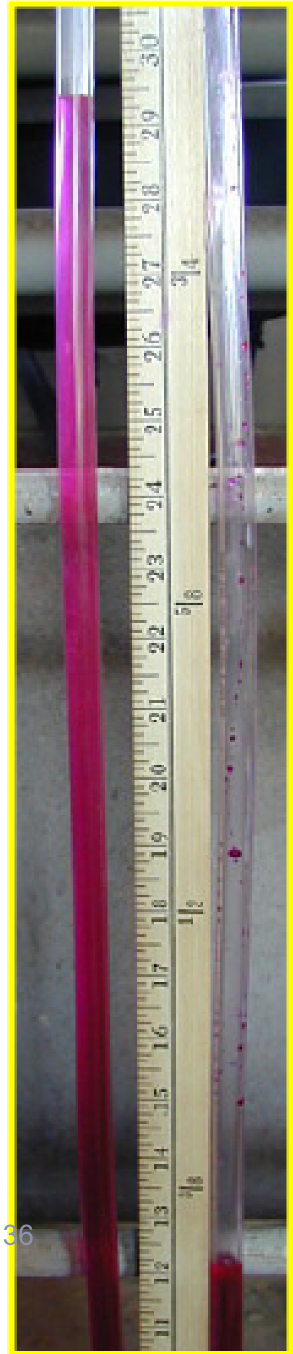
$$\frac{2.31(P_1 - P_2)}{SG} = \frac{V_2^2}{2g} + K\left(\frac{V_2^2}{2g}\right)$$

Where: P (psig); V (ft/sec)

$$P_1 - P_2 = \frac{V_2^2}{2g}(1 + K)$$

Where: P (ft Hd); V (ft/sec)

An exercise: estimate the flow rate for the previous slide



Suction header: 18" standard (17.25-inch ID)

Individual pump suction lines: 16" standard (15.25-inch ID)

Suggested loss assumptions:

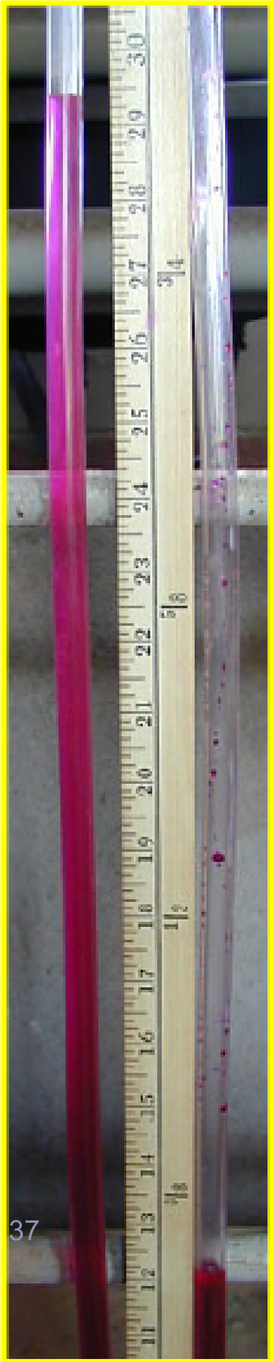
Branch tee:	0.45
18-16 reducer:	0.11
16-inch gate valve:	0.04

$$z_1 - z_2 = \frac{V_2^2}{2g}(1+K)$$

$$\text{gpm} = 2.448\ V\ d^2$$

Where z is elevation in feet, g is 32.174 ft/s^2, V is velocity in ft/s, and d is the pipe inside diameter in inches

An exercise: estimate the flow rate for the previous slide



16" standard (15.25-inch ID)

Branch tee:	0.45
18-16 reducer:	0.11
16-inch gate valve:	0.04
	0.60

$$z_1 - z_2 = \frac{V_2^2}{2g}(1+K) \qquad \frac{17.5}{12} = \frac{V_2^2}{2 \times 32.174}(1.6) \Rightarrow V = 7.66 \text{ ft/s}$$

$$\text{gpm} = 2.448\, V\, d^2 \qquad 2.448 \times 7.66 \times 15.25^2 = 4360 \text{ gpm}$$

Where z is elevation in feet, g is 32.174 ft/s^2, V is velocity in ft/s, and d is the pipe inside diameter in inches

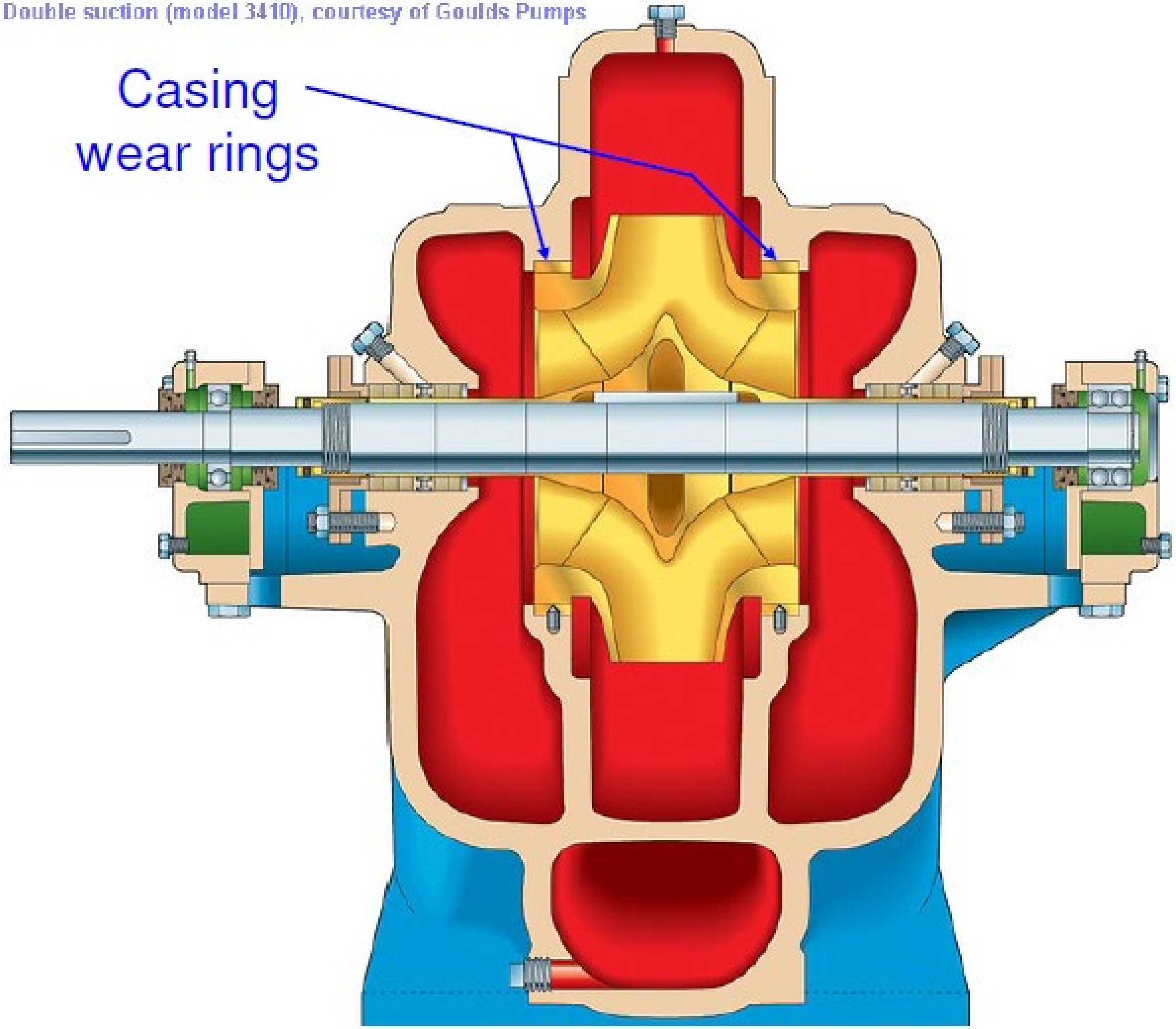
Reasons a pump optimization rating could be low

Optimization Rating = Actual Pump Efficiency/Optimal Pump Efficiency

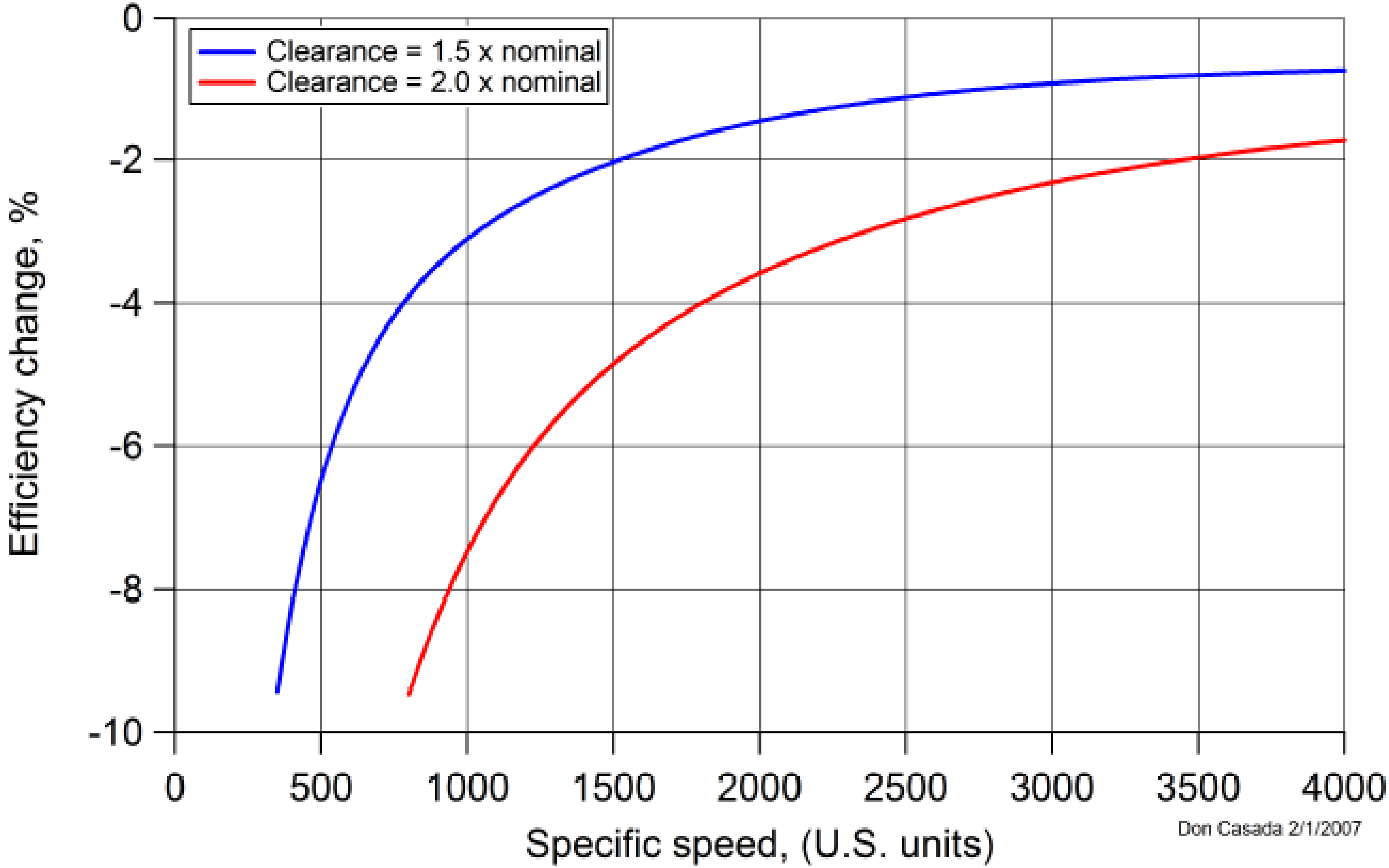
1. Pump operates away from its best efficiency point.
2. Pump parts (wear rings, impeller, volute, diffuser) are worn or have corroded/eroded.
3. Installation or operating problems, particularly on suction side.
4. Pump or motor simply aren't top of the line units.
5. Adjustable speed, belt, gear, or other drive is used.
6. Hydraulic Institute screwed up in their achievable efficiency estimates.
7. I screwed up in my measurements, or I missed a flow path (such as a leaking check valve on a parallel pump).

Tight wear ring clearances avoid excessive recirculation

Double suction (model 3410), courtesy of Goulds Pumps



HI 1.3 generic efficiency loss associated with increased wear ring clearances



$$N_s = \frac{N\sqrt{Q}}{(gH)^{0.75}}$$

Ns = specific speed
N = rotational speed
Q = flow rate
H = head (per stage)
g = acceleration due to gravity

Specific Speed

Specific speed is a numerical representation of the impeller shape. When appropriate units are used, it is a dimensionless quantity. However, standard industry practice is to use common engineering units that do not resolve to a dimensionless quantity.

The standard U.S. (metric) units used to calculate specific speed are:

Rotational speed: rpm (rpm)
Flow rate: gpm (m^3/hr)
Head: ft (m)

It is also standard practice to simply eliminate the gravitational constant term.

The specific speed equation is to be applied to only one condition, namely the best efficiency point (BEP).

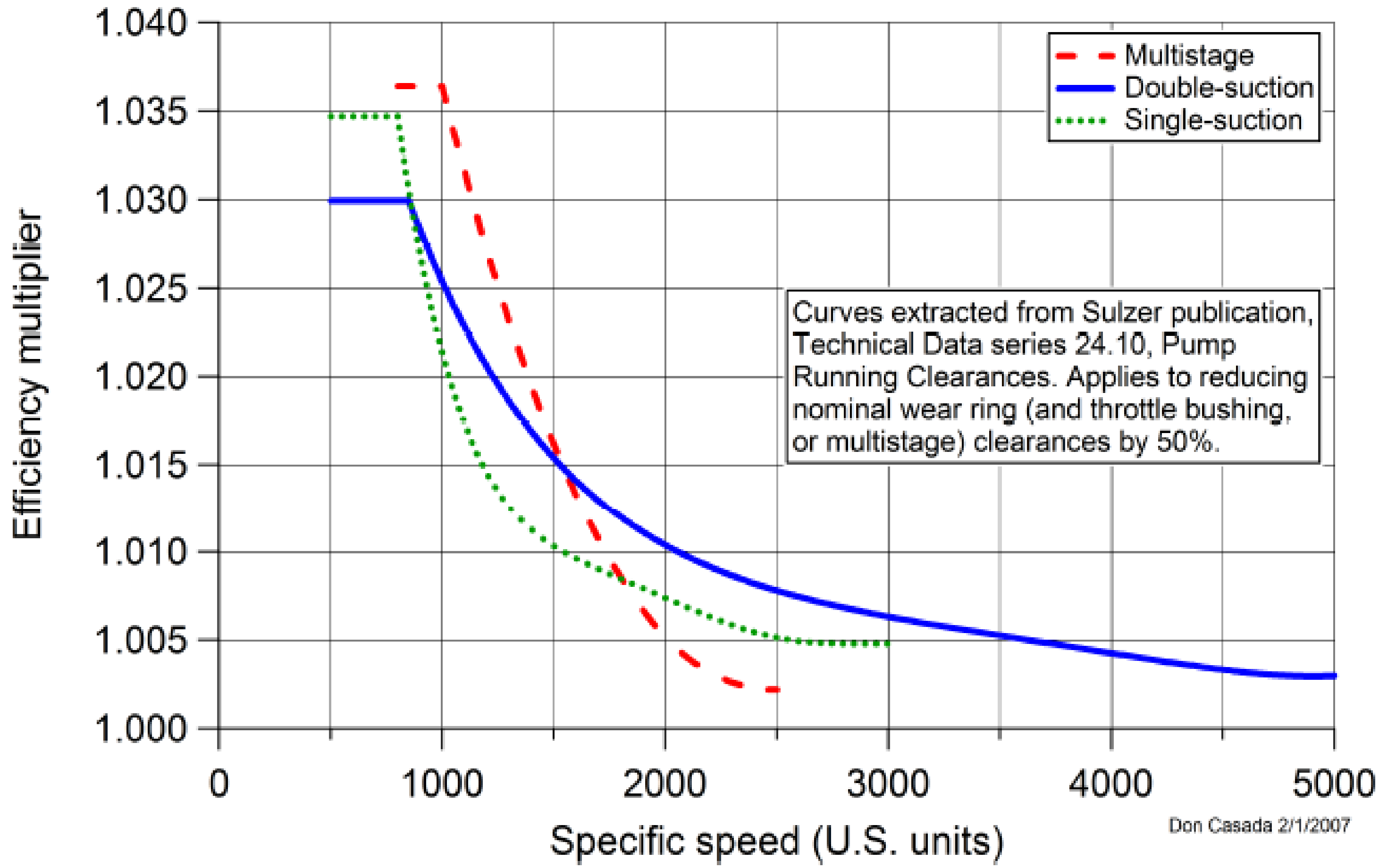
Example:

BEP flow rate = 5000 gpm (1135.6 m^3/hr)
BEP head = 120 feet (36.6 m)
Rotating speed = 1780 rpm
Single stage pump

=> U.S. Specific speed = 3472; metric specific speed = 4033

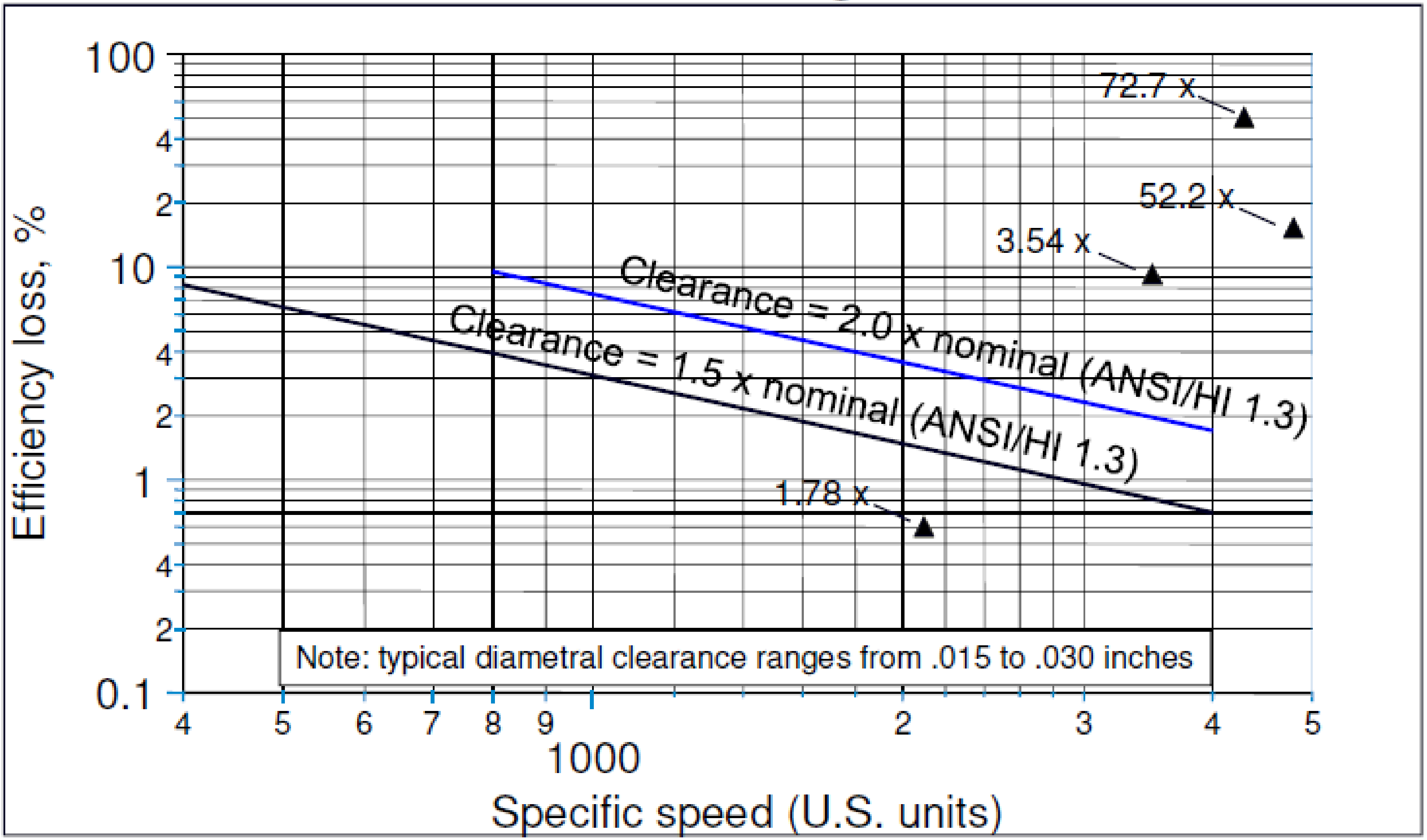
Most centrifugal pumps in common industrial applications will have specific speed values in U.S. (metric) units of between 500 (581) and 7000 (8132). Pump impellers toward the lower end of the range have a pronounced radial flow, while impellers toward the upper end of the range approach an axial flow profile.

Sulzer estimates for efficiency improvements from ***reducing*** wear ring clearances



These curves are used to multiply times the nominal pump efficiency for API or generic Sulzer clearances if the clearance is cut in half using approved non-galling materials such as PEEK (polyetheretherketone).

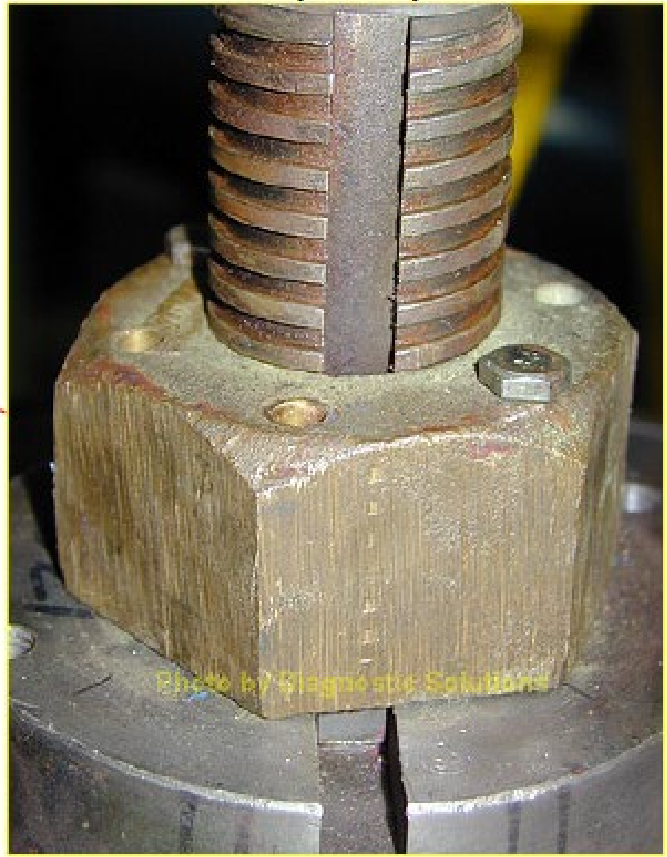
Generic estimates, reported data on the effects of increased wear ring clearances



Sources
- Curves: ANSI/HI 1.3, Hydraulic Institute (www.pumps.org)
- Data: Flowserve, as reported in Pump Handbook, 3rd edition, Karassik et al, McGraw-Hill

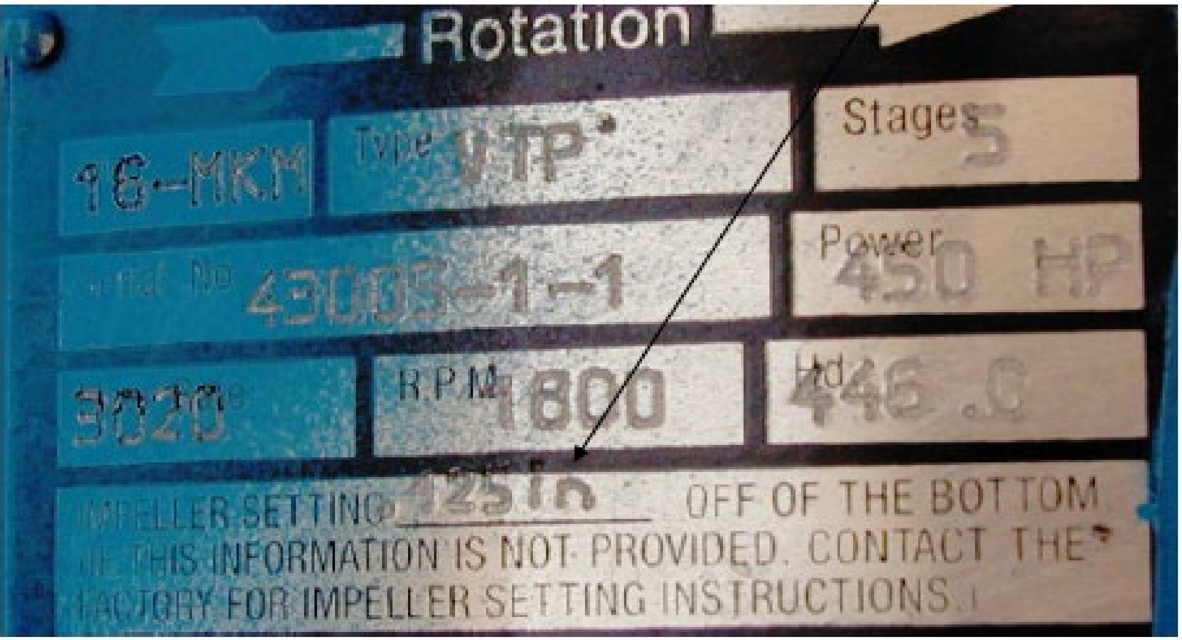
Clearances on pumps with open/semi-open impellers can normally be adjusted

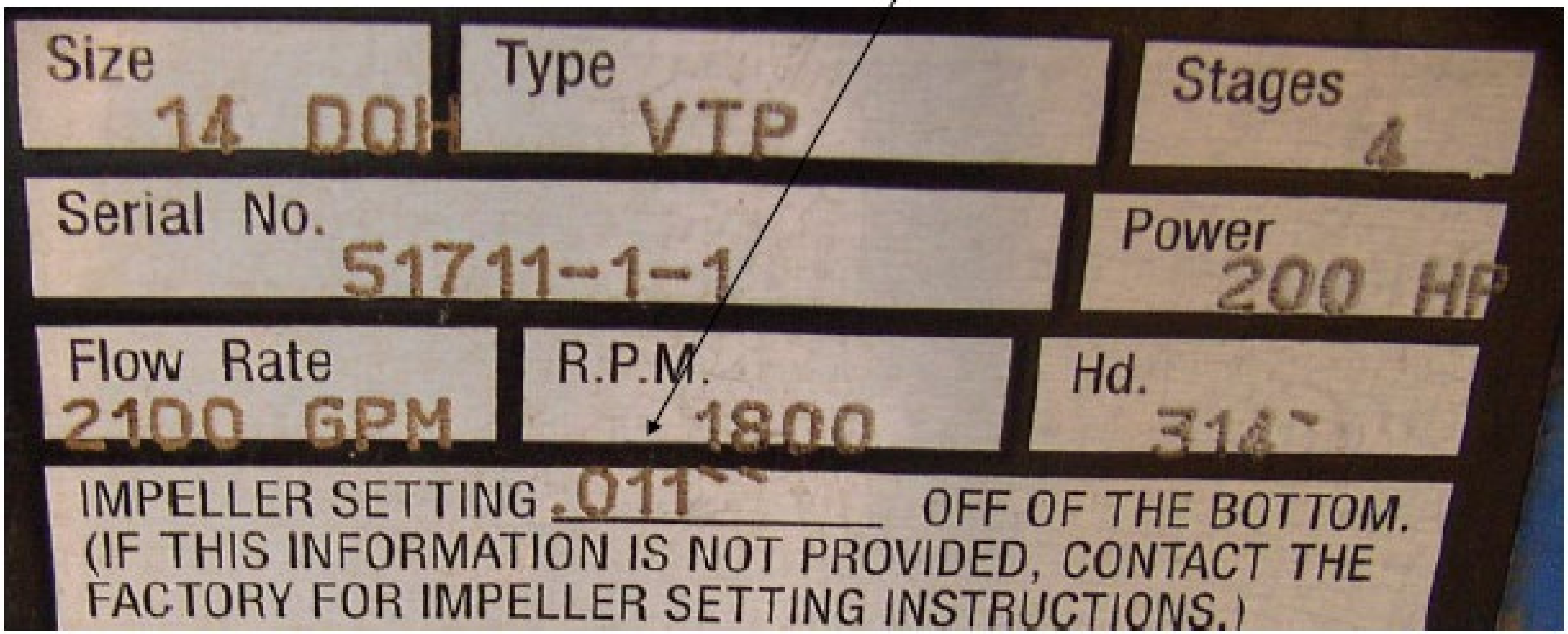
Vertical turbine pump lift setting



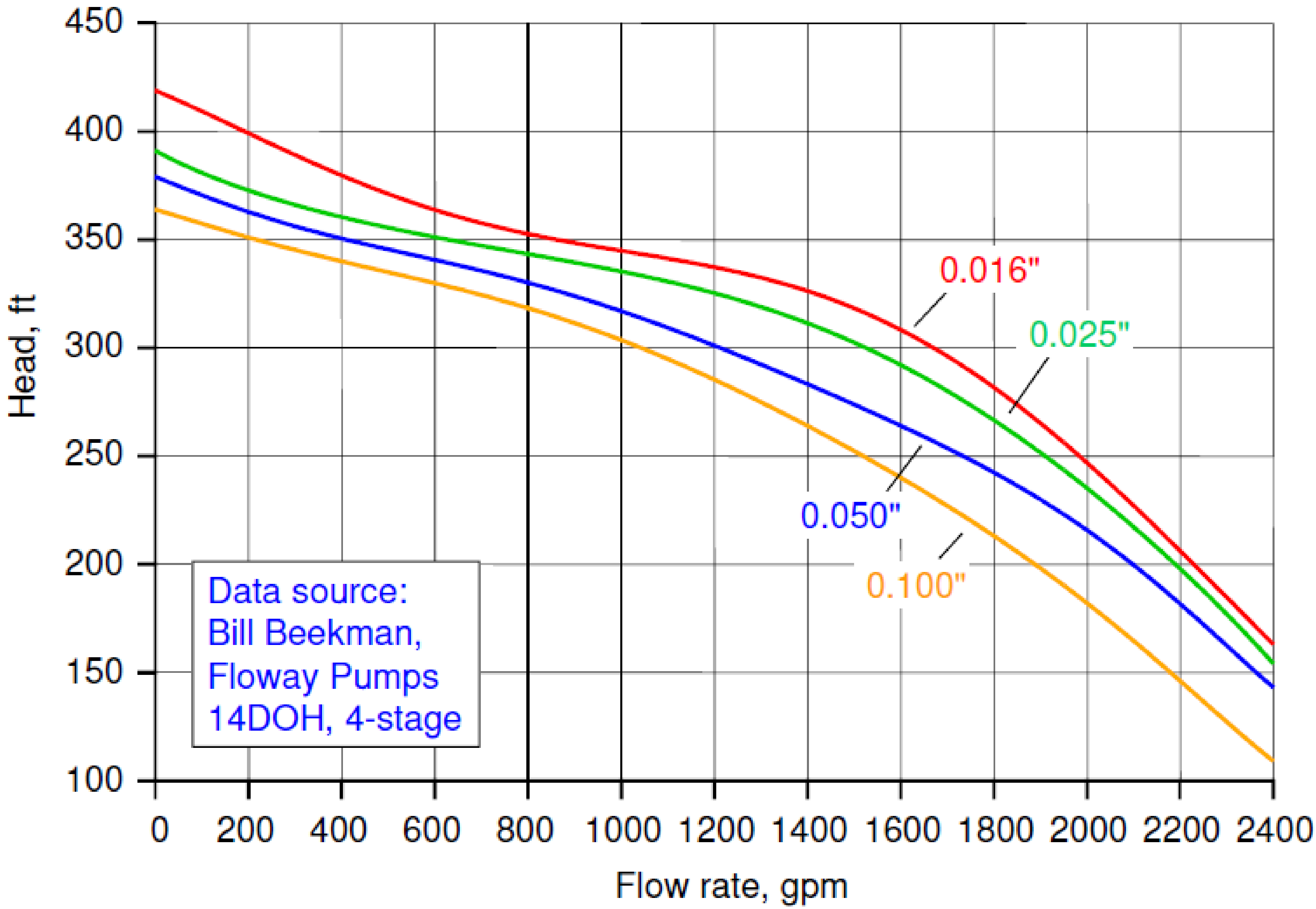
Adjustment in this case increased flow rate by 10%, efficiency by 5%

Enclosed impeller: 0.125"; Semi-open impeller: 0.011"

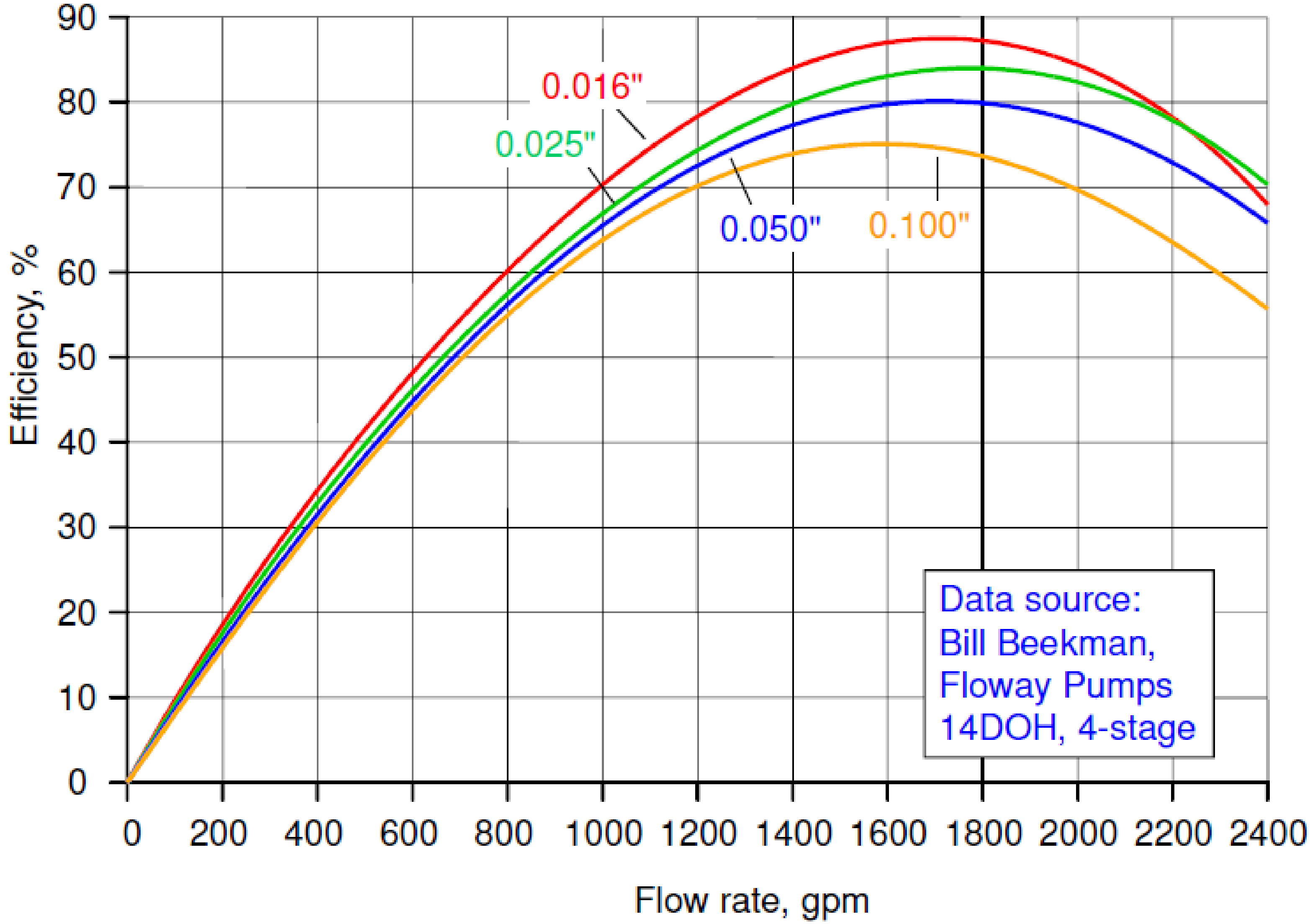




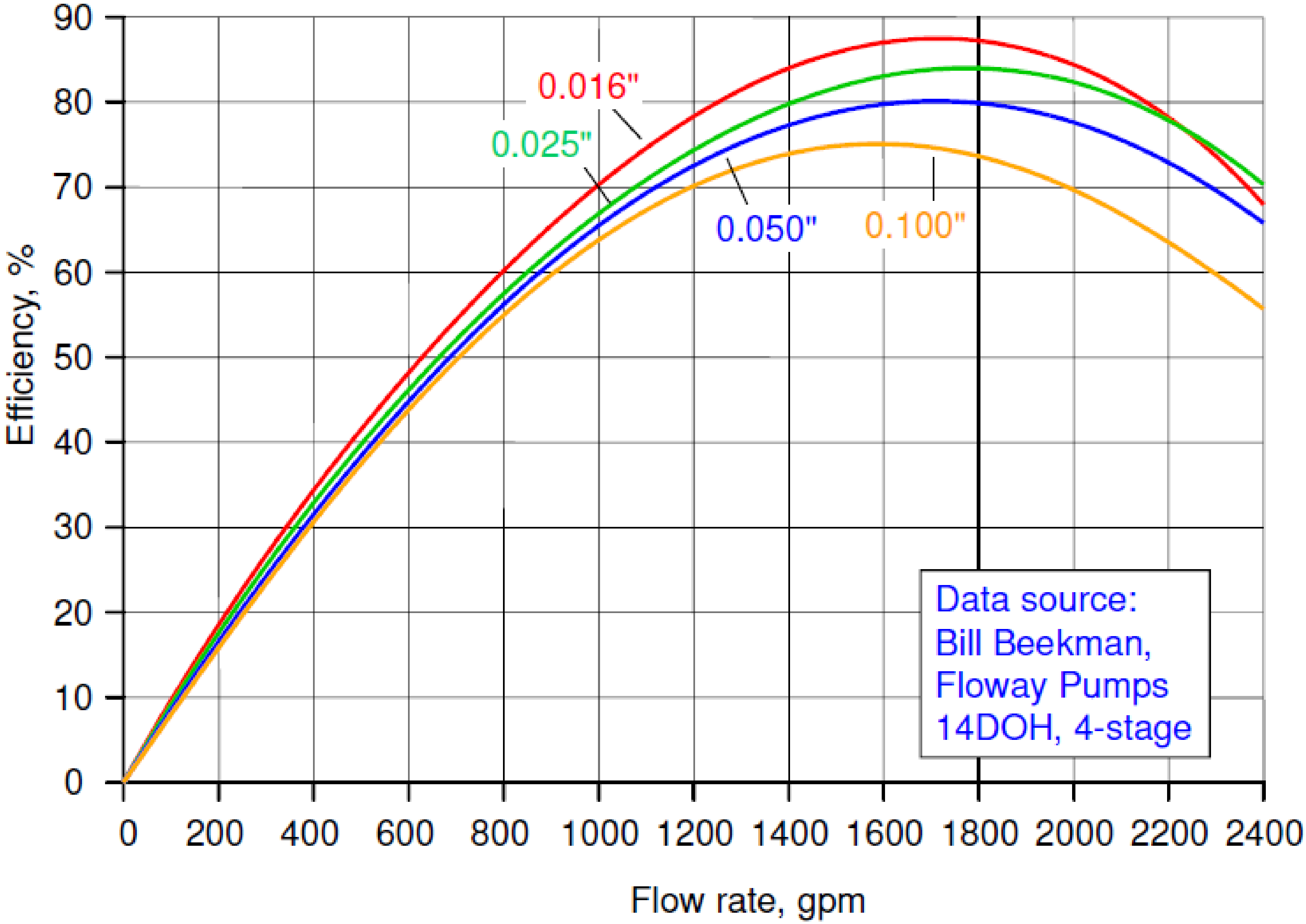
Axial clearance head-capacity effect: semi-open impeller



Axial clearance efficiency effect: semi-open impeller



Axial clearance efficiency effect: semi-open impeller

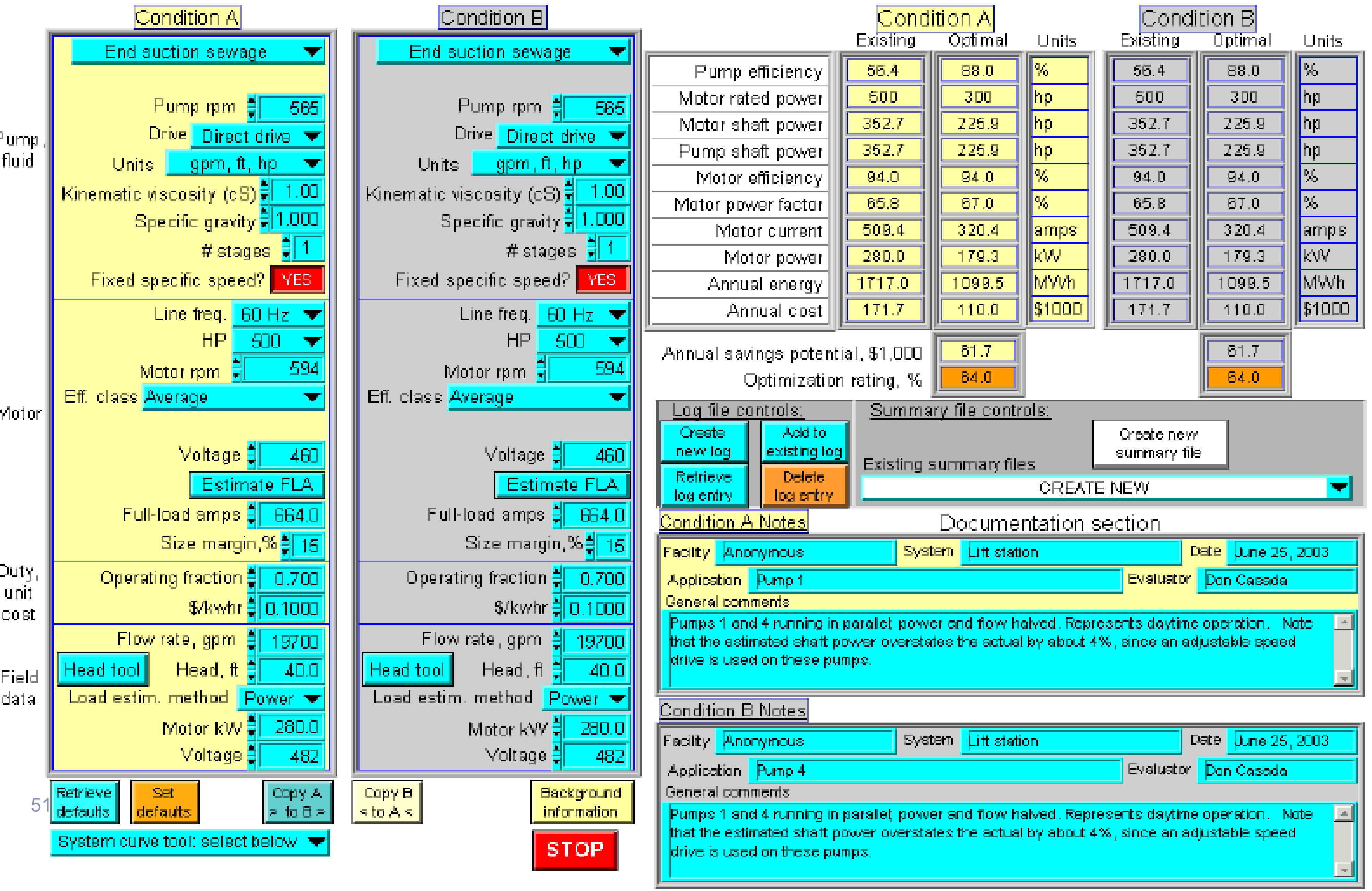


An interesting wear ring clearance episode

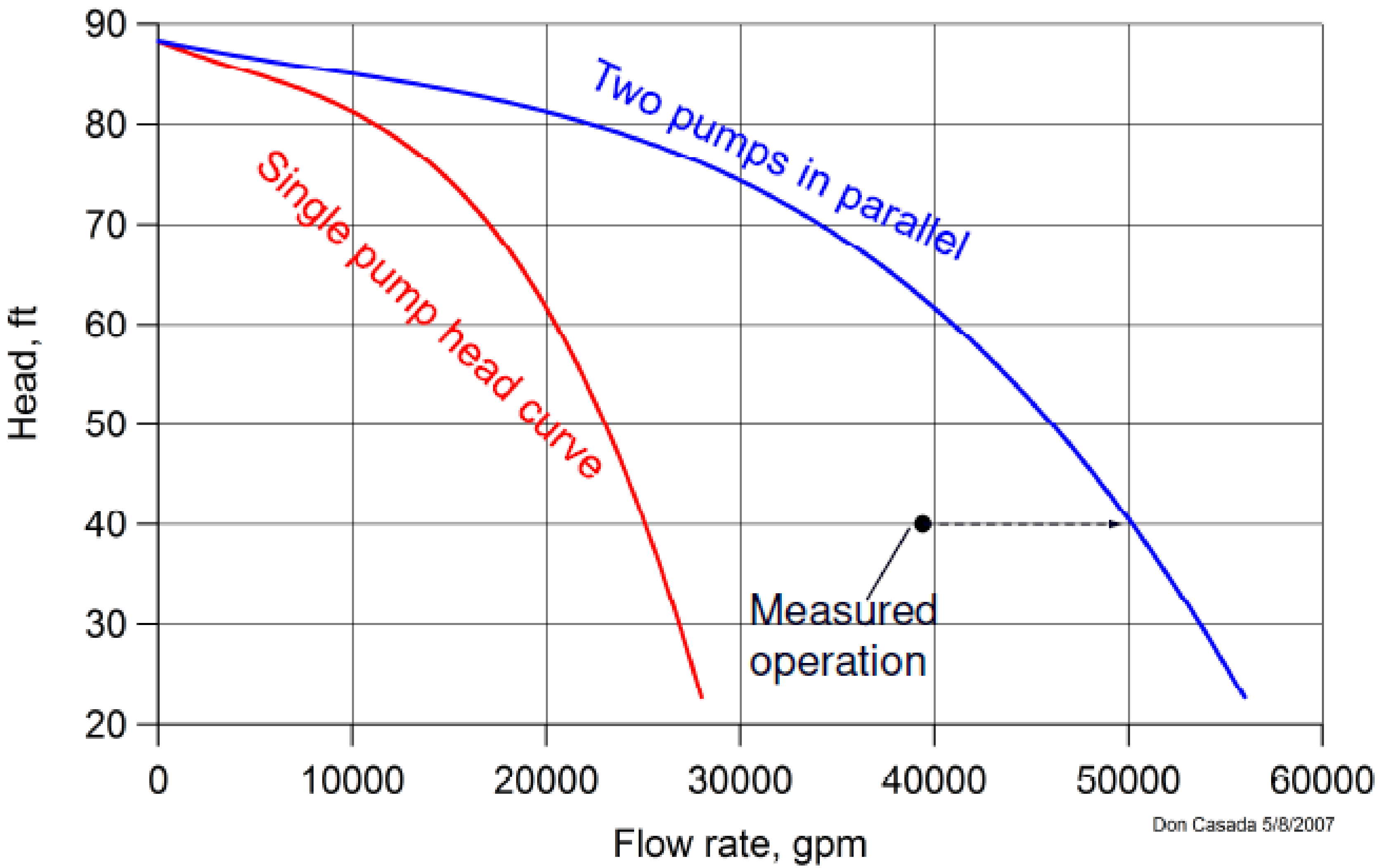
A very clean, well instrumented station

Motors: 500-hp, 12-pole, variable frequency drives, power meters in switchgear, station flow meter

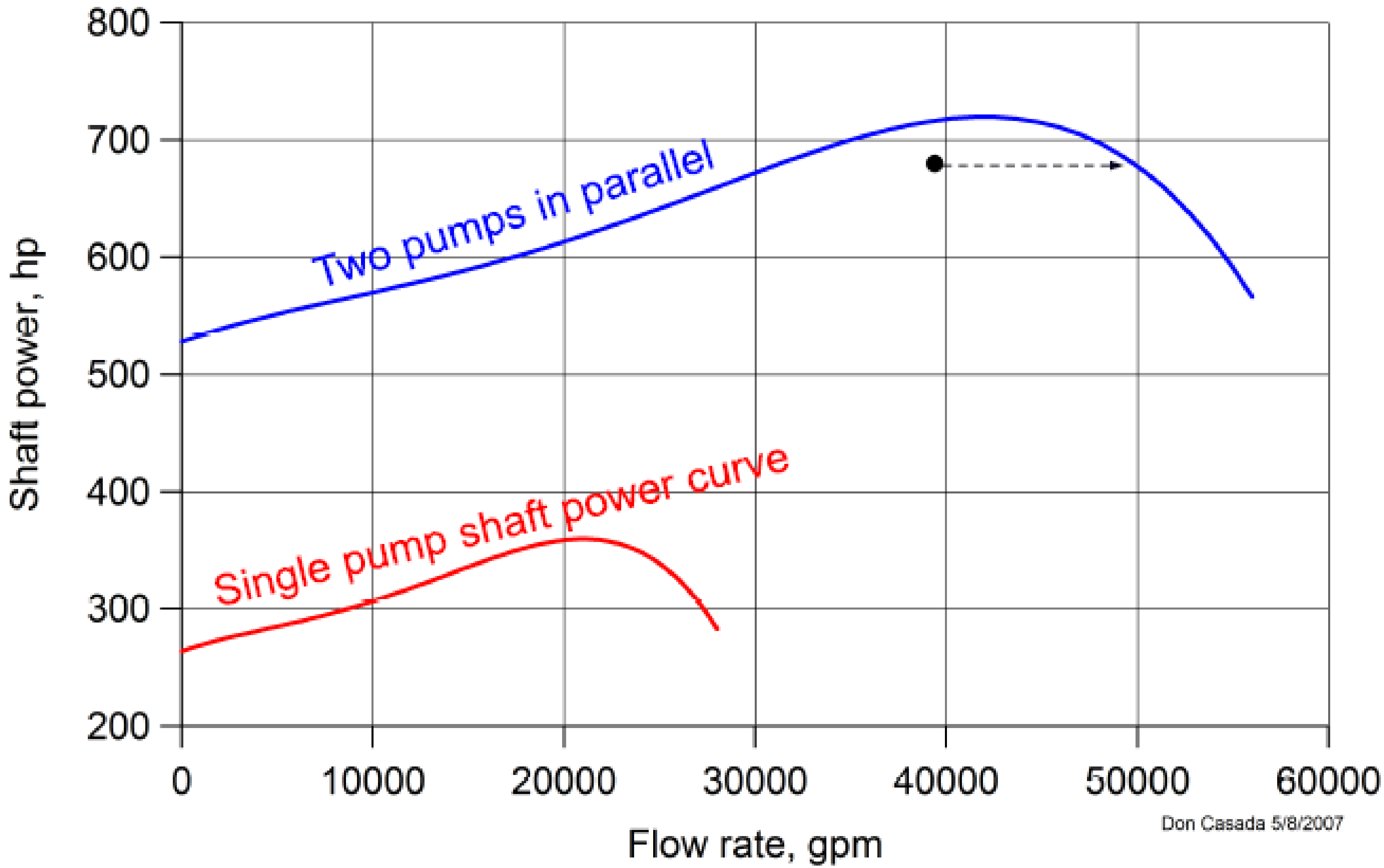
However, PSAT ratings were poor



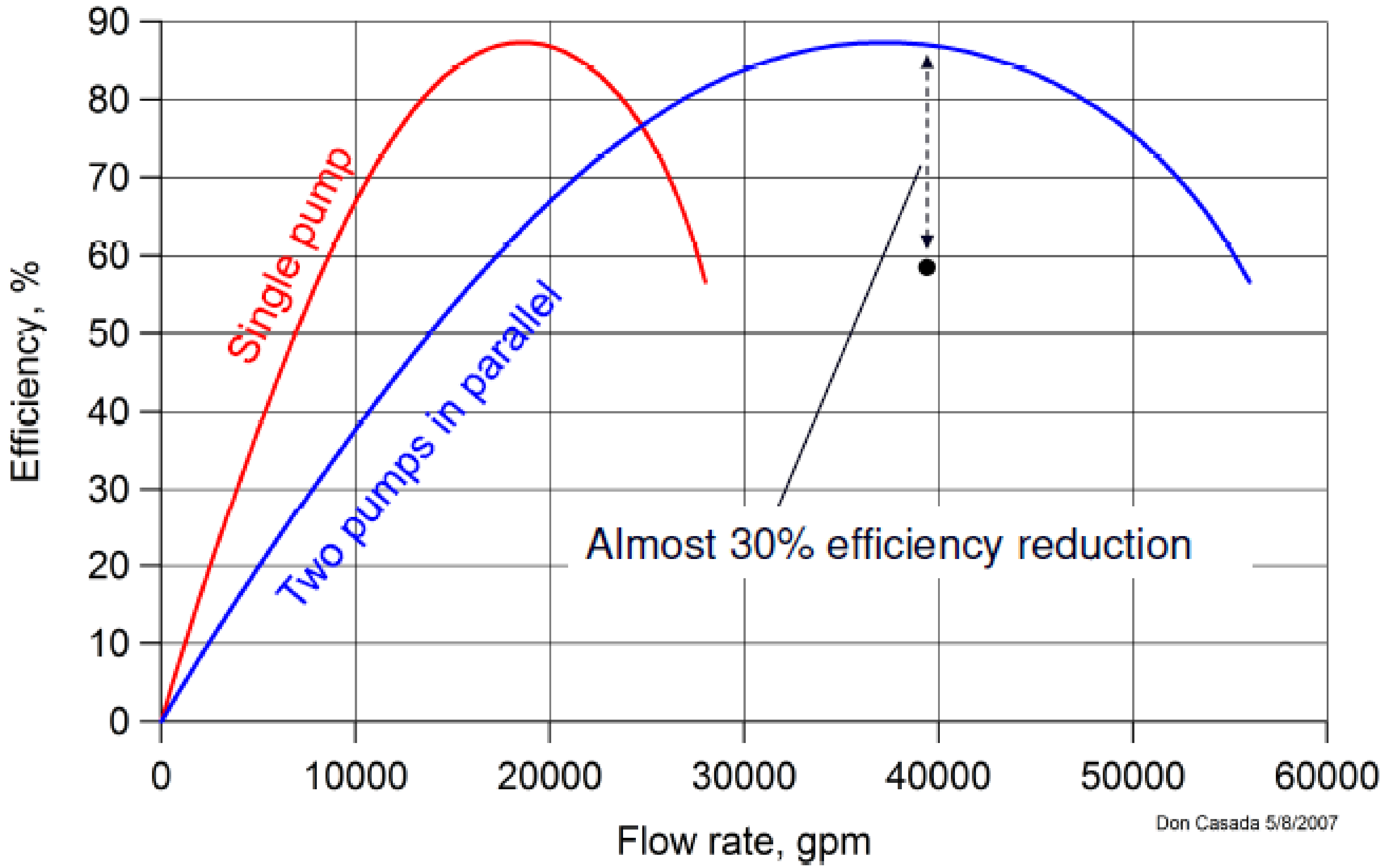
Head capacity curves and measured data



Power-capacity curves and estimated shaft power from PSAT (adjusted to account for VFD losses)



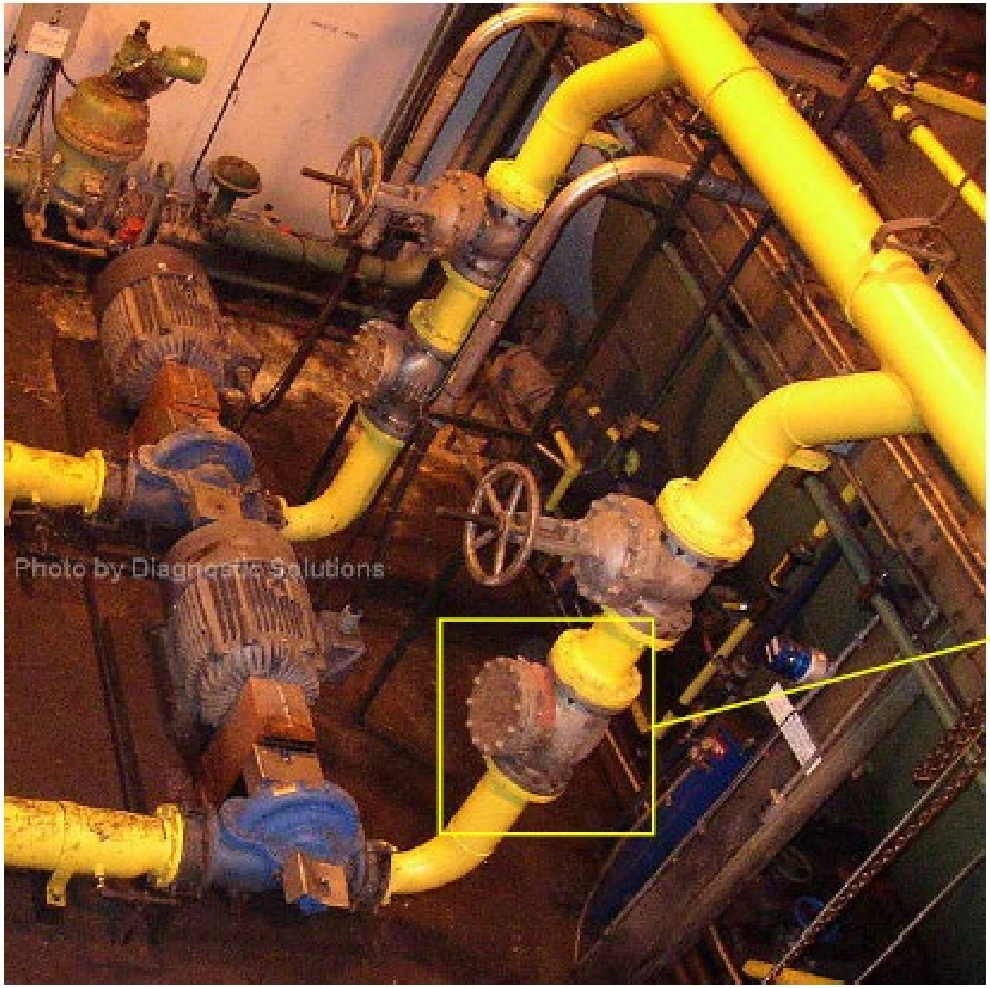
Efficiency-capacity curves and estimated pump efficiency (adjusted to account for VFD losses)



Original manufacturer wear ring diametral clearance = 0.030 inches

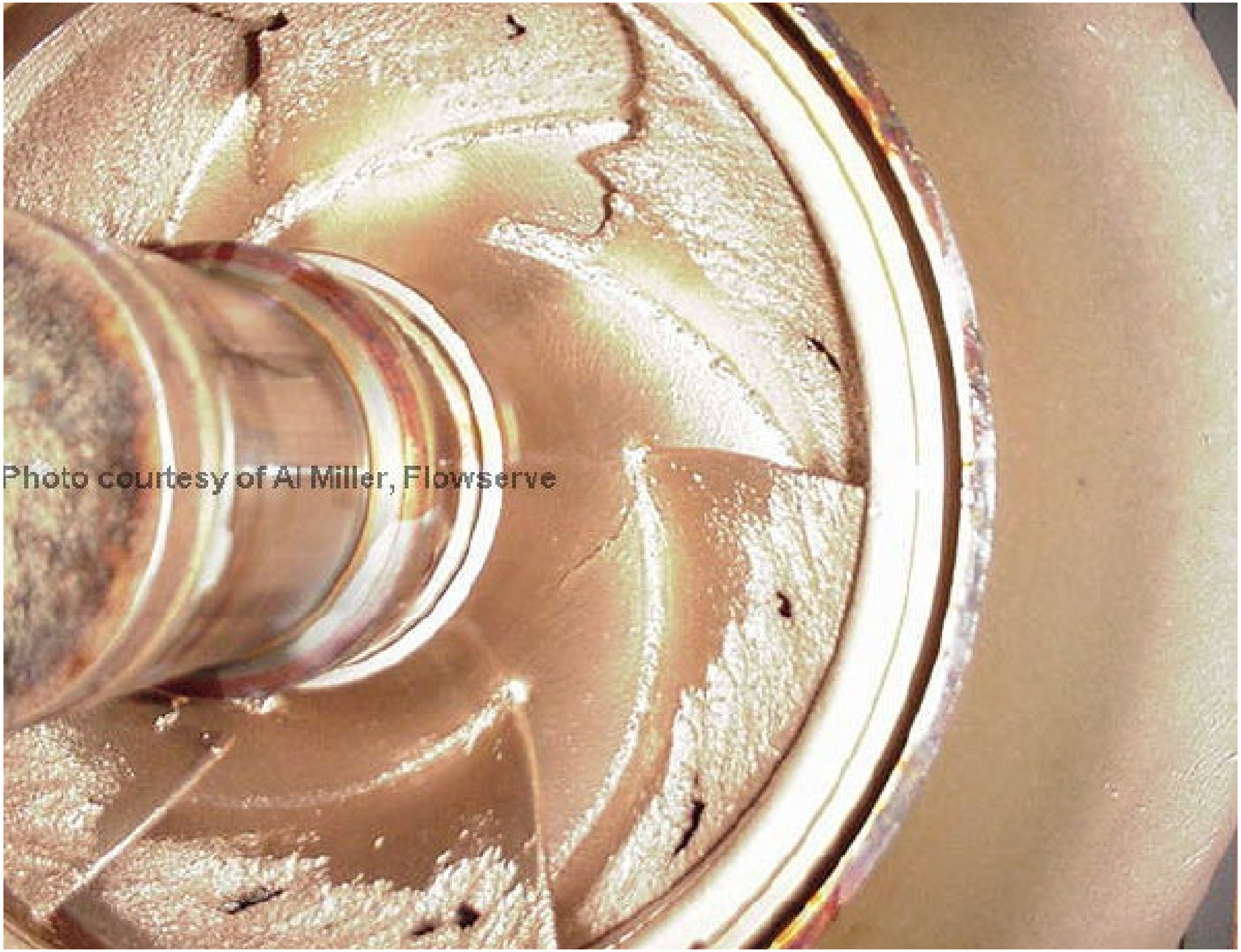
Existing clearance = 0.780 inches (nominal + 2x wear ring thickness)

Pump the fluid up and down it falls……….



Erosion/Corrosion Damage

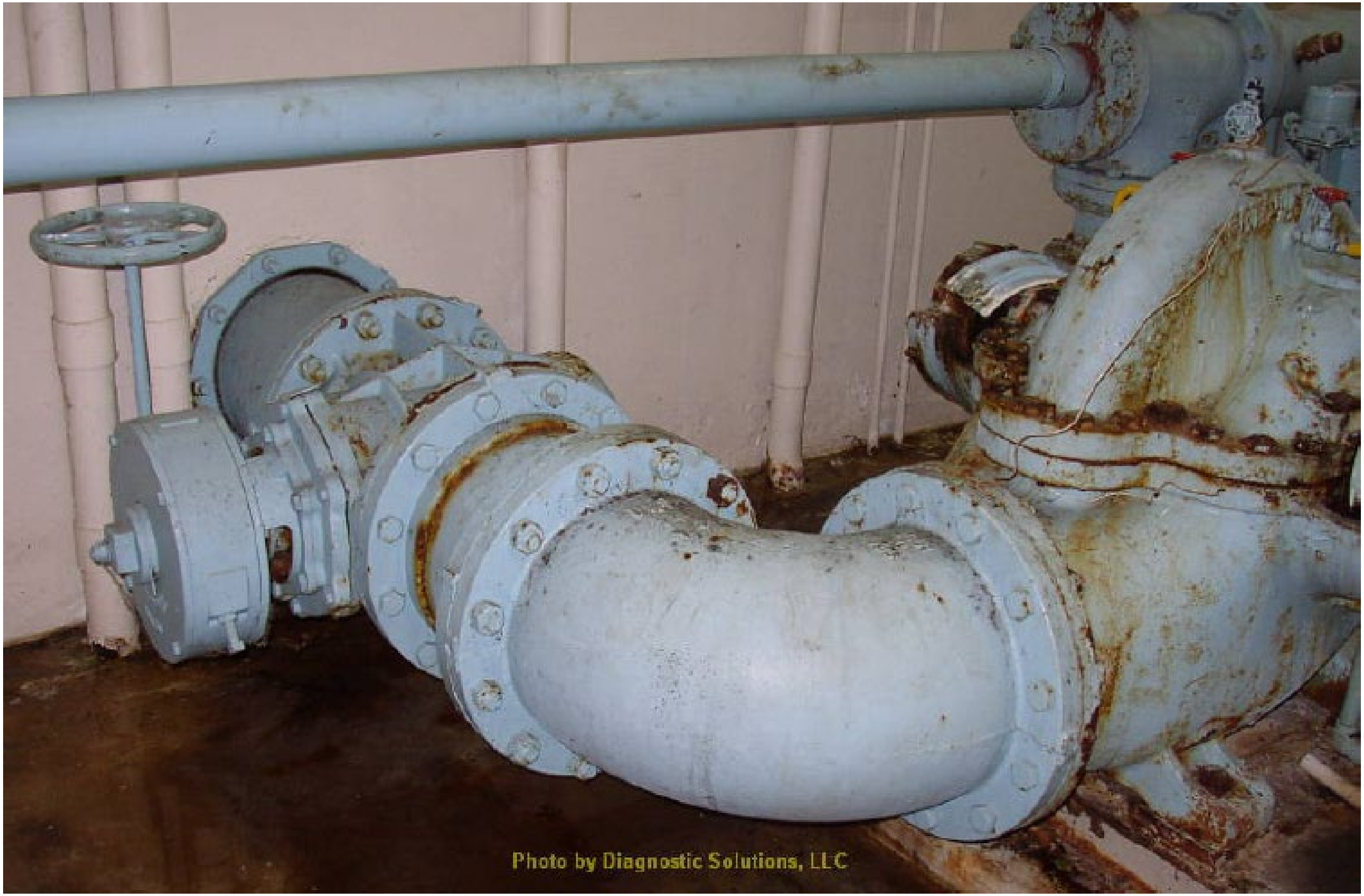
Cavitation Damage



Cavitation Damage – Waste Lift Station Pump

Bad Suction Geometry

Bad Suction Geometry




The End