



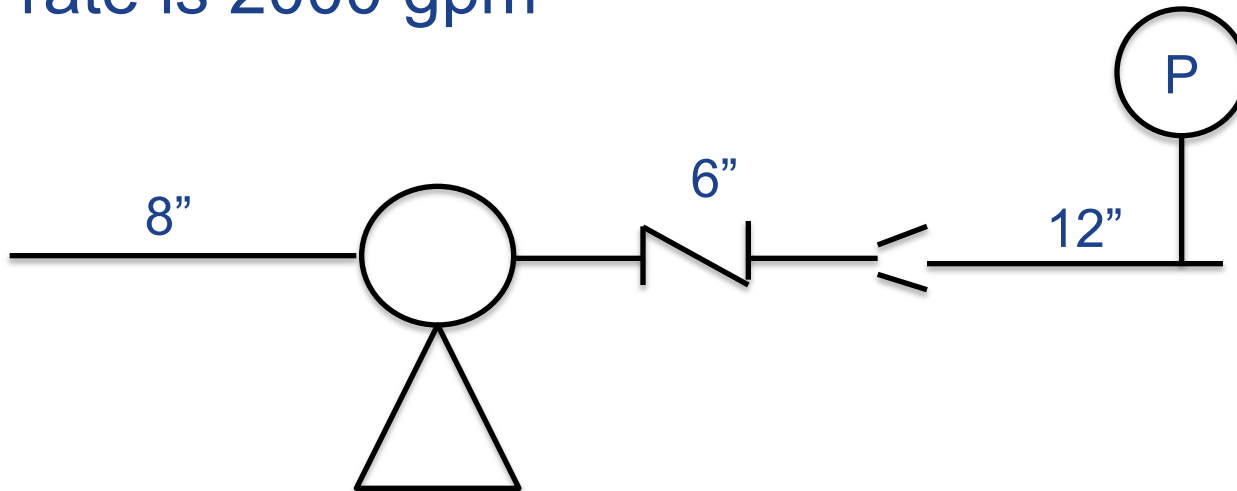
Pumping System Assessment

Week 5: Expanded Discussions



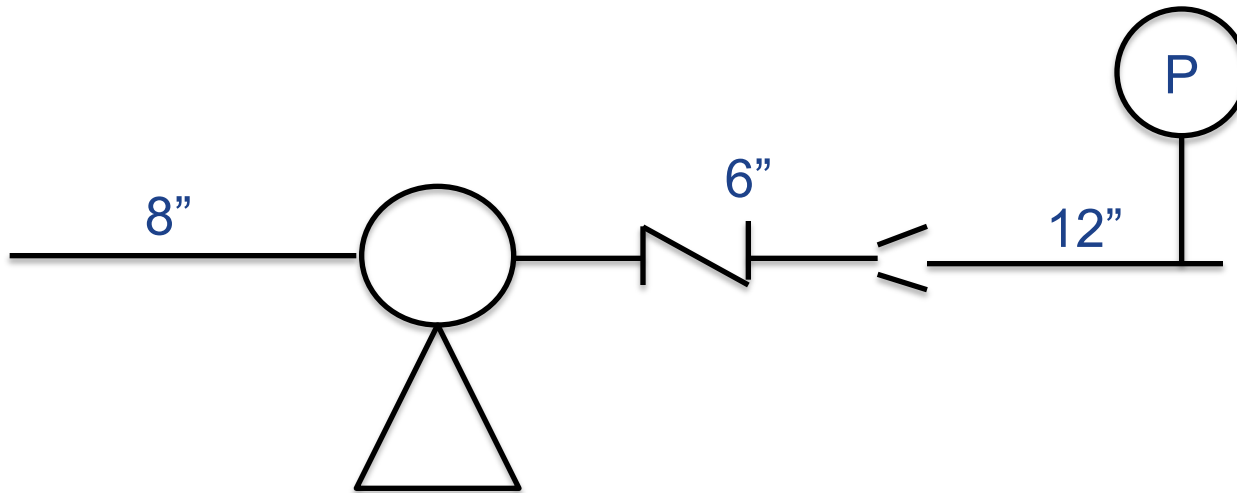
Scaling Loss Coefficients

- 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



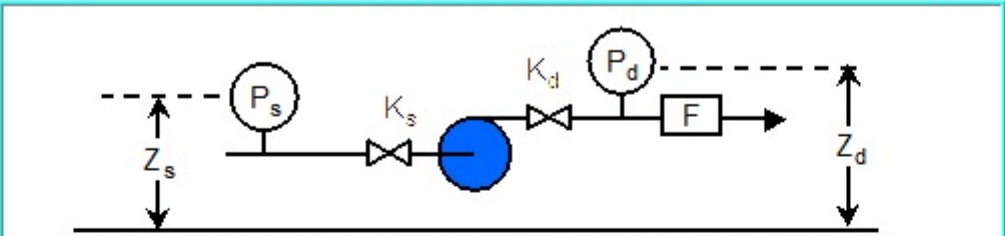
Scaling Loss Coefficients

- 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



Scaling Loss Coefficients

Type of measurement configuration
Suction and discharge line pressures



Click to access units converter tool

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID)	8.000	inches	Discharge pipe diameter (ID)	6.000	inches
Suction gauge pressure (P_s)	5.00	psig	Discharge gauge pressure (P_d)	75.00	psig
Suction gauge elevation (Z_s)	5.00	ft	Discharge gauge elevation (Z_d)	5.00	ft
Suction line loss coefficients, K_s	0.00		Discharge line loss coefficients, K_d	1.00	

Fluid specific gravity 1.000 Flow rate 2000.00 gpm

Don't update	Accept and update	Differential elevation head	0.00	ft
Click to leave the main panel head unchanged	Click to Accept and return the calculated head	Differential pressure head	161.76	ft
		Differential velocity head	5.47	ft
		Estimated suction friction head	0.00	ft
		Estimated discharge friction head	8.00	ft
		Pump head	175.23	ft

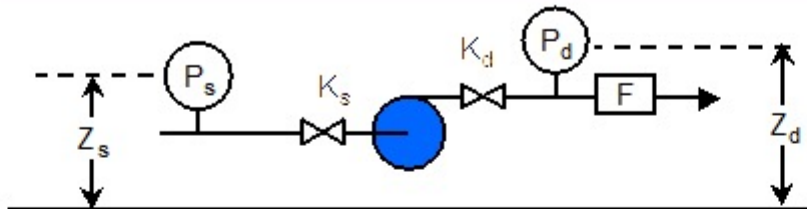
System of units: gpm, ft, hp

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

Check valve $k = 2$: valve friction loss = $2 \times 8.00 = 16.00$ ft

Scaling Loss Coefficients

Type of measurement configuration
Suction and discharge line pressures



Click to access units converter tool

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID)	8.000	inches	Discharge pipe diameter (ID)	12.000	inches
Suction gauge pressure (P_s)	5.00	psig	Discharge gauge pressure (P_d)	75.00	psig
Suction gauge elevation (Z_s)	5.00	ft	Discharge gauge elevation (Z_d)	5.00	ft
Suction line loss coefficients, K_s	0.00		Discharge line loss coefficients, K_d	1.00	

Fluid specific gravity 1.000 Flow rate 2000.00 gpm

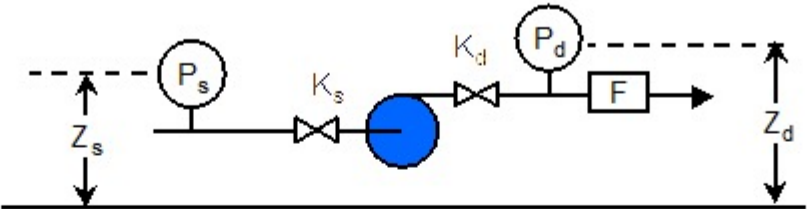
Differential elevation head	0.00	ft
Differential pressure head	161.76	ft
Differential velocity head	-2.03	ft
Estimated suction friction head	0.00	ft
Estimated discharge friction head	0.50	ft
Pump head	160.22	ft

System of units: gpm, ft, hp

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

Type of measurement configuration
Suction and discharge line pressures



Click to access units converter tool

K_s represents all suction losses from gauge P_s to the pump
 K_d represents all discharge losses from the pump to gauge P_d

Suction pipe diameter (ID)	8.000	inches	Discharge pipe diameter (ID)	12.000	inches
Suction gauge pressure (P_s)	5.00	psig	Discharge gauge pressure (P_d)	75.00	psig
Suction gauge elevation (Z_s)	5.00	ft	Discharge gauge elevation (Z_d)	5.00	ft
Suction line loss coefficients, K_s	0.00		Discharge line loss coefficients, K_d	32.00	

Fluid specific gravity 1.000 Flow rate 2000.00 gpm

Don't update Accept and update

Click to leave the main panel head unchanged Click to Accept and return the calculated head

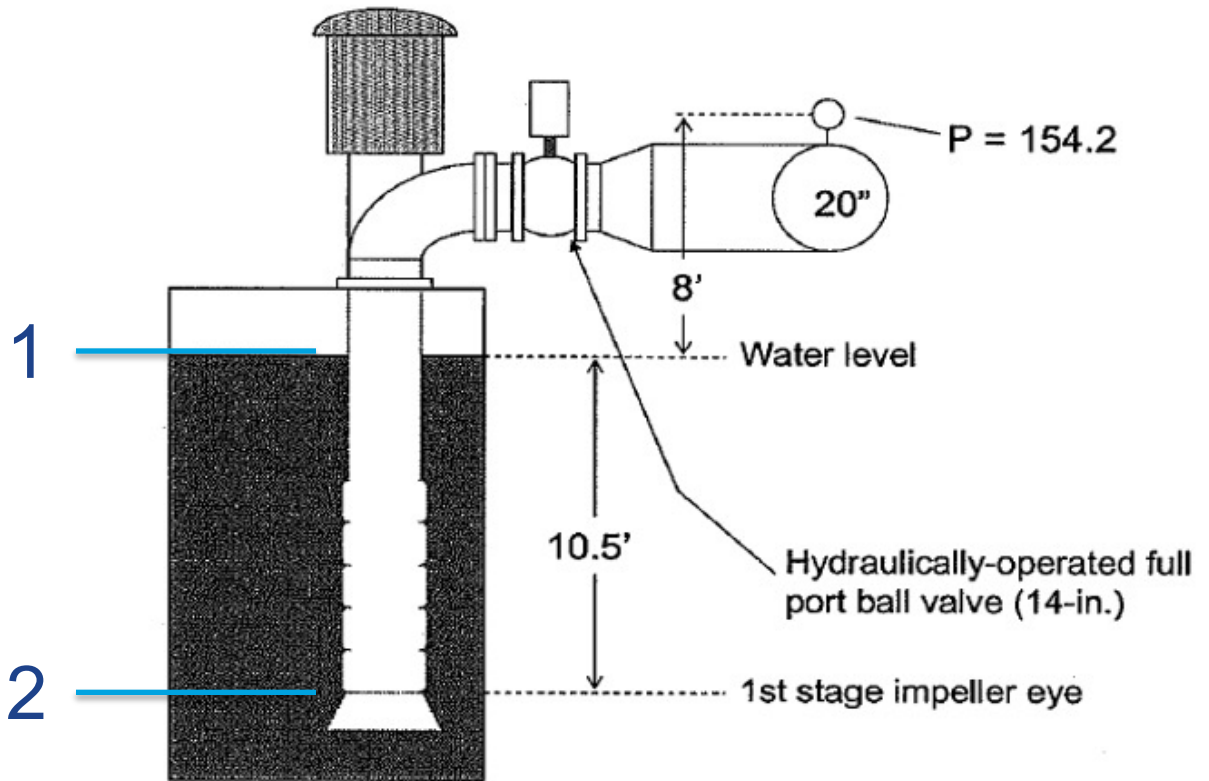
Differential elevation head	0.00	ft
Differential pressure head	161.76	ft
Differential velocity head	-2.03	ft
Estimated suction friction head	0.00	ft
Estimated discharge friction head	16.01	ft
Pump head	175.73	ft

System of units: gpm, ft, hp

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

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

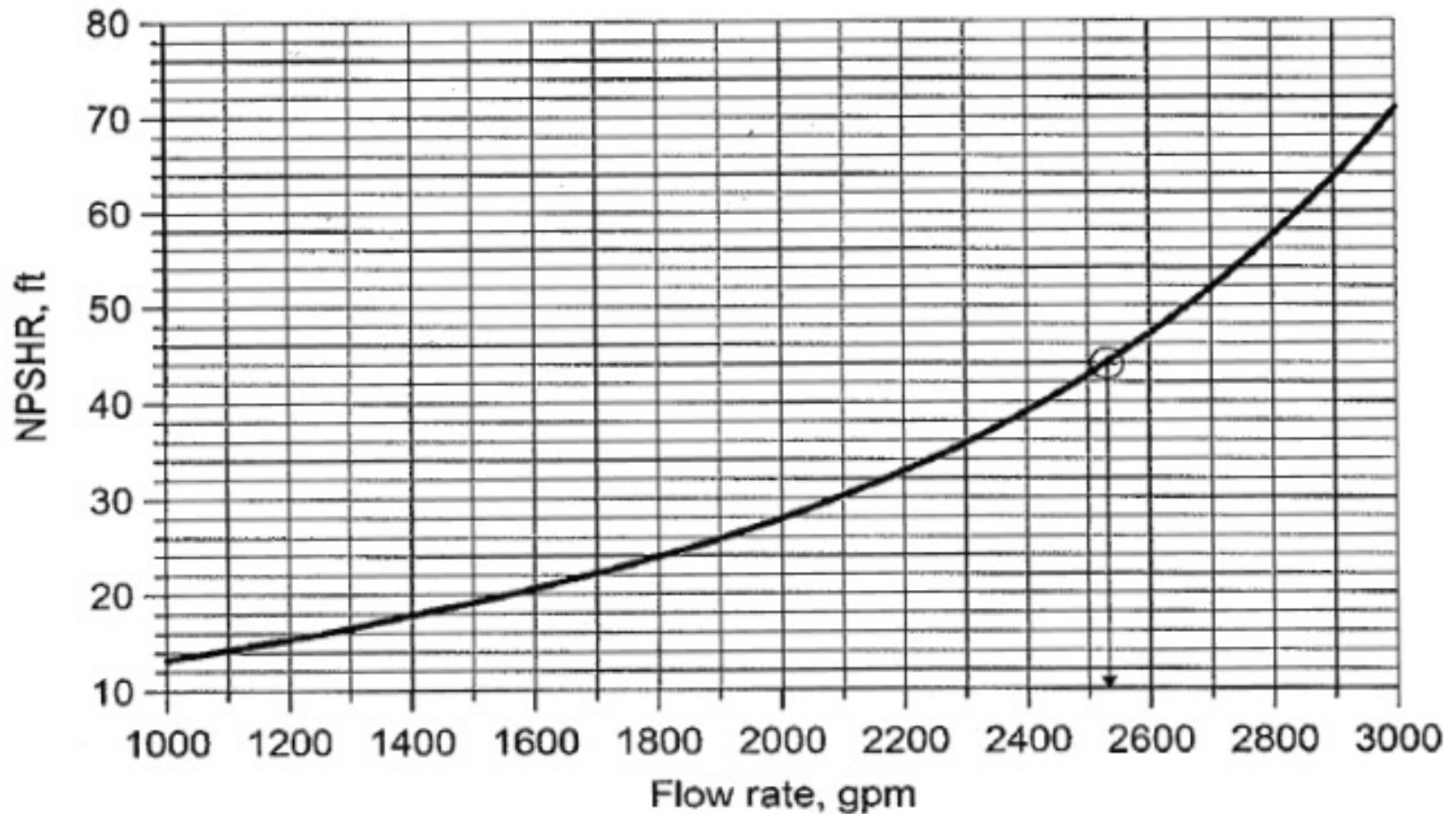
Water saturation vapor pressure at 60 F= 0.26 psia

Reference location for suction head determination is the water surface

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

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

Answer: NPSHR would exceed NPSHA at just over 2500 gpm

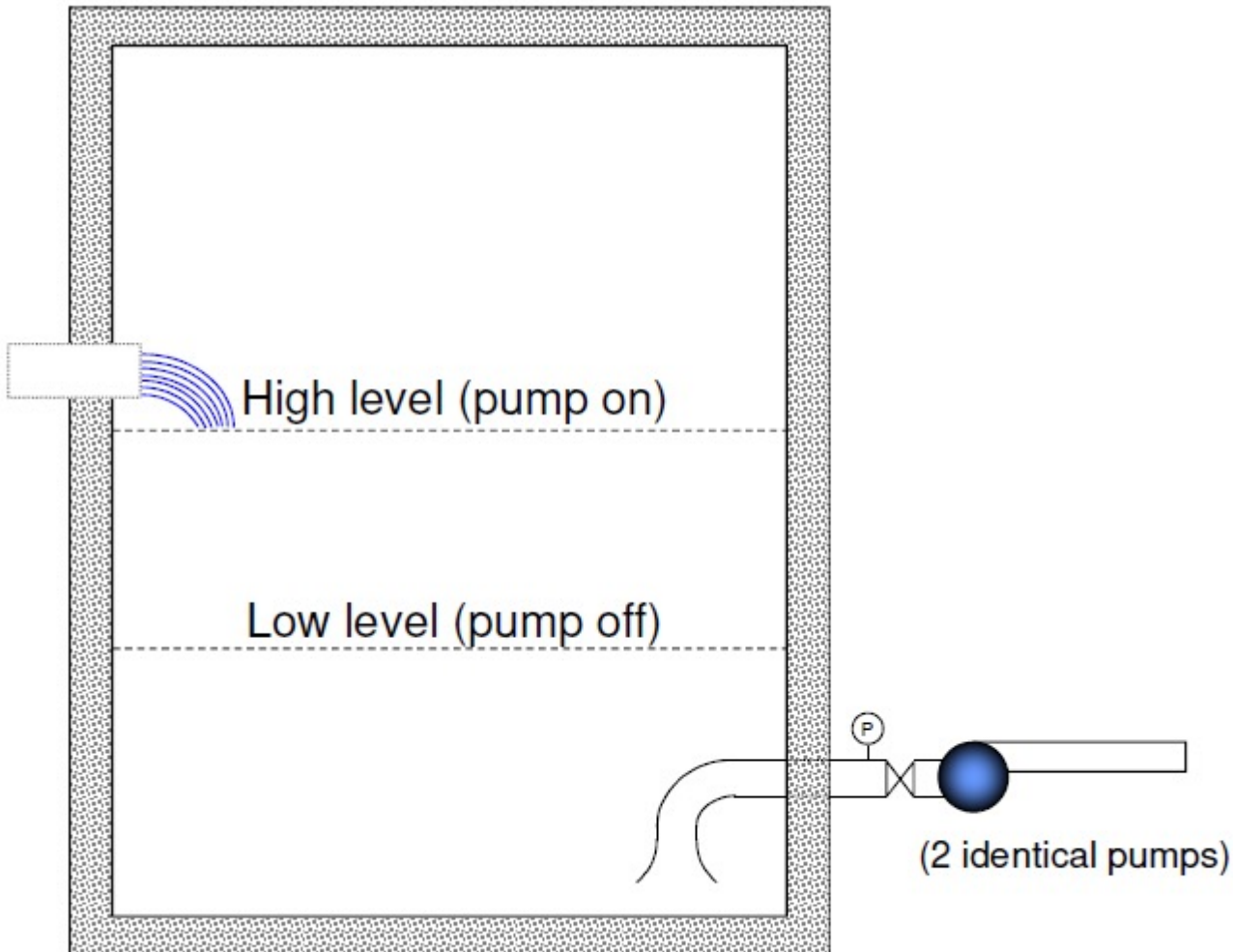


Flow estimation from suction pressure measurements

Using velocity and friction head components - two approaches

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

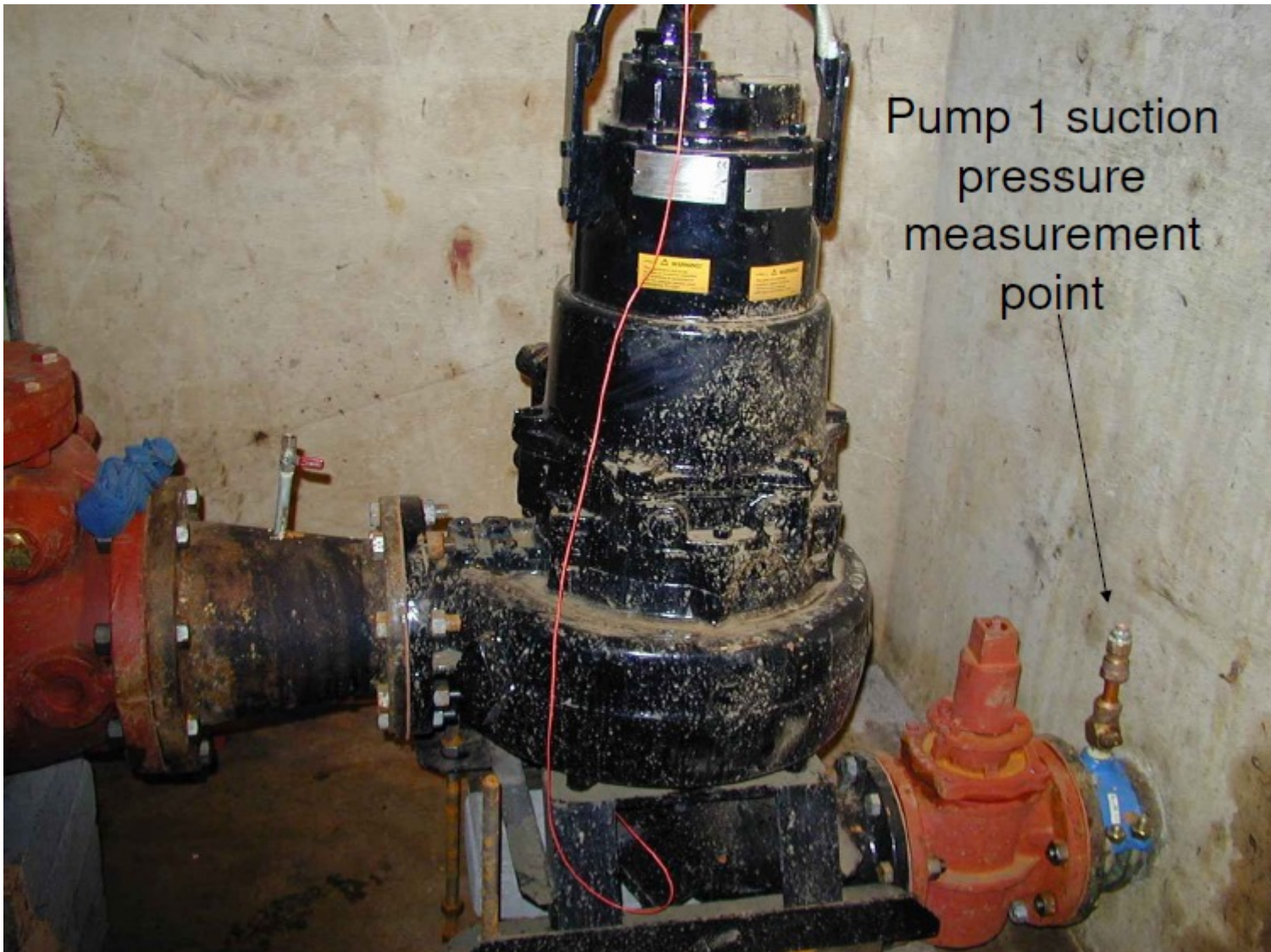
A standard wastewater lift station



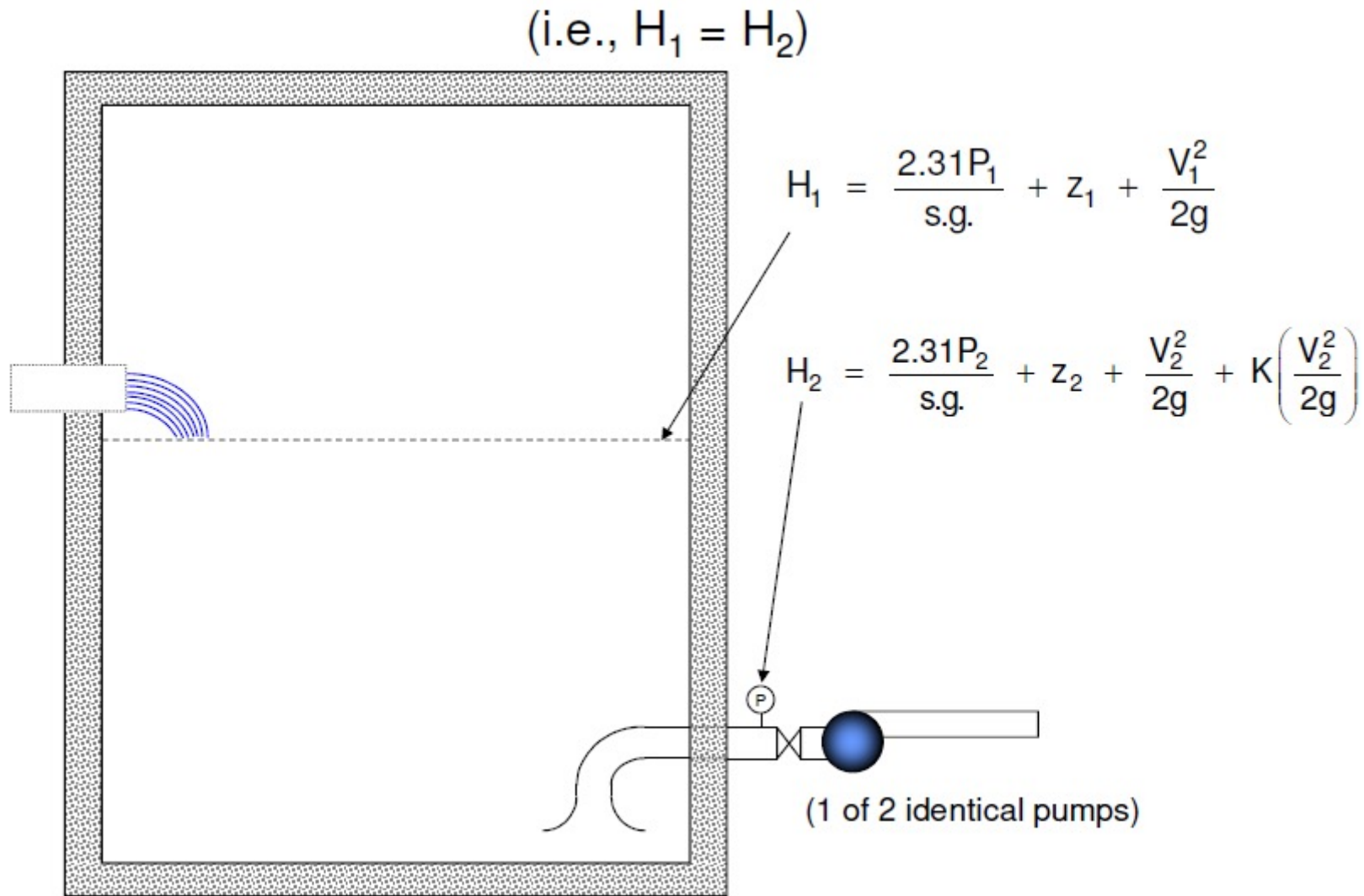
Incoming fluid spilling into the wet well



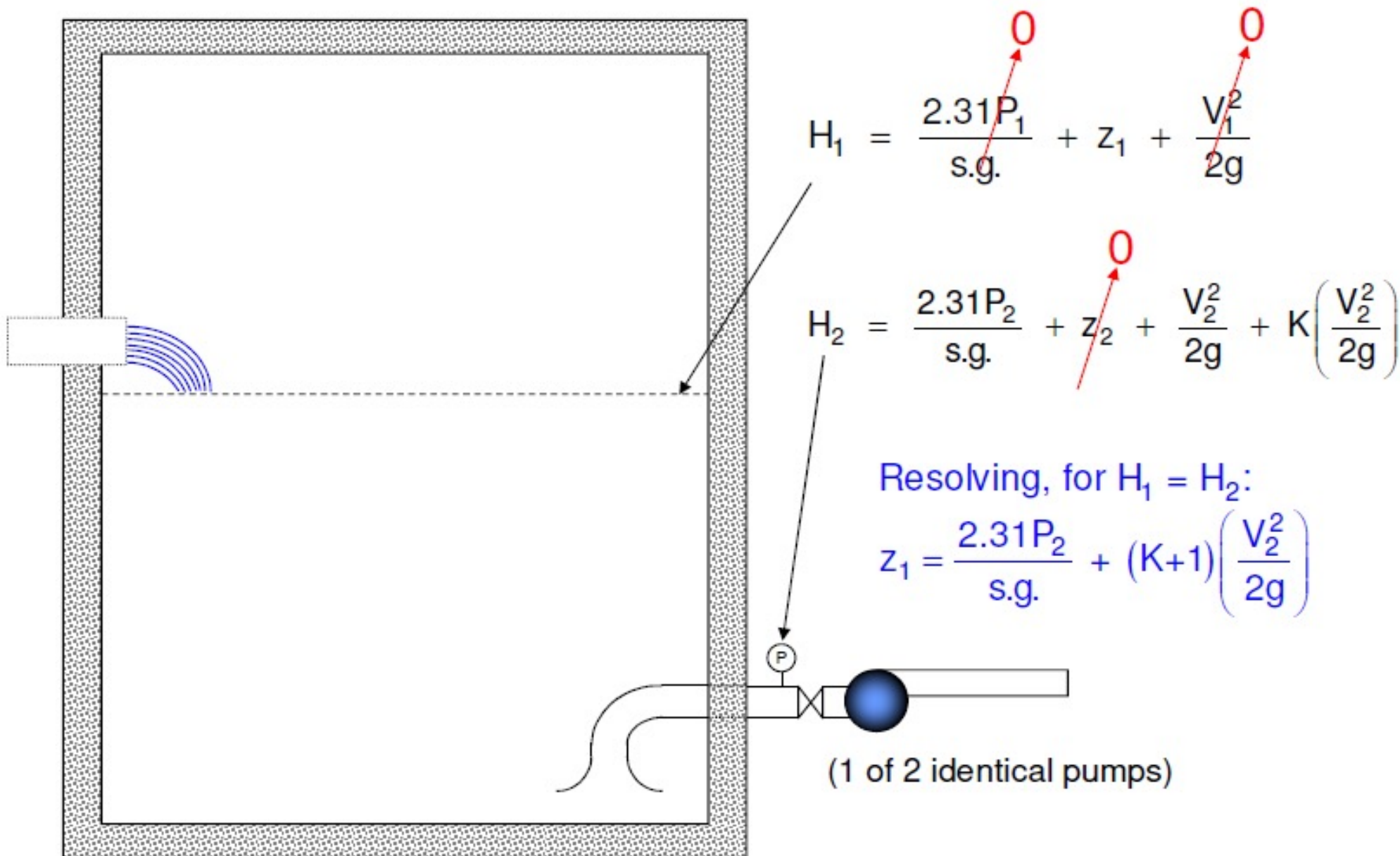
Pump 1 suction pressure measurement point



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



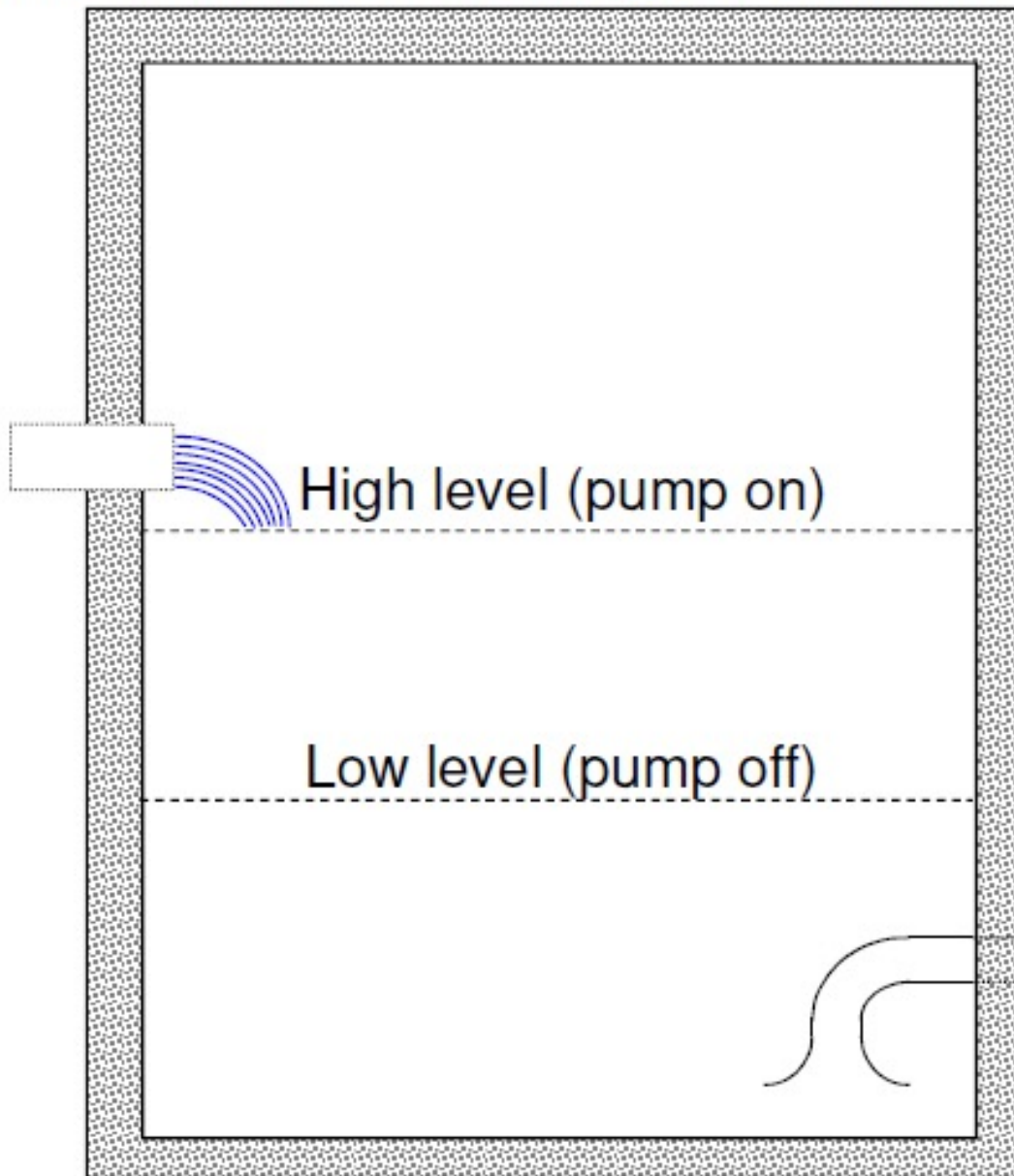
We can relate terms and arrive at:
Two knowns, one guess, and one unknown



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

An alternative perspective:

If level doesn't change instantaneously on pump start/stop.....

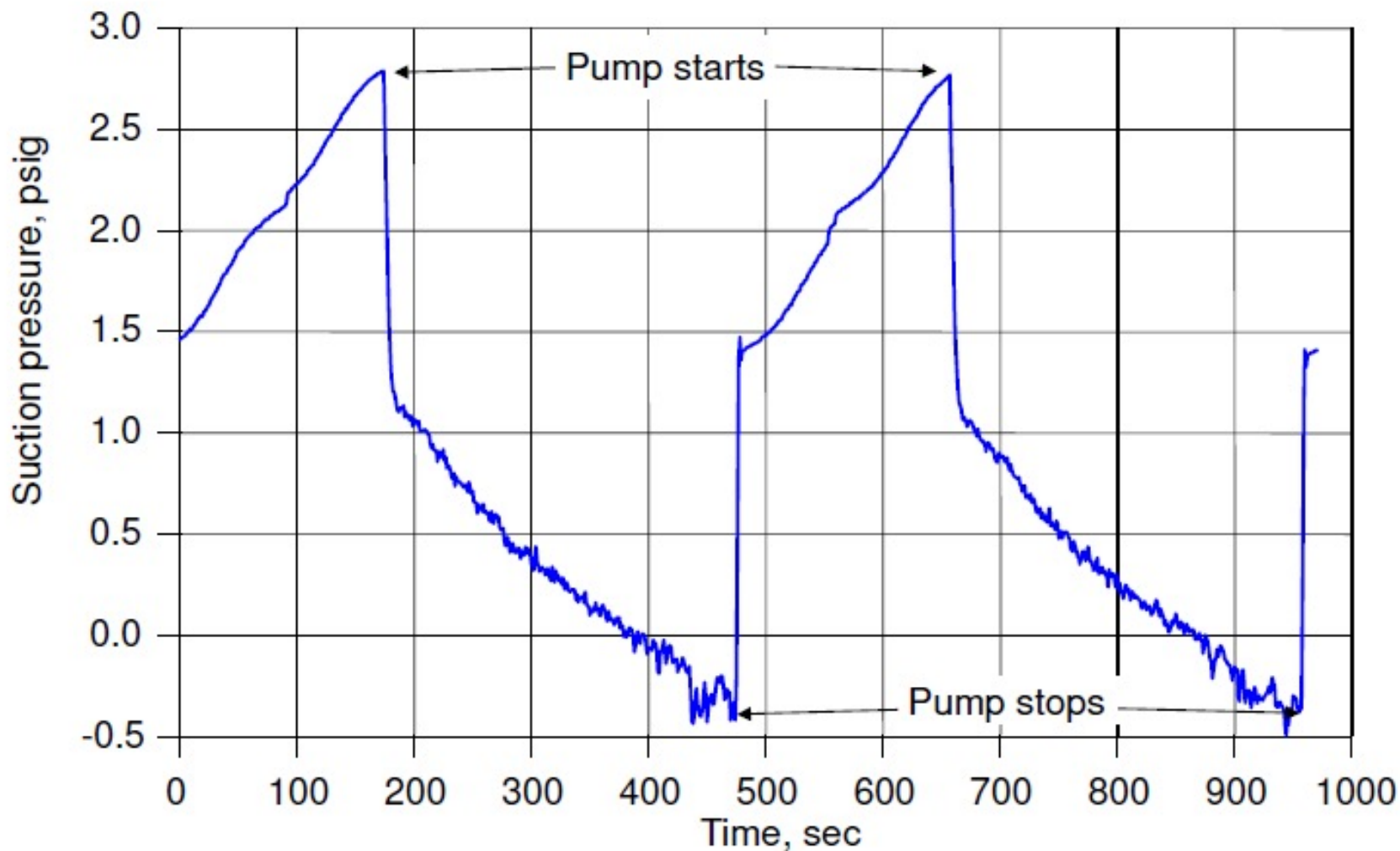


$$\frac{2.31P_{2bef}}{s.g.} + (K+1) \left(\frac{\cancel{V_{2bef}^2}}{\cancel{2g}} \right) = \frac{2.31P_{2aft}}{s.g.} + (K+1) \left(\frac{V_{2aft}^2}{2g} \right)$$

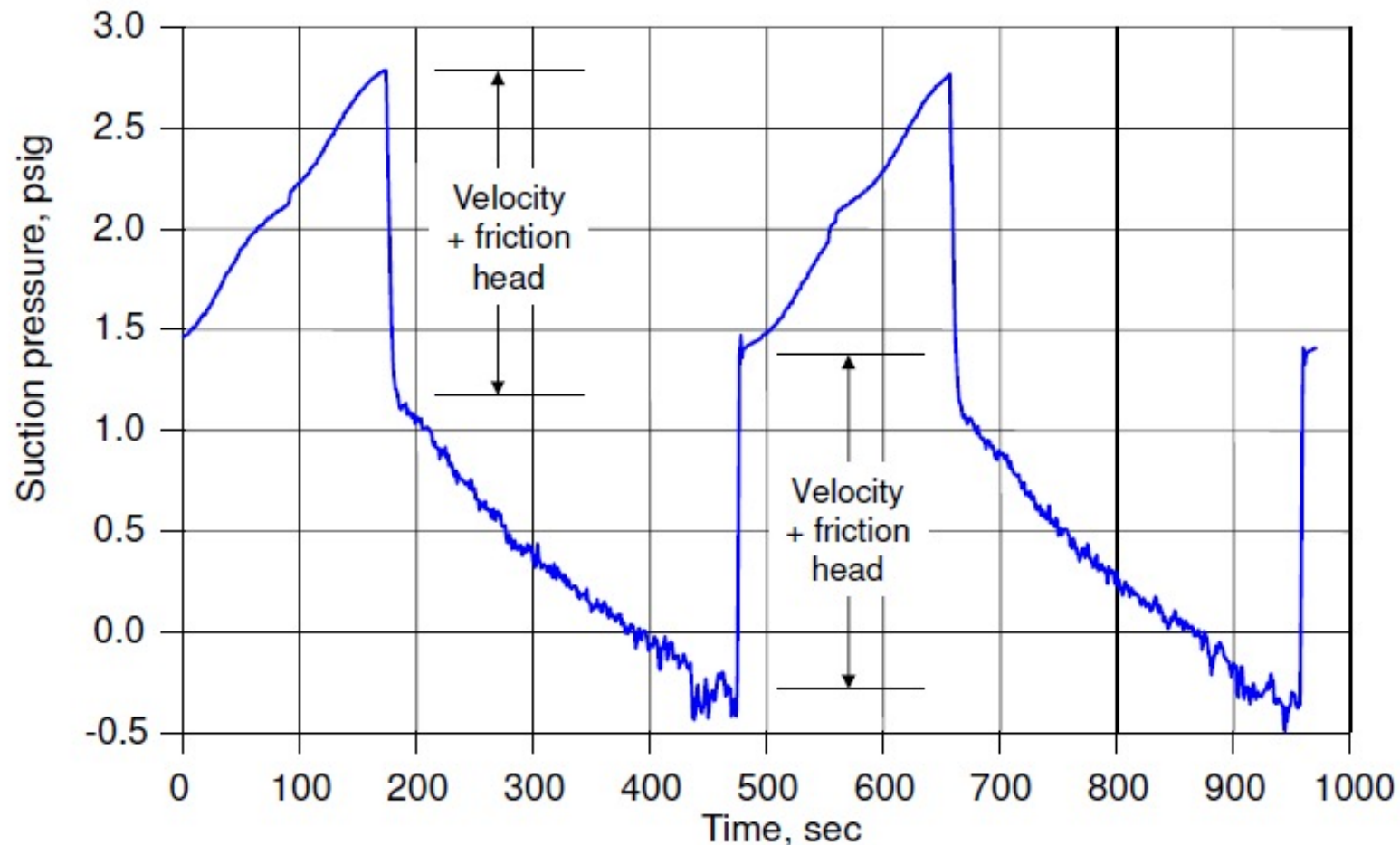
$$\frac{2.31(P_{2bef} - P_{2aft})}{s.g.} = (K+1) \left(\frac{V_{2aft}^2}{2g} \right)$$

∴ 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: $K = 0.23$

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

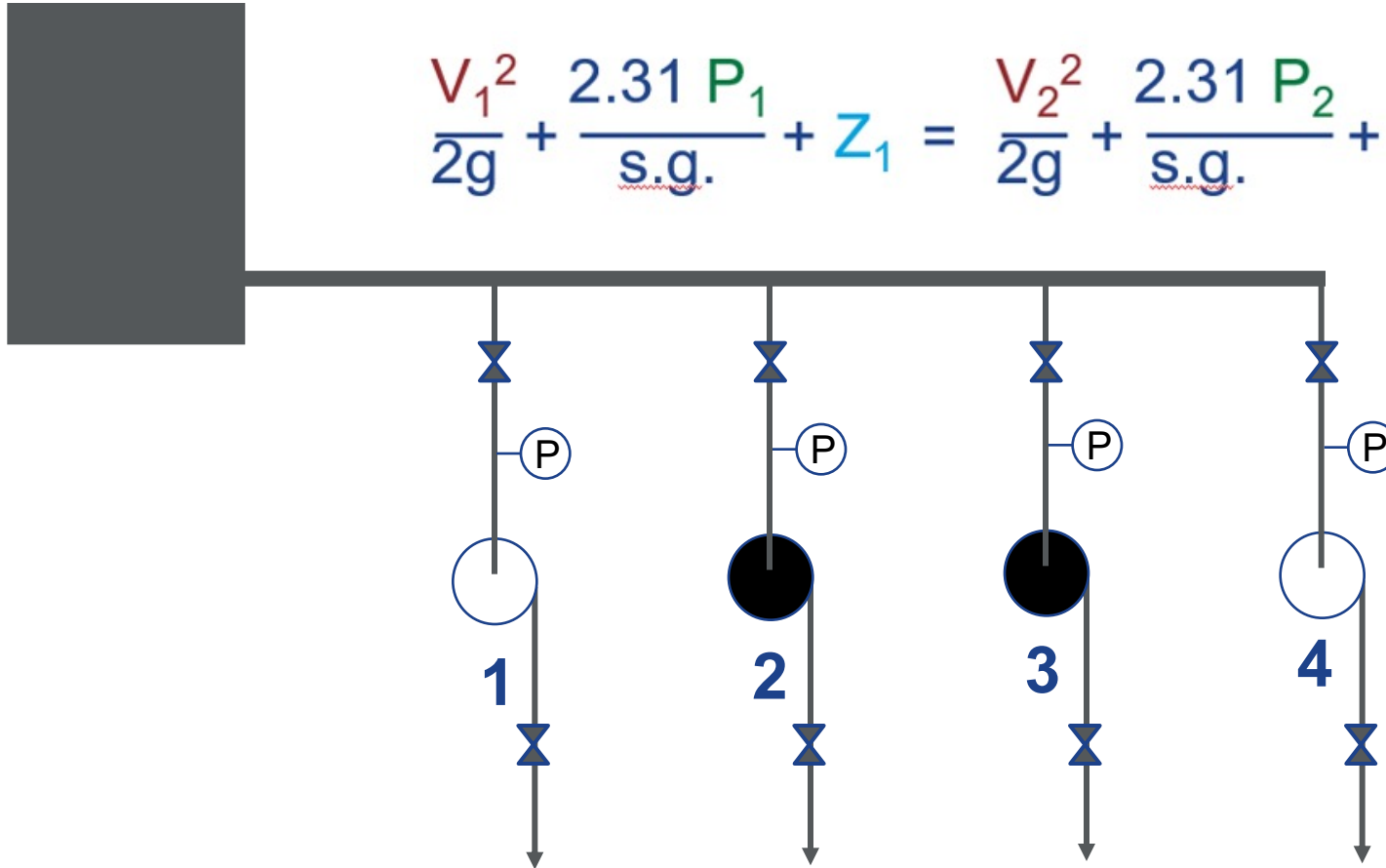
$$\Rightarrow \frac{V_2^2}{2g} = 3.14 \text{ ft}$$

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

$$\Rightarrow \text{calculated flow rate} = 1209 \text{ gpm}$$

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

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



● Off
○ On

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: $1 \text{ mV} = 1 \text{ kPa}$)



Pump 1 suction pressure
(Pump on)

Pump 2 suction pressure
(Pump off)

Differential = $11.453 \text{ kPa} = 1.66 \text{ psig} = 3.84 \text{ ft}$
(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

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							
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	1	6.69	17.5	10.80	3.62	2.94	13.76	1170
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	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
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	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

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

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

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

Manometers – even the home made kind – provide excellent accuracy



Photo by Y-12/ORNL reject

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

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

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

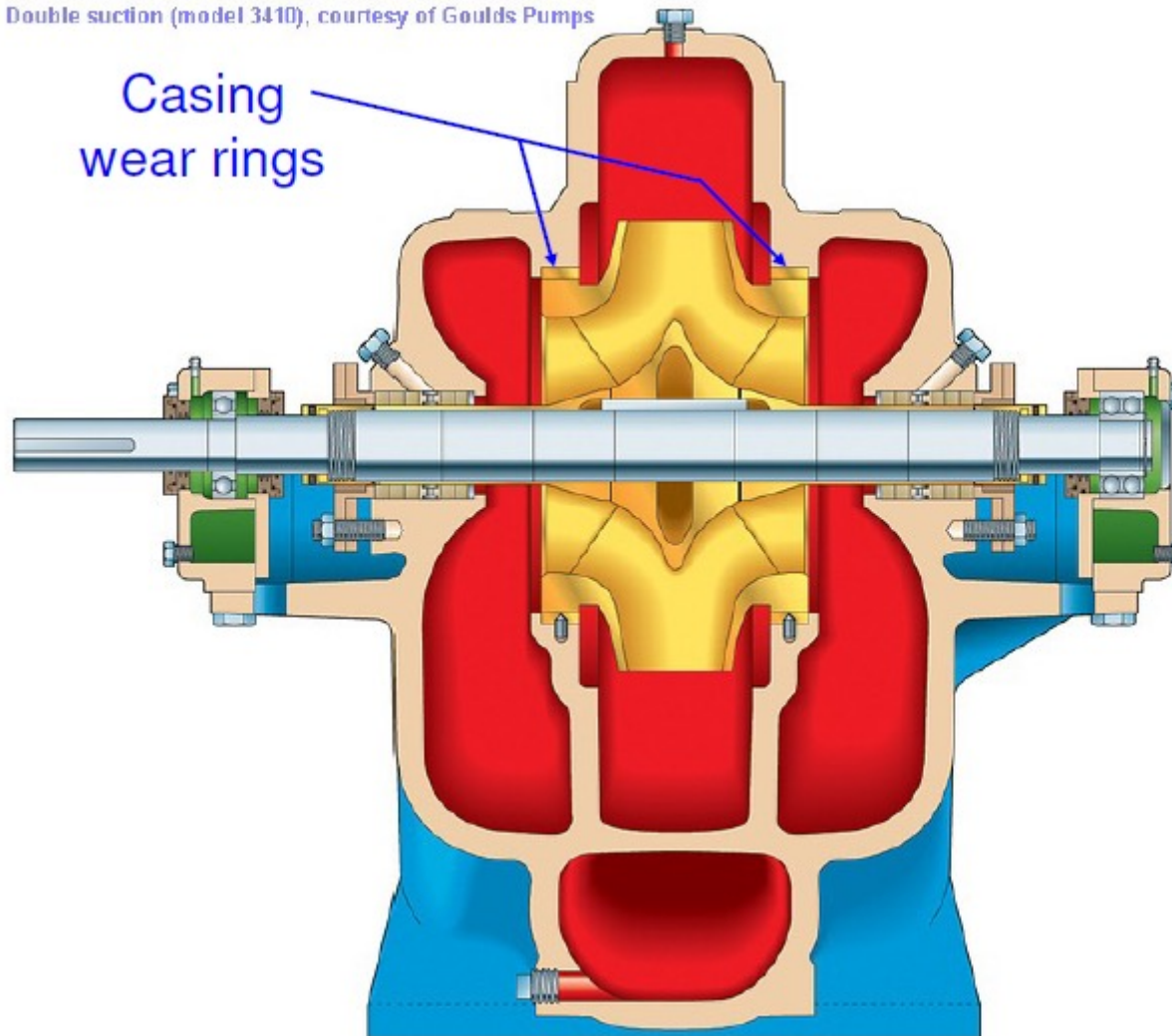
Where z is elevation in feet, g is 32.174 ft/s^2 , V is velocity in ft/s , and d is the pipe inside diameter in inches

Reasons a pump optimization rating could be low

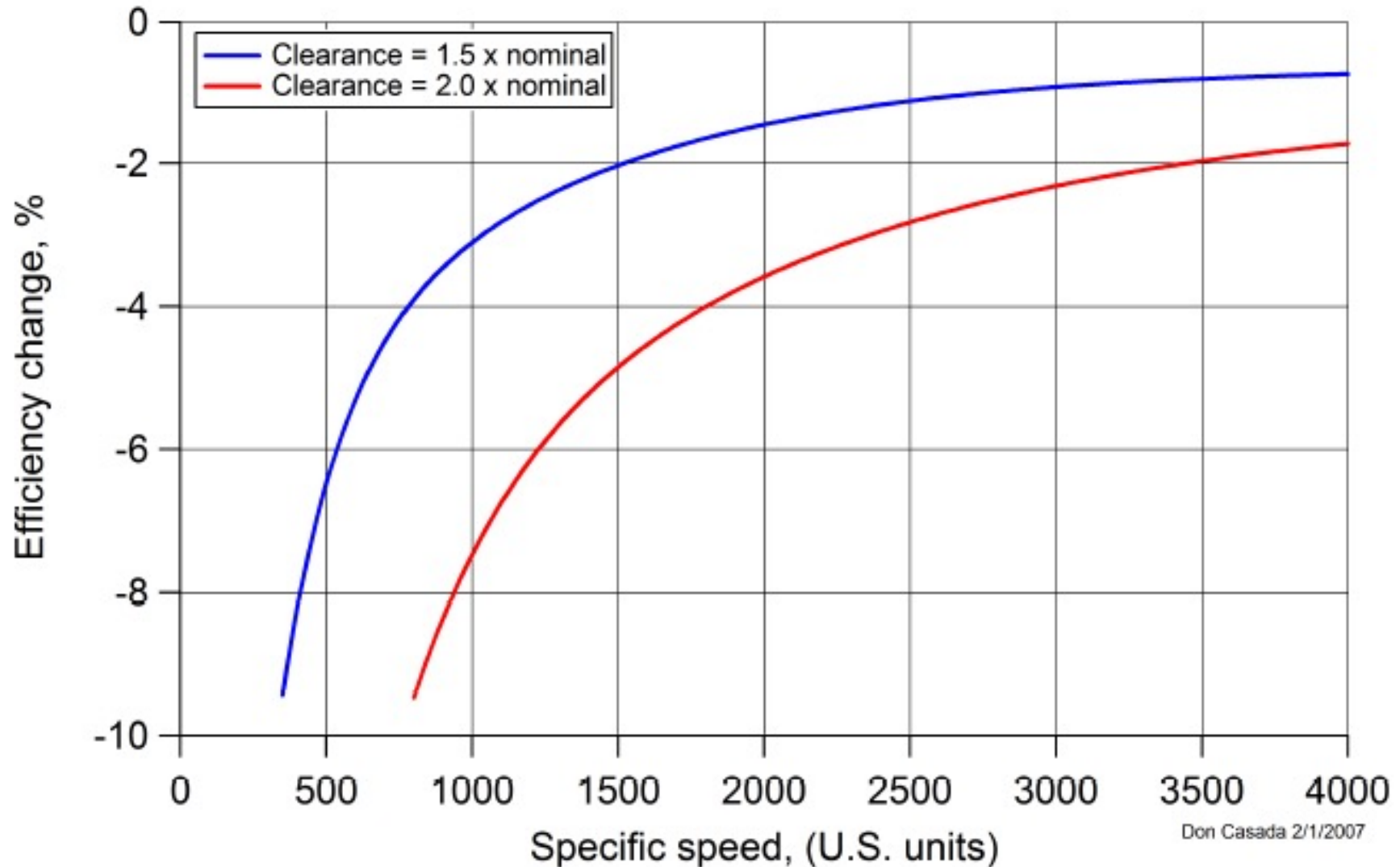
1. Pump operates away from its best efficiency point.
2. Pump parts (wear rings, impeller, volute, diffuser) are worn or have corroded/eroded.
3. Installation or operating problems, particularly on suction side.
4. Pump or motor simply aren't top of the line units.
5. Adjustable speed, belt, gear, or 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



Specific Speed

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

N_s = specific speed

N = rotational speed

Q = flow rate

H = head (per stage)

g = acceleration due to gravity

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)

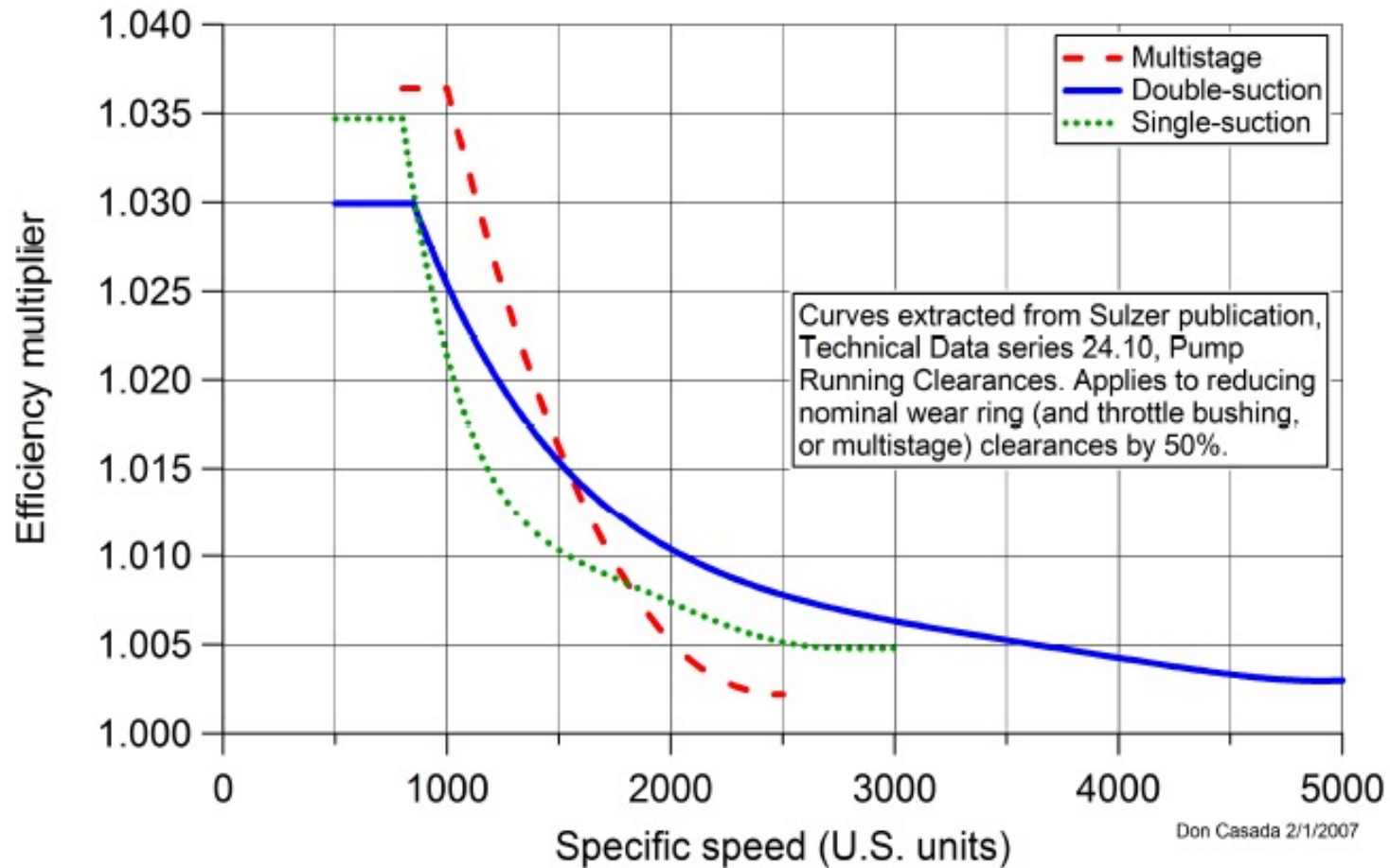
Rotating speed = 1780 rpm

Single stage pump

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

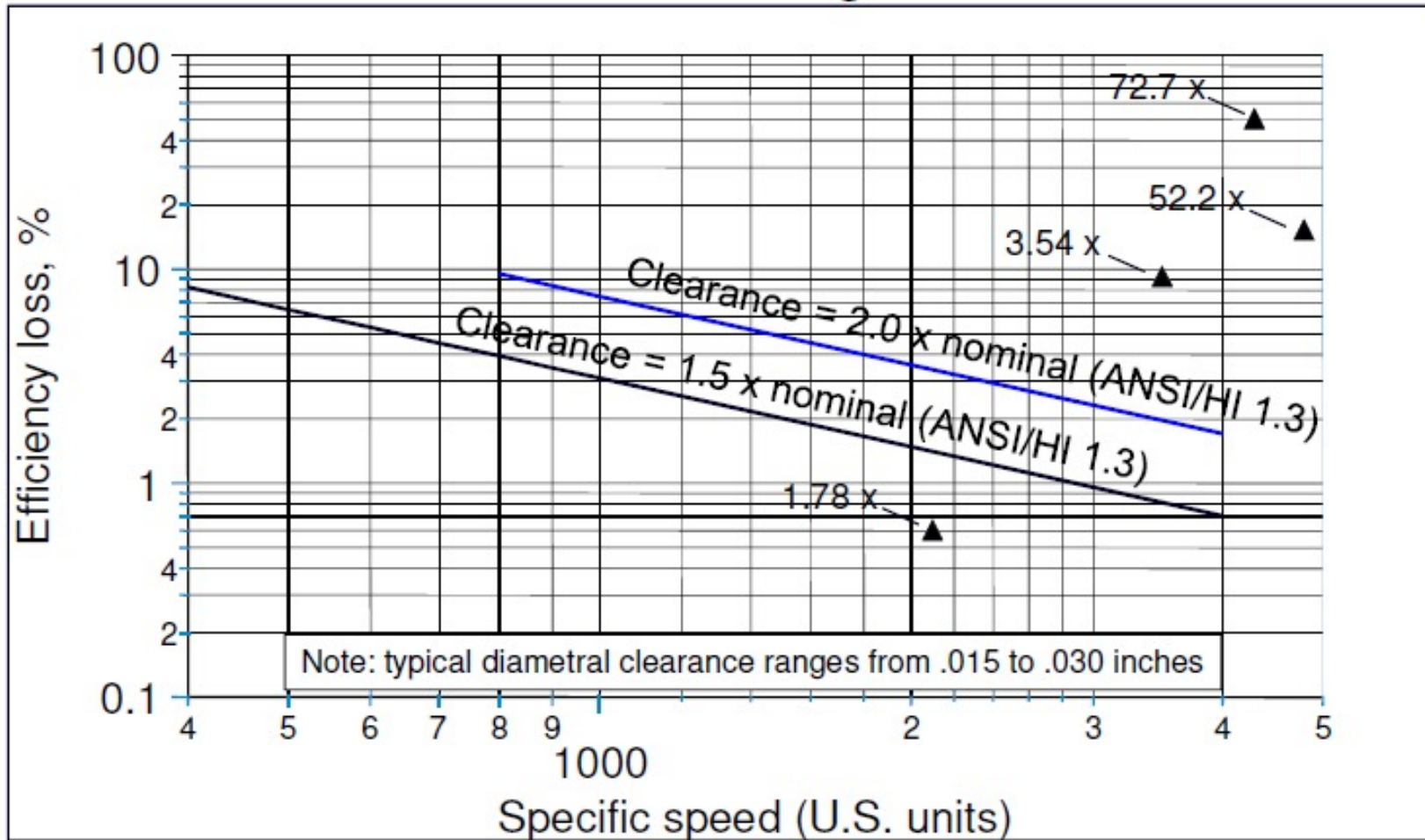
Most centrifugal pumps in common industrial applications will have specific speed values in U.S. (metric) units of between 500 (581) and 7000 (8132). Pump impellers toward the lower end of the range have a pronounced radial flow, while impellers toward the upper end of the range approach an axial flow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

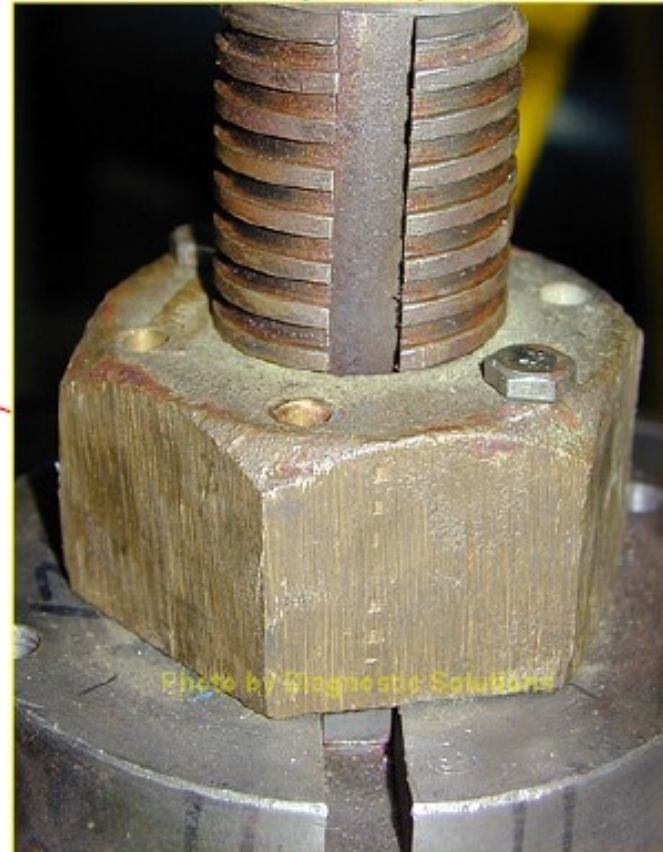


Sources { Curves: ANSI/HI 1.3, Hydraulic Institute (www.pumps.org)
Data: Flowserve, as reported in
Pump Handbook, 3rd edition, Karassik et al, McGraw-Hill

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



Vertical turbine pump lift setting



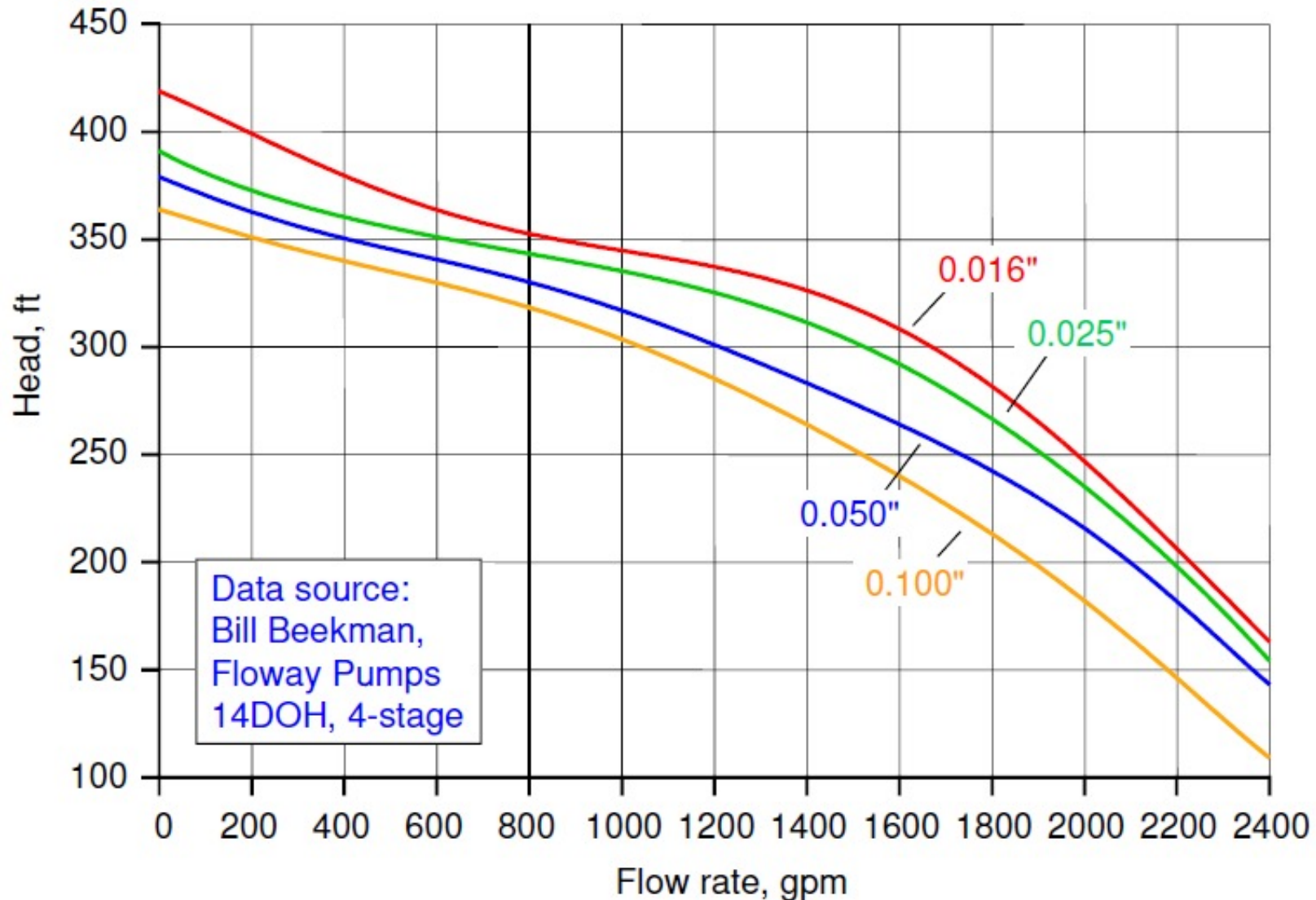
Adjustment in this case increased flow rate by 10%, efficiency by 5%

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

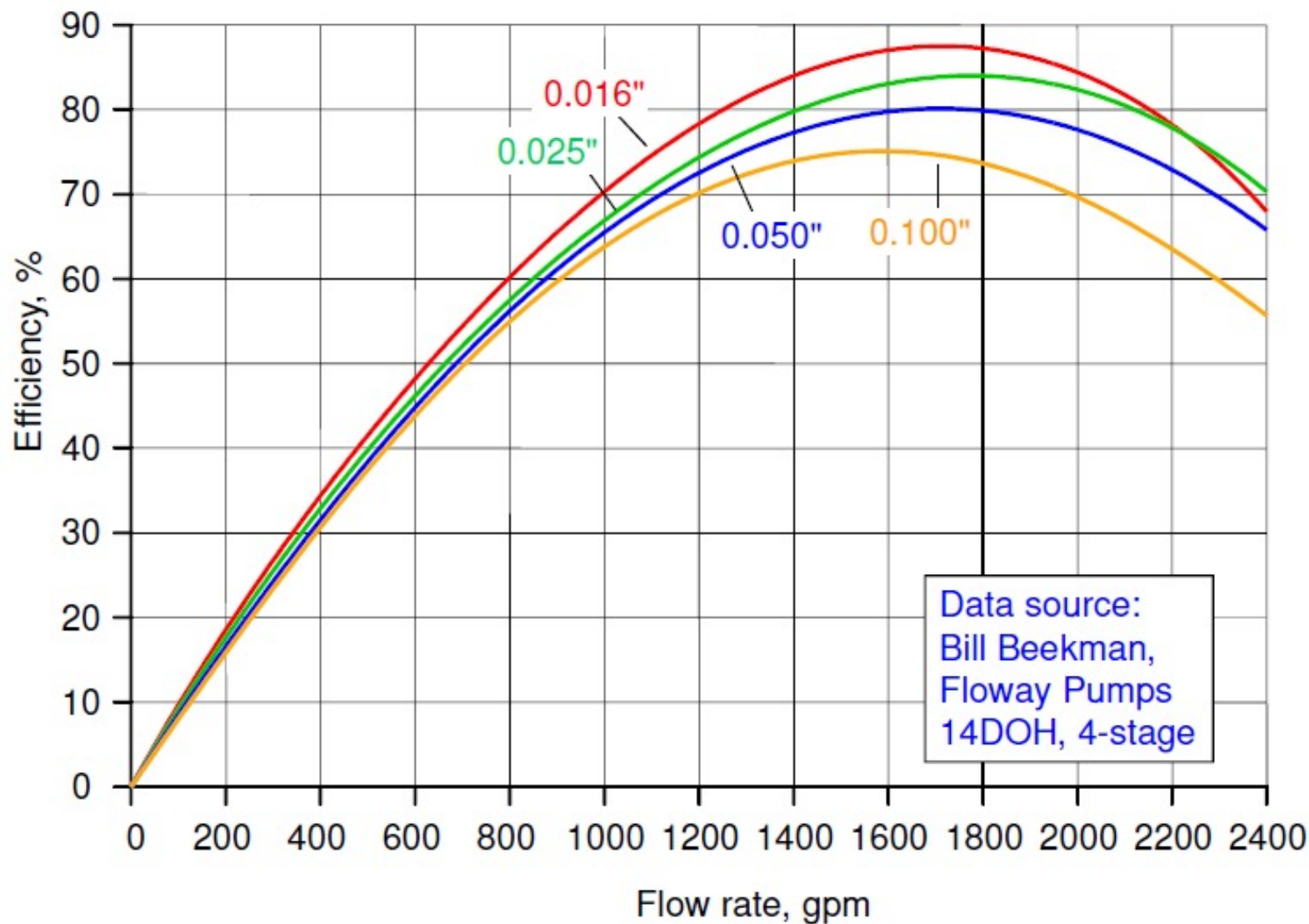
Rotation		
16-MKM	Type VTP	Stages 5
Serial No. 43005-1-1	Power 450 HP	
3020	R.P.M. 1800	Hd. 446.0
IMPELLER SETTING <u>0.125 in</u> OFF OF THE BOTTOM. (IF THIS INFORMATION IS NOT PROVIDED, CONTACT THE FACTORY FOR IMPELLER SETTING INSTRUCTIONS.)		

Size 14 DOH	Type VTP	Stages 4
Serial No. 51711-1-1	Power 200 HP	
Flow Rate 2100 GPM	R.P.M. 1800	Hd. 314
IMPELLER SETTING <u>0.011 in</u> OFF OF THE BOTTOM. (IF THIS INFORMATION IS NOT PROVIDED, CONTACT THE FACTORY FOR IMPELLER SETTING INSTRUCTIONS.)		

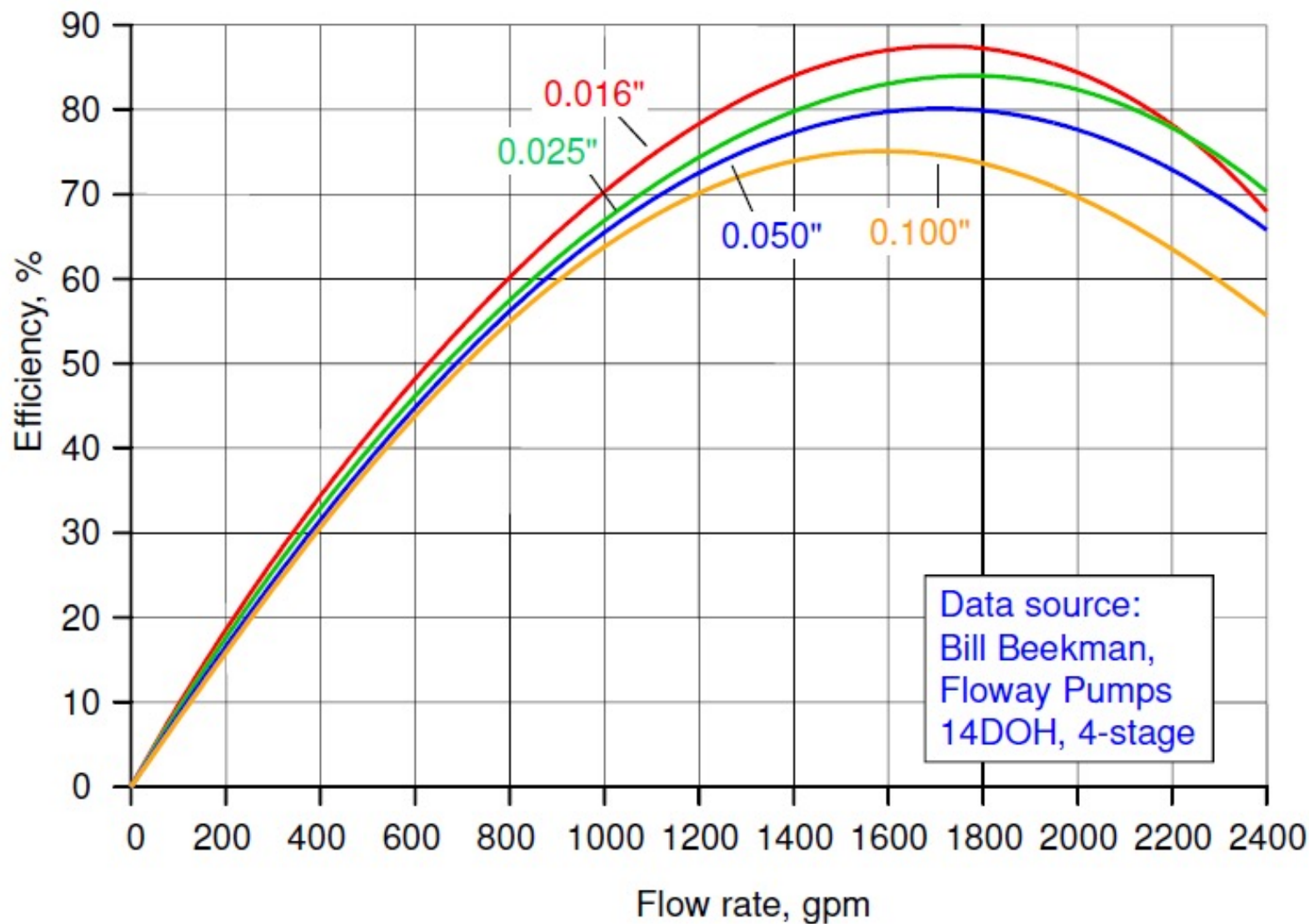
Axial clearance head-capacity effect: semi-open impeller



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



Photo by Diagnostic Solutions, LLC

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

However, PSAT ratings were poor

Condition A

End suction sewage

Pump rpm: 565

Drive: Direct drive

Units: gpm, ft, hp

Kinematic viscosity (cS): 1.00

Specific gravity: 1.000

stages: 1

Fixed specific speed? **YES**

Line freq: 60 Hz

HP: 500

Motor rpm: 594

Eff. class: Average

Voltage: 460

Estimate FLA

Full-load amps: 664.0

Size margin, %: 15

Operating fraction: 0.700

\$/kwhr: 0.1000

Flow rate, gpm: 19700

Head tool: Head, ft: 40.0

Load estim. method: Power

Motor kW: 280.0

Voltage: 482

Retrieve defaults | Set defaults | Copy A > to B > | System curve tool: select below

Condition B

End suction sewage

Pump rpm: 565

Drive: Direct drive

Units: gpm, ft, hp

Kinematic viscosity (cS): 1.00

Specific gravity: 1.000

stages: 1

Fixed specific speed? **YES**

Line freq: 60 Hz

HP: 500

Motor rpm: 594

Eff. class: Average

Voltage: 460

Estimate FLA

Full-load amps: 664.0

Size margin, %: 15

Operating fraction: 0.700

\$/kwhr: 0.1000

Flow rate, gpm: 19700

Head tool: Head, ft: 40.0

Load estim. method: Power

Motor kW: 280.0

Voltage: 482

Copy B < to A < | Background information | **STOP**

	Condition A		Units	Condition B		Units
	Existing	Optimal		Existing	Optimal	
Pump efficiency	56.4	88.0	%	56.4	88.0	%
Motor rated power	500	300	hp	500	300	hp
Motor shaft power	352.7	225.9	hp	352.7	225.9	hp
Pump shaft power	352.7	225.9	hp	352.7	225.9	hp
Motor efficiency	94.0	94.0	%	94.0	94.0	%
Motor power factor	65.8	67.0	%	65.8	67.0	%
Motor current	509.4	320.4	amps	509.4	320.4	amps
Motor power	280.0	179.3	kW	280.0	179.3	kW
Annual energy	1717.0	1099.5	MWh	1717.0	1099.5	MWh
Annual cost	171.7	110.0	\$1000	171.7	110.0	\$1000
Annual savings potential, \$1,000		61.7		61.7		
Optimization rating, %		64.0		64.0		

Log file controls:

Create new log | Add to existing log

Retrieve log entry | Delete log entry

Summary file controls:

Create new summary file

Existing summary files: CREATE NEW

Condition A Notes

Facility: Anonymous | System: Lift station | Date: June 25, 2003

Application: Pump 1 | Evaluator: Don Casada

General comments: Pumps 1 and 4 running in parallel power and flow halved. Represents daytime operation. Note that the estimated shaft power overstates the actual by about 4%, since an adjustable speed drive is used on these pumps.

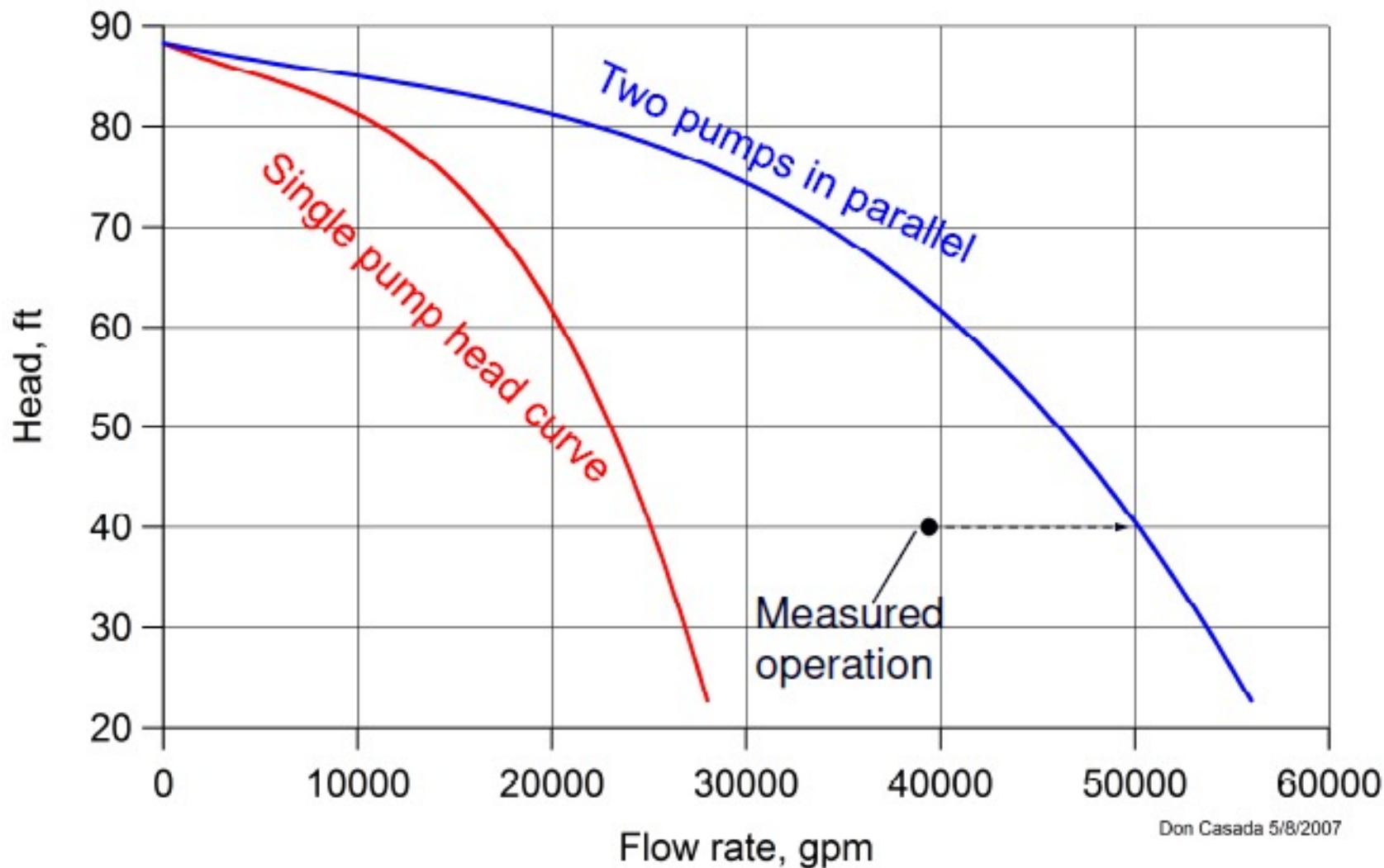
Condition B Notes

Facility: Anonymous | System: Lift station | Date: June 25, 2003

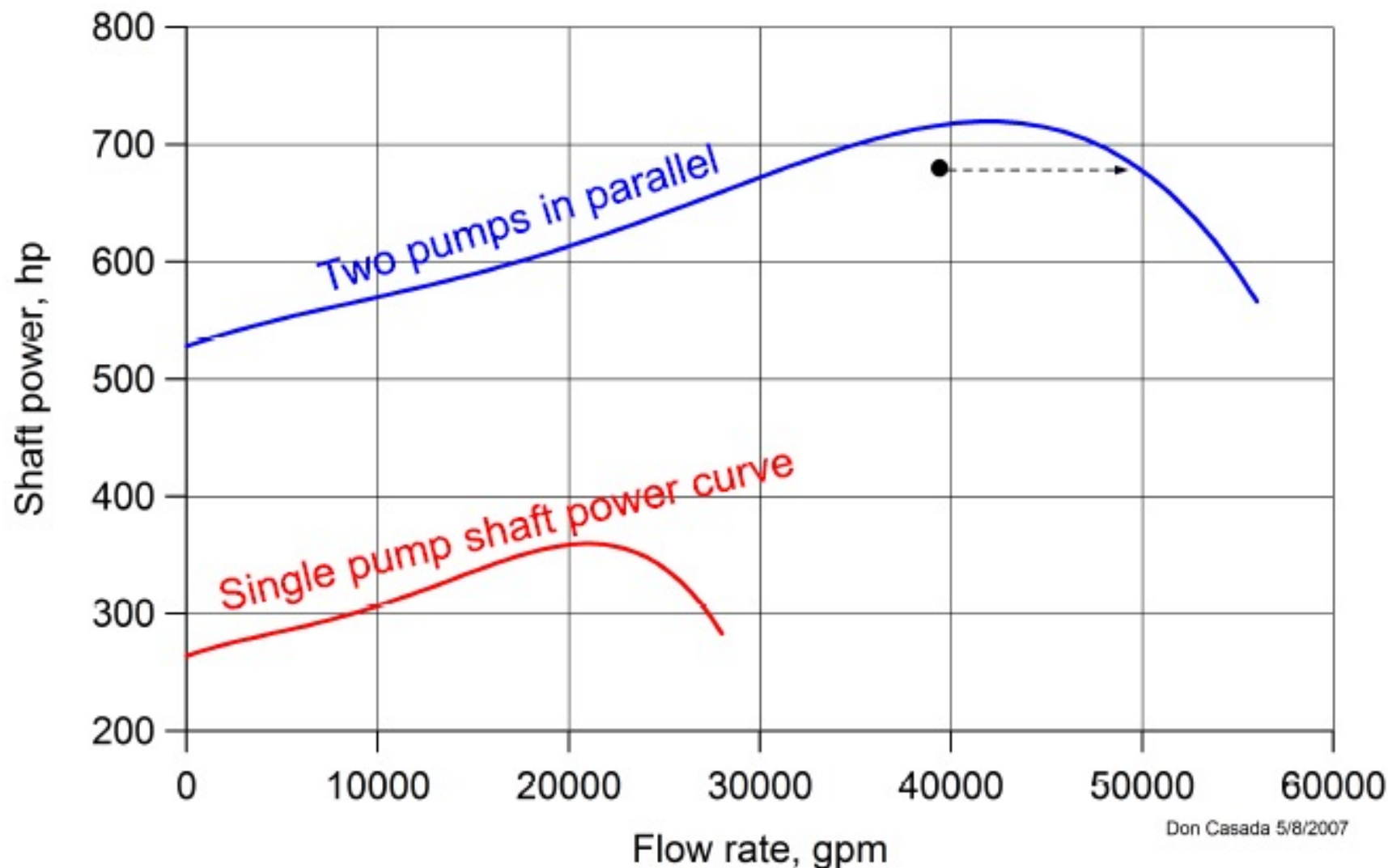
Application: Pump 4 | Evaluator: Don Casada

General comments: Pumps 1 and 4 running in parallel power and flow halved. Represents daytime operation. Note that the estimated shaft power overstates the actual by about 4%, since an adjustable speed drive is used on these pumps.

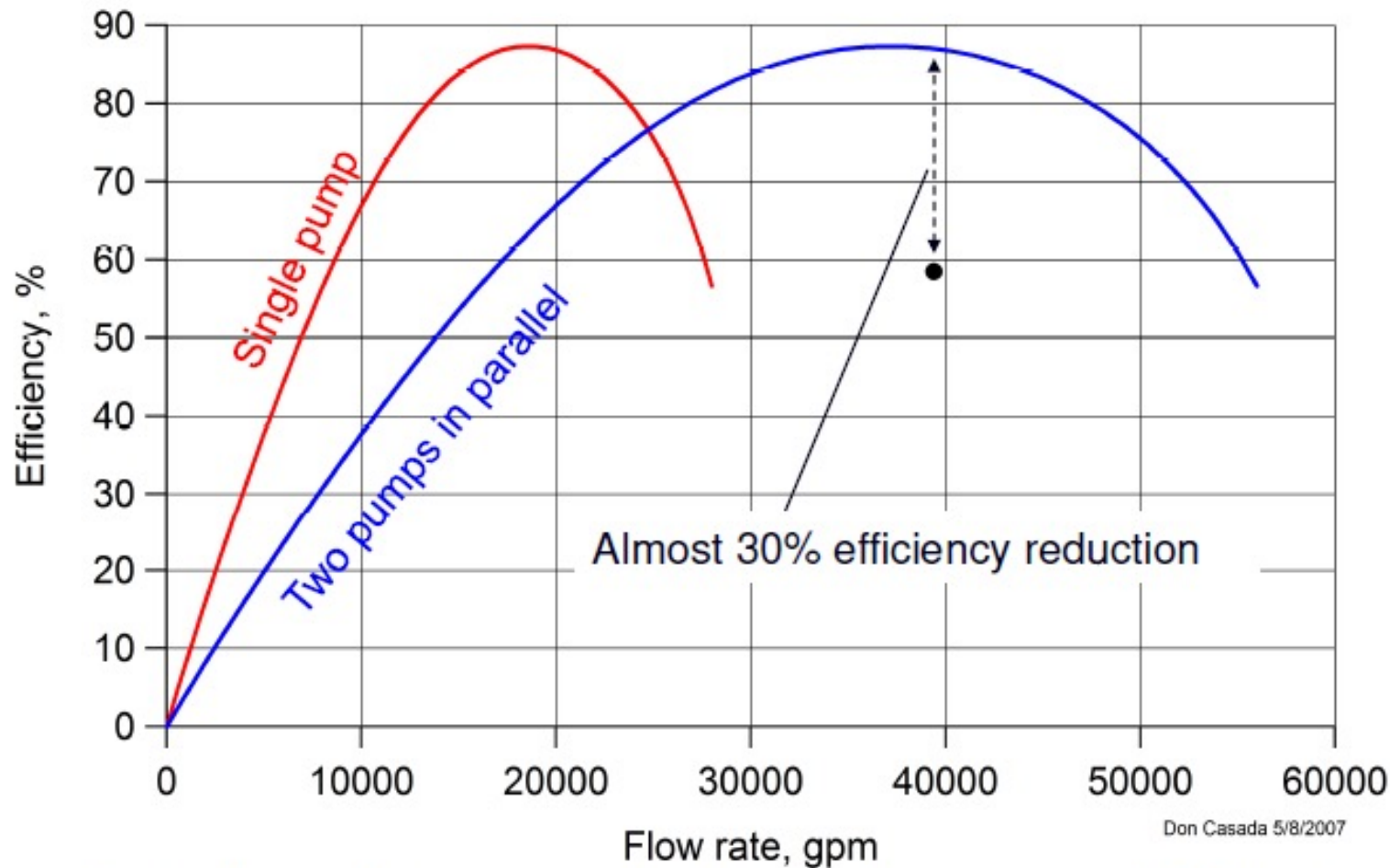
Head capacity curves and measured data



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



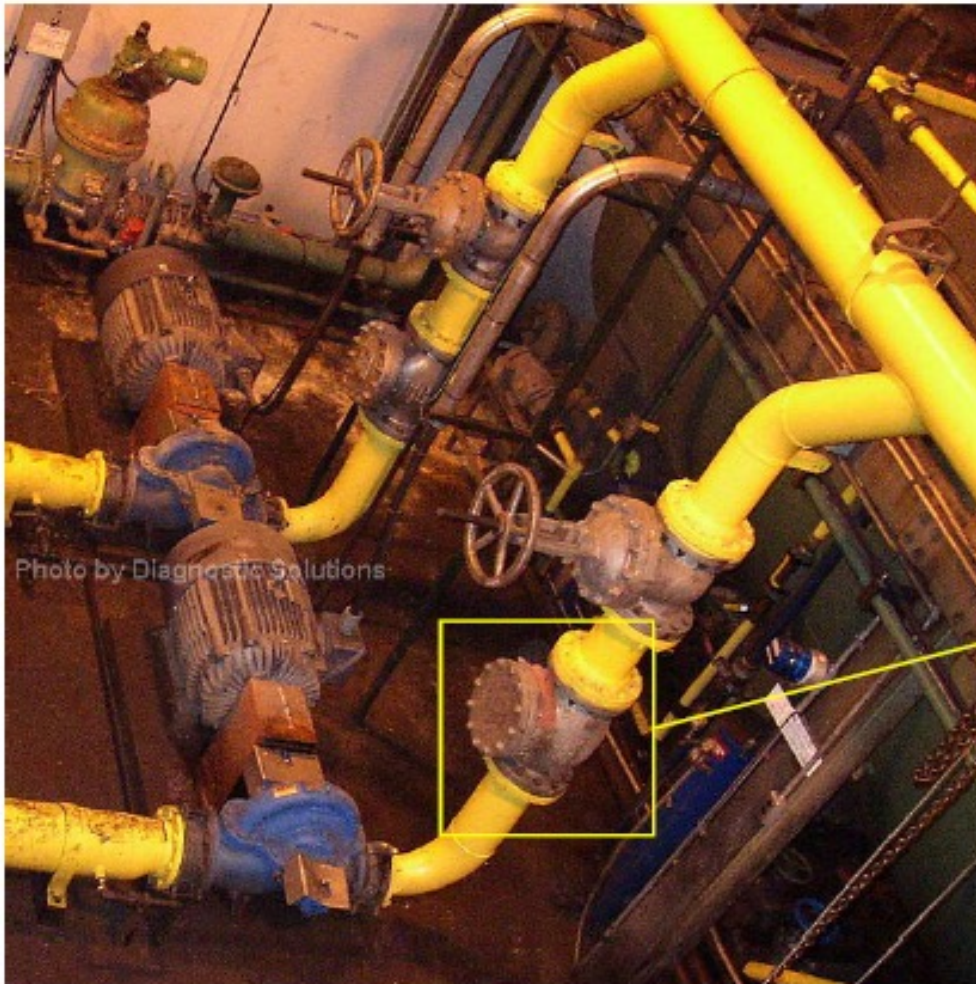
Efficiency-capacity curves and estimated pump efficiency (adjusted to account for VFD losses)



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

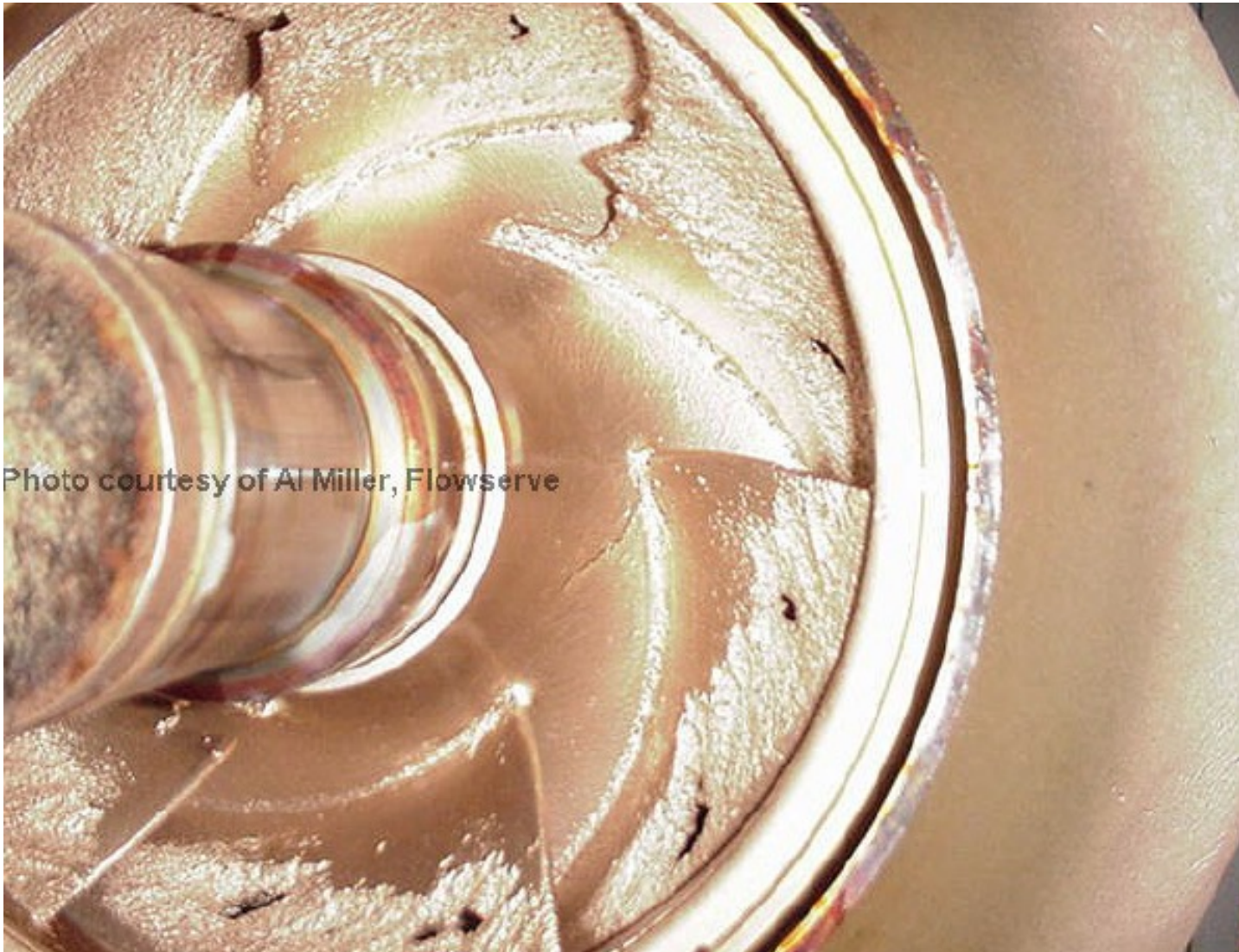


Photo courtesy of Al Miller, Flowserve

Cavitation Damage – Waste Lift Station Pump



Photo by: Diagnostic Solutions, LLC

Bad Suction Geometry



Bad Suction Geometry



Photo by Diagnostic Solutions, LLC



The End