The Little Red Schoolhouse

Large Chilled Water System Design Seminar

Courtesy of Oslin Nation Company

Centrifugal Pump Fundamentals

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The Pump Performance (Impeller) Curve

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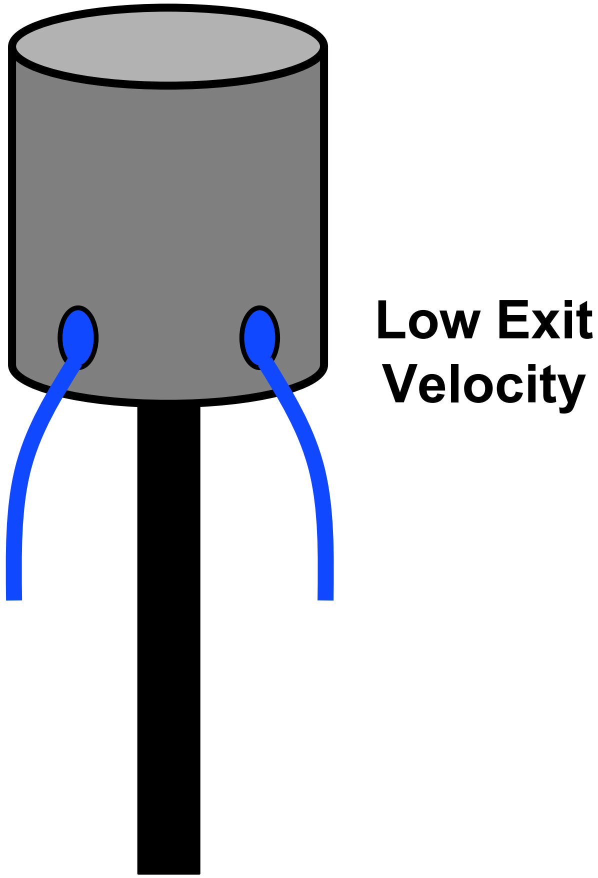
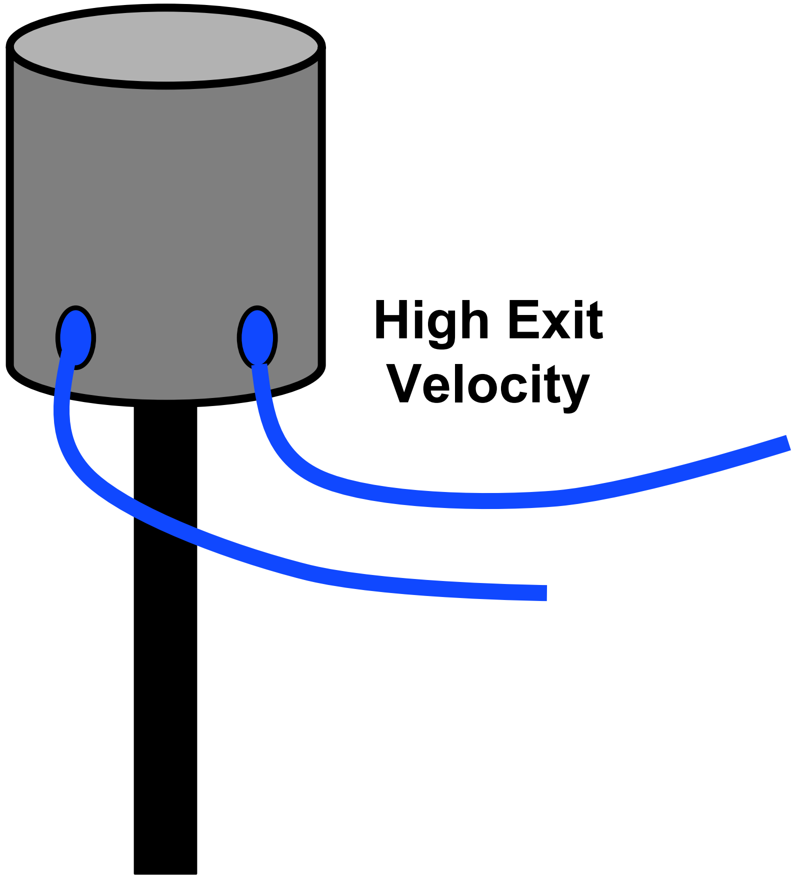


No Rotation



Rotating

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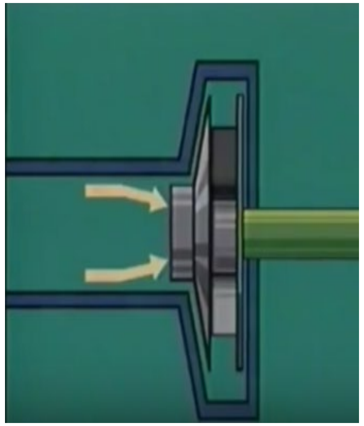
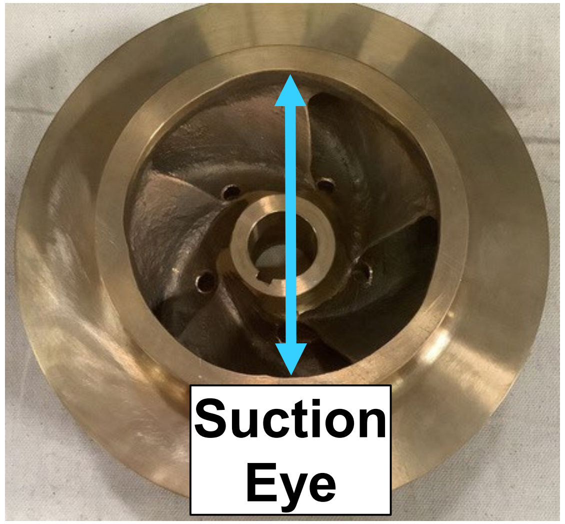
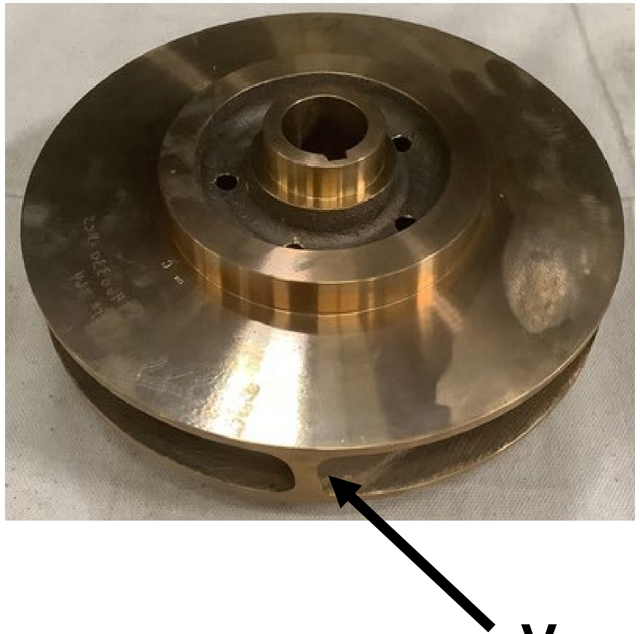




Shrouds



Vane

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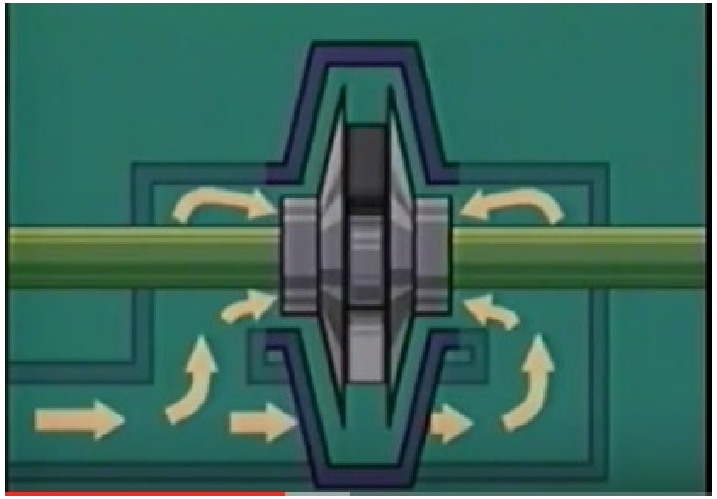

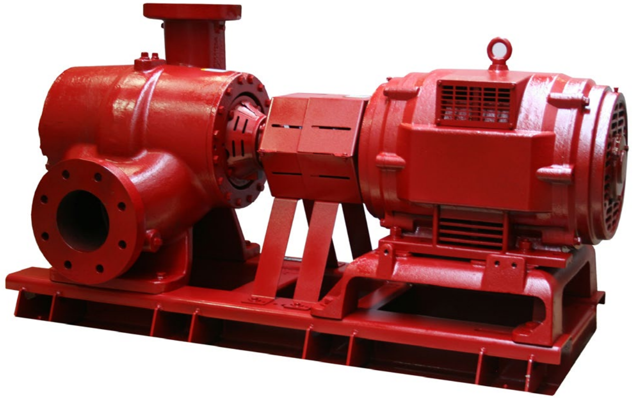

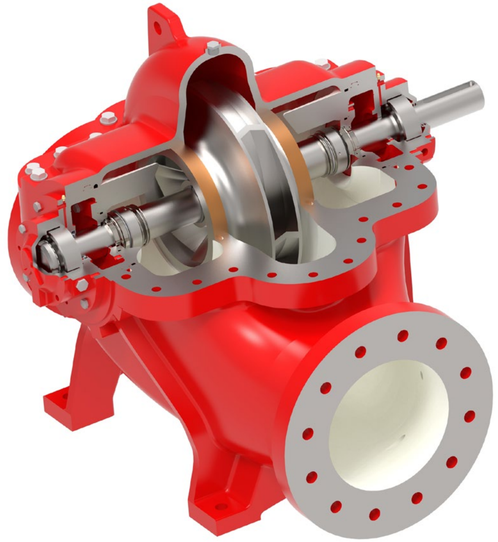

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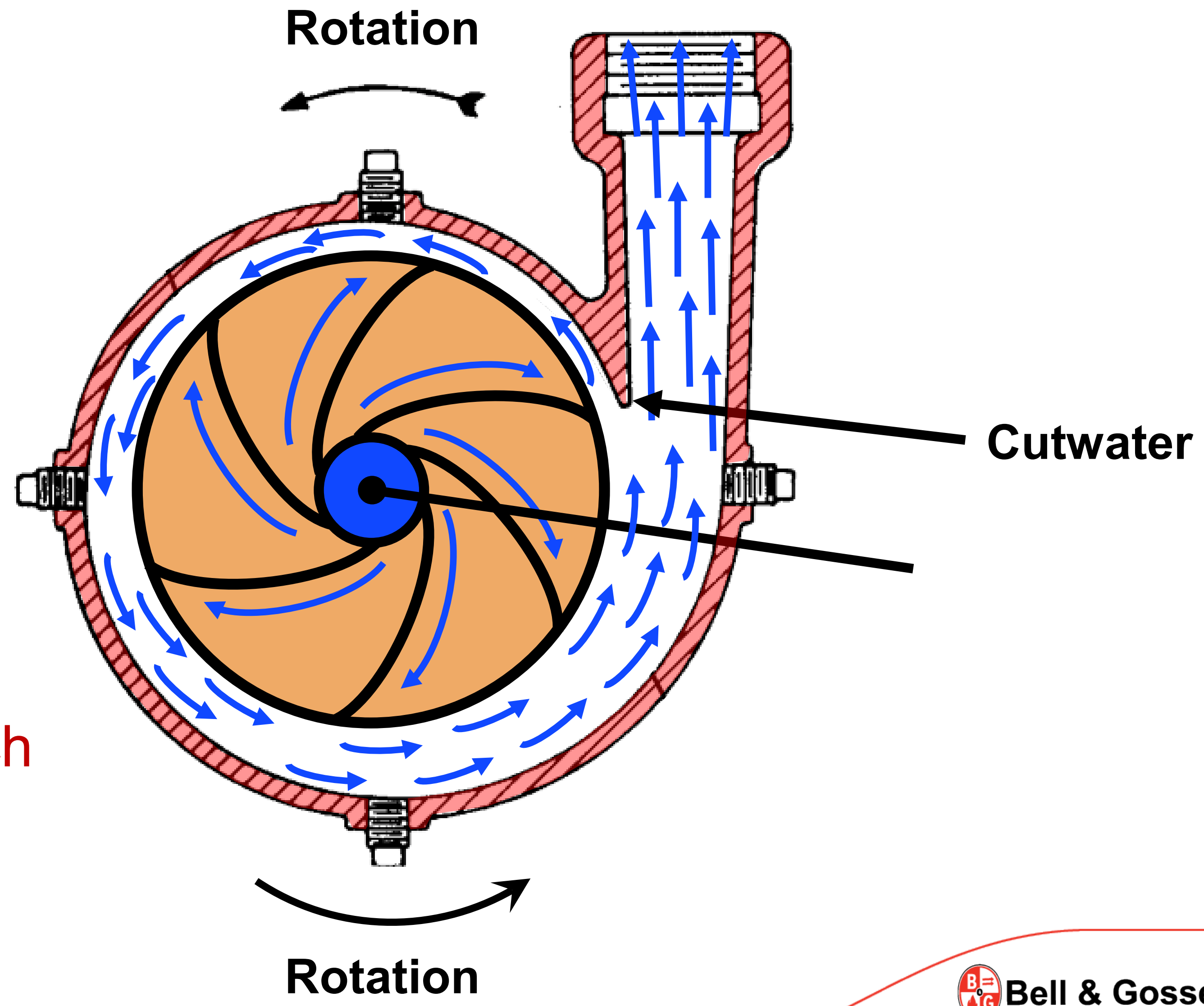
Transforming Velocity to Pressure – **Pump Head**

Impeller and Volute

Centrifugal Pump Components:

- Impeller
- Volute
- Driver *(Motor)*

The proper rotation of impeller is such that vanes are **“Slapping”** the fluid.



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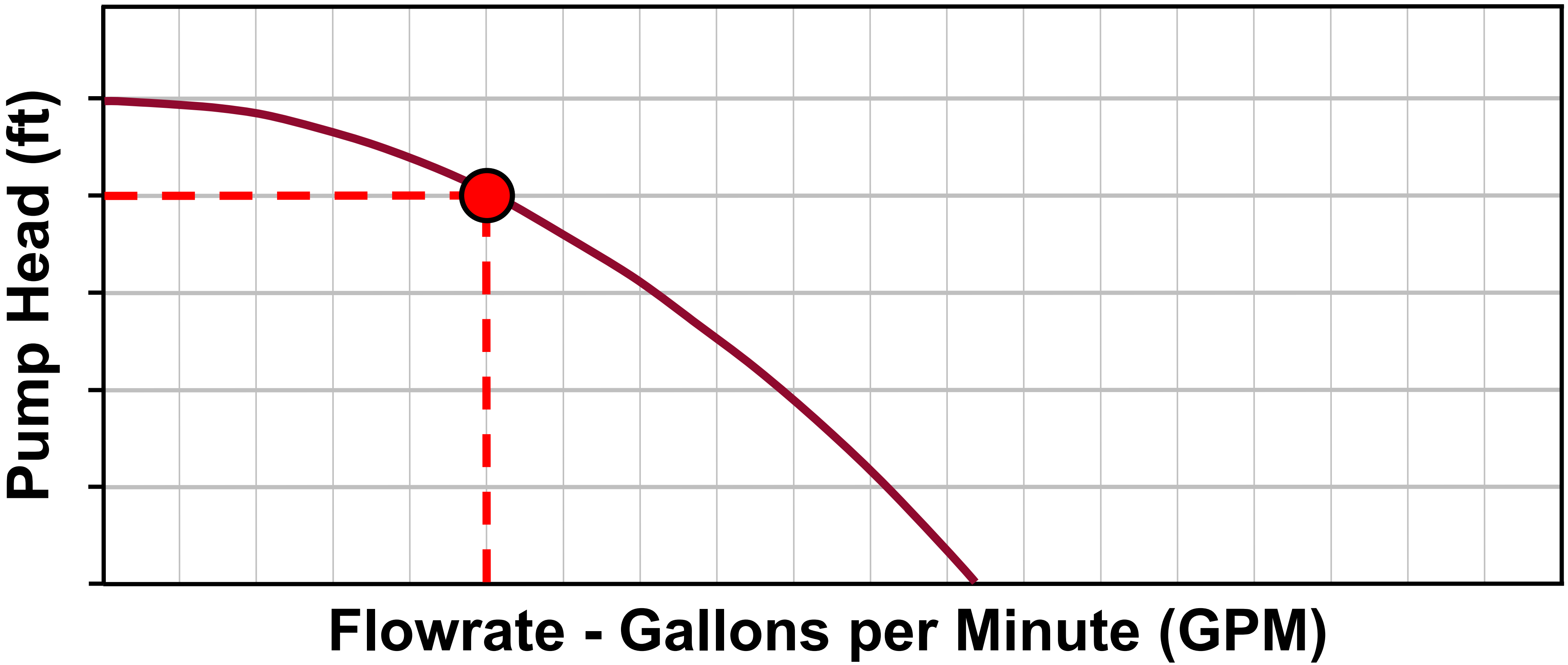

- Any combination of flow and head **must** intersect on the Pump Impeller Performance Curve

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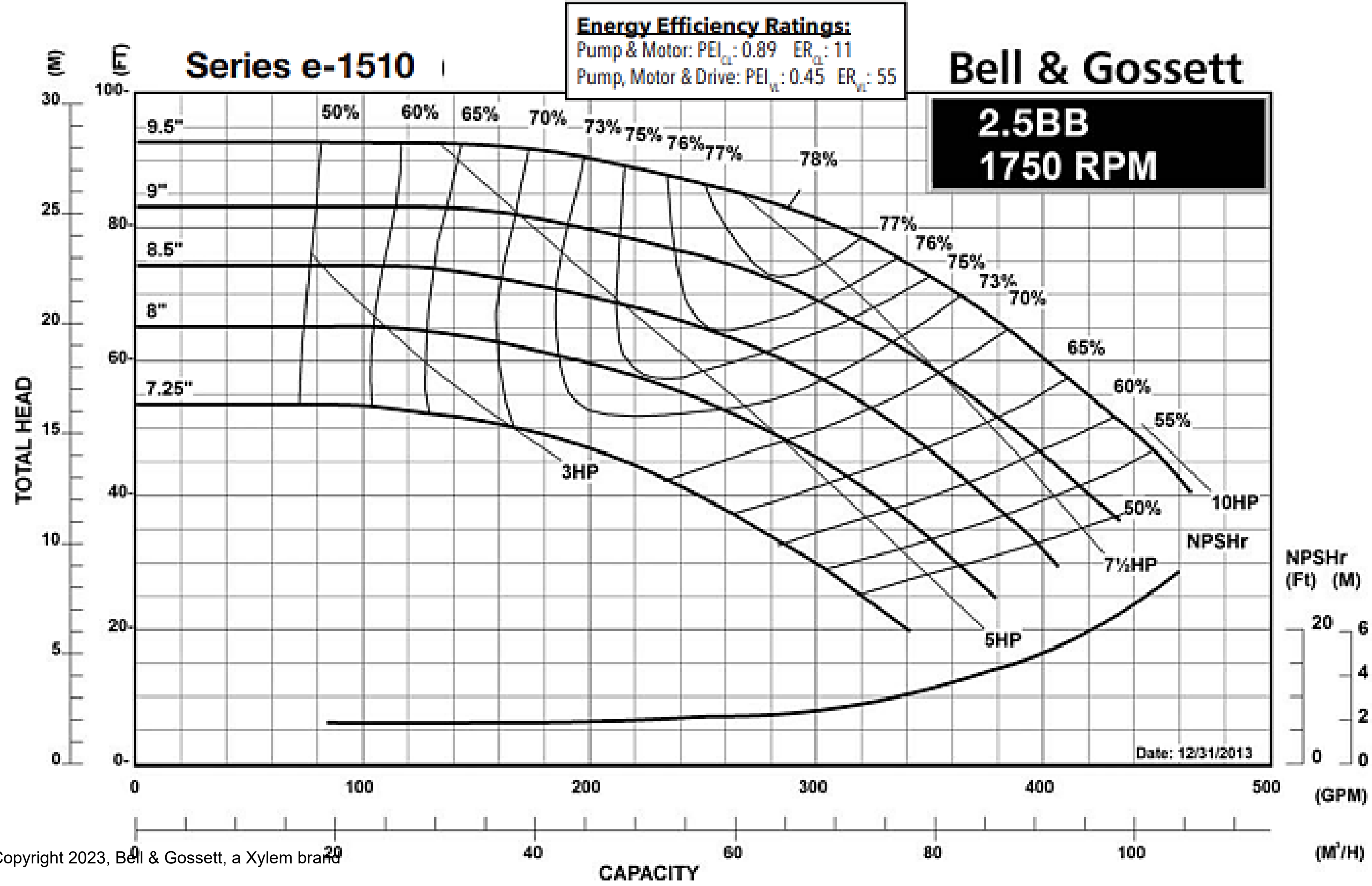

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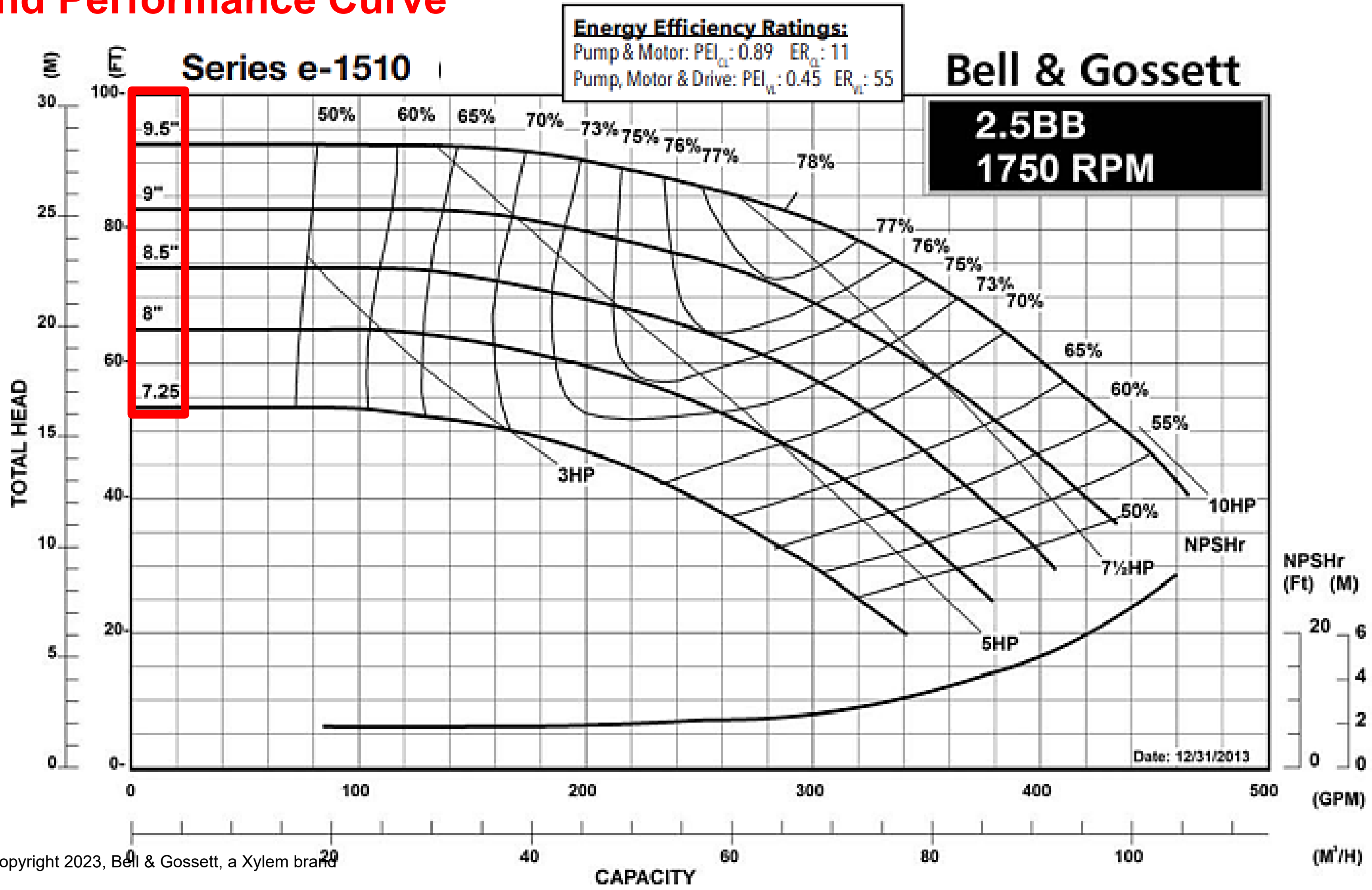
Impeller Diameter and Performance Curve



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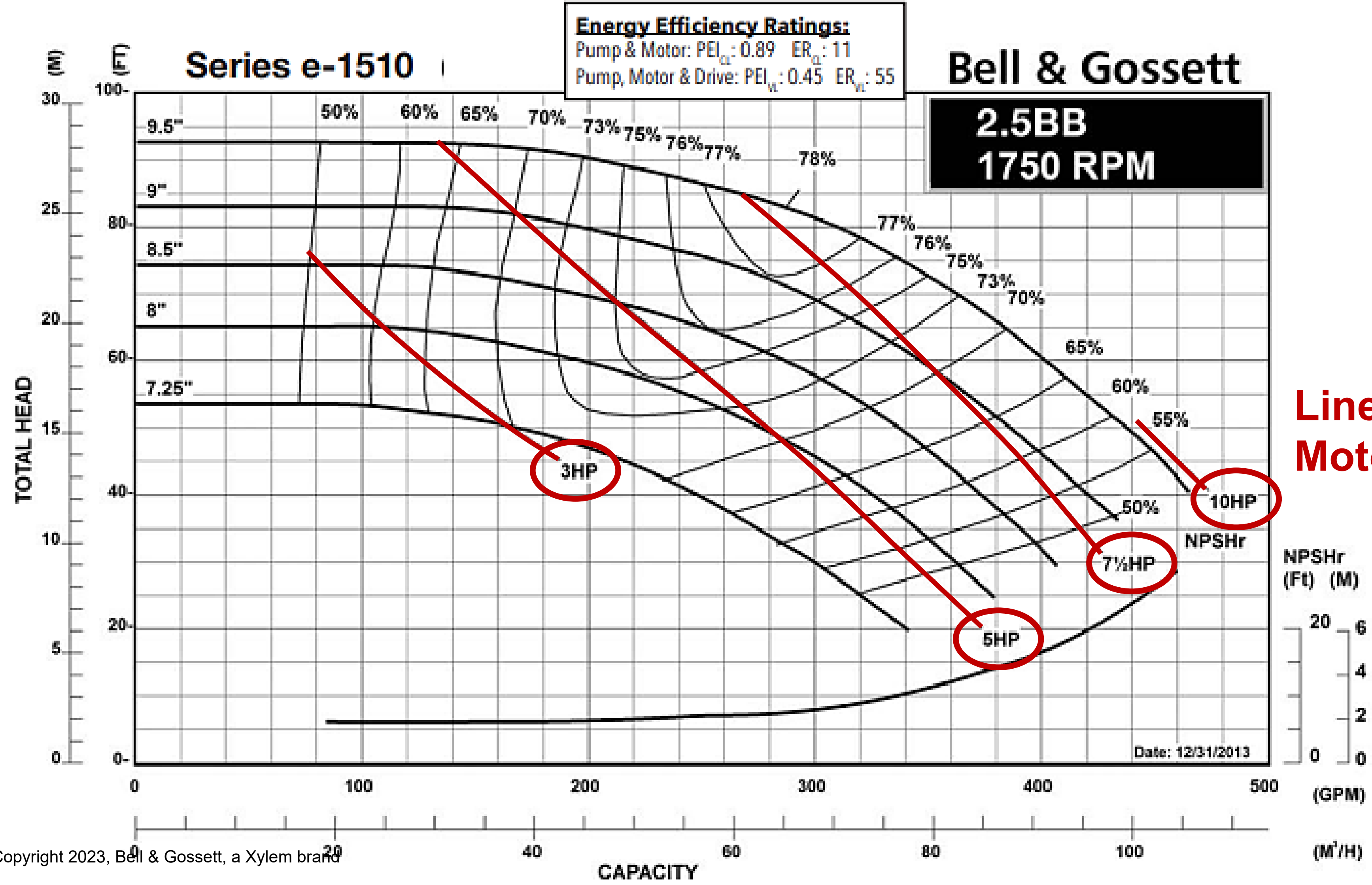

Lines of Constant Motor HP

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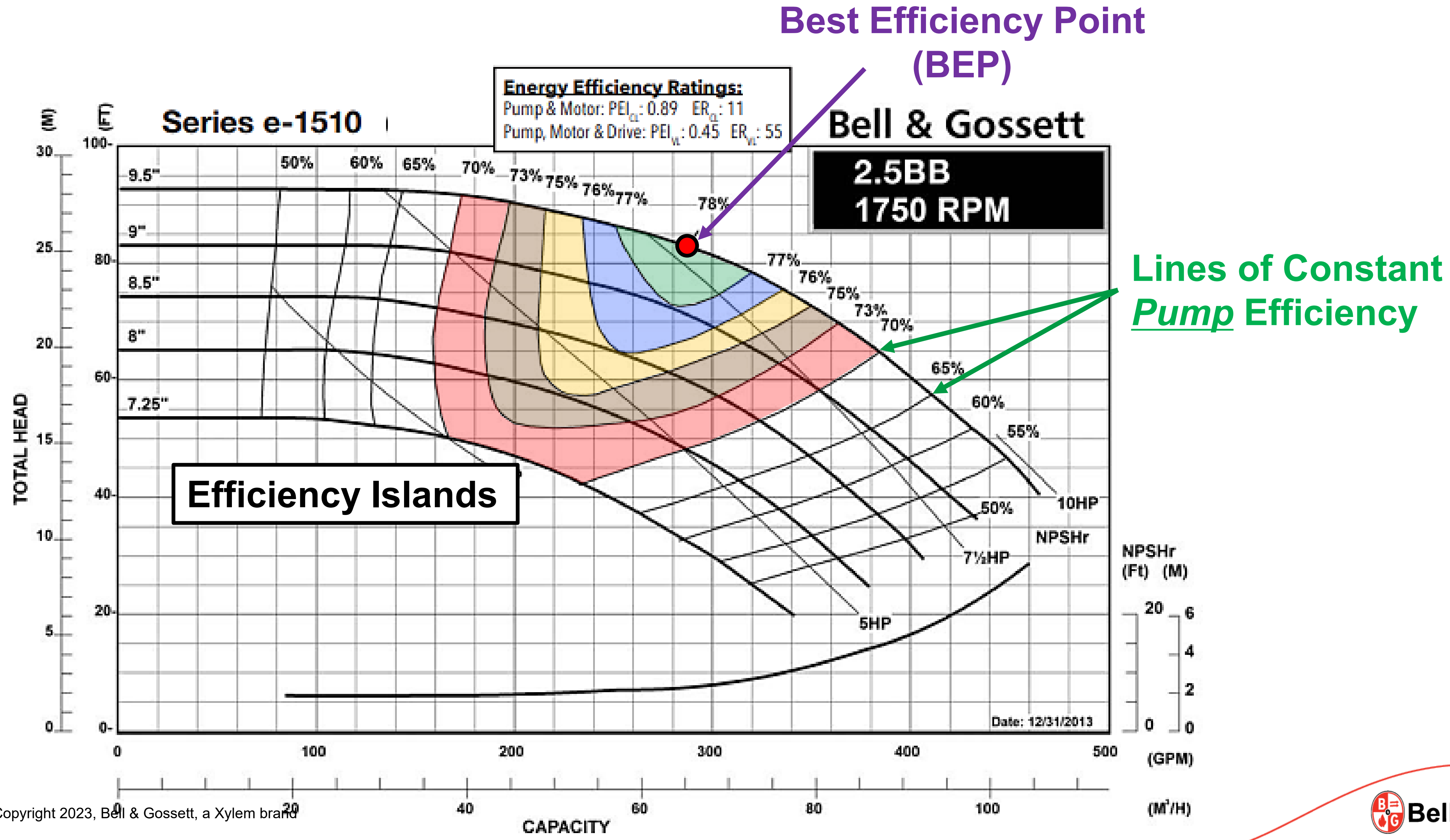

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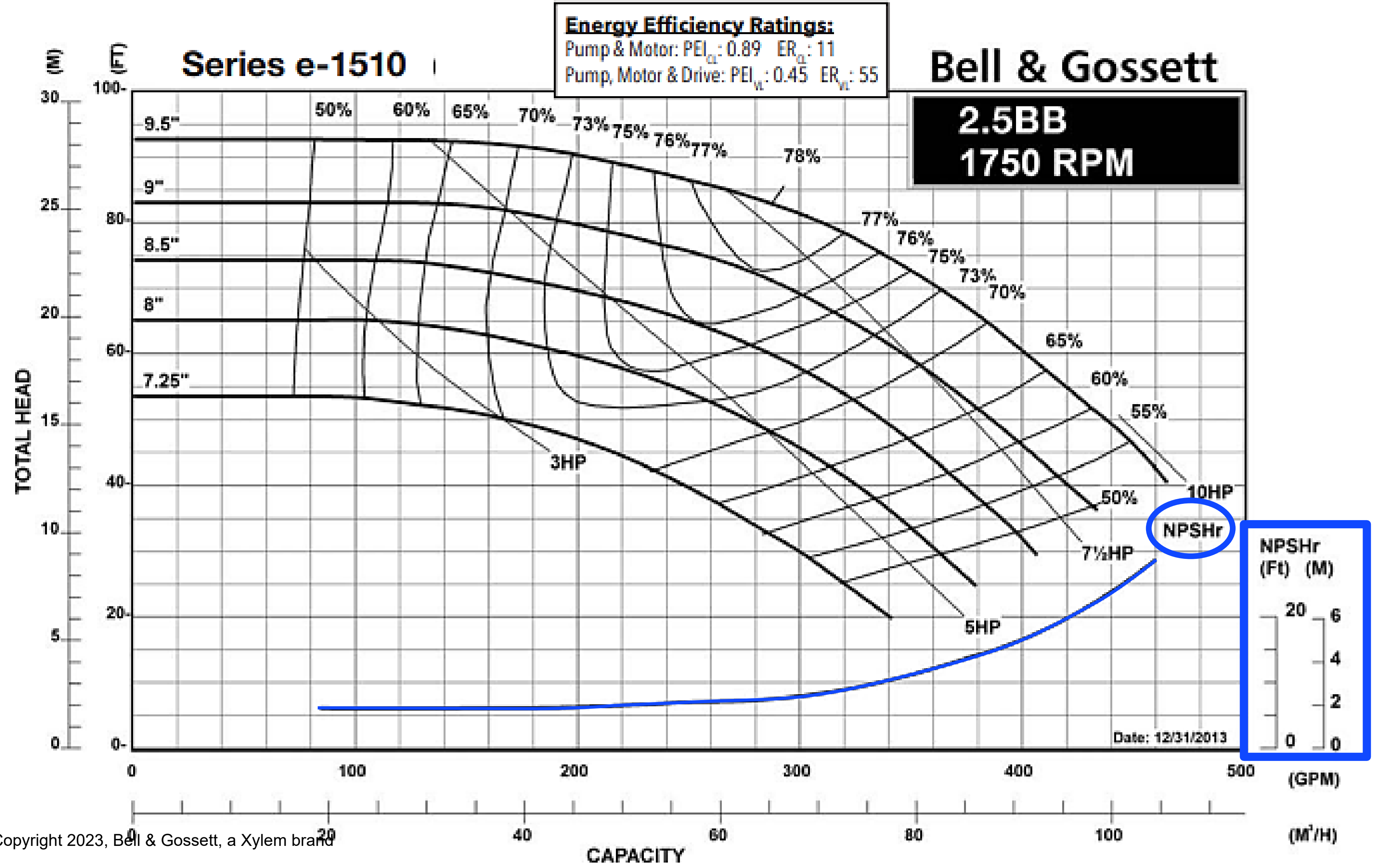

Net Positive Suction Head ***required***

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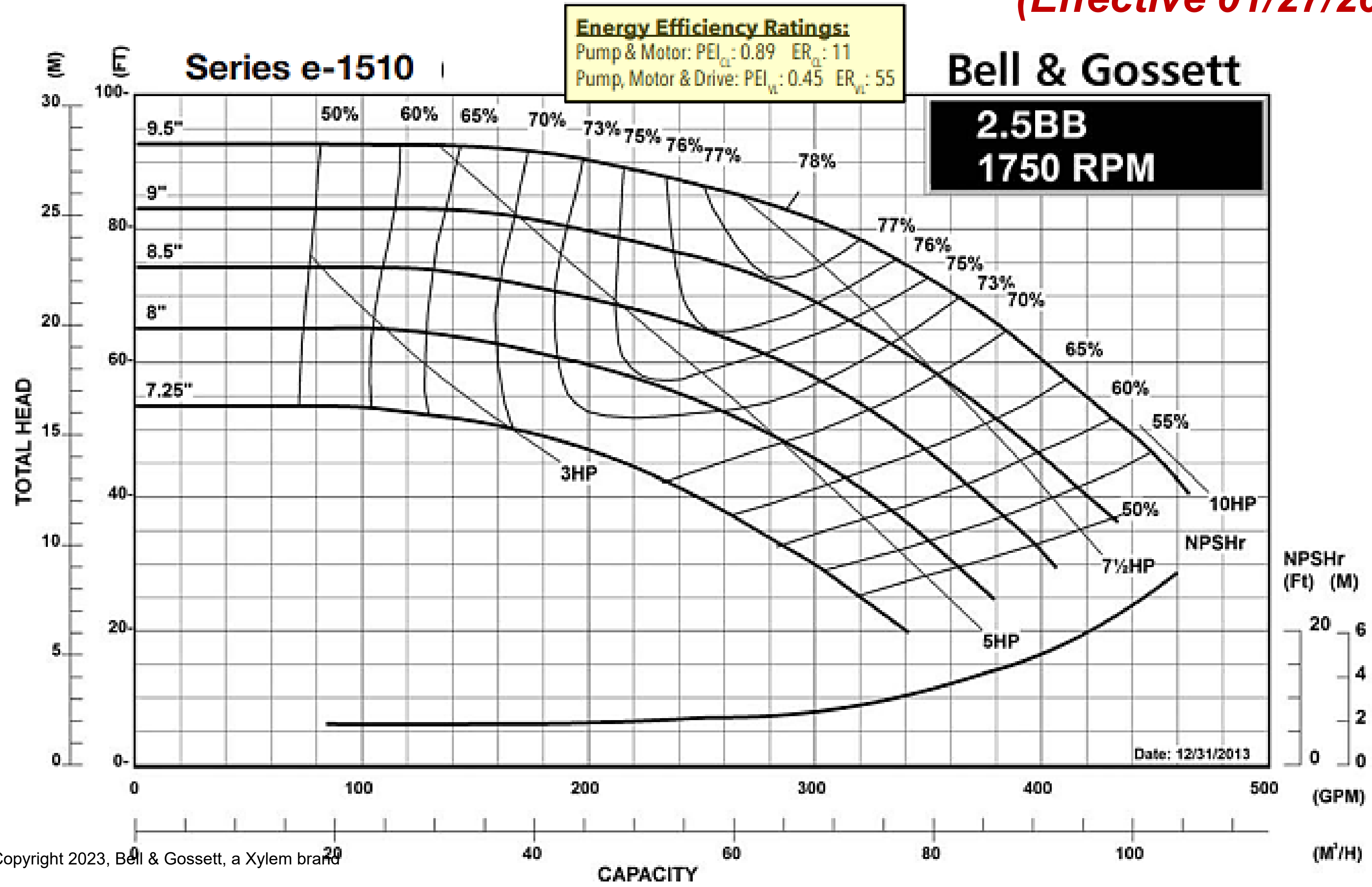
Department of Energy (DOE) Energy Conservation Standards *(Effective 01/27/20)*



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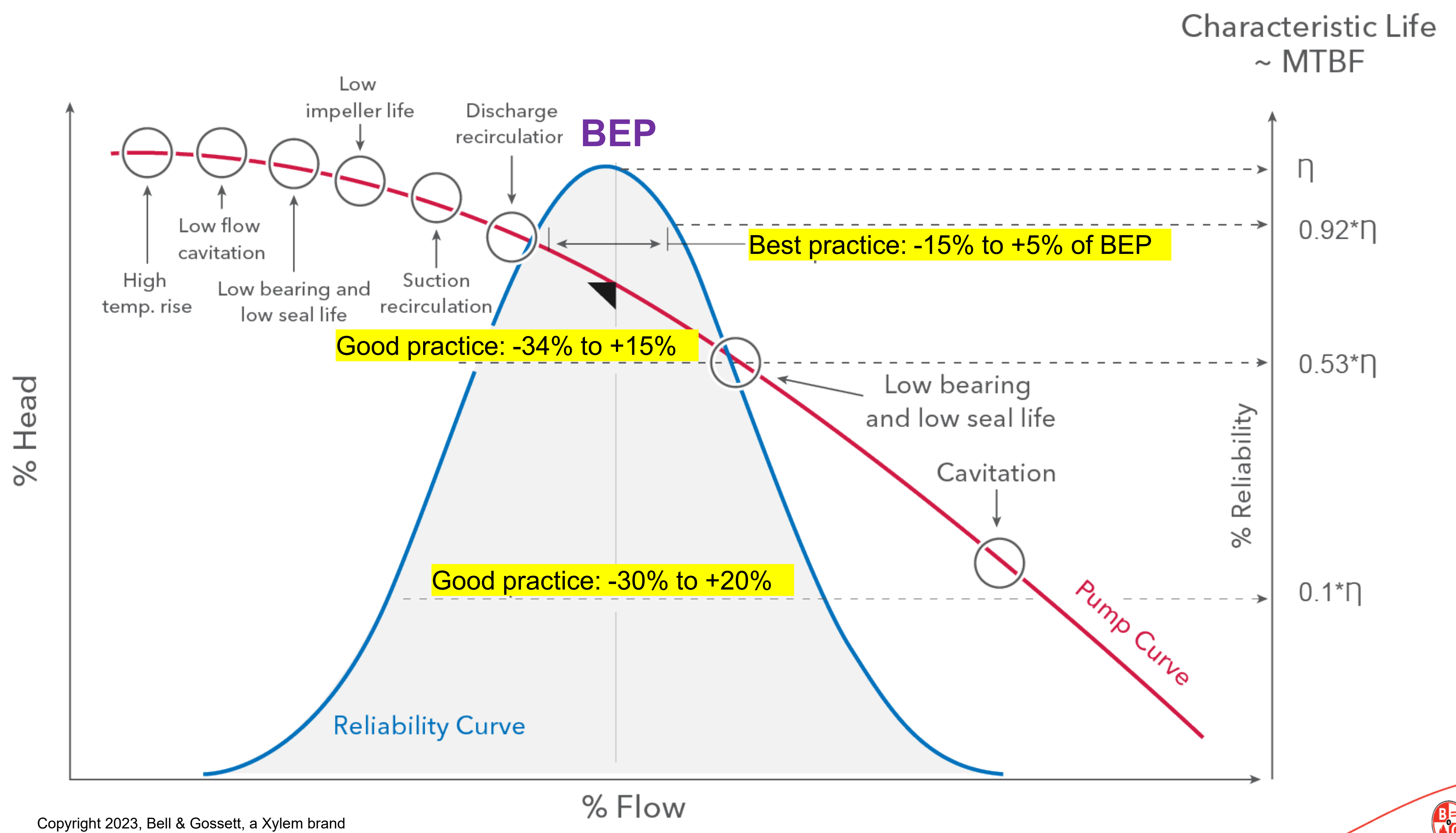

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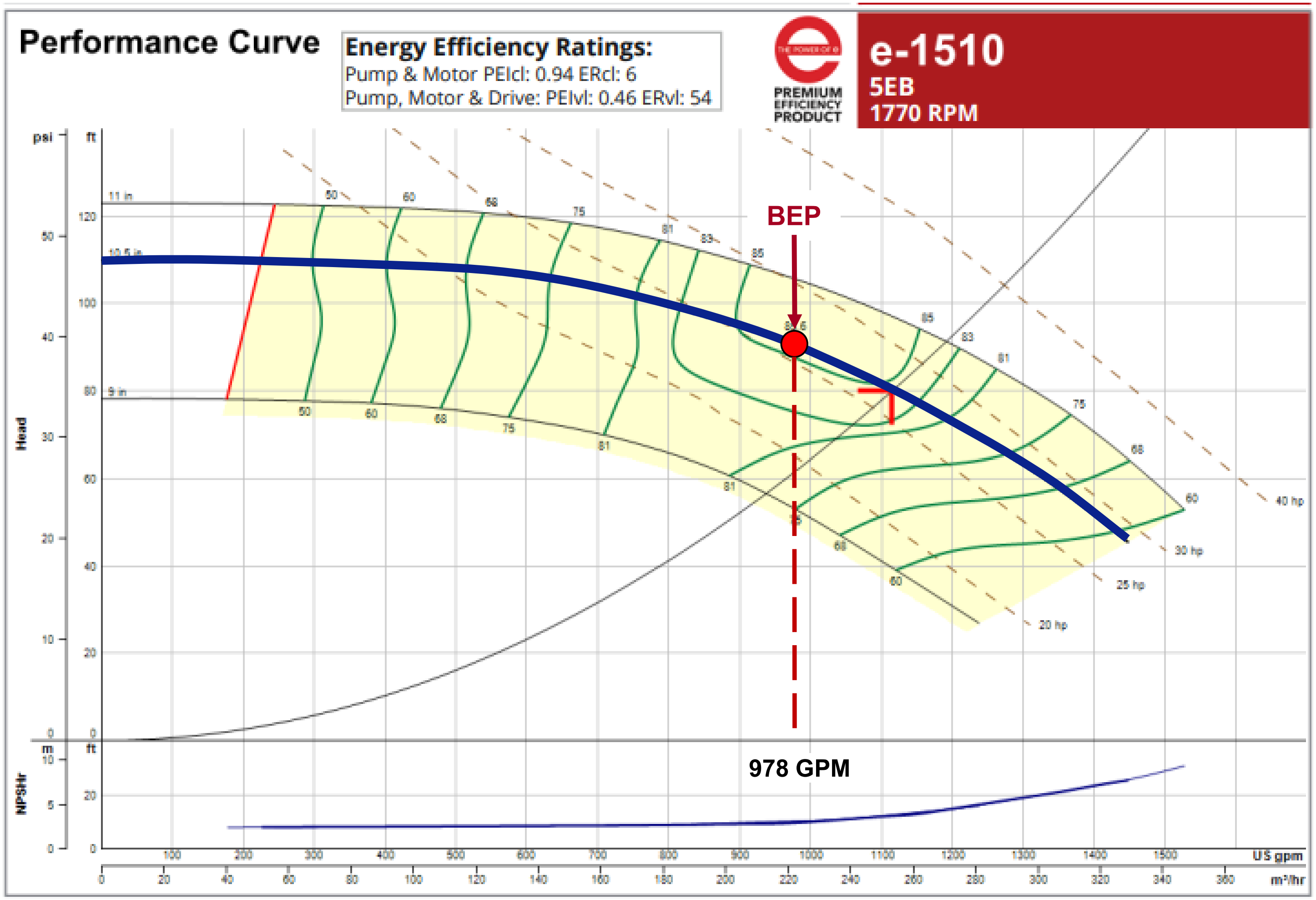
Defining the *Preferred* and *Acceptable* Operating Regions



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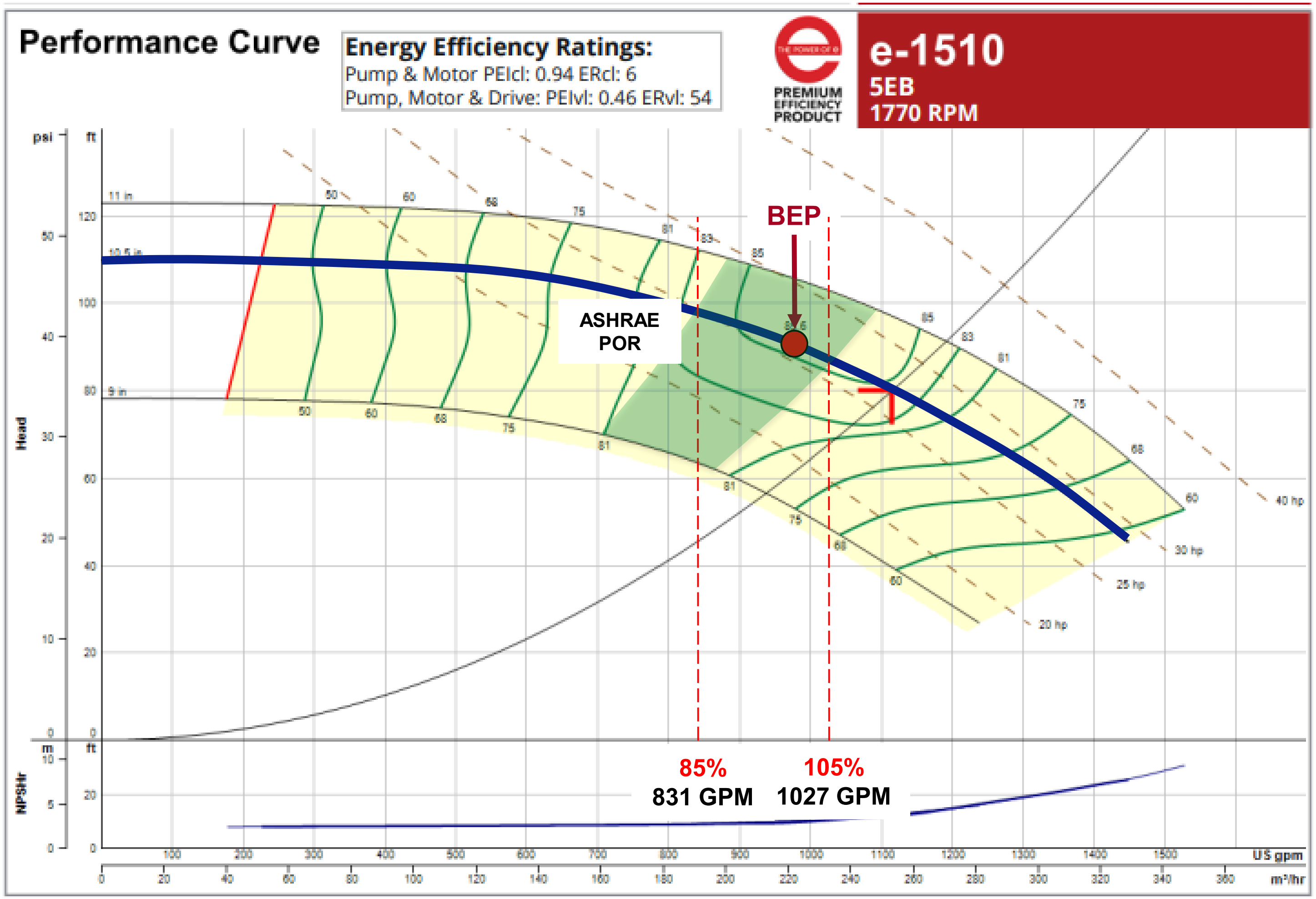
Defining the *Preferred* and *Acceptable* Operating Regions



- **ASHRAE 90.1** defines the Preferred Operating Region (POR) for a pump as **85%-105%** of the flowrate at BEP on the performance curve of installed impeller.

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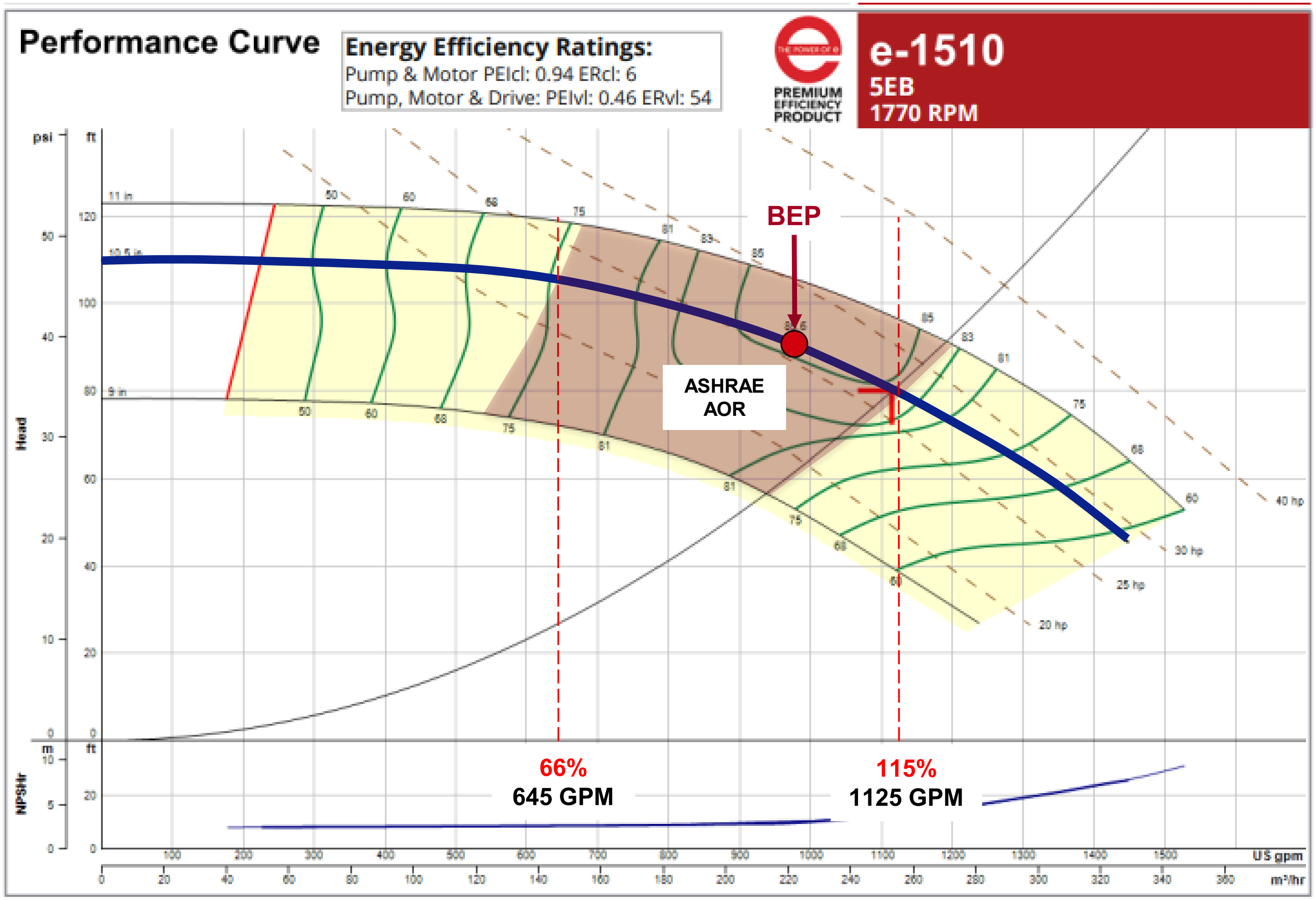
Defining the *Preferred* and *Acceptable* Operating Regions



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- **ASHRAE 90.1** defines the Acceptable Operating Region (AOR) for a pump as **66%-115%** of the flowrate at BEP on the performance curve of installed impeller.

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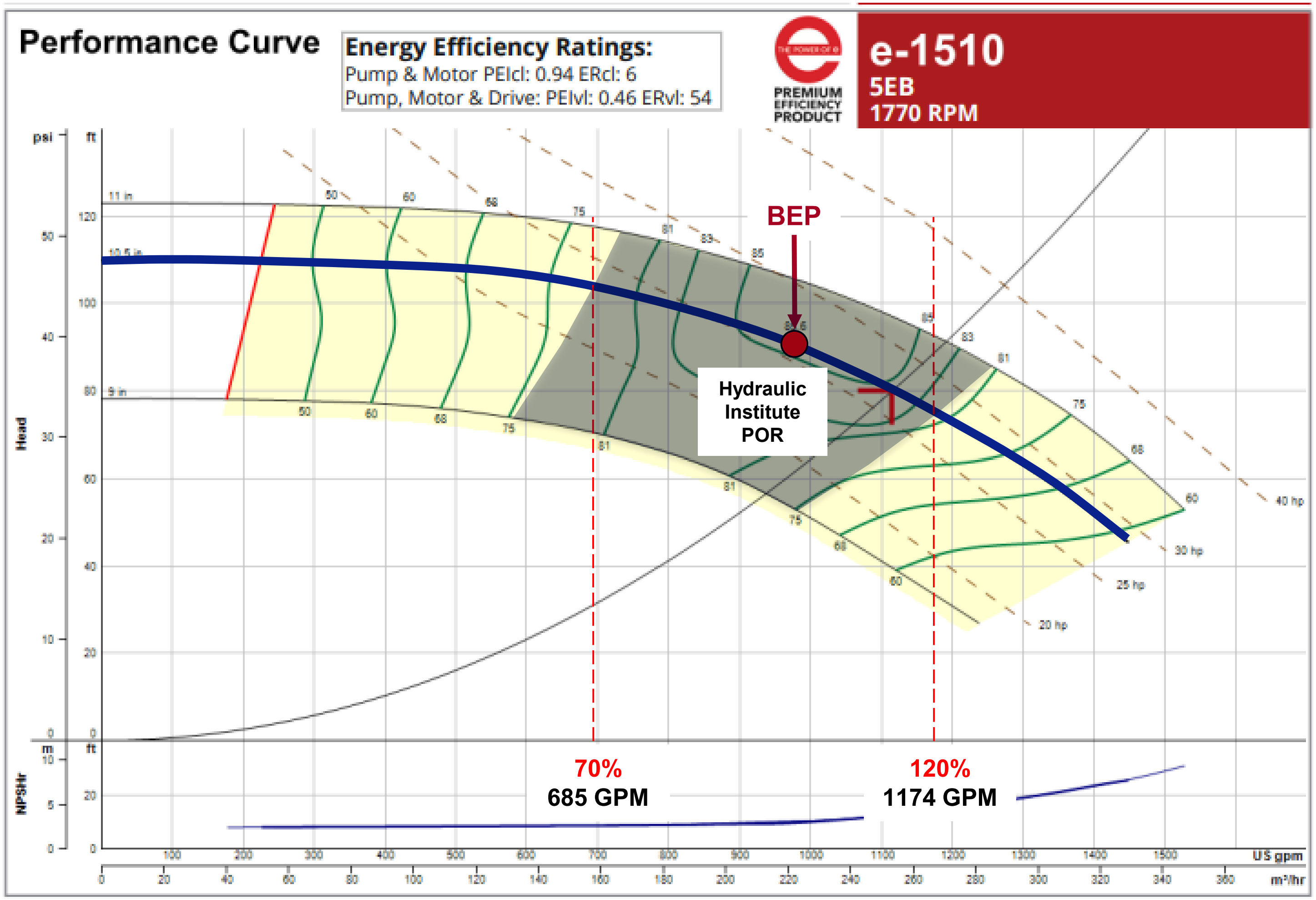
Defining the *Preferred* and *Acceptable* Operating Regions



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- **ASHRAE 90.1** defines the Acceptable Operating Region (AOR) for a pump as **66%-115%** of the flowrate at BEP on the performance curve of installed impeller.
- **Hydraulic Institute** defines the Preferred Operating Region (POR) for a pump as **70%-120%** of the flowrate at BEP on the performance curve of installed impeller.
- **Pump Manufacturers** can allow flowrates up to maximum of **85%** of pump curve capacity. This tends to align with the Hydraulic Institute definition of POR.

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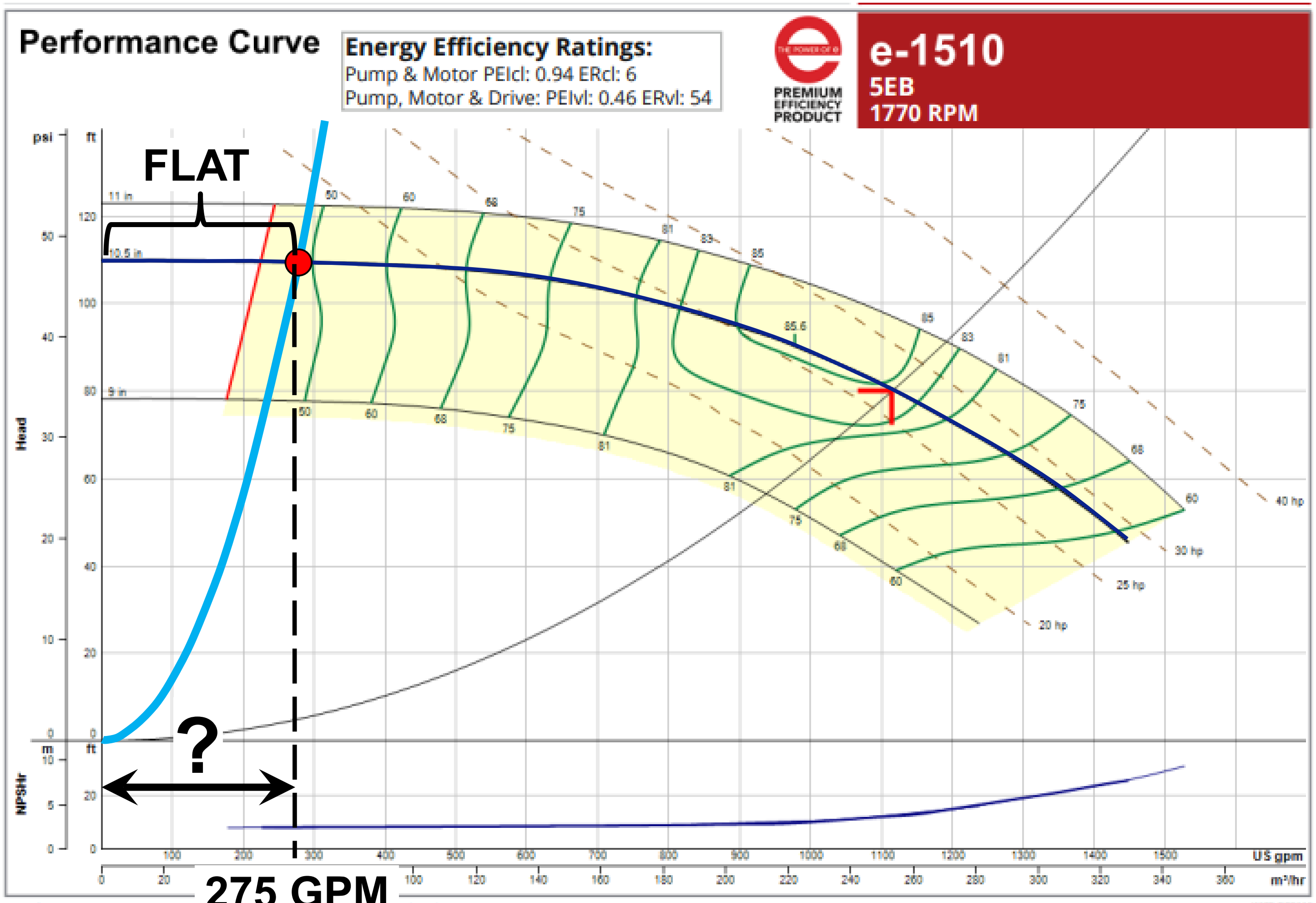
The Flat Portion of the Pump Curve: What's the Flow?

Avoid Continuous Operation in this area



- **Many possible flowrates at the same differential pump head**

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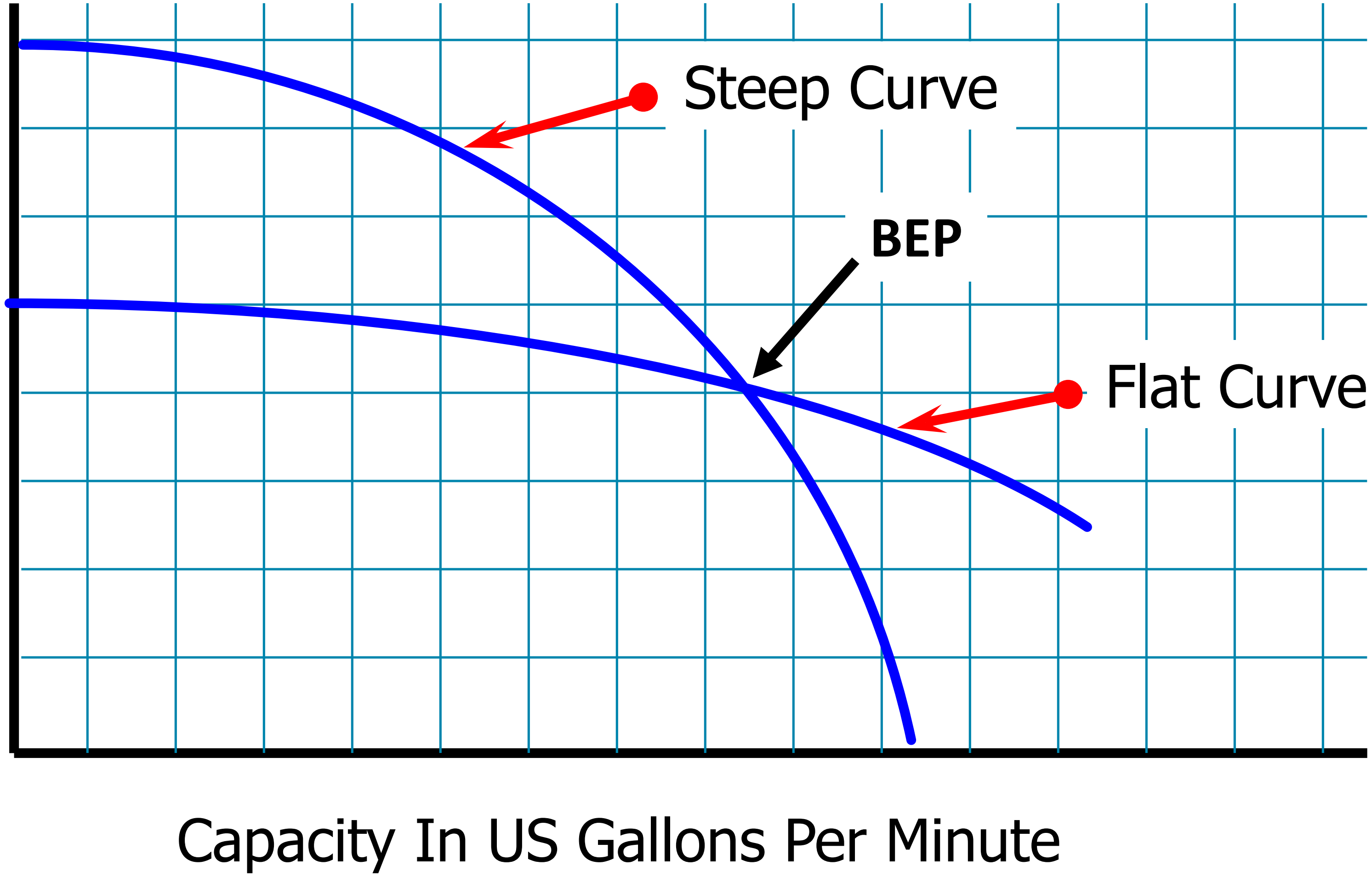
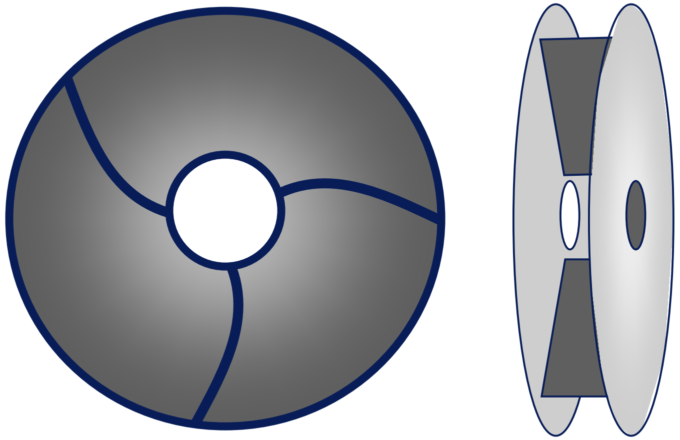
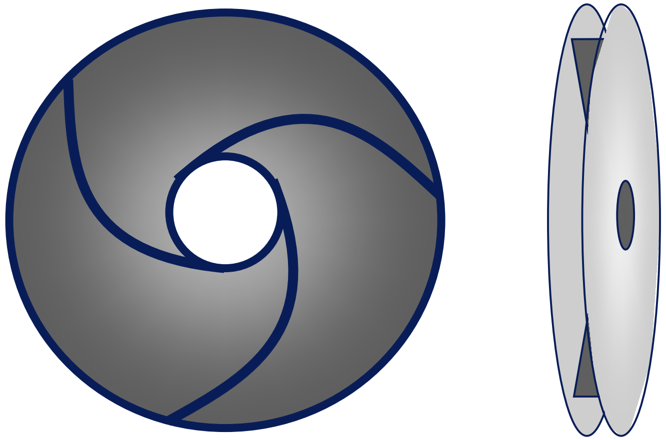


Flat Curve



Steep Curve



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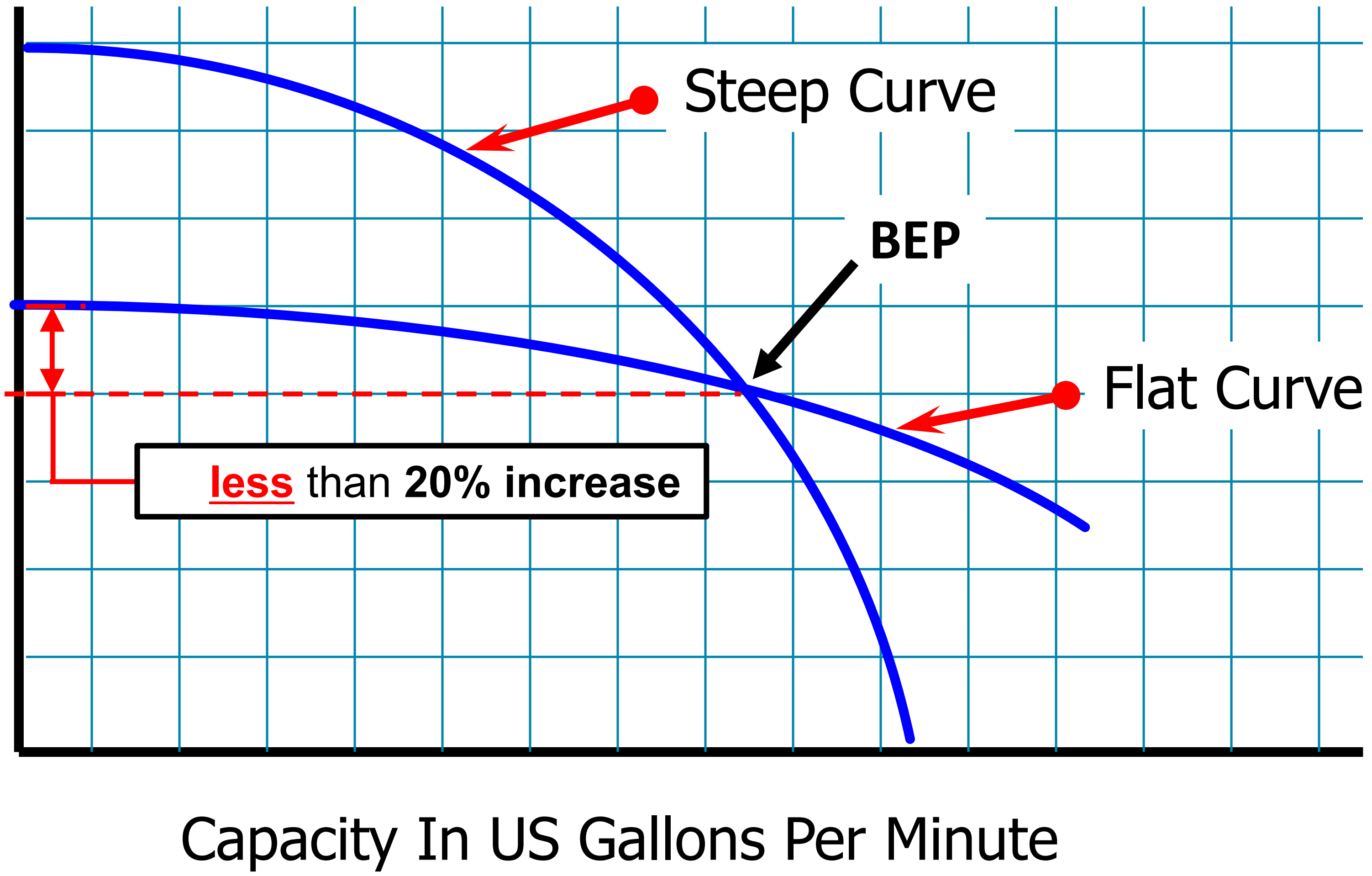

- **Flat Curve**

  Rise in Head from BEP to No Flow **less** than **20%**

- **Steep Curve**

  Rise in Head from BEP to No Flow **greater** than **20%**

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Pump delivering 30 Ft head @ design flow

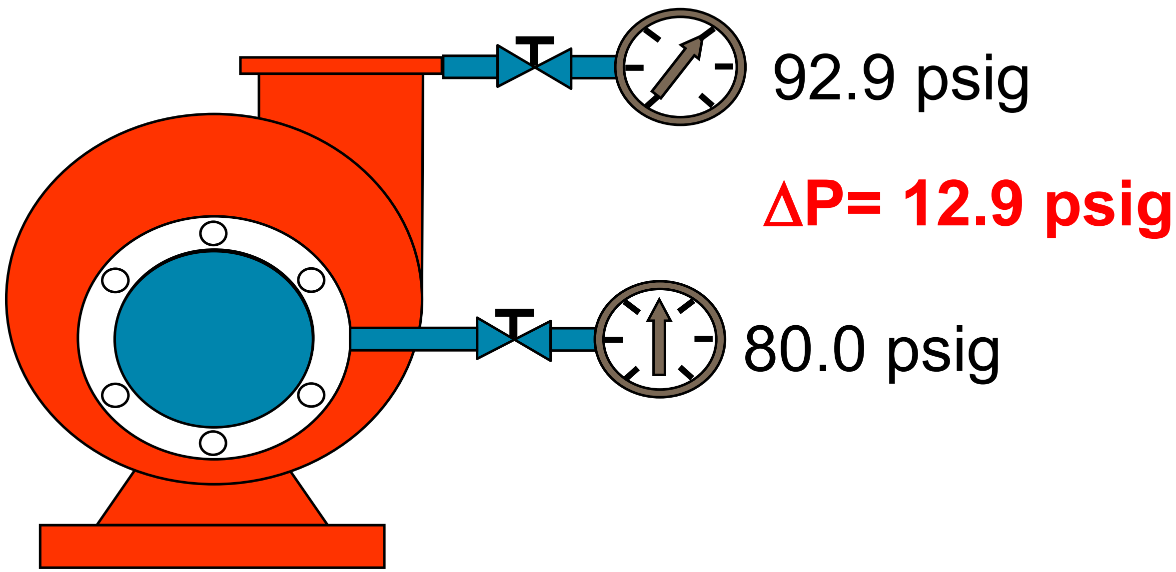
Density = 62.36 lbs/ft³

62.36 ÷ 144 = 0.43 psi/ft

144÷ 62.36 = 2.3 ft / psi

30 ft X .43psi/ft =12.9psi

12.9 psi X 2.3 ft/psi = 30 ft



Water @ 60° F

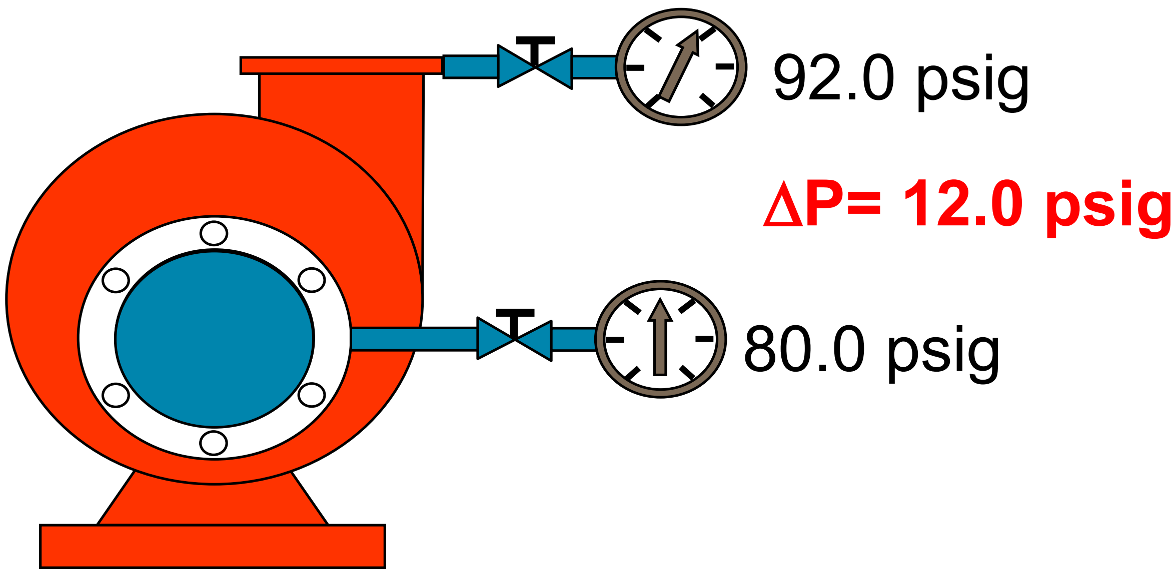
Density = 57.29 lbs/ft³

57.29 ÷ 144 = 0.40 psi/ft

144÷ 57.29 = 2.5 ft / psi

30 ft X .40psi/ft =12.0psi

12.0 psi X 2.5 ft/psi = 30 ft



Water @ 300° F

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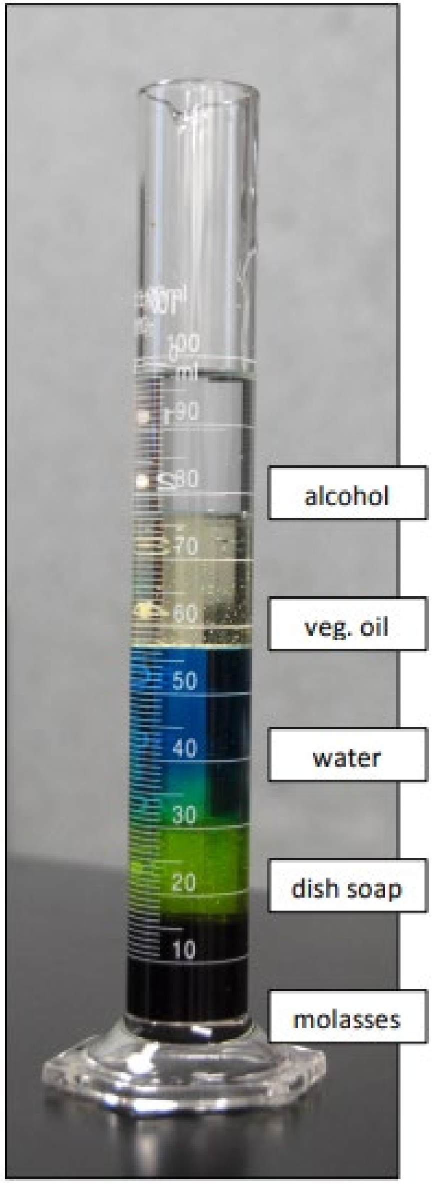
Density (ρ): How heavy a fluid is for the amount measured **(lbs./ft³)**

Specific Gravity (SG): Ratio of a fluid density compared to density of water, at a specific temperature and pressure

$$SG = \frac{\rho \text{ of Fluid}}{\rho \text{ of Water}}$$

Pump Head (ft-~~lb~~/~~lb~~)

$$\text{Pump Head (ft)} = \text{PSI} \times \frac{2.31}{SG}$$



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Let's Solve Water

Hydronic System Flow Determination

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______ Carrying?

$$Q = m\,C_p\,(t_s - t_r)$$

$$\text{Btu/Hr} = \frac{\text{lb}}{\text{Hr}} \cdot \frac{1\ \text{Btu}}{\text{lb}\ {}^{\circ}\text{F}} \cdot {}^{\circ}\text{F}$$

*Mass Flowrate

$$\text{Btu/Hr} = 500 \times \text{GPM} \times \Delta T$$

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_____ (Based on ΔT)

$$Btu/Hr = 500 \times GPM \times \Delta T$$

$$Btu/Hr = 500 \times GPM \times (20°F)$$

$$Btu/Hr = 10{,}000 \times GPM$$

$$\frac{Btu/Hr}{10{,}000} = GPM$$

$$\frac{Btu/Hr}{12{,}500} = GPM$$

(25°F)

$$\frac{Btu/Hr}{15{,}000} = GPM$$

(30°F)

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Water C_p = *1.0 Btu/lb °F* ***(@ 45 °F)***

Water C_p = *1.0 Btu/lb °F* ***(@ 68 °F)*** **Std. per* ***HI 3.6.3.30***

Water C_p = *0.98 Btu/lb °F* ***(@ 160 °F)***

In comparison:

40% P.G. C_p = *0.85 Btu/lb °F* ***(@ 45 °F)***

40% P.G. C_p = *0.91 Btu/lb °F* ***(@ 160 °F)***

40% E.G. C_p = *0.82 Btu/lb °F* ***(@ 45 °F)***

40% E.G. C_p = *0.87 Btu/lb °F* ***(@ 160 °F)***

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A new small commercial office building heating load calculation determines **650,000 Btu/hr** is required to maintain occupant comfort. A hydronic system has been chosen using a supply **water** temperature of **160°F** and an expected temperature drop of **20°F** across the terminal units. A single circulating pump will be used. Determine the required system flow rate.

Answer: $$\frac{650,000}{10,000} = 65\ \text{GPM}$$

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A new small commercial office building heating load calculation determines **650,000 Btu/hr** is required to maintain occupant comfort. A hydronic system has been chosen using a supply **water** temperature of **160°F** and an expected temperature drop of **20°F** across the terminal units. A single circulating pump will be used. Determine the required system flow rate.

Answer: $$\frac{650,000}{10,000} = 65 \text{ GPM}$$

What if we used 40% Ethylene Glycol instead of water?

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Water $\rho = 62.42\ lb/ft^3$ ***(@ 45 °F)***

Water $\rho = 62.31\ lb/ft^3$ ***(@ 68 °F)*** **Std. per* ***HI 3.6.3.30***

Water $\rho = 61.00\ lb/ft^3$ ***(@ 160 °F)***

40% E.G. $\rho = 66.51\ lb/ft^3$ ***(@ 45 °F)***

40% E.G. $\rho = 65.28\ lb/ft^3$ ***(@ 160 °F)***

REMINDER!!

$$SG = \frac{\rho \text{ of Fluid}}{\rho \text{ of Water}}$$

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Answer: $$\frac{650,000}{10,000} = 65 \text{ GPM}$$

What if we used 40% Ethylene Glycol instead of water?

Density of Water @ 160°F = 61.00 Lbs/ft³

Density of 40% EG @ 160°F = 65.28 Lbs/ft³

$$SG = \frac{\rho \text{ of Fluid}}{\rho \text{ of Water}} = \frac{65.28}{61.00} = 1.07$$

Cp of 40% EG @ 160°F = **0.87** Btu/lb °F

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A new small commercial office building heating load calculation determines **650,000 Btu/hr** is required to maintain occupant comfort. A hydronic system has been chosen using a supply **water** temperature of **160°F** and an expected temperature drop of **20°F** across the terminal units. A single circulating pump will be used. Determine the required system flow rate.

Answer: $$\frac{650{,}000}{10{,}000} = 65 \text{ GPM}$$

What if we used 40% Ethylene Glycol instead of water?

Answer: $$\frac{650{,}000}{\underbrace{(10{,}000)(0.87)(1.07)}_{9309}} = 70 \text{ GPM}$$

Density of Water @ 160°F = 61.00 Lbs/ft³

Density of 40% EG @ 160°F = 65.28 Lbs/ft³

$$SG = \frac{\rho \text{ of Fluid}}{\rho \text{ of Water}} = \frac{65.28}{61.00} = 1.07$$

Cp of 40% EG @ 160°F = **0.87** Btu/lb °F

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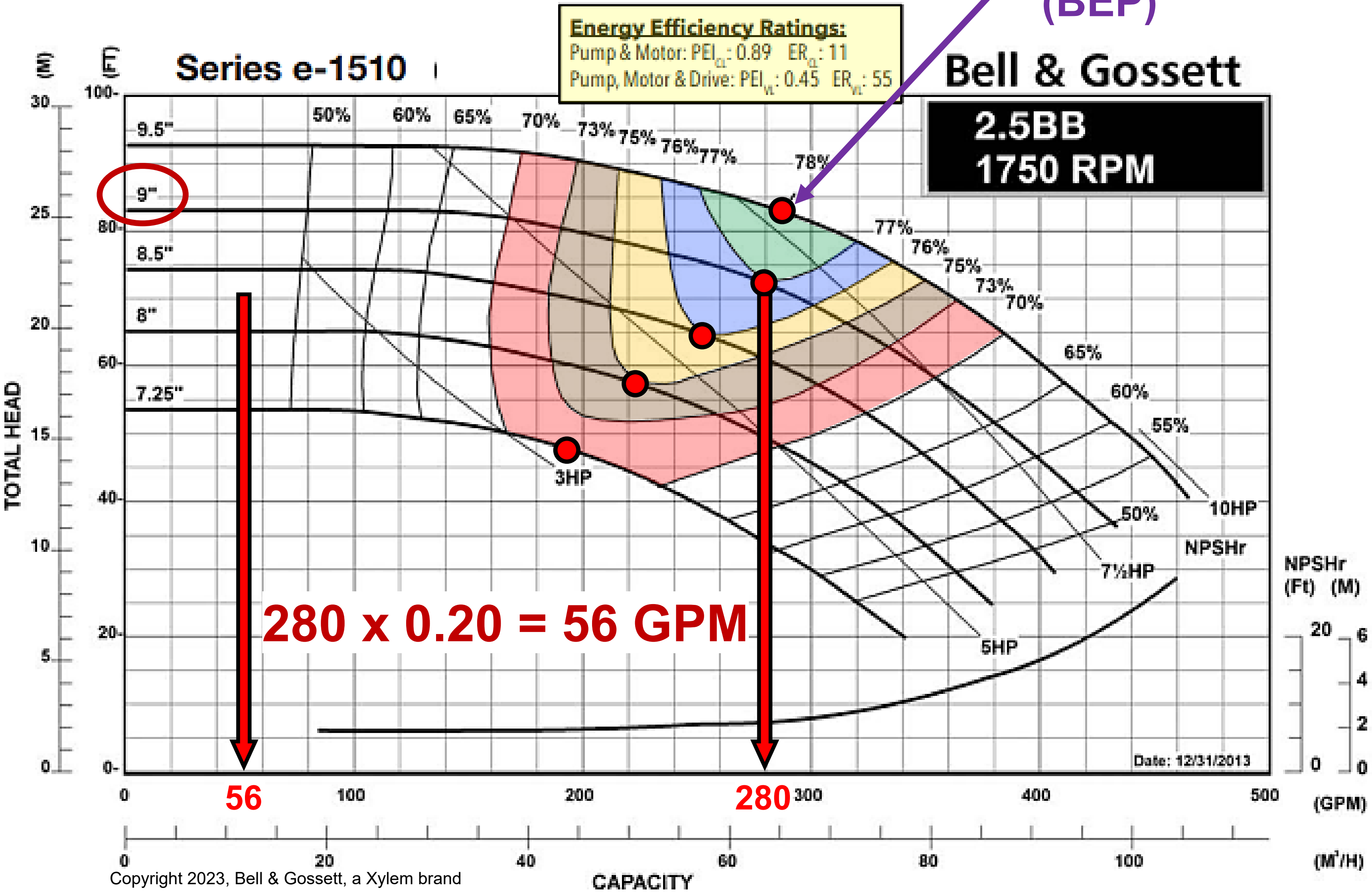
Best Efficiency Point (BEP)



Pump Impeller Type	RULE	% of BEP Flow
Double-Suction	All Split-Case Pumps	35%
Single-Suction	Flow@ BEP > 2500 GPM	25%
Single-Suction	Flow@ BEP > 800 GPM	23%
Single-Suction	Flow@ BEP > 100 GPM	20%
Single-Suction	Flow@ BEP > 10 GPM	15%
Single-Suction	Flow@ BEP > 1 GPM	10%

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Pump Head – Bernoulli's Theorem

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Pressure: A force applied against a unit of Area **(lb/in²)**

Fluid Head: Total mechanical energy contained in a pound of fluid **(ft-lb)**

Pump Head: Energy *(work)* added by a Pump as fluid passes through it **(ft-lb/lb)**

In a Hydronic System, with liquid flowing, three (3) factors influence how mechanical energy *(head)* in the fluid is *transformed*:

- The Force of static **pressure**
- The Head due to an **elevation** change above a reference point *(Potential Energy)*
- The Head due to the **velocity** of the liquid *(Kinetic Energy)*

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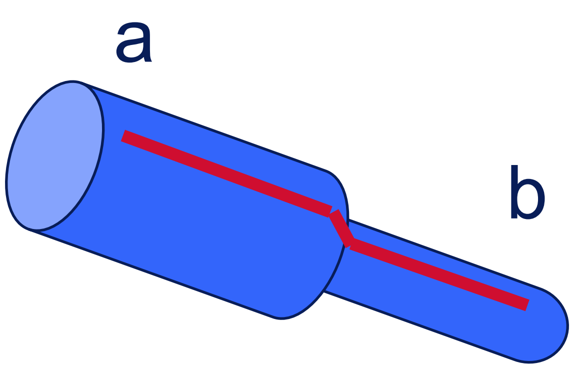

$$\frac{P_a}{W} + Z_a + \left(\frac{V_a^2}{2g}\right) = \frac{P_b}{W} + Z_b + \left(\frac{V_b^2}{2g}\right)$$

Total Head of a fluid at point **"a"** is equal to the total head of the fluid at point **"b"**

provided:

- There is **no head lost** due to friction or work
- There is **no head gained** due to the application of work

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Bernoulli's Theorem as applied for Pumped Hydronic Systems

Closed Loop

Energy (Fluid Head) lost due to friction

$$\frac{P_a}{W} + Z_a + \frac{V_a^2}{2g} + E_p = \frac{P_b}{W} + Z_b + \frac{V_b^2}{2g} + h_f$$

Total Pump Head $E_p = \left(\frac{P_b}{W} - \frac{P_a}{W}\right) + \left(Z_b - Z_a\right) + \left(\frac{V_b^2}{2g} - \frac{V_a^2}{2g}\right) + h_f$

<u>For "Closed Loop" Systems Only</u>

$$E_p = h_f$$

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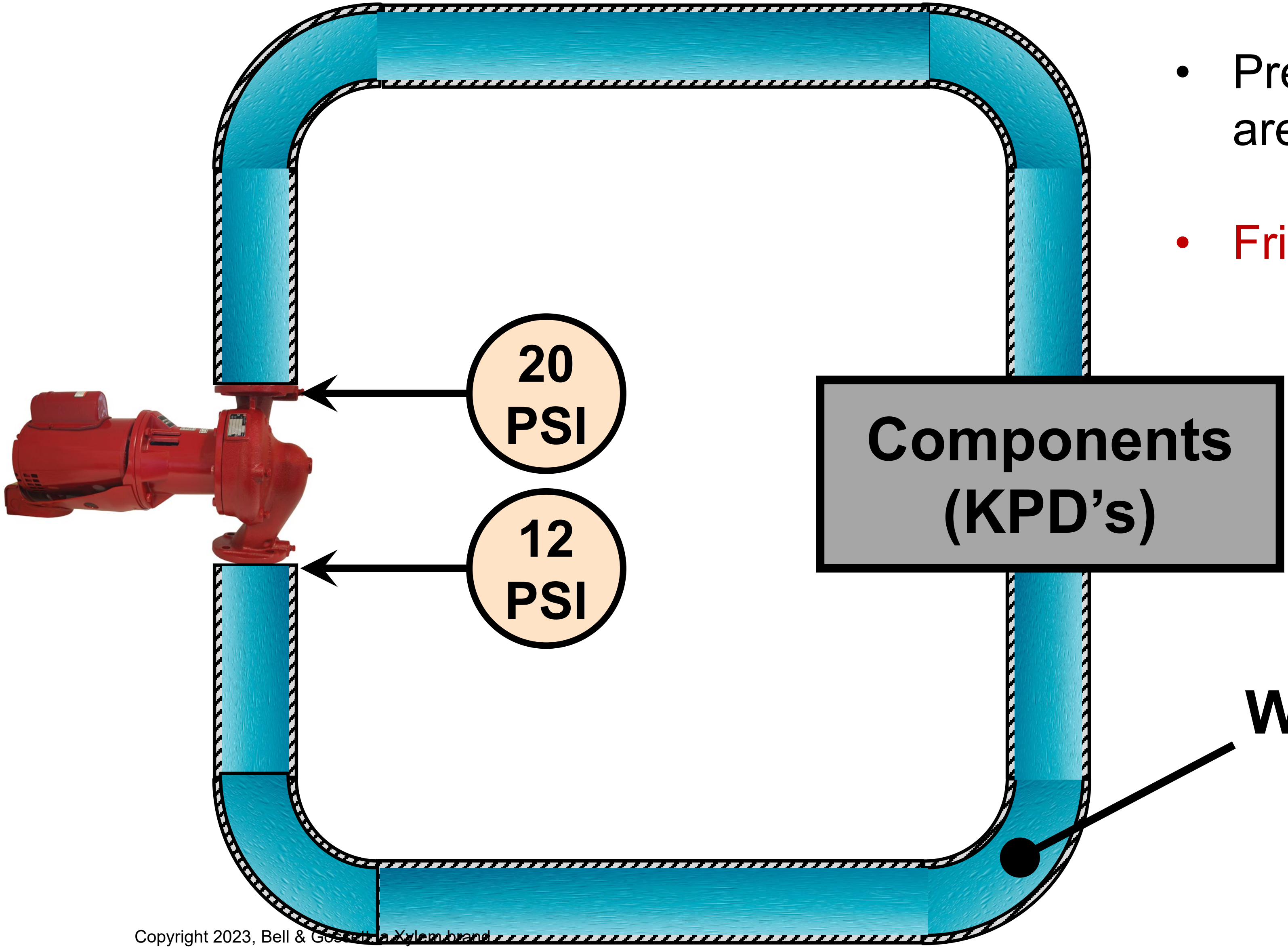

- Pressure, Velocity, and Elevation are equalized at all points
- Friction Head Loss = Pump Head

8 Psig x 2.31 = 18.5 ft

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Determining Fluid Head losses due to friction

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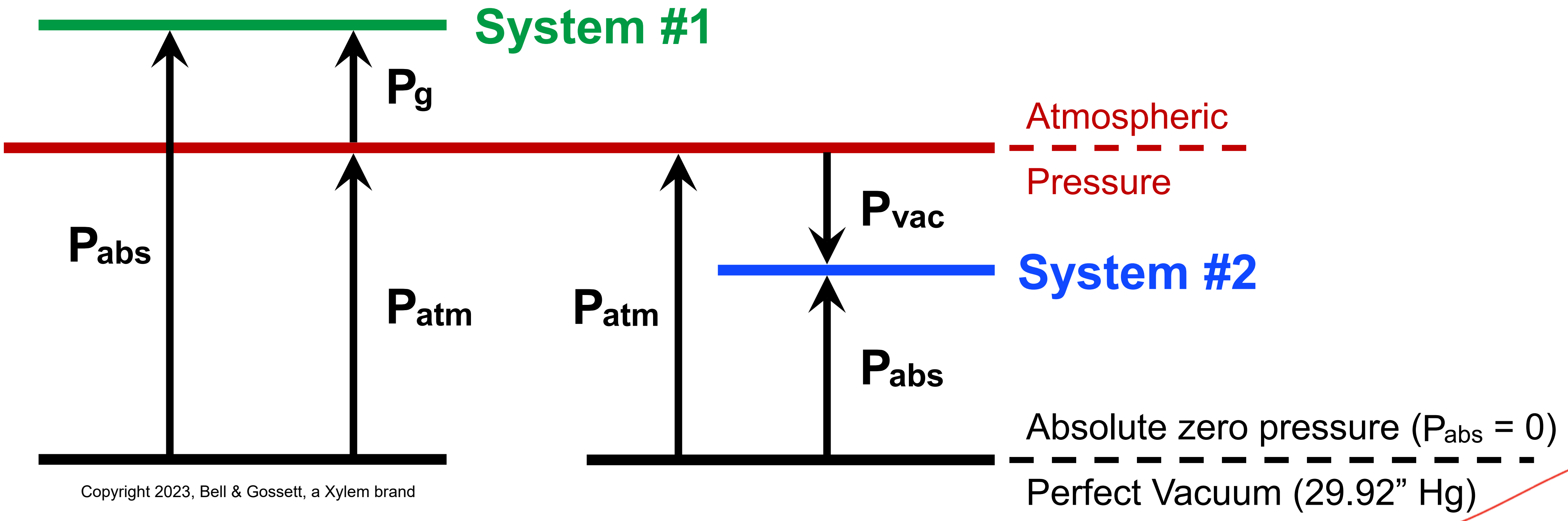
Absolute Pressure (P_{abs}): Measured with reference to Perfect Vacuum

Atmospheric Pressure (P_{atm}): Average air pressure at given elevation

Gauge Pressure (P_g): Measured **above** Atmospheric Pressure

Vacuum Pressure (P_{vac}): Measured **below** Atmospheric Pressure



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Absolute Pressure (P_{abs}): Measured with reference to Perfect Vacuum

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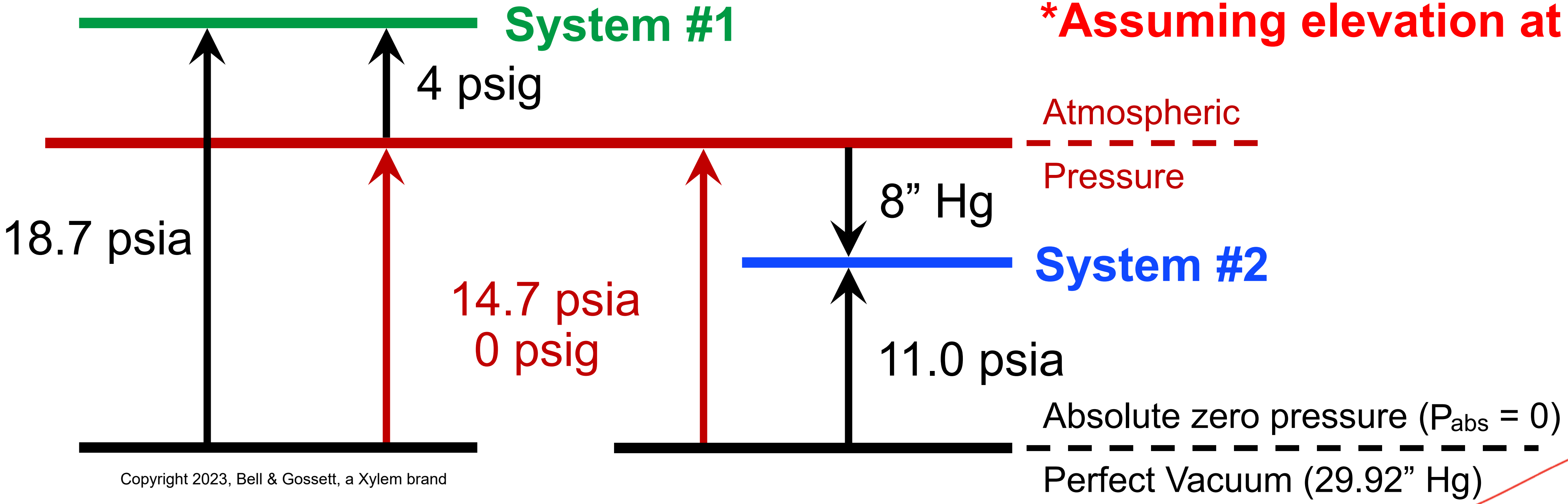
Gauge Pressure (P_g): Measured **above** Atmospheric Pressure

Vacuum Pressure (P_{vac}): Measured **below** Atmospheric Pressure

***Assuming elevation at Sea Level**



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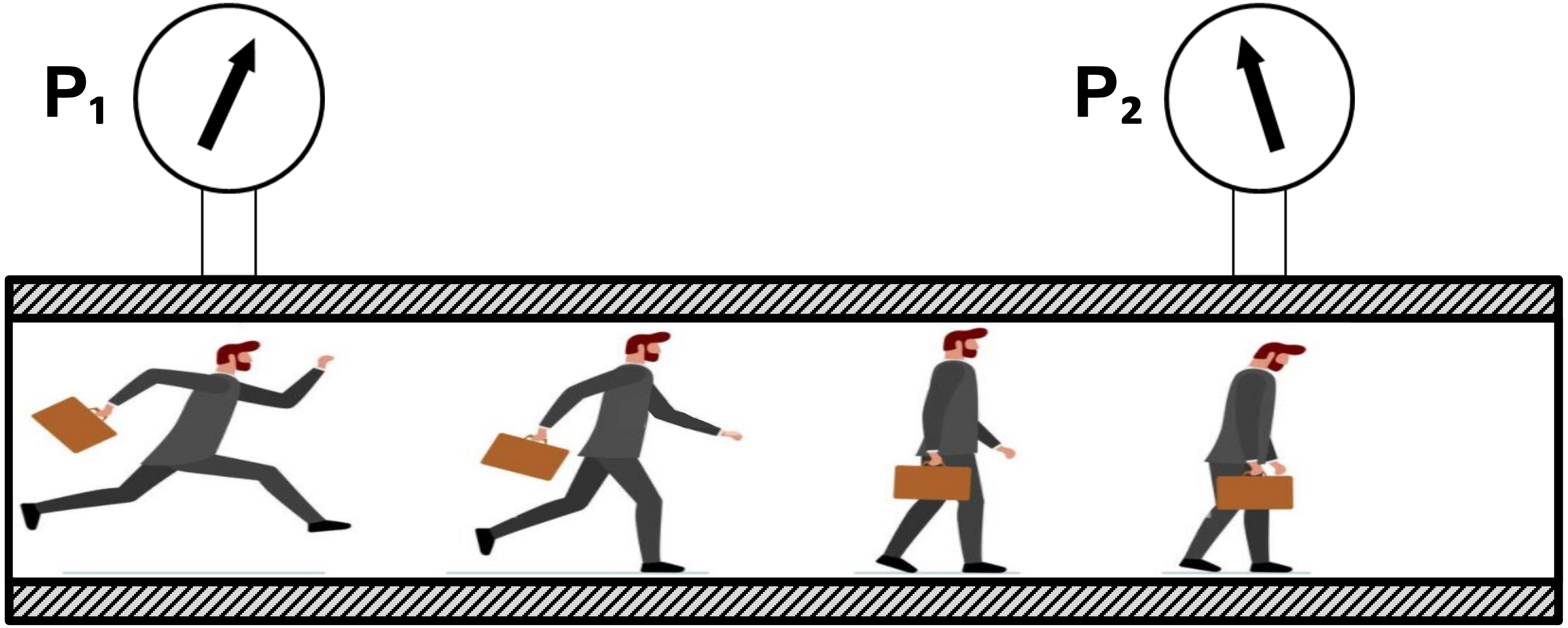
“Head” or “Friction” Loss



Pressure Drop is evidence energy (Pump Head) is consumed (used up) to move a fluid

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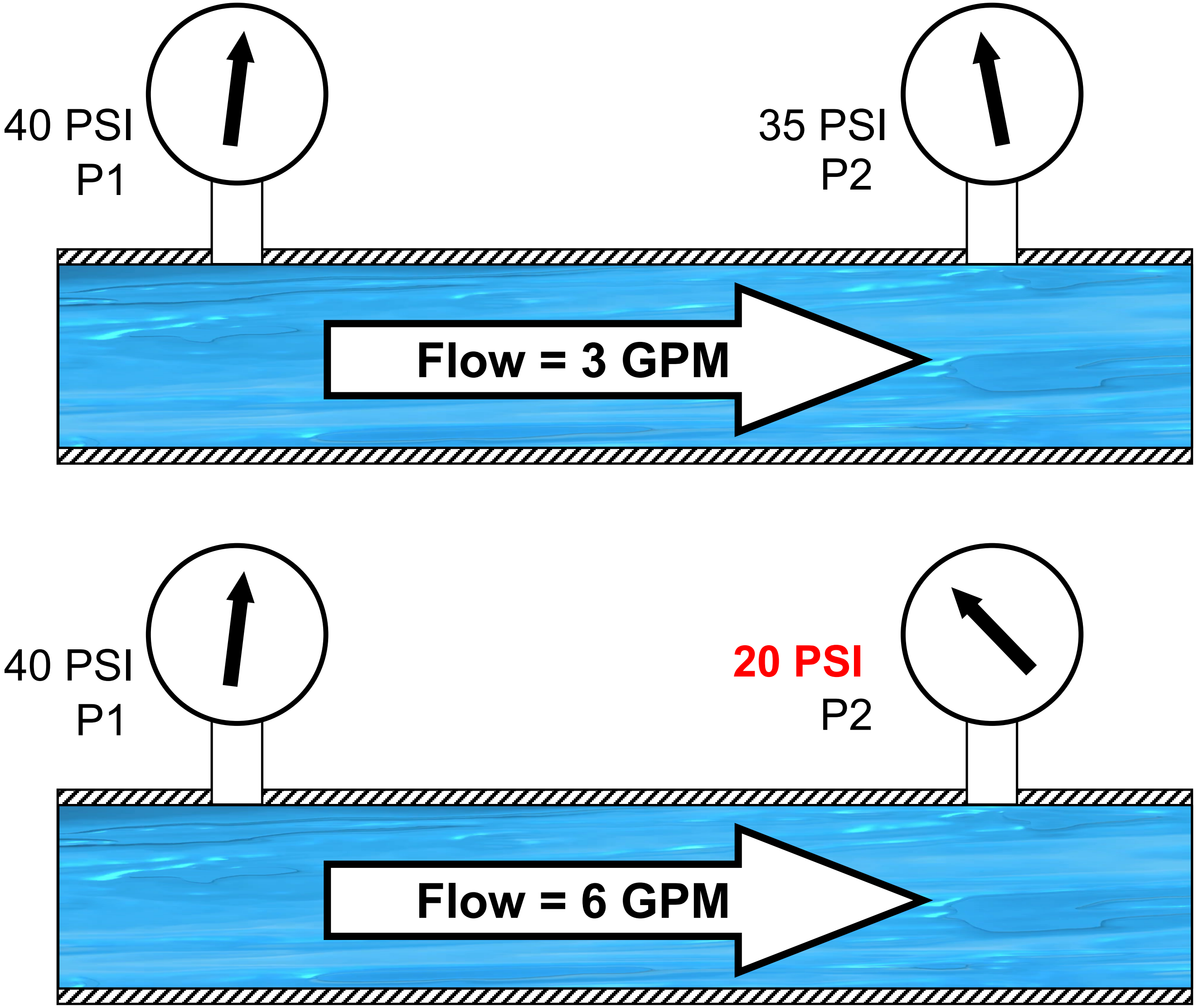
$$\left(\frac{Q_2}{Q_1}\right)^2 = \left(\frac{h_2}{h_1}\right)$$

- Q_1 = Known (design) Flow
- Q_2 = Final Flow
- h_1 = Known (design) Head
- h_2 = Final Head

$$h_2 = \left[\frac{Q_2}{Q_1}\right]^2 x\ h_1$$

$$20\ \text{PSI} = \left[\frac{6}{3}\right]^2 x\ 5$$



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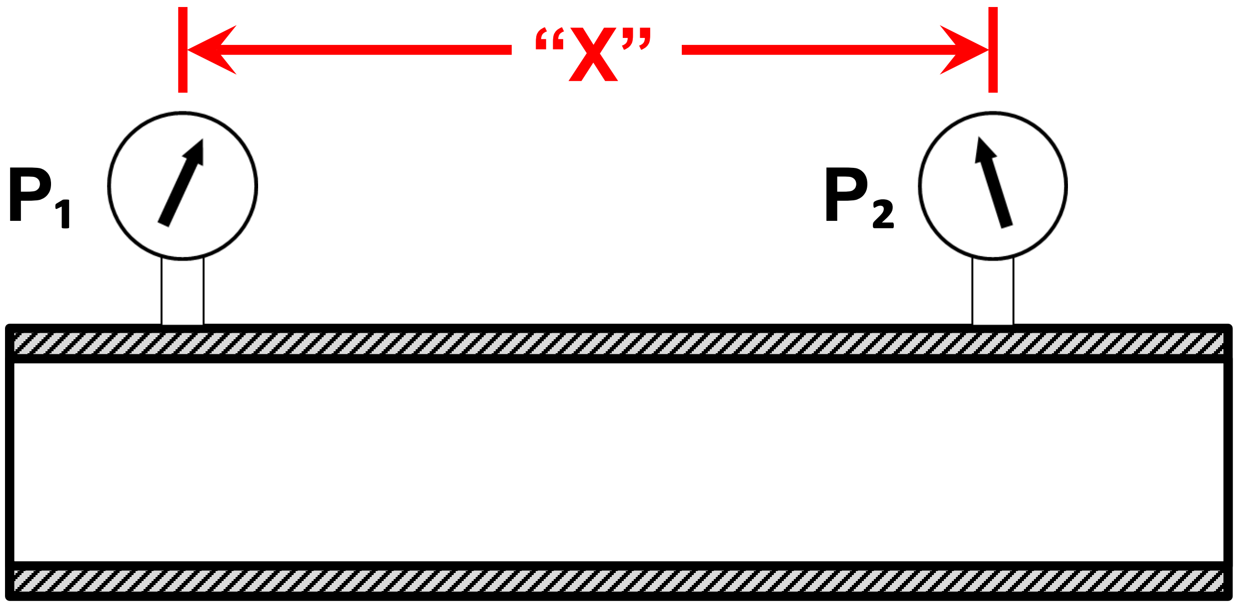

<u>Rate</u> of Head loss: Measured per **100ft** of <u>equivalent</u> straight pipe length *(Industry Standard)*

<u>Example</u>: In a 2" Copper pipe, a flow rate of 30 GPM will experience a pressure drop (friction loss) <u>rate</u> of 2ft per 100ft of <u>equivalent</u> straight pipe it travels through.

NOTE: When selecting pipe size, a friction loss rate greater than 4.0 - 4.5 ft per 100ft should be avoided

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Operating Hours/Year	<2000 Hours per Year				<2000 and <4000 Hours per Year				>4400 Hours per Year			
Nominal Pipe Size, in.	Other (GPM)	Friction Loss Rate (Ft/100 Ft)	Variable Speed (GPM)	Friction Loss Rate (Ft/100 Ft)	Other (GPM)	Friction Loss Rate (Ft/100 Ft)	Variable Speed (GPM)	Friction Loss Rate (Ft/100 Ft)	Other (GPM)	Friction Loss Rate (Ft/100 Ft)	Variable Speed (GPM)	Friction Loss Rate (Ft/100 Ft)
2-1/2	120	10.01	180	21.78	85	5.2	130	11.66	68	3.42	110	8.48
3	180	7.26	270	15.78	140	4.5	210	9.74	110	2.86	170	6.51
4	350	6.55	530	14.56	260	3.72	400	8.46	210	2.48	320	5.52
5	410	2.84	620	6.25	310	1.67	470	3.68	250	1.12	370	2.34
6	740	3.47	1100	7.44	570	2.11	860	4.63	440	1.30	680	2.96
8	1200	2.20	1800	4.79	900	1.27	1400	2.95	**700**	**0.79**	1100	1.86
10	1800	1.52	2700	3.30	1300	0.82	2000	1.86	1000	0.50	1600	1.21
12	2500	1.18	3800	2.63	1900	0.70	2900	1.57	1500	0.45	2300	1.01
Max. Velocity for pipes 14”-24”	8.5 ft/sec		13 ft/sec		6.5 ft/sec		9.5 ft/sec		5.0 ft/sec		7.5 ft/sec	

*** Created from 2021 ASHRAE Handbook – *“Fundamentals”*, Chapter 22, Pg. 22, Table 21**

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Equivalent Length: Length of pipe, of same size as the fitting, that would result in the same pressure drop as the fitting.

TABLE 2 FITTING EQUIVALENT LENGTH TABLE

Nominal Pipe Size	90° El or Tee: Flow Thru		Tee: Side Branch Flow In or Out		90° Miter	45° Miter	Valves		
							Gate	Globe	Plug
	Screw	Cu or Weld	Screw	Cu or Weld	Weld	Weld	All	All	All
½"	1	½	2	1	2½	½	½	15	1
¾"	2	1	4	2	4	¾	½	20	1½
1"	3	1½	6	3	5	1	¾	25	2
1¼"	3½	1¾	7	3½	6	1¼	1	30	2½
1½"	4	2	8	4	7½	1½	1¼	40	3
2"	5	2½	10	5	10	2	1½	50	4
2½"	6	3	12	6	12½	2½	2	80	5
3"	8	4	16	9	15	3	2½	90	6
4"		5½		12	20	4	3	110	8
5"		8		15	25	5	3½	140	10
6"		9		18	30	6	4	170	12
8"		11		24	40	8	5	240	16
12"		18		36	60	12	8	320	24

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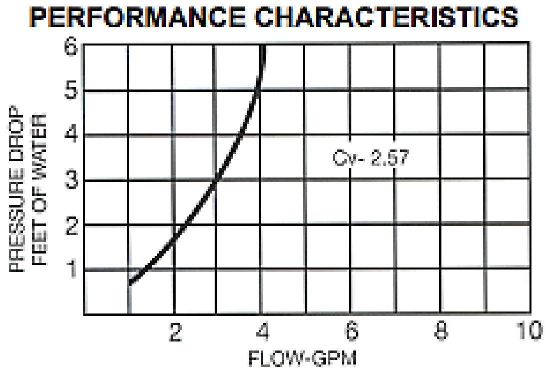
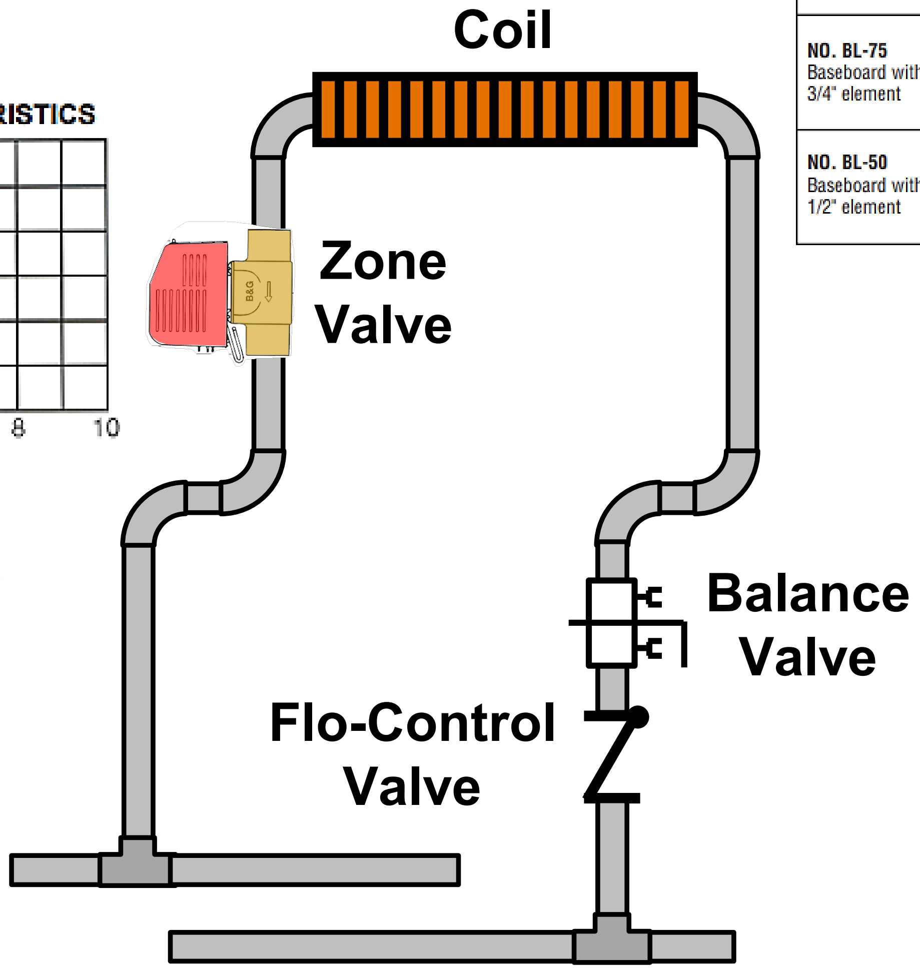
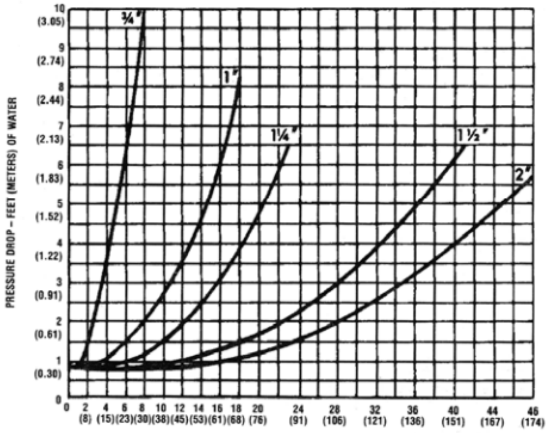
Known Pressure Drops (KPD's)

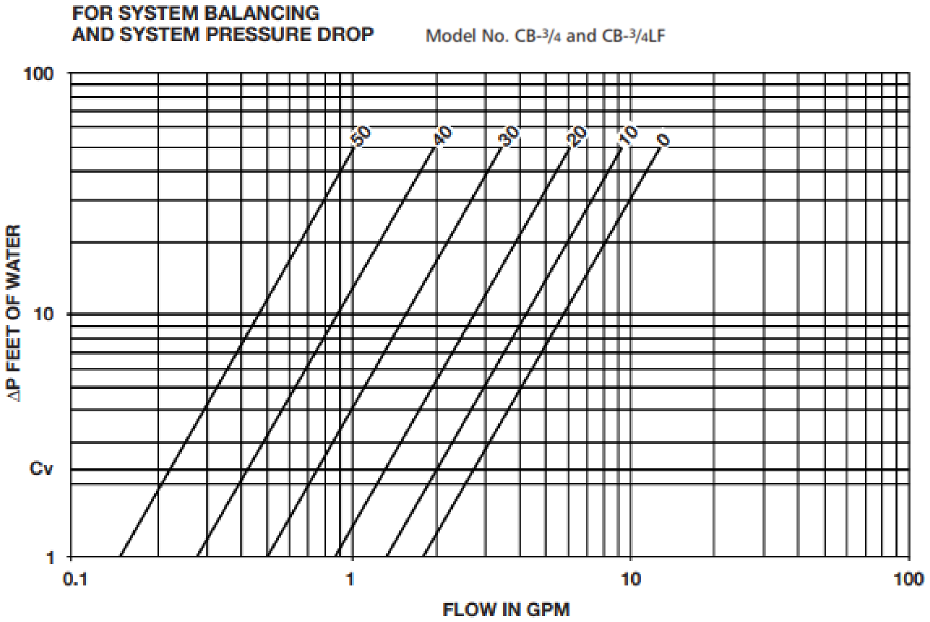






ELEMENT	WATER FLOW	PRESSURE DROP†	HOT WATER RATINGS BTU/HR. per linear ft. with 65˚F entering air												
			110˚F	120˚F	130˚F	140˚F	150˚F	160˚F	170˚F	180˚F	190˚F	200˚F	210˚F	215˚F	220°F
NO. BL-75 Baseboard with 3/4" element	1 GPM	47	150*	200*	250*	300*	360	430	500	570	630	700	770	810	840
	4 GPM	525	160*	210*	260*	320*	380	450	530	600	670	740	810	860	890
NO. BL-50 Baseboard with 1/2" element	1GPM	260	150*	200*	250*	300*	360	430	500	570	630	700	770	810	840
	4 GPM	2880	160*	210*	260*	320*	380	450	530	600	670	740	810	860	890

† Millinches per foot



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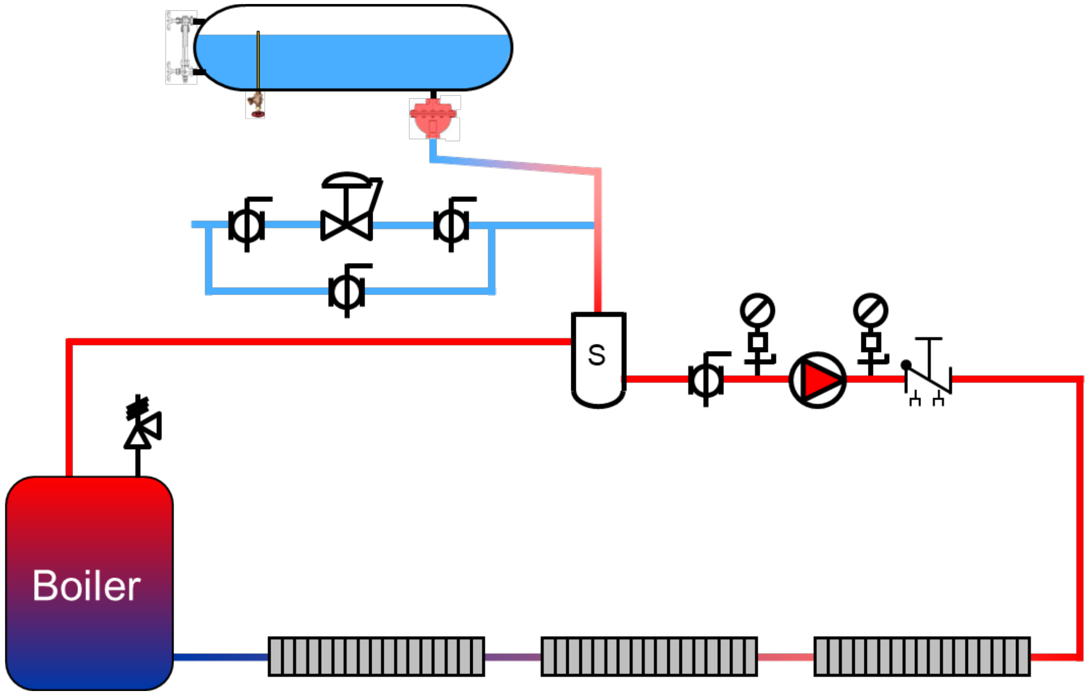
Series Loop *(Typically Heating Only)*



- Measure or calculate the Longest Circuit **TEL** ***(Pipe & Fittings)***
- Multiply **TEL** by Friction Loss Rate *(Ft. per 100')* + **KPD's = Req'd Pump Head (ft)**

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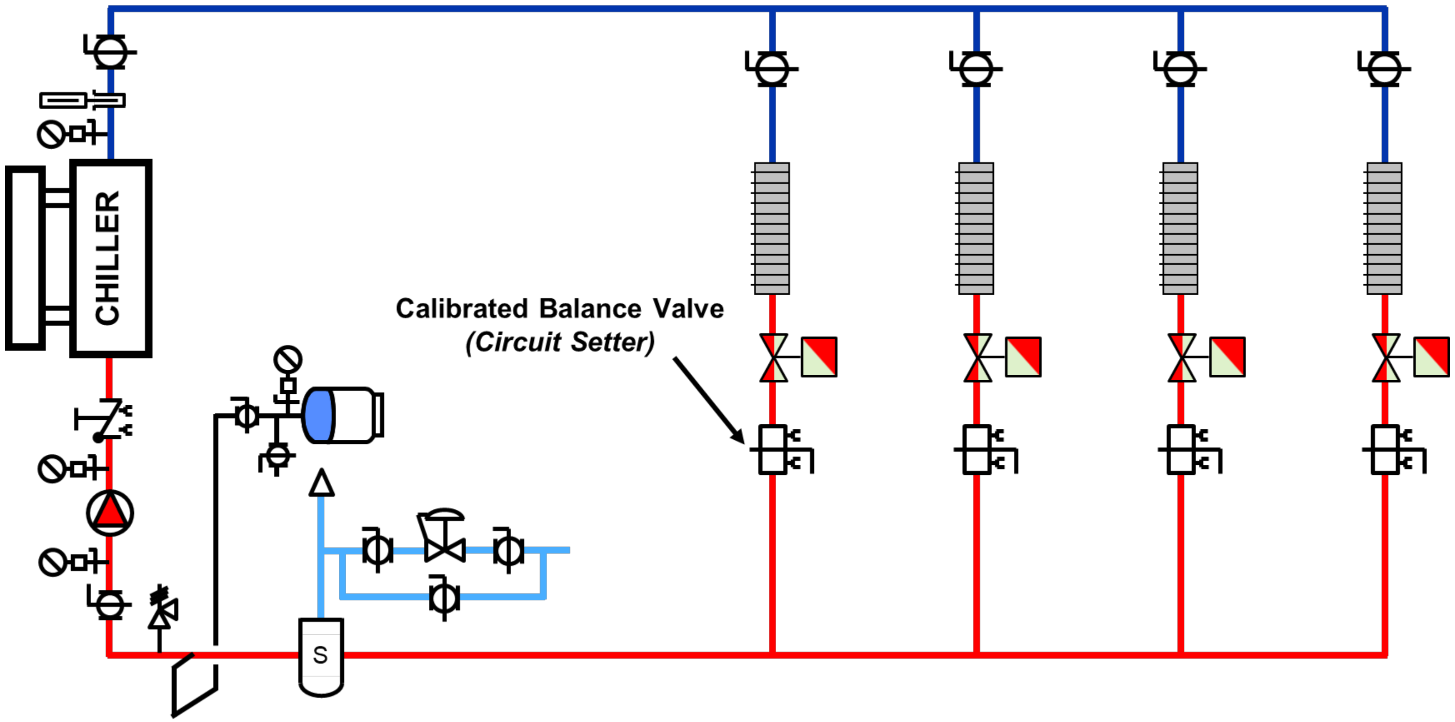
Total System Head Loss for Closed Loop Pump Sizing

Parallel Loops



Total System Head Loss = B+C for ***Critical Circuit***

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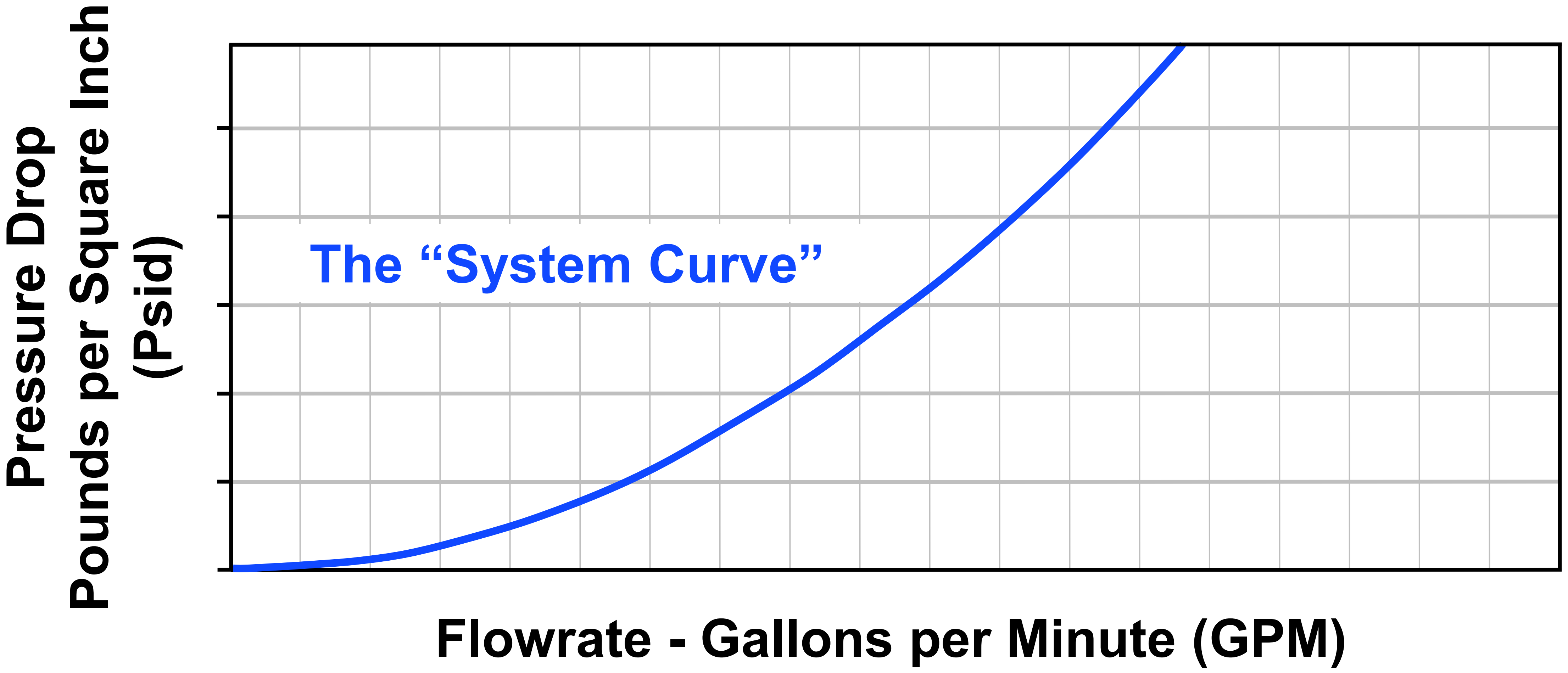

$$\textbf{Pressure Drop (ft)} = \textbf{Psid} \times \frac{2.31}{SG}$$

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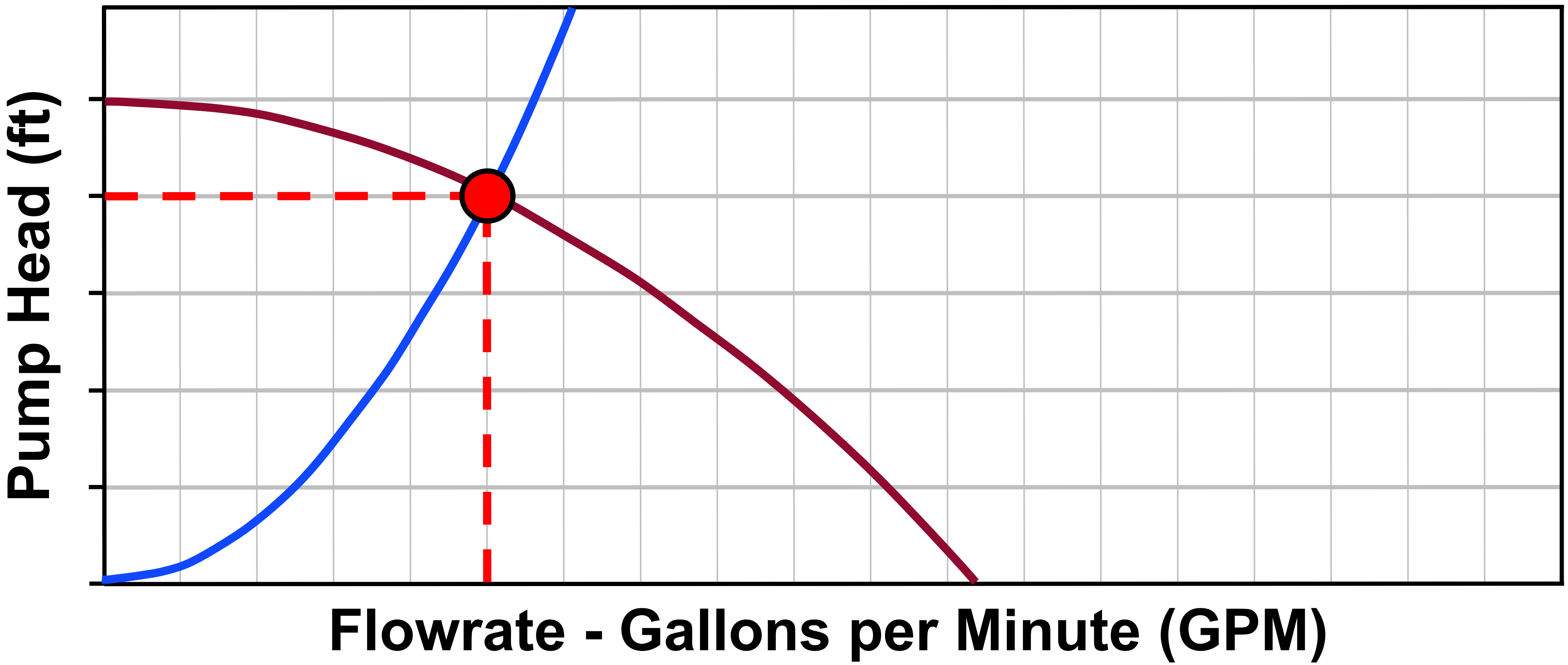
The Law of Conservation of Energy – Part 1



- Intersection of **Pump** and **System** curve is the point where Pump Head and System Friction Losses are equal. ***"All the pump head must be consumed"***

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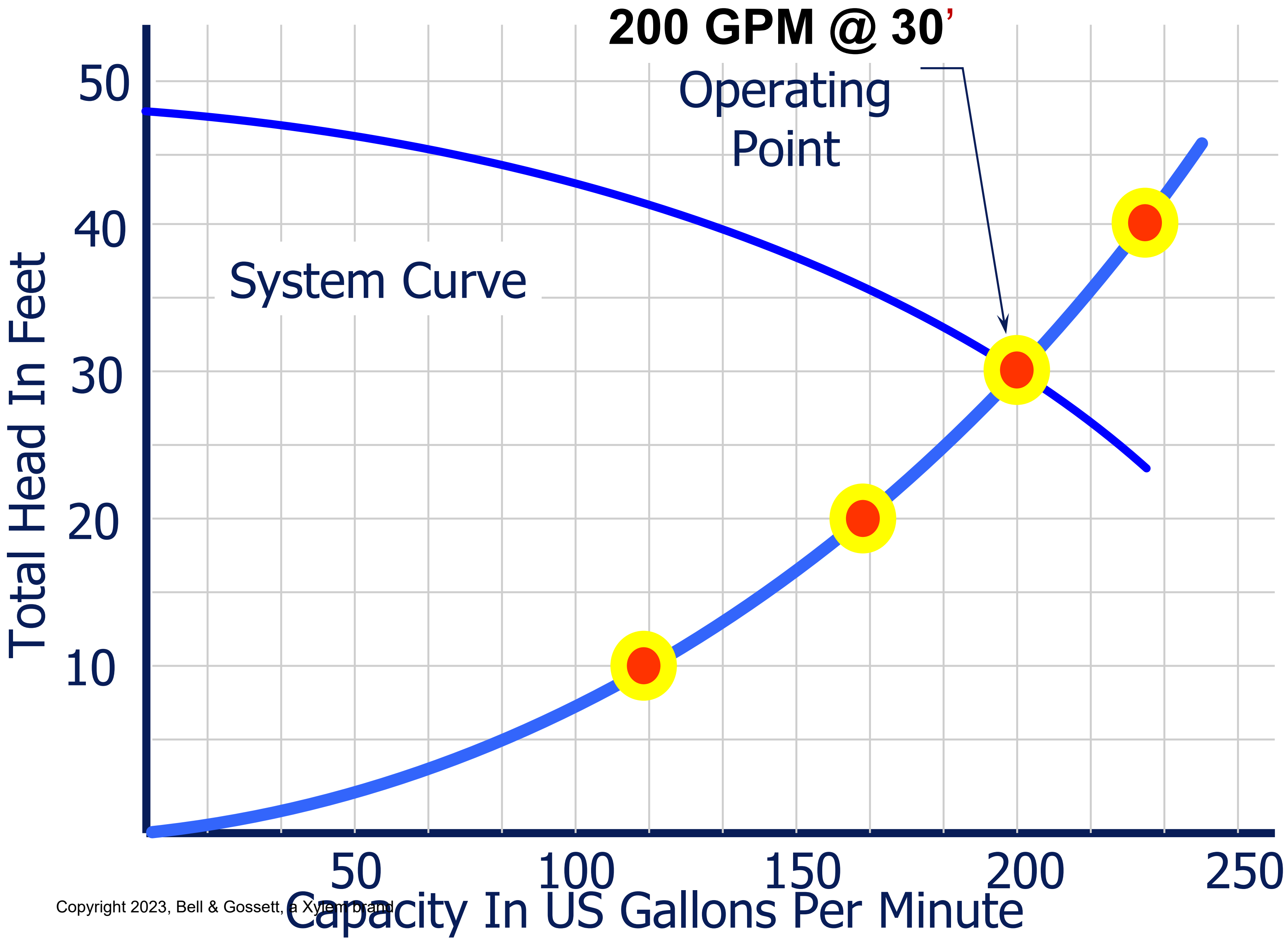
The Variable Head Loss



$$\left(\frac{Q_2}{Q_1}\right)^2 = \left(\frac{h_2}{h_1}\right)$$

$h_2 = (Q_2/Q_1)^2 \times h_1$

- Q_1 = Known (design) Flow
- Q_2 = Final Flow
- h_1 = Known (design) Head
- h_2 = Final Head

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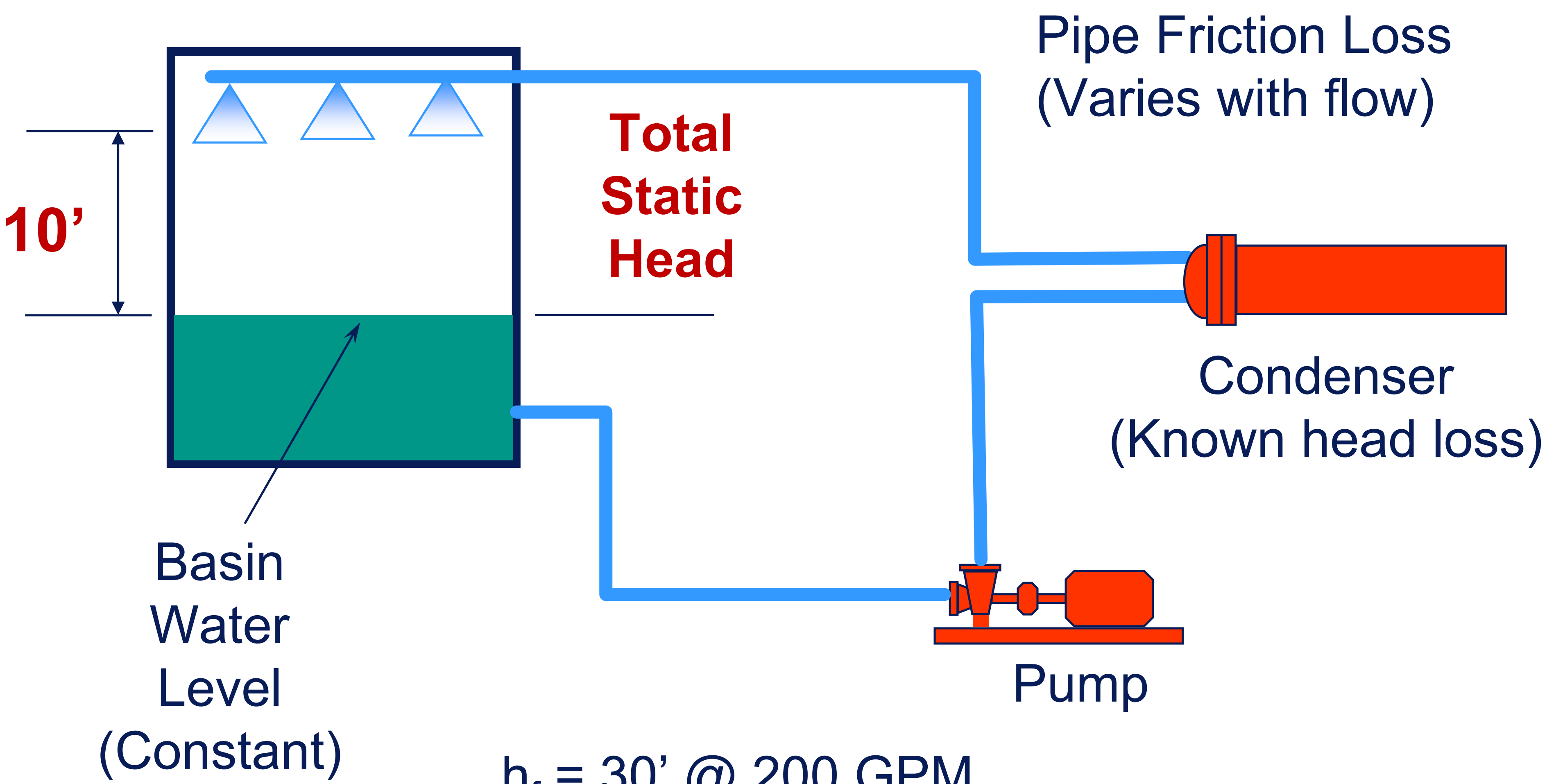
Must account for Constant (Static) Head Loss



h_f = 30’ @ 200 GPM

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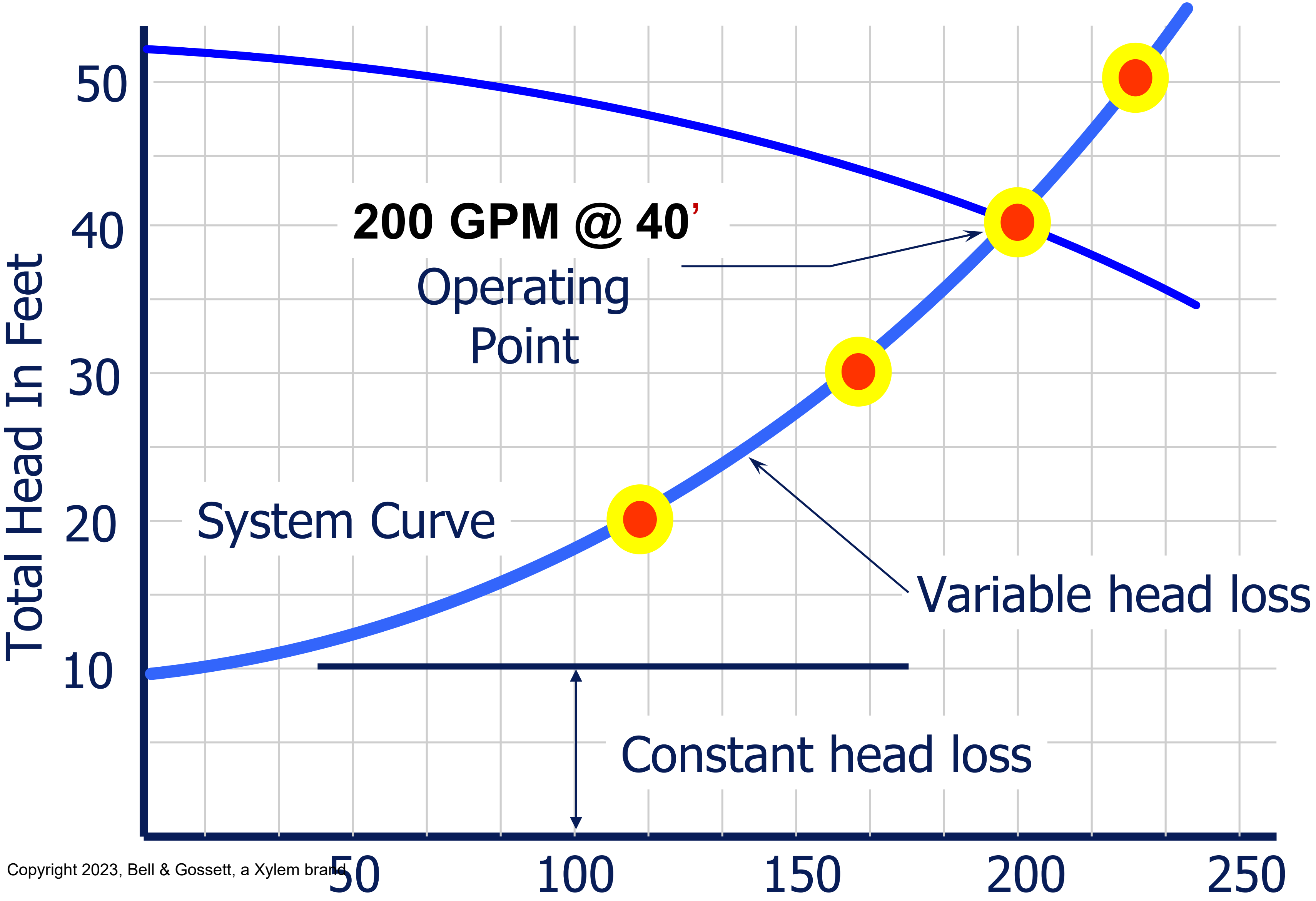
Must account for Constant (Static) Head Loss



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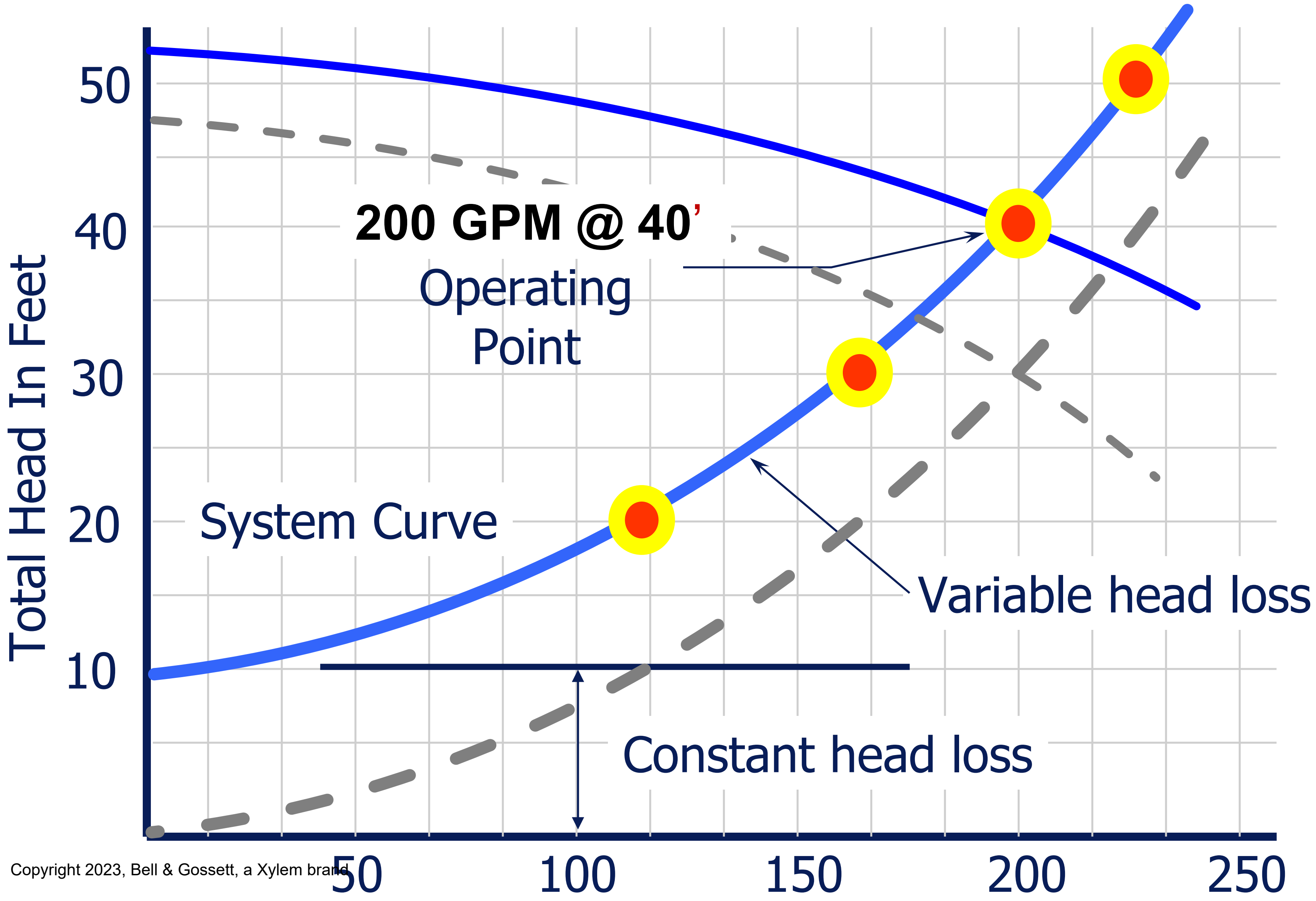
Must account for Constant (Static) Head Loss



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Net Positive Suction Head (NPSH)

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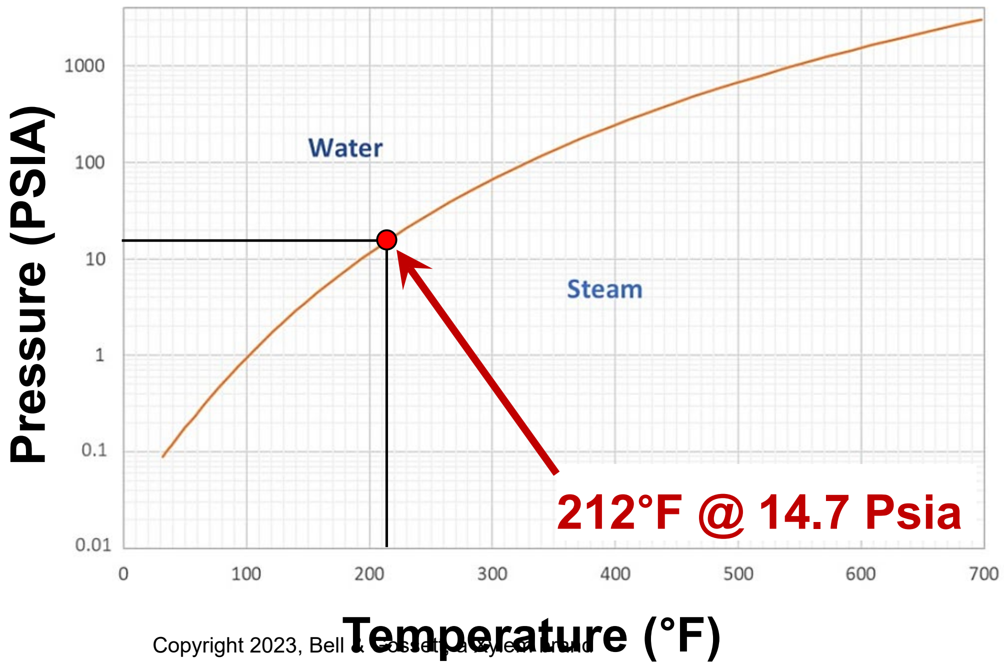
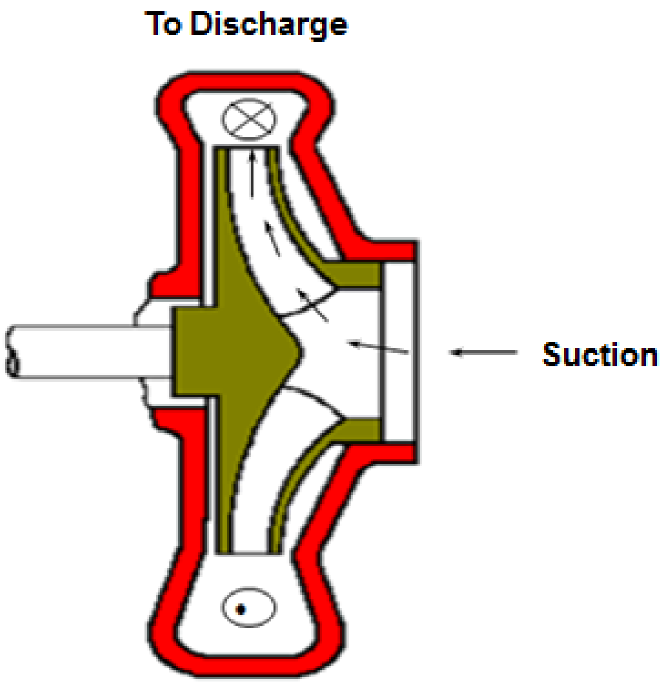
Avoiding

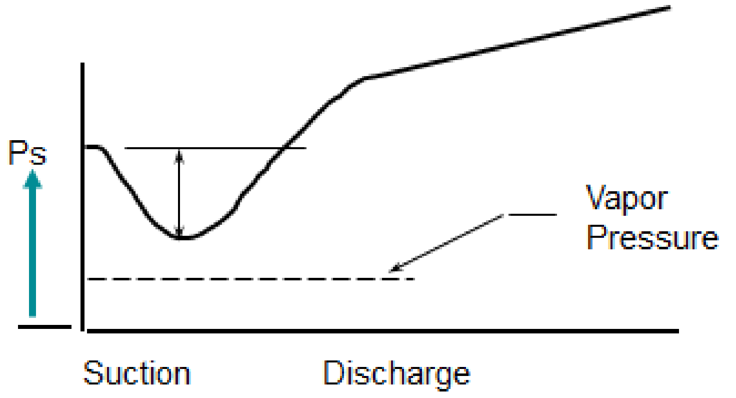
Fluid Vapor Pressure Details:

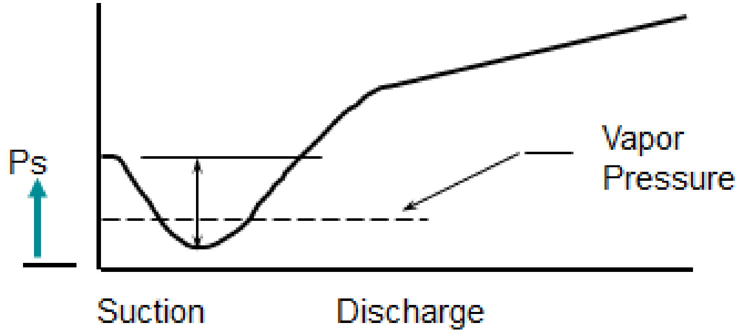
- Type of Fluid?
- Operating Temperature?
- Operating Pressure?

Water Saturation (Vapor) Pressure









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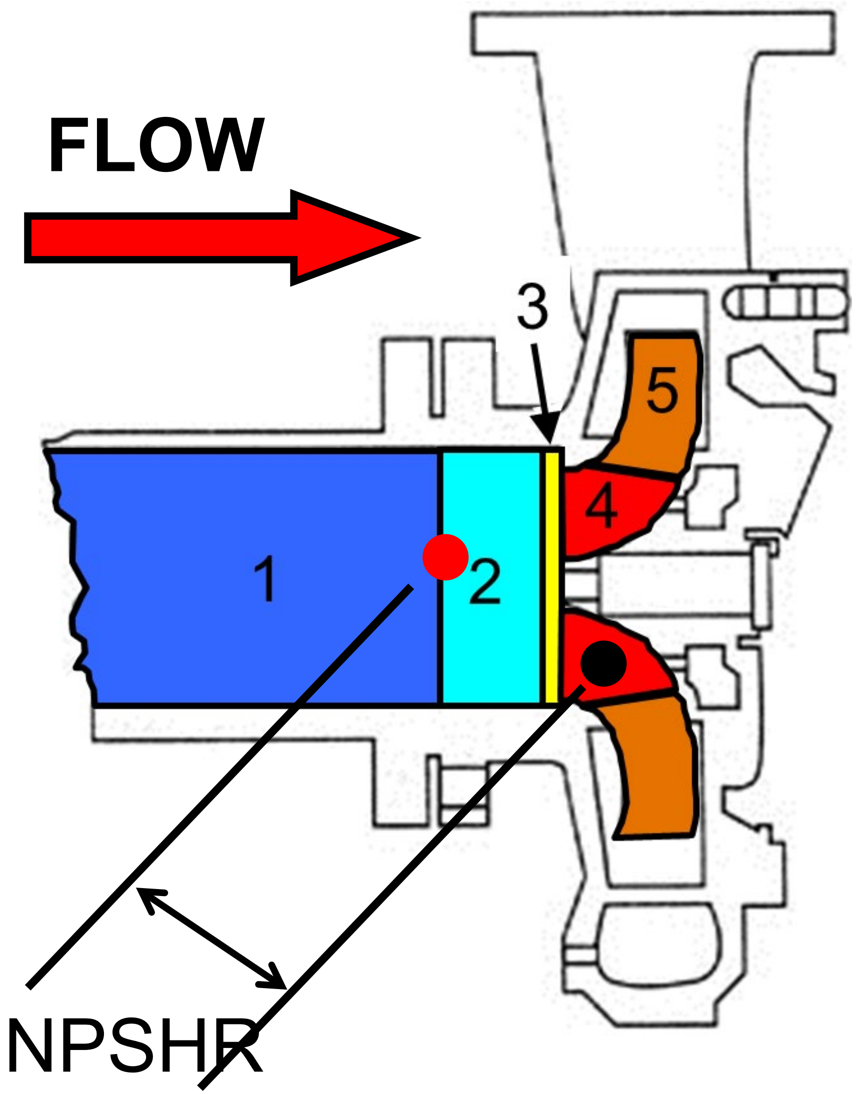
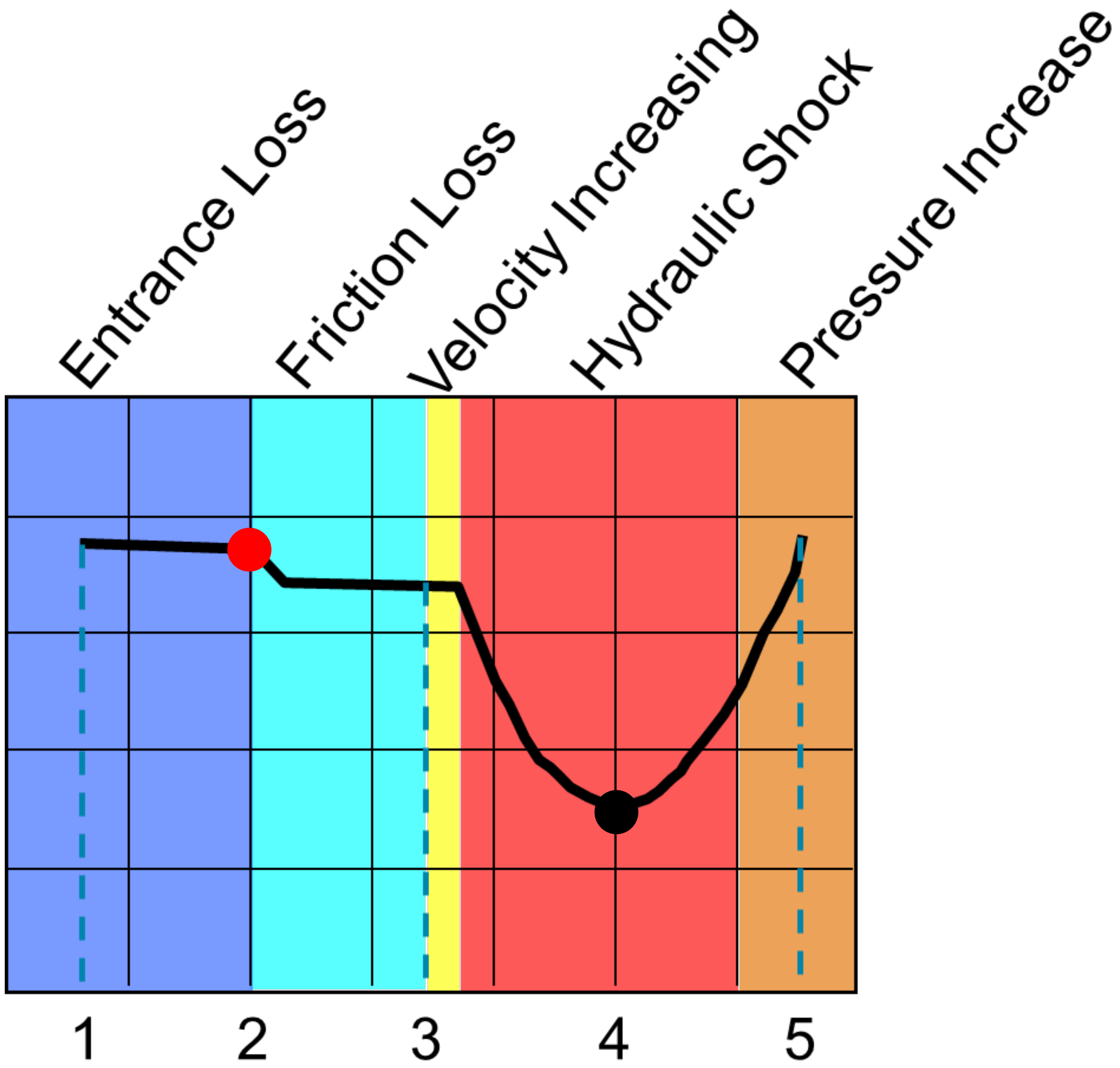
Net Positive Head Required (NPSHR)





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$$NPSHA = \frac{2.31(P_a - P_v)}{spgr} + (H_e - H_f)$$

Where:

P_a pressure in the receiver, (psia)
P_v vapor pressure of the liquid at its maximum temperature (psia)
H_e elevation head (feet)
H_f friction losses in the suction piping at the required flow rate (feet)

NOTE:

(psia): Gauge pressure reading + Atmospheric pressure *(Elevation reference to Sea Level?)*

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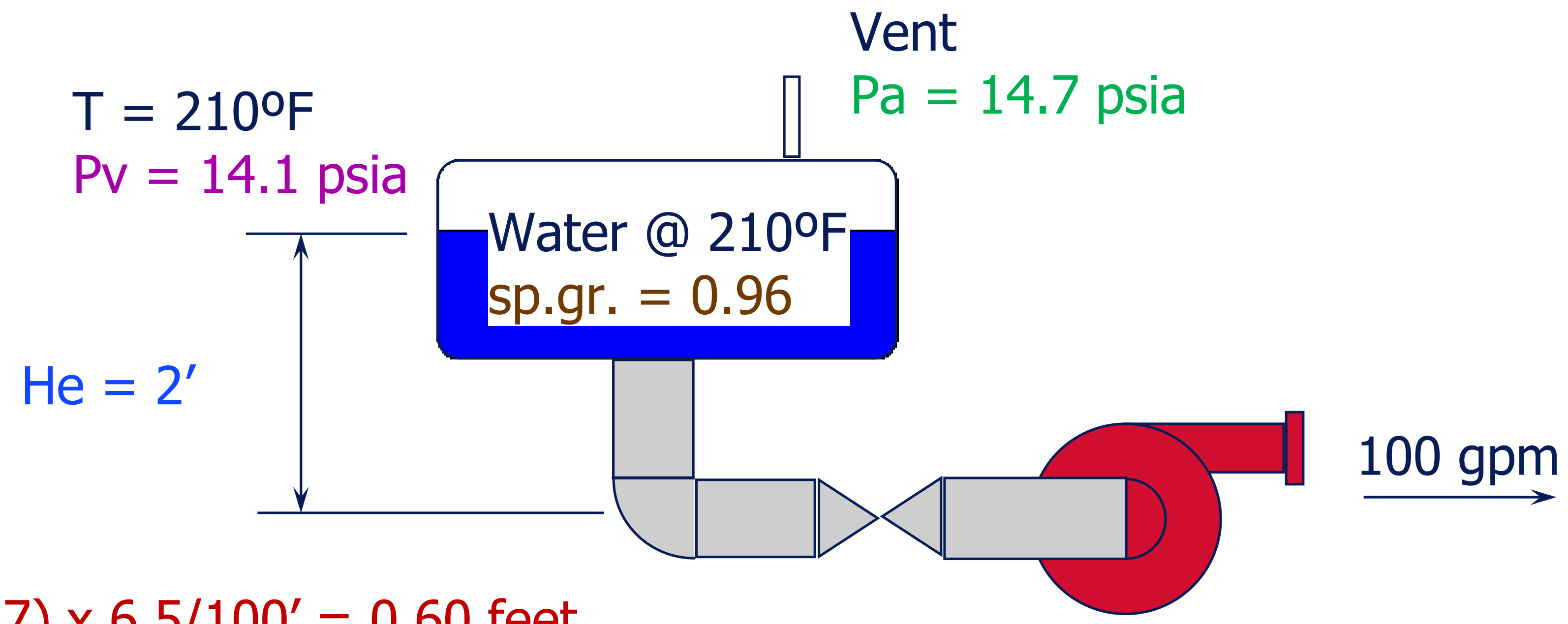

Hf = (4.0 + 3.6 + 1.7) x 6.5/100′ = 0.60 feet

100 GPM: 2½″ Iron Pipe

FLR: 6.45′/100′ (Water @ 210°F)

2½″ Measured Straight Pipe: 4.0′

2½″ Iron Reg. 90° Elbow: 3.6′ TEL

2½″ Screwed Gate Valve: 1.7′ TEL

$$NPSHA = \frac{2.31(Pa - Pv)}{sp\ gr} + (He - Hf)$$

$$= \frac{2.31(14.7 - 14.1)}{0.96} + (2 - 0.60)$$

= 2.84 feet

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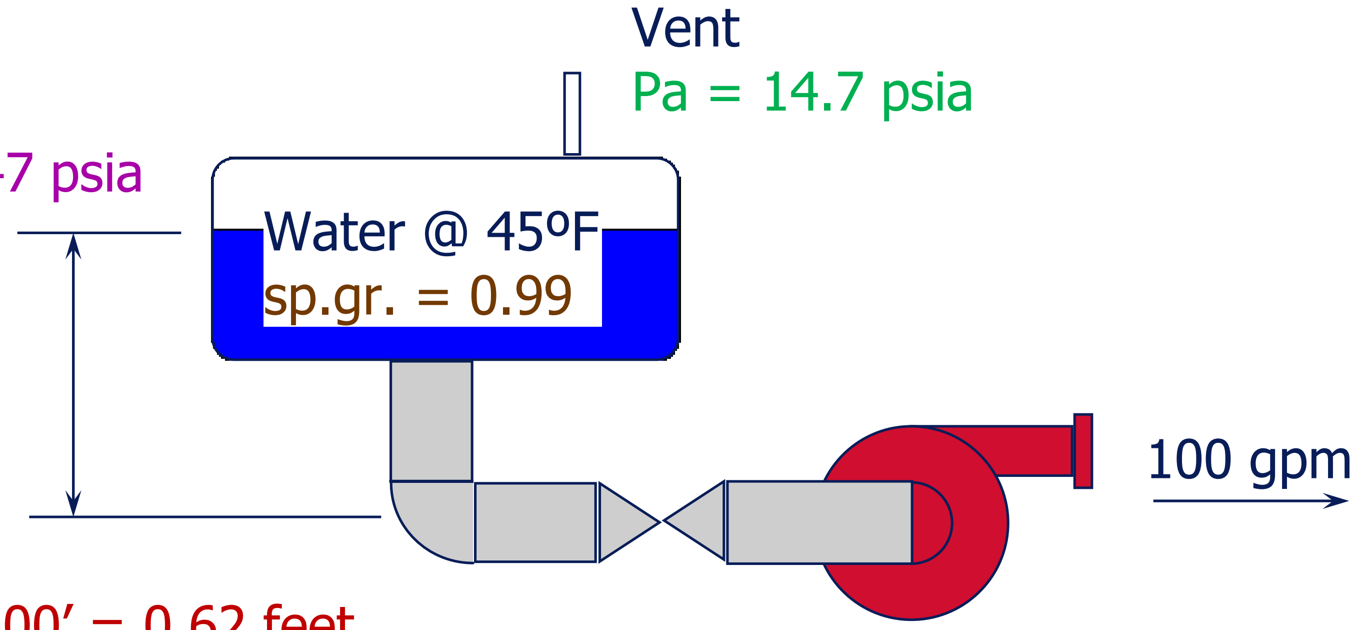

Hf = (4.0 + 3.6 + 1.7) x 6.7/100′ = 0.62 feet

100 GPM: 2½″ Iron Pipe

FLR: 6.7′/100′ (Water @ 45°F)

2½″ Measured Straight Pipe: 4.0′

2½″ Iron Reg. 90° Elbow: 3.6′ TEL

2½″ Screwed Gate Valve: 1.7′ TEL

$$NPSHA = \frac{2.31(Pa - Pv)}{sp\ gr} + (He - Hf)$$

$$= \frac{2.31(14.7 - 0.147)}{0.99} + (2 - 0.62)$$

= 35.34 feet

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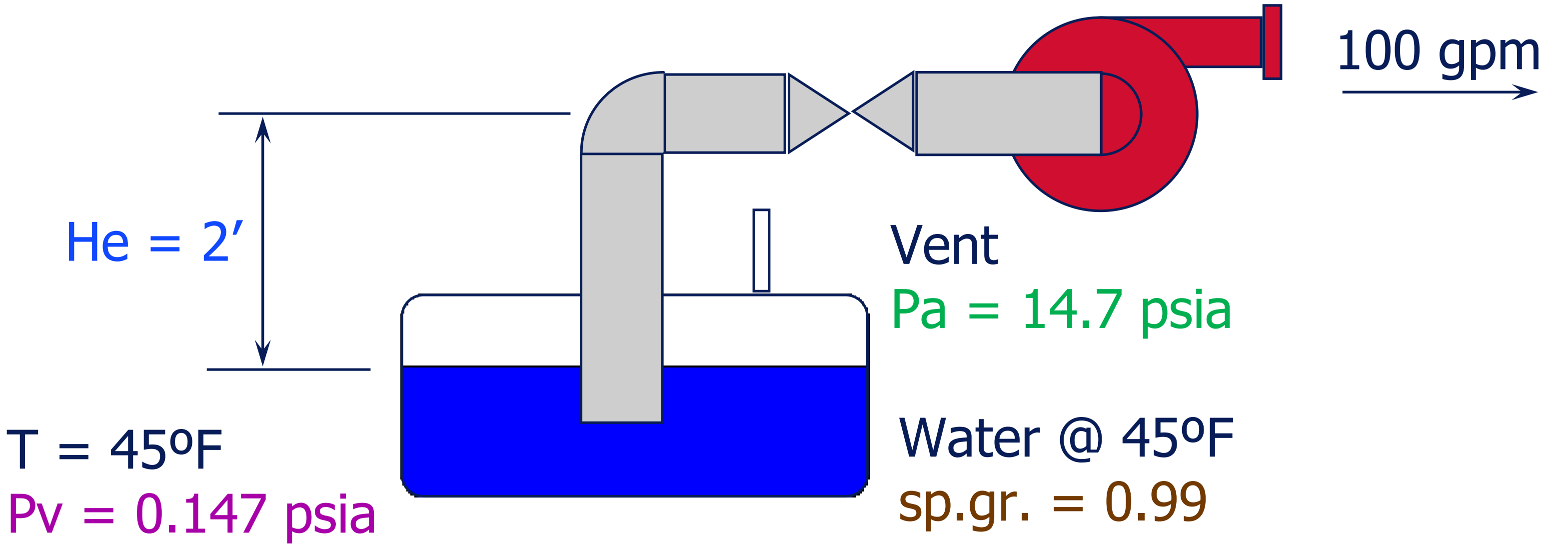

Hf = (4.0 + 3.6 + 1.7) x 6.7/100′ = 0.62 feet

100 GPM: 2½″ Iron Pipe

FLR: 6.7′/100′ (Water @ 45°F)

2½″ Measured Straight Pipe: 4.0′

2½″ Iron Reg. 90° Elbow: 3.6′ TEL

2½″ Screwed Gate Valve: 1.7′ TEL

$$NPSHA = \frac{2.31(Pa - Pv)}{sp\ gr} + (He - Hf)$$

$$= \frac{2.31(14.7 - 0.147)}{0.99} + (-2 - 0.62)$$

= 31.34 feet

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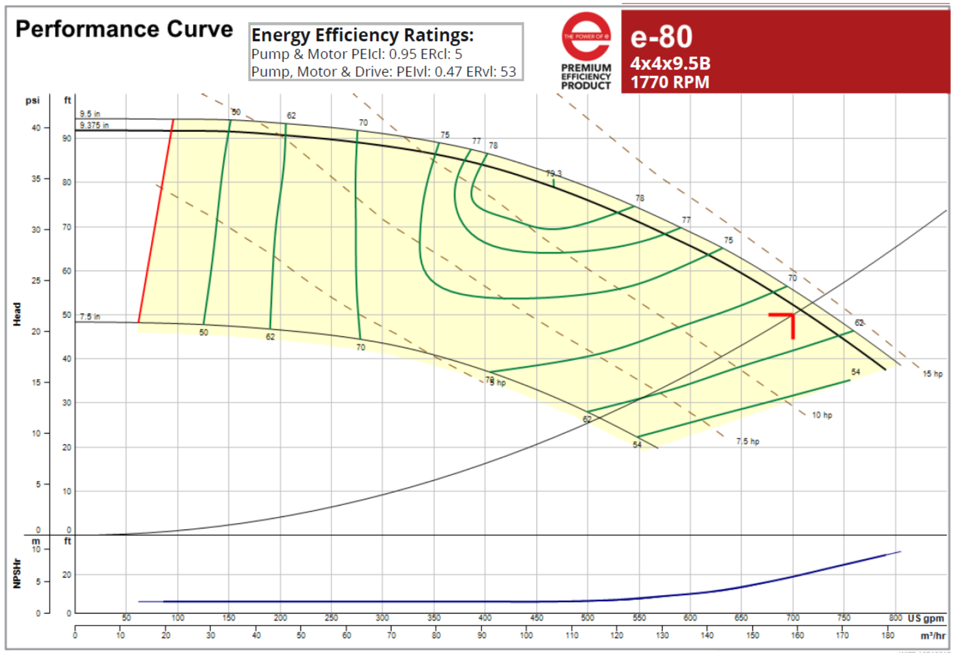

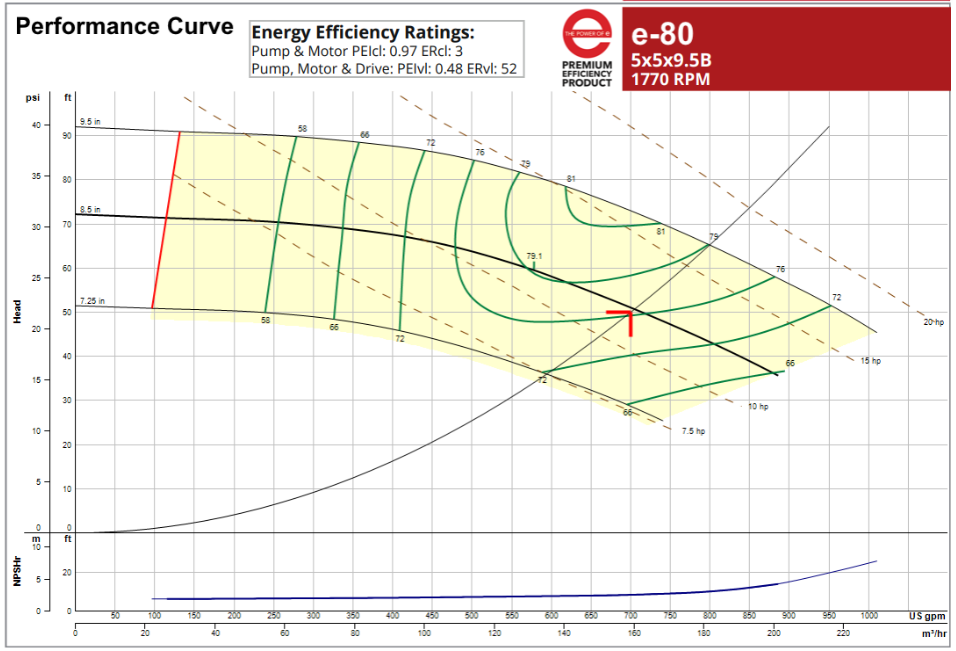

Close Coupled In-Line Centrifugal Pump

Series: e-80

Model: 4x4x9.5B

Features & Design
Best in Class Hydraulic Performance
Low Operating and Maintenance Cost
Horizontal or Vertical Installation

The Series e-80 is a highly efficient, heavy duty, close coupled pump designed for horizontal or vertical in-line mounting. The e-80 is available in stainless steel fitted construction, with flows up to 2500 GPM, heads to 380 feet.

http://bellgossett.com/pumps-circulators/in-line-pumps/series-e-80/

Pump Selection Summary	
Duty Point Flow	700 US gpm
Duty Point Head	50 ft
Control Head	0 ft
Duty Point Pump Efficiency	66.7 %
Part Load Efficiency Value (PLEV)	0.0 %
Impeller Diameter	9.375 in
Motor Power	15 hp
Duty Point Power	13.6 bhp
Motor Speed	1800 rpm
RPM @ Duty Point	1770 rpm
NPSHr	19.8 ft
Minimum Shutoff Head	91.9 ft
Minimum Flow at RPM	93.4 US gpm
Flow @ BEP	467 US gpm
Fluid Temperature	68 °F
Fluid Type	Water
Weight *(approx. - consult rep for exact)*	525 lbs
Pump Floor Space Calculation	4.44 ft²

Close Coupled In-Line Centrifugal Pump

Series: e-80

Model: 5x5x9.5B

Features & Design
Best in Class Hydraulic Performance
Low Operating and Maintenance Cost
Horizontal or Vertical Installation

The Series e-80 is a highly efficient, heavy duty, close coupled pump designed for horizontal or vertical in-line mounting. The e-80 is available in stainless steel fitted construction, with flows up to 2500 GPM, heads to 380 feet.

http://bellgossett.com/pumps-circulators/in-line-pumps/series-e-80/

Pump Selection Summary	
Duty Point Flow	700 US gpm
Duty Point Head	50 ft
Control Head	0 ft
Duty Point Pump Efficiency	76.4 %
Part Load Efficiency Value (PLEV)	0.0 %
Impeller Diameter	8.5 in
Motor Power	15 hp
Duty Point Power	11.8 bhp
Motor Speed	1800 rpm
RPM @ Duty Point	1770 rpm
NPSHr	8.47 ft
Minimum Shutoff Head	72.2 ft
Minimum Flow at RPM	116 US gpm
Flow @ BEP	578 US gpm
Fluid Temperature	68 °F
Fluid Type	Water
Weight *(approx. - consult rep for exact)*	405 lbs
Pump Floor Space Calculation	5.3 ft²

NPSHr = 19.8’

NPSHr = 8.5’

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- NPSH Required
  - impeller design, shape, materials
  - plotted on pump curve
  - increases with flow

- NPSH Available
  - Positives
    - » Static suction head
    - » Lower vapor pressure
    - » Higher system pressure
  - Negatives
    - » Friction losses
    - » Suction lift

To avoid cavitation: NPSHA > NPSHR

*** Suggest minimum 5% more NPSHA than NPSHR**

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Let's Solve Water

Pump Head – Bernoulli's Theorem for Open Loop Systems

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Open Loop

Work applied by pump (Pump Head)

Energy (Fluid Head) lost due to friction

$$\frac{P_a}{W} + Z_a + \frac{V_a^2}{2g} + E_p = \frac{P_b}{W} + Z_b + \frac{V_b^2}{2g} + h_f$$

Total Pump Head

$$E_p = \left(\frac{P_b}{W} - \frac{P_a}{W}\right) + \left(Z_b - Z_a\right) + \left(\frac{V_b^2}{2g} - \frac{V_a^2}{2g}\right) + h_f$$

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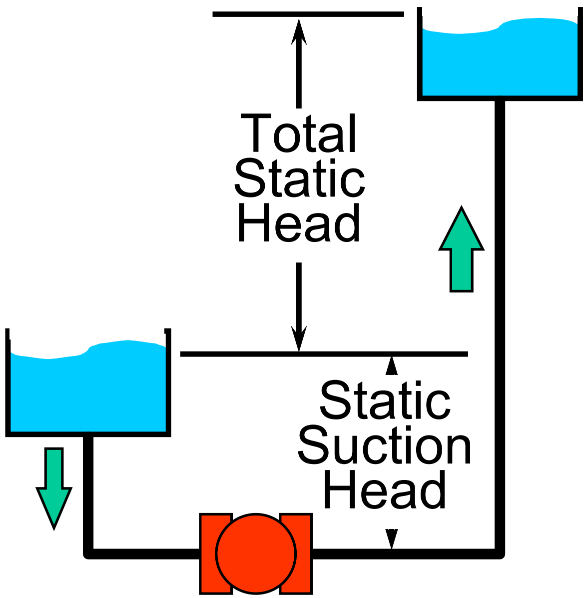
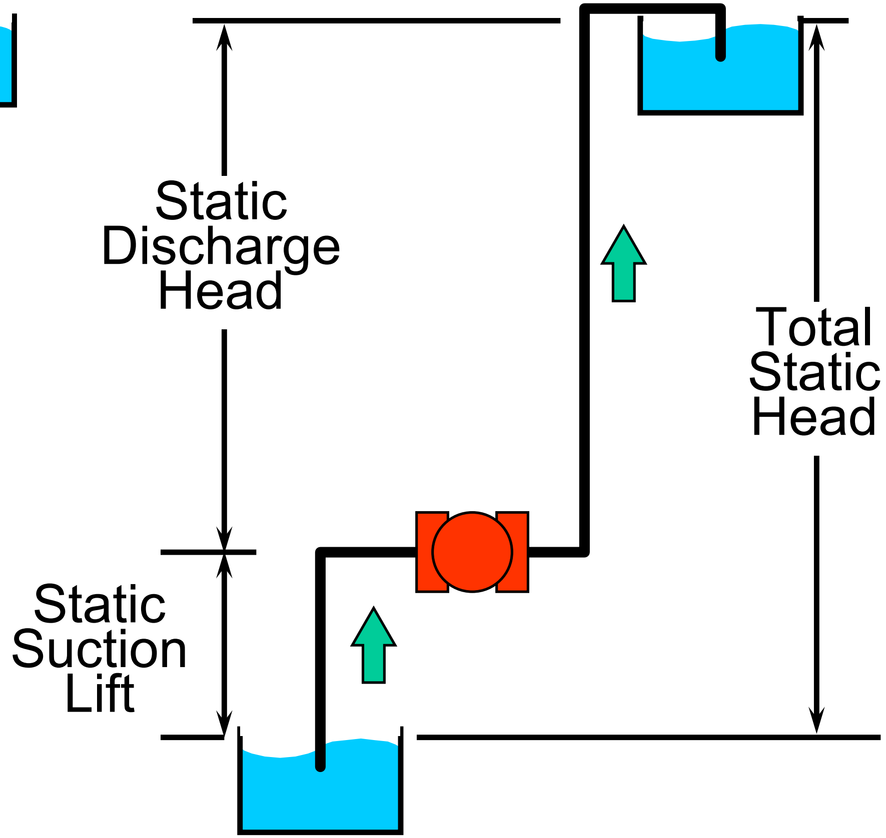
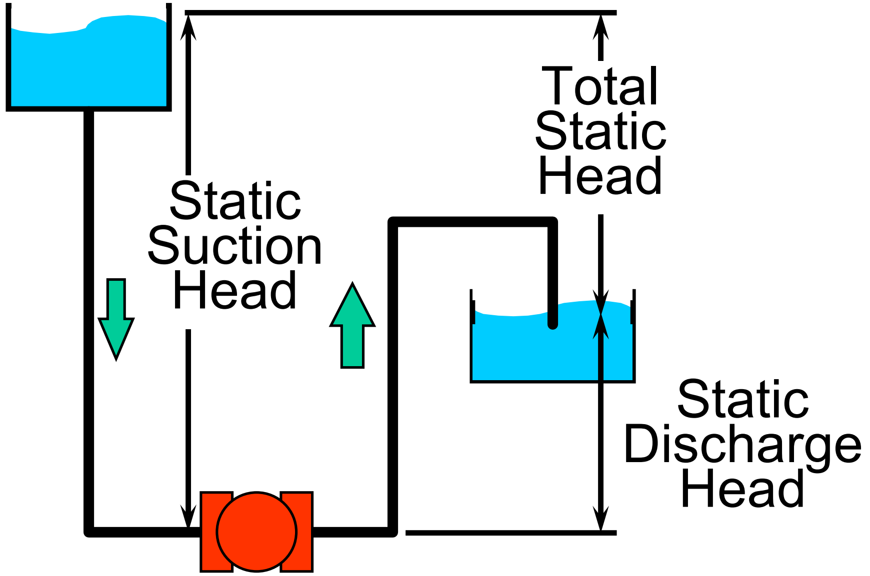
Open Loop System - **“Total Static Head”**



Static Suction Head
Less Than
Static Discharge Head



Static Suction Lift
Plus
Static Discharge Head



Static Suction Head
Greater Than
Static Discharge Head

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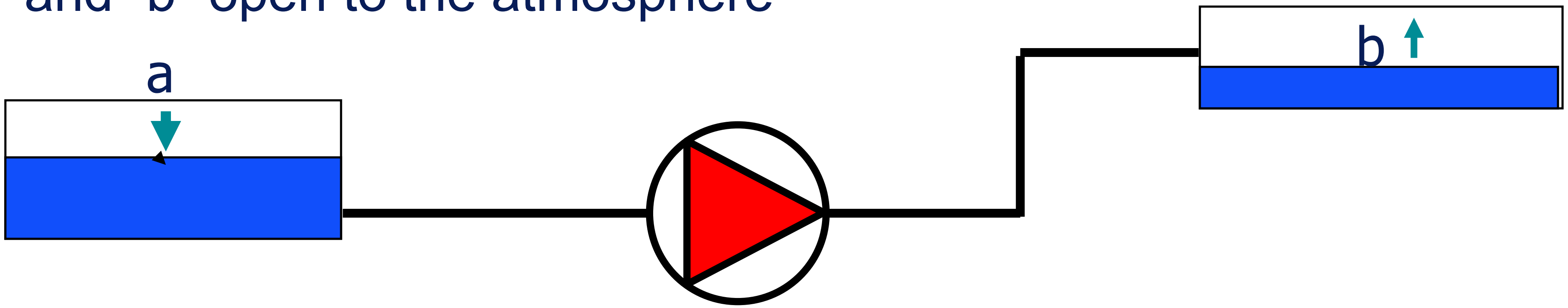
Calculating required Pump Head: ***Open Loop*** System Scenario #1

Tanks “a” and “b” open to the atmosphere



$$\frac{P_a}{W} + Z_a + \frac{V_a^2}{2g} + E_p = \frac{P_b}{W} + Z_b + \frac{V_b^2}{2g} + h_f$$

$$E_p = \left(\frac{P_b}{W} - \frac{P_a}{W}\right) + (Z_b - Z_a) + \left(\frac{V_b^2}{2g} - \frac{V_a^2}{2g}\right) + h_f$$

- Pressure differences don’t exist.
- Pump head is determined by elevation, velocity differences and friction in pipes, fittings.

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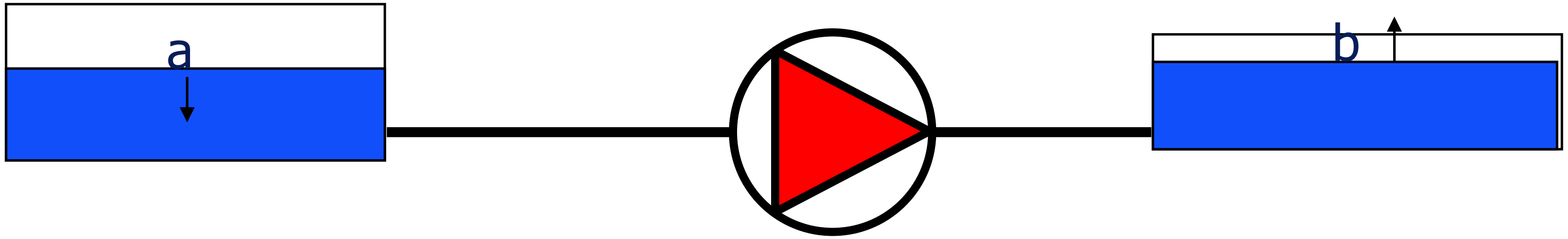
Calculating required Pump Head: ***Open Loop*** System Scenario #2

Tanks “a” and “b” are at the same level, however at different pressures



$$\frac{P_a}{W} + Z_a + \frac{V_a^2}{2g} + E_p = \frac{P_b}{W} + Z_b + \frac{V_b^2}{2g} + h_f$$

$$E_p = \left(\frac{P_b}{W} - \frac{P_a}{W}\right) + (Z_b - Z_a) + \left(\frac{V_b^2}{2g} - \frac{V_a^2}{2g}\right) + h_f$$

- Elevation differences don’t exist.
- Pump head is determined by pressure, velocity differences and friction in pipes, fittings.

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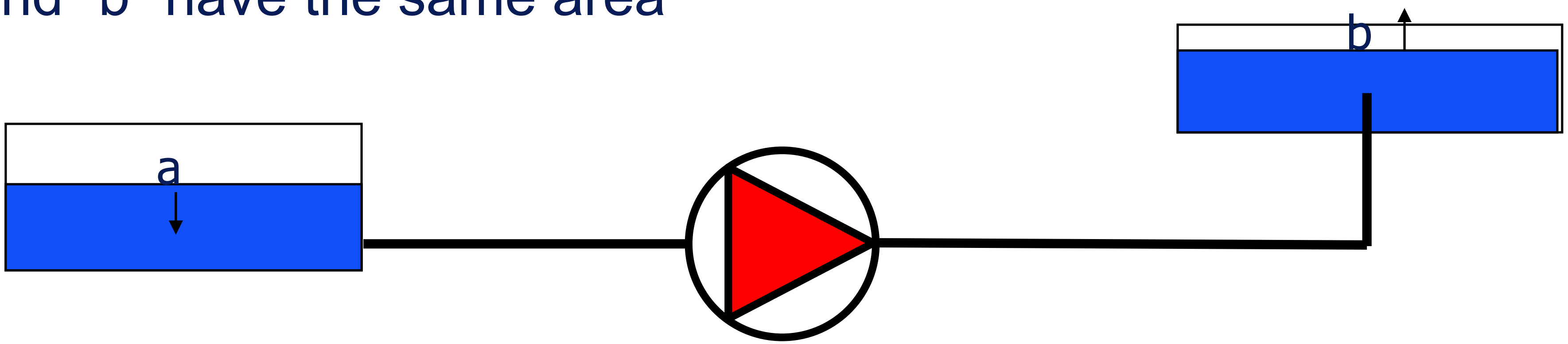
Calculating required Pump Head: ***Open Loop*** System Scenario #3

Tanks "a" and "b" have the same area



$$\frac{P_a}{W} + Z_a + \frac{V_a^2}{2g} + E_p = \frac{P_b}{W} + Z_b + \frac{V_b^2}{2g} + h_f$$

$$E_p = \left(\frac{P_b}{W} - \frac{P_a}{W}\right) + (Z_b - Z_a) + \left(\frac{V_b^2}{2g} - \frac{V_a^2}{2g}\right) + h_f$$

- Velocity differences don't exist.
- Pump head is determined by pressure, elevation differences and friction in pipes, fittings.

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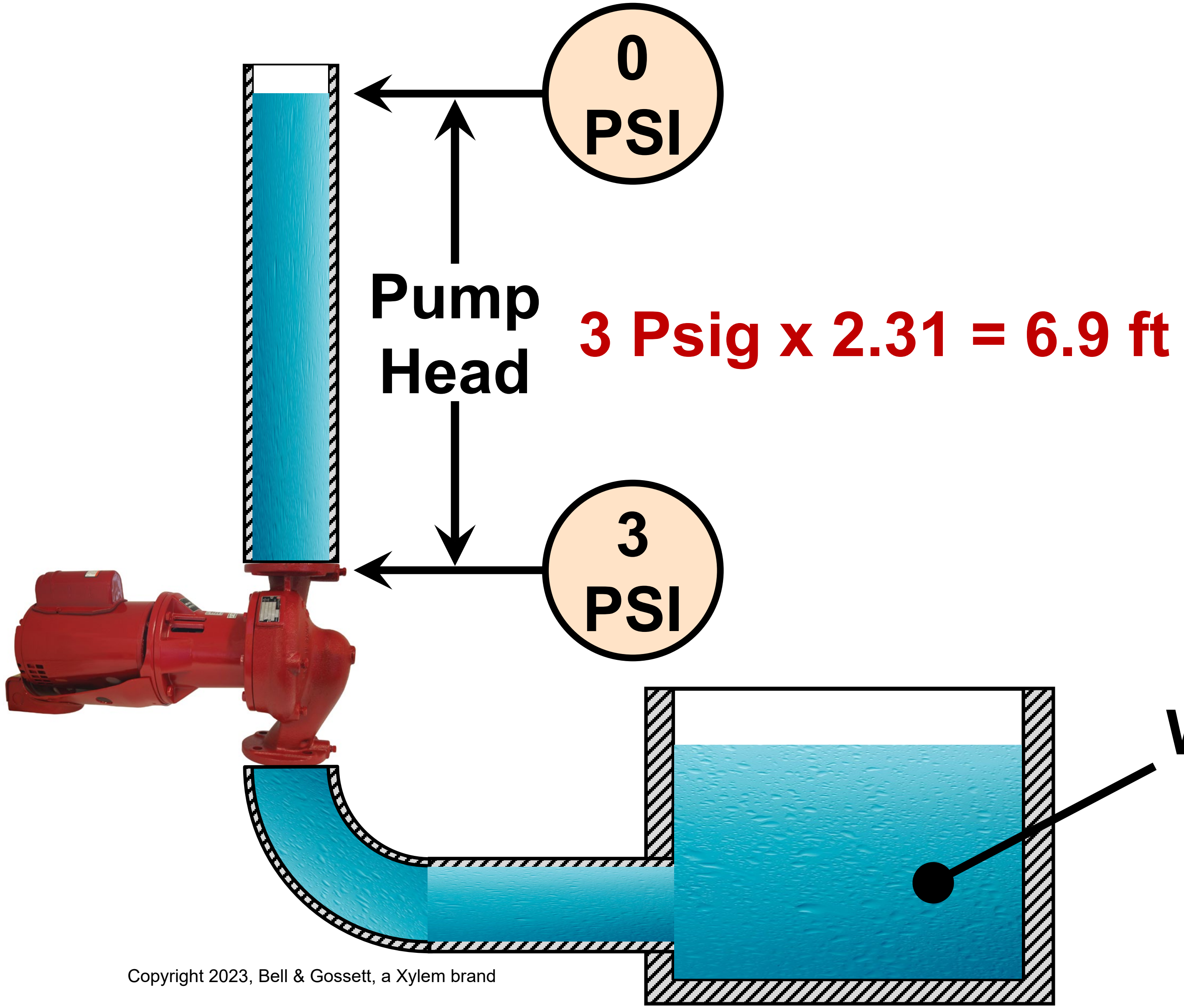
- Water column **"Elevation Head"** equals **"Pump Head"** at 6.9 ft
- Pump running, but no flow occurs



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Select an **e-1510** base mounted pump, operated at **1800 RPM constant speed**, for the condenser/cooling tower loop that supplies a **100 Ton** water-cooled chiller. The tower will be an Induced Draft Counterflow type, supplying **85°F** cold water, have a range of **10°F**, and require a hot water inlet pressure of **4 PSI.** At design flow, the chiller will have a **22'** pressure drop.

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Step 1: Required Pump Flow

30/ΔT °F = 30/10 = 3 GPM/Ton x 100 Ton = 300 GPM

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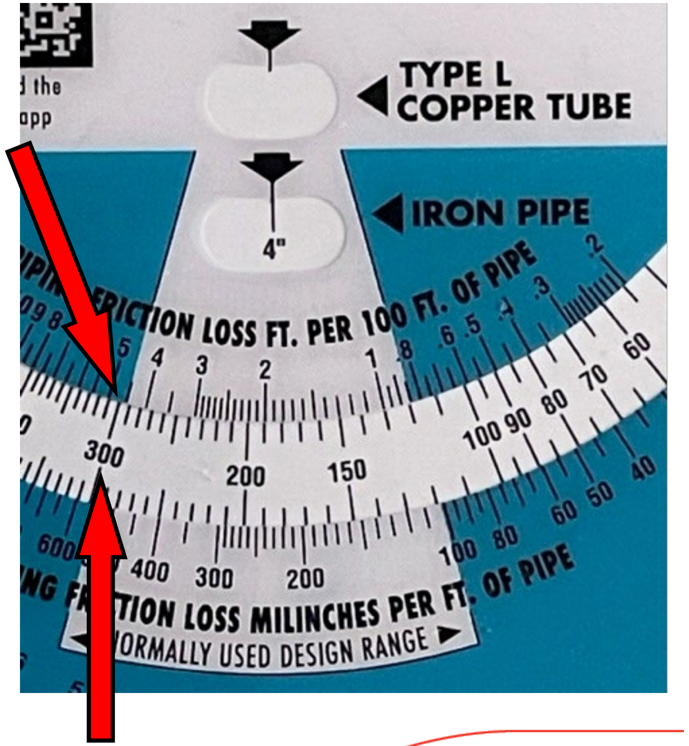
Step 1: Required Pump Flow

30/ΔT °F = 30/10 = 3 GPM/Ton x 100 Ton = 300 GPM

Step 2: Required Pipe Size & Friction Loss Rate

(Scale #2 on System Syzer, Iron Pipe)

4” Pipe, Friction Loss Rate 4.75’/100’ TEL



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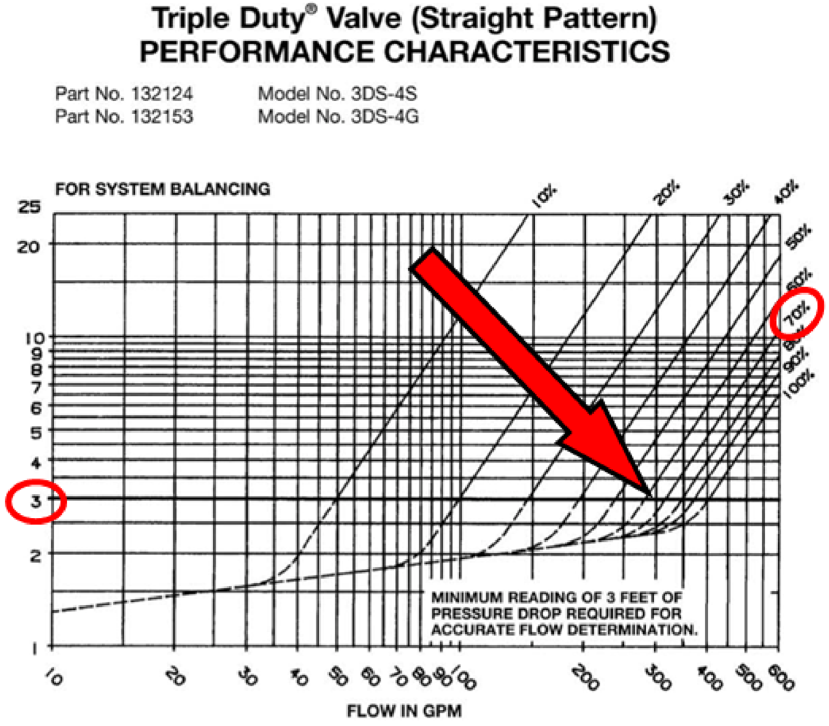
Step 2: Required Pipe Size & Friction Loss Rate

(Scale #2 on System Syzer, Iron Pipe)

4" Pipe, Friction Loss Rate 4.75'/100' TEL



Step 3: Select Triple Duty Valve, VerifyPressure Drop

300 GPM, 3DS-4S, 3.0' (70% Open)

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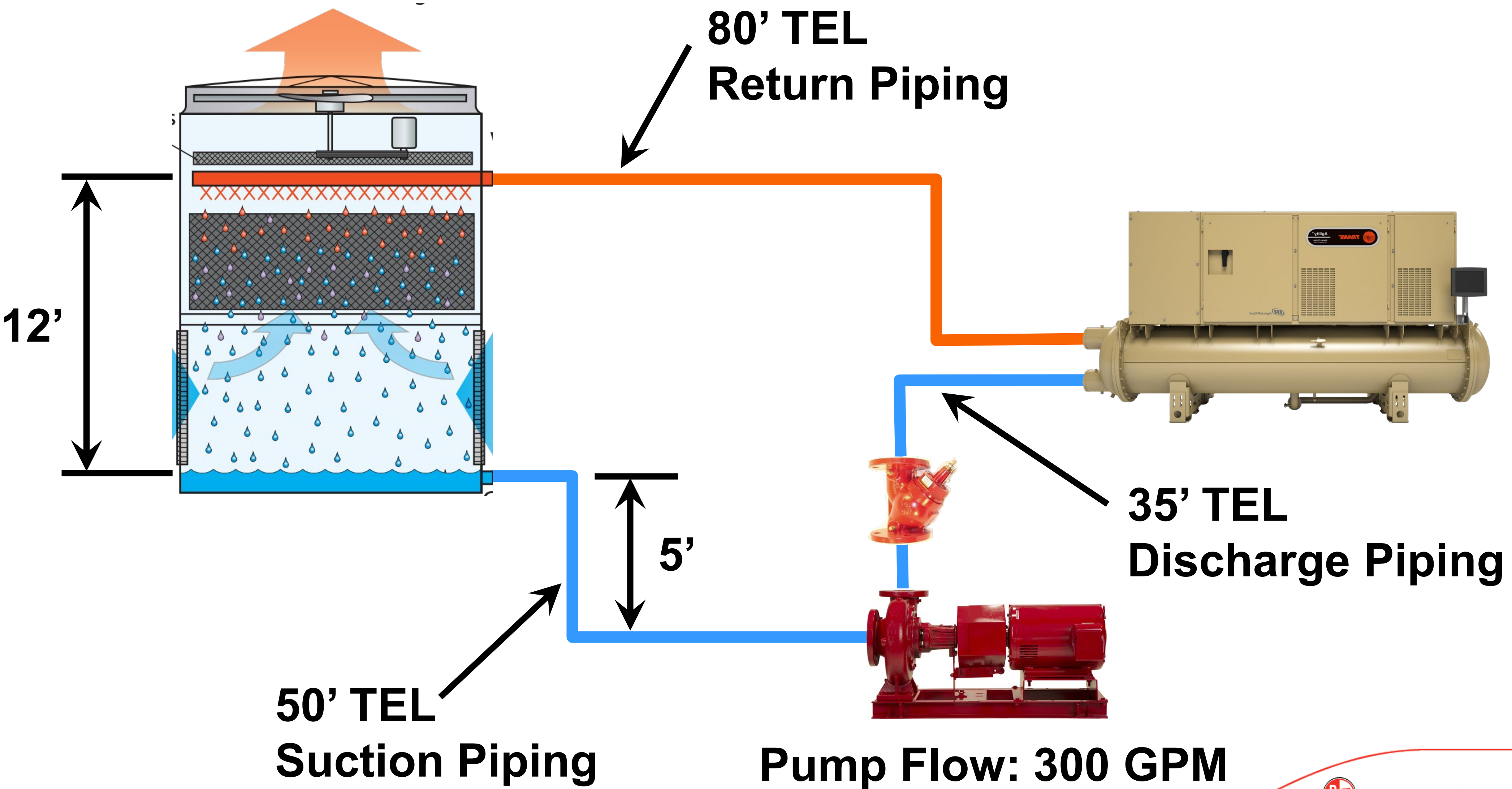
NOTE: Jobsite at Sea Level



Pump Flow: 300 GPM

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Step 5: Determine Piping Friction Losses

35' + 80' = 115/100 = 1.15 x 4.75 = 5.5'

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Step 5: Determine Piping Friction Losses

35’ + 80’ = 115/100 = 1.15 x 4.75 = 5.5’ *(Scale #4)*

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Step 5: Determine Piping Friction Losses

35’ + 80’ = 115/100 = 1.15 x 4.75 = 5.5’ *(Scale #4)*

Step 6: Add Known Pressure Drops (KPD’s)

Condenser: 22’

Tower Inlet: 5 PSI x 2.31 = 11.5’

Triple Duty Valve: 3’

Static Head: 12’

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Step 7: Total Pump Head

5.5' + 22' + 11.5' + 3' + 12' = 54'

Step 8: Select Pump for 300 GPM @ 54'

www.esp-systemwize.com

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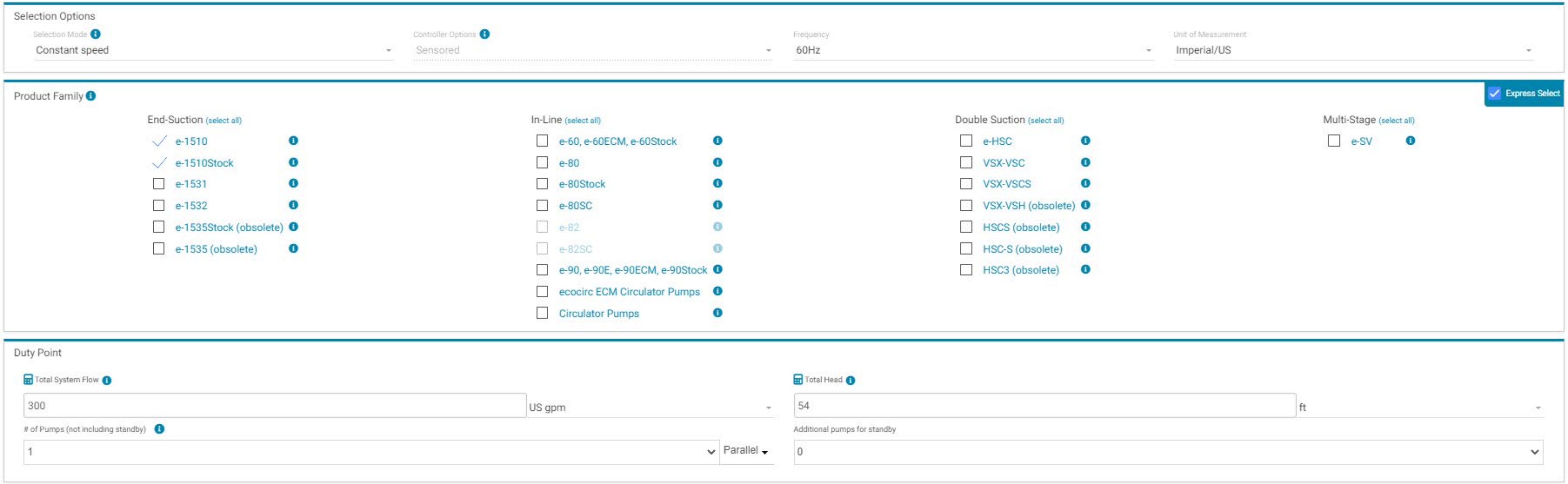
Pump Selection



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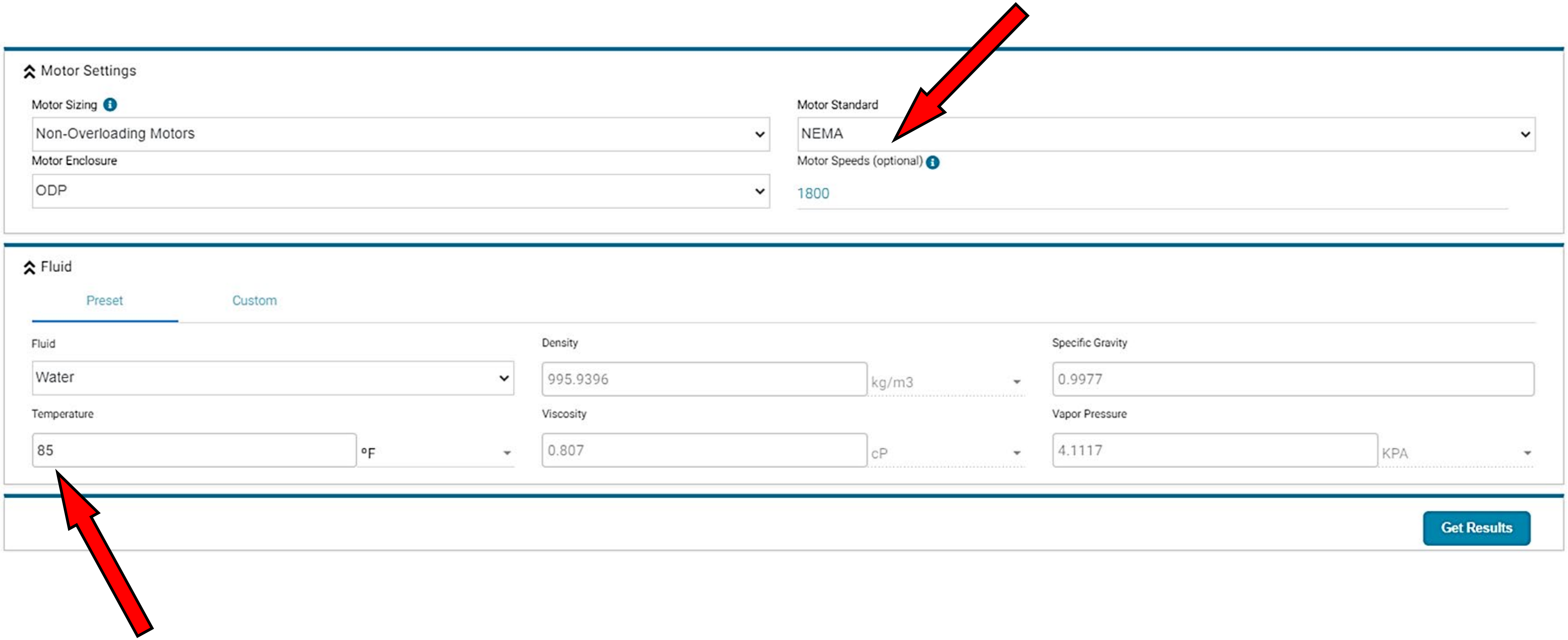

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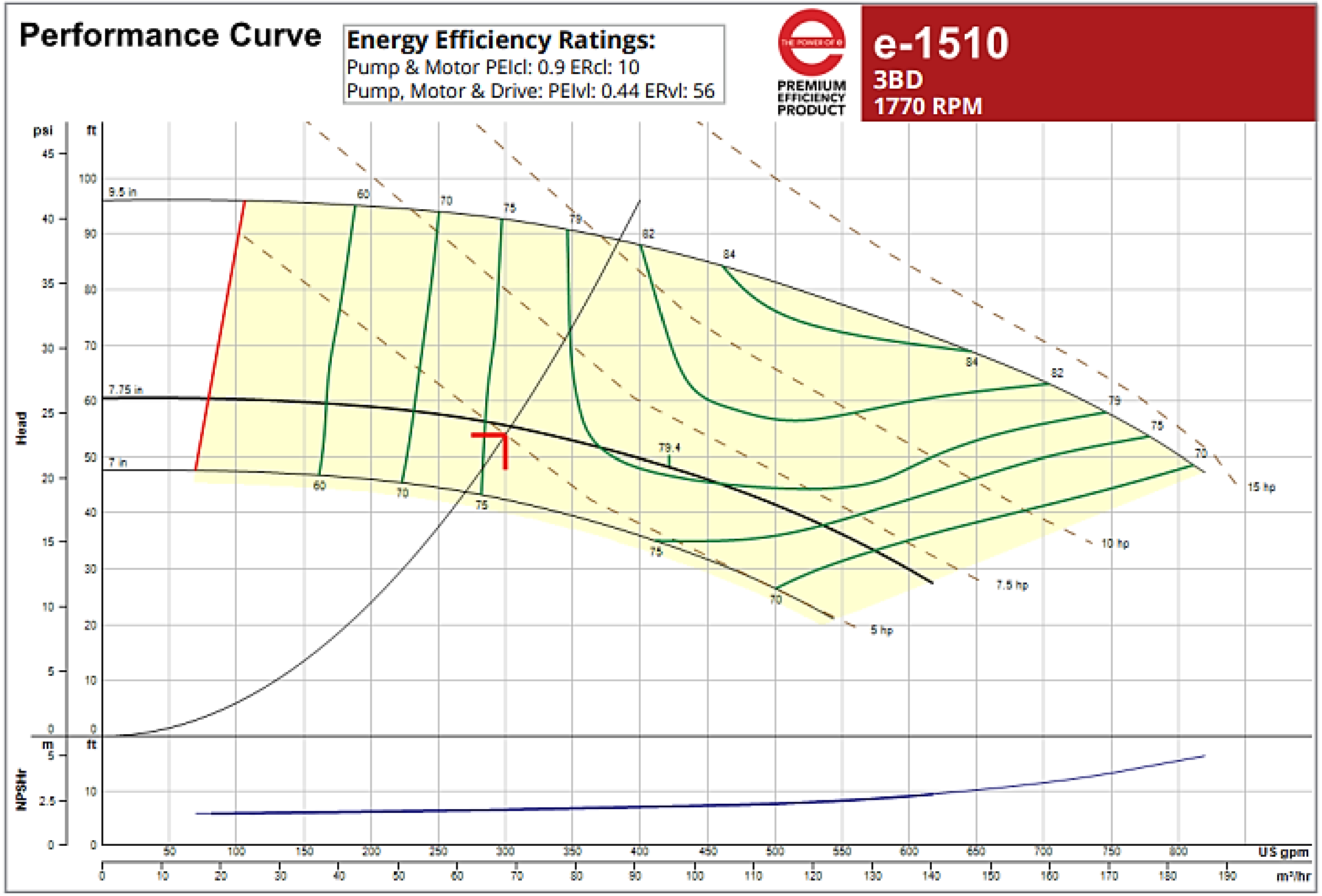

Performance curve meets 14.6 / ISO 9906 acceptance criteria | Available Phase: 1, 3, Available Voltage(s) [V]: 230, 230/460, 575, 200

Pump Selection Summary

Duty Point Flow	300 US gpm
Duty Point Head	54 ft
Control Head	0 ft
Duty Point Pump Efficiency	75.9 %
Part Load Efficiency Value (PLEV)	0.0 %
Impeller Diameter	7.75 in
Motor Power	7.5 hp
Duty Point Power	5.55 bhp
Motor Speed	1800 rpm
RPM @ Duty Point	1770 rpm
NPSHr	6.55 ft
Minimum Shutoff Head	60.6 ft
Minimum Flow at RPM	84.3 US gpm
Flow @ BEP	421 US gpm
Fluid Temperature	85 °F
Fluid Type	Water
Weight *(approx. - consult rep for exact)*	377 lbs
Pump Floor Space Calculation	4.75 ft²

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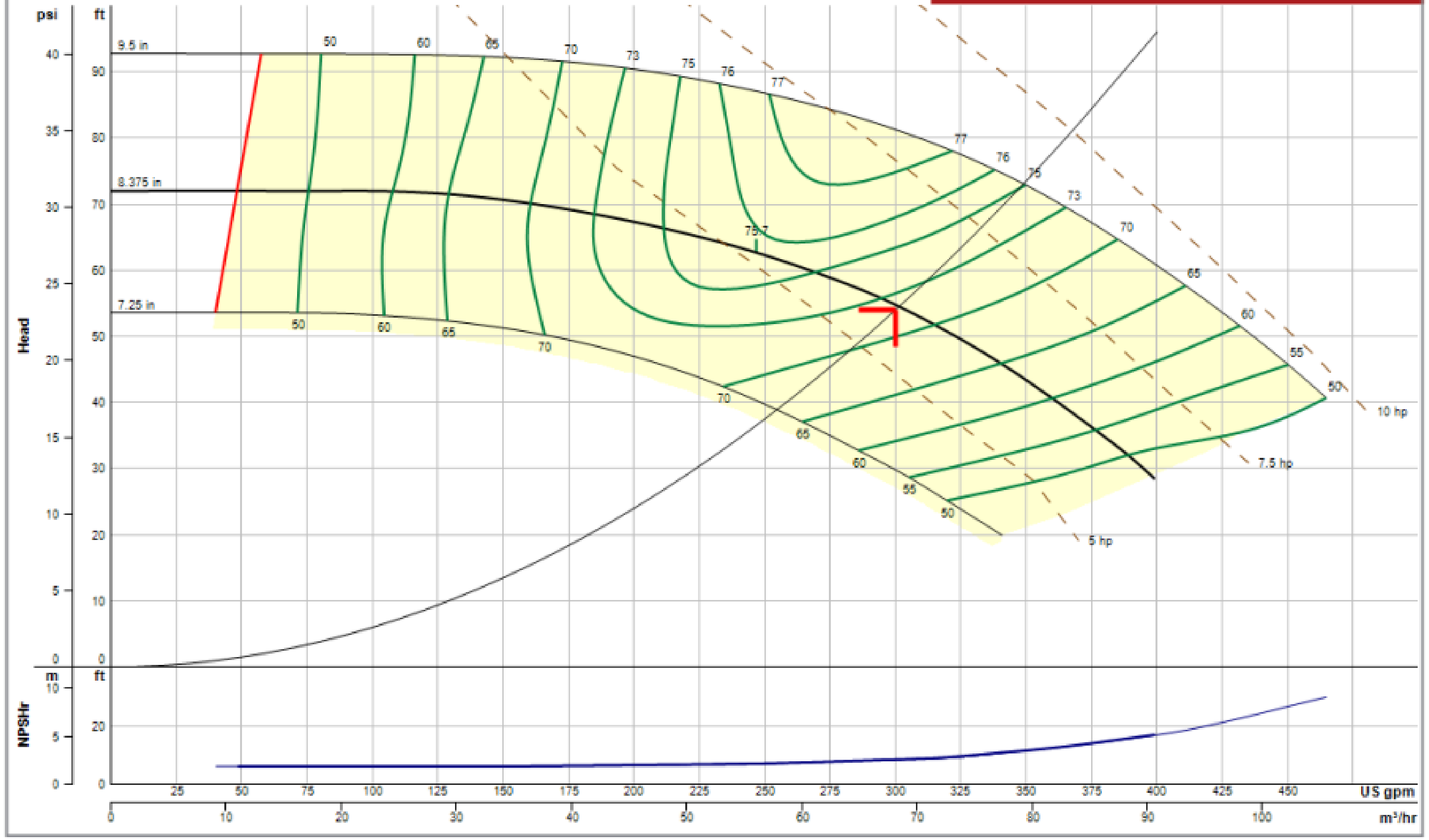
Performance Curve

Energy Efficiency Ratings:
Pump & Motor PEIcl: 0.89 ERcl: 11
Pump, Motor & Drive: PEIvl: 0.45 ERvl: 55

PREMIUM EFFICIENCY PRODUCT

e-1510
2.5BB
1750 RPM



Performance curve meets 14.6 / ISO 9906 acceptance criteria | Available Phase: 1, 3, Available Voltage(s) [V]: 230, 230/460, 575, 200

WIZE-D432D189

Pump Selection Summary

Duty Point Flow	300 US gpm
Duty Point Head	54 ft
Control Head	0 ft
Duty Point Pump Efficiency	72 %
Part Load Efficiency Value (PLEV)	0.0 %
Impeller Diameter	8.375 in
Motor Power	7.5 hp
Duty Point Power	5.73 bhp
Motor Speed	1800 rpm
RPM @ Duty Point	1750 rpm
NPSHr	8.52 ft
Minimum Shutoff Head	72 ft
Minimum Flow at RPM	49.4 US gpm
Flow @ BEP	247 US gpm
Fluid Temperature	85 °F
Fluid Type	Water
Weight *(approx. - consult rep for exact)*	367 lbs
Pump Floor Space Calculation	4.35 ft²

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Performance Curve

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Pump & Motor PEIcl: 0.89 ERcl: 11
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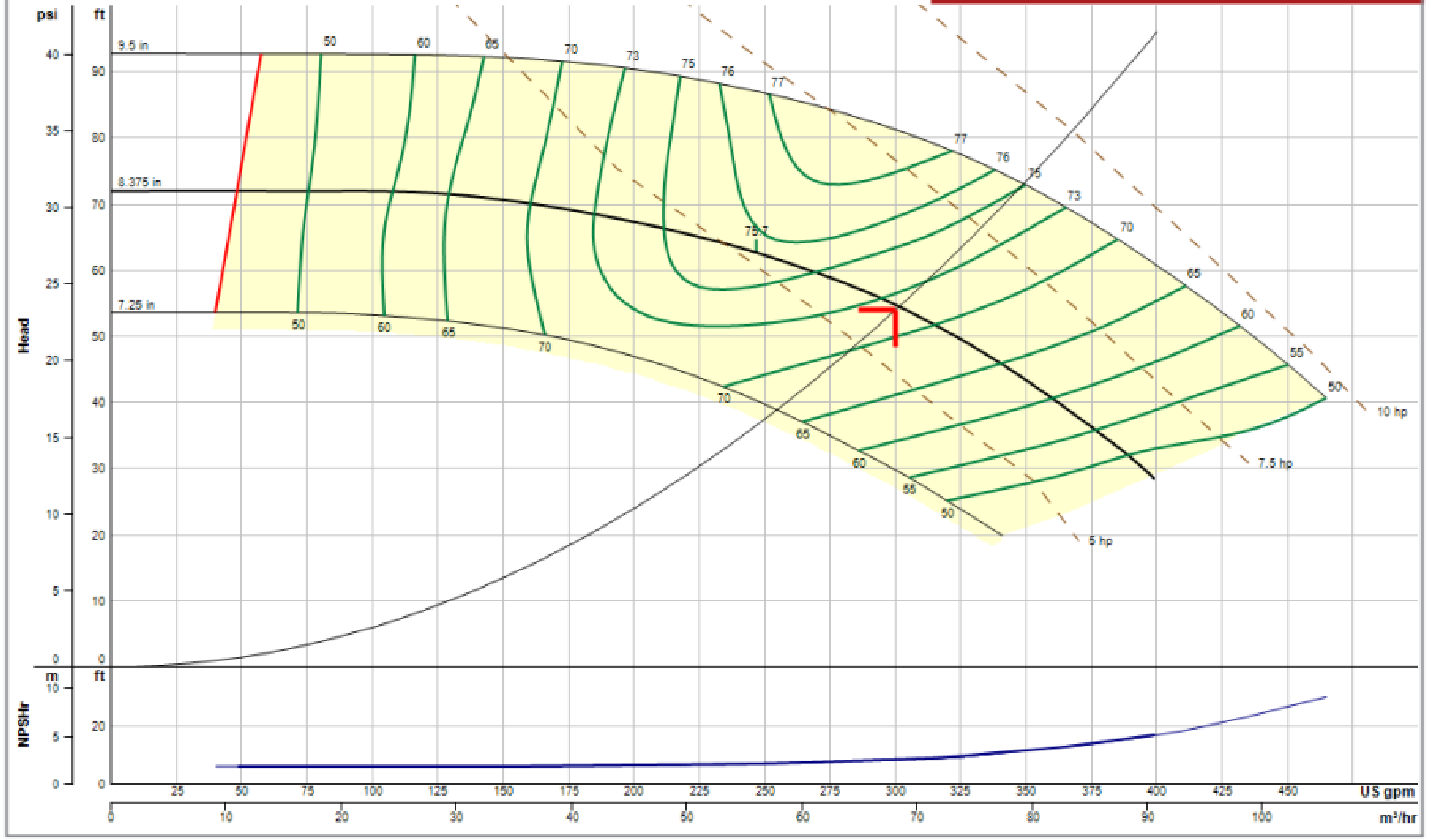

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Why does smaller pump have higher duty point bhp?

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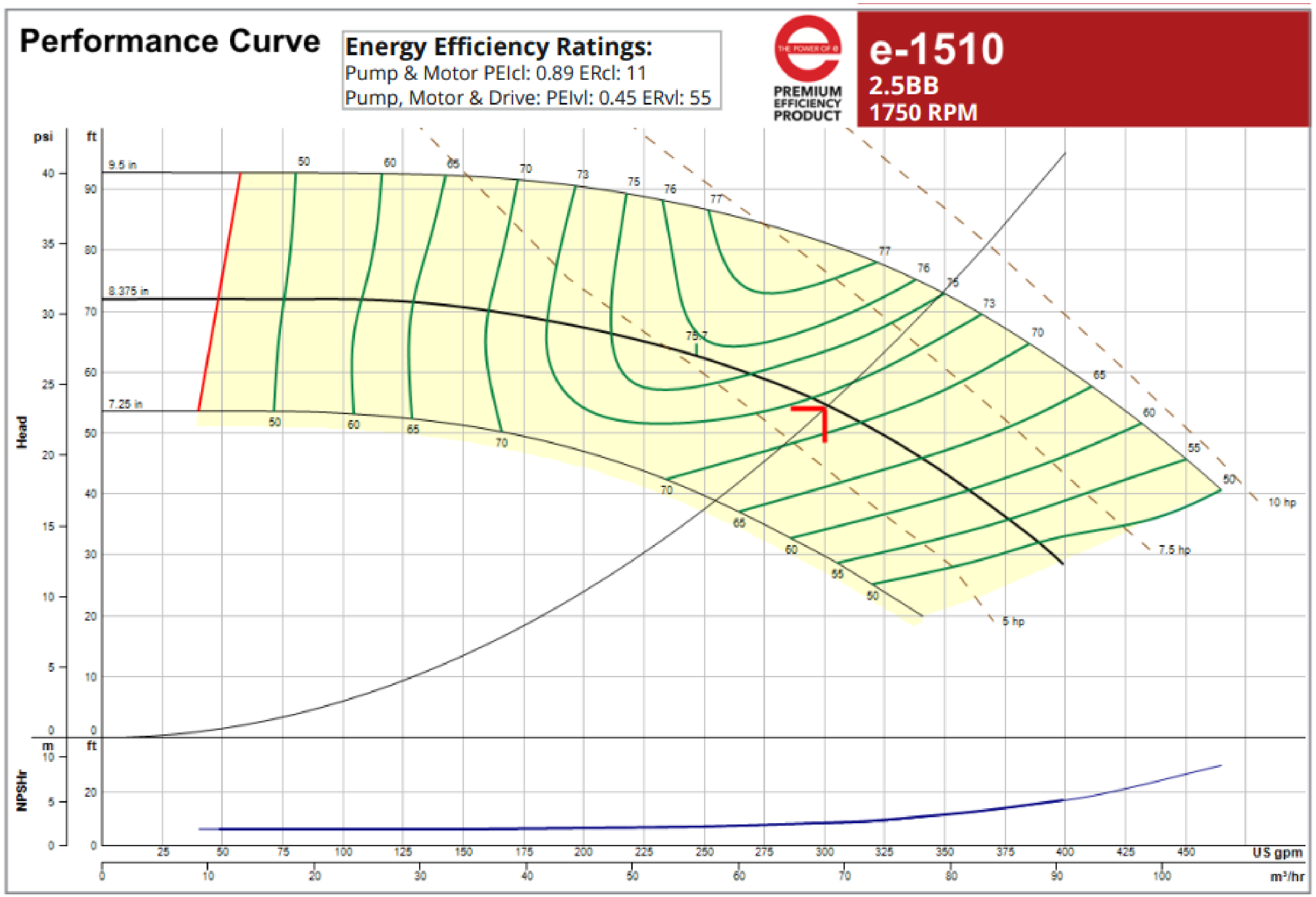

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Why does smaller pump have higher duty point bhp?

Why does smaller pump have higher required NPSH?

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$$NPSHA = \frac{2.31(P_a - P_v)}{spgr} + (H_e - H_f)$$

P_a = 14.7 Psia

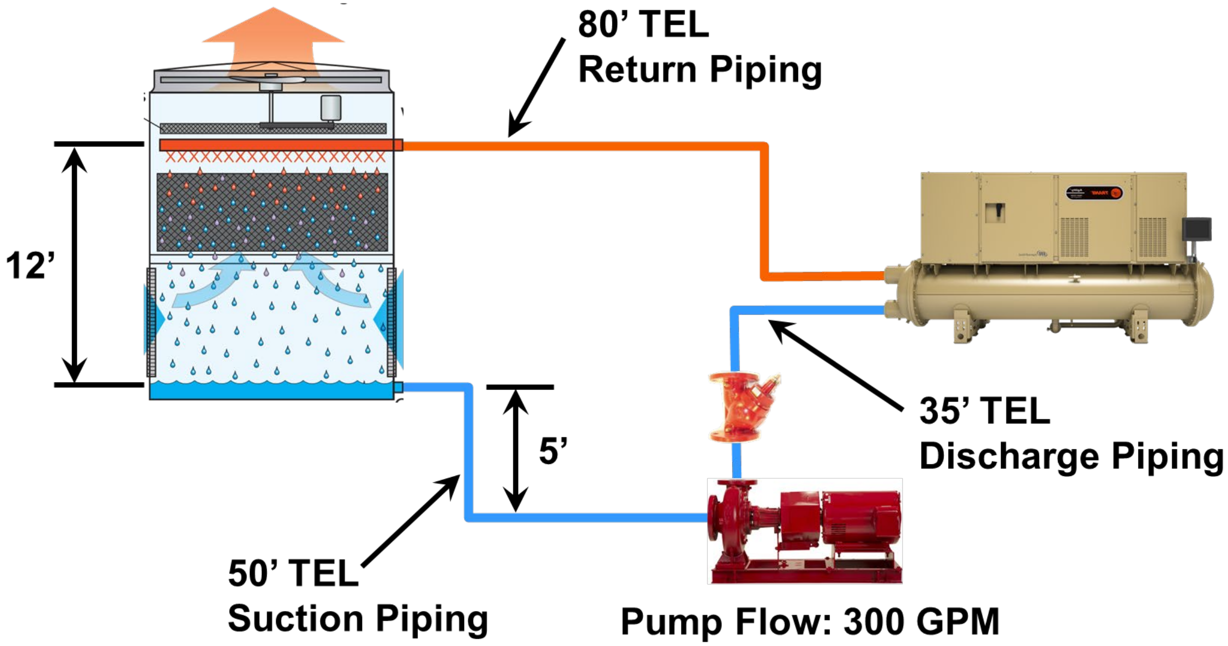
P_v = 0.597 Psia (@ 85°F Water)

sp.gr. = 0.9977 ≈ 1.0

FLR: 4.75′/100′ (Water @ 85°F)

H_e = 5′

H_f = 50′/100 x 4.75 = 2.4′



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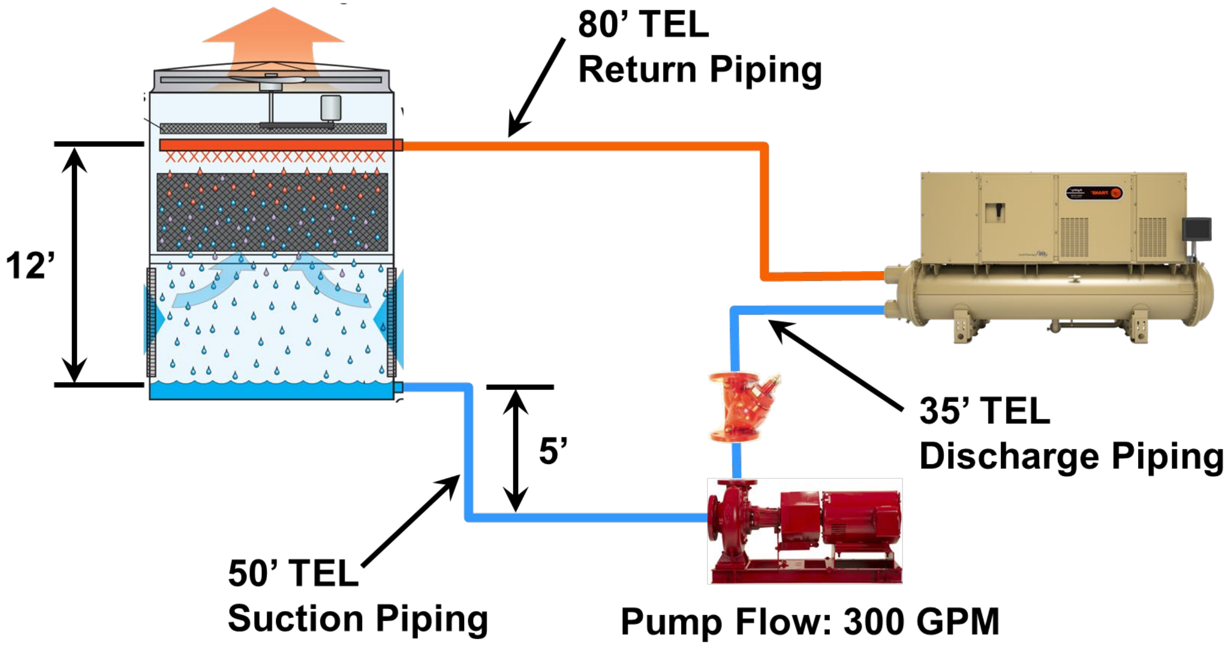

$$NPSHA = \frac{2.31(Pa - Pv)}{sp\ gr} + (He - Hf)$$

$$= \frac{2.31(14.7 - 0.597)}{1.0} + (5 - 2.4)$$

= 35.18 feet

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Understanding Horsepower

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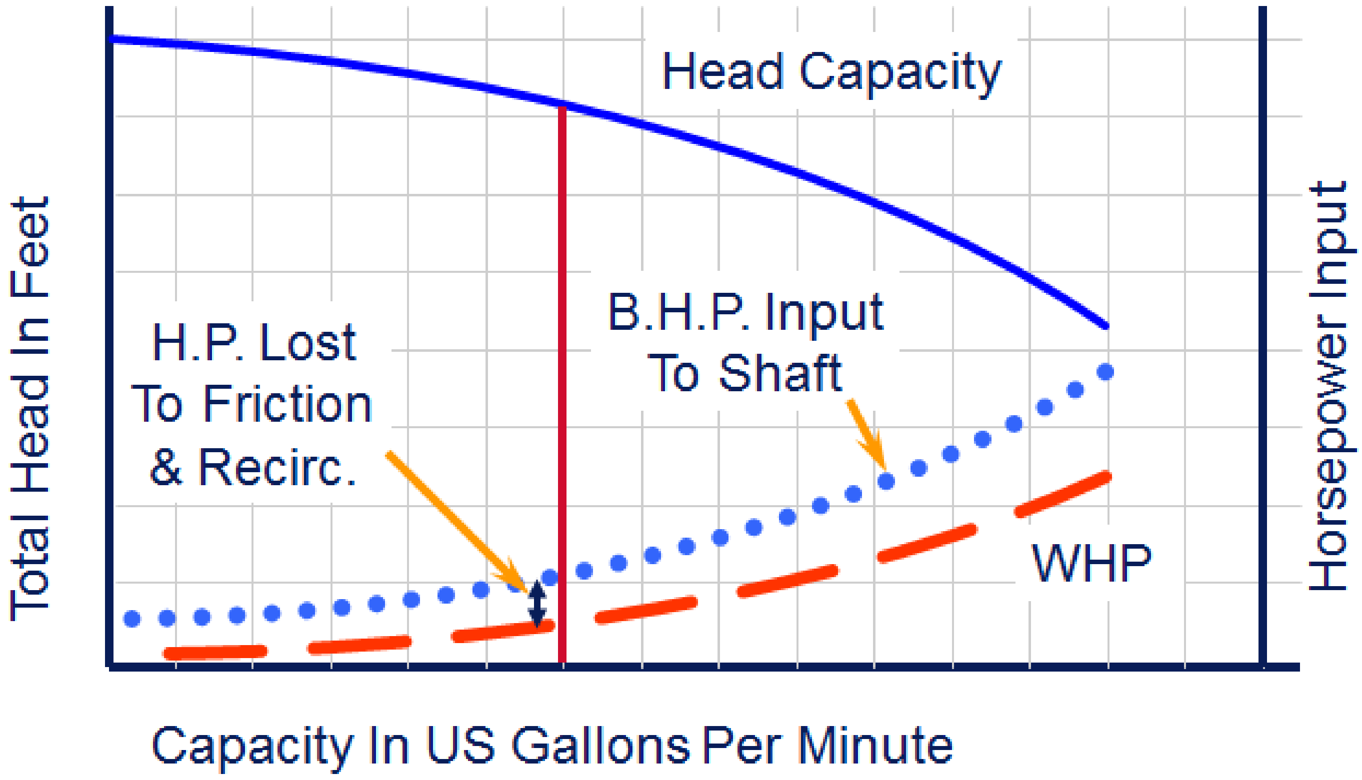
$$WHP = \frac{\text{Flow x Head x Sp. Gr.}}{3960}$$

Buy the pump to provide WHP

$$BHP = \frac{\text{Flow x Head x Sp. Gr.}}{3960 \times \eta_{Pump}}$$

Pay the electric bill for BHP

$$\eta_{Pump} = \frac{WHP}{BHP}$$



3960: Divide the number of ft-lbs for 1 HP (33,000) by the weight of 1 Gal. of water (8.33 Lbs.)

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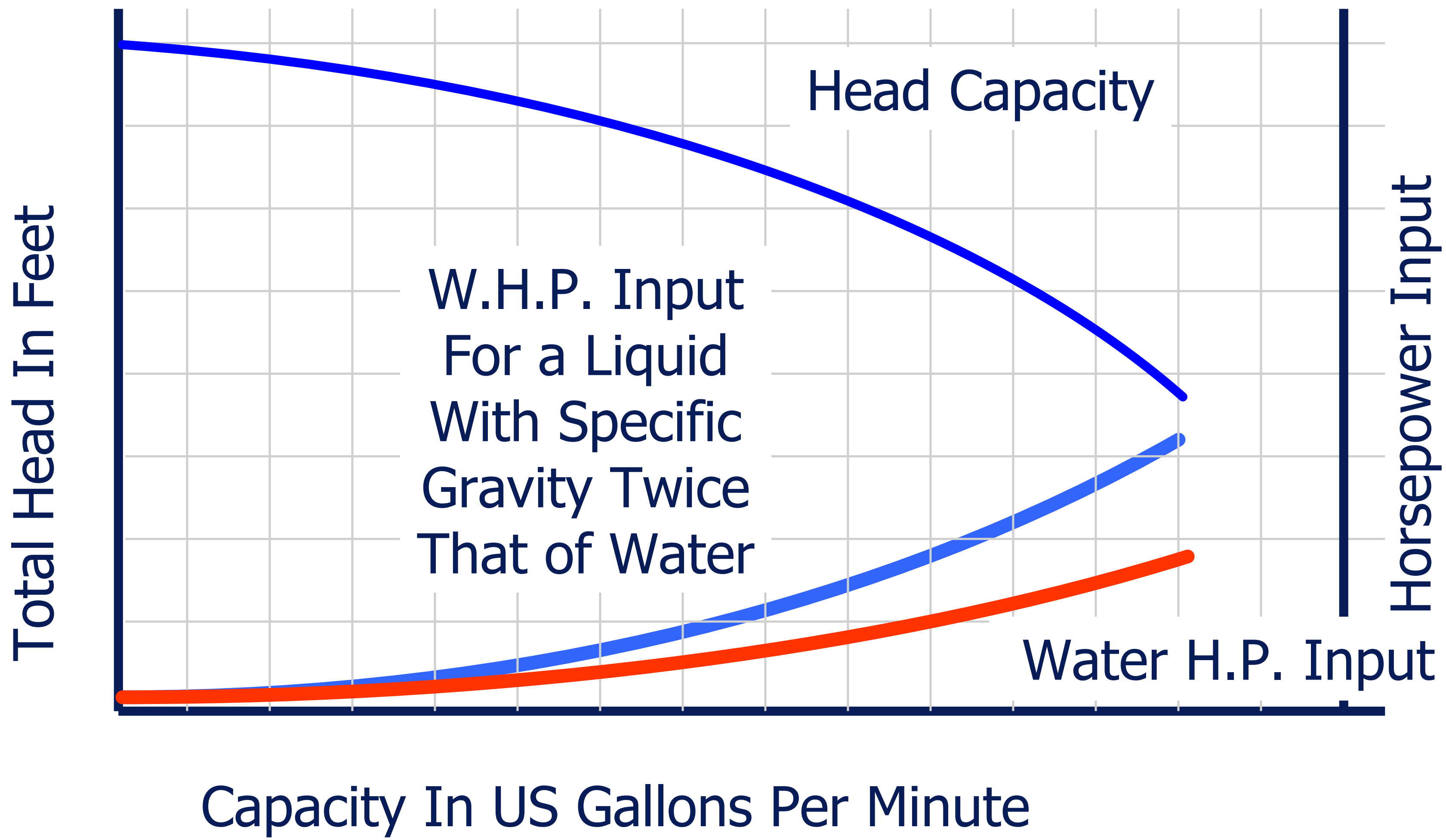

REMINDER!!

$$SG = \frac{\rho \text{ of Fluid}}{\rho \text{ of Water}}$$

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Rotating Speed (RPM):

- **1000** (50 Hz)/**1150** (60Hz): High Flow/Low Head *(low NPSH)*
- **1500** (50 Hz)/**1800** (60Hz): Most common for all applications
- **3000** (50 Hz)/**3600** (60Hz): Low Flow/High Head

Enclosure Type:

- ODP: Open Drip-Proof *(When mounted horizontally!)*
- TEFC: Totally Enclosed Fan Cooled
- TEAO: Totally Enclosed Air-Over
- XPRF: Explosion Proof

"Optional" drip cover when mounted Vertically

Image courtesy of WEG

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- Customers must **take or ship** defective motor to Manufacturer Approved Repair Shop
- Use of any **Non-Approved** Repair Shop must be authorized by the motor manufacturer **prior** to work being started
- Problems as a result of incorrect installation, foreign debris, or physical damage to any motor components are **not** covered

Protect Motors from Dust!!

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Introduction to the Pump Affinity Laws

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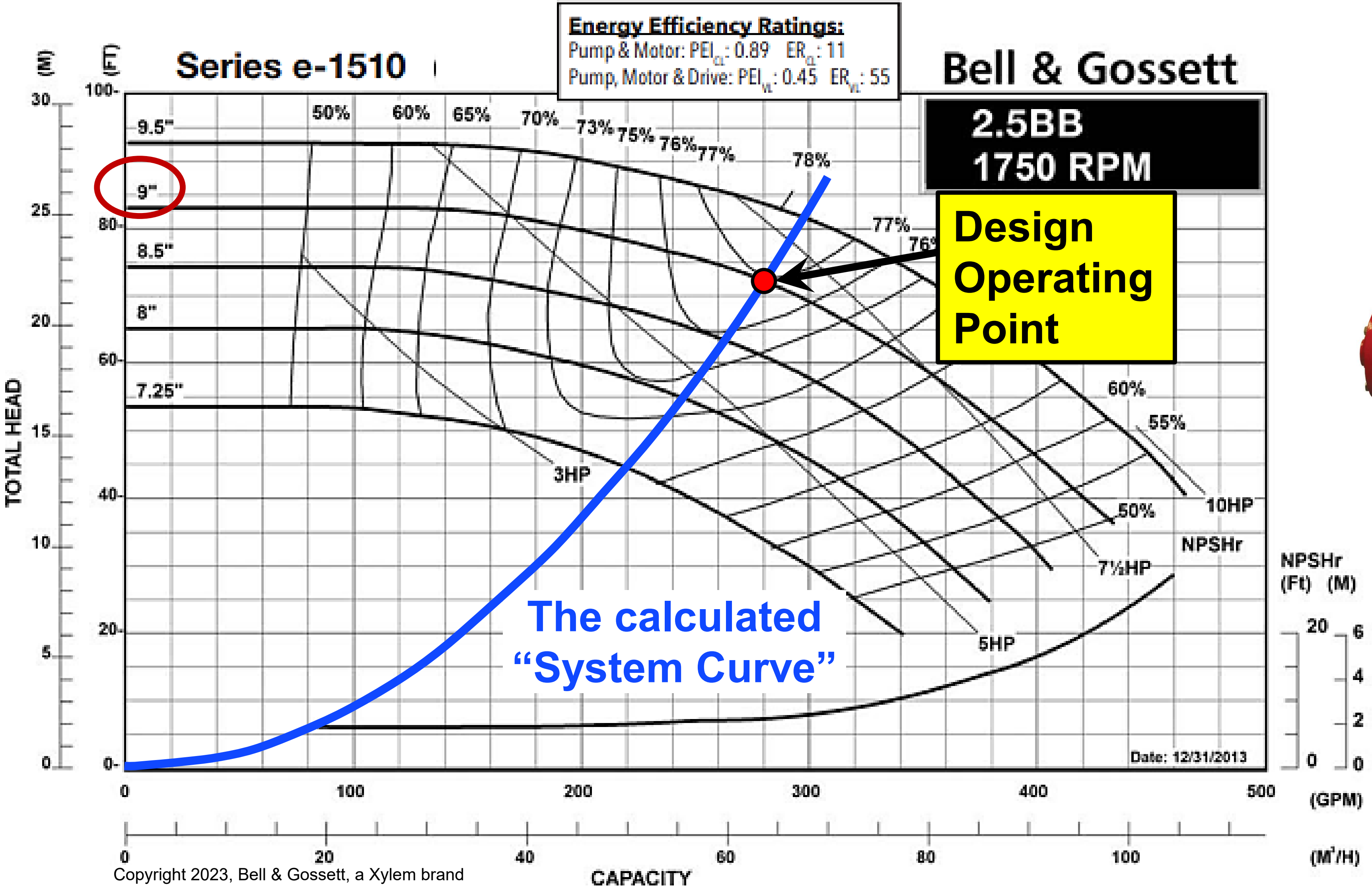
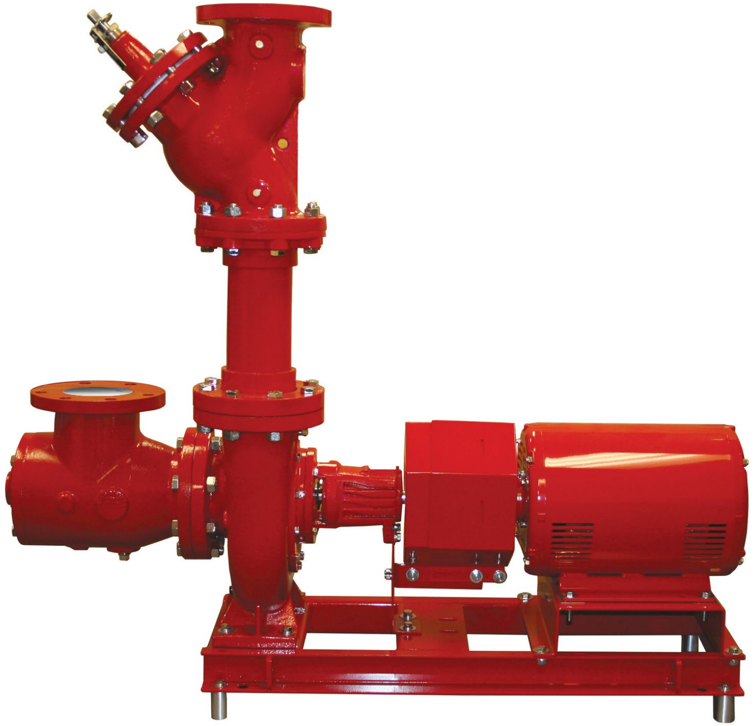
The Law of Conservation of Energy – Part 2





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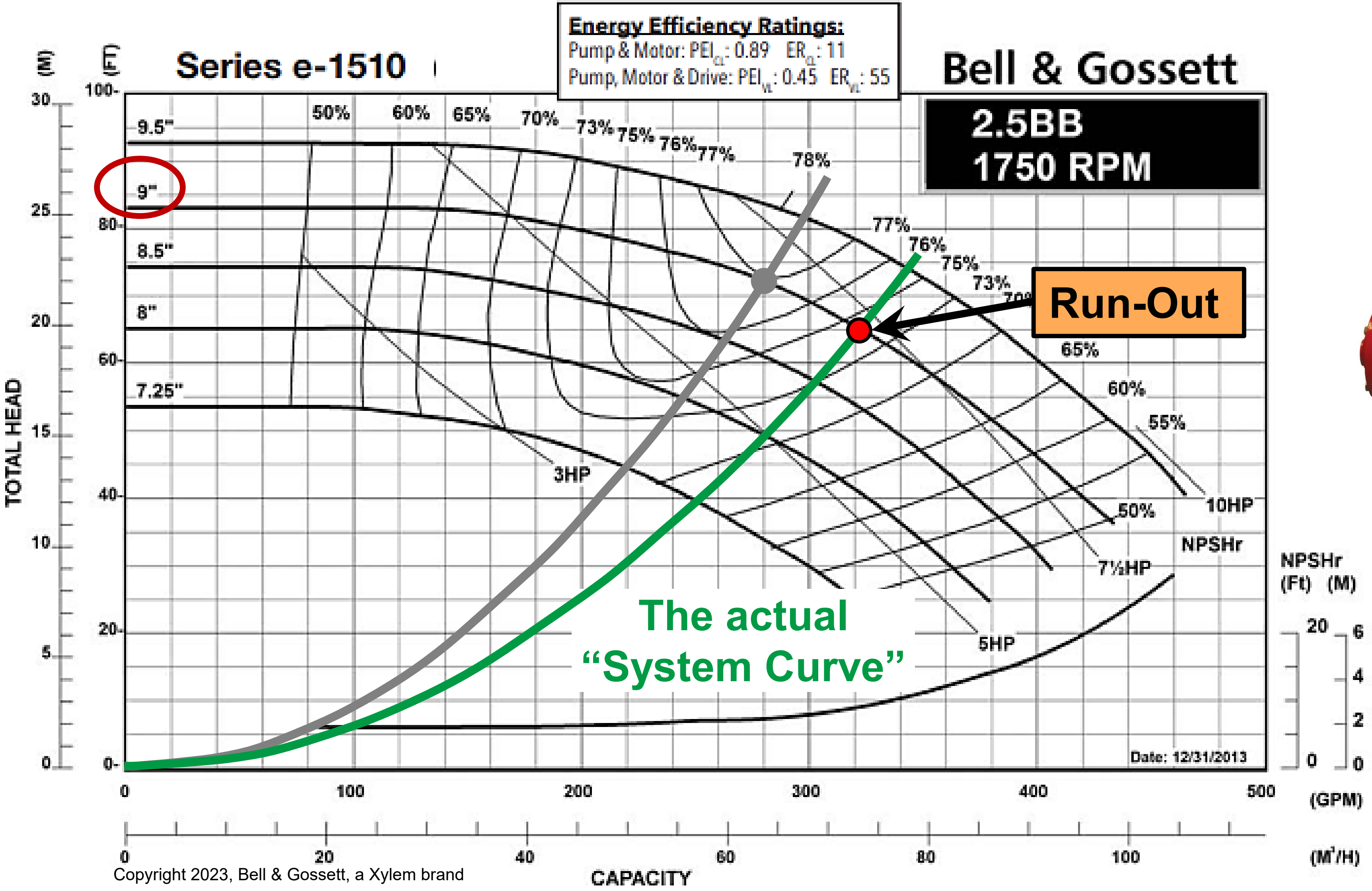
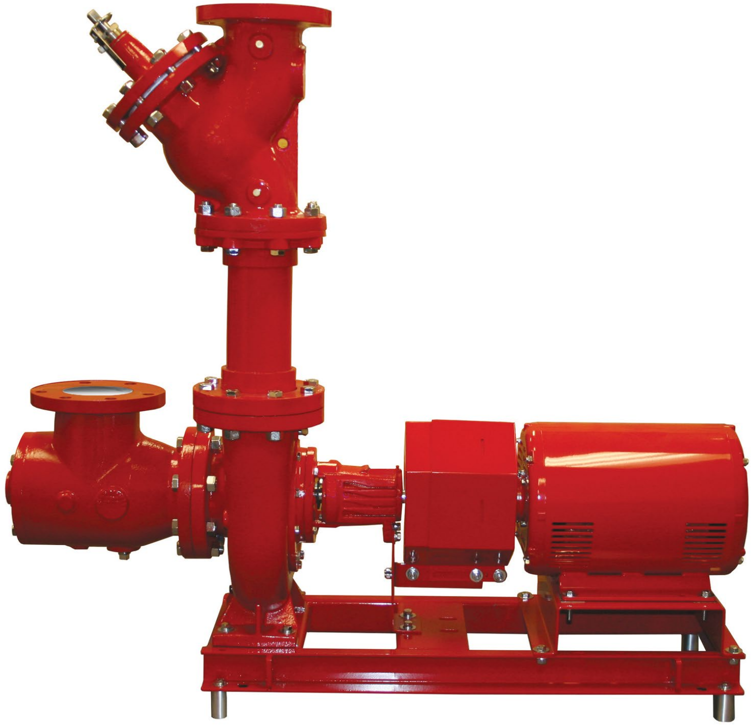
The Law of Conservation of Energy – Part 2





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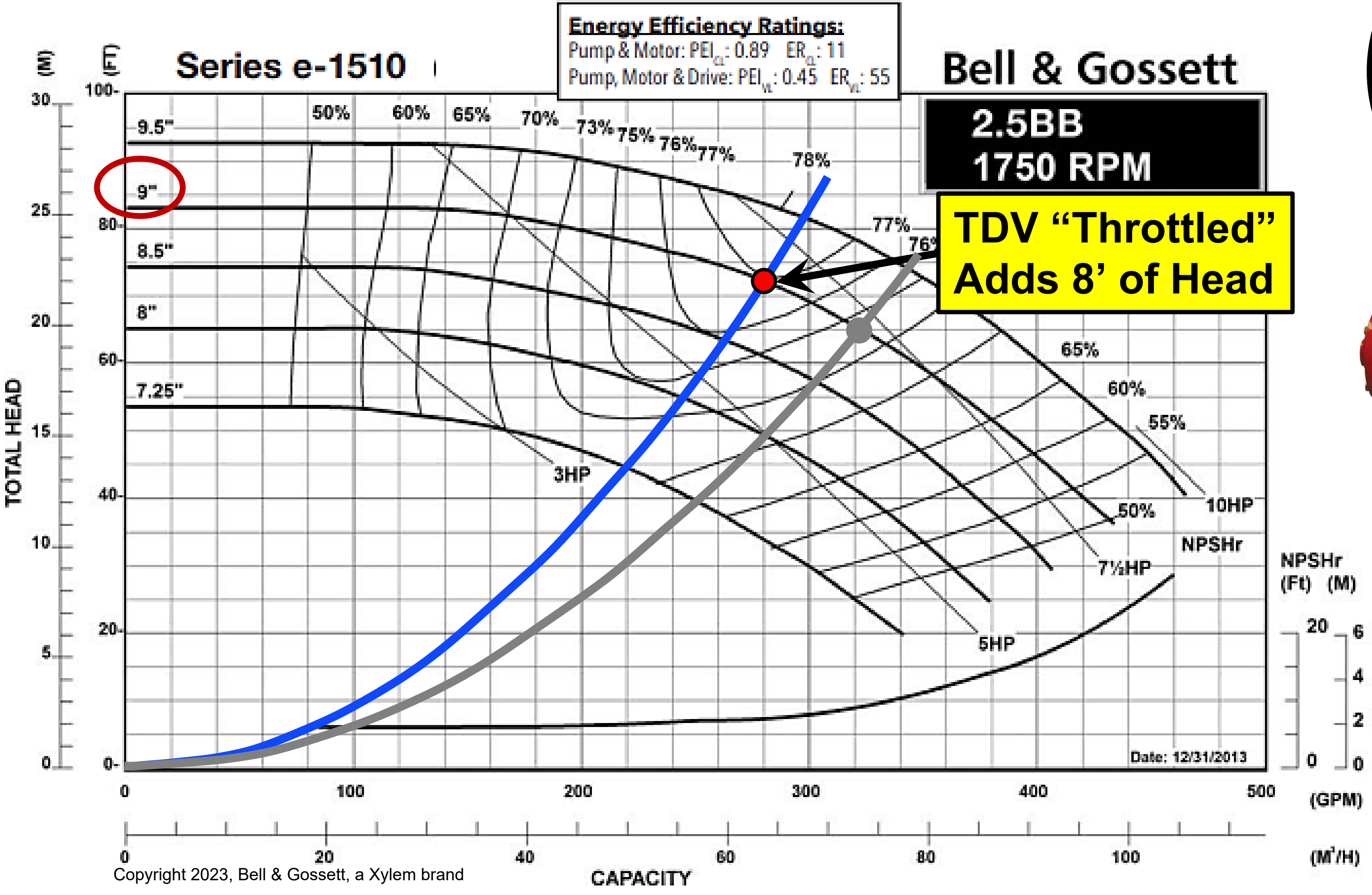

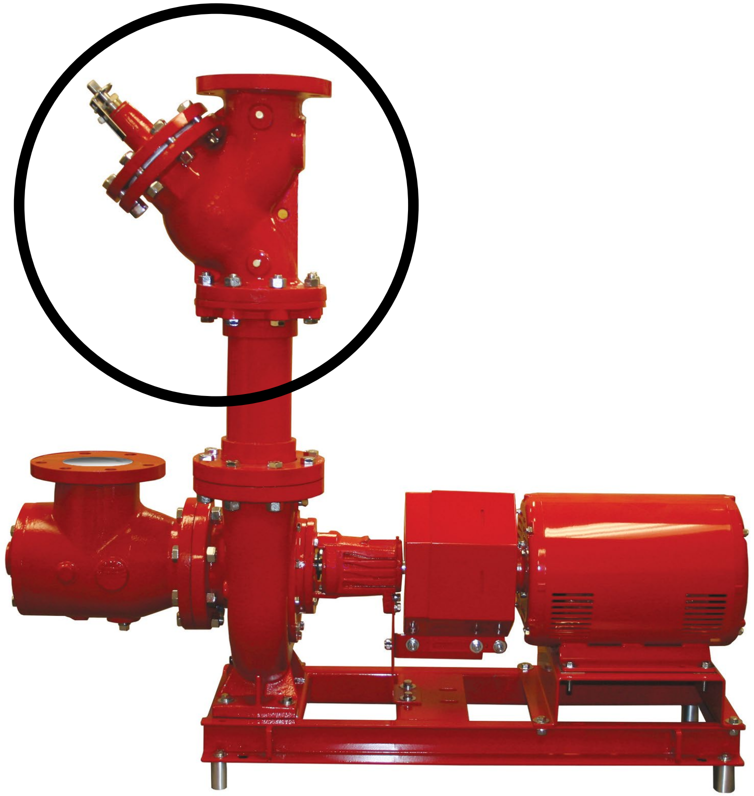

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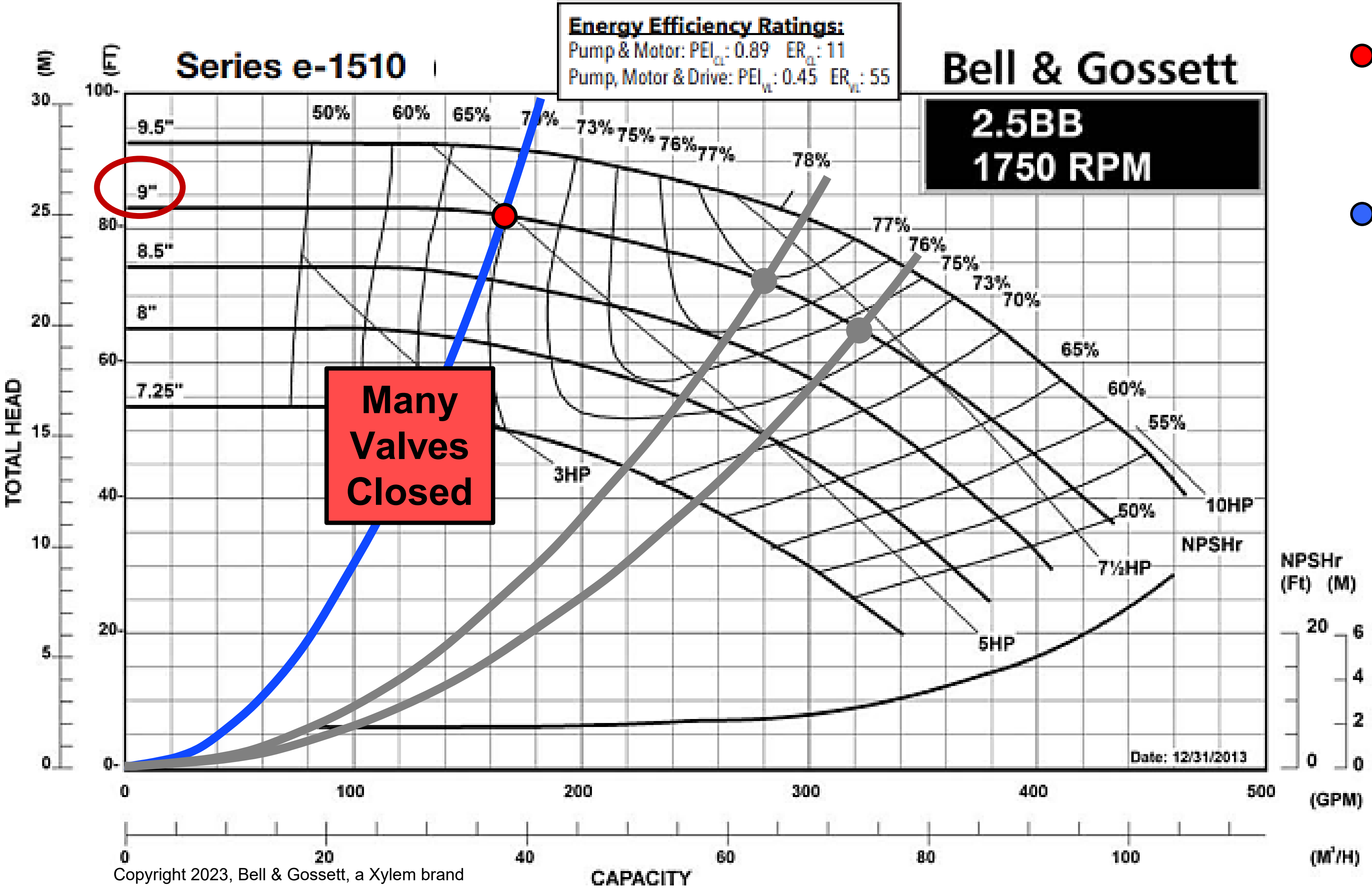

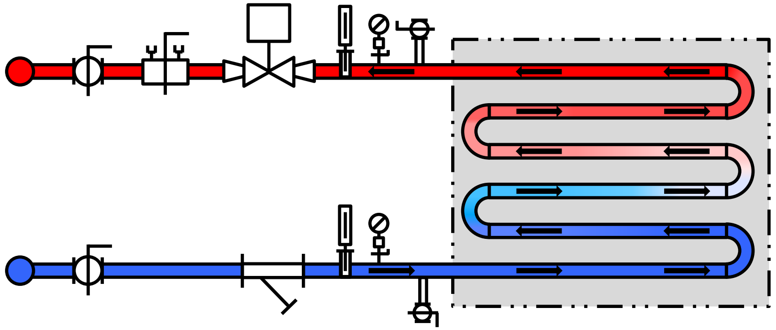

Typical Coil Piping Detail

2-Way Modulating Valve

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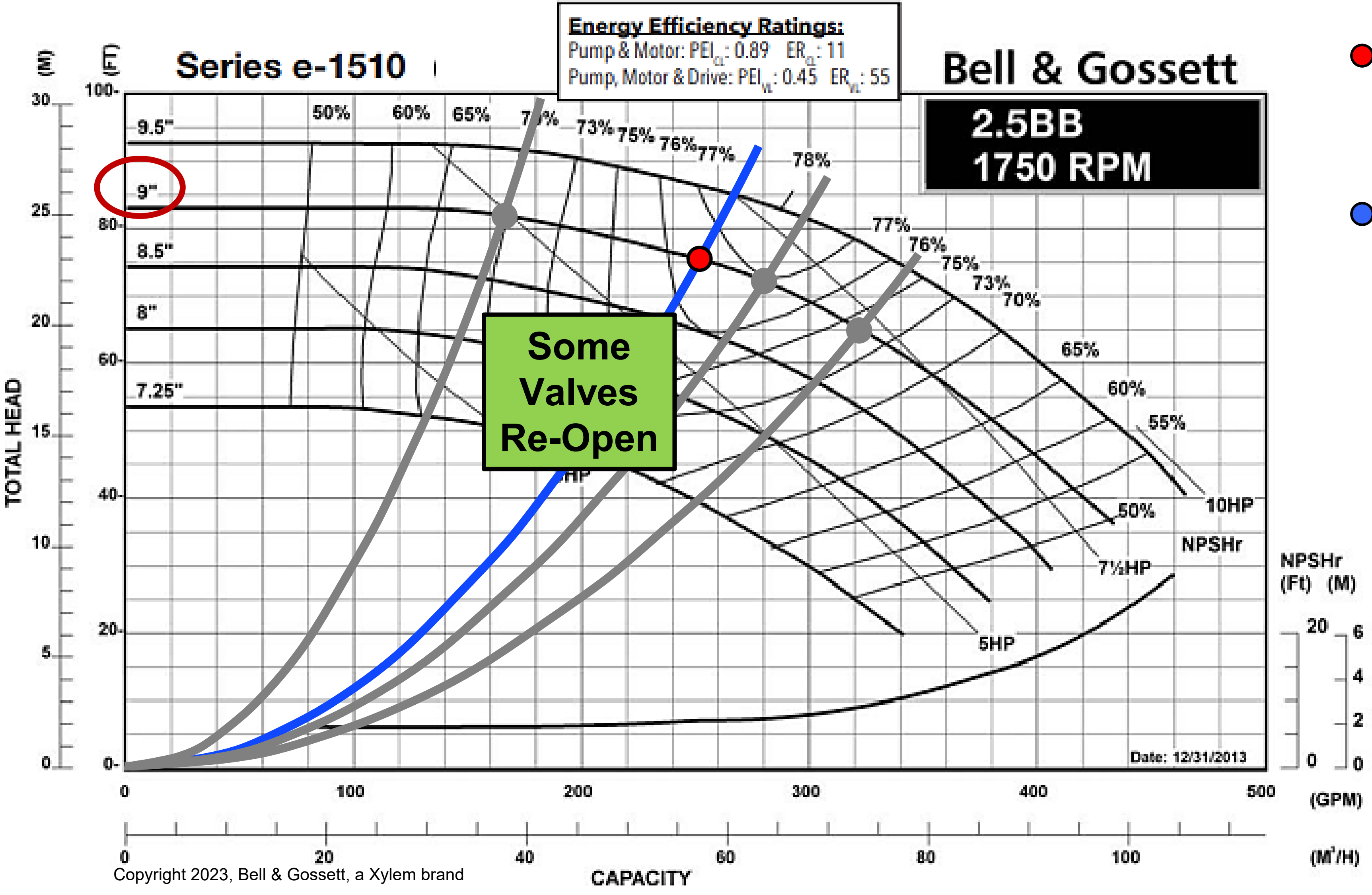
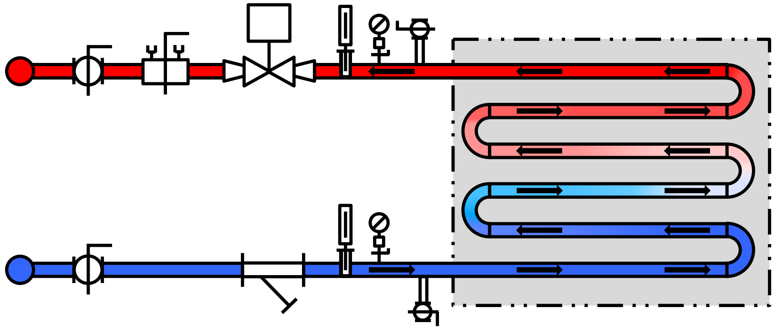
The Law of Conservation of Energy – Part 2





Typical Coil Piping Detail

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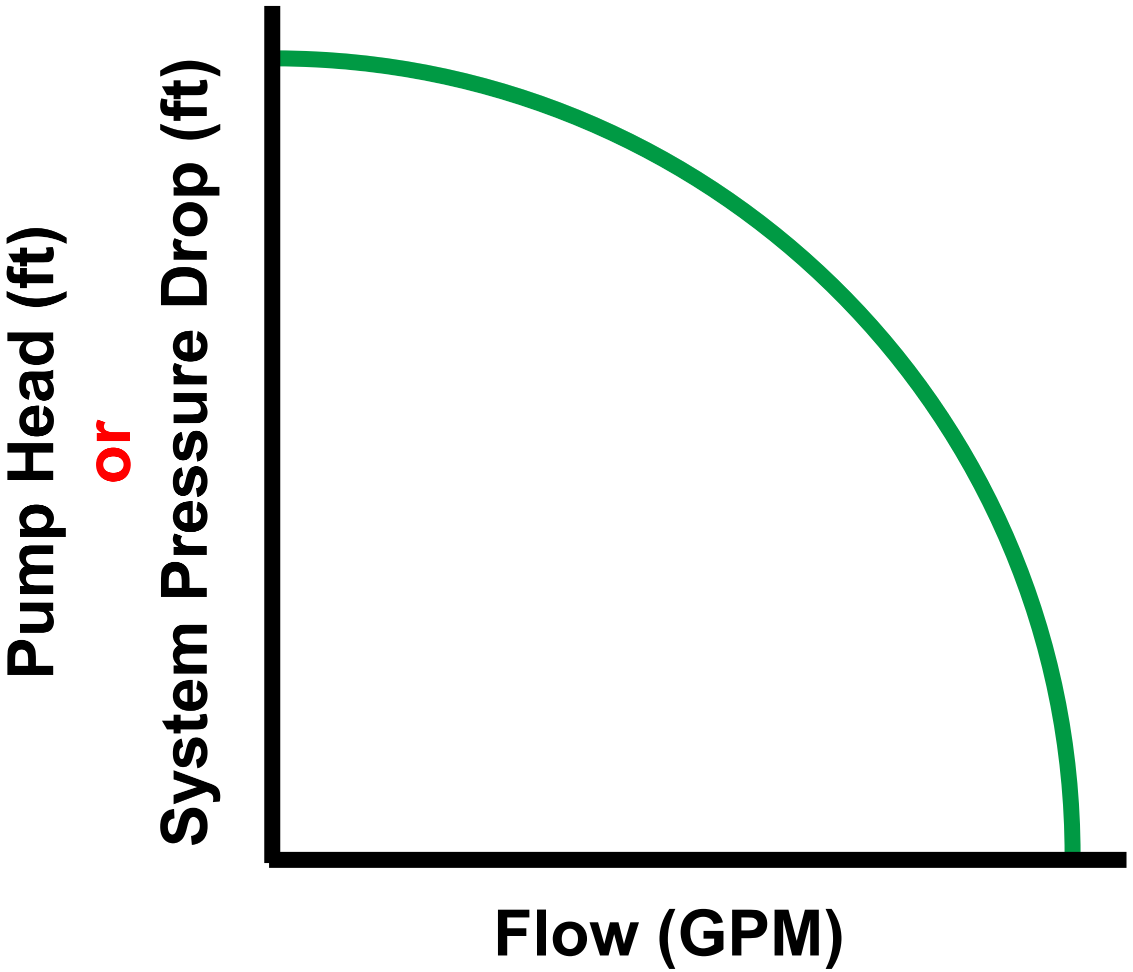

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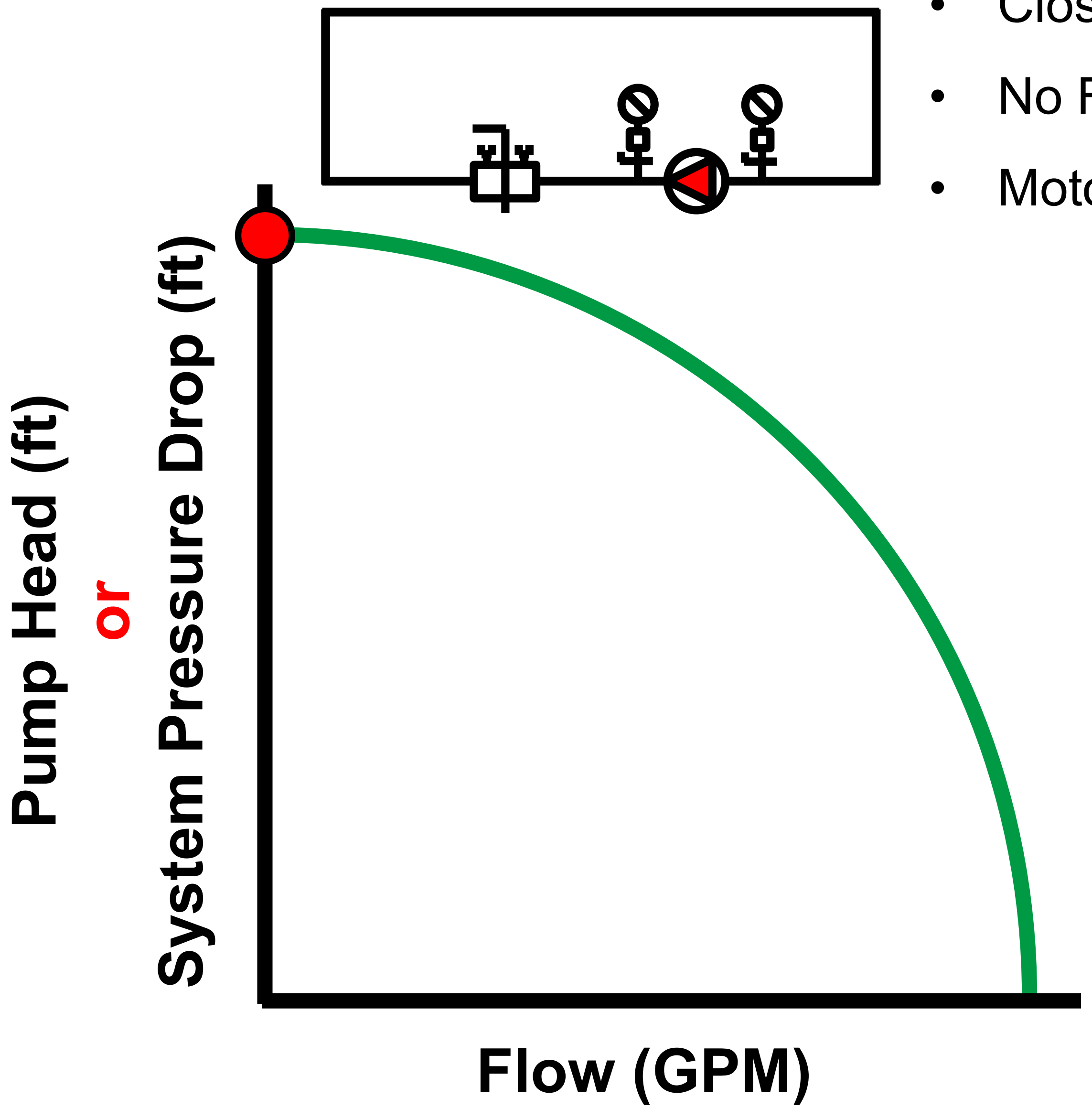

- Closed Valve *(Deadhead)*
- No Flow, Maximum Pump Head
- Motor Lightly Loaded *(Low Amp Draw)*

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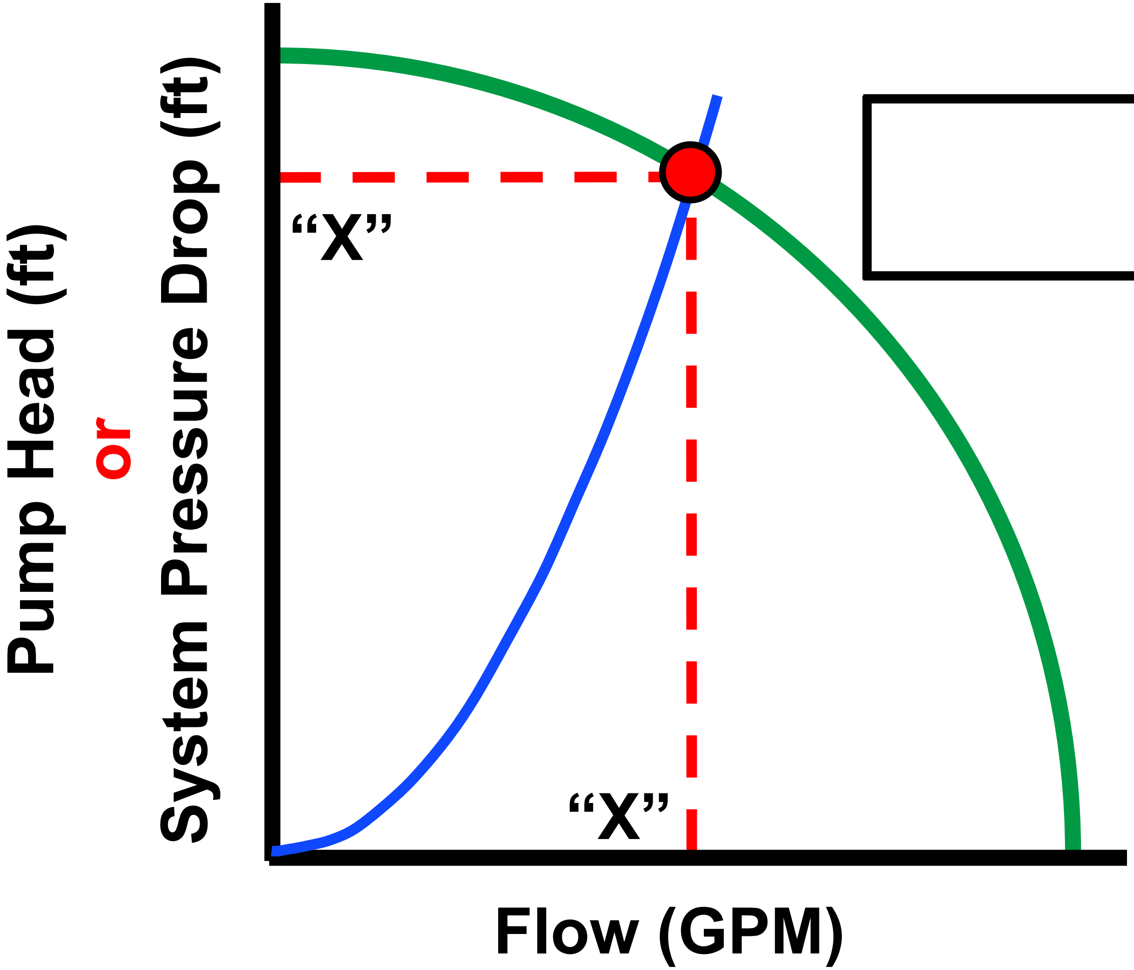

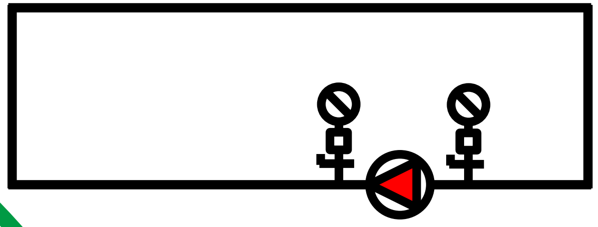

- "X" System Pressure Drop
- "X" Flow
- Increased Motor Load

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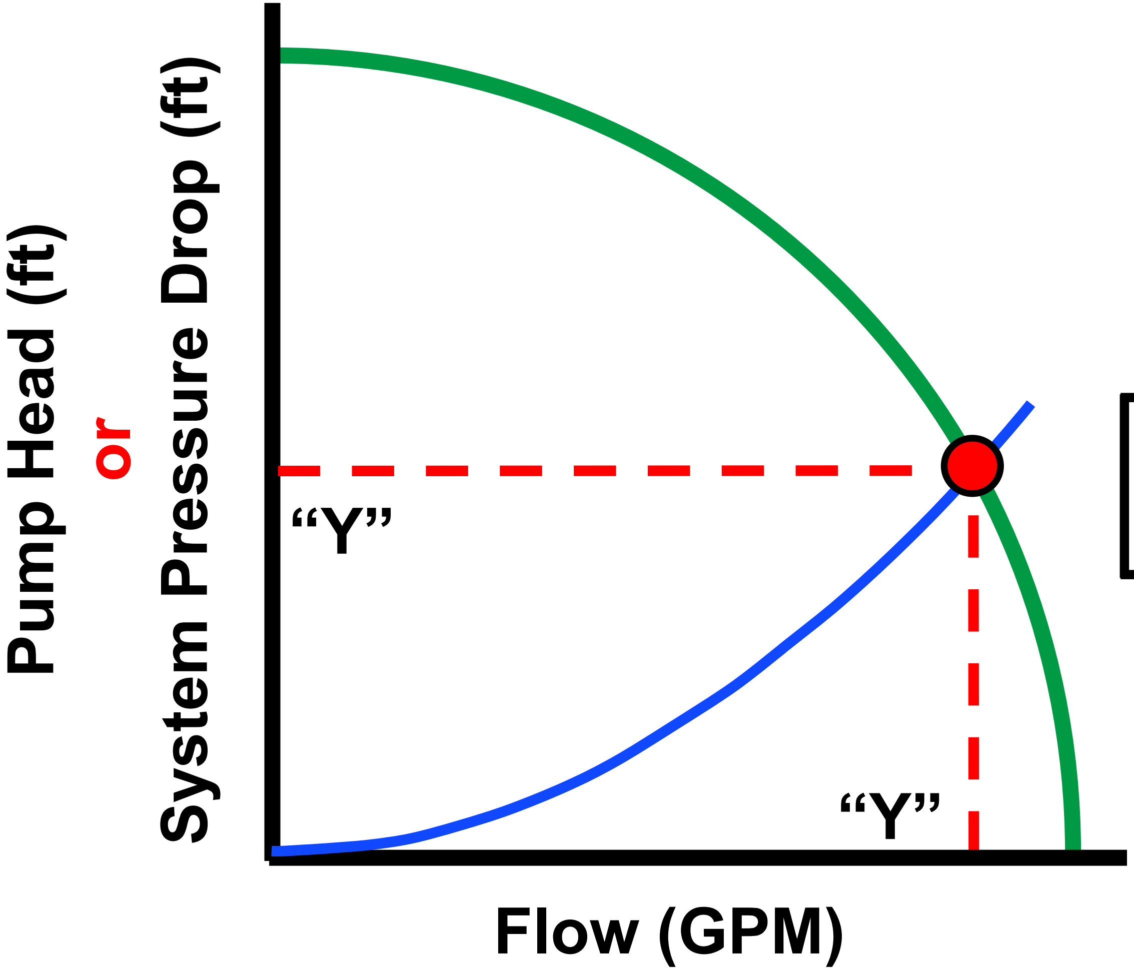

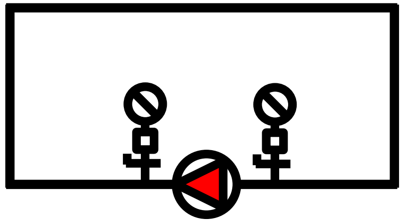

- "Y" System Pressure Drop
- "Y" Flow
- Increased Motor Load

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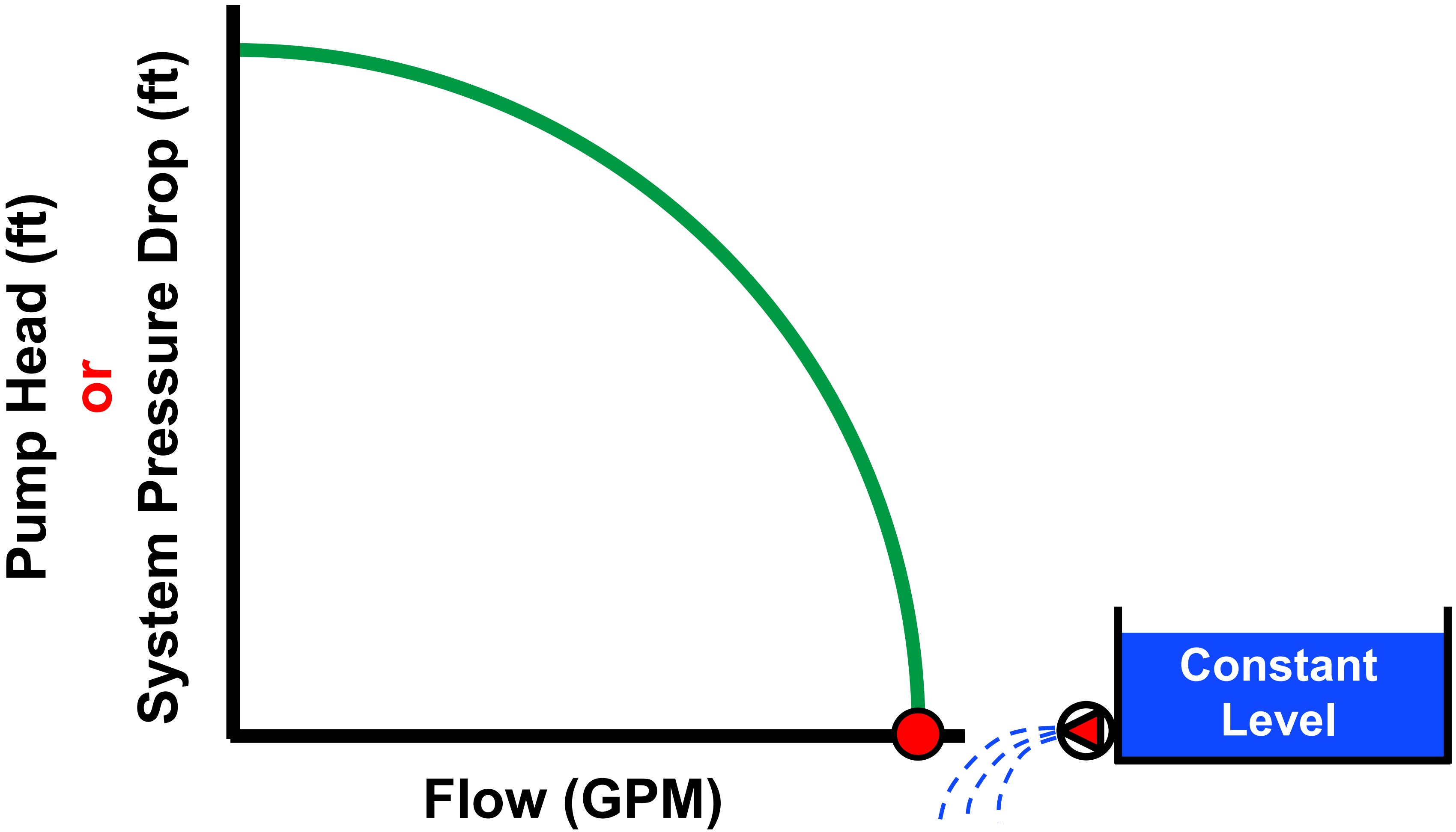

- **No** System Pressure Drop
- Maximum Flow
- Maximum Motor Load

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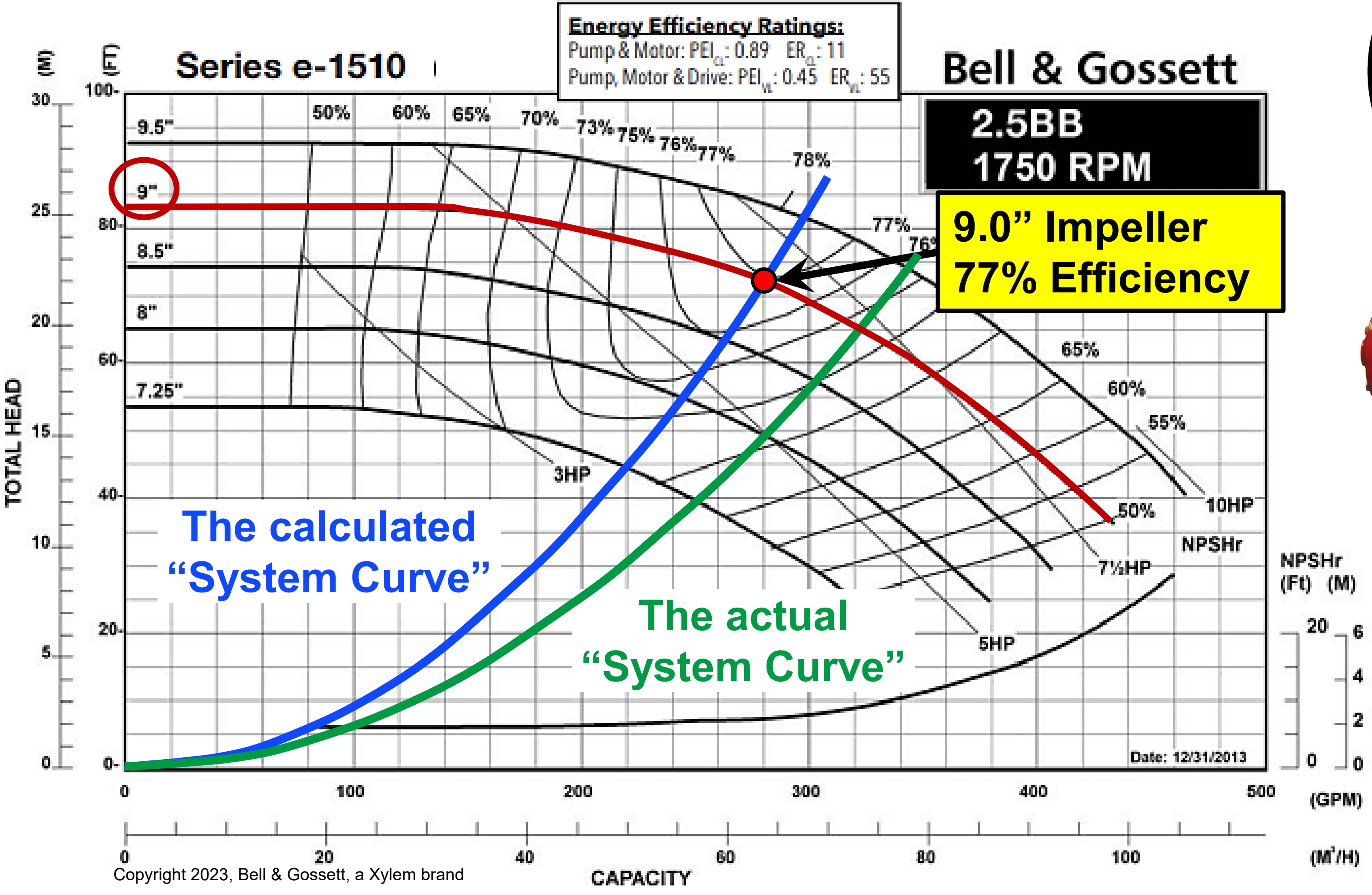
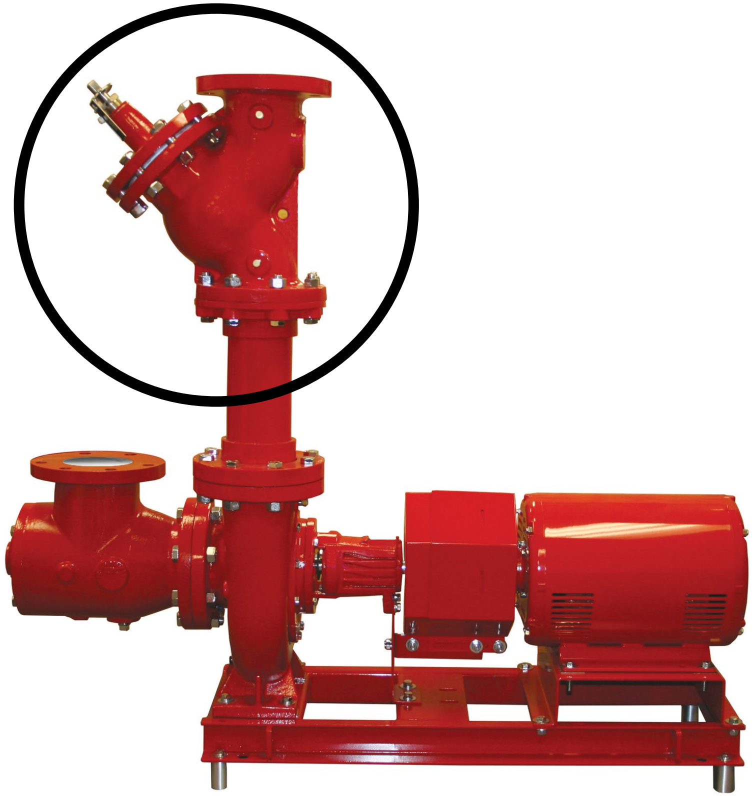
Impeller Trimming: Why and When?





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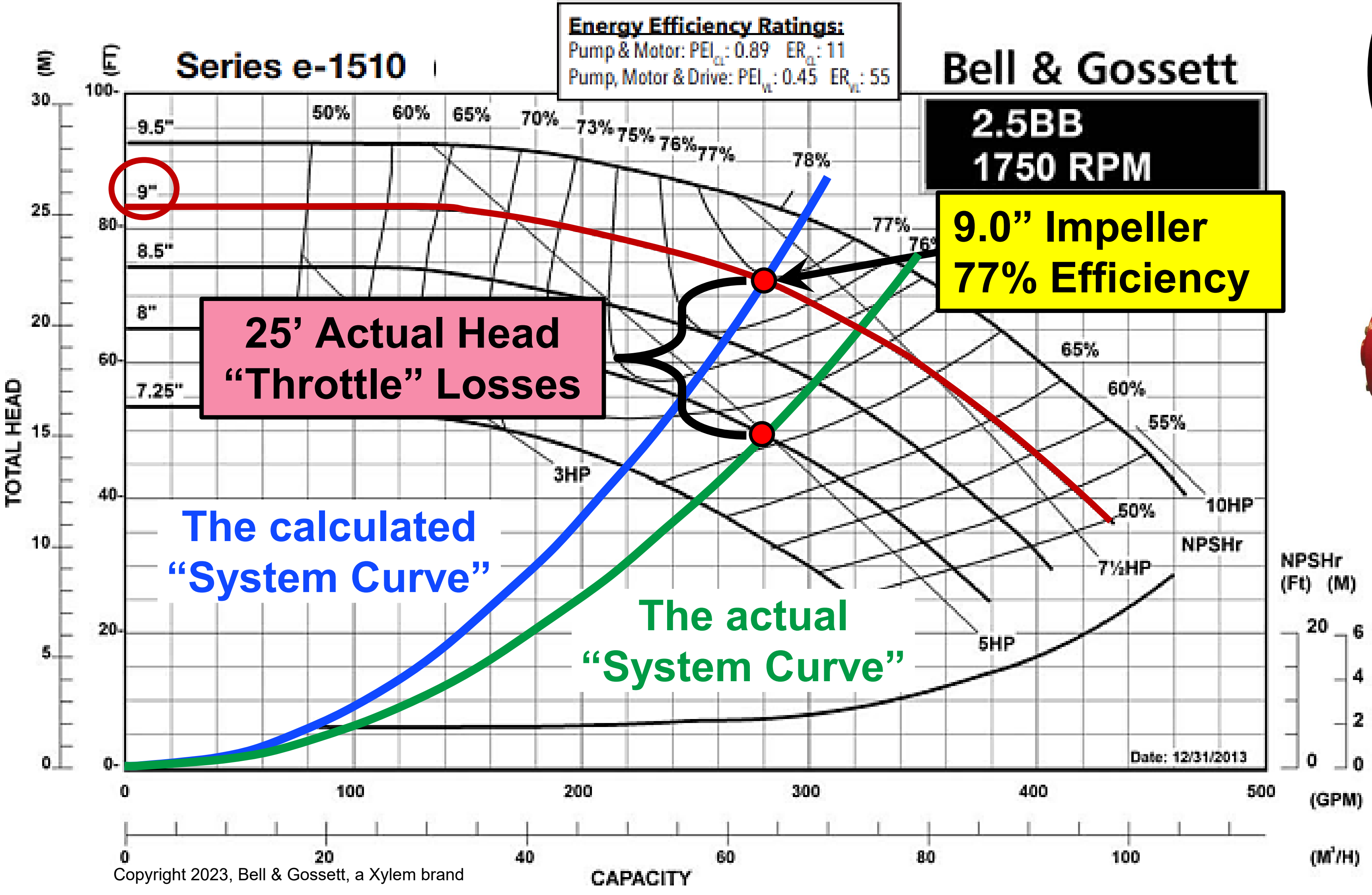
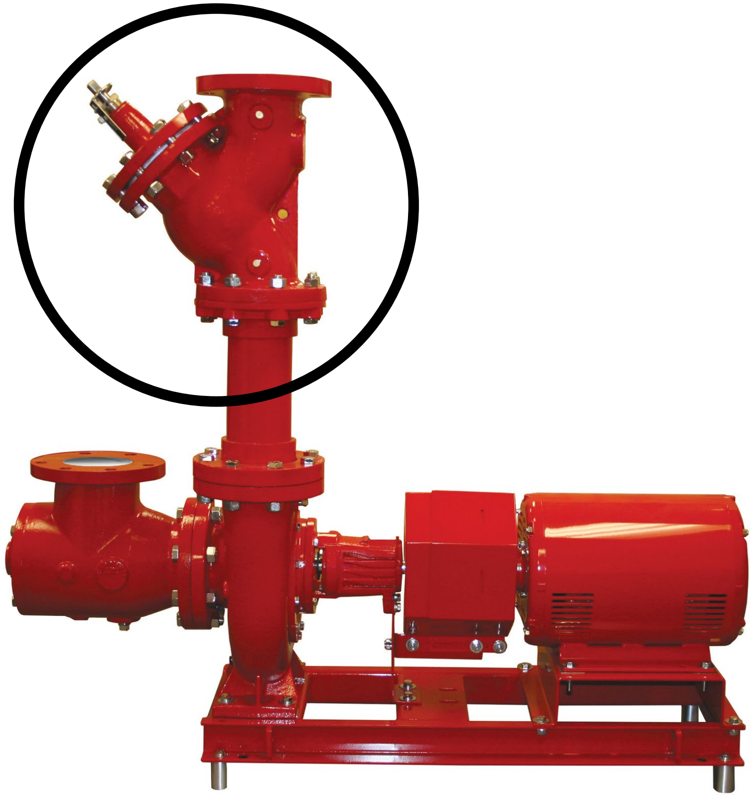
Impeller Trimming: Why and When?





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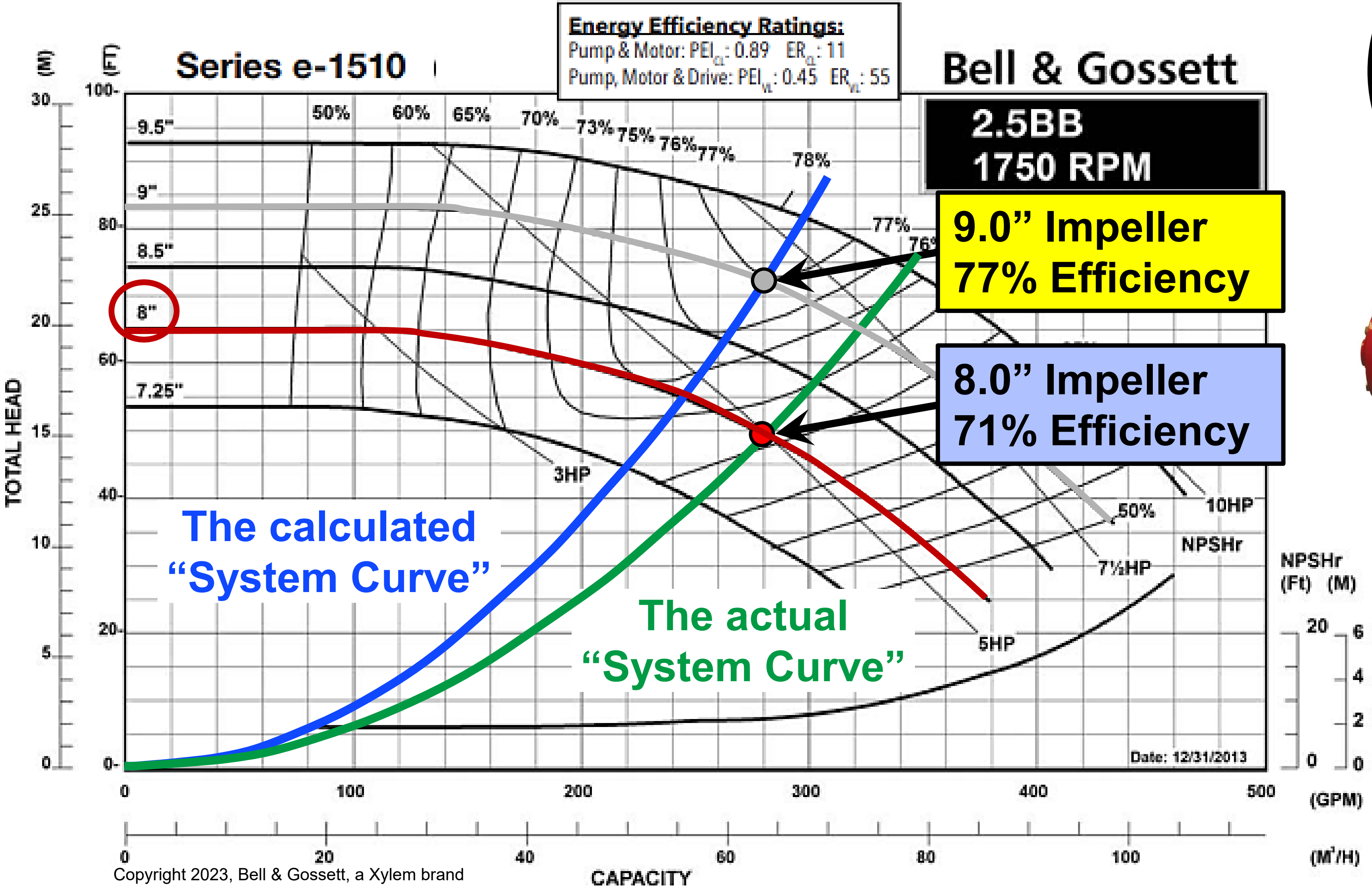
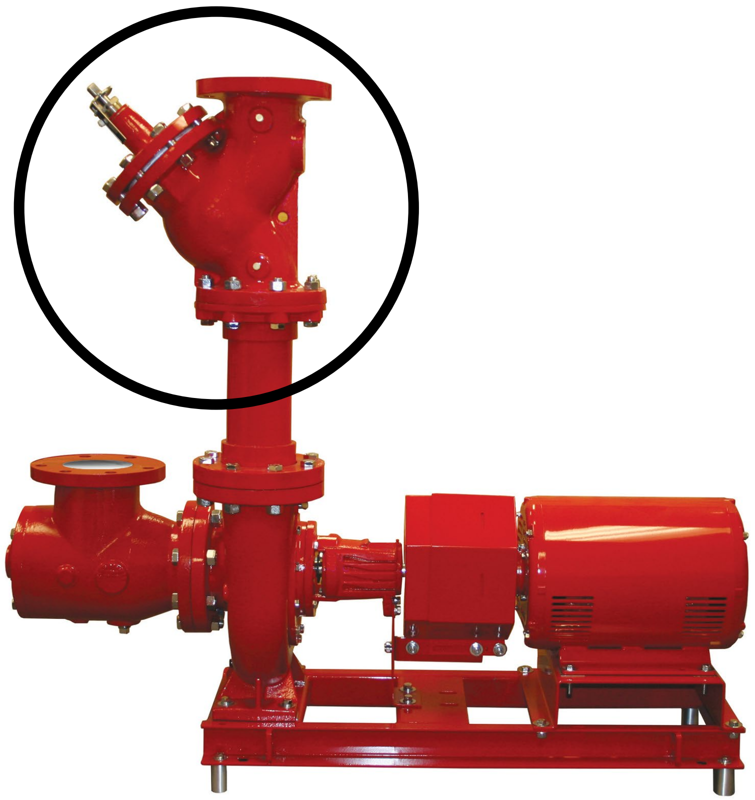
Impeller Trimming: Why and When?





- Trim Impeller
- Open Triple Duty

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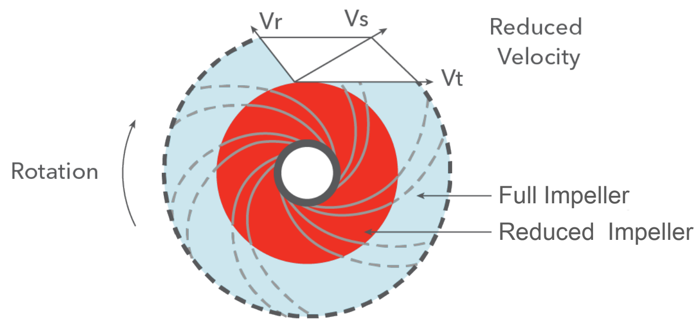
- Flow
  - $Q_2 = Q_1 (D_2/D_1)$
- Head
  - $h_2 = h_1 (D_2/D_1)^2$
- Power
  - $bhp_2 = bhp_1 (D_2/D_1)^3$

Q = Flow
D = Diameter
h = head
bhp = Horsepower

Subscript 2 indicates “new condition”

Subscript 1 indicates “old condition”



Diameter	Flow/ Volume	Head	Horsepower Required
100%	100%	100%	100%
90%	90%	81%	73%
80%	80%	64%	51%
70%	70%	49%	34%
60%	60%	36%	22%
50%	50%	25%	13%
40%	40%	16%	6%
30%	30%	9%	3%
20%	20%	4%	-
10%	10%	1%	-
0%	0%	0%	-

- These “Laws” *assume* pump efficiency remains constant

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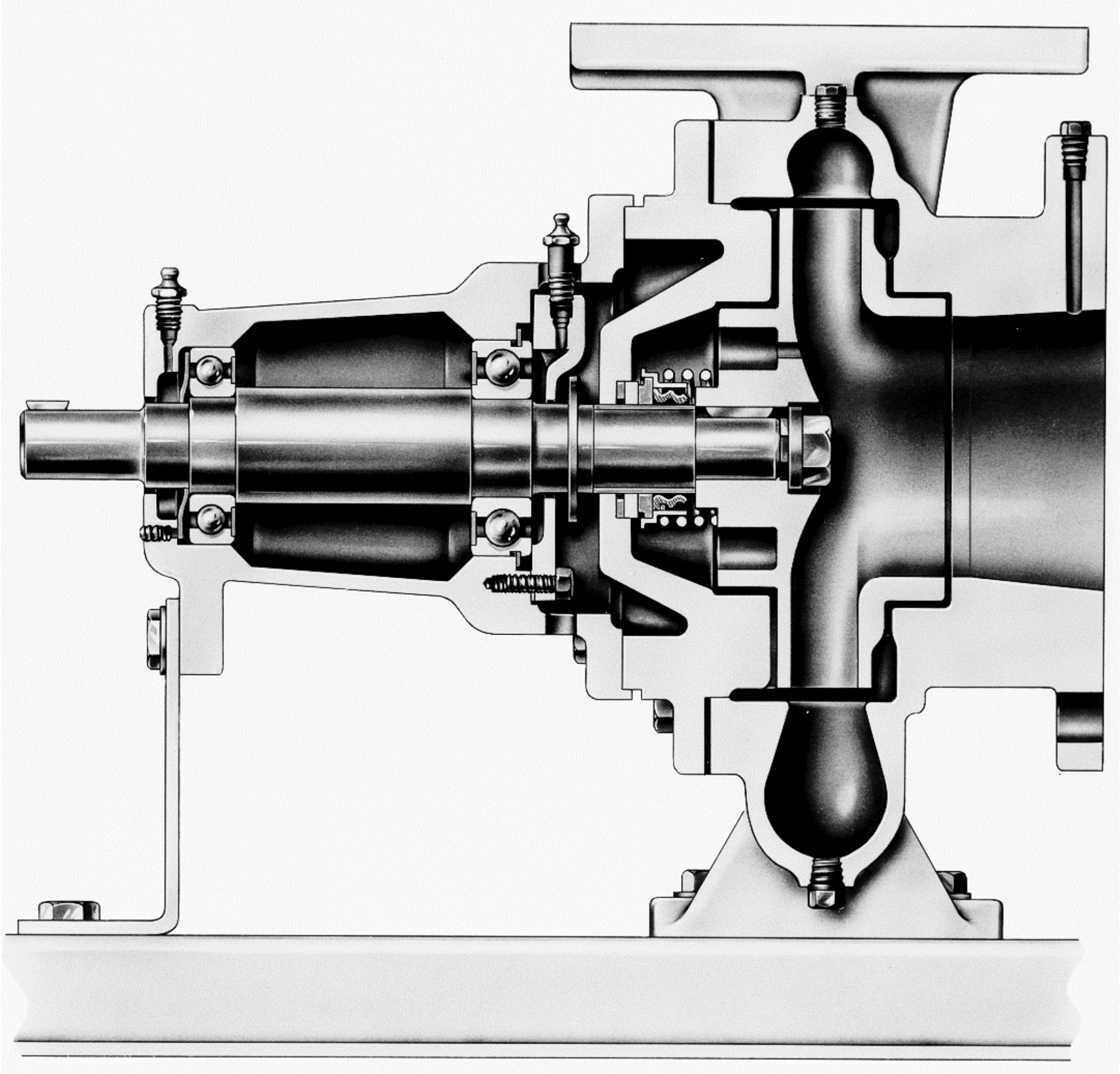

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- Bearing Friction
- Mechanical Seal
- Fluid Friction
- System Fluid Recirculation
- Shock Losses (Axial & Radial)

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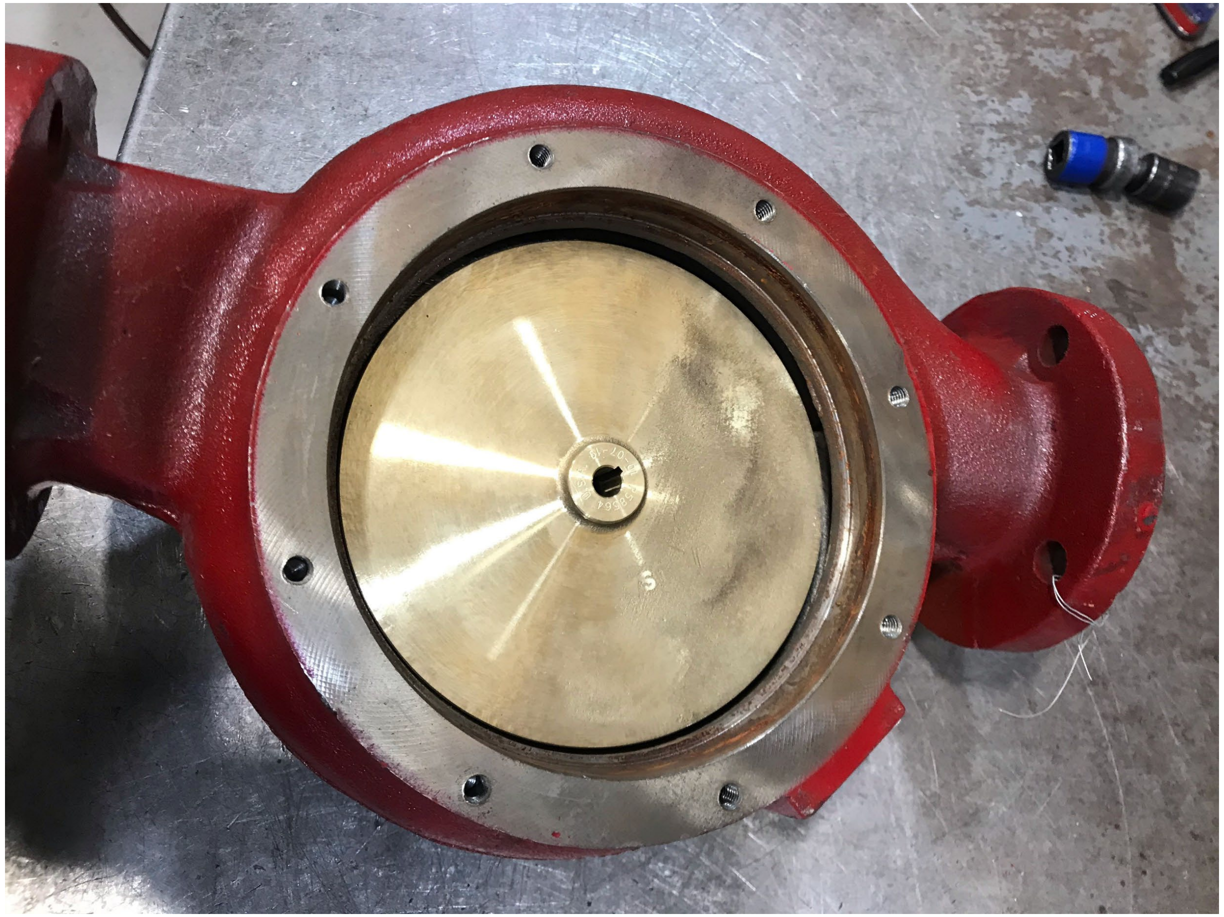
Smaller Impeller, Greater Internal Recirculation



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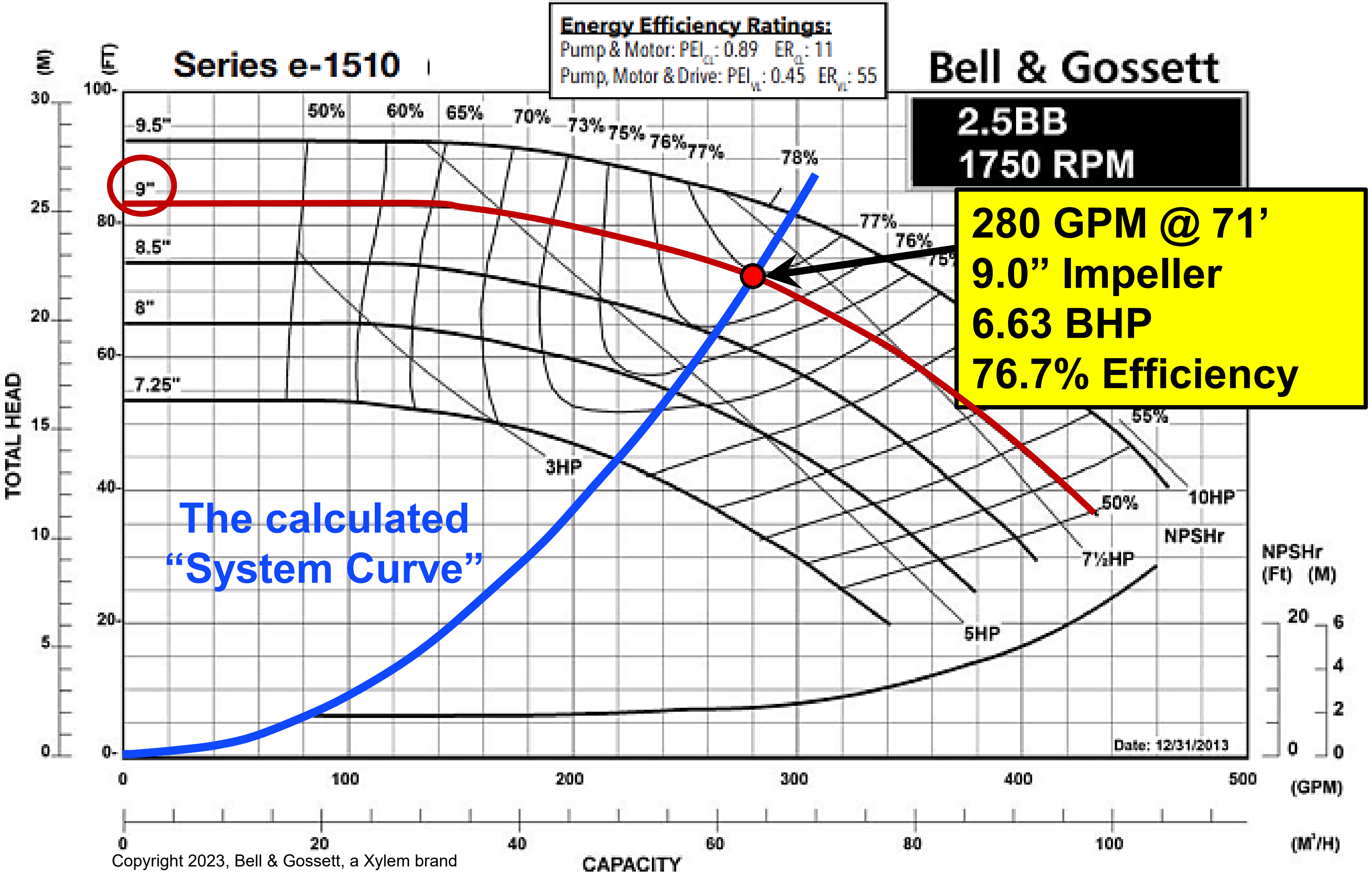
Impeller Trimming and the Affinity Laws



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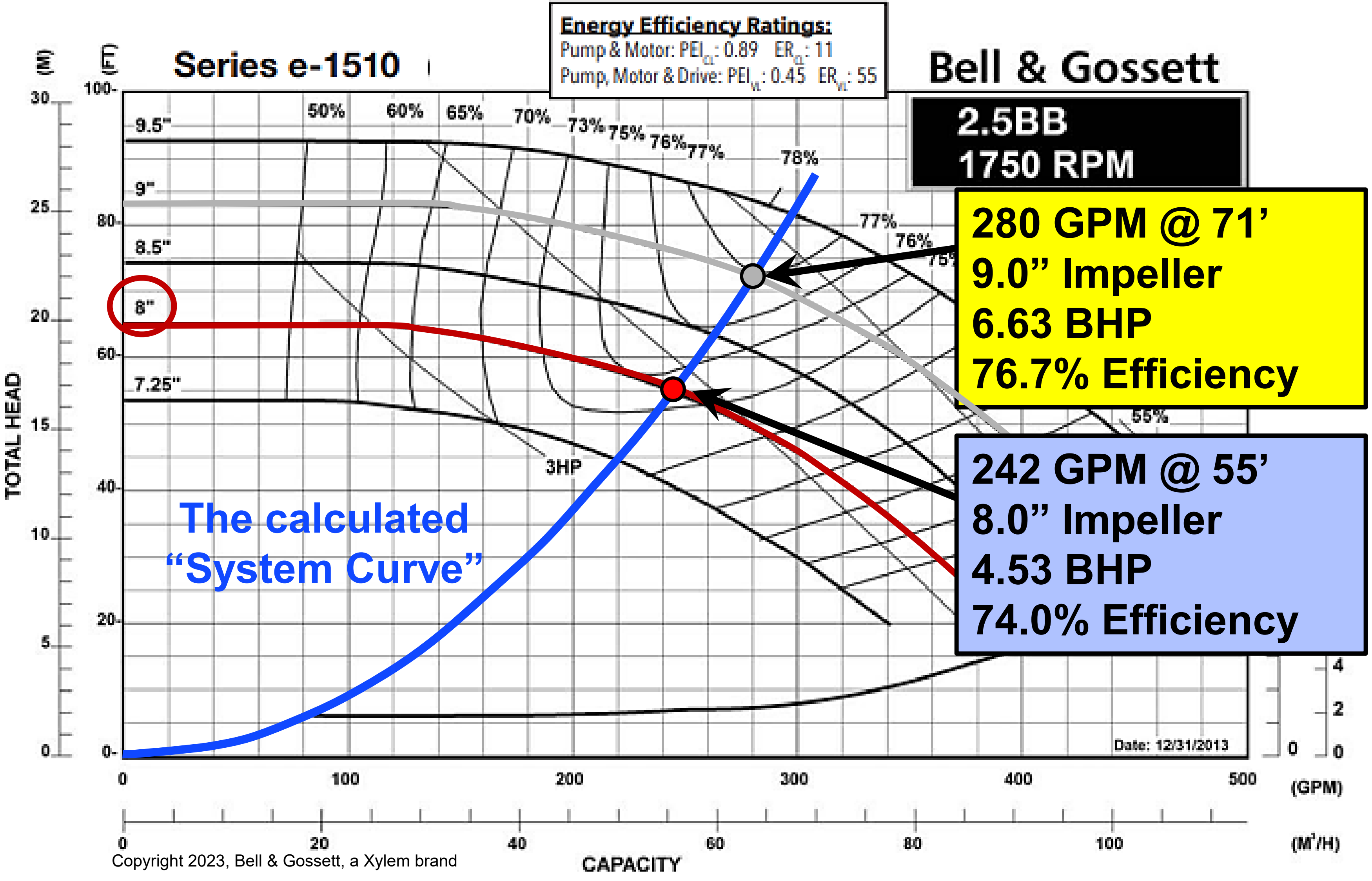
Impeller Trimming and the Affinity Laws



- Impeller Trimmed 12.5%
- Efficiency reduced 2.7%
- Flow reduced 13.5%
- Head reduced 23.0%
- BHP reduced 31.5%

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