Egyptian program for promoting Industrial Motor Efficiency
SAVE TODAY ... POWER TOMORROW

Pump Systems Optimisation
Expert Level Training

(Egypt Edition – version 8 December 2021)

Presented by:

Albert Williams & Siraj Williams

Ministry of Trade & Industry
وزارة التجارة والصناعة

Egypt National Cleaner Production Center
مركز تكنولوجيا الإنتاج الأنظف

مركز تحديث الصناعة
INDUSTRIAL MODERNISATION CENTRE

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Contents of this Course

1. Systems Approach
2. Efficiency & Process Demands
3. Pump System Fluid Relationships
4. System Curves
5. Pump Performance Curves
6. Pump System Energy Use
7. ASME EA-2 Standard & Guidance Document
8. ASME Chapter 6 Analysing the Data
9. ASME Chapter 7 Reporting & Documentation
10. Case Studies
11. MEASUR Software
12. Specific Energy
13. Pre-screening
14. Reliability & Maintenance
15. Motors
16. Control Methods
17. Collect Data & Field Measurements

Agenda: Day 1

PSO Expert Day 1	
TIME	**DESCRIPTION**
09:00 – 09:30	Welcome and registration
09:30 – 11:15	Introductions 1. Systems Approach 2. Efficiency & Process Demands
11:15 – 11:45	TEA
11:45 – 13:45	3. Pump System Fluid Relationships 4. System Curves
13:45 – 14:45	LUNCH
14:45 – 16:45	5. Pump Performance Curves 6. Pump System Energy Use 7. ASME EA-2 Standard & Guidance Document
16:45 – 17:00	SUMMARY OF DAY 1

Agenda: Day 2

PSO Expert Day 2

TIME	DESCRIPTION
09:30 – 11:15	8. ASME Chapter 6 Analysing the Data 9. ASME Chapter 7 Reporting & Documentation 10. Case Studies
11:15 – 11:45	TEA
11:45 – 13:45	11. MEASUR Software
13:45 – 14:45	LUNCH
14:45 – 16:45	12. Specific Energy 13. Pre-screening 14. Reliability & Maintenance
16:45 – 17:00	SUMMARY OF DAY 2

Agenda: Day 3

PSO Expert Day 3	
TIME	**DESCRIPTION**
09:30 – 11:15	Preparation for site visit Discussion of WTS-8 pump system processes
11:15 – 11:45	TEA
11:45 – 13:45	Site walkthrough (Hot transfer pumps group 1, Filter feedpumps group 2) Site walkthrough (Hot transfer pumps group 2, Filter feedpumps group 1)
13:45 – 14:45	LUNCH
14:45 – 17:00	Metering installations

Agenda: Day 4

PSO Expert Day 4	
TIME	**DESCRIPTION**
09:30 – 11:15	Discussions and process flows development
11:15 – 11:45	TEA
11:45 – 13:45	Remove equipment, extract data Development of MEASUR basecase models
13:45 – 14:45	LUNCH
14:45 – 17:00	Development of basecase MEASUR models (cont'd) Development off pump and system curves Development of proposed cases MEASUR models

Agenda: Day 5

PSO Expert Day 5	
TIME	**DESCRIPTION**
09:30 – 11:15	14. Reliability & Maintenance (cont'd) 15. Motors
11:15 – 11:45	TEA
11:45 – 13:45	16. Control Methods 17. Collect Data & Field Measurements
13:45 – 14:45	LUNCH
14:45 – 16:45	Development of proposed cases MEASUR models (cont'd) EEM Recommendations Business Case Development
16:45 – 17:00	COURSE WRAP UP

Welcome

- Name
- Organisation
- Energy management experience
- What do you expect to learn over these few days?

1. The Systems Approach

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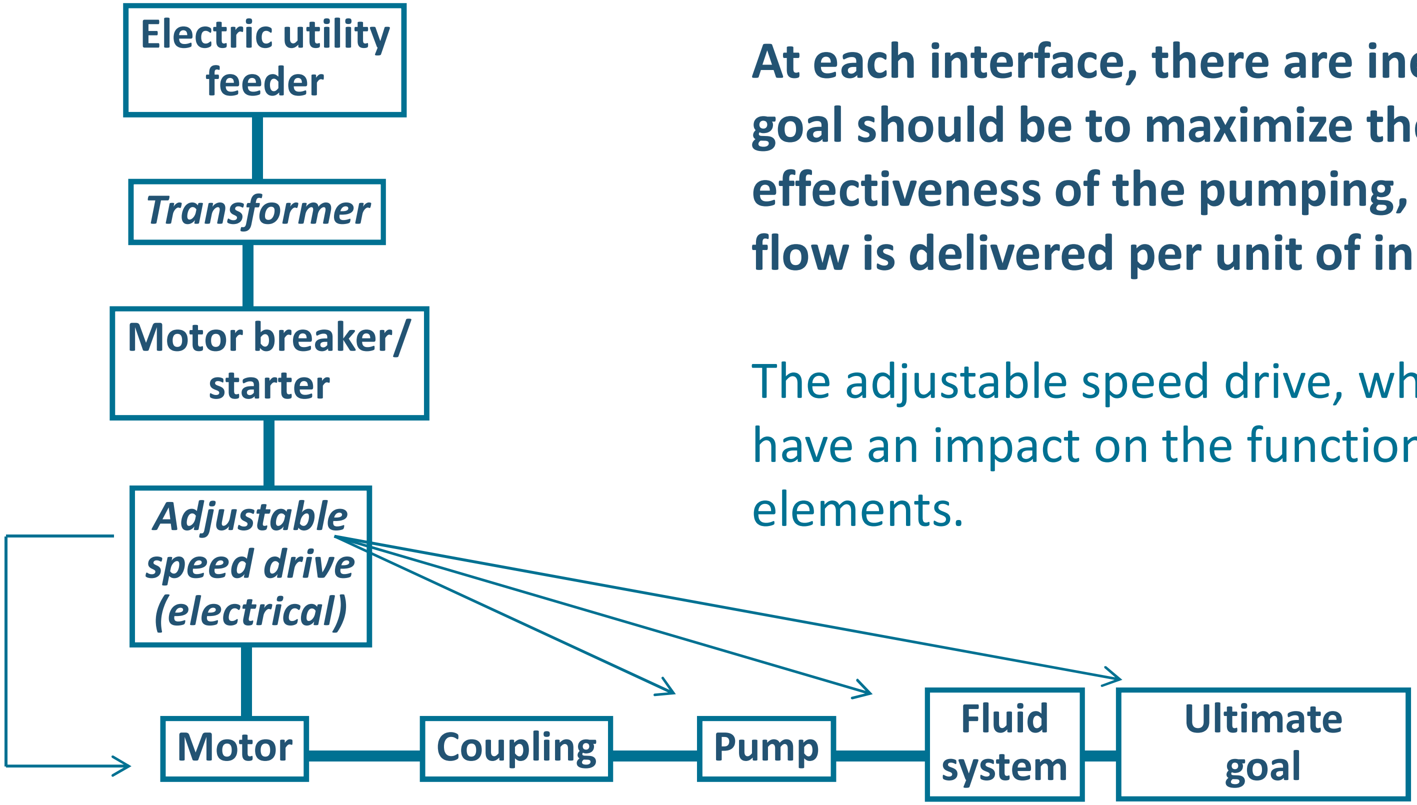
Systems Approach



At each interface, there are inefficiencies. The goal should be to maximize the overall cost effectiveness of the pumping, or how much flow is delivered per unit of input energy.

The adjustable speed drive, when present, will have an impact on the function of several elements.

Slide Courtesy of Oak Ridge National Laboratory

2. Efficiency & Process Demands

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Efficiency

Standard Efficiency definition = Energy out / Energy in

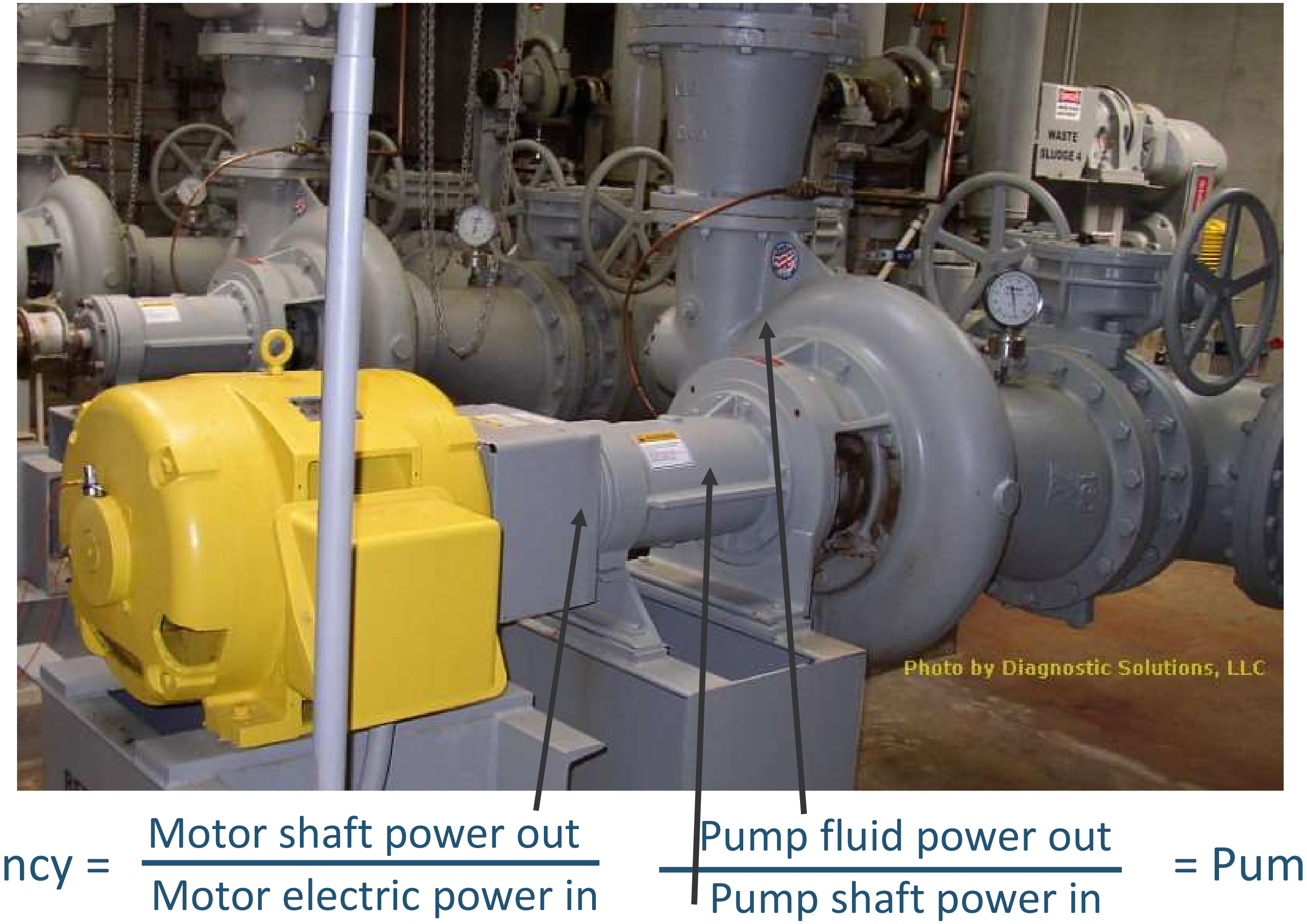
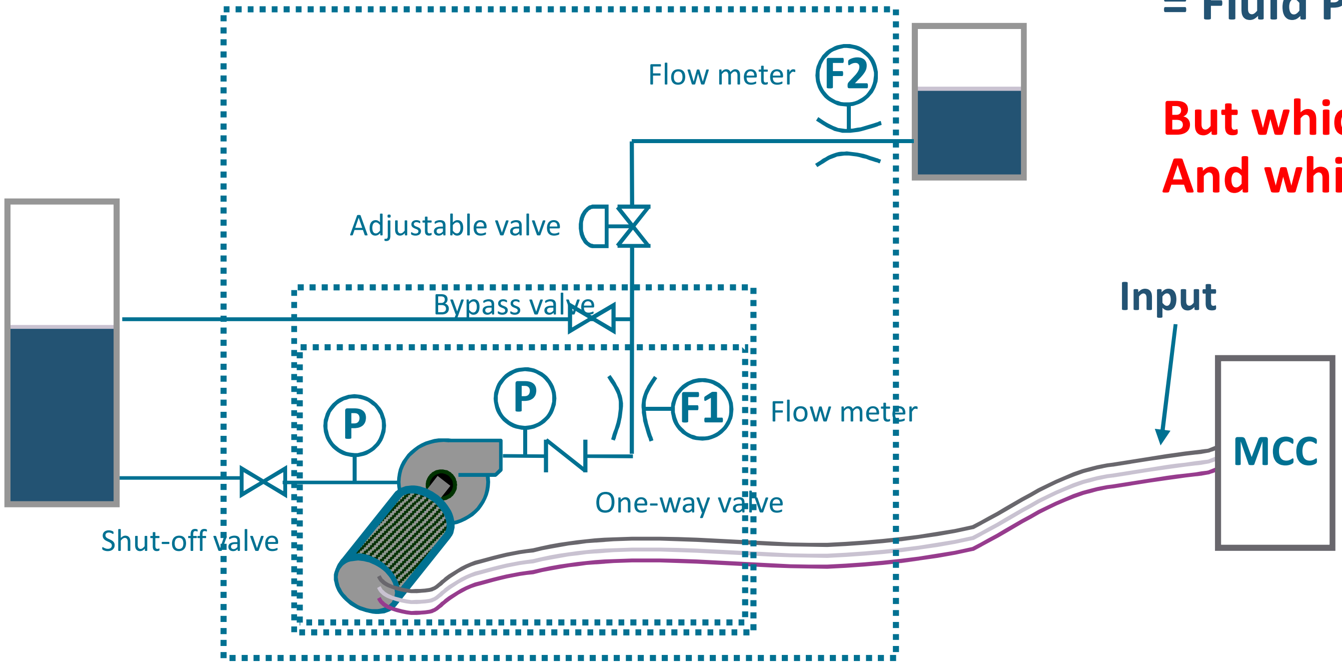


Photo by Diagnostic Solutions, LLC

$$\text{Motor efficiency} = \frac{\text{Motor shaft power out}}{\text{Motor electric power in}} \qquad \frac{\text{Pump fluid power out}}{\text{Pump shaft power in}} = \text{Pump efficiency}$$

Defining the System



Output:
= Flow Rate x Head x Constant
= Fluid Power

But which flow rate?
And which head?

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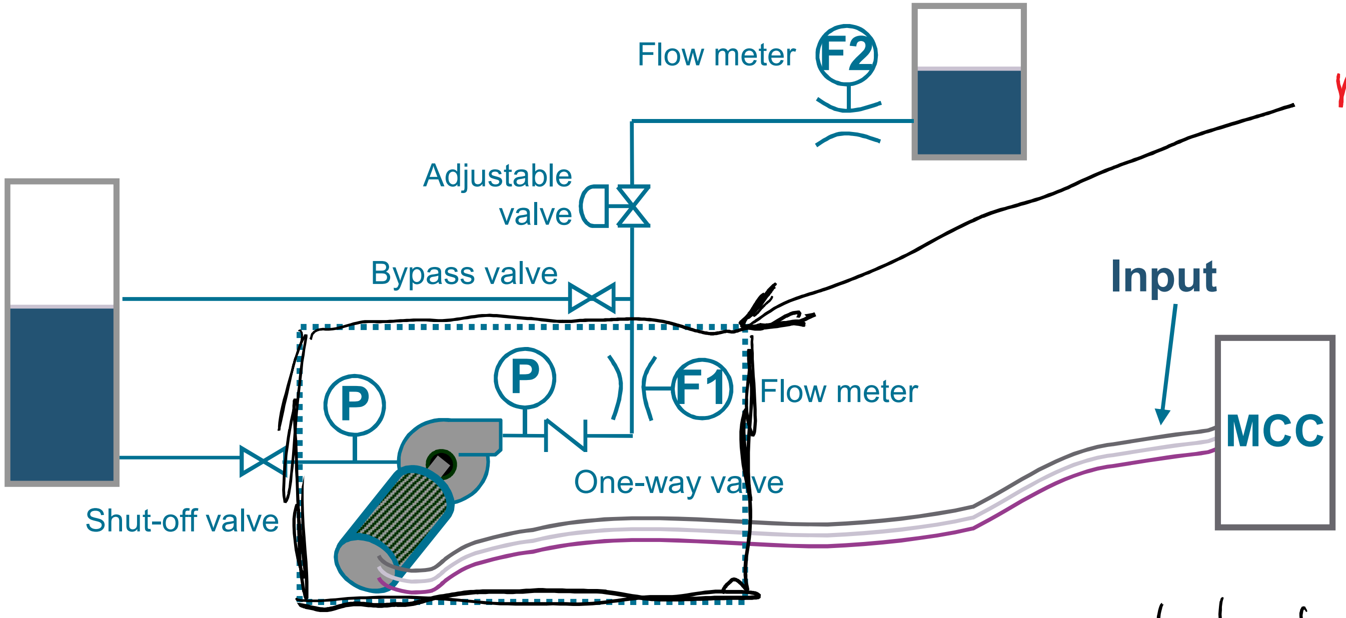
System Efficiency

$\eta_{motor} = 90\%$

$\eta_{pump} = 78\%$

$\eta_{sys} = ?$



* What about bypass?

* And FCV?

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Important Fundamental Relationships

$$\text{Fluid Power (kW)} = \frac{\text{Flow rate (l/s) x Head (m) x Rel. Density}}{102}$$

$$\text{Fluid Energy (kWh)} = \text{Fluid Power x run time}$$

$$\left.\begin{array}{l}\text{Reduce the run time}\\ \text{Reduce the flow rate}\\ \text{Reduce the head}\end{array}\right\}\quad \text{Reduce energy use, cost}$$

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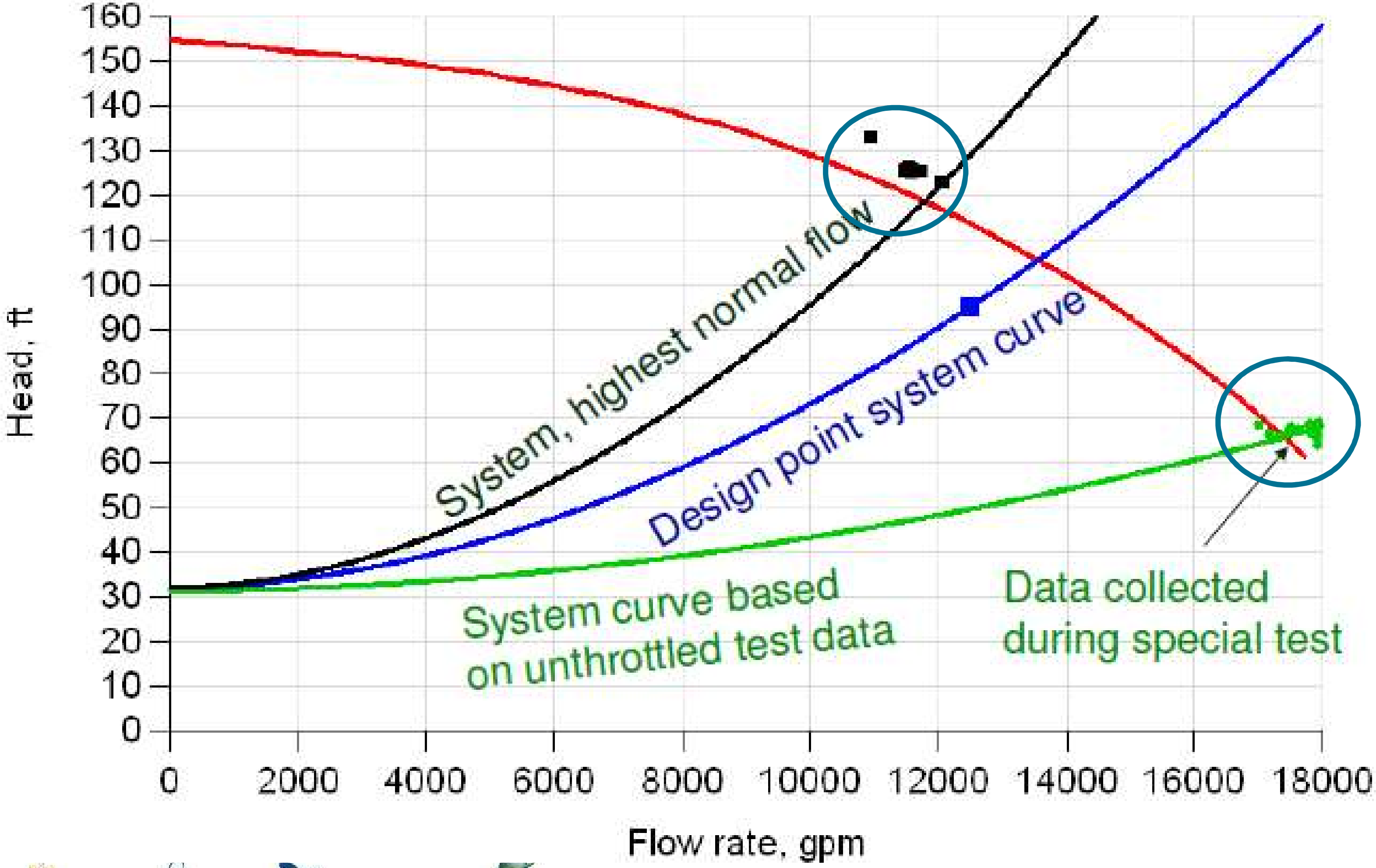
Example – Fluid Power

An industrial pumping system delivers water flow of 125 l/s at a total head of 29m.

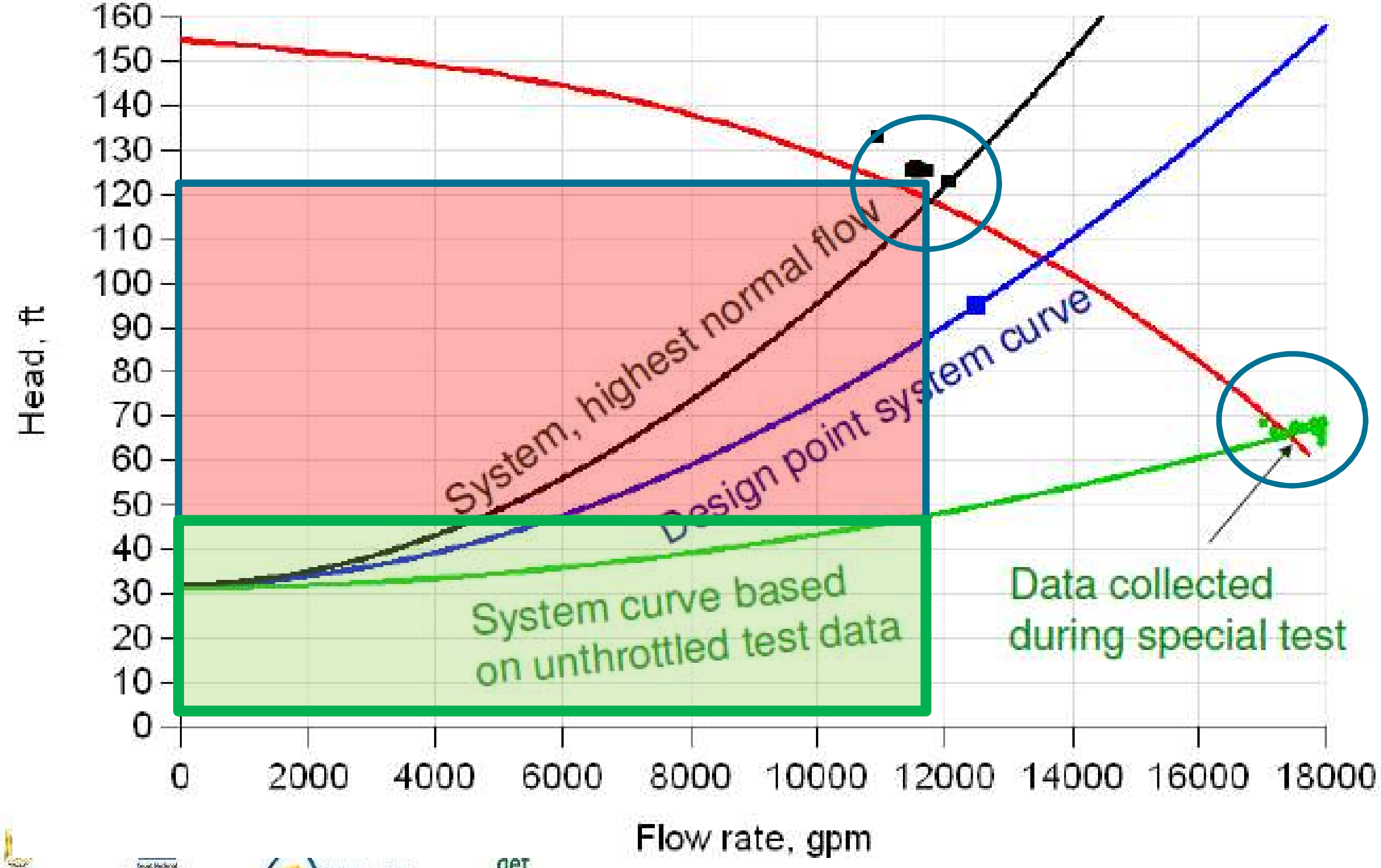
What is the fluid power required?

A Real Example



Delivered versus Needed?

The delivered fluid power is about
2.8 times larger than needed



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3. Pump System Fluid Relationships

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Basic System Knowledge

- Static and dynamic head
- Losses and loss coefficients
- System curves

Two types of Resistance to overcome

- Static
- Dynamic

Pressure Resistance from a System

- It takes **Energy** to lift fluid from one level to another
  - The **Pressure** used to ***lift*** the fluid is: - *Static head*
  - The **Energy** used to ***lift*** the fluid is: - *Independent of velocity*
  - The **Power** to ***lift*** fluid is: - *Linear function of velocity*

- It takes **Energy** to move fluid though a system of pipes and other equipment.
  - The **Pressure** used to overcome friction is: - *Dynamic head*
  - The **Pressure** required is: - *Proportional to the square of the fluid velocity*
  - The **Power** is: - *Proportional to the cube of the velocity*

Pumping Effort (Output)

- The ability for pumps to move water is based on the energy contained in a mass of water
- Pump output is measured in meters of head. The three common terms used to express this energy in water is:

  - Elevation / Pressure Head (Static Head or H_s) → Lift the fluid
  - Velocity Head (H_v) → Create kinetic energy
  - Head loss due to Frictional Losses (H_f) → overcome friction

Total Head (TH) = $H_s + H_v + H_f$

Velocity Head

Velocity head (H_v) is the amount of energy required to cause the water to move at a given velocity. This is represented by the following relationship:

$H_v = v^2/2g$ v = Velocity in meters/second

g = acceleration due to gravity ($9.8 \, m/sec^2$)

To determine velocity, the following equation can be used:

$v = Q/A$ Q = Flow in m^3/sec

A = the cross sectional Area of the inside of the pipe in m^2

Velocity head is usually below 0.5 m and can often be considered <u>minimal</u> for many water pumping systems

Example – Velocity Head

An industrial pumping system delivers a flow of 125 l/s at a head of 29m. If the pipe diameter is 250 mm, what is the velocity head?

Head Loss due to Piping Frictional Losses

- Frictional Head loss (H_f) is the loss of energy due to the friction of the piping materials and is expressed in meters of head. This can be determined theoretically using:

The Darcy Weisbach Equation

or

The Hazen-Williams Equation

- H_f can be determined more accurately in the field using actual pressure measurements

Estimating Pipe Friction Loss

- Pipe friction loss estimates are usually based on an equation referred to as Darcy-Weisbach
- This equation is very useful to understand what parameters influence ***frictiona***l losses in piping

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

H_f = pressure drop due to friction (ft or m)
f = Darcy friction factor
L = pipe length (ft or m)
d = pipe diameter (ft or m)
$\frac{v^2}{2g}$ = velocity head (ft or m)

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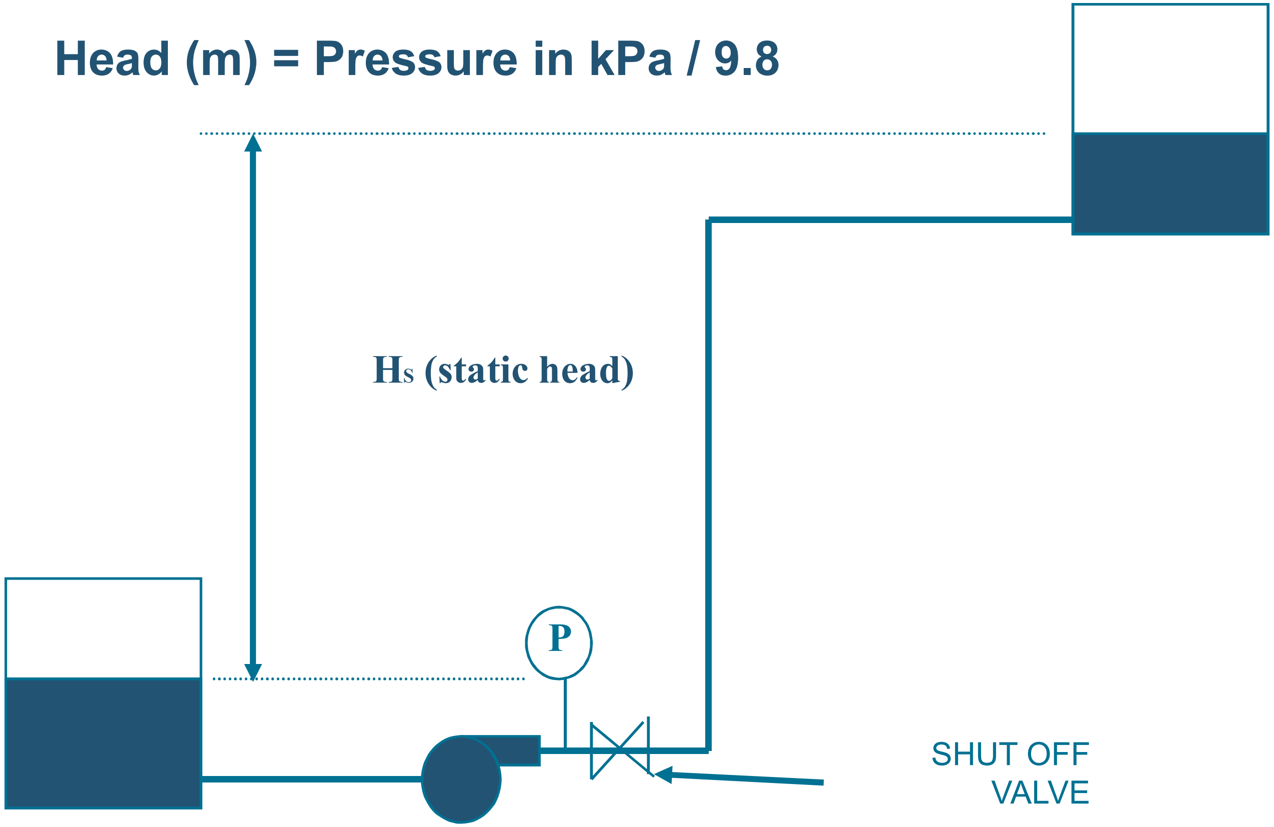
The Friction head loss:

- Is a function of fluid velocity
- Lower flow (lower velocity) results in lower head loss
- Head loss is proportional to the square of velocity - v^2
- Reduced to 25% when velocity is cut by 50%
- Increases by a factor of 4 when velocity is doubled

Static Head

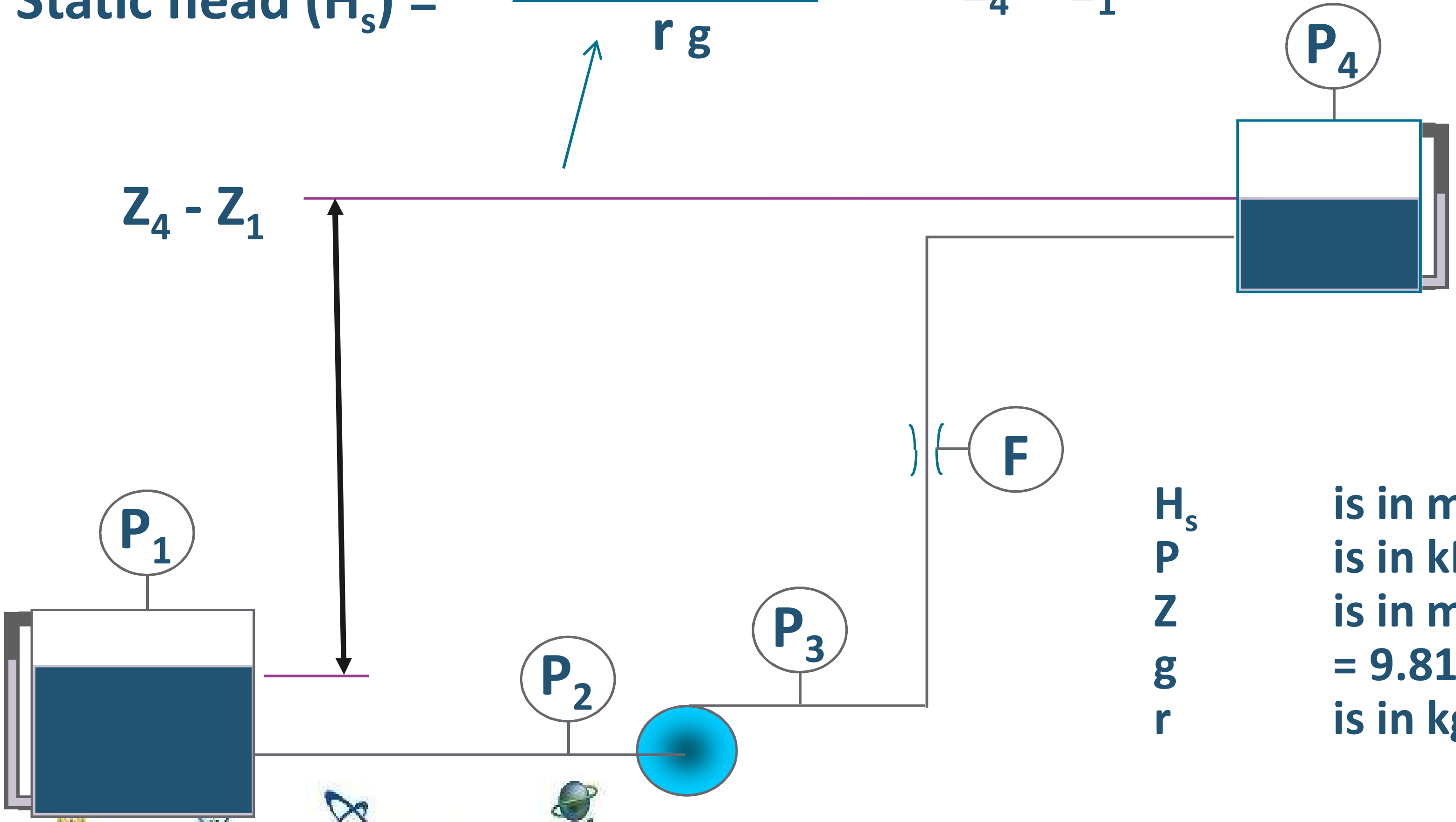
Head (m) = Pressure in kPa / 9.8



Static Head

The static head is made up of elevation, and sometimes pressure components

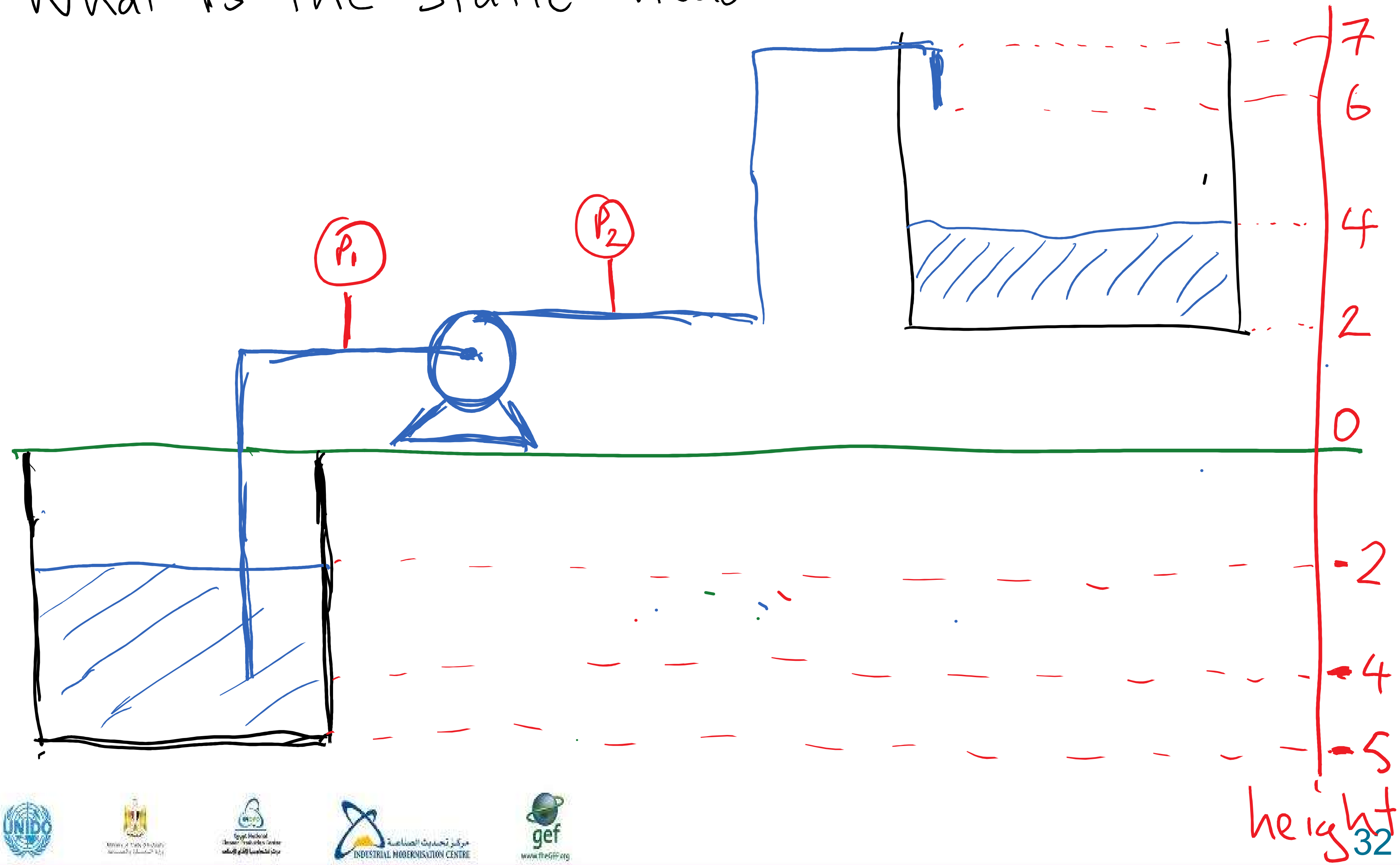
$$\text{Static head } (H_s) = \frac{(P_4 - P_1)}{r\,g} + Z_4 - Z_1$$



H_s is in m
P is in kPa
Z is in m
g = 9.81 m/s²
r is in kg/m³

Example - Head

What is the static head?



4. System Curves

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The System Curve

How do we show the pressure or head needed to push flow through a system?

- The answer is with a *System Curve*
- The two types of *Flow* resistance, **Static** and **Dynamic**, are added

System Head Curve: All Frictional System

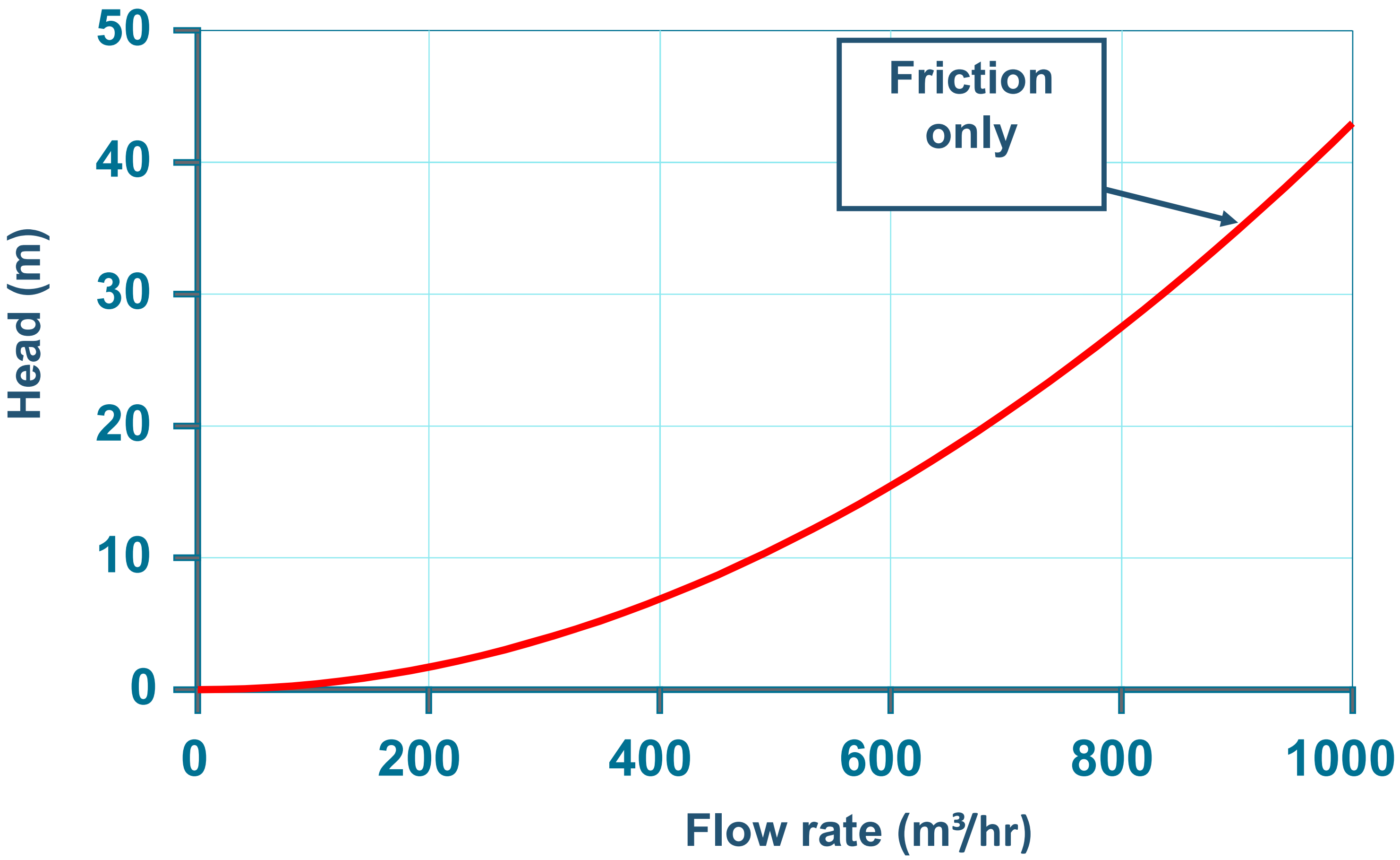


Figure Courtesy of Oak Ridge National Laboratory

System Head Curve: All Static System

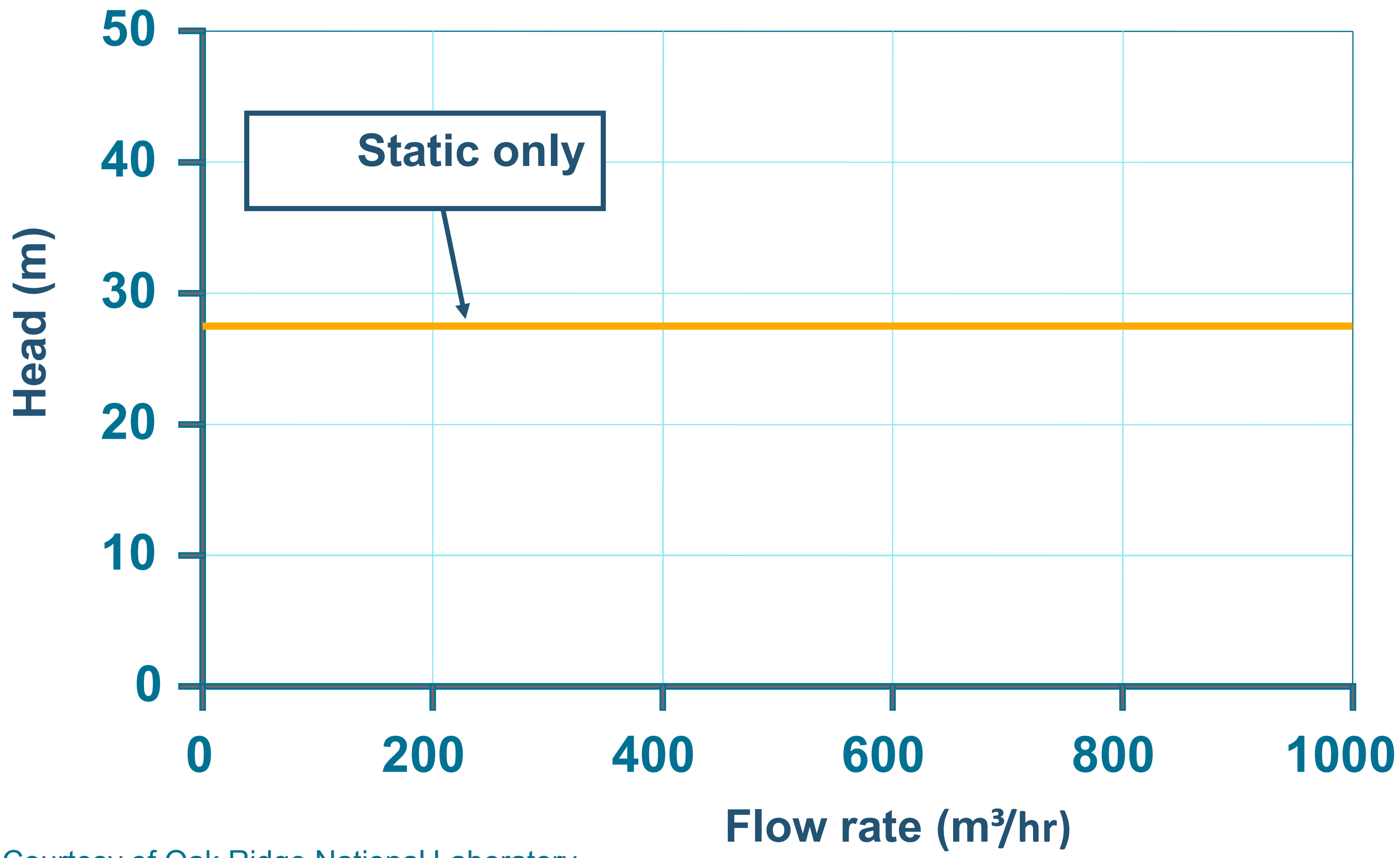
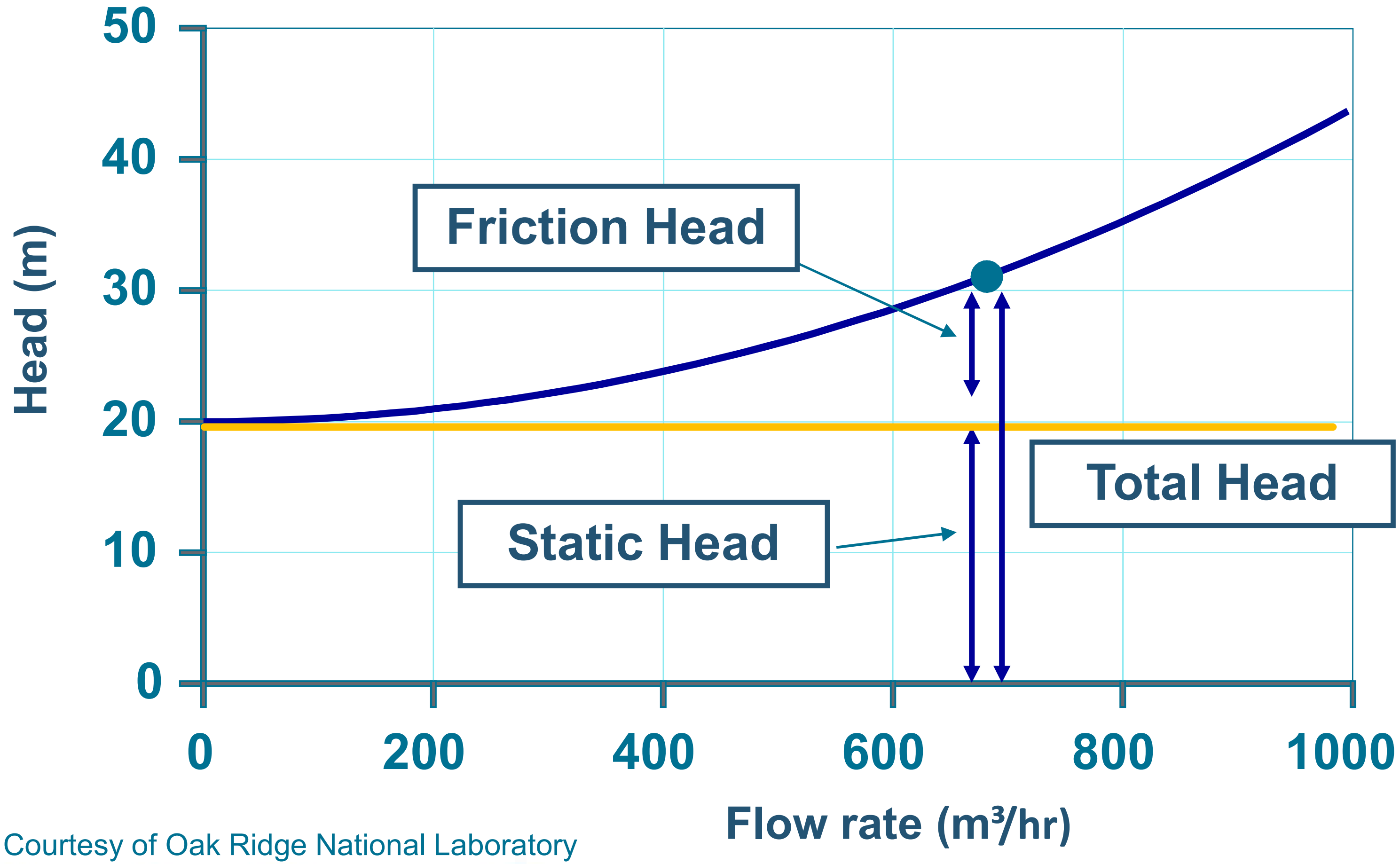


Figure Courtesy of Oak Ridge National Laboratory

System Head Curve: Combined Static and Frictional System



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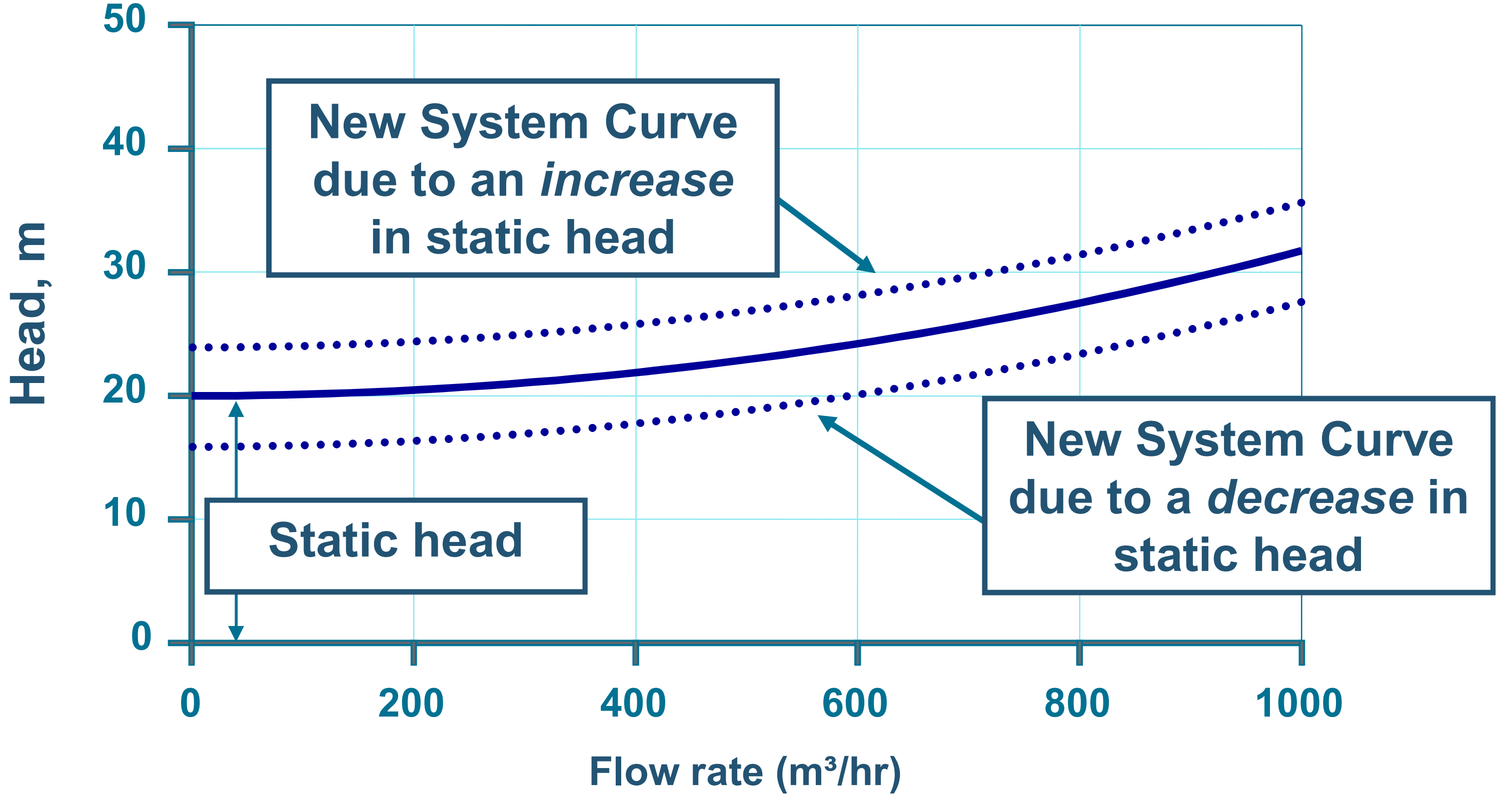
System Effects: Changes in Static Head

The effect on the system head curve when the static head changes



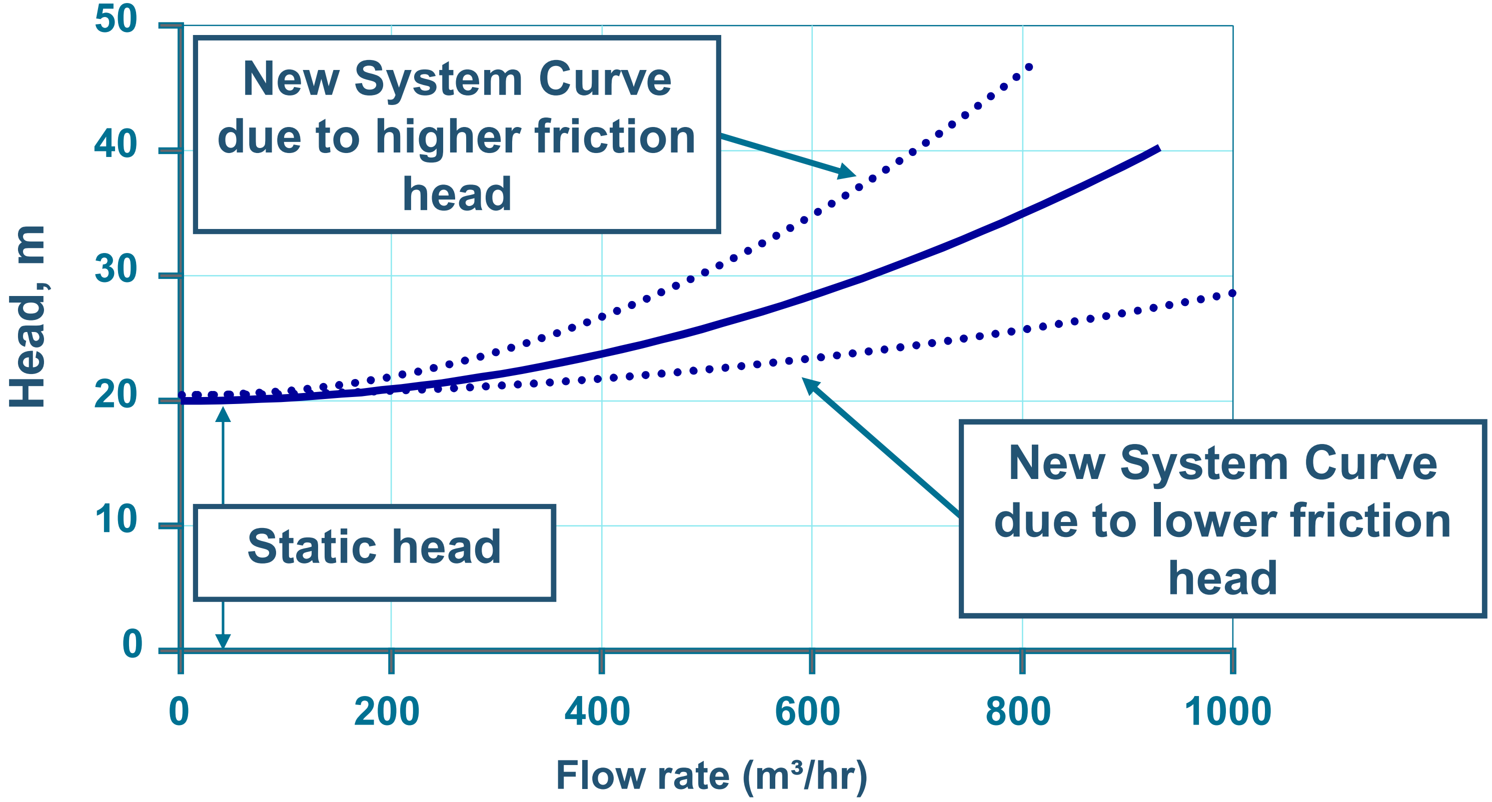
New System Curve due to an *increase* in static head

Static head

New System Curve due to a *decrease* in static head

System Effects: Changes in Friction Head

The effect on the system head curve when system friction changes

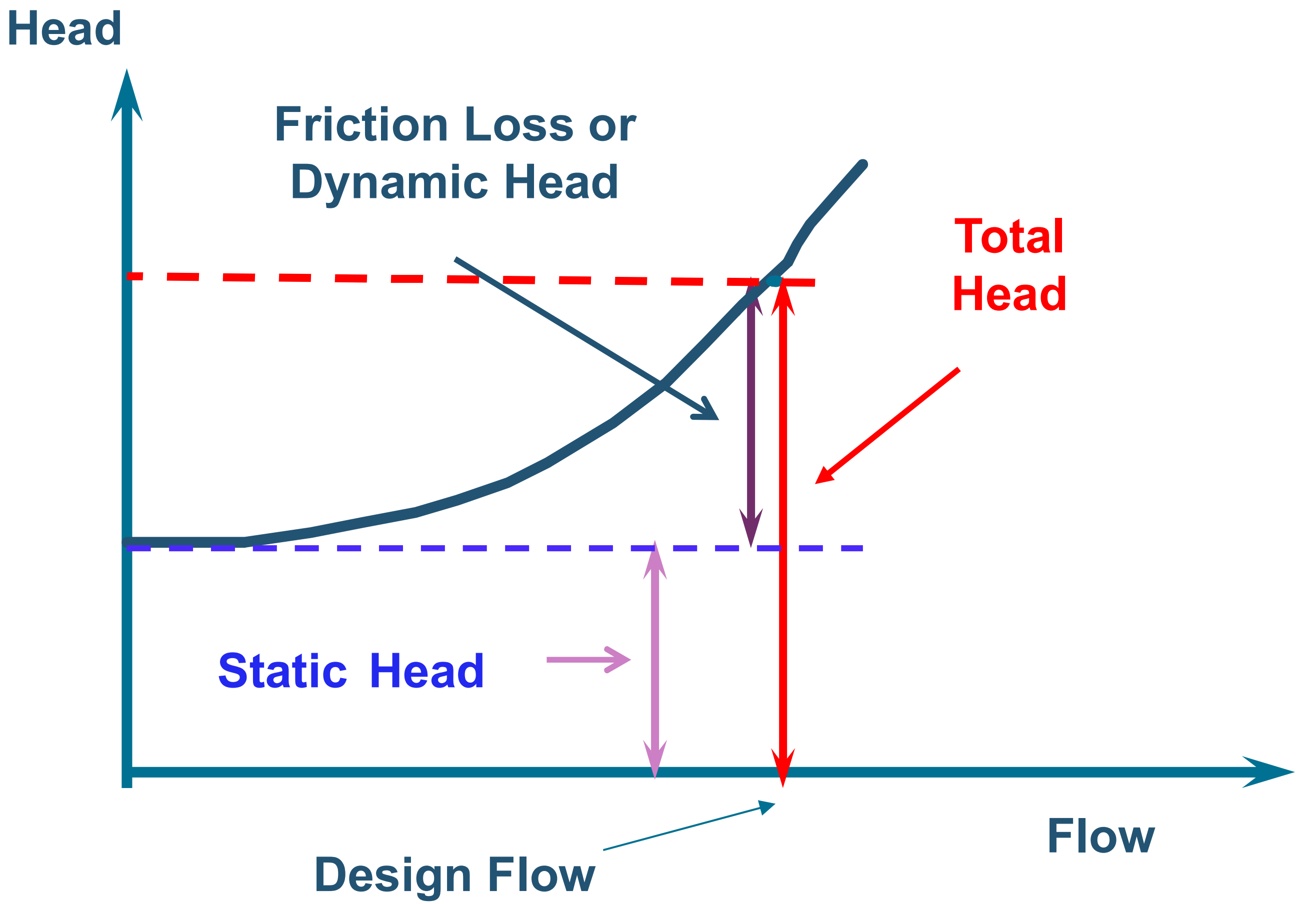


New System Curve due to higher friction head

Static head

New System Curve due to lower friction head

Two Components of System Curves: Static and Friction or Dynamic Head



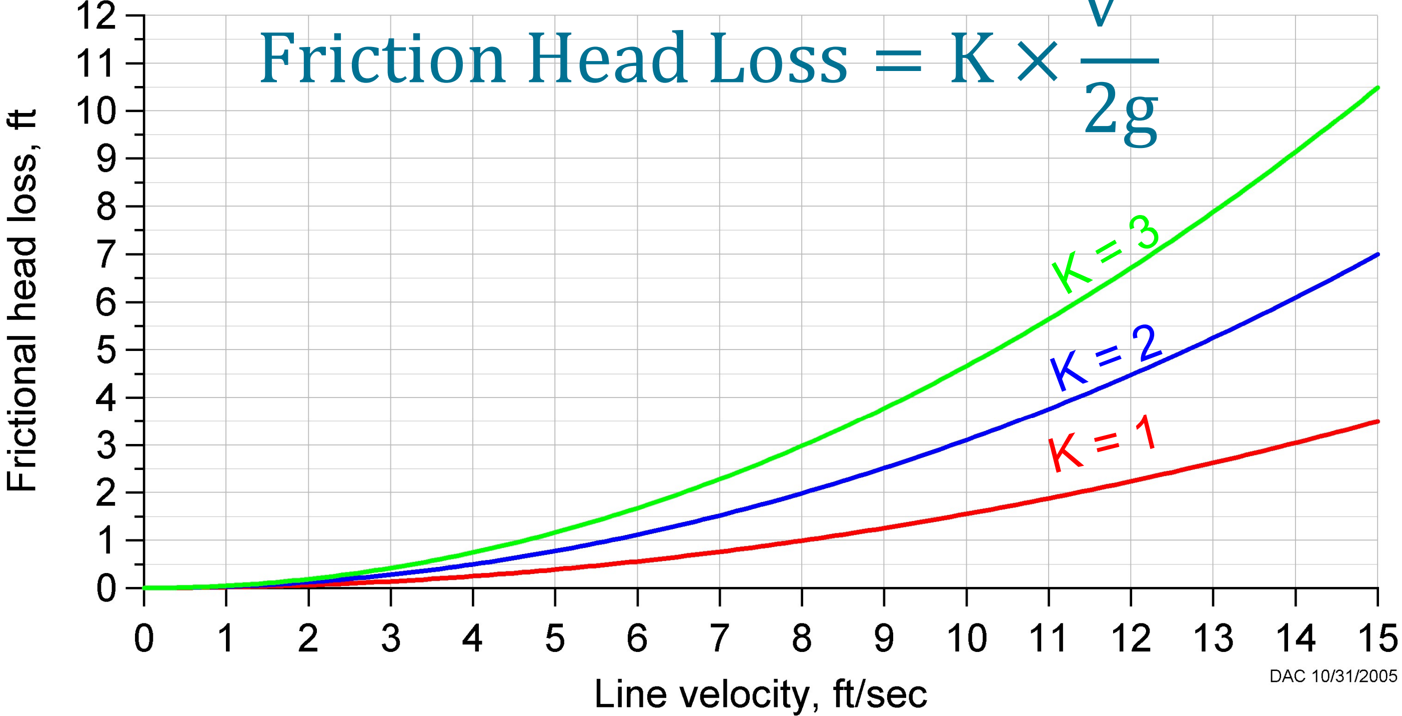
Friction head varies with approximately the square of the flow rate

Static head is the sum of pressure + elevation head differences from start to finish

Loss coefficient (K)

- All fluid movement results in frictional losses.
- The head loss can be estimated using this relationship.
- The K value is called the Loss Coefficient.
- It is multiplied by the velocity head to estimate the frictional head loss of one or more components and/or fittings.

$$\text{Friction Head Loss} = \mathrm{K} \times \frac{\mathrm{v}^2}{2\mathrm{g}}$$



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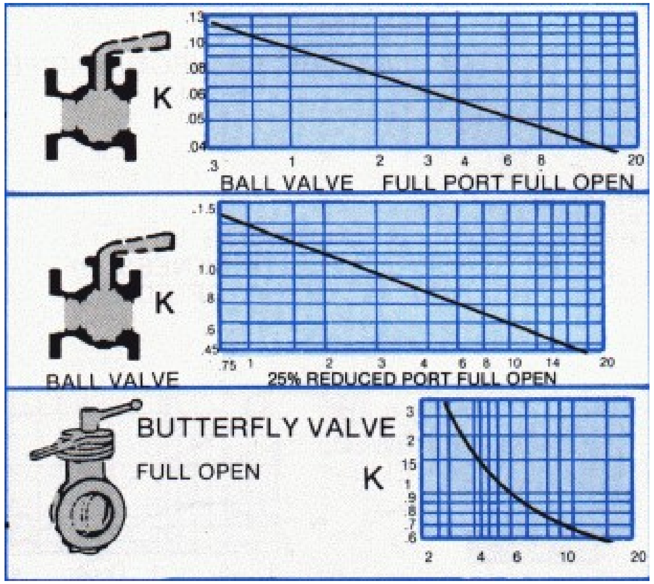
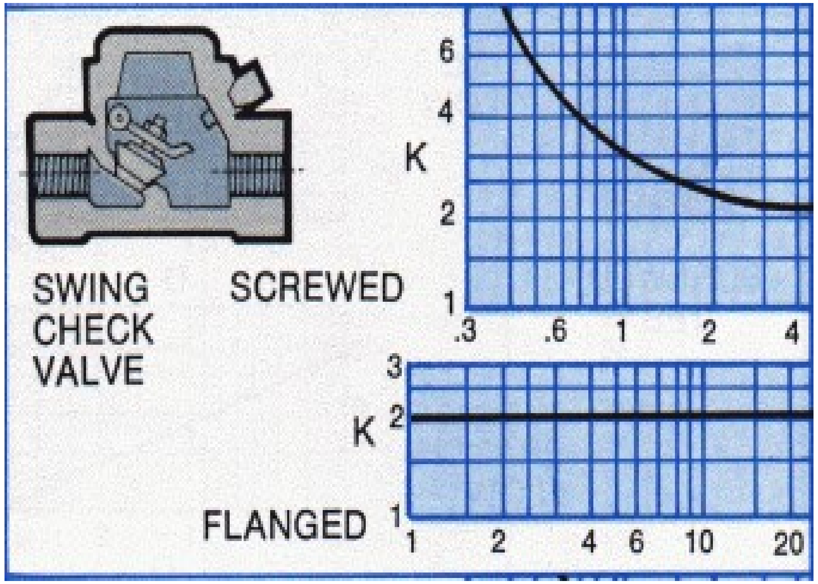
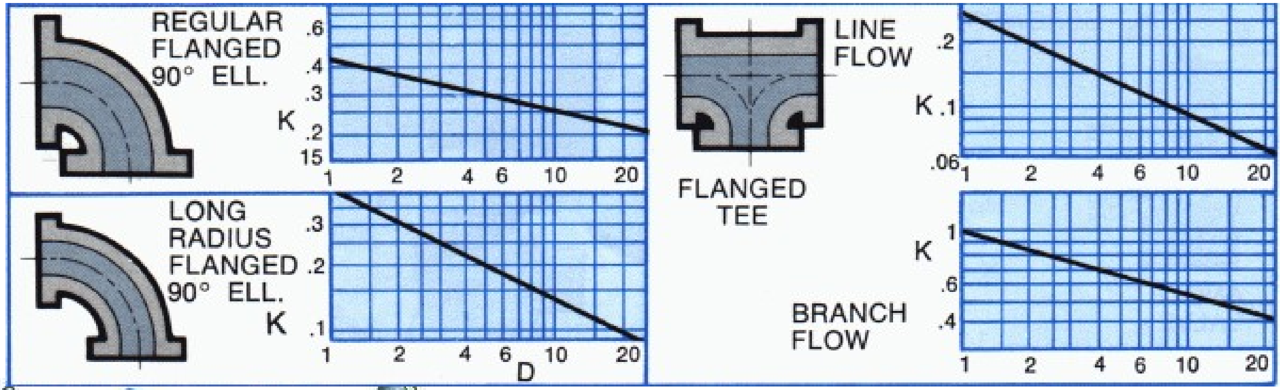
Generic loss K's

- available from several sources;
examples from HI Engineering Data Book

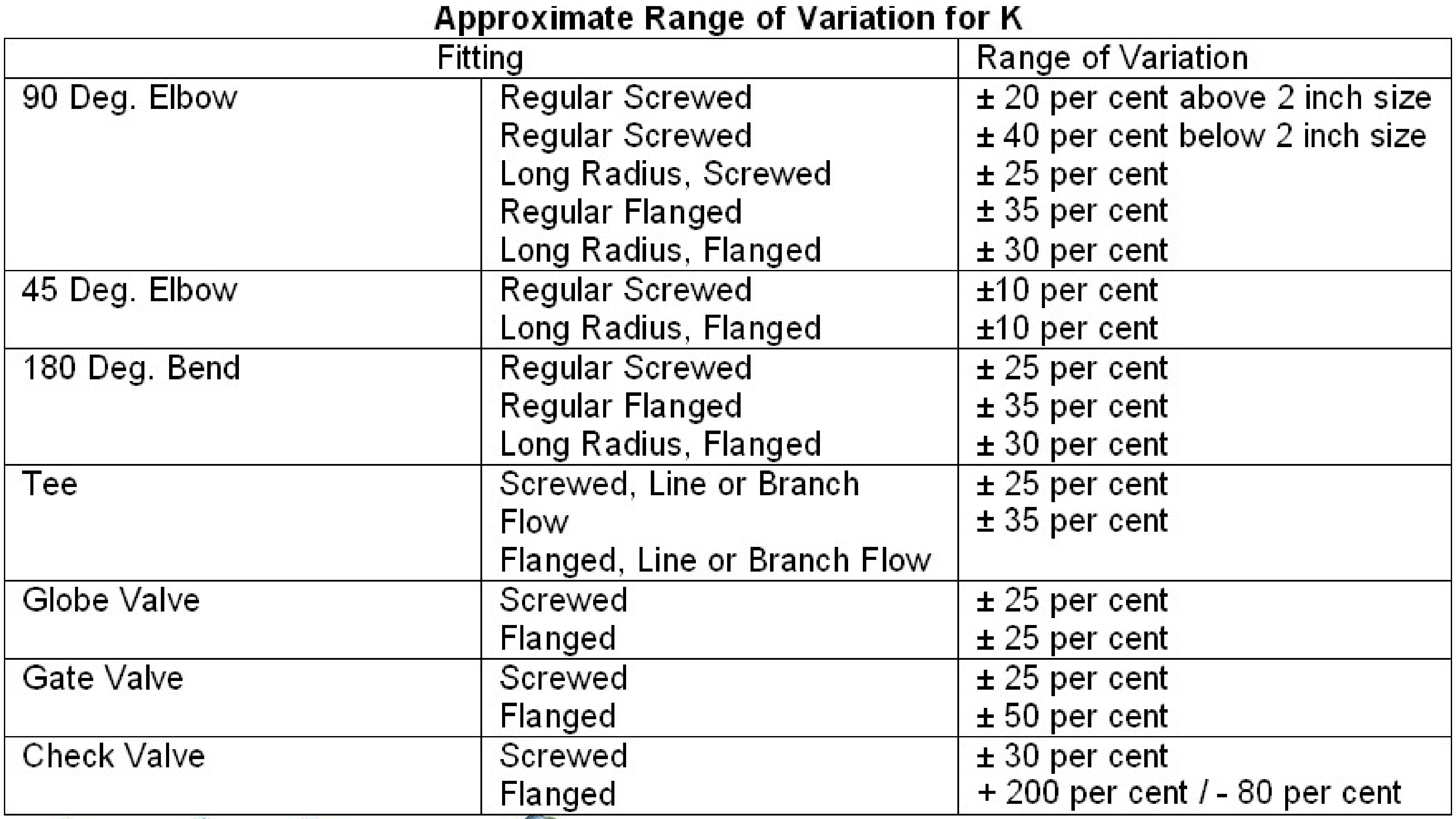






Loss coefficient (K)

This is not like generic medicine; there are huge differences between generic and specific



Approximate Range of Variation for K

Fitting		Range of Variation
90 Deg. Elbow	Regular Screwed Regular Screwed Long Radius, Screwed Regular Flanged Long Radius, Flanged	± 20 per cent above 2 inch size ± 40 per cent below 2 inch size ± 25 per cent ± 35 per cent ± 30 per cent
45 Deg. Elbow	Regular Screwed Long Radius, Flanged	±10 per cent ±10 per cent
180 Deg. Bend	Regular Screwed Regular Flanged Long Radius, Flanged	± 25 per cent ± 35 per cent ± 30 per cent
Tee	Screwed, Line or Branch Flow Flanged, Line or Branch Flow	± 25 per cent ± 35 per cent
Globe Valve	Screwed Flanged	± 25 per cent ± 25 per cent
Gate Valve	Screwed Flanged	± 25 per cent ± 50 per cent
Check Valve	Screwed Flanged	± 30 per cent + 200 per cent / - 80 per cent

5. Pump Performance Curves

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Nameplate Operating Point

Nameplate data applies to one particular operating point

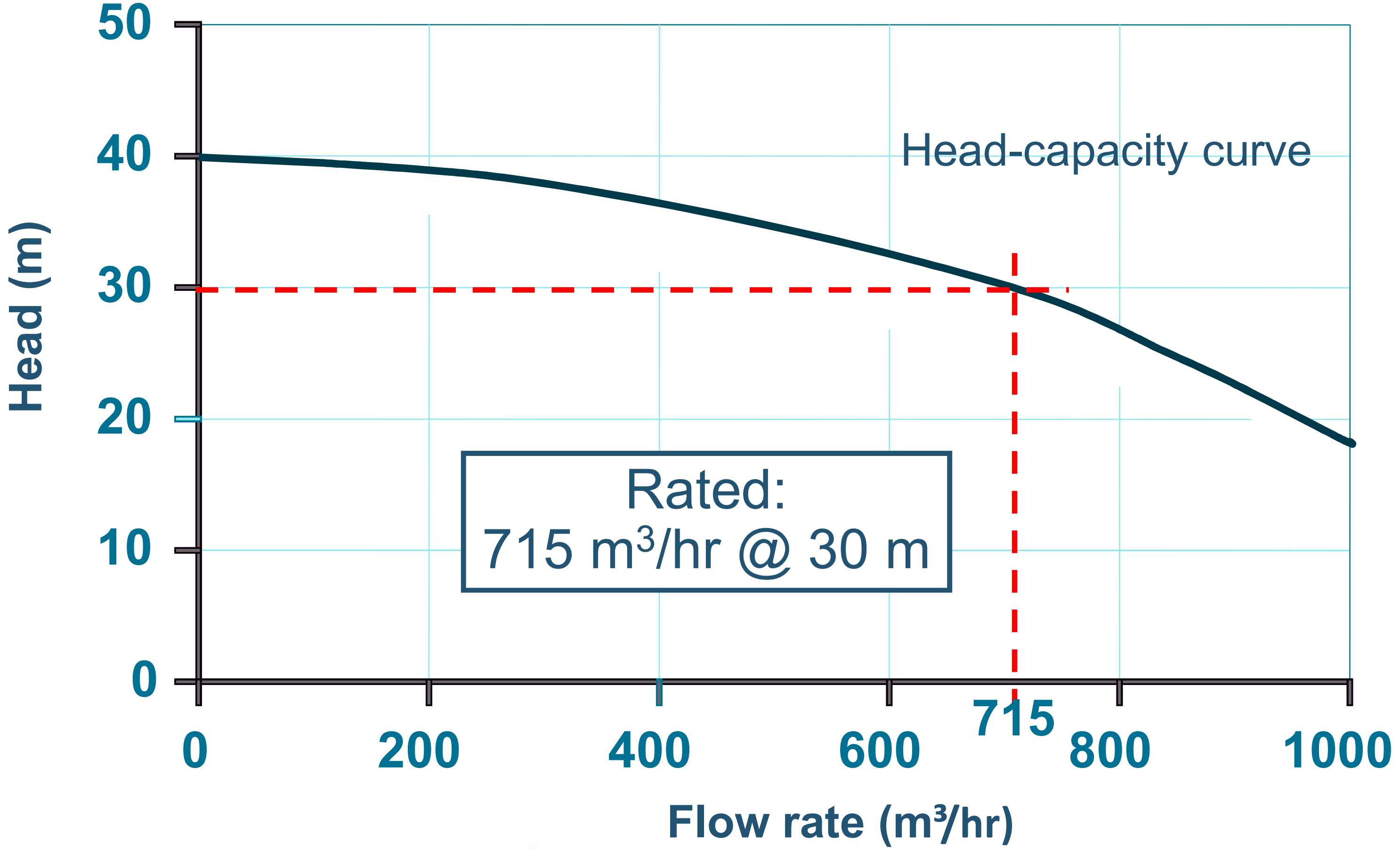
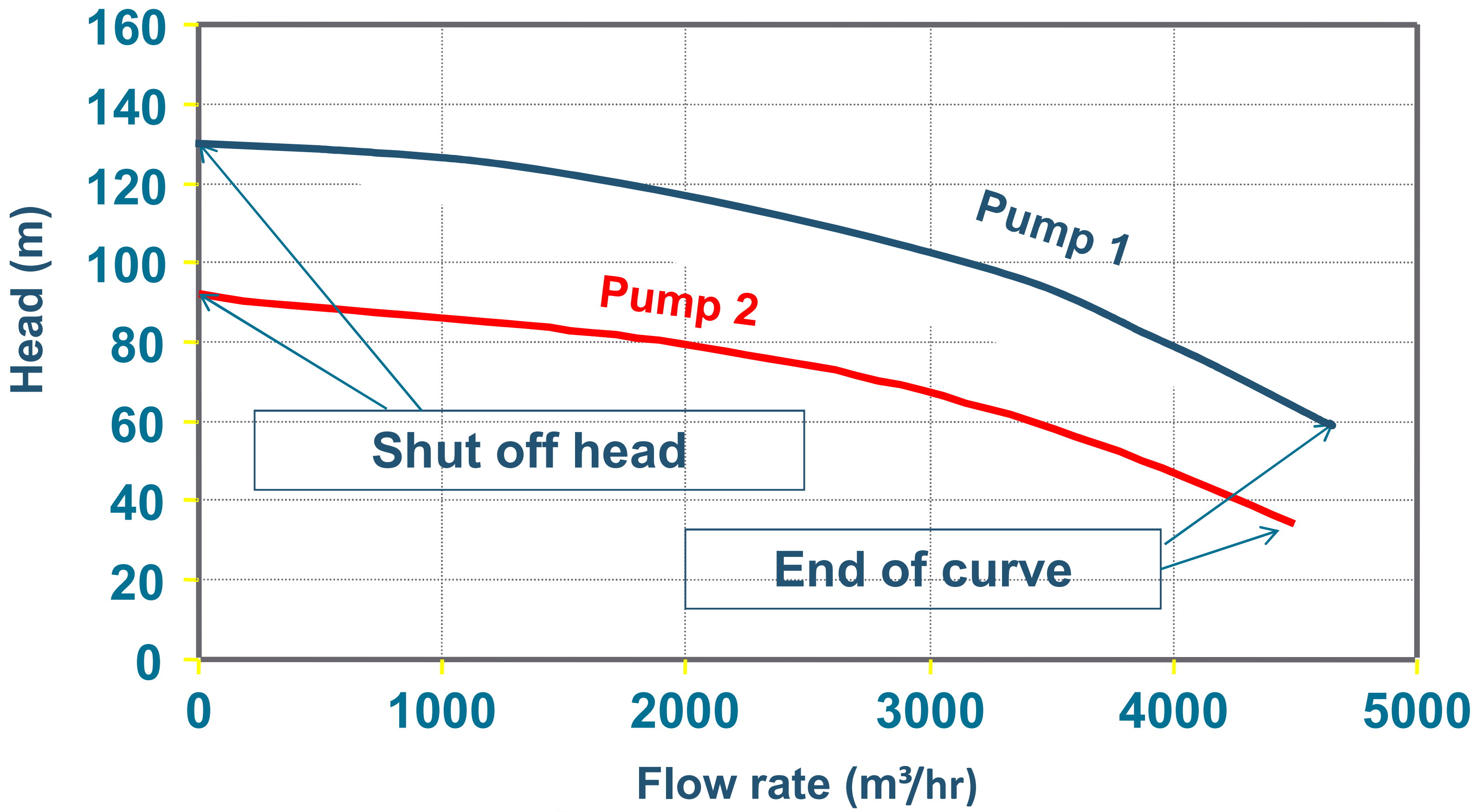


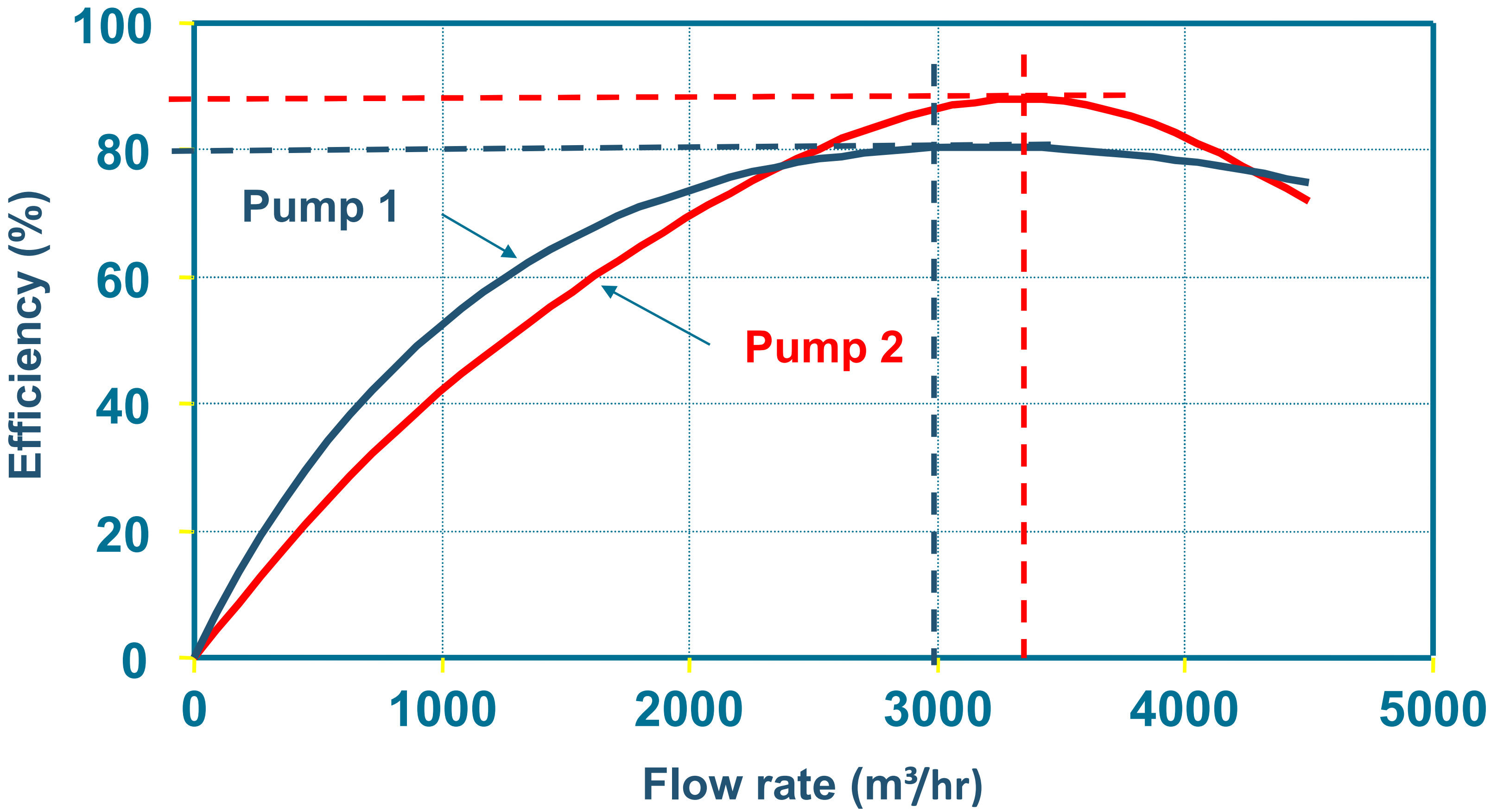
Figure Courtesy of Oak Ridge National Laboratory

Pump curve shapes vary

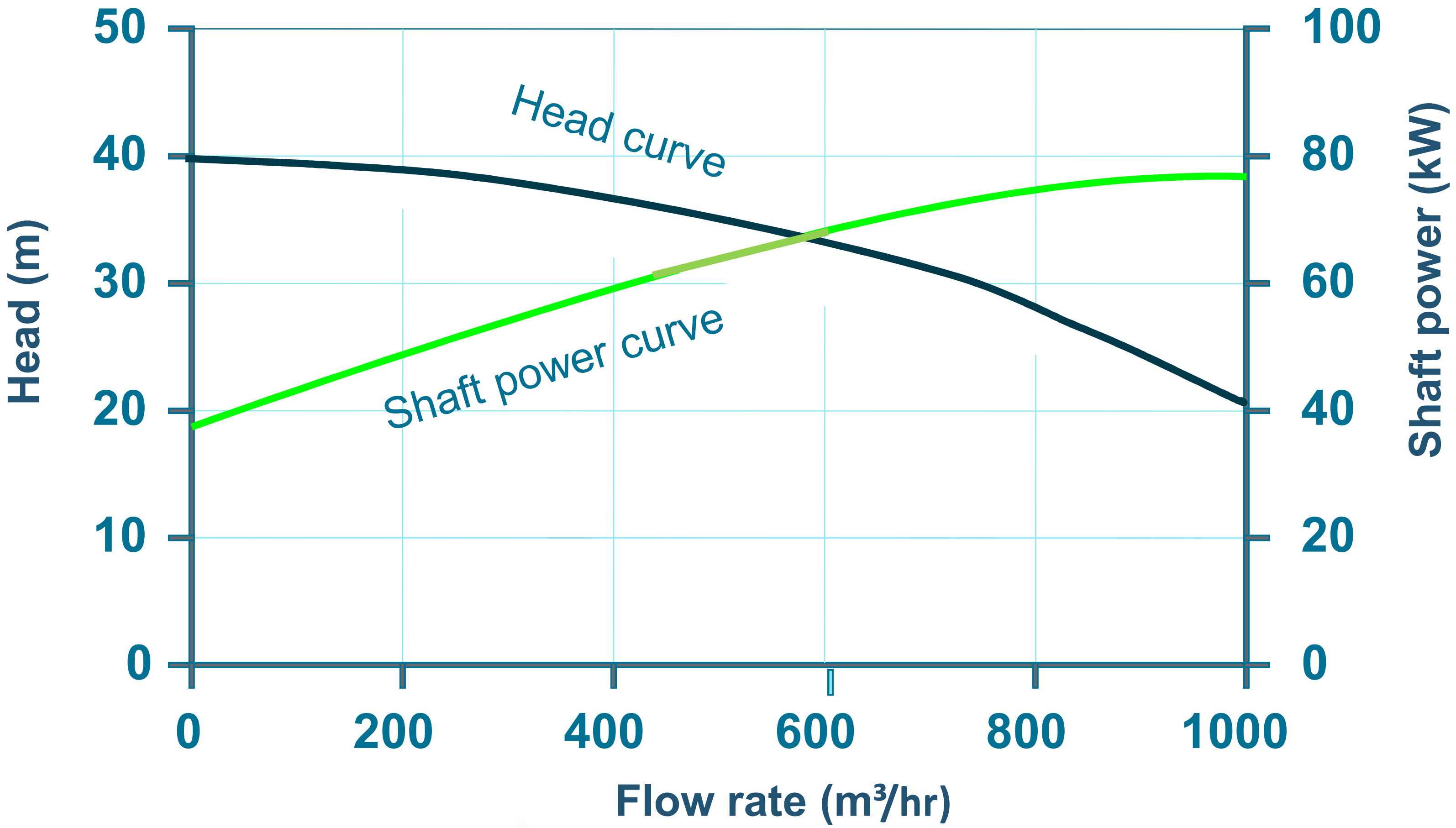
Head curves for two pump designs



Efficiency curves for the two pumps



Shaft power as a function of flow rate



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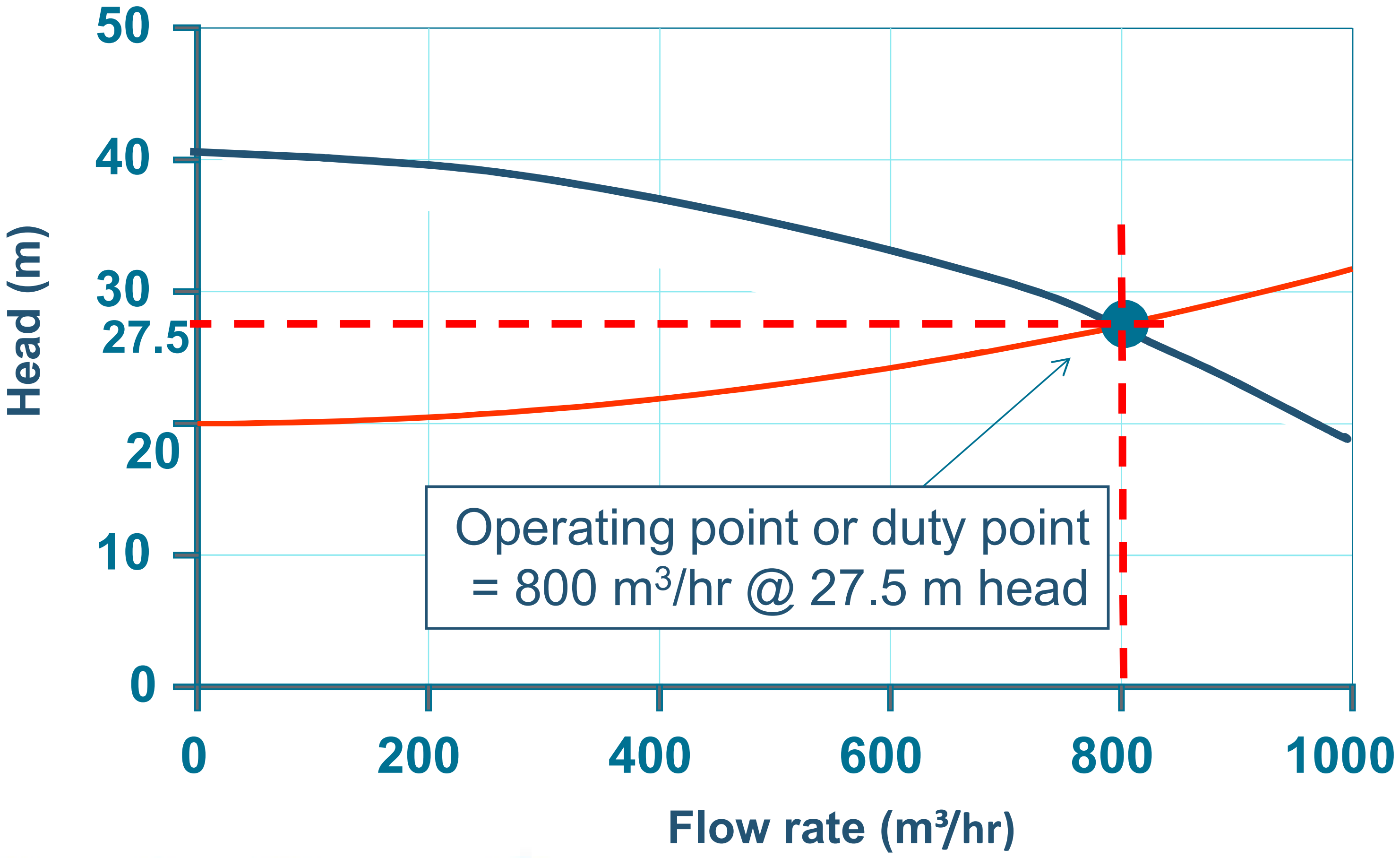
Operating point

The intersection between the Pump and System head capacity curves defines the operating point



Operating point or duty point
= 800 m^3/hr @ 27.5 m head

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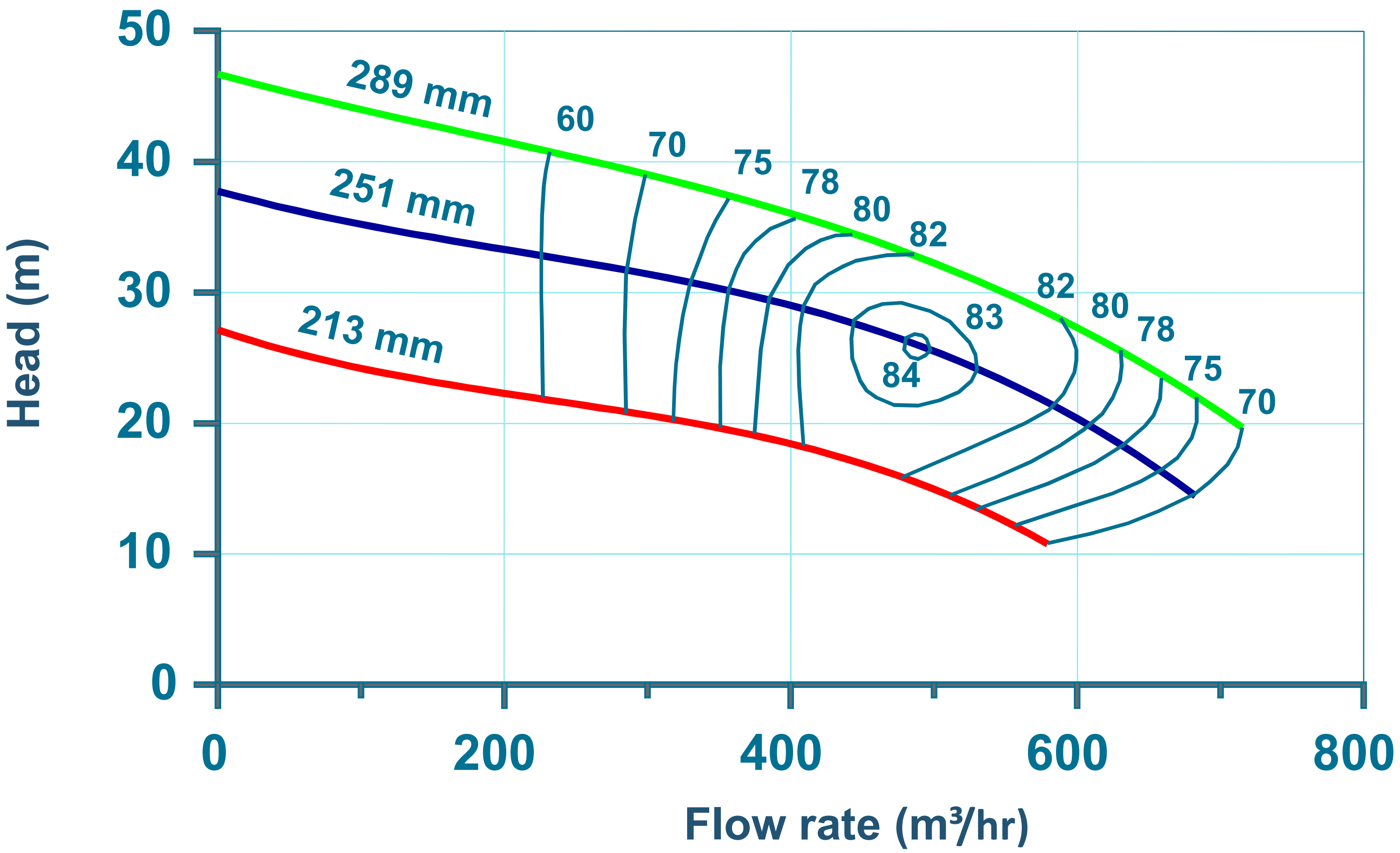
ISO-efficiency lines

ISO-efficiency lines are frequently overlaid onto head-capacity curves for multiple impeller diameters



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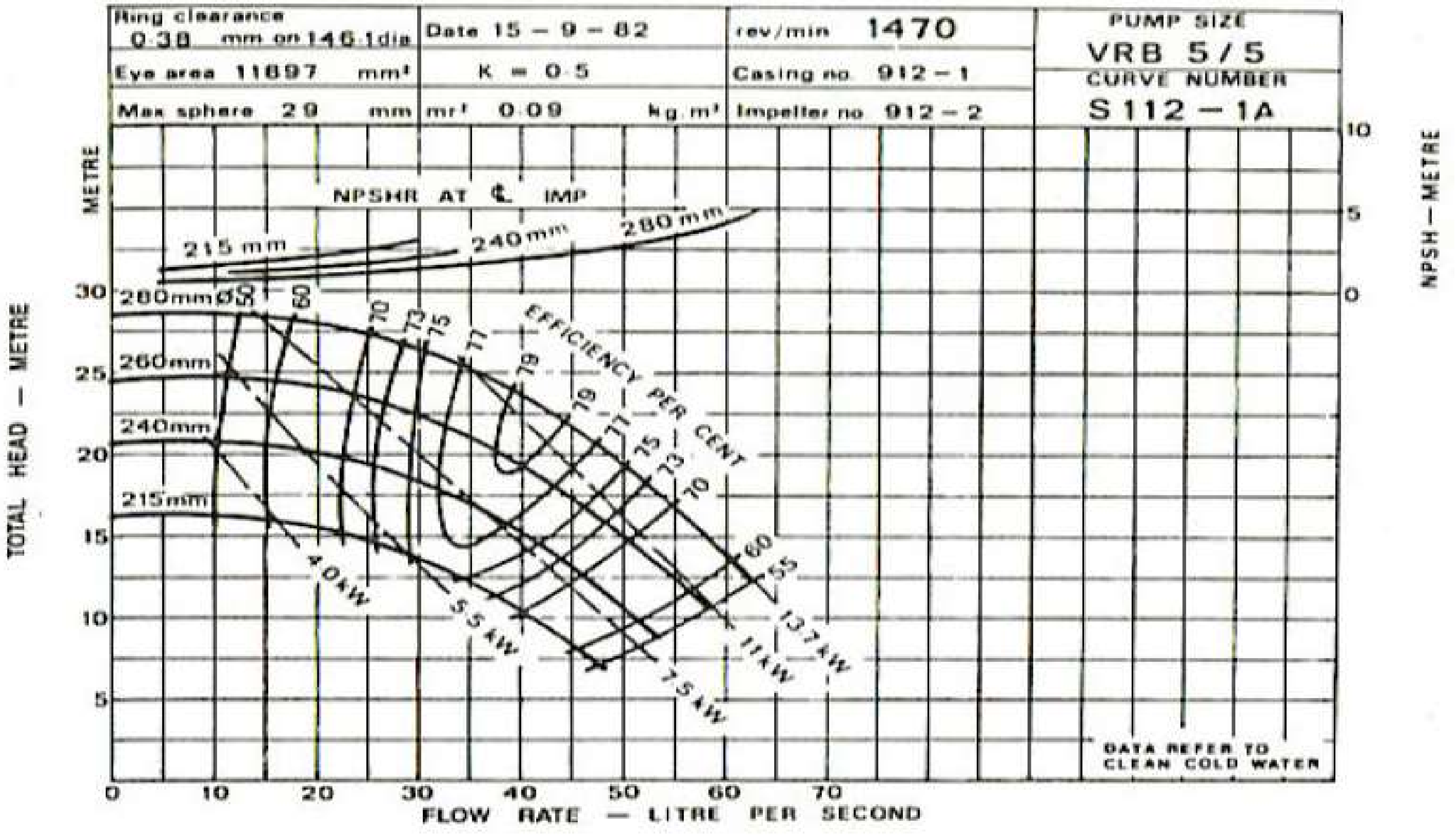
Typical performance curves for impeller trims

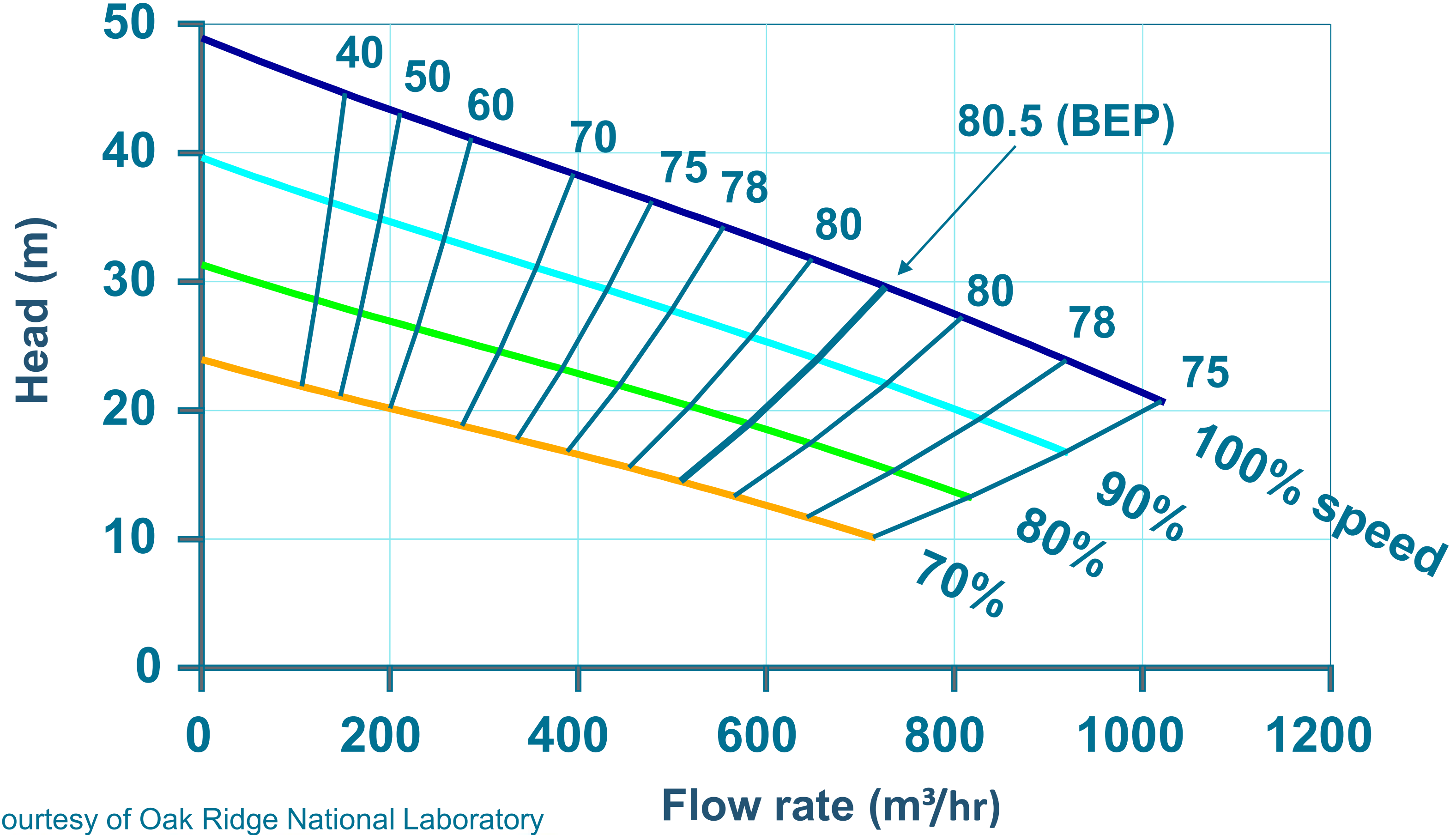


Ring clearance 0.38 mm on 146.1 dia	Date 15 – 9 – 82	rev/min 1470	PUMP SIZE VRB 5 / 5
Eye area 11897 mm²	K = 0.5	Casing no 912 – 1	CURVE NUMBER
Max sphere 29 mm	mr² 0.09 kg m²	Impeller no 912 – 2	S 112 – 1A

TYPICAL PERFORMANCE CURVES OF A CENTRIFUGAL PUMP AT 1470r.p.m

Speed changes

For speed changes, the efficiency
lines have a different pattern and all go through zero



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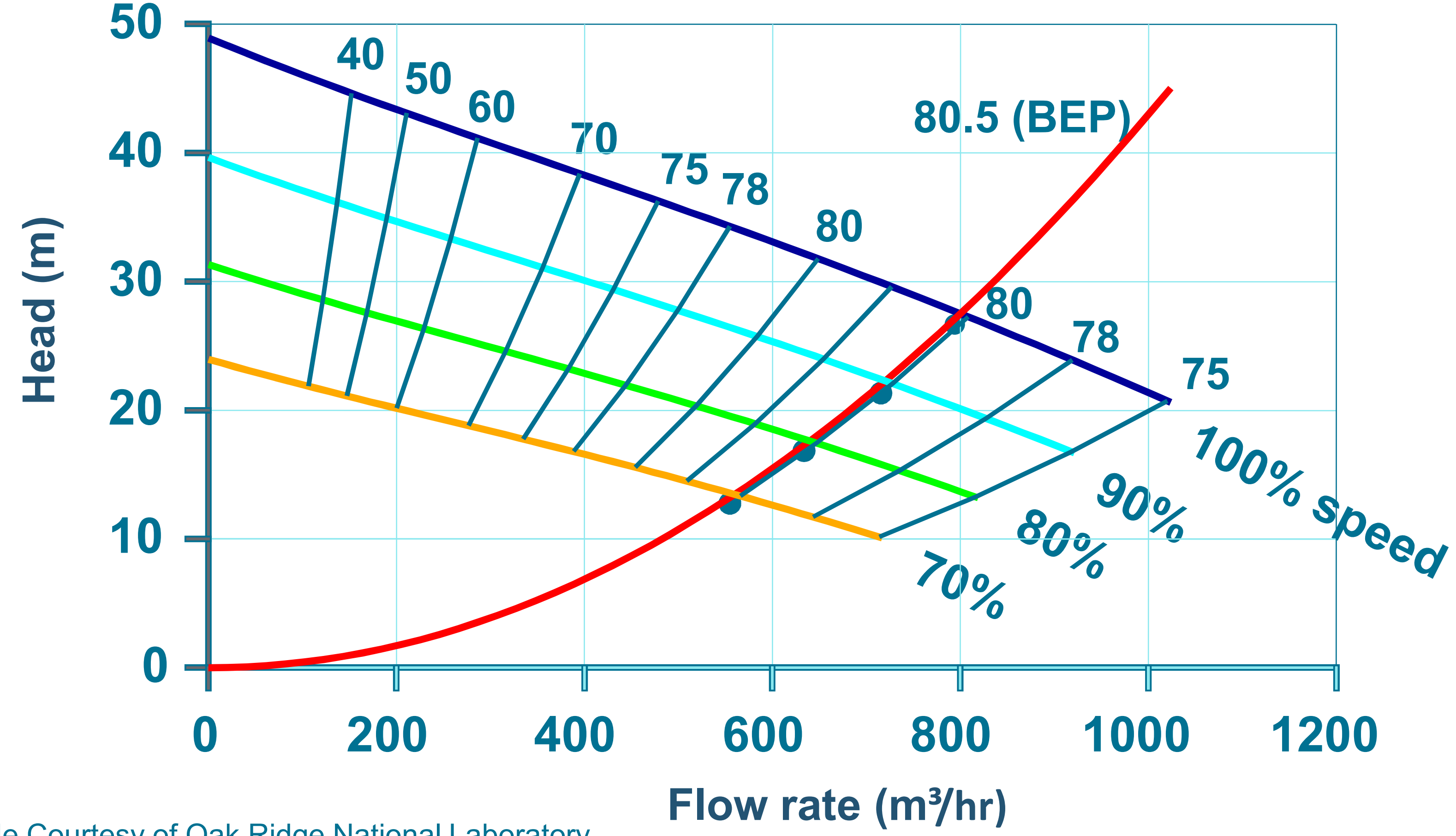
Lets review what happens if we operate a pump with reduced flow rate speed change, and with 3 different system curves

Change in Speed: All Frictional System

Change in speed for the All Frictional System results in maintenance of constant pump efficiency



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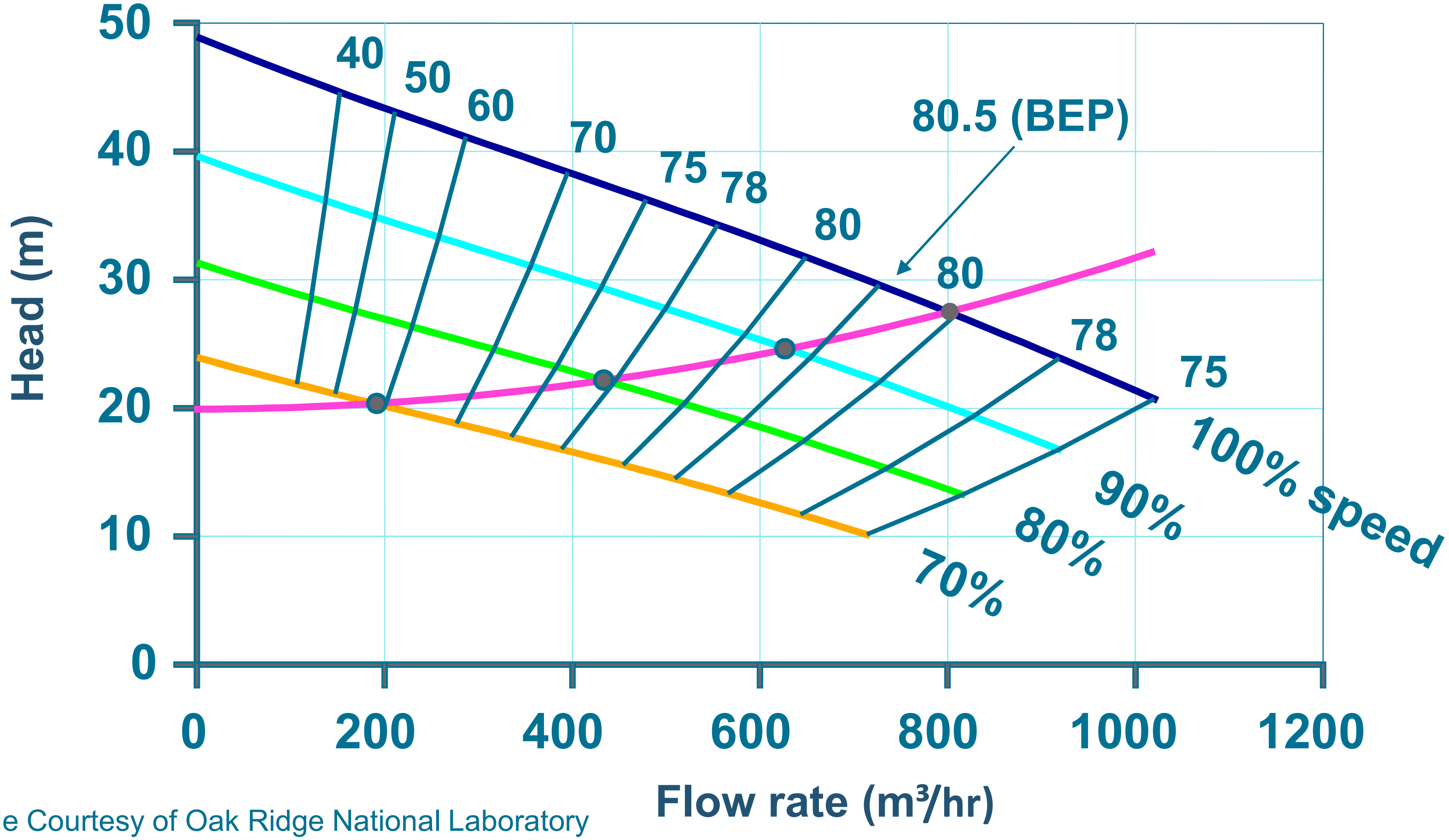
Change in Speed: Static & Frictional System

In a system with static head, pump efficiency *Does Not Remain Fixed* as speed changes



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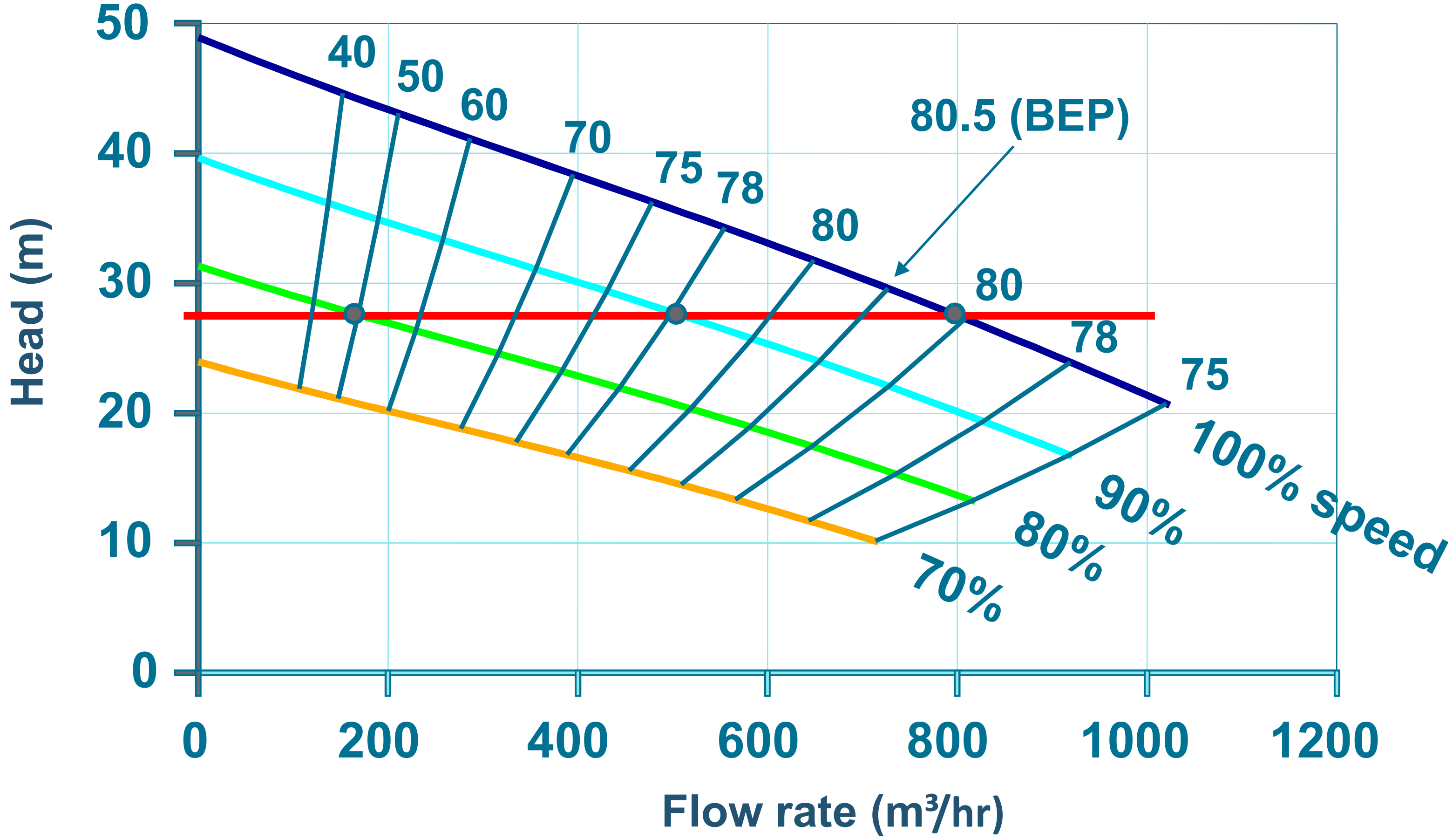
Change in Speed: All Static System

In a system with *ONLY Static Head*, the effect is even more dramatic



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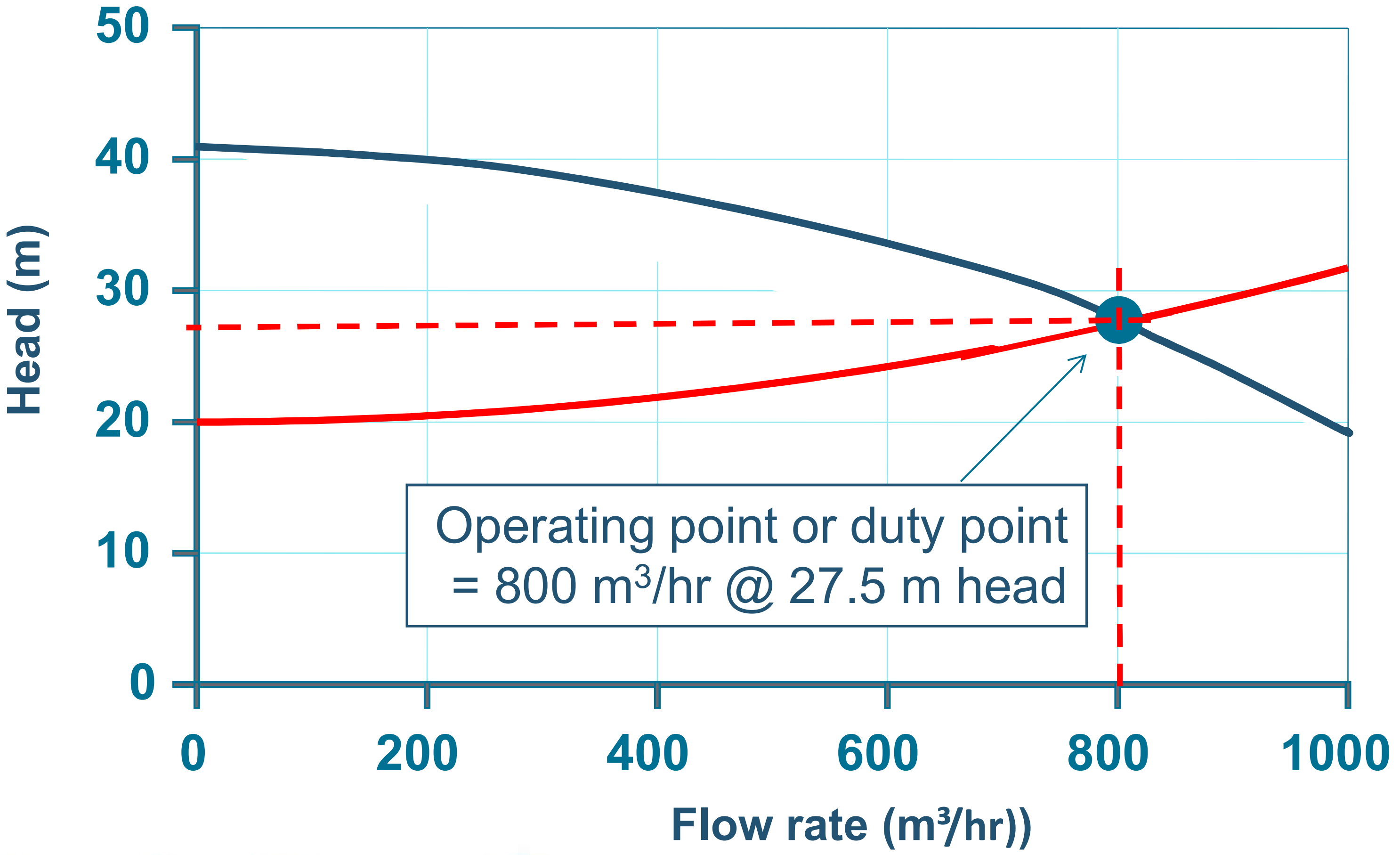
Now let's hook a Pump up to a System

Pump Operating Point

- The pump will ***always*** operate where the system and pump curves intersect
- At that point, we have balance between what the system demands and what the pump can deliver.

Pump Operating Point

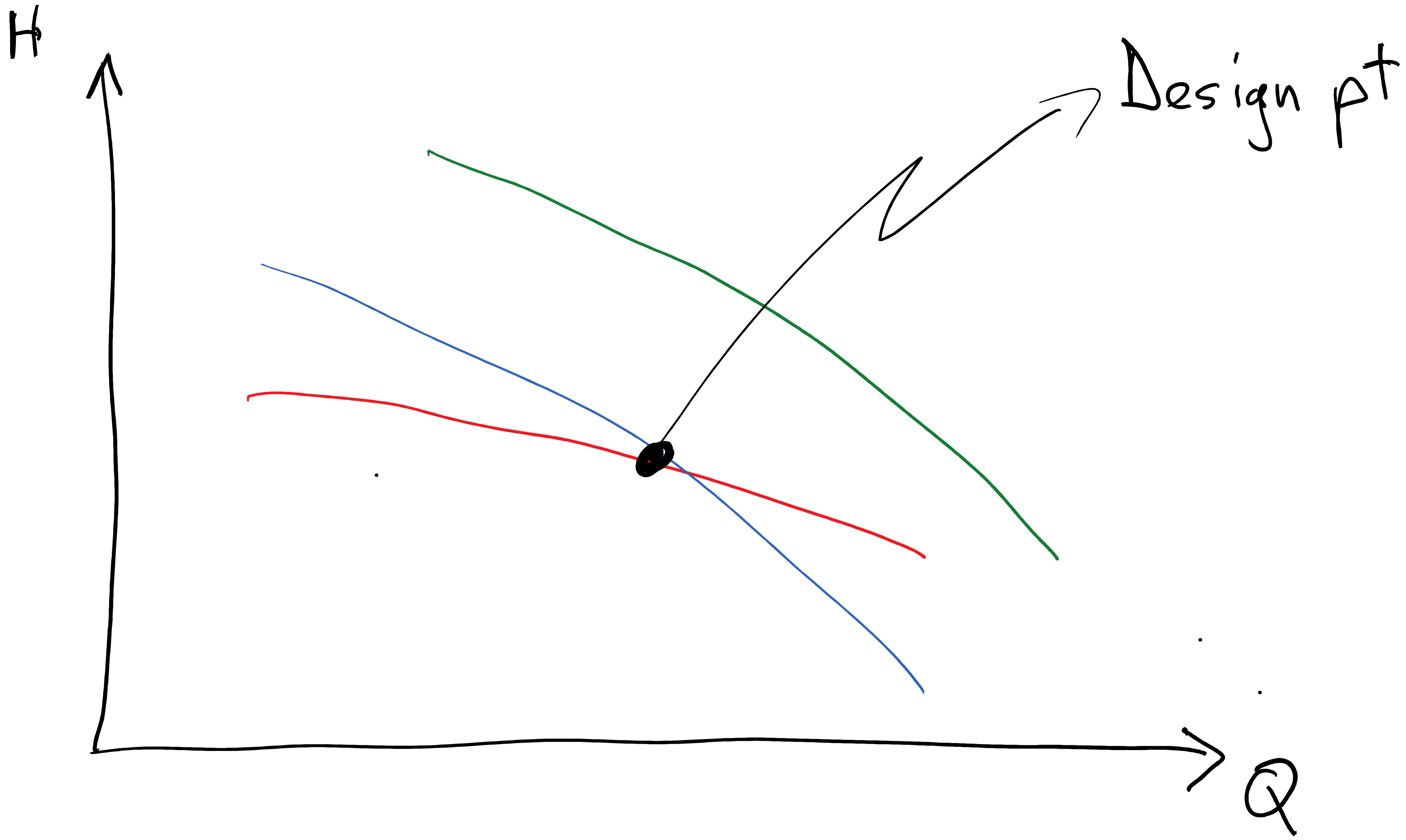
The intersection between the pump and system head capacity curves defines the operating point



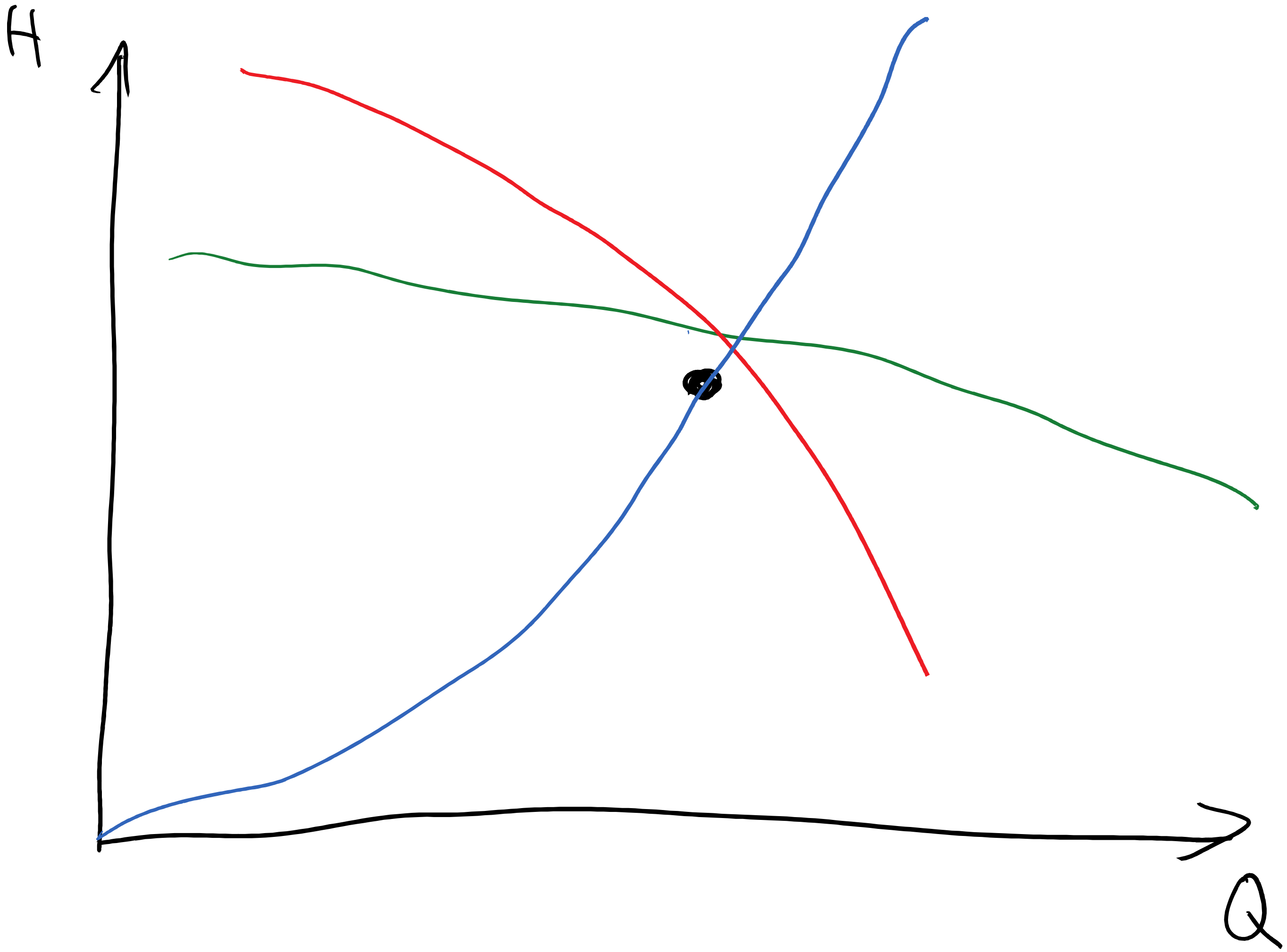
Operating point or duty point
= 800 m^3/hr @ 27.5 m head

Slide Courtesy of Oak Ridge National Laboratory

Which pump to buy?



Which pump to buy?



Which pump to buy?

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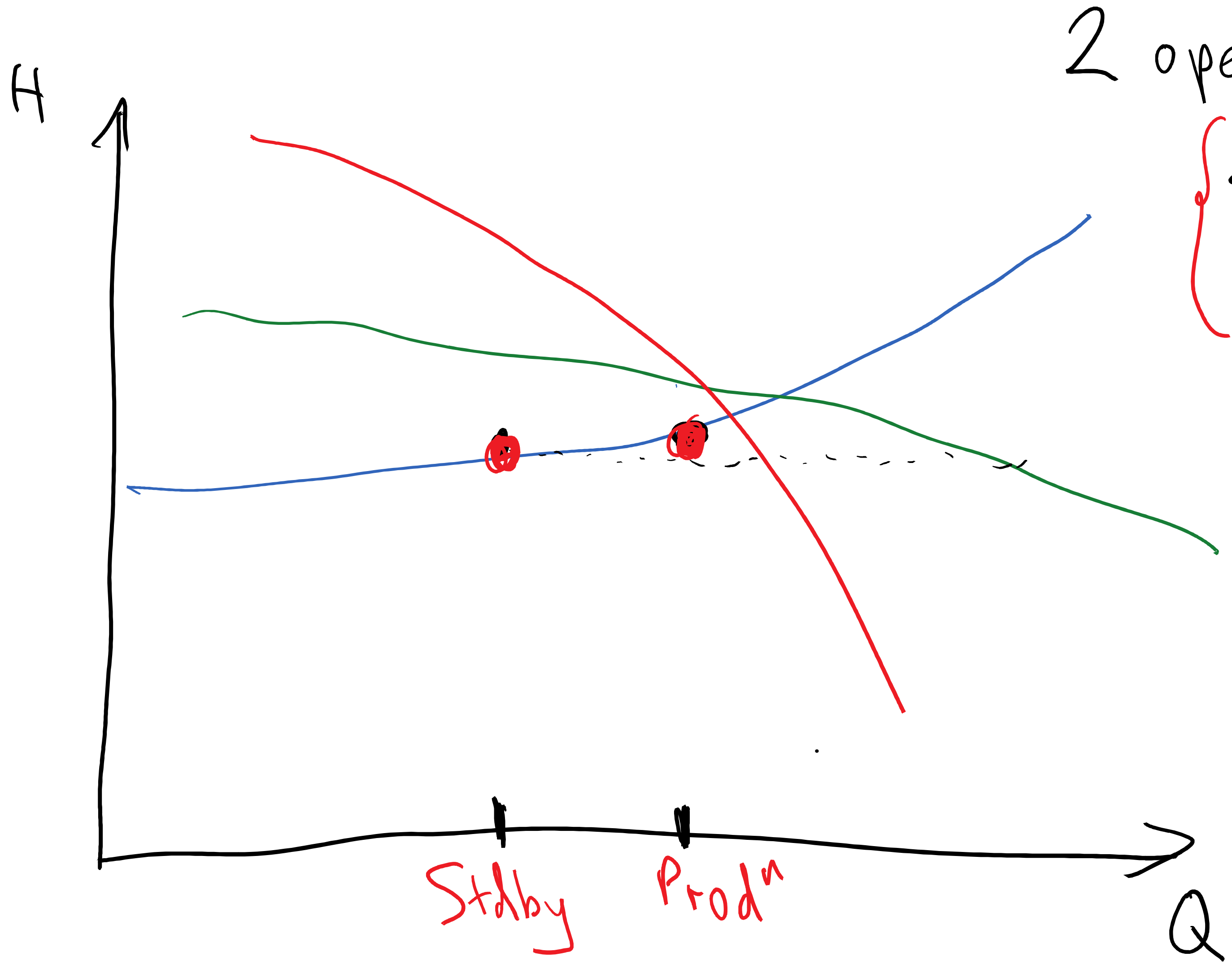
2 operating pts
STDBY
PRODⁿ



Affinity Laws

Affinity Laws

Pump Affinity Laws can be used to predict pump curves for different Speeds and Impeller Diameters

Speed

$$\left(\frac{Q_1}{Q_2}\right) = \left(\frac{N_1}{N_2}\right)^1$$

$$\left(\frac{H_1}{H_2}\right) = \left(\frac{N_1}{N_2}\right)^2$$

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{N_1}{N_2}\right)^3$$

Diameter

$$\left(\frac{Q_1}{Q_2}\right) = \left(\frac{D_1}{D_2}\right)^1$$

$$\left(\frac{H_1}{H_2}\right) = \left(\frac{D_1}{D_2}\right)^2$$

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{D_1}{D_2}\right)^3$$

Q = flow rate H = head P = power N = speed D = diameter

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Affinity Law Exercise 1 (rookie)

A rebar quenching centrifugal pump supplies water at 210 m^3/hr, and the motor electrical load is 25 kW. If the supply was reduced to 150 m^3/hr what would the new power consumption be expected to be?

A. 17.85 kW
B. 12.7kW
C. 9.1 kW
D. 7.6 kW

$P = [(150)/(210)]^3 \times 25$

Affinity Laws Exercise 2 (amateur)

A 3-phase motor is driving a centrifugal pump to pump water around a closed circuit. The power demand on the motor is 17.3kW. If we install a VSD to reduce the speed of the motor and the flow of water around the circuit is reduced by 25%, what is the likely reduction in power demand?

A. 4.33 kW
B. 7.3 kW
C. 10 kW
D. 11.2 kW

Old P = 17.3 kW, new P = 0.75 x 0.75 x 0.75 x 18.3 kW = 7.298 kW. Reduction in power demand is 18.3kW - 7.3 kW

Affinity Laws Exercise 3 (professional)

Egyptian program for promoting Industrial Motor Efficiency
SAVE TODAY ... POWER TOMORROW

A 75kW motor pumping water around a closed circuit is fully loaded. We reduce the speed of the pump using a VSD and the pressure difference across the pump (read by reading the suction and discharge gauges fitted) is seen to reduce to 0.36 of the original pressure difference. What can we say about the new flow rate and the expected power savings (assume motor efficiency is 95% and constant over the range used)

A. 16.2 kW
B. 58.8 kW
C. 61.9 kW
D. 66.3 kW

New pressure is 0.36 of old pressure.
Thus (0.36/1) = (new flow rate / old flow rate)^2.
Thus new flow rate = 0.6 of the old flow rate.
Thus $kW_{new}/kW_{old} = (0.6/1)^3$.
Thus kW_{new} = 21.6% of the old power consumption.
New power consumption = 0.216 x 75/0.95 = 17.05 kW.
Thus power saving is 78.95 – 17.05

Affinity Laws Exercise 4 (grandmaster)

Egyptian program for promoting Industrial Motor Efficiency
SAVE TODAY ... POWER TOMORROW

A system fed by a pump has a static head of 3m and a friction head of 10m at a flow of 100 m^3/h. What is the electrical power consumption of the pump system at a flow of 60m^3/h? The pump is driven directly and at 60 % flow the motor efficiency is 91% and the pump efficiency is 78%.

a) 1.1 kW b) 1.5 kW c) 2.7 kW d) 5.2 kW

100 m^3/hr = 27.77 lps
kW_{fluid} = QHp / 102 = (27.77 x 13 x 1) / 102 = 3.54 kW_{fluid}
Affinity law: $(Q_{old}/Q_{new})^3$ x kW_{old} = kW_{new} = 0.77kW_{fluid}
kW_{elec} = 0.77kW_f / (91% x 78%) = 1.1 kW_e

Considerations for Affinity Laws

- It's fine to use the affinity laws to explore the possibilities with impeller trimming for better pump and system matching, but don't get carried away. Get **actual** performance curves from the manufacturer, especially if the trim change being considered is large.

- The affinity laws will generally ***not tell you*** where on the curve the pump will operate or give you correct estimates of possible energy savings, *except for systems* ***without static head***

Parallel and Series Pumps

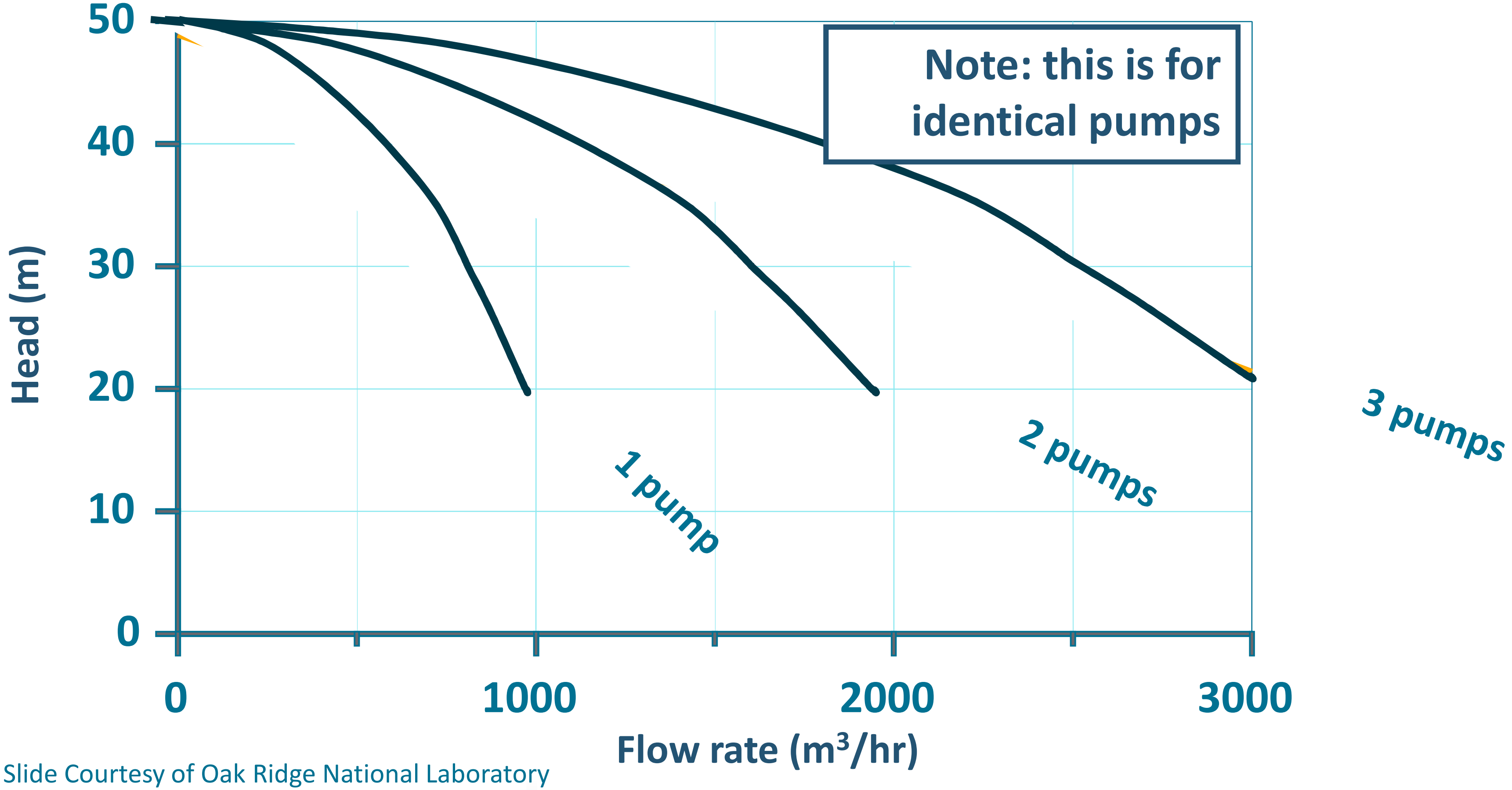
Parallel and Series pumping "laws", like the pump affinity laws apply to the *pump curves only*

Parallel pumps - **Sum** the **Flow** rates at a given head

Series pumps - **Sum** the **Heads** at a given flow rate

Parallel Pumps

Parallel pumps can help adapt to changing system requirements and provide redundancy



Note: this is for identical pumps

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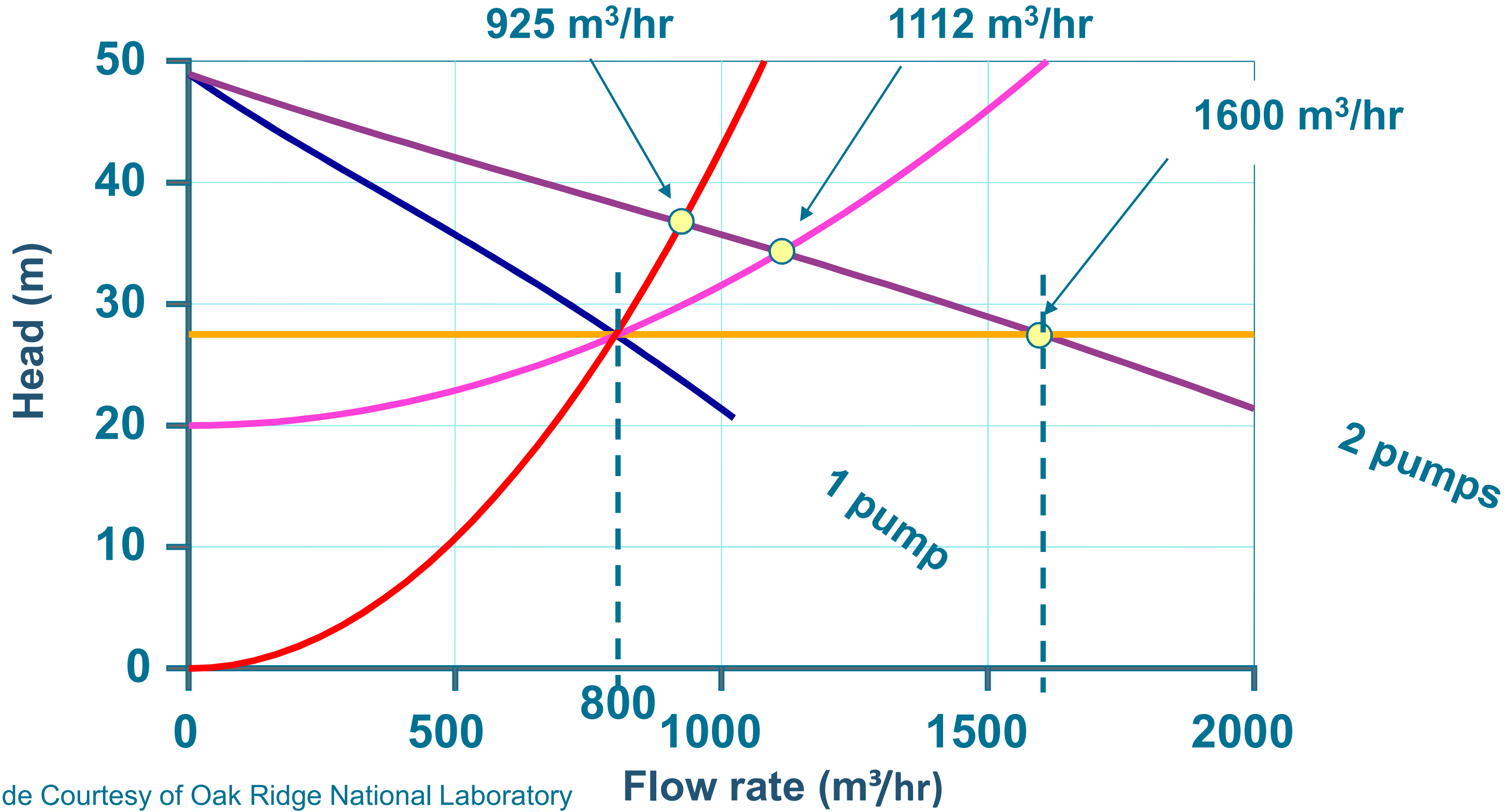
How about Parallel pump operation with different System types?

Parallel Pumps

The effect of turning on a parallel pump also depends on the nature of the system



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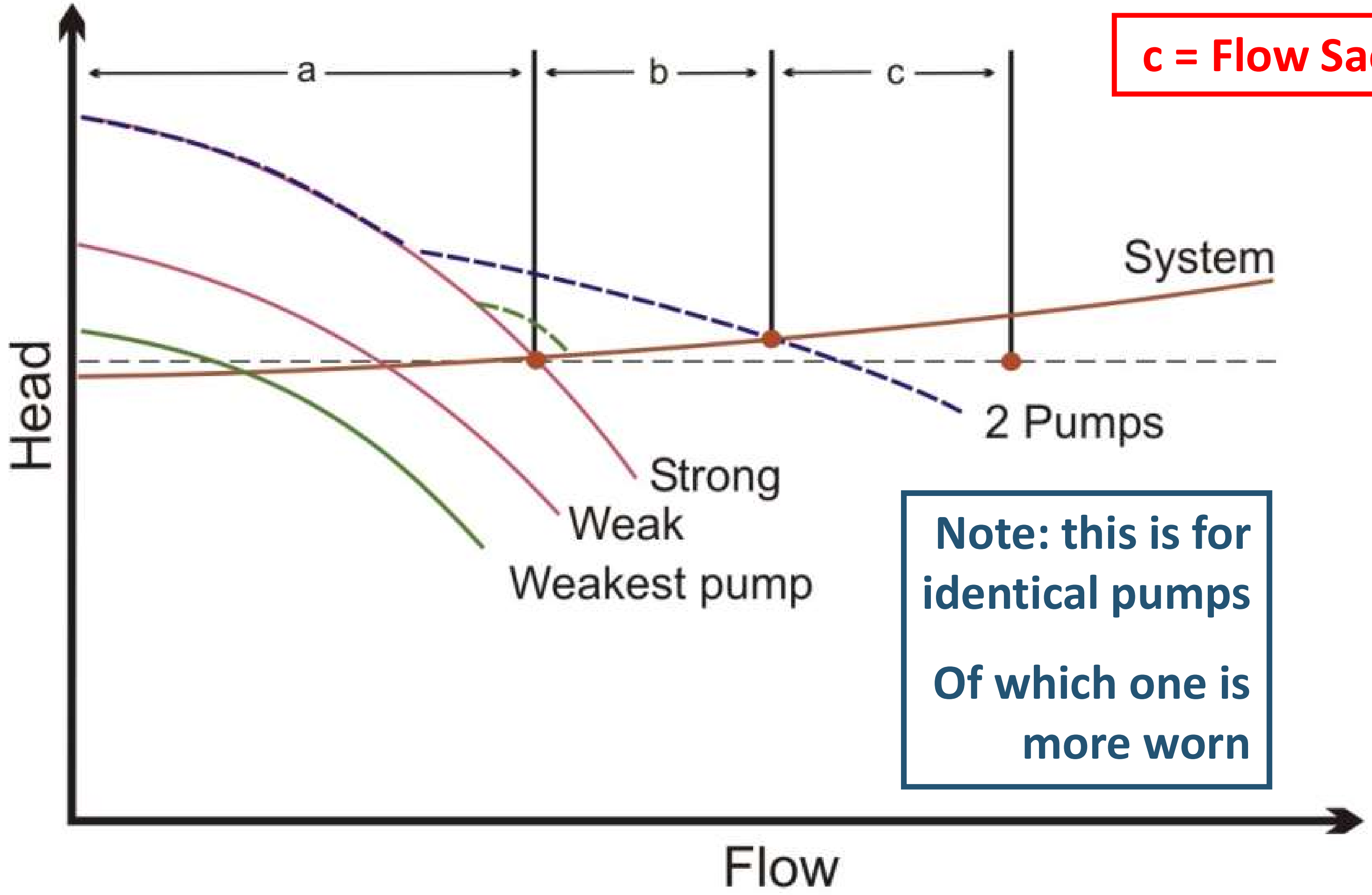
Parallel Pumps

Effect of wear on pumps in parallel
One pump will dominate the other

c = Flow Sacrifice

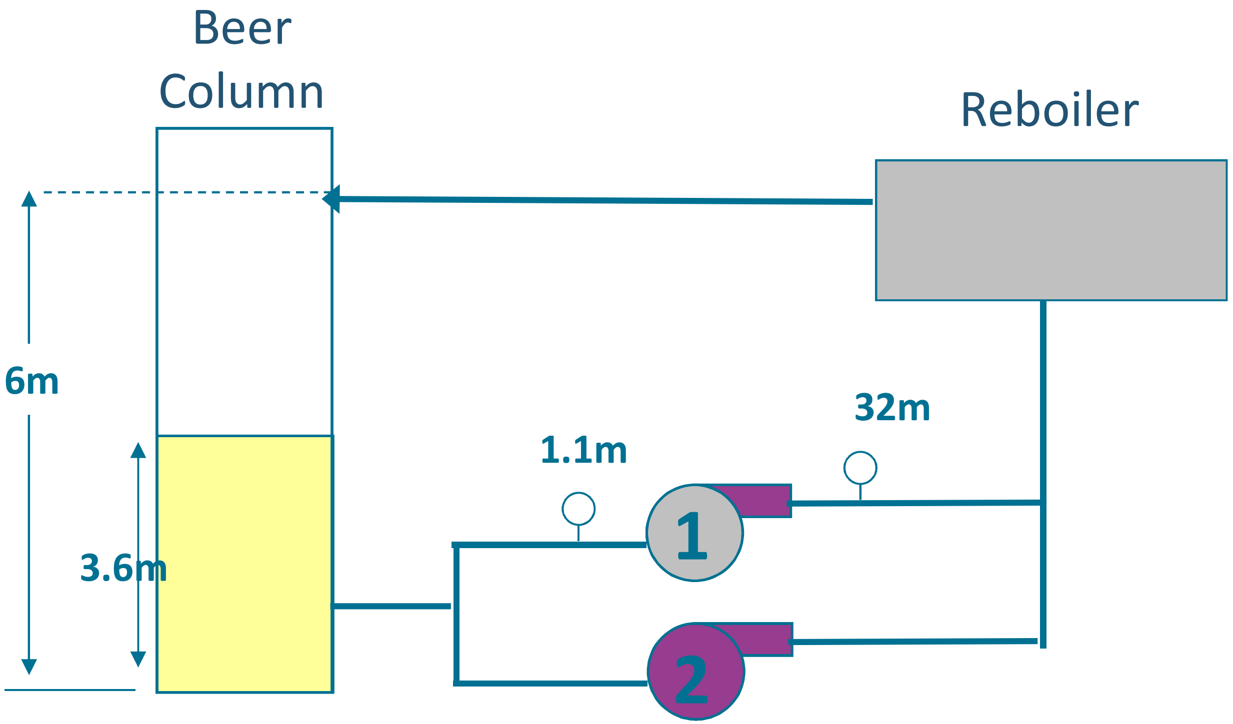


Note: this is for identical pumps

Of which one is more worn

Pumps in Parallel
Ethanol Plant Example

Reboiler Pumps #1 and #2



BC Boiler Pump measured Power:

#1 = 119.6 kWe

#2 = 120.4 kWe

(kWe = Electrical Power)

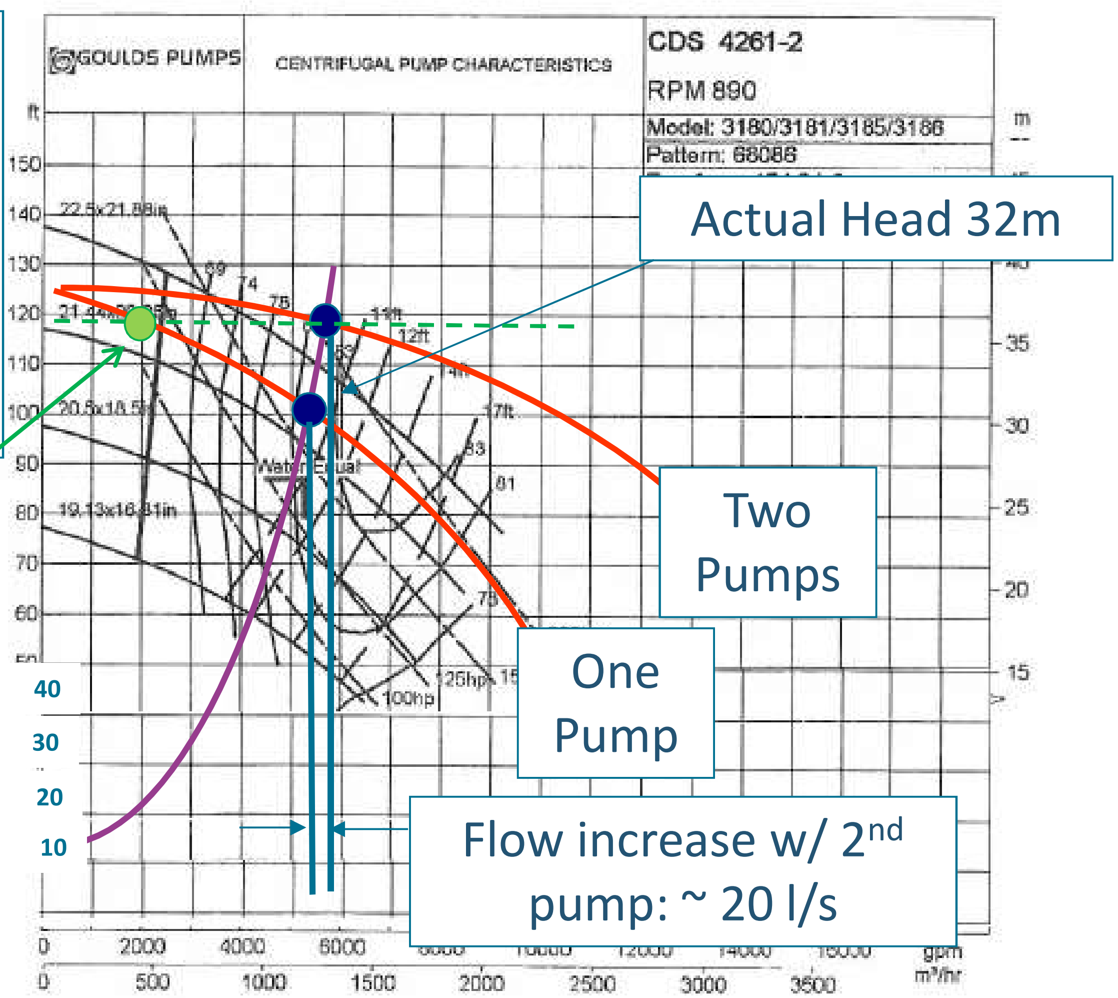
Energy Cost for Pump #2 : 120.4 kWe x 8 500 hours

= 1 023 400 kWh @ EGP1.00/kWh = EGP 1 023 400/ year

Reboiler Pumps #1 and #2

Operating two pumps instead of one only increases flow by 6% in this case, *but increases system annual energy costs by EGP 1 023 400*

Efficiency point of each pump when 2 are running



Actual Head 32m

Two Pumps

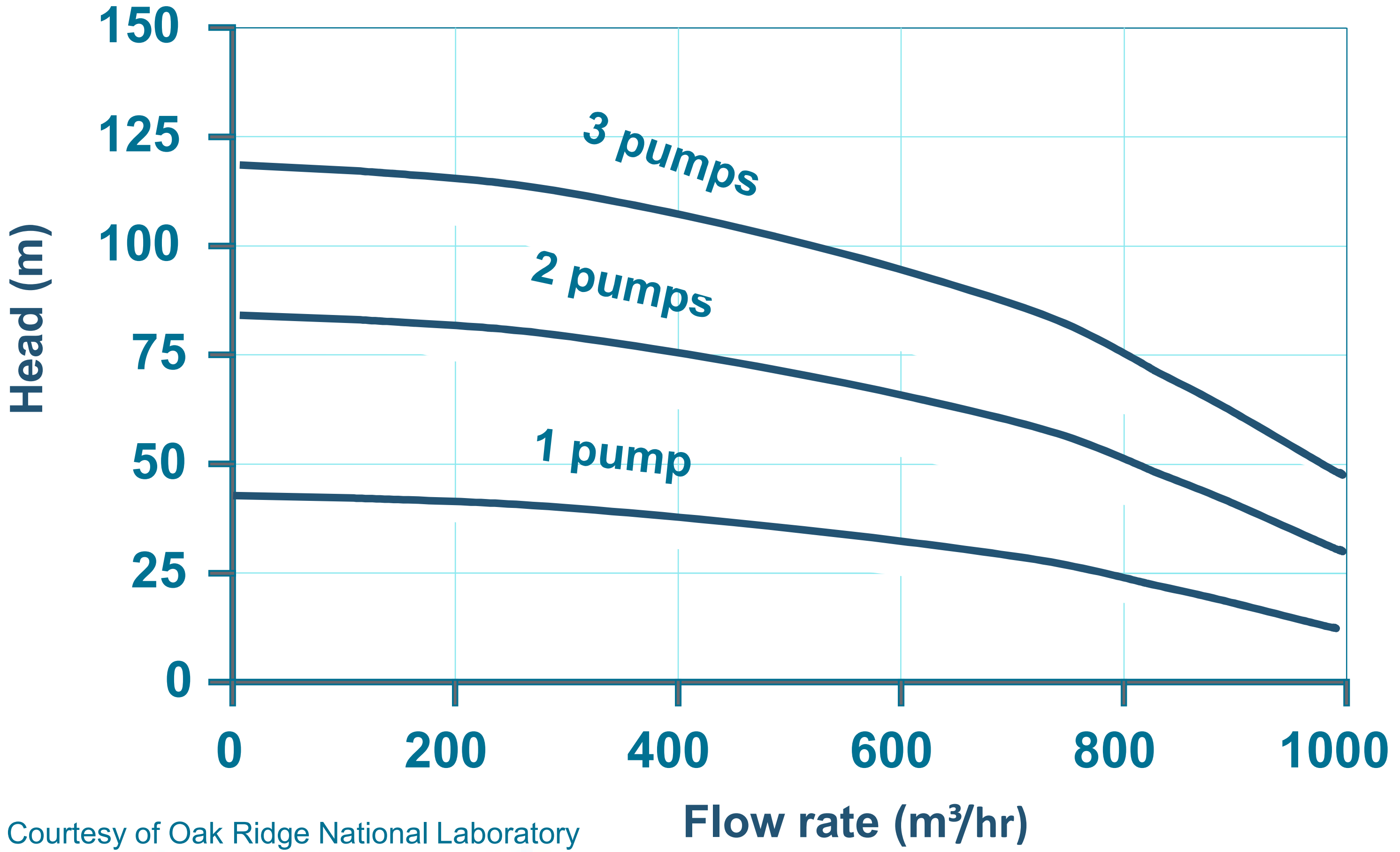
One Pump

Flow increase w/ 2nd pump: ~ 20 l/s

Pumps in Series

Pumps in Series

Identical pumps in series Add Head at a given flow rate to estimate overall performance



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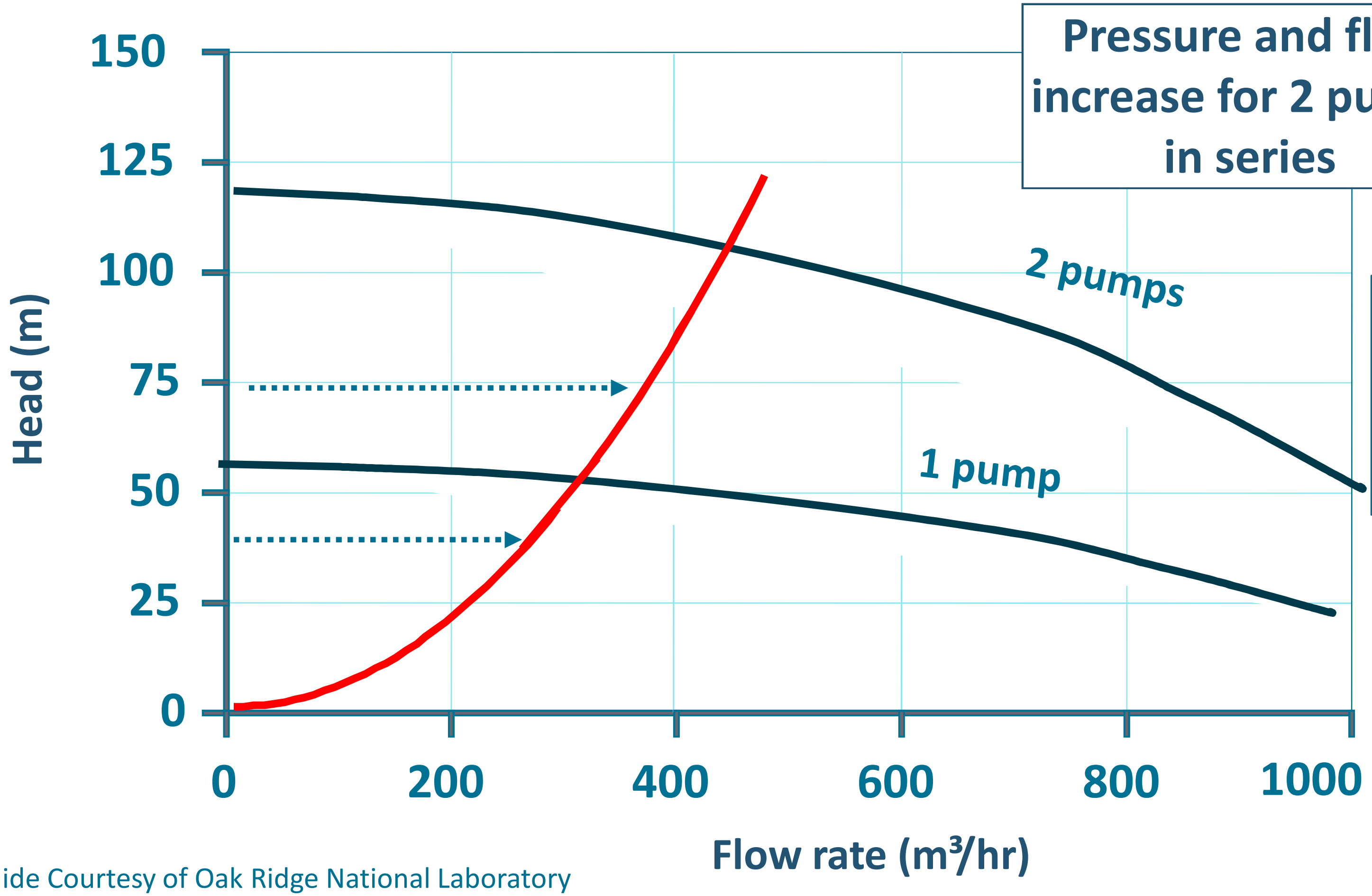
Pumps in Series

Two identical pumps in series with system curve



Pressure and flow increase for 2 pumps in series

Best result with all friction system

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6. Pump System Energy Use

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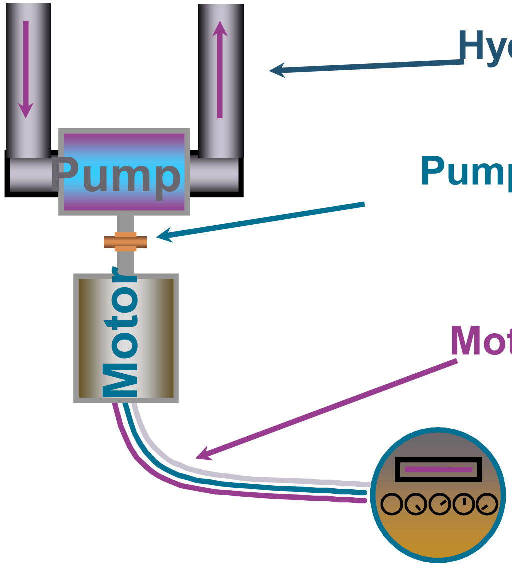
Power Used by a Pump



$$\text{Hydraulic power } (kW_{hyd}) = \frac{\text{l/s x m x relative density}}{102}$$

$$\text{Pump input power } (kW_{mech}) = \frac{\text{Hydraulic power}}{\text{Pump efficiency}}$$

(*or motor output*)

$$\text{Motor input power } (kW_{elec}) = \frac{\text{Motor output power}}{\text{Motor efficiency}}$$

Motor Operating Cost = Motor input power x Operating hours x per unit electricity cost

Slide Courtesy of Oak Ridge National Laboratory

Expanding the equation…

$$kW_{elec} = \frac{\text{Flow (l/s)} \times \text{Total Head (m)} \times \text{Relative Density}}{102 \times \eta_p \times \eta_m \times \eta_{vsd}}$$

Flow

Total Head

System-level Opportunities

η_p = pump efficiency

η_m = motor efficiency

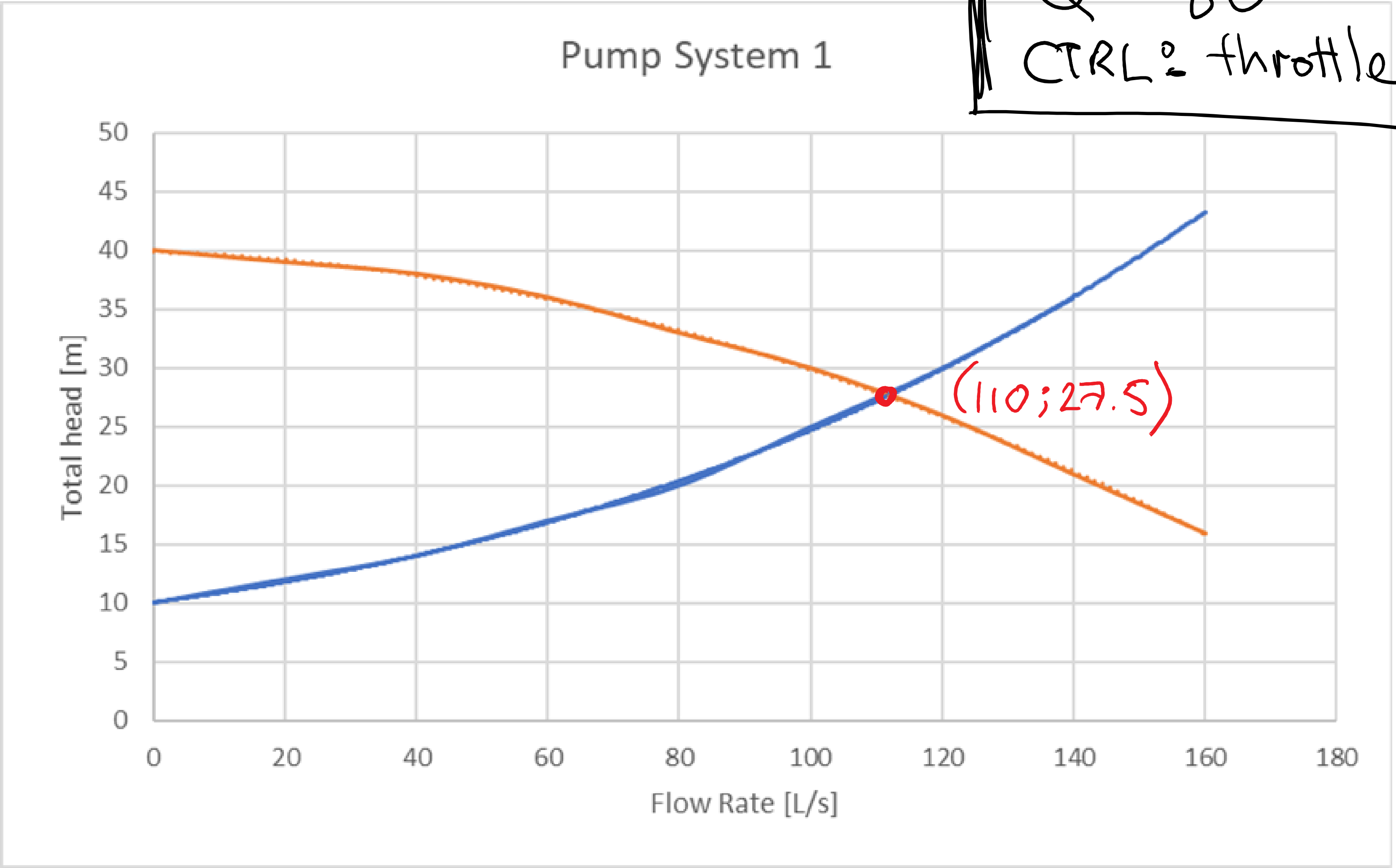
η_{vsd} = VSD efficiency

Component-level Opportunities

kWh = kW x Hours

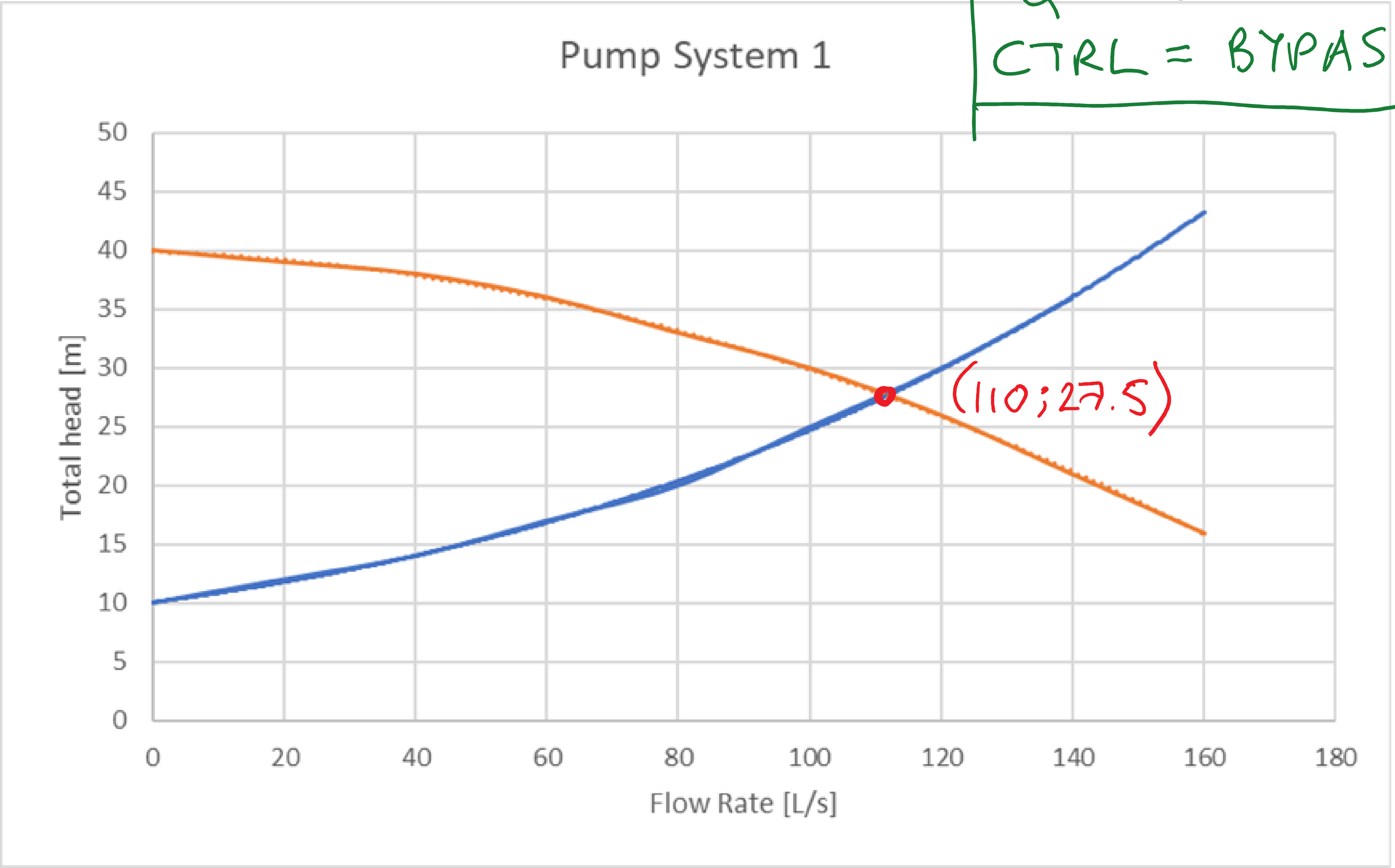
Estimate Power Losses

Q ≈ 80
CTRL: throttle



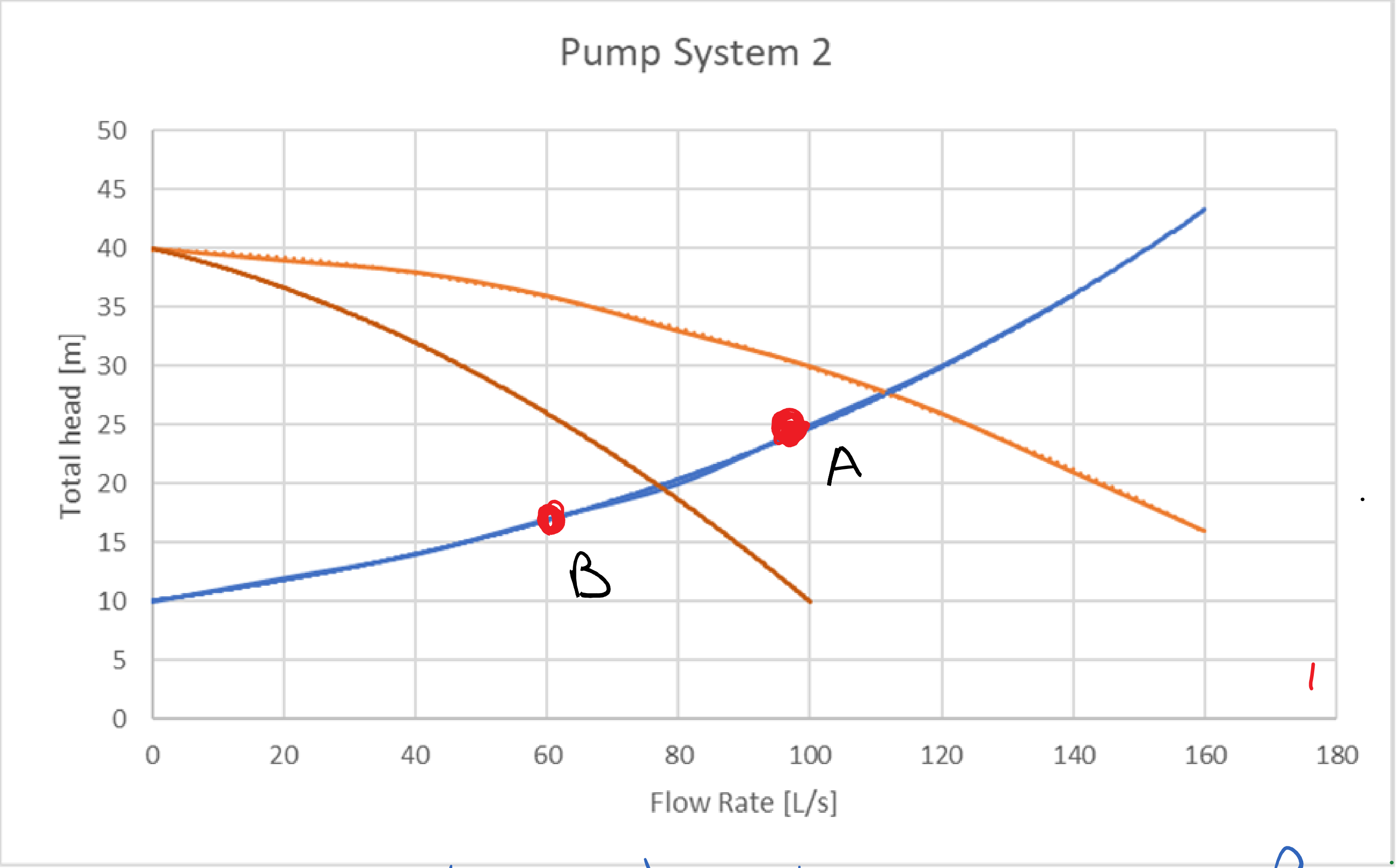
Estimate Power Losses

Q = 80
CTRL = BYPASS



How to Control?

2 OPERATING CONDITIONS : PRODUCT A & PRODUCT B



What control philosophy should I use?

7. ASME EA-2-2009

Energy Assessment Standard for Pumping Systems

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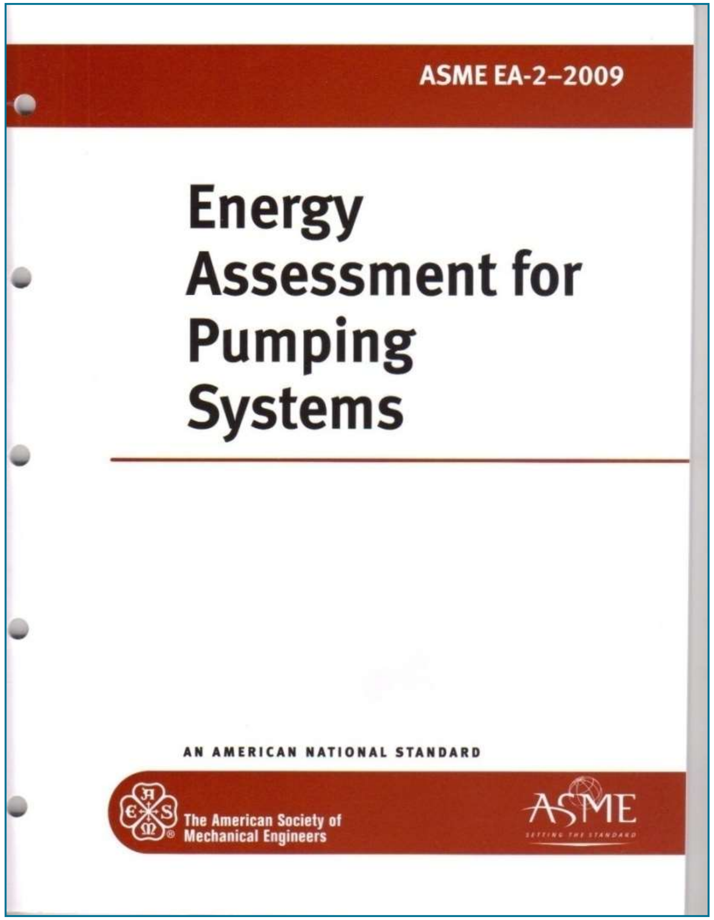
ASME Standards & Guides



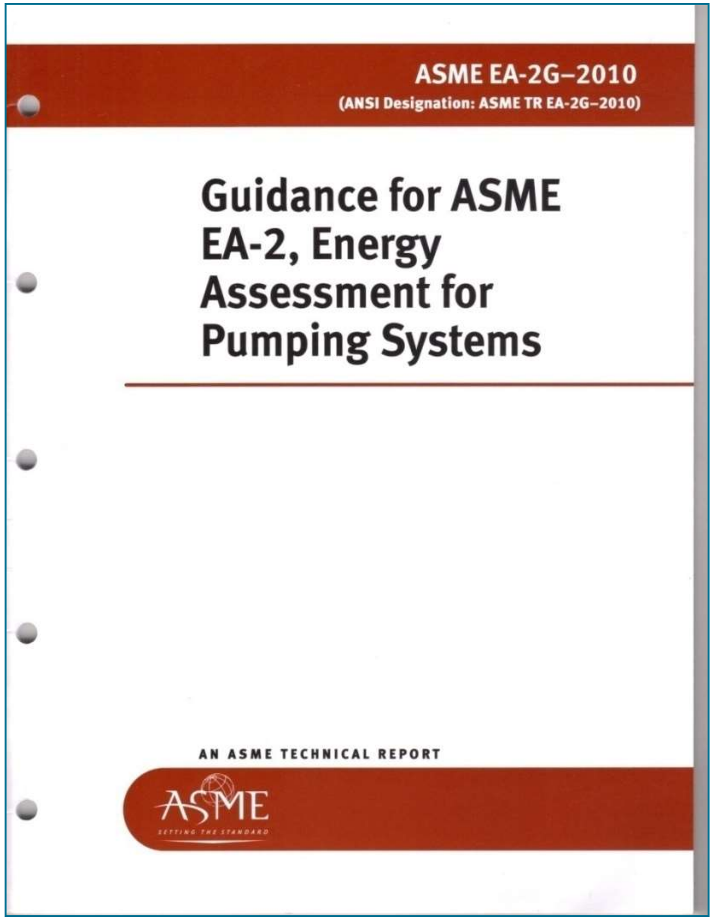
ASME EA-2–2009

Energy Assessment for Pumping Systems

AN AMERICAN NATIONAL STANDARD

The American Society of Mechanical Engineers

ASME SETTING THE STANDARD



ASME EA-2G–2010
(ANSI Designation: ASME TR EA-2G–2010)

Guidance for ASME EA-2, Energy Assessment for Pumping Systems

AN ASME TECHNICAL REPORT

ASME SETTING THE STANDARD

Standard vs Guide

Standard EA-2-2009

- Provides a common understanding of what should be included in a pump system assessment to replace the lack of a standardization for pump systems previously evaluated as part of an energy evaluation, audit, survey or energy study.
- Defines specific requirements that must be performed for different assessment levels.

Guidance Document EA-2G-2010

- Provides technical background and application details to help the user apply the standard.
- Includes rational for the technical requirements, application notes, alternative approaches, tips, techniques and examples.

Objectives of Pump Standard/Guidance Documents

- Provide a step by step approach to perform a pump system energy assessment.
- Identify energy assessment levels and the effort required for each type of assessment.
- Emphasize the importance of taking a systems approach.
- Review equipment data that should be collected for pump system evaluations.
- Become familiar with solutions for pump system optimization.
- Present the results in a suitable format.

Standard/Guidance Document Sections

ASME EA-2-2009 Energy Assessment Pump Systems Sections:

1. *Scope & Introduction*
2. *Definitions*
3. *References*
4. *Organizing the Assessment*
5. *Conducting the Assessment*
6. *Analyzing the Data*
7. *Reporting & Documentation*

Areas to be discussed

8. ASME Chapter 6

Analysing the data

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6. Analysis from the Data of the Assessment

6.1 Common Causes and Remedies for Excessive Energy Use

6.1.1 Reduce System Head

6.1.2 Reduce System Flow Rate

6.1.3 Ensuring that Components Operate Close to BEP

6.1.4 Change Pumping System Run Time

6.2 Basic Energy Reduction Opportunity Calculations

6.2.1 Comparing Existing and Optimal Energy Use

6.2.2 Excess System Energy Use

Solutions to Excessive Energy Use

Reduce System Head:

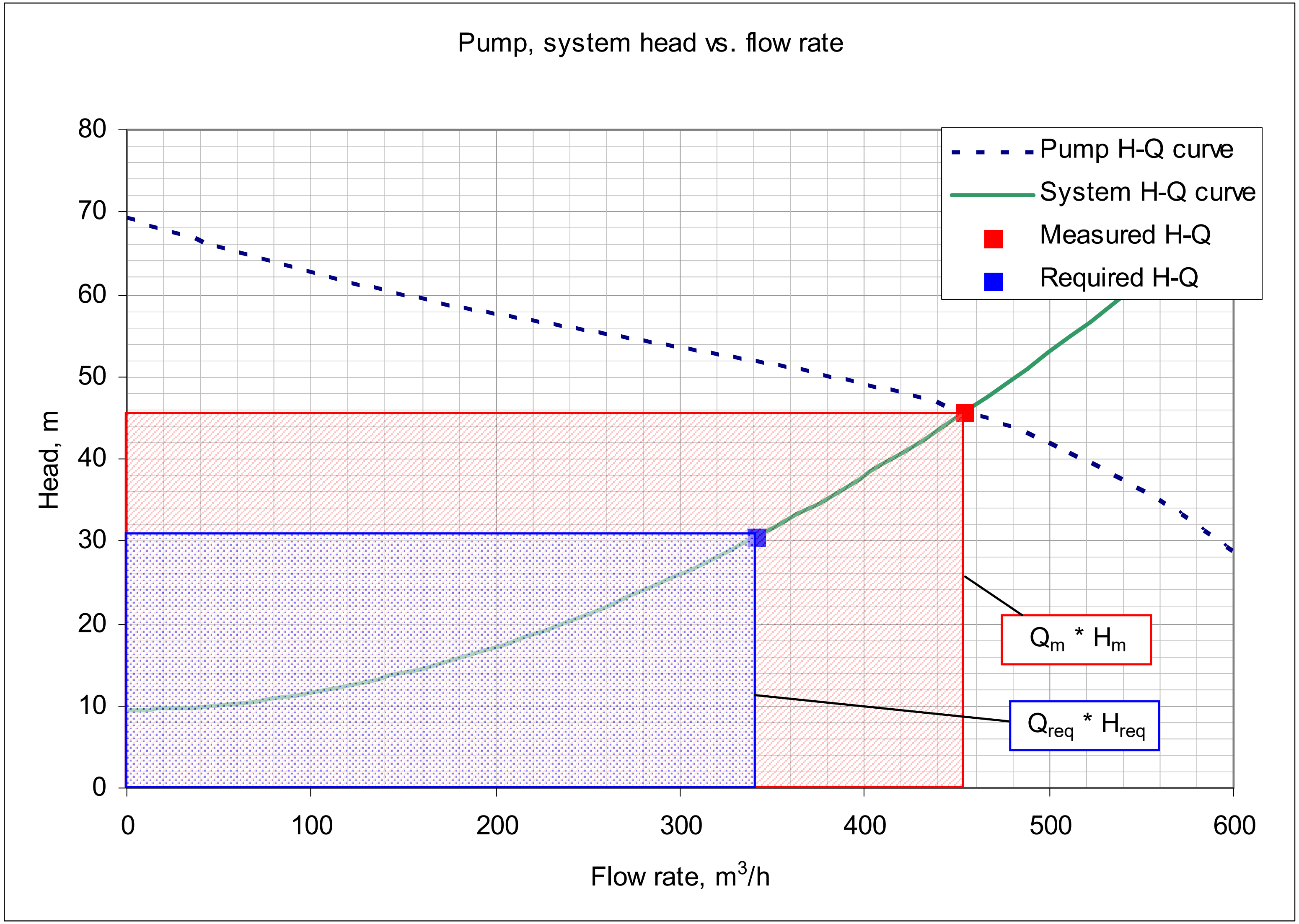
- Remove / reduce unnecessary throttling
- Clean fouled or partially blocked components
- Isolate unnecessary flow paths
- Replace old or corroded piping
- Up-size piping
- Reduce number of valves and fittings
- Increase suction tank level

Solutions to Excessive Energy Use

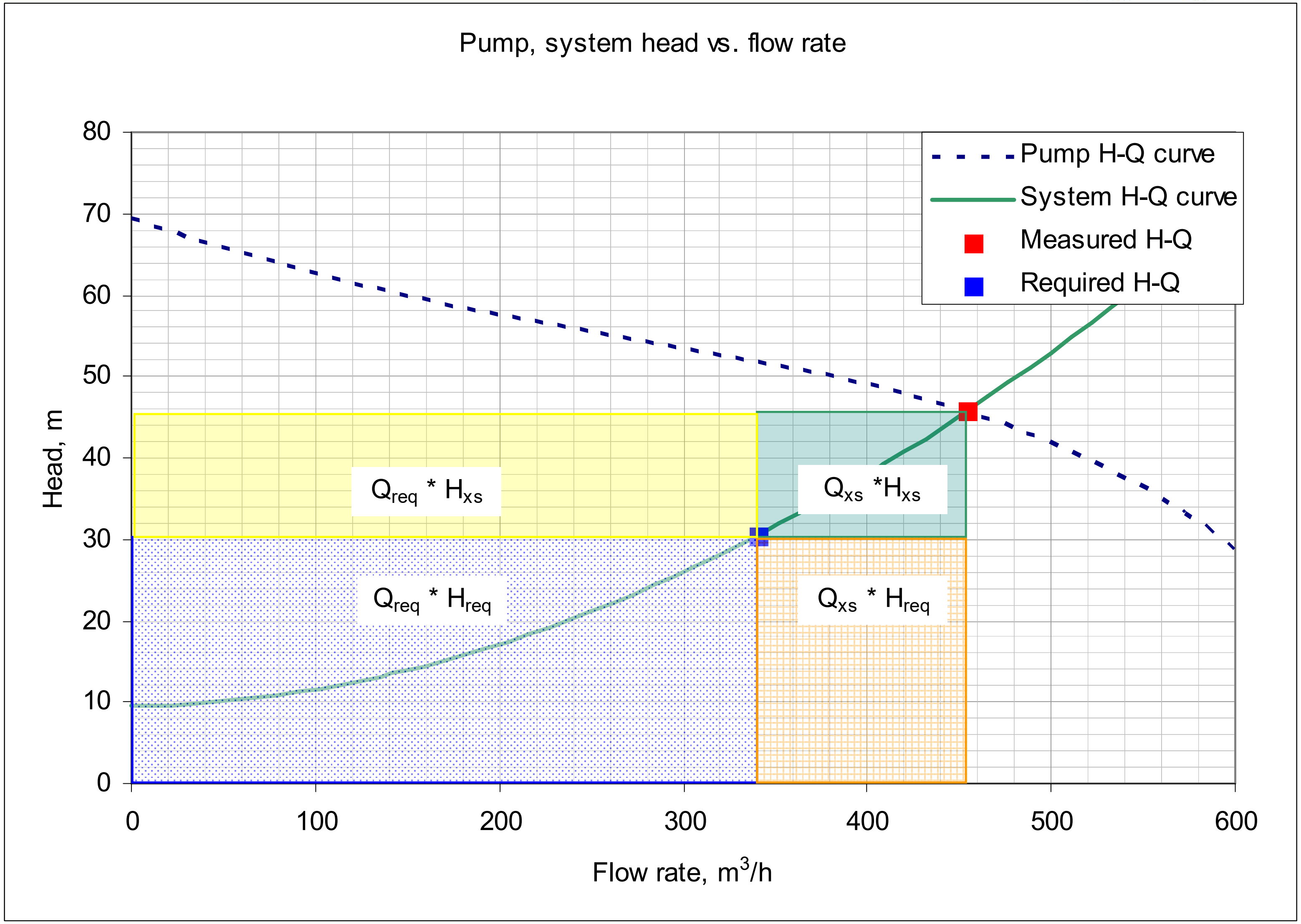
Reduce System Flow Rate:

- Maintain appropriate heat exchange differential temperatures by reducing cooling water flow.
- Isolate unnecessary flow paths.
- Extend batch process fill and drain times.
- Turn off/reduce flow when not needed.

Measured versus Required H-Q



The excessive power(s) delivered



9. ASME Chapter 7

Reporting and Documentation

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7. Reporting and Documentation

7.1 Introduction

7.2 Report Contents

7.2.1 Executive Summary and Project Summary Table

7.2.2 General Facility Information

7.2.3 Assessment Goals & Scope

7.2.4 Description of Systems

7.2.5 Data Collection Methods

7.2.6 Data Analysis

7.2.7 Energy Baseline

7.2.8 Savings Opportunities Identified

7.2.9 Implementation Recommendations

7.2.10 Appendices

7. Reporting and Documentation:

7.3 Provision for Third Party Review

7.4 Review of Final Report by Assessment Team Members

Executive Summary

- Summary of existing energy use
- Presentation of identified energy saving projects with annual kWh savings, cost savings, estimated project cost, SPB, NPV, IRR, CO_{2eq} savings. Projects typically presented as:
  - Implement Immediately (Low cost)
  - Longer Term (Full assessment required)
  - Pump vs System savings
- Summarize percent savings and environmental benefits.

Some Typical Examples....

Summary Chart Example: (EGP0.90/kWh)

	Proposed Cost saving Measures	Annual Energy Savings (kWh)	First Year Annual (EGP)	Initial Cost (EGP)	Simple Payback (yrs)
	OPERATIONAL MEASURES				
OM1	Initiate Efficiency Management Program	--	--	--	--
OM2	Install New Flow Meter at Scenic Station	--	--	EGP26 000	--
OM3	Lochrem Well Speed Adjustment	59 953	EGP53 957	--	--
OM4	Tutt Pump Speed Adjustment	9 665	EGP8 698	--	--
OM5	Scenic Pump Speed Adjustment	48 646	EGP43 781	--	--
OM6	Install Low Temperature Thermostats	--	--	--	--
	ENERGY CONSERVATION MEASURES				
ECM1	Airport Well #1 Pump/VSD Replacement	58 897	EGP53 007	EGP400 400	7.5
ECM2	Airport Well #2 Efficiency Improvements	150 650	EGP135 585	EGP257 400	1.9
ECM3	Union Street Pump Improvements	72 024	EGP64 821	EGP328 900	5.0
	ENERGY SUPPLY MEASURES				
ESM1	Prevent Two Pump Operation at Tutt	--	EGP41 522	--	--
ESM2	Switch Rate Schedules	--	EGP228 605	--	--
	Electric Energy Cost and Savings	**399 835**	**EGP629 976**	**EGP1 012 700**	**1.6**

1.1.1. Immediate implementation, no cost or low cost

PUMP SYSTEM	SYSTEM SAVINGS		
	R/yr	%	OPPORTUNITY
Staves Cooling Water Supply Pumps	R 1632 960	50%	All pumps highly throttled, open valves until reach max current. Should be able to run 2 pumps instead of 3 to achieve same flow.
Supply Pump – Scrubbers	R 876 000	13%	Remove and check NRV's in order to reduce friction loss
System 5	R 815 665	40%	Remove and check NRV's, reduce friction loss
Secondary Cooling Water Supply Pumps	R 557 200	30%	Remove throttling by opening valves until max current reached. Trim impellers.
System 4	R 551 124	20%	Reduce pressure to 9 Bar and 6 Bar by removing throttling
Cold Water Pumps	R 414 180	4%	Repair NRV and gate valve on pumps 4 and 5
Supply Pumps - Critical User	R 326 592	30%	Check max current for pump and only throttle to required current. Investigate trimming impeller to further reduce throttling
System 6	R 220 450	10%	Remove and check NRV's, reduce friction loss
System 8	R 218 245	15%	Remove and check NRV's, reduce friction loss
Cooling Tower Pumps	R 216 295	9%	Repair NRV and gate valves on pump 1 and 2
System 8	R 145 500	10%	Clean out heat exchangers
Secondary Cooling Water Supply Pumps	R 130 004	7%	Fix Pump #1 NRV and prevent reverse flow
TOTAL SAVINGS	**R 6 104 215**	-	-

1.1.3. Full assessment required

PUMP SYSTEM	SYSTEM SAVINGS		
	R/yr	%	OPPORTUNITY
Compressor	R 3 810 240	2%	Improve compressor efficiency by lowering cooling water temperature
Auxilliary Pumps	R 1 208 700	35%	Turn pumps off during idle time. Install additional accumulators to act as buffer, pumps will only start to top-up accumulator
Servo Pumps	R 998 100	4%	Turn pumps off during idle time. Install additional accumulators to act as buffer, pumps will only start to top-up accumulator
System 1	R 771 574	10%	Increase inlet temp to furnace. Run one pump less at least 50% of the time
System 21	R 727 484	40%	Remove solids from water, prevent blocking of pump suction, no more clean out of sumps required
System 11	R 396 809	15%	Synchronise both sets of pumps. Investigate leakage. Throttling used to balance flow to filters
System 23	R 303 118	25%	Ideal opportunity for VSD. Match flow to demand
System 3	R 275 562	20%	Investigate overall efficiency of pumps and cooling towers
Backup Pumps	R 201 398	80%	Use alternate heat source, boiler only operational 20% of time. Reduce pumping by 80%
TOTAL SAVINGS	**R 8 692 985**	**-**	**-**

Note: Rand values based on average tariff of 90c / kWh.

Rank Existing Pump Stations

Top 5 stations were assessed

Pump stations ranked by wastage

Site	Pump	Installed Power	Annual Wastage (R)	Wastage (%)	Wear Wastage %	% Wastage - Rank	Annual Wastage (R) - Rank	Wear Wastage %- Rank
Mrt Dam	1	676	R 0	0.0%	-	1	1	-
Mrt Dam	2	676	R 203 788	20.1%	5.1%	3	3	1
Mrt Dam	3	676	R 291 390	45.3%	34.4%	5	3	5
Groenkloof (High lift)	1	580	R 1 187 412	26.1%	34.3%	4	5	5
Groenkloof (High lift)	2	580	-	-	-	-	-	-
Groenkloof (High lift)	3	580	R 1 028 099	22.8%	29.4%	4	5	4
Wartburg	1	160	R 117 652	17.8%	9.0%	3	2	2
Wartburg	2	160	R 135 803	19.1%	16.8%	3	2	3
Wartburg	3	160	R 115 825	17.1%	7.2%	3	2	1
Bruynshill	1	55	R 47 211	19.4%	14.9%	3	1	2
Bruynshill	2	55	R 31 371	12.5%	6.0%	2	1	1
Dingle	1	75	R 43 454	16.9%	16.1%	3	1	3
Dingle	2	75	R 42 244	16.5%	15.2%	3	1	3
Thornville	1	45	R 92 142	34.9%	31.4%	5	1	5
Thornville	2	45	R 86 865	34.3%	36.0%	5	1	5
Thornville	3	45	R 93 573	35.4%	35.2%	5	1	5
DV Harris- Backwash	1	15	R 2 181	14.0%	-	2	1	-
DV Harris- Backwash	2	15	R 2 181	14.0%	-	2	1	-
DV Harris- Backwash	4	15	R 2 181	14.0%	-	2	1	-
DV Harris- Domestic	1	30	R 18 876	33.6%	1.5%	5	1	1

10. Case Studies

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Waste Water System Example

General Facility Information

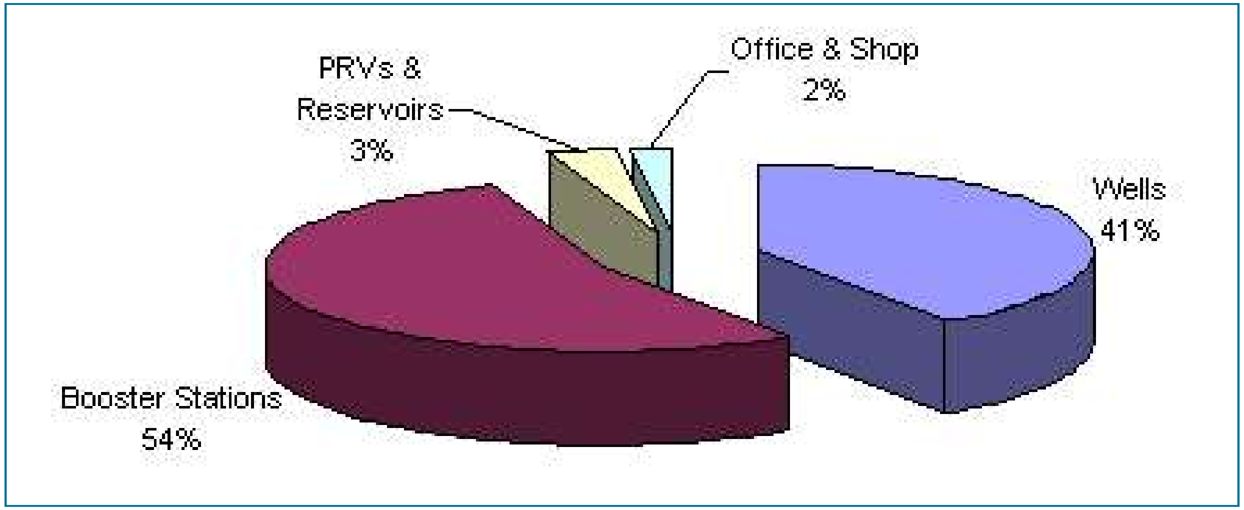
Pump stations feeding a waste water plant

The top 5 stations were assessed

Electric Acct #	Location	Station Name	2009 Cost
3734550273	Corner of Old Vernon/Hwy 97	Airport Wells	$60,355
8400427215	1595 Glenmore Rd	Tutt Pump Station	$30,806
8626205671	2141 Quail/Lochrem Rd	Lochrem Well	$25,888
3438735241	Scenic Rd	Scenic Booster Station	$19,228
8437034171	1850 Union Rd	Union Road Booster Station	$19,084
8784406113	Postill Lake	Postill Pump Station	$14,120
2028257738	Country Club Drive	Quail Pump Station	$13,971
7196445516	2052 Dewdney Rd	OK Lake Pump Station	$10,160
37892[illegible]	[illegible]	Ellison Well	$9,181
137025[illegible]	[illegible]	istrano Booster Station	$6,285
353290[illegible]	[illegible]	Kinley Pump Station	$5,310
639285[illegible]	[illegible]	Intake Screen	$3,904
68295[illegible]	[illegible]	ulach Bstr Pump Station	$3,508
8493235872	445B Glenmore Rd	Office	$2,922
4217787985	47192 Country Club Drive	UBCO Reservoir	$2,473
5303004369	833 Big Rock Court	Big Rock Booster Station	$2,380
2325753172	445A Glenmore Rd	Shop	$1,497
3280893061	McKinley Rd	Arthur Court Reservoir & Pump Station	$1,213
4616000173-4	2329 Rojem	Bulach Bstr Station Aux	$743
5700429059	540 Reynolds Rd	Cook Dom Pump Station	$635
6409641365	550 Valley Rd	Raisenen Rd PRV	$632
2448745798	800 Packinghse Rd	Scenic Reservoir	$522
3203030962	1248 Reynolds Rd	Cooks Irr Pump Station	$451
6081212108	2635 Dry Valley Rd	Dry Valley PRV	$241
4710548990	70877 Rifle	Rifle Rd Pump Station	$221
5286062985	127205 Sexsmith Rd	Sexsmith Road Well	$194
4366273409	1210 University Way / Concass	Vector Well #1	$0
Total			**$235,924**

The highest energy use stations representing 66% of GEID 2009 energy costs were reviewed to evaluate potential energy savings.

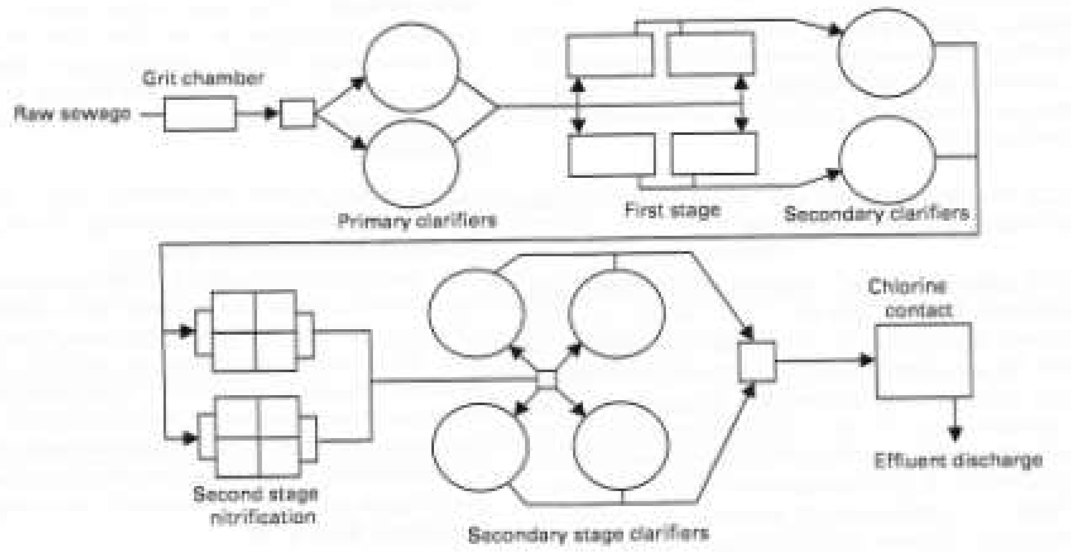
General Facility Information



Distribution of energy usage for pumping within a municipality

Layout of the waste water treatment plant

Fig. 11 Example of Process Diagram



Cooling Water System Example

Site Description

- The site has a large integrated cooling water system.
- There are 6 Cooling Tower (CT) sites with multiple cells per site which are interconnected with cooling water supply and return loops.
- Usually 3 pumps per site (2 operating, 1 reserve).
- There are 20 pumps in the cooling water system.
- ± 13 pumps (225 - 450kW each) are running at any given time to supply the necessary cooling water flows.
- CT 4 has only two 450kW pumps = focus of assessment
- The circulation rate is ± 9 500 l/s

Assessment goal and scope

Assessment of a cooling system. What to look for:

- Pump and Motor efficiencies
- Regulation methods
- Throttling and/or by-pass losses
- Cooling tower operation
- Cooling needs for served processes
- Supply equals demand?
- Cooling tower operation
- Water levels
- Fans
- Assess the system and suggest improvements

PSO Assessment Activities

During the assessment the following tasks will be performed:

- Review the operation of CT4 water cooling systems
- Estimate energy use
- Perform field measurements of power, pressure and flow to identify sources of energy loss
- Use MEASUR to quantify the opportunity
- Review energy use and pump reliability issues

Data Collection Methods

- Measurement of pressures and amperage was done on the chosen systems. This was a challenge due to lack of pressure taps, but the plant people were very helpful.
- The PSO expert worked closely with the plant personnel to examine and input the data collected into MEASUR.
- In all cases except one (where pump curves were not available), the results were compared to pump curves and the flow estimated from power and pressure measurements.

Energy Baseline

The Pumps at CT4 are running continuously drawing 216.9 kW each

Identified Saving Opportunities

Results:

- Saving opportunity ± EGP 2 600 000 /y at no cost by turning one pump off.
- Net 216.9 kW reduction in cooling water system power consumption (90 A at 2 410 V);
- Overall system pressure not affected
- Throttling valves at other pump stations were adjusted to accommodate for the change

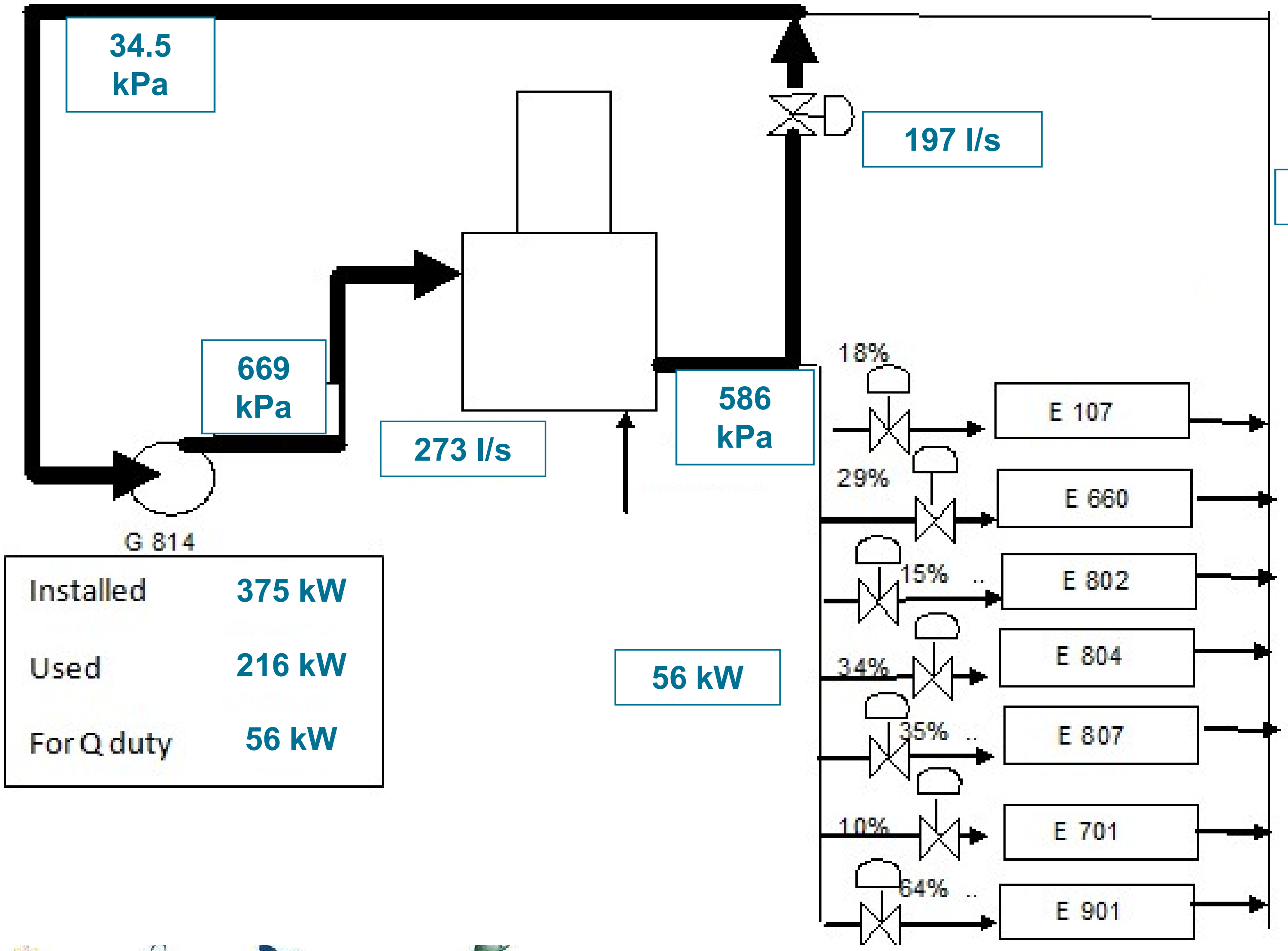
Recommendations

The proposed and executed change was:

- Shut down one of the 450kW water pumps at CT4
- Increase flow from the other pump at CT4 by opening up the throttling valve
- Assess load increase on the other pumps connected to the plant cooling system

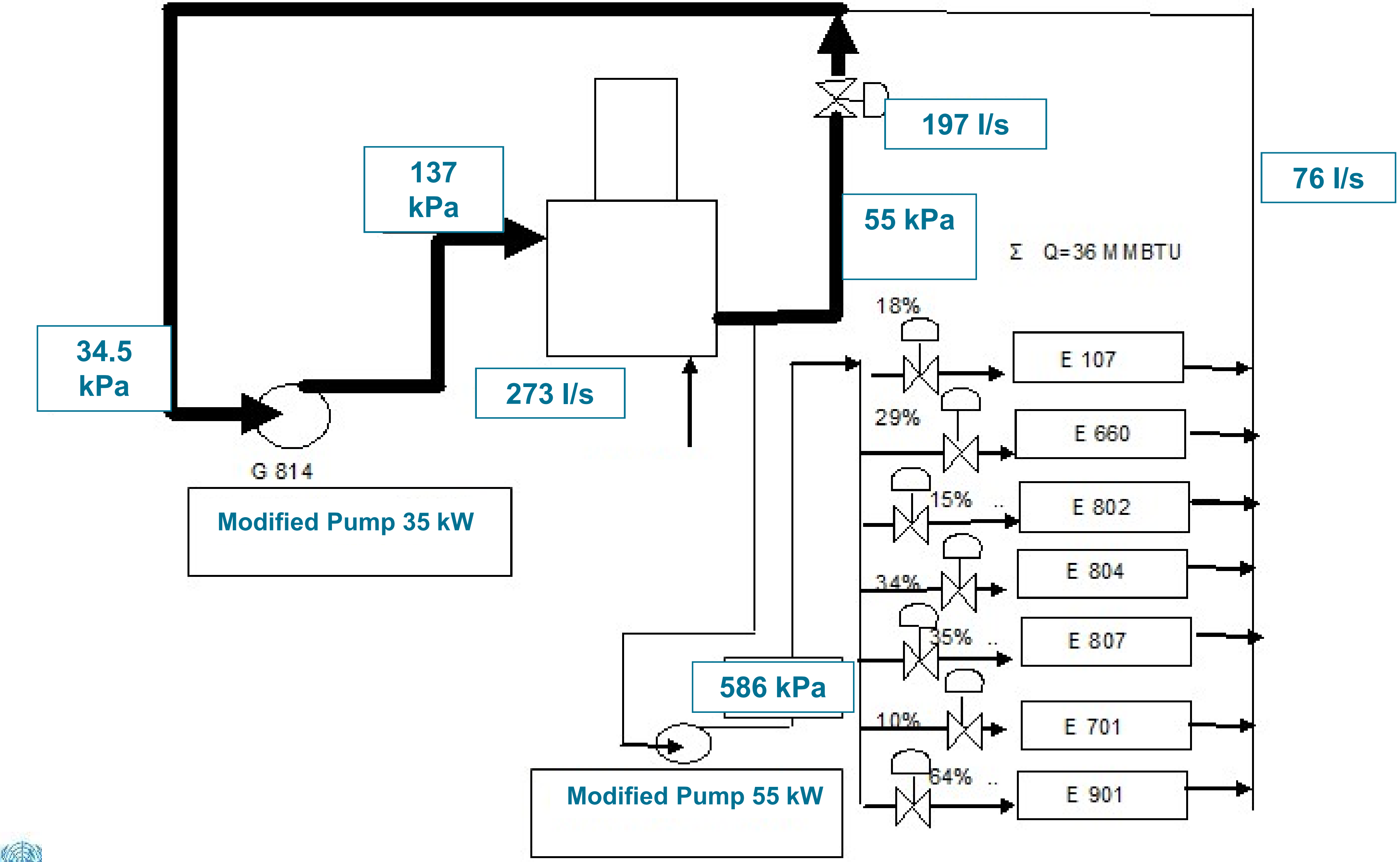
Hot Oil Circulation System Example

Installed Hot Oil System for Process Heating



Installed	375 kW
Used	216 kW
For Q duty	56 kW

Proposed System



137 kPa

197 l/s

76 l/s

55 kPa

Σ Q=36 MMBTU

34.5 kPa

273 l/s

18%

29%

15%

34%

35%

10%

64%

E 107

E 660

E 802

E 804

E 807

E 701

E 901

G 814

Modified Pump 35 kW

586 kPa

Modified Pump 55 kW

11. MEASUR Software

Manufacturing Energy Assessment Software for Utility Reduction

Pump Systems Optimisation (PSO) Expert Level Training
(Egypt Edition – Dec 2021)

Albert Williams
Siraj Williams

MEASUR

- Goal: to assist pump users in identifying pumping systems that are the most likely candidates for energy and cost savings
- Requires field measurements or estimates of flow rate, pressure, and motor power or current
- Uses pump and motor performance data from Hydraulic Institute standard ANSI/HI-1.3 and MotorMaster+ to estimate existing, achievable performance

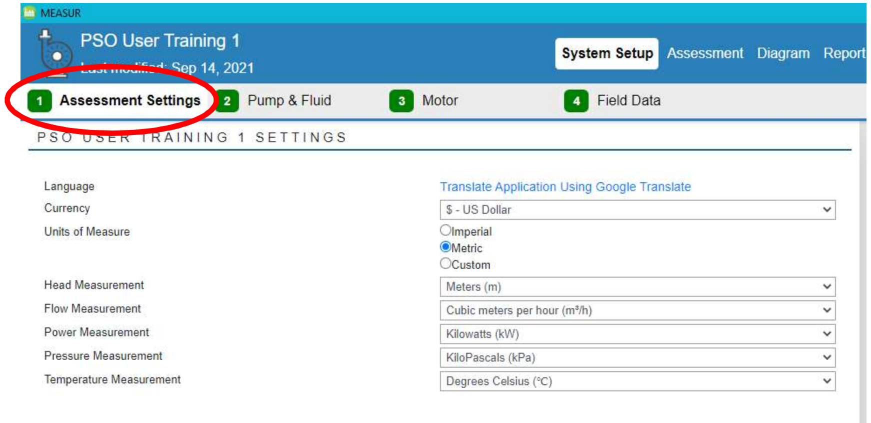
How to Use MEASUR

MEASUR: Can be used both as a Component Tool and as a System Tool

- For a given operating point, MEASUR searches for the highest pump efficiency possible at that point.
- It also searches for the highest motor efficiency available to drive the found pump at that point.
- It calculates the cost of operating at the point in terms of kWh used and $ saved.
- MEASUR can also be used as a system tool if the minimum flow and pressure needed for the process are entered instead of current head and flow.

https://www.energy.gov/eere/amo/measur

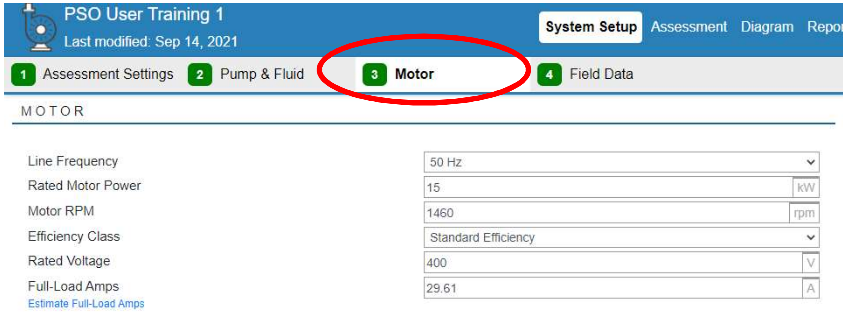
MEASUR: Input Fields



PSO User Training 1
Last modified: Sep 14, 2021

PSO USER TRAINING 1 SETTINGS

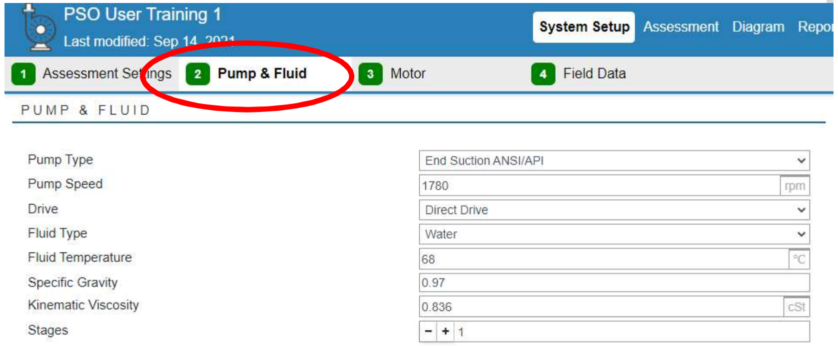
Language	Translate Application Using Google Translate
Currency	$ - US Dollar
Units of Measure	○ Imperial ● Metric ○ Custom
Head Measurement	Meters (m)
Flow Measurement	Cubic meters per hour (m³/h)
Power Measurement	Kilowatts (kW)
Pressure Measurement	KiloPascals (kPa)
Temperature Measurement	Degrees Celsius (°C)



PSO User Training 1
Last modified: Sep 14, 2021

MOTOR

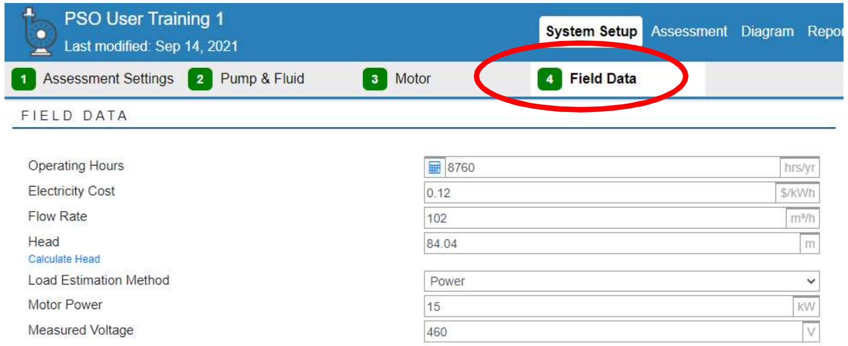
Line Frequency	50 Hz
Rated Motor Power	15 kW
Motor RPM	1460 rpm
Efficiency Class	Standard Efficiency
Rated Voltage	400 V
Full-Load Amps	29.61 A



PSO User Training 1
Last modified: Sep 14, 2021

PUMP & FLUID

Pump Type	End Suction ANSI/API
Pump Speed	1780 rpm
Drive	Direct Drive
Fluid Type	Water
Fluid Temperature	68 °C
Specific Gravity	0.97
Kinematic Viscosity	0.836 cSt
Stages	− + 1



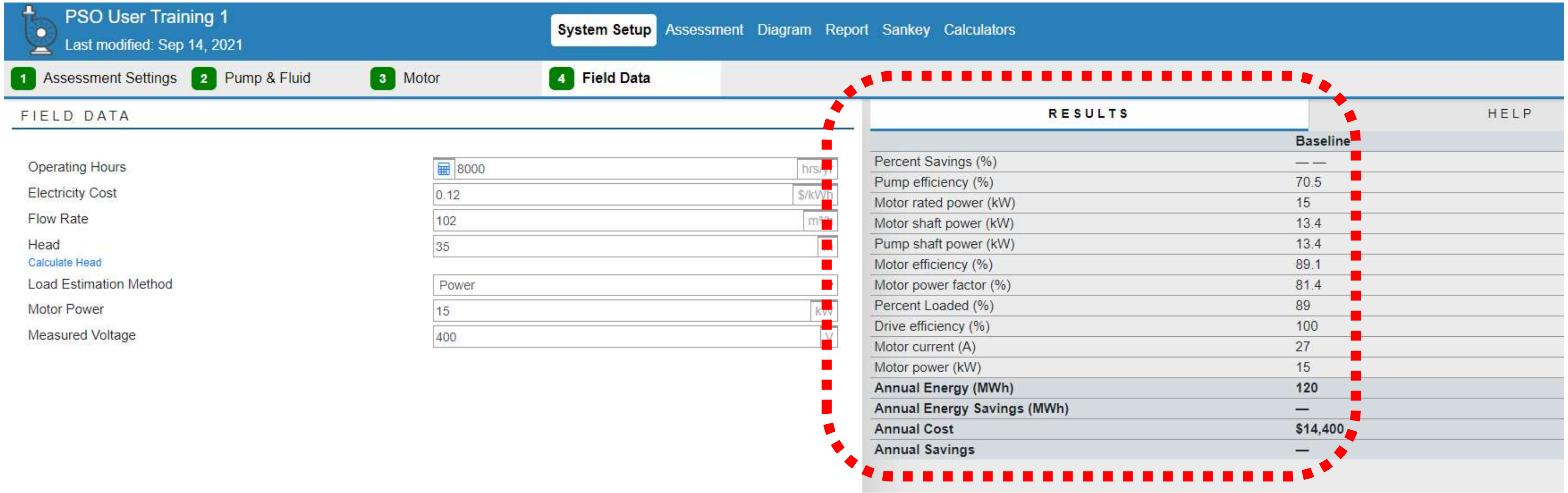
PSO User Training 1
Last modified: Sep 14, 2021

FIELD DATA

Operating Hours	8760 hrs/yr
Electricity Cost	0.12 $/kWh
Flow Rate	102 m³/h
Head	84.04 m
Load Estimation Method	Power
Motor Power	15 kW
Measured Voltage	460 V

MEASUR: Output Result

- Results from initial output provide the baseline energy consumption.
- Next step is to add in saving opportunities and evaluate energy savings against the baseline



FIELD DATA

Operating Hours	8000	hrs/yr
Electricity Cost	0.12	$/kWh
Flow Rate	102	m³/
Head Calculate Head	35	
Load Estimation Method	Power	
Motor Power	15	kW
Measured Voltage	400	V

	Baseline
Percent Savings (%)	— —
Pump efficiency (%)	70.5
Motor rated power (kW)	15
Motor shaft power (kW)	13.4
Pump shaft power (kW)	13.4
Motor efficiency (%)	89.1
Motor power factor (%)	81.4
Percent Loaded (%)	89
Drive efficiency (%)	100
Motor current (A)	27
Motor power (kW)	15
Annual Energy (MWh)	**120**
Annual Energy Savings (MWh)	**—**
Annual Cost	**$14,400**
Annual Savings	**—**

Important: Demand and Supply

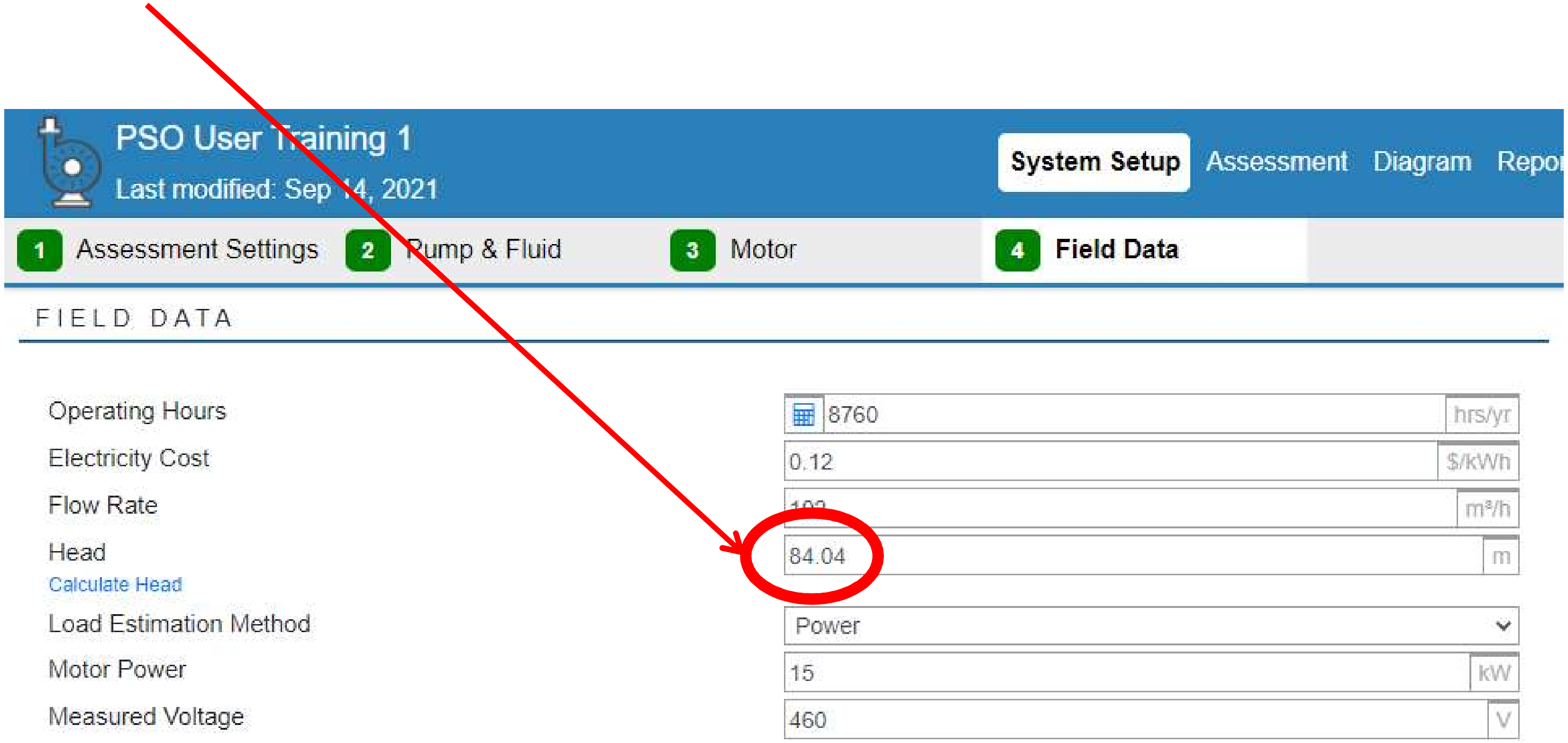
- There is often a difference between what the pump is providing the system and what the system really needs

- Try to think in terms of **Demand**, not **Supply**

Slide Courtesy of Oak Ridge National Laboratory

Using MEASUR Head Tool

Head is a required input, where does it come from?



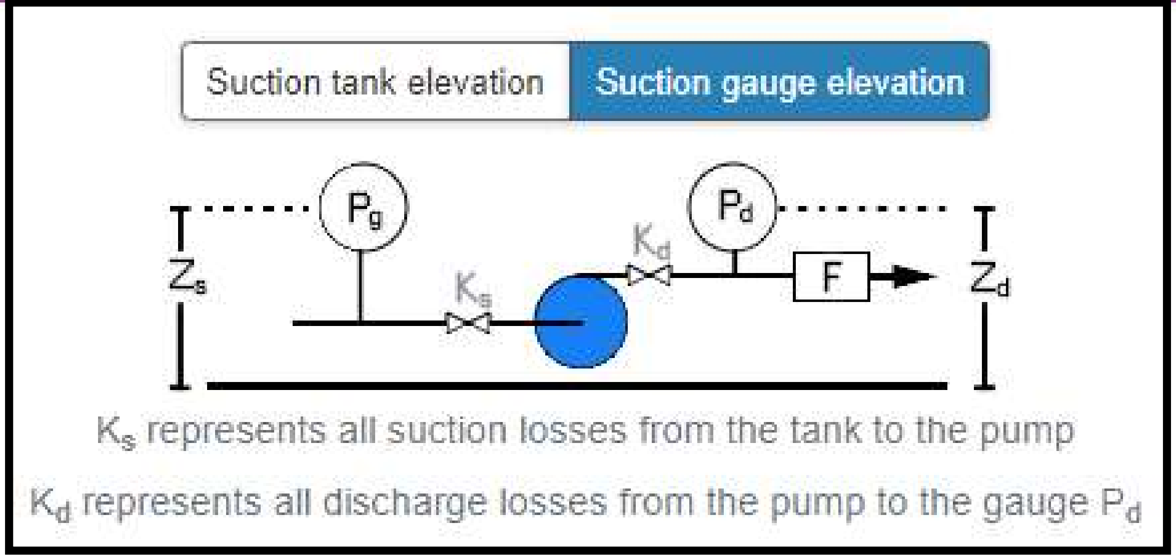
FIELD DATA

Operating Hours	8760 hrs/yr
Electricity Cost	0.12 $/kWh
Flow Rate	102 m³/h
Head	84.04 m
Load Estimation Method	Power
Motor Power	15 kW
Measured Voltage	460 V

Head

- It is obtained from the head calculator built into MEASUR
- Based on standard methods (i.e., Bernoulli), but also provides a method to adjust for non-ideal field conditions
- Information needed:
  - Suction pressure measurement
  - Discharge pressure measurement
  - Elevations of the pressure measurement locations
  - Line sizes at the same locations
  - Flow rate*
- Two basic layouts are supported…

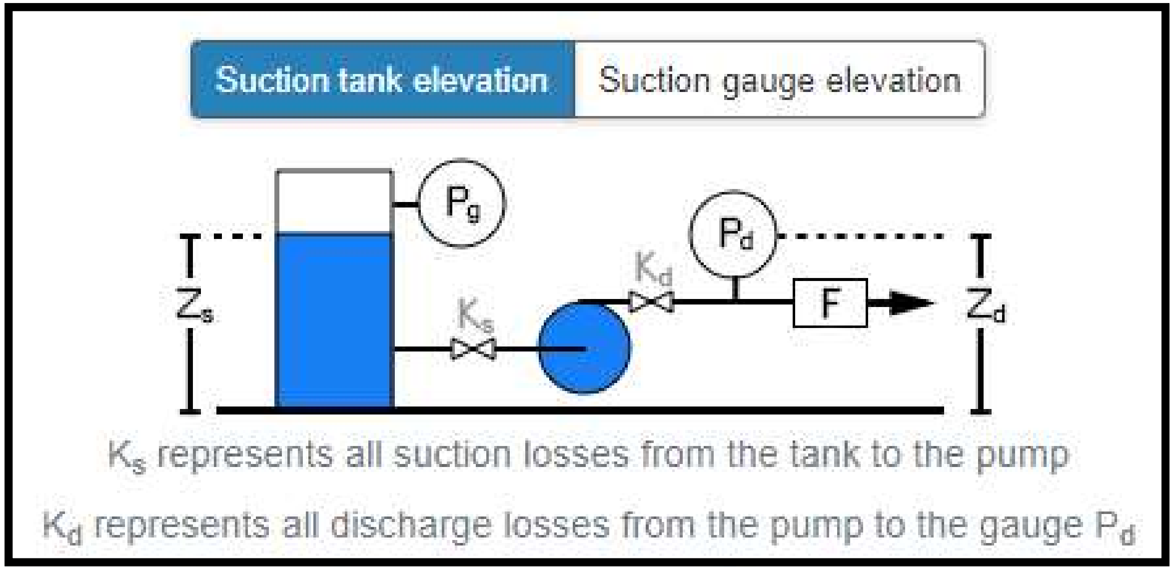
Two situations for calculating pump head



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Method 1:
Pressure measured in pump suction and discharge lines

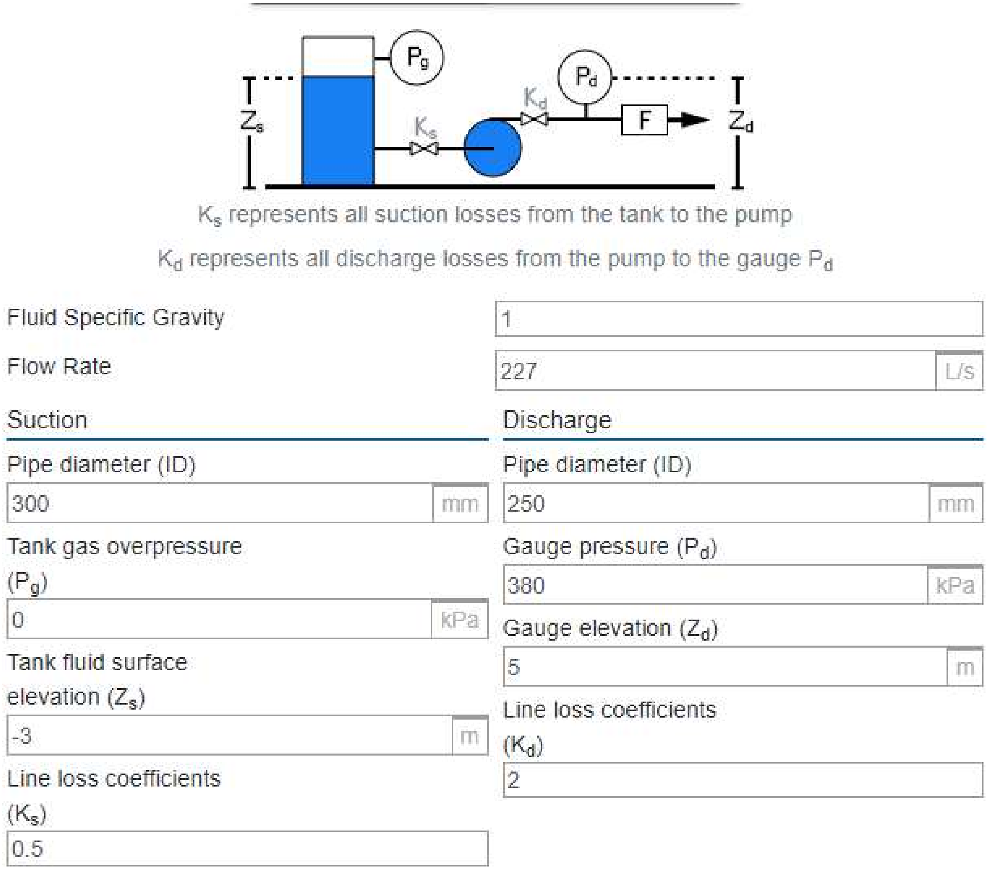


K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Method 2:
Pump draws suction from a tank (or well), with or without gas overpressure

Method 2: Example (hypothetical)



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Fluid Specific Gravity: 1

Flow Rate: 227 L/s

Suction

Pipe diameter (ID): 300 mm

Tank gas overpressure (P_g): 0 kPa

Tank fluid surface elevation (Z_s): -3 m

Line loss coefficients (K_s): 0.5

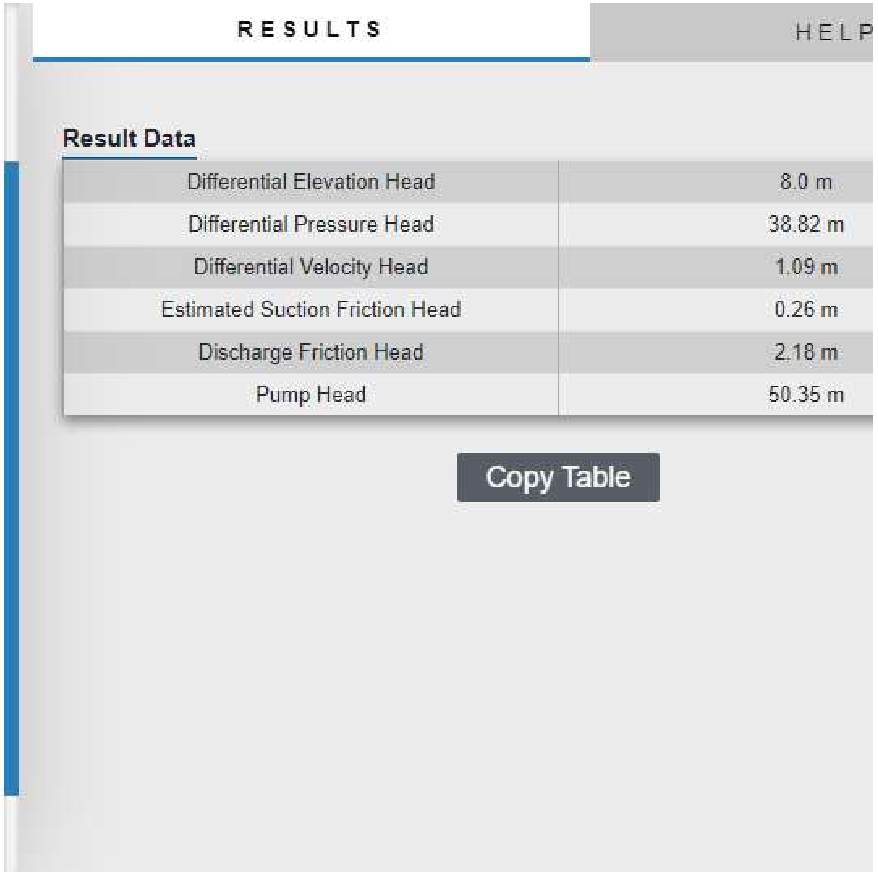
Discharge

Pipe diameter (ID): 250 mm

Gauge pressure (P_d): 380 kPa

Gauge elevation (Z_d): 5 m

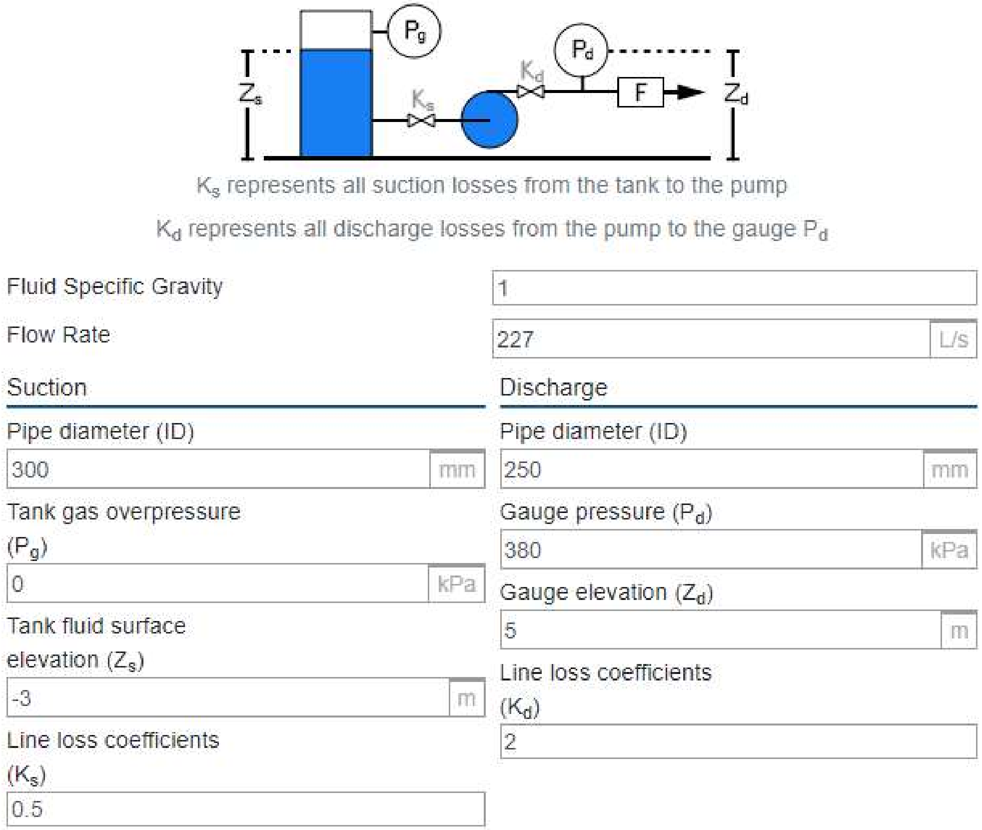
Line loss coefficients (K_d): 2



Result Data

Differential Elevation Head	8.0 m
Differential Pressure Head	38.82 m
Differential Velocity Head	1.09 m
Estimated Suction Friction Head	0.26 m
Discharge Friction Head	2.18 m
Pump Head	50.35 m

Method 2 example (situation just covered)



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Fluid Specific Gravity: 1

Flow Rate: 227 L/s

Suction

Pipe diameter (ID): 300 mm

Tank gas overpressure (P_g): 0 kPa

Tank fluid surface elevation (Z_s): -3 m

Line loss coefficients (K_s): 0.5

Discharge

Pipe diameter (ID): 250 mm

Gauge pressure (P_d): 380 kPa

Gauge elevation (Z_d): 5 m

Line loss coefficients (K_d): 2

- The 380 kPa discharge pressure corresponds to the average pressure in the pump discharge column head.
- For cases involving long columns, you must address the column friction losses in the discharge line loss coefficients entry.

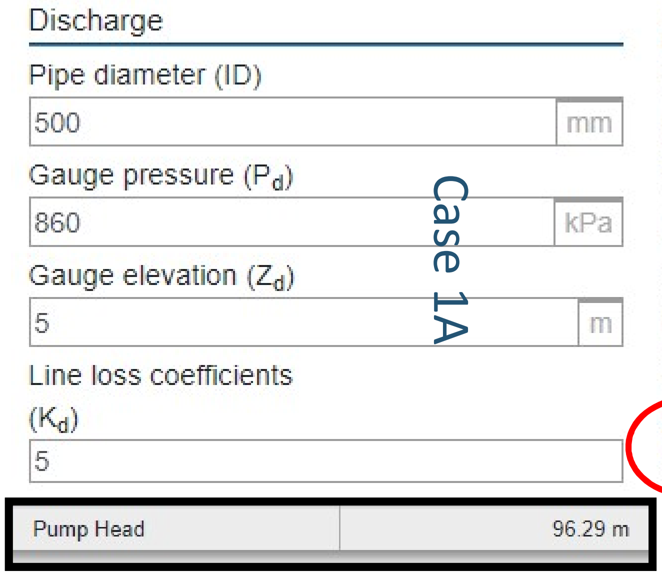
- Note: suction tank fluid surface elevation = -3.00 m; the level in a clear-well from which the pump drew suction was 3m below floor level, which was used as a reference.
- The discharge pressure gauge was on the pump base, about 5m above the floor level.

When, Why, and How questions related to the loss coefficient

- **When** should it be used:
  - Any time there are fittings between the pressure measurement reference points and the pump that may introduce friction losses
- **Why** use it:
  - Failure to account for those losses will understate the actual pump head
- **How:**
  - Use component-specific loss coefficients *(excellent)*
  - Use generic loss coefficients *(poor)*
  - A very helpful how to: use the MEASUR head calculator to get a handle on whether it is important or not

MEASUR Head Calculations

- can be used to get a sense of uncertainty importance



Case 1A

Discharge

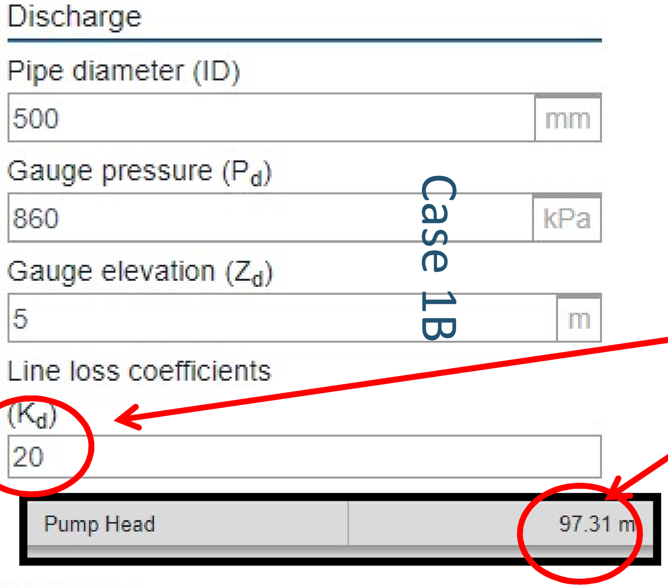
Pipe diameter (ID)
500 mm

Gauge pressure (P_d)
860 kPa

Gauge elevation (Z_d)
5 m

Line loss coefficients (K_d)
5

Pump Head 96.29 m



Case 1B

Discharge

Pipe diameter (ID)
500 mm

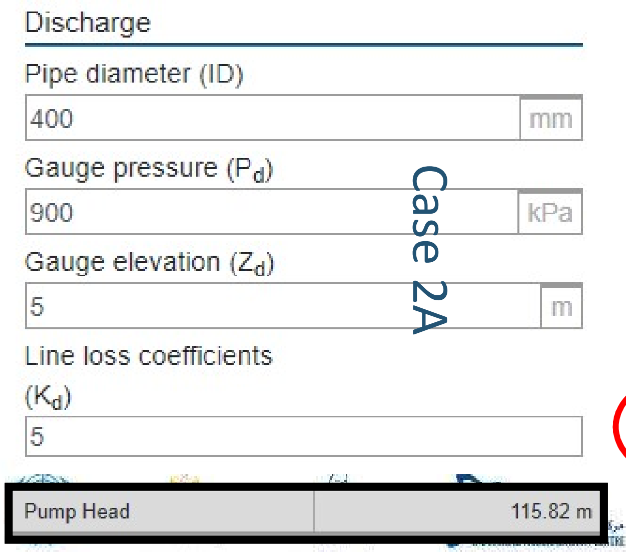
Gauge pressure (P_d)
860 kPa

Gauge elevation (Z_d)
5 m

Line loss coefficients (K_d)
20

Pump Head 97.31 m

4 x loss K =>
1% change in head



Case 2A

Discharge

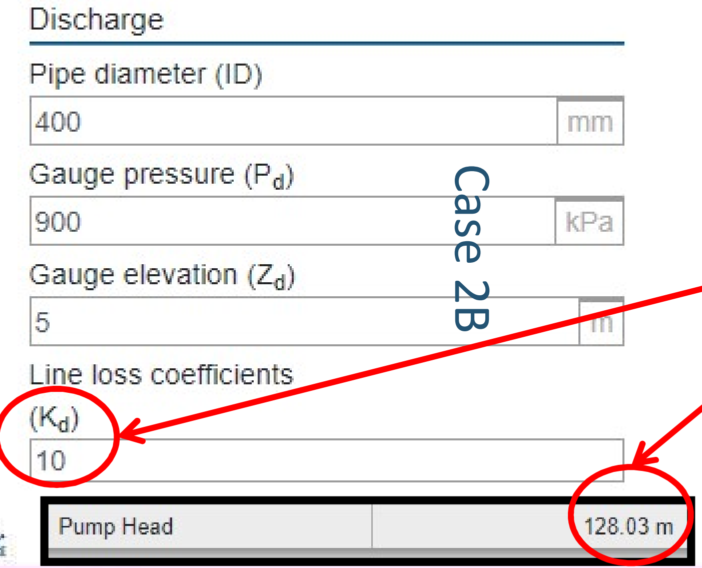
Pipe diameter (ID)
400 mm

Gauge pressure (P_d)
900 kPa

Gauge elevation (Z_d)
5 m

Line loss coefficients (K_d)
5

Pump Head 115.82 m



Case 2B

Discharge

Pipe diameter (ID)
400 mm

Gauge pressure (P_d)
900 kPa

Gauge elevation (Z_d)
5 m

Line loss coefficients (K_d)
10

Pump Head 128.03 m

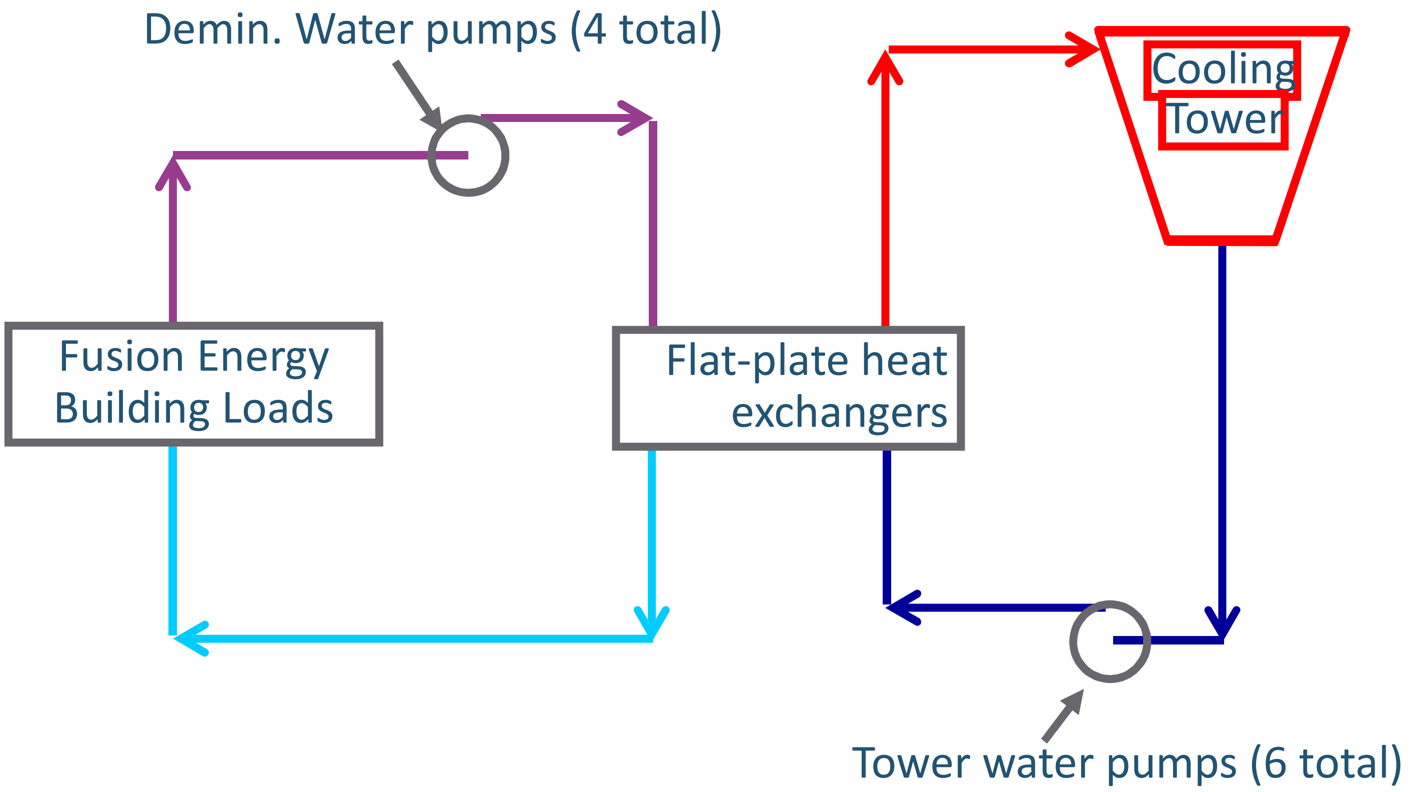
2 x loss K =>
10% change in head

MEASUR Example 1

Cooling Water System

De-mineralised Water Pumps

Simplified Flow Diagram



Demin. Water pumps (4 total)

Cooling Tower

Fusion Energy Building Loads

Flat-plate heat exchangers

Tower water pumps (6 total)

Slide Courtesy of Oak Ridge National Laboratory

A Friction dominated system

Application:

Demineralized water for process cooling

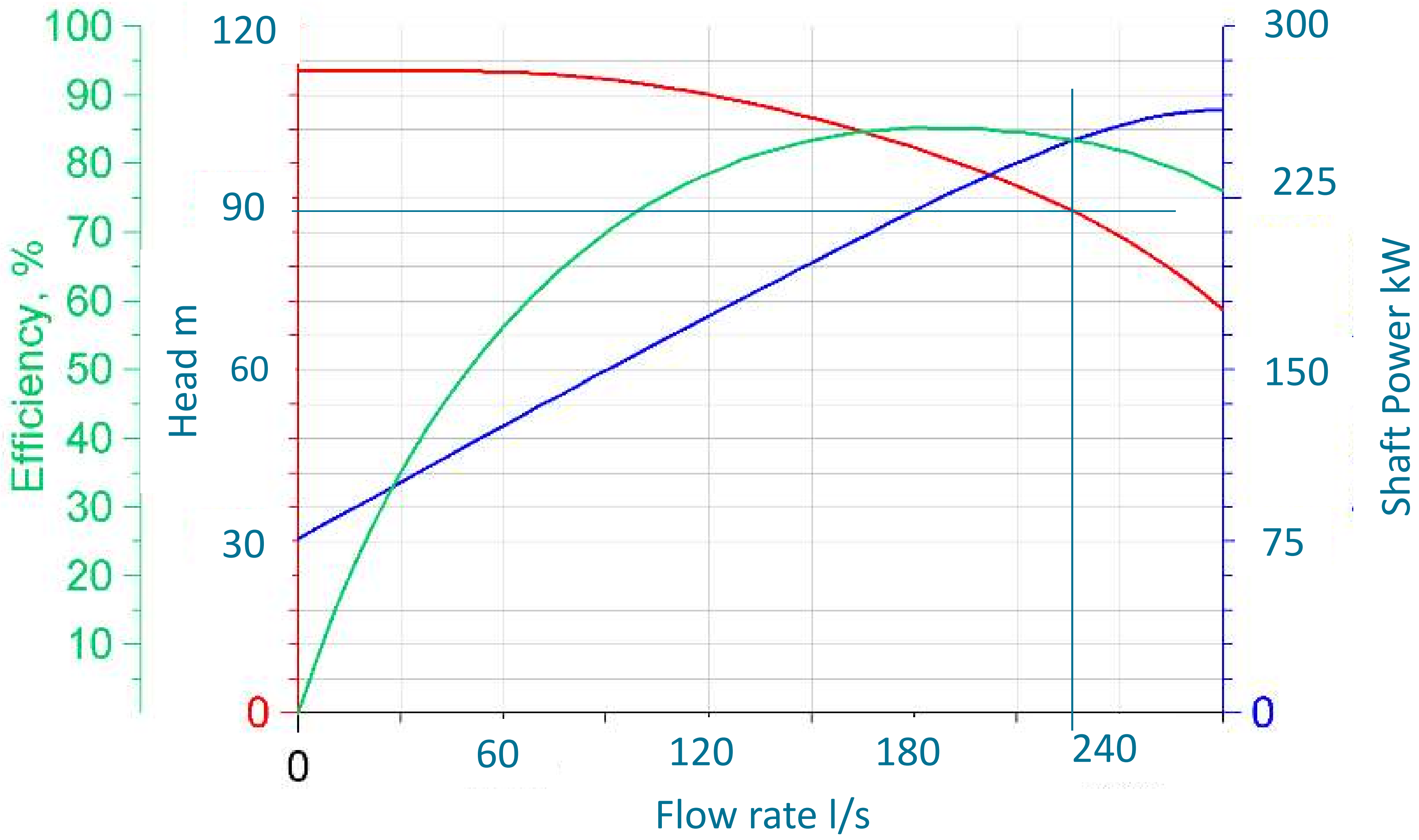
Original pump and motor design: 4 parallel pumps:

233 l/s @ 89 m, motor 1 785 rpm, 260 kW, 2300 V

Current system requirements:

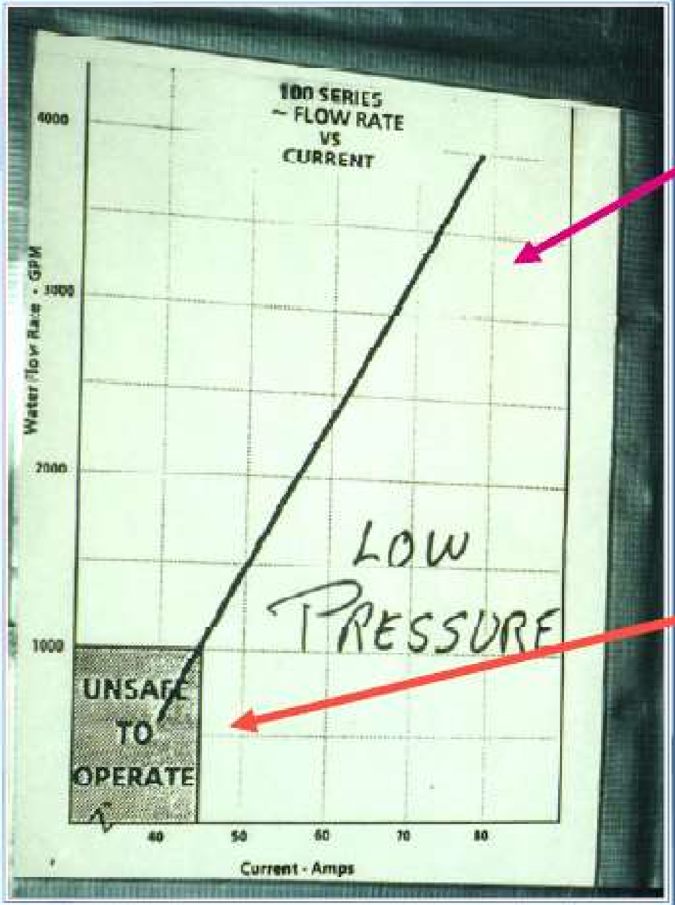
76 l/s @ 43 m head

Original pump curves



Warning on motor control box

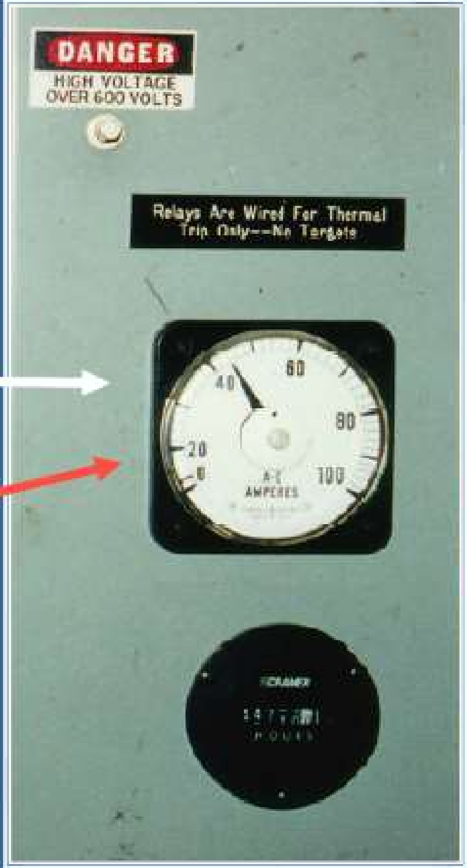
Operators can’t always accommodate outdated engineering (i.e., changed facility demands)



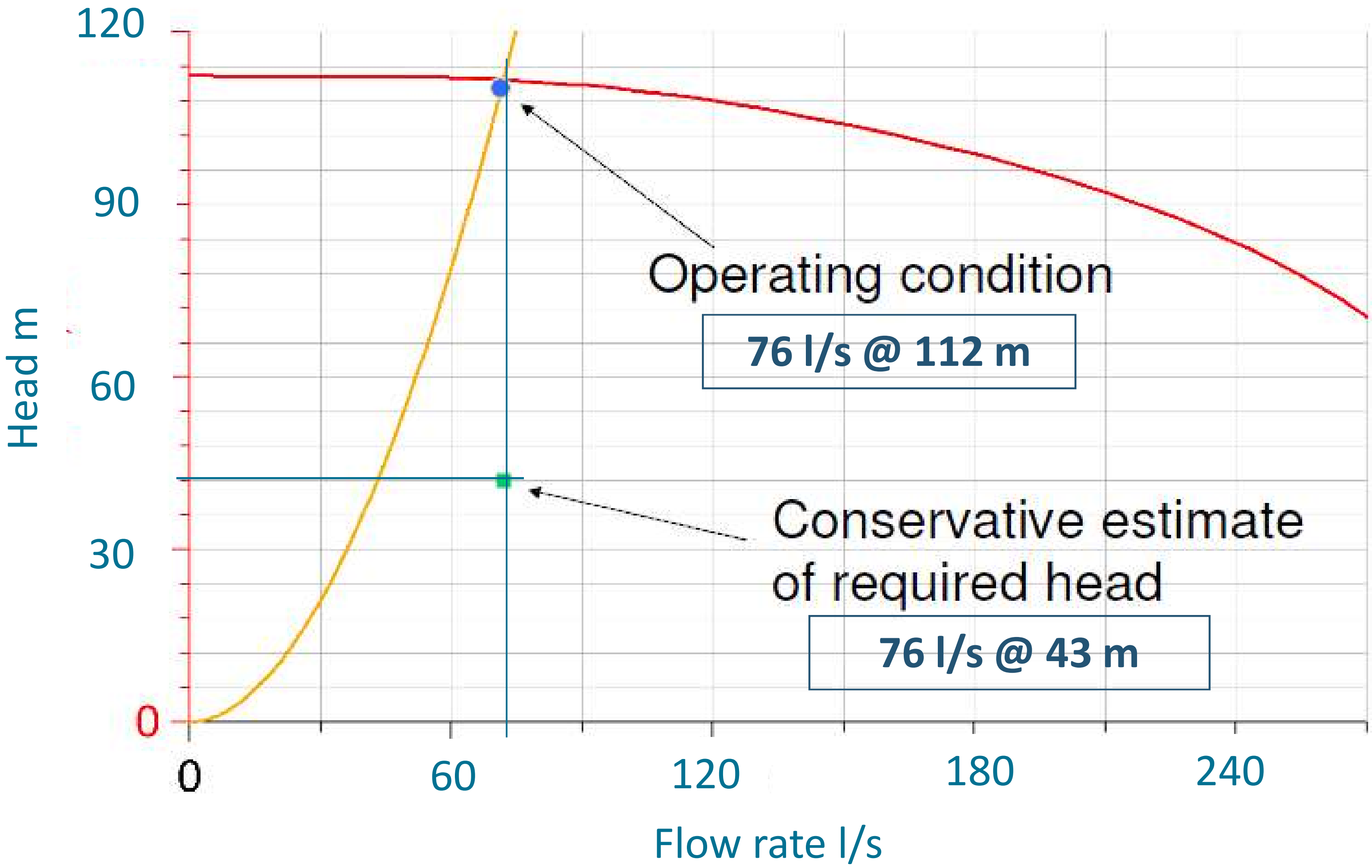
Sign on motor control center cabinet based on maintenance experience

Ammeter on the same cabinet (typical operating condition)

Normal condition was "unsafe"



Actual vs Required operating point



Operating condition

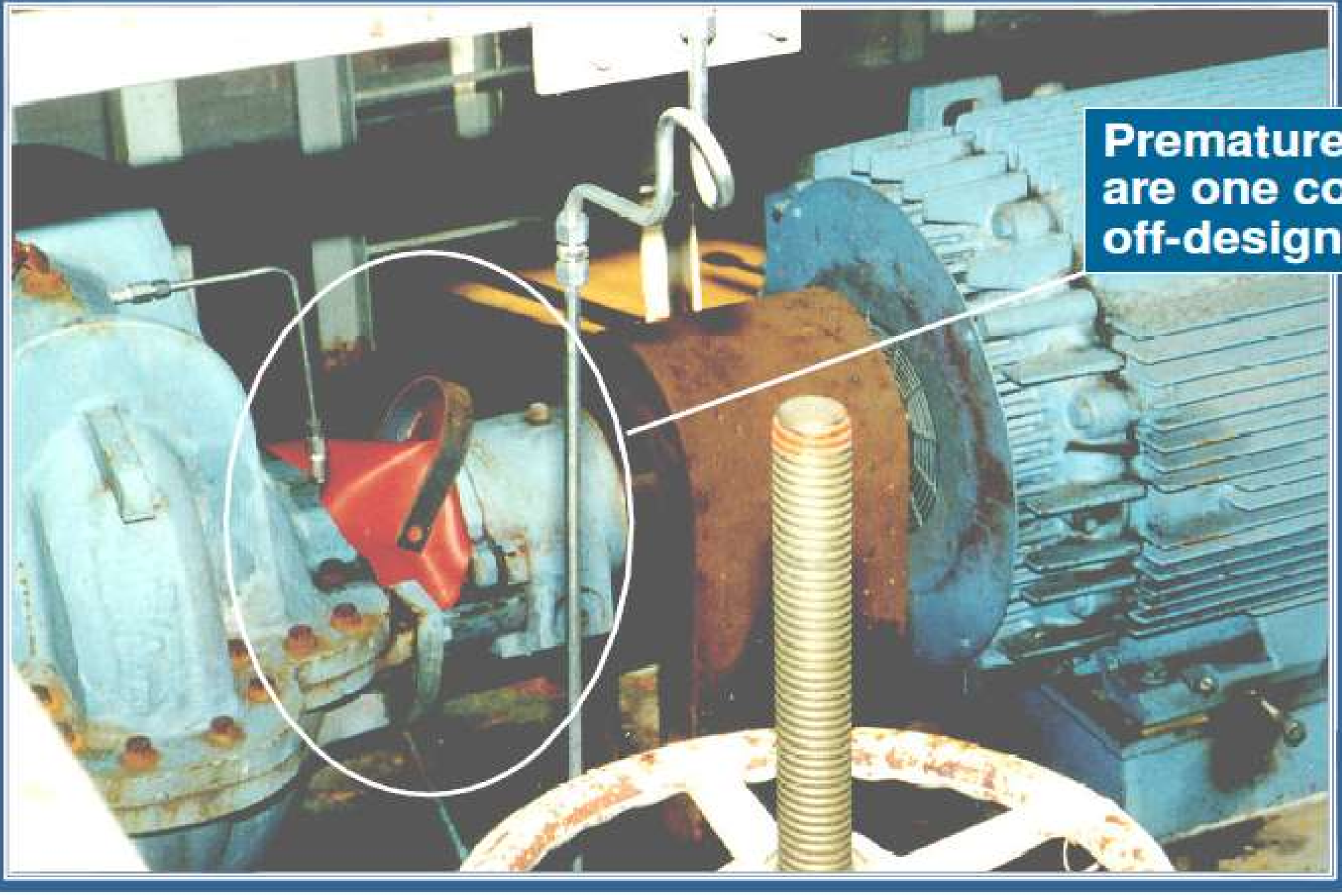
76 l/s @ 112 m

Conservative estimate of required head

76 l/s @ 43 m

Maintenance issues

Off-design operation of pumps will result in increased operating AND maintenance costs

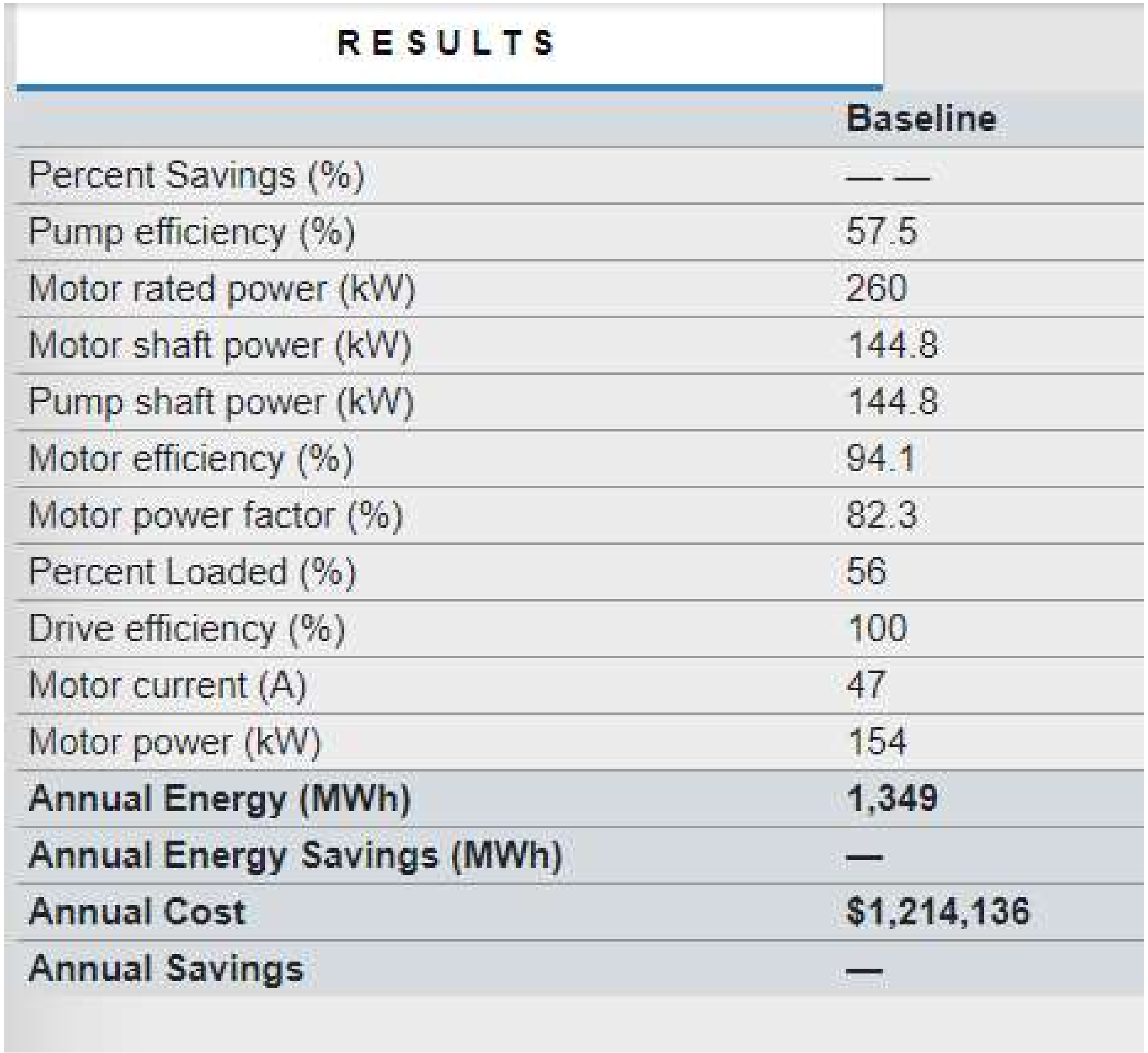


Premature seal failures are one consequence of off-design operation

Other Information

- Single stage API Double Suction Pump
- Direct Drive
- IE1 Motor
- EGP 0.9 / kWh
- 8760 hrs/y

MEASUR applied to existing operating point



RESULTS

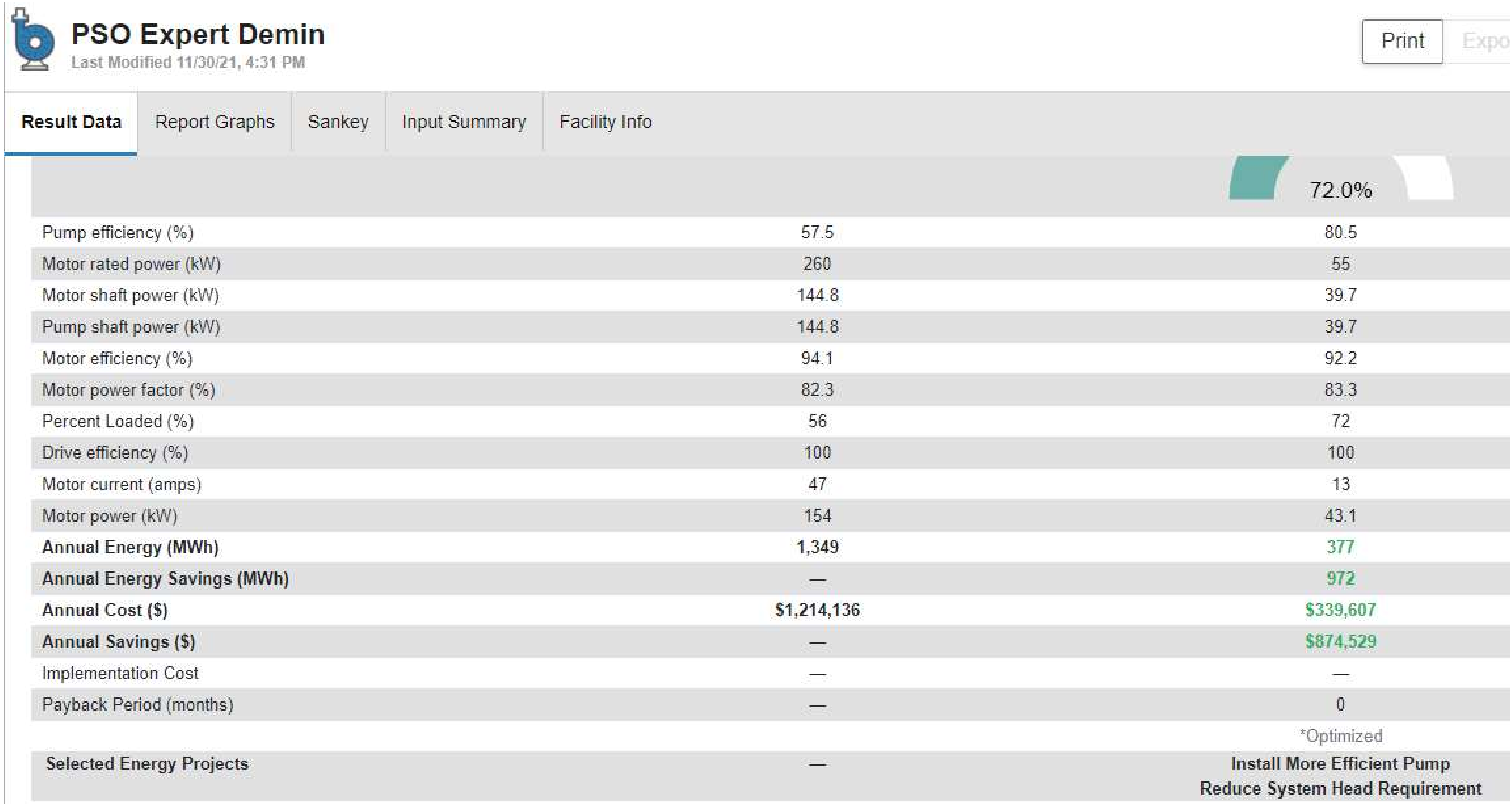
	Baseline
Percent Savings (%)	— —
Pump efficiency (%)	57.5
Motor rated power (kW)	260
Motor shaft power (kW)	144.8
Pump shaft power (kW)	144.8
Motor efficiency (%)	94.1
Motor power factor (%)	82.3
Percent Loaded (%)	56
Drive efficiency (%)	100
Motor current (A)	47
Motor power (kW)	154
Annual Energy (MWh)	**1,349**
Annual Energy Savings (MWh)	**—**
Annual Cost	**$1,214,136**
Annual Savings	**—**

Considering what was really required to fulfil the system's

Ultimate goal

Cast an entirely different light on the opportunity

MEASUR applied to new *Process* requirements



PSO Expert Demin

Last Modified 11/30/21, 4:31 PM

Pump efficiency (%)	57.5	80.5
Motor rated power (kW)	260	55
Motor shaft power (kW)	144.8	39.7
Pump shaft power (kW)	144.8	39.7
Motor efficiency (%)	94.1	92.2
Motor power factor (%)	82.3	83.3
Percent Loaded (%)	56	72
Drive efficiency (%)	100	100
Motor current (amps)	47	13
Motor power (kW)	154	43.1
Annual Energy (MWh)	**1,349**	**377**
Annual Energy Savings (MWh)	—	**972**
Annual Cost ($)	**$1,214,136**	**$339,607**
Annual Savings ($)	—	**$874,529**
Implementation Cost	—	—
Payback Period (months)	—	0
		*Optimized
Selected Energy Projects	—	**Install More Efficient Pump** **Reduce System Head Requirement**

Options Considered

- Trim the pump impeller
- Get a new, smaller pump
- Add a VSD

But what was finally decided was a little unconventional

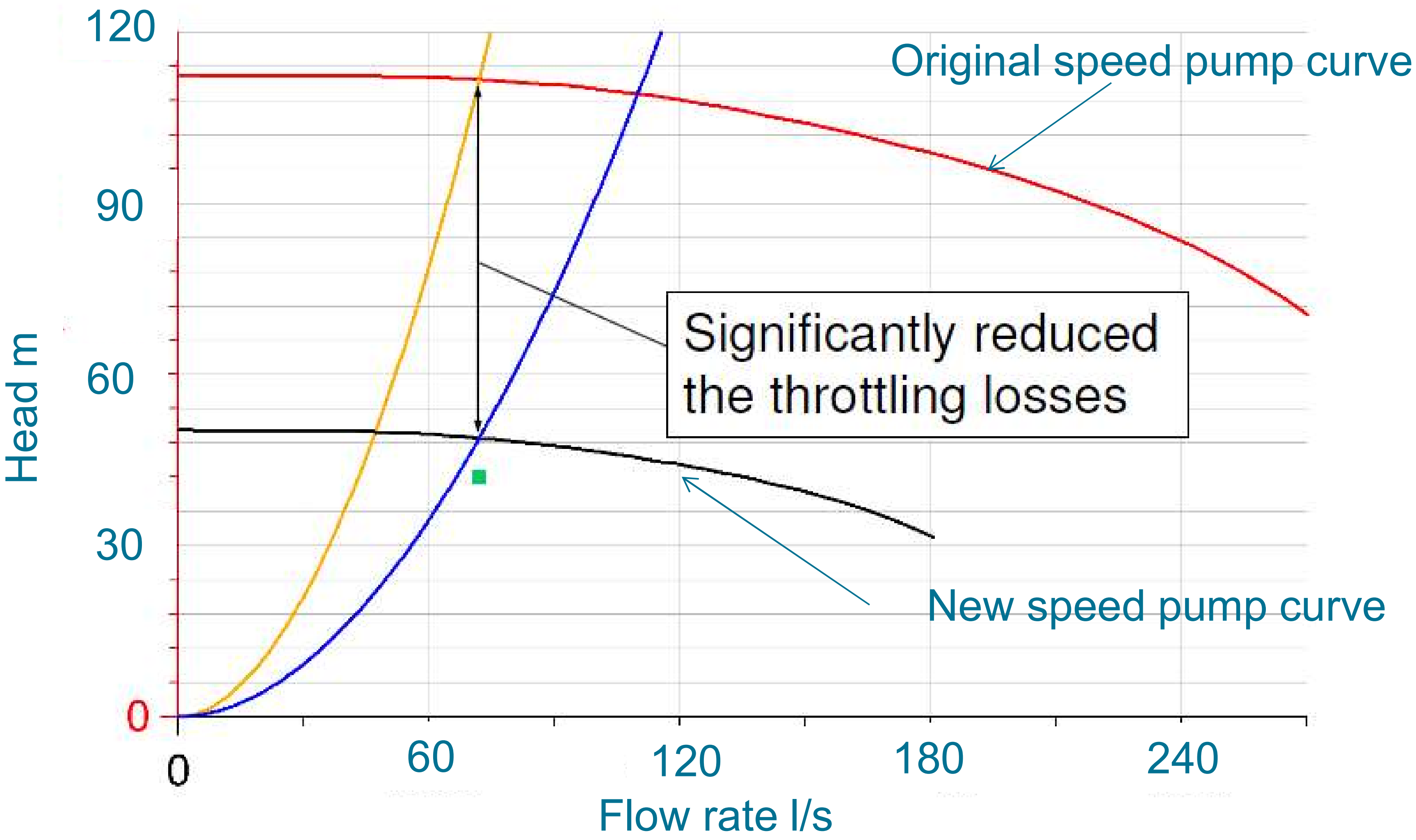
A Novel Solution…

Change of motor

A 93 kW 6-pole (1190 rpm) motor was installed on an existing demineralized water pump

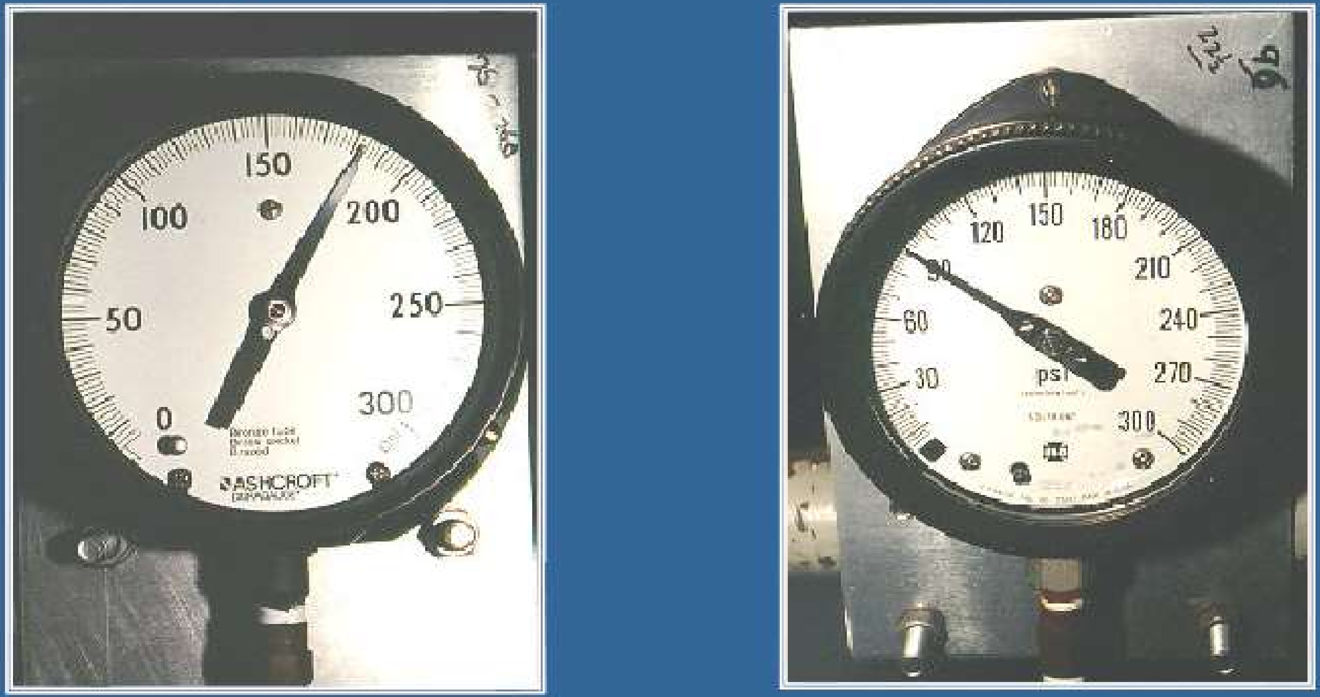
A motor with a broken foot was replaced

Avoided Throttle Losses



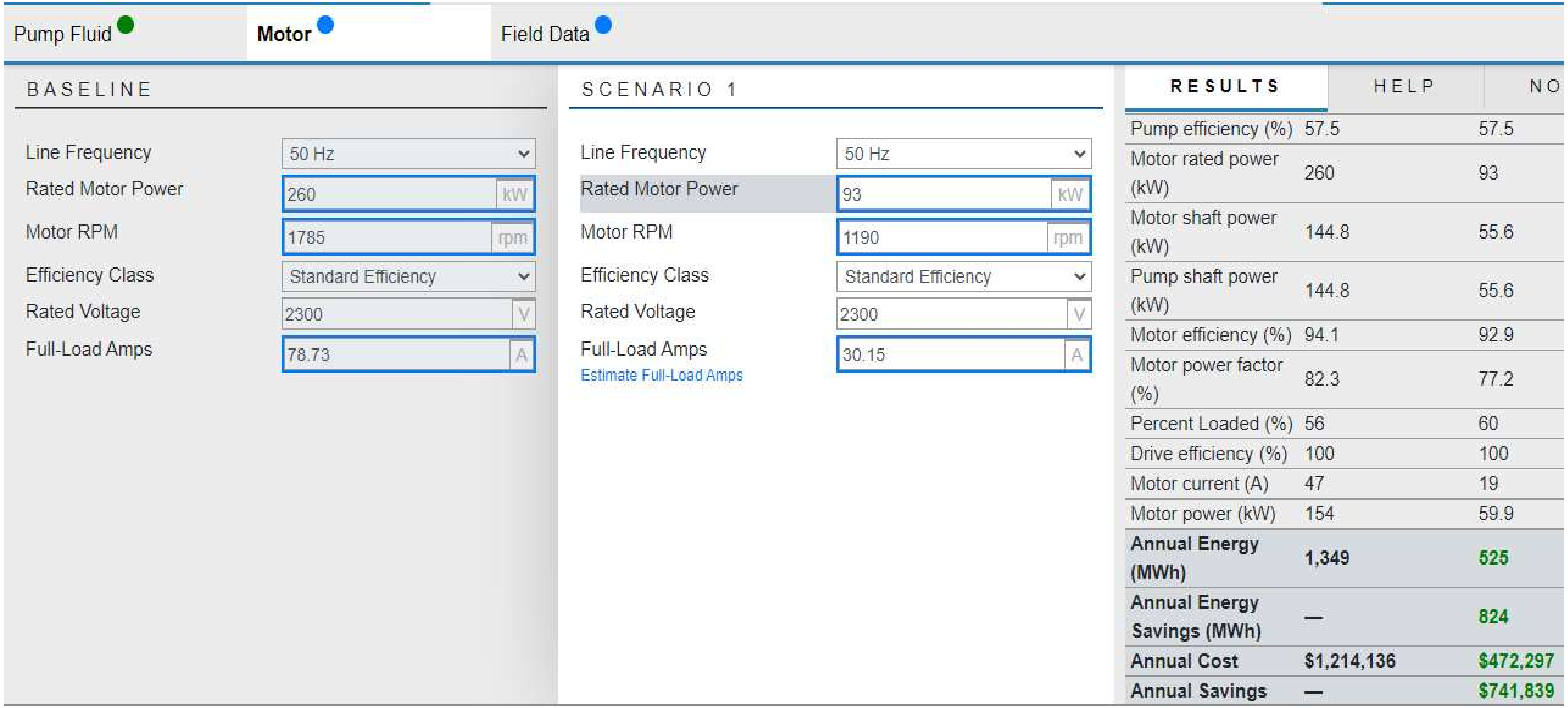
Head Reduced (psig)

By slowing the motor down, the operating head was dramatically reduced, even at the same flow rate



Discharge gauges on identical parallel pumps; left gauge is for a pump driven by a 4-pole motor, right gauge is for the pump with a 6-pole motor. Note: suction is ~ 25 psig.

Actual Implementation Simulated in MEASUR



BASELINE

Line Frequency: 50 Hz
Rated Motor Power: 260 kW
Motor RPM: 1785 rpm
Efficiency Class: Standard Efficiency
Rated Voltage: 2300 V
Full-Load Amps: 78.73 A

SCENARIO 1

Line Frequency: 50 Hz
Rated Motor Power: 93 kW
Motor RPM: 1190 rpm
Efficiency Class: Standard Efficiency
Rated Voltage: 2300 V
Full-Load Amps: 30.15 A
Estimate Full-Load Amps

Pump efficiency (%)	57.5	57.5
Motor rated power (kW)	260	93
Motor shaft power (kW)	144.8	55.6
Pump shaft power (kW)	144.8	55.6
Motor efficiency (%)	94.1	92.9
Motor power factor (%)	82.3	77.2
Percent Loaded (%)	56	60
Drive efficiency (%)	100	100
Motor current (A)	47	19
Motor power (kW)	154	59.9
Annual Energy (MWh)	**1,349**	**525**
Annual Energy Savings (MWh)	**—**	**824**
Annual Cost	**$1,214,136**	**$472,297**
Annual Savings	**—**	**$741,839**

Savings

Energy and EGP Savings:

- Annual electricity cost reduction from this change is almost EGP 742 000 (other changes made to the system)
- Reduction in energy is 824 000kWh/y
- Installed new motor and cable cost was EGP 200 000
- Capital cost repaid in < 3 months

Other Benefits

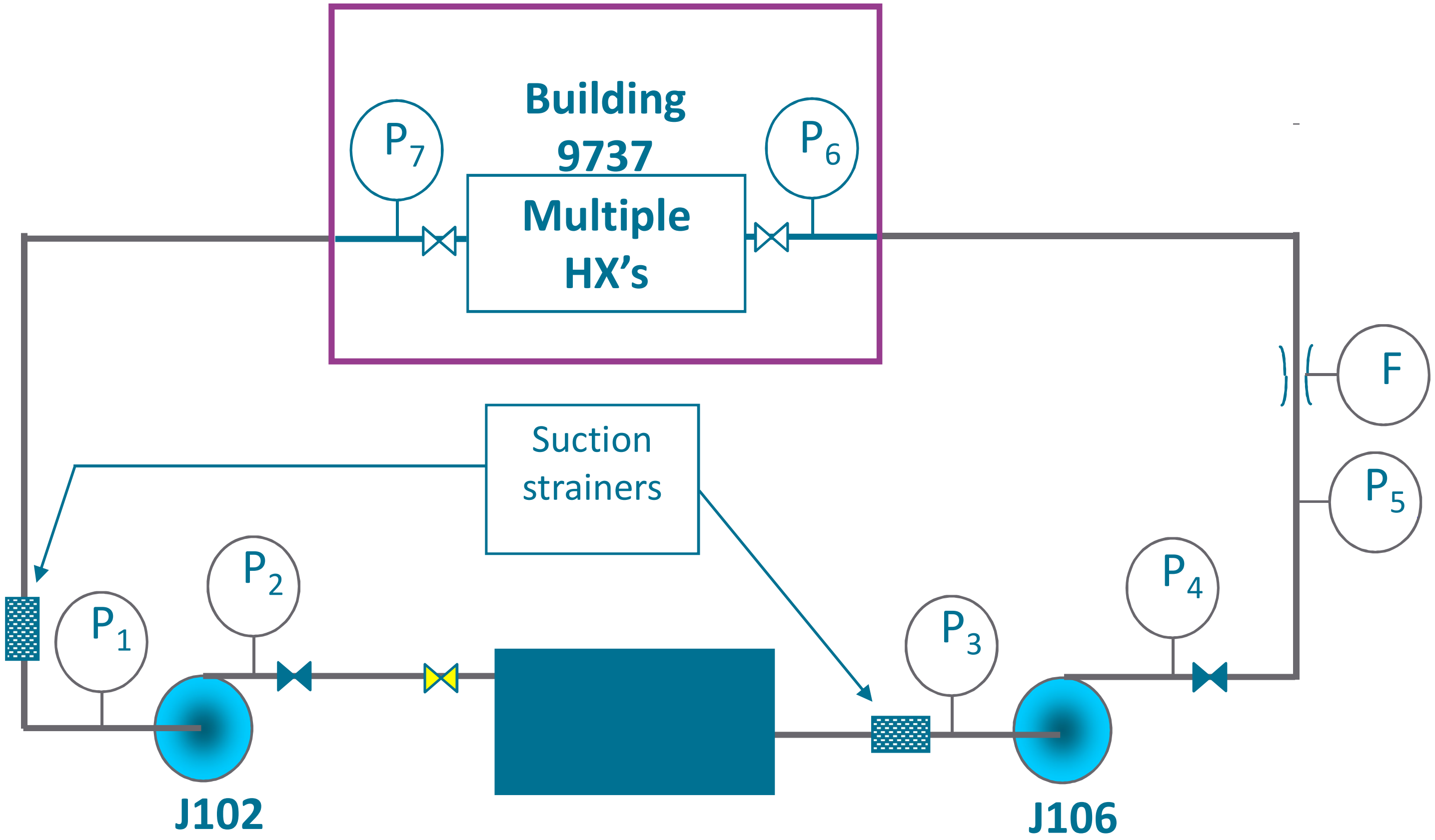
There were some other important tangential benefits:

- Seal face speed is reduced, seal life being extended
- The pump is more hydraulically stable, which means fewer maintenance problems are expected
- Noise levels are reduced – both in the pump house and in the main Fusion building (hearing protection is no longer required)

MEASUR Example 2

Chilled Water Secondary Pump J106

MEASUR: Example 2



- *8760 hrs / year*
- *US$0.1 / kWh*

Slide Courtesy of Oak Ridge National Laboratory

Pump Data

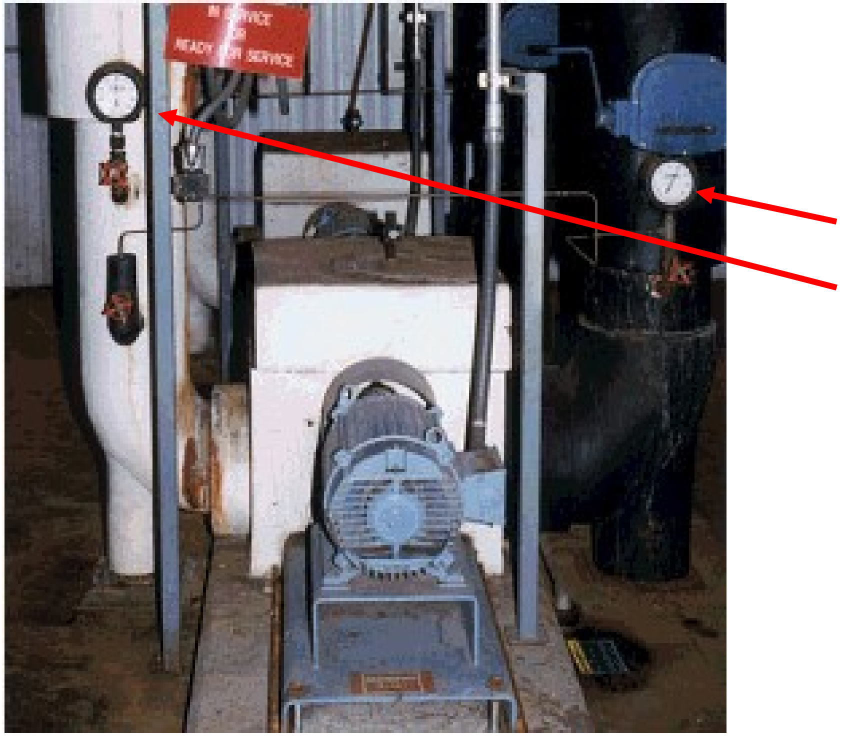


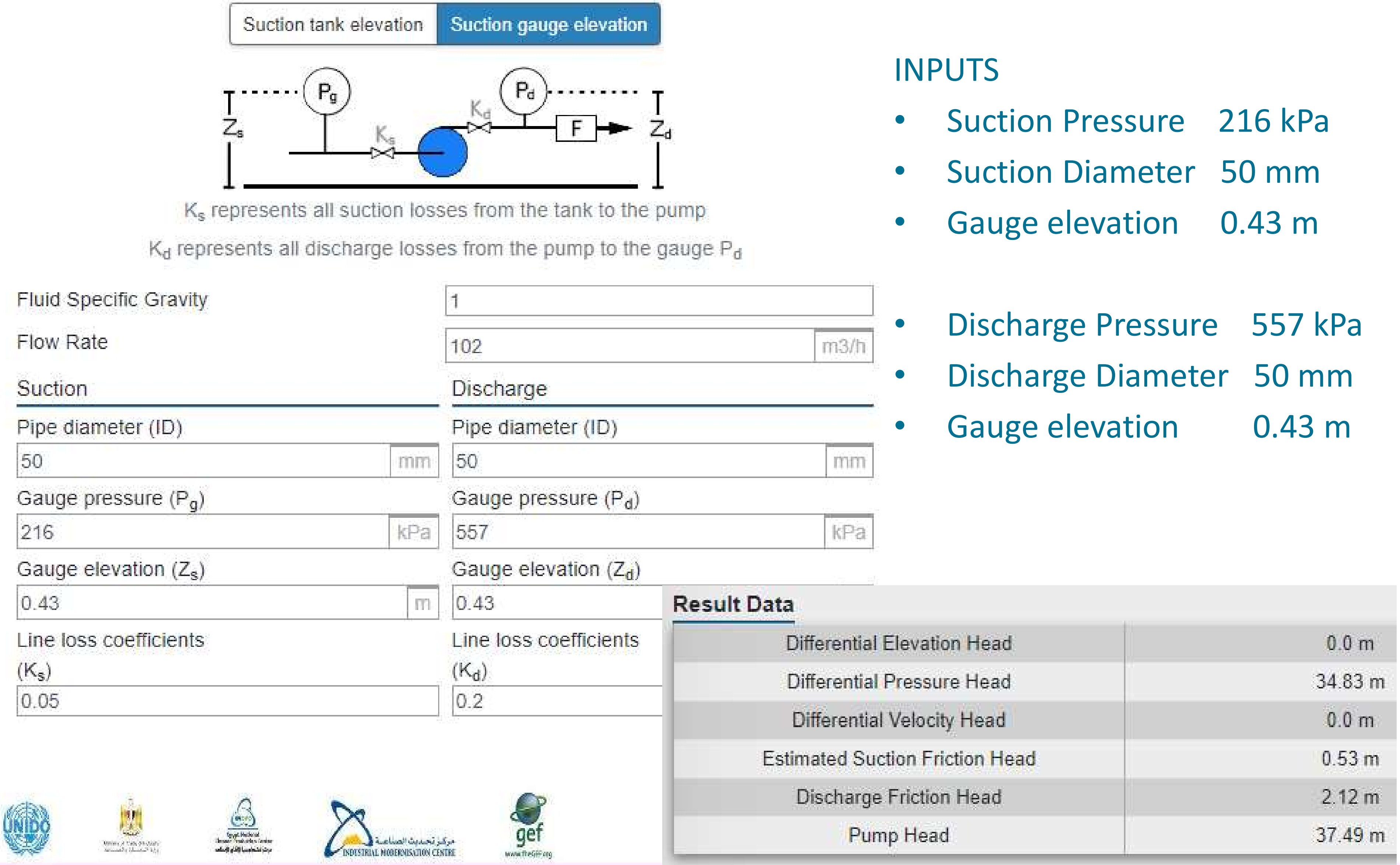
Photo Courtesy of Oak Ridge National Laboratory

Observed:

- Suction Pressure 216 kPa
- Discharge Pressure 557 kPa
- Gauge elevation 0.43 m

- Total head 37.5 m
- Flow rate 102 m^3/h

Using MEASUR Head Tool



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Fluid Specific Gravity	1
Flow Rate	102 m3/h

Suction	Discharge
Pipe diameter (ID) 50 mm	Pipe diameter (ID) 50 mm
Gauge pressure (P_g) 216 kPa	Gauge pressure (P_d) 557 kPa
Gauge elevation (Z_s) 0.43 m	Gauge elevation (Z_d) 0.43
Line loss coefficients (K_s) 0.05	Line loss coefficients (K_d) 0.2

Result Data

Differential Elevation Head	0.0 m
Differential Pressure Head	34.83 m
Differential Velocity Head	0.0 m
Estimated Suction Friction Head	0.53 m
Discharge Friction Head	2.12 m
Pump Head	37.49 m

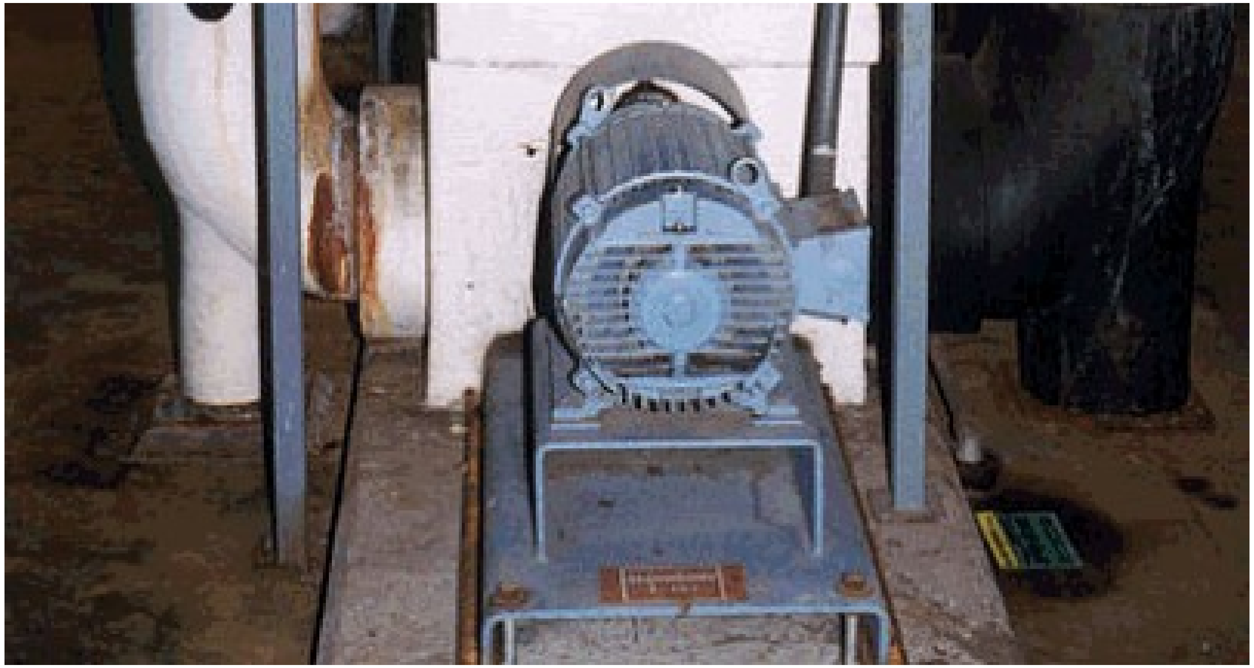
INPUTS

Suction Pressure	216 kPa
Suction Diameter	50 mm
Gauge elevation	0.43 m
Discharge Pressure	557 kPa
Discharge Diameter	50 mm
Gauge elevation	0.43 m

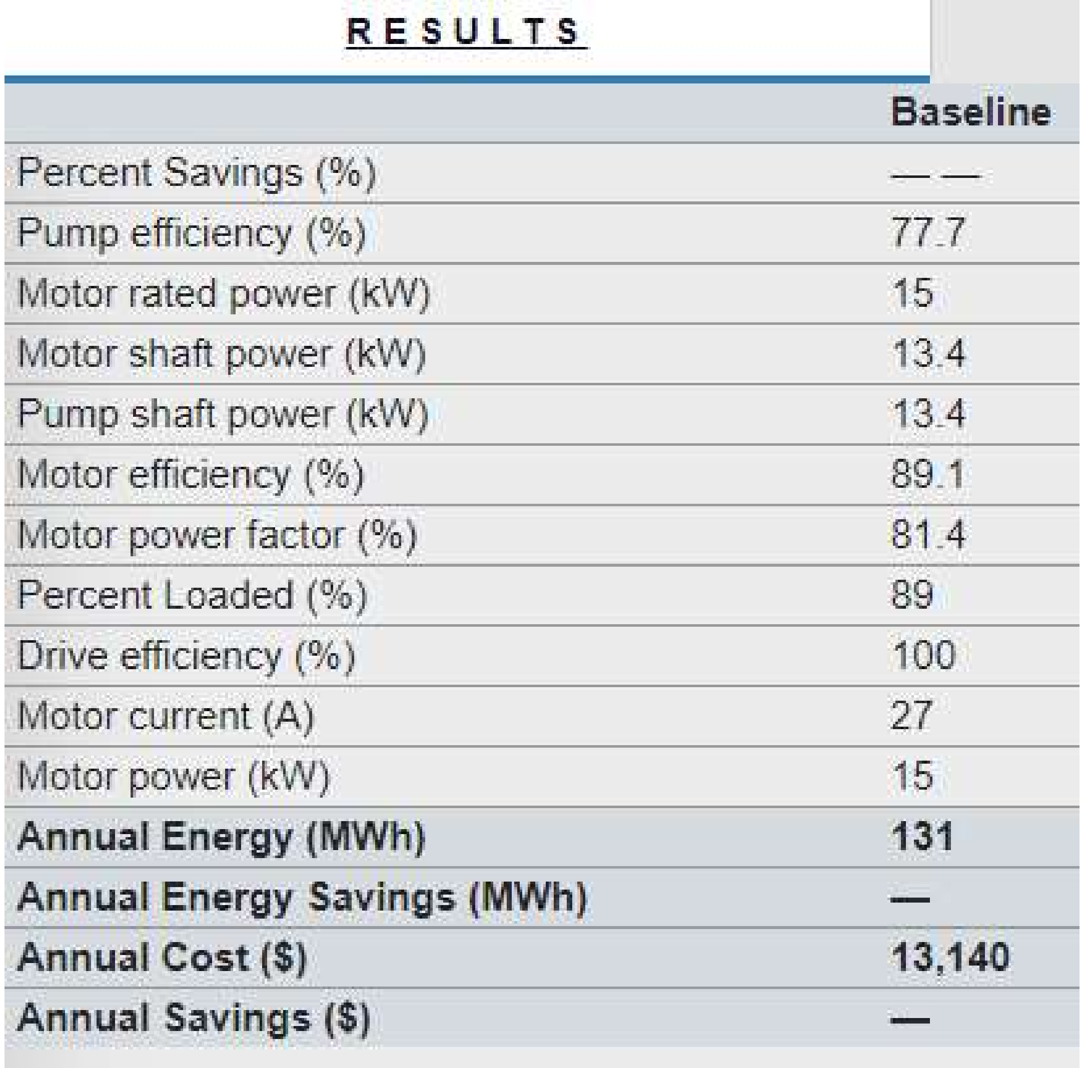
Motor Data (J106)

Nameplate:

- 15 kW
- 1460 rpm @ 50 Hz
- 400 V
- 29.6 A (full load)
- IE1



Baseline Results



RESULTS

	Baseline
Percent Savings (%)	— —
Pump efficiency (%)	77.7
Motor rated power (kW)	15
Motor shaft power (kW)	13.4
Pump shaft power (kW)	13.4
Motor efficiency (%)	89.1
Motor power factor (%)	81.4
Percent Loaded (%)	89
Drive efficiency (%)	100
Motor current (A)	27
Motor power (kW)	15
Annual Energy (MWh)	**131**
Annual Energy Savings (MWh)	**—**
Annual Cost ($)	**13,140**
Annual Savings ($)	**—**

BUT supply and demand are unbalanced

There is > 158 kPa pressure drop across the throttled valve; the downstream pressure was measured to be 379.2 kPa (3 meters above floor)

Suction gauge:	216.5 kPa
Discharge gauge:	379.2 kPa

Total pump head: 18.6 m

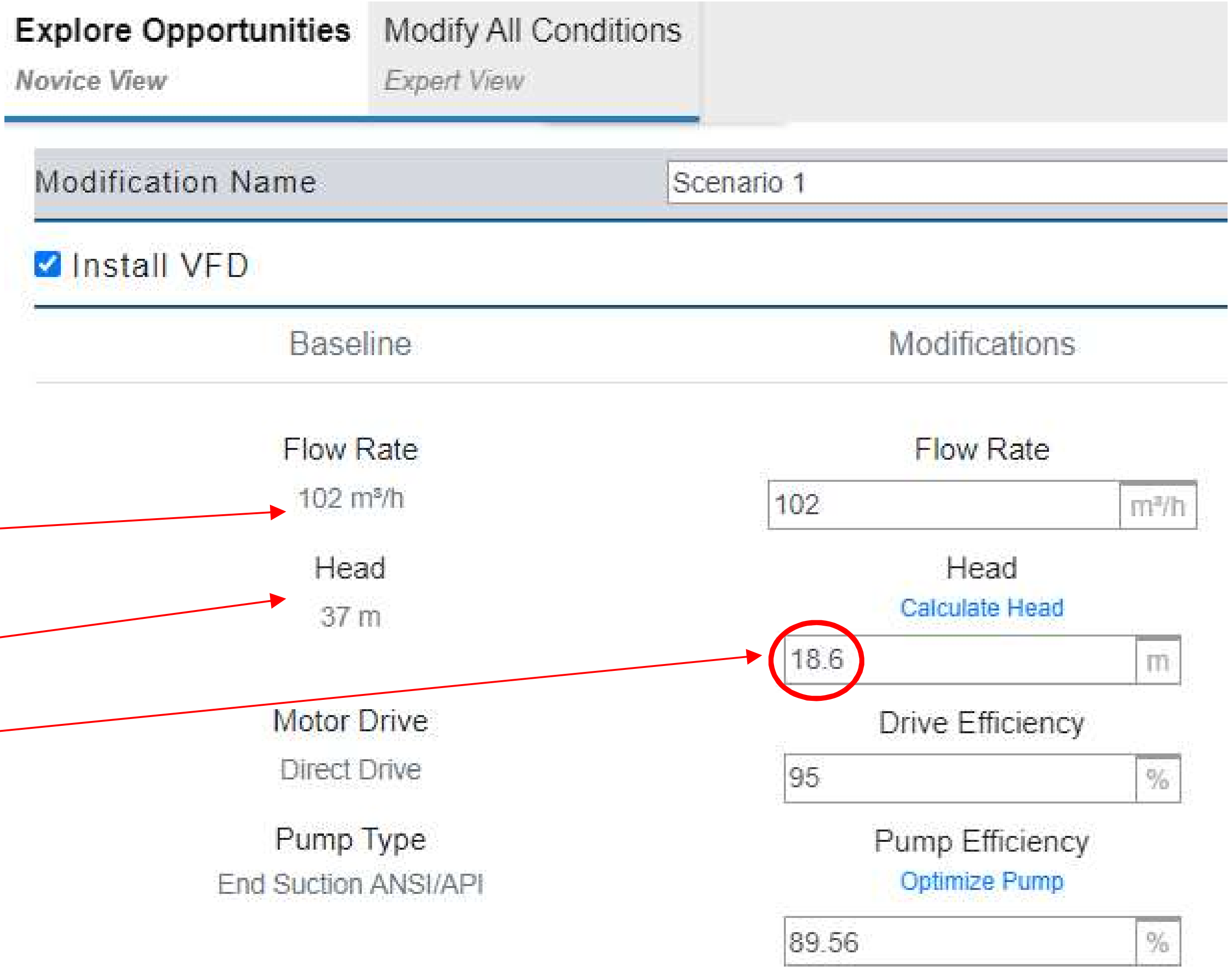
*This is the **net** required head*

Slide Courtesy of Oak Ridge National Laboratory

Opportunity: Install VSD instead of Throttle

Opportunity:

- Use a VSD instead of a throttle valve for flow control
- Flow is the same
- Head required with throttle is 37 m
- Head required with VSD and no throttle is 18.6 m

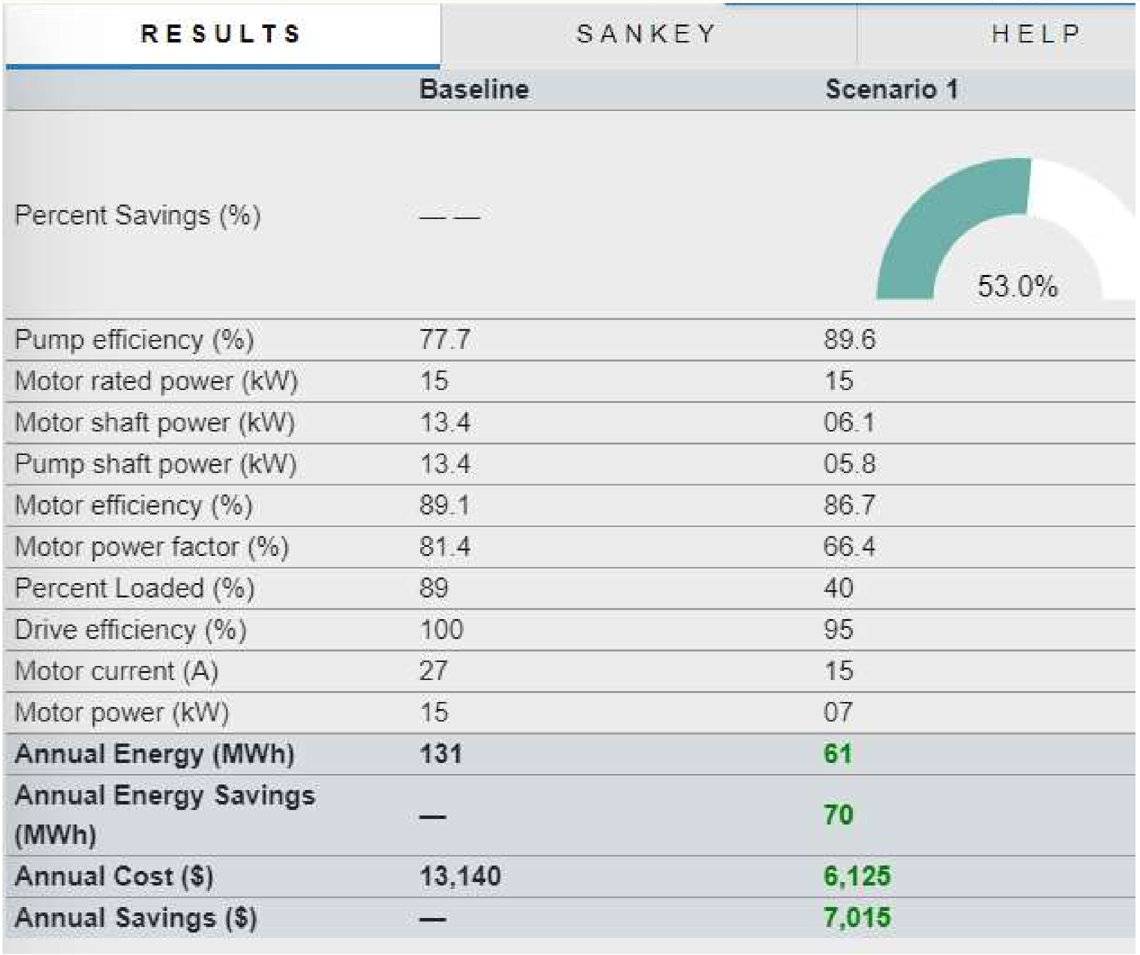


Modification Name: Scenario 1

☑ Install VFD

Baseline	Modifications
Flow Rate 102 m³/h	Flow Rate 102 m³/h
Head 37 m	Head Calculate Head 18.6 m
Motor Drive Direct Drive	Drive Efficiency 95 %
Pump Type End Suction ANSI/API	Pump Efficiency Optimize Pump 89.56 %

Output Results



	Baseline	Scenario 1
Percent Savings (%)	— —	53.0%
Pump efficiency (%)	77.7	89.6
Motor rated power (kW)	15	15
Motor shaft power (kW)	13.4	06.1
Pump shaft power (kW)	13.4	05.8
Motor efficiency (%)	89.1	86.7
Motor power factor (%)	81.4	66.4
Percent Loaded (%)	89	40
Drive efficiency (%)	100	95
Motor current (A)	27	15
Motor power (kW)	15	07
Annual Energy (MWh)	**131**	**61**
Annual Energy Savings (MWh)	**—**	**70**
Annual Cost ($)	**13,140**	**6,125**
Annual Savings ($)	**—**	**7,015**

Savings:

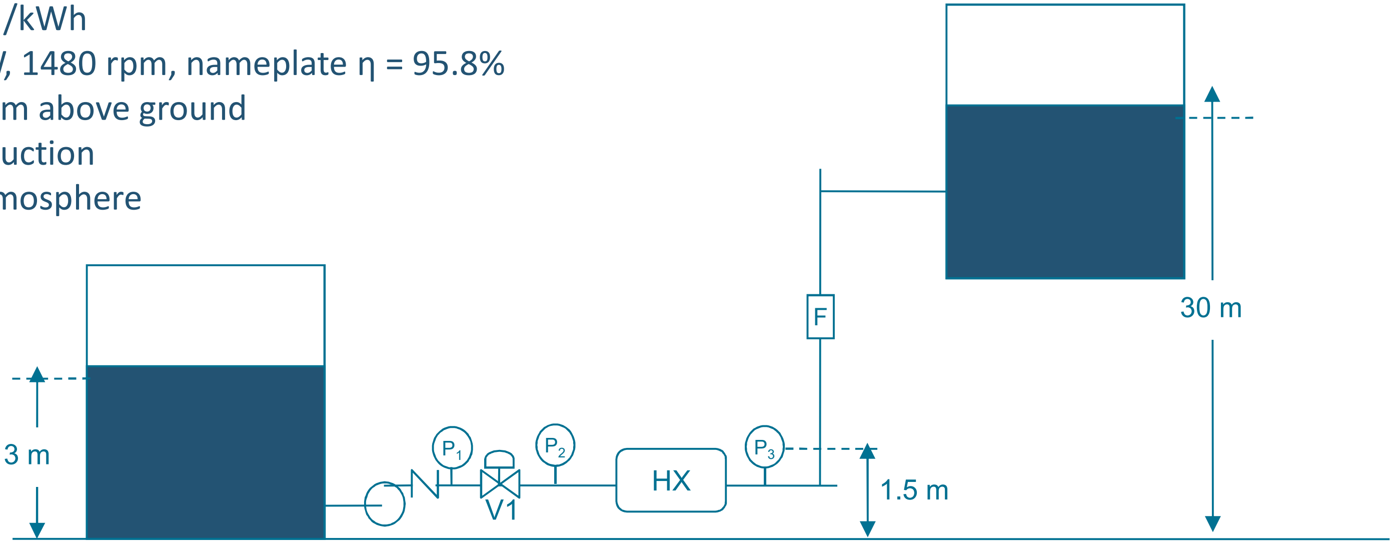
- Original operating cost = $13,140. New operating cost = $6,125
- Will save 47% of baseline consumption, savings of 70 000 kWh

MEASUR Example 3

Throttled Control Valve Losses

System Configuration and Operating Data

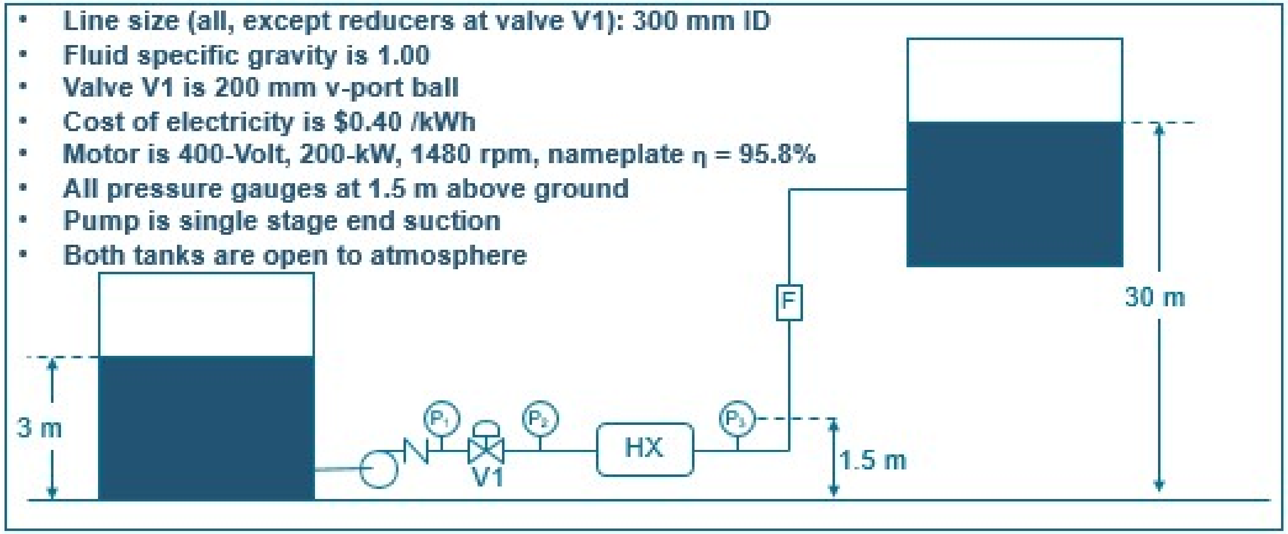
- Line size (all, except reducers at valve V1): 300 mm ID
- Fluid specific gravity is 1.00
- Valve V1 is 200 mm v-port ball
- Cost of electricity is $0.40 /kWh
- Motor is 400-Volt, 200-kW, 1480 rpm, nameplate η = 95.8%
- All pressure gauges at 1.5 m above ground
- Pump is single stage end suction
- Both tanks are open to atmosphere



Measured Operating Data

Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

Static Head



- Line size (all, except reducers at valve V1): 300 mm ID
- Fluid specific gravity is 1.00
- Valve V1 is 200 mm v-port ball
- Cost of electricity is $0.40 /kWh
- Motor is 400-Volt, 200-kW, 1480 rpm, nameplate η = 95.8%
- All pressure gauges at 1.5 m above ground
- Pump is single stage end suction
- Both tanks are open to atmosphere

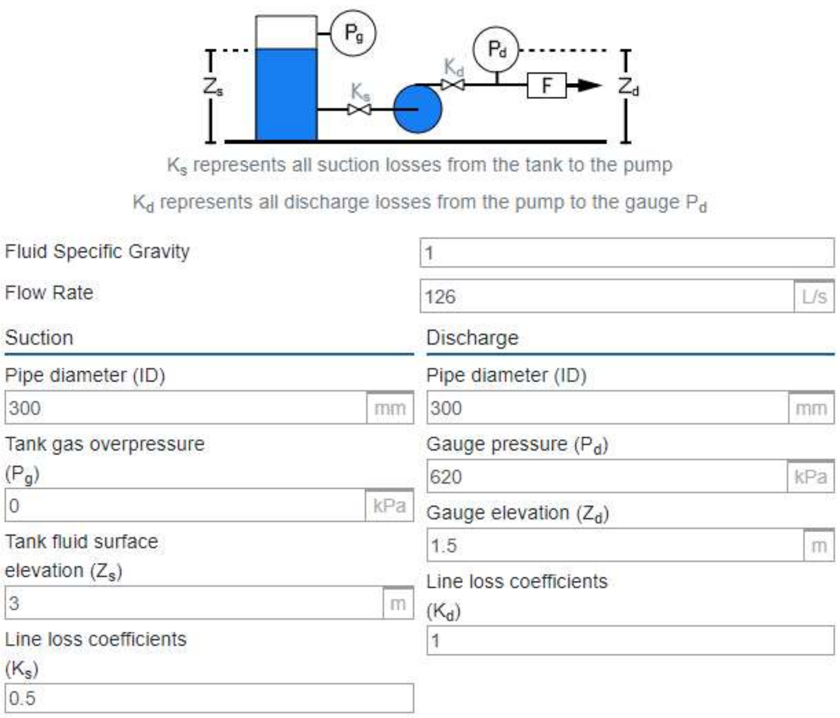
1. What is the static head for this system?

2. What pressure would you expect at P1 with the pump off?

Pump Head Condition A

Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

3. For Condition A, estimate the **actual pump head** (use the MEASUR head calculator). Calculate the optimised % savings and annual energy costs of operation. *(assume Ks = 0.5 loss for suction side and Kd = 1.0 loss for discharge side)*



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Fluid Specific Gravity: 1

Flow Rate: 126 L/s

Suction

Pipe diameter (ID): 300 mm

Tank gas overpressure (P_g): 0 kPa

Tank fluid surface elevation (Z_s): 3 m

Line loss coefficients (K_s): 0.5

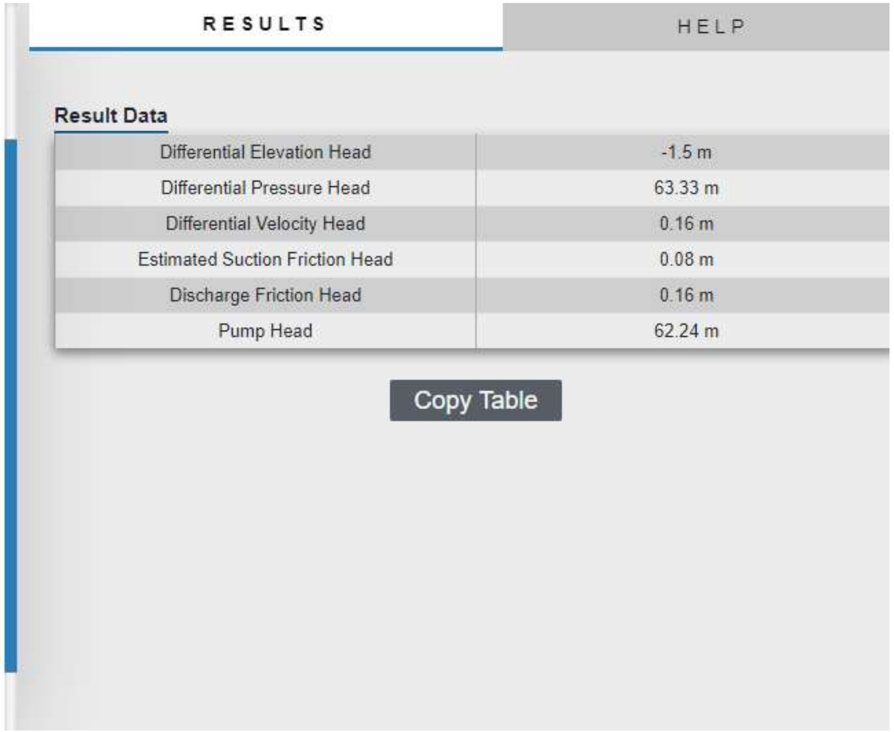
Discharge

Pipe diameter (ID): 300 mm

Gauge pressure (P_d): 620 kPa

Gauge elevation (Z_d): 1.5 m

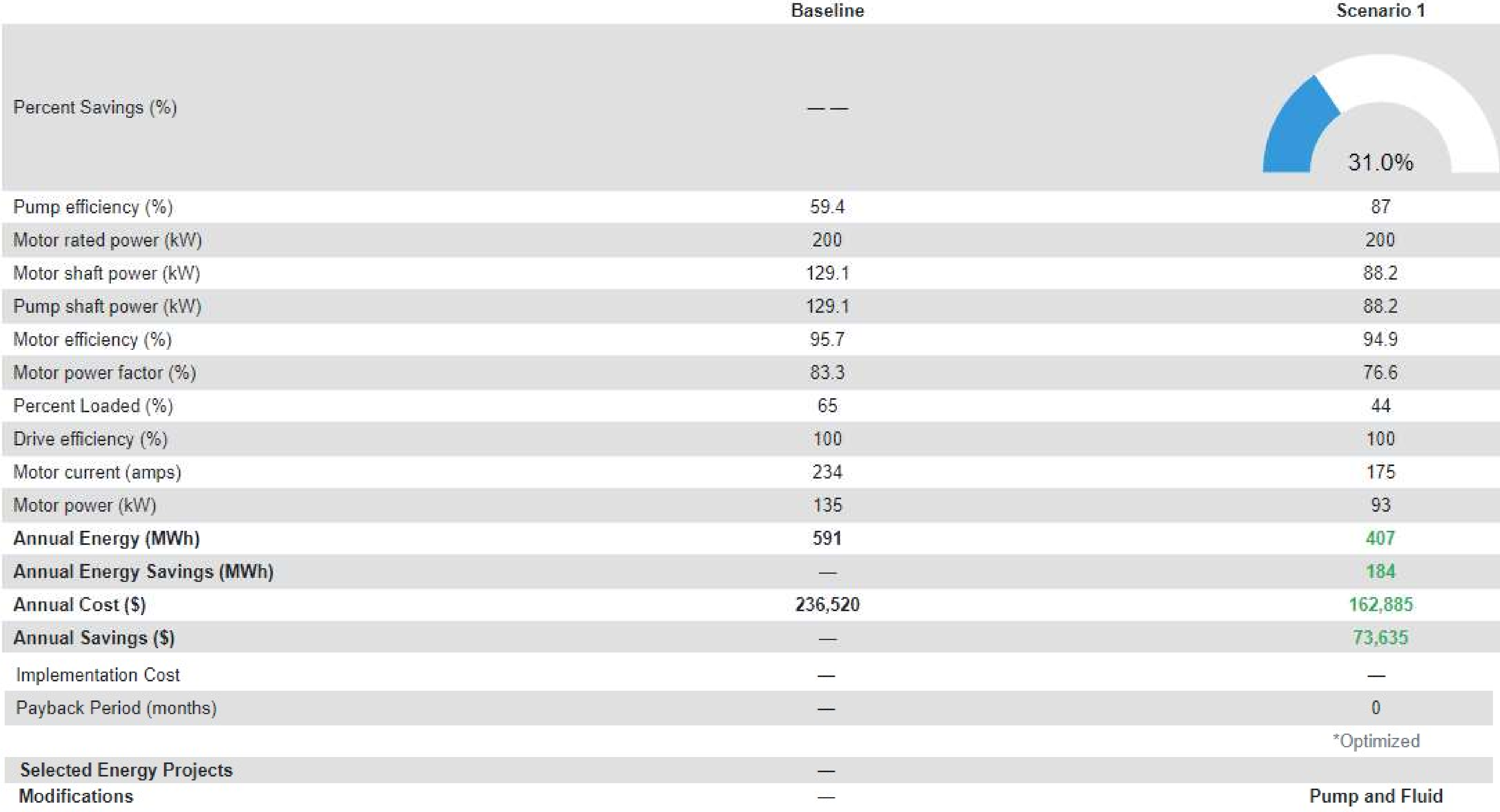
Line loss coefficients (K_d): 1



Result Data

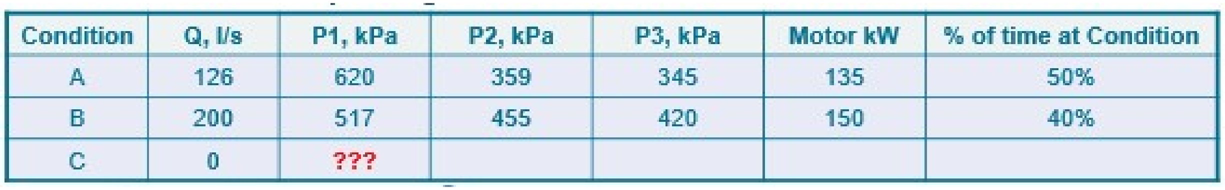
Differential Elevation Head	-1.5 m
Differential Pressure Head	63.33 m
Differential Velocity Head	0.16 m
Estimated Suction Friction Head	0.08 m
Discharge Friction Head	0.16 m
Pump Head	62.24 m

Cost and Savings Condition A



	Baseline	Scenario 1
Percent Savings (%)	— —	31.0%
Pump efficiency (%)	59.4	87
Motor rated power (kW)	200	200
Motor shaft power (kW)	129.1	88.2
Pump shaft power (kW)	129.1	88.2
Motor efficiency (%)	95.7	94.9
Motor power factor (%)	83.3	76.6
Percent Loaded (%)	65	44
Drive efficiency (%)	100	100
Motor current (amps)	234	175
Motor power (kW)	135	93
Annual Energy (MWh)	**591**	**407**
Annual Energy Savings (MWh)	—	**184**
Annual Cost ($)	**236,520**	**162,885**
Annual Savings ($)	—	**73,635**
Implementation Cost	—	—
Payback Period (months)	—	0
		*Optimized
Selected Energy Projects	—	
Modifications	—	**Pump and Fluid**

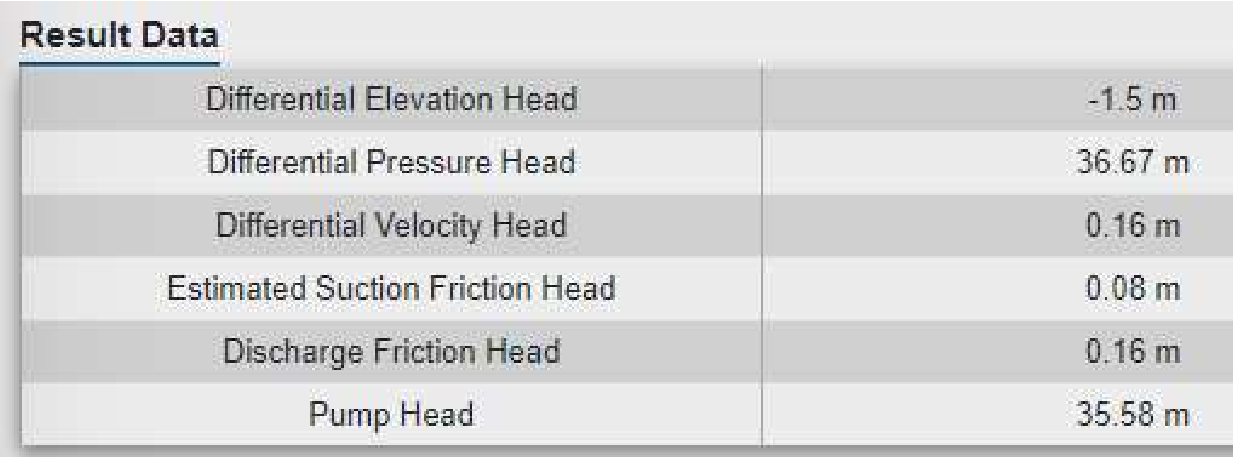
Pump Head Condition A



Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

4. For condition A, what is the system head as measured after the valve (at P2)?

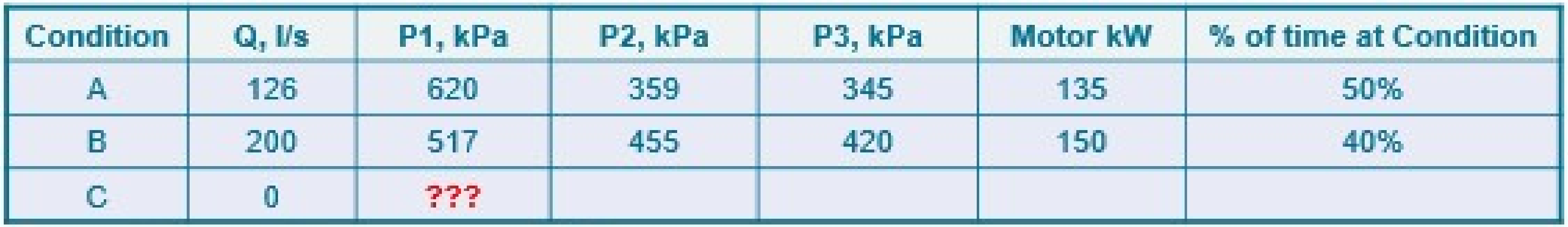
Answer:



Result Data

Differential Elevation Head	-1.5 m
Differential Pressure Head	36.67 m
Differential Velocity Head	0.16 m
Estimated Suction Friction Head	0.08 m
Discharge Friction Head	0.16 m
Pump Head	35.58 m

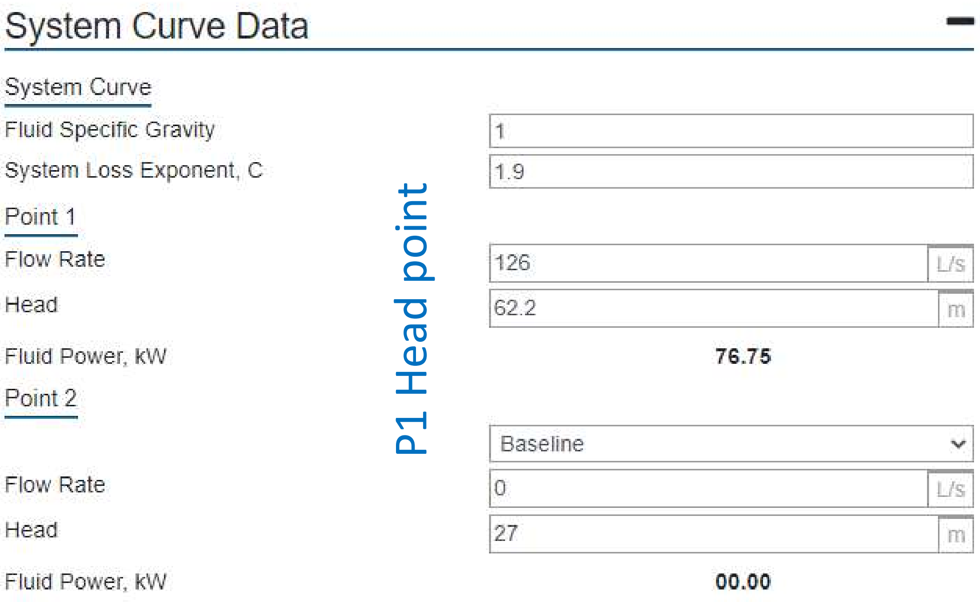
System Curve Condition A



Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

5. Using the pump curve calculator in MEASUR, develop two system curves for **Condition A** :

- System curve for P1 Head point (which is the current throttled valve operation)
- System curve for P2 Head point (if the control valve were removed and a VSD used)



System Curve Data

System Curve

Fluid Specific Gravity: 1

System Loss Exponent, C: 1.9

Point 1

Flow Rate: 126 L/s

Head: 62.2 m

Fluid Power, kW: 76.75

Point 2

Baseline

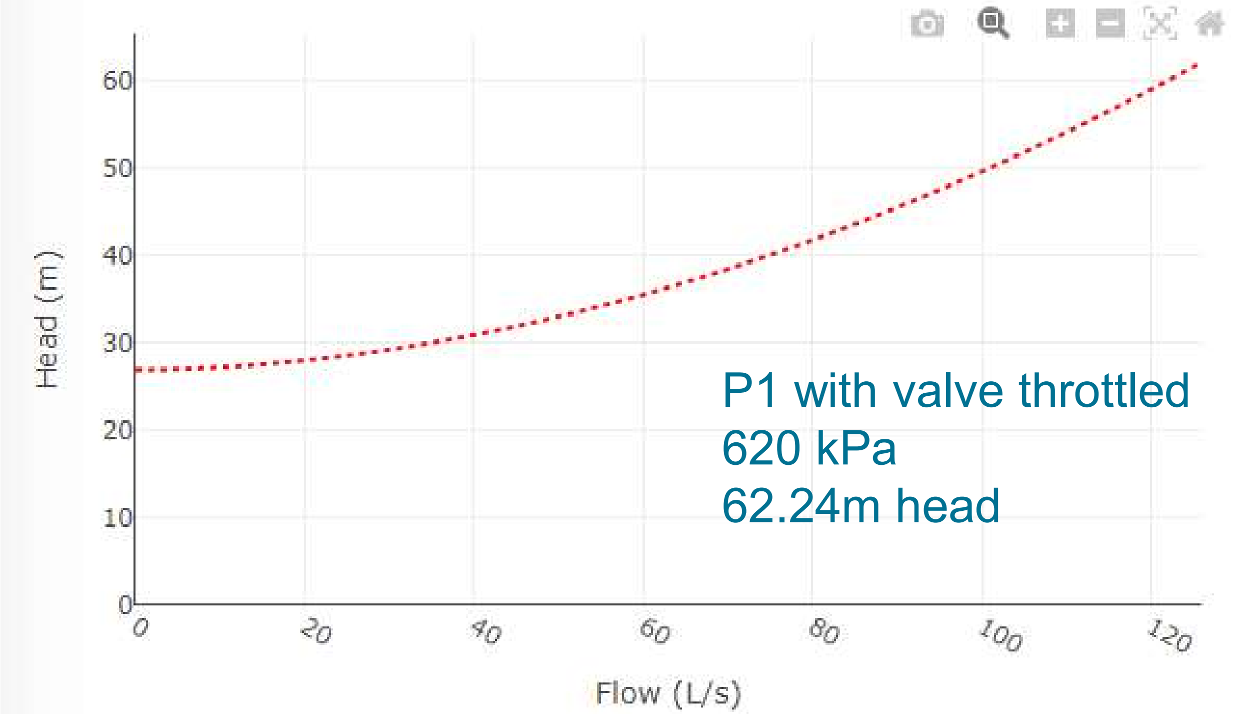
Flow Rate: 0 L/s

Head: 27 m

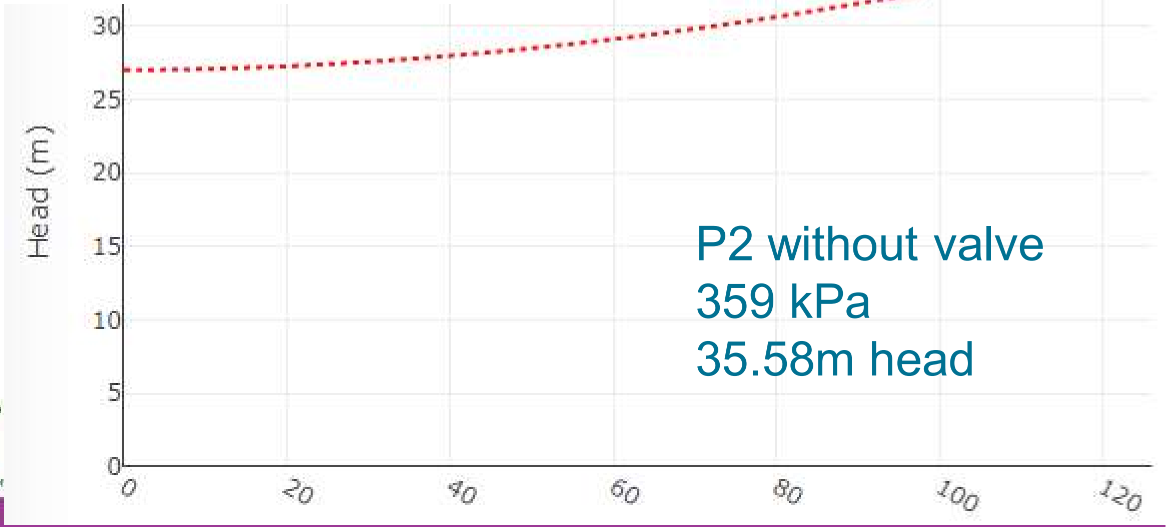
Fluid Power, kW: 00.00

P1 Head point

System Curve Condition A



P1 with valve throttled
620 kPa
62.24m head

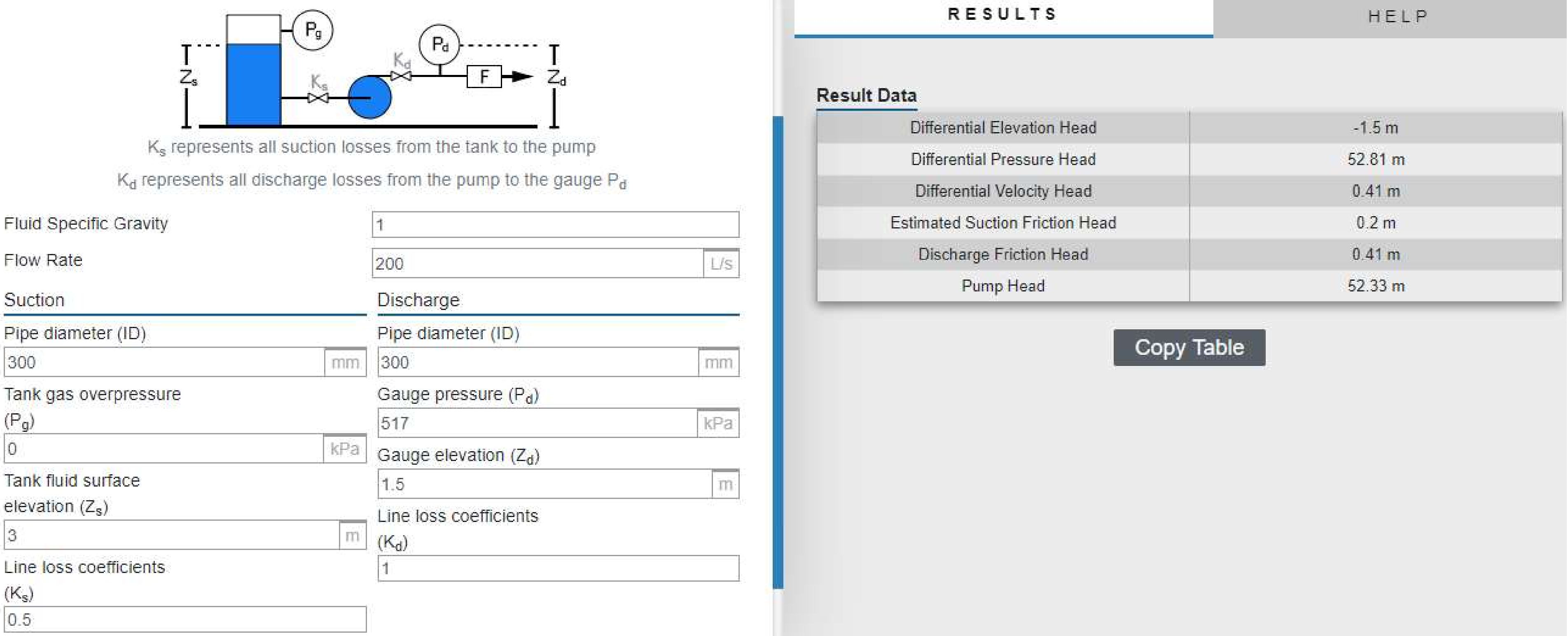


P2 without valve
359 kPa
35.58m head

Pump Head Condition B

Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

6. For Conditions B, estimate the actual pump head, optimised % savings and annual energy costs of operation.
(assume Ks = 0.5 loss for suction side and Kd = 1.0 loss for discharge side)



K_s represents all suction losses from the tank to the pump

K_d represents all discharge losses from the pump to the gauge P_d

Fluid Specific Gravity: 1
Flow Rate: 200 L/s

Suction
Pipe diameter (ID): 300 mm
Tank gas overpressure (P_g): 0 kPa
Tank fluid surface elevation (Z_s): 3 m
Line loss coefficients (K_s): 0.5

Discharge
Pipe diameter (ID): 300 mm
Gauge pressure (P_d): 517 kPa
Gauge elevation (Z_d): 1.5 m
Line loss coefficients (K_d): 1

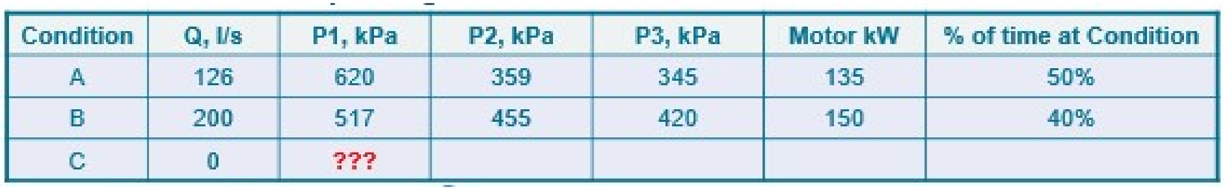
Result Data

Differential Elevation Head	-1.5 m
Differential Pressure Head	52.81 m
Differential Velocity Head	0.41 m
Estimated Suction Friction Head	0.2 m
Discharge Friction Head	0.41 m
Pump Head	52.33 m

Cost and Savings Condition B

Percent Savings (%)	— —	19.0%
Pump efficiency (%)	71.3	88
Motor rated power (kW)	200	200
Motor shaft power (kW)	143.6	116.3
Pump shaft power (kW)	143.6	116.3
Motor efficiency (%)	95.8	95.4
Motor power factor (%)	84.6	81.8
Percent Loaded (%)	72	58
Drive efficiency (%)	100	100
Motor current (amps)	256	215
Motor power (kW)	150	121.9
Annual Energy (MWh)	**526**	**427**
Annual Energy Savings (MWh)	—	**98.5**
Annual Cost ($)	**210,240**	**170,831**
Annual Savings ($)	—	**39,409**
Implementation Cost	—	—
Payback Period (months)	—	0
		*Optimized
Selected Energy Projects	—	
Modifications	—	**Pump and Fluid**

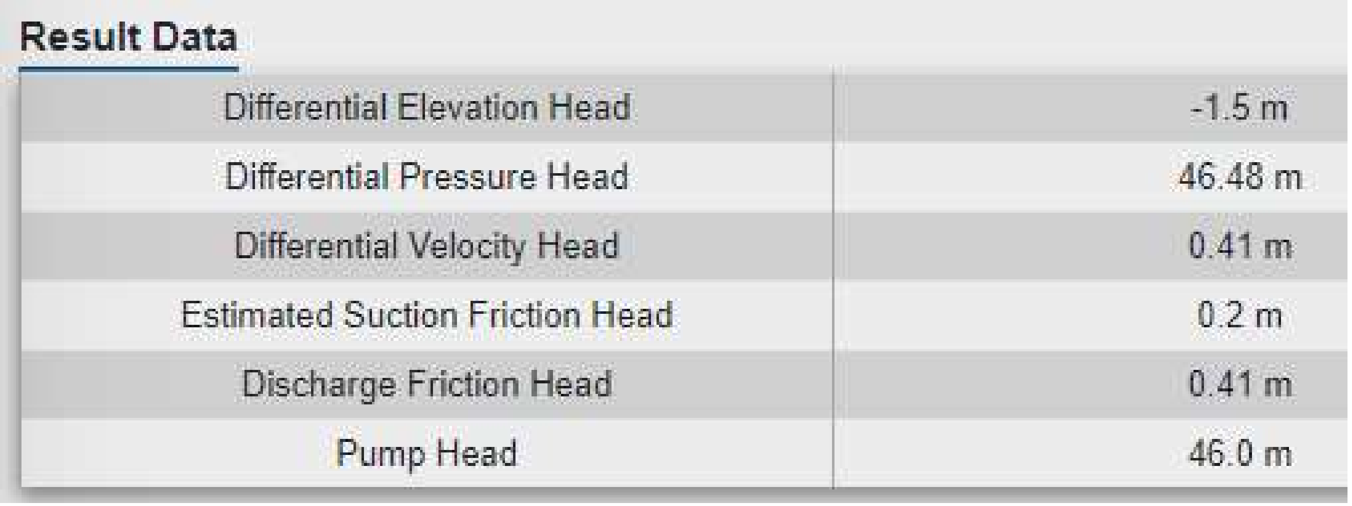
Pump Head Condition B



Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

7. For condition B, what is the system head as measured after the valve (at P2)?

Answer:



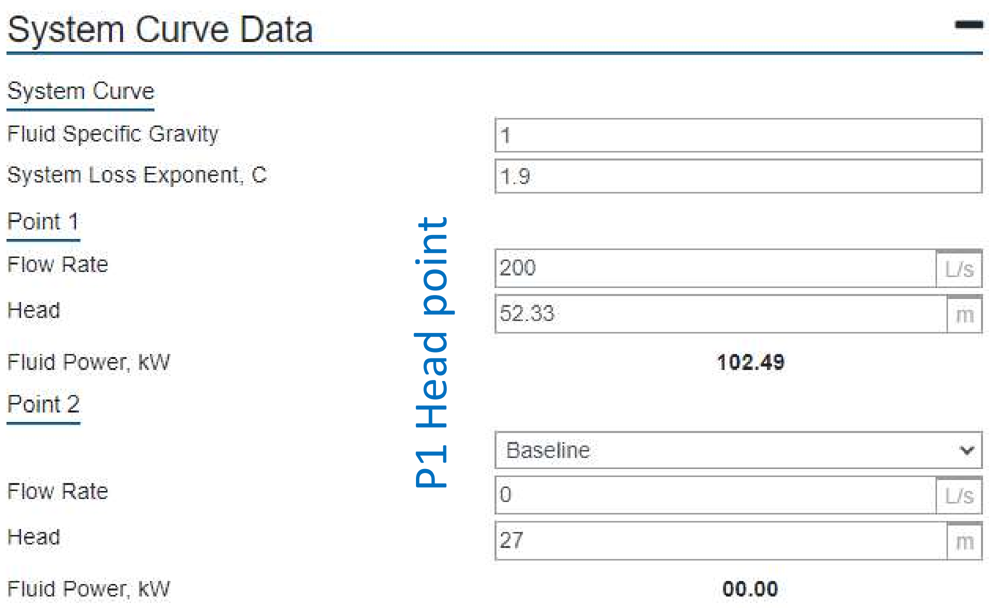
Result Data

Differential Elevation Head	-1.5 m
Differential Pressure Head	46.48 m
Differential Velocity Head	0.41 m
Estimated Suction Friction Head	0.2 m
Discharge Friction Head	0.41 m
Pump Head	46.0 m

System Curve Condition B

Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

8. Using the pump curve calculator in MEASUR, develop system curves based on the static head and the **Condition B** flow and both P1 and P2 head points.



System Curve Data

System Curve

Fluid Specific Gravity: 1

System Loss Exponent, C: 1.9

Point 1

Flow Rate: 200 L/s

Head: 52.33 m

Fluid Power, kW: 102.49

Point 2

Baseline

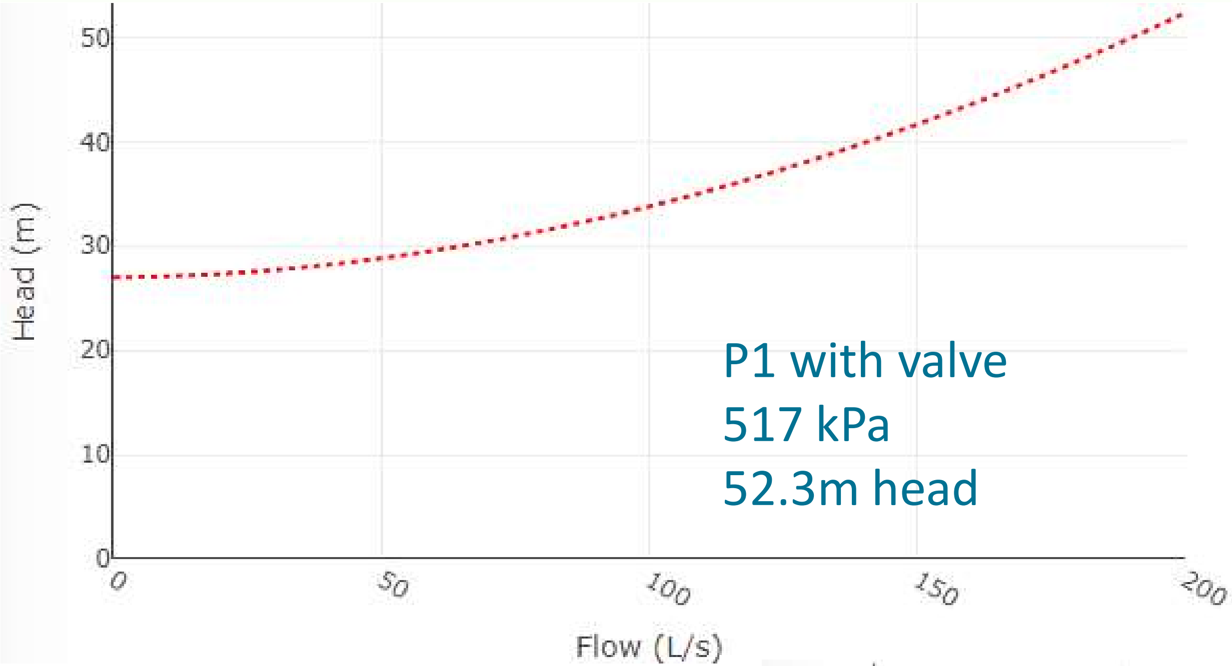
Flow Rate: 0 L/s

Head: 27 m

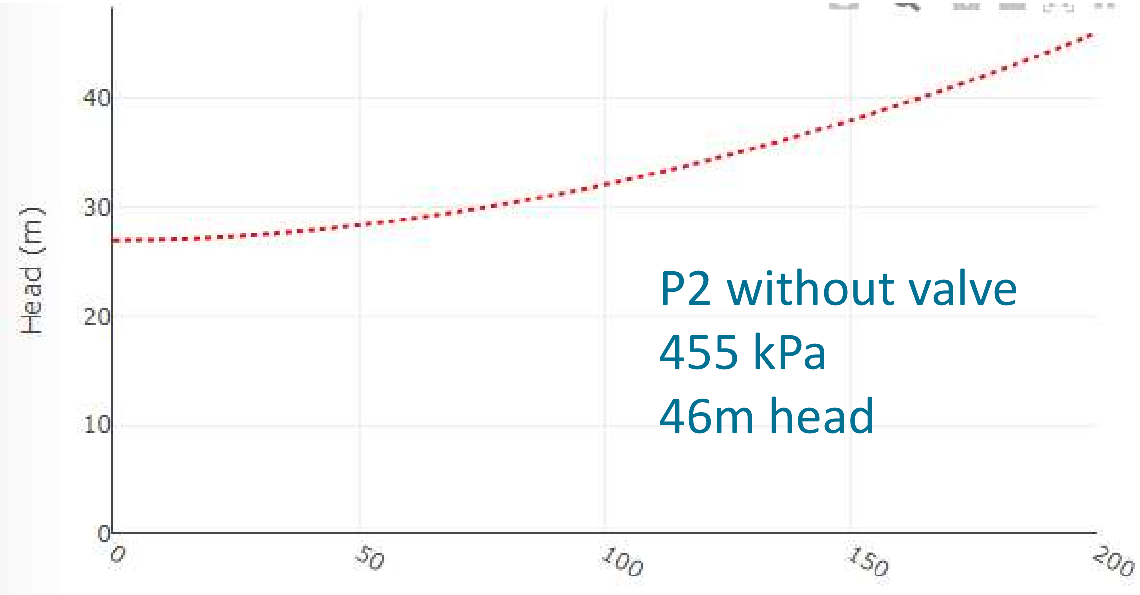
Fluid Power, kW: 00.00

P1 Head point

System Curve Condition B

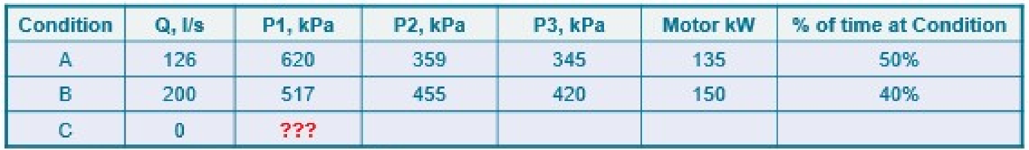


P1 with valve
517 kPa
52.3m head



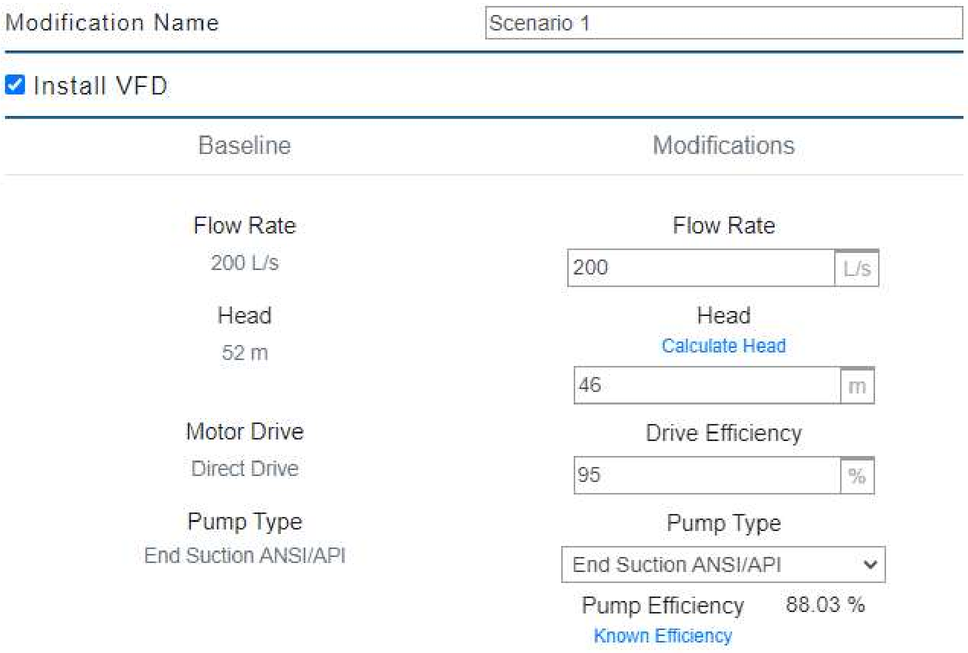
P2 without valve
455 kPa
46m head

Eliminate Control Valve Savings Condition B



Condition	Q, l/s	P1, kPa	P2, kPa	P3, kPa	Motor kW	% of time at Condition
A	126	620	359	345	135	50%
B	200	517	455	420	150	40%
C	0	???				

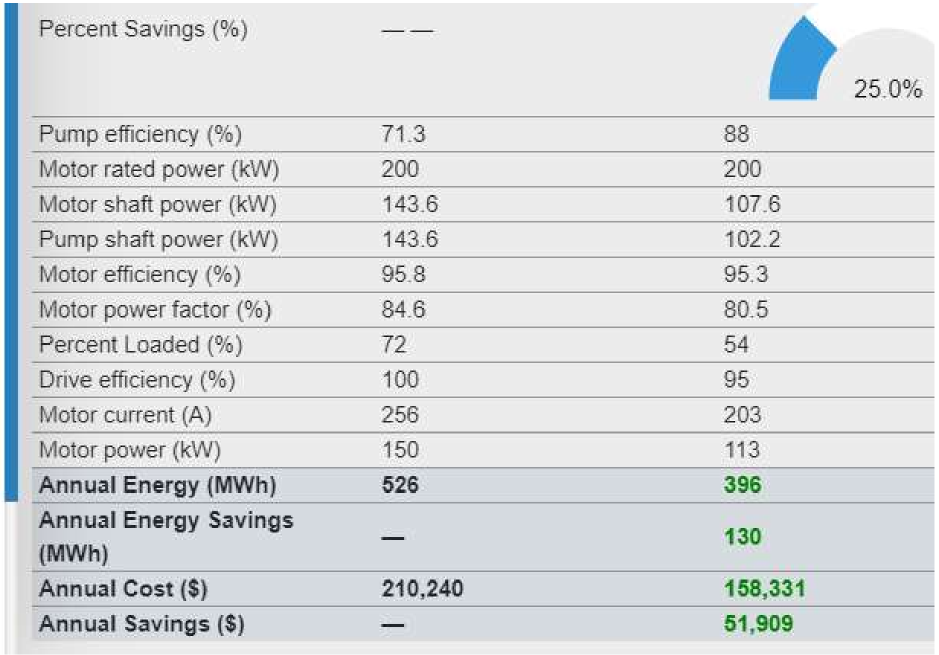
9. If the artificial head losses across the control valve could be eliminated, and a VSD is used to regulate flow, what would the energy savings be with an optimised pump?



Modification Name: Scenario 1

☑ Install VFD

Baseline	Modifications
Flow Rate 200 L/s	Flow Rate 200 L/s
Head 52 m	Head (Calculate Head) 46 m
Motor Drive Direct Drive	Drive Efficiency 95 %
Pump Type End Suction ANSI/API	Pump Type End Suction ANSI/API
	Pump Efficiency 88.03 % (Known Efficiency)



Percent Savings (%) — —

Pump efficiency (%)	71.3	88
Motor rated power (kW)	200	200
Motor shaft power (kW)	143.6	107.6
Pump shaft power (kW)	143.6	102.2
Motor efficiency (%)	95.8	95.3
Motor power factor (%)	84.6	80.5
Percent Loaded (%)	72	54
Drive efficiency (%)	100	95
Motor current (A)	256	203
Motor power (kW)	150	113
Annual Energy (MWh)	**526**	**396**
Annual Energy Savings (MWh)	**—**	**130**
Annual Cost ($)	**210,240**	**158,331**
Annual Savings ($)	**—**	**51,909**

12. Specific Energy

Pump Systems Optimisation (PSO) Expert Level Training
(Egypt Edition – Dec 2021)

Albert Williams
Siraj Williams

Specific Energy (E_s)

- The amount of energy needed to pump one unit volume through the system
- The specific energy varies with flow-rate
- A good way of comparing pump system performance

    ie. how much bang for your buck

- $$E_s = \frac{P_{in} \times Time}{V} = \frac{P_{in}}{Q}$$

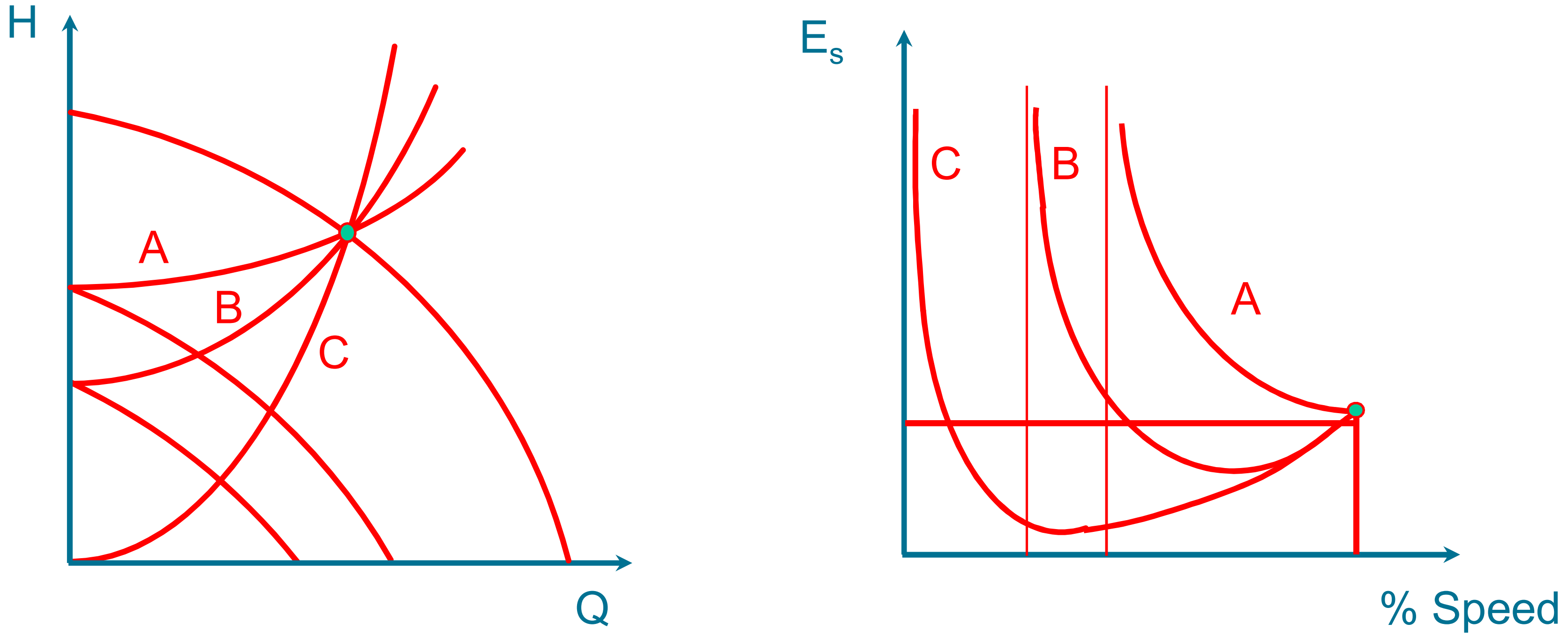
- $$\frac{\text{Energy Consumed}}{\text{Pumped Volume}} = \text{Specific Energy}$$

Variations in the understanding of the concept of Specific Energy

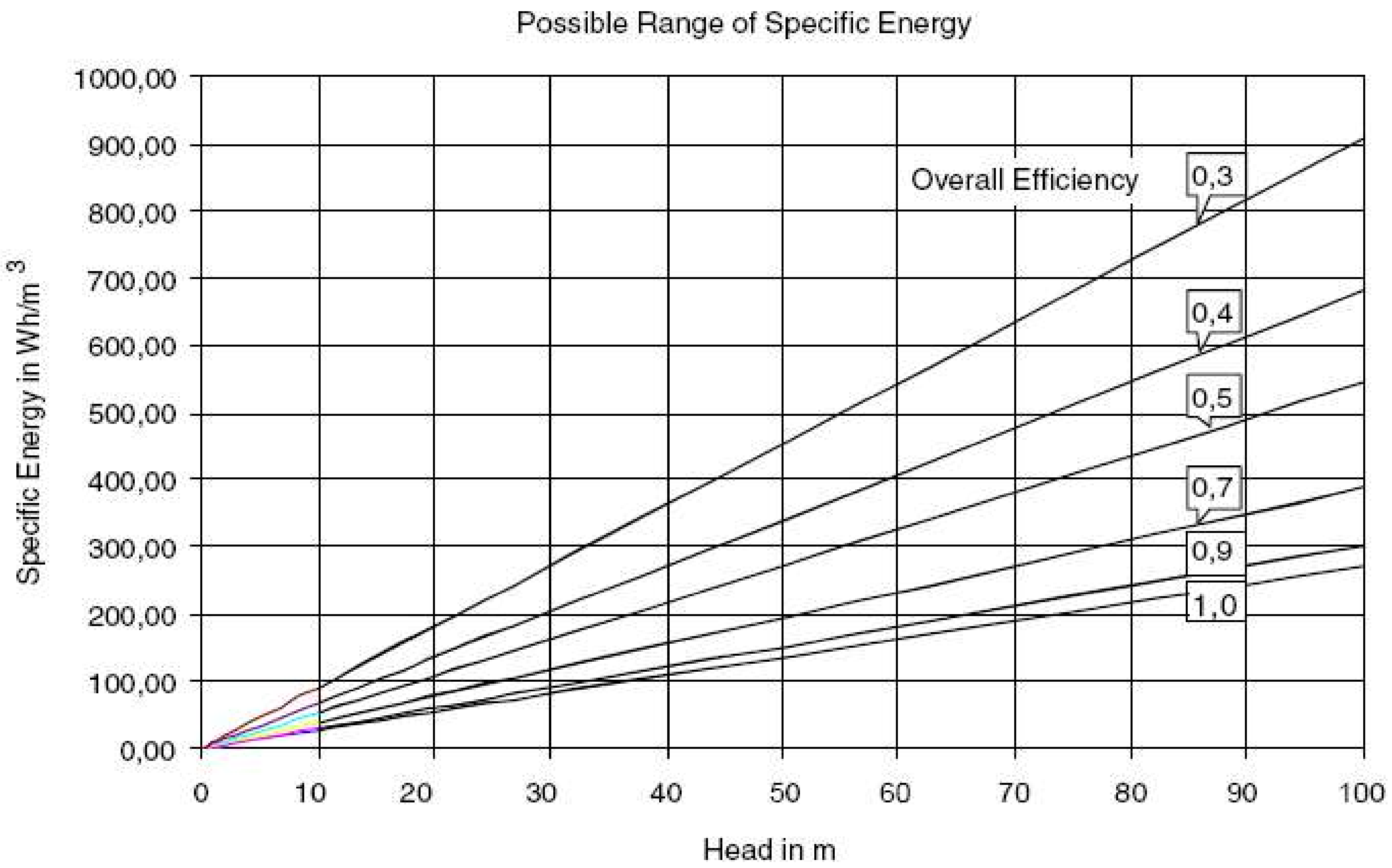
Examples

Specific Energy E_s

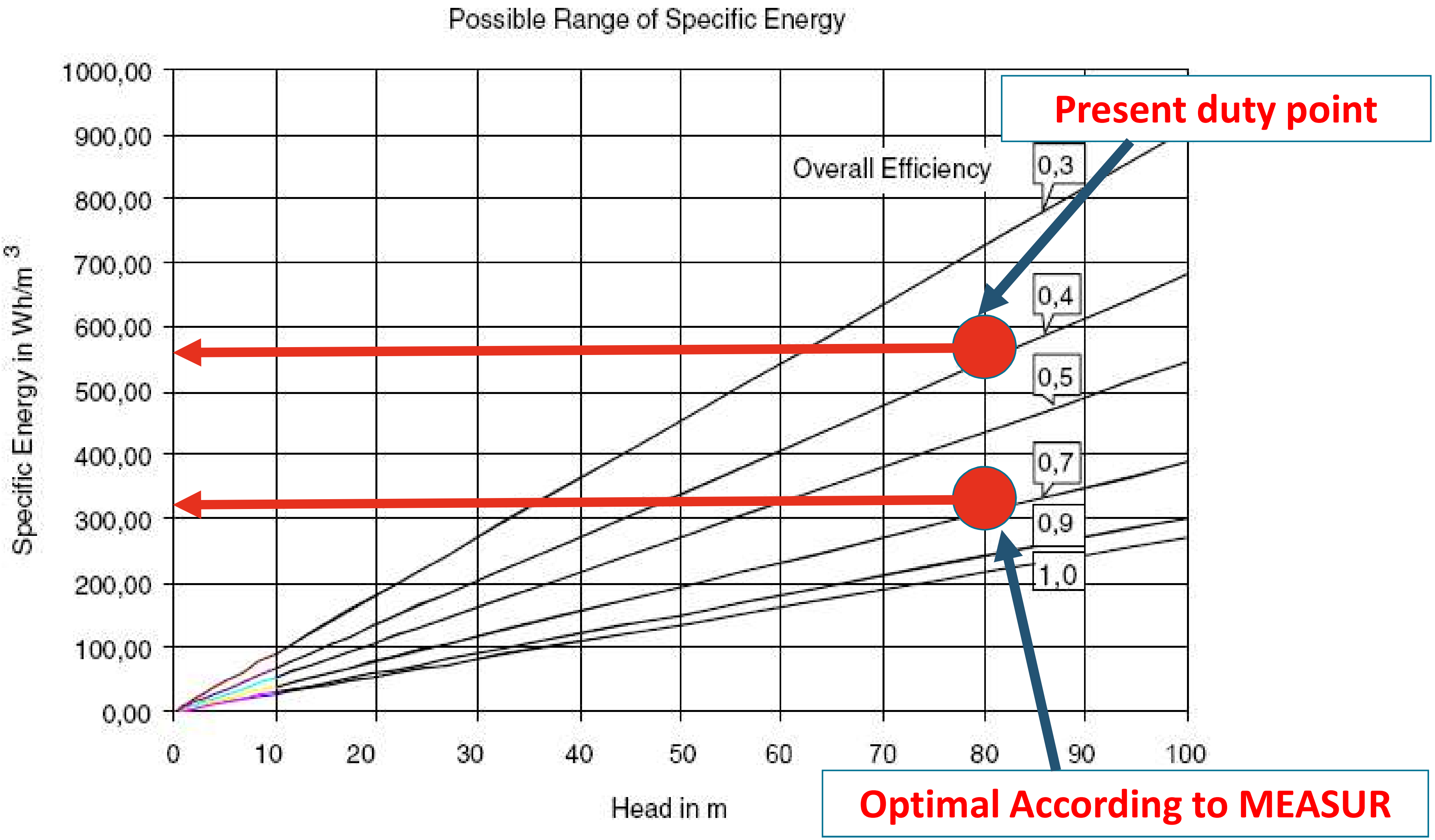
Specific Energy for 3 systems with VSD pumps and different static heads



Specific Energy is a function of Head



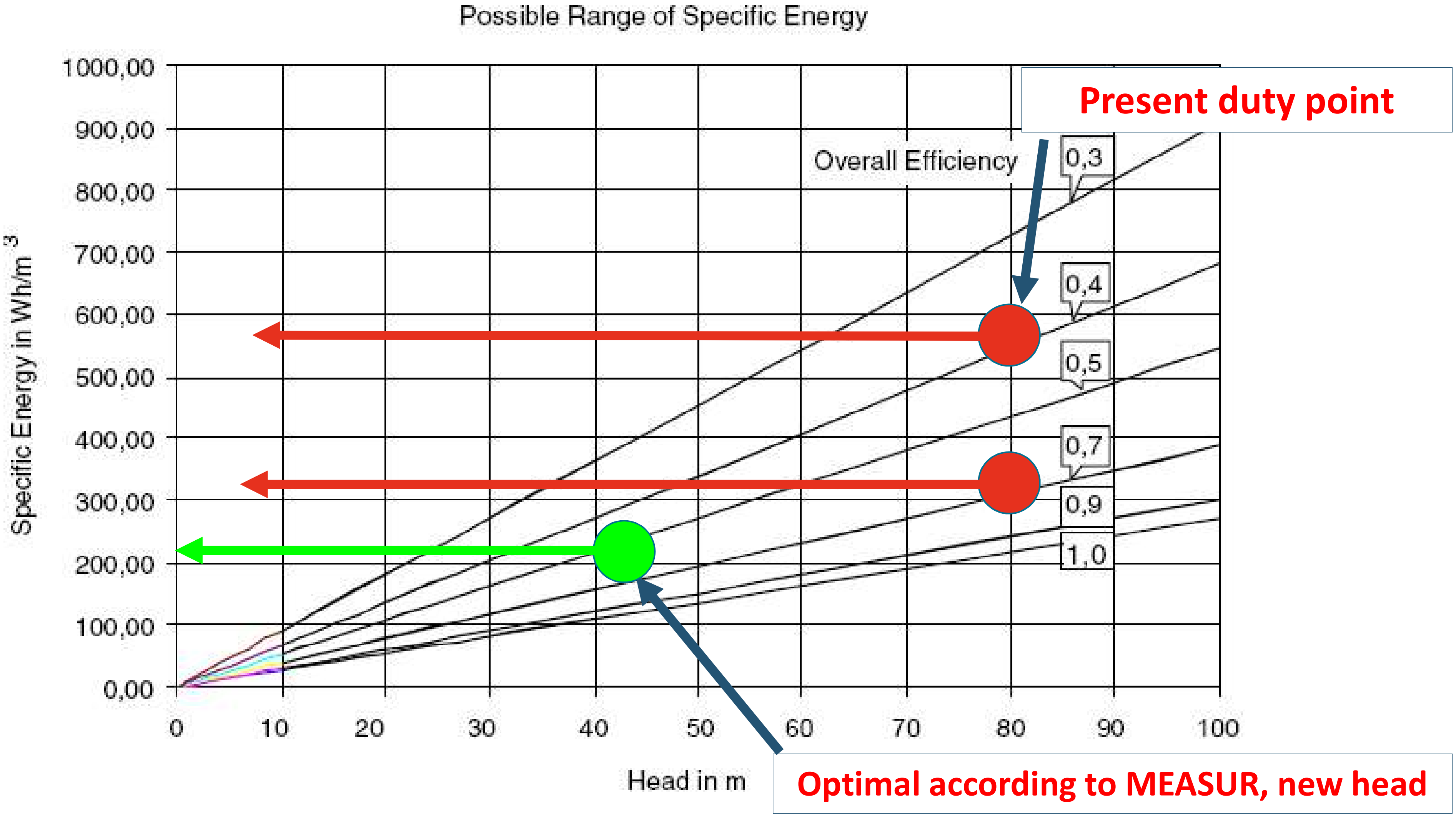
Specific Energy is a function of Head



Present duty point

Optimal According to MEASUR

Specific Energy is a function of Head



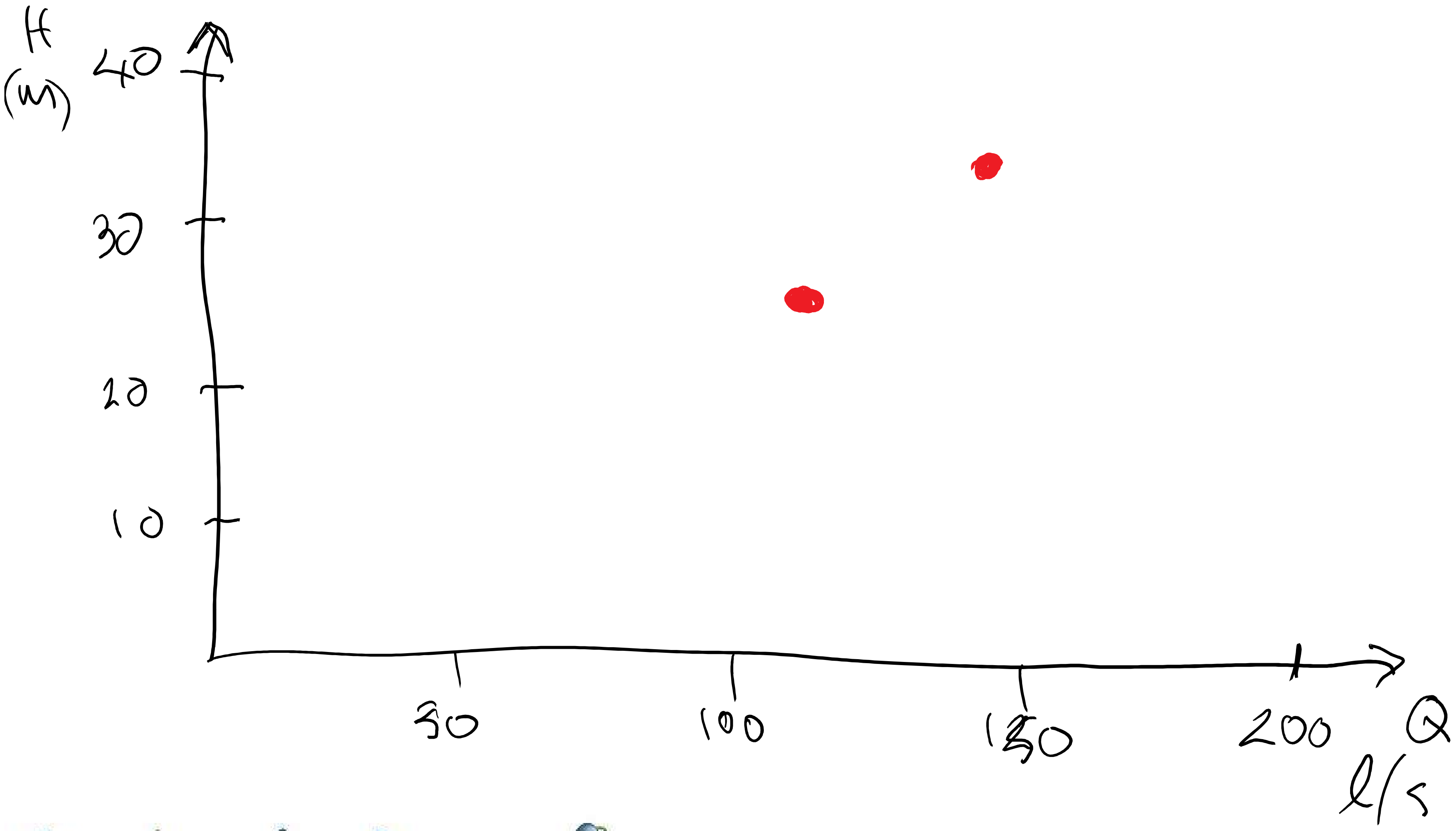
Present duty point

Optimal according to MEASUR, new head

Example – Specific Energy

- A centrifugal pump operates close to its best efficiency point (BEP =84%) while providing a flow rate of 110 l/s at a total head of 25 m.
- When an identical parallel pump is switched on, the composite system operating point shifts to 150 l/s at 35 m of head. Each pump now operates at 80% efficiency.
- What is the contribution of each pump in flow?
- What is the power consumption when one pump is operated?
- What is the power consumption when 2 pumps are operated?
- What is the specific energy consumption for each of the 2 conditions?

Given operating conditions



Specific energy

13. Pre-Screening

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Siraj Williams

What is Pump System Optimization?

- PSO is a systematic approach to evaluate high energy use pumps to identify EEM's
- After prescreening pump systems, potential savings of the selected pumps are determined by measuring pressure, flow and power in the field.
- This data is combined with pump system operational data to determine an energy use baseline and the true system requirements.
- MEASUR can be used to provide a preliminary savings analysis. If there is a good opportunity, a more advanced analysis can be performed to determine the most cost effective improvement for PSO.

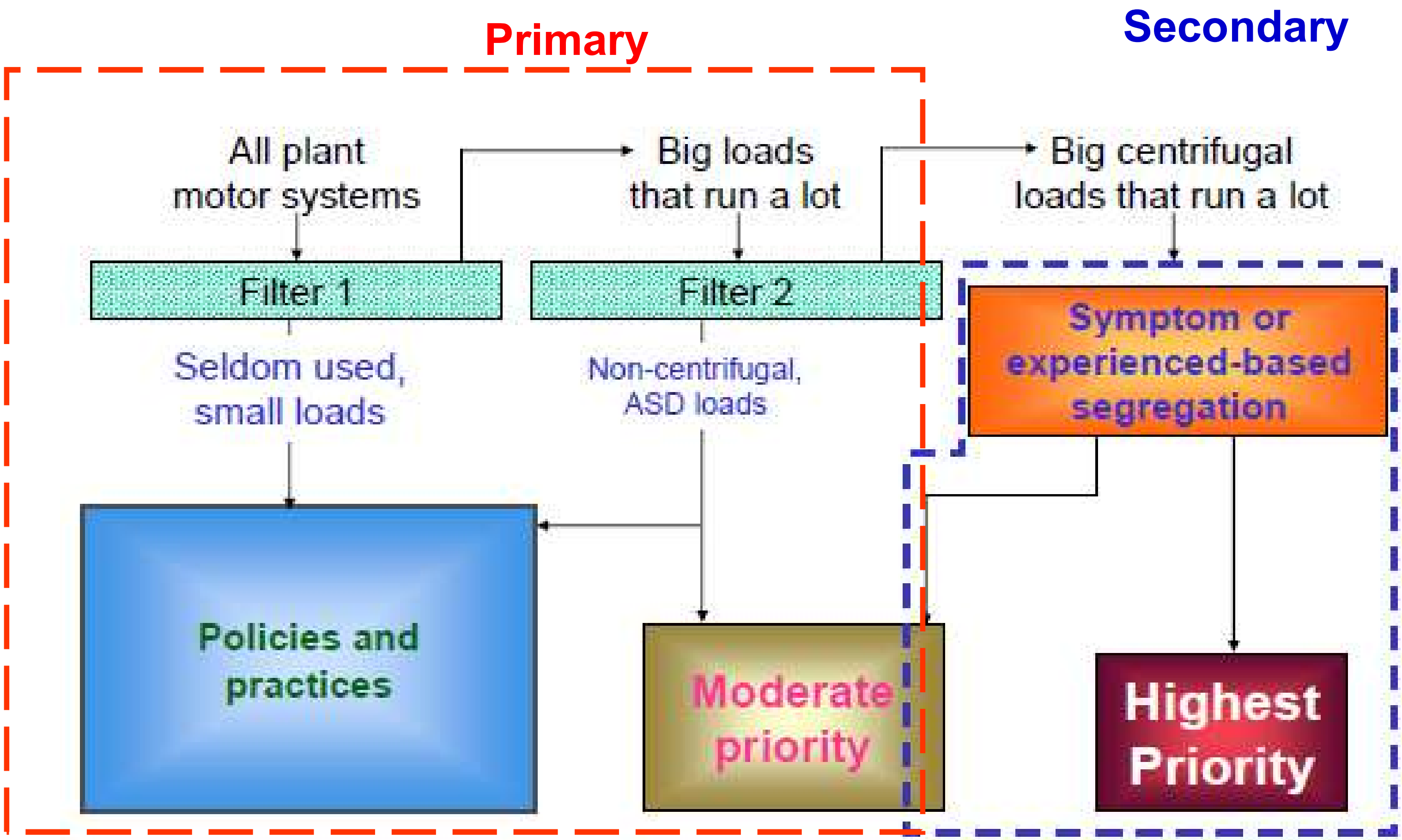
Pre-screening

The DOE Best Practices Program encourages a three tiered pre-screening and assessment approach that includes:

- Initial pre-screening based on size, run time and pump type.
- Secondary pre-screening to narrow the focus to systems where significant energy saving opportunities are more likely.
- Evaluating the opportunities and quantifying the potential savings.

Slide Courtesy of Oak Ridge National Laboratory

Primary & Secondary Prescreening



Primary

Secondary

All plant motor systems

Filter 1

Seldom used, small loads

Policies and practices

Big loads that run a lot

Filter 2

Non-centrifugal, ASD loads

Moderate priority

Big centrifugal loads that run a lot

Symptom or experienced-based segregation

Highest Priority

Slide Courtesy of Oak Ridge National Laboratory

Identifying Potential Savings Opportunities

4 common causes of less than optimal pump system performance:

- Installed components are inefficient at the typical operating condition
- The efficiency of the pump system components have degraded
- More flow or more head is being provided than the system requires
- The pump is being operated when it is not required by the system

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Using Field Observations to ID ESO's

- Valves throttled to control flow
- Bypass (re-circulation line) normally open
- Multiple parallel pump system with same number of pumps always operating
- Constant pump operation for a batch process
- Cavitation noise (at the pump or elsewhere in the system)
- High system maintenance
- Systems that have undergone a change in function

Slide Courtesy of Oak Ridge National Laboratory

Sample Pre-screening Form

Equipment Information								Control schemes (check all that apply)						Operating Parameters (provide if readily available, otherwise indicate with check if it is acquirable)									Other symptoms		Additional Information (is acquirable?)			
System name/description	Pump Type [MC, PD, Vacuum, Centrifugal]	Pump ID/process area	Installed motor hp	Service (e.g. utility, process, etc.)	Time in service (years)	Indicate shared duty pump systems/ in service spares	Voltage	Adjustable speed drive	Throttled (% open if available)	Bypass/Re-circ	On/off	More than one pump/split duty	Not controlled (pumps just run)	Operating hours or % of time equipment operates	Power or Current	Flow requirements have changed or are expected to change	Design flow rate	Operational flow rate	Design head	Operational head	Upstream pressure	Downstream pressure (after control valve, or bypass line, etc)	Cavitation at pump or in system?	System maintenance level (Hi/Med/Lo)	Typical flow rates and variation thereof	Duration diagrams	Maintenance Costs	PID / DCS screen-shots
Bldg 83 cold well	Centrifugal	13 A, B	200						x					80		No		x						Med				
Raw water	Centrifugal	42 A, B, C	125						X	x				95		No								Med				

Example of a list of pumps to be populated by a company prior to the assessment (shaded fields are mandatory).

14. Reliability & Maintenance

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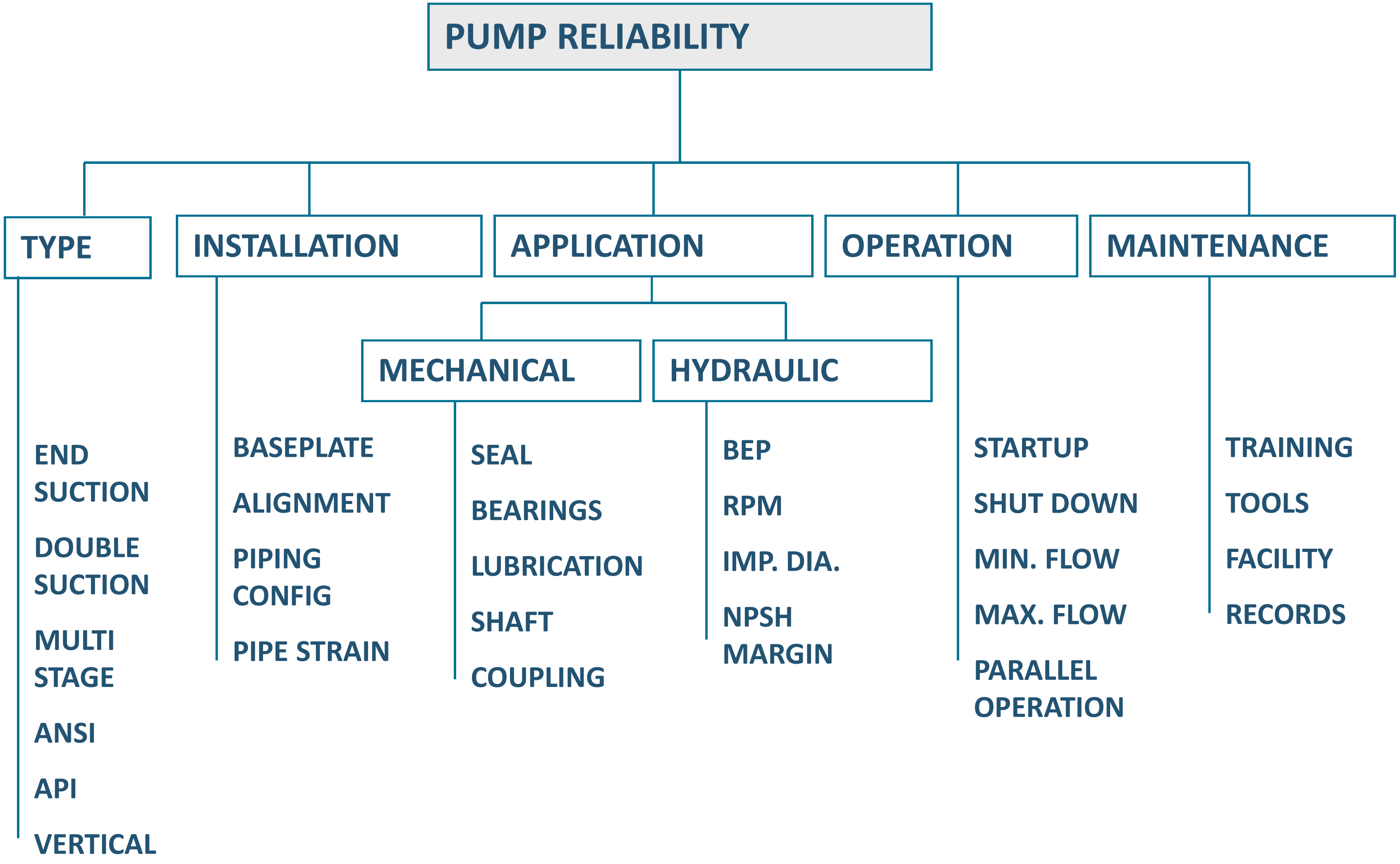
Factors that influence pump reliability



- **PUMP RELIABILITY**
  - **TYPE**
    - END SUCTION
    - DOUBLE SUCTION
    - MULTI STAGE
    - ANSI
    - API
    - VERTICAL
  - **INSTALLATION**
    - BASEPLATE
    - ALIGNMENT
    - PIPING CONFIG
    - PIPE STRAIN
  - **APPLICATION**
    - **MECHANICAL**
      - SEAL
      - BEARINGS
      - LUBRICATION
      - SHAFT
      - COUPLING
    - **HYDRAULIC**
      - BEP
      - RPM
      - IMP. DIA.
      - NPSH MARGIN
  - **OPERATION**
    - STARTUP
    - SHUT DOWN
    - MIN. FLOW
    - MAX. FLOW
    - PARALLEL OPERATION
  - **MAINTENANCE**
    - TRAINING
    - TOOLS
    - FACILITY
    - RECORDS

Selecting a pump type

Proper pump selection requires a through knowledge of the service the pump will be used in:

- Fluid characteristics
- Temperatures
- Flows and pressure
- System limitations

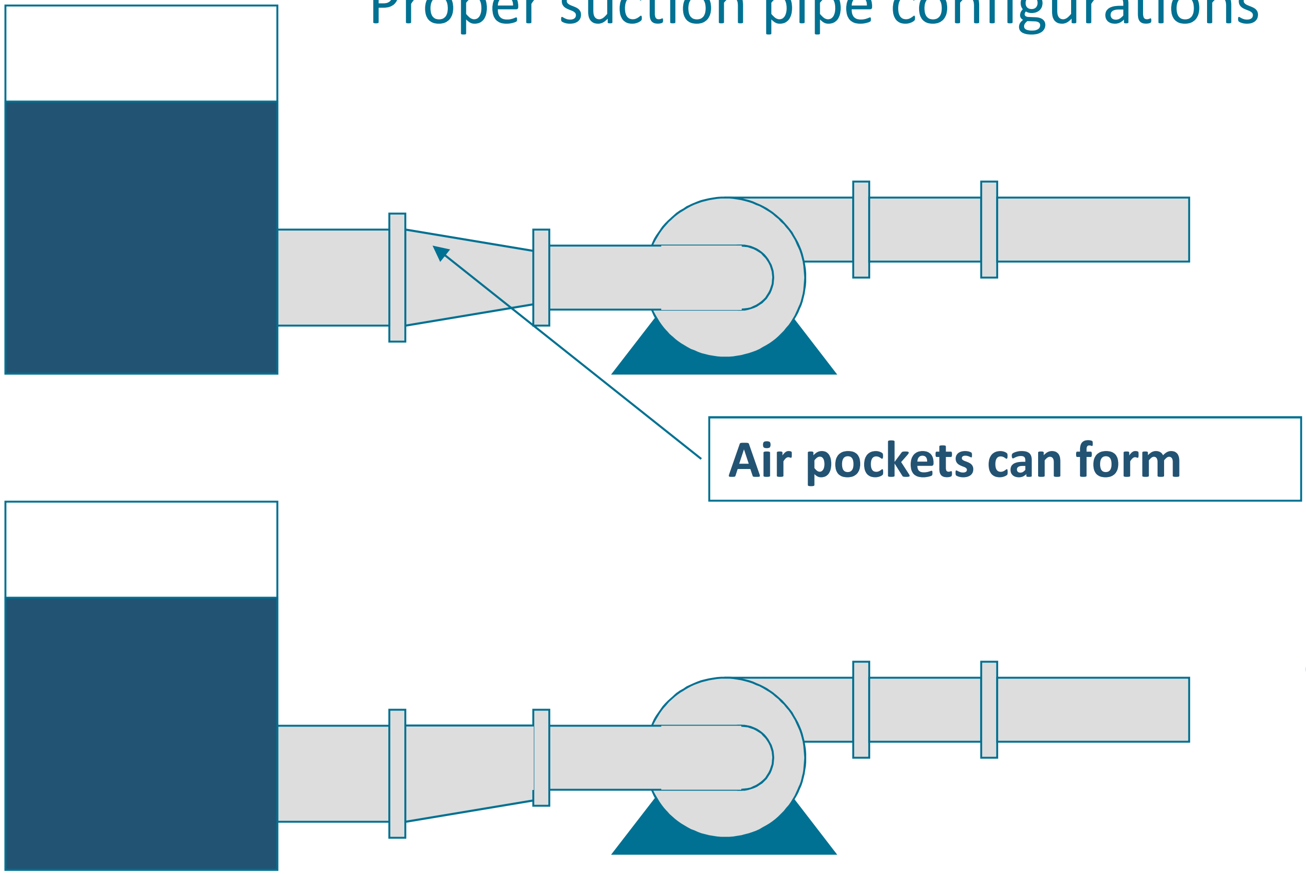
Some standards, commonly used, provide guidance on pump types for various applications

Importance of Proper Installation

- Pump installation is critical to long term pump system reliability and efficiency. Standards for each of the areas below should be reviewed to insure a proper installation;
    - Motor/pump coupling alignment
    - Pump hold down bolts, mounting, grouting, bedplate construction
    - Proper piping size, component installation (ANSI/HI 9.6.2)
- One company realized a 10 fold increase in reliability by instituting new installation specifications relating to base-plate, piping, and grouting.

Piping Configurations

Proper suction pipe configurations



Air pockets can form

Incorrect installation

Correct installation

Uneven flow into the pump

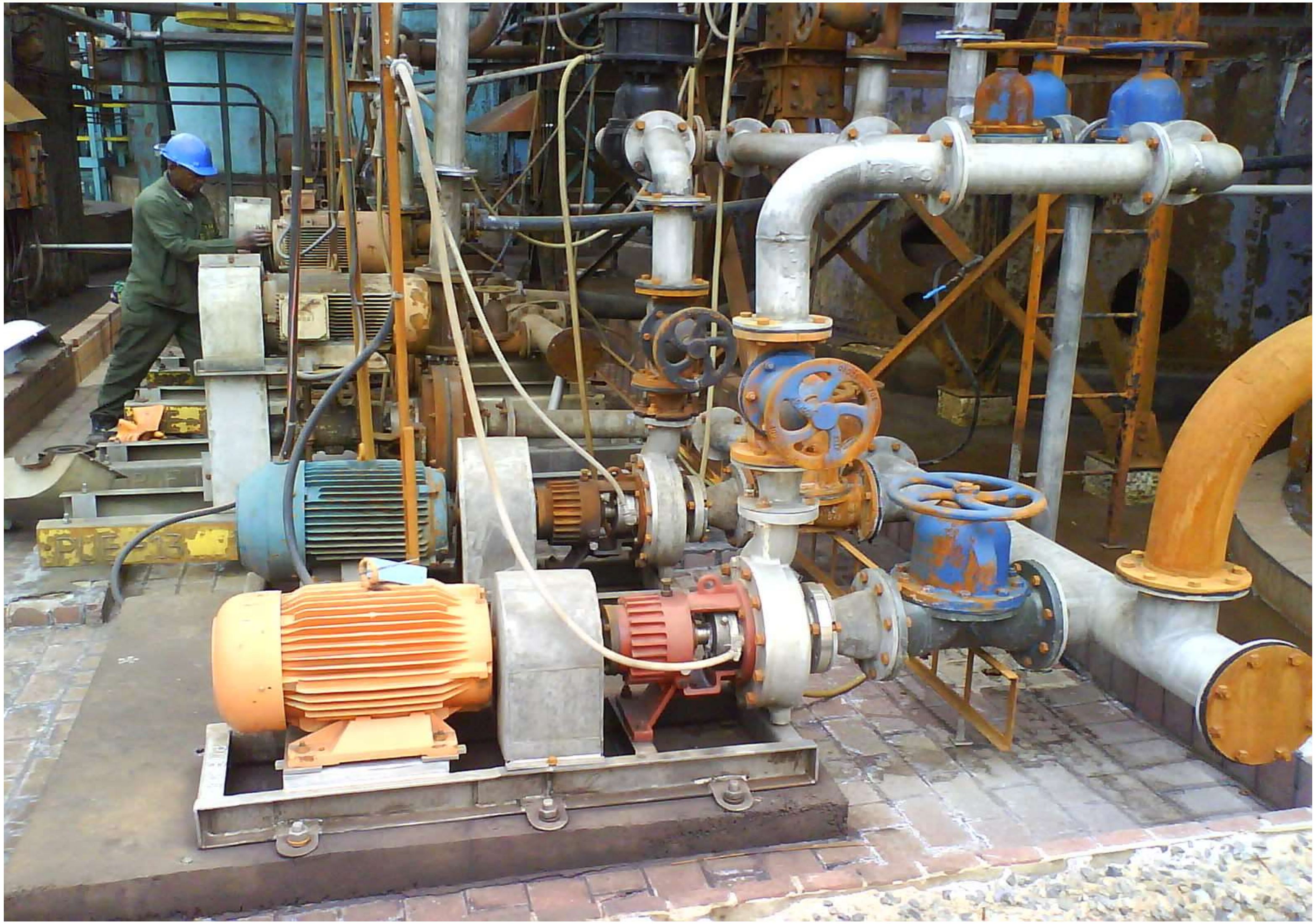
A pipe fitters dream?

Not the greatest inflow conditions

Throttled inlet valve

Throttled discharge valve

Piping Configurations



Piping Configurations

Piping Configurations

Piping Configurations

Piping Configurations

Cavitation - Theory

- An area of **low pressure** is always present at the impeller eye
- When this pressure is low enough, the liquid flashes
- Vapour bubbles form
- These bubbles collapses when moving further into the impeller in high pressure areas
- This is called cavitation, which is harmful to pump operation, performance and causes structural damage

Cavitation Damage

Impeller cavitation regions

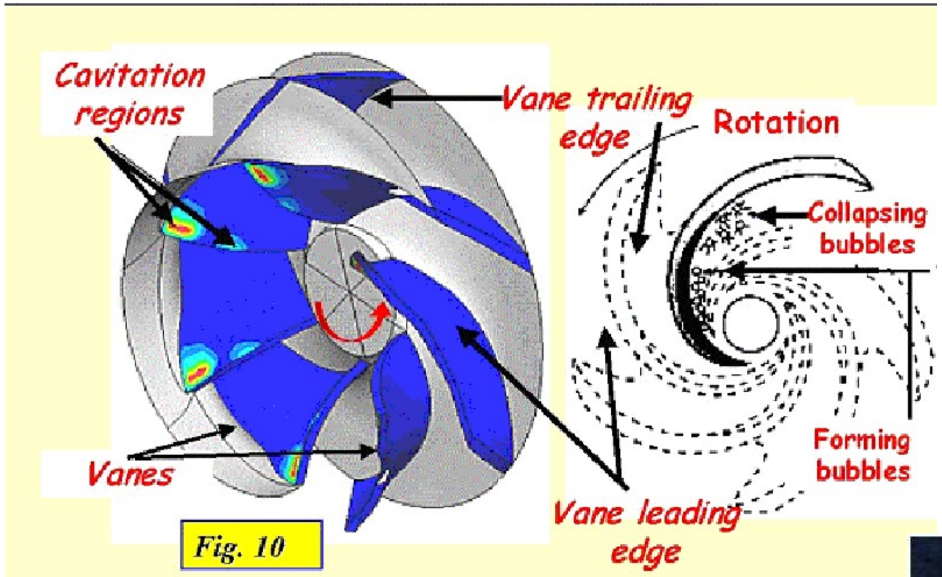


Fig. 10

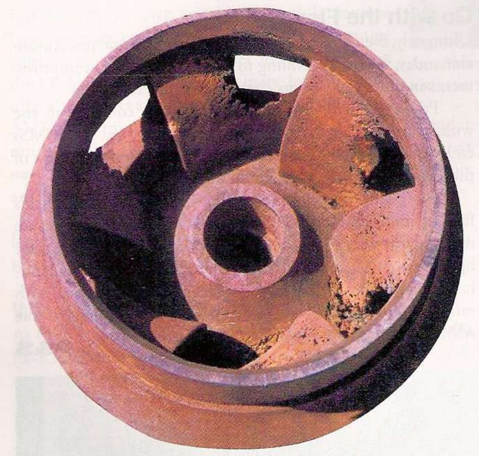


Photo by: Diagnostic Solutions, LLC

NPSH$_a$ should be greater than NPSH$_r$

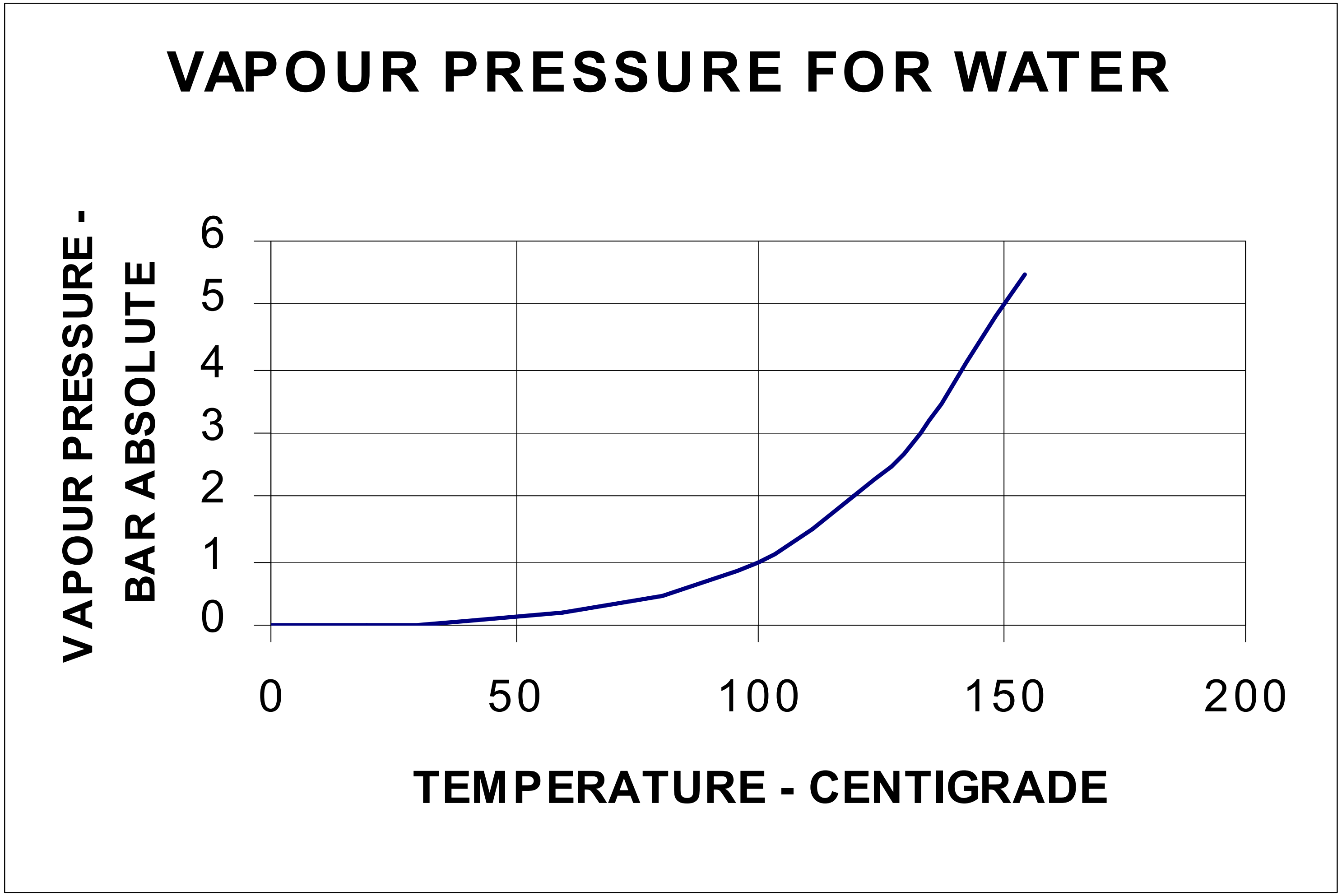
- Centrifugal pumps require enough pressure on the suction side of the pump to prevent flashing in the impeller eye.
- The amount of pressure required for a specific pump is determined by pump manufacturer during the design of the impeller and is confirmed during performance tests.
- This is called the Net Positive Suction Head Required. (NPSH$_r$)
- At that pressure cavitation is already taking place by 3%.
- The available suction pressure NPSH$_a$ has to be 3% higher than the NPSH$_r$ in order to avoid cavitation.

Determining NPSH Available ($NPSH_a$)

$$NPSH_a = H_a \pm H_z - H_f + H_v - H_{vap}$$

- H_a = Atmospheric pressure (absolute pressure, includes tank pressure, dependent on altitude)
- H_z = Vertical height between suction side water level and pump centreline
- H_f = friction loss through the suction pipe & fittings (always negative)
- H_v = Velocity head at pump suction (kinetic energy of the water, generally negligible)
- H_{vap} = Vapour pressure of water (pressure required to keep water in its liquid state)

Vapour Pressure



Cavitation Symptoms

- The pump sounds like it is pumping rocks!
- High vacuum reading on suction line
- Low discharge pressure
- High flow rate

Cavitation Remedies

To increase NPSH available in the system:

- Unblock suction line (remove debris in pipe, clean strainer, clean out suction tank)
- Increase suction line diameter
- Raise liquid level or lower the pump
- Move pump closer to tank
- Fully open suction line valve
- Use a booster pump
- Sub-cool the liquid

Cavitation Remedies

To reduce NPSH required by the pump:

- Move duty point left on curve
- Use oversize pump
- Run pump at slower speed
- Use a double suction impeller (two eyes)
- Use a larger impeller eye diameter (higher suction specific speed impeller). Lower inlet velocity due to increased area for the same flow.
- Use an inducer (special type of impeller)

Pump Operation

- Pump system reliability is compromised when pump flow rate increases or decreases away from the BEP due to higher (or lower) system pressures.

- Be careful when a VSD is used, since the forces inside the pump generally are reduced and seal face speed is lower, **but if the pump is operating in a *High Static Head* application, these forces could increase and lead to shaft failures.**

Pump Operation / Reliability

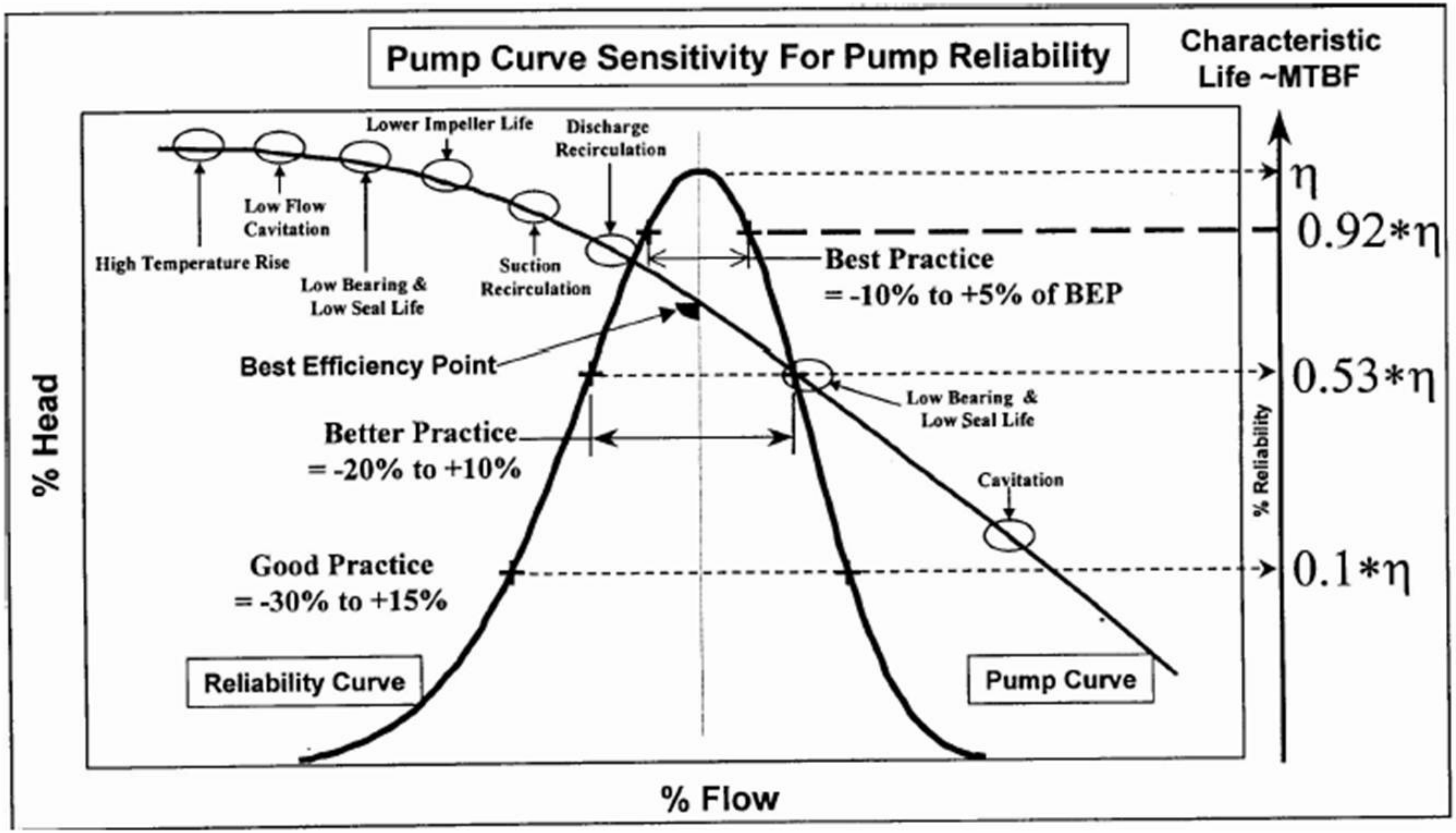
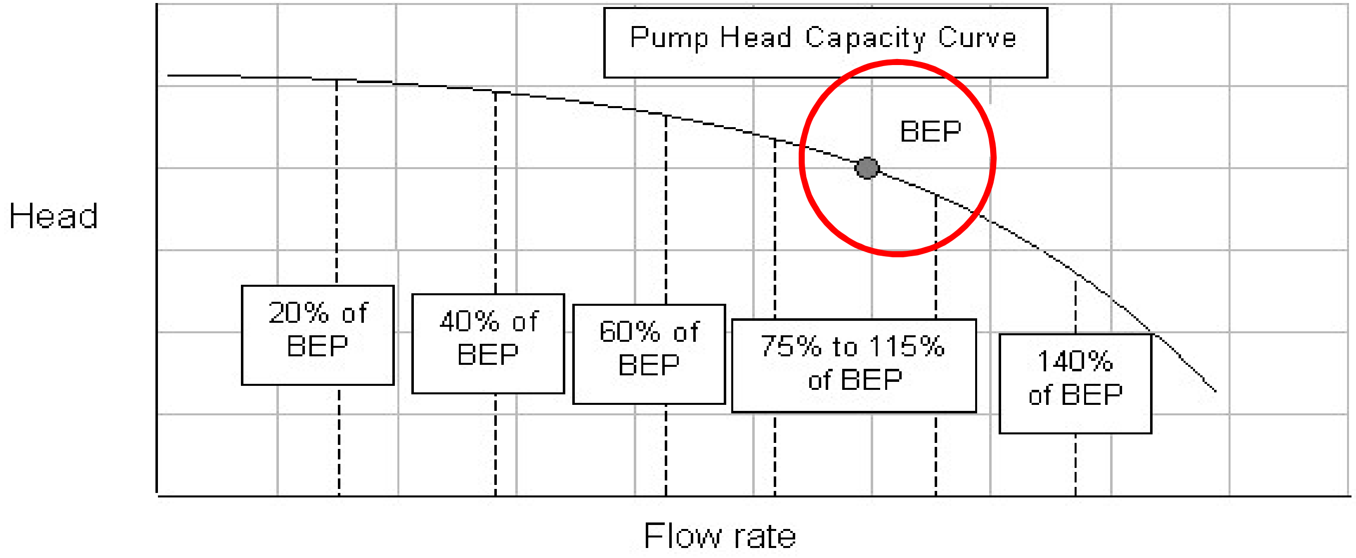


Figure Courtesy of P. Barringer.

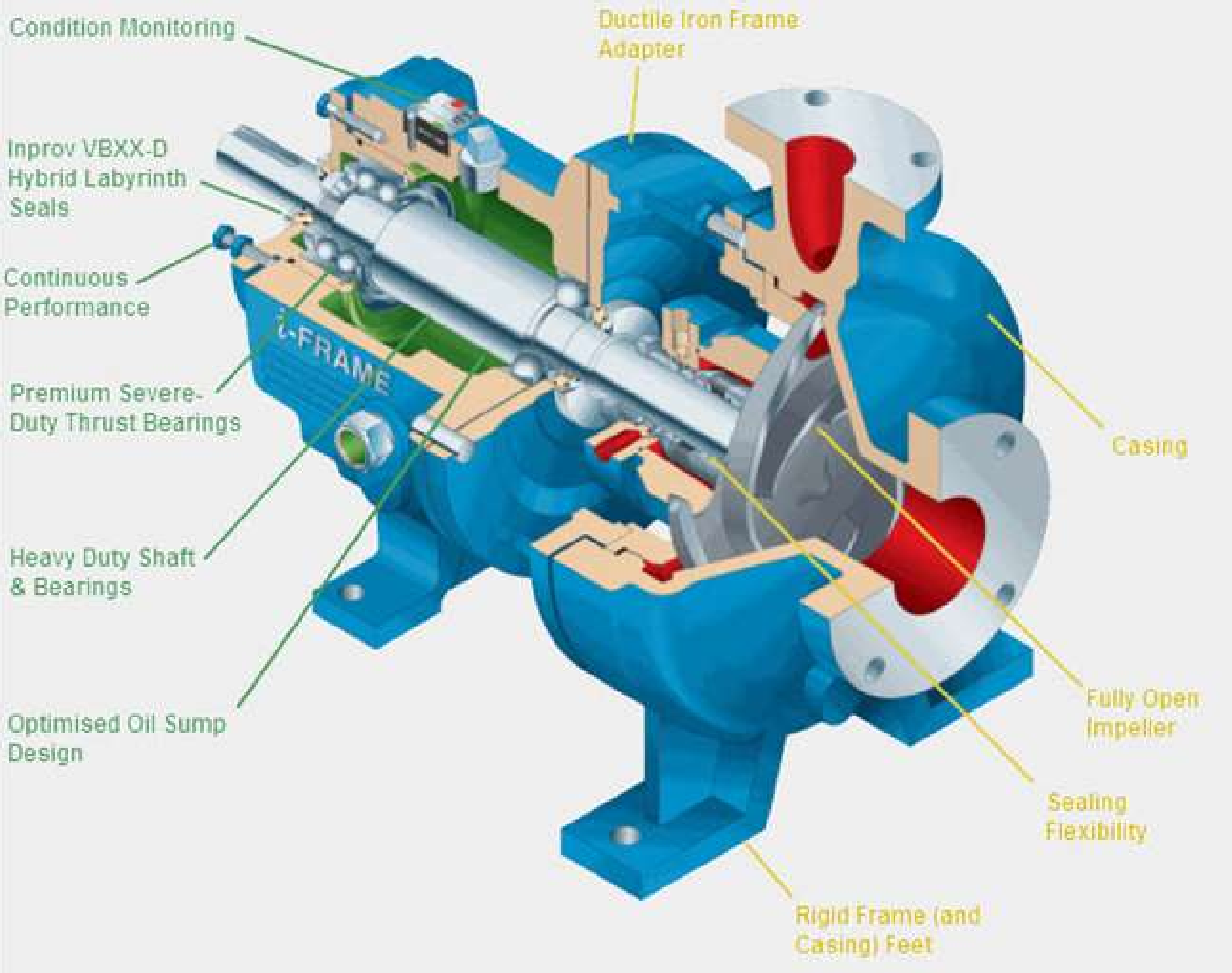
Maintenance Costs Relative to Distance from BEP



	%of BEP	20%	40%	60%	75-115%	140%	$/Failure	
Seals:	Life	2 Months	4 Months	1 Year	2 year	2 months	$1,000	Parts
	Failure/Year	6	3	1	0.5	6	$500	Labor
	Cost/Year	$9,000	$4,500	$1,500	$750	$9,000		
Bearings:	Life	1 Year	3 Year	4 Year	5 Year	1 Year	$500	Parts
	Failure/Year	1	0.33	0.25	0.2	1	$500	Labor
	Cost/Year	$1,000	$333	$250	$200	$1,000		
Casing/Impeller:	Life	2 Year	5 Year	7 Year	10 Year	2 Year	$2,000	Parts
	Failure/Year	0.5	0.2	0.014	0.1	0.5	$0	Labor
	Cost/Year	$1,000	$400	$285	$200	$1,000		
Total Cost/Year		$11,000	$5,230	$2,040	$1,150	$11,000		

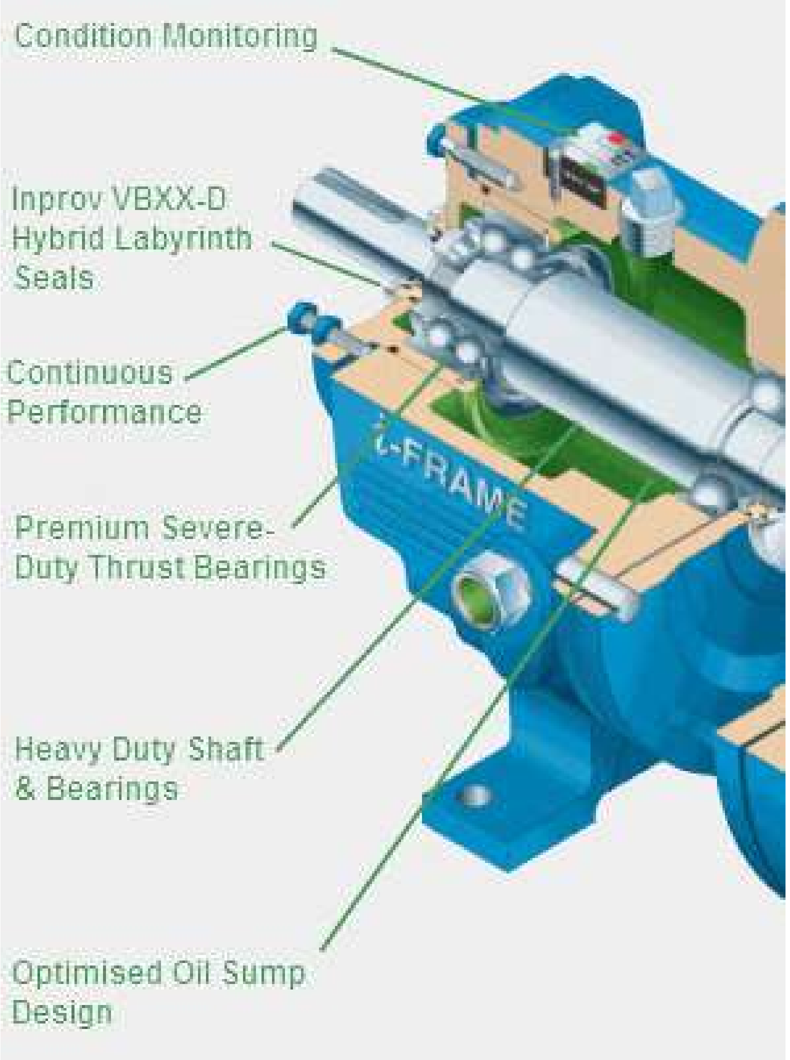
Table Courtesy of J. Hodgson.

Main Components of Centrifugal Pumps

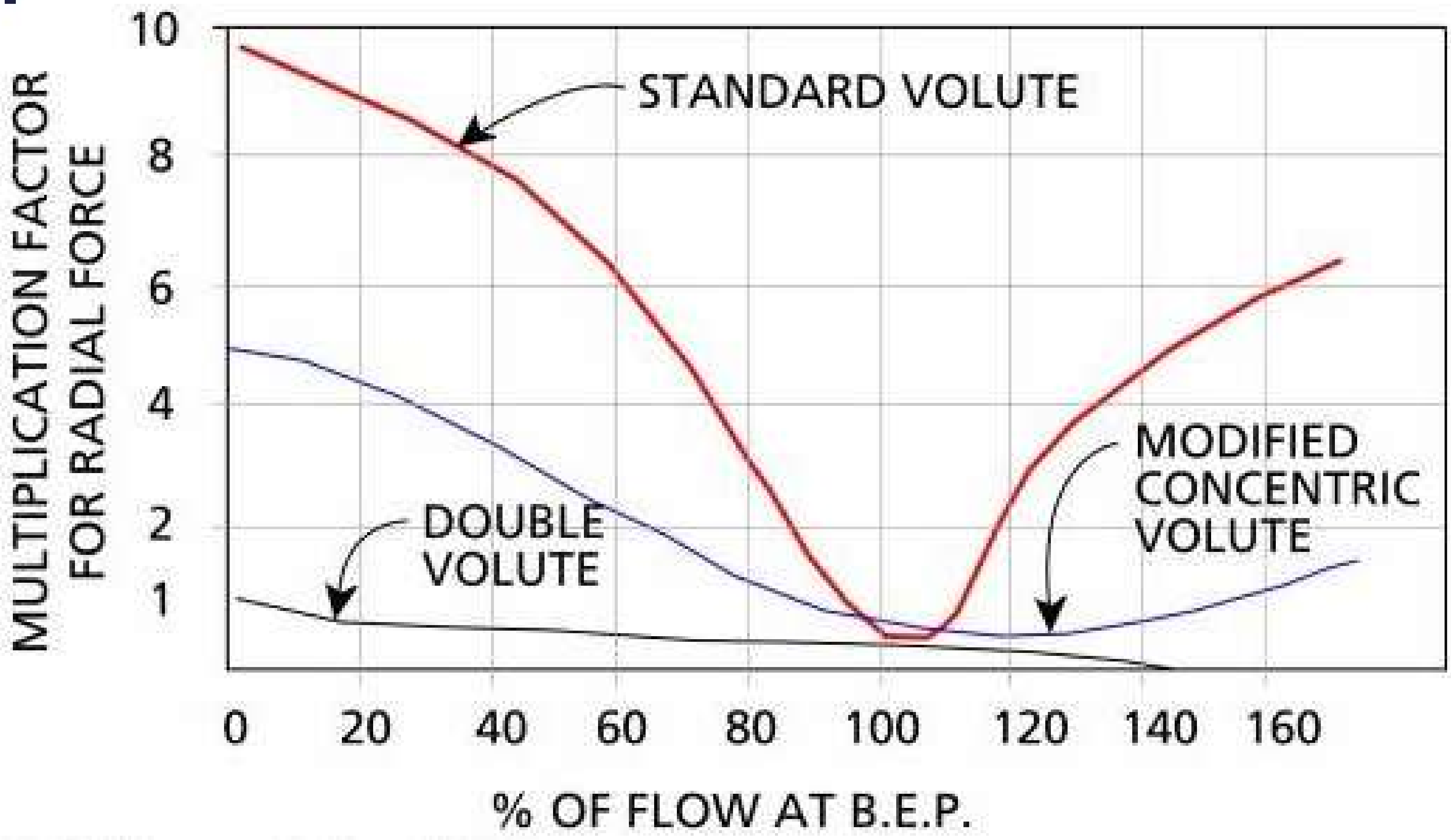


Bearing Housing

- Radial and thrust bearing
- Lubrication system



Bearing Life



Bearing Life

- End suction, overhung open impeller pump
- Optimum Duty
  - F_R = 1 800 N, F_A = 4 050 N
  - P = 6 156 N, L_{na} = 24 755 hrs (bearing life)
  - Approximately 5-10 years operation life
- Worn pump operating off BEP
  - Radial thrust 300% increase above normal
  - Axial thrust 50% increase
  - P = 16 767 N, L_{na} = 1 225 hrs (bearing life)
  - Bearing failure after approximately 1.8 months

Bearing Life

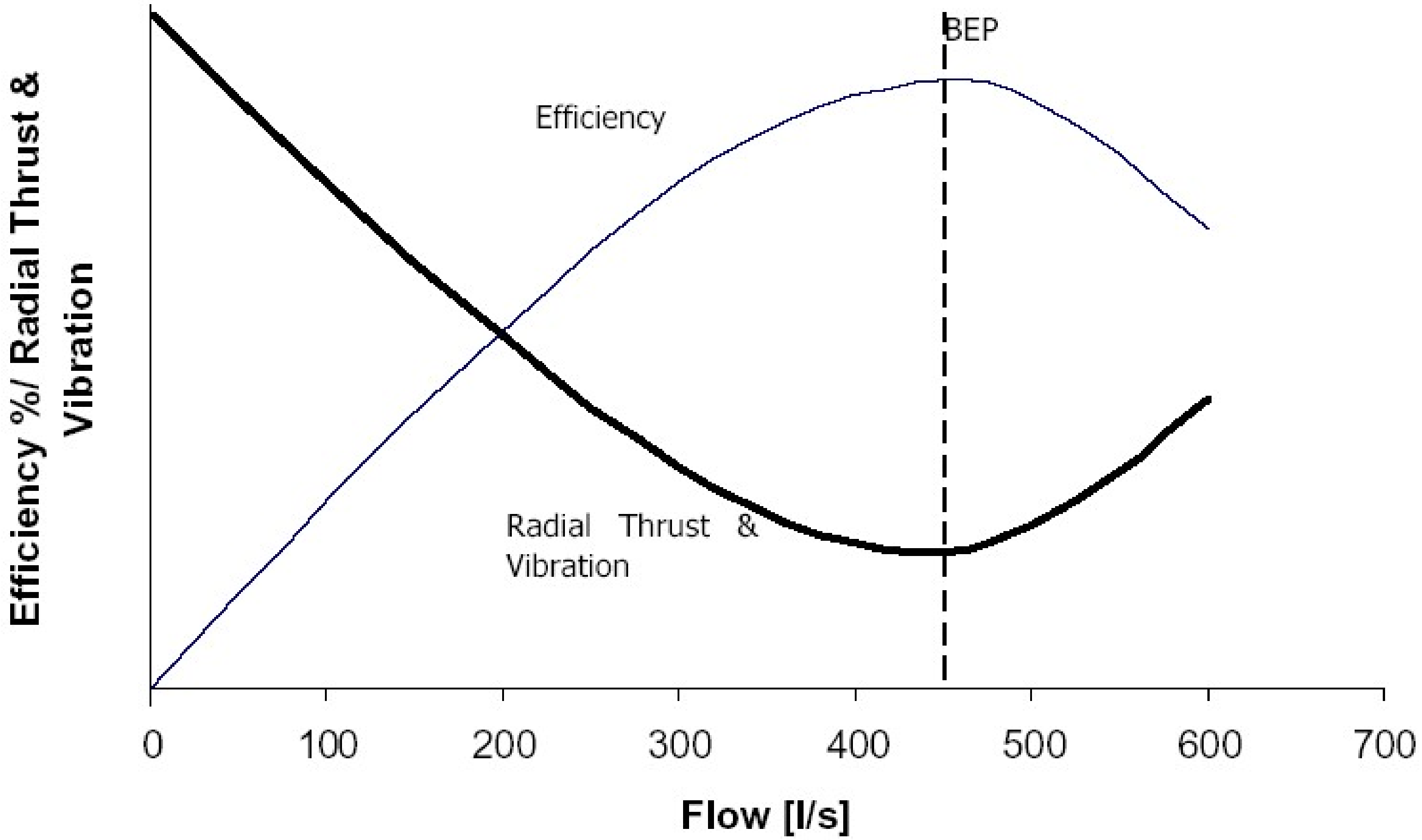
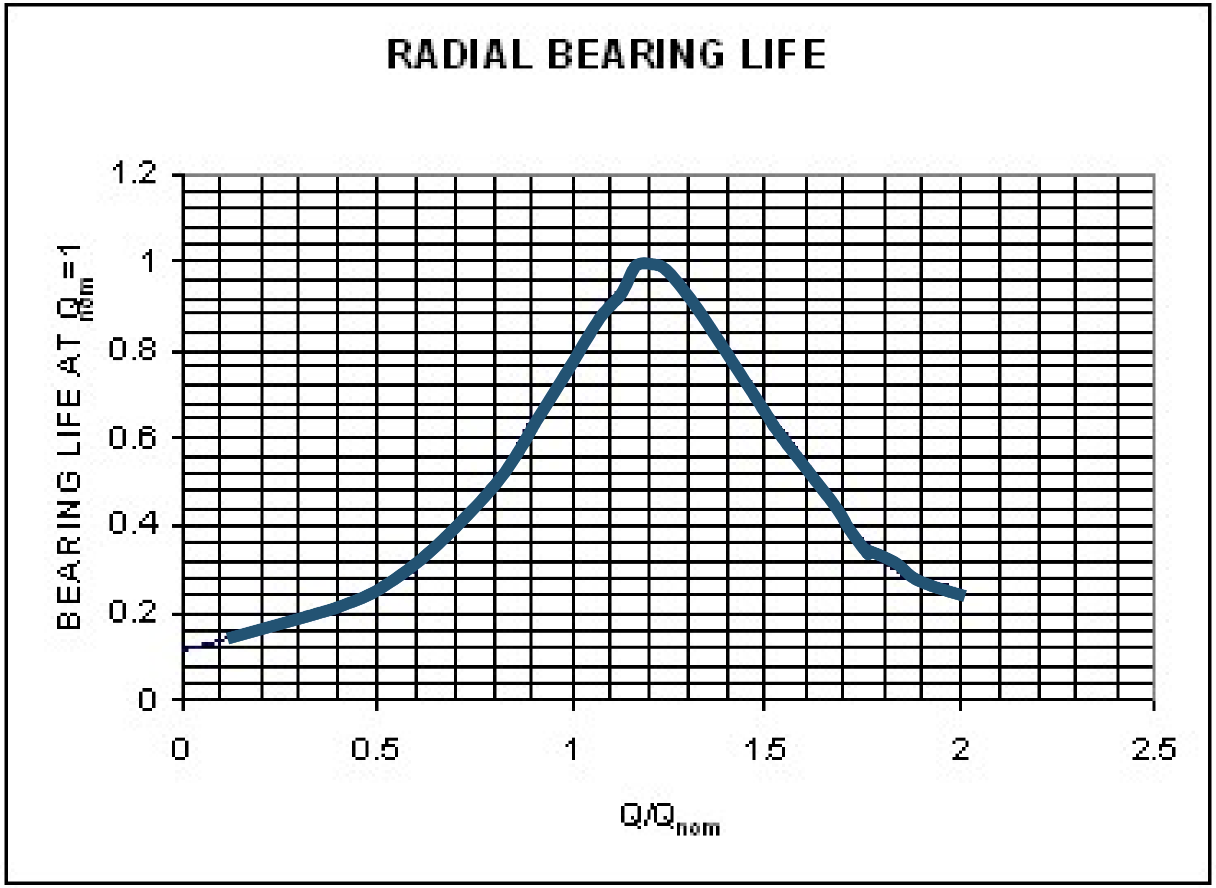


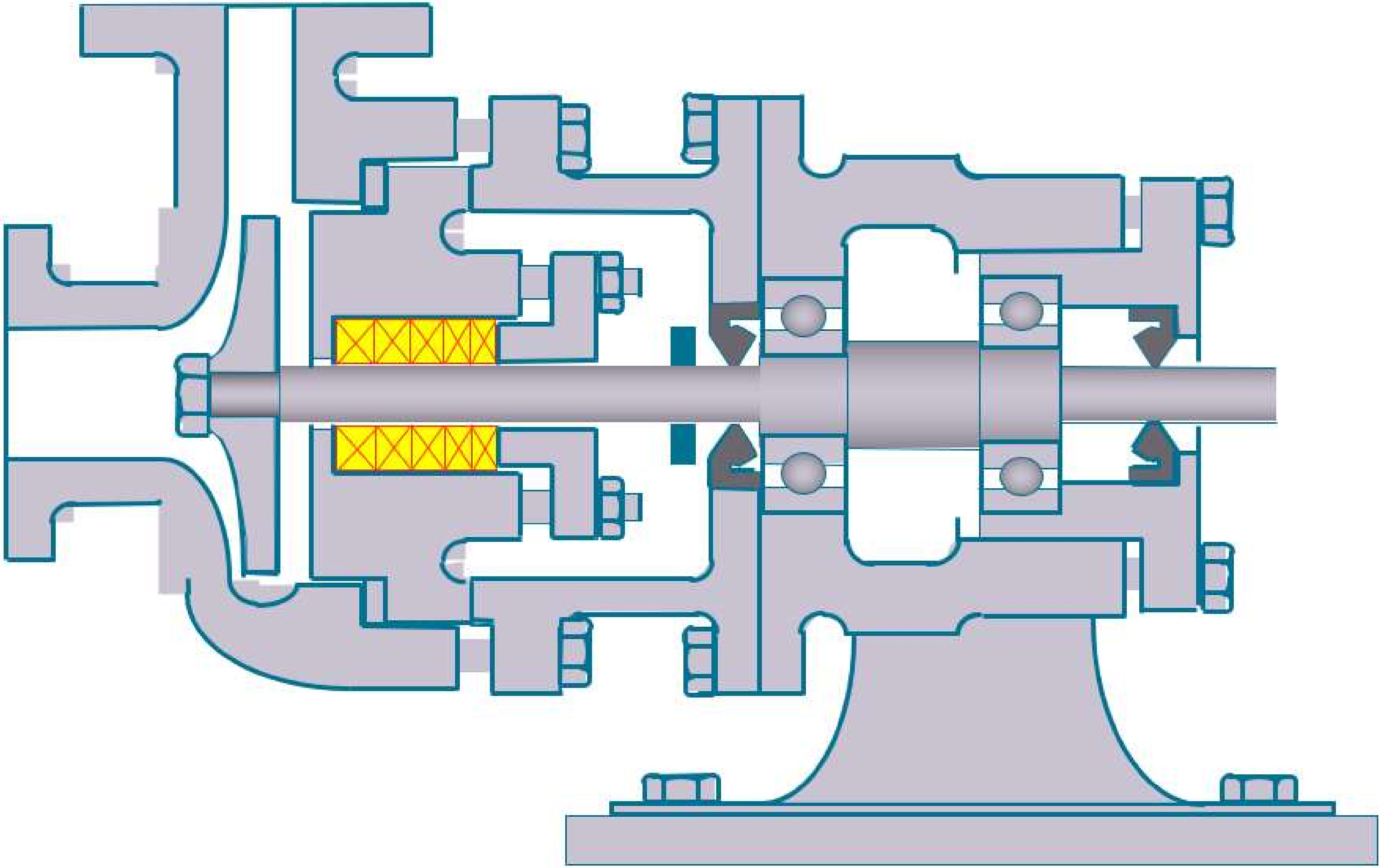
Fig 17: Radial Thrust and Vibration Increases as Pump Operation Moves Away From BEP

Stress on bearing

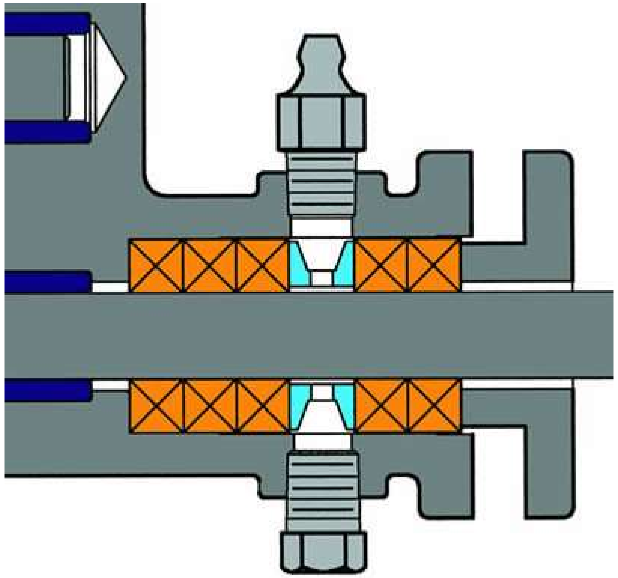


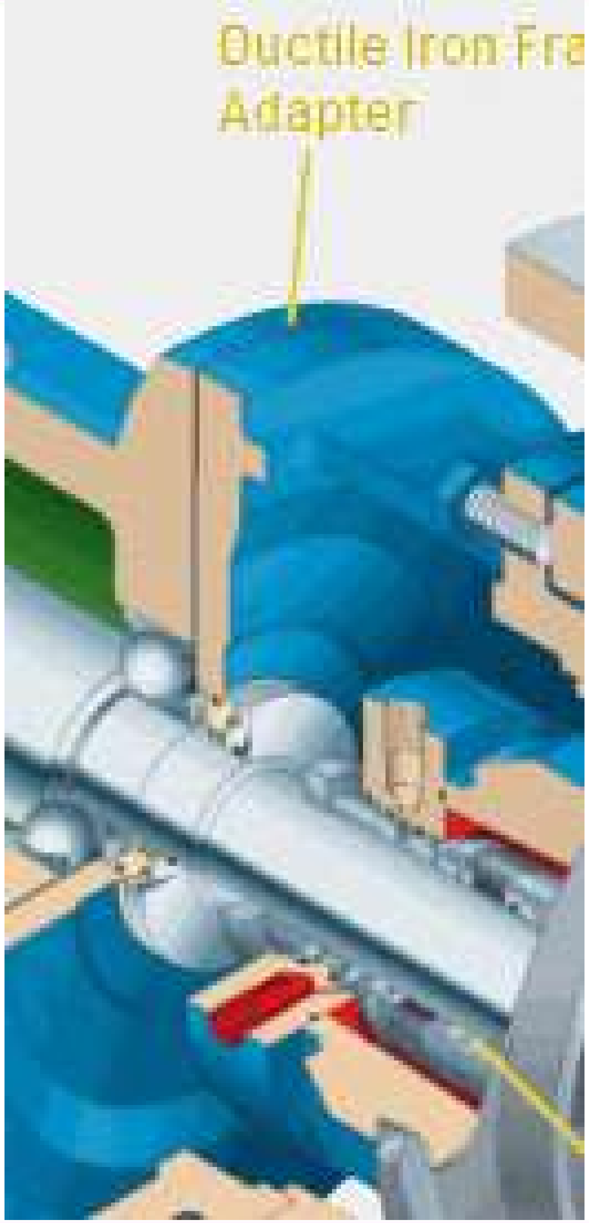
The further the pump is operated away from design flow (Q_{nom}) versus the actual flow (Q) the greater the stress on the bearings (for a full speed pump)

Seal Life

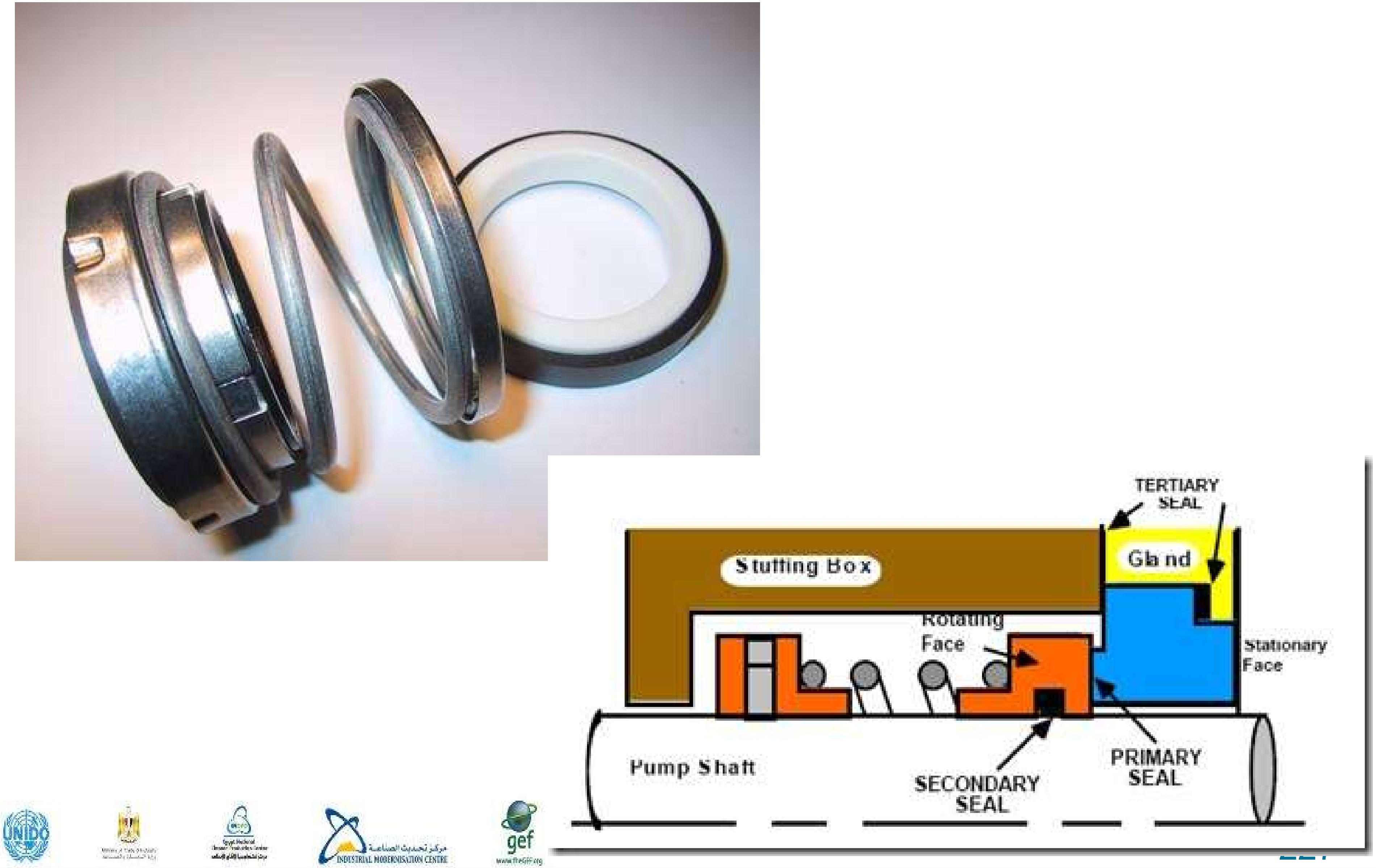


Gland Packing Seals

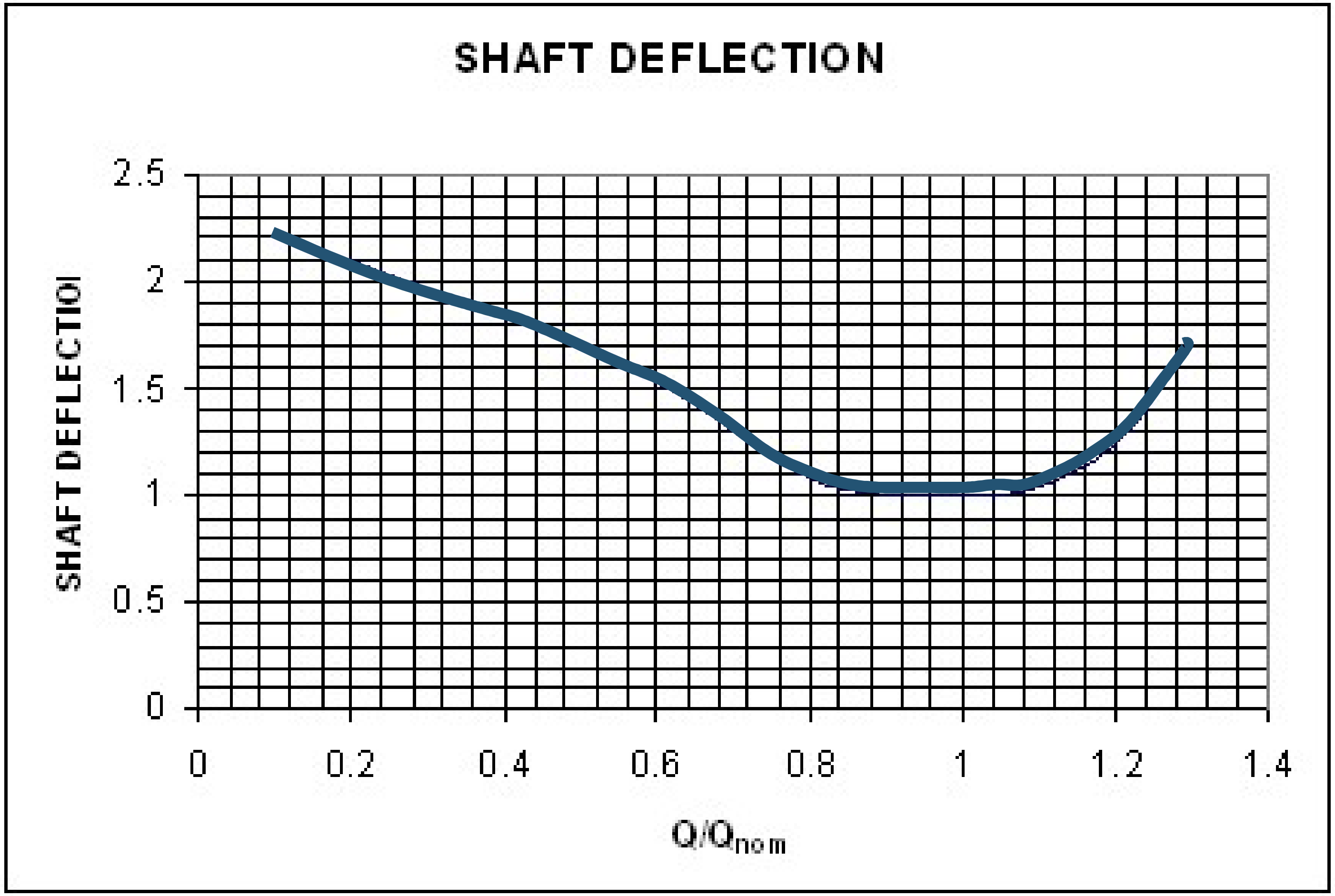




Mechanical Seals



Stress on shaft and seals

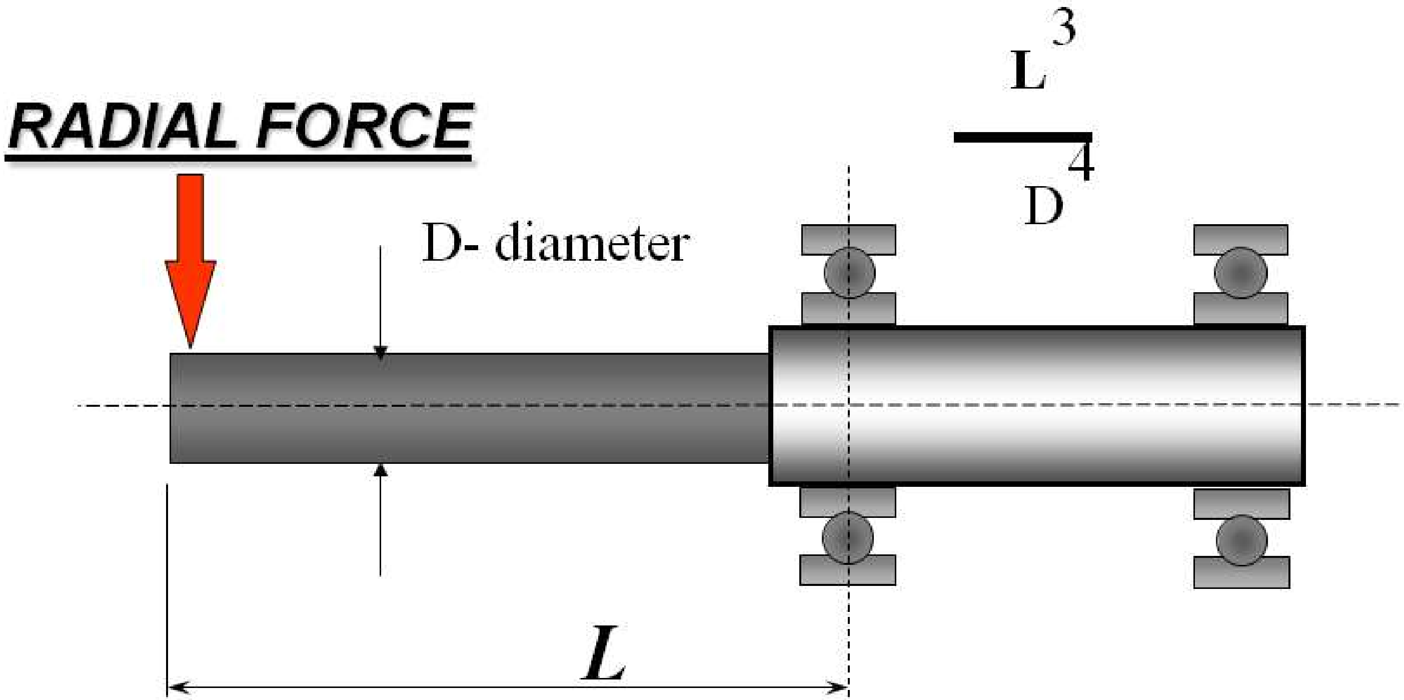


The further the pump is operated away from design flow (Q_{nom}) versus the actual flow (Q) the greater the shaft deflection and stress on the seals (for a full speed pump)

Seal Life when operating 20-30% away from BEP

- Increased radial load and shaft deflection
- 50% increase in radial load
- Angular misalignment exceeds 0.05 mm
- Exponential deterioration of seal life
- Less than 6 months life

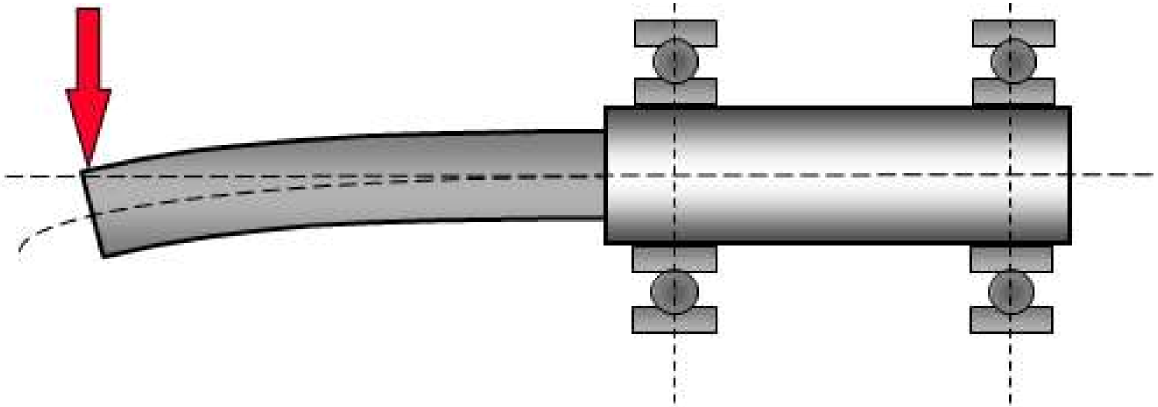
Shaft Deflection



Seal Life

Shaft Deflection

Throttled Discharge

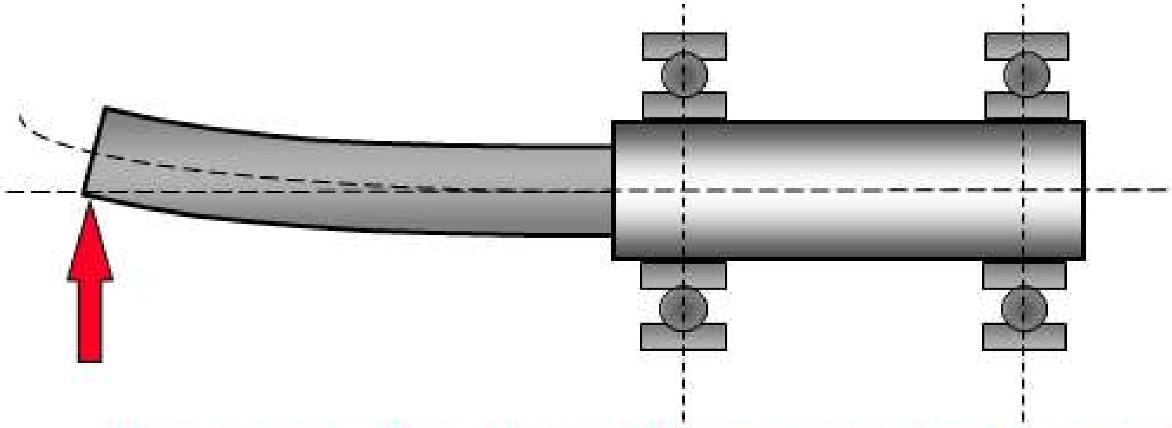


Springs operating 3000 x Min. to keep the faces together.

Seal Life

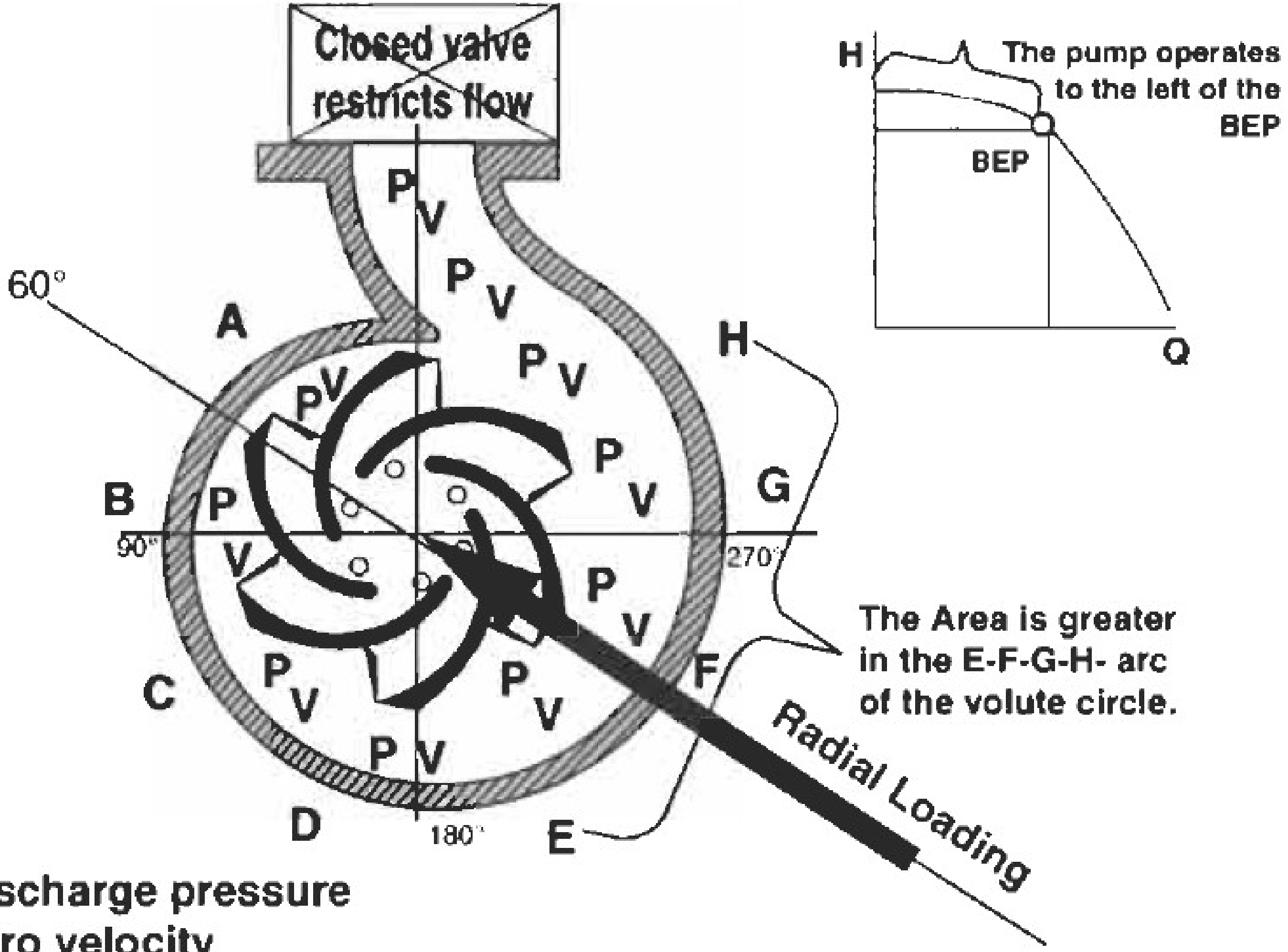
Shaft Deflection

Maximum Discharge
Too much Capacity



Springs operating 3000 x Min. to keep the faces together.

Shaft deflection - Throttled Discharge



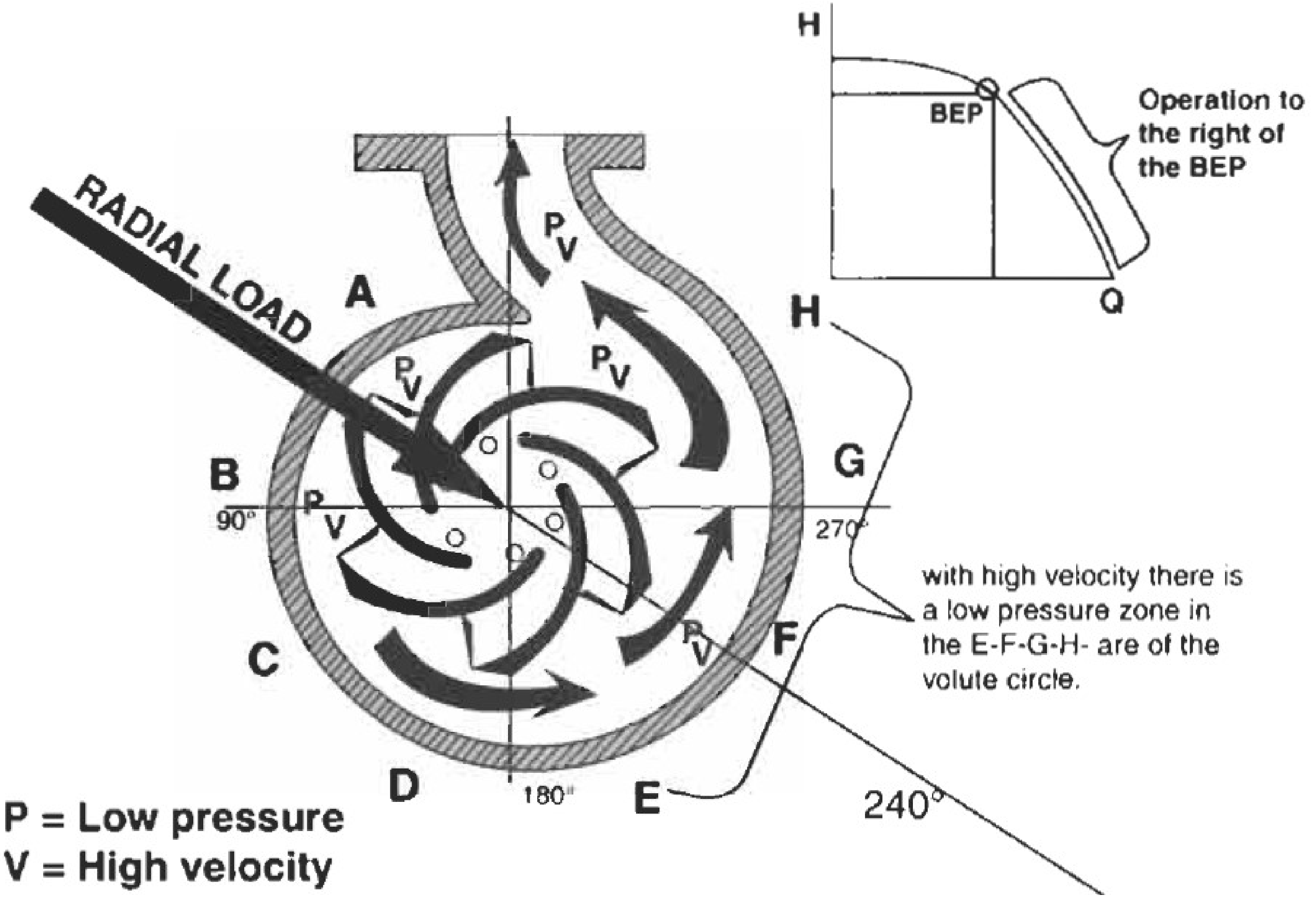
The pump operates to the left of the BEP

The Area is greater in the E-F-G-H- arc of the volute circle.

P = Discharge pressure
V = Zero velocity

Source: Mohamed Afifi

Shaft deflection - Too much Capacity



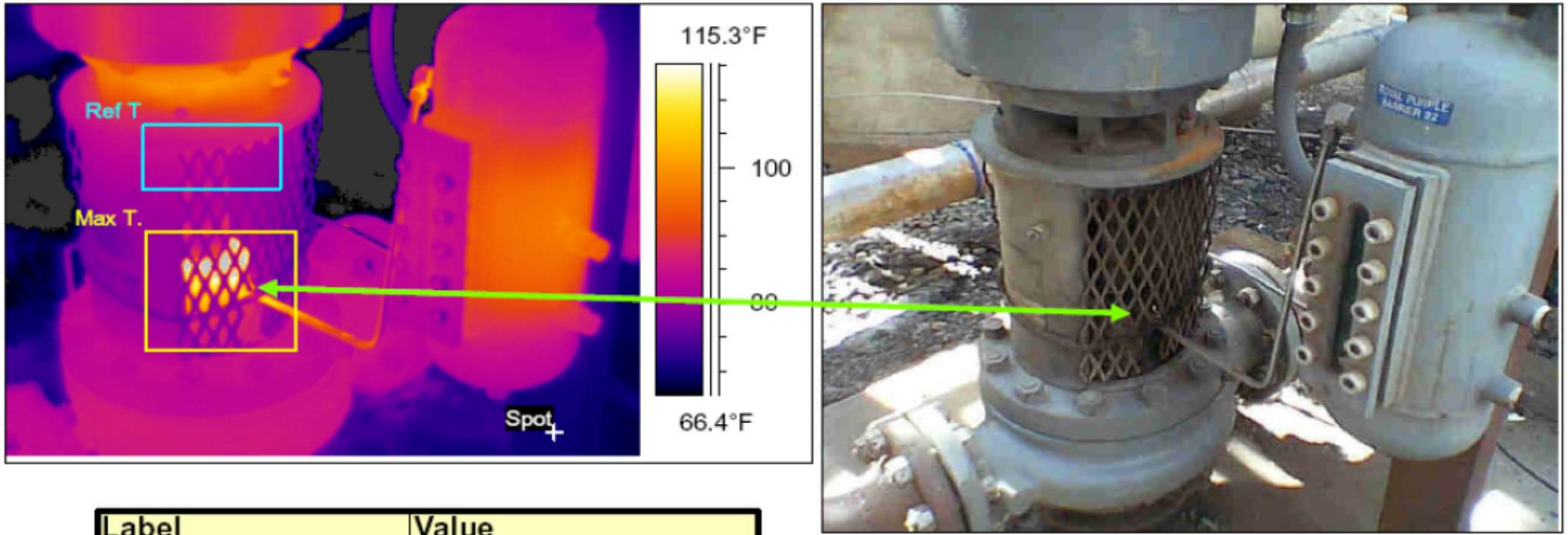
Operation to the right of the BEP

with high velocity there is a low pressure zone in the E-F-G-H- are of the volute circle.

P = Low pressure
V = High velocity

Source: Mohamed Afifi

Verify correct functioning of seals and bearing cooling



Label	Value
Spot	70.1°F
Max T. : max	178.9°F
Ref T : max	89.1°F
Delta T	**89.78°F**

IR information	Value
Date of creation	10/27/2005
Time of creation	12:52:07 PM

Fault Description and Conditions

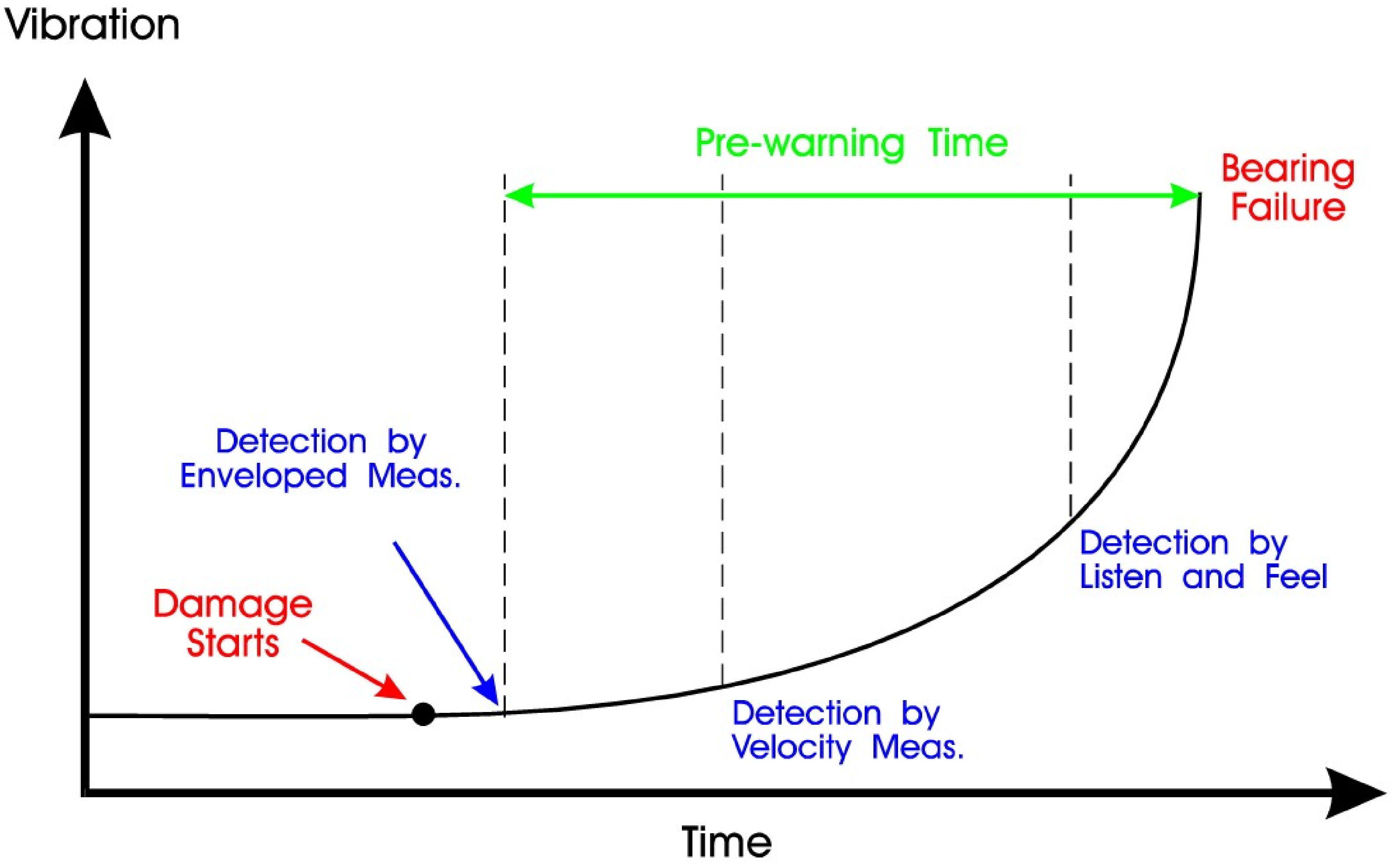
Seal is operating at higher than normal temperatures.
It appears that the seal lubrication is not circulating as it should

Maintenance Practices

- Five levels of maintenance operating practices:
  - ***Lowest Level:*** Fix it when it breaks, few maintenance records or spare parts, lack of training/capabilities
  - ***Second Level:*** Short range fixes, better maintenance records, some spare parts maintained
  - ***Third Level:*** Planned preventive maintenance, routine inspections, lubrication and adjustments made, good maintenance records, input from operations and engineering for maintenance problem solving
  - ***Top Level:*** Predictive maintenance techniques used (vibration, thermography), performance monitoring, problems are anticipated, computerized maintenance management system fully utilized

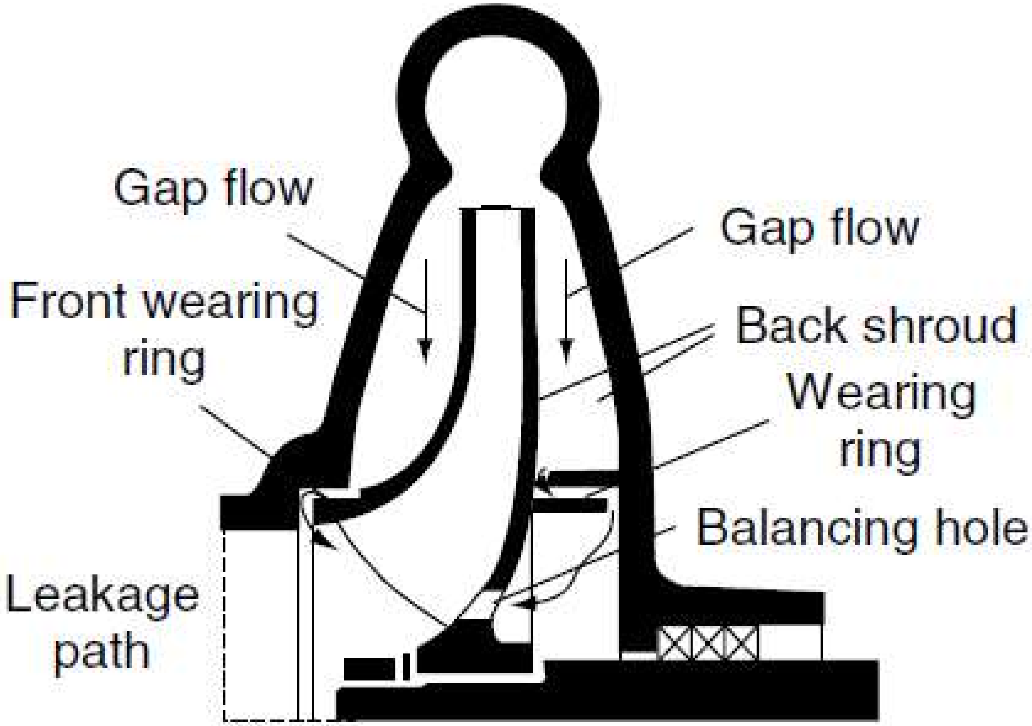
Typical Condition Monitoring

Describes advantage of Performance Monitoring

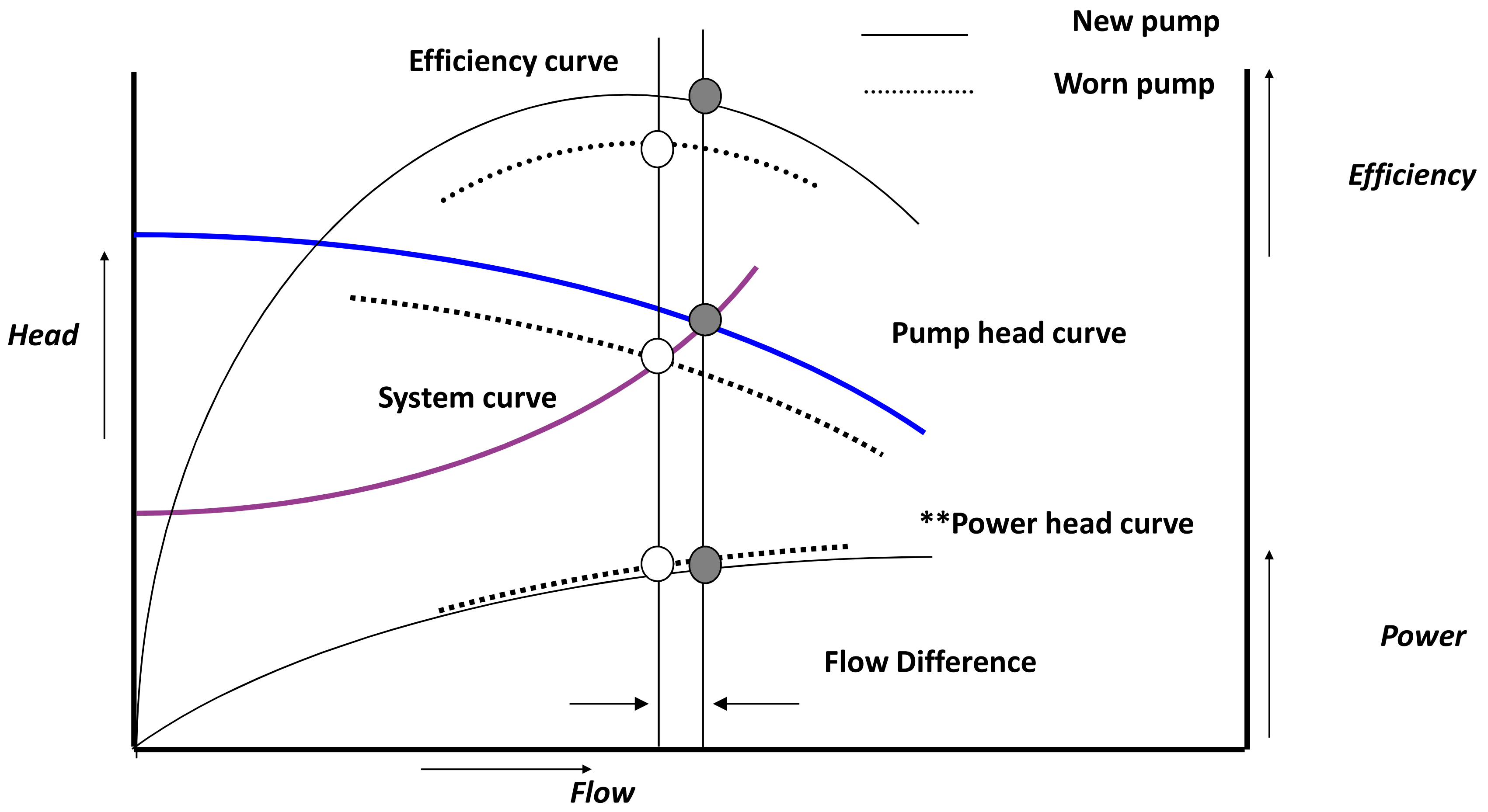


Wear Rings

- Wear rings - to provide proper clearances (between impeller and casing)
- Over time these clearances increase, recirculation the fluid from the high pressure side of an impeller to the low pressure side.

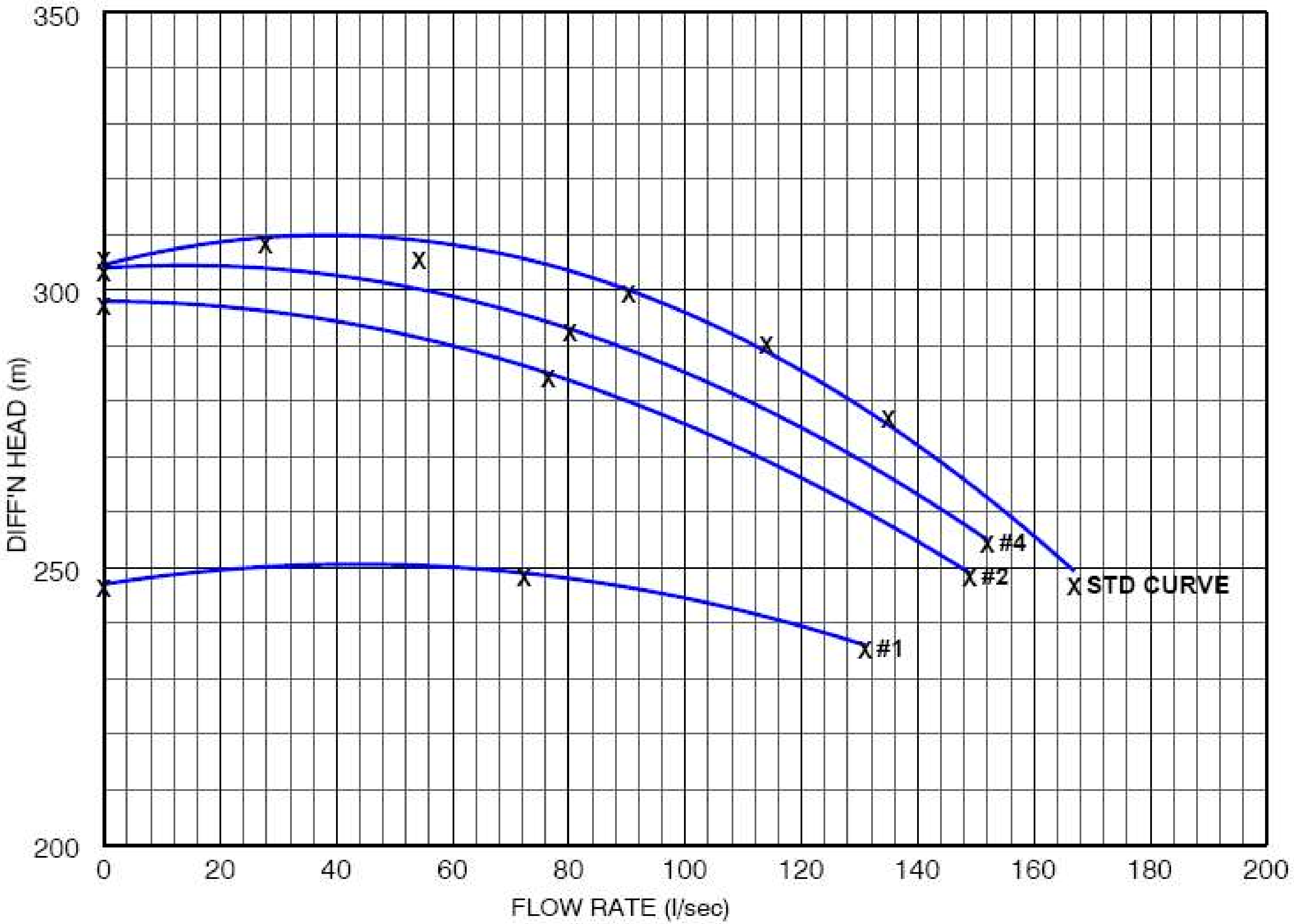


How pump wear can impact pump performance



Power may also ***decrease* when efficiency is reduced – but will also produce less flow

Drop off in pump performance due to wear



Making clearance adjustments to improve pump efficiency

Pump efficiency can be improved by adjusting impeller clearances for hollow shaft motors with semi-open impellers

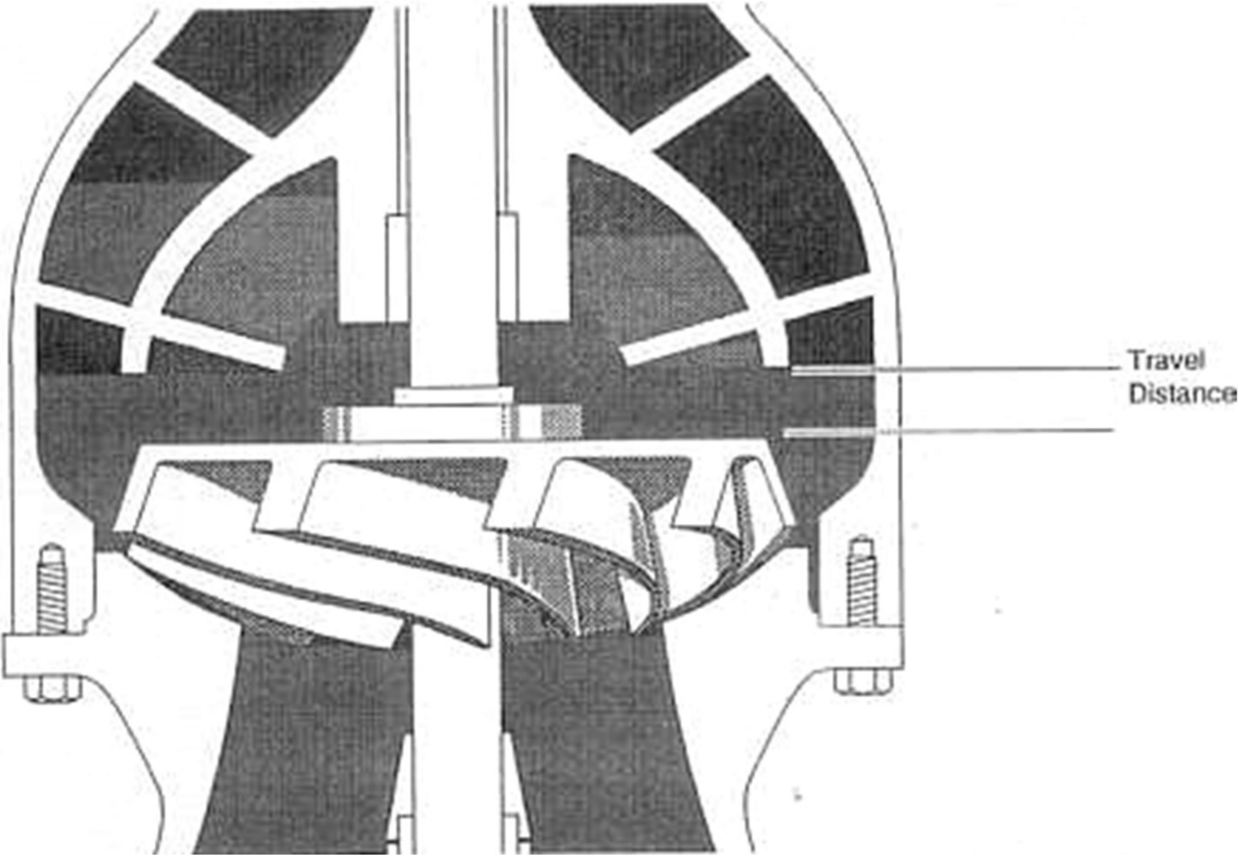


Figure Courtesy of ACR Publications

De-rating due to pumping Slurries

Deviations for Slurry Curves

- **Slurry De-rating Factors**

$$\text{Head Ratio} = \frac{\text{Total head developed on slurry}}{\text{Total head developed on water}}$$

$$\text{Efficiency Ratio} = \frac{\text{Pump efficiency on slurry}}{\text{Pump efficiency on water}}$$

- **Theoretical Data**
  - Density of solids
  - Particle size distribution
  - Average particle size D50
  - Concentration of solids in slurry, CV
  - Impeller diameter

Actual Slurry Performance

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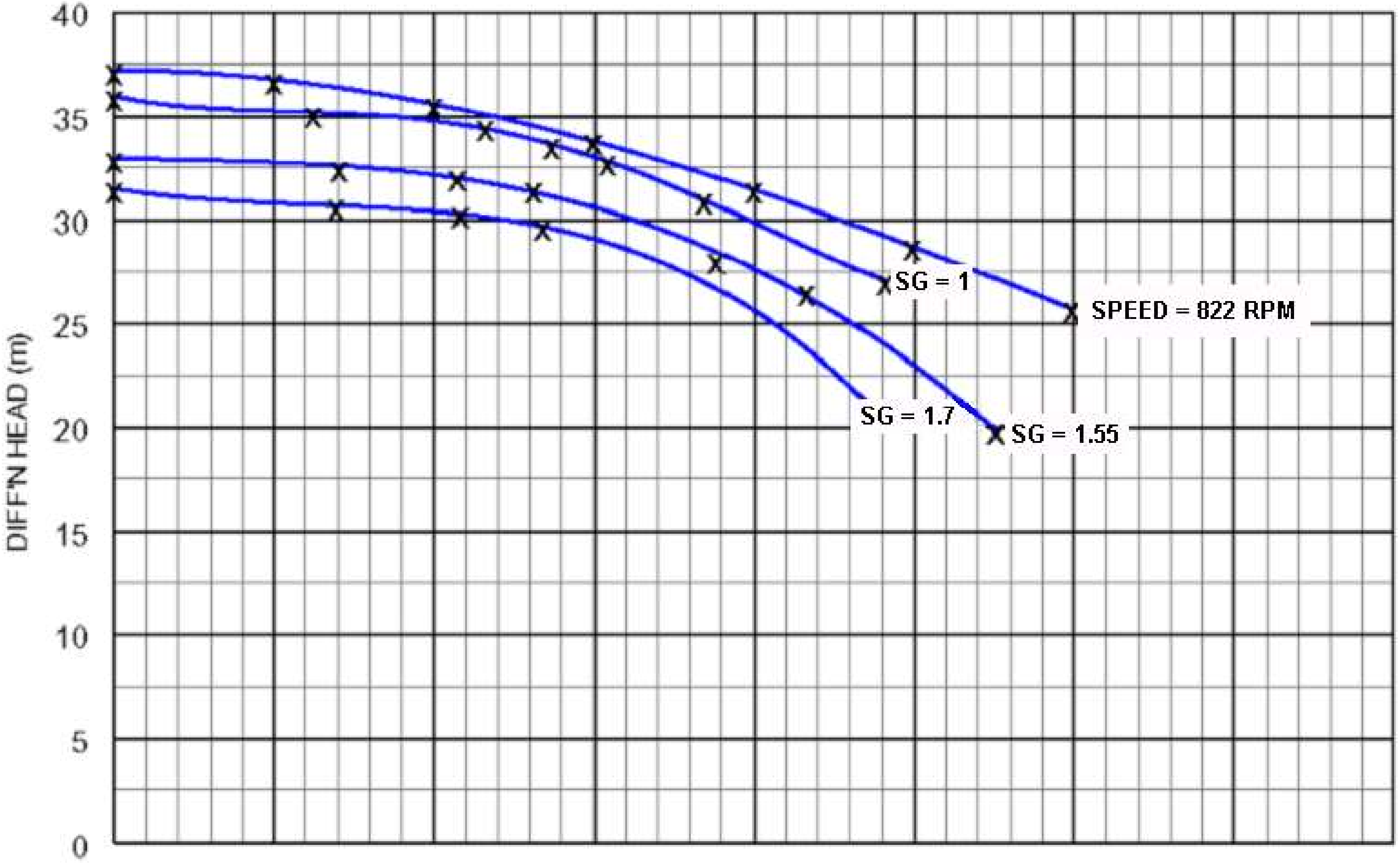
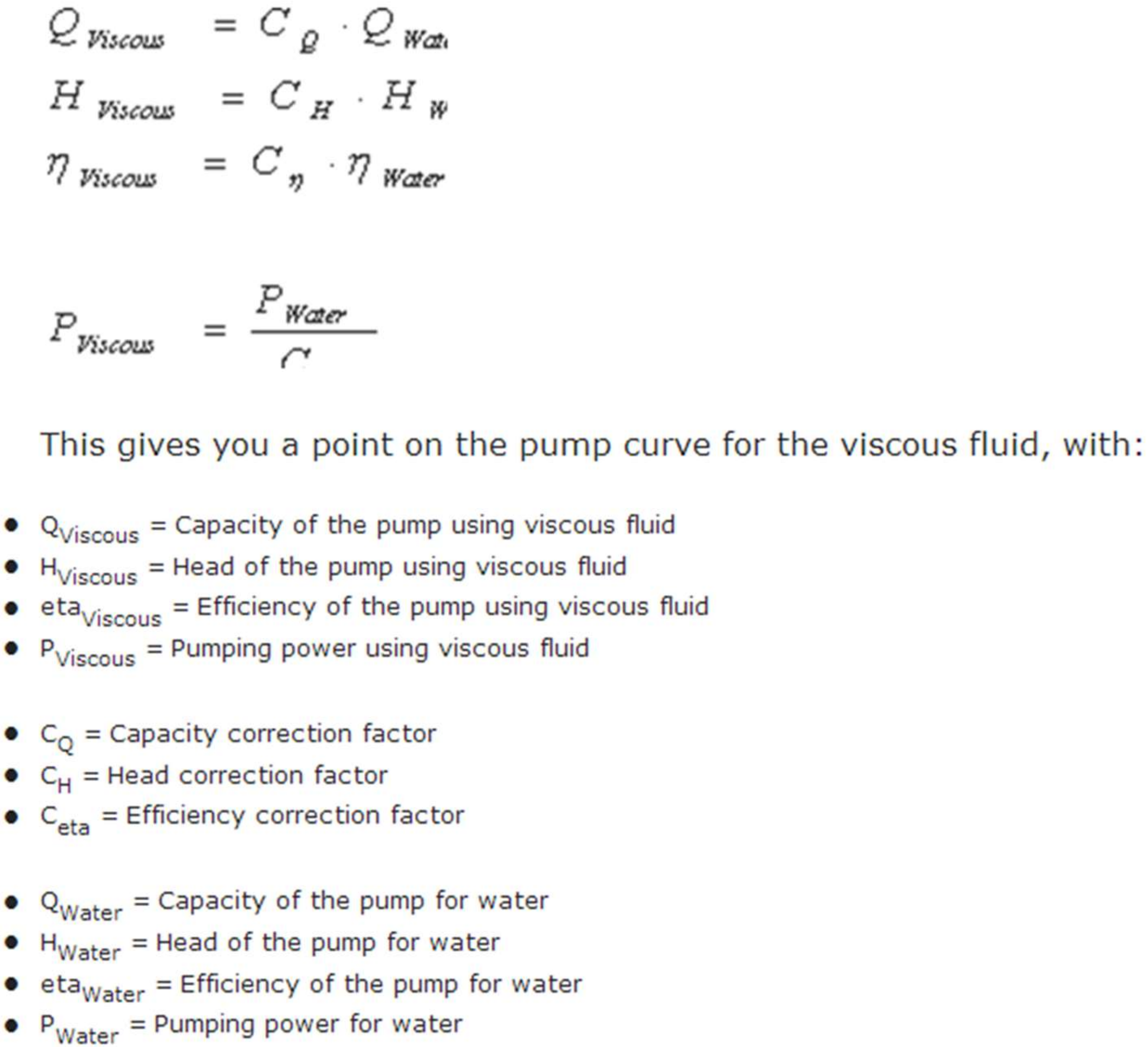
Determining Head and Efficiency Ratios Through Testing



De-rating due to Viscous effects

Viscous Liquid Correction Factors



$$Q_{Viscous} = C_Q \cdot Q_{Wat}$$

$$H_{Viscous} = C_H \cdot H_W$$

$$\eta_{Viscous} = C_\eta \cdot \eta_{Water}$$

$$P_{Viscous} = \frac{P_{Water}}{C}$$

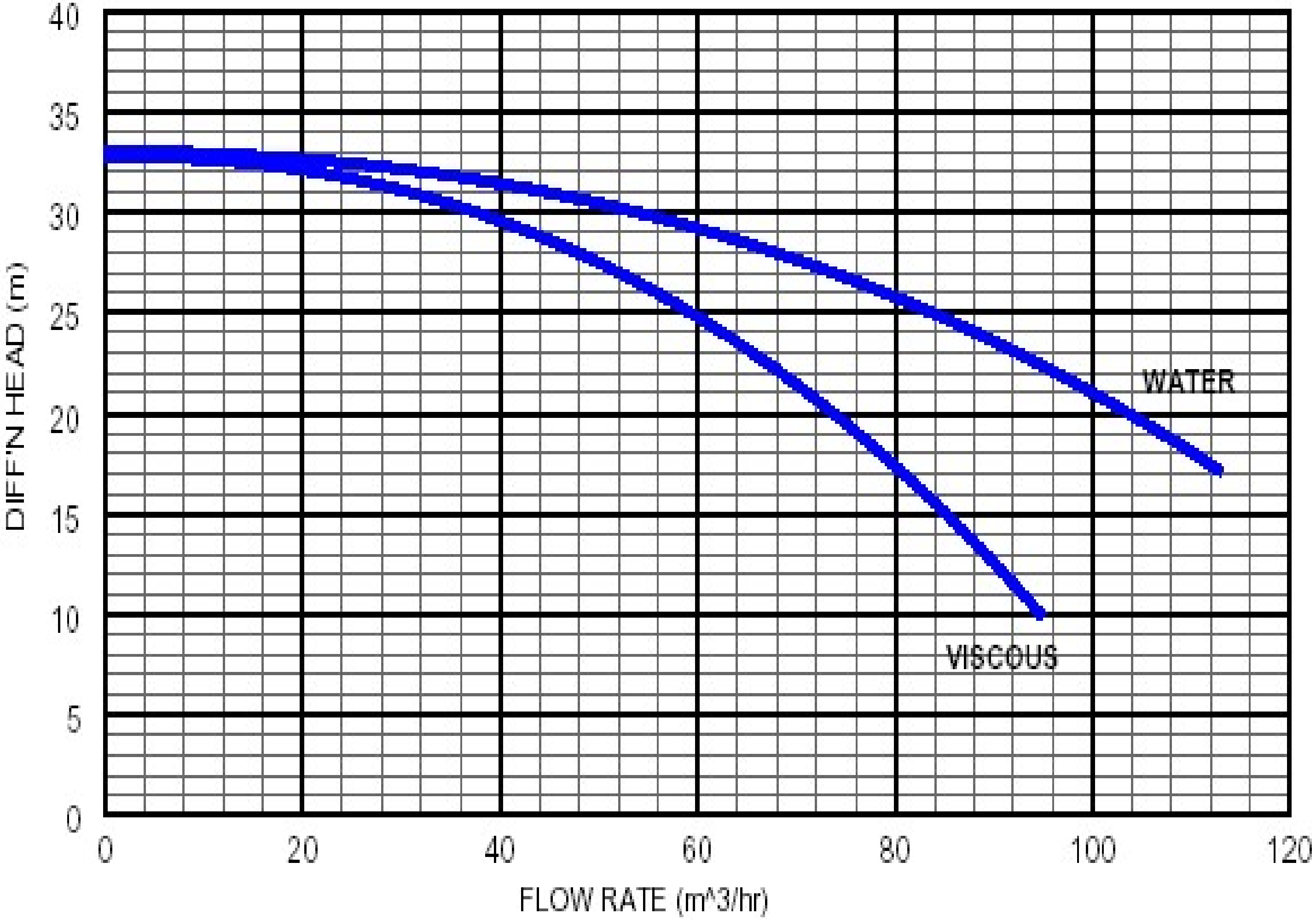
This gives you a point on the pump curve for the viscous fluid, with:

- $Q_{Viscous}$ = Capacity of the pump using viscous fluid
- $H_{Viscous}$ = Head of the pump using viscous fluid
- $eta_{Viscous}$ = Efficiency of the pump using viscous fluid
- $P_{Viscous}$ = Pumping power using viscous fluid

- C_Q = Capacity correction factor
- C_H = Head correction factor
- C_{eta} = Efficiency correction factor

- Q_{Water} = Capacity of the pump for water
- H_{Water} = Head of the pump for water
- eta_{Water} = Efficiency of the pump for water
- P_{Water} = Pumping power for water

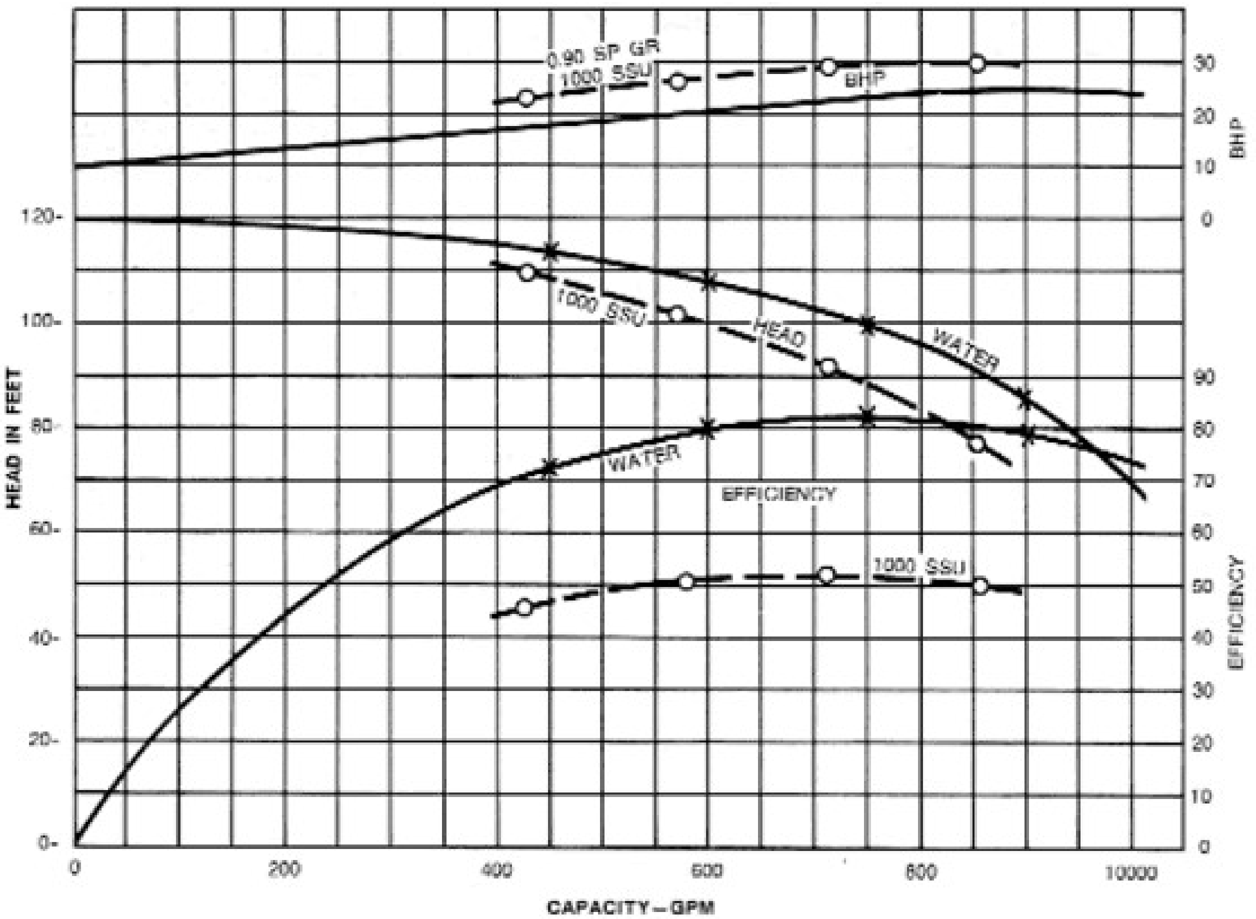
Viscous Liquids



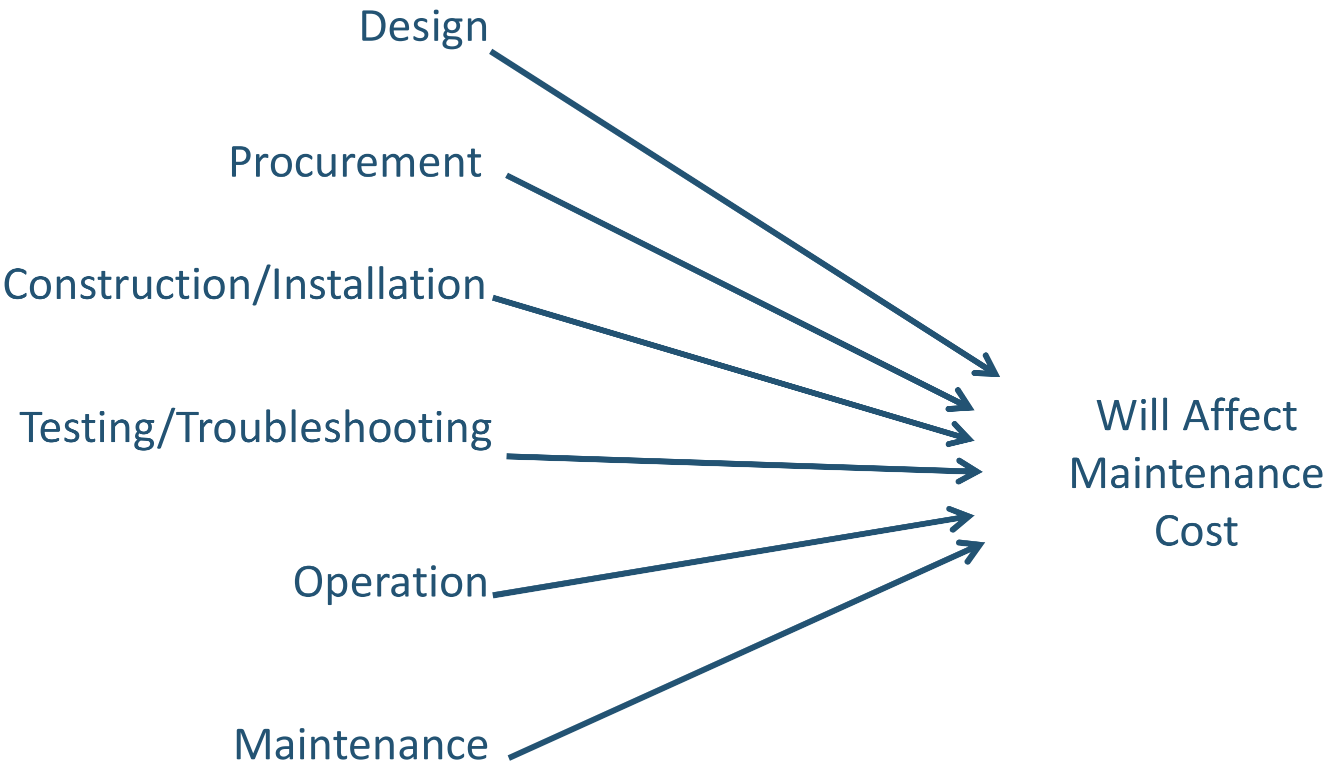
Viscous Liquid Correction Factors

Section D -- Properties of Liquids

D-4 Viscosity Corrections for Capacities of 100 GPM or Less Fig. 6 Sample Performance Chart



Many factors affect maintenance



Design

Procurement

Construction/Installation

Testing/Troubleshooting

Operation

Maintenance

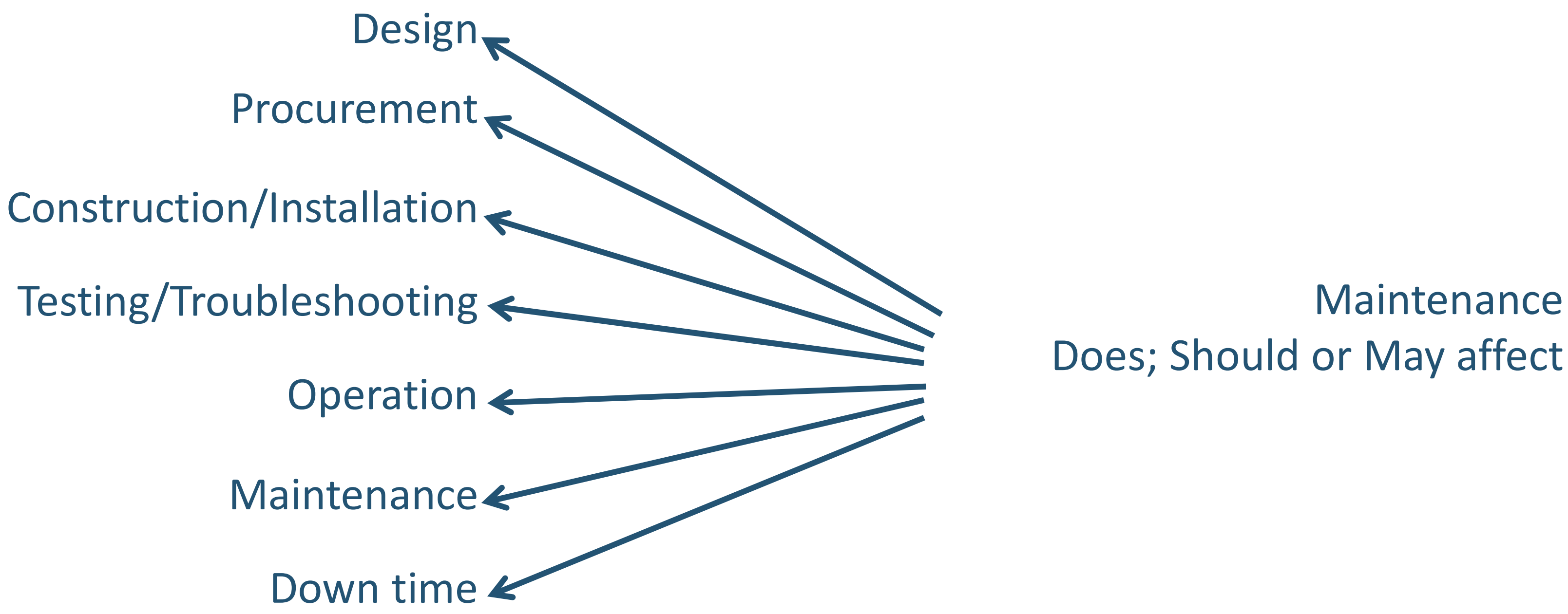
Will Affect
Maintenance
Cost

Slide Courtesy of Oak Ridge National Laboratory

Maintenance feedback

Just like any stable control system, optimal asset management requires feedback

Unfortunately, feedback is often weak or non-existent



Design

Procurement

Construction/Installation

Testing/Troubleshooting

Operation

Maintenance

Down time

Maintenance
Does; Should or May affect

Slide Courtesy of Oak Ridge National Laboratory

15. Motors

Pump Systems Optimisation (PSO) Expert Level Training
(Egypt Edition – Dec 2021)

Albert Williams
Siraj Williams

Common Motor Types

- **Types**
  - Motors used for high torque applications
  - General purpose Motors
  - Special purpose Motors
- **Enclosures:**
  - Open drip proof (ODP)
  - Totally enclosed fan cooled (TEFC)
- **Mounting**
  - Vertical or horizontal
  - Close coupled or shaft mounted
- **Service**
  - Inverter duty
  - Variable or constant torque

Motor nameplate data terms

- **Rated Voltage**: Motors are typically designed to be operated +/- 10% of the rated voltage.
- **Rated Full Load Amperage**: This is the value reached when full load torque and power is applied. FLA is used to select the correct wire size and overload protection devices.
- **Rated full load Speed**: This is the motor speed under full load conditions.
- **Insulation Class**: Insulation class can be B,F,H and is a measure of how hot the windings can get without shortening the life of the motor.

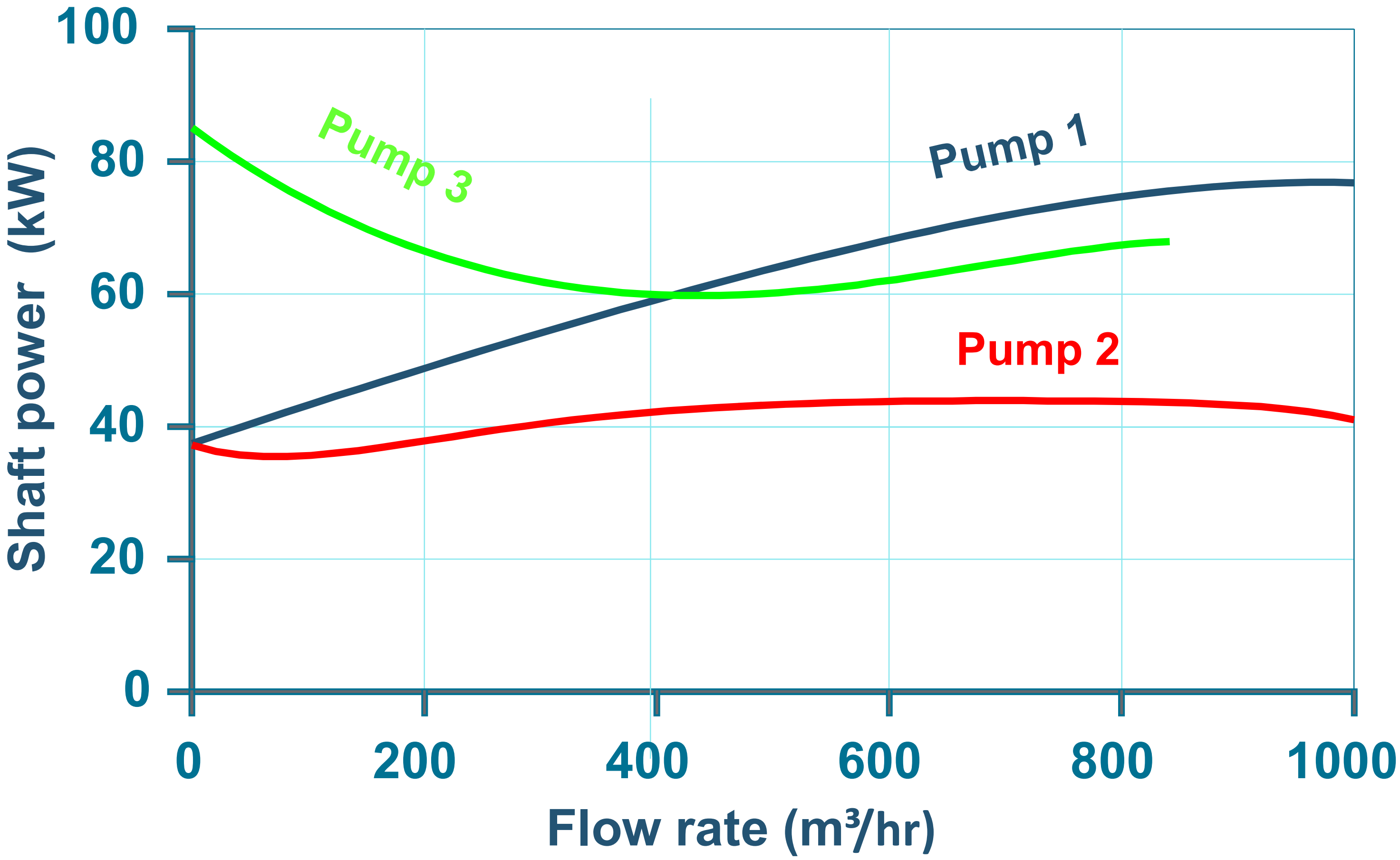
Motor nameplate data terms

- **Rated Power Output**: The rated shaft power output at the rated voltage, current and frequency. Units are kW for IEC motors, and HP for NEMA motors.
- **Service factor**: Service factor is an indication of how much overload a motor can take. Motors should not be operated in the service factor continuously.
- **Full load efficiency**: This is often given as “nominal” or “guaranteed minimum” and provides an indication of motor efficiency.

Conditions that can affect motor performance

- **Frequent starts and stops**
- **Power Quality (i.e. harmonics)**
- **Application of VSD's**
- **Operating in the service factor**
  - NEMA recommends that motors should be de-rated when operating in the SF area
- **Voltage unbalance or under/over voltage**
  - Creates additional heat
  - Increases motor internal losses
  - Motor is de-rated for high voltage unbalance
- **Environmental conditions**
  - Poor cooling due to high ambient temperatures
  - Partially clogged motor vents
  - Dirty/wet application

Different motor loads for different types of centrifugal pumps



Slide Courtesy of Oak Ridge National Laboratory

Typical high efficiency motor curves

Egyptian program for promoting Industrial Motor Efficiency
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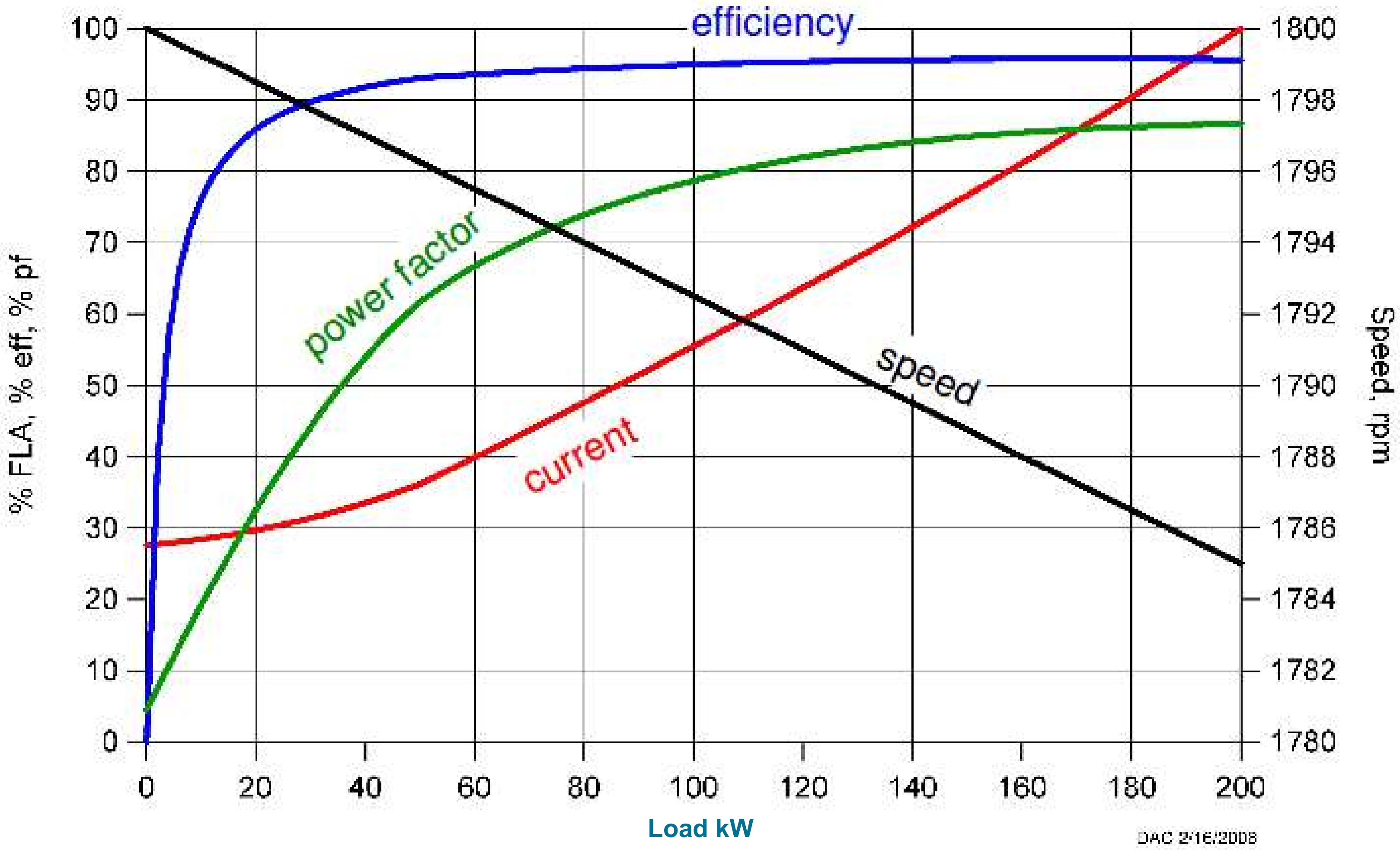
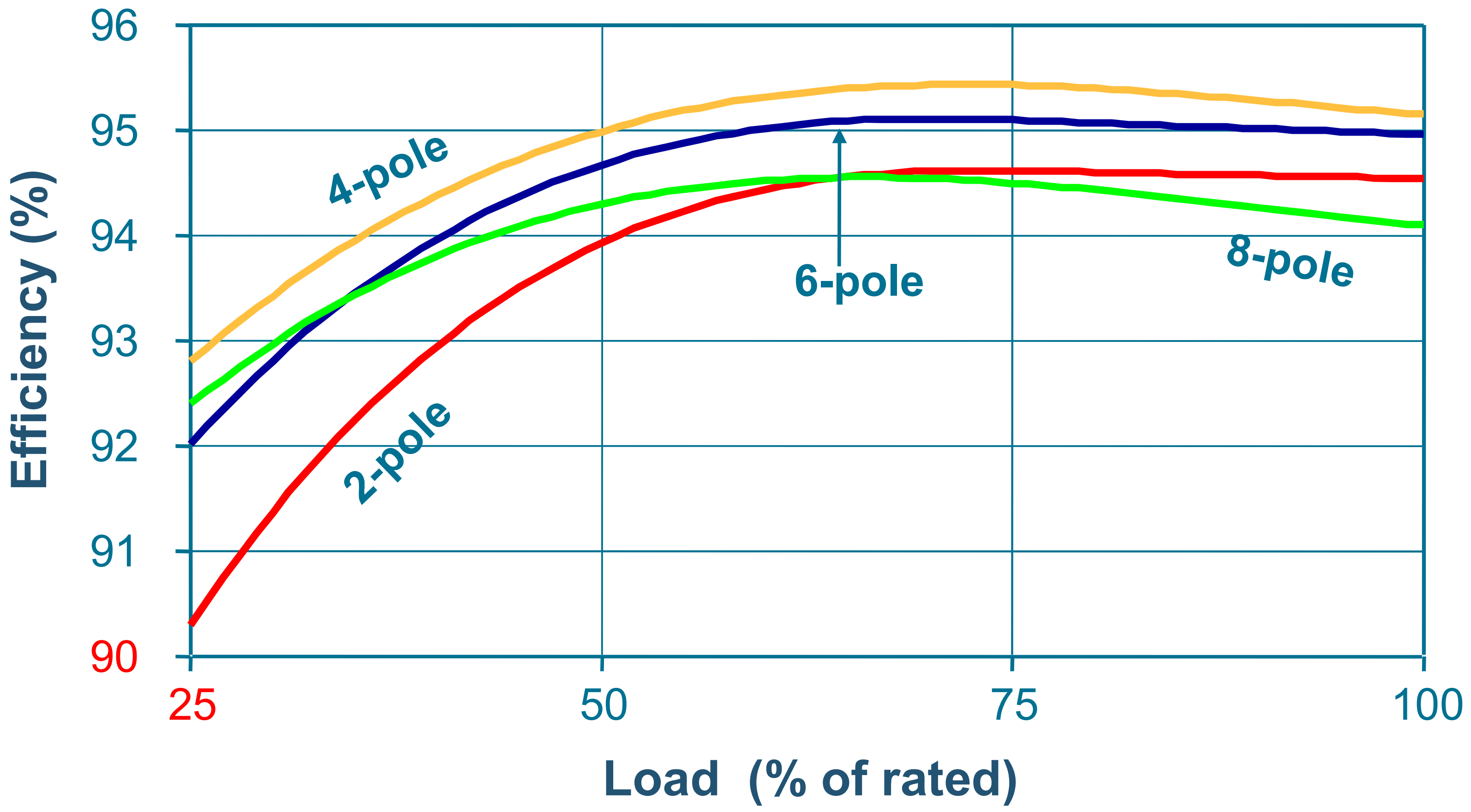
(150 kW, 4-Pole)



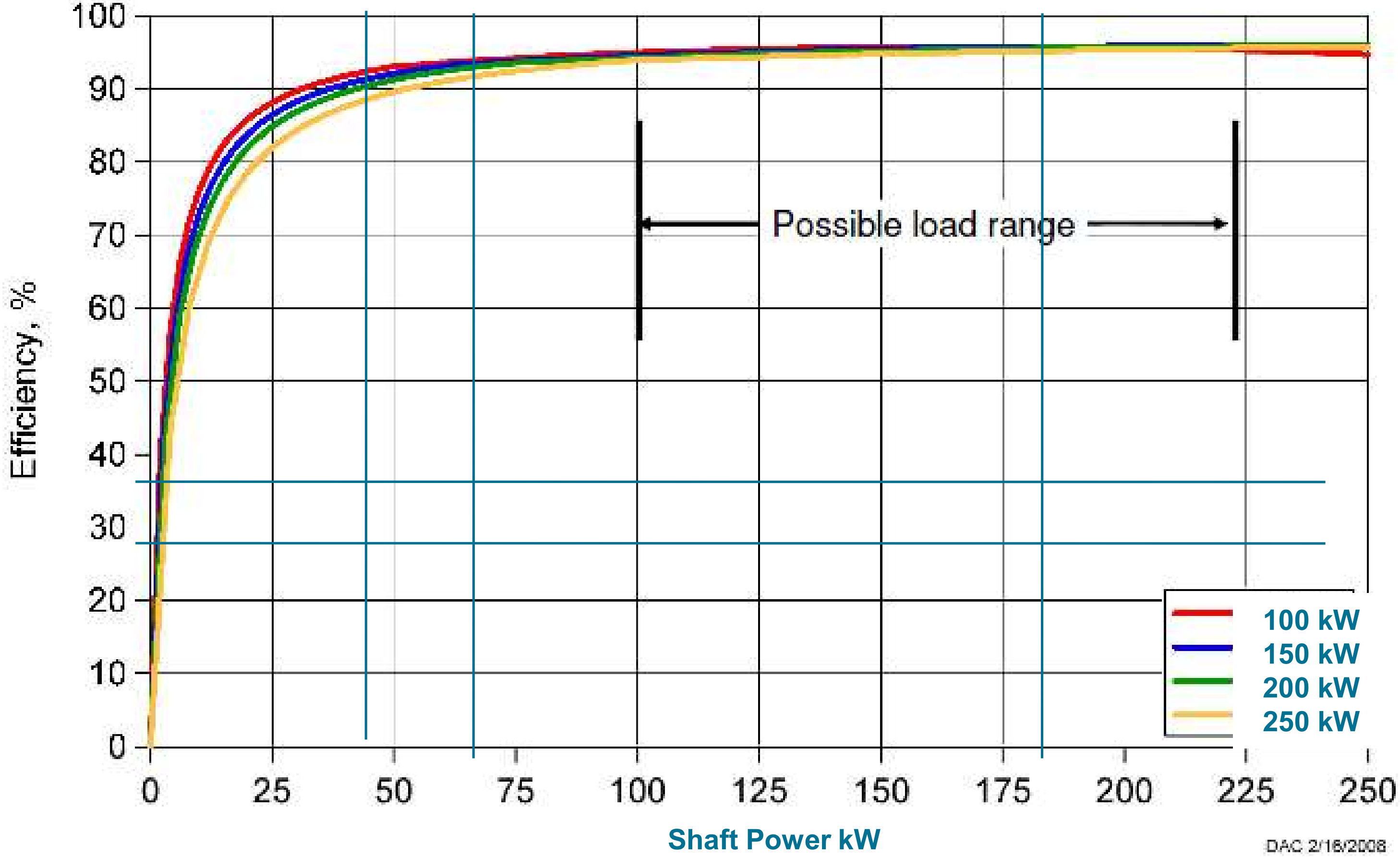
Motor efficiencies for 75 kW motors

- typical performance curves over normal pump load range



Slide Courtesy of Oak Ridge National Laboratory

Effect of an oversized motor (virtually nothing)



Motor improvements

- **Determine existing condition of motors**
  - Electrical measurements
  - Motor circuit analysis, infrared thermography
  - Efficiency

- **Plan ahead for repair/replace options**
  - Use MEASUR or *MotorMaster* software to evaluate savings by upgrading to a premium efficiency motor
  - Application of VSD will impact new motor type

Motor improvements

- **Evaluate how pump upgrades may effect the motor**
  - Opportunity to resize the motor
  - Effect of VSD
  - Impact on motor service factor
- **Determine how motor upgrades will impact pump performance**
  - The higher RPM of premium efficiency motors will increase pump capacity and power
  - The absorbed power is ± proportional to the cube of the speed!!!
  - A 2% speed increase could lead to 8% higher power useage

Egyptian program for promoting Industrial Motor Efficiency
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16. Control Methods

Pump Systems Optimisation (PSO) Expert Level Training
(Egypt Edition – Dec 2021)

Albert Williams
Siraj Williams

Flow Control in Pumping Systems

Process Types:

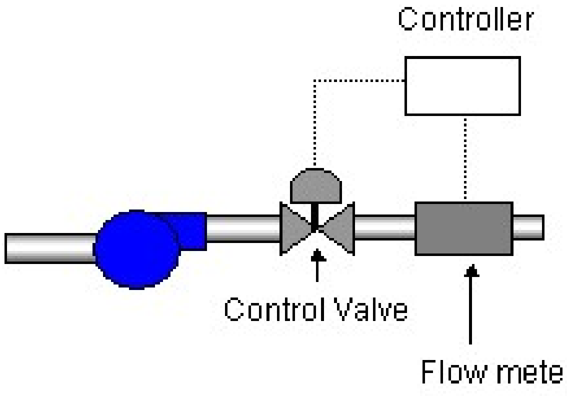
- Continuous
- Batch
- Combination

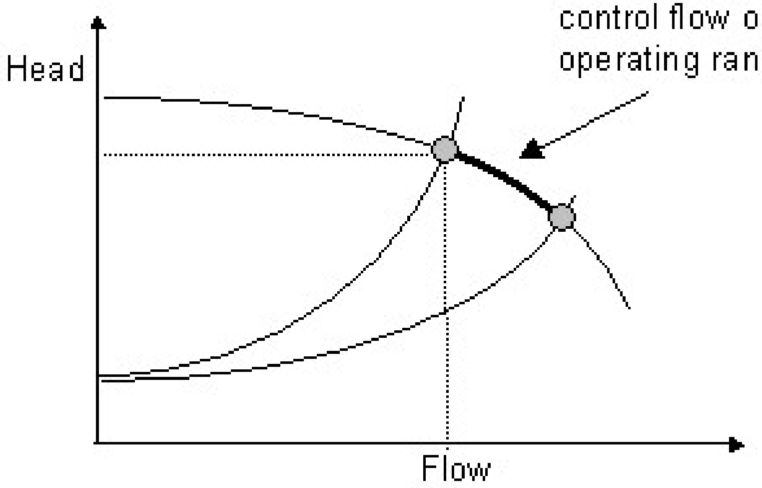
Can be either Steady or Variable flow

Control Strategies:

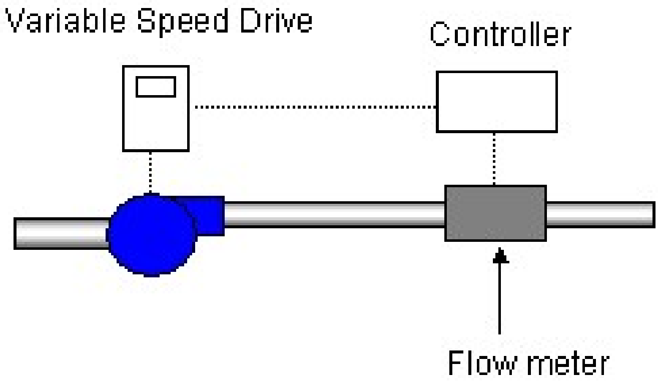
- On/off
- Valve throttling
- Bypassing
- VSD
- Combinations

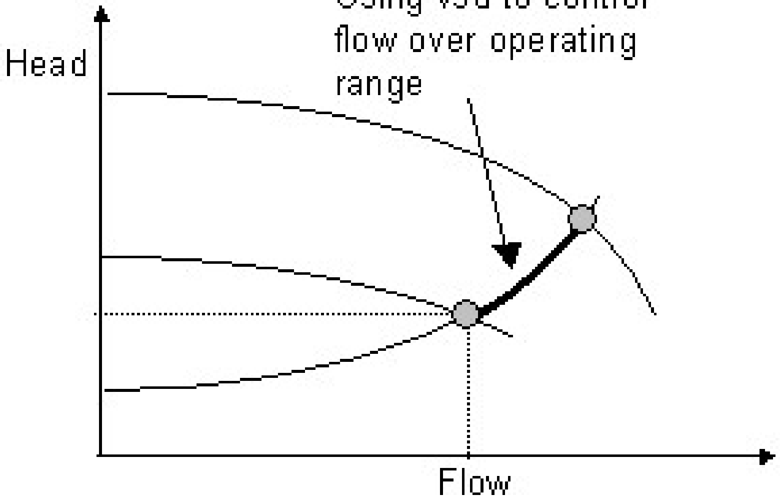
Flow Control Methods





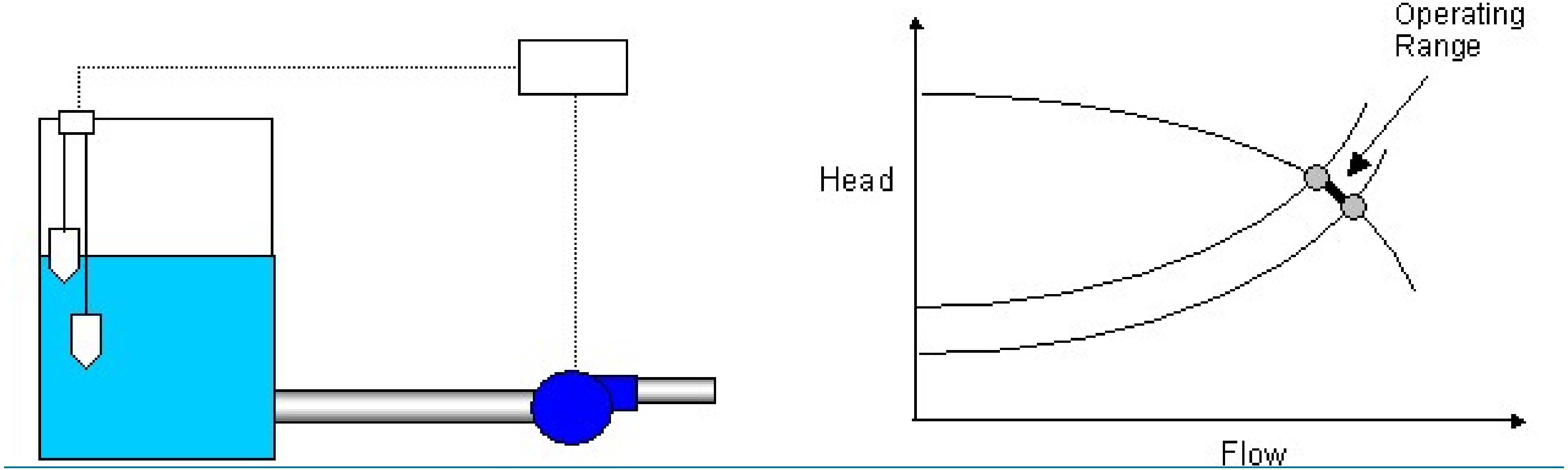
Throttled valve to control flow over operating range

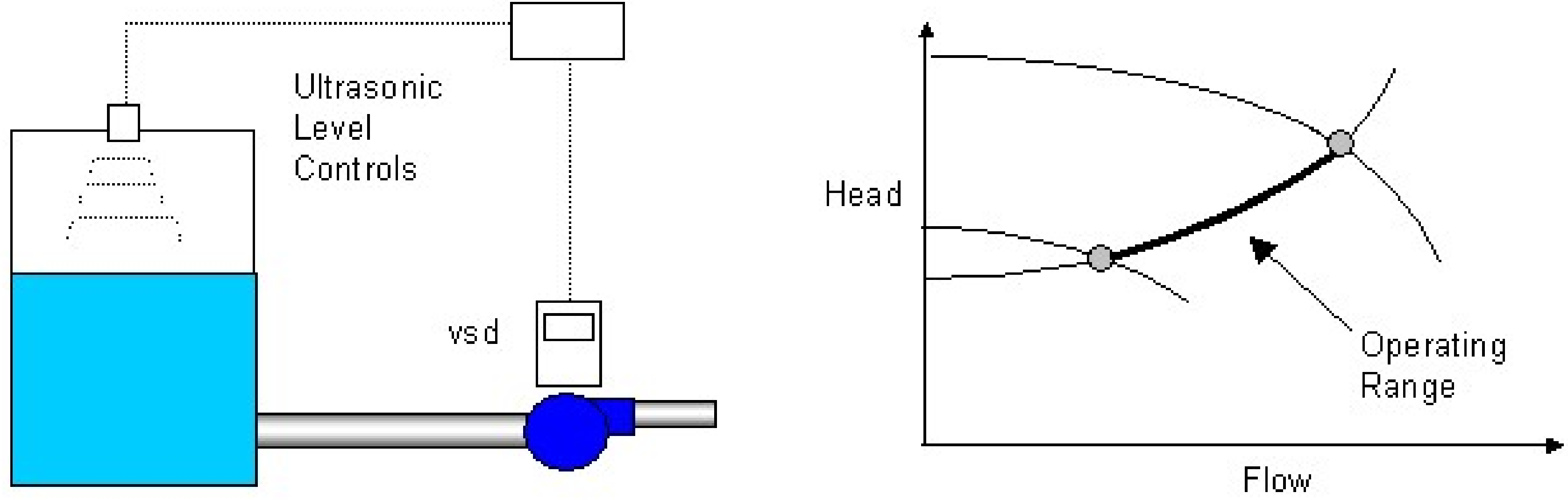




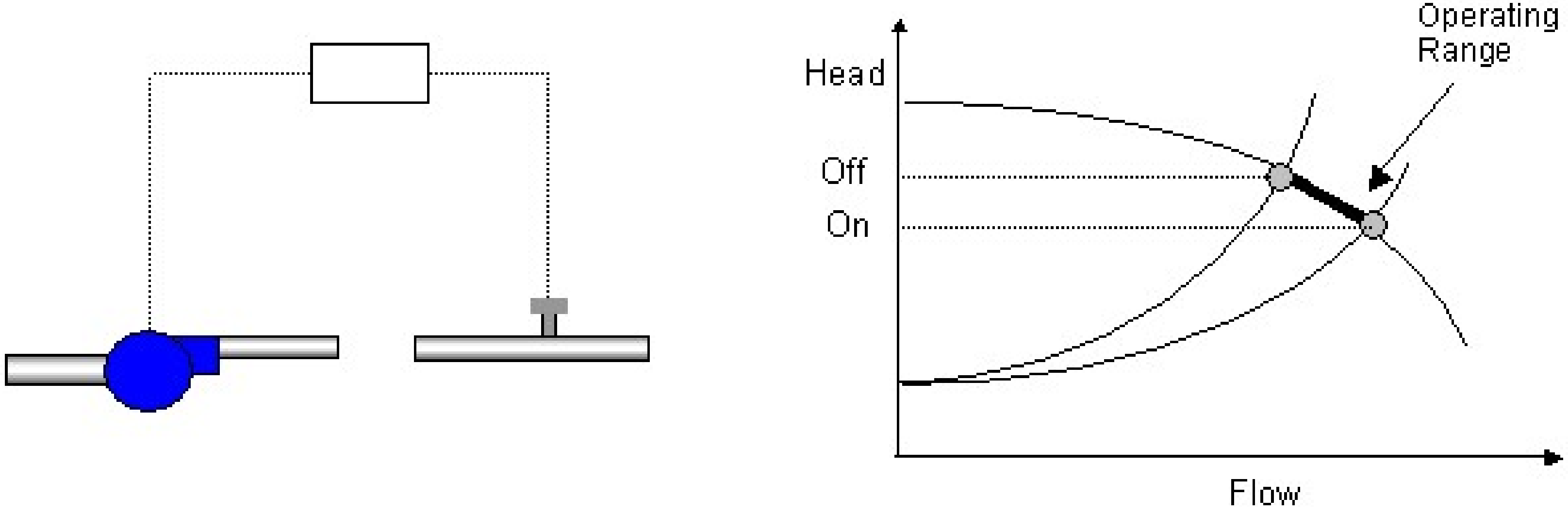
Using vsd to control flow over operating range

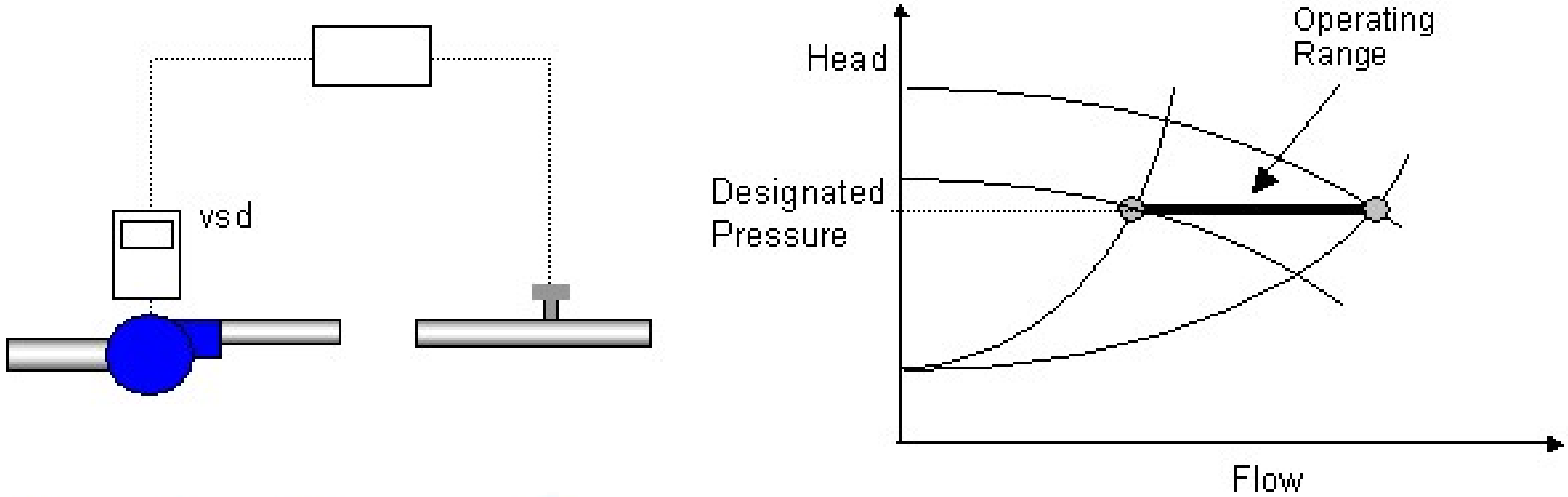
Level Control Methods





Pressure control methods

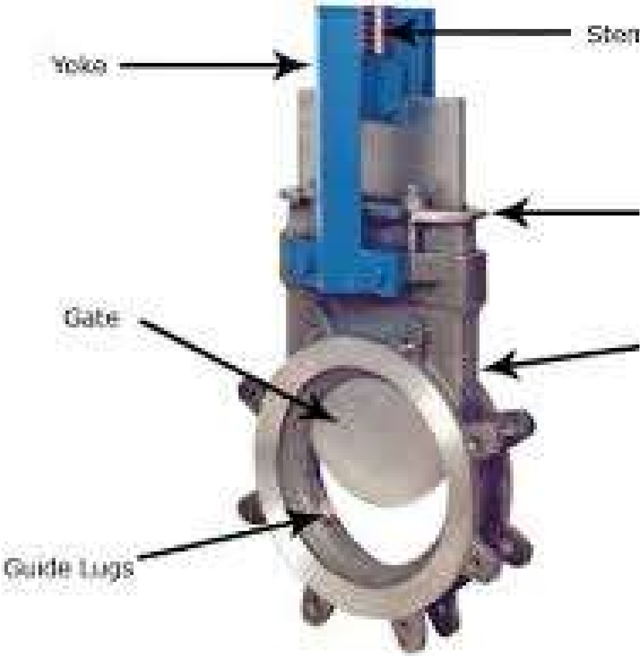




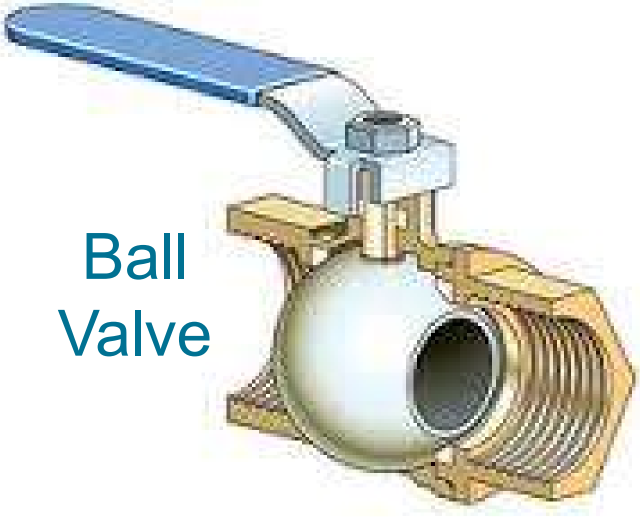
Mechanical Flow Control

Butterfly Valve

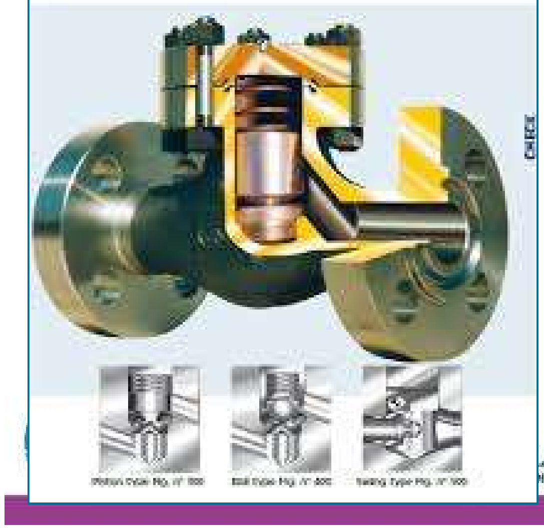
Gate Valve



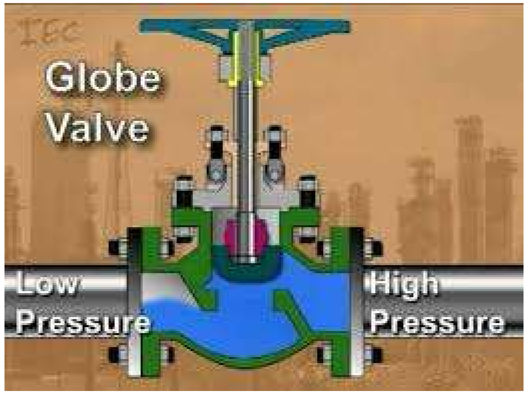
Ball Valve



Check Valve



Globe Valve

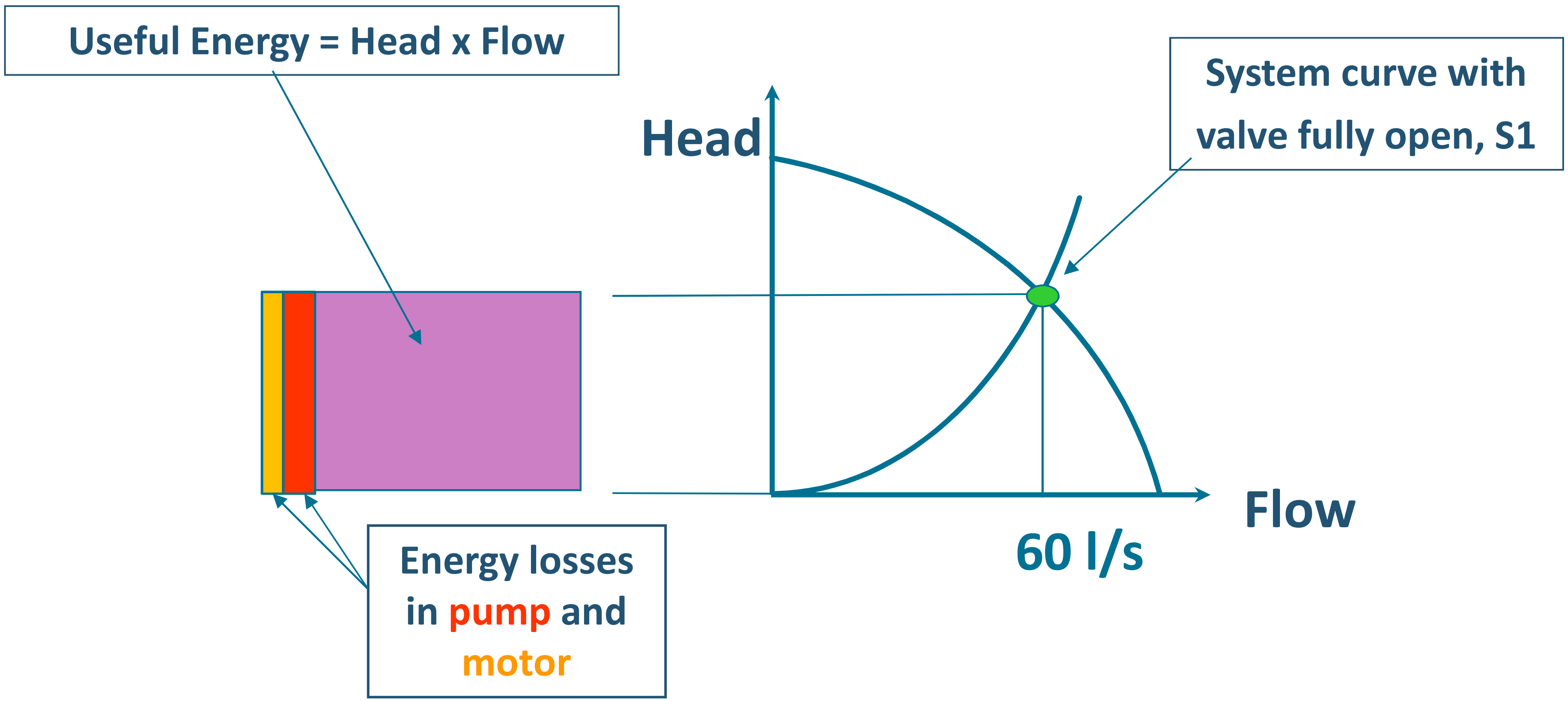


Flow Control in Pumping Systems

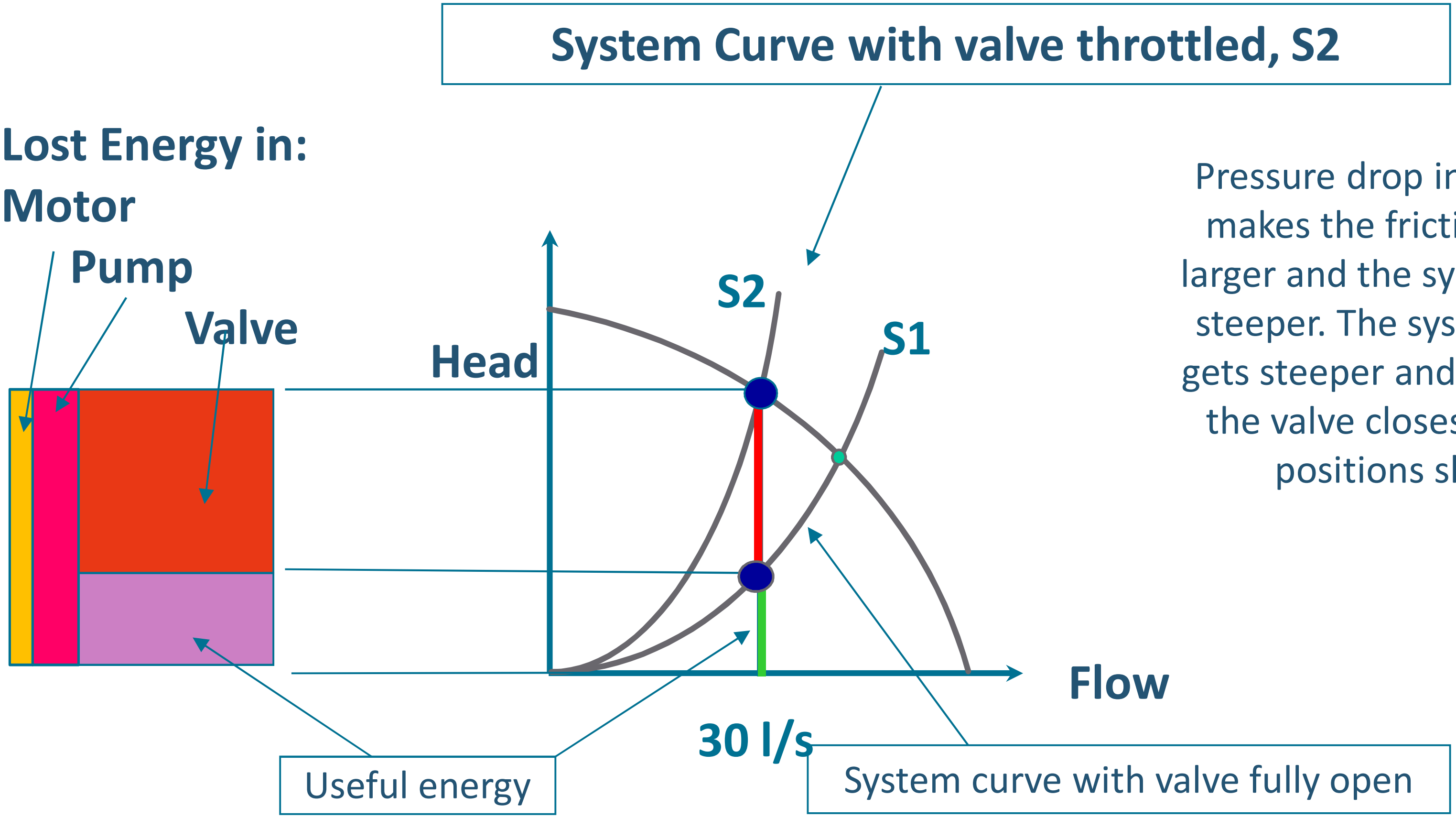
We will look at two options to achieve flow control in pumping systems:

- Change the **system curve** (valve throttling, bypass control, process demand change);
- Change the **pump curve** by
  - Changing the pump speed (by using a VSD or changing the motor)
  - Trimming the impeller or
  - Downsizing the pump/motor

Pump System Energy Representation



Throttling: Duty Point Moves to Left on the Pump Curve



System Curve with valve throttled, S2

Lost Energy in:
Motor
Pump
Valve

Head

S2

S1

Flow

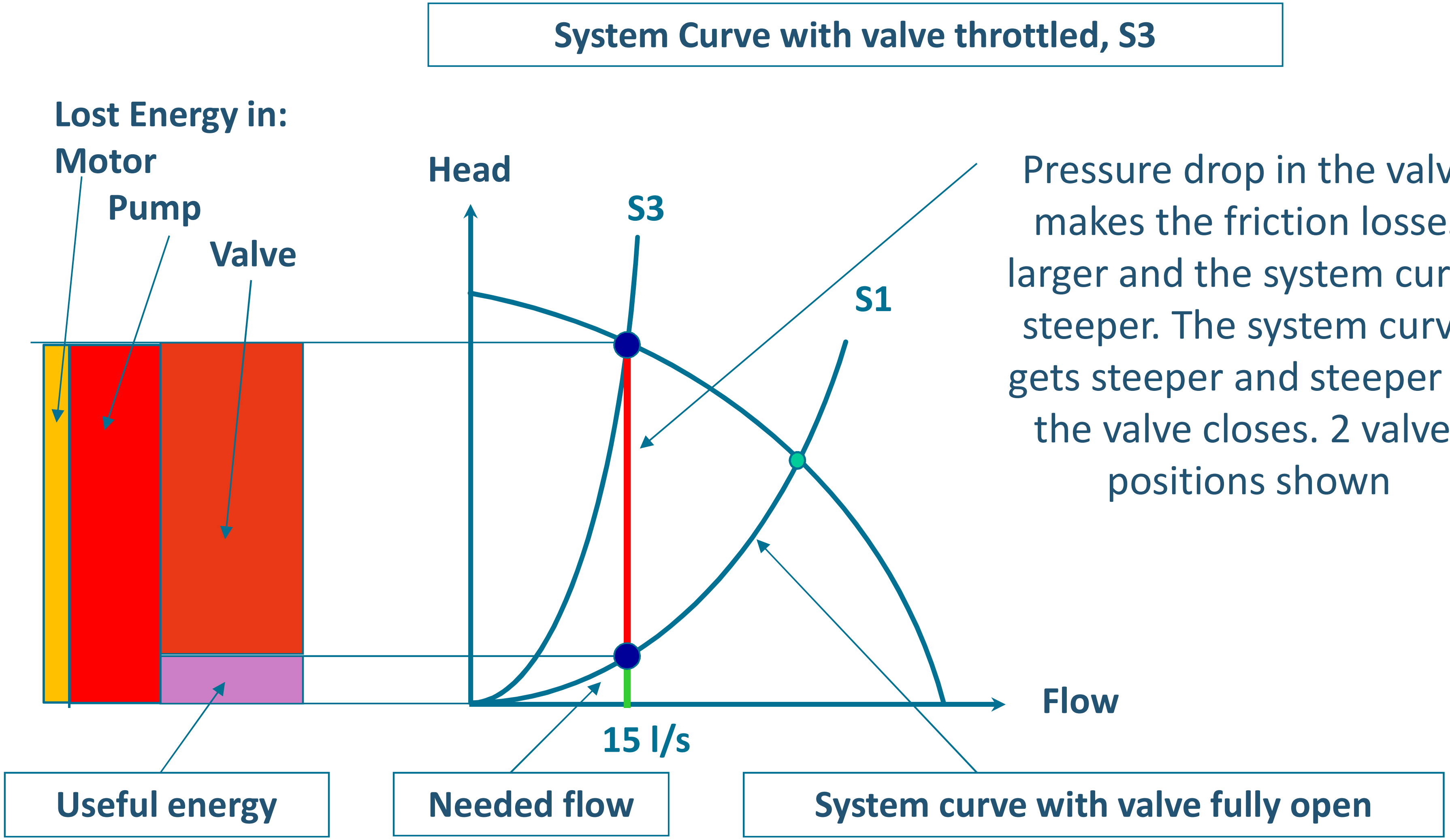
30 l/s

Useful energy

System curve with valve fully open

Pressure drop in the valve makes the friction losses larger and the system curve steeper. The system curve gets steeper and steeper as the valve closes. 2 valve-positions shown

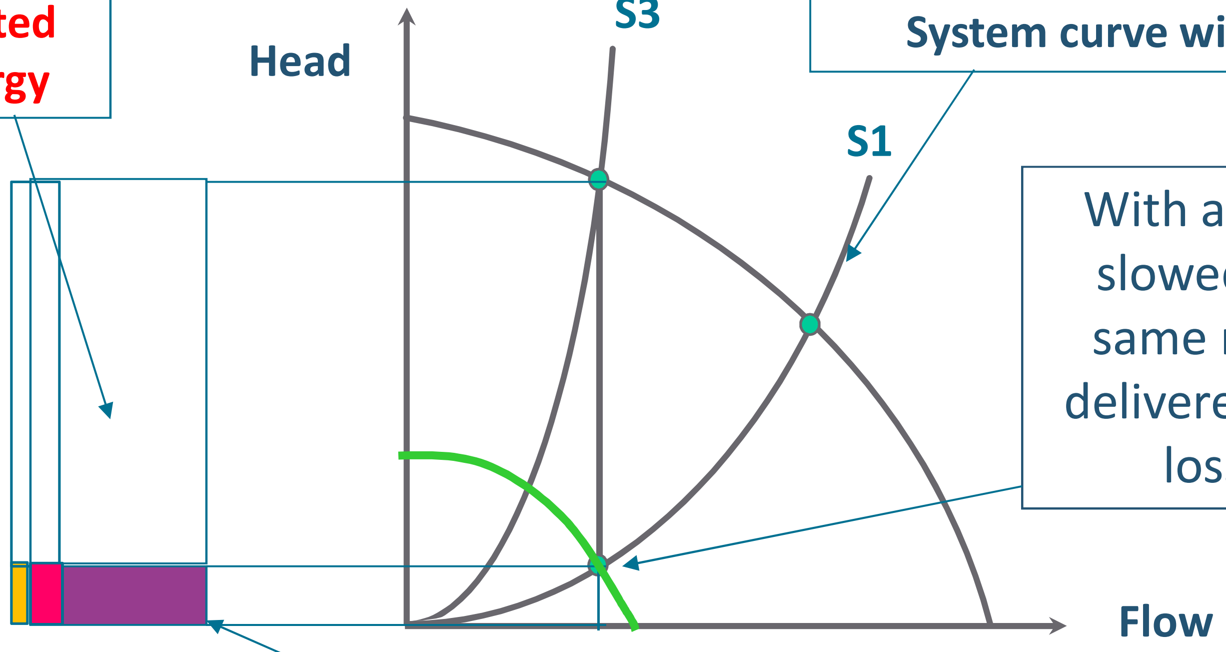
Throttling: Duty Point Moves to Left on the Pump Curve



Pressure drop in the valve makes the friction losses larger and the system curve steeper. The system curve gets steeper and steeper as the valve closes. 2 valve-positions shown

How does a VSD save energy?

The pump curve changes, ***not*** the system curve



NO Wasted Energy

System curve with valve fully open

With a smaller pump or a slowed down pump, the same rate of flow can be delivered without the large losses in the valve

Useful energy the same as before. Valve losses are eliminated

Pump Throttling Alternatives

- There are a number of alternatives that can be examined as an alternative to throttling;
  - Speed regulation
  - Impeller change or trimming
  - Multiple pumps of same or different sizes
  - Combinations of the above
  - Process change

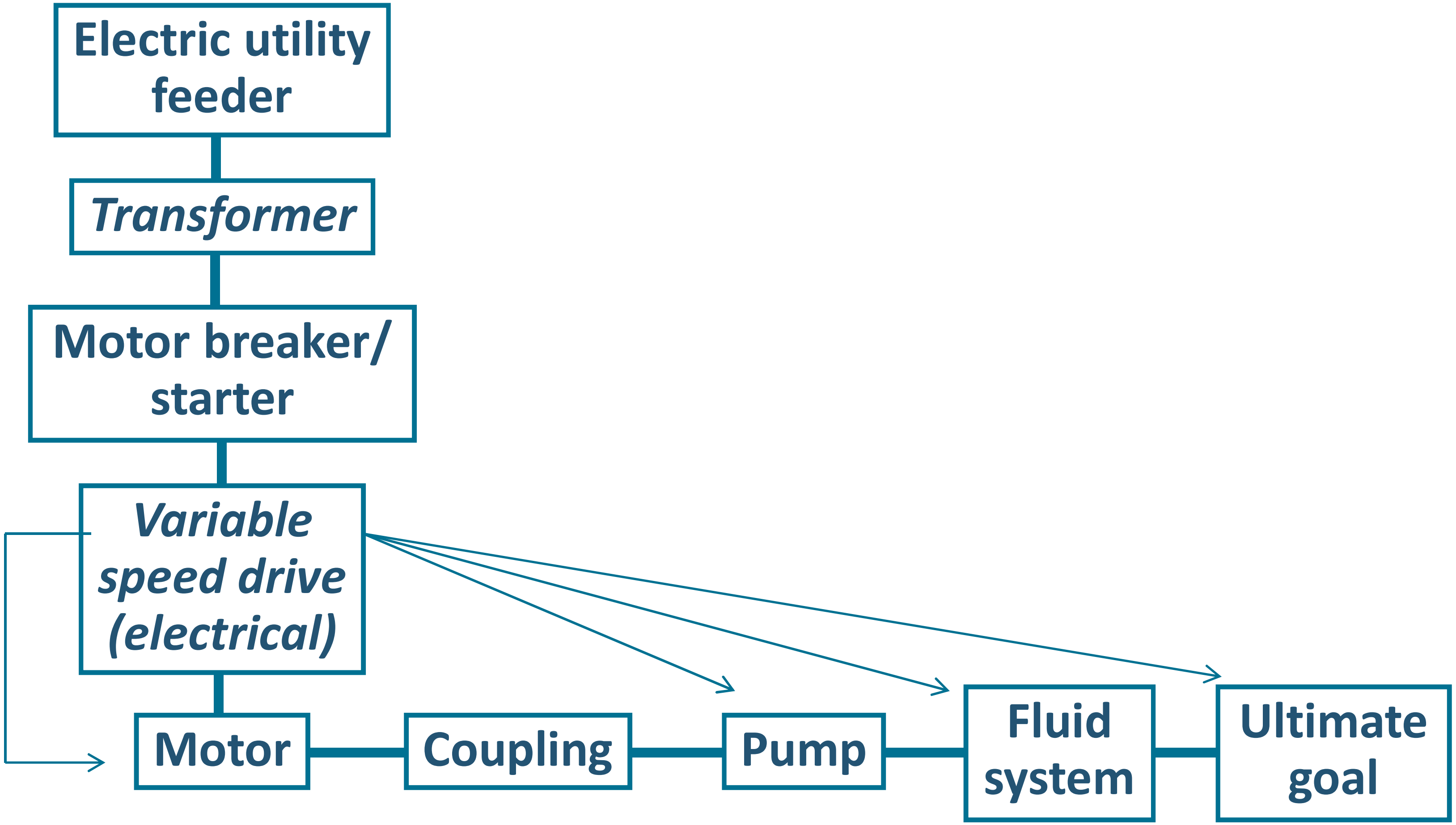
- A **Life Cycle Cost (LCC)** analysis could be used to find the best alternative – instead of the **Lowest Delivered Cost**.

Role of a VSD

- The true value of a VSD is the ability to precisely match motor and pump output to process requirements.
- Potential benefits of precise process speed control:
    - Improved product quality
    - Improved process throughput
    - Improved process control
    - Energy savings

Role of a VSD

The VSD will have an impact on the function of several elements


Electric utility feeder

Transformer

Motor breaker/ starter

Variable speed drive (electrical)

Motor

Coupling

Pump

Fluid system

Ultimate goal

Slide Courtesy of Oak Ridge National Laboratory

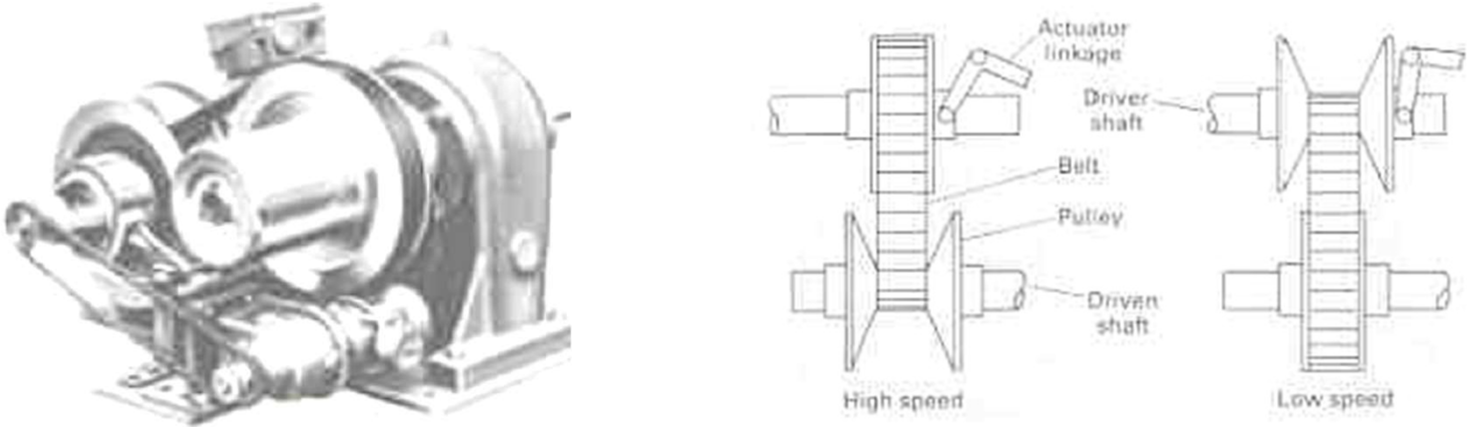
Different Variable Speed Control Devices

- Pulse Width Modulated VSD (PWM)
- Magnetic Coupling
- Mechanical Drive

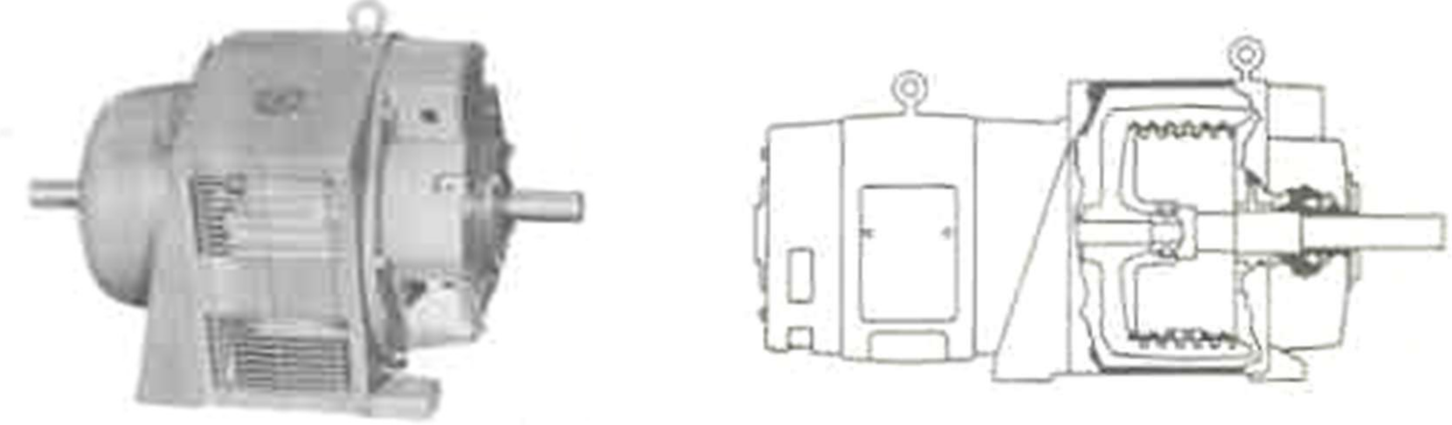
Other types:

- DC Drives
- Variable Voltage Inverter (VVI)
- Current Source Inverter (CSI)

Different Variable Speed Control Devices



Mechanical Belt Drive



Eddy Current Magnetic Clutch

Figures Courtesy of Hi-Lo Manufacturing and Eaton Drives

Unique behavior of VSD controlled induction motors

- Control of motor torque
- Control of motor speed
- Reduced starting current
- Improved efficiency over a range of operating conditions

VSD benefits

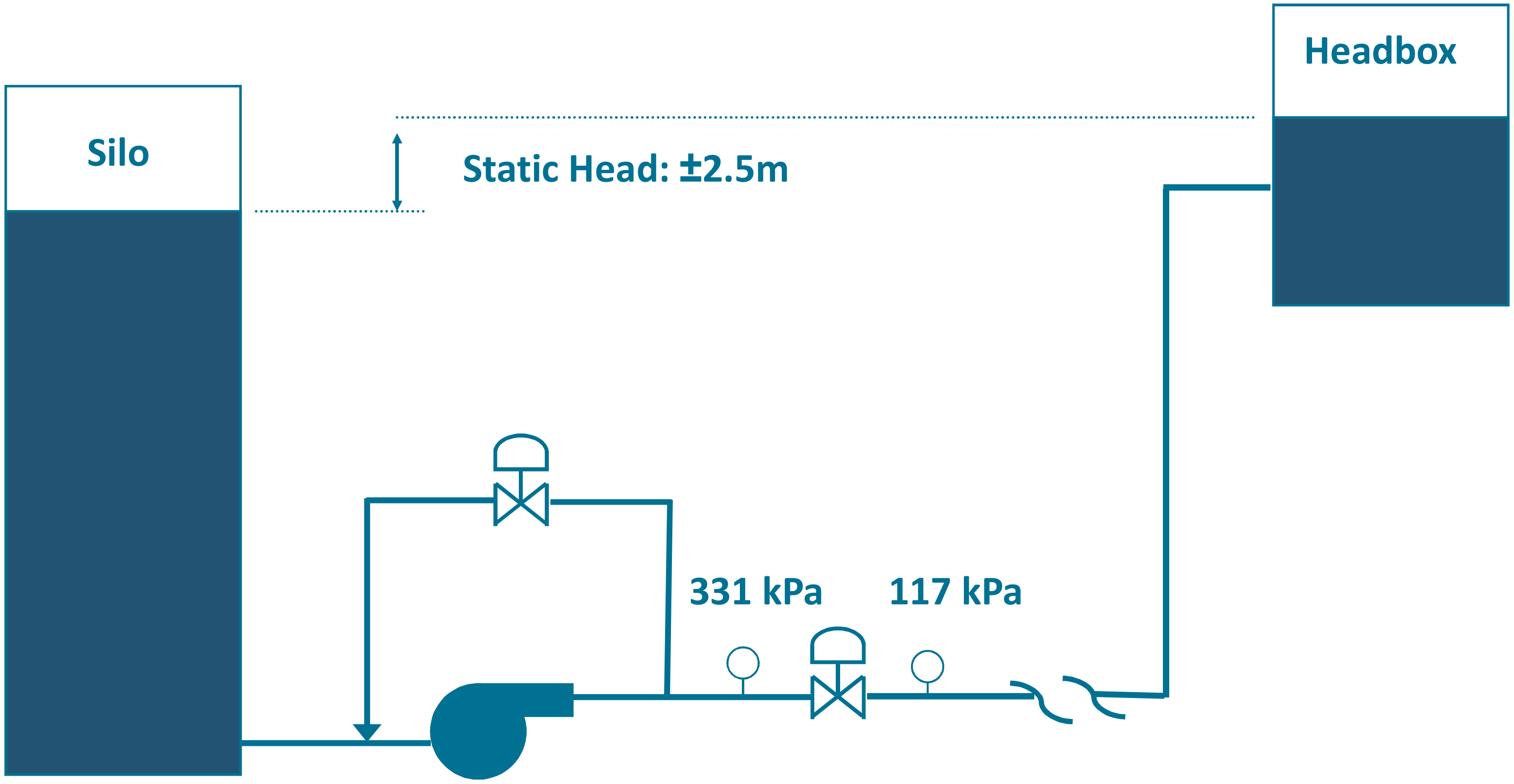
- Controls speed variations
- Provides mechanical control
- Eliminates startup impacts causing system vibration
- Provides fault tolerance
- Supports soft starts
- Restarts spinning load
- Controls speed swings
- Enhances product quality
- Can conserve energy in some systems
- Improves power factor (with active front-end)

Potential VSD issues

- Static head considerations
- Harmonics could effect instrumentation
- Fault-out (equipment shut-down) when power quality varies
- Bearing currents
- Mechanical vibrations
- Increased noise (acoustical)
- May need to include a full voltage starter as a bypass control

VSD Pulp and Paper Application at Paper Mill

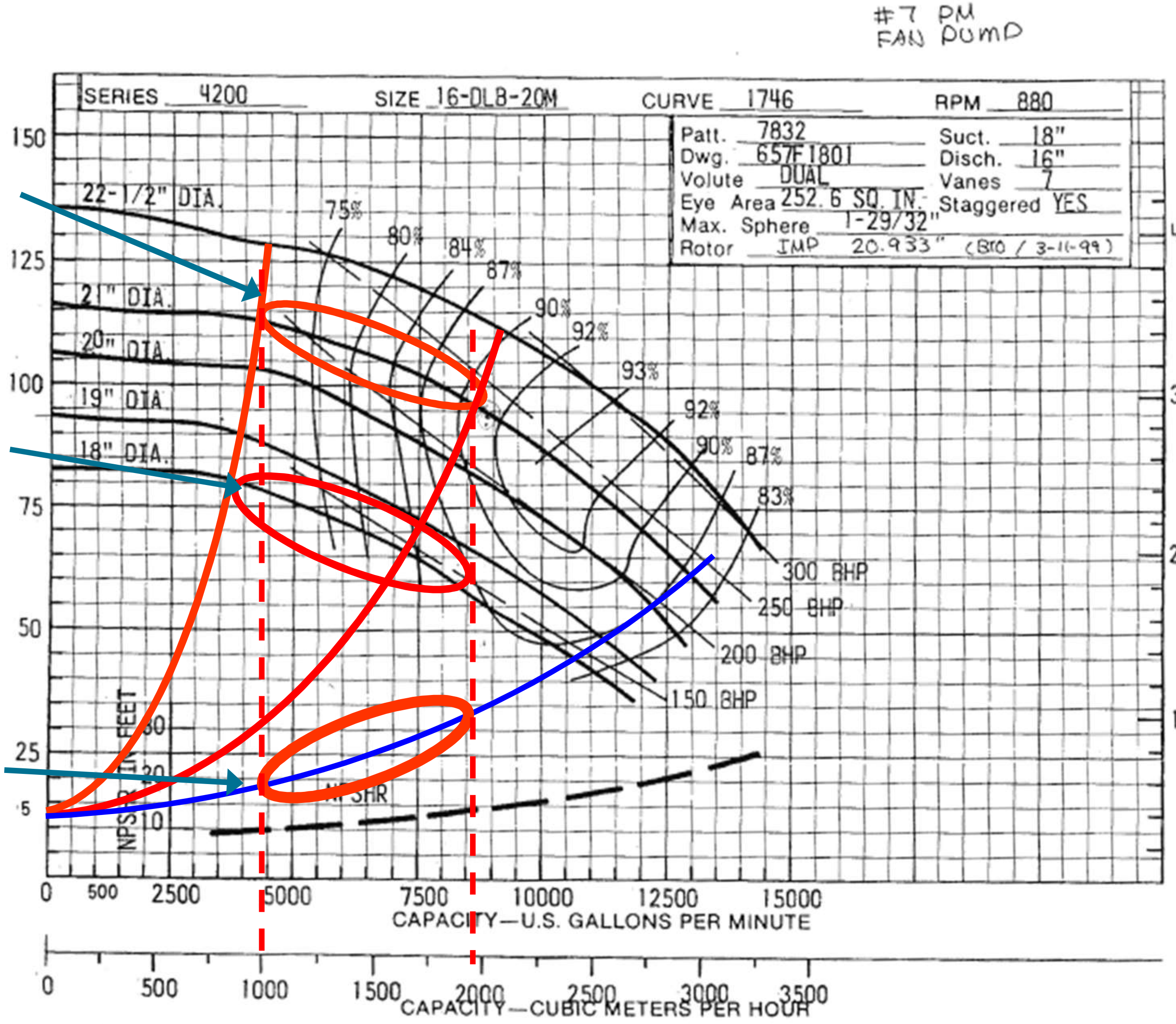
Overview of Pump System



225 kW Pump (Split case, double volute type)

Flow Range: (1 022 to 1 930 m³/hr)

Pump Curve



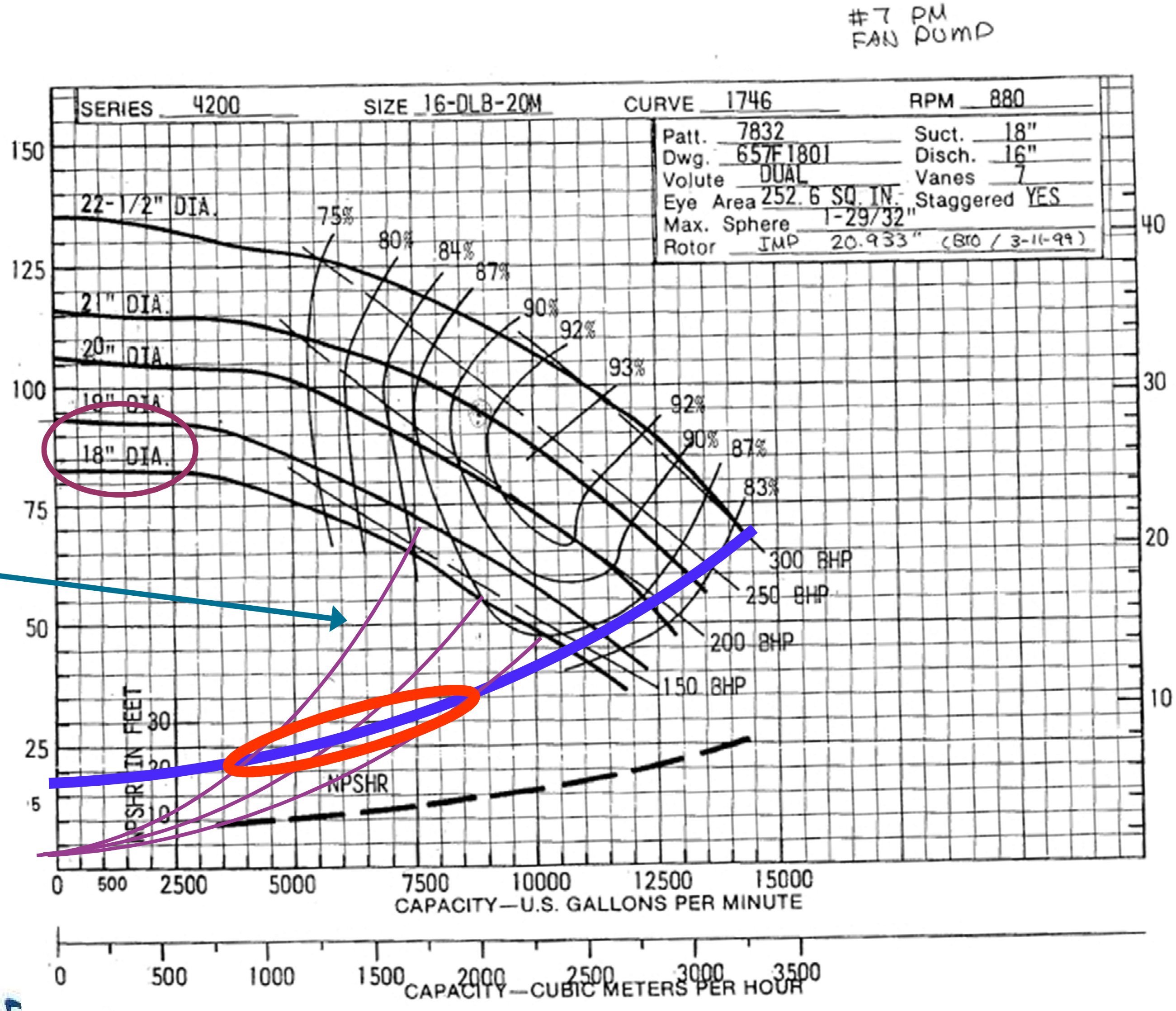
Using control valve for flow range of 1022 to 1930 m³/hr

Trim impeller and use control valve to adjust flow

Using variable speed drive for flow range of 1022 to 1930 m³/hr (with control valve open)

Pump Curve

Efficiency Curves using VSD with 18" impeller



Savings Analysis

Interval	Hours	Flow (m^3/hr)	TDH (m)	Pump Eff. (%)	VSD TDH (m)	Trimmed Pump Eff. (%)	AC Drive Eff. (%)	Trimmed Impeller TDH (m)
1	1 758	1 022	34	70	6.4	70	90	22.9
2	2 628	1 249	33	75	7.7	74	91	22.2
3	876	1 476	32	79	8.8	80	92	21.3
4	3 504	1 930	20	90	11.6	86	92	17.4

Impeller Trim to 18”			
Interval	**kW Existing***	**kW Proposed***	**Savings**
1	142	96	80 868
2	158	108	131 400
3	173	113	52 560
4	182	112	245 280
Total kWh Saving			**510 108**
Annual Cost Savings (EGP 0.90/kWh)			**EGP 459 097**

AC PWM VSD			
Interval	**kW Existing***	**kW Proposed***	**Savings**
1	142	25	205 686
2	158	36	320 616
3	173	47	110 376
4	182	80	357 408
Total kWh Saving			**994 086**
Annual Cost Savings (EGP0.90/kWh)			**EGP 894 677**

* Including assumed motor efficiency of 95%

Pump Recommendations

- Verify Efficiency, Flow and calculated kW data
- Evaluate control variation of each option
- Perform detailed cost estimates

	AC Drive	**Impeller Trim**
Energy Savings per year:	EGP 894 677	EGP 459 097
Estimated Project Cost:	EGP 3 600 000	EGP 230 000
Simple Payback:	4 years	6 months

NPV, IRR?

17. Collect Data & Field Measurements

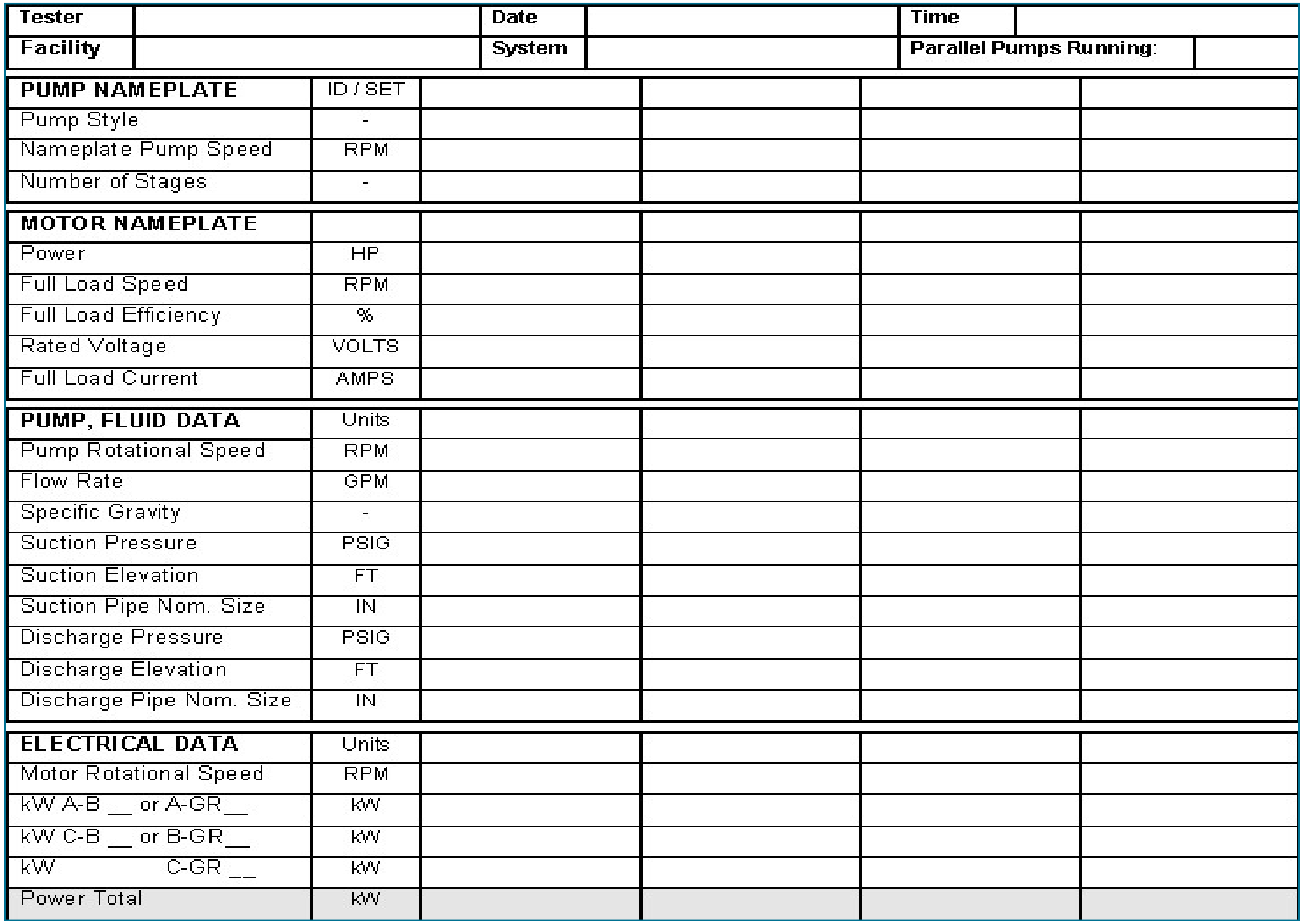
Pump Systems Optimisation (PSO) Expert Level Training
(Egypt Edition – Dec 2021)

Albert Williams
Siraj Williams

Collection of Equipment and Fluid Data

- **Driver information** *(the ASME standard focuses on motor-driven pumps)*
    - Motor nameplate: Type, Voltage, Frequency, Full Load Amps, rated Power, Speed, Efficiency, Power Factor, Service Factor.
- **Pump**
    - Type, Number of stages, Speed, Flow and Head design point, Impeller diameter, Pump curve, Maintenance records, presence of Cavitation.
- **Fluid Properties**
    - Temperature, Viscosity, Density or Specific Gravity, presence of Solids

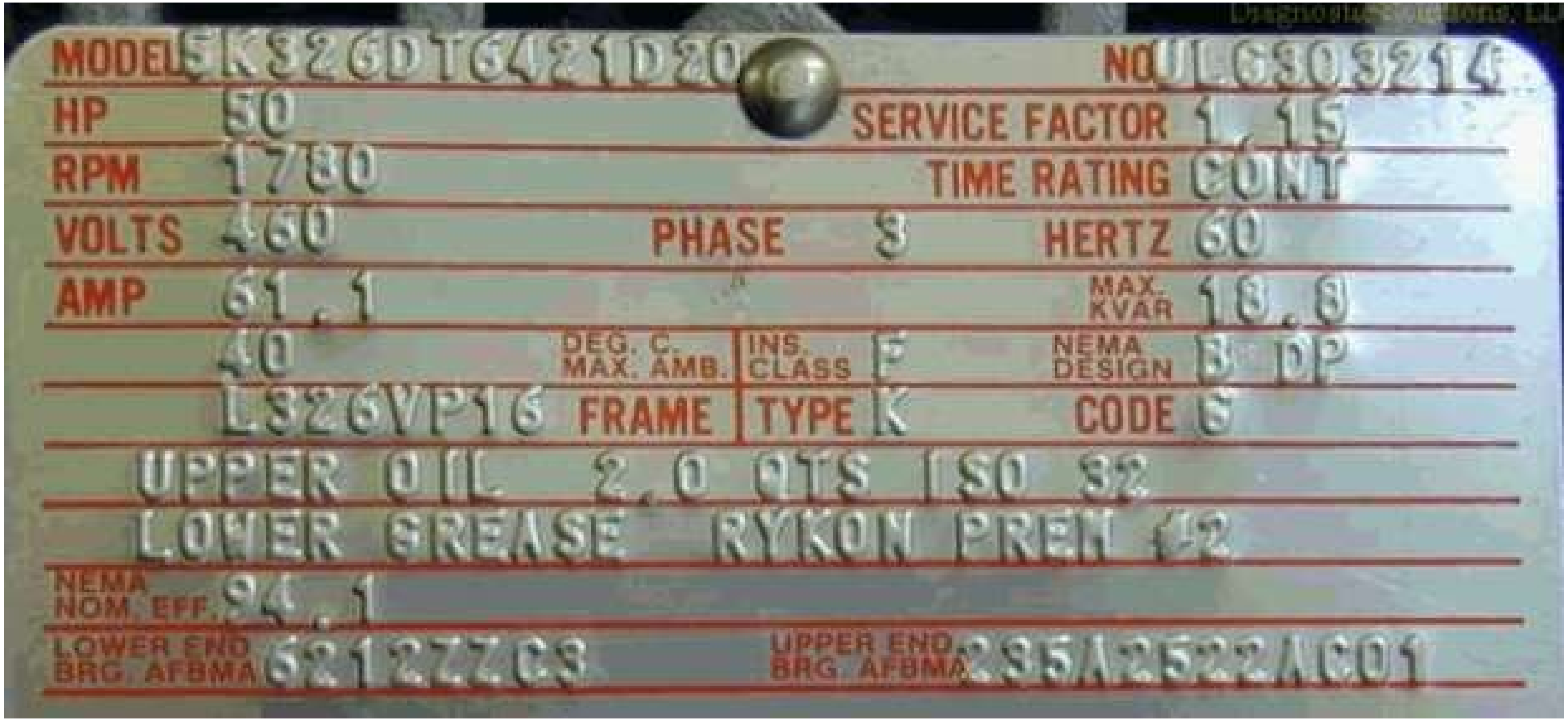
Example of Equipment Data Collection Form



Tester		**Date**		**Time**	
Facility		**System**		**Parallel Pumps Running:**	

PUMP NAMEPLATE	ID / SET				
Pump Style	-				
Nameplate Pump Speed	RPM				
Number of Stages	-				
MOTOR NAMEPLATE					
Power	HP				
Full Load Speed	RPM				
Full Load Efficiency	%				
Rated Voltage	VOLTS				
Full Load Current	AMPS				
PUMP, FLUID DATA	Units				
Pump Rotational Speed	RPM				
Flow Rate	GPM				
Specific Gravity	-				
Suction Pressure	PSIG				
Suction Elevation	FT				
Suction Pipe Nom. Size	IN				
Discharge Pressure	PSIG				
Discharge Elevation	FT				
Discharge Pipe Nom. Size	IN				
ELECTRICAL DATA	Units				
Motor Rotational Speed	RPM				
kW A-B __ or A-GR__	kW				
kW C-B __ or B-GR__	kW				
kW C-GR __	kW				
Power Total	kW				

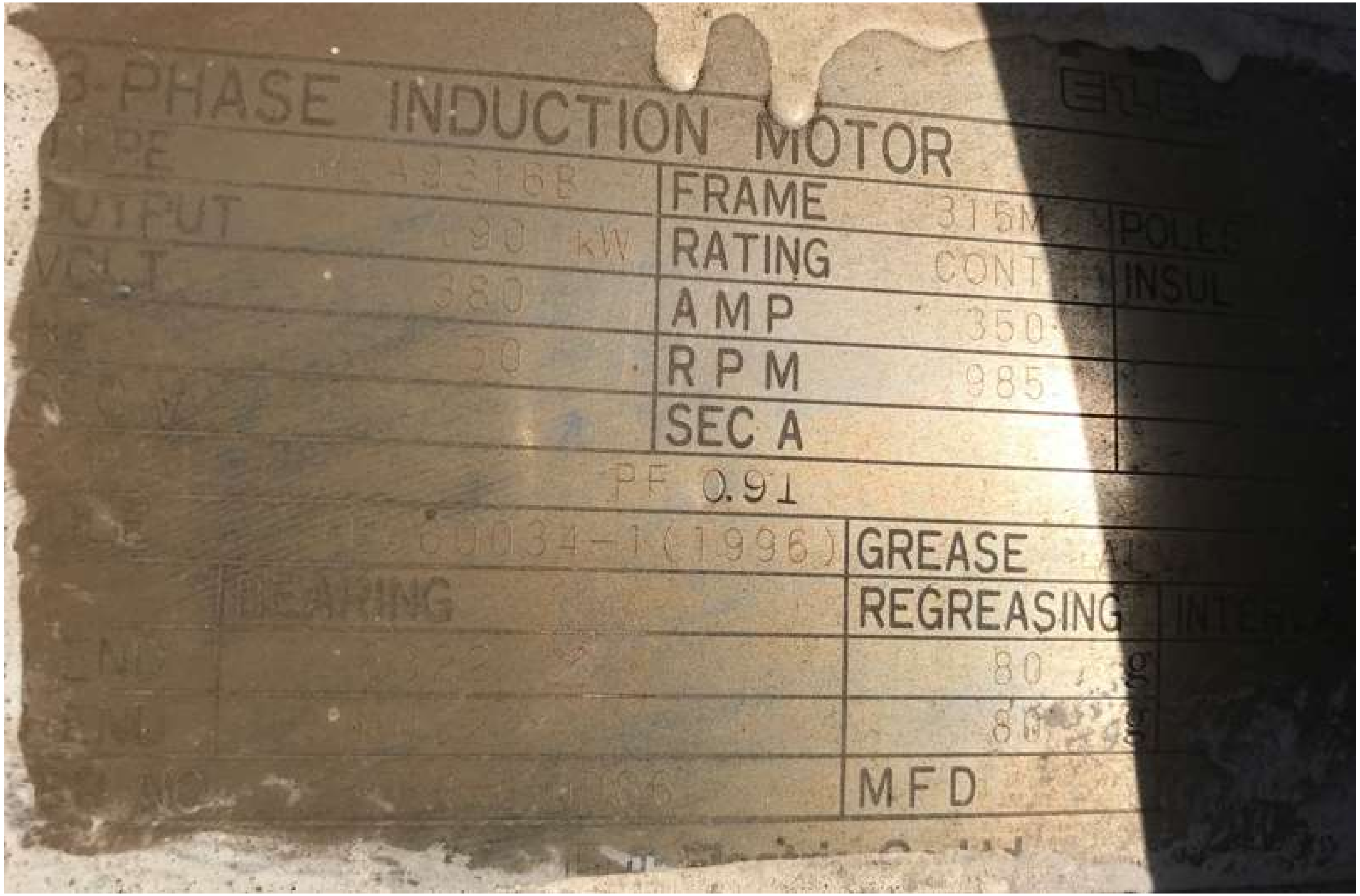
A typical motor nameplate



MODEL 5K326DT6421D20 NO UL6303214
HP 50 SERVICE FACTOR 1.15
RPM 1780 TIME RATING CONT
VOLTS 460 PHASE 3 HERTZ 60
AMP 61.1 MAX. KVAR 18.8
40 DEG. C. MAX. AMB. | INS. CLASS F NEMA DESIGN B DP
L326VP16 FRAME | TYPE K CODE G
UPPER OIL 2.0 QTS ISO 32
LOWER GREASE RYKON PREM #2
NEMA NOM. EFF. 94.1
LOWER END BRG. AFBMA 6212ZZC3 UPPER END BRG. AFBMA 285A2522AC01

Slide Courtesy of Oak Ridge National Laboratory

A typical motor nameplate



Slide Courtesy of Oak Ridge National Laboratory

Pump nameplate data

EUT 1906F

TORISHIMA PUMP

CDM450X OHN

RODUCT No AP028151 No. 6-6

OTAL HEAD 35 m SPEED 980 min

CAPACITY 1420 m³/H DRIVER 190 kW

BRG. No. 1 PIECES

6318 1 PIECES

ELIVERY TIME OCT 1998

Slide Courtesy of Oak Ridge National Laboratory

Pump nameplate data

BOWL	10M55	DATE	7 26 96
IMPELLER	10M 55	RPM	1800
TRIM	2 7.213	H.P.	50
TDH	276	GPM.	500
SER.NO.	014203	STAGES	6

Nameplate speed here (1800 rpm) is NOT consistent with flow rate and head, it is the ***nominal synchronous speed***

Slide Courtesy of Oak Ridge National Laboratory

Next… get a copy of the pump curve

Three types of pump curves:

- **Generic** curve for pump model - usually from a manufacturers catalog
- **Certified** factory curve – where the pump was tested at the factory
- **Field certified** curve – where the pump was tested after installed in the field.

Getting a certified factory test curve for the specific pump you're buying should be encouraged as a standard practice for pumps above 50 kW; a field certified curve should be pursued for pumps above 150 kW

Pump Curve with Impeller Trims

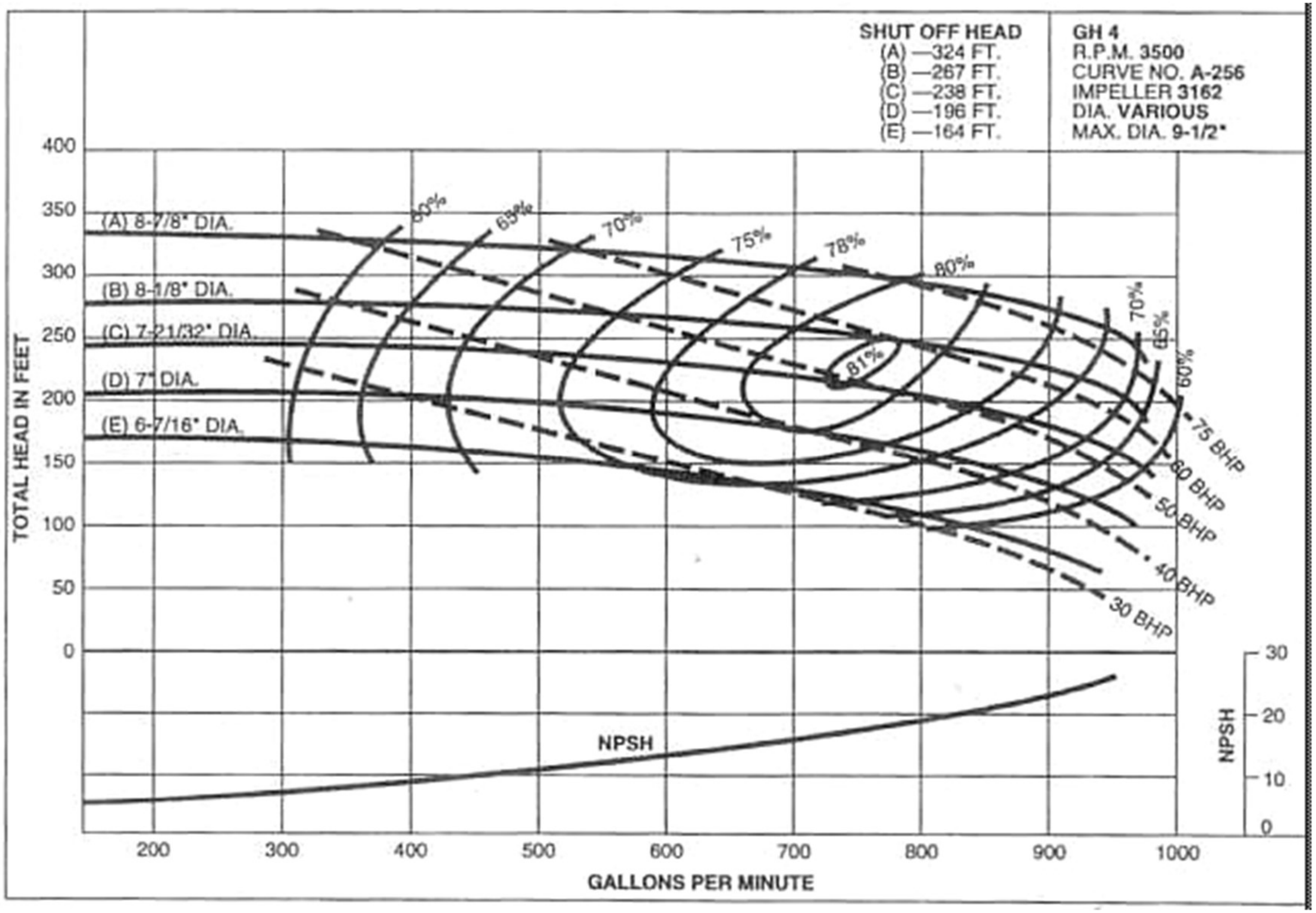


Figure Courtesy of ACR Publications

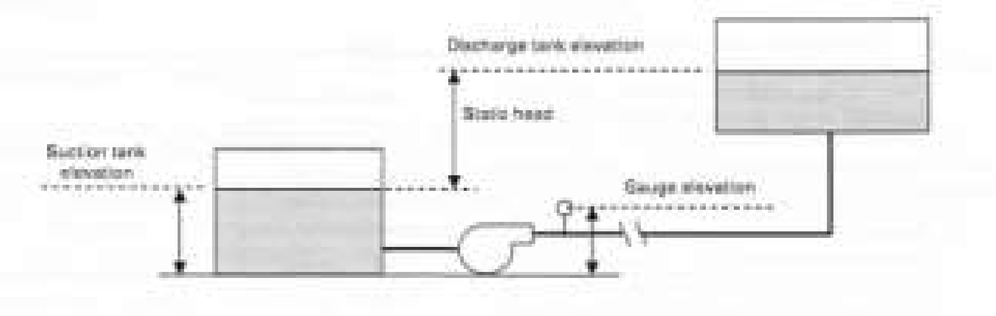
Develop a simplified flow diagram

- **Capture the Critical elements of the system**
- **How do you do that?**
  - Review P&ID and piping isometrics
  - Talk with operators
  - Walk the system down (nice to have a P&ID when you do)
  - Take notes !!

Slide Courtesy of Oak Ridge National Laboratory

Simple Drawing with Elevations

Fig. 14 Simple Pumping System Schematic



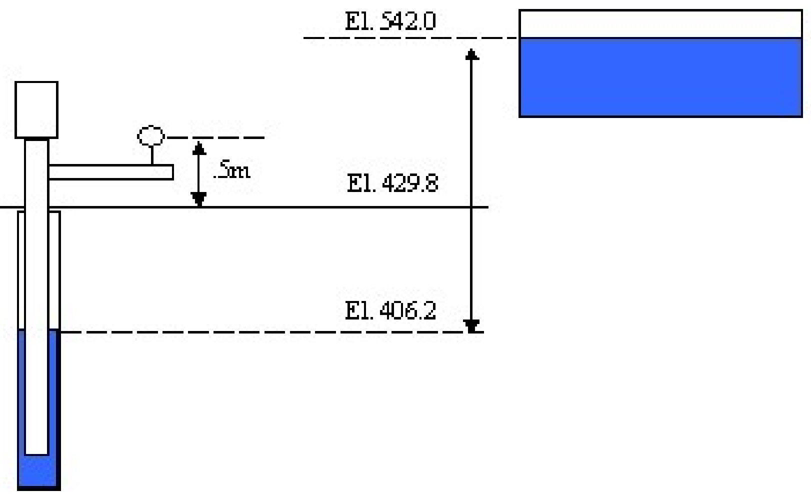
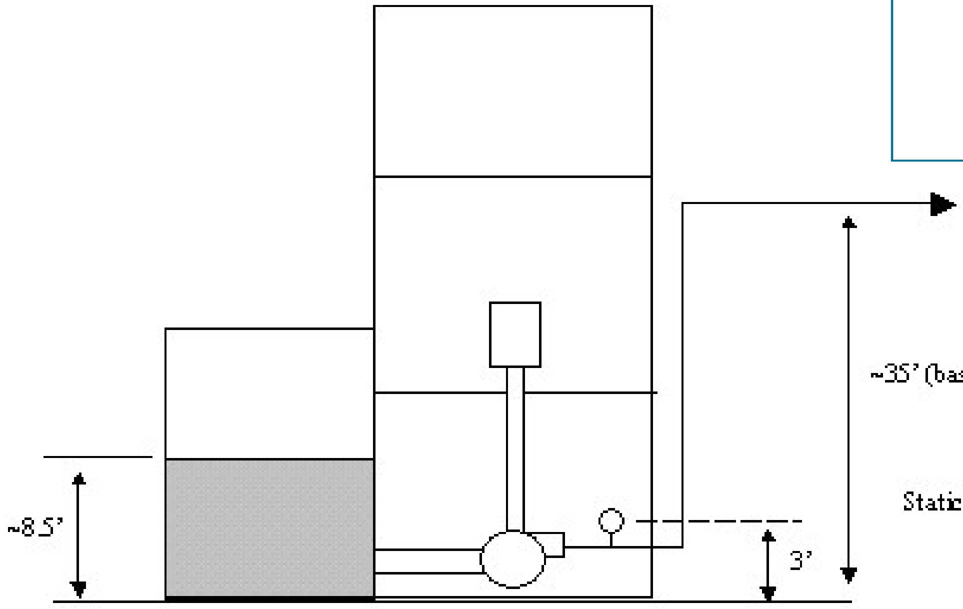


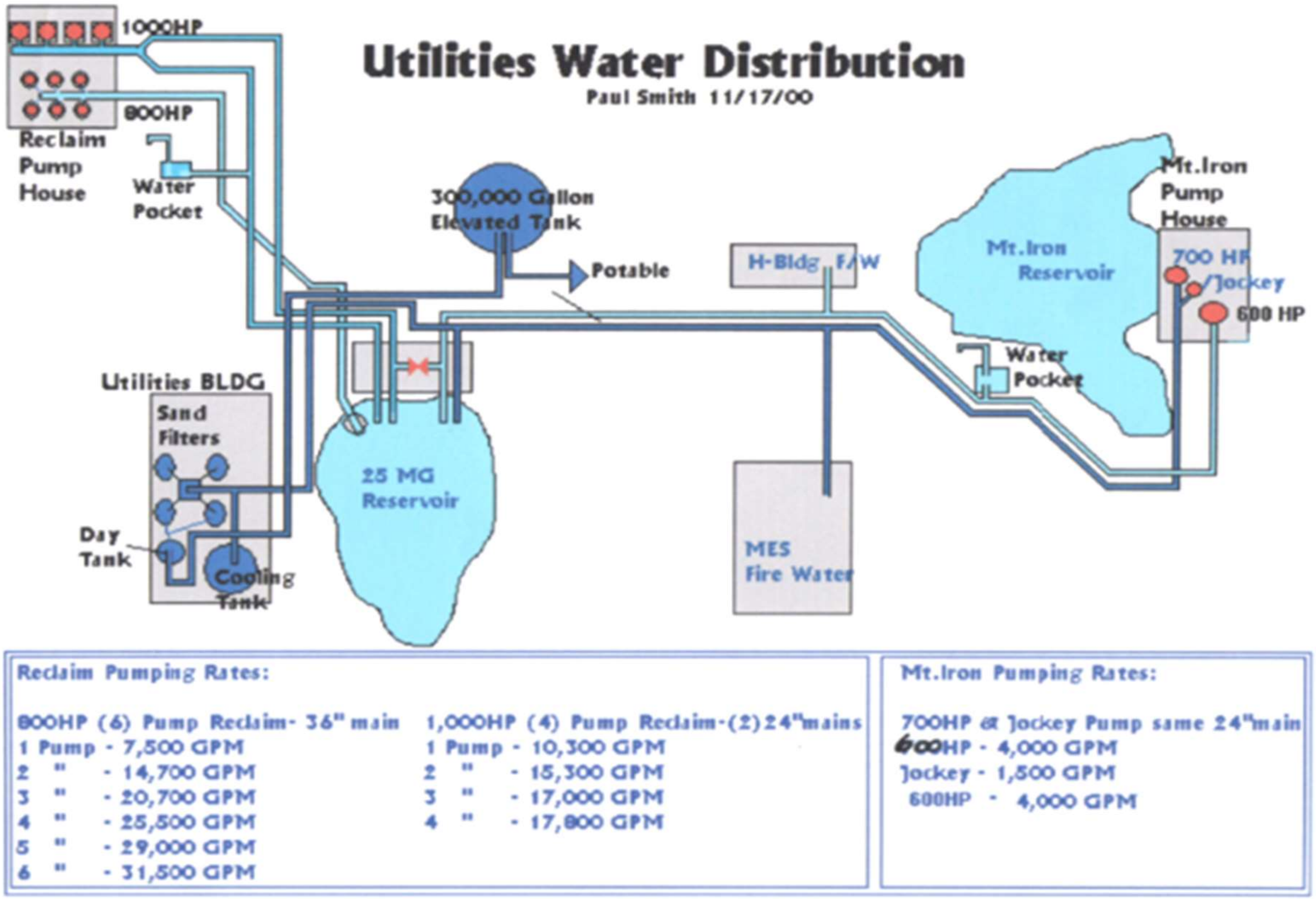
Figure 1: Pump Station Elevations



~35' (based on low flow pressure)

Static Head: 35' – 8.5' = 26.5'

Overall System Layout



Utilities Water Distribution

Paul Smith 11/17/00

Reclaim Pumping Rates:

800HP (6) Pump Reclaim- 36" main
1 Pump - 7,500 GPM
2 " - 14,700 GPM
3 " - 20,700 GPM
4 " - 25,500 GPM
5 " - 29,000 GPM
6 " - 31,500 GPM

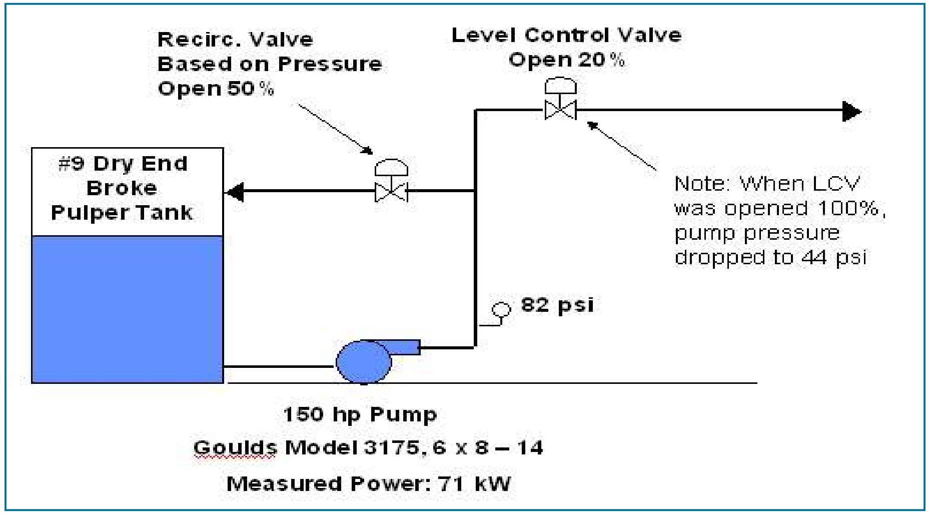
1,000HP (4) Pump Reclaim-(2)24"mains
1 Pump - 10,300 GPM
2 " - 15,300 GPM
3 " - 17,000 GPM
4 " - 17,800 GPM

Mt.Iron Pumping Rates:

700HP & Jockey Pump same 24"main
600HP - 4,000 GPM
Jockey - 1,500 GPM
600HP - 4,000 GPM

Collection of System Data

- **Data gathered using installed plant instrumentation or portable instruments:**
  - Motor power or voltage and current
  - Pump flow rate, suction and discharge pressure
  - Flow rates to system loads
  - Pressures at system loads
  - Fluid temperature, density, viscosity
- **Additional System Data:**
  - Static head
  - Operating hours
  - Pump control method:
    - VSD, Throttled valve
    - By-pass or recirculation, etc



Data Collection Tips

- Determine if data collected is a **representative** snapshot or if the system needs to be evaluated over a longer period of time or if historical process control data is available.
- Pressure measurements should be taken with calibrated, reliable gauges or transmitters.
- Flow measurements should be taken with properly installed, calibrated meters.
  - If using portable flow meters, confirm measurement at alternative locations
  - May use dP across a component and component curve

Data Collection Tips

- **Motor input power**
  - Preferably measure power directly with a power meter
  - MEASUR can calculate motor input power using measured voltage and current, and estimating the power factor
- **Cross-validation**
  - Flow rate, pressure, and power measurements may not be available but can be determined using cross-validation
  - Use dP pump differential pressure (total head) and pump curve to estimate flow rate
  - Use motor input power and efficiency to calculate shaft power, then use pump curve to estimate flow rate
  - Use valve position, flow rate, and K_v data to estimate dP
  - Measure drawdown and fill times to estimate flow rate

Primary parameters of interest

- Flow rate
- Pressure
- Elevations
- Electric power

Volumetric Flow Rate Measurement

Flow rate, velocity, and area relations

$$A = \pi \text{ x } \left(\frac{d}{2}\right)^2$$

Volumetric flow rate = Velocity x Area

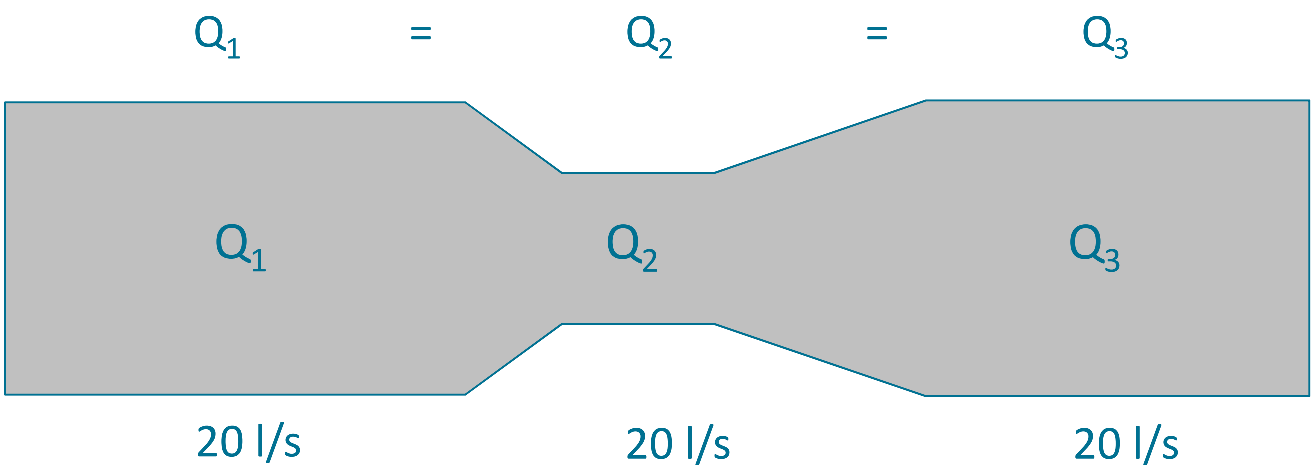
Q = v x A

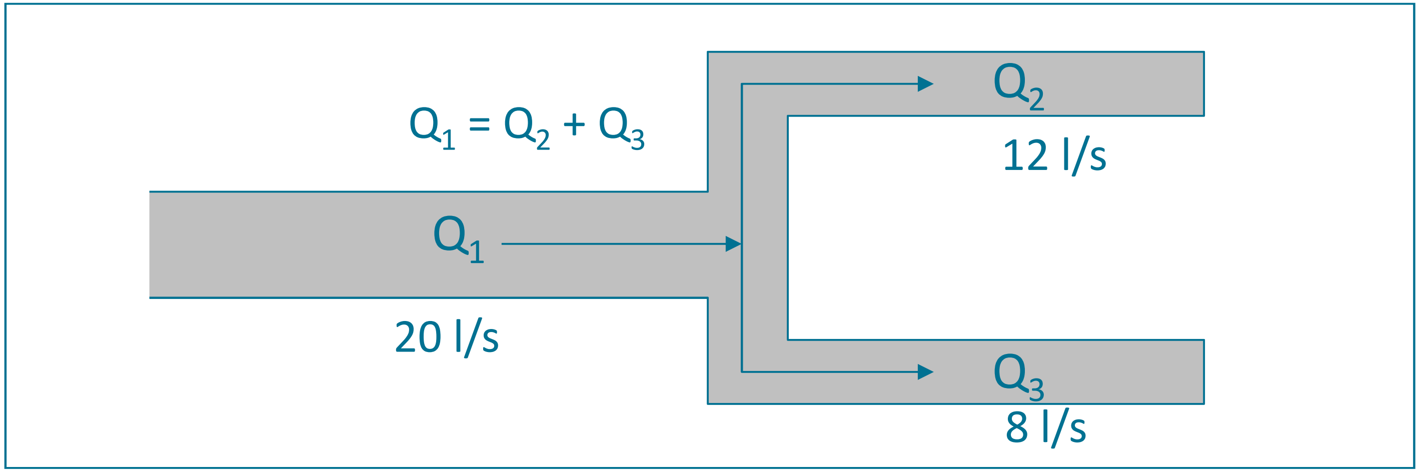
<u>Continuity</u>

$Q_1 = Q_2$

A practical principle for filled liquid piping systems: The flow rate at two points at any particular point in time will be equal

More on flow rate, continuity

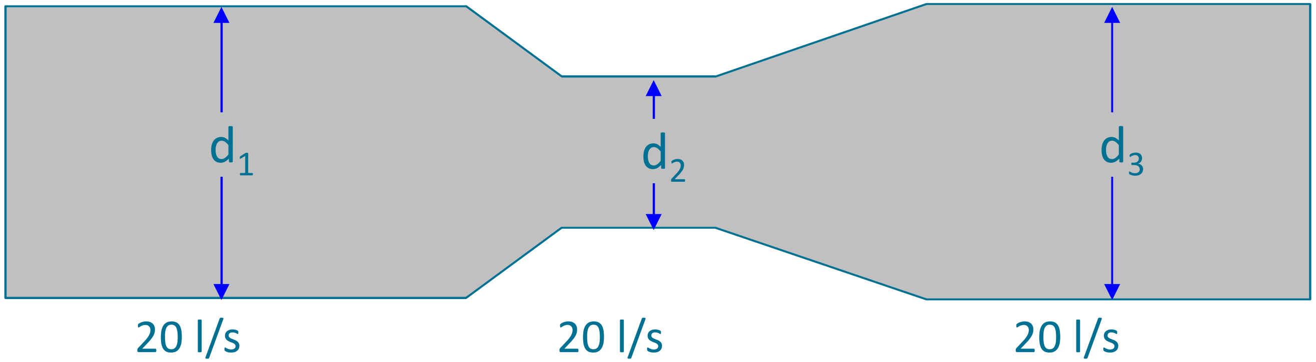




Velocities vary with diameter

$$Q_1 = Q_2 = Q_3$$

$$V_1A_1 = V_2A_2 = V_3A_3$$



$$A_1 = \pi \text{ x } \left(\frac{d_1}{2}\right)^2 \qquad A_2 = \pi \text{ x } \left(\frac{d_2}{2}\right)^2$$

$$V_1 = V_2\left(\frac{A_2}{A_1}\right) = V_2\left(\frac{d_2}{d_1}\right)^2$$

If d_1 is 2 x d_2, then V_2 will be 4 x V_1

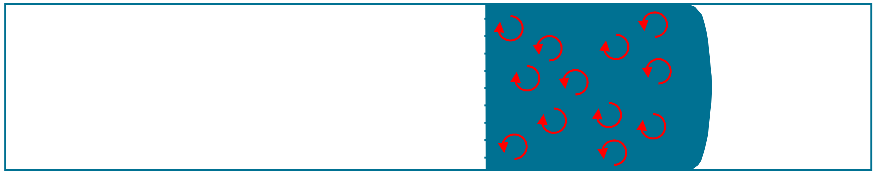
Flow meter considerations

- Proper flow profile and installation
- Range
- Calibration
- Wear
- Corrosion, scale, foreign material
- Sensing line issues (similar to pressure)

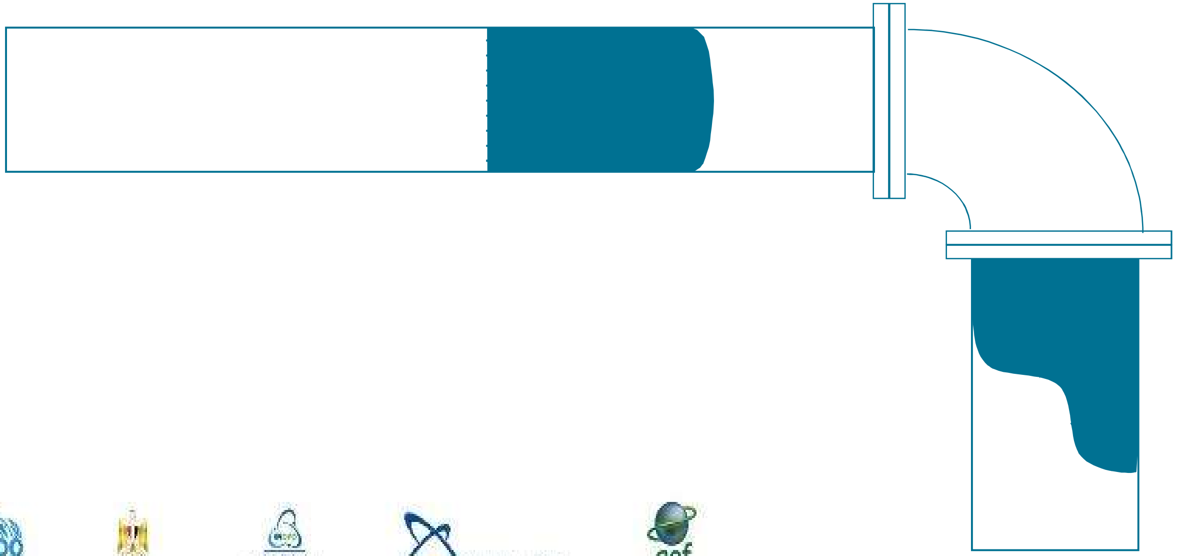
Slide Courtesy of Oak Ridge National Laboratory

The flow regime and upstream geometry affect the velocity profile

Laminar flow (very uncommon in significant energy-using pumping systems)



Turbulent flow (commonly the case in significant energy-using pumping systems)



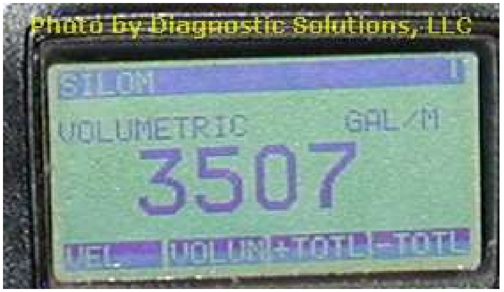
The flow profile can be significantly distorted after disturbances, such as elbows, valves, etc.

Permanently-installed ultrasonic flow meter in wastewater plant (single path)

This system measures the average velocity across the full pipe diameter; the transducers are in contact with the fluid.

Clamp-on portable ultrasonic

Ductile and cast iron (top), carbon and stainless steel pipe (bottom)



Portable Ultrasonic Flow Meter

Slide Courtesy of Oak Ridge National Laboratory

2-channel meter

For less-than-desirable geometric conditions, 2-channel meter provides a consistency check

```
CH1 VOLUMETRIC GAL/M
   36.53
CH2 VOLUMETRIC GAL/M
   36.52
```

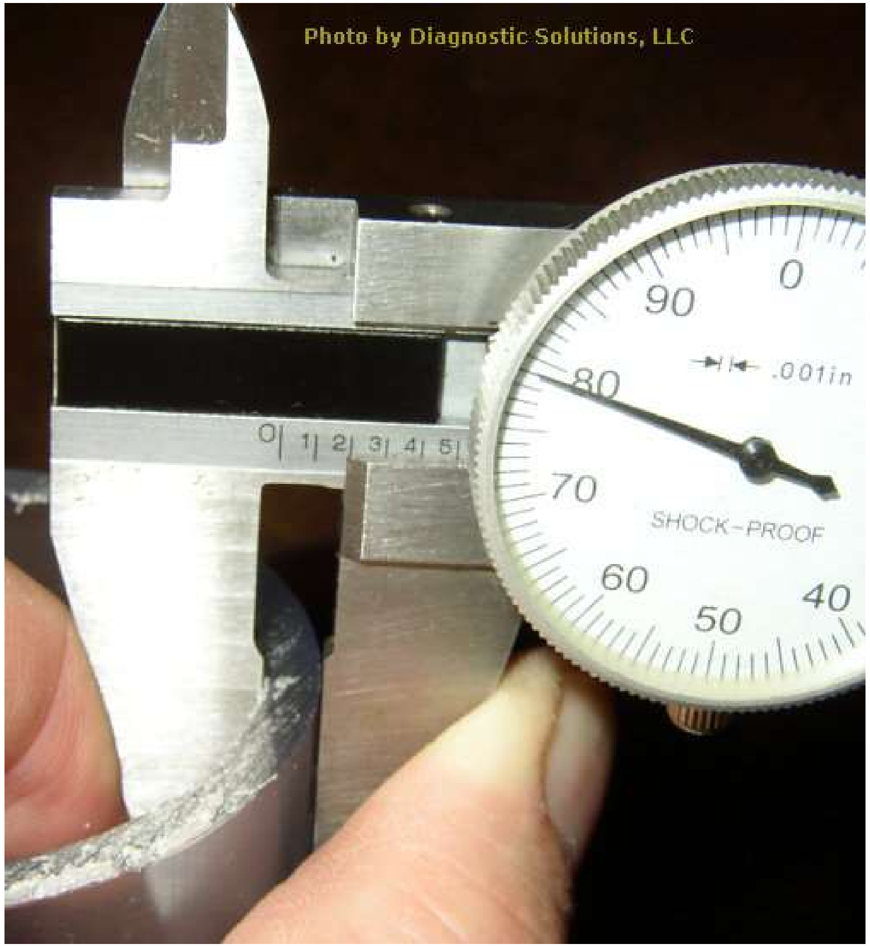
Photo by Diagnostic Solutions, LLC

Photo by Diagnostic Solutions, LLC

Note that the most upstream transducer is only about 2.25 pipe diameters downstream of the tee. The pair of ultrasonic units are set up about 90 degrees apart, circumferentially, thus sensing perpendicular velocity profiles.

Wall thickness

Wall thickness is a common source of uncertainty or error in all flow measurements

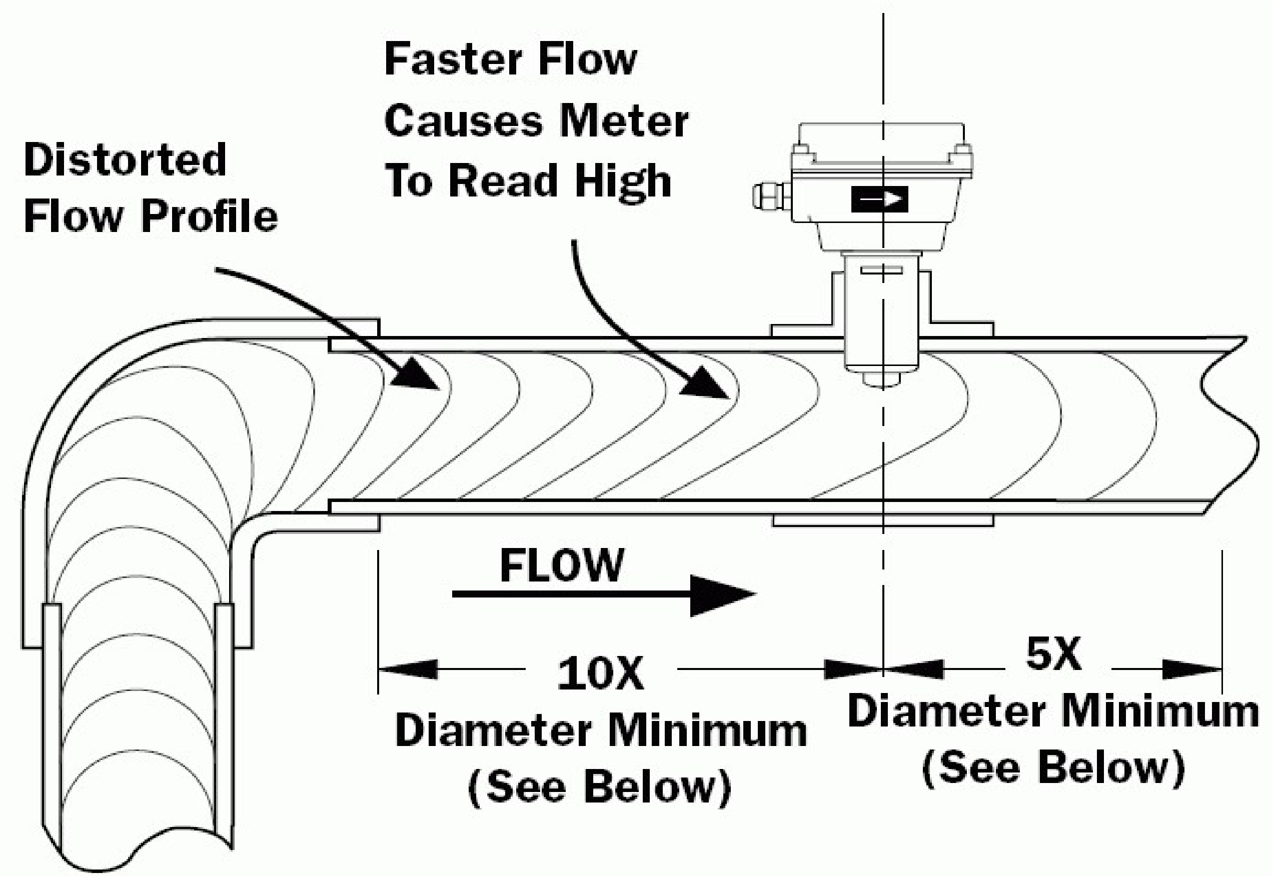


Problem applications for time of flight ultrasonic meters

- Slurries
- Medium/high density stock
- Aerated fluid
- Considerable scale buildup
- Good quality meters give the user an alert when the meter diagnostics suggest that the data is likely to be erroneous.
- Not all meters fit the "good quality" characterization.

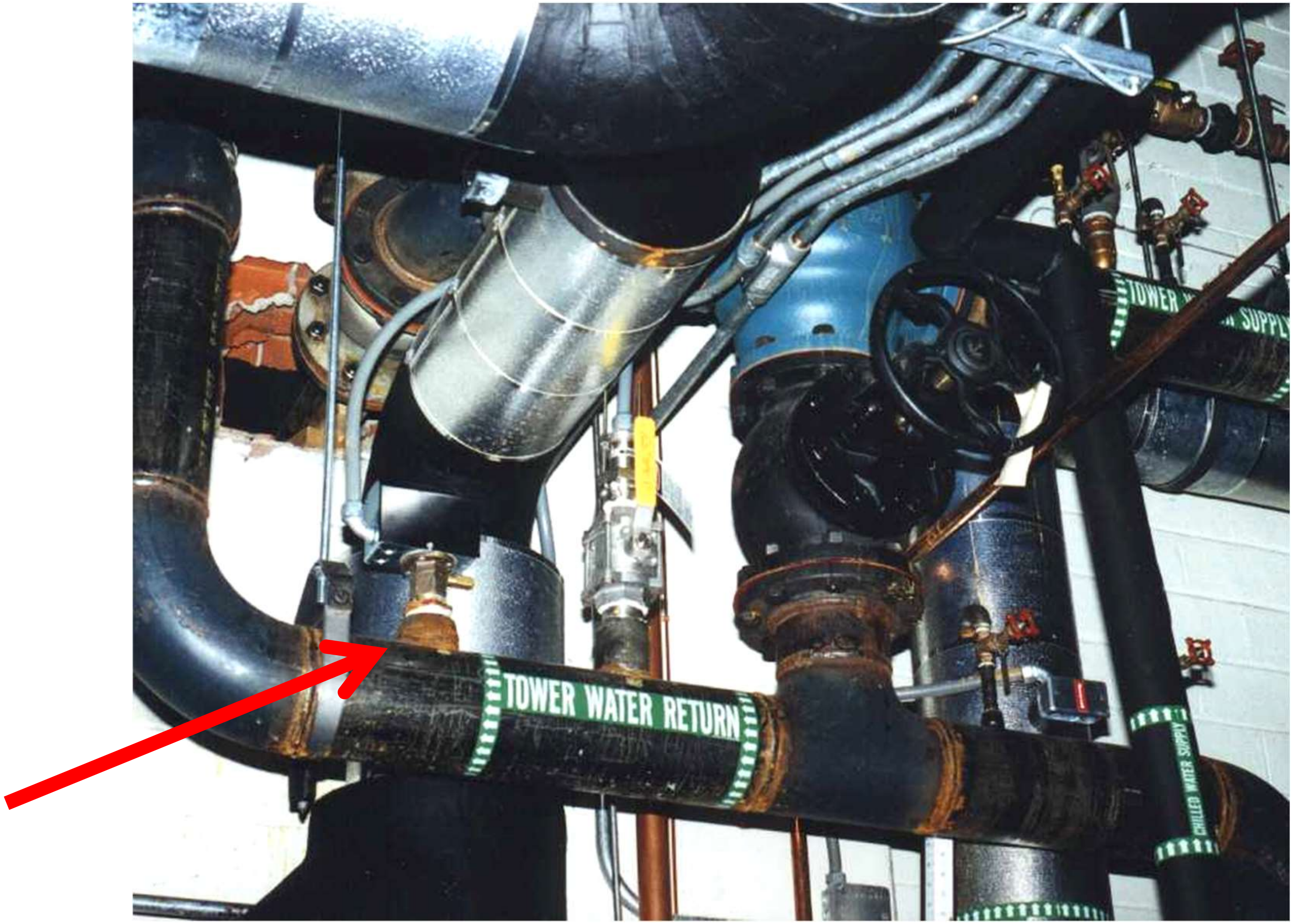
Single radial point transducers

Single radial point transducers are particularly susceptible to disturbed flow-induced errors



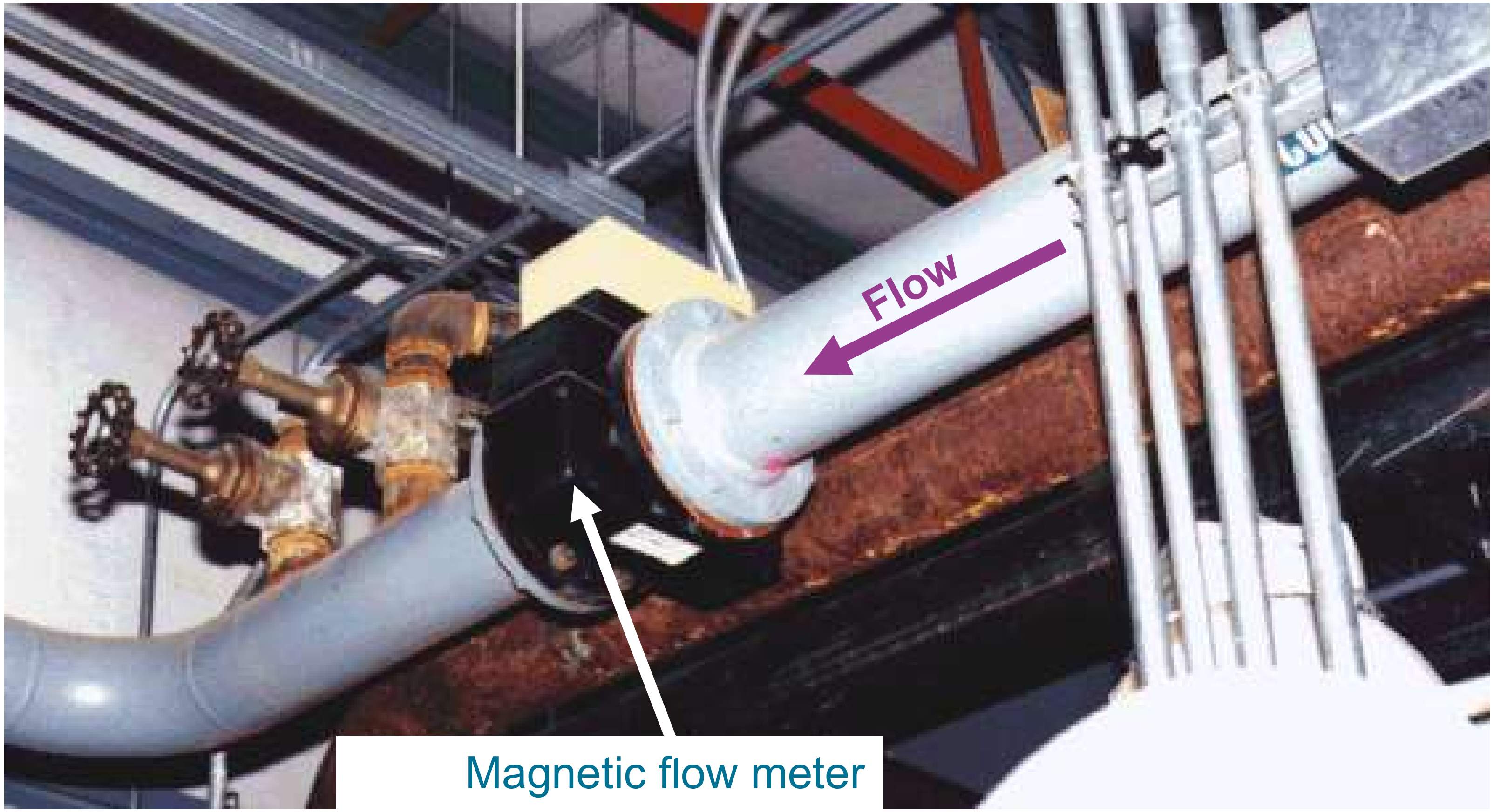
Source: SeaMetrics EX80 Series Electromagnetic Flow Sensor Instructions manual

Bad flow meter installation



A Better Configuration

(downstream conditions could be improved)



Magnetic flow meter

Slide Courtesy of Oak Ridge National Laboratory

Full diameter magnetic flow meter

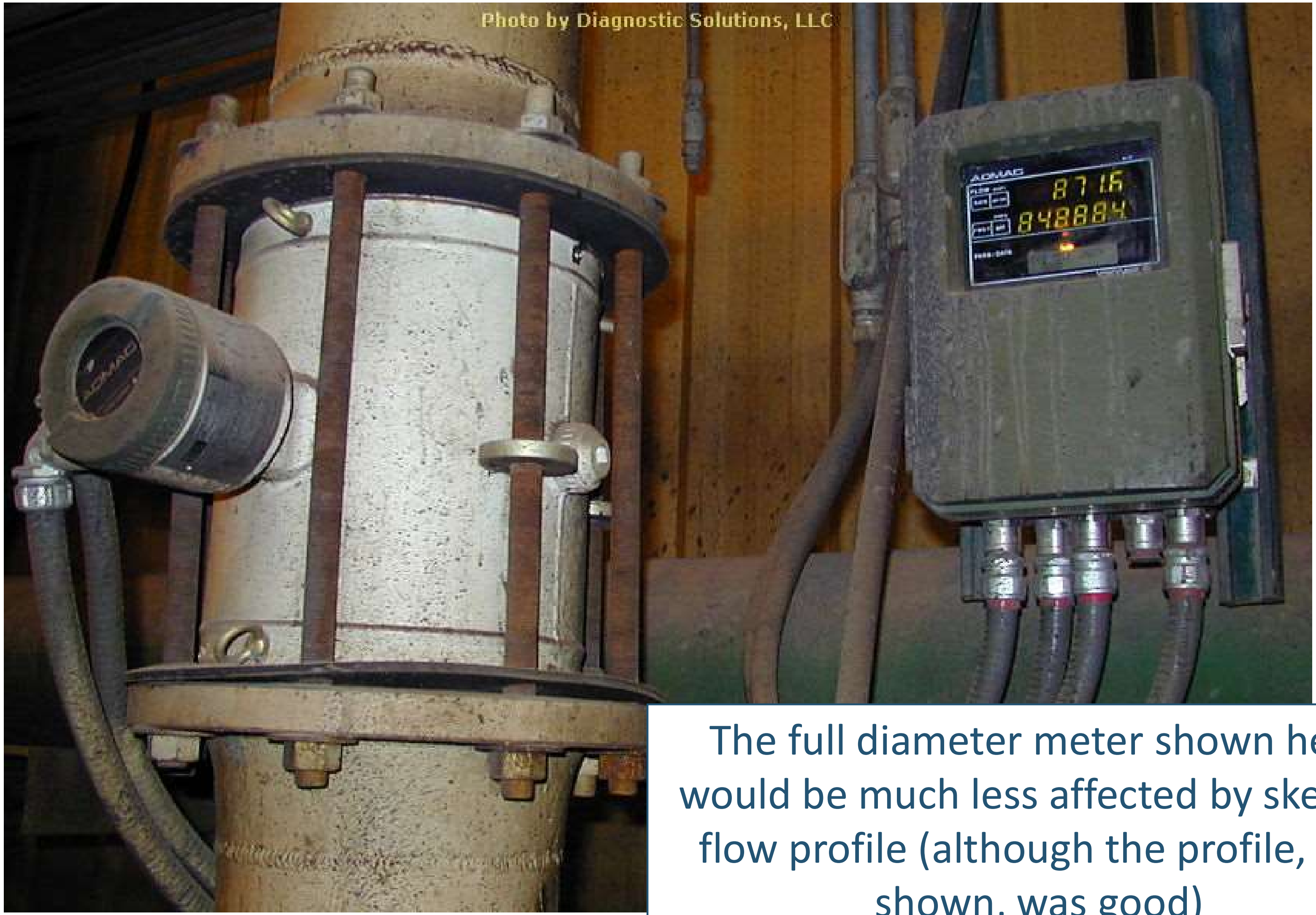


Photo by Diagnostic Solutions, LLC

The full diameter meter shown here would be much less affected by skewed flow profile (although the profile, not shown, was good)

Magnetic flow meter

Three magnetic flow meters used in a slurry application with decent pipe geometry

Pressure and Head Measurement

Pressure

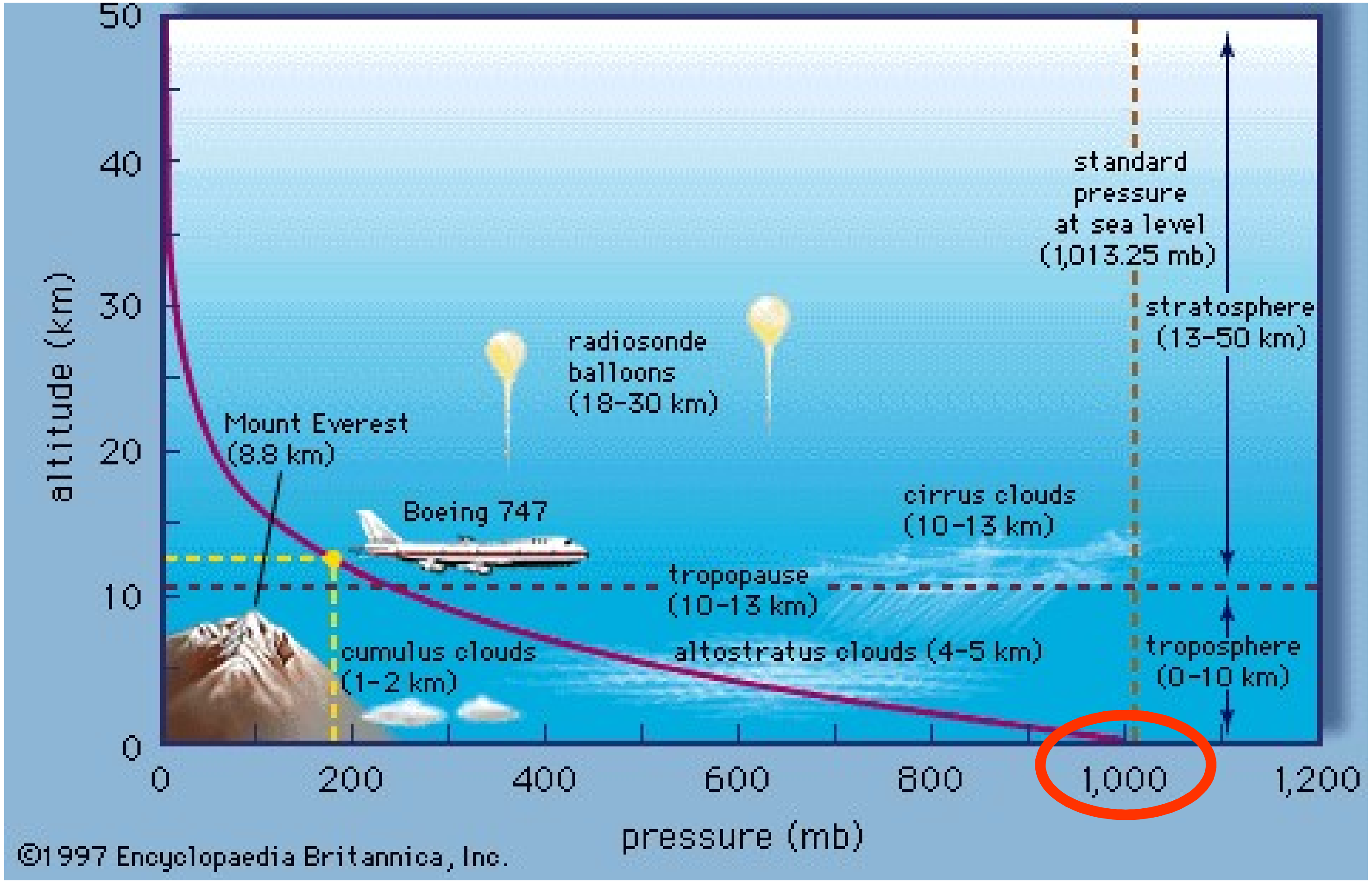
Pressure is normally measured relative to the local atmospheric condition

The SI-unit for pressure is kPa.

Imperial units are psig and inches of mercury vacuum

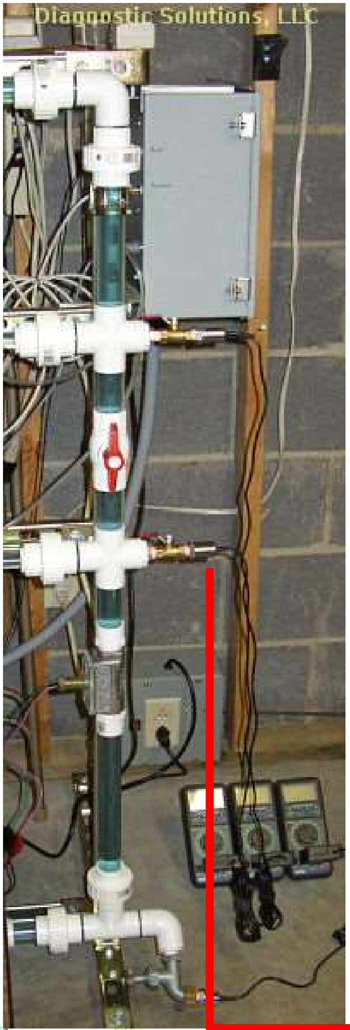
Abolute pressure

The absolute pressure in the atmosphere is a function of elevation



Gauge pressure

Gauge pressure is also a function of the gauge elevation



Diagnostic Solutions, LLC

Elevations of transducers in the riser at left (above floor)

1.08 m 0.77 m 0.06 m

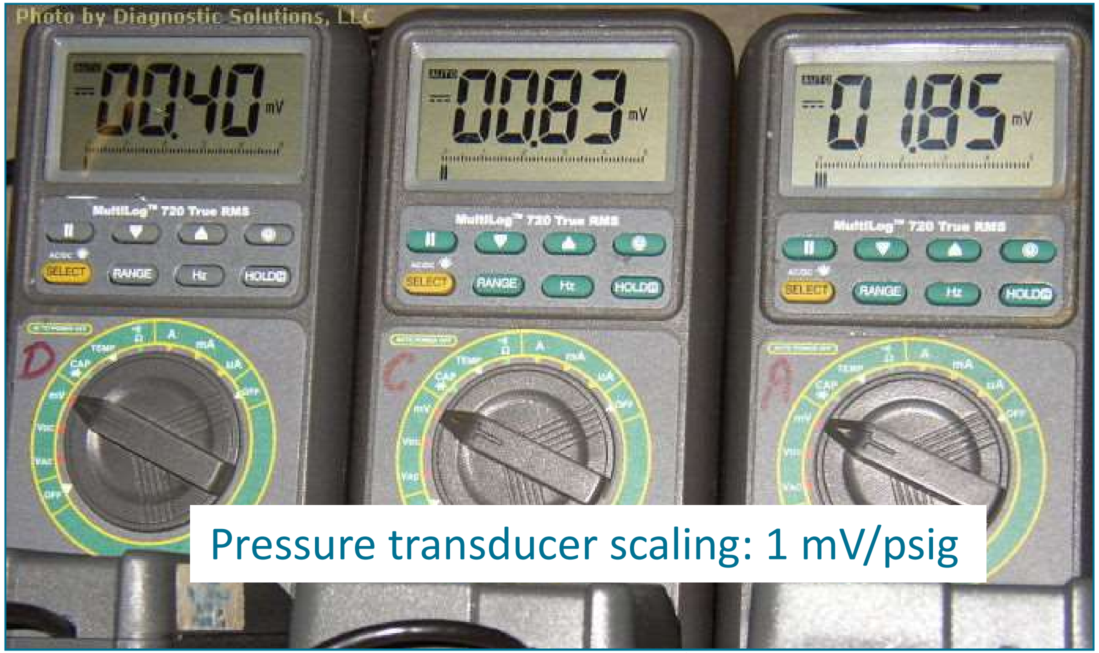
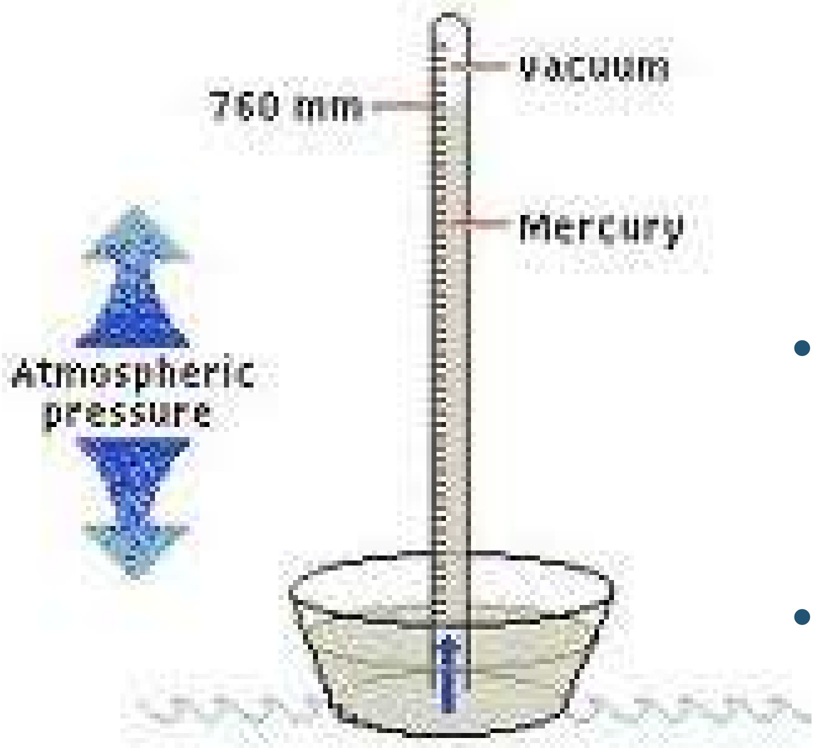


Photo by Diagnostic Solutions, LLC

Pressure transducer scaling: 1 mV/psig

Note:
Pump was off during this set of measurements

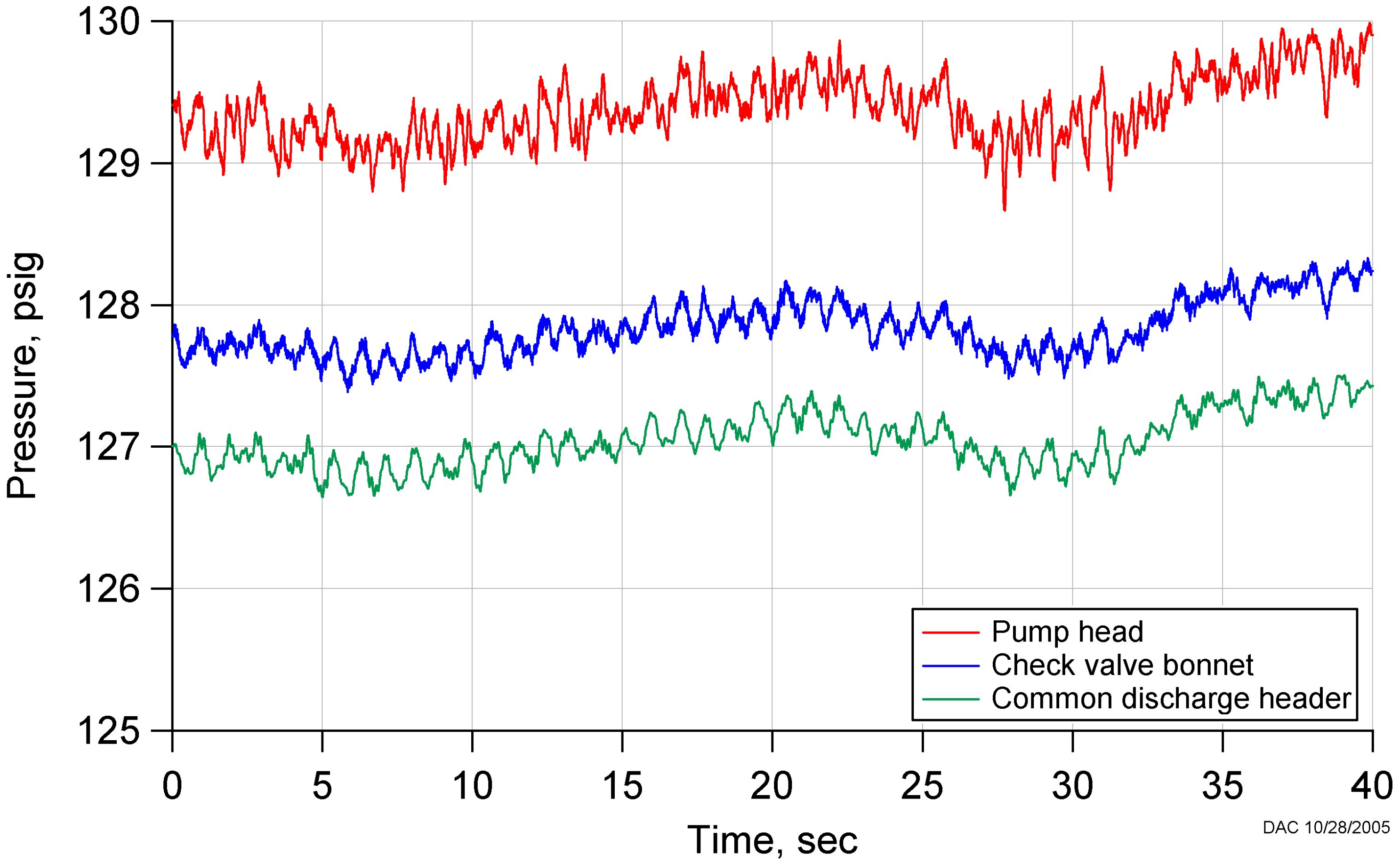
Gauge and Absolute Pressures



- **Average *sea-level pressure* is 101.325 kPa (1013.25 mbar, or hPa) or 760 millimeters (mmHg)** that is, the pressure of the air relative to a perfect vacuum
- Gauge pressure measurements are always relative to the ambient atmosphere
- Absolute pressure is an important factor in one pump performance attribute: NPSH

Gauge Pressure

Gauge pressure in pumping systems varies with location and time



DAC 10/28/2005

Some practical considerations

- **Service environment, history**
  - Water hammer
  - Calibration
- **Instrument range**
  - Accuracy
  - Overpressure capability
- **Physical location, setup**
  - Process connection point
  - Accounting for sensing element elevation
  - Proper instrument line fill & vent

Slide Courtesy of Oak Ridge National Laboratory

Common pressure transducers

The two most common pressure-measuring devices are the Bourdon tube and diaphragm-based strain gauge transducers

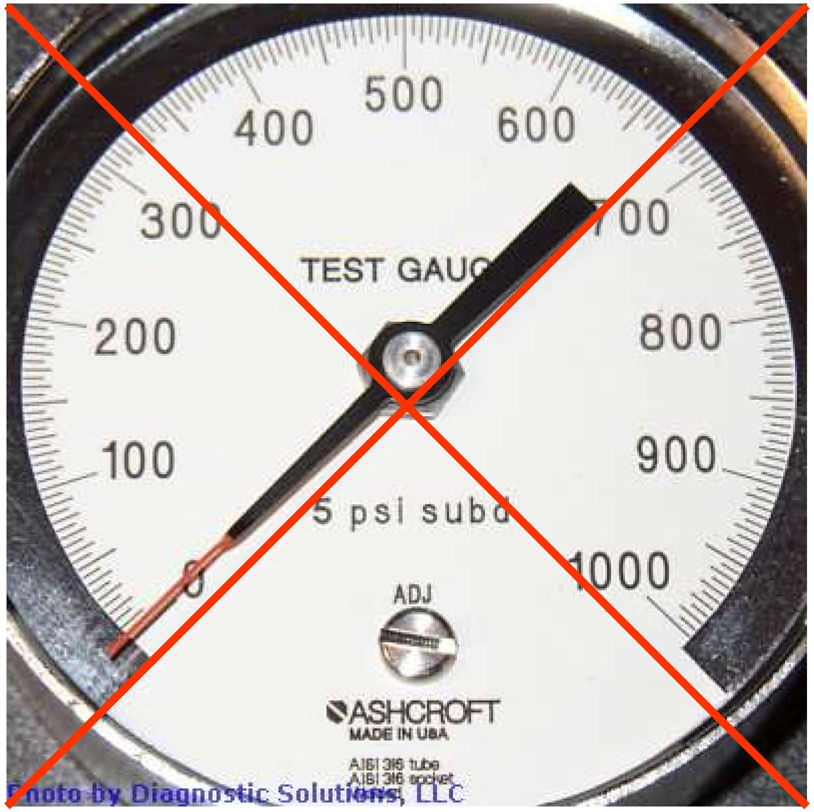


Photo by Diagnostic Solutions, LLC

Photo by Diagnostic Solutions, LLC

Calibration is desirable - but not sufficient

- Picture taken on 10/15/2004; note the calibration sticker was applied only three months before.
- This gauge is actually disconnected and still gives a reading of 70 PSI

Power Measurements

Power Measurements

Power can be measured:

- Directly
- By measuring Voltage, Amperage and estimating Power Factor
- MEASUR has a built in Power Factor estimator

End of Course

Thank you for your participation

Please complete the course evaluation

Contact Details

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