Better Plants®
U.S. DEPARTMENT OF ENERGY

Pumping System Assessment

U.S. DEPARTMENT OF ENERGY

Energy-saving Opportunities in Pumping Systems:

Where they are and how to recognize them

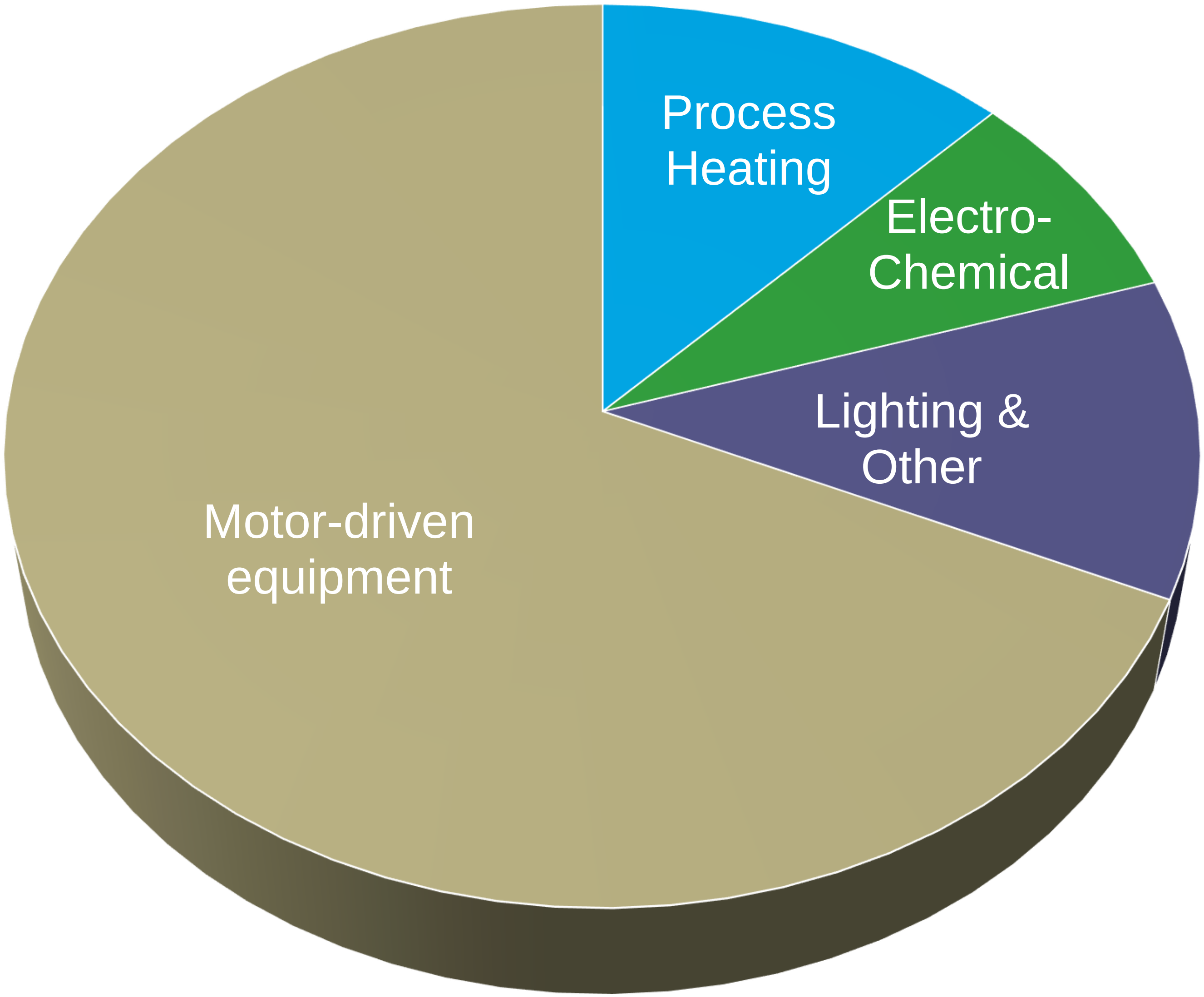
U.S. DEPARTMENT OF ENERGY

Big Picture Perspectives: Industrial Motor Systems

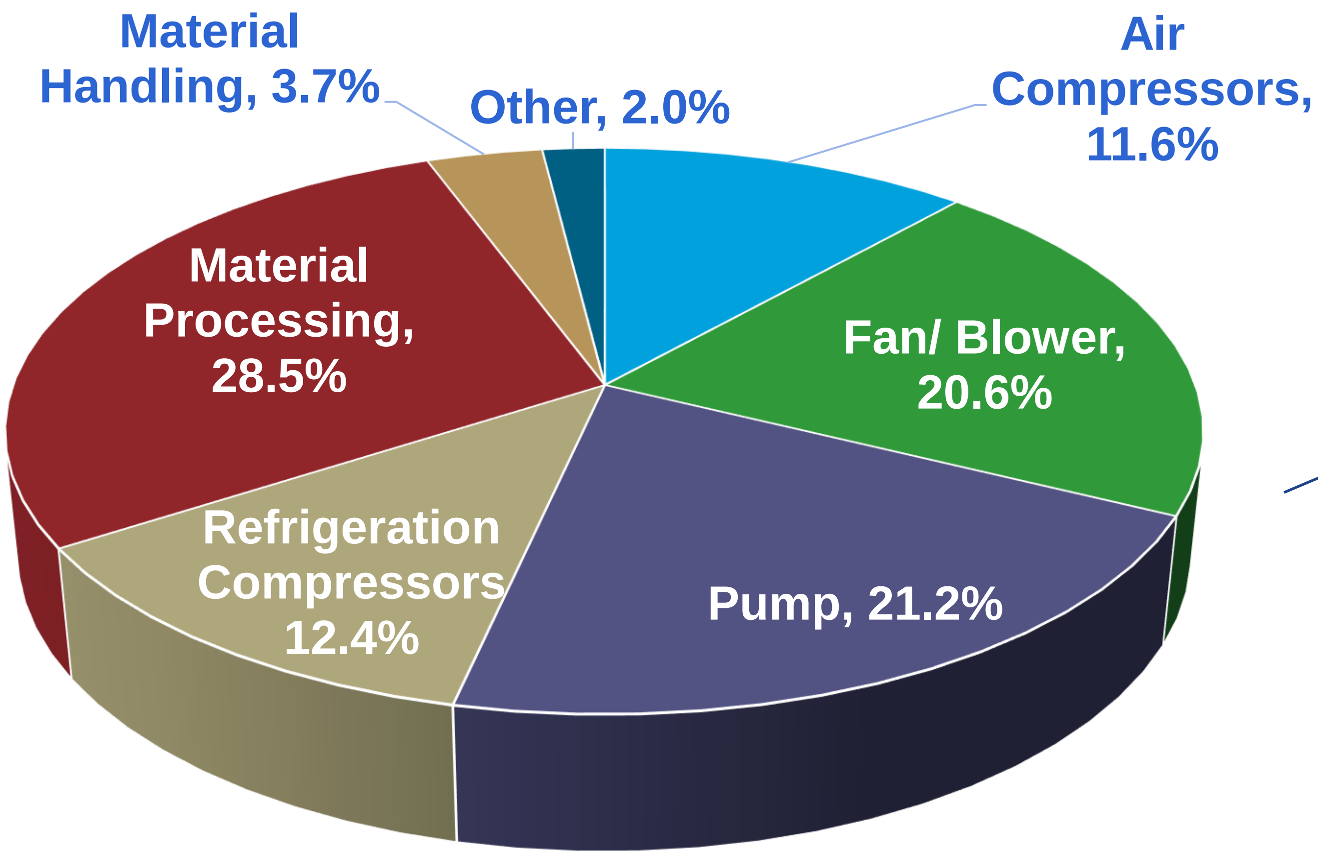
Industrial motor systems:

- are the *single largest electrical end use* category in the American economy
- account for 25% of U.S. electrical sales
- 950 million MWh in 2019

Motor loads dominate industrial electrical energy consumption



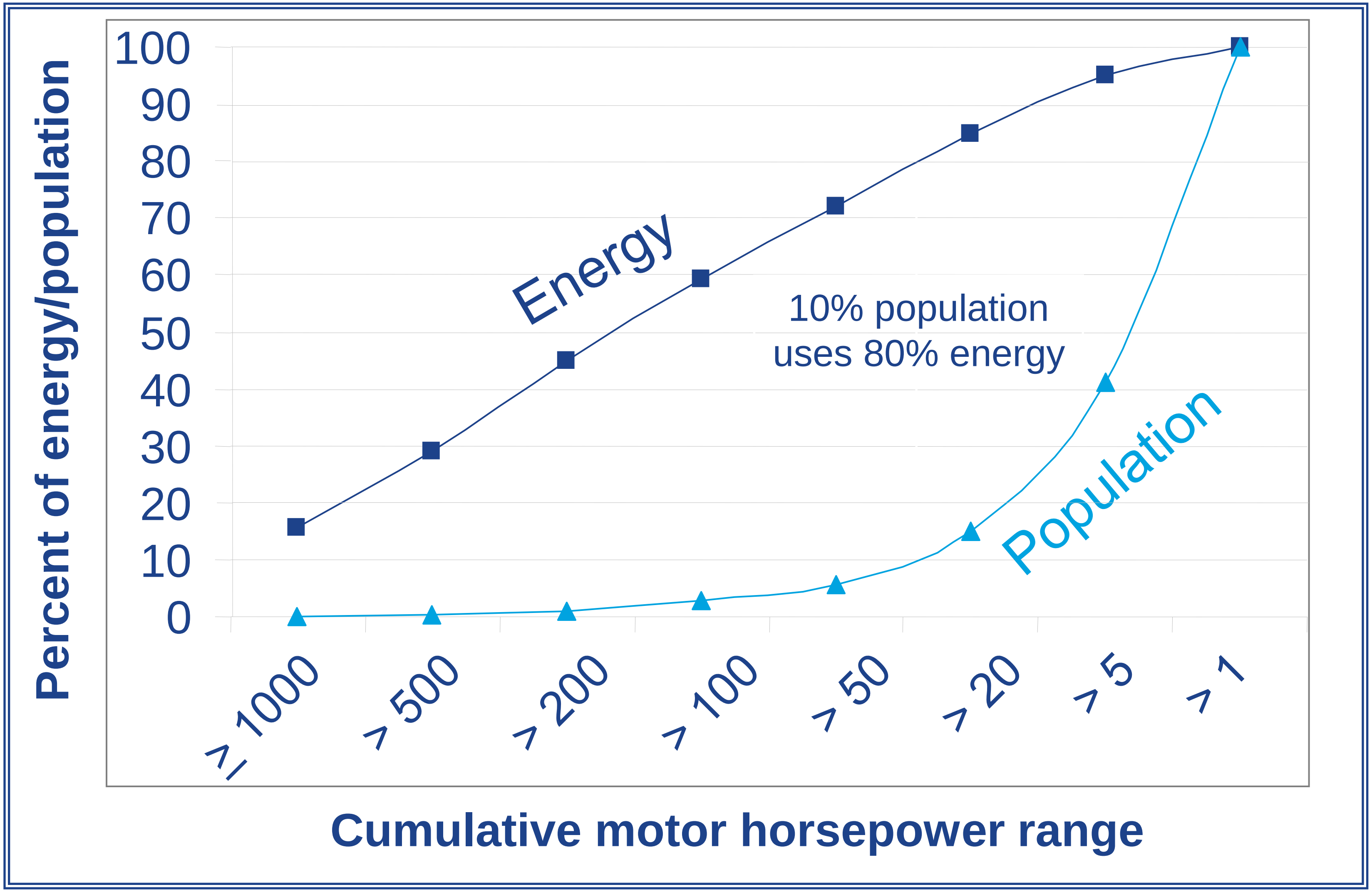
66% of industrial motor-system energy consumption involves fluid handling



Half of the motor population accounts for 2/3 of the energy

A large portion are centrifugal devices

A small fraction of the motor population is responsible for most of the energy consumption



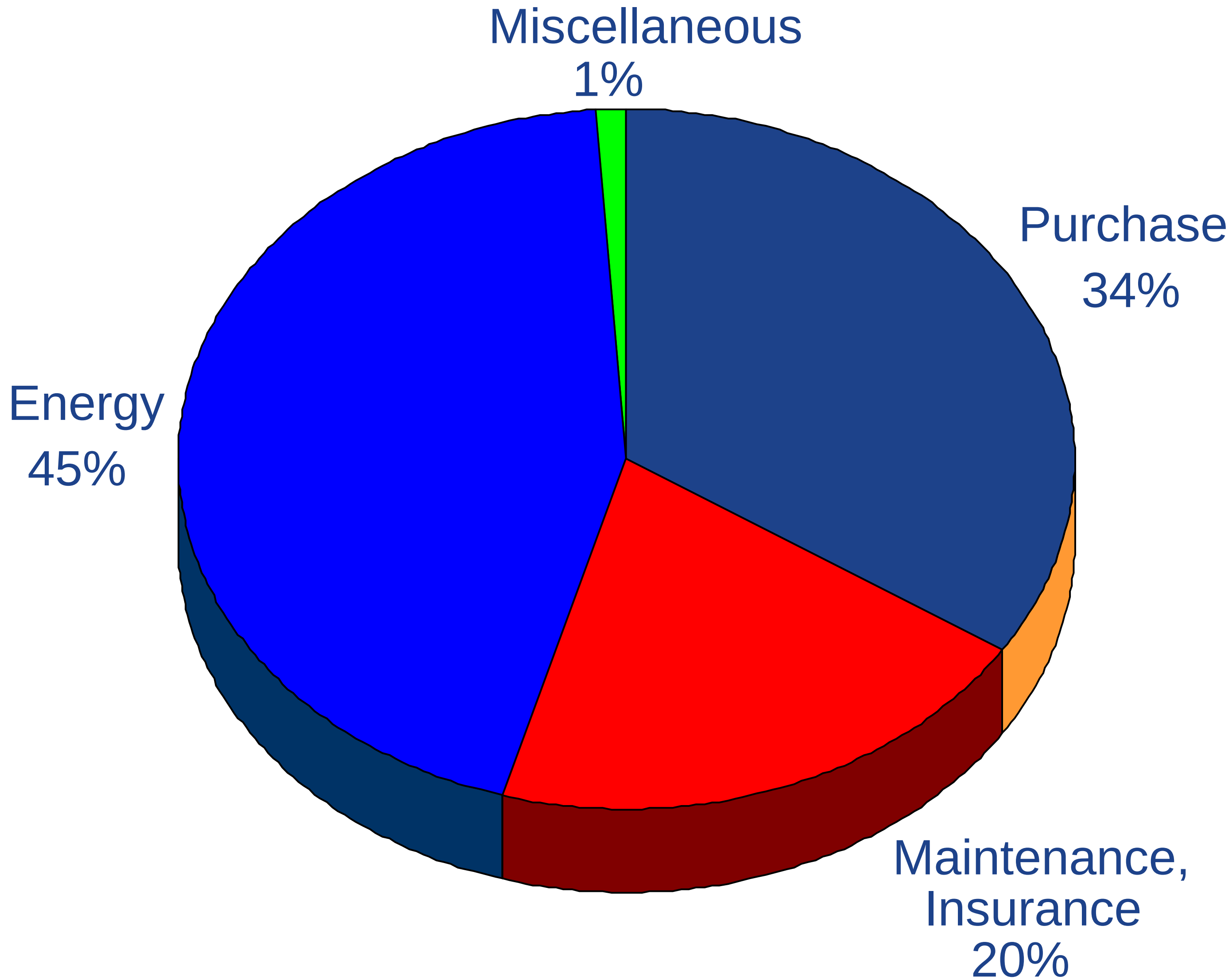
Note the descending order (left to right)

Comparing life cycle costs: automobile and pump/motor combination

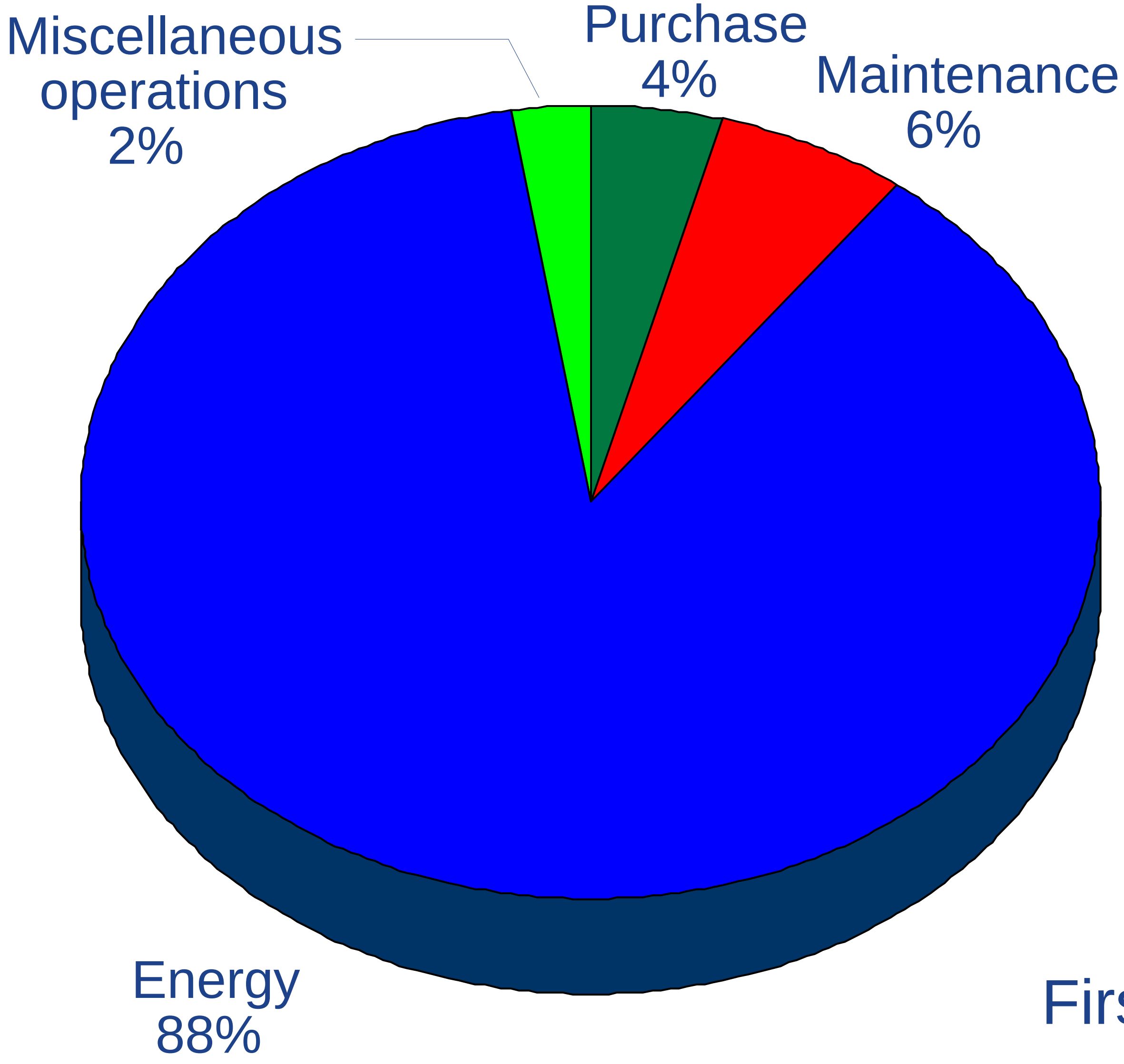
Common assumptions
Discount rate = 8%
Non-energy inflation rate = 4%
Lifetime = 10 years

Item	Automobile	Pump & motor
Initial energy cost rate	$4.00/gal	5 cents/kWhr
Energy inflation rate	10%/yr	5%/yr
Operating extent	20,000 miles/yr	7000 hr/yr (80%)
Maintenance/Insurance	$2,000/yr	$5,000/yr

Life cycle cost - example automobile
$28,000 purchase, 24 mpg, $4.00/gal, 20,000 miles/yr

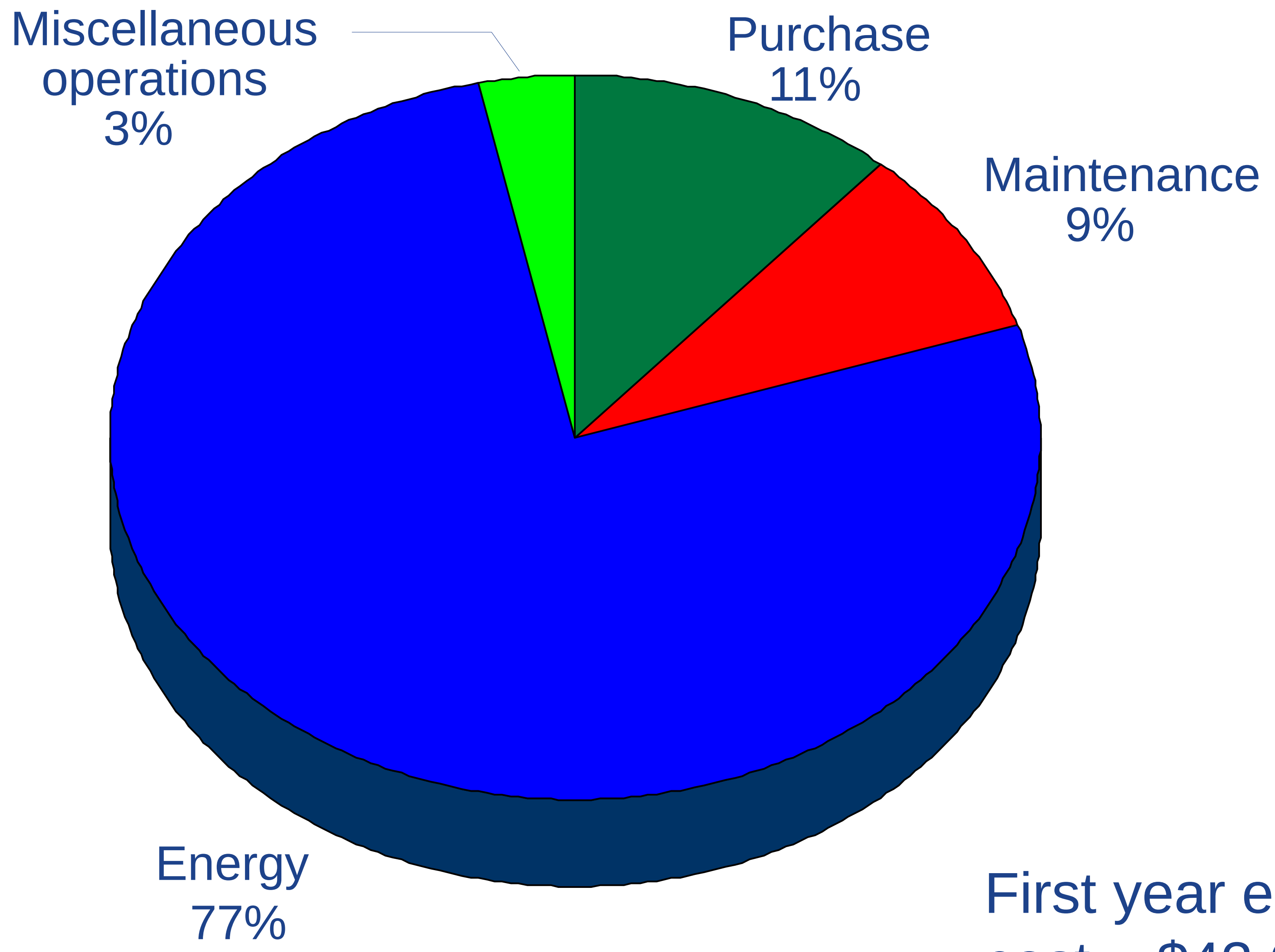


Life cycle cost - 250-hp pump and motor $28,000 initial cost, $5,000/yr maintenance



First year energy cost = $69,000

Higher first cost pump and motor ($56K), low service time (4,380 hrs/year)



First year energy cost = $43,000

Pump and motor efficiencies: Seventy+ years of progress

Year	Pump efficiency (%)	Motor efficiency(%)
1928	80	87.5
1955	85	90.5
2006	88	95.4

Achievable efficiency estimates for commercially available 75-hp pump and motor

With pumping <u>systems</u>, motor and pump performance is just part of the bigger picture

Better Plants® U.S. DEPARTMENT OF ENERGY

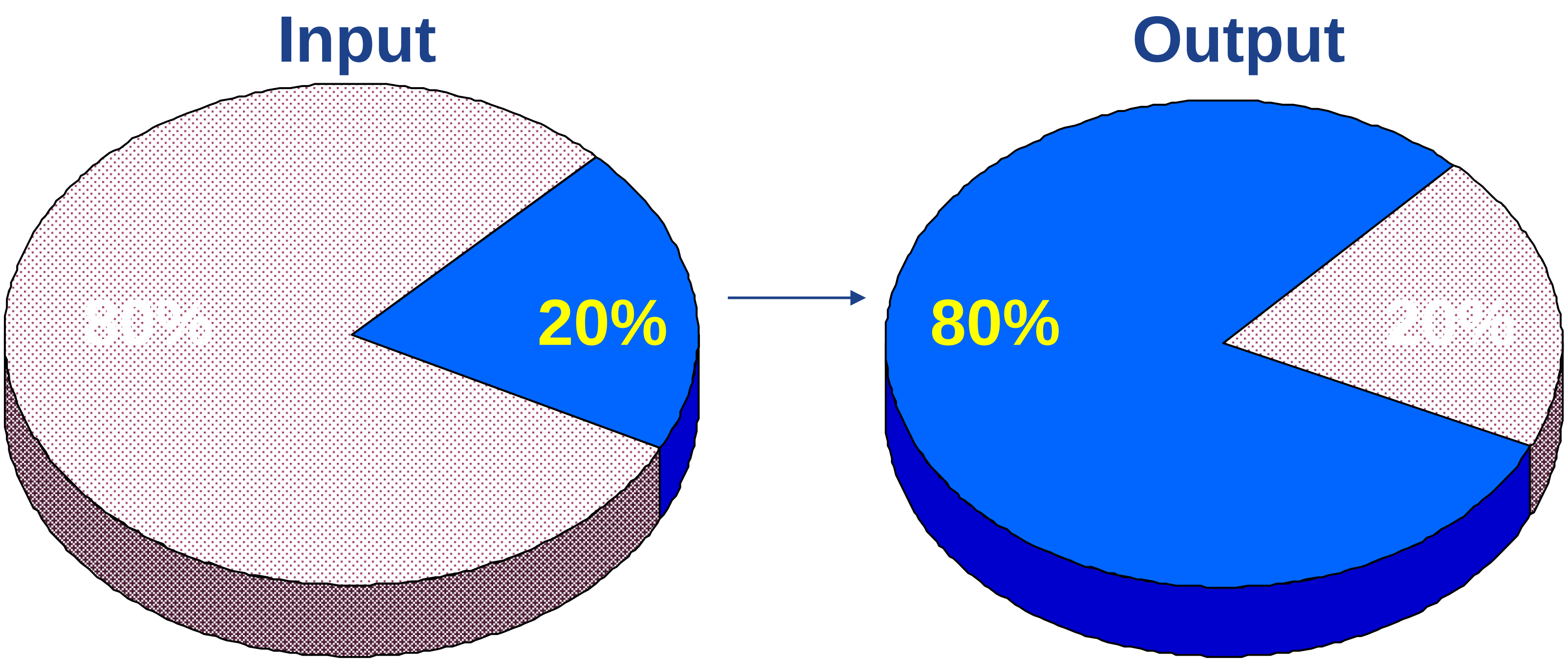
U.S. DEPARTMENT OF ENERGY

The Pareto Principle or "the vital few and trivial many"

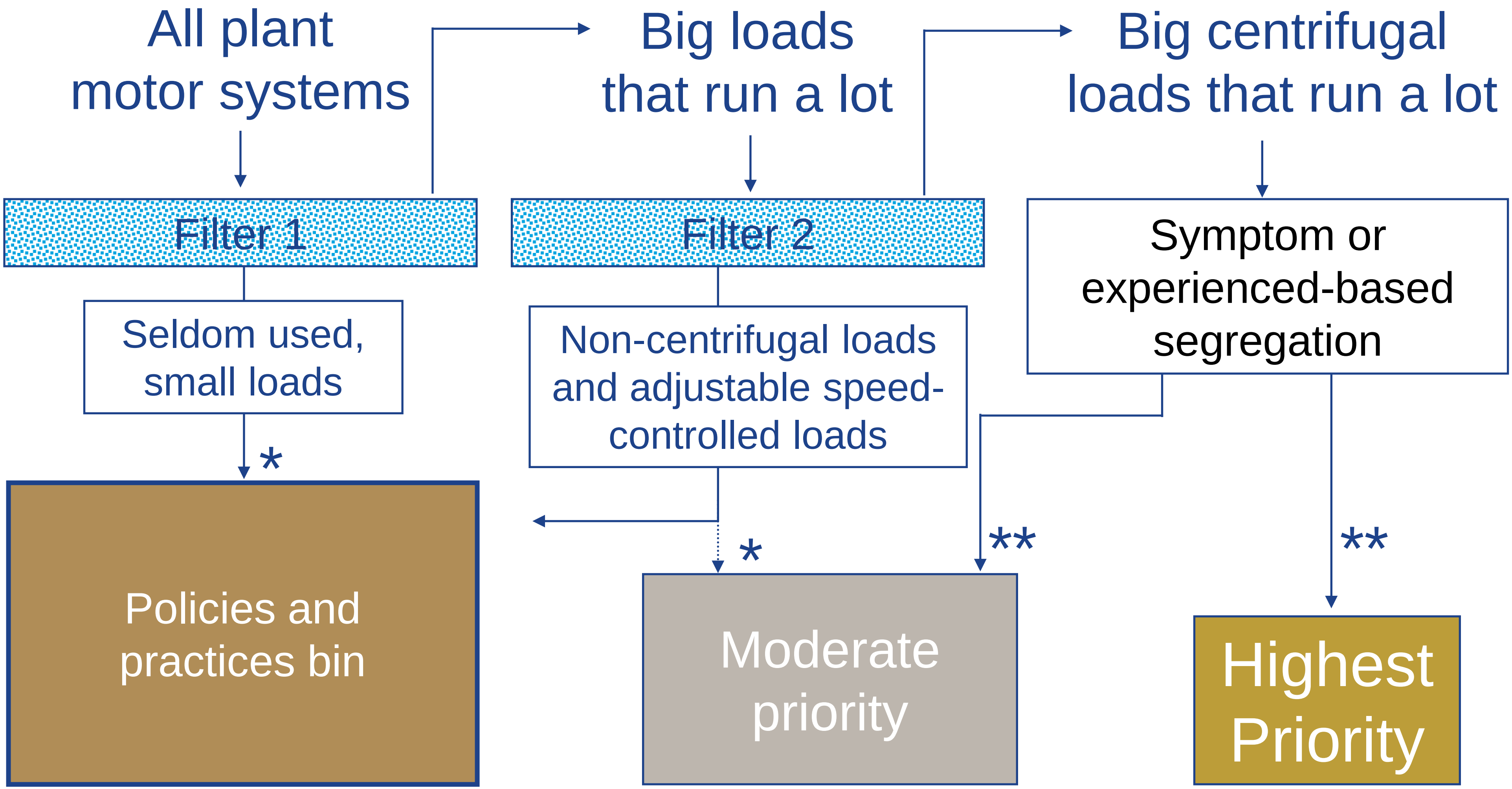
J. M. Juran, who first used the term "Pareto Principle" also coined a more descriptive phrase:

"The VITAL FEW and the trivial many"

(Relatively few are responsible for relatively much)



Prescreening to narrow the field of focus - i.e., to select the VITAL FEW for further review



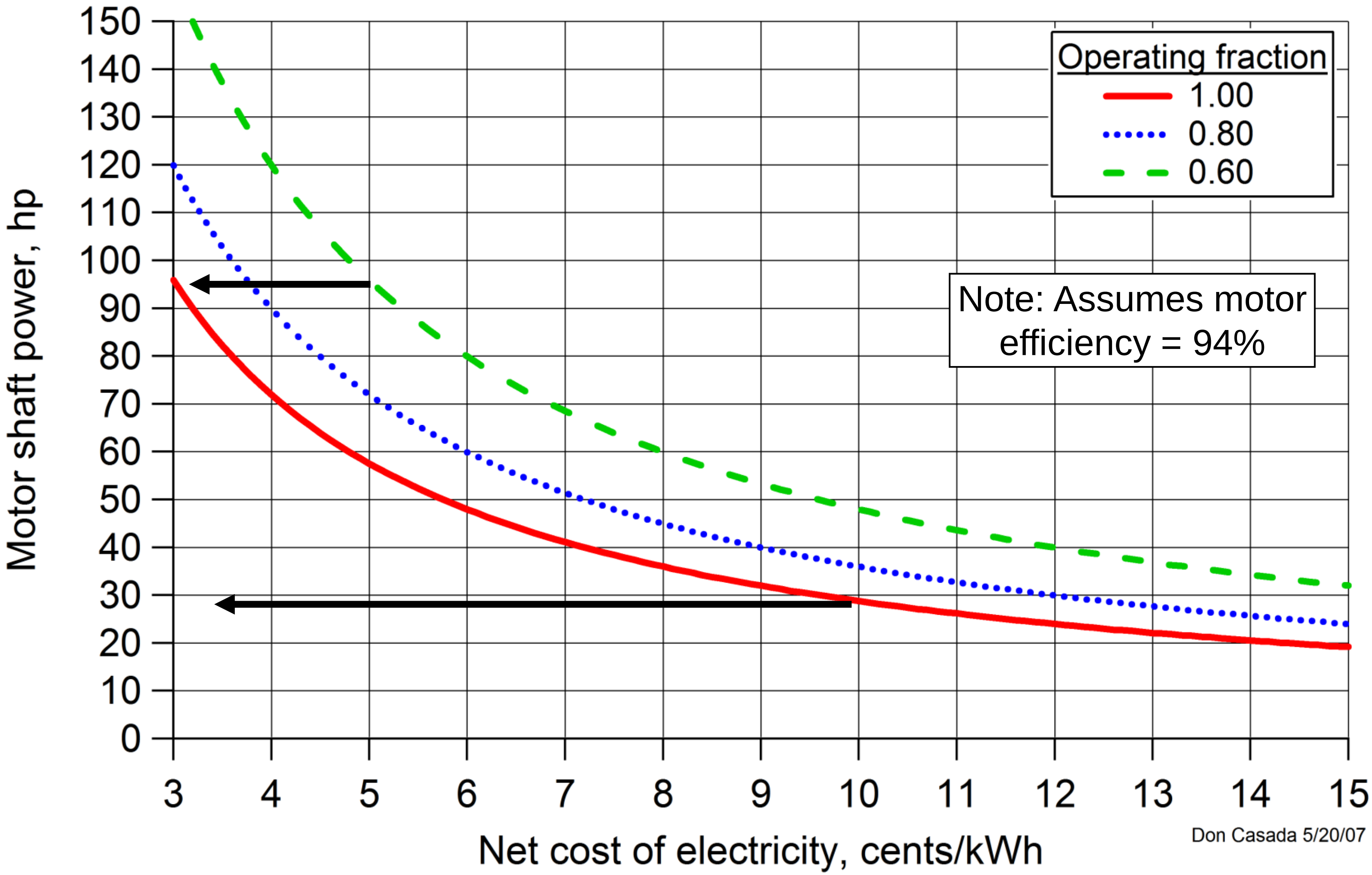
* Productivity/reliability-critical systems sent to higher priority levels

** Policies & practices also apply to moderate & highest priority applications

Electric rates help define what qualifies as "big"

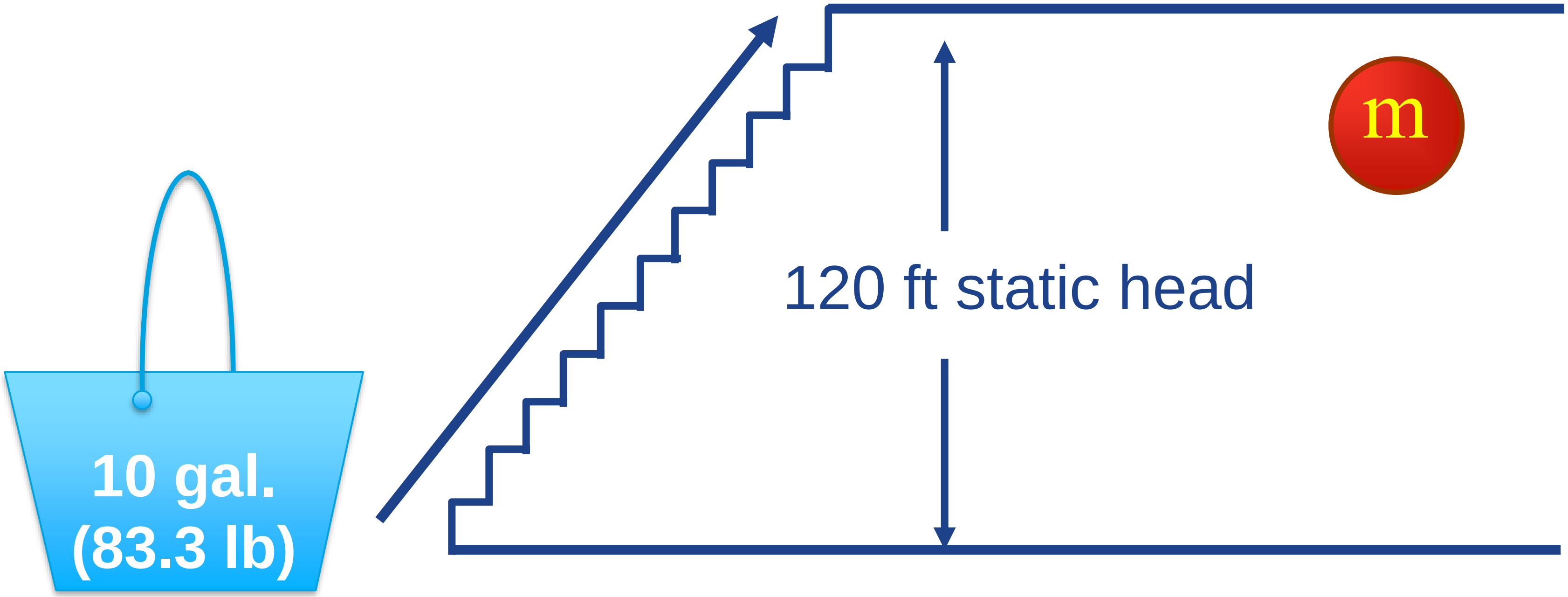
- Common rate elements
  - Energy ($/kWh)
  - Demand ($/kW)
  - Fixed charge ($/meter)
  - Reactive power ($/kVAR)
- Schedule options, misc. considerations
  - Incredible array of variations (time-of-use, seasonal, etc.)
  - Expect increasing attractiveness to time of use, and more variations of it as the narrow margin between capacity and demand shrinks.
- Suggested threshold: $20,000/yr electric cost
- But also consider low run time equipment that may set the demand charge (run off-peak instead of on-peak)

Motor load for $20,000/year cost as a function of net electrical cost rate



Examples: at 10 ¢/kWh, a continuously-operated 30-hp shaft load would cost more than $20K/yr.
At 5 ¢/kWh and 80% load factor, a 95-hp load would cost the same.

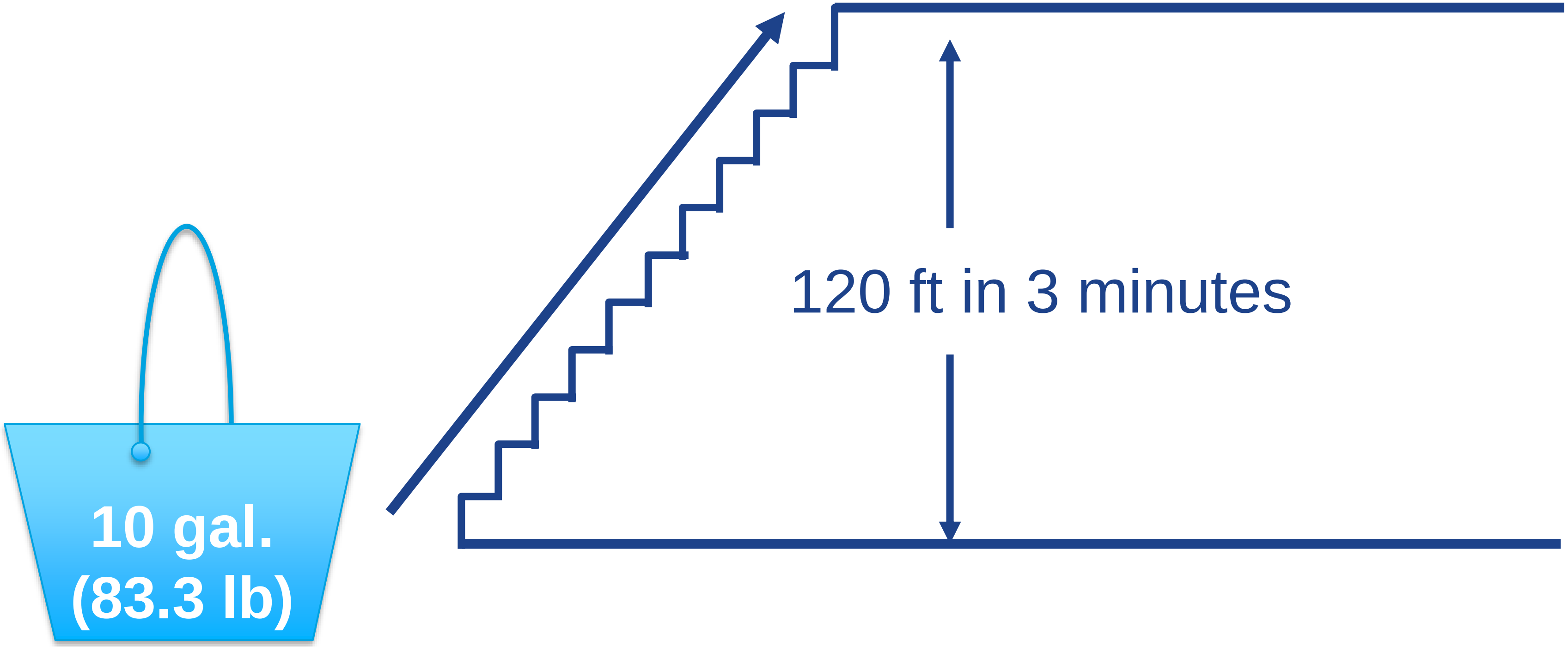
To provide direction for secondary prescreening, let's first review fluid energy basics



Ideal fluid <u>movement</u> energy requirements are proportional to weight and head

Ideal energy above =	10000 ft-lb, or 3.24 calories (less than one M&M)

Ideal power depends on how fast it is moved



Power required = 0.1 horsepower, or
65 calories per hour

An important – and practically useful – fluid power relationship

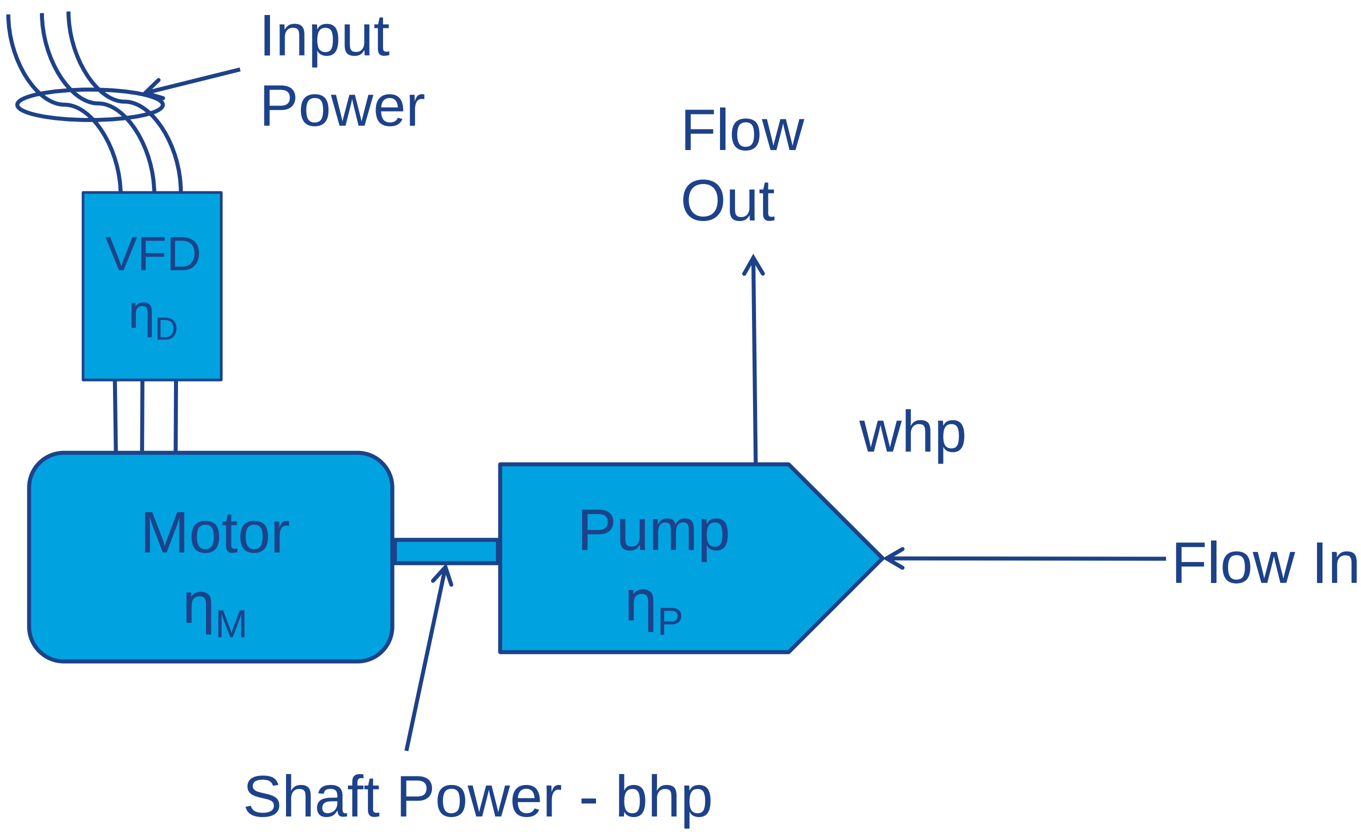
$$\text{Fluid power (hp)} = \frac{\text{Flow rate (gpm)} \cdot \text{Head (ft)} \cdot \text{specific gravity}}{3960}$$

Pumping system energy basics are fundamental to secondary prescreening

$$E = \frac{Q \cdot H \cdot T \cdot sg}{5308 \cdot \eta_{pump} \cdot \eta_{motor} \cdot \eta_{drive}}$$

E	energy, kilowatt-hours	
Q	flow rate, gpm	System-level opportunities
H	head, ft	
T	time, hours	
sg	specific gravity, dimensionless	
5308	Units conversion constant	
η_{pump}	pump efficiency, fraction	Component-level opportunities
η_{motor}	motor efficiency, fraction	
η_{drive}	drive efficiency, fraction	

VFD on Pump Motor



$$\text{VFD Input Power} = whp/(\eta_D * \eta_M * \eta_P)$$

Example symptoms in pumping systems that indicate potential opportunity

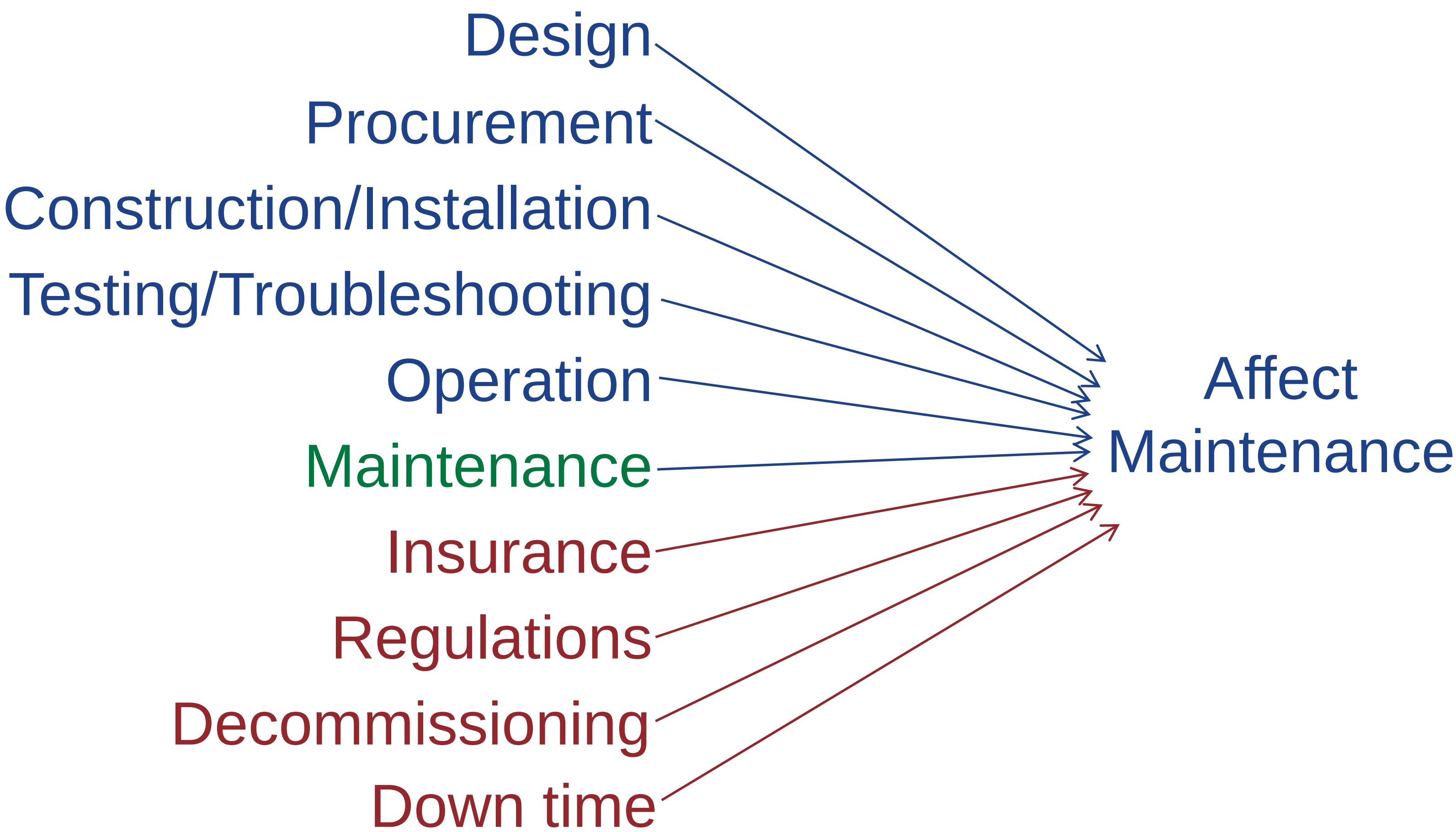
- Throttle valve-controlled systems
- Bypass (recirculation) line normally open
- Multiple parallel pump system with same number of pumps always operating
- Constant pump operation in a batch environment or frequent cycle batch operation in a continuous process
- Cavitation noise (at pump or elsewhere in the system)
- High system maintenance
- Systems that have undergone change in function

Many life cycle elements influence reliability, cost, and productivity of motor-driven systems

- Design
- Procurement
- Construction/Installation
- Testing/Troubleshooting
- Operation
- Maintenance
- Insurance
- Regulations
- Decommissioning
- Down time
- etc.…

Most of these elements are interdependent

Example: other factors do or may affect maintenance



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

Design
Procurement
Construction/Installation
Testing/Troubleshooting
Operation
Maintenance
Insurance
Regulations
Decommissioning
Down time

Maintenance does, should or may affect

Life cycle elements are an integrated system, much like the physical systems themselves

- The elements can be treated as components
  - For example, procurement can be on lowest first cost basis, without regard to the effect on maintenance or operations
- The elements can be treated as a system
  - For example, procurement considers all the elements of cost and is based on lowest total life cycle cost

Contingency planning - making the change when a failure occurs

- The alternatives evaluation picture changes dramatically when failures occur
- Changes that couldn't be justified when the system was functional may well be after failure
- The alternative may actually be less costly than simple repair/replace of the existing component
- Contingency planning is common for motors, but not for pumps

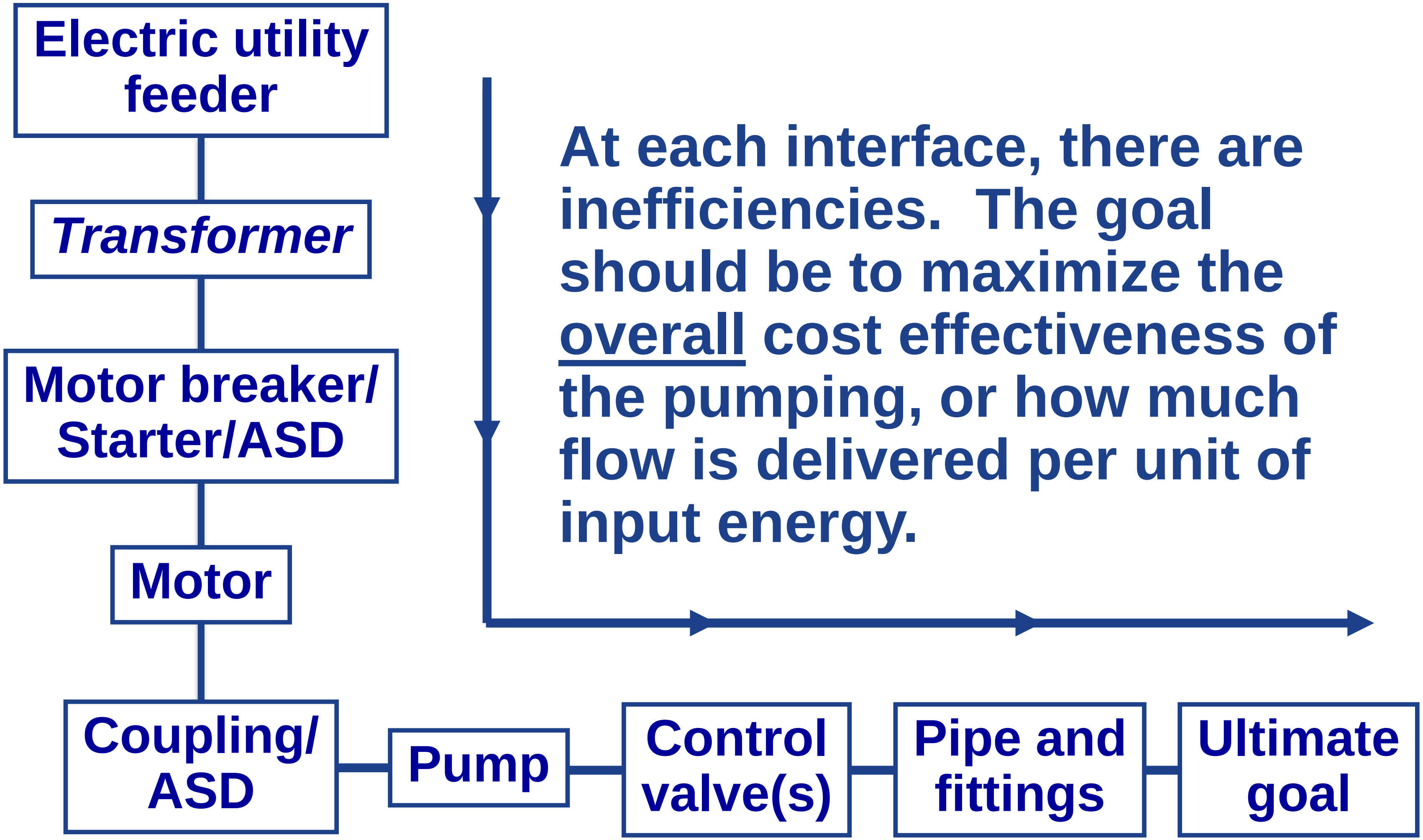
Break

Field monitoring of pumping systems

and

Application of MEASUR software tool

First, let's try to get a big picture perspective of energy flow for pumping systems



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

Observations on losses in some of the power train components

Utility grid – line losses; not our problem (or are they?)
Transformer – Very efficient (typically upper 90's %)
Breaker/starter – Negligible
Electrical ASD – Minor
Motor – Minor
Coupling – Minor
Mechanical ASD – Minor to moderate
Pump – Important
Control valves – Zero to major
Pipe and fittings – Minor to major
Ultimate goal – Always important

The optimization path is the other side of a two way street

Both in the design phase and in assessments of existing systems, the starting point is the ultimate goal.

This approach is NOT for cookie cutters and rules of thumb practitioners.

Electric utility feeder

Transformer

Motor breaker/ Starter/ASD

Motor

Coupling/ ASD

Pump

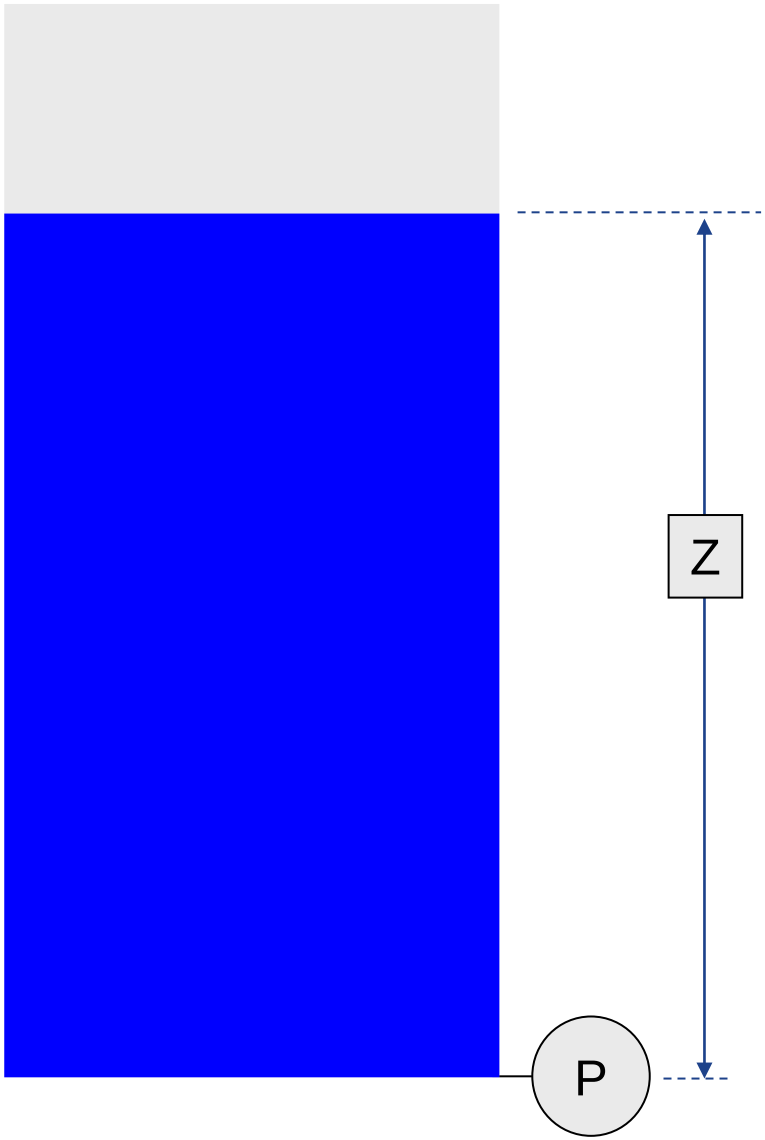
Control valve(s)

Pipe and fittings

Ultimate goal

System performance characteristics

Fluid statics: pressure, elevation, density



$$P = \frac{Z \bullet s.g.}{2.31}$$

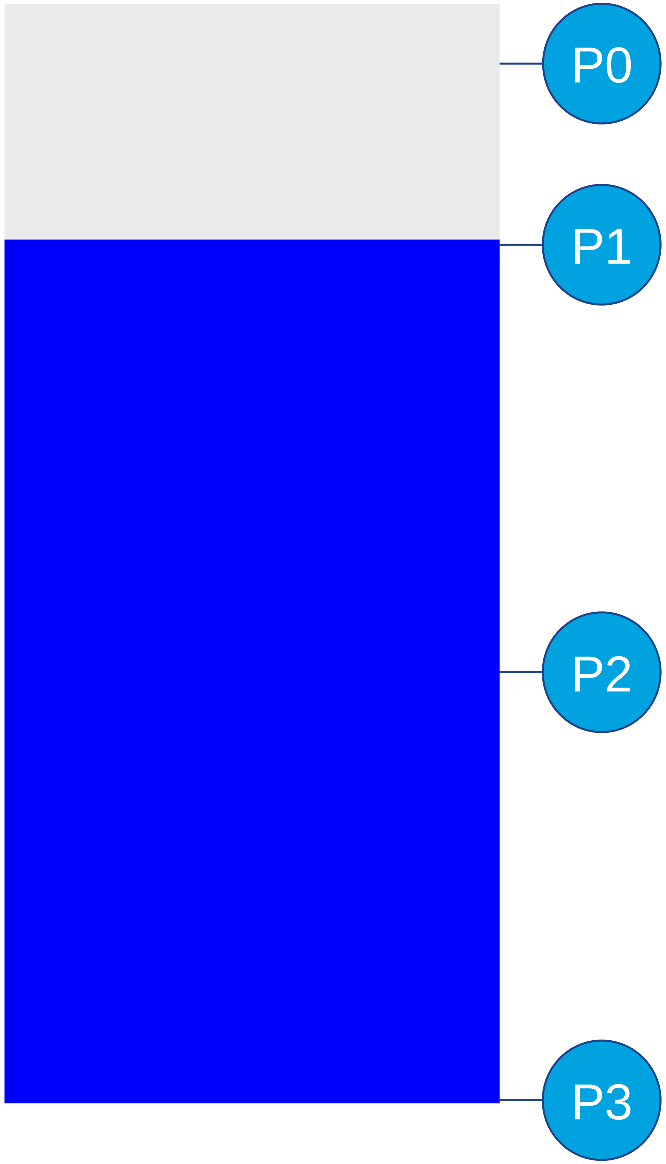
P = pressure, psig

Z = elevation, ft

s.g. = fluid specific gravity

If Z = 23.1 ft, and the fluid is normal temperature water (s.g. = 1.0), then P = 10 psig

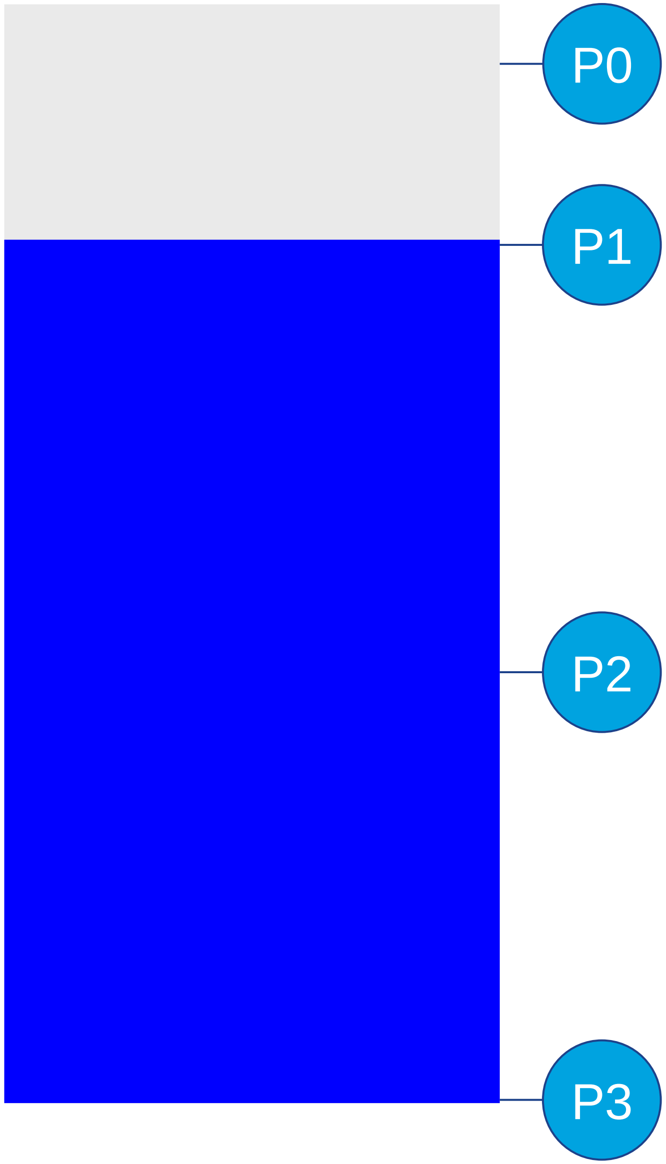
The head is constant throughout a static system



Gauge Location	Pressure psig	Gauge Elev, ft	Total head, ft
P1	0.00	23.10	23.10
P2	5.00	11.55	23.10
P3	10.00	0.00	23.10

s.g. = 1.00

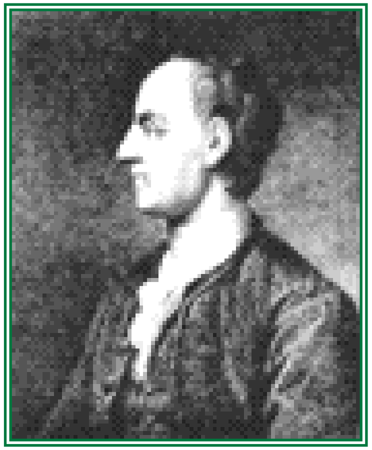
If we change to a more dense fluid, but at the same level, pressure changes, but head does not



Gauge Location	Pressure psig	Gauge Elev, ft	Total head, ft
P1	0.00	23.10	23.10
P2	6.00	11.55	23.10
P3	12.00	0.00	23.10

s.g. = 1.20

One of the single most important fluid energy relationships was identified in the 1700's



Euler

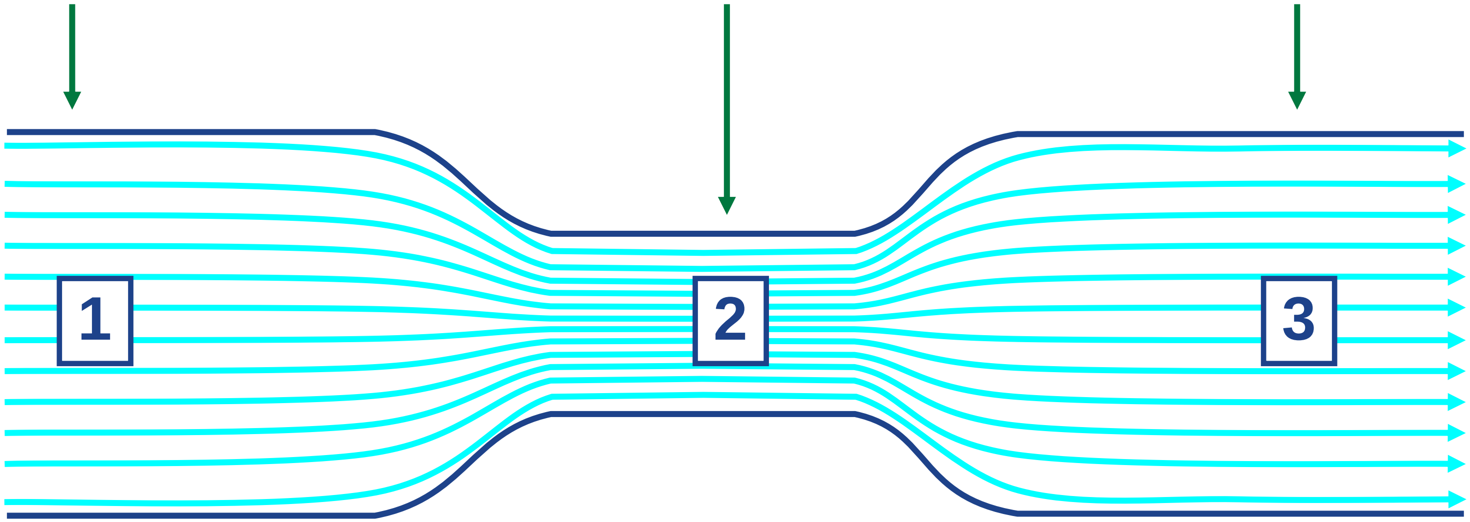
Bernoulli

Component	Symbol
Velocity	V
Pressure	P
Elevation	Z

The combined energy, or head, associated with the fluid velocity, pressure, and elevation along a frictionless streamline is constant, even if the individual components aren't.

Bernoulli's principle: Total energy is constant along a *frictionless* streamline

$$\frac{V_1^2}{2g} + \frac{P_1}{\gamma} + Z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\gamma} + Z_2 = \frac{V_3^2}{2g} + \frac{P_3}{\gamma} + Z_3$$



γ = fluid specific weight (lb_f/ft^3)
g = gravitational acceleration, 32.2 ft/sec^2 (9.81 m/s^2)
Note: Units are in ft (m) of *head*

A useful analogy to Bernoulli

The Bernoulli relation applies to frictionless, steady, streamline flow

In the real world, friction exists (both within the fluid itself and between the fluid and pipe walls).

So how much does friction cause the real world to deviate from the relationships of the Bernoulli principle?

It depends. Sometimes a little, sometimes a lot.

What are some sources of friction in pumping systems?

Pipe walls

Valves

Elbows

Tees

Reducers/expanders

Expansion joints

Tank inlets/outlets

(i.e., almost everything that the pumped fluid passes through, as well as the fluid itself)

Pipe friction loss estimates are usually based on an equation referred to as Darcy-Weisbach

This equation is very useful to examine to understand what parameters influence *frictional* losses in piping:

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

H_f = head loss due to friction (ft)

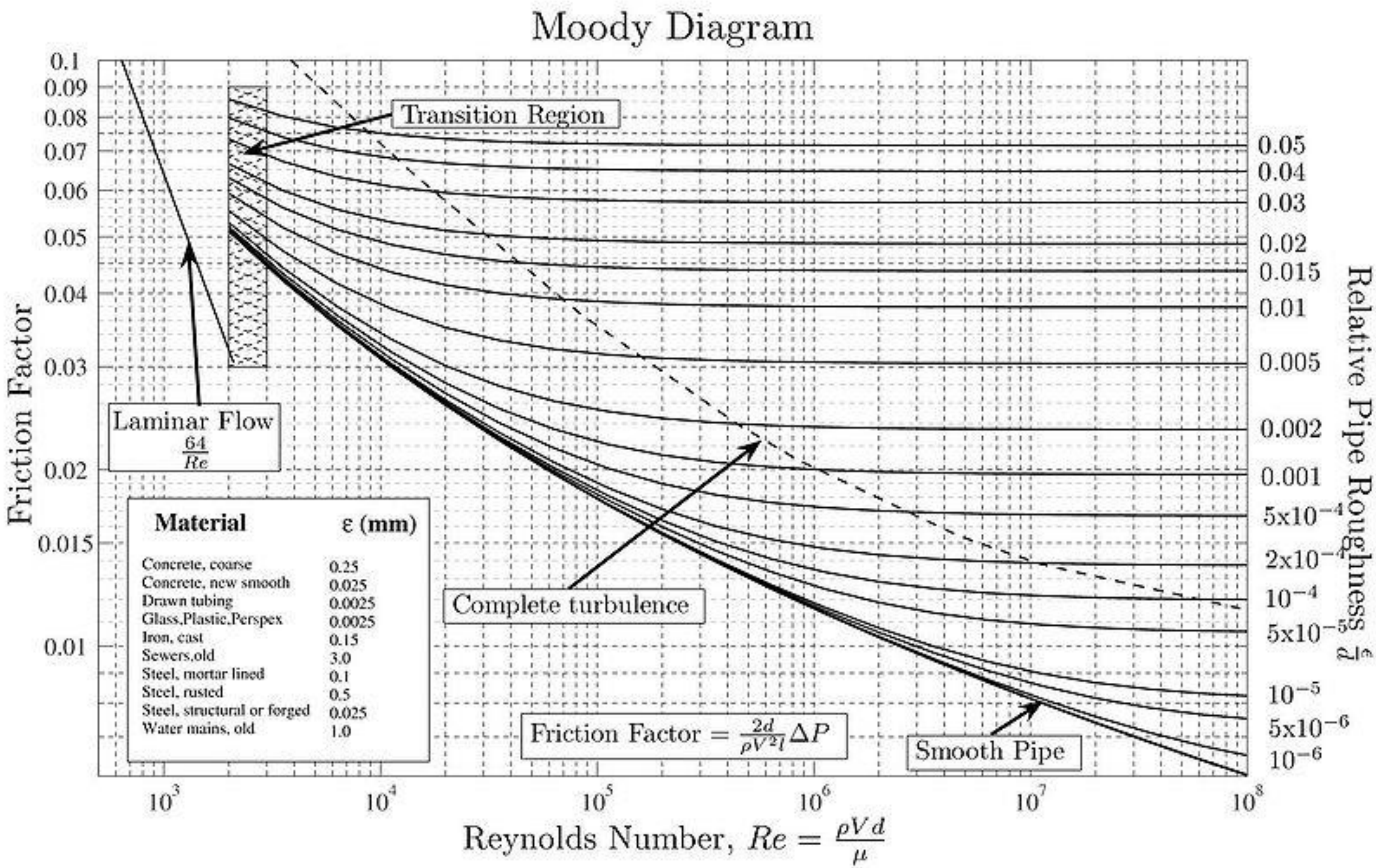
f = Darcy friction factor

L = pipe length (ft)

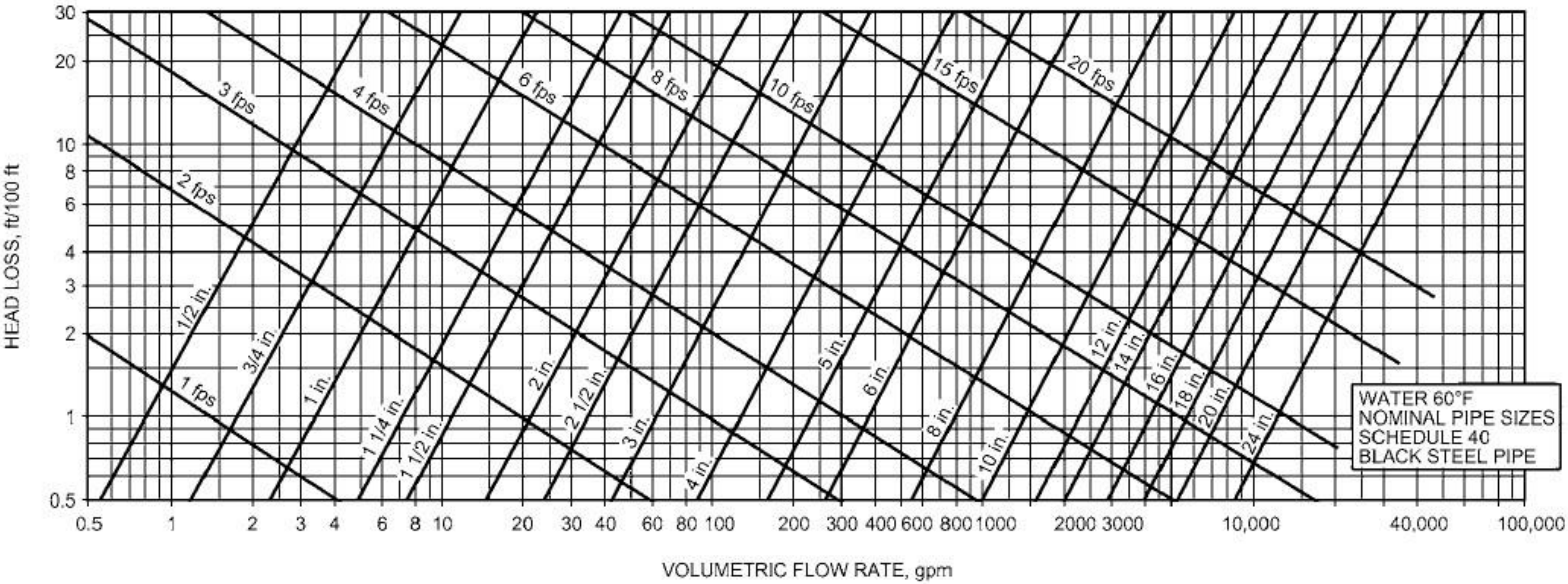
d = pipe diameter (ft)

$\frac{V^2}{2g}$ = velocity head (ft)

Moody Diagram for Determining Friction Factor



Schedule 40 Steel Pipe Sizing



Piping component frictional losses are also primarily dependent on experimental data

For pipe components, frictional losses have generally been estimated based on the velocity head.

$$H_f = K \cdot \frac{V^2}{2g}$$

K = Loss coefficient

$\frac{V^2}{2g}$ = velocity head

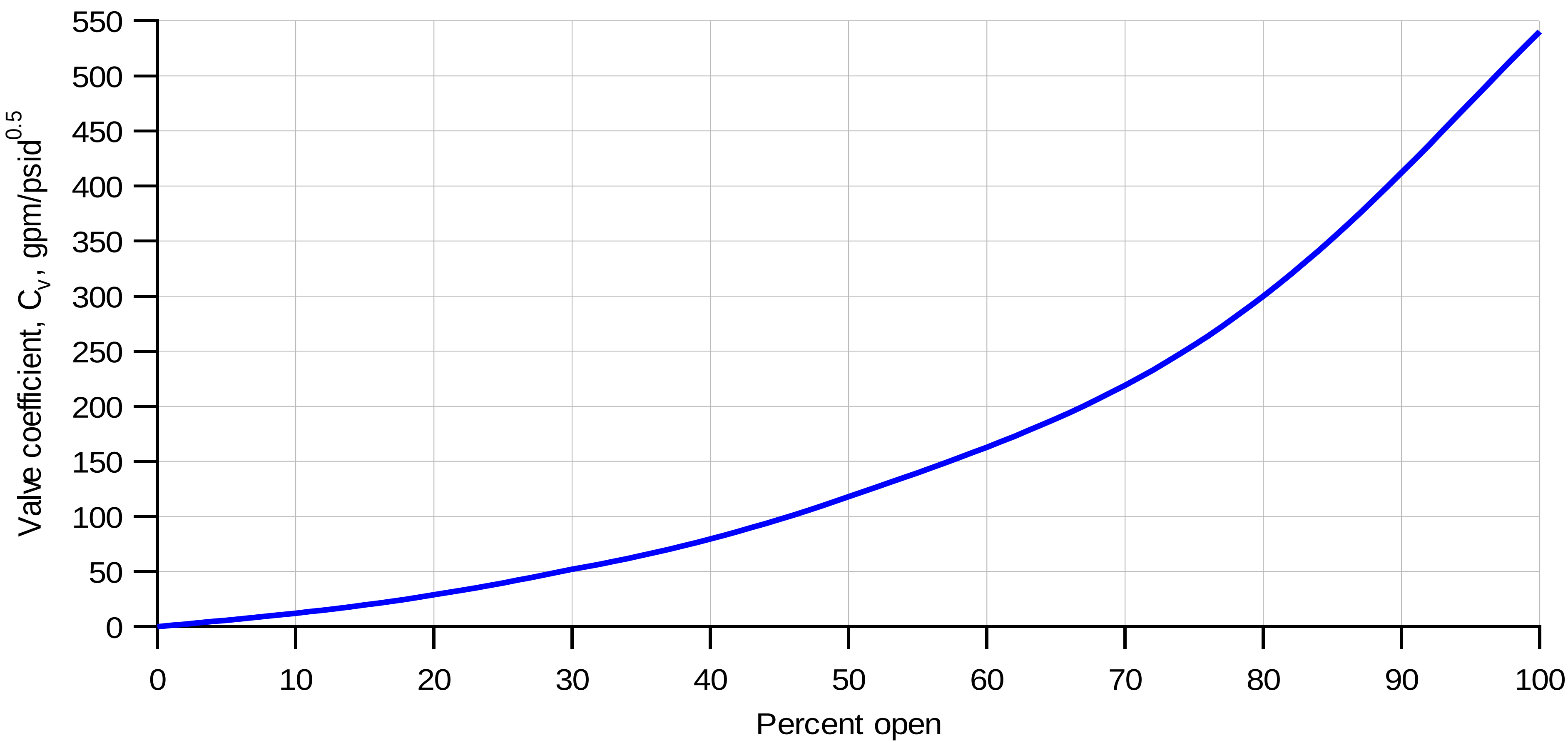
K is a function of size, and for valves, the valve type, and valve % open.

Some *typical* K values for miscellaneous pipe components

Component	Component K
90° elbow, standard	0.2 - 0.3
90° elbow, long radius	< 0.1 - 0.3
Square-edged inlet (from tank)	0.5
Discharge into tank	1
Check valve	2
Gate valve (full open)	0.03 - 0.2
Globe valve (full open)	3 - 8
Butterfly valve (full open)	0.5 - 2
Ball valve (full open)	0.04 - 0.1

Valve Performance Measures

Flow coefficient versus valve position, for the rotary valve



Unfortunately, "typical" is not very definitive

(From Hydraulic Institute Engineering Data Book, 2nd edition)
Approximate Range of Variation for K

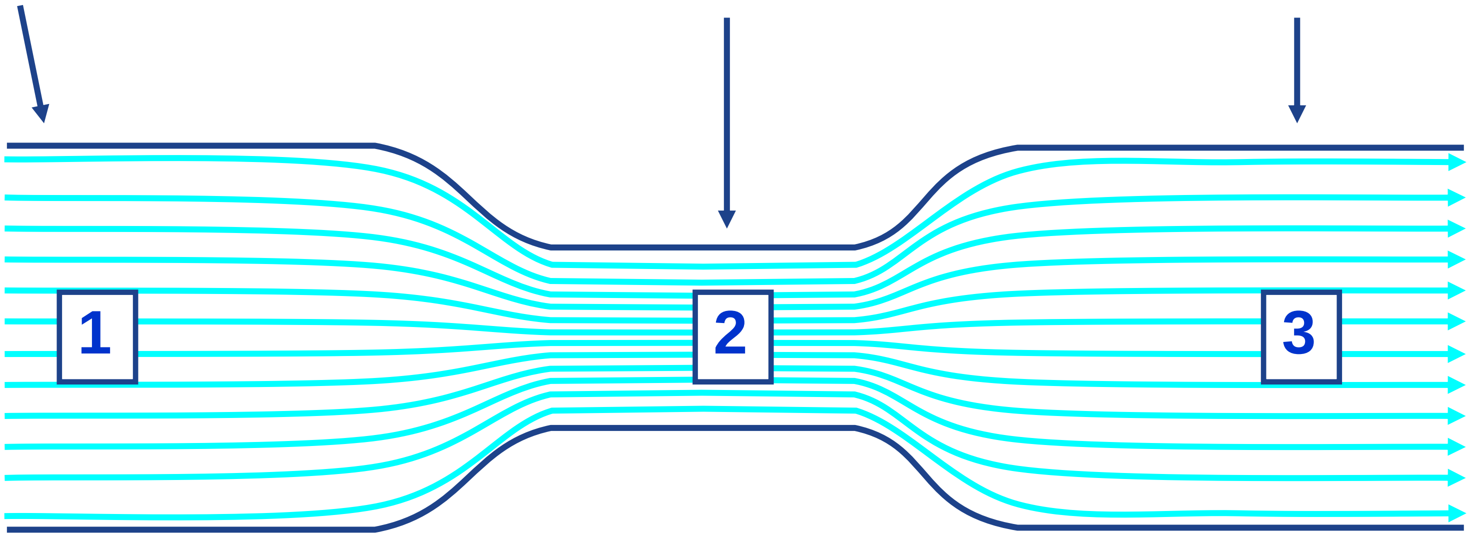
Fitting		Range of Variation
90 Deg. Elbow	Regular Screwed	± 20 per cent above 2 inch size
	Regular Screwed	± 40 per cent below 2 inch size
	Long Radius, Screwed	± 25 per cent
	Regular Flanged	± 35 per cent
	Long Radius, Flanged	± 30 per cent
45 Deg. Elbow	Regular Screwed	±10 per cent
	Long Radius, Flanged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

Design Engineer Perspective – Pump Selection

- If the pump is too small – I've got a big problem!
- If the pump is too large – The testing & balancing company will pinch the flow control valves and everything is fine. **(Except for the operating cost!!)**
- How will the engineer select the K's for his pressure drop calculations which leads to the pump selection?
- Gate valve (wide open) 0.03 – 0.20? Engineer used 0.20
- Globe valve (wide open) 3.0 – 8.0? Engineer used 8.0
- Butterfly valve (wide open) 0.5 – 2.0? Engineer used 2.0
- Standard 90º elbow 0.2 – 0.3? Engineer used 0.3
- Thus, the pump will likely be oversized
- The contractor may have added an additional safety factor to the pump sizing

We can slightly modify the Bernoulli equation to account for friction

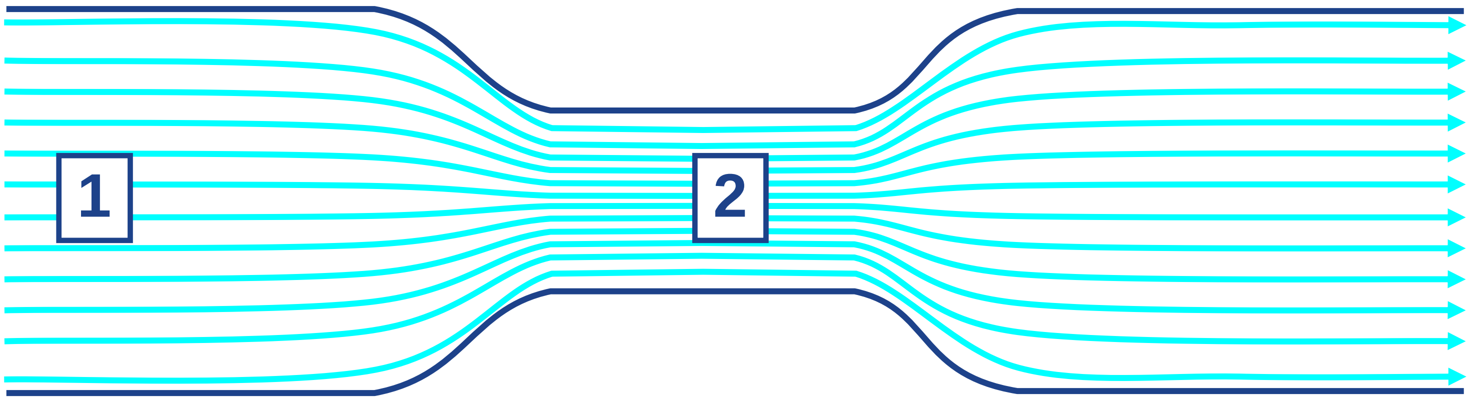
$$\frac{V_1^2}{2g} + \frac{P_1}{\gamma} + Z_1 = \frac{V_2^2}{2g} + \frac{P_2}{\gamma} + Z_2 + H_{f1\text{-}2} = \frac{V_3^2}{2g} + \frac{P_3}{\gamma} + Z_3 + H_{f1\text{-}3}$$



There is less hydraulic head/energy available at points 2 and 3 because of frictional losses

Modified again to accommodate the normal units of pressure in the U.S.A.

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

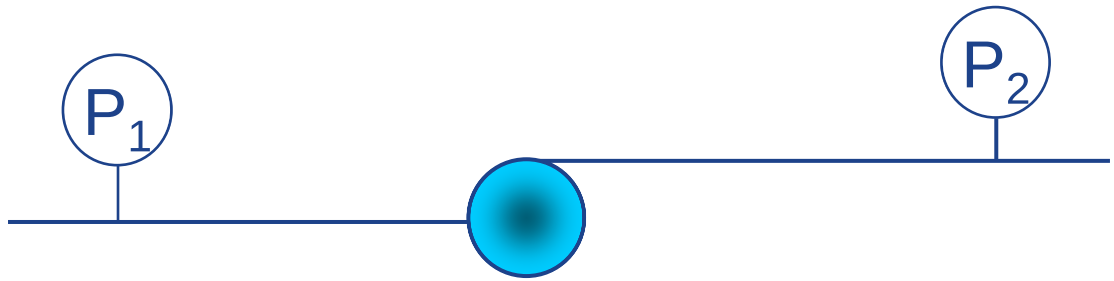


Symbol	Represents	Units
V	Velocity	ft/s
g	gravitational acceleration constant	ft/s^2
P	pressure	psig
s.g.	specific gravity	none
Z	Elevation	ft
H_f	Frictional head loss	ft

Pump Types

- Pumps are generally grouped into two broad categories—positive displacement pumps and dynamic (centrifugal) pumps. Positive displacement pumps use a mechanical means to vary the size of the fluid chamber to cause the fluid to flow. On the other hand, centrifugal pumps impart momentum to the fluid by rotating impellers that are immersed in the fluid. The momentum produces an increase in pressure or flow at the pump outlet.
- Positive displacement pumps have a constant torque characteristic, whereas centrifugal pumps demonstrate variable torque characteristics.
- A centrifugal pump converts driver energy to kinetic energy in a liquid by accelerating the fluid to the outer rim of an impeller. The amount of energy given to the liquid corresponds to the velocity at the vane tip of the impeller. The faster the impeller revolves or the bigger the impeller, then the higher the velocity of the liquid at the vane tip and the greater the energy imparted to the liquid.

The Bernoulli relationship is slightly modified to define the pump head



$$\frac{V_1^2}{2g} + \frac{2.31\,P_1}{s.g.} + Z_1 + \mathrm{H_{pump}} = \frac{V_2^2}{2g} + \frac{2.31\,P_2}{s.g.} + Z_2$$

or

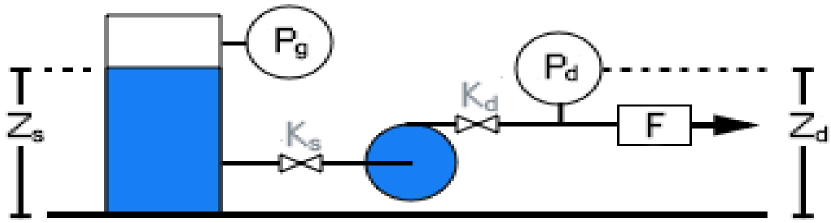
$$\mathrm{H_{pump}} = \left(\frac{V_2^2}{2g} + \frac{2.31\,P_2}{s.g.} + Z_2\right) - \left(\frac{V_1^2}{2g} + \frac{2.31\,P_1}{s.g.} + Z_1\right)$$

H_{pump} = Pump head at a given flow rate

MEASUR Pump Head Calculator

PUMP HEAD TOOL

Suction tank elevation | Suction gauge elevation



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	1.002
Flow Rate	3000 gpm

Suction		Discharge	
Pipe diameter (ID)	12 in	Pipe diameter (ID)	12 in
Tank gas overpressure (P_g)	0 psi	Gauge pressure (P_d)	124 psi
Tank fluid surface elevation (Z_s)	10 ft	Gauge elevation (Z_d)	10 ft
Line loss coefficients (K_s)	0.5	Line loss coefficients (K_d)	1

Generate Example | Reset Data

MEASUR Pump Head Calculator Results

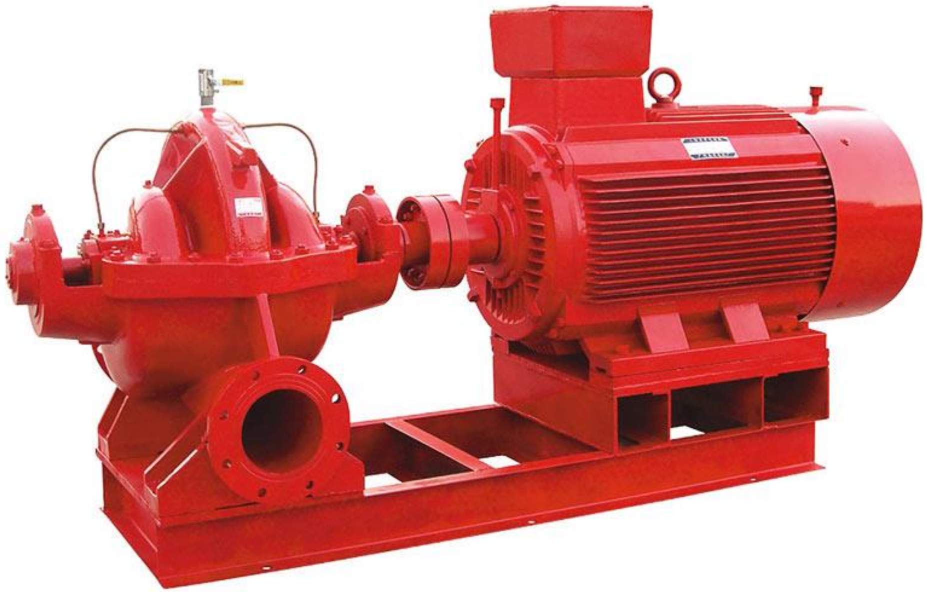
RESULTS | HELP

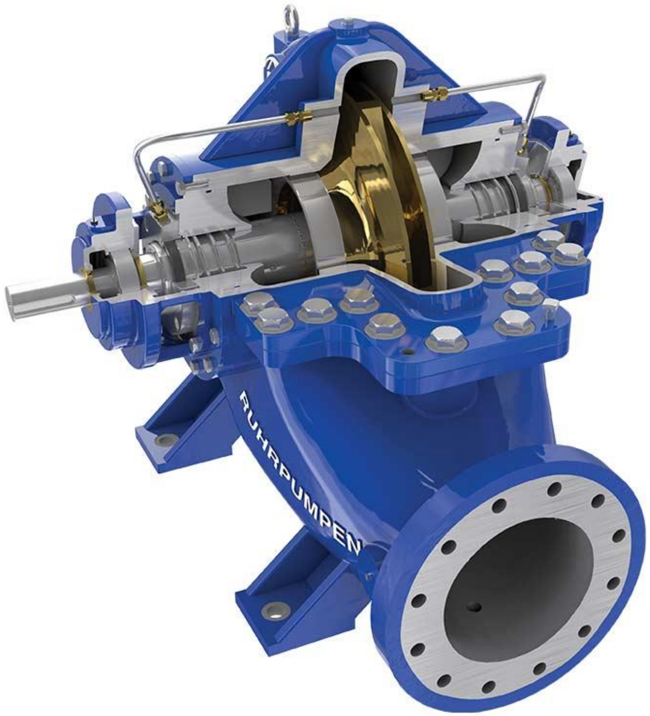
Result Data

Differential Elevation Head	0.0 ft
Differential Pressure Head	285.97 ft
Differential Velocity Head	1.13 ft
Estimated Suction Friction Head	0.56 ft
Discharge Friction Head	1.13 ft
Pump Head	288.78 ft

End Suction Vertical Discharge Centrifugal Pump

Horizontal Split Case Centrifugal Pump

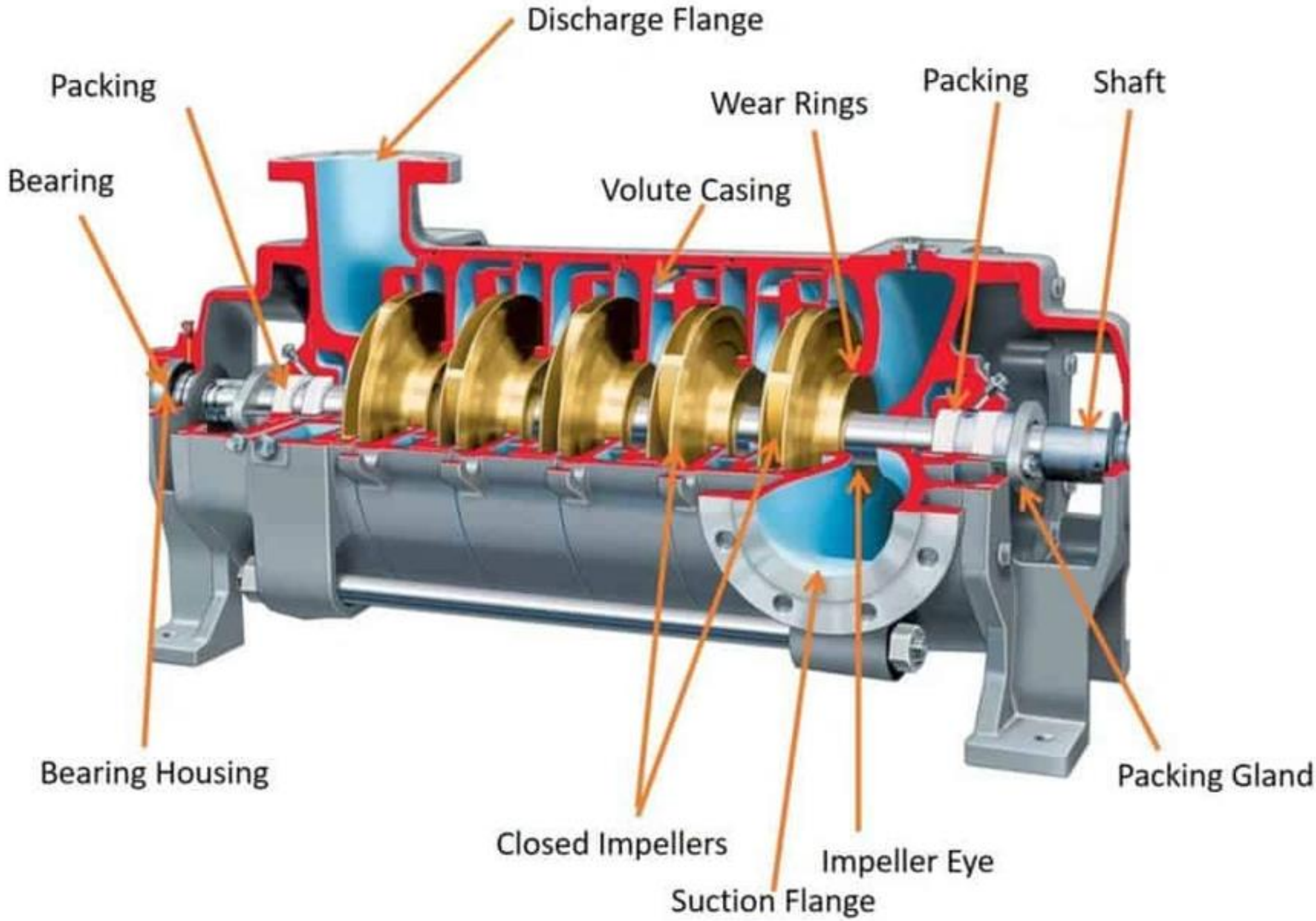




Vertical Turbine Pump

Vertical turbine pumps are commonly used in all types of applications, from moving process water in industrial plants to providing flow for cooling towers at power plants, from pumping raw water for irrigation, to boosting water pressure in municipal pumping systems, and for many other pumping applications.

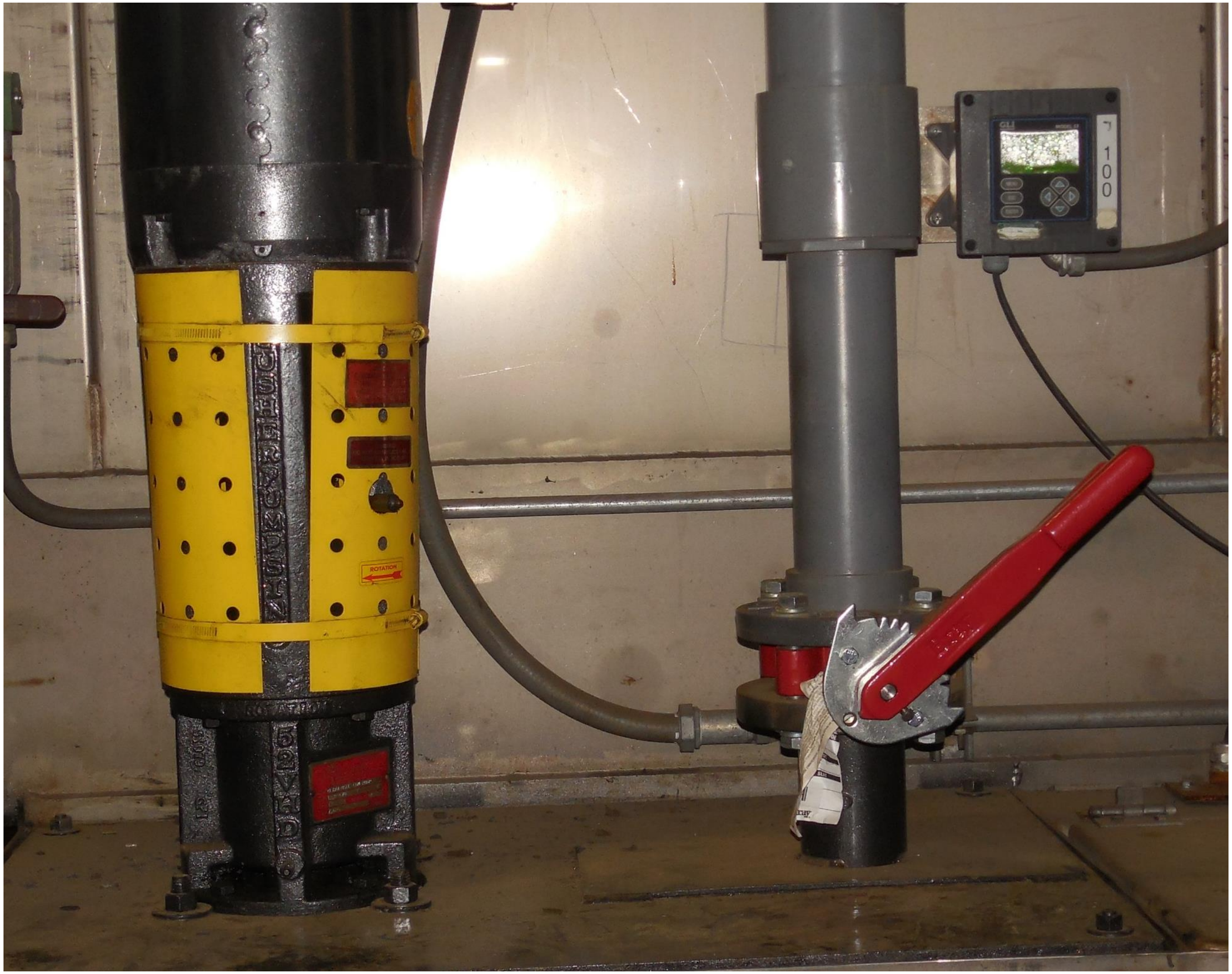
Multistage Boiler Feedwater Pumps



Typical pump isolation valves

Non-rising stem gate valve and swing check valve at the pump discharge.

Butterfly valves can isolate or throttle



The End for Session 1