Pumping System Head Estimation

These spreadsheets highlight losses, uncertainties, and value of power used, helping you optimize pumping system designs and save on operating costs.

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To design a pumping system well, you need to accurately estimate the total head, available suction total head, driver power, and energy cost.

You can estimate these quantities and their uncertainties for pumping systems with as many as 12 piping runs using a spreadsheet I've written for centrifugal pumps or a companion spreadsheet for positive displacement pumps, both of which are available at www.cepmagazine.org *(1)*. The spreadsheets handle flows in all Reynolds number regimes — turbulent, intermittently turbulent, and laminar — for fluids for which the viscosity and density can be taken as constant along a given piping run.

In this article I will describe the formulas in the spreadsheets, give tips on choosing input values, and review a sample calculation.

Formulas

First I'll cover the formulas used to calculate total head, usable head, friction head, uncertainties, the piping complexity factor, available suction total head, power required, and energy cost.

The total head h_t that a pump provides is the usable head h_u plus the friction head h_f.

$$h_t = h_u + h_f \tag{1}$$

The total head is also the discharge total head h_{td} minus the suction total head h_{ts}.

$$h_t = h_{td} - h_{ts} \tag{2}$$

The suction total head is the source usable head h_{us} minus the suction friction head h_{fs} consumed on the way to the pump.

$$h_{ts} = h_{us} - h_{fs} \tag{3}$$

The discharge total head is the discharge friction head h_{fd} consumed on the way to the destination, plus the destination usable head h_{ud}.

$$h_{td} = h_{fd} + h_{ud} \tag{4}$$

The usable head is sum of the elevation head h_{ue}, the pressure head h_{up}, and the velocity head h_{uv}.

$$h_u = h_{ue} + h_{up} + h_{uv} \tag{5}$$

The usable head is also the destination usable head h_{ud} minus the source usable head h_{us}, and its components are defined similarly. Here, the subscript "*d*" stands for "destination" and the subscript "*s*" stands for "source."

$$h_u = h_{ud} - h_{us} \tag{6}$$

$$h_{ue} = h_{ued} - h_{ues} \tag{7}$$

$$h_{up} = h_{upd} - h_{ups} \tag{8}$$

$$h_{uv} = h_{uvd} - h_{uvs} \tag{9}$$

The source usable head is the sum of the source elevation head, the source pressure head, and the source velocity head.

$$h_{us} = h_{ues} + h_{ups} + h_{uvs} \tag{10}$$

The source elevation head equals the source elevation z_s, which is the elevation of the source fluid above the eye of the pump impeller.

$$h_{ues} = z_s \tag{11}$$

The source pressure head is the head due to the source pressure p_s above the source fluid. This depends on the liquid density ρ, given the gravitational acceleration g.

$$h_{ups} = p_s/\rho g \tag{12}$$

The source velocity head is the head due to the velocity at the source vessel or branch. This depends on the volumetric flow q and the source diameter d_s.

$$h_{uvs} = \frac{\left(\dfrac{q}{\pi\left(\dfrac{d_s}{2}\right)^2}\right)^2}{2g} \tag{13}$$

In this expression, the velocity head *(2)* is calculated from the average flow without multiplying by the kinetic energy correction factor α, which is said to be approximately 1.07 for turbulent flow and is exactly 2 for laminar flow *(3)*, because α can be safely neglected when calculating total heads of pumping systems. At optimum flow velocities, α is close enough to 1 to allow α to be omitted without changing the total head *(4)*. At lower velocities, the velocity head approaches zero before α increases enough to change the total head.

The destination usable head h_{ud}, destination elevation head h_{ued}, destination pressure head h_{upd}, and destination velocity head h_{uvd} are determined by the destination elevation z_d, destination pressure p_d, liquid density, flow, and destination diameter d_d.

$$h_{ud} = h_{ued} + h_{upd} + h_{uvd} \tag{14}$$

$$h_{ued} = z_d \tag{15}$$

$$h_{upd} = p_d/\rho g \tag{16}$$

$$h_{uvd} = \frac{\left(\dfrac{q}{\pi\left(\dfrac{d_d}{2}\right)^2}\right)^2}{2g} \tag{17}$$

The friction head h_f is the sum of the equipment, instrument, and piping specialty head h_{fe}, the straight pipe head h_{fp}, and the fitting and valve head h_{ff}.

$$h_f = h_{fe} + h_{fp} + h_{ff} \tag{18}$$

The friction head is also the suction friction head h_{fs} plus the discharge friction head h_{fd}, and its components are defined similarly. Here, the subscript "*s*" stands for "suction" and the subscript "*d*" stands for "discharge."

$$h_f = h_{fs} + h_{fd} \tag{19}$$

$$h_{fe} = h_{fes} + h_{fed} \tag{20}$$

$$h_{fp} = h_{fps} + h_{fpd} \tag{21}$$

$$h_{ff} = h_{ffs} + h_{ffd} \tag{22}$$

The equipment, instrument, and piping specialty heads are generally available as known pressure drops at given flows and densities. In preliminary design, you might allow a 10 psi pressure drop for a heat exchanger, control valve, or spray nozzle; in detailed design, you generally know the pressure drop of a piece of equipment, an instrument, or a piping specialty component at conditions close to those in the pumping system. If the pumping system conditions exactly match the known conditions, you can use the known conditions directly.

If the pumping system conditions don't exactly match the known conditions, you can usually count on the number of velocity heads of the component — its k value — staying about the same, because two conditions will usually be met. First, the component's characteristic variation in head with the Reynolds number Re usually is more like that of a fitting or valve than it is like that of straight pipe. Second, the component flows will usually stay within a flow regime where k varies little, which is the case either in the turbulent regime ($Re > 5{,}000$) *(5)* or at the high end of the laminar regime ($500 < Re < 2{,}100$) *(5, 6)*. (Re is defined in Eq. 28.) Then, you can safely extrapolate from the known pressure drop Δp_0, known flow q_0, and known density ρ_0 to the head at a new flow q.

$$h_{fe} = (\Delta p_0/(\rho_0 g))(q/q_0)^2 \tag{23}$$

The straight pipe head of a given pipe run h_{fp} is calculated from the Fanning friction factor f, length l, diameter d, and velocity v *(5)*.

$$h_{fp} = (4f\, l/d)\, v^2/2g \tag{24}$$

The Fanning friction factor is estimated from the relative equivalent sand roughness Δ_r and Re using Chen's relatively accurate explicit representation of the Colebrook equation for turbulent flows *(7)* and using Churchill's interpolating formula for intermittently turbulent flows and laminar flows *(8)*.

For $Re > 5{,}000$:

$$f = 0.25\left(-2\log\left(\frac{\Delta_r}{3.7065} - \frac{5.042}{Re}\times \log\left(\frac{\Delta_r^{\,1.1098}}{2.8257} + \frac{5.8506}{Re^{0.8981}}\right)\right)\right)^{-2} \quad (25)$$

For $Re \le 5{,}000$:

$$f = 2\times\left[\left(\frac{8}{Re}\right)^{12} + \left(\left(2.457\ln\left(\frac{1}{\left(\frac{7}{Re}\right)^{0.9} + 0.27\Delta_r}\right)\right)^{16} + \left(\frac{37,530}{Re}\right)^{16}\right)^{-\frac{3}{2}}\right]^{\frac{1}{12}} \quad (26)$$

The relative equivalent sand roughness is calculated from the absolute equivalent sand roughness ε and the diameter.

$$\Delta_r = \varepsilon/d \quad (27)$$

The Reynolds number is calculated from the density, diameter, velocity, and viscosity μ.

$$Re = \rho dv/\mu \quad (28)$$

The fitting and valve heads are the entrance and exit heads h_{ffn} and h_{ffx}, increaser and reducer heads h_{ffi} and h_{ffr}, and heads for other fittings and for valves other than throttling control valves h_{ffo}.

The entrance head is estimated from the Reynolds number and the velocity *(9)*. The exit head — defined logically here as the velocity head lost to friction when the exiting flow discharges above the liquid surface or enters a vessel below the liquid surface — is calculated from the flow and the exit pipe diameter.

For a flush entrance:

$$h_{ffn} = \left(\frac{160}{Re} + 0.50\right)\frac{v^2}{2g} \quad (29)$$

For a dip pipe entrance:

$$h_{ffn} = \left(\frac{160}{Re} + 1.0\right)\frac{v^2}{2g} \quad (30)$$

For an exit:

$$h_{ffx} = \frac{\left(\dfrac{q}{\pi\left(\dfrac{d}{2}\right)^2}\right)^2}{2g} \quad (31)$$

The increaser head or reducer head of a given reducer is estimated from the inlet Fanning friction factor f_1, inlet and outlet diameters d_1 and d_2, inlet velocity v_1, and inlet Reynolds number Re_1 *(10)*.

For $Re_1 > 4{,}000$:

$$h_{ffi} = \left(1 + 3.2f_1\right)\left(1 - \left(\frac{d_1}{d_2}\right)^2\right)^2 \frac{v_1^2}{2g} \quad (32)$$

For $Re_1 \le 4{,}000$:

$$h_{ffi} = 2\left(1 - \left(\frac{d_1}{d_2}\right)^4\right)\frac{v_1^2}{2g} \quad (33)$$

For all Re_1:

$$h_{ffr} = \left(0.1 + \frac{60}{Re_1}\right)\left(\left(\frac{d_1}{d_2}\right)^4 - 1\right)\frac{v_1^2}{2g} \quad (34)$$

The heads for other fittings and for valves other than throttling control valves are estimated from Re, d, and v using the constants k_m, k_i, and k_d *(11)*, the constants k_1 and k_{inf} *(9)*, or constant k values *(12–14)*.

$$h_{ffo} = \left(\frac{k_m}{Re} + k_i\left(1 + \frac{k_d}{\left(\frac{d}{in.}\right)^{0.3}}\right)\right)\frac{v^2}{2g} \quad (35)$$

$$h_{ffo} = \left(\frac{k_1}{Re} + k_{inf}\left(1 + \frac{1}{\left(\frac{d}{in.}\right)}\right)\right)\frac{v^2}{2g} \quad (36)$$

$$h_{ffo} = kv^2/(2g) \quad (37)$$

The term $d/in.$ is the dimensionless diameter; when $d = 2$ in., for example, $d/in. = 2$ in.$/in. = 2$. The constants used in the spreadsheets are listed in Table 1.

The uncertainties in straight pipe heads and in fitting and valve heads are estimated as percentages of the heads *(14)*. The uncertainty percentage estimates used in the spreadsheets are listed in Table 1.

The piping complexity factor F_c is calculated from the diameter, straight pipe head, head for other fittings and for valves other than control valves, entrance head, exit head, velocity, Fanning friction factor, and length l *(15)*.

$$F_c = \frac{\dfrac{d\left(h_{fp} + h_{ffo} - h_{ffn} - h_{ffx}\right)g}{2v^2 f l} - 1}{0.347\left(\dfrac{d}{in.}\right)^{\frac{1}{2}} + 0.216} \tag{38}$$

The piping complexity factor is greater than one when the piping is relatively dense with fittings and valves, and less than one when the piping is relatively free of fittings and valves *(16)*. Typical values for F_c are 0.25 for a utility supply line outside battery limits, 0.5 for a straight pipe run, 1 for normal piping, 2 for a typical manifold, and 4 for a complex manifold.

The available suction total head h_{tsa} is the suction total head (Eq. 3) minus the vapor pressure head h_v *(2, 17)*. The vapor pressure head is calculated from the vapor pressure p_v and the liquid density.

$$h_{tsa} = h_{ts} - h_v \tag{39}$$

$$h_v = p_v/\rho g \tag{40}$$

The pumping power P_p is calculated from the density, estimated total head (Eq. 1 or 2), and flow.

$$P_p = \rho g h_t q \tag{41}$$

Table 1. Fitting and valve constants and uncertainties.

Item	3*k* Constants *(11)*			2*k* Constants *(9)*		Uncertainties *(14)*	
	k_m	k_i	k_d	k_1	k_{inf}	Value Used	Ref. Value if Different
Straight Pipe						±10%	–5%, +10%
Elbows							
90-deg long radius, *r*/*D* = 1.5, flanged				800	0.20	±30%	
90-deg long radius, *r*/*D* = 1.5, threaded	800	0.071	4.2			±25%	
90-deg standard radius, *r*/*D* = 1, flanged	800	0.091	4.0			±35%	
90-deg standard radius, *r*/*D* = 1, threaded	800	0.14	4.0			±40% for ≤2 in.	±40% for <2 in.
						±20% for >2 in.	±20% for >2 in.
Increasers						±50%	No value given
Reducers						±50%	
Tees							
Branch flow, flanged	800	0.28	4.0			±35%	
Branch flow, threaded	500	0.274	4.0			±25%	
Run-through, flanged	150	0.017	4.0			±35%	
Run-through, threaded	200	0.091	4.0			±25%	
Valves							
Ball	300	0.017	4.0			±50% for flanged	No value given
						±25% for threaded	No value given
Butterfly				800	0.25	±50% for flanged	No value given
						±25% for threaded	No value given
Check, swing-type	1,500	0.46	4.0			±30%	–80%, +200% for flanged
							±30% for threaded
Diaphragm	1,000	0.69	4.9			±50% for flanged	No value given
						±25% for threaded	No value given
Gate	300	0.037	3.9			±50% for flanged	
						±25% for threaded	
Globe	1,500	1.70	3.6			±25%	

The brake horsepower P_b is the pumping power divided by the pump efficiency η_p.

$$P_b = P_p/\eta_p \tag{42}$$

The energy cost present value e is calculated from estimates of the fraction of time the pumping system is running at design conditions t, brake horsepower, driver or motor efficiency η_d, utility rate r, real interest rate i, and design life n *(18)*.

$$e = t\frac{P_b}{\eta_d}r\frac{(1+i)^n - 1}{i(1+i)^n} \tag{43}$$

When comparing different design options, it's best to compare their total costs, including both capital and operating costs. Since the capital costs are present values, it makes sense to convert the operating costs to present values. The operating cost that is largest and that can be estimated the most accurately is the energy cost. You can convert the energy cost to a present value using the equation above by entering the long-term stock market real return as the real interest rate, entering the present utility rate, and entering the presently-allowed depreciation life, or by entering better estimates if available. The present value of the operating cost for different design options tells you how much capital cost you are justified in spending for different design options.

Choosing input values

You should choose the flow for the rating case so that it's the maximum flow needed in the near term, with no safety factor.

If the rated flow exceeds the near-term need, the operating cost rises quickly. Increasing the flow by 10% increases the friction head by 21% in turbulent flow, and by 10% even in laminar flow.

If the rated flow is too high, the reliability falls also, because wear increases and pump internal recirculation increases. Wear increases because the pump produces more head and, in unthrottled systems, produces more flow. Recirculation increases because the flows at the lower-flow operating points become lower fractions of the rated flow. Recirculation is more of a problem for pumps that have higher heads per stage, because it begins closer to the rated flow and builds more quickly to damaging levels.

When a pumping system is operated, the source level might be 5 ft higher than the value used for pump rating, and the like-new heat exchanger and straight pipe heads might be just 70% of the values used for pump rating. So the total head that the system will actually need in order to operate might be around 90% of the rating estimate, and might stay that way for years — maybe for the lifetime of the system. Besides that, the pump will very likely deliver a little more head than the rated head. In a throttled system, the extra head will need to be taken up by the throttling valve. In an unthrottled system, the flow will need to increase also. Such excesses in head and flow waste energy.

You can provide for uncertainty, wear, and future requirements easily enough without using a flow safety factor. The uncertainty is often provided for, to some extent, by allowances for corrosion in heat exchangers and in piping, which often provide more conservative safety margins than turn out to be needed for a given system. Wear is greatest for slow, low-flow, high-head pumps of soft construction that handle abrasive flows; for systems susceptible to wear, wear can be managed by choosing pumps of suitable construction or by restoring internal clearances, if needed over time *(19)*. Future capacity increases can be provided for by installing larger impellers. When you make reasonable plans to provide for future needs, and you match present designs to present needs, you'll get guaranteed savings up front.

You'll need to run the highest temperature case routinely. Temperature-induced changes in the density and viscosity do not change the total head much, but temperature-induced changes in the vapor pressure can affect the available suction total head substantially.

Increased densities at lower temperatures decrease the pressure head and friction head, but increase the power. Although the total head usually doesn't change much, the power does — it increases almost directly with the density.

Increased viscosities at lower temperatures increase the friction head, but so little that for low-viscosity flows, order-of-magnitude estimates of viscosity are normally all that are needed. Consider the system in the Figure at startup, taking the pump's head as approximately constant. For this low-viscosity system, reducing the temperature by 190°C increases the viscosity 250 times but reduces the flow only by 1/2. For high-viscosity systems with viscosities exceeding 100 cP, though, reducing the temperature increases the viscosity faster, and as a result, more accurate estimates of viscosity are needed.

Increased vapor pressures at higher temperatures decrease the available suction total head significantly. For both water and heptane, the increase in the vapor pressure head with temperature is on the order of 2% per degree F. Because of this, you need to be careful to identify the maximum temperature.

An effective vapor pressure for dissolved gas must be entered in many cases *(20)*:

• For systems with air dissolved in water supplied from above the pump, and similar systems, you can ignore the dissolved gas.

• For systems with a single dissolved gas at a low pressure, or for steam-stripping systems, you can enter the source-vessel pressure as the effective vapor pressure.

• For systems with a low-solubility inert gas and low levels of a high-solubility component, like ammonia wash systems, you can enter the low-solubility inert gas's pressure as the effective vapor pressure.

• For systems with air dissolved in water lifted from below the pump, or systems with a single dissolved gas at a higher pressure, you can explicitly calculate an effective vapor pressure following the procedure described in Ref. 20.

• For systems with more than one major dissolved-gas component, such as synthesis-gas systems, you will need a process simulation estimate of the effective vapor pressure.

You can avoid entrained gas problems in most services by eliminating pockets that trap gas *(21)*. You can usually avoid problems in other services by eliminating gas sources, by venting pumps *(22)*, or by choosing pumps designed to accommodate entrained gas *(23)*.

For pump rating, make the source elevation the lowest level in normal operation. Make the destination elevation the highest level in normal operation, or make it the nozzle height if the destination is a top nozzle. Make the destination elevation the piping high point, if this is what you need to do in order to make sure that the pump can get the flow started. Make the source pressure the lowest pressure in normal operation and the destination pressure the highest pressure in normal operation.

To reduce operating costs, consider selecting equipment, instrument, and piping specialty components that have lower pressure drops, At 100 gal/min, every 5 psi of pressure drop has a present value of $1,000, given a 75% pump efficiency, 90% motor efficiency, 0.05 $/kWh average power cost, 100% onstream time fraction, 10-yr design life, and 6%/yr real interest rate.

When applicable *(24)*, a compact heat exchanger that has ribbed surfaces might need a pressure drop of only 5 psi to transfer heat without significant fouling, while a conventional tubular heat exchanger that has smooth surfaces might need to be allowed 10 psi, and even then might experience significant fouling. When operation is intermittent, the optimum pressure drops are considerably higher — more like 20 psi for a compact heat exchanger *(25)*.

A rotary control valve with a fairly straight flow path needs just 5 psi at full flow, where a globe valve with a more tortuous flow path needs 10 psi *(26)*. A variable speed drive uses about the same power as a fixed speed drive when running at the rated flow and head, but uses less power when running at lower-flow operating points or lower-head operating points.

A spray nozzle that has seven times more orifices, each 50% smaller, needs just 10 psi to atomize a given flow to a given average droplet size, whereas a spray nozzle with fewer large orifices needs 40 psi *(27)*.

After entering the diameter of a pipe run, compare the resulting velocity to the guidelines listed in the spreadsheets. Although guidelines vary, suction velocities are best kept in the neighborhood of 3 ft/s for subcooled liquids or 1.5 ft/s for boiling liquids upstream of the pump suction reducer. The slow velocity and the reducer act to straighten the flow entering the pump, which ensures that the pump can deliver its rated performance and best reliability. Guideline discharge velocities for water-like fluids vary from 6 ft/s in carbon steel piping to 10 ft/s in stainless steel piping. For intermittent processes operated only one third of the time at rated flow, optimum velocities are 50% higher.

Check the absolute equivalent sand roughness data in the spreadsheets for guidelines on your piping type and surface condition. For the conditions of the sample calculation, a change from the like-new condition to a typical moderately-corroded condition would have increased the straight pipe head by 80%.

Note the calculated piping complexity factor to check the reasonability of fitting and valve counts.

Given a preliminary pump selection, compare the available suction total head to the required suction total head h_{tsr}. For most pumps, it's safe to choose a suction total head ratio h_{tsa}/h_{tsr} of 1.3 or more. For a 3,600-rev/min pump with an inlet nozzle diameter greater than 6 in. or for an 1,800-rev/min pump with an inlet nozzle diameter greater than 10 in., it's safe to use a suction total head ratio of 2.5 or more *(28)*.

Estimate the brake horsepower and the energy cost present value using the pump efficiencies given by the spreadsheets *(29)* or entering a pump efficiency, and using the reasonable values listed in the spreadsheets for driver or motor efficiency *(30)*, utility rate *(31)*, real interest rate *(32)*, and design life *(33)*, or using better estimates if available.

When pumps are quoted, compare the efficiencies and motor sizes of the quoted pumps to the pump efficiency and brake horsepower in the spreadsheets and check whether either the more-efficient of the pumps or premium motors would save enough energy to make a larger investment worthwhile.

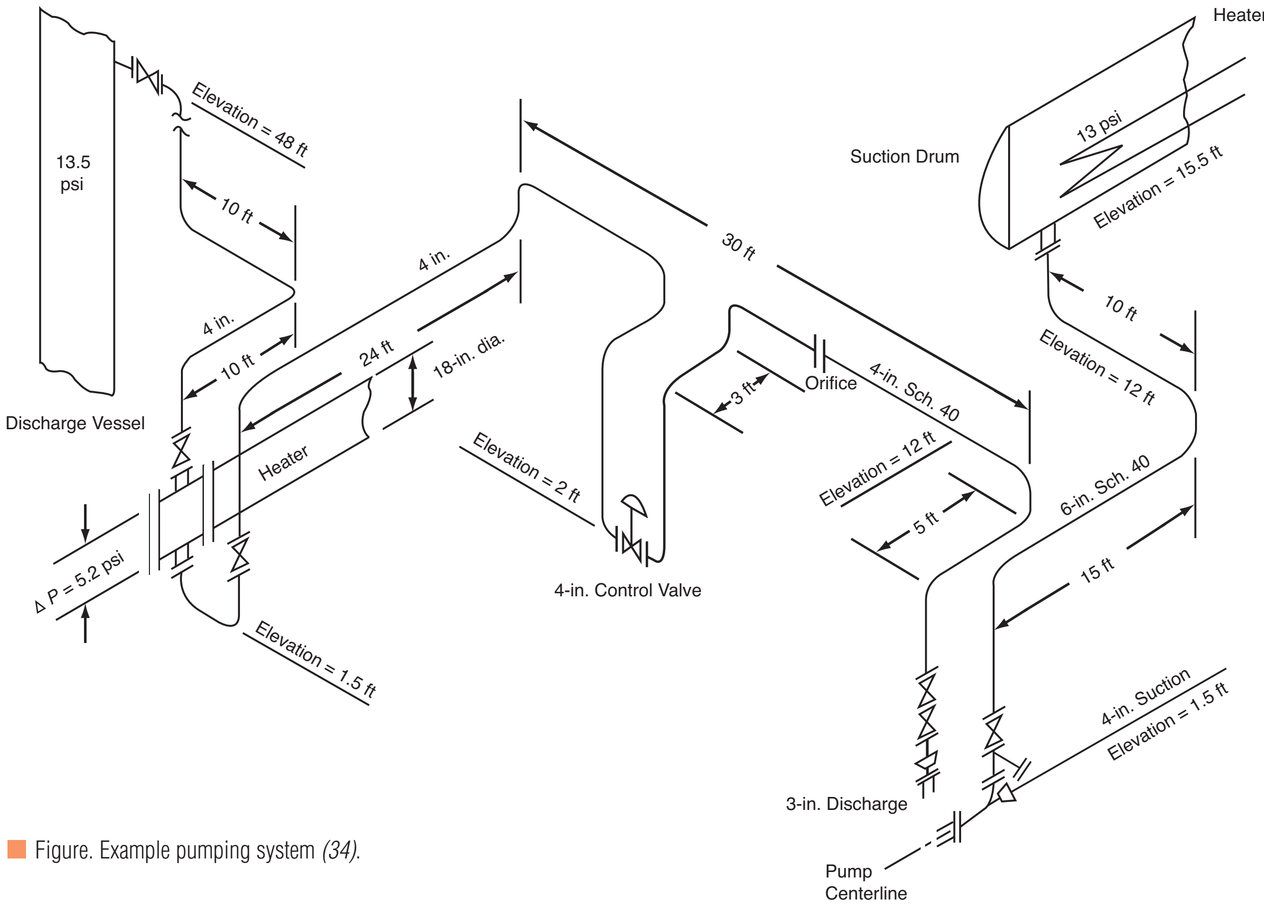

Figure. Example pumping system *(34)*.

Sample calculation

In the Figure, a centrifugal pump transfers gas oil from a suction drum to another vessel. Table 2 shows the inputs and the results.

I took the rated flow to be 250 gal/min, with no safety factor. The fluid temperature is 555°F; ρ is 64.87 lb/ft^3; μ is 0.6 cP; z_s is 14 ft; p_s is 13 psig. I took the source diameter d_s to be 100 in., which is much larger than the pump suction line. All the piping is flanged schedule 40 carbon steel. I took its absolute equivalent sand roughness ε to be 0.2 mm, which is the value listed in the spreadsheet for corroded steel pipe carrying oil. The suction line diameter is 6 in. and its length is 39 ft. The line has one flush entrance, five standard radius elbows, one gate valve, one strainer, and one 6-in. × 4-in. reducer. I took the strainer's number of velocity heads k from Ref. 14 to be 0.88. The suction line is of normal complexity.

On the discharge side, z_d is 46.5 ft and p_d is 13.5 psig. I took the destination diameter d_d to be 100 in., which again is much larger than the pipeline. The orifice plate, control valve, and heat exchanger pressure drops are 1.52 psi, 5.2 psi, and 10 psi, respectively. The discharge line diameter is 4 in. and its length is 177 ft. The line has one 4-in. × 3-in. increaser, five gate valves, 20 standard radius elbows, and a 4.026-in.-dia. exit pipe where the velocity head is lost. The discharge line is a little simpler than normal piping.

The rated total head is 83 ±3 ft. The usable head is 40% of the total head and the friction head is 60% of the total head. The available suction total head is 76 ft. At the spreadsheet-estimated pump efficiency of 66%, the brake horsepower is 8.3 hp. If the pump is onstream all but 11 days a year (t = 97%), and if the motor efficiency is 90%, the utility rate is 4.45 ¢/kWh, the design life is 9.5 yr, and the real interest rate is 6%/yr, then per Eq. 43, the present value of the lifetime energy cost is $19,000.

Conclusion

The methods described here will help you be sure of the magnitude of the head losses due to corrosion and be sure of the magnitude of the uncertainties in your head estimates, so you won't have to use safety factors you don't need.

Since you can quickly estimate operating energy costs, you'll find it easier to evaluate variable speed drives that can save up to 80% of energy costs, or

Table 2. Worksheet for single suction and single discharge *(1).* †

Atmospheric pressure	14.7	psia								
Fluid	Gas Oil									
Case	Preliminary Design Rating									
Pipeline	Suction				Discharge				Overall	
Flow	250	gal/min			250	gal/min			**250**	**gal/min**
Temperature	555	°F			555	°F			555	°F
Density	64.87	lb/ft^3			64.87	lb/ft^3			64.87	lb/ft^3
Viscosity	0.6	cP			0.6	cP			0.6	cP
Vapor pressure		psia							0.0	psia
Total head			76.1 ±0.1	ft			159.5 ±2.6	ft	**83.4 ±2.7**	**ft**
Usable head			75.5	ft			109.1	ft	33.6	ft
Elevation of source or destination above pump	14.0	ft	*14.0*	*ft*	46.5	ft	*46.5*	*ft*	*32.5*	*ft*
Pressure at source or destination	13	psig	*61.5*	*ft*	13.5	psig	*62.6*	*ft*	*1.1*	*ft*
Velocity head			*0.0*	*ft*			*0.0*	*ft*	*0.0*	*ft*
Dia. of source or destination vessel or branch	100	in.			100	in.				
Friction head			0.6 ±0.1	ft			50.4 ±2.6	ft	51.0 ±2.7	ft
Equipment, instruments, and piping specialties			*0.0*	*ft*			*37.1*	*ft*	*37.1*	*ft*
Sizing basis flow	250	gal/min			250	gal/min				
Sizing basis pressure drop		psi			16.7	psi				
Sizing basis density	64.87	lb/ft^3			64.87	lb/ft^3				
Straight pipe			*0.2 ±0.0*	*ft*			*7.8 ±0.8*	*ft*	*8.0 ±0.8*	*ft*
Nominal size	6	in.			4	in.				
Schedule 5S, 10S, 40S, 80S, PTFE, Tube, -	40S				40S					
Diameter	6.065	in.			4.026	in.				
Corroded or like new	Corroded				Corroded					
Roughness	0.2000	mm			0.2000	mm				
Relative roughness	0.00130				0.00196					
Velocity	2.8	ft/s			6.3	ft/s				
Reynolds number	225,743				340,073					
Fanning friction factor	0.0055				0.0060					
Length, ft	39	ft			177	ft				
Fittings and valves			*0.4 ±0.1*	*ft*			*5.5 ±1.8*	*ft*	*6.0 ±1.9*	*ft*
Complexity factor (0.5 simple, 1 normal, 2 complex)	1.6				0.7					
Flanged or welded, or threaded	Flanged				Flanged					
Elbows										
Long radius, r/D=1.5										
Standard radius, r/D=1	5		0.2 ±0.1	ft	20		4.1 ±1.4	ft	4.3 ±1.5	ft
Entrance ki = 0.5 if flush, 1 if dip pipe	0.5		0.1	ft					0.1	ft
Exit pipe diameter if exit velocity head is lost					4.026	in.	0.6	ft	0.6	ft
Increasers					1		0.4 ±0.2	ft	0.4 ±0.2	ft
Inlet diameter	6.065	in.			3	in.				
Outlet diameter	6.065	in.			4.026	in.				
Inlet relative roughness	0.00130				0.00262					
Inlet velocity	2.8	ft/s			11.3	ft/s				
Inlet Reynolds number	225,743				456,377					
Inlet Fanning friction factor	0.0055				0.0064					
Reducers	1		0.1 ±0.0	ft					0.1 ±0.0	ft
Inlet diameter	6.065	in.			4.026	in.				
Outlet diameter	4.000	in.			4.026	in.				
Inlet velocity	2.8				6.3	ft/s				
Inlet Reynolds number	225,743				340,073					
Tees										
Branch-flow										
Run-through										
Valves										
Gate	1		0.0 ±0.0	ft	5		0.4 ±0.2	ft	0.4 ±0.2	ft
Globe										
Other fitting or valve k value	0.88		0.1	ft					0.1	ft
Available suction total head			**76.1 ±0.1**	**ft**						
Brake horsepower									**8.3 ±0.3**	**hp**
Pumping power									5.5 ±0.2	hp
Pump efficiency									66	%
Energy cost present value									**18,527 ±608**	**$**
Fraction of time at design conditions									97	%
Motor efficiency									90	%
Power cost									0.0445	$/kWh
Facility design life									9.5	yr
Real interest rate									6	%/yr

† Underlines prompt users to enter values, such as fitting quantities, or to check values suggested by the spreadsheet. Calculated values of pumping system heads are shown at the right in the "Suction" and "Discharge" columns and are summarized in the "Overall" column. The spreadsheets include additional header lines, valve types, and optional and explanatory worksheets that are not shown in this table.

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unequal-size or same-size parallel pumps that can save up to 50%, or retrofit pumps that can save up to 25%, or larger pipe sizes and lower-head components that can save up to 20% *(35, 36)*.

By improving pumping systems, a typical plant will save energy worth at least $700,000 — that is, $90,000/yr *(35)* over a 10-yr design life *(33)* at an average 6%/yr real interest rate *(32)*. Your plant should be saving this much, too. CEP

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