

Pressure Changers

In this section, we cover two model libraries: the pressure changers. The modules in this library deal with parts of the process that has an effect on changing the pressure either directly, such as pumps or compressors, or through its operation, such as pipes and pipelines.

It is usually required when handling fluid streams to change its pressure for different reasons. For example, in the gas sweetening plants, an absorption column might be needed to operate at high pressures to improve the absorption and a pump might be used to compress the inlet fluids to the desired value. On the other hand, the regeneration column must be operated at near atmospheric conditions to allow the adsorbed gases to leave the liquid. The liquid which comes from the absorption column at high pressures is passed through a valve to achieve the design pressure.

Other operations require transportation of fluids. During transportation, hydrostatic pressures and friction causes a pressure drop. Therefore, pumps and compressors are needed to provide the required energy, by increasing the pressure, to the required level. The friction losses in pipes are a common cause for pressure drop, especially when with long pipelines. Therefore, pressures drops must be calculated to determine the required pumping through the pipelines, which is a common calculation when dealing with pipe networks (specifically for oil and gas collecting systems).

Pumps

In general, a pump is a device used to transport liquids, gases, and slurries. However, the term pump is usually used to refer to liquid handling equipment (this is true with Aspen Plus). The purpose of the pump is to provide a certain pressure at certain flow rate of a process stream. The pressure requirement is dictated by the process and piping involved, while the flow rate is controlled by the required capacity in the downstream units.

There are several types of pumps used for liquid handling. However, these can be divided into two general forms: positive displacement pumps (including reciprocating piston pump and the rotary gear pump), and centrifugal pumps. The selection of the pump type depends on many factor including the flow rate, the pressure, the nature of the liquid, power supply, and operating type (continuous or intermittent). Figure 30 shows a general guideline to selecting pump type based on flow rates and discharge pressure. Centrifugal pumps, such as the one shown in Figure 31, are by far the most widely used type in the chemical process industry, with other types employed for special process specifications (*e.g.*, high pressures).

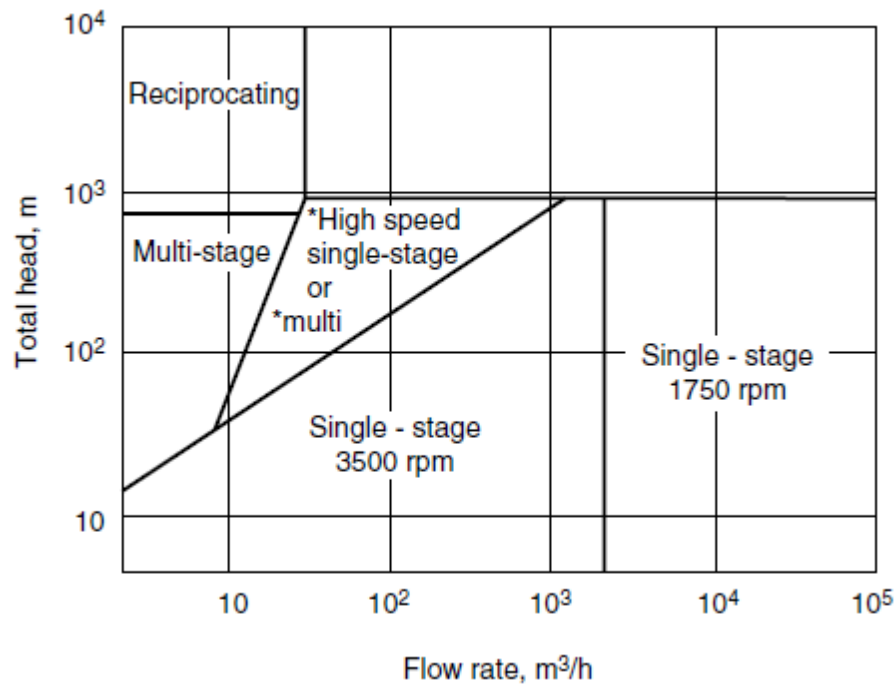


Figure 30. Pump selection guide. ²

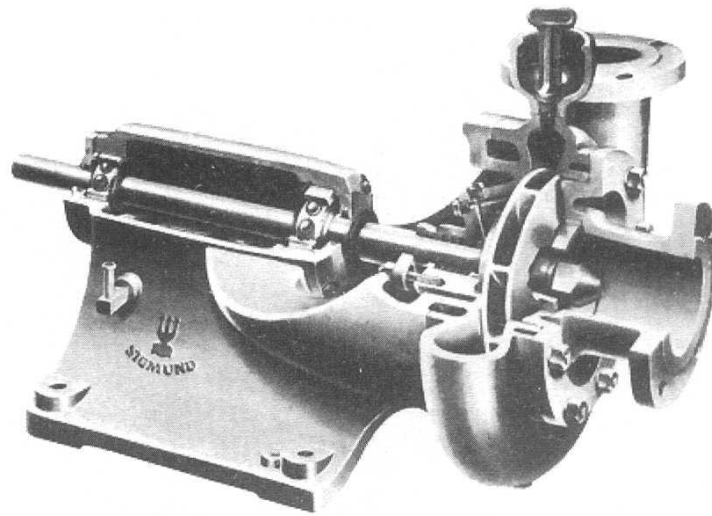


Figure 31. Section of centrifugal pump showing the inlet (horizontal), the outlet (vertical), the impeller, and the shaft connecting the impeller to the motor. ³

The power requirement for a mechanical system, like pumps and compressors, is given by the general mechanical balance equation:

$$\mathbf{P} = -mW_s = m \left(\Delta \left(\frac{u^2}{2\alpha} \right) + g\Delta z + \int_{P_1}^{P_2} v dP + F \right) \quad (1)$$

² R. K. Sinnott, John Metcalfe Coulson, and John Francis Richardson, *Coulson & Richardson's Chemical engineering Design*, vol. 6, 4th ed. (Butterworth-Heinemann, 2005).

³ J R Backhurst et al., *Chemical Engineering Volume 1: Fluid Flow, Heat Transfer and Mass Transfer* v. 1, 6th ed. (A Butterworth-Heinemann Title, 1999).

All terms in this equation take their normal meaning with m being the mass flow rate, and α a coefficient used to take into account the velocity profile inside the pipe (for laminar $\alpha = 0.5$, while for turbulent $\alpha = 1$). The required work (or power) given by $\mathbf{P}$ is the total work that needs to be delivered to the fluid. This work will be drawn from a motor (operated with electricity or engines). The conversion between the motor and pump power is not complete and an efficiency is defined to describe the power conversion. The efficiency is given by:

$$\eta = \frac{P_{out}}{P_{in}} \quad (2)$$

The input power can be measured from the source. For example, if the pump is operated with electricity, the input power will be $I \times V$ (current times voltage). The outlet power can be determined using Equation (1).

Notices that Equation (1) takes into account all power requirements: from the kinetics, potential, pressure differences, and any friction losses. The term "Head" is used to express the different parts of the power, or energy, requirements. The head is a measure of how high the fluid can be reached and is usually expressed in length units (m, ft) which. The pump head is generally divided into three parts:

1. Static head (Δz term): the height to which the fluid will be pumped.
2. Pressure head ($\int v dP$ term): the pressure to which the fluid will be delivered (in a pressurized vessel for example). The pressure units must be converted to length units using $P = \rho gh$ relation.
3. System or dynamic head (F term): the energy lost due to friction in pipes, valves, fittings, etc.

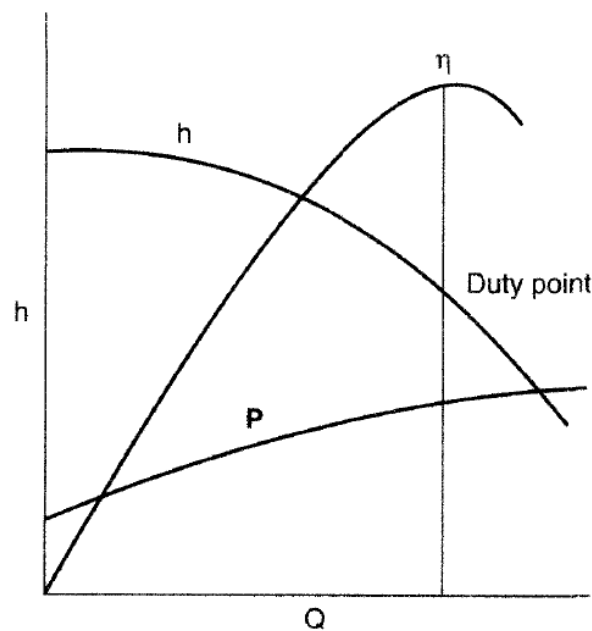


Figure 32. Typical characteristic curve for a pump.

As the pump works, it can convert its energy into kinetic energy (velocity) or pressure head. The relation between the flow (kinetic) and the pressure head is usually expressed using a pump characteristic curve that shows the head developed against the flow rate. The characteristic curve is a function of the pump design (impeller diameter and rpm). A typical characteristic curve is shown in Figure 32. The figure also shows the power requirement and pump efficiency of the pump as a function of flow rate. At a certain flow rate, the pump efficiency will be a maximum, and this will be referred to as the best efficiency point (B.E.P). This point represents the ideal combination of flow rate and head at which the pump can be operated. In other words, the maximum amount of power input is converted into the fluid.

Pump manufacturers usually supply a characteristic diagram with the pump model. A typical diagram is shown in Figure 33. The diagram shows different head-flow rate curves and the corresponding efficiencies. You can see from this diagram that the efficiency raises as the pressure (or head) is increased then it falls again, as indicated earlier. These curves can be used to select the best pump for a given operation.

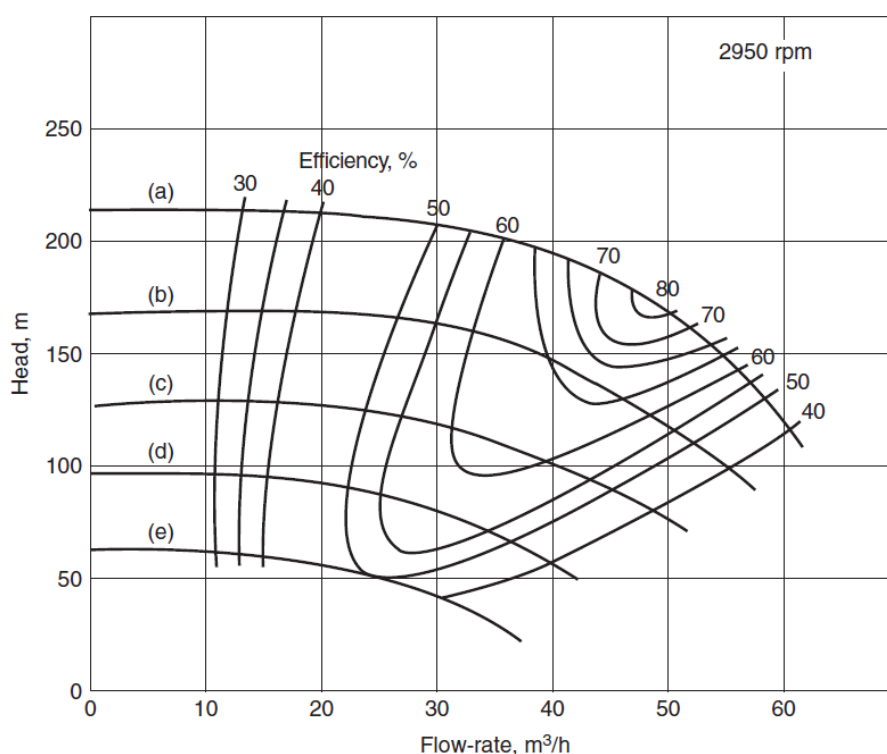


Figure 33. A typical characteristic diagram. Lines (a) through (e) represent decreasing impeller diameter. Notice the curve is given at 2950 rpm operation.

Consider, for example, a process in which you want to pump $40 \text{ m}^3/\text{h}$ of a fluid against a 150 m head. Then, according to Figure 33, impeller (b) will give an efficiency of about 62% with almost the required conditions. If we increase the head to 190 m , impeller (a) can be used with slightly higher efficiency. The choice of the impeller will thus depend on the process conditions.

Another important definition for pump operation the net positive section head (NPSH). To understand the importance of the NPSH, let us consider first the pump operation. The inlet of the pump is referred to as the suction side while the outlet is referred to as the discharge side. At the suction side, we should ensure that we have enough pressure to prevent vaporization of the liquid by ensuring the pressure is higher than the vapor pressure. This is especially important since the movement of the impeller will create low pressure regions that can drop below the vapor pressure. Presence of vapors in the pump will cause cavitation, which can damage the impeller.

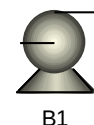
The NPSH is defined as the pressure at the suction side of the pump above the vapor pressure of the fluid. This pressure comes from the tanks and piping arrangements upstream of the pump. This can be expressed as:

$$NPSH_{avail} = \frac{P}{\rho} + H - \frac{P_f}{\rho} - \frac{P_v}{\rho} \quad (3)$$

where the NPSH term is designated as "avail" which refers to the available suction. The H term refers to the hydrostatic pressure available at the suction, the P_f is the friction losses in the suction side piping, and the P_v is the vapor pressure of the liquid. In contrast, the required NPSH, referred to as $NPSH_{req}$, is a design parameter calculated based on the process conditions. The $NPSH_{avail}$ must be higher than the $NPSH_{req}$ under all operating conditions, and the piping configuration must be designed to achieve this.

Defining Pumps in Aspen Plus

Pumps can be added from the Pressure Changers library. A pump takes one or more input streams and one product stream. It can also take a work stream. The pump specification is given through the Setup windows (accessed by double clicking on the pump). As shown in Figure 34, there are three different ways to define the pump performance: pressure, power, or performance curves. The first two options can be input directly, while the third is done in a separate window (Performance Curve window). The performance curve can be input as either data (usually available from the manufacturer), a 4th degree polynomial, or written as an equation using Fortran.



The minimum input needed for a pump is one of the direct specifications (either pressure related or power). With this input, Aspen Plus will calculate many of the pump parameters including:

1. Fluid horsepower (FHP): this is the weight of fluid being pumped multiplied by the head across the pump.
2. Brake horsepower (BHP): is the actual power required by the pump and is equal to FHP divided by the pump efficiency.
3. Electricity: electric power needed to drive the pump, and is equal to BHP divided by the drive efficiency.
4. NPSH available.

5. NPSH required: in order to calculate this value, you have to supply some information about the specific speed (roughly, the rpm of the impeller multiplied by the flow rate and divided by the head) and the suction specific speed (which is an index used for centrifugal pumps with values ranging from 6,000 to 12,000, where 8,500 is a typical value). These can be input in the **Setup | Calculation Options** tab.

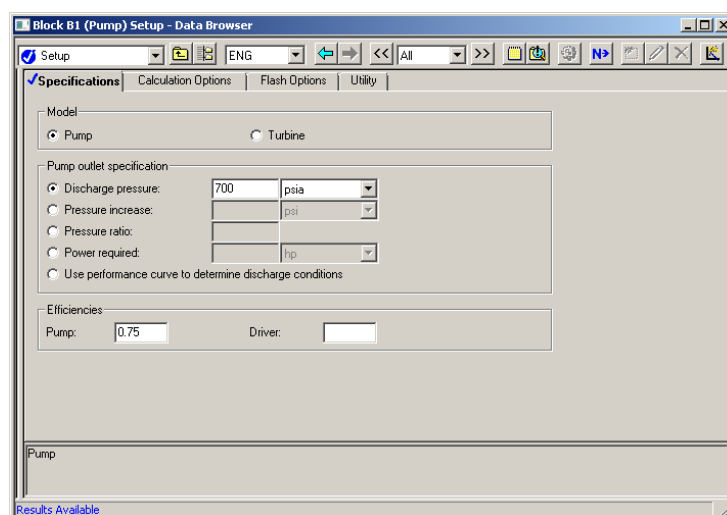


Figure 34. Setup window for pumps.

Let us consider setting up a pump where the manufacturer has supplied the following information: pump efficiency: 60%, drive efficiency: 90%, and a characteristic curve given in the table to the right. Benzene is being fed to the pump at a rate of 8000 kg/hr at 40.0 °C and 1 bar. Try inputting these data into Aspen Plus simulation (what property method do you suggest using here?).

Head (m)	Flow (m ³ /hr)
40	20
250	10
300	5
400	3

Once the simulation runs you will see that results obtained in the Results page. The results show the $NPSH_{avail}$ to be about 9.02 m. If you want to convert this into pressure use the $P = \rho gh$ relation, with ρ for benzene at the inlet conditions (which can be looked up as we learned previously). If you do the calculation you get 0.77 bar. The vapor pressure of benzene at the inlet condition is approximately 0.1 bar. The results also show that 10.3 kW of electricity is needed. We can calculate how much this will cost us based on electricity in Jordan as follows.

First, let us define a unit price of JD/kW-hr. To do so, go to the **Setup | Custom Units**, and define a new unit for electricity price (Elec-Price) as JD/kWhr based on \$/kWhr with a multiplier of 0.71 (conversion between \$ and JD). Also, create a cost rate (Cost-Rate) unit as JD/hr based on \$/hr with a multiplier of 0.71. Now, we can define the electric price. This is done in the Pump | Setup window by clicking on the utility tab and defining a new utility (call it Elec). Once you define the new utility, go to the **Utilities | Elec** page and input the prices for electricity in the newly defined units as 0.05 (the price of electricity for small industrial facilities in Jordan is

currently 50 fils/kW-hr). Now run the simulation and check the results for how much this pump will cost per year (you should get about 4500 JD/year).

Additional information can be added if we know more about the pump or process. For example, if we know that the pump has a 3" suction opening, then we can input the suction area in the Setup | Calculation Options tab as 45.6 cm². This value will be used to modify the NPSH for the kinetic effect (*i.e.* calculating the velocity at suction). Doing so will increase the $NPSH_{avail}$ to 9.03 meters (which means that the kinetic energy gives the fluid more head). If we know how much hydraulic static head is available, then we can take this pressure into account in our calculations.

You will notice that the required $NPSH$ has not been calculated. This is, as mentioned before, is because we need to input the suction specific speed that must be supplied by the manufacturer. Input a typical value of 9,000 for this item and check the required $NPSH$. Compare the values to the recommendations from the Hydraulic Institute shown in the table to the right.

Suction Energy	$NPSH_{Margin}$ Ratio ($NPSH_A/NPSH_R$)
Low	1.1 - 1.3
High	1.2 - 1.7
Very High	1.7 - 2.5

Compressors

The principles of gas compression are similar to that of liquid pumping. The main difference lies in the mechanical design of the equipment due the differences in the physical properties between gases and liquids Common compressor types are shown in Figure 35. Since gases are much more compressible than liquids, the compression process gives raise to higher temperature and volume changes in the gas.

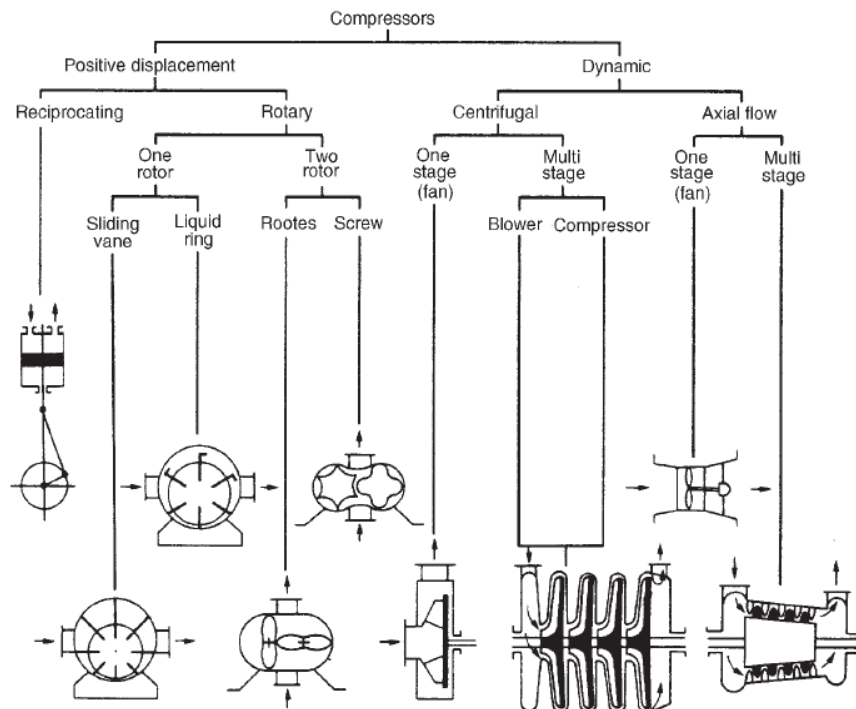


Figure 35. Types of compressors.⁴

In Equation (1) the term $\int v dP$ was calculated for liquids by assuming v is independent of P , thus yielding the $\Delta P/\rho$ term. For gases, this assumption cannot be made, and the path (relation between v and P) must be specified. This path is commonly expressed in terms of the relation:

$$PV^n = \text{constant} \quad (4)$$

This relation between P and V is termed a polytropic process. Special cases of this equation are the isothermal process ($n = 1$) and the isentropic process ($n = \gamma (= C_p/C_v)$). Both the polytropic and isentropic process are ideal processes and must be corrected to account for real processes. This is done through the use of efficiencies (polytropic and isentropic).

The isentropic efficiency refers to the deviation from the reversible, adiabatic work (constant entropy work), and is calculated as:

$$(5) \quad \eta_{is} = \frac{(P_2/P_1)^{(\gamma-1)/\gamma} - 1}{T_2/(T_1 - 1)}$$

where the subscripts 1 and 2 refers to the inlet and outlet conditions, respectively. We notice here that the efficiency will depend on the pressure ratio (P_2/P_1), which will make it necessary to adjust the efficiency for each pressure ratio. The polytropic efficiency can be used to overcome this drawback. In this case, a value of n is specified for the compressor and the polytropic efficiency is defined as:

$$(6) \quad \eta_p = \frac{n-1}{n} \frac{\gamma}{\gamma-1}$$

In both cases, the efficiency takes care of the irreversibilities and losses in the compressor. The efficiency can be used to calculate the energy balance as:

$$(7) \quad \eta \Delta H = \int v dP$$

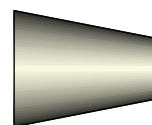
Once the enthalpy change is calculated, the brake horsepower (*BHP*) is given by:

$$(8) \quad BHP = m(\eta \Delta H)/\eta_m$$

where η_m is the mechanical efficiency of the compressor. The term $m(\eta \Delta H)$ is called the indicated horsepower (or *IHP*), and is the energy required to pressure the fluid.

Defining Compressors in Aspen Plus

A compressor in Aspen Plus takes similar inputs and output material and work streams as that of a pump. The specifications sheet for the compressor is shown in Figure 36. Unlike pumps, here you need to specify the compressor operating conditions, based on



⁴ Sinnott, Coulson, and Richardson, *Coulson & Richardson's Chemical engineering Design*, vol. 6. Refer to page 478 of this reference for typical operating conditions of each type.

experience or manufacturer recommendations and information. The outlet conditions can be either input directly as pressure specification, determined from input power, or determined from performance curves. If the last option is used, the performance curve must be given in Performance Curves sheet in a similar manner to that for the pump.

The pump efficiency can be input directly as a scalar quantity, or it can be calculated from efficiency curves. In both cases, the efficiency will have a direct effect on BHP calculations for the compressor.

Figure 36. Compressor specifications sheet.

Consider for example an isentropic compressor with a performance curve as shown in the table to the right. The operating and reference shaft speed for the pump is 5000 rpm. If we setup such compressor such that it processes $70 \text{ m}^3/\text{hr}$ of O_2 at -10°C and 1 bar, we will see that Aspen Plus calculates the net work required as 0.21 kW at 72% efficiency (try to figure out how Aspen Plus calculated this number), and a outlet pressure of 7.4 bar, temperature 292°C , and an isentropic temperature of 213°C .

Head (m)	Flow (m^3/hr)
10000	1
20000	2
30000	3
40000	4

Valves

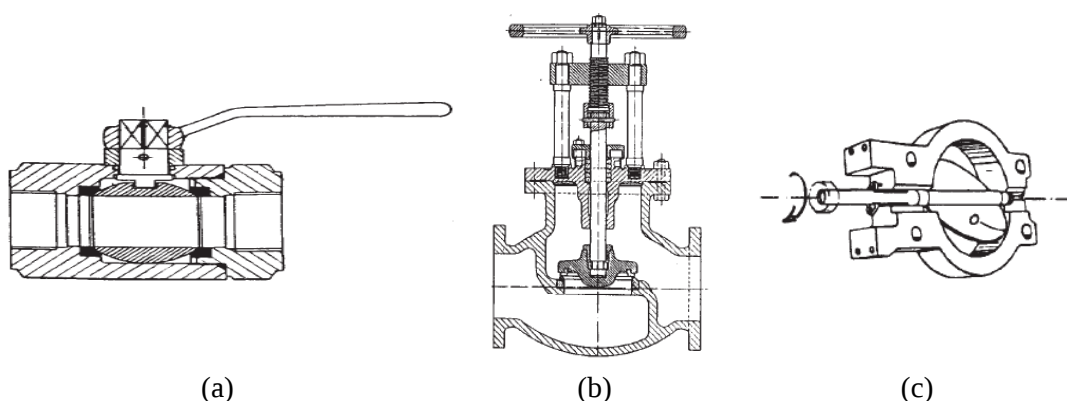


Figure 37. Different types of valves: ball (a), globe (b), and butterfly (c).

Valves used in the process industry are either shut off valves (used to close the flow) or control valves (used to adjust the flow either manually or automatically). Several types of valves exist such as the gate, plug, ball, globe, diaphragm, and butterfly valves. Some examples are shown in Figure 37. Shut off valves are designed to give good mechanical sealing when closed while having low resistance to flow when open. Gate, plug, and ball valves are typically used for this application. For flow control, it is important to be able to control the flow smoothly from fully open to fully close. For this purpose, globe valves are commonly used. Butterfly valves are used for vapor and gas flows.

As the fluid passes through the valve, there will be a direct relation between the pressure drop across the valve and the flow rate. This relation is usually expressed using the valve flow coefficient (C_v) which is defined as the number of gallons per minute of water at 60 °F that will pass through the valve with a pressure drop of 1 psi. The relation between flow rate and pressure drop is given by:

$$(9) \quad W = F_p C_v \sqrt{\rho_{in} (P_{in} - P_{out})}$$

where W is the mass flow rate and F_p is a piping geometry factor. A similar expression is used for gases with an expansion factor (Y) used in place of C_v .

According to Equation (9), the flow rate will increase with increasing pressure drop. However, there is a limit for pressure drop after which the flow will not increase any more. At this level of operation the valve is said to be choked, and the flow rate is termed the choked flow. A representation of flow dependence of pressure drop is shown in Figure 38. This phenomenon can be useful sometimes as it allows control of flow rate independent of downstream pressure.

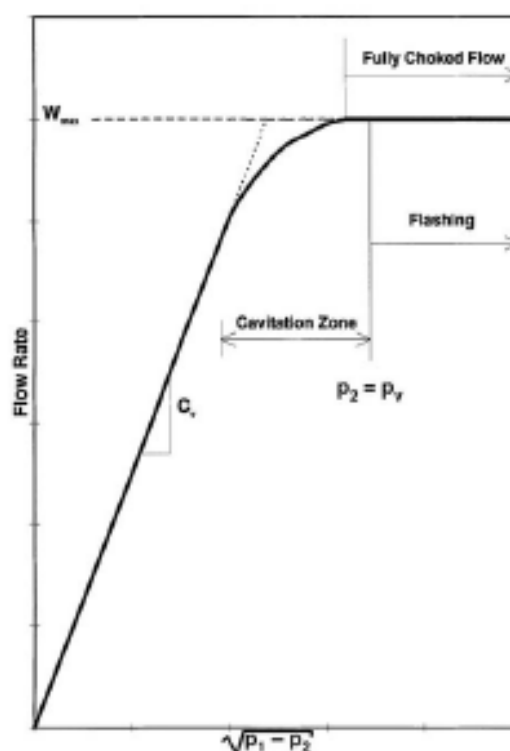
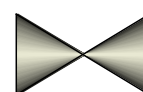


Figure 38. Liquid flow rate versus pressure drop (assuming constant p_1 and p_v).⁵

Valves in Aspen Plus

The valve module is accessible from the pressure changers library in Aspen Plus. The valve takes only one input and one output streams. The specifications for the valve are made in its **Input | Operation** sheet as shown



⁵ Don Green and Robert Perry, *Perry's Chemical Engineers' Handbook, Eighth Edition*, 8th ed. (McGraw-Hill Professional, 2007), 8-82.

in Figure 39. As you can see from in Operation tab, there are three options when modeling valves: direct specification of pressure and performing adiabatic calculations, valve sizing or design, or valve rating.

In the first case, you need to input the outlet pressure from, or the pressure drop across, the valve. The calculations will then be made assuming adiabatic operation ($\Delta H = 0$). In this case, the valve will act as a pressure changer with out any insight into the flow restricting action fo the valve. Try, for example, to model a valve with 1000 kg/min of water inlet at 200 °C and 10 bar, and specify the pressure drop as 1.5 bar (use ASME steam table correlations). The program will calculate the exit temperature of the stream assuming an adiabatic operation giving 192 °C.

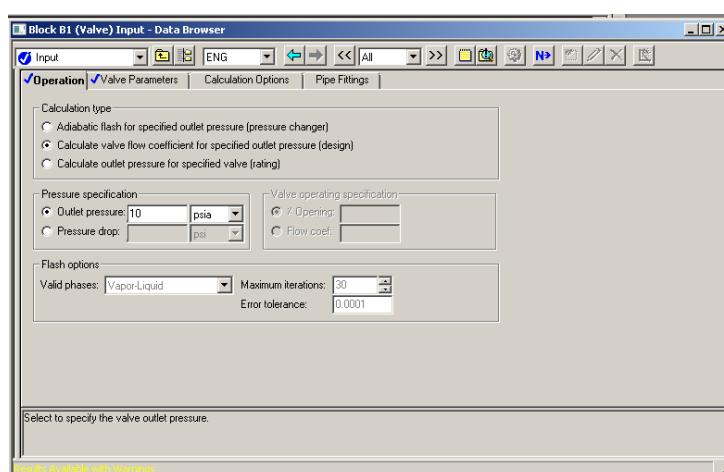


Figure 39. Valve specifications sheet.

In the second case, a specific outlet pressure is given by the user together with certain characteristic values for the valve, and the program will caculcate the valve flow coefficient, the choked outlet pressure, and the valve openining percentage. These values can then be used to specify the needed valve for the process based on its characterisitcs. If you repeat the previous system and select the "Calculate valve flow coefficient..." option, the **Input | Valve** Parameters tab will become available. In this tab you need to input C_v , pressure drop ratio factor (X_T), and pressure recovery factor (F_L). The pressure drop ratio factor accounts for the effect of the internal geometry of the valve on the change in fluid density as it passes through the valve, while the pressure recovery factor accounts for the effect of the internal geometry of the valve on its liquid flow capacity under choked conditions. Either of these values can be input as a table versus valve opening, or as a constant value.

Repeat the last simulation of water using the following information this time: $C_v = 1500$, $x_T = 0.8$, and $F_L = 0.90$. Use a linear characteristic equationo for C_v (check out the help to see what does this mean). If you examine the results you will see that the pressure drop for this case was calculated to be 2.96 bar, about 0.90 bar above chok pressure, and the required valve opening is 65%. As in the first case, the outlet temperature decreased to 192 °C.

In the third case, similar information is needed for the valve parameters. However, here the valve opening or flow coefficient are supplied by the user and the program will calculate the pressure drop across the valve. You can see how this is used by simulating the flow of 5000 gal/min of water at 50 °C and 10 bar through a valve at 60% opening that has similar factors as the previous case. In this case, we see that the valve will induce a 1.6 bar of pressure drop (still above the chock pressure).

Pipes and Pipelines

The flow of fluids inside pipes causes friction losses which affects the fluid pressure. Therefore, it is important to take into account the effect of piping systems and accommodate proper pumping operation for downstream units. In addition, as fluid flows into the pipe, it can exchange heat with the surrounding which affects its temperature.

The calculations of pressure drop across pipes depend on the pipe length and geometry, connections, fittings, and any contractions or expansions in the pipe. Correlations are available to describe the frictional losses in the pipe to which the pressure drop is related. Fittings effect on pressure drop is usually considered in friction calculations as an increase in the pipe length.

When the fluid inside the pipe is a single phase, it will fully fill the inside of the pipe. The calculations of the friction loss in this case will depend mainly on the flow conditions, physical properties of the fluid, and the pipe characteristics. On the other hand, if two phases exist inside the pipe, then it is important to determine the type of flow. There is large number of studies on the subject that attempts to identify the flow pattern based on flow characteristics. An example of flow patterns is shown in Figure 40. As shown in the figure, the flow becomes segregated between the two

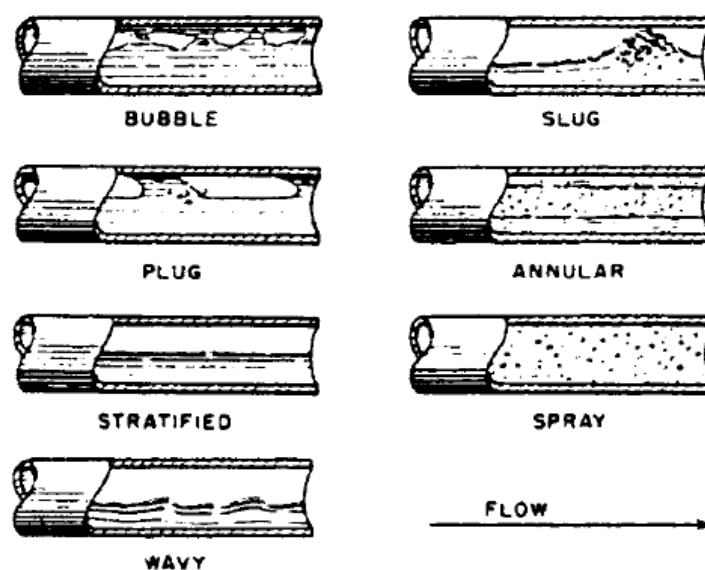


Figure 40. Gas/liquid flow patterns in horizontal flow.⁶

⁶ Ibid., 6-26.

phases which affect friction calculations. Also, since the gas has much lower density than the liquid, its superficial velocity can be higher resulting in liquid hold up in the pipe. Correlations present to calculate the liquid hold up. Of special importance when dealing with multi-phase flow is the orientation of the pipe (horizontal, vertical, and inclined) which should be taken into account in the calculations.

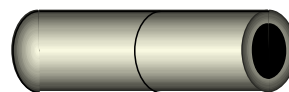
The heat effect on fluids passing through a pipe can be determined based on energy balance. The energy balance is simply given by:

$$(10) \quad \Delta \dot{H} = UA\Delta T + \dot{Q}$$

for which the heat transfer coefficient and ambient temperature (for ΔT calculations) must be specified.

Pipes & Pipelines in Aspen Plus

There are two modules available for modeling pipes in Aspen Plus: Pipes and Pipelines. The pipes module is used to model single segments of pipes with any associated fittings. Pipelines, on the other hand, are used to model larger pipes networks such as those encountered with oil and gas collecting systems.



Pipe specifications can be made in the **Setup | Pipe Parameters** tab for the pipe block. This sheet takes basic input about the pipe geometry. Temperature calculations are setup in the **Setup | Thermal Specification** tab where the choice on how to calculate the temperature profile is made. In this terms to include in the energy

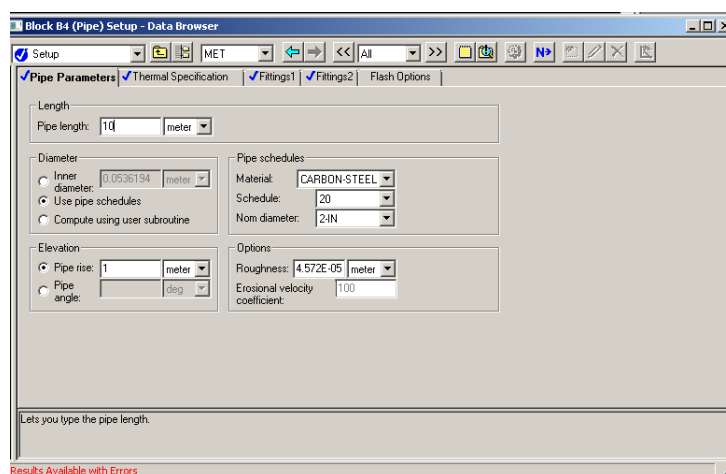


Figure 41. Gas/liquid flow patterns in horizontal flow.⁷

balance calculations is made (*i.e.*, $UA\Delta T$ and Q terms). Any fittings in the pipe can be defined in the **Setup | Fittings 1** tab, which also gives the ability to modify the factors used in the calculations, if needed. Finally, the **Setup | Fittings 2** tab gives the ability to include the entrance and exit effects and any contraction, expansion, or orifice presents in the pipe. Further calculations options for the pipe module can be made

⁷ Ibid., 6-26.

through the **Advanced | Calculation Options** and **Advanced | Methods** tabs available for the pipe block.

Consider for example a stream of natural gas (containing 80% C1, 10% C2, 5% C3, 3% C4, and 2% C5, mole fractions) flowing at a rate of 60 MSCFD, 150°F, and 100 psi. The stream will pass through 100 ft, carbon steel, 5", 40# pipe that is tilted 10° from the horizontal level. The pipe is exposed to the atmosphere at 68°F, and has a mass transfer coefficient of 1.5 Btu/ft²·hr·°F. The valve contains a butterfly valve and a 90° elbow. If you input all these information into the program, Aspen Plus gives the following results: 76.2 psi exit pressure, 64.5°F. You can notice that the outlet velocity of the gas is 432 ft/s, compared to 329 ft/s at the inlet (can you think why is this?). Finally, the results also show the flow regime for the system at its inlet and outlet. This is important to consider since the presence of mist in vapor streams can damage the pipelines. In this case the flow was completely in the vapor phase since the temperature is will above that of any of the components boiling points. Try decreasing the temperature to -60°F and increasing the pressure to 900 psi and see what is the flow regime for this case.

Exercise 1: Pump Performance Based on Data Sheet Specifications

The performance curve for the CL 125-160 (3 kW, $\Phi 183$) centrifugal pump is given in the attached sheet. The pump has an rpm of 1450. The pump is to be used to process $95 \text{ m}^3/\text{hr}$ of water at 100°C and 2.5 bar. Model this system using the STEAM-TA property method.



Johnson Pump

Questions:

4. What is the outlet pressure and temperature from the pump?

Temp:....., Pres:.....

5. What is the pump efficiency?

.....

6. How much energy does the pump consume? How much will that cost at a rate of 50 fils per kW-hr (define this in the utility section)?

Energy:....., Cost:.....

7. Using a 9000 suction specific speed, how much NSPH is required and how must it be available?

.....

8. What is the minimum operating inlet pressure needed to prevent cavitation in the pump?

.....

9. Does the efficiency increase, decrease, or remain the same as we increase the flow rate?

.....

Exercise 2: Compressor Operation

Two compressors are used to compress 4 m³/hr of oxygen at 40°F and 1 bar. The flow is divided between the two compressors equally. The specification for each compressor is given below:

Compressor 1

- Polytropic (use GPSA method) with $\eta = 50\%$.
- The performance is given by the following data:

Head (m)	Flow (m ³ /hr)
10000	1
20000	2
30000	3
40000	4

- Operating and shaft speed = 5000 rpm.

Compressor 2

- Isentropic with $\eta = 65\%$
- The performance is given by the following polynomial:

$$H(m) = 0.0001Q(m^3/hr)$$
- Operating and shaft speed = 5000 rpm

Questions:

1. What is the exit temperature and pressures from both compressors?

	Compressor 1	Compressor 2
T []		
P []		

2. How much electricity is needed for each compressor?

Compressor 1: _____, Compressor 2: _____

3. What is the isentropic temperature for each compressor? Does it differ from the outlet temperature? Why?

Compressor 1: _____, Compressor 2: _____

Exercise 3: Valve Specifications

Consider the valve datasheet supplied by Metso given below. In the top table on page 2, the value of C_v is given as a function of valve diameter. You can assume that the given diameter is the opening of a 20" valve. Also assume that X_T and F_L are 0.82 and 0.90 respectively.



Metso Valve

Ethanol is to be depressurized using this valve from 3 bar and 100°C at a flow rate of 360 SCMH.

Questions:

1. What will be the valve opening required to reduce the pressure to 2.5 bar? Do you think this valve is suitable for this process?

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2. What is the choked pressure for this valve?

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3. What is the cavitation index for this valve? What does this mean?

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4. If the valve is already installed and operated at 35% opening, what will be the outlet pressure?

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Exercise 4: Flow through Pipes

Natural gas and condensate are mixed and sent through a 27 km long pipeline to the processing facility. The conditions for the natural gas and condensate are shown below.

Condensate Conditions		Gas Conditions	
Temperature 24 °C		Temperature 38°C	
Pressure 42 bar		Pressure 42 bar	
Flow Rate 121 lpm		Flow Rate 410 kmol/h	
Composition (mole %)		Composition (mole %)	
Carbon Dioxide 0.0	n- Butane 14.0	Carbon Dioxide 0.2	n-Butane 1.5
Nitrogen 0.0	i-Pentane 11.0	Nitrogen 0.5	i-Pentane 0.7
Methane 3.5	n-Pentane 11.0	Methane 84.3	n-Pentane 0.3
Ethane 3.5	n-Hexane 23.0	Ethane 5.0	n-Hexane 0.5
Propane 15.0	n-Heptane 4.0	Propane 5.0	n-Heptane 0.5
i-Butane 15.0		i-Butane 1.5	

After the first 15 km, another gas stream joins the pipeline and has the conditions and composition show in the table to the right.

The following information is known about the piping system:

- The first 15 km is divided into three equal length segments each with 8 inch pipe.
- The first and third segments of pipe have no elevation change; however, the second segment has a 152 m increase in elevation.
- Add a 90° elbow after the first segment.
- The ambient temperature for the first segment is 10°C, 5°C for the second, and -1°C for the third.
- After the second gas stream joins the pipeline, the new mixture is compressed to 40 bar in a 65% polytropic efficient compressor.
- The compressed stream is pumped through a 12 inch line and continues for another 12 km to the processing facility.
- This pipe is well insulated.
- The heat transfer coefficient for the first three pipes is 10 W/m²·K.

Conditions	
Temperature 80°C	
Pressure 35 bar	
Flow Rate 9440 Nm ³ /d	
Composition (mole %)	
Carbon Dioxide 0.1	n-Butane 1.0
Nitrogen 0.8	i-Pentane 0.6
Methane 85.0	n-Pentane 0.4
Ethane 6.5	n-Hexane 0.4
Propane 3.7	n-Heptane 0.5
i-Butane 1.0	

Questions:

1. What is the temperature and pressure at the end of the piping system?

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2. How much energy is needed in the compressor?

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3. What happens to the simulation if the compressor is removed? Why?

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