Compressor Efficiency Definitions

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Many standard efficiency definitions exist that qualify the mass flow and power performance characteristics of a compressor. These different definitions are often used interchangeably throughout the compressor industry, although each is based on fundamentally different assumptions. This often results in incorrect comparisons between compressors offered by different companies, which precludes the customer from obtaining a fair and accurate assessment of their compressor needs.

Here, we present a brief description of some efficiency definitions to aid in understanding differences between them. To do so we first introduce a simple theoretical thermodynamic model of the compressor, and characterize its states during operation. Theoretical mass-flow equations are then presented, and theoretical compressor work are power are obtained. Next, power definitions, taken from standard thermodynamics texts and industry usage, are defined and discussed. Finally, these powers are used to define compressor efficiencies and recommendations are provided regarding their use, including which efficiency definitions are used by Vairex when characterizing compressors.

1 Thermodynamic States of an Ideal Piston Compressor

Prior to defining compressor efficiencies, we must quantify the theoretical thermodynamic states of the gas during compressor operation. This requires the determination of pressure (P), volume (V), and temperature (T) during a compression cycle. The remainder of the discussion in this section follows from standard thermodynamics texts such as [3] or [4].

A theoretical pressure-volume (PV) curve that displays the operating characteristics of a single cylinder compressor may be plotted given suction pressure P_1, discharge pressure, P_2, and the geometry parameters of swept volume, V_S, and clearance volume, V_C. In piston compressors, the swept volume and clearance volume are equal to the bore area multiplied

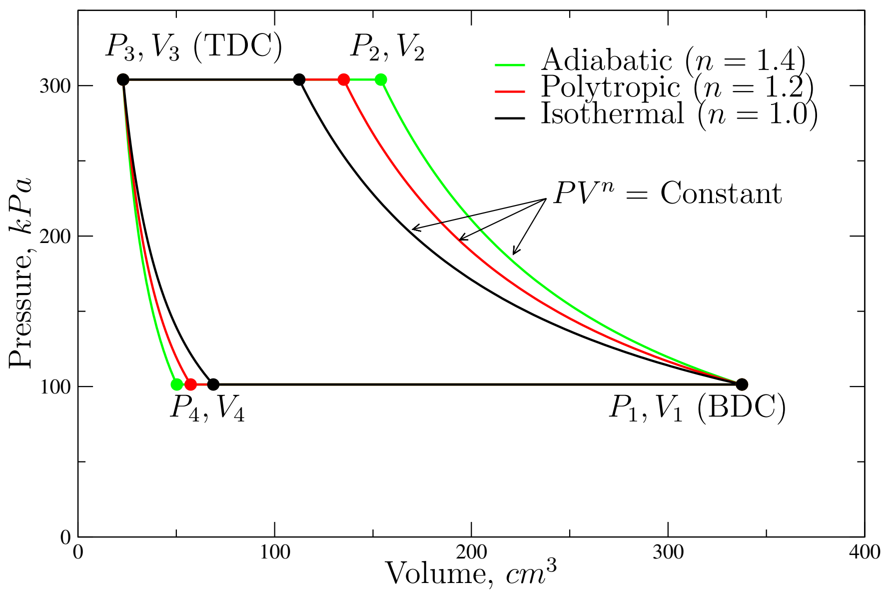


Figure 1: Theoretical PV curve for an example single cylinder compressor with pressure ratio 3.0 and clearance 7%. The positions 1, 2, 3, and 4 occur at BDC (Bottom Dead Center), discharge valve opening, TDC (Top Dead Center), and suction valve opening, respectively.

by the stroke length and the clearance length, respectively. The pressure ratio $P_R = P_2/P_1$ is adjusted by changing flow resistance in the discharge outlet pipe.

A PV curve is shown in Figure 1 for an example compressor with $V_S = 337.6cm^3$ (a cylinder of bore $= 11.89cm$ and stroke $= 3.05cm$), $V_C = 22.9cm^3$ (clearance=7%), $P_1 = 100kPa(1bar)$ (standard pressure at sea level), and pressure ratio $P_R = 3.0$. The suction temperature T_1 is assumed to be $15°C$ (standard temperature in Celsius).

In the Figure, V_1 and V_3 occur at the cylinder positions known as Bottom Dead Center (BDC) and Top Dead Center (TDC) respectively and are the maximum and minimum air volumes that occupy the cylinder. These volumes may be written as

$$V_1 = V_S + V_C \qquad V_3 = V_C. \tag{1}$$

The clearance, CL, is defined as a ratio of the clearance volume to the swept volume as [2]

$$CL = \frac{V_C}{V_S}. \tag{2}$$

Here, we assume that temperature and pressure remain constant during suction and discharge, so that

$$\begin{aligned} P_1 &= P_4, \qquad T_1 = T_4, \qquad &\text{(Suction)} \\ P_2 &= P_3, \qquad T_2 = T_3. \qquad &\text{(Discharge)} \end{aligned} \tag{3}$$

Also shown in the figure are the cylinder volume at discharge valve opening, V_2, and the cylinder volume at suction valve opening, V_4, both of which are process dependent. They may be calculated from the pressures and volumes at states 1 and 3 using the polytropic equation of compression, $PV^n = \text{Constant}$, in which n is a process-dependent constant:

$$V_2 = V_1 \left(\frac{P_2}{P_1}\right)^{-1/n}, \qquad V_4 = V_3 \left(\frac{P_3}{P_4}\right)^{-1/n} = V_3 \left(\frac{P_2}{P_1}\right)^{-1/n}. \tag{4}$$

The gas temperature at states 2 and 3 may be found from the ideal gas equation of state,

$$PV = mRT, \tag{5}$$

in which m is the mass of gas, R is the ideal gas constant for air, and T is the absolute temperature. Because the mass is constant between states 1 and 2,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}. \tag{6}$$

Using (6) to eliminate the ratio V_2/V_1 from (4a) the temperature at discharge, T_2, is found as

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \tag{7}$$

With (7), Figure 1 can be expanded to show the temperature behavior as a third axis in the 3D PVT curve. This third axis is shown in Figure 2, where we see the temperature change due to compression is highest in the adiabatic process, while there is no temperature change with $n = 1$ (*i.e.* an isothermal process). The thermodynamic states for the air in the compressor are now characterized throughout the cycle.

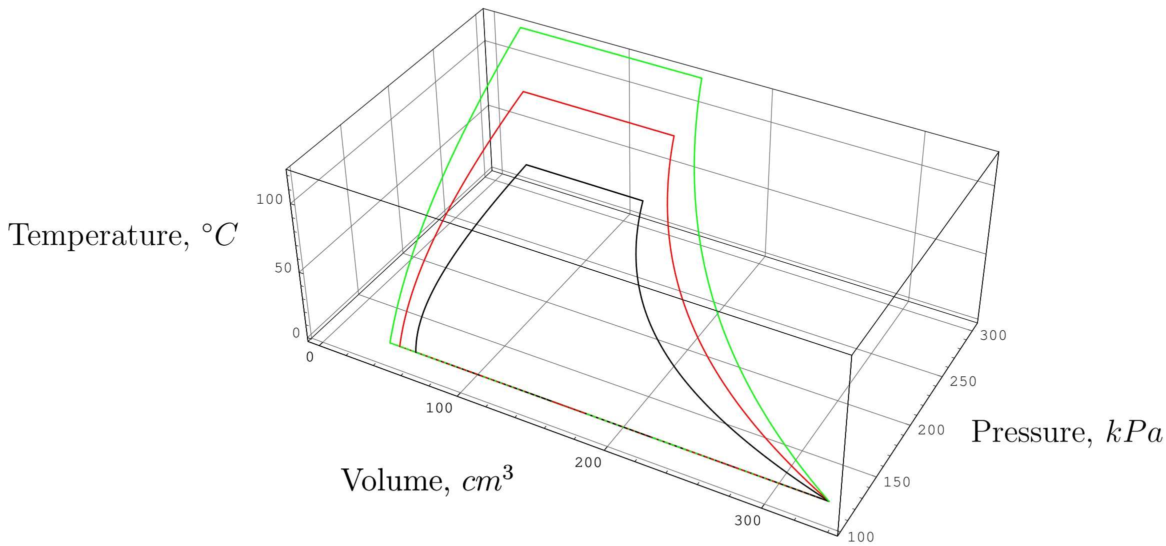


Figure 2: Theoretical PVT curve for an example single cylinder compressor with pressure ratio 3.0 and clearance 7%. The green, red, and black curves shown correspond to $n = 1.4, 1.2, 1.0$, respectively.

2 Mass Flow Definitions and Volumetric Efficiency

With the ideal gas equation of state (5), the theoretical mass contained within a single cylinder compressor can be formulated in terms of suction volume, $V_1 - V_4$, or discharge volume, $V_2 - V_3$, assuming no leakage during a cycle. This mass is

$$m = \frac{P_1(V_1 - V_4)}{RT_1} = \frac{P_2(V_2 - V_3)}{RT_2}, \tag{8}$$

and is process-dependent, because the values of V_2 and V_4 are functions of n. The mass-flow rate, $\dot{m}$, is simply

$$\dot{m} = f_c \frac{P_1}{RT_1}(V_1 - V_4) = f_c \frac{P_2}{RT_2}(V_2 - V_3), \tag{9}$$

where f_c is the compressor rotational frequency in Hz (RPM/60). The terms P_1/RT_1 and P_2/RT_2 in (9) can be viewed as a density at a specific suction state or a specific discharge state; the product of this density with the volume yields the mass flow per cycle.

We can simplify (9) by introducing the volumetric efficiency, η_V, which is defined by [2]

$$\eta_V = \frac{V_1 - V_4}{V_1 - V_3}. \tag{10}$$

Rearranging (9) and introducing (10) into the resulting equation yields

$$\dot{m} = f_c \frac{V_1 - V_4}{V_1 - V_3} \frac{P_1(V_1 - V_3)}{RT_1} = f_c \eta_V \frac{P_1 V_S}{RT_1} = \eta_V \dot{m}_S, \tag{11}$$

in which we recall that V_S is the swept volume and

$$\dot{m}_S = f_c \frac{P_1 V_S}{RT_1} \tag{12}$$

is the theoretical swept-volume mass-flow rate, which is independent of the pressure ratio and compression process, and can be viewed as the ideal compressor mass-flow rate for a given geometry with no clearance. Rearranging (11) we obtain

$$\eta_V = \frac{\dot{m}}{\dot{m}_S}. \tag{13}$$

To determine how the theoretical mass-flow rate, $\dot{m}$, depends on compressor geometry and process, we manipulate (10) such that it is a function of only the clearance, CL, the pressure ratio, P_2/P_1, and the polytropic exponent, n. The result is [1]

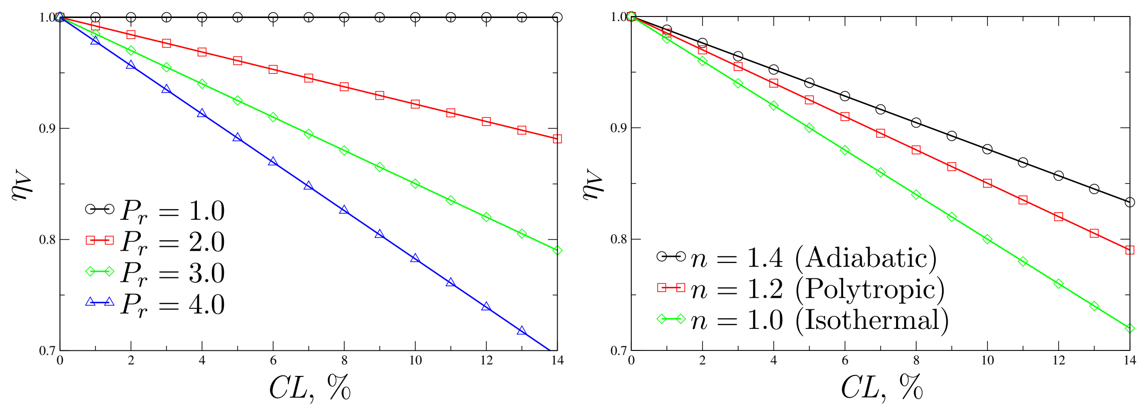
$$\eta_V = 1 - CL \left[\left(\frac{P_2}{P_1} \right)^{1/n} - 1 \right]. \tag{14}$$

Equation (14) is plotted as a function of CL and P_R for constant $n = 1.2$ in Figure 3a. Figure 3b shows (14) as a function of CL and n for constant $P_R = 3$. Clearly, the value of volumetric efficiency, and thus, the theoretical mass-flow rate, decreases with increasing clearance and decreasing n value. This degradation is greater for larger pressure ratios and for smaller values of n.

The actual mass flow is further limited by the imperfect nature of the valve, as well as wall friction and other frictional losses. In view of equation (13), the experimental volumetric efficiency, $\eta_{V,e}$, can be defined as

$$\eta_{V,e} = \frac{\dot{m}_e}{\dot{m}_S}, \tag{15}$$

in which $\dot{m}_e$ is the experimentally measured discharge mass-flow rate, which should be measured directly, *i.e.* in terms of mass rather than derived from volume flow and density, if possible. This volumetric efficiency is reported by Vairex.



(a) Constant polytropic exponent, $n = 1.2$

(b) Constant pressure ratio, $P_r = 3$

Figure 3: Volumetric Efficiency vs Clearance

If only a volumetric measurement of the discharge flow rate is available, $\eta_{V,e}$ may be calculated by

$$\eta_{V,e} = \frac{(ACFM)_e}{\dot{V}_S}, \tag{16}$$

in which $(ACFM)_e$ is the experimentally measured volume flow in actual cubic feet per minute (ACFM) and $\dot{V}_S$ is the swept-volume volumetric-flow rate, equal to $f_c V_S$.

A commonly used mass-flow ratio, γ_m, is the ratio of the experimentally measured mass-flow rate to the theoretical mass-flow rate,

$$\gamma_m = \frac{\dot{m}_e}{\dot{m}} = \frac{\eta_{V,e}\dot{m}_S}{\eta_V \dot{m}_S} = \frac{\eta_{V,e}}{\eta_V}. \tag{17}$$

This ratio should not be referred to as a volumetric efficiency because it is not a ratio of a flow rate to the maximum theoretical rate, as are η_V and $\eta_{V,e}$. Finally, because the actual flow $\dot{m}_e$ is always less than the theoretically predicted flow $\dot{m}$, $\gamma_m \leq 1$.

3 Work and Power Definitions

The theoretical work required for gas compression, W, can be calculated by integrating the PV curve shown in Figure 1 with respect to the volume in a clockwise direction,

$$W = \oint P(V)dV, \tag{18}$$

in which $P(V)$ is given by the polytropic equation or is constant, depending on which part of the curve is being integrated. Multiplying (18) by f_c and performing the integration over the volume yields the theoretical power [2]

$$\dot{W} = \begin{cases} f_c \dfrac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\dfrac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] & n \neq 1, \\ f_c P_1 (V_1 - V_4) \ln \left(\dfrac{P_2}{P_1} \right) & n = 1. \end{cases} \tag{19}$$

Figure 4 shows the work computed from (19) with the compressor geometry parameters given in Section 1 and $f_c = 1000RPM/60$. We see that as the process exponent n increases, so too does the required work. Processes in which $n < 1$ are not shown, as they are unattainable without impractical cooling of the compressor. Because the isothermal process requires the least work input, it is the most desirable process.

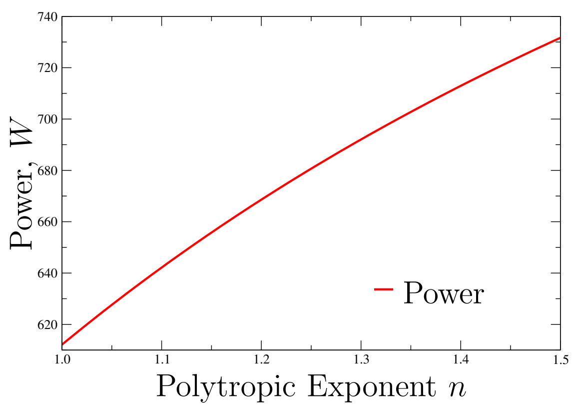


Figure 4: Theoretical power for an example single cylinder compressor with pressure ratio 3.0 and clearance 7% at $f_c = 1000RPM/60$.

The power may be related to the heat transfer and the flow work through conservation of energy for the compressor control volume,

$$\dot{W} = \dot{m} c_p (T_2 - T_1) \begin{cases} +Q_{removed} & \text{(heat leaving system, } n < k) \\ -Q_{added} & \text{(heat entering system, } n > k) \end{cases} \tag{20}$$

in which c_p is the specific heat at constant pressure for air, the values of heat transfer and $Q_{removed}$ and Q_{added}, are both positive. Equation (20) provides an additional method for quantifying the power, in terms of suction and discharge temperatures and an estimate of the heat transfer. This is valuable in that the process exponent n cannot be directly measured. Simplifications may be made to the equations (19) and (20) in order to define specific types of power. These types are defined below.

3.1 Adiabatic Power

The power may be written in a simpler form for an adiabatic process, defined by $n = k$, where $k = c_p/c_v$ and c_v is the specific heat of air at constant volume. We first rewrite (19b) in terms of c_p, c_v, and R (recall that $R = c_p - c_v$); then combining (19a) with (9), we obtain, through the displayed sequence of operations,

$$\begin{aligned}\dot{W}_{ad} &= \frac{c_p}{c_p - c_v} f_c P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \\ &= \frac{c_p}{R} f_c P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] = f_c \frac{P_1 (V_1 - V_4)}{R T_1} c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \\ &= \dot{m}_{ad} c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right] = \dot{m}_{ad} c_p (T_2 - T_1), \end{aligned} \tag{21}$$

in which $\dot{m}_{ad}$ is (9) evaluated for a particular compressor size and with $n = k$, the ratio $(k-1)/k = 2/7$ for air ($k = 1.4$), and T_1 and T_2 are the theoretical suction and discharge temperatures, respectively. Clearly, this expression may also be obtained from (20) by neglecting the heat transfer Q. We note that (21) represents the isentropic, or internally reversible, power required for compression.

A more commonly used definition of adiabatic power among compressor companies is based on the experimental mass-flow rate. It is

$$\dot{W}_{ad,e} = \dot{m}_e c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right]. \tag{22}$$

This form of experimentally based adiabatic power can be interpreted as the power required to adiabatically compress and expand the air that produces the experimental mass-flow rate. Therefore, the theoretical reference to particular compressor geometry is lost.

With (17), a relationship between these two adiabatic powers can be derived as

$$\dot{W}_{ad,e} = \dot{m}_e c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right] = \frac{\eta_{V,e}}{\eta_V} \dot{m}_{n=k} c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right] = \gamma_M \dot{W}_{ad}. \tag{23}$$

Because $\gamma_m \leq 1$, $\dot{W}_{a,e}$ is typically smaller than $\dot{W}_a$.

3.2 Isothermal Power

For cooled or water-injected air compressors with low discharge temperatures, the theoretical isothermal power is a reasonable estimate of the required input power; it is given by

$$\dot{W}_{iso} = f_c P_1 (V_1 - V_4) \ln \left(\frac{P_2}{P_1} \right) = \dot{m}_{iso} R T_1 \ln \left(\frac{P_2}{P_1} \right), \tag{24}$$

in which $\dot{m}_{iso}$ is (9) evaluated for a particular compressor size and with $n = 1$.

Unlike the adiabatic power, an experimental analogue does not appear in the thermodynamic literature or in industry usage. However, duplicating the concept used to obtain (22) from (21), we may define, by modifying (24),

$$\dot{W}_{iso,e} = \dot{m}_e R T_1 \ln \left(\frac{P_2}{P_1} \right). \tag{25}$$

The utility of such a definition is unknown.

3.3 Actual Power

Actual power is defined as the power required for gas compression only [2]. It is the indicated power integrated from an experimentally measured PV curve, which may be obtained through the difficult and expensive process of conducting compressor tests with pressure transducers within the compression chamber.

The actual power can be estimated, however, in terms of an enthalpy gain of the control volume flow and the heat transfer rate, assuming we can measure the discharge and suction temperatures accurately. This estimate is

$$\dot{W}_{actual,e} = \dot{m}_e c_p (T_2 - T_1)_e \begin{cases} +Q_{removed} & \text{(heat leaving system)} \\ -Q_{added} & \text{(heat entering system)} \end{cases} \tag{26}$$

in which $(T_2 - T_1)_e$ is the experimentally measured difference between the discharge and suction temperatures. It is common to assume that the heat transfer in (26) is negligible due to difficulty in its quantification and because, in the event of rapid compression, the heat flux per cycle is typically small with respect to the change in gas enthalpy. This assumption can potentially produce gross errors in the estimate of actual power.

3.4 Shaft Power

Shaft power is the experimentally measured power required to run a compressor and is calculated from measured shaft torque and speed. It includes all the frictional losses of a compressor and may be written as the sum

$$\dot{W}_{shaft,e} = \dot{W}_{actual,e} + \dot{W}_{friction}, \tag{27}$$

in which $\dot{W}_{friction}$ is work done against mechanical friction. In practice, it is the most easily measured power. It is always greater than actual power or adiabatic power.

3.5 System Power

The system power is the power required for an entire compressor system and is the sum of the shaft power and potentially several additional power requirements due to the presence of controllers and motors, *vis.*,

$$\dot{W}_{system} = \dot{W}_{shaft,e} + \dot{W}_{controllers} + \dot{W}_{motors} + \ldots \tag{28}$$

in which the ellipsis (...) indicates the potential presence of additional power terms.

4 Power Efficiency Definitions

A wide variety of efficiency definitions exist that characterize the volumetric, energetic, and thermal qualities of a compressor (see, *e.g.* [5]). Presented below are a subset of these definitions that we find most useful. We note that all experimental quantities are typically normalized to STP prior to their use in efficiency equations.

4.1 Adiabatic Efficiency

Adiabatic efficiency is defined as a ratio of an adiabatic power and the actual power. Because of the aforementioned definitions of adiabatic power, it is possible to define two adiabatic efficiencies,

$$\eta_{ad} = \frac{\dot{W}_{ad}}{\dot{W}_{actual,e}}, \qquad \eta_{ad,e} = \frac{\dot{W}_{ad,e}}{\dot{W}_{actual,e}}, \tag{29}$$

in which η_{ad} is the theoretical adiabatic efficiency, and $\eta_{ad,e}$ is the experimental adiabatic efficiency. The former has also been referred to as both the *adiabatic compression efficiency* and the *isentropic compression efficiency* [2].

By recognizing that $(T_1)_e = T_1$, the experimental adiabatic efficiency may be expanded as

$$\eta_{ad,e} = \frac{\dot{m}_e c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right]}{\dot{m}_e c_p (T_2 - T_1)_e} = \frac{\left[\left(\frac{P_2}{P_1} \right)^{2/7} - 1 \right]}{\left(\frac{T_2}{T_1} - 1 \right)_e}, \tag{30}$$

is sometimes referred to as the temperature efficiency, which is then used in lieu of adiabatic efficiency.

However, because of the difficulty in accurately measuring experimental suction and discharge temperatures, these forms of efficiency tend to be erroneous. Vairex does not report compressor performance with either of (29).

4.2 Isothermal Efficiency

Two isothermal efficiencies may be defined,

$$\eta_{iso} = \frac{\dot{W}_{iso}}{\dot{W}_{actual,e}}, \qquad \eta_{iso,e} = \frac{\dot{W}_{iso,e}}{\dot{W}_{actual,e}}, \tag{31}$$

which are the theoretical isothermal efficiency and the experimental isothermal efficiency, respectively. The former has also been referred to as the *isothermal compression efficiency* [2].

As seen in figure 5, the isothermal power is the least power required to compress a gas. Thus, the theoretical isothermal efficiency is always smaller than the corresponding theoretical adiabatic efficiency. The use of either isothermal efficiency to characterize a compressor is only sensible when the compressor is cooled, resulting in the gas undergoing a polytropic process with exponent n such that $1 \leq n < k$. The adiabatic efficiency for such a process can be greater than one. Vairex does not report these isothermal efficiencies, due to the fact that they are not widely used today, and because of error associated with the determination of $\dot{W}_{actual,e}$.

4.3 Mechanical Efficiency

Mechanical Efficiency is defined as

$$\eta_{mech} = \frac{\dot{W}_{actual,e}}{\dot{W}_{shaft,e}}, \tag{32}$$

and is a measure of losses due to mechanical friction in a system. As mentioned above, the Actual Power is difficult to determine experimentally. Because of this difficulty, Vairex does not use mechanical efficiency.

4.4 Overall Efficiency

Overall efficiency is defined as the ratio of an adiabatic or isothermal power to shaft power. These ratios have also been called *compressor efficiencies* [2]. Because of the multiple aforementioned definitions of adiabatic power and isothermal power, it is possible to define two adiabatic overall efficiencies,

$$\eta_{overall,ad} = \frac{\dot{W}_{ad}}{\dot{W}_{shaft,e}}, \qquad \eta_{overall,ad,e} = \frac{\dot{W}_{ad,e}}{\dot{W}_{shaft,e}} \tag{33}$$

and two isothermal overall efficiencies,

$$\eta_{overall,iso} = \frac{\dot{W}_{iso}}{\dot{W}_{shaft,e}}, \qquad \eta_{overall,iso,e} = \frac{\dot{W}_{iso,e}}{\dot{W}_{shaft,e}}, \tag{34}$$

in which a subscript $,e$ indicates that the numerator of the efficiency is based on experimentally measured quantities. The first and third of these are ratios of a theoretical power to the shaft power; as such, are easy to calculate. One problem associated with the use of all four of these efficiencies is that they do not account for the effects of flow reduction in actual systems. As a result, Vairex does not report these efficiencies.

4.5 Specific Overall Efficiency

The way to account for a decrease in mass flow in a reported efficiency is to introduce the specific theoretical power,

$$\dot{W}/\dot{m} = \begin{cases} \dfrac{n}{n-1} R T_1 \left[\left(\dfrac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] & n \neq 1, \\ R T_1 \ln \left(\dfrac{P_2}{P_1} \right) & n = 1, \end{cases} \tag{35}$$

which is the theoretical work per unit mass, and the specific shaft power, $\dot{W}_{shaft,e}/\dot{m}_e$, which is the experimental shaft work per unit mass. These are then used to define two theoretical specific overall efficiencies

$$\bar{\eta}_{overall,ad} = \frac{\dot{W}_{ad}/\dot{m}_{ad}}{\dot{W}_{shaft,e}/\dot{m}_e} \qquad \bar{\eta}_{overall,iso} = \frac{\dot{W}_{iso}/\dot{m}_{iso}}{\dot{W}_{shaft,e}/\dot{m}_e} \tag{36}$$

in which an over bar indicates a specific quantity, and $\dot{m}_{iso}$ is (9) evaluated for a particular compressor size and with $n = 1$. The former has also been referred to as the *isentropic compressor efficiency* [3, 4].

In Vairex specifications for compressors, the specific adiabatic overall efficiency, $\bar{\eta}_{overall,ad}$ is the only power-related compressor efficiency listed. By inserting $n = k$ into (35a), placing the resulting expression into (36a), and simplifying, we find

$$\bar{\eta}_{overall,ad} = \frac{\dot{m}_e c_p T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]}{\dot{W}_{shaft,e}} \tag{37}$$

in which we recall that $\dot{m}_e$ is the experimental discharge mass flow, T_1 is the ambient inlet temperature, P_2/P_1 is the theoretical pressure ratio, and $k = 1.4$. Interestingly, this specific efficiency is equal to the experimental adiabatic overall efficiency, equation (33b).

This efficiency can be calculated with non-standard conditions as long as those conditions are consistent in both numerator and denominator. For example, if the experimental test is done in Colorado, $\dot{m}_e$, T_1, and $\dot{W}_{shaft,e}$ may be all experimentally obtained (non-standard) values. The resulting efficiency will be a very close approximation of the efficiency at STP.

4.6 System Efficiency

A compressor system may consist of a compressor, a motor, a controller and other devices such as a water pump for the purpose of water injection. Therefore, a system efficiency of a compressor system, η_{sys}, can be defined as a series product of compressor overall efficiency, $\eta_{overall}$, motor efficiency, η_{motor}, controller efficiency, $\eta_{controller}$, and other efficiency of an auxiliary device, $\eta_{auxiliary}$, as:

$$\eta_{sys} = \bar{\eta}_{overall,ad} \cdot \eta_{motor} \cdot \eta_{controller} \cdot \eta_{auxiliary} \cdot \ldots \tag{38}$$

in which the ellipsis indicates the possible presence of additional component efficiencies.

5 Conclusion

Vairex reports three quantities when characterizing compressors. These are

- the experimental volumetric efficiency, equation (15), which is the ratio of the experimental exit mass-flow rate to the theoretical swept-volume mass-flow rate;
- the specific overall adiabatic efficiency, equation (37), which is the ratio of the specific theoretical adiabatic work to the specific experimental shaft work;

- the system efficiency, equation (38), which is the series product of several component efficiencies.

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