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Isentropic and Volumetric Efficiencies for Compressors with Economizer Port

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ABSTRACT

The economizer principle is a well-known method to reduce the expansion losses of a refrigeration system. Since the beginning of the last century, compressors with an additional suction port for the admission of gas at higher pressure than suction pressure have been in use. A comprehensive literature review has shown that no definitions of efficiencies for this type of compressor are available. However, for the communication of the compressor's behaviour in science and applications, such efficiencies are of value.

In this paper, several thermodynamic ideal reference processes of the compression process with gas admission are presented. Based on these ideal processes, efficiencies are defined.

1. INTRODUCTION

Since the early last century, the economizer principle has been applied in vapour compression refrigeration systems raising the efficiency and cooling capacity (Figure 1). In these days, reciprocating piston compressors were predominant. They allow two different economizer system layouts: two stage compression with flash gas admission at intermediate pressure or the flash gas admission into the cylinder while the piston reaches its bottom dead centre. The latter was introduced and patented by Voorhees in 1905. However, it gives only a minor increase in cooling capacity and therefore fell into oblivion in the middle of last century. The market penetration of screw compressors on the chiller business has made the economical use of the economizer system layout with flash gas admission into the compression chamber competitive. Also, chillers with multi-stage turbo compressors are equipped with vapour admission between the compression stages. Lately, scroll compressors with connectors for additional gas admission have been entering the market, allowing the design of smaller one-compressor economizer systems. Common definitions of compressor efficiencies are insufficient to describe such compressors because they don't account for the additional suction gas flow with a pressure above suction pressure. This paper suggests new definitions of compressor efficiencies that can handle the additional gas stream.

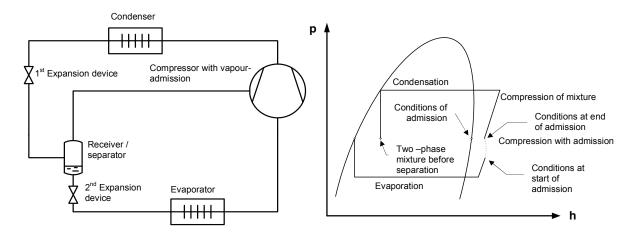
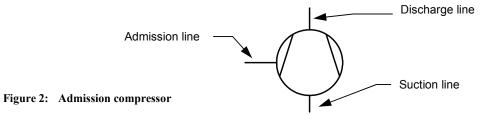


Figure 1: Economizer system with compressor with vapour admission

2. COMPRESSION WITH INNER GAS ADMISSION

As mentioned before, there are two ways of compressing the two gas flows at different pressures to one common discharge pressure. Option one is the compression by several compressors acting in parallel or in a two-stage layout. Option two is the economical compression of the suction gas with additional admission of a gas flow into the compression chamber of a single positive displacement compressor. However, admittedly, the potential for efficiency enhancement is lower using a single compressor with additional gas admission.

In the following, such compressors will be called admission compressor. The process of feeding additional gas into the compression chamber just after the compression process of the suction gas started is called inner admission in this paper. In principle, all positive displacement compressors can be modified to allow inner admission. Figure 2 shows the scheme of such an admission compressor.



In the following, the compression process with inner admission of on-the-market screw and scroll compressor designs is described. In general, it equals all positive displacement admission compressors. The compressor compresses suction gas from suction volume to the admission start volume at which the admission port opens. Then, admission gas is throttled to pressure in the compression chamber and mixed with the pre-compressed suction gas. While the admission gas is flowing in, the volume of the compression chamber reduces continuously. The admission stops as soon as the compression chamber reaches the volume at which the admission port closes. In many cases, at this state, the pressure in the compression chamber is above admission pressure. In this case, gas leaves the compression chamber through the admission port which results in a loss in efficiency. The compression process with admission is accompanied by heat transfer from the housing to the gas and leakage through the seals of the compression chamber and the valves.

3. CALCULATION OF A THERMODYNAMIC REFERENCE PROCESS

In the following, an idealized thermodynamic reference process is introduced to allow the definition of efficiencies of compressors with inner admission. It consists of suction phase, pre-compression, inner gas admission and final compression.

In the suction phase, the compression chamber expands; simultaneously, the suction gas enters. Hereby, the thermodynamic properties do not change. After this, in the simplified model, isentropic pre-compression takes place from suction volume to the volume at which the inner admission starts. All conventional admission compressors have a design defined inner volume ratio determining the ratio of the compression chamber's volume when the admission port opens to its volume at the end of the suction phase.

It follows the inner admission of the admission gas. Normally, the compression chamber's volume declines in the phase of admission; reaching the admission end volume, the admission port closes. This decrease in volume taking place simultaneously with the inner admission is neglected by the model.

The mixing process of the two gas portions is calculated by means of the mass and the energy balance. Since this process is not a steady flow process but a periodical, recurring, closed process instead, a control volume is encompassed by a system boundary as shown in Figure 3. At the beginning of the inner admission process, the admission gas expands isenthalpicly to the admission start pressure present in the compression chamber after precompression of the suction gas. Eventually, the pressure in the compression chamber adapts admission pressure.

Figure 3 shows all relevant details for the model of the mixing process. The left scheme displaces the compression chamber with volume V containing the recompressed suction gas $m_{s \text{ ads}}$ at admission start temperature $T_{s \text{ ads}}$ and admission start pressure $p_{s \text{ ads}}$. To its left, an infinite container with admission gas at admission conditions is indicated. An imaginary throttle valve simulating the opening and closing of the admission port allows connecting the container with the compression chamber.

The pipe leading to the admission port contains the admission gas mass ma with its volume Va at admission

temperature T_a and admission pressure p_a . The control volume for the mass and energy balance contains the two gas masses just before merging as displayed by the scheme to the left. The energy of the balanced system is comprised of the inner energy of the admission gas u_a and the inner energy of the gas in the mixing chamber u_s ads. The right picture of Figure 3 shows the final state of the mixing process. The left system boundary moved to the right to its final position when thermodynamic equilibrium was reached. For this illustration, the flow profile is assumed to be homogeneous.

After admission, volume V is filled by the gas mass of the mixture m_{mix} which comprises admission gas and the suction gas. The mixing process results in the mixing temperature T_{mix} at mixing pressure p_{mix} which, in the ideal case, equals the admission pressure p_a .

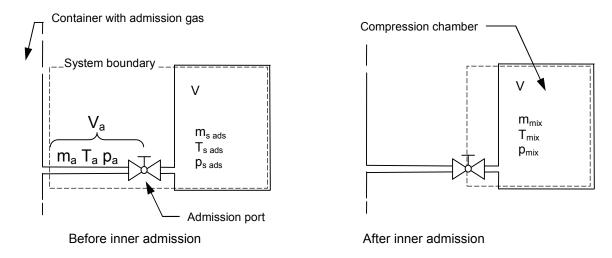


Figure 3: Closed system of mixing chamber and admission gas, left before admission and right after admission

At the system boundary, the gas of the container shown to the left transfers work to the balanced gas causing its change in volume. The imaginary throttle valve keeps the pressure at the system boundary constant at admission pressure. Therefore, the transferred work to relocate the admission gas and the system boundary is easy to calculate. It equals the volume of the admission gas multiplied with its pressure.

Consequently, the system energy at the end of the admission process equals the sum of the mentioned inner energies and the transferred work for relocation. The following equation system allows the calculation of the unknown variables of the mixing process, assuming a constant volume in the compression chamber during admission.

Mass balance: The masses correlate with the corresponding volumes and specific volumes as follows:

$$m_1 + m_2 = m_2$$
 $m_1 = \frac{V}{V}$ $m_2 = \frac{V}{V}$ (1)(2)(3)

A fourth equation results from applying the energy balance of non-stationary processes in open, adiabatic systems without considering kinetic and potential energy. The energy balance equation can be converted to a term containing only enthalpies instead of inner energies by means of the equation defining enthalpy.

$$\begin{split} & m_{_{s\,ads}} \cdot u_{_{s\,ads}} + m_{_{a}} \cdot u_{_{a}} + V_{_{a}} \cdot p_{_{a}} = m_{_{mix}} \cdot u_{_{mix}} \quad \rightarrow \quad m_{_{mix}} \cdot u_{_{mix}} - m_{_{s\,ads}} \cdot u_{_{s\,ads}} = m_{_{a}} \cdot h_{_{a}} \\ & \rightarrow \quad m_{_{mix}} \cdot h_{_{mix}} = m_{_{s\,ads}} \cdot h_{_{s\,ads}} + m_{_{a}} \cdot h_{_{a}} + m_{_{s\,ads}} \cdot v_{_{s\,ads}} \cdot \left(p_{_{a}} - p_{_{s\,ads}}\right) \end{split}$$

Six variables can be achieved from a fluid property database: v₁; v_a; v₂; h₁; h_a; h₂

One boundary condition: $p_2 = p_a$

Five known values: V, p_1 , T_1 , p_a , T_a

Eleven unknown values: m_1 ; m_a ; m_2 ; v_1 ; v_a ; v_2 ; h_1 ; h_a ; h_2 ; p_2 ; T_2

Thus there are four describing equations of the mixing process, six values to be determined by fluid properties, one boundary condition and five known values. Therefore, the system of equations can be solved and the eleven unknown values can be determined.

The main results of this calculation are the ratio of the admission mass-flow to the suction gas mass-flow and the temperature at the end of the mixing process which is needed for the idealized calculation of the ideal compressor's power consumption.

The mixing process is followed by isentropic compression of the mixture.

4. EFFICIENCIES OF AN ADMISSION COMPRESSOR

The result of an extensive literature research is that there is no definition of efficiencies to describe the behaviour of a compressor with inner admission. However, such characteristic numbers are of great importance for calculating economizer systems. In the following, two different approaches are introduced to define volumetric and energetic efficiencies of an admission compressor. Their content is identical; however, their philosophy differs substantially. In both approaches, the definitions are based on a publication describing the compression process of positive displacement compressors, written by the author (Lambers et al. 10/2007; 11/2007; 12/2007) which can only be cited, to keep this paper short.

4.1 Volumetric and Isentropic Efficiency of a Compressor with Inner Admission

Normally, efficiencies express the ratio of a real value based on measurements to an ideal reference. Usually, an ideal or maximum achievable value is chosen as a base, which is sometimes calculated with a model of an ideal reference process.

To keep it simple, this section deals with compressors that are not capacity controlled (constant speed, constant displacement and no internal bypass) even if the defined efficiencies apply for capacity controlled compressors as well. Therefore, the operating conditions are defined as soon as three parameters are known. Usually, the three variables suction pressure, suction temperature and discharge pressure are considered to be known. In this case, the relevant dependent variables are power consumption, mass-flow and discharge gas temperature. Less important dependent variables such as sound level, oil circulation rate, housing temperature, electrical power factor, working frequency and torque course are not considered at this stage. Also ignored are other environmental conditions such as ambient temperature and grid voltage; they are considered to be constant.

It is common to express the three relevant parameters, power consumption, mass-flow and discharge gas temperature by the three characteristic numbers isentropic efficiency, volumetric efficiency and isentropic discharge efficiency. Those dimensionless characteristic numbers base the dependent variables, or a value derived from the dependent variables, on a reference value. This value results from a reference process calculation with the independent variables. The above mentioned publication (Lambers et al. 12/2007) deals with those efficiencies in detail.

The operating conditions of an admission compressor with a fixed admission port position are defined by five variables. Usually, suction pressure, suction temperature, admission pressure, admission temperature and discharge pressure are expected to be known. Even in the case of a compressor with a variable admission port position, the dependency on five variables does not change provided that the position of the admission port is coupled to one of the parameters. This, for instance, applies if the positioning of the admission port is controlled depending on the admission pressure. Four dependent variables can be considered to be of interest: power consumption, suction gas mass-flow, admission gas mass-flow and discharge gas temperature. Reference processes must be defined to describe these four variables as efficiencies as described for common compressors with their three relevant dependent variables above. Those idealized reference processes are described in the following.

4.2 Volumetric Efficiency

It makes sense to apply the definition of the volumetric efficiency as it is used for conventional compressors to admission compressors.

$$\lambda = \frac{\dot{m}_s}{\dot{m}_{s \text{ ref}}} = \frac{\dot{m}_s}{V_h \cdot n \cdot f} \tag{5}$$

In the formula above, \dot{m}_s represents the suction gas mass-flow. The reference mass-flow \dot{m}_s ref can be calculated using the volume of the compression chamber V_h , the number of compression chambers n, the operation frequency of the compressor f and the specific volume of the suction gas at the compressor inlet v_s .

4.3 Volumetric Admission Efficiency

Generally, the volumetric admission efficiency λ_a can be defined as the ratio of the real admission gas mass-flow \dot{m}_a to the admission gas mass-flow of the reference process \dot{m}_a ref.

$$\lambda_{a} = \frac{\dot{m}_{a}}{\dot{m}_{a,ref}} \tag{6}$$

Several different idealized processes can be defined as reference processes for calculating the reference admission mass-flow. Generally, there are two conditions that need to be predefined to calculate a mass-flow with the model of the mixing process introduced above. Varying the possible conditions basically leads to six possible definitions of the volumetric admission efficiency. At present, it is not clear which definition will be most practical for use. That's why different alternatives are introduced in the following, indicated with a capital letter for distinction.

Definition of the Reference Process for Calculating the Reference Mass-Flow

The two above mentioned conditions to be predefined are made clear by having a closer look at the compression process with admission:

The compressor compresses suction gas from suction volume to the admission start volume at which the admission port opens. Then, admission gas is throttled to pressure in the compression chamber and mixed with the gas. As said before, the simplified model does not consider the continuing reduction of the compression chamber volume. In this model, the compression chamber reaches admission pressure as soon as the compression chamber reaches the volume at which the admission port closes. Heat transfer from the housing to the gas and leakage through the valves and seals of the compression chamber as well as any clearance volume are not considered in the idealized reference process either.

First Condition to be Predefined

To solve the model of the mixing process described above to calculate the reference mass-flow, the gas mass contained by the compression chamber at admission start must be defined. The following three options seem to be practical:

- a) The ideal transported suction mass (volumetric efficiency of one)
- b) The actual transported suction mass of the compressor without admission
- c) The actual transported suction mass of the compressor with admission

It seems wise to assume adiabatic reversible compression for calculating the thermodynamic conditions of the suction gas at admission start. Therefore, in case a), at admission start, the entropy of the gas in the compression chamber equals the entropy of the suction gas. In case b) and c), the entropy can be calculated from the transported suction gas mass, suction pressure and the compression chamber's volume at the end of the suction phase.

Second Condition to be Predefined

In common admission compressors, the volume of the compression chamber declines simultaneously with the admission process. As said before, to keep the definition of the characteristic numbers simple, the calculation of the reference processes should be based on a constant volume. Two volumes can be considered:

- a) The volume at start of admission
- b) The volume at end of admission

The reference admission mass-flow can be calculated with the introduced mixing model assuming the parameters of the admission gas, suction mass-flow, working frequency, number of compression chambers and entropy before the admission starts are known.

Six Possible Resulting Definitions of the Volumetric Admission Efficiency

The combination of the different possible conditions introduced above results in six different reference admission mass-flows allowing the definition of six volumetric admission efficiencies based on equation (6). In the following, those six cases are distinguished by capital letters from A-F.

It needs to be said that the volumetric admission efficiency can exceed the value of one in all these definitions. Internal leakage increases the volumetric admission efficiency while it decreases the volumetric efficiency and the suction mass-flow. The volumetric admission efficiencies A and B are based on an ideal compressor. Therefore, the reference admission mass-flow can be calculated without knowing the volumetric efficiency of the compressor.

Volumetric Admission Efficiencies A and B

The volumetric admission efficiencies A and B are calculated by the ideal transported suction mass:

Volumetric admission Efficiency A:

1. a) Ideal transported suction mass 2. a) Volume at start of admission

Volumetric admission Efficiency B:

1. a) Ideal transported suction mass 2. b) Volume at end of admission

Volumetric Admission Efficiencies C and D

The volumetric efficiency of the compressor without admission must be known to calculate the reference admission mass-flows on which the volumetric admission efficiencies C and D are based. This is so because it is used to calculate the entropy in the compression chamber at admission start.

Volumetric admission efficiency C:

1. b) Actual transported suction mass of the compressor without admission 2. a) Volume at start of admission

Volumetric admission efficiency D:

1. b) Actual transported suction mass of the compressor without admission 2. b) Volume at end of admission

Volumetric Admission Efficiencies E and F

To calculate the reference mass-flows to determine the following volumetric admission efficiencies, the volumetric efficiency of the compressor with admission is required.

Volumetric admission efficiency E:

1. c) Actual transported suction mass of the compressor with admission 2. a) Volume at start of admission

Volumetric admission efficiency F:

1. c) Actual transported suction mass of the compressor with admission 2. b) Volume at end of admission

4.4 Isentropic Admission Efficiency

The isentropic admission efficiency can be defined as the ratio of the theoretical power consumption for the compression of suction gas P_{Ref} to discharge pressure to the power P actually consumed at the same mass-flow:

$$\eta_{\rm a} = \frac{{\rm P}_{\rm Ref}}{{\rm P}} \tag{7}$$

Two reference processes seem to be appropriate to calculate the reference power consumption of the compressor. Their results differ only slightly. Either the reference process can be seen as a two-stage compression with admission at intermediate pressure or it can be seen as a one-stage compression of suction gas to discharge pressure with a parallel compression of the admission gas to discharge pressure. Both alternative reference processes assume isentropic compression. The first variant resulting in the isentropic admission efficiency A implies entropy generation due to mixing two gasses of different temperature at admission pressure. This reference power can be calculated as follows:

$$P_{\text{per }A} = \dot{m}_{s} \cdot \left(h(s_{s}; p_{a}) - h(s_{s}; p_{s})\right) + \dot{m}_{\text{nes}} \cdot \left(h(s_{\text{miv}}; p_{d}) - h(s_{\text{miv}}; p_{a})\right) \tag{8}$$

The second reference process resulting in the isentropic admission efficiency B leads to entropy generation due to

the temperature difference of mixed gasses at discharge pressure. It can be calculated as follows:

$$P_{RefB} = \dot{m}_{s} \cdot (h(s_{s}; p_{d}) - h(s_{s}; p_{s})) + \dot{m}_{a} \cdot (h(s_{a}; p_{d}) - h(s_{a}; p_{a}))$$

$$(9)$$

Even in the case of an ideal compressor, the isentropic admission efficiencies A and B do not reach one because the isenthalpic throttling, affected by exergetic losses, is not considered in the ideal reference process.

4.5 Isentropic Admission Discharge Efficiency

The two reference processes of the isentropic admission efficiency can also be used to define the isentropic admission discharge efficiency, which can be calculated analogous to the isentropic discharge efficiency of a conventional compressor (Lambers et al. 12/2007)

$$\eta_{ha} = \frac{P_{Ref}}{\dot{m}_{oes} \cdot h_d - (\dot{m}_s \cdot h_s + \dot{m}_a \cdot h_a)}$$
 (10)

The dominator is formed by the balance of all enthalpy flows lead in and out. The sum of these energy flows equals the consumed power of an adiabatic compressor. The numerator consists of the calculated reference power as defined before and used for the isentropic admission efficiency. Because there are two processes for calculating the reference power, there are also two defined isentropic admission discharge efficiencies, indicated with capital letters A and B, as above.

5. EFFICIENCIES DESCRIBING THE GOODNESS OF ADMISSION BASED ON AN IMAGINARY AUXILIARY COMPRESSOR

There is another way to describe the process of admission by characteristic numbers. This approach is applicable for compressors mainly designed for use in conventional systems equipped with the option of inner gas admission. With this introduced approach, the efficiency of the basic compressor and the efficiency of the admission process can be distinguished. For this approach, the compressor without admission is taken as a given. Its isentropic efficiency is expected to be known and it will be named 'main compressor' in the following. In operating with inner admission, an imaginary auxiliary compressor compresses the admission gas in parallel to the main compressor compressing the suction gas. Therefore, the admission can be described by conventional compressor efficiencies of the imaginary auxiliary compressor.

Indeed, this approach requires an additional characteristic number for a complete description of the admission process; in return, it is possible to contrast the efficiencies offhand with efficiencies of conventional compressors.

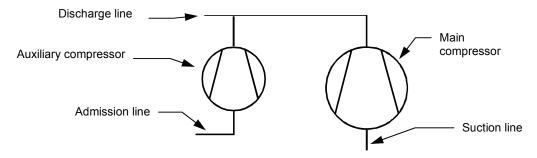


Figure 4: Diagram of an admission compressor as a combination of a main compressor and an auxiliary compressor

5.1 Isentropic Auxiliary Efficiency

In the following approach of stating the goodness of the admission process, the isentropic efficiency of the compressor without admission is required. In the working model, it is assumed that the compressor, named 'main compressor' in Figure 4 compresses the actual suction mass-flow with the known isentropic efficiency from suction pressure to discharge pressure. The resulting power consumption is to subtract from the actual power consumption. The resulting difference is taken as the power consumption of the imaginary auxiliary compressor for the

compression of the admission gas. With this power consumption, the mass-flow of the admission gas, its pressure and temperature as well as the discharge pressure, a conventional isentropic efficiency can be calculated. This efficiency is here called isentropic auxiliary efficiency.

5.2 Volumetric Auxiliary Efficiency

Another rather simple efficiency can describe the admission mass-flow. It relates the volume flow at the admission port to a volume flow calculated with the admission start volume, the working frequency of the compressor, the number of compression chambers and the specific admission volume.

5.3 Admission Mass Ratio

Volumetric admission efficiency and volumetric auxiliary efficiency are two characteristic numbers expressing the admission mass-flow. A third one is the admission mass ratio α_a . It defines the ratio of admission gas mass-flow to suction gas mass-flow and depends on the design of the admission compressor as well as the operation conditions. For many system calculations, this figure is easier to handle because it can be used in calculations with specific mass-flows without knowing the actual system size.

CONCLUSION

It is important to know the behaviour of compressors with admission (economizer) port, to be able to calculate the behaviour of economizer systems utilizing admission compressors. The efficiencies introduced in this paper facilitate the exchange of the compressor's performance data.

Two sets of efficiencies are introduced. One set bases the measured values on values calculated by an idealizing reference process. This has the advantage that the values of the efficiencies are expected to be similar at a wide range of operation conditions, as known from efficiencies of conventional compressors. This should also help to generate relatively accurate curve fits.

The efficiencies of the other set, based on an imaginary auxiliary compressor, result in values which depend much more upon the operating conditions. However, they can be compared to the efficiencies of common compressors directly. This can be done to evaluate if it is economical to build a two-stage or parallel compression economizer system instead of an economizer system with an admission compressor.

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