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On Suction Gas Heating in Hermetic Compressors (A Technical Note)

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INTRODUCTION

The phenomenon of suction gas heating in hermetic refrigerating compressors is very well recognized. In fact, sometimes it is even utilized to the advantage of the compressor; for instance, in superheating the suction gas to avoid any "liquid slugging" in the compressor cylinder, and in cooling the motor windings. However, it has also been observed that the refrigerant mass flow rate decreases as the temperature of the suction gas increases, thereby reducing its mass density [2,3,4,6] (by suction gas, here, we mean the refrigerant gas before it enters the compressor cylinder). The energy requirements of the compressor to maintain the same mass flow rate will also increase with increase in suction gas temperature [6]. This change in mass flow or energy requirements has generally been obtained either experimentally or using a pressure-enthalpy diagram provided the rise in temperature is known [3,6]. In this technical note, a simple thermodynamic relationship will be shown that will give us the change in mass flow rate and energy requirements if the heat transfer to the suction gas is known. For design estimation of the performance of a compressor, this relationship might be valuable if, through a simplified heat transfer model of the compressor shell, an approximate relationship for heat transfer to the suction gas can be established [see also reference 8].

HEAT TRANSFER TO SUCTION GAS

From the time the refrigerant enters the compressor shell until the time it enters the compressor cylinder, it undergoes temperature changes due to heat transfer from several sources. This also depends on the system being a low-side or high-side one. A low-side system is one where incoming refrigerant is open to the shell atmosphere and is thus exposed to heat transfer from all sources inside the shell. A high-side system is one where incoming gas is fed to the cylinder directly through a suction tube and is therefore exposed only to the suction tube, suction manifold and suction valves. The purpose of this note is not to go into a detailed heat transfer model of the compressor for which reader should refer to references [4,5,9]. The purpose here is to show how a particular aspect of it (namely, heat transfer to suction gas) affects the compressor performance and how to take it into account in estimating the performance analytically.

EFFECT ON MASS FLOW

As we mentioned earlier, the sources of heat transfer to suction gas vary according to the design and type of the compressor. Let us suppose for this analysis, however, that the average heat transfer rate to the suction gas ($\dot{Q}_s$) is known to us. During the passage of the gas from the shell inlet to the cylinder inlet, there is a slight pressure drop also involved along with heat transfer. For the present analysis, however, we shall assume that this pressure drop is negligible and, therefore, the heat transfer takes place at constant pressure (i.e., at suction pressure P_s). Other assumptions are: (1) the suction gas behaves as an ideal gas and (2) the polytropic coefficient of expansion (n) remains constant, thereby making the net cylinder displacement (V_{net}) constant (Fig. 1). Then, first law of thermodynamics gives:

$$\dot{Q}_s = C_p \dot{m}(T_c - T_s) \tag{1}$$

where C_p is the specific heat of refrigerant vapor at constant pressure, $\dot{m}$ is the average mass flow rate, and T_c and T_s are the temperatures of the gas at the inlet to the cylinder and the inlet to the shell respectively. But,

$$\dot{m} = \rho_c V_{net} \frac{N}{60}$$

$$= (\frac{P_c}{RT_c}) V_{net} \frac{N}{60} = (\frac{P_s}{RT_c}) \frac{V_{net} N}{60} \qquad (2)$$

Eliminating T_c from (1) and (2), we get:

$$\dot{Q}_s = C_p \dot{m} \left(\frac{P_s V_{net} N}{60 \dot{m} R} - T_s \right) \qquad (3)$$

or

$$\dot{m} = \frac{P_s V_{net} N}{60 R T_s} - \frac{\dot{Q}_s}{C_p T_s} \qquad (4)$$

The first term on the right hand side of the equation (4) can be easily recognized as the ideal mass flow rate had there been no suction gas heating since the quantity (P_s/RT_s) is actually the density ρ_s at the shell inlet conditions. The second term is the loss in the mass flow rate due to suction gas heating ($\dot{m}_{SHL}$). Or,

$$\dot{m}_{SHL} = \frac{\dot{Q}_s}{C_p T_s} = \frac{\dot{Q}_s (\gamma - 1)}{\gamma R T_s} \qquad (5)$$

where γ is the specific heat ratio of the suction gas. Thus, for a given suction gas heat transfer rate, $\dot{Q}_s$, the mass flow rate loss, $\dot{m}_{SHL}$ can be calculated, since T_s, the temperature at the inlet to the shell, is known.

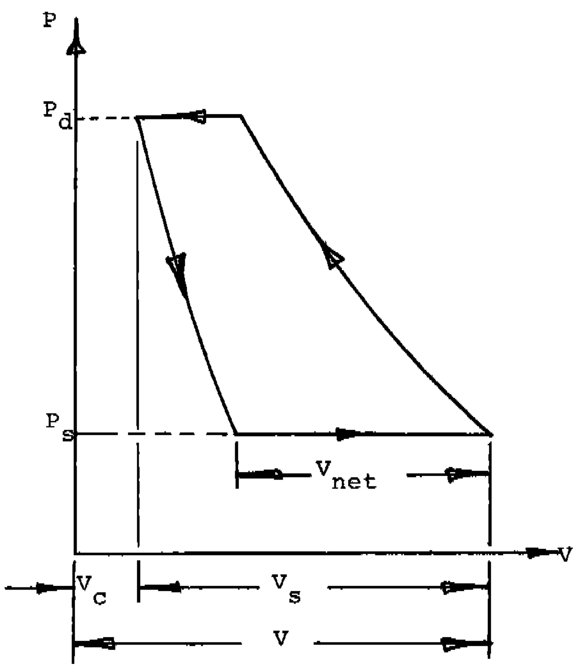

Fig. 1: Theoretical P-V diagram of a compressor

EFFECT ON ENERGY REQUIREMENTS

Referring to the theoretical P-V diagram of a compressor, shown in Fig. 1, the indicated compressor work required per cycle is given by the area of the P-V diagram and the corresponding formula as given in any standard textbook [2] is:

$$W = P_s V_{net} \left[\frac{n}{n-1} \left\{ \left(\frac{P_d}{P_s} \right)^{\frac{n-1}{n}} - 1 \right\} \right] \qquad (6)$$

Since P_s and P_d are controlled by valve design (for self-acting valves), they are constant. V_{net} will depend upon the clearance volume V_c and the coefficient of polytropic expansion n. Ideally, n should remain constant. In actual practice, however, the increase in the temperature of the suction gas entering the cylinder will affect the heat transfer to and from the cylinder walls and hence will change the value of n. This, in turn, will change V_{net}. But, these changes will be so small (because of the high speed of the compressor) that it is reasonable to assume that n and, therefore, V_{net} remain constant. Then, relationship (6) indicates that the work per cycle, W, remains constant. Thus, there is no change in the power requirements of the compressor. However, since the total mass flow rate is reduced, as given by Equation 5, the energy required per unit mass pumped will increase. Thus, the overall performance of the compressor will suffer as a result of suction gas heating.

EXAMPLE: Estimate of Suction Gas Heating as Function of Motor Loss

Since motor loss is one of the main contributors of the heat generated inside a compressor, it is interesting to see how it affects the suction gas heating. Letting $\dot{Q}_m$ and $\dot{Q}_r$ denote the rate of heat generation by motor and the remaining components of the compressor respectively, we can write:

Total heat generation rate,

$$\dot{Q}_T = \dot{Q}_m + \dot{Q}_r \qquad (7)$$

A fraction of $\dot{Q}_T$ (say $k\dot{Q}_T$), where k is less than 1, goes to heat the suction gas. Often times we are interested in knowing how a change in the efficiency of the motor will affect the suction gas heating. To do this, let us consider two different motors with respective motor losses $\dot{Q}_{m1}$ and $\dot{Q}_{m2}$ and the corresponding heat transfer to suction gas $\dot{Q}_{s1}$ and $\dot{Q}_{s2}$ respectively. Assuming that $\dot{Q}_r$ remains the same in both cases as we are considering the same compressor, and the same fraction of total heat transfer goes to suction gas in both the cases (i.e., k is constant), which does not seem to be too unreasonable, we can write:

$$\dot{Q}_{s_1} = k\dot{Q}_{T_1} = k(\dot{Q}_{m_1}+\dot{Q}_r) \tag{8}$$

$$\dot{Q}_{s_2} = k\dot{Q}_{T_2} = k(\dot{Q}_{m_2}+\dot{Q}_r) \tag{9}$$

Equations (8) and (9) can be solved to give the following relationship:

$$c = \frac{a+b}{1+b} \tag{10}$$

where

$$a = \frac{\dot{Q}_{m_2}}{\dot{Q}_{m_1}} \tag{11}$$

$$b = \frac{\dot{Q}_r}{\dot{Q}_{m_1}} \tag{12}$$

$$c = \frac{\dot{Q}_{s_2}}{\dot{Q}_{s_1}} \tag{13}$$

or

$$\dot{Q}_{s_2} = \left[\frac{a+b}{1+b}\right] \dot{Q}_{s_1} \tag{14}$$

Relationship (14) gives us the heat transfer rate to suction gas when motor No. 1 is replaced by motor No. 2, provided the losses of both the motors are known and the remaining losses within the compressor ($\dot{Q}_r$) are also known. For example, for a typical compressor the following results were obtained:

$$\dot{Q}_{m_1} = 580 \text{ [Watts]}$$

$$\dot{Q}_{r_1} = 250 \text{ [Watts]}$$

$$\dot{Q}_{m_2} = 700 \text{ [Watts]}$$

$$\dot{Q}_{r_2} = 250 \text{ [Watts]}$$

We notice that $\dot{Q}_{r_1}$ and $\dot{Q}_{r_2}$ are almost the same in both cases and therefore agree with our assumption. Then, substitution in (11), (12) and (14) yields:

$$a = 1{\cdot}2$$

or $$\dot{Q}_{m_2} = 1{\cdot}2\dot{Q}_{m_1}$$

$$b = {\cdot}43$$

$$\dot{Q}_{s_2} = 1{\cdot}14\dot{Q}_{s_1}$$

Thus, we find that, with the assumptions as mentioned earlier, for a 20% increase in motor loss we get an increase of 14% in suction gas heating in this case. Since the reduction in mass flow rate is proportional to suction gas heat transfer (see Equation 5), we deduce that in this particular case the loss in mass flow rate will be approximately 14% for a 20% increase in motor losses. The result is quite interesting.

CLOSURE

In this technical note, the importance of suction gas heating on mass flow rate and also work per unit mass delivered was pointed out. An example was given to illustrate in an approximate way this influence.

NOMENCLATURE

a	Motor loss ratio ($\dot{Q}_{m_2}/\dot{Q}_{m_1}$)
b	Ratio of rest of the losses to motor loss ($\dot{Q}_r/\dot{Q}_{m_1}$)
c	Suction gas heat transfer ratio ($\dot{Q}_{s_2}/\dot{Q}_{s_1}$)
C_p	Specific heat of gas at constant pressure [$\frac{Nm}{kg\text{-}^oK}$]
k	Constant
$\dot{m}$	Mass flow rate [kg/sec]
$\dot{m}_{SHL}$	Loss in mass flow rate due to suction gas heating [kg/sec]
n	Polytropic coefficient of expansion and compression
N	RPM of the motor
P_c	Pressure at the inlet to the cylinder [N/m^2]
P_d	Discharge pressure [N/m^2]
P_s	Suction pressure [N/m^2]
$\dot{Q}_m$	Motor loss [Watts]
$\dot{Q}_{m_1}$	Loss in motor No. 1 [Watts]
$\dot{Q}_{m_2}$	Loss in motor No. 2 [Watts]
$\dot{Q}_r$	Remaining compressor losses [Watts]
$\dot{Q}_{r_1}$	Remaining compressor losses in first case [Watts]
$\dot{Q}_{r_2}$	Remaining compressor losses in second case [Watts]
$\dot{Q}_s$	Heat transfer rate to suction gas [Watts]
$\dot{Q}_{s_1}$	Heat transfer rate to suction gas in first case [Watts]

$\dot{Q}_{S_2}$ Heat transfer rate to suction gas in second case [Watts]

$\dot{Q}_T$ Total compressor loss [Watts]

$\dot{Q}_{T_1}$ Total compressor loss in first case [Watts]

$\dot{Q}_{T_2}$ Total compressor loss in second case [Watts]

R Gas constant [Nm/kg-^{O}K]

T_c Temperature of the gas at the inlet to the cylinder [^{O}K]

T_s Temperature of the gas at the inlet to the shell [^{O}K]

V_c Clearance volume [m^3]

V_{net} Net effective cylinder displacement [m^3]

W Indicated compressor work per cycle [Nm]

ρ_c Mass density of gas at the inlet to the cylinder [kg/m^3]

ρ_s Mass density of the gas of the inlet to the shell [kg/m^3]

γ Ratio of specific heats of the suction gas

REFERENCES

1. Chlumsky, V., Reciprocating and Rotary Compressors, E & F N Spon, Ltd., London, 1965

2. Jordon, R. C. and Priester, G. B., Refrigeration and Air Conditioning, Prentice-Hall, Inc., Clifton, N. J., 1956

3. Jensen, O., "Investigation of the Thermodynamics of a Reciprocating Compressor," Purdue Compressor Technology Conference, 1972

4. Prakash, R. and Singh, R., "Mathematical Modelling and Simulation of Refrigerating Compressors," Purdue Compressor Technology Conference, 1974

5. Davis, G. L. and Scott, T. C., "Component Modelling Requirements for Refrigeration System Simulation," Purdue Compressor Technology Conference, 1976

6. Jacobs, J. J., "Analytic and Experimental Techniques for Evaluating Compressor Performance Losses," Purdue Compressor Technology Conference, 1976

7. Shaffer, R. W. and Lee, W. D., "Energy Consumption in Hermetic Refrigerator Compressors," Purdue Compressor Technology Conference, 1976

8. Pandeya, P. and Soedel, W., "A Generalized Approach Towards Compressor Performance Analysis," Purdue Compressor Technology Conference, 1978

9. Adair, R. P., Qvale, E. B. and Pearson, J. T., "Instantaneous Heat Transfer to the Cylinder Wall in Reciprocating Compressors," Purdue Compressor Technology Conference, 1972