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X. Dagan

X. Zenan

Y. Yongzhang

J. F. Hamilton

THE COMPUTER SIMULATION OF OIL-FLOODED REFRIGERATION TWIN-SCREW COMPRESSORS

Dagang Xiao, Zenan Xiong and Yongzhang Yu

Chemical Machinery & Equipment Division, Xi'an Jiaotong University, Xi'an, Shaanxi, China

James F. Hamilton

Ray W. Herrick Labs. , Purdue University, U. S. A.

ABSTRACT

A mathematical model considering the effect of real gases is developed to predict the effect of some actual factors, such as gas leakage, heat exchange between gas and flooded oil, and flow resistance at discharge port, etc. , on the performance of oil-flooded twin-screw compressors. A prediction program has been written to be available to ammomia and 12 kinds of freons including R22, R12, R502, etc. . The analytical formulas of geometrical characteristics such as control volume, sealing line length and discharge port area, etc. , for the sample profile are obtained. Some calculated results of P-V diagram, volumetric efficiency and adiabatic efficiency are enumerated in this paper.

NOMENCLATURE

A	area; centre distance between male and female rotors
C_l	specific heat of liquid (oil)
C_v	vapor specific heat at constant volume
D_o	nominal diameter
E	mechanical energy
h	specific enthalpy
i	transmission ratio
k	ratio of C_p to C_v

L length of sealing line

m mass

n_1 speed of male rotor

P pressure

P_1, P_2 characteristic parameters of screw line

t time; profile line parameter

t_o evaporating temperature

t_1 suction temperature

t_k condensing temperature

T absolute temperature

u specific internal energy

v specific volume

V volume

w work

α_t heat transfer coefficient

α flow coefficient

β blockage coefficient of oil-flooded

δ clearance

γ_{cr} the critical ratio of pressures

τ screw line parameter

φ turning angle

Subscripts

1 male rotor

2 female rotor

12 engaging rotors

g gas

i in

o out

s surface; suction

Z_1 male rotor lobe number

Z_2 female rotor lobe number

INTRODUCTION

Rotary twin-screw compressors are widely used in industry for gas compression and refrigeration. It is urgently required due to high cost energy that the machines be high efficient in operation, which would be achieved only when machine performance could be much better enderstood and be predictable. By the manufacturer's past experience and experimental researches om the machine, there are many limits to realize this objective. The computer simulation to its performance by means of mathematical model, however, can meet the need metioned above.

Some studies on the computer simulation of performance for oil-free and oil-flooded compressors (1-4) have been published, but they have not accounted for the effects of thermodynamic parameters of real gas.

The purpose of this paper is to develop a more sophisticated simulation model for analysing oil-flooded twin-screw refrigeration compressor performance, considering the sealing and cooling effects of oil-flooded on the compression cycle and taking the working medium —— refrigerant as real gas. The thermodynamic parameters of freons and ammonia are calculated through Martin-Hou and Rombusch state equations respectively. At the beginning of simulation, it is necessary to calculate the geometrical characteristics of screw compressor. As an example, the standard non-symmetrical profile in our country is discussed in this paper. Because of the complexity of rotor profile, the relations of control volume and discharge port area with turning angle can hardly be expressed in simple and explicit formulas. In this paper, the analytical expressions of them with turning angle are obtained by numerical approximation method. The computer simulation programmes presented in this paper are available to ammonia and 12 kinds of freons.

HYPOTHESES

One basic compressor volume pair including one male groove and one female groove is taken as a control volume to be studied. Since screw compressors are positive displacement machines, the control volume can be modeled by a chamber composed of a piston and a cylinder, connected to suction and discharge valves, leakage paths, and oil-flooded nozzle, as shown in Fig. 1. The temperature and pressure in the control volume are computed from the mass and energy conservation equations with the consideration of:

(1) Volume change due to rotor rotation.

(2) Mass and enthalpy flows of refrigerant and oil, entering or leaving the control volume through the suction port, discharge port and leakage paths.

(3) Heat transfer between refrigerant and oil.

These equations are expressed in the form of a set of differential equations which are simultantaneously solved at small steps using the Milne-Hamming method, which is the method with less amount of computation, higher precision and more stability, but more complication than the fourth order R-K.

In order to simplify the solution of the differential equations, the hypotheses below are made out:

(1) Oil and refrigerant are hypothesized to be seperate fluids. Oil-gas interaction takes place only through heat transfer.

(2) Oil and refrigerant states are homogeneous throughout the control volume at any moment.

(3) Oil is an incompressable fluid.

(4) Oil-refrigerant heat transfer is assumed to be proportional to the control volume and oil-refrigerant temperature difference.

(5) The phases of oil and refrigerant never change.

THEORY

Fundamental Equations

For the thermodynamic system shown in Fig. 2, the First Law of Thermodynamics is expressed as

$$dU = \delta Q + dE_{in} - \delta W - dE_{out} \tag{1}$$

Specifically, for the gas in control volume we have

$$dm_g = \sum_{k=1}^{q} dm_{igk} - \sum_{k=1}^{n} dm_{ogk} \tag{2}$$

$$\delta Q = \alpha_t \, V \, (T_1 - T_g) \, dt$$

$$\delta W = P \, dV + P \, dV_{ol} - P_i \, dV_{il}$$

$$dE_{in} = \sum_{k=1}^{q} h_{igk} \, dm_{igk}$$

$$dE_{out} = h_g \sum_{k=1}^{n} dm_{ogk}$$

$$dU = m_g \, du_g + u_g \, dm_g$$

$$du_g = C_v \, dT_g + \left[T_g \, (\partial P/\partial T)_v - P \right] dv_g$$

$$m_g \, dv_g = d(V - V_l) - V_g \, dm_g$$

$$d\varphi_1 = \omega_1 \, dt$$

The eq.(1), then, can be written to

$$\frac{dT_g}{d\varphi_1} = - T_g \left(\frac{\partial P}{\partial T} \right)_v \frac{1}{m_g \, C_v} \left(\frac{dV}{d\varphi_1} - \frac{dV_l}{d\varphi_1} \right) + \frac{1}{m_g \, C_v \omega_1} \sum_{k=1}^{q} \left(h_{igk} \frac{dm_{igk}}{dt} \right) + \frac{1}{m_g \, C_v \omega_1} \left\{ \frac{(V - V_l)}{m_g} \left[T_g \left(\frac{\partial P}{\partial T} \right)_v - P \right] - u_g \right\} \sum_{k=1}^{q} \left(\frac{dm_{igk}}{dt} \right) - T_g \left(\frac{\partial P}{\partial T} \right)_v \frac{(V - V_l)}{m_g^2 \, C_v \omega_1} \sum_{k=1}^{n} \frac{dm_{ogk}}{dt} + \frac{P_i - P}{m_g \, C_v} \frac{dV_{il}}{dt} - \frac{\alpha_t \, V}{m_g \, C_v \omega_1} (T_g - T_l) \qquad (3)$$

As to the oil in the control volume, we obtain

$$dm_l = \sum_{k=1}^{m} dm_{ilk} - \sum_{k=1}^{r} dm_{olk} \qquad (4)$$

$$\delta Q = \alpha_t \, V \, (T_g - T_l) \, dt$$

$$\delta W = 0$$

$$dE_{in} = \sum_{k=1}^{m} h_{ilk} \, dm_{ilk}$$

$$dE_{out} = h_l \sum_{k=1}^{r} dm_{olk}$$

$$h_l = u_l = C_l \, T_l + B \text{ (a constant)}$$

$$dU = u_l \, dm_l + C_l \, m_l \, dT_l$$

and the eq. (1) can be changed into

$$\frac{dT_l}{d\varphi_1} = \frac{1}{m_l \, \omega_1} \sum_{k=1}^{m} (T_{ilk} - T_l) \frac{dm_{ilk}}{dt} + \frac{\alpha_t \, V}{m_l \, C_l \, \omega_1} (T_g - T_l) \qquad (5)$$

As another simplification, the pressure during suction process is assumed constant:

P = const.

Leakage Flow Computation

The program accounts for several sources of leakage flow into and out of the control volume as enumerated below:

* Leakage through the engaging clearance.
* Leakage through the rotor tip-housing clearance both for leading and trailing control volumes.
* Leakage through blow-hole both for leading and trailing control volumes.
* Leakage through discharge end clearance.

It is difficult to estimate exact leakage values, especially in oil-flooded screw compressors. The leakage is calculated in this paper according to the following simplification assumptions: The refrigerant and oil are uniformly mixed and isolated from heat exchange with their surroundings; refrigerant flow rate is calculated using the well-known formula of flow through a convergent nozzle, and adjusted by flow coefficient and blockage coefficient which identify the effects of refrigerant viscosity and the sealing function of oil-flooded respectively. The refrigerant mass flow rate is

$$\frac{dm}{dt} = \begin{cases} \beta \alpha A P_1 \sqrt{\frac{2k}{k-1} \frac{1}{R T_1} \left(\gamma^{\frac{2}{k}} - \gamma^{\frac{k+1}{k}} \right)} & \gamma_{cr} < \gamma < 1 \\ \beta \cdot \alpha A P_1 \sqrt{\frac{k}{R T_1} \left(\frac{2}{k+1} \right)^{\frac{k+1}{k}}} & 0 < \gamma < \gamma_{cr} \end{cases} \qquad (6)$$

where,

$$\gamma = \frac{P_2}{P_1} \; ; \qquad \gamma_{cr} = \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}}$$

The mass flow through discharge port is also calculated using eq. (6). The discharge port area varies with turning angle. This variation and other geometrical parameters are determined from next part.

GEOMETRICAL CHARACTERISTICS

Geometrical characteristics must be calculated before the computer simulation to compressor working processes. Below are outlined the solution steps for the relations of control volume, sealing line length and discharge port area with turning angle.

Control Volume

Because of the complexity of profiles of male and female rotors, a certain groove section area $F_s(\varphi_1)$ occupied by each other is hardly expressed in a simple and explicit formula; and then, the explicit relation of control volume and its change rate with turning angle can not be obtained. In this paper the analytical relation of $F_s(\varphi_1)$ is gotten through cure fit, i.e. , numerical approximation to the numerical relationship between F_s and φ_1. Therefore, using the following relation:

$$\frac{dV}{d\varphi_1} = \begin{cases} -P_1\left[F_s(\varphi_1) + F_s(\varphi_1+\Phi_1) - F_o\right] & 0 < \varphi_1 < (\tau_{1z}-\Phi) \\ -P_1 F_s(\varphi_1) & (\tau_{1z}-\Phi) \leq \varphi_1 < \varphi_{1k} \\ -P_1 F_o & \varphi_{1k} \leq \varphi_1 \leq \tau_{1z} \\ -P_1 F_{s1}(\varphi_1) & \tau_{1z} < \varphi_1 \leq (\tau_{1z}+\varphi_{1k}) \end{cases} \tag{7}$$

where,

$$F_o = \frac{C_n D_o^2}{4} \; ; \qquad \Phi_1 = \varphi_{1k} - (\tau_{1z} - \Phi)$$

$$F_{s1}(\varphi_1) = F_o - F_s(\varphi_1 - \tau_{1z})$$

and

$$V(\varphi_1) = -\int_{\varphi_1}^{\tau_{1z}+\varphi_{1k}} \left(\frac{dV}{d\varphi_1}\right) d\varphi_1 \tag{8}$$

we can solve the problem metioned above.

Sealing Line Length

A knowledge of the sealing line length surrounding the control volume is essential to calculate the vapor-leakage. For this purpose, the correlation of the length with turning angle is deduced; and the solution steps are simply shown below.

According to the engaging principle, the general expression of instantaneous meshing line can be obtained from the equation of groove surface plus their engaging condition.

The general expression for screw groove surface are

$$X_{sk} = X_k(t)\ \mathrm{COS}(\tau) - Y_k(t)\ \mathrm{SIN}(\tau)$$

$$Y_{sk} = X_k(t)\ \mathrm{SIN}(\tau) + Y_k(t)\ \mathrm{COS}(\tau)$$

$$Z_{sk} = \pm P_k\, \tau_k$$

where,

$X_k(t)$, $Y_k(t)$ are the coordinates of rotor section profile and k=1, 2 identify the male rotor and female rotor respectively.

And the engaging condition is generally expressed

$$\frac{\partial Y_{s2}}{\partial t} (K Y_{s2} + A \sin\varphi_2) + \frac{\partial X_{s2}}{\partial t} (K X_{s2} - A \cos\varphi_2) = 0$$

where,

$$K = i + 1$$

Therefore, we obtain the parameter equation of instantaneous meshing line

$$x_k = x_k(\tau,\varphi_2)$$
$$y_k = y_k(\tau,\varphi_2)$$
$$z_k = z_k(\tau,\varphi_2)$$

and its length at turning angle $_2$

$$L(\varphi_2) = \int_b^e \sqrt{(\frac{\partial x_k}{\partial\tau})^2 + (\frac{\partial y_k}{\partial\tau})^2 + (\frac{\partial z_k}{\partial\tau})^2} \quad d\tau$$

We associate the inatantaneous meshing curve with the other sealing units consisting of the control volume, and can obtain the length of the other sealing lines with turning angle.

Discharge Port Area

Discharge port area includes a radial and an axial. The former can easily be obtained, but the later is not so easy since the complicated rotor profile. Numerical approximation is also used to solve the problem. The numérical relationship between characteristics area and turning angle is obtained according to the following steps:

Using the relation between rotation coordinates (Fig. 3a):

$$X = x(t) \cos\alpha - y(t) \sin\alpha$$
$$Y = x(t) \sin\alpha + y(t) \cos\alpha$$

we obtain the equation of curve $C_2'D_2'$ at turning angle α; therefore, we write the parameters of intersecting point G:

$$t = t(\alpha)$$

$$t_c = t_c(\alpha)$$

between curve $C_2^{\prime}D_2^{\prime}$ and a certain characteristic curve C_2D_2:

$$X_c = X_c(t_c)$$

$$Y_c = Y_c(t_c)$$

According to the area integration formula

$$f(\alpha) = \frac{1}{2}\int_b^e (\dot{Y}X - \dot{X}Y)\,dt$$

the discharge port area is obtained.

NUMERICAL EXAMPLE OF THE SIMULATION

In order to calculate a leakage flow rate, the refrigerant state in the grooves beyond the leakage paths must be known. However, the state of refrigerant in the leading control volume is not certain at the beginning of simulation calculations. Therefore, an initial state may be assumed under the condition of no leakage. Sebseguently, the leakage flow rate is calculated and the state is corrected with the Milne-Hamming procedure. This calculation is iterated until the system converges. Finally, volumetric efficiency, adiabatic efficiency, etc. are calculated.

Calculating Conditions

In this paper, performance is calculated for oil-free or oil-flooded refrigeration compressors with geometrical and working parameters of:

$$D_o = 160 \text{ mm},\ \tau_{1z} = 300^\circ,\ Z_1 = 4,\ Z_2 = 6;\ n_1 = 2960 \text{ rpm}$$

Effect of Leakage and Oil-flooded

Compression process is taken as an example to analyse the effect of leakage and oil-flooded on the compression cycle. The calculated results are illustrated in Fig. 4 and Fig. 5, which are under the hypothesis that the beginning of compression is the end of oil-flooded.

It is generally thought that the adiabatic compression process line is below the isentropic line at same volume or same turning angle since the gas leakage. That is true only when there exists external leakage or the external leakage is much greater than the internal leakage from the leading control volume. Some situations, however, are opposite in screw compressors since the higher temperature gas leaking from leading control volume heats the gas in the control volume, and the gas mass in the control volume increases during the beginning period of compression process, as shown in Fig. 4. When rotor tip-housing clearance, i.e. , the gas leakage from the leading control volume is small to a certain extent, a phenomina appears that the adiabatic compression process line is above the isentropic during the beginning period of compression process and becomes below during the following period, as shown in Fig. 5.

From Fig. 4 and Fig. 5, we know that the change rate of pressure with control volume is greater during the beginning period than the others since the function of oil-flooded heating; and the rate becomes smaller since the function of oil-flooded cooling.

The calculated volumetric efficiency and adiabatic efficiency for R22 are listed in Table 1.

Table 1

R22; $t_o = -20^\circ C$, $t_1 = -10^\circ C$, $t_k = +35^\circ C$

$\delta_{12} = 0.1$ mm, $\delta_1 = 0.22$ mm, $\delta_2 = 0.24$ mm, $\delta_e = 0.06$ mm

$\tau_{1z} = 300^\circ$, $\varphi_{1c} = 290^\circ$:

$\eta_v = 86.8\%$, $\eta_{ad} = 90.7\%$

CONCLUSIONS

A simulation model, in which the gas and oil leakage, and heat transfer between oil and gas are accouted for, has been developed for evaluating the performance of oil-free or oil-flooded screw compressors with the working medium of perfect gas or real gas. A generalized performance prediction computer programme is available to the refrigerants of 12 kinds of freons and ammonia; and is also available to various rotor profiles and geometrical parameters to optimize them.

According to numerical approximation theory, the control volume and discharge port area, etc. can be written in simple analytical formulas.

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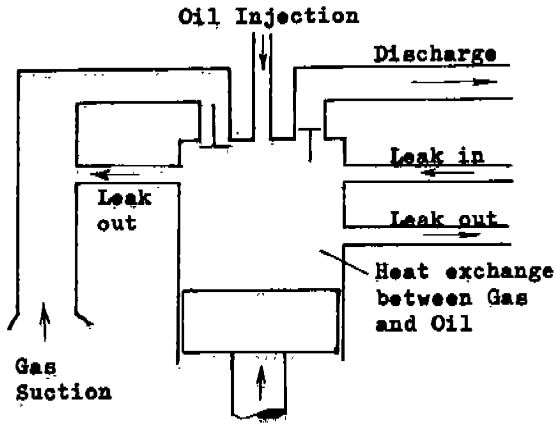

Fig. 1 Working Space Model

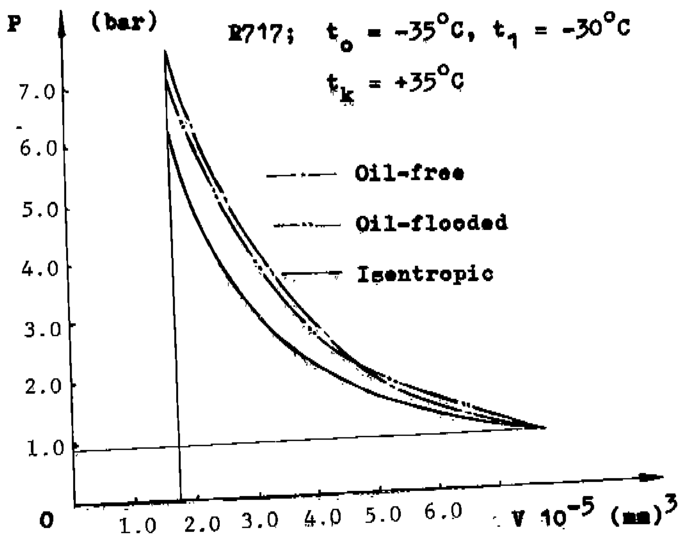

Fig. 4 P-V Diagram of Compression Processes

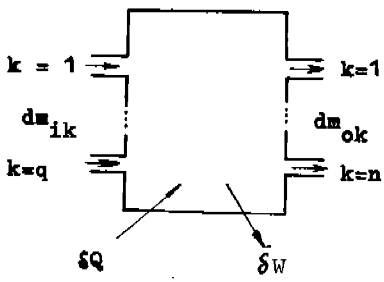

Fig. 2 Thermodynamic System

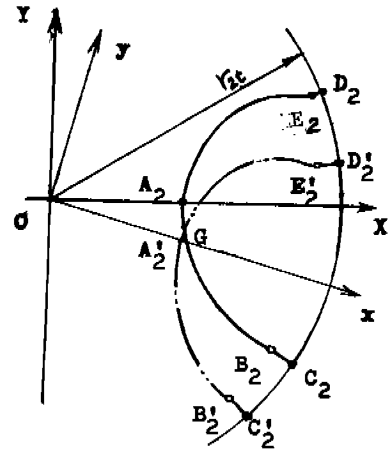

Fig. 3 INtersecting Point

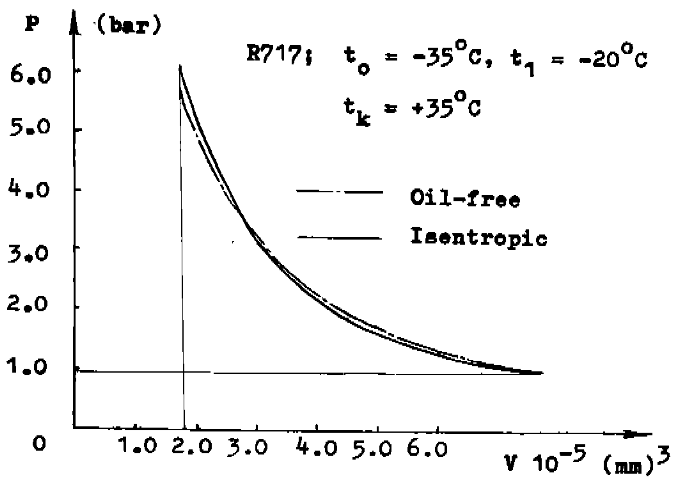

Fig. 5 P-V Diagram of Compression Processes

THE DEVELOPMENT OF A LONGER LIFE AND SUPERIOR PERFORMANCE COMPRESSOR PLATE VALVE

Robert E. Drews

Ingersoll-Rand Company
Gas Compressor Group
Painted Post, NY 14870

ABSTRACT

The constant desire to minimize machine downtime while obtaining peak performance of the equipment is the objective of every manufacturer and end user. One component of reciprocating compressors which has a definite affect on both of these is the compressor valve. The author reviews the development program of a reciprocating compressor valve that delivers a longer average life ratio along with an increase in the valves' flow coefficient.

The first subject area will be that of the valve life. Included will be:

1. The evaluation of materials and hardness levels used in the manufacturing of valve plates. This consisted of placing a cyclic load on samples of identical dimensions with varying material and material hardness. The process was also used to evaluate several manufacturing processes, such as laser cutting, water jet cutting, milling, and die cutting of the valve plate ports.

2. The results of valve plate motion tests which utilized proximity probes to monitor valve plate motion and also indicated the plates' impact velocity.

3. The results of valve plate stress tests which consisted of attaching strain gauges to the valve plate.