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SIMULATION OF THE WORKING PROCESS OF AN OIL FLOODED HELICAL SCREW COMPRESSOR WITH LIQUID REFRIGERANT INJECTION

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Abstract

In this paper a mathematical model for the working process of a refrigeration helical screw compressor is presented and discussed. The manner in which geometric parameters such as cell volume, sealing line length, blow hole area, discharge port area etc. are related to rotation angle is established and used in the thermodynamic model which is based on real gas laws. The influence of factors such as the degree of oil flooding, liquid refrigerant injection, vapour charge from the economizer, different refrigerants and partial loading etc., are considered simultaneously and separately in the model. The theoretical basis of the program is described in the paper. Moreover in order to confirm the computer model the performance predicted by it is compared with the results from compressor tests. The results are compared for different operating conditions.

INTRODUCTION

The helical screw compressors, which are applied in process refrigeration systems, usually operate over a range of conditions. Simulating the working process of a refrigeration helical screw compressor involves a number of important considerations as follows:

1. A quantity of oil much greater than the minimum quantity needed for proper lubrication is supplied to the compression chamber. This additional oil seals the clearances between the rotors and compression chamber walls, affects the discharge temperature, and permits the simple mechanical arrangement of one rotor driving the other. With oil injection, screw compressors can work at higher compression ratios. This has permitted their application to process refrigeration.

2. Liquid refrigerant is injected directly into the oil/refrigerant stream flowing through the compressor. As the liquid refrigerant is heated by the gas/oil, it evaporates and cools both the oil and the vapour being compressed so as to eliminate the need for separate oil cooling. Refrigerant injection is not used in place of, but rather in conjunction with, oil injection.

3. The vapour generated in superfeed, normally called flash, is injected into the compression chamber through a port in the compressor casing, the site of which is chosen to ensure that injection takes place at an early stage in the compression process. The superfeed circuits are similar in design to the economizers often used in multi-stage compression systems to cool the liquid refrigerant from the condenser prior to its being supplied to the evaporators.

4. Capacity control is needed to enable the helical screw compressor to meet the wide range of load demands which are a feature of modern refrigeration plants. The most common method makes use of a slide valve which allows a measured quantity of the compressed cavity volume to "blow out" back to suction.

5. The refrigerants often used in process refrigeration systems, such as R12, R22, R717, and R134a etc., cannot be considered as perfect gases. In this paper the real gas equations are

used to calculate their properties for use in the mathematical model.

A mathematical compressor model capable of handling a variety of conditions simultaneously, will be useful for research purposes and the design of refrigeration helical screw compressors.

GEOMETRICAL CALCULATION

In order to simulate the working process of a screw compressor, all the geometrical relationships must be calculated first. A geometrical calculation program has been developed for this purpose, the flow diagram for which is shown in Figure 1.

The program uses a subprogram to generate the male and female rotor profiles. There are many choices in this subprogram, for example between equal and unequal rotor diameters, whether the profile is single-sided or double-sided, the male rotor lobe number, the female rotor flute number, and the other profile basic parameters.

The geometrical calculation program can read the coordinates of any profile if given in the appropriate data file.

The following data are needed for the geometrical calculations and must be entered by the user:

1. Wrap angle and screw pitch.
2. Built-in volume ratio.
3. Slide-valve's geometry and position parameters (if a slide-valve is used).
4. Construction and position parameters of the liquid refrigerant injection port, oil injection port and superfeed port.
5. Machining parameters (if cutter profiles are needed).

All the geometrical characteristics used in the mathematical model are expressed in terms of the angle of the rotation of the male rotor. The start angle is the angle where the cavity volume equals zero. A great many geometrical relationships can be calculated and output as data files.

MATHEMATICAL MODEL

Assumptions for the Mathematical Model

1. The vapour flow through the suction port, discharge port, superfeed port, slide-valve by-pass port and every leakage path is assumed isentropic, and the vapour flow through the suction, discharge and slide-valve by-pass ports is assumed incompressible.

2. The pressures in the suction and discharge chambers and before the superfeed port, liquid refrigerant port and oil injection port are constant.

3. Gas, oil and liquid refrigerant are assumed to be separate fluids, and gas leakage through any leakage path does not result in mass changes of oil or liquid refrigerant in the considered cavity volume.

4. Heat transfer between the vapour and the compressor rotors or housing is treated as being negligible since it is much smaller than the energy exchanges which occur within the cavity as a result of the compression process and the flashing of refrigerant, whether injected as neat liquid or dissolved in the injected lubricating oil.

5. Flash rate of the liquid refrigerant injected and dissolved in oil is constant. A flash coefficient is used to represent this rate.

Basic Equations for the Mathematical Model

The control volume of a screw compressor is its cavity volume. For the control volume the conservation equation for internal energy in terms of the rotation angle can be written as:

$$\begin{cases} \frac{dU}{d\varphi} &= \left(\frac{dI}{d\varphi}\right)_{in} - \left(\frac{dI}{d\varphi}\right)_{out} + \frac{dQ}{d\varphi} - \frac{dL}{d\varphi} \\ \left(\frac{dI}{d\varphi}\right)_{in} &= \left(i_{in}\left(\frac{dm}{d\varphi}\right)_{in}\right)_{gas} + \left(i_{in}\left(\frac{dm}{d\varphi}\right)_{in}\right)_{FR} \\ \left(\frac{dI}{d\varphi}\right)_{out} &= \left(i_{out}\left(\frac{dm}{d\varphi}\right)_{out}\right)_{gas} \\ \frac{dQ}{d\varphi} &= \frac{A\alpha(T_{oil} - T_{gas})}{\omega} \\ \frac{dL}{d\varphi} &= p\frac{dV}{d\varphi} \end{cases} \tag{1}$$

where,

U Internal energy in the control volume, $[J]$.

I Fluid enthalpy. $[J]$.

Q Heat exchanged by gas with oil droplets injected through their surface area, $[J]$.

L Work exchanged by the system with the surroundings through the control surface area, $[J]$.

φ Rotation angle of the male rotor, $[^\circ]$.

i Fluid specific enthalpy, $[J/kg]$.

m Fluid mass, $[kg]$.

A Surface area of the oil droplets injected, $[m^2]$.

α Heat transfer coefficient, $[W/(m^2 \cdot K)]$.

T Fluid temperature, $[K]$.

ω Angular speed of the male rotor, $[^\circ/s]$.

p Gas pressure in the control volume, $[N/m^2]$.

V Cavity volume, $[m^3]$..

in All energies or masses which come into the control volume through suction or discharge port, superfeed port, injection ports and all leakage paths.

out All energies or masses which go out the control volume through discharge port, slide-valve by-pass port, and all leakage paths.

gas Vapour which comes into, goes out of or is in, the control volume.

oil Lubrication oil both drained and injected into the cavity volume.

FR Flashed refrigerant, due either to injected liquid flashing or to refrigerant degassing from oil.

The conservation equation for the mass in the control volume in terms of the rotation angle can be written as:

$$\frac{dm}{d\varphi} = \left(\frac{dm}{d\varphi}\right)_{in} - \left(\frac{dm}{d\varphi}\right)_{out} \tag{2}$$

The energy and mass conservation equations are used for the suction, compression and discharge processes. They are the most important equations for the mathematical model.

In order to calculate any flow rate through any port or any leakage path, the following continuity equation is needed:

$$\frac{dm}{d\varphi} = \rho W A/\omega \tag{3}$$

where,

ρ Fluid density, $[kg/m^3]$.

A Flow area, $[m^2]$.

W Fluid flow speed through the flow area, $[m/s]$.

When this equation is used to calculate flow rates through the suction port and discharge ports, their effective flow areas are needed, so modification coefficients are used to modify the areas obtained from the geometrical calculation program.

For the discharge process and slide-valve by-pass gas flow, the fluid flow speed through the flow area can be calculated by the following momentum equation:

$$WdW + \frac{dp}{\rho} = 0 \tag{4}$$

For the suction process, the flow speed can be calculated by equation 3, and then from equation 4 the suction resistance can be obtained. Equation 4 can also be used to calculate the injection speeds of oil and liquid refrigerant, but when substituting these speeds into equation 3 to yield the oil and liquid refrigerant injection rates, modification coefficients, obtained according to experience or from test results, are needed.

The gas flow through any leakage path or superfeed port can not be considered as incompressible since the Mach number is often large and sometimes equals one. The speed should be calculated by the following energy conservation equation:

$$WdW + di = 0 \tag{5}$$

The maximum flow speed through any leakage path can approach local sound speed if the pressure difference over the path is enough large. The local speed of sound can be calculated by the following equation:

$$A = \sqrt{\left(\frac{\partial p}{\partial \rho}\right)_s} \tag{6}$$

Besides the equations mentioned above, many equations used to calculate the thermodynamic properties of refrigerants in the mathematical model are needed. These equations are:

- Equation of liquid density.
- Equation of vapour pressure.
- Equation of state.
- Equation of latent heat of vaporization.
- Equation of vapour enthalpy.
- Equation of vapour entropy.

The above equations are different for different refrigerants. Four refrigerants, R12, R22, R143a and R717, are considered in the model.

The above equations form a system, which can be solved by the fourth-order Runge-Kutta procedure and the recursion procedure. The calculated results are accepted after the iteration procedure has given a difference between two consecutive pressure volume graphs which is smaller than a small value prescribed in advance.

Computer Program for the Mathematical Model

Figure 2 shows the flow diagram of the computer program developed for the mathematical model.

Besides the geometrical characteristics which are either calculated or supplied, which are read by the computer program for the mathematical model, the following data are needed and should be entered:

1. The rotational speed of the male rotor.
2. Rotor tip clearances, rotor end clearances and the clearance between the two rotors.
3. Refrigerant. There are four refrigerants, R12, R22, R134a and R717, which can be chosen.
4. Operating condition: evaporating temperature, condensing temperature, discharge pressure.
5. Gas state in superfeed, pressure before injection and flow rate or flow rate coefficient(if superfeed is on duty).
6. Liquid refrigerant state before injection and flow rate or flow rate coefficient (if liquid refrigerant is injected into the compressor).
7. Oil pressure and temperature before injection and flow rate or flow rate coefficient.
8. How much refrigerant is dissolved in the oil.
9. Flash coefficient, suction port area modification coefficient and discharge port area modification coefficient.

The program can calculate and output pressure, specific volume, temperature, entropy, specific internal energy, mass, sound speed in the cavity volume, suction and discharge flow speeds. All these calculated results are related to the rotation angle, so the plots of them versus rotation angle or cavity volume can be drawn. In addition the following characteristics for the working process can be obtained:

1. Gas flow rates through superfeed and slide-valve by-pass port.
2. Liquid refrigerant flow rate through the injection port.
3. Oil flow rate through the injection port.
4. Flow rates of unflashed liquid refrigerant injected or in oil.
5. Leakage rate through certain leakage paths.
6. Theoretical capacity, real capacity and volumetric efficiency.
7. Indicated power, isentropic power and isentropic indicated efficiency.
8. Other useful information.

SIMULATION AND TEST RESULTS COMPARED

In order to confirm the computer model the performance predicated by it is compared with the results from compressor tests. Some results are presented here.

Figure 3 shows some calculation and test results of an oil flooded screw compressor without liquid refrigerant injection. The refrigerant is $R22$, its rotation speed is $3000rpm$, the condensing temperature is $25°C$ and the suction superheat temperature is $30°C$.

Figure 4 shows some calculation and test results of an oil flooded screw compressor with liquid refrigerant injection. The compressor has the same operating condition as Figure 7 except that liquid refrigerant is injected and the condensing temperature is now $35°C$.

In both Figure 3 and Figure 4 the predicted and test volumetric efficiencies show the same trends and almost same values. The curves of predicted indicated efficiency and test total efficiency have same shape. For different pressure ratios the ratio of the total efficiency to the indicated efficiency which is the mechanical efficiency of the compressor is very nearly constant. This is expected since the rotational speed is constant and the mechanical efficiency depends predominantly on it.

When a screw compressor is on a partial load condition and its slide-valve by-pass port is opened, both the capacity and volumetric efficiency will be reduced. Figure 5 shows the relationship between the capacity and volumetric efficiency when the slide-valve is located at different positions. The operating condition is the same as for Figure 3. From Figure 5 it can be seen that the computer model also has very high prediction accuracy for partial load condition.

Figure 6 shows four $p-V$ diagrams corresponding to the following four conditions, oil injection being common to all:

- Oil injection alone.
- Liquid refrigerant injection.
- Superfeed on duty.
- Slide-valve on duty.

The other operational conditions are the same for the four $p - V$ diagrams.

CONCLUSIONS

1. A geometrical calculation program for screw compressors has been developed. The program can generate various profiles and allows all basic parameters to be varied. The coordinates of any profile and its basic parameters can be read by the program. After the profile is generated or its coordinates and basic parameters are entered, the program will calculate all the characteristics in terms of rotation angle, which are needed by the mathematical model, and all the geometrical characteristics, which are useful for both design and research purposes.

2. A computer program for the mathematical model of the working process of a refrigeration screw compressor has been developed. In the model, which is based on real gas laws, the influence of factors such as the degree of oil flooding, liquid refrigerant injection, vapour charge from the superfeed, different refrigerants and partial loading etc., are considered simultaneously and separately. The program results show excellent agreement with the test data available to date. The program predicts the performance of a refrigeration screw compressor of any condition accurately, so it is very useful for compressor design and research.

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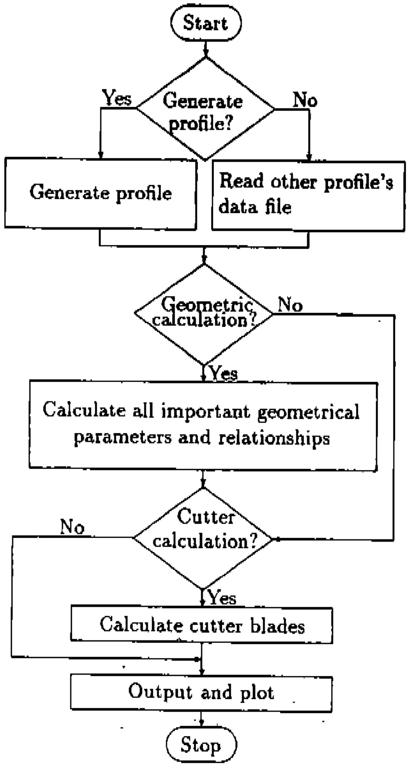

Figure 1 Geometrical calculation flow diagram

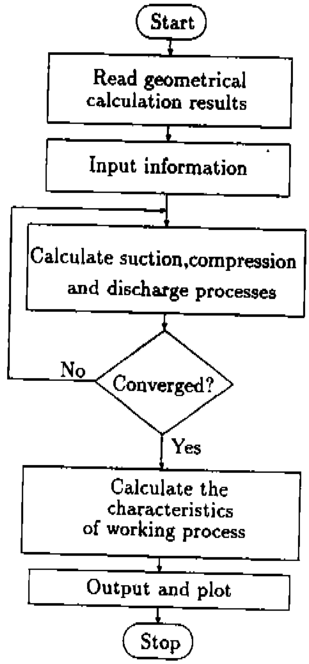

Figure 2 Process calculation flow diagram

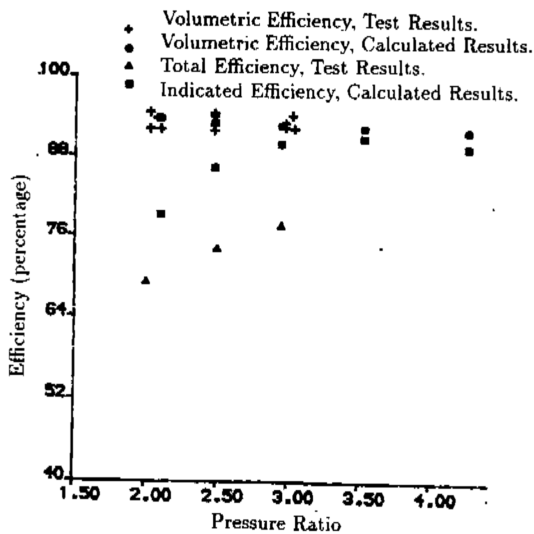

Figure 3 Prediction and test results for oil injection

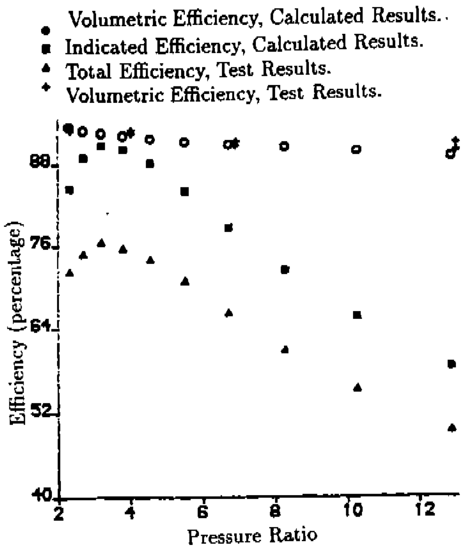

Figure 4 Prediction and test results for oil and refrigerant injection

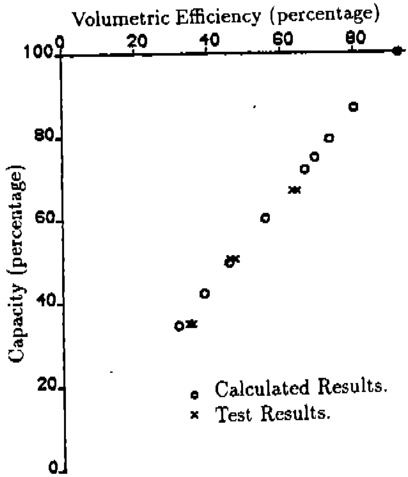

Figure 5 Prediction and test results for partial load condition

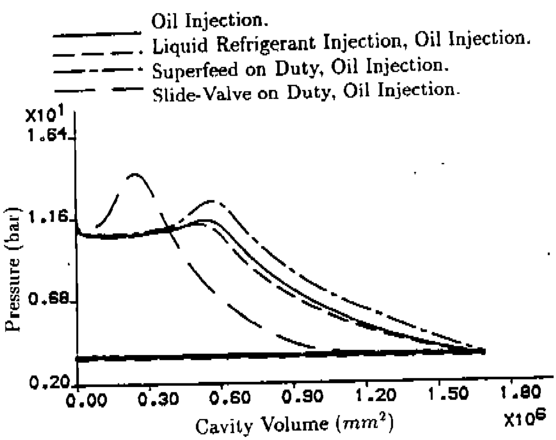

Figure 6 $p-V$ diagrams