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Computer Modeling of Single -Screw Oil Flooded Refrigerant Compressors

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COMPUTER MODELING OF SINGLE-SCREW OIL-FLOODED REFRIGERANT COMPRESSORS

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ABSTRACT

A computer model of an oil-flooded single-screw refrigerant compressor operating at 100% capacity and at matched discharge pressures was developed which predicts internal conditions using empirically determined values. The model gives detailed thermodynamic information concerning the internal processes. It calculates the total required shaft power, the refrigerant preheat and compression powers, and the oil shear displacement powers. It also calculates the contributions to volumetric inefficiency from preheat, recirculating oil and refrigerant leakage. Finally, it predicts the isentropic and volumetric efficiencies of the machine.

The results of the computer model are compared with available data and the effect of oil and refrigerant leakage on the computer performance is shown.

INTRODUCTION

The single-screw compressor is a rotary positive-displacement compressor used in air and refrigeration applications. The compressor gets its name from the single-screw rotor that meshes with two sealing rotors called star rotors or gaterotors (Figure 1). Its basic advantages of compactness, versatility, reliability, maintainability, weight, noise and vibration as compared to existing reciprocating units have prompted an investigation into this mechanism (1). At this point in the investigation, there is an interest in developing a model that can be used to evaluate design changes that might lead to better performance for the refrigerant compressor.

The operational principle of the single-screw compressor is straight forward and has been discussed by various authors (2-6). Gas enters the suction chamber and fills the portions of the threads that are exposed. After a thread is completely filled with suction gas, a star rotor, driven by the screw rotor, rotates a tooth into that thread and closes it off. The gas is trapped in a pocket formed by the three sides of that thread, the housing, and the newly engaged tooth. Further rotation of the screw rotor, or main rotor as it is called in practice, causes the tooth to sweep through that thread, reducing the pocket volume and compressing the trapped gas. The compression process continues until the leading edge of the thread uncovers a discharge port located in the casing. At that instant, the compression process ends and the gas in the pocket flows out radially through the fixed discharge port and into the discharge line. The gas is forced out by the final sweeping action of the tooth. It is important to realize that as the gas is being compressed on one side of the star tooth, gas is filling the thread on the other side. Once the discharge process for a thread is over, the thread is already completely filled with suction gas and is ready for the next star rotor to engage a tooth to repeat the process.

Figure 1 Single-Screw Air Compressor

The type of single-screw compressor investigated is oil flooded. Oil is injected into each thread on the compression side of the tooth.

The injection starts when the tooth closes the thread and ends when the thread passes the injection orifices. The oil has a silencing effect and also serves as a coolant, lubricant and sealant. The presence of clearances within the machine makes it necessary to inject oil. The oil fills these clearances, elmininating them as possible refrigerant leakage paths and thus increasing the machine's delivery capacity (mass flow rate). Oil is rejected with refrigerant through the discharge ports. This oil/refrigerant mixture then passes to a large separator tank. The oil is driven from the separator tank and into the compressor by the system's discharge pressure so no external pump is necessary.

The objective of this paper is to present the development and results of a computer model that is capable of predicting the performance of a single-screw oil-flooded refrigerant compressor operating at 100% capacity and at matched discharge pressures. A matched discharge pressure is defined as one in which the compression process ends exactly at the discharge pressure. The model depends on two empirical values before it can be applied and so the method of determining these empirical values for a given compressor will be outlined. The development of the model will then be presented in the sections entitled "overall compression system, modeling assumptions and physics". The computational scheme and results for the model will be given in the section entitled "computer model". The results are for a commercially available compressor and will be compared with available data.

OVERALL COMPRESSION SYSTEM

Since the physical processes that are described in this paper are done so in reference to components used in the overall compression processes, it is necessary to describe these components and their relative positions. Figure 2 shows these basic components. The single-screw compressor shown receives the incoming refrigerant and injected oil and discharges them into a common discharge line. The discharge line contains a valve that controls the discharge pressure. In this investigation, it is assumed that the discharge pressure matches the pressure achieved in a pocket (closed thread) just before the pocket is opened to a discharge port. The oil cooler shown controls the oil injection temperature. The oil injection temperature and flow rate can significantly affect the performance of the machine. The oil/refrigerant separator shown receives the discharged oil/refrigerant mixture and by the use of gravity separates the two. The evaporator and condenser are shown because it is standard practice to quote their respective saturation temperatures as an indication of the inlet and discharge pressures.

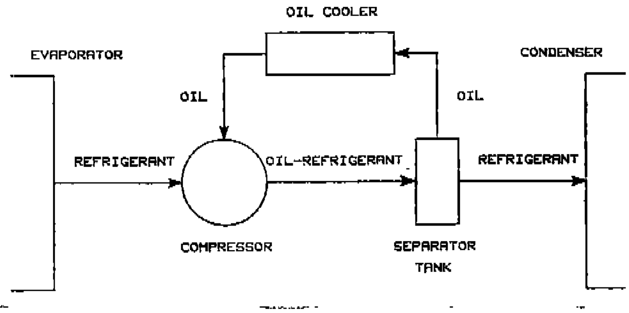

Figure 2 Basic Components of the Compression System

MODELING ASSUMPTIONS

The modeling of the overall compression process centers around four refrigerant processes and a treatment of oil and refrigerant leakages. The four refrigerant processes that make up the overall compression process are the suction process, the closed compression process, the discharge process and the separation process. The assumptions used in this analysis are:

(1) Potential and kinetic energies are negligible;
(2) The inlet refrigerant is preheated to a uniform temperature entering a thread;
(3) There is negligible pressure loss in the suction and separation process;
(4) Only oil leaks from a thread;
(5) The heat transfer between the refrigerant and oil in the closed compression process is negligible;
(6) The heat transfer between the discharge oil and refrigerant will continue until they reach thermal equilibrium;
(7) Refrigerant leakage is restricted to discharge port leakage paths;
(8) The refrigerant properties in a thread will be the same as in the other threads when they are at the appropriate phase angle;
(9) The refrigerant leakage process is an isentropic expansion process;
(10) The injection oil temperature equals the discharge oil temperature.

PHYSICS

Four Processes for the Refrigerant

The overall compression process can be broken down into a series of four smaller processes. The first of these processes is called the suction process. This process starts when the gas leaves the evaporator and ends when the gas has made its way through the sheets of centrifuged oil into an open thread. Tests have shown that during this process the gas receives heat and so the name preheat has been affixed to this phenomena.

The second process is called the closed compression process. The beginning of this process, termed suction closure, is marked by the trailing thread meeting the trailing flank of the star tooth thereby closing off the inlet port to form a pocket of trapped gas. The end of this process is marked by the point where further rotation of the main rotor will uncover the discharge port. As the star tooth sweeps through the closed compression process, the pressure increases and some of the oil that is trapped at suction closure and some of that which is injected into the process is forced out of the pocket through the running clearances.

The discharge process starts when the closed compression process ends. During this third process, refrigerant and oil are expelled through the discharge port while some oil continues to leak back to the suction chamber. The fourth process termed the separation process occurs after the oil and refrigerant leave the discharge port area and enter the discharge line. This constant pressure process continues until the refrigerant and oil exit the separator tank in their respective lines. The time required for this final process allows the oil and refrigerant to reach thermal equilibrium in the tank.

The refrigerant states corresponding to the four component processes of the overall compression system are shown in Figure 3. State 1 is at the exit of the evaporator; State 2 is the thermodynamic state of the refrigerant that is in a thread at suction closure; State 3 is the thermodynamic state of the refrigerant at discharge and State 4 is at the exit of the separator tank. State 5 represents the refrigerant that is throttled back to the suction chamber through discharge port leakage paths; this throttling process is not considered one of the component processes because it is not in series with these processes.

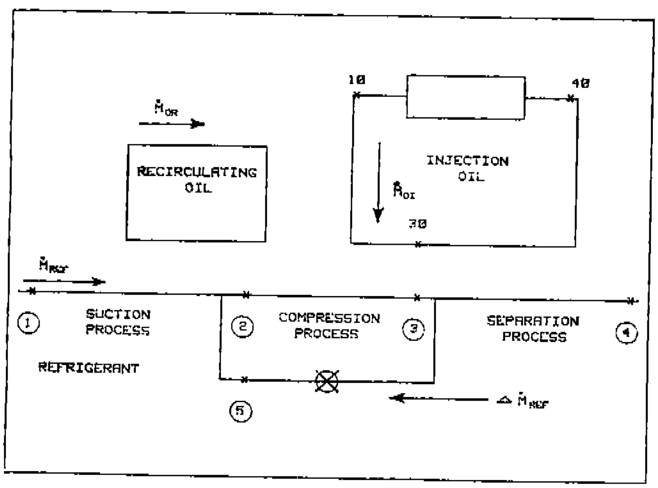

Figure 3 Four Component Process

Injection and Recirculating Oil Flows

Figure 3 also shows the oil that is injected ($\dot{m}_{oi}$) into the compression process. The recirculating oil ($\dot{m}_{or}$), oil that is present within the machine to seal clearances, is shown separated from the injected oil. The separation of these two basic oil flows is for analysis purposes only. The separation of the discharge oil and refrigerant is also for analysis purposes.

Figures 4 and 5 show the nine oil leakage paths and the gas leakage path indentified by Bein (5). The Bein oil leakage model is incorporated with slight modification in this computer model. The presence of oil affects the volumetric and isentropic efficiences of the compressor. Oil present at suction closure takes up available volume for refrigerant and so degrades the volumetric efficiency. It is hoped that the loss of volumetric efficiency due to this cause is more than offset by the reduction of refrigerant leakage through the clearances. The oil that leaks through the clearances is assumed to be present at suction closure displacing refrigerant.

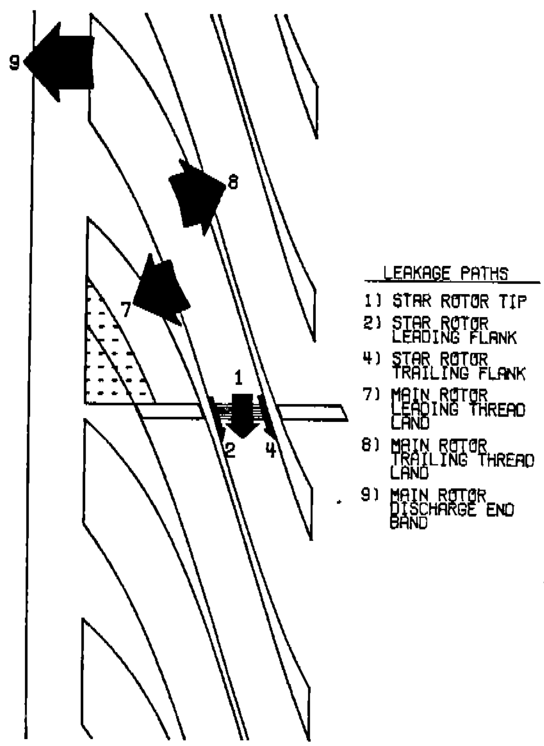

Figure 4 Bein Model Oil Leakage Path Definition - Main rotor View

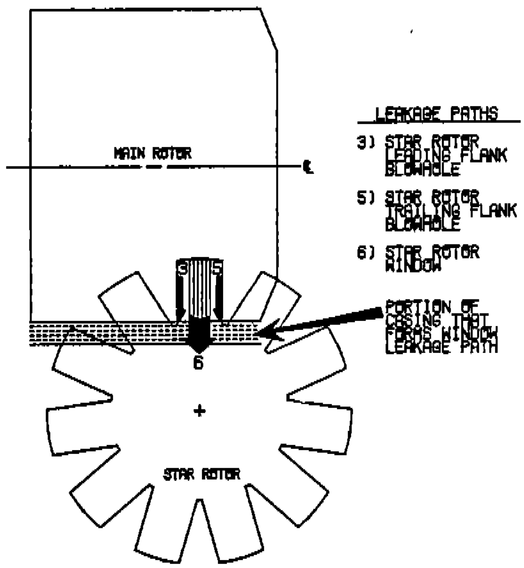

Figure 5 Bein Model Oil Leakage Path Definitions - Star Rotor View

The oil within the compressor absorbs some shaft power and so degrades the isentropic efficiency of the compressor. This oil absorbs shaft power through the mechanisms of viscous shear, displacement, bearing lubrication and churning. The oil can also transfer its absorbed power to the incoming refrigerant causing a preheat loss in both volumetric and isentropic efficiencies of the compressor. The increase in specific volume due to preheat causes an increase in compressor work and a decrease in trapped refrigerant mass. The equations for the power absorbed by the recirculating oil through the mechanisms of oil shear and oil displacement are:

$$P_{sh} = \frac{N^2 \ (2\ \pi)^2}{60^2} \quad \frac{\mu}{a} \ h' \ \frac{D}{2}^2 \ A_{sh} \qquad \text{Eq 1}$$

$$P_d = 2 \ \ NM \ \frac{N}{60} \int_0^{V_{or}} (p_p - p_1) \ dV_{or} \qquad \text{Eq 2}$$

The power absorbed by the injected oil is given approximately by

$$P_{oi} \approx \dot{V}_{oi} \ (P_4 - P_1) \qquad \text{Eq 3}$$

Refrigerant Leakage

Figure 3 shows refrigerant leaking ($\Delta\dot{m}_{ref}$) from the discharge side of the compressor to the suction side of the compressor. In order to be consistent with the Bein leakage model assumption that only oil leaks during the closed compression process, refrigerant leakage is then limited to the discharge port leakage paths. This means refrigerant leakage goes from the discharge conditions to the suction conditions as indicated by Figure 3. The refrigerant leakage is assumed to occur adiabatically. The leakage flow rate $\Delta\dot{m}_{ref}$ is calculated assumming isentropic expansion to the throat and using a discharge leakage path coefficient F_{DP}. In all the cases investigated, the refrigerant leakage flow was choked. The refrigerant leakage equation is

$$\Delta\dot{m}_{ref} = F_{DP} \ \rho_5 \ \ A_5 \ \ V_5 \qquad \text{Eq 4}$$

Control Volume Analysis

Figure 6 shows the process diagram and the six control volumes that are used to analyze the overall process. The shaft power, P_s, is shown divided into power absorbed directly by the oil and power absorbed directly by the gas. In this computer model, the power absorbed by the recirculating oil is passed on to the incoming refrigerant in the form of heat (Q_{cv2}) to preserve the steady state condition of this oil and to account for a part of the observed preheating of the incoming gas. In order to satisfy assumption 10, no heat is transferred between the recirculating oil and injected oil. No heat is shown transferred to the refrigerant from oil during the closed compression process (assumption 5) because there is relatively little time (6 millisec) for this to occur. Heat (Q_{cv4}) is shown being transferred between the discharge refrigerant and oil because the discharge temperature of the refrigerant is greater than the discharge temperature of the oil. Figure 6 shows heat (Q_{cv1}) transferred from the oil heat exchanger to the environment which indicates the overall compression process is nonadiabatic.

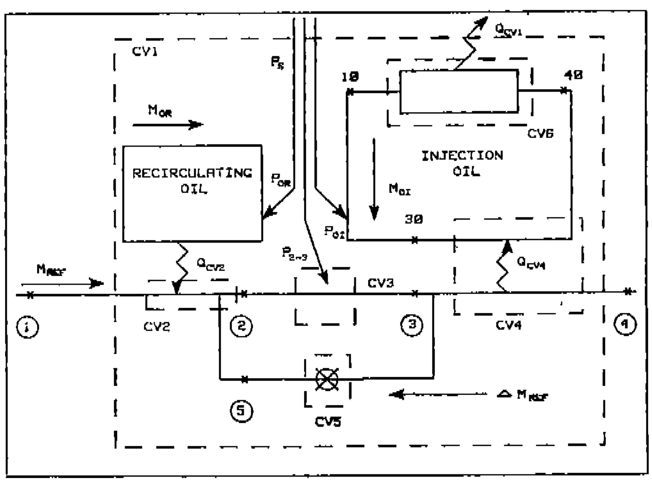

Figure 6 Control Volumes and Energy Transfers

The governing equations used throughout this analysis can be found by performing a steady-state steady-flow first law analysis for each control voume shown in Figure 6. For control volumes 1-6 respectively, these equations are:

$$P_s = \dot{m}_{ref} (h_4 - h_1) + Q_{cv1} \qquad \text{Eq 5}$$

$$Q_{cv2} = \dot{m}_{ref} (h_2 - h_1) + \Delta\dot{m}_{ref} (h_2 - h_5^o) \qquad \text{Eq 6}$$

$$P_{2-3} = (\dot{m}_{ref} + \Delta\dot{m}_{ref}) (h_3 - h_2) \qquad \text{Eq 7}$$

$$\dot{m}_{oi} C_p (T_{40} - T_{30}) = \dot{m}_{ref} (h_3 - h_4) \qquad \text{Eq 8}$$

$$\Delta\dot{m}_{ref} h_5^o = \dot{m}_{ref} h_3 \qquad \text{Eq 9}$$

$$Q_{cv1} = \dot{m}_{oi} C_p (T_{40} - T_{10}) \qquad \text{Eq 10}$$

These equations are incorporated into the computer model and are consistent with the assumptions previously mentioned.

COMPUTER MODEL

The computer model is a FORTRAN program (3,300 cards) that makes use of machine geometry and clearance information along with data at State 1 and on the injected oil to calculate performance. There are two steps in using the computer model. Step 1 is to find two unknown factors, η_p, polytropic compression efficiency, and F_{DP}, the discharge port leakage factor, as functions of volume ratio. The volume ratio is here defined as the total thread volume to the pocket volume at the start of discharge. The unknown factors are found by choosing their values, running the program, and checking to see if the resulting isentropic and volumetric efficiencies match those measured. Once a match has been found, the factors are assumed correct for all sizes of machines with the same volume ratio.

The second step is to make use of η_p and F_{DP} in the program to make performance predictions for other machines and new operating points. The following types of information are available from step 2.

(1) The compression history in terms of thermodynamic properties (p, T, h, v, and s)
(2) States 2, 3, and 4
(3) The relative contributions of oil leakage, refrigerant leakage and refrigerant preheat to volumetric inefficiency
(4) The internal powers and heat transfers
(5) The shaft power (P_s) required by the overall process
(6) The volumetric (η_{vol}), overall isentropic (η_{1-4s}), and compression (η_c) efficiencies

One unique aspect of this model is that the closed compression process is broken down into a series of pressure increment processes. The Bein model uses a main rotor angle increment approach; that approach requires that the pressure after an increment be chosen and then corrected until the gas volume available for the gas are matched. If that approach was adopted for this model, it would mean many calls of the refrigerant property subroutines which are time consuming.

Conditions for a Successful Run

The following conditions constitute a successful run for step 2.

(1) Internal Oil Mass Balance - The amount of oil leakage from a pocket equals the amount of oil trapped at suction closure.
(2) External Oil Mass Balance - The amount of oil injected equals the amount of oil rejected through the discharge port.
(3) Stabilized Pressure - The pressure history in the trailing, current, and leading threads are equivalent but out of phase by 360°/NM.

The following additional conditions must be satisfied for a successful step 1 run.

(5) Matched Volumetric Efficiency - The calculated volumetric efficiency is within 1% of the actual value.
(6) Matched Overall Compression Efficiency - The overall compression efficiency is within 1% of the actual value.

Computer Program Input

The computer program required a compressor operating point as input. The following variables (with sample values) define an operating point.

NR = Freon-12
P_1 = 219.1 kPa
T_1 = -5°C
N = 2980 rpm
$\dot{m}_{oi}$ = 1.47 kg/sec
T_{10} = 40°C

The computer program also requires six primary variables and three secondary variables to define the machine's geometry (8). The clearances and oil leakage factors for the nine oil leakage paths also must be given. The oil leakage factors and clearances are required by the Bein (5) oil leakage model. The following variables (with sample values) must be assumed (step 1) or known (step 2) before the computer model can run.

F_{DP} = 0.25
η_p = 0.8915

These sample values were determined for a compressor with a volume ratio of 3.5 and a main rotor diameter of 280 mm.

Block Diagram of Computational Algorithm

The computational method is quite involved and so it is best to step through the major building blocks of the program to gain an understanding of it. These building blocks are shown in Figure 7, a flow chart of the computer program.

Block 1 - Initialization block - The geometry of the machine is set including clearances, characterisitic lengths, and volume ratio. The operating condition is established as defined earlier. The oil flow factors are set. The oil properties and injection mass flow rate are set. Estimations are made for T_2, p_4, and V_{or}. The pocket volume is calculated as a function of main rotor rotation. The pressure history for a thread is set to a linear pressure history based upon the input values of p_1 and p_4. This initial pressure history is needed for oil leakage calculations. State 2 is determined from the estimated T_2 and suction pressure p_1.

Block 2 - The mass of trapped refrigerant at suction closure is calculated. This amount is a function of thread volume, the volume of oil present at suction closure and the specific volume at state 2. It is assumed that no refrigerant leaks from a closed pocket and so this mass is constant until redefined in this block during the next iteration.

Block 3 - The refrigerant properties at the start of the closed compression process are set to those at State 2.

Block 4 - The next compression state (n + 1) is found. Knowing p_{n+1} (the next pressure in the pocket after the pressure increment), knowing State n (the thermodynamic state before the pressure increment), and knowing η_p (the polytropic efficiency, it is possible to find u_{n+1}. The equation is

$$u_{n+1} = u_n + \frac{u_{(n+1)s} - u_n}{\eta_p} \qquad \text{Eq 11}$$

where $u_{(n+1)s} = f\,(p_{n+1}, s_n)$. With u_{n+1} and p_{n+1} determined, the other thermodynamic properties (T_{n+1}, v_{n+1}, and s_{n+1}) can be found.

Block 5 - The amount of main rotor rotation must be found. This is done by equating the volume of refrigerant found by knowing the trapped refrigerant mass and v_{n+1} to the volume available for the refrigerant. Mathematically,

$$m_{ref}\ v_{n+1} = V_t\,(\theta_n + \Delta\theta) - \left(V_{or_n} - \frac{\dot{V}_{or}}{(N\ 2\ \pi\ /60)}\right) \qquad \text{Eq 12}$$

The amount of rotation is found by finding the $\Delta\theta$ needed to satisfy this equation. Once $\Delta\theta$ is known, then the displacement work done by the refrigerant on the oil is

$$\Delta w_d = (p_p - p_1)\ \dot{V}_{or}\ \Delta\theta\ /(N\ \ 2\ \pi\ /60) \qquad \text{Eq 13}$$

(See Eq 2) The oil volume leakage rate, $\dot{V}_{or}$, used in Eq 13 is found using the Bein oil leakage model which requires the leading and trailing thread pressures.

Block 6 - If the net main rotor rotation is sufficient to uncover the discharge port, then the compression process is finished. (COMPRESSION PROCESS FINISHED - STATE 3 IS FOUND, see Figure 3). If this is not the case then the pressure increment loop continues.

Block 7 - (DISCHARGE PROCESS). Calculations are made for the oil leakage and consequent displacement work after the discharge port is open until the tooth leaves the thread.

Block 8 - Calculations are made for the net amount oil rejected and net amount of oil leaked.

Block 9 - The amount of oil leakage is compared with the amount of oil trapped at suction closure. If these amounts are within 1% of each other then execution continues (SUCCESSFUL RUN CONDITIONS 1 AND 2 ARE MET). If this is not the case, then the amount of oil leaked replaces the amount of oil assumed trapped at suction closure and a new pressure history found from the pressure increment loop replaces the old one. The program then begins at block 2.

Block 10 - The amount of power absorbed by the recirculating oil is found by summing the incremental displacement works and dividing by the time that it takes to sweep through a thread. The oil shear power is found from Eq 1. The power into the injection oil by Eq 3.

Block 11 - The refrigerant leakage rate, $\Delta\dot{m}_{ref}$, is found by the method outlined earlier.

Block 12 - Calculations are made for a new State 2. The power absorbed by recirculating oil equals Q_{cv2}. Equations 6 and 9 are solved for T_2 using h_3 found from the last increment in the closed compression process.

Block 13 - If the new T_2 is within 0.01°C of the old T_2 then the execution continues. Condition 3, stabilized pressure, is automatically achieved at this point with no special effort. (SUCCESSFUL RUN CONDITIONS

3 AND 4 ARE MET - STATE 2 FOUND). If this is not the case, the new T_2 replaces the old one and execution goes to block 2.

Block 14 - (STATE 4 FOUND). Equation 8 combined with p_3 and the assumptions $T_{40} = T_4$ and $T_{30} = T_{10}$ gives state 4. Performance calculations, including volumetric efficiency and overall compression efficiency are made.

Block 15 - User Decision (only for Step 1) (Not shown) If the volumetric efficiency and overall compression efficiency match the known values to within 1%, then successful run conditions 5 and 6 are met and it is assumed that the guessed values F_{DP} and η_p are correct. (SUCCESSFUL RUN CONDITIONS 5 AND 6 ARE MET AND F_{DP} AND η_p ARE FOUND).

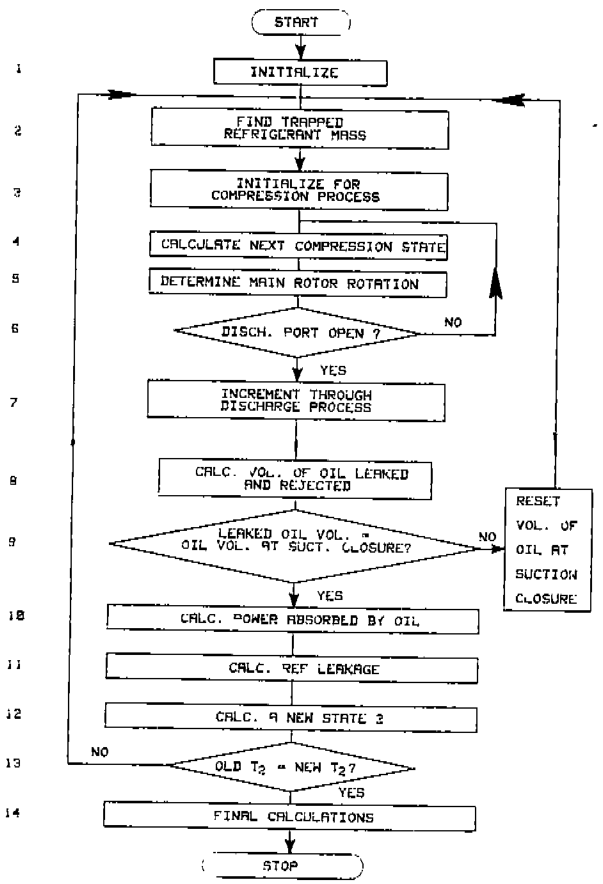

Figure 7 Flow Chart of Computational Algorithm

Step 1 - Determining F_{DP} (v_r) and η_p (V_r)

Step one is to take η_{1-4s} (V_r) and η_{vol} (V_r) for one machine, a machine being characterized by its main rotor diameter, and calculate F_{DP} (V_r) and η_p (V_r). The calculation scheme involves assuming F_{DP} (V_r) and η_p (V_r) and running the computer program until the resulting η_{vol} and η_{1-4s} are within 1% of the known values. This was done for a 280 mm diameter main rotor commercially available machine and the operating point given earlier. The results are then assumed valid for all machines of the same volume ratio at all other operating points. the results are shown in Figures 8 and 9.

Step 2 - Calculating Performance

The second step in using the computer model is to make performance calculations. The required input is the same as for step 1 except F_{DP} and η_p are not assumed but found from Figures 8 and 9.

To illustrate the above, the results presented below and in the rest of this section are for a 245 mm diameter commercially available machine, a volume ratio of 3.5, the operating point given earlier except $\dot{m}_{oi}$ = 1.08 kg/sec, η_p = 0.8915 and F_{DP} = 0.25.

p_2 = 219.1 kpa	P_s = 99.9 kw
T_2 = 8.6°C	P_{1-2} = 14.8 kw
p_3 = 919.1 kPa	P_{2-3} = 85.1 kw
T_3 = 64.9 kPa	P_{sh} = 8.1 kw
p_4 = 919.1 kPa	P_d = 4.3 kw
T_4 = 52.2°C	P_{oi} = 0.85 kw
Q_{cvl} = 26.0 kw	$\dot{m}_{ref}$ = 2.74 kg/sec
$\dot{m}_{ref}$ = 0.05 kg/sec	$\dot{m}_{or}$ = 7.0 kg/sec
η_c = 87.4%	η_{1-4s} = 71.5%

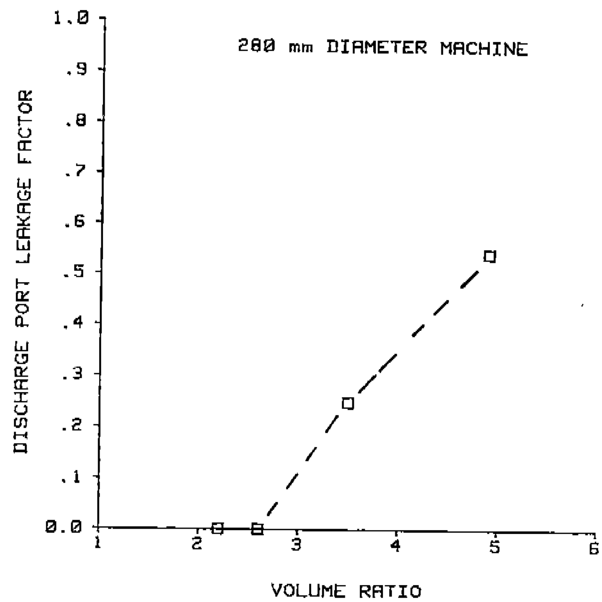

Figure 8 Discharge Port Refrigerant Leakage Factor

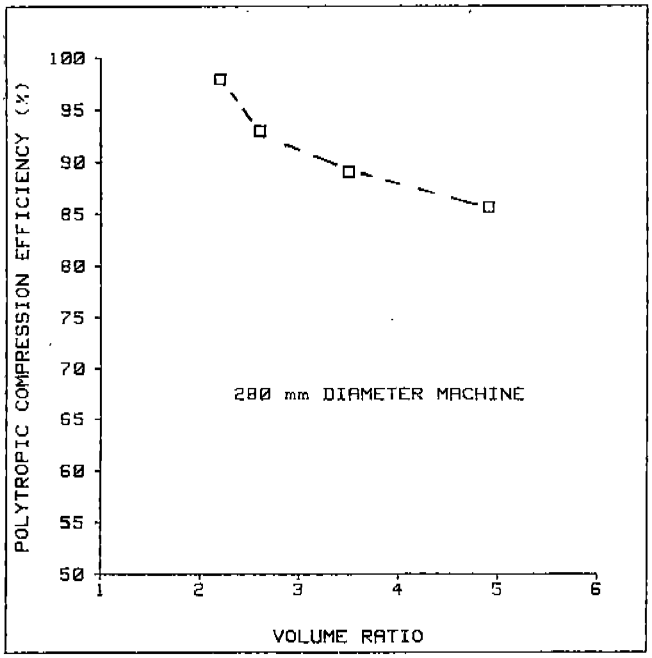

Figure 9 Polytropic Compression Efficiency

The relative contibutions made by the recirculating oil, preheat, and refrigerant leakage to the volumetric inefficiency (8.7%) can be determined. The largest contributor is preheat (3.75%). Recirculating oil contributes 3.3% and refrigerant leakage contributes 1.65%. Figure 10 shows the relative contribution of each oil leakage path to the volumetric inefficiency. This type of information is useful in design as it shows where the most significant improvements in volumetric efficiency can be won. In this case, cutting the tip clearance in half will halve the oil flow through this path thus increasing the volumetric efficiency by a minimum of 1.3%; but the power required for oil shear will be doubled (Eq 1). Cutting the clearance in half at the discharge port will halve the refrigerant leakage thus increasing the volumetric efficiency by a minimum of 0.8%. Reducing refrigerant leakage and oil leakage not only reduces their contributions to the volumetric inefficiency but reduces the preheat contribution because these leakages transmit energy to the incoming refrigerant.

Performance for Several Machines

Figure 11 is compressor efficiency curves published by Chan (7) with the calculated efficiencies η_{1-4s} and η_{vol} added. The calculations for η_{vol} and η_{1-4s} were made for the operating point given earlier except the oil mass rate, $\dot{m}_{oi}$, was allowed to vary with diameter. The values used were 1.08, 1.47, and 2.84 kg/sec which correspond to machine diameters of 245, 280, and 350 mm, respectively. The values of F_{DP} and η_p were found by using Figures 8 and 9. Figure 11 shows close agreement between the calculated and the measured efficiencies.

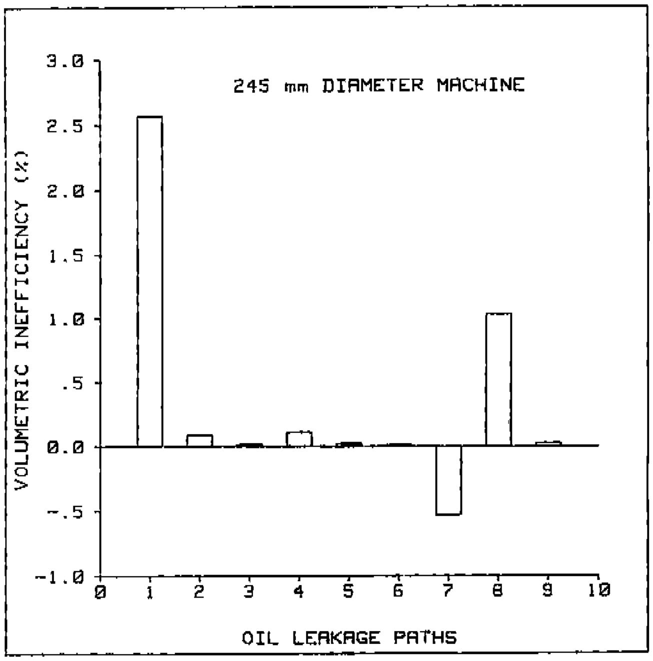

Figure 10 Breakdown by Oil Leakage Paths of the Oil Contribution to Volumetric Inefficiency

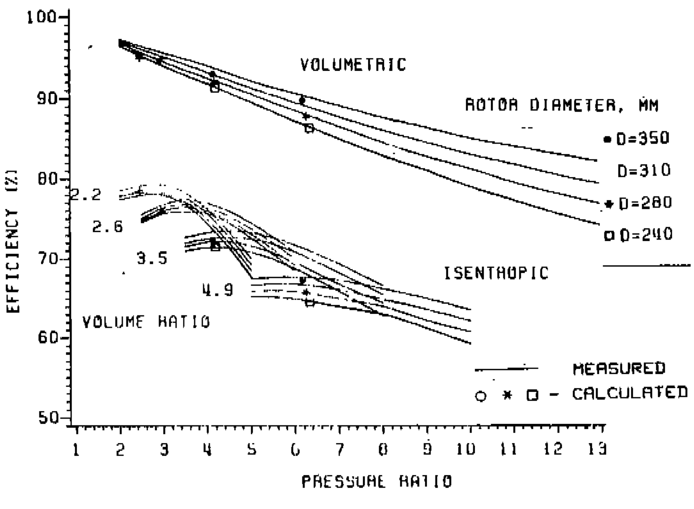

Figure 11 Calculated Compressor Efficiencies Compared with Measured Performance Curves

CONCLUSIONS

(1) It is possible to model the overall compressor process of an oil-flooded single-screw refrigerant compressor as a series of four component processes; the suction process, closed compression process, discharge process and separation process.

(2) The present calculations gave the following detailed results which are representative of the type of information obtainable.

a) The total shaft power, P_s (99.9 kw), is consumed by the refrigerant during the suction process (14.8 kw) and the compression process (85.1 kw) through the mechanisms of preheat and non-ideal compression, respectively. Oil and refrigerant leakage transmits some of the shaft power to preheat the incoming refrigerant. Oil absorbs power through shear (8.1 kw), displacement (4.3 kw), and external pumping (0.85). The refrigerant leakage also transmits a small portion of the shaft power (1.6 kw) upstream.

b) The volumetric inefficiency (8.7%) is split between oil present at suction closure (3.3%), preheat (3.75%), and refrigerant leakage (1.65%).

c) The oil injection mass flow is given (1.08 kg/sec). The refrigerant mass flow rate through the compressor (2.47 kg/sec), the refrigerant leakage mass flow rate (0.05 kg/sec), and the oil recirculating flow rate (7.0 kg/sec) are calculated.

d) The individual contributions of each oil leakage path that make up the total oil contribution to volumetric inefficiency (8.7%) can be found.

e) The polytropic compression efficiency (η_p = 89.15%), the closed compression efficiency (η_c = 87.4%), the overall compression efficiency (η_{1-4s} = 71.5%) and the volumetric efficiency (η_{vol} = 91.3%) can be found. The polytropic efficiency is found as a function of volume ratio only.

f) The factor F_{DP} (0.25) which scales the total available area for refrigerant leakage at the discharge port can be found as a function of volume ratio.

NOMENCLATURE

A	- cross-section area
A_{sh}	- shear area on main rotor
a	- gap distance between two plates
C_p	- specific heat of oil
D	- diameter of the main rotor
F_{DP}	- correction factor for discharge port
h	- correction factor for shear area
h_5	- stagnation enthalpy at state 5.
$\dot{m}$	- mass flow rate
N	- main rotor rotation rate (rpm)
NM	- number of main rotor threads
p	- pressure
P_{1-2}	- power absorbed between states 1 and 2
P_{2-3}	- power absorbed between states 2 and 3
P_d	- oil displacement power
P_s	- total shaft power
P_{oi}	- power into injected oil
P_{or}	- power into recirculating oil
P_{sh}	- oil shearing power
P_p	- pressure in a pocket
Q	- heat transfer rate
s	- entropy
T	- temperature
u	- internal energy
V	- velocity through a leakage path
$\dot{V}_{oi}$	- injected oil volume flow rate
V_{or}	- oil volume present at suction closure
$\dot{V}_{or}$	- oil volume leakage rate
V_r	- volume ratio
V_t	- volume of a thread
v	- specific volume
η_{1-4s}	- overall compression efficiency
η_c	- closed compression efficiency
η_p	- polytropic compression efficiency
η_{vol}	- volumetric efficiency
ρ	- oil density
μ	- absolute viscosity

Subscripts

1-5	- states 1-5 refrigerant properties
10	- injection oil
30	- discharge oil
40	- separator tank oil
CV (1-6)	- control volumes

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