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# MODELING AND SIMULATION OF ROTARY SCREW COMPRESSORS

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## ABSTRACT

The objectives of this work on modeling and simulation of rotary screw compressors were to develop a mathematical model and computer simulation having the necessary detail and sufficient accuracy required to investigate the performance sensitivity of screw compressor design parameters, to predict screw compressor performance, and to evaluate effects on efficiency of clearances, component design, and geometrical changes over a range of operating conditions.

Control volumes considered in this model included expansion through the compressor inlet, compression in the screw thread volume, and expansion through the compressor discharge. Expansion through internal leakage paths will be a future consideration. Homogeneity of temperature and pressure is assumed to exist in each control volume. All threads of the screw compressor are assumed identical and equally spaced. In steady state operation, each thread is assumed to respond in the same manner, and one characteristic thread can thus be used to represent the performance of the device.

The mathematical model consists of a set of interrelated differential equations coupled by mass flows between control volumes. Flows are modeled by compressible flow orifice equations with experimentally determined orifice coefficients. The mass flows are calculated products of orifice coefficients, areas dependent on the design geometry and mass flux. A polytropic process with experimentally determined exponents is used to model the compression process.

The characteristic thread technique offers several advantages. First, it reduces the number of differential equations to be integrated and, correspondingly, the amount of computer time required to obtain the simulated steady stage solution. Second, the model can be changed to represent any number of threads, all having a common suction and discharge plenum, each thread identically constructed and equally spaced relative to shaft angle.

The modular design of the simulation allows any of the subroutines to be updated, modified or eliminated without affecting the remainder of the computer program. This is most important when investigating the performance of alternate rotor profiles and capacity controls.

## INTRODUCTION

Using the high speed calculating capability of the digital computer, simulation of a screw compressor can become a powerful design and analysis tool. A good amount of work has been done previously on reciprocating and vane compressor simulation, especially in the area of refrigeration. Similar methods may be applied to any positive displacement compressor. This paper describes how previously established modeling methods have been applied to the screw compressor, also a positive displacement machine.

The program described is not intended to be a final version, but rather a foundation on which a more sophisticated model can be built. Additional sophistication will be gradually built in, proving out each phase as it is implemented. It should be noted that the model will be built only to the sophistication required to obtain desired results with sufficient accuracy.

The main computer program manipulates control between itself and several subroutines. This modular approach allows the program to be developed for several stages of sophistication. The subroutines can be independently changed or updated without modifying the rest of the program. This modular approach also requires setting up large blocks of "common" variables. This may seem to be a disadvantage, but actually forces the documentation of the program and its variables.

## THE MATHEMATICAL MODEL

The following processes have been considered in the model: (1) Expansion through the suction, (2) compression in the thread volume, (3) expansion through the discharge. Expansion through leakage paths and the effects of coolant injection

will be added to the model at a later date. The inner suction and discharge plenums are assumed to be at constant pressure. As experience is gained and the model becomes more sophisticated, the plenums can be added to model the pressure fluctuations existing at the inlet and discharge.

In the screw compressor, there are a multiple number of threads depending on the rotor lobe configuration. Each thread is identically constructed and equally spaced with respect to shaft angle. Pressure and temperature are assumed to be uniform throughout the thread volume. In steady state operation, each thread responds in the same manner. Thus, to simulate performance, only one thread is modeled. This is called the characteristic thread. Schwerzler used a similar approach in modeling a multiple cylinder refrigeration compressor (3). There are several advantages to the characteristic thread technique. First, it reduces the number of differential equations that must be integrated and the amount of computer time necessary for a solution. Second, the model can be easily changed to include any number of threads enabling the investigation of alternate rotor profiles.

The mass flows into and out of the characteristic thread have been modeled by a compressible flow orifice equation which assumes one dimensional, isentropic flow. This equation requires the use of experimentally determined orifice coefficients. The compression process has been modeled by a polytropic process. This approach is less complex than a first law approach because any heat transfer is accounted for in the polytropic exponent. One disadvantage is that the polytropic exponent must be experimentally determined. Also, varying compressor operating conditions will change the polytropic exponent. It has yet to be determined whether or not this approach will provide sufficient accuracy. Thermodynamic equations for the air-oil mixture may eventually be substituted.

## COMPUTER PROGRAM

A flow chart for the computer program is shown in Figure 1. All rectangles in the flow chart represent subroutines. The letters on the flow chart correspond with the letters in the margin below. Subroutines for leakage and coolant injection are enclosed by dotted rectangles, because they have not yet been added to the model. Initially, the input data is read in.

(A) The purpose of the volume subroutine is to furnish the characteristic thread volume as a function of rotation angle $\theta$. The volume curve has been represented by polynominals using a curve fit routine. This subroutine gives the thread volume for any shaft angle.

(B) The increase in volume reduces the pressure and temperature of the characteristic thread volume. Ideal gas behavior is assumed and the instantaneous temperature and pressure are obtained.

(C) As the axial and radial suction ports are uncovered, their area vs. rotation angle is required. These have been analytically represented by polynominals.

Since a pressure differential exists across these areas, mass will tend to flow into the thread volume. This flow is modeled by an orifice equation for ideal compressible flow, as previously described. The mass flow through the suction is a product of an orifice coefficient, a geometric area, and a mass flux.

(D) Next, the total instantaneous thread mass is calculated from suction port mass flows. The effects of leakages and coolant injection have not been considered at this time. They will be added at a later date.

(E) By integrating and summing the suction port mass flow, the instantaneous mass is computed. The integration is done in a separate subroutine, using the Runge-Kutta technique with variable step size. Runge-Kutta procedures are relatively stable and self starting.

(F) The values of $\theta$, T, P and M are then stored. These are some of the values which will later be delayed to simulate the other thread volumes. The shaft rotation is then incremented and the process repeated until compression begins.

(G) After compression begins, thread volume is obtained from the same volume subroutine as is used for the filling process. To compute the temperature and pressure, a polytropic process is assumed. This process will be verified experimentally by measuring interlobe pressure vs. time and fitting a polytropic exponent to the data. Different portions of the compression process may require different polytropic exponents due to variations in heat transfer during compression.

(H) A compressible flow orifice equation is used to model backflow through the suction port. This flow is integrated to get total mass.

(I) As thread pressure builds, the leakage in or out of the thread must be computed. This will be done in the leakage subroutine. Also to be added at a later date are the effects of coolant injection. Since the compression is modeled by a polytropic process, the coolant injection will affect the thread temperature and pressure by a change in the polytropic exponent. The coolant will also block the clearances, thereby decreasing the leakage rate. The temperatures and pressures are stored before the shaft rotation angle is incremented.

(J) Next, the mass flow through the discharge port is computed. This is done in the same manner as the mass flow through the suction port. The leakage subroutine will compute leakage

flow rate. The mass flows are summed and integrated to get instantaneous total mass.

(K) During the discharge process, thread volume is calculated from the volume subroutine. Temperatures and pressures are calculated from perfect gas laws.

(L) At this time, the characteristic thread mass flow, temperature, and pressure are checked for stabilization by comparing the current iteration with the previous iteration.

(M) If the properties have not stabilized, the characteristic thread is reset to its initial position and the properties of the other threads are obtained by delaying the properties of the characteristic thread by the appropriate crank angle.

(N) If stabilization has occurred, the output data is computed and printed or plotted as required.

## RESULTS

The results obtained from this preliminary model are not yet representative of an actual compressor, but they are good results considering the limitations of the model. Sample results for a 127.5mm (rotor diameter) compressor are shown in Figure 2. It can be seen in the sample results that the volumetric efficiency is unrealistically high. This is due to the absence of leakages in the computer model. Results also show a discharge temperature of 600°F. In an actual oil injected screw compressor running at these conditions, a discharge temperature of about 170°F. would be expected. Temperature calculations in the model are based on perfect gas equations and the effects of oil injection have not yet been added. Hence, the excessive discharge temperature. It is interesting to note that flow losses due to the discharge process are higher than thoseof the suction process. This is evident in the pressure profile curve. This can be attributed to overpressuring in the thread before the discharge opens, which increases horsepower. Power is also lost in the pressure fluctuations occurring during the discharge process. Similar discharge pressure fluctuations have been observed on an actual machine. This is probably caused by the irregular shape and small area of the discharge port during the last few degrees of rotation. It is expected that the addition of leakages to the model will attenuate these pressure spikes. Additional work is required in this area. The mass profile shows the accumulation of mass in the thread. The mass remains constant as it undergoes compression. When leakages are incorporated, the horizontal portion of the curve would assume an increasingly negative slope. The mass falls off to zero during the discharge process.

## FUTURE WORK

Although this screw compressor model is in its initial stage, it shows potential for becoming a valuable design tool. The simulation will be highly dependent on data obtained from testing an actual machine. Some of the data which will be necessary are:

1) Orifice coefficient values for suction and discharge ports.

2) Values for the polytropic exponent for compression. This will need to be done for various inlet temperatures and oil flows because the polytropic exponent is dependent on the amount of heat transfer occurring.

3) The effects of clearances (radial, contact line, blowhole, discharge, inlet) on performance needs to be experimentally determined to build and substantiate a leakage subroutine.

4) The effects of oil injection on leakages, heat transfer, and air end torque will be necessary to build and substantiate an oil injection subroutine.

In order to obtain this test data, a special test machine is being built. Basic components include a Joy 127.5mm screw compressor driven by a 100 hp variable speed D.C. motor. The air end is coupled to the motor by a torque transducer capable of monitoring instantaneous torque. In addition to the usual systems for oil separation and filtration, viscosity and density cells have been added for monitoring the oil. Special care is being taken with this machine to assure accurate results, which is essential to the accuracy and usefulness of the computer model.

## CONCLUSIONS

Computer modeling of the screw compressor makes possible the accumulation of data to aid in the design process. The simulation process is especially useful in evaluating machines for which experience is limited; such as expanders, refrigeration compressors, process gas compressors, or alternate rotor profiles (just to name a few).

Of immediate concern is the determination of what effect the various clearances have on performance. It may be possible to relax some of the tolerances without adversely affecting performance. In addition, the pressure instabilities during discharge bear looking into. Changes in the port design may minimize these instabilities resulting in a more efficient machine.

By studying these and other questions with a computer model, much could be learned about a machine which formerly could be answered only by prototype testing.

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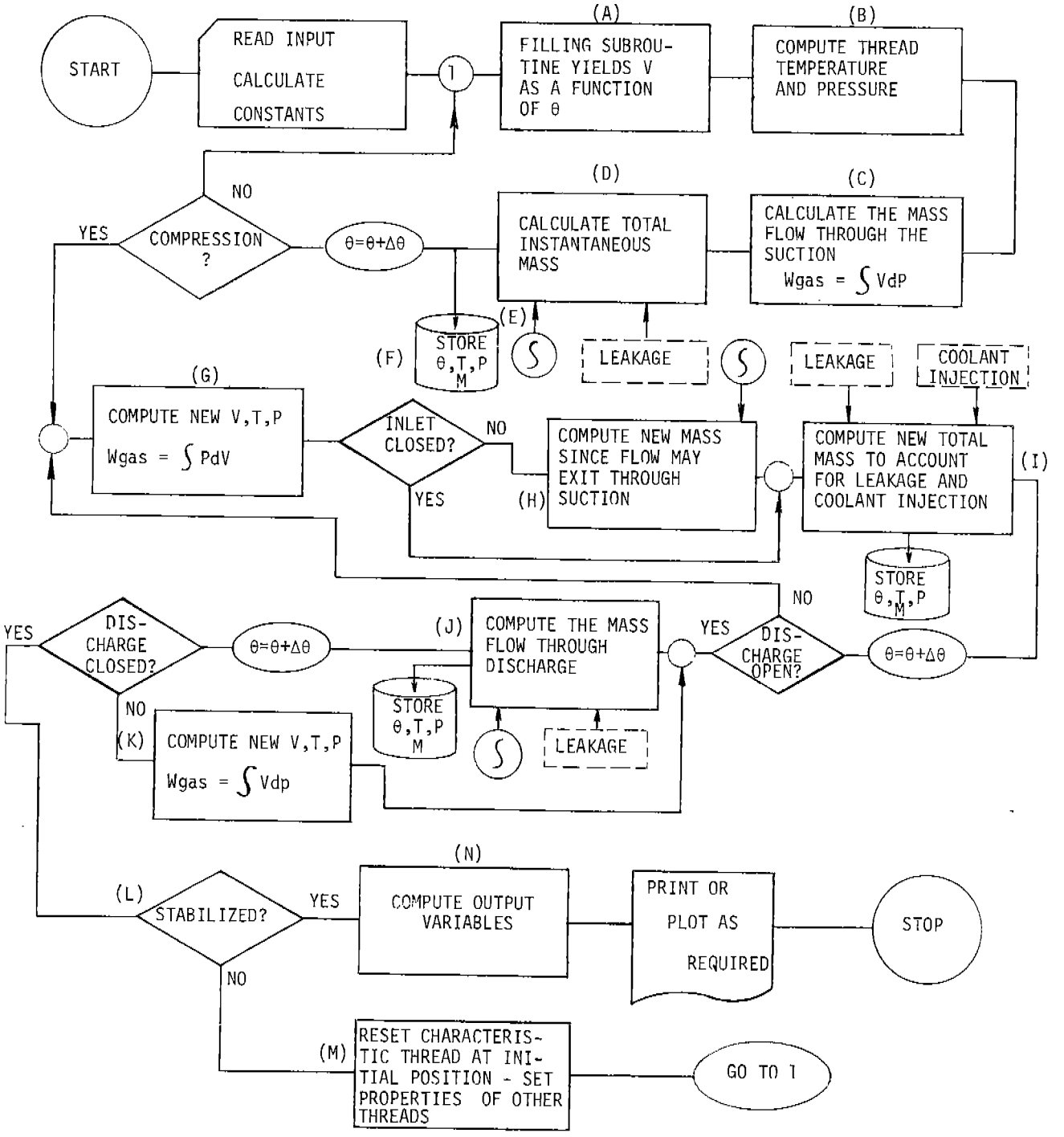
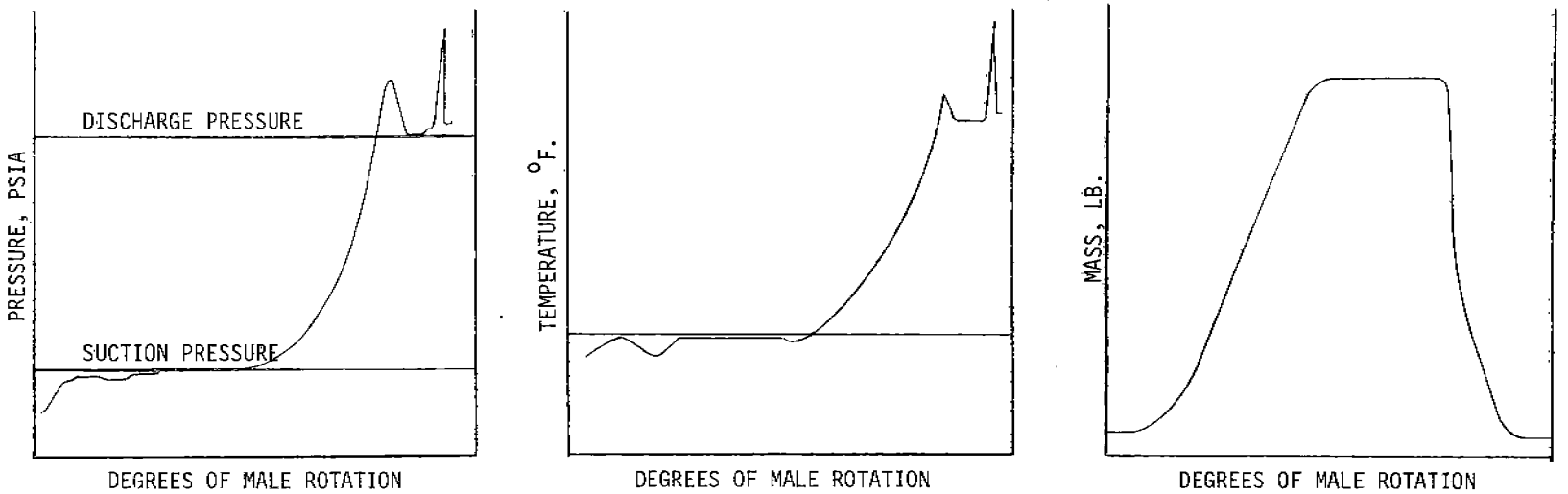
FIGURE 1

COMPUTER SIMULATION FLOW DIAGRAM






INPUT DATA

| | |
|---|---|
| Rotor Diameter | - 127.5mm |
| L/D Ratio | - 1.65 |
| Built-in Pressure Ratio | - 8.00 |
| Inlet Configuration | - Radial only |
| Discharge Configuration | - Both radial and axial |
| Ratio of Specific Heats | - 1.40 |
| Gas Constant | - 1545 ft-lb/(lb.-mole.-°R) |
| Critical Pressure Ratio | - 0.52828 |
| Polytropic Exponent | - 1.50 for compression |
| Suction Pressure | - -0.10 psig |
| Suction Temperature | - 70°F. |
| Discharge Pressure | - 100 psig |
| Barometric Pressure | - 29.910 in. hg. |
| Male Rotor Speed | - 5000 rpm |
| Inlet Flow Coefficient | - .10 |
| Discharge Flow Coefficient | - .90 |

OUTPUT DATA

Total delivered mass = 21.78468 lb/min
Percent mass entering radial inlet = 100
Percent mass entering axial inlet = 0.0
Percent mass exiting radial discharge = 17.375
Percent mass exiting axial discharge = 82.625
Total volume flow at inlet conditions = 292.68 cfm
Indicated power (gas only) = 57.338 hp
Suction process power loss = .756 hp
Discharge process power loss = 1.551 hp
Specific power = 19.591 hp/100 cfm
Volumetric efficiency = 99.512%
Adiabatic efficiency = 91.227%
Isothermal efficiency = 66.976%

FIGURE 2 - COMPUTER SIMULATION SAMPLE OUTPUT