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COMPONENT MODELING REQUIREMENTS FOR REFRIGERATION SYSTEM SIMULATION

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INTRODUCTION

The determination of the operating or balance point of a refrigeration system is a difficult problem to solve for off-design conditions. It can be done graphically if performance data for each component is available [11]* but to assemble performance data for all possible superheats, air flows, compressor speeds, etc. by testing is far too complex a task. Besides, the number of graphs required to completely model a system would be excessive. What is needed is a set of analytical models for predicting component behavior over a wide range of conditions along with a technique for finding the system operating point.

While many analytical models of compressors and heat exchangers have been developed and programmed for computer solution, most are research and development oriented [4],[5],[9]. For design of a single component, these programs are extremely useful. Once a component is designed, however, the problem is different. In most system design problems the desire is to predict system capacity, refrigerant flow rate, annual average EER, etc. for a given set of system components with different air flows, air temperatures, compressor speeds, and so forth. Questions such as how condenser fins/inch or hermetic compressor line voltage influence system capacity are the type asked. For these questions, information on valve motion, compressor bearing loads, or refrigerant pressure profiles in the evaporator are often superfluous. Also, since computer programs for system analysis may be run many times by the system designer, the computing time rapidly becomes excessive if research and development type models are used. What is required is a set of analytical models which achieve reasonable accuracy with minimum complexity. Such models can be created by combining basic principles with well chosen empirical parameters correlated with a few tests on the component. This paper discusses several such computer models, how they are correlated, and how they may be combined to create a system model which consumes a reasonable amount of computing time.

* Numbers in [ ] are references at the end of this paper.

A SIMPLE HERMETIC COMPRESSOR MODEL

For a hermetic unit of fixed design, the performance is governed by the following parameters.

1. Refrigerant type
2. Inlet temperature
3. Inlet pressure
4. Outlet pressure
5. Line voltage
6. Line frequency
7. Surrounding ambient air temperature
8. Surrounding ambient air flow over the unit

and the resulting parameters of interest to the system designer are

1. Outlet temperature
2. Refrigerant flow rate
3. Motor killowatts
4. Motor current
5. Heat loss from the casing

It is possible, of course, to create a series of graphs of the five "output" parameters as functions of the eight "input" parameters by testing alone. It is also possible to construct a computer model of the unit which calculates the output by a detailed analysis of all the processes taking place inside the unit.

The first method is obviously uneconomical. The second method, if successful, would require large computing times. In addition, the second method would require rather involved testing for correlation especially if temperature profiles, valve motion, bearing loads, etc. were to be predicted accurately as a necessary condition for overall performance predictability. What is needed is a simpler set of relations between input and output which can be correlated with a minimum number of tests using basic instrumentation. Such a model is discussed below.

Figure (1) shows a diagram illustrating the hermetic compressor model along with a list of the terms shown. To illustrate the model we begin with a general outline of the processes after which the details of the model will be clarified.

Refrigerant enters at T_i, P_i and a flow rate $\dot{m}$. Through the inlet fitting it experiences a pressure drop

$$P_i - P_{sh} = K_i \left[\frac{\dot{m}}{A_i}\right]^2 v_i \tag{1}$$

While passing over the motor, the refrigerant (now called the shell gas)

1. Picks up heat $\dot{Q}_m$ from the motor
2. Picks up heat $\dot{Q}_o$ from the oil
3. Picks up heat $\dot{Q}_c$ from the compressor
4. Rejects heat $\dot{Q}_{ct}$ to the casing top by convection

$$\dot{Q}_{ct} = h_{sh} A_{ct} \left[T_{sh} - T_{ct}\right] \tag{2}$$

The hot shell gas then passes through the suction line (including suction muffler) in an isenthalpic process, $h_{sh} = h_s$ and suffers a pressure loss

$$P_{sh} - P_s = K_s \left\{\frac{\dot{m}}{A_s}\right\}^2 v_{sh} \tag{3}$$

The shell gas temperature must be such that the energy balance on the shell gas is satisfied

$$\dot{m} h_i + \dot{Q}_m + \dot{Q}_c + \dot{Q}_o = \dot{m} h_{sh} + \dot{Q}_{ct} \tag{4}$$

and the heat flows through the casing top are equal

$$\dot{Q}_{ct} = h_{sh} A_{ct} \left[T_{sh} - T_{ct}\right] = h_\infty A_{ct} \left[T_{ct} - T_\infty\right] \tag{5}$$

Satisfaction of equations (4) and (5) is an iterative procedure since $\dot{Q}_m$, $\dot{Q}_c$, $\dot{Q}_o$ and $\dot{m}$ are not initially known. Also, h_∞ may be dependent on T_{ct} (natural convection) if ambient air flow over the unit is low.

Equation (3) along with $h_{sh} = h_s$ then defines the suction gas state.

The compressor then compresses the gas to the discharge pressure P_d in such a manner that

$$\dot{m} \left[h_d - h_s\right] = \eta_m \, SHP \tag{6}$$

and also

$$T_d = \bar{R}^{\frac{\gamma - 1}{\gamma}} \tag{7}$$

where γ is the real gas ratio of specific heats at suction conditions and $\bar{R}$ is a pseudo-pressure ratio which is a function of γ and P_d/P_s. The compressor heat loss is then

$$\dot{Q}_c = SHP\left[1 - \eta_m\right] \tag{8}$$

The flow rate must also satisfy

$$\dot{m} = \eta_v \left[DISP\right]\left[RPM\right] / v_s \tag{9}$$

An iterative procedure on the motor-compressor combination is required to satisfy these equations.

The gas then flows through the shock loop where its temperature drops by

$$T_d - T_o = \epsilon \left[T_d - T_{oil}\right] \tag{10}$$

The shock loop also imposes a pressure drop

$$P_d - P_o = K_o \left[\frac{\dot{m}}{A_o}\right]^2 v_m \tag{11}$$

where

$$v_m = \frac{v_d + v_o}{2} \tag{12}$$

and the gas losses heat to the oil

$$\dot{Q}_s = \dot{m}\left[h_d - h_o\right] \tag{13}$$

The oil also losses heat by convection to the casing bottom

$$\dot{Q}_{cb} = h_o A_{cb} \left[T_{oil} - T_{cb}\right] \tag{14}$$

and $\dot{Q}_{cb}$ must also satisfy

$$\dot{Q}_{cb} = h_\infty A_{cb} \left[T_{cb} - T_\infty\right] \tag{15}$$

The shock loop effectiveness is related to the flow rate, $\dot{m}$, and h_o depends on T_{oil} and $P_d - P_s$. The oil heat loss to the shell gas then follows from an energy balance on the oil

$$\dot{Q}_s = \dot{Q}_o + \dot{Q}_{cb} \tag{16}$$

Again, an iteration on these equations is required.

In order to solve these equations additional information is required. To obtain it, a series of steps both analytical and experimental are carried out as follows.

The geometric parameters A_{ct}, A_{cb}, A_i, A_s, A_o, RE, DISP, and L are known or easily measured. The loss factors K_i, K_s, and K_o may be estimated or measured by air flow tests.

The motor performance is taken from motor performance curves. This data gives $\dot{Q}_m$, η_e, KW, AMPS, RPM, SHP, vs τ at various line voltages E and line frequencies f. Two computer routines are written as follows

HMOT1 = Finds AMPS, KW, η_e, $\dot{Q}_m$, SHP and RPM given τ, E, f, or AMPS, KW, η_e, $\dot{Q}_m$, SHP, and τ given RPM, E, f.

HMOT2 = Finds τ, RPM, AMPS, η_e, $\dot{Q}_m$, and SHP given KW, E, f, or τ, RPM, KW, η_e, $\dot{Q}_m$, and SHP given AMPS, E, f.

There are several ways of doing this. One is to simply digitize motor performance data. But this restricts the program to those values of E and f at which the motor has been tested. A better way is to apply some empirical constants to basic motor theory such as outlined in texts [7]. For example, the speed-torque curve for an induction motor is modeled quite well by

$$\frac{\tau}{\tau_b} = \frac{2}{\frac{S_b}{S} + \frac{S}{S_b}}$$

where S = slip, τ_b = breakdown torque, S_b = slip at breakdown. Similarly, the breakdown torque is given by

$$\frac{\tau_b}{E^2} = \frac{c}{2 N_s}$$

where N_s is the synchronous speed and C is a constant for a given motor. From motor performance data, one can calculate C.

This type of analytical model which augments basic theory with a few well chosen empirical constants works quite well as Figure (2) demonstrates. The computing time is low and the accuracy is certainly within the spread between individual production motors.

Next one must have a system of computer programs for rapid evaluation of refrigerant properties. Such a system is described elsewhere in these proceedings*.

The remaining parameters to evaluate are h_{sh}, h_∞, h_o, η_m, η_v, ϵ, and $\bar{R}$. These may be found from a series of tests on the unit in which one measures P_i, T_i, T_s, T_d, T_o, P_o, T_{oil}, T_{ct}, T_{cb}, T_∞, AMPS, KW, $\dot{m}$, and air flow over the unit. All of these parameters except T_s, T_d, and T_{oil} are normally measured in standard calorimeter tests. Adding T_s, T_d, and T_{oil} thermocouples is a relatively simple process.

A computer program can be written to determine the above remaining parameters from the test data. The program HMOT2 is used here. For example, the total heat loss from the casing during test is

$$\dot{Q}_{ct}+\dot{Q}_{cb} = KW - \dot{m}\left[h_o - h_i\right] = h_\infty A_{ct}\left[T_{ct}-T_\infty\right] + h_\infty A_{cb}\left[T_{cb}-T_\infty\right]$$

from which one may find h_∞ and correlate it against the Reynolds number for the air flow over the unit.

η_m, η_v, and $\bar{R}$ correlate well as functions of γ and P_d/P_s. Figure (3) shows $\bar{R}$ vs the parameter

$$\frac{\gamma-1}{\gamma}\,\frac{P_d}{P_s}$$

The data is from several units of different rated capacity within the same series and demonstrates that one need not test every model of basically similar units which differ only in stroke or clearance volume.

There are many proposed correlations for volumetric efficiency which account for the individual effects of valve losses, heat transfer, re-expansion, and leakage. Since most of these factors require complex instrumentation to detect, we can only legitimately treat re-expansion effects. Figure (4) shows the correlation of

$$1-\eta_v-\frac{RE}{100}\left\{\left[\frac{P_d}{P_s}\right]^{1/\gamma}-1\right\} \quad vs \quad \frac{\gamma-1}{\gamma}\,\frac{P_d}{P_s}$$

again for several units within the same series.

Heat transfer between the shell gas and upper casing casing is correlated with

$$Nu = a\,R_e^{\,b}\,P_r^{\,1/3}$$

where a and b are constants and

$$Nu = \frac{h_{sh}D}{k} \qquad Re = \frac{\dot{m}}{\pi D \mu} \qquad Pr = \frac{\mu c_p}{k}$$

k, μ, and c_p are the shell gas conductivity, viscosity, and specific heat and D is the casing diameter. Figure (5) shows this correlation for two typical units. For this correlation, a and b depend on the compressor and motor since the motor winding size and shape influences the gas flow in the shell.

The oil temperature correlates quite well with the mean compressor temperature

$$T_m = \frac{1}{2}\left[T_s + T_d\right]$$

as shown in Figure (6), Again, the data is for several units within the same series.

The other parameters can be correlated by similar techniques but space does not permit discussion of the details. The important point is that by using dimensionless terms whenever possible, the compressor model can be extended to other refrigerants with reasonable confidence.

The structure of the resulting computer routine for analysing a hermetic compressor is outlined in Figure (7). Subroutine HERM1 is the main routine which calls upon the others to solve the problem. Details of the solution logic are shown in Figure (8).

The argument list for HERM1 is as follows

SUBROUTINE HERM1(IR,CM,CC,HERM)

where

IR = Code specifying type of refrigerant
CM = Array of empirical parameters for motor
CC = Array of empirical parameters for COMP1
HERM = Input-output array
HERM(1)-HERM(22) = input T_i, P_i, P_o, etc. plus empirical parameters for h_∞, T_{oil}, h_{sh}, h_o, ϵ, areas and loss factors, E, f.
HERM(23)-HERM(60) = Resulting T_o, flow rate, efficiencies, heat flows, internal temperatures, KW, AMPS, etc.

As shown in Figure (3), some of the correlations are not what one would call excellent. This results from the necessary simplification of the real device as well as the limitations on the accuracy of the supporting test data. The important factor, or course, is how well the model predicts the behavior. In this regard, the accuracy is within 5% on the average which is certainly good enough for systems design work.

This computer model allows the system designer to rapidly perform many functions which previously required hours of hand calculations. Standard rating curves for compressor specification sheets can be generated by computer. Figure (9) shows a sample created by varying the evaporating temperature from 10 to 60°F in 10° steps and the condensing temperature from 90 to 160°F in 10°

* Scott, T.C., and Davis, G.L., "Construction of a Library of Computer Routines for Refrigeration Systems Analysis"

steps, a total of 48 runs of subroutine HERM1. Computer CPU time for this job is less than one minute demonstrating the speed of the routine.

Since volumetric efficiency and other data is also determined, many special studies which previously required extra tests can be run on the computer. The system designer may thus respond rapidly to customer requests for performance data at different line voltages, air flows over the unit, etc.

This hermetic compressor model is admittedly an oversimplification of the real processes taking place inside the unit. The most drastic deviation is the failure to differentiate between heat loss from the compressor to the shell gas and heat loss from the compressor to the oil. The mechanical efficiency and the relation of T_{oil} to T_m are compensations for this which result in a model accurate enough for the purpose intended.

HEAT EXCHANGER MODELS

For an air cooled heat exchanger of fixed design, the performance is governed by

1. Refrigerant flow rate
2. Refrigerant inlet state
3. Air flow rate
4. Air inlet state

and the parameters of interest to the system designer are

1. Refrigerant outlet state
2. Air outlet state
3. Coil capacity
4. Air and refrigerant side pressure drops

Of equal interest are the effects of

1. Fin type (plain, louvered, etc.)
2. Fin spacing
3. Tube diameter
4. Fin bond
5. Tube spacing
6. Tube arrangement (in-line, staggered)
7. Number of rows high and deep

The system designer requires an analytical model which can quickly determine the effects of all these parameters. As with the compressor, a model based on fundamental principles augmented by carefully chosen empirical parameters yields low computing times and acceptable accuracy. Details of the solution scheme for air cooled condensers and evaporators have been presented elsewhere [3] so that only a brief discussion will be given here.

The refrigerant flow in an air cooled condenser is modeled by breaking the process down into three regions; de-superheating, condensing, and subcooling. The de-superheating and subcooling regions are quite easily modeled using standard heat transfer and pressure drop relations for single phase flow coupled with exact fluid properties. The air side heat transfer and pressure drop correlates with methods outlined by Kays and London [6].

In the condensing region the refrigerant side heat transfer and pressure drop is more complex. Many relationships for these may be found in the literature [1] for both local and overall average values. Each generally is restricted to certain refrigerants, tube orientations, ranges of flow rates, etc. The use of local values with numerical integration down the length of the condensing region is too time consuming and offers little increase in accuracy. The most efficient technique is to store a large number of expressions for overall average heat transfer coefficients and pressure drops and let the computer select the one applicable to the set of conditions existing in the condenser being studied. Improvement in the accuracy of these overall average relations can be made if the coefficients in them are refined through comparison with tests on actual condensers. This is often a necessity since many of the relations are based on research type test data with uniform heat fluxes or temperatures, situations which do not occur in actual heat exchangers.

The same principle applies to the modeling of air cooled evaporators. Expressions for overall heat transfer and pressure drop are augmented with empirical constants and stored in the computer.

SYSTEM SIMULATION

Figure (10) illustrates the iteration scheme for finding the operating or balance point of a system composed of a compressor, air cooled heat exchangers, a thermostatic expansion device, and a receiver. The solution scheme for the compressor is as previously discussed and the method used for the heat exchangers is as presented previously [3] with the exception that multiple row coils are analysed on a row-by-row basis. Row-by-row analysis is especially beneficial in the simulation of air cooled evaporators under partial "wet" coil conditions.

This computer model may be used by the system designer for a wide range of studies. For example, system component selection is generally made for one set of air temperatures and air flows to match a given capacity. After installation the system will be subjected to a range of air flows and temperatures depending on ambient conditions. System capacity, EER, etc. will thus change. Figure (11) shows one of the many types of studies of the effects of these changes which may be made.

The system designer may also use the computer model to select components based on maximizing annual average EER instead of maximizing EER at some standard rating point. By using local weather data in conjunction with the computer model, EERs can be determined over the entire cooling season and components selected for best annual performance.

Because of the dimensionless nature of the correlations used in the model, system performance with other refrigerants can be evaluated rapidly without the need for extensive tests. For example, the well publicized "Ozone Depletion Hypothesis" has been concerned with atmospheric effects of R-12 used in automotive systems. If a decision were made to switch to R-22, the computer model can be used to quickly evaluate some of the consequences.

Table (1) shows such a comparison for a typical automotive system (a different compressor model, that is, an automotive type, not hermetic, was used for this study).

The resulting system balance point with R-22 indicates higher compressor head pressures and greater capacity. This points out that a cost saving is possible if evaporator size is decreased to match R-22 capacity with R-12 capacity. Using the model, the system designer can select the evaporator design required. It should be pointed out, however, that any such change to a different refrigerant involves many other factors such as material changes to withstand higher pressures and effusion losses, re-design of system controls, and so forth.

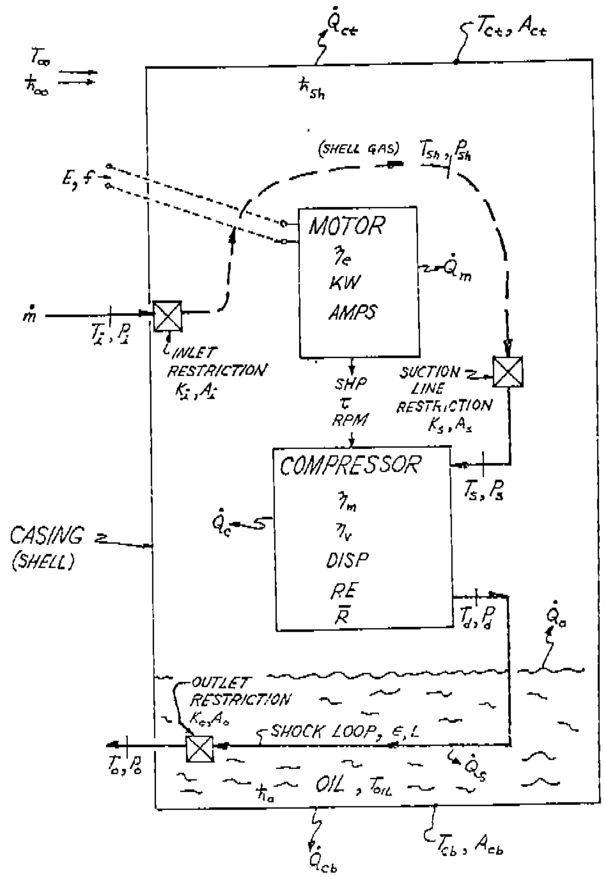
If new, untested refrigerants become available, all that is needed to run the system model with them is the thermophysical property information (equation of state, etc.). A reasonable estimate of system performance and/or required modifications can thus be made rapidly.

CONCLUSIONS

The availability of a system model gives the system designer expanded capability to rapidly study off-design conditions, select components based on annual average EER optimization, and quickly estimate the effects of different refrigerants. Because such a model must be run many times by the designer, it must achieve reasonable accuracy with low computing times. This is accomplished by constructing general component models based on fundamental principles augmented by carefully chosen empirical parameters whose values can be determined through relatively simple tests on the component.

REFERENCES

1. AHSRAE Handbook of Fundamentals, 1967 ed., Chap. 4
2. Chlumsky, V., Reciprocating and Rotary Compressors, E & F N Spon Ltd., London, 1965
3. Davis, G.L., Chianese, F., and Scott, T.C., "Computer Simulation of Automotive Air Conditioning - Components, System, and Vehicle" SAE paper 720077, 1972
4. Gatecliff, G.W., "A Digital Simulation of a Reciprocating Hermetic Compressor Including Comparison with Experiments", Ph.D. Thesis, University of Michigan, 1969
5. Hai, S.M., and Squarer, D., "Computer Simulation of Multicylinder Compressors", Purdue Compressor Technology Conference, 1974, Session TA2, p. 178
6. Kays, W.M., and London, A.L., Compact Heat Exchangers, McGraw-Hill, New York, 1964
7. Matsch, L.W., Electromagnetic and Electromechanical Machines, Int. Textbook Co., 1972
8. Scheel, L.F., Gas Machinery, Gulf Pub. Co., Houston, 1972
9. Schwerzler, D.D., "Mathematical Modeling of a Multiple Cylinder Refrigeration Compressor", Ph.D. Thesis, Purdue University, 1971
10. Smutny, F., "Study of Factors Influencing the Volumetric Efficiency of Reciprocating Compressors", Prog. in Refrig. Sci. and Tech., 11th Int. Conf., Vol. 1, 1963, p. 445
11. Stoecker, W.F., Refrigeration and Air Conditioning, McGraw-Hill, New York, 1958
12. York, R., "Compressor Design for the Process Industries", Ind. and Eng. Chem., Vol. 34, No. 5, 1942, p. 535



DEFINITION OF SYMBOLS

E = Line voltage
f = Line frequency
KW = Motor killowatts
AMPS = Motor current
SHP = Shaft horsepower
τ = Shaft torque
RPM = Shaft speed
$\dot{Q}_m$ = Heat loss from motor
η_m = Mechanical efficiency
η_v = Volumetric efficiency
DISP = Displacement/revolution
RE = % Re-expansion volume
$\bar{R}$ = Pseudo-pressure ratio
$\dot{Q}_c$ = Heat loss from compressor to shell gas
h_∞ = Convection coefficient to ambient air
h_{sh} = Convection coefficient, shell gas to casing top top
h_o = Convection coefficient, oil to casing bottom
A_{ct} = Surface area of casing top
A_{cb} = Surface area of casing botton
T_{ct} = Casing top metal temperature
T_{cb} = Casing bottom metal temperature
T_{oil} = Oil temperature
T_∞ = Surrounding ambient air temperature
$\dot{Q}_{ct}$ = Heat loss from casing top
$\dot{Q}_{cb}$ = Heat loss from casing bottom
$\dot{Q}_o$ = Heat flow from oil to shell gas
$\dot{Q}_s$ = Heat flow from shock loop to oil
L = Shock loop length
ϵ = Shock loop effectiveness $=[T_d - T_o] / [T_d - T_{oil}]$
$\dot{m}$ = Refrigerant flow rate

K_i, K_s, K_o = Pressure loss coefficients for inlet fitting, suction line, and shock loop.
A_i, A_s, A_o = Cross sectional areas of inlet fitting fitting, suction line, and shock loop
T_i, P_i = Inlet refrigerant state
T_{sh}, P_{sh} = Shell gas state
T_s, P_s = Suction gas state
T_d, P_d = Discharge gas state
T_o, P_o = Outlet gas state

Also, h = enthalpy, v = specific volume

FIG. 1: Hermetic Compressor Model and Related Nomenclature

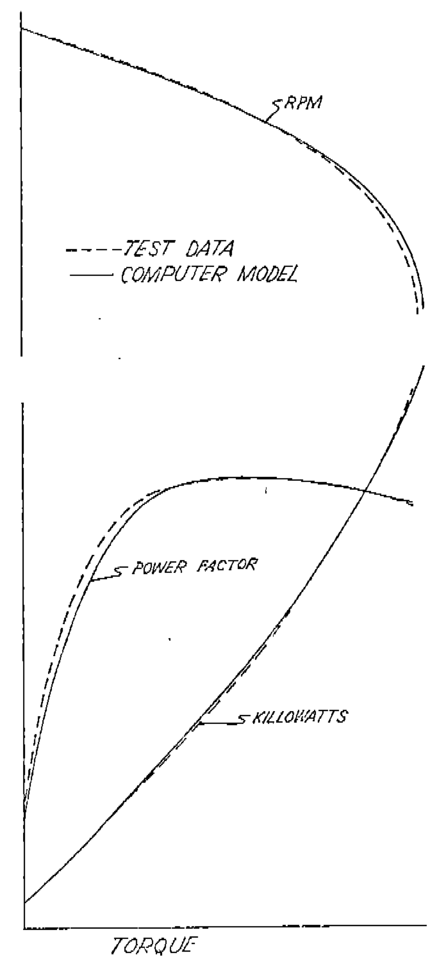


FIG. 2: Computer Prediction of Motor Performance

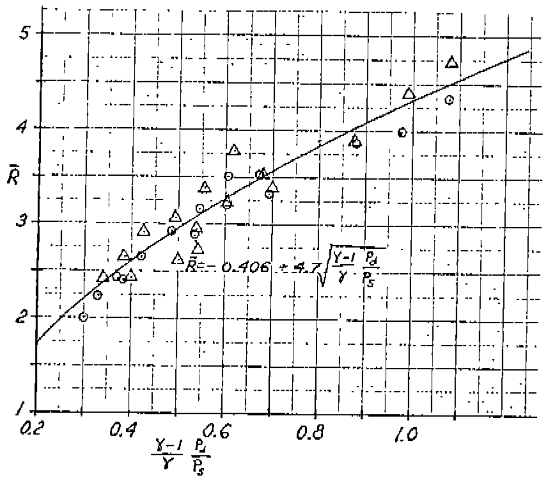


FIG. 3: Pseudo-Pressure Ratio Correlation

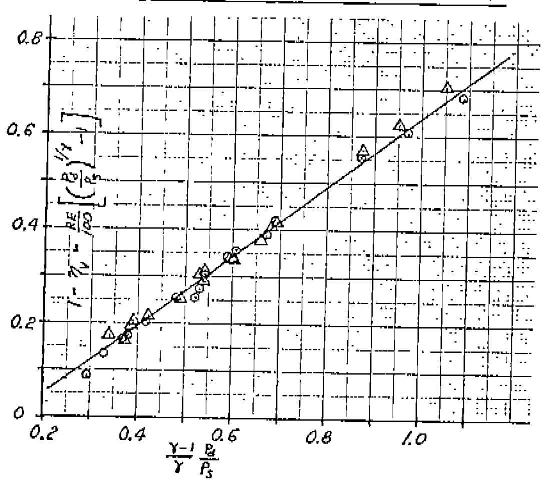


FIG. 4: Volumetric Efficiency Correlation

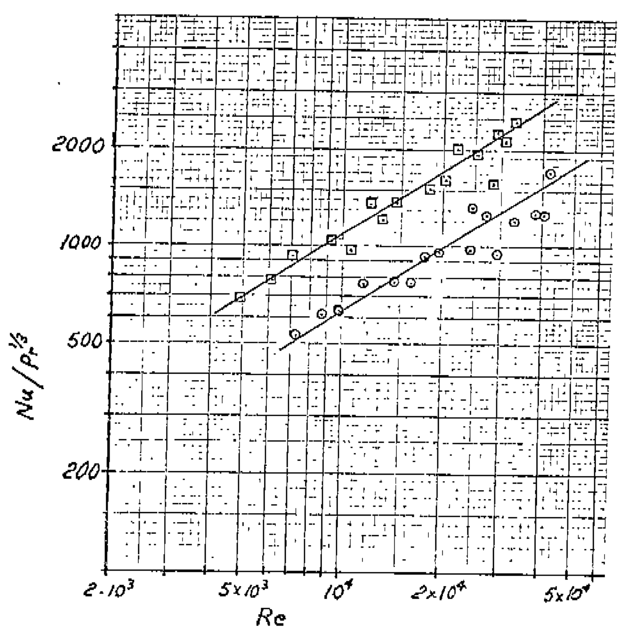


FIG. 5: Shell Gas Heat Transfer Correlation

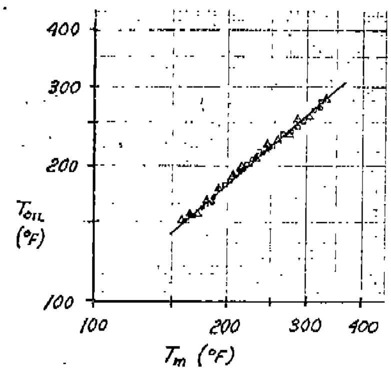


FIG. 6: Oil Temperature Correlation

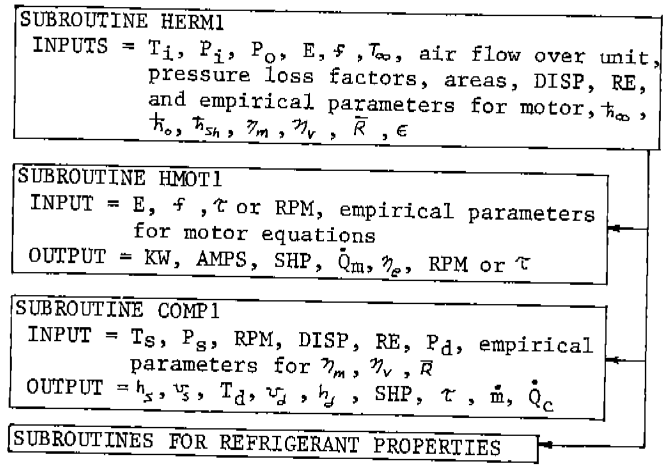


FIG. 7: Subroutine Structure for Hermetic Compressor Model

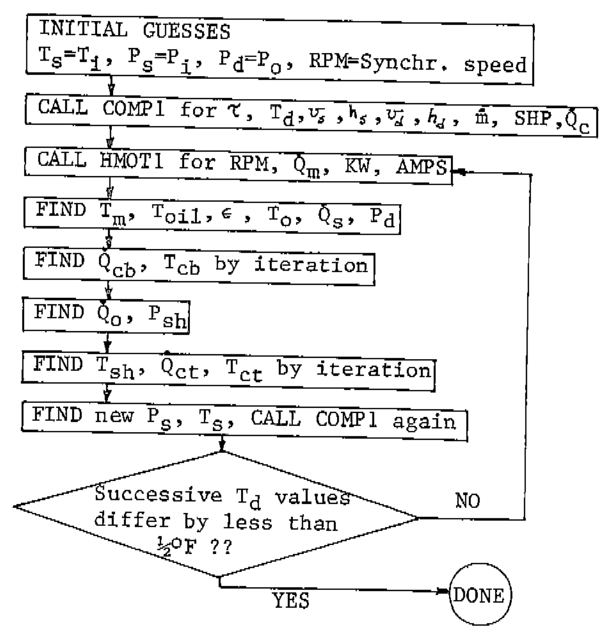


FIG. 8: Solution Logic for Hermetic Compressor

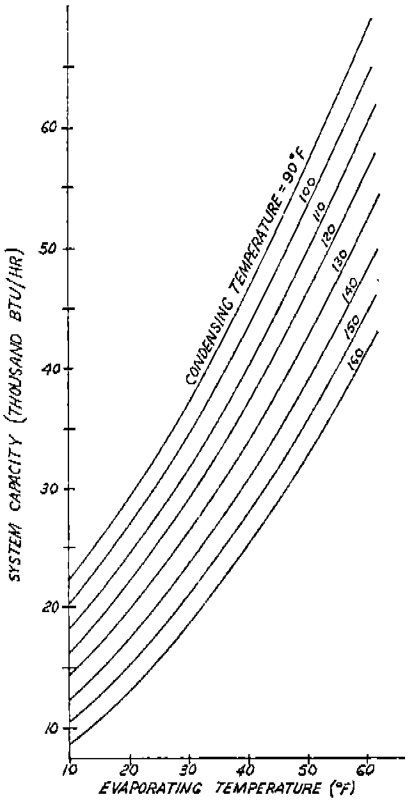


FIG. 9: Typical Capacity Curves from the Compressor Model

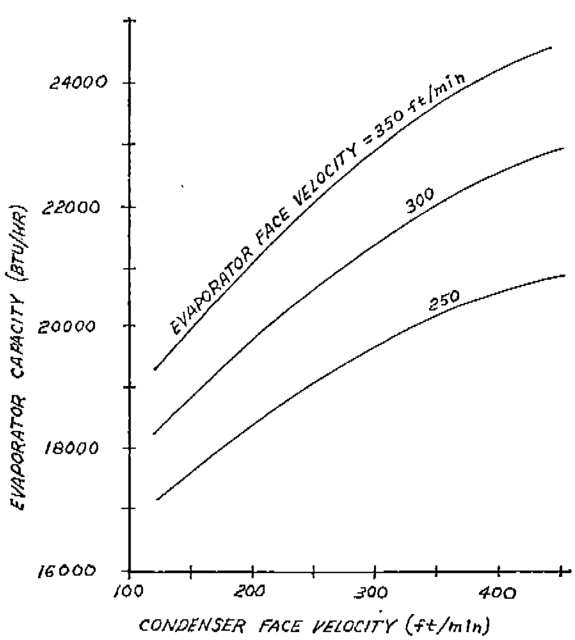


FIG. 11: Typical Study Possible with the System Model - Effect of Air Flows on System Capacity

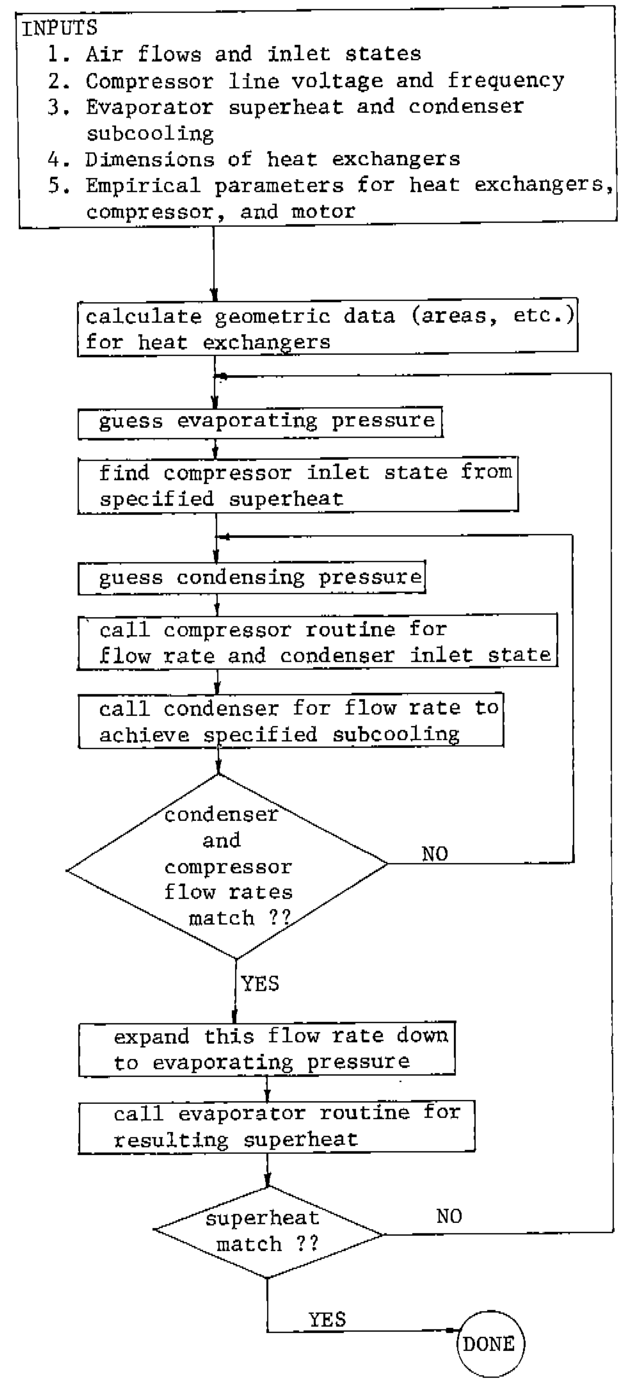


FIG. 10: Iteration Scheme for System Balance

TABLE 1: Effect of Refrigerant Change on Typical Automotive System Performance at Equal Air Flow, Same Components

	R-12	R-22
Evaporating Pressure (psia)	56.4	81.4
Condensing Pressure (psia)	260.	450.
Evaporator Capacity (Btu/hr)	18320	23680