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INTRODUCTION

Five different calorimeter configurations exist which provide results acceptable for rating small positive displacement compressors in accordance with ASHRAE practices [1]. When using the ASHRAE configurations and procedures in practice, from two to four hours are required to obtain compressor pumping capacity. From a manufacturer's viewpoint, an accelerated calorimeter test is necessary to permit early detection of production quality trends, satisfy customer demands for compressor with known characteristics and to permit the rapid evaluation of compressor design and assembly modifications.

A calorimeter and test procedure have been developed which enable compressors to be rated in less than twenty minutes. The essential physical features of the calorimeter are: 1) a recirculating refrigerant flow measurement loop consisting of the compressor, a heat exchanger, control valves and interconnecting piping, and 2) a minicomputer which monitors the loop and provides all control, logic and bookkeeping functions for the system.

This paper describes the general approach used and significant results from a project undertaken to develop a computer controlled calorimeter.

CALORIMETER TESTS

A compressor calorimeter is a device which measures the refrigerating effect of a compressor operating under a particular set of steady state conditions. The compressor capacity can be determined either by measuring the total heat transfer rate from the refrigerant in an evaporator or condenser, or by direct measurement of the compressor mass flow rate.

The variables which must be controlled in the calorimeter are:

1. Compressor discharge pressure.
2. Compressor suction pressure.
3. Compressor suction temperature.
4. Voltage at compressor terminals.
5. Ambient temperature surrounding compressor.
6. Air velocity or flow over the compressor.

The variables to be measured are:

1. Compressor discharge temperature.
2. Compressor speed.
3. Compressor current and power.
4. Compressor flow rate if not inferred from calorimetric measurements on an evaporator or condenser.
5. Heat input to the calorimeter.
6. Others as required for manufacturer's records.

Of the five ASHRAE calorimeter configurations mentioned previously, four of these use an energy balance around the evaporator to infer compressor flow rate. These tests are characterized by refrigerant phase changes in both the evaporator and condenser. In the fifth method, the gaseous refrigerant flow meter method, compressor flow rate is directly measured. The configuration of this method is shown schematically in Figure 1.

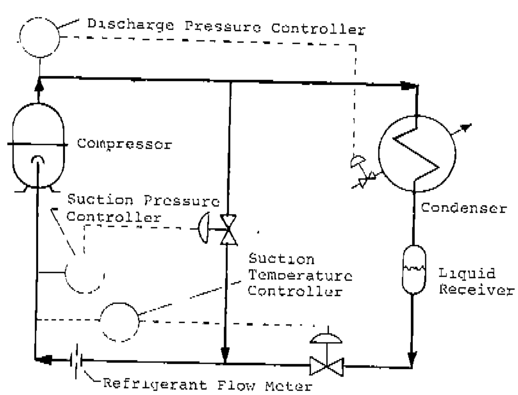

Figure 1. Schematic of Gaseous Refrigerant Flow Meter Calorimeter

The major drawback to these accepted methods is the time required to attain steady state operating conditions. A mathematical model of a hermetic compressor and a modified gaseous flow meter calorimeter were developed to study calorimeter transient operation. This model provides the basis of a digital computer based control system which accelerates the entire calorimeter test procedure, thereby attaining essentially steady-state compressor operation in about fifteen minutes.

COMPRESSOR AND CALORIMETER SYSTEM MODELS

A number of good reciprocating compressor mathematical models have been developed such as described in references [2-8]. Most of these models are oriented towards studying compressors at the microscopic level. Major attention is focused on valve dynamics and other parameters of interest to the compressor designer. In general, the models do not readily lend themselves to studies of transient compressor behavior. Thus, there was a need for a simplified compressor model that could be used for studying overall performance in a refrigeration system.

The work of Marriott [9] represents the first total system dynamic model which can be used to analyze and control a hermetic compressor and calorimeter system. Space precludes a complete discussion here of the mathematical model used. In general, however, the calorimeter and model are based upon the gaseous refrigerant flow meter calorimeter, modified as shown in Figure 2. The calorimeter consists of the compressor, heat exchanger, compressor discharge and suction pressure control valves (PCV-1 and PCV-2), a suction temperature control valve (TCV-1), turbine flow meter and interconnecting piping.

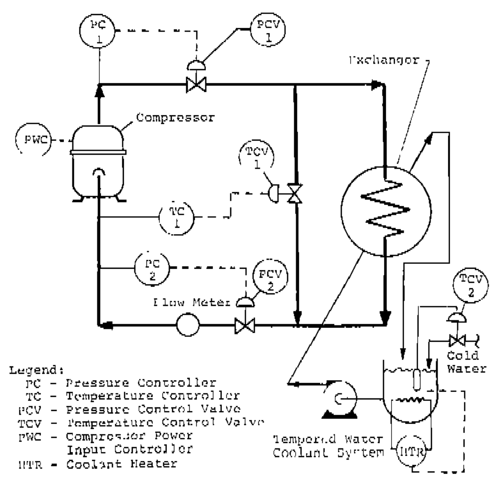

Figure 2. Schematic of Proposed Calorimeter System

The selected calorimeter configuration was based on several considerations. The more important were: 1) that system hardware should be minimal to reduce thermal inertia, and 2) that no refrigerant phase changes should occur within the cycle. This reduces thermodynamic lags and also minimizes the refrigerant charge required in the system.

The heat exchanger and coolant system are sized such that a nearly constant refrigerant temperature is maintained exiting the exchanger. About 25% of the refrigerant flow is bypassed through valve TCV-1 around the exchanger during steady state operation. The refrigerant pressure in the exchanger floats between the compressor discharge and suction pressures at a value dictated by both the total refrigerant charge in the system and instantaneous operating conditions.

The modeling method consisted of assigning lumped control volumes to the system components and expressing the governing differential and algebraic equations over a small time interval Δt. The compressor was represented by six control volumes and the exchanger by three. Applicaton of the Laws of Continuity and Energy yielded thirteen ordinary coupled differential equations. Augmenting the differential equations were nine algebraic equations which express the instantaneous relationships between the variables of the differential equations. Real refrigerant properties as well as actual compressor motor characteristics were used in the mathematical model. The model equations were solved on a digital computer using a Runge-Kutta integration technique to obtain the dynamic response of the calorimeter system. Calculated versus experimental results are presented in a later section. For a complete discussion of the mathematical model the reader is referred to Reference [9].

OPERATING PHILOSOPHY FOR THE PROPOSED CALORIMETER

The relatively large masses of calorimeter system components act as thermal inertias which slow system response and are the chief reason for the long times required to stabilize calorimeters at steady state conditions. While the proposed calorimeter system has a minimum number of components, it still suffers from the same malady. After extensive mathamatical modeling, the following operating steps were selected to give a practical and easily implemented approach towards overcoming the thermal inertias and accelerating the test.

The operation of the system is divided into three stages. During Stage I, full rated voltage is applied to the single phase compressor motor run winding to heat the compressor with "stalled rotor" power.

All control valves are in the open position during this stage. During Stage II, the compressor is started and its discharge pressure controlled by valve PCV-1. Since control valve TCV-1 is open during this stage, much of the hot refrigerant from the compressor will bypass the heat exchanger. This hot refrigerant helps to warm the compressor during this period.

After the compressor is heated to temperatures near its operating point as indicated by its shell temperature, Stage III is initiated. Compressor discharge refrigerant is diverted through the exchanger, temperature control valve TCV-1 and suction pressure control valve PCV-2. During this stage, the suction pressure and temperature come under control of their respective control loops.

EXPERIMENTAL EQUIPMENT

A schematic of the complete calorimeter test apparatus is shown in Figure 3. The compressor is an air conditioning model rated nominally at 18,500 BTU/HR and manufactured by Tecumseh Products Company. The refrigerant loop and coolant system are located in a constant temperature room which maintains the ambient test temperature for the compressor.

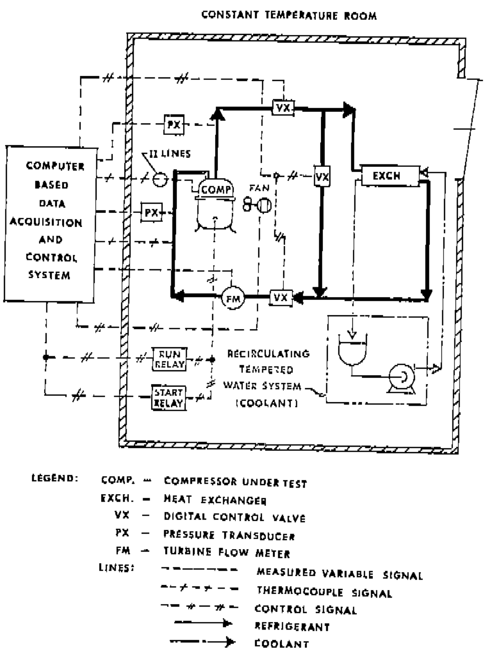

Figure 3. Schematic of Experimental Calorimeter System

A computer based data acquisition and control system is used to moniter all the system variables indicated in the CALORIMETER TEST section except the room and coolant temperatures which have separate controllers. The computer automatically starts the test, sequences through the three operational stages described above, determines when steady state is achieved and terminates the test.

The computer package is a Digital Equipment Corporation INDAC-8 data acquisition and control system. This package consists of a PDP-8/E computer with two high speed disk storage units, analog-to-digital converter, digital-to-analog converter, Teletype and INDAC software provided by DEC [10]. The INDAC software is a BASIC - like language written specifically for the scheduling of events as a function of time, sequence or external event. An executive operating system provides automatic overlays from disk to core of program units according to user-established priorities. The variables and functions of the calorimeter system monitored and/or controlled are indicated in Table 1. The program constructed for implementing the calorimeter system is shown in block diagram form in Figure 4.

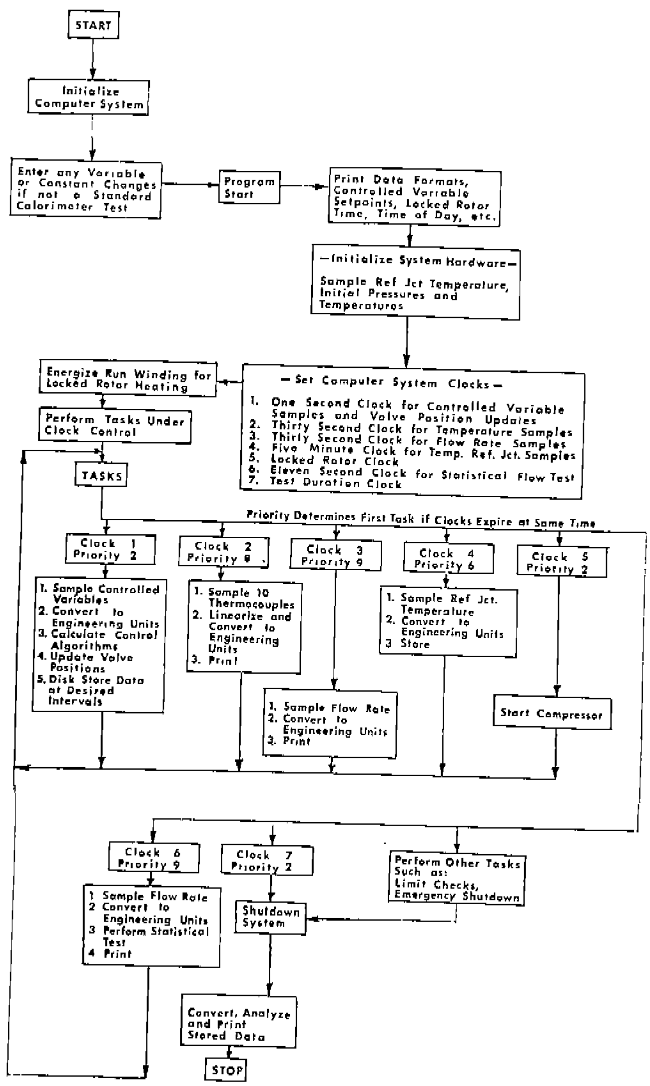

Figure 4. Block Diagram of Control Computer Functions and Tasks

TABLE I

Functions of the Computer Control System

DATA SAMPLING

1. Discharge Pressure
2. Suction Pressure
3. Suction Temperature
4. Nine System Temps.
5. Volume Flow Meter
6. Compressor Speed
7. Temperature Ref. Junction

STORAGE

1. Logical and real variables.
2. Constants such as controller gains, set point, switch temperatures, compressor data, equations of state for refrigerant, etc.
3. Temperature and pressure history of run.

CONTROL

1. Discharge Pressure
2. Suction Pressure
3. Suction Temperature

LOGIC

1. Apply locked rotor power.
2. Start compressor.
3. Shutdown compressor on high discharge pressure.
4. Shutdown system at test completion.

OUTPUT

1. Real time data.
2. Stored data at run completion.

CALCULATION

1. Conversion of sampled data to engineering units.
2. Updated control valve settings.
3. Mass flow conversion using equations of state.
4. Linearization and reference junction correction of thermo-couples.
5. Indices for storage and logic decisions.
6. Statistical test of flow data for steady state determination.

EXPERIMENTAL RESULTS

Comparisons between predicted and typical experimental results are shown in Figures 5, 6, 7 and 8. These figures show the transient behavior of the four more important variables monitored and/or controlled during the calorimeter test.

Figure 5 shows the temperature of the compressor shell bottom and is of interest because this temperature is used as the variable indicative of the status of the compressor during warmup. The experimental data are seen to behave quite similarly to the calculated temperature trends but lag from two to four minutes behind. The time lags are caused by the relatively slow heat conduction processes within the compressor.

The peak in the temperature data when stage III is initiated is the result of the change in heat transfer modes from the shell occurring at this time. During stages I and II, the fan which normally circulates air over the compressor in this particular test is off. Heat losses from the compressor occur by natural convection during this period. After the compressor shell is heated to a value near the pre-determined steady state temperature, the fan is started, forced convection begins, and the shell temperature drops. Eventual return to the steady state operating temperature occurs during stage III.

Compressor flow rate, shown in Figure 6, is important for two reasons. Its steady state value is the end product or result that is desired from the calorimeter test, and it has also been selected as an indicator of steady state. After the compressor starts, a high flow rate exists because the compressor operates initially under no load. The flow rate decreases during the first three or four minutes as the discharge pressure comes under control of its control loop. This is followed by a gradual increase in flow during stage II operation due to increasing suction refrigerant density. The flow rate decreases sharply as the suction pressure is reduced during stage III operation. Both the calculated and experimental results then show a very gradual asymptotic approach to the final steady state flow rate.

During the latter portion of the test compressor flow rate is sampled at eleven second intervals. A moving average of these samples is made to detect the trend of the flow rate versus time. A statistical t-test is performed by the computer to determine, within given confidence limits, when the mean flow rate ceases to change. For the calorimeter configuration and compressor used, this time occurs about 17-1/2 minutes after start-up. The steady state time as predicted by the mathematical model is

FIGURES 5, 6, 7 & 8. Typical Computer Controlled Calorimeter Performance

Data:

Model AJT - 15 Compressor
Initial Conditions:
Refrigerant - 130 psia, 95°F
System Temp. - 95°F
Ambient Temp. - 95°F
Operation:
Stage I - Stalled rotor power input, 25 seconds
Stage II - Hot gas bypass warmup
Stage III - System lineout to steady-state
Dimensionless Ratio= (Variable value - Initial value)/(Steady state value - Initial value)

Legend:
Calculated Values ————
Experimental Values o----o----o

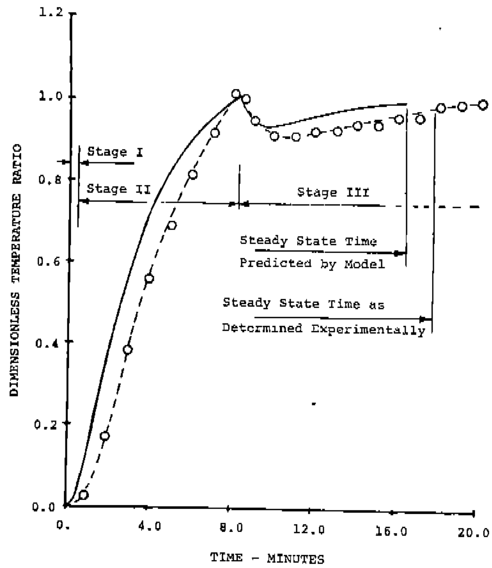

Figure 5. Compressor Shell Bottom Temperature vs Time

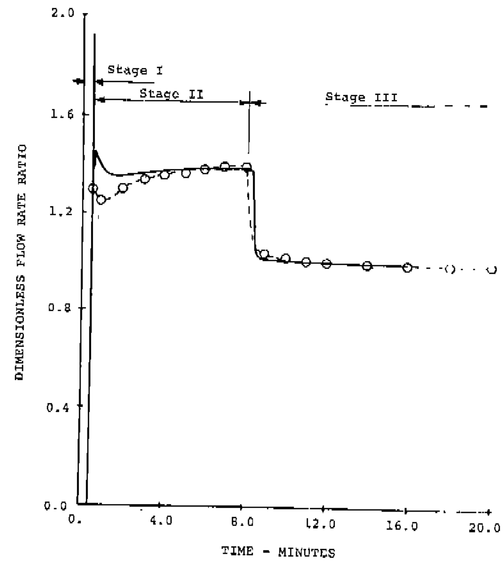

Figure 6. Compressor Flow Rate vs Time

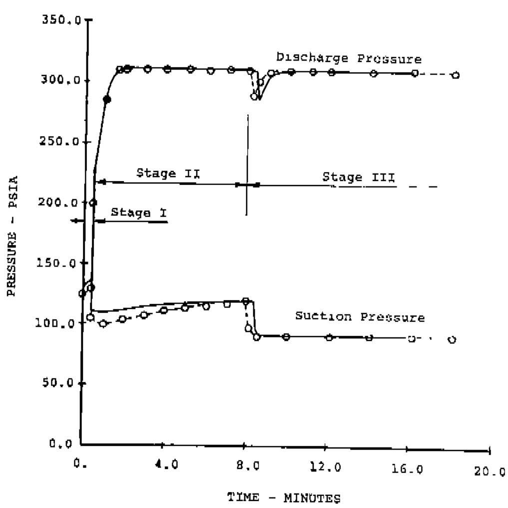

Figure 7. Compressor Discharge and Suction Pressures vs Time

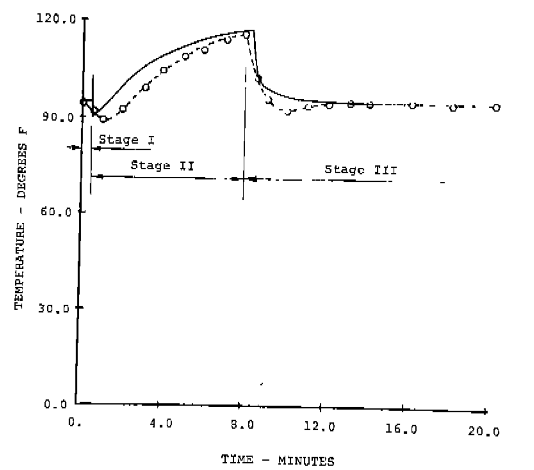

Figure 8. Compressor Suction Return Gas Temperature vs Time

about 13-1/2 minutes.

The controlled compressor variables of discharge and suction pressures are plotted in Figure 7.

During the stalled rotor stage, both the discharge and suction pressures increase slightly because heat generated in the motor is transferred to the refrigerant. Since all control valves are open at this time, the pressures in the system remain equal. After the compressor starts, the suction pressure drops as a flow becomes established in the test loop. The discharge pressure comes under control at its set point.

Once the discharge pressure reaches its control set point, the suction pressure magnitude is determined by the original refrigerant charge in the system and the refrigerant pressure drops across the valves, piping and exchanger in the loop. The suction pressure gradually increases due to the increasing average refrigerant temperature in the system.

When Stage III is initiated, the suction pressure control valve closes rapidly to bring the suction pressure to its control set point. This action upsets the discharge pressure but its controller rapidly returns the discharge pressure to the control set point.

The other controlled variable, suction return temperature, is shown in Figure 8. The exchanger bypass valve controlling this temperature is open until stage II is entered. After flow becomes established, the suction temperature decreases, as would be expected with the decreasing suction pressure. This is followed by a gradual temperature rise during stage II operation because much of the hot compressor discharge gas is being bypassed around the heat exchanger. Another sharp drop in suction temperature occurs as the end of stage II as the compressor suction pressure decreases to its control set point. Finally, the suction temperature is brought to its set point during stage III.

SUMMARY

A system for rapidly determing the steady state pumping capacities of hermetic refrigeration compressors has been presented. The system provides the same data as test stand calorimeters which measure the heat removal ability of compressors by thermodynamic means.

The essential physical features of the system are: 1) a recirculating refrigerant flow measurement loop consisting of the compressor, heat exchanger, control valves and interconnecting piping and 2) a mini-computer which monitors the loop and provides all control, logic and bookkeeping functions for the system.

The calorimeter system is completely automatic with the major compressor variables of discharge pressure, suction pressure and suction return temperature maintained at the proper set points by the controlling computer. Control logic is divided into three operational stages which function to drive the compressor quickly to the desired rating conditions. The compressor is first resistively heated by a brief application of line voltage to the motor run winding. Next, the compressor is operated with hot discharge gas returned to the suction port so that, in essence, the compressor helps heat itself to operating temperatures. Discharge pressure only is controlled during this stage. During the third stage, compressor suction pressure and temperature are controlled as the compressor is brought to the desired rating conditions.

A criterion was established to determine at what elapsed test time a compressor can be considered operating at its rated conditions. A statistical t-test is performed in real time by the controlling computer to determine, within given confidence limits, when the mean flow rate ceases to change. The supposition is made that conditions external to the compressor, such as ambient temperature, pressures, etc., are closely controlled.

CONCLUSIONS

1. It is concluded that reliable steady state compressor rating data can be obtained with the proposed system within approximately 17 minutes of startup. This represents nearly an order of magnitude decrease in the test times necessary in conventional calorimeter systems.

2. The simulation model indicated that the three major loops controlling compressor discharge pressure and suction enthalpy could be handled by analog controllers. However, the tasks of performing the logic decisions of the staging process, control of ambient conditions around the compressor, collection and conversion of data to engineering terms and flexibility in general, makes a computer based system imperative in order to meet the objective of a twenty minute calorimeter test.

BIBLIOGRAPHY

1. "Methods of Testing for Rating Positive Displacement Refrigeration Compressors" ASHRAE Standard 23-67, 1967.

2. M. Wambsganss, and R. Cohen, "Dynamics of Reciprocating Compressors with Automatic Reed Valves", Part 1: Theory and Simulation: Part II: Experiments and Evaluation, Proceedings of the XII International Congress of Refrigeration, Madrid, Spain, August 30 - September 6, 1967, Paper 3.06.

3. G. W. Gatecliff, "A Digital Simulation of a Reciprocating Hermetic Compressor Including Comparisons with Experiment", Ph.D. Dissertation, University of Michigan, 1969.

4. W. Brunner, "Simulation of a Reciprocating Compressor on an Electronic Analog Computer", presented at ASME Winter Annual Meeting, 1949.

5. J. D. Haseltine, "Part-Load Performance of Reciprocating Compressors", M.S.E.E. Thesis Mechanical Engineering Dept., Purdue University, 1970.

6. B. D. Kotalik, "Computer Simulation of a Five Horsepower High Speed Reciprocating Compressor", M.S.M.E. Thesis, Mechanical Engineering Dept., Purdue University, 1969.

7. J. F. T. MacLaren and S. V. Kerr, "Valve Behavior in a Small Refrigerating Compressor Using a Digital Computer", Journal of Refrigeration, Volume 11, No. 6, June 1968.

8. H. J. Blankespoor, and S. Touber, "Computer Simulation of a One Cylinder Reciprocating Compressor Using a Hybrid Computer", Proceedings of the 1972 Purdue Compressor Technology Conference, July 25-27, 1972.

9. L. W. Marriott, "Control of a Refrigeration Compressor Calorimeter for Minimum Testing Time", Ph.D. Dissertation, University of Michigan, 1973.

10. "PDP-8/E and PDP-8/M Small Computer Handbook-1972", Digital Equipment Corporation.