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Detailed Modeling and Computer Simulation of Reciprocating Refrigeration Compressors

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INTRODUCTION

The compressor model presented here has been developed to simulate overall performance of reciprocating hermetic and non-hermetic refrigeration compressors with a high degree of accuracy, while requiring a minimum of inputs. Unlike other models developed in recent years(1),(2),(3), the present model was created for use in total systems studies. Prerequisites were that the model, while containing sufficient detail to permit accurate simulation of internal compressor modifications, be easily manipulatable, rapid, and inexpensive to run. The present model can be easily used to study the effects of many design changes, such as capacity control, or internal heat transfer modifications, on overall compressor and overall system performance. The above is achieved through the use of approximate representations of valve dynamics, manifold pressure pulsations, and manifold heat transfer, and hence the present model is not suited to study factors related specifically to valve dynamics. If dynamic valve motion is the topic of interest, more complex models must be used.(4),(5),(6)

Factors accounted for in the present model are as follows:

- Real gas properties, rather than ideal-gas.
- Approximate representations of valve dynamics and manifold pressure pulsations.
- Approximate manifold heat transfer.
- Motor cooling.
- Friction losses due to bearings, pistons, and rings.
- Motor efficiency and speed variation with load.
- Effect of oil circulation.
- Various capacity control schemes.

DESCRIPTION OF MODEL

Discussion of the compressor model can be divided into three major sections:

1. Cylinder processes, valve, and manifold modeling,
2. Motor cooling, friction, and suction-discharge heat transfer, and
3. Oil circulation effect on capacity.

Cylinder Processes, Valve, and Manifold Modeling

All positive displacement compressors can be modeled by a four-step cylinder process, as shown in Figure 1:

1. Intake and mixing with residual mass,
2. Compression,
3. Discharge, and
4. Re-expansion of residual mass.

The present model treats the compression and re-expansion processes as non-isentropic, through the use of an isentropic efficiency term. It is important to note that, since the above isentropic efficiency is concerned only with specific portions of the total compressor processes, values usually greater than 90 percent are to be expected. By comparison, the overall compressor isentropic efficiency is typically 60 percent or less. Specific losses, such as motor cooling, and others to be mentioned later, contribute the major portions of the overall low efficiency.

Correctly modeled, the intake and discharge processes should include the effects of valve dynamics and manifold pressure pulsations. Dynamic valve simulation, however, would considerably complicate the model, and computational time would be prohibitive in overall performance simulations. A two-part representation is used for valve dynamics and manifold/cylinder pressure interactions in the present model. First, the cylinder pressure behavior seen in Figure 2 is modeled as a constant pressure overshoot ΔP_D or undershoot ΔP_S above or below average discharge and suction pressures. Second, the closing delay of the discharge valve, θ_D, is modeled as an increase in the effective clearance volume, and the closing delay of the suction valve, θ_S, is modeled as a decrease in the effective displacement volume.

Actual effective values of ΔP_D, ΔP_S, θ_D, and θ_S will vary not only with compressor speed, size, valve design, and manifold design, but also with pressure ratio, refrigerant, and flow rate. It has been found, however, that satisfactory results can be obtained, except near extreme operating limits as discussed later, by using constant values for a given compressor. The normal ranges to be expected for ΔP_D, ΔP_S, θ_D, θ_S, and other parameters have been established from experimental measurements by a variety of investigators(1),(2),(7), and are summarized in Table 1. When data on a particular compressor of interest, or on one similar to the one of interest, is not available, values near the limits of Table 1 should be used to obtain the most conservative results.

Motor Cooling, Friction, and Suction-Discharge Heat Transfer

The effect of motor cooling, internal friction, and suction/discharge manifold heat transfer is to heat the suction gas, producing a decrease in refrigerant density entering the cylinders, and reducing the mass flow rate. The present model iterates to find the rise in suction gas temperature due to the above effects. Mechanical efficiency of reciprocating compressors, accounting for friction in bearings, pistons, and rings can be expected to range between 90-98 percent for medium and large compressors, and could be somewhat less in small, fractional horsepower compressors.(8) All simulations to date with the present model have used 96 percent mechanical efficiency. Motor efficiency and speed variation as a function of percent load have been included in the model, with typical efficiencies being 85-89 percent and typical speeds being 95-99 percent of synchronous speed. Most of the heat generated by motor inefficiency and friction in hermetic and semi-hermetic compressor is given to the suction gas. A small portion, however, usually less than 20 percent(10), is lost to the ambient by convection and radiation from the compressor shell.

The evaluation of heat transfer between suction and discharge manifolds is done in an approximate manner because there are a variety of manifold designs, and data on heat transfer coefficients inside of compressor passages is rare and not well correlated (11),(12). The present model for suction/discharge manifold heat transfer is for heat flow from hot to cold gas streams separated by a thin metal wall. The wall is modeled as a flat plate with negligible resistance to heat flow, and some simple assumptions are made concerning relative flow areas on suction and discharge sides of the manifold. The purpose of the model is not to simulate the flow passages inside the compressor exactly, since even when details of the flow passages are known, exact simulation of the heat transfer would still be difficult at best. Rather, the purpose of the heat transfer model is to allow the investigator to simulate a desired temperature rise in the suction gas at a given condition, and to study the variation of that temperature rise with flow conditions. Table 1 gives some approximate values for rise in suction gas temperature due to suction/discharge heat transfer, learned from literature on the subject.(13)

Oil Circulation Effect on Capacity

Cooper(14) has pointed out that circulation of lubricating oil with the refrigerant can reduce available compressor capacity by as much as 20 percent. The reduction of capacity results from some refrigerant remaining in solution with the oil as it leaves the evaporator. Solubility of oil-refrigerant mixtures has been discussed by Bambach(15) and Spauschus(16). The present compressor model is equipped to determine oil-refrigerant solubilities as a function of temperature and pressure, for refrigerants 12 and 22. Oil circulation rates for particular compressors have been obtained from the manufacturers, and are typically between 0 and 15 percent of the total oil-refrigerant mixture flow rate by weight.

VERIFICATION OF MODEL

Three different compressors have been studied for the purpose of verifying the present compressor model:

1. Carrier 06D-824, a relatively large, semi-hermetic refrigeration compressor of nominal 9-ton capacity.
2. Carrier 06D-537, a large, semi-hermetic refrigeration compressor of nominal 14-ton capacity. The 537 is a larger version of the 824 compressor above, having the same bore, but a longer stroke.
3. A relatively small, nominal 3-ton, fully hermetic refrigeration compressor. (Manufacturer wishes to remain unidentified.)

All necessary data for the above compressors has been supplied by the manufacturers. Comparisons of actual and predicted performance for the above compressors are given in Figures 3, 4, and 5, respectively.

It can be seen that the simulations of the 06D-824 compressor are highly accurate. The worst error for predicting power consumption is about 8 percent, occurring at the extreme limit of low suction temperature, and improves rapidly to within 5 percent over most of the operating range. Similarly, the worst error for predicting capacity is about 15 percent at the extreme limit of low suction temperature, and improves rapidly to within 6 percent over most of the operating range.

The accuracy of the 06D-537 simulations is not quite as good as the 06D-824 simulation. The worst error for predicting power consumption is about 21 percent, occurring at the extreme limit of high suction temperature and low condensing temperature, and improves rapidly to within 7 percent with either increasing condensing temperature or decreasing suction temperature. The worst error for predicting capacity is about 11 percent at the extreme limit of low suction temperature, and improves rapidly to within 5 percent over most of the remaining operating range.

It is worthwhile to study why there is a difference in accuracy between the 06D-824 and 537 simulations. Both models are of similar design, differing primarily in the length of the stroke. The head plate, valve, and manifold design is very similar, if not identical, in both compressors, because they are of the same model series. As noted, the region of greatest inaccuracy for the 537 simulation is at low condensing temperatures and high suction temperatures, indicating a high refrigerant flow rate. A possible explanation is that the manifold and valve design are adequate for the 824 compressor under the above conditions, while they are not large enough for the 537 compressor, with its higher mass flow, causing a flow restriction which the present model does not account for. Moreover, as shown in the parametric studies to be discussed shortly, compressor power requirements are highly sensitive to increased head pressure in the low head pressure - high suction pressure region.

The accuracy of the 3-ton compressor simulations is also within acceptable limits. The worst error for predicting power consumption is about 16 percent, occurring at the extreme limit of high suction temperature and low condensing temperature, and increases to within 8 percent at higher condensing temperatures. Accuracy for predicting capacity is within 1 percent over the entire operating range.

There are several important differences in modeling the smaller 3-ton compressor compared to the larger semi-hermetic units. One important difference is that the smaller unit runs at 3500 RPM compared to 1750 RPM for the larger units. When running at higher speeds, the amount of closing delay for suction and discharge valves becomes more pronounced. The larger surface-to-volume ratio of smaller compressors also makes cylinder heat transfer more significant than in larger compressors and causes smaller compressors to have lower isentropic compression and expansion efficiency than larger units. Larger surface-to-volume ratio also increases suction/discharge manifold heat transfer. The percentage of oil circulating with the refrigerant is often greater in smaller compressors for a similar reason.

SIMULATING CAPACITY CONTROL

Capacity control via clearance volume control is easily simulated by changing the clearance volume as input to the model. Capacity control via late suction valve closing is easily simulated by specifying the closing delay parameter θ_S for the suction valve. A slight modification would be desirable, however, to account for throttling of the gas as it is forced back through the suction valve.

In order to simulate capacity control via motor speed control, the efficiency of the speed control device and its effect on motor waste heat must be considered. Furthermore, possible effects on valve dynamics should be explored. The speed control method has not been included in the present model.

The present model has been specially equipped to model the early suction valve closing (or "cut-off") method of capacity control.(17) One additional parameter is required to indicate the amount of capacity reduction desired. The expansion of the gas in the cylinder after cut-off is modeled in a way similar to the re-expansion portion of the stroke.

PARAMETRIC STUDIES

Results of parametric studies, showing effects of changing the parameters given in Table 1 on capacity, power, and overall efficiency, over the entire operating range of the 06D-537 compressor, are summarized in Table 2. It is important to note that compressors which are designed to have low values of θ_S and θ_D normally have high values of ΔP_S and ΔP_D, and conversely, compressors with low values of ΔP_S and ΔP_D normally have higher values of θ_S and θ_D. The effect of varying oil circulation from 0 to 10 percent by weight has little or no effect on flow and power of the compressor. Rather, the effect is to reduce cooling capacity in the evaporator by reducing the amount of refrigerant available for evaporation, since some of the refrigerant remains dissolved in the oil as it leaves the evaporator. The effect is strongly a function of evaporator superheat, percent oil circulation, and refrigerant. The higher the superheat leaving the evaporator, or the lower the oil circulation rate, the less the capacity reduction will be.

Technical details of the present compressor model are available in Reference 18.

CONCLUSIONS

The most important input parameters to the simulation model, as seen from Table 2, are ΔP_D, ΔP_S, η_{is}, and η_{mech}. The effects of both η_{is} and η_{mech} are relatively constant over the normal compressor operating range. The effect of ΔP_S, however, becomes very large at the extreme limit of low suction pressure, due to greater density change per unit pressure drop, and is the cause of simulation inaccuracies in the above region. Likewise, simulation inaccuracies in the high suction pressure - low discharge pressure region of operation result from the increased sensitivity to ΔP_D in that region. In the latter region increases in ΔP_D add substantially to the total pressure difference across which the compressor must pump, and the high mass flow rates associated with the latter region often cause increased internal pressure drops which raise the effective value of ΔP_D.

The above inaccuracies in the compressor simulation model are usually tolerable because, during normal operating in refrigeration or heat pump systems, the compressor does not operate in the extreme regions. The present compressor model can, therefore, be a valuable design or trouble-shooting tool for assessing the effects of internal compressor modifications, system modifications, non-standard applications, refrigerant changes, and more.

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Table 1

SUMMARY OF NORMAL RANGE OF VALUES FOR INPUT PARAMETERS OF COMPRESSOR SIMULATION

	1800 RPM OR BELOW			3600 RPM	
VARIABLE	LARGE	MEDIUM	SMALL	SMALL	VERY SMALL
θ_D	0 - 10°	0 - 10°	0 - 10°	5 - 20°	UNKNOWN
θ_S	0 - 20°	0 - 20°	0 - 20°	5 - 30°	
ΔP_D	10 - 30 psi	10 - 30 psi	10 - 30 psi	10 - 50 psi	
ΔP_S	1 - 5 psi	1 - 5 psi	1 - 5 psi	1 - 5 psi	
η_{is}	.94 - .98	.90 - .94	.85 - .90	.88 - .95	
η_{mech}	.94 - .98	.94 - .98	.92 - .96	.90 - .96	
%MC	.80 - 1.0	.80 - 1.0	.80 - 1.0	.80 - 1.0	
MORE SUBJECT TO DESIGN VARIATIONS					
%Oil	0 - 5%	0 - 10%	0 - 10%	0 - 10%	
Suct-Disc. Heat Trans.	$\Delta T_S \leq 20°F$	$\Delta T_S \leq 30°F$	$\Delta T_S \leq 50°F$	$\Delta T_S \leq 50°F$	

Where:

SVR = Surface to Volume Ratio of Cylinder

LARGE = SVR < 2.8 $\frac{1}{in}$

MEDIUM = 2.8 < SVR < 3.2 $\frac{1}{in}$

SMALL = 3.2 < SVR < 4 $\frac{1}{in}$

VERY SMALL = 4 < SVR

θ_D = Discharge Valve Closing Delay (Degrees after TDC)

θ_S = Suction Valve Closing Delay (Degrees after BDC)

ΔP_D = Equivalent Cylinder Pressure Overshoot on Discharge

ΔP_S = Equivalent Cylinder Pressure Undershoot on Intake

η_{is} = Compression and Expansion Isentropic Efficiency

η_{mech} = Compressor Mechanical Efficiency Due to Friction

%MC = Percent of Motor and Friction Heat Given to Suction Gas

%Oil = Weight Percent of Oil Circulating in System

ΔT_S = Additional Suction Gas Superheat Due to Suction-Discharge Heat Transfer at Low Suction Pressure-High Discharge Pressure (High Pressure Ratio) Condition

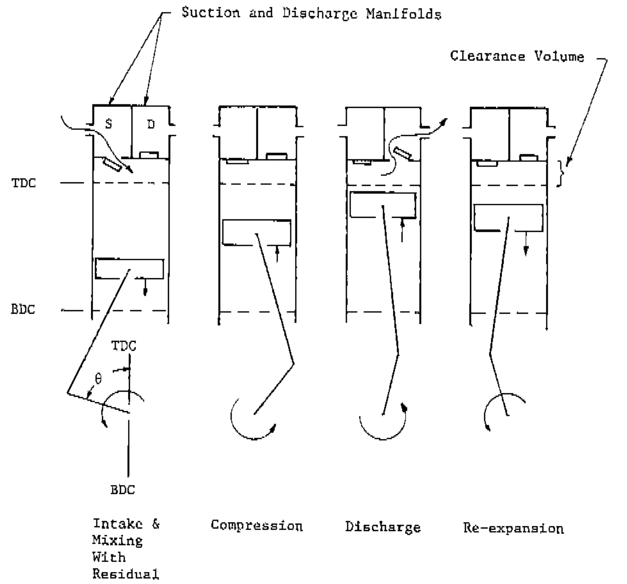
Table 2

SUMMARY OF EFFECTS OF VARYING INPUT PARAMETERS ON CARRIER MODEL 06D-537 COMPRESSOR PERFORMANCE

PARAMETER	VARY FROM - TO	CHANGE IN FLOW (%)	CHANGE IN POWER (%)	CHANGE IN OVERALL EFFICIENCY (%)	REMARKS
θ_D	0 - 10° ATDC	-4%	-3%	0%	Inversely related to ΔP_D
θ_S	0 - 20° ABDC	-4%	-4%	0%	Inversely related to ΔP_S
ΔP_D	10 - 30 psi	-3%	+3%	-3%	More significant effect at low pressure ratios-see Appendix H
ΔP_S	1 - 5 psi	-10%	-3%	-4%	More significant effect at low suction pressures-see Appendix H
η_{is}	94 - 98%	+1%	-7%	+6%	
η_{mech}	94 - 98%	+2%	-4%	+4%	

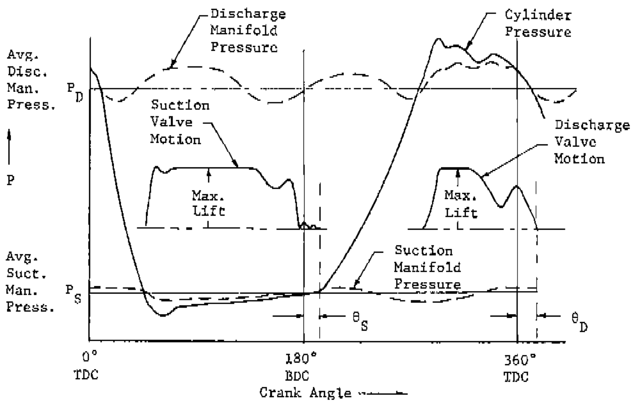
OTHER EFFECTS

%MC	80 - 100%	- Negligible
%Oil	0 - 10%	- Negligible effect on flow, power, or overall efficiency, but large effect on evaporator capacity (10% capacity reduction)
Suct.-Disc. Heat Trans.		- 30°F additional superheat at low suction pressure and high discharge pressure reduces flow by 7% with negligible effect on power, and hence reduces overall compressor efficiency by 4%. At lower pressure ratios, the additional superheat is much less, and the effect of suction-discharge heat transfer is negligible.



FOUR STEP CYLINDER PROCESS

Figure 1



TYPICAL VALVE, CYLINDER PRESSURE, AND MANIFOLD PRESSURE BEHAVIOR OF RECIPROCATING COMPRESSORS WITH PRESSURE ACTUATED VALVES

Figure 2

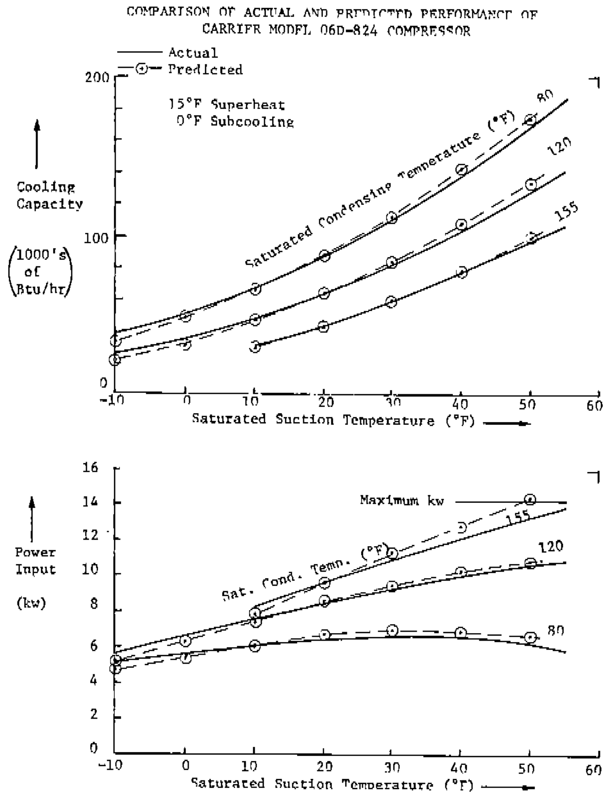


Figure 3

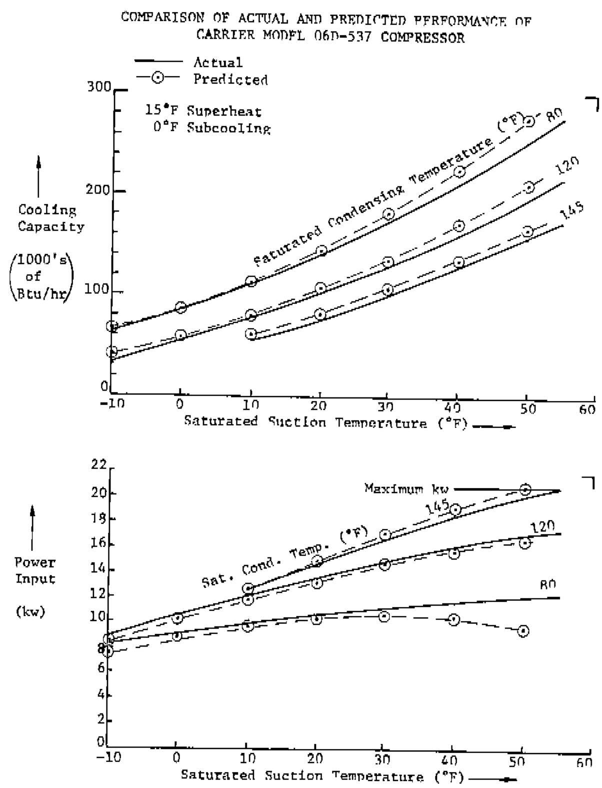
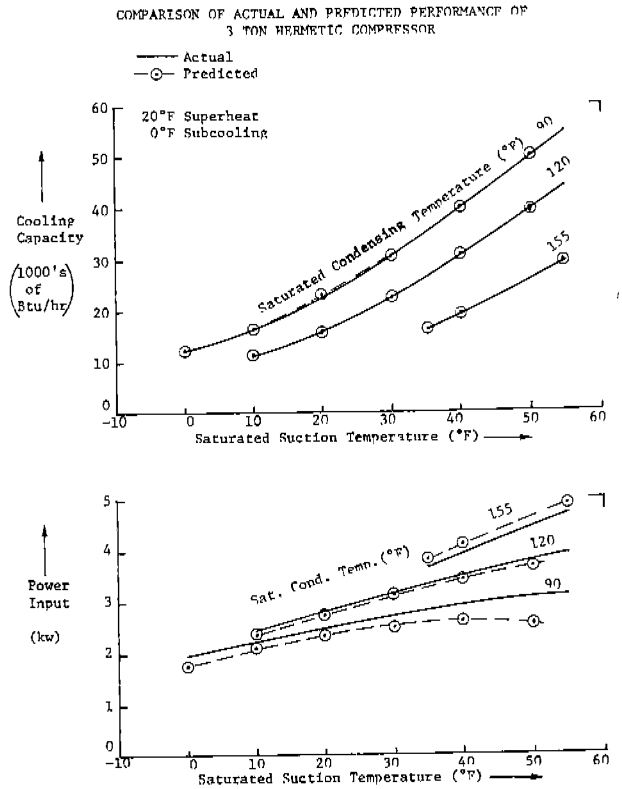


Figure 4



COMPARISON OF ACTUAL AND PREDICTED PERFORMANCE OF
3 TON HERMETIC COMPRESSOR
Actual
Predicted
20°F Superheat
0°F Subcooling
Cooling Capacity
(1000's of Btu/hr)
Saturated Condensing Temperature (°F)
90
120
155
Saturated Suction Temperature (°F)
Power Input
(kw)
Sat. Cond. Temp. (°F)
155
120
90
Saturated Suction Temperature (°F)

Figure 5