Purdue University
Purdue e-Pubs

International Compressor Engineering Conference School of Mechanical Engineering

1976

Conceptual Design of a Better Heat Pump Compressor for Northern Climates

D. Squarer

F. Sisk

S. Veyo

CONCEPTUAL DESIGN OF A BETTER HEAT PUMP COMPRESSOR FOR NORTHERN CLIMATES

D. Squarer, F. J. Sisk and S. E. Veyo
Westinghouse Research Laboratories
1310 Beulah Road
Pittsburgh, Pennsylvania 15235

ABSTRACT

Most heat pumps have been designed primarily as summer air conditioners that upon cycle reversal will provide winter heating service. Their use for winter heating, especially in northern climates has not been widespread. With the advent of an insecure convenience fuel supply as well as higher prices for both these fuels and electricity, the heat pump should very rapidly replace resistance electrical heat and should make dramatic inroads in geographical areas where heating has historically been supplied by burning oil or natural gas. A strong incentive therefore exists to improve the efficiency and low temperature capacity of heat pump systems. As the compressor is the heart of a heat pump system any improvement in its efficiency and low evaporator temperature performance should make possible a concomitant improvement in the performance of the system.

This paper documents an attempt to use analytical techniques to quantify the degree of compressor performance improvement possible through perturbations in the design of a reciprocating compressor. A four ton compressor representing those currently applied to heat pump service was first simulated with a previously developed reciprocating compressor computer model. The mass flow rate and input power estimated with the model were tuned to give good agreement with actual performance through judicious choices for the effective clearance volume and time of discharge valve closure. A sensitivity study was then conducted in order to determine the validity of certain assumptions concerning other parameters.

Several potential compressor design modifications were incorporated within the computer model in an effort to ascertain their value in improving compressor capacity and efficiency, especially at low evaporator temperatures. A characteristic map for the modified and improved compressor was then generated with the computer model. The computer model was also used to extend the map for the unmodified compressor to evaporator temperatures lower than those for which experimental data was available. These maps are to be used in subsequent studies of system performance. The use of a computer model to extend to low evaporator temperatures the compressor characteristic data now available is particularly significant as it should encourage more system analysis of the heat pump and through this optimization of the heat pump for winter heating service. This work was funded by the Electric Power Research Institute under contract RP 544-1.

INTRODUCTION

The compressors used in most heat pumps today were originally designed and refined for service in air conditioners. They were applied to service in heat pumps without major redesign. As low evaporator temperature performance was not significant for air conditioners this was thought to be an area amenable to improvement. Under the sponsorship of the Electric Power Research Institute Contract RP 544-1 we have studied ways to improve the low ambient temperature performance of heat pump systems through modifications to the compressor.

This study was analytical and the study of potential compressor improvement involved two steps: First, a representative currently used reference compressor was simulated by a computer model and performance estimates compared with available data. Second, potential modifications to this compressor were studied through the modification and exercise of the computer model.

COMPRESSOR DATA AVAILABLE

Performance data for the reference compressor of interest were available for all conditions of interest for air conditioning and southern climate heating. These data included a map of the motor input and mass flow as a function of both the evaporator and condensor temperature, a data point at the zero mass flow rate point, and the performance characteristics of the drive motor. Certain line pressure drops were also available.

COMPRESSOR COMPUTER MODEL

The compressor computer model which was used in the study is a version of a previously developed proprietary computer program.[(1),(2)] Briefly, the model has the following features: thermodynamic and transport properties of refrigerant-22 or refrigerant-12, pressure drop and heat transfer calculations in any piping configuration as well as the necessary compressor internal logic. This program simultaneously determines the temperature, mass, volume, pressure and heat flow rate through a control volume which encloses the vapor inside the cylinder. The calculations progress in small steps (approximately one crank-angle degree) through one cycle -- a cycle which includes compression, gas discharge, expansion and gas intake. The initial conditions for the first cycle are assumed, and the results at the end of the first cycle are used to compute the initial conditions for the second cycle, and so on. Between three and five computational cycles are necessary to achieve convergence between initial and final conditions.

The version of the compressor model which was used in the present study incorporates a simple valve analysis. The discharge valve is assumed to open instantaneously at a given pressure in excess of the discharge plenum pressure and to close instantaneously at a given crank-angle position after top-dead-center. This behavior has been observed in many compressors. Suction valve behavior, however, is usually more complex, and an assumed constant "pressure undershoot" required to open the suction valve may introduce an error because this pressure may depend upon operating conditions. Consequently, it was assumed that the suction valve lift is proportional to the pressure difference across the valve. The maximum valve lift is limited by the valve stop, and the valve closes instantaneously when the pressure in the cylinder exceeds the pressure in the suction head plenum. The above assumption is, strictly speaking, applicable only to an inertialess valve. However, such an approximation is much closer to actual valve behavior than the assumption of an instantaneously opened valve at a given pressure "undershoot" and instantaneously closed valve at a given crank-angle position after bottom-dead-center (BDC). The mass flow rate through the valves is proportional to an "effective flow area," KA, where K is an orifice coefficient and A is flow area. This "effective flow area" was evaluated in a detailed analysis of the flow path through the valve opening and the valve plate. In this analysis, a flow path, which is composed of parallel and series combinations of individual flow paths, was considered, and a general "effective flow area" coefficient was derived for the discharge valve and for the suction valve, the latter being a function of the valve lift. Another feature of the computer model was a provision for induction wall ports. Wall ports are effective in relieving the pressure difference across the suction valve near BDC and may improve the compressor performance. In the computer model, the wall port area was calculated as a function of crank angle, and a maximum wall port area equal to the suction valve area was assumed. No attempt was made to optimize either the port area or the port shape.

RESULTS

Computer Model Verification

The first step of the study was model verification. All relevant data on the reference compressor were input to the model, and an effective clearance was selected to fit the model prediction to the data. The effective clearance volume required was higher than the actual clearance volume quoted by the manufacturer. However, once the effective clearance volume was selected for this compressor, it remained fixed throughout the study of the compressor.

The available performance map of the reference compressor does not cover the complete temperature range which was investigated in this study. Specifically, no performance data were available at low evaporator temperatures. It was necessary, therefore, to extrapolate the performance map to complete the model verification.

Table I summarizes the model verification with respect to the motor input and the mass flow rate. (All the data which were extrapolated are marked with an asterisk.) The data it contains clearly show that the agreement between the predicted performance and the actual performance is very close for most of the range. The deviation between prediction and data is greatest for high condenser temperatures (120°F, 130°F) and a low evaporator temperature (-20°F). It should be stressed again that the data in this range were extrapolated.

Modified Compressor

The reference compressor can be modified with some redesigning to include the first five modifications given in Table II. The sixth modification (stroke/bore ratio) requires a major redesigning of the compressor.

To assess the effect of the above modifications on compressor performance, several performance maps were generated, and the calculated results were normalized with respect to the model prediction of the reference compressor. The evaluation was made primarily in terms of motor input and mass flow rate.

The performance of the compressor with five modifications (level A of Table II) is summarized in Table III. It is evident that, in addition to a visible improvement in capacity (mass flow rate) at any operating condition, the compressor operating range is extended to low evaporator and high condenser temperature, where the reference compressor has no capacity. The improvement in capacity at a given condenser temperature increases with decreasing evaporator temperature. Another comparison between the modified and reference compressors is shown in Table IV, where the efficiency ratio is tabulated. (Efficiency is defined as the isentropic reversible work relative to the required motor input.) The calculated efficiency of the reference compressor is shown in Figure 1 whereas the efficiency of the modified compressor is shown in Figure 2. A marked improvement in efficiency is clearly evident only at the very low evaporator temperatures below -20°F.

The importance of each of the modification levels listed in Table II was assessed by generating separate performance maps for each modification level. These results indicated that the modification levels may be listed in the following order of importance: 50% clearance volume reduction (most important modification), adiabatic suction line, wall ports, reduced ΔP in suction line and reduced ΔP in discharge line. The effect of the modifications considered is pronounced only at low evaporator temperatures. Compressor performance across the most frequently encountered evaporator temperatures (0°F to +40°F) is not significantly improved. Typical contributions of each of the modifications are 5 to 15 percent improvement in performance at evaporator temperature of 40°F to -20°F due to adiabatic suction gas; 1 to 2 percent at all operating conditions due to reduced suction line pressure drop; 1.6, 4.6 and 50 percent capacity increase at condenser temperature of 130°F and evaporator temperatures of 0°F, -20°F and -40°F, respectively, due to wall ports. A smaller capacity increase is obtained at lower condenser temperatures.

The sixth modification, which is listed in Table II, involves a major compressor redesigning effort. Its main advantage of course is the allowance for an additional reduction in clearance volume. Thus, if we convert the present three cylinder compressor into a two cylinder compressor and simultaneously increase the stroke and the discharge port area by 50%, we will maintain virtually the same capacity when the clearance volume is unchanged. However, since the percentage clearance volume of a longer stroke cylinder can be reduced by an additional 50%, the result is a further improvement in compressor performance due to an even lower clearance volume. The efficiency of a modified level E compressor is shown in Figure 3.

The compressor computer model was also employed to study two capacity control methods, one continuous which involves a clearance pocket and second, a discontinuous two speed compressor which is accomplished with a two speed motor. The efficiency of the latter method is higher but it has the distinct disadvantage of a step-change in capacity. It is also much simpler to accomplish without a major compressor redesigning effort.

CONCLUSIONS

The performance improvement of heat pump systems as applied to heating in northern climates is logically served by certain compressor modifications which improve the compressor efficiency and capacity at low evaporator temperature. Since the computer model used to simulate the reference compressor yields estimates of performance in good agreement with available data, we conclude that the predictions of performance for the modified compressor considered are of adequate precision to be used in the analysis of heat pump systems. These modifications should be conducive to improved system performance at low temperature. Experimental verification will be required if the improvement in system performance indicates the economic viability of the compressor modifications considered. It must be emphasized that the data used is typical rather than representing the best performing compressor available on the market today. Similar performance improvement can be expected however in any compressor of this type.

ACKNOWLEDGMENT

This work was sponsored by the Electric Power Research Institute under Contract RP-544-1, "An Investigation of Methods to Improve Heat Pump Performance and Reliability in a Northern Climate." The authors are grateful for the valuable comments of Professor J. F. Hamilton, Purdue University, who served as a consultant on this project.

REFERENCES

(1) D. Squarer and R. E. Kothmann, "Digital Computer Simulation of a Reciprocating Compressor - A Simplified Analysis," Proceedings of the 1972 Purdue Compressor Technology Conference, July 25-27, 1972, pp. 502-505.

(2) S. M. Hai and D. Squarer, "Computer Simulation of Multicylinder Compressors," Proceedings of the 1974 Purdue Compressor Technology Conference, July 10-12, 1974, pp. 178-185.

TABLE I

PERFORMANCE COMPARISON: RATIO OF COMPUTER MODEL ESTIMATES TO AVAILABLE DATA

Temperature [°F] Condenser	Evaporator	Electrical Input Ratio	Capacity Ratio
80	40	1.014*	0.9976*
	20	1.033*	1.039*
	0	1.015*	1.062*
	-20	1.062*	1.019*
	-40	1.098*	1.024*
90	40	1.010*	1.007
	20	1.016*	1.032
	0	1.030*	1.046
	-20	1.056*	0.9920*
	-40	1.018*	0.9966*
100	40	1.015	1.018
	20	1.014	1.022
	0	1.021	1.032
	-20	1.028*	0.9561*
	-40	0.9096*	0.6083*
110	40	0.9992	1.028
	20	0.9958	1.028
	0	1.000*	1.023
	-20	0.9767*	0.8772*
	-40	0.8933*	-
120	40	0.9825	1.037
	20	0.9738	1.020
	0	0.9532*	0.9754
	-20	0.8785*	0.7044*
130	40	0.9518	1.017
	20	0.9336	1.002
	0	0.8876*	0.9877*
	-20	0.7590*	0.370*

* compressor performance data extrapolated

TABLE II

COMPRESSOR MODIFICATIONS

	Modification Levels				
	A	B	C	D	E
50% Clearance Volume Reduction	+	+	+	+	+
Wall Ports	+	+	+	0	+
Adiabatic* Suction Line (motor not cooled with suction gas)	+	0	0	+	+
Reduced ΔP in Suction Line	+	0	+	+	+
Reduced ΔP in Discharge Line	+	+	+	+	+
Increased Stroke/Bore Ratio	0	0	0	0	+

Note:
+: modification is included
0: modification is not included
*: suction gas includes only 10% of motor electrical losses

TABLE III

COMPARISON OF ESTIMATED PERFORMANCE: LEVEL-A MODIFIED COMPRESSOR/REFERENCE COMPRESSOR

Temperature [°F] Condenser	Evaporator	Electrical Input Ratio	Capacity Ratio
80	40	1.0355	1.0876
	20	1.0703	1.1324
	0	1.1203	1.2333
	-20	1.1826	1.3942
	-40	1.3300	2.2686
90	40	1.0497	1.0987
	20	1.0887	1.1496
	0	1.1380	1.2406
	-20	1.2324	1.5040
	-40	1.4332	3.411
100	40	1.0659	1.1152
	20	1.1033	1.1954
	0	1.1701	1.2891
	-20	1.2978	1.6742
	-40	1.5746	11.6493
110	40	1.0840	1.1319
	20	1.1385	1.2255
	0	1.5206	1.3557
	-20	1.3862	1.9699
	-40	1.930*	66.6*
120	40	1.1038	1.1484
	20	1.1592	1.2313
	0	1.2702	1.4545
	-20	1.5170	2.6608
	-40	1.731*	43.6*
130	40	1.1316	1.1939
	20	1.1904	1.2663
	0	1.3415	1.6033
	-20	1.7266	5.4672
	-40	1.487*	17.8*

*Actual predicted value of the modified compressor. Electrical input is in [kW] and the capacity is in [lb/hr].

TABLE IV

COMPARISON OF ESTIMATED COMPRESSOR EFFICIENCY: LEVEL-A MODIFIED COMPRESSOR/REFERENCE COMPRESSOR

Temperature [°F] Condenser	Evaporator	Relative Efficiency
80	40	1.050
	20	1.058
	0	1.1007
	-20	1.174
	-40	1.706
90	40	1.0467
	20	1.0576
	0	1.0902
	-20	1.220
	-40	2.380
100	40	1.046
	20	1.084
	0	1.1017
	-20	1.2901
	-40	7.408
110	40	1.044
	20	1.0764
	0	1.116
	-20	1.421
	-40	.3157*
120	40	1.0405
	20	1.062
	0	1.1451
	-20	1.7538
	-40	.2419*
130	40	1.055
	20	1.0579
	0	1.1954
	-20	3.1672
	-40	.1214*

*Estimated efficiency of the modified compressor.

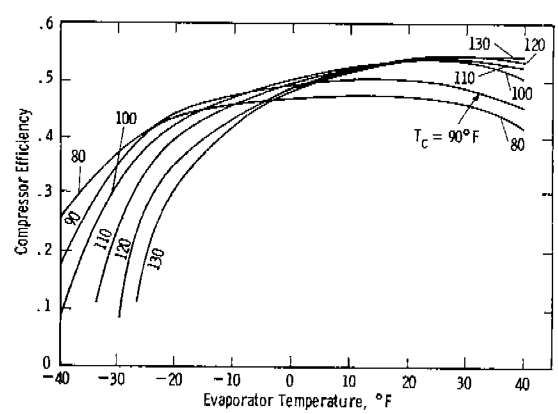

Fig. 1—Efficiency of present compressor

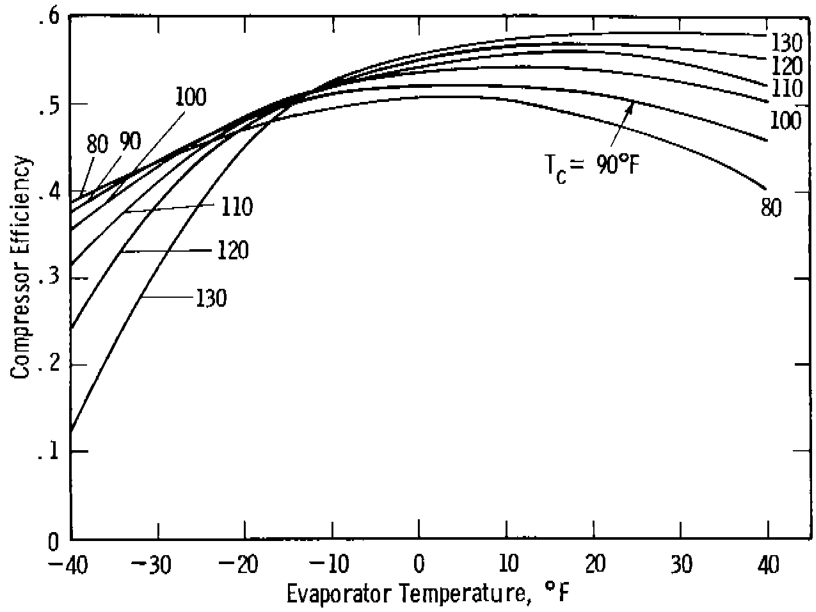

Fig. 2—Efficiency of modified level-A compressor

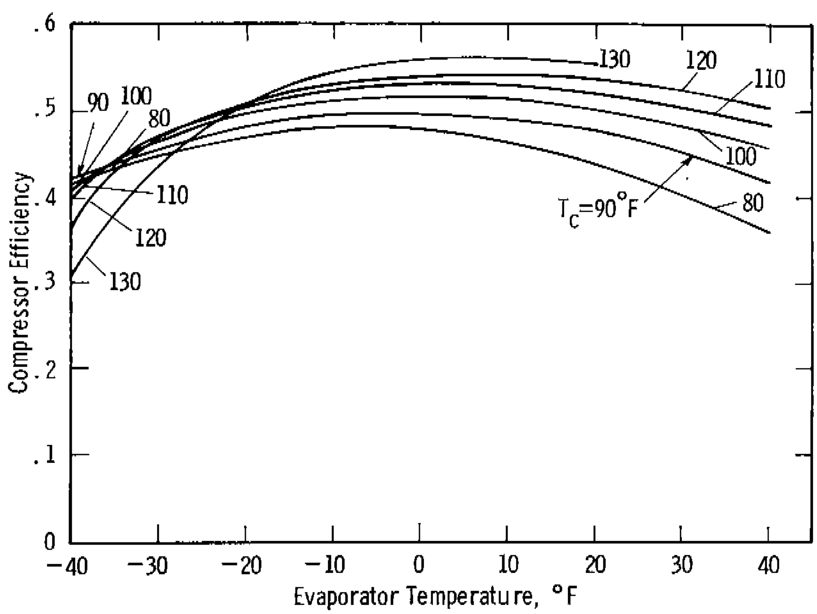

Fig. 3—Efficiency of modified level-E compressor