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MODELING ON THE PERFORMANCE OF AN INVERTER DRIVEN SCROLL COMPRESSOR

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ABSTRACT

Thermodynamic modeling of low-pressure scroll compressor was developed by combining continuity with energy conservation equation. Suction gas heating was considered using energy balance inside the low pressure shell. Pressure, temperature and mass of refrigerant-22 as a function of orbiting angle were calculated by solving the governing equations using fourth order Runge-Kutta scheme. Motor efficiency as a function of frequency was estimated using experimental results. The developed model was applied to an inverter driven scroll compressor with a variation of frequencies and major operating parameters. To confirm validity of the model, simulation results were compared with experimental results at the same operating conditions.

INTRODUCTION

Due to increasing concern for energy saving and being comfort problem on the rise, the compressor satisfying high efficiency, low noise and vibration has been demanded. Furthermore, the compressor operating wide speed range with high performance that is suitable for an inverter-driven heat pump has been required. The scroll compressor gratifying this necessity has been highlighted and many studies were performed

Inverter heat pump makes the system operate continuously by via of controlling frequency of motor that is corresponding to building load. Several researches were conducted to analysis system performance of an inverter driven scroll compressor. Ishii, et. al.[1] investigated performance of a small capacity inverter driven scroll compressor and compared dynamic characteristics with an inverter driven rotary compressor. Hirano, et. al.[2] surveyed performance of an inverter driven scroll compressor with applying self-sealing mechanism.

In the present study, a numerical model was developed to simulate dynamic characteristics of an inverter driven heat pump as a function of frequency. The developed model was derived from geometrical relations and basic thermodynamic equations for a control volume of a scroll compressor with some experimental data.

MODELING AND ANALYSIS

Based on geometrical analysis of a scroll compressor, variation of displacement volume (V) with respect to change of orbiting angle (θ) is given as:

$$\frac{dV}{d\theta} = Br_b r[2\phi_e - 2\theta - \pi - 2(\phi_e - \pi + \alpha)\cos\theta - (\pi - 2\alpha)\cos 2\theta + 2\sin\theta] - 4\pi B r_b r - Br_b r(2\phi_1 - \pi) \quad (1)$$

where B, r_b, and r represent wrap height, radius of basic circle and radius of cutter, respectively. The ϕ_e is ending involute angle, and α is initial involute angle of scroll.

Leakage from compressor chamber was determined by the equation for one-dimensional compressible flow through nozzle. Leaking process was assumed to be isentropic.

$$\frac{dm}{dt}=AC_d\frac{P_{up}}{\sqrt{T_{up}}}\sqrt{\frac{2\gamma}{R(\gamma-1)}(P_r^{2/\gamma}-P_r^{\gamma+1/\gamma})} \tag{2}$$

The equation (2) can be applied to suction and discharge port as well. The subscript "up" and "down" means high pressure and low pressure, respectively. The reduced pressure P_r is the ratio of low to high pressure, and is replaced with the critical pressure when it is greater than the critical pressure. The flow coefficient C_d was set to 0.08 in the present study. The A in equation (2) is flow area of suction or discharge port.

The flow area of suction port (A_{suc}) was calculated by the following equation:

$$A_{suc}=r(1-\cos\theta) \tag{3}$$

The flow area of discharge port was too complicate to calculate using a simple geometric equation. A graphical approach was adopted in determination of discharge area by considering pocket A and B separately. Figure 1 shows graphically determined discharge area for the selected scroll compressor in the present study.

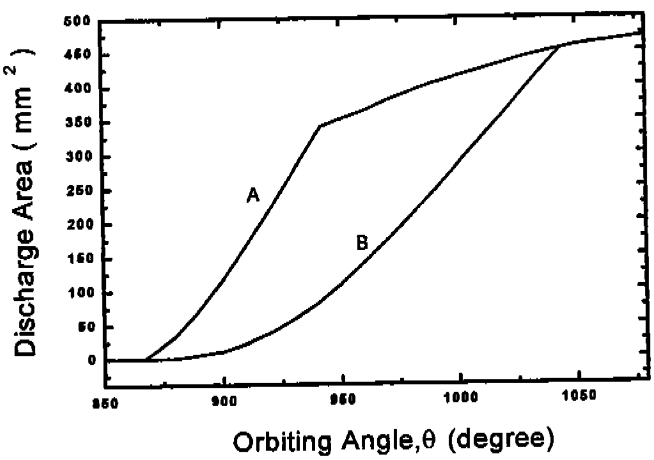

Fig. 1 Discharge area of the pocket A and B

Governing equations of the thermodynamic model were derived from mass and energy conservation equations with the equation of state for real gas. Pressure(P), temperature(T) and mass(m) were calculated by the following differential equations with respect to orbiting angle:

$$\frac{dP}{d\theta}=\left(\frac{\partial P}{\partial T}\right)_v\frac{dT}{d\theta}+\left(\frac{\partial P}{\partial v}\right)_T\frac{dv}{d\theta} \tag{4}$$

$$\frac{dT}{d\theta}=\frac{\frac{1}{m}\left[\frac{dm_i}{d\theta}(h_i-h)-\frac{dm_i}{d\theta}(h_o-h)\right]-\left[\left(\frac{\partial h}{\partial v}\right)_T-\left(\frac{\partial P}{\partial v}\right)_T v_c\right]\frac{dv}{d\theta}}{\left[\left(\frac{\partial h}{\partial v}\right)_T-\left(\frac{\partial P}{\partial T}\right)_v v_c\right]} \tag{5}$$

$$\frac{dm}{d\theta}=\frac{dm_i}{d\theta}-\frac{dm_o}{d\theta} \tag{6}$$

where h represents enthalpy and v means specific volume. The subscripts "i" and "o" represent inlet and outlet of a control volume, respectively.

Motor efficiency was measured as a function of frequency and applied to the model. It was assumed that mechanical efficiency is 92.3% at the rated frequency[3, 4] and is non-linearly decreased as frequency is increased over 60 Hz. Suction gas heating was included in the model by considering energy balance inside the low pressure shell.

Capacity of low pressure scroll compressor used in the present study was 2.8 kW. Simulation starts with input data of geometric conditions, operating conditions and frequency. The governing equations for pressure and temperature were solved by applying the fourth-order Runge-Kutta method because the variables were coupled each other. Convergence of calculation was determined by comparing adiabatic efficiency of a cycle with the value of a previous cycle.

RESULTS AND DISCUSSIONS

Figures 2 and 3 show effects of pressure ratio (PR) on performance of a scroll compressor at the standard conditions of cooling mode. For the pressure ratio of 2.5, there exists over compression, while for the pressure ratio of 3.5 there is reverse flow into a control volume (under compression). Under-compression loss results in reduction of adiabatic efficiency. Over-compression losses for the pocket B is greater than that for the pocket A at the pressure ratio of both 2.5 and 3.0.

Figures 4 and 5 show pressure-volume diagram of the pocket A and B respectively with a variation of frequency. As frequency was increased, compression processes of pocket A and B were approached to adiabatic compression. It was due to the decrease of leakage as frequency was increased.

Figure 6 shows calculated total mass including the pocket A and B. As frequency was increased, variation of leakage during compression process was decreased. Reversed flow during discharge process was increased with an increase of frequency.

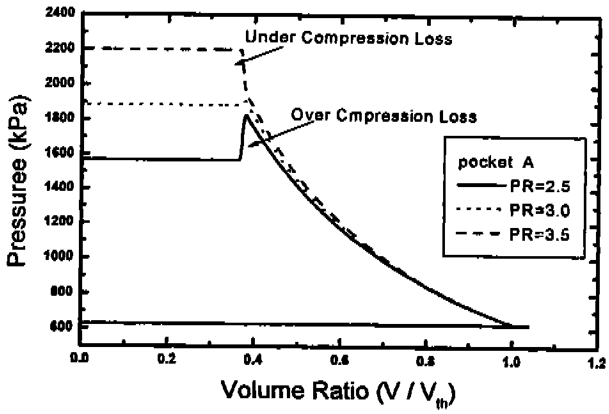

Fig. 2 P-V diagram of pocket A as a function of pressure ratio(PR).

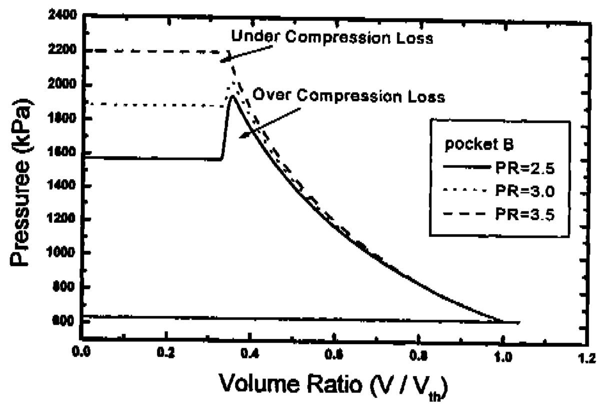

Fig. 3 P-V diagram of pocket B as a function of pressure ratio(PR).

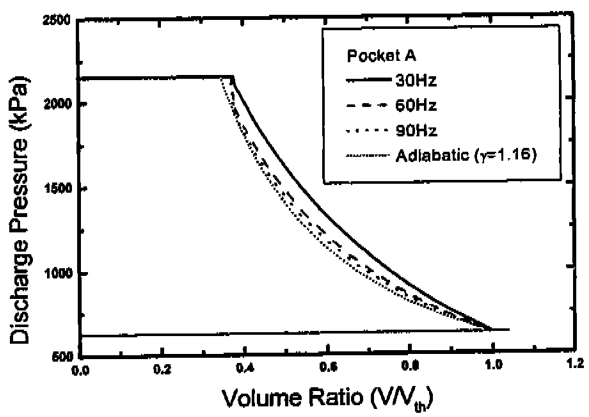

Fig. 4 P-V diagram of Pocket A as a function of frequency

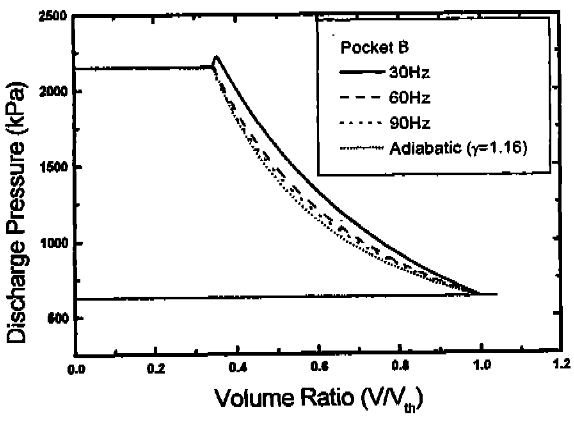

Fig. 5 P-V diagram of pocket B as a function of frequency

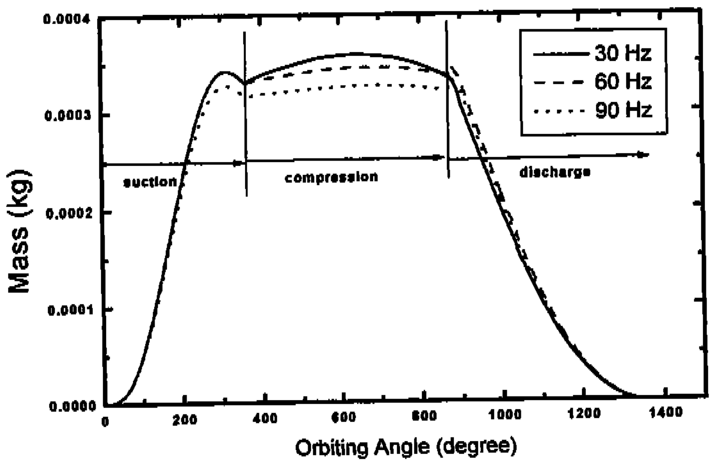

Fig. 6 Mass with orbiting angle.

Discharge temperature showed U shape pattern as a function of frequency as shown in the Fig. 7. The minimum temperature was estimated around the rated frequency of 60 Hz. The main reason for higher discharge temperature at the low frequency region was leakage into a control volume. Whereas, for the high frequency region where leakage is very low, discharge temperature was increased due to the higher temperature of suction gas that come from reduction of efficiency and increment of compressor power input.

As frequency is increased, mass flow rate was linearly increased and after 60 Hz the slope of flow rate was gradually decreased (Fig. 8). Since compression process was approached to adiabatic process as frequency was increased, adiabatic efficiency was enhanced but the slope of adiabatic efficiency was gradually diminished as a function of frequency.

Compressor power input with respect to frequency showed linear relationship up to the frequency of 75 Hz as shown in the Fig. 9. However, for the high frequency region above 75 Hz, an increase of power input to the compressor was significant because of reduction of mechanical efficiency. The maximum value of COP was estimated near 60 Hz as shown in the Fig. 10. As frequency was deviated from the rated value (both higher and lower frequency region than the rated value), COP was reduced.

To verify the model, characteristics of an inverter driven scroll compressor were measured in the compressor calorimeter. As shown in the Figs. 11 and 12, the measured values of power input and cooling capacity were consistent with the calculated values using the present model. The maximum difference of power input between the measured and the estimated was 9.1%. The maximum deviation of the predicted value of cooling capacity from the measured value was approximately 12%. The maximum difference was observed at the frequency of 105 Hz.

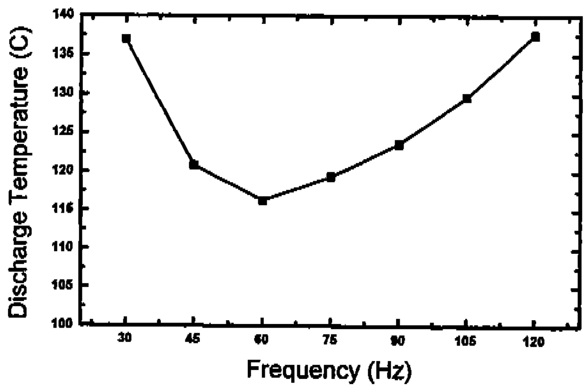

Fig. 7 Discharge temperature as a function of frequency.

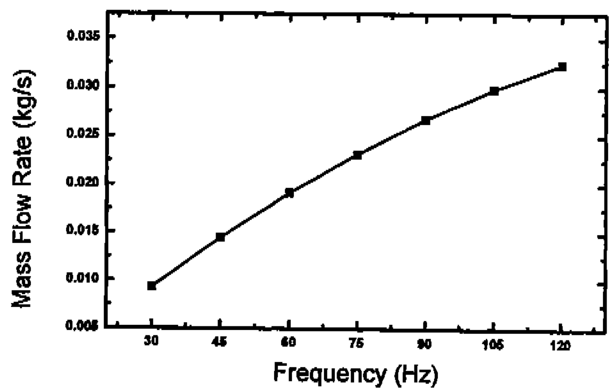

Fig. 8 Mass flow rate as a function of frequency

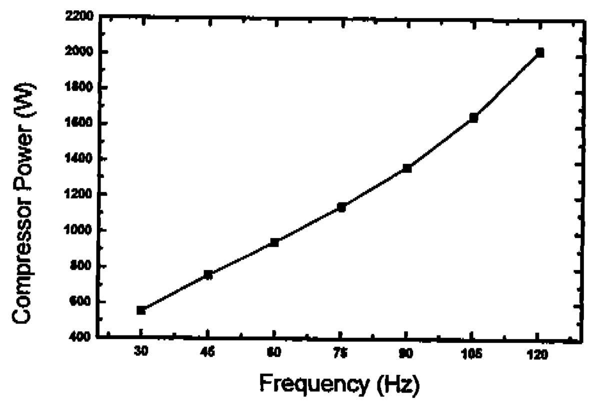

Fig. 9 Compressor power input as a function of frequency.

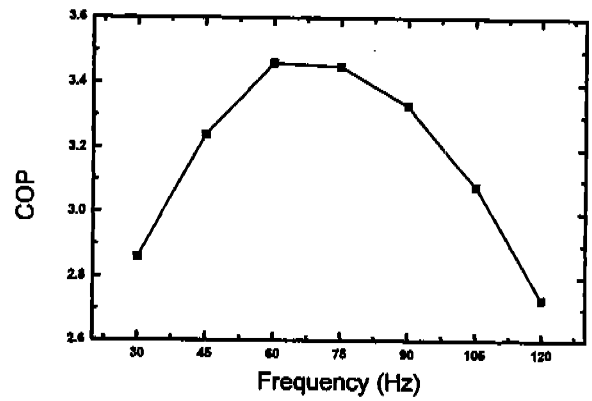

Fig. 10 COP as a function of frequency

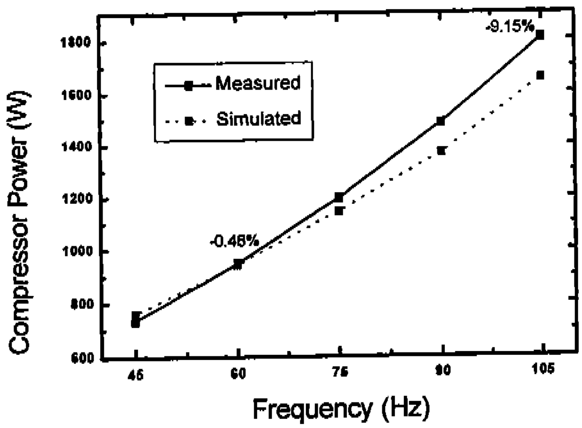

Fig. 11 Comparison of simulated results with measured value of compressor power

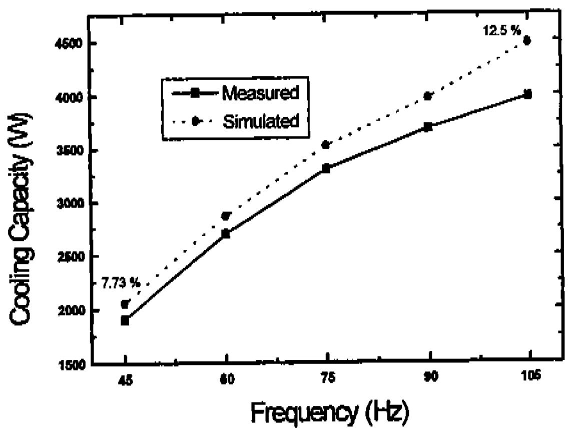

Fig. 12 Comparison of the simulated with the measured value of cooling capacity

CONCLUSIONS

A numerical model of an inverter driven scroll compressor was developed and tested at the ASHRAE-T conditions with a variation of frequency. The simulated results were compared with the experimental data to validate the developed model. Generally, the developed model was adequate to predict performance of an inverter driven scroll compressor as a function of frequency. Calculated parameters from the model were discharge temperature, mass flow rate, power input, COP, and thermodynamic properties with respect to orbiting angle. To enhance the performance of a scroll compressor, it is essential to diminish leakage at low frequency level and improve the mechanical efficiency at high frequency level.

ACKNOWLEDGMENTS

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