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EXPERIMENTAL EVALUATION OF SUCTION VALVE BEHAVIOR IN A VARIABLE SPEED HEAT PUMP COMPRESSOR

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ABSTRACT

An experimental procedure used to determine suction plenum pressure fluctuations and valve forces for a nominal 2 1/2-ton variable speed reciprocating compressor is outlined and discussed. Laboratory data are compared with predictions generated by a simplified compressor model and those obtained by using measured, instead of computed, valve forces.

INTRODUCTION

Automatic valves of the type studied here are stamped from relatively thin (.012-.020 in) sheets of spring steel. The length, width and form of these elements is determined by the number of ports and their geometric distribution. Motion is imparted to the valve as a result of uneven surface and elastic force distributions.

The timing of the various valving events is a principal determinant of both compressor capacity and efficiency. Both "opening" and "closing" are most significant in this regard as are impacts between the valve and structures limiting its motion (valve stops and the valve plate). Impacts are characterized by energy dissipation and changes in the shape and magnitude of the velocity distribution within the body of the valve. Each impact alters the timing and nature of subsequent valve events. Designers attempt to space and distribute valve stops so as to facilitate efficient operation and timely closing.

The resonant frequencies of the valve, valve stop and valve plate structure are determined by geometry and material properties. As a result, the linear dynamic characteristics of the valve and its mounting structure are not affected by variations in crankshaft speed.

The forces giving rise to interaction between the valve, valve stops and the valve plate are a function of the state of the working fluid in both the cylinder and plenum, in addition to the geometric distribution of the port area. Fluid properties change as a function of cyclic variations in cylinder volume. These occur at frequencies equal to and in multiples of the rotational speed of the crankshaft. As a result, the harmonic content of the forcing function producing valve motion is a function of crankshaft speed.

VALVE GEOMETRY

The valves studied here are of the following type.

Discharge (Flapper)	- Spring steel leaf of uniform thickness and arbitrary shape secured to the valve plate with a clamped-free mounting arrangement.
Suction (Floater)	- Spring steel leaf of uniform thickness and arbitrary shape confined to a fixed volume with a free-free mounting arrangement.

The valves and valve plate of interest are shown in Figure 1.

CAPACITY MEASUREMENT

Standard condensing, secondary refrigerant calorimeter equipment and methodologies were employed during the testing process.

The test conditions were -

Air Conditioning: 30-90 Hz in 15 Hz increments

Heat Pump: 60-120 Hz in 15 Hz increments

Air conditioning tests followed the ARI Standard rating point conditions, i.e., 45 F/130 F, 20 F superheat of return gas and 15 F subcooling of liquid condensate.

Heat pump test conditions were taken as -10 F/90 F with the same superheat and subcooling as above.

ANALYTICAL/EXPERIMENTAL CORRELATION

Air Conditioning (Figure 2). The experimental capacity data for the compressor approximate a straight line when plotted vs. crankshaft speed in the air conditioning regime. However, there is a consistent tendency for the computer to predict a lesser capacity above 60 Hz when compared to the experimental data.

Heat Pump (Figure 3). Measured capacity varies as a function of speed in the heat pump mode. It exhibits a near-linear trend line in the 60-90 Hz region and a substantial loss in output followed by a recovery in the 90-120 Hz region.

Analysis. The analytical predictions shown in Figures 2 and 3 were generated by a flow model utilizing only the basic energy equation (First Law of Thermodynamics). This program has proven adequate to predict the forcing function on the valve with reasonable accuracy at 60 Hz A/C rating point conditions. It appears that a more sophisticated model employing fluid dynamic effects may be necessary to predict flow phenomenon at speeds above 90 Hz. The resolution of discrepancies between analytical and experimental performance was undertaken as a means of clarifying the important parameters relevant to variable speed operation.

Measurement of Suction Valve Motion. A Bently Nevada proximity probe was used to measure actual suction leaf opening, closing and bounce timing, for comparison with the simulation's predictions for those events.

The curves shown in Figure 4 for 105 Hz heat pump conditions are characterized by two distinct open-close events. The first closing takes place at or near bottom dead center (180° A.T.C.). The second occurs at approximately 260° A.T.C., or approximately 100° B.T.C. (before top center) of the next cycle.

The late closing of the valve at 105 Hz accounts for the loss in capacity at this frequency and is one of the reasons the capacity curves fail to exhibit a linear relationship with crankshaft speed.

The proximity probes used to generate the voltages plotted on the ordinate of Figure 4 do not produce output that is a linear function of displacement. This nonlinear nature inhibits a direct comparison between measured and computed data discussed below.

Measurement of Instantaneous Cylinder and Suction Plenum Pressures. Additional tests were undertaken to ascertain the roll of gas-born pressure waves. These tests involved experimental determination of the dynamic forcing functions acting on the suction valve.

Figure 4 shows both the pressure differential acting across the valve and the motion of the valve for heat pump operation at 105 Hz. The second relative maxima of differential pressure is much higher, compared to the first relative pressure maxima for the 105 Hz case.

The second peak in the pressure curve for 105 Hz can be observed to occur at or near bottom dead center at a point in the cycle where the valve is closed. Suction valve closing at B.D.C. would normally be expected to produce an ideal situation because the volume rate of change in the cylinder would be negligible at this point and the pressure rate of rise in the cylinder small. As a result, the valve should remain closed.

However, the valve reopens almost immediately in this case due to the relatively large differential pressure acting on it. The increase in differential pressure at B.D.C. is the result of a traveling pressure wave in the suction plenum that arrives at the suction ports at a time when the rate of rise of pressure on the cylinder side of the valve is small. As a result, the valve is dislodged from the seat and a second opening is initiated. This event begins at or about 180° A.T.C. (i.e., B.D.C.) and lasts until approximately 260° A.T.C. or 100° B.T.C. of the next cycle. Thus a resonant-like fluid wave in the suction plenum begets a second valve opening and results in a loss of pumping capacity.

The 105 Hz data shown in Figure 4 illustrate the anomaly in the motion of the suction valve which was first noticed as a loss in capacity at 105 Hz. It was possible to eliminate this behavior at 105 Hz by raising the discharge pressure, i.e., changing the operating conditions of the compressor. This procedure shifted (but did not eradicate) the resonant-like behavior off the 105 Hz test point.

CONCLUSIONS

The pressure measurements provide clear evidence that resonant-like pressure pulsations occur in the suction plenum of the test unit under heat pump conditions at crankshaft speeds in the 100-110 Hz range. The pressure fluctuations were found to arrive at the suction ports at or near the bottom dead center point in the cycle and dislodge the valve from its near-closed position, thereby begetting a second open-close event. Conditions which facilitate this behavior, in addition to speed and operating conditions, include: 1) the valve being of the floater or "free-free" type and 2) relatively small nominal pressure differentials acting across the suction valve.

The experimentally-measured pressure function acting on the valve was input to the computer model and used therein to force the analytical valve. The resultant predictions of valve motion are shown in Figures 5 through 9. The correlation between Figures 5 through 9 and Figures 10 through 14 is reasonable. In comparing these two sets of graphs, it's important to remember the difference in linearity which exists between the experimentally-determined Bently Nevada proximity data and the computer-generated output.

The agreement between the two sets of curves is not perfect. The prediction at 105 Hz is shown in Figure 8. The model is able to predict a second opening of the valve as a result of this change. In addition, the plots pertaining to frequencies both above and below 105 Hz show a much smaller second opening.

These results support the conclusion that the valve model was not the cause of the disagreement noted earlier between theory and experiment. Instead, the fluid model, which has been shown to produce inadequate predictions of the dynamic forces acting on the suction valve, is believed to be the primary cause of this discrepancy.

The correlation between observed and predicted valve motion was substantially improved when the measured pressure differentials were used to force the valve model in place of their analytically-determined counterparts. The corresponding improvement noted between calorimeter and simulation capacity data is shown in Figure 15. This result indicates the need for a more sophisticated fluid model than the energy equation at crankshaft speeds above 90 Hz.

ACKNOWLEDGMENT

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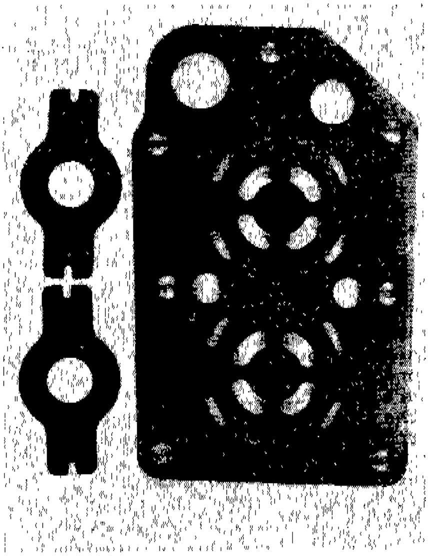
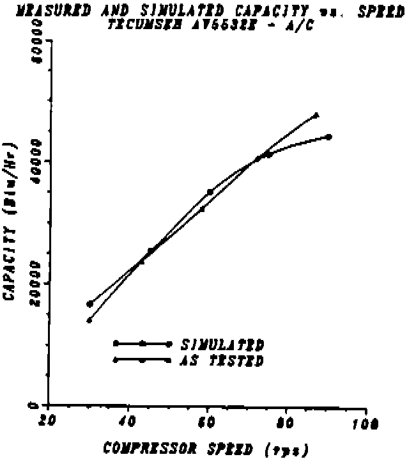


Figure 1. Valves and Valve Plate



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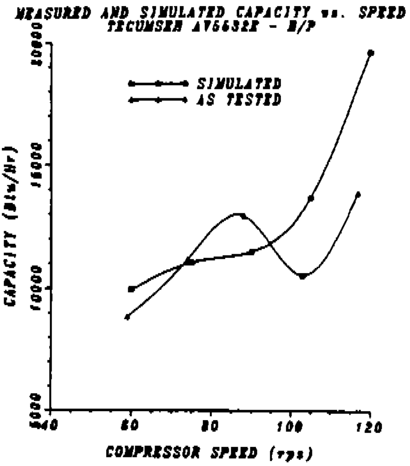

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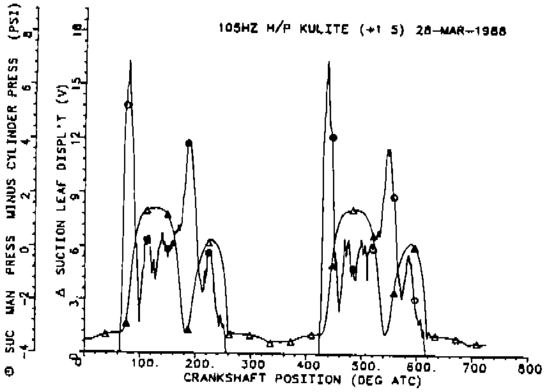

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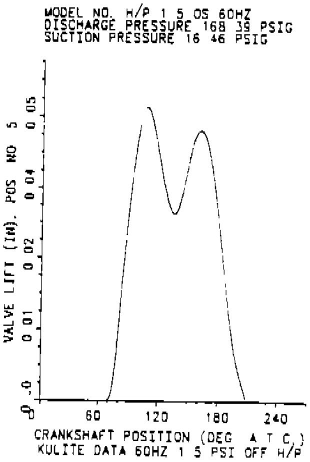

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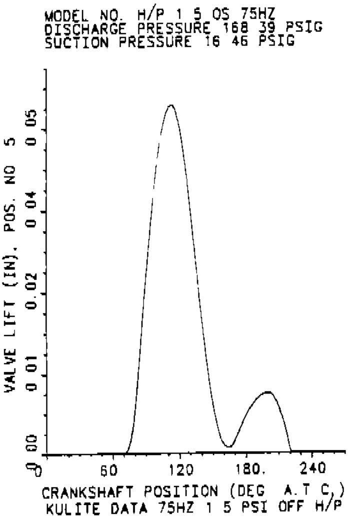

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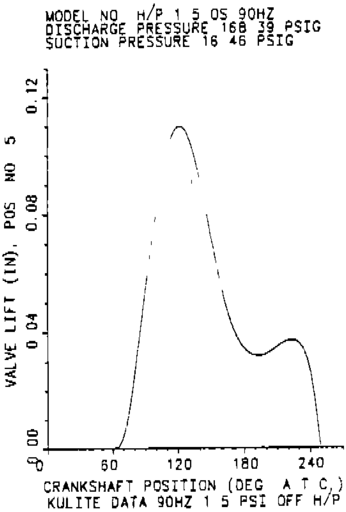

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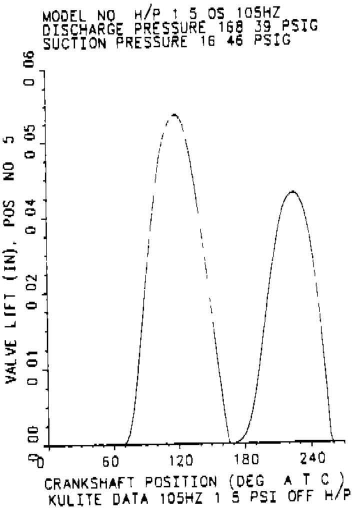

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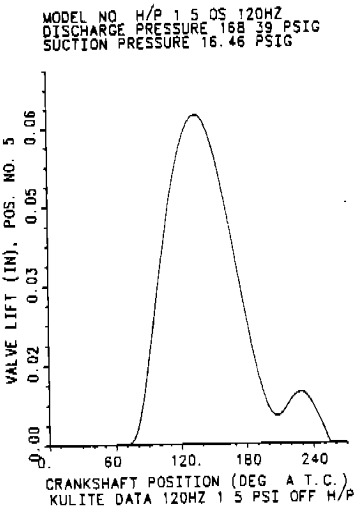

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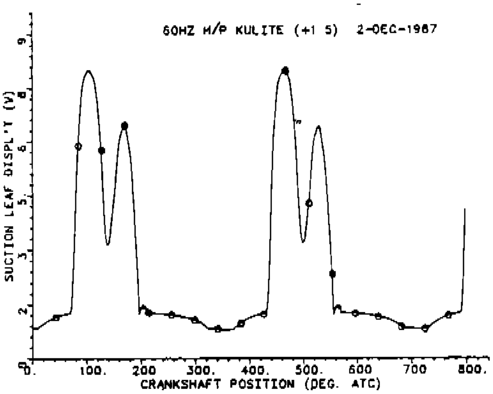

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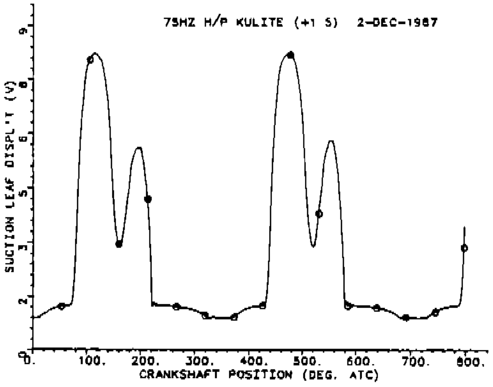

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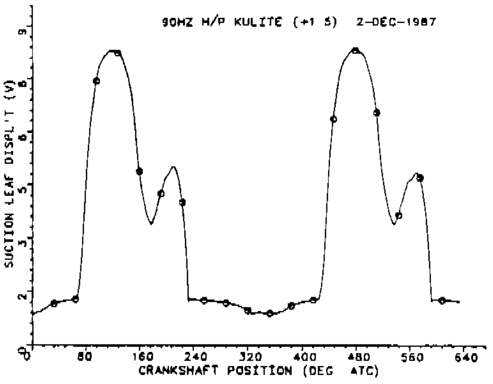

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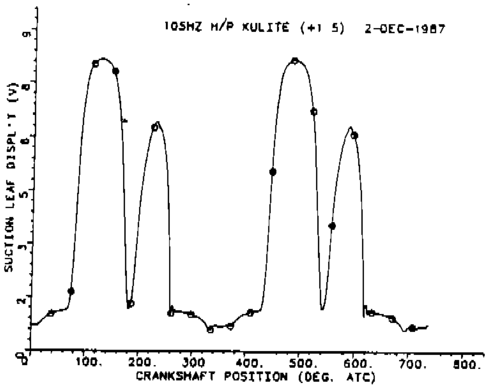

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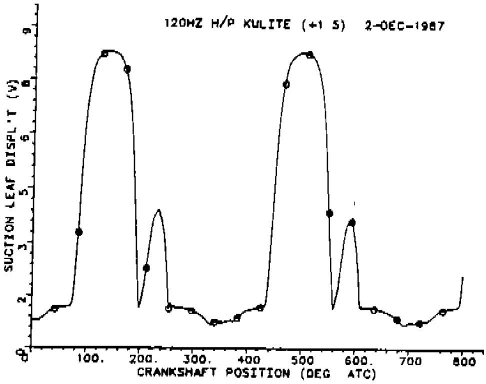

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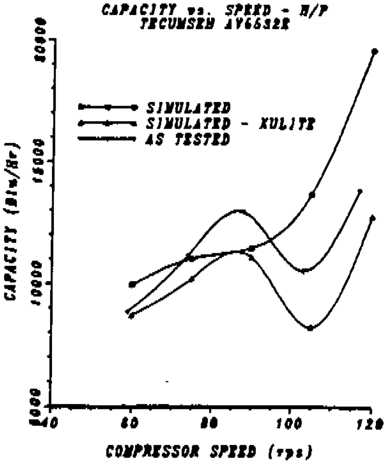

Figure 15