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Energy Consumption in Hermetic Refrigerator Compressors

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INTRODUCTION

A generalized approach for measuring and partitioning the energy losses of hermetic refrigeration compressors will be developed. Mechanical losses, valve losses, suction gas heating losses, leakage and heat transfer losses are dealt with separately and their effect on refrigerator EER is discussed. An experimental program to verify the general approach was undertaken and the results are presented. This experimental program involved testing a standard 1/4 hp hermetic reciprocating refrigeration compressor and was carried out, using the manufacturer's test facilities.

GENERALIZED APPROACH

The overall efficiency for hermetic compressors is defined as:

$$\eta_o = \frac{M\Delta h_s}{\omega} \qquad (1)$$

where:

- M - is the actual delivered mass flow per cycle
- Δh_s - is the isentropic enthalpy change based on suction line(s) conditions (see Figure 1)
- ω - is the actual shaft work per cycle and is equal to the motor power times the motor efficiency (η_e)

Unfortunately, overall efficiency does not reveal much information about specific losses inside the compressor. The loss in compressor performance due to suction gas heating and mechanical losses could be measured by breaking the overall efficiency into components:

$$\eta_o = \frac{\oint pdv}{\omega} \times \frac{\Delta h_s}{\Delta h_{sc}} \times \frac{M\,\Delta h_{sc}}{\oint pdv} = \frac{M\,\Delta h_s}{\omega} \qquad (2)$$

where:

- $\oint pdv$ - is the pressure-volume gas compression work per cycle or the work corresponding to the area in the pressure-volume diagram.
- $\frac{\oint pdv}{\omega}$ - is the usual definition of mechanical efficiency (η_m) and is the ratio of the indicated or gas compression work per cycle to the shaft work per cycle.
- Δh_{sc} - is the enthalpy change from suction cylinder conditions, isentropically to discharge pressure.
- $\frac{\Delta h_s}{\Delta h_{sc}}$ - is the suction gas heating efficiency and indicates compressor losses due to suction gas heating.
- $\frac{M\,\Delta h_{sc}}{\oint pdv}$ - can be defined as the gas compression efficiency (η_c). It is the ratio of the theoretical work per cycle to compress the delivered mass isentropically from suction cylinder conditions to discharge line pressure, to the actual indicated gas compression work per cycle.

The gas compression efficiency gives some indication of how good the compression process is compared to an isentropic ideal process, but does not break down specified losses such as valve losses, heat transfer losses, and leakage losses. However, valve losses can be found by comparing the actual pressure-volume diagram to an ideal valveless compressor-volume diagram. In Figure 2 the area enclosed by the p-v (pressure-volume) curve and greater than the discharge pressure represents the work loss per cycle for discharge porting. Likewise, the area enclosed by the suction pressure and the p-v curve represents the work loss per cycle for suction porting.

There are also losses during the compression and expansion stroke due to heat transfer and leakage. Figure 2 shows a pressure oscillation during the expansion process. This is probably caused by discharge valve bounce or flutter, which allows high pressure gas to leak into the cylinder. This leakage represents a loss. To determine the loss from the compression and expansion portion of the stroke, a test was performed involving no net mass flow which eliminates the discharge and suction process. Assuming that the heat transfer process and the discharge valve flutter is the same as during the performance test, the gas compression work or the enclosed area in the p-v curve excluding porting loss will approximate the losses during compression and expansion. Figure 3 illustrates this p-v curve. The test procedure was to first close the suction line valve and allow the compressor to draw a vacuum. Next, the discharge line valve was closed and gas was bled into the compressor until a differential pressure across the compressor equal to that during operation was achieved.

TEST PROCEDURE

A standard production 1/4 hp hermetic refrigeration compressor with a resistance start, induction run motor was selected for testing. A standard calorimeter performance test was run on the test compressor and checked against a large sample of similar compressors to insure the compressor was representative. The only changes made to the compressor were:

- A pressure tap with an Entran Devices EPS #1032 pressure transducer. The increase in clearance volume was estimated to be less than 1%.
- A suction cavity thermocouple
- A thermocouple placed on the discharge valve retainer.
- A small electrical contact placed on the crankshaft to indicate top dead center.
- Two Conax feed-throughs were added for thermocouple and pressure transducer leads.

The pressure transducer output was connected to an oscilloscope with a Polaroid camera attachment. The pressure transducer zero was calibrated by recording the equilibrium pressure (compressor stopped) with the compressor at operating temperature. Compressor speed was determined from the pressure-time trace on the oscilloscope. This is probably accurate to within $\pm$ 10 rpm. Compressor speed was assumed to be constant. A computer simulation of the compressor indicated a maximum change in compressor speed of 75 rpm/revolution or $\pm$ 2% of the time average. Shaft power was calculated from the motor input power and the motor efficiency. Standard manufacturers' motor curves were used with a correction made for operating temperature. Tests were run in the compressor manufacturer's calorimeter test cell.

TEST RESULTS

Tests were run at three conditions, - 15, - 10, and + 15°F evaporator temperature with a 100, 120 and 110°F, respectively, condensing temperature. Results of these tests are shown in Table 1. A loss breakdown for the - 10°F evaporator temperature test condition is shown in Table 2, and the p-v curve is given in Figure 4.

The power to compress the delivered gas given in Table 2 (149 watts) is the ideal gas compression power for a lossless compressor. This agrees well with the isentropic compression power from suction cylinder conditions ($M\ \Delta h_{sc}$) which is 147 watts.

SUMMARY

A test program was performed to analyze the energy breakdown in a refrigeration compressor. The complete elimination of the individual loss elements in the compressor will affect the compressor's performance substantially as shown in Table 3. These energy savings represent the ultimate potential and are far from being practical design goals. Some areas to investigate energy saving options for these compressors are:

1. Improvement of motor efficiency from about 70% to 80-85% with capacitor run windings, and higher grade silicon steel laminations.
2. Larger discharge port area and reduced valve bounce.
3. Reduction of suction gas heating by direct injection of suction gas into suction muffler with alternative compressor cooling.

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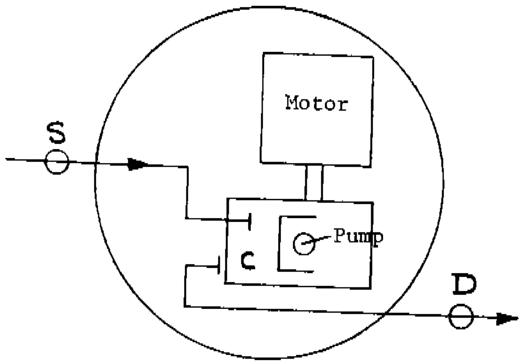

Figure 1

Hermetic Compressor

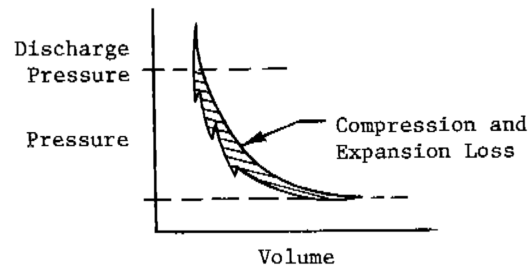

Figure 3

Zero Mass Flow p-V Curve

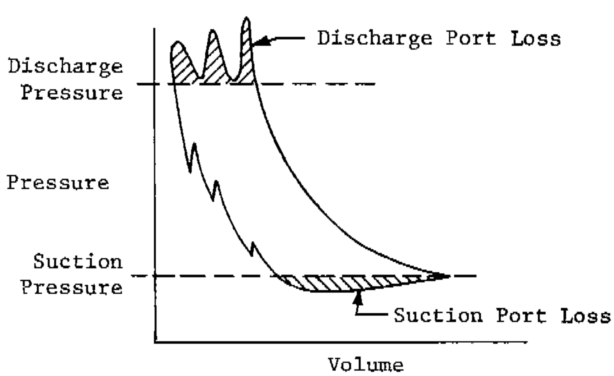

Figure 2

Pressure-Volume Diagram for

Reciprocating Compressor

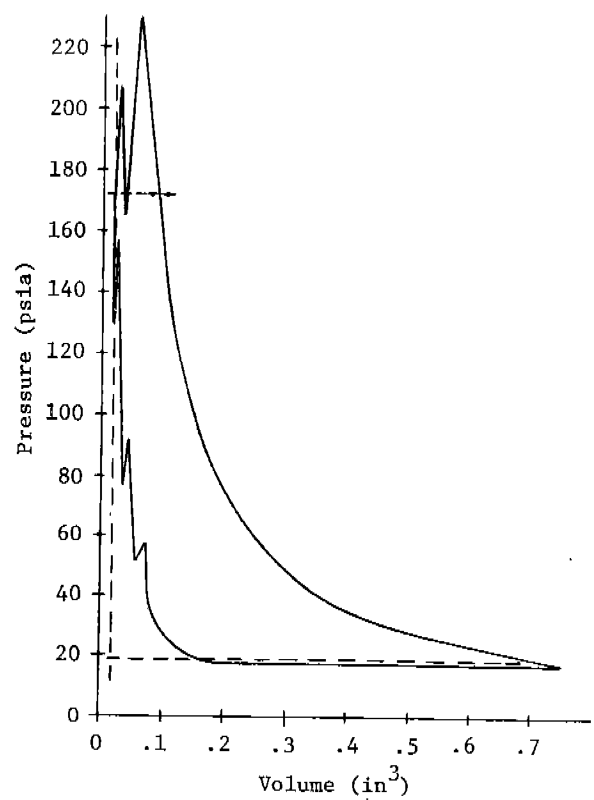

Figure 4

- 10°/120°F p-v Curve

Table 1

Test Results

DATA			
Evaporator Temperature (°F)	- 15	- 10	+ 15
Condensing Temperature (°F)	100	120	110
Suction Pressure (psia)	17.1	19.3	32.0
Discharge Pressure (psia)	131.3	172.0	150.3
Suction Line Temperature (°F)	90	90	90
Suction Cylinder Temperature (°F)	223	229	213
Mass Flow (lb/hr)	17.02	17.91	36.46
Motor Power (watts)	287	309	437
Refrigerant		R-12	
CALCULATIONS			
Speed (rpm)	3,510	3,510	3,488
Gas Compression Power (watts)	170	179	248
Shaft Power (watts)	195	210	309
Isentropic Compression Power (watts)	98	110	152
MOTOR EFFICIENCY			
Motor Efficiency (η_e)	.68	.68	.71
PUMP EFFICIENCY			
Mechanical Efficiency (η_m)	.87	.85	.80
Suction Heating Efficiency $\left(\frac{\Delta h_s}{\Delta h_{sc}}\right)$	.80	.76	.80
Compression Efficiency (η_c)	.73	.82	.77
Overall Efficiency (η_o)*	.50	.53	.49

*Overall efficiency refers to the pump alone $\eta_o = \eta_c \frac{\Delta h_s}{\Delta h_{sc}} \eta_m$, Equation 2.

Table 2

Power Breakdown for - 10°F Evaporator/120°F Condenser

	Power Loss in Watts	Power Input at Each Stage
Power into Motor		309
Motor Loss	99	
Shaft Power to Compressor		210
Mechanical Losses	31	
Power to Compress Cylinder Gas		179
Discharge Porting Losses	12	
Suction Porting Losses	5	
Compression and Expansion Losses	13	
Power to Compress Delivered Gas		149

Table 3

Effect of Complete Elimination of Individual Loss Components on Refrigerator Performance

Individual Loss				
Description	Watts	Electrical Power to Compressor Motor* (Watts)	Ultimate Potential Savings for Refrigerator† (KWH/Day)	EER
No change	0	309	0	3.2
Motor Losses	99	210	1.1	4.7
Pump Mechanical Losses	31	263	.5	3.75
Discharge Porting Losses	12	291	.2	3.4
Suction Porting Losses	5	302	.1	3.3
Compression and Expansion	13	290	.25	3.4
All of the Above Total Losses	160	120	1.7	6.6
Elimination of Suction Gas Heating	-	-	1.1	4.2
All of the Above	-	-	2.4	8.7

*Based on a motor efficiency of 68%.

†Based on a standard refrigerator operating 60% on time and normally consuming about 4.6 KWH/day total, and standard compressor performance 750 Btu/hr capacity and system EER = 3.2 Btu/watt-hr.