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Analytic and Experimental Techniques for Evaluating Compressor Performance Losses

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ANALYTIC AND EXPERIMENTAL TECHNIQUES FOR EVALUATING COMPRESSOR PERFORMANCE LOSSES

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INTRODUCTORY REMARKS

This paper is presented to aid the engineer who must identify the performance losses of a reciprocating hermetic compressor. It is also the aim of this paper to present an overview of all the major performance losses rather than concentrate on a few specific areas. The compressor performance losses considered are due to suction gas superheating, mean manifold pressure drop, manifold pressure pulsations, valve dynamics, piston ring blow-by and clearance volume. Losses due to motor inefficiency and dynamic friction are not considered. An experimental analysis of each category has been conducted to present typical instrumentation techniques, and give a range for the magnitude of each loss. A detailed theoretical analysis of each area is beyond the scope of this paper. However, a short theoretical analysis of the losses due to the suction gas superheating, mean manifold pressure drop, piston ring blow-by and clearance volume are considered.

SUCTION GAS SUPERHEATING

Suction gas is superheated as it passes over the motor and the crankcase. As the amount of suction superheating increases, the effective performance of the compression process will decrease. At a condition of 45°/130°/65° there is approximately a 2% loss in compressor performance for every 10°F rise in suction temperature. The rise in suction temperature can easily be determined by placing a thermocouple in the suction ports.

Theory

Performance is affected by the amount of suction gas superheating. As the suction gas superheat within the shell increases the power to maintain a constant flow rate at a given operating pressure differential increases. This can be shown on the following p-h diagram.

The ideal power to compress the gas is defined as the product of the flow rate and the change in enthalpy from p_s to p_d along a given entropy line. With a minimum amount of superheat the compression process follows the isentropic line S_2 rather than S_1. Since the change in enthalpy for path 2 is less than that for path 1 the power to compress the gas is reduced.

The following example illustrates the effect of superheat on the performance.

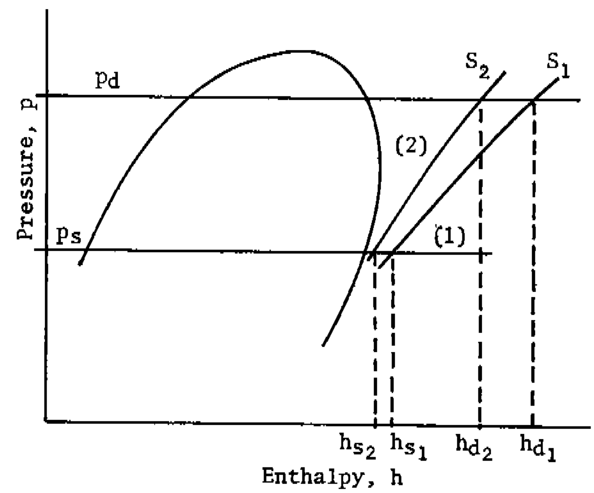

Figure 1: Superheat Effect on the Vapor Compression Cycle

Assumptions:

Refrigerant: R22
Compressor Operating Condition: 45°/130°/65°
Discharge Manifold Pressure: 315 psia
Suction Manifold Pressure: 90 psia
Discharge Overpressure: 20%
Maximum Cylinder Pressure: p_d = (315-90)(.20)+315 = 361 psia
Suction Underpressure: 2%
Minimum Cylinder Pressure: p_s = 90-(315-90)(.02) = 85 psia
Clearance Volume: (Min Cyl Volume/Max Cyl Volume)
Clearance Volume: 5%

Flow Rate = [(Mass at Max Cyl Volume) - (Mass at Min Cyl Volume)] X (Compressor Speed)

Power = Flow Rate X $(h_d - h_s)$

For path 1

T_{s1} = 150 °F, p_{s1} = 85 psia, S_1 = .254 Btu/Lb°R,
h_{s1} = 127.36 Btu/Lb, T_{d1} = 293°F, p_{d1} = 361 psia,
h_{d1} = 147.85 Btu/Lb
Specific Power = 360 W/Lb. of Refrigerant Compressed

For path 2

T_{s2} = 100°F, p_{s2} = 85 psia, S_2 = .239 Btu/Lb°R,
h_{s2} = 118.6 Btu/Lb, T_{d2} = 243°F, p_{d2} = 361 psia,
h_{d2} = 137.15 Btu/Lb
Specific Power = 325.9 W/Lb of Refrigerant Compressed

In addition to reducing the specific power a reduction in superheat will increase the flow rate. For an ideal compression process a 50°F drop in T_S at 45°/130°/65° will increase the flow rate by 11%. The performance improvement will be 9.4% for the 50°F reduction in superheat (entering the cylinder), or a 1.9% for every 10°F drop. Experimentally, both the performance and the flow rate changes will be shown.

Experimental Results

The performance improvement theoretically predicted by reducing the suction superheat can be verified by cooling the suction gas just prior to compression with a counter flow heat exchanger as shown in Figure 2.

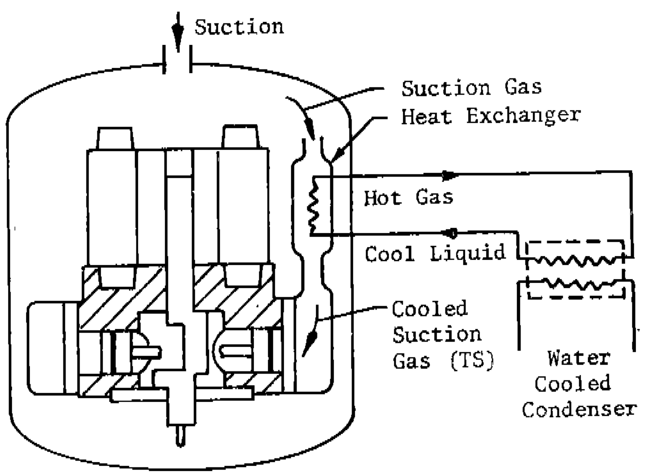

Figure 2: Compressor System Used to Verify the Suction Superheat Performance Effects

The counter flow heat exchanger is coupled to a water cooled condenser. Hot gas (R22) is drawn from the heat exchanger to the condenser by the lower vapor pressure in the condenser. Liquid refrigerant in the condenser drains by gravity back to the heat exchanger to cool the suction gas. Results from the test at 45°/130°/65° are as follows:

Table 1

	Without Heat Exchanger	With Heat Exchanger
Capacity (Btu/Hr)	17087	18168
Power (Watts)	2038	2017
Performance (Btu/W-Hr)	8.38	9.0
Flow Rate (Lb/Min)	4.5	4.8
Suction Temp. Entering Cylinder Cover (°F) (TS)	118	76
Compressor Discharge Temp. (°F)	225	194
Heat Rejected by Heat Exchanger (Btu/Hr)	0	2195
% Performance Improvement		7.5
% Performance Improvement per 10° Drop in Suction Temp.		1.8

The test results verify the predicted performance improvements within 5%. The flow rate predictions are only within 25% of the theoretical value. The large flow rate discrepancy occurred because the theory calculated the change in flow rate due to a 50°F drop in TS at the maximum cylinder volume and neglected all flow losses and valve dynamic changes.

MEAN MANIFOLD PRESSURE LOSSES BEFORE AND AFTER THE COMPRESSION PROCESS

A mean pressure drop through the suction and/or discharge manifolds will superimpose a higher pressure ratio on the cylinders. The new operating conditions in the cylinder which are produced by the increased manifold pressure loss will reduce the performance of the compressor. It can be shown that the major performance loss per psi manifold drop will occur when the flow restrictions are on the low side of the compressor. The magnitude of these flow restrictions can be measured with either a manometer tube or two transducers capable of measuring absolute pressure.

Theory

In this analysis all energy losses due to flow friction will be neglected. Therefore, all performance losses will be due to a change in the operating condition within the cylinders. As an example consider a compression process operating with a 20% overpressure and a 2% underpressure. The operating conditions will be the same as those in the section on "Suction Superheat".

The ideal power to compress the gas is 360 Watts per pound of refrigerant compressed as derived in the previous section. The power will increase to 362 Watts per pound of refrigerant compressed if a 4 psi pressure drop is imposed on the discharge manifold. On the other hand, if a 1.5 psi pressure drop is imposed on the low side of the compressor the power will change from 360 Watts to 369 Watts per pound of refrigerant compressed. These results have been experimentally verified. (See Table 2)

Table 2

Percent Change in Effective Performance

Condition = 45°/130°/65°	Discharge Drop = 4 psi	Suction Drop = 1.5 psi
Theory:% Change in Effective Performance	.5%	2.5%
Experimental:% Change in Effective Performance	1%	1.5%

The above experimental results confirm that a mean manifold pressure drop on the low side of a reciprocating compressor is more detrimental to compressor performance than the same pressure drop on the high side. The difference between the theoretical and experimental results is due to the ideal assumptions for the compression process.

SUCTION PRESSURE PULSATIONS

Pressure pulsations in the suction manifold can be used to improve the performance. One can easily determine if the pulses are detrimental to performance by examining a pressure transducer trace of the manifold pressure which is correlated with the valve motion. These pulsations affected the performance of the test compressor by 12%.

Experimental Evaluation

Pressure pulsations affect the valve motion, underpressure, power and capacity. They can delay the valve motion and/or reopen the valve after it has closed. A theoretical discussion of predicting, the effects of pressure pulsations on performance is beyond the scope of this paper. Experimentally two characteristic effects are shown. The first effect deals with tuning out the major performance controlling pulsations and the second deals with correlating the major pulsations with the suction valve motion.

a. Detuning the Suction Manifold - Compressor performance can be improved by tuning out the major pulsation. If all pulsations are eliminated the suction manifold is essentially an infinite plenum volume. The amplitude and frequency content of the pressure pulsations within the suction plenum can be adjusted by a tunable tube. (See Figure 3)

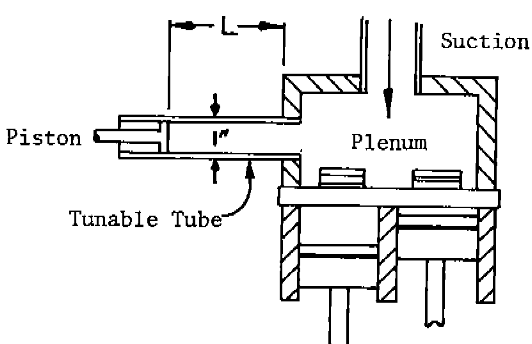

Figure 3: Tunable Suction Chamber

The pulsations in the suction manifold with a tuned tube of length "L" equal to zero is shown in Figure 4.

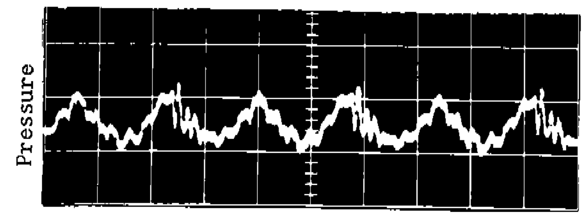

Figure 4: Suction Manifold without Tuned Tube

The major frequency component of the pulsations shown in Figure 4 is 116 Hz. This frequency was eliminated with a 1/4 wavelength tube. A scope trace of the manifold pressure with the quarter wavelength tube is shown in Figure 5. Note that the 116 Hz content has been eliminated. This technique improved the capacity by 4.5% and the performance by 2%.

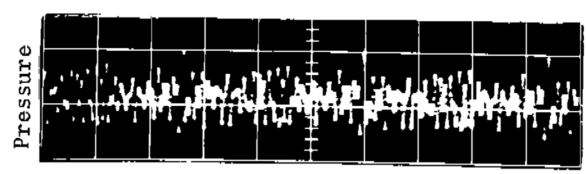

Figure 5: Suction Manifold Trace with 1/4 Wavelength Tuned Tube

b. Correlating the Valve Motion and the Pressure Pulsations - Compressor performance can be improved by correlating the amplitude and frequency of the suction pressure pulsations with the valve motion. If the pressure within the suction plenum is at its peak as the valve opens the cylinder will be "supercharged". This is accomplished by changing L in Figure 6 until the dynamic characteristics of the system produce the wave forms shown in Figure 7.

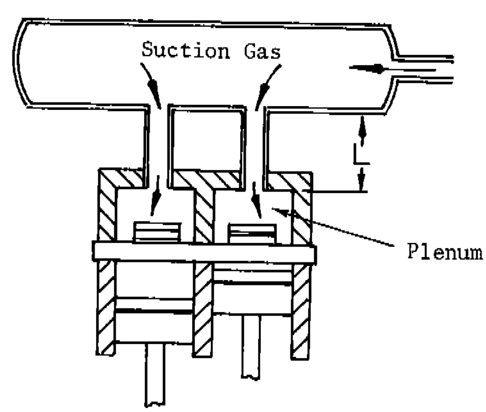

Figure 6: Supercharging Test System

The performance improvement obtained with this technique was equal to that of the tuned tube. An adverse effect on performance and capacity occurred when "L" was modified to produce the wave pattern shown in Figure 8. Under these circumstances the valve opens at the minimum plenum pressure. The plenum pressure also reopens the suction valve after it closed. These two effects reduced the capacity by 23% and the performance by 12%.

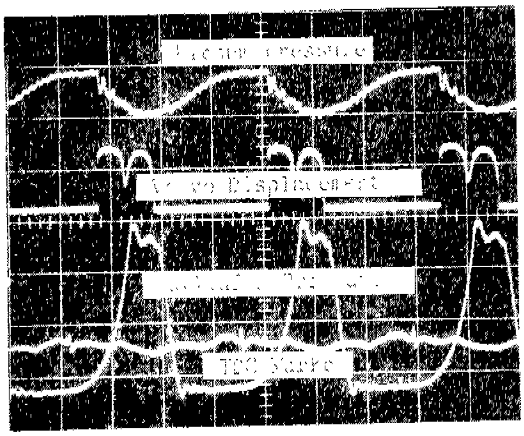

Figure 7: Suction Valve Opens Near Peak Plenum Pressure Pulse (Optimum Performance)

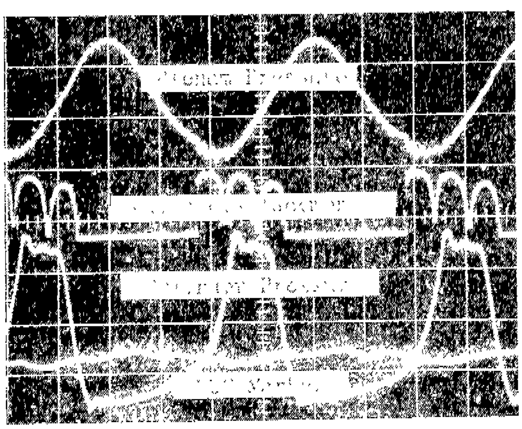

Figure 8: Suction Valve Opens at Minimum Plenum Pressure (Minimum Performance)

COMPRESSION LOSSES

In this discussion compression losses will include losses due to valve porting, valve dynamics, and piston ring blow-by. This section will present experimental techniques for accurately determining these losses.

The most important measurement required is the instantaneous cylinder pressure. This can be accomplished with a strain gage type pressure transducer capable of measuring absolute pressure. For accurate results ($\pm$ 3%) the signal from the transducer must be compensated for the temperature change of the transducer. The temperature compensated signal is transmitted to an analog to digital converter (See Figure 9). The output from the A to D converter can be processed by a mini computer for power and valve losses and results can be plotted on an x-y plotter. A typical p-v diagram of an actual compression process plotted by this technique is shown in Figure 10. The area within the p-v diagram which is above and below the manifold pressures can be calculated to determine the total power loss due to valve dynamics and gas restrictions.

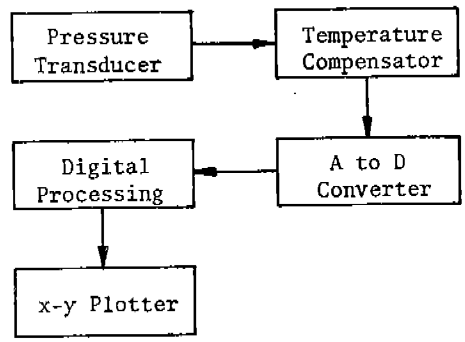

Figure 9: Schematic of the Cylinder Pressure Measurement Technique

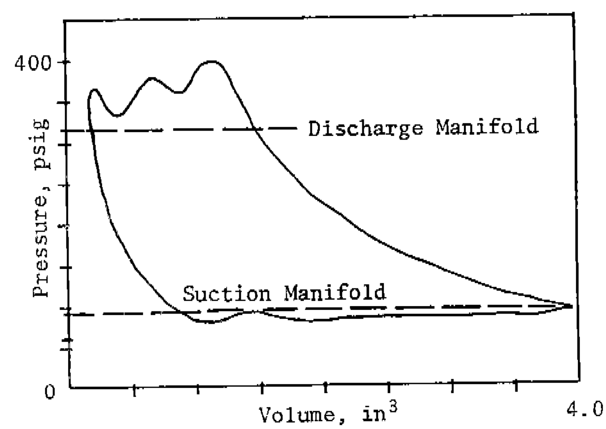

Figure 10: p-v Diagram

Valve displacements for both the suction and discharge valves are required for a complete analysis Valve motion can be measured with an eddy current probe transducer. A piston top dead center location marker will also make results more complete. This can be achieved with a single magnetic pick up which indicates off of the crankshaft.

Valve Port Losses

Compressor performance losses can be reduced by altering either the valve port area and/or the port entrances. An experimental analysis of the port entrance losses will be discussed in this section. All measurements were made before and after the port optimization with strain gage type pressure transducers mounted in a compressor and tested on a calorimeter. Both discharge and suction port entrances were modified from a square entrance to 1/8" radius. The exits of the discharge ports were also counterbored. (See Figure 11)

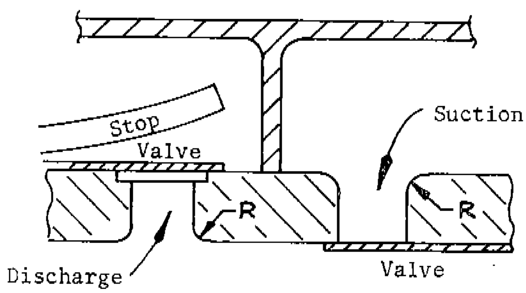

Figure 11: Modification to Valve Ports

The test results show that the cylinder overpressure was reduced from 78 psig to 54 psig and the underpressure from 25 psig to 22 psig. The net effect on performance was a 2-1/2% increase at 45°/130°/65°. A p-v diagram of the change is shown on Figure 12.

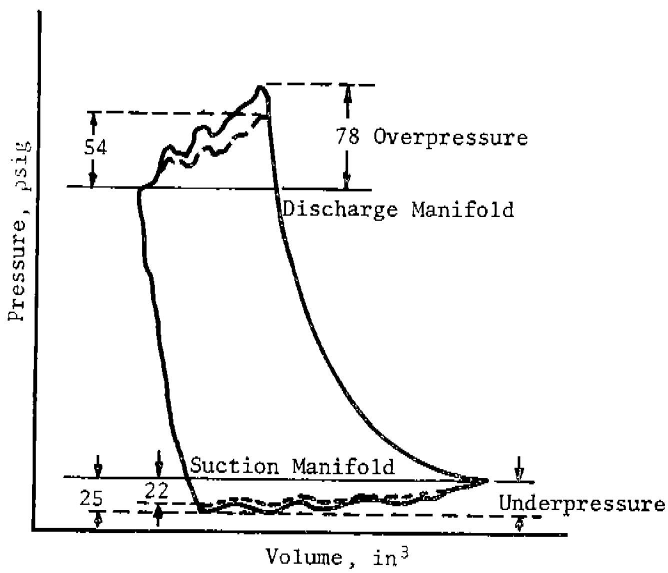

Figure 12: p-v Diagram Showing Valve Port Entrance Effects on Cylinder Pressure

Piston Ring Blow-By

Piston ring leakage reduces compressor performance. Theoretical results indicate that a compressor with a 2.9% leakage rate has a performance loss of 2.5% at 45°/130°/65°. Actual leakage on a reciprocating compressor operating at 45°/130°/65° has been measured between 2 to 4%.

Theory

Theoretically the piston ring leakage can be calculated by assuming an isothermal expansion past the piston ring shown in Figure 13. It can be shown that:

$$\text{Flow rate} = \left[\frac{2g_c(\pi D)^2 h^3 (p_1{}^2 - p_2{}^2)}{FLRT_2}\right]^{1/2} \qquad (1)$$

Where:

D = Diameter of piston
h = Equivalent distance between ring and piston due to the irregular surface
p_1 = Cylinder pressure
p_2 = Crankcase pressure
F = 64/RE Where RE = Reynolds Number
L = Ring width
T = Temperature of gas
R = Gas constant
g_c = Gravitational constant

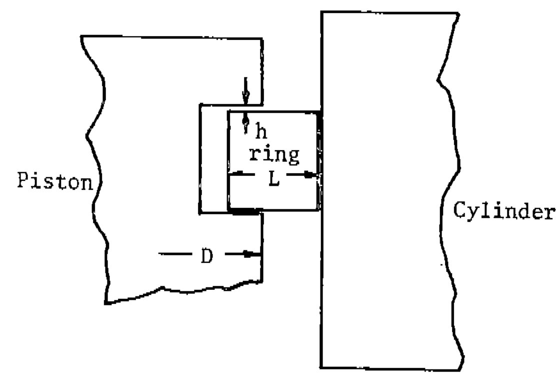

Figure 13: Piston Ring Leakage Parameters

This equation is the flow rate at a given instant during the compression process. To determine the actual leakage rate equation (1) must be evaluated over a complete cycle. The theoretical results for h = 0, .004 and .008 are shown below.

h	% Leak Rate	Capacity	Performance
.000	0.0	31886	8.65
.004	2.9	31116	8.45
.008	4.7	30626	8.31

Notice that when the leakage rate increased from 0 to 2.9% the performance was reduced by 2.5%. Theoretically this indicates that the major portion of the leakage occurs after the gas has been compressed and the discharge valve opened. Therefore implying that the capacity and power can be improved if the leakage rate is reduced.

Experimental Results

Experimentally the leakage rate of a compressor was measured with the following system:

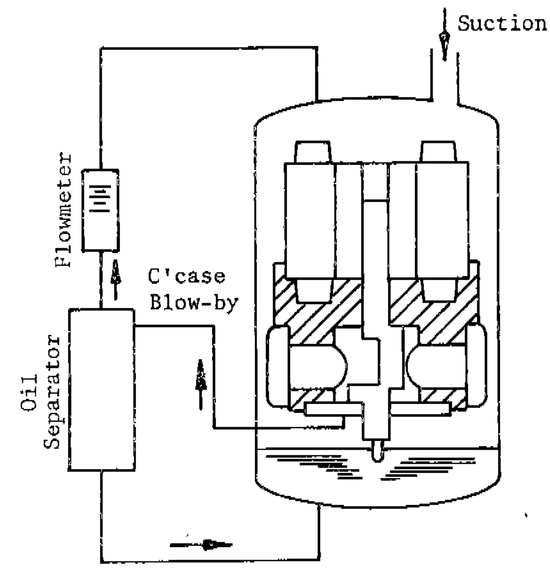

Figure 14: Piston Blow-By Measurement Technique

All blow-by gas leaving the crankcase is allowed to pass through the flowmeter. Leakage rates were found to be 2 to 4% depending on the operating conditions of the compressor. Data has not been taken to substantiate the specific theoretical losses.

Valve Dynamics

This section will present an experimental study involving a ring type suction valve thickness change to illustrate a typical valve dynamics problem. The suction valve thickness was changed from .018 to .025" thick. All of the data from this change was acquired and processed by a computer based digital system. Calorimeter results show that increasing the thickness reduced the capacity by 1.8% and reduced the performance by 2.3% as illustrated in Table 4.

Table 4

Condition 45°/130°/65°	.018 Suction Valve	.025 Suction Valve
Capacity, Btu/Hr	44540	43717
Power, Watts	5360	5381
Performance, Btu/W-Hr	8.31	8.12
Min Cyl Pressure, psig	72	66
Max Cyl Pressure, psig	364	358
*SV Opens, C'angle°	236	238
*SV Closes, C'angle°	40	41
*DV Opens, C'angle°	131	132
*DV Closes, C'angle°	198	203
Suction Valve Lift (% of .018 SV)		40

SV = Suction Valve
DV = Discharge Valve
* 0° = BDC

The major effect of the .025 suction valve is to reduce the minimum cylinder pressure and cause a reduction in both the capacity and performance. Test results indicate that the valve was open for a shorter period and that the effective lift was reduced by 60%. Figure 15 presents the instantaneous valve motion of both the .018 and .025 thick suction valves correlated with a piston top dead center marker.

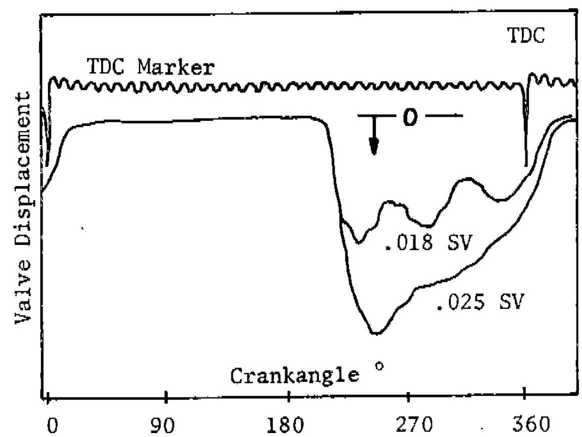

Figure 15: Instantaneous Valve Displacement

CLEARANCE VOLUME

Clearance volume can be optimized to produce a high performing compressor for any given piston and valve plate assembly design. To provide the optimum performance the losses due to re-expansion and trapped gas must be minimized.

Theory

Performance and flow rate are controlled by clearance volume. The flow rate of a compressor is given by:

$$F = RPM \times N \times \left[\frac{SV + CV}{v} \Big|_{BDC} - \frac{CV}{v} \Big|_{TDC} \right]$$

F = Flow rate
N = No. cylinders
SV = Swept volume
CV = Clearance volume
v = Specific volume
RPM = Compressor speed

From this equation it can be seen that as the clearance volume increases the flow rate must decrease. The gas remaining in the cylinder $\frac{CV}{v}\Big|_{TDC}$ reduces the flow rate, releases energy to the crankcase during re-expansion and superheats incoming suction gas. The energy released to the crankcase and the superheating of the suction gas will also reduce the flow rate. It is important to note that as the pressure ratio increases the term $\frac{CV}{v}\Big|_{TDC}$ becomes more significant.

Performance can be optimized by controlling the clearance volume. There are two opposing performance criteria. As the clearance volume is decreased the power loss due to trapped gas increases and the power loss due to superheating decreases. (See Figure 16) The two effects combine to produce an optimun performance.

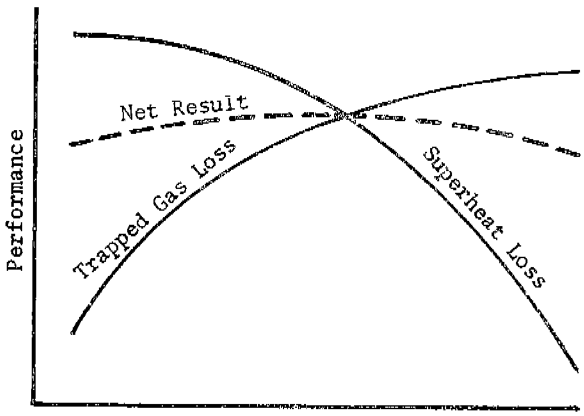

Figure 16: Major Performance Controlling Parameters Affected by CV

Furthermore, the valve motion is also affected by clearance volume. The discharge and suction valve openings are delayed as the clearance volume increases. This does not appear to have a significant effect on performance.

Experimental Data

Experimentally the overall effect of clearance volume can be evaluated by shaving the head of a piston in a reciprocating compressor. The effects of clearance volume on performance and capacity are shown on Figures 17 and 18. It is clear that for this particular compressor operating at 125°F condensing, the optimum performance occurs with a clearance volume of approximately 10%. The minimum operating clearance volume due to tolerance stack up and thermal expansion is approximately 6% for this compressor. However, to minimize trapped gas losses the clearance volume had to be increased to 10%. The requirement for optimum performance therefore reduced the capacity by approximately 8%. (See Figure 18)

To verify that the change in valve motion due to an increase in clearance volume has a negligible effect on performance the cylinder pressures were evaluated at 6 and 8% clearance volume. As shown in table 5, the maximum and minimum cylinder pressures within the cylinders did not change significantly. This indicates that there is no additional power loss due to the delayed valve motion.

Table 5

Effect of Clearance Volume on Valve Motion

Condition: 35°/130°
Refrigerant: R22

Clearance Volume, %	6%	8%
Suction Valve Opens, C'angle°	236	245
Closes, C'angle°	40	43
Discharge Valve Opens, C'angle°	132	141
Closes, C'angle°	204	202
Min Cylinder Pressure, psig	58.5	57.0
Max Cylinder Pressure, psig	367	368

0° = BDC

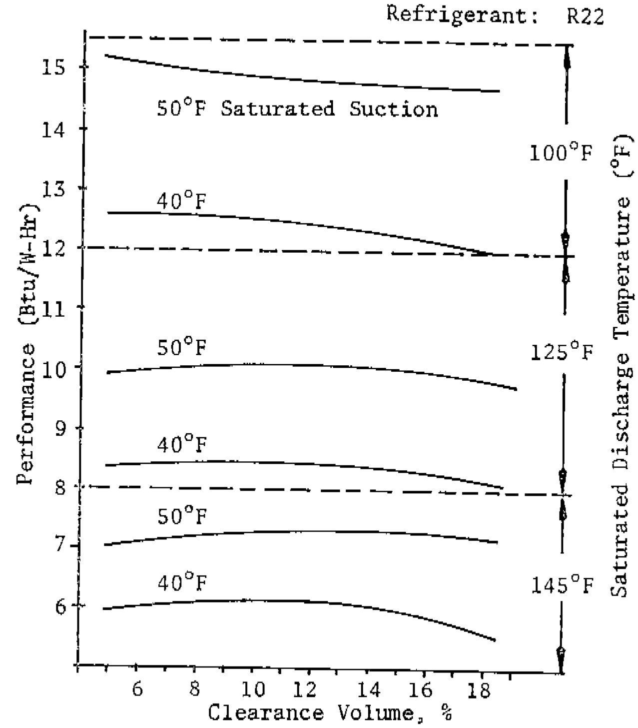

Figure 17: Performance versus Clearance Volume

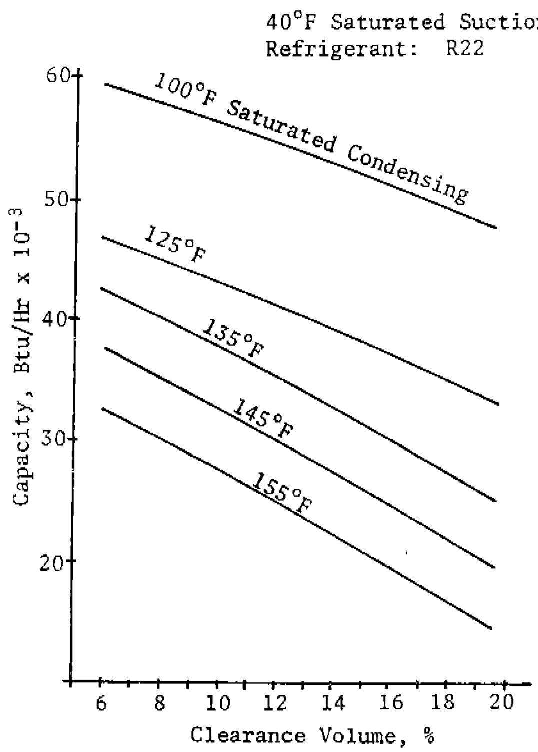

Figure 18: Capacity versus Clearance Volume

CONCLUDING REMARKS

Major performance losses, their relative magnitude and the experimental techniques to evaluate them are summarized in Table 6. Estimated motor and friction losses are included to make the summary complete. The losses described in Table 6 are typical examples and will vary with each design. This chart should be used for comparison purposes in evaluating the losses for full hermetic, two pole, reciprocating compressors.

Table 6

Typical Compressor Losses

Refrigerant: R22
Conditions: 45°/130°/65°
RPM: 3450

Category	Perf. Btu/W-Hr	% Loss	Measurement Technique
Theoretical Maximum	15.2		
Motor Efficiency	12.5	18	
Superheat @ 140°F	10.7	12	Thermocouples located at the suction port
Suction Manifold @ 1.5 psi Drop	10.5	1.3	Pressure transducers
Underpressure	9.8	4	Pressure transducers
Overpressure	8.9	6	Pressure transducers
Clearance Volume @ (8% CV)	8.8	.6	Calorimeter
Leakage (2%)	8.7	.6	Flowmeter
Discharge Manifold @ 4 psi Drop	8.6	.6	Pressure transducers
Friction (Estimated)	8.4	1.3	
TOTAL	8.4	44.4	