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High Efficiency Refrigerator Freezer Reciprocating Compressors

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HIGH EFFICIENCY REFRIGERATOR FREEZER RECIPROCATING COMPRESSORS

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ABSTRACT

The question of how to design a higher efficiency reciprocating refrigerator freezer compressor is addressed. This question is examined in light of the advent of alternate refrigerants and alternate lubricating oils. Some analytical results and also some experimental results are presented.

The relationship between alternate refrigerants, oil viscosity and compressor geometry is examined from an analytical point of view and some experimental results are presented.

INTRODUCTION

The question of how to build a more efficient refrigerator freezer compressor has become more intense recently. The question has become somewhat more complex as a result of the advent of alternate refrigerants and alternate lubricants. There is still some question as to what the refrigerant and lubricant of the future will be. The physical properties of these alternate refrigerants and lubricants are continually being re-measured and revised.

This paper examines the question of compressor efficiency. It examines efficiency in light of the differing physical properties of three different refrigerants, the presently used R12 and two leading candidates of the future, R134a and R152a.

The paper also presents the results of an objective study, both theoretical and experimental, of bearing friction power loss as a function of oil viscosity and as a function of some variations in bearing geometry.

Although more subjective than objective the paper briefly examines the effect that such things as pressure drop through valves and residual gas re-expansion and suction gas heating have on efficiency.

HIGH EFFICIENCY RECIPROCATING COMPRESSORS

The question of compressor efficiency has meaning only if the operating or test conditions are specified, but these test conditions are a moving target. Homes have changed over the past twenty years. As a collective average they are cooler in summer than they once were. Ten years ago we had a standard ARI rating point and there was not a lot of emphasis on testing or rating a refrigerator freezer compressor at anything other than this rating point. We still have this ARI rating point and it is still valuable as one common point of comparison, but there are other important test points. As an example one compressor may be best when tested at 130 degrees Fahrenheit condensing temperature, the ARI rating point, but another one may be better when tested at 110 degrees, and 110 degrees may be more significant to the refrigerator freezer designer than 130 degrees.

Table 1 (based on NIST data available February 1992) presents some ideal performance calculations of three different refrigerants R12, R134a and R152a when operating over a range of conditions. An expanded scale of the part of this data that pertains to the normal operating range of the refrigerator freezer compressor, including the ARI rating point, is presented in Figure 1.

Cond Temp	Return Gas Temp	Ideal COP (100% Isentropic Comp Efficiency) R12	R134a	R152a
90 Deg F	-10 Deg F	3.502	3.435	3.629
"	0	3.511	3.438	3.603
"	30	3.518	3.519	3.608
"	60	3.567	3.561	3.594
"	90	3.610	3.606	3.613
110	-10	2.969	2.967	3.101
"	0	3.003	2.968	3.084
"	30	3.019	3.023	3.082
"	60	3.055	3.050	3.066
"	90	3.086	3.093	3.084
130	-10	2.646	2.623	2.725
"	0	2.661	2.622	2.711
"	30	2.657	2.670	2.706
"	60	2.696	2.696	2.701
"	90	2.719	2.735	2.709

-10 Deg Evap Temp, 90 Deg Liquid Temp
(results based on 1992 NIST data.)

Table 1
Ideal COP

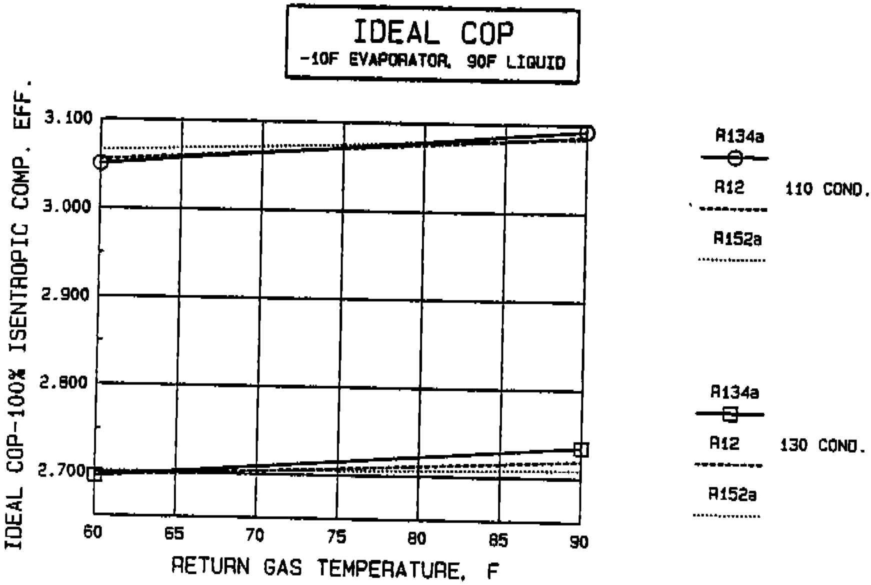


Figure 1

These ideal performance calculations are useful as a baseline for comparison but they do not explain the actual test results. Figure 1 taken by itself would suggest that there is no significant difference in the three refrigerants as far as EER is concerned but this is contrary to actual test results.

Actual test results do indicate that there is relatively little difference between R12 and R134a but this is not true in the case of R152a. Test results suggest that R152a results in from about 2 to 4 percent lower EER than either R12 or R134a. For a relative comparison the ideal results assumed an isentropic compression efficiency of 100%. The relative comparison is equally valid if a more typical isentropic compression efficiency of 59% is assumed, or if any isentropic compression efficiency is assumed provided it is the same for each refrigerant.

But the isentropic compression efficiency is not the same for each refrigerant. Table 2 lists some parameters that are useful in comparing the refrigerants. For relative comparison a 750 BTU/hr compressor operating at the ARI test point (-10 degrees Fahrenheit evaporator temperature, 130 degrees Fahrenheit condensing temperature, 90 degrees Fahrenheit liquid temperature and 90 degrees Fahrenheit return gas temperature) is assumed.

	R12	R134a	R152a
v -lb/hr	12.149	9.326	5.93
ρ -lb/ft cu	0.4022	0.2948	0.1731
V -ft cu/hr	30.208	31.63	34.26
cyl vol -ideal - in cu	0.2485	0.2603	0.2819
cyl vol - 70% vol eff	0.355	0.3718	0.4027
disch press - psia	195.6	213.7	190.8
suction pressure - psia	19.14	16.7	15.19
differential pressure - psi	176.5	197.0	175.6
piston dia - in *	0.9667	0.9819	1.008
piston area -in sq.	0.734	0.757	0.7986
crank peak load - lb	130.1	149.9	140.9
compression ratio	10.23	12.80	12.56

*Assumes a bore to stroke ratio of 2 to 1 and a connecting rod to crank throw length of 6 to 1.

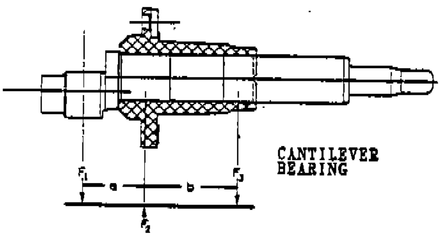
Table 2
Performance Parameters

From Table 2 we observe the compression ratio is the lowest for R12 and the highest for R134a. This factor, taken by itself, would suggest that R12 is a little better than the other two refrigerants but there are other things that also contribute to efficiency.

The peak bearing load is highest for R134a and lowest for R12. In equally optimized machines the bearing friction power loss would be less with R12 than it would be with either of the other two refrigerants.

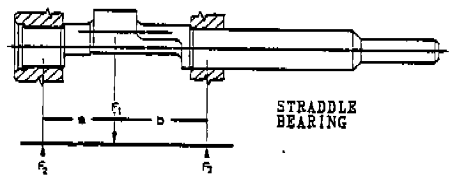
The required cylinder volume for a compressor operating with R152a is 13 percent greater than it is for one operating with R12 and 8 percent greater than it is for one operating with R134a. This results in a larger piston being required hence higher friction and also a large volume flow into the cylinder thus there is more pressure drop through the suction valve port. Possibly this reduced capacity, per a given cylinder volume, is the indirect cause of the lower efficiency actually measured in the compressor operating with R152a.

Figures 2 and 3 illustrates schematically two different bearing configurations used in refrigerator freezer compressors.



TOTAL MAIN BEARING LOAD = 3 X CRANK LOAD

Figure 2



TOTAL MAIN BEARING LOAD = CRANK LOAD

Figure 3

The compressor main bearing system illustrated in Figure 2 is often referred to as a single main bearing but more accurately it would be referred to as a cantilever bearing pair. The bearing pressure distribution curve for this type of bearing is complex but it is roughly as is illustrated in the figure. As a result of the cantilever effect F_2 is roughly equal to twice the crank bearing load F_1, and F_3 is roughly equal to the crank bearing load F_1. This is approximately equivalent to the main bearing supporting three times the crank bearing load and this results in high bearing friction power loss.

Figure 3 illustrates schematically a split or straddle bearing configuration. Each half of the main bearing supports half of the crank bearing load. The total equivalent load of the main bearings in the straddle bearing compressor is equal to the crank bearing load, not three times the crank bearing load as it is in the cantilever bearing compressor and this has a significant effect on the total compressor bearing friction power loss and in turn on compressor efficiency.

Table 3 presents calculated bearing friction power loss for the two different bearing configurations for different oil viscosities. Table 4 extrapolates these results to the compressor EER.

Oil Viscosity	Total Machine Bearing Friction Power Loss	
	Cantilever Bearing	Straddle Bearing
4 Cs at 100°C	23.3 watts	14.46 watts
3	20.2	12.59
2	14.7	9.16
1	7.5	4.67
0	0.0	0.0

Table 3
Bearing Friction Power Loss

Oil Viscosity	EER	
	Cantilever Bearing	Straddle Bearing
4 Cs at 100°C	5.35*	5.66
3	5.45	5.73
2	5.65	5.86
1	5.93	6.05
0	6.26	6.26

*Reference for extrapolation to other bearing configurations assumes: 856 BTU/hr, 160 W total, 23.3 watt bearing friction power, 4 CS at 100°C, Cantilever Bearing

Table 4
Effect of Oil Viscosity and Bearing Geometry on Compressor EER

A simplified theory for calculating bearing friction power loss suggest that the more the oil viscosity is decreased the greater the efficiency becomes. This is what actually happens but there are limits.

Lower oil viscosity results in significantly less bearing friction power loss which translates into higher efficiency. Both an analytical analysis and experiments confirm this, but there are some limits as to how low the viscosity can go. If the viscosity is too low, an oil film sufficiently thick to protect the bearings is not developed and a very rapid bearing wear rate results. There are other effects also. If the oil viscosity is too low the piston to cylinder oil seal may not be fully developed and there may be gas blow-by. There is not a linear relationship between oil viscosity and gas blow-by. If the viscosity is above a minimum threshold level there is almost no gas blow-by but after the viscosity is reduced below this threshold the blow-by becomes very great.

There is a similar situation in regard to the valves. If the viscosity is above a certain threshold there may be almost no leakage by the valves. If the viscosity is below this threshold the valve leakage, and valve noise, and valve wear rate may become very great.

Figure 4 presents some results of actual efficiency measurements of two refrigerator freezer compressors tested at the standard ARI rating point.

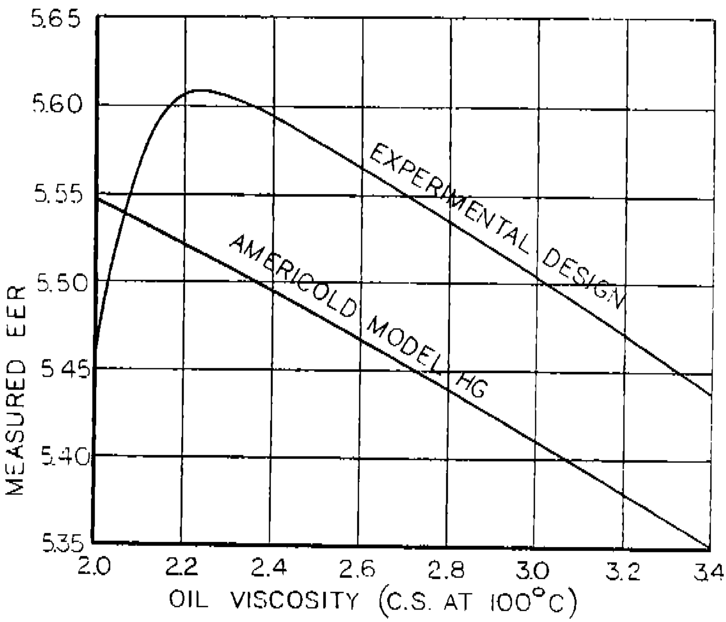


Figure 4
Measured Compressor Efficiency

Apparently even with an oil viscosity as low as 2 centistokes at 100 degrees centigrade the threshold for one of the compressors was not reached but for the other it was. From the test results it is not apparent if the problem was gas blow-by or if it was valve leakage. The absence of a significant increase in sound level would lead to the conclusion that the problem was gas blow-by.

Cylinder diametrical clearances are normally about 0.0003 inches to 0.0005 inches but dimensions of this magnitude are difficult to control and to measure precisely. It is probable that the compressor represented by the curve labeled as "experimental design" had a slightly greater piston to bore clearance than the other compressor. In any event the test results would suggest that it may not be practical to design a compressor to operate with an oil with a viscosity lower than about 2.2 centistokes at 100 degrees centigrade.

An isentropic compression process is normally used as a basis for comparison when dealing with refrigerator freezer compressors. An isothermal compression process is often used as a basis for comparison in dealing with air compressors. An ideal isothermal compression process always takes less power than an ideal isentropic compression process. As a bench mark for comparison, isentropic compression efficiency is a good yardstick for relative comparison but this is a bench mark for comparison only and not an absolute limit. Isentropic compression efficiencies of about 59% are typical in the refrigerator freezer compressor.

Further it should be noted that the ideal COP of an actual refrigeration system depends on the heat of vaporization of the refrigerant and the slope of the constant entropy lines. This does not relate directly to the classical Carnot COP, but once again, the Carnot COP is often used as a yardstick for relative comparison. Modern compressors have a COP of about 50% of the ideal Carnot COP.

There are things in addition to bearing failure that are known to have, a significant effect on compressor efficiency. Some of these things include the following:

There is always some residual gas remaining between the piston and the discharge valve when the piston is at top dead center. Work was done on this residual gas to compress it. When the suction stroke begins some of this energy that was stored in this residual gas is returned to the piston, but there is some loss associated with compression and re-expansion. There is a hysteresis loop, all of the energy is not returned to the piston. It is difficult to measure precisely how much is lost by compression and re-expansion. It is known that reducing the residual volume increases efficiency. It is estimated that in the case of the approximately 10 to 1 compression ratio refrigerator freezer compressor about 2/3 of the energy that we put into compressing this residual gas is recovered and that 1/3 of it is lost.

There are practical limits as to how much the residual volume can be decreased. If the piston comes extremely close to the valve plate then there is a significant restriction to the gas as it flows from the outer periphery of the piston surface to the discharge port. Different piston geometry and different discharge valves can be useful in reducing this residual gas volume and its flow path. This will result in improved compressor efficiency.

It is difficult to separate the effect of suction gas heating from other things that happen in the compressor but experience has shown that suction gas heating is harmful. The constant entropy lines slope more as we move farther to the right, toward higher temperatures, on our PH diagram and both theory and experiment suggest that this reduces compressor efficiency.

Ideally there should be no pressure drop through the valve. It is obvious that if the peak cylinder pressure is greater than the discharge pressure or the minimum cylinder pressure lower than the suction pressure the compressor does more work than would be required with ideal valves. A large bore to stroke ratio allows room for larger valves and this is good. But a large bore to stroke ratio results in increased residual gas volumes and increased bearing loads, both of which are not good. Improved valve types and improved head configurations can allow for a more effective use of space thus permitting a smaller bore to stroke ratio to be used thus reducing residual gas re-expansion and bearing loads and hence higher efficiency.

Summary

In summary it is known that there are things that can be done to increase compressor efficiency. Much of what is done will be trial and error experiment but to examine the problems analytically can shorten the experimental procedures.

Up to a certain threshold, decreasing the oil viscosity significantly improves the efficiency but a lot of testing is required before a safe lower limit is established.

From an efficiency point of view a straddle bearing arrangement is better than the more commonly used cantilever bearing.

Reducing residual gas volume, pressure drop through the valves and/or suction gas heating will increase compressor efficiency. Considerable experimentation will be required in order to determine how far we can go.