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An Experimental Investigation Into Heat Transfer to the Suction Gas in a Low-Side Hermetic Refrigeration Compressor

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AN EXPERIMENTAL INVESTIGATION INTO HEAT TRANSFER TO THE SUCTION GAS IN A LOW-SIDE HERMETIC REFRIGERATION COMPRESSOR

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ABSTRACT

Heat transfer to the suction gas in a hermetic compressor is known to have adverse effects on compressor performance parameters. This can be explained by thermodynamic cycle analysis. The extent of suction gas heating inside a compressor and the areas in which this heating occurs were determined experimentally in the present investigation.

A commercial, 1/3 horsepower, low-side, reciprocating, hermetic compressor was instrumented internally with thermocouples and pressure transducers. Freon flow rates and power consumption were also measured. The measurements were used to determine the magnitude of suction gas heating at various stages between the suction inlet and the suction manifold.

Individual contributions due to gas circulation inside the compressor shell, heat transfer to the gas in the suction muffler, and heat transfer to the gas in the suction plenum were measured. Quantitative results describing the temperature increases in the suction passage are presented.

In addition, the effect of suction passage configuration on suction gas heating was investigated. The different configurations included misalignment of the suction inlet and muffler inlet, and baffling to restrict the gas circulation inside the compressor shell. Data for five different suction passage configurations are presented and discussed.

Finally, the effect of suction gas temperature on specific power consumption for the various configurations is presented.

INTRODUCTION

This paper deals with the operating efficiency of small, hermetically sealed, low side refrigeration compressors. More particularly, it concerns the design of the refrigerant flow passages between the suction line leaving the evaporator and the intake manifold to the compression cylinder.

The efficiency of small refrigeration compressors is of great concern since they are manufactured and produced by the tens of millions each year. Thus, the savings of just a few cents per compressor per year means an overall savings of millions of dollars per year in energy costs.

Any increase in suction gas temperature between the outlet of the evaporator and the intake to the compressor cylinder can be considered as wasted cooling capacity which reduces cycle efficiency. Theoretically, a simple, well insulated, direct connection between the suction line and the intake manifold would be the most efficient design. However, other considerations make the direct connection undesirable. Those considerations include noise propagation, the need to separate the compressor oil from the refrigerant gas, and slugging problems associated with start-up. In addition, the suction gas is used to cool the electric motor and compressor in some designs.

One popular design technique that is the base line for this study is to dump the suction gas into the compressor shell. The gas is then drawn into an acoustic muffler, through flow passages, and into the intake manifold. This design has the desirable characteristic of damping the compressor noise. The design also provides an easy solution to the liquid-gas separation need, and to the slugging problems.

For this design only a part of the suction gas entering the shell goes directly into the muffler. The remainder is "spilled," and circulates inside the shell, and is superheated by the relatively hot motor and compressor. Part of the superheated gas is drawn into the muffler on each intake stroke of the compressor, and is mixed with the unspilled flow. Theoretical estimates show that for the small compressor considered here, a 10F increase in suction gas temperature results in approximately a one percent decrease in compressor efficiency.

This paper is the result of an experimental investigation that was undertaken to study the heat transfer to the suction gas at various stages inside the compressor shell. A compressor was installed in an experimental test stand which simulates a typical refrigeration cycle. The compressor was instrumented with thermocouples to measure suction gas temperature at strategic points in the gas passage. Thermocouples were also used to measure the temperature of the compressor's internal components.

Pressure transducers measured the suction and discharge pressures. Freon flow rates and power consumption were also measured. The measurements were used to determine the magnitude of suction gas heating at various stages between the suction inlet and the suction manifold.

Individual contributions due to gas circulation inside the compressor shell, heat transfer to the gas in the suction muffler, and heat transfer to the gas in the suction plenum were measured. Measurements were made for five different compressor steady-state operating configurations. These configurations enable the identification of major heat transfer sources inside the compressor and their relative magnitudes. The details of the study are contained in Ref. 1.

A companion study (see Ref. 2 and 3) used flow visualization to investigate the flow pattern of the suction gas in transit between the suction inlet and the muffler inlet. The flow pattern was recorded on video tape and compared for the following cases:

1. The muffler inlet in alignment with the suction inlet,

2. The muffler inlet misaligned relative to the suction inlet,

3. Different offset between the aligned muffler inlet and suction inlet,

4. The use of a shroud around the muffler to partially baffle the flow in the vicinity of the muffler, and

5. The use of a larger volume muffler instead of the standard muffler, with an increased inlet hole diameter and increased outlet passage diameter.

Reference 2 describes the flow-visualization study in detail, and also contains the video tape produced as part of that study.

To quantify the effects of misalignment of the suction gas inlet and the muffler inlet, which were indicated by the flow-visualization study, the operating temperatures were measured for several misalignments and with the use of a shroud to baffle the flow. From the measurements a flow mixing parameter, δ , is determined for the various geometries and flow conditions considered. δ is the fraction of unspilled flow so that a value of $\delta = 1$ represents no flow spillage (a direct connect). Measurements indicate that $\delta = 0.5$ is typical of off the shelf compressors, and that values of 0.8 to 0.9 could be achieved by some design improvements.

As an extension of the experimental program an analytical heat transfer model was developed which utilizes energy balance on the internal components of the compressor to predict both component and fluid temperatures. The analytical model is described in a companion paper, Ref. 4. The model is used to investigate the effects of gas circulation inside the compressor shell and also heat transfer to the gas in the muffler. The experimental results were used to determine heat transfer coefficients and compressor volumetric efficiency for use in the model.

EXPERIMENTAL SYSTEM DESCRIPTION

Figure 1 is a schematic of the compressor test stand. The lines in the system are 1/4 in. copper tubing with flare fittings and connections. The evaporator and condenser are shell-and-tube heat exchangers. The evaporator is 18 in. long and consists of 1/4 in. copper tubing inside 3/4 in. copper tubing. The flow is concurrent with hot tap water at about 110° F as the heating fluid. The condenser is counter-flow with cold tap water at about 70° F as the cooling fluid.

The two 1/4 in. valves V1 and V3 control the flow of refrigerant. Valves V6 and V7 isolate the system from the external Freon-12 storage tank and are used to charge and/or discharge the system. Valve V2 is the expansion valve. The flow of heating and cooling water is controlled by valves V4 and V5. Suction and discharge pressures are monitored using 150 and 500 Psi Heise gauges, respectively. These gauges serve as visual references to establish the desired operating conditions. Calibrated transducers (not shown in Fig. 1) are used to enable recording of the suction and discharge pressures.

Mass flow rate of refrigerant in the system is measured using a calibrated Micro Motion Model D6 mass flow meter (not shown in Fig. 1) in the discharge line from the compressor. Input power to the compressor is monitored using a Scientific Columbus dilogic watt transducer.

Temperature measurements were made using alumel-chromel thermocouples. The measurements at the ten stations numbered 100 thru 109, shown in Fig. 1, were used to monitor the overall system performance. Nine other temperature measurements were made. They are the suction gas temperature (007), discharge temperature (008), compressor shell temperature (not shown in Fig. 1), and six measurements inside the compressor shell. Figure 2 is a schematic of the suction muffler and the suction and discharge plenums for the compressor, and shows the location of four of the internal temperature measurements. The two additional internal temperature measurements were of the motor surface and of the gas at a representative point in the gap between the motor and the shell.

The main component for these tests is an off-the-shelf, 1/3 hp, reciprocating, hermetic compressor which was modified to permit access to the interior. A standard compressor was cut in half, a 1/2 in. of the shell was removed from each half, and flanges were welded to each half of the shell. A one inch aluminum ring was bolted between the flanges.

The normal test procedure consisted of assembling and evacuating the system (30 minutes on the vacuum pump), charging the system to about 35 psig with Freon-12, operating the system for about 2 hours to achieve steady-state operation, making the fine adjustments to the desired operating point, and recording the 30 measurements at one minute intervals for a 30 minute period. An Acurex Autodata 10/5 datalogger and Zenith P.C. are used to acquire, record and store the data. Data reduction and plot preparation is done on the VAX 11/780.

Tests were run for five different compressor configurations:

1.) Normal Configuration
2.) Offset Configuration
3.) Opposite Side Configuration
4.) Normal Configuration with the Shroud
5.) Offset Configuration with the Shroud

The normal configuration represents a typical compressor as it is manufactured. In this case, the suction line and the muffler inlet are aligned with a gap of about 1/4 in. between them. In the offset configuration the suction line and the muffler inlet are offset horizontally by about 1/4 in. The opposite side configuration used a suction line at the back side of the shell so that none of the gas entered the muffler directly from the suction line. In the final two configurations a rubber shroud was used to form an enclosure around the muffler which limited the gas flow around the top and sides of the muffler. The shroud also covered the back of the muffler to help insulate it from the adjacent, hot, electric motor. The gap at the bottom of the muffler between the muffler and compressor shell was left open.

For each of the five configurations data were recorded at four different suction line temperatures ranging from 60° F to 90° F. Thus, a total of twenty independent sets of data were recorded. Each test was repeated three times so that overall sixty sets of steady-state data were obtained. For all tests, the suction and discharge pressures were maintained as near as possible to 4.9 psig and 177 psig respectively.

EXPERIMENTAL RESULTS

Table 1 is a summary of the data for test 1-a, the normal configuration, and is representative of the measurements for each of the five configurations tested. The values tabulated are averages of 30 measurements over a half-hour steady-state run. The standard deviations for each measurement are an indication of the experimental uncertainty in the measurement. Four sets of data are shown at four different suction line temperatures. Each set of runs (configuration) was repeated three times at nearly the same suction line temperatures.

The bottom four values in each column in Table 1 are the temperature rise values in degrees F between various points on the flow path. The temperature rise between the suction line and the muffler inlet is 35 to 40° F and is a direct result of flow spillage and mixing of the cold inlet gas with the hot gas inside the compressor shell. In the normal configuration that temperature rise accounts for a little over half of the overall temperature rise between the suction line inlet and the suction plenum. To characterize the flow circulation and mixing inside the shell a mixing parameter, δ, is defined as the fraction of inlet gas that enters the muffler without mixing. A value of $\delta = 1$ represents a direct connect (no mixing) between the inlet and the muffler, while a value of $\delta = 0$ represents complete mixing. Thus,

$$\dot{m}_{tot} = \dot{m}_{sp} + \dot{m}_{dir} \qquad (1)$$

$$\delta = \frac{\dot{m}_{dir}}{\dot{m}_{tot}} = 1 - \frac{\dot{m}_{sp}}{\dot{m}_{tot}} \qquad (2)$$

If it is assumed that the spilled portion of the gas, $\dot{m}_{sp}$, is heated to the shell gas temperature before it enters the muffler, that the specific heat of the gas does not vary over the temperature range, and that the temperature measurement at the muffler inlet is representative of a mixture of the direct and spilled gas, then from an energy and mass flow balance on the gas entering the muffler one obtains,

$$\delta = \frac{T_{circ} - T_{muf}}{T_{circ} - T_{suc}} \qquad (3)$$

where T_{circ} is the temperature of the gas circulating inside the compressor shell, T_{muf} is the temperature of the gas mixture entering the muffler, and T_{suc} is the temperature of the suction gas entering the compressor shell.

Table 2 tabulates the mixing parameter, δ, for each of the twelve runs for each of the five flow configurations. Average values for each configuration are tabulated along with the standard deviation. Except for one series of four runs on the opposite configuration, the measurements are consistent and show the dramatic effect of both inlet misalignment which increases the mixing and the use of a shroud to reduce the flow circulation and mixing. δ appears to be a weak function of the inlet temperature.

Figure 3 is a plot of the muffler inlet temperature for the various suction line temperatures for the five configurations tested.

Based on these results alone one could conclude that:

a.) There is about 50% mixing of the inlet gas with the shell gas in the normal configuration. This represents a 35 to 40° F temperature increase and a 3.5 to 4% efficiency loss.

b.) A slight misalignment of the inlet line and muffler inlet results in an additional efficiency loss, estimated at 2 to 3%.

c.) A very simple shroud around the muffler can reduce the mixing and temperature rise to less than half that of the normal configuration. The shroud also compensates for slight misalignment of the inlet line and muffler inlet.

Unfortunately one or more of Murphy's laws comes into play here and this potential improvement from the shroud is mostly offset by higher heat transfer rates in the intake plenum. Figure 4 is a plot of the overall temperature rise between the suction line and the suction plenum verses suction line temperature for the five configurations tested. These results indicate that there are indeed large losses associated with introduction the suction gas opposite to the muffler inlet, and substantial losses associated with misalignment of the inlet line and muffler inlet. However, the hoped for improvement from using a shroud is largely canceled by a much higher temperature rise in the intake plenum. These relationships are illustrated in Fig. 5 which is a summary of the various contributions to the overall temperature rise for the five configurations tested.

The measured power consumption did not show any discernible difference between the various tests. Unfortunately the measurement uncertainty is so large that the measured power consumption cannot be used to determine the 1% to 2% efficiency differences that are of interest here.

The measured mass flow rate is plotted verses the measured gas temperature in the suction plenum in Fig. 6. The solid lines on the figure are at a theoretical slope of -0.0254. The points above 180 F are for the opposite side configuration. Most of the points between 160 and 180 F are for the misaligned configuration. Although there is a great deal of scatter in the data the trend is clear. The lower the suction plenum temperature the higher the mass flow rate. For a constant power consumption, the theoretical slope of -0.0254 represents an efficiency increase of about 1.5% for each 10 degrees F reduction in the suction gas plenum temperature.

CONCLUSIONS AND RECOMMENDATIONS

1. Murphy's laws work in compressors. That is, any simple change that should yield improved performance will be offset by factors that are very difficult to control.

2. A simple shroud around the muffler to reduce flow circulation, for the type of compressors considered here, is recommended.

3. Methods of reducing the heat transfer to the suction gas in the suction plenum need to be studied in more detail.

ACKNOWLEDGMENT

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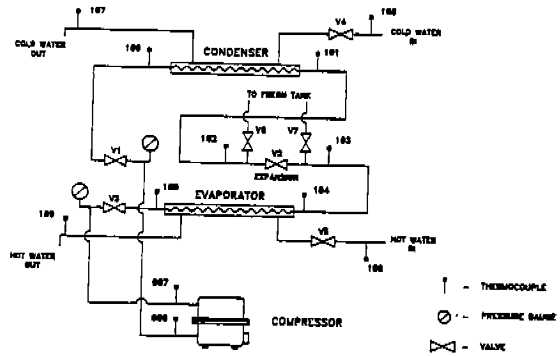


Figure 1: Schematic of Compressor Test Stand

Ambient Temperature (F)	77.00
Ambient Pressure (psia)	14.50

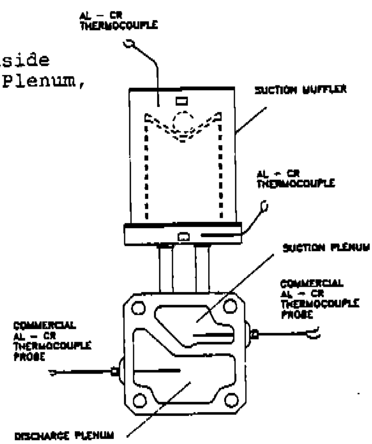
	Steady State	Std. Dev.	Steady State	Std. Dev.	Steady State	Std. Dev.	Steady State	Std. Dev.
Power (watts)	256.56	2.20	253.14	1.29	254.85	0.93	254.09	1.34
Muffler In (F)	114.75	0.73	117.64	0.75	120.18	0.74	122.76	0.77
Muffler Out (F)	126.89	1.29	129.59	0.77	132.05	1.26	132.99	0.38
Suction Plenum (F)	149.38	0.48	153.28	0.44	155.54	0.29	157.39	0.31
Motor Surface (F)	191.32	0.53	192.09	0.34	192.48	0.24	193.91	0.22
Gas in Shell (F)	156.18	0.49	156.41	0.52	156.89	0.40	158.22	0.38
Discharge Plenum (F)	281.85	0.53	284.15	0.49	285.94	0.30	287.46	0.34
Discharge Line (F)	191.92	0.54	191.41	0.49	192.71	0.24	193.84	0.24
Top of Shell (F)	120.00	0.71	120.31	0.48	120.67	0.55	121.49	0.36
Suction Pressure (psig)	5.13	0.23	4.95	0.11	4.98	0.08	4.89	0.14
Discharge Press. (psig)	173.15	0.55	177.15	0.57	177.47	0.21	175.92	0.65
Mass Flow Rate (lbm/hr)	17.29	0.26	16.76	0.13	16.76	0.13	16.76	0.13
Suction Line (F)	74.21	0.39	79.34	0.41	84.47	0.31	87.80	0.22
Temperature Rise Between Suction Line and Muffler Inlet (F)	40.54		38.30		35.71		34.96	
Temperature Rise in Muffler (F)	12.14		11.35		11.87		10.23	
Temperature Rise Between Muffler Outlet and Suction Plenum (F)	22.49		23.69		23.49		24.40	
Overall Temperature Rise (F)	75.17		73.94		71.07		69.59	

Table 1: Steady-State Data from Test 1-A (Normal Configuration)

	NORMAL		OFFSET		OPPOSITE		SHROUD-Aligned		SHROUD-Offset	
	Ts	Delta	Ts	Delta	Ts	Delta	Ts	Delta	Ts	Delta
	74.21	0.512	71.50	0.242	67.05	0.000	69.07	0.783	63.59	0.739
	79.34	0.507	76.20	0.245	72.72	0.000	74.36	0.791	71.29	0.755
	84.47	0.513	83.86	0.253	79.07	0.000	84.90	0.819	82.18	0.774
	87.80	0.509	88.92	0.256	85.68	0.000	90.13	0.822	90.29	0.785
	67.16	0.505	70.05	0.253	64.90	0.166	66.47	0.780	64.25	0.745
	70.30	0.516	74.42	0.254	71.81	0.229	75.25	0.796	73.16	0.760
	75.94	0.511	79.14	0.255	80.32	0.211	83.60	0.809	80.10	0.773
	82.36	0.507	83.56	0.259	90.38	0.242	89.57	0.822	89.30	0.790
	74.62	0.509	69.32	0.243	75.57	0.000	64.33	0.782	62.71	0.749
	80.25	0.511	75.07	0.248	80.29	0.000	72.43	0.792	71.21	0.762
	85.06	0.511	79.57	0.246	84.90	0.000	80.44	0.807	80.10	0.777
	89.19	0.509	84.31	0.249	90.98	0.000	88.32	0.822	88.21	0.796
Avg.		0.510		0.250		0.071		0.802		0.767
Std. Dev.		0.003		0.005		0.101		0.016		0.018

Table 2: Values of Delta Parameter for Various Test Configurations

Figure 2: Thermocouple Locations Inside Suction Muffler, Suction Plenum, and Discharge Plenum



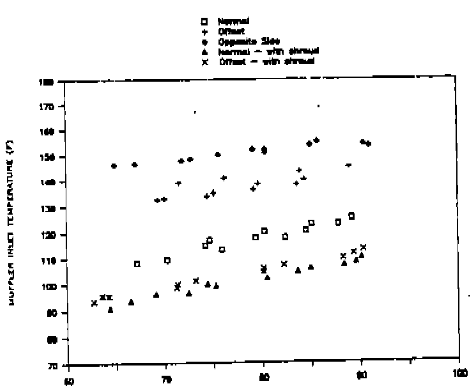


Figure 3: Muffler Inlet Temperature vs. Suction Line Temperature for the Five Tested Configurations

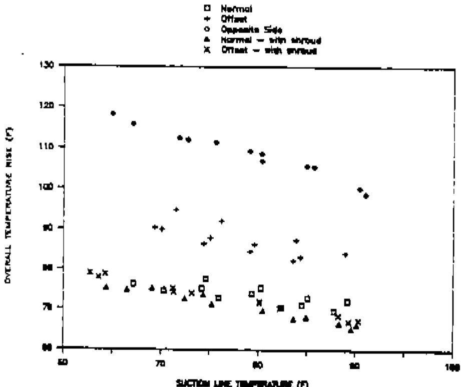


Figure 4: Overall Temperature Rise vs. Suction Line Temperature for the Five Tested Configurations

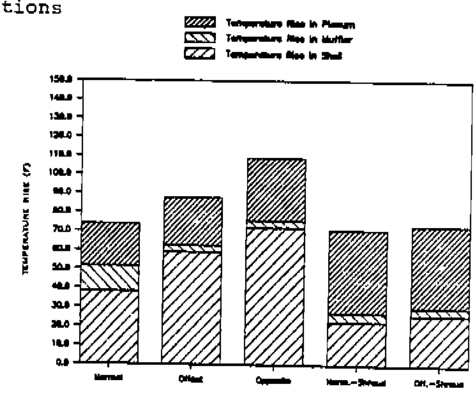


Figure 5: Summary of Contributions to Overall Temperature Rise for Five Tested Configurations

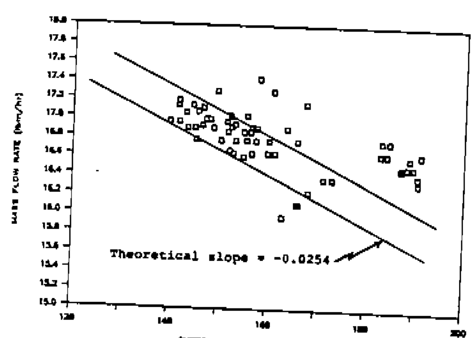


Figure 6: Mass Flow Rate vs. Suction Plenum Gas Temperature (Experimental Data)