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Use of Lubricants to Improve Compressor Efficiency

R. F. Connor

USE OF LUBRICANTS TO IMPROVE COMPRESSOR EFFICIENCY

R. F. Connor, Product Engineer

Whirlpool Corporation
Refrigeration Product Engineering
Evansville, Indiana 47727

ABSTRACT

Current world conditions require compressors to be both efficient and economical to manufacture.

One means of improving efficiency in a rotary vane (or any) compressor is to use the lubricant as a heat transfer agent.

A rotary vane refrigerator compressor was modified and instrumented for calorimeter testing so that the performance improvement could be identified. The changes and testing were controlled so that performance and unique physical parts changes were accomplished by the opening and closing of external valves only, thereby providing the most reliable data possible.

To provide dependable lubrication rotary compressors normally have high-side-pressure housings resulting in a lubricant/refrigerant mixture controlled by pressure and temperature. This mixture was also observed.

The results show the lubricant mixture can be used to provide performance improvement and simultaneously advantageously change the lubricant/refrigerant mixture. And, the relative cost is less than the equivalent required motor improvement costs requiring minor tooling changes only.

INTRODUCTION

The use of lubricant to improve compressor efficiency is neither new nor unique for large commercial refrigeration units commonly use "oil" coolers. But, smaller appliance-type compressors are limited as to lubricant type/quality efficiency improvements.

An appliance manufacturer built into the piston compressor crankshaft a small eccentric pump to circulate cooled lubricant from the sump through the shaft bearing and over the heated cylinder head. The aluminum head gasket, pan formed, retained the lubricant and heated it before it gravitated over the assembly frame in returning to the sump. Concurrently, lubricant was circulated and pooled in the stator mounting frame, (Fig. 1). A large diameter lubricant circulating "U" tube in the bottom housing using an externally attached defrost water disposal pan for cooling completed the heat transfer cycle.

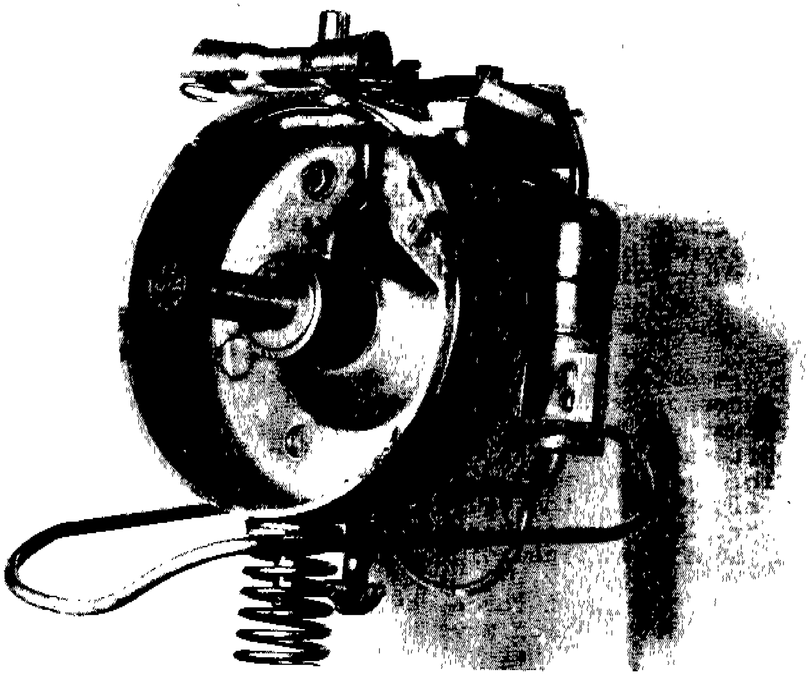

FIG. 1

For low-temperature applications, rotary vane compressors use a precooler (or desuperheater) whose primary purpose is to separate and return to the sump lubricant flowing with the discharge gas. Simultaneously, gas and lubricant are cooled, but excessive cooling can entrain refrigerant in the lubricant, decreasing performance.

BACKGROUND

Rotary vane (or rotary) compressors are unique for the housing is normally at discharge/high pressure to simplify the lubrication system. Also, the cylinder discharge gas contains significant amounts of lubricant which needs to be retained in the housing, while the sump lubricant contains much varying amounts of dissolved refrigerant with pressure/temperature/time.

The gas compression process of a rotary vane compressor is like that of any positive displacement compressor but uses a rotary mechanism causing a unidirectional refrigerant flow, (Fig. 2). The suction flow is continuous with varying volume while the discharge flow is cyclically intermittent. Therefore, rotary compressors do not require suction valves or mufflers but can and do use a simple inherent check valve to prevent reverse off cycle refrigerant flow. Suction volumes to prevent slugging and/or aid starting are commonly used for some applications.

The lubricant flow for a rotary vane compressor is significantly different for it can be comprised of a number of different forces that vary with design and housing pressure/temperature. The primary lubricant flow force is differential pressure leakage--the difference between the housing and cylinder suction pressure reduced by the leakage path length and clearance, (Fig. 3). Pressures of 14 inches leakage lift have been measured.

Secondary lubricant pressure results from the compressor rotor centrifugal force, which is the product of the lubricant-dissolved refrigerant mass, the shaft radius and the shaft rotational velocity squared, (Fig. 3). The mass variation with dissolved refrigerant is minor while the shaft radius and RPM are design controlled. Experimental values of 80-90% theoretical are easily obtained, ignoring time.

The smallest force results from rotating helical shaft grooves which move the lubricant proportional to the shaft rotational velocity while groove friction and leakage negate the flow. Using a combination of properly designed secondary force and groove pumping can provide a self lubricating rotary vane compressor. Vacuum pump/low pressure pumps with 10 inches lift have been life tested successfully.

The rotary vane lubrication system differs also since it not only carries bearing lubricant but carries the lubricant that flows with the discharge gas. That lubricant/refrigerant mixture leaks across the faces of

TOP VIEW
ROTARY VANE COMPRESSOR
(TOP HOUSING REMOVED)

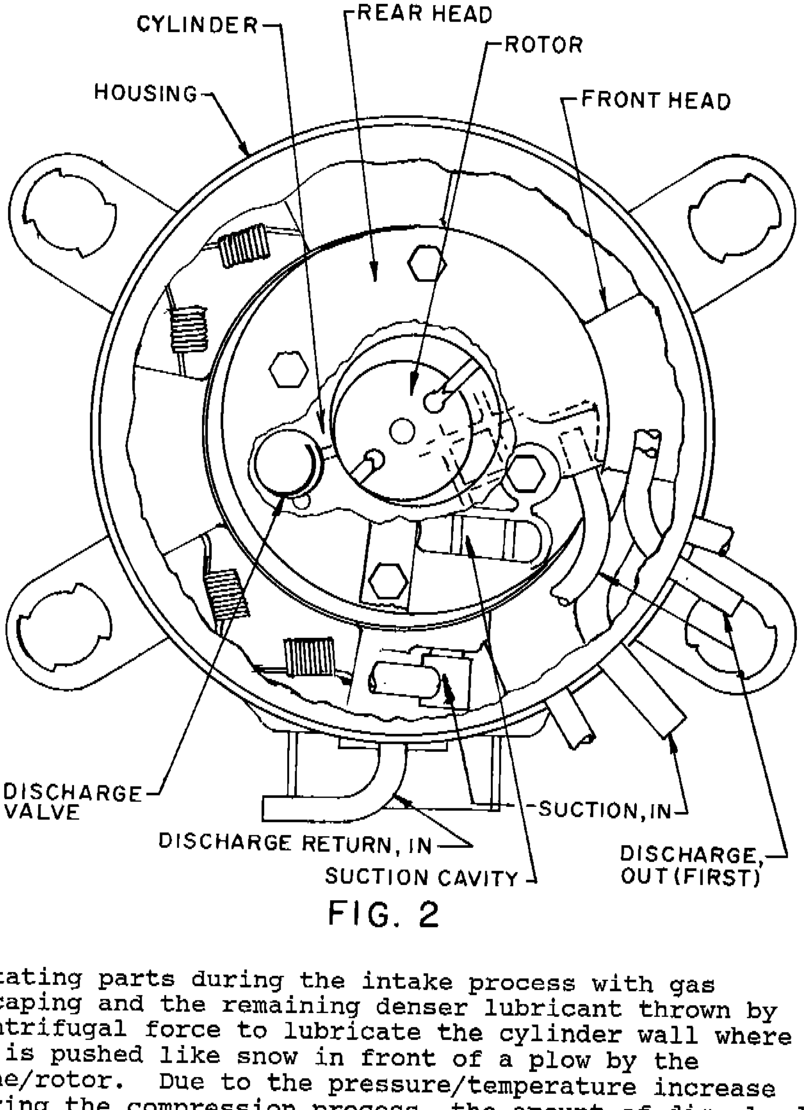

FIG. 2

rotating parts during the intake process with gas escaping and the remaining denser lubricant thrown by centrifugal force to lubricate the cylinder wall where it is pushed like snow in front of a plow by the vane/rotor. Due to the pressure/temperature increase during the compression process, the amount of dissolved gas in the lubricant increases with pressure and as the heat of compression is adsorbed. Some refrigerant becomes entrained in the lubricant so that physical

CROSS SECTION
ROTARY VANE COMPRESSOR

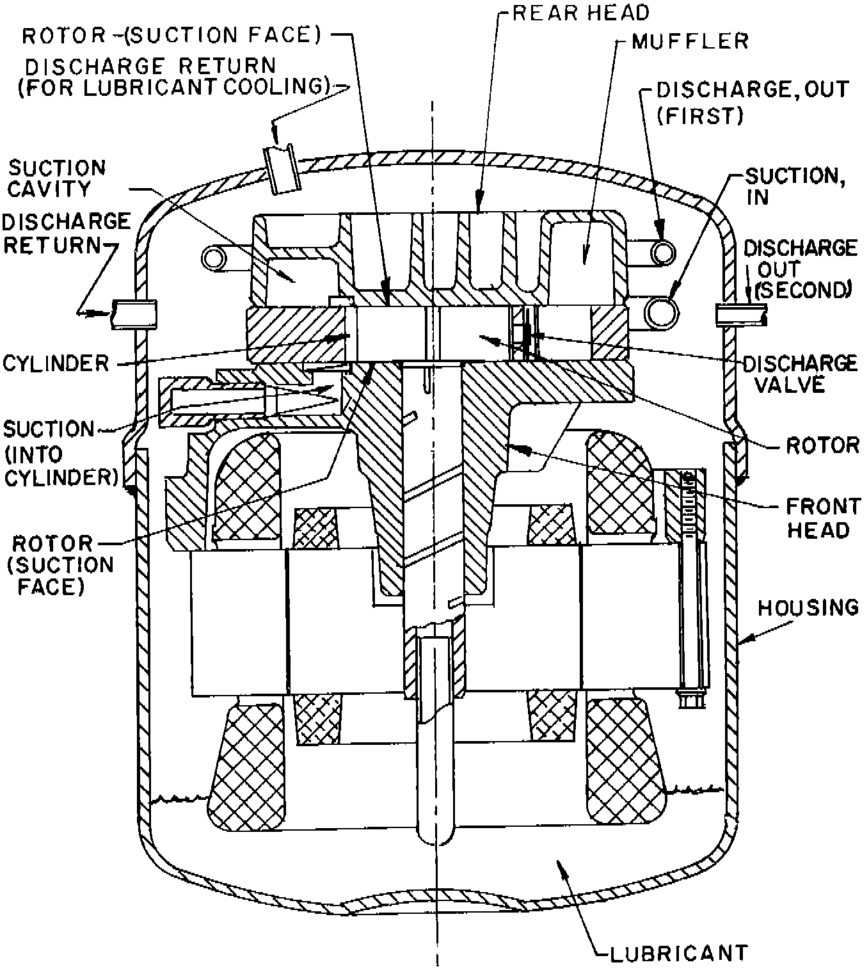

FIG. 3

separation is required to remove it. This whole process is readily observable using a sight glass bearing surface with a strobe light, high-speed photography or TV equipment.

For low temperature refrigeration, the high pressure cylinder solution--gas and lubricant--are piped to the precooler, the desuperheated lubricant and gas returned to the housing and are separated. The lubricant with dissolved/entrained refrigerant flows to the compressor sump and the gas flows over the motor/compression assembly gaining heat as it moves to the condenser. In a well-designed compressor/ refrigeration system the condenser gas contains no lubricant or very little. Typically the cylinder discharge will contain 20-40 cc/min. lubricant flow while the condenser inlet normally contains no measurable amount but at maximum approximately 0.1 cc/min. flow.

High suction pressure rotary compressor applications can omit precoolers because the heat of compression from the 3:1-4:1 compression ratio versus the 10:1-12:1 of low suction pressure can be readily rejected by other means. But, the compressor must be well designed to prevent discharge lubricant flow into the system as the lubricant flow increases with increased refrigerant flow.

CONCEPT

Since the desuperheated lubricant is cooled and contains dissolved refrigerant, heat and agitation can release it. And, if the heat of compression can be adsorbed the efficiency of the compressor can be increased. Any decrease in the dissolved/entrained refrigerant will further increase efficiency through, (1) the increase in return discharge gas available for system use and (2) by decreasing the displacement reduction of gas flashing from the leaking lubricant into the cylinder during the suction process.

The rotary vane compressor with an integral rear head discharge muffler and an external structure having pockets can easily be redesigned to control flow-heat transfer required for this concept, (Fig.'s 4 and 5).

TESTS

1. To examine the concepts and viability, production-type compressors (Fig. 3) were tested at both standard and special rating conditions with only the simplest modifications providing these generalizations:
a. An efficiency improvement of 5% minimum.
b. Increased improvement with larger capacity compressors.
c. Inconsistent improved performances with higher lubricant flows and lower temperature desuperheated return flows.
d. Unexplained performance improvement variations of 5-30% for same and compressors of same capacity.
e. Need for an instrumented compressor to define process and change in process.

REAR HEAD-EXTERIOR
LUBRICANT FLOW FACE

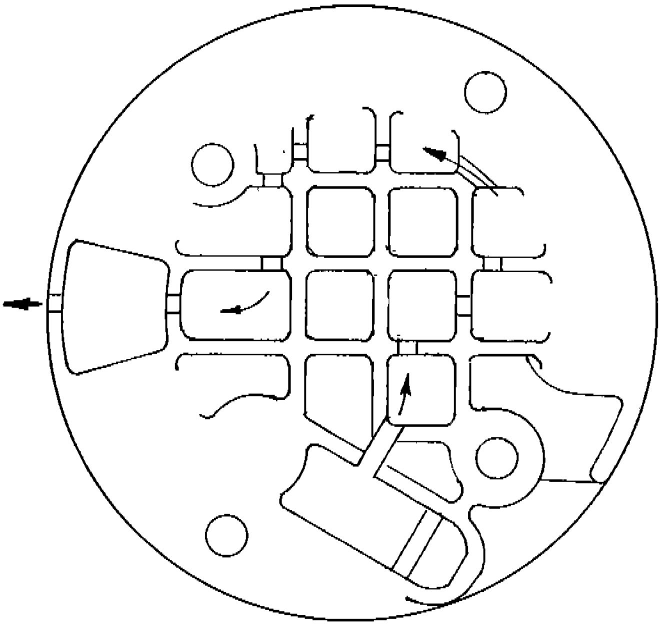

FIG. 4

2. The preliminary tests indicated that lubricant flows per se varied widely even though the cylinder discharge lubricant flow rates did not do so. Therefore, samples of sump lubricant and precooler lubricant flows were taken to ascertain refrigerant/ lubricant ratios. Because the samples are difficult to obtain and control, only limited analysis were made. Generalizing:

a. Within normal BTUH variations, larger variations of lubricant flow per se exists.

b. Sump refrigerant concentration can vary widely with constant controlled conditions.

c. The lubricant is readily saturated but not readily undersaturated; with the best means of attaining undersaturation being preventing saturation.

d. The probable actual lubricant/refrigerant

REAR HEAD - CYLINDER FACE MUFFLER FLOW

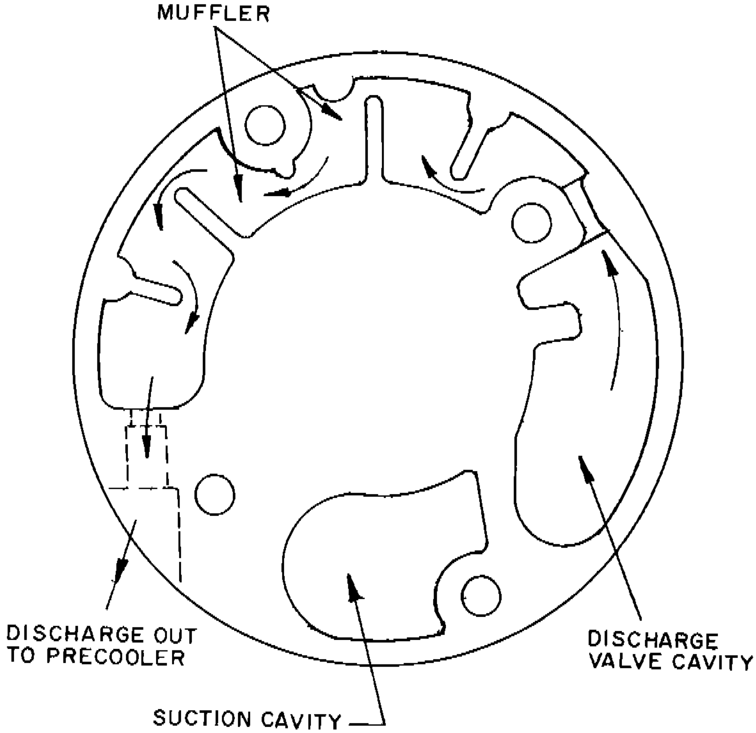

FIG. 5

relationship is a hysteresis-type curve with saturation occurring in the cylinder and precooler and undersaturation with heat/agitation/time.

3. Variations in performance improvements indicated a controlled lubricant flow path would provide more consistent performance improvements. Concurrently with performance data several flow routes were observed using a sight glass, concluding:
a. Desuperheated return flow should start at the cavity over the rear head suction volume, then
b. Counterflow to discharge muffler flow, and
c. Follow longest distance/time flow path possible.

4. For simplicity the instrumented nominal capacity 4-pole compressor used 22 thermocouples to track the pressure-enthalpy curve and verify heat transfers. To minimize controlled test condition variation the standard precooler return tube and others

were included with valving so that the compressor need not be stopped between tests. Also, test procedures were selected to minimize rebuilds. Sight glasses were included to observe flow patterns.

TESTS DATA

1. Much data was recorded for three types of rear head lubricant flow patterns: (1) the original, (2) a modified, (3) the final design. Table 1 is an abstracted summary of the tests.

SUMMARY OF TESTING

	ITEM	WITHOUT LUBRICANT COOLING	FINAL DESIGN	TEMP'TURE CHANGE DEG. F.
1	% Performance Improvement, EER	---	+11	---
2	% Refrigerant in Sump Lubricant	29	26	---
3	% Refrigerant in Sump Lubricant (Per DuPont/Sun Oil Charts)	48	42	---
4	Discharge Pressure, psig	181	181	---
	(Cond'sing Temperature, deg. F.)	130	130	---
5	Suction Pressure, psig	4.5	4.5	---
	(Evap'ting Temperature, deg. F.)	-10.0	-10.0	---
6	Dome Temperature, deg. F.	150	150	0
7	Suction Temperature, deg. F.	90	90	0
8	Ambient Temperature, deg. F.	90	90	0
9	Suction in Rear Head, deg. F.	175	141	-34
10	Discharge Valve Cavity, deg. F.	246	217	-29
11	Muffler Out, deg. F.	204	162	-42
12	Discharge to Precooler, deg. F.	182	154	-28
13	Second Discharge, deg. F.	150	150	0
14	Sump Lubricant, deg. F.	158	153	-5

TABLE 1.

2. Data from Table 1 was plotted on a Refrigerant-12 Pressure-Enthalpy Diagram to illustrate the compression process for compressors with and without lubricant cooling, (Fig. 6).

NOTE: This P-H diagram does not depict a constant entropy process because of the amount of lubricant circulated with the refrigerant.

COMMENTARY

1. Application tests confirmed the performance improvement but since the system normal operating conditions are not as severe, the performance improvement was less and varied with the application.

PRESSURE-ENTHALPY R-12 DIAGRAM FOR ROTARY VANE COMPRESSOR WITH & WITHOUT LUBRICANT COOLING

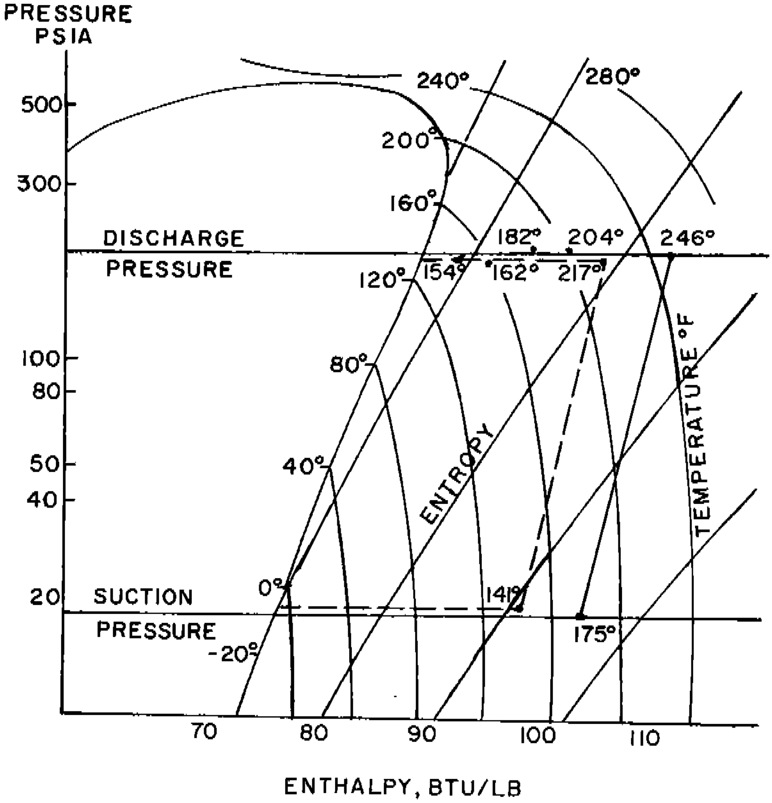

FIG.6

2. A cost study showed an equivalent motor improvement cost 350% more than the costs for adding this type of lubricant cooling.

3. Additional work with centrifugal lubricant/refrigerant cooling/separation confirmed the pressure/temperature/time undersaturation shown in this testing.