Oil Circulation-Its Effects on Compressor Capacity, Theory and Experiment

K. W. Cooper
Borg-Warner Corporation

A. G. Mount
York Division

Follow this and additional works at: http://docs.lib.purdue.edu/icec

http://docs.lib.purdue.edu/icec/9

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
INTRODUCTION

This paper is an outgrowth of automotive air conditioning system analysis done at the York Division of Borg-Warner Corporation. One of the conditions for an acceptable automobile air conditioner is that it be able to quickly cool the car interior after a period of standing in the sun. To simulate this condition, the automotive industry uses a so-called "pulldown test". This test consists of "soaking" a car with a simulated sun load and at an elevated temperature until a given inside temperature is reached. The car is then started and the air conditioner engaged. Selected temperatures are monitored as a function of time.

It has been found that the performance of the compressor with a given system is important for the initial minute or two of the pulldown. If for some reason there is a deterioration of performance at this time, the system cannot seem to catch up to an initially better performing compressor even though the capacities of the two are the same after several minutes.

During one of the pulldown tests, this phenomenon was apparent. In attempting to analyze the test, selected calorimeter tests were performed. It was found that the system had inadvertently been overcharged with oil causing a high oil circulation rate. When these conditions were duplicated on the calorimeter, a difference in capacity of nearly 30% was noted. In order to explain this difference, the thermodynamics of R12-oil mixtures was investigated. This paper presents the results of that study.

THERMODYNAMICS

A positive displacement compressor will induct a given volume of fluid at a given speed and pressure ratio. Figure 1 is a sketch of the refrigeration system. The capacity of the evaporator will be given by:

$$Q = m \cdot (h_2 - h_1) = \rho s V_D (h_2 - h_1) \gamma_v$$  \hspace{1cm} (1)

If there is only refrigerant flowing, the capacity will be:

$$Q_r = \rho s V_D (h_2^e - h_1^e) \gamma_{v_r}$$  \hspace{1cm} (2)

and if there is an oil-refrigerant mixture

$$Q_m = \rho s_m V_D (h_2^m - h_1^m) \gamma_{v_m}$$  \hspace{1cm} (3)

Experience has indicated that volumetric efficiency may be somewhat affected by oil circulation due to the cooling and expansion effect. However, for the first approximation, we will assume that it is not changed. Thus:

$$\gamma_{v_r} = \gamma_{v_m}$$  \hspace{1cm} (4)

Taking the ratio of (3) to (2) and using (4);

$$\frac{Q_m}{Q_r} = \frac{\rho s_m (h_2^m - h_1^m)}{\rho s_r (h_2^e - h_1^e)}$$  \hspace{1cm} (5)

It is known that $$(h_2^m - h_1^m)$$ is less than $$(h_2^e - h_1^e)$$ and that $$\rho s_m$$ is greater than $$\rho s_r$$; thus it cannot be told how much reduction will occur without some actual calculations.

When the liquid mixture enters the evaporator, it contains circulating oil (x lbs. of oil per lb. of mixture). As it leaves the evaporator, some of the refrigerant (w lbs. of refrigerant per lb. liquid mixture) remains in the oil as a liquid. This reduces the amount of refrigerant...
vapor. The amount of oil and refrigerant liquid (z lbs. oil and refrigerant liquid per lb. mixture) is given by:

\[ z = \frac{x}{1 - w} \]  \hspace{1cm} (6)

The amount of refrigerant vapor is \((1 - z)\).

In order to find the amount of refrigerant liquid in the oil leaving the evaporator \((w)\), it is necessary to have an equation of oil-refrigerant solubility. This study is concerned with R12-oil mixtures. G. Bambach, has developed the following equations to represent paraffin base R12-oil solubility. These equations are written in the MKS system of units following European custom. No attempt has been made to convert to English units, as it is a simple matter to make the required conversions for pressure and temperature.

For temperatures below 0°C. \((32°F.)\):

\[ \log_{10}(P') = 5.0057 - 0.558 w^{-\frac{1}{2}} - 1177.67 - 98.753 w^{-\frac{1}{2}} \]  \hspace{1cm} (7)

and for temperatures above 0°C. \((32°F.)\):

\[ \log_{10}(P') = 5.0057 - 0.558 w^{-\frac{1}{2}} - 1177.67 - 98.753 w^{-\frac{1}{2}} \]  \hspace{1cm} (8)

\[ -(0.002338(w - 0.6)^2 - 0.000075)(T' - 273.16) \]

Where:

- \(P'\) pressure--Kg/cm\(^2\) = 0.07031(Psia)
- \(T'\) temperature--Kelvin

For oil the enthalpy to the same base as R12, i.e., liquid at \(-40°F.,\) is found by considering the definition:

\[ \frac{dh}{dT} = C_p \]  \hspace{1cm} (9)

Thus:

\[ \int_{0}^{h_o} dh = \int_{-40}^{T} C_p dT \]  \hspace{1cm} (11)

or:

\[ h_o = \int_{-40}^{T} (0.403 + 0.0005(T)) dT \]  \hspace{1cm} (12)

Integrating and substituting the limits:

\[ h_o = 0.403(T) + 0.00025 T^2 + 15.72 \]  \hspace{1cm} (13)

The enthalpy of the liquid leaving the evaporator is:

\[ h_z = (1 - w) h_{O_{\text{rL}}} + wh_{2_{\text{rL}}} \]  \hspace{1cm} (14)

where, \(h_{2_{\text{rL}}} = \) liquid R12 enthalpy at \(T_2\).

Then the enthalpy of the total mixture leaving the evaporator is:

\[ h_m = z h_z + (1 - z) h_{2_{\text{rv}}} \]  \hspace{1cm} (15)

where,

\[ h_{2_{\text{rv}}} = \text{enthalpy of R12 vapor at } P_2 \text{ and } T_2. \]

Reference to Bambach \(^1\) indicates that the heat of mixing for R12-oil mixtures is small and thus has been neglected in equation (15).

The enthalpy of the liquid mixture entering the evaporator is:

\[ h_{1_{\text{m}}} = x h_{1_{\text{O}}} + (1 - x) h_{1_{\text{rL}}} \]  \hspace{1cm} (16)

where, \(h_{1_{\text{rL}}} = \) enthalpy of R12 liquid at \(T_1\).

The mixture density at the compressor is found in a similar manner. For the liquid portion of the mixture:

\[ \frac{1}{\rho_{z}} = \frac{w}{\rho_{S_{\text{rL}}}} + \frac{1-w}{\rho_{S_{\text{O}}}} \]  \hspace{1cm} (17)

where, \(\rho_{S_{\text{rL}}} = \) density of refrigerant liquid at \(T_s\) and \(\rho_{S_{\text{O}}} = \) density of oil at \(T_s\).

For a common refrigeration oil:

\[ \rho_{O} = (0.923 - 0.00035(T - 60)) 62.4 \]  \hspace{1cm} (18)

The mixture density is then found by a second application of equation (17);

\[ \frac{1}{\rho_{sm}} = \frac{z}{\rho_{z}} + \frac{1-z}{\rho_{s_{rv}}} \]  \hspace{1cm} (19)

where, \(\rho_{s_{rv}} = \) density of refrigerant vapor at \(P_s\) and \(T_s\).

Again, reference to Bambach \(^1\) indicates that the volume contraction for R12-oil mixtures is small and therefore has been neglected in equation (19).

Equations (6) through (19) may be used to find the numerator of equation (5). The denominator is composed of the pure refrigerant properties at the indicated conditions. It should be noted that the density ratio calculation is based on the properties at the compressor suction while the enthalpy ratio is calculated using...
properties entering and leaving the evaporator. The next section presents some of these calculations in parametric form.

RESULTS

In order to obtain a feeling for the magnitude of the compressor capacity correction, Figures 2, 3 and 4 have been drawn assuming no change in pressure and temperature between the evaporator outlet and compressor suction. These curves have been calculated for a range of conditions typical of automotive air conditioning system operation.

Figure 2 indicates that the capacity correction is not a strong function of liquid temperature. However, the correction factor becomes sizable with increased oil circulation when evaporating temperature and superheat are held constant.

As the evaporating temperature increases, the capacity correction factor increases. Figure 3, for constant liquid temperature and superheat, shows that the capacity correction does not change greatly with evaporating temperatures. The dependence on oil circulation is about the same as exhibited by Figure 2.

Figure 4 shows that superheat has by far the greatest influence on capacity reduction for a given oil circulation rate. E.g., for a 5% oil circulation the capacity correction factor decreases from 0.956 to 0.810 as the superheat is reduced from 30°F to 3°F. The reason for this is that the solubility of R12 in oil is much greater at low superheats. Therefore, less refrigerant is available for evaporation.

In an actual system, the conditions leaving the evaporator are different than those entering the compressor. This means that the compressor capacity correction must be broken up into a density correction and an enthalpy correction as indicated in equation (5). The density correction is calculated using the pressure and temperature at the compressor suction, whereas, the enthalpy correction uses the liquid temperature before expansion and the pressure and temperature leaving the evaporator. The two factors should be multiplied to obtain the compressor capacity correction.

Figure 5 is a graph of the density correction. Enter the graph with the oil circulation, draw a line vertically until it intersects the compressor superheat. From this point, draw a horizontal line to intersect with the compressor saturated temperature and then down to determine the density correction factor.

Similarly, Figure 6 shows the enthalpy correction as a function of oil circulation, liquid temperature entering the expansion valve, superheat and saturated temperature leaving the evaporator. To facilitate construction of Figures 5 and 6, some minor approximations have been made. This has been done for illustrative purposes only. For best accuracy, the equations presented above along with refrigerant property subroutines may be easily solved using a digital computer.

The example lines in Figures 5 and 6 represent the conditions of an actual test. For 7% oil circulation, the density correction is 1.30 and enthalpy correction is 0.54 resulting in a compressor capacity correction of 0.702. When the same compressor was retested with the correct oil charge, the oil circulation rate was nearly zero. The measured capacity ratio between the tests was 0.715. This compares favorably with the calculated correction. A second test with an oil circulation of 2% gave a calculated capacity correction of 0.934 and a measured correction of 0.92.

CONCLUSION

This paper has shown that the thermodynamic effects of R12-oil mixtures are important when measuring compressor capacity or when designing systems. This is indicated by the agreement between test and calculation.

It has been seen that low superheat has a large effect on performance when there is a significant oil circulation rate. The evaporating temperature and liquid temperature exhibit a lesser influence. This points out the need for careful monitoring of oil circulation rates.

It should be noted that this is independent of the effect of oil circulation on heat transfer.

ACKNOWLEDGMENT

We would like to express our appreciation to the York Division of Borg-Warner Corporation for the use of their facilities and data. We are grateful for the help and cooperation of all the people involved in the testing and preparation of this paper.

NOMENCLATURE

\[ \begin{align*}
\text{C}_p & \quad \text{Specific heat of oil} \\
\text{h} & \quad \text{Enthalpy} \\
\text{m} & \quad \text{Mass flow rate} \\
\text{P}' & \quad \text{Pressure} \\
\text{P} & \quad \text{Pressure} \\
\end{align*} \]
Q  Capacity          Btu/hr
T  Temperature       F
T' Temperature       K
V_D Compressor displacement cu ft/hr
w Liquid refrigerant fraction 

x Oil circulation rate 

z Liquid mixture fraction 

\( \rho \) Density 1b/cu ft

\( \gamma_v \) Volumetric Efficiency 

Subscripts

1 entering evaporator

2 leaving evaporator

L liquid

m mixture

o oil

r refrigerant

s compressor suction

v vapor

z liquid mixture in total mixture

BIBLIOGRAPHY


Figure 1. Refrigeration System
Figure 2. Compressor Capacity Correction. Liquid Temperature, R12-Oil Mixtures

Figure 3. Compressor Capacity Correction, Saturated Temperature, R12-Oil Mixtures
LIQUID TEMPERATURE ~ 130°F
SATURATED TEMPERATURE ~ 30°F

Figure 4. Compressor Capacity Correction, Superheat, R12-Oil Mixtures
Figure 5. Density Correction, R12-Oil Mixtures
Figure 6. Enthalpy Correction, R12-Oil Mixtures