Performance Comparison of R-502 Replacements in a Commercial Scale Low Temperature Refrigeration Plant with a Two Stage Compressor

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PERFORMANCE COMPARISON OF R-502 REPLACEMENTS IN A COMMERCIAL SCALE LOW TEMPERATURE REFRIGERATION PLANT WITH A TWO STAGE COMPRESSOR

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ABSTRACT

An existing commercial scale low temperature R-502 refrigeration plant has been extensively instrumented for evaluating R-502 long term replacements R-404A, R-407A and R-507. This installation is part of the refrigeration plant located at the Thermal Technology Centre. The R-502 retrofit test facility has an internally compounded two stage 40 HP reciprocating compressor equipped with desuperheating and subcooling. The evaporator used was a standard direct expansion shell and tube heat exchanger with R-502 on the tube side, and R-11 (used as a brine) on the shell side. The condenser was also a shell and tube heat exchanger with R-502 on the shell side and water on the tube side.

A series of performance tests were carried out under controlled conditions at two different condensing temperatures: 40.6°C (105.0°F) and 29.4°C (85.0°F). The refrigerant evaporating temperature was varied over a range of -45.0°C to -25.0°C (-49.0°F to -13.0°F).

Test data were used to compare important system performance characteristics with R-502 including evaporator capacity, cooling coefficient of performance (COP), and compressor discharge temperature. A comparison of system operating refrigerant pressure drops and refrigerant mass flowrates is also included.

INTRODUCTION

Three refrigerant blends, R-407A, a zeotrope of HFC-32/HFC-125/HFC-134a (20/40/40 wt.%), R-404A, a near azeotrope of HFC-125/HFC-143a/HFC-134a (44/52/4 wt.%) and R-507, an azeotrope of HFC-125/HFC-143a (50/50 wt.%) were compared to R-502 in a commercial scale low temperature refrigeration plant with a two-stage reciprocating compressor and shell and tube heat exchangers. All three refrigerants were developed to replace R-502 without requiring any major equipment changes. The main difference as compared to R-502, is the requirement of using a polyol ester lubricant with the replacements.

DESCRIPTION OF THE COMMERCIAL REFRIGERATION TEST FACILITY

This installation is part of the M-17 low temperature refrigeration plant and uses commercially available two stage water cooled condensing units and refrigeration equipment that is typical in a design of this type. A detailed refrigeration piping schematic of the R-502 retrofit test facility is shown in Figure 1. The installation consists of tandem six cylinder internally compounded two-stage water cooled reciprocating compressors. The two compressors share a common suction/stator housing and are driven by twin 20 HP electric motors. The condensing unit is equipped with inter-stage desuperheating expansion valves and liquid sub-coolers. The condenser is a shell and tube heat exchanger with R-502 on the shell side and water on the tube side. The evaporator used was also a standard direct expansion shell and tube heat exchanger with R-502 on the tube side, and R-11 (used as a brine) on the shell side. The tube side was configured with two passes in a single refrigerant circuit. The heat exchanger was constructed with 71 tubes of 16 mm (0.625 in.) OD and a tube length of 2438 mm (100 in.), giving a total heat exchange surface area of 8.63 m^2 (92.9 ft^2).

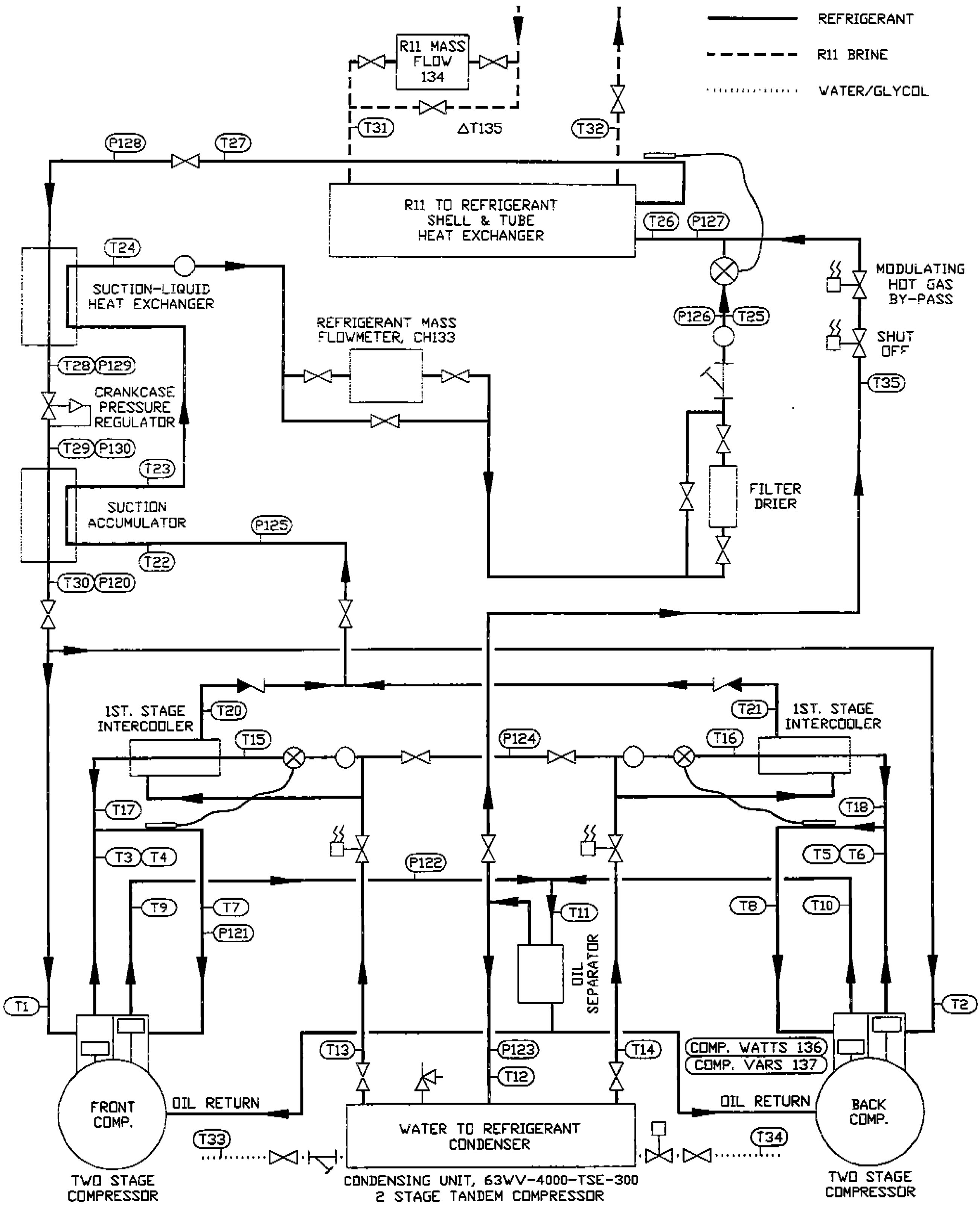

Figure 1. Schematic of the Test Facility

METHOD OF TESTING

Baseline testing was performed first using R-502 and alkylbenzene type oil with a viscosity of 32 mm^2/s at 40°C (104.0°F). The condensing temperatures, 40.6°C (105.0°F) and 29.4°C (85.0°F) were set by adjusting the condensing pressure actuated water-regulating valve. The refrigerant evaporating temperature was varied over a range of -45.0°C to -25.0°C (-49.0°F to -13.0°F) by adjusting the temperature of the R-11 brine using the outdoor room of the Calorimetric Test Facility as the load. Test data were recorded for each condensing temperature and each corresponding evaporating temperature. After completing the R-502/alkylbenzene tests, the system was retrofitted to a polyol ester oil with a viscosity of 32 mm^2/s at 40°C (104.0°F). This required three complete flushes to bring the alkylbenzene oil content down to less than 5% in the system, as measured with a refractometer. The tests were repeated with R-502 and polyol ester oil. The system was then evacuated and refilled with the test refrigerants and the tests repeated under the same conditions for each refrigerant.

TEST MEASUREMENTS

The evaporator capacities reported were obtained from mass flow measurements in the R-11 brine circuit and from accurate differential temperatures of the R-11 brine flow between the inlet and outlet of the evaporator. The specific heat of the R-11 brine was determined for the average of the evaporator inlet and outlet temperatures.

The mass flow of the refrigerant was measured in the liquid line before the expansion valve so that the capacity of the evaporator and condenser could be calculated independently. The capacity of the evaporator calculated by the refrigerant enthalpy method was compared to the capacity calculated on the R-11 brine side as a check of the accuracy of the instrumentation.

To achieve a fair comparison of a zeotrope to a single refrigerant or near azeotrope, the refrigerant cycle operating conditions need to be defined. The evaporating temperature was defined as the mean of the evaporator outlet pressure dew point and the evaporator inlet temperature, and the condensing temperature as the mean of the condenser inlet pressure dew point and the condenser outlet pressure bubble point. The superheat and subcooling were measured from the evaporator outlet pressure dew point and the condenser outlet pressure bubble point respectively.

The refrigerant thermodynamic properties of all the test refrigerants and the specific heat for the R-11 brine were determined from a commercially available software program (REFPROP V4). These values were confirmed with property data sheets provided by the refrigerant manufacturers.

The refrigerant charge required for the complete system was 53.1 kg (117 lb) for R-502, 45.6 kg (100.5 lb) for R-507, 45.4 kg (100 lb) for R-404A and 44.5 kg (98 lb) for R-407A (or approximately 15% less charge by mass for the alternatives).

SYSTEM PERFORMANCE

The system performance evaluation has been divided into two main parts. The first part analyzes the effect of the alternative refrigerants on the performance of the condensing unit. The second part discusses the performance of the evaporator operating with the different refrigerants and its effect on overall system performance. The first part would be applicable to any retrofit using this particular condensing unit and these refrigerants. The second part deals with this particular retrofit application and may not be typical of other retrofit situations that incorporate different evaporator configurations.

Part 1. Condensing Unit Performance

There was no measurable difference in performance between R-502 using either alkylbenzene or the polyol ester oil, and this paper reports the results for R-502 with polyol ester oil. The condensing unit was tested at two condensing temperatures: 40.6°C (105.0°F) and 29.4°C (85.0°F). At the lower condensing temperature, all the refrigerants tested had an increase in evaporator capacity of about 7% and a reduction in compressor power of 9%. Only the results for the higher condensing temperature are shown in this paper.

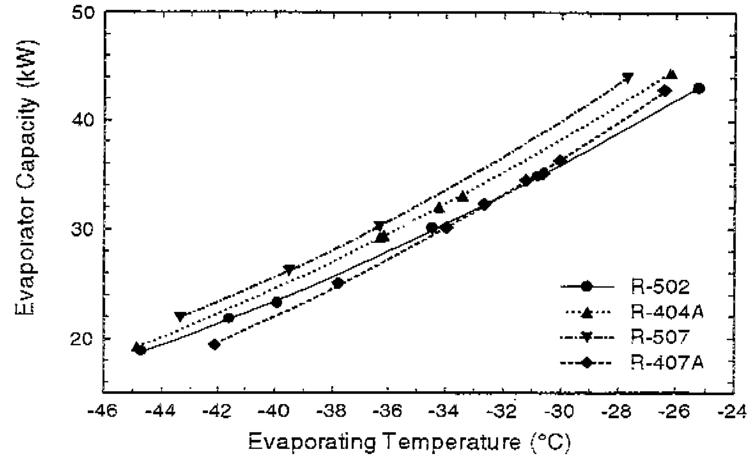

Figure 2. Evaporator capacity of condensing unit

The evaporator capacity of the condensing unit at a condensing temperature of 40.6°C (105.0°F) is shown in Figure 2 R-507 showed the greatest increase in capacity compared to R-502, ranging from 8.5% greater at -40.0°C (-40.0°F) to 11.0% at -26.0°C (-14.8°F). R-404A was next in line with an increase of 4% at -40.0°C (-40.0°F) and up to 6.5% at -26.0°C (-14.8°F). R-407A had a cross-over at -32.0°C (-25.6°F) with a decrease in capacity of 6% at -40.0°C (-40.0°F) and an increase of 3% at -26.0°C (-14.8°F).

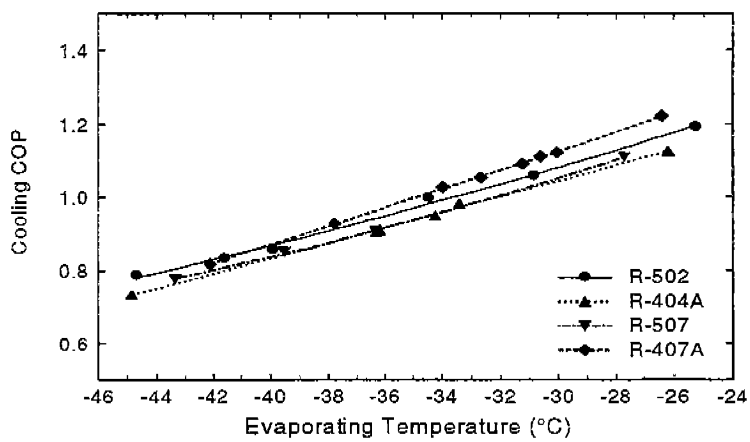

Figure 3. COP of condensing unit

The coefficient of performance (COP) for the condensing unit at 40.6°C (105.0°F) condensing temperature is shown in Figure 3. R-407A had the same COP as R-502 at -40.0°C (-40.0°F) and was about 4% higher at -26.0°C (-14.8°F). R-507 and R-404A had similar COPs that were approximately 3% lower than R-502 over the entire range of evaporating temperatures tested.

The discharge temperature of R-407A was 6°C to 11°C (10.8°F to 19.8°F) higher than R-502, while the other two refrigerants had a discharge temperature that was 4°C to 6°C (7.2°F to 10.8°F) lower than R-502.

The refrigerant mass flowrate through the condensing unit for R-507 was very close to the flowrate of R-502. The flowrate for R-404A was approximately 8% lower than R-502 while R-407A was 28% to 34% lower.

The pressure drop measured between the evaporator inlet and compressor inlet for R-507 and R-404A were about 5% higher than that of R-502, while the pressure drop of R-407A was about 11% lower than R-502.

Part 2. Evaporator and Overall System Performance

The evaporator is the main link between the process fluid being cooled (in this case R-11 brine) and the refrigerant that performs the cooling. In the real world it is the process fluid outlet temperature that must be maintained, and the refrigerant evaporating temperature will vary to suit the operating conditions. Figure 4 shows the required condensing unit evaporating temperature for each refrigerant for a given R-11 outlet temperature. The resulting evaporating temperatures are lower for all three replacement refrigerants as compared to R-502, which is an indication of reduced overall heat transfer performance of the heat exchanger/refrigerant combination.

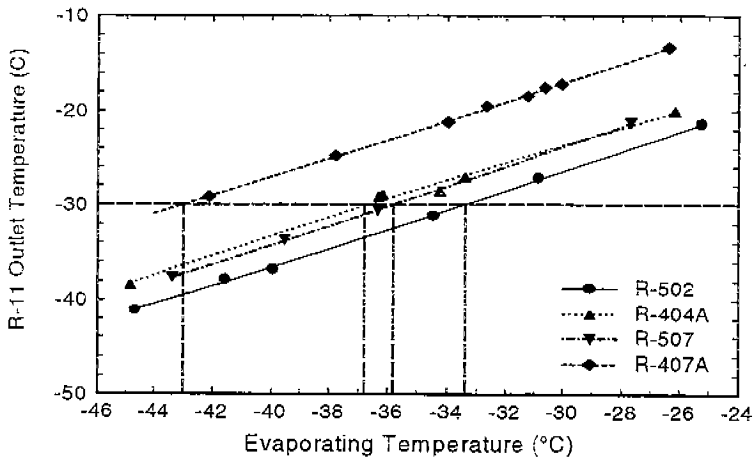

Figure 4. R-11 outlet temperature vs. evaporating temperature

As an example, Figure 4 shows that if the required R-11 outlet temperature for the system was -30.0°C (-22.0°F) and the refrigeration system was operating with refrigerant R-502, the average evaporating temperature would be -33.4°C (-28.1°F). Similarly for R-507, R-404A and R-407A, the evaporating temperatures would be -35.8°C (-32.4°F), -36.8°C (-34.2°F), and -43.0°C (-45.4°F) respectively. From Figures 2 and 3 the overall system capacity and COP can be determined for each refrigerant at the evaporating temperature required to maintain an R-11 outlet temperature of -30°C (-22.0°F). These results are shown in Table 1.

Test Refrigerant	R-11 Outlet Temperature	Evaporating Temperature	System Capacity	COP
R-502	-30.0°C (-22.0°F)	-33.4°C (-28.1°F)	31.2 kW (106.4 Mbtu/h)	1.00
R-507	-30.0°C (-22.0°F)	-35.8°C (-32.4°F)	31.2 kW (106.4 Mbtu/h)	0.92
R-404A	-30.0°C (-22.0°F)	-36.8°C (-34.2°F)	28.9 kW (98.6 Mbtu//h)	0.90
R-407A	-30.0°C (-22.0°F)	-43.0°C (-45.4°F)	18.5 kW (63.1 Mbtu/h)	0.80

Table 1. System capacity and COP for R-11 evaporator outlet temperature of -30.0°C (-22.0°F)

Table 1 shows that the system capacity for R-507 is the same as R-502. This is the result of the increased capacity of the condensing unit with R-507 being offset by the reduced performance of the evaporator. For R-404A the increased capacity of the condensing unit is more than offset by the reduced performance of the evaporator. The evaporator performance with refrigerant R-407A was the lowest of the test refrigerants and had the greatest impact on the overall system capacity. The poor evaporator performance of R-407A may have been caused in part by the lower mass flowrate of R-407A compared to the other refrigerants and its effects on refrigerant distribution and heat transfer in the evaporator. Another contributing factor affecting R-407A evaporator performance is the two-pass tube design of the shell and tube heat exchanger which is not counter-flow and cannot take advantage of the temperature glide of R-407A.

CONCLUSIONS

Three refrigerant blends: R-407A, R-404A and R-507 were evaluated in a commercial scale low temperature refrigeration plant with a two-stage compressor and shell and tube heat exchangers and were compared to R-502.

The performance of the condensing unit operating with R-507 showed the greatest increase in cooling capacity compared to R-502, ranging from 8.5% to 11.0% greater. R-404A was next in line with an increase in capacity of 4% to 6.5%. R-407A had a cross-over at -32.0°C (-25.6°F) with a decrease in capacity of 6% at -40.0°C (-40°F) and an increase of 3% at -26.0°C (-14.8°F) evaporating temperatures. A comparison of the COP of the condensing unit showed that R-407A had the same COP as R-502 at -40.0°C (-40.0°F) and was about 4% higher at -26.0°C (-14.8°F). R-507 and R-404A had similar COPs that were approximately 3% lower than R-502. These conclusions would be applicable to any retrofit application using this type of condensing unit and these refrigerants.

The second part of the evaluation concerned the performance of the two pass, single circuit, shell and tube evaporator and its effect on the overall system performance. The following conclusions deal with this particular evaporator design and may not be typical of other retrofit situations that incorporate different evaporator configurations. All three refrigerant blends showed a reduction in heat transfer performance in this evaporator compared to R-502. When a comparison of the R-502 replacements was made using the evaporator R-11 brine outlet temperature of -30.0°C (-22.0°F) as a reference, the system capacity for R-507 was the same as R-502. For R-404A the increased performance in the condensing unit was more than offset by the reduced performance of the evaporator. The overall system performance with R-407A was the lowest of the test refrigerants, with the design of the evaporator having the greatest impact on the overall system cooling capacity and COP.

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REFPROP, NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database, Version 4, NIST, Gaithersburg, Maryland.