The knowledge hub for refrigeration, air conditioning and heat pumps

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# Comparative Performance of HFO Blends in a Condenser

## Paper 2 – research laboratory tests

## Why this topic?

Users and designers of commercial refrigeration systems are considering alternatives to traditional HFC refrigerants due to the F Gas Phase down.

This paper will present the results of an independent investigation of performance of R448a and R449a in a condenser at typical operating conditions for commercial systems. The performance is compared to R404a, the refrigerant which they are designed to directly replace without any significant plant overhaul.

Learn what system adjustments may be necessary, such as charge levels and expansion device settings when using these refrigerants as a retro-fit for R404a.

### 1 Introduction

With increased pressures for reduction in global warming, more chemicals are under investigation for use as refrigerant. Often front runners are compromised in some way. With HFO refrigerants the disadvantage is flammability. Even though it is at a low level, flammability is a significant issue in commercial applications. To overcome this a number of HFO blends have been developed to provide an acceptable level of GWP with an A1 safety rating.

Two of the leading alternatives to R404a are R448A and R449A

**Table 1 - Constituents of Refrigerants Studied**

| Refrigerant Number | Manufacturer | Constituents | | | | | | GWP |
|---|---|---|---|---|---|---|---|---|
| | | R143a | R125 | R134a | R1234yf | R-32 | 1234ze (E) | |
| R404A | - | 52% | 44% | 4% | - | - | - | 3922 |
| R448A | Honeywell | - | 26% | 21% | 20% | 26% | 7% | 1273 |
| R449A | Chemion | - | 25% | 26% | 25% | 24% | - | 1397 |

As shown in Table 1, the alternatives have 5 and 6 constituents. One of the consequences of this mixture is a significant amount of temperature glide between the saturated liquid and saturated gas state during phase change at a constant pressure. Table 2 gives examples of the value of glide for refrigerants in this study and other established refrigerants with relatively high glide.

**Table 2 - Examples of Glide Values**

| TEMPERATURE | R404A | R407A | R407C | R407F | R448A | R449A |
|---|---|---|---|---|---|---|
| -31 | 0.7 | 6.5 | 6.9 | 6.2 | 6.1 | 6.0 |
| -8 | 0.6 | 6.0 | 6.4 | 5.8 | 5.8 | 5.7 |
| 5 | 0.5 | 5.7 | 6.1 | 5.5 | 5.6 | 5.5 |
| 40 | 0.3 | 4.5 | 5.0 | 4.5 | 4.6 | 4.5 |

It can be seen that the glide values for the HFO blends are very close to each other and to those of R407A and R407F, refrigerants that have recently been in common usage. It would be optimistic to say that the implications of glide are widely understood, but at least many in the refrigeration industry have experienced them before.

For condenser operation the main considerations are:

- Make selections of components using consistent conditions – all mid-point or all dew point.
- When using legacy operating conditions originating from non-glide refrigerants (e.g. R404A or R22) consider values as mid-point.
- Glide subtracts from the driving temperature difference for sub-cooling in a condenser, so the inclusion of a separate sub-cooling section is recommended, particularly when the operating TD is low.

The primary purpose of the testing is to calibrate the manufacturer's coil design software for condenser design with R448A and R449A. Test results will be compared with software results for existing refrigerants with glide to enable inclusion of these new refrigerants in the software and at them to Product Selector software.

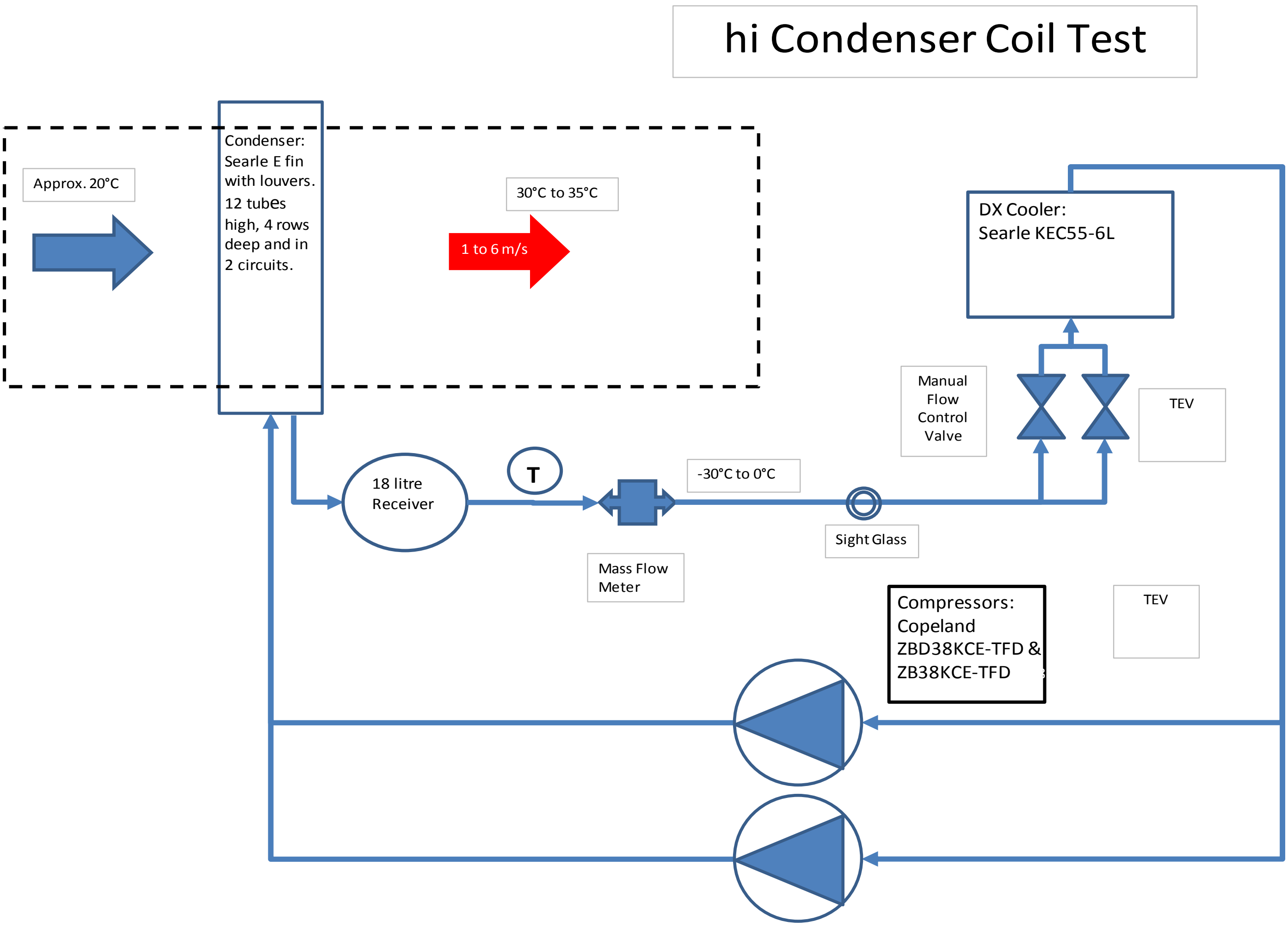
While condenser performance is primary, other implications of using HFO blends in a system will be investigated.

The second stage of the study will be to repeat the process while concentrating on the evaporator performance.

## 2 Test Equipment

The refrigeration system for the study was designed with the primary function of measuring condenser performance. However, component were selected as typical for a commercial system and instrumentation provided to enable recording of the complete system performance.
The major components and instrumentation are shown in Figure 1

**Figure 1 - Refrigeration Test Circuit**



The condenser coil was a typical small condenser coil: 3/8" inner grooved copper tube with louvered aluminium fin. To match the range of operating conditions during the study the coil had 2 circuits

**Figure 2 - Condenser Coil in External Heat Transfer Rig**


For optimum performance with glide refrigerants good counter-flow is required. However, as the HFO blends are intended to be "retrofit" replacements for R404A and it is unlikely that all condensers in the field will not have been circuited for high glide refrigerants, the test condenser circuit arrangement was not optimised in that way.

Air flow through the condenser was delivered by fitting it in the external heat transfer rig. This provided for accurate air flow control and ample instrumentation for air flow and temperatures. The air inlet to the rig is not directly controlled but within the R&D lab building the ambient temperature is controlled and allows for suitably stable inlet air temperatures.

Three different scroll compresors were used. The inclusion of a digital compressor provided for flexible capacity control.

**Figure 3 - Evaporator and Condensing Unit**

The compressors were housed in a condensing unit with the internal condenser and fan disconnected. The condensing unit provided oil separation, a receiver, compressor capacity control and switchgear.

The evaporator was fitted with a variable speed version of usual fansets. As with the digital compressor, the variable speed fans gave flexibility in control.

The evaporator was fitted with two types of expansion devices. One was a typical thermostatic expansion valve (qq), for investigation of any adjustment required during the retrofit process and the other a manual needle valve for performance control.

Instrumentation was fitted to the rig to enable measurement of items in Table 3.

**Table 3 – Instrumentation**

| Parameter | Instrument Type | Details |
|---|---|---|
| Air on Condenser (°C) | Thermocouple | T Type 4 off |
| Air off Condenser (°C) | Thermocouple | T Type 4 off |
| Condenser Gas Inlet (°C) | Thermocouple | T Type 1 off |
| Condenser Liquid Outlet (°C) | Thermocouple | T Type 1 off |
| Evap Liquid Inlet (°C) | Thermocouple | T Type 1 off |
| Evap Suction Outlet (°C) | Thermocouple | T Type 1 off |
| Evap Air On (°C) | Thermocouple | T Type 4 off |
| Condenser Gas In (bar) | Pressure Transducer | UNIK 5000 0 to 40bar |
| Condenser PD (bar) | Pressure Transducer | UNIK 5000 0 to 2bar |
| Cooler Inlet (bar) | Pressure Transducer | Emerson PT5-30M 0 to 30bar |
| Cooler Outlet (bar) | Pressure Transducer | UNIK 5000 0 to 10bar |
| Condenser Air PD (Pa) | Pressure Transducer | Furness Model 332 0 to 1 kPa |
| Refrigerant Flow (kg/h) | Flow Meter | Krohne Optimass 7300C T06 |
| Coil Face Velocity (m/s) | HO Test Rig Orifice Plate | EN BS 848 |
| Compressor Amps | Power Meter | Crompton C13-01-R5 |
| Compressor Watts | Power Meter | Crompton C13-01-R5 |
| Compressor digital % | Pack Controller | Emerson EC2-552 |

## 3 Refrigerant Charge

The system was charged first with R404A. Refrigerant was charged conventional and as a liquid. The refrigerant charge was measured using a set of commercial bottle scales – the same set for each refrigerant. Running the system at minimum evaporating and condensing temperatures the superheat value was reduced – to increase the liquid content in the evaporator – until there was gas visible in the liquid line sight glass. The conditions were recorded so that they could be repeated for the other two refrigerants.

During the remainder of testing, the sight glass was monitored for gas.

**Table 4 - Refrigerant Charge**

| Refrigerant | Charge (kg) |
|---|---|
| R404A | 9.7 |
| R448A | 8.9 |
| R449A | 9.2 |

## 4 Condenser Performance Testing

For a range of operating conditions with coil face velocities from 1.5 to 6 m/s, condensing temperatures from 30°C to 50°C and operating temperature differences between 8 K and 25 K. Coil face velocity was controlled directly by the variable speed fan in the external heat transfer rig. Condensing temperature and operating TD (the difference between condensing temperature and air inlet temperature) are controlled using the manual expansion valve to control refrigerant flow and, at lower flow rates, allowing the compressor to offload.

When testing with R448A and R449A, condensing temperature and TD were considered for mid-point condition. Superheat was always calculated from the dew-point condition and sub-cooling from the bubble point condition. Mid-point temperatures were calculated as the average of dew-point and bubble-point.

A comparison of the test capacity measurement with those from the coil design program is given in Figure 4. The lower capacity data points were not repeated for R448A and R449A as the circuit loading was below what would normally be considered practical.

The primary method for determination condenser capacity was with a mass flow meter and temperature and pressure measurements at the coil inlet and outlet. Pressure and temperature enable the enthalpy difference to be determined and the heat transfer capacity is the product of this value and the mass flow rate. The result of this primary method was validated using the measured and theoretical air off temperatures given by the design software for the test capacity. A chart showing the comparison is in Figure 5.

**Figure 4**

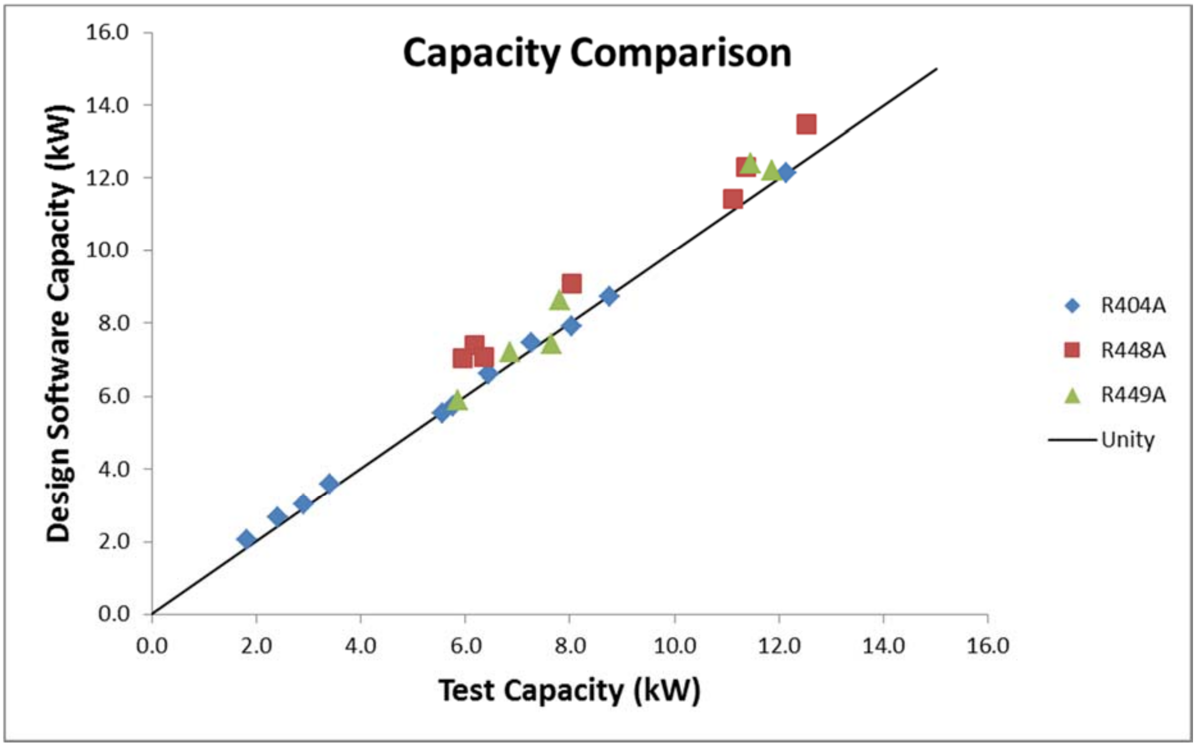
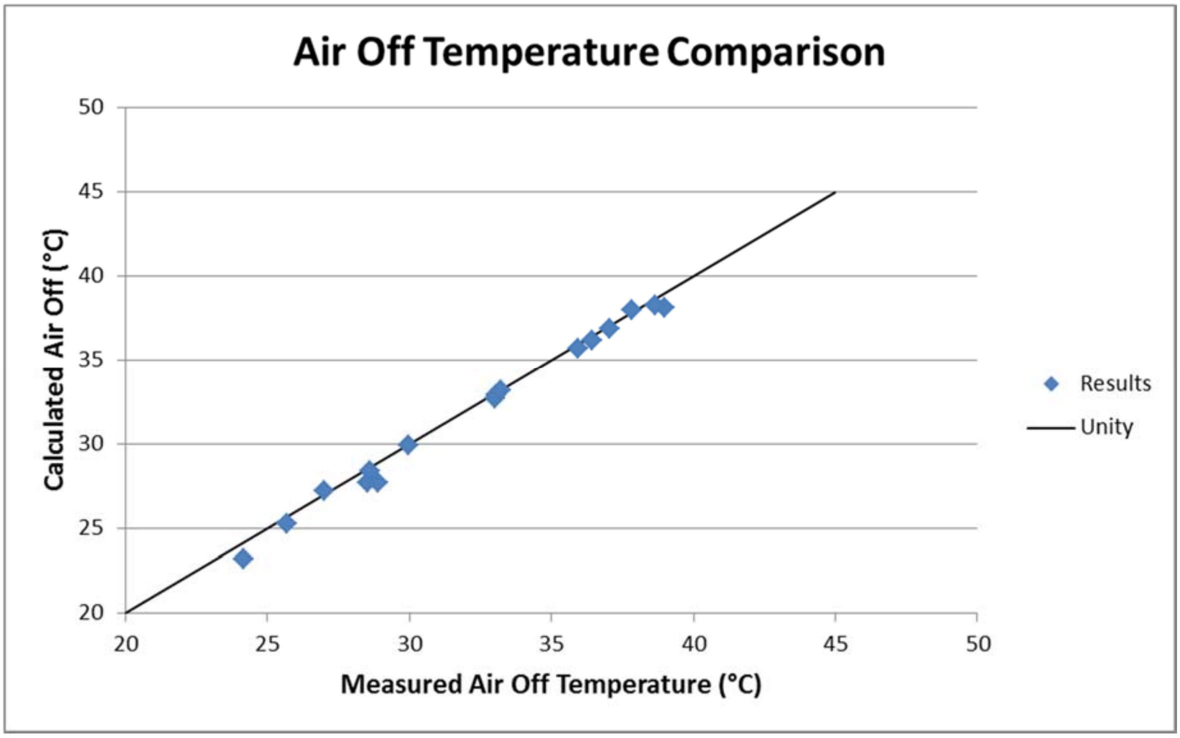
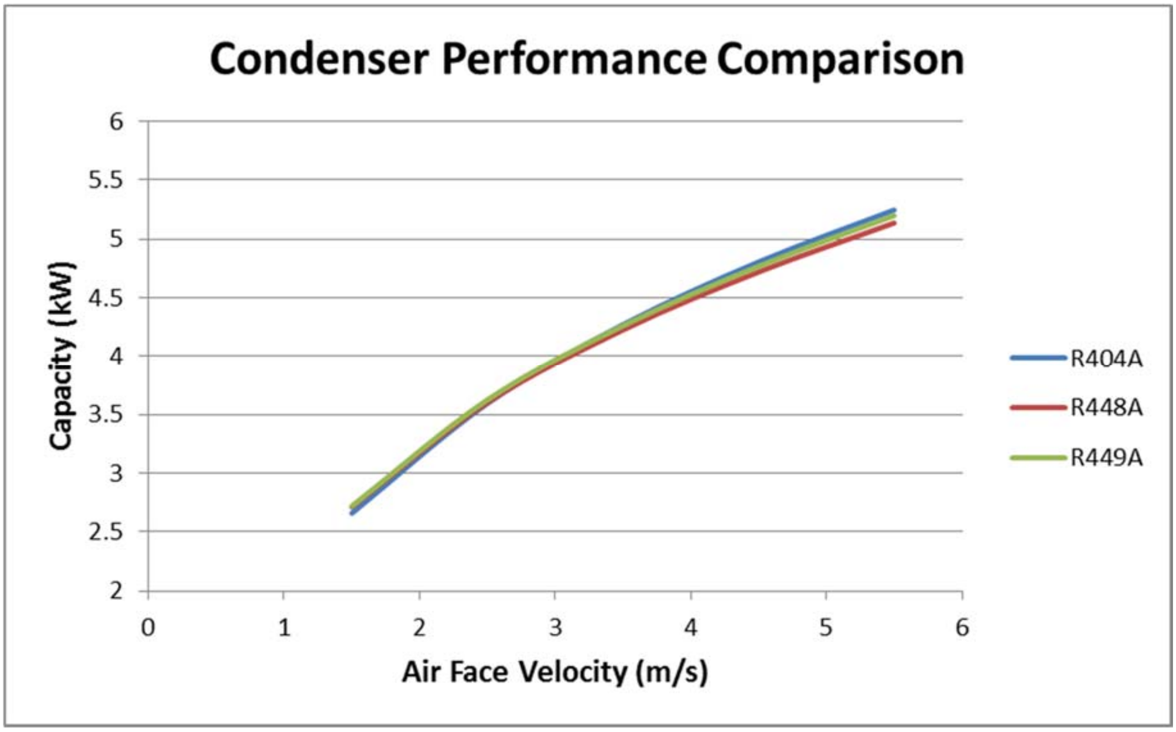


**Table 5 - Capacity Validation**




The coil design software was modified to model the condenser performance with each of the HFO blends. Figure 3 gives a comparison of performance for the test condenser with each of the refrigerants. The conditions for calculation are generally as specified for EUROVENT certification (25°C air inlet temperature, 40°C dew condensing and 65°C gas inlet temperature) except that the HFO capacities were calculated with 10 K higher inlet gas temperature and 40°C mid-point condensing temperature. A chart of results is shown in Figure 6. The maximum difference at any point is 2%.

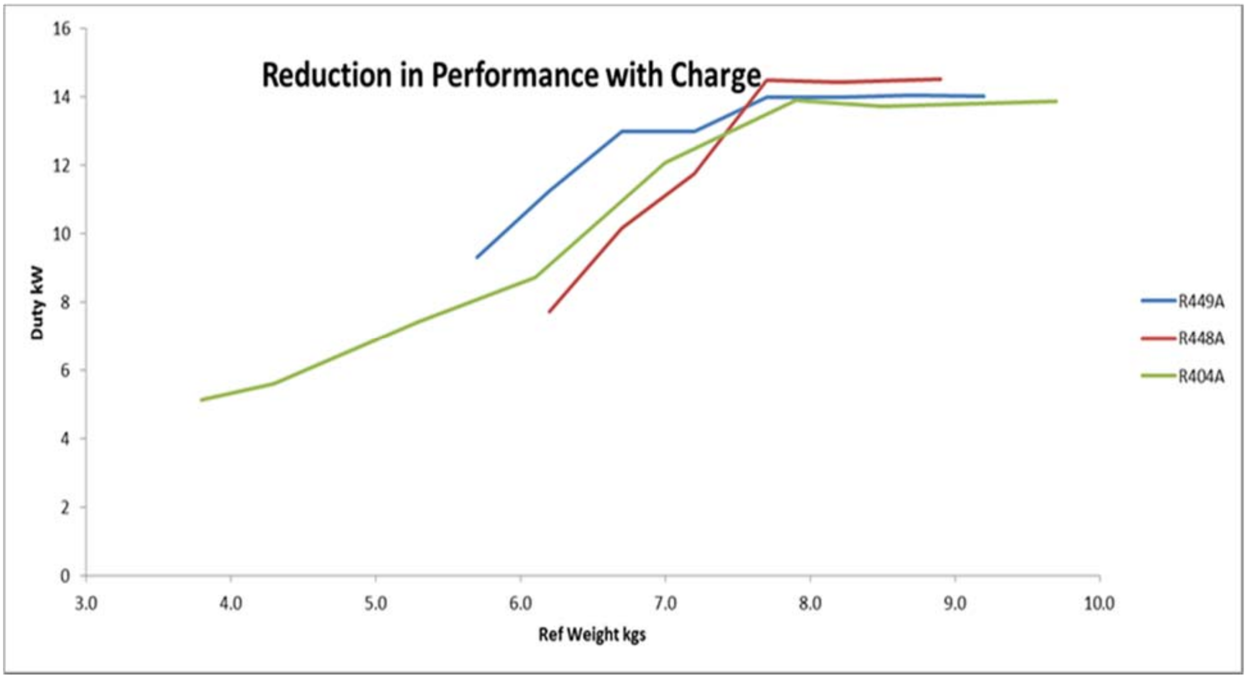
**Table 6**



The setting of the TEV was compared for each refrigerant. At 0°C evaporating it was necessary to adjust the setting in 2 turns for the R448A compared to the R404A setting and for the R449A 3 turns compared to R404A. However, this should be used as no more than a guide for initial setup. TEV valves should be set according to superheat measured as different valves and operating conditions will require different settings. On a system with DX evaporator, superheat setting is one of the most significant factors in operational efficiency.

The final test in the program for each refrigerant was to measure the system performance as the refrigerant was recovered. Refrigerant was recovered in steps to simulate a gas leak. The chart in figure 7 shows the relative rates of performance reduction. The differences between the 3 refrigerants are not significant for this consideration. It is hoped that leakage will be prevented before differences are relevant.

**Figure 7 - Simulated Leakage**




## 5 Conclusion

Condenser performance, when using mid-point condition for the HFO blends, is not significantly different with any of the refrigerants tested. Also, the performance R448A and R449A are unlikely to be significantly different from R407A or R407F.

From the refrigeration changes and system monitoring carried out during the test program nothing was found to discourage any qualified refrigeration engineer from retrofitting R448A or R449A in an existing well maintained R404A system. Differences noted were:

- the requirement to adjust the expansion valve setting (bearing in mind the superheat should be measured from the vapour saturation condition).
- an increase of approximately 10 K in compressor discharge temperature;
- small reduction in charge weight.

### References:

R404A Thermodynamic Refrigeration Data – NIST Standard Reference Database 23 version 9.1
R448A Thermodynamic Refrigeration Data – Honeywell (R448A Properties 16/4/2015)
R449A Thermodynamic Refrigeration Data – Chemours (XP40 NIST database add-in 15/5/2014)
Refrigeration system performance using alternative refrigerants - PETER ERIKSSON - München, 2015-06
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### About Nick Atkins

Currently Technical Manager at Kelvion Searle in Fareham, UK. He has been part of the R&D team there for 27 years. As well as product development and support, he represents the company at EUROVENT and FETA and the UK on CEN-TC110-WG1, responsible for test standards for heat exchangers used in refrigeration and air conditioning. Previously employed at F & R Cooling Limited. He has an honours degree in Mechanical Engineering from the University of Bath.

## Discussion report for Alway and Atkins papers combined

**David Gibson commented that the author had used 1.5K subcooling in the thermographic picture and asked what the subcooling would have been if R404a had been used? How much subcooling are you loosing?**
Nick Atkins replied that the TD they are working on is not sufficient to measure the actual subcooling. This is only an estimate. It depends on drainage on the condenser outlet. With glide you lose 2K operating TD at the outlet so subcooling will be slightly reduced. The 1.5 was only an estimated figure not an actual measurement.

**David Gibson noted that a subcooling coil was recommended for new equipment. What about retrofits?**
Nick replied that this was not generally necessary. Perhaps if a supermarket was operating with 8 K TDit should be considered, but larger organisations with engineering resource should be able to manage that. For medium to small system that tend to operate on larger TDs the amount of subcooling lost is not significant.

**Andrew Gigiel asked about the process of reducing the refrigerant charge during testing which had been shown on the graph. The tests had shown the same results for both refrigerants but why had the system been overcharged?**
Nick responded that there was surplus charge in the receiver in order to cope with all of the conditions that the test system would be subject to. During the operating tests they pushed components to their maximum charge condition, which would have been pushing the superheat on the cooler to a lower level than you would normally use, with a high condensing temperature and a low evaporating temperature. But during the discharging exercise, the system wasn't being placed under the worst operating condition, which would have been pushing the superheat on the cooler to a lower level than you would normally use, with a high condensing temperature and a low evaporating temperature. The same conditions were used for all of the refrigerants tested.

**Colin Vines asked between the two refrigerants one needed two turns on the valve and one needed three. Did you have any thoughts as to why…**
Nick said they hadn't investigated that any further.

**John Austin Davies by webinar asked whether the authors had any advice on which refrigerant to choose R448A or R449A as they seem very similar?**
Paul Alway responded that they had started with one and will probably progress to the other, as the results were very similar across both refrigerants. At the moment R449 seemed to offer a saving in one case.. Industry would have to try them both and consider availability and price. To a certain extend it depended on how the market responds and if everyone make the same choice this would bring the production costs down. Paul had been happy to share his results, but this only represents one supermarket, and there are a lot more systems with R404A out there. Colin confirmed that both price and availability were essential considerations.

**Colin Vines asked why there were such significants differences in results between the three cases used in the first paper?**
Paul said that this was probably down to design issues. One of the cases had a second compressor that didn't start as much as the others due to design, and its base load was drawing less energy. We think that in changing the refrigerant it just tipped it over into using the second compressor. All of the cases are on control circuits and they get a lot of information about performance. They were continuing to monitor and review long term results of comparisons of the three cases to identify the impact of compressors not running as much.