Performance of an Industrial Refrigeration System Using R-407A in a Flooded Evaporator

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ABSTRACT

The performance of an industrial refrigeration system operating on R-407A has been studied by measurement and by computer modelling. The system used an economised cycle with a screw compressor and flooded-bundle refrigerant evaporator. The operating conditions were typically -30°C (-22°F) evaporating temperature and up to 35°C (95°F) condensing temperature. The refrigerant composition in circulation around the system was determined over a range of operating conditions and compared to the predictions of the system model. The performance of the system evaporator was evaluated by measurement of the heat duty and system temperatures. System stability was assessed by comparison of plant data records over an extended period of operation.

INTRODUCTION: DESCRIPTION OF SYSTEM

System History

This paper describes the use of a zeotropic refrigerant, R-407A, in a typical industrial refrigeration system. The refrigeration plant provides several hundred kilowatts of cooling at -30 to -35°C (-22 to -31°F) to a main process plant distillation column. It is located on an ICI manufacturing site in the UK and has operated on R-407A for several years. This paper discusses the practical operating experience with the refrigerant and presents some comparisons of modelled *versus* observed performance parameters.

The refrigeration system was originally designed to use R-22 as the working fluid and ran using R-22 for some years, until changes in the process requirements brought about a need to alter the unit's capability. A new process cooler unit was then needed to enable the refrigeration system to match the increased capacity requirement. The new unit was designed to meet these requirements using R-22 as the working fluid. It was then recognised that this change offered an opportunity of using zeotropic HFC refrigerants in an industrial refrigeration plant with a flooded evaporator. After assessment of the possible process implications it was decided to replace R-22 with R-407A in an extended trial. This would determine whether any significant difference in behaviour of the system arose from use of a zeotrope in place of a single-component fluid. The trial performance was acceptable and the unit is still running on R-407A today.

Design Features

The refrigerant feed to the evaporator was modified to give four feed points, equally spaced along the shell and angled slightly down towards the tube bundle. Each feed branch was fitted with a manual isolation valve and thermowell. This modification was intended to help mix incoming refrigerant equally along the length of the heat transfer surface.

The capacity of pressure relief valves and the vapour pressure relative to design pressure were checked as part of the conversion. An inhouse computer program [1] was used to estimate refrigerant compositions as part of this work. Guidance was also obtained from the compressor manufacturers on the selection of appropriate elastomer seals, lubricant viscosities, instrument calibration and control system alterations.

Refrigeration Cycle

A simplified sketch of the refrigeration cycle is shown in Figure 1.

The system is driven by a single-screw compressor with oil cooling, hot-gas bypass and capacity slide control. The refrigerant is evaporated on the shell side of a kettle-type heat exchanger, the heat being provided by a condensing process vapour. The compressed refrigerant vapour is condensed against cooling tower water in the shell-side of a 1-2 shell/tube exchanger. The unit is also fitted with an economiser. The system charge is approximately 3 tonnes of refrigerant.

The system is controlled to maintain fixed refrigerant evaporating pressure and liquid level. A degree of hot-gas recycle is maintained in addition to adjustment of the capacity slide valve. The condensing pressure floats depending on the cooling water supply temperature: the condenser water flow is set at a fixed rate. The economiser is controlled to a fixed liquid exit temperature by adjustment of the flash liquid rate.

The typical system operating conditions are: evaporator temperature -31 to -35°C (-22 to -31°F), condenser temperature 25-35°C (77-95°F) (dependent on ambient temperature), economiser exit temperature 0°C (32°F), compressor suction temperature -5°C (23°F), compressor discharge temperature 65°C (150°F).

MODELLING THE SYSTEM

Composition Shift

It has been known for some time that when a zeotropic refrigerant is charged to a system the composition of fluid in circulation may differ from that of the bulk charge [1,2]. For systems where the refrigerant charge is significantly larger than that of the lubricant the size of this difference has been shown to be determined principally by two effects:

- The *volumetric* effect, that is, the relative ratios of volumes of liquid and vapour in evaporator and condenser units
- The *temperature* effect, that is, the influence of evaporator and condenser temperatures on the partitioning of the components between liquid and vapour

Corr et al. [1] developed a PC-based computer model to investigate the relative magnitude of these phenomena. These effects may be summarised:

- As the proportion of liquid to vapour in the evaporator rises, the circulating composition is enriched in more volatile components. Conversely a drop in liquid level reduces the proportion of volatile component in the circulating composition.
- As the temperature in the evaporator falls, the circulating composition is enriched in more volatile components.

Cycle Model

A new computer model of the refrigeration system was developed as an extension of the work of Corr et al. This model was written in a proprietary process simulation package, allowing extension of the cycle from the "four-block" model used in the original work. The model includes process and cooling water flows, the compressor oil cooler and the economiser unit.

The model used a block of FORTRAN code together with a constraint function to determine the circulating composition. The calculation algorithm was similar to that used previously except that the effect of condenser vapour on circulating composition was omitted. This simplification was justified by analysis of the impact of condenser liquid fill variations over the observed range on the plant.

The refrigerant thermodynamic properties used in the simulation were calculated using the MHV-2 method as described by Morrison et al. [3]. These property correlations had previously been verified by laboratory measurement [2,3] over the temperature range of interest.

EXPERIMENTAL WORK

Circulating Composition Analysis

The circulating refrigerant composition was measured with two objectives: verification of the composition shift model, and assessment of the effect of a change in evaporator operating level on the circulating composition.

Three sample points were fitted to the refrigeration system: one on the evaporator vapour offtake line; one on the compressor suction (after mixing of economiser and evaporator vapour), and one on the liquid offtake from the high-pressure liquid receiver line. Gas and liquid samples were acquired over a period of several days of operation and analysed by gas chromatography, using the same procedures and methods as for quality control during zeotropic refrigerant manufacture.

Three sets of samples were taken with the refrigerant level in the evaporator at its normal operating level. These data are presented in Table 1, together with the prediction of the simulation model. The agreeement is about 1 - 1.5% w/w on each component composition which is in line with the accuracy observed in earlier work [1,3]. The samples are also consistent with each other, showing that the plant was operating at steady state.

Table 1: Circulating refrigerant composition at normal operating level

Sample Set	Sample point location	R-32 %w/w	R-125 %w/w	R-134a %w/w
1	Evaporator outlet (vap)	25.2	49.1	25.8
1	Compressor suction (vap)	24.9	48.9	26.2
1	HP receiver outlet (liq)	25.7	49.5	24.9
1	Average over all samples	25.3	49.2	25.6
2	Evaporator outlet (vap)	25.1	49	25.8
2	Compressor suction (vap)	25	49.1	26
2	HP receiver outlet (liq)	25.6	49.5	24.9
2	Average over all samples	25.2	49.2	25.6
3	Evaporator outlet (vap)	25.1	49.1	25.8
3	Compressor suction (vap)	25.1	49	25.9
3	HP receiver outlet (liq)	26.3	49.8	23.9
3	Average over all samples	25.5	49.3	25.2
MODEL 1-3	Predicted by model	24.8	47.6	27.5

The refrigerant level setpoint was then adjusted so that the evaporator was running with the level just above the low alarm point. The plant was allowed to stabilise and a set of samples were taken. The level setpoint was then adjusted upwards so that the evaporator was running just below the high level alarm point, allowed to stabilise and the refrigerant composition was sampled again. In each case the change in evaporator mass holdup was about 30% of the normal holdup. The data from these measurements, together with the model predictions are shown in Table 2.

The measured compositions show that as the level in the evaporator rises the circulating composition is enhanced in R-32 and R-125. This is in line with the earlier predictions of Corr et al. as outlined above. The agreement with the predictions of the model is also good , giving confidence that the model may be used to assess plant performance with reasonable accuracy.

Table 2: Circulating composition for high and low evaporator liquid holdup

Sample Set	**Sample point location**	**R-32** %w/w	**R-125** %w/w	**R-134a** %w/w
4 (low level)	Evaporator outlet (vap)	20.9	49.7	29.4
4 (low level)	Compressor suction (vap)	20.7	50.1	29.2
4 (low level)	HP receiver outlet (liq)	22.9	46.5	30.5
4 (low level)	Average over all samples	21.5	48.8	29.7
MODEL 4	Predicted by model	23.5	45.8	30.6
5 (high level)	Evaporator outlet (vap)	26.3	50.5	23.2
5 (high level)	Compressor suction (vap)	26.4	50.5	23.1
5 (high level)	HP receiver outlet (liq)	29.9	52.1	18
5 (high level)	Average over all samples	27.5	51	21.4
MODEL 5	Predicted by model	26.1	49.3	24.7

Performance Measurement

The refrigeration cycle performance was estimated by performing a heat and mass balance over the plant for several sets of operating data. The system Coefficient of Performance (COP) was calculated to be a value of ~2.1 and was invariant over the range of circulating compositions studied. (This encompassed a variation of ~20% in delivered duty from normal operating point). The estimated COP at the design point with R-22 was 2.05 (design conditions -31/34°C (0.6/12.2 barg).The system has historically been operated at loads ranging from 50% to over 100% of design with no noticeable problems either in ability to meet load or in stability.

CONCLUSIONS

An industrial refrigeration system designed for R-22 has been successfully operated with R-407A over a period of years. The system was modelled to incorporate composition shift effects, and the model was verified by measurement of the circulating composition over a range of operating conditions. The predictions of the model agreed well with the measured data. Performance of the unit is close to that which would have been expected using R-22.

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FIGURES

Figure 1: Schematic Of Refrigeration Plant