1996

Wide-Boiling Refrigerant Mixtures - Technical and Commercial Challenges

G. G. Haselden
University of Leeds

WIDE-BOILING REFRIGERANT MIXTURES - TECHNICAL AND COMMERCIAL CHALLENGES

Geoffrey G. Haselden - University of Leeds

ABSTRACT

The scope for large power saving by the use of wide-boiling refrigerant mixtures is illustrated by reference to air conditioning. A novel flow circuit is described which is self-optimising for any mixture and operating load. It uses a proportional liquid injection system coupled with a modulating float valve. The system is modelled using the NIST REFPROP data base, and a number of refrigerant blends. Modest increases of heat transfer surface are needed to exploit mixed refrigerants, and these give COP improvements of about 30%. This prediction was confirmed by a laboratory test using the R407C blend.

Similar improvement are predicted for water chillers especially if allowance is made for part-load operation by reducing compressor displacement. Other fields of application are vehicle air conditioning and process industry duties.

Introduction

Wide-boiling mixed refrigerants are potentially attractive when the duty involves cooling a flowing fluid through a temperature range of at least 5°C and preferably 10 to 20°C. An additional bonus arises if heat rejection in the condenser is to a fluid which, for independent reasons, should rise in temperature by a similar amount. Wider temperature ranges or dissimilar temperature ranges (in the evaporator and condenser) can be handled by multi-staging if the application is on a large enough scale.

Practical realisation of the thermodynamic benefits of wide-boiling mixtures requires more than just the matching of temperature profiles in the evaporator and condenser. It requires very close attention to the heat transfer processes and a new control system which optimises the performances of all the plant components over the required range of operating conditions. The refrigerant mixture must be environmentally acceptable, and the overall unit should cost little more than the pure refrigerant unit it replaces.

Air conditioning offers many potential applications of wide-boiling mixtures which meet the stated requirements. However the first cost of a mixed refrigerant unit will normally be larger because the condenser and evaporator will generally be larger and their extra cost will exceed the savings with the compressor and drive. The value of the power saved will be greatly influenced by the demand place on the unit. An air conditioner operating in a tropical climate, and running at full capacity for most of the year, will yield great savings. A similar unit operating in a moderate climate requiring cooling mostly in daylight hours for only a few months of the year may give only a trivial saving. The relative cost per kWh of electric power and the type of tariff will also be relevant. At the present time most air conditioning units are installed in areas of temperate climate and first cost dominates, but the balance is sure to change.

Mixed Refrigerant Flowsheet and Control System

Figure.1 illustrates the wide-boiling mixed refrigerant system, developed by the Author at the University of Leeds, applied to packaged air conditioning (1, 2). Superheated refrigerant vapour mixture leaving the compressor passes via an oil separator to a manifold which distributes vapour equally between tubes in the entry row of a fin-coil condenser. Usually there will be multiple passes in the first row prior to the refrigerant vapour passing to the second and subsequent rows. The example given has five rows of vertical tubes passing through horizontal louvered fins. The air enters at the opposite face and the resulting geometry provides essentially countercurrent contact between refrigerant and air.

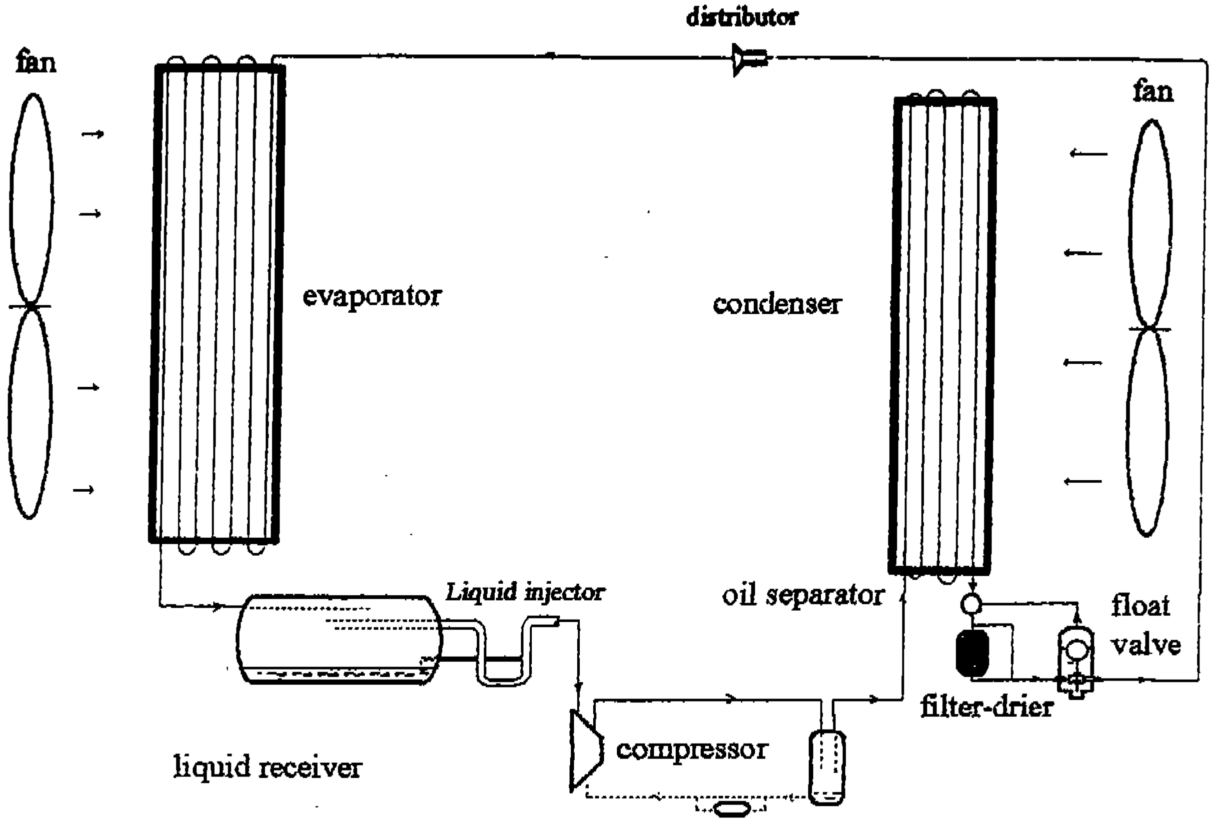

Figure 1. Flow diagram of a mixed refrigerant air conditioning unit using the new control strategy

Vertical tubes are used to encourage the refrigerant phases to travel together with good mixing, and this objective is further encouraged by internal rifting of the tubes, and by Tee-ing tubes together between rows. Thus the effective flow area is reduced as the refrigerant volumetric flowrate goes down. It is vital that the flow through the final tubes is downwards, and that the rate of condensation in them is sufficient to drag condensate over the final return bends.

The condensate flows to a novel modulating float valve which ensures that the condenser is fully utilised, whatever the duty, and that the evaporator receives a steady 2-phase feed. It employs the double orifice, opposed flow, principle to balance the pressure thrust on the valve shaft, leaving the float to carry only the gravitational load. The double orifice principle is widely used in motorised valves, but not (to the Author's knowledge) in float valves. In the valve built and tested in this work the two identical tapers were 20mm long and discrimination was within ± 0.2mm. The needle shaft incorporated a dash-pot action to reduce instabilities.

The evaporator design is similar to that of the condenser. The tubes are vertical and the 2-phase velocity was chosen to be sufficient to ensure the maintenance of annular flow. The refrigerant feed is divided amongst 16 tubes in the first row. This was achieved using conical upward-flowing distributors in 2 stages of 4 to 1 each.

The evaporator has seven rows, each with 16 internally rifled tubes. Since the volumetric flowrate changed by a factor of less than 5, no T-junctions were needed. At turn-down instability would start in the first row of tubes, so it is desirable that the flow direction in these should be downwards.

The control system was designed so that the evaporator discharge had a wetness of about 2%. Experience with the 2-section evaporator (1), previously described by the Author, had shown that this liquid content was sufficient to maintain a high boiling heat transfer coefficient with rifled tubes. The challenge was to control the discharge wetness without the complexity and potential instabilities of the 2-section evaporator. The chosen method (4) employed a proportional liquid injection system coupled with a low pressure separator/receiver into which the evaporator discharge drains (Figure.1).

The exit vapour line from the liquid receiver is designed so that in normal operation the pressure drop along it, to the point of liquid injection, is about 1.3 kPa (or 100mm liquid head). Liquid injection occurs through a length of capillary tube and at a vertical height a little above the normal liquid level in the receiver. The length and bore of the capillary tube are chosen so that, when subjected to a pressure differential of 1.3 kPa, it transmits a liquid flow equal to 2% of the total mass flow through the compressor. Thus the injection system maintains a constant proportional liquid content in the compressor suction independent of the duty. Due to the U-bend in the exit vapour line, there is no possibility of uncontrolled draining of receiver liquid into the compressor, even at shut-down. The injection system also provides for the return of oil to the compressor.

Referring to Figure.1, because the proportional injection system is operating in a closed system, under steady state conditions the intake to it from the evaporator must equal the discharge from it to the compressor. Thus the evaporator pressure must adjust itself automatically to achieve 2% wetness - or whatever wetness fraction is designed into the system.

For maximum thermodynamic efficiency the controlled injection of liquid into the compressor suction should allow liquid droplets to penetrate into the actual compression space and not be evaporated in the suction manifold, and certainly not in the drive motor windings. In this way the superheat at the end of compression will be reduced and the effective suction volume will be increased. A wet suction is thermodynamically sound.

This novel control system is also applicable to pure refrigerant systems and would lead to smaller evaporators, condensers and compressors for the same duties or power reductions with the same sized units. It is even more attractive with mixed refrigerants because it is composition independent as well as self-optimising. Variants (4) are possible with even higher efficiencies.

Mixture Selection

To obtain the maximum power saving with wide-boiling mixtures the refrigerant temperature glide in the evaporator must approximately equal the required cooling range. An exact fit is not important, and it is better that the glide should be marginally less and not more than the duty. The evaporator glide does not start from the bubble point due to flashing after the expansion valve. For economic reasons it is desirable that the evaporator pressure should be in the range 5-7 bar. Another relevant factor is that for an approximately equimolar mixture of components, obeying ideal solution laws, the temperature span between the dew and bubble points is approximately one third of the difference between the boiling points of the pure components at the same pressure. It is also necessary to take into account the temperature difference for heat transfer (say 5°C). Thus if the duty involves cooling air from 26 to 13°C the refrigerant glide needs to be from 21 to 8°C (13°C). So the high boiling component needs to have a boiling point of 21 + 13 = 34°C, and the low boiling component 8 - 13 = -5°C, both at evaporator pressure of 6 bar. Suitable components might be R125 for the low boiling component but no common refrigerant has a b.p of 6 bar at 34°C. Instead an equimolar mixture of R152a and R600a would give this b.p, so that the feed mixture might be 50% R125, 25% R152a and 25% R600a.

Matching the temperature profiles in the condenser is not necessarily important since the achievement of minimum power will normally require that the maximum amount of coolant is used, determined by fan or pumping power. Thus the optimum condenser design will often involve the coolant rising in temperature by a smaller amount that the condensing range of the refrigerant. The need to remove vapour superheat further exacerbates the matching challenge. Even so the irreversibility of the mixed refrigerant condenser will generally be less than with a pure refrigerant.

Modelling Studies

The mixed refrigerant model developed at Leeds was pioneered by Juifa Chen (3) and revised by Bensafi and Robert Lamb (present student) who has installed the NIST REFPROP database on-line. For a specified cooling duty and coolant availability it can be used as a design program to calculate power requirement and heat transfer areas using specified heat transfer coefficients and pinch values. Alternatively heat transfer areas can be specified and the performance calculated. Counter-current contacting is assumed. In both procedures the % wetness of the evaporator discharge must be specified, leading to wet compression at the same value. The program generates temperature profiles in the condenser and evaporator. It also handles oil return.

The rating program applied to a standard cooling duty is illustrated for a range of mixtures in Table.1. Normally the duty involves cooling air from 26.7°C and 50% RH to 14.1°C with moisture deposition. The rate of air flow is adjusted to bring the refrigerant compressor suction volume to a standard value of approximately $13.5\,m^3/h$. The condenser airflow is generally taken as 2.0kg/s at 35°C. In many cases the fin areas of the condenser and evaporator are each taken as $68\,m^2$ - these being the values for the test rig in the Authors' laboratory. Thus the effect of mixture choice on performance can be tested for the same hardware in similar duties, and can be verified experimentally. For these duties a normal R22 A/C unit will have a COP of about 3.5.

Table 1.
Predicted performance data for the laboratory rig with various blends and heat transfer areas.

REFRIGERANT COMPOSITION Wt. %	EVAP. AREA m^2	COND. AREA m^2	EVAP.AIR FEED kg/s	COOLING DUTY kW	COMP. WORK kW	COP	SUCTION VOLUME m^3/h
R125, 50% R152a, 25% R600a, 25%	68	68	0.90	13.6	2.49	5.5	13.1
R407C	68	68	1.25	17.4	3.74	4.6	13.5
R32, 31% R125, 32% R134a, 37%	68	68	1.25	18.9	4.46	4.2	13.6
R32, 35% R227, 65%	68	68	1.27	19.2	4.24	4.5	13.4
ditto	50	68	1.20	18.1	4.19	4.3	13.5
ditto	80	80	1.31	19.8	4.10	4.8	13.3
ditto	100	100	1.38	20.8	4.04	5.2	13.4
ditto	100	68	1.27	13.9	2.02	6.9	7.8

Line 1 is for the refrigerant mixture derived above. Due to the high specific volume of this blend the air feed to the evaporator is only 0.9kg/s. The assumed overall heat transfer coefficients, based largely on laboratory measurements with fin coils, are:

Removal of vapour superheat: $10\mathrm{W/m^2}$ DEGC
Condensation : $65\mathrm{W/m^2}$ DEGC
Evaporation : $41\mathrm{W/m^2}$DEGC

It is seen that this blend gives a cooling duty of only 13.6kW instead of about 19kW for R22 in a standard unit with the same compressor displacement, and a total fin area of about $100\mathrm{m^2}$, but the COP has risen to 5.5 - more than 50%.

By contrast the second line tests a narrower boiling 407C blend in the same hardware. Due to the lower specific volume and higher evaporator pressure (7.35 compared to 5.78 bar) the cooling duty has risen to 17.4 but the COP has dropped to 4.6 (though still a gain of more than 30% compared to pure R22. These calculations assume that the composition of the fluid circulating is the same as that of the feed, which will not be true. Depending on the relative amounts of refrigerant stored in the receiver, and in circulation, the composition of the latter will change.

The third line of the table uses an estimate of the composition of the circulating refrigerant (from an R407C feed). A laboratory test rig using an R407C feed gave a COP of 4.5 under fairly similar operating conditions, and confirmed the reliability of the model.

The remaining lines in the table are for a thermodynamically desirable blend of R32 and R227 with different surface areas for heat transfer. With the same heat transfer areas it is marginally better than R407C. With reduced area the performance suffers, but with areas of $100\mathrm{m^2}$ for both units a COP of 5.2 is possible. Similar advantages could be achieved by using enhanced heat transfer coefficients without increasing the area.

The last line introduces an additional way of making large power saving, using the proportional liquid injection system and wide-boiling mixtures, if the cooling duty at part load is matched by control of compressor displacement.

In this example it is assumed that ambient temperature has dropped from 35 to 33°C and the smaller cooling load is met by maintaining the same air flowrate through the evaporator but allowing the return temperature to rise from 14.1 to 16°C. This could be achieved by reducing the compressor displacement to about 60% of its maximum value. Under these conditions, where all the heat transfer surface is being used effectively for a smaller duty, the COP rises to 6.9 - double the normal pure refrigerant value.

Thus the possibilities for exploiting mixed refrigerants to achieve power savings in air conditioning are exciting.

Water Chilling

On the face of it water chilling appears to offer little scope for wide-boiling mixtures because the cooling range is small (typically 11 - 6 = 5°C. However this is compensated by the higher heat transfer coefficients so that pinches are smaller.

A typical water chiller with the following specification is examined.

Cooling capacity 70kW
Chilled water in 11°C, out 6°C, flowrate 3.35kg/s
Condenser water in 30°C, flowrate 2.2kg/s
Surface area, evaporator $7\mathrm{m^2}$, condenser $14\mathrm{m^2}$
COP 3.2

An appropriate mixed refrigerant for this duty is R134a 55% and R32 45%. The predicted performance of a water chiller based on these data and with a few surface areas is presented in Table 2.

Table 2.
Predicted performance of water chiller using a 45% R32/55% R134a blend for the specified duty.
The bottom line is for 70% loading, the compressor displacement being reduced to 60%.

EVAP. AREA m^2	COND. AREA m^2	EVAP. PRESSURE bar	COND. PRESSURE bar	COOLING DUTY kW	COMP. WORK kW	COP	COMP. VOLUME m^3/h
7	14	5.2	20.1	70	22.0	3.2	76
14	27	6.1	16.9	70	15.2	4.6	60
18	23	6.3	17.2	70	15.0	4.7	58
18	23	6.8	15.0	49	7.9	6.2	36

With the original areas the performance with mixed refrigerants shows no improvement. Virtual doubling of the areas reduces the power consumption by 30%, whilst redistributing the same total area more in favour of the evaporator raises the saving to 32%. The final line again shows the very large savings possible under part-load conditions if the compressor displacement is reduced to match the lower cooling load, whilst the water feed to the condenser drops from 30°C to 28°C. Here again the proportional liquid injection system ensures that all the heat transfer surface if fully exploited despite the lighter load.

Other Applications

Vehicle air conditioning is a difficult application because the emphasis must be on achieving the most compact possible designs of condensor and evaporator, whilst the cost of fuel needed to drive the compressor is not apparent to the driver. This will change when electric vehicles become common. Possible designs for the condenser and evaporator for mixed refrigerant vehicle use have been proposed (5), using the vacuum brazed aluminium plate and fin construction methods currently employed.

More obvious applications of mixed refrigerants arise in process industries. In the chemical industry there are many plants in which a flowing fluid must be cooled before being admitted to a distillation column or reactor. These plants normally run continuously and the exit temperature must be precisely controlled despite changes of ambient temperature. The author is exploring such an application in which a compressed gas stream, leaving the aftercooler at any temperature between 15 and 25°C, is to be cooled to 5°C with deposition of moisture. The available cooling water temperature also varies throughout the year. The existing ammonia unit removes all the heat at a temperature of about 2°C and uses conventional control methods with a fixed head pressure.

The use of mixed refrigerants coupled with the control system described in this paper indicate that the power cost over the year might be halved.

References

1. Chen J., Haselden G.G. Mixed refrigerants for air conditioning and heat pumping, Proc. Inst. Refrig., (1982/3) 89, 27-38.
2. Bensafi A., Haselden G.G. Wide boiling refrigerant mixtures for energy saving, Int. Journal Refrig., 1994, 17, 469-474.
3. Haselden G.G., Chen J. A computer simulation program for mixed refrigerant air conditioning, Int. Journal Refrig., 1994, 17, 343-350.
4. Int. Patent Application No. PCT/GB/02982.
5. Int. Patent Application No. PCT/GB/02983