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CONTRACT NUMBER   96/001

Phasing out ODS at Alaska, Helwan Company for Metallic Appliances, Iberna, El Nasr Company for Electr and Electr Apparatus, Siltal and Super Bosh Domestic Refrigeration Plant in Egypt (Project No MP/EGY/95/038)

Project report by J. Ellis, F.Inst.R., F.I.Diag.E., LCGI
Ellis Training Consultancy

Saturday 13th January 1996

Travelled from UK to Cairo. Briefing meeting with Cameron Murdoch.

Sunday 14th January 1996

Accompanied Cameron Murdoch and Mrs. Salwa El Tayeb to the Egyptian Environmental Affairs Agency (EEAA). In discussion the programme for the workshop was formulated. Visits to the Iberna and Super Bosh factories were made followed by a meeting at the offices of Dr. Abdel Rahman Fahmi a consultant to UNDP and EEAA to discuss the training workshop and the industrial situation in Egypt.

Monday 15th January 1996

Delivered training workshop sessions as per the attached programme. The delegates took an extremely active part in the discussion sessions, asking a wide range of questions to which full answers were given. Each delegate was issued with a copy of the training material enclosed.

Tuesday 16th January 1996

Accompanied Cameron Murdoch and Mrs Salwa El Tayab to Helwan Company for Metallic Appliances to provide consultancy and to discuss training need. Attended Dr Abdel Rahman Fahmi's office together with Mrs Salwa El Tayeb, Cameron Murdoch and Dr Tomas Grof to discuss the feedback from the training workshop and further training needs in the domestic refrigerator sector during the implementation of the project.

Wednesday 17th January 1996

Returned from Cairo to UK

UNIDO Programme
for Domestic Refrigeration Workshop
****

Monday, 15 January, 1996
Pyramisa   Hotel, Cairo.

09:00 - 09:30	Registration

Opening Session:

09:30 - 09:45	Presentation by EEAA Representative
09:45 - 10:00	Presentation by UNIDO Representative

Workshop Sessions

10:00 - 11:00	Introduction of Refrigerant 134-a, its use and a comparison between R134-a and R-12
11:00 - 11:10	*Discussion Session by Dr. O. El-Kholy*
11:10 - 11:30	**Coffee Break**
11:30 - 12:30	Lubricants Necessary with R134-a Handling Refrigerant134-a and Polyolester Lubricants
12:30 - 12:45	*Discussion Session By Dr. T. Grof*
12:45 - 13:45	Good Production and Service Practices
13:45 - 14:00	*Discussion Session By Dr. A. Fahmy*
14:00 - 15:00	**Lunch**
15:00 - 16:15	Optimizing Refrigerator Systems for Energy Efficiency Capillary Tube Restrictor
16:15 - 16:30	*Discussion Session*
16:30 - 17:00	Conclusion Session

ELLIS
TRAINING CONSULTANCY

HFC-134a

TRAINING WORKSHOP

for

DOMESTIC

REFRIGERATION

15/16 January 1996

REFRIGERANT R134a

R134a is a hydrofluorocarbon (HFC) refrigerant developed to replace R12 while producing no risk to the stratospheric ozone layer.

It is necessary to be aware of the hazards to health created by this refrigerant in order that safety of personnel can be assured.

POTENTIAL HAZARDS

Asphyxiation

Burns

Decomposition in naked flames

Narcotic sensitisation

ASPHYXIATION

Because R134a is heavier than air it would sink to the floor and build up concentration in confined spaces. This may cause displacement of breathable air leading to dizziness, nausea and possible fainting. If a person becomes unconscious in heavy concentrations of refrigerant death may follow.

BURNS

When liquid refrigerant R134a contacts the skin or eyes it boils off at -26.5°C leading to severe burns to the skin and possible blindness if contact with the cornea is prolonged.

DECOMPOSITION IN NAKED FLAMES

Although R134a is non-flammable toxic decomposition products may be given off when in contact with naked flames. Acid fumes which are irritant are given off first, then phosgene which is highly toxic.

NARCOTIC EFFECTS

At high concentrations, normally in excess of 15% by volume, exposure to the vapours can cause a feeling of well being that is, or can be dangerously misleading.

CARDIAC SENSITISATION

At high concentrations, normally in excess of 15% by volume, exposure to the vapours can cause irregularities in the heart beat which in severe cases can cause heart attack and possibly death.

Proper use of personal protective equipment such as goggles and rubber gloves help to reduce the hazards, also ensuring that the workplace is properly ventilated helps to reduce the risks to health that can otherwise be caused by losses of refrigerant into the workplace.

Cylinders should be stored in well ventilated areas, out of direct sunlight and away from local heat sources. These cylinders should be restrained and not standing in water. Adequate warnings should be displayed to warn people of their presence.

Refrigerant should not be deliberately released into the atmosphere at any time. During service work it is necessary to use a recovery machine to withdraw the refrigerant from the system into a safe cylinder so that it can be re-used or disposed of safely.

When handling refrigerants from systems that have suffered electrical burn out, be aware that refrigerant and oils may be acidic and thus pose a risk to health.

REFRIGERANT/REFRIGERATION OIL COMPATIBILITY

R134a is NOT miscible with the traditional mineral oils that were popular with CFC refrigerants. Polyolester lubricants must be used, the compressor will already be charge with polyolester lubricant if it is designed for use with R134a.

R134a is very hygroscopic and polyolester lubricants are approximately 100 times more hygroscopic than mineral oils.

With domestic refrigeration systems utilising a capillary tube restrictor as the expansion device it is very important to dehydrate the system before the refrigerant is charged into it. The main problem caused by moisture in such a system is a blockage of the capillary tube which can lead to failure of the system compressor.

In the presence of moisture R134a hydrolyses with the lubricating oil to form acids which attack the motor winding insulation. Also moisture, oil and dirt particles stick together causing blockages which lead to failure.

Cleanliness of the system is always extremely important, it becomes even more critical with capillary tube systems particularly when these systems are utilising R 134a and polyolester lubricating oil.

Attention to detail during the manufacturing process is critical, systems should not be left open to the atmosphere at any time, careful control of the quality of polyolester oil should be maintained. Compressors should be sealed until the last possible moment and then careful dehydration and evacuation should be implemented to reduce the moisture content in the system to below 50 PPM.

WORKING WITH R134a

LEAKAGE AND LEAK TESTING

Because of its physical makeup, R 134a is more searching than R 12. That is, it leaks more easily and therefore joints have to be extremely 'tight' before the refrigerant is charged into the system.

Traditional leak detectors searched for Chlorine in the process of leak detection, the Chlorine reacted to give a visual or audible signal of its presence.

There is no Chlorine in R134a and consequently different leak detectors are required. R 134a is a Fluorinated Hydrocarbon and consequently leak detectors that search for Fluorine are required.

Traditional leak detectors do NOT find leaks on R134a systems.

It is recommended that the system is leak tested prior to charging the refrigerant. Oxygen free Nitrogen can be used to provide the pressure, with joints tested using soap and water. Trace refrigerant leak testing is acceptable as a last resort if necessary.

EVACUATION AND DEHYDRATION

Reducing the system pressure sufficiently to allow the heat from the atmosphere to vaporise any moisture left in the system is necessary before carrying out the charging process.

The saturated water and steam table included show the relationships between temperature and saturated pressure. It will be necessary to allow for a reasonable difference between the ambient temperature and the saturation temperature so that the moisture can vapourise and be withdrawn by the vacuum pump.

The efficiency of the vacuum pump is affected by:-

    quality of the oil in the vacuum pump

    length and resistance of the connecting lines

    temperature of the system being evacuated
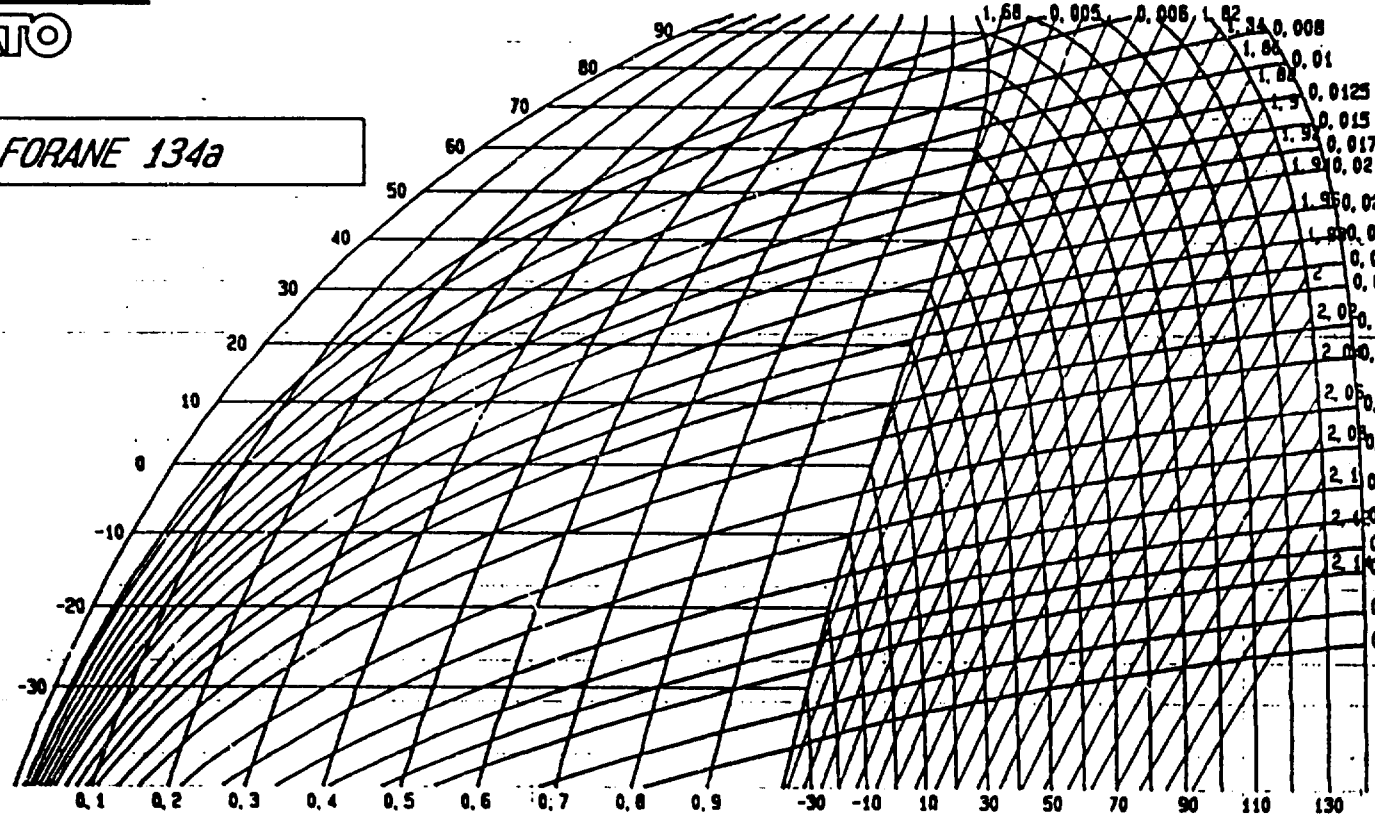
elf atochem

ATO

FORANE 134a

UNITES   P : bar   T : °C   REFERENCES   T=0°C   h=200kJ/kg   s=1kJ/ kg.K

h : kJ/kg   s : kJ/kg.K   v : m^3/kg   T eb : -26,4   Pc : 40,7   Tc : 101   d (25°C) : 1,210

REFRIGERANT PROPERTY COMPARISON

R134A V R12

		R134a	R12
CHEMICAL FORMULA		$CH_2\ F\ CF_3$	$CCl_2\ F_2$
Molecular Weight	9/mol	102	120.9
Boiling Point at 1 atm	°C	-26.2	-28.9
Freezing Point	°C	-101	-158
Critical Temperature	°C	101	112
Critical Pressure	bar	40.55	41.15
Critical Volume	m^3/kg	0.00196	0.00179
Specific Heat of Liquid at 25°C	kj/kg/k	1.42	0.98
Specific Heat of Vapour at 25°C	kj/kg/k	0.86	0.68
Density of Liquid at 25°C	kg/m^3	1206	1310
Density of Saturated Vapour at Bpt	kg/m^3	5.26	6.33
Latent Heat of Vapourization at Bpt	kj/kg	177.33	165.26
Thermal Conductivity Liquid at 20°C	W/mlk	0.0871	0.0727
Thermal Conductivity of Vapour at 20°C	W/m/k	0.0145	0.0097
Viscosity of Liquid at 20%	cP	0.2225	0.256
Viscosity of Vapour at 20°C	cP	0.0124	0.0127
Surface Tension	kN/m	8.6	9.0
Solubility in Water at 20°C 1 atm	%wt	0.077	0.028
Solubility of Water in Refrigerant at 20°C	%wt	0.056	0.009
Critical Density	kg/m^3	509.2	558
Vapour Pressure at 25°C	bar	6.621	6.516

REFRIGERANT RECOVERY

When refrigerators are to be serviced it is necessary to remove or recover the refrigerant from them.

A recovery machine, suitable for R134a should be used to withdraw the refrigerant via the process tube and to pump it into an appropriate cylinder ready for recycling.

Note that the capillary tube slows the process of recovery and must be taken into account.

REFRIGERANT RECYCLING

This is the process of cleaning the refrigerant by filtration and separation to remove oil, acids and moisture to improve its quality prior to re-use.

REFRIGERANT RECLAIMATION

This is the process of returning the refrigerant to its original virgin condition in a specially built machine. Before and after processing the refrigerant would be analysed in a specialist laboratory.

CAPILLARY TUBES

The capillary tube serves almost all small refrigeration systems with capacities of up to about 10kw. Lengths vary from 1m to 6m and inside diameters or bores from 0.5mm to 2mm. In reality the bore is too large to permit capillary action.

Liquid refrigerant enters the capillary tube from the condenser, and as it flows through the tube the pressure drops because of friction and acceleration of the refrigerant. Some of the liquid flashes to vapour as the refrigerant flows through the tube.

Numerous combinations of bore and length are available to obtain the desired restriction, but of course once the capillary tube has been selected and installed it cannot itself adjust to variations in discharge pressure, suction pressure or load.

The compressor and capillary tube must arrive at suction and discharge conditions which allow the compressor to pump from the evaporator the same flow rate of refrigerant that the capillary feeds to the evaporator i.e. in balance. Any condition of unbalanced flow will of course be temporary.

The mass flowrate of the capillary can be plotted on the same graph as the mass flow rate of the compressor, as shown in the diagram, Fig.1. The dashed line represents the flow through the capillary, the solid line the compressor mass flow rate.

At high condensing pressures the capillary feeds more refrigerant to the evaporator than it does at low condensing pressure because of the increase in pressure différence across the tube.

There are other factors which affect this balance however which complicate the picture. If the evaporator heat transfer is not satisfied at the compressor/capillary balance point, an unbalanced condition results which can starve or overfeed the evaporator.

Starving the evaporator results when the suction pressure rises and the capillary does not feed sufficient refrigerant to refrigerate the evaporator surfaces adequately. Fig. 2 shows a balance point for a constant condensing pressure between the compressor and the capillary tube at suction pressure A. A heavy heat load, when received at the evaporator causes the suction temperature and pressure to rise to B. At suction pressure B the compressor can remove more refrigerant from the evaporator than the capillary can

supply, so the evaporator becomes short of refrigerant and the suction pressure starts to drop. In the meantime there is less refrigerant entering the capillary and liquid starts to back up in the condenser, reducing the effective surface area and causing the condenser pressure to rise. With the increased condensing pressure, the compressor capacity is reduced and the capillary tube rate of flow is increased and the balance is restored.

If the refrigeration load drops off to less than the refrigeration capacity at the balance point, the suction pressure and temperature drop to some point C. At this point the capillary tube can feed more refrigerant to the evaporator than the compressor can draw out. The evaporator fills with liquid and at some point the condenser has drained and vapour enters the capillary to reduce the flow and restore the balance at D. Although D represents balanced flow, it is not satisfactory because the state of the refrigerant entering the capillary is two phase which reduced the refrigerating effect compared with when saturated or subcooled liquid enters the capillary.

Fig. 3 shows that although each kilogram of refrigerant provides a reduced refrigerating effect, the work done or energy input per kilogram remains the same.

Capillary tubes are often installed in heat exchange with the suction line to retard the flashing of refrigerant flowing through the capillary tube.

There are advantages and disadvantages with capillary tubes.

Advantages - simple, no moving parts, allow pressure equalisation during off cycle so that low start torque motors can be used.

Disadvantages - not adjustable to changing load conditions, susceptible to clogging or blockage by foreign matter and require that the mass of refrigerant charge is critical (within close limits)

The last disadvantages mean that the capillary is only suitable for hermetically sealed systems manufactured in carefully controlled environments and to high quality standards.

Capillary tubes are designed for one set of operating conditions and change in the applied heat load or condensing temperature will result in a decrease in operating efficiency.

SELECTION OF CAPILLARY TUBES

Although calculations are available to design the correct length and bore of capillary tubes, the optimum length will be found by 'cut and try' tests carried out on the actual refrigerating system.

Capillary tubes are specified according to the inside diameter, length and volume flow rate with an inlet (condensing) pressure of 10 bar in relation to the compressor size and the evaporating temperature range. (DANFOSS)

The volume flow rate V (in l/min) can be worked out according to the following formula:

$$V = 2.5 \sqrt{120} \times \frac{d^{2.5}}{L^{0.5}} \text{ l/min}$$

where   L = length in metres
         d = inside diameter in mm

Alternative capillary tubes can be selected according to the diagram on the following page on the basis of volume flow.
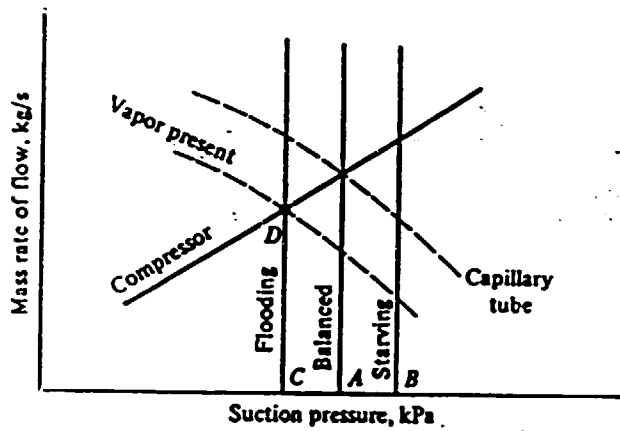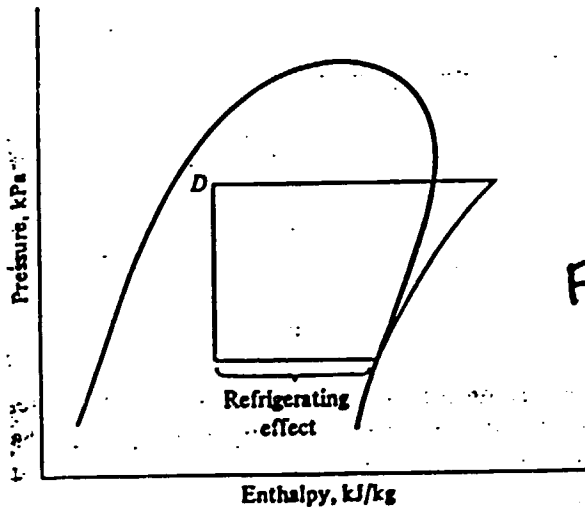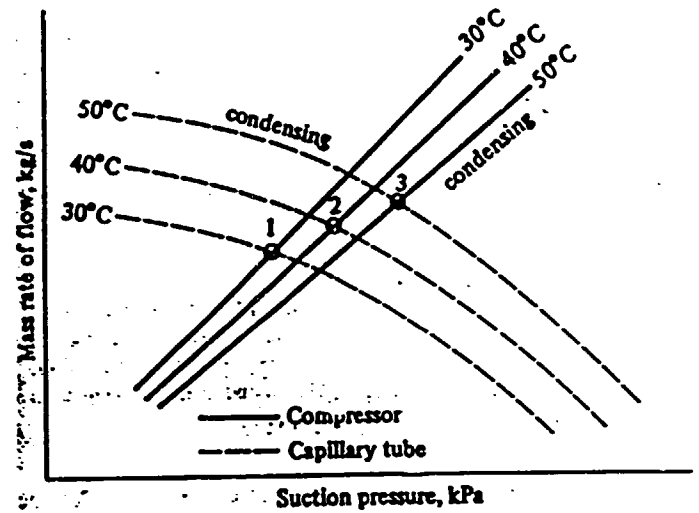
Fig 2

Fig 1

Fig 3

Capillary-tube calculations

Position	Temperature, °C	Pressure, kPa	x	Specific volume, m³/kg	Enthalpy, kJ/kg	Velocity, m/s	Increment length, m	Cumulative length, m
1	40	1536.4	0.000	0.000885	249.85	4.242		
2	39	1498.8	0.008	0.000995	249.84	4.769	0.2306	0.231
3	38	1461.9	0.016	0.001110	249.84	5.320	0.2013	0.432
4	37	1425.8	0.023	0.001230	249.84	5.895	0.1770	0.609
5	36	1390.3	0.031	0.001355	249.83	6.496	0.1565	0.765
6-31	....	....	....	....	....	....	....	....
32	9	657.65	0.194	0.007660	249.18	36.71	0.0097	2.089
33	8	637.90	0.199	0.008048	249.11	38.57	0.0085	2.098
34	7	618.61	0.204	0.008452	249.03	40.51	0.0075	2.105
35	6	599.78	0.209	0.008873	248.95	42.52	0.0066	2.112
36	5	581.38	0.213	0.009309	248.86	44.61	0.0049	2.118
42	-1	479.97	0.239	0.01231	248.11	59.00	0.0017	2.137
43	-2	464.50	0.243	0.01288	247.95	61.73	0.0012	2.138
44	-3	449.41	0.247	0.01347	247.77	64.56	0.0007	2.139
45	-4	434.71	0.250	0.01409	247.58	67.50	0.0003	2.139
46	-5	420.38	0.254	0.01472	247.37	70.55	-0.0001	

COMPATIBILITY WITH SYSTEM MATERIALS

In a refrigeration system the oil/refrigerant mixture will be in direct contact with elastomers and plastics contained in system components.

Initially sealed tube tests were carried out by the chemical companies on R134a and various system components and a basis of information was established.

Subsequent tests with an oil refrigerant mixture, however, gave different results depending on the type of oil.

The most promising results so far have been with a combination of mPOEs and R134a.

Hermetic Motor Windings

One of the reasons why hermetic and semi-hermetic compressor manufacturers prefer mPOEs to PAGs is because of their superior compatibility with motor winding insulation materials and varnishes.

Problems were experienced with embrittlement of PET (polyethylene terephthalate) insulation material when PAG lubricants were used with R134a. Sealed tube tests confirmed that PET films must be dried to less than 0.1% wt of moisture content, which is not practical during compressor assembly. Hence the move to mPOE's, which so far are proving compatible with existing materials.

Seals

The testing of seal materials can be misleading. The test results will depend on the manufacturer of the refrigerant, the oil and the seal with regard to their individual specifications.

There are already some discrepancies and it would be advisable to ensure that the compressor manufacturer has verified test results.

A general guide however, is to select 'O' ring seals and other seals manufactured from Neoprene as opposed to Nitrile to prevent adverse swelling.

Flexible Hoses

Nylon should be used as opposed to Nitrile due to the greater permeation of R134a compared with R12.

Reflon '62 hose is fully compatible with R134a.
The permeation rate for the standard ¼" hose is 0.017 gr/24 hours/m at 20 bar pressure difference and at 20°C external ambient.

Initially Polyalkylene glycols (PAGs), used in air compressors were evaluated and found suitable for automobile applications, however, unacceptable system test results were obtained.

The main disadvantages of the PAGs tested were:

o   Separation of R12/oil mixture at 60/80°C.

o   Depolymerisation phenomenon at high temperatures.

o   Poor lubricating properties for aluminium/steel contacts.

o   Incompatibility with current hermetic winding varnishes.

o   High Hygroscopicity (water vapour retention) leading to copper plating and low dielectric properties.

o   Corrosion trends with aluminium, zinc and cadmium.

Current Status for Domestic and Commercial Refrigeration Systems

The current trend is towards synthetic ester technology.

Tests to date indicate that hermetic compressor manufacturers favour the lubricants that have been developed from polyol ester base stocks - POEs and modified POEs (include additives which provide lubricity stability and are non corrosive to copper and other vital system materials).

mPOEs have the following advantages:

o   Good miscibility with R134a from -50 to +150 C.

o   Good thermal stability at high temperatures.

o   Excellent lubricating properties.

o   Low pour points.

o   Low hygroscopicity and good dielectric properties.

o   Compatible with system materials with the exception of some elastomers.

o   Toxicity tests have shown that POEs and mPOEs produce no more adverse effects on health when properly handled and used. The normal precautions of attention to personal hygiene such as washing skin - contact areas with soap and water are adequate.

LUBRICATION

Traditionally, mineral oils have been used for lubrication in refrigeration equipment, particularly R12 systems. They are generally completely miscible in R12 over a wide temperature range. On domestic, automobile and small commercial applications the mineral oil can move in a free manner around the system and return to the compressor without unduly affecting overall heat transfer.

With the requirement to replace R12 with R134a, it was found that due to the immiscibility of R134a with mineral oil it would be necessary either to redesign the system or look for an alternative lubricant which would be miscible with R134a.

The latter alternative was chosen but there is a view by some oil companies that a mineral oil may still be satisfactory on larger plants incorporating oil separators and where there is scope for changing the system design.

There has however been considerable development in alternative lubricants for use in the smaller equipment.

Requirements

A re-evaluation of the requirements for an alternative lubricant was made as follows:

a. Satisfactorily lubrication of the compressor at the lowest cost.

b. Miscibility with R134a equal to that between mineral oil and R12.

c. Good thermal stability of the refrigerant/oil mixture.

d. Compatibility of the mixture with the system materials (metals, plastics, and elastomers).

e. Low toxicity.

Investigations

Synthetic oils were already commercially available and some of the major chemical companies worked in conjunction with the oil and compressor manufacturers, concentrating initially on the main potential markets - domestic refrigeration and automobile air conditioning.
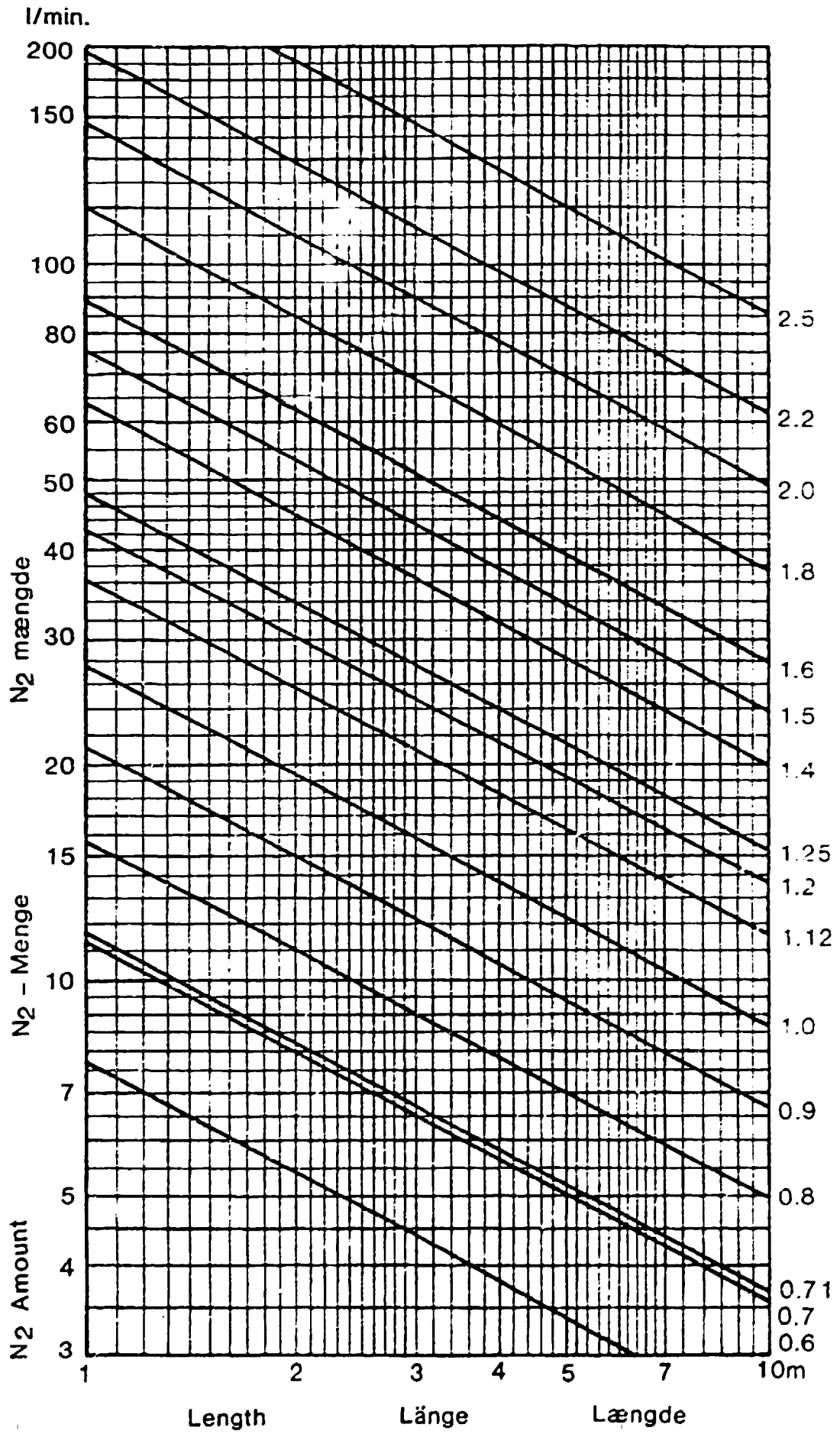
REFRIGERANT / REFRIGERATION OIL COMPATIBILITY

With the increasing number of new Forane® refrigerants now in use, it is important to ensure compatibility between these products and various compressor oils. Care should be taken that the correct lubricant is always used in routine servicing of refrigeration plant and when retrofitting to new products. The following table is a general guide relating to refrigerant/oil compatibilities - users should consult the original equipment manufacturer for specific recommendations if required.

FORANE® REFRIGERANT	LUBRICANT TYPE			
	mineral oil (MO)[1]	alkylbenzene (AB)[2]	MO/AB blend (MO/AB)[3]	polyolester (poe)[4]
12	P	A	A	A
502	A	P	P	A
22	P	A	A	A
134a	NC	NC	NC	P
FX70 (R-404A)	NC	NC	NC	P
407C	NC	NC	NC	P
FX10 (R-408A)	A	P	P	A
FX56 (R-409A)	A	P	P	A
FX57 (R-409B)	A	P	P	A

Legend: P = preferred lubricant choice   A = acceptable alternative
       NC = not compatible

1. typical mineral oils: Suniso 3GS, Texaco WF32, Shell Clavus 68
2. typical alkylbenzene oils: Zerice S68, Zerol 150
3. MO/AB blends: Shell SD, Shell MS 2212
4. polyolester oils: Mobil Arctic EAL 22, Castrol Icematic SW32

If compatible, different mineral oils may be mixed and used in both high and low temperature systems. Similar considerations apply for alkylbenzene oils and MO/AB blends. Polyolesters must be employed if HFC refrigerants, such as FORANES® 134a, FX70 (R-404A), 407C and 23, are used in the system. Manufacturer's guidelines regarding the use of specifically approved polyolester lubricants should always be followed. Polyolesters are also acceptable for use with CFC and transitional products and are compatible with mineral oils. They can therefore be mixed with mineral oils when used in systems with CFC or HCFC refrigerants. The hydroscopic nature of polyolesters means that care must be taken to avoid moisture absorption during handling.

"The complete refrigerant service"

CLEANLINESS

Cleanliness of plant is associated with the expertise of
the initial installation and its subsequent need for
servicing other than routine.

The procedures for new R134a plant are if anything
more stringent than those used for R12, attention being
drawn, particularly to good evacuation and charging
techniques.

It is vital that the system is not contaminated with chlorinated
residues from charging manifolds and vacuum pumps previously
used with R11 or R12, or any refrigerant containing chlorine.

The safe level of chlorinated residues is in the region of
200ppm, but every endeavour should be made to eliminate traces
of chlorine to zero.

In applications where the plant is to be converted from R12
to R134a the situation is more critical as the installation
will contain chlorine and possible chlorine residues
(see section 9).

CONDENSER AND EVAPORATOR PERFORMANCE

R134a compared with R12 has a better thermal conductivity, higher latent heat and higher coefficients of heat transfer. The specific volume of the vapour however, is greater as the evaporation temperature falls below about +5°C (see figure 4).

The consequence of these properties is:

o   A reduction in capacity of about 10-25% at low back
     pressures if an R12 compressor displacement is used,
     due to the higher specific volume.

o   At high back pressures there should be an improvement in
     performance due to the higher thermal properties and lower
     viscosities (possibly up to 10% at + 10% evaporating).

o   Similarly there should be an improvement in performance
     at low back pressure if a compressor with a higher
     displacement is used.

o   With a new installation, optimisation of the evaporator
     and condenser will be necessary to acheive the best balance
     with the compressor.

o   The effect of the lubricant on the above properties
     is however, still under review and the tendency is to
     keep the same evaporator and condenser as selected for R12.

     Friga Bohn recommend that the R12 correction factors
     are applied relative to the compressor manufacturer.

o   Pressure drops in both evaporators and condensers will
     be less than R12.

THE ABILITY OF COMMERCIAL LEAK DETECTORS TO DETECT R134a

New leak detectors have been developed for use with alternative refrigerants including R134a.

This was necessary because the majority of existing types were designed to monitor for the presence of chlorine in a refrigerant molecule. With the absence of chlorine in the hydrofluorocarbon refrigerants the detector has to be designed to be sensitive to a different chemical.

CPS have advised that their model L-790a Leak Seeker is suitable for R134a and detects the fluorine component of the molecule.

TIF have developed a range of three models detecting the fluorine atom. See the addendum for more information.

VACUUM PUMP SIZING

There are three basic parameters to consider other than the commercial and practical considerations when selecting a vacuum pump.

a)   Free air displacement
b)   Vacuum performance
c)   Water vapour tolerance

FREE AIR DISPLACEMENT:

Measured in various units, eg. c.f.m., m^3/h etc. This is usually determined in the original design and subsequently verified in the laboratory using a flowmeter. It is important to appreciate this is measured directly at the pump intake using the manufacturers recommended sized fitting. This is a theoretical figure and is dramatically affected by many factors including length and diameter of connecting hose. Frictional losses through the manifold, fittings, etc. also have an important effect - always negative.

VACUUM PERFORMANCE

Again, measured in various units. Low vacuum pumps are usually rated in inches Hg, medium vacuum pumps in TORR or mm Hg, high vacuum pumps are usually rated in microns. It is interesting that whereas all units are related to the 30" x 1"^{2} column of mercury inches Hg are measured from the top of the column, 29.92" absolute vacuum, whereas microns are measured from the bottom. Another way of expressing it is ins. Hg is the distance from the top of the column to the surface of the mercury while microns is the height of the mercury remaining.

Vacuum Performance is important as it is directly related to temperature at which water will be converted to seam (see steam table). The steam table is commonly misunderstood in that the comparative temperatures are not ambient other than possibly before the dehydration process commences. The change of state from water to vapour involves the transfer

of latent heat as a result of which, once dehydration is initiated, the temperature of the remaining water and surrounding surfaces fall dramatically. Providing the vacuum performance of the chosen vacuum pump is adequate, this presents little problem. When the vacuum developed in the system reaches 4579.6 microns, any remaining water will freeze. Again, this is not a serious difficulty when sufficient vacuum capability is available. Increasing vacuum will convert the ice directly into steam without passing through the liquid phase, this process is called sublimation. However, if the ultimate vacuum capability of the vacuum pump is inadequate, it is recommended to resort to triple evacuation. This is a lengthy procedure involving the breaking of vacuum by admitting refrigerant. The process is repeated twice with a final vacuum.

One of the attendant problems in dehydration work is that as water is converted to steam, a great expansion in volume takes place eg. 1lb of water vapour at sea level occupies a volume of approx. 27 ft^3. Reducing the pressure to 0.18" Hg will cause this to expand to 3300 ft^3.

Should entrained water freeze, the subsequent sublimation further increases the volumetric expansion in the order of nine fold.

This is one of the main reasons why refrigeration compressors are unsuitable for dehydration work, their effective volumetric capacity falls off dramatically with increasing vacuum to a point where no pumping capacity exists. True high vacuum pumps are usually positive displacment machines.

The actual measurement of vacuum is a subject in its own right. The refrigeration industry normally employs three distinct types of gauge: bourdon tube which are relatively inaccurate, Capsule Gauges - more commonly referred to as TORR gauges which represent a considerable improvement in terms of accuracy and the Thermistor Gauge which is particularly accurate. Whichever type is employed, it should be recognised that the indicated vacuum is that which exists at the gauge sensing point. Nearer the vacuum pump, the vacuum will be greater and the furthest point of the system will be at a higher pressure. When the vacuum pump is isolated, the vacuum nearest the pump will deteriorate while the vacuum at the furthest point will normally increase. The exception is when ice has formed within the system. In this situation, the effect of continuing vacuum will convert some, or all, of the ice into vapour with the attendant increase in volume and consequent rise in pressure throughout the system. Ice formation can usually be

identified when using a Thermistor Vacuum Gauge but would be difficult or impossible to detect by reference to less accurate forms of gauge.

WATER VAPOUR TOLERANCE

The refrigeration industry employs vacuum pumps purely for dehydration work. The largest user of vacuum pumps is the production industry, closely followed by the pharmaceutical and food industries, with a significant demand from laboratories and research applications. It is unlikely that in any of these industries the amount of water passing through the vacuum pump is anywhere near as great as in refrigeration applications. For this reason true refrigeration vacuum pumps have been developed specifically to cope with relatively large amounts of water. Industrial vacuum pumps are rarely satisfactory, although they may be perfectly adequate in other terms.

Irrespective of manufacturer and with the exception of the oil less type, which are not true refrigeration vacuum pumps, the internal vapour or vacuum seal is effected by thin oil film. Contamination or breakdown of this oil film will have a serious adverse effect on the actual vacuum capability. It is for this reason that reputable pump manufacturers link their stated performance figures to use of a particular oil specification having an extremely low vapour pressure and freedom from potentially harmful additives.

Water is, of course, a major source of contamination. When this is absorbed by the oil in the vacuum pump, the oil seal tends to break down and performance deteriorates. Modern vacuum pumps incorporate a Gas Ballast arrangement which is simply an air admittance valve into the compression side of the vacuum pump. During one revolution of a vacuum pump gases passing through the pump are subjected to a rapid change from vacuum at the intake port to compression made up of atmospheric pressure, exhaust valve spring load and frictional losses. This rapid increase in pressure encourages some of the water vapour to condense internally, thereby contaminating the oi. By admitting atmospheric air when the gases are compressed, internal condensation is reduced.

As a general rule it is reasonable to assume that reputable manufacturers of vacuum pumps specifically intended for the refrigeration trade design and develop their products to have tolerance to water vapour. Presumably these manufacturers have laboratory figures for their products, although some may not be prepared to divulge these.

PRACTICAL/COMMERCIAL CONSIDERATIONS

As in many fields, the purely theoretical facts can be over-whelmed by other quite valid factors. Insofar as free air displacement is concerned, while no one with practical experience would agree that a smaller capacity pump can dehydrate a system more quickly than a bigger unit, the law of diminishing returns certainly applies. HRP Sales Limited have, for many years, supplied Thorn with 2.5 c.f.m. Robinair Vacuum Pumps for use exclusively on domestic fridges and these have proved eminently satisfactory. However, due to frictional losses between the vacuum pump and the fridge, the time taken would not double by using a pump having only half the cubic capacity. This is clearly illustrated in the Robinair Vacuum Equipment Handbook by reference to the Table I overleaf. Far greater time savings can be obtained by using a Schrader Valve Core Remover, see Table II or, by shortening the connecting hose and/or increasing its diameter.

SINGLE STAGE vs TWO STAGE

Single stage versus two stage format is frequently misunderstood. It is commonly thought that the only advantage of a two stage pump is that it will develop a better vacuum. There is also the important added advan-tage that the two stage model will develop vacuum more quickly than its single stage counterpart. The second stage acts as a 'supercharger' to the first stage, which is in effect a "roughing" pump. In a single format, the vacuum pump develops an increasing vacuum condition at the inlet which pumping against a fixed positive back pressure as mentioned earlier. A second stage converts the exhaust back pressure to a vacuum, thereby increasing the pumping efficiency of the first stage.

When selecting a vacuum pump, commercial consideration such as price, availability and, especially after sales service facilities, should be taken into account. The finest pump in the world at the lowest price will become a useless chunk of metal and a bad investment if it cannot be kept in service for lack of spare parts. The larger Robinair R.V.P. pumps feature field replacement pumping modules, enabling customers to completely recondition their pumps without losing their use.

Again, individual customers may have special requirements - dual voltage for use on building sites, an integral TORR

Gauge or even as part of a complete Service Station. At the
end of the of the day, the prospective customer must take
all factors into consideration but put simply, large enough
for all anticipated applications (1 c.f.m. for domestics,
bottle coolers etc.) 2 c.f.m. for 5 - 15 ton systems and 3
c.f.m. for larger systems. Thereafter, physical size,
weight and cost may outweigh any advantage of larger pumps.

TABLE 1.

¼" TUBING 1.3 CFM TWO STAGE BELT DRIVE		
Length Ft.	Vacuum Microns	Time min.
10	50	94
6	50	60
3	50	34
2	50	24
1	50	14

3.17 CFM TWO STAGE DIRECT DRIVE		
Length Ft.	Vacuum Microns	Time min.
10	50	88
6	50	54
3	50	28
2	50	22
1	50	13

TABLE 2.

Vacuum (Microns)	Core Depressed	Core Removed
500	23.25	2.92
400	25.50	3.25
300	29.00	3.83
200	34.00	5.00
150	38.00	5.92
100	43.00	7.50
90	44.50	7.92
80	45.75	8.42
70	47.50	9.17
60	50.25	10.00
50	52.50	10.75

A vacuum was pulled on a 25 pound refrigerant cylinder by a
3 CFM pump through a flexible hose and depressed valve core
and again with the core removed by an inline valve core
remover.

VACUUM: UNITS OF MEASURE

1 inch = 2.540 cm Hg. = 25.40 mm Hg. = 25,400 microns
1 torr = 1000 microns or .039 in. or 1.0 mm or 0.1 cm
Vacuum scientists have divided the vacuum world into five categories: Low, Medium, High, Very High and Ultra high vacuum.
LOW VACUUM = 759,993 microns to 25,400 microns 29.92" Hg. to 1" Hg.   (COURSE VACUUM)
MED. VACUUM = 25,400 microns to 1 micron 1" Hg. to .0000394" Hg. (REFRIGERATION)
HIGH VACUUM = 1 micron to .001 micron .0000394" Hg. to .0000000394" Hg.

High vacuum equipment for refrigeration service will perform at one micron and below only under controlled conditions as in a laboratory. Such equipment cannot be expected to operate in the high vacuum range in the field.

STEAM TABLE

F Deg.	C Deg.	lb/in^2	in Hg.	mm Hg.	Microns
212	100.00	14.696	29.921	759.993	759993.4
194	90.00	10.169	20.704	525.881	525881.6
176	80.00	6.8699	13.987	355.269	355269.8
140	60.00	2.8900	5.8842	149.549	149458.7
104	40.00	1.0700	2.1786	55.336	55336.4
76	24.44	.44435	.90472	22.980	22979.9
69	20.56	.35084	.71432	18.144	18143.7
59	15.00	.24720	.50330	12.784	12783.8
32	0.00	.08858	.18030	4.580	4579.6
21	-6.11	.5293	.10780	2.738	2738.1
6	-14.44	.02521	.05134	1.304	1304.0
-24	-31.11	.004905	.009987	0.254	253.7
-60	-51.11	.0004972	.001012	0.0257	25.7

Saturated Water and Steam

t [°C]	p_s [bar]	v_g [m³/kg]	h_f	h_{fg} [kJ/kg]	h_g	s_f	s_{fg} [kJ/kg K]	s_g
0.01	0.006112	206.1	0*	2500.8	2500.8	0†	9.155	9.155
1	0.006566	192.6	4.2	2498.3	2502.5	0.015	9.113	9.128
2	0.007054	179.9	8.4	2495.9	2504.3	0.031	9.071	9.102
3	0.007575	168.2	12.6	2493.6	2506.2	0.046	9.030	9.076
4	0.008129	157.3	16.8	2491.3	2508.1	0.061	8.989	9.050
5	0.008719	147.1	21.0	2488.9	2509.9	0.076	8.948	9.024
6	0.009346	137.8	25.2	2486.6	2511.8	0.091	8.908	8.999
7	0.01001	129.1	29.4	2484.3	2513.7	0.106	8.868	8.974
8	0.01072	121.0	33.6	2481.9	2515.5	0.121	8.828	8.949
9	0.01147	113.4	37.8	2479.6	2517.4	0.136	8.788	8.924
10	0.01227	106.4	42.0	2477.2	2519.2	0.151	8.749	8.900
11	0.01312	99.90	46.2	2474.9	2521.1	0.166	8.710	8.876
12	0.01401	93.83	50.4	2472.5	2522.9	0.180	8.671	8.851
13	0.01497	88.17	54.6	2470.2	2524.8	0.195	8.633	8.828
14	0.01597	82.89	58.8	2467.8	2526.6	0.210	8.594	8.804
15	0.01704	77.97	62.9	2465.5	2528.4	0.224	8.556	8.780
16	0.01817	73.38	67.1	2463.1	2530.2	0.239	8.518	8.757
17	0.01936	69.09	71.3	2460.8	2532.1	0.253	8.481	8.734
18	0.02063	65.08	75.5	2458.4	2533.9	0.268	8.444	8.712
19	0.02196	61.34	79.7	2456.0	2535.7	0.282	8.407	8.689
20	0.02337	57.84	83.9	2453.7	2537.6	0.296	8.370	8.666
21	0.02486	54.56	88.0	2451.4	2539.4	0.310	8.334	8.644
22	0.02642	51.49	92.2	2449.0	2541.2	0.325	8.297	8.622
23	0.02808	48.62	96.4	2446.6	2543.0	0.339	8.261	8.600
24	0.02982	45.92	100.6	2444.2	2544.8	0.353	8.226	8.579
25	0.03166	43.40	104.8	2441.8	2546.6	0.367	8.190	8.557
26	0.03360	41.03	108.9	2439.5	2548.4	0.381	8.155	8.536
27	0.03564	38.81	113.1	2437.2	2550.3	0.395	8.120	8.515
28	0.03778	36.73	117.3	2434.8	2552.1	0.409	8.085	8.494
29	0.04004	34.77	121.5	2432.4	2553.9	0.423	8.050	8.473
30	0.04242	32.93	125.7	2430.0	2555.7	0.436	8.016	8.452
32	0.04754	29.57	134.0	2425.3	2559.3	0.464	7.948	8.412
34	0.05318	26.60	142.4	2420.5	2562.9	0.491	7.881	8.372
36	0.05940	23.97	150.7	2415.8	2566.5	0.518	7.814	8.332
38	0.06624	21.63	159.1	2411.0	2570.1	0.545	7.749	8.294
40	0.07375	19.55	167.5	2406.2	2573.7	0.572	7.684	8.256
42	0.08198	17.69	175.8	2401.4	2577.2	0.599	7.620	8.219
44	0.09100	16.03	184.2	2396.6	2580.8	0.625	7.557	8.182
46	0.1009	14.56	192.5	2391.8	2584.3	0.651	7.494	8.145
48	0.1116	13.23	200.9	2387.0	2587.9	0.678	7.433	8.111
50	0.1233	12.04	209.3	2382.1	2591.4	0.704	7.371	8.075
55	0.1574	9.578	230.2	2370.1	2600.3	0.768	7.223	7.991
60	0.1992	7.678	251.1	2357.9	2609.0	0.831	7.078	7.909
65	0.2501	6.201	272.0	2345.7	2617.7	0.893	6.937	7.830
70	0.3116	5.045	293.0	2333.3	2626.3	0.955	6.800	7.755
75	0.3855	4.133	313.9	2320.8	2634.7	1.015	6.666	7.681
80	0.4736	3.408	334.9	2308.3	2643.2	1.075	6.536	7.611
85	0.5780	2.828	355.9	2295.6	2651.5	1.134	6.410	7.544
90	0.7011	2.361	376.9	2282.8	2659.7	1.192	6.286	7.478
95	0.8453	1.982	398.0	2269.8	2667.8	1.250	6.166	7.416
100	1.01325	1.673	419.1	2256.7	2675.8	1.307	6.048	7.355

† u and s are chosen to be zero for saturated liquid at the triple point.

HRPR - PRODUCT INFORMATION

HARP® 134a

(1,1,1,2-tetrafluoroethane)

HARP® 134a is a zero ozone depletion (ODP) hydrofluorocarbon refrigerant with properties very similar to R-12. It can be used both as a pure refrigerant in a number of traditional R-12 applications, and as a component in alternative refrigerant blends targeted for R-502 and R-22 applications.

Compressor and system manufacturers now supply equipment which has been specifically designed for *HARP® 134a*. In addition, laboratory testing, field trials and applications experience have confirmed that *HARP® 134a* can work in the retrofit of many existing R-12 installations.

New Systems

Industries which are successfully making the transition from R-12 to *HARP® 134a* include automotive air-conditioning, other specialised air-conditioning or climate control applications, positive pressure centrifugal chillers, medium temperature commercial refrigeration, domestic refrigeration appliances, industrial refrigeration plants and transport refrigeration.

Retrofit

Applications where *HARP® 134a* has been proved reliable in the retrofitting of R-12 systems include centrifugal chillers, semi-hermetic, reciprocating and screw refrigeration applications of all kinds, and industrial refrigeration plants. See later section for specific considerations.

HARP® 134a: Basic Property Data

Chemical Formula:	CH_2FCF_3
Molecular weight:	102.0
Boiling point at 1 atmosphere:	-26.5°C
Density of saturated vapour at boiling point:	0.0053 g/cm³
Density of saturated liquid at 25°C:	1.21 g/cm³
Critical temperature:	101 °C
Critical pressure:	40.7 bara
Latent heat of vaporisation at boiling point:	210.0 kJ/kg
Specific heat of liquid at 25°C:	1.46 kJ/kg.°C
Specific heat of vapour at 1 atmosphere, 25°C:	0.874 kJ/kg.°C
Flammability limits in air:	none*
Ozone depletion potential (ODP):	0
Halocarbon global warming potential (HGWP):	0.27
Occupational exposure limit	1000 ppm

* *HARP® 134a* does not propagate flame in ASTM E-681-95 at test temperatures up to 100°C.

"The complete worldwide refrigerant service"

USE OF *HARP* 134a IN NEW REFRIGERATION SYSTEMS

Lubrication

For all *HARP* 134a applications, lubrication is a very important consideration. Miscibility between refrigerant and oil is critical for many equipment designs and is required to ensure adequate oil return to the compressor. R-134a is not miscible with mineral oils traditionally found in R-12 systems. Polyolester and polyalkylene glycol lubricants have been recommended by various equipment manufacturers for use with *HARP* 134a. Both of these new types of lubricant will absorb moisture quickly. They must be handled carefully to avoid prolonged exposure to air. Generally, new equipment will be shipped by the manufacturer with the compatible lubricant already charged. All of the manufacturers' recommendations should be followed.

System Performance

Climate Control

Chillers and specially designed air-conditioning systems have been engineered to use *HARP* 134a while providing energy efficiency comparable to that of R-12. Manufacturers have successfully introduced products for mobile air-conditioning and positive pressure chillers using *HARP* 134a.

Refrigeration:

Low Temperature Applications

At lower evaporating temperatures, the pressure ratio of R-134a exceeds that of R-12 and the refrigeration capacity, or duty, may be significantly reduced. Equipment manufacturers should be consulted for specific recommendations regarding the use of equipment with R-134a at lower application temperatures. One alternative for low temperature applications is the use of HFC blends, such as *HARP* 134a, which have been specifically designed to replace R-502 in low temperature refrigeration applications.

Medium and High Temperature Applications

HARP 134a can be used in most medium and high temperature R-12 applications. An ideal theoretical analysis using the thermodynamic properties of *HARP* 134a shows a slight decrease in capacity and efficiency. When improvements, such as liquid subcooling, are introduced into the equation, the performance of *HARP* 134a becomes equal to that of R-12. These improvements are being taken advantage of by equipment manufacturers.

CONSIDERATIONS FOR RETROFITTING R-12 SYSTEMS TO *HARP* 134a

Retrofit projects should be included as part of an overall refrigerant management programme. All retrofit applications will require a change in lubricant from that found in R-12 semi-hermetic or open drive systems. The mineral or alkylbenzene oil in the system will need to be replaced with a polyolester lubricant (or in some specific instances a PAG) to a residual mineral oil level of less than 5%. This will require several oil flushes with polyolester depending upon system design. Recommendations from equipment manufacturers regarding the compatibility of materials of construction, components and driers with R-134a should be obtained. Replace any materials that are not suitable. Obtain procedures, material specifications, and any other recommendations from equipment, component and lubricant manufacturers, as well as from HRPR. Hermetic units cannot be retrofitted the R-134a. The construction of these systems does not allow the existing lubricant to be adequately drained and replaced with a polyolester.

HARP* 134a: engineering data

temperature (°C)	absolute pressure (bara)	density		enthalpy (kJ/kg)	
		liquid (kg/dm³)	vapour (kg/m³)	liquid	vapour
-40	0.5122	1.4148	2.7705	148.57	374.16
-30	0.8436	1.3859	4.4256	161.10	380.45
-26	1.0164	1.3741	5.2740	166.16	382.94
-22	1.2160	1.3622	6.2461	171.26	385.43
-18	1.4454	1.3502	7.3643	176.39	387.89
-14	1.7074	1.3380	8.6133	181.56	390.33
-10	2.0052	1.3256	10.037	186.78	392.75
-6	2.3418	1.3130	11.640	192.03	395.15
-2	2.7206	1.3002	13.441	197.33	397.51
0	2.9269	1.2937	14.420	200.00	398.68
2	3.1450	1.2871	15.456	202.68	399.84
6	3.6186	1.2738	17.705	208.08	402.14
10	4.1449	1.2602	20.210	213.53	404.40
14	4.7276	1.2463	22.999	219.03	406.61
18	5.3706	1.2321	26.089	224.59	408.78
22	6.0777	1.2175	29.516	230.21	410.89
26	6.8531	1.2026	33.300	235.90	412.95
30	7.7008	1.1872	37.495	241.65	414.94
34	8.6250	1.1713	42.123	247.47	416.85
36	9.1172	1.1632	44.623	250.41	417.78
40	10.165	1.1465	50.025	256.35	419.58
44	11.300	1.1292	55.991	262.38	421.28
48	12.527	1.1113	62.578	268.49	422.88
52	13.852	1.0926	69.930	274.71	424.35
56	15.280	1.0730	78.125	281.04	425.68
60	16.815	1.0524	87.260	287.49	426.86
70	21.165	0.9956	115.34	304.29	428.89
80	26.331	0.9274	154.80	322.41	429.02
85	29.259	0.8862	181.82	332.27	427.91
90	32.445	0.8369	216.92	343.01	425.48

To calculate the latent heat of vaporisation, subtract the liquid enthalpy from the vapour enthalpy at the desired temperature.

HARP* 134a: temperature - pressure relationship comparison with R-12

temperature (°C)	134a pressure (bara)	R-12 pressure (bara)	temperature (°C)	134a pressure (bara)	R-12 pressure (bara)
-30	0.8436	1.0044	8	3.8749	3.9765
-28	0.9268	1.0929	10	4.1449	4.2276
-26	1.0164	1.1872	12	4.4289	4.4903
-24	1.1127	1.2878	14	4.7276	4.7651
-22	1.2160	1.3949	16	5.0413	5.0523
-20	1.3268	1.5088	18	5.3706	5.3521
-18	1.4454	1.6296	20	5.7159	5.6651
-16	1.5721	1.7578	22	6.0777	5.9914
-14	1.7074	1.8937	24	6.4566	6.3315
-12	1.8516	2.0374	26	6.8531	6.6857
-10	2.0052	2.1893	28	7.2676	7.0544
-8	2.1684	2.3498	30	7.7008	7.4379
-6	2.3418	2.5190	32	8.1530	7.8366
-4	2.5257	2.6974	34	8.6250	8.2509
-2	2.7206	2.8851	36	9.1172	8.6811
0	2.9269	3.0827	38	9.6301	9.1277
2	3.1450	3.2902	40	10.165	9.5909
4	3.3755	3.5082	42	10.721	10.071
6	3.6186	3.7368	44	11.300	10.569

THERMODYNAMIC EQUATIONS FOR *HARP* 134a

1. Vapour pressure

$$\ln P = \frac{a_1}{T} + a_2 + a_3 T + a_4 \ln T$$

where T is in Kelvin and P is in bar.
Constants:

a_1	=	-3.218946E+03	a_3	=	2.780783E-03
a_2	=	2.732727E+01	a_4	=	-2.714900E+01

2. Density of the saturated liquid

$$d = b_0 + \sum_i^4 b_i (1 - T_r)^{i/3}$$

where d is in kg/m^3 and T is in Kelvin. T_c is the critical temperature and $T_r = T/T_c$.
Constants:

b_0	=	5.119758E+02	b_3	=	6.591549E+02
b_1	=	9.475580E+02	b_4	=	-4.215000E+02
b_2	=	1.714503E+02	T_c	=	374.15 K

3. Ideal gas heat capacity

$$C_v^o = \frac{C_1}{T} + C_2 + C_3 T + C_4 T^2$$

where C_v^o is in J/mol.K and T is in Kelvin.
Constants:

C_1	=	1.582170E+01	C_3	=	3.296570E-03
C_2	=	-8.674560E-02	C_4	=	-2.017321E-06

4. Equation of State

$$P = \frac{RT}{V - b} + \sum_{i=2}^{5} \frac{A_i + B_i T + D_i \exp(-kT)}{(V - b)^i}$$

where T is in Kelvin and P is in bar.
Constants:

R	=	8.148816E-04	B_4	=	0.000000E+00
A_2	=	-1.195051E-03	D_4	=	0.000000E+00
B_2	=	1.137590E-06	A_5	=	-6.953904E-14
D_2	=	-3.531592E-02	B_5	=	1.269806E-15
A_3	=	1.447797E-06	D_5	=	-2.051369E-11
B_3	=	-8.942552E-10	b	=	3.455467E-04
D_3	=	6.469248E-05	k	=	1.462928E-02
A_4	=	-1.049005E-09			