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Experimental Performance Analysis of Reciprocating Compressor Working with Non-Aceotropic Refrigerant Mixtures

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EXPERIMENTAL PERFORMANCE ANALYSIS OF RECIPROCATING COMPRESSOR WORKING WITH NON-AZEOTROPIC REFRIGERANT MIXTURES

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A B S T R A C T

Much work has been done in the last years to analyse the performance of refrigeration systems with non-azeotropic refrigerant mixtures. In all cases results for the total system are presented. No further information of compressor performance has been available up to now.

For this reason experimental work with a reciprocating compressor has been done to analyse the compressor behaviour with various non-azeotropic refrigerant mixtures.

Experimental results concerning refrigeration capacity, shaft power, isentropic indicated efficiency and volumetric efficiency will be presented for different mixture concentrations of R 22 / R 114.

INTRODUCTION

As stated by the International Energy Agency /1/ much work has been done in the last years to analyse the possibilities of energy conservation by applying non-azeotropic refrigerant mixtures in refrigeration systems or heat pumps.
Nearly all investigations have been done concerning energy consumption of the whole system, or concerning measurements of heat transfer coefficients at special test facilities.

Besides the heat exchangers the compressor is responsible for most of the losses occuring during the refrigerant cycle. For this reason it is neccessary to get knowledge of compressor behaviour when working with non-azeotropic refrigerant mixtures.

For this purpose measurements have been done at a secondary fluid calorimeter with an open type reciprocating compressor variable in speed by a DC electric motor. The refrigerant mass flow rate, calculated by an energy balance at the heat exchangers, the shaft power and the p-V-diagram have been measured to compare the influence of different mixture concentrations on the isentropic efficiencies.

TEST FACILITY

According to ISO 917 the tests have been made at a secondary fluid calorimeter (Fig.1). The test facility consists mainly of an open type compressor (displacement volume 120 cc/rev) (1), a water cooled counterflow coaxial condenser (2), a liquid subcooler (3), dryer (4), three expansion valves (9-11) different in size and an electrical heated boiler, filled with R11, that contains the evaporator. To get no fractionation of the refrigerant mixtures inside the circuit no accumulator is used.

The cooling water for the condenser is circulated by a pump (29) out of the tank hanging above the condenser, to get constant water pressure at the suction side of the pump. At the high pressure side of the pump the water flow through the condenser is regulated by a pressure controlled valve (23) or by a hand valve (22). By opening the bypass valve (21) it is possible to reduce mass flow rate through the pump. After passing the condenser the water is transfered to the tank again. The water temperature at condenser input can be regulated by a thermostatic water valve (27), that connects the water tank with the fresh water circuit of the laboratory.

The electrical power input to the calorimeter boiler is regulated manually with a transformator in the range 0-1000 W or 0-2000 W. In addition several heaters can be switched on having a capacity of 1000 W each.

Several shut-off valves (3,7,15,17) are located at the circuit to get gas samples of refrigerant mixtures out of the cycle during operation. The samples can be analysed with a gas chromatograph.

The compressor is driven by a DC electric motor allowing variable speed. The shaft power is calculated by the measured reaction moment of the motor and the speed of the compressor.

The cylinder pressure during the working cycle of the compressor is measured with piezo-electric pressure transducer. In future strain gauge pressure transducer will be used in order to calculate the suction and discharge valve losses from the p-V-diagram. In addition the valve lift curves of the suction and the discharge valves will be recorded by inductive displacement pick-ups.

DATA RECORDING

All temperatures and pressures that are illustrated by the letters "t" respectivly "p" in figure 1 have been measured with help of a computer controlled data aquisition consisting of a multiplexer, an A/D converter and an interface transferring the data to the computer. Beside these all other data like power input to the calorimeter, reaction moment of the electric motor, the mass flow rate of the cooling water and others are fed into the computer manually or via the data aquisition.

To record the transient signals of the pressure and the valve lift a so called " multiprogrammer" is used, having an A/D converting speed of 33 kHz and transfering the data directly to the computer.

DATA PROCESSING

In order to carry out an energy balance at the heat exchangers it is neccessary to calculate caloric data of the various mixtures as a function of concentration. For this purpose a computer program was developed which is based on the Redlich-Kwong-Soave equation of state /2/.

As described above the test facility is connected with a micro-computer that manage the data recording and the data processing by calculating the mass flow rate of the compressor and the p-V-diagram. These data are partly

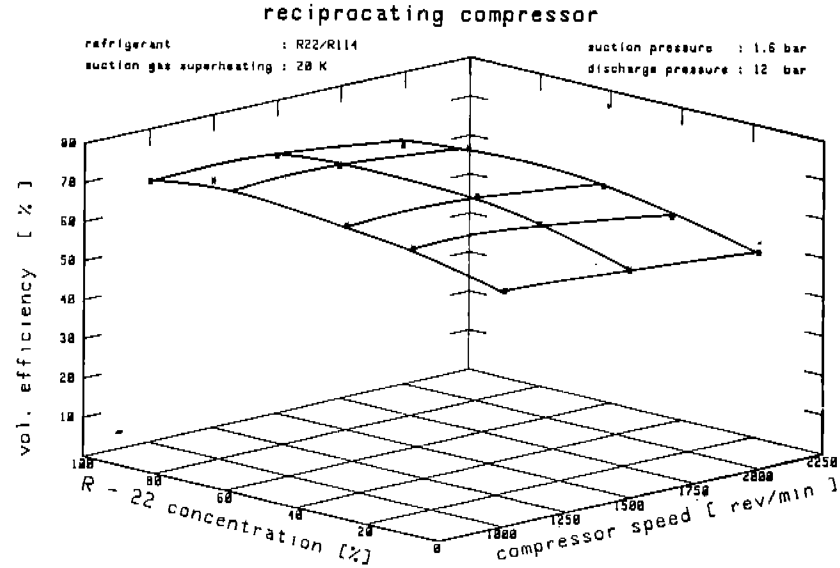

figure 12 : volumetric efficiency as a function of compressor speed and R - 22 concentration

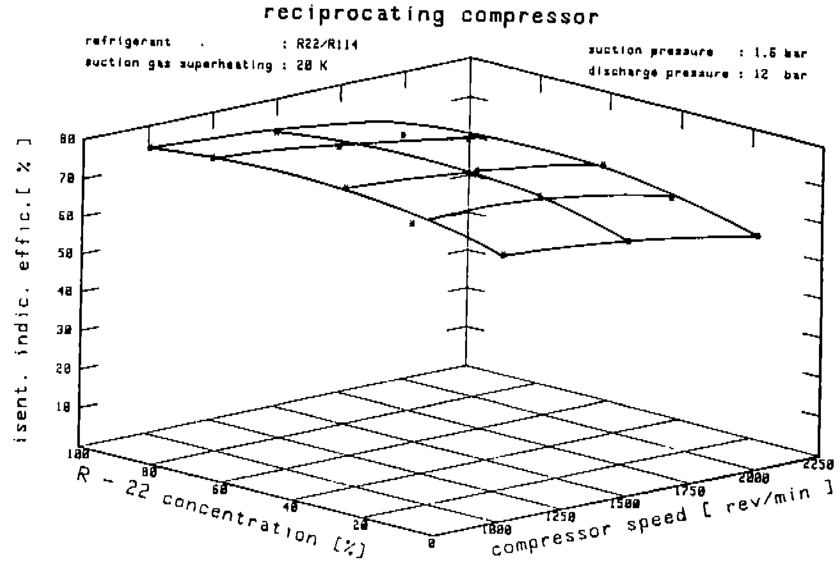

figure 13 : isentropic indicated efficiency as a function of compressor speed and R - 22 concentration

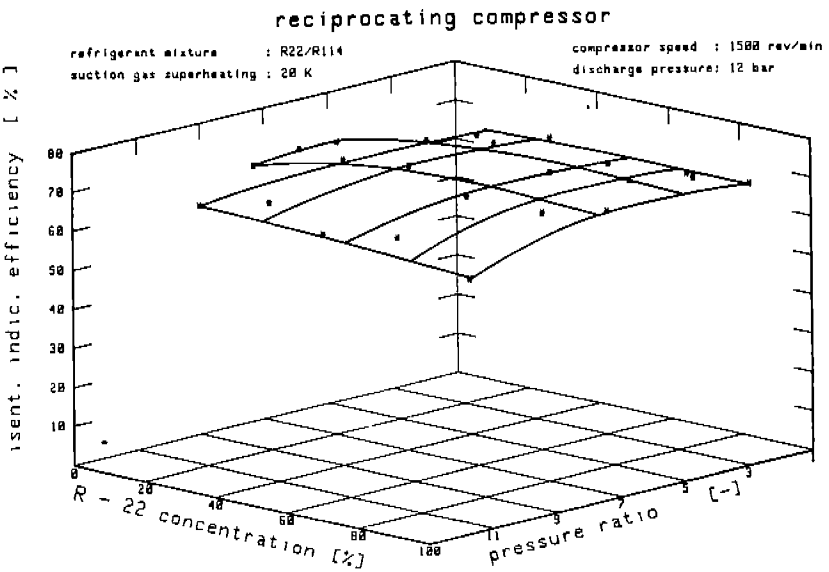

figure 10. isentropic indicated efficiency as a function of pressure ratio and R - 22 concentration

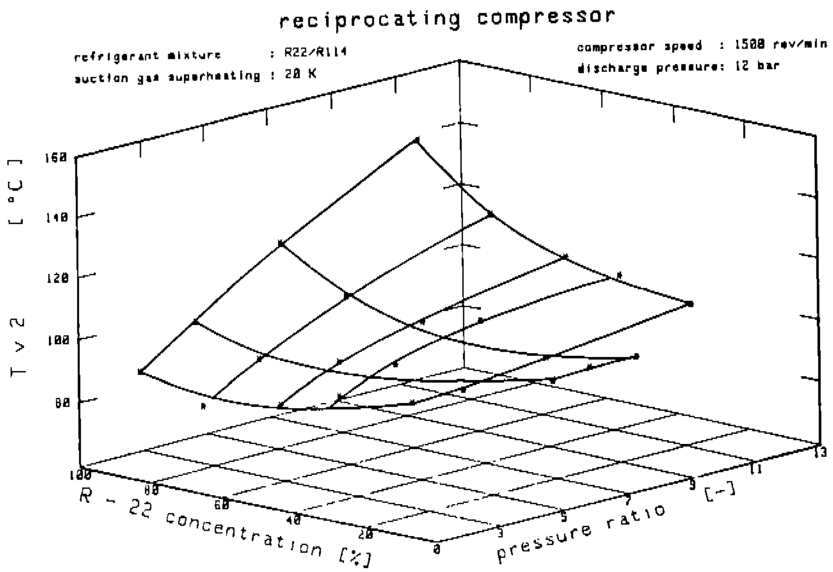

figure 11 discharge temperature as a function of pressure ratio and R - 22 concentration

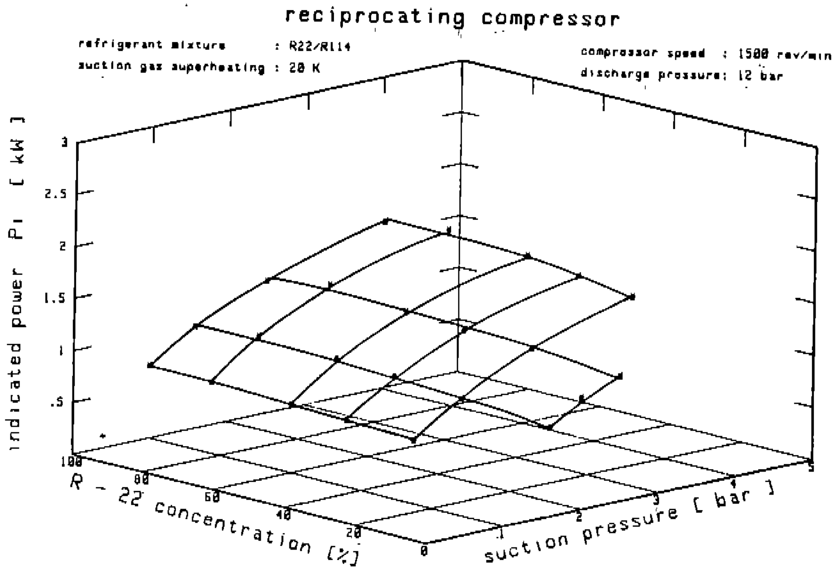

figure 8. indicated power as a function of suction pressure and R - 22 concentration

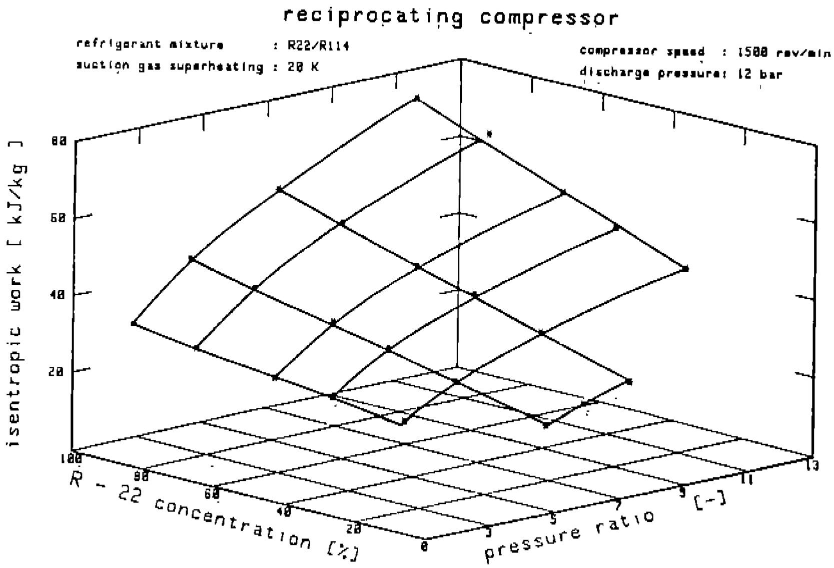

figure 9: isentropic specific work as a function of pressure ratio and R - 22 concentration

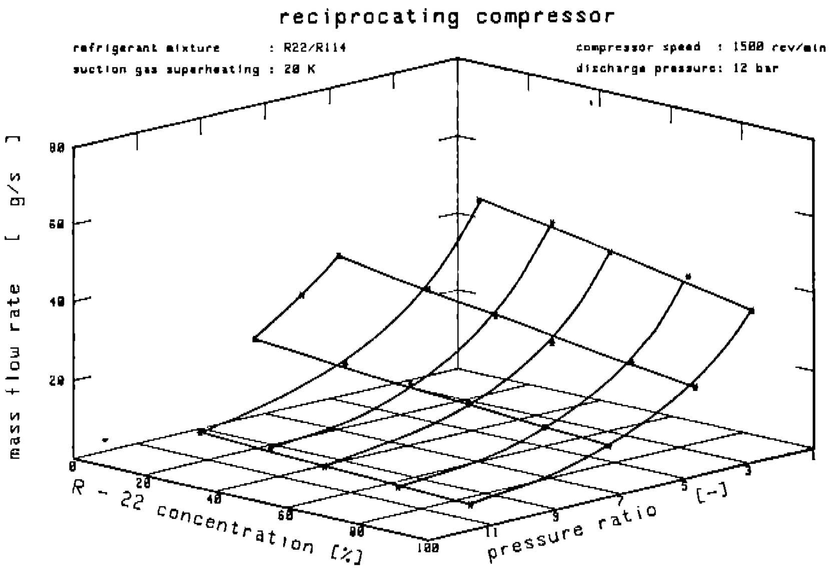

figure 6. mass flow rate as a function of pressure ratio and R - 22 concentration

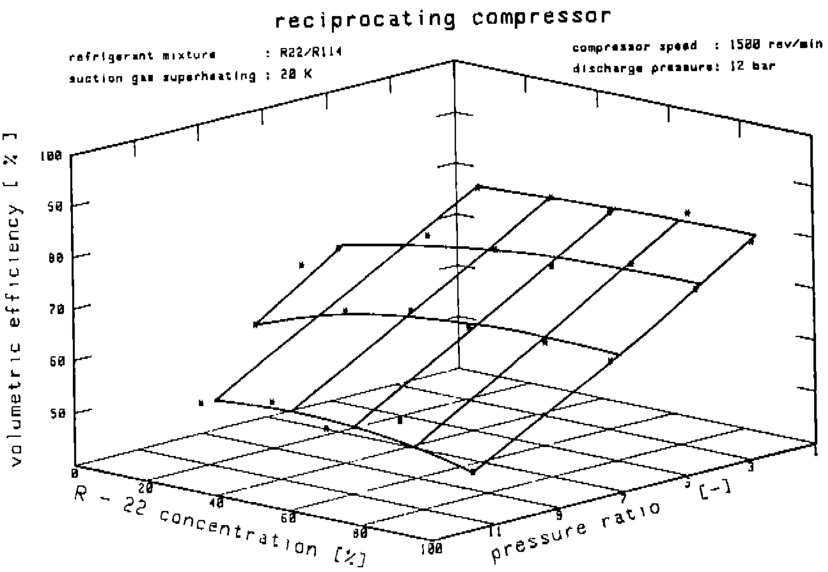

figure 7. volumetric efficiency as a function of pressure ratio and R - 22 concentration

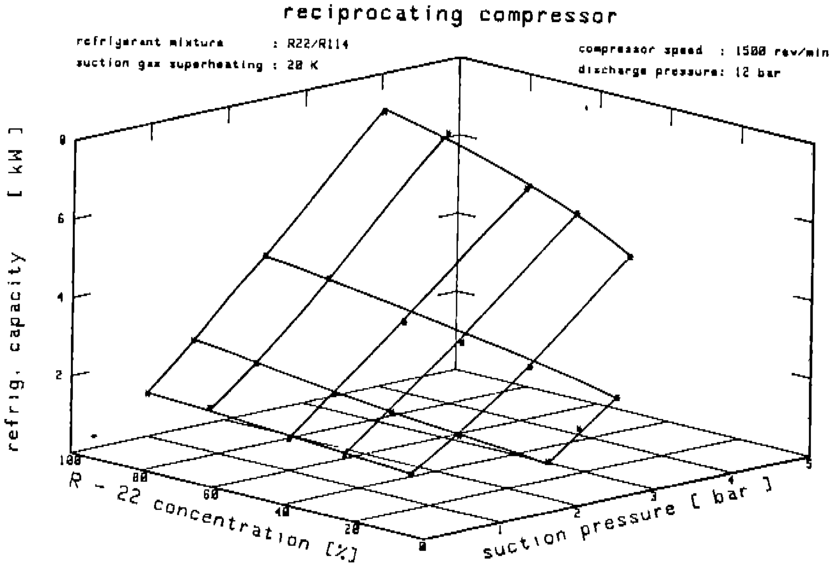

figure 4: refrigeration capacity as a function of suction pressure and R - 22 concentration

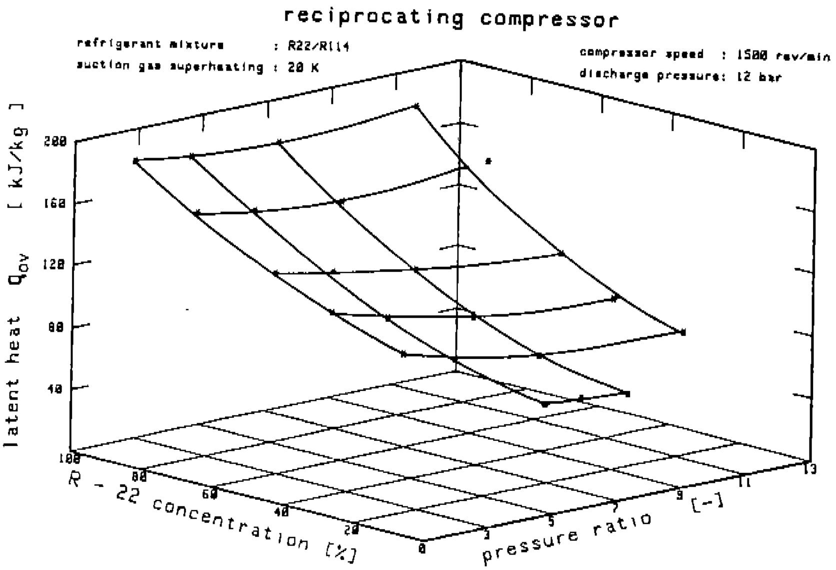

figure 5: latent heat of evaporation q_{ov} as a function of pressure ratio and R - 22 concentration

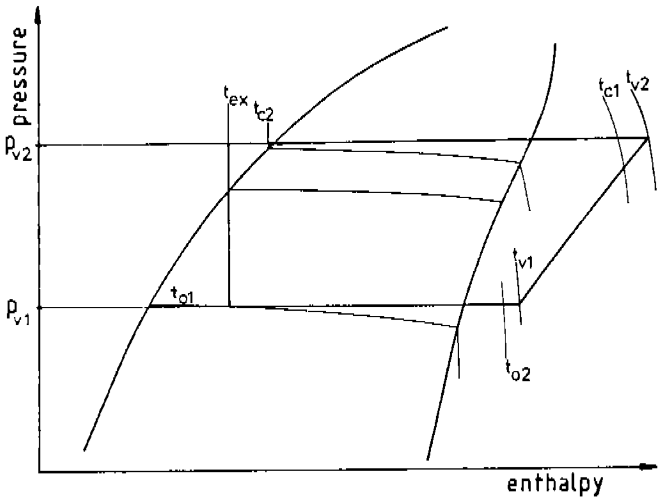

figure 2: lg p–h diagram of the mixture cycle according diagram 1

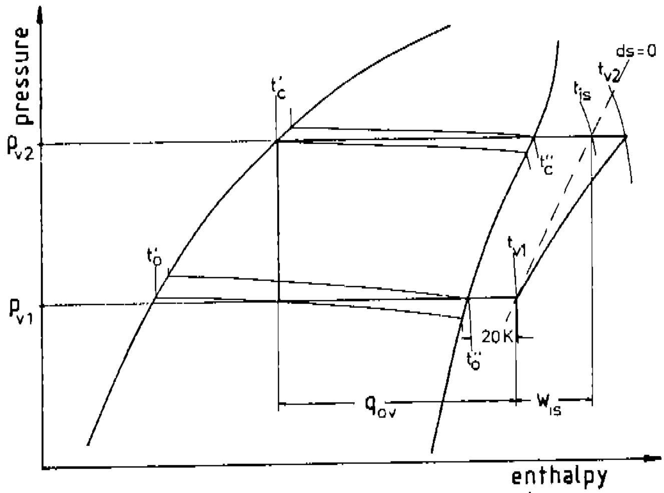

figure 3: temperatures and enthalpy values according fig. 4–13 in a lg p–h diagram

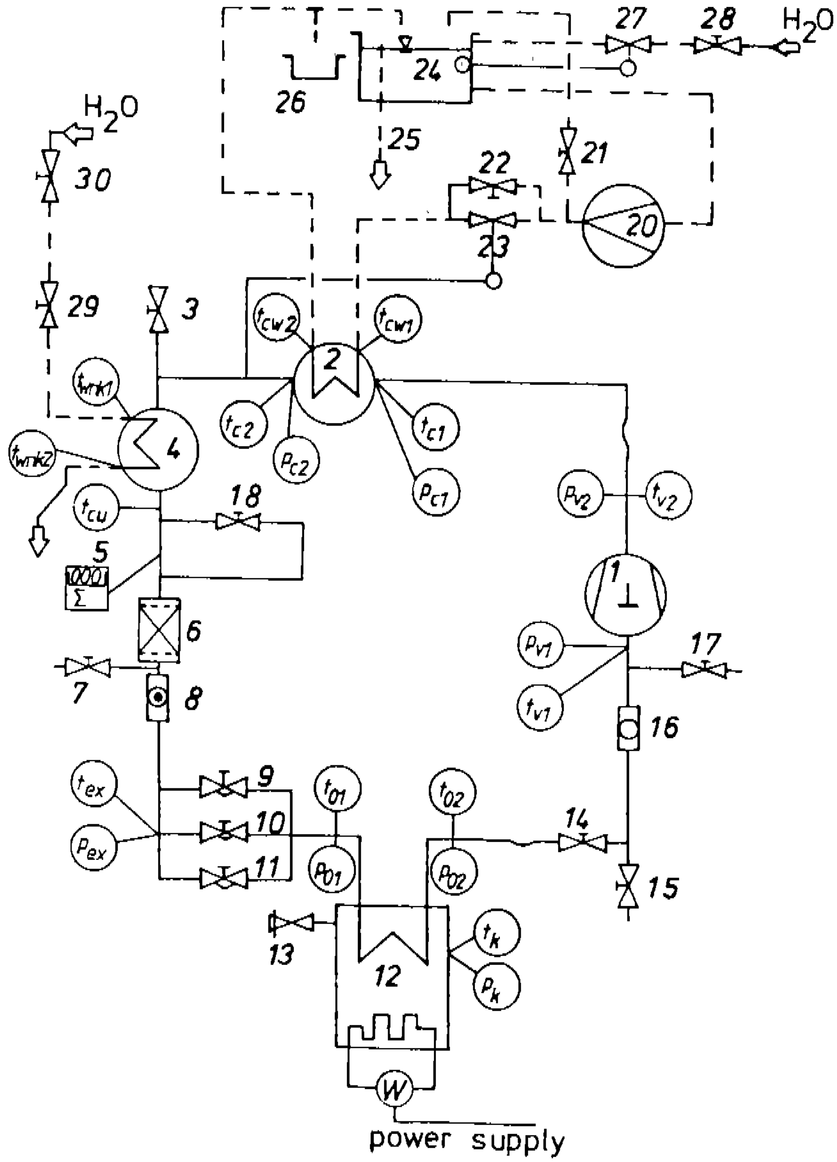

figure 1 : test facility

the results of the computer simulation will be controlled then.

R E F E R E N C E S

/1/ IEA — Bibliography : "Non-Azeotropic Refrigerant Mixtures As Working Fluid In Compression Heat Pumps" IEA Heat Pump Center
HPC-B1 Edition.1 Oct 84

/2/ Soave — Chem. Eng. Science 35; p. 1725

/3/ Küver, Kruse — The Use of Non-Azeotropic Refrigerant Mixtures in Two Temperatures Refrigerators.
Proc. of the Int.Inst.of Refrig. Com. B1
Purdue 1986

/4/ Vakil — Performance of Reciprocating Compressors with Non-Azeotropic Mixtures
Proc.Purdue Comp. Tech. Conf. 1984

/5/ Röttger, Kruse — Analysis of the working cycle of single stage refrigeration compressors using digital computer
Proc.Purdue Comp. Tech. Conf. 1976 p.18

speeds demonstrate only slight differences in compressor performance. In a real refrigeration system working either with a pure or a mixed refrigerant the compressor behavior can be quite more different because of changing pressure ratio and sometimes speeds for various refrigerants. This shall be demonstrated by the following example of system test results gained with the same compressor type as used here.

Experiments at a test facility /3/ working with pure R 12 and refrigerant mixture of 60 % R 22 / R 114 respectively operating at the same secondary fluid temperatures and the same cooling capacity show that the measured COP of the plant increases from 1.15 (R 12) to 1.42 (R 22 / R 114). The COP calculated with the isentropic compressor power increases only from 2.38 (R 12) to 2.56 (R 22 / R 114) showing an improvement of 8 % while the real plant shows an improvement of about 23 %. The reason for this difference is the better efficiency of the compressor when running at the same outer temperature conditions at a lower pressure ratio (11.9 for R 12, 10.4 for R 22 / R 114) and a lower compressor speed for the same refrigeration capacity (1300 min^{-1} for R 12, 900 min^{-1} for R 22 / R 114) with refrigeant mixtures as compared to pure R 12. It can be seen in this example that most of the improvements are caused by the better operation conditions for the compressor when working with a refrigerant mixture. On the other hand this example shows how the datas of the compressor tests that are presented here can be used for a judgement of test results that have been determined at a real plant.

CONCLUSION

Vakil /4/ stated from his measurements that the performance of a compressor working with non-azeotropic refrigerant mixtures shows no fundamental difference compared to pure refrigerants. Our own measurements done with the same refrigerant mixture of R 22 / R 114 lead to the results that the efficiency decreases slightly with decreasing R 22 concentrations. At R 22 concentrations of less than 40 % this effect is greater than at higher R 22 concentrations. The behaviour of other mixtures e.g. R 13B1 / R 114 is under investigation now.

Because it is not possible to run such tests in the laboratory for all refrigerant mixtures a compressor simulation program with refrigerant mixtures is under work now. The program is based on the work of Röttger /5/ taking the real gas behaviour of non-azeotropic refrigerant mixtures under account by using the Redlich-Kwong-Soave equation of state. With the measured data decreasing R 22 concentrations compensates the decreasing isentropic work partly and this leads to the smooth slope of the indicated power with decreasing R 22 concentration.

Figure 10 shows the performance graph concerning the isentropic indicated efficiency. The plot looks similar to that of the volumetric efficiency. The indicated efficiency decreases slightly with decreasing R 22 concentrations. In the region with smaller R 22 concentrations this tendency is greater than in the range with R 22 concentrations greater than 25 %.

The effective isentropic efficiency evaluated from the measured shaft power has been calculated in order to get the mechanical efficiency. This shall not be shown here since there is no special additional influence of refrigerant mixture against that already inherent indicated efficiency.

The graph of the discharge temperature (Fig. 11) shows that even at low R 114 concentrations of about 25 % the discharge temperature is lowered by about 25 K.

Test results at variable compressor speed

Suction pressure	: p_{v1}	= 1.6 bar
Discharge pressure	: p_{v2}	= 12.0 bar
Suction gas superheating	: Δt_{oh}	= 20 K
Compressor speed	: n	= 1000 - 2000 min^{-1}

The test results with variable compressor speed are illustrated for the volumetric and the indicated isentropic efficiency in figure 12 and figure 13 respectively. The volumetric efficiency decreases with higher compressor speeds and decreasing R 22 concentrations. It can be stated that the slope of this graph is not constant. At smaller R 22 concentrations the increasing compressor speed causes a more falling tendency than at higher R 22 concentrations. The same fact shows the graph of the isentropic indicated efficiency illustrated in figure 13. The compressor operates in this range best with pure R 22 at 1000 min^{-1} and worse with pure R 114 at compressor speed of 2000 min^{-1}.

Compressor performance working in a refrigeration system

The test results shown here for fixed pressure ratios or speed, and second at variable speed at a constant pressure ratio .

Test results with constant compressor speed

Suction pressure	: p_{v1} =	1.0 - 4.0 bar
Discharge pressure	: p_{v2} =	12.0 bar
Suction gas superheating	: Δt_{oh} =	20 K
Compressor speed	: n =	1500 min^{-1}

The graph of the refrigeration capacity as a function of R 22 concentration and suction pressure is shown in figure 4. It can be stated that the capacity of the compressor decreases with decreasing R 22 concentration. The reason for this is the lower latent heat of R 114 as compared to R 22 (Fig. 5). The mass flow rate of the compressor is illustrated in figure 6. The graph shows that the mass flow rate is increasing with decreasing R 22 concentrations. The increasing mass flow rate is not able to compensate the decreasing latent heat of R 114, so that the absolut cooling capacity is decreasing with higher R 114 concentrations as illustrated in figure 4.

To judge the compressor concerning its refrigerant delivery it is the best way to calculate the volumetric efficiency (Figure 7). As can be seen the volumetric efficiency as a function of mixture concentration is nearly constant for all mixture concentrations between 60 to 100% R 22. There is only a small diminution at the high pressure ratio for pure R 22, caused by the high body temperatures in consequence of discharge temperatures of about 410 K (140 °C). On the other side the diagram shows especially at medium pressure ratios that the volumetric efficiency decreases at lower R 22 concentrations of about 0 - 60 %, caused by higher pressure drops in the valves due to higher molecular weight compared to mixture concentrations that are rich of R22. The diminution increases at mixture concentrations of about 0 - 25 % R 22.

The graph of the indicated power (Fig. 8) as a function of suction pressure and R 22 concentration shows the same tendency as the plot of the cooling capacity. The power input decreases with decreasing R 22 concentrations. As illustrated in figure 9 the theoretical isentropic work has the same tendency but the slope is greater with changing concentration. In this case the increasing refrigerant mass flow rate (Fig. 6) with stored at a micro-disc. Plotts of the measured and analysed data can be produced with help of another software.

T E S T   C O N D I T I O N S

Normally refrigeration compressors are tested at defined evaporation and condensing temperatures. Consequently the performance graph is usually presented as a plott containing the cooling capacity and the power input of the compressor as a function of a evaporation saturation temperature with condensing saturation temperature as a parameter.

Because of the gliding temperatures of an evaporating or condensing non-azeotropic refrigerant mixture it is not possible to go this way. It is even impossible to measure in such a way, because the starting temperature of evaporation depends on the liquid subcooling of the condensate (Fig 2).

For this reason the measurements have been made at various constant suction and discharge pressures with a constant suction superheating of 20 K.

The suction and discharge pressures were choosen out of preferred numbers according ISO R3 and shown in table 1.

Discharge pressure (bar) :	8.0	12.0	18.0	
Suction pressure (bar) :	1.0	1.6	2.5	4.0
Suction gas superheating : 20 (K)				

Table 1 : Suction and discharge test conditions

The first refrigerant mixture that has been investigated was R 22 / R 114 . This has been done because parallel to this test another research project at a two evaporator refrigeration cycle using the same type of compressor and the same refrigerant mixture was carried out /3/. The measured compressor data are helpful to analyse that cycle further.

T E S T   R E S U L T S

The illustration and nomenclature of the thermodynamic data that will be presented here is done according to figure 3. The discussion of the results is done first for a discharge pressure of 12.0 bar with constant