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AN ANALYTICAL COMPARISON BETWEEN THE PERFORMANCE OF A HOT WATER HEAT PUMP WITH A NON-AZEOTROPIC REFRIGERANT MIXTURE AND A PURE REFRIGERANT

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ABSTRACT

The applications of hot water in the industrial, domestic and mining applications are numerous, and these are only a few of the core areas of use. In these applications fossil fuels and electrical resistance systems are usually used to heat water to temperatures near boiling point. The refrigerant R22, that is currently being used in hot water heat pumps, delivers hot water temperatures from 60 °C to 65 °C. This limits the applications of hot water heat pumps.

This analytical study uses three comparison methods to investigate and compare the potential of a non-azeotropic refrigerant mixture consisting of R22 and R142b. From the results different advantages of non-azeotropic refrigerant mixtures are evident. Depending on the application, if the results of a non-azeotropic refrigerant mixture are compared with a pure R22 heat pump, an increase in hot water temperatures to above boiling point, an increase in coefficient of performance, an increase in capacity and a decrease in compressor pressure ratio are possible. Unfortunately, not all these advantages are valid for each application. For instance, extremely high hot water temperatures are obtained, whilst the heating capacity is excessively low.

KEY WORDS

Hot water, Heat pump, non-azeotropic, heating, R22, R142b

1. INTRODUCTION

One can almost not imagine life without the use of hot water without even considering an attempt to name the number of applications of hot water in domestic, industry, food processes and mining practices. Several methods are used for water heating. A few of these methods include electrical heating elements, coal heating, solar panels and gas heating. The detriment of these methods are that more input power is needed, than the amount of heating that is done. For example if energy is supplied at a rate of 1 kW to an electrical element, no more than 1 kW of heating may be done.

With hot water heat pumps the amount of heating is greater than the electrical input. Currently hot water heat pumps use R22 as refrigerant which deliver hot water temperatures up to 60 to 65 °C (Meyer and Greyvensteyn 1991). Although this temperature is sufficient for domestic applications, higher water temperatures will increase the applications of hot water heat pumps.

The purpose of this study is to investigate the influence of a non - azeotropic refrigerant mixture on the performance of a hot water heat pump. To make this possible, detailed heat pump design methodologies have to be developed in order to predict performances. Although the performances could be calculated by developing a computer programme, hand calculations will be used in this study to develop a better grasp of the interaction of the different variables involved. To make the design more elementary, the heat pump systems selected are water-to-water heat pumps with counterflow, horizontal pipe-in-pipe heat exchangers, with refrigerant on the inside and water in the annulus. It is not the aim of this study to optimise the systems: this can be done at a later stage, preferably after the development of a computer programme, if the results of this study are positive.

The outline of this paper is as follows: Three methods are used to compare the influence of a non - azeotropic refrigerant mixture on the performance of a hot water heat pump. These methods are discussed, where after results are represented in graphical form, the results are discussed and finally conclusions are made.

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2. COMPARISON OF PERFORMANCE

A design methodology was developed (Smit 1996) to determine the performance of a pure R22 and a non-azeotropic heat pump. Boundary conditions of each method are used separately and results are generated as function of different concentrations. The condenser outer pipe diameters are 9.53 mm and 15.88 mm, with the water mass flow rate of 0.04 kg/s and 15.88 mm and 25.4 mm for the evaporator with the water mass flow rate of 0.4 kg/s. It was found through experiments that these geometries give acceptable pressure drops (ASHRAE 1985). The only unknown geometry is therefore the lengths of the evaporator and condenser for different concentrations R22 and R142b.

There are several methods of making comparisons (Högberg *et al* 1992) between heat pumps with mixtures and with pure working fluids. Each method sheds new light on the subject. In this study three methods are investigated, namely 1) Pressure ratio, 2) Bubble and dew point temperatures and 3) Maximum hot water and constant bubble point temperature.

2.1. Method 1: Pressure ratio

Since the compressor is the heart of the heat pump and also the most expensive component, its influence must also be taken into account. The performance of the compressor is influenced by the inlet refrigerant density, suction and discharge pressure. By keeping the suction and discharge pressures of the compressor constant (and thus keeping the inlet condenser and inlet evaporator pressure constant), the first method investigates the influence of the refrigerant at these constant pressures for different concentrations of R22 with R142b. The evaporator pressure is 497.7 kPa and the condensing temperature is 2 738 kPa. These values were selected as they represent an evaporating temperature of 0 °C and a condensing temperature of 65 °C for pure R22.

2.2. Method 2: Bubble and dew point temperatures

Another popular method (Högberg *et al* 1992, Johannsen 1992) used in research studies is the comparison between the Carnot and Lorentz cycles. In this method the dew point temperature in the condenser and the bubble point temperature in the evaporator are kept constant for the two cycles. These two temperatures are influenced by the temperatures of the heat sink and heat source.

In most heat pump applications the heat source is water or air at atmospheric conditions i.e. at a temperature of approximately 20 °C. If a maximum temperature glide of 7 °C and a superheat of 11.11 °C is assumed for the mixture (the superheat is a recommended value provided by the compressor manufacturer), an adequate average evaporating temperature would be in the region of 0 °C.

Unfortunately, at such a low temperature the pressure of R142b is very low. The gas trapped in the clearance volume of the compressor must expand to a pressure low enough for the suction valves to open and draw more gas in. If the pressure at the inlet of the compressor is too low (as in this case) no gas would be sucked into the compressor and the mass flow of the refrigerant would be zero. To ensure a mass flow of 0.00126 kg/s which is only 3% of the maximum mass flow of the selected compressor, the suction pressure must be higher than 321 kPa. For this pressure it can be observed from Figure 1 that as the concentration of R142b is increased, the evaporating temperature increases, the result being that higher heat source temperatures are needed.

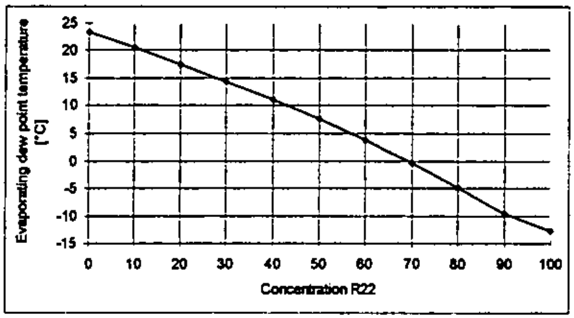

Figure 1. Evaporator dew point temperature of a mixture of R22 and R142b at different concentrations and a pressure of 321 kPa.

Although the heat source temperature is excessively high for a case where 100% R142b is used as refrigerant, the restriction to this temperature is due to the practical limitations of the selected compressor. In this study the heat

Although the heat source temperature is excessively high for a case where 100% R142b is used as refrigerant, the restriction to this temperature is due to the practical limitations of the selected compressor. In this study the heat source temperature will be chosen high to compensate for the low pressure in the compressor, but for practical applications a compressor with a high pressure ratio and low suction pressure would be better suited. At the time of this study such a compressor was not available but the results will still show the same tendency as in a case with low heat source temperatures. Therefore, for the purposes of this study, the dew point temperature in the condenser will be 65 °C and the bubble point temperature in the evaporator will be 25 °C.

2.3. Method 3: Maximum hot water and constant bubble point temperature

After analysing the heat pump at constant pressures and constant temperatures, the third method compares the performance of the heat pump for a more practical application, by determining the maximum water temperature that the heat pump can deliver while the heat source temperature is kept constant. The reason for this is that in practical situations the user of the heat pump would like to have control over the maximum hot water temperature. It is also known that the temperature of the heat source, which is usually the ambient air, stays relatively constant (in Johannesburg for example the average ambient temperature fluctuates with only 10 °C from summer to winter), (Climate of South Africa 1988), making this a suitable method. In the view of the problem experienced in the previous section (paragraph 2.2) with the dew point temperature, the same dew point temperature of 25 °C is used while the condensing pressure is kept constant at 2 738 kPa (which is the same pressure used in the first method, in section 2.1).

3. RESULTS

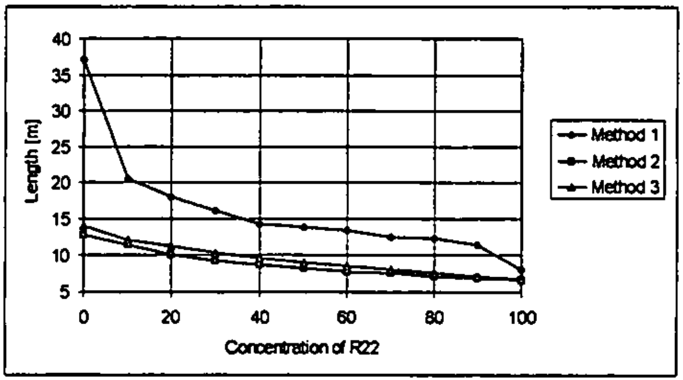

Figure 2. Length of condenser.

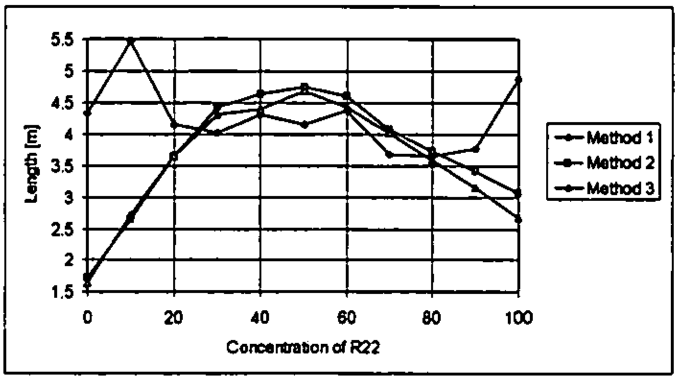

Figure 3. Length of evaporator.

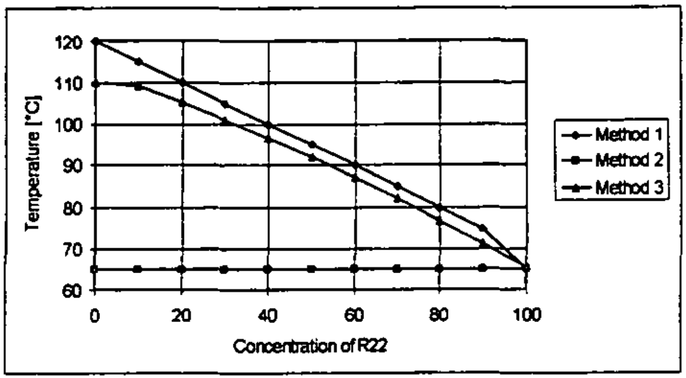

Figure 4. Condenser water outlet temperature.

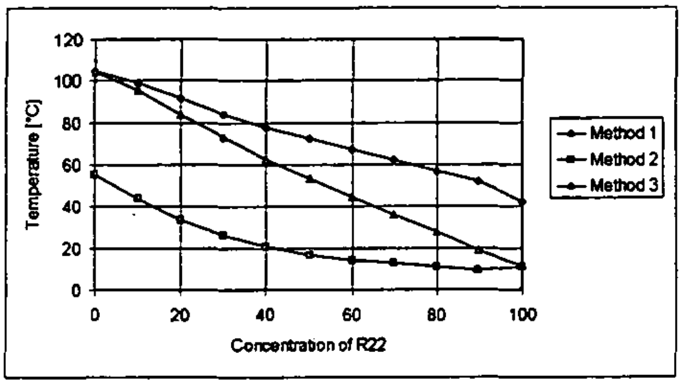

Figure 5. Condenser water inlet temperature.

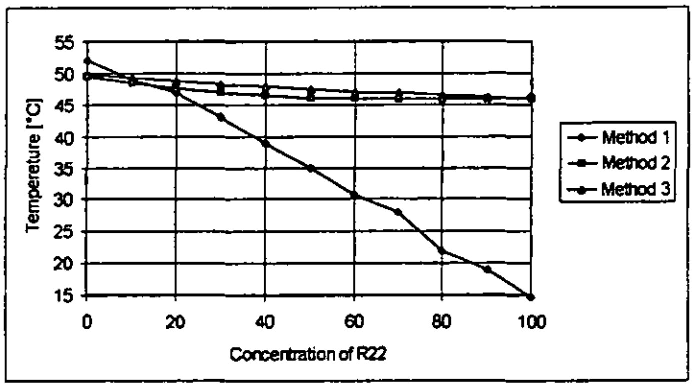

Figure 6. Evaporator water inlet temperature.

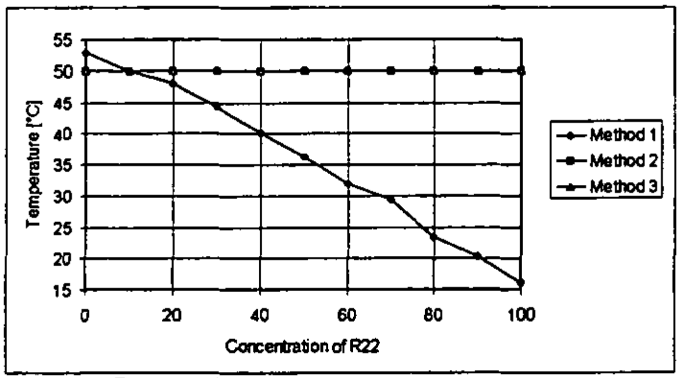

Figure 7. Evaporator water outlet temperature.

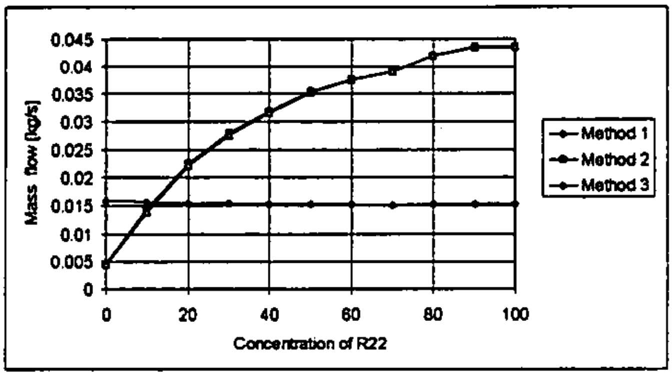

Figure 8. Refrigerant mass flow.

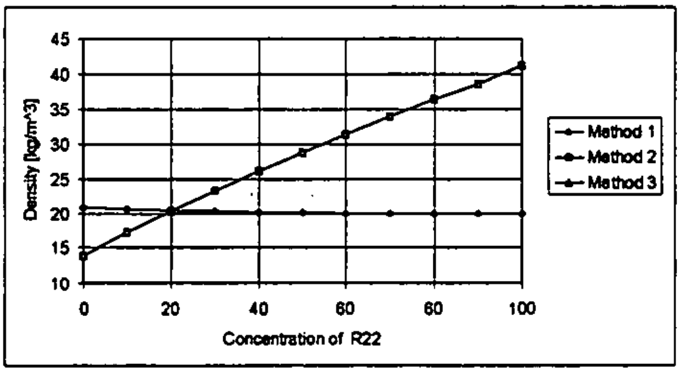

Figure 9. Density of refrigerant mixture at compressor inlet.

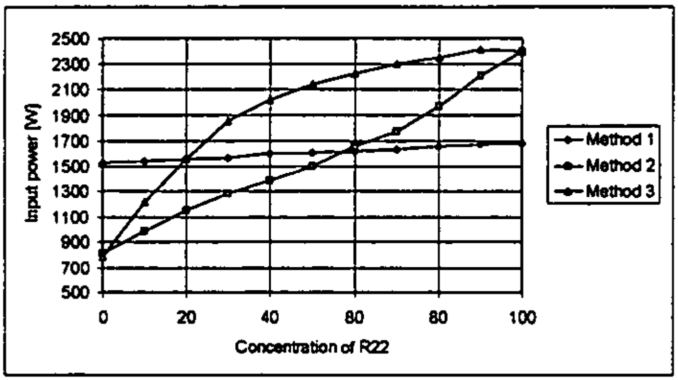

Figure 10. Input compressor power.

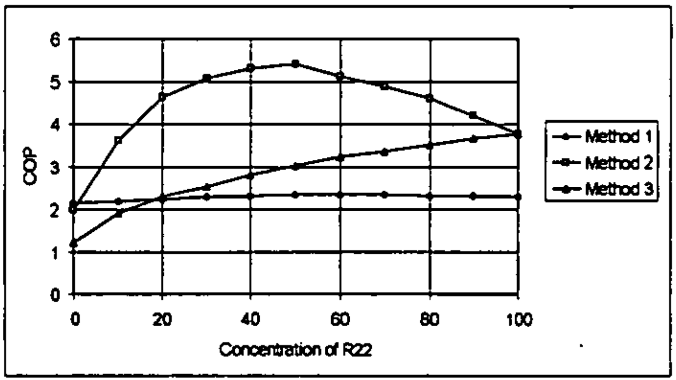

Figure 11. Heating COP.

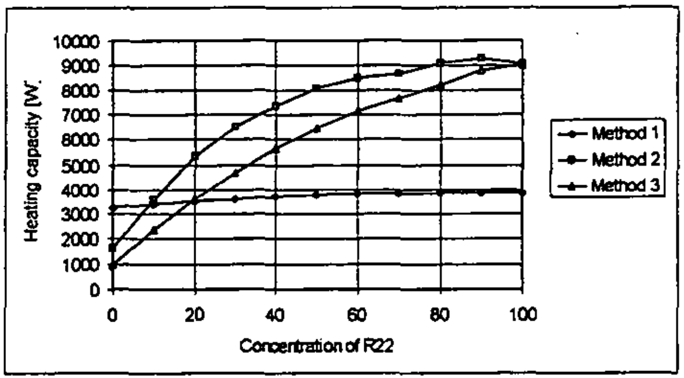

Figure 12. Heating capacity.

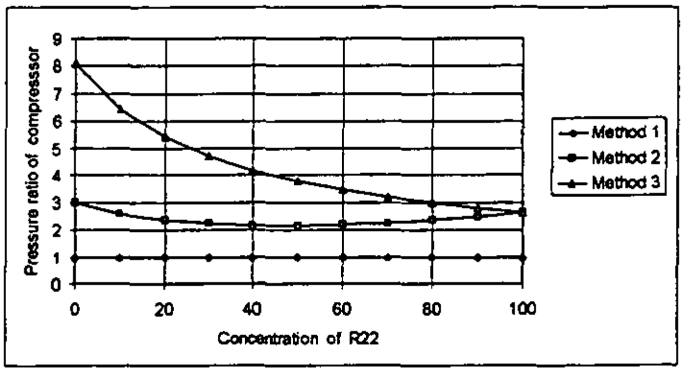

Figure 13. Compressor pressure ratio.

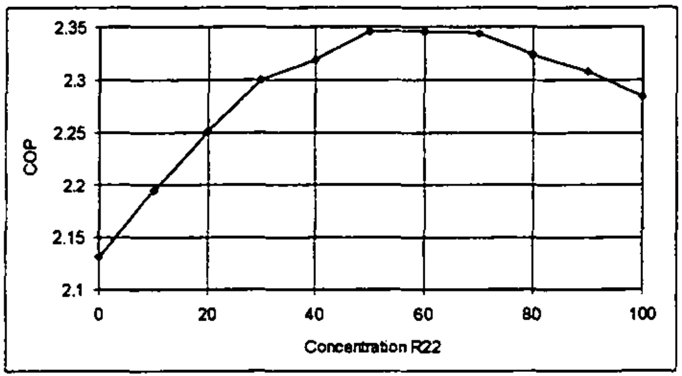

Figure 14. Heating COP of Method 1.

4. DISCUSSION

Figures 2 to 14 represent the results at different concentrations of R22 and R142b. The concentration refers to the mixture of R22 and R142b on a mass basis. To avoid unnecessary repetition in the text, an increase in a property would mean the increase of the property with the increase of R142b in the mixture.

It can be seen that for all 3 methods in Figure 2 the, length of the condenser increases. For low concentrations of R142b in Method 1, the increase is not substantial, but for high concentrations (i.e. approximately 80% R142b and more) the length of the condenser is substantially longer than for low concentrations.

The length of the evaporator fluctuates in Method 1, as can be seen from Figure 3, while methods 2 and 3 has the same tendency, i.e. an increase in length up to a concentration of 50% R22 and 50% R142b. With further addition of R142b in the mixture, these two methods indicate a reduction of the evaporator length.

Figure 4 shows the water outlet temperature of the condenser. This temperature is part of the boundary conditions in the condensers. In Method 1 it was chosen as 75 °C at a composition of 90% R22. The water outlet temperature was then increased with 5 °C for each 10% reduction of R22 in the mixture. Therefore, for an 80% R22 mixture, the maximum hot water temperature was taken as 80 °C. In methods 2 and 3 this temperature was chosen as the dew point temperature of the refrigerant.

The condenser water inlet temperatures is shown in Figure 5. Methods 1 and 3 show a constant increase. At low concentrations R142b (60% R22 and 40% R142b), the water inlet temperature in Method 2 is nearly constant. At higher concentrations R142b the temperature increase substantial.

In Figure 6 the evaporator inlet water temperature is shown. Just as in the case of the condenser, this temperature is part of the boundary conditions of the evaporator. This temperature chosen for all three methods was 5 °C higher than the refrigerant bubble point temperature.

The evaporator water outlet temperature is shown in Figure 7. Method 1 shows a linear increase while Methods 2 and 3 show a near constant temperature.

The mass flow of the mixture for the three methods is presented in Figure 8. Method 1 presents a decrease in mass flow up to a concentration of 70% R22, whereafter the mass flow increases. In contrast to this, Methods 2 and 3 show only a decrease in mass flow. The reason for the tendency of the refrigerant mass flow to decrease may be attributed to the tendency of the refrigerant density at the compressor inlet to decrease as shown in Figure 9.

All three methods indicate a reduction in input compressor power as indicated in Figure 10. Methods 1 and 2 show a near linear decrease while Method 3 indicates a more drastic decrease.

One may think that the reduction in compressor input power would imply a increase in heating COP, but this is not the case as shown in Figure 11. Method 1 shows an increase in the heating COP up to a concentration of 60% R22. As the concentration of R22 in the mixture is reduced, a reduction in heating COP follows. In Method 2 the same tendency is evident, except that the maximum heating COP occur at a concentration of 50% R22 and 50% R142b. A decrease in heating COP only occurs in Method 3.

The reason for the behaviour in the heating COP in Methods 1 and 2 can be understood if one notices the high reduction in heating capacity at low concentrations R22 in Figure 12. At first the reduction in heating capacity is less than the reduction in compressor input power leading to an increase in heating COP. As the concentration in R22 is reduced in the mixture, the decrease in heating capacity is higher than the decrease in compressor input power leading to a lower heating COP. In Method 3 the reduction in heating capacity is higher than the reduction in compressor power resulting in a decrease in heating COP.

At high concentrations R22, Method 1 and Method 2 indicate an increase in heating capacity. The maximum occur at concentrations of 80% R22 and 90% R22 respectively. Although an increase in heating capacity occurs at high concentrations R22, the increase is quite small.

The pressure ratio of the three methods is indicated in Figure 13. In Method 1 the pressure ratio stays constant due to the boundary condition of this method. Method 2 shows a decrease in compressor power up to a concentration of 50% R22 and 50% R142b. With further addition of R142b in the mixture, the compressor pressure ratio increases. Method 3 shows an increase in compressor pressure. This is expected since the condenser pressure was kept constant at 2 738 kPa and a reduction in R22 concentration in the mixture lead to a reduction in evaporator pressure.

The heating COP of Method 1 is again given in Figure 14. This figure showns more clearly, on a different scale, that the COP increases when R142b is added to R22 up to a concentration of approximately 40% R142b and 60% R22 where the maximum COP occurs. At higher concentrations of R142b the COP decreases from the maximum to the lowest point with 100% R142b. At the maximum COP point, the COP is 2.35 which is associated with hot water temperatures of 90 °C (Figure 4) and a composition of 60% R22. This should be compared to a 100% R22 heat pump with a COP of 2.28 and a maximum hot water temperature of 65 °C (Figure 4), and even to a hot water heat pump with a concentration of 25% R22 and R75% R142b, which would give the same COP as a 100% R22 heat pump, with temperatures of up to 108 °C (Figure 4). At this point however, the heating capacity will be 12.5% lower. It therefore seems that a point exists at approximately 80% R22 and 20% R142b where the COP, heating capacity and maximum hot water outlet temperature will be higher than a 100% R22 heat pump.

In Method 2 the optimum point of the heating COP is at a concentration of 50% R22 and 50% R142b which is 30% higher than the heating COP of pure R22. At this point the compressor pressure ratio has been reduced to its minimum value which is a decrease of 21% leading to longer compressor life.

In order to obtain water at 100 °C, Method 3 show that a concentration of 40% R22 and 60% R142b is adequate. This has a 38% increase in hot water temperature. Unfortunately the heating COP decreases with 35% while the pressure ratio of the compressor increases with 38%.

5. CONCLUSION

Three methods are used to investigate the influence of a non-azeotropic refrigerant mixture on the performance of a hot water heat pump. These methods are named: 1) Pressure ratio, 2) Bubble and dew point temperatures and 3) Maximum hot water and constant bubble point temperature.

The advantage of a non-azeotropic refrigerant depends on the method of comparison. In methods 1 and 3 an increase in hot water temperature is evident at low concentrations R22. There is also a price that must be paid, namely a decrease in heating capacity. Method 3 show the worst case scenario: A decrease in heating COP and an increase in compressor pressure ratio.

The hot water temperature in Method 2 is kept constant. A reduction of 21% in compressor pressure ratio occur at a concentration of 50% R22 and 50% R142b resulting in longer compressor life. The heating COP also increased with 30% resulting in energy savings.

It is difficult to find an optimum concentration of R22 and R142b. The optimum concentration depends on the application.

6. ACKNOWLEDGEMENTS

The financial support of the FRD is acknowledged.

7. REFERENCES

ASHRAE (1985), *"Advances in non-azeotropic mixture refrigerants for heat pumps"*,ASHRAE Technical Bulletin, Vol. 1, No. 9, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Climate of South Africa (1988), *"Climate statistics up to 1984"*, WB40, Weather Bureau, Department of Environmental Affairs.

Högberg M, Vamling L, and Berntsson T (1992), *"Calculation methods for comparing the performance of pure and mixed working fluids in heat pump applications"*, International Journal of Refrigeration, Vol. 16, No. 6, pp. 403-412.

Johannsen A F B (1992), *"Potential of non-azeotropic refrigerant mixtures for water-heating heat pumps in South Africa"*, Energy Branch, Department of Mineral and Energy Affairs, South Africa, Report nr.: ED 8807.

Meyer J P and Greyvenstein G P (1991), *"Hot water for homes in South Africa with heat pumps"*, Energy the International Journal. Vol. 16, No. 7, pp. 1039-1044.

Smit F J (1996), *"The influence of a non-azeotropic refrigerant mixture on the performance of a hot water heat pump."*, M. Eng. dissatations, Rand Afrikaans University, Johannesburg, South Africa.