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# A STUDY OF A WATER-TO-WATER HEAT PUMP USING FLAMMABLE REFRIGERANTS

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## ABSTRACT

This investigation compared the performance of R22 to the performance of R290 and that of flammable zeotropic mixtures, R32/290, and, R32/152a. The tests were conducted in a test apparatus specially designed to simulate a residential water-to-water heat pump. Brazed plate heat exchangers used as the evaporator and condenser allowed for counter-flow heat transfer between the refrigerant and indoor-loop and outdoor-loop heat-transfer fluids. The performance of the system was characterized by air-side capacity and Coefficient of Performance (COP) at different compressor speeds. Performance comparisons at a constant capacity were also provided. In the cooling mode at a constant-capacity, the R32/290 (50/50) mixture produced the highest COP; 8% higher than R22. In the heating mode, the COP of R290 was the highest; from 6% to 8% higher than that of the remaining fluids. The cooling COP of R32/290 was 13% lower and the heating COP of R290 was 1% higher than the respective COPs of R22 in the reference water-to-air system.

## INTRODUCTION

With the success of the Montreal Protocol effectively regulating the use of chlorine-containing refrigerants, the trapping of infrared radiation by anthropogenic gases and the resulting increase of the earth's temperature is being recognized as the most critical global environmental problem (Albritton, 1997). A heat pump or refrigeration system has a direct and indirect impact on the earth climate. The direct impact is related to the Global Warming Potential (GWP) of the refrigerant molecule when released to the atmosphere. The indirect impact is related to the emission of carbon dioxide by a fossil-fuel power plant producing electricity, and hence is affected by the electricity generation mix. The direct impact may be mitigated by constructing a leak-proof system, and the indirect impact can be reduced by increasing the system efficiency. Both impacts are combined in the concept of Total Equivalent Warming Impact (TEWI) (Calm, 1993).

Concerns about global warming have caused a re-evaluation of hydrofluorocarbons (HFCs) and their mixtures because their GWPs are considered to be excessive in some countries. This unfavorable sentiment may affect the two most touted R22 replacements, R407C (R32/125/134a) and R410A (R32/125), whose non-flammable HFC components, R134a and R125, contribute the most to those mixture's GWP values.

Flammable refrigerants, including flammable HFCs, have short atmospheric lives and low GWPs. So far, flammables have received limited attention in studies for residential heat pump applications because of liability concerns for refrigerant piping passing through the inhabited space. The chance of acceptance of flammable refrigerants would improve if they remain outdoors. The water-to-water heat pump satisfies this requirement and for this reason was selected as a system in which flammable refrigerants were evaluated.

Table 2: Selected properties of tested refrigerants at 4.4 °C (40 °F) saturation temperature

| Refrig. | GWP[a] | ASHRAE Standard 34 Safety Group | Vapor Pressure[b] kPa (psia) | $T_{dew}$ - $T_{bubble}$ °C (°F) | Vapor Absolute Viscosity kg/(m s) (lbm/(ft h)) | Liquid Thermal Conductivity W/(m °C) (Btu/(h ft °F)) | Volumetric Capacity kJ/m$^3$ (Btu/ft$^3$) |
|---|---|---|---|---|---|---|---|
| R32/290 (50/50)[c] | 335 | A2/A3 | 977 (141.7) | 6.4 (11.5) | 127.5 (23.77x10$^{-3}$) | 122.3x10$^{-3}$ (70.64x10$^{-3}$) | 8532.3 (229.0) |
| R32/152a (50/50)[c] | 395 | A2/A2 | 552 (80.0) | 7.8 (14.0) | 139.0 (25.92x10$^{-3}$) | 133.4x10$^{-3}$ (77.10x10$^{-3}$) | 5059.7 (135.8) |
| R22 | 1500 | A1 | 574 (83.3) | 0 | 154.1 (28.73x10$^{-3}$) | 94.7x10$^{-3}$ (54.70x10$^{-3}$) | 4892.1 (131.3) |
| R290 | 20 | A3 | 542 (78.6) | 0 | 101.3 (18.89x10$^{-3}$) | 103.9x10$^{-3}$ (60.06x10$^{-3}$) | 4344.4 (116.6) |

[a] integrated time horizon 100 years; $CO_2$ used as a reference
[b] zeotropic mixture pressure at a 4.4°C (40°F) average of dew-point and bubble-point temperatures
[c] mass fraction , %

Table 3: Test conditions for water-to-water heat pump tests

| Location | Cooling | Heating | Tolerance |
|---|---|---|---|
| Indoor dry-bulb temperature | 26.7 °C (80.0 °F) | 21.1 °C (70.0 °F) | ±0.3 °C (±0.5 °F) |
| Indoor wet-bulb temperature | 19.4 °C (67.0 °F) | 10.0° C (50.0 °F)[a] | ±0.3 °C (±0.5 °F) |
| Outdoor-loop HTF inlet temperature[b] | 25.0 °C (77.0 °F) | 0.0 °C (32.0 °F) | ±0.3 °C (±0.5 °F) |
| Refrigerant evaporator superheat and condenser subcooling | 3.9 °C (7.0 °F) | 3.9 °C (7.0 °F) | ±1.7 °C (±3.0 °F) |

[a] maximum allowed
[b] for condenser in the cooling mode; for evaporator in the heating mode

In addition, R22 was tested at the conditions simulating the operation of a water-to-air direct-expansion heat pump. For these tests, the saturation temperature of R22 in the evaporator was controlled to match the average HTF temperature across the air coil during the water-to-water cooling tests. Water-to-air heating tests were conducted in a similar manner with the condenser saturation temperature matching the average air-coil HTF temperature during a water-to-water heating test. These water-to-air heat pump tests were conducted to provide an indication as to what extent glide matching between the indoor HTF and zeotropic mixture could mitigate the penalty associated with the indoor-loop HTF/refrigerant heat exchanger needed to isolate flammable refrigerants outdoors.

## 4. RESULTS

### 4.1 Cooling Mode Performance

Figure 2 shows air-side capacities in the cooling mode as a function of compressor speed. The R22-REF designation is for tests representing the operation of a water-to-air heat pump. Other data points, including those designated by R22, are for tests of a water-to-water system. Straight lines are used to indicate the trend of test data for different fluids. The locations of the capacity lines for different fluids are in the order of their volumetric capacities. For R22 at a given compressor speed, the capacity of the water-to-water heat pump is lower by more than 20% than that for the water-to-air system. This capacity penalty is caused by the lower evaporator temperature (and pressure) needed to transfer heat in the R22/indoor HTF heat exchanger. Because of the lower pressure, compressor and system capacities are lower as well.

Additionally, capacity was measured on the HTF-side in the indoor and outdoor loops. The HTF temperature change was measured by 10-junction thermopiles located in oil filled wells in the copper HTF lines, and the volumetric flowrate was measured using turbine flow meters. HTF density and specific heat values were supplied by the fluid manufacturer. A 3.5% agreement or better was obtained between the air-side and HTF-side capacities. Table 1 shows measurement uncertainty for the most important measurements.

Table 1: Measurement uncertainties (95% confidence level)

| Quantity | Range | Relative Uncertainty |
|---|---|---|
| Temperature | -21 °C to 93 °C (-15 °F to 200 °F) | ±0.5 °C (±0.9 °F) |
| Refrigerant Composition | 0 to 100 % mass fraction | ±1.0%* |
| RPM | 0 RPM to 3000 RPM | ±5 RPM |
| Torque | 0 to 39.5 N·m (0 to 350 in·lbf) | ±1.0% |
| Compressor Shaft Power | 0.5 kW to 2.5 kW (1700 Btu/h to 8500 Btu/h) | ±1.1% |
| Air-Side Capacity | 2 kW to 11 kW (6800 Btu/h to 37500 Btu/h) | ±3.3% |
| Air-Side COP | 3.5 to 7.3 | ±3.4% |
| Heat Transfer Fluid Mass Flow | 226.8 to 1632.9 kg/h (500 to 3600 lbm/h) | ±0.8% |
| Refrigerant Mass Flow | 27.2 to 226.8 kg/h (60 to 500 lbm/h) | ±0.8% |

* Indicates percent of the reading.

## 2. REFRIGERANTS

Table 2 shows the refrigerants tested and their selected properties. R22 was tested as a reference fluid. The mixtures were selected based on previous studies and simulations performed with a semi-theoretical model, CYCLE-11. The tested fluids included flammable HFCs, R32 and R152a, because they have very favorable thermophysical properties and small GWPs. R290 (propane) was also tested as a single component fluid because it has the closest saturation pressure to that of R22. With the exception of R22, all of the tested fluids were flammable. A naphthenic mineral oil was used for R22, R290, and all mixtures containing hydrocarbons, while a polyol ester synthetic oil was used for the R32/152a mixture.

## 3. EXPERIMENTAL PROCEDURE

Cooling and heating tests were performed at the operating conditions shown in Table 3 as specified by ARI Standard 330-93. A test series for a given refrigerant started in cooling. The indoor and outdoor-loop HTF mass flowrates were adjusted to obtain 5.6 °C (10.0 °F) HTF temperature difference across the brazed plate evaporator and condenser. At the same time evaporator superheat and condenser subcooling were set to 3.9 °C (7.0 °F) by adjusting the refrigerant charge and setting of the needle valve expansion device. Charge was not adjusted in the heating mode. In the heating mode the indoor-loop HTF was routed through the condenser and the outdoor-loop HTF through the evaporator. The HTF volumetric flowrates in both loops were left unchanged from those established in cooling, but now the cooling evaporator flow was supplied to the condenser; and the condenser HTF flow from the cooling mode was supplied to the evaporator. Heating HTF temperature glides resulted from heat transfer in the brazed plate evaporator and condenser. The described procedure was repeated for each test series at different compressor speeds. Airflow across the finned tube coil was established in cooling to obtain a 11.1 °C (20 °F) temperature change and was left unchanged for heating tests.

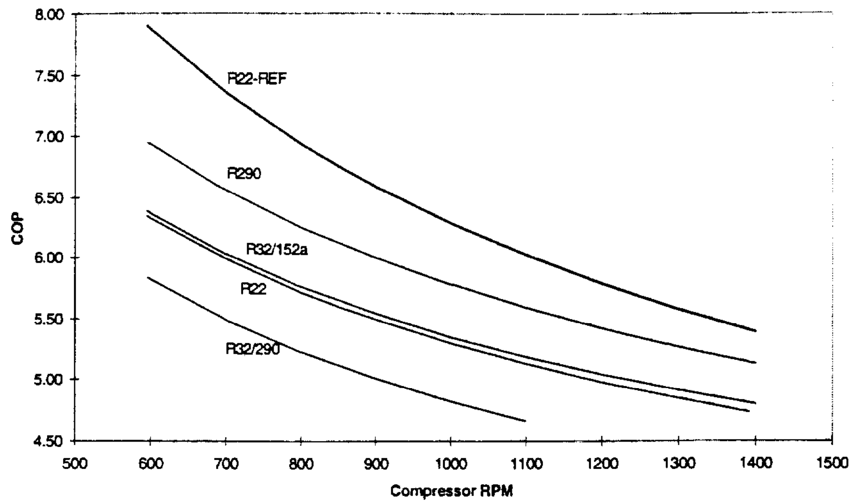


Figure 3 - Cooling COP as a function of compressor speed

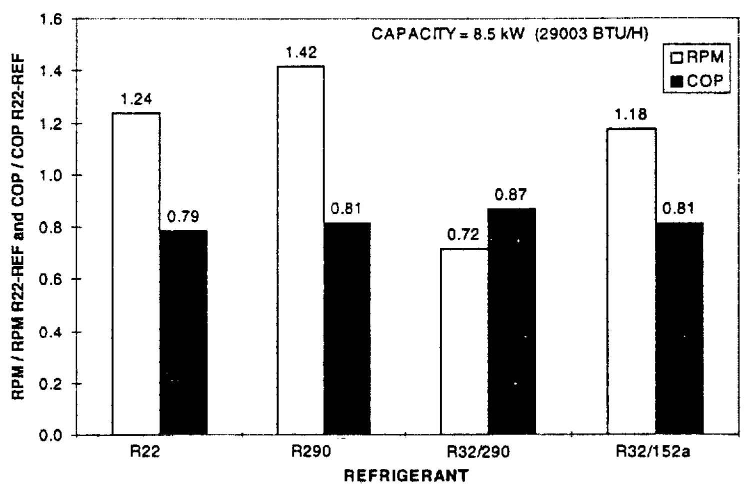


Figure 4 - Cooling COP and compressor speed comparison at constant capacity

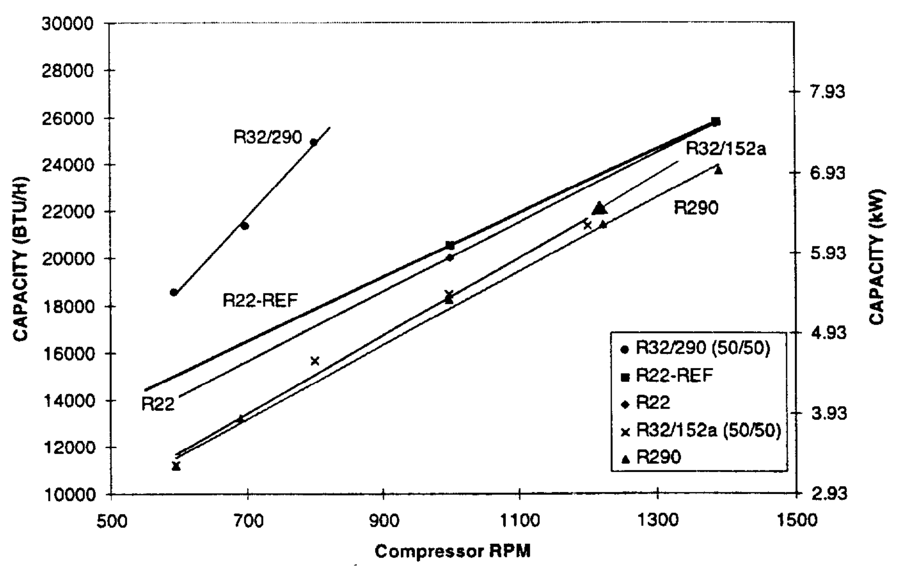


Figure 5 - Heating capacity as a function of compressor speed

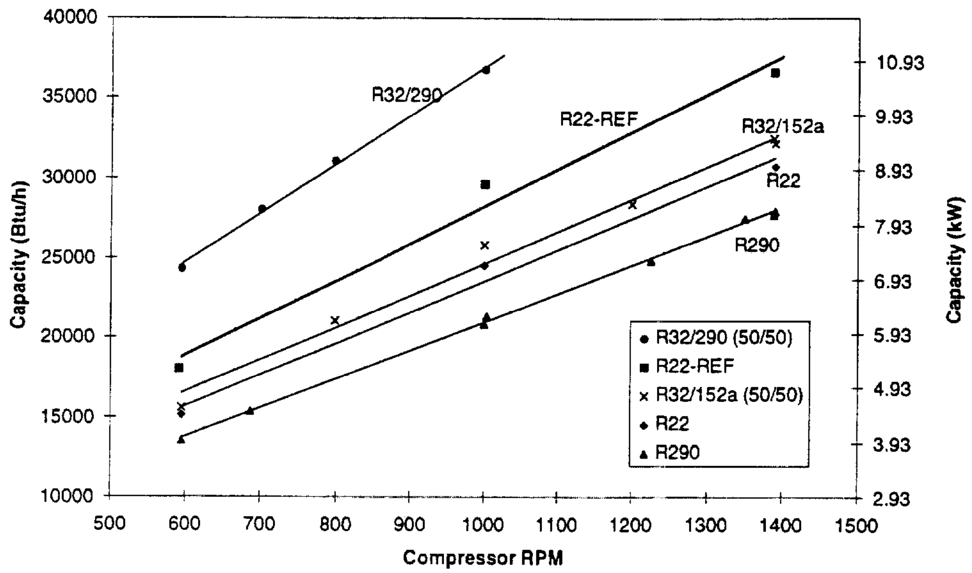


Figure 2 - Cooling capacity as a function of compressor RPM

Figure 3 presents cooling mode COPs as a function of compressor speed. The COP lines were generated by the algorithm incorporating the functional relationship between compressor speed, system capacity, and energy input to the compressor. The spread between R22 COPs for the water-to-air and water-to-water systems is due to the thermodynamic penalty associated with the R22/indoor HTF heat transfer.

The goal of this study was to compare COPs of different refrigerants at the same duty, in this case at the same air-side capacity. This approach is in accord with the postulate by McLinden and Radermacher (1987) that a fair comparison between different fluids has to be done at the same heat exchanger heat flux. COP ratios at a constant capacity shown in Figure 4 were calculated from capacity and power input characteristics correlated with compressor speed. The figure shows that none of the flammable refrigerants could achieve the COP of R22 in the water-to-air system. The COP of the best refrigerant in the water-to-water heat pump, R32/290, is 13% lower than the COP of the R22 water-to-air system. The COPs of the remaining refrigerants are comparable.

**4.2 Heating Mode Performance**

Figures 5 and 6 show heating capacities and COPs, respectively, as a function of compressor speed. One noticeable difference between the cooling and heating results for R22 is that heating water-to-air and water-to-water system capacities are similar, while in cooling the difference between these capacities is greater than 20%. This can be explained by the fact that in heating both systems have approximately the same compressor suction saturation temperature because their evaporators use the same heat source (ground-loop HTF), while in cooling the suction saturation temperature is higher in the water-to-air system. In heating, the water-to-water heat pump has a higher discharge saturation temperature than the water-to-air system, however, this has a small impact on performance. Another difference between the cooling and heating results is that the heating capacities of R290 and R32/152a are similar while the cooling capacity of R32/152a was significantly higher than the capacity of R290. The converging trend in theoretical volumetric capacities of R290 and R32/152a with lowering saturation temperature can serve as one explanation of this result.

### 4.3 Entropic Average Temperature Difference

The use of zeotropic refrigerants makes the heat transfer process occurring in the evaporator and condenser more difficult to characterize due to the varying temperature heat transfer process. By introducing the concept of an entropic average temperature, the heat transfer process between the zeotropic refrigerant and HTF may be simplified to an equivalent heat transfer process between two isothermal reservoirs (Herold 1989, Alefeld 1987). The entropic average temperature is defined by the following equation:

$$T_{sa} = \frac{\dot{Q}}{\int \frac{\delta \dot{Q}}{T}} \tag{1}$$

Equation (1) states that the entropic average temperature is equivalent to the temperature of an isothermal reservoir which is exchanging the same amount of heat energy as the variable temperature fluid while generating the same amount of entropy. Assuming an incompressible heat transfer fluid with constant specific heat, the entropic average temperature for a temperature change is determined by $T_{sa}=(T_2-T_1)/ln(T_2-T_1)$. Applying the definitions in eq. (1) to the refrigerant side yields an entropic average temperature given by $T_{sa}=\Delta h/\Delta s$.

Figure 8 illustrates the use of the concept by showing the entropic average temperature differences in the evaporator for the cooling tests. The most noticeable feature of this figure is the difference between the pure refrigerants and the zeotropic mixtures. Between the two mixtures, the glide of R32/290 more closely matched that of the HTF, which resulted in the lowest temperature difference.

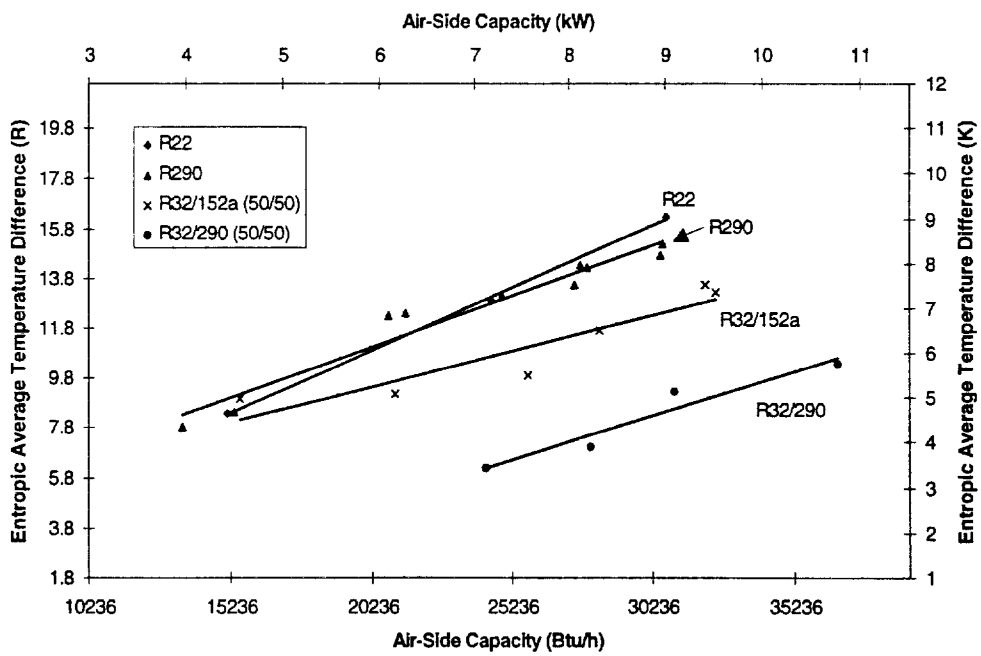


Figure 8 - Cooling mode entropic average temperature differences in the evaporator

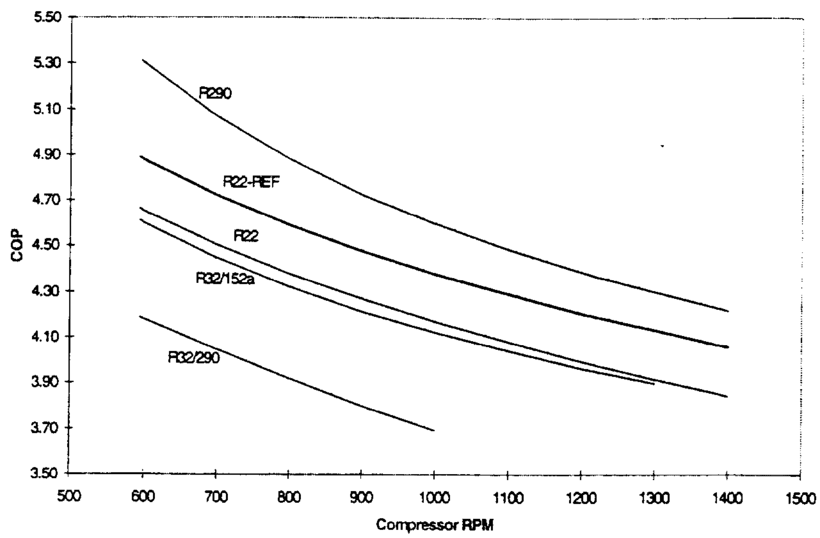


Figure 6 - Heating COP as a function of compressor speed

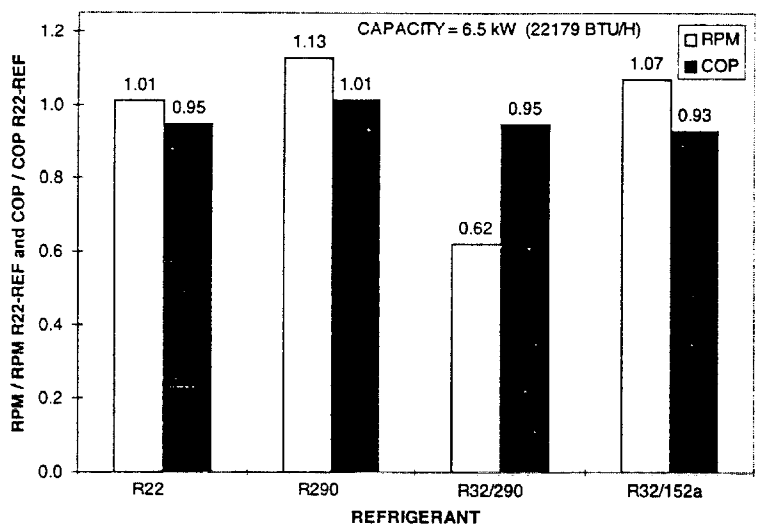


Figure 7 - Heating COP and RPM comparison at constant capacity

Since the mixtures' glide was in the order of 10 °C (18 °F), the mixtures experienced over-gliding with respect to the HTFs. This apparently had a similar impact on performance for the mixtures as under-gliding had on the single-component fluids, R290 and R22.

R290 COP is approximately 8% higher than that for R32/152a, yet their capacities are nearly equal. Compressor performance was investigated to explain this occurrence. The factor which was the primary contributor to the improved performance of the R290 was its lower shaft torque. Discharge temperatures for the R32/152a mixture averaged 28 °C (50 °F) higher than those seen with R290. The lower shaft torque for R290 could also be due to oil effects. R290 would tend to remain in solution with the napthenic mineral oil much more readily than R32/152a would remain in solution with the polyol ester lubricant at the lower temperatures seen during heating. With the glide-matching effect mitigated during the heating tests, R290 showed the best performance among all fluids because of its best overall thermophysical properties and oil solubility.

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L' ÉTUDE SUR UNE POMPE À CHALEUR, DU TYPE EAU-EAU, UTILISANT DES RÉFRIGÉRANTS INFLAMMABLES


RESUME : Cette enquête compare la performance du R22 avec celle du R290 et avec celle des mélanges inflammables zéotropic, R32/290 et R32/152a. Les tests furent exécutés dans un appareil spécialement conçût pour simuler une pompe à chaleur pour habitation individuelle. Les plaques cuivrées des échangeurs de chaleur utilisées pour l'évaporateur et le condensateur permirent un transfert contre-courant de la chaleur entre le réfrigérant et la boucle-interne et la boucle-extérieure des fluides échangeurs de chaleur. La performance du système est caractérisée par la capacité du coté air et le Coefficient de Performance (COP) à différentes vitesses du compresseur. Comparaisons des performances à capacité constante sont aussi données. Dans le régime de refroidissement à capacité constante, le mélange (50/50) du R32/290 produisit le COP le plus élevé: 8% plus élevé que celui du R22. Dans le régime de chauffage, le COP du R290 était le plus élevé: jusqu'à 8% plus élevé que celui du restant des fluides. Le COP de refroidissement du R23/290 était de 13% plus bas et le COP de chauffage du R290 était 1% plus élevé que les COP respectifs pour le R22 dans le système de référence eau-air.

## 5. CONCLUDING REMARKS

The study allows one to make a relative performance comparison of the tested fluids in the water-to-water system to R22 in the water-to-air system. If the water-to-water heat pump alone is considered, R32/290 showed the best cooling COP, which was greater by 6% to 9% than the COPs of R32/152a, R290, and R22. The glide of the R32/290 mixture well matched the glide of the HTFs in the cooling mode. The poorer COP of the other mixture can be explained by R32/152a over-gliding the HTFs and the expected lack of refrigerant/lubricant miscibility in the evaporator.

When comparing results for water-to-water and water-to-air heat pumps, the cooling glide matching of R32/290 did not overcome the thermodynamic penalty associated with refrigerant/indoor HTF heat transfer. The cooling COP for R32/290 was 13% lower than the COP of the water-to-air system charged with R22. In the heating mode, the COP of R290 was within 1% of the R22 water-to-air system. It can be speculated that the COP of R32/290 could exceed that of R22 by up to 5% if the mass flow of HTFs were optimized for the heating mode. Considering the substantially lower cooling COP of the water-to-water heat pump, its TEWI will be higher than that of the R22 water-to-air heat pump under most probable calculation assumptions

A note has to be made that the presented results do not include the COP penalty associated with HTF pumping power. The percent degradation in COP caused by adding a 5.2 m (17 ft) pumping head loss at a rate of 3.6 W/(L/min) (13.6 W/GPM) (ARI 1993) is approximately 5.5%. Therefore the average performance of the R32/290 mixture, which was within 13% of the R22 water-to-air system, should be lowered by this amount to place it 18.5% below the performance of the R22 water-to-air system at a given capacity. In heating, the R32/290 mixture has the potential to match the performance of the water-to-air R22 system if HTF mass flowrates are optimized.

Practical reasons make it difficult to reduce temperature difference between the refrigerant and HTF ($\Delta T$) to below 5 °C (8 °F). Small $\Delta T$s require large heat exchangers. To avoid large pressure drop penalty, low fluid velocities are required. Since the heat-transfer coefficient is detrimentally affected by low flow velocity, the capacity of the heat exchanger will be also reduced. Fluid flow velocity may be also limited at the low end by the lubricant return consideration. Hence, a designer will approach the economic and practical limit for the size of the heat exchangers he can specify. Another difficulty in attaining low $\Delta T$s stems from the fact that conventional vapor-compression systems operate with a few degrees of evaporator superheat and condenser subcooling, and this puts a constraint on the minimum attainable $\Delta T$. An intra-cycle heat exchanger for heat transfer between high-quality refrigerant leaving the evaporator and low-quality refrigerant leaving the condenser would facilitate lower $\Delta T$s. For zeotropic mixtures, such heat transfer would provide an additional benefit of increased evaporator pressure and improved COP.

## NOMENCLATURE

COP = coefficient of performance
GPM = volumetric flowrate, U. S. gallons per minute
HTF = heat-transfer fluid
$Q$ = heat-transfer rate
RPM = revolutions per minute
$T_{sa}$= entropic average temperature
$T_2$, $T_1$ = heat-transfer fluid exit and inlet absolute temperatures, respectively
$\Delta h$ = change in specific enthalpy of refrigerant
$\Delta s$ = change in specific entropy of refrigerant
$\Delta T$ = average effective temperature difference between refrigerant and HTF

The use of zeotropic mixtures in the water-to-water heat pump was originally proposed by Didion (Mulroy et al., 1988) as a method for improving heat pump performance for low-lift/high-glide applications. A later simulation study showed that the Coefficient of Performance (COP) of the water-to-water system charged with a zeotropic mixture can approach the COP of the direct-expansion R22 system if large counter-flow heat exchangers are used as the evaporator and condenser (Domanski et al., 1994). The follow-up experiment in a general-purpose mini-breadboard heat pump (MBHP) showed the importance of adequate sizing of the heat exchangers (Choi et al., 1996). The evaporator and condenser of the MBHP did not have sufficient capacity to obtain a sizable gain in COP due to glide matching between the zeotropic mixture and heat-transfer fluid. The current study was performed in a specially designed apparatus with a large brazed plate evaporator and condenser to ensure adequate capacity.

## 1. EXPERIMENTAL APPARATUS

Figure 1 shows a schematic of the experimental set-up. The vapor-compression system consisted of a reciprocating compressor, brazed plate condenser and evaporator, manually adjustable expansion device, and an accumulator. The brazed plate heat exchangers were installed in a counter-flow configuration. The compressor was a two-cylinder, open-drive reciprocating design with total piston displacement of 121 $cm^3$ (7.4 $inch^3$) and a speed range of 300 RPM to 3000 RPM. It was driven by an explosion-proof motor, whose speed was controlled by a variable frequency drive. The compressor and electric motor were coupled with a torque and RPM transducer to allow energy input to the compressor to be measured at the compressor shaft. The system was designed for a cooling load of 10.6 kW (36 kBtu/h).

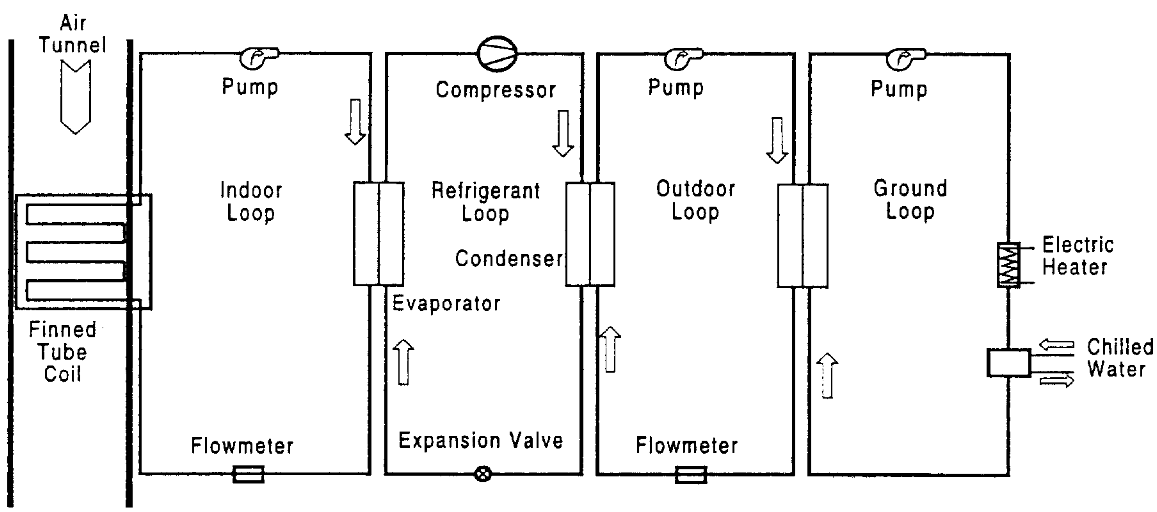


Figure 1 - Configuration of the apparatus in the cooling mode

The indoor-loop, outdoor-loop, and ground-loop were filled with the same heat-transfer fluid (HTF); a 70/30 (mass) mixture of water and ethylene glycol. The ground was simulated by the outdoor-loop/ground-loop heat exchanger with the ground-loop HTF being chilled by in-house chilled water and heated by an electric heater. The schematic shown in Figure 1 corresponds to the cooling mode arrangement. For the heating mode, the valve and piping system (not shown in the figure) allowed rerouting the flow of the outdoor-loop HTF through the evaporator and that of the indoor-loop through the condenser.

System capacity was measured on the air-side using the air-enthalpy method. A 25-junction thermopile measured the air temperature change, and dew-point instruments measured the change of moisture content. A venturi nozzle was used for measuring the volumetric flowrate of air.