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EVALUATION OF FLAMMABLE REFRIGERANTS FOR USE IN A WATER-TO-WATER RESIDENTIAL HEAT PUMP

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1. SUMMARY

This paper evaluates the performance of R-22, R-290, R-290/600a (70/30), and R-32/152a (50/50) for application in a water-to-water residential heat pump for space cooling and heating. The tests were performed in a laboratory apparatus at different compressor speeds to allow the comparison of refrigerants for the constant-compressor-speed and the constant-capacity criteria.

Comparison of results for the same system capacity showed R-32/152a to be the best performer due to good glide matching in the heat exchangers and its excellent thermodynamic and transport properties. The hydrocarbon mixture, R-290/600a, had the highest Coefficient of Performance (COP) at a given compressor speed, but its COP at the constant-capacity criterion was the lowest. This low COP was in disagreement with an earlier simulation study where R-290/600a and R-32/152a had a similar COP. The test data showed that the low-volumetric-capacity R-290/600a had an excessive pressure drop at the constant-capacity comparison. This was related to the high compressor RPM needed for R-290/600a to reach the target capacity. With an optimized compressor and heat exchangers using larger diameter tubes, R-290/600a should have a COP comparable to that of R-32/152a. The study demonstrates the need for both simulations and laboratory methods in evaluating alternative refrigerants.

2. INTRODUCTION

The phase-out of R-22 by the Montreal Protocol /1/ spurred a notable research effort to find a replacement fluid. The elimination of R-22 is of particular consequence for manufacturers of split residential air conditioners and heat pumps, in which R-22 has been used exclusively. The main thrust of research has been directed toward finding a chlorine-free, nonflammable R-22 substitute for use in the conventional air-to-air system, without departing from the conventional system design. The leading candidates for this application are R-32/125/134a, R-32/125, and R-32/134a. All three components of these mixtures have zero ozone depletion potential (ODP). However, R-125 and R-134a have significant global warming potential (GWP), which could make them a possible target of future international regulations.

Flammable refrigerants have short atmospheric lives and low GWPs but thus far have received limited attention in studies for residential heat pump application. Previously one heat pump manufacturer made a considerable effort to develop a propane (R-290) air-to-air heat pump. Propane has excellent transport properties, a similar vapor pressure to that of R-22, and could replace R-22 without much technical difficulty or added expense were it not for its flammability. It was estimated that the Underwriters Laboratory (UL) safety requirements could increase the cost by at least 25 percent /2/.

The main concern when using flammable refrigerants in a split residential system is that the refrigerant's piping passes through the conditioned space. The safety concerns would be minimal if the flammable working fluid remained outdoors. This can be accomplished if a secondary fluid is used to exchange heat between the conditioned space and refrigerant, as is done in the water-to-water heat pump shown schematically in Fig. 1. With this design dictated by safety concerns, the authors decided to focus on a water-to-water heat pump (fig. 1), which

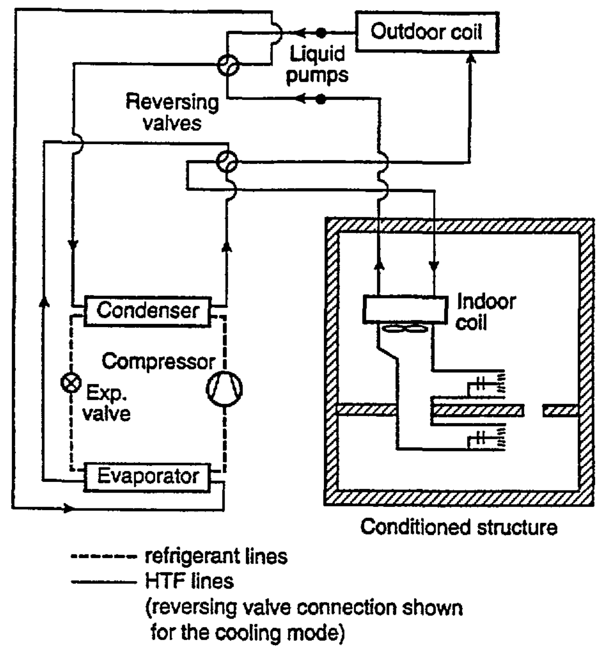

Fig. 1 - Schematic of a water-to-water heat pump*.

also has a secondary fluid on the outdoor side. A water-to-air conventional earth-coupled heat pump charged with R-22 was also considered, to provide reference performance data. This paper is an abridged version of a more comprehensive NIST Interagency Report to the U.S. Department of Energy /3/. The laboratory work described here follows a computer simulation study that evaluated the water-to-water system charged with flammable refrigerants /4/.

3. REFRIGERANTS

The refrigerants used in this study and their selected properties are shown in Table I. The compositions shown for mixtures are mass-based. The (70/30) composition of the R-290/600a mixture was selected to produce a zeotropic mixture with a meaningful temperature glide without significantly diminishing the volumetric capacity due to the presence of R-600a. The 50/50 composition of R-32/152a was chosen because, based on simulation results, this mass ratio was expected to have a capacity similar to R-22. The hydrocarbons, R-290 and R-290/600a have *zero GWPs and* those for R-22 and R-32/152a (50/50) are 1700 and 370, respectively (ITH=100). Except for R-22, the tested fluids are flammable.

TABLE I: Selected properties of studied refrigerants at atmospheric pressure.

Refrigerants	Molecular Weight (g/mol)	T_{dew} (°C)	$T_{dew}-T_{bub}$ (°C)	Liquid [1] Thermal Conductivity (W/(m°C))	Liquid [1] Absolute Viscosity (mPa·s)
R-22	86.47	-40.8	0.0	0.1286	3369
R-290	44.10	-42.1	0.0	0.1523	1999
R-290/600a (70/30)	47.54	-30.8	6.7	0.1448	2171
R-32/152a (50/50)	58.34	-35.3	8.1	0.1798	3039

[1] Obtained from REFPROP /5/ at the normal bubble point

* Originally proposed by D.A. Didion at the 1987 U.S. Department of Energy program review as a method for improving a heat pump's performance by wide glide mixture

All four working fluids were used for the water-to-water tests, but only R-22 was used for the water-to-air tests. The Carnahan-Starling-DeSantis equation of state routines /5/ were used for calculating thermodynamic properties.

4. TEST FACILITY AND PROCEDURE

This study was performed using a laboratory apparatus known as the Small Breadboard Heat Pump (SBHP). It has been employed in other NIST investigations and was originally described by Pannock and Didion /5/. Figure 2 shows a schematic of the SBHP. The system employed an open two-cylinder reciprocating compressor with total displacement of 45 cm^3. The inverter allowed to maintain compressor speed at the selected RPMs. Commercially available transducers measured compressor speed and torque for calculating compressor power.

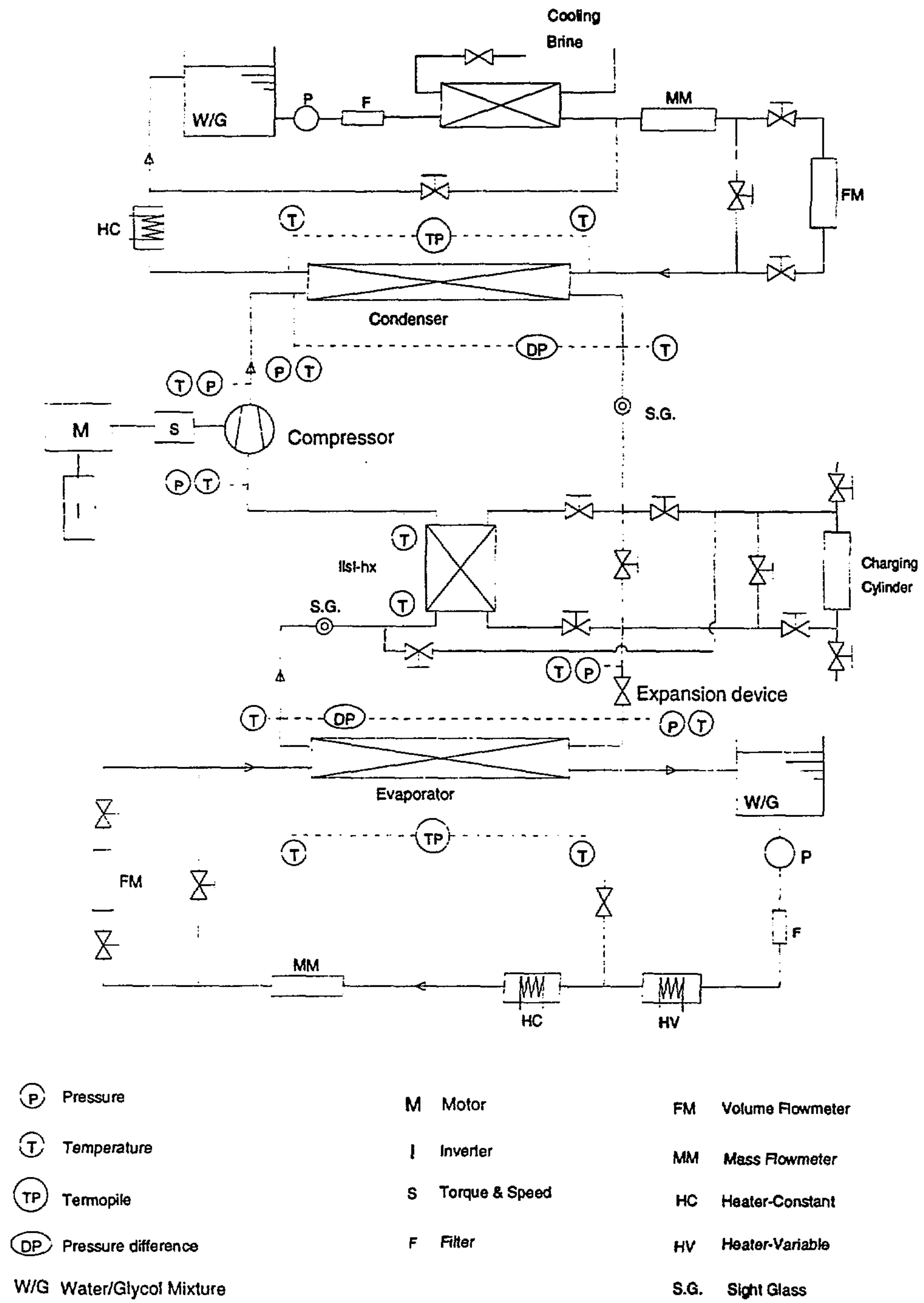

Fig. 2 - Schematic of the Small Breadboard Heat Pump.

The condenser and evaporator were counterflow heat exchangers in which the phase-changing refrigerant flowed in the inner tube, and the heat-transfer fluid (HTF) flowed through the annulus. The HTF was a 60/40 mixture of water and ethylene glycol. Some tests employed the liquid-line/suction-line heat exchanger (llsl-hx), in which the condensed liquid was further subcooled by superheating the refrigerant vapor leaving the evaporator.

Capacity was measured on the HTF side. A Coriolis-type mass flow meter provided HTF mass flow rate. Temperature change of the HTF across each heat exchanger was measured by a ten-junction thermopile located in wells at the inlet and outlet of the condenser and evaporator. All temperature measurements were made using type-T thermocouples. For measuring pressure, five pressure transducers and two differential pressure transducers were installed. The combined standard uncertainty at 68% confidence level for capacity and COP was 3.5% and 3.7%, respectively, based on the cooling mode test results of R-32/152a at 1000 compressor RPM /4/.

Test conditions, shown in Tables II and III, were set to allow comparison of the performance of the R-22 water-to-air (reference) heat pump with that of the water-to-water heat pump charged with different refrigerants. For the water-to-air system, the temperature of R-22 in the indoor coil was maintained at 10 °C in the cooling mode and 42 °C in the heating mode. For the water-to-water system, the temperature of the HTF fluid was controlled at the specified inlet and outlet values to ensure the same HTF average temperature as that of R-22 in the water-to-air system. The HTF temperature in the outdoor coil was the same for both systems.

After the refrigerant was charged, the compressor speed was fixed to a specific value. The inlet temperatures of the HTFs were maintained by electric heaters, and the mass flow rates were adjusted by automatic valves to provide the desired outlet temperatures. The HTF temperatures were maintained within ±0.1 °C of the target temperatures. The condenser subcooling and evaporator superheat were set to 5 °C ±2 °C. To obtain the specified superheat and subcooling, the expansion valve opening and amount of refrigerant charge were adjusted. The data acquisition system took measurements after the system maintained the above parameters for at least 15 minutes. If the system was charged with a refrigerant mixture, its composition was measured with a gas chromatograph. The analysis was performed using a small amount of vapor taken from the discharge line of the compressor after the system reached steady-state operation. This procedure was repeated for every compressor speed. The measured composition was within 1.5% of the target composition (70/30 for R-290/600a and 50/50 for R-32/152a).

TABLE II: Test conditions for the R-22 water-to-air (reference) system.

		Cooling Condition	Heating Condition
HTF of evaporator	Inlet	To maintain R-22 saturation temperature of 10.0 °C	5.0 °C
	Outlet		1.0 °C
HTF of condenser	Inlet	29.0 °C	To maintain R-22 saturation temperature of 42.0 °C
	Outlet	35.0 °C	

TABLE III. Test conditions for the water-to-water system.

		Cooling Condition	Heating Condition
HTF of evaporator	Inlet	13.0 °C	5.0 °C
	Outlet	7.0 °C	1.0 °C
HTF of condenser	Inlet	29.0 °C	39.0 °C
	Outlet	35.0 °C	45.0 °C

5. RESULTS

Test results are presented in three categories: (1) for different compressor speeds as they were obtained in the laboratory, (2) for selected compressor speed, and (3) for selected capacity.

5.1 Results versus Compressor Speed

Fig. 3, 4, 5, and 6 show capacity and COP for the cooling and heating operating conditions. For both cooling and heating, the basic trend is the same in that the highest capacity was measured at the highest compressor speed for each refrigerant, while the highest COP was obtained at the lowest RPM. The noticeable difference between cooling and heating is that the cooling capacity and COP of R-22 are higher in the conventional water-to-air heat pump than in the water-to-water heat pump, while in the heating mode they are almost the same. The performance difference in the cooling mode was caused by the thermodynamic penalty associated with heat transfer in the evaporator between the refrigerant and HTF in the water-to-water system. In the heating mode, this heat-transfer penalty was on the condenser side and did not degrade performance significantly.

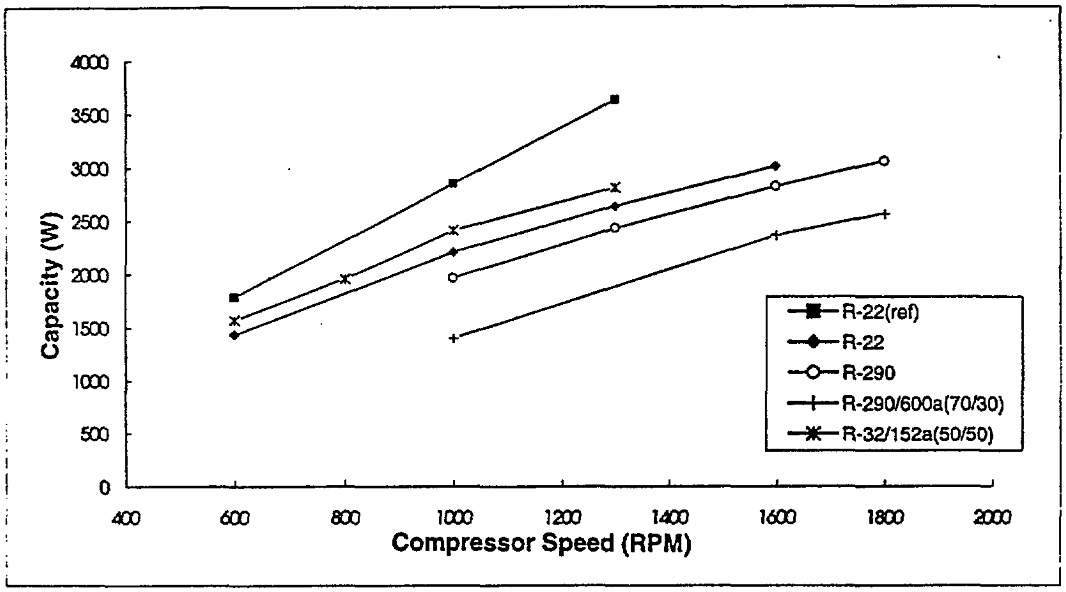

Fig. 3 - Capacities for cooling condition

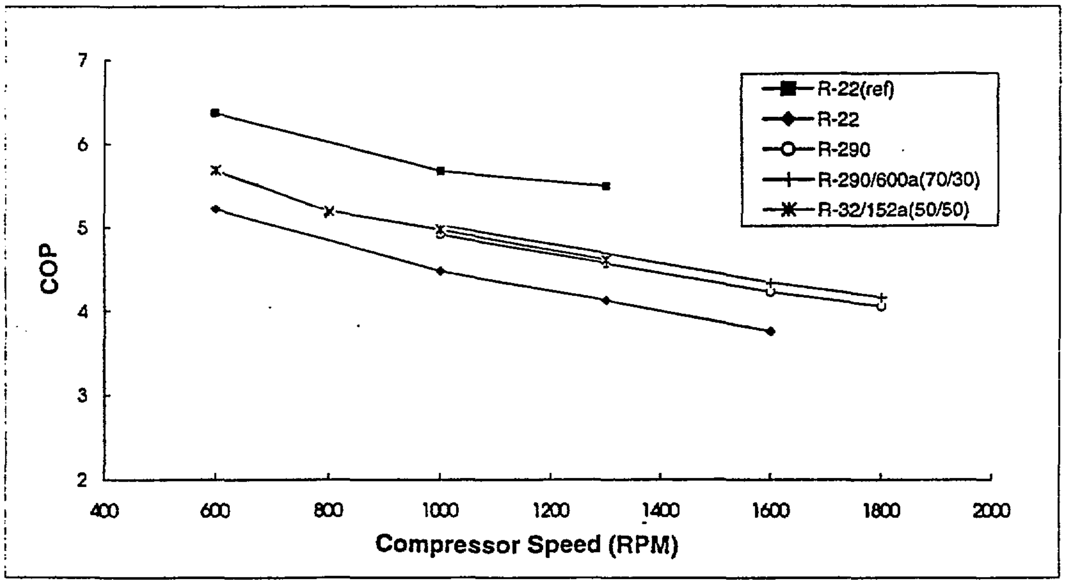

Fig. 4 - COPs for cooling condition

Fig. 7 presents temperature glides in the evaporator for the cooling mode. The glides were calculated by subtracting the evaporator inlet temperature from the saturated vapor temperature at the evaporator exit. Since refrigerant flow is associated with a pressure drop, some temperature change could be observed even for single-component refrigerants because of a change in refrigerant saturation pressure. The temperature glides for the R-290/600a and R-32/152a mixtures are a result of their pressure drop and a difference between their dew-point and bubble-point temperature. In the evaporator, the saturation temperature change due to pressure drop is opposite to the mixture's glide temperature. In the condenser, the saturation temperature drop due to pressure drop will add to the mixture's temperature glide.

In the evaporator, the glides for the mixtures have the opposite sign from that for R-22 and R-290 because the mixtures' glides are dominated by the zeotropic effect while the R-22 and R-290 glides are due to a pressure drop only. For this reason, a parallel flow arrangement in the evaporator would result in a better glide matching for R-22 and R-290 than the counter-flow arrangement used during the tests.

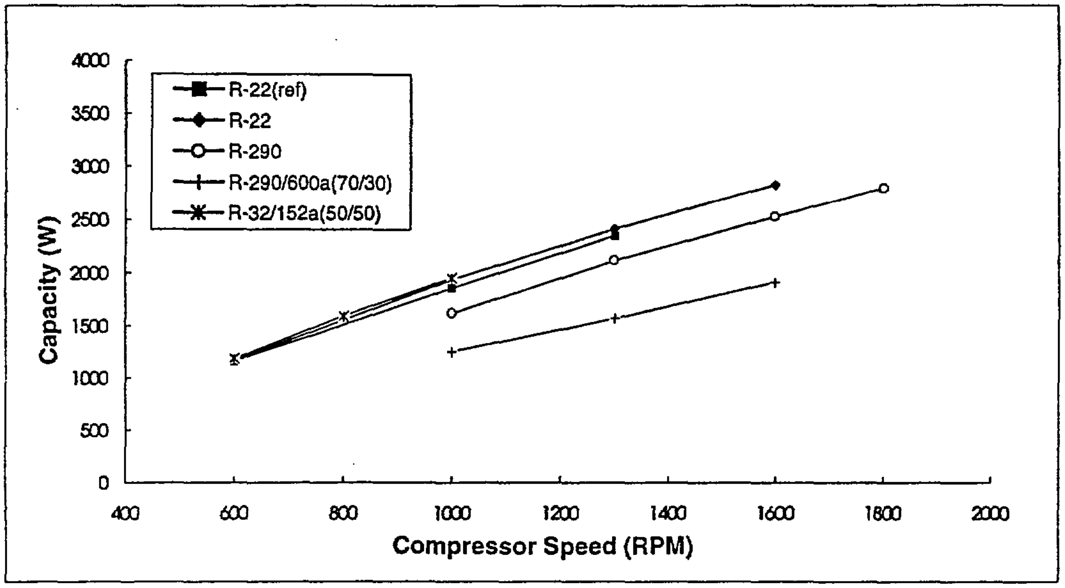

Fig. 5 - Capacities for heating condition

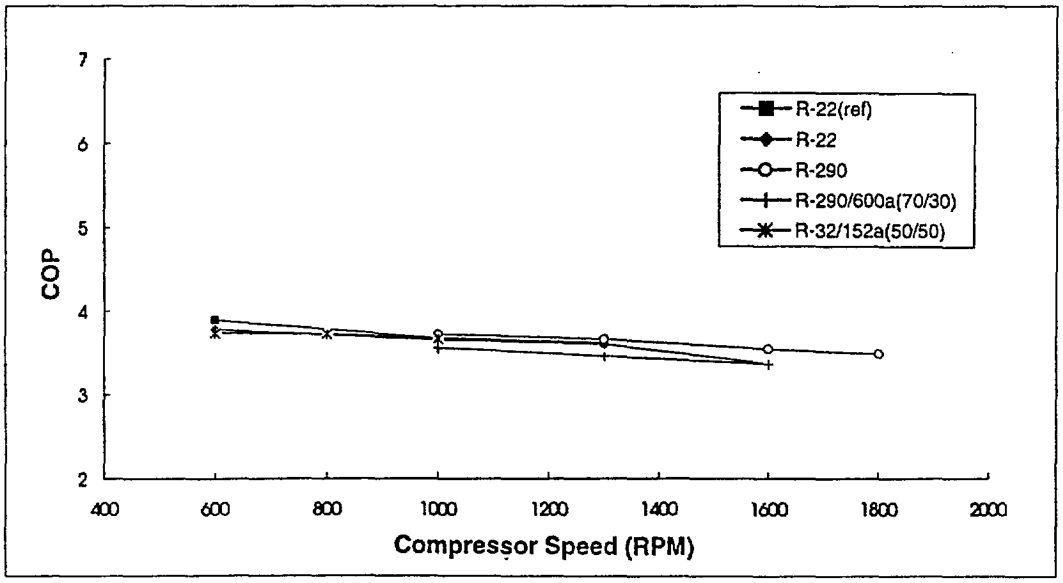

Fig. 6 - COPs for heating condition

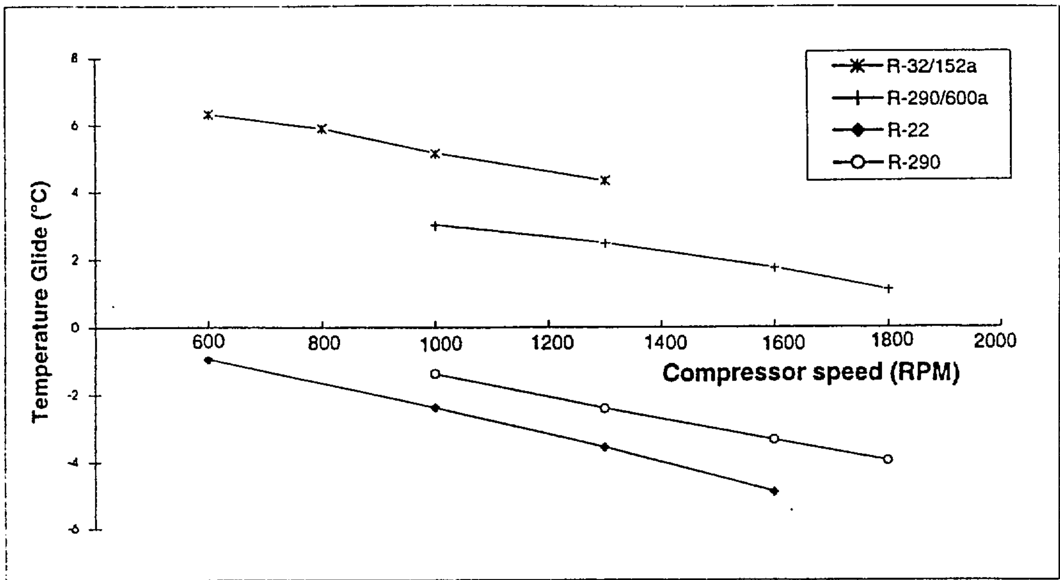

Fig. 7 - Temperature glide in evaporator, cooling condition

The glides displayed some sensitivity to the compressor speed, which was distinct in the evaporator and was weak in the condenser. This can be explained by considering that a certain increase in RPM (and refrigerant mass flow rate) requires an increase of saturation temperature in the condenser and a decrease in the evaporator by a similar value (since the original ΔTs were the same). The resulting change in pressure drop will depend on the change in refrigerant mass flow rate (the same for both heat exchangers) and on the change in vapor density; the latter will be negative for the condenser and positive for the evaporator. This will result in a larger increase in pressure drop for the evaporator than for the condenser. Considering further that dT/dP is larger for the evaporator (i.e., farther from the critical point) than for the condenser, the change in pressure drop and saturation temperature is larger in the evaporator.

5.2 Capacity and COP at 1000 Compressor RPM

Comparing results obtained at the same compressor speed corresponds to a "drop-in" evaluation for a given system. Figure 8 shows capacity and COP obtained for the water-to-water system referenced to those of the water-to-air system. The cooling capacities and COPs were below those of the reference system with the lowest capacity measured for R-290/600a. In the heating mode, all COPs were approximately at the same level. Also, the capacity of R-22 and R-32/152a were comparable to that of the reference system, but capacities of R-290 and R-290/600a were lower by 13% and 32%, respectively.

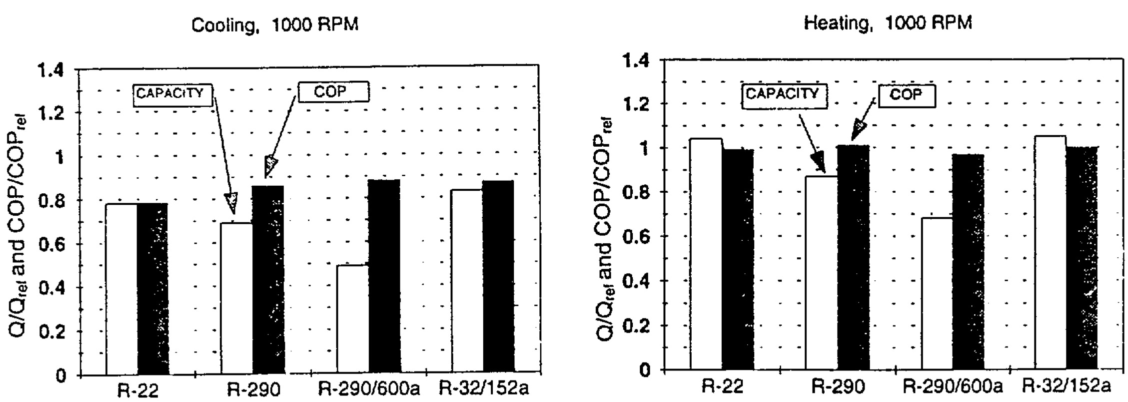

Fig. 8 - Capacity and COP at RPM=1000 for water-to-water heat pump referenced to capacity and COP for water-to-air system charged with R-22

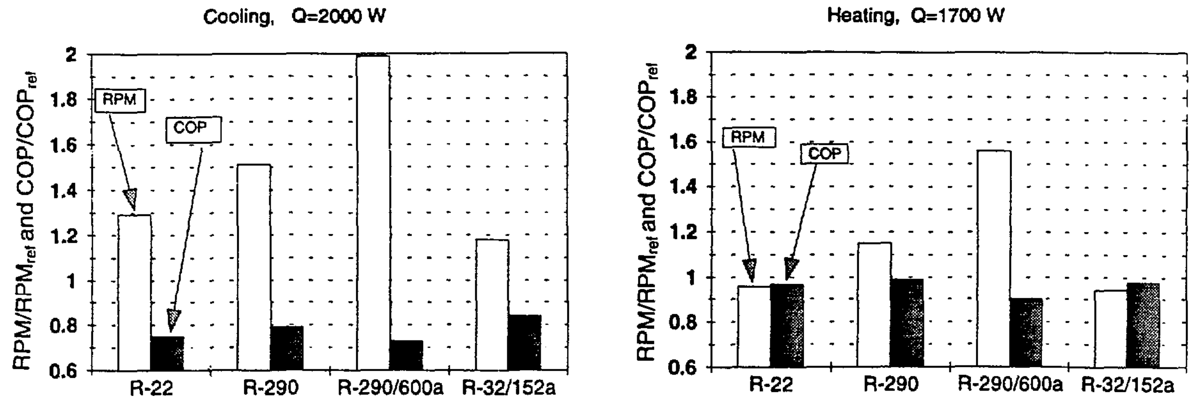

Fig. 9 - Capacity and COP at constant capacity for water-to-water heat pump referenced to capacity and COP for water-to-air system charged with R-22

The "drop-in" evaluation has a drawback because with this method different fluids assume different temperatures in the evaporator and condenser as a consequence of their different capacities. Different capacities will, in turn, require different mass flow rates of the HTF to maintain the specified HTF temperature at the outlet of the evaporator and condenser. This will affect the heat-transfer coefficient on the HTF side and the overall heat-transfer coefficient. Therefore a "drop-in" evaluation does not provide a fair comparison of the fluids' potentials since the merits due to physical properties are blurred by different refrigerant/HTF temperature differences and heat-transfer coefficient on the HTF side.

5.3 COP and Compressor Speed at Constant Capacity

When comparing the performance of different refrigerants at the same selected capacity, the heat flux and mass flow rate of HTF through the heat exchangers are the same for all refrigerants. Therefore, refrigerant temperature in the evaporator and condenser is determined by refrigerant properties alone. This "constant capacity" ("constant heat flux") comparison method is preferred to the "drop-in" test for evaluating the performance potential of different refrigerants /7/. Testing different fluids at the same system capacity requires using a different displacement compressor for each fluid. This was accomplished by using the same compressor running at different compressor speeds bracketing the selected capacity. Numerical values for the selected capacity were obtained using polynomial equations that correlated the test data.

Figure 9 shows compressor RPM and system capacity referenced to those for the reference water-to-air system charged with R-22. In the cooling mode for capacity of 2000 W, COPs of all fluids were lower than the reference COP (25% for R-22, 21% for R-290, 27% for R-290/600a, and 16% for R-32/152a). The water-to-water system required a higher-displacement compressor for all fluids. In the heating mode, the COPs were more uniform (1%-3% below the reference COP for R-22, R-290, and R-32/152a) except R-290/600a (10% below the reference COP). The required compressor displacement for R-22 and R-32/152a was similar to that of the reference system, but for R-290 and R-290/600a was higher by 15% and 56% , respectively.

Installation of a liquid-line/suction-line heat exchanger had a marked impact on the heating mode tests with R-290 and R-290/600a. The additional compressor displacement requirement decreased to 9% for R-290 and 49% for R-290/600a. The COP of R-290 exceeded the reference COP by 3%, and that of R-290/600a was just 3 percent below the reference value. This COP improvement for R-290 and R-290/600a can be explained by their large molar heat capacity /8/.

5.4 Discussion

It is of interest to compare the laboratory results with the simulation results from the preceding study /3/. Cross checking the simulation and laboratory results may indicate inadequacies of the model and some design aspects of the laboratory rig that could have dominated the outcome of the tests. Ideal agreement between the experimental and simulation results could not be expected because the simulations assumed a constant compressor efficiency for all fluids and did not include any design specifics of the SBHP. In spite of obvious differences between the simulation model and the test rig, it is appropriate to compare relative results (i.e., referenced to

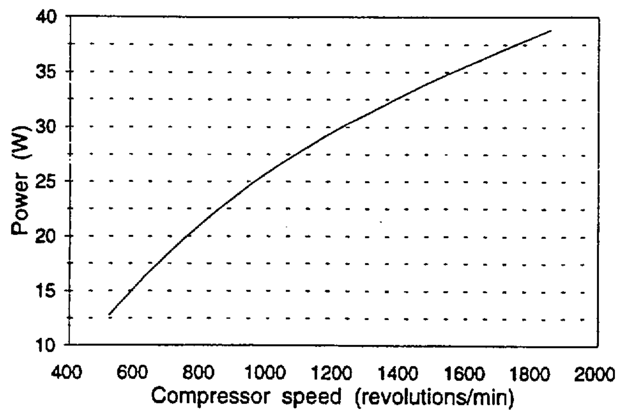

Fig. 10 - Compressor power during vacuum tests.

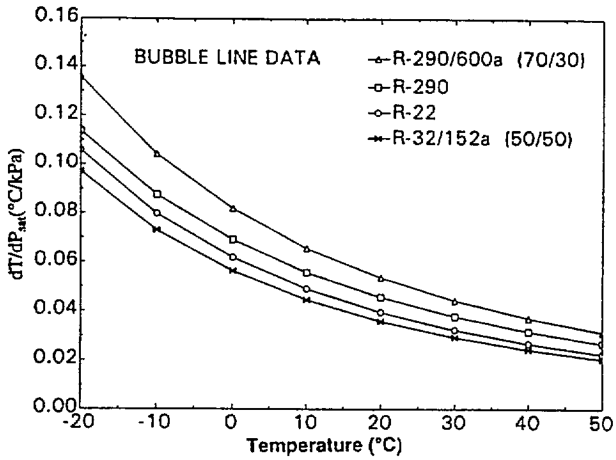

Fig. 11 - Change of saturation temperature with change of pressure at different saturation temperature levels.

the results for the water-to-air system) for both studies. Since for the test data at 2000 W capacity the evaporator ΔT was approximately 5 °C, these results were compared with the simulation results for simulations performed at ΔT equal 5 °C. A note must be made that the simulation study at ΔT equal 5 °C used R-32/152a (59/41). This (59/41) composition results in a higher capacity than the (50/50) composition. The impact on COP should be insignificant.

A reasonable agreement (within 5%) exists among the COP values obtained by the tests and simulations performed at ΔT equal 5 °C except for to R-290/600a, where the tested COP was 11% lower than the simulated COP. In the simulations, the COPs of R-32/152a and R-290/152a were the highest of all fluids and within 2% of each other, while the test-obtained COP of R-290/600a was 11% below that of R-32/152a.

One reason for the poor test performance of R-290/600a could be high friction losses of the compressor. During tests with R-290/600a the compressor had to be run at a high RPM to obtain the target capacity, and RPM-related losses in the compressor were higher with R-290/600a than with other fluids. We evaluated the RPM-related friction losses by measuring torque at a wide range of RPMs for the compressor evacuated to $2.5 \cdot 10^{-3}$ kPa abs. The original compressor was not available for the vacuum tests. Therefore, another compressor--the same model but with the ISO VG 100 oil instead the ISO VG 32 oil--had to be used. A third degree polynomial fit to the obtained compressor power (fig. 10) is representative of the friction losses. Correcting for these losses improved the COP of R-290/152a (in relation to that of the reference system) by 1.4%. This small change in COP suggests that R-290/600a suffered an additional penalty during laboratory tests from other sources. Other measurements showed that R-290/600a had a significant pressure drop, particularly in the evaporator. For 2000 W cooling capacity, compressor RPM was 1344 for R-290/600a and only 799 for R-32/152a, and the respective pressure drops in the evaporator were 33 kPa and 13 kPa. We may also speculate that R-290/600a realized an excessive pressure drop on the suction valves in the compressor. This excessive pressure drop for R-290/600a resulted in a higher compressor pressure head. The pressure drop in the evaporator also reduced the available glide of R-290/600a to approximately 2 °C providing a limited benefit of glide matching with the 6 °C glide heat-transfer fluid, while the remaining glide of R-32/152a was 6 °C. The larger loss of glide is also related to a higher dT/dP for the lower-pressure R-290/600a, as shown in fig. 11. Since the simulations were performed using the same compressor efficiency and the same mass flux in the heat exchangers for all refrigerants, the simulated pressure drop penalty for R-290/600a and R-32/152a was similar. It is reasonable to conclude that the performance of R-290/600a would be much better in a system with a compressor and heat exchangers optimized for this mixture.

Although more confidence is usually put in laboratory test data than in simulation results, our experience teaches us that experimental results comparing the performance of different refrigerants must also be evaluated with caution. It is probably impractical to develop a single laboratory apparatus in which performance potentials of different fluids can be fairly compared.

It appears that a simultaneous evaluation using a laboratory experiment and simulation model may provide a balanced solution.

6. CONCLUSIONS

The concept of the water-to-water heat pump was evaluated using R-22, R-290, R-290/600a (70/30), and R-32/152a (50/50). The study covered the cooling and heating mode of the vapor compression cycle (excluding fan and water pump powers). The combined standard uncertainty (68% confidence level) was 3.5% and 3.7% for capacity and COP, respectively. Comparison of results for the same system capacity showed R-32/152a to be the best refrigerant in cooling. During the tests with approximately 5 °C ΔT in the evaporator, the cooling COP of R-32/152a was 9% better than that of R-22 in the water-to-water system, but it was 16% lower than the COP of R-22 in the conventional water-to-air (reference) system. In the heating mode, the COP of R-32/152a was only 2% below that of the reference water-to-air unit. The heating COP of R-290 was measured to be just 1% below that of the reference COP. Installation of the llsl-hx markedly benefited the COPs of R-290/600a and R-290 in the heating mode. The COP of R-290 exceeded the reference COP by 3%, and that of R-290/600a was only 3% below the reference value.

R-290/600a had the lowest COP at the constant-capacity criterion in the basic (no llsl-hx) system; lower than R-32/152a by 11% in cooling and 6% in heating, while in the preceding simulation study the performance of both fluids was comparable. The analysis of test data showed that the poor performance of R-290/600a in the SBHP was caused by its excessive pressure drop that would be avoided in an optimized system. This exemplifies the difficulty in conducting an experiment that would fairly evaluate refrigerants of different volumetric capacities. This problem can be to some degree alleviated by conducting a parallel simulation study.

Comparison of the simulation predictions with the laboratory results points to the possibility of COP improvement for mixtures by increasing the size of the heat exchangers /3/. There are benefits of the water-to-water system that go beyond the evaluation covered here. For example, this studied concept allows the vapor-compression unit to be fully assembled, charged, and hermetically sealed in the factory, assuring a proper charge and reliable operation. Since COP is one of the deciding factors affecting the system's sales, large-capacity compact heat exchangers and zeotropic mixture hold the key to a better COP and the system's acceptability.

7. NOMENCLATURE

COP	$= Q/W$, Coefficient of Performance
GWP	Global Warming Potential
HTF	heat-transfer fluid
ITH	Integration Time Horizon
Q	capacity
P	pressure
RPM	revolutions per minute
SBHP	Small Breadboard Heat Pump
T	temperature
ΔT	average effective temperature difference
W	compressor power

Subscripts:

bub	bubble point
dew	dew point
ref	reference (water-to-air) system charged with R-22

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