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Experimental transient performance of a heat pump equipped with a distillation column

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Abstract

A series of experiments were conducted on a heat pump equipped with a distillation column. The system was operated with R32 and with a 30/70% by mass mixture of R32/134a to examine the difference between the transient performance trends with a pure fluid (R32), and those with a zeotropic mixture (R32/134a). Additionally, the effects of varying heat transfer fluid mass flow, compressor speed, and accumulator sump heat input were examined. Each test was 1 h in duration. The heat pump capacities did not generally achieve steady state during the R32/134a tests. Steady state was generally achieved during the R32 tests. As a percentage of the final (end-of-test) capacity, the rate of capacity increase was greater during the R32/134a tests than during those conducted with the pure fluid. The R32/134a tests exhibited capacity oscillations early in each transient that were not present during the R32 tests. The results show that circulating refrigerant mass and composition are the primary controlling factors with regard to transient capacity.

Keywords: Heat pump; Experiment; Refrigerant; R32; Binary mixture; R134a; Transient; Distillation; Performance

Performance transitoire expérimentale d'une pompe à chaleur munie d'une colonne de distillation

Mots clés : Pompe à chaleur ; Expérimentation ; Frigorigène ; R32 ; Mélange binaire ; R134a ; Régime transitoire ; Distillation ; Performance

1. Introduction

The steady-state measure of how well a heat pump utilizes electrical energy input to heat or cool an occupied space is the coefficient of performance (COP). In heating mode, this is typically calculated from

$$\mathrm{COP} = \frac{\dot{q}_{\mathrm{cd}}}{\dot{w}_{\mathrm{cp}}} \tag{1}$$

But COP is only a partial indication of the overall performance of a heat pump. Over the course of a season, heat pumps cycle on and off numerous times. Better insight to the impact of cycling on a customer's electrical consumption

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Nomenclature

COP	Coefficient of performance
c_{p_f}	Constant pressure specific heat of the heat transfer fluid [J kg^{-1} °C^{-1}]
HSPF	Heating seasonal performance factor [Btu W^{-1} h^{-1}]
HTF	Heat transfer fluid
$\dot{m}_f$	Heat transfer fluid mass flow rate [kg s^{-1}]
$\dot{m}_{f_{cd}}$	Heat transfer fluid mass flow rate through the condenser [kg s^{-1}]
$\dot{m}_{f_{ev}}$	Heat transfer fluid mass flow rate through the evaporator [kg s^{-1}]
$\dot{q}$	Rate of heat transfer (heat exchanger capacity) [W]
$\dot{q}_{cd}$	Rate of heat output from the condenser [W]
$\dot{q}_{sp}$	Rate of heat input to the sump [W]
rpm_{cp}	Compressor speed [rpm]
τ_{cp}	Compressor drive shaft torque [N m]
$T_{f_{in}}$	Heat transfer fluid inlet temperature [°C]
$T_{f_{out}}$	Heat transfer fluid outlet temperature [°C]
$\dot{w}_{cp}$	Compressor power [W]

is gained by evaluating the heating seasonal performance factor (HSPF).

$$\mathrm{HSPF} = \frac{\text{Seasonal building heat load (Btu h}^{-1}\text{)}}{\text{Seasonal power input (W)}} \qquad (2)$$

HSPF is calculated using a procedure established by the National Institute of Standards and Technology [1].

Generally, the HSPF calculation procedure requires experimental heat pump startup/shutdown testing to determine the expected cyclic losses in terms of a ‘‘cyclic degradation coefficient,’’ which is then included in the HSPF calculation. The HSPF calculation procedure also allows the use of a default degradation coefficient that represents an experience-based estimate of the aggregate effects of cycling. The value of the default degradation factor was established using R22, a single component refrigerant, and was designed to be conservative.

Mulroy and Didion [2] showed that cyclic losses are largely attributable to the lack of circulating charge during startup. In fixed area expansion device systems, the refrigerant migrates to the accumulator during the off-cycle, and the time required to reach maximum performance depends on how long it takes the refrigerant to re-establish its steady-state distribution throughout the system.

Because of the lack of single component direct R22 replacements, refrigerant mixtures have received attention as alternatives in heat pumps. Zeotropic refrigerant mixtures, in particular, present certain thermodynamic advantages over pure fluids. Mulroy et al. [3] and Domanski et al. [4] showed that the average temperature differences between the heat transfer fluid (HTF) and the refrigerant in counterflow condensers and evaporators can be reduced when using zeotropic mixtures. This leads to reduced irreversibilities and, therefore, improved COP. Additionally, Cooper and Borchardt [5] and Gromoll and Gutbier [6] showed that zeotropic mixtures could be used to modulate heat pump capacity by varying the volatility of the circulating refrigerant. Furthermore, Rothfleisch [7] demonstrated that capacity modulation could be passive. Rothfleisch conducted experiments using a zeotropic mixture in a system equipped with a fixed area expansion device in heating mode. As the outdoor temperature decreased, more and more refrigerant was stored in the accumulator. Due to vapor–liquid equilibrium, this caused the circulating composition to become richer in the more volatile component. As a result, the heating capacity decrease associated with decreasing outdoor temperature was attenuated. Rothfleisch also incorporated a heater and distillation column to enhance the composition shift further.

Because of the impact of circulating composition on steady-state performance, it is reasonable to expect that transient changes in circulating refrigerant composition also would affect transient performance. As is the case for circulating mass [2], this should have an impact on the HSPF cyclic degradation factor. However, no experimental work was found in the open literature comparing transient heat pump characteristics when charged with zeotropic mixtures with the characteristics when charged with a pure refrigerant. Also, no work examining heat pump transient behavior due to changing circulating composition was found.

The goal of the present investigation was to experimentally examine the transient characteristics of a heat pump equipped with a distillation column charged with a single component refrigerant and with a zeotropic mixture (R32 and R32/134a, respectively). The experiments were conducted using variations in four design parameters.

2. Experimental apparatus and procedures

The distillation heat pump consisted of a refrigerant loop and two water–ethylene glycol HTF loops (Fig. 1). A variable-speed reciprocating compressor pumped the refrigerant through the condenser and evaporator, which were counterflow heat exchangers composed of annular tubes. The heat exchangers were of the same design as those described by Kedzierski and Kim [8,9]. The expansion device was a needle valve. A distillation column, refrigerant storage accumulator (sump), and electrical heat source were

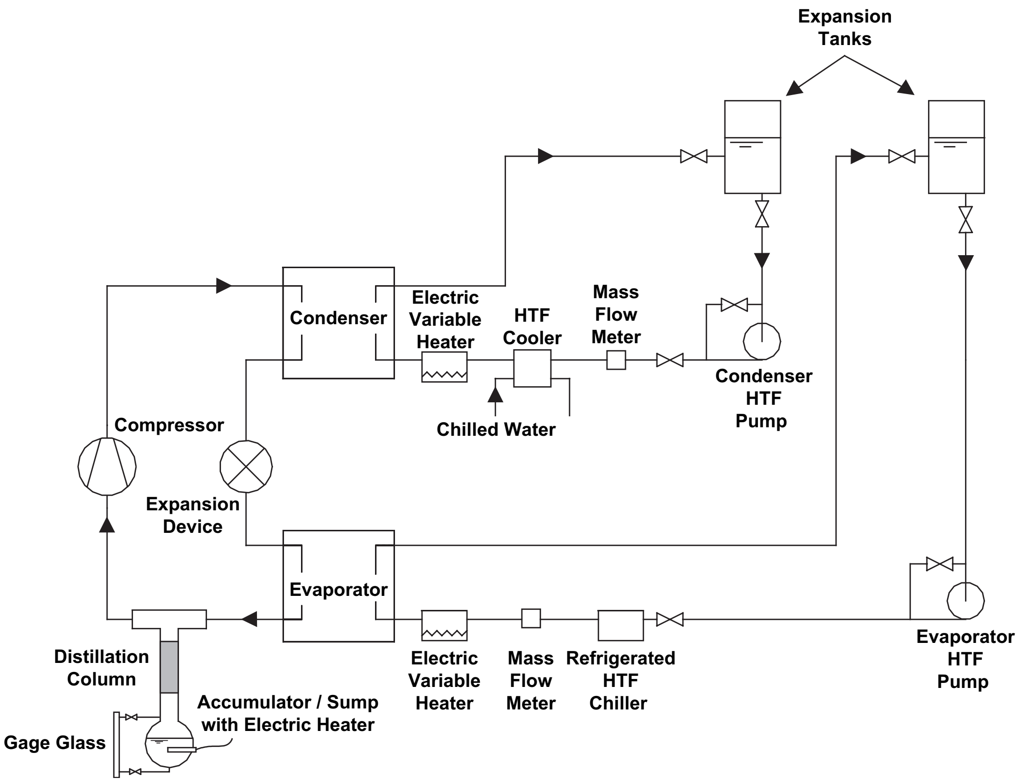

Fig. 1. Experimental distillation heat pump.

incorporated into the refrigerant loop in the manner described by Rothfleisch [7]; the distillation column design details may be found in Ref. [7]. The distillation accumulator sump level was monitored with a gage-glass. An electric cartridge heater was mounted in the bottom of the sump.

Ten sampling ports were located on the compressor discharge line. Samples were drawn from this location to ensure that the refrigerant was a single-phase vapor, as required for sample analysis by gas chromatography.

A turbine pump provided flow through each HTF loop and flow rates were measured using Coriolis effect flowmeters.

HTF inlet temperatures to the condenser and evaporator were controlled by overcooling, using chilled water in the condenser HTF loop and a chiller unit in the evaporator loop, then re-heating using variable electric immersion heaters.

Temperature measurements throughout the system were made using type-T thermocouples. Thermopiles, consisting of 10 type-T thermocouples immersed directly in the HTF, were used to measure sensible heat change across the heat exchangers.

A full factorial design experiment [10] with four factors and two levels per factor was used for both tested fluids (R32 and R32/134a). The four independent variables tested were: sump heater input power, evaporator HTF mass flow rate, condenser HTF mass flow rate, and compressor speed. The low/high sump heater power, condenser and evaporator HTF mass flows, and compressor speed settings were 25/100 W, 0.08/0.16 kg/s and 800/1000 rpm, respectively. The values of these settings were chosen as a result of preliminary system tests using R134a [11].

All tests were performed at a low evaporator HTF inlet temperature heating condition, a moderate room ambient condenser HTF inlet temperature, and with the same charged mass of R32. The nominal evaporator HTF inlet temperature, condenser HTF inlet temperature, and R32 charge were −7 °C, 20 °C, and 1500 g, respectively.

Prior to each transient test, the experimental system was placed in a repeatable initial condition as follows. The heat pump was operated at each factorial condition to be tested until the capacity and sump level were relatively steady. The refrigerant loop was then shutdown, but the HTF flow rates and inlet temperatures were kept constant through the condenser and evaporator. Then, the compressor was intermittently ‘‘jogged’’ to pump as much liquid refrigerant as possible to the sump. This procedure was judged [12] to give repeatable transient results. Each test was run for 1 h.

Some of the data were read directly from the instrumentation used (sump heater power, sump liquid level, and compressor speed), while others required supplemental calculations. Details can be found in the work of Gebbie [12].

A thermal conductivity type gas chromatograph (GC) was used to determine the mass fractions of the constituents in the refrigerant samples. During each test, visual sump gage-glass observations were made and refrigerant samples were taken approximately every 10 min.

With the torque and speed of the compressor shaft available from the transducer data, compressor power was calculated using

$$\dot{w}_{cp} = \frac{\tau_{cp} \cdot \mathrm{rpm}_{cp}}{60} \quad (3)$$

The heat exchanger capacity calculations were based on the sensible heat change and mass flow of the HTF.

$$\dot{q} = \dot{m}_f c_{p_f} \left(T_{f_{out}} - T_{f_{in}} \right) \quad (4)$$

The RMS uncertainty [13] and experimental repeatability of the data were examined. These evaluations were made based on the state of the system at the completion of each test. The uncertainties were: condenser and evaporator capacities, and compressor power, $\pm 1.5\%$; R32 mass fraction in an R32/134a mixture, $\pm 0.5\%$ by mass R32. The experimental repeatability in condenser capacity, compressor power, and COP were $\pm 1\%$, and the repeatability in evaporator capacity was $\pm 2.5\%$.

The uncertainty of the independent variables ($\dot{q}_{sp}$, $\dot{m}_{f_{cd}}, \dot{m}_{f_{ev}}$, and rpm_{cp}) were: sump heater power, $\pm 0.5\%$; HTF flow rates for both heat exchangers, $\pm 0.15\%$; HTF inlet temperatures, ± 1 °C. The compressor speed data were considered to be exact because the readings were derived directly from a rotation counter. The repeatability of the independent variables were: sump heater power, $\pm 9\%$ at 25 W and $\pm 5\%$ at 100 W; condenser HTF mass flow, $\pm 1.5\%$; evaporator HTF mass flow, $\pm 6\%$; compressor speed, $\pm 2.5\%$. Details of the RMS analysis and experimental uncertainty determination, as well as a more detailed description of the experimental apparatus and procedures, can be found in the work of Gebbie [12].

3. Results

Transient results for each fluid were compared by plotting the condenser capacity, evaporator capacity, sump level, and, in the case of the mixture tests, circulating R32 concentration as a function of time. Comparisons were made for each parametric variation.

An example of the pure R32 condenser capacity, evaporator capacity, and sump level transient behavior changes due to increased sump heater power with pure R32 is shown in Fig. 2. Increasing sump heater power decreased the time needed for the heat pump to reach steady state. The test performed at a sump heater power of 25 W required more than the 1 h test time to achieve steady condenser capacity and sump level. The evaporator capacity during that test was nearly steady after 30 min. The test that was performed at a sump heater power of 100 W required approximately 28 min, 35 min, and 20 min to achieve steady condenser capacity, sump level, and evaporator capacity, respectively. These results are consistent with those of Mulroy and Didion [2] showing that the overall system transient is controlled by the time necessary to introduce refrigerant from the sump into circulation. Furthermore, the results suggest that the majority of the refrigerant will be held-up in the condenser. Fig. 2 shows that, during the low sump heater power test, only the condenser capacity and sump level are changing after 30 min. Thus, the energy and mass transfer is exclusively between the sump and the condenser.

Comparison of the results from all of the pure R32 tests can be summarized as follows [12]:

1. The early ($< \sim 5$ min) transient capacity trends are similar to the step response of a first-order system. Some tests, most often those at low sump heater power, exhibited a linear increase in condenser capacity after the initial "first-order" exponential type response (Fig. 2). After a short period, the transient heat pump capacity was a function of circulating mass alone.
2. Increasing sump heater power decreased the time necessary to reach steady-state capacity and sump level.
3. Increasing the evaporator HTF flow rate increased the time necessary to reach steady state.
4. Increasing the condenser HTF flow rate increased the condenser capacity gradient at first, but the total time required to reach steady state was unaffected.
5. Increasing compressor speed did not affect the time necessary to reach steady state.

Example data from two of the mixture tests, as they varied with sump heater power, are compared in Figs. 3 and 4. Fig. 3 shows the condenser capacity, evaporator capacity, and circulating refrigerant composition from the two tests. Fig. 4 shows the transients sump levels.

The high concentrations of R32 at the beginning of each test (Fig. 3) were typical. This was due to the procedure used to set up for each test. As discussed above, each test was initiated after pumping as much refrigerant as possible to the sump. As the compressor was jogged to pump liquid refrigerant to the sump, the remaining vapor had a higher concentration of R32.

The rapid decrease in R32 concentration (Fig. 3) was a consequence of the refrigerant flashing in the sump, due to the rapid low-side pressure decrease at the start of each test. Visual observations of the flow-sight-glass downstream of the evaporator indicated that eventually there was liquid flow to the sump during the high sump heater power tests, but none during the low power tests. Flow of liquid to the sump, accompanied by vapor flow due to the sump heater, satisfied the conditions necessary for component separation in the distillation column. Therefore, after the initial rapid decrease, the circulating R32 concentration increased during the high sump heater power test, but continued to decrease (mildly) during the low sump heater power test.

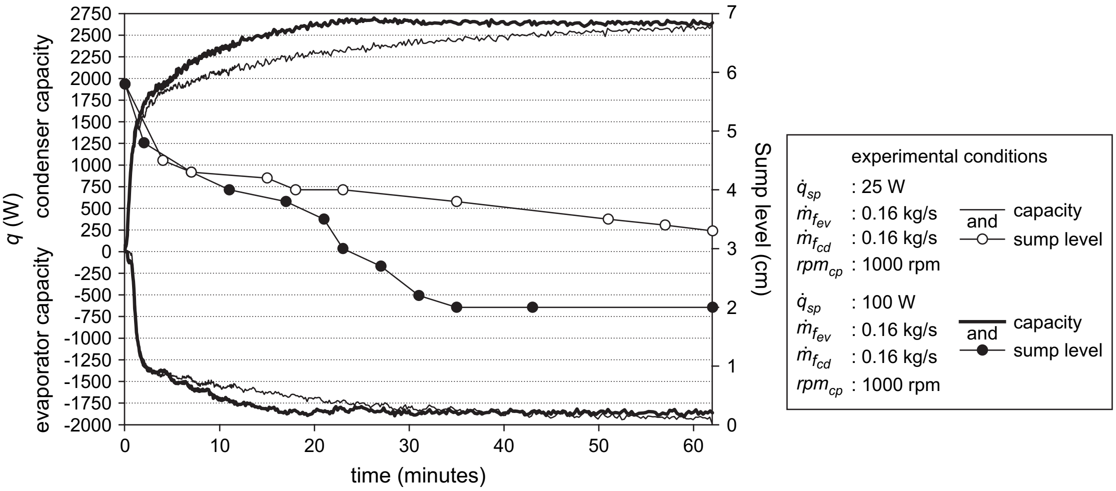

Fig. 2. Transient capacity and sump level: R32 example.

Fig. 4 shows that the sump level during the low sump heater power test was nearly steady after approximately 2 min. However, these data were based on visual observations of the sump gage-glass. It seems likely, in light of the decreasing R32 concentration (Fig. 3) that refrigerant from the sump was being added to circulation. Gage-glass observations made during the high sump heater power test showed the sump level to be steady after approximately 20 min.

Except for early transient oscillations, within the first 10 min, Fig. 3 shows that the evaporator and condenser capacities increased throughout the high and low sump heater power tests. The rate of capacity increase during the low sump heater power tests was small and probably due to the addition of refrigerant as suggested above. The capacity increase during the high sump heater power test was due to the increase in the circulating R32 concentration.

Comparison of the results from all of the zeotropic R32/134a mixture tests can be summarized as follows [12]:

1. The capacity data showed an oscillatory character early in the transients. This suggests that the distillation heat pump was a second-order system (or higher).
2. Steady-state was not generally achieved during the R32/134a tests. This was due to continuing changes in circulating mass and/or circulating composition that persisted until the end of each 1 h.
3. Due to test procedures, the R32 mass fraction circulating through the heat pump just after startup was

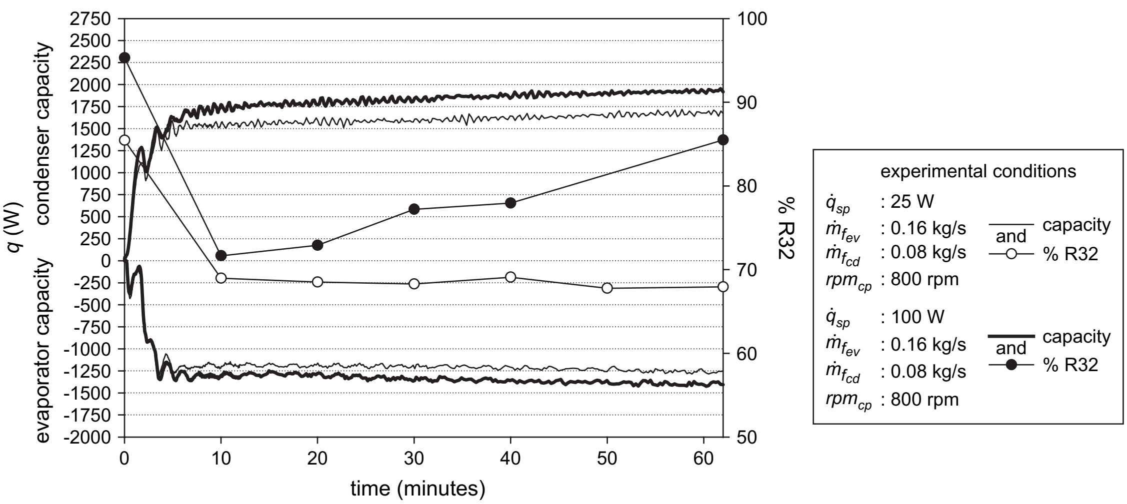

Fig. 3. Transient capacity and R32 mass fraction: R32/134a example.

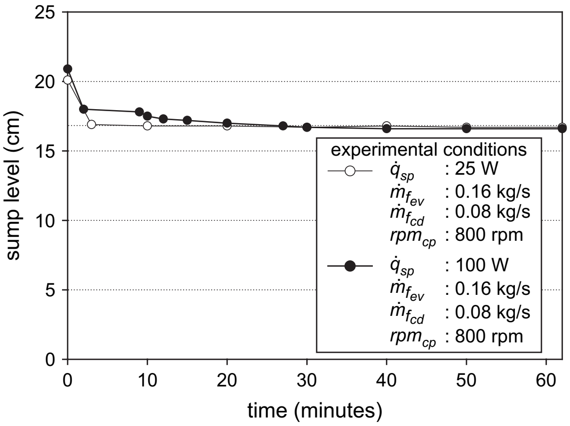

Fig. 4. Transient sump level: R32/134a example.

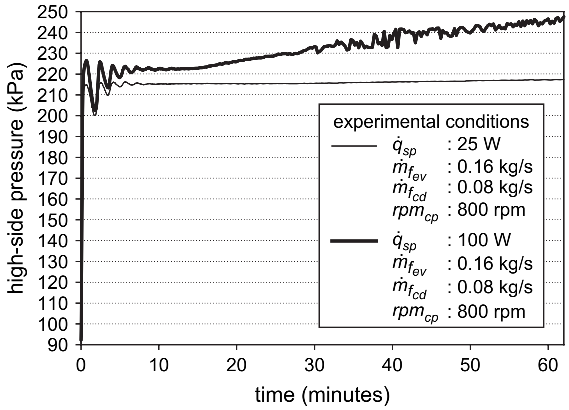

Fig. 5. Transient high-side pressures: R32/134a example.

significantly higher than expected. The R32 mass fraction decreased rapidly due to refrigerant flashing in the sump.

4. Increasing sump heater power
   a. Had little effect on the time required for the rate of capacity increase to become linear (10 min in the case of the tests shown in Fig. 3).
   b. Caused the evaporator outlet to become two phase during the test duration. The resultant liquid down-flow in the distillation column, combined with the vapor up-flow, caused an increase in circulating R32 composition.
5. Increasing evaporator HTF flow rate
   a. Caused a slight increase in the early transient capacity gradient in the evaporator
   b. Had no apparent effect on the condenser capacity transient.
   c. Had no consistent effect on the transient sump level or circulating composition data.
6. Increasing condenser HTF flow rate increased the amplitude but decreased the frequency of early transient condenser capacity oscillations.
7. Increasing compressor speed
   a. Had no substantial effect on the transient capacity data.
   b. Increased the rate of sump level decrease, and the time necessary for the sump to reach its steady level.
   c. Delayed the point in the transient where composition shifting began.
8. Overall, the mixture results indicate that, shortly after startup, transient heat exchanger capacities were a function of circulating mass and composition alone. This conclusion is based on the data that showed
   a. The time dependence of the capacity was limited to the time dependence of the composition in those cases when the circulating mass was steady.
   b. The time dependence of the capacity was limited to the time dependence of the circulating mass in those cases when the circulating composition was steady.

The clearest difference between the pure R32 tests and the R32/134a mixture tests is that there were capacity oscillations present during the R32/134a tests. The capacity rise during the pure fluid tests was nearly monotonic.

The reason for the capacity oscillations during the mixture tests is thought to have been due to the rapid addition of R134a at the beginning of the transients. The data show (Fig. 3) that the capacity oscillations were generally restricted to the period of rapidly decreasing R32 concentration. During this time, the pressure on the high-side of the refrigerant loop was rising rapidly (Fig. 5). Had the R32 concentration remained constant, the pressure would have continued increasing. However, the rapid addition of R134a with its lower volatility (low vapor pressure) tended to decrease the pressure. During the early transient, the increasing concentration of R134a was occasionally enough to dominate, and decrease, the high-side pressure (Fig. 5). Subsequently, the R134a condensed and the pressure began to rise again, which increased the R32 vapor concentration in the condenser. These pressure oscillations caused changes in the saturation temperature in the two-phase region of the condenser, and this caused the refrigerant-to-HTF temperature difference and, thus, the condenser capacity, to oscillate. High-side pressure/capacity oscillations were responsible for oscillations in the evaporator due to the coupling of the high and low sides.

The data also show that startup transients with refrigerant mixtures may generally be shorter than those with pure fluids. The low sump heater power test shown in Fig. 3 is nearly steady after 10 min. It would appear that, had it not been for the distillation column, the high sump heater power test would have been steady after approximately 15 min. The pure fluid tests took substantially longer to reach steady state (Fig. 2). The difference was because of the capacity ‘‘boost’’ given to the mixture tests by the high R32 concentration at the start (Fig. 3). Although the steady-state capacities of the mixture tests were less than those of the pure fluid, they seem to reach steady state more quickly. If the HSPF degradation was determined following the procedure by Parken et al. [1], then the impact of cycling might be substantially less for a mixture.

4. Conclusions

An experimental investigation was conducted into the effects of using either a zeotropic refrigerant mixture (30/70% by mass R32/134a) or a pure fluid (R32) on the transient performance of a heat pump. Due to its potential for enhancing capacity modulation when a heat pump is charged with a zeotropic mixture, a distillation column was installed in the manner prescribed by Rothfleisch [7].

Sixteen experiments were performed using pure R32, and sixteen were performed using an R32/134a mixture. Each set of sixteen experiments consisted of a 2^4 factorial design in which the factors were sump heater power, evaporator HTF mass flow rate, condenser HTF mass flow rate, and compressor speed.

In general, the pure R32 experiments achieved steady state during the 1 h allotted for each test; often, the R32/134a mixture tests did not. This was due to the time necessary for the distillation column to separate R32 from R134a during the mixture tests. The transient R32/134a mixture experiments showed distinct, repeatable, oscillations during startup. These oscillations, not present during the pure R32 experiments, were hypothesised as being due to the rapid addition of R134a to circulation early in the transients. The mixture tests received a capacity boost due to the relatively high R32 concentration at the start of each test. As a result, the mixture tests operated closer to the steady-state capacity for more of the test duration than the pure fluid tests. This means that heat pumps operating with mixtures may have significantly less cyclic degradation. Therefore, the default cyclic degradation coefficient of Parken et al. [1] may excessively penalize such systems.

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