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Effect of refrigerant charge on the performance of air conditioning systems

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SUMMARY

An air conditioning system operates in an optimal condition if the system is fully charged with a specified amount of refrigerant. Poor field maintenance or refrigerant leakage causes low level of charge resulting in a lower thermal performance and higher operating cost. An experimental investigation was conducted to study the effect of low charge level of R-22 on the performance of a 3-ton residential air conditioning system. The experimental results show that if a system is undercharged to 90 per cent then the effect is small: a 3.5 per cent reduction in cooling capacity and a 2 per cent increase in the coefficient of performance (COP). However, the system performance suffers serious degradation if the level of charge drops below 80 per cent. An ice layer formed on the outer surface of the cooling coil impedes the heat transfer between the warm air and cold refrigerant vapour. An economic analysis shows that the cost of properly charging an undercharged system which is at an 85 per cent charge level, can pay for itself in savings in a short period of 3–4 months. Copyright © 2001 John Wiley & Sons, Ltd.

KEY WORDS: refrigerant charge; air conditioning systems; performance of HVAC systems

1. INTRODUCTION

For each air conditioning system, a specified amount of refrigerant is required to charge the system to operate in an optimal condition at the design discharge and suction pressures. Field systems usually show some variations from the specified charge conditions. Small variations possibly due to the negligence of the field installers may not affect the system performance significantly. However, serious performance degradation can occur when the level of charge is below a certain limit. Periodic maintenance of the system can identify and correct the problem. However, in many cases the problem may persist for a long period of time because of poor or no maintenance, resulting in lower system performance and higher energy usage.

Farzad and O'Neal (1994) conducted experiments to compare the effects of R-22 charge between capillary tube and short-tube orifice expansion devices at a range of outdoor

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temperature. The results indicated that the maximum cooling capacity and energy efficiency ratio (EER) occurred near the full-charge condition. Exceedingly undercharging or overcharging the system caused decreases in both capacity and EER, and the effect on capacity for the low charge was stronger than the overcharged conditions. Compared with the capillary tube expansion device, the short-tube orifice device showed a relatively small variation with the refrigerant charge but a strong dependence on the outdoor temperature. Robinson and O'Neal (1994) studied the refrigerant charge effect on the performance of a system using R-32/R-134a blends. The measured capacity and COP with various levels of refrigerant charge were found to be similar to those reported in the previous study (Farzad and O'Neal, 1994). The effects of charge level were similar for both R-22 and the blends, about 15 per cent decrease in capacity for an 85 per cent charged system. In the theoretical studies, Damasceno *et al.* (1991) developed a theoretical model to predict the performance of a heat pump in both the cooling and heating modes at various levels of charge. It was found that precise measurements of internal volume of the system and slight modification of void fraction model yielded accurate prediction of refrigerant mass distribution within the various system components. In turn, the predicted heat pump capacity agreed well with the measured values for a system under different levels of charge.

Although R-22 is being phased out and gradually replaced with HFC refrigerants, existing air conditioning systems largely operate with R-22. The purpose of this research was to determine the performance penalty of a 3-ton residential air conditioning system associated with various low R-22 charge levels. The test environments were maintained at constant temperatures and humidities to meet the ASHRAE and ARI standards. Thermocouples, pressure transducers, humidity sensors, and power transducers were properly installed to monitor the performance of the system. The cooling capacity and COP were determined by both air enthalpy and refrigerant enthalpy methods. Based on the experimental results, an economic analysis was performed to evaluate the savings if an air conditioning system is properly maintained at full-charge level.

2. EXPERIMENTAL FACILITY AND PROCEDURES

The air conditioning system test facility illustrated in Figure 1 was set up at the Solar Energy and Energy Conversion Laboratory at the University of Florida (Goswami *et al.*, 1993). The facility serves as a non-biased industry-independent laboratory capable of testing HVAC systems, of up to 7 tons capacity, and other related equipment and products. It was configured to accommodate the guidelines specified in ASHRAE Standard 37-1988 (ANSI/ASHRAE, 1998) and ARI Standard 210 (ANSI/ASHRAE, 1989) for testing unitary equipment.

The test air conditioning system consisted of a 3 ton high-efficiency split-system heat pump condensing unit and a fan/coil air handling unit. The condensing unit, equipped with a single-speed hermetically-sealed reciprocating compressor, was installed in Room A (see Figure 1), which simulated the outdoor air conditions maintained at 95°F dry-bulb and 75°F wet-bulb temperatures. The air handler, equipped with an orifice-type expansion device, was installed in Room B, which simulated the indoor air conditions maintained at 80°F dry-bulb and 67°F wet-bulb temperatures. A 7 ton refrigeration system, electric heaters (30 kW total), and spray nozzle humidifiers were used to maintain the above test conditions. Thermocouples and humidity sensors were installed at the locations indicated in Figure 1 to measure the air properties at the inlet and outlet of the evaporator. In addition, thermocouples and pressure transducers were

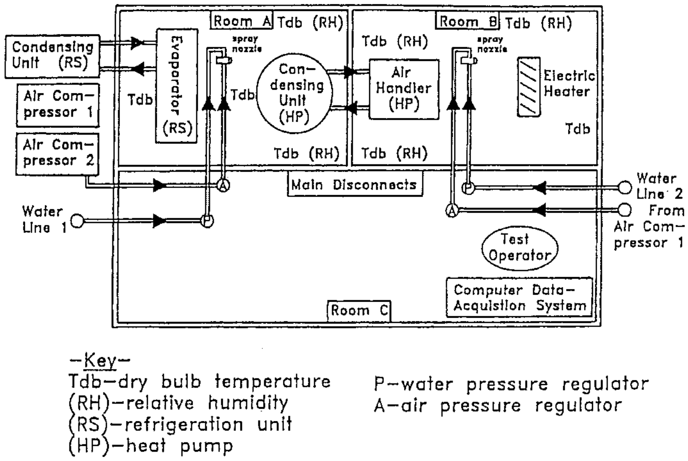

Figure 1. Schematic diagram of the air conditioning test facility.

used to measure the thermodynamic states of refrigerant entering and exiting the evaporator. Room C served as a control room facilitated with a control panel and a data acquisition system.

The test system was initially fully charged with 8.5 lb of R-22. The system was run at the simulated conditions described above for a 3 h period under a quasi-steady operation. During the test, the room temperatures and humidities were carefully maintained within $\pm 2°$F and ± 2 per cent, respectively. The power consumption, air temperatures and humidities, and refrigerant temperatures and pressures were recorded every 2 min. The results obtained were used to establish the base-line performance of a fully charged system. Then, 0.85 lb of the refrigerant charge was removed and the system of 90 per cent charge was tested again. Similarly, the above procedure was repeated for the refrigerant charge levels of 80, 70, and 50 per cent.

3. PERFORMANCE ANALYSIS

The air and refrigerant enthalpy methods, specified in ASHRAE Standard 37-1988 (ASHRAE, 1993), were employed to evaluate the performance of the system. For the *air enthalpy method*, cooling capacity is determined from knowledge of the evaporator inlet and exit air enthalpies, and the corresponding air flow rate over the cooling coil. Air enthalpies are determined from air dry-bulb temperature and humidity data collected via the data acquisition system.

Using the principle that the total enthalpy of a mixture of perfect gases equals the sum of the individual component enthalpies, the moist air enthalpy can be found as

$$h_a = 0.240T_a + \omega(1061 + 0.444T_a)\,\text{Btu lb}_m^{-1} \tag{1}$$

where T_a is the dry bulb temperature and ω is the humidity ratio which is calculated from the following equation:

$$\omega = \frac{0.622 p_v}{p_0 - p_v} \tag{2}$$

where p_0 and p_v are the atmospheric and vapour pressures, respectively.

The first term on the right-hand side of Equation (1) is the dry air enthalpy, and the other term is the specific enthalpy for saturated water vapour at the temperature T_a of the mixture. Accordingly, the expression for the total cooling capacity based upon air enthalpies is obtained by writing the energy equation for a control volume enclosing the evaporator. Assuming that no energy storage or generation occurs inside the control volume, and changes in air potential and kinetic energy are negligible, the steady-state energy equation becomes

$$\dot{q}_a = \dot{m}_a (h_{a,i} - h_{a,o}) \tag{3}$$

where $\dot{q}_a$ is the total heat pump cooling capacity based upon evaporator inlet and outlet air enthalpies determined from Equation (1). The term $\dot{m}_a$ in Equation (3) is the air mass flow rate across the cooling coil, and is found by dividing the measured volumetric air flow rate $\dot{Q}_a$ by the calculated air specific volume v_a from Equation (4):

$$v_a = \frac{R T_a}{p_0 - p_v} \tag{4}$$

where R is the gas constant for air, T_a is the dry-bulb absolute temperature, p_0 is the local atmospheric pressure, and p_v is the partial pressure of water vapour. The water vapour partial pressure p_v is found from the relative humidity Φ and the water vapour saturation pressure p_g evaluated at the dry bulb temperature as

$$p_v = \phi p_g \tag{5}$$

Coefficient of performance, defined as the refrigeration (cooling) capacity divided by compressor power input, is therefore,

$$\mathrm{COP_a} = \frac{\dot{q}_a}{P_c} \tag{6}$$

where P_c is the measured compressor power.

For the *refrigerant enthalpy method*, the total system cooling capacity is found using calculated evaporator inlet and exit refrigerant enthalpies, and the measured liquid refrigerant flow rate. Refrigerant enthalpies are determined directly using the computer program REFPROP (REFPROP, 1998).

Inputs for REFPROP are sets of refrigerant pressure and temperature data. As discussed previously, each set of data corresponds to measurements taken at particular locations in the vapor compression cycle: the compressor inlet and outlet, the throttling-valve inlet, and the evaporator outlet. Program output includes refrigerant thermodynamic properties evaluated for each set of pressure and temperature data (four total).

The expression for total cooling capacity based upon refrigerant enthalpies is obtained by writing the energy equation for a control volume enclosing the evaporator. Assuming that no energy storage or generation occurs inside the control volume, and changes in refrigerant potential and kinetic energy are negligible, the steady-state energy equation becomes

$$\dot{q}_{\mathrm{r}} = \dot{m}_{\mathrm{r}}(h_{\mathrm{r,o}} - h_{\mathrm{r,i}}) \tag{7}$$

where $\dot{q}_{\mathrm{r}}$ is the total cooling capacity based upon the evaporator-outlet and valve-inlet refrigerant enthalpies. The term m_{r} in Equation (7) is the liquid refrigerant mass flow rate, and is found by dividing the measured liquid refrigerant volumetric flowrate $\dot{Q}_{\mathrm{r}}$ by the refrigerant specific volume v_{r}, where v_{r} is calculated by REFPROP. Note that Equation (7) assumes that the refrigerant expansion across the throttling valve is isenthalpic. Accordingly, the heat pump coefficient of performance based upon refrigerant enthalpies becomes simply

$$\mathrm{COP} = \frac{\dot{q}_{\mathrm{r}}}{P_{\mathrm{c}}} \tag{8}$$

4. RESULTS AND DISCUSSION

The operating conditions measured in terms of the refrigerant temperatures, air temperatures and humidities, and compressor power are shown in Figures 2–6, respectively. As expected in Figure 6, at lower charge levels the compressor requires lesser power input. Based on these measurements, the cooling capacity is determined. Figure 7 clearly reveals that both the air

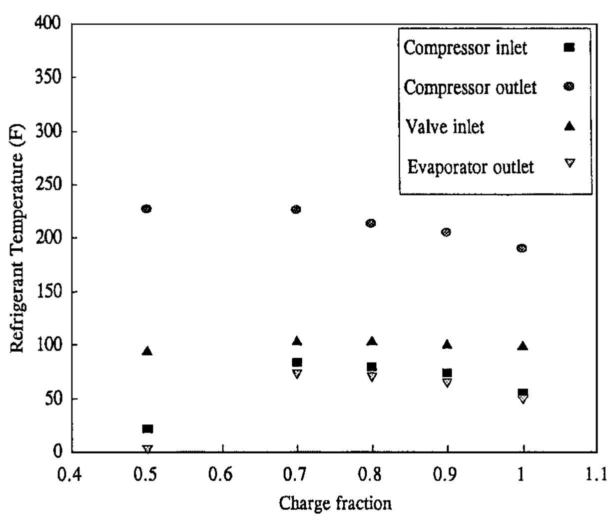

Figure 2. Refrigerant temperature versus charge fraction.

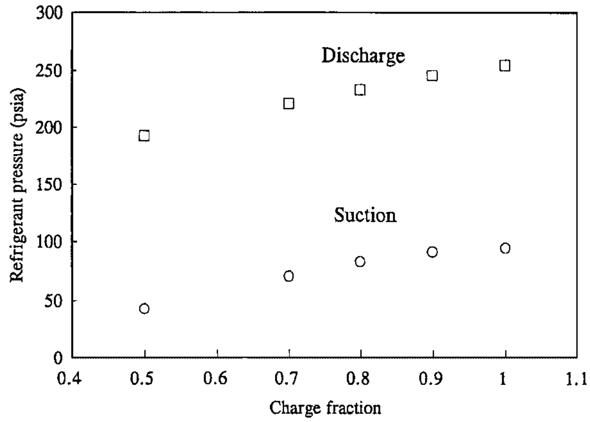

Figure 3. Discharge and suction pressures versus charge fraction.

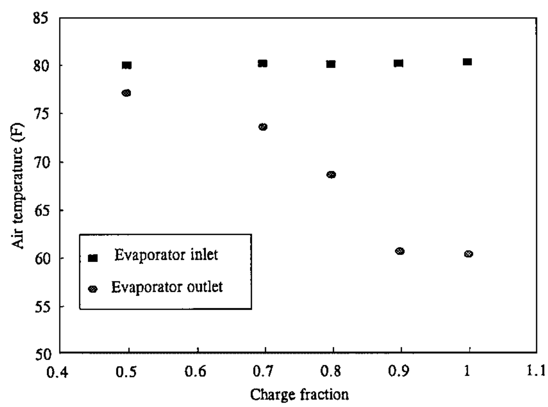

Figure 4. Air temperature at the evaporator versus charge fraction.

enthalpy and refrigerant enthalpy methods yield similar results at charge levels down to 90 per cent. For further reduction in charge level to 80 per cent and below, while the refrigerant enthalpy method shows a gradual decrease in the cooling capacity, the air enthalpy method shows a drastic decrease to zero and even negative values. The corresponding COP is plotted in Figure 8.

A zero cooling capacity means that there is no net cooling of the air flowing over the evaporator coil. Even though the temperature of the air decreases as it flows over the coil, its absolute humidity goes up because of the moisture that is absorbed by air and the coil surface. Since the total enthalpy change is the sum of changes due to both temperature and humidity, the overall effect may be zero or even a negative change in the air enthalpy. This happens because at low charge levels ice forms on the coil due to the very low-temperature refrigerant coming out of the expansion valve. As time progresses the layer of ice continues to increase. The ice acts as a thermal insulator between the refrigerant and the air, and also blocks the air flow over the coil. It must be

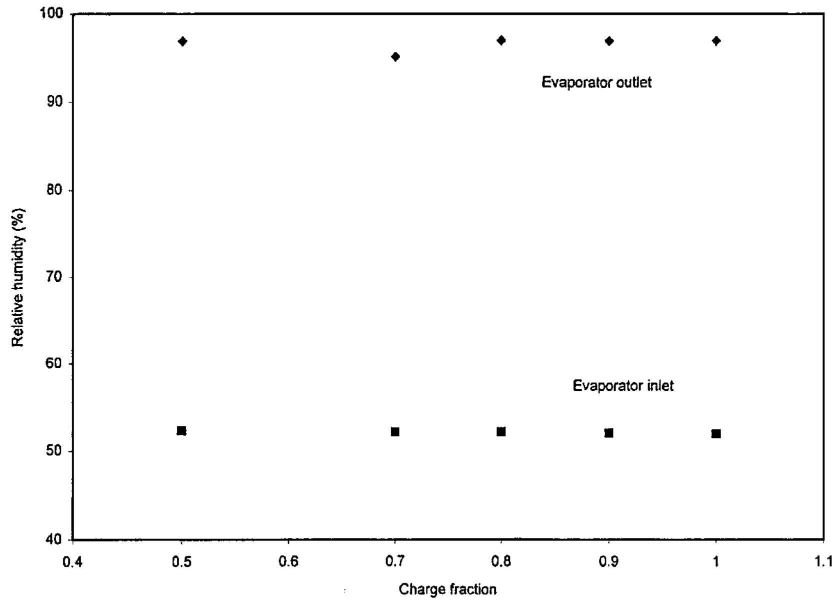

Figure 5. Relative humidity of air at the evaporator versus charge fraction.

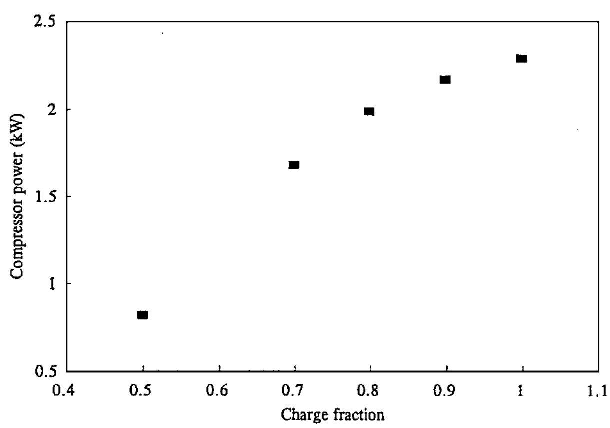

Figure 6. Compressor power versus charge fraction.

pointed out that even at low charge levels as the system is first turned on it does cool the air until a layer of ice forms on the coil. The results presented in this paper show only the system performance at a quasi-steady state.

In general, it is more accurate to estimate the performance based on the refrigerant enthalpy method because the errors in measuring liquid flow are lower than that in air flow. However, under the low charge conditions (below 80 per cent), measurements on the refrigerant side may

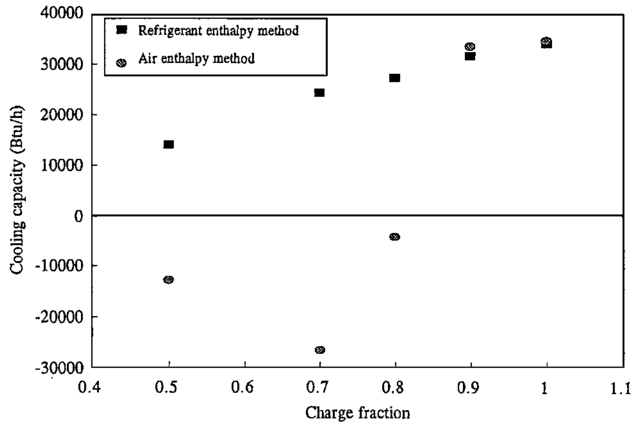

Figure 7. Cooling capacity versus charge fraction.

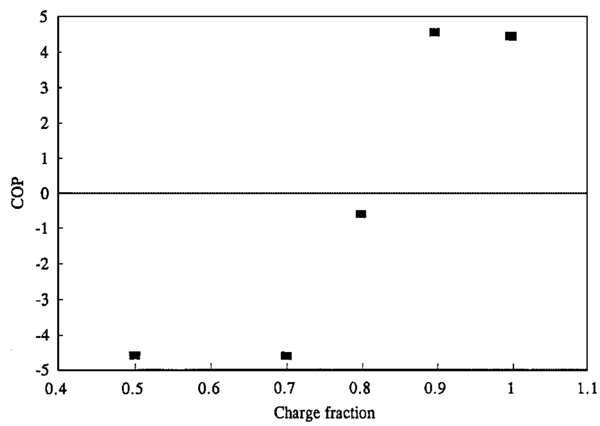

Figure 8. Coefficient of performance versus charge fraction.

show cooling capacity being obtained, but in reality the cooling is used up in just increasing the thickness of ice on the coil, instead of cooling the air. It is clear that the estimation of the system performance under these conditions by the refrigerant enthalpy method would be misleading. Therefore, at low charge levels the system performance should be estimated by the air enthalpy method.

5. ECONOMIC ANALYSIS

An economic analysis was performed on an 85 per cent undercharged system. From the experimental results carried out under the standard condition (95°F dry bulb, 75°F wet bulb) it was found that the EER (or COP) for such a system decreases by about 56 per cent. Thus, at other ambient conditions the same decrease in EER has been assumed.

Calculations were based on a 3-ton residential air conditioning unit (Carrier model 50QJ004). The EER data for the cooling performance of such a system were obtained from the manufacturer. It was assumed that the 3 ton unit is sized to provide the complete design load of 36,000 Btu h^{-1} with a temperature difference of 20°F (95°F dry bulb outside, 75°F dry bulb inside).

Thus, the total annual energy consumption of a fully charged and an undercharged system could be estimated by the Bin Method as

$$Q = \sum_i \frac{\dot{q}_{\mathrm{load}} \times (T_{\mathrm{bin,i}} - T_{\mathrm{bal}}) \times N_{\mathrm{bin,i}}}{\mathrm{EER} \times \Delta \mathrm{T} \times 1000} \tag{9}$$

where EER is the energy efficiency ratio of the system (Btu $W^{-1} h^{-1}$), $N_{\mathrm{bin,i}}$ the number of hours of average temperature $T_{\mathrm{bin,i}}$ (h), $\dot{q}_{\mathrm{load}}$ the design cooling load (36,000 Btu h^{-1}), Q the annual energy consumed by the system(kW h), T_{bal} the balance point temperature (assumed as 72°F), $T_{\mathrm{bin,i}}$ the average bin outdoor dry bulb temperature (°F), and ΔT the design temperature difference (20°F).

The annual energy savings are

$$Q_{\mathrm{SAV}} = Q_{\mathrm{UC}} - Q_{\mathrm{FC}} \tag{10}$$

where Q_{SAV} is the annual energy savings (kW h), Q_{UC} the annual energy consumption of an undercharged system (kW h), and Q_{FC} the annual energy consumption of a fully charged system (kW h).

The cost savings evaluated at an electricity rate of \$0.07643 $kW^{-1} h^{-1}$ (FPL, 1996). The maintenance costs included 2 h of labour at \$60 h^{-1} and one pound of refrigerant charge at \$8 lb^{-1}. Based on the information obtained from the Engineering Weather Data (1978), the calculations have been applied to a number of major U.S. cities in which the cooling demand is high. The results are summarized in Table I.

Table I. Summary of savings and payback period for different cities.

City	Energy savings, Q_{SAV} (kW h)	Payback period (months)
Tucson, AZ	7615	2.6
Sacramento, CA	3704	5.4
Jacksonville, FL	4945	4.0
Miami, FL	7006	2.9
New Orleans, LA	4771	4.2
Oklahoma City, OK	3993	5.1
Houston, TX	5490	3.7
San Antonio, TX	6511	3.1

6. CONCLUSIONS

Based on this experimental study, it can be concluded that the level of charge greatly affects the performance of air conditioning systems. For charge levels down to 90 per cent the effect is negligible. However, for charge levels of 80 per cent or below the steady-state cooling capacity and the COP may be zero or even negative. For a brief transitional period, a system with a lower charge level will provide cooling when it is first turned on. After that an ice layer forms on the evaporator coil resulting in drastic reductions in the cooling capacity and COP. The refrigeration used up in forming the ice layer represents wasted energy. The consumer may not realize that the system is not working until the charge level goes below 80 per cent, especially where the system cycles on and off frequently. Routine maintenance may avoid the problem and prevent the wastage of energy. An economic analysis shows that the cost of charging an 85 per cent undercharged system can pay for itself in savings over a period of 3–4 months.

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