Experimental Study of a Liquid Overfeeding Window Air Conditioner

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ABSTRACT

An off-the-shelf, two-ton window air conditioner with an energy efficiency ratio (EER) of 10 was modified for liquid overfeeding (LOF) operation by adding a recuperative accumulator-heat exchanger (AHX) and an additional 15% of R-22 charge (Mei and Chen 1993). No original component was replaced. The unit was tested in an environmental chamber with outdoor chamber temperatures varying from 82°F to 110°F and the indoor chamber temperature at 80°F. The relative humidity was 52%. The unit was first tested without the LOF feature (baseline test) and then tested with LOF. Both air-side and refrigerant-side measurements were collected. It was found that air-side and refrigerant-side cooling capacities, for most tests, were within 5% of each other.

The test results showed that at 82°F ambient, the LOF cooling capacity and coeffecient of performance (COP), compared with baseline test data, improved by about 14% and 10%, respectively. At 95°F ambient, the LOF's cooling capacity and COP are about 12% and 7.5% higher than that of the baseline test. For ambient temperature below 100°F, LOF operation will have both higher sensible and latent cooling capacities. The only additional cost is for the AHX and for a rerouting of liquid and suction lines. For units that have accumulators in the system design, the added cost may be minimal.

INTRODUCTION

Liquid overfeeding (LOF) systems have been used on ammonia refrigeration systems successfully for many years (ASHRAE 1990; Richards 1970). The LOF refrigeration system design floods the evaporator coils for easy operation, higher refrigeration capacity, and less maintenance. The LOF system is, however, too complicated for small window air conditioners and residential heat pumps. In this study, an accumulator-heat exchanger (AHX) was added and additional refrigerant was charged to a window air conditioner to achieve an LOF effect. Figure 1 shows the schematic of the system. When the system is charged with additional refrigerant, it starts accumulating in the AHX. Warm liquid from the condenser exit flows through the heat exchanger coil in the AHX and boils the low-pressure liquid in the AHX. The warm (high-pressure) liquid acquires additional subcooling, and the compressor suction inlet will have saturated, or near-saturated vapor, and thus a higher mass flow

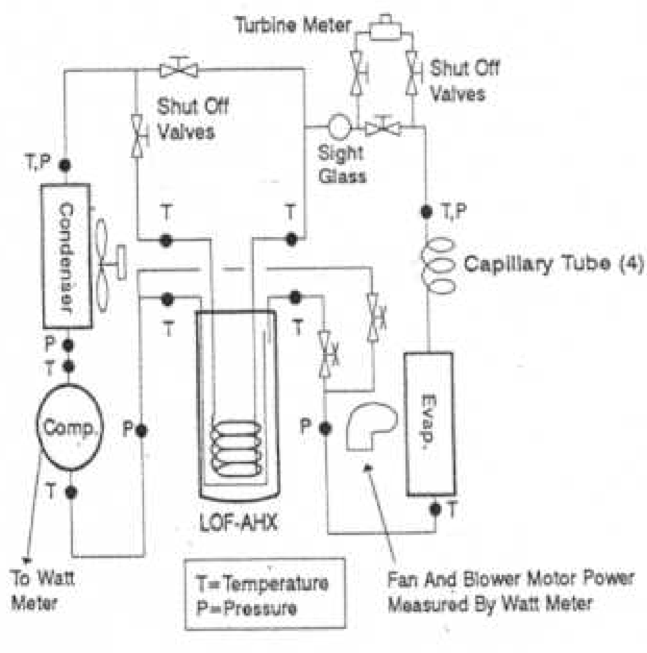

Figure 1 Schematic of air-side test setup.

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rate. The high-pressure refrigerant flows through the expansion device to be evaporated. Because the liquid is highly subcooled and the refrigerant mass flow rate is higher, refrigerant is not fully evaporated in the evaporator coils. Two-phase fluid flows to the AHX, and the extra low-pressure liquid is boiled off by the warm liquid from the condenser.

The LOF cycle accomplishes several purposes: (1) it provides additional liquid subcooling; (2) the evaporator is all wet, or 100% utilized; (3) it improves compressor efficiency because saturated, or near-saturated, vapor is supplied to the compressor suction, thus increasing refrigerant mass flow rate; and (4) it lowers compressor discharge temperature, which has a positive impact on the compressor (this is particularly important for units operating at high ambient temperatures). High discharge temperatures could carbonize oil, which could result in loss of lubrication for reciprocating compressors.

A two-ton, EER 10, off-the-shelf window air conditioner was modified and tested with and without (baseline) the LOF feature. Both air-side and refrigerant-side data were collected. The test results show that at 82°F ambient, the cooling capacity and COP of LOF operation are about 14% and 10% higher than those of the baseline test. The improvement of cooling capacity and COP drops to 12% and 7.5% at 95°F ambient. The LOF has higher power consumption, lower compressor discharge temperature, higher refrigerant mass flow rate, and higher discharge pressure.

TEST SETUP AND PROCEDURE

A two-ton window air conditioner was modified by adding an AHX and then tested in the environmental chamber. No original component was removed or modified. Figure 1 shows the refrigerant-side schematic. For refrigerant-side measurement, a turbine meter, located just before the capillary tubes, was used for refrigerant flow rate measurement. Pressure transducers and thermocouples were used for pressure and temperature measurements. A watt-meter was connected to the power source for power consumption measurement. The air conditioner specified a 52-oz. R-22 charge. However, because of the additional piping involved for the LOF and instrumentation, 67 oz. was charged for baseline test. An additional 11 oz. was charged, for a total of 78 oz., for the LOF operation.

Figure 2 shows the air-side schematic. Two thermocouple piles were used for evaporator inlet and outlet air dry-bulb temperature measurements. Two hygrometers were used for evaporator inlet and outlet air dew-point measurements. A ductwork airflow rate measurement device was used for air volumetric flow rate measurement. A data-acquisition system, coupled with a personal computer, was used for data collection and reduction. All the tests were performed at steady-state operating conditions.

For the baseline test, the AHX was valved out of the refrigerant flow circuit, the indoor chamber was set at 80°F dry bulb and 52% relative humidity, and the outdoor chamber kept constant at 82°F. The air conditioner was run until steady-state operation was reached, and then data were collected over a period of 5 to 10 minutes. The outdoor chamber temperature was then raised to 85°F, and the test was repeated at 5°F increments until the outdoor chamber temperature reached 110°F. For LOF tests, the above process was repeated with the AHX in the circuit.

TEST RESULTS AND DISCUSSION

Figure 3 shows the cooling capacities for both baseline and LOF operation. Air-side and refrigerant-side measurements are in good agreement. At 82°F ambient, the cooling capacity of the LOF system is about 14% higher than that of the baseline test. However, when the ambient temperature increases, the improvement due to LOF operation is reduced. At 95°F, the capacity

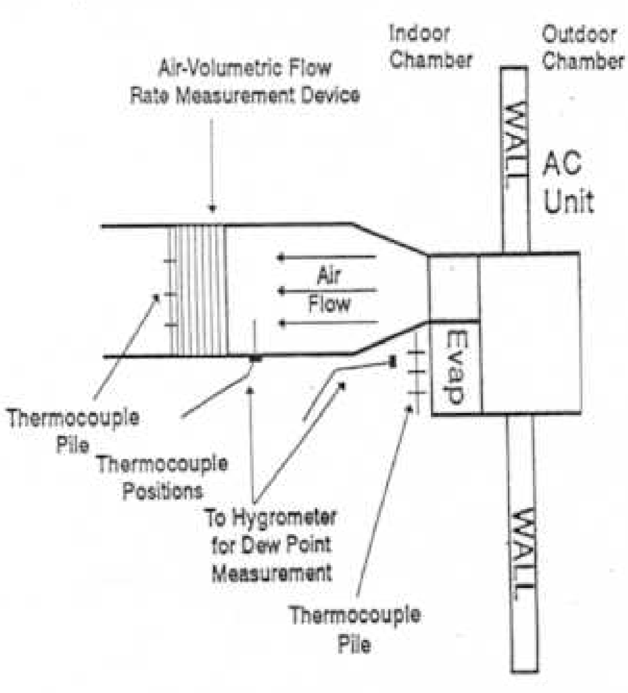

Figure 2 Cooling capacity: baseline vs. LOF.

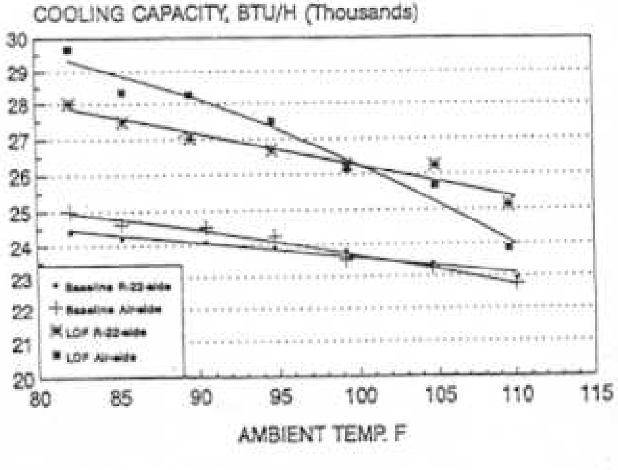

Figure 3 Mass flow rate: baseline vs. LOF.

improvement drops to 12%. This is expected because, as the ambient temperature increases, the compressor discharge pressure increases, and the refrigerant mass flow rate increases which reduces the evaporator coil dry-out section. Thus, the LOF system will have less potential for improvement. Figure 4 shows the comparison of refrigerant mass flow rate. LOF's mass flow rate is more than 10% higher than that of the baseline test. Baseline operation has a slightly higher mass flow rate increase at higher ambient temperature than LOF. Figure 5 shows the comparison of power consumption. The LOF does consume more power. However, it is not proportional to the mass flow rate increase. One reason is that for LOF operation the compressor discharge temperature is lower, as shown in Figure 6. Lower discharge temperature will have a positive effect on compressor power consumption. Figure 7 shows the comparison of system COPs. Even though LOF consumes more power, it also provides more cooling capacity. The net effect is that LOF has higher system COPs until ambient temperature is about 110°F. At 82°F ambient, the COP improvement of LOF over that of baseline operation is about 10%. At 95°F, the COP improvement is around 7.5%. Figure 8 shows the compressor high-low pressure ratio, which is a good indication of compressor efficiency. The LOF's high-low pressure ratio is about 7.7% lower than that of the baseline test, which means compressor operation with LOF is more efficient. Figure 9 shows the comparison of evaporator air outlet dry-bulb and dew-point temperatures. LOF has lower dew point up to 100°F ambient. With constant evaporator inlet air conditions, lower outlet air dew point means drier outlet air. LOF also has lower evaporator outlet air dry-bulb temperature across the tested ambient temperatures. With dryer and cooler air from the LOF system, the comfort level is improved. Figure 10 shows the comparison of refrigerant subcooling before the capillary tubes. The LOF has a higher level of liquid subcooling across the test range of ambient temperature. At 82°F and 95°F ambient, LOF's subcooling is 8°F and 4.7°F higher than that of the baseline data. The difference becomes smaller when the ambient temperature gets higher.

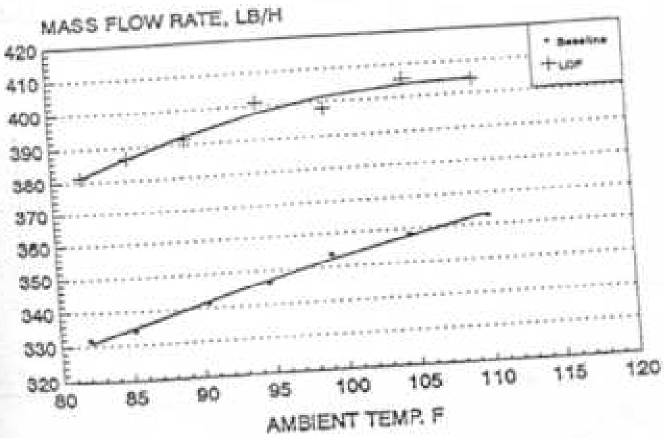

Figure 4 Power consumption: baseline vs. LOF.

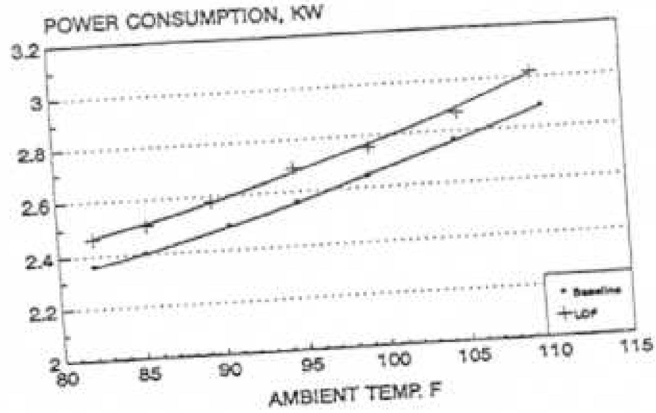

Figure 5 Compressor discharge temperature: baseline vs. LOF.

For LOF operation, an optimum refrigerant charge seems to exist. The preliminary test results indicate that when an additional 0.22 pound of R-22 was charged into the system, the air-side cooling capacities remained close to those of optimum charge, but became lower when the ambient temperature exceeded 100°F. However, the discharge temperature for overcharged LOF operation was about 4°F to 8°F lower than that of optimum charged LOF. The suction pressure for the overcharged LOF was about 1 psi to 2 psi higher. The power consumption for the overcharged LOF was close to that of the optimum charged system, except that at 100°F and higher ambient conditions, the power consumption for the overcharged LOF operation became lower. When the system was further charged with additional R-22, the air-side cooling capacities deteriorated, suction pressure increased, and the power consumption decreased at high ambient conditions. The net effect of overcharging an LOF system is that it will have a lower system cooling capacity and COP.

CONCLUSIONS

An off-the-shelf, two-ton window air conditioner, rated EER 10, was modified so it could be tested with or without LOF features. Both baseline tests (without LOF) and LOF tests were performed in an environmental chamber at Air Conditioning and Refrigeration Institute (ARI) indoor rating conditions, and with the outdoor ambient temperature ranging from 82°F to 110°F. The test results indicated that LOF operation improved the system cooling capacity and COP. At 82°F, LOF improved cooling capacity by 14% (and COP by 10%) over the baseline. When the ambient temperature increases, the improvement decreases. At 110°F, the system COP becomes equal to, or less than, that of the baseline. Other than higher cooling capacity and system COP, LOF has a lower compressor high-low pressure ratio, lower compressor discharge temperature, higher refrigerant mass flow rate, slightly higher power consumption, and slightly higher suction pressure. There is an optimum refrigerant charge for LOF operation. Refrigerant overcharge could lead to lower cooling capacity and system COP.

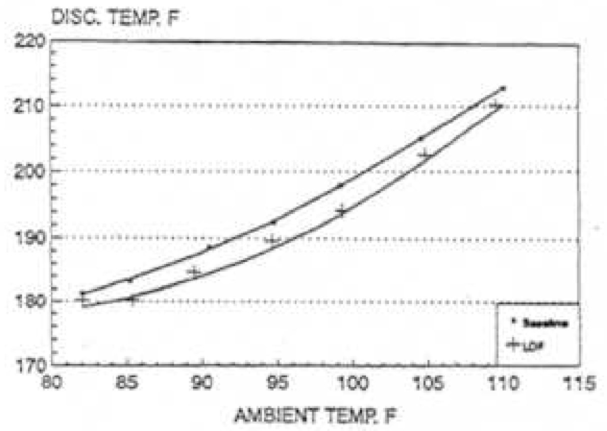

Figure 6 *COP: baseline vs. LOF.*

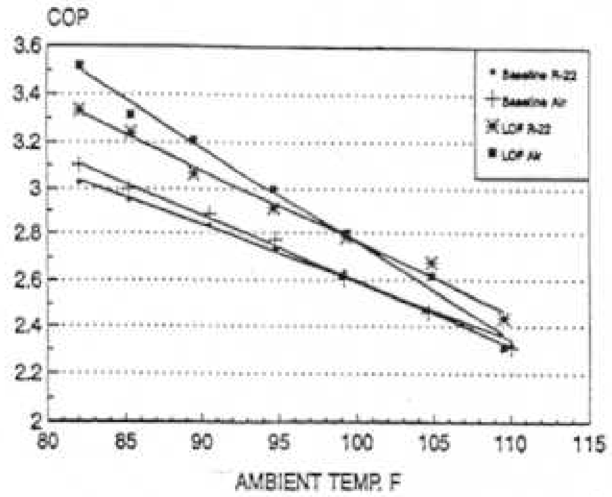

Figure 7 *Compressor high-low pressure ratio: baseline vs. LOF.*

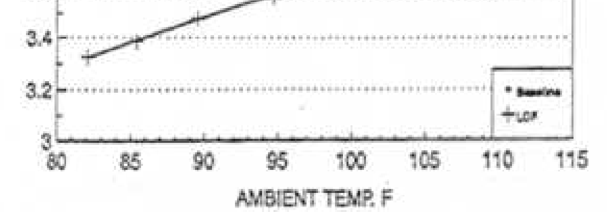

Figure 8 *Compressor high-low pressure ratio: baseline vs. LOF.*

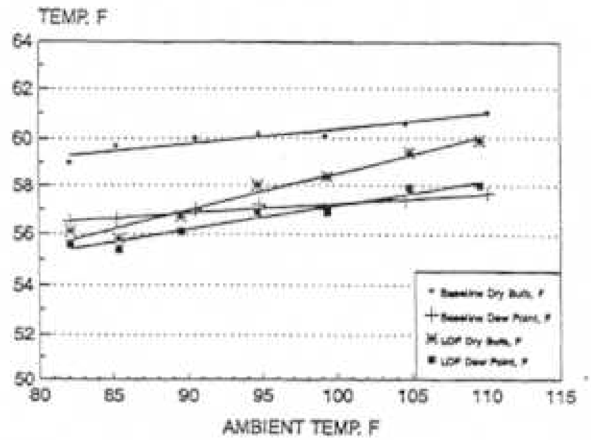

Figure 9 *Evaporator exit dry-bulb and dew-point temperatures: baseline vs. LOF.*

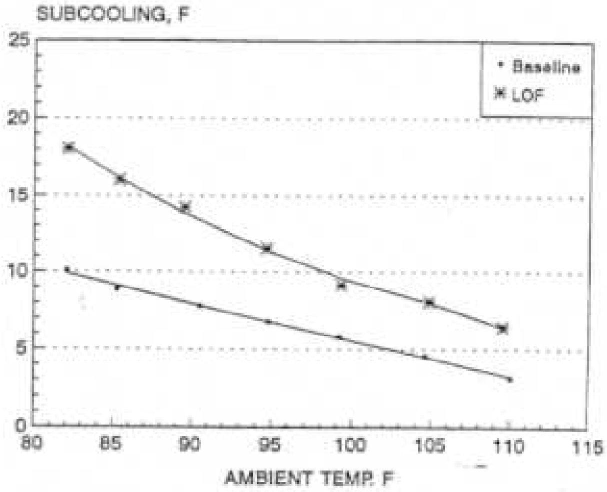

Figure 10 *Refrigerant subcooling: baseline vs. LOF.*

Overall, the LOF system performed as expected. The LOF design is simple and easy to implement. For units already designed with accumulators, additional costs for LOF operation could be minimal. If this is not an option, the added cost of an AHX and rerouting of the liquid and suction lines should be considered.

ACKNOWLEDGMENTS

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