A LIQUID OVER-FEEDING MILITARY AIR CONDITIONER

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ABSTRACT

A 3.3-ton military air conditioning unit has been studied experimentally in both baseline (as received) and as modified for liquid over-feeding (LOF) operation. The baseline test, using a proper refrigerant charge, showed the measured cooling capacity to be less than 1% off the rated capacity at 95°F ambient temperature. The test results indicate that LOF operation outperforms the baseline case over a wide ambient temperature range in terms of cooling capacity, power consumption, and system coefficient of performance (COP). At a 95°F test point, LOF operation has a cooling capacity of 51,100 BTU per hour, which is a 28.9% improvement over the baseline operation capacity of 39,600 BTU per hour. The COP for LOF at 95°F is 2.62, which is 29% better than the baseline COP of 2.03. However, an optimal refrigerant charge is essential for LOF to work properly.
1. INTRODUCTION

The army has unique requirements for compact, rugged air conditioners. These units are used to cool command, control, communications, and intelligence electronics in mobile shelters. These "window-type" units are available in a family of standardized designs ranging from 6,000 to 60,000 BTU per hour and include both horizontal and vertical configurations. These units use R-22 as their refrigerant. Their efficiency and performance have been constrained by the compact design and rugged construction necessary for them to fit and operate on mobile tactical shelters. Despite the relatively high cost of these units, small incremental increases in performance have been rejected in the past as non-cost-effective. However, the performance gains promised by the LOF technology (Richards 1970) developed at the Oak Ridge National Laboratory (ORNL) for air conditioning applications (Mei and Chen 1993) offered an opportunity to dramatically improve performance with a minimum increase in per-unit cost. This report discusses the results of tests on an Army air conditioner with 39,600 BTU per hour rated capacity (Keco Model F36T-2S) modified for LOF operation. The test data indicate that LOF operation improves system performance over baseline performance by a good margin in power consumption, cooling capacity, and COP at 95°F ambient temperature and indoor conditions of 80°F and 52% relative humidity. The results indicate that LOF technology could possibly change the 33-ton unit to a 4.25-ton unit.

2. BACKGROUND AND TEST SETUP

LOF equipment has been used on large refrigeration systems for many years because of its high capacity and efficiency. However, the system and equipment needed for LOF operation—such as a surge drum, floating valves, and other related components—are too complicated and costly for application in conventional air conditioning. But the patented LOF technology developed at ORNL is easy to implement on an air conditioner. Basically LOF operation requires adding an accumulator-heat exchanger (AHX) in the refrigerant circuit and charging the system adequately with refrigerant. When the system is properly charged, refrigerant from the evaporator will accumulate in the AHX. Liquid refrigerant from the condenser flows through the heat exchanger coil inside the AHX and boils the low pressure refrigerant in the accumulator. This heat exchange between low-side and high-side refrigerant results in highly subcooled high-side refrigerant at the entrance to the expansion device. In addition, the refrigerant boiling in the AHX results in saturated (or near saturated) vapor going into the compressor, which improves compressor volumetric efficiency and results in increased refrigerant mass flow. With higher refrigerant mass flow and higher liquid subcooling, the evaporator cannot evaporate all the refrigerant and therefore has two-phase refrigerant flow which accumulates in the AHX as mentioned above. The result is that 100% of the evaporator is wetted instead of the 85% or so in conventional systems, increasing the cooling capacity. For this particular unit, the compressor efficiency is high. But because part of the liquid has been used to quench the suction line superheat, the overall system efficiency is compromised.
Figure 1 is a schematic of the original unit. Figure 2 is a schematic of the unit modified for LOF operation; the modifications permit switching back and forth between normal and LOF operation. An additional expansion device is present in the original system to inject liquid refrigerant into the suction line to reduce superheat and hence discharge temperature (commonly known as a quench valve). Because the compactness of the compartment makes it difficult to reach the liquid line, holes were drilled on the side of the unit for additional piping required for the LOF test. The turbine meter, which measured the total refrigerant volumetric flow rate, had to be installed upstream of the quench valve. For the tests, refrigerant-side flow rate calculation could not be performed because part of the liquid flows through the quenching valve to lower the suction line superheat. Therefore refrigerant-side measurements are basically for temperature and pressure. The flow rate measurement is for reference only.

Figure 3 shows the outside appearance of the unit after the modifications. The major components were not altered except for the thermal expansion valve, which was by-passed by two metering valves connected in parallel that served as the expansion devices. An LOF system requires either capillary tubes or orifice plates as expansion devices. Conventional thermal expansion valves that control refrigerant flow based on superheat cannot be used, as LOF operation generates little or no superheat. In this study, two 0.125-in.-diameter orifice metering valves were installed in parallel as the expansion devices.

Figure 4 shows the air-side test setup. One thermocouple pile (6 thermocouples) was installed on the air inlet and another thermocouple pile (9 thermocouples) was at the air outlet for dry bulb temperature measurements. Wet bulb temperatures were measured with a thermocouple wire covered with a wetted wick. All measurements were on the air side so that dehumidification capability could be estimated. A three-phase power meter was used to measure the system power consumption.

Before any modification, the unit was checked for leaks and then evacuated and charged with 6.0 lb of R-22, the amount of refrigerant charge specified on the name plate (the specification sheet of the model called for 6.2 lb, see Appendix B). The baseline tests were performed at an ambient temperature of around 95°F and indoor conditions of 80°F and 52% relative humidity. The cooling capacity measured was 38,400 BTU per hour, which was about 3% below the rated capacity of 39,644 BTU per hour. The unit was then modified with the LOF feature. The baseline tests were repeated with about 7.9 lb of refrigerant charge for the extra piping involved. The cooling capacity for baseline operation was improved to 39,360 BTU per hour, which is within 1% of the rated capacity. An additional 2.2 lb (a total charge of 10.1 lb) of R-22 was added into the system and the baseline tests were repeated again. LOF tests were also conducted at both charge levels (7.9 lb and 10.1 lb).
Fig. 1. Refrigerant-side schematic - original design
Fig. 2. Refrigerant-side schematic - modified design
Fig. 3. LOF air conditioner
Fig. 4. Air-side schematic
The tests were performed over a range of ambient temperatures from 80 to 110°F. The unit was operated at an ambient temperature until it reached steady state operation, and then the data were collected over 3 to 4 min. The raw data were averaged over that period of time.

3. Test Results

The baseline test data before and after the system modifications were almost identical, indicating that the modification of the system did not affect its baseline performance. The specification of the unit shows that it was designed for 39,644 BTU per hour at 95°F ambient. The baseline test results showed a cooling capacity of 39,360 BTU per hour at 7.9 lb charge, which is less than 1% off the design capacity.

3.1 R-22 Charging at 7.9 lb

Figure 5 shows the cooling capacities of LOF and conventional system operation. The cooling capacity using LOF is about 8.4% higher than that for the baseline. At high-ambient conditions, the improvement decreases. This is because the refrigerant mass flow rate for baseline operation increases as a result of increased suction pressure and consequent higher vapor density, which results in a smaller dry portion of the evaporator coil. There will thus be less potential for LOF to improve cooling capacity. However, if the refrigerant charge were optimized at a higher ambient temperature, of, say, 110°F, we expect that LOF operation would outperform the baseline case by a good margin.

Figure 6 shows the evaporator exit dry bulb and wet bulb temperature comparison. LOF operation has lower dry bulb and wet bulb temperatures than baseline operation, indicating that LOF has higher sensible and latent load capacities. LOF also has better dehumidification capability.

Figure 7 compares system power consumption, 5.83 kW for baseline and 5.70 kW for LOF operation at 95°F ambient. When the ambient temperature is 83°F or higher, LOF operation actually consumes less power. This is because the compressor suction takes in nearly saturated vapor, resulting in lower discharge temperature and lower discharge vapor enthalpy. At higher ambient temperatures, the power consumption for both operations becomes closer.

Figure 8 compares system COP. At 95°F, the improvement for LOF over baseline operation is about 11.1% because of the combined effect of lower power consumption and higher cooling capacity.
Fig. 5.  Cooling capacity, LOF vs baseline - 7.9 lb R-22 charge

Fig. 6.  Evaporator exit dry and wet bulb temperatures, LOF vs baseline - 7.9 lb R-22 charge
Fig. 7. System power consumption, LOF vs baseline - 7.9 lb R-22 charge

Fig. 8. System COP, LOF vs baseline - 7.9 lb R-22 charge
Figure 9 is a comparison of system discharge pressures. The discharge pressure for baseline operation is actually equal to or slightly higher than that of LOF operation at 95°F ambient. This is encouraging, because it indicates that LOF operation does not penalize the compressor.

3.2 R-22 CHARGING AT 10.1 lb

When the unit was operated with LOF at the higher charge, the evaporator refrigerant outlet temperature was equal to or lower than the evaporator refrigerant inlet temperature (after the expansion devices). The performance for both baseline and LOF operation changed at this charge. General observation showed that the compressor discharge pressures approached 400 psia (at 110°F ambient), and the cooling capacity improved for both baseline and LOF operation. Because the pressure relief valve was set at around 420 psi, the system was not tested at any temperature above 110°F to avoid any pressure relief.

Figure 10 is a comparison of the cooling capacity for baseline and LOF operation. The baseline and LOF cooling capacities have increased to over 42,500 and 51,100 BTU per hour, respectively, at 95°F ambient, showing an LOF advantage of over 18%. The potential to improve the cooling capacity of the unit from the original rating of 39,644 BTU per hour to over 51,100 BTU per hour, a 28.9% improvement in cooling capacity, has been demonstrated with the LOF technology.

Figure 11 shows the dry bulb and wet bulb temperatures at the evaporator air inlet and outlet for LOF and baseline operation. Wet bulb and dry bulb temperatures for LOF operation are much lower than for baseline operation, about 3°F lower at 95°F ambient. This means the LOF operation has both higher sensible and latent load cooling capacities. However, the advantages of LOF operation slowly disappear at higher ambient temperatures because the refrigerant charging for LOF operation was optimized at 95°F ambient. If we had optimized the charging at a higher ambient, we would expect that LOF operation would outperform the baseline at a higher ambient; but then when the ambient was lower, the advantages of LOF would have been reduced.

Figure 12 shows the power consumption of the air conditioner in both modes. Baseline operation actually consumes more power. One reason is that the quench valve, which shunts part of the liquid from the condenser to the suction line, lowers the compressor discharge temperature. However, this liquid bypassing also causes the compressor to pump some extra liquid without directly adding any cooling capacity at the evaporator. During LOF operation, the quench valve is not bypassing as much liquid as during baseline operation because there is little superheat at the suction line. At 95°F ambient, the power consumption for LOF is about 6.5% lower than for baseline operation. Again, LOF operation was optimized at 95°F ambient, and the power saving is a maximum at that temperature.
Fig. 9. Compressor discharge pressure, LOF vs baseline - 7.9 lb R-22 charge

Fig. 10. Cooling capacity, LOF vs baseline - 10.1 lb R-22 charge
**Fig. 11.** Evaporator exit dry and wet bulb temperatures, LOF vs baseline - 10.1 lb R-22 charge

**Fig. 12.** System power consumption, LOF vs baseline - 10.1 lb R-22 charge
Figure 13 compares system COPs. At 95°F, the system COP for LOF operation is at 2.623. The percentage improvement of LOF over baseline system COP is 29.3%—an even greater improvement than in cooling capacity because not only is cooling capacity for LOF much higher, but also power consumption for LOF is lower. If we compare LOF COP with the air conditioner's baseline (7.9 lb charge) COP of 1.975 (see Fig. 8), the improvement is 32.8%.

Figure 14 is a comparison of the compressor discharge pressures. The discharge pressures for LOF and baseline operation are very close over a wide range of ambient temperatures. LOF discharge pressure is slightly lower than baseline operation pressure at low- and high-end ambient temperatures.

4. DISCUSSION

A 3.3-ton military air conditioner was tested for both baseline and LOF operation. The tests were performed at two levels of LOF refrigerant charging. In one case, refrigerant was charged at 95°F ambient into the system until LOF started taking effect—at 7.9 lb. In that case, LOF had 8.4% higher cooling capacity, lower power consumption, and an improvement of more than 11% in system COP over baseline operation. In the second case, refrigerant was charged into the system, again at 95°F ambient, during LOF operation until the refrigerant temperature at the evaporator outlet was equal or slightly lower than the temperature at the evaporator inlet. The cooling capacity for baseline operation improved from 39,600 to 42,500 BTU per hour. However, the cooling capacity for LOF operation improved to 51,100 BTU per hour, and system dehumidification capability also was greatly improved. The compressor discharge temperatures and pressures for the two operating modes in both cases were very close.

These encouraging results indicate that the cooling capacity of the unit can be improved from 39,600 to 51,100 BTU per hour and the system COP improved from 2.03 to 2.62, both 29% gains. All these improvements can be achieved cost-effectively.

The quench valve interfered with the performance of LOF operation by shunting liquid to the suction line. We believe compressor power consumption will be lowered even further if the quench valve can be bypassed or closed.

In addition, LOF operation tends to increase the suction pressure slightly because of refrigerant boiling in the AHX. The combined effect of lowering the compressor discharge pressure and slightly increasing the suction pressure reduces power consumption and improves compressor reliability.
Fig. 13. System COP, LOF vs baseline - 10.1 lb R-22 charge

Fig. 14. Compressor discharge pressure, LOF vs baseline - 10.1 lb R-22 charge
5. SUGGESTIONS FOR MODIFYING MILITARY AIR CONDITIONER DESIGN

The following suggestions for possible military air conditioner design modifications could lead to improved cooling capacity and compressor reliability and lower system cost.

- Because LOF already has an AHX in the system, the receiver used in the current design can be removed.

- Because LOF will have little or no suction line superheat, the quench valve can be removed.

- Because LOF will have little or no suction line superheat, inexpensive capillary tubes or orifice plates can be used instead of the expensive thermal expansion valve, reducing the system cost.

6. CONCLUSIONS

The experimental data clearly showed that LOF outperformed baseline operation over a wide range of ambient temperatures in terms of much higher cooling capacity and system COP. The test results are extremely encouraging. A unit normally rated at 39,600 BTU per hour could be modified with LOF to have a cooling capacity output of 51,100 BTU per hour, a 29% improvement. COP is further improved because power consumption using LOF is lower than in baseline operation. However, the compressor would have to be operated at around 300 to 400 psi, or even higher for higher ambient temperatures. We think the conventional compressor can easily stand this pressure.

Because the refrigerant charge was optimized for 95°F operation, the improvement of LOF over baseline operation decreases at higher ambient temperatures, indicating that optimal refrigerant charging is very important for LOF operation. If the charge were optimized at, for example, 110°F, we would expect better LOF system performance at 110°F. However, the improvement in system performance would then probably be smaller at lower ambient temperatures. Overcharging the system excessively might result in a loss of cooling capacity: Excessive suction pressure backup caused by more refrigerant boiling in the AHX at high ambient temperatures would result in higher suction temperatures and thus higher evaporator coil temperature.

If compactness is important, LOF can be used in smaller evaporator and condenser sizes and maintain the same cooling capacity. The addition of the AHX means that the liquid receiver can be eliminated. LOF operation also can replace expensive expansion valves with low-cost orifice plates or capillary tubes, thus lowering the overall cost, and improving reliability. For future air conditioners using R-22 replacement refrigerants, LOF provides a even more promising improvement (Mei et al. 1995). Currently, all the promising R-22 substitutes are hydrofluorocarbon mixtures. For nonazeotropic refrigerant mixtures (NARMs), a higher subcooling level means lower evaporator inlet
temperature after expansion because of the temperature glide. LOF thus will enhance system performance with NARMs. LOF presents an opportunity to update the design of future Army air conditioning units by improving performance cost-effectively.

For the mixed refrigerant test with Du Pont AC-9000, a promising R-22 replacement, the compressor will have to be cleaned with new oil. Because the air conditioner compressor compartment was practically full, it would have been nearly impossible to change the compressor oil on this unit. Our technical manager at Fort Belvoir decided that the mixed refrigerant would be tested on an off-the-shelf window air conditioner. That work will be performed at a later date. However, a window air conditioner with an EER 10 rating has already been modified with LOF. An addendum will be delivered to the project technical manager once this testing is completed.

7. RECOMMENDED FUTURE WORK

Based on the experimental data acquired through the tests on the air conditioner and discussions with our technical manager at Fort Belvoir, it is recommended that the following work be performed on Army environmental control units (ECUs).

- Optimize the LOF refrigerant charge at an ambient temperature range of 110 to 120°F.
- Optimize the AHX at the smallest possible size for acceptable results.
- Test a nonazeotropic mixed refrigerant such as Du Pont AC-9000 in an air conditioner with a counter-cross-flow evaporator to take advantage of the temperature glide of the refrigerant mixture and further enhance system performance.
- Tests be performed at lower than 80°F ambient temperatures.
- Tests be performed at higher than 120°F ambient temperatures, if the unit can be operated at such ambient conditions.
- Design and develop next generation of ECUs with LOF technology.
8. REFERENCES


Appendix A

COMPARISON OF AIR-SIDE AND REFRIGERANT-SIDE CALCULATION

Refrigerant-side measurement is usually very accurate because the measurement of refrigerant volumetric flow rate with a turbine meter is accurate. Measuring temperatures and pressures is much easier than measuring the air dew point or wet bulb temperatures. In this study, however, we found that because the refrigerant flow rate measurement was upstream of the quench valve (because of the difficulty in cutting into the liquid line downstream of the quench valve), the refrigerant flow to the evaporator could be smaller than the measured value. Part of the liquid bypassed the evaporator and went directly to the suction line through the quench valve. In a liquid overfeeding (LOF) system, because the level of superheat is lower, the amount of liquid bypassing the evaporator could be smaller, but it would not necessarily be zero. For LOF operation at low ambient temperature, the suction line superheat is lower than at higher ambient temperatures. It is possible that the quench valve could be closed at the low ambient temperature.

Figure A.1 shows the refrigerant volumetric flow rate, at 6.2 and 7.9 lb R-22 charge, as a function of ambient temperatures. The baseline refrigerant volumetric flow rate is higher than the flow rate in LOF operation, which could not be explained unless the quench valve was wide open. Our past experience in air conditioner testing has always shown a refrigerant flow rate increase in LOF operation over the flow rate in baseline operation. We are not sure at what ambient temperature the quench valve opens for LOF operation. However, the calculated cooling capacity for LOF operation might provide some information about the influence of the quench valve. Figure A.2 shows the cooling air-side and refrigerant-side cooling capacities for LOF operation, at 7.9 lb R-22 charge, as a function of ambient temperatures. The cooling capacities tell more about the quench valve opening. If the quenching valve were closed, the refrigerant-side and air-side cooling capacities should be very close to each other. Figure A.2 shows that the air-side cooling capacity, as expected, decreases as the ambient temperature increases. The refrigerant-side shows the opposite trend, indicating interference from the quench valve. At the low end of the ambient temperature, the quench valve could be closed.

Overall, we think the refrigerant-side calculated cooling capacity is not dependable. The air-side cooling capacity measured at 95°F was 39,360 BTU per hour, less than 1% off the rated cooling capacity of 39,600 BTU per hour. This proves that the air-side cooling capacity measurement is more dependable. Throughout this report, the cooling capacities and COPs calculated by the data collected on the air-side measurement are presented.
Fig. A1. Refrigerant volumetric flow rate, LOF vs baseline

Fig. A2. LOF cooling capacity, air-side vs refrigerant-side
Appendix B

SPECIFICATION OF THE MILITARY AIR CONDITIONER TESTED
(KECO UNIT MODEL F36T-2S)
Manufacturer: KECO, Florence, KY

Model: F36T-2S

1. Cooling

A. Capacity at 120°F outdoor, 90°F DB and 75 °F WB indoor:
   a. Net cooling (minimum): 37,800 BTUH
   b. Net cooling (maximum): 42,165 BTUH
   c. Net Sensible (minimum): 22,880 BTUH
   d. Net Sensible (maximum): 28,003 BTUH

B. Capacity at 95°F outdoor, 80°F DB and 67°F WB indoor:
   a. Net cooling (minimum): 37,800 BTUH
   b. Net cooling (maximum): 39,644 BTUH
   c. Net Sensible (minimum): 24,570 BTUH
   d. Net Sensible (maximum): 28,600 BTUH

2. Heating

   a. Minimum: 28,600 BTUH
   b. Maximum: 31,681 BTUH

3. Evaporator air flow: 1250 SCFM (dry coil at 0.0" static pressure).

4. Compressor: Hermetic, reciprocating, 208 V, 3 phase, 50/60 HZ.

5. Fan motor: Two evaporator, one double shaft condenser, 208 V, 3 phase, 50/60 HZ.

6. Power consumption: 8.5 kW.
   b. Power Factor: 0.84

7. Air filter:
   a. Duct holding capacity: 33 c/100 in.sq.
   b. Air Filter Institute Test Dust (AFITD): 57% minimum.

9. Refrigerant: R-22


11. Condenser coil: Aluminum fins with tin plated copper tubes.

12. Evaporator coil: Aluminum fins with tin plated copper tubes.


14. Operating controls:
   a. Selector switch.
   b. Cooling/heating temperature control.
   c. Fresh air control.

15. Mode of operation:

   Ventilating, cooling, and heating (high/low)


17. Starting currents: 103 Amps.