A Liquid Overfeeding Military Air Conditioner with a Quench Valve

Vince C. Mei, Ph.D., P.E.

Fang C. Chen
Member ASHRAE

Chris W. Bolton

ABSTRACT

A 3.3-ton rated military air-conditioning unit has been studied experimentally for baseline and liquid overfeeding (LOF) operation. The tests before any modification, using the nameplate-specified refrigerant charge, showed the measured cooling capacity to be less than 1% off the rated capacity at 95°F (35°C) ambient temperature. The test results, after modification, 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 (35°C) test point, LOF operation has a cooling capacity of 51,100 Btu per hour, which is an 18% improvement over the baseline operation capacity of 42,500 Btu per hour. The COP for LOF at 95°F (35°C) 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.

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 (Btu/h) 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 not cost-effective. However, the performance gains promised by the LOF technology (ASHRAE 1994; Richards 1970) developed for air-conditioning applications (Mei et al. 1995a) offered an opportunity to dramatically improve performance with a minimum increase in per-unit cost. This paper discusses the results of tests on an Army air conditioner with a rated capacity of 39,644 Btu/h 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 (35°C) ambient temperature and indoor conditions of 80°F (26.6°C) and 52% relative humidity (RH). The results indicate that LOF technology could possibly change the 3.3-ton unit to a 4.25-ton unit.

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 LOF technology can be applied to the air conditioner relatively easy. 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, the refrigerant starts accumulating in the AHX. The liquid refrigerant from the condenser flows through the heat exchanger coil inside the AHX and boils the 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 nearly 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 at its exit. The liquid refrigerant is trapped in the AHX and boiled off by the hot liquid line from the condenser. The result is that 100% of the evaporator is wetted, thus increasing the cooling capacity. This approach has been tried before on heat pumps (Farmer 1985)

Vince C. Mei and **Fang C. Chen** are with the Oak Ridge National Laboratory, Oak Ridge, Tenn. **Chris W. Bolton** is with the U.S. Army CECOM RD&E Center, Fort Belvoir, Va.

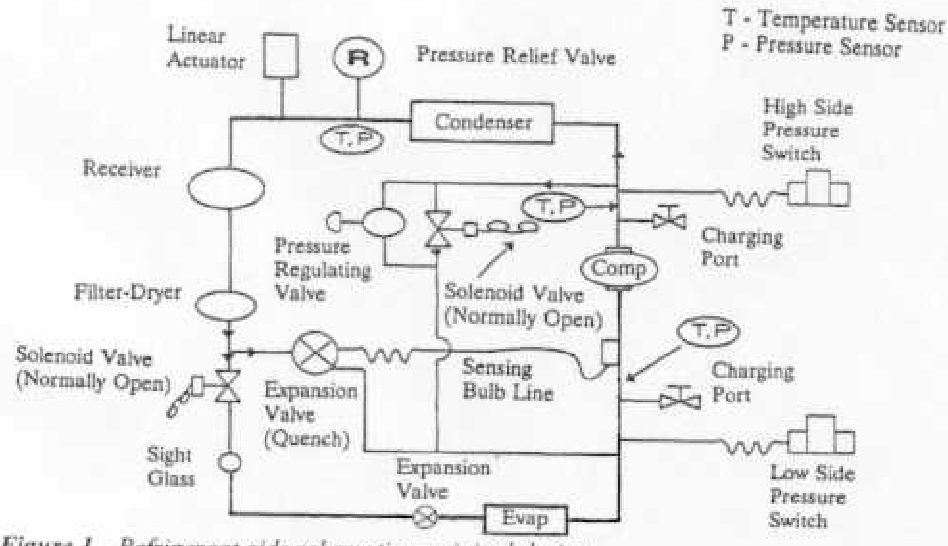

Figure 1 Refrigerant-side schematic—original design.

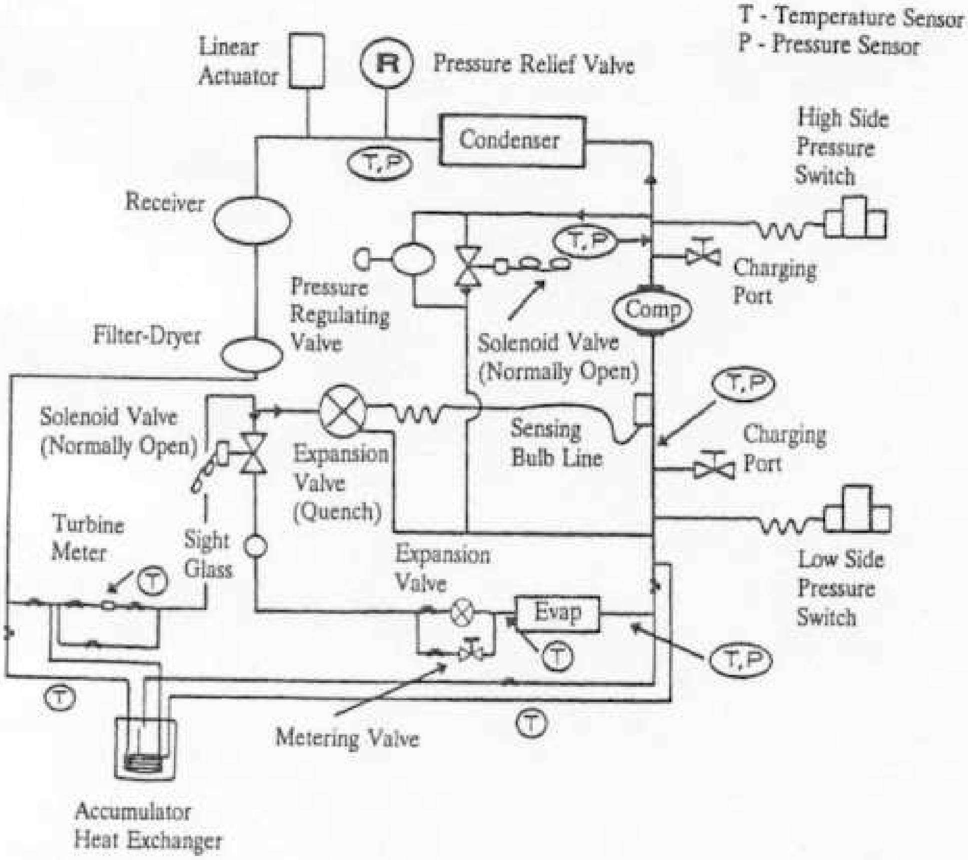

Figure 2 Refrigerant-side schematic—modified design.

with liquid subcooling control. The liquid subcooling control could potentially result in unstable compressor pressure swing. In the present approach, no liquid subcooling control was used. 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, as no original components were modified or changed. 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 the 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, evaporator 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 was for reference only. The major components were not altered except for the thermal expansion valve, which was bypassed 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. (0.317 cm) diameter orifice metering valves were installed in parallel as the expansion devices.

Figure 3 shows the air-side test setup. One thermocouple pile (six thermocouples) was installed on the air inlet and another thermocouple pile (nine thermocouples) was at the air outlet for dry-bulb temperature measurements. Wet-bulb temperatures

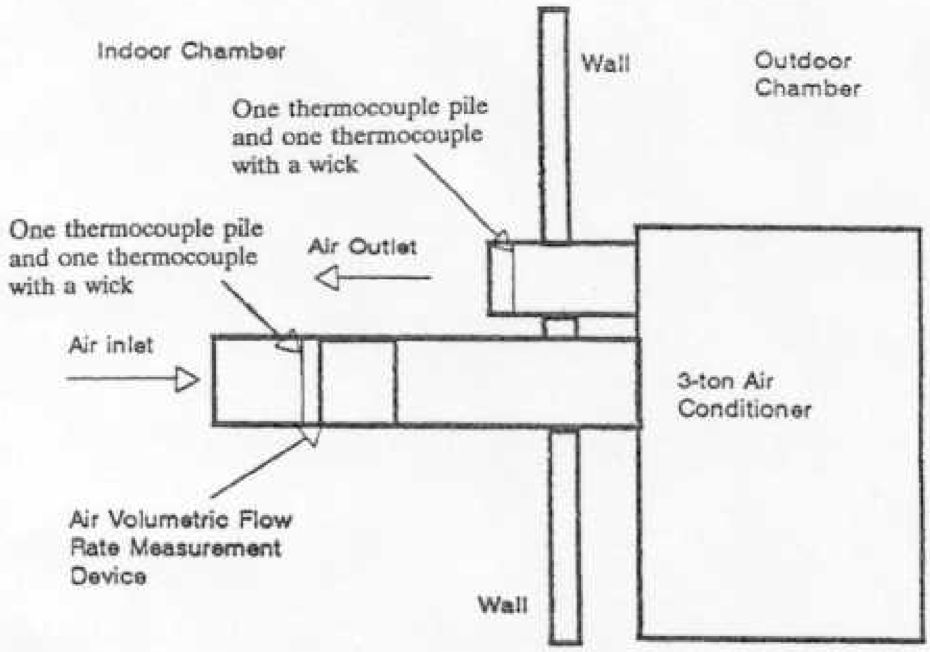

Figure 3 Air-side schematic.

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 6.2 pounds of R-22, the amount of refrigerant charge specified by the manufacturer. Tests were performed at an ambient temperature of 95°F (35°C) and indoor conditions of 80°F (26.6°C) and 52% RH. The cooling capacity measured was 38,400 Btu/h, which was about 3% off the rated capacity of 39,644 Btu/h. The unit was then modified with the LOF feature. The baseline and LOF tests were performed with about 7.9 pounds of refrigerant charge for the extra piping involved and for the LOF to start working (Mei et al. 1995a). The cooling capacity for baseline operation was improved to 39,360 Btu/h, which is less than 1% off the rated capacity. The results showed that at 95°F (35°C) ambient, the cooling capacity and COP for LOF operation were 8.4% and 11.1%, respectively, higher than those of the baseline operation (Mei et al. 1995a).

An additional 2.2 lb (a total charge of 10.14 lb) of R-22 was again charged into the system and the baseline and LOF operation tests were repeated. The reason for the additional charge was to have evaporator inlet (after the expansion device) and exit temperatures equal to each other for LOF operation. This operating condition was considered as the optimal refrigerant charge. The experimental findings of 10.14-lb charge, both for baseline and LOF operations, are reported in the paper.

The modified system has completed approximately 300 hours of testing without any observed failures or negative performance impacts. Oil return through the metering hole on the "J" tube inside the AHX was working fine; no oil accumulation inside the AHX was observed.

The tests were performed over a range of ambient temperatures from 80°F to 110°F (26.6°C to 43.3°C). The unit was operated at an ambient temperature until it reached steady-state operation; then the data were collected over three to four minutes. The raw data were averaged over that period of time.

TEST RESULTS

When the unit was operated with LOF at a charge of 10.14 pounds R-22, the evaporator refrigerant outlet temperature was equal to or lower than the evaporator refrigerant inlet temperature (after the expansion devices). The performances for both baseline and LOF operation were measured. General observation showed that as the compressor discharge pressures approached 400 psig (at 110°F [43.3°C] ambient), the cooling capacity decreased for both baseline and LOF operation. Testing above 110°F (43.3°C) was not conducted because of chamber limitations.

Figure 4 is a comparison of the cooling capacity for baseline and LOF operation. The baseline and LOF cooling capacities have increased to more than 42,500 and 51,100 Btu/h, respectively, at 95°F (35°C) ambient, showing an LOF advantage of more than 18%. The potential to improve the cooling capacity of the unit from the original rating of 39,360 Btu/h to

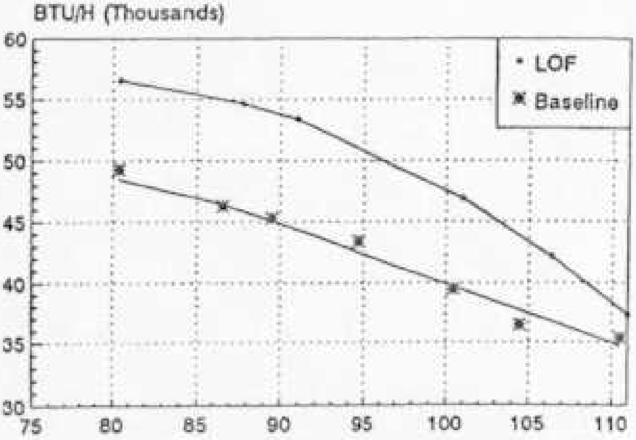

Figure 4 Cooling capacity, LOF vs. baseline.

more than 51,100 Btu/h, a 29.8% improvement in cooling capacity, has been demonstrated with the LOF technology.

Figure 5 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 (35°C) 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 (35°C) ambient. If the charging had been optimized at a higher ambient, LOF operation would be expected to outperform the baseline at a higher ambient; but then when the ambient was lower, the advantages of LOF would have been reduced.

Figure 6 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 refrigerant 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 (35°C) ambient, the power consumption for LOF is about 6.5% lower than for baseline operation. Again, LOF operation was optimized at 95°F (35°C) ambient and the power saving is a maximum at that temperature.

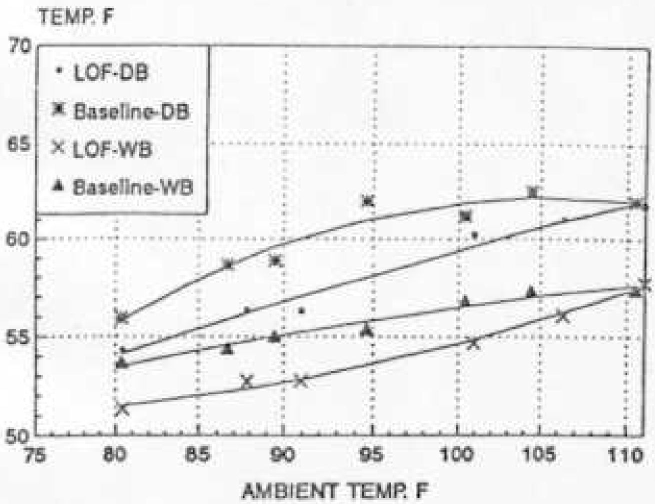

Figure 5 *Evaporator exit dry- and wet-bulb temperatures, LOF vs. baseline.*

Figure 7 compares system COPs. At 95°F (35°C), the system COP for LOF operation is 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 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 (Mei et al. 1995a), the improvement is 32.8%.

Figure 8 is a comparison of the compressor discharge pressures. The discharge pressures for LOF and baseline operation are 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.

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 encouraging. A unit normally rated at 39,360 Btu/h could be modified with LOF to have a cooling capacity output of 51,100 Btu/h with a 10.1-lb R-22 charge, a 29.8% improvement. However, if comparing with the baseline performance of 42,500 Btu/h at higher refrigerant charge, the improvement is about 18%. Because additional piping and an additional accumulator-heat exchanger were added to the system, addi-

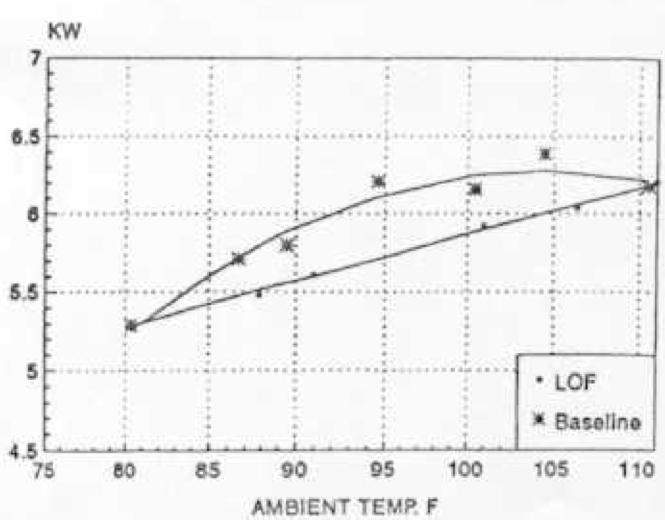

Figure 6 *System power consumption, LOF vs. baseline.*

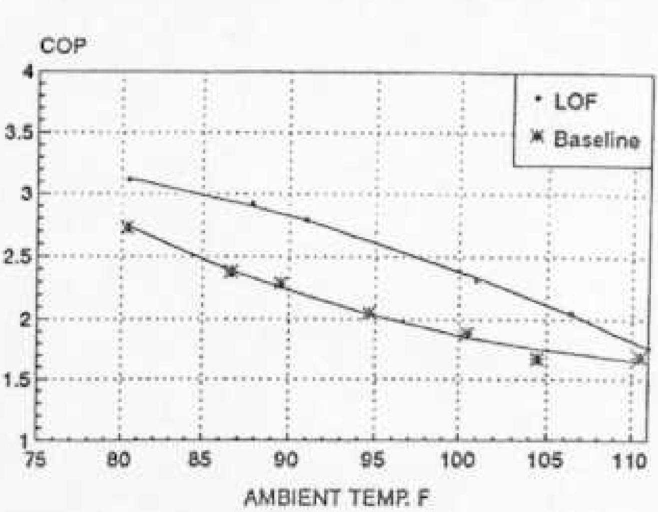

Figure 7 *System COP, LOF vs. baseline.*

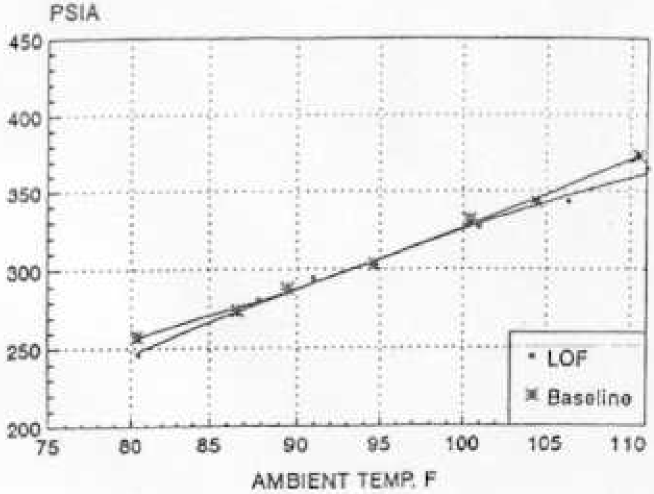

Figure 8 Compressor discharge pressure, LOF vs. baseline.

tional refrigerant was charged. COP is further improved because power consumption using LOF is lower than in baseline operation. Compressor discharge pressure is in the 300- to 600-psia range or higher with the 10.1-lb refrigerant charge. This is within the normal operating range for this system at high ambient temperatures and does not present a problem.

Because the refrigerant charge was optimized for 95°F (35°C) operation, the improvement of LOF over baseline operation decreases at higher ambient temperatures, a good indication that optimal refrigerant charging is important for LOF operation at a specific ambient temperature. Test data (unpublished) for various refrigerant charging showed that overcharging the system excessively might result in a loss of cooling capacity: excessive suction pressure backup caused by more refrigerant boiling in the AHX resulted in higher suction temperatures and thus nigher 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 potentially lowering the overall cost and improving reliability. For future air conditioners using R-22 replacement refrigerants, LOF provides an even more promising improvement (Mei et al. 1995b). Currently, all the promising R-22 substitutes are hydrofluorocarbon mixtures. For nonazeotropic refrigerant mixtures (NARMs), a higher subcooling level means a 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.

ACKNOWLEDGMENTS

The authors are in debt to Dr. A. Patil of the U.S. Army CECOM Research, Development, and Engineering Center for his support of this study. This project was sponsored by the Army CECOM Research, Development, and Engineering Center at Fort Belvoir, Va., under agreement no. 244 CJ with the U.S. Department of Energy, contract no. DE-AC05-84OR21400 with Lockheed Martin Energy Systems, Inc.

REFERENCES

ASHRAE. 1994. *1994 ASHRAE handbook—Refrigeration*, chapter 1. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

Farmer, J. 1985. Personal communication at Oak Ridge National Laboratory about Westinghouse Hy/Re/Li heat pumps.

Mei, V.C., et al. 1995a. A liquid over-feeding military air conditioner. *IECEC Conference 1995 Proceedings*, Aug. 27-30, Orlando.

Mei, V.C., F.C. Chen, D.T. Chen, and E.-P. HuangFu. 1995b. Performance tests of R-22 and R-32/R-125/R-134a mixture for baseline air conditioning and liquid overfeeding operations. *ASHRAE Transactions* 101(2): 1072-1077.

Richards, W.V. 1970. Pumps and piping in liquid overfeed systems. *ASHRAE Symposium Bulletin KC-70-3*. Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

DISCUSSION

Brian Webb, General Manager, Envirothermics, Inc., Winnipeg, Manitoba, Canada: Why not install the HGBP/quench valve assembly upstream of the accumulator/heat exchanger and so not compromise the compressor life span by liquid slugging and excessive suction superheat? (Either could be the case with malfunction of the HGBP assembly.)

Vince C. Mei: It is a good suggestion. But, the quench valve was in the unit as one of the original components. The liquid over-feeding (LOF) feature was added to the system later, based on the concept that no original components, other than expansion devices (by-passed), would be replaced or taken out. The military air conditioner was built so tight that we could not reach the quench valve without taking the unit apart (which we did not do). LOF operation was to have two-phase refrigerant flow at the exit of evaporator. We thought the quench valve will be closed anyway because the suction line superheat would be minimum. The location of the quench was, therefore, not really a concern in our study. For new generation military air conditioners, if LOF is adopted, the quench valve could be eliminated.

Ron Cole, President, R.A. Cole and Associates, Inc., Champaign, Ill.: The presenter reported a reduction in measured compressor power with the LOF evaporator. What were the factors that made that happen? Unless the compressor is operating to the right of the peak of the power vs. the suction pressure curve, the power must increase with an increase in suction pressure.

Mei: One major reason for the compressor power to decrease is that LOF actually has lower refrigerant flow rate than that of the baseline operation. The quench valve for baseline operation was found to be constantly in operation. For LOF operation, the refrigerant by passing the quench valve was much less. For example, at 95°F ambient, one set of out test data showed that baseline refrigerant flow was at about 1.48 gpm, and LOF at about 1.3 gpm. Also, our test results showed that LOF has smaller compression ratio than that of baseline operation, which indicate that LOF operated at higher compressor efficiency.

Will Stoecker, Professor Emeritus, University of Illinois, Urbana: Rather than the orifice valves to control the liquid flow, could a modulating valve sensing the liquid level in the accumulator control the level?

Mei: The liquid Level control valve, usually used in flooded evaporator type systems, can be used for LOF design. However, the cost will be substantial and the system will be complicate for residential air conditioners. The metering valves used in this study was for research study only. If LOF is included in the future military air conditioner design, the metering valve will be replaced by an orifice plate. Because we don't know what size of the orifice plate to be used on this specific unit, metering valves (two in parallels in this study) provide the flexibility needed to optimize the refrigerant expansion process. To match the orifice size with the air conditioner size will require further study.

Steven Lowe, President, Lowe Temperature Solution, Inc., Canterbury, N.H.: How much was the charge increased in your lab unit and projected in a production unit?

Mei: The nameplate of the air conditioner calls for 6.2 lb of R-22 charge. Because of the additional piping and an off-the-shelf accumulator heat exchanger involved in the LOF design, 7.9 lb was charged before LOF started working. The extra piping was substantial in this engineering unit in order to keep all the existing components. An additional 2.2 lb was charged (10.1 total) for LOF to be optimized. In the real production unit, with properly sized accumulator heat exchanger, less additional piping, and possibly no receiver, we project that an additional 10% refrigerant charge (about 0.6 lb) will be sufficient.