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# Test of an Improved Gas Engine-Driven Heat Pump

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## ABSTRACT

*A new generation of natural gas engine-driven heat pumps (GEHPs) was introduced to the marketplace recently. While the units installed have performed exceptionally well and earned rave reviews for comfort and savings on utility bills, the higher initial cost and relatively long payback time have affected the wide commercialization of this advanced technology. According to a study done for the southeastern U.S. in the Atlanta metropolitan area, the annual operating cost of the GEHP is less than that of a baseline system consisting of a 92% efficiency gas furnace and a seasonal energy efficiency ratio (SEER) 12 air conditioner. The estimated payback time is about ten years to cover the difference in initial equipment price between the new and the baseline system.*

*It has been projected that a liquid overfeed (LOF) recuperative cycle concept can simplify the hardware design of a GEHP, resulting in reduced cost and improved performance. Laboratory tests have shown that LOF would improve the energy efficiency of a vapor compression unit by 10%. In addition, LOF will reduce the compressor pressure ratio and thereby improve equipment reliability. Based on the assumed performance improvements and cost reduction, a simple payback calculation indicates LOF can reduce the payback time for an improved GEHP considerably in the Atlanta metropolitan area. Laboratory testing of an improved GEHP has been carried out. This paper reports on the equipment design modifications required to implement LOF and the results of performance tests at steady-state conditions. The preliminary cooling test results have indicated that the LOF, in conjunction with an orifice-type expander, can be applied to a GEHP for cost and performance enhancements. The improvements in energy efficiency will be dependent upon several controlling parameters, including the proper refrigeration charge, the selected ambient temperature, and the system operating condition.*

## INTRODUCTION

A new generation of natural gas engine-driven heat pumps (GEHPs) for residential applications that offer better energy efficiency and thermal comfort was introduced to the marketplace recently. A GEHP differs from a conventional air conditioner in that the Rankine cycle vapor compression system is driven by a natural gas engine instead of by an electric motor. Because of the availability of a gas engine as the drive, a GEHP can easily vary its speed to provide a better load following ability for space conditioning and it can utilize the engine waste heat for winter heating, thus reducing the need for supplemental heating. These salient features result in improved energy efficiencies and thermal comfort and reduced operating costs.

While these installed GEHP units have performed exceptionally well and earned rave reviews for comfort and savings on utility bills, the higher initial cost and relatively long payback time have affected the wide commercialization of this advanced technology. According to a study done for the southeastern United States in the Atlanta metropolitan area (Wolfe and Getman 1995), the annual operating cost of the GEHP is less than that of a baseline system consisting of a 92% efficiency gas furnace and a seasonal energy efficiency ratio (SEER) 12 air conditioner. While the overall operational costs for the GEHP were calculated $135 less than the baseline system, the estimated payback time is about ten years to compensate for the difference in initial equipment price between the GEHP and the baseline system. Innovative means of achieving cost reduction while maintaining its performance and comfort levels appear to be the required development

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needs to make this new space conditioning technology more affordable to the public.

In our research in the developments of energy-efficient and cost effective space conditioning technologies in recent years, we have patented a liquid overfeed (LOF) recuperative cycle concept (Mei et al. 1993). In an LOF recuperative cycle, high-pressure warm liquid refrigerant goes from the condenser through a heat exchanger in an accumulator. The warm liquid in the heat exchanger boils off the low-pressure liquid refrigerant in the accumulator resulting in subcooled liquid. The subcooled liquid then goes through an orifice expander and to the evaporator without being fully evaporated so that the evaporator will be flooded and a two-phase flow will exit at the evaporator exit. The carry-over liquid will be trapped in the accumulator, only to be boiled off by the heat from the warm liquid heat exchanger in the accumulator. In addition, the vapor in the accumulator is almost saturated before it goes into the intake of the compressor, which reduces the compressor pressure ratio and thereby improves compressor volumetric efficiency and equipment reliability. Laboratory tests (Mei et al. 1993) have shown that LOF would improve the energy efficiency of a vapor compression unit by 10% to 15%. Since the GEHP under consideration already has an accumulator, modifying it to include the LOF option is a simple hardware redesign resulting in improved performance at potentially reduced cost. Based on the assumed performance improvements and cost reduction, a simple payback analysis indicates LOF can reduce considerably the payback time for an improved GEHP in the Atlanta metropolitan area. Laboratory testing of an improved GEHP has been carried out. This paper reports on the equipment design modifications required to implement LOF and the results of performance tests at steady-state conditions for assessing the potential of an improved GEHP.

## UNIT MODIFICATION AND TEST SETUP

### Heat Pump Modification

Figure 1 shows the schematic of the heat pump original design. The engine has 17 speeds, the outdoor fan has two speeds, and the indoor fan has five speeds (York 1996). Figure 2 is the schematic of the heat pump modified with a liquid over-feeding feature.

The accumulator was replaced with an accumulator heat exchanger. For laboratory test purposes, four metering valves with a 0.125 in. orifice diameter, two each for the indoor and outdoor units, were installed in parallel to the original expansion valves. The metering valves provide the flexibility needed to optimize the expansion process during laboratory tests. No other major components were replaced or modified.

### Temperature, Pressure, and Air and Gas Flow Rate Measurements

Pressures at compressor inlet and outlet, at condenser outlet, before expansion device, after the expansion device, and at evaporator outlet were measured by six pressure transducers. Dry-bulb temperatures for evaporator air inlet and outlet were measured with two thermocouple piles. Evaporator outlet air wet-bulb temperatures were measured with four thermocouples with wicks. Average wet-bulb temperatures were calculated. Evaporator inlet air wet-bulb temperature was measured with a thermometer covered with a wick. Airflow was measured by a ductwork airflow measurement device that measured the average total head across the ductwork area and then subtracted the average static head, just like

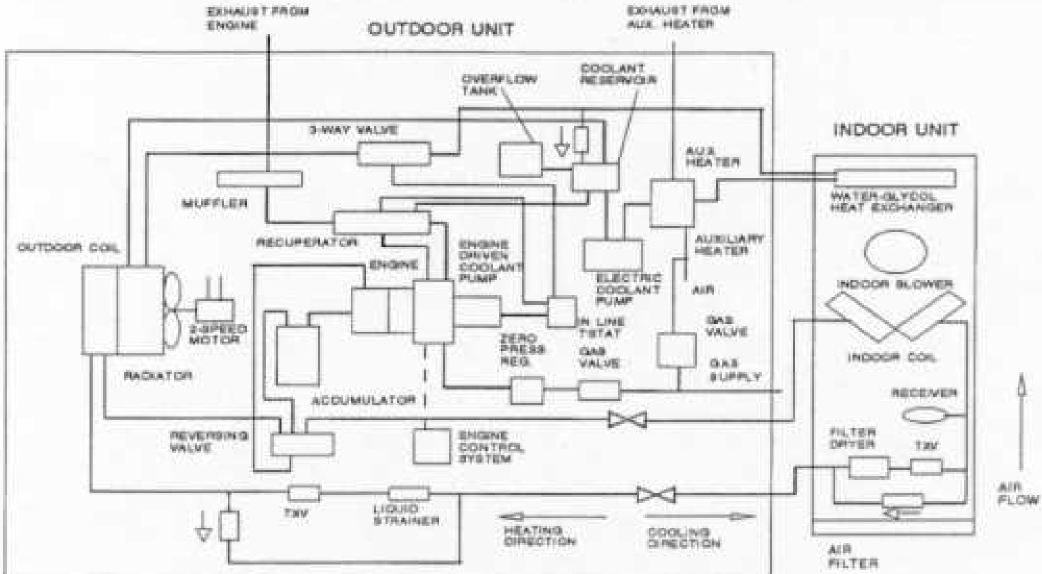


*Figure 1 Schematic of original heat pump design.*

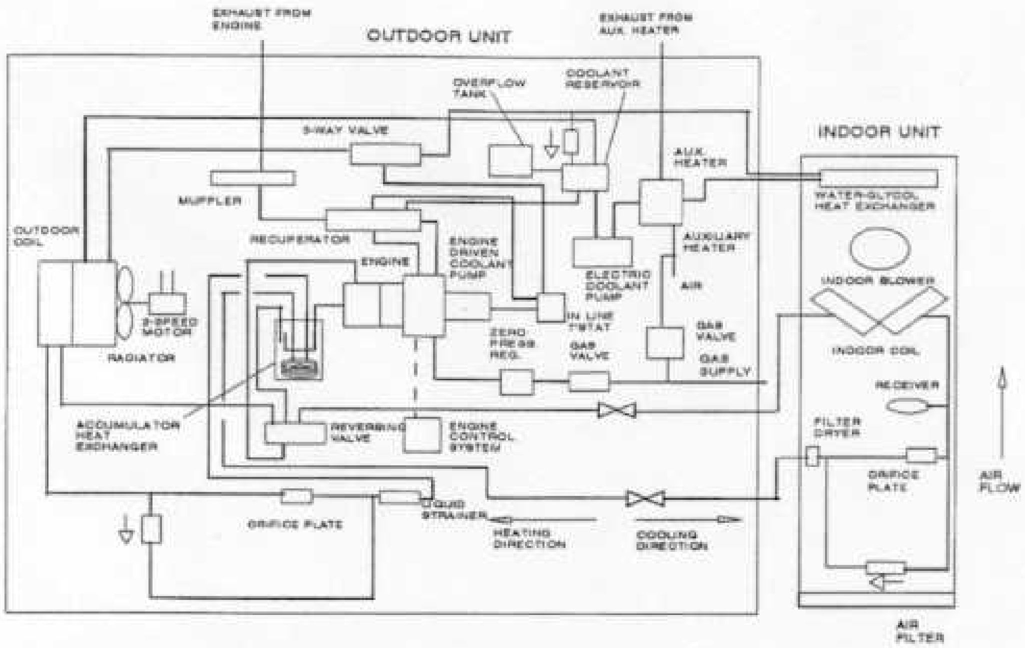


***Figure 2** Schematic of LOF heat pump design.*

a pitot tube. A micromanometer was used to measure the velocity head. Gas consumption was measured with a gas meter. The gas heating value was found to be between 1000 Btu/ft$^3$ and 1030 Btu/ft$^3$. For calculation, 1000 Btu/ft$^3$ was adopted.

## TEST PROCEDURE

### Baseline Cooling Tests

A service analyzer, which allows a service person to obtain system information and to control the heat pump, was attached to the heat pump so the engine speed could be dialed. There are 17 speeds for the engine, ranging from 1200 rpm to 3000 rpm. For this experiment, five engine speeds were selected for testing: 1200 rpm, 1652 rpm, 2104 rpm, 2556 rpm, and 3000 rpm. The ambient temperatures tested ranged from 80°F to 110°F with 5°F interval, including an ARI rated outdoor condition at 95°F. The environmental chamber was set at proper indoor (80°F and 52% RH) and outdoor conditions, and the heat pump was allowed to run until steady-state operation was achieved at a selected engine speed. For each ambient temperature setting, the heat pump performance was tested for five engine speeds.

### LOF Cooling Mode Tests

**Optimization of Metering Valve Opening and Refrigerant Charge.** One of the features for the LOF test is that the thermal expansion valve should not be used because there is little or no superheat at the suction line. Fixed opening expansion devices, such as an orifice plate or a capillary tube will have to be used. The metering valve (as orifice plate) opening becomes crucial. For single-speed compressor heat pumps, the optimization of the metering valve opening will simply be set at 95°F ambient for cooling mode operation, which is straight forward. For variable-speed compressor heat pumps, as the one tested in this study, the optimum setting is becoming difficult because the setting will have to work at all engine speeds and ambient temperatures. The metering valve was expected to be setting in such a way that the performance of the heat pump would be optimized at high engine speeds at high ambient temperatures and at low speeds at low ambient temperatures.

The preliminary test results indicated that at a fixed metering valve opening and ambient temperature, increasing the engine speed will cause the liquid to dry out in the accumulator heat exchanger and decreasing the engine speed will cause the liquid to accumulate in the accumulator heat exchanger.

It was determined by trial and error that the heat pump charging and metering valve opening should be optimized at 95°F ambient and at an engine speed of 2104 rpm. Initially, the system performance was optimized at 95°F and 3000 rpm. However, it was soon found that when the ambient temperature increases (engine speed maintained at 3,000 rpm), the evaporator coils were flooded so that the suction temperature increased and the cooling capacity reduced rapidly; when the engine speed was reduced, the coils were again flooded and the performance decreased. We selected the engine speed at 2104 rpm (middle speed) as the point for system optimization. It was found that one metering valve was enough for the indoor unit. The other metering valve remained shut throughout the test. First, the metering valve was open wide, additional R-22 was charged into the heat pump until the liquid over-feeding effect started, and then the metering valve was slowly adjusted down on its orifice opening until maximum cooling performance was achieved, which could be easily detected by the indication of the evaporator outlet dry- and wet-bulb temperatures.

**LOF Heat Pump Cooling Mode Test.** Once the system optimization is completed, the heat pump test procedure with the LOF feature is the same as the baseline test.

**Data Collection.** The heat pump was run at a fixed indoor and outdoor condition and at a fixed engine speed until steady-state operation was achieved. Data were collected for about five minutes and averaged.

## TEST RESULTS AND DATA ANALYSIS

### Capacity Performance in Baseline Testing

Baseline cooling capacity is shown according to engine speed and outdoor ambient temperature in Table 1 and according to latent cooling capacity in Table 2. The outlined diagonal highlights in each table represent probable engine speeds for the given outdoor temperature. At such conditions, the latent cooling capacity is approximately one-fifth of the total capacity. Latent capacity, as a percentage of total capacity, increases above the highlight and decreases below the highlight due to corresponding increases and decreases in engine speed. The highest latent capacity value, which is also the highest value as a percentage of total capacity, occurs with an engine speed of 3000 rpm and an outdoor temperature of 80°F. This is expected when the evaporator (indoor) coil is at its coldest, giving the highest $\Delta T$ between return air temperature and coil temperature. Correspondingly, the lowest absolute value of latent capacity occurs at 1200 rpm and an outdoor temperature of 110°F, as is expected since such conditions give rise to the lowest $\Delta T$ between return air temperature and evaporator coil temperature. Supplementally, Figure 3 gives a graphical representation of total baseline cooling capacity.

Cooling coefficient of performance (COP) values for baseline testing are shown in Figure 4. These values represent total system COP including natural gas energy input to the engine and electrical energy input to the indoor and outdoor fans and associated control circuits. As can be seen from Figure 4, the total system COP decreases both with rising outdoor temperature and increased engine speed. The highest calculated COP, realized at 1200 rpm and 80°F, is approximately 1.47, and the lowest calculated COP, realized at 3000 rpm and 110°F, is approximately 0.67.

**TABLE 1**
**Total Cooling Capacity—Baseline (kBtu/h)**

| Outdoor Temp | Engine Speed 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
|---|---|---|---|---|---|
| 80°F | 24.5 | 29.1 | 35.7 | 39.3 | 41.5 |
| 85°F | **22.4** | 26.3 | 33.5 | 37.3 | 39.6 |
| 90°F | 20.9 | **25.5** | 31.3 | 35.5 | 38.1 |
| 95°F | 20.1 | 24.7 | **29.8** | 33.3 | 35.4 |
| 100°F | 18.5 | 23.5 | 27.2 | **30.2** | 34.2 |
| 105°F | 17.0 | 22.0 | 25.6 | 28.7 | **30.2** |
| 110°F | 16.4 | 21.5 | 25.2 | 27.4 | 29.5 |

**TABLE 2**
**Latent Cooling Capacity—Baseline (kBtu/h)**

| Outdoor Temp | Engine Speed 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
|---|---|---|---|---|---|
| 80°F | 6.06 | 7.57 | 10.35 | 11.22 | 12.49 |
| 85°F | **5.03** | 5.46 | 9.47 | 10.60 | 11.01 |
| 90°F | 4.17 | **5.18** | 8.43 | 9.78 | 10.52 |
| 95°F | 3.24 | 5.32 | **7.34** | 8.32 | 8.55 |
| 100°F | 2.80 | 4.09 | 5.69 | **6.08** | 8.95 |
| 105°F | 2.10 | 3.61 | 4.76 | 5.12 | **4.92** |
| 110°F | 1.96 | 3.47 | 4.39 | 4.74 | 4.80 |

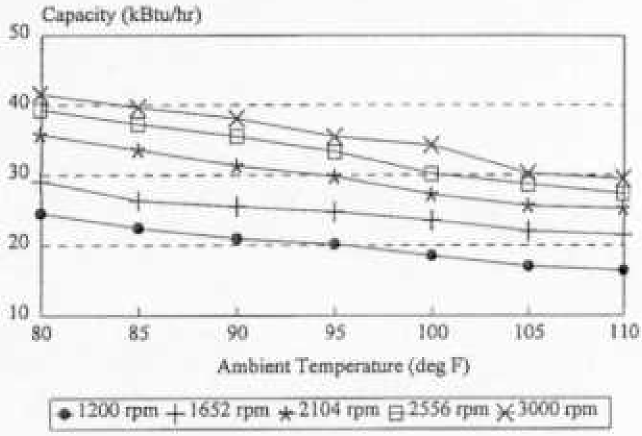


*Figure 3 Total cooling capacity—baseline.*

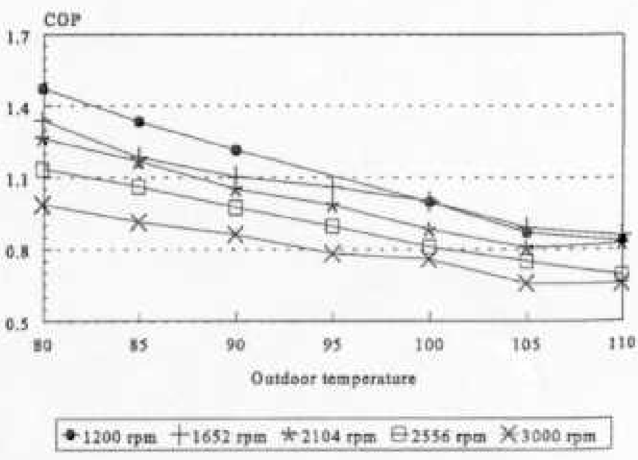


*Figure 4 Cooling COP—baseline.*

## Results of Liquid Over-Feed Testing—With Comparisons to Baseline

Total cooling capacity and latent cooling capacity for LOF tests are shown in Tables 3 and 4, respectively. The outlined areas mark the conditions at which results from the LOF tests exceeded those from the baseline tests. At low temperatures and speeds, LOF results range from slightly under to slightly over results for the same conditions in baseline testing. As outdoor temperature and engine speed increase, the positive differential in performance between LOF results and baseline results increases—both for total cooling capacity and for latent cooling capacity. At the charging condition of 2104 rpm and 95°F, LOF cooling capacity realizes a 13.1% increase over baseline cooling capacity. However, at 3000 rpm and 110°F, LOF cooling capacity is increased by 26.1% over baseline cooling capacity. LOF cooling capacity is likewise shown graphically in Figure 5.

Figure 6 gives the COP values from LOF testing. At low engine speed and low outdoor temperature, LOF values fall just below to just above corresponding baseline values. At

### TABLE 3
### Total Cooling Capacity—LOF (kBtu/h)

| Outdoor Temp | Engine Speed | | | | |
|---|---|---|---|---|---|
| | 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
| 80°F | 21.5 | **30.3** | 32.2 | 32.2 | N/A |
| 85°F | 19.7 | **29.5** | 32.7 | 33.0 | 33.9 |
| 90°F | 17.9 | **27.3** | **33.4** | 32.9 | 35.1 |
| 95°F | 16.9 | 24.7 | **33.7** | **34.6** | **35.6** |
| 100°F | N/A | 22.3 | **31.3** | **34.9** | **37.2** |
| 105°F | N/A | 20.7 | **27.4** | **34.4** | **36.5** |
| 110°F | N/A | 18.2 | 25.1 | **32.4** | **37.2** |

Outlined area shows conditions with increased total capacity.

### TABLE 4
### Latent Cooling Capacity—LOF (kBtu/h)

| Outdoor Temp | Engine Speed | | | | |
|---|---|---|---|---|---|
| | 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
| 80°F | 3.85 | **8.34** | 10.20 | 10.84 | N/A |
| 85°F | 3.14 | **7.72** | **9.82** | 10.24 | 10.04 |
| 90°F | 1.56 | **6.02** | **9.46** | **10.09** | **11.72** |
| 95°F | 0.59 | 4.46 | **8.91** | **9.67** | **10.86** |
| 100°F | N/A | 2.91 | **7.74** | **9.41** | **10.95** |
| 105°F | N/A | 1.68 | **5.27** | **9.36** | **8.92** |
| 110°F | N/A | 0.30 | 3.72 | **7.14** | **9.27** |

Outlined area shows conditions with increased latent capacity.

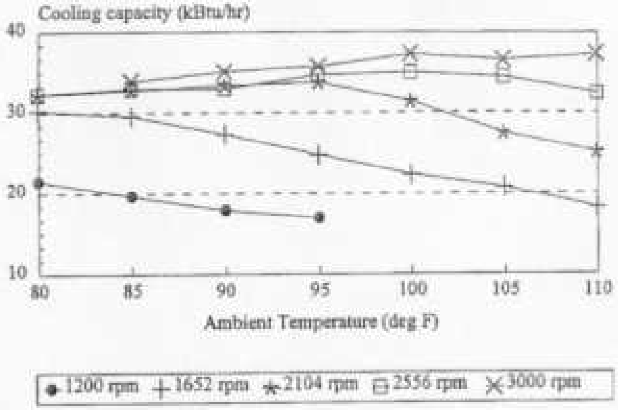


*Figure 5 Total cooling capacity—LOF.*

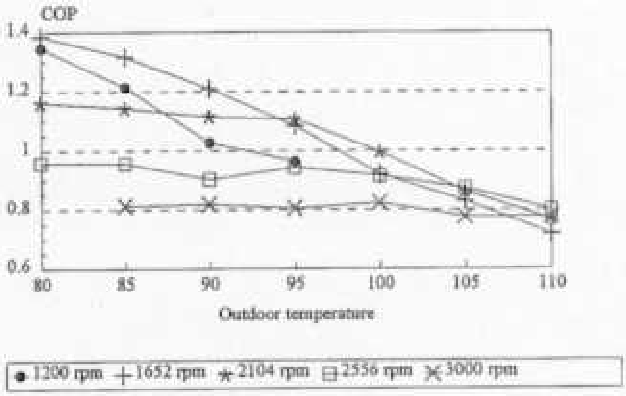


*Figure 6 Cooling COP—LOF.*

high engine speed and high outdoor temperature, LOF values increasingly exceed the corresponding baseline value.

A comparison of the supply-air states for the array of engine speeds and outdoor temperatures is given in Table 5, indicating good agreement with the trends exhibited in the comparisons of total cooling capacity, latent cooling capacity, and system COP. The outlined area marks conditions at which the LOF measured supply air temperatures (dry bulb/wet bulb) are lower than those of baseline tests.

Compressor discharge pressures at three engine speeds—1200 rpm, 2104 rpm, and 3000 rpm—for both baseline tests (dashed lines) and LOF tests (solid lines) are shown in Figure 7. LOF pressures are lower than their baseline counterparts at 80°F, but increase more rapidly as outdoor temperatures rise. In general, LOF discharge pressures are more dependent on engine speed than are those of baseline.

## DISCUSSION AND CONCLUSIONS

In single-speed liquid over-fed systems, the general behavior of refrigerant inventory is to tend toward an under-charged state at ambient temperatures below the design point and to tend toward an over-charged state at ambient temperatures above the design point. An associated deviation from optimum performance comes with these tendencies. Recalling that this LOF test was optimized for the central point of 2104 rpm

## TABLE 5
## Supply Air Conditions

Baseline
(Dry Bulb/Wet Bulb °F)

| Outdoor Temp | Engine Speed | | | | |
|---|---|---|---|---|---|
| | 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
| 80°F | 58.4/57.8 | 57.4/56.7 | 56.2/55.6 | 56.6/55.9 | 58.0/56.4 |
| 85°F | 59.0/58.1 | 58.1/57.1 | 57.5/56.4 | 57.8/56.6 | 58.1/56.8 |
| 90°F | 59.7/58.7 | 59.1/58.4 | 58.2/57.0 | 58.6/57.1 | 59.1/57.4 |
| 95°F | 60.2/59.4 | 59.8/58.6 | 59.1/57.9 | 59.5/58.1 | 59.8/58.3 |
| 100°F | 61.8/60.3 | 60.9/59.6 | 60.3/59.0 | 61.0/59.2 | 61.4/59.4 |
| 105°F | 63.0/60.7 | 61.7/59.8 | 60.7/59.2 | 60.9/59.3 | 61.4/59.5 |
| 110°F | 63.4/60.9 | 62.0/60.1 | 61.2/59.7 | 61.6/59.8 | 62.2/60.1 |

LOF
(Dry Bulb / Wet Bulb °F)

| Outdoor Temp | Engine Speed | | | | |
|---|---|---|---|---|---|
| | 1200 rpm | 1652 rpm | 2104 rpm | 2556 rpm | 3000 rpm |
| 80 °F | 59.2/58.8 | 57.3/56.6 | 59.5/57.0 | 62.7/58.6 | N/A |
| 85 °F | 59.9/59.4 | 57.4/56.8 | 58.7/56.8 | 61.5/58.2 | 62.9/58.7 |
| 90 °F | 60.8/60.2 | 58.6/58.0 | 57.9/56.7 | 61.5/58.3 | 62.6/58.5 |
| 95 °F | 61.5/60.9 | 59.7/59.1 | 57.5/56.8 | 60.1/57.9 | 62.0/58.5 |
| 100°F | N/A | 60.4/59.7 | 58.5/57.6 | 59.8/58.0 | 60.7/58.0 |
| 105°F | N/A | 61.1/60.3 | 59.9/59.0 | 59.3/57.8 | 60.0/58.0 |
| 110°F | N/A | 62.0/61.0 | 60.4/59.5 | 59.8/58.4 | 59.6/57.9 |

Outlined area shows conditions with decreased wet-bulb temperature.

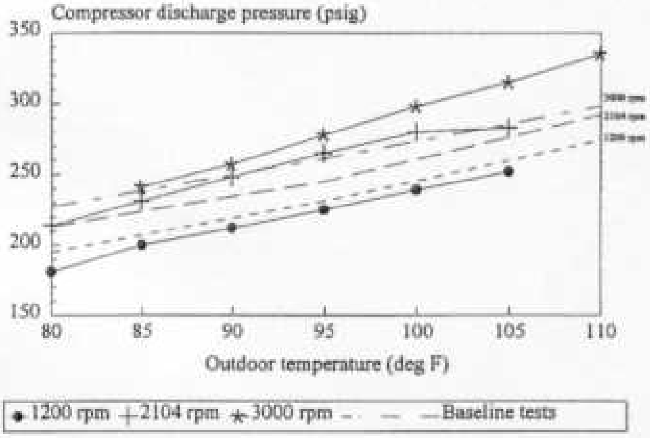


*Figure 7 Compressor discharge pressure.*

at 95°F, the above mentioned tendencies toward over-charging and under-charging are evident. However, the multi-speed nature of the unit adds another variable that acts to compensate for these tendencies to a large degree. For example, when the ambient temperature is above 95°F (at 2104 rpm) the system tends to become over-charged; this, however, can be compensated with an increase in engine speed, with the result being that there are optimum points at each ambient temperature, albeit at different engine speeds.

Since, in general, the call for cooling capacity is small at low ambient temperatures and greater at high ambient temperatures, it follows that the lower the ambient temperature, the lower the necessary engine speed and the higher the ambient temperature, the higher the necessary engine speed. As shown in the data on cooling capacity, LOF optimum points tend to follow the same pattern, yielding higher capacity at the optimum speed for a given temperature. Recalling that only 5 of the possible 17 engine speeds were tested, interpolation of the cooling capacity data approximates the optimum engine speed for 80°F ambient to be 1540 rpm and, similarly, for 110°F ambient to be 2888 rpm, with a linear correlation at intermediate speeds and temperatures.

The COP data reveals the same trend as cooling capacity: for each ambient temperature there is a corresponding engine speed that optimizes LOF performance. As with cooling capacity, particularly good improvement in system COP is realized at high ambient temperature conditions with high compressor speeds. Compressor discharge pressure at optimum LOF points is slightly higher than for similar points at baseline, particularly at high ambient temperatures. This is a consequence of using a fixed size expansion device. The higher pressures are still well within acceptable levels for maintaining compressor integrity.

At conditions well away from optimum—high engine speed and low ambient temperature or low engine speed and high ambient temperature—two distinct paths are followed. In the case of low engine speed coupled with high temperature, the system tends to be substantially overcharged, creating an abnormally warm evaporator. In such a case, there is a risk of sending too much liquid refrigerant into the accumulator and, hence, potentially to the compressor inlet. In the case of fast engine speed coupled with low temperature, the system tends to be substantially undercharged and, hence, operates with a high level of evaporator exit refrigerant superheat. While this is not efficient, as reflected in the system COP, it is not particularly detrimental to the system. These extreme conditions, while tested in the laboratory, are not conditions for which the tested unit was originally designed and, thus, are not of great concern in implementation considerations.

An inherent deficiency in vapor compression air-conditioning systems is the deterioration of cooling capacity at high ambient temperatures, conditions where the most capacity is needed. This previously explained exploration of the effectiveness of liquid over-feeding in a multi-speed system has revealed that such a system has complementary properties that provide much greater cooling capacity at high ambient temperatures. Additionally, the added capacity does not come with a tax on efficiency; in fact, efficiency, measured as total system COP, increases as well.

## ACKNOWLEDGMENTS

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## DISCUSSION

**Vinton Wolfe, Senior Research Engineer, Atlanta Gas Light Co., Atlanta, Ga.:** In the field, refrigerant charge is often determined by the length of the refrigerant lines. Did you do any sensitivity analysis on the effects of reduced or overcharged systems with respect to the optimized charge?

**Fang Chen:** In this reported test, only the point at 2104 rpm and 95°F is optimized and, thus, all other points deviate from optimum--some only slightly and some more significantly. However, even with deviation from optimum performance, there are still many points with increased performance—as noted by blocked regions in the paper. By adjusting the refrigerant charge or the expansion device opening, the optimum performance point can be changed thereby shifting the range of increased performance accordingly.

**John Burgers, Team leader, Long Manufacturing Ltd., Oakville, Ontario:** Have you noticed any negative control aspects associated with transient system behavior (defrost, stop, start, speed modulation). In particular, the trapping of liquid in the accumulator could occur in situations when there is not enough heat transfer from the liquid line to boil liquid out. Could this yield a situation where the suction pressure progressively decreases and the system never recovers?

**Chen:** No such runaway behavior was noticed in this system. It was, however, only tested for steady-state cooling conditions and thus transients imposed by defrosting were not encountered.

**Bob Bach, Senior Consultant, Engineering Interface Ltd., Toronto, Ontario:** (a) Is liquid slugging a potential problem with your system? (b) Have you considered using a value such as the HiReli cycle to better control the potential for liquid slugging?

**Chen:** (a) Liquid slugging is not a problem if the accumulator is sized properly. At some extreme conditions (high compressor speeds and very low condenser side temperatures) the liquid level in the accumulator became high, but at no point did slugging occur. (b) No; the tested system is open-loop whereas the HiReli is closed-loop.

**Carl Hiller, Electric Power Research Institute, Palo Alto, Calif.:** What is the effect on heating mode operation? Is compressor damage a greater risk?

**Chen:** Heating mode tests have not yet been extensively explored. It is expected that performance improvement will occur in heating mode, the degree of which is not yet known. Because the cooling and heating expansion devices are independent, the heating expansion device can be tuned such that liquid slugging is no greater a risk with LOF than without, provided the accumulator is properly sized.

**Thomas L. Davis, Research Project Engineer, Carolina Power & Light Co., Raleigh, N.C.:** (a) Did you look at electric expansion valve for the throttling device? (b) Could you elaborate on your optimization criteria?

**Chen:** (a) No. (b) Optimization was performed at the selected central point of the tested range—2104 RPM and 95°F. The criteria for optimization was maximum air-side cooling capacity.

**Lawrence C. Hoagland, VP, Airxchange, Inc., Rockland, Mass.:** What type and how large is the heat exchanger coil required in the suction accumulator for this liquid overfeed system? Is it likely to add much cost to this system?

**Chen:** A properly sized standard suction accumulator, which has a built-in heat exchanger, will suffice for the liquid overfeed system. The heat exchanger coil consists of a couple turns of copper tubing inside the suction accumulator, and the added cost is deemed to be minimum compared to the total system cost and in view of the measured energy performance improvements of the system.