AC-02-5-2 (4543)

A Frost-Less Heat Pump

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

Ronald E. Domitrovic

Fang C. Chen, Ph.D., P.E.
Member ASHRAE

Joe K. Kilpatrick
Member ASHRAE

ABSTRACT

Frost accumulation on the outdoor coil of a heat pump is one of the basic concerns about heat pump operation. Defrosting by cycle reversing is energy consuming, compromises indoor thermal comfort, and causes excessive and unnecessary use of electric resistance heating.

By adding a moderate amount of heat to the refrigerant stream in the accumulator, the evaporator coil temperature can be raised by several degrees. This concept can be used to retard frost accumulation on the outdoor coil of the heat pump and, thus, keep the coil clean. The frequency of defrosting cycles can be drastically reduced by a factor of five in the Knoxville, Tennessee, area. This study describes the development of a frost-less heat pump.

Also discussed is a new defrosting technology that does not rely on heat taken from the conditioned space of the house by adding heat to the liquid refrigerant in the accumulator. The accumulator becomes an evaporator, providing heat for coil defrosting. In this system, the indoor fan is not energized during the defrosting period. Two-phase refrigerant flowing through the indoor coil will remain two-phase as it enters the accumulator where it then boils. An obvious advantage is that there will not be a cold blow phenomenon during the defrosting period. Energy consumption is modest—around 1 kW—compared to an average of 10 kW resistant heating coil fully energized.

Test data on several engineering units showed that the frost-less concept and the new defrosting scheme worked as expected. Results are presented in this paper.

INTRODUCTION

A heat pump heats a home thermodynamically. It will absorb heat from lower temperature ambient air—or any other heat source—and delivers it to the higher indoor temperature to maintain indoor thermal comfort. If designed right, a heat pump can be very energy efficient for both heating and cooling.

However, there are two major concerns about the heat pump operation during the heating mode operation. One is the "cold blow" that the heat pump supply air temperature is generally in the range between 90ºF and 100ºF, which can cause a sensation of cold air blowing across the skin. When frost starts accumulating on the outdoor coil, the supply air temperature will be even lower. Another concern is the requirement for periodical defrosting of the outdoor coil, which results in taking heat out from the house to defrost the outdoor coil. Not only the resistance heating will be fully energized—the level of "cold blow" could be more severe, particularly near the end of the defrosting period where the supply air temperature could be below the house's indoor air temperature. Also, each defrosting cycle will subject the heat pump to thermal and pressure shock twice.

Research and development work on the cold blow problem has been successfully performed so that air delivery temperature was above 105°F, except the added manufacturing cost to achieve the goal was too high (in the order of $200 to $550)—well over the goal of $100 (TVA 1998). As for defrosting, much work has been reported on heat pump frosting and defrosting performance (Miller 1989; Kondepudi and O'Neal 1987, 1989; Nutter and O'Neal 1996). Work was also reported on shortening the defrosting cycle by adding heat to the accumulator during the defrosting period (Nutter and O'Neal 1996). However, there is practically no reported work

Viung C. Mei is a senior researcher and **Fang C. Chen** is a program manager at Oak Ridge National Laboratory, Oak Ridge, Tenn. **Ronald E. Domitrovic** is a senior research associate at the University of Tennessee, Knoxville, Tenn. **Joe K. Kilpatrick** is a project manager at the Tennessee Valley Authority's Public Power Institute, Chattanooga, Tenn.

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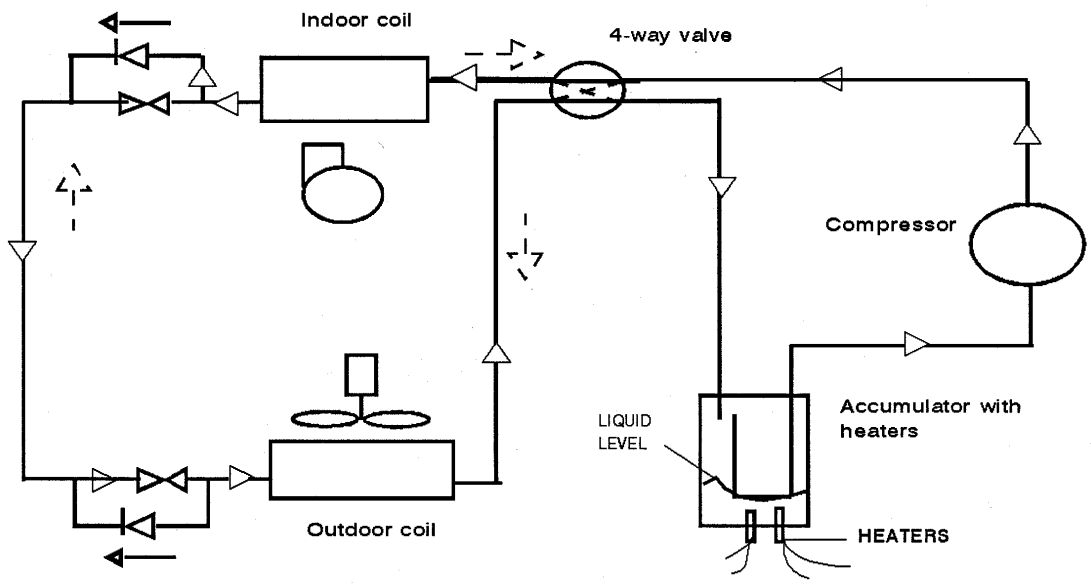


Figure 1 *Refrigerant-side schematic.*

in the public domain discussing the possibility of reducing the number of defrosting cycles by frost retardation on the outdoor coil and the possibility of reducing the severity of cold blow even during the defrosting cycle.

In this study, a novel way to retard the frost formation and a new defrosting technology were discussed and the concept was experimentally proved. These two technologies can be used together or separately. Laboratory test results showed that not only was the frost buildup on the outdoor coil severely retarded, but the heat pump supply air was also warmer than that of the baseline operation by 4ºF to 6ºF. Better yet, there will be no cold air blow even during the defrosting period. The laboratory experiment was performed with a two-ton split heat pump system with a rated EER of 10. The tests were over an outdoor temperature range from 41ºF to 32ºF and with the relative humidity set at 75% to 80%, in which the frost is most likely to form, while the indoor conditions were kept at ARI rated conditions of 70ºF dry-bulb and 60ºF wet-bulb (maximum).

THE FROST-LESS HEAT PUMP CONCEPT

The "frost-less" heat pump concept was derived from the experimental results of the liquid overfeeding (LOF) air-conditioning work (Mei et al. 1996). During an LOF air-conditioner laboratory experiment, the suction pressure observed was higher than that during the baseline operation by 6 to 8 psi, which was caused by the addition of heat to the accumulator from the condensed liquid from condenser. In the same way, if heat were added to the accumulator during the winter heating mode operation (see Figure 1), the outdoor coil temperature should be increased, and frost formation on the coil would then be retarded because of the elevated coil temperature. However, the majority of the heat added to the accumulator will be delivered through more efficient compressor operation with higher suction pressure and mass flow rate to the house in terms of higher supply air temperature. However, it should be noted that "frost-less" is not "frost-free." If the heat pump is operated over a long time at low ambient temperature, frost will accumulate on the outdoor coil even with heat added into the accumulator.

There is a concern that the heat transfer between outdoor coil and ambient air will be reduced because of higher coil temperature for the frost-less operation. When the outdoor coil is clean, the heat transfer between the outdoor coil and ambient air will be less efficient than that of baseline operation. However, once the frost accumulation on the outdoor coil starts to deteriorate the heat transfer process as in the baseline operation, the outdoor coil for a frost-less heat pump is still clean and, thus, the heat transfer will be more efficient in this stage of operation.

Because the heating capacity is higher with the frost-less system, the heat pump should be cycled off before the defrosting cycle would be initiated. It is estimated that the frost-less heat pump will reduce the defrosting cycle by a factor of five in the Knoxville, Tennessee, area.

NEW DEFROSTING METHOD

Cycle reversing is the standard heat pump defrosting method. The heat pump tested was equipped with a standard timer, and refrigerant temperature sensor for defrosting. The time was originally set for 60-minute defrosting. It was reset for 90-minute defrosting for this study. During the defrosting period, heat is taken from indoors to defrost the outdoor coil, which resulted in cold blow even with the indoor resistance heating coil fully energized.

The frost-less concept retards frost accumulation on the outdoor coil and increases the heating capacity, thus poten-

tially reducing the amount of cycle reversing for defrosting. However, when ambient temperature is below 32°F, the heaters to add heat to the accumulator will be shut off in heat pump heating mode operation because adding a modest amount of heat to the accumulator will not retard frost accumulation on the outdoor coil. Eventually, a cycle-reversing defrosting will be needed.

However, during the cycle-reversing defrosting period, the indoor blower will be shut off. Liquid refrigerant passing the indoor expansion device will be two-phase throughout the indoor coil without absorbing heat from the house. The two-phase refrigerant will flow to the accumulator. The heater in the accumulator will be energized during the defrosting period. In other words, the accumulator is taking over the function of the evaporator, where refrigerant is evaporated to provide vapor to the compressor for outdoor coil defrosting.

This design allows cycle-reversing defrosting but without indoor cold air draft and without indoor resistance heaters being energized. The advantages of this defrosting scheme are that (1) the cold blow is eliminated during the defrosting period because the indoor blower is shut off, (2) the defrosting is very efficient because the accumulator is close to the outdoor coil where heat loss is minimum, and heat input to the accumulator will be much less than the 10 kW (in the order of 1 kW) requirement of the indoor resistance heating elements, and (3) the power consumption by the indoor blower motor can be saved.

TEST SETUP

A two-ton split heat pump was selected for the test because it was considered the most popular model in the region. Figure 1 shows the refrigerant-side schematic and Figure 2 shows the accumulator-heater assembly. Figure 3 shows the air-side schematic. The setup is to measure both the air-side and refrigerant-side performance. Pressure transducers and thermocouples were used for pressure and temperature measurements. However, air-side measurements were adopted because refrigerant-side mass flow rate measurements would not be accurate during the transient operating period, such as at the beginning and at the end of the defrosting cycle.

The indoor blower motor was wired in a way that during the cycle-reversing defrosting period for the frost-less operation, the motor would not be energized. Power measurements for the compressor and condenser fan motor, indoor blower motor, and resistance heating elements were recorded separately by power transducers.

From the measured data, the heat pump heating capacity, supply air temperature, power consumption, system COP, etc., could be calculated. Summarization of the test setup is as follows:

Air side:

- Heat pump indoor unit inlet and outlet air dry- and wet-bulb temperatures with thermocouple piles.
- Outdoor dry-bulb and wet-bulb temperatures with thermocouples
- Indoor air volumetric flow rate with a micromanometer.

Refrigerant side:

- Pressure measurements with pressure transducers—compressor inlet and outlet, condenser (indoor coil) outlet, and suction before accumulator.
- Temperature measurements with thermocouples—compressor inlet and outlet, condenser (indoor coil) outlet,

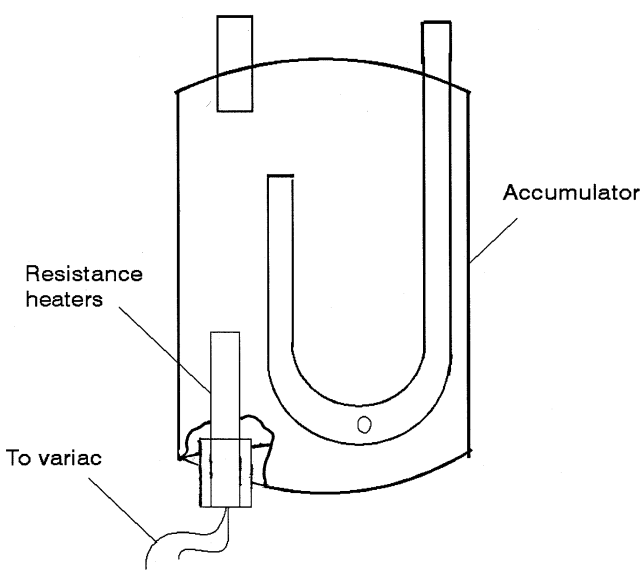


Figure 2 *Accumulator with heaters.*

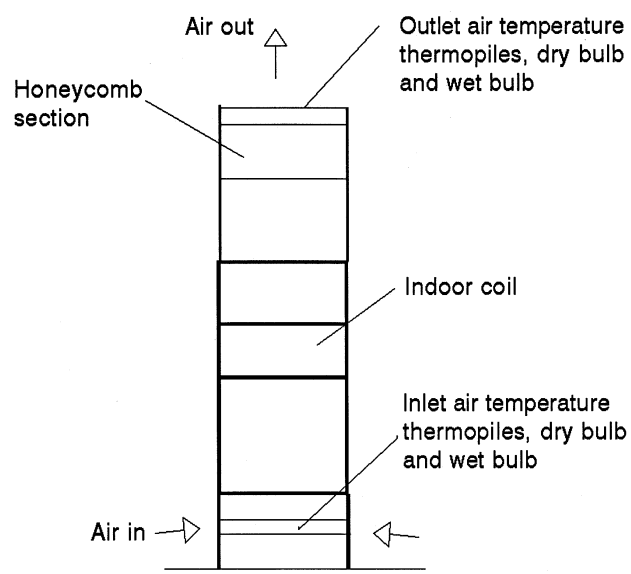
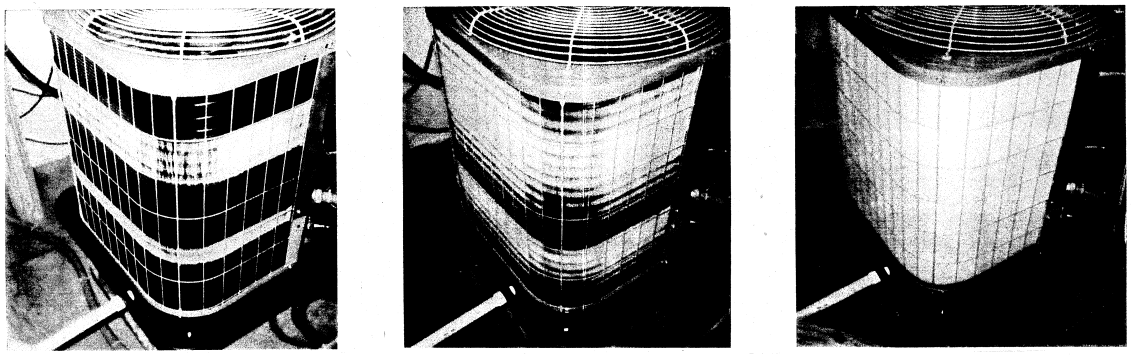


Figure 3 *Air-side schematic.*



(a) 10 minutes (b) 50 minutes (c) 90 minutes

Figure 4 *Baseline operation at 35ºF ambient and 75% RH.*

before expansion device (on outdoor coil), after expansion device, before and after accumulator.

- Refrigerant volumetric flow rate measurement is with a turbinemeter located just before the expansion device (on outdoor coil).

Power:

- Compressor and condenser fan power, indoor blower motor, and resistance heater power inside the accumulator were measured.

TEST PROCEDURES

Frost-Less Operation

In this study, a two-ton cooling heat pump has been tested extensively for baseline and frost-less operation over an outdoor temperature range of 33ºF to 41°F at 75% relative humidity.

Baseline tests were performed first. After going through a frosting and defrosting period at a specified ambient temperature (normally a 90-minute cycle), the heaters in the accumulator were turned on for frost-less operation. The heat pump was operated continuously with the heaters on until frost finally accumulated on the outdoor coil. Tests were performed at simulated outdoor temperatures of 41ºF, 39ºF, 37ºF, 35ºF, and 33°F.

Test of New Defrosting Method

The heat pump was operated with simulated outdoor ambient temperatures below 30°F to allow frost to accumulate on the outdoor coil without energizing the heaters. Once the heat pump called for defrosting after a 90-minute operation, the heat pump cycle was reversed and the heaters in the accumulator were energized and the indoor blower motor was turned off. The two-phase refrigerant flowed past the indoor coil without been evaporated. The accumulator served as the evaporator during the defrosting period. The amount of power input to the accumulator was recorded and the outdoor coil temperatures were measured. The test data were then compared with the baseline defrosting test data.

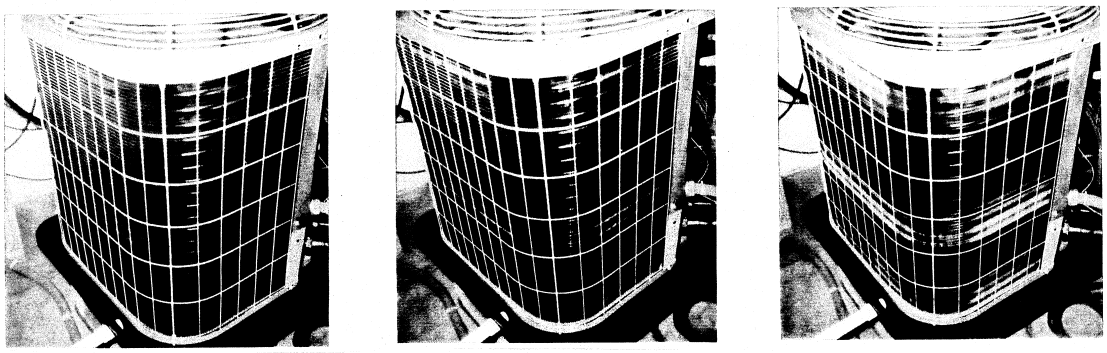
TEST RESULTS AND DISCUSSION

Comparing the test data for the two operations, it is clear that the frost-less concept provides higher average supply air temperature of 4ºF to 6°F over that of baseline. The supply air temperatures by the frost-less concept are also very stable, while the supply air temperature for baseline operation varies over the operating cycle. The indoor thermal comfort is, thus, greatly improved.

Figures 4a through 4c show the coil frost accumulation over a 90-minute operation at 35°F ambient and 75% RH. It is evident that the coil was almost covered with frost at the end of the 90-minute operation. However, when the heaters in the accumulator were energized in the same heat pump (as shown in Figures 5a through 5c), there was little or no frost on the outdoor coil. Figures 6 and 7 show the heat pump supply air temperature and heating capacity comparisons between the baseline and frost-less operations. It is clear that the frost-less operation provided higher and more stable air supply air temperature and heating capacity. The test data proved the validity of the frost-less concept.

As for low ambient temperature defrosting tests, cycle-reversing coil defrosting was performed with the indoor fan off and accumulator heaters energized (1.0 kW). It was found that the coil defrost was just as fast as the conventional cycle reversing defrosting. The new defrosting technology actually saves energy for defrosting.

Tables 1 and 2 show the spreadsheet analysis of the heating seasonal performance in the Knoxville, Tennessee, area. The calculation was based on the temperature bin method (ASHRAE 1983). The temperature bins were from ASHRAE (1989). The tables show that once the energy consumed during the defrosting period is counted, FLHP is actually 9% more efficient to operate than the baseline line system.



(a) 10 minutes (b) 50 minutes (c) 90 minutes

Figure 5 *Frostless operation at 35ºF ambient and 75% RH.*

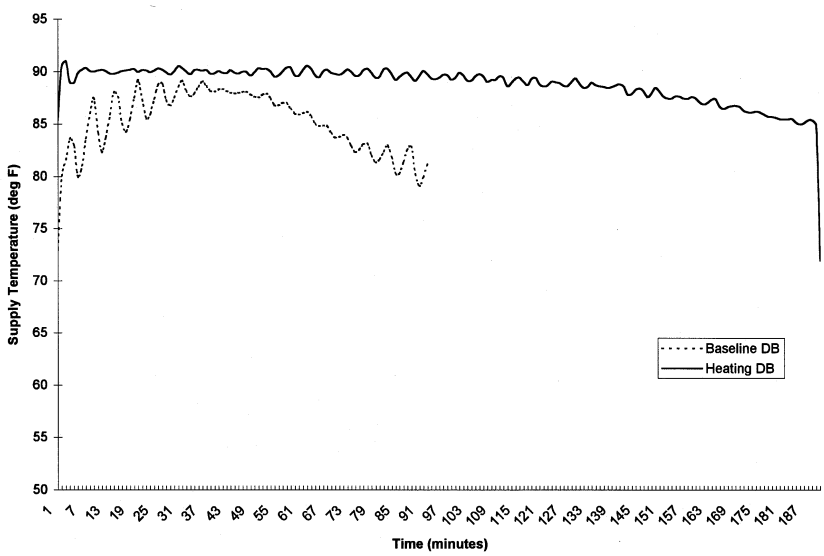


Figure 6 *Supply air temperature comparison between baseline and frostless operation at 35ºF and 75% RH.*

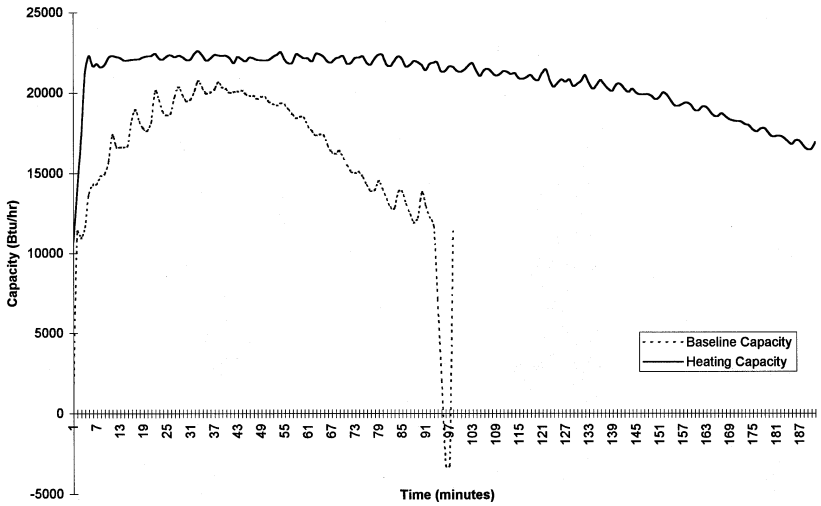


Figure 7 *Capacity comparison between baseline and frostless operation at 35ºF and 75%.*

TABLE 1 *
Baseline Heat Pump

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
ODT BIN	**Seasonal Heating Hours**	**Btu/h Loss/ Deg**	**ODT Diff (65-ODT)**	**Heat Loss, Btu/h**	**HP Cap, Btu/h**	**HP Input, KW**	**HP Load Factor**	**Partial Load Factor**	**Run Time**	**HP, kWh**	**Resist Heat, KW**	**Res. Ht, kWh**	**Btu/h Loss/ Year**	**Defrost kWh/ Year**
62	746	642	3	1925	28305	2.250	0.068	0.767	9%	148.83	0.00	0.00	1436	
57	675	642	8	5133	26500	2.199	0.194	0.798	24%	360.12	0.00	0.00	3465	
52	672	642	13	8342	24752	2.148	0.337	0.834	40%	583.11	0.00	0.00	5606	
47	689	342	18	11550	23060	2.098	0.501	0.875	57%	827.24	0.00	0.00	7958	
42	648	642	23	14758	21432	2.050	0.689	0.922	75%	991.97	0.00	0.00	9563	
37	590	642	28	17967	18546	1.887	0.969	0.992	98%	1087.04	0.00	0.00	10600	655.56
32	456	642	33	21175	18343	1.954	1.000	1.000	100%	891.02	0.83	378.37	9656	506.67
27	217	642	38	24383	16885	1.908	1.000	1.000	100%	414.04	2.20	476.75	5291	241.11
22	101	642	43	27592	15486	1.862	1.000	1.000	100%	188.06	3.55	358.24	2787	112.22
17	41	642	48	30800	14144	1.817	1.000	1.000	100%	74.50	4.88	200.09	1263	45.56
12	21	642	53	34008	12874	1.773	1.000	1.000	100%	37.23	6.19	130.04	714	23.33
7	7	642	58	37217	11635	1.729	1.000	1.000	100%	12.10	7.50	52.47	261	7.78
2	2	642	63	40425	10467	1.687	1.000	1.000	100%	3.37	8.78	17.56	81	2.22
								Totals =		**5619**		**1614**	**58680**	**1594**

Outdoor Design Temperature = 17ºF
Design Heating Load Requirement = 30800 Btu/h
Balance Point Outdoor Temperature = 35ºF

HSPF = Sum N / ((Sum K + Sum M + Sum O))
HSPF = 6.65

* Notes:
1) Baseline HP defrost is assumed a 10 KW resistance heating coil on for 10 minutes. Every 90 minutes, defrost once.
2) Knoxville weather data.

TABLE 2 *
Frostless Heat Pump

A	B	C	D	E	F	G	H	I	J	K	L	M	N	O
ODT BIN	**Seasonal Heating Hours**	**Btu/h Loss/ Deg**	**ODT Diff (65-ODT)**	**Heat Loss, Btu/h**	**HP Cap, Btu/h**	**HP Input, KW**	**HP Load Factor**	**Partial Load Factor**	**Run Time**	**HP, kWh**	**Resist Heat, KW**	**Res. Ht, KWH**	**Btu/h Loss/ Year**	**Defrost kWh/ Year**
62	746	642	3	1925	28305	2.250	0.068	0.767	9%	148.83	0.00	0.00	1436	
57	675	642	8	5133	26500	2.199	0.194	0.798	24%	360.12	0.00	0.00	3465	
52	672	642	13	8342	24752	2.148	0.337	0.834	40%	583.11	0.00	0.00	5606	
47	689	642	18	11550	23060	2.098	0.501	0.875	57%	827.24	0.00	0.00	7958	
42	648	642	23	14758	21432	2.050	0.689	0.922	75%	991.97	0.00	0.00	9563	
37	590	642	28	17967	22742	3.024	0.790	0.948	83%	1487.62	0.00	0.00	10600	
32	456	642	33	21175	16261	1.815	1.000	1.000	100%	827.64	1.44	656.54	9656	50.67
27	217	642	38	24383	13976	1.742	1.000	1.000	100%	378.01	3.05	661.70	5291	24.11
22	101	642	43	27592	15486	1.862	1.000	1.000	100%	188.06	3.55	358.24	2787	11.22
17	41	642	48	30800	14144	1.817	1.000	1.000	100%	74.50	4.88	200.09	1263	4.56
12	21	642	53	34008	12874	1.773	1.000	1.000	100%	37.23	6.19	130.04	714	2.33
7	7	642	58	37217	11635	1.729	1.000	1.000	100%	12.10	7.50	52.47	261	0.78
2	2	642	63	40425	10467	1.687	1.000	1.000	100%	3.37	8.78	17.56	81	0.22
								Totals =		**5920**		**2077**	**58680**	**94**

Outdoor Design Temperature = 17ºF
Design Heating Load Requirement = 30800 Btuh
Balance Point Outdoor Temperature = 35ºF

HSPF = Sum N / ((Sum K + Sum M + Sum O))
HSPF = 7.25

* Notes:
1) Frostless HP defrost is assumed a 1 KW resistance heating coil inside the accumulator on for 10 minutes. Every 90 minutes, defrost once.
2) Knoxville weather data.

CONCLUSIONS

A two-ton heat pump was modified for both baseline and frost-less operation. The new defrosting method was also tested. The test data showed that both concepts worked as expected. The following conclusions can be summarized from the tests:

1. Adding heat to the accumulator during the winter heating season will retard frost accumulation on the outdoor coil. The frost-less concept worked as expected. However, when the ambient temperature drops below 32°F, adding a moderate amount of heat will not prevent frost accumulation on the outdoor coil and, thus, the heat input to the accumulator should be stopped. Heat should be added to the accumulator only when the ambient temperature is between 41ºF and 32°F, where frost is most likely to accumulate on the coil.
2. The majority of heat added to the accumulator to retard frost accumulation would be delivered indoors via higher heat pump supply air temperature because compressor suction pressure was increased. The indoor thermal comfort would be improved because the heat pump supply air temperature was elevated by 4ºF to 6°F. Because frost accumulation on the outdoor coil was severely retarded, the supply air temperature was also very stable for frost-less operation, while the baseline operation showed a big swing of supply air temperature.
3. The new defrosting method for low-ambient temperature operation worked as expected. The moderate amount of heat supplied to the accumulator during the cycle-reversing defrosting period, making the accumulator the de facto evaporator, was enough for outdoor coil defrosting. No heat would be taken from indoors because the indoor blower was off during the defrosting period.
4. If the amount of heat used for defrosting was considered, frost-less operation out performed baseline operation by providing better indoor thermal comfort and higher HSPF.

The frost-less concept is simple, efficient, and can potentially be low-cost. It improves indoor thermal comfort even

during the defrosting period. It can potentially eliminate the need for cycle-reversing defrosting by a factor of five in the Knoxville, Tennessee, region and, thus, can potentially increase the reliability of the heat pump. Because indoor thermal comfort and efficiency have always been the concerns of the heat pump manufacturers, this technology addresses both concerns in a very simple but effective way.

ACKNOWLEDGMENTS

This project was sponsored by the Tennessee Valley Authority. Oak Ridge National Laboratory is managed by UT-Battelle under contract DE-AC05-00OR22725 with the U.S. Department of Energy.

CONVERSION TABLE

1 kJ = 1 Btu/1.055
°C = (°F – 32)/1.8

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