A new heat pump defrost control cycle is described which is shown to dramatically reduce the number of defrost cycles needed by heat pumps in the author's environment. The costs involved in this control cycle implementation if done on a production basis would probably be less than $25 which would be easily recovered in less than one year of operation.

Some Preliminary Information
In the summer of 2007, I installed a Carrier Heat Pump for a client in the Atlanta, Georgia area. This change-out was designed to improve the efficiency of the heating and cooling equipment. The 25HNA9 heat pump was a 48000btu/h unit with SEER rated 19 and a COP in heating mode of about 3 at 27°F and about 4 at 37°F. As an example, this means that for a total input of about 3.3KW (equivalent to about 10,800btuh) at 34°F, the heating system outputs about 40,000btuh into the building.

According to Carrier data, at a temperature of 37°F, the cost per therm of heat (100,000btu) is about 2.5x 3.3KW/H = 8.25KW/H. Here in Georgia, the "all up" cost per KW/H on the residential TOU rate in winter is about 8cents per KW/h. So the cost to move 1 Therm of heat from outside (37°F) to the inside of the house is about 8.25kwh x 8cents per KWH = $0.66 at a nominal outside temperature of 37°F. This compares to about $1.50 per therm as an "all up" cost per therm for Natural Gas (95% efficiency furnace). For fuel oil at about $2.50 per gallon you get about 140,000btu per gallon and with a 90% efficiency Oil Furnace the cost per therm is about $1.98 per therm. Similarly for electric RESISTANCE heat which is often used as supplemental heat for heat pumps, the cost per therm is again about 8cents per KW/H and you get 3413 Btu/h per KW/H. Thus 100,000btuh will cost (100000/3413) x 8c = $2.35 per therm.

In summary, with the SEER =19/COP = 4 (at T=37°F) heat pumps available now AND with a reasonable electric energy cost such as we have here in Georgia, the obvious solution for home heating is a heat pump with whatever backup heat (Nature Gas/Propane/Fuel Oil/Electric) that is best for your area. NOTE: Heat pumps WILL NOT be as efficient in colder northern climates, so you must run a calculation similar to the above to determine costs in your area at various temperatures.

With the above in mind, we installed the heat pump system and an adjunct high efficiency (SEER=17) Air Conditioner and the client enjoyed a summer with about a 25% lower overall electric energy cost compared to that using the older SEER 12 air conditioners. Cheers!
The Carrier 25HNA9 48,000btuh "Ultra" Heat Pump System.
This Heat pump system is a LITTLE unusual. The unit was purchased with the thinking that it would be rather simple to connect it up to the existing furnace and zoned control system. What a surprise when I opened the manual and it said, “This equipment can only be used with Carrier Furnaces and Infinity Thermostats"... After mulling things over for a day, we found that there were no readily available air source heat pump systems with the efficiency of the one we had bought. So... Since I am an Electrical Engineer by trade, we set about to figure out WHY this heat pump could not be used with "ordinary" furnaces. The answer was: A marketing decision was made to integrate the furnace, heat pump, thermostats and zones so that they would "talk" only over a pair of wires. I am sure that somewhere along the way, the marketers figured this was a good way to force anyone wanting Carrier's premium heat pump product to "Go Carrier All The Way". I think that perhaps this is a great marketing strategy for Carrier and it eliminates any potential equipment compatibility problems and allows certain system optimizations as well.

BUT! I wanted to use the 25HNA9048 heat pump with the existing 12 year old stainless firebox 95% efficiency Amana Furnace. The installed Building Automation System is a system designed and manufactured by HI Solutions Inc. The system was designed for use in LARGE commercial buildings but it uses a distributed architecture that makes it quite adaptable (by an HVAC Engineer) and cost effective to use for buildings of any size, including medium to large sized homes. Since this system was already up and running in the home and grounds, we chose the HI Solutions model UUC-8 Universal Controller to REPLACE the Carrier Heat Pump Controller Card inside the heat pump. This meant we had to adapt the basic heat pump to operate with the new UUC-8 controller AND program this controller so it could both control and protect the heat pump and also communicate and interface with the remainder of the building automation system. This process took several days. I was able to procure product literature from Carrier which described in some detail how their controller functioned and I simply wrote a 40 line program for the UUC-8 universal controller (in a language similar to BASIC) to operate and monitor the heat pump equipment.

While I was at it, I added a few instrumentation "niceties" so we could monitor the detailed performance of the heat pump from my remote computers. The UUC-8 controller has 8 Analog Inputs, 12 Switch Closure Inputs, 3 Analog Outputs, and 8 ports with Triac Switchable 24vac at up to 2 amps. I set up the new UUC-8 controller to control the scroll compressor, reversing solenoid, and the High/Low capacity solenoid in the heat pump compressor unit itself. Then we added a "furnace run (enable GAS)" signal, a furnace Fan High/Low signal, an overall FURNACE ENABLE signal, and a Heat Pump/Furnace select signal. The technician built a simple relay tree interface for the furnace end so that any single wire or signal failure could not put the furnace (or the heat pump) in RUN and leave it there. As an added safety feature, a standard Honeywell thermostat was connected in series with the furnace system's gas valve so that if return air ever gets over 80°F, the gas valve in the furnace cannot be turned on. On a "one of a kind" system such as this, MULTIPLE safeguards are an essential ingredient because
"errors happen" and you do not want a small "program error" to overheat your client's house!! One of the analog outputs go to control the GE variable speed fan in the heat pump unit. Analog input sensors monitor Outside Air temperature, Evaporator refrigerant OUTLET temperature, and Suction Line temp on the line coming from the furnace evaporator/condenser. UUC-8 switch closure inputs include Puron Over-Pressure, Puron Under-Pressure, plus an input from the furnace air vane switch which proves furnace air flow. (The compressor is never allowed to run unless air flow in the connected furnace air handler is "proved".). Then a current transformer was added so as to be able to monitor heat pump compressor and outside fan unit current draw. This, along with the other data available in the UUC-8 allows the computation of a good quality estimate of the system SEER, COP, and to allow the program to monitor for over-current, over-temperature, overpressure and various other abnormal conditions as may arise.

What about the DEFROST CYCLE is so interesting?
In November, North Georgia weather turned rather cool. Night low temperatures were going from 27°F to about 45°F which has allowed a good laboratory for getting the heat pump right down to freezing for a few hours and then quickly bringing the temperature into the 50 to 70 range in the daytime. During this interval, I noticed via the remote instrumentation that the heat pump was going into defrost mode in the 3am to 8am time period but we found that the defrosts were producing little water. I was basically using Carrier's defrost algorithm which goes like this: You power up and the system resets the defrost interval to a value of 30 minutes of compressor running time in heating mode. Then, after this 30 minutes expires, the temperature of the evaporator refrigerant outlet line is measured and if it is less than 32°F a defrost cycle is initiated. Then, when a defrost cycle is initiated, if the time it takes to defrost the outside evaporator coil (as evidenced by the outside coil exhaust temperature being greater than 65°F) is less than 3 minutes, the next defrost interval is set to 120 minutes, if between 3 and 5 minutes set the defrost interval to 90 minutes, if between 5 and 7 minutes, set the defrost interval to 60 minutes, and if it takes greater than 7 minutes to defrost the evaporator, set the next defrost interval to 30 minutes. The decision to defrost (or not) is that AFTER the compressor runs for the defrost interval, check the outside coil refrigerant outlet temperature. If it is below 32°F, then defrost, if not, reset the timer and continue operations. This algorithm leads to unneeded defrost cycles where the coil is really NOT iced up to an extent affecting efficiency in a significant way.

We noticed that a good many defrosts (1 to 3 a day) produced little water from the defrost cycle. I was a bit curious as to the "real" amount of ice being developed and so we temporarily disabled the defrost part of the program but left in a "freeze up shutoff" if the coil temperature ever got down to 15°F. The equipment was watched remotely for another week and the outside refrigerant outlet temperature never got lower than 22°F and never differed from the outside air temperature by more than about 8 degrees. (Did I mention that the UUC-8 also has trend logging for analog and digital inputs and most any internal variable you may wish to log for later review? The log comes complete with a date/time stamp.) In this case, we logged what would have been defrosting cycles. There were 19 in a 10 day period but NOT ONE was actually needed. I decided that there had to
be a better way to determine WHEN to defrost so as not to waste the energy used in the defrost cycle. A typical defrost cycle lasts about 4 minutes and pumps "cool" air into the house and warms up the outside evaporator thus causing any ice to melt. Then when the heat pump is reversed back to heating mode, it takes LONGER to pump the heat back into the house than the defrost cycle took. All in all you are "defrosting and getting back where you were" to the tune of maybe 10 minutes or more for each defrost cycle. The energy used is in the range of 3KW (at 33°F) for 1/6 hour or about 4 cents per defrost. This is not a great deal of money per cycle, but you ALSO lose the heat pump capacity for those 10 minutes which amounts to perhaps 7000btu for the average defrost cycle. Note that even more energy is used for the defrost cycle when electric heat is used to compensate for the cooling effect in the home due to the defrost cycle. In fact, a defrost cycle compensated by electric heat will cost about 9 cents for each defrost cycle...

Another reason for wanting to try and optimize the defrost cycle was to try to be able to operate the heat pump at lower temperatures than the 30 or so degrees typical of these systems. While the heat output goes down significantly below 30°F, the heating capacity is still substantial. This 48000btu HP system capacity is approximated by the formula BTU OUTPUT = 1000(0.63T+17). Where T is the outside air temperature in degree F. This equipment is rated to produce HALF its rated capacity at about 15°F. Though this BTU output will likely be inadequate to maintain comfort, even at 0°F, the cost per therm is less using the heat pump than that cost using Natural Gas or any of the other optional fuels. In this setup, we are able to use the 48,000BTU/H Natural Gas furnace simultaneously with the 48,000 BTU/H heat pump as needed since the two furnaces share a common plenum system which feeds the entire home. The furnace serving as air handler for the Heat Pump is a 90,000BTU nature gas (NG) Amana unit.

Now to the discuss the new DEFROST CYCLE implementation
As an example of what the new defrost scheme can do, see Figure 10 (below) where the heat pump was run at temperatures between 20°F and 30°F for 12 hours and required ZERO defrost cycles.

We gave a lot of thought about options for deciding when a defrost was needed. In the end, it seemed to me that "freezing up" would be accompanied by a reduced air flow produced by the heat pump outdoor unit fan. This reduced air flow would be accompanied by a reduced air pressure (vacuum) inside the heat pump fan/evaporator enclosure. We installed a 0-1 inch differential air pressure transducer inside the heat pump enclosure for experiments. The low pressure side connected by rubber tube through a hole to the inside of the heat pump enclosure and the high pressure side was led by a similar tube to a place exterior to the enclosure. Tubes were installed so it was downhill from the transducer to prevent rain from entering the tube and running into the transducer. With this transducer and other measurement tools, we discovered that, over another week, the Carrier algorithm wanted to defrost 14 times and my algorithm wanted to defrost once. During this interval, I allowed NO defrosts and at no time did the evaporator inlet/outlet differential temperature exceed 9°F (except during system startup). The
"ice free" differential air pressure across the coil is nominally 0.11 inches. On just one occasion during the week long test, the differential air pressure drop across the coil got up to 0.4". Based on this (rather sparse) data, I have initially set up the defrost algorithm with a) NO timers, but instead, b) the algorithm is programmed to defrost the system whenever the air pressure drop exceeds 0.37" -or- if the "evaporator input to evaporator output" refrigerant temp differential across the coil exceeds 15°F.

The differential temperature (Outside Air minus Evaporator Coil Outlet Temperature) is allowed to excursion to 25°F for up to 10 minutes following a startup. This transient on startup is normal and does not call for a defrost.

The above defrost scheme (in this moderate weather) is definitely doing a much more frugal job of handling system defrost and without any apparent reduction in efficiency. In fact, due to fewer defrost cycles, the system's overall efficiency is increased by maybe 2 to 3% plus a similar increase in system heating capacity due to fewer defrost cycles.

As the winter gets colder and the data folder gets thicker, I am certain to have refinements to the above scheme... Stay Tuned!

Below is the client's Building Automation System's Main Floor Display. Note the extensive instrumentation of the HVAC equipment in the Top-Upper Right. Special data display for the heat pump experiments include Defrost OFF/ON, delta-T across the liquid inlet to vapor outlet of the outside evaporator, Fan Speed, VACuum (x100) across the outside evaporator, HPbtu/h for given OAT, computed from balance curves furnished by Carrier, Heat Pump Amps, computed instantaneous SEER for heat pump, Heat Pump Ambient(OAT), HP-OCT (HP outside coil outlet temp, Inside HP condenser INPUT air temp (BYPT), HP supply side (heated) air temp, Duct Supply Pressure, Furnace or Heat Pump (Stage), AC and HP high/low capacity commanded and some other related parameters. This display is a touchscreen and settings are controlled by touching the desired parameter (such as OCCupied) for a particular room and turning it on or off. Setting individual room temperatures is handled similarly or by individual room digital thermostats.
FIGURE 1 (above)
A photo of the HI-Solutions model UUC-8 Controller Board is shown below in figure 2. This board was substituted for the OEM Carrier Heat Pump Controller.

The following Figure 3 is not a typical day in the life of a heat pump in heating mode, but it is a great show of how the new defrost algorithm works in real life. Notice that the blue graph in the temperature group is outside air temperature at the Heat Pump and the green is outside humidity. The black graph in the Heat Pump Tons group is heating Tons and the green graph is "stage number" for the heating system. Stage 0 is OFF, Stage 2 is Heat Pump ON and Stage 3 is Heat Pump plus Auxiliary Gas heating. In the Total Amps graph group, the RED graph is the differential pressure across the Outside Evaporator/Condenser (VAC is in units of 100 times the pressure difference in inches of water.), the black graph is the total amps drawn by the Heat Pump unit and the green graph represents the Outside Coil Refrigerant Exit Temperature DIFFERENCE from the Outside Air Temperature.

Notice that the Outside air temperature was below 35°F for much of the night, rising to 40°F only around 2PM. The outside humidity was about 97% during this entire interval. During this time, and beginning about 6AM when the heat pump begin running continuously for about 3.5 hours, the Outside Coil began to "ice up". This is evidenced by the increase in "VAC" from about 0.11 inches at 6:15AM to about 0.25 inches at 9:30AM. During this time, the coil was slowly freezing up and the coil temperature was constantly below freezing. Thus, the Carrier Defrost Algorithm would have defrosted about every hour during this interval. The freezing conditions continued until 2PM when the coil finally thaws out due to the warming temperature. The important thing to note is that at no time during this 10 hours of coil icing was a defrost cycle called by the new defrost algorithm BECAUSE the pressure differential never got up to the trip point of 0.35 inches. The graphs of amps drawn and Supply Air Temperature (TSAT) indicate little if any capacity reduction during this interval. We know that some capacity
reduction occurred because the DELT (Outside Air minus Outside Coil Temperature) slowly rose from about 6°F to 10°F during this interval. However my calculations are that the very small reduction in capacity was MUCH less than the amount of energy which would have been expended and lost by defrosts cycles. Not to mention the fact that whiles the equipment is defrosting, it is not only not delivering HEAT to the building, but is, in fact, delivering COLD into the structure! It is for these reasons that the new defrost algorithm is an important improvement in Heat Pump Operation.

Below is a typical graph of Heat Pump operation with outside air in the range of 35°F with the new algorithm. Note that the machine ran for a full 12 hours in this instance without a single defrost cycle being required. The VAC never got above about 0.13” in this period because the outside humidity (HUMO) was less than 78% for the entire interval. The pressure differential measurement across the outside coil automatically takes this lower humidity into account because the icing amount was less
resulting in less pressure drop through the coil. In the graph below, OAIR is outside air temp, BTUH is computed Heat Pump capacity in the selected Low/High mode, DELT is the outside evaporator coil temperature difference and VAC is the pressure drop across the outside evaporator coil in inches x 100. (VAC=10 indicates 0.10” pressure drop across outside coil.), TAMP is the total amps drawn by the fan and compressor in the heat pump unit. Note: A defrost cycle occurred when TAMP shows the compressor is running and VAC is zero showing the outside fan is NOT running as in figure 5 at about 8:50AM.
In the figure 4, note that over this 12 hour period where the outside coil temperature was frequently below freezing for short periods (and the Carrier algorithm would have run about 4 defrost cycles), the new algorithm did not call for a defrost even once. Note also that the pressure drop never got above about 0.12" during the period and there was no drop off in TAMPs such as you see with the capacity reduction which accompanies a freeze up situation. This was a situation where there was a) medium humidity, and b) the normal heat pump shutoffs at frequent intervals allowed some defrosting to occur at intervals without running a machine defrost cycle.

Another phenomena demonstrated by the above graph is the value of good home insulation. Note that the Outside Air Temperature (OAIR) was down to about 37°F at 23:00hours. But the system continued its cycling on/off at a pretty steady duty cycle until about 04:00 when you notice the first transition of BTUH above the LOW capacity during the night. Starting at 06:00 the system goes to high capacity and stays there for 1.5 hours. Then at 08:00, the family room is "turned on" and the system then runs in HIGH capacity for several more hours. During this sequence, I locked out the auxiliary gas heat so the overall effectiveness of the 48,000btu heat pump in the home (all rooms set to occupied) could be evaluated. As shown in the graph, the actual heat pump output at 35°F outside was in the range of 37,000btu/h during the night.

In figure 5 below, Notice that there was obviously increasing icing of the outside coil from about 6AM. However the icing only got the pressure drop across the outside coil to 0.35" (red VAC graph line) at about 8:45AM and so there was only one defrost cycle during the entire night with the temperature below 40°F for the entire period. The OEM carrier defrost algorithm would have caused a defrost cycle about four times. It is interesting to note the DELT (green) curve in the lower curves of figures 5 and 6. It can be seen that when significant icing is occurring, the value of DELT stays POSITIVE during the normal heat pump compressor OFF cycle. But if there is no significant icing, the value of DELT reverses and goes negative during the OFF cycle. This results because with little or no icing, the small amount of heat left contained in remnants of the refrigerant in the inside condenser coil moves outside to the colder evaporator coil and is enough to heat up the outside coil to above outside ambient temperatures between cycles unless significant ice is present.
The image in figure 6 below shows another night when the temperature hovered around freezing. Note that from about 5AM the temperature was around 35°F and there was a lot of running of the heat pump from 6AM until 9AM. Yet, not a single defrost cycle was requested by the new algorithm and none was needed. The OEM Carrier Algorithm called for 3 defrost cycles during this interval. Note the multiple momentary power failures at about 5:20AM. These had no effect on the operation of the new algorithm. But the OEM Carrier Algorithm would have reset the defrost timer to 30 minutes as a result of the power fail and this would have perhaps generated a fourth defrost cycle in the Carrier OEM defrost algorithm for this period.

FIGURE 6 (above)
The figure below shows a sustained period when the outside temperature was between 33°F and 38°F. The humidity was in the range of 70% so there was not as much icing as you might expect in that temperature range. However, between 6AM and 8AM we do see the red VAC curve arch up to a maximum pressure drop of about 15 (0.15"). The graph shows that on this day there was no defrost cycle called for. The OEM carrier defrost cycle algorithm would have called for a defrost 2 or 3 times during this interval. The BTUH curve shows that the system was alternating between the low and high output modes in the 6AM to 8AM period finally going to high output mode continuously at about 8:30AM. At about 8:30am, there is a small "pip" on the STGW up to stage 3 and BTUH graphs indicating that, for a few minutes, auxiliary gas heat was turned on.

---

**FIGURE 7 (above)**
In Figure 8 (below) is shown another near freezing night without even one defrost cycle being called. The OEM algorithm would have called for 2 or 3 defrost cycles during this interval. Note that the upper group of graphs has been changed to include Outside Humidity, Supply Air Temperature as well as Outside Air Temperature. Again, note the multiple changes from low to high heat pump capacity as shown in the BTUH and TAMP graphs. The STGW graph should be ignored as my data collection equation was not operating properly.

FIGURE 8 (above)
In Figure 9 (below) we see a night when the temperature went to freezing at about 4AM and stayed there until about 9AM. Only one defrost cycle was called for and that was at about 9:10AM when the air pressure drop across the outside coil finally made it to 0.35". By comparison, the OEM Carrier algorithm called for defrosts three times.

Figure 10, below, shows Heat Pump operation on a day when the minimum temperature was 20°F and the maximum was 28°F. Note that the humidity on this day was below 80% and that NO DEFROST CYCLES were required or run. During the night,
the heat pump mostly ran in "low" heating mode only going to "high" heat mode after 6AM as the living quarters zones were turned on. Note also that the heat pump ran continuously for more than 5 1/2 hours from 6:30AM without any appreciable VAC pressure drop across the outside evaporator and was producing between 2.5 and 3.0 tons (30,000 to 36,000BTU/h) of heating during this period. The standard carrier algorithm a) would have defrosted probably 4 times during this interval and would not have allowed operation below about 32°F. This is a dramatic showing of the improvement in operation of the new heat pump defrost control scheme. Note also the green curve of STGW. This shows that the auxiliary heat (STGW=3) came on for only two short periods during the night but it did run pretty continuously from 7:45am through 10:15am adding 45,000btuh to the heat pump's 30,000+ BTU/h. Note also that the 90,000BTU/h supplemental furnace (STGW=4) never came on line.

FIGURE 10 (above)
Figure 11 (below) shows a full day of operation of the heat pump system with the temperatures ranging from about 13°F up to a high of 40°F. Note that from the time the temperature went below 20°F the system switched over to the backup (natural gas) furnaces. But except for that period from about 2AM until 10AM the heat pump was running and almost continuously and without any sign of a freeze-up as shown by the VAC reading staying in the normal range of just about .11" to .13". The evidence is becoming pretty strong that whenever the humidity is below about 80% or so, defrost cycles are seldom needed. Even above 80% RH, the number of defrost cycles used by the Time + Temperature algorithm vs. the Pressure Difference used here is about 5 to 1. This is more (and pretty conclusive) evidence that the time + temperature defrost algorithm used by Carrier and most other manufacturers is causing MANY MANY wasted defrost cycles. This graph Figure 11 also demonstrates that the heat pump provides useful heat output of about 2.5 tons (30,000btu) all the way down to 20°F. Unfortunately, in this case, the amount of heat required to maintain "balance" at 20°F is about 80,000BTU/H and so at 20°F the switch had to be made automatically from the 30,000 BTU/H Heat Pump + 48,000 BTU/H NG to the 90,000BTU/H NG + the 48,000BTUH NG.
Figure 12 (below) shows another interesting characteristic of the icing of the outside evaporator. This scatter plot has data on Outside Humidity as the Horizontal Axis and Intensity of occurrences of VACuum measurements (differential pressure across the outside air coil) on the vertical axis. This data was accumulated over the period from mid November to 5 January 2008. Notice that ALL of the significant icing (as indicated by red dots above VAC= 12) of the outside heat pump coil has occurred when the outside HUMIDITY was greater than 85%. Below this humidity level, no significant icing occurred at any outside air temperature between 20°F and 40°F. During this period, the heat pump was run for up to 6 hours continuously in temperatures from 20°F to 40°F. I consider this factor very significant as it is another measurable parameter that could be used in a decision process as to when defrosting of the Heat Pump outside coil is necessary. This is vivid evidence that the current time + temperature algorithms are causing the World's Heat Pumps to enter the defrost cycle FAR too frequently.

Questions? Contact joe@mehaffey.us

FIGURE 12 (above)