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EXPERIMENTAL COMPARISON OF CONTINUOUS VS. PULSED FLOW MODULATION IN VAPOR COMPRESSION SYSTEMS

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ABSTRACT

Continuous capacity modulation, for example by varying compressor speed, is only one alternative to conventional cycling of vapor compression systems. Cycling the compressor at higher frequencies (~0.01–0.1 Hz) than conventional systems (~0.001 Hz) offers a potentially simpler and less costly alternative. Experiments were conducted to quantify effects of steady vs. pulsed flow on the performance of other components of the system. Rapid-cycling results at a runtime fraction of 0.56 were compared to continuous (simulating variable speed) operation of a smaller compressor providing the same cooling capacity. Rapid-cycling degraded the performance of other components enough to reduce COP by 2.5-7.5% at cycle periods of 10-80 sec, respectively. The corresponding COP loss for conventional cycling was 11.5%, all on the condenser side, because evaporator surface temperature was held constant to ensure that sensible and latent capacities were comparable.

NOMENCLATURE

A	Area, m^2	ΔT_{sat}	Temperature lift, °C
c	Specific heat, kJ/kg-K	ΔP	Pressure drop, kPa
C	Thermal capacitance, kJ/K	h	Heat transfer coefficient, kW/m^2-K
COP	Coefficient of performance	hA	Product of h and corresponding area, kW/K
ΔT	Temperature difference, °C	μ	Runtime fraction
ΔT_c	Condenser side degradation term, °C	Q	Heat exchanger capacity, kW
ΔT_e	Evaporator side degradation term, °C	t	Time, s
ΔT_{loss}	Lift loss term, °C	τ	Cycle period, s

Subscripts and abbreviations

air	Air	off	Off-cycle
avg	Average	on	On-cycle
c	Condenser	ref	Refrigerant
e	Evaporator	rht	Refrigerant heat transfer
m	Metal	sat	Saturation
mid	Middle	SC	Short-cycling
nonlin	Nonlinearity	VS	Variable speed

INTRODUCTION

There is a need for capacity regulation in any air-conditioning system in order to maintain specified room comfort as sensible and latent loads change. Most residential air-conditioning systems in North America are designed to meet the maximum heat load, sized to operate continuously at a specified design condition. During its lifetime an a/c system seldom operates at design conditions. To meet actual loads, less than the maximum, they operate in cycles of on and off operation, where the cycle length (~20 min) is determined by a deadband thermostat controlling indoor room temperature. In this conventional cycling (CC) type of capacity regulation, both evaporator capacity and compressor power have maximum values during a large portion of on-cycle and are zero during a large portion of off-cycle. The heat exchangers (evaporator and condenser) are practically inactive in off-cycle, which causes very high pressure lift during the large portion of the on-cycle and thus low efficiency. Refrigerant migration and component thermal masses have been identified as the important contributors to cycling losses in the work of Coulter and Bullard (1997), Krause and Bullard (1996), Wang and Wu (1990), Mulroy and Didion (1985), Murphy and Goldschmidt (1986).

One alternative to conventional cycling is variable speed (VS) capacity regulation, in which the compressor runs continuously, but at lower speed. The room thermostat is connected to the compressor controller such that when the heat load increases (indoor temperature increases) compressor speed increases maintaining constant indoor temperature. The surface area of heat exchangers is used during the whole period of continuous operation, reducing pressure lift and achieving high cycle efficiencies. The findings of Bahel and Zubair (1989), Tassou et al. (1983), Marquand et al. (1984), Umezu and Suma (1984) show that VS systems can achieve seasonal energy savings of 15% to 40%. The cost is high due to a need for inverter for changing compressor speed.

Another way of regulating system capacity is by compressor short-cycling (SC). The study on short-cycling capacity regulation method has not yet been published in literature. Short-cycling is characterized by short cycles of compressor on and off operation having duration on the order of 5-20 seconds. The room thermostat would regulate the runtime fraction of short-cycling just as it regulates compressor speed in VS method. Thus, when the heat load increases (indoor temperature increases) compressor runtime fraction increases maintaining constant indoor temperature. Heat exchangers are used during the whole period of short-cycling operation, reducing pressure lift during the on-cycle and ideally achieving the cycle efficiency of VS capacity regulation. The initial cost of using short-cycling capacity regulation method is potentially lower, since there is no need for inverter.

The purpose of this paper is to experimentally compare short-cycling and continuous operation of the a/c system. After briefly explaining the factors causing SC efficiency losses: refrigerant side heat transfer resistance, pressure drop and heat exchanger thermal capacitance, an experimental comparison between short-cycling and continuous operation will be presented. The system performance at μ =0.56 and τ=10-80 sec. cycle periods, will be compared to continuous (simulating variable speed) operation of a smaller compressor providing the same cooling capacity. The results for both regulation methods will be compared to conventional cycling regulation method.

EXPERIMENTAL FACILITY AND PROCEDURE

Experimental Facility

The heat exchangers (condenser and evaporator) of the residential air conditioning system tested were taken from a two-ton, R-22, unitary rooftop air conditioning system (Trane model TCH024100A). Both are made of copper tubing and plate fins, and were placed in separate environmental chambers. The heat exchangers were not modified from the original unitary a/c design in any way, but compressor, piping and expansion device were replaced. A two-ton hermetic scroll compressor (Copeland model ZR22K3-TF5) was fitted with a solenoid valve to enable more precise measurement and system control during the experiments. The valve ensured that no refrigerant vapor leaked back to the evaporator during the off-cycle. The compressor was connected to a switch and timer that controlled lengths of on and off compressor operation during short-cycling. The existing commercial implementation of this concept has a clutch mechanism installed in the compressor for the purpose of engaging and disengaging scrolls, enabling power efficient short-cycling without high starting currents and potential reliability risks which exist when a switch is used. A switch was used in our experiment since our focus has been on the influence of short-cycling on the rest of the system, not the compressor. To facilitate application to normal operation the high starting currents were excluded from the analyses presented in this paper. The measured start-up power peak lasted about 0.2 seconds and accounted for about 0.5-10 % of the cycle-average measured power, for the longest (56 sec) and shortest (3 sec) on-cycles, respectively. With our apparatus, it was not possible to vary the speed of our compressor, so continuous operation was achieved by installing a nominal one-ton rotary compressor (Kenmore a/c unit, model 253.8783110, compressor part# A445513).

The system was designed to enable liquid flow into the expansion device during the whole off-cycle, by adding a receiver holding enough liquid refrigerant to last during off-cycles of up to 40 seconds. A manually adjusted expansion device replaced the original TXV to enable more precise control of the experiments by eliminating the uncontrollable lags inherent in TXV's. Following Wang and Wu (1990) a solenoid valve was installed upstream of the expansion device to synchronize valve closing with compressor shut-down and to isolate evaporator from condenser during off-cycle.

Temperatures, pressures, powers and flow rates were measured throughout. Air side and surface temperatures were measured by T-type thermocouples, and the refrigerant side temperatures were measured by T-type immersion thermocouple probes. Thermocouples were adjusted for tare values to provide the same output at the room temperature. Thermocouple measurement uncertainty is estimated to be +-0.1^oC, using manufacturer's specifications. A low range, 0 to 1" water (+/-0.4% FS), differential air pressure transducer measured the air side pressure drop across the nozzle, and for calculating the evaporator air flow rate. The refrigerant pressures and pressure drops were measured by using absolute 0 to 500 psia (+/-0.1 % FS) and differential 0 to 50 psid (+/-0.25% FS) pressure transducers. Watt transducers 0 to 8 kW (+/-0.5% FS) measured the compressor, blower and heater powers. Coriolis-type 0 to 680 kg/h (+/-0.15% of reading) mass flow meter was used for measuring the refrigerant mass flow rate.

Test Procedure

Experiments were performed with dry indoor coil, at 26.7^oC indoor air, and 35^oC outdoor air temperatures. They were repeated at 50% indoor humidity, but the conclusions did not change. The results at other experimental conditions and for wet coil tests are presented in Ilic et al. (2001). The design cooling capacity of the a/c system with two-ton compressor at μ=1 was 7.0 kW. The two-ton compressor was short-cycled at μ=0.56 to match exactly the capacity achieved during the continuous compressor operation of smaller one-ton compressor. These short-cycling experiments can therefore be compared to continuous operation. The following analysis compares short-cycling experiments done at μ=0.56, and at τ=10-80 seconds, to continuous operation of nominal one-ton compressor to meet a load of about 4.0 kW.

The main method of regulating the evaporator capacity in short-cycling systems is to vary μ . The evaporator design air flow rate was 0.378 m^3/s. The air flow rate over the condenser was kept constant for all experiments at 1.180 m^3/s. The air flow rate over evaporator was decreased to about 0.21 m^3/s along with the decrease in μ from 1 to 0.56, in order to maintain the evaporator surface temperature, $T_{m,e}$, at about 12^oC, as would be required for dehumidification during actual wet coil operation. The air flow rate over evaporator and condenser during continuous operation of smaller compressor was the same as during the SC operation at μ=0.56.

An EEV was adjusted manually to achieve the desired fixed opening for each operating condition. A fixed opening proved capable of providing the desired average on-cycle superheat at the evaporator exit (of about 4^oC) for experimental points at τ~10 sec. However, for τ~60 sec, the fixed opening caused large fluctuations of superheat (1-15^oC), largest at the end of on-cycle, causing significant degradation of evaporator capacity. A fast-responding EEV, which would maintain constant superheat at the exit of evaporator during whole cycle was not available. Therefore to ensure that evaporator capacity was repeatable, an expansion device opening was found which enabled maintenance of the two-phase refrigerant evaporator exit throughout the whole on-cycle for each experimental point. This eliminated evaporator capacity degradation. To protect compressor from liquid slugging a tape heater was installed on the suction line. In a real system, a fast-responding EEV or suction line heat exchanger would serve this purpose with no additional energy input.

RESULTS AND DISCUSSION

Factors affecting short-cycling performance

System performance impacts due to refrigerant flow modulation occur solely on the refrigerant side. It is therefore possible to define an "ideal" baseline system as a system operating – continuously – between heat source and sink temperatures defined by the tube temperatures of the real heat exchangers, which have finite size and finite airflow rate. Any method of capacity regulation: conventional cycling, variable-speed, or short-cycling, can be compared to this ideal.

Viewed from the air side, the requirement for sensible and latent cooling capacities to be equal for all three systems implies that the cycle-averaged evaporator surface temperatures must also be equal. The numerator of COP is therefore constant, so the amount of degradation depends solely on the power, which in turn is a function of on-cycle temperature lift. Therefore to compare short-cycling system efficiencies with those of ideal system, and with other systems, the increase in temperature lift was monitored. In the evaporator and condenser, the increase is equal to the difference between the whole-cycle mean tube temperature and the on-cycle mean saturation temperature, $\Delta T = | T_m - T_{sat,on} |$, where $T_{sat,on}$ is measured at compressor inlet on the evaporator side, and compressor exit on the condenser side. Detailed analysis of ΔT_e, by Ilic et al. (2001), identified the three most important factors affecting COP as capacity is modulated by short-cycling: refrigerant side heat transfer resistance, pressure drop and thermal capacitance influencing the nonlinearity of a heat exchanger metal temperature oscillations.

Refrigerant side heat transfer

The existence of a finite refrigerant side ΔT is one factor increasing the on-cycle temperature lift, and thereby degrading COP relative to the ideal case where $\Delta T_{ref} = 0$. Therefore, the on-cycle refrigerant side temperature difference, $\Delta T_{ref} = | T_{m,on} - T_{sat,on,mid} |$, is one factor contributing to degradation of temperature lift ($T_{sat,on,mid}$ is measured in the middle of the heat exchanger). In both heat exchangers its magnitude differs among the various flow modulation schemes, because $\Delta T_{ref} > 0$ in all cases.

Pressure drop

Refrigerant pressure drop increases temperature lift by $\Delta T_{sat}(\Delta P_{ref})$. Since pressure drop in a crossflow heat exchanger also increases capacity, the net effect of pressure drop on system temperature lift is $\Delta T_{sat}(\Delta P_{ref}/2) = | T_{sat,on} - T_{sat,on,mid} |$ for both evaporator and condenser. This term was expected to be almost independent of τ, due to the almost constant on-cycle refrigerant mass flux resulting from the constant on-cycle compressor speed.

Thermal capacitance

Although the air side of the heat exchangers is used during the whole cycle period, there are SC losses due to exponential (nonlinear) heat exchanger metal temperature oscillations. Due to the inherently exponential nature of the heat transfer process, the evaporator metal temperature during the off-cycle approaches the air temperature nonlinearly. The same phenomenon occurs during the on-cycle as the evaporator surface temperature is pulled down towards the steady-state refrigerant saturation temperature. As a result, the on-cycle average evaporator metal temperature is always lower than the whole-cycle average, and this difference, $T_{m,e}-T_{m,e,on}$, was defined as the evaporator temperature lift degradation due to nonlinearity. Short-cycling truncates these processes, but the fundamental asymmetry remains. Since the compressor sees only the lower evaporator on-cycle metal temperature, there is a penalty in pressure lift for long τ where the nonlinearities are the greatest. The curvature of the evaporator surface temperature oscillation depends on its time constant (C_m/hA_{air}), and the magnitude of oscillation is inversely proportional to the evaporator's thermal capacitance, $C_m=m*c$. The same temperature lift degradation term is defined on the condenser side, since the condenser on-cycle average metal temperature is always higher than the whole-cycle average. This nonlinearity degradation term does not exist for continuous operation where on-cycle average metal temperature equals the whole-cycle average.

Short-cycling losses at μ=0.56 compared to continuous operation

Figure 1 shows how temperature lift increases with τ, causing the COP decrease shown in Figure 2. The difference between T_m and $T_{sat,on}$ was monitored in the evaporator and condenser. The larger those differences, the greater the COP degradation relative to the ideal baseline operation at a given capacity. Evaporator and condenser temperature lift degradation terms (ΔT_e and ΔT_c) were compared at τ=10, 30, 50, 65 and 80 seconds.

For continuous operation, the temperature lift degradation relative to ideal baseline operation is 2.0°C on evaporator and 2.1°C on condenser side (as can be seen in Figures 1, 3 and 4). In addition to degradation from the ideal, we analyze the "cycling loss", ΔT_{loss}, relative to continuous operation; it is defined as an incremental increase in temperature lift attributable to short-cycling relative to variable-speed operation, where continuous operation of the smaller compressor simulates ideal VS operation without inverter losses. This increment is defined as the difference between the short-cycling and VS lift degradation terms, $\Delta T_{loss} = \Delta T_{sc}(\tau) - \Delta T_{vs}(\tau=0)$. ΔT_{loss} can be seen in Figures 3 and 4 increasing the lift by 0.7°C on evaporator and 0.6°C on condenser side as τ increases from 0 to 10 seconds. Compared to continuous operation, SC at this relatively high frequency reduces COP by about 2.5% as shown in Figure 2. As cycling period increased to 80 seconds, the COP declined about 7.5% due to the "cycling loss" of 2.3°C on evaporator and 2.1°C on condenser side shown in Figures 3 and 4.

In order to compare cycle COP's, the effect of potentially different compressor efficiencies was eliminated by calculating the smaller compressor's steady-state specific power using steady-state compressor calorimeter data provided by the manufacturer of the larger scroll compressor. The experimental points at τ=0 represent continuous operation.

Causes of lift degradation for τ=0, 10 and 80 seconds

Figures 5 and 6 quantify the factors affecting performance loss relative to the ideal baseline in the evaporator and condenser respectively. Figure 7 illustrates how these degradation terms are calculated for the evaporator at τ=80 seconds, where $\Delta T_e(nonlin)=T_{m,e}-T_{m,e,on}$, $\Delta T_e(rht)=T_{m,e,on}-T_{sat,e,on,mid}$, and $\Delta T_e(\Delta P)=T_{sat,e,on,mid}-T_{sat,e,on}$. The terms are defined so that their sum is equal to $\Delta T_e=T_{m,e}-T_{sat,e,on}$.

Figures 8 and 9 show the detailed breakdown of the lift "cycling loss" term at τ=10 and τ=80 seconds on condenser and evaporator side. It is by definition zero for continuous operation. Recall from Figure 2 that this additional temperature "lift loss" caused COP losses of about 2.5% and 7.5% at τ=10 and τ=80, respectively, compared to continuous operation.

Figures 8 and 9 also show that cycling losses at τ=10 seconds are caused primarily by the refrigerant side parameters, refrigerant side heat transfer resistance and pressure drop, while the effect of thermal capacitance is small due to the shortness of the on-cycle.

Compared to continuous operation the short-cycling refrigerant side ΔT (difference between the on-cycle average metal and mean saturation temperatures) is 0.4^oC and 0.1^oC greater, respectively, for evaporator and condenser. This loss occurred because the low off-cycle refrigerant side heat transfer rate required a higher on-cycle ΔT_{ref} to achieve the same capacity. This loss was only partially offset by the reduction in ΔT_{ref} due to higher on-cycle refrigerant side heat transfer coefficient with a higher refrigerant mass flux.

An additional loss occurred because the higher refrigerant mass flux increased pressure drop as well. Pressure drop increased temperature lift by 0.3^oC on the evaporator side and 0.2^oC on the condenser side, compared to the case of continuous operation. The effect of thermal capacitance (nonlinearity of temperature oscillations) is by definition zero for continuous operation at τ=0. It is negligible on the evaporator side for short-cycling operation at 10 seconds, and is 0.3^oC on the condenser side. However it is significant for longer τ as discussed below.

For τ=80 seconds, Figures 8 and 9 show that cycling losses during the longer cycles are dominated by refrigerant side heat transfer resistance and the nonlinearity of heat exchanger surface temperature oscillations. As expected, short-cycling losses due to pressure drop relative to continuous operation are essentially independent of cycle period because the mass flux is the same for all cycling conditions.

The losses due to refrigerant side heat transfer were about equal at τ=10 seconds in the evaporator and condenser. However at τ=80 seconds, the on-cycle average heat transfer resistance in the evaporator increased much more than in the condenser. One possibility is that off-cycle evaporation persists for short time after compressor shut-off, which may have a significant effect at τ=10 but is clearly negligible for τ=80 seconds as shown in Figure 7. A more reasonable hypothesis is that dry-out occurs over most of the evaporator during the longer off-cycles, in which also contributes to a lower refrigerant heat transfer rate during the first part of the on-cycle. Such an effect would not be expected in the condenser, where the walls would always be wet during the on and off-cycles, and even the normally superheated zone would be condensing during the off-cycle. Figure 9 shows only a small increase in condenser side ΔT, from 0.1^oC at τ=10 to 0.3^oC at τ=80 seconds, consistent with this hypothesis.

As expected, the effect of heat exchanger thermal capacitance becomes dominant as $\tau\rightarrow$80 seconds. The on-cycle average evaporator surface temperature was 0.7^oC lower than the cycle-average (Figure 8), and the loss due to effect of thermal capacitance was 1.6^oC in the condenser (Figure 9). It is expected that the temperature lift penalty due to nonlinearity would be greatest on the condenser side. Ilic et al. (2001) noted that the increased magnitude and curvature of metal temperature oscillations are the causes of increased nonlinearity. The magnitude of metal temperature oscillations is proportional to Q/C_m and curvature is related to exponential rate quotient, hA_{air}/C_m. The thermal capacitance, C_m, of the condenser was about equal to that of the evaporator, but its 25% higher capacity, Q, increased the magnitude of metal temperature oscillations by the same percentage. The condenser side hA_{air} was about 4 times greater than the evaporator side, so the correspondingly higher exponential rate quotient hA_{air}/C_m increased metal temperature curvature, which increased nonlinearity. However this temperature lift penalty was more than offset by the beneficial effect of this quadrupled hA_{air}, which decreased condenser saturation temperature, thus temperature lift, substantially (order of magnitude higher beneficial effect). Therefore, maximization of hA_{air} on condenser side is beneficial because it decreases temperature lift by minimizing air side ΔT, even though it slightly increases nonlinearity of T_m oscillations.

In summary, the refrigerant side heat transfer and pressure drop are main contributors to short-cycling temperature lift cycling losses at τ=10 seconds causing about 2.5% COP loss. The metal temperature nonlinearity and refrigerant side heat transfer are primary contributors to short-cycling temperature lift cycling losses at τ=80 seconds, causing the largest part of 7.5% COP loss.

CONCLUSIONS

As a point of reference, Figure 10 shows an 11.5% COP difference (at τ=0) between continuous operation of the small 1-ton compressor, simulating the variable-speed case, and the nominal 2-ton compressor, simulating conventional cycling. It also shows how short-cycling reduced this loss to 2.5% and 7.5% at cycle periods of 10 seconds and 80 seconds, respectively. Of course this comparison is approximate, since it neglects cycling losses in the CC case, and difference between blower power requirements, both of which should favor SC over CC.

The relatively small cycle COP losses of about 2.5 % at τ=10 seconds, compared to continuous operation, were caused primarily by higher refrigerant side heat transfer resistance and pressure drop. At longer τ, short-cycling losses of about 7.5% are dominated by refrigerant side heat transfer resistance, and by the nonlinearity of heat exchanger surface temperature oscillations. The experiments showed that the evaporator refrigerant heat transfer resistance part of COP degradation term increased significantly at longer τ. This may have been due to tube dry-out over most of the evaporator during the longer off-cycles resulting in lower heat transfer rate during the first part of the on-cycle. As expected, ΔP losses, relative to variable-speed, were essentially independent of τ because the mass flux did not change. The effect of thermal capacitance, metal temperature nonlinearity degradation term, becomes significant for evaporator at long τ (τ=80 seconds), and earlier for the condenser.

The short-cycling loss of 2.5% and 7.5% at τ=10 and 80 seconds, respectively, can be compared to the maximum 11.5% COP loss that would occur for large τ, corresponding to conventional cycling operation. The 11.5% COP loss difference between two compressors (continuous operation of the small 1-ton simulating the variable-speed, and the nominal 2-ton, simulating conventional cycling without cycling losses) is due mainly to the savings on condenser side, since the requirement of keeping the constant latent/sensible ratio of ~25/75% required maintenance of a constant evaporator surface temperature, in all comparisons presented here.

ACKNOWLEDGEMENTS

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FIGURES

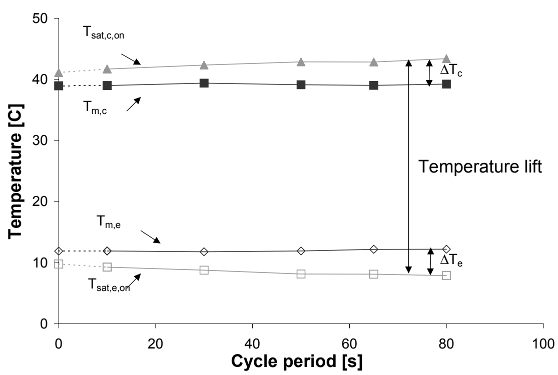

Figure 1. Saturation temperature lift increase with τ

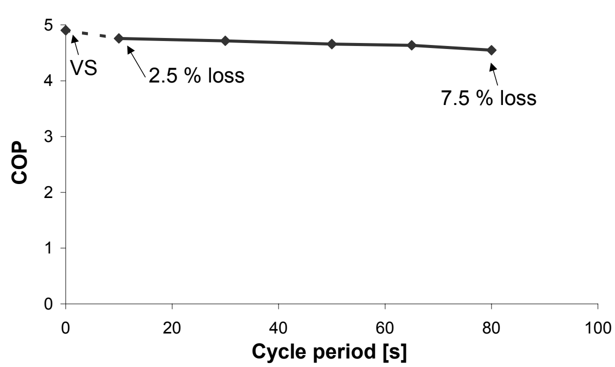

Figure 2. Coefficient of performance decrease with τ

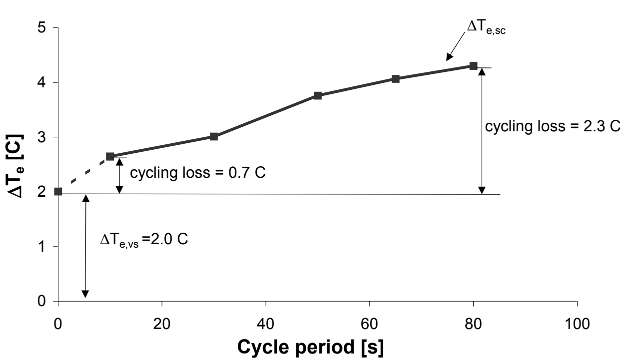

Figure 3. Evaporator side lift degradation term

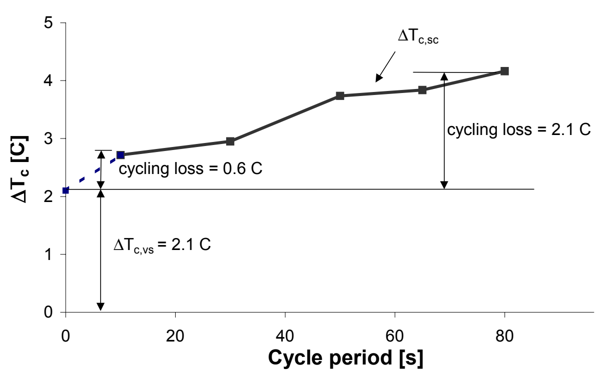

Figure 4. Condenser side lift degradation term

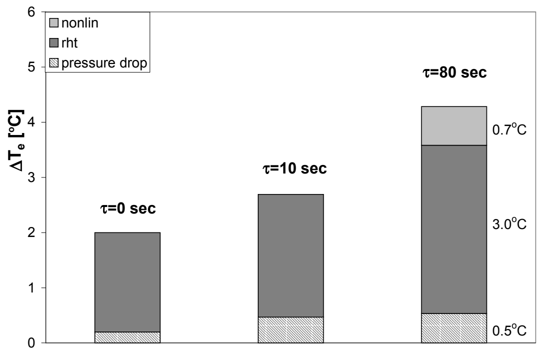

Figure 5. Evaporator lift degradation term breakdown

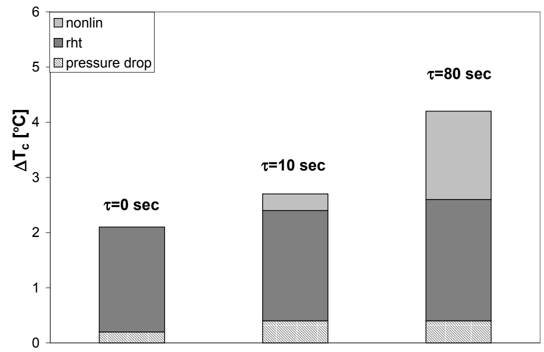

Figure 6. Condenser lift degradation term breakdown

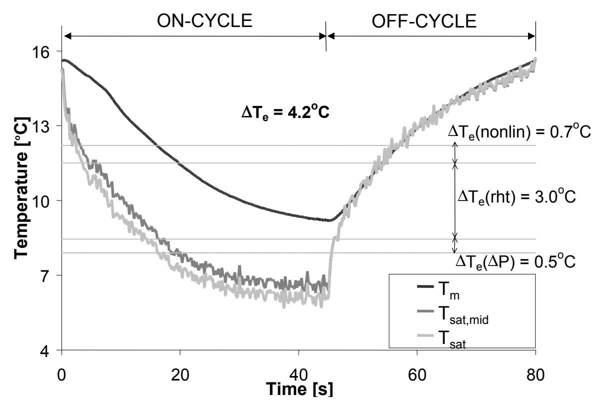

Figure 7. Evaporator degradation term for SC (μ=0.56)

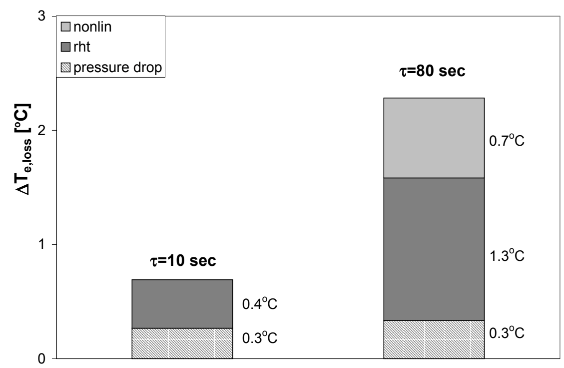

Figure 8. Evaporator cycling loss term breakdown

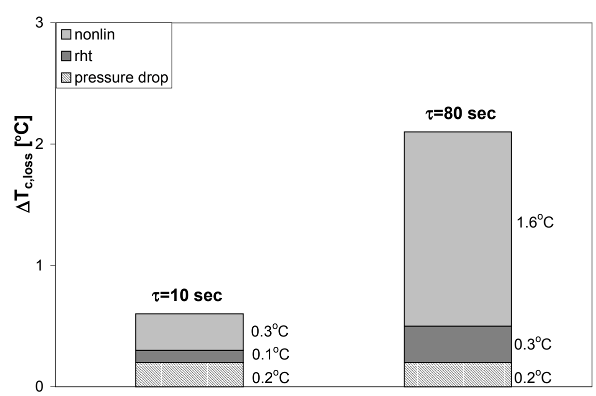

Figure 9. Condenser cycling loss term breakdown

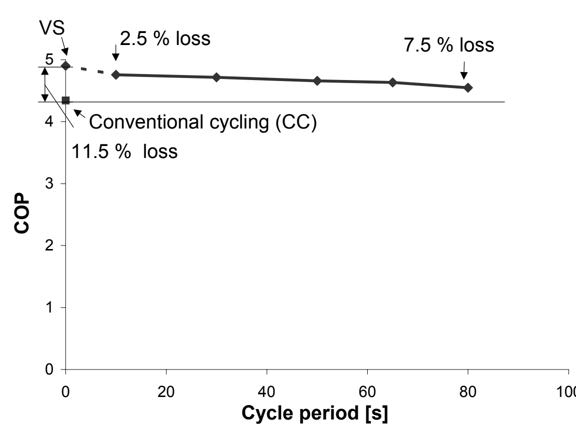

Figure 10. COP comparison to conventional cycling