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POTENTIAL COEFFICIENT OF PERFORMANCE IMPROVEMENTS DUE TO GLIDE MATCHING WITH R-407C[1]

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ABSTRACT

Potential improvements of the Coefficient of Performance (COP) were investigated for R-407C at cooling and heating conditions of a residential heat pump. The study used CYCLE-11, a semi-theoretical vapor-compression cycle model, which was upgraded to simulate cross-flow, counter-flow, and parallel-flow evaporator and condenser with no restrictions on specifying superheated vapor and subcooled liquid regions.

The COP benefit of using counter-flow heat exchangers in a low-lift air-conditioning application was found to be 7.1% when compared to a system with a cross-flow evaporator and condenser. For heating conditions (high-lift application), the predicted benefit was 3.6%. These results are representative of a water-to-water system, where counter-flow heat exchangers are suitable for implementing glide matching. Since air-to-refrigerant heat exchangers are not counter flow but rather cross-counter flow (in the best case), the potential to improve COP in a practical air conditioner is lower . The COP penalty caused by glide mismatch (parallel flow) is approximately twice as severe as the benefit that can be realized from glide matching.

NOMENCLATURE

A - area
COP - coefficient of performance
c - heat capacity
p - pressure
Q - heat
s - entropy
T - temperature
U - overall heat-transfer coefficient
UA - heat exchanger conductance
V - volume
Δ - difference

Subscripts:
ave - average
evap - evaporator
cond - condenser
hx - heat exchanger
p - constant pressure
v - constant volume
σ - saturation line
1, 2, ..., i - denotes a section of the heat exchanger

INTRODUCTION

The use of zeotropic mixtures as CFC and HCFC substitutes has brought about a discussion of performance benefits due to matching of the temperature profile of the zeotrope with the temperature profile of the heat-source and heat-sink fluids. The matching of temperature profiles, also referred to as glide matching, results in smaller irreversibilities of the heat-transfer processes, which, in turn, results in an improved cycle efficiency. This concept is often graphically explained using the Carnot cycle (for fluids with constant temperature evaporation and condensation processes) and the Lorenz cycle (for zeotropes), as shown in Figure 1. The efficiency advantage of the Lorenz cycle corresponds to the reduction in the area between the refrigerant and heat-sink and heat-source fluids. Didion and Bivens (1989) presented a comprehensive discussion of glide matching, overgliding, undergliding, temperature profile non-linearity, and pinching.

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The benefit of glide matching has been an elusive point in discussions of zeotropic mixtures because the COP improvement very much depends on operating conditions and heat exchanger design. For this reason, results of different experimental studies do not provide consistent information on the COP improvement potential. An interesting theoretical methodology for estimating the COP increase due to glide matching was developed by Cavallini (1995), who based his method on thermodynamic fundamentals of the Carnot cycle and the Lorenz cycle. Because of this fundamental approach, the Cavallini analysis does not involve fluid properties and does not take into consideration some real cycle characteristics such as evaporator superheat, condenser superheat, and condenser subcooling. To avoid these limitations, a semi-theoretical model, CYCLE-11, was used in this study with the goal of examining the COP improvement potential for a residential heat pump charged with R-407C.

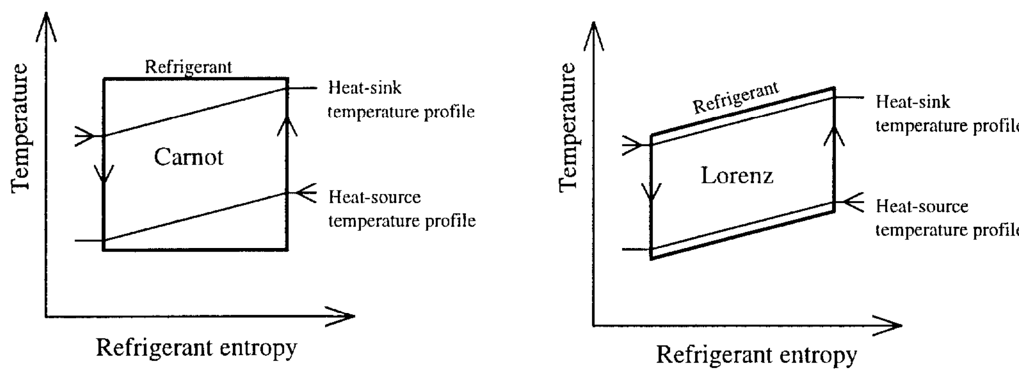

Figure 1. The Carnot and Lorenz cycles on the temperature-entropy coordinates

SIMULATION MODEL

CYCLE-11 (Domanski and McLinden, 1992, Domanski et al., 1994a) is a semi-theoretical model for evaluating the performance of refrigerants or refrigerant mixtures in the vapor-compression cycle. Simulation runs are performed for user-specified temperature profiles of the heat source and heat sink. This feature makes CYCLE-11 suitable for examining the effect of glide matching on the system performance. The recent modifications of CYCLE-11 upgraded its modeling capabilities to include cross-flow, counter-flow, and parallel-flow heat exchange with refrigerant superheat and subcooling where appropriate. The program employs FORTRAN subroutines from REFPROP (Huber et al., 1996) to calculate refrigerant thermodynamic properties.

Two versions of CYCLE-11 were used in this study: the DT version and the UA version. Both versions use the same internal algorithms and are equivalent except for some differences in the input data required and resulting simulation constraints. The most important difference is in the input data for representation of the heat exchangers. In the DT version, the evaporator and condenser are represented by their ΔT, an average effective temperature difference between the refrigerant and the heat-exchange fluid. Specifying the same ΔT for different simulation runs results in the same heat flux through the heat exchangers, an important condition for a fair comparison of different refrigerants (McLinden and Radermacher, 1987). This type of evaluation is highly impractical to implement in a laboratory because it would require changing the evaporator and condenser size for any refrigerant and operating condition to satisfy the constant heat-flux constraint. In the UA version, the evaporator and condenser are represented by their UA, a product of the overall heat-transfer coefficient (U) and heat-transfer area (A). Simulations using the UA version correspond to tests in a given system with fixed heat exchangers. In both DT and UA versions a constant heat-transfer coefficient is assumed for single-phase and two-phase refrigerant regions.

SIMULATION RESULTS

Comparison of CYCLE-11 and Cavallini Algorithm

This section reports simulation results obtained using the DT version of CYCLE-11 and compares them with the results obtained from the algorithm developed by Cavallini (1995). Table 1 shows the two operating conditions simulated; a low-temperature lift condition representing the cooling mode and a high-temperature lift condition representing the heating mode. For CYCLE-11 simulations 70% polytropic efficiency, zero pressure drop in heat exchangers, and 5 °C superheat at the evaporator outlet were used. Subcooling at the condenser exit was optimized for each simulation case. Different combinations of cross-flow and counter-flow evaporator and condenser were simulated to evaluate the effect of glide matching on performance.

The Cavallini method uses the Carnot and Lorenz cycles to estimate the improvement in the cycle performance due to a change from a constant saturation temperature in the evaporator and condenser to a specified refrigerant temperature glide. The final relation in the Cavallini algorithm has the following form:

$$\frac{\Delta(COP)}{COP} = \frac{1}{\overline{T}_c - \overline{T}_e}\left(\Delta T_c^* + \frac{T_c}{T_e}\Delta T_e^*\right) \qquad (1)$$

where T_c and T_e are the saturation temperatures of the Carnot cycle, $\overline{T}_c$ and $\overline{T}_e$ are the average saturation temperatures of the Lorenz cycle, and ΔT_c^* and ΔT_e^* are the changes in saturation temperature due to glide matching when changing from the Carnot to Lorenz cycle.

Table 2 presents the obtained results. For all simulated cases, CYCLE-11 predicted COP improvement approximately twice as high as the Cavallini method. For both methods, the COP improvement in the cooling mode is higher than in the heating mode. This can be explained by a difference in temperature lifts for these cases. As can be best observed for the cycle outlined on the T-s diagram, the reduction in work and increase in capacity due to glide matching represent a larger percentage of the original work and capacity for lower temperature-lift applications. In both operating conditions, CYCLE-11 predicted a higher benefit due to a change from a cross flow to a counter flow for the condenser than for the evaporator.

Table 1. Operating conditions of the simulations using the DT version of CYCLE-11 [°C]

Mode	Outdoor Temp.	Indoor Temp.	Effec. ΔT Cond.	Effec. ΔT Evap.	Sink Fluid Temp. Glide	Source Fluid Temp. Glide	Subcool. Cond. Outlet	Superheat Evap. Outlet
Cooling	27.8	26.7	9.8	13.0	8.5	13.0	8.8	5.0
Heating	8.3	21.1	7.9	5.0	8.9	4.4	8.0	0.0

Table 2. COP improvement for different condenser and evaporator configurations over cross-flow evaporator and cross-flow condenser configuration by CYCLE-11 and Cavallini method for conditions of Table 1

Configuration	Cooling		Heating	
	COP increase by CYCLE-11	COP increase by Cavallini	COP increase by CYCLE-11	COP increase by Cavallini
Cross-Flow Evaporator Counter-Flow Condenser	3.9%	1.4%	2.3%	1.1%
Counter Flow Evaporator Cross-Flow Condenser	2.9%	2.1%	1.2%	0.7%
Counter-Flow Evaporator Counter-Flow Condenser	7.1%	3.5%	3.6%	1.8%

Glide Matching Effect on COP

To study the influence of glide matching in greater detail, a series of simulations were performed using several glides of heat-sink and heat-source fluids with their average temperatures kept constant. The average effective temperature differences between the refrigerant and heat-exchange fluids, condenser subcooling, and evaporator superheat were held constant as listed in Table 1. The average temperature of the heat-sink fluid was kept 32.0 °C (based on the 27.8 °C inlet temperature and 8.5 °C temperature glide). The average temperature for the heat-source fluid was kept 20.2 °C (based on the 26.7 °C inlet temperature and 13.0 °C).

Figure 5 shows simulation results for a counter-flow condenser and cross-flow evaporator system. The maximum COP occurs at the highest glide in the condenser, limited in this case by the low temperature lift of the application (higher temperature glide would result in the outdoor temperature being lower than the indoor inlet temperature). Figure 6 shows COPs for the inverse case with a counter-flow evaporator and cross-flow condenser. The optimum temperature glide of 6 °C corresponds to the glide of R-407C. Surprisingly, there is also an optimum temperature glide for the heat-sink fluid interacting with a cross-flow condenser. This is due to the superheating and subcooling regions in the condenser. The cross-flow evaporator case did not benefit from a temperature glide because refrigerant superheat was small. Additional simulations with high values of evaporator superheat indicated that a temperature glide would result in an improved COP in these cases. Comparing the COP predictions in Figures 5 and 6 for large heat sink glides, the COP of the counter-flow condenser system is higher than the COP of the counter-flow evaporator system regardless of the temperature glide of the heat source.

An alternative way to analyze the system performance is to examine its entropy production for different cases. Figure 7 presents the entropy generation for the system with a cross-flow evaporator and cross-flow condenser at the operating conditions listed in Table 1. The figure shows minimum entropy generation at approximately 6 °C glide of the heat-sink fluid and at 0 °C to 2 °C glide of the heat-source fluid.

Reconfiguring one heat exchanger, either an evaporator or condenser, from a cross-flow to a counter-flow configuration results in a change in entropy production in both heat exchangers. Switching to a counter-flow evaporator yielded reduced entropy production in both the evaporator and condenser. On the other hand, switching to a counter-flow condenser reduced entropy production in the condenser and increased entropy production in the evaporator. However, the overall effect of reducing irreversibilities in this case was stronger with the counter-flow condenser than with the counter-flow evaporator.

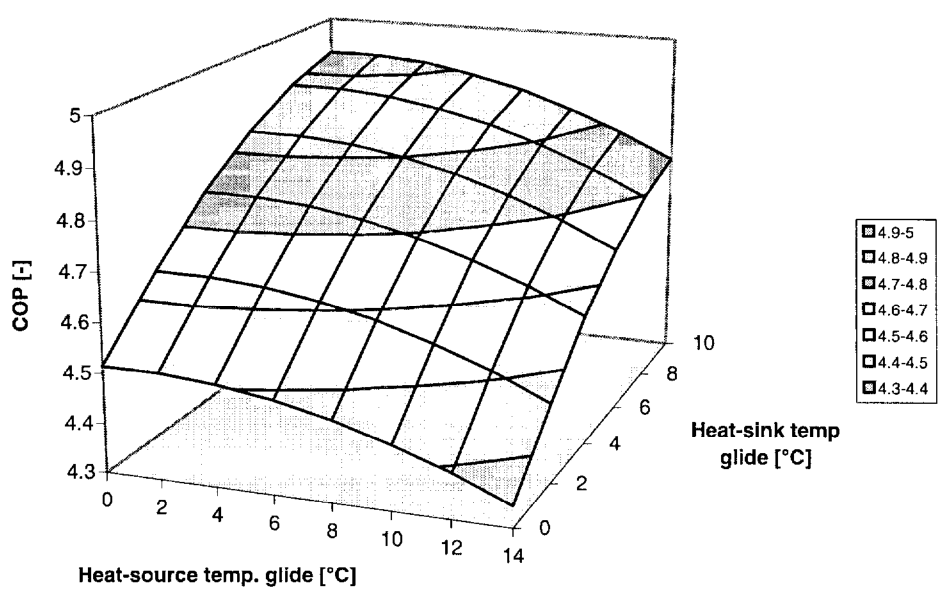

Figure 5. COP versus heat-sink and heat-source glides (DT version of CYCLE-11, cooling, $T_{outdoor,ave}$=32.0 °C, $T_{indoor,ave}$=20.2 °C, cross-flow evaporator, counter-flow condenser)

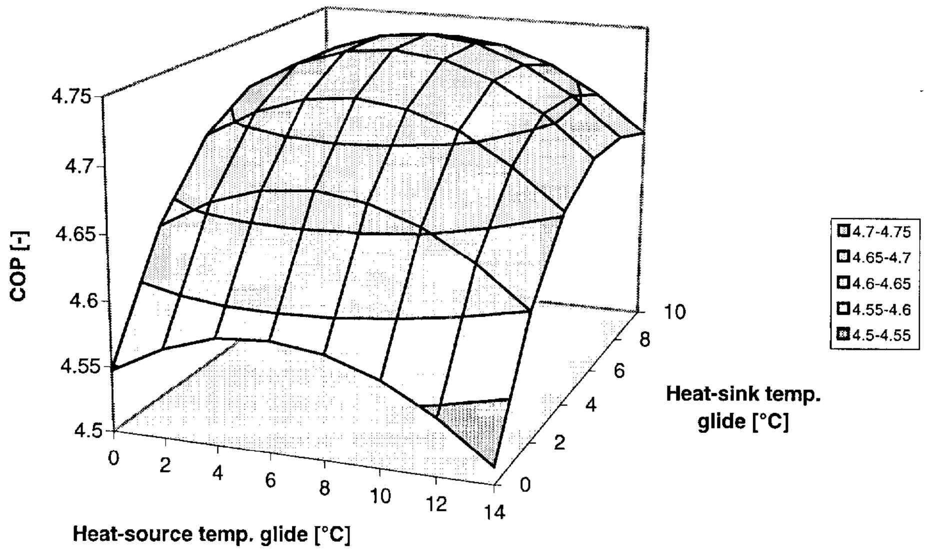

Figure 6. COP versus heat-sink and heat-source temperature glides (DT version of CYCLE-11, cooling, $T_{outdoor,ave}$=32.0 °C, $T_{indoor,ave}$=20.2 °C, cross-flow condenser, counter-flow evaporator)

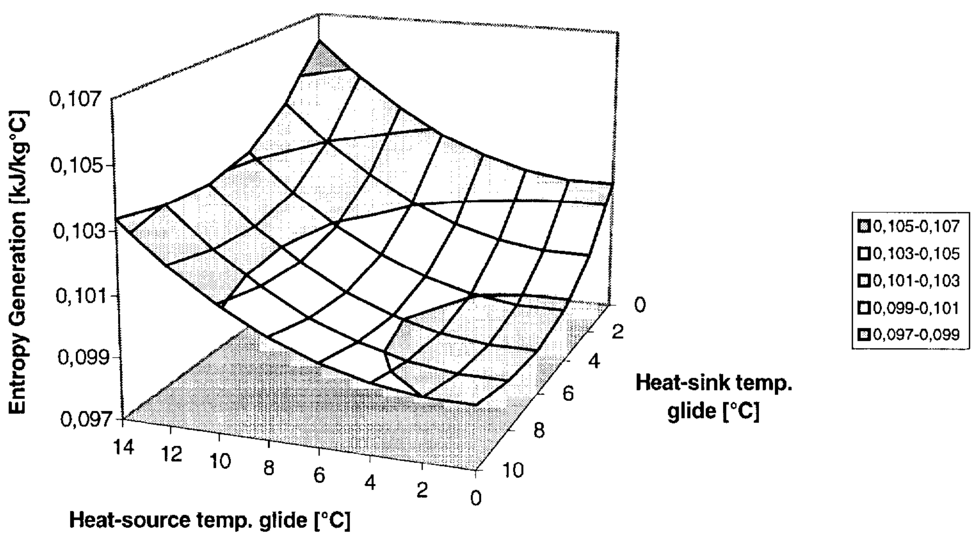

Figure 7. Entropy generation versus heat-sink and heat-source temperature glides (DT version of CYCLE-11, cooling, $T_{outdoor,ave}$=32.0 °C, $T_{indoor,ave}$=20.2 °C, cross-flow evaporator and cross-flow condenser)

The average effective temperature difference, ΔT, also affects the performance improvement when a heat exchanger is reconfigured from cross flow to counter flow. As a rule, the smaller the temperature difference, the greater the benefit of changing from a cross-flow to counter-flow configuration.

COP at a Range of Ambient Conditions

The simulation results presented in the previous sections were obtained with the DT version of CYCLE-11. In the DT version, CYCLE-11 adjusts heat-transfer area in the evaporator and condenser to implement the constant-heat-flux constraint. This section presents simulation results obtained with the UA version of CYCLE-11. Since the UA version simulates a system with fixed heat exchanger UAs, the results are representative of the performance changes affected by reconfiguring from cross flow to counter flow for fixed heat-transfer-area heat exchangers. The simulations started with the baseline conditions listed in Table 3. The simulations assumed zero pressure drop in heat exchangers, 5 °C superheat at the evaporator outlet, and 70% polytropic efficiency. For each configuration the subcooling at the condenser was optimized for maximum COP. This subcooling was kept constant for all other outdoor temperatures to approximate the control of the system by a thermostatic expansion valve. Glides of the heat-sink and heat-source fluids were changed according to the capacities of the heat exchangers as if the mass flow rate and the heat capacity of these fluids were constant. The UA values were selected based on the cooling mode simulations that resulted in 7.2 °C average evaporation temperature and 42.8 °C average condensation temperature for a system with a cross-flow evaporator and condenser, and optimum condenser subcooling.

Figure 8 shows the cooling COPs of systems with different heat exchanger configurations referenced to the COP of the system equipped with the cross-flow evaporator and cross-flow condenser. The two limiting cases are the systems with both counter-flow and both parallel-flow heat exchangers, respectively. The performance of the system with both counter-flow heat exchangers is the upper limit in the figure. Its COP is closer to that of the system with both cross-flow heat exchangers (baseline) than the performance of the lower-limit case, the system with both parallel-flow evaporator and condenser. This indicates that the potential benefits of glide matching are smaller than the possible performance penalty caused by a glide mismatch. The cooling results shown in Figure 8 are representative of the results obtained for the heating mode.

The COP improvements predicted by the UA version are lower than those predicted by the DT version and shown in Table 2. This is not an unexpected outcome because the results in Table 2 represent the COP increase in systems with the same heat exchanger heat flux.

Table 3. Baseline operating conditions of the simulations using UA version of CYCLE-11 [°C]

Outdoor temperature	Indoor temperature	Outdoor temperature glide	Indoor temperature glide
27.8	26.7	8.5	13.0

Note: Condenser subcooling was optimized for each heat exchanger configuration for this baseline condition

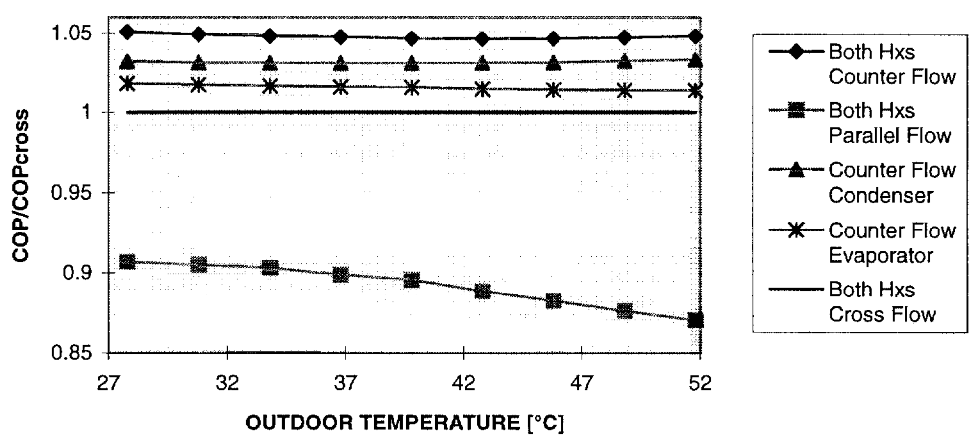

Figure 8. COP referenced to COP of a system with cross-flow evaporator and condenser (UA version of CYCLE-11, cooling, T_{indoor}=26.7 °C)

CONCLUDING REMARKS

The COP benefit of using counter-flow heat exchangers in R-407C systems in a low-lift air-conditioning application was found to be 7.1%. For heating conditions (high-lift application), the predicted benefit was 3.6%. These predictions were obtained for a constant-heat-flux scenario where the average effective temperature difference between refrigerant and heat-transfer fluid was held constant. Smaller COP improvement was predicted for a system with a fixed heat exchanger size. For this case cooling mode simulations indicated a COP improvement of 5%.

Since air-to-refrigerant heat exchangers are not counter flow but rather cross-counter flow (in the best case), the potential to improve COP in a practical air conditioner is lower than the values obtained for a system with counter-flow evaporator and condenser. The presented result are more representative of a water-to-water system, where counter-flow heat exchangers are suitable for implementing glide matching.

The improvement of COP due to glide matching predicted by the Cavallini theoretical method based on the Carnot and Lorenz cycles is approximately half the improvement predicted by CYCLE-11.

The COP penalty caused by a glide mismatched (parallel flow) is approximately twice as severe as the benefit that can be realized from glide matching. Hence, optimization of refrigerant circuitry in a serpentine heat exchanger may be a crucial factor for obtaining high system efficiency. This applies not only to zeotropic mixtures but also to single-component refrigerants that may have a temperature glide due to pressure drop.

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