Residential Space Conditioning and Water Heating with Transcritical CO2 Refrigeration Cycle

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ABSTRACT

The high heat rejection temperature of the transcritical CO_2 cycle creates opportunities for meeting domestic water heating needs in addition to space heating and air conditioning. This paper presents a thermodynamic cycle analysis that accounts for the presence of realistic heat exchanger sizes in a residential air to air heat pump system. The analyses presented here explore a variety of options for integrating domestic water heating into the heating and cooling cycles. Results suggest the following operating strategies for maximizing energy efficiency. When heating or cooling loads are relatively high, compressor speed and power are increased to heat water within the specified time constraint. During periods of low space conditioning load when the compressor is cycling, water is heated during the off-cycle at a faster rate. Multiple hardware configurations were also explored: a single water heater located at the compressor discharge and in series with the gas cooler air coil; a second water pre-heater in series upstream of the expansion device; and a tandem mode alternating between space conditioning and water heating. Since the heat sink provided by the cold water competes with the internal heat exchanger (IHX), seasonal IHX bypass options are also analyzed. The results illustrate optimal hardware configurations and operating strategies for a range of outdoor air temperatures and water heating demands.

INTRODUCTION

The use of CO_2 refrigerant for simultaneous space conditioning and water heating has been the focus of renewed research and testing since Lorentzen re-introduced the idea of the transcritical cycle (Lorentzen, 1994). Neska *et al* have investigated several CO_2 systems under standard test conditions, including solo heat pump water heaters, and dual purpose systems that simultaneously provide space conditioning and domestic water heating (Neska, 2002). Several authors have published results of prototype CO_2 systems operating over wide ranges of test conditions. This paper presents an idealized residential system that meets domestic water and space conditioning demands. The ideal system presented here heats water to U.S. domestic storage temperatures of 60°C, in addition to meeting residential heating and cooling loads. Using simplified thermodynamic calculations and assumptions, optimal system operation strategies and configurations were identified for both heating and cooling seasons. The results provide a basis for more detailed system simulations that can focus on optimizing actual heat exchanger hardware design.

The optimal operating strategy varies throughout the year because the transcritical CO_2 cycle exhibits a COP-maximizing discharge pressure, with a discharge temperature above or below the desired hot water temperature. Optimization algorithms are used to identify the most efficient operating condition. Water heating costs can be reduced to a third of electric resistance at $T_{amb} > -5°C$ and the efficiency increases with T_{amb}. After reviewing the assumptions made, an overview will be given of the various operation strategies and hardware configurations selected by climate season. Each operating strategy based on ambient climate will be examined in detail. Summaries of system parameters, such as high and low end CO_2 pressures and temperatures, will provide ideal system parameters which provide the starting point for more detailed system simulation and control studies of the space conditioning and water heating system.

1. ASSUMPTIONS

A number of assumptions were made in order to greatly simplify cycle calculations and the optimizations, which were performed using Engineering Equation Solver (EES). Heating and cooling loads, based on an overall UA value from a moderately insulated house, are summarized in Figure 1, with loads approximated as linearly dependent on ambient temperature (Richter et al, 2001). The system provides 10.5 kW of cooling at the standard test condition, $T_{amb} = 35^oC$. The heating and cooling lines meet at 18^oC, and it is assumed that ventilation suffices between $15\text{-}20^oC$. A minimum water heating capacity of 4 kW is desired, matching common electric resistance heaters that meet the 12.1 kWh/day average US domestic hot water demand in 3 hours. The compressor was sized for maximum displacement at $T_{amb} = 40^oC$, with a speed range of $30 < \omega_c < 120$ Hz. The isentropic efficiency is taken from a 10.5 kW Dorin compressor and expressed as a function of P_{dis}/P_{suc}.

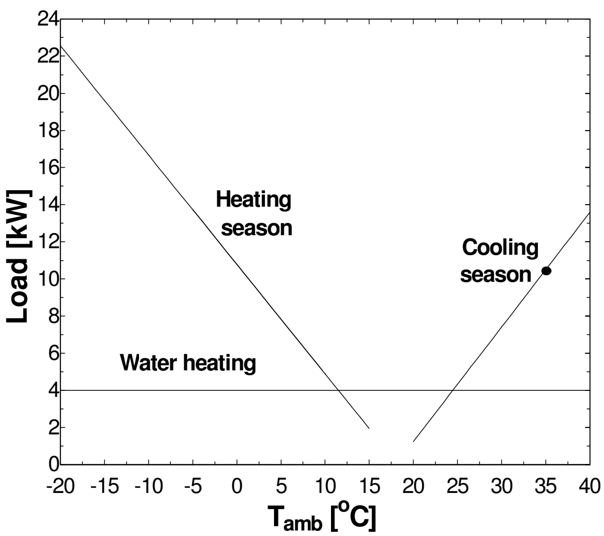

Figure 1. Space conditioning and water heating loads.

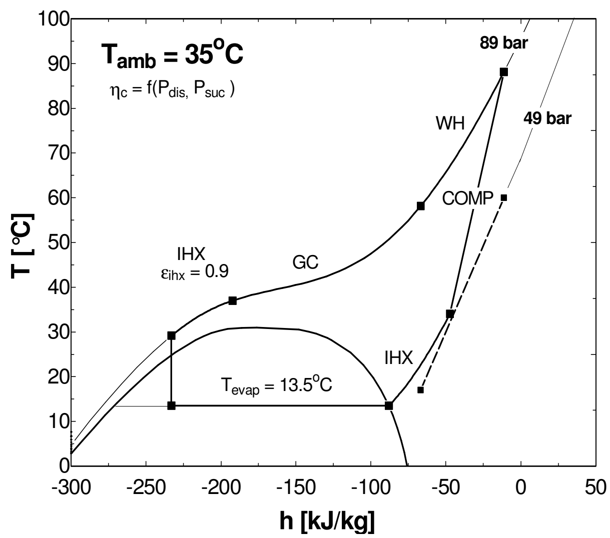

Figure 2. "Ideal" simultaneous a/c and water heating cycle.

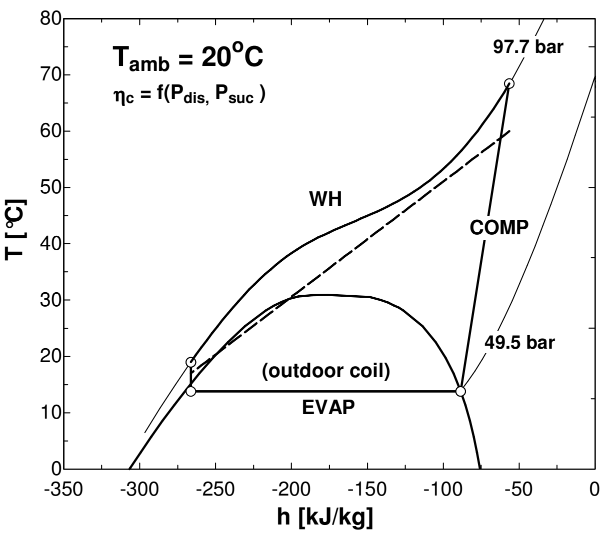

Figure 3. The water heating-only cycle.

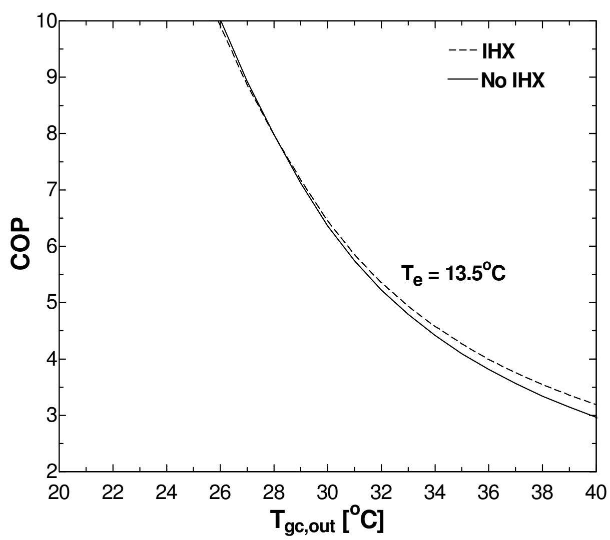

Figure 4. Effect of IHX on a/c-only cycle COP.

Assumptions for the cooling season are summarized in Figure 2, which shows a CO_2 T-h diagram for simultaneous a/c and water heating (a/c & WH). A pinch limit, $T_{CO2} \geq T_{H2O\text{ or air}} + 2^oC$, was enforced for the counter-flow water and indoor air heat exchangers. For a/c season the T_{evap} is maintained at 13.5^oC by modulating blower speed, to meet sensible heat ratio target of 0.75 (Kim, 2002). The outdoor coil is cross-flow and assumed large enough to be able to cool CO_2 to a 2°C approach at any ambient air temperature ($T_{gc,out} = T_{amb} + 2$°C). An internal heat exchanger (IHX) is assumed to have a 0.9 effectiveness, and proves beneficial as cooling loads increase as shown in Figure 4. Pressure drop is neglected in all heat exchangers. A simplified heat pump cycle can be used for space heating (SH) or water heating (WH). Figure 3 shows a T-h diagram of a WH cycle where all heat is rejected to the water. During heating season, a UA = 3.6 was assumed for the outdoor coil, based on an air flow rate and geometry from a previously simulated coil (Kim, 2002). For space heating, air is supplied to the room at 40°C and returned at 20°C; and water is heated from 17°C supply to 60°C delivery temperatures.

It is recommended that the IHX be bypassed in the heat pump cycle, because the highly superheated suction gas produces a higher compressor discharge temperature as observed in Figure 5. Because of concerns about compressor oil breakdown, and the negligible improvement in COP at low evaporating temperatures as shown in Figure 6, the IHX can be bypassed during the heating season.

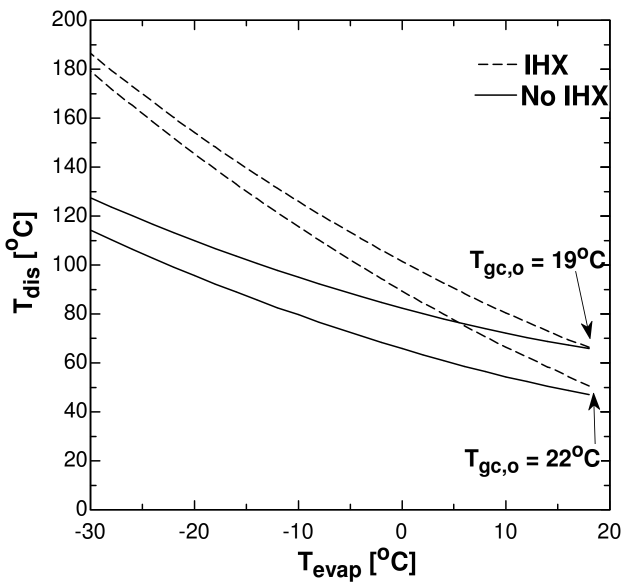

Figure 5. Affect of IHX on T_{dis} of HP cycle

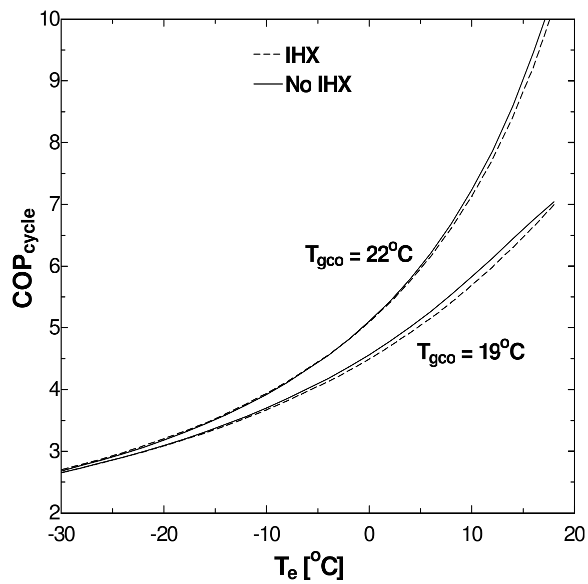

Figure 6. Affect of IHX on COP of cycle

2. OVERVIEW OF OPTIMIZED SYSTEM

Figure 7 shows the minimized total operating cost of space conditioning and water heating for a wide range of ambient temperatures. Five distinct operating strategies are needed for this optimized system, as shown in Figure 8. The first two strategies apply to the a/c & WH cycle. For $T_{amb} > 23.5$°C, indoor air is the sole heat source for water heating. The water inlet valve to the heater is simply opened for simultaneous a/c & WH, and closed during a/c-only operation. At $20 < T_{amb} < 23.5$°C, insufficient heat available in the indoor air; therefore, a combination of indoor and outdoor air heat sources may be used. Thus, a third operation strategy is needed, using the CO_2 system as a water heat pump during a/c off-cycle time. The system continues to act as a water heat pump during the ventilation season, $15 < T_{amb} < 20$°C. The fourth strategy, tandem operation between SH & WH cycles, is employed when $-4 < T_{amb} < 15$°C. At $T_{amb} < -4$°C, the system reaches maximum capacity and space heating loads must be met using a fifth strategy, combining the heating output from the SH cycle and an electric resistance heater. Water heating load is provided using electric resistance.

Adding water heating to the overall CO_2 system reduces the water heating cost by 2/3 or more compared to electric resistance. Figure 9a shows the cost of adding water heating load to the CO_2 refrigerant system and Figure 9b expresses that cost in terms of water heating COP. Cost declines further with rising evaporating temperature as the outdoor temperature increases. One-stage water heating was found to be optimal or nearly optimal for all seasons, with the CO_2-to-H_2O heat exchanger placed immediately downstream of the compressor. Where the water source is colder, it may be cost-effective to add a preheater because a colder heat sink increases cycle efficiency. Water heating times shown in Figure 10 are influenced by compressor speeds, shown in Figure 11.

At $T_{amb} > 29^oC$ ($P_{dis} > 75$ bar), there is abundant heat rejection to the air due to the sizable a/c loads, and the COP-maximizing discharge pressure for the a/c only cycle is high enough to heat water to 60°C, as observed in Figures 12 and 13. Therefore, water heating is free. The 3-hour time constraint determines how much of the high-side refrigerant temperature profile is needed for water heating (4 kW). The rest of the heat is rejected to air in the gas cooler downstream. As a/c load decreases with T_{amb}, high side pressure decreases, and when it reaches 75 bar at 29°C, the temperature difference between CO_2 and water approaches the pinching limit and water heating is still free, because no extra compressor work is needed. CO_2 flows through the refrigerant side of the water heater continuously, while water flows through it for 3 hours per day and is then shut off for the remaining 21 hours, (see Figure 8). Rejecting all heat to the water would result in much faster heating times, but it is less efficient because the high-side discharge pressure would be 94.4 bar**,** regardless of T_{amb}, which is inefficient.

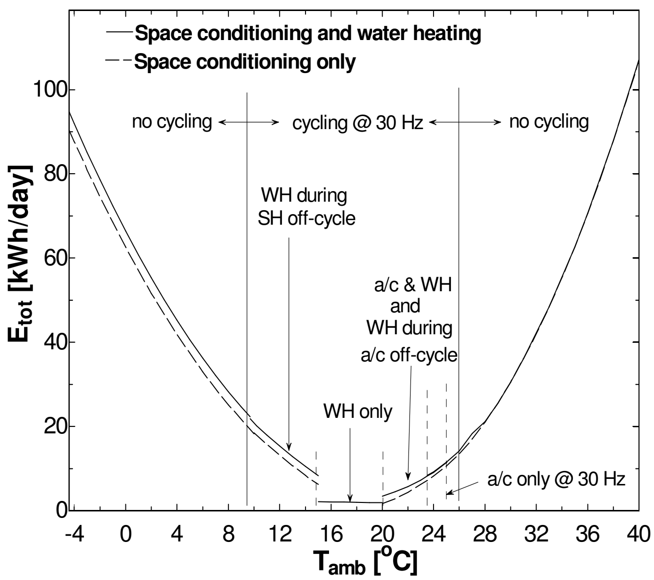

Figure 7. Total cost of space conditioning and water heating

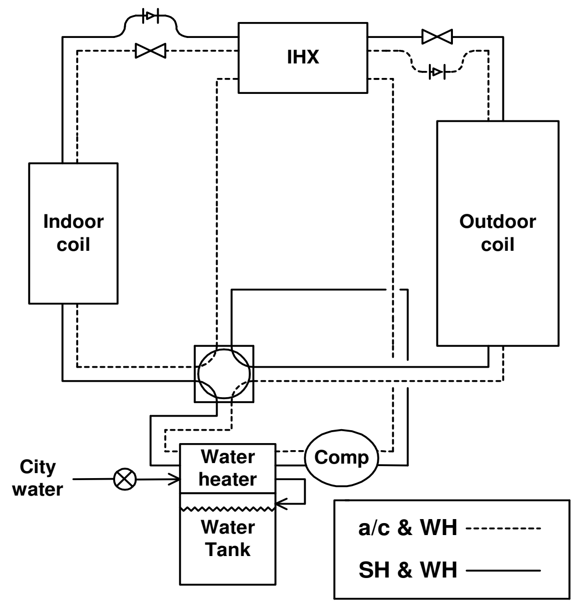

Figure 8. Space conditioning and water heating cycles

At $T_{amb} < 29^oC$, the COP-maximizing discharge pressure at these moderate-load conditions is too low to heat water to 60°C. Therefore, added compression energy is required in order to limit water heating time to 3 hours. Figure 14 demonstrates (at $T_{amb} = 26^oC$) how high-side pressure is raised 12 bar above the optimal a/c-only high-side pressure, raising T_{dis} from 55 to 70°C. The kWh cost for the added compressor power is very small, only 0.72 kWh, compared to the required 12.1 kWh/day of domestic hot water produced, yielding a water heating COP_{wh} of ~ 17. At 26°C, the compressor speed reaches its (lubrication limited) minimum of 30 Hz. Compressor cycling occurs at $T_{amb} < 26^oC$, meeting the daily cooling load in less than 24 hours. The P-h diagram is identical for steady and cycling operation. The heat rejection profile in Figure 14 shows the majority of the heat being rejected to water, reflecting the decreasing size of the indoor air heat source as T_{amb} drops. At $T_{amb} = 23.5^oC$, the a/c & WH system runs only 3 hours per day, with all 4 kW of heat rejection going to water heating.

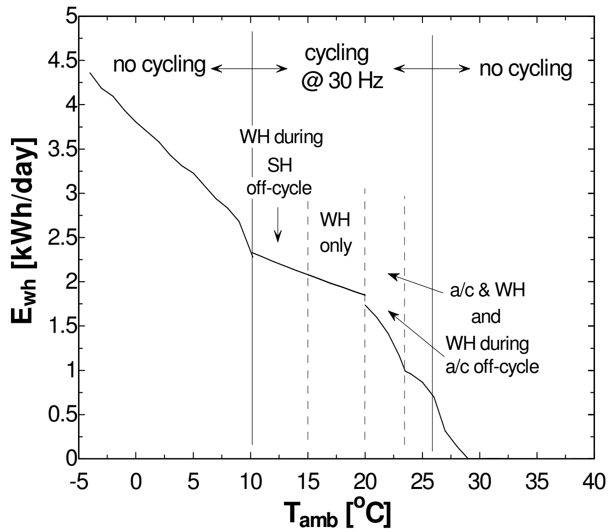

Figure 9a. Water heating costs.

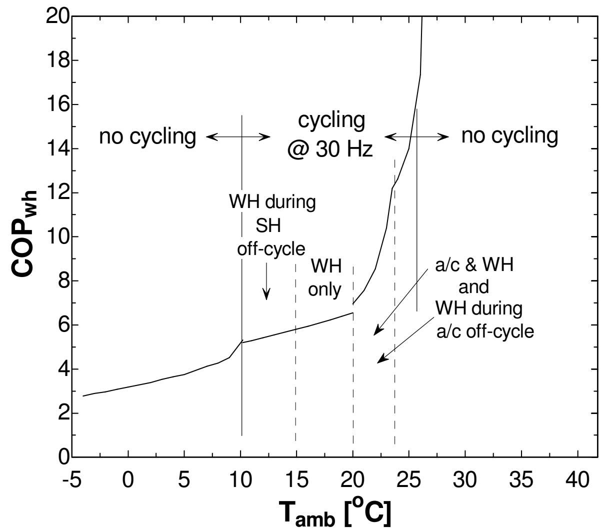

Figure 9b. Water heating COP

At T_{amb} < 23.5°C, the cooling load is small and not enough heat is rejected to heat water in 3 hours. But the outdoor air is available for use as a heat source and can heat hot water faster than 3 hours. Therefore, a combination of the indoor and outdoor heat sources can be used when 20 < T_{amb} < 23.5°C. The total WH time will still be 3 hours, using the outdoor air only as much as needed because it is colder than the indoor air being used as the heat source in a/c mode. Off-cycle time from space cooling is abundant, as shown in Figure 11, with enough time for optimal water heating at the minimum 30 Hz compressor speed. When the solo WH cycle operates, the CO_2 refrigerant can flow through the indoor coil with the blower fan turned off, without heating the indoor air significantly.

When T_{amb} < 20°C, the outdoor air is the sole heat source for heating water and air. During the ventilation range, 15 < T_{amb} < 20°C, the CO_2 system is used only as a water heat pump. When space heating demands start at T_{amb} < 15°C, water heating at 30 Hz may be performed in tandem with space heating, because heating loads are small at mild temperatures, off-cycle time is abundant. The water heating time from running the WH cycle continuously at 30 Hz is very reasonable, taking 2 hours or less. Figure 10 shows that total space heating and water heating (30 Hz) times sum to 24 hours at 10.2°C.

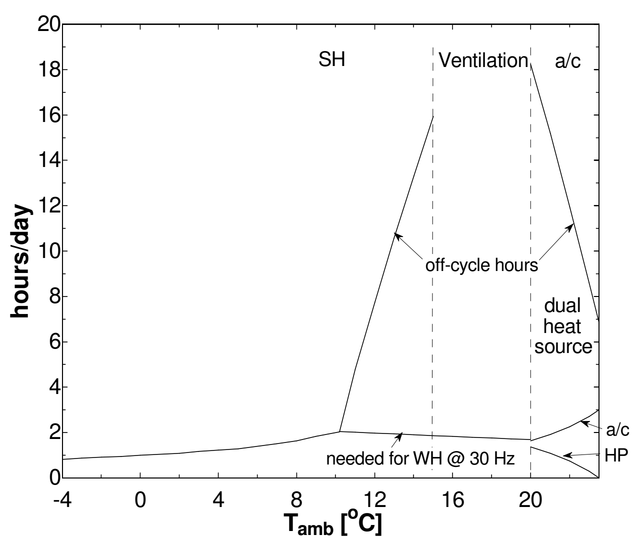

Figure 10. Water heating times

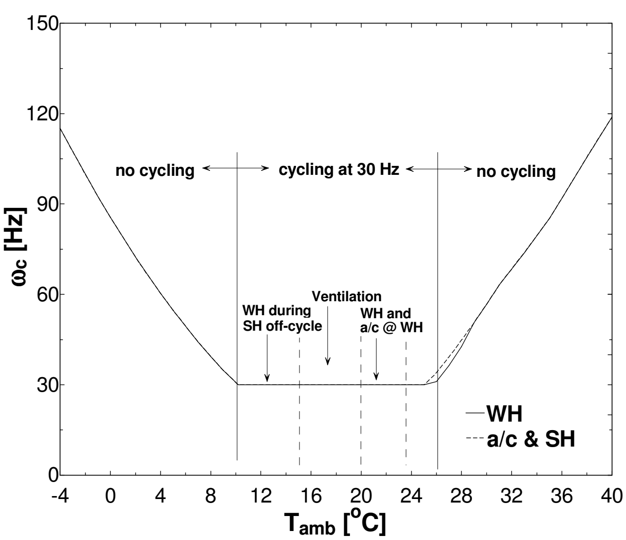

Figure 11. Compressor speeds

At -4 < T_{amb} < 10.2°C, the space and water heating system configurations operate continuously in tandem at a common minimum compressor speed until the compressor reaches the maximum 120 Hz speed limit at around -4°C.

Consequently, as compressor speed increases to meet higher SH demands on increasingly colder days, the water heating time will decrease, as shown in Figure 15. If water heating time is increased above the optimum, the SH cycle, which operates during the majority of the day, will be forced to run less efficiently at a higher capacity. The cycle state points are dictated by the compressor speed ω_c, which determines water and space heating capacity. If ω_c is increased, T_{evap} decreases because the outdoor coil UA is finite. Figures 12 and 13 show the discharge temperatures and pressures resulting from the 2°C pinch constraint on tandem SH and WH operation at their respective 40°C and 60°C delivery temperatures. While the compressor runs at constant speed, the transitions are accomplished by the back-pressure valve adjusting P_{dis} from approximately 75 to 90 bar in order to maximize COP for SH and WH, respectively. At colder temperatures than -4°C, system capacity is exceeded, and electric resistance heaters are used to supplement the space heating and to meet the entire water heating load.

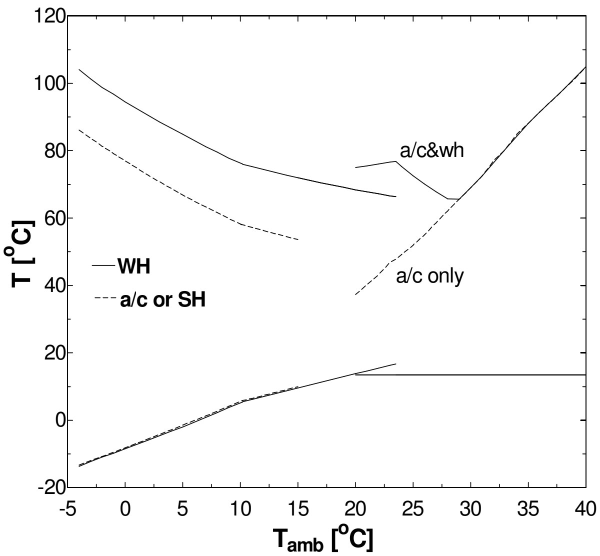

Figure 12. Discharge and evaporating temperatures

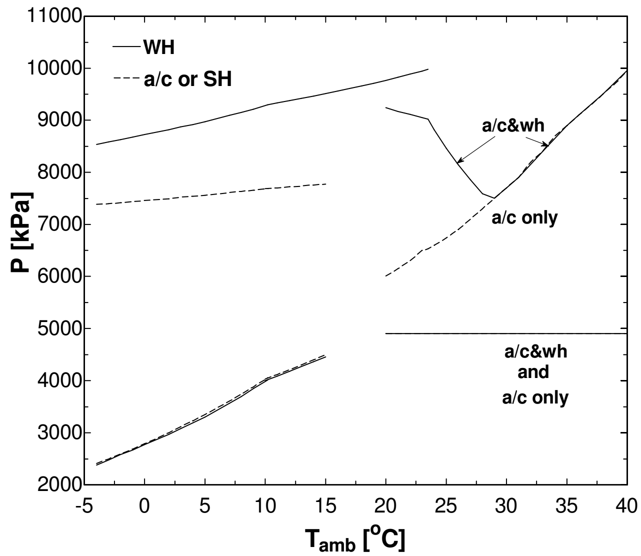

Figure 13. Discharge and suction pressures

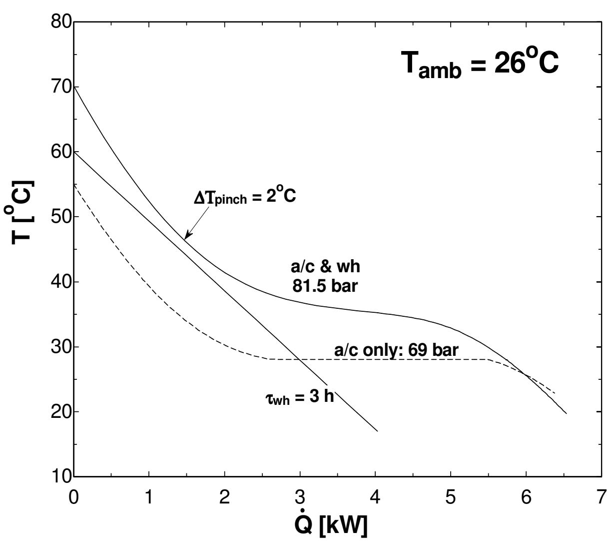

Figure 14. Changes in heat rejection profile diagram in a/c season as T_{amb} falls.

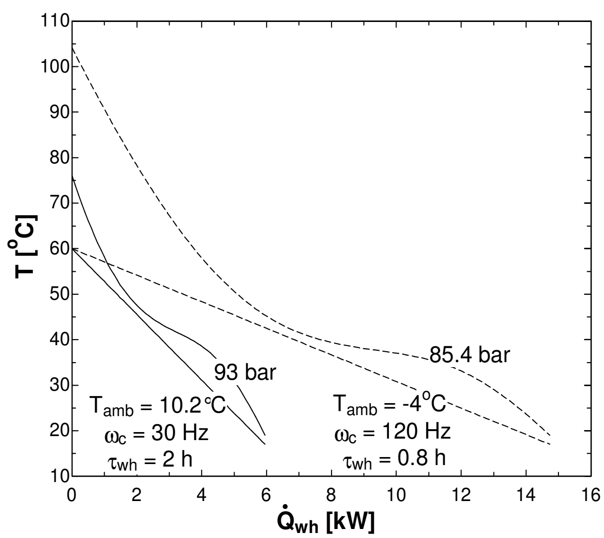

Figure 15. Impact of compressor speed on refrigerant and water temperature profiles.

The optimized system meets the 3 hour water heating limit during all climate ranges. At T_{amb} > 23.5°C, water is heated in 3 hours using the indoor air as the heat source, with P_{dis} being increased at T_{amb} < 29°C to maintain the 60°C water delivery temperature. At ambient temperatures between 20 and 23.5°C, the total time remains 3 hours, with the outdoor air heat source being used for part of that time. Between 10.2°C and 20°C, the hp-WH cycle at a 30 Hz compressor speed can be used, with WH times under 2 hours. In the -4 < T_{amb} < 10.2°C cold season range, water heating time decreases as compressor speed is rising to supply the increasing space heating loads. These various water heating times during different climate conditions assume that the water heater is sized for minimum operating costs. The CO_2 to water heat exchanger will be sized at T_{amb} = 23.5°C, where the average refrigerant-to-water temperature difference is smaller than either of the profiles shown in Figure 15.

3. CHOOSING OPTIMAL SYSTEM CONFIGURATIONS

The total cost of water heating using 1 or 2 stage water heating strategies is very similar, as shown in Figure 16. For 2 stage water heating, in which refrigerant flows through a preheater downstream of the outdoor coil, system efficiency is increased by the presence of this additional heat sink. At T_{amb} > 28°C,the extra heat sink improves the a/c & WH cycle efficiency and lowers the total operating cost by approximately 1 kWh/day, because simultaneous a/c and WH occurs for only 3 hours per day. Only in very hot climates would this small benefit offset the cost and complexity of the extra refrigerant circuit through the water heater. Benefits would also be larger in cities where the water supply temperature is lower or hot water capacity demands are large. During cooler ambient conditions there is very little heat rejection to preheat sink because the CO mass flow decreases along with the temperature difference between the water inlet temperature and that of the CO_2 exiting the outdoor coil.

For the heating season, a simultaneous 1 stage SH & WH strategy was compared to a tandem SH & WH strategy. Over the comparable T_{amb} range, tandem water heating costs an average of about 0.5 kWh less than the simultaneous SH & WH strategy, as shown in Figure 17. In simultaneous mode, extra heat must be generated to meet the simultaneous SH and WH loads; therefore the maximum 3 hour limit is always optimal. Tandem mode is more efficient because the discharge pressure can be optimized for WH and SH independently. Since water is heated more quickly in tandem mode, the compressor will hit maximum speed when T_{amb} is a few degrees colder than simultaneous mode. Thermal comfort effects from tandem operation can be controlled by more frequent alternation between SH and WH.

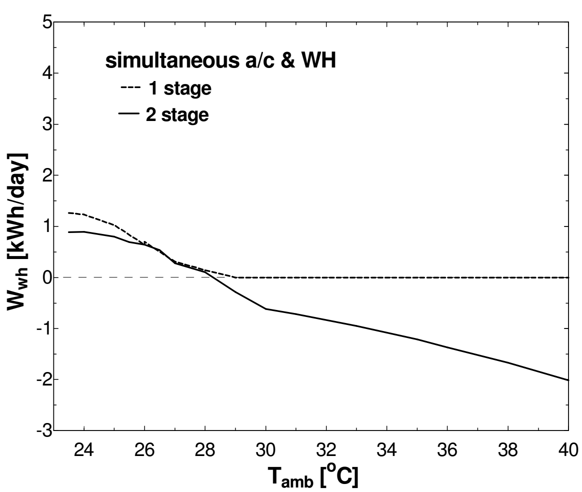

Figure 16. Comparison of 1 and 2 stage water heating for the cooling season

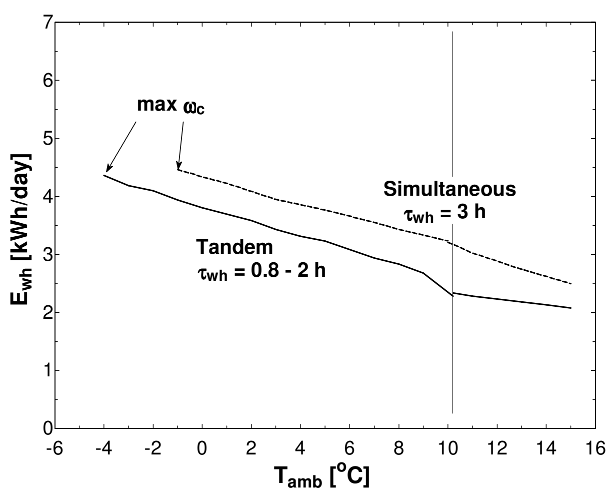

Figure 17. Comparison of tandem and simultaneous space and water heating

4. SUMMARY AND CONCLUSIONS

Based on the preceding cycle analyses, it is apparent that a complex system configuration is not required to approach ideal water heating efficiency while satisfying residential a/c and heating loads. A rather simple single stage water heater will suffice, due to the mild 17°C city water heat sink. Therefore, future hardware-specific simulation analyses can be focused on this simple configuration to refine component and system designs. Conservative approaches to preventing compressor oil breakdown necessitate bypassing the IHX during the heating season, because of high discharge temperatures. The optimal operating and control strategies appear to be more complex, but this is partly an artifact of the need to make specific assumptions for initial cycle analysis. In practice, the selection of sensors will determine the complexity of the control system.

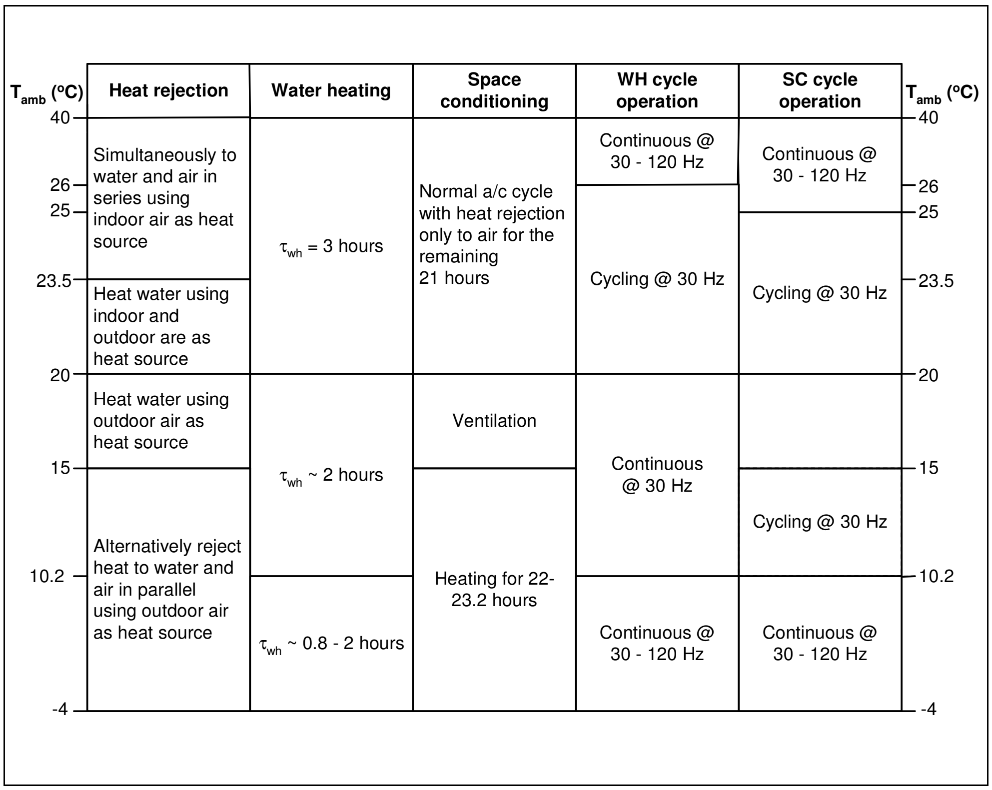

Figure 18. Summary of idealized space conditioning and water heating strategies

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ACKNOWLEDGEMENT

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