Heat pump water heaters for restaurant applications

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SYNOPSIS
Heat pump water heaters (HPWH) are ideally suited for water heating in restaurant applications, since restaurant kitchens have both a high demand for hot water and a nearly year round space cooling load. The heat pump can be used to heat the water while cooling the kitchen air. To account for the performance of the heat pump as both a water heater and an air conditioner, a fundamental model of a heat pump was developed which accounts for changes in performance caused by changes in the condenser and evaporator environments. This model was used in conjunction with a restaurant model to determine the space conditioning savings and the cost of water heating compared to conventional gas and electric water heaters.

INTRODUCTION

Heat pump water heaters (HPWH) use a vapour compression refrigeration cycle to transfer heat from the surrounding space into the hot water tank. Restaurants are a potentially attractive application of HPWH since large amounts of hot water are required for dishwashing and clean-up. Further, since the water heater is often located in the kitchen area, a HPWH may provide a beneficial air conditioning effect.

In this paper models are developed of a vapour compression cycle, a water storage tank, the combined HPWH, and of building space conditioning loads. The HPWH and building models are combined in a simulation program to calculate the annual performance and net savings of the heat pump. The performance, space conditioning load, water heating savings, and air conditioning savings or heating expense are determined for each hour of the year. Results from this simulation program are given for a typical restaurant.

The HPWH system studied in this paper is patterned after the Dairy Equipment Company (DEC) water heater HP-120-27. The HPWH is shown schematically in Figure 1. In the DEC system, the compressor is a reciprocating compressor, the first condenser is a wrap-around heat exchanger on the storage tank. The second condenser is a water-to-refrigerant counterflow tube-in-tube heat exchanger, with the cooling fluid either the inlet water to the storage tank if hot water is being drawn, or water recirculated from the bottom of the tank. The evaporator is an air-to-refrigerant crossflow heat exchanger located in the air space above the HPWH. The storage tank volume is 120 gallons [Note: Some conversion factors to S.I. units are given at the end of this paper.].

HEAT PUMP WATER HEATER MODEL

The vapour compression cycle of the HPWH is modelled by individually modelling each of the

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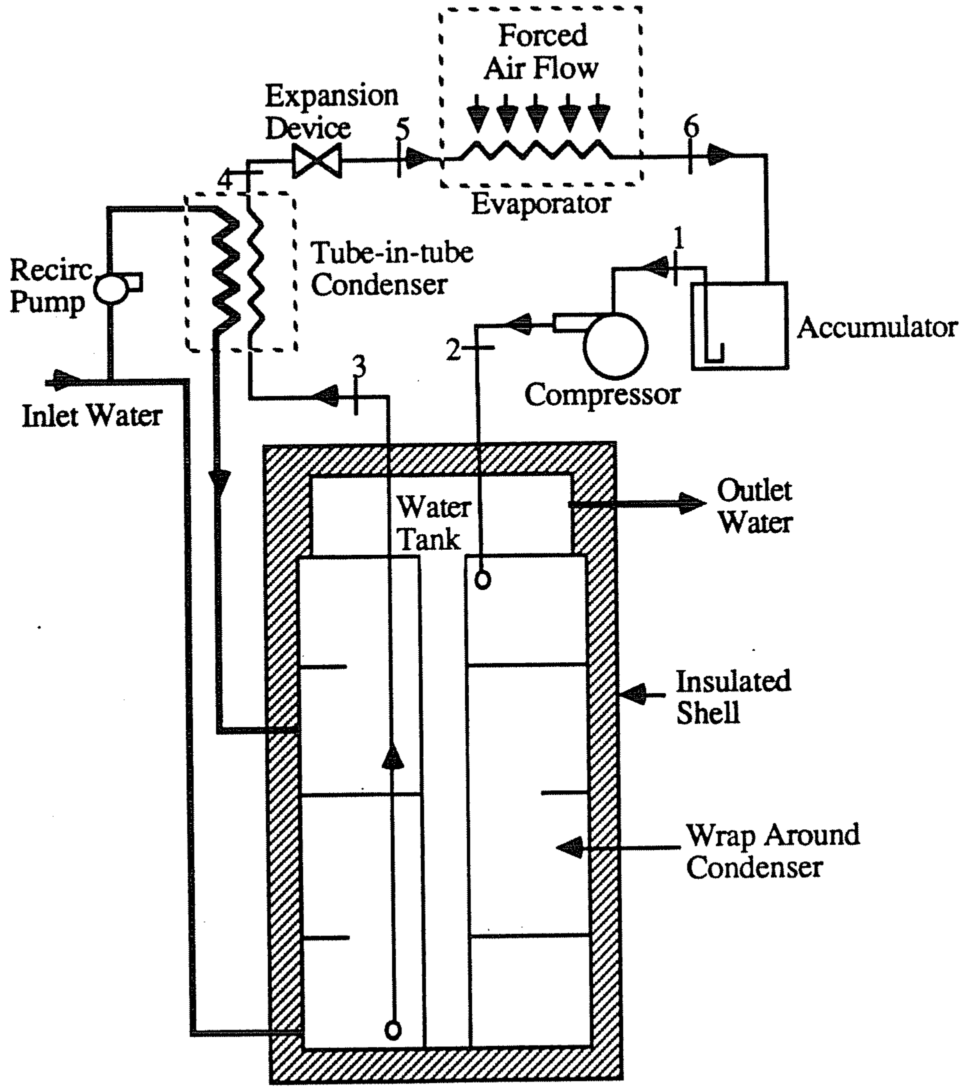

Figure 1
Schematic of the DEC HP-120-27 vapour compression cycle.

major components (compressor, condensers, expansion device and evaporator). The complete vapour compression cycle model, which is used to find the steady state performance of the heat pump, connects the component models together in order of their physical occurrence.

A hermetically-sealed reciprocating compressor is used in the DEC heat pump cycle. In developing the compressor model, the suction and discharge pressures and the inlet enthalpy of the refrigerant are taken as inputs and the refrigerant flow rate, compressor work, compressor input power, and outlet enthalpy are outputs.

The inlet refrigerant is first used to cool the electric motor and then enters the cylinder where it is compressed and discharged. The energy loss of the electric motor goes partially to heating the inlet refrigerant and partially to heating the environment. The heating effect equals the difference between the electric energy supplied to the motor and the mechanical energy supplied to the piston; it is assumed that the heat gained by the refrigerant is a fixed percentage of this difference. The DEC heat pump loses approximately 10% of its work to heating the environment [1] and has a motor efficiency of about 60%. Using these two values, the heat transferred to the refrigerant and the environment was estimated to be 75% and 25% of the motor losses respectively. The enthalpy of the refrigerant entering the cylinder is found from an energy balance on the refrigerant.

The flow rate of refrigerant through the compressor is dependent upon the inlet and outlet states of the cylinder and is described in terms of a volumetric efficiency. The ideal volumetric efficiency is given [2] by:

$$\eta_v = 1 - m\left(\frac{V_{suction}}{V_{discharge}} - 1\right) \quad (1)$$

where the clearance volume fraction is:

$$m = \frac{V_c}{V_{cyl} - V_c} \quad (2)$$

It is assumed that the actual volumetric efficiency is equal to the ideal efficiency.

The mass flow rate through the compressor is calculated by:

$$\dot{m}_r = \frac{PDR\,\eta_v}{V_{suction}} \quad (3)$$

where the piston displacement rate (PDR) is the rate of volume swept by the piston. It is assumed that the polytropic efficiency is known and constant. The polytropic efficiency is defined for an ideal gas as:

$$\eta_{poly} = \frac{\left(\frac{k-1}{k}\right)}{\left(\frac{n-1}{n}\right)} \quad (4)$$

where k and n are the isentropic and polytropic indices respectively. The isentropic index is determined from the inlet and outlet conditions and the polytropic index is estimated using this equation. Using the value of the polytropic index, the isentropic efficiency is given as:

$$\eta_{isentropic} = \frac{h_{discharge,s} - h_{suction}}{h_{discharge} - h_{suction}} = \frac{\frac{k}{k-1}}{\frac{n}{n-1}}\left[\frac{\left(P_{discharge}/P_{suction}\right)^{\frac{k-1}{k}} - 1}{\left(P_{discharge}/P_{suction}\right)^{\frac{n-1}{n}} - 1}\right] \quad (5)$$

The enthalpy difference across the compressor is calculated using equation (5). It is assumed that the loss by the electric motor is constant, and the compressor input power is calculated by:

$$\dot{E}_{motor} = \dot{W}_{comp} + \dot{W}_{losses} \quad (6)$$

The properties (temperature, pressure, enthalpy, entropy, quality, specific volume, and internal energy) of the refrigerant are calculated using a Fortran program (FREON).

The DEC heat pump condenses the refrigerant in two separate heat exchangers: a wrap-around heat exchanger and a tube-in-tube heat exchanger. The energy transfer from the refrigerant in the wrap-around heat exchanger was modelled as occurring in three separate heat exchangers: a heat exchanger which cools the superheated vapour to a saturated gas; a condenser which condenses the refrigerant to a saturated liquid; and a heat exchanger which subcools the refrigerant. The heat transfer in each sub-exchanger is calculated assuming that the refrigerant exchanges heat with the mean tank water temperature.

The heat transfer in each region is calculated using a log mean temperature difference (LMTD) approach. Since the refrigerant in each region is assumed to exchange heat with the mean water temperature in the tank, the numerator of the LMTD reduces to the difference between the inlet and outlet refrigerant temperature. In the two-phase region both the inlet and outlet refrigerant and water temperatures are equal, and the LMTD reduces to the temperature difference between the refrigerant and water.

It is assumed that the conductance is constant over the height of each heat exchanger and does not vary with flow rate. The range of refrigerant flow rates is small and the major portion of the energy transfer occurs in the two phase region where the conductance is approximately constant.

An energy balance on the refrigerant in each sub-exchanger section yields the heat transfer:

$$q = \dot{m}_r\,(h_{r,in} - h_{r,out}) \quad (7)$$

The equations describing the heat transfer in each region are solved simultaneously. Since the intermediate refrigerant states are implicit in the equations, a secant iteration method is used. The total energy delivered to the storage tank is equal to the sum of the heat transfers in the three regions.

The refrigerant exiting from the wrap-around heat exchanger enters the tube-in-tube heat exchanger where it is cooled by a counterflow of water. The entering refrigerant is either a two phase mixture or a subcooled liquid, and thus the exiting refrigerant may be either two-phase or subcooled. The heat transfer in this condenser is modelled as a condensing heat exchanger and a subcooling heat exchanger. Since the outlet and intermediate water temperatures are unknown, the effectiveness method instead of the LMTD method is used. The heat transfer between two streams a and b is given by:

$$q = \varepsilon\,(\dot{m}\,c_p)_{min}\,(Ta_{in} - Tb_{in}) \quad (8)$$

where the effectiveness relation for a single-pass counter-flow heat exchanger is employed [3]. The solution is iterative since the intermediate water temperature and refrigerant state are unknown and implicit in the equations; a secant iteration method was used to obtain a solution.

The fluid exiting the condenser is a subcooled liquid. The pressure drop in the tube between the inlet and the length at which the fluid becomes a saturated liquid is determined by a momentum balance on the fluid. A constant enthalpy expansion process was assumed. The quality and enthalpy at that length are equal to zero and the inlet enthalpy respectively. These two properties completely define the thermodynamic state, and therefore, the pressure can be determined. The steady state pressure drop in the two phase portion of the tube is calculated using a finite difference solution to the mass, energy, and momentum equations as outlined by Stoecker and Jones [2].

The DEC evaporator is a forced-air crossflow heat exchanger. It is assumed that the entering refrigerant is a two phase mixture, but that it may leave as either two phase or superheated. The evaporator can be viewed as two heat exchangers, an evaporator and a conventional heat exchanger. It is assumed that the moisture in the air stream does not condense on the outside surface of the evaporator. This is reported to be true in practice except for a few summer months. The evaporator model, therefore, assumes a completely dry exterior surface for all conditions, with the effectiveness for a crossflow heat exchanger given by Kays and London [3].

The component models are combined to determine the steady-state performance of the HPWH system. The model calculates the performance of each component in the order in which the refrigerant flows. The solution is iterative, and a combination of a secant method and a half interval method is used.

Input values were determined from catalogue information. Six measured data points of the performance of the HPWH were available from DEC. The input values which were found to give the best comparison to these test data are given in Table 1. These input values were found to predict the measured heating capacity and required power to within 3%. Table 2 shows the comparison of the predicted condenser and evaporator pressures to data. The model closely predicts the pressures for the higher water temperatures, but is off by up to 15% for the lower temperatures.

Table 1 Input values to vapour compression cycle model.

Variable	Units	Input
Refrigerant		22
m		0.08
PDR	ft^3/s	0.0458
η_{poly}		0.8
W_{losses}	Btu/h	2350
$(UA)_{WA}$	Btu/h-°F	450
L_{TT}	ft	14.5
$(UA)_{TT}/L$	Btu/h-ft-°F	140
m_{H_2O}	lbm/h	2000
A_{CT}	ft^2	0.0000191
L_{CT}	ft	2.5
$(UA)_{evap}$	Btu/h-°F	380
m_{air}	lbm/h	5625

Table 2 Comparison of condenser and evaporator pressure to data.

Average tank temperature (°F)	Condenser pressure actual/ predicted (psi)	Evaporator pressure actual/ predicted (psi)
136.3	370/361	98/97
128.7	340/331	95/93
112.3	285/272	88/87
95.2	235/219	85/81
76.7	191/170	83/77
57.4	152/129	75/73

The vapour compression cycle model is time-consuming, and it is impractical to incorporate the HPWH model into an annual simulation program which requires that the heat pump performance be calculated at least once an hour for each hour of the year. To allow rapid simulation, performance values were generated over a range of operating conditions, and curve fitted using regression analysis [4].

Since the environmental conditions at the evaporator are nearly constant year-round, the performance of the heat pump is only affected by changes in the water temperatures in the two condensers. The heat pump performance data of interest are the heat transfers in the heat exchangers and the compressor input power. The model was used to generate data over a range of condenser water temperatures.

The motor power was curve fitted using a quadratic expression:

$$\dot{E}_{motor} = a_1 + a_2 T_{TT} - a_3 T_{TT}^2 + a_4 T_{TT} T_{WA} \quad (9)$$

The two heat transfers, q_{TT} and q_{WA}, were fitted by:

$$q = b_1 3 b_2 T_{TT} + b_3(T_{WA} - T_{TT})^{-1/2} + b_4 T_{TT}^2 + b_5 T_{WA} + b_6 T_{TT} \quad (10)$$

where the coefficients are given in Table 3. These equations all have R^2 values ranging between 0.995 and 0.998 and are therefore good fits to the generated data.

A schematic of the storage tank is shown in Figure 2. In addition to the heat pump heater, the unit also has two 6 kW electric resistance back-up heaters. The wrap-around heat exchanger of the heat pump covers approximately the bottom 85% of the tank. When hot water is being drawn from the tank, an equal amount of cold water is supplied to the tank. While the heat pump is operating, the recirculation pump draws water

Table 3 Coefficients to HPWH performance relations.

Coefficient	E_{motor}	Coefficient	Q_{WA}	Q'_{TT}
a_1	1980	b_1	2880	13940
a_2	29.3	b_2	55.0	−22.5
a_3	−0.0214	b_3	264	−259
a_4	0.01607	b_4	−0.1005	−0.258
		b_5	−37.2	38.7
		b_6	0.0484	−0.0830

through the tube-in-tube heat exchanger to heat the water.

The HPWH is designed to supply the coolest water available to this heat exchanger. When the heat pump is operating and hot water is being drawn, the tube-in-tube heat exchanger uses the cold supply water; otherwise the tube-in-tube uses water from the bottom of the tank. For low hot-water draw rates, a combination of the supply water and water drawn from the bottom of the tank is used to meet the required flow for the tube-in-tube heat exchanger. Each heater in the tank has an individual controller, located near the top of the tank for the two back-up heaters and at the bottom of the tank for the heat pump heater.

The water storage tank is modelled as a stratified tank by dividing the tank into several horizontal sections, or nodes [5]. Each node is modelled as fully mixed and accounts for heat addition by heaters, for losses to the environment, and for energy exchanges between adjacent nodes caused by water entering and leaving the node. The water temperature in any node is less than the water temperature in the node above it. Therefore, the model assumes that energy supplied to the tank by heaters is added into the node specified as having that heater, until the temperature of that node is equal to that of the node above it. These nodes are then considered to be mixed, and energy is added to them equally.

A three-node tank model was chosen. The first node is the bottom of the tank and includes the inlet and outlet water from the tube-in-tube condenser, lower back-up element, lower portion of the wrap-around condenser, heat pump controller, and cold water inlet. The second node

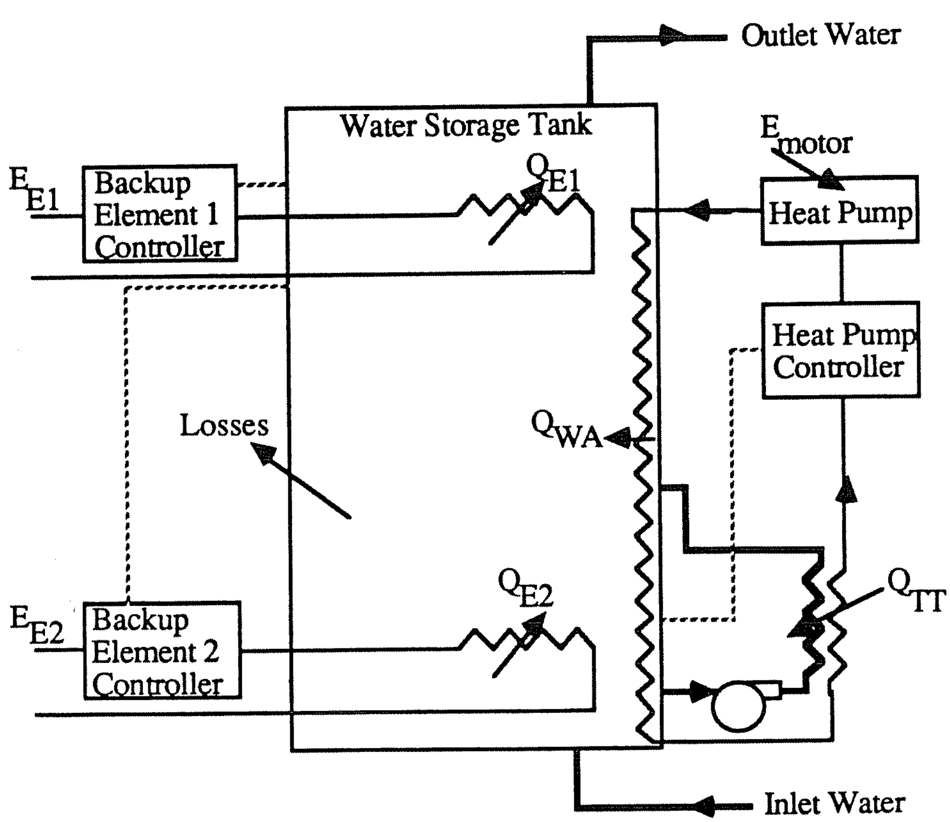

Figure 2
Schematic of the HP-120-27 HPWH. Water lines are shown in bold lines, refrigerant in other lines.

is the middle of the tank and includes the top portion of the wrap-around condenser and the controller for the bottom back-up element. The third node is the top of the tank and includes the top back-up element, the controller for the top element, and the hot water outlet.

It was not possible to check the accuracy of this stratified tank model due to the limited amount of data available. The performance of the heat pump increases with increased stratification, and therefore, the performance would be better than predicted if the stratification is being under-predicted.

The mean water temperature of the wrap-around condenser is equal to the average temperature of nodes 1 and 2. The temperature of the water supplied to the tube-in-tube condenser depends upon the operating conditions: it is equal to the supply water temperature if hot water is being drawn, it is equal to the bottom node temperature if hot water is not being drawn, and it is equal to the flow weighted temperature of these two temperatures if hot water is being drawn but at a rate less than the water flow rate through the condenser.

The overall coefficient of performance of the HPWH is given by:

$$COP_{HPWH} = \frac{Q_{HPWH} + E_{heater1} + E_{heater2}}{E_{motor} + E_{heater1} + E_{heater2}} \qquad (11)$$

The measured performance data of the unit was obtained during heating of a tank of water from 57.4°F to 136.3°F, using only the heat pump heater [1]. The model predictions were within 1% of the test data.

RESTAURANT MODEL

A model of a "fast food" restaurant was developed to predict cooling and heating loads. The building model developed calculates hourly space-conditioning requirements considering effects such as inside and outside temperatures, envelope losses/gains, ventilation, infiltration, and internal gains caused by cooking, people, and electric lighting and appliances. Data from several restaurants of monthly electric, gas, and water usage from several restaurants were used to estimate the magnitude of the heating, cooling, and hot water loads.

The model restaurant is assumed to be open for 18 hours per day, opening at 6 o'clock in the morning and closing at 12 o'clock in the evening. The building is assumed to have one heating zone that is perfectly controlled to a specified temperature. It is assumed that the HPWH has a negligible effect on the latent load. The input values for the base case building model are summarized in Table 4.

Whether the HPWH cooling is a saving or an expense is dependent on when the building load occurs in the day relative to the hot water load. In

Table 4 Inputs for base case building model.

Variable description	Units	Input value
% oven gains	(0 to 1)	0.2
% electric gains	(0 to 1)	0.3
$(UA)_{Bld,overall}$	Btu/h°F	800
Ventilation	CFM	1000
Inside temperature	°F	75
η_{furn}	(0 to 1)	0.6
η_{GWH}	(0 to 1)	0.55
C_{kWh}	$/kWh	0.0613
C_{Therm}	$/Therm	0.43

a restaurant the building load is largely controlled by the internal gains, and therefore, the modelling of hot water draw and internal gains are important considerations. If hot water is only needed during hours when there are no gains, then the HPWH cooling is generally a heating expense; otherwise it is generally an air conditioning saving. In general, however, fast food restaurants' gains are fairly uniform during the period that they are open.

Hot water use schedules having one, two and three periods of uniform draw throughout each day were studied. It was determined that the performance was insensitive to the number of periods of draw, but sensitive to the rate at which the water is drawn. The draw schedules considered in this paper have two periods of uniform draw with at least two hours of no draw between them to allow the storage tank to be completely reheated. Various draw rates are investigated, two of which are shown in Figure 3, along with the building gains schedule. In each case the first draw period ends at 3 p.m. and the second draw period ends at 2 a.m. and the total daily hot water draw is 500 gallons. The building's electric gains and oven gains are evenly distributed over the open hours (7 a.m. until 1 a.m.).

PERFORMANCE INDICES

The HPWH and building models are combined in an annual simulation program which performs hourly calculations of the building's space conditioning load and the HPWH performance. For each hour the program determines whether the space cooling produced by the heat pump is an air conditioning saving or a heating requirement. The HPWH cooling is an air-conditioner saving if the net cooling load energy is greater than the cooling. The amount of economic saving depends upon the cost of electricity (C_{kWh}) and the coefficient of performance of the air conditioner for that hour, and is calculated by:

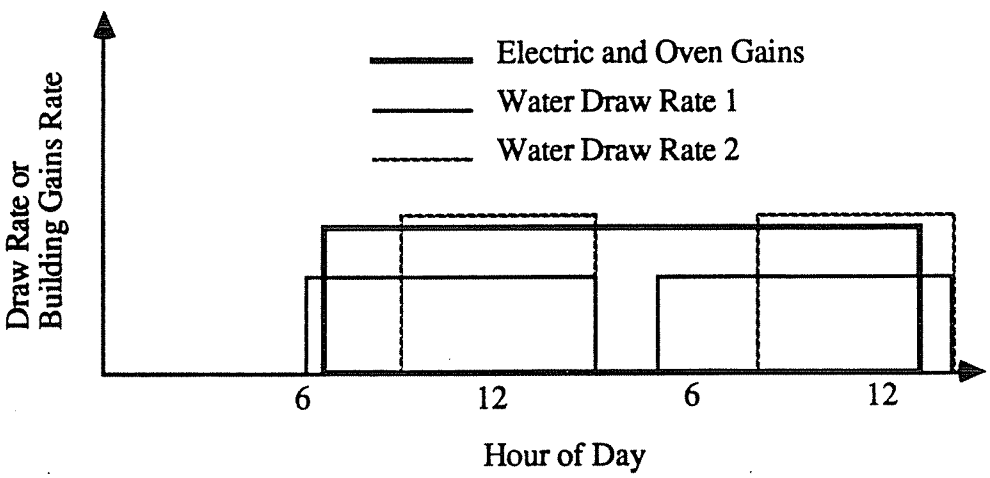

Figure 3
Water draw schedule and building electric and oven gains schedule, which show the relative positioning of the schedules.

$$Sav_{A/C} = \frac{Q_{A/C}}{COP_{A/C}} C_{kWh} \qquad (12)$$

The COP of the air conditioner varies with changing temperature and is calculated using a fixed percentage of the Carnot COP assuming fixed temperature differences across the evaporator and condenser. A typical summer COP predicted by this method is between 2.5 and 3.

The HPWH cooling is a heating expense if the net building energy is negative. The expense associated with this cooling depends upon the efficiency of the furnace and the price of the heating fuel, and is calculated by:

$$Exp_{SH} = \frac{Q_{A/C}}{\eta_{furn}} C_{Therm} \qquad (13)$$

The furnace efficiency is assumed to be a constant 60%. The annual space conditioning saving associated with the HPWH cooling is determined by summing all the hourly savings and expenses throughout the year.

In addition to determining the air conditioning saving, the water heating savings of the HPWH compared to electric or gas water heaters were determined. The annual water heating savings of the HPWH compared to an electric water heater is calculated by:

$$Sav_{WH} = C_{kWh}\left[Q_{H_2O} - \int \dot{E}_{motor}\, dt\right] \qquad (14)$$

For a gas water heater the savings depend upon the thermal efficiency of the heater and the savings are calculated by:

$$Sav_{WH} = \frac{Q_{H_2O}}{\eta_{GWH}} C_{Therm} - C_{kWh} \int \dot{E}_{motor}\, dt \qquad (15)$$

The total savings of the HPWH relative to the electric or gas water heaters is the sum of the space conditioning and water heating savings.

The electricity and gas charge rates used were for small commercial users (i.e. restaurants) of \$0.061/kWh and \$0.43/therm, respectively.

Annual simulations of the restaurant together with the various water heating alternatives were performed for the Madison location. The annual cooling and heating savings compared to operation with electric and gas water heaters were determined. In addition to the base case, a number of variations were studied to determine the sensitivity of the results to alter parameters.

For the base case, the daily total savings of the HPWH, when compared to an electric resistance heater, range from between \$2 and \$4, and amount to between \$700 and \$1400 annually. The variation in the total daily savings is due to changes in the amount of the HPWH cooling that is useful, which depends on the internal gains. The air cooling produced by the HPWH is a net air conditioning saving for about 7 months of the year, but is a net heating requirement the rest of the year.

The HPWH compared to a gas water heater is found to be economic only during the summer months. The daily total savings of the HPWH is approximately \$0.60 during the summer, but a daily loss of about \$1.20 during the winter. Over the year the total saving compared to gas is about a \$30 loss.

The effect of water draw rates other than 36 gallons/hour was also evaluated. For each new draw rate, the period over which water is drawn was also changed so that the total daily draw was the same as the base case (500 gallons/day).

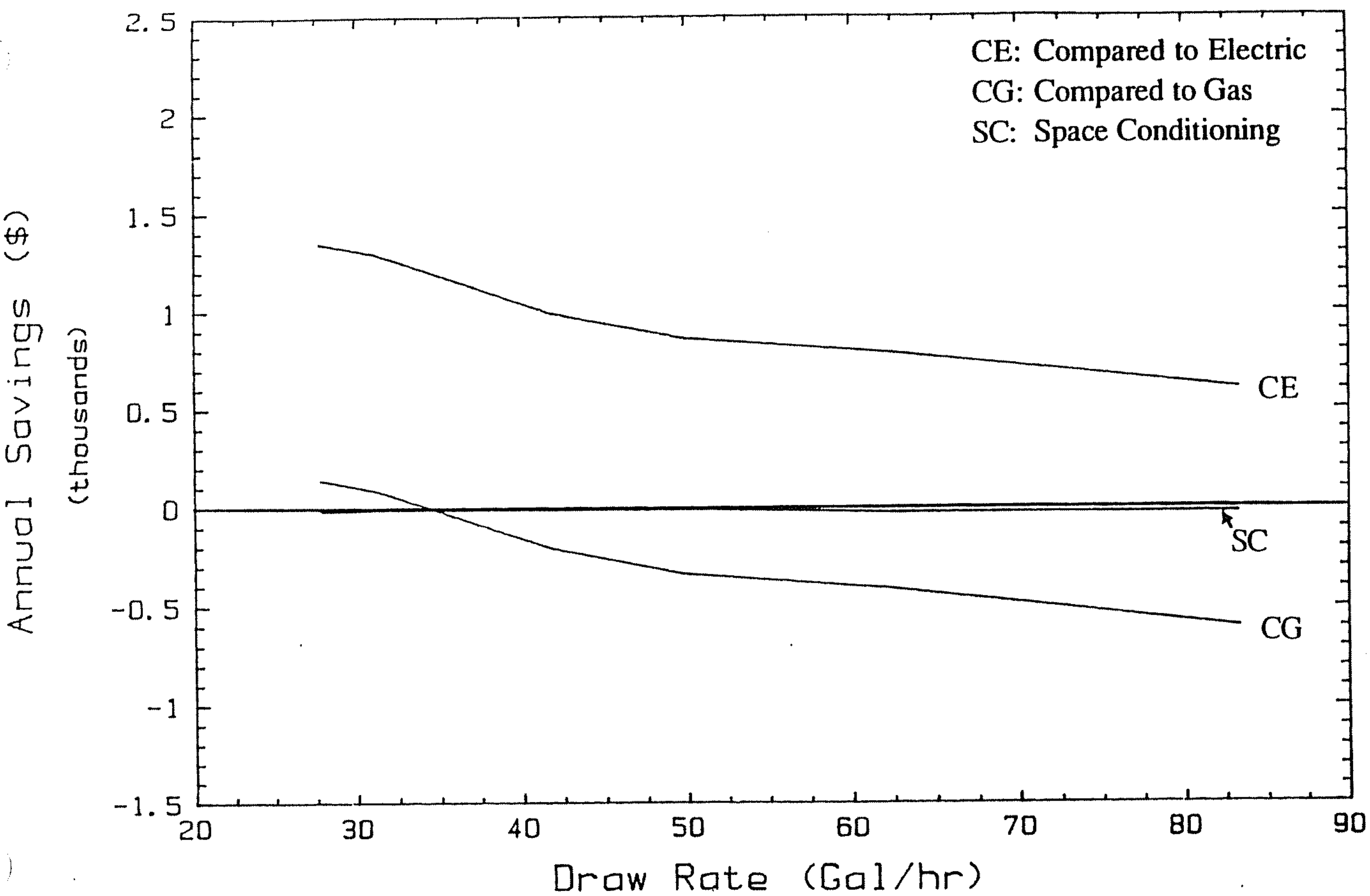

Figure 4 Annual savings versus hot water draw rate for the base case conditions.

Figure 4 is a plot of the predicted HPWH annual savings versus hot water draw rate. This plot shows that for the base case building the HPWH is more economical than the electric water heater for all draw rates, and that the gas water heating is less costly than the HPWH at low draw rates, but more costly at high draw rates. In practice, the HPWH is sized so that the back-up elements rarely turn on. This is equivalent to low flow rates so that the HPWH is the least expensive.

For the base case building, the air conditioning savings produced by the HPWH are approximately equal to the additional heating expense, and therefore, the air cooling of the HPWH has a negligible contribution to the total savings. For restaurants having larger space conditioning loads than the base case, the HPWH space conditioning savings will be larger.

The effects of different HPWH and building model parameters on annual savings were examined. Table 5 summarizes the results. There is a large effect of internal gains on the savings. With greater gains, there is increased benefit to the air conditioning, and the HPWH system is economic. Similarly, locations with a high air conditioning load (e.g. Nashville, Miami), allow a better performance of the HPWH.

The gas water heater efficiency significantly affects the relative savings, with high efficiency water heaters reducing the energy savings. Increased storage tank volume somewhat increases savings.

Increased space temperatures allow the HPWH to operate more efficiently, and produce lower costs for the same water heating. The reduced compressor power results in increased air conditioning effect. Increased temperature which might be achieved by positioning the evaporator near a heat source such as an oven also increases the savings.

In comparing HPWHs to conventional water heaters, two additional items which may affect the savings comparison are electric demand charges and construction requirements for gas water heaters. These were not thoroughly investigated since they did not apply to the restaurants being considered in this study. In locations where restaurants have electrical demand charges, the savings of the HPWH compared to electric water heaters would increase because of the decreased

Table 5 Effects changing various parameters have on annual savings.

Change	Average increased savings ($)		
	Comp. to gas	Comp. to electric	Net A/C
50% of base case gains	−110	−110	−110
150% of base case gains	110	110	110
60% gas heater efficiency	−130	0	0
45% gas heater efficiency	200	0	0
85°F evap. air temperature	140	140	0
50% of base case storage	−70	−70	0
150% of base case storage	70	70	0
Miami weather	225	225	225
Nashville weather	100	100	100

electrical demand, while the savings compared to gas water heaters would decrease.

Some states require that gas water heaters be in fireproof rooms. The additional construction expense of this room could be avoided by using a HPWH instead of the gas water heater. This saving would only apply when gas and electric water heaters are being considered for new construction; it would not affect the comparison when considering replacement water heaters for restaurants already having a fireproof room.

Gas and electric water heaters having the same heating capacity as the HPWH currently cost about $1300, whereas the HPWH costs about $3000. When considering the installation of HPWH instead of conventional water heaters, it is the incremental cost of the HPWH that is important in an economic comparison. The payback period is the time needed for the cumulative fuel savings caused by using a HPWH instead of a conventional water heater to equal the total initial investment (i.e. selling price). The economic parameters used in this study are given in Table 6. For an incremental equipment cost of $1700, the annual saving required to give a three year payback are calculated to be $850 [6].

In Figure 4 it was seen that the annual savings of the HPWH compared to electric water heaters range between $600 and $1400, and between $150 saving and a $600 loss compared to a gas water heater. For the base case conditions, the HPWH compared to an electric water heater could have a three year payback, but would not pay back compared to a gas water heater. If the HPWH is to replace an existing water heater for efficiency purposes only, the equipment cost is equal to the price of the HPWH. For an equipment cost of $3000, the required annual saving to give a three year payback is calculated to be $1500. For the base case it is unlikely that either the gas or electric water heaters could have a three year payback. These calculations neglect electric demand and construction savings. In restaurants where these savings occur, the payback period would decrease.

Table 6 Economic parameters used in this study.

Variable	Value
i	4%
d	8%
t_i	Federal 34%
	State 8% (WI)
	Total 42%
t_p	2.7%
N_d	5 years

CONCLUSION

In a restaurant, heat pump water heaters have the potential to both heat water and provide air conditioning. However, if air conditioning requirements are low, or occur at times during which hot water heating is required, the air conditioning effect may offset the purchased air conditioning requirement.

Compared to electric hot water heaters, HPWH saves money annually for all of the conditions considered here. The amount saved pays for the HPWH within three years. However, if gas water heaters are the alternative, the HPWH is a marginal investment.

NOMENCLATURE

C_{kWh}	electricity cost per kWh.
COP	coefficient of performance.
c_p	specific heat at constant pressure.
C_{Therm}	gas cost per Therm.
E	input energy to the HPWH motor or back-up elements.

$\dot{E}$ rate of energy input to motor.
Exp additional space heating expense.
h enthalpy
k isentropic index.
$\dot{m}$ mass flow rate.
m percent clearance in the piston cylinder assembly of the compressor.
n polytropic index.
P pressure.
PDR piston displacement rate of the compressor.
q heat transfer rate.
Q total heat transfer.
Sav annual economic savings.
t time.
V_c clearance volume.
V_{cyl} cylinder volume.
W work.

Greek

ε heat exchanger effectiveness.
η_{furn} furnace efficiency.
η_{GWH} gas water heater efficiency.
$\eta_{isentropic}$ isentropic compressor efficiency.
η_{poly} polytropic efficiency in the compression process.
η_v volumetric efficiency of the compressor.
v specific volume.

Subscripts

A/C air conditioning.
Comp compressor
discharge. discharge from the compressor.
HPWH heat pump water heater.
H_2O water.
heater auxiliary water heater.
in inlet to a component.
losses heat losses from the compressor motor.
motor compressor motor.
out outlet to a component.
r refrigerant.
s outlet condition if the process took place isentropically to the same pressure.
suction inlet to the compressor.
TT tube-in-tube condenser.
WA wrap-around condenser.
WH water heating.

INTERNATIONAL SYSTEM OF UNITS

Some conversion factors.

Temperature °F − 32 = 9/5°C.
Volume 1 US gallon = 3.785 litres.
Energy 1 Btu = 1.055 kJ (1 Therm = 10^5 Btu).
Pressure 1 psi = 6.894 kPa (1 bar = 10^5 Pa).

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