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System Design Optimization and Validation for Single-Speed Heat Pumps

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ABSTRACT

Computer models for predicting the steady-state and seasonal performance of electric driven, air-source heat pumps are described. Laboratory data on the heating and cooling performance of two heat pumps (under steady-state conditions) are compared with model predictions. Measurements of the seasonal and annual energ consumption for an electric heat pump installed in an unoccupied home in Knoxville, TN, are presented and also compared with model predictions.

A method for combining the steady-state and seasonal performance calculations and placing them under the control of a numerical optimization procedure is described. This combined code was used to determine the design of a heat pump which minimized annual energy consumption and yet met a design cooling capacity and had fan and compressor efficiencies and total heat exchange surface area comparable to one of the heat pumps used for the laboratory tests. The optimization results indicated a 16% reduction in annual energy consumptions but it was determined that about half of the energy savings was due to assumptions or the steady-state calculations. The true energy savings were found to result from reduced fan powers and lower cycling losses. Similar analyses were performed for a central air-conditioner and a heat pump with idealized refrigerant flow control.

INTRODUCTION

Recently, more use is being made of computer models for air-source heat pumps, in conjunction with numerical optimization software, to aid in the design of advanced, high efficiency heat pumps [1-7]. In most cases the predictions have had to stand on their own although at least one project has led directly to the development and testing of a prototype unit [8].

This paper begins with a brief overview of the computer models that were used in the present study (i.e., the steady-state heat pump simulations and the seasonal and annual performance calculations) and includes some validation of each of them. It also gives a general description of the optimization techniques that were applied to the heat pump calculations. Finally, results are presented for three system design optimizations and reasons are discussed for the predicted improvements in heat pump performance.

STEADY-STATE HEAT PUMP MODEL

The steady-state performance of single-speed heat pumps was simulated using a standard vapor compression "cycle analysis" model [9,10]. The program uses data about the components (e.g., compressor maps, fan efficiencies, heat exchanger geometry, and sizes) to predict heating or cooling capacity and electrical input to the heat pump under a given set of indoor and outdoor temperatures and relative humidities. The heat exchangers are modeled using simple circuiting assumptions and effectiveness versus NTU (Number of transfer units) relationships for cross-flow heat transfer. A relatively simple model is used for indoor fan power calculations. The outdoor fan power is computed using a representative curve of static efficiency versus specific speed for a propellor fan. Curve fits for the data on manufacturers' compressor maps were used to compute the refrigerant mass flow rate and compressor power consumption given the refrigerant saturation temperatures at the shell inlet and outlet. Information about the indoor air ducts and the refrigerant lines was supplied as part of the input data.

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There is one potentially significant limitation of the current steady-state model -- the lack of refrigerant charge inventory calculations. Without such refrigerant mass balance capability, any interaction effects between the refrigerant charge and the system operating conditions (the latter usually represented by low-side super-heat) cannot be accounted for.

For certain types of equipment and design analyses, these effects can be considered secondary and are satisfactorily ignored. The steady-state model contains the implicit assumption that the refrigerant contains an accumulator (low-side receiver) and that there is sufficient refrigerant charge and accumulator volume so that the accumulator is always partially filled [3,9,10]. With this assumption, a low degree of superheat is maintained at the compressor inlet (a desirable ideal condition) and the superheat level is thus uncoupled from refrigerant flow control and charge mass balance consideration.

However, for systems with no accumulators or with insufficiently large accumulators, the current model is generally inadequate. The model can only be used in these cases for validation tests with the low-side superheat specified a priori from exerimental measurements or design specifications. Consequently, steady-state design calculations cannot be performed away from design or validation points for heat pumps that use superheat-controlled expansion devices such as thermostatic expansion valves (TXVs) or for cases where the accumulators run dry at some operating conditions. Possible effects of these limitations are discussed later with regard to the design optimization results.

This model has been validated using laboratory data on two different heat pumps; a low first cost (low-bid) unit of moderate efficiency and a high efficiency heat pump [10,11]. These results are summarized in Table 1. The compressors in the heat pump studied in the laboratory and that for the published compressor map performed somewhat differently. The COP and capacity comparisons at 17F and 47F would be even closer if corrections were applied to compensate for discrepancies between the published refrigerant flow rates and power consumptions and those actually observed at the same compressor conditions.

The cooling performance comparisons on the one heat pump are not as good as we would like, but the poorer agreement can be explained for the most part. The heat pump in the laboratory ran in cooling mode with more than 75% of the evaporator filled with superheated vapor (a rather inefficient condition). The heat exchanger models used contain assumptions about the extent and location of the single-phase region in the heat exchangers relative to the two-phase region that were not applicable to the unit tested in cooling mode. The models assume that such inefficient conditions would not occur normally; the operating assumptions used in the analysis and consistent with these assumptions. Consequently the program cannot do a good job modeling the heat exchangers under those conditions.

ANNUAL PERFORMANCE MODEL

The Annual Performance Model that was used to predict seasonal and annual energy consumption is being documented and will be the subject of a future report [12]. It uses a "bin type" of analysis to apply steady-state heat pump performance data to given or calculated house heating and cooling loads. It uses empirical degradation factors to account for cycling, frosting, and defrosting losses [13-15]. House loads are calculated by a program developed by Ballou et al. [16] and have been validated against detailed field measurements on one house located in Knoxville, Tennessee.

The Annual Performance Factor (APF) Model has also been validated against two sets of field measurements. These data are for the same building and heat pump in Knoxville, Tennessee, but for two different heating and cooling seasons (i.e., 79-80 and 82-83). In the intervening years, changes were made to the heat pump at the test site that altered its steady-state performance. Consequently two different sets of measured heat pump steady-state data were used as input to the APF program for the validations. Degradation factors presented by Parken and Kelly [13] and defrosting data obtained by Baxter [17] were used to account for frosting and defrosting losses. Cycling losses were computed using the manufacturer's DOE Cd degradation factors. Measured, rather than calculated, house loads were used for the APF validations. These results are summarized in Table 2.

LOAD AND COMPONENT ASSUMPTIONS

The APF and SPF calculations that were performed for this study, as well as an earlier one [6], used data for an average weather year for Nashville, Tennessee and building data for a light-frame HUD minimum insulation, 1800 square foot (167 m^2), ranch-style house.

Some "basic" assumptions were made about the heat pumps to be optimized in these studies in order to place some engineering constraints on component sizes and efficiencies. First, compressor performance was based on the maps for a three-ton compressor used in a high efficiency heat pump by a major manufacturer. It was assumed that the total displacement of the compressor could be scaled up or down, to simulate different capacities, without altering the overall isentropic and volumetric efficiencies. The indoor fan was modeled assuming a constant, combined blower/blower-motor efficiency of 22%. The outdoor fan was modeled using a motor efficiency of 55% and a fan static efficiency curve (as a function of specific speed) that gave a potential maximum combined efficiency of 25%. And finally, the "total heat exchanger area" was fixed at 13 square feet per ton of design cooling capacity (at a 95F (35°C) ambient). The "total heat exchanger area" was taken to be the sum of the frontal areas times the number of refrigerant tube rows for the indoor and outdoor heat exchangers. This value was proportional to true total heat exchanger area because the internal heat exchanger geometries were held fixed at a major manufacturer's production value.

The steady-state heat pump model and the APF model discussed earlier were converted into subroutines that could be invoked by an "external" program (see Figure 1). In most cases, that "external" program was a numerical optimization subroutine. The "optimizer" would assign some combination of heat pump design variables (in the lowermost block in Figure 1) and call the APF model. The APF model would in turn use the steady-state model to calculate the COP and capacity for a heat pump with that combination of components for each of four different outdoor ambient temperatures (17F (-8.3°C) and 47F (8.3°C) for heating performance curves and 82F (28°C) and 95F (35°C) for cooling curves). The APF model then used linear interpolations and extrapolations from the calculated steady-state data along with the prescribed house loads and dynamic loss factors to calculate an annual energy consumption and APF (frosting and defrosting losses were computed using data presented by Parken and Kelly [13] and cycling losses were computed based on a C_d of 0.24). The APF and design cooling capacity were then passed back to the optimization code for comparison with corresponding values for different heat pump configurations.

OVERVIEW OF THE OPTIMIZATION PROCEDURE

The optimization process in general is a systematic approach to modifying the independent variables in a function, or model, to maximize or minimize the function. The search algorithm used in this study is similar to the standard method of steepest descent (or ascent). It also allows the user to impose constraints or design criteria that must be satisfied for a combination of the design variables to be acceptable or feasible. These constraints can be nonlinear functions of the independent variables.

Figure 2 is an illustration showing a surface in three dimensions as a function of two independent variables and also a projection of that surface into a two dimensional contour plot. The optimization routine would try to find the highest point on the perspective plot (or correspondingly the center of the highest projected contours in two dimensions) by following the steepest path to the top (this corresponds to following a path perpendicular, or orthogonal, to the contours). This path, or search direction, is found by starting at some initial guess for the independent parameters, taking a small step in each coordinate direction, and examining the "rise" or "fall" of the surface. This provides the path of steepest descent and the current estimate of the "best" combination of independent variables is "moved" this direction until it is deemed worthwhile to reassess the search direction. This corresponds to following a straight path uphill for a while and then looking around again and seeing if there is a more direct, steeper, way to the top.

A redefinition of the goal, or objective, of the optimization process is shown in Figure 3. In this case, a qualification, or constraint, has been placed upon the process. A dashed line representing a constraint function of the design variables has been superimposed on the surface, and now, instead of trying to find the highest point of the surface, the goal is to find the highest point on this curve. This corresponds to the point in the contour plot where the projected constraining line is tangent to a contour line (shown by the dotted contour). The search direction, or path, is thus constrained to follow this line (typically, the search follows close to the constraint but not right on it, and a periodic adjustment is made after it "strays" too far to bring it back to a "feasible" point).

APPLICATION OF OPTIMIZATION TO HEAT PUMP DESIGN

The goal of the optimizations in these studies was to find the combination of heat pump components that gave the best annual performance factor (i.e., maximum APF or equivalently the lowest annual energy consumption). The constraint that was imposed was the requirement that design cooling load at a 95F (35°C) ambient be satisfied in order for the combination of components to be a feasible set.

Three different optimizations were performed:

— a heat pump with capillary tubes for expansion devices,

— a heat pump with idealized flow control, and

— a central air conditioner with a capillary tube for an expansion device.

The predicted performance levels are compared to base case systems and some of the differences in design and performance between the optimized systems were explored.

A concerted effort was made to compare the optimized heat pumps with an existing high efficiency unit on as fair a basis as possible. The base system naturally available as the starting point for the analysis was the high efficiency unit that was tested in the laboratory. It had been used for the steady-state validation and represented the state of the art performance of single-speed heat pumps reasonably well. This "base case" heat pump was a three-ton unit, though, and was not the best match for the assumed house loads. These loads had been used in another study and it was felt that they represented a more typical house size than would the loads appropriate for the heat pump in the laboratory. Consequently, the components of the base case heat pump were scaled linearly to approximate a comparable 2.25 ton (7.9 kW) heat pump that matched the loads.

This scaling was done in a manner that:

— maintained the compressor efficiency levels but altered its capacity,

— held the ratio of total heat exchanger area to the systems rated size constant at 13 sq. ft./ton of cooling at 95F (0.34 m^2/kW at 35°C), and

— kept the fan and blower efficiencies unchanged.

This effectively reduced the steady-state capacity of the heat pump at each ambient by the ratio of the desired design capacity (2.25 tons (7.9 kW)) to the design capacity of the base case unit (3 tons (10.6 kW)) without changing the COP at each ambient.

Ten to twelve design parameters were controlled by the optimization routine. These were:

— compressor displacement,

— number of refrigerant circuits for each heat exchanger,

— number of refrigerant tube rows for each heat exchanger,

— volumetric air flow rate for each heat exchanger,

— fraction of total heat exchanger area used for the indoor coil, and either

— capillary flow factors for heating and cooling operation, or

— coefficients for specified condenser subcoolings as linear functions of the outdoor temperature for both heating and cooling (two coefficients each for heating and cooling).

The frontal areas of each heat exchanger were calculated using Equations 1 and 2 where $A_{x,indoor}$ and $A_{x,outdoor}$ are the frontal areas of the indoor and outdoor heat exchangers, N_{indoor} and $N_{outdoor}$ are the number of tube rows in each heat exchanger, and F is the fraction of the total heat exchanger area used for the indoor coil. The constant 29.25 is the assumed total available heat exchanger area.

$$A_{x,indoor} = 29.25F/N_{indoor} \tag{1}$$

$$A_{x,outdoor} = 29.25(1.0-F)/N_{outdoor} \tag{2}$$

Each of the various design parameters was held within a reasonable specified range so that during the optimization process they would not become either too large or too small and make the resulting heat pump configuration impractical.

RESULTS

All of the results for the heat pump analysis are given in Tables 3 and 4. Table 3 lists the components for the scaled base case, the optimized heat pump with capillary tubes, and the optimized central air conditioner. The calculated power consumptions and performance factors for the scaled base case and the optimized heat pumps are in Table 4. The calculated heating and cooling seasonal performance factors and the annual performance factors are given in the first three rows of Table 4. The remainder of the table contains seasonal and annual energy consumption breakdowns in kilowatthours for

— the steady-state operation of the heat pumps; how much energy would be used if the heat pump could run at its steady-state COP and capacity with no dynamic losses,

— the cycling losses -- how much additional energy is required due to the COP degradation and on times resulting from on/off cycling of the heat pump,

— frosting and defrosting losses -- how much additional energy is required because of degraded COPs and capacities in the frosting region and energy needed during the defrost cycle, and

— the resistance heat used below the balance point.

The five columns of data in Table 4 are calculated quantities for the different heat pumps and variations, which are

— the high efficiency heat pump tested in the laboratory using the measured steady-state COPs and capacities (capacities were scaled to meet the design cooling load at 95F (35°C)),

— the high efficiency heat pump from the first case but with calculated steady-state performance using the measured superheats at the compressor inlet (capacities were again scaled),

— the high efficiency heat pump but this time with specified low compressor inlets superheats,

— the optimized heat pump configuration with specified low superheats and optimized capillary tubes for flow control, and

— the optimized heat pump configuration with optimal subcool flow control and specified low evaporator superheats.

The differences between the first two columns reflect the effect of computing steady-state data with the heat pump model. There is a difference of about 600 kWh. Much of this is due to differences in the estimated resistance heat requirements, 480 kWh, due to heating capacity under predictions and the rest is in the steady-state and loss factor terms.

The assumption that there is a low level of compressor inlet superheat has a significant effect on the annual power consumption and performance factors. This is evident in the differences between the second column, based on measured superheats for the base case system, and the third column, which used low superheats. The steady-state power consumptions for each season show large differences (14%) and there is a 21% reduction in the required resistance heat. The annual power consumption was reduced by 1345 kWh by assuming a constant, low (7F) superheat at all heating and cooling ambients.

Although the first column in Table 4 is the system that most nearly resembles a high efficiency heat pump on the market, the third column is the data that should be used for a comparison with the optimized heat pumps. It includes the effects of using the Heat Pump Design Model (HPDM) for the steady-state calculations and also the assumption of low compressor inlet superheat.

Table 5 shows the steady-state COPs and capacities for the comparably-modeled base case and the optimized heat pump with capillary tubes (columns 3 and 5 in Table 4) . The optimized heat pump has higher steady-state COPs in both heating and cooling modes. It has higher low temperature capacities and roughly the same cooling and milder ambient heating capacities. The consequences of this, as shown in Table 4, are

— lower resistance heat use (100 kWh),

— better steady-state performance, when accumulated or totaled over the year, (465 kWh),

— lower frosting and defrosting losses, because of higher evaporating temperatures (20 kWh), and

— lower cycling losses (95 kWh).*

The difference in resistance heat is the result of the higher heating capacity and the drop in cycling losses comes from the slightly lower cooling capacity.

HEAT PUMPS WITH CAPILLARY TUBES

There is more information within the steady-state data in Table 4 that is worth examining. Besides accumulating steady-state power consumption over the year, the APF model also kept separate the power consumptions for the compressor and indoor and outdoor fans. These results are tabulated in Table 6 for the comparably modeled base case and the optimized heat pump with capillary tubes. These show a significant savings in fan powers for the year, 485 kWh, but also an increase in compressor power for winter time operation, 135 kWh.

* The influence of the type of flow control device on the cyclic degradation (C_D factors) was not addressed in this study.

The steady-state power consumptions for each component were examined by temperature bins to see when these seasonal and annual differences occurred. This showed that there was a uniform energy savings for each of the fans throughout the year. This is the result of the lower air velocities across each coil (see Table 4) and the reductions in air-side pressure drops (in both the coils and the ducts). Although the indoor fan/fan motor efficiencies were the same for the base case heat pump and the optimized system, the fan powers are quite different. The indoor fan of the optimized heat pump draws 156 W in heating mode and 175 W in the cooling mode versus 285 W and 355 W for the base case. The corresponding pressure drops are 0.34 and 0.39 inches of water (89 and 97 kPa) (for heating and cooling modes, respectively) for the optimized heat pump and 0.53 and 0.66 inches of water (132 and 164 kPa) for the base case (for heating and cooling).

The outdoor fan/fan motor of the base case heat pump runs at an efficiency of 24% in both heating and cooling modes; almost the maximum combined efficiency of 25%. The optimized system has lower fan powers, though, while operating at combined efficiencies of 17-22% in heating and 14-15% in cooling. Its pressure drops are 0.07-0.12 inches of water in heating mode (17-55 kPa) and 0.06 inches in cooling (15 kPa) compared to 0.15-0.28 (37-70 kPa) and 0.13 (32 kPa) for the base case. The net effect is lower fan powers in heating, particularly at the higher ambients where there are a lot of operating hours (60-90 W above 30F (-1°C)). The outdoor fan of the optimized heat pump draws 6-11 W more in cooling mode than the base case. The seasonal and annual effects of these changes in fan powers are shown in Table 5.

The optimized heat pump has higher compressor powers in heating mode (80-230 W) and lower in cooling mode (108-225 W). In heating mode, the optimized heat pump operates at evaporating temperatures that are up to 3F (1.6°C) higher than those for the base case and condensing temperatures from 4-15F (2.2-8.3°C) higher. This results in higher suction and discharge pressures, capacities, and power consumptions. In cooling mode, however, the evaporating temperatures differ by less than 1F (0.6°C) while the optimized system has condensing temperatures that are 4-5F (2.2-2.8°C) lower than those for the base case. This gives slightly lower head pressures, lower capacities below the design point temperature, and lower compressor powers.

IDEALIZED REFRIGERANT FLOW CONTROL

It is reasonable to consider how much better a system could perform that used a more sophisticated expansion device than a simple capillary tube. Although the steady-state heat pump model that was used cannot directly simulate a superheat-controlled device such as a TXV, it can be used to find the "best" level of condenser subcooling at a constant level of evaporator superheat. Given a specified level of condenser subcooling, the model can predict the COP and capacity at a particular ambient temperature. A best, or "ideal", subcooling control scheme was investigated by expressing the desired subcooling at each ambient as a linear function of the outdoor temperature (one function each for heating and another for cooling) and letting the optimization routine determine the best slopes and intercepts for those curves.

Even though the optimization started out with a very different set of components than the optimized system with capillary tubes, the iterations moved to heat exchanger and compressor parameters that are virtually the same as those from the previous case. The maximum annual performance factor for this case with idealized flow control is 2.37 compared with 2.35 for the system with the best capillary tubes. This difference amounts to only 81 kWh. Table 7 shows the amount of net reduction in seasonal and annual power consumption obtained by replacing the capillary tubes with idealized flow control. The differences in the power consumptions are practically insignificant. The most significant fact about the differences in Table 7 is that they are so small.

This is an unexpected result and merits further investigation. It may only mean that to approach the more ideal control, subcooling should be controlled as a nonlinear function of the outdoor temperature. As they are, these results for both subcool and capillary tube control may be only valid when compressor inlet superheat is held at a low value, and this may require an unreasonably large accumulator to prevent the accumulation from going dry in the cooling mode. It is not known if similar results would be obtained for a reasonable sized accumulator used with a charge inventory model. If this result does remain under further analysis and testing, however, it would underscore the significance of proper selection of the capillary tube. Selecting the capillary tube for best seasonal performance, rather than best steady-state performance at a particular design temperature, could perhaps result in seasonal efficiencies for single-speed heat pumps obtained with much more sophisticated flow control devices. However, experience with equipment to-date suggests that the primary causes of the close performance are more likely the linear subcool or control assumptions and the charge inventory limitations of the current model.

OPTIMIZATION FOR SUMMER-TIME OPERATION

Another question of interest is how well does a heat pump perform as an air conditioner; how has the cooling performance been compromised to provide heating in the winter. The objective function of the optimizations was changed to investigate this so that now the optimizer was searching for the best cooling season performance instead of the best annual performance. The steady-state model was used just in cooling mode to simulate a central air conditioner and the APF program was used only with the cooling loads. The results of the optimization were compared to the optimized heat pump.

It is important to note that the steady-state model does not include losses due to the reversing valve in the heat pump, neither those due to heat transfer or pressure drops nor due to refrigerant mass leakage. Hence the performance differences given here do not include any deleterious effects of the reversing valve on heat pump performance and any efficiency gain the air conditioner would derive from not requiring one.

A comparison of the components for the optimized air conditioner and heat pump (both with capillary tube control) were given in Table 3 with a comparison of the power consumptions shown in Table 8. The scaled (comparably-modeled) base case has been included in Tables 3 and 8 as a point of reference for current heat pump technology. These show a seasonal energy efficiency ratio (SEER) of 9.65 for the optimum central air conditioner, 9.39 for the optimized heat pump, and 8.50 for the scaled base case. The indoor and outdoor fan powers (steady state) are virtually the same (5.5 and 8.5 W higher for the heat pump) which leads to small (5-10 kWh) differences over the cooling season. The steady-state compressor power for the air conditioner is 60-80 W lower, resulting in a 90 kWh savings over the summer. This is the result of 3-4F (1.7-2.2°C) lower condensing temperatures.

Optimizing the design for cooling mode results in only a minor improvement, therefore, in cooling season performance over a heat pump designed for best annual performance. The differences in optimum design configuration are more significant than the differences in cooling season power consumption. One conclusion from this is that the cooling season performance, as modeled, is somewhat insensitive to the considered design changes about the optimal configurations. This suggests that efficient air-conditioning does not have to be compromised for heating duty, unless charge inventory effects are a factor. It therefore suggests that the larger SEER's obtained on the average by central air conditioners are likely due to reversing valve effects and charge inventory considerations rather than any compromises in choice of heat exchanger, compressor, and air flow configuration.

SUMMARY AND CONCLUSIONS

This study has examined the use of computer simulation models and numerical techniques to aid in the design of single-speed heat pumps. Although it has not predicted any staggering improvements in annual performance, it has shown the viability of these computational tools in evaluating design tradeoff questions. These techniques may only be "useful" in conventional designs, but they should prove invaluable in the design of future generations of heat pumps. Sophisticated controls and modulation strategies will both complicate the design process and also necessitate finding the best design in order to make full use of and justify their capabilities. Computer-aided design is a viable approach for the HVAC industry and can shorten the design process while leading to more energy efficient systems.

The specific conclusions of this study are somewhat clouded by the limitations of the steady-state heat pump model. The lack of charge inventory calculations and the requirement of low, fixed compressor inlet superheats have hampered more definitive predictions. Useful analysis can be performed, though, even within the current limitations of the models.

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TABLE 1
Steady-State Validation Results

Unit	Mode	Ambient temperature (F (°C))	Observed COP	Predicted COP	% diff	Observed capacity (MBtu/hr (kW))	Predicted capacity (Mbtu/hr (kW))	% diff
low bid	H	42.0 (5.6)	2.04	2.16	+	36.26 (10.62)	35.92 (10.53)	-
low bid	H*	51.0 (10.6)	2.14	2.18	+	38.41 (11.25)	39.36 (11.53)	+
hi-eff	H	17.00 (-8.3)	2.04	1.83	-	19.86 (5.82)	16.05 (4.70)	-
hi-eff	H	47.00 (8.3)	3.09	2.90	-	36.00 (10.55)	33.41 (9.79)	-
hi-eff	C	75.00 (29.0)	2.54	2.76	+	29.45 (8.63)	34.78 (10.19)	+
hi-eff	C	95.00 (35.0)	2.27	2.31	+	30.81 (9.03)	32.72 (9.59)	+

* No cooling tests were performed on the low-bid unit.

TABLE 2
Validation of SPF Calculations for Knoxville, TN.

Season	Observed SPF	Predicted SPF	Difference
Heating 79-80	1.99	1.86	-7 %
Heating 82-83	1.97	1.90	-5 %
Cooling 80	2.19	2.36	+8 %
Cooling 83	2.17	2.28	+5 %

TABLE 3
Comparison of Components in Scaled Based Case, Optimized Heat Pump, and Optimized Central Air Conditioner With Capillary Tubes

	Scaled Base Case	Optimized Heat Pump	Optimized Air Cconditioner
Compressor Displacement	2.91	2.68	2.68 cu. in.
	47.7	44.0	44.0 ml.
Indoor Heat Exchanger:			
Air Flow Rate	1000	850	880 cfm
	470	400	415 l/s
Face Velocity	275	170	145 ft/min
	1.40	0.86	0.74 m/s
Frontal Area	3.64	5.07	6.09 sq. ft.
	0.34	0.47	0.57 m^2
Refrigerant Tube Rows	4.0	2.5	2.0
Refrigerant Circuits	2.4	3.0	3.0
Fraction of HX Area	0.49	0.43	0.41
Outdoor Heat Exchanger:			
Air Flow Rate	1820	2430	2525 cfm
	860	1150	1190 l/s
Face Velocity	365	285	290 ft/min
	1.85	1.45	1.47 m/s
Frontal Area	5.01	8.51	8.76 sq. ft.
	0.46	0.79	0.81 m^2
Refrigerant Tube Rows	3.0	2.0	2.0
Refrigerant Circuits	2.4	3.0	2.0
Fraction of HX Area	0.51	0.57	0.59
Capillary Flow Factors:			
Heating	3.04	2.25	*****
Cooling	3.67	3.53	4.03

TABLE 4
Comparison of Seasonal and Annual Performance of Base Case and Optimized Systems for Nashville, TN

	Base Case with Measured Steady-State Data	Base Case Steady-State Model Data with Measured Superheats	Base Case with Steady-State Model Data using Low Superheats	Optimized Heat Pump Capillary Tubes	Optimized Heat Pump Subcool Flow Control	
Performance Factors:						
Heating	1.91	1.74	2.00	2.11	2.11	
Cooling	2.22	2.20	2.49	2.75	2.83	
Annual	2.03	1.92	2.18	2.35	2.37	
Power Consumption:						
Heating Season						
Steady-State	4,345	4,455	3,900	3,725	3,735	kWh
Cycling Losses	755	665	615	590	580	kWh
Frosting Losses	560	650	615	595	605	kWh
Resistance Heat	610	1,090	860	760	760	kWh
Total	6,270	6,860	5,990	5,670	5,680	kWh
Cooling Season						
Steady-State	3,575	3,550	3,120	2,830	2,760	kWh
Cycling Losses	575	625	580	510	490	kWh
Total	4,150	4,175	3,700	3,340	3,250	kWh
Annual						
Steady-State	7,920	8,005	7,020	6,555	6,495	kWh
Cycling Losses	1,330	1,290	1,195	1,100	1,070	kWh
Frosting Losses	560	650	615	595	605	kWh
Resistance Heat	610	1,090	860	760	760	kWh
Total Annual	10,420	Q1,035	9,690	9,010	8,930	kWh

TABLE 5
Steady-State Performance of the Scaled Base Heat Pump and the Comparable Optimized Heat Pump with Capillary Tubes

Ambient Temperature (F (°C))	Scaled Base Case	Optimized Heat Pump	Percent Difference (%)
COP			
17 (-8.3)	2.35	2.51	6.8
32 (0)	2.79	2.97	6.5
47 (8.3)	3.20	3.36	5.0
82 (28)	2.97	3.27	10.1
95 (35)	2.56	2.79	9.0
Capacity	Btu/h (kW)	Btu/h (kW)	
17 (-8.3)	15,325 (4.49)	16,220 (4.75)	5.8
32 (0)	21,520 (6.21)	21,810 (6.39)	1.3
47 (8.3)	28,250 (8.28)	28,310 (8.30)	0.2
82 (28)	30,365 (8.90)	29,575 (8.67)	-2.6
95 (35)	27,610 (8.09)	27,260 (7.99)	-1.3

TABLE 6
Comparison of Steady-State Power Consumption by Components for the Scaled Base Case and the Optimied Heat Pump

	Scaled Base Case	Optimized Heat Pump with Capillary Tubes	Difference
Heating Season:			
Compressor	3,095	3,230	+135 kWh
Indoor Fan	T85	R65	-220 kWh
Outdoor Fan	320	225	-95 kWh
Total	3,900	3,720	-180 kWh
Cooling Season:			
Compressor	2,635	2,515	-120 kWh
Indoor Fan	365	185	-180 kWh
Outdoor Fan	120	130	+10 kWh
Total	3,120	2,830	-290 kWh
Annual:			
Compressor	5,730	5,740	+10 kWh
Indoor Fan	850	450	-400 kWh
Outdoor Fan	440	355	-85 kWh
Total	7,020	6,550	-65 kWh

TABLE 7
Comparison of Power Consumptions for Optimized Heat Pumps with Capillary Tubes and Idealized Flow Control for Nashville, TN

	Optimized Heat Pump With Capillary Tubes	Optimized Heat Pump With Idealized Flow Control	Difference
Heating Season			
Steady-State			0
Compressor	3,230	3,250	20 kWh
Indoor Fan	265	240	-25 kWh
Outdoor Fan	225	240	15 kWh
Subtotal	3,720	3,730	10 kWh
Dynamic Losses			
Cycling	590	580	-10 kWh
Frosting/Defrosting	595	605	10 kWh
Subtotal	1,185	1,185	0 kWh
Resistance Heat	760	760	0 kWh
Total	5,665	5,680	15 kWh
Cooling Season			
Steady-State			
Compressor	2,515	2,460	-55 kWh
Indoor Fan	185	160	-25 kWh
Outdoor Fan	130	140	10 kWh
Subtotal	2,830	2,760	-70 kWh
Cycling Losses	510	490	-20 kWh
Total	3,340	3,250	-90 kWh
Annual			
Steady-State			
Compressor	5,745	5,710	-35 kWh
Indoor Fan	450	400	-50 kWh
Outdoor Fan	355	380	25 kWh
Subtotal	6,550	6,490	-60 kWh
Dynamic Losses			0
Cycling	1,100	1,070	-30 kWh
Frosting/Defrosting	595	605	10 kWh
Subtotal	1,695	1,675	-20 kWh
Resistance Heat	760	760	0 kWh
Total	9,005	8,925	-80 kWh

TABLE 8
Comparison of Optimized Central Air Conditioner and the Cooling Performance of an Optimized Heat Pump and the Scaled Base Case Heat Pump for Nashville, TN

	Scaled Base Case	Optimized Heat Pump	Optimized Central Air Conditioner
SEER	8.50	9.39	9.65
CSPF	2.49	2.75	2.83
Steady-State:			
Compressor	2,635	2,515	2,425 kWh
Indoor Fan	365	185	190 kWh
Outdoor Fan	120	130	140 kWh
Total Steady-State	3,120	2,830	2,755 kWh
Cycling Losses	580	510	495 kWh
Total Power Consumption	3,700	3,340	3,250 kWh

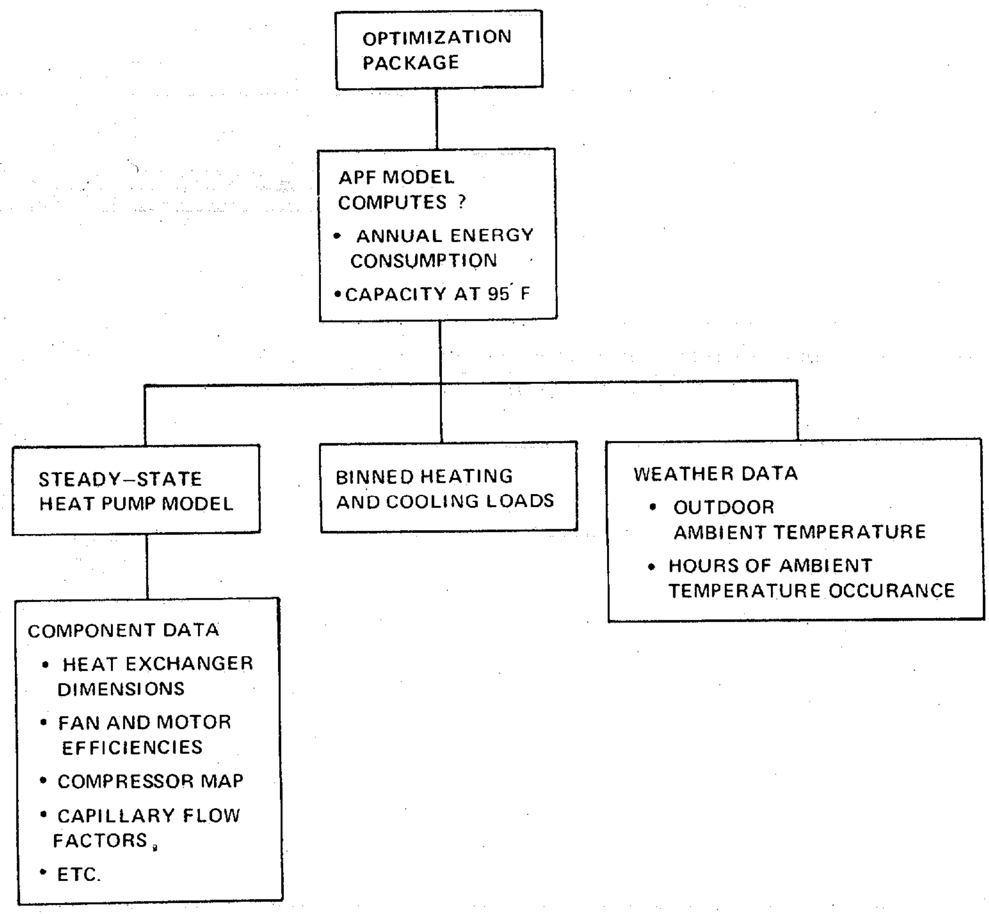

Figure 1. Program and data structure for annual performance factor (APF) optimization

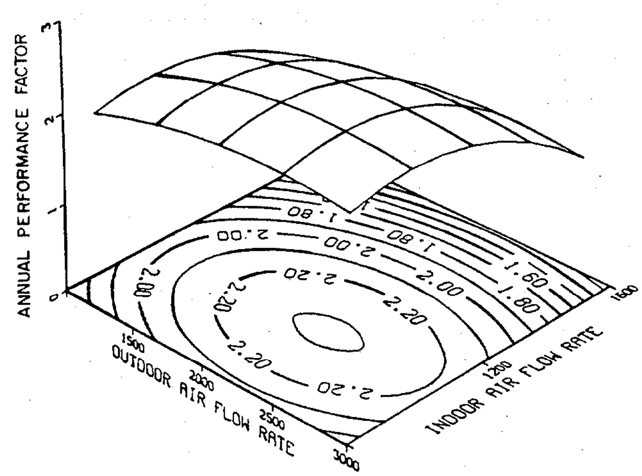

Figure 2. An illustration of the unconstrained maximum of a function of two variables using perspective and contour plots

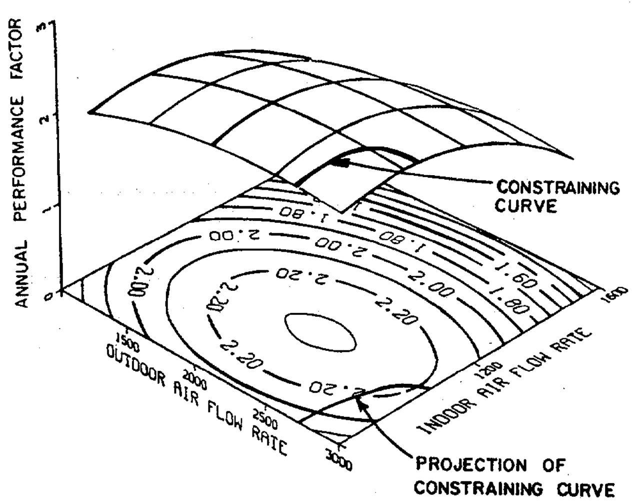

Figure 3. An illustration of the constrained maximum of a function of two variables using perspective and contour plots

Discussion

J.M. CALM, Institut CERAC S.A., Switzerland: The model presented overpredicts cooling performance and underpredicts heating performance relative to actual performance. The concluding comparison of optimized to base case performance reflects annual energy savings of 7% to 8% with specific component energy use breakouts. Are these savings reliable in light of the overprediction in the predominant load mode of operation, or are some of the savings attributable to analytical error?

FISCHER: Yes, I think they are. I have a great deal of confidence in our predicted energy savings because they come from model to model comparisons. The discussion in the paper of the data in Table 4 focuses on the model-to-model comparison and points out the uncertainties in making a predicted-to-observed comparison. The model-to-model predicted energy savings are then substantiated by the energy use breakdowns.

As you say the model results for both the base case system and the optimized heat pump overpredict cooling performance and underpredict heating performance. The magnitudes of these errors will be the same, though, for "model-to-model" comparisons and will pretty much wash out in the end.

CALM: Will future studies include optimization based on unequal energy weighting between the heating and cooling modes (rather than an annual performance factor to reflect utility concerns with different generation mixes between seasons or similarly seasonal electrical rates to consumers)?

FISCHER: I can't say definitively what directions future work will take. An interesting possibility, which we have looked at a little but which merits further investigation, would be to optimize the design for best cooling performance alone and just see how well it does in heating. An increasing share of our work is going to modulating heat pumps and we ought to consider the balance between heating/cooling demands, different utility rate structures, peak demand charges, etc., in setting goals for the project. Minimum annual power consumption is certainly not the only way to go.

W.E. MURPHY, Texas A&M Univ., College Station: Your comparisons with experimental data are for COP only. How did the absolute capacities compare?

FISCHER: The paper includes comparison of experimental and predicted data for both COP and capacity for two different heat pumps in Table 1. I split these data between three or four different slides, and then chose the most "representative" one to use in the presentation. But all of the data, including capacity results, are in the paper.

MURPHY: I understand that your capillary tube model uses curves taken from the ASHRAE Handbook. Do you expect those curves to be accurate for operation between 17 F and 95 F? Could you elaborate on your expected accuracy of those curves over a wide range of operating conditions?

FISCHER: Our capillary tube model is based on data in the ASHRAE Handbook, as you say, and we have done very well with it in our limited validations to date when compared to laboratory measurements. We are aware of a few other more involved approaches, but so far we have been satisfied with the simplified treatment in the Handbook. We feel that the ASHRAE approach will be satisfactory for predicting performance over the range of heat pump operation, provided the approach is coupled with a good charge inventory model.