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A SIMPLIFIED EXPERIMENTAL MODEL TO SIMULATE AN AIR CONDITIONER

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Abstract. *A simplified model based on experimental data has been developed to be used in air-conditioning systems simulations. Cycle thermodynamic relationships, along with experimental data, were used to mathematically model the air-conditioning system. Characteristic parameters of the equipment were determined from the results of a limited number of testing points. The following parameters were used: the clearance factor C, the piston displacement PD, the efficiency of the compressor* η_{comp}*, the sensible heat factor SHF and the product of the overall heat transfer coefficient times area (UA value) for the heat exchangers. Component performance simulation data was validated using experimental data from a 3 ton air conditioner. The model calculates the following parameters: coefficient of performance (COP), system capacity (*q_e*), system heat rejection (*q_c*) and compressor power (W) at any operating condition. A heat exchanger performance model was developed using overall heat transfer coefficients and first law energy balance principles. A semi-empirical compressor model developed by Popovic and Shapiro (1995) was incorporated to the air-conditioner simulation. The simulation model consists of heat exchanger energy balance relationships for the evaporator, condenser, expansion device and compressor model. The purpose of the model is to reduce the amount of experimental data needed to characterize a system while keeping an acceptable level of accuracy in predicting its performance. The model predicted system heat rejection (*q_c*) and system capacity (*q_e*) within 10% of experimental data. The required compressor power (W) was predicted within 5% of uncertainty. However, the coefficient of performance (COP) had an uncertainty slightly higher than ±13% when compared to experimental data. These ranges of uncertainty are quite acceptable for experimental models. These differences relate to equipment characterization of the current model, which is determined from the results of a few testing points. The accuracy of the model is quite acceptable, despite the simplifying assumptions. An additional advantage is the use of the simulation model in equipment certification minimizing the required number of testing points.*

Key Words : *Air Conditioning, Systems Simulation, Modeling*

Symbols

A	Curve-fitting coefficients for polytropic exponent dependence on compression ratios

C	Clearance factor
c_p	Specific heat, kJ/kg°K
f	Thermodynamic property of refrigerant at saturation temperature
h	Specific enthalpy , kJ/kg
m	Mass flow rate, kg/s
m_c	Condenser air mass flow rate, kg/s
m_e	Evaporator air mass flow rate, kg/s
n	Polytropic exponent
P	Pressure, kPa
PD	Piston displacement volume, m^3
q	Rate of heat transfer, W
SHF	Sensible heat factor
T	Temperature, °C
v	Specific volume, m^3/kg
W	Energy rate (power), kW
UA	Overall heat transfer coefficient, W/°K

Greek Symbols

η	Efficiency
φ	Thermodynamic property of refrigerant at saturation temperature
ψ	Thermodynamic property of refrigerant at saturation temperature
θ	Thermodynamic property of refrigerant at saturation temperature

Subscripts

c	Condenser
ci	Air side condenser inlet
comp	Compressor
e	Evaporator
ei	Air side evaporator inlet
dis	Discharge, state of refrigerant leaving the compressor
ref	Refrigerant
suct	Suction, state of refrigerant entering the compressor

Introduction

The performance of an air conditioner depends not only on the performance of its individual components but also on how they interact. The intricate relationships among components are linked to operating conditions and design parameters. Due to this complexity, it is too expensive and time-consuming to perform field experiments to determine the effects of all possible changes either in the components or in the operating conditions on the performance of the air conditioner. As a result, computer models are used to predict system response to changes in sizing, materials, ambient conditions, etc.

Model Description

The pressure-enthalpy diagram in figure 1 shows the standard vapor-compression cycle. Process is as follows: isentropic compression of saturated vapor (state point 1) to the condenser pressure (state point 2). Change of phase at constant presuure inside the condenser (2 to 3). Saturated liquid leaves the condenser (state point 3) and, expands at constant enthalpy to gas at evaporator pressure (state point 4). Evaporation at evaporator pressure to saturated vapor (state point 1).

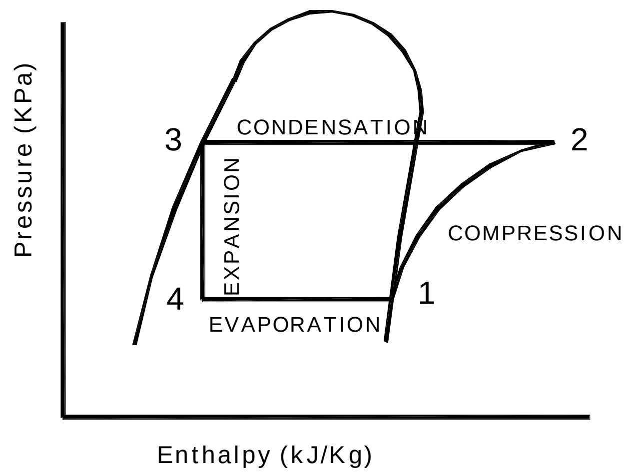

Figure 1. Pressure enthalpy diagram of the standard vapor compression cycle

In order to reduce the complexity of the model the following simplifying assumptions are made: Steady-state operation (same mass flow of refrigerant in any part of the loop). Pressure drops are neglected, except at the expansion valve. No subcooling takes place at the condenser outlet. There is no superheating at the evaporator outlet. Compression is a polytropic process. The polytropic exponent is a function of the refrigerant type and compression ratio only. Lubricating oil has negligible effects on refrigerant properties and compressor operation. There are no pressure drops at the compressor suction and discharge lines. Refrigerant charge effects are neglected.

Compressor

An exact analytical model of a reciprocating compressor is extremely complex. There are two different compressor models for system simulation. The first compressor model requires a compressor performance chart. These charts are generated by a manufacturer for a fixed superheat, and curve-fitted to compressor inlet and outlet temperatures (Fisher and Rice (1983)). In the second model, the compressor performance is modeled through a control volume approach in which the overall energy transfer is evaluated. Reciprocating and centrifugal compressors have been modeled using this approach with very good results (Popovic and Shapiro (1995); Popovic and Shapiro (1998)). This semi-theoretical model is based upon displacement, operational speed, volumetric efficiency and isentropic efficiency. The refrigerant flow rate through the compressor can be expressed as:

$$m_{ref} = \frac{PD}{v_{suct}}\left(1 + C - C\left(\frac{P_{dis}}{P_{suct}}\right)^{1/n}\right) \quad (1)$$

This equation shows that the refrigerant mass flow rate depends on compressor geometry, compression ratio, refrigerant type (through the polytropic exponent), and the condition of the refrigerant vapor at the beginning of compression. Popovic and Shapiro (1995) analyzed extensive experimental data and proposed a model for the polytropic exponent as a function of the refrigerant type and compression ratio. This function is of the following form:

$$n = A_1 + \frac{A_2}{\left(\frac{P_{dis}}{P_{suct}}\right)} + \frac{A_3}{\left(\frac{P_{dis}}{P_{suct}}\right)^2} \quad (2)$$

The coefficients used in this equation for various refrigerants are shown in table 1.

Table 1. Coefficients for polytropic exponents as a function of the compression ratio for various refrigerants

Refrigerant	A_1	A_2	A_3
R-22	1.2094	-0.2931	0.7802
R-134a	1.0428	0.1270	0.1603
MP-39	1.0022	1.1272	-1.5753
MP-52	1.0627	0.4177	-0.0991
R-12	0.9975	0.8477	-0.5327

The enthalpy balance for the compressor is given by:

$$h_2 - h_1 = \frac{n}{n-1} P_{suct}\, v_{suct}\left(\left(\frac{P_{dis}}{P_{suct}}\right)^{\frac{n-1}{n}} - 1\right) \quad (3)$$

For an isentropic compression process, under steady state conditions, the compressor power is given by:

$$W_{comp} = m_{ref}\left(h_2 - h_1\right) \quad (4)$$

The actual compressor power required can be found by dividing the isentropic power by the compressor efficiency:

$$W = \frac{W_{comp}}{\eta_{comp}} \quad (5)$$

Evaporator Model

A wide range of models for heat exchangers are currently available, and can be differentiated by the degree of complexity and empiricism incorporated. Most of them are highly empirical and require few geometric specifications (Rabehl et al (1999)). There are two methods of heat exchanger analysis that are commonly used. One, the *log-mean-temperature-difference* method, is extensively used in the process industries; the other, the *effectiveness-NTU* (Number of Transfer Units) method is more often employed for gas-flow heat exchangers, especially in thermal power systems (Incropera and Dewitt (1996); Kays and London (1984)). Our model uses the first scheme to simulate heat exchanger performance.

It is possible to derive a set of equations representing the fluid heat transfer when one of the fluids flowing through a heat exchanger changes phase. This fluid will remain at a constant temperature, provided that its pressure does not change. As air moves across the evaporator its temperature decreases from T_{ei} to T_{eo} while the refrigerant temperature, T_e, remains constant. The characteristic shape of the temperature curves of the two fluids is shown in figure 2

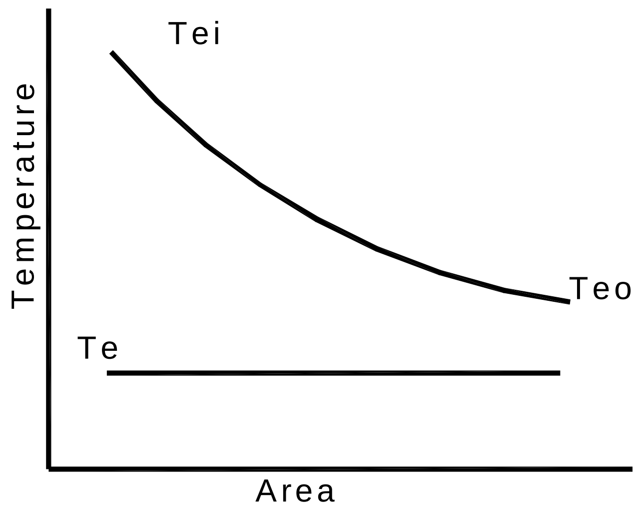

Figure 2. Fluid temperature variation in an evaporator

The cooling and dehumidifying processes that take place in the evaporator involve both sensible and latent heat transfer. The total heat is the sum of both the sensible and latent heats. The sensible heat ratio is defined as the ratio between the sensible heat and the total heat across the evaporator.

From a refrigerant energy balance on the evaporator, the energy given by the refrigerant can be expressed as:

$$q_e = m_{ref} \left(h_1 - h_4\right) \tag{6}$$

The evaporator refrigerant temperature can be found from an air-side heat balance on the evaporator and can be expressed as:

$$T_e = T_{ei} - \frac{q_e \; SHF}{m_e \; c_p \left[1 - \exp\left(-\frac{(UA)_e}{m_e \; cp}\right)\right]} \tag{7}$$

The suction pressure corresponds to the saturation pressure at the evaporator refrigerant temperature:

$$P_{suct} = f\left(T_e\right) \tag{8}$$

and the suction specific volume is also a function of the evaporator temperature:

$$v_{suct} = \varphi(T_e) \tag{9}$$

The refrigerant enthalpy at the evaporator exit is a function of the saturation temperature:

$$h_1 = \psi(T_e) \tag{10}$$

Condenser Model

The condenser model is a very simple approach similar to that of the evaporator model. The fluid will then remain at a constant temperature, provided that its pressure does not change. As air moves through the condenser its temperature increases from T_{ci} to T_{co}, while the refrigerant temperature, T_c, remains constant. The fluid temperature profiles for the condenser are shown in figure 3.

These profiles are valid only in the condensing zone of the condenser and not in the desuperheating and subcooling zones. Stoecker and Jones (1982), however report that this representation of the heat exchanger provides reasonably accurate results throughout the entire condenser. The condenser refrigerant temperature can be found from an air-side heat balance on the condenser and can be expressed as:

$$T_c = T_{ci} + \frac{q_c}{m_c\ c_p\left[1 - \exp\left(-\frac{(UA)_c}{m_c\ cp}\right)\right]} \tag{11}$$

The discharge pressure corresponds to the saturation pressure at the condenser refrigerant temperature:

$$P_{dis} = f(T_c) \tag{12}$$

From a refrigerant energy balance on the condenser, the energy absorbed by the refrigerant can be expressed as the refrigerant mass flow rate times a change in enthalpy of the refrigerant.

$$q_c = m_{ref}\,(h_2 - h_3) \tag{13}$$

Expansion Device

In predicting system performance so far it has tacitly been assumed that the expansion device is able to regulate the flow of refrigerant into the evaporator so that the heat transfer surfaces on the refrigerant side of the evaporator are wetted with liquid refrigerant. Saturated liquid is entering our expansion device. Assuming that there is no heat transfer or external work.

$$h_3 = \psi(T_c) \tag{14}$$

Finally, the process from state 3 to state 4 through the expansion device is assumed to be isoenthalpic

$$h_4 = h_3 \tag{15}$$

The vapor compression system can be now modeled by the steady state representation of each of its components. This steady state model yields to a set of 15 equations with 15 unknowns. These unknowns are:

$$m_{ref}\ , v_{suct}\ , P_{suct}\ , P_{dis}\ , n\ , h_1\ , h_2\ , h_3\ , h_4\ , W_{comp}\ , W\ , q_e\ , q_c\ , T_e\ , T_c$$

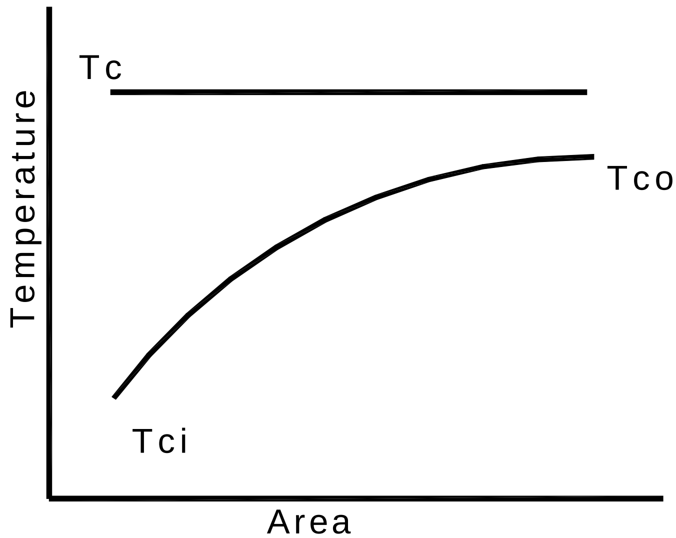

Figure 3. Fluid temperature distribution in a condenser

These 15 equations are nonlinear, and they will be solved simultaneously and numerically. However, the equations include certain parameters that need to be determined before a solution can be sought. These parameters are: for the compressor C , PD , η_{comp} and for the heat exchangers SHF , $(UA)_e$, $(UA)_c$

Results

Component performance was validated using experimental data from a 3 ton air conditioner unit based upon the test data taken by Rosario (1999). Characteristic parameters of the equipment were determined from the results of three testing points. The air-conditioning parameters studied were: coefficient of performance *COP*, system capacity q_e, system heat rejection q_c, and compressor power *W*. Figures 4 through 7 present comparison between experimental and predicted parameters.

Figure 4 presents a comparison of simulated and measured coefficient of performance *COP*. The majority of data points as seen in figure 4 lies within 13% relative error of the calculated coefficient of performance. Analyzing these results, It can be concluded that the model overpredicted the coefficient of performance COP. However, predicted COP values are within reasonable limits.

The values obtained for the system capacity prediction are shown in Figure 5. In a majority of cases the system capacity was predicted within a ±10% difference of the measured values. These results are relatively consistent especially if we considered that they were calculated based on airside measurements only.

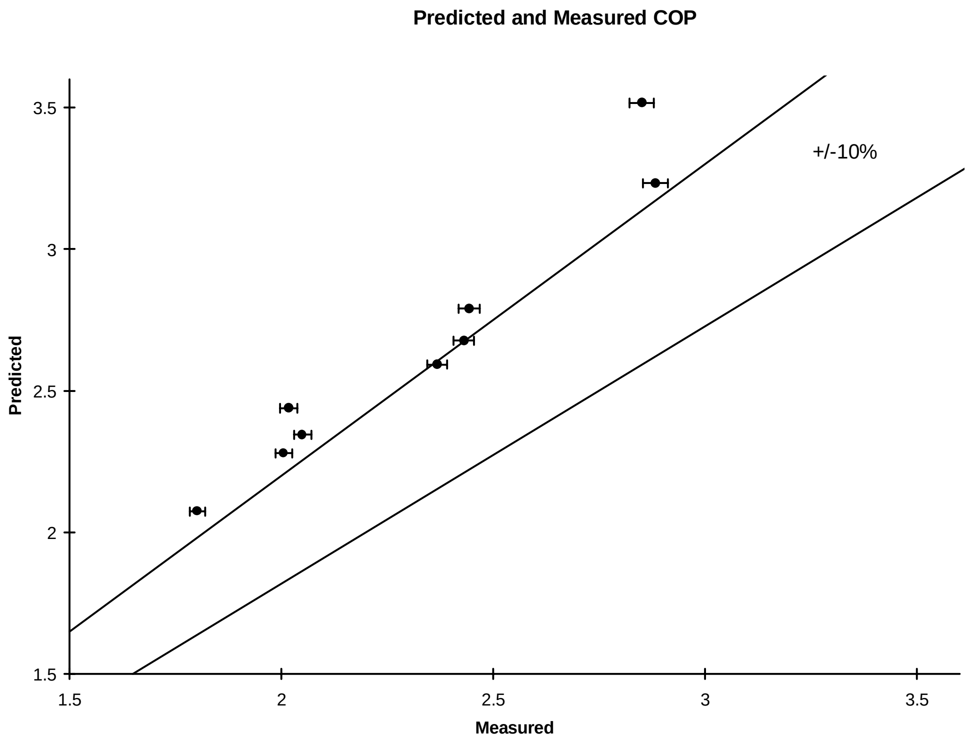

Figure 4.- Predicted and measured coefficients of performance (COP)

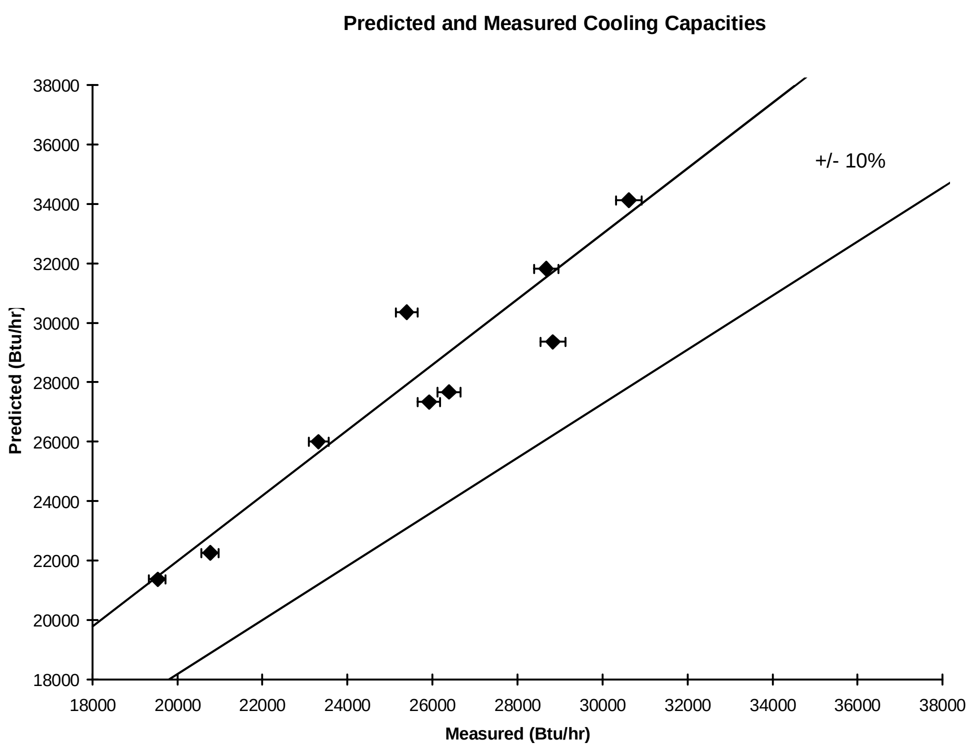

Figure 5.- Predicted and measured system capacities

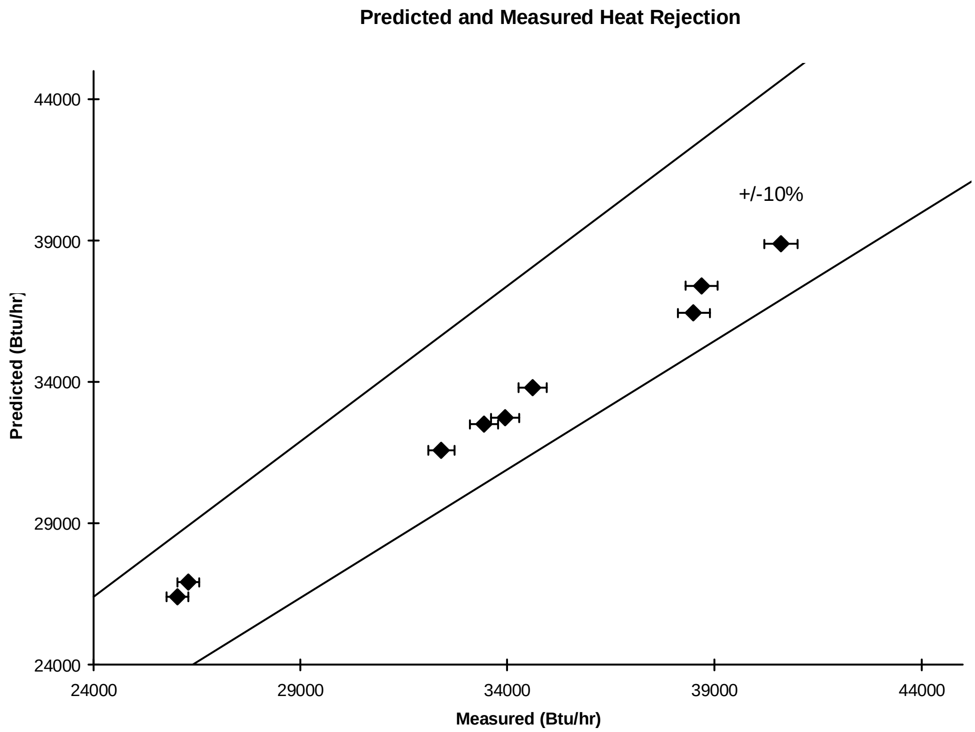

Figure 6.- Predicted and measured system heat rejections

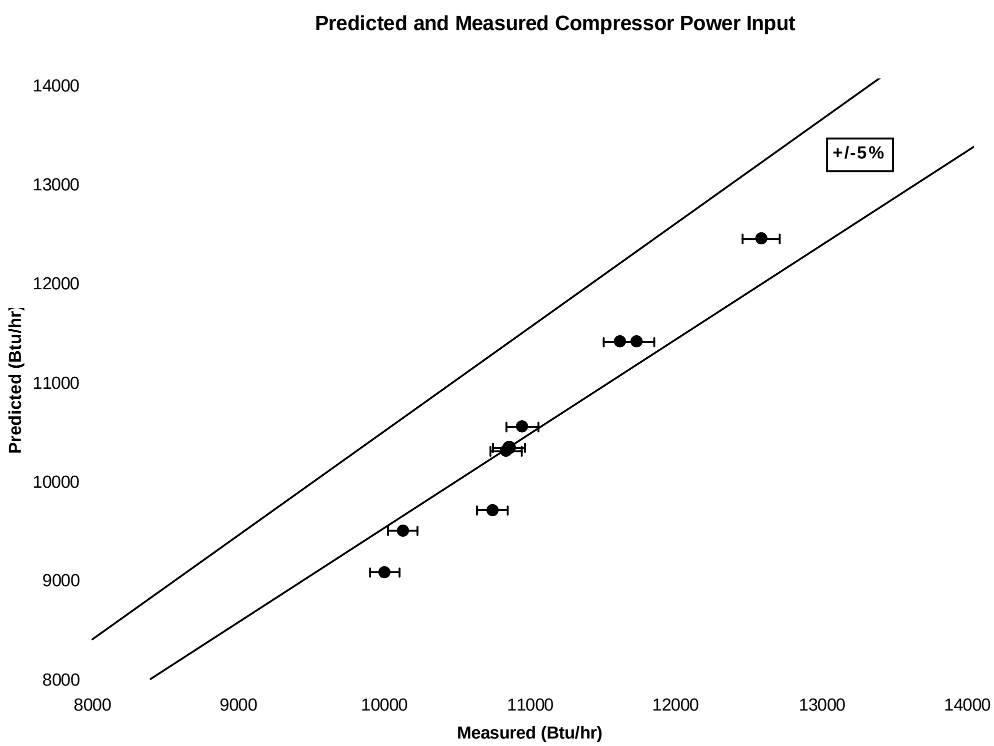

Figure 7.- Predicted and measured compressor power

Figure 6 presents results of the comparison between the calculated and predicted system heat rejection for the entire set of experimental data. These results have an uncertainty of about ±10%. These comparisons show that the model accurately predicted the system heat rejection. All predicted data points showed good agreement with experimental data.

Figure 7 compares predicted and measured compressor powers for our entire set of data. In this case the uncertainty is about ±5%. Three out of nine data points were underpredicted within 10% by the model. This observation may indicate the importance of selecting experimental points for characterization compressor components. This selection should take into account very different compression ratios.

Conclusions

A simple model for an air conditioner has been developed.. Figures 4 through 7 have shown that the present model predicted parameter results in reasonable agreement with the experimental observations for all conditions tested.. The accuracy of the air-conditioner model is quite satisfactory since used characteristic parameters are determined by using three testing points. This computer model would be valuable to air conditioning manufacturers because system model can be generated quickly by testing few operating points. The present model is easy to use while keeping good accuracy. The ability to select any combination of input indoor and outdoor air conditions increases the value of the model as a design tool. This model will be relevant to manufacturers in the developing countries that have limited experimental and financial resources at their disposal.

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