MODIFICATION AND VALIDATION OF A UNIVERSAL THERMODYNAMIC CHILLER MODEL USED TO EVALUATE THE PERFORMANCE OF WATER-COOLED CENTRIFUGAL CHILLERS

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ABSTRACT

This paper presents modifications made to the procedure used to develop a universal thermodynamic model of chillers (Gordon and Ng, 1995). The modifications were necessary to capture physical phenomena involved when water is the coolant fluid at the condenser. The model was developed as a tool for analyzing chiller performance. It was designed to predict the coefficient of performance (COP) and the total electrical energy consumed by the compressor. The input parameters included time-series values of condenser temperatures, evaporator temperatures, and evaporator loads. The modifications were validated with data from monitoring two chillers operating at two institutional campuses: one a South Texas high school, and the other, a Northwest Arkansas university. The results showed that the

paper presents a literature review, model development, experimental set-ups, results, and summary and conclusions.

INTRODUCTION

In the integral part of building design, knowing the performance as well as the capabilities and limitations of its systems is essential. Moreover, instrumentation and monitoring of system components is expensive. Therefore, the designer needs to make use of modeling tools. For the present study, the procedures used to develop a universal thermodynamic model of chillers were modified to capture the physics of chillers that are commonly found in space cooling applications: the water-cooled centrifugal chiller.

The "efficiency" of chillers is expressed in terms of the coefficient of performance (COP), which is defined as the cooling capacity of the chiller (at the evaporator side) divided by the power input to the compressor (generally electrical power). Recent studies by Gordon and Ng (1994, 1995) and by Gordon et. al (1995) showed that the coefficient of performance and total energy consumption could be predicted using a "universal" thermodynamic model for all chillers. However, the procedures followed to predict COP and total energy consumption proved to work for water cooled centrifugal chillers only if modifications to the development procedure were made (Figueroa, 1997). The objective of this paper is to present the modifications made to the existing model. In addition, step-by-step procedures used in the development of the model are formulated.

LITERATURE REVIEW

Thermodynamic modeling of chillers constitutes a relatively new field. The major studies are summarized below. Leverenz and Bergan (1983)

simulation models were used to analyze chiller performance. The first type was a detailed model that took into account the involvement of all chiller components, such as the compressor and heat exchangers. The second type was an input/output model described as the "blackbox" model. This model related the input and output parameters of the chiller by algebraic equations without directly considering internal components. These algebraic equations were derived from the performance data and/or from the outputs of the detailed model. The input parameters included the load on the chiller, the leaving chilled water temperature, and the condenser inlet water temperature. The output parameters included the chiller's electrical power consumption and the load on the cooling tower. Leverenz and Bergan determined linear regression coefficients by least squares fit analysis. They found that the coefficients appeared to be independent of the size of the chiller and that these could be used as default parameters to describe reciprocating chiller performance. The model was validated using experimental data. They concluded that the input/output ("blackbox") model accurately calculated the actual chiller performance in an HVAC configuration.

Gordon and Ng (1994) developed a simple thermodynamic model for predicting the coefficient of performance and total energy consumption of reciprocating chillers. The model was also developed for a better understanding of chiller performance and diagnostic purposes. The simple thermodynamic model was expressed as follows:

$$\frac{1}{COP} = -1 + \frac{T_{cond}^{in}}{T_{evap}^{out}} + \frac{1}{\dot{Q}_{evap}}\left(\dot{q}_{cond} + \dot{q}_{evap}\frac{T_{cond}^{in}}{T_{evap}^{out}}\right) \qquad (1)$$

This model predicted a linear relationship between 1/COP and 1/cooling capacity. They also determined that for reciprocating chillers the characteristic curves depended on condenser and evaporator temperatures. The model was developed combining the first law of thermodynamics, the second law of thermodynamics, heat transfer processes of the evaporator and condenser, irreversibilities in the evaporator and condenser, and the definition of coefficient of performance for refrigeration devices. Two assumptions were made: (1) cyclic operations were considered steady state, and (2) heat transfer processes from and to the heat exchangers were linear and isothermal.

Further research carried out by Gordon and Ng (1995) resulted in a universal thermodynamic model that was assumed to work for all chillers if the same procedure used for reciprocating chillers were to be followed. The purpose of their research was also to provide a procedure for predicting chiller performance and to develop a diagnostic tool. This model was validated using reciprocating chillers, air-cooled centrifugal chillers, and absorption chillers.

Research by Brandemuehl et al. (1996) provided uniform and systematic procedures for in-situ field measurements and evaluations of centrifugal chiller performance in order to evaluate annual energy consumption and peak demand characteristics. The chiller model was developed for centrifugal chillers using the same procedure of Gordon et al. (1995). The drawback of this study was that the procedures, which were developed for air-cooled chillers rather than for water-cooled chillers were used. The research presented in this paper aimed at developing a thermodynamic model to predict chiller performance for water-cooled centrifugal chillers.

THEORY AND MODEL DEVELOPMENT

The model was developed by first analyzing the main components of a water-cooled centrifugal chiller. The refrigeration cycle for water-cooled centrifugal chillers is shown in Figure 1. The vapor compression refrigeration cycle is used to take heat from one space and transfer it to a warmer space at the expense of electrical power input at the compressor. Typical commercial-applications refrigeration-cycles consist of three main loops (two water loops and one refrigerant loop). The two water loops consist of the evaporator-cooling load assembly and of the condenser-cooling tower assembly. The refrigerant loop consists of the condenser, evaporator, compressor, and expanding device. Therefore, there is an indirect communication between the two water loops through the refrigerant loop.

At the evaporator the refrigerant is vaporized and extracts heat from a surrounding medium (water in this case). The condenser rejects the heat absorbed previously. Under ideal circumstances, the energy given off by the refrigerant at the condenser is the energy that was gained at the evaporator plus energy added during the compression stage. Common operating conditions for commercial comfort air conditioning systems frequently have chilled water entering the evaporator at 13.3 °C (56 °F) and leaving at 6.7 °C (44 °F). The temperature of the water leaving and entering the condenser depends on the ambient conditions. Industry testing standards, however, use 29.4 °C (85 °F) entering and 35 °C (95 °F) leaving the condenser.

Since chillers operate in a cyclic manner and under steady-state conditions, Gordon and Ng (1994, 1995) and Gordon et. al (1995) noted that both the change in energy and change of entropy of the system's working fluid would be zero. In addition, they proposed that the heat transfer processes at the condenser and evaporator were the major contributors to irreversibilities and therefore had to be included in any modeling effort. Therefore, if the first and second laws of thermodynamics, together with the definition of the coefficient of performance were combined, an expression for the COP in terms of evaporator load and heat exchanger temperatures could be obtained. Furthermore, this expression would include the effects of irreversibilities. The following formulations are after Gordon and Ng (1994). Taking into account steady-state conditions, the first law of thermodynamics (energy balance on the system) for cyclic devices yields the expression given in Equation (2).

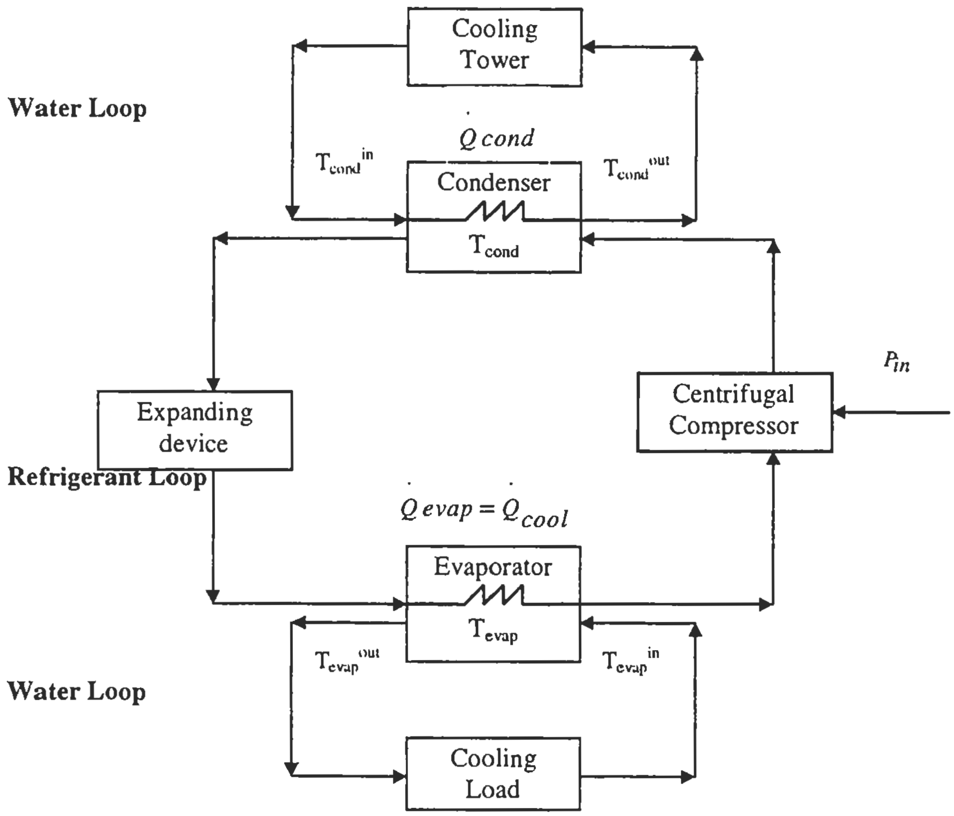

Figure 1. Refrigeration cycle of centrifugal chillers.

$$0 = P_{in} - \dot{Q}_{cond} + \dot{Q}_{evap} \qquad (2)$$

where,

P_{in} = Power input to the compressor (usually electrical power),

$\dot{Q}_{cond}$ = Rate of heat rejection at the condenser

$\dot{Q}_{evap}$ = Rate of heat gain at the evaporator (cooling capacity).

The second law of thermodynamics for cyclic devices can be expressed as:

$$0 = \frac{\dot{Q}_{cond} - \dot{q}_{cond}}{T_{cond}^{in}} - \frac{\dot{Q}_{evap} + \dot{q}_{evap}}{T_{evap}^{out}} \qquad (3)$$

where

T_{cond}^{in} = Condenser inlet water temperature (return from the cooling tower)

T_{evap}^{out} = Evaporator outlet water temperature (supply to the building)

$\dot{q}_{cond}$ = Rate of internal heat loss in the condenser

$\dot{q}_{evap}$ = Rate of internal heat loss in the evaporator.

The heat losses/gains occur from heat leaks in the evaporator and condenser. Next, the coefficient of performance (COP) for refrigeration devices is defined as:

$$COP = \frac{\dot{Q}_{evap}}{P_{in}} \qquad (4)$$

After combining Equations (2), (3), and (4), one obtains

$$\frac{1}{COP} = -1 + \frac{T_{cond}^{in}}{T_{evap}^{out}} + \frac{1}{\dot{Q}_{evap}}\left(\dot{q}_{cond} + \dot{q}_{evap}\frac{T_{cond}^{in}}{T_{evap}^{out}}\right) \qquad (5)$$

Furthermore, if one assumes throttling, fluid pipe friction, and desuperheating losses to be small when compared to heat leaks, and if the heat transfers are approximated linear while the device temperature is kept constant, then the heat losses could be expressed as:

$$\dot{q}_{cond} = -A_0 + A_3 T_{cond}^{in} \quad (6)$$

$$\dot{q}_{evap} = -A_2 + A_4 T_{evap}^{out} \quad (7)$$

These heat transfer rates are very difficult to measure, but must be accounted for. Therefore, after combining Equations (5), (6) and (7), and defining

$$A_1 = A_3 + A_4 \quad (8)$$

the final functional expression of 1/COP was:

$$\frac{1}{COP} = -1 + \frac{T_{cond}^{in}}{T_{evap}^{out}} + \frac{1}{\dot{Q}_{evap}}\left(-A_0 + A_1 T_{cond}^{in} - A_2 \frac{T_{cond}^{in}}{T_{evap}^{out}}\right)$$

(9)

where A_0, A_1, and A_2 are linear regression coefficients that characterize the heat gains/losses in the evaporator and condenser. The three coefficients are determined using actual performance data of an installed, monitored chiller. After obtaining actual performance data, Equation (9) is rearranged as

$$\left(\frac{1}{COP} + 1 - \frac{T_{cond}^{in}}{T_{evap}^{out}}\right)\dot{Q}_{evap} = -A_0 + A_1 T_{cond}^{in} - A_2 \frac{T_{cond}^{in}}{T_{evap}^{out}}$$

(10)

and treated as a linear expression ($Y = mX + b$) where

Y = $\left(\frac{1}{COP} + 1 - \frac{T_{cond}^{in}}{T_{evap}^{out}}\right)\dot{Q}_{evap}$, [kW]

m = $-A_2$, [kW]

X = $\frac{T_{cond}^{in}}{T_{evap}^{out}}$

b = $-A_0 + A_1 T_{cond}^{in}$, [kW].

After plotting Y against X, a linear regression trendline is inserted on a scatter plot format and the slope of the line provides the coefficient $-A_2$. The coefficients A_1 and $-A_0$ are similarly determined. Equation (10) is then rearranged to get the following expression:

$$\left(\frac{1}{COP} + 1 - \frac{T_{cond}^{in}}{T_{evap}^{out}}\right)\dot{Q}_{evap} + A_2 \frac{T_{cond}^{in}}{T_{evap}^{out}} = -A_0 + A_1 T_{cond}^{in}$$

(11)

also treated as a linear expression ($Y' = mX' + b$) where

Y' = $\left(\frac{1}{COP} + 1 - \frac{T_{cond}^{in}}{T_{evap}^{out}}\right)\dot{Q}_{evap} + A_2 \frac{T_{cond}^{in}}{T_{evap}^{out}}$, [kW]

m = A_1, [kW/K]

X' = T_{cond}^{in}, [K]

b = $-A_0$ (kW).

After plotting Y' against X', a linear regression trendline is inserted on a scatter plot format and the slope of the line results in the coefficient A_1 and y-intercept represent $-A_0$.

During the course of this study, it was found that one straight line was produced as shown in Figure 2 and described in the results section. Previous research by Gordon and Ng (1994, 1995), Gordon et al. (1995), and Brandemuehl et al. (1996) indicated that many parallel lines for the linear regression coefficients were produced in similar plots of Figure 2. It was their understanding that, for water-cooled centrifugal chillers, many parallel lines were supposed to be produced out of which $-A_2$ would be determined. This study found otherwise. One of the reasons given for the discrepancy was that for water-cooled condensers, there exists a limited range of condenser inlet water temperatures as opposed to the many temperatures experienced by the condenser of an air-cooled chiller system.

Once the three linear regression coefficients ($-A_0$, A_1, and $-A_2$) are determined, the coefficient of performance may be predicted using Equation (9) and the total energy consumption may also be predicted using Equation (4).

EXPERIMENTAL SET-UP

The experimental aspect of the research began by selecting installed water-cooled centrifugal chillers. It was necessary to obtain experimental data to compare measured and predicted coefficient of performance and total energy consumption in order to determine the validity of the model modifications. The data came from two chillers operating at two institutional campuses: one a South Texas high school, and the other, a Northwest Arkansas university. Both are described below.

One of the water-cooled centrifugal chillers served a high school campus in South Texas where the climate zone is hot and humid. The average annual temperature is 21.7 °C (71 °F). Temperatures in January range from an average low of 6.7 °C (44

°F) to an average high of 17.8 °C (64 °F) and in July range from 23.3 °C (74 °F) to 34.4 °C (94 °F). The average relative humidity is 89 percent at 6 a. m. and 63 percent at 6 p. m. The sun shines during the year on the average 60 to 65 percent of the daylight hours.

The high school campus was approximately 23,885 m^2 (257,000 ft^2) with a main building, academic building, and field house. All HVAC equipment operated only during occupied hours and was turned on manually at 6:00 a. m. and turned off at 8:00 p. m. on weekdays. The main building was 5,019 m^2 (54,000 ft^2) with two stories, brick slab/grade construction, and a flat roof. Space cooling and heating was provided with 17 rooftop units (4 - 7.5 tons each), 70 fan-coil units (0.17 hp each), and one 675 kW (192-ton) water-cooled centrifugal chiller (the one used for this study). The academic wing was 5,641 m^2 (60,700 ft^2) with two stories, brick slab/grade construction, and a flat roof. Space cooling and heating were provided with 118 fan-coil units (112 - 0.05 hp and 6 - 0.17 hp), and one 640 kW (182-ton) centrifugal chiller. The 675 kW (192-ton) water-cooled centrifugal chiller was a commercial chiller and was located in the main building mechanical room. The chiller was built with a multi-stage compressor, a two-stage economizer, and an evaporator-condenser assembly. The chiller worked in conjunction with an evaporative cooling tower. The compressor had fully modulating variable inlet guide vanes that provided capacity control. The compressor was hermetically sealed, two-pole, induction type motor with a low-slip squirrel cage rotor.

For the testing, sensors were installed to monitor the evaporator load (water flow rate, supply and return chilled water temperatures), chiller electrical power consumption (kWh per minute or kWh per hour), and condenser water temperature. The monitored variables were recorded as, one-minute and hourly values. The one-minute data were collected from initial start-up to approximately four hours (236 minutes) of operation. The hourly data were obtained from an existing database (Energy Systems Laboratory, Texas A&M University). The hourly data analyzed represented approximately 6 months of recordings.

The second chiller was located at a university that has approximately 15,000 students, 800 faculty, and 3000 staff, who occupy 270,000 m^2 (2.9 million ft^2) of conditioned building space in more than 160 buildings. The buildings are used for classrooms, dormitories, offices, laboratories, and other ancillary purposes. The physical plant operates centralized mechanical plants to provide both chilled water and steam for building conditioning. There are four primary water-cooled chillers, having a total nominal capacity of 36,225 kW (10,300-tons), used to meet the campus chilled water needs. There is a need for chilled water for all weather seasons of the year. This study focused on a 10,550 kW (3000-ton) centrifugal chiller with a 225 kW (300 hp) circulation pump supplying condenser water.

The average annual temperature for the northwest corner of Arkansas is 8.9 °C (48 °F). Temperatures range from an average low of -2 °C (28 °F) in January to an average high of 32.4 °C (90.3 °F) in August. Temperatures reach 32.2 °C (90 °F) or higher for an average of 60 days each year.

Performance data for the chilled-water system were measured using the physical plant monitoring system. The data presented in this study was collected hourly from May 15, 1997 through July 2, 1997 totaling 1,152 hours. Ambient air conditions ranged between 0.6 °C (33 °F) and 37 °C (99 °F) during the days of interest. Metered points included chilled water flows, evaporator and condenser flow temperatures, chiller electrical power requirements, and ambient air conditions.

RESULTS

The evaluation of the model was based on comparisons of measured data for the COP and total energy consumption to the model predictions. The theoretical COP and total energy consumption predictions agreed with actual data with high accuracy. The results showed that the average measured and predicted coefficient of performance differed by no more than 2 percent. Similarly, the measured and predicted total energy consumption by the compressor differed by no more than 0.1 percent for both cases.

Figures 2 through 7 represent the development and results of the model using data from the chiller serving the South Texas high school campus. In Figure 2, *Y* was plotted against *X* (for every ratio of $\frac{T_{cond}^{in}}{T_{evap}^{out}}$) resulting in a scatter plot. Next, a best-fit line was inserted in which the slope of the line yielded the linear regression coefficient $-A_2$. The coefficient $-A_2$ was calculated to be -763.98 kW (-2.61 x 10^6 Btu/hr). Also, it can be seen at the lower end of the scatter plot, where *X* ranged from approximately 1.065 to 1.085, that a small inconsistency developed. This inconsistency was produced by the "on-off" cycling of the chiller. In Figure 3, Y' was plotted against X' (for every T_{cond}^{in}) also resulting in a scatter plot. A best-fit line was inserted in which the slope of the line yielded the linear regression coefficients of $-A_0$ and A_1, where A_1 represents the slope and $-A_0$ the y-intercept of the line, respectively. In this case, the coefficient $-A_0$ and A_1 were calculated to be -433.95 kW (-1.48 x 10^6 Btu/hr) and 4.36 kW/K (8270 Btu/hr°R), respectively. After the linear regression coefficients were determined these were substituted back into Equation (9). This yielded the following expression:

$$\frac{1}{COP}=-1+\frac{T_{cond}^{in}}{T_{evap}^{out}}+\frac{1}{\dot{Q}_{evap}}\left(-433.95+4.363326T_{cond}^{in}-763.98\frac{T_{cond}^{in}}{T_{evap}^{out}}\right)$$

(12)

Equation (12) could now be used to predict the COP and total energy consumption for this particular test. The initial start-up was neglected since there were no readings of measured power input to the compressor. The measured COP was determined by dividing the measured evaporator load into the measured power input. As for the predicted COP, it was determined by substituting the linear regression coefficients into Equation (12) and taking the inverse of the equation. In Figure 4 it can be seen that the COP was predicted, with good accuracy, close to the measured COP. The average predicted and measured COP were determined by taking an average over the entire test data. The average predicted COP was determined to be 2.87, which was -2.14 percent off of the average measured COP of 2.81.

It can be seen that for the first 60 minutes the COP, was under and over predicted. Also, after approximately one hour of operation the COP was predicted very closely to the measured COP. The reason is that this model was developed specifically for steady-state chiller operation. Therefore, since the chiller did not reach steady-state operation until after one hour of operation, the chiller COP was not predicted accurately during the initial start-up. In Figure 5 it can be seen that the total energy consumption was predicted, also with good accuracy, close to the total measured energy consumption. The average predicted and measured total energy consumption were determined by taking an average over the entire test data. The average predicted total energy consumption was determined to be 15,605.3 kWh (53.248 x 10^6 Btu) which was -0.00311 percent off of the average measured total energy consumption of 15,604.8 kWh (53.247 x 10^6 Btu).

Figures 6 and 7 are characteristic plots that represent the performance of chillers. The kW/ton parameter (the power per ton of refrigeration) should vary linearly with the evaporator load and the inverse of COP (1/COP) should also vary linearly with the inverse of the evaporator load $\left(\frac{1}{\dot{Q}_{evap}}\right)$. Figure 6 shows that the model agreed with measured data in that the kW/ton decreases linearly as the evaporator load increases. Figure 7 also shows that the model agreed with measured data in the fact that 1/COP increased linearly as the inverse of the evaporator load $\left(\frac{1}{\dot{Q}_{evap}}\right)$ increased.

The same approach as described above was used to model the performance characteristics of the 10,550 kW (3000-ton) chiller serving the northwest Arkansas university campus. The resulting expression for 1/COP is as follows:

$$\frac{1}{COP}=-1+\frac{T_{cond}^{in}}{T_{evap}^{out}}+\frac{1}{\dot{Q}_{evap}}\left(-1288.3+12.405T_{cond}^{in}-1309.7\frac{T_{cond}^{in}}{T_{evap}^{out}}\right)$$

(13)

Like in the first case, the average predicted and measured COP were determined by taking an average of the entire performance test data. The average predicted COP was determined to be 4.22, which was 0.45 percent off of the average measured COP of 4.24. Also, the average predicted total energy consumption for the 48 days of operation was within 0.1 percent of the measured energy requirements.

Figures 8 and 9 show the chiller characteristic plots. The first, Figure 8, gives the performance characteristic in terms of kW/ton versus evaporator load (in tons of refrigeration). The predicted linear regression deviated slightly from the measured data resulting in an R-squared value for the predicted linear regression of 0.76. Similarly, Figure 9 shows that 1/COP varies linearly with $\left(\frac{1}{\dot{Q}_{evap}}\right)$. The R-squared value for the predicted linear regression was 0.84 against the measured data.

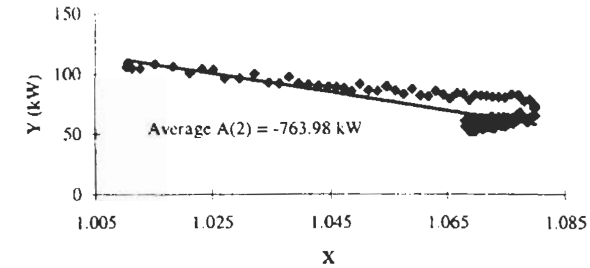

Figure 2. Plot and linear regression to calculate coefficient A_2.

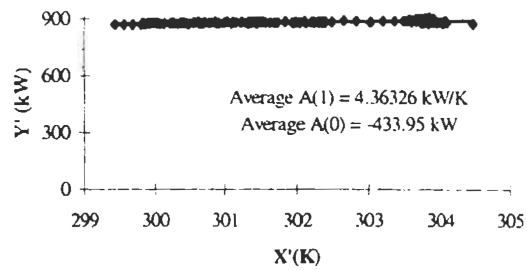

Figure 3. Plot and linear regression to calculate coefficients A_1 and A_0.

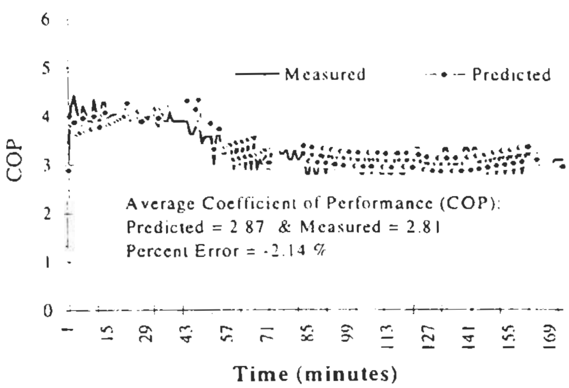

Figure 4. Model prediction of coefficient of performance of the South Texas High School 144 KW (192-ton) chiller.

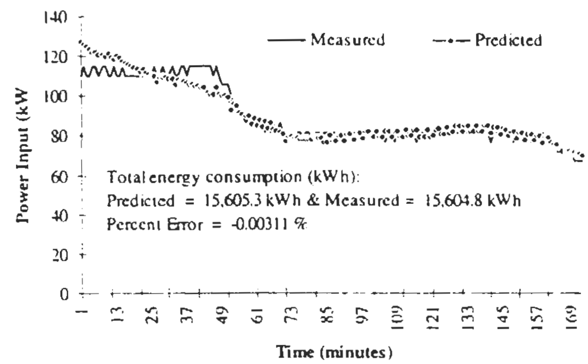

Figure 5. Model prediction of total energy consumption of the South Texas High School 144 KW (192-ton) chiller.

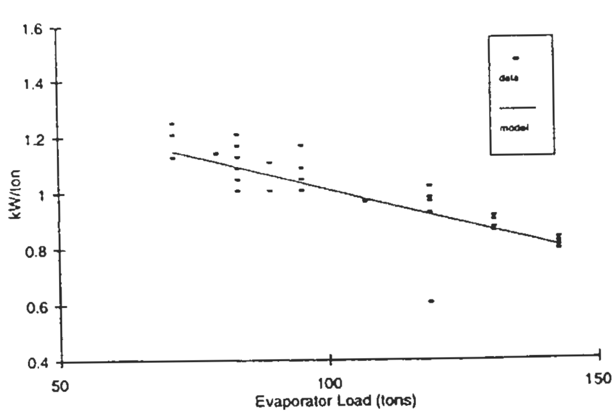

Figure 6. Model predictions of kW/ton vs. evaporator load (in tons) of the South Texas high school 675 kW (192-ton) chiller.

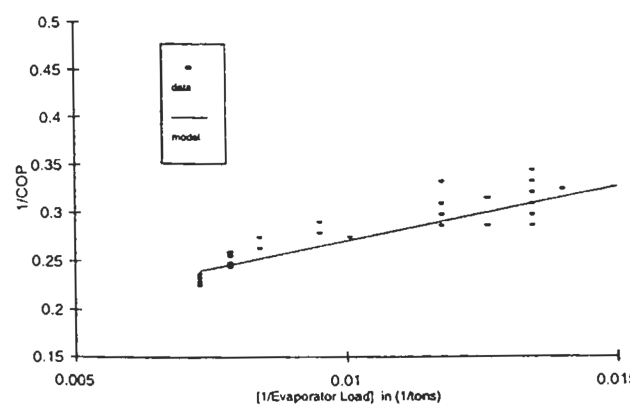

Figure 7. Model predictions of 1/COP vs. [1/evaporator load] (in 1/tons) of the South Texas high school 675 kW (192-ton) chiller

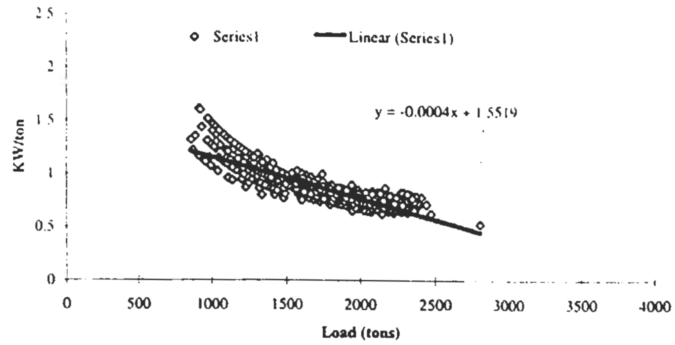

Figure 8. Model predictions of KW/ton versus 1/Qevap for the 10,550 KW (3000-ton) chiller.

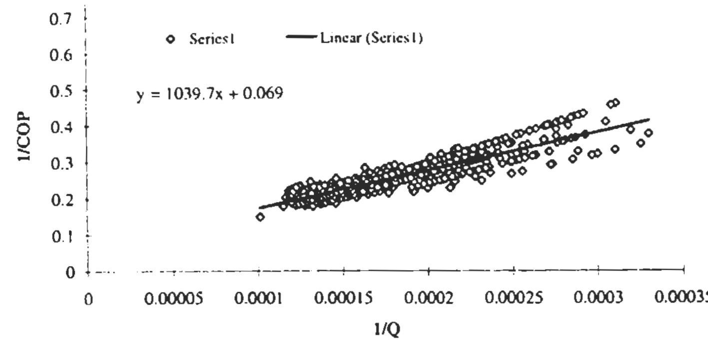

Figure 9. Model predictions of 1/COP versus 1/Qevap for the 10,550 KW (3000-ton) chiller.

SUMMARY AND CONCLUSIONS

The purpose of this research was to develop a thermodynamic model to predict the coefficient of performance and total energy consumption of water-cooled centrifugal chillers for a wide range of operating conditions. It was also developed for diagnostic purposes and for analyzing chiller performance. The study of chiller thermodynamic modeling is fairly new and this model was based on published works that began in 1983. This research also provides a review of the most relevant literature in chiller thermodynamic modeling and a step-by-step procedural analysis that could be used to re-develop the model for future study.

The study described within this paper compared the predictions of the model with actual performance test data to determine the validity of the model. The model had direct inputs such as the cooling capacity (evaporator load), condenser inlet temperature, and evaporator outlet temperature. The output of the model predicted the coefficient of performance and total energy consumption. The thermodynamic model was basically developed by incorporating the first law of thermodynamics, the second law thermodynamics, and the heat transfer processes of the evaporator and condenser into the basic definition of the coefficient of performance of refrigeration devices (chillers). The final functional expression predicted the coefficient of performance and total energy consumption of water-cooled centrifugal chillers. The model predictions were validated by comparing the prediction model with the actual performance test data from a 675 kW (192-ton) water-cooled centrifugal chiller located in south Texas and a 10,550 kW (3,000-ton) centrifugal chiller located in Arkansas. Data collected from the chiller included the evaporator load, water flow rate, supply and return chilled water temperatures, chiller power consumption, and condenser water temperature. It was found that the average COP and total energy consumption predictions were within ± 2 percent and ±0.1 percent, respectively. It should be noted that the numeric coefficients shown in this paper are only valid for the specific chillers and at the conditions described. It is the methods and procedures that are transferable.

Two conclusions were drawn from this work. First, the linear regression coefficient $-A_2$ can be determined from a single regression for water-cooled centrifugal chillers. This differs from previous studies and from air-cooled chillers. Secondly, the modeling approach described in this paper can be used as an accurate method for the prediction of different size water-cooled centrifugal chiller performance. This modeling method can be used as an effective diagnostic tool for chiller performance and / or analysis. For example, the reduction of the condenser inlet water temperature could be studied to determine its effect on the power requirements and energy usage of an installed chiller. Further extension of this work should begin to include the influences of ancillary equipment such as pumps and cooling towers.

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