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Historical Data Based Models for Chilled Water Production from Waste Heat of Turbine

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Abstract: Performance analysis, optimization and environmental load assessment of an absorption system requires accurate but simplified models. The objective of the present study is to develop such models based on non-dimensional parameters (part load ratio, part-load factor and diverter damper position) and ordinary least squares. Since for the case study, the hourly, daily and monthly load demands data are available, the method of averaging over eight years is considered. The models are developed for a system comprised of two heat recovery steam generators and two steam absorption chillers. It was observed that the proposed method is effective in providing better picture of the relationships between the supplied heat and the amount of energy recovered by each subsystem. The maximum cooling load experienced by the two chillers was about 2392RT, which is 4.32% lower than the design capacity. The steam generators were found operating at part load ratio of about 0.41 only. Both chillers deteriorated in performance within the study period. This was confirmed by the part load ratio of 0.8 and steam consumption higher than that required for a new chiller. A generalized model was also identified for the two chillers with the correlation coefficient (R^2), chi-square (χ^2) and Root Mean Squared Error (RMSE) equals to 0.9996, 1.9765e-5 and 0.0044, respectively. The model was found accurate for cooling water temperatures in the range of 29 to 32°C.

Key words: Waste heat recovery, absorption process, steam absorption chiller, heat recovery steam generator

INTRODUCTION

In a cogeneration plant, the waste heat from the turbine can be used to generate steam for driving either steam turbines or absorption chillers. Recovering the energy from the gas turbine waste heat increases overall thermal efficiency of the cogeneration plant to as high as 73 to 90% (Hordeski, 2011; Dincer and Zamfirescu, 2011). For the case of a district cooling plant, the steam is used to energize Steam Absorption Chillers (SAC) to generate chilled water. Among the advantages of using SAC compared to vapor compression chillers are:

- SAC need less electricity to drive the solution and refrigerant pumps
- SAC involve few moving parts hence reduced noise and vibration
- SAC use water and LiBr as working fluids and this assist in protecting the environment

The performances of the SAC are monitored from time to time for reasons of safety, greenhouse gas emissions, reliability and economic concerns. In cases where the performances deteriorate significantly, the SAC are retrofitted or overhauled. Studies on performances of SAC covering various aspects on technical and economics have been undertaken by various authors. Some of the studies are highlighted below.

Kato *et al.* (2001), have developed the environmental load and part load performance assessment that involved optimization. In modeling the cooling load demand, they considered the trend for a typical summer day. Linear relationships between input and output variable was assumed. This assumption fails to recognize the nonlinear trend in off-design operation.

Chow *et al.* (2002) adopted neural network models to capture the relationship between fuel consumption and cooling load. Separate models were produced for the coefficient of performance as well. The neural network models were trained by Levenberg-Marquardt (LM) algorithm. The training data was a collection of experimental data and data collected from manufacturer's catalogue.

Braun (2006) discussed the different models common in control optimization for absorption chillers. Models are also developed in terms of part load ratio and part load factor. While assuming multivariable linear regression, the models considered cooling water inlet temperature and cooling load as independent variables.

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Most reported studies on cooling loads used data for a new system. This assumption is valid in the optimization of operating strategies for a new system. However it is not applicable for old systems whose performance dropped by some percent due to fouling and malfunctions. Hence if cooling load demand models for an old system are to be formulated, recent operation data should be used.

The objective of the study is to develop a model for an absorption process using waste heat from a cogeneration plant. The model uses actual plant operation data collected over eight years. However, for the generalized model, it uses data from literatures.

MATERIALS AND METHODS

The system adopted for modeling is an absorption system for a district cooling plant. The system configuration is as shown in Fig. 1. The system consists of two Heat Recovery Steam Generators (HRSG), one unit Auxiliary Gas Boiler (AGB) and two units SAC. AGB will be used for the case maximum cooling load is required and when one of the HRSG is down for maintenance.

The approach adopted is to treat each subsystem separately. Conservation of mass and energy equations are used to derive a governing equation for each subsystem. It is assumed that all data for working fluid properties are available. The final models are formulated in terms of normalized quantities. To normalize each variable, the design point data for a new system is considered.

Heat recovery steam generator: The HRSG use the energy in the exhaust gas from two gas turbines rated 5.2 MW at ISO condition. The function of the HRSG is to produce saturated steam at 0.85 MPa by taking feed water at a temperature of about 90°C. For i-th HRSG, assuming a steam flow rate $\dot{m}_s^{(i)}$, the energy transferred to the feed water is calculated as (Treado *et al*., 2011):

$$\dot{Q}_{H,st}^{(i)} = \dot{m}_s^{(i)}(h_g-h_f), \qquad i = 1,2 \tag{1}$$

where, h_g is enthalpy of saturated vapor at 0.85 MPa; h_f is enthalpy of saturated liquid at 90°C. Disregarding the effect of the diverter damper, the energy available to the system that includes the gas turbine is:

$$\dot{Q}_{total}^{(i)} = \dot{m}_{f,gas}^{(i)}LHV_{gas}+\dot{m}_{f,oil}^{(i)}LHV_{oil} \tag{2}$$

where, $\dot{m}_{f,gas}^{(i)}$ and $\dot{m}_{f,oil}^{(i)}$ are flow rate of gas fuel and oil fuel, respectively; LHV is lower heating value in kJ kg^{-1} K^{-1}.

Performance of the HRSG can be analyzed based on the ratio between Eq. 1 and 2. The approach is suitable to quantify the amount of input energy recovered by the gas turbine and HRSG. However, it hides the exact amount of energy going to the HRSG. The energy truly available for the HRSG is:

$$\dot{Q}_g^{(i)} = \dot{m}_g^{(i)}(h_{g,in}^{(i)}-h_{g,out}^{(i)}) \tag{3}$$

where, $\dot{m}_g^{(i)} = \dot{m}_{exg}^{(i)}-\dot{m}_{g,bypass}^{(i)}$ is mass flow rate of the waste heat used in the HRSG; $\dot{m}_{exh}^{(i)} = \dot{m}_{air}+\dot{m}_f$; $\dot{m}_f$ and $\dot{m}_{air}$ are mass flow rate of waste heat and air, respectively. h_g is enthalpy of the waste heat:

$$h_g = \int_{T_0}^{T} C_{p,g}(T)dT$$

T is temperature in K; $C_{p,g}$ is specific heat of the waste heat in kJ kg^{-1} K^{-1}. For the calculation of properties of steam and waste heat, empirical equations adopted from Hinrichen and Pritchard (2011) and Widiyanto *et al*. (2003), respectively, are applied.

In cases where mass flow rate of the waste heat at part load is not known, the energy available to the HRSG can be estimated by the following relation:

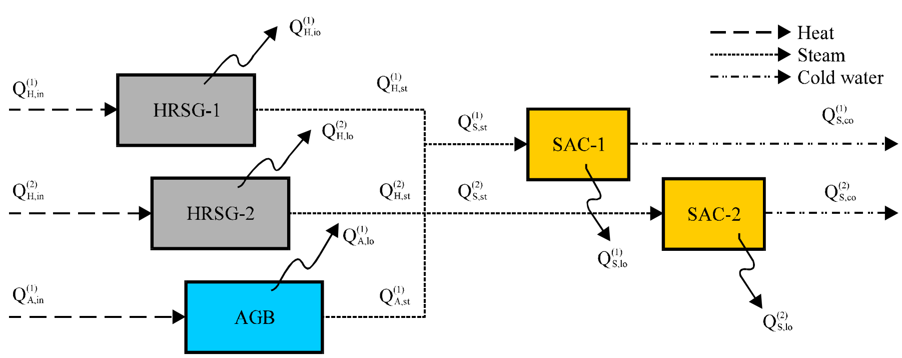

Fig. 1: Flow sheet for steam and absorption process for the district cooling plant

$$\dot{Q}_{H,in}^{(i)} = [1-\eta_{GTG}(1-\alpha)]\dot{Q}_{total} \quad (4)$$

where, η_{GTG} is efficiency of the Gas Turbine Generator (GTG) connected to the HRSG; α is a constant assumed to account for the energy lost in the GTG. Note that, Eq. 4 assumes all the lost energy in the GTG is available to run the HRSG. η_{GTG} varies in the range of 0.15 to 0.29.

Auxiliary gas boiler: The AGB is a direct fired one through boiler. It is comprised of evaporator, economizer and blow-down heat exchanger. Since it is a dual fuel boiler and produces saturated steam, Eq. 1 and 2 can be used to calculate the energy transferred to the feed water $\dot{Q}_{A,st}$ and the input energy $\dot{Q}_{A,in}$, respectively.

Steam absorption chiller: The SAC produce chilled water using steam generated by HRSG. The SAC being studied is a two-stage absorption chiller of LiBr and H_2O type. Part load operation of an SAC can be evaluated applying Eq. 5 and 6:

$$\dot{Q}_{S,st}^{(i)} = \dot{m}_s^{(i)}(h_g-h_f), \qquad i = 1,2 \quad (5)$$

$$\dot{Q}_{S,co}^{(i)} = \dot{m}_{chw}^{(i)}(h_{in}-h_{out}), \qquad i = 1,2 \quad (6)$$

where, $\dot{Q}_{S,st}^{(i)}$ is the steam energy available to drive the SAC; $\dot{Q}_{S,co}^{(i)}$ the cooling effect actually available from the system; $\dot{m}_s^{(i)}$ and $\dot{m}_{chw}^{(i)}$ are the mass flow rate of steam and chilled water, respectively. h_g and h_f and h_{in} and h_{out} are enthalpies of steam and chilled water, respectively.

The model adopted for instant energy usage for each subsystem in the absorption process has been simplified for analysis. For i-th (i = 1, 2, ..., n_s) subsystem the energy usage are defined as:

$$\dot{Q}_{in}^{(i)}(k) = \left(\frac{\dot{Q}_{rated}^{(i)}}{K_{rated}^{(i)}}\right).PLF^{(i)}.\gamma^{(i)}(k) \quad (7)$$

where, $\dot{Q}_{rated}^{(i)} \in R$ is the rated capacity of the subsystem; $\dot{Q}_{in}^{(i)}(k) \in R$ is hourly, weekly or yearly energy consumption; $PLF^{(i)}$ is the part load factor; $\gamma^{(i)}(k) \in \{0,1\}$ is the control variable; $K_{rated}^{(i)}$ is performance at the rated condition. For the case of HRSG and ABG, it is equal to thermal efficiency. While for SAC it is equal to Coefficient of Performance (COP).

The Part Load Factor (PLF) for a given system is often correlated with the Part Load Ratio (PLR) (Treado *et al*., 2011) and it is defined as the ratio between assumed load and rated capacity. The relationship is normally determined from experimental data. For a new system, the manufacturers may provide the graph for PLF versus PLR. For the system in service, the graph has to be redefined as the system may deteriorate with time.

Parameter estimation: The model relating the PLF with PLR is considered as the following form:

$$y^{(i)} = X^{(i)}\theta^{(i)} \quad (8)$$

Where, i = {1, 2, ..., n_s} is the index for a subsystem in the absorption process. In the present work, since two HRSG and two SAC are available, i is limited to a maximum of 2. $y^{(i)}$ is the output vector; $X^{(i)}$ is the regression matrix constructed from input vectors; $\theta^{(i)} \in R^{m+1}$ is vector of model parameters:

$$X^{(i)} = \begin{bmatrix} 1 & x_{1,1} & \cdots & x_{1,m} \\ 1 & x_{2,1} & \cdots & x_{2,m} \\ \vdots & \vdots & \ddots & \\ 1 & x_{Nd,1} & \cdots & x_{Nd,m} \end{bmatrix} \text{ and } y^{(i)} = [y_1, y_2, \ldots, y_{Nd}]^T$$

The solution for Eq. 8 is obtained applying ordinary least squares. It can be shown that (Hinrichen and Pritchard, 2011), the optimum estimate for the model parameters is given by:

$$\hat{\theta}^{(i)} = [X^{(i)T}X^{(i)}+\alpha I]^{-1}X^{(i)}.y^{(i)} \quad (9)$$

where, $I \in R^{(m+1)\times(m+1)}$ is an identity matrix and α is the regularization parameter assumed to avoid singular condition.

Model performance parameters: There are several statistical methods applied to evaluate the performance of a model. In the present work, three of the most commonly used methods are selected. These include correlation coefficient (R^2), the reduced chi-square (χ^2) and Root Mean Squared Error (RMSE).

RESULTS AND DISCUSSION

Universiti Technologi PETRONAS gas district cooling plant has been considered as a case study for testing the proposed approaches. The setup has two HRSGs, each with separate supply of exhaust gas. The steam from the two HRSGs is collected in the steam header. The steam from the steam header is supplied to SAC. Nominal capacity of one HRSG is 12 ton h^{-1}. Design point data for all subsystems are given in Table 1. Schematic diagram for the HRSG is shown in Fig. 2. As can be seen from Fig. 2, the HRSG is a natural circulation drum boiler with economizer and blow-down heat exchanger. Flow rate of the exhaust gas going to the HRSG is controlled by varying the position of the diverter damper.

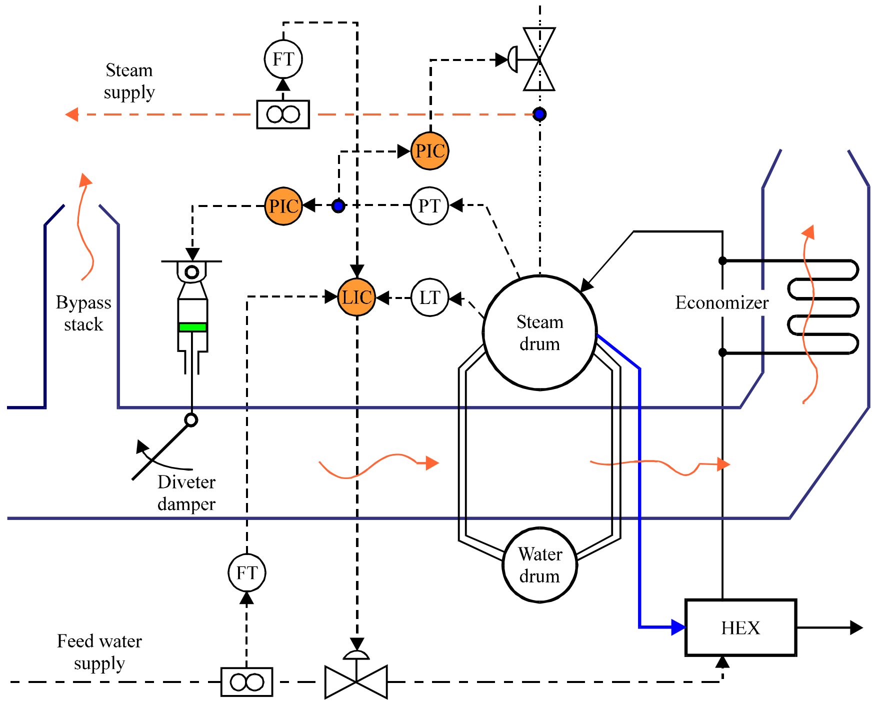

Fig. 2: Schematic diagram for the heat recovery steam generator

Table 1: Design point data for the main components in the steam-chilled water system

Subsystem	Capacity
HRSG	12 ton h^{-1}, saturated steam at 0.85 MPa
HRSG-pump	15.5 m^3 h^{-1} (liquid temp. less 100°C) and 11 kW
AGB	6 ton h^{-1}, saturated steam at 0.85 MPa
AGB-pump	7.2 m^3 h^{-1} (liquid temp. less 100°C) and 5.2 kW
SAC	1250 RT
SAC-pump	920 m^3 h^{-1} (liquid temp. less 100°C) and 7.5 kW

RT: Ton of refrigeration

Table 2: Monthly average (μ) and standard deviation (σ) for the year 2004 to 2011

Month	μ	σ	Month	μ	σ
January	1593	331	July	1766	200
February	1767	284	August	1976	325
March	1840	185	September	1813	247
April	1934	202	October	1769	183
May	1949	211	November	1780	154
June	1818	166	December	1655	190

Models for cooling loads: The models for cooling load demands are constructed applying statistical analysis on actual data collected over eight years. The calculated trends on hourly, daily and monthly basis are shown in Fig. 3a-c, respectively. The peak hours were between 8:00 a.m. and 6:00 p.m. In the monthly case, the peak in a particular month of the year is included for the estimation of average value over eight years. The mean μ and standard deviation σ of the cooling load demand corresponding to each month in a year is depicted in Table 2. The maximum cooling load experienced by the system is about 2392 RT, Fig. 3c. The trend over the year is useful in addressing optimization of operating strategies as chance constrained problem, which is an assumption closer to the reality on the ground. In (Kato *et al.*, 2001; Widiyanto *et al.*, 2003), the hourly heat and cooling demands are set to a typical model without mentioning how it is calculated. It can be said that, given the eight years data, the proposed models are better representations of the variations in the cooling load demands. Since environmental load and part load performance assessment involves these kinds of models, the result is seen as a good contribution.

Models for the SAC, HRSG and AGB: Kato *et al.* (2001), assumed that any output variable is linearly correlated to the corresponding input. The linear assumption, though easy to apply, is not always true as verified by data from actual plants. Manufacturer supplied performance map for a 1250 RT steam absorption chiller and other similar designs (Braun, 2006; Yamaguchi and Shimoda, 2010; Petchers, 2003; Matsushita *et al.*, 2002) is shown in Fig. 4. The actual steam needed for a certain refrigeration effect is a function of cooling water temperature. As can be seen from Fig. 4, the trend at some operating points could be better approximated by higher order polynomials. For the 1250 RT chillers, model parameters estimated according to

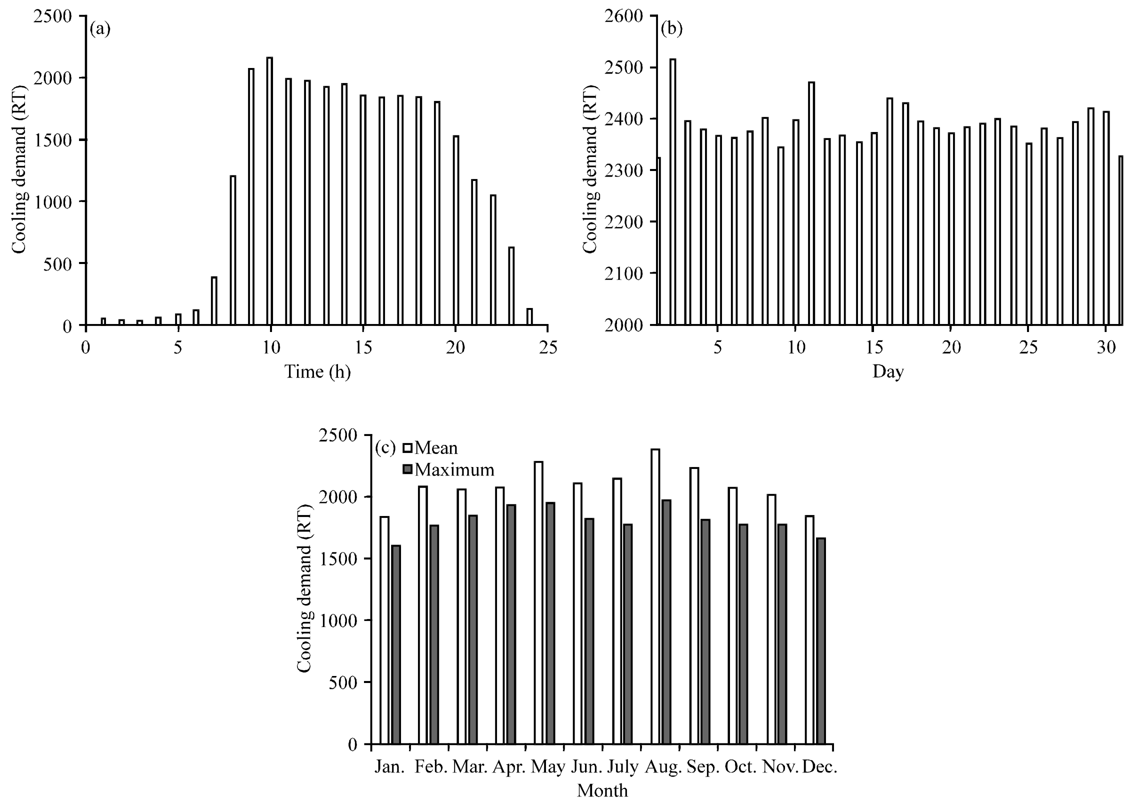

Fig. 3(a-c): Cooling load demand at (a) Hourly, (b) Daily and (c) Monthly bases

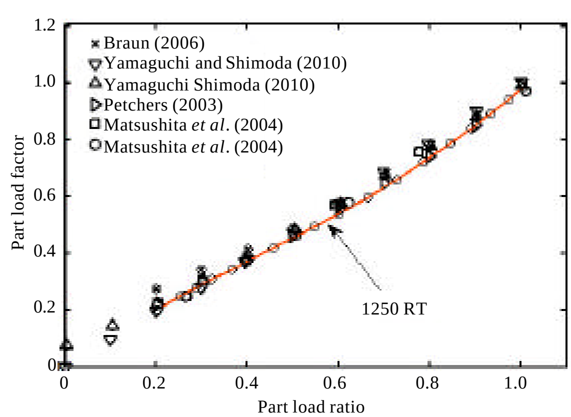

Fig. 4: Performance map for various designs

Eq. 9 is given in Table 3. The R^2 value is close to 1 and χ^2 and RMSE are both small indicating high correlation between the actual and predicted result. A generalized model is also produced fitting all the data in Fig. 4 by a single second order model. The estimated model parameters are θ_0 = 0.0532, θ_1 = 0.7313 and θ_2 = 0.1942. The corresponding performance parameters are correlation coefficient (R^2) 0.9938, chi-square (χ^2) 4.6098e-4 December 15, 2012 and Root Mean Squared Error (RMSE) 0.0210. However, the developed models could not be used for a deteriorated SAC.

Table 3: Part-load model parameters for HRSGs and SACs

Model parameter	HRSG-1	HRSG-2	SAC-1	SAC-2
θ_0	-0.0873	-0.0873	0.0954	0.0954
θ_1	1.5211	1.5211	0.5482	0.5482
θ_2	-	-	0.3233	0.3233

θ_i (i = 1, 2, 3) are constants in the regression Eq. 8

Based on year 2006 data, for SAC-1, 88% of the operating points indicate that the system was run at part load ratio of 0.8835. In case of SAC-2, 80% of the operating points were located at part load ratio of 0.93. Both chillers were running with steam consumptions higher than normally required when they were new. The deteriorated performance could be attributed to fouling in the cooling water loop, crystallization due high temperature in the generators and air leak in to the system. Based on the observations, it can be said that direct use of a performance map supplied by a manufacturer to study SAC that served for more than three to eight years may lead to erroneous result. To overcome the problem, models were developed based on recent actual operation data. The new models are presented as follows:

Table 4: Models performance parameter

Parameter	HRSG-1	HRSG-2	SAC-1	SAC-2
R^2	0.7901	0.7901	0.9996	0.9996
χ^2	0.0021	0.0021	1.9765e-5	1.9765e-5
RMSE	0.0446	0.0446	0.0044	0.0044

- For SAC-1: θ_0 = -0.1100, θ_1 = 2.2183 and θ_2 = 0.6625. The corresponding performance parameters are R^2 = 0.9084, χ^2 = 0.0063 and RMSE = 0.0795
- For SAC-2: θ_0 = -0.2171, θ_1 = 3.5709 and θ_2 = -2.3406, with corresponding performance parameters of R^2 = 0.8958, χ^2 = 0.0061 and RMSE = 0.0783

For the new models, the lower value of R^2 and higher values of χ^2 and RMSE, respectively, are due to measurement noise. Besides, it was assumed that the cooling water inlet temperature is in the range of 29 to 32°C, which may be true all the time.

For the HRSG, the heat input is a function of the fuel flow rate and variable inlet guide vane position in the GTG supplying the waste heat to the heat recovery steam generator. Since the waste heat temperature and flow rate were also involved, the final model for the heat going to the heat recovery system must be a function of these variables. It is possible to develop the performance map for the HRSG relying on data obtained by simulating thermodynamic models for the GTG and HRSG. However, in the present work, the steam flow rate from the HRSG was modeled as a function of the diverter damper position (θ_{DD}) using actual plant operation data. The parameters estimated using Eq. 9 are given in Table 3 while the corresponding performance parameters are listed in Table 4. For the two chillers, the developed models are featured by a correlation coefficient close to one indicating higher accuracy. The models identified in this section are applicable for thermo-economic and environmental load assessment.

The AGB uses gas fuel directly. Since there is no reliable data available, model for AGB could not be formulated.

CONCLUSION

The objective of this study is to develop models for an absorption process that is easy to use in thermo-economic and environmental load assessment and optimization of operating strategies. From the analysis results, the following conclusions are made:

- For cooling water temperatures in the range of 29 to 32°C, part load performance of SAC can be accurately described by a single generalized curve
- The HRSG and SAC are all seen deteriorated in performance. Hence, the models provided by the manufacturers and the generalized model could not be used directly. Here, it is recommended to work on a fault detection and diagnostic system which can be applied to early identify the cause for performance drop
- The maximum cooling load experienced by the absorption system is 2392RT, which is 4.32% lower than the design capacity. Since the system is underutilized, it is necessary to implement better operating strategies

The method discussed in the present work can be extended to electric chillers and gas turbines. Future work will focus on the use of the developed models in thermo-economic, environmental load and performance optimization studies.

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