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Analytical Models for Energy Eudit of Cogeneration Plant

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Abstract: The energy losses and mismanagement can degrade the performance of the cogeneration plant as well as increasing the economical cost of energy usage. Energy analysis models can be used as an effective mean for energy audit and management of the cogeneration plant. This paper presents the development of analytical models for energy audit of a cogeneration plant. Most of the energy losses occurred in a cogeneration plant are due to the imbalances of energy in the sub-systems of the plant. An analytical tool is developed and written in a computer program using the available specifications of component from an existing cogeneration plant. Actual operating data of the plant is used to verify the models and the energy utilization of the plant. Comparison between the actual operation performance and the optimum operating conditions provides means to improve the real time performance of the overall plant. The sub-systems in the cogeneration plant analyzed are, gas turbine engine, heat recovery steam generator, and steam absorption chiller. Implementation of the analytical tool found that energy loss in gas turbine engine could be minimised by changing the inlet air temperature and air mass flow rate.

Key words: Energy audit, analytical models, cogeneration plant, heat recovery steam generator, steam absorption chiller, plant optimization

INTRODUCTION

Detail energy audit is an important part in the operation of a power plant since it can identify energy losses due to the operating parameters variations and imbalances of energy quantities in the sub-system of the utility. Detail audit can be carried out by analyzing the operating data of the utility, which can be undertaken by a computer program (Bhatt, 2000). The analysis enables identifying areas of energy imbalances and losses, which could provide the opportunity for measures to be taken for energy recovery.

This paper discusses the analytical models for energy audit of a cogeneration plant consisting of a Gas Turbine Engine (GTE) with a Heat Recovery Steam Generator (HRSG) and a Steam Absorption Chiller (SAC) to produce both power and heat simultaneously. Plant performance improvement using analytical models has been reported by several researchers. Modeling and numerical optimization presented by (Bassily, 2005) found that 1% higher cycle efficiency can be gained by a dual-pressure reheat combine-cycle plant compared to the standard designed combine cycle. Casarosa *et al*. (2004) developed analytical and numerical mathematical methods for optimization of HRSG operating in a combined cycle plant. The tool was able to record an efficiency of 60% when the HRSG is optimized. Second law analysis of a waste HRSG, an equation for entropy generation number, was derived and used for analysis (Bucher and Reddy, 2007). Observation revealed that a particular non-dimensional inlet gas temperature difference and for other fixed parameters, entropy generation number is minimum at particular number of transfer unit. The entropy generation number increases with increase in non-dimensional hot flue gas inlet temperature difference ratio due to higher temperature difference between stream to steam (flue gas and water/steam) which increases total irreversibility. An energy analysis of technological systems of natural gas fired combined heat-and-power plants found that, not only the plant have high energy efficiency but also have low harmful impact on the environment (Zaporowski and Szczerbowski, 2003; Xu and Dai, 1997) reported on the optimization of absorption chiller of both parallel and series flow type based on thermodynamic study. It is found that the optimum results of the series flow type are similar to that of parallel flow, though the parameters at various state point and components are different (Xu *et al*., 1996). A general thermodynamic model for prediction performance that can be used as a diagnostic tool for absorption chiller performance is presented by (Gordon and Choon, 1995).

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In the current research work, analytical models for several components of a cogeneration plant are developed to simulate the energy utilization of the components. Actual recorded data from the plant were used to verify the models. Results generated by comparing the energy utilization between the models and the actual data provided means to optimize the plant performance.

DESCRIPTIONS OF THE COGENERATION PLANT

The cogeneration plant located on a site called the Gas District Cooling (GDC) produces both electric power and chilled water for space cooling in Universiti Teknologi PETRONAS (UTP). The GDC plant consists of gas turbine engine, heat recovery steam generator, steam absorption chiller, air cooled chiller, cooling tower, and thermal energy storage. The waste heat from the exhaust of the gas turbine is utilized for steam production that is used for heating in the steam absorption chiller. The chilled water produced from the steam absorption chiller is used for air conditioning of the buildings in UTP. Cogeneration is considered as a form of energy conservation process, because of the heat energy recovery that would have been lost from the gas turbine exhaust.

ENERGY MODELS DEVELOPMENT

The analytical tool for energy audit of the cogeneration plant energy models is developed based on the principle of first law of thermodynamics, mass and energy balance models. The analytical method formulated for the systems include the Gas Turbine Engine (GTE), Heat Recovery Steam Generator (HRSG) Steam Absorption Chiller (SAC). The analytical models for all the systems are written in script files in the matlab program. Specifications of the components were obtained from the manufacturer of the component. The validation of the models was done based on the actual operating data collected from the Gas District Cooling (GDC). The results from the model are compared with the actual data for each of the sub-systems. The results are displayed to indicate the variation of energies when the plant is operated at the local condition. This information provides the state of efficiencies of the sub-systems and the overall performance of the plant. The program enables intervention by the operator to take corrective measures to optimize the sub-system in the plant. The governing equation for the various systems models are described in the following sections.

Gas turbine engine model: The GTE energy model consists of three sub-systems, i.e., air compressor model, combustion chamber model, and the turbine model. The assumptions made here are as follows:

- The turbine engine is at steady state
- Air is an ideal gas
- The pressure ratio is constant both in compressor and turbine
- Heat lost due to lubrication is eliminated

Air compressor energy model: Compressor work can be determined from first law of thermodynamics as:

$$\dot{Q}-\dot{w}_c=\dot{m}_a\left[(h_{2a'}-h_a)+\frac{v^2_{2a'}-v_a^2}{2}+g(z_{2a'}-z_a)\right] \quad (1)$$

At steady state the kinetic and potential energy can be ignored and the compressor work can be obtained by:

$$\dot{w}_c=\dot{m}_a(Cp_{2a'}T_{2a'}-Cp_aT_a) \quad (2)$$

The specific heat of air at ambient temperature, T_a and compressed temperature, T_{2a} after compressor are respectively calculated as a function of temperature only and the constant values are taken from (Alhazmy and Najjar, 2004).

Where $T_{2a'}$ is the temperature leaving the air compressor at isentropic efficiency, η_c, calculated by;

$$T_{2a'}=T_a+\frac{T_a}{\eta_c}\left(r_c^{\frac{Y-1}{Y}}-1\right) \quad (3)$$

Isentropic efficiency η_c can be calculated by the correlation given in (Alhazmy and Najjar, 2004) as:

$$\eta_c=1-\left(0.04+\frac{r_c-1}{150}\right) \quad (4)$$

Knowing the input parameters of $\dot{m}_a$, T_a, r_c the program computes values η_c, $T_{2a'}$, $Cp_{2a'}$, CP_a and finally w_c. These provide the energy required to compress the air.

Combustion chamber energy model: The combustion chamber is supplied with energy of the compressed air and energy of the fuel, which would produce combustion of the mixture. The energy balance equation of the first law requires that this energy to equal the energy of the output hot gases. The difference between the inlet and outlet total energy would indicate the losses in the system. Assuming that the combustion of fuel is complete and the product of combustion is CO_2, H_2O, N_2, O_2 only.

Energy input to combustion chamber by the compressed air and fuel are:

$$\dot{Q}_{in,2a'} = \dot{m}_a CP_{2a} T_{2a'} \tag{5}$$

$$\dot{Q}_{in,f} = \dot{m}_f LHV \tag{6}$$

The mass balance is calculated by:

$$\dot{m}_a + \dot{m}_f = \dot{m}_g \tag{7}$$

Specific heat of hot gases, C_{pg} is again calculated as a function of temperature only the constant are obtained from (Cengel and Boles, 2007). Knowing the gas temperature, T_g at the outlet of the combustion chamber, exit energy of the combustion chamber can be determined by:

$$\dot{Q}_{g,out} = \dot{m}_g Cp_g T_g \tag{8}$$

Hence energy lost in combustion chamber is calculated by;

$$\dot{Q}_{cc,los} = \dot{m}_a Cp_{2a'} T_{2a'} + \dot{m}_f LHV - \dot{m}_g Cp_g T_g \tag{9}$$

Turbine energy model: The energy input to the turbine is supplied by the hot gases from combustion chamber, to convert thermal energy to mechanical energy in the form of work. However, not all of the thermal energy is utilized and subsequently some of the energy is lost and escape with the hot gases in the flue. A portion of the work produced by the turbine is used to drive the air compressor. Thus, the difference between the turbine work output and air compressor work input is known as the net work.

Temperature of hot gas entering turbine is estimated by:

$$T_g = \frac{\dot{m}_f LHV \eta_{cc} + \dot{m}_a Cp_{2a'} T_{2a'}}{\dot{m}_g Cp_g} \tag{10}$$

Energy input to the turbine is the same energy which exits the combustion chamber, Eq. 8. Energy lost from the turbine is calculated by:

$$\dot{Q}_{t,los} = \dot{m}_g Cp_{gTET} T_{TET} \tag{11}$$

Turbine work is calculated by:

$$\dot{w}_t = \dot{m}_g (Cp_g T_g - Cp_{gTET} TET) \tag{12}$$

The turbine exhaust temperature, TET can be determined from the polytropic relations for ideal gases as:

$$TET = T_g + \eta_t T_g \left(\left(\frac{1}{r_c} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \tag{13}$$

where, the isentropic efficiency of the turbine η_t is estimated using correlation given in (Alhazmy and Najjar, 2004):

$$\eta_t = 1 - \left(0.03 + \frac{r_c - 1}{180} \right) \tag{14}$$

The network and total energy lost of the gas turbine engine are calculated by:

$$\dot{W}_{net} = \dot{W}_t - \dot{W}_c \tag{15}$$

$$\dot{Q}_{GTE,los} = \dot{Q}_{cc,los} + \dot{Q}_{t,los} + \dot{W}_{c,los} \tag{16}$$

Heat Recovery Steam Generator model (HRSG): Hot flue gas from the GTE flow through HRSG section before exhausting to the atmosphere. HRSG consists of several sub-systems, including evaporator, economizer, and steam drum. Returning water from the SAC flow through the economizer during which some of the exhaust gas is transferred to the water and exit as warm water used as the boiler feed water. The water from the steam drum goes through the evaporator which will receive heat from the hot gases before returning to the steam drum at saturated temperature. The assumptions made for the HRSG are as follow:

- The pressure variations in components are neglected
- The system is at steady state
- There is no leakage of hot gases and steam

HRSG evaporator energy model: The energy balance in the evaporator is the heat energy supplied by the flue gases which must be equal to energy gained by the steam plus energy leaving the evaporator and the energy losses in the evaporator.

Energy entering HRSG from gas turbine engine in calculated by:

$$\dot{Q}_{HRSG} = \dot{m}_g cp_{gTET} TET \tag{17}$$

Energy supplied by hot gases in evaporator in found from:

$$\dot{Q}_{ev,s} = \dot{Q}_{ev,in} - \dot{Q}_{ev,exit} = \dot{m}_g (h_{ev,in} - h_{ev,exit}) \tag{18}$$

Heat energy in saturated water is calculated by:

$$\dot{Q}_{sw} = \dot{m}_{sw} cp_{sw} T_{sw} \quad (19)$$

Energy gained by the steam in evaporator:

$$\dot{Q}_{steam,g} = \dot{Q}_{steam} - \dot{Q}_{sw} = \dot{m}_{steam}(h_{steam} - h_{sw}) \quad (20)$$

Energy lost in the evaporator:

$$Q_{ev,los} = (\dot{Q}_{ev,in} - \dot{Q}_{ev,exit}) - (\dot{Q}_{steam} - \dot{Q}_{sw}) \quad (21)$$

HRSG economizer energy model: Due to energy inequality in the economizer, the energy supplied by the flue gas would not be equal to the energy gained by the warm water; hence some energy would be lost.

Energy entering the economizer:

$$\dot{Q}_{ec,in} = \dot{m}_g cp_{ec,in} T_{ec,in} \quad (22)$$

Energy supplied by the hot gases in the economizer:

$$\dot{Q}_{ec,s} = \dot{m}_g (h_{ec,in} - h_{fg,los}) \quad (23)$$

Energy gained by warm water in economizer:

$$\dot{Q}_{ww} = \dot{m}_{ww}(h_{sw} - h_{ww}) \quad (24)$$

The energy lost can be calculated by:

$$Q_{ec,los} = (\dot{Q}_{ec,in} - \dot{Q}_{fg,los}) - (\dot{Q}_{sw} - \dot{Q}_{ww}) \quad (25)$$

Energy lost by the flue gases from the economizer:

$$Q_{fg,los} = \dot{m}_g cp_{fg} T_{fg} \quad (26)$$

Total energy lost in the HRSG system is calculated by:

$$Q_{HRSG,los} = Q_{ev,los} + Q_{ec,los} + Q_{fg,los} \quad (27)$$

Steam Absorption Chiller models (SAC): The steam absorption chiller consists of sub-systems such as; high and low temperature generator, condenser, evaporator, absorber, high and low temperature heat exchanger, refrigerant pump, and solution pump. The energy exchange in the sub-systems of the SAC is between the refrigerant Lithium bromide/water and the cooling water in the absorber and condenser, chilled water in the evaporator and steam in the high temperature generator. The assumptions made for SAC are as follows:

- The system is at steady state
- Heat lost due to friction in the components are neglected
- Refrigerant lost is neglected

Mass balances in SAC are as follows: The refrigerant mass balance in absorber is given by:

$$\dot{m}_{1s} = \dot{m}_{cs,ab} + \dot{m}_r \quad (28)$$

The mass balance in low temperature generator:

$$\dot{m}_{cslg} = \dot{m}_{vlg} - \dot{m}_{1s} \quad (29)$$

The vapor mass exit high generator is given by:

$$\dot{m}_{vhg} = 0.5\dot{m}_r \quad (30)$$

The mass of solution exit low generator is given by:

$$\dot{m}_{ws} = \dot{m}_r + \dot{m}_{vlg} \quad (31)$$

The mass balance in the high generator is given by:

$$\dot{m}_{vhg} = \dot{m}_{1s} - m_{cs,hg} \quad (32)$$

Energy balance in the steam absorption chiller can be drawn as follows,

Absorber energy model: Energy rejected by cooling water in absorber:

$$\dot{Q}_{ab,cw} = \dot{m}_{cw}(h_{1'} - h_1) \quad (33)$$

Energy added by the refrigerant in absorber:

$$\dot{Q}_{ab,r} = \dot{m}_r h_{3'} + \dot{m}_{cs,ab} h_{7'} - \dot{m}_{1s} h_4 \quad (34)$$

Energy lost in absorber can be calculated by:

$$Q_{ab,los} = \dot{Q}_{ab,r} - \dot{Q}_{ab,cw} \quad (35)$$

High Temperature Generator (HTG) energy model: Energy gained in high temperature generator is calculated by:

$$\dot{Q}_{hg} = \dot{m}_{vhg}h_6 + \dot{m}_{csg}h_{5'} - \dot{m}_{1s}h_{6'} \quad (36)$$

Energy added to high temperature generator by the steam:

$$\dot{Q}_{hg,d} = \dot{m}_s h_{11'} - \dot{m}_d h_{11} \quad (37)$$

Energy lost in high temperature generator:

$$Q_{hg,los} = \dot{Q}_{hg} - \dot{Q}_{hg,d} \quad (38)$$

SAC condenser energy model: Energy added by refrigerant in condenser is calculated by:

$$\dot{Q}_{cd} = \dot{m}_{vlg}h_{10'} + \dot{m}_{ws}h_{10} - \dot{m}_r h_3 \quad (39)$$

Energy rejected by cooling water in condenser:

$$\dot{Q}_{cd,cw} = \dot{m}_{cw}(h_{1'} - h_2) \quad (40)$$

Energy lost in condenser:

$$Q_{cd,los} = \dot{Q}_{cd} - \dot{Q}_{cd,cw} \quad (41)$$

SAC evaporator energy model: Energy absorbed by refrigerant in evaporator is calculated by:

$$\dot{Q}_{lc} = \dot{m}_r(h_3 - h_{3'}) \quad (42)$$

Energy rejected by chilled water in evaporator is calculated by:

$$\dot{Q}_{chw} = \dot{m}_{chw}(h_{12} - h_{12'}) \quad (43)$$

Energy lost in evaporator;

$$Q_{ev,los} = \dot{Q}_{lc} - \dot{Q}_{chw} \quad (44)$$

Coefficient of performance of the SAC is calculated by:

$$COP = \frac{\dot{Q}_{lc}}{\dot{Q}_{hg}} \quad (45)$$

Total energy lost in SAC system:

$$Q_{SAC,los} = Q_{ab,los} + Q_{hg,los} + Q_{SAC,ev,los} + Q_{cd,los} \quad (46)$$

RESULTS AND DISCUSSION

Validation of the models: Validation of the model is performed based on the existing operating data of GDC. In Fig. 1, the actual net work obtained from the records of GDC and the calculated net work from the analytical model is plotted against part load ratio. The calculated net work is shown to have close relation with the actual net work with a maximum difference of +1kW of the actual net work in the gas turbine engine.

Validation for the HRSG model is shown in Fig. 2, where the steam mass flow rate is plotted against the heat gain by steam. The trend also exhibits a close relationship between the calculated and the actual mass flow rate. The calculated mass flow rate of steam departed at point 1.5 kg sec^{-1} of steam mass flow rate and increase rapidly before closing back at point 3 kg sec^{-1} steam mass flow rate. The increased of steam mass flow rate at pont 3 kg sec^{-1} could be due to the increase of the supplied heat energy to the evaporator.

While the validation of steam absorption chiller is performed by comparing the cooling load and COP in the steam absorption chiller. The cooling load is graphed against the mass flow rate of the steam as shown in Fig. 3. The difference between the calculated and the actual coefficient of performance is ±0.01.

Energy analysis: The main operating parameters that have effect on the gas turbine engine are; air mass flow rate, inlet air temperature, fuel flow rate and the compression ratio. The energy losses experienced in the gas turbine engine components are shown in the Fig. 4. These parameters obtained from the recorded data from the plant were used as inputs to the analytical models. The results from the models were compared to the actual data to determine the energy losses. From the operating data, the net work of the turbine is 3334 kW when the mass flow rate of fuel is 0.26 kg sec^{-1} and mass flow rate of air is 15 kg sec^{-1}. Increasing the air temperature from 301 to 309 K, while maintaining the other parameters constant, i.e., $\dot{m}_a, r_c, \dot{m}_f$, the model shows that an optimum net work of 3533 kW can be produced. This is an increase of 5.64% from the actual net work. When the actual net work is 2902 kW at inlet temperature of 302.4 K and mass flow rate of fuel is 0.227 kg sec^{-1}, it is found that the net work decreases by 17.9% from the optimum point. To maintain the performance of the gas turbine at optimum, the mass flow rate of air must be maintained at 14 kg sec^{-1} and the fuel flow rate at 0.26 kg sec^{-1}.

Energy lost in air compressor: The work input to drive the air compressor in gas turbine engine is obtained from

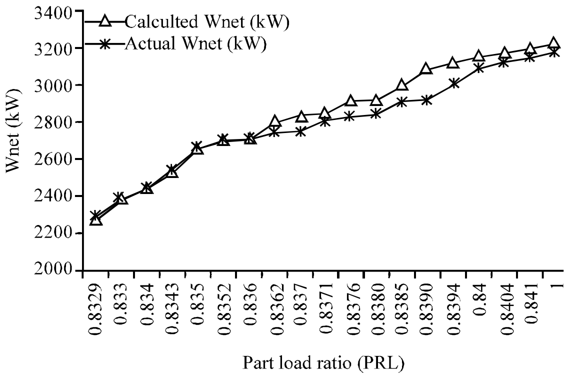

Fig. 1: Net work (kJ sec^{-1}) against part load ratio

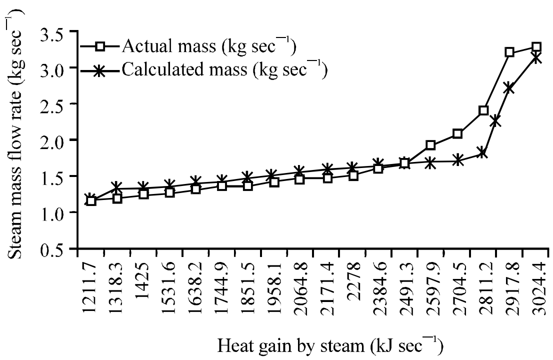

Fig. 2: Steam mass flow rate against heat gained by steam (kJ sec^{-1})

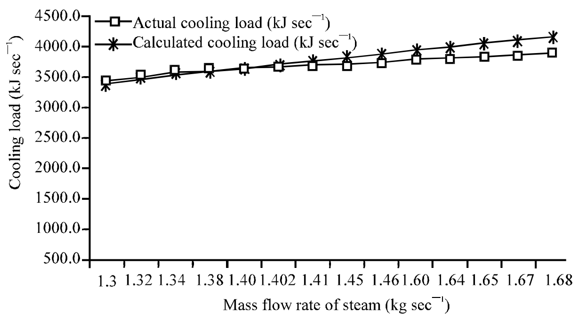

Fig. 3: Cooling load (kJ sec^{-1}) against mass flow rate of steam (kg sec^{-1})

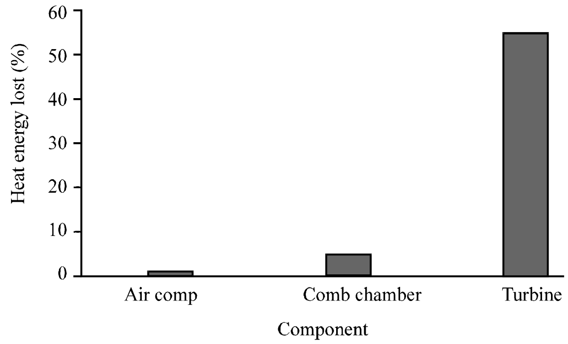

Fig. 4: Heat energy lost (%) against gas turbine engine components

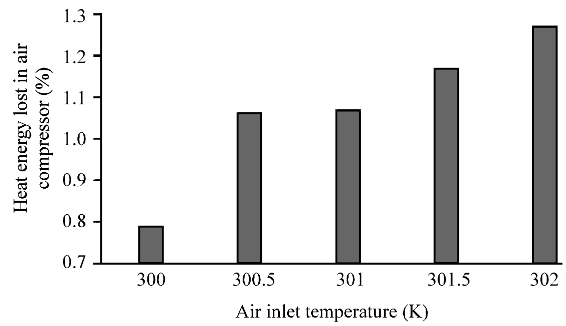

Fig. 5: Heat energy lost (%) in air compressor against air inlet temperature (K)

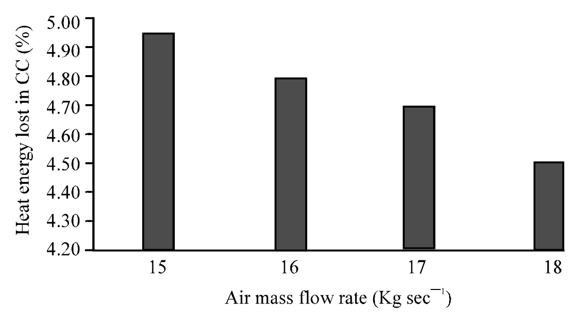

Fig. 6: Heat energy lost in combustion chamber against airmass flow rate

the turbine work output. In Fig. 5, energy lost in the air compressor is related against the ambient temperature. The energy losses in air compressor increases at high ambient temperature. This is because as the air mass flow rate increases, the content of heat energy lost in the air compressor also increases. Air compressor work can be minimize when the air inlet temperature and mass flow rate is regulated. The energy that can be saved from air compressor is 0.77 kW, at the inlet air temperature of 304.3 K and air mass flow rate of 15 kg sec^{-1}.

Energy lost in combustion chamber: The energy losses in the combustion chamber is found to be 5%. High mass flow rate of air can minimize the energy losses in the combustion chamber as this would introduce more air required for combustion. The unburn air in combustion chamber acts as a coolant. In Fig. 6, the energy losses decrease as the temperature of mass flow rate of hot gases is decreased. This is due to the high quantity of air mass flow rate, which lower the temperature of the hot gases.

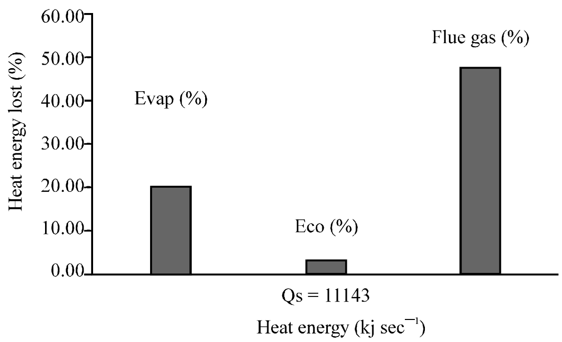

Fig. 7: Heat energy lost against heat quantity in HRSG components

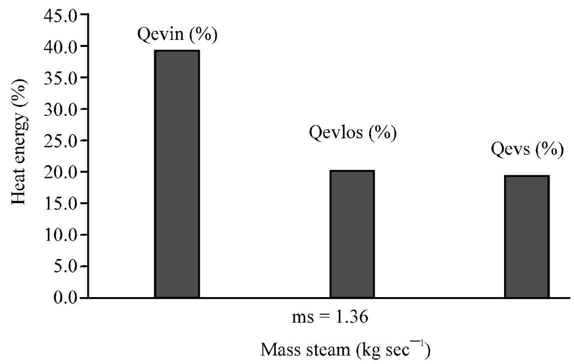

Fig. 8: Heat energy (%) against mass of the steam (kg sec^{-1}) in HRSG evaporator

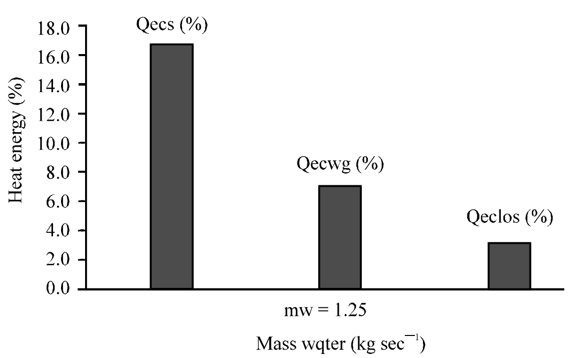

Fig. 9: Heat energy (%) and mass of warm water (kg sec^{-1}) in economizer

Results from Heat Recovery Steam Generator (HRSG) model

Analysis: The percentages of energy quantities are computed by taking the ratio of inlet/outlet energy and the total energy into the component. Energy supplied to the evaporator is 38.94% of heat energy supplied for steam generation, 10.88% is gain by the steam and 28.06% is lost in the evaporator.

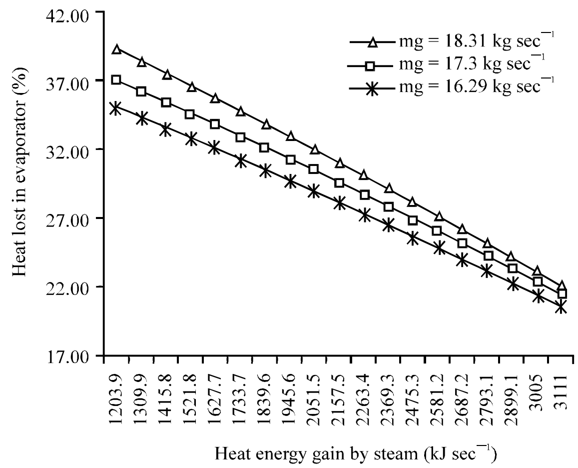

Fig. 10: Heat energy lost in evaporator (%) against heat gained by steam (kJ sec^{-1})

Energy entering the economizer is 16.8%, where the energy supplied to heat the warm water is 7.06% and 3.15% of total energy in the economizer is lost. Energy lost in economizer decreases by 0.78% when the mass flow rate of warm water is increased by 0.16 kg sec^{-1}. The percentage increased of energy gain by warm water in economizer is 0.84% at the same mass flow rate of warm water.

Figure 7 shows the heat energy lost in the evaporator and economizer. Heat supplied to the heat recovery steam generator is calculated as 11.143 MW. In Fig. 8 the percentages of heat in the HRSG evaporator is plotted against the steam produced at 1.36 kg sec^{-1}. Heat lost in the economizer is found 3.15% of the total heat supplied to the economizer. While, in Fig. 9 the percentages of heat energy calculated in the economizer is graphed against the mass flow rate of warm water 1.25 kg sec^{-1}.

Heat lost in HRSG evaporator: The heat energy in the evaporator is not gained completely by the saturated water entering the evaporator. A portion of the heat in the evaporator is lost and the lost in the evaporator increases as the parameters such as the saturated water mass flow rate through the evaporator is not regulated. In Fig. 10, the heat lost and the heat gained by the steam is related, and it can be seen that heat lost decreases as the heat gained by the steam increase. The heat gained by the steam increased only when the mass flow rate of the saturated water is increase and saturated water temperature is low.

Heat lost in HRSG economizer: In the economizer the heat leaving the evaporator is utilised to heat the warm

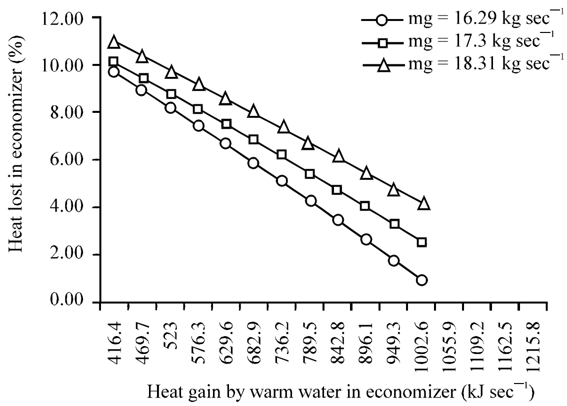

Fig. 11: Heat energy lost in economizer (%) against heat gained by warm water (kJ sec^{-1})

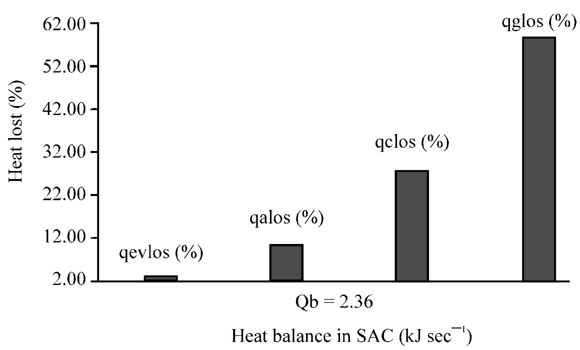

Fig. 12: Heat lost (%) against heat balance (kJ sec^{-1}) in SAC

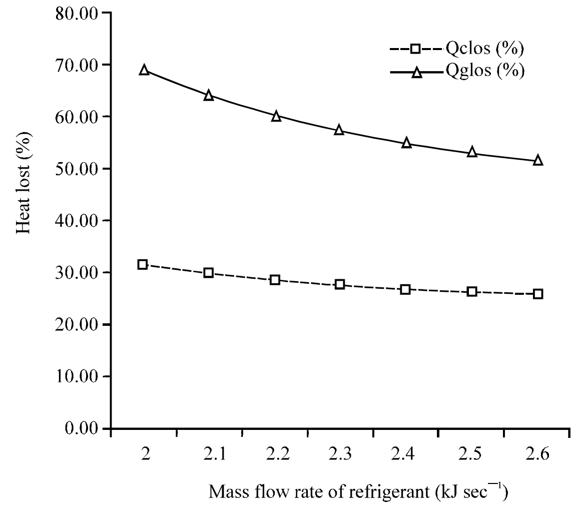

Fig. 13: Heat energy lost (kJ sec^{-1}) and mass flow of refrigerant (kg sec^{-1})

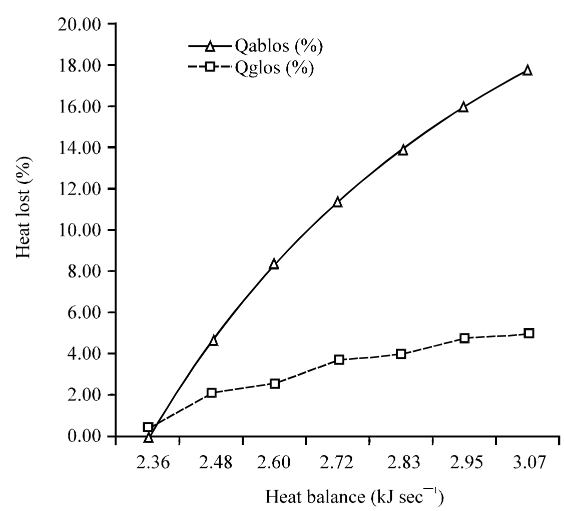

Fig. 14: Heat energy lost (%) against heat balance (kJ sec^{-1})

water that will be supplied to the economizer. Figure 11, is showing the relationship between the heat lost in the economizer and the heat gained by the warm water in the economizer. It can be seen that heat energy lost in the economizer decreases at high heat gained by the warm water. When the temperature leaving the evaporator is not corresponding with the mass flow rate of warm water then heat lost in the economizer can increase.

Results from steam absorption chiller model analysis: The performance of Steam Absorption Chiller (SAC) is characterized by its cooling load and COP. Energy exchanges in SAC involved heat rejection in absorber and condenser which is equal to the heat gain by the cooling water. The energy lost in the absorber is found to be 17.78% and in the condenser is 25.8% of the total energy lost in the SAC system. Energy lost in the evaporator is found to be 4.49% and in the high temperature generator is 51.35% of the total energy lost in the system.

Heat lost in SAC components: The percentage of heat losses in the steam absorption chiller components are shown in Fig. 12. High heat energy is lost in the high generator of SAC as the source of heat energy is generated in the high generator.

Figure 13 shows the heat lost in high generator and condenser, the main two components that experienced high heat lost.

Heat balance in absorber and evaporator of SAC can be seen in Figure 14, as it is shown the heat lost in the components increases as the heat balance between the components of the SAC is increasing.

CONCLUSION

Gas turbine engine and HRSG: The analytical model showed that energy flow in the cogeneration plant can be monitored and optimization of the plant can be achieved.

The energy lost in combustion chamber of gas turbine is approximately 5% of total heat energy lost in the engine at the optimum operation of the gas turbine engine. Increase in the heat lost in the combustion chamber decreases the part load ratio and as result decreases the net work. Air compressor work consumption from the turbine is calculated 27.9% of the turbine inlet heat energy. The work of air compressor can be reduce by 1.1% of the turbine work if the inlet air temperature is increase to 305 K instead of at 298 K. Increase in inlet air temperature of air compressor decreases the compressor work.

- The exhaust energy lost from the turbine is 54.9% of the turbine inlet heat energy, and energy consumed by the turbine is 45.08% of the turbine inlet energy.
- Energy supplied to the evaporator for steam production is 38.94% of the total energy entering the HRSG and 10.88% of the energy supplied is gained by the steam and 28.06% is energy lost in the evaporator.
- Increase in mass flow rate of steam by 0.11 kg sec^{-1} decreases the energy lost in the evaporator by 0.88% and steam gain energy 0.92% increase.
- Energy supplied to economizer is 16.81% and energy gained by warm water is 3.15% of the energy entering economizer. Energy lost in the economizer is 7.06% of the total energy entering economizer and energy lost by the flue gases is 6.59%.
- Increasing the mass flow rate of warm water by 0.16 kg sec^{-1} decreases the energy lost in economizer by 0.78% and warm water energy gain increases by 0.84%.

Steam Absorption Chiller (SAC): It was found that energy loses and imbalance can decrease the performance of the refrigeration systems. In SAC the performance of absorber is high than condenser base on heat rejection, the heat rejected by the absorber heat 61-69.6% of the total heat to be rejected from the SAC, where the heat rejected by the condenser is 10.17-11.5% of the total heat to be rejected from the SAC.

Energy lost in high temperature generator is found 51.35% and in condenser is 25.8% of the total heat energy lost in SAC. The energy lost in absorber is found to be 0.01-17.78% of the total heat energy lost in SAC. The minimum energy lost in SAC is in evaporator which is 0.44-4.9% of the total heat energy lost in the SAC.

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