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Simulation of CHP Energy Conversion System

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ABSTRACT

Simulation of energy conversion systems is to increase resource utilisation efficiency (RUE) which in tern is dependent upon the maintained capability of individual components to use the fuel (natural gas in present case) efficiently. Therefore, an assemblage of mathematical equations of each and every component of a combined heat and power (CHP) is simulated in real time situation implemented in MATLAB computer programming tool software are to evaluate optimum cycle pressure ratio. RUE of steam turbine directly affected by steam turbine RUE. For the present analysis steam turbine RUE is varied from 80% to 95% and it is found that combined cycle RUE increases from 38.14% to 39.83%. For the present analysis combustion chamber RUE is varied from 80% to 95%. As the combustion chamber RUE will be increasing more enthalpy will be going to gas turbine and more heat will be transferred to HRSG. Due to which combined cycle RUE will be increasing. With increase in combustion chamber RUE by 15% there is an increase of combined cycle RUE by 6.6%.

Keywords: CHP, System modeling, Thermodynamics, Flow chart

INTRODUCTION

Current commercially available CHP system has RUE typically in the 55-58% LHV range if power is generated alone [1]. In case of bother power and heat generation RUE is increased and found to more than 90% [2]. If power is generated alone then flue gases coming out at high temperatures are directed to the heat recovery steam generator. As the steam generated in HRSG is at high temperature and pressure, therefore, the flue gases coming out of HRSG are also at high temperature. The flue rejection at high temperature to the environment is thermodynamic loss to the plant. In other case utility steam is generated at lower temperature. In lower temperature steam generation thermodynamic losses are decreased and flue gases entering the environment are also at lower temperature. The maximum amount of useful work which may be extracted from a system is known as availability.

CHP systems are generally very large and complex systems with a lot of different configurations. Therefore, it is imperative to develop power plant system structure and analyze it with some suitable mathematical tool. In a CHP system CNG, air and purified water are the input and electricity and flue gases are the output. Although in nature water and steam properties are not changed but the accuracy of the mathematical modeling is to be improved from time to time. In 1997, the International Association for the Properties of Water and Steam (IAPWS) proposed a new set of mathematical modeling for evaluating thermodynamic properties of water and steam for industrial and research purpose. This new formulation, called IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam (IAPWS-IF97), replaces the previous industrial formulation, IFC-67, that had formed the basis for power-plant calculations and other applications in energy engineering since the late 1960's. Information flow diagram for the programming of water properties is shown in Fig. 1. As the accuracy and speed of computation is increasing so is the thermodynamic

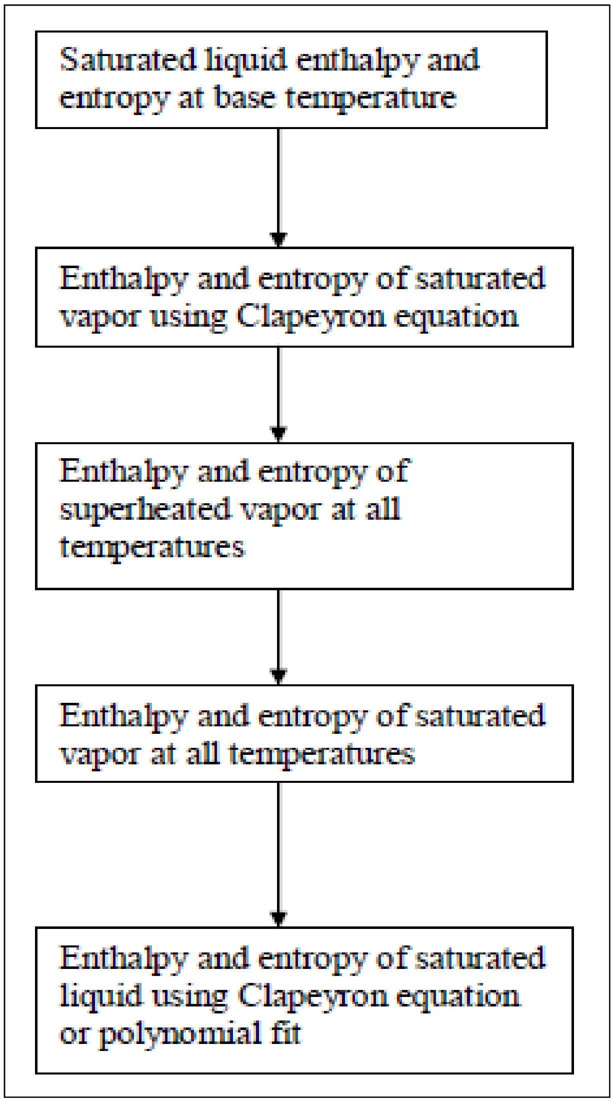

Fig. 1. Information flow diagram for the property calculation based upon IAPWS-97

systems are also being improved. With the help of computer programming tools it possible to analyze large and complex thermodynamic systems as represented in Figure 2. It is the outline structure of triple pressure steam generation system used in more than 250 power plants installed worldwide between 1975-2015 [3-7]. Real time technology is the balance of expenditure and return. It is expected that payback period for a CHP system is expected to be nearly three years. Afterwards maintenance costs are increased and share of revenue generated from tariff for maintenance is also increased. In a real life operating power plant steam recovered at single pressure, double pressure and triple pressure is with increasing cost and increasing efficiency respectively. Therefore, before designing it is suggested to take care of cost and efficiency. Simulation procedure for such systems is as represented in Figures 1 and 3. Mathematical modelling is described hence forth.

MATHEMATICAL MODELING AND COMPUTER SIMULATION STRATEGY

The flue gas side pinch point temperature (T_P) and economizer exit steam temperature (T_{EO}) are calculated by assuming the drum saturation pressure (P_{Drum}) with the expression as following:

$$T_P = T_{DRUM} + PP \qquad (1)$$

$$T_{EO} = T_{DRUM} - AP \qquad (2)$$

The steam generated for each kg/sec of exhaust gases can be determined by applying mass and energy conservation principles across the super-heater and evaporator.

$$M_W = \frac{M_{GEX} \times C_{PG}(T_{GEX} - T_P)}{(h_{ST} - h_{EO})} \qquad (3)$$

The steam generated with the use of waste heat is predominantly by convection. It is customary to neglect the radiative heat transfer, particularly because the reduction in heat transfer due to soot deposition/ fouling etc. is also ignored and it is assumed that these two approximately compensate each other. The heat across each section of boiler can be estimated as follow:

$$Q_{ECON} = M_W(h_{EO} - h_{FW}) \qquad (4)$$

$$Q_{EVAP} = M_W[h_{FG} + C_{PW}(T_{DRUM} + T_0)] \qquad (5)$$

$$Q_{SUPR} = M_W(h_{ST} - h_G) \qquad (6)$$

The flue gas temperature in the stack can is estimated on the basis of the heat balance across economizer.

$$T_{STACK} = T_P - \frac{M_W(h_{LPEO} - h_{FW})}{M_{GEX} \times C_{PG}} \qquad (7)$$

A low stack temperature is always desirable from the point of waste recovery which otherwise will be lost to environment [3]. However to avoid the corrosion from moisture formation in economizer, the minimum temperature should always be kept higher than the acid dew point temperature. Also, the size of economizer depends on the stack temperature which has therefore to be justified on the economic consideration. The mass and energy balance for the various processes of steam turbine cycle, heat transfer and RUE can be calculated by the following equations:

$$W_p = v_f (P_{DRUM} - P_{COND}) \tag{8}$$

$$h_{FW} = h_f + \frac{h_{FW} - h_f}{\eta_p} \tag{9}$$

Work done by the steam turbine is

$$(W.D)_{ST} = h_{MST} - h_{EX} \tag{10}$$

Net wok done will be

$$(W.D)_{net} = (W.D)_{ST} - W_p \tag{11}$$

$$Q_{SC} = h_{MST} - h_{FW} \tag{12}$$

The associated bottoming cycle and plant efficiencies are:

$$\eta_{SC} = \frac{(W.D)_{net}}{Q_{SC}}$$

$$\eta_{CC} = \frac{W_{GT} + W_{SC}}{Q_{in}} = \frac{W_{CC}}{M_f LHV} \tag{13}$$

A computer simulation model in MATLAB has been developed to simulate the performance of CHP system. Results obtained from the program are discussed in the following section.

RESULTS AND DISCUSSION

Gas turbines used in the CHP system are having pressure ratios below 15 to optimize efficiency and cost. For the present analysis cycle compression ratio is varied from 8 to 20. In actual practice a gas turbine gives maximum RUE at different pressure ratio than that for maximum work output. Therefore compression ratio is kept in between these two ratios. As the compression ratio is increased the maximum temperature at the outlet of compressor is increased. The fuel requirement is decreased because turbine inlet temperature is fixed. Gas turbine inlet temperature is fixed by the thermal stress bearing limit of the turbine

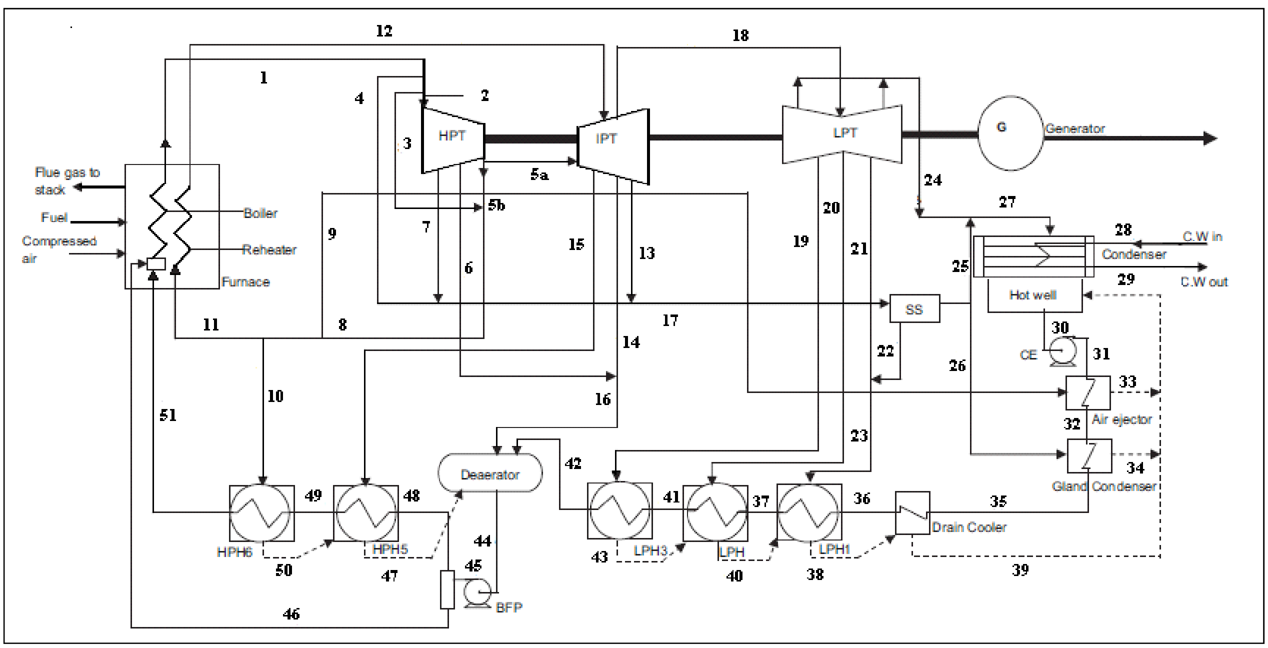

Fig. 2. Thermal Power Plant Process flow diagram (High Pressure Turbine (HPT); Intermediate Pressure Turbine (IPT); Low Pressure Turbine (LPT); Steam Seal Regulator (SSR); Cooling Water (CW); Low Pressure Heater (LPH); High Pressure Heater (HPH); Condensate Extraction Pump (CEP); Boiler Feed Water Pump (BFP); Stream Numbers (1,...,51)).

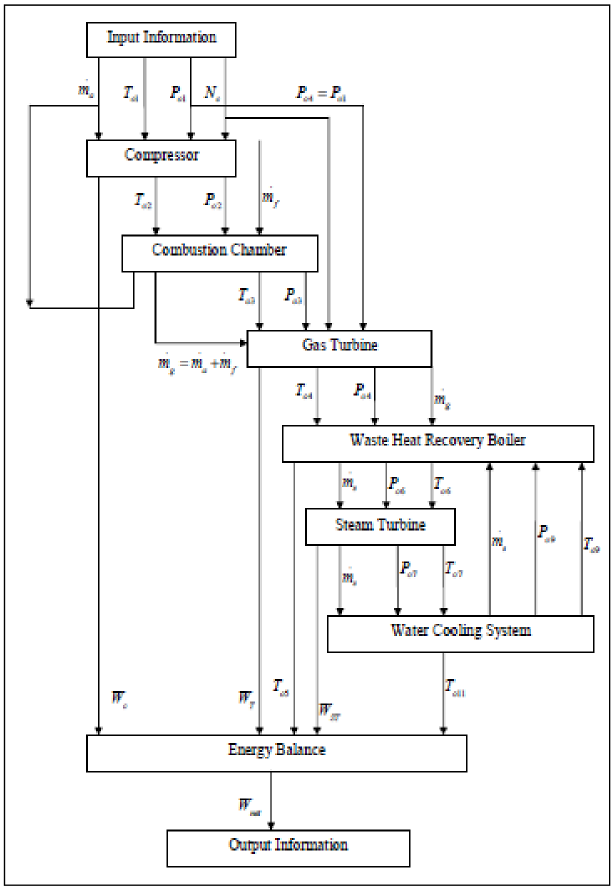

Fig. 3. Information flow diagram of the present CHP

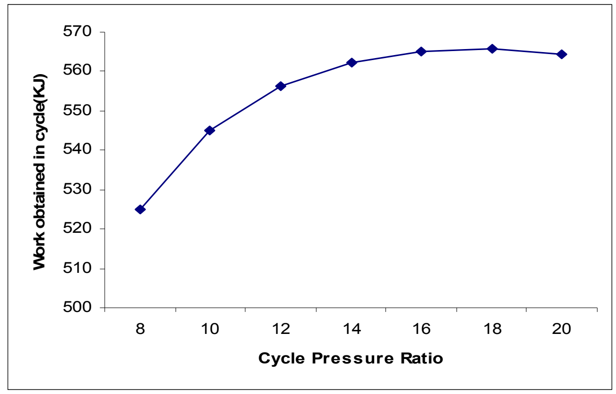

Fig. 4. Change in work obtained from combined cycle with change in gas turbine compression ratio

blade material. If the compression ratio of the gas turbine has to be increased then size of the blade will be larger and inertia force will increase. To bear this larger inertia force, a strong blade base is required. In a CHP systems waste heat coming out from gas turbine system is utilized in heat recovery steam generator to generate process steam.

Cycle compression ratio directly effect on gas turbine outlet temperature. As compression ratio increases, gas turbine outlet temperature decreases due to higher expansion in turbine section which make lesser heat available for pressurized water in HRSG. Due to this steam turbine work output is decreased and after the pressure ratio of 18 a decreased work output from the cycle is obtained (Fig. 4). For the lower cycle pressure ratio sufficient heat is available to convert the pressurized water into steam.

As it may be observed that there is not much gain in work output by changing the compression ratio but RUE gain is reasonable. To calculate the cycle RUE a ratio of work output and energy supplied is taken. As the increase in pressure ratio lower the fuel consumption so the energy supplied to the cycle decreases and RUE increases.

In a CC plant, the power output varies with the ambient temperature. During summer months, a CC plant could lose as much as 10 to 15 percent of its rated ISO output. Ambient air temperature never remains constant. Now from the analysis it is being found that as the IAT will increase fuel requirement will decrease. This is due to the fact that TIT is fixed for this case and if the IAT increases then combustion chamber inlet temperature will also increase. But combustion chamber outlet temperature is fixed. So the fuel requirement decreases.

For the design conditions if the TIT is fixed then, as the gas turbine inlet temperature will keep on increasing then the fuel requirement will decrease. But due to the increase in the ambient temperature the mass flow rate of the air to the compressor also decrease which leads to the lesser work output and lesser RUE. Inlet air cooling may bring the ambient air to the designed condition. Effect of increase in compression ratio is positive on the combined cycle RUE but that of IAT is negative. With the increase in IAT combined cycle RUE comes down. As the inlet air temperature increases, fuel consumption is not decreased much but decrease in work output is high. Due to this reason RUE of combined cycle decreases with increase in ambient air temperature.

In combined cycle work is obtained from the gas turbine and as well as from the steam turbine. With increase in turbine inlet temperature (TIT) work output of gas turbine is not changed but gases come out from gas turbine at higher temperature. So more heat is available in HRSG for the water to be converted into steam. Due to this reason work output of steam cycle increases and hence the net output of combined cycle.

RUE of steam turbine directly affected by steam turbine RUE. As the steam turbine RUE increases, RUE of combined cycle also increases. For the present analysis steam turbine RUE is varied from 80% to 95% and it is found that combined cycle RUE increases from 38.14% to 39.83%. Effect of steam turbine parameter is less pronounced than that of gas turbine parameters.

Effect of combustion chamber RUE is linear on the combined cycle RUE. Combustion chamber is having many types of losses. For the present analysis combustion chamber RUE is varied from 80% to 95%. As the combustion chamber RUE will be increasing more enthalpy will be going to gas turbine and more heat will be transferred to HRSG. Due to which combined cycle RUE will be increasing. With increase in combustion chamber RUE by 15% there is an increase of combined cycle RUE by 6.6%.

In the present analysis lower heating value of the fuel is taken into consideration. It is being found that with increase in calorific value of fuel cycle RUE is increased. Fuel supplied to the plant is natural gas and its composition varies from place to place and hence its calorific value. In the present analysis it is being found that with change in calorific value from 40000 KJ/kg to 45000 KJ/kg combined cycle RUE is varied by only 0.10%. So the calorific value is having very less effect on the combined cycle performance.

Back pressure is the pressure at which steam comes out of the steam turbine. For the higher RUE it is kept as low as possible. In the present work value of back pressure is varied from .080 atmosphere to .105 atmosphere and it is being found that combined cycle RUE is decreased by 0.18%. Pinch points and approach temperatures are important HRSG design parameters. The pinch point strongly influences the amounts of heat transfer surface in the evaporating section. Current HRSG designs use pinch points in the 8 to 14°C range. In general, these boilers have 50% more surface in the evaporating section than boilers with pinch points of 22°C to 28°C. The approach temperature also influences the amount of surface required for an economizer section, with exponentially increasing amounts required for very low approach temperatures. Current HRSG economizers have approach temperatures in the 8°C to 14°C range at the design point. Many other operating conditions can occur at off-design points, including start-up. Some conditions will result in steaming at the exit of the economizer, such that it acts as an evaporative surface. As the drum pressure is increased, combined cycle RUE comes down. With the change in drum pressure from 12 bar to 17 bar cycle RUE is decreased by 0.85%. Pinch point (PP) temperature is generally kept 8°C and an approach point of 2°C is taken. So we can take a pinch point of 10°C. In figure.18 it is shown that with increase in pinch point from 5°C to 25°C, combined cycle RUE is decreased by 0.18%. As the PP is increased, decrease in RUE is lesser but after that it is increased.

CONCLUSION

In the present work mathematical modelling of a CHP system is carried out so as to study the effect of different parameters. With increase in gas turbine pressure ratio, work obtain from this cycle is increased. As the compression ratio is increased, it results into increase in expansion ratio also. Result of which is the decrease in gas turbine outlet temperature. As the turbine outlet temperature is decreased so is the efficiency of steam cycle is decreased. Mathematical modelling developed in the present work is helpful in analysing both the cycles.

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