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Analysis and Modeling of a Combined Cycle Power Plant using Graph Theory

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***Abstract*: Graph theory is a well-established tool for the analysis of any system. For the graph theory system has to be divided in to sub systems and factors have to be identified for the analysis. In the present work some of the attributes are identified for the application of graph theory. A proper identification of the attributes helps in precise evaluation of selection of combined cycle power plant systems. As the number of attributes increases, the performance characteristics of such systems are representing precisely and the selection of gas turbine system becomes more reliable. However, with increase in number of attributes, usage of dedicated computer programs becomes more indispensable for efficient evaluation and selection of combined cycle power plant systems. Generally, the attributes of combined cycle power plant systems can be classified either by different subsystem of the combined cycle power plant system or by the different performance characteristics of the combined cycle power plant system as well as matching of its subsystems. For the identification of attributes two methods are adopted. Some of the attributes are identified from the literature and for some attributes a computer program was made. Results obtained from the program also give us new attributes.**

***Keywords*: (amb) Ambient, (avg.) (aph) Air Preheater, (IAt) Inlet air temp., (COP) Coefficient of Performance, (CCPP) Combine cycle power plant, Diameter (m), (E) Energy (KJ), (Ed) Energy Destruction, (ΔEc) Energy Change in Combine System.**

1. Introduction

The installed power generation capacity of India stood at 147,000 MW while the per capita power consumption stood at 612 kWH. The country's annual power production increased from about 190 billion kWH in 1986 to more than 680 billion kWH in 2006. The Indian government has set an ambitious target to add approximately 78,000 MW of installed generation capacity by 2012. The total demand for electricity in India is expected to cross 950,000 MW by 2030. Current installed capacity of Thermal Power (as of 12/2008) is 93,392.64 MW which is 63.3% of total installed capacity. Current installed base of Coal Based Thermal Power is 77,458.88 MW which comes to 53.3% of total installed base. Current installed base of Gas Based Thermal Power is 14,734.01 MW which is 10.5% of total installed base. Current installed base of Oil Based Thermal Power is 1,199.75 MW which is 0.9% of total installed base. The electricity generation growth rate in India was 6.4% in the 2007-08 and 2.8% in the year 2008-09 with respect to base year 1993-94 (The Hindu, 13.05.08). The state of Maharashtra is the largest producer of thermal power in the country.

2. Mathematical Modeling and analysis of the WHRB

In recent years a great deal of attention is focused on the efficient utilization of energy resources with minimum heat loss. The continuous need to save the fast depleting energy resources, and to use them in an optimum efficient manner has renewed the interest in devices that can use heat from waste flue gases from various source. The flue gases on the virtue of being at a higher temperature relative to the surroundings and having a higher mass flow rate, possess considerable amount of available energy, which if not utilized properly will lead to huge undesirable energy loss. During the last two decades there has been more attention, on to use heat from flue gases for various applications and to optimize the units which are used to absorb heat from waste gases. Waste heat recovery boiler is the component of the bottoming steam cycle, which absorbs energy of exhaust gas of the gas turbine and produces steam best for the process or for further electricity generation by a steam turbine.

Table 1
WHRB energy distribution

Surface	% of Total Area of Pressure Parts	% of Total Heat transfer to Pressure Parts
Economizer	19.6	7.28
Boiler (Including water walls)	28.8	66.97
Super heater	51.6	25.75
Total	100.0	100.0

The gas side pinch point temperature (T_P) and economizer exit temperature (T_{EO}) are calculated by assuming the drum saturation pressure (P_{Drum}).

$$T_p = T_{DRUM} + PP \quad (1)$$

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

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.

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

The WHRB, being considered is a non-firing boiler. Therefore the heat transfer is mainly 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:

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

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

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

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

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

A low stack temperature is always suitable from the point of waste recovery. However to eliminate the corrosion from moisture formation in economizer, the minimum temperature should always be kept higher than the acid dew point temperature. As well as, the size of economizer depends on the stack temperature which has therefore to be justified on the economic consideration. Mathematical modeling of single steam extraction and double steam extraction may be had from any standard power plant text book

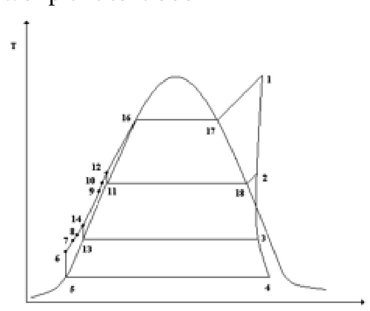

Fig. 1. T-s diagram of non-ideal superheat Rankine cycle with two closed-type feed water heaters with drains pumped forward

On energy balance at high and low-pressure heaters we have

$m_2 (h_2-h_{11}) = (1- m_2)(h_9-h_8)$ (8)
$m_3 (h_3-h_{13}) = (1- m_2-m_3)(h_7-h_6)$ (9)

The value of h_9 and h_7 are obtained from temperature T_9 and T_7. These temperatures are equals to the saturation temperature of the steam in each heater minus its terminal temperature difference (TTD).

$T_9 = T_{11} - TTD$ HP Heater (10)
$T_7 = T_{13} - TTD$ LP Heater (11)

Further we have

$$h_{12} = h_{11} + v_{11} \frac{P_{12} - P_{11}}{\eta_p} \quad (12)$$

and for h_{14} we have

$$h_{14} = h_{13} + v_{13} \frac{P_{14} - P_{13}}{\eta_p} \quad (13)$$

$h_{10} = m_2 h_{12} + (1- m_2)h_9$ (14)
$(1- m_2)h_8 = m_3 h_{14} + (1- m_2-m_3)h_7$ (15)

Turbine work
$W_T = (h_1 - h_2) + (1- m_2)(h_2 - h_3) + (1- m_2-m_3)(h_3 - h_4)$ (16)

Pump work
$\sum W_p = m_3 (h_{14} - h_{13}) + m_2 (h_{12} - h_{11}) + (1- m_2-m_3)(h_6 - h_5)$ (17)

Heat added $Q_A = h_1 - h_{10}$ (18)

Thermal efficiency $$\eta_{th} = \frac{W_T - \sum W_P}{Q_A} \quad (19)$$

3. Result analysis

In a gas turbine system, the inlet air is initially compressed in the compressor to increase its pressure and temperature. The performance of the compressor significantly affects the overall performance of the gas turbine system. Since, the energy transfer during the flow of air compressor takes place, the thermodynamic as well as aerodynamic and mechanical design and factor effecting these parameters play a significant role in the performance of the compressor.

The air with high pressure and temp. as received from the exit manifold of the compressor is fed to the combustions chamber where in the fuel in atomized form is fed. due to high temp. of air, the fuel droplets get ignited and generate thermal energy due to its burring, subsequently, increasing the temp. of the hot fluid medium in the combustor. higher is the peak temp. achieved in the combustor. It is desired that NOx emission level in the plant should be as low as possible. From the figure below it may be seen that cycle pressure ratio effects the emission of the nitrogen in the cycle. As the cycle pressure ratio increases concentration of nitrogen get changed.

The Heat Content of the gases obtained at the exit of the combustor is converted into work output through energy transformation in the gas turbine.

Steam power plants lose 0.5 to 1.5 %of the flow rate of their water steam circuit because of leakage from fittings & bearings, esc. with non-condensable gases in de-aeration process boiler blow down & the other causes. raw water may include suspended solids & turbidity, org, hardness, calcium & magnesium, alkalinity (bicarbonates, carbonates, hydrates) other dissolved ions (sodium, sulfate, chloride etc.), silica dissolved gases CO_2, CO_2 as impurities. As the requirement of raw water for a CCPP is very high, it may be accomplished from a surface source, such as a river or lake. Raw water is firstly chlorinated to prevent fouling of the equipment

4. Future scope

Combine cycle power plant is a developing system so that the new attributes are identified with new researches. A Survey can be done to identify new factors taking role in the performance of Combine cycle power plant.

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