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Performance of Regenerative Air Preheater of Pulverized Coal Fired Boilers

M. Ravi kumar[1], D. Kulandaivel[2], K. Ramesh[3].
[1]*PG Student, Thermal Engineering, GCT, Coimbatore*
[2]*Assistant Professor, Department of Mechanical Engineering, GCT, Coimbatore*
[3]*Professor, Department of Mechanical Engineering, GCT, Coimbatore*

Abstract: The air preheater (APH) is the final heat trap in the boiler's flue gas path. Effective pre-combustion coal drying and efficient combustion in the boiler are required for APH to operate efficiently. The effectiveness of heat transfer in APH is significantly influenced by the characteristics of relative humidity and ambient air temperature (AAT). Relative humidity (RH) and air density are also impacted by variations in AAT, which in turn alters the heat capacity of the air. If there is insufficient reserve heat exchange capacity in APH, an increase in AAT will diminish the possibility of waste heat recovery. Additionally, a change in AAT will affect the power consumption of the fans and the leakage through various seals.
Keywords: Air Preheater, Ambient air temperature, mass flow rate, Air preheater Efficiency

I. INTRODUCTION

In pulverised coal fired power generation, moisture in coal is a major problem. Due to its hygroscopic nature, coal acquires a lot of surface moisture with seasonal changes. The performance of the pulverizer is impacted by coal moisture. Effective coal pulverisation and pneumatic transport depend on efficient coal drying. Traditional large pulverised coal-fired boilers use a coal drying mechanism that incorporates waste heat recovery from hot flue gas before discharge through stack by producing hot air stream (regenerative or recuperative APH) for coal drying purposes. Additionally, the need for cost- and energy-effective coal drying processes is critical because, in some cases, the combined weight of coal ash and moisture exceeds 50% of the coal "as received." Therefore, an appropriate and effective drying system on flue gas waste heat recovery will increase profitability while lowering the cost and emissions of power generation. The capacity for drying, however, is constrained by the hot primary air (PA) temperature and available hot airflow. The air preheater (APH) inlet flue gas temperature determines the PA temperature, which in turn limits the mill drying capacity. An increase in this temperature will reduce the boiler's overall efficiency, and too much hot PA may start a fire in the mill when the amount of coal fed is decreased to match a decrease in the demand for steam. Once more, any indirect heat exchange mechanism for coal drying within boiler combustion PA intake - preheating - combustion and exhaust circuit has an inherent limit on coal moisture removal capacity in the pulverising process that compromises overall boiler efficiency as heat of evaporating moisture is consumed from fuel energy.

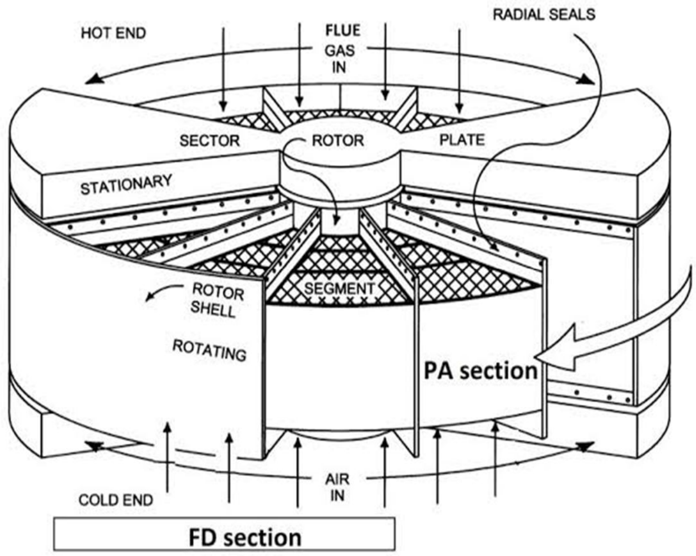
The heat retained in APH is consumed by the moisture from evaporated coal, and extra accessible heat from secondary air aids in achieving the furnace's combustion conditions. AAT also influences how much heat is available for steaming, which has an impact on how well APH works. The performance of the APH is impacted by the relative humidity (RH) and AAT, which determine heat loss resulting from moisture in the air.

II. AIR PREHEATER

The last heat trap in the boiler's heat transfer route, known as the APH, raises the ambient air temperature (AAT) by transferring heat from flue gas that would otherwise be squandered by being released through the stack. It functions as a heat exchanger, heating the air exiting the APH. APH typically provides 10% to 12% of the boiler's thermal efficiency and 10% to 15% of the total fuel heat trapping. In a boiler, air is warmed since it is utilised to dry coal and expedite the achievement of the pulverised coal's ignition temperature for effective combustion. While the heat content of hot primary air is crucial for determining the pulverised coal milling system's drying capacity, the heat content of hot secondary air dictates how quickly ignition temperature is reached and how stable the flame front velocity is at the burner mouth. Insufficient hot PA heat content will restrict raw coal fed to the pulverizer's ability to dry, which will lower the pulverised coal throughput capacity. Similar to this, inadequate hot secondary air temperature and heat content may cause the flame front to hunt, occasionally losing ignition and increasing carbon in ash loss from unburned coal particles. The amount of heat rejected to the chimney may be greatly decreased by using APH, which can successfully recover heat from flue gas at lower temperature levels than the economiser, enhancing the boiler's efficiency.

The boiler efficiency rises by around 1% for every 22°C decrease in the temperature of the flue gas discharge. By using hot air, the APH increases stability, intensifies, and enhances combustion. Additionally, it improves the boiler's efficiency by recovering waste heat, burning inferior fuel effectively, and speeding up heat transmission in the boiler, which minimises the need for heat transfer space. Complete combustion is achieved as a consequence, there are less unburned fuel particles in the flue gas, and enhanced combustion allows for quicker load variation. Hot air may burn coals of lower quality effectively. As a result, APH is essential in lowering fuel and auxiliary power use.

Figure 1 Arrangement of standard tri-sector regenerative rotary APH



The two primary categories of APHs are recuperative and regenerative. The early power boilers had recuperative shell and tube APHs with air flowing horizontally through the tubes' baffle-plated outer surfaces and flue gas flowing vertically through them. Ash fouling, especially at the hot end of the flue gas passage via tubes, has a significant negative impact on the efficiency of this type of tubular recuperative APH.

In a radially split cylindrical shell known as the rotor, hundreds of high efficiency heat-exchanging metal components are compactly and densely distributed within sector-shaped compartments. Heat transfer is facilitated by the corrugated design of the heating components and the tight packing of the baskets. Sector plates and a sealing system in the APH separate the gas side from the air side. The housing surrounding the rotor has connections for ducts on both ends and is sufficiently sealed by radial and circumferential sealing members that create a secondary air passage through one (bi-sector) or primary and secondary air passage through two (trisector) sectors of the APH and a gas passage through a different sector. Between these two sides, the rotor revolves. Heat is collected by the elements as they go through the hot flue gas stream and released as they move through the air flowing passage(s) as the rotor gently spins the elements through the air and gas passageways, raising the temperature of the air utilised in combustion. Flue gas and air flow in typically opposing directions.

Despite having a surface area per unit volume and volume per unit load (m^3/kW) that are superior to tubular APHs (350 m^2/m^3 and m^3/kW, respectively), rotary APHs nevertheless have substantial air leaks from the air side (pressure: +6.5 to 7.5 kPa) to the flue gas side (-0.3 to -0.8 kPa). This is due to tip sealing. Due to inefficient or delayed combustion in the furnace, air leakage to the flue gas side of the APH may result in a fire danger. Despite the fact that a lot of fan power is expended on it, the tramp air leakage cannot be used for combustion. In a regenerative rotary APH, a minimum leakage of 5%–7% is obviously inevitable. In this study, the tri-sector regeneration type APH is the only one whose performance is evaluated in relation to the AAT variation.

For high moisture coal drying, the regenerative APH's usual rotational direction matches the direction of heat transfer from flue gas to PA preheating, followed by secondary air preheating. For high volatile, low moisture coal, the rotational direction must be reversed to avoid a fire danger in the APH caused by unburned carbon in the ash and an unwelcomely high, hot PA temperature at the mill intake.

III. AIR PREHEATER PERFORMANCE

The performance of auxiliary equipment is becoming more crucial as the efficiency of pulverised coal burned thermal power production is progressively stressed. Due to inadequate waste heat recovery and beneficial combustion air leakage into exhaust flue gas, the regenerative APH is a cause of lost thermal efficiency. Furthermore, it is challenging to gauge the temperature of the APH exit flue gas and the effectiveness of the air heater due to tramp air intrusion, APH bottom ash hopper leaks, and intake air duct leaks.

Various routes between the rotor and stator may be the source of the leakage in regenerative APH. Due to damage to the circumferential seal, air and flue gas may bypass the rotor through the APH. Due to no appreciable change in total air or flue gas flow, they may reduce APH heat transfer efficiency but have no impact on induced draught (ID), forced draught (FD), or PA fan power consumption. Larger diameter regenerative rotary APHs with greater hot end and cold end temperature differences (on average, 185–200°C) and with provisions for thermal expansion have a higher prevalence of circumferential seal leakages. The radial seal leakage may happen at either the hot end or the cold end of the regenerative APH, causing, respectively, hot air entrance and cold air intrusion in the flue gas route. The power consumption of the ID and FD fans is increased by the hot and cold air leakage through the radial seals, but the boiler's thermal efficiency is not improved. Insufficient total combustion air flow from excessive radial seal leaking may result in lack of ignition. When expressing APH air ingress as a percentage of O_2 in APH exit flue gas flow, radial seal leakages are expressed as a fluctuation in O_2 concentration in flue gas before and after the regeneration APH. However, as was already said, neither the American Society of Mechanical Engineers' (1991) standard nor any performance guarantee test describe the flue gas or air traversing the APH rotor owing to circumferential seal leakage properly, nor have they characterised it adequately.

Always monitor APH air ingress in the flue gas route based on changes in boiler exit flue gas flow rather than changes in overall air flow through the APH. For the purpose of adjusting the flue gas exit temperature, which must be ensured during the performance guarantee test and is necessary for assessing the overall boiler efficiency, it is crucial to evaluate changes in the total flue gas flow via the APH. Although it shows a guaranteed pressure drop over the APH that would have otherwise been significantly greater than the test result, it ignores the heat transfer loss associated with bypassing the APH and the impact of air and flue gas bypass. More air and flue gas flow will be observed with less of system resistance and the pressure drop across the APH will be lessened with more APH bypass.

The weight of air moving from the air side to the gas side of the air heater is known as the APH leakage percentage. This index serves as a gauge for the health of the rotor post and radial seals on the air heater. Air heater leakage rises when air heater seals deteriorate. The increase in air heater leakage raises the PA, FD, and ID fans' station service power needs, raising unit net heat rate and potentially restricting unit capacity. An indicator of the internal health of the air heater is the gas side efficiency index. The air heater's gas side efficiency declines when internal problems like ash pluggage and basket wear get worse. Typically, there is also an increase in exit gas temperature and a decrease in air heater air outlet temperature, resulting in an increase in unit heat rate.

The amount of heat energy (q_g) received by APH from flue gas may be specified as:

$$q_g = m_g . C_{pg} . (T_{ge} - T_{gl}) \quad (1)$$

The total amount of heat transferred by APH to the primary and secondary air (q_a) may be specified by:

$$q_a = m_a . C_{pa} . (T_{al} - T_{ae}) \quad (2)$$

where C_{pa} = the mean specific heat of air between T_{ae} and T_{gl}; T_{ae} = temperature of air entering APH; T_{al} = temperature of air leaving APH; T_{ge} = temperature of gas entering air heater; T_{gl} = measured temperature of gas leaving air heater; C_{pg} = the mean specific heat of flue gas between temperature T_{gl} and T_{ge}.

The method to determine air heater leakage as per this procedure is the volumetric method. This is an empirical approximation of air heater leakage with an error in accuracy within ±1% and is given by:

$$A_L = \frac{(CO_{2\,ge} - CO_{2\,gl}) \times 0.9 \times 100}{CO_{2\,gl}} \quad (3)$$

where A_L is the percent of APH air leakage with respect to total air intake to APH; CO_{2ge} is percent CO_2 in flue gas entering APH. CO_{2gl} is percent CO_2 in flue gas leaving APH. The CO_2 values may be determined stoichometrically from the measured O_2 entering and leaving the APH. Alternatively, the APH air leakage may be determined also from the following equation:

$$A_L = \frac{(O_{2gl} - O_{2ge}) \times 100}{21 - O_{2gl}} \quad (4)$$

where O_{2ge} = percent O_2 in flue gas entering air heater, O_{2gl} = percent O_2 in gas leaving air heater.

The gas side efficiency is defined as the ratio of the temperature drop, corrected for leakage, to the temperature head, expressed as a percentage. Temperature drop is obtained by subtracting the corrected gas outlet temperature from the gas inlet temperature. Temperature head is obtained by subtracting air inlet temperature from the gas inlet temperature. The corrected average gas outlet temperature is defined as the outlet gas temperature calculated for 'no APH leakage' and is given by the following equation:

$$T_{gnl} = \{ A_L \cdot C_{pa} (T_{gl} - T_{ae}) / C_{pg} \} + T_{gl} \quad (5)$$

where T_{gnl} = flue gas outlet temperature corrected for no air leakage; the APH gas side efficiency (η_{APHg}) is represented by:

$$\eta_{APHg} = (T_{ge} - T_{gnl}) / (T_{ge} - T_{ae}) \quad (6)$$

Similarly, the air side efficiency η_{APHa} can be represented as:

$$\eta_{APHa} = (T_{al} - T_{ae}) / (T_{ge} - T_{ae}) \quad (7)$$

IV. IMPACT OF AMBIENT AIR TEMPERATURE

A. Variation in ATT

The majority of tropical nations experience excessive AAT excursions both in the winter and the summer because to the altering monsoon patterns brought on by climate change. Any APH is created using a set of criteria. Performance is impacted if one of these factors deviates. The performance of APH is impacted by a number of elements, including mass flow rate, coefficient of heat capacity, density, humidity ratio, etc. We'll go through each of these components' consequences in more depth.

B. Effect Of ATT In Mass Flow Rate Variation

The volumetric flow rate of air (V_a) or flue gas (V_g) remain almost constant as the space available for fluid flow through the APH device is constant. In certain occasion mass flow rate is affected by APH bottom ash hopper seal leakage or duct leakage in air flow passage causing variation in pressure drop across the APH flue gas and air path. In general gaseous fluid flow terms, air mass flow through APH may be represented by:

$$m_a = P_a V_a / R T_{ae} \quad (8)$$

where P is the atmospheric pressure, V_a is the volumetric flow rate of air through the APH, R is the gas constant, T_{ae} is measured temperature of air entering the APH. As already discussed without any significant error Tae represents the AAT. An increase in AAT is reflected in higher value of T_{ae} and as a result, mass flow rate of air (m_a) through the APH decrease

C. Effect of AAT on RH or Humidity Ratio

The specific humidity of intake cold air and that of hot air leaving APH does not change and measurable through dry and wet bulb temperature, expressed as kg of moisture per kg of wet air. The moisture content of air may be determined with a hygrometer at an observed barometric pressure. The RH or humidity ratio is an important parameter providing maximum moisture holding capacity of ambient air at a given temperature. As AAT increases for the constant RH, Humidity Ratio increases with AAT. This means with the increase of AAT, the air becomes more capable for holding moisture. Therefore, enthalpy of air entering the APH changes with humidity ratio. With the increase of humidity ratio enthalpy of air also increases. The enthalpy (h_1) of air at AAT being Tae may be expressed by the following equation

$$h^1 = h_a{}^1 + w h_g{}^1 = C_{pa} * T_{ae} + w * h_g{}^1 \quad (9)$$

where w = RH, h_a^1 = enthalpy of dry air at T_{ae} °C; h_g^1 = enthalpy of saturated air at T_{ae}°C; C_{pa} = specific heat capacity of air at temperature, Tae.

According to Nag (2007), the RH (w) can be calculated as

w □ 0.622 * P_v/□ P □ P_v □□(10)

where P = total pressure of air at given AAT and P_v = saturation pressure of air at temperature T_{ae}. As humidity ratio increases moisture content of air also increases, thus loss of fuel heat energy occurs to dry up this moisture, evaporated within the system. Considering the humidity ratio be 'x' per kg of dry air, heat loss (H_L) due to moisture per kg of air may be given by

$$H_L = \{ x.C_m/ (x+1)\}(T_{al} - T_{ae}) \tag{11}$$

where C_m = specific heat content of moisture.

D. Effect Of AAT in APH Efficiency

The efficiency of APH (η_{APH}) as already discussed may be represented on heat transfer and mass flow basis considering the effects on air and gas flow circuit of regenerative

APH as: $$\eta_{APH=} [m_a.C_{pa}(T_{al}-T_{ae})]/[\{(m_a.C_{pg})-(m_A.C_{pA})\}.(T_{ae}-T_{gl}) \tag{12}$$

where each notation has its usual significance and m_a = mass of fly ash in flue gas exiting the air heater, C_{pA} = heat capacity of fly ash, kJ/kg/°C.

V. FIELD STUDY TO DETERMINE THE EFFECT OF AAT IN APH EFFICIENCY

A. Plant Configuration Choosen Foe Field Study

A pulverised coal fired thermal power plant with a modified Rankine steam cycle, operating in the subcritical regime with a gross power generation capacity of 135 MWe, and fitted with a regenerative, reheat boiler having a corner firing arrangement was selected in order to assess the performance variation with reference to design criteria and running plant performance of a regenerative APH with variation in AAT. During a field investigation of the plant in central India, which has a significant variation in AAT throughout the year, a collection of operational metrics pertinent to the assessment of APH performance was gathered. To eliminate bias, all data were collected during a 24-hour span to avoid variation in RH factor on the same day. at virtually identical plant generating load conditions (with a variance in plant load of less than 0.5%).

On the field research day in question, the RH in the area was close to 40%. Table 1 lists the information gathered on the mass of air flow (m_a), flue gas flow (m_g), and flow of fly ash (m_A) in flue gas at the APH's input and outlet temperatures as determined by the plant data acquisition system. As indicated in Table 1, the mass flow rate is expressed in tonnes per hour (TPH), whereas the temperatures of the air leaving the APH, the AAT, the flue gas entering the APH, and the flue gas exiting the APH are expressed in degrees Celsius (°C).

Table 1 Operating parameters of APH at 135 MWe gross power generation with different AAT

Observations	m_a *(TPH)*	m_g *(TPH)*	m_A *(TPH)*	T_{al} *(C)*	T_{ae} *(°C)*	T_{ge} *(°C)*	T_{gl} *(°C)*
1	235	347.49	38.61	304	42	349	143
2	228	342	38	305	42.5	359	146
3	219	346.5	38.5	304	45	363	149
4	210	351	39	302	47.5	360	155
5	200.8	346.5	38.5	306	50	363	159

The variation of mass flow rate of air when plotted against AAT is shown in Figure 2. The graph in Figure 2 plotted against given data matches very close to the relation as shown for ma and AAT (T_{ae}). Therefore, it is seen that due to 8°C increase in temperature there is about 35 TPH decrease in air flow which is almost 4.37 TPH/°C change in AAT. Table 2 provides the enthalpy of air at various AAT along the calculated energy loss to evaporate air – moisture at RH of 40%. The specific humidity is expressed as moisture content in kg per kg of wet air. The results of Table 2 are used to represent graphically the relationship between AAT with fuel energy lost to evaporate air – moisture per kg of intake air in APH as shown in Figure 3.

Figure 2 Variation air flow rate with increase in AAT

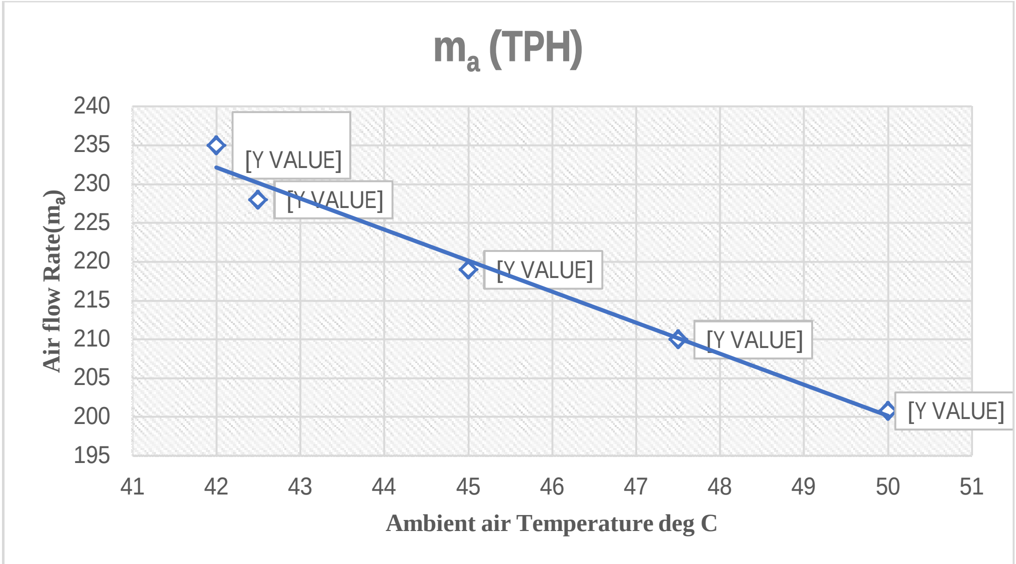
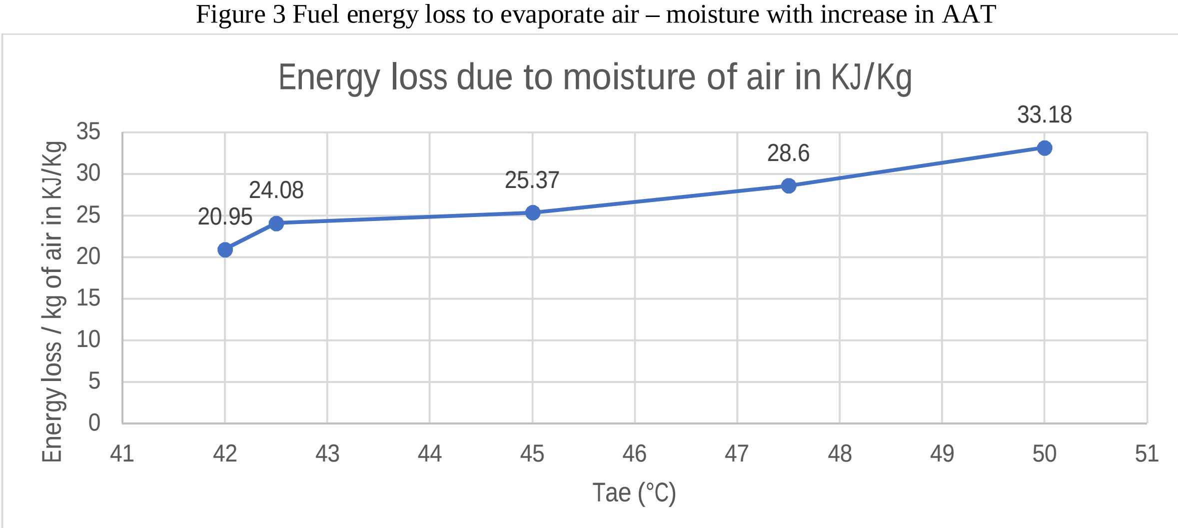


Table 2 Energy loss due to moisture at various AAT with 40% RH

Observations	*T_{al} (°C)*	*Tae (°C)*	*Specific Humidity*	*Energy loss due to moisture of air in KJ/Kg*	*h^1: enthalpy of air at T_{ae}(^{O}C)*
1	304	42	0.0195	20.95	92
2	305	42.5	0.0225	24.08	95
3	304	45	0.024	25.37	105
4	302	47.5	0.027	28.6	110
5	306	50	0.032	33.18	133

Figure 3 Fuel energy loss to evaporate air – moisture with increase in AAT



Therefore, it is observed that due to 8°C increase in AAT, energy loss is almost 12 KJ/kg of air. On an average, there is energy loss of 1.53 KJ/kg of air for each degree centigrade increment in inlet air temperature.

B. Variation Of APH Efficiency With ATT

Efficiency calculation of APH is done on the basis of above operating parametersoutlined in Table 1, according to the equation (15). The result of the APH efficiency obtained for different AAT is shown in Table 3. The change in APH efficiency due to thevariation of AAT and at a given RH is outlined graphically through Figures 4 and 5. Figure 4 represents the relationship of APH efficiency with AAT and Figure 5 representsthe variation in gross overall efficiency with AAT. The different values adopted for specific heat capacity for air, flue gas and fly ash are C_{pa} = 1.047 KJ/kg/°C @ 300°C, C_{pg} = 1.05 KJ/kg/°C, C_{pA} = 0.8KJ/kg/°C respectively as used for calculation of efficiency values and on the basis of which the results are given in Table 3.

Table 3 APH efficiency at various AAT

Observations	T_{ae} (°C)	η_{APH} (%)	$\eta_{overall}$ (%)
1	42	89.94	77.29
2	42.5	86.15	73.02
3	45	79.99	65.54
4	47.5	77.8	62.5
5	50	76.04	55.38

Figure 4 Variation of APH efficiency with AAT

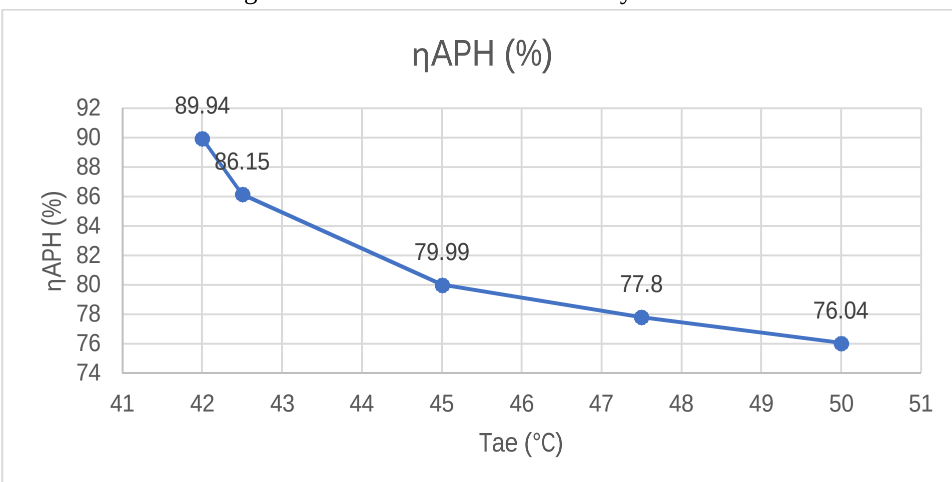
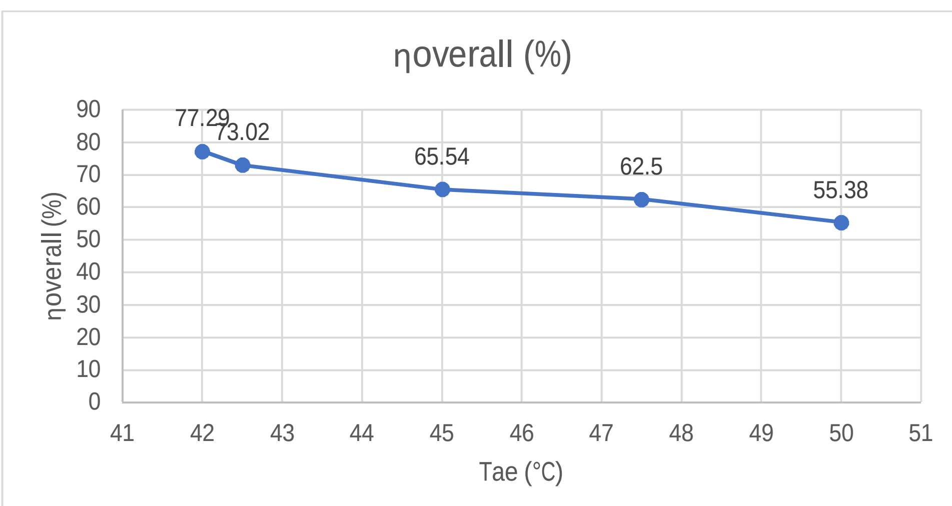


Figure 5 Variation of APH overall efficiency with AAT



VI. CONCLUSION

Deterioration in APH performance inevitably results in a loss of unit capacity that causes commercial losses, a reduction in boiler efficiency because of incomplete combustion, increased dry flue gas loss, and finally a rise in energy use in the draught system. The loss of ID fan margin and increase in ID fan power consumption due to air leakage in the flue gas path within the APH eventually limits the amount of extra O2 levels available in the furnace to ensure combustion completeness, limiting unit capacity. As a result, it is determined that using regenerative APH in tropical regions with greater AAT is not an energy-efficient alternative since significant fuel energy is wasted to remove moisture from the air in addition to the high moisture levels low-rank coal used for pulverised coal fired power generation system. In reality, hot PA drying will result in a greater loss of fuel energy since it dries out low-rank, high moisture coal, which leaves less heat available for steam production. As a result, atmospheric fluidized bed dryers that use waste heat from flue gas downstream of ID fans and before exhaust through stacks can more economically accomplish high moisture coal drying (Bhattacharya and Banerjee, 2011). The partial flue gas recirculation through pulverizer (PFGR) system can lower the APH heat transfer loading because the capacity of coal drying can occasionally be limited by an inadequate hot PA temperature, which in turn affects the capacity of power production. Additionally, a decrease in the demand for coal drying results in a rise in secondary air temperature at APH, which ensures improved combustion with a decreased risk of NOx formation and less extra air.

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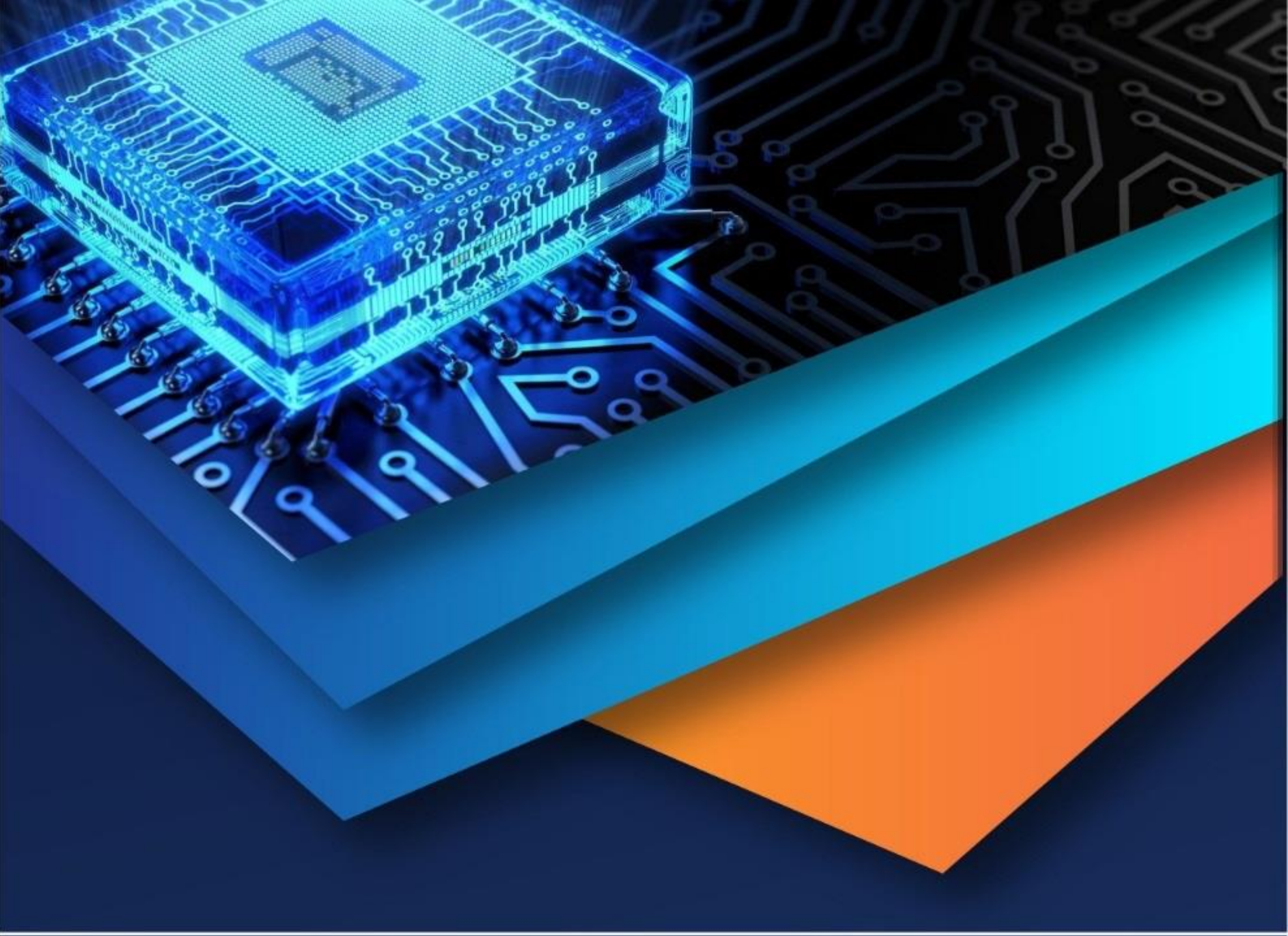

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