

Localization of Exergy Destruction in Industrial Thermal Oil Heater

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Abstract— Thermodynamic and technology parameters of green carbon anode manufacture should be optimized in order to decrease energy consumption. Exergy Efficiency or second law of thermodynamics determinates the degree of sophistication of the process. Its definition of system heat-using elements in green carbon anode manufacture allows revealing some units defined by considerable irreversibility. Exergy analysis allows accounting and identifying the destruction of Exergy through the green carbon anode plant elements.

Further than the aluminum industry has the highest Exergy efficiency in the total aluminum plant of 52.5%. Mean ambient environment

temperature is also proposed as a tool for understanding the degree of quality of energy required in this industry and

consequently better quality matching which leads to better energy quality utilization. The purpose of this study is to assess the use of principal of availability of energy in Egyptian industrial sector. The Exergy analysis has been performed along with energy analysis, in order to gain deeper and more realistic understanding of the sector's condition. The sources of energy degradation and the mechanisms which cause degradation of quality of energy have been identified. Moreover remedial actions for better utilization of availability of energy are proposed.

Index Term--Combustion,Exergy, Energy, Green Anode, Coke, Thermal System.

NOMENCLATURE

A	Area [m ²]	S	Distance [m]
A _{Abs}	The absolute availability of a system [-]	S _o	Entropy of a system at environmental state [J/kg]
C _{p, c}	Specific heat capacity [J/kgK] or heat capacity [J/kg]	S _i	Specific entropy, of substance I, [J/kgK]
e, E	Specific exergy [J/kg] or available work [J]	T	Temperature [K]
E _f	Fuel exergy [kJ/kg]	u, U	Specific internal energy [J/kg] or internal energy [J]
e ^{ch} = A _{ch}	Chemical exergy [kJ/kg]	v, V	Specific volume [m ³ /kg] or volume [m ³]
E/Q	Exergy factor [no unit, %]	σ	Stefan-Boltzmann constant [W/m ² K ⁴]
h, H	Specific enthalpy [J/kg] or enthalpy [J]	ε	The emissivity [-]
m _a	Mass flow rate of air [kg/s]	ω	Exergetic efficiency [-]
m _f	Mass flow rate of fuel [kg/s]	q, Q	Specific heat [J/kg] or heat [J]
P _o	Environment pressure [bar]		

I. INTRODUCTION

The most common thermodynamic assessment parameter is energy efficiency. It is considered an indicator how well an energy conversion or transfer process is accomplished. However, this definition is not sufficiently precise, as energy efficiency does not take into account internal irreversibility. Exergy analysis is a universal method for evaluating the rational use of energy. It can be applied to any kind of energy

conversion system or chemical process. An exergy analysis identifies the location, the magnitude and the causes of thermodynamic inefficiencies and enhances understanding of the energy conversion processes in complex systems, moreover exergy destruction accounting give us the chance to put our hands on the points where our available energy destroyed and create more efficient machines and systems

which can use the energy by improved and optimized methods.

The analysis of the real thermodynamic inefficiencies in a system and the system components is valuable for improving an energy-intensive operation. Exergy destruction accounting may be used for make the systems more efficient for saving energy, environment consideration, and less fuel consumption.

Exergy analysis allows accounting and identifying the destruction of Exergy through the thermal oil heating plant components. The plant is evaluated on the basis of the first and second laws of thermodynamics, expressed by the mass, energy and exergy balances of the system. Exergy rate of each process stream is calculated and the exergetic balance of each plant's component is presented, in order to identify the amount and the place where losses and irreversibility occur. Rational and simple exergetic efficiency are calculated for each component of the plant and compared.

Review of previous work

Exergy analysis is particularly valuable tools in the design and analysis of thermal systems since they help in identifying sources of irreversibility in the system. The second law has been applied to analysis of internal combustion engines, turbines, boilers, heaters, and power plants. As stated previously, the current work aims to understand Exergy losses during combustion regimes from the viewpoint of implementing these strategies for more efficient combustion in Direct Fired Heaters.

Significant work has been done with regard to applying the second law to heaters and boilers simulations in the past.

One of the first works to include a rigorous treatment of the chemical Exergy component in Exergy analysis of the engines was conducted by Van Gerpen and Shapiro . Van Gerpen and Shapiro outline the need for including the chemical Exergy component for Exergy analysis. They also conclude that the chemical Exergy contribution is significant and needs to be incorporated into the calculations to obtain an accurate estimate of the irreversibility of the processes. The work by Van Gerpen and Shapiro showed that the chemical Exergy increases in significance with increasing temperature and pressure and for richer equivalence ratios. This is mainly due to the increased concentrations of species such as carbon monoxide and hydrogen which are not present in the

atmosphere and have a significant chemical energy associated with them.

II. EXPERIMENTAL TEST REG

Thermal fluid systems Figure1 provides an efficient means of supplying indirect heat to one or more process systems. Such systems offer both high temperature and low pressure, making them ideal for a wide variety of process heating application. The heat transfer fluid firstly heated by means of a direct fired heater then circulated through a closed loop systems to the users. Heat from the fluid is transferred to the user and then re-circulated for reheating and the cycle repeated. However, organic media has become more common and often replaces a classic steam-water operation. The heaters are made with coils made of seamless tubes. The thermal fluid is heated during the flow through the tubes. The heat is transferred to the fluid as radiant heat in the combustion chamber, where the inner cylindrical tube coil and a flat tube coil forms the chamber wall and the bottom respectively.

III. EXPERIMENTAL BURNER

We will use the (Direct Fired Heater) which used in EGYPTALUM company as test rig, and will describe this heater consists here as follow:

Design of Directly Fired Heaters

Because most heat carriers decompose from a certain temperature, this temperature must not be exceeded at any point in the heater. A circular furnace cross-section is almost exclusively chosen to maintain a constant ratio of flame diameter to furnace diameter because of the axially symmetrical flame shape. At our test rig the furnace have a cylindrical furnace with a heating surface consisting of tubes of constant diameter. The maximum permissible internal tube wall temperature exists in this design only over a small range. The life of the heat carrier depends on the maximum residence time and the participating volume in the thermal boundary layer, as will describe in next section. A heat carrier exposed only partially to the maximum permissible temperature has a longer life, Figure 1 show the construction of the Direct Fired Heater.

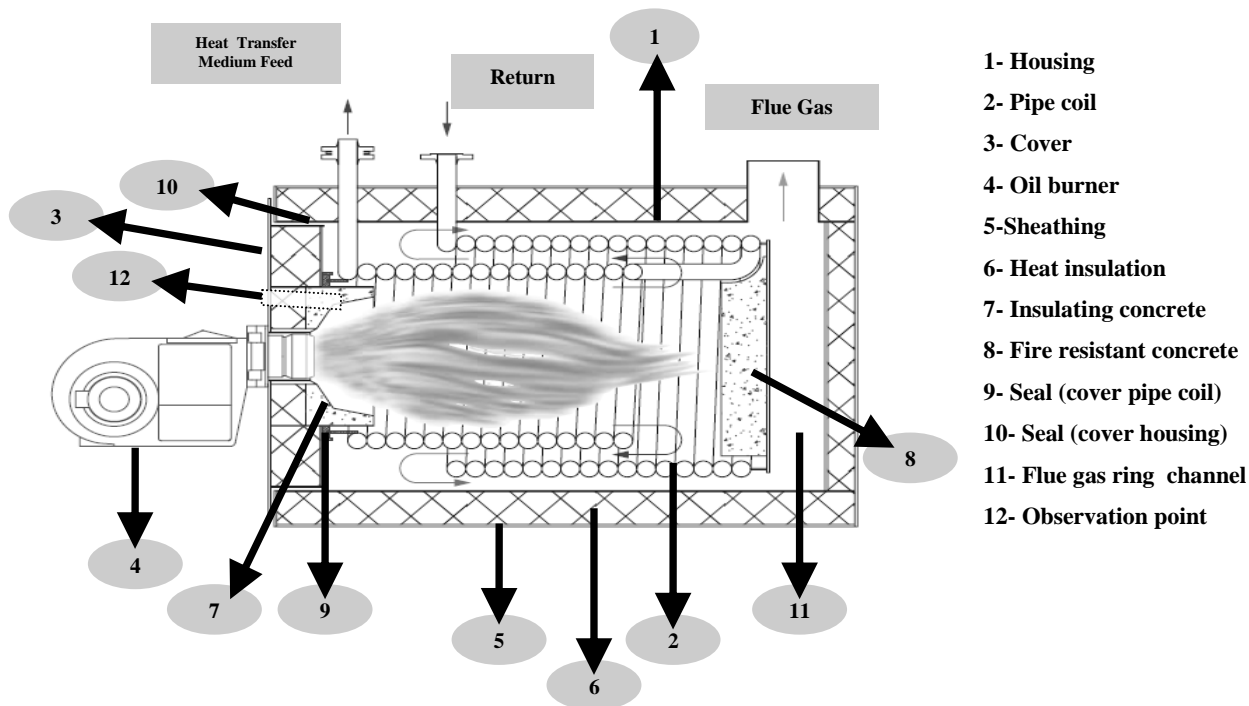


Fig. 1. Direct Fired Heater (Thermal Oil Heater)

EXPERIMENTAL FUEL

The fuel that will be used in this test will be Natural gas. Natural gas is a subcategory of petroleum that is a naturally occurring, complex mixture of hydrocarbons, with a minor amount of inorganic compounds. Table 1-1 shows composition of a typical natural gas. It indicates that methane is a major component of the gas mixture. The inorganic compounds nitrogen, carbon dioxide, and hydrogen sulphide are not desirable because they are not combustible and cause corrosion and other problems in gas production and processing systems. Depending upon gas composition, especially the content of inorganic compounds, the heating value of natural gas usually varies from: 33,000 to 38,000 kJ/m³.

Table I
Composition of a typical natural gas:

Compound	Mole Fraction
Methane	0.8407
Ethane	0.1586
Propane	0.0220
i-Butane	0.0035
n-Butane	0.0058
Nitrogen	0.0345
Total	1.0000

SUPPLYING, CONDUCTING AND MONITORING THE PRIMARY AIR

The radial fan impeller supplies and compresses the air and conducts it, via the diffuser, to the hollow ring. The primary air flow is about fifteen per cent of the required combustion air flow. The pressure increase caused by the primary air fan is monitored by a differential pressure monitor. By means of an electromagnetic valve connected in parallel the proper operation of the monitor can be checked.

Supplying, Conducting and Distributing the Combustion Air

The fan unit suctions the combustion air through the air inlet elbow unit from the ambient atmosphere and conducts it through the air proportioning assembly and the carrying and air guiding unit into the combustion air annulus which is formed by the outer and inner air annulus rings. From there, the combustion air is guided through several, axially arranged vanes uniformly into the furnace. Part of the combustion air passes via the ring body into the space between the shroud and the inner air annulus ring. This air then flows past the vanes of the axial vane ring unit, which impart a whirling motion to it, and passes into the furnace. For monitoring the fan unit, a pressure monitor is installed.

START UP IGNITION, DESCRIPTION OF OPERATION AND DESIGN

The igniter operates on the principle of pressure jet oil atomization. It is designed for firing light fuel oil of a maximum viscosity of 17 mm²/s at 20°C. The fuel oil atomized by the nozzle is ignited by a high-voltage spark drawn between the electrode tips Figure 2. The combustion air is either taken from the windbox.

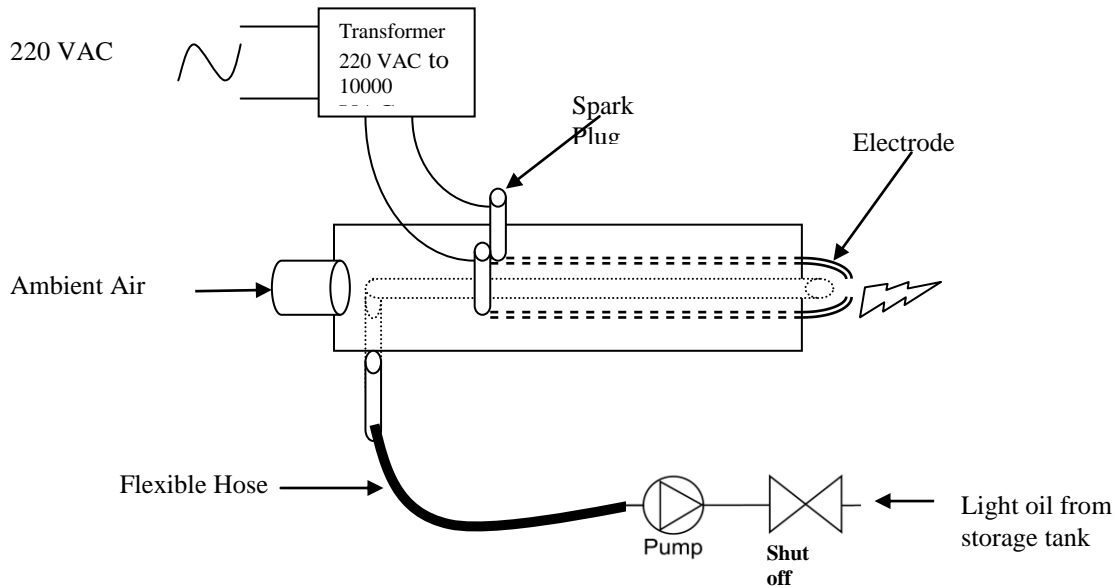


Fig. 2.Schematic diagram for Ignition system.

The ignition transformer generates from the power supply voltage the high voltage required for the formation of the ignition spark in the igniter. The high tension lead to the ignition electrode is firmly attached to the ignition transformer and installed at its free end with a connection plug for the ignition electrode. The igniter assembly comprises essentially the igniter head, stabilizing disk and ignition electrode. For presetting the ignition oil flow a setting screw is installed. No special air supply system is required.

EXPERIMENTAL PROCEDURES

Measurement of excess air

One of the most critical operating parameters for attaining good combustion is excess air. Too little air can be a source of excessive unburned combustibles and can be a safety hazard. Too much excess air increases stack gas losses. The total gaseous products of combustion are referred to as wet flue gas. Solid products or residue are excluded. The wet flue gas flow rate is used for heat transfer calculations and design of auxiliary equipment. The total gaseous products excluding moisture are referred to as dry flue gas; this parameter is used in the efficiency calculations and determination of flue gas enthalpy. The wet flue gas is the sum of the wet gas from fuel (fuel less ash, unburned carbon and sulfur captured), combustion air, moisture in the combustion air, additional moisture such as atomizing steam and, if sorbent is used, carbon dioxide and moisture from sorbent. Dry flue gas is determined by subtracting the summation of the moisture terms from the wet flue gas.

Flue gas analysis

The major constituents in flue gas are CO_2 , O_2 , N_2 and H_2O . Excess air is determined by measuring the O_2 and CO_2 contents of the flue gas. Before proceeding with measuring techniques, consider the form of the sample. A flue gas sample may be obtained on a wet or dry basis. When a sample is extracted from the gas stream, the water vapor normally condenses and the sample is considered to be on a dry basis. The sample is usually drawn through water near ambient

temperature to ensure that it is dry. The major constituents of a dry sample do not include the water vapor in the flue gas. When the gas is measured with an in situ analyzer or when precautions are taken to keep the moisture in the sample from condensing, the sample is on a wet basis. The amount of O_2 in the flue gas is significant in defining the status of the combustion process. Its presence always means that more oxygen (excess air) is being introduced than is being used. Assuming complete combustion, low values of O_2 reflect moderate excess air and normal heat losses to the stack, while higher values of O_2 mean needlessly higher stack losses.

The Testo 350S unit, which measures (CO_2 & H_2O) and O_2 on a dry volumetric basis, remains a trusted standard for verifying the performance of electronic equipment. The Testo 350S uses chemicals to absorb the CO_2 and O_2 , and the amount of each are determined by the reduction in volume from the original flue gas sample.

Temperature measurements

Thermal oil inlet, thermal oil outlet, ambient temperature, flue gas temperature and outer surface temperature, all are measured in this study for more accurate calculation. Temperature of thermal oil inlet, thermal oil out, and flue gas will be measured by PT -100 Thermocouple.

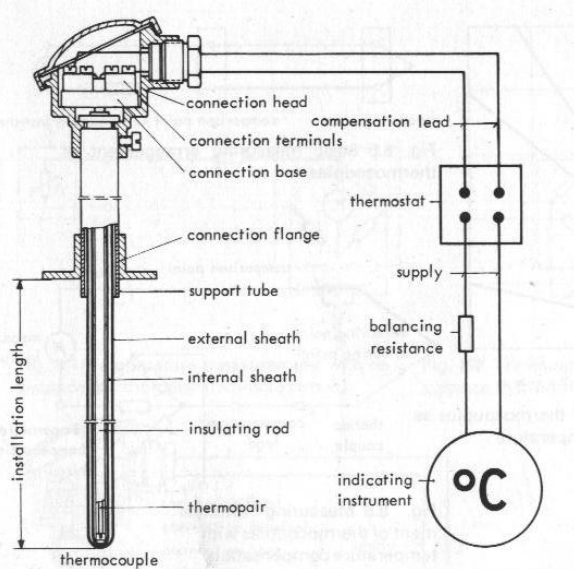


Fig. 3. PT100 Thermocouple

Temperature of outer surface of heater will be measured by using Raytek® MiniTemp device Model MT2 which use Infrared sensor technology to measure temperature of surfaces without contact, Figure5 photo of Raytek® MiniTemp device.

$$\omega = \text{Exergetic efficiency} = \frac{\text{Exergy output}}{\text{Exergy input}} \quad (1)$$



Fig. 4. Raytek® MiniTemp device Model MT2

EXPERIMENTAL PROCEDURES AND METHOD OF CALCULATIONS

The most commonly used indicator for the efficiency of energy conversion process is the ratio of the output of useful energy to the total energy input. This ratio is called first law efficiency. It is based on a quantitative accounting of energy, which reflects recognition of the first law of thermodynamics and the law of conservation of energy.

It is well known that the second law of thermodynamics defines the availability of energy more restrictively than the first law. Principally, first law is silent on the effectiveness

with which availability is concerned. Analysis in terms of the second law of thermodynamics more closely describes the effectiveness with which systems or processes use available energy.

Exergy losses are calculated by making Exergy balance for each component of the system. Unlike energy balance where the inflow is equal to outflow (when there is no internal energy generation or consumption), in Exergy balance due to reasons of irreversibility, Exergy inflow is always greater than the Exergy outflow and their difference gives the Exergy loss or Exergy destruction. Ratio of Exergy output to Exergy input gives the Exergetic efficiency of a system.

Tests and readout measurements for eight adjustment natural gas flow rate, and for all natural gas flow rate adjustment, we will execute five tests at different excess air ratio. The summations of tests which will be run are forty tests. The eight adjustment for natural gas flow rate are controlled by of servomotor of burner which is responsible about adjustment the air fuel ratio for all loads and start up conditions for burner. For all test we will measure the following:

- 1- Thermal fluid inlet temperature °C.
- 2- Thermal fluid outlet temperature °C.
- 3- Flue gas temperature °C.
- 4- Ambient Air temperature °C.
- 6- Natural gas flow rate m³/hr
- 7- Thermal fluid flow rate m³/hr
- 8- Thermal fluid pressure bar.
- 9- Excess air %.
- 10- Flue gas compositions.
- 11- Heater outer surface temperature °C.

Energy Calculations (First law of thermodynamics)

In this section we will apply the first law of thermodynamics on our system; the traditional method will be used in energy analysis by using energy balance equations and energy conservation principle. The tendency of this analysis is to obtain energy analysis of Direct Fired Heater which will be compared after that with Exergy analysis which will obtain in next section.

Energy Analysis Formula

The energy balance can be expressed in symbols as

$$E_2 - E_1 = Q - W \quad (2)$$

$$\Delta KE + \Delta PE + \Delta U = Q - W \quad (3)$$

which shows that an energy transfer across the system boundary results in a change in one or more of the macroscopic energy forms: kinetic energy, gravitational potential energy, and internal energy.

Energy enter the control volume = Energy Out from system
In this experimental, Energy enter represented as natural gas only, and Energy Out represented as

$$E_{in} = E_{oil} + E_{stack} + E_{loss, surface} + Irriv. \quad (4)$$

Energy transfer to thermal fluid, Energy losses from heater surface to surrounding, and Energy loss at stack gases therefore :

Where,

E_{in} = Energy enter with natural gas=LHV of fuel = 37033kJ/kg

E_{oil} = Energy transfer to thermal fluid

E_{stack} = Energy loss at stack gases

$E_{loss, surface}$ = Energy losses from heater surface to surrounding

Energy enter to system represented as chemical energy in natural gas equal to Lower heating value of fuel, the LHV of natural gas (37033kJ/m³).

Energy transfer to thermal fluid is the useful part of total of energies which exit from the system, equation (5) used to identify the quantity of heat which transfer to thermal oil.

$$Q_{oil} = \dot{m} \cdot C_p \cdot \Delta T \quad (5)$$

Where,

$Q_{oil} = E_{oil}$

\dot{m} = Thermal fluid flow rate kg/hr

C_p = Specific heat capacity of thermal oil

ΔT = different temperature between inlet and outlet of thermal fluid

Stack losses as hot gases exit from the system to surrounding equal to the quantity of heat content in flue gas which can be calculated by using equation (6)

$$Q_{stack} = \dot{m} \cdot C_p \cdot \Delta T \quad (6)$$

Where,

$Q_{stack} = E_{stack}$

\dot{m} = mass flow rate of products of combustion kg/hr

C_p = Specific heat capacity of products of combustion

ΔT = different temperature between flue gas and surrounding.

Heat transfer from the surface area A of heater to surrounding (surface losses) which is quantified macroscopically by a modified form of the Stefan–Boltzmann law equation (7)

$$Q_e = \varepsilon \sigma A (T_s^4 - T_o^4) \quad (7)$$

Which shows that a quantity of heat that transfer to surrounding by thermal radiation Q_e is associated with the fourth power of the absolute temperature of the surface, T_s .

ε = The emissivity is a property of the surface that indicates how effectively the surface radiates ($0 \leq \varepsilon \leq 1.0$), and for Egyptalum Direct Fired Heater the outer surface is made of polished aluminum which have $\varepsilon = 0.08$.

σ is the *Stefan–Boltzmann* constant = $5.67 \cdot 10^{-8}$ [W/(m². K⁴)],

and A surface area of heater , for this study the area of outer surface of heater = 48m².

In general, the *net* rate of energy transfer by thermal radiation between two surfaces involves relationships among the properties of the surfaces, their orientations with respect to

each other, the extent to which the intervening medium scatters, emits, and absorbs thermal radiation, and other factors.

Energy efficiency of the system is evaluated by using equation (8)

$$\eta = (E_{oil} + E_{stack} + E_{loss, surface}) / E_{in} \quad (8)$$

Where, E_{oil} = Energy transfer to thermal fluid, calculated by equation (5)

E_{in} = Energy enter with natural gas = LHV of fuel = 37033 kJ/kg.

Exergy calculation (second law of thermodynamics)

The objective of this section is to introduce Exergy analysis, a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the investigate and analysis of combustion process and thermal systems. Another term frequently used to identify Exergy analysis is availability analysis.

This difference in composition between the restricted dead state and the environment can be exploited to further obtain work from the system. This work, obtained by allowing the restricted dead state to come to chemical equilibrium with the environment, is referred to as the chemical Exergy, A_{Ch} , of the system.

$$A_{Ch} = \sum_{k=1}^n N_k (\mu_k^0 - \mu_{k,0}) \quad (11)$$

Where N_k is the number of moles of the respective species (k), $\mu_{k,0}$ and μ_k^0 are the chemical potentials of the respective species in the restricted dead state and the environment, respectively.

The chemical potentials may further be expressed as:

$$\mu_k = g_k(T_0, P_0) + \bar{R} T_0 \ln \left(\frac{p_k}{p_0} \right) \quad (12)$$

Where g_k is the Gibbs energy of the kth species in the mixture, R is the Universal gas constant and p_k is the partial pressure of the kth species in the mixture. If the restricted dead state and the environment, both had the same constituent species, differing only in their respective compositions, the Gibbs energy term would cancel out, leaving a simpler expression for the chemical availability of the system

$$A_{Diff} = \bar{R} T_0 \sum_{k=1}^n N_k \ln \left(\frac{p_k^0}{p_{k,0}} \right) \quad (13)$$

The difference in concentrations of the various species in the system and the atmosphere may be exploited by first separating the various components in the mixture (using devices such as semi-permeable membranes) and then allowing them to expand or compress to the atmospheric partial pressures, as the case may be. Work may be gained or lost during this process and this creates an additional potential for work. Since this term may be attributed to the work obtained by allowing the species in the system to diffuse to the atmospheric concentrations, it would be appropriate to refer to

this as the “diffusion availability”. It may be noted that diffusion availability of a system can be positive or negative, depending on the concentrations of the various species in the system.

The diffusion availability of a system is largely ignored since its contribution is often small relative to the thermo-mechanical availability A_{TM} of the system. Also, it is not easy to extract the diffusion availability component of the availability since it would require the use of semi-permeable membranes to extract the various species in the mixture before allowing them to diffuse to atmospheric concentrations.

It is also evident from the expression for the diffusion availability of a system that it depends on the composition of the environment. The assumed composition of the atmosphere therefore, makes a difference on the diffusion availability of the system. The current work uses a standard wet atmospheric composition unless otherwise stated. The availability of a system, A_{Total} , incorporating the various components would then be

$$A_{Total} = (U - U_0) - T_0(S - S_0) + P_0(V - V_0) + \sum_{k=1}^n N_k (\mu_k^0 - \mu_{k,0}) \quad (14)$$

The above expression for availability is valid for closed systems. For open systems, the flow availability, $A_{Total,f}$ needs to be considered. This is defined as

$$A_{Total,f} = (H - H^0) - T_0(S - S^0) + \sum_{k=1}^n N_k (\mu_k^0 - \mu_{k,0}) \quad (15)$$

$$E_{oil} = \text{Exergy flow to thermal oil} = \dot{m} \cdot c_p (T_{out} - T_{in} - T_o \ln \frac{T_{out}}{T_{in}}) \quad (19)$$

where, \dot{m} = Thermal fluid mass flow rate kg/hr
 c_p = average specific heat capacity kJ/(kg.k)
 T_{out} = Thermal fluid outlet temperature Kelvin.

$$E_{stack} = \underline{h} - \underline{h}_0 - T_0(\underline{s} - \underline{s}_0) + e^{ch} \quad (20)$$

T_{in} = Thermal fluid inlet temperature Kelvin.

E_{stack} in equation (18) represents Exergy flow to surrounding with flue gases and calculated here from equation (20):

where in eq. (4-11), h and s represent the specific enthalpy and entropy, respectively, at the inlet or exit under consideration; h_0 and s_0 represent the respective values of these properties when evaluated at the dead state. Values of h , s , h_0 and s_0 are from standard tables of thermodynamics.

Where the underlined term is the thermomechanical contribution of Exergy in combustion products e^{ch} is the chemical contribution evaluated as following :

$$\bar{e}^{ch} = \bar{R}T_0 \sum_i y_i \ln \left(\frac{y_i}{y_i^e} \right) \quad (21)$$

Where, \bar{R} = Universal Gas Constant = 8.314 kJ/kmol. K, and y_i and y_i^e denote, respectively, the mole fraction of component i in the mixture of combustion products at T_0 , P_0 and in the

Where H and H^0 are enthalpies of the system and the restricted dead state respectively.

In general, then, the availability of a system, A_{Total} , may be expressed as a sum of the thermo-mechanical availability and chemical availability.

$$A_{Total} = A_{TM} + A_{Ch} \quad (16)$$

The chemical availability term may further be split into constituents, the reactive availability and diffusive availability as:

$$A_{Total} = A_{TM} + A_{Reactive} + A_{Diff} \quad (17)$$

Exergy analysis formula

The start point in the Exergy analysis is Exergy balance for a system, Exergy balance for this system can be symbolized as:

Where, E_{in} represent chemical Exergy involved in natural gas

$$E_{in} = E_{oil} + E_{stack} + E_{s,loss} + E_{Destruction} \quad (18)$$

enter to combustion chamber, also Exergy of fuel = LHV * 1.04.

that means $E_{in} = 37033 * 1.04 = 38514$ kJ/m³ natural gas
 E_{oil} in equation (18) represent Exergy flow to thermal oil, and calculated here from equation (19):

environment, with assumption that products of combustion are modeled as an ideal gas mixture at all states considered. $E_{s,loss}$ in equation (18) represents Exergy flow to surrounding by radiation from the surface of heater and calculated here from equation (22):

$$E_{s,loss} = (1 - [T_o / T_{surf}]) * Q_e \quad (22)$$

where Q_e calculated from equation (4-7).

$E_{Destruction}$ in equation (18) represents Exergy Destruction inside furnace because irreversibility and are calculated by making Exergy balance for control volume in this study.

RESULTS AND ANALYSIS

In fig. (5) below, which represents the variation of energy efficiency with variation of excess air at different levels of fuel flow rate, it is obvious that energy efficiency trends to decrease while excess air level increased. In microscopic view for trend line, clear that high value of energy efficiency achieved in range of 8% to 20% for excess air. For all levels of fuel flow rate, energy efficiency values limited in range of 60% to 82% for all runs.

Exergy Efficiency

Exergy Efficiency Variation with Variation of excess air is presented in the same figure (5) below at variation of natural gas flow rate. From fig. (5) below, it is clear that the trend line is the same as appear in energy efficiency representation in

fig. (5) but change in value of exergy efficiency than energy efficiency, since energy efficiency in range of 68% to 87%, the exergy efficiency in range of 26% to 34% for the same

runs, and the higher value of exergy efficiency is achieved at excess air in range of 8% to 20%.

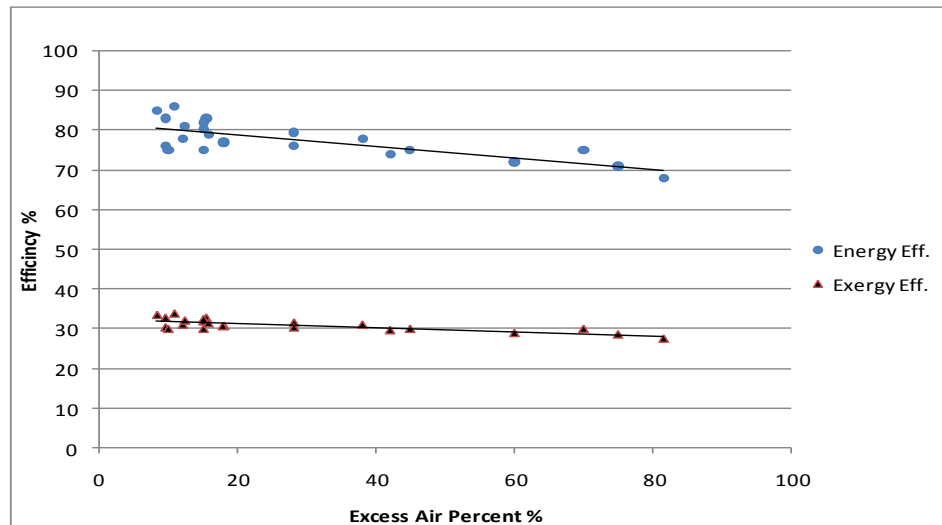


Fig. 5. The Variations of energy efficiency and Exergy efficiency with variation of excess air ratio

Exergy losses through surface emission

Radiation and convection are the two modes that create surface losses. The emission through these two modes depends on the surface temperature, surface area, and condition of the surface and wind velocity. In general, heaters are operated inside a heaters room and the average wind velocity is almost constant everywhere and it is around 1.5 m/s. The surface area depends on the capacity of the heater but the surface condition (the surface emissivity) is almost constant and in average the surface emissivity is in the range of 0.08. The percentage Exergy loss varies between 0.004% and 0.008%. Whereby, this achieves all possibilities to reduce the Exergy loss through the surface emission, whereas the surface temperature of the heater is in the range of 50°C to 70°C more than the ambient. In the case of high losses from the surface it is possible to bring the surface temperature closer to the ambient. But the insulation cost will get increased while reducing the surface temperature. Therefore, there should be a compromise between these two factors and as a rule of thumb it can be said that the surface temperature should be in the range of 50°C more than the ambient.

The effect of variation of Flue Gas Temperature on Energy efficiency with different levels of fuel flow rate:

In fig. (6) which represent the variation of energy efficiency with variation of flue gas temperature, energy efficiency decrease when flue gases temperature increase nearly in linear relation as obvious in figure for natural gas variation between 134 m³/hr and 376 m³/hr as listed above. Therefore, by implementing a proper operation and maintenance program,

the flue gas temperature can be maintained at a low level. But the temperature reduction should be limited to 180°C.

According to this there should be an optimum limit for flue gas parameters in Direct Fired Heaters. The equivalent O₂ concentration is 1.5% - 3.0%. If the combustion is complete the equivalent CO₂ concentration is 11% - 13%.

The effect of variation of Flue Gas Temperature on Exergy efficiency with different levels of fuel flow rate:

In fig. (6) which represent the variation of exergy efficiency with variation of flue gas temperature, exergy efficiency decreases when flue gases temperature increase nearly in linear relation as obvious in figure for natural gas variation between 136 m³/hr and 376 m³/hr. The effects of variation of Flue Gas Temperature on Exergy efficiency with different levels of fuel flow rate are the same as on energy efficiency with different in the magnitude, also quality of the flue gas is a measure of the combustion and the heat transfer. If the flue gas temperature is high more Exergy is lost through the stack.

The variation of Energy Stack Losses with variation of Excess air at variation of natural gas flow rate:

Fig. (7) below shows the variation of energy stack losses with variation of excess air at different levels of fuel flow rate, in this figure it is clear that general trend line go to high level of energy which go to surrounding with flue gases with increase of excess air level, but in range of 8% to 20% excess air stack losses decrease and return to increase again with excess air. Energy stack loss value limited by the range 13% to 32% for all runs.

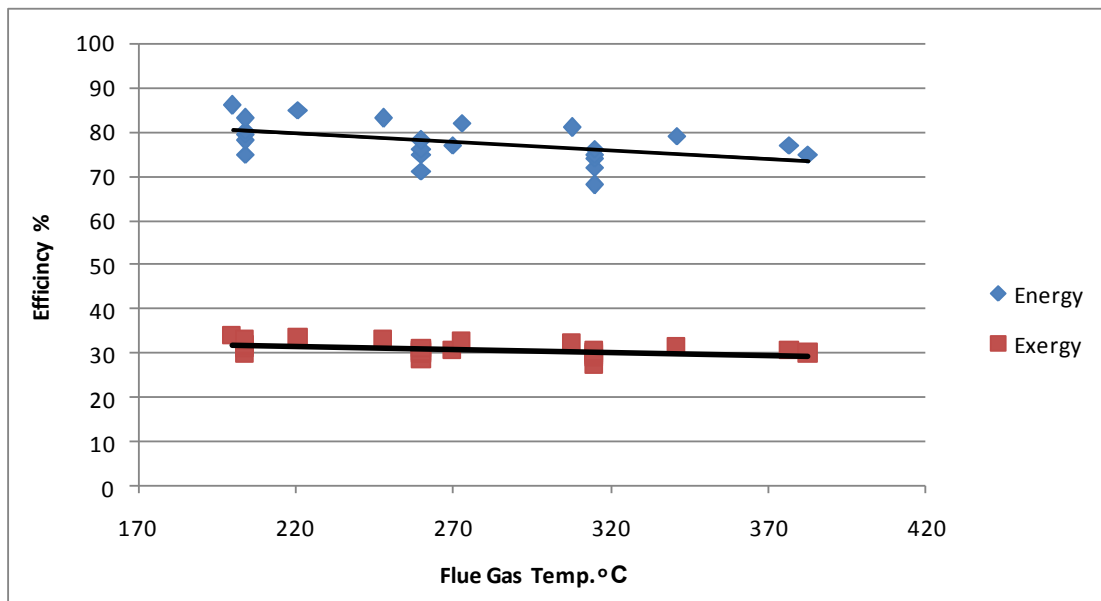


Fig.6.The variation of Energy efficiency and Exergy efficiency with variation of Flue Gas Temperatur

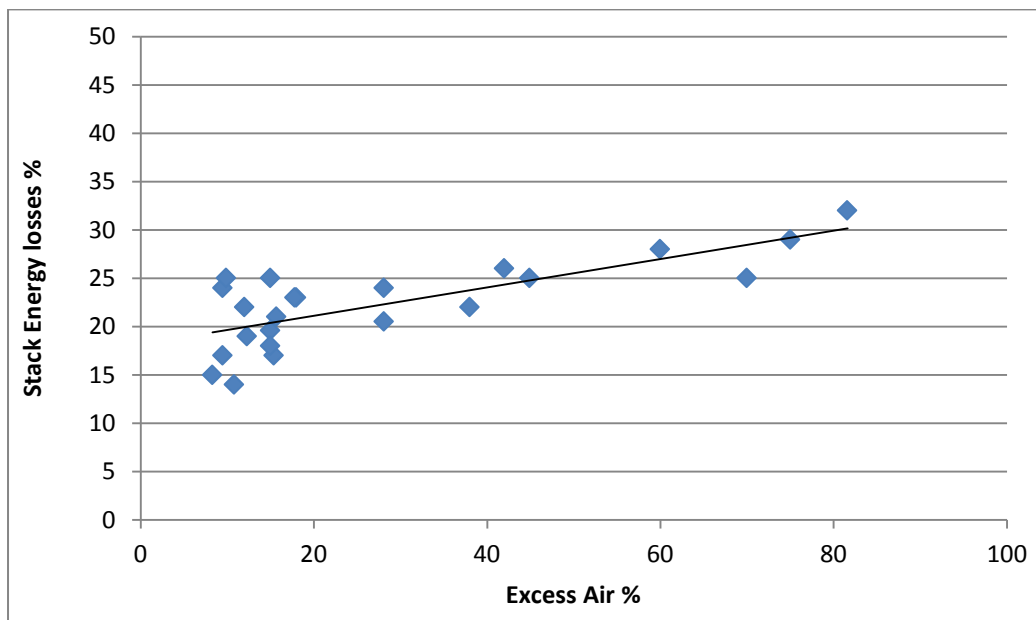


Fig. 7.The variation of Energy Stack Losses with variation of Excess air at variation of natural gas flow rate.

The variation of Exergy Stack Losses with variation of Excess air at variation of natural gas flow rate:

Fig.(8) below shows the variation of exergy stack losses with variation of excess air at different level of fuel flow rate, it is obvious that the trend line of curve are the same as appear in fig.(7) but with changes in value of exergy stack losses than energy stack losses , since energy stack losses vary in range of 13% to 32%, while exergy stack losses vary in range 2% to 12% for the same runs. According to the measurements taken from the stack the flue gas temperature varies between 200°C and 450°C. The excess air level varies between 0 % and 120%. The Exergy loss of the flue gas varies between 2 % & 12 % and the average is 7 %. Reference to the figures given below the Exergy loss gets increased with increase of excess air

level. That means the increase of excess air is most greatly affected to the thermodynamics irreversibilities associated in the flue gas.

5.9 The Relation between Exergy Stack Losses and energy stack losses :

Variation of exergy loss with energy loss in flue gas follows a linear relationship Fig.9 Exergy loss is always lower than the energy loss and approximately energy loss is three times greater than the exergy loss in terms of MJ. The following figures illustrate the variation pattern. The approximate variation can be given by the following formula:

$$\text{Energy loss (MJ)} = 2.7 \times \text{Exergy loss} + 132$$

$$\text{Percentage energy loss} = 2.3 \times \text{Percentage Exergy loss} + 5.2$$

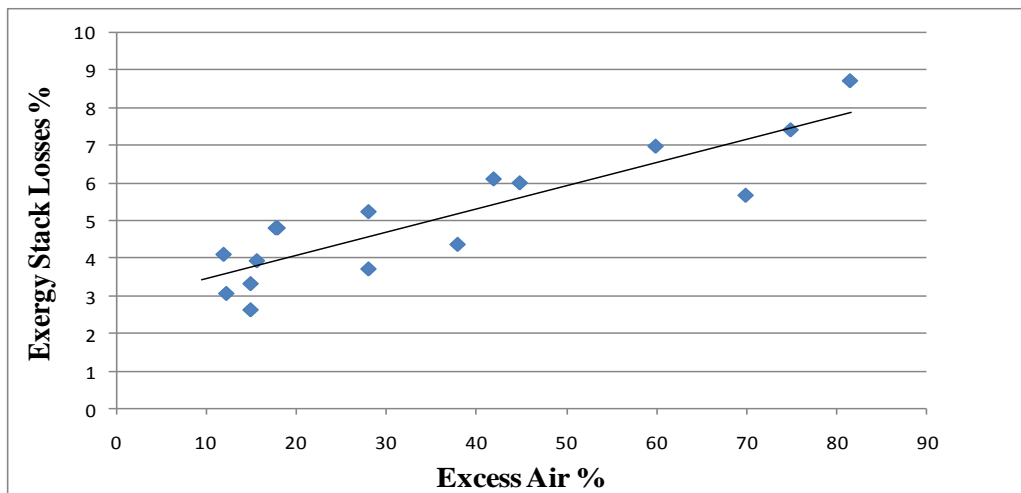


Fig. 8. The variation of Exergy Stack Losses with variation of Excess air at variation of natural gas flow rate.

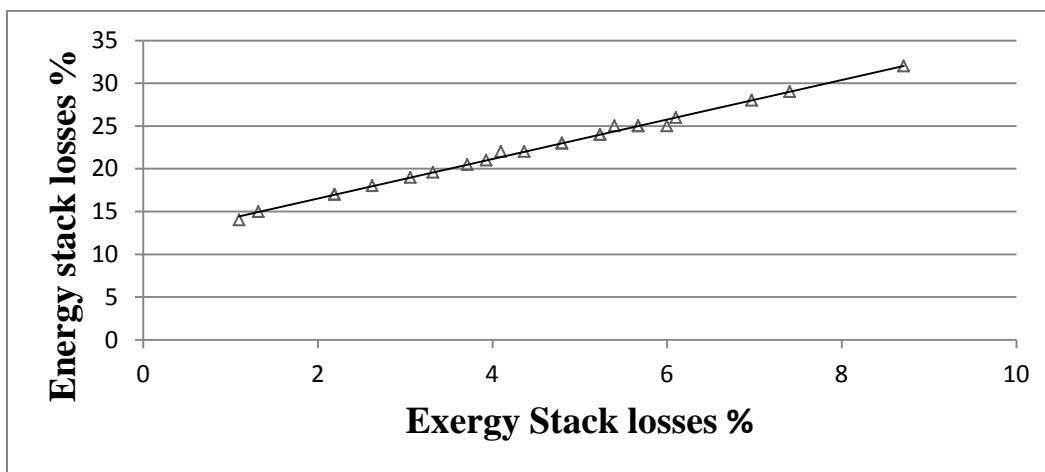


Fig. 9. Variation of exergy loss with energy loss in flue gas

Exergy Destruction

When a system undergoes an irreversible process, the entropy always increases and anything that generates entropy always destroys exergy. Fig. (10) below shows the variation of exergy destruction with variation of excess air at different level of fuel flow rate. From figure (10), appears that exergy destruction within the system under investigation decrease while excess air level increase for all runs, that means the irreversibility within the system decrease with increase of excess air levels. The Exergy of input fuel is at a higher level. The temperature of the combustion products is in the range of 1500°C and the energy in this mixture transfers to thermal fluid which is in low temperature. Mainly Direct Fired Heater is operated in range of 200 to 400°C , and in this study the range is 200 to 280°C for thermal fluid. There is a rapid temperature reduction in this process and due to this reason

Exergy is destroyed rapidly. Apart from the temperature variation, the irreversibility in combustion reaction also causes Exergy destruction. Reference to the figures (11) & (12) represented below which show the variation of Exergy destruction with Exergy loss in flue gas we note that the Exergy loss through the flue gas gets increase with reduce of percentage Exergy destruction and inverse is correct. When the flue losses are low, that means there are more Exergy remaining in the combustion chamber to transfer to the thermal fluid. While transferring Exergy from combustion products to thermal fluid, Exergy gets destroyed rapidly due to sudden temperature reduction between the two states. If the flue gas losses are high, fewer amounts of Exergy transfers to thermal fluid and thus the Exergy destruction is less, but the Exergy loss at the flue gas is high. References to these there are limitations to get the maximum Exergitec efficiency.

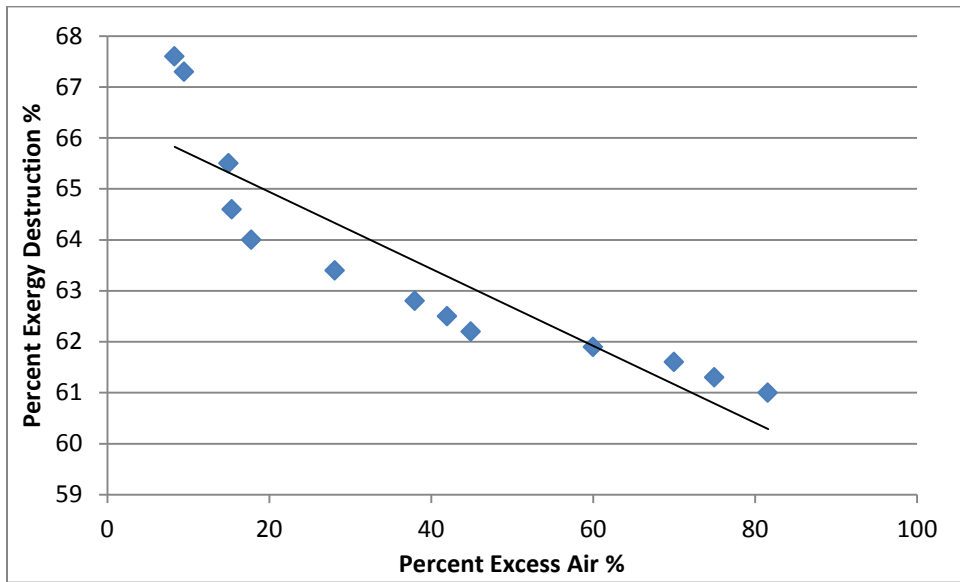


Fig. 10. variation of exergy destruction with variation of excess air at different level of fuel flow rate.

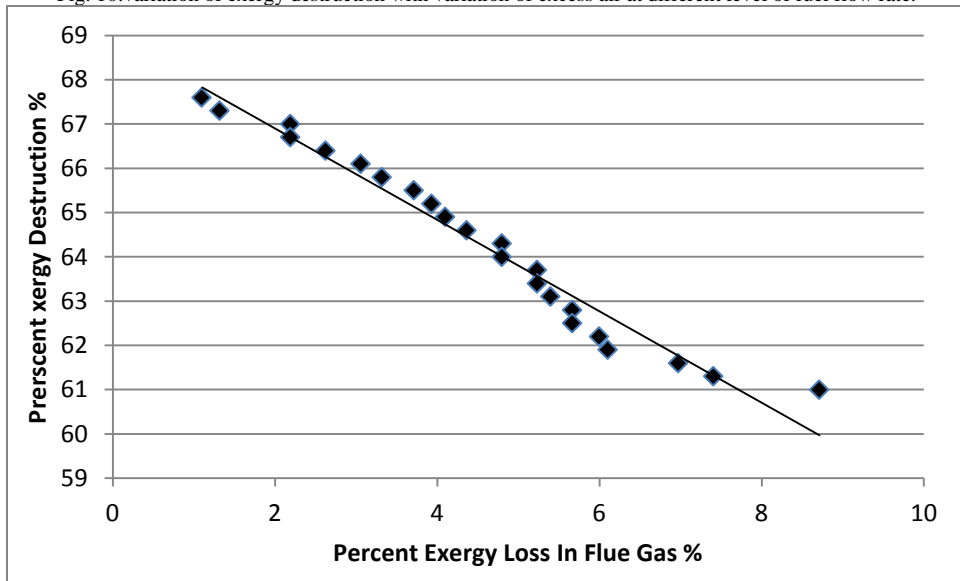


Fig. 11. Variation of exergy destruction with variation of Exergy Loss in Flue Gas at different level of fuel flow rate.

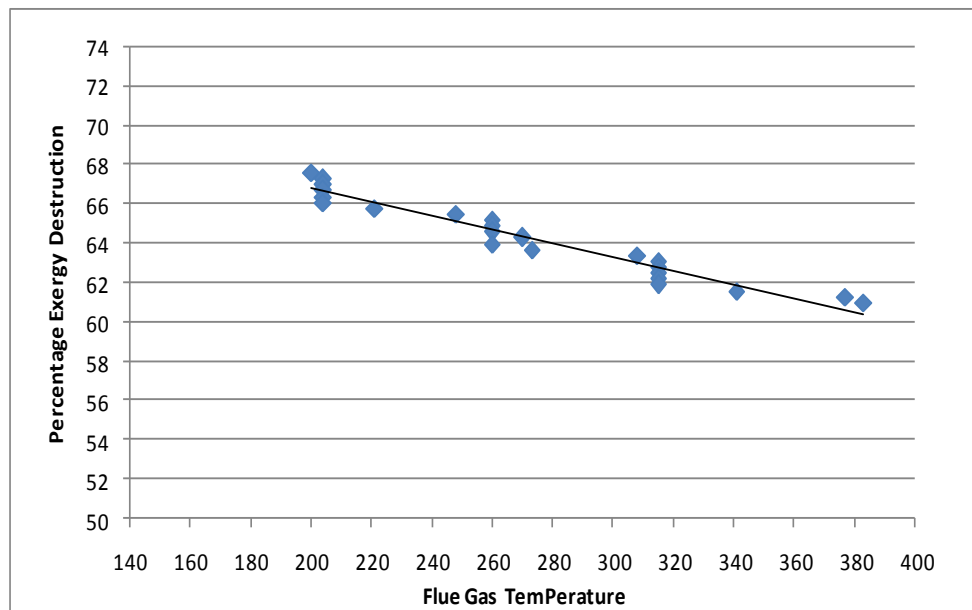


Fig. 12.The variation of Exergy Destruction with variation of flue gas Temperature.

CONCLUSION

Exergy means the real work output that can be extracted from a given form of energy. According to this analysis the work output of the thermal fluid is in a very low level compared to the work input. Therefore the Exergitec efficiency is very low comparing the energy efficiency in Direct Fired Heater. The Exergy destruction is in the range of 12 to 60 % in Direct Fired Heater. Figures above show Exergetic efficiency with flue gas temperature at different excess air levels.

The Exergy loss through the flue gas is in the range of 20 to 65 % in Direct Fired Heater, and it is in the range of 0.004 to 0.008 % for surface emission. Depending on the previous figures the average Exergetic efficiency becomes 33 % in Direct Fired Heater. According to this analysis the minimum possible Exergy losses in a Direct Fired Heater should be within the following limits:-

Percentage Exergy loss through flue gas - 20 %.Percentage Exergy loss through surface emission - 0.005 %.

With reference to these limits the percentage Exergy destruction is 60 % and the Exergetic efficiency is 33 %. This is the maximum possible Exergetic efficiency that can be taken by maintaining the optimum running condition.

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