9

Exergy, the Potential Work

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1. Introduction

The exergy method is an alternative, relatively new technique based on the concept of exergy, loosely defined as a universal measure of the work potential or quality of different forms of energy in relation to a given environment. An exergy balance applied to a process or a whole plant tells us how much of the usable work potential, or exergy, supplied as the input to the system under consideration has been consumed (irretrievably lost) by the process. The loss of exergy, or irreversibility, provides a generally applicable quantitative measure of process inefficiency. Analyzing a multi component plant indicates the total plant irreversibility distribution among the plant components, pinpointing those contributing most to overall plant inefficiency (Gorji-Bandpy&Ebrahimian, 2007; Gorji-Bandpy et al., 2011)

Unlike the traditional criteria of performance, the concept of irreversibility is firmly based on the two main laws of thermodynamics. The exergy balance for a control region, from which the irreversibility rate of a steady flow process can be calculated, can be derived by combining the steady flow energy equation (First Law) with the expression for the entropy production rate (Second Law).

Exergy analysis of the systems, which analyses the processes and functioning of systems, is based on the second law of thermodynamics. In this analysis, the efficiency of the second law which states the exact functionality of a system and depicts the irreversible factors which result in exergy loss and efficiency decrease, is mentioned. Therefore, solutions to reduce exergy loss will be identified for optimization of engineering installations (Ebadi&Gorji-Bandpy, 2005). Considering exergy as the amount of useful work which is brought about, as the system and the environment reach a balance due to irreversible process, we can say that the exergy efficiency is a criterion for the assessment of the systems. Because of the irreversibility of the heating processes, the resulting work is usually less than the maximum amount and by analyzing the work losses of the system, system problems are consequently defined. Grossman diagrams, in which any single flow is defined by its own exergy, are used to determine the flow exergy in the system (Bejan, 1988). The other famous flow exergy diagrams have been published by Keenan (1932), Reisttad (1972) and Thirumaleshwar (1979). The famous diagrams of air exergy were published by Moran (1982) and Brodianskii (1973). Brodianskii (1973), Kotas (1995) and Szargut et al. (1988) have used the exergy method for thermal, chemical and metallurgical analysis of plants. Analysis of the technical chains of processes and the life-cycle of a product were respectively done by Szargut et al. (1988) and Comelissen and Hirs (1999). The thermoeconomy field, or in other words, interference of economical affairs in analyzing exergy, has been studied by Bejan (1982).

In this paper, the cycle of a power plant and its details, with two kind fuels, natural gas and diesel, have been analysed at its maximum load and the two factors, losses and exergy efficiency which are the basic factors of systems under study have been analysed.

2. Methodology

When a system is thermodynamically studied, based on the first principle of thermodynamics, the amount of energy is constant during the transfer or exchange and also, based on the second principle of thermodynamics, the degree of energy is reduced and the potential for producing work is lessened. But none of the mentioned principles are able to determine the exact magnitude of work potential reduction, or in other words, to analyse the energy quality. For an open system which deals with some heat resources, the first and second principles are written as follows (Bejan, 1988):

$$\frac{dE}{dt} = \sum_{i=0}^{n} \dot{Q}_i - \dot{W} + \sum_{in} \dot{m}h^0 - \sum_{out} \dot{m}h^0 \tag{1}$$

$$\dot{S}_{gen} = \frac{dS}{dt} - \sum_{i=0}^{n} \frac{\dot{Q}_i}{T_i} - \sum_{in} \dot{m}s + \sum_{out} \dot{m}s \geq 0 \tag{2}$$

In the above equations, enthalpy, h°, is $h + (V^2/2) + gz$, T_0 is the surrounding temperature, E, internal energy, S ,entropy, and $\dot{W}$ and $\dot{Q}$ are the rates of work and heat transfer.
For increasing the work transfer rate $(\dot{W})$, consider the possibility of changes in design of system. Assumed that all the other interactions that are specified around the system $(\dot{Q}_1, \dot{Q}_2, ..., \dot{Q}_n$, inflows and outflows of enthalpy and entropy) are fixed by design and only $\dot{Q}_0$ floats in order to balance the changes in $\dot{W}$. If we eliminate $\dot{Q}_0$ from equations (1) and (2), we will have (Bejan, 1988):

$$\dot{W} = -\frac{d}{dt}(E - T_0 S) + \sum_{i=0}^{n} \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i + \sum_{in} \dot{m}(h^0 - T_0 s) - \sum_{out} \dot{m}(h^0 - T_0 s) - T_0 \dot{S}_{gen} \tag{3}$$

When the process is reversible $(\dot{S}_{gen} = 0)$, the rate of work transfer will be maximum and therefore we will have:

$$\dot{W} = \dot{W}_{rev} - T_0 \dot{S}_{gen} \tag{4}$$

Combination of the two principles results in the conclusion that whenever a system functions irreversibly, the work will be eliminated at a rate relative to the one of the entropy. The eliminated work caused by thermodynamic irreversibility, $(\dot{W}_{rev} - \dot{W})$ is called “the exergy lost”. The ratio of the exergy lost to the entropy production, or the ratio of their rates results in the principle of lost work:

$$\dot{W}_{lost} = T_0 \dot{S}_{gen} \tag{5}$$

Since exergy is the useful work which derived from a material or energy flow, the exergy of work transfer, $\dot{E}_w$, would be given as (Bejan, 1988):

$$\dot{E}_w = \dot{W} - P_0\frac{dV}{dt} = -\frac{d}{dt}\left(E + P_0V - T_0S\right) + \sum_{i=1}^{n}\left(1 - \frac{T_0}{T_i}\right)\dot{Q}_{in} + \sum_{in}\dot{m}\left(h^0 - T_0s\right) - \sum_{out}\dot{m}\left(h^0 - T_0s\right) - T_o\dot{S}_{gen} \quad (6)$$

In most of the systems with incoming and outgoing flows which are considered of great importance, there is no atmospheric work, $(P_0(dV/dt))$ and $\dot{W}$ is equal to $\dot{E}_w$ (Bejan, 1988):

$$\left(\dot{E}_w\right)_{rev} = \dot{W}_{rev} - P_0\frac{dV}{dt} = -\frac{d}{dt}\left(E + P_0V - T_0S\right) + \sum_{i=1}^{n}\left(1 - \frac{T_0}{T_i}\right)\dot{Q}_{in} + \sum_{in}\dot{m}\left(h^0 - T_0s\right) - \sum_{out}\dot{m}\left(h^0 - T_0s\right) \quad (7)$$

The exergy lost, which was previously defined as the difference between the maximum rate of work transfer and rate of the real work transfer, can also be mentioned in another way, namely, the difference between the corresponding parameters and the available work (Figure 1):

$$\dot{W}_{lost} = \left(\dot{E}_w\right)_{rev} - \dot{E}_w = \left(\dot{E}_w\right)_{lost} \quad (8)$$

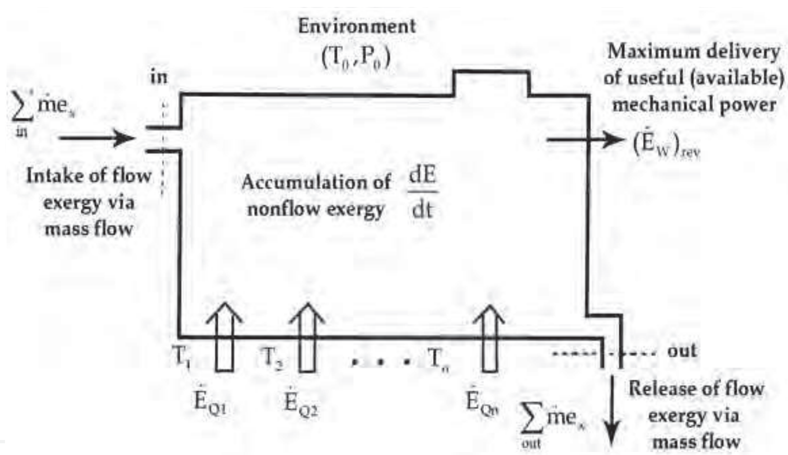

Fig. 1. Exergy transfer via heat transfer

In equation (6), the exergy transfer caused by heat transfer or simply speaking, the heat transfer exergy will be:

$$\dot{E}_Q = \dot{Q}\left(1 - \frac{T_0}{T}\right) \quad (9)$$

Using equation (1), the flow availability will be introduces as:

$$b = h^0 - T_0s \quad (10)$$

In installation analysis which functions uniformly, the properties do not changes with time and the stagnation exergy term will be zero, in equation (6):

$$\dot{E}_w = \sum_{i=0}^{n}\left(\dot{E}_Q\right)_i + \sum_{in}\dot{m}b - \sum_{out}\dot{m}b - T_0\dot{S}_{gen} \tag{11}$$

The flow exergy of any fluid is defined as:

$$e_x = b - b_0 = h^0 - h_0^0 - T_0(s - s_0) \tag{12}$$

Substituting this definition into equation (11), we will have:

$$\dot{E}_w = \sum_{i=0}^{n}\left(\dot{E}_Q\right)_i + \sum_{in}\dot{m}e_x - \sum_{out}\dot{m}e_x - T_0\dot{S}_{gen} \tag{13}$$

The flow exergy (e_x) is the difference between the availabilities of a flow (b), in a specific condition and in the restricted dead state (in balance with the environment). Equation (13) is used to balance the exergy of uniform flow systems. The mechanisms which lead to the production of entropy and the elimination of exergy are listed as follows:

- heat transfer caused by limited temperature difference (Bejan, 1988):

$$\dot{S}_{gen} = \frac{\dot{Q}}{T_H T_L}(T_H - T_L) \tag{14}$$

- frictional flow (Reistad, 1972):

$$\dot{S}_{gen} = \dot{m}\int_{out}^{in}\left(\frac{v}{T}\right)_{h=const.} dp \tag{15}$$

- combining (Stepanov, 1955):

$$\frac{1}{\dot{m}_3}\dot{S}_{gen} \cong x\left[\frac{1}{T}(h_3 - h_1) - \frac{v}{T}(P_3 - P_1)\right] + (1 - x)\left[\frac{1}{T}(h_3 - h_2) - \frac{v}{T}(P_3 - P_2)\right] \tag{16}$$

The efficiency of the second law that determines used exergy is divided into two groups: Elements efficiency (Pump and Turbine) and Cycle efficiency (Thermal efficiency and coefficient of performance). The definition of the efficiency of the second law is (Wark, 1955):

$$\eta_{II} = \frac{\text{availability of useful outgoing}}{\text{incoming availability}} \tag{17}$$

The definition of the efficiency of the second law is more practical for the uniform flow systems, and is determined as follows (Bejan, 1988):

$$\eta_{II} = \frac{\text{outgoing exergy rate}}{\text{incoming exergy rate}} \tag{18}$$

The second law emphasizes the fact that two features of the same concept of energy may have completely different exergies. Therefore, any feature of energy is defined by taking into account its own exergy. The efficiency of the second law will be used in calculating the reduction of ability in performing a certain amount of work.

3. Case study

In order to analyse the above theories, the consequences have been analysed on the Shahid Rajaii power plant in Qazvin of Iran. This power plant has an installed capacity of 1000 MW electrical energy, which consists of four 250 MW steam-cycle units (Rankin cycle with reheating and recycling) and has been working since 1994. The major fuel for the plant is the natural gas and is augmented with diesel fuel.

The Shahid Rajaii power plant consists of three turbines: high pressure, medium pressure, and low pressure. The 11-stage high pressure turbine has Curtis stage. The number of the stages in the medium pressure turbine is 11 reactionary stages and in the lower pressure, 2×5 reactionary stages. All of the turbines have a common shaft with a speed of 3000 rpm. The boiler is a natural circulation type in which there is a drum with no top. Other properties of the boiler are that the super-heater is three-staged and that the reheater and the economizer are both two-staged. Figure 2 illustrates the plant diagram overlooking the boiler furnace, cooling towers, attachment (circulation and discharge pumps, blowers, etc), turbine glands condenser and regulator lands, expansion valves and governors and feed water tanks.

In Table 1, properties of water and vapour in the main parts of the cycle have been shown.

Maximum losses of cycle water in this plant are 5 kg/hr, which is negligible due to the minute amount. In analyzing the cycle and drawing diagrams, it is assumed that the temperature is $30^{\circ}C$, pressure is $90\,kPa$ and relative humidity is 30% as environmental conditions. Other assumptions are:

- kinetic and potential energies are neglected because they are not so important
- all elements of the cycle are considered to be adiabatic
- in this part, the combustion process of the boiler has been ignored.

With Figure 2, the conversion equation and energy balance of boiler will be written as (Jordan, 1997):

$$\dot{m}_1 = \dot{m}_2 \text{ and } \dot{m}_3 = \dot{m}_4 \tag{19}$$

Description	$P\left(kg/cm^2\ abs\right)$	$T\left(^{\circ}C\right)$
Feed water incoming to boiler	150	247-202
Vapour incoming to HP turbine	140	838
Vapour incoming to reheater	17-40	358-287
Vapour incoming to IP turbine	15.2-37.3	538
Vapour incoming to LP turbine	8-3.5	320
Vapour incoming to condenser	0.241-0.960	64-45
Water outgoing from condenser pump	16-7	63-44

Table 1. Properties of water and vapour in cycle

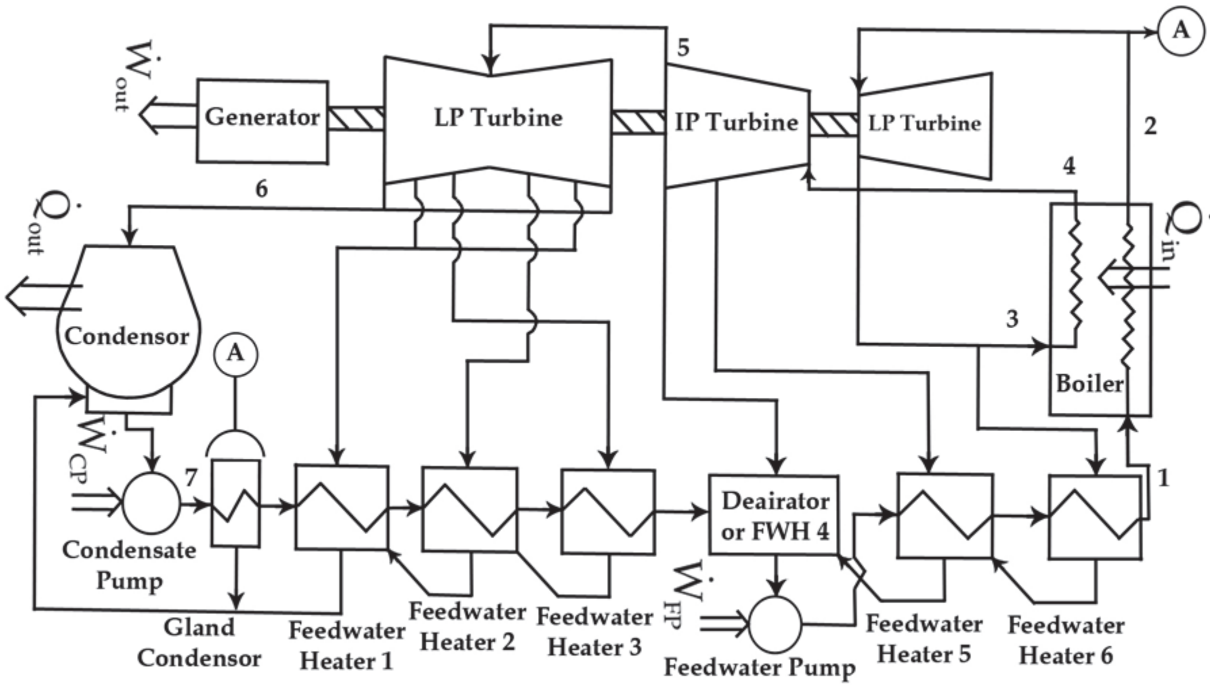

Fig. 2. The diagram of flow cycle of the plant

$$\dot{Q}_{in} = \dot{m}_1 (h_2 - h_1) + \dot{m}_4 (h_4 - h_3) \tag{20}$$

Energy efficiency is written as (Jordan, 1997):

$$\eta_I = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{21}$$

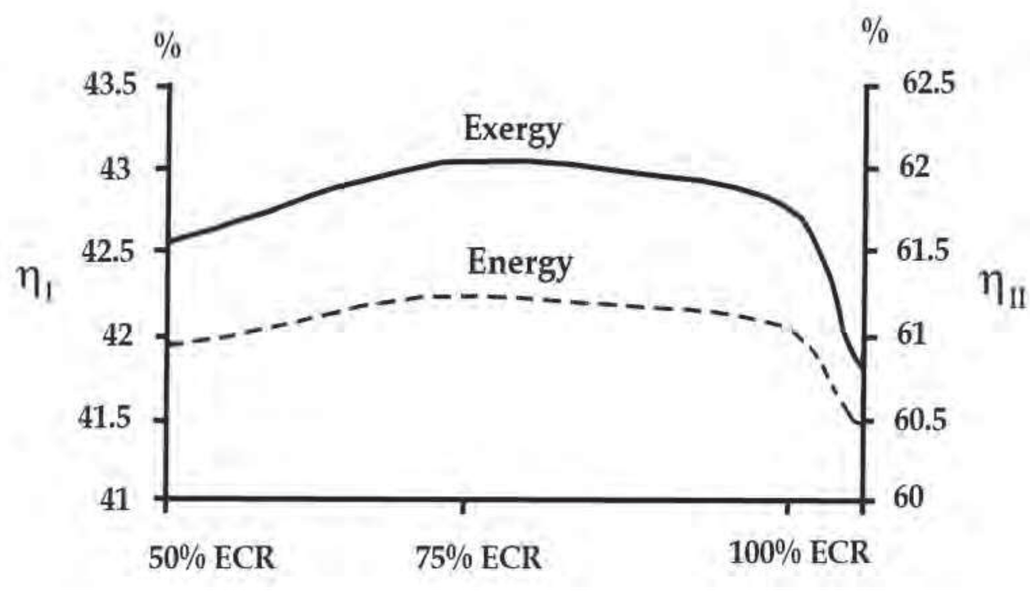

Fig. 3. The diagram of Qazvin power plant efficiency under different loads (natural gas)

If we use the lost work law for the closed cycle:

$$\dot{W}_{net} = \dot{E}_{Qin} - \dot{E}_{w} \tag{22}$$

$$\left(\dot{E}_{w}\right)_{lost} = \dot{E}_{Qin} - \dot{W}_{net} = T_0 \dot{S}_{gen} \tag{23}$$

Exergy efficiency of plant is:

$$\eta_{II} = \frac{\dot{E}_{w}}{\dot{E}_{Qin}} = \frac{\dot{W}_{net}}{\dot{E}_{Qin}} \tag{24}$$

Exerting energy and exergy balance equations for the plant cycle, and calculating the energy and efficiencies, Figure 3 consequently results.
As can be seen in Figure 3, under the maximum load, the exergy efficiency is 60.78% and the energy efficiency is 41.38%, relative to different minimum loads. Therefore, boiler analysis is done at maximum load.

3.1 Analysing the different elements of the cycle

Using energy balance, which is the basis of exergy balance, and implementing equations (9) and (13) and assuming the warm source temperature to be 950 K, the results of exergy lost and efficiency of all components of the plant cycle are shown in Table 5.

3.2 Energy and exergy efficiencies of the plant

In part one, the power plant efficiency has been calculated, overlooking the boiler combustion process and losses under different loads and Figure 3 was therefore mapped out. In order to more accurately calculate the efficiencies, it is necessary to consider the combustion process. The energy efficiency of the plan is the amount of produced net work divided by the fuel energy. In Table 2, percentage of mass for both Natural gas and Diesel fuel in this study has been shown.

Element	Natural gas	Diesel fuel
C	75.624	85
H	23.225	12
O	0	0.4
N	0.206	0.2
S	0	2.4
Ash	0	0
Moisure	0	0
Co_2	0.945	0

Table 2. Percentage of mass for both natural gas and diesel fuel

$$\eta_{I} = \frac{\dot{W}_{net}}{\dot{m}_f \times LHV} \tag{25}$$

where LHV is the fuel low heating value. The exergy efficiency of the plant is the amount of outgoing exergy (produced net work) divided by the fuel exergy.

$$\eta_{II} = \frac{\dot{E}_w}{\dot{m}_f e_{ch.f}} \tag{26}$$

Where, $e_{ch.f}$ is the specific chemical exergy of the fuel.
The natural gas and diesel fuel consumption are respectively 50010 kg/hr and 59130 kg/hr under the maximum load. The low heating values of the natural gas and diesel fuel are 41597 kj/kg and 48588 kj/kg. Assuming the natural gas as a perfect gas and using the tables of standard chemical exergy, the chemical exergy of the natural gas is calculated as (Szargut et al., 1988):

$$e^0_{ch.NG} = \sum Y_i e^0_{ch.i} = 50403 \frac{kj}{kg} \tag{27}$$

And implementing the Szargut method, the chemical exergy of diesel fuel is calculated to be 45540 kj/kg (Szargut et al., 1988):

$$\begin{aligned} e_{ch.oil} &= \left(LHV + \dot{m}_{in,f} h_{fg}\right) \\ &\times \left[1.0401 + 0.1728\frac{H}{C} + 0.0432\frac{O}{C} + 0.2196\frac{S}{C}\left(1 - 2.0628\frac{H}{C}\right)\right] \end{aligned} \tag{28}$$

Where LHV is the fuel's low heating value, h_{fg} and $m_{m,f}$ are the vaporization temperature of the hot water and the mass of the moisture content and S/C, H/C and O/C are the mass ratio of sulphur, hydrogen and oxygen, to carbon, respectively. The electric power needed for the attachments of the boiler such as fans and pumps is 3.83 MW or 4.28 MW for natural gas and diesel fuels. Feed water pumps and the condenser and other helping elements of the plant also respectively use 9.926 MW and 70.06 MW of the electrical energy. So the produced net work will be:

$$\dot{W}_{net,NG} = 263.53 - 6.926 - 3.83 = 252.774\,MW \tag{29a}$$

$$\dot{W}_{net,oil} = 261.95 - 7.06 - 4.28 = 250.61\,MW \tag{29b}$$

The heating and exergy efficiencies of the plant using the two fuels will be:

$$\eta_{I,NG} = \frac{252.774}{(50010/3600)\times 11605(4.1868)} = 37.45\% \tag{30a}$$

$$\eta_{I,oil} = \frac{250.610}{(59130/3600)\times 41597} = 36.68\% \tag{30b}$$

$$\eta_{II,NG} = \frac{252.774}{(50010/3600)\times 50403} = 36.10\% \tag{31a}$$

$$\eta_{II,oil} = \frac{250.610}{(59130/3600)\times 45540} = 33.50\% \tag{31b}$$

As is obviously seen, the heating efficiency of the power plant changes from 36.68% to 37.45% and the exergy efficiency from 33.5% to 36.1%, when natural gas is replaced by diesel fuel. Therefore the exergy efficiency change is greater than that of energy efficiency.

3.3 Analysing the boiler

The boiler of this plant is designed based on the natural circulation, and high pressure cold water flow furnace and the water pipes have been appointed vertically. The design pressure of the boiler is 172 kg/cm², the design pressure of the reheating system is 46 kg/cm² and the capacity is 840 ton/hrs. Two centrifugal fans (forced draught fan (FDF)) provide the needed air for the combustion.

The boiler is modeled for thermodynamic analysis. The air and gas fans, discharge pumps, and generally, the utilities which are work consuming are not considered in the model. Their effect is the total work which enters the control volume $(\dot{W}_{m,B})$. Also the heaters and the de-super-heaters within the control volume have been ignored. The heat losses to the surrounding environment are introduced as $\dot{Q}_{out.B}$. The energy and exergy balance of the boiler referring to the equations (1) and (13) are written as:

$$\dot{m}_f h_f + \dot{W}_{\dot{m},B} + \dot{m}_a h_a - \dot{m}_g h_g + \sum_{in} \dot{m}_w h_w - \sum_{out} \dot{m}_w h_w - \dot{Q}_{out,\mathbf{B}} = 0 \tag{32}$$

$$\dot{m}_f e_{ch,f} + \dot{W}_{\dot{m},B} + \dot{m}_a e_{x,g} - \dot{m}_g e_{x,g} + \sum_{in} \dot{m}_w e_{x,w} - \sum_{out} \dot{m}_w e_{x,w} - T_0 \dot{S}_{gen,B} = 0 \tag{33}$$

Indices a, g and respectively used for air, gas (combustion products) and water (vapour).
The energy and exergy efficiencies of the boiler are defined as:
In Table 3, thermodynamic properties of water and vapour for both Natural gas and Diesel fuel have been summarized.

fuel	property	Feed water	Super heated vapour	Hot reheated vapour	Cold reheated vapour
Natural Gas	$T(^\circ C)$	247.7	538	358.1	538
	$P(kPa)$	15640	13730	3820	3660
	$h(kj/kg)$	1075.6	3430.2	3116.2	3534.9
	$e_x(kj/kg)$	241.9	1439.3	1109.8	1345.9
	$\dot{m}(kg/h)$	840000	840000	751210	751210
Diesel Fuel	$T(^\circ C)$	247.4	538	357.1	538
	$P(kPa)$	15640	13730	3800	3640
	$h(kj/kg)$	1074.7	3430.2	3115.5	3535.0
	$e_x(kj/kg)$	241.8	1439.0	1109.1	1346.0
	$\dot{m}(kg/h)$	840000	840000	747010	747010

Table 3. Thermodynamic properties of incoming water and outgoing vapour for the boiler at maximum load

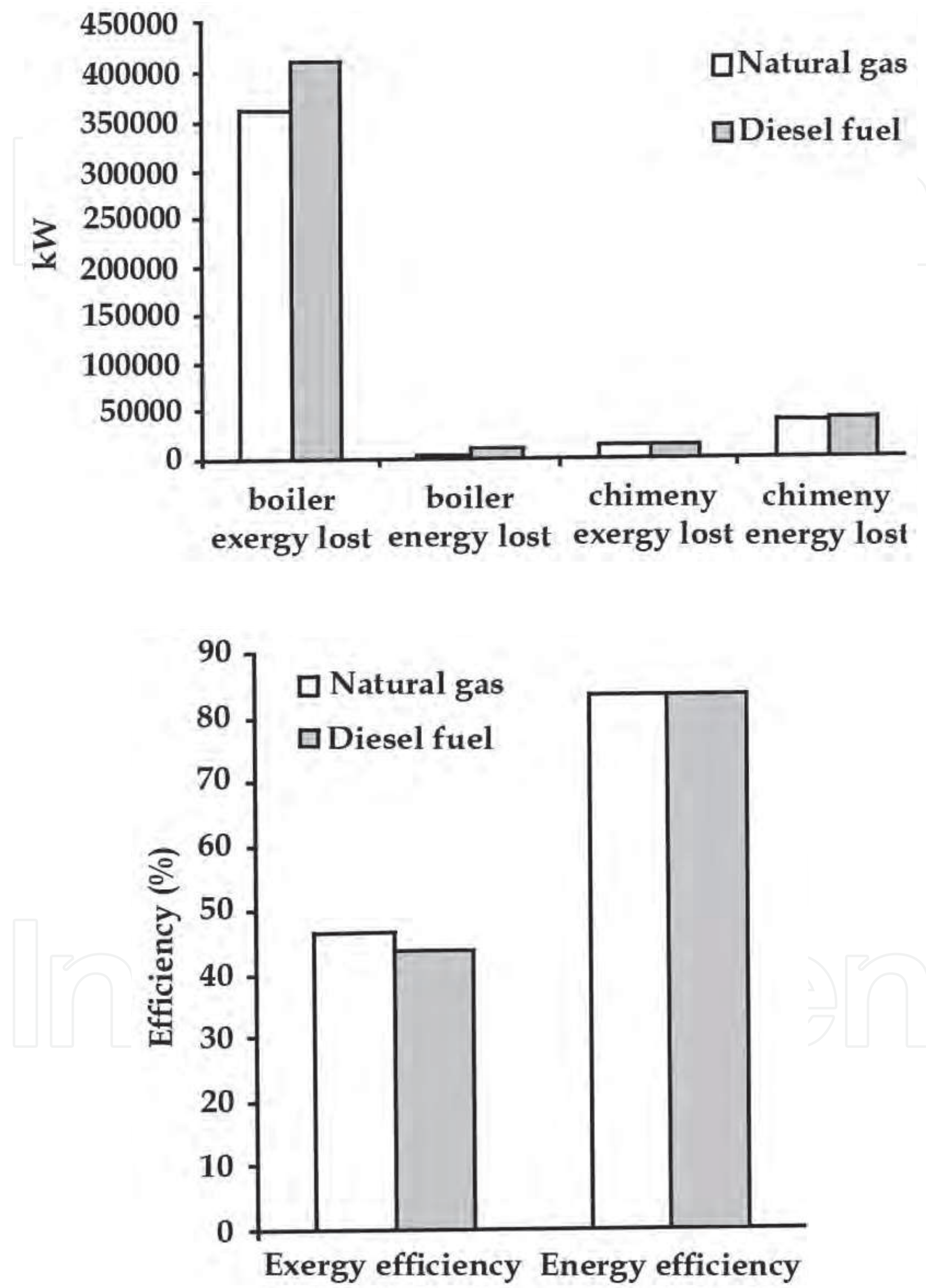

Fig. 4. The diagram of boiler exergy losses and efficiency

$$\eta_{I,B} = \frac{\sum_{out}\dot{m}_w h_w - \sum_{in}\dot{m}_w h_w}{\dot{m}_f LHV + \dot{m}_{dry\,air} h_a + \dot{w}_{in,B}} \tag{34}$$

$$\eta_{II,B} = \frac{\sum_{out}\dot{m}_w e_{x,w} - \sum_{in}\dot{m}_w e_{x,w}}{\dot{m}_f e_{ch,f} + \dot{w}_{in,B} + \dot{m}_{dry\,air} e_{x,a}} \tag{35}$$

In Table 4, the calculations of the enthalpy and exergy of the dry combustion gases, vapour and wet combustion gases have been summarized. Combustion gases do not have CO. Since the chemical exergy of CO is high, the exergy of the combustion products is negligible.

Fuel	Natural gas ($M_{dry\ gas}$ = 29.77)		Diesel fuel ($M_{dry\ gas}$ = 30.36)	
	Enthalpy	Exergy	Enthalpy	Exergy
Dry combustion gases	384.8	10.15	409.0	17.42
Vapour	740.4	314.30	801.6	417.19
Wet combustion gases	474.3	48.15	472.3	50.34

Table 4. Thermodynamic properties of dry and wet combustion gases and vapour outgoing from pre-heater

The result of calculating the last four equations is briefly shown in Figure 4.

3.4 The boiler processes

Two important processes happen in the boiler: Combustion and heat transfer. Therefore the internal exergy losses of the boiler $(T_0\dot{S}_{gen,B})$ are the losses of both exergy and heat transfer. Of course, there is a small exergy losses caused by friction which is calculated in the exergy caused by heat transfer. In the boiler, the exergy losses caused by friction have two different reasons; one is the pressure losses of the combusting gases (at most 4.4 kPa) which is to be neglected and the other is the pressure losses of the actuating fluid. These kinds of exergy losses, as was previously mentioned in the first part, are negligible compared to the losses due to heat transfer. Thus, there will be no specific analysis of friction; these two kinds of exergy losses together called the exergy losses due to heat transfer.

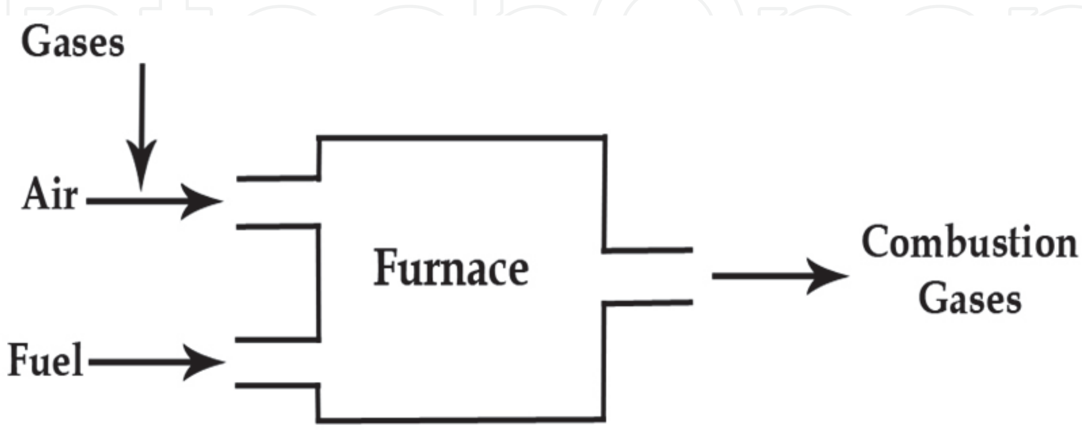

Fig. 5. The model of the boiler furnace

3.4.1 Combustion

To study the furnace, we assume that the combustion is in an isolated control volume. According to Figure 5, using the energy balance, we will get the enthalpy of the combustion gases:

$$\dot{Q}_{out,\,comb} = \dot{m}_f h_f + \dot{m}_a h_a + \dot{m}_r h_r - \dot{m}_p h_{p,\,ad} = 0 \tag{36}$$

By determining the temperature of the combustion products, using the iteration method, we can write the exergy balance equation of the furnace in order to determine the chemical exergy losses of the furnace.

$$T_0 \dot{S}_{gen,\,comb} = \dot{m}_f e_{ch,\,f} + \dot{m}_a e_{x,\,a} + \dot{m}_r e_{x,\,r} - \dot{m}_p e_{x,\,p,\,ad} \tag{37}$$

Therefore the combustion exergy efficiency will be:

$$\eta_{II,\,comb} = \frac{\dot{m}_p e_{x,\,p,\,ad}}{\dot{m}_f e_{ch,\,f} + \dot{m}_r e_{x,\,r} + \dot{m}_a e_{x,\,a}} \tag{38}$$

3.4.2 Heat transfer

In the boilers, heat transfer to the actuating fluid is classified into four categories:

- heat transfer in the pipes of the first and secondary economizers
- heat transfer in the pipes of the furnace walls
- heat transfer in the first, secondary and final super-heaters
- heat transfer in the first and secondary reheaters

Exergy losses caused by heat transfer occur in five main parts of the boiler; the evaporator, the economizer, the super-heater, the reheaer and the air preheater. Exergy losses of the main five elements together with the furnace losses are the total losses of the boiler which were mentioned in Section 3.3.

$$T_0 \dot{S}_{gen,\,Q} = T_0 \dot{S}_{gen,\,B} - T_0 \dot{S}_{gen,\,comb} \tag{39}$$

3.5 The boiler elements

Each element of the boiler is, in fact, a heat exchanger. Therefore, when there is more one inlet or outlet to the heat exchanger, exergy balance is written as follows:

$$T_0 \dot{S}_{gen,\,element} = \sum_{in} \dot{m}_g e_{x,\,g} - \sum_{out} \dot{m}_g e_{x,\,g} + \dot{m}_w \left(e_{x,\,w,\,in} - e_{x,\,w,\,out} \right) \tag{40}$$

Therefore the heat transfer exergy efficiency will be:

$$\eta_{II,\,element} = \frac{\dot{m}_w \left(e_{x,\,w,\,out} - e_{x,\,w,\,in} \right)}{\sum_{in} \dot{m}_g e_{x,\,g} - \sum_{out} \dot{m}_g e_{x,\,g}} \tag{41}$$

The results of exergy losses and efficiency of each element of the boiler are shown in Figure 6.

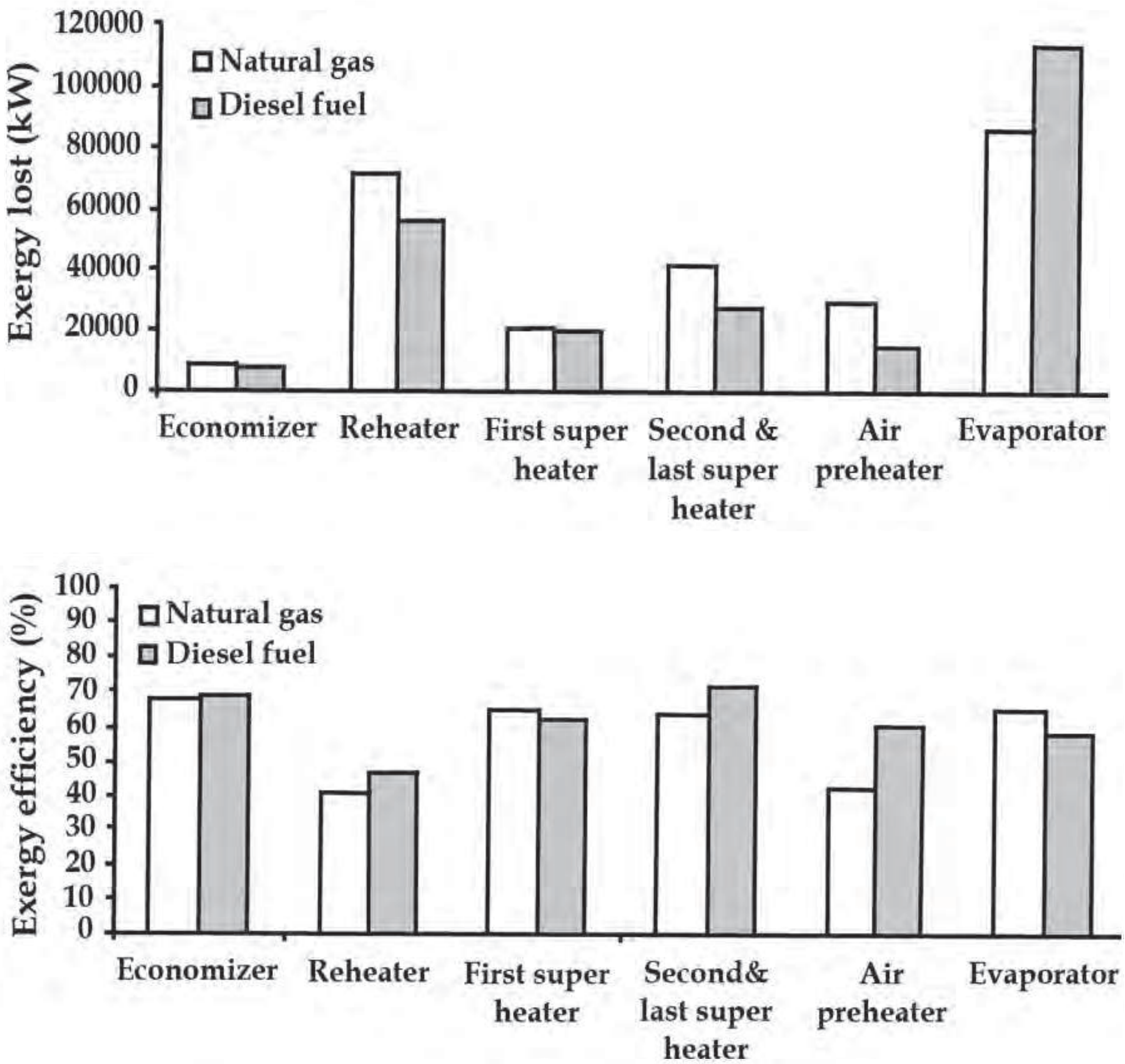

Fig. 6. The diagram of the exergy loss and efficiency

3.6 The correction factor of the boiler efficiency

All the measured data such as temperature, pressure, flow, etc and also the calculated magnitude of the exergy efficiency and losses were assumed under some specific environmental conditions. But the plant is not always under these specific conditions, and in fact there are some conditions under which the efficiency changes. These are divided into two parts: internal conditions such as excess air and moisture content and the external (environmental) conditions such as temperature and humidity.

An increase in the amount of air in the combustion process can easily decrease the adiabatic temperature of the flame so that the adiabatic exergy of the products is reduced and the exergy losses of the combustion increase. On the other hand, the combustion products have a lower temperature and greater mass flow as they flow inside the boiler, which leads to lower exergy losses in the heart transfer process.

The correction factor for the boiler exergy efficiency caused by the internal and external conditions is done using the equation (35). For instance, this factor due to the excess air is:

$$\Delta\eta_{II,B} = \left.\frac{\sum_{out}\dot{m}_w e_{x,w} - \sum_{in}\dot{m}_w e_{x,w}}{\dot{m}_f e_{x,f} + \dot{m}_a e_{x,a} + \dot{W}_{in,B}}\right|_{EAR}^{x} \tag{42}$$

Where EAR (Excess Air Ratio) is the ratio of the excess air to the time of the boiler test.

4. Results and discussions

As is shown in Table 5, the lowest efficiency belongs to the gland condenser which is 28.9%. Its exergy loss at 46 kW is among the least. The highest exergy efficiency belongs to the high pressure and medium pressure turbines, and is 94.95%. In these turbines, 8617 kW of the 616505 kW of incoming exergy disappears and the rest is changed to work. The low pressure turbine has more exergy losses at 15303 kW and its efficiency is 87.47%, which is less than the high pressure turbine. The plant boiler, which is one of the most important elements of the cycle, has one of the lowest exergy efficiencies, 46.94%, which should be optimized.

Cycle elements	$\left(\dot{E}_w\right)_{lost}(kW)$	η_{II} (%)
Boiler	371522	46.94
Low Pressure Turbine	15303	87.47
Condenser	12867	60.05
High Pressure Turbine	8617	94.95
Generator	5222	98.06
Feed water pump	2147	60.00
Heater 5	2026	86.04
Heater 6	1563	91.42
Heater 4	1555	85.38
Heater 3	456	88.71
Heater 2	730	81.72
Heater 1	449	81.29
Feed water motor pump	313	94.50
Main drain	120	31.94
Condenser pump	62	64.57
Gland condenser	46	28.90
Condenser motor pump	15	92.00
Total	423013	-

Table 5. Exergy losses and exergy efficiency of the cycle elements of the plant

In Figure 4, exergy analysis shows that the major part of the exergy losses are because of the internal irreversibility of the boiler, while the chimney losses are less than one thirtieth of the boiler losses. Conversely, energy analysis shows that chimney losses are four times greater than boiler losses (heat transfer of the boiler to the surrounding area) and this is the major cause of efficiency reduction. Therefore, in order to reduce the energy losses, chimney losses are to be decreased, whereas the reduction of the internal exergy losses is more effective in the increase of exergy efficiency.

Figure 4 proves that energy efficiency is not dependent on the fuel type. The difference between the efficiencies of the two different fuels is 0.22% which is very minute. But the exergy efficiencies with two fuels are 3.02% different, based on Figure 4.

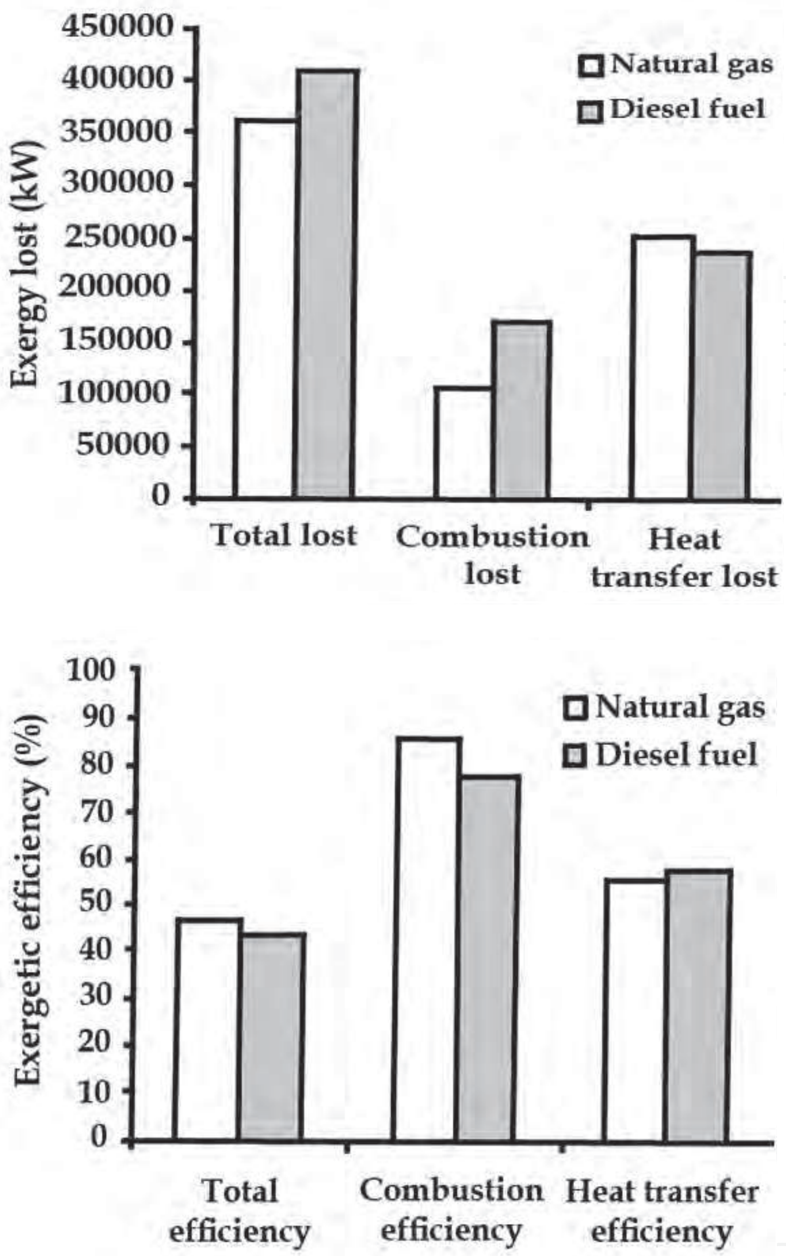

Fig. 7. The diagram of the exergy losses and exergetic efficiency of the different processes

Figure 7 demonstrates the comparison of the exergy losses and efficiencies of the different processes of the boiler.

The important result is that the exergy losses in the heat transfer process are greater than that in the combustion process. In other words, the heat transfer process is more irreversible. Exergy efficiency, as an evaluating standard for exergy losses, shows that the heat transfer process is more inconvenient than the combustion process. This result is generally true for all the plant boilers. In order to optimize the boilers, engineers should focus on heat transfer processes, optimization of the heat exchangers and increasing their exergy efficiencies. One of the methods to achieve this aim is Pinch which helps us in the exergy analysis and in determining the arrangement of the pipes in order to reduce the irreversibility of the heat transfer. When using diesel fuel, exergy losses in the combustion process are greater than

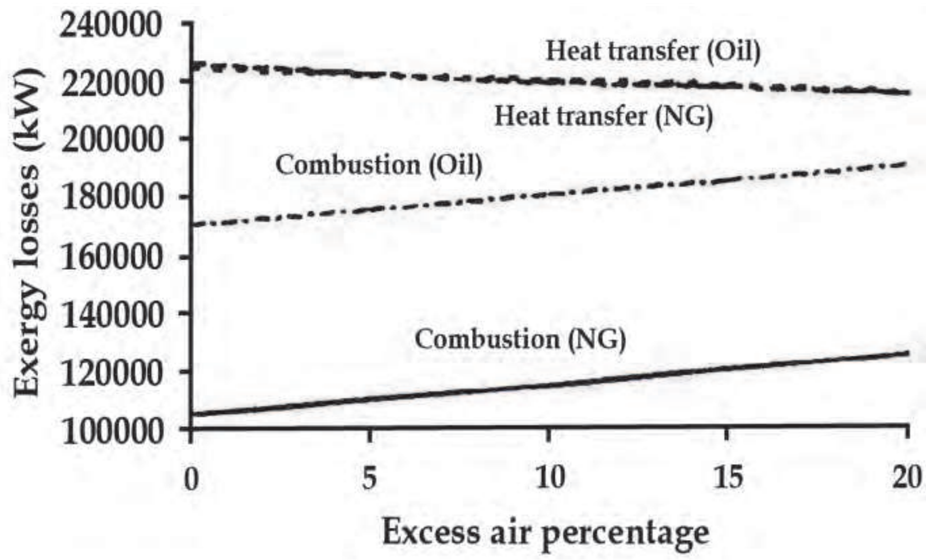

Fig. 8. The diagram of the internal exergy loss relative to the excess air percentage

In Figure 9, as the excess air percentage increases, the exergy efficiency of the boiler is reduced.

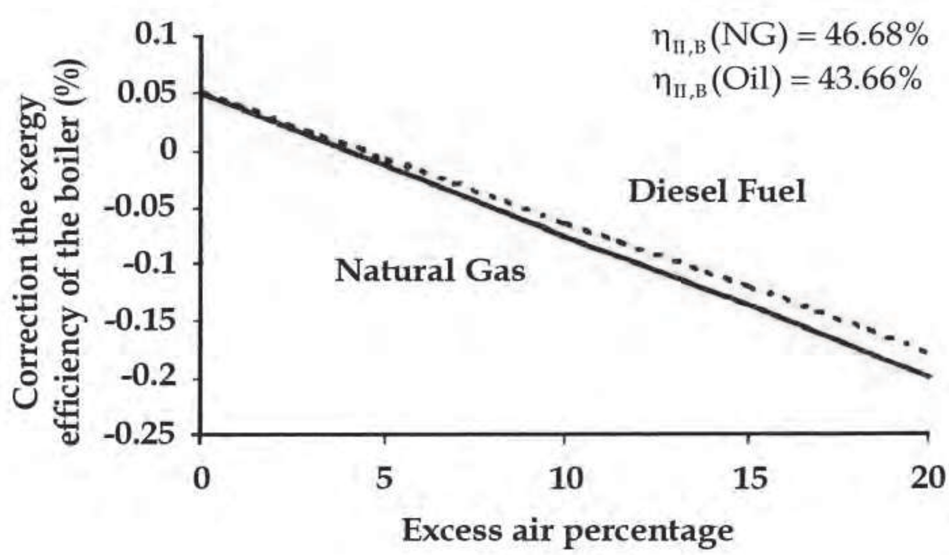

Fig. 9. The diagram of boiler efficiency based on the excess air percentage

when we use natural gas. But his is not the case for heat transfer. It is because the flow of combustion products is more when we use diesel fuel than when we use natural gas. On the other hand, the increase in the flow of combustion products is because of the great amount of incoming air to the furnace which leads to reduced exergy efficiency.In Figure 6, when we use natural gas, the exergy losses are greater in all elements than when we use diesel fuel, except the evaporator. The exergy efficiency, using diesel fuel, is less than when we use natural gas. The main reason behind the difference between the evaporator function and the other elements is that the combustion and heat transfer processes happen simultaneously in

the evaporator. The secondary and the final super-heaters have the highest exergy efficiency (71.67%) when using diesel fuel, with the exergy losses of 27657 kW. The second one is the evaporator using the natural gas, with exergy efficiency of 67.1% and exergy losses of 58800 kW. The reheater has the lowest efficiency using the natural gas (40.99%), with exergy losses of 70933 kW. The evaporator using diesel fuel has the highest exergy loss of 114071 kW and the economizer using diesel fuel has the lowest exergy losses of 7486 kW.
In Figure 8, as the excess air percentage becomes higher, the exergy losses of the natural gas increase at a lower rate. This reduction in the exergy losses of heat transfer based on the excess air percentage has a higher rate with natural gas than with diesel fuel. Generally, the gradients of exergy losses in the diagrams on combustion are more than in heat transfer.
In Figure 10, as the mass percentage of fuel humidity increases, the exergy efficiency of the boiler is reduced.

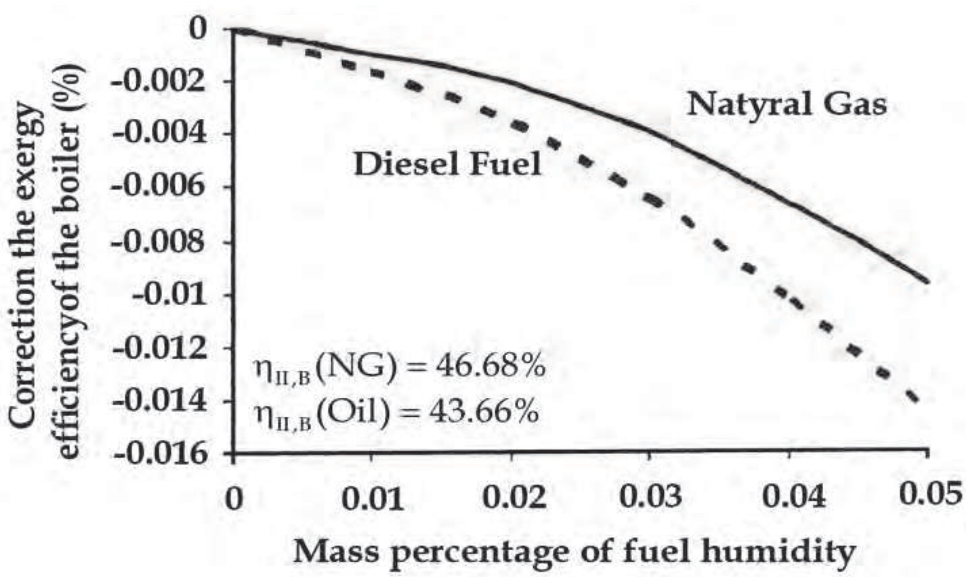

Fig. 10. The diagram of boiler efficiency changes based on the moisture content percentage

In Figure 10, humidity in diesel fuel causes more reduction in the boiler efficiency than in the natural gas. According to this Figure, it can be said that moisture content does not have a considerable effect upon the efficiency.
According to the Figure 11, we can say that as the relative air humidity percentage becomes higher, the boiler efficiency is reduced and the exergy efficiency reduction rate is more with diesel fuel than the natural gas.

Figure 12 shows that an increase in the air temperature from the assumed temperature of the environment $(30^{\circ}C)$ decreases the exergy efficiency, and under the other conditions, decrease or increase of the exergy efficiency is greater with the natural gas.

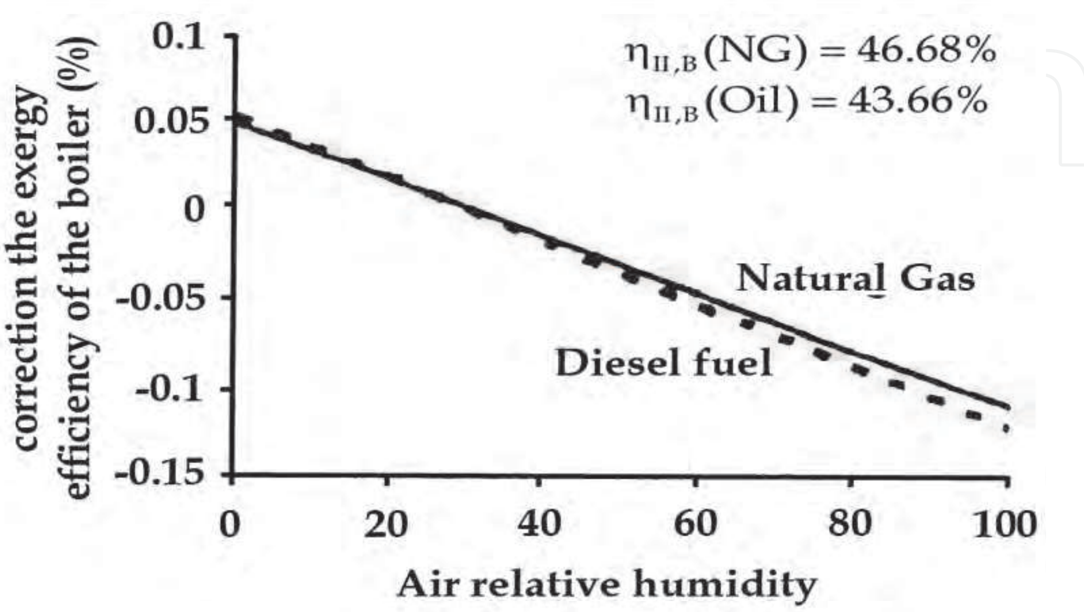

Fig. 11. The diagram of boiler efficiency changes based on the air relative humidity percentage

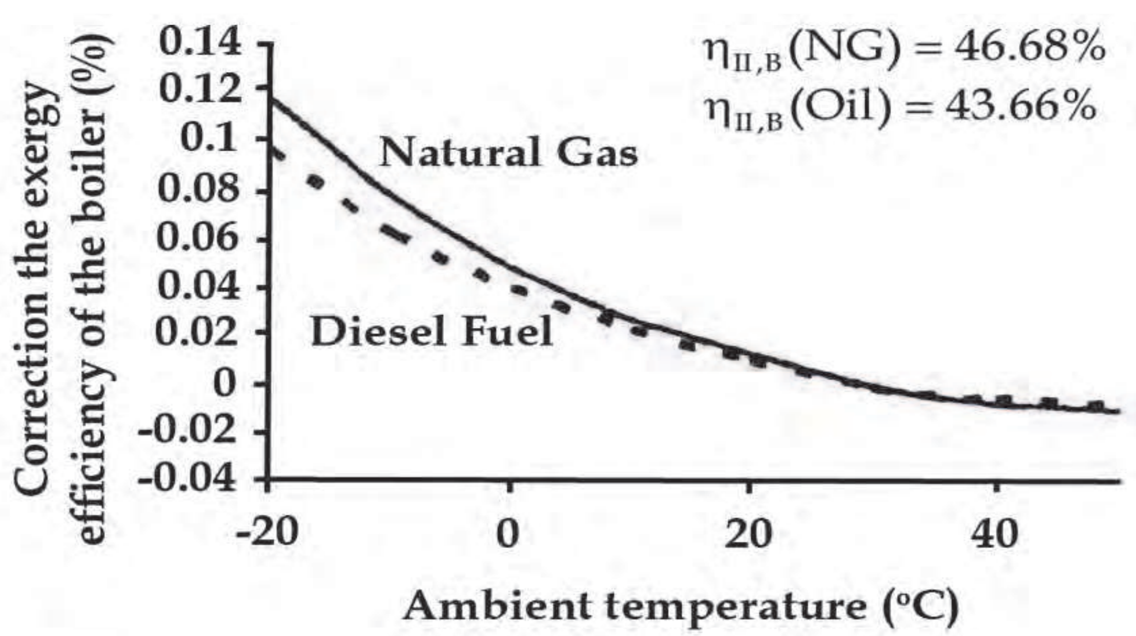

Fig. 12. The diagram of boiler efficiency changes based on the air temperature

5. Conclusions

The exergy analysis of the power plant cycle has shown that

- The total exergy efficiency of the plant is 36.1% for natural gas and 35.5% for diesel fuel.
- Among the main elements of the plant cycle, the greatest irreversibility (exergy losses or the least exergy efficiency) belongs to the boiler.
- The internal losses of the boiler which includes the heat transfer losses, the combustion losses and the friction losses, are 362899 kW for natural gas and 411127 MW for diesel fuel, and the exergy losses of the chimney, which are caused because of the combustion hot gases exiting it, are 12453 kW for the natural gas and 12668 kW for diesel fuel. The natural gas and diesel fuel, respectively, have chemical exergies of 700182 kW and 747995 kW.
- The comparison of the internal losses and the chimney losses shows that the outgoing exergy from the chimney is not considerable. Also, the exergy losses of the boiler are 46.68% for the natural gas and 43.66% for diesel fuel, under the assumed conditions ($T_0 = 30^{\circ}C$ and $P_0 = 90\ kPa$).
- Analysing the boiler processes shows that the exergy losses caused by heat transfer (255999 kW for the natural gas and 239302 kW for diesel fuel) are greater than the exergy losses of the combustion (106900 kW for the natural gas and 171835 kW for diesel fuel) and those against the heat transfer process in the combustion process of the natural gas are less irreversible than in the diesel fuel combustion.
- After proving that the main cause of the exergy losses are the heat transfer has the boiler, the exergy analysis of the boiler elements shows that the reheater has the lowest exergy efficiency (40.99% for the natural gas and 46.6% for diesel fuel) and that the evaporator has the highest exergy losses (85800 kW for the natural gas and 114071 kW for diesel fuel). All these reports show that natural gas is better than diesel fuel in producing super heater vapour in the boiler.

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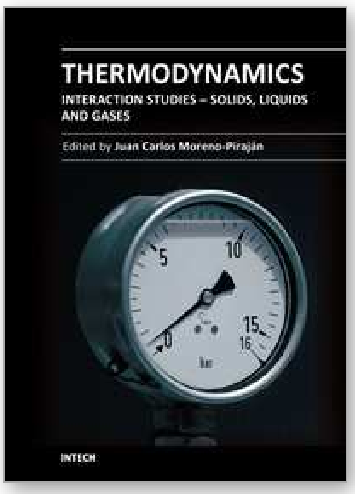

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