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# Investigation of Turbulators for Fire Tube Boilers Using Exergy Analysis

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**Abstract**

In this paper, an experimental study of five different types of "turbulator" inserts for fire tube boilers is presented. The experimental setup was constructed in the Department of Mechanical Engineering Laboratory of Karadeniz Technical University. It was tested to evaluate the boiler efficiency according to TS 4040 standard, and to evaluate the second law efficiency of thermodynamics for the cases with and without inserts under the same operating conditions. Four new types of turbulator consisted of truncated hollow cones. They were placed in tandem parallel to the gas flow direction to allow no contact with the fire tube wall. The half-apex angles of the conical turbulators used in the experiments were 14° and 20°. The number of the turbulators inserted in the of the boiler was 200, their installation arrangements in the fire tubes were periodic, and their enlarging positions were in accordance with the gas flow direction. It was found that turbulators increased the boiler efficiency from 8% to 12% and the second law efficiency of the boiler from 24% to 27%. It is also shown that the one with the half-apex angle of 20° gives better results, about 4% enhancement compared with the other ones. A fifth new turbulator, consisting of a truncated half-cylindrical surface and placed in tandem with flow direction, also periodically interrupted and transposed in the fire tube boilers, provided a 4% increase in the boiler efficiency. The second law efficiency of the boiler was improved from 24% to 25.2%. It was also shown that there was no need to use an excess fan for the flue gas in the chimney because of the very low pressure drops in the new types of turbulator.

**Key Words:** Boiler, Turbulator, Fire Tube, Efficiency, Exergy

## Ekserji Analizi Kullanılarak Duman Borulu Kazanlarda Türbülatörlerin İncelenmesi

**Özet**

Bu yayında, duman borulu kazanlar için beş yeni tip türbülatörlerin deneysel çalışması ele alınmıştır. Kazan deney düzeneği, Karadeniz Teknik Üniversitesi Makina Mühendisliği Bölümü Laboratuvar'ında kurulmuştur. Kazan, aynı işletme şartlarında türbülatörlü ve türbülatörsüz durumlar için TS 4040 standartlarına göre tanımlanan kazan verimi ve Termodinamiğin ikinci kanununa dayalı verimi belirlemek üzere kazan test edilmiştir. Dört yeni türbülatör, kesik halka koniden meydana gelmekte ve bunlar akış yönünde paralel ve boru cidarına değmeyecek şekilde yerleştirilmişlerdir. Kesik halka konilerin yarım tepe açıları 14° ve 20° seçilmiştir. Kazan içerisine yerleştirilen 200 adet türbülatörler duman boruları içerisine akış yönünde genişleyecek şekilde yerleştirilmiş olup, birinci kanun verimini türbülatörsüz duruma göre % 8 ila % 12 artırdığı ve ikinci kanuna dayalı kazan verimini de % 24 den % 27 ye yükseltiği bulunmuştur. Yarım tepe açısı 20° olan kesik halka konili türbülatörlerin diğerlerine göre daha iyi sonuç verdiği ve diğerlerine göre % 4 etkili olduğu gözlenmiştir. Beşinci yeni tip türbülatör, kesik yarım silindirik halka yüzeyli ve saptırılmış türbülatörler, gaz akız yönüne paralel yerleştirilerek yapılan deneylerde birinci kanuna dayalı kazan veriminde % 4 lük bir verim artışına neden olmuştur. İkinci kanuna dayalı kazan veriminde ise % 24 den % 25.2 ye kadar bir artış sağlamıştır. Yeni tip türbülatörler basınç kaybı çok az olduğundan, kazan tesisinde baca gazlarını çekmek için ilave bir fana gereksinim duyulmayacağı gösterilmiştir.

**Anahtar Sözcükler:** Kazan, Türbülatör, Alev borusu, Verim, Ekserji

## Introduction

In recent years, energy saving has been studied extensively, taking into consideration air pollution and shortage of energy sources. Investigations for these purposes show that boiler efficiency is an important parameter. Any improvement in the efficiency of boilers, where fossil-based fuels are used, causes energy savings and economical energy usages. The majority of Turkeys energy requirements Turkeys are satisfied by imported energy. Approximately 50 percent Turkeys of total energy will be supplied by foreign countries after 2010. In this situation, the importance of saving energy and reducing fuel consumption is obviously important.

The use of availability and irreversibility in an actual problems from a thermodynamics point of view is of growing importance for thermo-economic design. Theoretical and experimental analyses were carried out on a boiler to see what happened to the availability of the air-fuel mixture that entered the furnace and where the irreversibility occurred during the process. In many engineering decisions, other factors, such as the impact on the environment (for example, air pollution) and the impact on society, must be considered when developing the optimum design. In connection with the increased use of availability analysis in recent years, a term called "second law efficiency" has come into more common used. This involves a comparison with the cost, or input, in terms of the thermodynamic availability (Van Wylen and Sonntag, 1985).

The thermal efficiency of a boiler is calculated indirectly by thermal loss estimation or directly with the first law of thermodynamics. In reality, the amount of energy is as important as the energy quality and energy usage. Therefore, the second law of thermodynamics is sufficient for the optimum design for all processes.

In order to increase boiler efficiency, combustion efficiency is an important parameter. The research on the effects of improved combustion process on boiler efficiency shows that the efficiency was increased 5% for the complete combustion process in comparison with the incomplete combustion process (Tüter, 1974). A lot of research has been done on the use of different kinds of fuels in boilers. Research on boilers capable of using fuel oil and natural gas show that to produce the same amount of outputs under similar operating conditions, 470 kg/h of fuel oil and 370 kg/h of natural gas were consumed. When the flue gas temperature decreases in the chimney, the boiler efficiency increases, and also during the process, combustion thermal losses and fuel consumption decrease (Tenir and Kincay, 1992).

The effect of turbulators developed in the Mechanical Engineering Department of Karadeniz Technical University on fossil fuelled hot water boilers is investigated in this study.

## Exergy Balance

### Exergy concept

For a given environment, energy which is convertible into other forms of energy is called "Useful Energy" or "Exergy". Energy which is impossible to convert into other energy forms is called "Useless Energy" or "Anergy" (Büyüktür, 1986).

Energy = Exergy + Anergy

The determination of the useful part of a given amount of any energy form (just like heat, enthalpy), exergy, is introduced. Therefore, exergy is the maximum possible work for a reversible process from specified initial state to the state of its environment. In accordance with this definition, exergy can be calculated at the given temperature, pressure and composition of the environment and also a reversible process must be considered. For the exergy analysis, a form of reversible process is not needed, like other thermodynamic analysis. Knowing the conditions of the initial and final states of the process is enough to calculate the exergy. The exergy balance can be written for the steady-flow process (Arıkol, 1985) as follows:

$$\sum E_{\text{in}} - \sum E_{\text{out}} + \sum E_{\text{loss}} = 0 \qquad (1)$$

Lost work can be defined as the difference between maximum work and actual work so

$$W_{\text{loss}} = W_{\text{max.}} + W = E_{\text{loss}} \qquad (2)$$

At the same time this is equal to exergy loss (Tenir and Kincay, 1992; Arıkol, 1985).

## Experimental Setup

### Experimental boiler properties

The experimental setup was designed and built according to Turkish Standard TS 4041. Its schematic picture is given in Figs. 1 and 2. The low pressure and smoke tube hot water boiler is used in this experimental setup. The outer surface of the boiler is 7.5 $m^2$, the heating capacity of the boiler is 209,000 kJ/h and its tested pressure is 5.9 bars.

### Boiler fuel properties

Thin combustion oil called motorin, which does not need precombustion was used as fuel in the test. The maximum amount of $CO_2$ in the dry flue gas is 15.4%, the minimum amount of air $L_{\text{min}} = 11.04$ $m^3$/kg, and the amount of dry flue gas $V_{\text{min}} = 11.73$ $m^3$/kg.

The mass combustion of motorin consisted of 85% C, 13% H 1.7% O and 0.3% S. The caloric value of motorin was given as 42,636 kJ/kg (Telli, 1989; Özge, 1989).

### Turbulators

The somewhat complex geometries of the truncated conical ring turbulators tested are given in Fig. 3, where the nomenclature used is noted on each figure and a letter is used to identify each insert. Numerical values of the various geometric dimensions are also given in the same figure. The truncated conical ring turbulators were tested by Ayhan et al. (Karabay and Ayhan, 1988; Ayhan and Arıcı,1985). A 135% to 175% increase in the heat transfer coefficient at a Reynolds number of 10,000 was obtained with friction factor increases of nearly 1000%. In a study by Ayhan and Arıcı (1985), the half apex angles of 14° and 20° were used for truncated conical ring turbulators. They were made of 0.5 mm thick steel plate. Several commercial, arbitrary design, in-

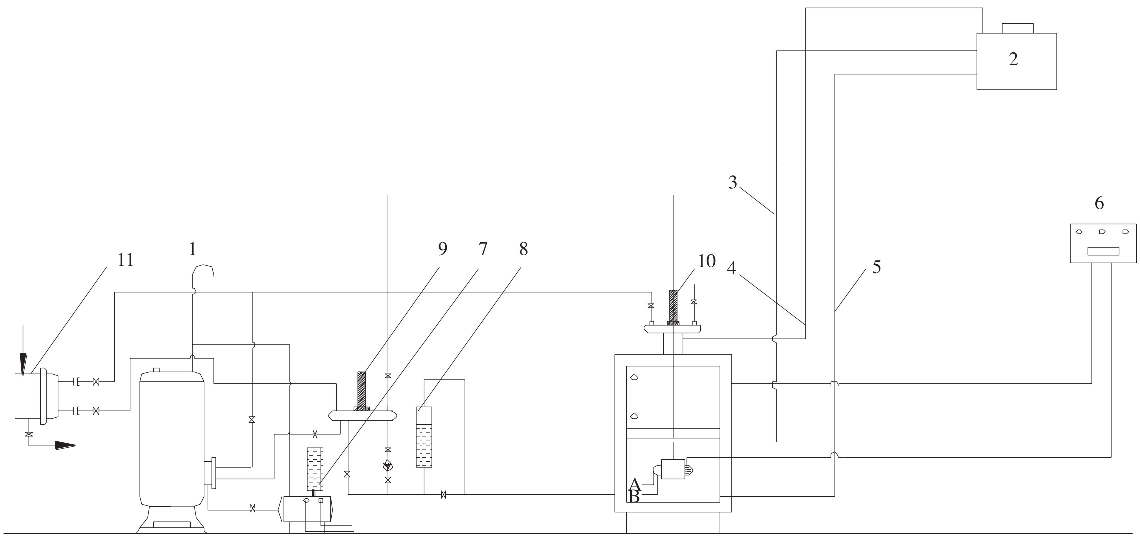
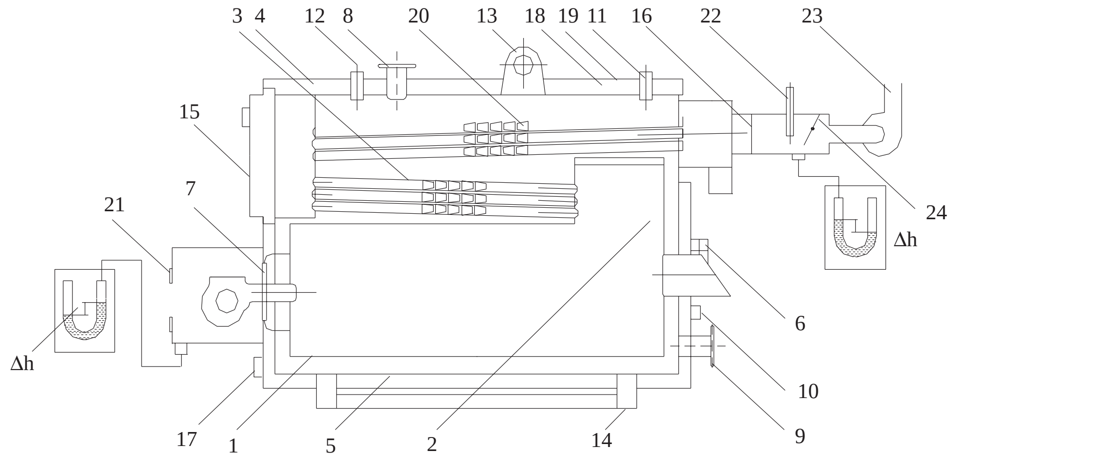



**Figure 1.** Schematic view of experimental setup




| | | | |
|---|---|---|---|
| 1. Fire door | 7. Burner | 13. Hanger | 19. Insulating shell |
| 2. Furnace | 8. Water outlet pipe | 14. Boiler emplacement | 20. Turbulators |
| 3. First smoke-tube | 9. Water inlet pipe | 15. Front smoke box | 21. Orifice |
| 4. Second smoke-tube | 10. Inlet safety | 16. Back smoke box | 22. Measuring canal of flue gas |
| 5. Boiler shell | 11. Outlet safety | 17. Filler and discharge pipe | 23. Fan |
| 6. Blowdown | 12. Junction of measuring instr. | 18. Insulating material | 24. Butterfly valve |

**Figure 2.** Schematic view of experimental setup.

terrupted and periodically transposed truncated half cylinder surface type turbulators were tested by Ayhan (1996). The fundamental work has been done to establish the heat transfer/flow friction basis for the design of inserts. The geometry and dimensions of the half cylinder surface type turbulator tested are given in Fig. 4, where the nomenclature used is noted on the figure and a letter is used to identify the insert. Their arrangements in boiler pipes are shown in Fig. 2. Turbulators are placed into smoke tubes in such a way that they increase the through flow of smoke gas. Contracting and enlarging conical ring surface turbulators are more effective for heat transfer enhancement in comparison with other types of turbulators (Ayhan, 1985; Bergles, 1983; and Junkham et al., 1985).

In a study by Ayhan (1996), electrically heated pipe flow facility was used to deliver heated air. Tube wall temperatures, fluid bulk temperatures, and flow rates were measured to obtain sectional average heat transfer coefficients for 10 segments of the tube. Reference data for the empty tube were in excellent agreement with the Petkow correlation for heat transfer and friction factor (Holman , 1997).

Heat transfer coefficient increases of 85% to 70% were recorded at a Reynolds number of 10,000, and the respective friction factor increases were very low in comparison with other types of insert.

Also, fabrications and applications of this kind of turbulator are easily comparable with other types of insert.

## Experimental Procedure

Before the experiment was performed, the boiler was operated for a while to reach steady state condition. Hot water coming from the boiler was sent to the water-cooled heat exchanger, and also to domestic heating convectors. The volumetric flow rate of cooling water, supplied by the waterworks system, is adjusted by means of a valve for the water cooled heat exchanger. The volumetric flow rate of hot water is adjusted by means of valves placed on the collector for supply lines and on the collector for receiving lines. The air flow rate for combustion and flow rate of flue gas are adjusted by means of a throttle valve, which is placed at the inlet of the chimney to control combustion.

Firstly, experiments were carried out for empty smoke tubes of the boiler. Data was obtained from experimental studies using the First Law of Thermodynamics. Reference data for the empty smoke tubes of the boiler was in excellent agreement with the TS 4041 value for boiler efficiency, which is defined in accordance with the First Law of Thermodynamics. Later, four geometrical variation of truncated conical ring inserts and one interrupted and periodically transposed truncated half cylinder surface type turbulator were tested. In order to achieve steady state condition of the boiler, fuel-oil consumption was kept constant but the mass flow rate of the circulated hot water was altered. In the experiments of truncated hollow cone inserted tubes, an additional fan was used for the flue gas in the chimney.

The air flow rate for combustion was kept constant for all experiments. In order to achieve this, the air flow rate for combustion and air leakage in the furnace were controlled. The air flow rate for combustion was adjusted by using a throttle valve at the inlet of the chimney.

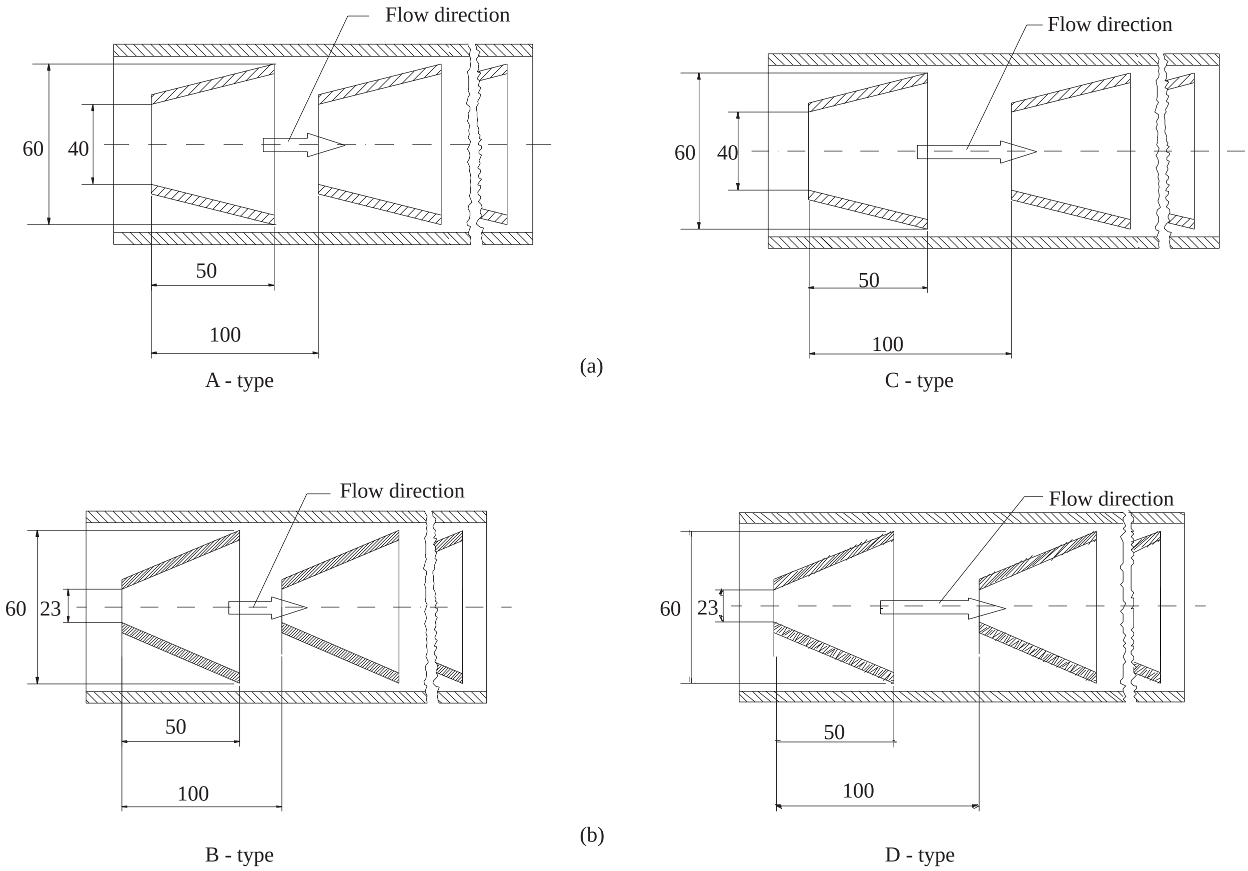



**Figure 3.** Schematic view of conical turbulators.

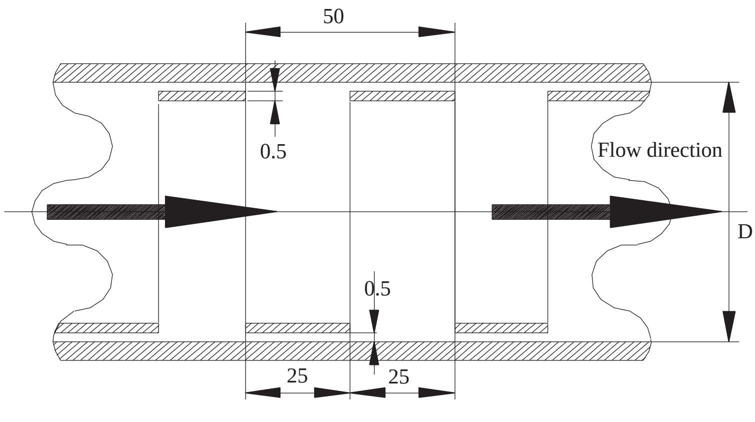



**Figure 4.** Schematic view of transposed truncated half cylindrical surface type turbulators.

### Instrumentation

Boiler inlet and outlet water temperatures were measured with liquid thermometers with an accuracy of 0.1°C. Flue gas temperatures were measured with thermocouples, installed at the inlet of the chimney. Volumetric flow rate of the circulated hot water was measured with a rotameter. Fuel-oil consumption was measured by calibrated flow meter, placed on the fuel line.

Flue gas components were analyzed by smoke tube gas analyzer and the volumetric percentages of $CO_2$, CO and $O_2$ gases were determined. The volumetric flow rate of air accompanying combustion was measured by orifice meter. Pressure losses were measured by U-type manometer. A pressure tap is placed at the entrance of the chimney. Measurements were taken at 10 minute intervals. The values obtained during the experiments are shown in Tables 1 and 2.

**Table 1.** Performance measurement of boiler.

| | M | $T_{\text{out}}$ | $T_{\text{in}}$ | Bh | $T_{\text{mean}}$ |
|---|---|---|---|---|---|
| Empty | 1906.71 | 88 | 66 | 4.845 | 34 |
| Type A | 2102.27 | 88.5 | 66.2 | 4.91 | 36 |
| Type B | 2248.94 | 86.4 | 65.1 | 4.935 | 40 |
| Type C | 2060 | 89 | 69 | 4.88 | 34 |
| Type D | 2170 | 87 | 67 | 4.91 | 37 |
| Type E | 2125 | 84 | 63 | 4.92 | 34 |

**Table 2.** Measurement of gas compositions.

| | $T_a$ [°C | $YCO_2$ [%] | $YO_2$ [%] | YCO [%] | $YN_2$ [%] | $YNO_2$ [%] | $YSO_2$ [%] | $P_a$ [Pa] | $V_{\text{meas}}$ [m$^3$/h] |
|---|---|---|---|---|---|---|---|---|---|
| Empty | 237.1 | 9.73 | 7.5 | 0.12 | 82.6 | 0.0027 | 0.0017 | 0.058 | 84.9 |
| Type A | 136.6 | 9.43 | 8.6 | 0.09 | 81.9 | 0.0371 | 0.0016 | 0.137 | 86.5 |
| Type B | 102 | 9.09 | 8.46 | 0.14 | 82.3 | 0.0378 | 0.0018 | 0.264 | 86.7 |
| Type C | 125.4 | 8.1 | 9.9 | 0.069 | 82 | 0.0003 | 0.0016 | 0.14 | 86.5 |
| Type D | 119.2 | 7.9 | 10.1 | 0.0966 | 82 | 0.0027 | 0.0015 | 0.23 | 87 |
| Type E | 178.4 | 9.3 | 8.2 | 0.023 | 82.5 | 0.038 | 0.0197 | 0.06 | 87.5 |

Uncertainties in first and second law boiler efficiencies from a typical run for Type A test, according to the method of Kline and McClintock (1953), using odds of 20 to 1. The following uncertainties are typical of those that can be expected in the other test runs: $\eta_1 = 5$ percent, $\eta_{11} = 5.3$ percent.

### Calculations

#### Excess air coefficient, and amounts of air with combustion and flue gas

The excess air coefficient is defined by

$$\lambda = \frac{\text{Totalamountofair}}{\text{Amountoftheoreticallyrequiredair}} = \frac{L}{L_{\min}} \quad (3)$$

Where L is the total amount of air and $L_{\min}$.. is the amount of theoretically required air for combustion. For the situation of incomplete combustion, $O_2$, $CO_2$ and $N_2$ volumetric rates will change as follows:

$$\dot{Y}O_2 = YO_2 - 0.5YCO \quad (4)$$

$$\dot{Y}CO_2 = YCO_2 - YCO \quad (5)$$

$$\dot{Y}N_2 = 100\% - \dot{Y}CO_2 - \dot{Y}O_2 \quad (6)$$

Using equations (4) to (5), the excess air coefficient is calculated as follows (Tüter, 1974);

$$\lambda - 1 = \frac{\dot{Y}O_2}{\frac{21}{79}\dot{Y}N_2 - \dot{Y}O_2} \quad (7)$$

The excess air coefficient values calculated are given in Table 3. The excess amount of air, total amount of air and also the total amount of flue gas were defined by the following equations:

$$\text{Excess amount of air } L_1 = (\lambda - 1) \quad (8)$$

$$\text{Total amount of air } L = L_{\min} + L_u \quad (9)$$

$$\text{Total amount of flue gas } V_{\text{cal}} = V_{\min} + L_u \quad (10)$$

#### First law efficiency calculations

Fuel and air temperatures at the entrance to the boiler are assumed equal to the surrounding temperature, which is $T_0 = 19$°C. Boiler efficiency is calculated by the amount of heat extracted from working fluid per hour which is $Q_{\text{water}}$, over heat load of fuel used. According to the first law of thermodynamics, boiler efficiency is calculated as follows (Tenir and Kincay, 1992; and TSE, 1983):

$$\eta_I = \frac{mC_p(T_{\text{out}} - T_{in})}{Bh\ Hu} \tag{11}$$

If combustion air is heated, that amount of heat will be considered a input together with fuel energy.

### Loss details and indirect method of efficiency calculation

It is not possible to use all the of heat realised as a result of combustion for any kind of fuel, due to losses. Boiler efficiency is calculated as follows by taking lasses into consideration (Özge, 1989):

$$Q_{\text{indirect}} = 100 - 3 \text{ Losses} \tag{12}$$

$$Q_{\text{indirect}} = 100 - (v_f + v_r + v_a) \tag{13}$$

Total losses are divided into three parts:

a) Combustion loss: Combustion loss varies depending on the place of combustion and the type of combustion. For liquid fuels, combustion loss is relatively small and is found to be $v_f = 0.01$ (Özge, 1989).

b) Residual losses: Radiational and convectional heat transfer rate from outer surface of boiler to surrounding is called Residual Loss. The heat transfer rates from the outer surface of the boiler depend on the type of boiler and heat insulation of the boiler. The temperature of the outer metal surface of the boiler should not exceed 40°C while boiler is being operated (TSE, 1983). Residual losses can be calculated depending on the mean surface temperatures of the boiler as $a_{\text{mean}} = 33.44$ kJ/m$^2$h°C, $T_0 = 19$°C,

$$Q_x = F_x a_{\text{mean}}(T_{c.\text{mean}} - T_0) \tag{14}$$

$$\eta = \frac{\sum Q_x}{Q_{\text{water}}} \tag{15}$$

The total heat transfer coefficients at the divided parts of the boiler surface can be chosen depending on surface temperature (TSE, 1983).

c) Flue gas losses: Approximate calculation of flue gas losses is performed using the expression given below where $Z$ is the loss factor. This loss factor depends on fuel type, fuel components and $YCO_2$ amount (TSE, 1983). The values calculated are given in Table 3.

$$V_a = Z\frac{T_a - T_0}{YCO_2} \tag{16}$$

### Second law efficiency calculations

The exergy balance expressions used in this study are given below. Exergy of smoke tube gas

**Table 3.** Values of boiler heat losses and amounts of burning air and flue gas.

| | Combustion air and Smoke-tube gas | | | Heat Losses | | |
|---|---|---|---|---|---|---|
| | $\lambda$ | $L$ | $V_{\text{cal}}$ | $v_r$ | $v_f$ | $v_a$ |
| | [%] | [m$^3$/h] | [m$^3$/h] | [%] | [%] | [%] |
| Empty | 1.52 | 81.43 | 82.12 | 0.5 | 1.9 | 13.3 |
| Type A | 1.56 | 8489 | 85.56 | 0.5 | 2.3 | 6 |
| Type B | 1.55 | 84.44 | 85.11 | 0.5 | 2.8 | 5.4 |
| Type C | 1.88 | 98 | 100 | 0.5 | 2 | 9 |
| Type D | 1.91 | 103 | 106 | 0.5 | 2.3 | 6.2 |
| Type E | 1.64 | 87.5 | 90.4 | 0.2 | 2 | 9.9 |

$$E_{ST} = \dot{m}\left\{c_p(T_a - T_0) - T_0 c_p \ln\frac{T_a}{T_0}\right\} + E_{\text{chem}}. \tag{17}$$

$$E_{\text{chem.}} = RT_0\left\{YN_2 \ln\frac{YN_2}{0.7893} + YO_2 \ln\frac{YO_2}{0.2099} + YCO_2 \ln\frac{YCO_2}{0.000345}\right\} + \\ YCO\left\{b^0_{ch_{co}} + \frac{T_a - T_0}{T_a}(C^0_{ch_{co}} - b^0_{ch_{co}})\right\} + RT_0 \ln YCO\frac{P_a}{P_0} \tag{18}$$

Exergy of fuel

$$E_{\text{fuel}} = Hu\left\{1.0401 + 0.1728\frac{H}{C} + 0.0432\frac{O}{C} + 0.2169\frac{S}{C}\left[1 - 2.0628\frac{H}{C}\right]\right\} \tag{19}$$

where H/C, O/C and S/C ratios are mass proportions of the elements (Szargut et al, 1988). Heat loss exergy from the surface (Tenir and Kincay, 1992)

$$E_Q = Q_x\left[1 - \frac{T_0}{T_{S_{\text{mean}}}}\right] \tag{20}$$

Exergy of water in the boiler (Arıkol, 1985)

$$\Delta E_{\text{water}} = (H - H_0) - T_0(S - S_0) \tag{21}$$

Exergy loss (Tenir and Kincay, 1992)

$$E_{\text{loss}} = E_{in} - E_{\text{out}} = E_f - \Delta E_w - E_{st-T_a} - E_{Q-x} \tag{22}$$

Additional exergies from the fan placed inside the smoke tube, water circulation pump and burner are calculated separately and added to fuel exergy in the denominator of the efficiency expression.

$$\eta_{II_{\text{boiler}}} = \frac{\Delta E_w}{E_f + E_{\text{fan}} + E_{wp} + E_{br}} \tag{23}$$

Data produced by using equations (17) to (23) are presented in Table 4.

**Table 4.** Values of energy, exergy and efficiency at experimental results obtained.

| | $Q_w$ [kJ/kg$_f$] | $E$ [kJ/kg$_f$] | $E_f$ [kJ/kg$_f$] | $E_Q$ [kJ/kg$_f$] | $E_w$ [kJ/kg$_f$] | $E_{\text{loss}}$ [kJ/kg$_f$] | $O_{\text{ind}}$ [%] | $O_I$ [%] | $O_{II}$ [%] |
|---|---|---|---|---|---|---|---|---|---|
| Empty | 36252 | 2267 | 45570 | 308 | 11269 | 31726 | 84 | 85 | 24 |
| Type A | 39939 | 789.9 | 45570 | 398.1 | 12667 | 31715 | 91 | 93 | 27 |
| Type B | 40602 | 701 | 45570 | 508 | 12643 | 31718 | 92 | 95 | 27 |
| Type C | 37274 | 803.7 | 45570 | 429.6 | 12598 | 31738 | 89 | 88 | 27 |
| Type D | 39024 | 765 | 45570 | 455 | 12608 | 31744 | 91 | 92 | 27 |
| Type E | 36505 | 2046 | 45570 | 235 | 11550 | 31739 | 85.7 | 87.6 | 25.2 |

## Results

It was first necessary to ensure that the experimental procedure and method of data processing were satisfactory. This was done by comparing data for the empty smoke tube of the boiler with Turkish Standard of Boiler TS 4040 values. Reference data for the empty smoke tube of the boiler was in excellent agreement with TS 4040 values. The boiler efficiency in accordance with the first law was found to be 84-85% percent for the empty smoke tube of boiler.

A, B, C, D and E types of turbulators were tested. Boiler efficiency increases from 85% to 93% for type A, to 95% for type B, to 89% for type C, to 91% for type D and to 89% for type E.

Experimental results based on the second law of thermodynamics to determine efficiency show that type A enhanced boiler efficiency 4% in comparison with type B and also type C increased boiler efficiency 3% in comparison with type D. Type E turbulators increases the second law efficiency of the boiler from 24% to 25.5%. The pressure drops in the E-type turbulator are smaller than those in the other types of turbulator which are introduced in this study. Therefore, the E-type turbulator is recommended.

Using the estimated heat loss values of the boiler, the indirect method of efficiency calculation gives almost the same boiler efficiency for the turbulator tested in the boiler. The results are presented in Table 4.

Types A, B, C and D increased pressure losses, even though they enhanced boiler efficiencies. Because of that an excess fan is used. But for type E a fan is not used, due to the very low pressure losses.

In order to evaluate boiler performance under the conditions of excess energy, usage of a fan and pressure drops in the smoke tube of the boiler, the second law efficiencies of the boiler were calculated for the turbulators. Increases in the second law efficiency of the boiler of 12.5% for A and B type turbulators, 11% for C and D type turbulators, and 4% E type turbulators were estimated, in comparison with results from the boiler without inserts under similar operating conditions.

## Concluding Remarks

The present results show that the commercial turbulator inserts tested provide considerable enhancement of gas-side convective heat transfer coefficients. Also, radiation heat transfer was occurred on the total heat transfer rate. Radiation heat transfer effects on the total heat transfer rate will be investigated in the future.

Design dimensions are presented for the turbulators. Although it may be possible to reduce fuel consumption through the use of turbulators, improvement in gas-side convection coefficients does not result in a one-to-one improvement in boiler efficiency. Rather, enhancement of gas-side coefficients is only one factor among many items such as fuel/air ratio adjustment, tube cleanliness, and air/gas system flow resistance that lead to reduced fuel consumption and improved boiler efficiency. The calculation of boiler efficiency with turbulators should consider also changes in the water-side flow distribution (and heat transfer coefficient) due to a large increase in the gas-side heat transfer coefficient.

## Nomenclature

| | |
|---|---|
| $b^o ch_{co}$ | [kJ/kg]: Standard chemical energy of carbon dioxide |
| Bh | [kg/h]: Mass flow rate of fuel |
| $CO_{2-max}$ | [%]: Maximum $CO_2$ ratio in flue gas |
| $c_p$ | [kJ/kg°K]: Specific heat |
| $C^o_{co}$ | [kJ/kg]: Formation enthalpy of $CO_2$ |
| $E_{st}$ | [kJ/kg]:Exergy for flue gas |
| $E_h$ | [kJ/kg]: Exergy for enthalpy |
| $E_{\text{loss}}$ | [kJ/kg]: Losses exergy |
| $E_{\text{chm.}}$ | [kJ/kg]: Chemical exergy |
| $E_{Q-x}$ | [kJ/kg]: Exergy of boiler surface temperature |
| $E_w$ | [kJ/kg]: Water exergy |
| $E_f$ | [kJ/kg]: Fuel exergy |
| $E_x$ | [m$^2$]: Outer surface of boiler |
| Hu | [kJ/kg]: Thermal capacity |
| $H_0$ | [kJ/kg]: Reference enthalpy |
| $L$ | [m$^3$/h]: Total amount of air |
| $L_{\text{min.}}$ | [Nm$^3$/kg]: Minimum mass flow rate of air for combustion |
| $L_u$ | [Nm$^3$/kg]: Excess air |
| $m$ | [kg/h]: Feed water mass flow rate |
| $n_i$ | [kmol]: Mole number |
| $P$ | [Pa]: Pressure |
| $Q_w$ | [kJ/kg]: Heat for feed water |
| $Q_x$ | [kJ/kg]: Heat loss from boiler surface |
| $R$ | [kJ/kg°K]: Universal gas constant |
| $S_o$ | [kJ/kg°K]: Reference entropy |
| $T_a$ | [°C]: Flue gas temperature |
| $T_{\text{s-mean}}$ | [0C]: Mean temperature of boiler surface |
| $T_{\text{out}}$ | [0C]: Outlet temperature of feed water |
| $T_{in}$ | [°C]: Inlet temperature of feed water |
| $T_0$ | [°C]: Reference temperature |
| $V_{\text{cal.}}$ | [m$^3$/h]: Calculated mass flow rate of flue gas |
| $V_{\text{min.}}$ | [m$^3$/h]: Minimum flow rate of flue gas |
| $V_{\text{meas.}}$ | [Nm$^3$/kg]: Measured mass flow rate of flue gas |
| $v_a$ | [%]: Flue gas loss |
| $v_f$ | [%]: Combustion loss |
| $v_r$ | [%]: Wasted heat loss |
| $W$ | [kJ/kg]: Work |
| $\dot{Y}O_2$ | [%]: Variation of volumetric ratio of oxygen for incomplete combustion. |
| $\dot{Y}CO_2$ | [%]: Variation of volumetric ratio of carbon dioxide for incomplete combustion. |
| $\dot{Y}CO$ | [%]: Variation of volumetric ratio of carbon monoxide for incomplete combustion. |
| $YN_2$ | [%]: Variation of volumetric ratio of nitrogen for incomplete combustion. |
| $Z$ | : Chimney loss factor |
| $l$ | : Excess air coefficient |
| $\eta_{\text{ind.}}$ | [%]: Efficiency |
| $\eta_I$ | [%: First law efficiency |
| $\eta_{II}$ | [%]: Second law efficiency |
| $:i0$ | [kJ/kmol]: Chemical potential |

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