Concept of Heat Recovery from Exhaust Gases

Maria Bukowska 1, Krzysztof Nowak 1, Danuta Proszak-Miąsik 1, Sławomir Rabczak 1

1Rzeszow University of Technology, al. Powstańców Warszawy 12, 35-959 Rzeszów, Poland

mbuk@prz.edu.pl

Abstract. The theme of the article is to determine the possibility of waste heat recovery and use it to prepare hot water. The scope includes a description of the existing sample of coal-fired boiler plant, the analysis of working condition and heat recovery proposals. For this purpose, a series of calculations necessary to identify the energy effect of exhaust temperature decreasing and transferring recovery heat to hot water processing. Heat recover solutions from the exhaust gases channel between boiler and chimney section were proposed. Estimation for the cost-effectiveness of such a solution was made. All calculations and analysis were performed for typical Polish conditions, for coal-fired boiler plant. Typicality of this solution is manifested by the volatility of the load during the year, due to distribution of heat for heating and hot water, determining the load variation during the day. Analysed system of three boilers in case of load variation allows to operational flexibility and adaptation of the boilers load to the current heat demand. This adaptation requires changes in the operating conditions of boilers and in particular assurance of properly conditions for the combustion of fuel. These conditions have an impact on the existing thermal loss and the overall efficiency of the boiler plant. On the boiler plant efficiency affects particularly exhaust gas temperature and the excess air factor. Increasing the efficiency of boilers plant is possible to reach by following actions: limiting the excess air factor in coal combustion process in boilers and using an additional heat exchanger in the exhaust gas channel outside of boilers (economizer) intended to preheat the hot water.

1. Introduction

The aim of the analysis performed was a concept of heat recovery from exhaust gases from the boiler room, determination of the possibility to recover waste heat and using it for preparing hot tap water in neighbouring residential buildings. This article describes a typical Polish boiler room in which bituminous coal is burnt, and the possibility and usefulness of heat recovery with that boiler room as an example.

2. Description of current condition of the analysed boiler room

2.1. Description of the boiler room

The housing estate boiler room of a Housing Cooperative is located on one of housing estates in the town of Radymno. It is intended for supplying heat for central heating and hot tap water purposes. The boiler room is equipped with 3 SWC „Rumia” boilers with the power rating of 900 kW each. The boilers are fired with fine coal. Two boilers operate with the purpose of central heating, and the third one to the needs of hot tap water. In the summer season only one boiler operates, supplying buildings with hot water. The boiler room has a closed cycle. Hot tap water is prepared in a JAD-D2-6/50 heat exchanger and gathered in two 2000/6 heat accumulators. Cold water is supplied from the water
supply system, heated in JAD exchangers, pumped to hot water tanks and then to consumers using pumps.

2.2. Description of exhaust gas dust extraction system
All boilers have individual fans carrying exhaust gases away to the flue. Three WPWs40/1.8 A+K centrifugal single-stream fans have been installed and they have the efficiency $Q_v=2.2$ m$^3$/s. Exhaust gases from boilers are conveyed to the flue through CE-4x400/0.5 multi-cell cyclones cooperating with the fans. Exhaust ducts are connected with the flue. A section area of the flue is 580x850 mm and its length is approx. 13500 mm.

2.3. Description of chimney
The exhaust gases with the catalogue temperature of 453 K are conveyed from each of the three „Rumia” SWC 900 boilers though the flue to a joint chimney. It is a steel, 35 m high chimney with an outlet diameter of 0.9 m. The chimney is free-standing and located outside the building.

3. Fuel policy
In the analysed local boiler room, II A fine coal is burnt to produce heat for the purpose of central heating and hot tap water. An amount of fuel purchased in specific months of the year and its basic parameters were presented in Table 1.

| Month | $m_p$ | $Q_w$ | $Q$ calculated | Coast | Price PLN/Mg | Price PLN/GJ | $Q_{measured}$ | Price PLN/GJ |
|-------|-------|-------|----------------|-------|--------------|--------------|----------------|--------------|
| I     | 4.69  | 24    | 112.56         | 115.51| 239.59       | 25.95        | 1955           | 75.60        |
|       | 109.69| 23    | 2522.87        | 2339.05| 213.26       | 21.27        | 1888           | 75.60        |
| II    | 95.89 | 24    | 2301.36        | 2358.33| 245.95       | 25.95        | 746            | 75.60        |
|       | 26.44 | 22    | 581.68         | 5870.73| 222.04       | 20.90        |                |              |
| III   | 149.45| 24    | 3586.80        | 3675.52| 245.95       | 25.95        | 1802           | 20.40        |
| IV    | 78.30 | 24    | 1879.20        | 1925.04| 245.95       | 25.95        | 1322           | 14.57        |
| V     | 65.29 | 24    | 1566.96        | 1605.82| 245.95       | 25.95        | 989            | 16.24        |
| VI    | 59.02 | 24    | 1416.48        | 1451.08| 245.95       | 25.95        | 746            | 19.46        |
| VII   | 39.44 | 24    | 946.56         | 9700.34| 245.95       | 25.95        | 779            | 16.17        |
| VIII  | 9.97  | 24    | 239.28         | 2893.67| 290.24       | 20.90        |                |              |
| IX    | 38.99 | 23    | 896.77         | 10084.37| 258.64       | 11.25        | 724            | 20.47        |
|       | 16.31 | 24    | 391.44         | 4733.78| 290.24       | 12.09        |                |              |
| X     | 57.38 | 23    | 1319.74        | 14840.76| 258.64       | 11.25        | 785            | 18.91        |
|       | 27.24 | 24    | 653.76         | 6699.73| 245.95       | 25.95        | 1509           | 15.59        |
|       | 36.08 | 23    | 1043.73        | 8349.77| 480.39       | 20.89        |                |              |
|       | 28.38 | 23    | 652.74         | 8476.80| 298.69       | 12.99        |                |              |
| XI    | 57.16 | 24    | 1371.84        | 14058.61| 245.95       | 25.95        | 2029           | 12.44        |
|       | 12.53 | 23    | 288.19         | 3240.75| 258.64       | 11.25        |                |              |
|       | 27.72 | 23    | 637.56         | 7937.17| 286.33       | 12.45        |                |              |
| XII   | 180.38| 24    | 4329.12        | 44364.82| 245.95       | 25.95        | 2458           | 18.05        |
| together | 1120.35| 23.675 | 26524.75 | 275965.03 | 246.32 | 10.40 | 16986 | 16.25 |
From information received from the boiler room and calculations made on the basis of consumed fuel, amounts of produced heat are obtained, as presented in Table no 1. The table shows also amounts of produced heat, as resulting from measurements of installed meters of thermal energy.

A gross annual value of purchased fuel according to calculations made was PLN 275 963.00. It is a value of purchased fuel. Costs of transport were added to it. So, finally, the gross annual cost of fuel was PLN 338 817.40.

4. Energetic assessment of current condition

A measure of the effectiveness of boiler room operation is a total watt-hour efficiency which includes all components of primary and ultimate energy. In the case of the boiler room, it is sufficient to assume thermal efficiency in which electric energy will be disregarded. The basic thermal balance is determined by the following equation:

\[ \eta = \frac{Q_u}{m_{pal} \cdot Q_w} \]  

where:

- \( Q_u \) – useful heat produced to the needs of heating and hot water,
- \( \eta \) – total thermal efficiency,
- \( m_{pal} \) – mass of burnt fuel,
- \( Q_w \) – fuel calorific value.

On the basis of Table 1, the calculated average annual calorific value of burnt coal was:

\( Q_w = 23,675 \text{ MJ/kg} \).

The total amount of coal burnt in that time was:

\( m_{pal} = 1120,35 \text{ Mg} \).

An amount of produced heat in the analysed year as indicated by heat meters was:

\( Q_u = 16 986 \text{ GJ} \).

It means that the total average annual efficiency in the analysed year was:

\[ \eta = \frac{Q_u}{m_{pal} \cdot Q_w} = \frac{16986 \cdot 10^3}{1120 \cdot 10^3 \cdot 23,675} = 0.64 = 64\% \]  

The efficiency achieved is relatively small taking into account the analysed type of the boiler room. The total efficiency is mainly affected by boiler efficiency which is influenced by thermal losses incurred during operation of the boilers. They include: carry-over (chimney) loss, incomplete combustion loss, imperfect combustion loss, radiation loss. The balance of losses is determined by the following relation:

\[ \eta_k = 1 - (s_{wy} + s_{nc} + s_{nz} + s_r) \]  

where:

- \( \eta_k \) – boiler efficiency,
- \( s_{wy} \) – carry-over loss,
- \( s_{nc} \) – fraction of incomplete combustion loss,
- \( s_{nz} \) – fraction of imperfect combustion loss,
- \( s_r \) – fraction of radiation loss.

5. Carry-over (chimney) loss

Most significant in the thermal balance of boiler is a carry-over loss and, therefore, most attention is given to reducing that loss. The carry-over loss results from conveying to the environment exhaust gases whose temperature is higher than the ambient temperature. Such thermal loss results from the following relation:

\[ Q_{wy} = V \cdot \rho_s \cdot c_{sp} \cdot (T_s - T_o) \]
where:
- \( Q_{\text{wyl}} \) – carry-over loss,
- \( V_s \) – amount (volume) of exhaust gases,
- \( \rho_s \) – density exhaust gases,
- \( c_w \) – specific heat of exhaust gases with constant pressure,
- \( T_s \) – temperature of exhaust gases,
- \( T_o \) – ambient temperature.

An amount of exhaust gases is connected with an amount of burnt fuel which is determined by the following relation:

\[
V_s = V_{s1} \cdot m_{\text{pad}}
\]  

(5)

where:
- \( V_{s1} \) - amount of exhaust gases received from a unit of fuel.

If the thermal balance of the boiler is not made using a measuring method, an amount of exhaust gases may be determined computationally in a sufficiently precise way [1].

For coal, a theoretical unit amount of exhaust gases may be calculated using the following formula:

\[
V_{s1} = 0,212 \cdot Q_w + 0,5 \left[ \frac{m^3}{kg} \right]
\]  

(6)

An actual unit amount of exhaust gases includes excess air and is determined by the following formula:

\[
V_{s1} = V_{s1} + (\lambda - 1) \cdot V_a
\]  

(7)

where:
- \( V_a \) is a theoretical unit amount of air.

The above-mentioned quantities refer to conditions of reference, i.e. for pressure \( p_0 = 1 \text{ bar} = 10^5 \text{ Pa} \) and temperature \( T_0 = 0^\circ \text{C} = 273.15 \text{ K} \).

For a calorific value of burnt coal \( Q_w = 23.675 \text{ MJ/kg} \), such quantities were calculated and they are, respectively:

\[
V_{s1} = 6,699 \left[ \frac{m^3}{kg} \right]
\]

\[
V_a = 229 \left[ \frac{m^3}{kg} \right]
\]

An excess air number \( \lambda \) describing air supplied in excess of what is theoretically necessary affects the total amount of exhaust gases and, consequently, the carry-over loss. Therefore, to determine the carry-over loss, only the temperature of exhaust gases and the excess air number are necessary. Other quantities may be assumed without making a major error.

The density of exhaust gases \( \rho_s \) results from the following formula:

\[
\rho_s = \frac{p \cdot M_s}{R \cdot T_s}
\]  

(9)
where:
\[ p \approx 10^5 \text{ Pa} \quad \text{– pressure,} \]
\[ \tilde{R} = 8.315 \text{kJ} / \text{kmol} \cdot K \quad \text{– gas constant,} \]
\[ M_s \approx 30.5 \text{ kg/kmol} \quad \text{– molar mass of exhaust gases.} \]

Also, specific heat of exhaust gases with constant pressure \( c_{sp} \) depends on the excess air number. To determine the excess air number, the knowledge of the exhaust gas composition is necessary, though it is sufficient to know either the fraction of carbon dioxide in exhaust gases or the fraction of oxygen in exhaust gases [2]. The fractions bear the following relation:
\[ y_{\text{co}_2} + y_{\text{co}} + y_{\text{o}_2} + y_{\text{N}_2} = 1 \]  \hspace{1cm} (10)

where:
\[ y_{\text{co}_2} \quad \text{– fraction of carbon dioxide,} \]
\[ y_{\text{co}} \quad \text{– fraction of carbon monoxide,} \]
\[ y_{\text{o}_2} \quad \text{– fraction of oxygen,} \]
\[ y_{\text{N}_2} \quad \text{– fraction of nitrogen.} \]

The fraction of carry-over loss \( S_{\text{wyl}} \) is defined in the following formula:
\[ S_{\text{wyl}} = \frac{Q_{\text{wyl}}}{m_{\text{pal}} \cdot Q_w} \]  \hspace{1cm} (11)

**Figure 1.** Fraction of carry-over loss depending on the temperature of exhaust gases and the excess air number

Such fraction may be also determined using the following approximate formula:
\[ S_{\text{wyl}} \approx \sigma \cdot \frac{T_s - T_o}{y_{\text{co}_2}} \]  \hspace{1cm} (12)
where:
\[ \sigma = 0.68 \] – factor of proportionality for coal.

In view of the above introduction, the fraction of carry-over loss was calculated depending on the temperature of exhaust gases and the excess air number. Such relation with reference to a unit of burnt coal was presented in Figure 1. It was also assumed that the ambient temperature is 10°C.

From the results of measurements made by the Provincial Environmental Protection Inspectorate Laboratory, it appears that the content of oxygen in exhaust gases is 18.8 % and the content of coal dioxide is 1.9 %, which corresponds to a very high excess air number \( \lambda = 10 \). When the operation is proper, the number for that type of boilers should not exceed 2.

With the presence of an excess air number and a measured temperature of exhaust gases of 110 oC, the fraction of carry-over loss is 26.4 %. Taking that into consideration and assuming the above-mentioned total efficiency of 0.64, the remaining losses, apart from the carry-over loss, are:

\[
(s_{nc} + s_{nz} + s_r) = 1 - \eta_k - s_{wyl} = 1 - 0.64 - 0.264 = 0.096
\]

Therefore, the remaining losses constitute only 9.6 % and are slightly changed with a changing load of the boiler.

6. Reduction of fuel consumption

As stated above, the largest loss in using fuel in the boiler is the carry-over loss and, therefore, some efforts should be made to reduce such loss. From the analysis made in the previous chapter it appears that two quantities have the greatest influence on reducing the carry-over loss: the excess air number and the temperature of exhaust gases.

6.1. Excess air number \( \lambda \)

Such number is a measure of the amount of air supplied for combustion in excess of the amount which is theoretically needed, as resulting from stoichiometric equations of combustion. The amount of air should be as little as possible but it should assure a proper process of combustion. It means that the condition of complete combustion should be met, i.e. as little as possible combustible particles (not burnt particles of coal) should be left in slag. At the same time, combustion should be perfect, i.e. there should be no carbon monoxide in exhaust gases. The fraction of carbon monoxide should not exceed 0.1 \( \% \). If coal is burnt on a moving grate, such condition is met when the excess air number is 1.5 \( \% \). The staff operating the boiler is responsible for establishing a proper excess air. A meter of oxygen and/or carbon dioxide content in exhaust gases may be helpful. A carbon dioxide content for the above-mentioned conditions should be 10.8 \( \% \). It is recommended to fix a meter or, even better, a recorder of oxygen content in exhaust gases for each boiler to achieve proper combustion in the boiler.

6.2. Temperature of exhaust gases

The temperature of exhaust gases is a result of exchange of heat between exhaust gases and a heated factor (water). It depends on a heat exchange area, that is a structure of the boiler. Also, the temperature of exhaust gases is affected by the amount of supplied air which does not take part in combustion and must be heated from the ambient temperature to the outlet temperature. Assuming that combustion is made properly and the excess air number is minimal, the temperature of exhaust gases may be reduced only by fixing an additional heat exchange area on the road of exhaust gases, most frequently, outside the boiler (of the recuperator). Thus, it is possible to additionally cool exhaust gases and use recuperation heat in an amount expressed by the following formula:

\[
Q_{rek} = V_s \cdot \rho_s \cdot c_{sp} \cdot (T_s - T_{rek})
\]

(14)
where:
Ts – temperature of exhaust gases in front of the recuperator (without recuperation),
Trek – temperature of exhaust gases behind the recuperator, other notations as above.

In fact, the final temperature of exhaust gases should not be lower than the temperature of the dew-point of exhaust gases. The temperature of dew-point is connected with the content of water in exhaust gases [4]. Taking into account the average moisture of coal and the most frequent properties of air, the temperature of the dew-point of exhaust gases is: TR ≈ 65°C. Assuming that exhaust gases are cooled down to the temperature of the dew-point TR, unit heat gains obtained from combustion of 1 kg of coal were calculated. Calculation results were placed in Figure 2.

![Figure 2](image)

**Figure 2.** Relation between unit heat gains in the recuperator, the outlet temperature of exhaust gases and the excess air number

7. **Modernization guidelines**
Organizational and technical actions should be undertaken to reduce fuel consumption. Organizational actions include the assurance of the proper process of combustion in the boiler with a minimal amount of air. Technical actions assume installation of an additional feedwater heater in the exhaust duct behind boilers to after cool exhaust gases, Figure 3.

A reduction of the temperature of exhaust gases affects a reduction of the carry-over loss. Approximately, it may be assumed that the reduction of the temperature of exhaust gases by 15°C will result in an increase of efficiency by approx. 1%. When choosing the size of the heater, the temperature of exhaust gases in front of and behind the heater and the boiler power should be taken into account. In front of the heater, there is usually a pass-through dust extractor which is a first stage of dust extraction and which protects the heater against ash erosion. Exhaust gases may be also cooled using a blast air heater. However, that is a less beneficial arrangement. The heat exchanger fixed in the exhaust duct is intended for preheating sanitary water [5].

8. **Economic effects of modernization**
The economic effect of applying the recuperator for heat recovery (economizer) from exhaust gases results from reducing heat consumption due to additional cooling of exhaust gases. A measure of economic effectiveness is a Simple Pay Bank Time (SPBT) described in the following relation:
$SPBT = \frac{N}{\Delta K}$

(15)

where:
N – outlays on implementing the undertaking,
$\Delta K$ – annual cost saving caused by reduction of heat consumption.

Figure 3. Process diagram of the boiler room with recuperator.

Financial outlays on implementing the undertaking include the purchase and assembly of equipment. A yearly cost saving results from the reduction of heat consumption due to application of a recuperation exchanger. The cost reduction is determined by the following formula:

$\Delta K = \Delta Q_a \cdot O_z + 12 \cdot Q \cdot O_m$

(16)

where:
$\Delta Q_a$ - annual reduction of heat consumption, $O_z$ - unit price of heat,
$Q_a$ - thermal power, $O_m$ - monthly constant fee for heat production.

Heat costs and heat production were determined on the basis of data from the boiler room. Yearly variability of heat production was presented in Figure 4. On the basis of the diagram (Figure 4), the maximum power for heating and hot water purposes was determined, assuming that in the summer heat is used only for preparing hot water. From the diagram, it appears that the required power for hot water purposes is approx. 300 kW. For heating purposes, it is 896.75 kW. After taking into account all data, the following was obtained:

$O_z = 18.48$ PLN/GJ - unit heat price,
$O_m = 10147.62$ PLN/(MW monthly) - monthly constant fee for heat production.
Table 2. Determination component of the levy fixed and variable rates of heat

| Component                              | Cost (PLN/year) |
|----------------------------------------|-----------------|
| Depreciation cost                      | 2706,00         |
| The costs of repairs                   | 5000,00         |
| The cost of servicing                  | 138024,00       |
| The annual fixed costs                 | 145730,00       |

Monthly fixed costs

- M-y constant fee for heat production: 10147,62 PLN/MW m-t
- Heat demand c.o.: 896,75 kW
- Heat demand c.w.: 300,00 kW
- Heat demand Σ: 1196,750 kW
- Boiler efficiency ηk: 80 %
- Calorific value of burnt coal Qw: 23 675 kJ/kg

The annual heat consumption

- Qa = jśr * Q * ta: 7525,2 GJ/year
- The annual heat consumption c.w.: 9460,8 GJ/year
- The annual heat consumption Σ: 16986,0 GJ/year
- The annual consumption of coal: 1120 Mg
- Annual cost the carbon tax: 275963,0 PLN/year

Annual electricity consumption in the boiler room

- En: 99661,0 kWh/year

Cost of electricity

- Ke = ke * En: 37871,18 PLN/year

The average price of electricity tax

- ke: 0,380 PLN/kWh

Average price of heat

- Oz = kc = ( Ko + Ke ) / Qa: 18,48 PLN/GJ

Figure 4. Yearly variability of heat production
To determine effects, it was assumed that the excess air number $\lambda = 1.6$ and the reduction of the temperature of exhaust gases was made by $50^\circ C$, the maximum thermal power of the boiler room is $1200$ kW.

An annual heat saving is determined by the following relation:

$$\Delta Q = V_{1s} \cdot m_{pal} \cdot C_{sp} \cdot \Delta T_s \cdot t_a$$  \hspace{1cm} (17)$$

where:

- $Q_{wyd}$ – carry-over loss,
- $V_{1s} = 10.407 \, m^3 / kg$ – unit amount of exhaust gases,
- $m_{pal}$ – amount of burnt fuel,
- $C_{sp} = 1.3 \, kJ / m^3$ - volumetric thermal capacity of exhaust gases with constant pressure,
- $\Delta T_s$ – reduction of the temperature of exhaust gases,
- $t_a = 8760 \, h$ – annual time of operation.

After replacing the above with the values, the following was calculated:

fuel consumption

$$m_{pal} = \frac{Q_u}{\eta_k \cdot Q_w} = \frac{1200}{0.65 \cdot 23675} = 0.0780 \, kg / s$$  \hspace{1cm} (18)$$

annual heat saving

$$\Delta Q_a = V_{1s} \cdot m_{pal} \cdot C_{sp} \cdot \Delta T_s \cdot t_a = 10,407 \cdot 0,0780 \cdot 1,3 \cdot 50 \cdot 8760 \cdot 3600 \cdot 10^{-6} = 1663.5 \, GJ / a$$  \hspace{1cm} (19)$$

thermal power of recuperator

$$Q = V_{1s} \cdot m_{pal} \cdot C_{sp} \cdot \Delta T_s = 10,407 \cdot 0,0780 \cdot 1,3 \cdot 50 = 52.75 \, kW$$  \hspace{1cm} (20)$$

Thus, annual savings are

$$\Delta K = \Delta Q_a \cdot Q_z + 12 \cdot Q \cdot Q_m = 1663.5 \cdot 18.48 + 12 \cdot 0.05275 \cdot 10147.62 = 37163.85 \, zl / a$$  \hspace{1cm} (21)$$

An expected cost of regeneration exchanger (recuperator) with the indicated thermal power will be approx. PLN 35 000. Taking into account assembly costs, an expected cost of modernization (financial outlays) will be: $N = PLN 45 000$.

The Simple Pay Back Time (SPBT) will be:

$$SPBT = \frac{N}{\Delta K} = \frac{45000}{37163.85} = 1.21 \, years$$

9. Conclusions

On the basis of data obtained from the boiler room user and the analysis performed, it was found that combustion in the boiler is with too high excess air numbers, which results in low efficiency of the boiler room. Measured temperatures of exhaust gases are relatively low; however, it is caused by an excessive amount of air and not sufficient cooling of exhaust gases.

Therefore, it is possible to increase the efficiency of boilers after undertaking the following action:

- reducing the amount of the excess air during coal combustion in the boiler to the value not exceeding $1.8 - 2.0$ (meters of oxygen content in exhaust gases outside the boilers),
- applying additional heat exchanger (economizer) with the power of $50 – 80 \, KW$ in the exhaust duct outside the boilers, intended for preheating hot water.

An expected increase of the efficiency of the boilers – approx. 4 %, whereas an expected increase of the average annual efficiency of boiler room operation is approx. 9.5 %. 
References

[1] Roman Zarzycki; “Heat exchange and mass movement in environmental engineering”. Wydawnictwo Naukowo-Techniczne, Warszawa, 2010.

[2] Bukowska M., Nowak K., Rabczak S., „Emission of air pollutants in the hot water production”, Technologia Wody, volume 51, issue 1, pages 38-44, 2017.”

[3] Krystyna Mizielińska, Jarosław Olszak – „Gas and oil heat sources of low power”. Oficyna Wydawnicza Politechniki Warszawskiej, 2011.

[4] Kazimierz Buczek – „Steam and water boiler operator”. Wydawnictwo Kabe, 2009.

[5] Wojdyga K., Niemyski O., „Hydraulic analysis for a district heating system supplied from two CHP plants”. Energy and Buildings 54 (2012) p.81-87. Elsevier 2012.