Design and Simulation of Waste Heat Recovery System for Heavy Oil Preheating in Dashen Brewery Company

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Abstract. Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use. In this paper an effort has made about waste heat recovery system design for Dashen Brewery Company. It is tried to identify source of waste heat and design a new plant layout for the purpose of waste heat recovery. In Dashen brewery company heat is basically lost at the boiler, process water heater pipe and Wort kettle chimney. The company uses heavy oil for boiler and this oil needs to be heated to convert in to light oil for better ignition. To heat this heavy oil the company still uses steam but this paper design a system to use exhaust steam at Wort kettle for heating of heavy oil instead of steam. Tasks performed in in this paper are direct temperature measurement on the waste heat source, heat loss calculation on the chimney and construct appropriate waste heat recovery routine. Head loss and other basic flow parameters had been considered. Furthermore from the analytical model it is possible to determine the amount of power needed to the boiler which is 56.947 MW from 6220 L/h of furnace oil and from exhaust steam 17.5 kW is gained which can burn 190 L/h of fuel; in terms of birr it is possible to save 456 L/day × 19.5 birr which is 8892 birr per day.

Keywords: Waste heat recovery · Head loss · Shell and tube heat exchange

1 Introduction

Industrial waste heat refers to energy that is generated in industrial processes without being put to practical use. Sources of waste heat include hot combustion gases discharged to the atmosphere, heated products exiting industrial processes, and heat transfer from hot equipment surfaces. The exact quantity of industrial waste heat is poorly quantified, but various studies have estimated that as much as 20 to 50% of industrial energy consumption is ultimately discharged as waste heat.

In Dashen Brewery Company there are three main sections; these are utility section or power house section, brew house and process flow section. Among these sections

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the utility section is the back bone of the company because it is a source of steam, process water and purified carbon dioxide which have significant impact on production. In this paper an effort has made about waste heat recovery system design for brewery companies. It is try to identify source of waste heat and design a new plant layout for the purpose of waste heat recovery. Waste heat recovery power generation (WHRPG) started in the 1980s in Japan, within the cement industry. It grew mainly in Eastern Asia (China and Japan) from 2000, at a time when energy prices increased [1].

2 Literature Review

There were many works previously carried out on the Shell and tube heat exchangers. Some of them are listed here: Presented in this paper are how to design, construct and test the waste heat recovery pipe for air-preheating used for furnace in a hot brass forging process, Yodrak et al. [6]. Muhammad Zeeshan et al. [7] carried out a Waste Heat Recovery system for Electric Power Generation from Cement Industry. Those scholars is therefore try to recover waste heat from Fecto Cement Plant installing a 6 MW power in order to save energy, reduce heat consumption and production cost of waste heat recovery plant. Selvaraj and Varun [8] both scholars try to recover waste heat during metal casting. The waste heat source in this paper is the knocked out casting which has heat energy stored in it and is wasted into atmosphere as the casting cools down in the shop floor. Remeli et al. [9] is also investigating a system how to generate power using heat pipes and thermo-electric generators.

Saidawat et al. [11] carried out the research work on the power generation from the waste heat extracted through clinker production in the cement industry. This study includes the power generation calculation for a cement plant and the different methodologies used to generate power (Table 1).

3 Materials and Methods

For the three sample tests above there is some variation of surface temperatures due to room temperature variation or seasonal variations. Therefore consider the April temperature data, and temperature variation along the chimney is uniform, and then average surface temperature for April is 109 °C (Fig. 1).

Table 1. Three months average surface temperatures

Tests	Months	Chimney's average surface temperature T_S in °C
1	Feb	111
2	April	109
3	Jun	108

Fig. 1. Infrared temperature measurement taken on wort kettle

3.1 Energy Supplied for Wort Boiling

Wort input to the wort kettle = 345 hl, Cast Wort to the whirlpool = 315 hl, Exhaust wort in the form of steam = 30 hl. From the input and output data it is possible to know the amount of heat energy wasted in the form of waste gas through the chimney.

Amount of wort changed to steam (wasted through the chimney) = Amount of boiling wort in the kettle- amount of boiled wort out to whirlpool (storage tank for cast wort) = 345 hl – 315 hl = 30 hl. Then conventionally to boil 1 hl of cast wort about 14 kWh of energy is required. Therefore for 30 hl of steam 420 kWh energy is required. The wort in the wort kettle tanker is boiled for 1–2 h and the hot finished casting wort is produced. In Dashen brewery factory the time required for cast wort is 2 h. The energy used can be expressed in terms of Power. Therefore power is the given energy per total hours which is 210 kw [2].

3.2 Mathematical Modeling

For a hallow pipe exposed to convection environment on its outer and inner surfaces, the overall heat transfer would be expressed by

$$q = \frac{T_{1\infty} - T_{2\infty}}{\frac{1}{h_{1\infty}A_i} + \frac{\ln(r_0/r_i)}{2\pi kL} + \frac{1}{h_{2\infty}A_0}} = h_{1\infty}A(T_1 - T_{1\infty}) \tag{1}$$

Where

T1∞	temperature of the steam in the chimney
T2∞	room temperature at the outer surface
ro	outer radius of the chimney
ri	inner radius of the chimney
L	height of the chimney

By direct measurements Outside Diameter of exhaust chimney D0 = 400 mm = 0.4 m, Inside diameter of exhaust chimney Di = 380 mm = 0.38 m, Thermal conductivity of stainless steel k = 16.2 W/m°C (Figs. 2 and 3).

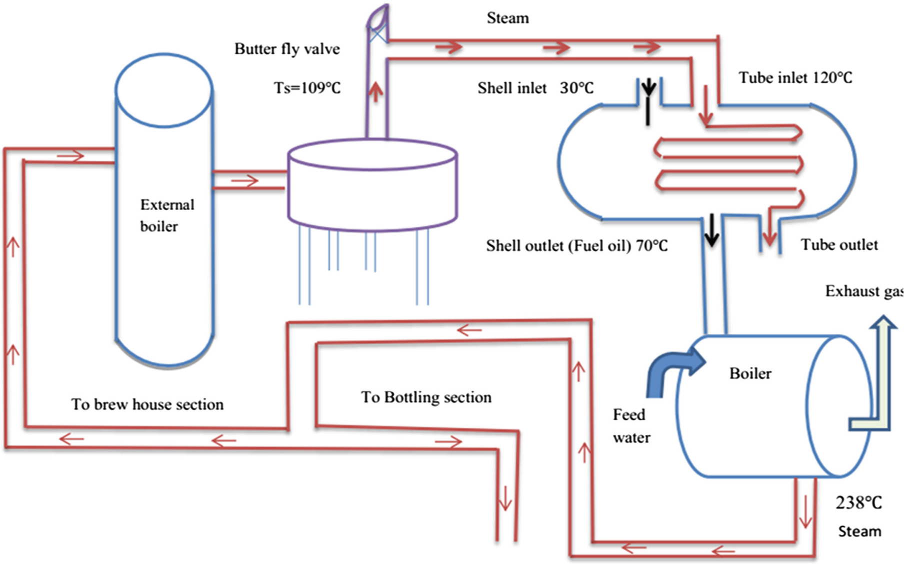

Fig. 2. Conceptual layout of the designed system

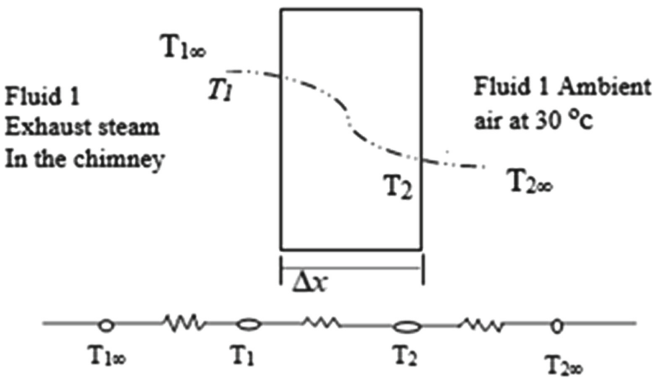

Fig. 3. Resistance circuit on the chimney

Fluid mean temperature on the chimney is:

$$T_f = \frac{T_s + T_\infty}{2} = \frac{109 + 30}{2} = 69.5\,^\circ\mathrm{C}$$

Properties of steam at mean temperature: Thermal conductivity k = 0.0296w/m.°C, Specific heat capacity Cp = 1.0075 kJ/kg, Density = 1.009 kg/m^3, Kinematic, viscosity = 2.00 10 − 5 m^2/s, Prandtl number Pr = 0.7, Reynolds number,

$$\mathrm{Re} = \frac{u \times D}{\nu} \tag{2}$$

Nusselt Number, $Nu = \frac{hD}{k}$

Volumetric flow rate, $\dot{V} = \frac{q}{c_p \rho dT} = 2.61\ \mathrm{m^3/s}$, where q = 210 kW

Velocity of the steam, $u = \frac{4\dot{v}}{\pi D^2} = 23\ \mathrm{m/s}$, $\mathrm{Re} = \frac{u \times D}{\nu} = 438079$, (Turbulent)

Convective heat transfer coefficient h at the outer surface, $h_{2\infty} = \frac{Nu \times k}{D} = 47\ \mathrm{W/m^2.k}$.

From Fourier's law of heat conduction equation $q_{net} = \frac{k \times (T_{in} - T_{out})}{\Delta x}$

$$18.56\,\mathrm{kW} = \frac{16.2 \times (\mathrm{T_1} - 109)}{0.01}, \mathrm{T_1} = 120\,^{\circ}\mathrm{C}$$

Assume inside surface temperature is equal to the fluid temperature, then $T_{1\infty} = 120\,^{\circ}\mathrm{C}$ inside convective heat transfer coefficient, $h_{1\infty} = \frac{q}{A_s(T_{1\infty} - T_{2\infty})} = 19\ \mathrm{W/m^2.k}$

3.3 Design of Waste Heat Recovery Route Equipment

Pipe line sizing: schedule 40 is most standard schedule and based on given pressure drop take Nominal Bore 10 in. (DN 250 mm), Outside Diameter 273.0 mm and thickness of the pipe t = 9.3 mm and internal diameter 254.4 mm (Fig. 4).

$$\text{Mass flow rate of steam } m = \frac{q_{net}}{h_{fg}} = 0.84\ \mathrm{kg/s}$$

$$\text{Average velocity of exhaust steam } u_a = \frac{4\,\dot{v}}{\pi D^2} = 4.24\ \mathrm{m/s}$$

$$\mathrm{Re} = \frac{4.24 \times 0.254}{2 \times 10^{-5}} = 53874\ \text{(turbulent)}$$

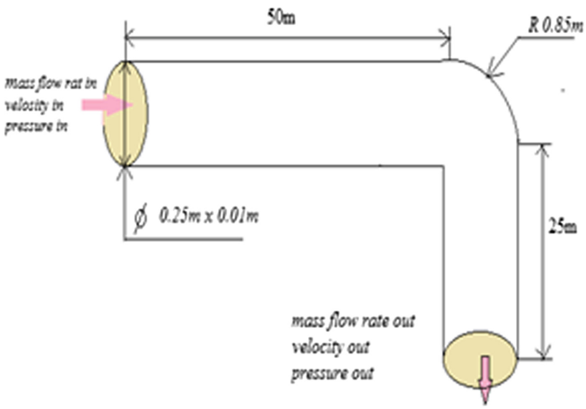

Fig. 4. Dimensions of pipeline route

3.4 Thermal Insulation

Insulation is defined as those materials or combinations of materials which retard the flow of heat energy by performing one or more of the following functions:

Conserve energy by reducing heat loss or gain, Control surface temperatures for personnel protection and comfort, Facilitate temperature control of a process, Prevent vapor flow and water condensation on cold surfaces (Fig. 5).

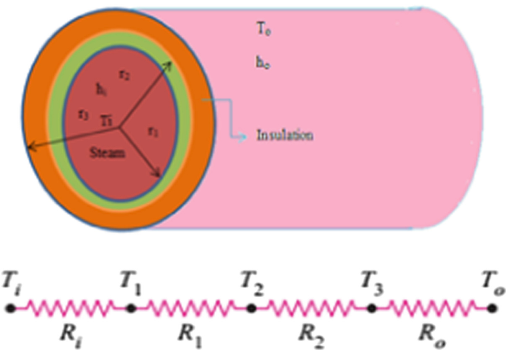

Fig. 5. Thermal insulation and circuit diagram

It should be realized that insulation does not eliminate heat transfer; it merely reduces it. The thicker the insulation, the lower the rate of heat transfers but also the higher the cost of insulation. Therefore, there should be an optimum thickness of insulation that corresponds to a minimum combined cost of insulation and heat lost. The determination of the optimum thickness of insulation is illustrated in Fig. 6.

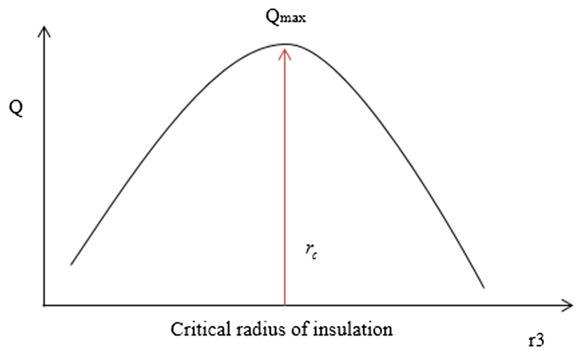

Fig. 6. Determination of the optimum thickness of insulation

Properties: Pipe internal temperature Ti = 120 °C, thermal conductivity of stainless steel material k = 16.2 w/m°C, Internal diameter di = 0.254 m, Maximum allowable temperature of the outer surface insulation T_0 = 30 °C.

Thermal conductivity of cellular glass insulation kinsulation = 0.038 w/m°C, Internal and external convective heat transfer coefficients ho = 47 w/m^2, hi = 19 w/m^2.

Letting r_3 represent the outer radius of the insulation, the areas of the surfaces exposed to convection for an L = 50 m long section of the pipe become

$$\begin{aligned} A_2 &= \pi D0L = \pi \times 0.273 \times 50 = 42.88\,\mathrm{m}^2 \; A1 = \pi DinL \\ &= \pi \times 0.254 \times 50 = 39.89\,\mathrm{m}^2 \\ A_3 &= 2\pi \times r_3 \times 50 = 354 r_3\,\mathrm{m}^2 \end{aligned}$$

Then the individual thermal resistances are determined to be

$$\begin{aligned} R_i &= R_{conv\,1} = \frac{1}{h_i A_1} = 1.32 \times 10^{-3}\,{}^\circ\mathrm{C/w} \; R_1 = R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k L} \\ &= 1.4 \times 10^{-5}\,{}^\circ\mathrm{C/w} \\ R_2 &= R_{ins} = \frac{\ln(r_3/r_2)}{2\pi k L} = 0.0837 \ln(r_3/0.136)\,{}^\circ\mathrm{C/w} \\ R_0 &= R_{conv\,2} = \frac{1}{h_0 A_3} = \frac{6 \times 10^{-5}}{r_3}\,{}^\circ\mathrm{C/w} \\ \dot{Q} &= \frac{T_i - T_0}{R_{tot}} = \frac{T_I - T_0}{\left(\frac{1}{2\pi r_1 h L} + \frac{\ln(r_2/r_1)}{2\pi k L} + \frac{\ln(r_3/r_2)}{2\pi k L} + \frac{1}{2\pi r_3 h L}\right.} \\ \dot{Q} &= \frac{T_i - T_0}{R_{tot}} = \frac{120 - 30}{(133.4) \times 10^{-5} + 0.0837 \ln\left(\frac{r_3}{0.136}\right) + \frac{6\times 10^{-5}}{r_3}} \end{aligned} \tag{3}$$

Hence the outer surface temperature of insulation is assumed to be 35 °C; the rate of heat loss over the routine can also be expressed as:

$$Q = \frac{T_3 - T_0}{R_0} = \frac{35 - 30}{\frac{6\times 10^{-5}}{r_3}} = 37790.7\, r_3 \tag{4}$$

From Eq. (1) to find the value of thickness t for which Q maximum should be equated to zero or denominator should be minimum.

$$\frac{d}{dr_3}(Q) = \left(\left(\frac{1}{2\pi\, r_1 h\, L} + \frac{\ln(r_2/r_1)}{2\pi k L} + \frac{\ln(r_3/r_2)}{2\pi k L} + \frac{1}{2\pi\, r_3 h\, L}\right) = 0\right.$$

Then by rearranging critical radius of insulation for cylinder becomes

$$r_3 = \frac{k}{h} = \frac{16.2}{47} = 0.344\,\mathrm{m}$$

The rate of heat loss from Eq. (2) becomes $Q_{loss} = 37790.7 r_3 = 13\,\mathrm{kW}$. Then the net heat transfer rate $Q_{net} = 18.56\,\mathrm{kW} - 13\,\mathrm{kW} = 5\,\mathrm{kW}$.

3.5 Major Loss, Minor Loss and Pressure Drop in Pipe Flow

Major loss across horizontal length, $h_f = 2 \times 0.0105 \frac{50}{0.254}\left(\frac{4.24^2}{9.81}\right) = 7.57\,\mathrm{m}$

Similarly for major head loss h_f across vertical pipe length is

$$h_f = 2 \times 0.0105 \frac{25}{0.254}\left(\frac{4.24^2}{9.81}\right) = 3.78\,\text{m}$$

Minor losses:

$$k_m = 0.015, h_m = 0.015 \frac{4.24^2}{2 \times 9.81} = 0.014\,\text{m}\ \Delta P = \rho g \times loss$$
$$= 1.009 \times 9.81 \times 0.014 = 1.4\,\text{pa}$$

Allowable Pressure drop: The pressure drop across the horizontal pipe can be calculated using the following equation:

The pressure drop $\Delta P = \rho g \times loss = \gamma h_f$, here, the specific weight $\gamma = \rho \times g$

$\Delta p = 2f \frac{L}{D}(\frac{\rho u^2}{g}) = \rho g h_f = 1.009 \times 9.81 \times 5.57 = 63\,\text{Pa}$ Similarly for vertical length of pipe pressure drop p = 38.4 pa, Total pressure drop p = 102.8 pa (Fig. 7).

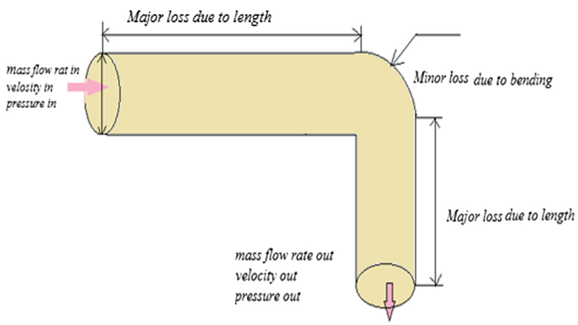

Fig. 7. Head losses

3.6 Shell and Tube Heat Exchanger Design for Preheating

Steam side properties
Inlet temperature of steam pipe line Tin = 120 °C, Inlet pressure of steam Pin = Psat = 198.5 kPa, Specific heat capacity of steam Cp = 1.075 kJ/kg.k, Pressure drop p = 102.4 pa, Mass flow rate of steam ms = 0.84 kg/s, Fouling factor for heavy oil 0.0009 (W/m^2°C) − 1, Fouling factor for steam 0.0001(W/m^2°C) − 1

Shell side Furnace oil properties, Inlet temperature of furnace oil tin = 30 °C Output temperature furnace tout = 70 °C, Density of furnace oil = 800 kg/m^3, Specific heat capacity of oil = 2.3 kJ/kg.k, Amount of fuel consumed = 6220 L/h, Mass Flow rate of fuel consumed mf = 1.382 kg/s, Specific heat capacity oil = 2.3 kJ/kg.k.

Heat exchanges between Steam-to-heavy fuel oil and overall heat transfer coefficient will be in the range 50–200 W/m^2°C so let's start with U = 100 W/m^2°C (Fig. 8).

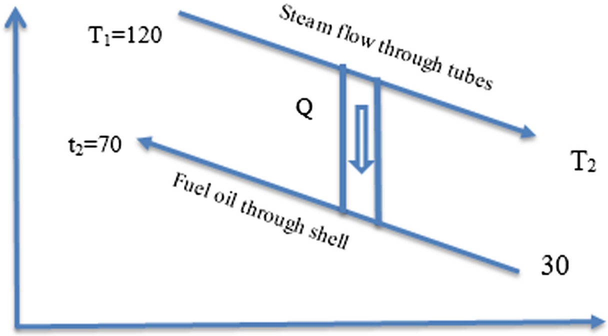

Fig. 8. Schematic diagram of counter flow heat exchanger

3.7 Heat Exchanger Type and Dimensions

The general tube layout is d0 = 20 mm, t = 2.0 mm, din = 16 mm, L = 2.44 m tubes on square pitch (Pt = 1.25d0) Area of one tube = 0.314 m^2

$$\text{Number of tubes Nt} = \text{As/At} = 2.99/0.314 = 9.5 \text{ say } 10$$

$$\text{Volume flow rate oil} = \frac{m}{\rho} = 0.84/1.009 = 0.00105\,\text{m}^3/\text{s}$$

$$\text{Tube side velocity Ut} = \frac{volume\ flow\ rate}{area\ per\ pass} = \frac{0.00105}{0.0001} = 10.5\,\text{m/s}$$

Overall heat transfer coefficient

$$\frac{1}{U_0} = \frac{1}{h_i} \times \frac{A_0}{A_i} + \frac{R_f, i}{A_i} + \frac{A_0 \ln(d_0/d_i)}{2\pi k L} + \frac{1}{h_s} + \frac{R_{f,o}}{A_0}$$

$$R_{f,i} = 0.0009\,\text{m}^2.°\text{C}/w \quad R_{f,0} = 0.0001\,\text{m}^2.°\text{C}/w$$

The need for calculating the overall heat transfer coefficient is to check whether the initial estimated value is safe or not, if it is safe proceed unless revise the estimation heat transfer coefficient.

4 Results and Discussion

From mathematical modeling of the waste heat recovery or exhaust steam it is possible to extract 17.5 kW power which can save 456 L/day of fuel. It increases the annual income of the company. But this result may partly affected due to measurement inaccuracy or improper material calibration because during testing of chimney temperature, the reading of thermocouple and infrared thermometer were having different reading values.

5 Conclusion

The main objective of this study was to design and simulate the waste heat recovery system for oil preheating system in Beverage Companies. Initially the study attempts to start the new designing process by constructing system design layout and proper material selection for each component. Designing of the system for exhaust steam were selected by comparing with exhaust flue gases at the boiler. At the beginning the validity of the study was checked by measuring the amount of temperature in the chimney. At the wort kettle chimney the amount of surface temperature was 109 °C which is enough to reheat heavy oil. Beside temperature measurement, amount of power wasted to the environment was determined. Wort input to the wort kettle = 345 hl, Cast Wort to the whirlpool = 315 hl, Exhaust wort in the form of steam = 30 hl. From the input and output data it is possible to know the amount of heat energy wasted in the form of waste gas through the chimney. Amount of wort changed to steam (wasted through the chimney) = Amount of boiling wort in the kettle- amount of boiled wort out to whirlpool (storage tank for cast wort) = 345 hl – 315 hl = 30 hl. According to company's manual to boil 30 hl exhaust wort with steam 210 kw power is required. The overall dimensions of the pipe line are 50 m horizontal length, 25 m vertical length, internal diameter 250 mm with thickness 10 mm and material insulation is needed with proper dimension. On the pipe line the main parameters determined first were head loss and pressure drop because these parameters are determinant factors for fluid flow through pipes. At the end of the pipe line the net amount of power was determined and made it ready for input of the preheater. Important assumptions were made during pipe flow, like steady state flow condition. At the preheater heat exchanging process between heavy oil and exhaust steam were taking place. Heavy oil has initial temperature 30 °C and reaches 70 °C. Quantity of Heavy fuel oil supplied to the preheater is 6220 L/h or 1.382 kg/s.

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