Performance Evaluation of Recuperative Heat Exchanger in Rotary Furnace

A.T. Oyelami, S.B. Adejuyigbe and M.A. Waheed

ABSTRACT

The need for increase in thermal efficiency, fuel economy and environmental friendliness of operation necessitated the usage of heat exchanger for the recycling of waste gases in the operation of a melting furnace. This work therefore focuses on the analysis of the developed recuperative heat exchanger for recycling the waste heat through modelling and simulation of heat distribution in the exchanger. This was done by modelling a conjugate heat transfer phenomenon through Turbulent Fluid-Thermal Interaction. The periodicity of the flow is taken into cognizance because a heat exchanger where the flow is fully developed is being modelled. Due to the inherent difficulty in making a periodic configuration converge from a homogeneous initial guess, an initial calculation with constant inlet velocity and fixed outlet pressure is first performed. The logarithmic wall function boundary condition for turbulent flow is used to model the solid-fluid interfaces. An algebraic relationship—the logarithmic wall function—describes the momentum transfer at the solid-fluid interface. This makes the solid-fluid boundaries in the model actually represent lines within the logarithmic regions of the boundary layers. Total number of triangular elements used during descritization/meshing is 21,892 while the number of degrees of freedom solved for was 183,200.

***Index Terms*— Conjugate-Heat-Transfer, Descritization, Fluid-Thermal-Interaction, Recuperative-Heat-Exchanger**

I. INTRODUCTION

Aluminium recycling can be said to be the traditional use of rotary furnaces. These furnaces are initially used for the production of Aluminium from scraps (Krivadin and Markov 1980). Today, besides their usage in processing a wide variety of scraps materials, from coated strip to used beverage cans and furnace dross having different composition and content of Aluminium, rotary furnaces now play a major role in the recycling of cast iron. In addition to this, it is also being adapted for the production of Austempered Ductile Iron (ADI) (EMDI 2008). A typical rotary furnace consists of a cylindrical, refractory lined, steel drum, supported by a structural steel frame, in between two conical frustums.

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The conical frustums are of two different end-diameters. The primary heat source of the furnace could either be a natural gas or a liquid–fuel burner. The burner is usually positioned at the smaller-diameter end of the frustum, while the exhaust, in this case aligned with the heat exchanger, is positioned at the larger-diameter end.

Fuel costs are normally taken as a small fraction of the total manufacturing cost, and in the past, fuel economy has often been considered unimportant. However, with increasing costs and decreasing availability of fuels, there is now a considerable financial and a strong conservational incentive to watch fuel economy aspects. For the purpose of fuel economy and increase in thermal efficiency, a radiation type recuperative heat exchanger will be appropriate for a medium sized rotary furnace. For an industrial rotary furnace however, there is the need for a significant increase in the efficiency of the heat exchange process to ensure high thermal efficiency and significant fuel consumption reduction through the usage of a hybrid recuperator comprising of both radiant and convective sections. It is therefore desirable to evaluate the effects of heat exchanger incorporation in rotary furnace both quantitatively and qualitatively.

II. HEAT EXCHANGER INCORPORATION IN ROTARY FURNACE

The effects of incorporation of heat exchanger in the operation of a liquid-fuel fired furnace can be expressed mathematically. For normal operation, the working space of a furnace must be supplied every hour with a specified quantity of heat, which includes the physical heat of preheated air (Q_{ph}) and the chemical heat of fuel (Q_{ch}), i.e

$$Q_{\Sigma} = Q_{ch} + Q_{ph} \qquad (1)$$

As can be inferred from equation (1), with Q_{Σ} taken as being constant, Q_{ch} can be diminished by increasing Q_{ph}. In other words, utilization of the heat of waste gases can save fuel. The quantity of saved fuel is dependent on the degree of heat utilization.

Apart from saving fuel, preheating of air increases the calometric temperature of combustion. This also forms one of the principal reasons for recuperation of the waste heat. At a constant value of the lower calorific value of the liquid fuel (i.e at Q_w^l = constant), an increase of Q_{ph} results in an increased temperature of combustion.

Additional advantage of waste heat recovery through recuperative heat exchanger is environmental friendliness of the operation. The presence of hazardous Carbon Monoxide is almost totally eliminated. This is evidenced in the blue flame obtainable from the operation, signifying complete combustion. Lagging of the ducting pipes also serves as noise absorber (besides its primary role of minimizing heat loss) thereby making the operating environment friendlier .

A. Recuperator Specific Design Calculations (For a 300kg Rotary Furnace)

The first step taken in the design of the recuperator is to determine the process conditions. This includes the analysis of the fluid (waste gases), determination of the temperatures of operation and the pressures drop. The second step was to obtain all the required physical properties over the temperature and pressure ranges of interest. The third stage involved choosing the type of recuperator needed while the fourth and the last stage was the real design and re-design (whenever necessary) to get the necessary dimensions and ensure that the process specifications with respect to both heat transfer and pressure drop are met (Robert and Don, 1998)

B. Determination of Operational Temperatures

The average temperature of the waste gases leaving the furnace is 1050°C and that of the surrounding air is taken to be at the room temperature of 27°C. Since one of the main purposes of designing the recuperator is to increase the temperature of combustion significantly, it is pertinent to determine the existing temperature of combustion (for the existing rotary furnace without recuperator), choose a new desired temperature to be accomplished as a result of the recuperator incorporation and calculate the final temperatures of both the air and the waste gases necessary for the attainment of the new combustion temperature.

C. Calculation of Final Temperature of Air Expected from the Recuperator

The maximum combustion temperature obtainable from the rotary furnace without a recuperator is 1500^{0}C. Now, it is desired to raise the temperature to 2000 ^{0}C through the incorporation of the recuperator. This is the basis used for calculating the final temperature of the air to be preheated in order to have the furnace temperature raised to 2000 ^{0}C.

The Calorific temperature t_c is the ratio of Combustion Temperature t_{comb} to the efficiency η

$$t_c = \frac{t_{comb}}{\eta} \quad , \qquad t_c = \frac{2000}{0.7} = 2857.14^0 C$$

Enthalpy of combustion products at 2857.14^{0}C is

I_{CO_2} = 0.10686 x 6303.53 = 673.60 kJ/m^3

I_{N_2} = 0.74429 x 3786.09 = 2817.95 kJ/m^3

I_{O_2} = 0.033 x 4014.29 = 132.47 kJ/m^3

I_{H_2O} = 0.11576 x 5076.74 = 587.68 kJ/m^3

I_0 = 4211.70 kJ/m^3

$$Q_0^l = I_0 \times v_{wg} = 4211.70 \times 12.324 = 51905 \text{ kJ/kg}$$

$$Q_0^l = Q_w^l + Q_{ph}^l$$

where $Q_{ph}^l = \dfrac{Q_a}{B}$ and $Q_w^l = \dfrac{Q_{ch}}{B}$ = 44000kJ/kg

$$Q_{ph}^l = Q_0^l - Q_w^l = 51905 - 44000 = 7905 \text{kJ/kg}$$

$$t_a = \frac{Q_a}{B} \cdot \frac{1}{n c_a v_a}$$

where

$\dfrac{Q_a}{B} = Q_{ph}^l = 7905 \times 10^3$ J/kg; n = 1.2;

c_a (at 400^{0}C) = 1.3302 x 10^3 J/m^3 ^{0}C; v_a = 11.481m^3/kg

$$t_a = 7905 \times 10^3 . \frac{1}{1.2 \times 1.3302 \times 10^3 \times 11.481} = 431.34^0\text{C}$$

Thus the final temperature of air t_a^f is 450^{0}C

D. Calculation of final Temperature of Waste Gases Expected from the Recuperator

This is done through the analysis of the heat balance in the recuperator. On the assumption that the recuperator is perfectly gas-tight, the heat balance is analyzed based on further assumption that 10 per cent of the heat supplied is lost to the surroundings.

$$0.9 V_{wg} (c_{wg}^{in} t_{wg}^{in} - c_{wg}^f t_{wg}^f) = V_a (c_a^f t_a^f - c_a^{in} t_a^{in})$$

(Krivadin and Markov 1980)

$t_{wg}^{in} = 1050^0 C$, $t_a^f = 450^0 C$, $t_a^{in} = 27^0 C$

at 1050^{0}C

$$c_{CO_2} = 0.127 \times [2.2266 + \frac{1050-1000}{1100-1000}(2.2593 - 2.2266)] = 0.2849$$
kJ/m^3 ^{0}C

$$c_{N_2} = 0.736 \times [1.3938 + \frac{1050-1000}{1100-1000}(1.4056 - 1.3938)] = 1.0302$$
kJ/m^3

$$c_{H_2O} = 0.137 \times [1.7133 + \frac{1050-1000}{1100-1000}(1.7397 - 1.7133)] = 0.2365$$
kJ/m^3

$c_{wg}^{in} = c_{\sum}$ = 1.5516 kJ/m^3

at 700^{0}C

c_{CO_2} = 0.127 x 2.1077 = 0.2677 kJ/m^3 c_{N_2} = 0.736 x 1.3553 = 0.9975 kJ/m^3

c_{H_2O} = 0.137 x 1.6338 = 0.2279 kJ/m^3

$c_{wg}^f = c_{\sum}$ = 1.4931 kJ/m^3

The specific heat capacity of air is

at 27^{0}C

$$c_a^{in} = 1.3009 + \frac{27-0}{100-0}(0.3051 - 1.3009) = 1.03203 \text{kJ/m}^3 \, ^0\text{C}$$

at 450^{0}C

$$c_a^f = 1.3302 + \frac{450-400}{500-400}(1.3440 - 1.3302) = 1.3371 \text{kJ/m}^3 \, ^0\text{C}$$

Using $1\frac{1}{2}$ hrs as the standard cycle period

V_{wg} =9324.72/1.5 = 6217m^3/h
V_a = 8684.52/1.5 = 5790m^3/h

$$0.9 \times 6217(1.5516 \times 1050 - 1.4931 t_{wg}^f) = 5790(1.3371 \times 450 - 1.03203 \times 27)$$

$$5595.3(1629.18 - 1.4931\, t_{wg}^f) = 3322476.8$$

$$t_{wg}^f = (1629.18 - \frac{3322476.8}{5595.3}) \times \frac{1}{1.4931} = 693.44^0\text{C}$$

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E. Average Temperature Difference in the Recuperator

Using the logarithmic mean temperature difference (LMTD) approach (Kern 1984)

$$\Delta t_{av} = \frac{(t_{wg}^{in} - t_a^{in}) - (t_{wg}^{f} - t_a^{f})}{\ln \dfrac{t_{wg}^{in} - t_a^{in}}{t_{wg}^{f} - t_a^{f}}} = \frac{(1050-27)-(693.44-450)}{\ln \dfrac{1050-27}{693.44-450}} = 543^0 C$$

F. Determination of the Basic Design Parameters

A typical radiant recuperator has an appreciable thermal load of the heating surface ranging from 250 to 335 MJ/(m^2.h) **(Alan 1984).** Taking the average of the two, the expected thermal load is

(250+335)/2 = 292.5MJ/(m^2.h) = 292.5 x 10^6/3600 = 81250W/m^2

Heat transfer coefficient is the thermal load per unit temperature.

Therefore, Heat Transfer of the heating surface is

$$K_c = \frac{ThermalLaod}{t_{wg}^{in}},$$

K_c = 81250/1050 = 77.38 W/m$^{2\,0}$C

This is the clean overall heat transfer coefficient, which shows that dirt has not been taken into account. The design overall coefficient is the dirty overall coefficient K_d.

$$\frac{1}{K_d} = \frac{1}{K_c} + R_d \quad \text{or} \quad K_d = \frac{K_c}{1 + R_d K_c}$$

(Oyelami and Adejuyigbe, 2007)

For diesel combustion products, R_d = 0.00175 m$^{2\,0}$C/W

$$K_d = \frac{77.38}{1 + 0.00175 \times 77.38} = 68.15 \text{ W/m}^{2\,0}\text{C}.$$

The heating surface covered by the waste gases before reaching t_{wg}^{f} is $A = \dfrac{Q}{K_d \Delta t_{av}}$

From which $A = \dfrac{Q}{K_d \Delta t_{av}} = \dfrac{46145.43}{68.15 \times 543} = 1.247 m^2$

III. QUANTITATIVE EFFECTS OF WASTE HEAT RECUPERATION

This centres on the quantitative resultant effects of recuperator-incorporation in the design and operation of the rotary furnace system. Particular emphasis is on the coefficient of heat utilization and on the rate of fuel consumption.

A. Increase in Coefficient of Heat Utilization

There are two forms of coefficient of heat utilization. They are the coefficient of useful heat utilization (η_{uhu}) and coefficient of total heat utilization (η_{hu})

$$\eta_{uhu} = \frac{Q_{ch} + Q_{ph} - Q_{wg} - Q_{los}}{Q_{ch} + Q_{ph}}$$

$$\eta_{hu} = \frac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}}$$ **(Krivadin and Markov 1980)**

For the purpose of finding the effect of recuperator incorporation on the coefficient of heat utilization in the rotary furnace, the *coefficient of total heat utilization* is used.

Without recuperator incorporation,

Q_{ph} is zero; $\eta_{hu}^{/} = \dfrac{Q_{ch} - Q_{wg}}{Q_{ch}}$

Using fuel consumption rate of 25litres/h (= $\dfrac{25 \times 10^{-3}}{10.44}$ = 2.395×10^{-3} kg/h)

$Q_{ch} = BQ_w^l = 2.395 \times 10^{-3} \times 44000 \times 10^3$ = 105.38×10^3 J/h

$Q_{wg} = Bv_{wg}c_{wg}t_{wg} = 2.395 \times 10^{-3} \times 12.324 \times 1493.1 \times 1050 =$ 46.27×10^3 J/h

$$\eta_{hu}^{/} = \frac{105.38 \times 10^3 - 46.27 \times 10^3}{105.38 \times 10^3} = 0.5609$$

With recuperator incorporation,

$$\eta_{hu} = \frac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}}$$

$Q_{ch} = BQ_w^l = 2.395 \times 10^{-3} \times 44000 \times 10^3$ = 105.38×10^3 J/h

$Q_{wg} = Bv_{wg}c_{wg}t_{wg} = 2.395 \times 10^{-3} \times 12.324 \times 1493.1 \times 693.44 =$ 30.56×10^3 J/h

$Q_{ph} = Bc_a t_a n v_a = 2.395 \times 10^{-3} \times 1337.15 \times 450 \times 1.2 \times 11.481$ $= 19.86 \times 10^3$ J/h

$$\eta_{hu} = \frac{105.38 \times 10^3 + 19.86 \times 10^3 - 30.56 \times 10^3}{105.38 \times 10^3 + 19.86 \times 10^3} = 0.7560$$

The percentage increase in the coefficient of heat utilization is

$$\%Increase\,\eta_{hu} = \frac{\eta_{hu} - \eta_{hu}^{/}}{\eta_{hu}^{/}} \times 100\% = \frac{0.7560 - 0.5609}{0.5609} \times 100\%$$

= 34.78%

This signifies a significant improvement on the thermal efficiency of the rotary furnace system as a result of the recuperator incorporation.

B. Reduction in Fuel Consumption

From $\eta_{hu} = \dfrac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}}$

$$0.5609 = \frac{44 \times 10^6 B^{/} + 8.29 \times 10^6 B^{/} - 30.56 \times 10^3}{44 \times 10^6 B^{/} + 8.29 \times 10^6 B^{/}}$$

$$= \frac{52.29 \times 10^6 B^{/} - 30.56 \times 10^3}{52.29 \times 10^6 B^{/}}$$

From where $B^{/} = \dfrac{30.56 \times 10^3}{22.96 \times 10^6} = 1.331 \times 10^{-3}$ kg/h

$= 1.331 \times 10^{-3} \times 10.44 \times 10^3 = 13.90$ litres/h

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$$\%FuelConsumption\,Reduction = \frac{B - B'}{B} \times 100\%$$

$$\frac{25 - 13.9}{25} \times 100\% = 44.4\%$$

This also signifies a significant reduction on the fuel consumption rate of the rotary furnace system as a result of the recuperator incorporation, thereby reducing the operational cost.

The developed recuperators are shown in Figures 1, 2 and 3 while the Overall Assembly of the developed 300kg Recuperative Rotary Furnace is shown in Figure 4.

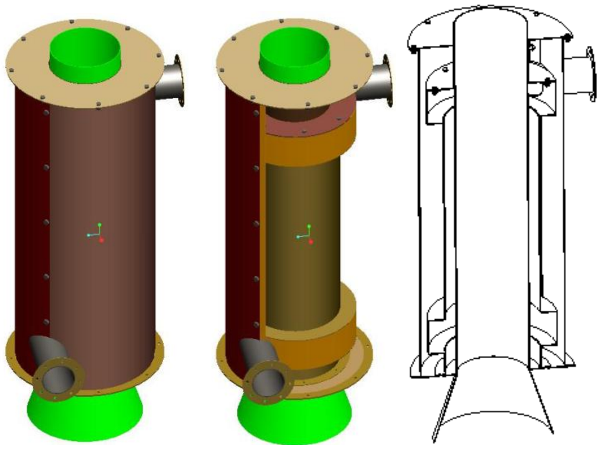

Fig 1: Radiation Recuperator

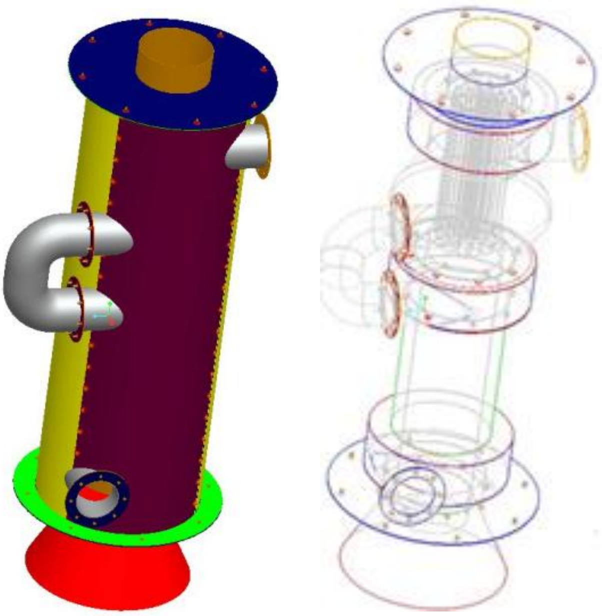

Fig 2: Unlabelled Hybrid Recuperator

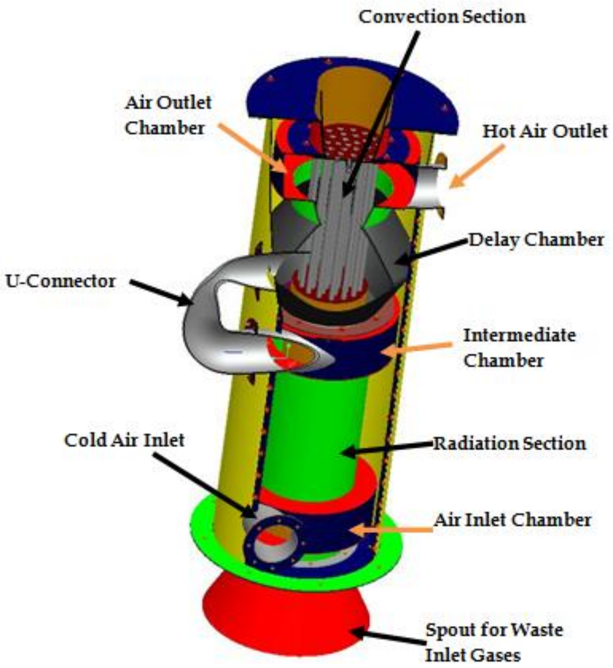

Fig 3: labelled Hybrid Recuperator

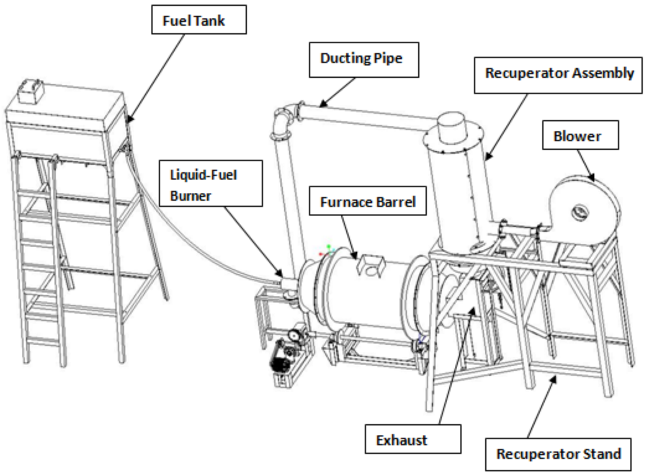

Fig 4: Labelled Overall Assembly of the developed Recuperative Rotary Furnace

IV. STATISTICAL ANALYSIS

Statistical analysis is done to establish a significant improvement in the temperature of combustion of the developed 300kg Capacity Rotary Furnace when it is incorporated with a heat exchanger and when it is without it. The data are as shown in table I.

The procedure for testing hypothesis/significance involves six steps. These are to

- Set up the null hypothesis (H_0)
- Set up the alternative hypothesis (H_1)
- Choose a level of significance (α)
- Carry out the test using appropriate test statistic
- Determine the decision rule
- Draw up the conclusion **(Grace, 2000)**

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Table I: Parameters Table for t-test between set of temperatures obtained from 300kg Rotary Furnace without a Recuperator (T1) and with a Recuperator (T2)

S/N	T_1	T_2	T_1^2	T_2^2
1	1093	1211.5	1194649	1467732.25
2	1154.5	1433	1332870.25	2053489
3	1183	1546	1399489	2390116
4	1286.5	1615	1655082.25	2608225
5	1366	1698.5	1865956	2884902.25
6	1392	1768.5	1937664	3127592.25
7	1422	1837.5	2022084	3376406.25
8	1447.5	1897.5	2095256.25	3600506.25
9	1465	1928.5	2146225	3719112.25
10	1474	1940	2172676	3763600
	$\sum T_1 =$ 13283.5	$\sum T_2 =$ 16876	$\sum T_1^2 =$. 17821952.75	$\sum T_2^2 =$ 28991682

Assuming the population variance $\sigma_1 = \sigma_2$;
$n_1 = n_2 = 10$
Null hypothesis $H_0: \mu_1 = \mu_2$
Alternative hypothesis $H_1: \mu_1 \neq \mu_2$

Test statistic is
$$t = \frac{\bar{T}_2 - \bar{T}_1}{S\sqrt{\frac{1}{n_1} + \frac{1}{n_2}}}$$

$$\bar{T}_1 = \frac{\sum T_1}{n} = \frac{13283.5}{10} = 1328.35$$
,
$$\bar{T}_2 = \frac{\sum T_2}{n} = \frac{16876}{10} = 1687.6$$

$$S_1^2 = \frac{\sum_{i=1}^{n}(T_{1_i} - \bar{T}_1)^2}{n-1} = [\frac{\sum_{i=1}^{n} T_{1_i}^2 - (\sum_{i=1}^{n} T_1)^2 / n}{n-1}]$$
,
$$S_1^2 = \frac{17821951.75 - 13283.5^2/10}{10-1} = 19646.06$$

$$S_2^2 = \frac{\sum_{i=1}^{n}(t_{2_i} - \bar{t}_2)^2}{n-1} = [\frac{\sum_{i=1}^{n} t_{2_i}^2 - (\sum_{i=1}^{n} t_2)^2 / n}{n-1}],$$

$$S_2^2 = \frac{28991681.5 - 16876^2/10}{10-1} = 56860.43$$

$$\text{S(pooled variance)} = \sqrt{\frac{(n_1-1)S_1^2 + (n_2-1)S_2^2}{n_1+n_2-2}}$$

$$\text{S(pooled variance)} = \sqrt{\frac{(10-1)*19646.06 + (10-1)*56860.43}{10+10-2}}$$

= 195.58
$$t = \frac{1687.6 - 1328.35}{195.58\sqrt{\frac{1}{10} + \frac{1}{10}}} = 4.107$$

At 1% level of significance, tabulated $t_{18,\ 0.01}$ is 2.878. Since the calculated t is greater than the tabulated t, it shows that there is a significant improvement in the temperature of combustion for a recuperator-incorporated rotary furnace over an ordinary furnace without a recuperator. Graphs of the temperatures of combustion (of the two scenarios) when plotted against time give the pattern shown in Figure 5

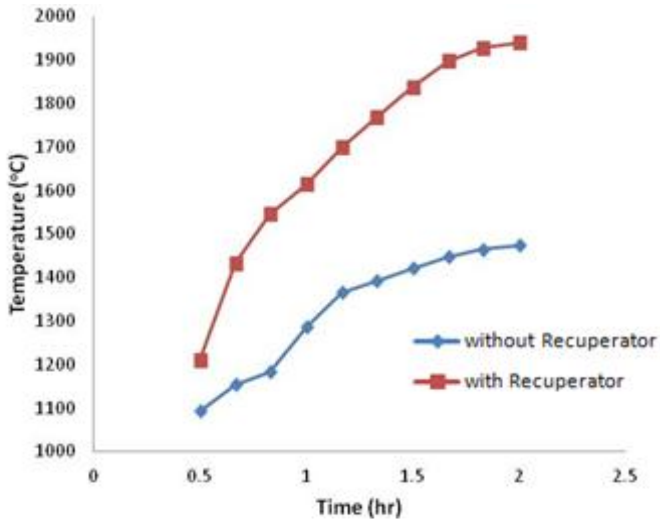

Fig 5: Comparative Graphs of Combustion Temperatures (Rotary Furnace with and without a Recuperator) vs Time)

V. FINITE ELEMENTS ANALYSIS

Figure 6 shows the discretized recuperator used for structural and thermal analyses through Finite Elements Analysis (FEA). The analyses were done through Pro/E Mechanica (Figure 6 and 7) and COMSOL Multiphysics (Figures 8 and 9). Pro-E Mechanica is a CAE (Computer Aided Engineering) product that allows simulation of the physical behaviour of a part or assembly, to understand and improve the mechanical performance of a design (Roger 2006). COMSOL Multiphysics on the other hand is a powerful interactive environment for modeling and solving all kinds of scientific and engineering problems based on partial differential equations (PDEs). With this software, conventional models for one type of physics can easily be extended into multiphysics models that solve coupled physics phenomena – and do so simultaneously.

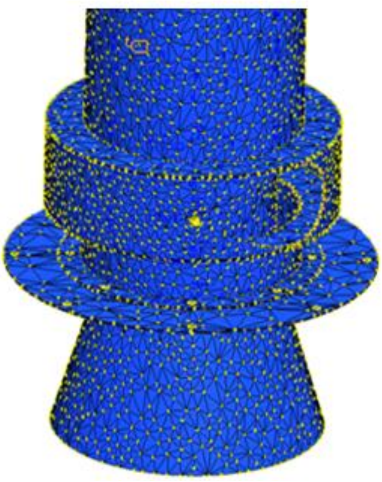

Fig 6: Meshing/Discretization of the developed Recuperative Heat Exchanger

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A. Thermal Analysis

The thermal analysis was done by modelling a conjugate heat transfer phenomenon using the Turbulent Fluid-Thermal Interaction predefined multiphysics coupling from the Heat Transfer.

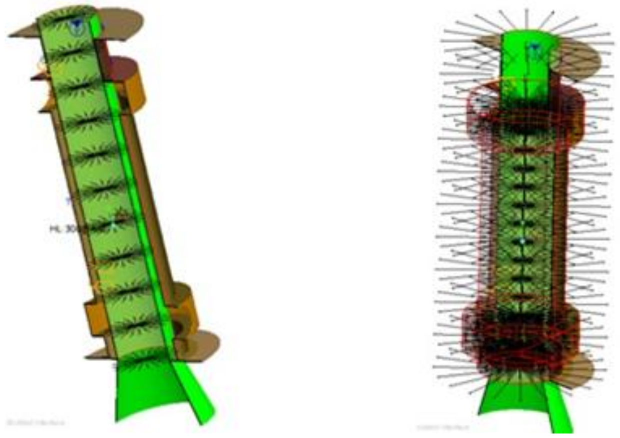

Figure 7: Developed Recuperator's Thermal Simulation

B. Model Definition – Solid and Fluid Heat Transfer

The Turbulent Fluid–Thermal Interaction predefined multiphysics coupling sets up the appropriate application modes together with applicable couplings, making it easy to model the fluid-thermal interaction. It is necessary to correct the fluid's thermal conductivity to take into account the effect of mixing due to eddies. The turbulence results in an effective thermal conductivity, k_{eff}, according to the equation

$$k_{eff} = k + k_T$$

where k is the fluid's physical thermal conductivity and $k_T = C_p \eta_T$ is the turbulent conductivity. η_T denotes the turbulent dynamic viscosity, and C_p is the heat capacity. It is easy to obtain the effective conductivity in COMSOL Multiphysics by using the predefined Fluid group in the fluid domain. In this group, the variable for turbulent conductivity is already given in the heat transfer application mode for the fluid.

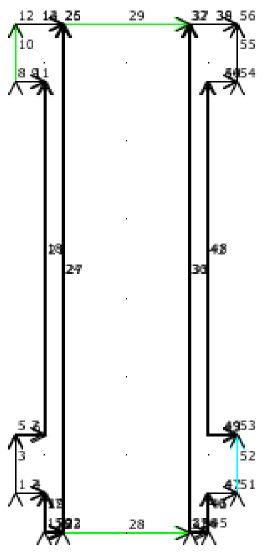

Fig 8: Model Representation with Boundary Conditions

C. Boundary Conditions and Thermal Simulation Results

The model representing the boundary conditions is shown in Figure 8. The boundary conditions describing the problem are:

i. *k*-ω equations in the fluid domain
ii. Specified mass flow through the domain
iii. Pressure difference between inlet and outlet given by the mass flow
iv. Normal flow at the inlet and outlet
v. Stream-wise periodic conditions for *u*, *v*, *k*, and ε.
vi. Logarithmic wall function at the pipes' surface boundaries
vii. Heat transport equations
viii. 1050 °C temperature at the hotter waste gases inlet
ix. Convection-dominated transport at the outlet
x. Thermal wall function at the pipe/water interfaces
xi. Fixed temperature at the inside of the heat pipes

The periodicity of the flow is important because a heat exchanger where the flow is fully developed is being modelled. It is hard to make a periodic configuration converge from a homogeneous initial guess, however, and therefore, an initial calculation with constant inlet velocity and fixed outlet pressure is first performed. The logarithmic wall function boundary condition for turbulent flow is used to model the solid-fluid interfaces. An algebraic relationship—the logarithmic wall function—describes the momentum transfer at the solid-fluid interface. This means that the solid-fluid boundaries in the model actually represent lines within the logarithmic regions of the boundary layers.

Figure 9 shows the final result of the thermal modelling and simulation of the incorporated radiation recuperator into the operation of a 300kg capacity rotary furnace while the mesh statistics is shown in Table II.

Table II: Recuperator Mesh Statistics

Number of degrees of freedom	183200
Number of mesh points	11332
Number of elements	21892
Triangular	21892
Quadrilateral	0
Number of boundary elements	2668
Number of vertex elements	46
Minimum element quality	0.626
Element area ratio	0.002

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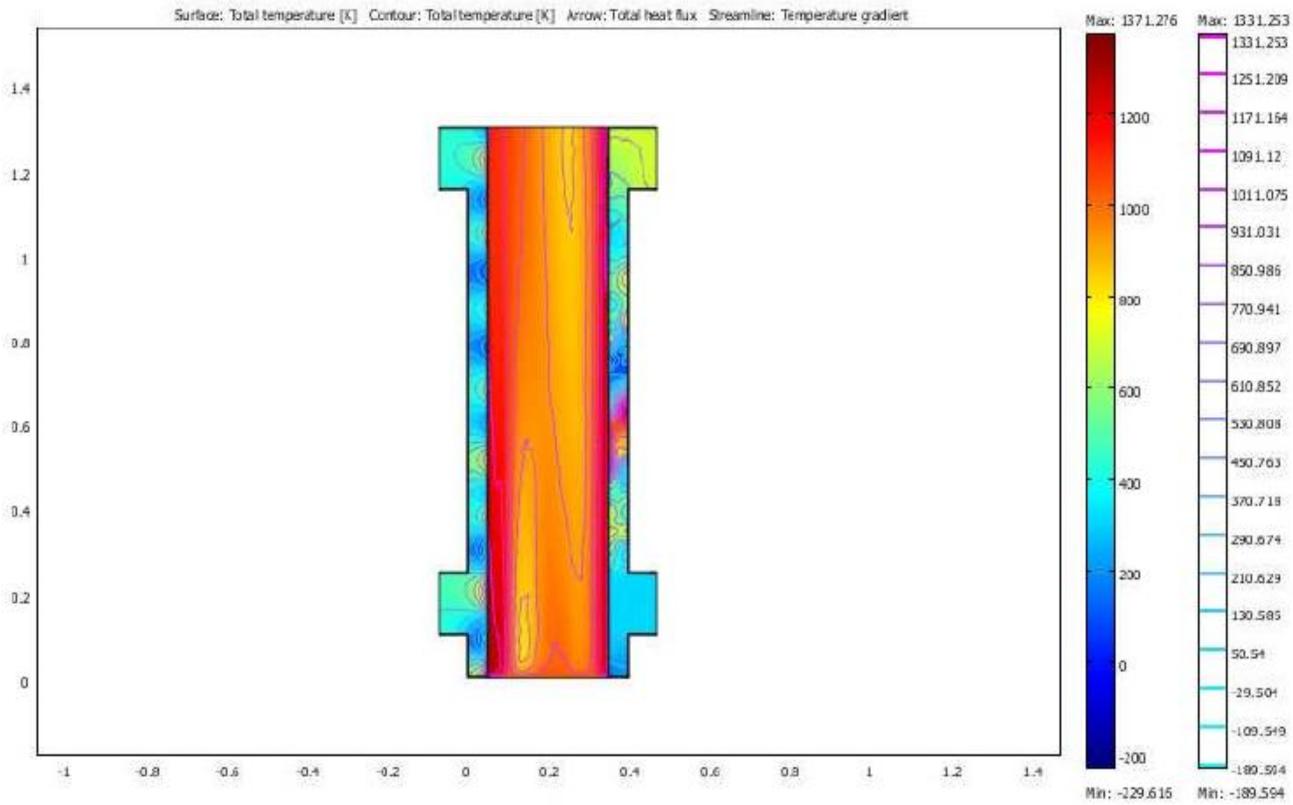

Fig 9: Temperature distribution in Recuperative Heat Exchanger used in Rotary Furnace Operation

VI. CONCLUSION

This work has been able to establish a significant quantitative and qualitative improvement in the performance of a liquid fuel fired melting furnace through the incorporation of a recuperative heat exchanger. Finite Elements Analyses were done for a better insight into the thermal and structural conditions of the system. Incorporation of recuperative heat exchanger into the design of a rotary furnace is very essential. A major reason being that a unit of physical heat recovered from waste gases and returned back into the furnace with air is much more useful than a corresponding unit of chemical heat produced in the furnace through combustion of fuel, since the heat of preheated air or gas involves no heat losses with waste gases. The value of a unit of physical heat is greater at a lower coefficient of fuel utilization and higher temperature of waste gases.

This paper has been able to establish that the recuperative heat exchanger designed, developed and incorporated into the rotary furnace system results in

i. Appreciable increase (34.78%) in the furnace thermal efficiency
ii. Considerable increase in the temperature of combustion (33.33%). This makes it possible to adapt the furnace for steel casting.
iii. Significant reduction (44.4%) in the fuel combustion thereby reducing operating cost and
iv. Aiding of complete combustion.

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