

Paper on Design and Analysis of Bending Furnace using Alternative Fuels

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Abstract

The aspect of heat transfer which in itself requires an elaborate model is considered here. Taken Into account are the convection and radiation between gas and brick wall, and the conduction from bricks to light diesel oil as a fuel used in this paper. by taking minimum heat loss rate (HLR) as optimization objective. Two boundary conditions of the insulation layers, convective heat transfer and combined convective and radioactive heat transfer, are taken into account. The optimal constructs of the insulation layers with the two boundary conditions are obtained. Improvement of energy efficiency of the most consuming facilities was achieved by improving the use of alternative energy minimization such as reducing the heat loss of hot gases, minimizing the heat radiated through refractory linings of metallurgical furnaces.

Keywords- Furnace Wall, Transient Heat Transfer, LDO, Coal, LPG, Insulation

I. INTRODUCTION

As energy crisis is becoming prominent, an effective way to make use of energy is what people chase for. Thermal insulation is an effective way to reduce the heat loss of a thermal system. Therefore, many scholars have shown great interests in the investigations of the thermal insulation problems, such as building enclosure structures, pipeline systems, etc. Constructed theory is a powerful theory for the optimal designs of various engineering applications, and many insulation problems had been solved by using this theory. For the constructal designs of insulation layers of the reheating furnaces, built the plane and cylindrical insulation layer models of a reheating furnace wall, and optimized the distributions of the layers with minimum heat loss rate (HLR).The results showed that the HLR of the insulation layer with optimal thickness was reduced by 12.5% as compared with that of the uniform one, and the heat loss reduction of the plane insulation layer with optimal thickness was larger than that of the cylindrical one.. Built a multilayer insulation model of the furnace wall with constant temperature boundary condition, and carried out constructal optimizations of the insulation layers by taking minimum HLR as optimization objective. The results showed that the heat loss of the insulation layer with optimal distributed thicknesses was greatly reduced compared with that of the uniform ones, and the decrement of heat loss would be more obvious when the temperature distribution of the furnace wall was convex. Moreover, they also investigated the optimal distributions of the heaters in the reheating furnace with convective and radiative heat transfers by taking minimal fuel consumption as optimization objective Based on the models in and applied constructal theory and entransy theory. Optimized the distributions of the single-layer insulation with constant temperature boundary condition and convective and radiative boundary condition, respectively[1].

They concluded that the optimal distributions of the insulation layers were different from those obtained by HLR minimizations. For the constructal designs of insulation layers of the hot fluid pipes, Bejan built a distributed insulation layer model for a convective heat transfer hot fluid pipe, and optimized the thickness of the cylindrical insulation layer with minimum HLR. The result showed that optimal thickness of the insulation layer was uniform when the amount of the insulation material was fixed. by using optimal control theory, and the results obtained were coincided with those obtained in Moreover, they further optimized the distributions of the insulation layers of the hot fluid pipe with radiative as well as combined heat transfer boundary conditions, and obtained some different optimal distributions of the insulation layers. For the constructal designs of vertical insulating walls, optimized the internal structures of the vertical insulation wall subjected to a fixed mechanical stiffness and obtained the maximum thermal resistance of the wall. Furthermore, reconsidered the insulation wall model, and optimized the number of air cavities by considering heat flow, strength and wall weight simultaneously[2].

The constructal optimization of multilayer insulation structures with constant temperature boundary condition was carried out in. Based on the multilayer insulation model in a more actual model of multilayer insulation structures with radiative as well as combined convective and radiative heat transfer boundary conditions will be considered in this paper. Constructal optimizations of the insulation layers of a reheating furnace wall with the two boundary conditions will be carried out by taking minimum HLR as optimization objective. The optimal insulation performance with minimum HLR will be compared to that with average thicknesses of the insulation layers as well as that with minimum maximum temperature gradient. The model will be more generalized, and several heat transfer boundary conditions of the insulation layers will become special cases of this paper.

A blast furnace is a key installation of a blast furnace system as well as an important device of the metallurgical production process. As the furnace wall is an important part of a blast furnace, the furnace wall plays an important role in the blast furnace system. Hence, it is significant to perform research on heat transfer performance (HTP) of blast furnace wall and optimizations for the structure of blast furnace wall. Many scholars have performed research on heat transfer problems of blast furnace wall. Performed optimization for the cross section shape of cooling channel of a cooling stove based on a convective heat transfer empirical formula, and the results showed that adopting oblate cooling channel could decrease the heat transfer quantity and cooling water usage. Performed analyses for the temperature field (TF) of a cast cooling stove, and analyzed the effects of different structure parameters on the TF. Established a simplified one-dimensional model of a blast furnace wall, and proposed a method to calculate the furnace wall erosion line and TF. Analyzed the effects of cross section shape of cooling channel on the thermal stress of cooling stove. Established a 3D model of a blast furnace wall, and analyzed the effects of slag skull thickness on the TF of blast furnace wall [3].

II. CONSTRUCTAL OPTIMIZATION OF MULTILAYER INSULATION STRUCTURES

Consider a simple model of steel rolling reheating furnace wall with multilayer insulation structures, as shown in. The billet steel is heated by the high temperature gas in the inner of the hearth, and part of heat is dissipated to the ambient (ambient temperature T_0) through the furnace wall. The temperature $T(x)$ ($0 \leq x \leq L$) of the internal furnace wall is specified. A number (N) of insulation layers (thermal conductivity k_i , thickness t_i , $i = 1, 2, 3, \dots, N$) are laid outside of the furnace wall to reduce heat loss from the furnace. The length and width of the insulation layers are L and W , respectively. For the simplification of the calculation, the parameters along the third dimension (width direction) are assumed to be not varied. In this case, the heat conduction model in the paper becomes two dimensional, and the width of the insulation layer is fixed at unit width, i.e. $W=1$. When the thicknesses t_i ($i = 1, 2, 3, \dots, N$) are much smaller than the length of the insulation layers, the heat transfer rate along the thickness direction is much larger than that along the length direction. In this case, the heat transfer along the length direction can be approximately ignored, and the heat conductions in the insulation layers can be simplified as one dimensional the models of the multilayer insulation structures with uniform and distributed thicknesses are shown in (a) and (b), respectively [4].

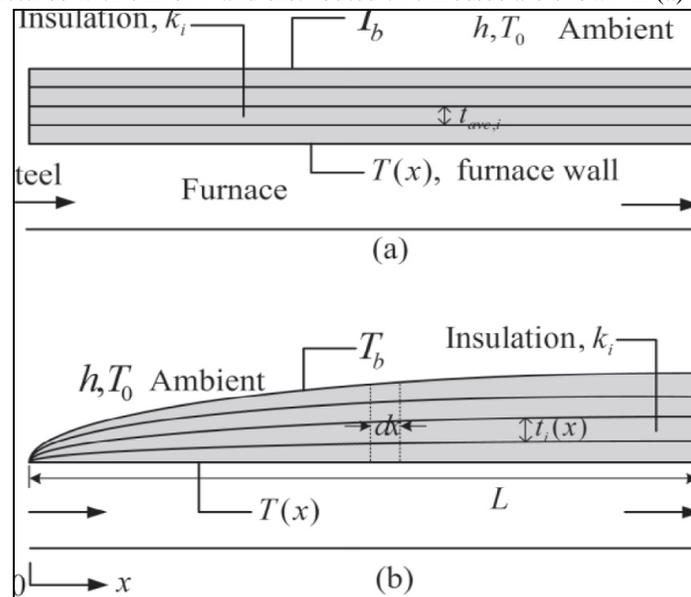


Fig. 1: Model of a reheating furnace wall with multilayer insulation structures:
(a) Insulation layers with uniform thicknesses (b) insulation layers with distributed thicknesses

III. FURNACE CONSTRUCTION

The modern industrial furnace design has evolved from a rectangular or cylindrical enclosure, built up of refractory shapes and held together by a structural steel binding. Combustion air was drawn in through wall openings by furnace draft, and fuel was introduced through the same openings without control of fuel / air ratios except by the judgment of the furnace operator. Flue gases were exhausted through an adjacent stack to provide the required furnace draft. To reduce air infiltration or outward leakage of combustion gases, steel plate casings have been added. Fuel economy has been improved by burner designs providing some control of fuel / air ratios, and automatic controls have been added for furnace temperature and furnace pressure. Completely sealed furnace enclosures may be required for controlled atmosphere operation, or where outward leakage of carbon monoxide could be an operating hazard. With the steadily increasing costs of heat energy, wall structures are being improved to reduce heat losses or heat demands for cyclic heating. The selection of furnace designs and materials should be aimed at a

minimum overall cost of construction, maintenance, and fuel or power over a projected service life. Heat losses in existing furnaces can be reduced by adding external insulation or rebuilding walls with materials of lower thermal conductivity.

To reduce losses from intermittent operation, the existing wall structure can be lined with a material of low heat storage and low conductivity, to substantially reduce mean wall temperatures for steady operation and cooling rates after interrupted firing. Thermal expansion of furnace structures must be considered in design. Furnace walls have been traditionally built up of preferred refractory shapes with bonded mortar joints. Except for small furnaces, expansion joints will be required to accommodate thermal expansion. In sprung arches, lateral expansion can be accommodated by vertical displacement, with longitudinal expansion taken care of by lateral slots at intervals in the length of the furnace. Where expansion slots in furnace floors could be filled by scale, slag, or other debris, they can be packed with a ceramic fiber that will remain resilient after repeated heating. Differential expansion of hotter and colder wall surfaces can cause an inward-bulging effect. For stability in self-supporting walls, thickness must not be less than a critical fraction of height[5].

IV. TYPES AND CLASSIFICATION OF DIFFERENT FURNACES

Based on the method of generating heat, furnaces are broadly classified into two types namely combustion type (using fuels) and electric type. In case of combustion type furnace, depending upon the kind of combustion, it can be broadly classified as oil fired, coal fired or gas fired.

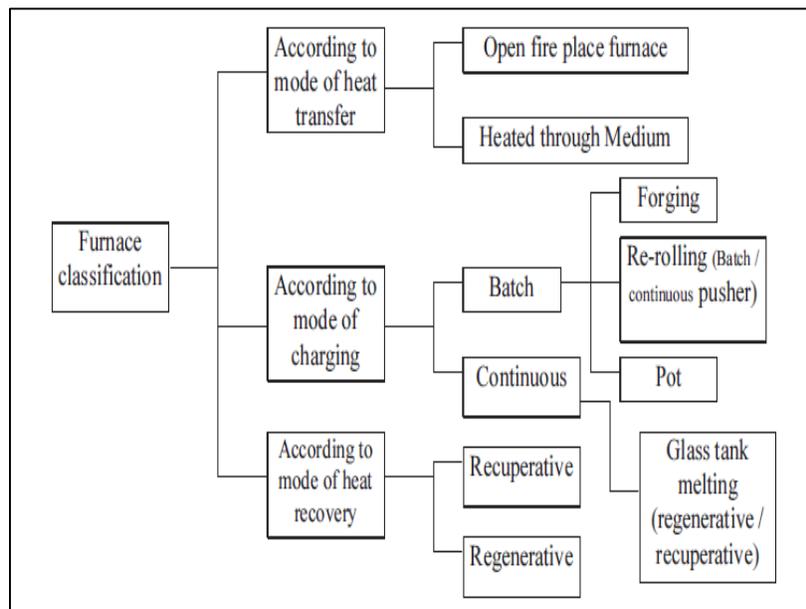


Fig. 2: Furnace Classification

Based on the mode of charging of material furnaces can be classified as (i) Intermittent or Batch type furnace or Periodical furnace and (ii) Continuous furnace.

- Based on mode of waste heat recovery as recuperative and regenerative furnaces.
- Another type of furnace classification is made based on mode of heat transfer, mode of charging and mode of heat recovery[6].

V. HEAT TRANSFER IN FURNACES

The main ways in which heat is transferred to the steel in a reheating furnace are shown in Figure 3. In simple terms, heat is transferred to the stock by:

- Radiation from the flame, hot combustion products and the furnace walls and roof.
- Convection due to the movement of hot gases over the stock surface. At the high temperatures employed in reheating furnaces, the dominant mode of heat transfer is wall radiation. Heat transfer by gas radiation is dependent on the gas composition (mainly the carbon dioxide and water vapors concentrations), the temperature and the geometry of the furnace[7].

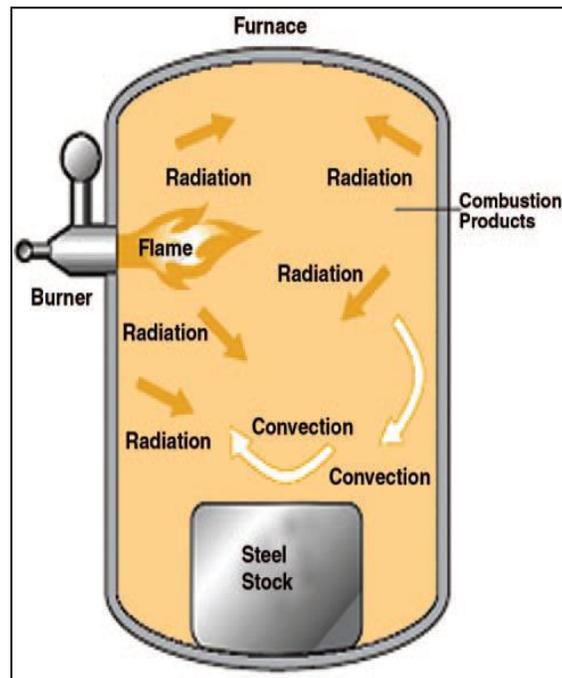


Fig. 3: Heat Transfer in Furnace

VI. FURNACE TEMPERATURE PROFILES

To predict heating rates and final load temperatures in either batch or continuous furnaces, it is convenient to assume that source temperatures, gas (T_g) or furnace wall (T_w), will be constant in time. Neither condition is achieved with contemporary furnace and control system designs. With constant gas temperature, effective heating rates are unnecessarily limited, and the furnace temperature control system is dependent on measurement and control of gas temperatures, a difficult requirement. With uniform wall temperatures, the discharge temperature of flue gases at the beginning of the heating cycle will be higher than desirable.

Three types of furnace temperature profiles, constant T_g , constant T_w , and an arbitrary pattern with both variables, are shown in Fig. 27. Contemporary designs of continuous furnaces provide for furnace temperature profiles of the third type illustrated, to secure improved capacity without sacrificing fuel efficiency. The firing system comprises three zones of length: a preheat zone that can be operated to maintain minimum flue gas temperatures in a counter flow firing arrangement, a firing zone with a maximum temperature and firing rate consistent with furnace maintenance requirements and limits imposed by the need to avoid overheating of the load during operating delays, and a final or soak zone to balance furnace temperature with maximum and minimum load temperature specifications. In some designs, the preheat zone is unheated except by flue gases from the firing zone, with the resulting loss of furnace capacity offset by operating the firing zone at the maximum practical limit[8].

VII. PURPOSE OF INSULATION

A thermal insulator is a poor conductor of heat and has a low thermal conductivity. Insulation is used in buildings and in manufacturing processes to prevent heat loss or heat gain. Although its primary purpose is an economic one, it also provides more accurate control of process temperatures and protection of personnel. It prevents condensation on cold surfaces and the resulting corrosion. Such materials are porous, containing large number of dormant air cells. Thermal insulation delivers the following benefits:

- Reduces over-all energy consumption
- Offers better process control by maintaining process temperature.
- Prevents corrosion by keeping the exposed surface of a refrigerated system above dew point
- Provides fire protection to equipment
- Absorbs vibration

A. Types and Application

The Insulation can be classified into three groups according to the temperature ranges for which they are used.

1) Low Temperature Insulations (up to 90 °C):

This range covers insulating materials for refrigerators, cold and hot water systems, storage tanks, etc. The commonly used materials are Cork, Wood, 85% magnesia, Mineral Fibers, Polyurethane and expanded polystyrene, etc.

2) *Medium Temperature Insulations (90 – 325 °C):*

Insulators in this range are used in low temperature, heating and steam raising equipment, steam lines, flue ducts etc. The types of materials used in this temperatures range include 85% Magnesia, Asbestos, Calcium Silicate and Mineral Fibers etc.

3) *High Temperature Insulations (325 °C – above):*

Typical uses of such materials are super-heated steam system, oven dryer and furnaces etc. The most extensively used materials in this range are Asbestos, Calcium Silicate, Mineral Fiber, Mica and Vermiculite based insulation, Fireclay or Silica based insulation and Ceramic Fiber.

4) *Heat Losses from Furnace Walls:*

In furnaces and kilns, heat losses from furnace walls, affect the fuel economy substantially. The extent of wall losses depends on:

- 1) Emissivity of walls.
- 2) Conductivity of refractories.
- 3) Wall thickness.
- 4) Whether furnace or kiln is operated continuously or intermittently.

Different materials have different radiation power (emissivity). The emissivity of walls coated with aluminum paint is lower than that of bricks. Shows the coefficient of heat dissipation for the following conditions:

- a) Rough vertical plane surface.
- b) Vertical aluminum painted walls

VIII. MANUAL CALCULATION

A. Calculation for LDO Fuel:

Oil: - LDO

Plate Material: - Mild Steel

1) Heat Supplied by fuel (Q_s):-

$$Q_s = m_f \times CV$$

Where:-

m_f = mass of Fuel consumption/hr

$$= 46710 \text{ Kg/month (30 days)}$$

$$= 46710 / (1 \times 26 \times 960 \times 60)$$

$$m_f = 0.0311 \text{ kg/sec}$$

CV= Calorific Value = 10700 K-Cal/kg

$$= 10700 \times 4.1868 \text{ (1KCal/kg=4.1868KJ/kg)}$$

$$= 44798.44 \text{ KJ/kg}$$

$$Q_s = 0.0311 \times 44798.44$$

$$Q_s = 1393.24 \text{ K-W}$$

1) *Step 1) To Calculate Sensible Heat Loss:*

$$\text{Sensible heat loss} = M_g \times C_{pg} \times (T_f - T_g)$$

$$= M_f \times (A/F + 1) \times C_{pg} \times (T_f - T_g)$$

$$= 0.0311 \times 15.2 \times 1.005 \times (410 - 52.5)$$

$$Q_{\text{sensible}} = 169.84 \text{ KW}$$

% of heat loss in flue gas = Q sensible/ heat supply by fuel

$$= 169.84 / 1393.24$$

$$= 12.19 \%$$

2) *Step 2) To Calculate The Heat Loss Due to Heat Carried Away in Stock:*

$$\text{Heat absorbed by plate} = M_{\text{plate}} \times C_{p \text{ plate}} \times (T_{p2} - T_a)$$

$$= 160 \times 0.416 \times (700 - 52.5)$$

$$= 43097.6$$

$$= 43097.6 / 20 \text{ min}$$

$$= 2154.88$$

$$Q_{\text{absorbed}} = 35.16 \text{ KW}$$

% of heat utilizing by plate = Q_{absorbed}/ heat supply by fuel

$$= 35.16 / 1393.24$$

$$= 2.57 \%$$

3) *Step 3) To Calculate the Heat Loss Due to Moisture:*

$$Q_{\text{moisture}} = M_m \times C_{pm} \times (T_f - T_a)$$

Where M_m = Mass of moisture

$$= 9 \times \% \text{ of hydrogen}$$

$$= 9 \times 15/100 \times 0.0311$$

$$= 0.041$$

$$= 0.041 \times 2.1 \times (410 - 52.5)$$

$$Q_{\text{moisture}} = 30.78 \text{ KW}$$

$$\% \text{ of heat loss in flue gas} = Q_{\text{Moisture}} / \text{heat supply by fuel}$$

$$= 30.78 / 1393.24$$

$$= 2.20 \%$$

4) Step 4) Calculate Heat Loss Due To Radiation And Convection Heat Loss From The Furnace Outer Surface :

$$Q_{\text{surface}} = [9.7 \times (T_{\text{wall}})^2 / 1000 - 1.42 T_{\text{wall}} + 164] - [(T_{\text{amb}} - 75) / 50 \times (0.085 \times (T_{\text{wall}} - 100) + 90)]$$

Reference W Trinks industrial furnace John Wiley & sons

where, A = surface area of furnace in m²

T_{wall} = wall surface temp. In F.

$$= [9.7 \times (1268.6)^2 / 1000 - 1.42 \times 1268.6 + 164] - [(52.5 - 75) / 50 \times (0.085 \times (1268.6 - 100) + 90)]$$

$$= 14171.019 - (-85.13)$$

$$= 14171.019 \text{ Kj /kg}$$

$$\% \text{ of heat loss in flue gas} = Q_{\text{surface}} / \text{CV of fuel}$$

$$= 14171.019 / 44798.76$$

$$= 31.63 \%$$

5) Step 5) Heat Loss Due to Opening :

The shape of the opening is Rectangular and ratio = D/X

$$= 1.5 / 0.6$$

$$= 2.5$$

With respect to value 2.5 from chart

Factor of radiation = 0.78

Black body radiation corresponding to 7000C = 8 Kcal/cm²/hr (from chart)

Area of opening = 100cm × 100cm

$$= 10000 \text{ cm}^2$$

Emissivity = 0.8

Total heat loss = Black body radiation × Area of opening × Factor of radiation × Emissivity

$$= 8 \times 10000 \times 0.78 \times 0.8$$

$$= 49920 \text{ Kcal/hr}$$

Equivalent heat loss = 49920/10700

$$= 4.66 \text{ Kg/hr}$$

$$\% \text{ of heat loss} = 4.66 / 64.87 \times 100$$

$$= 7.19 \%$$

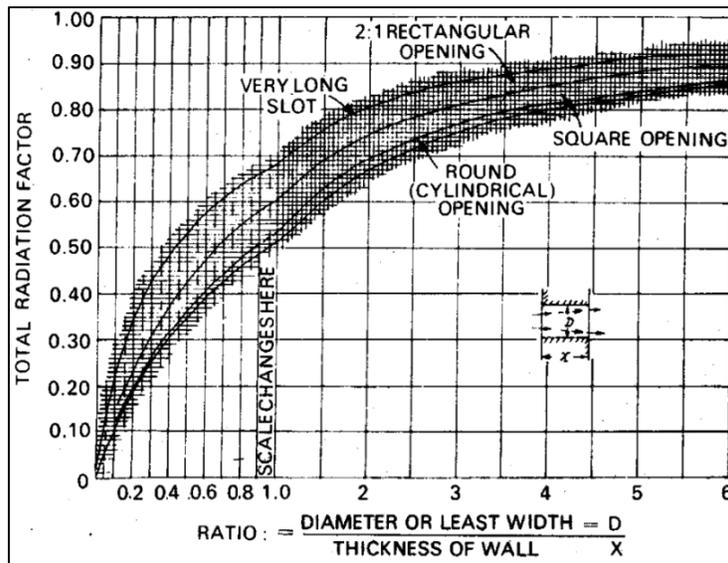


Fig. 2.4: Factor for Determining the Equivalent of Heat Release from Openings to the Quality of Heat Release from Perfect Black Body.

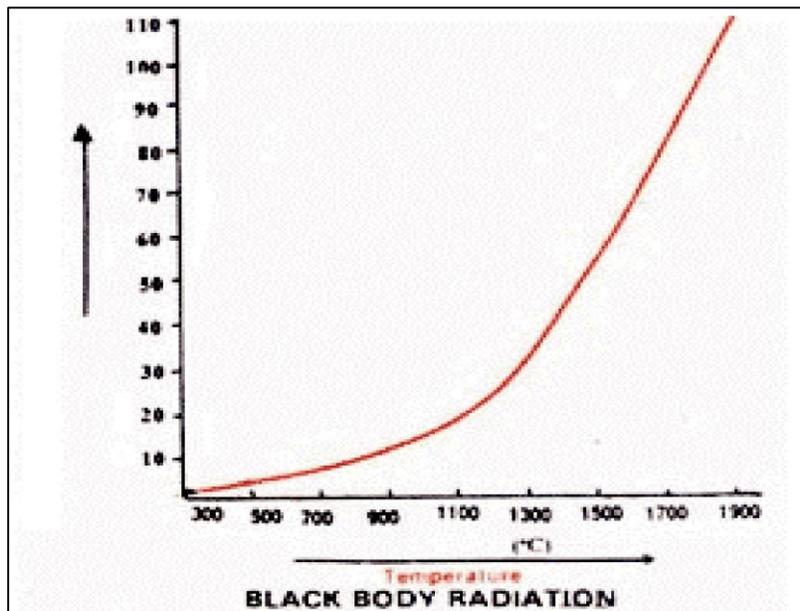


Fig. 2.5: Graph for Determining Black Body Radiation at a Particular Temperature

6) *Step 6) Unaccounted Heat :*

$$\begin{aligned} \text{Furnace efficiency} &= 100 - \text{Total losses} \\ &= 100 - 55.78 \\ &= 44.22 \% \end{aligned}$$

B. To Calculate The Length of Insulation (With or Without Insulation):

1) *Without Insulation*

$$\begin{aligned} R_{\text{total}} &= R_1 + R_2 + R_3 \\ &= 1/H_i A + L/KA + 1/H_o A \\ &= 1/8 \times 2.5 [(1/125) + (0.6/1.45) + (1/70)] \\ &= 4 \times 10^{-4} + 0.020 + 7.1428 \times 10^{-4} \\ &= 0.0211 \text{ oK/W} \\ Q \text{ without insulation} &= T_i - T_o / R_{\text{total}} \\ &= 700 - 52.5 / 0.0211 \\ &= 30.68 \text{ KW} \end{aligned}$$

2) *To Calculate Inside Wall Temp*

$$\begin{aligned} Q &= T_i - T_1 / R_1 \\ 30680 &= 700 - T_1 / 4 \times 10^{-4} \\ T_1 &= 687^\circ\text{C} \end{aligned}$$

To calculate the outside wall temp:

$$\begin{aligned} Q &= T_1 - T_2 / R_2 \\ 30680 &= 687 - T_2 / 0.020 \\ T_2 &= 73.8^\circ\text{C} \end{aligned}$$

3) *Heat Transfer through Furnace with Insulation*

1) Insulation : Gypsum powder (K= 0.4 W/M°C)

We have to reduced 50% of total heat

$$\begin{aligned} \text{i.e } Q_1 &= 0.5 \times Q \\ &= 0.5 \times 30680 \\ &= 15.34 \text{ KW} \\ \text{i.e } Q_1 &= T_2 - T_3 / R_{\text{insulation}} \\ 15340 &= 73.8 - 30 / R_{\text{insulation}} \\ R_{\text{insulation}} &= 2.85 \times 10^{-3} \\ L/KA &= 2.85 \times 10^{-3} \\ L &= 2.85 \times 10^{-3} \times 0.4 \times 8 \times 2.5 \\ L &= 22.84 \text{ mm} \end{aligned}$$

2) Insulation: Asbestos fiber ($K = 0.115 \text{ W/M}^0\text{c}$)

$$\text{i.e } Q_1 = T_2 - T_3 / R_{\text{insulation}}$$

$$15340 = 73.8 - 30 / R_{\text{insulation}}$$

$$R_{\text{insulation}} = 2.85 \times 10^{-3}$$

$$L/KA = 2.85 \times 10^{-3}$$

$$L = 2.85 \times 10^{-3} \times 0.115 \times 8 \times 2.5$$

$$L = 6.29 \text{ mm}$$

3) Insulation: Ceramic fiber ($K = 0.12 \text{ W/M}^0\text{c}$)

$$\text{i.e } Q_1 = T_2 - T_3 / R_{\text{insulation}}$$

$$15340 = 73.8 - 30 / R_{\text{insulation}}$$

$$R_{\text{insulation}} = 2.85 \times 10^{-3}$$

$$L/KA = 2.85 \times 10^{-3}$$

$$L = 2.85 \times 10^{-3} \times 0.12 \times 8 \times 2.5$$

$$L = 6.84 \text{ mm}$$

C. Calculation for Bituminous Coal Fuel

Fuel: - bituminous Coal

Plate Material: - Mild Steel

1) Heat Supplied by fuel (Q_s):-

$$Q_s = m_f * C.V.$$

Where:-

m_f = mass of Fuel consumption/hr

$$= 1200 \text{ kg/day}$$

$$= 36000 \text{ Kg/month (30 days)}$$

$$= 36000 / (1 \times 26 \times 960 \times 60)$$

$$m_f = 0.0240 \text{ kg/sec}$$

C.V = Calorific Value = 8001.33 K-Cal/kg

$$= 8001.33 \times 4.1868 \text{ (1K-Cal/kg = 4.1868KJ/kg)}$$

$$= 33,500 \text{ KJ/kg}$$

$$Q_s = 0.0240 \times 33500$$

$$Q_s = 804 \text{ K-W}$$

1) Step 1) To calculate sensible heat loss:

$$\text{Sensible heat loss} = M_g \times C_{pg} \times (T_f - T_g)$$

$$= M_f \times (A/F + 1) \times C_{pg} \times (T_f - T_g)$$

$$= 0.0240 \times 14.5 \times 1.005 \times (400 - 45)$$

$$Q_{\text{sensible}} = 124.15 \text{ KW}$$

% of heat loss in flue gas = $Q_{\text{sensible}} / \text{heat supply by fuel}$

$$= 124.15 / 804$$

$$= 15.44 \%$$

2) Step 2) To Calculate The Heat Loss Due To Heat Carried Away In Stock:

$$\text{Heat absorbed by plate} = M_{\text{plate}} \times C_{p \text{ plate}} \times (T_{p2} - T_a)$$

$$= 160 \times 0.416 \times (750 - 45)$$

$$= 46924.8$$

$$= 46924.8 / 20 \text{ min}$$

$$= 2346.24$$

$$= 39.10 \text{ KW}$$

% of heat utilising by plate = $Q_{\text{absorbed}} / \text{heat supply by fuel}$

$$= 39.10 / 804$$

$$= 4.86 \%$$

3) Step 3) heat loss due to moisture :

$$Q_{\text{moisture}} = M_m \times C_{pm} \times (T_f - T_a)$$

Where M_m = Mass of moisture

$$= 9 \times \% \text{ of hydrogen}$$

$$= 9 \times 3/100 \times 0.0240$$

$$= 6.48 \times 10^{-3}$$

$$= 6.48 \times 10^{-3} \times 2.1 \times (400 - 45)$$

$$= 4.83 \text{ KW}$$

% of heat loss in flue gas = $Q_{\text{Moisture}} / \text{heat supply by fuel}$

$$= 4.83 / 804$$

$$= 0.60 \%$$

4) Step 4) Calculate Heat Loss Due to Radiation and Convection Heat Loss from The Furnace Outer Surface :

$$Q_{\text{surface}} = [9.7 T_{\text{wall}}^2/1000 - 1.42 T_{\text{wall}} + 164] - [(T_{\text{amb}}-75)/50 \times (0.085 \times (T_{\text{wall}} - 100) + 90)]$$

Reference W Trinks industrial furnace John wiley & sons

Where, A = surface area of furnace in m²

T_{wall} = wall surface temp. In F.

$$= [9.7 \times (1358.6)^2/1000 - 75/50 \times 1.42 \times 1358.6 + 164] - [(45 - \times (0.085 \times (1358.6 - 100) + 90))]$$

$$= 16138.98 - (-118.15)$$

$$= 16257.16 \text{ Kj/kg}$$

$$\% \text{ of heat loss in flue gas} = Q_{\text{surface}} / \text{heat supply by fuel}$$

$$= 16257.16 / 33500$$

$$= 46.44 \%$$

5) Step 5) Heat Loss Due To Opening:

The shape of is opening is Rectangular and ratio = D/X

$$= 1.5/0.6$$

$$= 2.5$$

With respect to value 2.5 from chart

Factor of radiation = 0.78

Black body radiation corresponding to 750 Oc = 12Kcal/cm² /hr (from chart)

Area of opening = 100cm × 100cm

$$= 10000 \text{ cm}^2$$

Emissivity = 0.8

Total heat loss = Black body radiation × Area of opening × Factor of radiation × Emissivity

$$= 12 \times 10000 \times 0.78 \times 0.8$$

$$= 74880 \text{ Kcal/hr}$$

Equivalent heat loss = 74880/ 8001.33

$$= 9.35 \text{ Kg/hr}$$

$$\% \text{ of heat loss} = 9.35 / 50 \times 100$$

$$= 18.71 \%$$

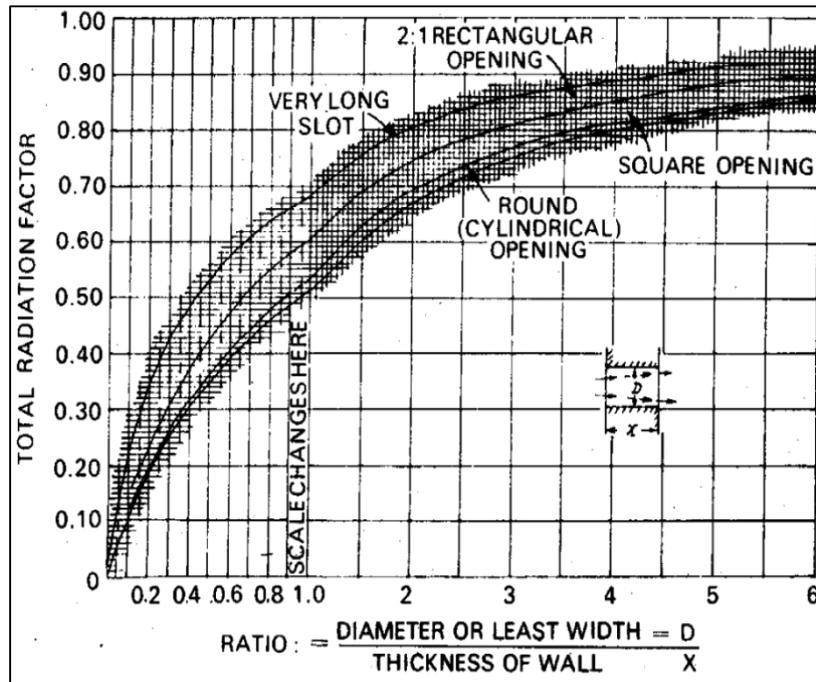


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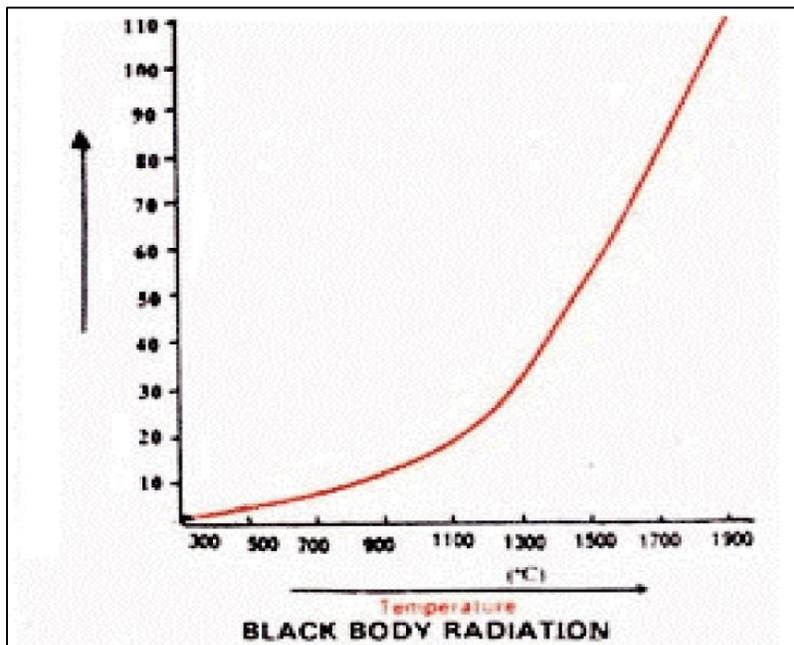


Fig. 2.5: Graph for Determining Black Body Radiation at a Particular Temperature

6) Step 6) Unaccounted Heat:

$$\begin{aligned} \text{Furnace efficiency} &= 100 - \text{Total losses} \\ &= 100 - 86.05 \\ &= 13.95\% \end{aligned}$$

1) To Calculate the Length of Insulation (With or Without Insulation)

Without insulation:

$$\begin{aligned} R_{\text{total}} &= R_1 + R_2 + R_3 \\ &= 1/H_i A + L/KA + 1/H_o A \\ &= 1/8 \times 2.5 [(1/125) + (0.6/1.45) + (1/70)] \\ &= 4 \times 10^{-4} + 0.020 + 7.1428 \times 10^{-4} \\ &= 0.0211 \text{ } ^\circ\text{K/W} \end{aligned}$$

$$\begin{aligned} Q \text{ without insulation} &= T_1 - T_o / R_{\text{total}} \\ &= 750 - 45 / 0.0211 \\ &= 33.41 \text{ KW} \end{aligned}$$

To calculate inside wall temp.

$$\begin{aligned} Q &= T_1 - T_1 / R_1 \\ 33412.32 &= 750 - T_1 / 4 \times 10^{-4} \\ T_1 &= 736.63 \text{ } ^\circ\text{C} \end{aligned}$$

To calculate the outside wall temp.

$$\begin{aligned} Q &= T_1 - T_2 / R_2 \\ 33412.32 &= 736.63 - T_2 / 0.020 \\ T_2 &= 68.43 \text{ } ^\circ\text{C} \end{aligned}$$

D. Heat Transfer through Furnace with Insulation

1) Insulation: Gypsum powder ($K = 0.4 \text{ W/M}^0\text{c}$)

We have to reduced 50% of total heat

$$\begin{aligned} \text{i.e } Q_1 &= 0.5 \times Q \\ &= 0.5 \times 33412.32 \\ &= 16.70 \text{ KW} \end{aligned}$$

$$\begin{aligned} \text{i.e } Q_1 &= T_2 - T_3 / R_{\text{insulation}} \\ 16700 &= 68.43 - 30 / R_{\text{insulation}} \end{aligned}$$

$$R_{\text{insulation}} = 2.30 \times 10^{-3}$$

$$L/KA = 2.30 \times 10^{-3}$$

$$L = 2.85 \times 10^{-3} \times 0.4 \times 8 \times 2.5$$

$$L = 18.4 \text{ mm}$$

2) Insulation: Asbestos fiber ($K = 0.115 \text{ W/M}^0\text{c}$)

$$\text{i.e } Q_1 = T_2 - T_3 / R_{\text{insulation}}$$

$$16700 = 68.43 - 30 / R_{\text{insulation}}$$

$$R_{\text{insulation}} = 2.30 \times 10^{-3}$$

$$L/KA = 2.30 \times 10^{-3}$$

$$L = 2.30 \times 10^{-3} \times 0.115 \times 8 \times 2.5$$

$$L = 5.08 \text{ mm}$$

3) Insulation: Ceramic fiber ($K = 0.12 \text{ W/M}^0\text{c}$)

i.e $Q_1 = T_2 - T_3 / R_{\text{insulation}}$

$$16700 = 68.3 - 30 / R_{\text{insulation}}$$

$$R_{\text{insulation}} = 2.30 \times 10^{-3}$$

$$L/KA = 2.30 \times 10^{-3}$$

$$L = 2.30 \times 10^{-3} \times 0.12 \times 8 \times 2.5$$

$$L = 5.52 \text{ mm}$$

E. Calculation for Bituminous LPG Fuel

Oil: - LPG

Plate Material: - Mild Steel

1) Heat Supplied by fuel (Q_s):-

$$Q_s = m_f \times C.V.$$

Where:-

m_f = mass of Fuel consumption/hr

$$= 2000 \text{ Lit/day}$$

$$= 1020 \text{ kg/day}$$

$$= 30600 / (1 \times 26 \times 960 \times 60)$$

$$M_f = 0.024 \text{ kg/sec}$$

C.V = Calorific Value = 10650 K-Cal/kg

$$= 10650 \times 4.1868 \text{ (1K-Cal/kg=4.1868KJ/kg)}$$

$$= 48776.22 \text{ KJ/kg}$$

$$Q_s = 0.024 \times 48776.22$$

$$Q_s = 1170.62 \text{ K-W}$$

1) Step 1) To calculate sensible heat loss:

$$\text{Sensible heat loss} = M_g \times C_{pg} \times (T_f - T_g)$$

$$= M_f \times (A/F + 1) \times C_{pg} \times (T_f - T_g)$$

$$= 0.024 \times 13.1 \times 1.005 \times (390 - 50)$$

$$Q_{\text{sensible}} = 107.43 \text{ KW}$$

% of heat loss in flue gas = $Q_{\text{sensible}} / \text{heat supply by fuel}$

$$= 107.43 / 1170.62$$

$$= 9.17 \%$$

2) Step 2) To calculate the heat loss due to heat carried away in stock:

$$\text{heat absorbed by plate} = M_{\text{plate}} \times C_{p \text{ plate}} \times (T_{p2} - T_a)$$

$$= 160 \times 0.416 \times (680 - 50)$$

$$= 41328$$

$$= 41328 / 20 \text{ min}$$

$$= 2066.4$$

$$= 34.44 \text{ KW}$$

% of heat utilizing by plate = $Q_{\text{absorbed}} / \text{heat supply by fuel}$

$$= 34.44 / 1170.62$$

$$= 2.94 \%$$

3) Step 3) heat loss due to moisture :

$$Q_{\text{moisture}} = M_m \times C_{pm} \times (T_f - T_a)$$

where M_m = Mass of moisture

$$= 9 \times \% \text{ of hydrogen}$$

$$= 9 \times 17.85 / 100 \times 0.0311$$

$$= 0.0385$$

$$= 0.0385 \times 2.1 \times (390 - 50)$$

$$Q_{\text{moisture}} = 27.48 \text{ KW}$$

% of heat loss in flue gas = $Q_{\text{Moisture}} / \text{heat supply by fuel}$

$$= 27.48 / 1170.62$$

$$= 2.34 \%$$

4) Step 4) calculate heat loss due to radiation and convection heat loss from the furnace outer surface :

$$Q_{\text{surface}} = [9.7 T_{\text{wall}}^2 / 1000 - 1.42 T_{\text{wall}} + 164] - [(T_{\text{amb}} - 75) / 50 \times (0.085 \times (T_{\text{wall}} - 100) + 90)]$$

Reference W Trinks industrial furnace John wiley & sons

Where, A = surface area of furnace in m²

T_{wall} = wall surface temp. In F.

$$= [9.7 \times (1234.4)^2/1000 - 1.42 \times 1234.4 + 164] - [(52.5 - 75)/50 \times (0.085 \times (1234.4 - 100) + 90)]$$

$$= 113191.46 - (-93.21)$$

$$= 13284.07 \text{ Kj/kg}$$

% of heat loss in flue gas = Q surface/ CV of fuel

$$= 13284.07/ 48776.22$$

$$= 27.23 \%$$

5) Step 5) Heat Loss Due To Opening:

The shape of is opening is Rectangular and ratio = D/X

$$= 1.5/0.6$$

$$= 2.5$$

With respect to value 2.5 from chart

Factor of radiation = 0.78

Black body radiation corresponding to 680 0c = 7 Kcal/cm² / hr (from chart)

Area of opening = 100cm × 100cm

$$= 10000 \text{ cm}^2$$

Emissivity = 0.8

Total heat loss = Black body radiation × Area of opening × Factor of radiation × Emissivity

$$= 7 \times 10000 \times 0.78 \times 0.8$$

$$= 43680 \text{ Kcal/hr}$$

Equivalent heat loss = 43680/10700

$$= 4.08 \text{ Kg/hr}$$

$$\% \text{ of heat loss} = 4.08/ 42.5 \times 100$$

$$= 9.60 \%$$

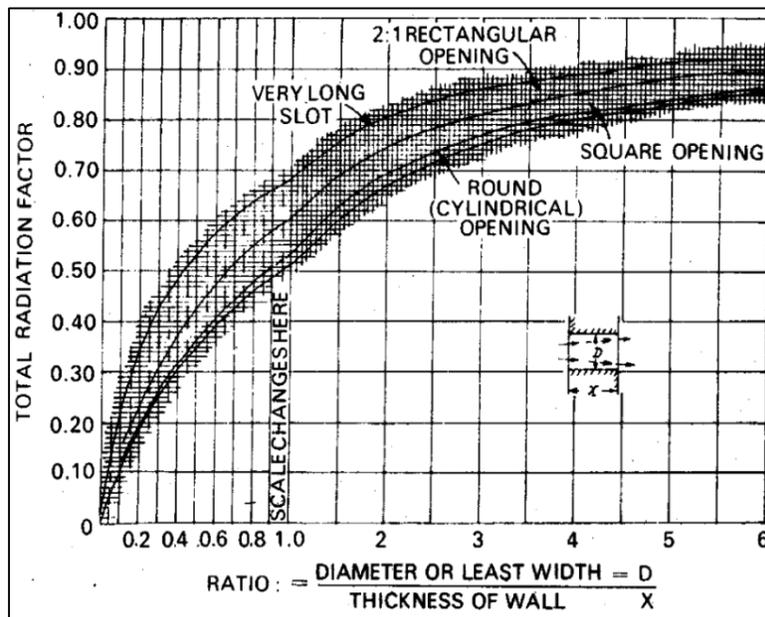


Fig. 2.4: Factor for Determining the Equivalent of Heat Release from Openings to the Quality of Heat Release from Perfect Black Body.

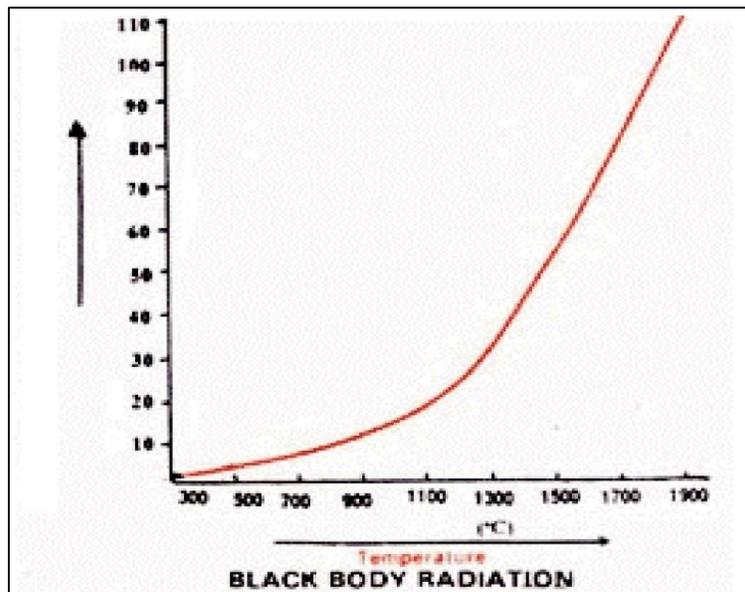


Fig. 2.5: Graph for Determining Black Body Radiation at a Particular Temperature

6) Step 6) Unaccounted Heat :

$$\begin{aligned} \text{Furnace efficiency} &= 100 - \text{Total losses} \\ &= 100 - 51.28 \\ &= 48.72 \% \end{aligned}$$

To Calculate The Length Of Insulation (With or Without Insulation):

Without insulation

$$\begin{aligned} R_{\text{total}} &= R_1 + R_2 + R_3 \\ &= 1/H_i A + L/KA + 1/H_o A \\ &= 1/8 \times 2.5 [(1/125) + (0.6/1.45) + (1/70)] \\ &= 4 \times 10^{-4} + 0.020 + 7.1428 \times 10^{-4} \\ &= 0.0211 \text{ } ^\circ\text{K/W} \end{aligned}$$

$$\begin{aligned} Q \text{ without insulation} &= T_i - T_o / R_{\text{total}} \\ &= 680 - 50 / 0.0211 \\ &= 29.85 \text{ KW} \end{aligned}$$

To Calculate inside wall temp.

$$\begin{aligned} Q &= T_i - T_1 / R_1 \\ 29857.81 &= 680 - T_1 / 4 \times 10^{-4} \\ T_1 &= 668.05 \text{ } ^\circ\text{C} \end{aligned}$$

To calculate the outside wall temp.

$$\begin{aligned} Q &= T_1 - T_2 / R_2 \\ 29857.81 &= 668.05 - T_2 / 0.020 \\ T_2 &= 71.05 \text{ } ^\circ\text{C} \end{aligned}$$

F. Heat Transfer through Furnace with Insulation

1) Insulation : Gypsum powder (K= 0.4 W/M⁰c)

We have to reduced 50% of total heat

$$\begin{aligned} \text{i.e } Q_1 &= 0.5 \times Q \\ &= 0.5 \times 29857.81 \\ &= 14.92 \text{ KW} \end{aligned}$$

$$\begin{aligned} \text{i.e } Q_1 &= T_2 - T_3 / R_{\text{insulation}} \\ 14920 &= 71.05 - 30 / R_{\text{insulation}} \end{aligned}$$

$$R_{\text{insulation}} = 2.75 \times 10^{-3}$$

$$L/KA = 2.75 \times 10^{-3}$$

$$L = 2.75 \times 10^{-3} \times 0.4 \times 8 \times 2.5$$

$$L = 22.01 \text{ mm}$$

IX. CONCLUSION

The analysis of the previous system is done to know the drawbacks of system on other fuel. To suggest alternative fuel like light diesel oil, coal, liquid petroleum gas and finally LDO is selected. Properties of alternative fuel suggested i.e. LDO is also studied like cost, availability, flash point etc. due to the use of LDO, the company is in profit as compare to previous fuel used in the industry.

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