Design and Fabrication of Heat Exchanger for Co-Generation

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Abstract— Heat Exchanger device which transfer of heat one place to other place with minimum losses. In this paper we calculate numerically heat obtain by CI engine at applied of different load. According to that we calculate design of compact type of heat exchanger and fabricated it. This fabricated heat exchanger used at exhaust of CI engine .That will useful for textile industry where we required hot water and steam water. The waste heat exchanger was designed to recover heat available in the exhaust at 75- 80% of full load of the engine because the efficiency of engine is maximum at this load. We calculate numerically recovery of heat.

Key words: Heat Exchanger, Heat Transfer, Co-generation, Heat Exchanger Design

I. INTRODUCTION

A heat exchanger is a device which transfers the energy from a hot fluid to a cold fluid, with maximum rate and minimum investment and running cost. A waste heat operated compact type heat exchanger was designed and fabricated for producing hot water. The waste heat exchanger was designed to recover heat available in the exhaust at 75- 80% of full load of the engine because the efficiency of engine is maximum at this load.

II. DESIGN OF COMPACT TYPE HEAT EXCHANGER

For designing of heat exchanger, at the beginning the data related to the properties of engine exhaust were observed and calculated, which are shown in Table. Here for calculation purpose, some of the properties of exhaust gases like viscosity, prandtl number etc. were taken for air due to easy availability in property tables.

Properties	Hot fluid (Exhaust gases)	Cold fluid (Water)
M	29.01 kg/hr	2 kg
Tinlet (^{0}C)	Tg_1= 370	Tw_1=30
Toutlet (^{0}C)	Tg_2= 205.62 (From energy balance)	Tw_2 =90
C_p	1.2	4.18
K_1 = Const. for bundle diameter = 0.319		
n_1 = Const. in bundle diameter equation = 2.142		

Table 1: Properties of Hot Fluid (Gas) and Cold Fluid (Water)

A. Steps for Heat Exchanger Design

1) Step: I

As per exhaust gas temperature and higher thermal conductivity copper (k_w = 378 w/m^{0}C) was selected for tube construction and finalized tube's diameter, length.

2) Step: II

Heat duty equation was applied for both fluids (hot and cold) for calculating actual heat exchange rate between two fluids; the equation is given as following

$$q = m_w c_{pw}(T_{w2} - T_{w1}) = m_g c_{pg}(T_{g1} - T_{g2})$$

Where, subscripts c and h refer to cold and hot streams. Then heat exchange rate obtained (q).

3) Step: III

LMTD (Log mean temperature difference) was calculated for counter flow type heat exchanger by using given equation.

$$\text{For counter flow LMTD} = \frac{\theta_1 - \theta_2}{\ln\left(\frac{\theta_1}{\theta_2}\right)}$$

Where θ_1 and θ_2 are temperature difference

4) Step: IV Temperature correction factor

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)}$$

$$S = \frac{(t_2 - t_1)}{(T_1 - T_2)}$$

Temperature correction factor was calculated by chart

5) Step: V The mean temperature difference was calculated by-

$$DT_m = F_t \times LMTD$$

Where Ft = Temp. Correction factor

LMTD = Log mean temperature difference

6) Step: VI Over all heat transfer coefficient for hot and cold fluid took from table.

$$U = 250 \text{ w/m}^2 \text{ c}$$

Then Area 'A' was calculated as below equation

$$q = U.A.DT_m$$

$$A = \frac{q}{U.D_m}$$

Now after calculating the area of heat transfer, diameter and length of tube can be finding out by using following relation [24]

$$A = n.\pi.d.l$$

Where,

n = No. of tubes

d = Diameter of tube

l = Length of tube

7) Step: VII Calculation for pitch and bundle diameter by using following equation

$$P_t = 1.25 d_o$$

$$\text{and } D_b = d_o \left(\frac{N_t}{K_1}\right)^{1/n1}$$

Where

N_t = Number of tubes

D_b = Bundle diameter, mm

D_o = Tube outside diameter, mm

Where, K_1 and n_1 are obtained from the table.

8) Step: VIII Calculation for Shell diameter and baffle spacing

$$D_s = Db + BDC$$

$$\text{and } B_s = 0.4 D_s$$

where

D_s = Shell diameter

BDC = Bundle diameter clearance (calculated from chart)

B_s = Baffle spacing

9) Step: XI Calculation for area of cross flow and shell side mass velocity

$$\text{For cross flow area, } A_s = \frac{(p_t - d_o) D_s . B_s}{p_t}$$

Shell- side mass velocity, $G_s = \frac{\text{Shell-side flowrate}\left(\frac{kg}{s}\right)}{A_s}$

10) Step: X Reynolds number (R_e) and Prandtle number (P_r) Reynolds number calculated by

$$R_e = \frac{G_s d_e}{\mu} = \frac{\mu_s d_e \rho}{\mu}$$

Prandtle number given by

$$Pe = \frac{\mu . C_p}{\mu}$$

III. Calculations for Heat Exchanger Design

A. *When Engine Operating At Load 9.9 Kg*

(a) Mean effective radius (R_e) = $\frac{d_B + 2d_R}{2}$ m, = $\frac{0.20+2(0.015)}{2}$ = 0.115 m Where, R_e = Mean effective radius, m

(b) Torque (T) = $(W_1\text{-}W_2)\times g\times R_e$

Where, W_1 and W_2 = Spring balance weight and Dead weight, kg T = (10-0.1) ×9.81×0.115 = 11.168 N-m

(c) Brake Power (B.P.) = $\frac{2\pi NT}{60\times1000}$ kW

Where N = Revolution per minute

T = Torque

B.P. = $\frac{2\pi\times1500\times11.168}{60\times1000}$ = 1.753 kW

(d) Specific fuel Consumption ($\dot{m}_f$) = $\frac{\text{Volume of fuel consumption(x),ml}}{\text{Time taken for x ml,in second}} \times \frac{\text{Density of fuel}}{10^6}$

$(\dot{m}_f) = \frac{x}{t} \times \frac{\rho_f}{10^6} = \frac{10}{51.44} \times \frac{804}{10^6} = 1.56\times10^{-4}$ kg/sec

(e) Brake specific fuel consumption ($\dot{m}_{sf}$) = $\frac{\dot{m}_f}{B.P.}$

$\dot{m}_{sf} = \frac{1.56\times10^{-4}}{1.753} = 8.89\times 10^{-5}$ kg/kW.sec

(f) Heat supplied by fuel (Q_{in}) = $C_v \times \dot{m}_f$ = 43500×1.56×10^{-4} = 6.786 kW

(g) Mass flow rate of water in cooling jacket ($\dot{m}_w$) = $\frac{V_E}{t_E} \times \frac{\rho_w}{10^3}$ $\dot{m}_w = \frac{1}{15.8} \times \frac{1000}{10^3} = 6.329\times10^{-2}$ kg/sec

(h)Heat carried out by water (Q_w) = $\dot{m}_w \times C_{pw} \times (T_2\text{-}T_1)$Here T_1 & T_2 Temp. of water inlet & outlet of engine , ^{0}C

Q_w = 6.329×10^{-2}×4.186× (36-30) = 1.589 kW

(i) Head causing flow of air through orifice (H) = $\frac{H_1-H_2}{100} \times \left(\frac{\rho_w}{\rho_a} - 1\right)$H = $\frac{22-9.8}{100} \times \left(\frac{1000}{1.293} - 1\right)$ = 94.232 m

(j) Area of orifice (A_o) = $\frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times (0.017)^2$ = 2.27×10^{-4} m^2

(k) So, Vol. rate of air (V_a) = $C_d\times A_o\times\sqrt{2gH}$

V_a = 0.64×2.27×10^{-4}×$\sqrt{2 \times 9.81 \times 94.23}$ = 6.246×10^{-3} m^3/sec

(l) Swept Volume (V_s) = $\frac{\pi}{4} \times \frac{(d)^2\times L\times NOC\times N}{60\times n}$

Where d = Dia. of cylinder, m

L = Length of stroke, m

NOC = No. of cylinder

N= Revolution per minute

$V_s = \frac{\pi}{4} \times \frac{(0.08)^2\times0.11\times1\times1500}{60\times2}$ = 6.908×10^{-3}m^3/sec

(m) Mass flow rate of air ($\dot{m}_a$) = $V_a\times \rho_a$ = 6.246×10^{-3}×1.293 = 8.076×10^{-3} kg/sec

(n) Mass flow rate of gas ($\dot{m}_g$) = $\dot{m}_a + \dot{m}_f$

$\dot{m}_g$ = (8.076+.156)×10^{-3} = 8.232×10^{-3} kg/sec

(o) Exhaust heat (Q_{exh}) = $\dot{m}_g \times Cpg \times (T_g - T_{atm})$

Where T_g and T_{atm} are temperature of exhaust gas and temperature of atmosphere

Q_{exh} = 8.232×10^{-3}×1.09×(164-27) = 1.229 kW

(p) Uncountable heat (Q_{un}) = Q_{in} – (Q_{exh}+B.P.+Q_w)

Q_{un} = 6.786 – (1.229+2.665+1.589) =

(q) Brake thermal efficiency ($\eta_{B.T.}$) = $\frac{B.P.}{Q_{in}} = \frac{1.753}{6.786}$×100 = 25.83%

(r) Volumetric Efficiency (V.E.) = $\frac{V_a}{V_s}$×100 = $\frac{6.246\times10^{-3}}{6.908\times10^{-3}}$×100 = 90.41%

B. *When Engine Operating At Load 19 Kg*

Torque (T) = $(W_1\text{-}W_2)\times g\times R_e$

Where, W_1 and W_2 = Spring balance weight and Dead weight, kg T = (20-1) ×9.81×0.115 =21.438 N-m

Brake Power (B.P.) = $\frac{2\pi NT}{60\times1000}$ kW

Where N = Revolution per minute

T = Torque

B.P. = $\frac{2\pi\times1450\times22.438}{60\times1000}$ = 3.25 kW

Specific fuel Consumption ($\dot{m}_f$) = $\frac{\text{Volume of fuel consumption(x),ml}}{\text{Time taken for x ml,in second}} \times \frac{\text{Density of fuel}}{10^6}$

$(\dot{m}_f) = \frac{x}{t} \times \frac{\rho_f}{10^6} = \frac{10}{37.34} \times \frac{804}{10^6} = 2.15\times10^{-4}$ kg/sec

Brake specific fuel consumption ($\dot{m}_{sf}$) = $\frac{\dot{m}_f}{B.P.}$

$\dot{m}_{sf} = \frac{2.15\times10^{-4}}{3.25} = 6.615\times 10^{-5}$ kg/kW.sec

Heat supplied by fuel (Q_{in}) = $C_v \times \dot{m}_f$ = 43500×2.15×10^{-4} = 9.3525 kW

Mass flow rate of water in cooling jacket ($\dot{m}_w$) = $\frac{V_E}{t_E} \times \frac{\rho_w}{10^3}$

$\dot{m}_w = \frac{1}{15.8} \times \frac{1000}{10^3} = 6.329\times10^{-2}$ kg/sec

Heat carried out by water (Q_w) = $\dot{m}_w \times C_{pw}\times (T_2\text{-}T_1)$

Here T_1 & T_2 Temp. of water inlet & outlet of engine , ^{0}C

Q_w = 6.329×10^{-2}×4.186× (38-31) = 1.85 kW

Head causing flow of air through orifice (H) = $\frac{H_1-H_2}{100} \times \left(\frac{\rho_w}{\rho_a} - 1\right)$

H = $\frac{22-10.3}{100} \times \left(\frac{1000}{1.293} - 1\right)$ = 88.05 m

Area of orifice (A_o) = $\frac{\pi}{4} \times d^2 = \frac{\pi}{4} \times (0.017)^2$ = 2.27×10^{-4} m^2

So, Vol. rate of air (V_a) = $C_d\times A_o\times\sqrt{2gH}$

V_a = 0.64×2.27×10^{-4}×$\sqrt{2 \times 9.81 \times 88.05}$ = 6.038×10^{-3} m^3/sec

Swept Volume (V_s) = $\frac{\pi}{4} \times \frac{(d)^2\times L\times NOC\times N}{60\times n}$

Where d = Dia. of cylinder, m

L = Length of stroke, m

NOC = No. of cylinder

N= Revolution per minute

$V_s = \frac{\pi}{4} \times \frac{(0.08)^2\times0.11\times1\times1450}{60\times2}$ = 6.677×10^{-3} m^3/sec

Mass flow rate of air ($\dot{m}_a$) = $V_a\times \rho_a$ = 6.038×10^{-3}×1.293 = 7.807×10^{-3} kg/sec

Mass flow rate of gas ($\dot{m}_g$) = $\dot{m}_a + \dot{m}_f$

$\dot{m}_g$ = (7.807+.215)×10^{-3} = 8.022134×10^{-3} kg/sec

Exhaust heat (Q_{exh}) = $\dot{m}_g \times Cpg \times (T_g - T_{atm})$

Where T_g and T_{atm} are temperature of exhaust gas and temperature of atmosphere

Q_{exh} = 8.0221×10^{-3}×1.09×(210-27) = 1.600 kW

Uncountable heat (Q_{un}) = Q_{in} – (Q_{exh}+B.P.+Q_w)

Q_{un} =9.3525 – (1.6+3.25+1.85) = 2.65 kW

Brake thermal efficiency ($\eta_{B.T.}$) = $\frac{B.P.}{Q_{in}} = \frac{3.25}{9.3525}$×100 = 34.75%

Volumetric Efficiency (V.E.) = $\frac{V_a}{V_s}\times100 = \frac{6.038\times10^{-3}}{6.677\times10^{-3}}\times100$ = 90.42%

C. *When Engine Operating At Load 26 Kg*

Torque (T) = $(W_1\text{-}W_2)\times g\times R_e$

Where, W_1 and W_2 = Spring balance weight and Dead weight, kg

T = (30-4) ×9.81×0.115 =29.3319 N-m

Brake Power (B.P.) = $\frac{2\pi NT}{60\times1000}$ kW

Where N = Revolution per minute

T = Torque

B.P. = $\frac{2\pi\times1428\times29.3319}{60\times1000}$ = 4.384 kW

Specific fuel Consumption ($\dot{m}_f$) = $\frac{\text{Volume of fuel consumption(x),ml}}{\text{Time taken for x ml,in second}}\times\frac{\text{Density of fuel}}{10^6}$

$(\dot{m}_f) = \frac{x}{t}\times\frac{\rho_f}{10^6} = \frac{10}{31.03}\times\frac{804}{10^6} = 2.59\times10^{-4}$ kg/sec

Brake specific fuel consumption ($\dot{m}_{sf}$) = $\frac{\dot{m}_f}{B.P.}$

$\dot{m}_{sf} = \frac{2.59\times10^{-4}}{4.384} = 0.5907\times10^{-4}$ kg/kW.sec

Heat supplied by fuel (Q_{in}) = $C_v\times\dot{m}_f$ = 43500×2.59×10^{-4} = 11.2665 kW

Mass flow rate of water in cooling jacket ($\dot{m}_w$) = $\frac{V_E}{t_E}\times\frac{\rho_w}{10^3}$

$\dot{m}_w = \frac{1}{15.8}\times\frac{1000}{10^3} = 6.329\times10^{-2}$ kg/sec

Heat carried out by water (Q_w) = $\dot{m}_w\times C_{pw}\times(T_2\text{-}T_1)$

Here T_1 & T_2 Temp. of water inlet & outlet of engine , ^{0}C

$Q_w = 6.329\times10^{-2}\times4.186\times(41\text{-}30) = 2.38$ kW

Head causing flow of air through orifice (H) = $\frac{H_1-H_2}{100}\times(\frac{\rho_w}{\rho_a}-1)$

H = $\frac{21.5-10.5}{100}\times(\frac{1000}{1.293}-1) = 84.96$ m

Area of orifice (A_o) = $\frac{\pi}{4}\times d^2 = \frac{\pi}{4}\times(0.017)^2 = 2.27\times10^{-4}$ m^2

So, Vol. rate of air (V_a) = $C_d\times A_o\times\sqrt{2gH}$

$V_a = 0.64\times2.27\times10^{-4}\times\sqrt{2\times9.81\times84.96} = 5.9314\times10^{-3}$ m^3/sec

Swept Volume (V_s) = $\frac{\pi}{4}\times\frac{(d)^2\times L\times NOC\times N}{60\times n}$

Where d = Dia. of cylinder, m

L = Length of stroke, m

NOC = No. of cylinder

N= Revolution per minute

$V_s = \frac{\pi}{4}\times\frac{(0.08)^2\times0.11\times1\times1428}{60\times2} = 6.575\times10^{-3}$ m^3/sec

Mass flow rate of air ($\dot{m}_a$) = $V_a\times\rho_a = 5.9314\times10^{-3}\times1.293$ = 7.6693×10^{-3} kg/sec

Mass flow rate of gas ($\dot{m}_g$) = $\dot{m}_a + \dot{m}_f$

$\dot{m}_g = (7.6693+.259)\times10^{-3} = 7.928\times10^{-3}$ kg/sec

Exhaust heat (Q_{exh}) = $\dot{m}_g\times Cpg\times(T_g - T_{atm})$

Where T_g and T_{atm} are temperature of exhaust gas and temperature of atmosphere

$Q_{exh} = 7.928\times10^{-3}\times1.09\times(286\text{-}27) = 2.238$ kW

Uncountable heat (Q_{un}) = $Q_{in} - (Q_{exh}$+B.P.+$Q_w)$

Q_{un} =11.2665 – (2.238+4.38+2.238) = 2.264 kW

Brake thermal efficiency ($\eta_{B.T.}$) = $\frac{B.P.}{Q_{in}} = \frac{4.384}{11.2665}\times100$ = 38.91%

Volumetric Efficiency (V.E.) = $\frac{V_a}{V_s}\times100 = \frac{5.9314\times10^{-3}}{6.575\times10^{-3}}\times100$ = 90.21%

D. *When Engine Operating At Load 32 Kg*

Torque (T) = $(W_1\text{-}W_2)\times g\times R_e$

Where, W_1 and W_2 = Spring balance weight and Dead weight, kg

T = (40-8) ×9.81×0.115 =36.1008 N-m

Brake Power (B.P.) = $\frac{2\pi NT}{60\times1000}$ kW

Where N = Revolution per minute

T = Torque

B.P. = $\frac{2\pi\times1413\times36.1008}{60\times1000}$ = 5.339 kW

Specific fuel Consumption ($\dot{m}_f$) = $\frac{\text{Volume of fuel consumption(x),ml}}{\text{Time taken for x ml,in second}}\times\frac{\text{Density of fuel}}{10^6}$

$(\dot{m}_f) = \frac{x}{t}\times\frac{\rho_f}{10^6} = \frac{10}{23.45}\times\frac{804}{10^6} = 3.428\times10^{-4}$ kg/sec

Brake specific fuel consumption ($\dot{m}_{sf}$) = $\frac{\dot{m}_f}{B.P.}$

$\dot{m}_{sf} = \frac{3.428\times10^{-4}}{5.339} = 0.64217\times10^{-4}$ kg/kW.sec

Heat supplied by fuel (Q_{in}) = $C_v\times\dot{m}_f$ = 43500×3.428×10^{-4} = 14.9139 kW

Mass flow rate of water in cooling jacket ($\dot{m}_w$) = $\frac{V_E}{t_E}\times\frac{\rho_w}{10^3}$

$\dot{m}_w = \frac{1}{15.8}\times\frac{1000}{10^3} = 6.329\times10^{-2}$ kg/sec

Heat carried out by water (Q_w) = $\dot{m}_w\times C_{pw}\times(T_2\text{-}T_1)$

Here T_1 & T_2 Temp. of water inlet & outlet of engine , ^{0}C

$Q_w = 6.329\times10^{-2}\times4.186\times(46\text{-}31) = 3.951$ kW

Head causing flow of air through orifice (H) = $\frac{H_1-H_2}{100}\times(\frac{\rho_w}{\rho_a}-1)$

H = $\frac{21.5-11}{100}\times(\frac{1000}{1.293}-1) = 78.01$ m

Area of orifice (A_o) = $\frac{\pi}{4}\times d^2 = \frac{\pi}{4}\times(0.017)^2 = 2.27\times10^{-4}$ m^2

So, Vol. rate of air (V_a) = $C_d\times A_o\times\sqrt{2gH}$

$V_a = 0.64\times2.27\times10^{-4}\times\sqrt{2\times9.81\times78.01} = 5.683\times10^{-3}$ m^3/sec

Swept Volume (V_s) = $\frac{\pi}{4}\times\frac{(d)^2\times L\times NOC\times N}{60\times n}$

Where d = Dia. of cylinder, m

L = Length of stroke, m

NOC = No. of cylinder

N= Revolution per minute

$V_s = \frac{\pi}{4}\times\frac{(0.08)^2\times0.11\times1\times1413}{60\times2} = 6.506\times10^{-3}$ m^3/sec

Mass flow rate of air ($\dot{m}_a$) = $V_a\times\rho_a = 5.683\times10^{-3}\times1.293$ = 7.348×10^{-3} kg/sec

Mass flow rate of gas ($\dot{m}_g$) = $\dot{m}_a + \dot{m}_f$

$\dot{m}_g = (7.348+.3428)\times10^{-3} = 7.6909\times10^{-3}$ kg/sec

Exhaust heat (Q_{exh}) = $\dot{m}_g\times Cpg\times(T_g - T_{atm})$

Where T_g and T_{atm} are temperature of exhaust gas and temperature of atmosphere

$Q_{exh} = 7.6909 10^{-3}\times1.09\times(370\text{-}27) = 2.875$ kW

Uncountable heat (Q_{un}) = $Q_{in} - (Q_{exh}$+B.P.+$Q_w)$

Q_{un} =14.9139 – (2.875+5.339+3.951) = 2.748 kW

Brake thermal efficiency ($\eta_{B.T.}$) = $\frac{B.P.}{Q_{in}} = \frac{5.339}{14.9139}\times100$ = 35.79%

Volumetric Efficiency (V.E.) = $\frac{V_a}{V_s}\times100 = \frac{5.683\times10^{-3}}{6.506\times10^{-3}}\times100$ = 87.35%

E. *Average Calculative Observations*

S.No.	Parameters name	Average value
1	Torque (T)	19.60 N-m
2	Brake Power (B.P.)	2.945 kW

3	Specific fuel consumption ($\dot{m}_f$)	2.12×10^{-4} kg/sec
4	Heat supplied by fuel (Q_{in})	9.239 kW
5	Heat carried out by water (Q_w)	2.218 kW
6	Mass flow rate of air ($\dot{m}_a$)	7.847×10^{-3}
7	Mass flow rate of gas ($\dot{m}_g$)	8.0598×10^{-3}
8	Exhaust heat (Q_{exh})	1.7628 kW
9	Uncountable heat (Q_{un})	2.1284 kW
10	Volumetric Efficiency (V.E.)	89.802 %
11	Temperature	230.4 ^{0}C

Table 2: Average Calculative Observations

F. Emission Performance Of Single Generation

CO	0.019	0.017	0.015	0.012
CO_2	2.09	2.48	2.86	2.89
O_2	17.59	17.09	16.85	16.8
NO_x	351	495	648	652

Table 3: Emission performance of single generation

IV. Calculation for Compact Type Heat Exchanger

1) Tube Diameter (d) = 25 mm
2) Tube Length (L) = 2.5 m = 2500 mm
3) Material for construction of tube = Cu
4) Thermal conductivity (k_w) = 378 w/m^{0}C)
5) Using heat duty equation $m_w\, cp_w\,(T_{w2} - T_{w1}) = m_g\, cp_w\,(T_{G1} - T_{G2})$
   $6.329\times10^{-2}\times4.186\times10^{3}\ (90\text{-}30) = 8.059\times10^{-2}\times1.2\times10^{3}(370\text{-}\ T_{G2})$
   $164.37 = 370\text{-}\ T_{G2}$
   $T_{G2} = 205.62\ ^0C$
6) For counter flow LMTD $(\theta_m) = \frac{\theta_1-\theta_2}{\lg(\frac{\theta_1}{\theta_2})}$
   Where θ_1 and θ_2 temperature difference of hot and cold fluid
   $\theta_1 = 370\text{-}90 = 280\ ^0C$
   $\theta_2 = 205.62\text{-}30 = 175.62\ ^0C$
   $(\theta_m) = \frac{280-175.62}{\lg(\frac{280}{175.62})} = 223.76\ ^0C$
7) Temperature correction factor
   $R = \frac{370-205.62}{90-30} = 2.73$
   F_t = .80 [From chart]
   Then the mean temperature difference
   $DT_m = F_t \times LMTD = .80\times223.76 = 179.08$
   Overall heat transfer coefficient
   U = 250 w/m^{2}c [for hot fluid gas & cold fluid water]
   =Then
   $q = UAT_m$
   $6.32\times10^{-2}\times60 = 250\times A\times179.08$
   A = 0.354 m
8) Calculations for Number of tubes
   $A = n\pi dl$
   $0.354 = n\times\pi\times(.2500)\times25\times10^{-3}$
   n = 1.80 = 2 tubes
   Where n = number of tubes,
   d = diameter of tube
   l = length of tubes
9) Tube pitch = $1.25d_0 = 1.25\times25 = 31.25$ mm
   Bundle diameter (D_b) = $d_0\left(\frac{N_t}{K_t}\right)^{1/n_1}$
   $D_b = 25\left(\frac{2}{0.319}\right)^{1/2.142} = 158.90$ mm
   Where K_t and n_1 are obtained from table [K_t = 0.319 and n_1 = 2.142]

   From the chart bundle diameter clearance for fixed and U- tube heat is 80 mm
10) Shell diameter D_s = Db+BDC = 158.90+80 = 238.90mm
11) Baffle spacing (B_s) = 0.4 D_s = 0.4×238.90 = 95.56 mm
12) Area for cross flow (A_s) = $\frac{(p_t-d_0)D_s.B_s}{p_t}$
    $A_s = \frac{(31.25-25)(138.90)(55.56)}{31.25} = 1543.45\ mm^2$
13) Shell side mass velocity (G_s) = $\frac{\text{shell-side flowrate}[\frac{kg}{s}]}{As}$
    $G_s = \frac{8.059\times10^{-2}}{1543.45} = .0552\ m^3 = 5.05\times10^{-2}$ m
14) Calculation for Reynolds number (R_e) and Prandtle number (P_r)
    $R_e = \frac{G_s d_e}{\mu} = \frac{5.05\times10^{-2}\times.13890}{32\times10^{-5}} = 56.11$
    $P_r = \frac{\mu\times C_p}{k} = \frac{.658\times10-6\times4186\times103}{385} = .715$
15) Pressure drop$(\Delta P_S) = 8j_f\left(\frac{D_s}{d_e}\right)\left(\frac{L}{l_B}\right)\frac{\rho u_s^2}{2}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$

$=8\times4.8\times10\left(\frac{.138}{.025}\right)\left(\frac{2.5}{.055}\right)\times1000\times\frac{(5.05\times10^{-2})^2}{2}\left(\frac{.20\times10^{-6}}{.658\times10^{-6}}\right)^{-0.14}$

    $J_f = 4.8\times10^{-1}$ From chart
    $\Delta P_S = .1792\ N/m^2$
16) Effectiveness = $\left[\frac{T_{w2}-T_{w1}}{T_{G1}-T_{G2}}\right] = \left[\frac{90-30}{370-205.62}\right] = .3650$

V. Effectiveness of Heat Exchanger at Different Load

A. When Engine Operating At 1.753 Kw

$T_{w2} = 75.3\ ^0C$ and $T_{w1} = 25.2\ ^0C$
$T_{G1} = 344.6\ ^0C$ and $T_{G2} = 173.37\ ^0C$
Effectiveness ($\in$) = $\left[\frac{T_{w2}-T_{w1}}{T_{G1}-T_{G2}}\right] = \left[\frac{75.2-25.2}{344.6-173.37}\right] = .2921$

B. When Engine Operating At 3.25 Kw

$T_{w2} = 86.8\ ^0C$ and $T_{w1} = 28.41\ ^0C$
$T_{G1} = 362.8\ ^0C$ and $T_{G2} = 201\ ^0C$
Effectiveness ($\in$) = $\left[\frac{T_{w2}-T_{w1}}{T_{G1}-T_{G2}}\right] = \left[\frac{86.8-28.41}{362.8-201}\right] = .3624$

C. When Engine Operating At 4.38 Kw

$T_{w2} = 92.4\ ^0C$ and $T_{w1} = 39.01\ ^0C$
$T_{G1} = 380.7\ ^0C$ and $T_{G2} = 234.94\ ^0C$
Effectiveness ($\in$) = $\left[\frac{T_{w2}-T_{w1}}{T_{G1}-T_{G2}}\right] = \left[\frac{92.4-39.01}{380.7-234.94}\right] = .3635$

D. When Engine Operating At 5.33 Kw

$T_{w2} = 94.6\ ^0C$ and $T_{w1} = 45.76\ ^0C$
$T_{G1} = 386.6\ ^0C$ and $T_{G2} = 247.32\ ^0C$
Effectiveness ($\in$) = $\left[\frac{T_{w2}-T_{w1}}{T_{G1}-T_{G2}}\right] = \left[\frac{94.6-45.76}{247.32-386.6}\right] = .3506$

VI. Observations of Parameters after Cogeneration

Load & Parameters	1.753 kW	3.25 kW	4.38 kW	5.33 kW
BTE (%)	25.7	34.5	37.9	34.1
BSFC (kg/kW-hr)	0.325	0.2399	0.2196	0.25
CO_2	2.08	2.46	2.84	2.87
O_2	17.58	17.07	16.81	16.6
NO_x	359	493	645	649
$\in$	0.2921	0.3624	0.3625	0.3506

Table 4: Observations of Parameters after Cogeneration

From the above mentioned results it is seen that waste heat recovery system is feasible.

VII. FABRICATION OF HEAT EXCHANGER

After getting all dimensions and material of shell, tube the fabrication was done. To fabricate the shell a pipe of diameter 250 mm was taken, according to availability in market and tube was made of copper. The copper tube arranged in shell and drilled at the entrance and exits of the shell for temperature measurement using thermocouples. The photographic view of the fabricated shell and waste heat operated heat exchanger.

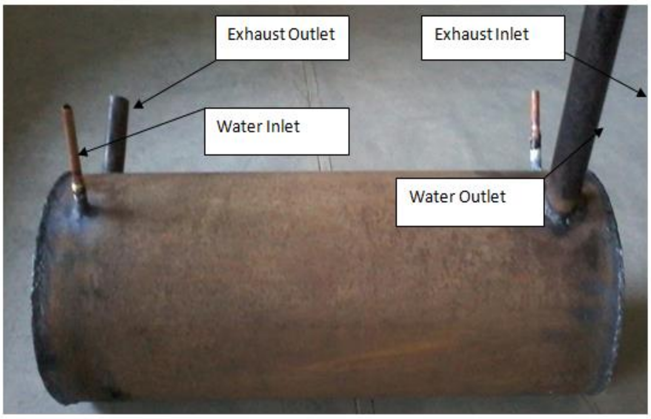

Fig. 1: View of Heat Exchanger in Laboratory

Fig. 2: Photographic views at Connection time

After the fabrication of waste heat operated heat exchanger insulation was done on the outer side of shell to reduce the heat loss from shell. After insulation the effectiveness of heat exchanger increases because heat loss to the surroundings decreases. Insulation was done by glass wool and insulation because these are cheap and easily available in the market.

After the fabrication and insulation the heat exchanger was coupled with the engine setup. The complete procedure of the setup is given in the next chapter.

VIII. INSULATION THICKNESS CALCULATION

It is required to reduce the heat loss through boundary. Thermal insulation thickness used for saving energy vary from condition to condition, but as in general rule, pipes operating at more-extreme temperatures exhibit a more heat flow and larger thicknesses are applied due to greater potential savings. Thickness of insulation was calculated as

Let,

Td = Desired or actual insulation surface temperature in ºC

K = Thermal conductivity of insulating material at mean temperature Tm (W/m-C)

tk = Thickness of insulation in mm

R_{st} = Surface thermal resistance = 1/h (oC-m2/W)

Rtr = Thermal resistance of insulation = tk/k (oC-m2/W)

$$T_m = \frac{(Tf1+ Td1)}{2}$$

$$R = (r+t_k)$$

The heat flow from the heat exchanger surface and the ambient can be expressed as

$$H = \frac{T_{f1}-T_{at}}{R_{tr}+R_{st}} = \frac{T_s-T_{at}}{R_{st}}$$

From the above expression for desired Td and Rtr can be calculated. From Rtr (Thermal resistance of insulation)and known value of thermal conductivity k, thickness of insulation can be calculated.

Equivalent thickness of insulation

$$(E_{tk}) = (r + t_k) \times \ln\left(\frac{R+t_k}{r}\right)$$

The above equation was used to find out the insulation thickness to reduce the heat loss from shell.

IX. MATERIAL SELECTION FOR HEAT EXCHANGER AND ITS FABRICATION

Material selection for heat exchangers, insulation depends upon the temperature of waste heat source. Corrosion and oxidation reactions, like all chemical reactions, are accelerated automatically by increasing in temperature. If the waste heat system contains corrosive substances, the heat recovery surfaces area can quickly become damaged. In addition, some materials like carbon steel at about above the 800°F [425ºC] and stainless steel above the 1,200°F [650ºC] begins to oxidize. Therefore, advanced alloys or composite materials must be used for high grade range temperatures. Mostly Metallic materials are not used at temperatures above 1,600°F [871ºC]. The choice of insulation material is generally depends upon cost and availability of the material.

X. RESULT

We succesfully calculate torque , break power, specific fuel consumption, heat supplied by fuel, heat carried out by waste ,mass flow rate of air, Exhaust heat,rate of volumetric efficiency ,temprature, emission performance of single generation by gas analysier, calculation of tube diameter, tube length, material for tube, number of tube, tube pitch , shell diameter , shell length.

After we fabricated compact type heat exchnger that used at exhaust of CI engine and calculate effectiveness of heat exchanger at different load emission performance of cogeneration.

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