FINAL REPORT ON SERB FUNDED PROJECT

Name of the Project: Development of an Innovative Model of Combined Heat and Power from Purely Producer Gas Based Engine Alternator System for Partial Conventional Energy Substitution of Tea Processing Industries in North-East India

File No: SR/FTP/ETA-52/2011 and 31/08/2012

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Annexure-IX

PROJECT COMPLETION REPORT

Notes:

1. Five copies of Project Completion Report (PCR) should be sent within one month or termination of project.

2. PCR should be in bound form.

3. The cover page should include the title of the project, file number, names and address of investigation.

1. Title of the project:

Development of an innovative model of Combined Heat and Power from Purely Producer Gas Based Engine Alternator System for Partial Conventional Energy Substitution of Tea Processing Industries in North-East India

2. Principal investigator & Co-investigator DR. PARTHA PRATIM DUTTA
3. Implementing institute and other collaborating institute : Tezpur University
4. Date of commencement : 20/09/2012
5. Planned date of completion : 31/03/2016
6. Actual date of completion : 31/03/2016
7. Objectives as stated in project proposal:

(a) Tea production and processing industries are next important to oil and gas industries in Assam and it alone contributes more than 50% of all India tea production from about 1000 numbers of medium and large tea estates. The share of electrical energy consumed by entire tea processing operation is approximately 12% of peak demand energy 1300 MW (2012-2013) in the state of Assam. Tea drying is highly energy intensive process and present fuel used is either natural gas or coal as per availability and ease of operation. From study, it is also observed that many tea factories, particularly in the North bank of the river Brahmaputra are equipped with very inefficient coal fired furnace.

(b) Almost all large and medium scale tea-processing units have stand-by diesel electrical generator set (300 kVA to 500 kVA) to supplement with state electricity utility due to scarcity of energy at peak hours.

(c) Application of biomass based producer gas fired engine-generator coupled with exhaust waste-heat recovery system for tea-drying concept is proposed. The new model of energy system may be either by partial modification of existing diesel-generator set for dual fuel operation or 100% producer gas operation coupled with waste-heat recovery system.

(d) Modified and standardized producer gas engine-generator is available from manufacturer whereas exhaust waste heat recovery system with dryer is designed and developed. Its performance is tested for waste heat to useful thermal energy conversion system. It is observed from the outcome of the project that around (10-20) % of total tea drying thermal energy requirement may be trapped from waste heat recovery system in engine-exhaust. The system is environment friendly due to the application of green fuel for both heat and power generation.

8. Deviation made from original objective if any, while implementing the project and reason there of:

Not deviated from original objectives

9. Experimental work giving full details of experimental set up, methods adopted, data collected supported by necessary tables, charts, diagrams and photographs

Experimental Setup: The present biomass gasifier model name is **GAS-11** is made by "Ankur". The "Ankur" biomass gasifier system includes reactor, hopper, venture scrubber unit, fine filter, safety filter, gas flare and power generation unit with other required auxiliary equipment. The reactor is having a charcoal bed for reduction of partially combusted (at 400^0C to 500^0C) product from the combustion zone. Biomass fuel of specific size is fed into the hopper and it can be stored inside the hopper during the operation of the system. The measured and controlled amount of atmospheric air enters through the air nozzles into the combustion zone where partial combustion of the biomass fuel takes place. The gaseous products from the partial combustion zone travel through the charcoal reduction bed where it is converted into useful gaseous fuel called producer gas. Perforated sheet helps in removing ash generated during gasification of biomass with the help of comb-rotor and also allows the gas to be taken out from the reactor through annulus. The ash discharge is collected in a box called dry ash collection box at the bottom while raw producer gas is scrubbed and cooled in scrubber with re-circulating cooling water in cooling pond with the help of AC scrubber water pump. Then gas is separated from water in drain box and allowed to pass through the fine filter and the safety filter. The cool and clean producer gas is pre-mixed with air using gas-air mixture valve before entering into the producer gas engine. Figure 1 shows the experimental setup of a 10 kW_e producer gas engine generator with biomass gasifier and gas cleaning system. Figure 2 shows the producer gas engine used for waste heat recovery.

Figure 1: Experimental setup for 10 kW$_e$ output producer gas engine generator

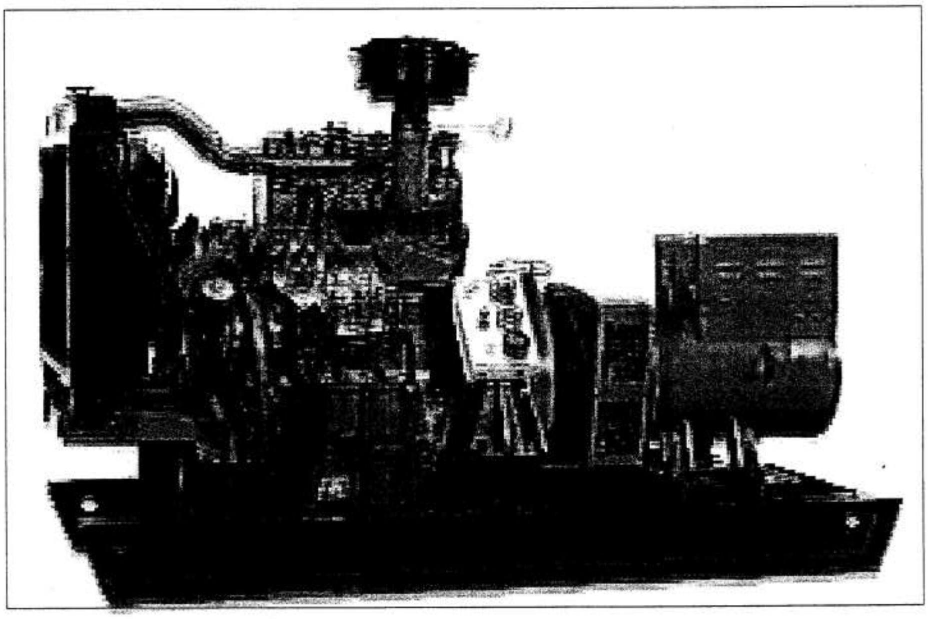


Figure 2: 10 kW$_e$ producer gas engine generator

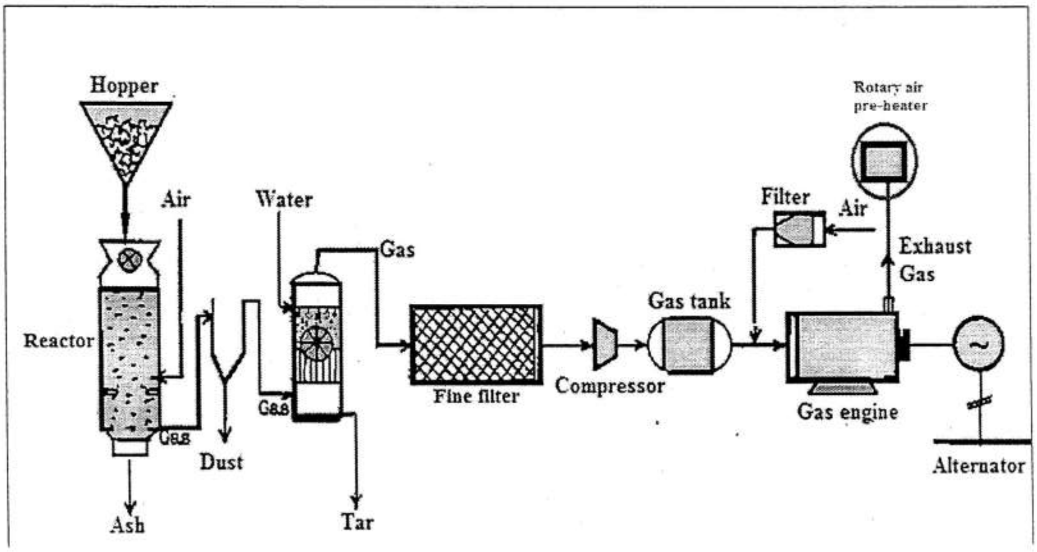


Figure 3: Schematic diagram of the rotary heat exchanger with the power generation system

Once the biomass gasifier coupled with gas engine generator is ready, a waste heat recovery compact heat exchanger is designed and developed to extract waste coming out from the exhaust of the producer gas engine. Based on the mass flow rate and average temperature of flue gas coming out, the final sizing of the compact plate heat exchanger is performed. The design methodology of compact plate heat exchanger is presented as follows.

DESIGN METHODOLOGY

The present work deals with mathematical modelling and the computer aided design (CAD) of different parts of the rotary heat exchanger. For developing a waste heat recovery device, the first step is to select an appropriate heat exchanger. A regenerative rotary heat exchanger is selected as a heat recovery device. In the second step, the mathematical modelling and appropriate sizing of the rotary heat exchanger is being discussed. Then the CAD model is developed by using the Autodesk Inventor 2014 version. Flow chart of the development process of the rotary heat exchanger is given below in Figure 4.

Technical information of the rotary heat exchanger

Different technical specifications of the rotary heat exchanger are given below in Table 1 below. *Material of the sheet used in the heat exchanger matrix:* Aluminium alloy AA3105.

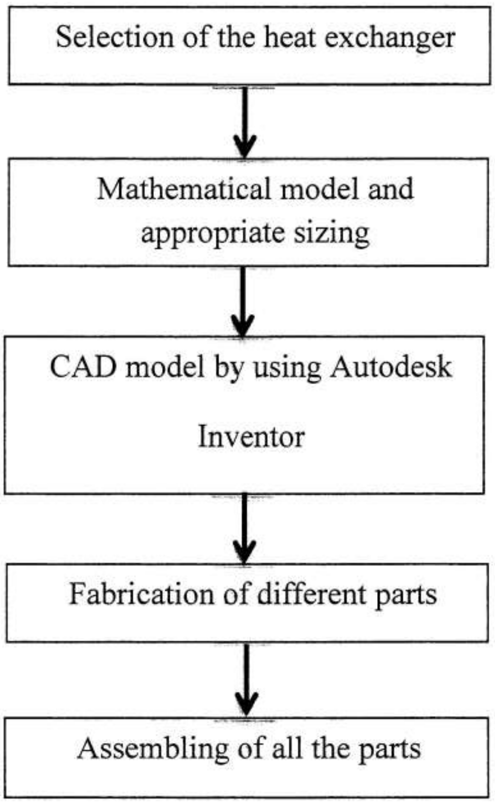


Figure 4: Flow chart of development of the rotary heat exchanger

Table 1: Specification of the Rotary Heat Exchanger

Rotor size	***Diameter***	***Height***	***Width***	***Depth***	***Rib***
900 mm	880 mm	1000 mm	1000 mm	300 mm	60 mm

The different mechanical properties of the heat exchanging matrix material are given below in Table 2 and Table 3.

Table 2: Mechanical properties of the matrix material (Courtesy: HINDALCO Ever-last Aluminium Roofing and Structural)

Alloy	***Temper***	***UTS(MPa) Min***	***Yield stress (0.2% proof stre***
AA3105	H18	195	160

Table 3: Chemical properties (Courtesy: HINDALCO Ever-last Aluminium Roofing & Structural)

Alloy	*Al*	*Cu*	*Mg*	*Si*	*Fe*	*Mn*	*Zn*	*Ti*	*Cr*
AA3105	Remainder	0.30	0.2-0.8	0.60	0.70	0.2-0.8	0.4	0.1	0.2

Specific heat capacity: 0.897 J/g°C (Courtesy: MatWeb)
Thermal conductivity: 171 W/mK (Courtesy: MatWeb)
Thickness of the sheet: 0.5 mm.
Materials used for making the housing and the frame: Mild steel and Aluminium alloy.

Drive of the Rotary Heat Exchanger

The wheel is driven by an electric motor and belt. Belt drive and worm gear drive is used for speed reduction of the rotary wheel. The Rotary Heat Exchanger will be used and examined to recover waste heat from the exhaust of a purely producer gas based 11 kW gas engine.

Rotational speed of the Heat Exchanger: About 1-15 rev/min.
Main components of the device:

- Rotary wheel
- Motor
- Worm gear
- Belt drive

Brief description of the gasifier power generation system

A schematic diagram of the biomass gasifier power generation system with the rotary air pre-heater is shown in the Figure 3. The present biomass gasifier model name is **GAS-11** is made by "Ankur". The biomass gasifier system includes reactor, hopper venture scrubber unit, fine filter, safety filter, gas flare and power generation unit with other required auxiliary equipment. The reactor is having a charcoal bed for reduction of partially combusted (at 400°C to 500 °C) product from the combustion zone.

Biomass fuel of specific size which is fed into the hopper can be stored during the operation of the system. The measured and controlled amount of atmospheric air enters through the air nozzles into the combustion zone where partial combustion of the biomass fuel takes place. The gaseous products from the partial combustion travel through the charcoal reduction bed and get converted into useful gaseous fuel called producer gas. Perforated sheet helps in removing ash, generated during gasification of biomass with the help of comb-rotor and also allows the gas to be taken out from the reactor through annulus. The ash discharge is collected at the bottom in a box, called dry ash collection box, while raw producer gas is scrubbed and cooled in scrubber with re-circulating cooling water in cooling pond with the help of scrubber water pump. Then gas is separated from water in drain box and allowed to pass through the fine filter and the safety filter. The cool and clean producer gas is pre-mixed with air using gas-air mixture valve before feeding into the engine. The gasifier is started with the help of a 12 V battery and an inverter which initially provide auxiliary power to the scrubber pump to start the gasifier system. Then the producer gas air mixture enters into the power generator and it starts. The rotary air pre-heater then rotates very slowly within the exhaust flue gas and the supply fresh air [1].

Figure 3 shows the schematics of 11 kW power generation systems, coupled with a rotary heat exchanger. The temperature of the exhaust gas is about 300^0 C. As the high temperature exhaust gas will pass through the rotary heat exchanger, the heat exchanging matrix will extract the heat energy from the exhaust gas in one half cycles and will release it to the fresh air in the other half cycle of its rotation. Corrugated aluminium alloy AA3105 of thickness 0.5 mm is used to make the heat transfer matrix of the heat exchanger.

The aluminium alloy AA3105 has a specific heat capacity of 0.897 J/g-^{0}C and thermal conductivity of 171 W/m-K. The different technical specifications and operating parameters are given below in Table 4.

Design of a speed reduction mechanism

One of the main significances of the rotary heat exchanger is that its effectiveness varies with its rotational speed. The desired rotational speed of the rotary heat exchanger is very slow, which is about 1-15 rpm. But the actual speed of the motor is 1426 rpm. So a mechanism has to be developed to reduce the speed of the motor to 1-15 rpm.

The speed reduction mechanism can be divided into four stages. The flowchart of the speed reduction mechanism is given in Figure 5.

Table 4: Technical Specification and important operating parameters of the gasifier (Courtesy "Ankur")

Model	*GAS-11 in Scrubbed, Clean Gas Mode*
Output power	11 kW in 100 % producer gas mode
Gas flow rate	35 m^3/hr
Type	Downdraft
Average Gas Calorific Value	1000 kcal/m^3
Gasification temperature	1050-1100^{0}C
Fuel Storage Capacity	125 kg.
Ash Removal	Dry ash discharge by manual
Start up	Through scrubber pump
Fuel type & size	Wood block with maximum dimension not exceeding 25mm x 25mm
Permissible moisture content in biomass	Less than 20%(wet basis)
Biomass charging	Online batch mode
Rated hourly consumption	14-15 kg
Rated hourly ash discharge	5 to 6 %
Typical conversion efficiency	>75%
Typical gas composition	CO – 19 to 22%, H_2 – 18 to 20% CO_2 – 10 to 13%, CH_4 –up to 3% N_2 – 50%

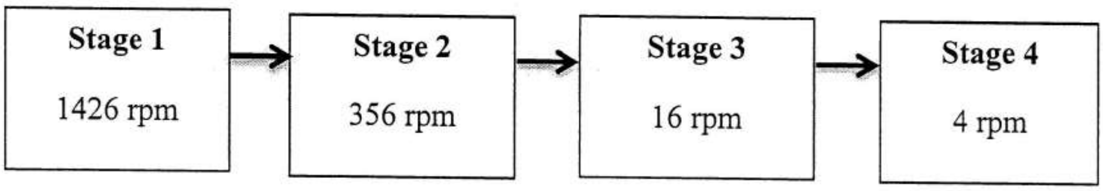


Figure 5: Flow chart of the speed reduction mechanism

Design of a worm gear

As the original speed of the single-phase AC motor which is used as a drive is 1426 rpm, it is very much necessary to design such a mechanism that will reduce its speed to a great extent up to 1-15 rpm. A worm gear is designed for reducing the rotational speed of the rotary heat exchanger. The worm gear with the belt and rope drive will bring down the rotational speed of the rotary heat exchanger to the required rotational speed which is about 4 rpm.

Different steps involved in designing the worm gear and the worm shaft is given below.

- ***Step 1:***

  No. of gear teeth, $N_g = 22$

- ***Step 2:***

  Module of the cutter, $m = 2$

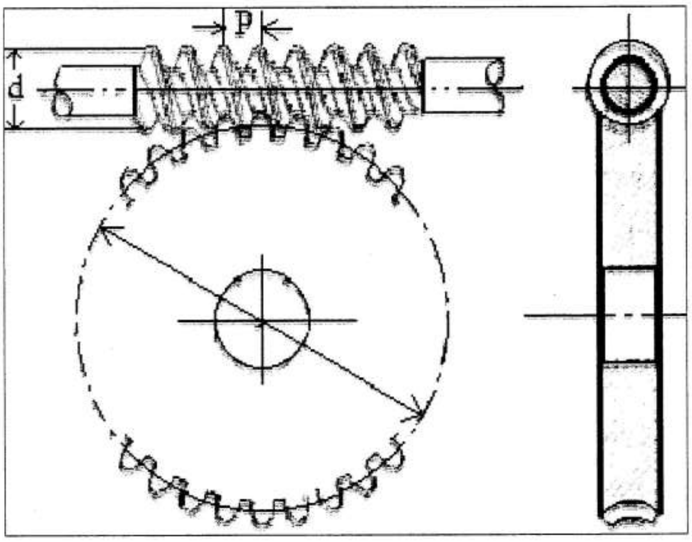


Figure 5a: Worm gear and the worm shaft

- ***Step 3:***

  Blank diameter, $BD = m\ (N_g + 2)$ (1)

- ***Step 4:***

  $PCD = m \times N_g = 2 \times 22$ (2)

  $= 44\ mm = D$ (Assumed)

- ***Step 5:***

  $CP = m\pi = 2 \times \frac{22}{7} = 6.28\ mm$ (3)

- ***Step 6:***

  We have

  - $PCD = \frac{N_g P_x}{\pi}$ (4)
  - $mN_g = \frac{N_g P_x}{\pi}$

- $P_x = m\pi = 2 \times 3.14 = 6.28$ mm ≈ 6 mm
- i.e. $P_x =$ Pitch of the worm threads

  = Pitch of the worm wheel

- ***Step 7:***

  Pitch diameter of the worm $d_1 = d_0 - 2m$ (5)

  Assume, $d_0 = 35\ mm\ (Outside\ diameter\ of\ the\ worm)$

  $$d_1 = 35 - 2 \times 2 = 31mm$$

- ***Step 8:***

  No. of threads on worm, $N_w = 1$

  Lead of the worm, $L = N_w \times P_x$ (6)

  $L = P_x = 6\ mm$

- ***Step 9:***

  $$\varphi = tan^{-1}\frac{L}{\pi d_1} = tan^{-1}\frac{m}{d_1} = tan^{-1}\frac{2}{31} = 3.69° = 3.7° \quad (7)$$

- The desired rotational speed of the rotary heat exchanger is about 1-15 rpm
- In stage 1 the actual speed of the motor is = 1426 rpm
- The belt drive from the motor to the worm shaft will reduce the speed to = 356 rpm
- The worm gear has 22 teeth so it will reduce the speed by 22 times
- Therefore, the speed becomes = 356/22 = 16 rpm
- The rope drive between the worm gear shaft and the thermal wheel will reduce the speed up to 4 rpm.

Calculation for length of the rope of the rope drive

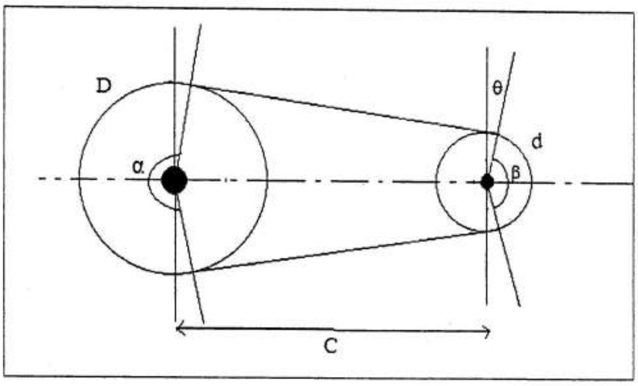


Figure 5b: The rope drive

Centre distance between big and small pulley, C = 1200 mm

Pitch diameter of the small pulley, d = 150 mm

Pitch diameter of the large pulley (wheel), D = 900 mm

$$\alpha = 180 + 2\theta \quad (8)$$

$$\beta = 180 - 2\theta \quad (9)$$

Where, θ is given by,

$$\theta = \sin^{-1}\frac{(D-d)}{2C} \quad (10)$$

Length of the belt [2],

$$L = \frac{\pi}{2}(D+d) + 2C + \frac{1}{4C}(D-d)^2 \quad (11)$$

The calculated rope length, L = 4166.5 mm

= 4.2 m

Mechanical design of shaft of the thermal wheel

Different criteria and design considerations that should be taken care of when designing a shaft for a rotary heat exchanger are discussed below.

Material selection

Different ferrous, non-ferrous materials and non-metals are used as shaft material depending on the application. The shaft used in the rotary heat exchanger is made up of mild steel rod of 25 mm diameter.

Design consideration of shaft

Design based on Strength

In this design methodology, design is carried out so that stress at any location of the shaft does not exceed the material yield stress [3]. The stepwise design of the shaft is given below.

Design of shaft subjected to only bending moment

Bending stress of the shaft, $$\sigma_b = \frac{M*r}{I} \quad (12)$$

Moment of inertia, $$I = \frac{\pi*d^4}{64} \quad (13)$$

Radius of the shaft, $$r = \frac{d}{2} \quad (14)$$

$$\sigma_b = \frac{32*M}{\pi d^3} \quad (15)$$

Where, M is the bending moment of the shaft.

Design of shaft subjected to only twisting moment

Torsion shear stress of the shaft, $\tau = \frac{Tr}{J}$ (16)

Polar moment of inertia, $J = \frac{\pi * d^4}{32}$ (17)

Torsion shear stress of the shaft, $\tau = \frac{16 * T}{\pi d^3}$ (18)

Design based on Stiffness

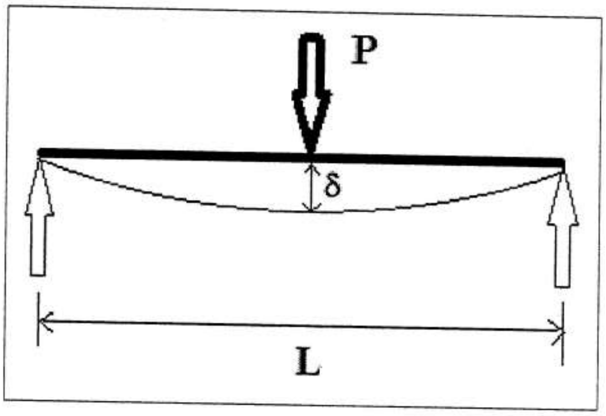


Figure 5c: Deflection of shaft

To design a shaft based on stiffness i.e. torsional rigidity, the limit of angle of twist should be known. The angle of twist is given as follows

$$\frac{T}{J} = \frac{G\theta}{L} \quad (19)$$

The angle of twist is given by,

$$\theta = \frac{TL}{GJ} \quad (20)$$

$$\theta = \frac{584TL}{Gd^4} \quad (21)$$

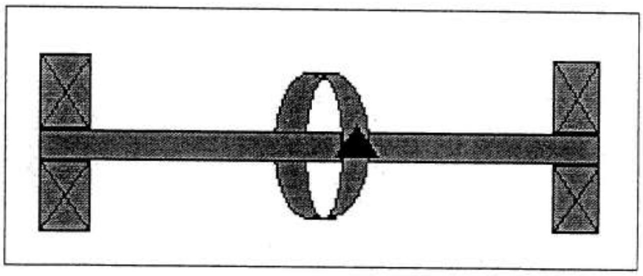


Figure 5d: Simply supported shaft

Critical speed of a rotating shaft is defined as the speed where the shaft becomes dynamically unstable. It can be shown that the frequency of free vibration of a non-rotating shaft is same as its critical speed.

The equation of lowest critical speed of a shaft on two supports is given by,

$$f_{critical} = \frac{30}{\pi} * \sqrt{\frac{g}{\delta}} \tag{22}$$

In our case the shaft is simply supported i.e. the ends rotate normal to the axis as shown in Figure 5d.

The final formulation for the shaft speed is given by,

$$f = \frac{\pi * n^2}{2} \sqrt{\frac{gEI}{wL^4}} \tag{23}$$

Where n is the number of nodes and w is the load on the shaft.

For the shaft used in the heat exchanger,

Diameter of the shaft, d = 0.025 m,

Load on the shaft, F=30×9.81 N = 294.3 N

Modulus of rigidity, G= 64000× 10^6 N/m^2

Moment of force, M = F×$\frac{l}{2}$ = 30×9.81×$\frac{0.5}{2}$ N-m = 73.58 N-m

Bending stress, $\sigma_b = \frac{32 \times M}{\pi d^3} = \frac{32 \times 30 \times 9.81 \times 0.5}{2 \times 3.14 \times 0.025^3}$ = 4.798× 10^7 N/m^2 = 47.98 N/mm^2

Torsional moment (torque), T = F×r = 30×9.81×$\frac{0.025}{2}$ N-m = 3.67 N-m

Shear stress, $\tau = \frac{16 \times T}{\pi d^3} = \frac{16 \times 30 \times 9.81 \times 0.025}{2 \times 3.14 \times 0.025^3}$ = 1.2×10^6 N/m^2 = 1.2 N/mm^2

Angle of twist, $\theta = \frac{584TL}{Gd^4} = \frac{584 \times 30 \times 9.81 \times 0.5}{64000 \times 10^6 \times 0.025^4}$ = 3.43^0

Calculation of the heat transfer area of the heat exchanger matrix

For mathematical design of a heat exchanger the calculation of the heat transfer surface area is very important. The calculation procedure of the surface area of the rotary heat exchanger is given below. For this calculation we have assumed the Reynolds number (*Re*) to be around 6000. According to the flue gas composition and range of *Re* we have calculated the *Cp* for flue gas and some of the parameters are assumed and some of them are found from manuals [4-5].

The considered Reynolds number,	*Re* = 6000
Prandtl number for the flue gas,	*Pr* = 0.74
Fouling factor,	R_f = 0.0004 m^2/KW
Cp for the flue gas,	*Cph*=1097 J/kgK

Cp of air,	Cpc=1005 J/kgK
Mass flow rate of flue gas,	$\dot{m}_f = 66.75\frac{kg}{s}$
Mass flow rate of air,	$\dot{m}_a = 2.3\frac{kg}{s}$
For gas to gas exchanger the	
Overall heat transfer coefficient,	U =60 W/m^{2}K

Calculation of exhaust gas outlet temperature,

$$\dot{m}_h \times C_{ph}(300 - T) = \dot{m}_c C_{pc} \times 77 \quad (24)$$

$$T = 250\,^0C$$

$$LMTD = \frac{\Delta T_i - \Delta T_e}{ln(\frac{\Delta T_i}{\Delta T_e})} \quad (25)$$

$$LMTD = 211.2$$

$$Q = UA_s LMTD \quad (26)$$

$$Q = \dot{m}_c C_{pc} * 77$$

$$Q = 177985.5 \text{ J}$$

$$\mathbf{A_s = 33.709\ m^2 = 34\ m^2} \quad (27)$$

The heat transfer surface area required for the efficient heat exchanging between the hot and the cold fluid is almost 34 m^2.

Effectiveness calculation procedure

The effectiveness of a rotary regenerator is given by:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (28)$$

Where, $\dot{Q}$ is the actual amount of heat transfer rate from the hot fluid to the cold fluid within the rotary regenerator and $\dot{Q}_{max}$ is the maximum possible heat transfer rate that can be achieved in an ideal rotary regenerator with an infinite heat transfer surface area.

Calculation of $\dot{Q}$ from hot fluid to the cold fluid requires the solution of governing differential equations for transport phenomena within the rotary regenerator. These partial differential equations are usually formulated with various simplifying assumptions. But still under classical assumptions, their solutions are very challenging to solve.

The effectiveness of a rotary heat exchanger can be expressed in terms of the effectiveness of a stationary counter flow heat exchanger (ε_0) as follows:

$$\varepsilon = \varepsilon_0 \varphi_r \varphi_c \tag{29}$$

Where, φ_r and φ_c represent the factors that take into account the influence of rotational speed and cleaning region respectively [6].

Calculation of the effectiveness of a stationary counter-flow heat exchanger (ε_0)

The effectiveness in NTU method can be expressed as:

$$\varepsilon_0 = \frac{1-e^{[-NTU(1-C^*)]}}{1-C^* e^{[-NTU(1-C^*)]}} \tag{30}$$

Where C^* is the heat capacity-ratio given by:

$$C^* = \frac{C_{min}}{C_{max}} \tag{31}$$

C_{min} and C_{max} are the smaller and the larger of the cold (C_c) and hot fluid (C_c) heat capacity rates respectively.

$$C_c = \dot{m}_c C_{pc} \tag{32}$$

$$C_h = \dot{m}_h C_{ph} \tag{33}$$

The number of transfer units (NTU) is a non-dimensional expression of the heat transfer size of the heat exchanger which can be expressed as:

$$NTU = \frac{UA}{C_{min}} \tag{34}$$

UA is the heat capacity of the exchanger. It can be calculated as:

$$\frac{1}{UA} = \frac{1}{h_h A_h} + \frac{1}{h_c A_c} \tag{35}$$

Where A is the heat transfer surface area and h is the convective heat transfer coefficient [6].

Analytical approach for fluid flow analysis

Since, the profile of proposed rotary heat exchanger is corrugated (Alûminum plate), therefore equivalent flow area and wetted perimeter of each channel is computed from following relationships.

$$A_0 = H \times W \tag{36}$$

$$P = 2(H + W) \tag{37}$$

Where, A is area (m^2) of duct, H is depth (m) of corrugation, W is channel width (m) and P is wetted perimeter. The hydraulic diameter (D_h) of the channel is computed from following expression.

$$D_h = \frac{2H \times W}{(H+W)} \quad (38)$$

Now thermal diffusivity (α) and specific flow rate (G) of the fluid flowing through the corrugated duct is given by following expressions.

$$\alpha = \frac{k}{\rho C_p} \quad (39)$$

$$G = \frac{\dot{m}}{A_0} \quad (40)$$

The dimensionless Reynolds number and Peclet number may be computed as:

$$Re = \frac{GD_h}{\mu} \quad (41)$$

$$Pe = \frac{uD_h}{\alpha} \quad (42)$$

The pressure difference required to push a stream of fluid through heat exchanger channel is a function of flow parameters and channel geometry. If density of the flowing fluid does not vary then pressure drop is given as:

$$\Delta p_s = \frac{f4LG\rho V^2}{2D_h} \quad (43)$$

The friction factor and pumping power are estimated for corrugated plate heat exchanger from following expression [7].

$$f = (2.9 + 5.6\emptyset + 0.12\emptyset^2)Pe^{-0.13} \quad (44)$$

$$P_c = \frac{m\Delta p}{\rho} \quad (45)$$

CAD model of the rotary heat exchanger

Before fabricating the rotary heat exchanger (RHE) the design is done both numerically and with the help of computer software. The CAD design of the rotary heat exchanger is done by using Autodesk Inventor 2014 version (Autodesk Educational product). Different parts are drawn separately using the software and then assembled together.

The two-dimensional view of the CAD model of the rotary heat exchanger is first drawn with different dimensions in 'mm'. The different views of the drawing are shown below in Figure 6a and Figure 6b.

Front View

The front view of the 2-D CAD drawing is shown below in Figure 6a with its different dimensions. All the dimensions are in "mm". The width, breadth and height of the rotary wheel casing are 300 mm, 700 mm and 700 mm respectively. The radius of the rotary thermal wheel is 300 mm. The height of the base frame with the drive is 572 mm and the breadth is 712 mm.

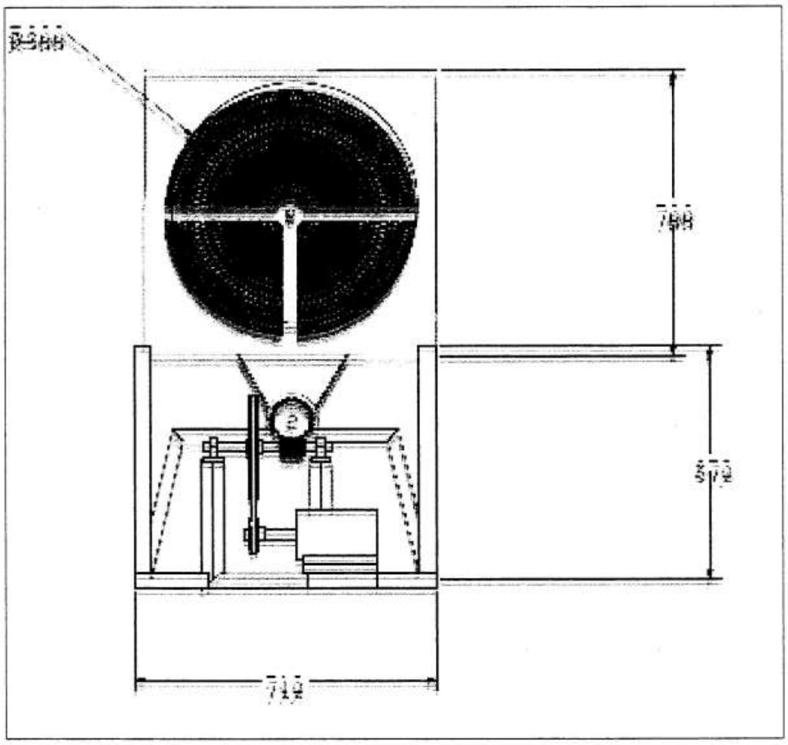


Figure 6a: Front view of the CAD model with dimensions in 'mm'

Side view

The 2-D side view of the Rotary Heat Exchanger is shown below in the Figure 6b. All the dimensions of the figure are in "mm". From this view we can see the length of the RHE base frame and the width of the rotary wheel which are 1062 mm and 300 mm respectively.

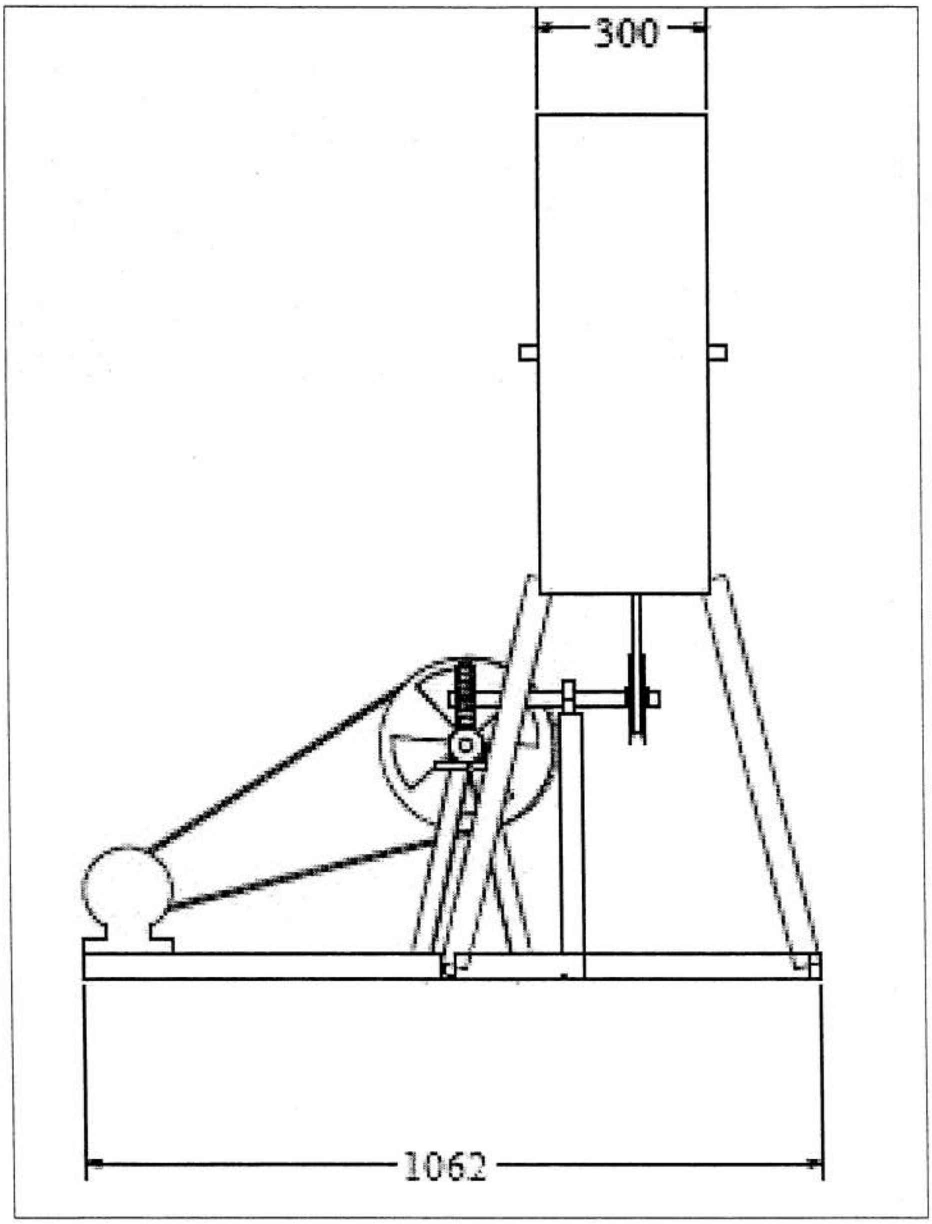


Figure 6b: Side view of the CAD model with Dimensions in 'mm'

Different 3-D wire frame views of the heat exchanger are shown below in Figure 6c and Figure 6d.

Wire frame isometric view

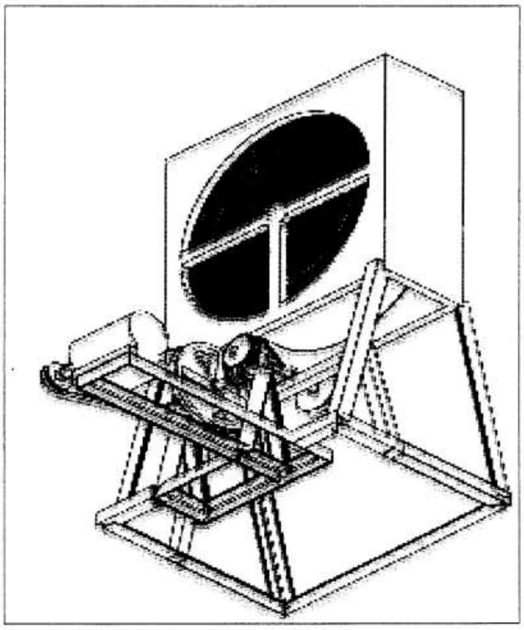


Figure 6c: Wire frame isometric view

Wire frame bottom view

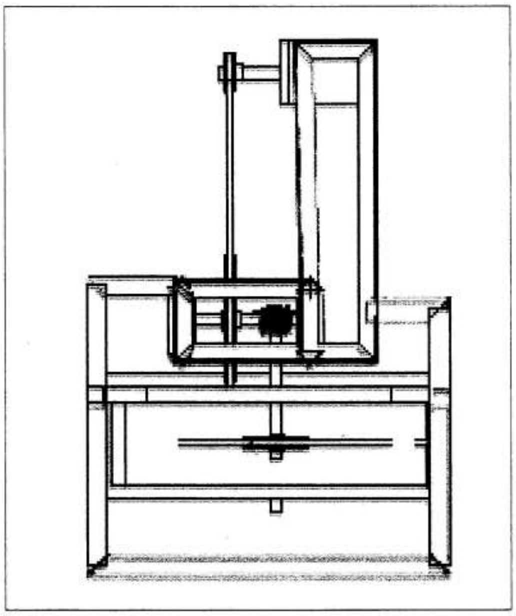


Figure 6d: Wire frame bottom view of the CAD model

The front view and side view of Rotary Heat Exchanger are shown below in Figure 6e and Figure 6f.

Front View

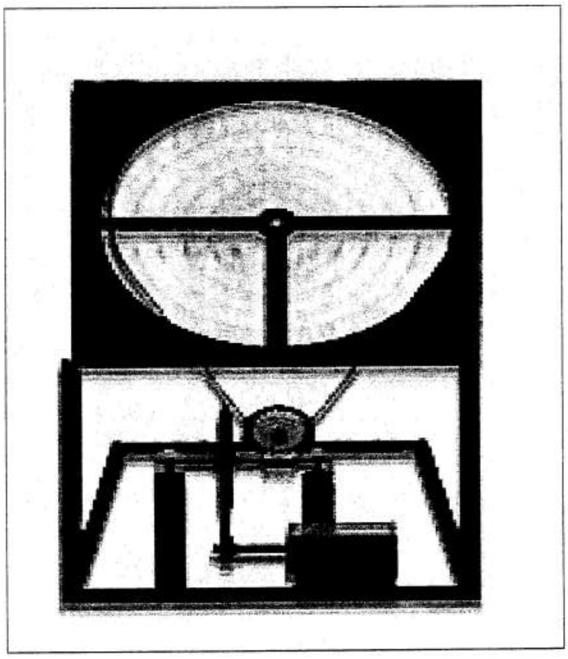


Figure 6e: The front rotary heat exchanger

Side View

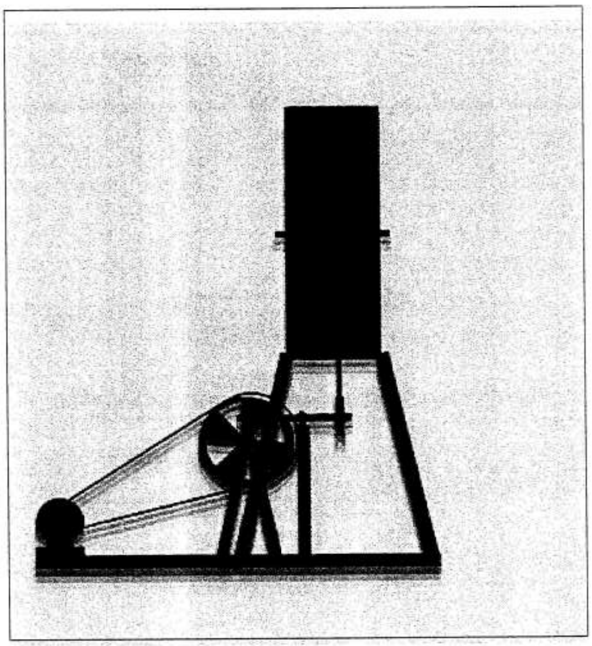


Figure 6f: The side view of the rotary heat exchanger

Isometric View

The isometric view of the rotary heat exchanger is shown in Figure 7 with its different parts.

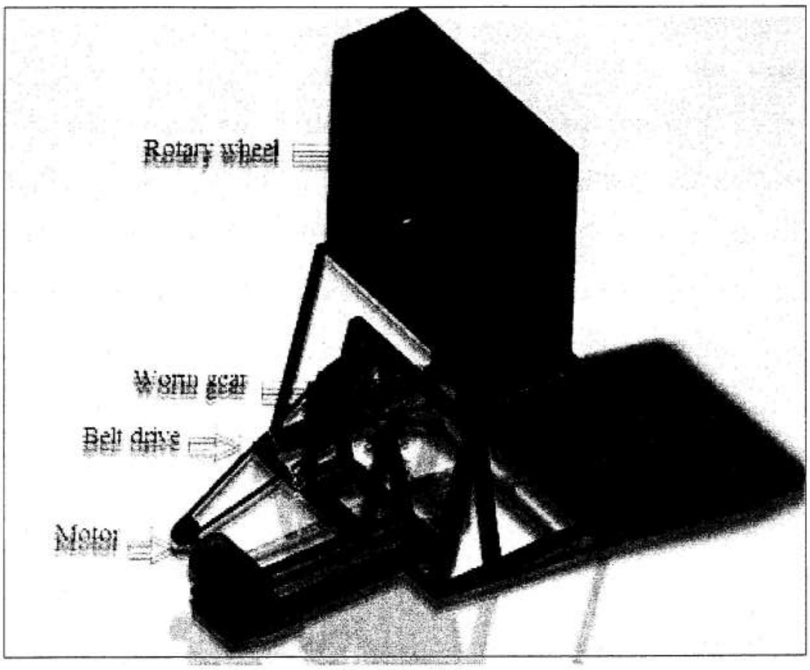


Figure 7: The isometric view of the rotary heat exchanger

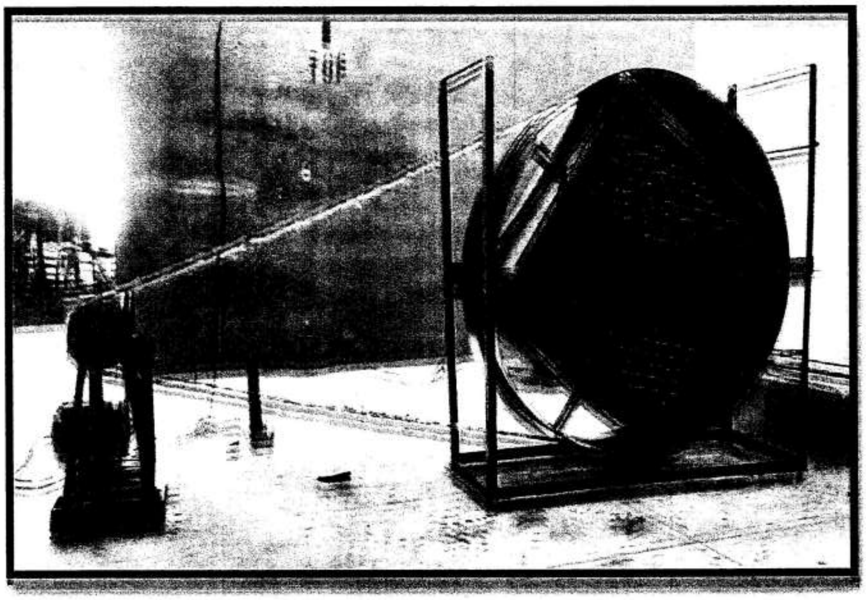


Figure 8: Complete assembly of rotary compact heat exchanger

10. Detailed analysis of results indicating contributions made towards increasing the state of knowledge in the subject

Results and Discussions:

Pressure Drop

Pressure drop of flue gas over length of a compact heat exchanger duct dictates the pumping power requirement in the blower. Figure 9 shows pressure drop of heat exchanger with increase in flue-gas inlet temperature. As the inlet temperature of gas side increases there is a minor increase in pressure drop. The pressure drop in air side almost remains constant. Due to change of physical properties of flue gas with temperature, there is a minor change in pressure drop. Similarly, with increase in flue gas mass flow rate, its pressure drop increases (Figure 10).

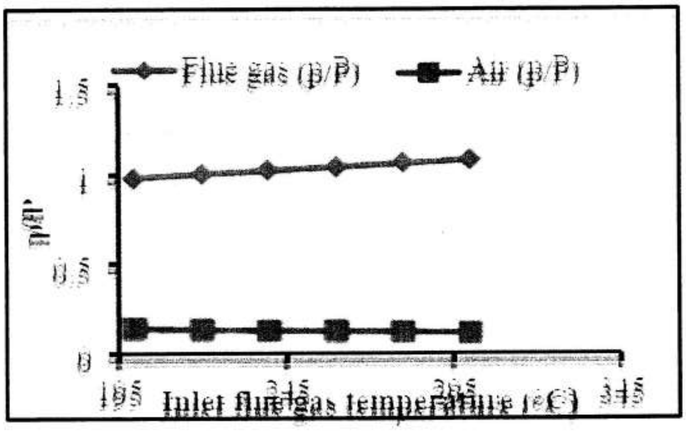
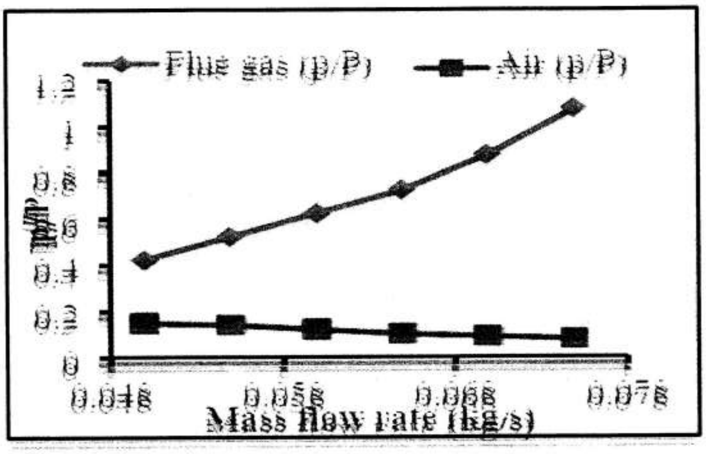


Figure 9 : Variation of pressure drop with flue gas temperature



Figue 10: Variation of pressure drop with mass flow rate

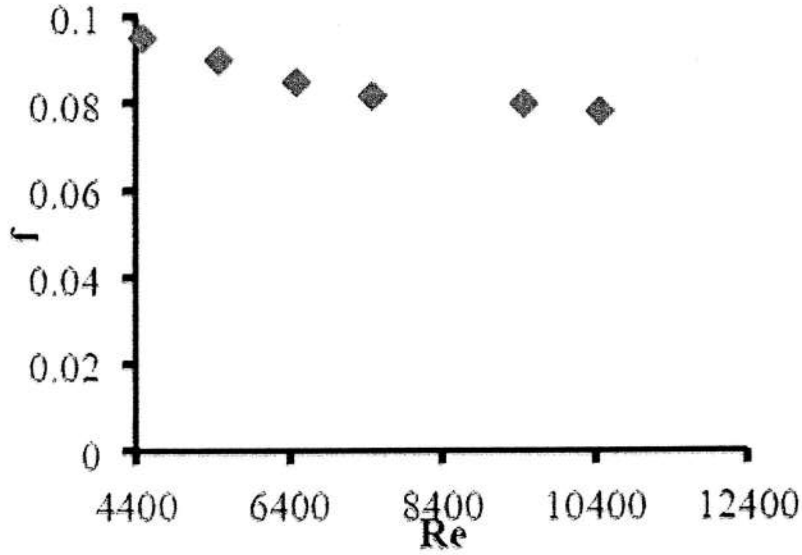


Figure 11: Variation of friction factor with Reylonds number

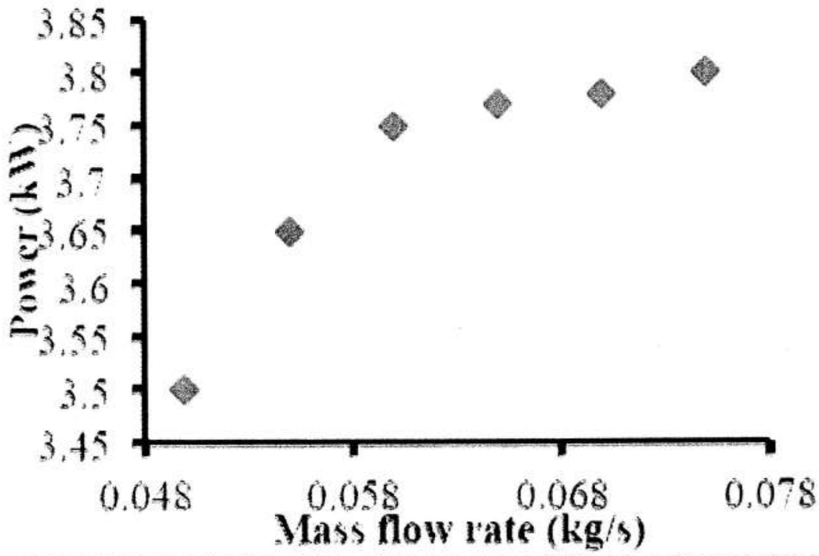


Figure 12: Variation of heat transfer co-efficient with mass flow rate

Friction Factor and Reynolds Number

In airside, this increase is sluggish. The average velocity of flue gas is much higher compared to air. Moreover, the mean temperature change of air was small compared to flue gas. Variation of friction factor with Reynolds number in flue gas side has been presented in Figure 11. It is clear from the graph that there is a noticeable change in the slope of the curve around Reynolds number 6000-7000. This means transition from laminar to turbulent flow takes place in this region. Due to variation of composition (density) of flue gas in combustion chamber, there may be scattered variation of friction factor near turbulent region.

Variation of Heat Transfer Coefficient with Mass Flow Rate

Figure 12 shows variation of heat transfer rate from flue gas with its mass flow rate. For constant air mass flow rate and inlet flue gas temperature the heat transfer rate from flue gas increases with increase in its mass flow rate. This is because with increased flue gas mass flow rate, there is a corresponding increase in its velocity that prompts enhanced heat transfer.

11. Conclusions summarizing the achievements and indication of future work

A rotary heat exchanger has been designed and developed for coupling with a waste heat recovery system from a purely producer gas fired engine exhaust. The flow performances of the flue gas and air have been studied theoretically as well as experimentally for the compact flue gas heat exchanger by using standard mathematical relationship with different known data of the system. As the inlet temperature of entry side of compact rotary air preheater increased, there is a minor increase in pressure drop and almost constant pressure is available for airside. With an increase in flue gas mass flow rate, a prominent increase in pressure drop is observed. Flue gas flow changed from laminar to turbulent around Reynolds number 6000-7000 where there is a prominent change in the slope of the curve of friction factor versus Reynolds number. With increase in mass flow rate of flue gas there is augmentation of heat transfer rate from flue gas to air. These fluid flow data are expected be useful for design improvement of compact rotary air preheater.

12. Science and technology benefits accrued

i. List of research publications:

Sl. No.	Authors	Title of paper	Name of the journal	Volume	Page	Year
01.	Dutta,P.P., Baruah D.C..	Gasification of up rooted Tea Shrubs for Process Heat generation in Tea Manufacturing.	*Biomass and Bioenergy (Elsevier)*	66	27-38	2014 (Impact factor: 4.146)
02.	Dutta, P.P., Baruah D.C.	Possibility of biomass gasification in tea	International Journal of Renewable Energy	5(4)	310-322	2014

		manufacturing industries in Assam, India	Technology (Inderscience)			
03.	Dutta, P.P., Pandey, V., Das, A.R., Sen, S., Baruah, D. C.	Downdraft gasification modelling and experimentation of some indigenous biomass for thermal applications	Energy Procedia *(Elsevier)*	54	21-34	2014

Publications in conference

Sl. No.	Authors	Title of paper	Conference Proceeding	Volume	Page	Year
01	Dutta, P.P., Das, A., Mishra, A., Kumar, B. and Singh, D.K.	Some studies on a plate type heat exchanger	National conference on Recent Advancements in Mechanical Engineering, NERIST, Itanagar, Nov.08-09, 2013	(ISBN: 978-93-82880-71-4)	170-176	2013
02	Dutta, P.P., Pandey, V., Das, A.R., Sen, S., Baruah, D	Downdraft gasification modelling of some indigenous biomass for thermal application	4th International Conference of Energy Research at IITB: 10-12 December 2013.	(ISBN: 978-81-928795-0-5)	166-175	2013
03	Dutta, P. P., Baruah, N.	Fluid flow study of a rotary compact air-preheater	International conference in frontier in mathematics, March:2015, Guwahati University	(ISBN: 978-81-928118-9-5)	142-145	2015
04	Dutta, P.P., Baruah, N.	Food drying opportunity from waste heat recovery	National seminar cum workshop IPFP:IBES:2015, Tezpur University	ISBN:978-93-84388-06-5	84-91	2015
05	Dutta, P. P., Dutta, P., Das, A., Kalita, P.	Heat transfer and pressure drop in V corrugated channels of different height applicable to waste heat recovery compact	5th annual international conference on sustainability, inspiration, innovation and inclusion, IIM Shillong: 17-			2016

		heat exchanger	19/03/2016			
06	Dutta P, Das, A, Dutta P.P.	Theromohydraulic study of a V corrugated heat exchanger	In the proceeding of 6th international congress on computational mechanics and simulation : 27 June to 1 July		142-145	2016

ii. Manpower trained on the project

(a) Research Scientists or Research Associates: Nil

(b) Number of PhDs produced : Nil

(c) Other technical personals trained : [Three M. Tech (Mechanical Engineering) students did thesis work and eight B. Tech (Mechanical Engineering) students completed final year project work.

iii. Patents taken if any? : Nil

13) Financial position:

No	Financial Position / Budget head	Fund sanctioned	Expenditure	% of total cost
I.	Salaries / Manpower costs	Nil	Nil	Nil
II.	Equipment	Rs.1200000/-	Rs.1190000/-	99.16%
III.	Supplies & Materials	Rs.50000/-	Rs.50000/-	100%
IV.	Contingencies	Rs.20000/-	Rs.20000/-	100%
V.	Travel	Rs.20000/-	Rs.18350/-	91.75%
VI.	Overhead Expenses	Rs.100000/-	Rs.99800/-	99.80%
VII.	Others, if any (Fabrication)	Rs.75000/-	Rs.75000/-	100%
VIII.	Total	Rs.1465000/-	Rs.1453150/-	99.19%

14. Procurement / Usage of equipment

a)

S. No	Name of equipment	Make / Model	Cost (FE/Rs)	Date of installation	Utilization rate (%)	Remarks regarding maintenance/ breakdown
01	Producer Gas Engine Generator	Ankur	1078776.00	30.11.2013	100%	University will be requested to fund in case breakdown
02	HP Desktop i7	HP	54256.00	18.03.2013	100%	

03	HP Colour Printer	HP	56500.00	08.11.2013	100%	

b) Plan of utilizing equipment in future: The equipment will be utilized for experimental studies of biomass gasification based waste heat recovery system for various B. Tech and M.Tech final year project works.

[signature] 05/01/2017
Name and Signature with Date
Dr. Partha Pratim Dutta
(Principal Investigator)

References:

[1] Ankur Scientific Energy Technologies Pvt. Ltd. http://www.ankurscientific.com/, accessed in 19-12-2014.

[2] Kulkarni, S.G. *Machine Design*, Tata McGraw-Hill Education, New Delhi, 2008.

[3] Design of shaft, http://www.nptel.ac.in/courses/112105125/pdf/mod8les1.pdf, *NPTEL*, version 2 ME, IIT Kharagpur, accessed in January 2015.

[4] Bejan, A., Tsatsarsonis, G. & Moran, M. *Thermal Design and Optimization,* John Wiley & Sons, Inc., USA, 1996.

[5] Cengel, Y.A. *Heat and Mass Transfer: A Practical Approach*, Tata Mcgraw-Hill, Third edition, 2007.

[6] Yilmaz, T., Buyukalaca, O. Design of regenerative heat exchangers, *Heat Transfer Engineering* **24**(4), 32-38, 2003.

[7] Pandey, S.V., Nema,V. Experimental analysis of heat transfer and friction factor of nanofluid as a coolant in a corrugated plate heat exchanger, *Experimental Thermal Fluid Science* **38**, 248–256, 2012.

UC pertains to appropriate box **UC for recurring**

Utilization certificate for the financial year ending 31st March (2015-2016)
To be given separately for each financial year ending 31st March

First release	Second release	Third release	Forth release	✓ Final

Is the UC is provisional : Yes / No (Final) ✓

1.	Title of the Project/ Scheme:		Development of an innovative model of Combined Heat and Power from Purely Producer Gas Based Engine Alternator System for Partial Conventional Energy Substitution of Tea Processing Industries in North-East India under FAST TRACK SCHEME FOR YOUNG SCIENTIST	
2.	Name of the Institution:		Tezpur University	
3.	Name of the Principal Investigator		Dr. Partha Pratim Dutta	
4.	Science & Engineering Research Board (SERB) Sanction order No & date sanctioning the project: (First financial sanction order)		No. SR/FTP/ETA-52/2011 and 31/08/2012	
5.	Head of account as given in the original sanction order			
	Equipment (Nonrecurring)		Rs.12,00000/-	
	Total Capital (A)		Rs.12,00000/-	
	Consumables	Recurring	Rs.15,0000/-	
	Contingency		Rs.60,000/-	
	Travel		Rs.60,000/-	
	Other (Fabrication)		Rs.75,000/-	
	Over Heads		Rs.3,00000/-	
	Total General (B)		Rs.6,45,000/-	
	Total (A+B)		Rs.18,45,000/-	
6.	Amount brought forward from the previous Financial year quoting SERB letter no and date in which the authority to carry forward the said amount was given		i.Amount	Rs. 7,112/-
			ii.Letter No.	
			iii. Date	
7.	Amount received during the financial year (Please give SERB Sanction order no and date)		vi. Amount	Rs.30,862/- (Interest earned from 2012-13 to 2016)
			v.Letter No.	
			vi.Date	
8.	Total amount that was available for expenditure (excluding commitments) during the financial year (Sr. No. 6+7)		Rs.37,974/-	
9.	Actual Expenditure (excluding commitments) Incurred during the financial year (upto 31st March 2016)		Rs.5,262/-	
10.	Balance amount available at the end of the financial year:		Rs. 32,712/-	
11.	Unspent balance refunded, if any (please give details of cheque no etc.)		Rs. 32,712/- (Cheque No:	
12.	Amount to be carried forward to the next financial year (if applicable)		Nil	

Finance Officer
Tezpur University

UTILISATION CERTIFICATE

(Financial Year 2015-2016)

Certified that out of Rupees Nil of grants-in-aid sanctioned during the year 2015-2016 in favour of Registrar, Tezpur University Rupees seven thousand one hundred twelve only Rs.7,112/- on account of unspent balance of the previous year and interest earned Rs.30,862/- only during 2012-13 to 2015-16, a sum of Rs. 5,262/- has been utilized for the purpose of implementation of the project (General head) for which it was sanctioned and that the balance of Rs. 32,712/- remaining unutilized at the end of the year. Rs.32,712/- (Rupees thirty two thousand seven hundred twelve only) has been refunded/returned to SERB (vide DD/Cheque No.: ________ dated: __________).

Certified that we have satisfied ourselves that the condition on which the grant in aid was sanctioned have been fulfilled / are being fulfilled and that we have exercised the following checks to see that the money was actually utilized for the purpose for which it was sanctioned.

Dutta	6/11/17	6-11-17
Signature of PI	Signature of Registrar/ Head of the Institute	Accounts Officer of the Institute
Date: 05/11/2017	Date: *Registrar Tezpur University*	Date: *Finance Officer Tezpur University*

Kind of checks exercised:

1.
2.

For SURAJIT CHAKRABORTY & CO.
CHARTERED ACCOUNTANTS
09-12-2017
CA. SURAJIT CHAKRABORTY
(Proprietor)
Membership No.- 305054

Science and Engineering Research Board
UC accepted has been accepted by:

Signature
Name of SERB Officer
Designation

UC pertains to appropriate box **UC for non-recurring**

Utilization certificate for the financial year ending 31st March (2015-2016)
To be given separately for each financial year ending 31st March

First release	Second release	Third release	Forth release	✓ Final

Is the UC is provisional : Yes / No (Final) ✓

1.	Title of the Project/ Scheme:		Development of an innovative model of Combined Heat and Power from Purely Producer Gas Based Engine Alternator System for Partial Conventional Energy Substitution of Tea Processing Industries in North-East India under FAST TRACK SCHEME FOR YOUNG SCIENTIST	
2.	Name of the Institution:		Tezpur University	
3.	Name of the Principal Investigator		Dr. Partha Pratim Dutta	
4.	Science & Engineering Research Board (SERB) Sanction order No & date sanctioning the project: (First financial sanction order)		No. SR/FTP/ETA-52/2011 and 31/08/2012	
5.	Head of account as given in the original sanction order			
	Equipment (Nonrecurring)		Rs.12,00000/-	
	Total Capital (A)		Rs.12,00000/-	
	Consumables	Recurring	Rs.15,0000/-	
	Contingency		Rs.60,000/-	
	Travel		Rs.60,000/-	
	Other (Fabrication)		Rs.75,000/-	
	Over Heads		Rs.3,00000/-	
	Total General (B)		Rs.6,45,000/-	
	Total (A+B)		Rs.18,45,000/-	
6.	Amount brought forward from the previous Financial year quoting SERB letter no and date in which the authority to carry forward the said amount was given		i.Amount	Rs.10,000/-
			ii.Letter No.	
			iii. Date	
7.	Amount received during the financial year (Please give SERB Sanction order no and date)		vii. Amount	Nil
			v.Letter No.	
			vi.Date	
8.	Total amount that was available for expenditure (excluding commitments) during the financial year (Sr. No. 6+7)		Rs.10,000/-	
9.	Actual Expenditure (excluding commitments) Incurred during the financial year (upto 31st March 2016)		Nil	
10.	Balance amount available at the end of the financial year:		Rs.10,000/-	
11.	Unspent balance refunded, if any (please give details of cheque no etc.)		Rs.10,000/- (Cheque No:	
12.	Amount to be carried forward to the next financial year (if applicable)		Nil	

Finance Officer
Tezpur Un[illegible]

UTILISATION CERTIFICATE

(Financial Year 2015-2016)

Certified that out of Rupees Nil of grants-in-aid sanctioned during the year 2015-2016 in favour of Registrar, Tezpur University, Rupees ten thousand only Rs.10, 000/- on account of unspent balance of the previous year, a sum of Rs. Nil has been utilized for the purpose of implementation of the project (Capital head) for which it was sanctioned and that the balance of Rs. 10,000/- remaining unutilized at the end of the year. Rs.10,000/- (Rupees ten thousand only) has been refunded/returned to SERB (vide DD/Cheque No.: ________ dated: ________)

Certified that we have satisfied ourselves that the condition on which the grant in aid was sanctioned have been fulfilled / are being fulfilled and that we have exercised the following checks to see that the money was actually utilized for the purpose for which it was sanctioned.

Signature of PI
Date: 05/11/2017

Signature of Registrar/
Head of the Institute
Date: 6/11/17
Registrar
Tezpur University

Accounts Officer of the Institute
Date: 6.11.17
Finance Officer
Tezpur University

Kind of checks exercised:

1.
2.

For SURAJIT CHAKRABORTY & CO.
CHARTERED ACCOUNTANTS
09.12.2018
CA. SURAJIT CHAKRABORTY
(Proprietor)
Membership No.- 305054

Science and Engineering Research Board

UC accepted has been accepted by:

Signature
Name of SERB Officer
Designation

STATEMENT OF EXPENDITURE

(2012-2013, 2013-2014, 2014-2015, 2015-2016)

Sr. No. (I)	Sanctioned Heads (II)	Total Funds Allocated (indicate sanctioned or revised (III)	Actual fund received (IV)	Expenditure Incurred				Total expenditure till. 31/03/2016 (IX = V + VI + VII+VIII)	Balance as on (date) (X =IV – IX)	Remarks
				1st Year 20/09/2012 to 31/03/2013 (V)	2nd Year 01/04/2013 to 31/03/2014 (VI)	3rd Year 01/04/2014 to 31/03/2015 (VII)	4th Year 01/04/2015 to 31/03/2016 (VIII)			
1.	Capital (Equipment)	12,00,000/-	12,00000/-	Nil	11,90,000/-	Nil	Nil	11,90,000/-	10,000/-	–
Non-recurring items (Capital)		**12,00,000/-**	**12,00,000/-**	Nil	**11,90,000/-**	Nil	Nil	**11,90,000/-**	**10,000/-**	
2.	Consumables	1,50,000/-	50,000/-	Nil	25000/-	25,000/-	Nil	50,000/-	Nil	–
3.	Travel	60,000/-	20,000/-	3,085/-	4,653/-	5,350/-	5262/-	18,350/-	1650/-	
4.	Contingencies	60,000/-	20,000/-	13,265/-	6,735/-	Nil	Nil	20,000/-	Nil	–
5.	Others (Fabrication)	75,000/-	75,000/-	Nil	Nil	75,000/-	Nil	75,000/-	Nil	
6	Overhead expenses	3,00,000/-	1,00000/-	Nil	99,751/-	49/-	Nil	99,800/-	200/-	–
Recurring Items (General)		**6,45,000/-**	**2,65,000/-**	**16,350/-**	**1,36,139/-**	**1,05,399/-**	**5,262/-**	**2,63,150/-**	**1,850/-**	Interest: Rs.30,862/- 2012-13 to 2015-16
Interest		-	-	-	-	-	-	-	**30,862/-**	Returned: **Rs.42,712/-**
7.	**Total**	**18,45,000/-**	**14,65,000/-**	**16,350/-**	**13,26,139/-**	**1,05,399/-**	**5,262/-**	**14,53,150/-**	**42,712/-**	

05/11/2017

Signature of Principal Investigator

6-11-17

Signature of Accounts Officer

Finance Officer
Tezpur University

Signature of the Head of Institute

Registrar
Tezpur University