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Experimental studies on the performance of Hamara ST-5 Stirling Engine and possibilities for performance improvement

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Abstract

This paper is concerned with experimental investigation on the *HAMARA ST-5* Stirling engine. The nature of the investigations are (a) the energy balance, (b) the determination of frictional power, (c) performance with original combustor, (d) heat transfer analysis, (e) vibration problem, and (f) a simpler combustor design. The study shows that 40 - 45% of the energy delivered by combustion is transferred to the engine, of which 91 to 93% goes into the cooling water including the frictional losses and the remaining 7 to 9 % is realised as useful power. The overall efficiency based on total energy supplied is about 3 to 3.3%. The frictional power was determined from the motoring tests which was found to be about 10% of the total energy supplied. Performance with the original combustor showed that at a power input of 65 kW and crankcase pressure of 4.5 bar (g), one could get 3.4 kW of delivered output. However, this single point output could not be realised normally and the output recorded in a less-than carefully attended operation resulted in

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0.8 - 1.2 kW output. A critical analysis of the heat transfer involved with this combustor showed radiant mode to be dominant. The problems were assessed to be a combustor of high inertia causing long startup (25 to 30 mins), consequent poor efficiency and if considerable attention is not provided, it would result in loss of power. There were also severe vibration problems probably inescapable in the present design, which were later suppressed to a large extent by providing a large, sturdy and expensive foundation. Further, the non-return valve and the internal compressor requires redesign for robust and reliable operation.

The positive contribution of the present work apart from the clear assessment of the performance is the development of an inexpensive powdery biomass combustor which can use sawdust and one needing very little attention except at start. The startup time is brought down to about 10 minutes and power delivered in excess of 2 kW. The installation cost of power generation is about 30,000 - 35,000 Rs/kW as of now. The authors see the possibility of enhancing the reliablity of the system by attending to the points noted earlier, but not in reducing the cost per kW.

Introduction

Stirling engines are external combustion engines working on a closed regenerative cycle[1] involving two isothermal and two constant volume processes. They are known for their silent operation and ability to use wide variety of fuels including biomass.

HAMARA ST-5 is one such engine using biomass such as saw dust and rice husk as fuel. Studies were earlier conducted on this engine and are as follows.

Mathur et al[2] have identified the limitations of the fuel feeder system and have suggested modified feeder systems to accommodate a variety of biomass. Tests were reported to have been conducted using the modified feeder for different applications. These test results however show high idling time of the order of 2.5 to 3.0 hours before the commencement of loading and moreover it is not clear whether the engine performed at the rated capacity or not.

Haridasan et al[3] have tested the engine for water pumping application using saw dust, rice husk and firewood as fuel. Various combinations of these fuels were tried and performance characteristics obtained. However the paper does not report the engine's performance characteristics as a function of time.

Vaithilingam[4] has conducted tests using rice husk and saw dust. Torque vs Speed charateristics were obtained at different crankcase pressures. The report however does not contain details of fuel consumption and overall efficiencies obtained.

Balasubramanian et al[5] discuss about the mathematical model due to Ureili for evaluating the engine's performance. Computational results based on actual parameters of the engine has been compared with only a few experimental results. However the paper does not discuss the reasons for the engine's poor performance and ways of improving it.

In the present work these engines are evaluated in order to estimate its performance and suggest suitable improvements upon it. Among the four engines obtained for testing, three engines were subjected to extensive tests and the fourth engine was dismantled for better understanding of the system. Performance tests were conducted on engine using wood gas generated by a wood gasifier developed inhouse. In the earlier stages, the estimate of the frictional losses was not known and motoring tests were conducted. These showed little possibility of reducing the high frictional losses in the present design.

Performance of the engine was also evaluated using the original combustor supplied by the manufacturer. The system performed well during a single run, but could not be repeated again due to problems of fuel feeding. These tests showed that the system required continuous attention throughout the operation.

Therefore it was decided to design a new combustor without the drawbacks of the original one and also one which would require very little attention once the system commences its operation.

The output of the engine being a function of mean pressure, increasing the output by pressurising the engine externally (due to limitations of built-in compressor) was also examined. These tests could not be conducted adequately due to the problems with the non return valve.

This paper is dealt in the following manner:

1. A description of the details of the engine and its specifications
2. Experiments relating to energy balance with performance on wood gas reactor which is used as a heat source
3. Frictional horse power determination
4. Performance with the original combustor
5. Heat transfer analysis
6. Vibration problem
7. Alternate combustor and performance with it.

The Engine

Hamara ST-5 engine is external heated, low pressure Stirling engine, working on a closed regenerative cycle using biomass such as sawdust and ricehusk as fuel.

It is supplied as a two component system, namely the combustor and the engine. The combustor is meant for burning the fuel and supplying the heat of combustion to the working fluid (air) of the engine by means of external heat transfer via the heater head of the engine.

Some of the major components of the engine are heater head, regenerator, cooler, crankcase, and hot and cold working spaces. The schematic illustration of the engine is shown in *Figure 1.* The engine is provided with two pistons, unlike that of an I.C. Engine, one being the displacer which displaces the working fluid between the hot and the cold working spaces, and the other being the power piston meant for delivering power.The arrangement of the displacer, is of beta-type, in which the displacer and the power piston are in line. The space between the domed face of the displacer and the inner surface of heater head forms the hot expansion working space, and the space between the flat face of power piston and the displacer forms the cold compression working space. The working fluid is conveyed between the hot and cold working spaces through the annular spaces around the cylinders via the heater, regenerator and cooler. The piston motion is determined by the crank-rocker mechanism, by which there exists a phase difference between the displacer and power piston. It is provided with a built-in compressor for pressurising the crankcase upto 5 bars (gauge).

The materials used, and the method of manufacture of the major components are shown in *Table 1.* There are two types of piston rings provided on the piston, one being the wear band ring made of teflon impregnated with bronze, which is meant for reducing friction between the cylinder liner and the piston by preventing metal to metal contact and the other being the seal ring made of teflon impregnated with graphite. The nominal parameters of the engine are given in *Table 2.*

The Energy balance

Performance with Wood gas

Among the four engines nomenclatured as E1, E2, E3 and E4, load test was carried out on E1, E2 and E3. The engines E1 and E2 were tested using wood gas generated by a Gasifier system. This mode was chosen because the energy supplied could be controlled and metered. A specially designed swirl chamber is used so as to allow the hot gases to pass close to heater head and allow maximum heat transfer to take place. The outer surface of swirl chamber is insulated with low density alumino-silicate blanket to reduce the heat loss.

The wood gas that is generated in the gasifier system is drawn using the suction of a blower and premixed with air before it is supplied to the swirl chamber for combustion. The gas and air mixture upon combustion transfer the heat to the engine which is realised as mechanical power at the engine shaft. Temperature of the gases at the inlet and the outlet of the swirl chamber were measured using thermocouple. The power delivered by the engine was estimated using a belt dynamometer. The heat loss to the coolant was estimated by measuring the coolant's mass flow rate and its temperature rise.

The test/measurement details are shown in *Table 3*, which is presented to indicate the method employed and typical range of the instruments used for measuring various parameters such as the air and gas flow rate, inlet and outlet temperatures of the gas and its calorific value. The method employed for measuring coolant flow rate, its temperature rise and power output is also given.

Results:

The engine E1 developed a maximum shaft power of 1 kW at a crank pressure of 4 bar (abs) giving a maximum overall efficiency of 3% in a series of tests conducted on it. Similarly engine E2 developed a maximum shaft power of 1.4 kW at 4 bar (abs) giving a maximum overall efficiency of 3.7 %. The low performance of the engine is attributed to the following reasons in the order of importance.

1) Input power being lower than specified.
2) Inability of the system to build higher crankcase pressure (5 bar (g) for rated power of 3.75 kW), which is due to the poor built-in compressor and non return valve design[7].
3) Possibility of leakage of working fluid past the piston, thus reducing the peak pressures in the cycle.
4) Inadequate heat transfer to the heater head.

Performance plot:

The performance plot for E1 and E2 are shown in *Figure 2*, in terms of crankcase pressure which shows that engine output increases with increase in input power and crankcase pressure, however there is a fall in output with increase in crank pressure for E2. This behaviour of engine (E2) is contradictory to the fact that the rise in crank pressure should result in higher output. The reason for the above behaviour may be the inability of the piston rings to provide effective sealing and high fluid frictional losses.

Energy flow:

Using the data of cooling water flow rates, the inlet and outlet temperatures, the heater inlet and gas outlet temperatures as well as load measurements energy balance is drawn up. It is seen from the energy balance represented in *Figure 3* that the maximum useful power obtained is about 1 kW at 550 RPM at an energy input of 30 to 33 kW, giving an overall efficiency of 3 to 3.3%. It shows that nearly 40 to 45% of the energy delivered by combustion goes into the heater head, of which 91 to 93% goes to the cooling water including the frictional losses and the remaining 7 to 9% is realised as shaft power. As the amount of heat lost due to friction was not known from the energy balance, it was proposed to conduct motoring test on the engine so as to determine the frictional losses.

Frictional horsepower determination

In order to estimate the frictional horsepower of the engine, motoring tests were conducted on the engine. A 7.5 kW, 3-phase induction motor was used to drive the engine through a belt drive. The power drawn by the motor is estimated by measuring the current, voltage and power factor across the three phases.

When a Stirling engine is motored, a refrigeration effect is obtained at the heater head. Therefore, to estimate the amount of heat extracted from the heater head, due to refrigeration effect, a chamber was mounted surrounding the heater head and coolant (in this case, water) circulated through it. The heat that is rejected to the cooler (in actual engine) is taken as the sum of heat extracted at the heater head end and the power input to the engine. Therefore it can be concluded that the power input to the engine is the actual *fhp* consumed by the engine.

This test was carried for 5–6 hrs till the system reached a steady state (no change in temperature of coolant flowing in either of the jackets). These tests were done at different crank pressures and at constant speed (530 RPM).

It can be seen from the *Table 4* that there are two sets of readings given as the function of crank pressure. The first set of readings represents the *fhp* consumed by the engine, which is based on thermal measurement. The other set of data refer to the power input to the engine without considering the motor and transmission losses. These two sets of readings are different. It is hypothesized that the motor efficiency and to a limited extent transmission efficiency accounts for the difference. The ratio of the two is presented in the last column. It can be seen that with increase in power, the efficiency also improves. The values like 65–70 % are typical of efficiencies at 50–60 % load for the motor. There is a possibility for the efficiency to be as low as 44% at lower loads, but is difficult to justify.

These data show that *fhp* increases with the increase in crank pressure. Among the two factors that contribute to the *fhp*, the fluid friction which is a function of density and sqaure of fluid velocity, increases *fhp* at higher crankcase pressure. The solid friction which is a function of speed does not seem to contribute to the increase in *fhp* as the tests were done essentially at constant speed.

Results and discussions

It is seen from the *Table 4* that the *fhp* value is fairly high which is about 3.2 kW at 4.0 bar pressure. There is a possibility for this value to be much higher at higher crankcase pressures and at higher speeds. The possibility of improving the engine's performance based on reduction of frictional losses is little in the present design. This conclusion was arrived at, after the inspection of engine's flow passages and working parts which showed that the flow passages cannot be altered in the present engine, so as to make it more aerodynamic and lead to reduction of fluid frictional losses. Similarly the possibility of reduction of solid frictional losses is little as the engine parts work in a totally oil free environment.

Performance with original combustor

The *HAMARA* Stirling engine was tested using the original combustor supplied by the manufacturer as shown in *Figure 4* with saw dust as the fuel. The combustor is provided with a screw feeder for feeding the fuel, run by a belt drive from the engine. The combustor is also provided with a blower which pumps the pre-heated air, heated by the outgoing burnt gases for combustion thus improving the efficiency of the system. The rate at which the fuel is fed to the combustor can be varied by means of a variable pulley arrangement. The inner surface of the combustor is lined with fire bricks and its front end is suitably shaped to receive the heater head of the engine. The combustor along with hopper weighs over 200 kg (dead weight), which poses problems during handling and transhipment.

Power output and efficiency:

When the engine (E3) was tested using the above arrangement as shown in *Figure 5*, the maximum power estimated was around 3.4 kW at 660 RPM and at 4.7 bar (gauge) crank pressure. The temperature at the top of heater head was around 875 to 900 K. The amount of energy supplied was around 65 kW at a fuel feed rate of 14 kg/hr (with some firewood to initiate and sustain combustion). Thus the maximum overall efficiency obtained was around 5.2%. These were the figures obtained during one single run which could not be realised again mainly due to problems with the feeder.

The problems encountered with the original combustor are as follows:

1. The feeder stops in presence of coarse fuel particles. This therefore directly limits the fuel to be used to be in a finely pulverised form (it is claimed that the system at SJCE., Mysore does not have such a problem, but understood to require some attention).
2. Requires almost continuous attention during its operation.

3. High thermal inertia, the time required to get the combustor sufficiently hot is about 25 to 30 min, before starting the engine. This also limits the response of the combustor to the requirements of the power variations of the engine.

For better understanding of the system it was felt necessary to conduct critical analysis of the original combustor so as to determine the contributions made by different modes in the total heat transfer process from the hot gases to the heater head of the engine. The analysis is as follows.

Heat transfer analysis

Analysis

The fuel feed rate and the temperatures of the gas, combustor walls and heater head is given in *Table 5*. The dimensions of the flow passages between the heater head and combustor required for velocity calculations were measured and the calculated surface area from these measurements are given in *Table 6*. The *Table 7* contains data like the constants required for heat transfer calculations along with the properties of air[6].

The equations used for the analysis are as follows which are based on[6]

Radiation $Q_r = \sigma = A_1 \varepsilon \, (T_c^4 - T_h^4)$

Convection

1) Sides

$$v_1 = m_{ft} / A_{e1}$$
$$Re_L = v_1 L / \nu$$
$$Nu = h_s L / k = 0.332 \, (Re_L)^{0.5} \, (Pr)^{0.33}$$
$$Q_s = h_s A_2 \, (T_g - T_h)$$

2) Dome

$$v_2 = m_{ft} / A_{e2}$$
$$Re_D = v_2 D / \nu$$
$$Nu = h_d D / k = 0.332 \, (Re_D)^{0.5} \, (Pr)^{0.33}$$
$$Q_d = h_d A_3 (T_g - T_h)$$

Results

Heat transferred by radiation $Qr = 5.66 \times 10^{-8} \times 0.332 \times 0.8 \, (1250^4 - 873^{4)}) = 27kW$

Heat transferred by convection $Q_c = Q_s + Q_d$

$= 8.3 \times 0.833 \, (1300 - 873)$

$+ 2.3 \times 0.09 \, (1300 - 873) = 3kW$

Discussion

From the first hand information obtained from the manufacturer, an impression was given that convection mode of heat transfer plays a dominant role in the heat transfer process, from the hot gases to the heater head of the engine. But the critical analysis made above shows that radiant mode not only contributes substantially but also plays a dominant role (87 – 90%) in the total heat transfer process. The contribution by convection mode is about 10–13%, this is mainly due low velocity of the gas (4.5 m/s), contributing towards low Reynold's number (5440) and low heat transfer coefficient (8.3 W/m^2 K). As the size of the combustor is small the effect of gas radiation has been neglected.

The sensitivity of temperature, velocity and dimension variations are considered, which is represented in *Table 8*, which refers to the contribution made by radiation and convective modes. These have been considered for different gas & combustor wall temperatures and different gas velocities. These only reaffirm radiant heat transfer to be dominant and a major contributor, which is truely a merit of this combustor.

Vibration problem

Due to the unbalanced forces resulting from lack of dynamic balancing of the heavy reciprocating masses, vibration is caused during the working of the engine. This lack of dynamic balancing is mainly due to the crank-rocker mechanism.

This engine requires a foundation bed for its installation: A light foundation bed will not suffice as the engine starts vibrating at low speeds. From our earlier experience, it was found that a foundation bed of size 1.2 m length × 0.78 m width × 1.5 m height (0.45 m deep in ground) would not suffice the requirement and hence a foundation bed of dimension 1.6 m length × 1.2 m width × 1.67 m height (0.9 m deep in ground) was constructed and this provided a base without vibrations.

This feature seems to be strengthened further by the fact that an engine installed at Wandiwash (100 km from Madras), under MNES field demonstration programme faced similar vibration problems, which was later suppressed by reinforcing the foundation bed, which finally has a dimension of 1.8 m × 1.5 m × 0.9 m (deep in ground).

The heavy foundation bed besides requiring more space also leads to higher expense for engine installation (around Rs 6000 at present). This is being brought out since proper information is not available in the manuals of the engine.

Alternate combustor

The maximum output obtained by using the original combustor during regular run was 0.8 to 1.2 kW as against the rated output of 3.75 kW. This problem was assessed to be due to insufficient heat input to the engine and low operating pressure. It was therefore decided to look into a new combustor using inexpensive powdery biomass such as sawdust, that would perform at the required power level and one without the drawbacks of the original combustor. A combustor which possesses the qualities of low thermal inertia, deliver the required power level and respond to the power requirement of the engine fast was set out to be designed and one requiring minimal or no attention after the commencement of operation.

In the new design shown in *Figure 6*, a chamber is used along with the combustor so as to allow the gases to pass closely along the heater head and ensure that maximum heat transfer takes place to the heater head. Provision is made to measure temperatures at various points around the heater head both axially and circumferentially by providing opening in the chamber for thermocouple insertion. The new combustor is essentially a cylindrical drum with passages at the bottom for air entry as shown in *Figure 7*. The outer surface of the combustor is insulated with low density alumino-silicate blanket so as to reduce the heat loss from the system. In order to ensure that complete combustion takes place, the stoichiometry ratio of A/F should be maintained around 6.0, which is accomplished by having the height to .which the combustor is loaded with fuel equal to about six times the diameter of the core used. A fuel vapour & air mixer was felt essential for proper mixing of air and fuel and to ensure complete combustion to take place[8]. This is done by placing perforated sheet having 6 mm diameter holes, on the top of the combustor.

As the amount of energy to put in was known, from the experience of the previous combustor, it was decided to design a combustor which would perform around the same level. Combustors of different configurations were tried and experiments carried out, to determine their power level. They are given in *Table 9* along with brief comments about their performance. The power level is estimated by determining the mass loss per unit time, after lighting the stove and taking calorific value of the fuel into consideration. The calorific value assumed for sawdust is 14 MJ/kg.

The *Figure 6* gives the arrangement for the load test with the new combustor, a single phase 3.5kVA alternator has been used to estimate the power output of the engine. The power output is estimated by recording the voltage and current (without considering the alternator and transmission efficiency).

The performance of the engine with different combustors is given in *Table 10*, which contains details of maximum heater head temperature and power output attained along with brief comments about their operation. From these experiments it can be concluded that the heat supplied by the *comb 5* almost satisfies the specified power requirement and the temperature obtained at the heater head is fairly constant and is above 825 K during its entire operation. But the problem encountered was the smoky exhaust, this being due to insufficient amount of air available for combustion which is caused by excess pressure drop at entry of the chamber. This problem was solved by raising the stack height to three metres.

Method of operation

The engine E3 was tested using *comb 5*. The arrangement of it is shown in *Figure 6*. The combustor was fired in the following manner. To start with initially two cores including the top surface was lit, after about 10–15 min when the head was hot enough (625 K) for the engine to be cranked, the remaining two cores were lit with a gap of 2 min. This is done in order to get the chamber and the draft column to be sufficiently hot such that natural draft can be induced resulting in a smokeless operation.

Performance

After about 10 min of start, the engine delivered a power of about 0.3 kW and it increased to 1.2 kW as the crank case was pressurised and maintained around 4.0 bar (g). This power output was fairly constant for about 30 min and then the output increased to 1.8 kW, which was constant for another 60–65 min. This sudden increase in output was attributed to the increased radiant effect from the red hot bed of char in the combustor, there was slight improvement in temperature and it reached 875 K. After this, the power declined to 1.2 kW which continued for about 40 min. The power vs time plot is represented in *Figure 6*, which shows the variation of the power generated as a function of time. The area under the curve gives the energy delivered and the product of mass of fuel used and its calorific value gives the energy supplied. The ratio of the two gives the overall efficiency which is about 2% for the above case.

The power output indicated in *Figure 8* is the electrical output from the alternator. A typical efficiency of 3.5 kVA alternator at the loads indicated above are estimated at 70%. This implies a delivered output of 2.5 kW from the engine. This result is argued to be consistent because the same test done with belt dynamometer and alternator keeping other elements same has shown the ratio of the input to the output power of the alternator to be 70%.

Discussion

It can be concluded from the above experiment that the new combustor almost gives a trouble free and smokeless performance and also requires the least attention. The output of the engine can be made more stable and also further enhanced by improving the power level of the combustor. The new combustor along with the chamber weighs about 75 to 80 kg (dead weight) and is relatively easier to handle and less expensive (in absence of blower, hopper and feeder) when compared to the original combustor.

Conclusions

This paper reports the investigations carried out on the indigenously manufactured Stirling engine. Tests and other investigative studies show that the engine has the capacity to deliver higher output provided there is sufficient heat input to the engine. As there are problems associated with the usage of original combustor a new combustor has been designed. Using this it is possible to realise mechanical power of the order of around 2.5 kW besides reduction in startup time. Moreover as the new combustor does not require a fuel feeder system when compared to original one, some amount of power is saved which is available at the engine crankshaft as useful power. The other limitations with the engine are the built-in compressor and poor non-return valve, if these could be solved it is possible to pressurise the engine to higher pressures and realise higher outputs. There are other problems associated with the engine like excess vibration which can be overcome only by providing a heavy and expensive foundation and this increases the overall cost of power generation. Now the estimated cost of power generation is about Rs. 30,000 to 35,000 per kW. There is a possibility of enhancing the reliability of the system by attending to the points noted earlier, but not in the cost per kW.

Nomenclature

m_f	:	Fuel feed rate, kg/hr
m_{ft}	:	Total mass flow rate, kg/hr
T_g	:	Gas temperature, K
T_c	:	Combustor wall temperature, K
T_h	:	Heater head temperature, K
A_1	:	Area considered for radiation calculation, m^2
A_2	:	Area considered for convective calculation, (sides including the fin area), m^2
A_3	:	Area considered for convective calculation,(dome), m^2
L	:	Length of heater head side, m
D	:	Diameter of the heater head dome, m
A_{e1}, A_{e2}	:	Effective area for gas velocity, m^2
v_1	:	Velocity of gas over sides of heater head, m/s
v_2	:	Velocity of gas over dome of heater head, m/s

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Table 1: Materials and Manufacture

Component	Material	Manufacturing Method
Heater Head	Stainless Steel SS 321	Pressed 3 mm shell and fins brazed in H_2 environment
Cooler	Al Alloy (LM 6)	Gravity Die Casting
Linkages	Al Alloy (LM 25)	Forging
Crank case	SG Iron	Casting
Regenerator	Stainless Steel	Tightly packed wool 100 μm dia

Table 2: Nominal Parameters

Power	5 hp/3.75 kW
Speed	720 rpm
Head Temperature	700°C max
Coolant Temperature	50°C max
Crank Pressure	5 atms, Gauge
Power Piston Dia.	270 mm
Displacer Dia.	300 mm
Stroke Length	100 mm
Fuel used	Sawdust, Rice Husk
Start	Fuel Wood

Table 3: Measurement Details

Parameters Measured	Techniques	Range
Combustion Source	Wood Gasifier	upto 4 g/s
Air Flow Rate	Venturi	0 - 11 g/s
Gas Flow Rate	Venturi	0 - 7 g/s
Combustor inlet and Outlet Temperature	Type K Thermocouple	upto 1275 K
Gas Cal Value	Junker's Calorimeter	5-6 MJ/kg
Engine Power	Belt Dynamometer	50 kg in 200 kg

Table 4: Results of *fhp* test

Crank Pr. atm, abs	fhp^{+} kW	Power * Inp, kW	η^{**} %
1.0	0.95	1.9	44
2.0	2.0	3.0	65
3.0	2.7	4.0	66
4.0	3.2	4.5	70

Table 5: Input for Analysis

Fuel Feed Rate (m_f)	14 kg/hr
A/F	6.0
T_{wall}	1250 K
T_g	1300 K
T_{head}	873 K

+ Frictional Power from thermal balance,* = Efficiency of motor and transmission taken as 100 %, η^{**} = Efficiency of motor and transmission works out to this value

Table 6: Dimensional Details

Length of heater side(m)	0.220
Diameter of heater head dome (m)	0.330
Area of radiation (A_1), m^2	0.332
Area for convection	
Sides (+fin area) (A_2), m^2	0.833
Dome (A_3), m^2	0.09
Effective area for velocity, A_{e1}, m^2	0.016
A_{e2}, m^2	0.132

Table 7: Data for heat transfer calculation

Emissivity of Fire Brick(ϵ)	0.8
Stefan-Boltzmann constant(σ)(W/m^2K^4)	5.66E-8
Properties of air, 1300 K	
Kinematic viscosity(ν) (m^2/s)	181E-6
Thermal conductivity	0.0837
(k_g), W/m K	0.0837
Prandtl number(Pr)	0.705
Density(ρ), kg/m^3	0.2707

Table 8: Contributions by different heat transfer modes

Parameters	Q_r, kW	Q_c, kW	Q_{total}, kW
Nominal, T_g, T_w = 1300, 1250 K	27	3	30
T_g, T_w = 1400, 1350 K	40	3.7	43.7
T_g, T_w = 1250, 1200 K	22	2.7	24.7
v_1, v_2 = 9, 1 m/s, at nominal T_g, T_w	27	4.25	31.25
Passage dimensions ± 10 %	29/24	3.4/2.7	32.4/26.7

Table 9: Combustor Types

Type	Max. Dia. (mm)	No of Cores	Core Size (mm)	Fuel Ht. (mm)	Fuel Wt. (kg)	Mean Power P, kW	Remarks
Comb 1	400	1	150	900	25	22	Stoich A/F, P low
Comb 2	400	3	80	480	15	35	Stoich A/F, P low
Comb 3	400	4	90	540	22	45	Stoich A/F, P low
Comb 4	400	45	25	125	3.2	45	Fuel Rich, P low
Comb 5	570	4	140	840	50	58	Stoich A/F, P OK

Table 10: Performance with Different Combustors

Type	Head Temp, K	Output P (kW)	Remarks
Comb 1	700	0.6	Temp. low, Constant, Low P
Comb 2	725	0.7	Temp. low, Constant Low P
Comb 3	800	1	Temp. Varying, Fuel rich, Low P
Comb 5	above 825	1.8	Temp. Constant, Combustion & P OK

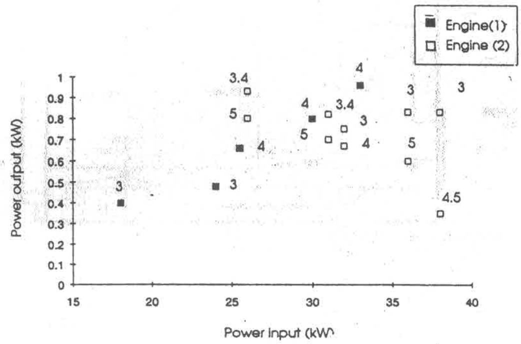

Figure 2 Performance plot

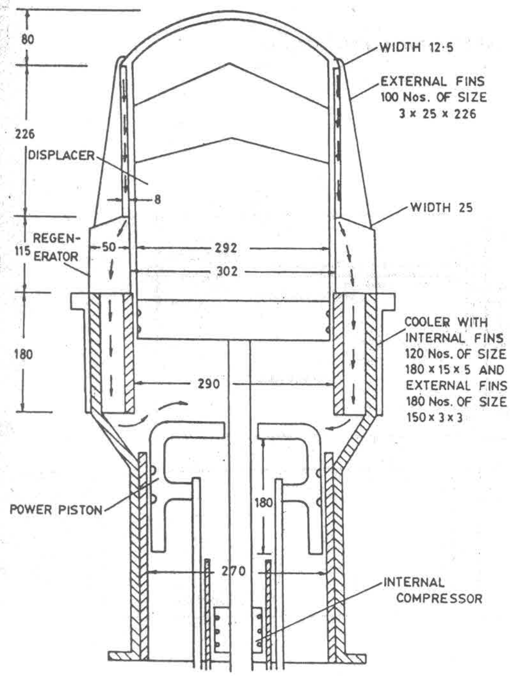

Figure 1 Schematic of ST-5 engine

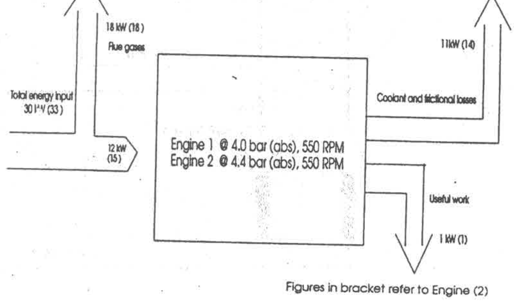

Figure 3 Energy balance for engine (1) & (2)

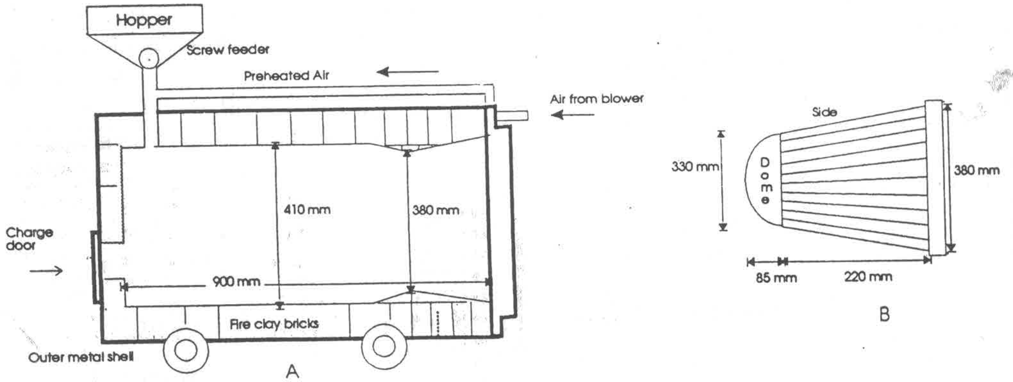

Figure 4 (A) Sectional View of Original Combustor (b) Heater Head

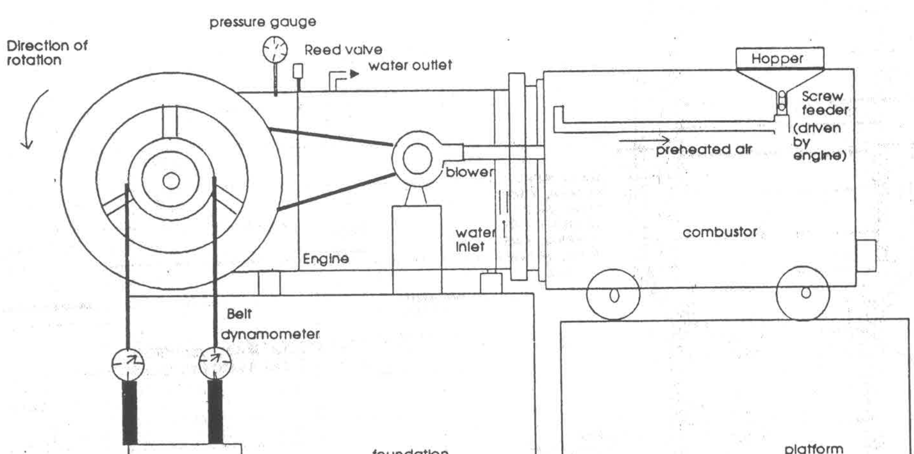

Figure 5 Experimental setup with original combustor

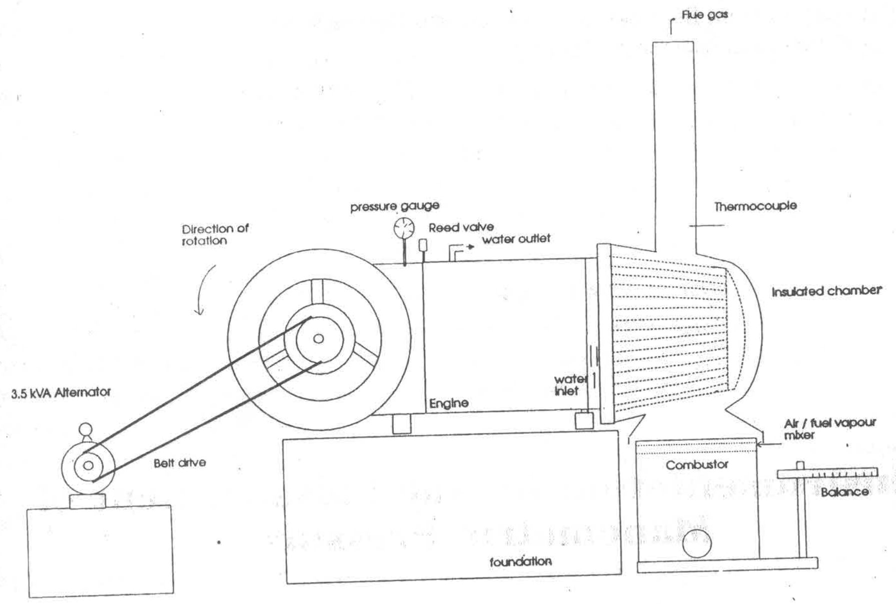

Figure 6 Experimental setup with new combustor

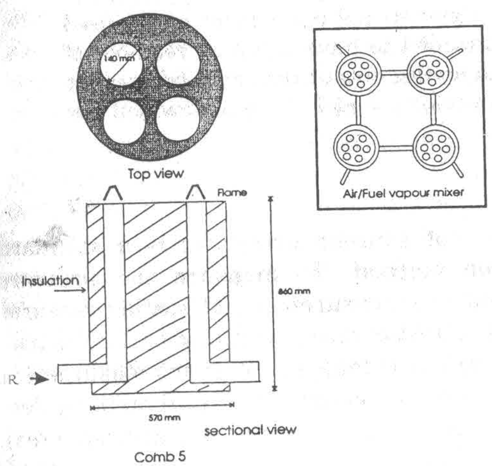

Figure 7 Alternate Cumbustor

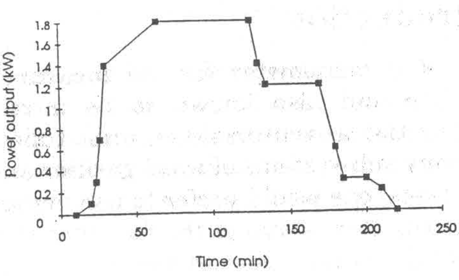

Figure 8 Power–Time Curve