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Waste Heat Recovery in Spark Ignited Internal Combustion Engines

Abhishek C.P

B.Tech Graduate, Department of Mechanical Engineering, Cochin University of Science and Technology, Cochin ,
Kerala, India

ABSTRACT: In this project , a 4 stroke single cylinder SI engine (Kawasaki Boxer 100) was utilized to conduct a comparative study on SI engines with and without regeneration. The basic scheme of the project is waste heat recovery. Majority of losses in IC engines come in the form of incomplete combustion and exhaust blow down losses. This problem was rectified by fabricating a heat exchanger and allowing the waste heat carried away by the exhaust gases to preheat the inlet air in a counter flow parallel pipe heat exchanger. A comparative study was done by conducting a standard brake load test on the engine with and without the preheat setup. The experimental setup involved mounting the engine (with the brake drum) and heat exchanger on laboratory floor .Fuel and temperature readings required a panel board. The design of the heat exchanger was basically done with the help of ANSYS FLUENT software . The overall setup was cheap and proved to be economical.

KEY WORDS: Regeneration , Preheat , Heat Exchanger , Brake Load Test, ANSYS FLUENT.

I. INTRODUCTION

We use a recuperative heat exchanger to increase the temperature of the inlet air before it enters the carburetor. The heated air provides more vapourisation to the fuel in the carburetor and also better mixing of the charge takes place. The benefit for the engine is during idling as well as high power ranges where high air fuel mixture is required . The waste heat in the exhaust gas provides better charge mixing as well as higher thermal efficiency because of better combustion inside the combustion chamber. Apart from these factors the emission characteristics were studied on the engine with and without preheating. The production of CO and HC's significantly reduces due to complete combustion. A comparison of the performance parameters for the engine with and without preheating reveals the advantages and disadvantages of the process.

II. LITERATURE SURVEY

Engine exhaust heat recovery is considered to be one of the most effective means of waste heat management and it has become a research hotspot recently. For example, Doyle and Patel have designed a device for recovering exhaust gas heat based on Rankine cycle on a truck engine. The commissioning experiment of 450 kilometers showed that this device could save fuel consumption by 12.5%. Cummins Company has also done some research on waste heat recovery on truck engines, and the results showed that engine thermal efficiency could improve by 5.4% through exhaust heat recovery. James Conklin and James P. Szybist have designed a six-stroke internal combustion engine cycle with water injection for in-cylinder exhaust heat recovery which has the potential to significantly improve the engine efficiency and fuel economy. R. Saidur et al Rankine bottoming cycle technique to maximize energy efficiency, reduce fuel consumption and greenhouse gas emissions.

III. SELECTION OF MATERIAL FOR HEAT EXCHANGER

The material selection for the heat exchanger has to be done considering the thermal conductivity, resistance to flow and strength at high temperature. The best available materials are copper ,aluminium, steel. Although copper and aluminium have higher conductivities than steel ,steel was selected as the material considering the fact that it is easily

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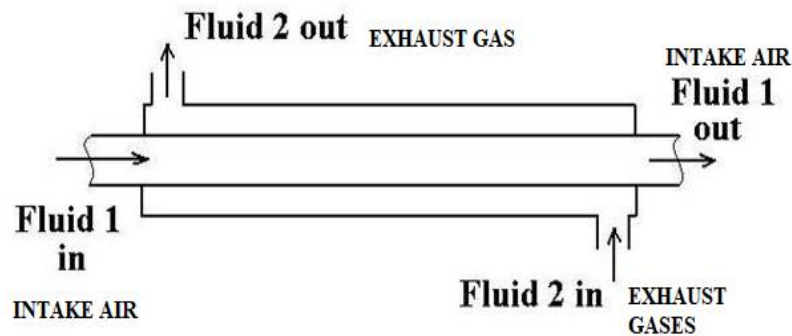
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available and easier to fabricate the heat exchanger using steel. Galvanized iron pipes were selected out as fouling would be a major problem with exhaust gases flowing through the pipes and degreasing and cleaning would be easier. Also GI pipes of a range of diameters were commercially available.

IV. DESIGN FOR HEAT EXCHANGER

The heat exchanger used was a Straight pipe counter flow heat exchanger. The inner diameter was .75 inch and the outer diameter being 2 inch .The total length of the heat exchanger used was 500mm.The material used for the heat exchanger was GI. The heat exchanger was so designed that it produced optimum effectiveness and was able to rise the temperature of the inlet air by 10 to 20degree.The inner pipe was used for inlet air and the outer one for the exhaust gases. The temperature rise was restricted to such values because a very high increase in temperature results in over expansion of inlet air and volumetric efficiency would considerably reduce. The reduction in volumetric efficiency would in turn reduce the power output of the engine .Previous experiments with electric inlet air pre heaters have shown that the optimum rise in temperature should be about ten degree without considerable loss in power output. Also the problem with high inlet temperature is that after the compression stroke the temperature of the mixture rises and the temperature shouldn't cross a particular value to prevent knocking. Knocking is a serious issue that damages the engine cylinder



Double Pipe Heat Exchanger Counterflow

Fig: Double pipe counter flow heat exchanger

A straight pipe setup was adopted so that any kind of pressure losses in the system are minimized and the engine runs steadily .Initially NTU method was employed to find out the effective length of the heat exchangers. The diameters were so selected that they wouldn't allow any large pressure drops in the flow, as that would cause pumping work in the engine. All the calculations were performed at 1500rpm .The effectiveness of a heat exchanger is the ratio of actual heat transfer to the maximum possible heat transfer. For heat capacities and mass flow rate being same for both hot and cold fluid effectiveness

$$e = NTU / (1 + NTU)$$

$$NTU = U * A / m * C_{min}$$

To find U we find the heat transfer coefficients of inlet and outlet. We employ Dittus-Boelter equation for h_i and h_o .

$$H \text{ eating} = .023(Re^{\cdot 8}) * (pr^{\cdot 4}) * k/d$$

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$$H_{cooling} = .023(Re^{.8}) * (pr^{.3}) * k/d$$

Refer to [7] for calculation of U from h_i and h_o .

Here Re is the Reynolds number calculated based on the hydraulic mean diameter. The velocities can be obtained from average flow rate at 2000 RPM. Following were the values obtained

Re(heating)= 1400

Re(cooling)= 3200

Prandtl number inlet= .711

Prandtl number outlet=.7

Overall heat transferr coefficient $U=23.7$ (w/(m²k))

Effectiveness $\epsilon = .11$

The purpose of the above calculations is to obtain the area of the heat exchanger. Since the diameters of the inlet and outlet pipes are fixed the calculation of area automatically provides the length for the heat exchanger. The obtained length is then used as an initial value .The length would be varied and simulation would be performed with the prerequisite boundary conditions until satisfactory results are obtained. The primary requirement of the heat exchanger to raise the temperature of the intake air by 10 to 20 degree Celsius.

V. SIMULATION AND RESULTS

Based on this calculation a rough effective length is obtained. We use this length as an initial bound to our problem and iterate varying the lengths above the value to get the required effectiveness. We finally choose a length of 500mm for the heat exchanger. The simulation was done on the heat exchanger with following boundary conditions.

Hot inlet temperature = 400k

Cold inlet temperature = 300k

Hot inlet static pressure =0 bar and constant mass flow inlet .00043kg/s Cold inlet static pressure =0 bar and constant mass flow inlet .00043kg/s.

The values of mass flow rate obtained from volume of the engine , RPM as 500 and assuming constant air density.

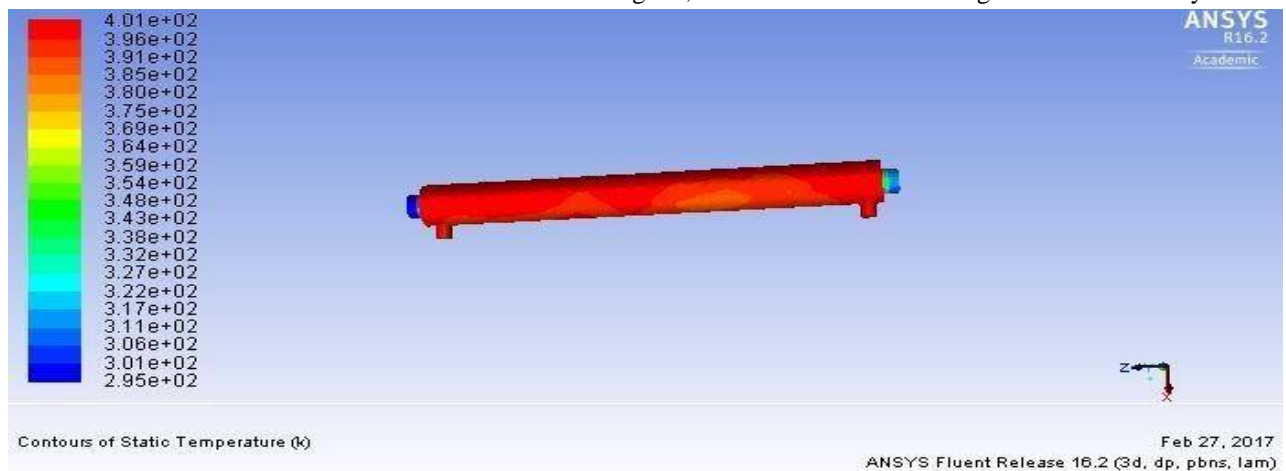


Figure: Temperature distribution in heat exchanger.

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OBSERVATIONS:

The average temperature rise obtained for the above heat exchanger is 10degree. If the length was infinite the rise would have been 100c. So the effectiveness of the heat exchanger is 10%. To be noted that the inlet temperature of the hot fluid was taken as 400k but in actual case it may be 500k. So choosing a heat exchanger with effective length 500mm was feasible and could produce more than 10c rise in temperature. As the experiment is conducted without pressuring the air we required only 10 to 20 degree rise in temperature, otherwise it would seriously affect the volumetric efficiency of the engine. So to summarize the result the effectiveness was found to be 0.11

COST EVALUATION:

The fabrication involved the assembly of GI pipes for the inner and outer pipes of the heat exchanger. The total fabrication cost was Rs 1500. The outer pipe was wound with insulating wool and covered with Aluminium foil to prevent radiation losses

VI. TEST SETUP

Basically, we conduct Brake load test on the engine. The components used in assembly of the rig are as follows: 100cc IC engine (spark ignition type), Load test display board consisting of thermocouple display, fuel burette, brake drum with spring loaded dynamometer, Counter flow Straight pipe type heat exchanger and Tachometer to read the engine speed.

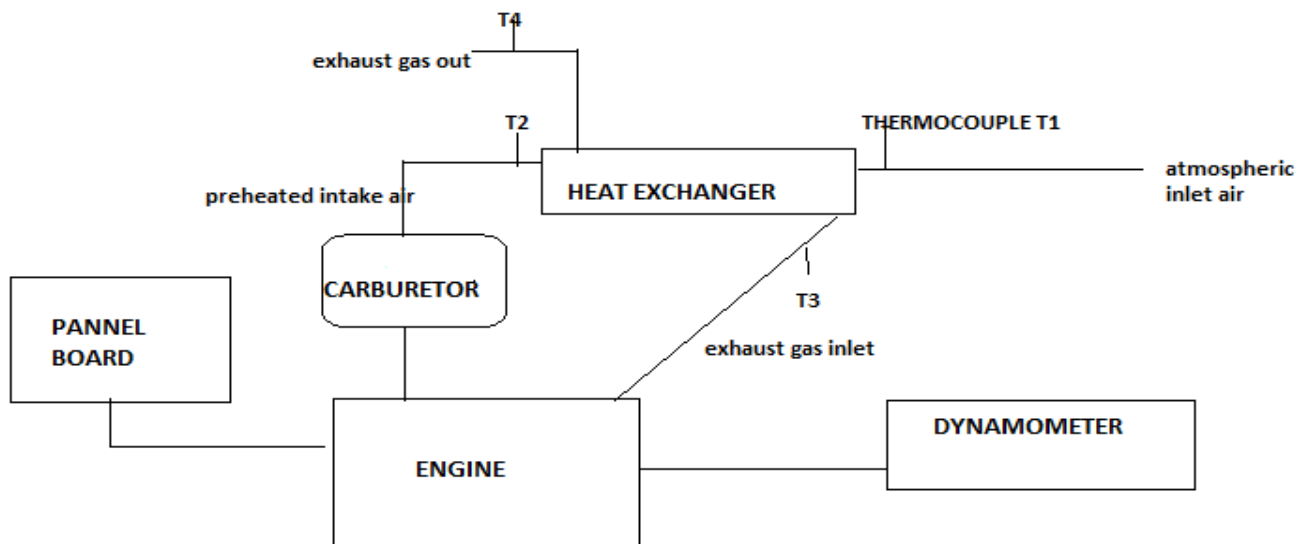


Fig: Experiment setup diagram

The above figure shows how the hot and cold fluids flow through the system and exchange heat at the heat exchanger.

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VII. TEST PROCEDURE

The test consisted of brake load test on the engine with and without the preheat setup. Values in both cases were then compared .

1. The engine is first checked for all the electrical connections to be correct. Ensure that the engine is at no load
2. The engine is started at no load and preheat connection is not established until steady state.
3. Ensure steady state is attained by looking at the thermocouple readings .when reached steady state establish the preheat connection and then taking all the no load values in the display board. Assure that the rated output shaft speed of 500 rpm is attained
4. Increase the load in the spring brake dynamometer in steps of 2kgs ,take the values of fuel flow time for 10cc and the thermocouple readings T1, T2,T3,T4
5. Reduce the load gradually to 0 I steps and turn off the fuel supply cock and other electric connections

Note that as the additional heat exchanger setup is added the engine takes some time to reach the steady state at a particular load .The steady state values can be judged from the thermocouple readings. When the readings become stagnant it is suggestive of steady state. Although in real life scenario the engine is never in steady state , it is important to take the steady state to compare the performance.

VIII. TEST RESULTS

Test 1 and test 2 were performed with and without the setup accordingly. The results were tabulated and performance characteristics obtained. Following is the table of test results.

	fuel flow cc	S1(kg)	S2(kg)	RPM	Time(s)	Flow rate(cc/s)	Input Power (w)	Brake power(w)	BSFC (cc/j)
without setup	10	0.6	2.8	500	42.85	0.233333	1100	57.56667	0.004053
	10	1.2	4.2	500	31	0.322581	1500	78.5	0.004109
	10	1.5	6	500	29	0.344828	2250	117.75	0.002928
	10	2.2	8	500	30	0.333333	2900	151.7667	0.002196
	10	2.7	12	500	36.75	.272109	4650	243.35	.001118
with setup:	0	0	0	500	52	0	0	0	0
	2	0	2	500	37.8	0.05291	1000	52.33333	0.001011
	2	0	4	500	43.9	0.045558	2000	104.6667	0.000435
	2	0	6	500	36.13	0.055356	3000	157	0.000353
	2	0	8	500	29.4	0.068027	4000	209.3333	0.000325
	2	0	10	500	27	0.074074	5000	261.6667	0.000283

Table.: Table of test results

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It can be clearly noted that the BSFC (brake specific fuel consumption) on an average reduced by at least half its original value on an average. The maximum difference in BSFC is .003(cc/j). This definitely points to the fact that the efficiency of the engine has increased with the setup.

T1	T2	T3	T4	BP(w)
31	49	144	115	261.667
31	48	140	112	209.333
31	45	135	110	157
31	45	135	110	104.667
31	45	135	110	52.333

Table: Table for temperatures vs brake power

Here T1 represents the cold inlet, T2 represents the cold outlet, T3 represents the hot inlet and T4 represents the hot outlet. All the temperatures are in degree Celsius. The temperature readings are tabulated against the brake power. We can observe that the temperature of exhaust gases increases with the brake power and proportionally the inlet air temperature goes on increasing. This increase in inlet temperature assists the engine to meet higher brake power at lower fuel consumption rate.

With comparison it can be established that the performance increased with setup. The performance plot shows the proper comparison between the two.

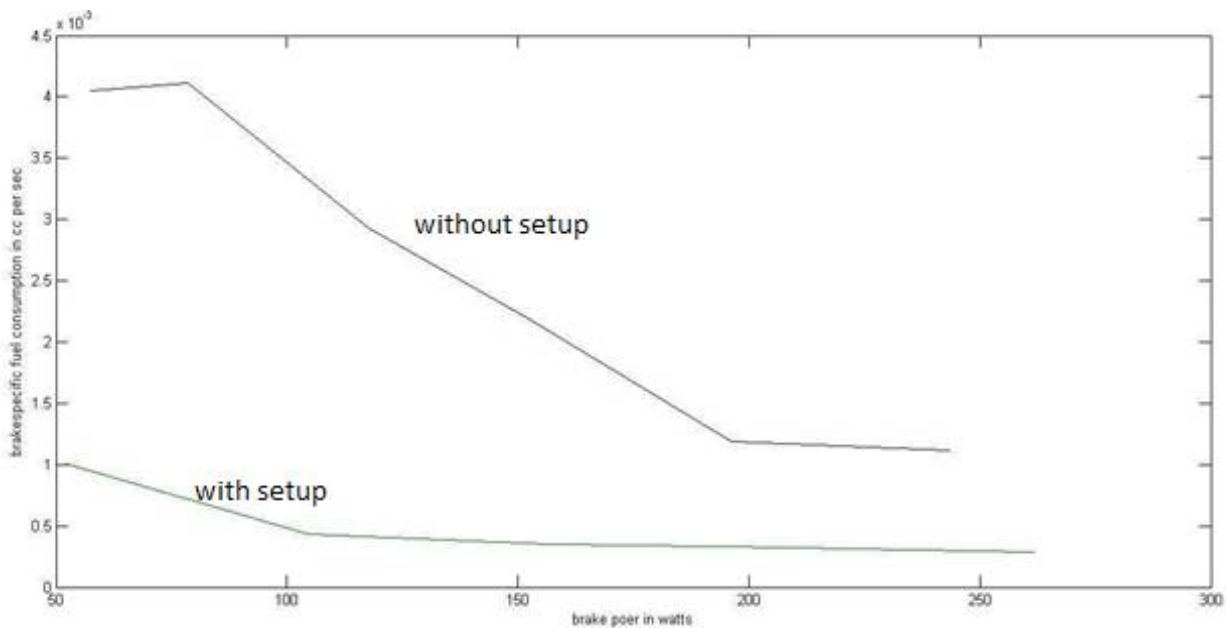


Fig: brake specific fuel consumption (cc per sec) vs brake power (w)

As we can see the BSFC (brake specific fuel consumption) difference is first higher and then decreases as the load on

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the engine keeps on increasing. This is because, as the load on the engine increases the engine becomes hotter and hotter and the inlet air without setup gets heated eventually by heat transfer through the engine walls and inlet manifold hence both tend to converge for the given domain of load.

IX. CONCLUSION

The brake specific fuel consumption for the above taken SI engine was found to increase with the suggested inlet air preheating technique. It can be concluded without doubt that the same can be expected for other spark ignited engines. Also the emissions show that the carbon monoxide and hydrocarbon levels are lower. This is suggestive of the fact that the increase in thermal efficiency is aided by the superior combustion characteristics. In an increasing energy demand scenario it is important to opt for a high quality energy. Adopting the above mentioned technique is proven to reduce the fuel demand for SI engine or an increase in mileage. Hence making this a commercial technique is called for and feasible because of its low cost of fabrication.

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