Thermodynamic Analysis of in-Cylinder Exhaust Heat Recovery in I.C.Engine using Water Injection

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***Abstract* - A concept of waste heat recovery from exhaust of a four stroke internal combustion engine is presented in this paper. It is done by adding an additional two strokes to a conventional Otto cycle or Diesel cycle which marks the conversion of four stroke cycle to six stroke cycle. Water is used for generation of power in the additional two strokes. Conversion of water to steam using the heat available from the recompression of exhaust gas is considered as a method to recover the heat wasted in a four stroke cycle. It can be considered as a partial exhaust heat compression event coupled with water injection to obtain an additional power stroke. An ideal thermodynamic model of an engine is made to analyze the variation of thermodynamic parameters in six stroke cycle. The mean effective pressure, power and thermal efficiency of six stroke engine were found using this model. From the various parameters obtained, the waste heat recovery by water injection is an efficient method to recover waste-heat from a four stroke engine and it can considerably increase the work done and mean effective pressure.**

Keywords - ***Six stroke engine, Mean effective pressure, Thermal efficiency, Water injection***

I. INTRODUCTION

The efficiency of a four stroke I.C.Engine is very low due to the loss of heat energy in the exhaust, balancing of the engine, cooling of the engine etc. Improving the efficiency of internal combustion engines is an ongoing area of active research. Numerous designs have been proposed based on the traditional Otto or Diesel cycles, and all of these include four sequential thermodynamic processes or ‘strokes’ of the piston. These are the following strokes: 1) air–fuel intake stroke 2) air–fuel compression stroke 3) post-combustion expansion stroke and 4) exhaust gas discharge.

The modified cycle proposed here adds two additional strokes that increase the work extracted per unit input of fuel energy. These additional strokes involve trapping and recompression of some of the exhaust from the exhaust stroke, followed by a water injection and expansion of the resulting steam/exhaust mixture. The residual exhaust gas is trapped in the cylinder by closing the exhaust valve earlier than usual, i.e., well before top dead center (TDC). Energy from the trapped recompressed exhaust gases is transferred to the liquid water, causing it to vaporize and increase the pressure. This added pressure then produces more work from another expansion process. The steam–exhaust gas mixture is expelled to ambient pressure near the point of maximum expansion [1].

II. IDEAL THERMODYNAMIC ANALYSIS OF THE SIX STROKE ENGINE

The analysis of each stroke is based on the First Law of Thermodynamics for a Control Volume which states that the difference between the energy absorbed to the energy released will be the change in stored energy. For every stroke of the engine cycle, it can be stated as [4]:

$$Q_{in} - Q_{out} + W_{in} - W_{out} + (m_{in} \times h_{in}) - (m_{out} \times h_{out}) = \Delta E$$

The above equation holds for every stroke in the cycle. Each stroke in the six stroke cycle is analyzed according to the above equation. The upper case letters denote the extensive and lower case letters denote the intensive properties. The combustion chamber (engine cylinder) is taken as a control volume and all the energy loss due to friction, coolant etc are not considered.

A computer program was written to solve the energy equation subjected to the appropriate initial conditions and assumptions described as follows for the cycle component processes. The state properties of the fuel, water and the water/exhaust gas mixture were calculated using MATLAB R2012b.

A. Fuel Compression Process

The engine is cranked to get the initial compression and the engine cycle is considered to start with the compression process. The piston moves from BDC to TDC compressing the charge which is initially available in the cylinder.

Initially, to start with, it is assumed that the values of residual exhaust from the previous cycle in the engine during compression stroke is zero. The initial temperature is taken as ambient atmospheric temperature neglecting the temperature due to vaporization, since both parameters depends on the previous cycle, which in turn, depends on the cycle before that, and so on. The initial temperature at the time of the engine valve closing can be considered as atmospheric pressure because the engine is going to start its first compression process. It is assumed that compression process is isentropic between the crank angles 0° to 180°. Since there is no mass flow out of the cylinder, the simplified form of the energy conservation equation for the isentropic compression process can be written as :

$W_{12} = Win = E_2 – E_1$

Since the initial temperature and pressure at the closing of the exhaust valve just before the actuation of initial compression process is specified, the state 1 is completely known. Since the process is assumed to be isentropic from state 1 to state 2, the complete thermodynamic properties at state 2 can be calculated using State Postulate of Thermodynamics for a simple compressible system. The crank angle during the exhaust valve closing and end of the compression strokes are known, giving the combustion chamber volume at that point which enables to calculate number of moles (mass) of charge compressed from state 1 to state 2. Thus the work required for the compression process is thus known for a given crank angle.

B. Fuel Combustion Process

It is assumed that the combustion takes place adiabatically and instantaneously, with piston at TDC. Since there is no work or heat transfer the energy equation can be written as [2]:

Internal energy of reactant = Internal energy of the product i.e.

$Ur\ (T_2) = Up\ (T_3)$

Since the internal energy at the end of the combustion and specific volume is known at the end of the state 3, thermodynamic state at point 3 can be uniquely determined. The pressure and temperature values are also known at state point 2 by which thermodynamic state at point 3 can be calculated.

C. Fuel Combustion Expansion Process

Similar to the analysis of the isentropic compression process, expansion process from state 3 to state 4 is determined. Isentropic expansion process occurs between the crank angles 180° to 360°. By simplifying the energy conservation equation assuming that there is no mass and heat transfer across the control volume:

$-W_{34} = Wout = E_4 – E_3$

Since the mass does not change during the isentropic expansion process to the cylinder volume at bottom dead center (BDC), the thermodynamic properties can be determined with an additional assumption that the expansion process is isentropic from state 3 to state 4. Then the thermodynamic properties at state 4 can be uniquely determined by state postulate since the thermodynamic properties at state point 3 are completely known. Thus the work output for expansion is calculated.

C. Heat Rejection Process

After the expansion stroke due to the combustion of the fuel, the exhaust valve is opened at crank angle of 360° (state 4). The pressure drops to ambient pressure. Temperature of the exhaust drops and the heat rejection process is assumed to be instantaneous (state 5). Simplified form of the energy equation for no mass flow out of the control volume is:

$E_5 – E_4 = -Qout$

The mass remains same as the end of the expansion stroke. Since the specific volume of the engine cylinder and mass of exhaust remaining in the engine cylinder are known at a particular crank angle, the thermodynamic properties at state point 5 can be calculated.

D. Partial Exhaust Rejection

The exhaust valve is open and the exhaust gas is rejected partially according to the movement of piston in the piston-cylinder arrangement. State point 5 is the end of the partial exhaust heat rejection by early closing of exhaust valve.

The partial exhaust heat rejection event starts from 360° CA (crank angle) but the crank angle for early closing of exhaust valve or the crank angle at which the partial exhaust heat rejection event ends is yet unspecified and it is to be taken as a parameter for investigation.

It is assumed that the exhaust gas mixture is uniform and the rejection of exhaust through the exhaust valve is also uniform. That is, when the piston moving from 360 to 450 CA (crank angle), the rotation is half of the total crank angle rotation from BDC to TDC. So assumption is made that half of the exhaust is driven out when the crank rotates half the total crank angle. That is, the rejection of exhaust gas is directly proportional to the rotation of crank. Through the assumption made above the mass remaining in the cylinder can be calculated by using the equation:

$m_{inside} = \mu \times m_{remining}$,

where m_{inside} denotes the mass in control volume after closing of exhaust valve at the end of partial exhaust rejection event, $m_{remaining}$ denote the mass available at the end of the fuel expansion stroke or mass available in the cylinder at the end of heat rejection process and μ denotes the factor by which crank angle rotates Pressure remains as ambient. Since the state points at the state 5 are known, the thermodynamic properties at the state point 6 (end of the partial exhaust rejection) can be calculated by using the state postulates. The work done by the engine to remove the exhaust partially can be calculated from energy conservation equation.

E. Recompression of Exhaust

Recompression of exhaust is done by early closing of exhaust valve. A yet unspecified actual crank angle (CA) for early closing for exhaust valve is unspecified and it is to be taken as a parameter for investigation. It is assumed that recompression process is isentropic. Since there is no mass flow out of the cylinder, the simplified form of the energy conservation equation for the isentropic recompression process can be written as [1]:

$W_{67} = E_7 – E_6$
Since the temperature and pressure at the closing of the exhaust valve just before the actuation of recompression process is specified, the state 6 is completely known. Since the process is assumed to be isentropic from state point 6 to state point 7, the complete thermodynamic properties at state 7 can be calculated using State Postulate of Thermodynamics for a simple compressible system. The crank angle during the early exhaust valve closing and end of the compression strokes are known, giving the combustion chamber volume at that point which enables to calculate number of moles of charge compressed from state 6 to state 7. Thus the work required for the compression process is thus known for a given crank angle closing.

F. Water Injection Process

The simple control volume analysis reduces to a fixed volume at TDC with an as yet unspecified entering mass of water with a given enthalpy or temperature. Water has a specific internal energy at a particular temperature which can be calculated if mass of the water injected and specific enthalpy of water at the temperature at which it is injected. That is, change in internal energy in the control volume from state point 7 to state point 8 (end of the conversion of water to steam) can be equated with the enthalpy water possess at the temperature at which it is injected. Assuming adiabatic conditions and that the water injection is instantaneous, the energy conservation equation reduces to [1]:
$m_{water} * h_{water} = (m_8*u_8) – (m_7*u_7)$
The mass conservation equation was used to equate the mass at state 8 to the mass at state 7 and the mass of the injected water. Now that the two properties of internal energy and specific volume are known at state point 8, the thermodynamic state is uniquely determined. Thus the temperature and pressure at the start of the additional power stroke are known.

G. Water Power Stroke Expansion Process

Water expansion process occurs between the crank angles 540° and 720° CA. The expansion process is assumed to be isentropic. Similar to the analysis of the isentropic recompression process, expansion process from state 8 to state 9 is analyzed. By simplifying the energy conservation equation assuming that there is no mass and heat transfer across the control volume [1]:
$-W_{89} = W_{out, water\ injection} = E_9 – E_8$
Since the mass does not change during the isentropic expansion process to the cylinder volume at bottom dead center (BDC), the thermodynamic properties can be determined with an additional assumption that the expansion process is isentropic from state 8 to state 9 (end of the expansion stroke produced by the conversion of water to steam). Then the thermodynamic properties at state 9 can be uniquely determined by state postulate since the thermodynamic properties at state point 8 are completely known. Thus the work output for expansion is calculated.

H. Exhaust and Intake Processes

At the end of the second power stroke by using water injection the exhaust valve is opened at the crank angle 720° and exhaust process is done from crank angle 720-900. Pressure drops to ambient and the temperature at the end of the exhaust stroke can be found from the equations of an ideal gas.
At the end of the exhaust valve intake valve opens and the intake of fresh charge is done from CA 900-1020 marking the completion of one cycle of six stroke engine. Pressure is taken as ambient and the residual exhaust gas is considered in the control volume for the next cycle of the engine.

III. CALCULATED RESULTS

As detailed above, there are a number of parameters that are to be considered to obtain the results for the analysis. The initial conditions, assumptions and constraints are provided in the table 1 and engine details in the table 2. However, the assumption that these processes are instantaneous, combined with the assumption of no heat transfer, sets an upper bound on the potential extra work of this engine cycle modification. Finally, the constraints are applied to the engine model.

Table 1
Initial conditions, assumptions, and constraints

Initial conditions	
Initial temperature	460 K
Initial pressure	1 bar
Fuel composition	Iso-octane (C_8H_{18}) Complete stoichiometric products of iso-octane (73.4% N_2, 12.5% CO_2, 14.1% H_2O)
Water injection temperature	298 K
Assumptions	
Fuel injection duration	Instantaneous at 180 CA
Fuel combustion	Instantaneous at 180 CA
Water injection duration	Instantaneous at 540 CA
Water vaporization	Instantaneous at 540 CA
Water and air mixing	Instantaneous at 540 CA
Constraints	
Pressure at 720 CA	≥ 1 bar
Temperature at 720 CA	≥ Dew point

Table 2
Engine geometry modelled

Bore (mm)	79.4
Stroke (mm)	111.2
Connecting rod length (mm)	233.4
Compression ratio	7.4

The constraints are applied for temperature and pressure at 720 CA is given to ensure that the in-cylinder pressure does not drop below ambient in order to effect the removal of the spent gases, and to ensure that liquid does not condense in the cylinder.

The pressure-volume diagram (figure 1) for fuel stroke shows that the pressure obtained from the adiabatic combustion of the fuel is 77 atm for the first cycle without considering the residual exhaust gas formation at the end of the cycle. The compression and expansion is considered to be isentropic. Heat rejection process and fuel combustion process is considered to be instantaneous and adiabatic. No mass is considered to be flowing out of the control volume during these four process.

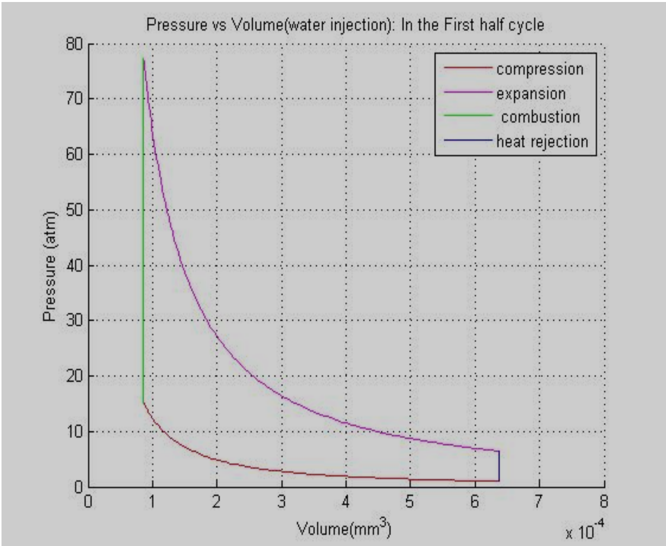

Fig 1: Pressure-volume diagram for the fuel using cycle in a six stroke engine

The pressure-theta diagram (figure 2) shows the variation of pressure with the crank angle for the fuel using cycle of engine. The constraints for the temperature (above dew point temperature) and pressure (above dew point pressure) of the fuel is applied here.

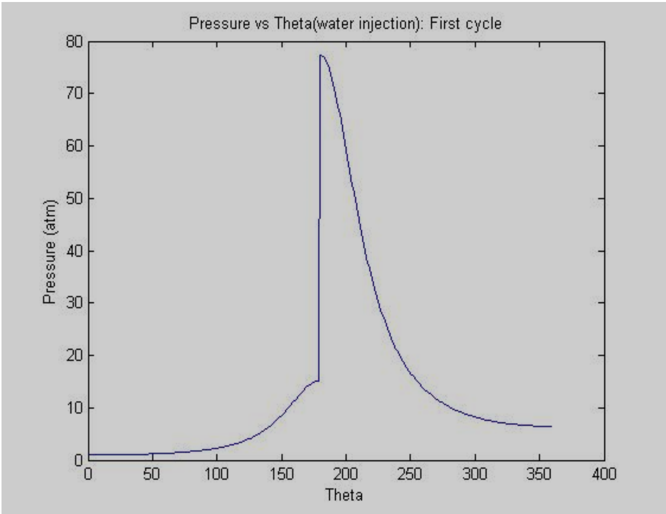

Fig 2: Pressure-theta diagram for fuel using cycle in a six stroke engine

The pressure-volume diagram (figure 3) for water injection cycle shows that the pressure obtained by the conversion of water to steam by using the heat from the recompression of the exhaust is 30 atm which is less than half of the pressure obtained from the power stroke of the fuel. S

The exhaust recompression angle is considered to be 450 CA and variation of properties with respect to the variation in the CA (crank angle) of early exhaust valve closing for the recompression of exhaust is yet to be studied. As the assumption taken, the variation of mass of exhaust remaining will be proportional to the CA, it is considered that half of the exhaust will remain in the engine for recompression if exhaust recompression is done up to 450 crank angle (CA).

The difference in the peak pressure at the end of the power stroke obtained from the iso-octane fuel and water leads to a major problem of balancing of the engine. When the engine runs at two different pressures for two power strokes of the same cycle the vibrations of the engine may be large. So balancing can be a major problem for the fabrication and running of this type of six stroke engines.

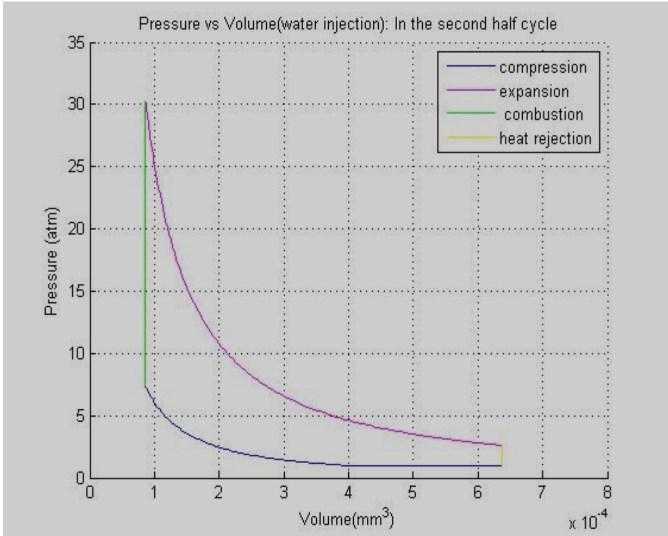

Fig 3: Pressure-volume diagram for water using cycle in a six stroke engine

The pressure-theta diagram (figure 4) for the water injection cycle shows the variation of pressure against different crank angle for the water injection stroke.

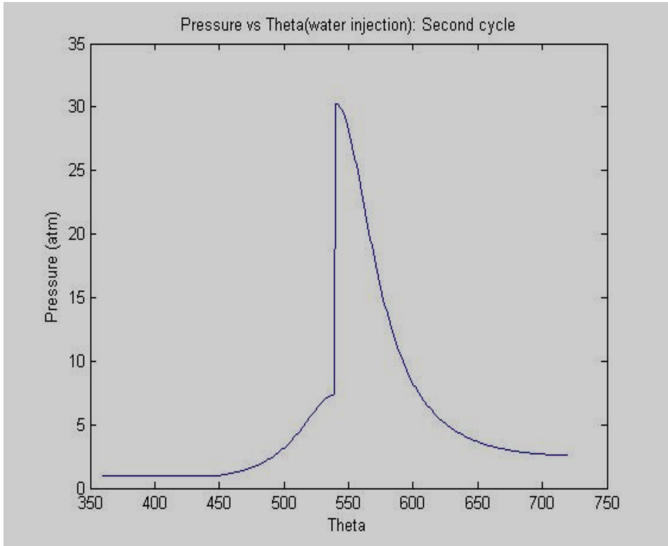

Fig 4: Pressure-theta diagram for water using stroke in a six stroke engine

Initial conditions given for the engine to obtain the desired output are shown in the table 3. Speed is considered as constant throughout the engine running cycle.

Table 3: Initial conditions given for six stroke engine

	Operating Conditions
Speed[rpm]	4000
Air/Fuel Ratio	59.5000
Intake manifold pressure (atm)	1
Exhaust manifold pressure(atm)	1
Fuel vapourisation model	neglected
Mixture temperature(K)	298

The air-fuel ratio was given as 12.5 (stoiometric ratio for iso-octane fuel) and the speed is given as 4000 rpm. Intake and exhaust manifold temperature was considered as 1 atmosphere. The mixture temperature in which fuel is given to the carburetor was taken as 298 K. Fuel vaporization model is neglected as shown in the table 3.

In all the cases of above shown figures, the intake and exhaust strokes were are not considered. The engine was running for a single cycle. So the residual exhaust gas, N_x was taken as zero.

As the engine continuously works for more than one cycle, the residual will build up in the engine decreasing the pressure obtained from the engine which is shown in the table 4.

Table 4: Results obtained from the simulation of a six stroke engine.

	Thermodynamic Results		
cycle	1	2	3
p1[atm]	1	1	1
p2[atm]	15.1974	13.9021	11.9422
p3[atm]	77.3835	56.9840	51.6595
p4[atm]	6.4599	4.7657	4.3182
p5[atm]	1	1	1
p6[atm]	1	1	1
p7[atm]	7.2891	7.1883	7.1829
p8[atm]	30.2888	34.4632	34.3288
p9[atm]	2.8144	3.3292	3.3164
p10[atm]	1	1	1
T1[K}]	298	422.3389	460.2665
T2[K]	612.0461	793.4827	742.8282
T3[K}]	3.1165e+03	3.2525e+03	3.2133e+03
T4[K]	1.9237e+03	2.0114e+03	1.9862e+03
T5[K}]	1.3390e+03	1.4866e+03	1.4959e+03
T6[K]	739.6219	821.4893	826.5307
T7[K}]	1.3886e+03	1.5141e+03	1.5219e+03
T8[K]	5.7700e+03	7.2590e+03	7.2733e+03
T9[K}]	3.9645e+03	5.1855e+03	5.1959e+03
T10[K]	3.3670e+03	4.3615e+03	4.3727e+03
kr	1.3596	1.3151	1.2392
kp	1.2410	1.2401	1.2404
kpw	1.1875	1.1681	1.1680
Nx / (Nm+Nx)	1.1804e-04	1.2915e-04	1.4038e-04
Net work done	1.1664	0.9643	0.9027
P[kW]	38.8805	32.1425	30.0890
p_imef[atm]	20.9074	17.2842	16.1800
thermal efficiency	81.1760	80.6048	80.6583

Table 4 shows the results obtained for a six stroke engine through the analysis of an ideal thermodynamic model of the engine.

The table above shows the running of six stroke engine in continuous three cycle till the same results are obtained in the consecutive cycles. Temperature and pressure at various salient points for consecutive cycles are shown in the table 4.. The residual exhaust gas N_x is taken into account. For the first cycle N_x as zero and then for the remaining cycle N_x is considered. Here the intake and exhaust of the engine is considered and the results are shown in the table.

Net work done is obtained from the pressure-volume diagram. From the net work done (W_{net}) the MEP is calculated using the formula:

MEP = (Net work done) / (volume displaced)

$$= (W_{12} - W_{34} + W_{67} + W_{89}) / \text{(volume displaced)}$$

The intake and exhaust strokes are considered. Pressure-volume and pressure-theta diagrams are plotted. The thermal efficiency obtained from the running of the six stroke engine is 80.65%. Mean effective pressure, work done, power and thermal efficiency is obtained from simulation of the petrol engine. The figure showing the pressure-theta diagram for the engine where the exhaust and intake strokes are considered. The pressure–theta diagram running of the engine from 0° to 900° is shown below.

The red line shows the cycle using fuel and blue line shows the cycle using water for obtaining power strokes in both pressure-volume diagram (figure 6) and pressure-theta (figure 5) diagram. The pressure generated for the fuel cycle is about 80 atmosphere and for water injection cycle it is about 30 atmosphere.

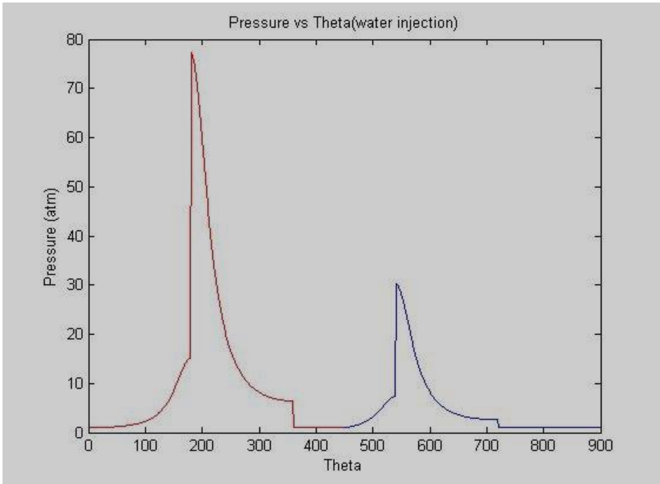

Fig 5: Pressure-theta diagram for six stroke engine using water injection

The shaded area shows the additional work obtained by water injection.

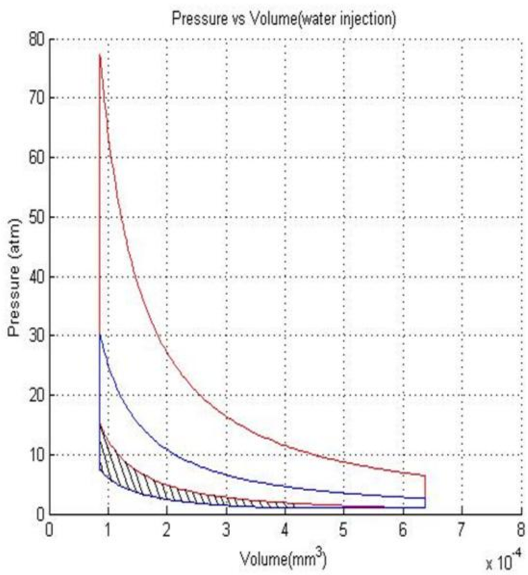

Fig 6: Pressure-theta diagram for six stroke engine using water injection

IV. Discussion

It is important to note that this is an idealized thermodynamic model and several assumptions were made that may not hold true in a real engine system.

1. For a real engine configuration the losses due to friction, balancing of the engine and various other losses account which are not considered in thermodynamic model of the engine. As for the same engine running cycle it produces

two peak pressure lighting a major problem of engine balancing that may have the potential for reducing the engine efficiency.

2. It is assumed that water injection, vaporization, and perfect homogeneity are instantaneous. In a real engine system, the vaporization and mixing processes will take a finite time. This could result in a lower power output.
3. Assumptions were made to prevent the in-cylinder temperature at the end of the expansion stroke for fuel expansion power stroke and water expansion power stroke from being lower than the dew point and the in-cylinder pressure from being less than 1 bar. These constraints are not realistic in a real engine in which higher temperatures are needed to prevent condensation throughout the exhaust, and to maintain proper function of exhaust.
4. The thermodynamic modelling does not consider the heat transfer between the combustion chamber walls and the cylinder contents. Some heat transfer will occur during the running of the engine cycle, but the thermodynamic modelling shows that there is a sufficient amount of heat in the exhaust gas for the steam cycle without extracting any heat from the walls directly.

The modified cycle proposed here adds two additional strokes that increase the work extracted per unit input of fuel energy. These additional strokes involve trapping and recompression of some of the exhaust from the fourth piston stroke, followed by a water injection and expansion of the resulting steam/exhaust mixture. The residual exhaust gas is trapped in the cylinder by closing the exhaust valve earlier than usual, i.e., well before top dead center (TDC). Energy from the trapped recompressed exhaust gases is transferred to the liquid water, causing it to vaporize and increase the pressure. This added pressure then produces more work, forms another expansion process. The steam-exhaust gas mixture is expelled to ambient pressure near the point of maximum expansion.

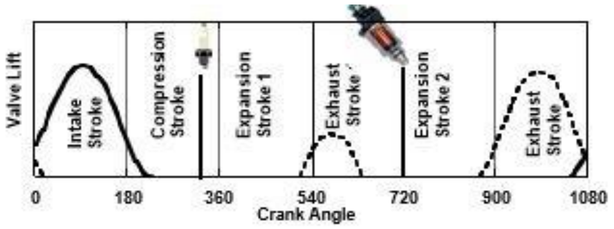

Fig 7: Valve lift diagram for six stroke engine[1]

To summarize in graphical form, figure 7 represents the valve lifts versus crank angle where the proposed exhaust recompression and water injection are explicitly shown.

Thus, the additional proposed power stroke produces more output work from the engine without any additional fuel, thereby increasing the fuel efficiency of the engine.

The value of this exhaust valve closing for maximum MEPsteam depends on the limiting conditions of 1 bar or the dew point temperature of the expansion gas/moisture mixture when the exhaust valve opens to eject the spent gas mixture out of the engine cylinder (control volume). The range of MEPsteam shown in figure is from 25 -30 atmosphere for one typical internal combustion engine running in Otto cycle having geometry as given in Table 2. Typical mean effective pressures of engines using iso-octane as the only fuel is only up to 80 atmosphere, thus this concept has the potential to show a very significant increase in engine efficiency and fuel economy.

Although the thermodynamic modelling presented here was performed for one set of engine conditions, similar increases in engine output are expected for a wide variety of engine geometries and operating conditions.

V. Conclusion

An ideal thermodynamics model of the exhaust gas recompression coupled with water injection at top dead center, and expansion was used to investigate a modification to recover waste heat energy from exhaust. The additional two strokes require substantial modifications in the exhaust valve operation, injection of water into the combustion chamber and the balancing of engine for the smooth running of the engine without vibration. This concept recovers waste heat energy from the exhaust which is normally discarded in current engine designs and converts the heat to useable power and work.

From the successful thermodynamic modelling and study, the six stroke engine as compared to four stroke engine shows a substantial increase in the work done, power, mean effective pressure and thermal efficiency than a conventional four stroke engine.

Form the study of various thermodynamic parameters, an ideal thermodynamic model for a particular engine geometry was generated. This concept has the potential for a substantial increase in fuel efficiency over existing conventional internal combustion engines without decreasing the power density significantly.

REFERENCES

[1]. James C. Conklin and James P. Szybist, A highly efficient six-stroke internal Combustion engine cycle with water injection for in-cylinder Exhaust heat Recovery. Energy, The International Journal for Mechanical Engineering, Volume 35, Issue 4, pp. 1658-1664, 2010.

[2]. V. Ganesan, Computer Simulation of Spark-ignition Engine Processes, Universities Press (India) Limited, 1996.

[3]. John B Heywood, Internal Combustion Engine Fundamentals, McGraw Hill Education (India) Edition, 2011.

[4]. Çengel Y A and Boles M A, Thermodynamics: an engineering approach. 5th ed., Boston: McGraw-Hill, 2005.

[5]. Pandiyarajan V., Pandian M. C., Malan E., Velraj R. Seenira R.V., "Experimental Investigation on Heat Recovery from Diesel Engine Exhaust Using Finned Shell and Tube Heat Exchanger and Thermal Storage System",Applied, Vol. 88, 2011.

[6]. Ravikumar, N., Ramakrishna, K., Sitaramaraju, A. V., Thermodynamic Analysis of Heat Recovery Steam Generator in Combined Cycle Power Plant, Thermal Science, 11. 4, pp. 143- 156, 2007.

[7]. Dyer LH. Internal combustion engine. United States patent 1,339,176; 1920.

[8]. Rohrbach H, Tamins R. Engine having alternate Internal-Combustion and fluid pressure power strokes. United States patent 2,671,311, 1954.

[9]. Tibbs RC. Six cycle combustion and fluid vaporization engine. United States patent 3,964,263; 1976.

[10]. Kellogg-Smith O. Internal combustion and steam engine. United States patent 4.143, 518; 1979.

[11]. Larsen G J. Engine with a six-stroke cycle, variable compression ratio, and constant stroke. United States patent 4,736,715; 1988.

[12]. Prater DM. Multiple stroke engine having fuel and vapor charges. United States patent 6,253,745; 2001.
[13]. Crower B. Method and apparatus for operating an internal combustion engine, United States patent application 20070022977; 2005.