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Evaluation of Performance Characteristics of Tri-Lobed Cam Mechanism with 8° and 9° Taper Turn in an IC Engine

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Abstract

Customarily, the crank shaft angle has been a fixed function of the intake and exhaust valve opening/lift in IC engines. Substantial fuel efficiency enhancements may be made, nevertheless, if these values are controlled by variable valve timing (VVT) systems that can be adjusted in response to the crankshaft's angular displacement. The main issues with VVT cam systems, however, stem from the noise and wear brought on by high contact velocities during valve opening and closure. The valve actuators for these kinds of applications now predominantly rely on resonant spring arrangements to provide the necessary valve dynamics, which is another significant issue. In the absence of a completely flexible valve actuation system, this results in a fixed amplitude of the valve trajectory and merely permits variable valve timing. In the current study, an effort is made to develop a novel "TRI-LOBED-CAM" mechanism that may be utilized in conjunction with an established cam operating mechanism to axially move the camshaft via a tiny displacement in response to the engine's operating circumstances, namely, maximum valve displacement at high loads and high engine speeds, minimum valve displacement at lean loads and low engine speeds, and medium valve displacement at intermediate loads. The article outlines an alternative technique for assessing the performance of newly created Tri-lobed cam mechanisms with 8° and 9° taper turns on internal combustion engines. This is used to compare the performance of newly created and commercially available IC engines and determine the effectiveness of the technical solution that is being suggested.

Keywords: *IC Engine; Performance Characteristics; Tri-Lobed Mechanism; CAM; Taper Turn.*

1.0 Introduction

In the recent years owing to the stringent environmental standards, enhancement in efficiency and lessening in the exhaust gas discharges have turned into the two noteworthy difficulties in the automotive segment. To satisfy the prerequisites of these directions, the IC engines have encountered noteworthy changes during the previous decades. Implementing the variable valve mechanisms in IC engines is one of the few approaches to enhance the performance of the internal combustion engines. For quite a long while, the modification of IC engine camshaft design has

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been the essential answer for valve incitation and timing. In the cam operated IC engine valves open and close with the fixed lift and timing. In spite of the fact that these mechanisms offer a dependable and precise valve activity, the IC engine can't be operated efficiently over a wide range of engine operations. The introduction of cutting edge innovations for air charging admission still accounts for one of the more promising methodologies to enhance IC engine performance, efficiency and, to some degree, emission discharges [Mianzo. 2007, Clearly et.al. 2007., Ghauri et.al. 2000., Bohac et.al. 2004., Anderson et.al. 1998., Salber et.al. 2002.].

In the last two decades, a few sorts of variable valve activation techniques have been executed in IC engines in various structures, extending from mechanical 2-step cam phase, consistently variable cam phase, modification of cam design, cam profile exchanging components. By these strategies, some adaptability for both timing and valve lift has been acquired. The genuine level of advantages accomplishment is unmistakably constrained by the genuine abilities of every particular kind of valve train system, be that as it may be, in general, this adaptability gives a considerable contribution in the exertion of overcoming the established internal combustion engine tuning trade-offs which includes power versus efficiency, power versus torque.

In the recent years, tremendous pressure has been applied on automotive industries for enhancement of IC engines efficiency to lessen CO_2 discharges constraining IC engine advancement towards the selection of new innovations and design criteria able to decrease fuel utilization. In this situation, the higher cost of IC engine components accounts for more decent than it was in the current past, a viable possibility could be discussed by the selection of completely adaptable variable valve actuation (VVA) systems.

Conventional IC engines are composed of various mechanically actuated valves. The position of the crankshaft and the profile of the camshaft decide the valve occasions (the timing of the opening and shutting of the admission and exhaust valves). Since, regular IC engines have valve movement that is mechanically reliant on the crankshaft position; the valve movement is fixed for every single operating condition. The perfect timing of the valve occasions, be that as it may, contrasts extraordinarily between various working conditions. This speaks to a noteworthy of modifying design of an IC engine [Fukuo. et.al. 1997., Brüstle and Schwarzenthal. 2001., Sellnau and Rask. 2003., Chaudhari et.al. 2014.].

Fuel saving in internal combustion engines facilitates to keep the environment, pollution and global weather variations in control. Multi valve fuel injection technology was extensively investigated for internal combustion engine design beside with variable valve timing method to improve engine productivity, power or torque. Generally, valves stimulate the breathing of IC engines. The timing of breathing plays crucial role in deciding performance of IC engines. Especially, total duration of air intake and exhaust is largely controlled by the shape and phase angle of cams present in the IC engine. Optimization of breathing time is an important aspect and IC engines are needed to be operated at variable valve timing at different engine speed. As there is increase in engine revolutions the total time required for intake and exhaust stroke reduces which causes the possibility of restricting fresh charge inside the engine cylinder completely and scavenging of exhaust gases may not take place entirely from the previous cycle.

Consequently, the finest solution to issue is to keep open the inlet valves in advance and exhaust valves have to be closed later. In the other words, the total duration of overlapping between exhaust and intake needs to be amplified as there is increase in engine revolution. Conventional IC engine design generally utilizes cam shaft having fixed or variable cam profile to accomplish a practical negotiation between idle speed stability, fuel saving, and torque performance. Individual control of valve timing leads to considerable improvement in the engine performance.

Kinematics profiles for IC engine valves, such as those showing valve position vs time, valve speed versus time, etc., have a defined form and are created with consideration for the crankshaft position of the engine. We claim that the engine valves are not controlled from the perspective of control frameworks. Instead, if we were able to independently alter the span, stage, and lift of the valves, we would observe a marked difference in discharges, effectiveness, highest power, and efficiency. Even though it is simple, the engine's mechanical design compromises its efficiency and highest level of power. However, any factor valve incitation system must be able to offer a variety of valve profiles without compromising the fundamental characteristics of a conventional IC engine valve profile. The productivity and greatest power may be obtained, and discharges can be managed, by using the variable valve lift component [Chang. et. .al. 2002., Pischger. et.al. 2000., Pischinger et.al. 1999.].

Engine valve relocations in typical IC engines are fixed in relation to the crankshaft position. The condition of the cams, which are mounted on a belt-driven camshaft and used to drive the valves, is determined by balancing the demands for engine speed, power, and torque with those for fuel efficiency of the vehicle. A very productive engine is now possible thanks to this development, but only under specific circumstances. Instead, significant improvements in efficiency - up to 20% - as well as improvements to torque, yield power, and discharges may be made if the engine valves are engaged as a variable capacity of crankshaft point [Chang. 2001., Levin and Schlecter. 1996., Barkan and Dresner. 1989].

Electromechanical camless valve trains are used in solenoid-controlled frameworks (EMCVs). In EMCV, a spring structure keeps the valve in the centre position. An armature positioned on the valve is then drawn into either the open position or the closed position by stimulating two loops once again. The relationship between power, position, and current becomes nonlinear as the armature moves toward either end. It is really difficult to control the sitting pace because of this. But recently, remarkable improvements have been achieved in the exhibiting and controlling of this device. Regardless, the seating speed is effectively controlled while taking temperature variations and valve wear into account [Schernus. et. al. 2002.].

2.0 Experimental Details

2.1 Modified Cam shaft and Rocker Configuration

In four-stroke cycle engines the valve timing is controlled by the camshaft. It can be shifted by altering the camshaft, or it can be fluctuated during engine task by variable valve timing. It is influenced by the change of the valve movement, and especially by the tapper freedom. Every valve that has to be operated receives a lowered cam from the modified camshaft. Through the use of timing gears, the crankshaft drives the camshaft. The profile of the adjusted camshaft is overhauled with decreased (slanting) surface is appeared in the Fig.1. The changed rocker having point contact with the particular cam and it is made of cast iron or forged steel. Figs.2 and 3 depict minimum, medium and maximum valve displacement positions on the cam profile. Fig 4 shows the profile of modified rocker that has been used in conjunction with the modified camshaft with a tapered surface that results in variable valve displacement as the modified cam shaft slides axially along the splines provided on the shaft. Fig.5 provides the 3D model of the cam shaft indicating minimum, medium and maximum position. Figs.6 and 7 show actual photograph of the cam shaft with modified with 8º and 9º taper turn. Fig.8 shows magnified view of the cam shaft depicting minimum, medium and maximum valve displacement positions. Fig.9 shows the actual photograph of the rocker placed at different position of the modified cam.

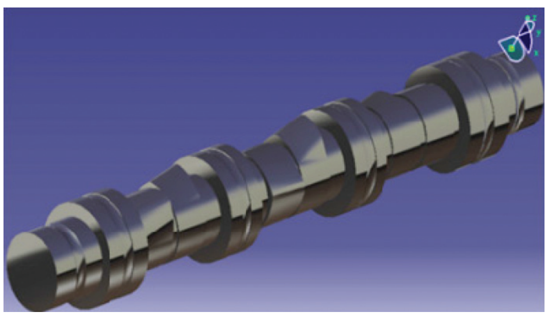


Figure 1: Modified cam Shaft

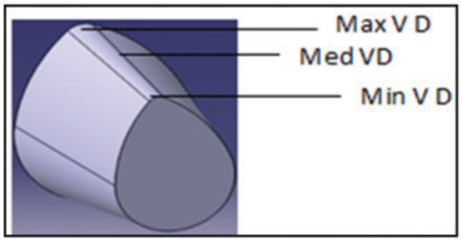


Figure 2: Valve displacement positions on cam profile

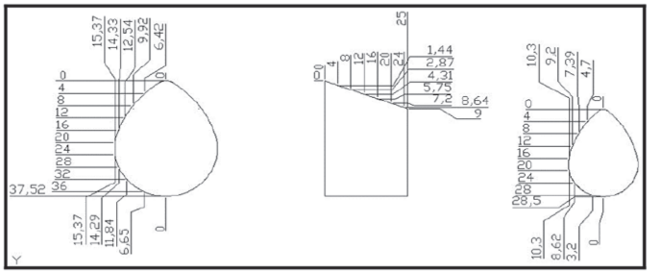


Figure 3: Angular positions of cam profile

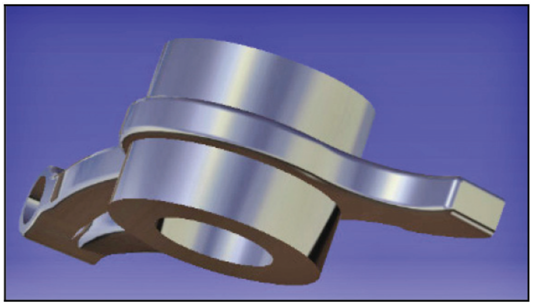


Figure 4: Modified Rocker

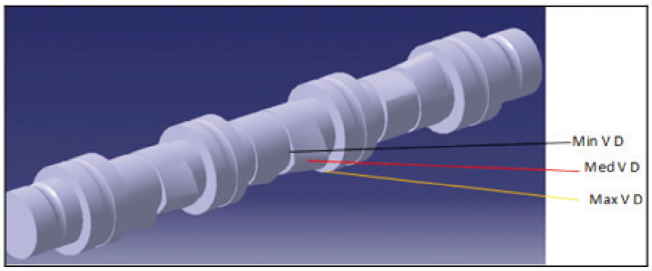


Figure 5: Modified cam shaft showing minimum, medium and maximum valve dispalcement psoitions

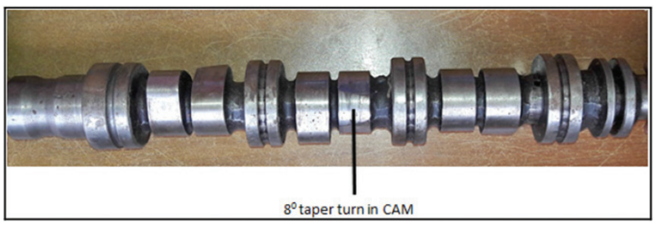


Figure 6: Modified shaft with 8º Taper turn in cam shaft

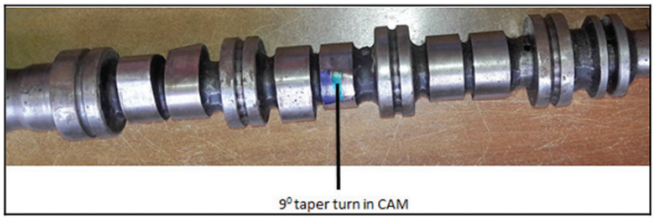


Figure 7: Modified shaft with 9^o Taper turn in cam shaft

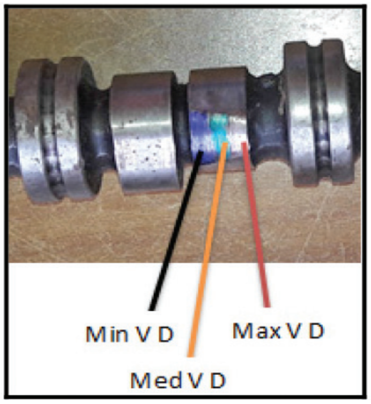


Figure 8 Magnified view of various diplacement positions

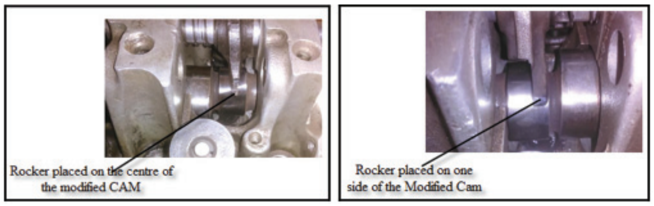


Figure 9: Rocker placed at different locations of tri lobed cam

3.0 Results and Discussions

3.1 Pressure-Crank Angle and Log P-Log V for 8^0 Taper Turn in Cam for Different Loading Conditions

For the 8^o taper turned cam in and IC engine, pressure-crank angle and Log P–Log V plots were experimentally derived for the different loading conditions like no-loading, quarter loading, half loading, 3/4th loading and full loading. Upon several investigations the following results were adopted to evaluate the change in the taper angle in cam for the different parameters like pressure, Log P, Log V etc., and are discussed as below.

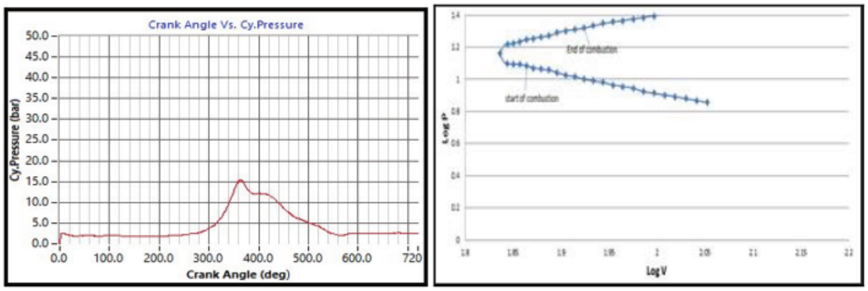
Figure 10(a) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to no load at medium valve displacement position of cam. Combustion pressure peaks at approximately 20 after TDC position of piston and achieves a maximum value of 15.8 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V diagram of Figure 10(a). The begining of combustion and finishing of combustion are approximately 341 degree of crank angle location and 378 degree of crank angle location and the overall combustion duration is approximately 370 of crank rotation for the set of operating condition.

Figure 10(b) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to Quarter load at medium valve displacement position of CAM. Combustion pressure peaks at approximately 30 after TDC position of piston and achieves a maximum value of 16 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Figure 10(b). The begin of combustion and end of combustion are approximately 343 degree of crank angle location and 382 degree of crank angle location and the overall combustion duration is approximately 390 of crank rotation for the set of operating condition.

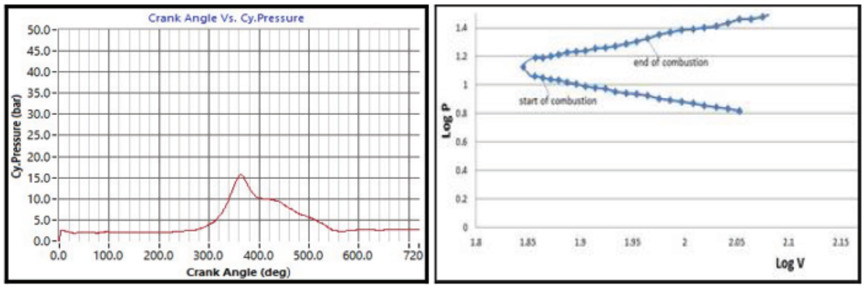
Figure 10 (c) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to half load at medium valve displacement position of cam. Combustion pressure peaks at approximately 250 after TDC position of piston and achieves a maximum value of 25.8 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V diagram of Figure 10(c). The begining of combustion and ending of combustion are approximately 348 degree of crank angle location and 392 degree of crank angle location and the overall combustion duration is approximately 440 degree of crank rotation for the set of operating condition.

Figure 10(d) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to full load at medium valve displacement position of cam. Combustion pressure peaks at approximately 220 after TDC position of piston and achieves a maximum value of 33.8 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V diagram of Figure 10(d). The begining of combustion and ending of combustion are approximately 350 degree of crank angle location and 397 degree of crank angle location and the overall combustion duration is approximately 470 of crank rotation for the set of operating condition.

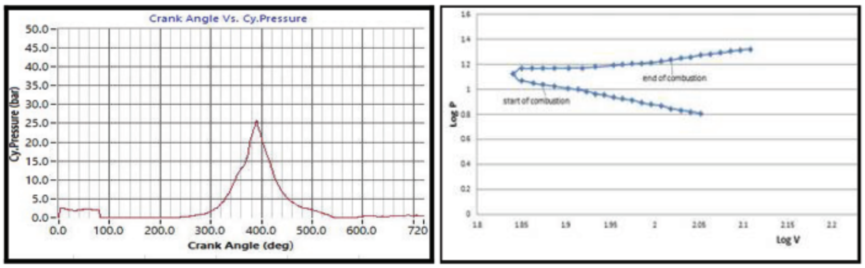
Figure 10(e) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to full load at medium valve



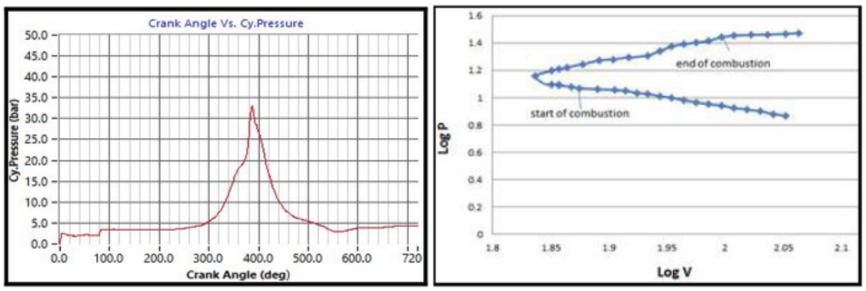
(a) Under no load condition



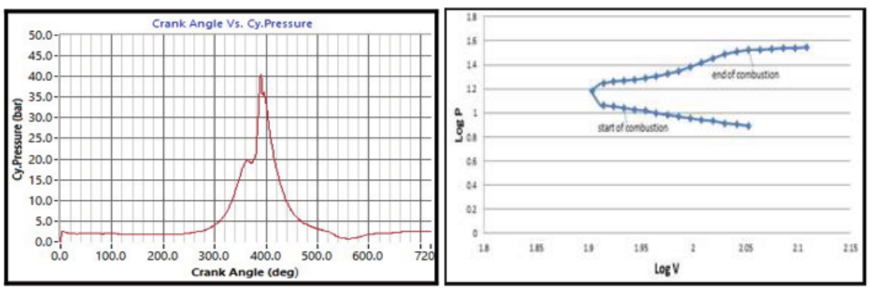
(b) Under quarter load condition



(c) Under half load condition



(d) Under 3/4th load condition



(e) Under full load condition

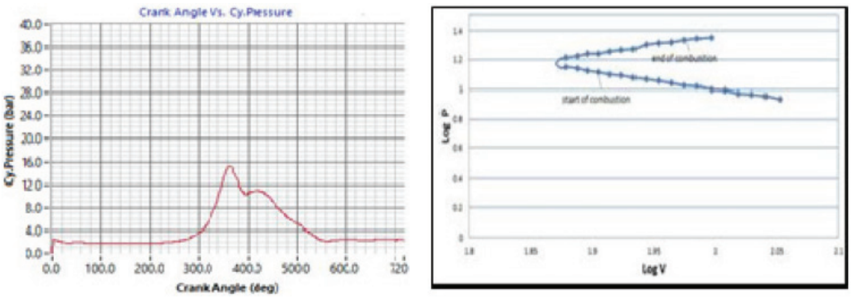
Figure 10: Pressure-Crank angle and Log P-Log V for 8^o Taper turn in cam for different loading conditions

displacement position of cam. Combustion pressure peaks at approximately 220 after TDC position of piston and achieves a maximum value of 40 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V diagram of Figure 10(e). The begining of combustion and ending of combustion are approximately 350 degree of crank angle location and 398 degree of crank angle location and the overall combustion duration is approximately 480 degree of crank rotation for the set of operating condition.

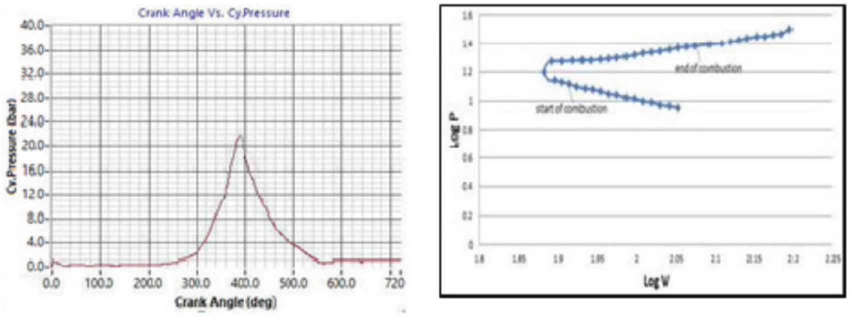
3.2 Pressure-Crank Angle and Log P-Log V for 90 Taper Turn in Cam for Different Loading Conditions

Similarly, for the 9^o taper turned cam in and IC engine, pressure-crank angle and Log P–Log V plots were experimentally derived for the different loading conditions like no-loading, quarter loading, half loading, 3/4th loading and full loading. Upon several investigations the following results were adopted to evaluate the change in the taper angle in cam for the different parameters like pressure, Log P, Log V etc., and are discussed as below.

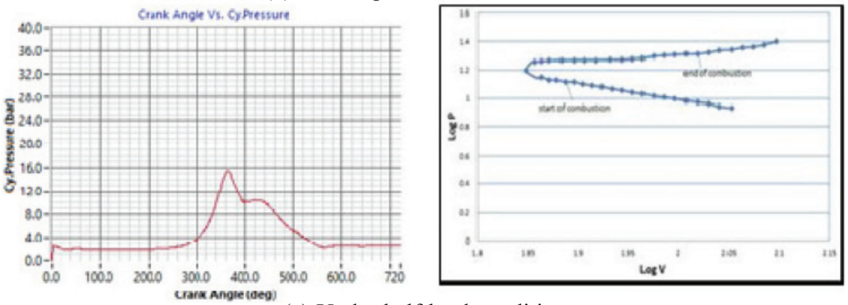
Figure 11(a) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to no load at minimum valve displacement position of cam. Combustion pressure peaks at approximately 20 after TDC position of piston and achieves a maximum value of 15.1 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V graph of Figure 11(a). The begining of combustion and ending of combustion are approximately 342 degree of crank angle location and 379 degree of crank angle location and the overall combustion



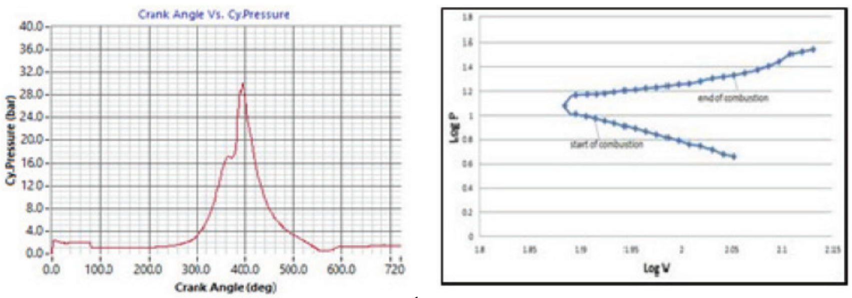
(a) Under no load condition



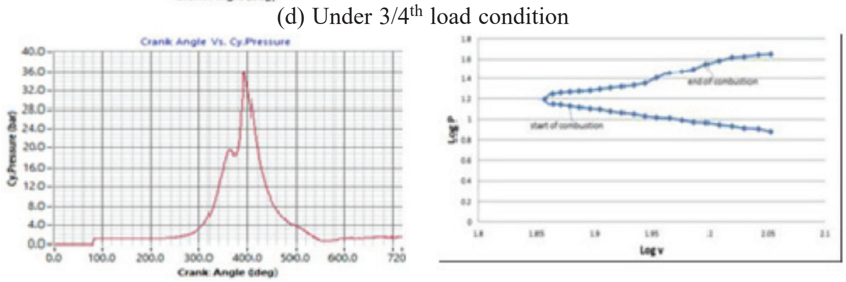
(b) Under quarter load condition



(c) Under half load condition



(d) Under 3/4th load condition



(e) Under full load condition

Figure 11: Pressure-Crank angle and Log P-Log V for 90 Taper turn in cam for different loading conditions

duration is approximately 370 of crank rotation for the set of operating condition.

Figure 11(b) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to quarter load at minimum valve displacement position of cam. Combustion pressure peaks at approximately 40 after TDC position of piston and achieves a maximum value of 15.6 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V graph of Figure 11 (b). The begining of combustion and ending of combustion are approximately 343 degree of crank angle location and 383 degree of crank angle location and the overall combustion duration is approximately 400 of crank rotation for the set of operating condition.

Figure 11(c) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to half load at minimum valve displacement position of cam. Combustion pressure peaks at approximately 230 after TDC position of piston and achieves a maximum value of 22 bars.

The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V graph of Figure 11(c). The begining of combustion and ending of combustion are approximately 348 degree of crank angle location and 393 degree of crank angle location and the overall combustion duration is approximately 450 of crank rotation for the set of operating condition.

Figure 11(d) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to 3/4th load at minimum valve displacement position of cam. Combustion pressure peaks at approximately 230 after TDC position of piston and achieves a maximum value of 30 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V graph of Figure 11(d). The begining of combustion and ending of combustion are approximately 347 degree of crank angle location and 395 degree of crank angle location and the overall combustion duration is approximately 480 degree of crank rotation for the set of operating condition.

Figure 11(e) shows a typical pressure crank angle (P-θ) and Log P–Log V diagram for the engine with modified camshaft at operating conditions corresponding to full load at minimum valve displacement position of cam. Combustion pressure peaks at approximately 240 after TDC position of piston and achieves a maximum value of 37 bars. The curves smooth without any tendency towards possibility of knocking and the combustion is nearly complete with a Mass Fraction Burnt ratio of approximately 98% as shown in Log P–Log V graph of Figure 11(e). The begining of combustion and ending of combustion are approximately 351 degree of crank angle location and 400 degree of crank angle location and the overall combustion duration is approximately 490 of crank rotation for the set of operating condition.

4.0 Conclusions

Current study accentuated the significance, design, influence of various parameters of modified variable cam actuator mechanism on performance characteristics of IC engine. Well remembering this, further work on variable valve actuator mechanism may be carried out as specified below, an optimum design needs to be fixed for commercialization by developing the mechanism and control systems for various types of IC engines. In response to strict environmental norms, it is likely that the internal combustion engine will remain as major one and it is expected that variable valve actuation system will have an undeniably vital part in enhancing the performance from IC engines. More work should be done to assess the environmental effect of design alterations under different working states of IC engines. In the present study, the experiment of an IC engine valve actuation system has received significant attention; nevertheless, the specifications and regulating methods of the proposed system as they apply to different types of engine valve applications are not covered. It is recommended that a thorough numerical analysis be done for multi-cylinder engines. The most effective design of the proposed system for multi-cylinder valve engines, as well as the system's repeatability, accuracy, and robustness, will be evaluated. The variable valve actuator mechanism and its corresponding performance characteristics may be recorded and analysed at various angles of taper turn and displacement position. Impact of modified variable valve actuator cam mechanism on overall power consumption and system performance needs to be explored.

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