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Experimental Measurement of Frictional Torque in End Pivoted Roller Finger Follower Valve Train

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Abstract: *Friction reduction is an effective way to achieve better fuel economy while maintaining reduced exhaust emissions in engines. The friction characteristics in engine valve train vary directly with the operating temperature, lubricant pressure, lubricant chemistry, camshaft speed and type of engine. Substantial experimental work has been carried out on direct acting bucket tappet valve train to measure friction between cam and tappet. However, little to no experimental work has been reported to investigate friction in an end-pivoted roller finger follower valve train. In this research work, end-pivoted roller finger follower valve train has been instrumented for the first time to experimentally measure the friction drive torque. Increase in friction at cam/roller interface with rise in oil inlet temperature was observed whereas it decreased considerably with increase in camshaft operating speed. The results of measured friction under actual engine operating conditions are presented in this manuscript in detail.*

Index Terms: *friction, drive torque, torque tube, end pivoted roller finger follower.*

1. Introduction

The bucket tappet is the most common type of valve train configuration being employed by the automotive industry. In recent years however, the use of roller finger follower valve trains configuration (Figure 1) has become common in modern passenger cars. This is due to its improved performance, better fuel economy and reduced friction of roller follower valve train.

At higher operating temperatures and lower engine speeds the frictional losses become more significant in engine valve train. The increased trend to reduce the sliding friction by using engine oils having lower viscosities has pushed the lubrication mode towards mixed/boundary region. Cam/roller interface operating in mixed and boundary lubrication regions increases more issues about the stability and reliability of the engine components.

Staron et al. [1] showed that more than 50% reduction in friction can be achieved with the help of roller type configuration instead of sliding contact. Sun and Rosenberg [2] also observed a similar trend and reported only 20% power loss by use of roller follower contact instead of flat tappets.

Some experimental work on friction was presented by Lee et al. [3, 4] on a heavy duty diesel engine having roller follower valvetrain configuration. However, these previous research does not show the complete picture as these experiments were performed on extensively simplified/modified test rigs. Moreover, the results of friction under real operating engine speeds and lubricant temperatures are also missing.

The friction of cam/roller pair is governed by numerous factors like viscosity of lubricant, lubricant temperature, contact loading, engine/camshaft speed, oil film thickness, lubricant formulation, surface roughness, lubricant pressure, etc. [6]. In this experimental research work, the friction at cam/roller interface has been experimentally

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investigated under actual engine operating conditions. A gasoline engine valve train Toyota INZK, as shown in figure 2 has been instrumented, for the very first time without making any modifications to the engine head to measure the drive torque.

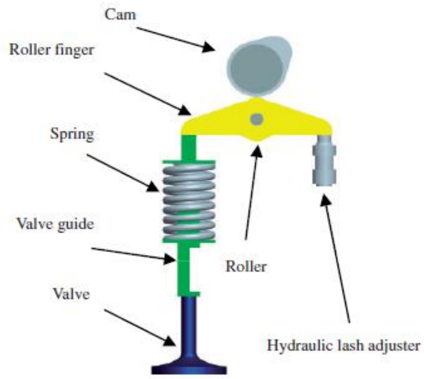

Fig. 1 Toyota 1NZK end pivoted roller follower valve train [5]

Fig. 2 Toyota INZK Engine Head

2. Test Rig Setup

The engine has twin cam system with 16 valves employing end pivoted roller finger follower valve train configuration with hydraulic lash adjuster. The valvetrain assembly is lubricated with the help of oil spray channels which are available in the engine head cover.

The engine test rig comprised of a three phase induction motor, an oil pump, oil heating and cooling unit, electrical controllers and an optical encoder. To drive the camshaft of the said engine head, it was attached with the induction motor with the help of a bellow coupling. The purpose of using this specific coupling was to transmit the torque without any backlash and to provide compensation for radial, angular and axial misalignments. The speed of the camshaft can be adjusted through a variable speed drive controller. The controller was setup in a vector mode close loop feedback mechanism with the help of an optical encoder so that 100 % standstill torque could be ensured. A high resolution push pull type optical encoder was used for this purpose which was mounted at one end of the induction motor. Camshaft angular position and speed were acquired through optical encoder that generated incremental pulses per half degree of cam rotation along with the index plus at 180 degree in a cycle.

An oil pump, driven by an electric motor, was provided in the test rig to circulate oil through the engine. The oil pump was controlled with the help of an electrical controller and a piezo-resistive pressure transducer that provides feedback to electrical controller so that pressure inside engine head could be retained at elevated temperature. The temperature of the oil was controlled with a plate type heat exchanger installed in line, at the end of which an external refrigerated and heating circulator was connected. A K-type thermocouple, installed in line, was used to monitor the current temperature of oil. Complete engine test rig can be seen in figure 3.

Fig. 3 Engine Valve Train Test Rig

The basic objective for the research was to investigate the instantaneous friction torque at the cam/roller interface in an end pivoted roller follower valvetrain. So a torque tube with specific stiffness & sensitivity was employed with the camshaft assembly. Special flange adopters were manufactured and the torque tube was connected to the camshaft with these flange adopters to avoid modification inside engine head. In order to measure the torsion or the drive torque, two-element 90 degree rosette strain gauge was installed at either sides of torque tube and it was attached in a full bride circuit configuration. A Michigan Scientific B series slip ring was used on the rotating shaft to connect the torque tube gauge to the data acquisition hardware. The slip ring assembly along with the torque tube coupled with the camshaft can be seen in Figure 4.

Fig. 4 Slip ring assembly attached with torque tube and camshaft

Before experimentation, the torque tube was mounted on a custom designed calibration bench, as shown in Figure 5. The self-aligned frictionless bearings on the calibration bench made it possible to align and hold the torque tube during calibration. Certain load was applied clock wise and then counter clock wise and the output data from torque tube was logged. A linear relationship between applied torque and output strain was confirmed by repeating the tests.

Fig. 5. Torque tube calibration bench

3. Instrumentation and Data Acquisition System

A high speed advanced data acquisition system was employed using National Instruments hardware and LabVIEW software. This data acquisition system not only monitored and log the torsional output data but it was also used to control and monitor other test rig parameters i.e. motor speed, oil pressure, lubricant temperature etc.

In order to control engine test rig parameters, NI Compact DAQ-9174, 4-slot USB chassis was used. NI 9401 digital I/O module, NI 9215 analog input module, NI 9219 universal analog input module and NI 9263 analog output module were attached to the cDAQ-9174 chassis. With the help of these modules, the engine camshaft speed, lubricant pressure and lubricant temperature were controlled and monitored, as each motor has its own variable electric drive (controller) that were set to remote access mode and interfaced with the said data acquisition cards.

For torque tube output data acquisition, a SCXI-1000 chassis was used that was connected to SCXI-1314, SCXI-1520 and SCXI-1600 modules. The SCXI-1600 digital and analogue I/O channels were used to acquire the signal from the optical encoder. Thus it helped in determining of the camshaft angular speed and position with the respect to time. The incremental pulses from the optical encoder also acted as an external sampling clock for the strain signal. The SCXI-1314 and SCXI-1520 was employed with the SCXI-1000 chassis for the torsional measurement of torque tube. The SCXI-1520 is a universal strain gage input module that has a built in amplifier and automatic null compensation system. While SCXI-1314 is a front-mounting terminal block for the SCXI-1520 module. An internal voltage was supplied to the full bridge strain gauge setup for excitation and mV/V was acquired through the SCXI cards which was then multiplied by the calibration factor obtained after calibration of the said torque tube. All the data was acquired using simultaneous sampling. Butterworth filter with a low pass filter was used in order to filter the noise from the signal. All the data acquisition was completed in a custom designed software built in LabVIEW.

4. Experimental Methodology

Experimentation was conducted at cam shaft speeds of 300 rpm, 600 rpm and 900 rpm at oil inlet temperatures of 25°C and 95°C. The reasons of choosing these testing conditions were to measure the friction under actual engine operating temperatures while covering a range of engine speeds. Group IV base oil having a viscosity of 4 centistokes was used to conduct these tests. Before the start of the each test, lubricant was heated to the test temperatures, it was circulated through engine head via oil pump while maintaining a constant lubricant pressure of 2 bar. Lubricant temperature was monitored and maintained throughout the tests. After an hour, motor driving the camshaft was switched on and was run for another hour to make sure that required engine operating conditions are achieved at each of the given test temperatures. The data acquired from the torque tube depicted a characteristic waveform which consisted of geometrical torque component and frictional torque component. By taking mean of the entire waveform obtained from torque tube canceled out the geometrical torque and left behind the mean frictional torque.

5. Repeatability Tests

Before experimentation, the repeatability of the results was ensured. The plot shown in figure 6 below shows the repeatability results by running the setup three times at 25°C. The maximum variation amongst the test was 0.012. Figure 7 shows the repeatability results by running the setup three times at 95°C. The maximum variation turned out to be 0.01.

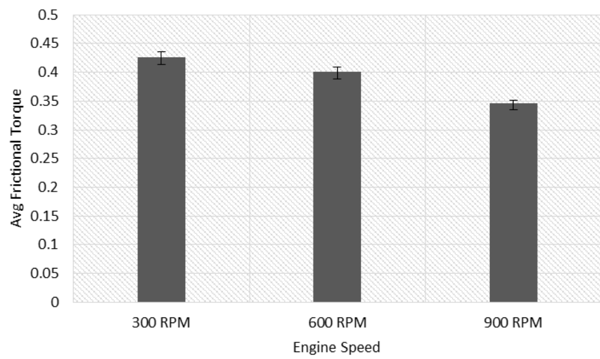

Fig. 6 Repeatability test at 25°C

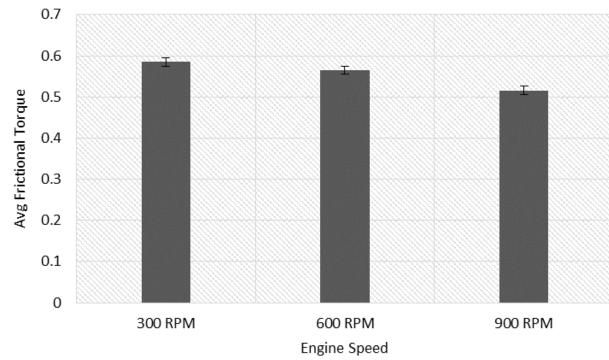

Fig. 7 Repeatability test at 95°C

6. Results and Discussion

Figures 8 and 9 show the graphs of the instantaneous drive torque recorded at 300, 600 and 900 RPM at 25°C and 95°C for the cam active cycle. It is evident from figures 8 and 9 that the instantaneous drive torque increases with increases in temperature and reduces with increase in cam shaft speed.

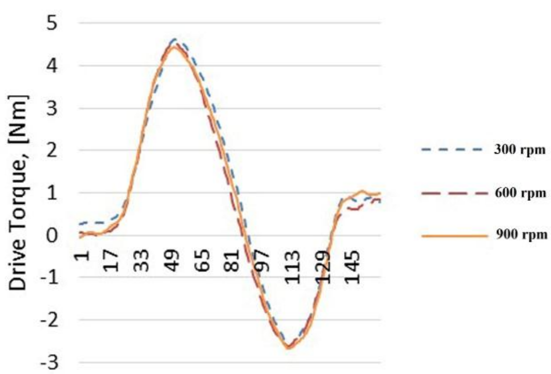

Fig. 8 Instantaneous Drive Torque at 25°C

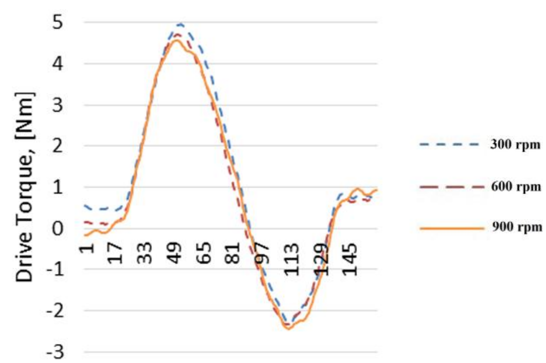

Fig. 9 Instantaneous Drive Torque at 95°C

At lower frequencies, the load on cam nose is strongly influenced by the valve spring compression which results in elevated friction around the cam nose area. At higher rotational frequencies of the camshaft, the mean frictional torque reduces. This is because at higher frequencies, the lubrication conditions improves considerably as a result of the reduction in load at the cam nose and the increase in the amount of lubricant entrained between cam and roller interface.

With increase in the lubricant temperature, the friction between at the cam and roller increases due to reduction in the lubricant viscosity which pushes the lubricant regime more towards the boundary lubrication. The oil film thickness between the cam and roller increases due to an increase in the entrainment velocities which shift the lubrication regime towards hydrodynamic lubrication; this reduces the friction at higher RPM at elevated temperature.

The data for each camshaft speed was recorded at 25°C and 95°C respectively. For every speed, fifty consecutive camshaft rotations were averaged to minimize errors. The averaged frictional torque results are shown figures 10 and 11.

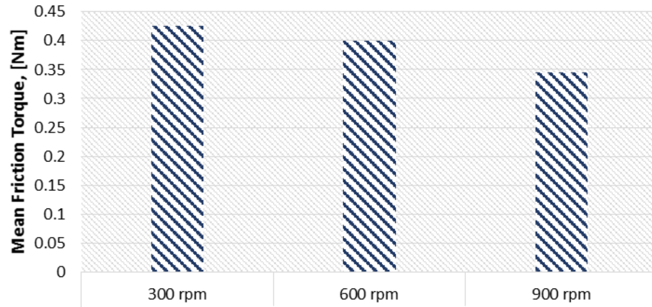

Fig. 10 Mean Friction Torque at Lubricant Temperature of 25^{o}C

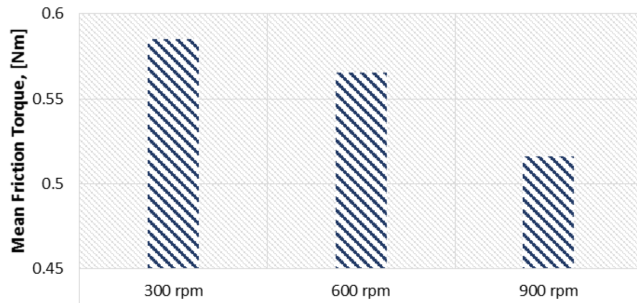

Fig. 11 Mean Friction Torque at Lubricant Temperature of 95^{o}C

The results obtained at 25^{o}C show that the mean friction torque has values of 0.426 Nm at 300 RPM, 0.400 Nm at 600 RPM and 0.345 Nm at 900 RPM. There is 6.1% reduction in friction as the RPM is doubled from 300 rpm to 600 rpm and the friction reduces further by 19.1% as the rpm is increased from 300 rpm to 900 rpm. The decrease in friction between the camshaft and roller is due to the increase in the oil film thickness at the cam and roller contact which results due to the increase in the lubricant entrainment velocities at higher camshaft operating speeds.

At 95^{o}C the mean friction torque has values of; 0.585 Nm at 300 RPM, 0.565 Nm at 600 RPM and 0.516 Nm at 900 RPM. There is 3.4% reduction in friction as the RPM is doubled from 300 RPM to 600 RPM and the friction reduces further by 11.8% as the RPM is tripled from 300 RPM to 900 RPM. These results show a similar reduction in friction with increase in camshaft speeds.

Figure 12 shows the comparison between the average friction torque at 300, 600 and 900 RPM as the temperature is raised by 70^{o}C from 25^{o}C to 95^{o}C.

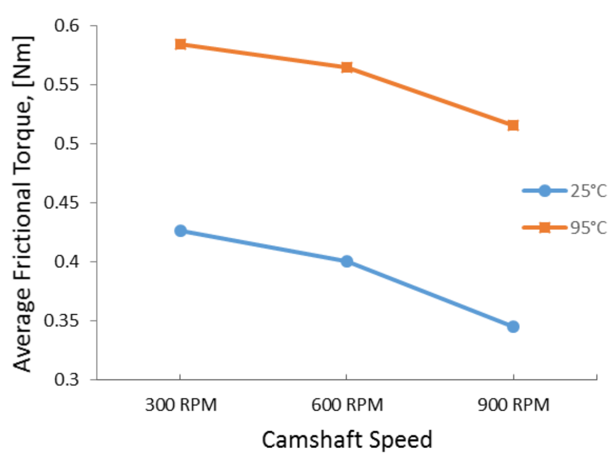

Fig. 12 Mean Friction Torque at Lubricant Temperature of 25^{o}C and 95^{o}C

Comparing the results of the mean friction torque represented in figure 10 show 37.3% increase in friction at 300 RPM, 41.3% increase in friction at 600 RPM and 49.6% increase in friction at 900 RPM, as the temperature is increased from 25^{o}C to 95^{o}C, figure 11. These results show a substantial increase in the friction with increase in lubricant temperature. The increase in temperature causes the oil viscosity to decrease substantially which results in the decrease in oil film thickness between the cam and the roller interface. This decreased oil film thickness brings the parts closer together and engine starts operating in mixed to boundary lubrication regime which results in the increase in friction under the influence of asperities interactions.

It can also observed from figure 12 that as the temperature increases from 25^{o}C to 95^{o}C, the difference between the measured frictions is increasing at higher frequencies. This is because at higher frequencies the lubrication conditions improves due to the reduction in load at the cam nose and the increment in amount of lubricant entrained between cam and roller interfaces. At lower frequencies the load on cam nose is strongly influenced by the valve spring compression which results in elevated friction around the cam nose. This influence diminishes at higher speed due to the inertial effects on the valve spring.

7. Conclusion

The research was conducted to experimentally measure the friction torque in Toyota 1NZK engine head having roller finger follower valvetrain configuration. Engine head was instrumented successfully and experimentation was carried out at different lubricant temperatures and engine speeds.

The results revealed that friction torque decreases with the increase in engine speed due to reduced load at cam nose resulting into improved lubrication conditions at cam/roller contact. However an increment in the valve train frictional torque was observed with the rise in lubricant temperature as the lubricant viscosity reduces and consequently film thickness decreases.

The developed experimental setup and technique can prove to highly beneficial to further improve the tribological performance of engine valve train. It can be used to study the performance of a roller follower valve train under different lubrication conditions in detail. Engine lubricants having various additives and different viscosities can be studied by using this setup and technique. The effects of various surface treatments and coating on the power losses in an end pivoted roller follower valve train can also be studied using this setup.

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References

[1] Staron, J. and P. Willermet, *An analysis of valve train friction in terms of lubrication principles*. 1983, SAE Technical Paper.
[2] Sun, D. and R. Rosenberg, *An experimental study of automotive cam-lifter interface friction.* ASLE transactions, 1987. **30**(2): p. 167-176.
[3] Lee, J. and D.J. Patterson, *Analysis of Cam/Roller Follower Friction and Slippage in Valve Train Systems*. 1995, SAE Technical Paper.
[4] Lee, J., et al., *Friction measurement in the valve train with a roller follower*. 1994, SAE Technical Paper.
[5] Khurram, M., et al., *Experimental measurement of roller slip in end-pivoted roller follower valve train.* Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology, 2015: p. 1350650115572198.
[6] Calabretta, M., et al., *Valvetrain Friction-Modeling, Analysis and Measurement of a High Performance Engine Valvetrain System.* SAE International Journal of Engines, 2010. **3**(2): p. 72-84.

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