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The Friction Behavior of Individual Components of a Spark-Ignition Engine During Warm-Up

C. C. Daniels\(^a\) & M. J. Braun\(^b\)

\(^a\) College of Engineering, University of Akron, Akron, OH, 44325-3900, USA
\(^b\) Mechanical Engineering Department, University of Akron, Akron, OH, 44325-3903, USA

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The research presented herein fills a void in the published literature through investigation of transient friction contributions by individual internal combustion engine components during simulated engine warm-up. Currently, engine manufacturers design internal combustion engines primarily for use at steady-state operating conditions with little design consideration for transient engine warm-up. Using the motoring torque waveform and cycle-averaged data of a spark-ignition internal combustion engine, the present work determined the friction behavior of individual engine component assemblies, including the valve train, pistons and connecting rods, oil pump, and crankshaft of a modern internal combustion engine. A common criticism of the standard motoring method is that the engine does not warm up, so lubricant temperature and viscosity does not model that of a fired engine. In the present study, the lubricant and coolant were warmed from 25 to 85\(^\circ\)C. Observations were presented as to the effect of engine speed and the temperature of the coolant and lubricant on total engine friction. Contributions of individual engine components to total engine losses were examined, as well as their variation with engine temperature. The added knowledge of the transient effects of engine temperature can help future designers to mitigate friction and component wear, thus improving overall maintenance costs, specific fuel consumption, and emissions.

**KEY WORDS**
Spark-Ignition Engine; Engine Warm-Up; Motoring Torque Waveform; Cycle-Averaged Data

**INTRODUCTION**
As concerns for the environment increase, automakers are faced with increasingly stricter government regulations for fuel economy and emissions. These environmental issues, coupled with the consumer’s appeal for efficiency, reliability, and performance, have driven the need for engine friction mechanism research. An engine’s mechanical efficiency is directly impaired by frictional losses, since power is wasted to overcome friction within the engine. Thus, reduction of engine friction becomes a very important element, whose management will lead to increased fuel economy. For new designs to be able to reduce engine friction, a better comprehension of engine component friction loss behavior is needed and their contribution to total engine friction must be understood.

Engine manufacturers are faced with a complex non-linear system of design considerations that change as the engine components’ temperatures vary. As engines become hot (due to friction and combustion gases generated within the cylinders), the lubrication conditions change (due to changes in oil viscosity and clearances between rubbing surfaces). As the lubrication conditions change, each individual component’s friction contribution to total engine losses varies. Presently, there is a dearth of literature pointing to an objective research need because designers rely heavily on steady-state experiments and numerical simulation (Barauh (1)), since data from transient experiments is more difficult to generate than from steady state. So, it is the focus of this investigation to study the changes of engine friction as the engine warms from ambient temperature (25\(^\circ\)C) to steady-state temperature (85\(^\circ\)C).

Manufacturers, and researchers at the behest of manufacturers, are using seven main methods to test complete engines and engine components for friction loss. The indicated pressure diagram method (Soejima, et al. (2); Gish, et al. (3); Bishop (4); Brown (5)), the instantaneous torque method (Ma, et al. (6); Ball, et al. (7)), the angular velocity measurement method, the motoring method (Soejima, et al. (2); Bishop (4); Ball, et al. (7); Kovach, et al. (8); Hoshi and Baba (9); El-Moneer (10); Paranjpe and Cusenza (11); Millington and Hartles (12); Hosonuma and Maeda (13); Fodor and Ling (14)), the misfire method, the Willans-Line...
method (Soejima, et al. (2)), and the run-out method (Soejima, et al. (2); Henein, et al. (15); Wakuri, et al. (16)) are all presently used to measure total engine friction and individual friction loss of engine components. Of these, the motoring method is the only experimental method available that is capable of measuring component friction contributions without altering the original design of the test engine. Further, there exists no literature that examines the behavior of friction during engine temperature transients.

SCOPE OF WORK

The conclusions of Ku (17) that indicated temperatures had the greatest effect on engine friction losses emphasize the importance of a fundamental understanding of the transient temperature effects on internal combustion engine friction losses. Though there are many other references to general friction loss studies, our literature review has found only a limited number (Barauh (1); Ma, et al. (6); Rakopoulos and Giakoumis (18); Daniels (19); Zweiri, et al. (20), (21)) of such studies in the area of transient behavior in internal combustion engines. Since only one researcher (Zweiri, et al. (20)) studied friction behavior during temperature transients, and did so on a diesel engine, it follows that there is a real scarcity of data producing meaningful insights into engine component behavior during spark-ignition engine start-up.

It was in this context that the work performed herein was focused on an experimental investigation of an internal combustion engine that quantified the start-up transient behavior of frictional torque generated by individual engine components. The study established the frictional torque characteristics as they correlated to increasing temperature of the engine, as a function of crank angle. The friction contribution of individual engine components was determined using the motoring (and removal) method.

An extension of the standard motoring method used herein more closely modeled actual operating temperature conditions, by heating both the lubricating oil and the coolant. The amount of friction contributed separately by the piston assembly, valve train assembly, crankshaft assembly, and oil pump of a production automobile engine were determined during this experimental investigation. The friction torque waveform results are presented as a function of crank angle. The effect of pumping losses on the complete assembly, along with the changes in component friction contributions with engine temperature, are also be investigated and correlated.

The primary reason for undertaking an engine friction study using the motoring method was that friction contributions of individual engine components could be determined by removing each, one at a time (referred to as the removal method). The measured friction torque difference between one experiment and the next was the friction contribution of the removed component. In this study, the following procedure was used to achieve a better understanding of component friction contribution. The engine frictional torque experiments were conducted on:

1. A complete engine minus the water pump;
2. A complete engine without the water pump, intake and exhaust manifolds, and spark plugs;
3. A complete engine minus the water pump, manifolds, spark plugs, and valve train;
4. A complete engine minus the water pump, manifolds, spark plugs, valve train, piston, and connecting rods;
5. The crankshaft only (a complete engine minus the water pump, manifolds, spark plugs, valve train, piston, connecting rods, and oil pump).

These tests were conducted at two engine-operating speeds of 1100 and 1700-rpm because of their proximity to idle speeds. Thus, using the method of engine component removal, the additive effects of the following were isolated:

1. Pumping losses;
2. Friction torque of the valve train assembly;
3. Friction torque of the pistons and connecting rod assembly;
4. Friction torque of the oil pump; and
5. Friction torque of the crankshaft only.

DESCRIPTION OF THE TEST INSTALLATION

An internal combustion engine was turned over, without the aid of combustion, using an electric motor, see Fig. 1. A belt drive system made the system capable of achieving speeds up to
6000 rpm. A torquemeter, installed between the driving motor and the driven engine, was used to measure the torque required to turn the engine.

**Internal Combustion Engine**

The internal combustion engine was a 2.0-L automobile engine for use in passenger vehicles. The four-cylinder gasoline engine had a compression ratio of 9.2:1 ± 0.25. The engine was installed in the test apparatus in brand-new condition, without a wear-in period.

**Water System**

The water system used the engine’s original 5.8-L capacity cooling system, including the water pump and jacket passages, to heat the engine components. Modifications were made to the water system by removing the radiator and replacing it with an external water reservoir. The closed system used the original water pump to supply the engine with water. This pump, normally driven through connection to the crankshaft pulley, was spun by an external electric motor. By using this configuration, the additional friction from this component was inherently removed from all of the results.

The external reservoir contained a 2 kW submersion heater and coils of copper tubing for cooling water. To accurately control the fluid temperature, the water heating system contained a solid-state relay, a high and low temperature limit controller, and a rheostat, along with the heater. A separate thermocouple was used to sense the liquid temperature and provide data to the temperature controller. The control system regulated the water temperature and allowed for the water temperature to increase as required for mimicking an engine throughout warm-up. The temperature of the water was raised at an approximate rate of 40°C/hour. The original connections to and from the test engine remained, and hose of the same diameter as the original

![Figure 2](image-url)
Tomonitor the engine during the experiments, two thermocouple probes. They were connected to the connector block of the data acquisition system using extension-grade thermocouple wire. The shielded connector block of the data acquisition system had a built-in electronic junction, referred to as a cold junction, which corrects for the open end of the thermocouple being at a non-zero temperature. This cold junction automatically defined the calibration of the thermocouple by the data acquisition software. The accuracy using this type of connection was ±2.2°C over the 0–200°C temperature range.

The experimental pressure data was obtained using a high-temperature ultra-miniature pressure transducer mounted in the spark plug orifice of the first engine cylinder. These pressure transducers have an absolute operating range of 500 psi and are capable of reading 1000 psi without loss of calibration. Each pressure transducer has internal temperature compensation over 25–204°C, eliminating any zero, span, and calibration drift. The 100-mV maximum output voltage was fed directly into the connector block of the data acquisition system through shielded wires. The accuracy of this device was estimated to be ±3.13 kPa (0.455 psi).

The device used to collect friction torque data was an inline shaft torquemeter mounted between the transmission sheaves and the driven engine. The torque sensing equipment was connected by two torsionally rigid single-flex couplings to remove any damping of instantaneous torque and allowed for accurate torque measurements as a function of crank angle with small angular resolution. The enhanced accuracy device had a maximum torque range of ±250 lb/in. The shielded torquemeter was connected to a signal conditioner, which provided amplification to a ±5-volt output to the data acquisition system. The signal conditioner had a selectable low-pass signal and was set to remove all signals above 500 Hz in frequency. Shielded wires connected the output voltage from the signal conditioner to the shielded connector block of the data acquisition system. After calibration, the readable output from the data acquisition system had a repeatable error of ±1.19 N-m (10.6 lb/in).

RESULTS AND DISCUSSION

Experimental Results at an Engine Speed of 1100 rpm

The complete engine was motored to determine baseline values for friction and pumping losses. All other experiments were compared to this established baseline. All of the production engine components were attached, including the crankshaft, oil pump and filter, pistons and connecting rods, intake camshaft and valves, exhaust camshaft and valves, timing belt, spark plugs, and intake and exhaust manifolds. The engine was motored at an engine speed of approximately 1100 rpm and was naturally aspirated through the intake and exhaust manifolds.

Starting from the baseline motoring results, (completed with all the engine components attached), each successive experiment was conducted with one less component assembly attached to the engine. First, the spark plugs and manifolds were removed, then the valve train, followed by the pistons and connecting rods, and finally the oil pump. Each experiment was compared to the previous and the difference in motoring torque was attributed to the removed engine component. The resultant motoring torque contributed by engine component was graphed versus engine...
As the engine coolant water was heated from 25 to 85°C, the speed of the engine remained steady, averaging 1081 rpm with random fluctuations $+68/-38$ rpm, (see Fig. 2a), and the oil temperature followed within $+0.0/-0.8$°C, (see Fig. 2b). To determine the increase in cylinder pressure due to the reduction in volume of the cylinder during each compression stroke, the pressure of the trapped gases within the cylinder was measured using the pressure transducer mounted in the spark plug hole. The cylinder pressure rose from atmospheric pressure to an average of 1910 kPa (277 psig). The cylinder pressure at top dead center (TDC) varied from a maximum of 1980 kPa (287 psig) to a minimum of 1760 kPa (255 psig), showing no trends with increasing temperature.

At approximately every 2.5°C rise in engine water coolant temperature, 2000 torque measurements were taken at 0.0001-second intervals. Using the timing mark attached to the flywheel to correlate each set of data to a common angular crankshaft position, these waveform measurements distinctly show four peaks and four troughs in the motoring torque, as shown in Fig. 3. The pistons worked in conjunction with each other, so that the pistons in cylinders 1 and 4 arrived at top dead center (TDC) as the pistons in cylinders 2 and 3 were at bottom dead center (BDC), and vice versa. The peaks in motoring torque arose approximately 180° apart at 17°aTDC and 204°aTDC. At these angular positions, the piston velocities were accelerating from motionless at top (cylinders 1 and 4) and bottom (cylinders 2 and 3) dead centers. The minimum torque occurred at 139°aTDC and 324°aTDC when all four pistons were at peak velocity at their upper mid-stroke. The second set of peaks and troughs were shown because of the second revolution of the engine needed to complete the four-stroke cycle. Ku (17) confirmed this torque behavior in an isolated piston component test facility. After top dead center, the compression rings are in contact with the lower surface of the piston groove due to high cylinder pressure. As the piston ring encounters larger oil film thickness, friction between the piston groove and the ring restricts the radial movement of the ring, causing a peak in torque just after top dead center.

Fig. 4—Comparison of mean effective pressure contributions of individual components to complete engine mean effective pressure at varying water temperature at an engine speed of 1100 rpm.
Overall, the motoring torque waveforms shift downward with increasing water temperature. This trend was reflected in the cycle-averaged motoring torque decreasing 60 lb/in from 254 lb/in at 25° C, see Fig. 2a. Small fluctuations in values of torque were attributed to irregularities in revolution-to-revolution torque behavior, as shown in Fig. 3.

Systematically, components of the engine were removed. Through subtraction of the current cycle-averaged friction value from that of the previous experiment build, the resulting friction contribution by the removed component was determined. The contributions of each component to the total friction are shown in Fig. 4, expressed as friction mean effective pressure. Friction mean effective pressure can be expressed as,

\[ f_{MEP} = \frac{4\pi \tau_f}{V_d} \]

where \( \tau_f \) is the motoring torque and \( V_d \) is the volume displaced. With the inherent scatter of the experimental data, the contribution of the oil pump appears null in the figure at 67.5° F.

After conducting experiments with all the engine components attached to the engine, the cycle-averaged motoring torque was observed to increase when the spark plugs and manifolds were removed from the engine. With the spark plugs attached to the engine, compression takes place in the engine every other revolution, aiding the downward piston movement on the expansion stroke. Without the spark plugs attached to the engine, pushing and pulling air through the spark plug holes caused increased resistance to rotation on every revolution. This problem is common to motoring torque measurements and highlights a deficiency of this method.

**Experimental Results at an Engine Speed of 1700 rpm**

The standard motoring method was again employed to determine baseline values for friction and pumping losses against which all other experiments were compared. All of the production engine components were attached, including the crankshaft, oil pump and filter, pistons and connecting rods, intake camshaft...
and valves; exhaust camshaft and valves; timing belt, spark plugs, intake and exhaust manifolds. The engine was motored at the normal idling speed of the engine, approximately 1700 rpm and was naturally aspirated through the intake and exhaust manifolds.

The speed of the engine remained steady averaging 1673 rpm with random fluctuations $+59/-25$ rpm as the engine coolant water was heated from 25 to $85^\circ$C, (see Fig. 5). The oil temperature followed within $-2.5/-7.2$ $^\circ$C of the engine water temperature. The cylinder pressure rose from atmospheric pressure to an average of 282 psig during the compression stroke. The cylinder pressure at top dead center (TDC) randomly varied between a maximum of 298 psig and a minimum of 264 psig.

The torque waveform measurements were recorded using the timing mark previously described and the measurements distinctly show four peaks and four troughs in the motoring torque, as shown in Fig. 6. The peaks in motoring torque arose approximately 180$^\circ$ apart at 22$^\circ$aTDC and 208$^\circ$aTDC. Similar piston ring–oil film interaction occurred at 1700 rpm as was attributed to the rise in motoring torque at 1100 rpm. The minimum torque occurred at 78$^\circ$aTDC and 246$^\circ$aTDC when all four pistons were at peak velocity (at their upper mid-stroke).
As expected, the motoring torque amplitudes decrease in magnitude with increasing water temperature as the viscosity of the oil decreased. This trend was reflected in the cycle averaged motoring torque decreasing 89 lb/in, from 274 lb/in at 25°C to 185 at 85°C, as shown in Fig. 5.

The cycle-averaged friction values were subtracted from the previous experiment build, resulting in a value of friction contributed by the removed component. The values of friction are shown in Fig. 7. To assess the variability between data sets, the standard deviation of the means was calculated. The largest value was obtained with the engine complete at 25°C (0.004 MPa) and the smallest (9 × 10^-5 MPa) with only the crankshaft installed at 85°C.

SUMMARY

The present work used a novel method of heating the lubricant and cooling water in order to experimentally determine the friction behavior of individual engine component assemblies (including the roller-follower valve train, pistons and connecting rods, oil pump, and crankshaft assemblies), over a range of operating temperatures from 25 to 85°C.

The two engine speeds examined had minimal effect on the required motoring torque and resulting mean effective pressure of the complete engine assembly. The required total engine friction torque recorded rose only slightly with the increase in engine speed, averaging a 4.9% increase over the range of temperatures. The fraction of friction mean effective pressure contributed by each component did not appreciably change with the engine speed.

The pistons and connecting rod assembly (including the oil ring, two compression rings, connecting rod wrist pin bearing, and connecting rod main bearing) contributed the most friction mean effective pressure to the total engine losses. The valve train assembly fraction of contributed friction mean effective pressure was far smaller, 19 and 13% at 1100 and 1700 rpm, respectively. The crankshaft and oil pump contributed only 11 and 9%, respectively, at both speeds.

The engine coolant temperature had a significant effect on friction mean effective pressure. At both engine speeds, mean effective pressure of the complete engine assembly dropped approximately 26% when engine coolant temperature was raised from 25 to 85°C. The largest contributors to total engine friction, the piston and connecting rod assemblies, contributed 76\% at 25°C than at 85°C. Though a significantly smaller proportion of total engine mean effective pressure, the contribution of the oil pump and crankshaft were significantly reduced at 85°C. The reductions of the oil pump and crankshaft were 35 and 84% less at 1100 rpm and 69 and 57% less at 1700 rpm, respectively. Only the valve train assembly did not show a significant drop in mean effective pressure contribution with increasing engine coolant temperature, as expected of a roller-follower type valve train.

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REFERENCES