Comparison of measured drive torque and average friction torque at cam/follower interface for various commercially available engine oils of the same SAE rating (10W-40)

Author

Naseh Jamal

Registration Number

320573

Supervisor

Dr. Jawad Aslam

DEPARTMENT

SCHOOL OF MECHANICAL & MANUFACTURING ENGINEERING

NATIONAL UNIVERSITY OF SCIENCES AND TECHNOLOGY

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Author

Naseh Jamal

Registration Number

320573

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Thesis Supervisor:

Dr. Jawad Aslam

Thesis Supervisor's Signature: ______________________________

DEPARTMENT

SCHOOL OF MECHANICAL & MANUFACTURING ENGINEERING

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Acknowledgments

First and foremost, I am thankful to Allah Almighty for providing me with the wisdom and intuition to carry out the successful completion of this thesis. Without his will it wouldn’t have been possible at all.

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Abstract

In recent times, automobile scientists have focused a lot upon the fuel economy and energy conservation in motor vehicles. The search for alternate fuels and the minimization of thermal and mechanical losses in motor engines has been a major topic of research for the past several decades. In an automobile, one of the most complicated systems in terms of mechanical losses is the engine valve train. Several theoretical and experimental models have been used to calculate the losses in engine valve train and the effect of various engine parameters on the frictional losses. The work presented in this article focuses upon the drive and friction torque values for four different commercially available oils of the same grade i.e., 10W-40. These oils are Shell, Castrol, Kixx and ZIC. The tests are carried out on a Suzuki swift RS413 gasoline engine cylinder head. The variation in drive and friction torques with oil temperature and camshaft speed is presented. The results show the comparisons of these four oils for the drive and friction torque values and based on these results, conclusions are given regarding the best oil in terms of friction performance of the given engine valve train. It is thus concluded in this thesis that Kixx is the oil with the least value of average friction torque for the given experimental conditions and is therefore the best one among the four oils compared.

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CHAPTER 1: INTRODUCTION

The research work done in this thesis is presented in four sections. The first section covers the background, literature review and the aim of this thesis. The second section explains the lubricants used and the experimental setup utilized for carrying out the research work. The third section covers the results and the discussions. Finally, the last section presents the conclusion of this research.

1.1 Background and Literature review

At the onset of the second half of the 20th century it became clear that fuel economy and overall energy conservation are of extreme importance for automobile scientists. The rate at which the fossil fuel was being consumed made it necessary to look for either alternate fuels or make automobiles more efficient to reduce the overall fuel consumption. Furthermore, strict rules and regulations were put in place to ensure that automobile scientists and engineers look for ways to tackle the problems. To solve the problem of fuel economy various advanced techniques have been developed by automobile manufacturers such as low viscosity oils, innovative valve timing techniques, direct fuel injection etc. [1]. Fuel consumption can also be reduced by minimizing the energy losses in an internal combustion engine. Therefore, a lot of research has been done to find out the major sources of energy losses in an internal combustion engine. The major contributor among these is the losses due to thermodynamic effects. However, the effect of mechanical losses in the various components of an engine cannot be underestimated [2]. The pure mechanical losses in an engine are in various components of the engine such as piston assembly, bearings (crankshaft and camshaft), valve train, pumps, and various other auxiliaries [3]. The major contributor towards friction losses are generally the bigger components like the pistons and the crankshaft bearings.

However the friction losses in valve train cannot be ignored especially at low engine speeds [2].

The first step in minimizing friction losses in an engine is to calculate the friction losses that are occurring in the engine. A lot of research has therefore been done in order to predict the friction losses in various components of an engine by utilizing various friction prediction models [7-12]. In case of a camshaft, a new technique for measuring the drive torque without modification of the drive train has been reported [2]. Similarly theoretical engine friction mathematical model like the one developed in a study at Leeds [5] are present which have also been experimentally verified [5].

One of the most technically complicated components of an engine in terms of energy losses is the engine valve train. The main components of an engine valve train include the camshaft, followers and the valves. The most stringent conditions in terms of forces and from a lubrication perspective are at the cam/ follower interface in an engine valve train [4]. In order to understand how to minimize the force of friction at the cam/follower interface and how to lubricate the interface, it is necessary to know what kind of contact takes place between the cam and the follower. It has been established that the lubrication mode transitions from elastohydrodynamic through mixed to full boundary lubrication [5]. Due to such a lubrication regime change, the design of the engine valve train and the subsequent reduction of friction forces have always been a challenging task.

The valve train friction has been measured in many ways by various researchers and the effect of various parameters on the friction losses has been reported. Using instrumented cam pulleys, it has been shown that valve train friction decreases with increasing the engine speed but at the same time increases with an increase in the oil temperature [6]. Similarly, for a roller/follower valve train an increase in the friction drive torque due to an increase in the lubricant oil temperature has been reported [1].

Furthermore, the effect of the type of lubricant on the friction losses has also been studied extensively. It has been shown that the use of a bio-lubricant (PAO+GMO+MoDTC+ZDDP) is effective in reducing the friction between the cam and tappet interface [13].

The losses due to friction between interacting surfaces can be reduced by using special coatings that protect the interacting surfaces [15-17]. Furthermore, these coatings also improve the other tribological characteristics of the interacting parts. In this context, the use of DLC coatings which are chemically inactive [18] and their interaction with base oils has been reported and a reduction in boundary friction values has been observed [14].

1.2 Aim of the thesis

As discussed in the above section, much work has been done in predicting the valve train friction as well as establishing the relationships between friction and various parameters such as the engine speed and the lubricant temperature. The aim of this thesis is to see the response of the drive and average friction torques of a particular engine oil namely 10W-40 of several commercially available brands to the two most important parameters i.e., the engine speed and the lubricant temperature.

The four oils selected for this research work are of the following brands.

1. Shell Helix HX6
2. Castrol MAGNATEC
3. ZIC X7
4. Kixx G1

This study also compares the friction values at different speeds and lubricant oil temperatures for the oils of different brands. The final goal is to establish which of these commercially available brands is the best one in terms of reduced friction losses in the valve train.

CHAPTER 2: LUBRICANTS AND EXPERIMENTAL SETUP

2.1 Lubricants

The following four lubricants were used in this research.

1. Shell Helix HX6
2. Castrol MAGNATEC
3. ZIC X7
4. Kixx G1

The following table lists the important characteristics of these oils used.

Brand	Nature	Grade	Viscosity@40°C (cSt)	Visocsity@100°C (cSt)	Flash point °C	Qty used (L)
Shell Helix HX6	Fully synthetic	10W-40	92.1	14.4	220	4
Castrol MAGNATEC	Fully synthetic	10W-40	94	13.6	202	4
ZIC X7	Fully synthetic	10W-40	92.7	14.1	240	4
Kixx G1	Semi Synthetic	10W-40	Not available in data sheet	14.9	240	4

Table 1. Properties of the four oils used

2.2 Experimental Setup

A Suzuki swift RS413 gasoline engine cylinder head was used for carrying out the experiments. This cylinder head was mounted on a direct acting valve train test rig as shown in Fig 1. All the experiments were conducted under motored conditions of the engine. The RS413 is a four-cylinder gasoline engine with two cams and eight intake and exhaust valves. An induction motor that can transmit back-lash free torque and is self-compensating for the shaft misalignments was connected to the exhaust camshaft of the engine. Moreover, this motor had the added functionality of being a variable speed unit and was feedback controlled. The entire control system comprising of the controller, the Data acquisition system (DAQ) and a custom-built software in the LabVIEW environment was used for real time monitoring and control of the variables like the lubricant oil temperature and the rotational speed of the motor. Furthermore, this system was also used to acquire the necessary experimental data from the engine. The lubricant pump responsible for supplying lubricant to the engine was operated by another motor. The pressure and flow rate of the lubricant were controlled by the DAQ system. The test rig consisted of an oil sump made of sheet metal in which the engine was assembled, and this sub assembly was then attached to the valve train test rig via bolts. The oil sump was used to collect the oil coming out of the engine which was then transferred to an external oil sump. This sump was a double walled container which was insulated to minimize the heat losses from the system as all the tests were conducted at oil temperatures above the ambient temperature. The external oil sump was the main reservoir for holding the lubricant and was the place where lubricant changes would take place. The experiments required the lubricants to be heated to different temperatures and subsequently maintained at these temperatures. To accomplish this, a heating and cooling unit as shown in Fig 1 was used. The heating of the lubricant was carried out indirectly. The heating and cooling unit had heat transfer oil in it. Initially, if the lubricant was to be heated, the heat transfer oil was heated up in the unit and then this oil was used to transfer heat to the lubricant in an in-line plate heat exchanger. At the place where the lubricant entered the engine a thermocouple was mounted, and it was used to monitor the temperature of the lubricant. The pressure of the lubricant entering the engine was monitored via an analogue pressure gauge. A pressure transducer (piezo-resistive) was used to convert pressure into voltage. Using this, the pressure of the lubricant entering the engine was controlled via the custom-built LabVIEW software. All the pipes in this system for carrying the lubricant were insulated to minimize the heat losses.

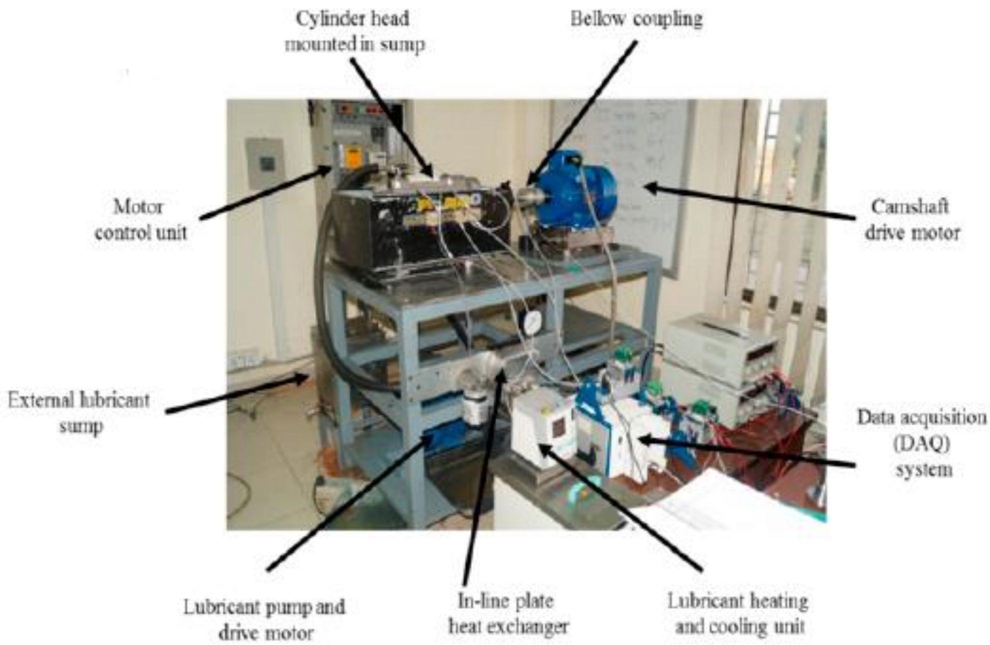

Figure 1. Test rig with the direct acting valve

Figure 2. Swift RS413 cylinder head and the cam/tappet interface

CHAPTER 3: RESULTS AND DISCUSSIONS

To fully understand the results of the experiments conducted, three important terminologies need to be explained. These are as follows.

- Measured Drive Torque
- Average Friction Torque
- Number of pulses

3.1 Measured Drive Torque

The drive torque consists of two components i.e., the geometric torque and the frictional torque (which is explained in the next section). In case of a cam/follower mechanism, the geometric torque represents the torque associated with the opening and closing of the valve. During the opening and closing of a valve, the numeric value of the geometric torque is equal. However, it is taken as positive during the opening motion and negative during the closing of the valve. Therefore, upon the addition of the geometric and the frictional torque (which is positive during the entire process) to obtain the drive torque, the positive part of the geometric torque increases and simultaneously the negative part becomes less negative. Consequently, the graphs that are presented in the following sections will appear offset from the mean position.

The term measured drive torque in the following text represents the drive torque that has been measured experimentally.

3.2 Average Friction Torque

The frictional torque on the other hand is associated with the frictional forces that are present at the cam/follower interface, the cam bearing, the valve guide etc.

The term average frictional torque in the following text represents the average value of the measured drive torque over the entire sample cycle.

3.3 Number of pulses

The term "number of pulses" will be used in the following sections as the title of the x-axis on all the graphs. This term refers to the number of pulses of the optical encoder as the camshaft goes through a complete rotation covering 360 degrees.

The results of the experiments conducted with the four different oils are first presented separately and later a comparative analysis has been shown as well.

3.4 Results and discussion on the measured drive torque values of the four oils

The measured drive torques of Shell Helix HX6, Castrol MAGNATEC, ZIC X7 and Kixx G1 at different temperatures and camshaft rotational speeds are shown in Figs 3-7., Figs 8-12., Figs 13-17. and Figs 18-22., respectively.

An important remark regarding these figures is that the geometric torque portion of the drive torque is only affected by the engine speed and the lubricant oil temperature has no effect on it.

The measured drive torque at a fixed temperature decreases as the camshaft rotational speed increases. This is visible in Figs 5-7., Figs 10-12., Figs 15-17. And Figs 20-22. for Shell Helix HX6, Castrol MAGNATEC, ZIC X7 and Kixx G1 respectively. This indicates that that as the camshaft rotational speed increases, the lubrication conditions at the cam/follower interface improve. This can be attributed to the increase in the lubricant film thickness at the interface due to more and more oil being pushed into the pressurized contact interface due to the high entrainment velocity. The entrainment velocity increases as the camshaft rotational speed is increased as reported by [1]. This increase in the lubricant film thickness shifts the lubrication regime between the cam and follower towards hydrodynamic lubrication and since there is less metal on metal contact thereby the tribological characteristics are improved. Furthermore, two types of loadings namely the spring loading and the inertial loading are important regarding the cam/follower mechanism. At low camshaft rotational speeds inertial loading is less dominant as compared to the spring loading. The spring loading is more pronounced at the cam nose region and so it affects the contact loading at the cam/follower interface. This results in increased drive torque values. At higher camshaft rotational speeds, the inertial loading takes over and instead of the cam nose, it is more pronounced at the cam flank regions. This results in a reduced contact loading at the cam/follower interface and so the drive torque values are reduced.

The measured drive torques at fixed camshaft rotational speeds with varying temperatures are interesting. At 300 RPM, the trend in the measured drive torque is straightforward for all four oils. As the temperature is increased at this camshaft rotational speed, the drive torque increases. This is shown in Fig 3, Fig 8, Fig 13, and Fig 18 for Shell Helix HX6, Castrol MAGNATEC, Zic X7 and kixx G1 respectively. The reason is that as the temperature increases, the viscosity of the oils decreases and so the fluid film thickness between the cam and follower decreases. This shifts the lubrication regime more towards mixed and boundary lubrication. Due to this change in the lubrication regime, the surface roughness and the asperities of the cam and follower become extremely important as they meet each other and so the result is an increase in friction and thereby an increase in the measured drive torque values. At camshaft rotational speed of 500 RPM, the drive torque values increase more prominently as the temperature is increased from 60°C to 90°C but the drive torque values stay almost the same at lower temperatures of 30°C and 60°C as shown in Fig 4, Fig 9, Fig 14, and Fig 19. A possible explanation for this trend may be that although the viscosity drops in increasing the temperature from 30°C to 60°C, the shift in the contact loading from spring to inertial loading as the camshaft rotational speed is increased, offsets that change.

Graphs of Shell Helix HX6

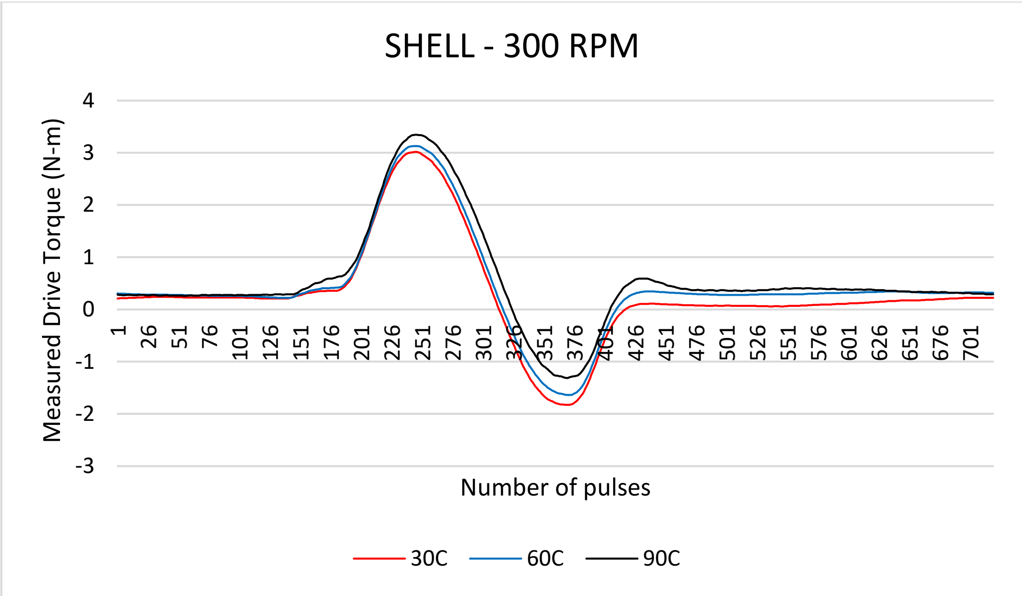

Figure 3. Measured drive torque at 300 RPM for Shell Helix HX6

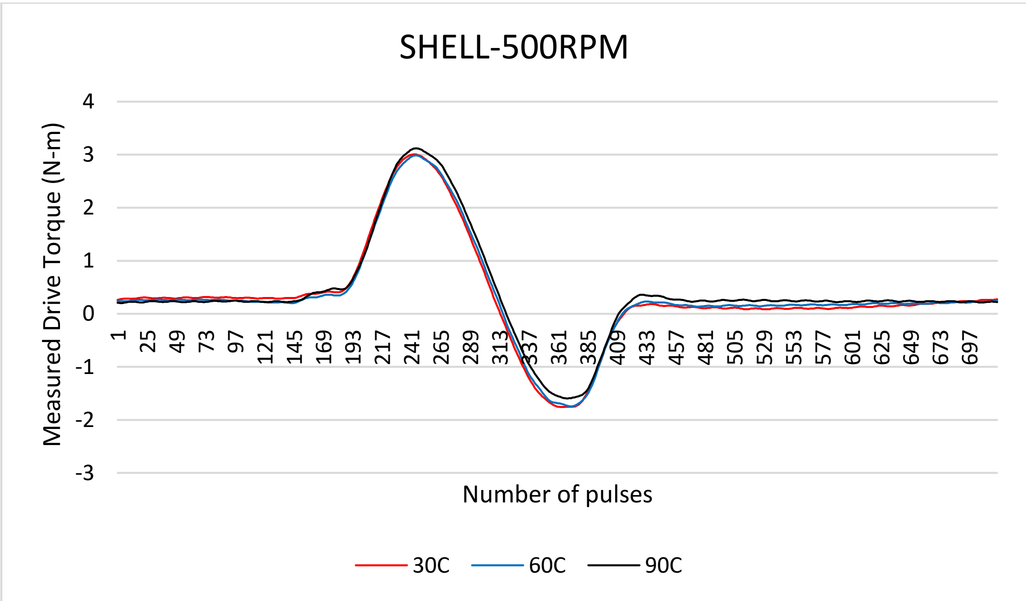

Figure 4. Measured drive torque at 500 RPM for Shell Helix HX6

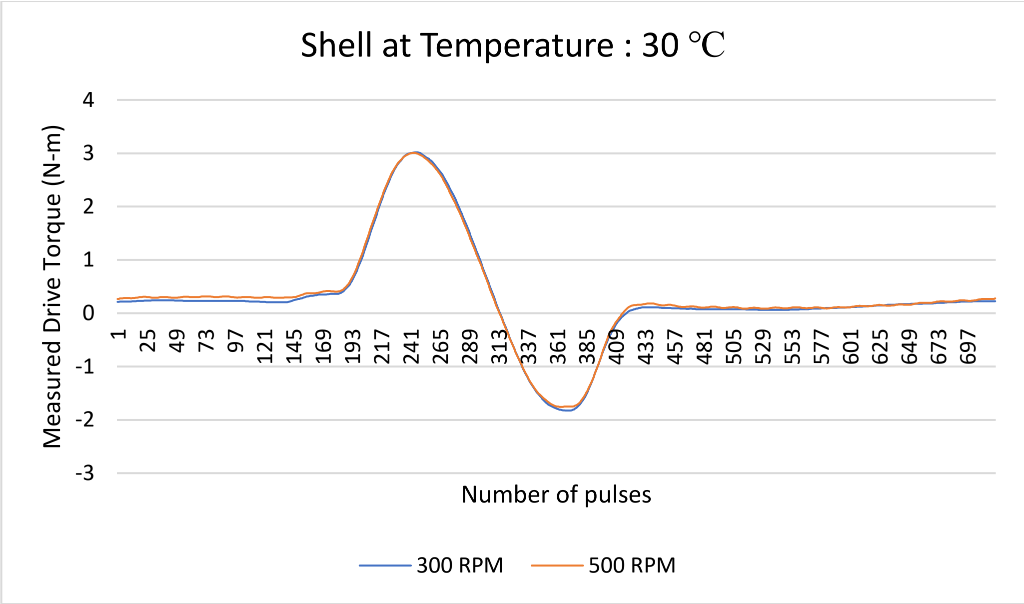

Figure 5. Measured drive torque at 30°C for Shell Helix HX6

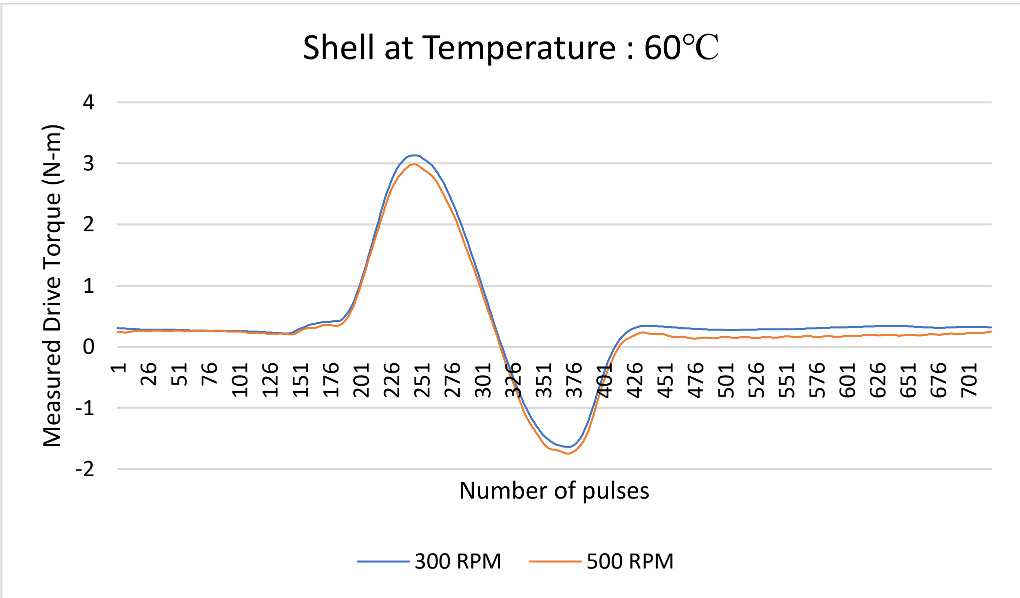

Figure 6. Measured drive torque at 60°C for Shell Helix HX6

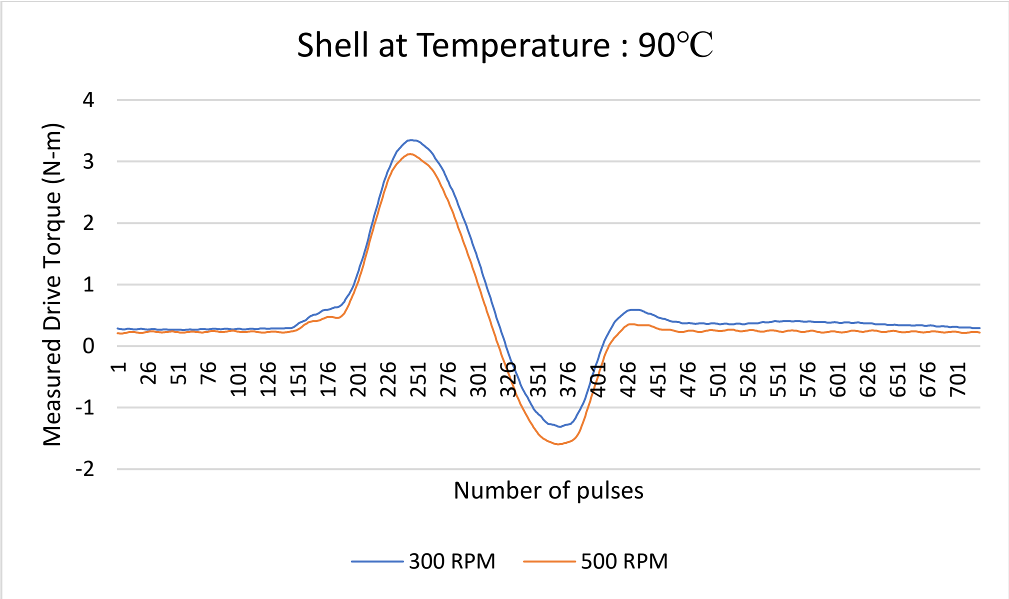

Figure 7. Measured drive torque at 90℃ for Shell Helix HX6

Graphs of Castrol MAGNATEC

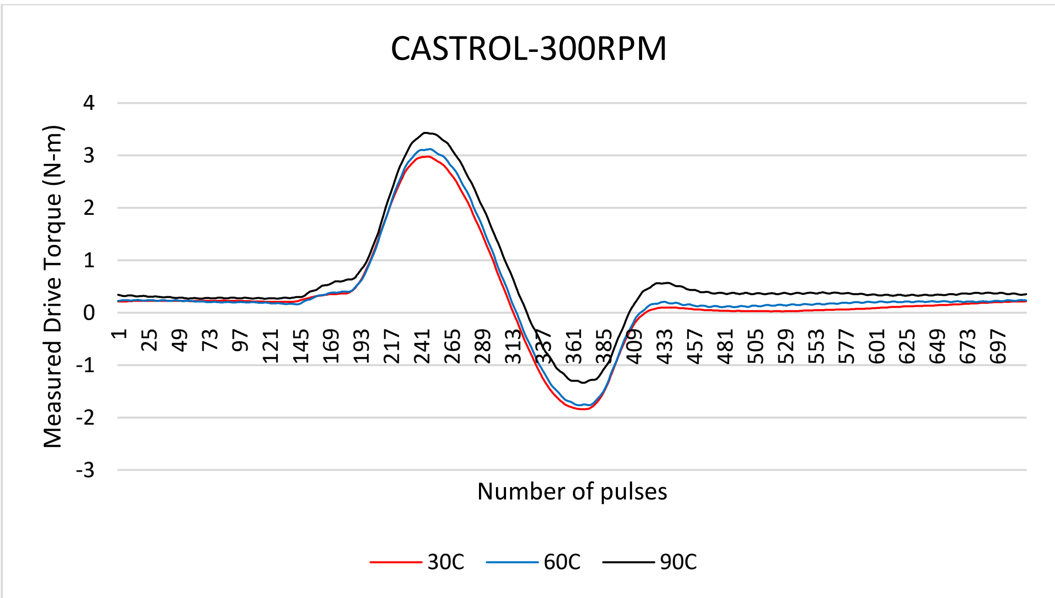

Figure 8. Measured drive torque at 300 RPM for Castrol MAGNATEC

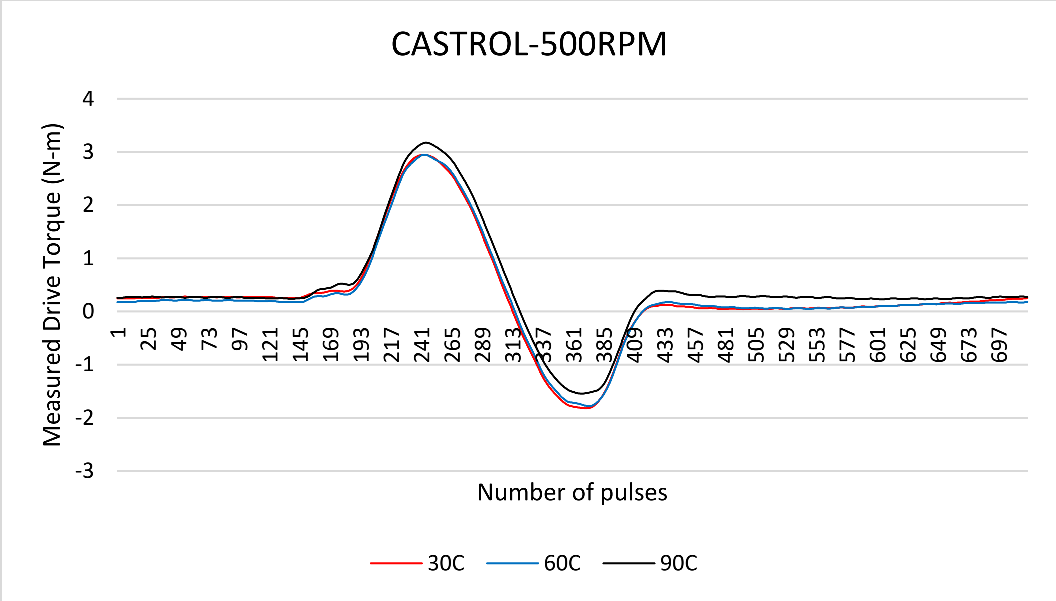

Figure 9. Measured drive torque at 500 RPM for Castrol MAGNATEC

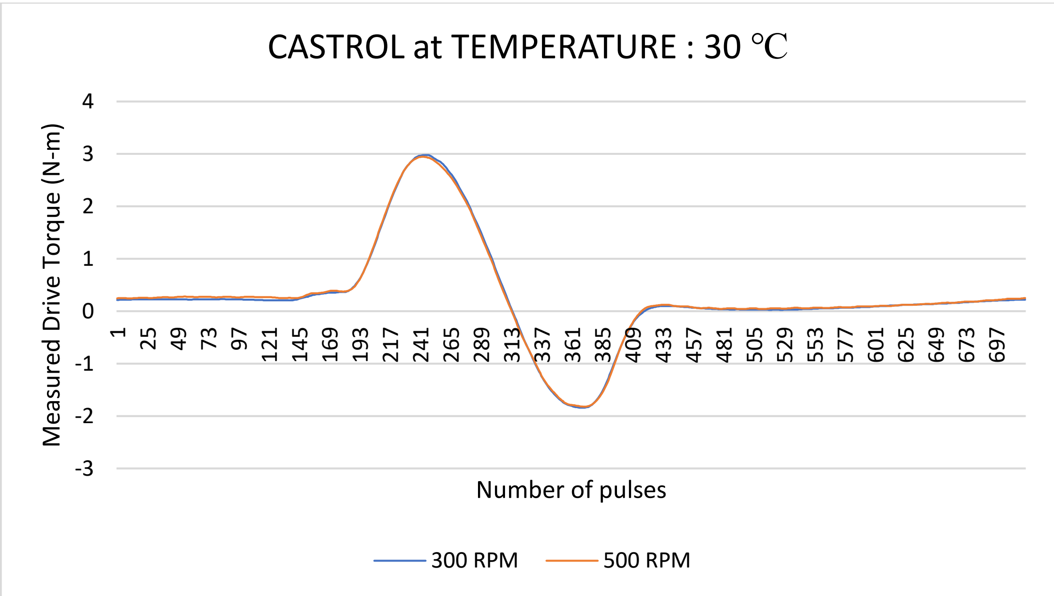

Figure 10. Measured drive torque at 30 ℃ for Castrol MAGNATEC

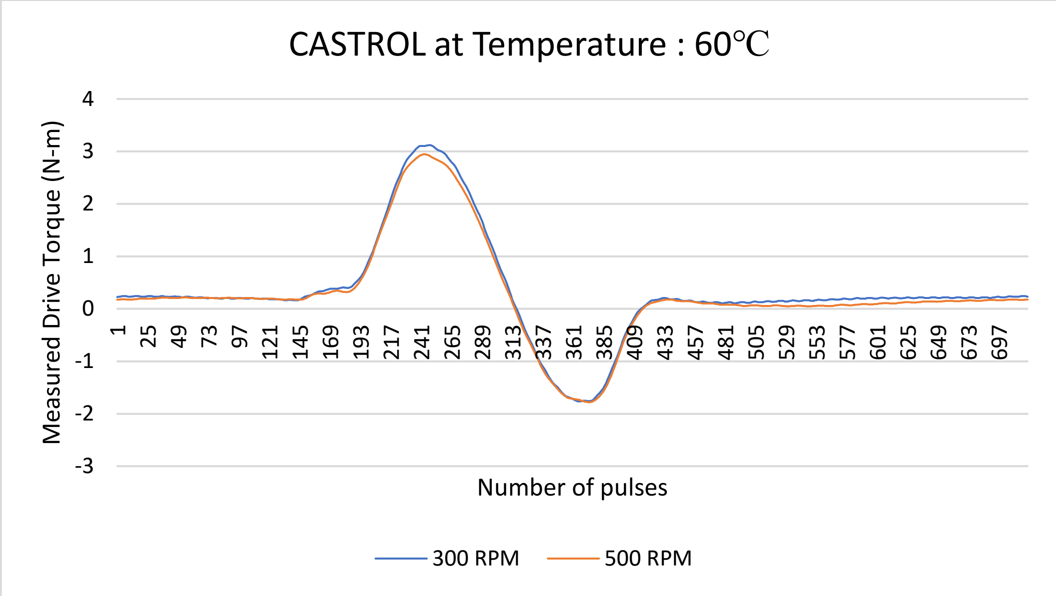

Figure 11. Measured drive torque at 60 °C for Castrol MAGNATEC

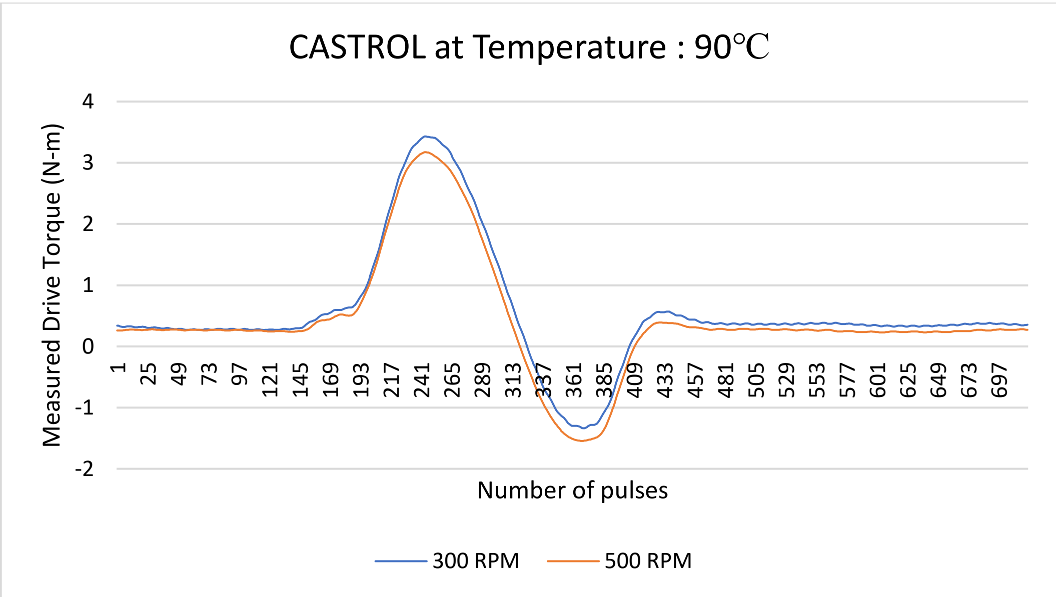

Figure 12. Measured drive torque at 90 °C for Castrol MAGNATEC

Graphs of ZIC X7

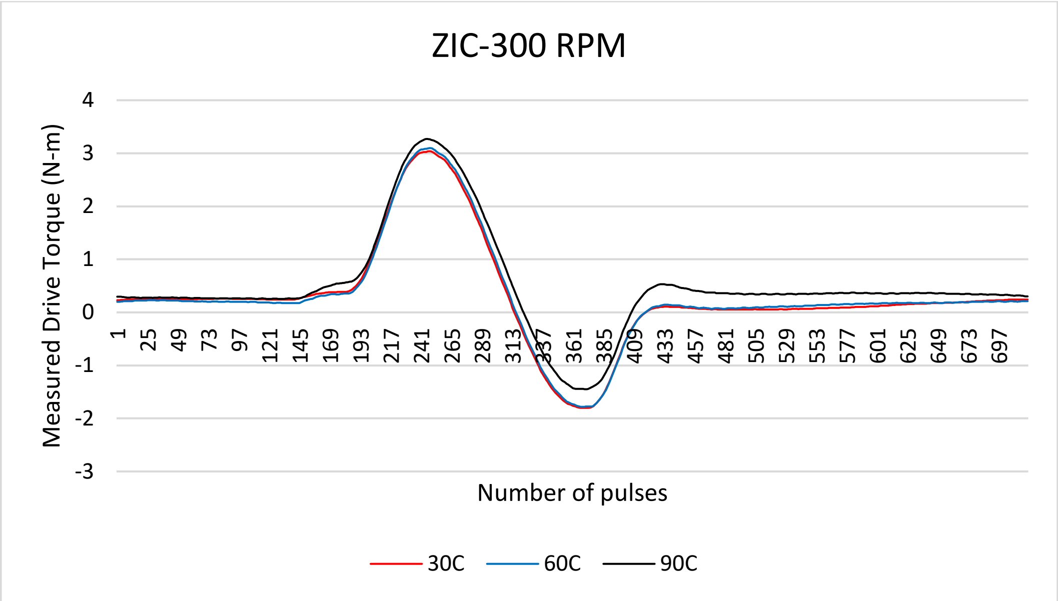

Figure 13. Measured Drive Torque at 300 RPM for ZIC X7

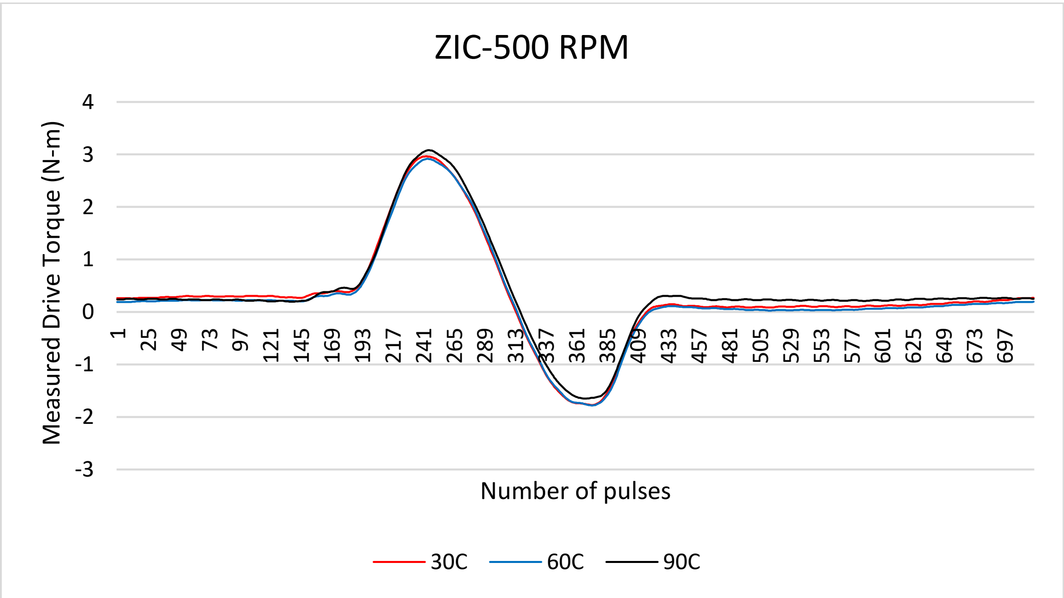

Figure 14. Measured Drive Torque at 500 RPM for ZIC X7

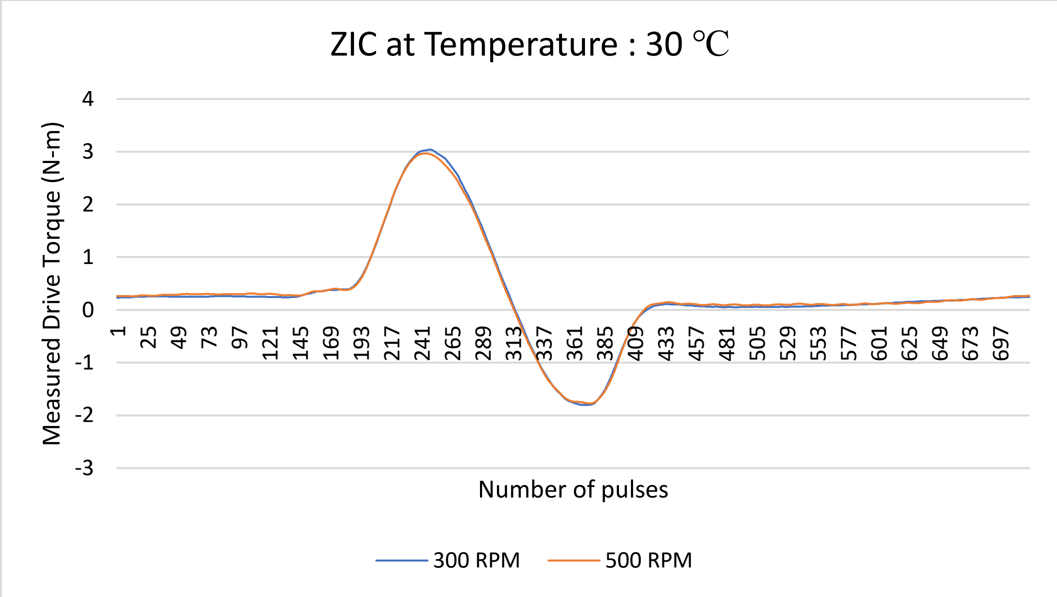

Figure 15. Measured Drive Torque at 30°C for ZIC X7

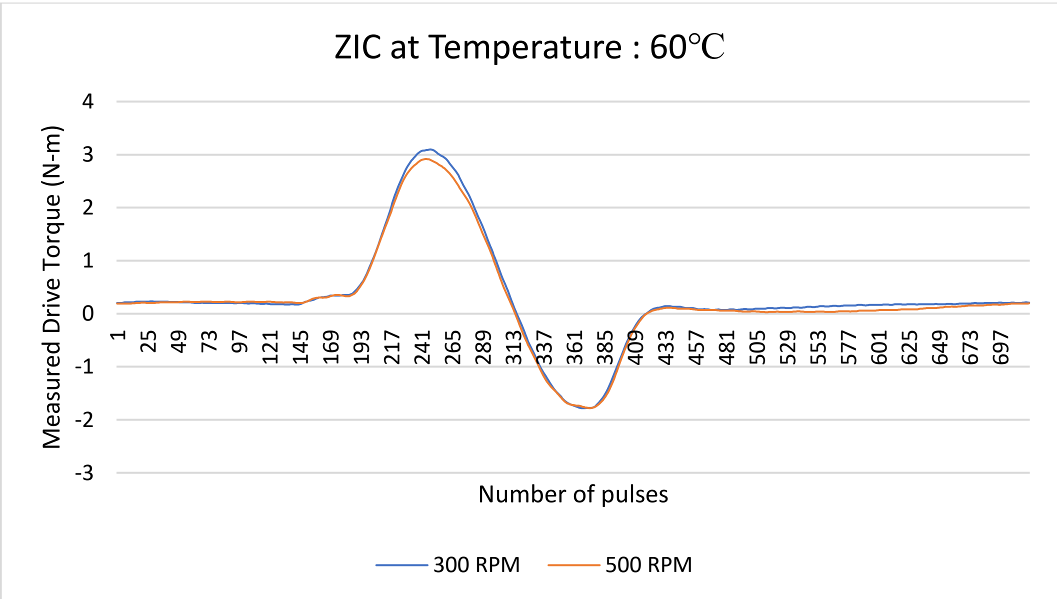

Figure 16. Measured Drive Torque at 60°C for ZIC X7

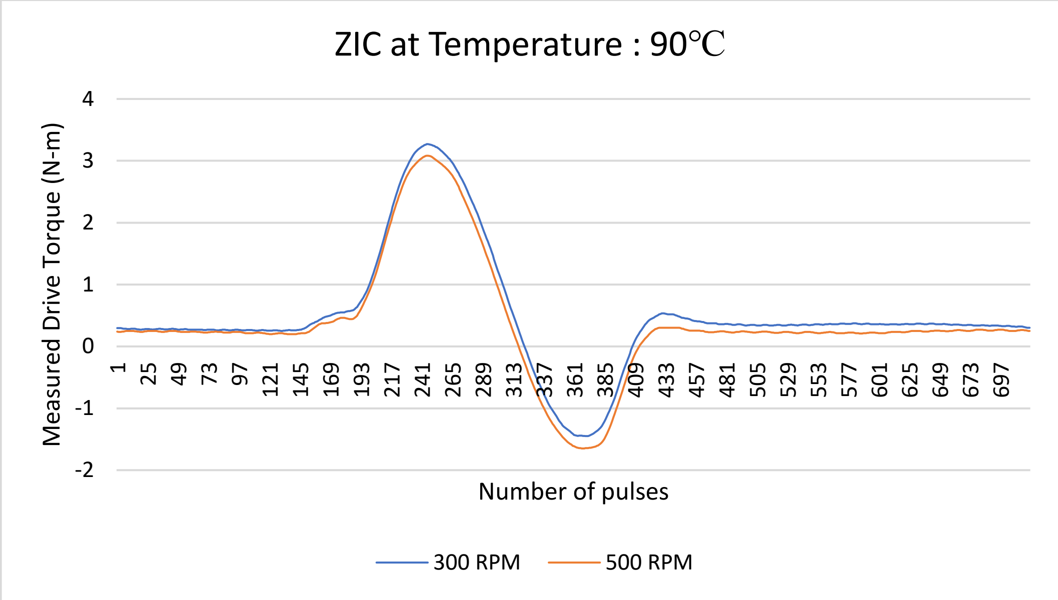

Figure 17. Measured Drive Torque at 90℃ for ZIC X7

Graphs of Kixx G1

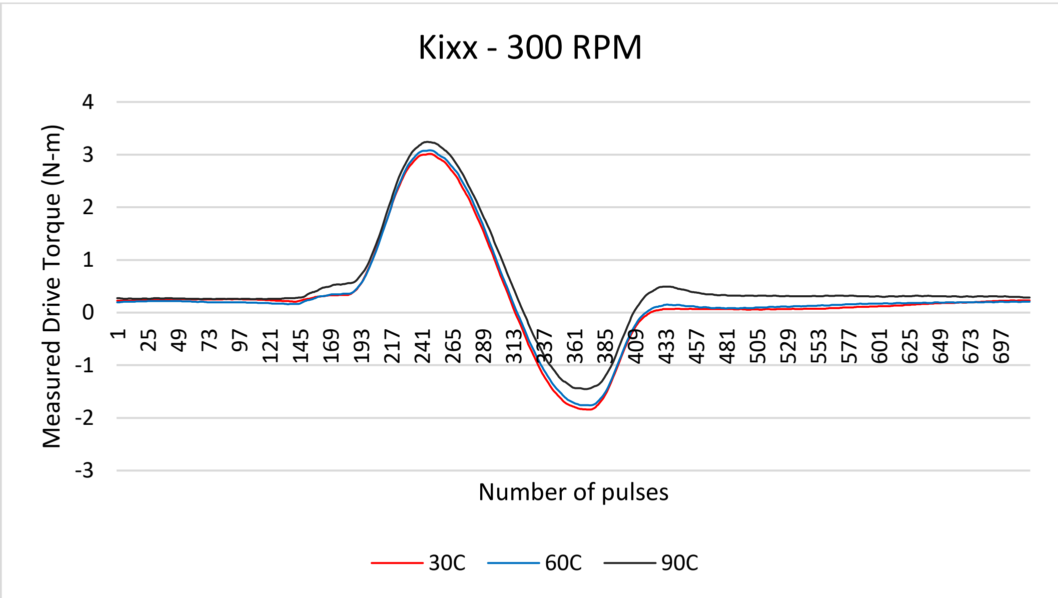

Figure 18. Measured Drive Torque at 300 RPM for Kixx G1

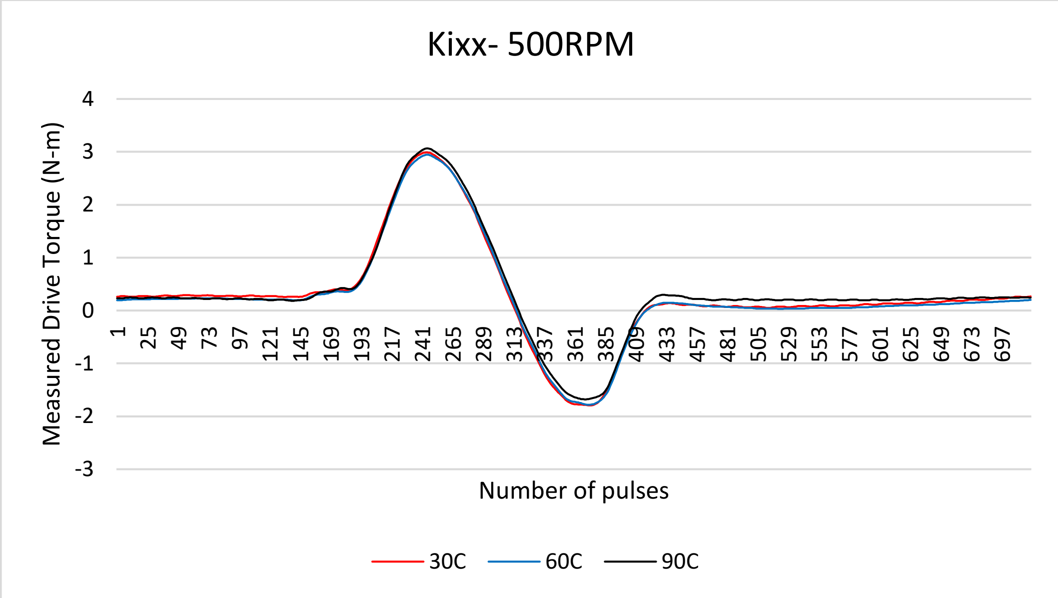

Figure 19. Measured Drive Torque at 500 RPM for Kixx G1

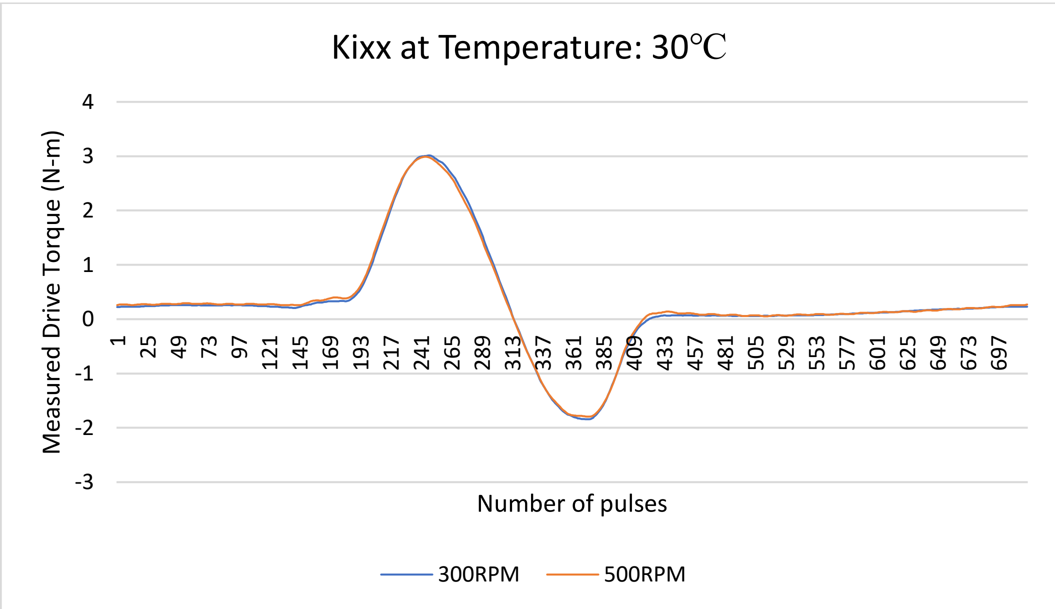

Figure 20. Measured Drive Torque at 30℃ for Kixx G1

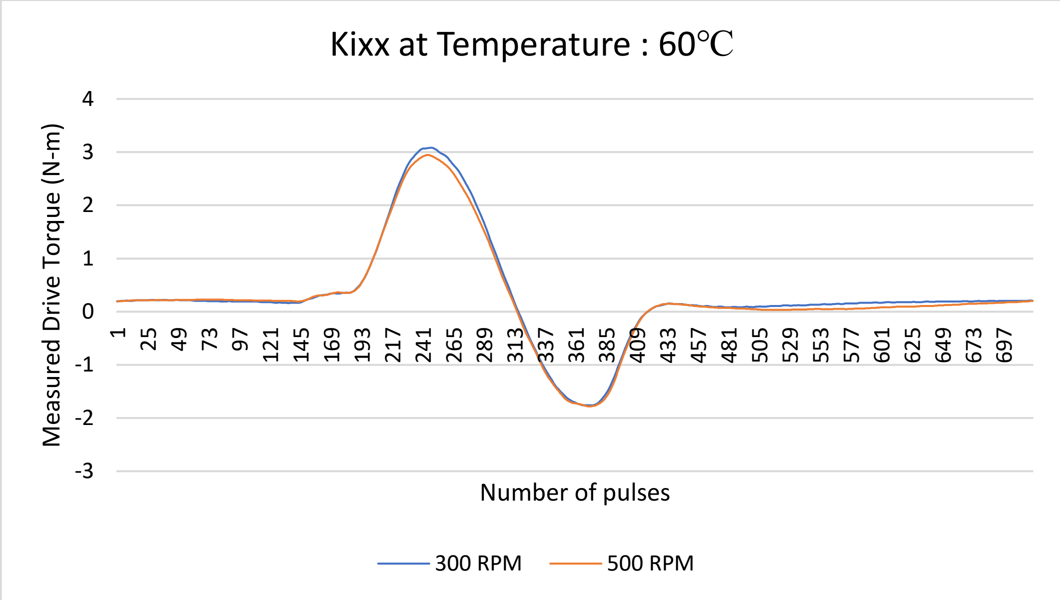

Figure 21. Measured Drive Torque at 60℃ for Kixx G1

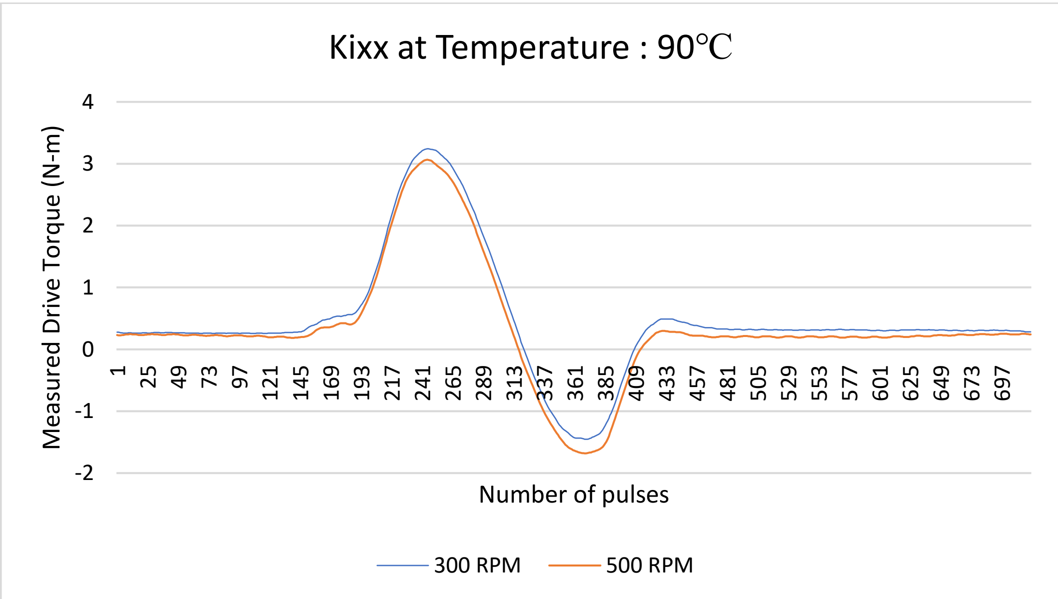

Figure 22. Measured Drive Torque at 90℃ for Kixx G1

3.5 Results and discussion on the average frictional torque values of all four oils

The average frictional torques for Shell Helix HX6, Castrol MAGNATEC, ZIC X7 and Kixx G1 at temperatures of 30℃, 60℃ and 90℃ are shown in Figs. 23-25. Furthermore, Fig. 26 and Fig. 27 shows the average frictional torque values at camshaft rotational speeds of 300 RPM and 500 RPM respectively.

The difference in average frictional torque values at a temperature of 30 for all four oils when changing the camshaft rotational speed from 300 RPM to 500 RPM is not significant as shown in Fig.23. The reason for this can be the fact that although increasing the rotational speed has the effect of improving the lubrication conditions as explained in the previous section, the temperature of 30℃ is low enough that the oils are already viscous and further improvement in the lubrication film thickness is resulting in increasing the frictional values instead of decreasing them. Therefore, the average frictional torque values for all four oils are slightly higher at 500 RPM than at 300 RPM. However, it is necessary to point out that the difference in frictional values is minimal between the two rotational speeds.

The average frictional torque values at Temperature of 60℃ and 90℃ for all four oils drop when the camshaft rotational speed is increased from 300 RPM to 500 RPM. This is shown in Fig. 24 and Fig. 25 respectively. This is consistent with the explanation given in the previous section that the increase in camshaft speed for each oil improves the lubrication conditions and therefore the frictional values drop.

The average frictional torque at 300 RPM and 500 RPM in Figs 26-27. shows that for each oil the increase in temperature increases the frictional torque values since the viscosity drops and the asperity contact increases. However, the frictional values for all four oils are different at every temperature. This is due to the different chemical composition of each oil. This chemical composition is generally not available in data sheets because it is a commercial trade secret and will vary among producers [19]. However, some remarks can be made in this regard. Firstly, the oil with lesser values of average friction torque may be showing this behavior because of the use of improved or altogether new friction inhibitors. Secondly, the two oils namely KIXX and ZIC which have overall lesser values of friction as compared to the other oils may be due to the use of such base oils and additives which maintain their structural integrity and chemical properties throughout the operating range.

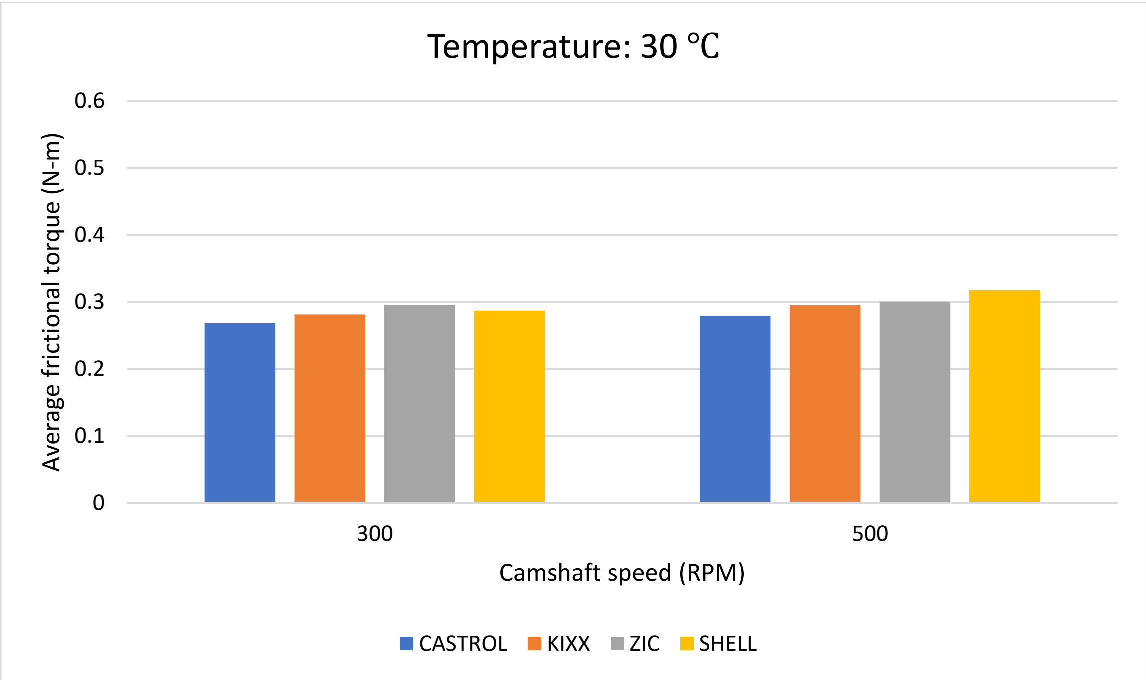

Figure 23. Average frictional torque at lubricant temperature: 30°C

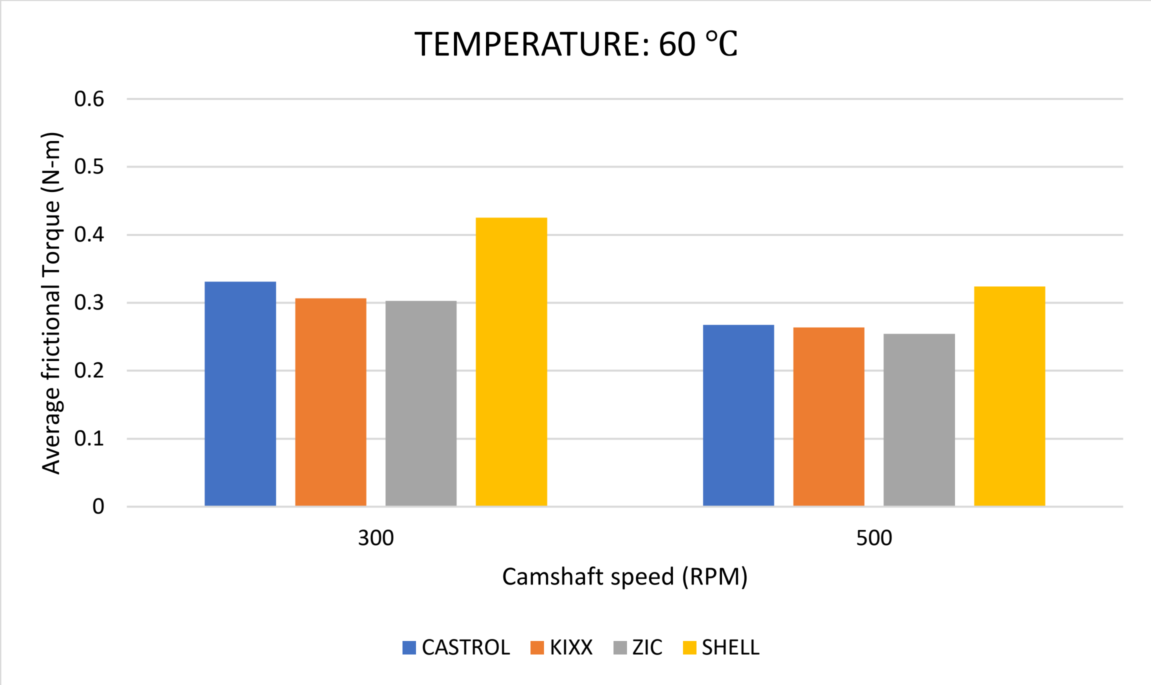

Figure 24. Average frictional torque at lubricant temperature: 60°C

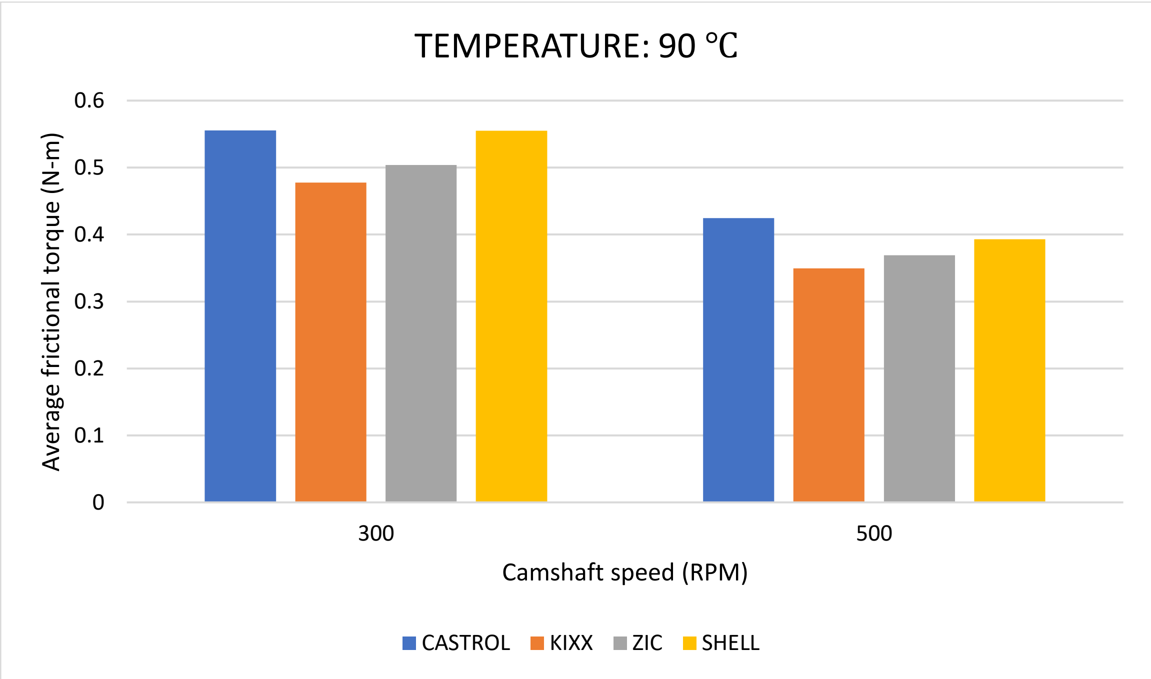

Figure 25. Average frictional torque at lubricant temperature: 90°C

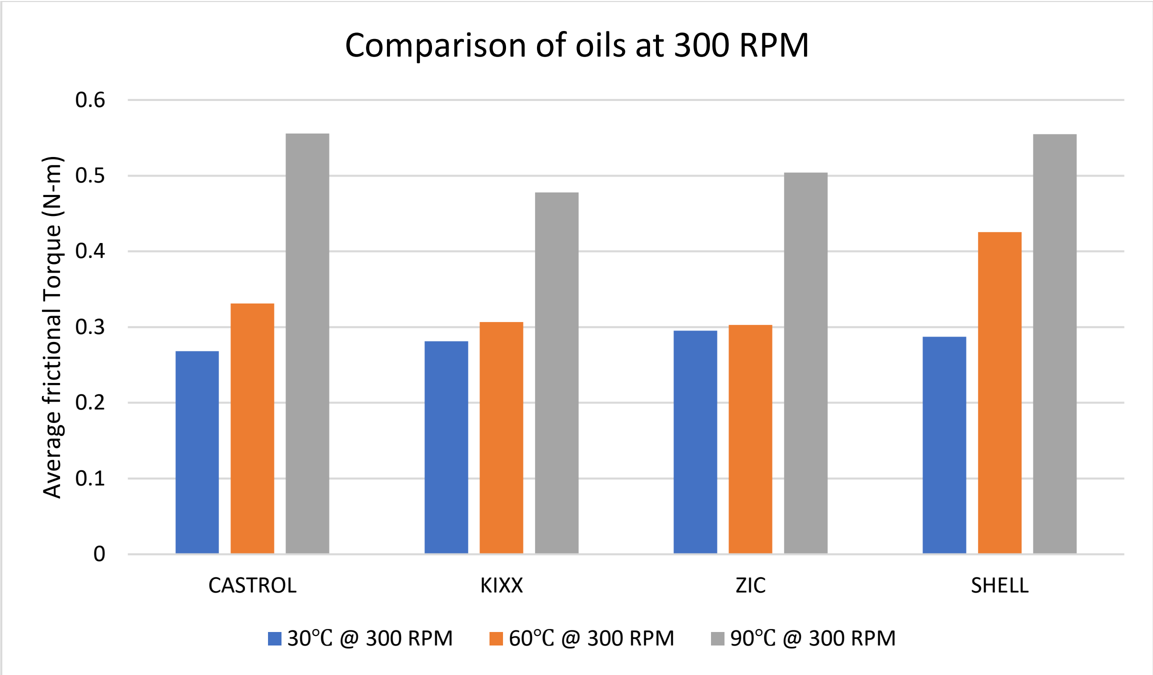

Figure 26. Average frictional torque at 300 RPM

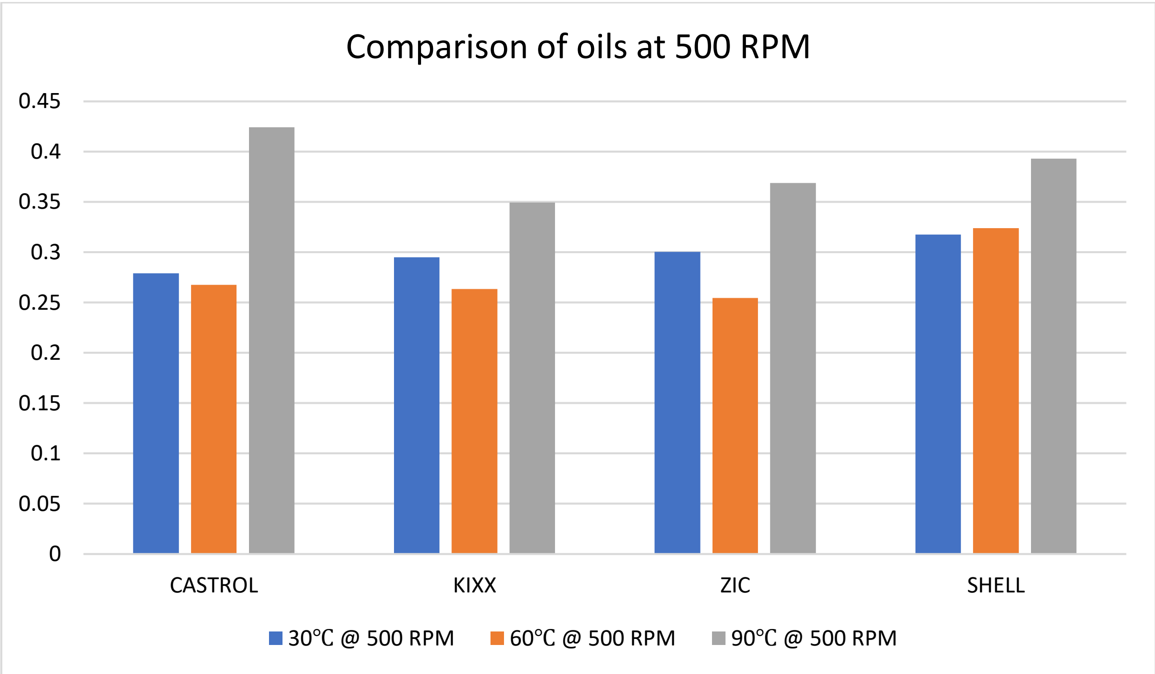

Figure 27. Average frictional torque at 500 RPM

Chapter 4: CONCLUSIONS

As shown in Fig 26 and Fig 27, Kixx G1 is the better oil in terms of improving the fuel economy of the engine because of the lower average frictional torque. Kixx G1 has lowest average frictional torque values at 90°C for both camshaft speeds of 300 RPM and 500 RPM. At 60°C and 300 RPM, Kixx G1 has lowest average frictional torque value alongside Zic X7. Although at 30°C and 300 RPM, Kixx G1 doesn't have the lowest frictional torque, it isn't much greater than the other oils. At 500 RPM, Kixx G1 is outperformed by ZIC X7 at temperatures of 30°C and 60°C but not by as big a margin as Kixx G1 outperforms ZIC X7 at 90°C. Since the temperatures of lubricating oil in real engine operation are much higher than 90°C and at the higher temperatures Kixx G1 seems to be outperforming the other oils, it is justified to say that it is the better of all the four oils to be used in real conditions.

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