

**IMPROVING EFFICIENCY IN DRIVE LINES: AN
EXPERIMENTAL STUDY ON CHURNING
LOSSES IN HYPOID AXLE**

by

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ABSTRACT

The research concerns improvement of the power efficiency of lubricated components in automotive drivelines such as transmissions and axles. Meshed gear pairs, rolling bearings, seals and oil churning by rotating components immersed in the oil are studied. The purpose of the research is to explore the most effective way of improving the efficiency in drivelines, focusing on an axle comprising hypoid gears and a differential assembly.

First, a study of the nature of losses affecting the efficiency of a simple spur gear box was carried out, and a model of friction and churning in a simple transmission was developed. Next, a detailed experimental study of oil churning losses in a hypoid axle from a four wheel drive road vehicle was carried out using the inertia run-down technique.

To perform the above experiments, a new test rig for measuring oil churning losses was designed, manufactured and commissioned. The test rig allowed a wide range of speed and lubricant parameters to be explored and was designed by the author for tests available at different roll and pitch attitudes. In addition, an "extended" housing, consisting of a modified gear case that accepted the same internal components as the production axle but which had much greater internal clearances, was designed and manufactured. This enabled the effects of different casing geometry and of internal baffles to be studied. Additionally, the extension housing was modified to investigate the oil flow inside the housing through its one transparent side for the understanding of the effect of oil flow on churning losses.

An investigation of design-related parameters influencing churning losses was carried out using the new test rig. Empirical equations for the churning losses, based on dimensional analysis, were developed to describe the test results. It was found that some combinations of baffles gave a significant reduction in losses. Supplementary tests were carried out using

transparent windows to visualise the oil flow. These identified some of the mechanisms responsible for the reduction in churning loss and suggested a number of practical methods by which churning could be reduced without compromising the lubricant supply to remote components. It is argued that these innovations can contribute to improving fuel efficiency and limiting oil temperature rise in all-wheel-drive vehicles.

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NOMENCLATURE

English Alphabets

A_c	Actual average contact area
A_{dfi}	Wetted face area of gear, $i=1,2$
A_o	Apparent contact area
A_v	Area of cavity between the teeth
A	Immersed surface area
A_1, A_2	Bottom area of oil sump in a parallelepiped rectangular housing
a	Addendum
b	Tooth face width of gear(Width of the crown wheel in Chapter 6)
C	Torque coefficient
C'	Gear shape factor which is relative to face width and tooth size
C_a, C_b	Factors
C_D	Dimensionless coefficient for the drag
C_{ch}, C_{churn}	Churning torque
C_m	Moment coefficient, Dimensionless drag torque
C_f	Dimensionless moment coefficient for front and rear face of windage loss of a spur gear
C_l	Dimensionless moment coefficient for teeth of windage loss of a spur gear
C_t	Dimensionless moment coefficient for windage loss of a spur gear
C_{sp}	Splash factor in Höhn's experimental formula
C_b, C_2	Constant
C_2	Constant from the laminar flow rate of incompressible fluid along a tube in Chapter 6, $C_2 = (\pi r^4)/8L$, Flow rate $\dot{Q} = \frac{\pi r^4}{8L} \frac{P}{\eta} = C_2 \frac{P}{\eta}$ where P is pressure, r is a radius of tube and L is a length of tube
c	Bearing clearance in Chapter 6
D	Element diameter
D_f	Root diameter of gear
D_p, d_m	Pitch diameter

D_M	Mean blade diameter
E_{total}	Total efficiency
E_{ef}	Efficiency
e_a	Axial clearance
e_{al}	Axial clearance between inclined surface of baffle and gear side of crown wheel
e_{ar}	Axial clearance between flat surface of baffle and rear side of crown wheel
e_r	Radial clearance
F	Force
F_{dpi}	Drag force acting on the periphery of a spur gear, $i=1,2$
F_w	Face width
Fr	Froude number ($= \omega^2 R_p / g$)
f_B, f_W	Average coefficient of friction
f_g	Gear dip factor (ration of dipping depth to element outer diameter)
f_l	A coefficient depending on bearing design and relative bearing load
f_o	A coefficient depending on bearing type and lubrication method
f	Natural frequency (cycle/second)
G	The centre of mass for a platform
G_r	Gear ratio
g	Acceleration of gravity
H	Internal height of housing
H_B	Blade height
H_{tooth}	Tooth height
h	Immersion depth into fluid
hc	Central lubricant film thickness
h_{max}	Maximum immersion depth
h_0	Reference value of immersion depth (=10mm) in Chapter 2
h_0	The distance between the centre of the crown wheel and the bottom of oil sump in Chapter 6
h_1, h_2	Immersion depth of pinion and gear respectively
h_1, h_2	Oil depth to the bottom of oil sump in Chapter 6
I_i	Moment of inertia with notation i [$\text{kg} \cdot \text{m}^2$]
I_o	Total polar moment of inertia

K	Experimental coefficient
k	Spring coefficient
L	Internal width of housing (Chap. 2), Length of suspension string (Chap.3)
L_c	Contact line length
L_{SB}	Friction loss of spiral bevel gear
L_W	Friction loss of worm gear
L_1	Distance between pitch point and intersection of contact path and base circle of spur pinion
L_2	Distance between pitch point and intersection of contact path and base circle of spur gear
M	Total torque loss of bearings, discs or gears
M_f	Fluid friction drag torque in Boness's formula
M_g	Windage torque loss of spur gear
M_l	Load friction torque of bearings
M_n, m	Mass
M_p	Torque of pinion
M_t	Windage drag torque in the formula presented by Mann and Marston
M_v	Viscous friction torque
M_w	Torque of crown gear
m_n	Module
mo	Average square value of the height of roughness
m_s	The strength of the source
N	Normal load
N_c	Average number of cavities
N_s	Rotational speed [revolution per a minute], RPM
P	Power loss
P_b	Base pitch
P_{ch}, P_{churn}	Churning power losses
P_{di}	Churning drag power losses associated with the interaction of the surrounding medium (oil), $i=1,2$
P_{dfi}	Churning power losses due to drag on the faces of a gear, $i=1,2$
$P_{dfi}^{(L)}$	P_{dfi} in laminar regime, $i=1,2$

$P_{dfi}^{(T)}$	P_{dfi} in turbulent regime, $i=1,2$
P_i	Power loss for i
P_{input}	Input power
P_{rfi}	Churning power losses due root filling inside one tooth cavity, $i=1,2$
P_{dpi}	Churning power losses due to drag on the periphery of a gear, $i=1,2$
P_{load}	Load dependent losses
P_{moy}	Mean contact pressure
$P_{non-load}$	Load-independent losses
P_{shaft}	Notional power loss from the oil in the bearings
P_{win}	Windage power losses
P_{wheel}	Wheel pumping loss of oil
R	Effective radius of curvature $R = r_1 r_2 / (r_1 + r_2)$
R_n	Radius of harmonic circular motion
R_o	Tip radius, Outer radius
R_p	Pitch radius
R_w	Radius of test wheels
Re	Reynolds number ($= \omega R_p^2 / \nu$)
Re^*	Critical Reynolds number
R^*	Critical radius separating laminar and turbulent zone
r	Radius of circular plate or disc
r	Radius of the bearing in Chapter 6
r_c	Ratio of contact
r_{oi}	Outer radius, $i=1,2$
r_{ri}	Root radius, $i=1,2$
r_i	Pitch radius of cylinder 1 and 2
S	Surface roughness (Chap. 2), the characteristic surface of the body (Chap. 6)
S_{rf}	Roughness factor
S_m	Immersed surface area
T	Total loss torque
T_i	Drag torque with notation i [N·m]
$T_{bearing}$	Bearing drag torque
T_{ch}, T_{churn}	Churning torque

T_{gear}	Gear torque at which efficiency is desired
$T_{no\ oil}$	Drag torque measured after draining the oil in the housing
T_{max}	Maximum gear torque
T_{period}	Period of suspension
T_{rfi}	Torque loss due to the swirling flow of the lubricant inside the cavity, $i=1,2$
T_{shaft}	Notional torque from the oil in the bearings
T_{seal}	Drag torque caused by seal
T_{total}	Total drag torque
T_{wheel}	Wheel pumping torque
t	Time [sec, min]
u	Speed ratio
V_g	Total immersed volume
V_l	Immersed volume of gear
V_m	Volume of the oil bath
V_o	Oil volume
V_r	Rolling velocity, $V_r = u_1 + u_2$
V_s	Relative sliding velocity
V_t	Pitch line velocity
V_{t0}	Reference value of pitch line speed (=10m/s)
V_0	Total oil volume
V_1, V_2	Oil volume of oil sump
$\dot{V}_1$	Oil pumping volumetric flow rate
$\dot{V}_2$	Oil volumetric flow rate through a capillary tube
v	The mean flow velocity
W'	Unit load
X_A	Profile shift coefficient
y	Height/depth of the flow
Z	Number of teeth

Greek Alphabets

α	Pressure viscosity coefficient
α_p	Pressure angle at pitch point for spur gears
α_A	Pressure angle at tooth tip
β	Helix angle
β_a	Arc of approach of equivalent helical gears
β_r	Arc of recess of equivalent helical gears
ϵ	Ratio of arc of admission to total circumference
ξ	Reduction factor for fluid flow analysis for spur gears
η, μ_d	Dynamic viscosity [cP=1mPa·s=0.001 Pa·s= 0.001 N·s /m ²]
η_0	Viscosity at zero pressure (usually taken to be 1atm)
η_p	Viscosity at pressure p
θ	Angular displacement [degree]
$\ddot{\theta}$	Angular acceleration [rotation/min ²]
γ	Experimental coefficient
γ_p	Angle of generatrix of pinion
λ	Box space function
ν	Kinematic viscosity [cSt=1mm ² /s]
μ	Friction coefficient
τ_{dry}	Shear stress due to dry friction
τ_{fluid}	Shear stress due to fluid friction
ρ	Fluid density
φ	Immersion angle (= $\cos^{-1}[1 - h/R_o]$)
Φ	Oil mixture function
Φ_T	Thermal correction factor
ϕ	Pressure angle (before degrees), Diameter (before numbers)
ϕ_n	<i>Normal pressure angle</i>
ω	Angular velocity [radian/second]
ω_p	Angular velocity of pinion
ω_w	Angular velocity of crown gear

ω_n	Angular frequency [radian/second]
Ω	Rotational speed [RPM]
Ψ	Spiral angel of spiral bevel gear at middle of tooth face
Ψ_G	Men gear spiral angle
Ψ_P	Mean pinion spiral angle
Ψ_l	Coefficient
Ψ_i	Exponential coefficient, i= 1,2, 3...

Subscripts

$1,2$	corresponding to 1 and 2
$ch, churn$	churning
win	windage

ACRONYMS AND ABBREVIATIONS

CCW	counter- clockwise, anticlockwise
CW	clockwise
EHD	elastohydrodynamic
EHL	elastohydrodynamic lubrication
EP	extreme pressure
Ext.	extension
etc.	and so forth
Dia.	diameter
FWD	forward direction
IC	internal combustion
LCV	low-carbon vehicles
MIN	minute
MSE	mean squared error
PC	personal computer
RPM	revolution per minute
rev/min	revolution per minute
RMSE	root mean squared error
RVS	reverse

CHAPTER 1

INTRODUCTION

1.1 Background

Machine efficiency has become an important issue in energy saving and for decades a lot of effort to improve efficiency and reduce fuel consumption in cars has been made in relation to combustion engines and transmission performance of vehicles. More efficient engine technology has been mainly developed to get higher efficiency and reduce harmful toxic emissions in addition to the search for new energy sources to substitute for fossil energy resources. To obtain this higher efficiency in vehicles, the power source such as combustion engine and its durability have been mostly studied by tribological measures of reducing friction and wear so far. On the other hand, the vehicle driveline, which connects the transmission to the road wheels and transfers the power from the energy generator to the final wheel axis, has recently been focused to improve fuel economy.

As well as the fuel economy in the view of high fuel prices, environmental pressures and government regulations based on the Kyoto Protocol (Black, 2009) on climate change urging many countries to cut their carbon dioxide emission are becoming more stringent in terms of allowable emissions in the form of hazardous gases and particles released to the environment from the use of vehicles (Taylor, 2002). This also leads to a need for low power-loss drivelines as well as powertrain. The tree diagram, Figure 1.1 (Bartz, 2000; Pinkus & Wilcock, 1977 and 1981) shows the energy distribution in passenger cars during an USEPA(U.S. Environmental Agency) driving cycle, combining urban and highway driving cycles and finally only 12 percent of the input energy remains at the wheels. It also reveals the tribologically originated losses are composed of 3 percent of losses caused from gear boxes (transmission 1.5% and axles 1.5%) and 7 percent of friction losses from engine, which are associated with engine and gear oils. If the losses in transmission and axles were reduced to zero, the useful available energy could be augmented by 3 %, from 12 % to 15 % - an increase of a quarter of final drive efficiency. Therefore, the efficiency of the automotive drivelines relative to that of the total vehicle has become no less important than any other components and has been of interest to mechanical engineers.

However, the continuous efforts to increase the efficiency of vehicles and to secure further reductions in carbon dioxide per car have been recently made, and the total efficiency of vehicles currently improved better than the figures in Figure 1.1. The best diesel IC (Internal Combustion) engines now have an overall efficiency of up to 45 % compared to only 30 % 15 years ago. Recent incremental improvements on the technologies for a conventional IC engine and other elements of a powertrain have been developed as presented in Table 1.1 (Report of IMechE, 2009). Furthermore, new technology solution for LCV(Low-Carbon Vehicles), such as advance petrol/diesel, full/plug-in hybrid, hydrogen, fuel cell and battery only electric cars, will substitute theconventional internal combustion engine in a near future (Report of IMechE,

2009). Accordingly, the efficiency of drivelines for LCV as well as the existing IC engine requires to be improved and its importance is increasing.

Table 1.1 Efficiency saving from new technologies for powertrain of vehicles (King, 2007)

Category	Technology	Efficiency saving
Engine	Direct injection and lean burn	10 ~ 13 %
	Variable valve actuation	5 ~ 7 %
	Downsizing engine capacity with turbocharging or supercharging	10 ~ 15 %
	Ricardo 2/4 stroke research engine	27 %
Transmission	Torotrak continuously variable transmission	15 ~ 20 %
	Dual clutch transmission	4 ~ 5 %
Mild hybrid and others	Stop-start	3 ~ 4 %
	Stop-start with regenerative braking	7 %
	Electric motor assist	7 %
	Reduced mechanical friction components	3 ~ 5 %

To evaluate the efficiency of drivelines, the prediction of power losses in lubricated rolling-sliding contacts such as those of the running components in the drive lines presents a significant challenge because it is coupled with heat transfer and lubricant properties, leading to a complex nonlinear problem. Moreover, this kind of efficiency model must include losses caused by tooth meshing, bearings, shaft seals, oil churning and pumping effects. Out of those factors influencing efficiency, the lubricant plays an important role in drivelines and is required to provide not only adequate means of cooling as well as a greater performance over a wide range of temperatures, but also a higher degree of wear protection of rotating components through a load-carrying lubricant film. It is also required to provide excellent lubrication even under extreme conditions of high load-carrying, high rotating speed and high

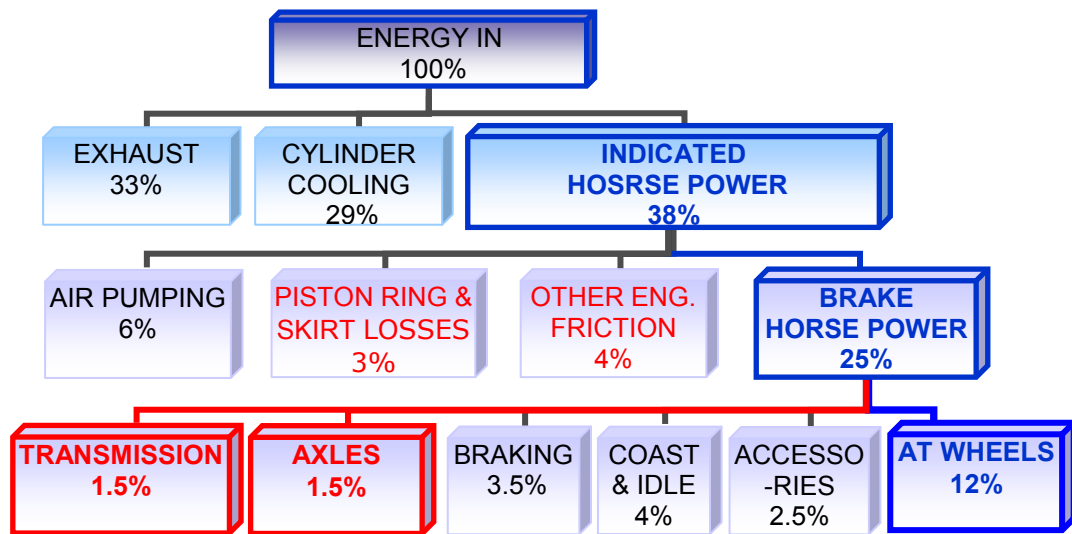


Figure 1.1 The energy distribution in a passenger car during an EPA combined cycle (Bartz, 2000; Pinkus & Wilcock, 1977 and 1981)

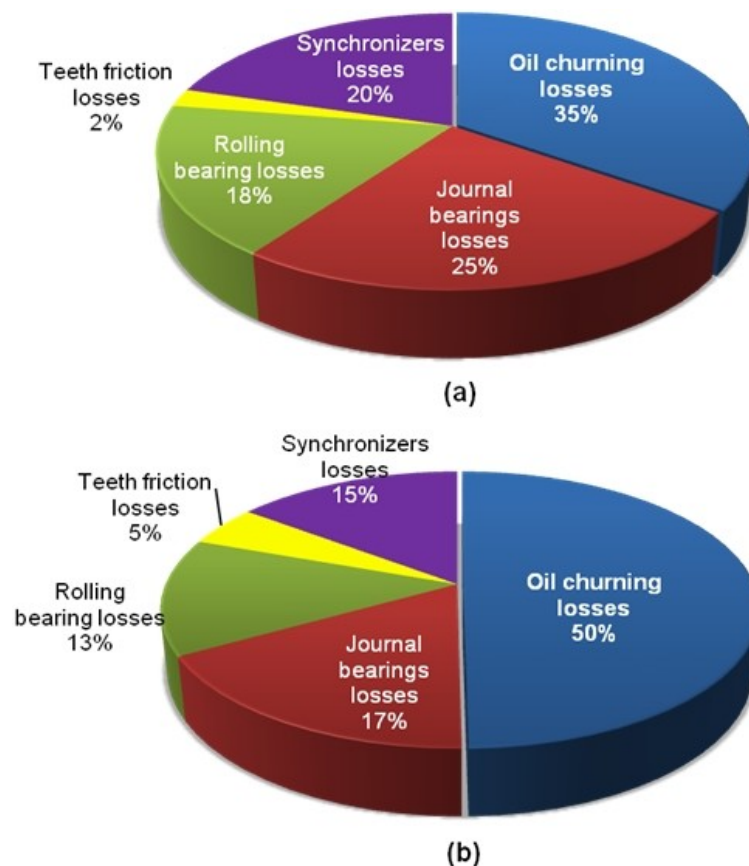


Figure 1.2 Power losses distribution in six-speed transmission under the condition of 5,000 rev/min, 30 Nm and 6th gear engaged: (a) Total power lost 4,082 W at oil bulk temperature 40°C (b) Total power lost 2,861 W at oil bulk temperature 80°C (Changenet, Oviedo-Marlot, & Velez, 2006)

or low temperatures. This lubrication of rotating components such as gears and bearings in gear boxes is achieved by dipping into an oil-filled sump, grease lubrication or forced feed system in which oil is introduced into the unit through jets under pressure. A dip lubrication, which is also called as splash type, is usually used in gear boxes rather than forced feed lubrication called jet lubrication because splash lubrication is not only much simpler in design and manufacturing, but also less expensive than jet lubrication. Moreover, the operation of the necessary feed pump for oil injection in the jet lubrication system needs to consume a certain additional amount of engine power. It is generally known that grease lubrication is appropriate for a low tangential speed of gears below 5 m/s while oil jet lubrication is suitable for speeds higher than 15 m/s.

Power losses occurring in splash lubricated gear boxes can be generally divided into load dependent losses and non-load dependent losses. Load dependent losses take place only in the friction components such as seals, bearings and gears that transmit the load. On the other hand, non-load dependent losses consist of churning losses, windage losses and rotating seal loss, and do not carry a transmitted power but occur through the rotating components. Accordingly, non-load dependent losses are often referred to as load-independent losses or spin power losses. Townsend (Townsend & Kubo, 1992) defined churning losses as the losses caused by oil drag and entrapment of lubricant in the gear mesh. The lubricant is trapped, squeezed and splashed as the gear teeth come to the meshing zone, squeezed lubricant is then splashed out from the meshing zone, and the lubricant will also cause resistance as the gears move into it. Windage loss refers to the spin losses coming from the frictional resistance to air on the rotating gear or disc due to the surrounding atmosphere comprised of only air or mixture of air and oil mist. Windage loss can be assumed to apply to a dry sump in which the rotating components are not in contact with oil, while churning losses refer to viscous drag of rotating components partially submerged in an oil bath.

Correspondingly, Yamamori et al. (Yamamori, Saito, & Yoneda, 2005) in Toyota Motor Corporation presented that the ratio of churning losses to overall energy losses in a manual transmission was as high as 38 % in EC mode driving and an attention to reduce the churning losses was required for the improvement of vehicle fuel economy. Oil churning losses also contribute mainly to overall inefficiency in the six gear-engaged transmission leading to a significant rise in a bulk temperature, and account for 35 and 50 % of total power losses at each temperature as illustrated in Figure 1.2 (Changenet et al., 2006). This figure shows that churning losses kept constant as 1430W at each temperature although the whole losses were changed. Thus, the portion of churning losses increased and that of bearing losses were reduced as the oil bulk temperature rose.

Although oil churning is thought of as the main energy dissipation source remaining to be improved, a large number of previous studies have been done regarding load-dependent losses so far, but there is little agreement as to research on churning losses. Furthermore, churning losses for hypoid gears running in a dip-lubricated system have not been studied, except only research on churning losses for spur gears.

The research reported in this thesis concerns the churning losses in dip lubricated drivelines, particularly axles composed of a pair of hypoid gears, which change the drive direction through a 90° angle and transmit engine power to the wheels, slowing the rotational speed of the transmission by means of the final reduction gear ratio.

1.2 Aims and Objectives

The research reported in this thesis concentrates on measurement of churning losses in gear boxes such as mechanical transmissions and especially axles, inspection of influencing

variables on them and prediction of them with a considerable accuracy. Through inspection of the several kinds of factors causing losses, a greater understanding of the mechanisms would be obtained; which would lead to higher efficiency of gear boxes, and an understanding of the underlying principles involved.

An experimental rig was built to measure the churning losses from rotating components dipped in oil in a gear box such as rear axles at these interfaces. Areas of interest include mainly churning losses caused by the gear pumping effect. In addition, frictional losses from gear teeth and bearing losses in gear boxes are considered, but a special attention will be focused on the churning losses from axles composed of a pair of hypoid gears.

It was planned to investigate how efficiency characteristics for churning losses are determined by means of experiments based on rig testing. The main objective of the research was to identify the key parameters influencing lubricant-related efficiency of a hypoid axle and finally to define design guidelines for improved lubricant efficiency. Furthermore, the ultimate goal is to explore the feasibility of improving the efficiency by means of changes to lubricant rheology and to the lubrication system.

The purpose of the research was as follows:

- (i) To build a test rig to enable a quantitative and qualitative measurement of churning losses in gear boxes, especially axles.
- (ii) To identify the detailed churning behaviour of axles composed of a pair of hypoid gears.
- (iii) To investigate the controlling variables and their influences on churning losses.

- (iv) To develop an empirical formula for churning losses for hypoid axles.
- (v) To find a way of decreasing churning losses.

1.3 Outline of the Thesis

The structure of the thesis is organised as follows:

- (i) Chapter 2 reviews the literatures for load-dependent losses such as gear teeth friction losses and bearing losses, and non-load dependent losses. Previous reports of churning losses models in spur gears are presented.
- (ii) Chapter 3 discusses the experimental details of the present research. The description of the newly built rig and shrouded housing, the preparation of the test components and the details of the experimental set-ups and procedures are presented.
- (iii) Chapter 4 presents and discusses the results obtained from the experiments on the mass production axle housing to identify dominant variables influencing on churning losses.
- (iv) Chapter 5 presents and discusses the results obtained from the experiments on the newly built-up extension housing to identify dominant variables influencing on churning losses.

- (v) Chapter 6 covers the results explained in Chapters 4 and 5, and presents the discussion through the comparison of the results.
- (vi) Chapter 7 states the main conclusions of the present research, and suggests some recommendations for future work.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

In chapter one, the background of the research was described and the research objective was defined. The current chapter reviews some of the important principles and theories, relevant background and experimental studies on power losses that have been carried out so far, which are required to understand how the investigation and research on churning losses of hypoid axles will be guided and developed.

In this chapter, an overview of automotive drivelines is presented, followed by a brief introduction to friction losses for gears and bearings as load-dependent power losses. The latter is reviewed, since it is an important loss under the condition of a high-transmitted torque in studying the efficiency of automotive drivelines, before turning to examine spin losses. Thereafter, the experimental studies to date on windage losses of discs and spur gears are

summarised. Furthermore, the experimental studies of churning losses for spur gears based on Von Karman's theory of the rotating disc are reviewed as well as the effects of parameters on churning losses. Finally, churning losses performed in helical gears and hypoid gears are shortly reviewed in respect of their affecting variables and churning losses relationships.

2.2 Efficiency and Geometry in Automotive Drive Lines

The power from a car engine is transferred through the transmission to the axle, via the drive shaft, which is connected to the pinion gear inside the axle housing, which is filled with lubricant. The pinion gear rotates according to the driveshaft, and turns the ring gear, which is connected to both the shafts extended to the ground wheels. Those drivelines are schematized according to driving wheels in Figure 2.1. Nowadays all-wheel-drive vehicles are widely used having a differential located between each set of drive wheels for both on and off-road driving. The axles compose of a pair of hypoid gears that not only transfer the power from the engine to the wheels, but also allow the wheels to rotate at different speeds while a car is turning. This function of axles for differentials is shown in Figure 2.2. The drive pinion gear and the ring gear rotating inside their axle housing are supported by bearings. Therefore, power losses in the axle are mainly caused by the bearings; churning of lubricant inside the axle housings; and frictional losses in the contacts between pinion and ring gear, that usually use spiral bevel gears or hypoid gears.

As previously referred in Chapter 1, the power losses in drivelines, P_{total} are generally decomposed into load-dependent losses, P_{load} and load-independent losses called spin power losses, $P_{non-load}$, and so the total power losses can be described as the following equation.

$$P_{total} = P_{non-load} + P_{load} \quad (2.1)$$

Those power losses can be respectively divided as shown below.

$$P_{non-load} = P_{churn} + P_{win} \quad (2.2)$$

where P_{churn} are the churning power losses caused by the viscous drag from lubricant in a dip

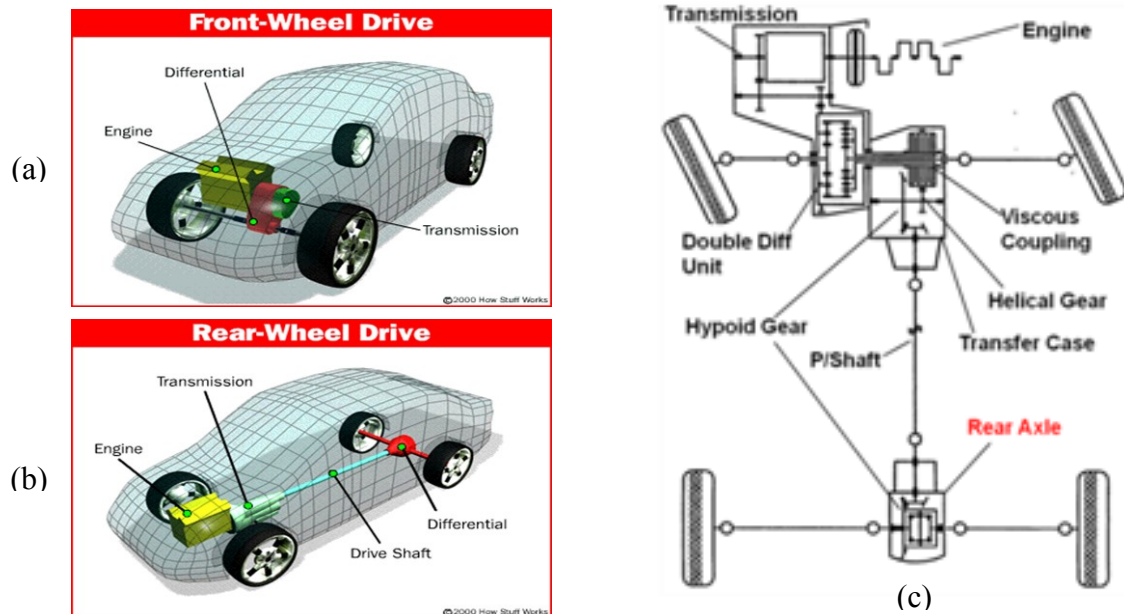


Figure 2.1 Geometry of automotive driveline (a) Front wheel drive (b) Rear wheel drive (<http://auto.howstuffworks.com/>) (c) a schematic of power train and drivelines in four-wheel-drive cars

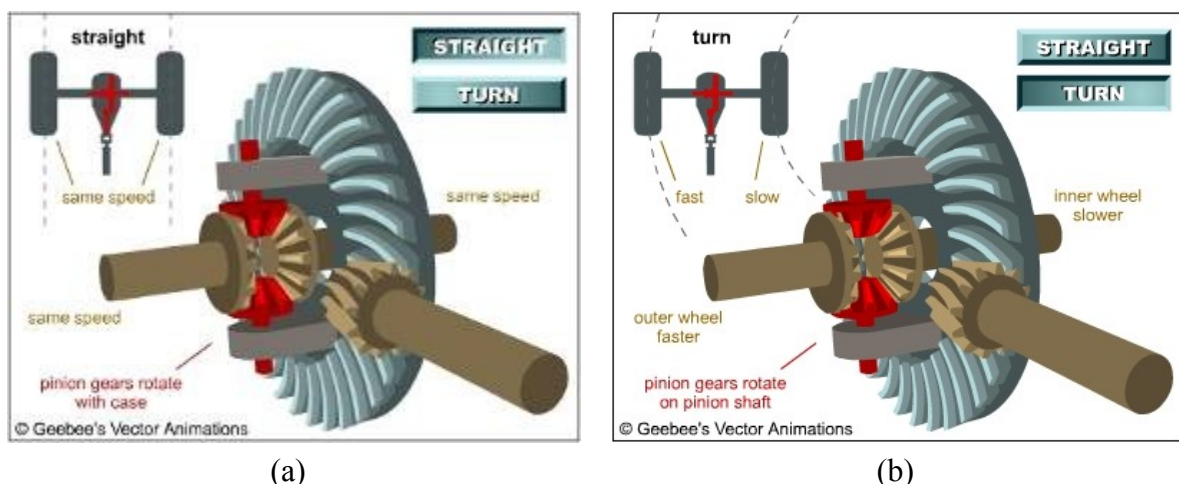


Figure 2.2 Function of axles as differential (a) Straight driving (b) Turn driving (<http://auto.howstuffworks.com/>)

lubrication, and P_{win} are the air windage power losses caused by air friction from a high-speed rotation of a disc or gear. Load independent power losses can be divided largely into churning losses and windage losses, whereas load dependent losses can be expressed by Equation 2.3.

$$P_{load} = P_{gear\ friction} + P_{bearing\ friction} + P_{auxiliary} \quad (2.3)$$

where $P_{auxiliary}$ are the losses caused by auxiliaries. Load dependent losses, which are generally based on the friction of components under the load, consist of gear friction loss and bearing loss. However, load independent losses are still present when power is transmitted.

Churning power losses are one source of non-load power losses predominant under the condition of high speed and low torque. Barzaghi (Barzaghi, Berti, & Gommellini, 1995) experimentally measured the mechanical efficiency and churning losses for an automotive manual transmission with a reference mineral oil operating in four regimes as shown in Table 2.1. This table represents the relative contribution of churning losses to total losses, which is largest in low load and high-speed conditions compared with other operating conditions. The lowest mechanical efficiency at low load and high-speed conditions represents how much churning losses influence the total efficiency and shows the demand to reduce churning losses in axles for increased power density and fuel economy of drivelines.

Therefore, these energy losses in Equations 2.2 and 2.3 having an important influence on the efficiency of gear boxes such as axles of vehicles, can be also classified traditionally into two groups: oil churning losses and sliding & rolling friction losses. Oil churning losses are mainly affected by the rotational speed of the gears and the properties of lubricant (Bronshteyn & Kreiner, 1999; Michaelis & Höhn, 1994), while the sliding & rolling friction losses are dependent on the transferred load. In addition, from simple physics, torque loss times angular speed generally equals power loss.

Table 2.1 Mechanical efficiency and relative churning losses of a gear transmission under different regimes (Barzaghi et al., 1995)

Parameter / Regime	Low load Low speed	High load Low speed	Low load High speed	High Load High speed
Temperature of oil in steady state condition [°C]	50	85	66	108
Mechanical efficiency [%]	91.4	94.5	87.33	95.02
Relative churning losses (if 100% assigned to high load, high speed regime) [%]	75	40	165	100

Thus, the efficiency of a pair of gears can be made up from the input and output torques combined with the speed ratio of the engaged gear, which is expressed by

$$\text{Efficiency of a pair of gears} = \frac{\text{Output torque}}{\text{Input torque}} \times \text{Ratio of the engaged gears} \quad (2.4)$$

Output power is obtained by subtracting power losses such as sliding power losses and churning power losses from input power. Therefore, the efficiency can be derived by

$$E_{total} [\%] = \frac{P_{input} - (P_{gear\ friction} + P_{bearing\ friction} + P_{seal} + P_{churning} + P_{windage})}{P_{input}} \times 100 \quad (2.5)$$

2.3 Friction Losses

Friction losses, as mentioned above, are mainly divided into gear teeth friction loss and bearing friction loss. As shown in Table 2.1, such losses are always load-dependent. This

friction force is proportional to the normal load according to Amontons' friction law as shown below.

$$F = \mu \times N \quad (2.6)$$

where F is a friction force, μ is a friction coefficient and N is a normal force of body acting on the plane. This equation means that friction loss of the engaged gears in gear trains is dependent on load, contact surface roughness and lubricant viscosity. The low friction system with a good durability accordingly requires lower capacity of lubrication for gears. As a result of this reduction in demand on lubrication, load-independent losses of lubricant may be also minimised slightly by decreasing the friction losses. Conversely, the increasing distribution of pumped lubricant in the oil sump for better lubrication and durability, can lead to increased churning losses despite the reduction in friction losses. Thus, friction losses should be minimized, which is associated with the need for reducing spin losses. This is the focus of the author's literature review of gear friction losses.

A large part of previous theoretical and experimental studies on geared box efficiency focus primarily on friction losses caused by the rolling and sliding contacts between meshing gears. This is because gear efficiency can be found from the level of sliding friction that occurs between gear teeth. Therefore, theoretical research on friction losses has been dedicated to investigating friction coefficients in gear mesh, with some being experimentally validated. For this reason, a review of the literature on gear friction and bearing friction losses is presented in the next sections.

2.3.1 Contact in Spur Gears: Gear Friction Model

Initially, the equations for tooth efficiency of spur gears were derived assuming the coefficient of friction to be constant. Tuplin (1962) derived equations for instantaneous and overall tooth efficiency when one pair of teeth is in contact. In the following year, an equation for overall efficiency for two pairs of teeth including the effect of variation of coefficient of friction across the path of contact and different values over the path was given by Buckingham (1963). Following this, Martin (1972) considered different load ratios and speeds and produced a simplified equation for a spur gear friction model by assuming a constant coefficient of friction μ when the contact ratio was bigger than 1.0.

A study (Benedict & Kelly, 1961) using an empirical friction coefficient was also done. Since no sliding exists in the direction of the contact line for involute teeth in spur gears, the sliding velocities in the empirical formula of the friction coefficient are normal to the line of the contact and are uniform along it due to their geometry. However, in all gears including spur, helical and hypoid gears, the radius of curvature changes along the surface of contact of teeth.

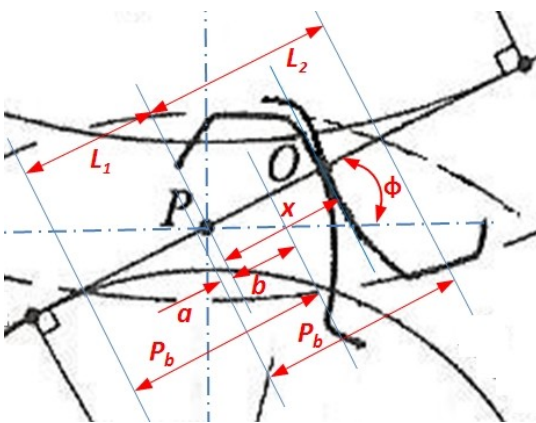


Figure 2.3 The notation for friction loss of an involute spur gear (Johnson, 1985;

To simplify the analysis of spur gear contact, it is assumed that the coefficient of friction in the contact area is constant. This constant friction coefficient and the normal force acting at a given contact point were used to calculate a tangential friction force by Denny (1998) and Pedrero (1999). In general, the friction loss for a pair of

mating spur gears can be calculated theoretically. Figure 2.3 (Johnson, 1992) represents the

notations for obtaining a friction loss of spur gear tooth. It is assumed that the work transmitted is $N \times P_b$. The notations L_1 , L_2 and contact ratio r_c in Figure 2.3 are defined regarding base circle pitch P_b as shown below.

$$a = P_b - L_1, \quad b = P_b - L_2 \quad (2.7)$$

$$r_c = (L_1 + L_2)/P_b \quad (2.8)$$

Where P_b is base circle pitch, subscript 1 refers to the pinion and subscript 2 the gear. When two pairs of involute teeth and one pair of spur gears are in contact, the dissipation losses are calculated respectively as shown below using sliding velocity V_s and $V_c = dx/dt = \omega_1 r_1 \cos \phi = \omega_2 r_2 \cos \phi$. The equation (Johnson, 1992) for friction power loss is calculated as,

$$\int F |V_s| dt = \int F \frac{|V_s|}{V_c} dx = 0.25 \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \sec \phi \times N \times \mu_2 [(L_1^2 + L_2^2) - (a^2 + b^2)] \quad (2.9)$$

$$\int F |V_s| dt = \int F \frac{|V_s|}{V_c} dx = 0.5 \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \sec \phi \times N \times \mu_2 \times (a^2 + b^2) \quad (2.10)$$

Then, the frictional power loss can be developed as power dissipation for involute spur gear can be divided by power transmitted.

$$\frac{\text{Power dissipated}}{\text{Power transmitted}} = 0.5 \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \sec \phi \times \left[\frac{\mu_1 (L_1^2 + L_2^2) - (2\mu_1 - \mu_2) P_b^2 (r_c - 1)}{P_b} \right] \quad (2.11)$$

This equation shows that the efficiency decreases as base pitch P_b increases. Moreover, the increased addendum of gear teeth by higher involute contact ratio, which is known to reduce vibration amplitude and noise, contributes to higher sliding velocities, which make it a more significant friction loss as a result, although face contact ratio influences nearly nothing on the efficiency due to same applied load. Namely, a good efficiency of gears is achieved by making μ and $L_1^2 + L_2^2$ as small as possible, and pitch circle radius r and base pitch P_b as large as possible.

The gear contact analysis model uses gear design parameters such as surface finish, predicted load distribution, gear geometry and lubricant parameter to determine the instantaneous friction coefficient value at every point of contact line in the teeth surface. In actuality, the coefficient of friction is known not to remain constant during the movement of tooth along the path of contact and changes according to the varied speed and load. It also takes place in gear trains that different coefficients of friction occur according to the regime of lubrication being either mixed lubrication or mainly EHL(Elastohydrodynamic lubrication). Therefore, most of empirical formulae for coefficient of friction, as shown below, are deduced from curve fitting of data, which are collected from an EHL test rig such as two-disc machine consisting of two independently driven discs mounted on the spindle of a motor. The empirical formulae for the coefficient of friction μ are followed by the general form as shown below.

$$\mu = f(S, \eta, V_s, V_r, W', R) \quad (2.12)$$

where S is surface roughness, η is dynamic viscosity, V_s is relative sliding velocity, V_r rolling velocity and W' is unit load and R is effective radius curvature. These variables were used in the empirical formula of μ are shown in Table 2.2. O'Donoghue and Cameron (1966) found that friction coefficient was a function of surface finish, therefore S is the initial surface of roughness in their formula. The Benedict and Kelly's formula (1961), which is widely used in gearing, not only gives an overestimate of tooth friction in vicinity of the pitch line of gears as the sliding velocity V_s approaches zero, but also under predicts values for high slide-to-roll ratios because of the log function. They considered the oil sump temperature, the ambient pressure, but the piezoviscosity and limiting shear stress were ignored. The prediction of instantaneous efficiencies comparing the measured values was also tried by Yada (1973, 1974) and Radzimovsky (Radzimovsky & Mirarefi, 1975). Yada measured the friction loss by the heat transmitted from the gear to the oil in which the gear was immersed whereas

Radziovsky made measurements of the average coefficient of friction for gear speeds up to 1000 rev/min. Thereafter, Anderson and Loewenthal (1982) calculated gear mesh loss for a wide range of gear geometries and operating conditions by using the sliding and rolling velocities occurring at the point along the path of contact and using Benedict-Kelley's friction coefficient (given in Table 2.2).

Table 2.2 Friction coefficient formulae

Author	Equation	Application range
O'Donoghue and Cameron (1966)	$\mu = 0.6[(S + 22)/35][V_s^{1.8} V_r^{1/3} V_r^{1/6} R^{0.5}]^{-1}$	
ISO TC60	$\mu = 0.12[W'S/(R V_r \eta)]^{0.25}$	
Kelly and Lemanski (1967)	$\mu = \left[\frac{0.0099}{(1 - S/45)} \right] \log_{10} \left[\frac{50.8 \times W'}{\eta V_s V_r^2} \right]$	
Benedict and Kelly (1961)	$\mu = 0.0127 \left[\frac{50}{(50 - S)} \right] \log_{10} \left[\frac{3.17 \times 10^8 W'}{\eta V_s V_r^2} \right]$	$\frac{50}{(50 - S)} \leq 3$
Diab, Ville and Velex (2006)	$\mu = \frac{1}{P_{moy}} \left[\frac{A_c}{A_o} \tau_{dry} + \left(1 - \frac{A_c}{A_o}\right) \tau_{fluid} \right]$ $\frac{A_c}{A_o} = \frac{1}{2} \left[\frac{1}{2} \left(1 - \operatorname{erfc}\left(\frac{\Phi_T hc}{\sqrt{2mo}}\right)\right) \right]$	

Then, the calculation of spur gear mesh loss was also extended to apply to the non-standard geometry of spur gears with modified addendums by Anderson and Loewenthal (1986). Recently, Diab et al. (2006) suggested an alternative friction law based on Greenwood-Tripp-Mikic theory (Greenwood & Tripp, 1971; Mikic, 1974), accompanied by a large number of experimental traction curves, which combined the shear stress τ_{fluid} caused by the fluid friction with the shear stress τ_{dry} generated by roughness interactions. They also revealed that

Benedict-Kelley formula gives a good result with some discrepancies near the pitch line, while Kelley-Lemanski formula appears to overestimate the real value of friction coefficient.

With the concern that these models are only as good as their empirical formula of friction coefficient, other studies have tried to predict the traction in an EHD (Elastrohydrodynamic) contact using a number of EHL models with regard to the prediction of the friction coefficient μ in sliding of gear teeth. Martin (1981) suggested surface shear stress distribution caused by the fluid film assuming contact from smooth surface for a spur gear pair, and computed the friction coefficient from it. In addition, a transition EHL model analysis for an involute spur gear with smooth surfaces was conducted by Larsson (1997), and Olver and Spikes (1998) developed a simple algorithm based on Maxwell-Eyring rheology (Evans & Johnson, 1986) for predicting fluid traction in EHD contacts. Then, Mihailidis et al. (Mihailidis, Bakolas, Panagiotidis, & Drivakos, 2002) considered thermal effects and non-Newtonian lubricant behaviours including the influence of the asperity contacts by using numerically generated roughness profile in calculating the friction coefficient μ . More recently, a close study on the friction model was made by Xu and et al. (Xu, Kahraman, Maddock, & Anderson, 2007). They used a validated non-Newtonian thermal EHL model in conjunction with a multiple linear regression analysis to develop and compute the friction model and the new friction formula including sliding and rolling friction under typical operating, surface and lubricant conditions. In the same laboratory, Petry-Johnson and et al. (Petry-Johnson, Kahraman, Chase, & Anderson, 2008) developed a new test methodology for measurement of spur gear efficiency under high-speed and variable torque conditions and made the recommendation for the design of spur gear pairs, surface roughness and lubricant selection for better efficiencies.

2.3.2 Contact in Helical Gears

Helical gears have a parallel axis like spur gears, but the gear teeth lie on a spiral and do not run parallel to the gear axis. There have been less studies of the efficiency of helical gears compared to those of spur gears due to the mechanical efficiency of helical gear pairs being considered as high as 99.5 % to 99.7 % (Xu et al., 2007), although they have been widely used in transmission of automotive drive trains. Simons (1988) suggested increasing the radii of tooth profile curvatures and making the better surface finish of teeth for lowered power loss through the full EHL analysis of lubrication of helical gears. Then, Heingartner and Mba (2005) divided sliding friction loss and rolling friction loss for load-dependent losses in helical gears. They used Benedict and Kelly's formula as a coefficient of friction that was independent of gear surface temperature to calculate the sliding friction loss, and an equation for rolling friction loss was suggested considering the instantaneous rolling velocity, working pressure angle, lubricant property, gear material, tooth load and the instantaneous lubricant film thickness. They concluded that the sliding friction losses were heavily dependent on load while the rolling friction losses decreased slightly as load increased due to a reduction of oil film thickness. Xu and Kahraman (2007) argued the question of friction coefficient described in Table 2.2 and pointed out that sliding velocity V_s approaches zero when the contact nears the pitch point. Namely, those formulae predict very large value of friction coefficient while the friction coefficient should be zero under no sliding. Thus, a new friction coefficient formula for spur and helical gears, which is dependent on lubricant type, was suggested by performing a multiple linear regression analysis of EHL test results. In a recent paper published on the mechanical efficiency of helical gears, Li et al. (Li, Vaidyanathan, Harianto, & Kahraman, 2009) proposed a helical gear mechanical efficiency model based on a mixed EHL formulation and the selection of influencing design parameters such as tooth profile.

2.3.3 Contact in Hypoid Gears

Several types of gears are used for power transfer between intersecting shafts as shown in Figure 2.4. The teeth of spiral bevel gear are curved and oblique in contrast to straight bevel gear, and an analogy for the relationships between them can be thought as that between spur and helical gear. A pair of hypoid gears is similar to spiral bevel gears except that the pinion axis is offset above or below the axis of crown wheel, as shown in Figure 2.4 (c).

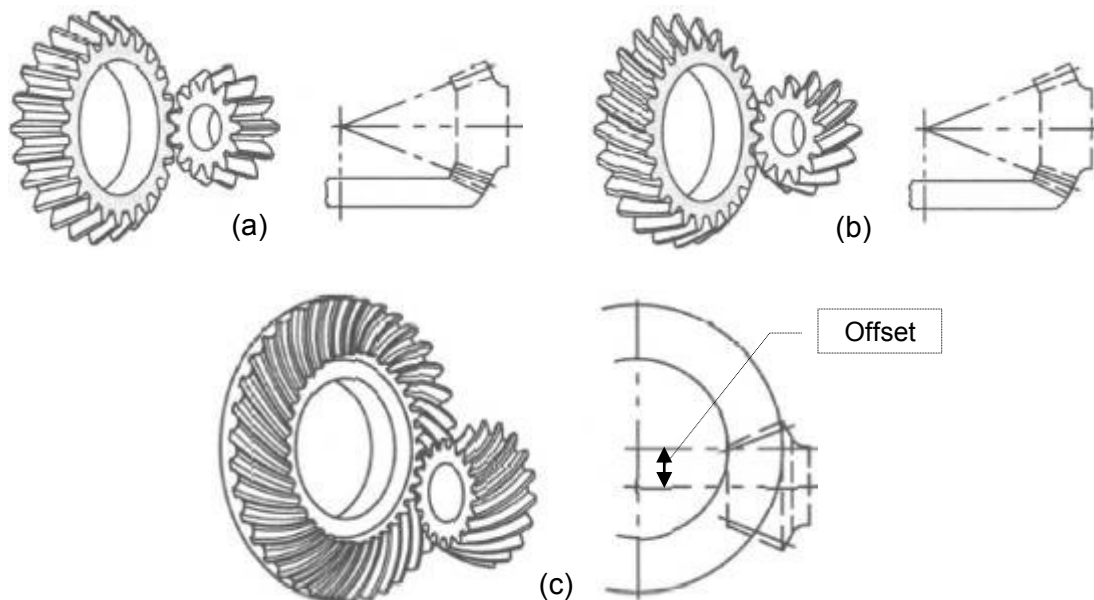


Figure 2.4 Several type of intersecting-shaft gear (a) Straight bevel gear (b) Spiral bevel gear (c) Hypoid gear (AGMA, 2005)

In other words, power is transferred to nonparallel and nonintersecting shaft in hypoid gears. Straight bevel gears are used for relatively low-speed applications, whereas spiral bevel and hypoid gears are mainly applied to large speed reduction ratio systems in the gear boxes of helicopters and the differential axle of cars due to their high capacity of torque transmitted.

2.3.3.1 Structure of Hypoid Gear

The structure of hypoid gears is described in Appendix 1.

2.3.3.2 Friction Model by Buckingham

The efficiency of hypoid gears was earlier described into the equation by E. Buckingham (1963) and was determined by

$$\text{Efficiency of hypoid gear} = \frac{1}{L_{SB} + L_W + 1} \quad (2.13)$$

where L_{SB} is friction loss of spiral bevel gear and L_W is friction loss of worm gear. The friction loss of hypoid gears were regarded as the sum of the friction losses from worm and spiral bevel gears.

$$L_{SB} = \frac{\cos \phi}{\cos \phi_n \cos \psi} \left[\frac{1 + (1/G_r)}{(\beta_a + \beta_r) \cos \psi} \right] \left(\frac{f_B}{2} \right) (\beta_a^2 + \beta_r^2) \quad (2.14)$$

where ϕ is pressure angle in plane of rotation of equivalent helical gear, ϕ_n is normal pressure angle at middle of gear face, ψ is spiral angle, G_r is gear ratio of equivalent helical gears, β_a is arc of approach of equivalent helical gears, β_r is arc of recess of equivalent helical gears and f_B is average coefficient of friction from experiments.

$$L_W = \frac{2f_W}{\cos \phi_n \sin 2\gamma_p + 2f_W} \quad (2.15)$$

where f_W are average coefficient of friction from experiments and γ_p is angle of generatrix of pinion. In 1975, another friction efficiency formula of hypoid gears was suggested by Coleman (Coleman, 1975) as shown below

$$E_{eff} = \frac{100}{1.0 + \sqrt{T_{max}/T} (0.01 + \mu \sec \phi \sqrt{(\tan \psi_P - \tan \psi_G)^2 + 0.0225})} \quad (2.16)$$

where ψ_G is mean gear spiral angle and ψ_P is mean pinion spiral angle. From this equation, the efficiency of a hypoid gear can be increased by reducing the pressure angle ϕ and spiral angle as well as increasing operating load.

Recently, Xu and Kahraman (2007) proposed a model for the prediction of efficiency in hypoid gears employing a hypoid gear contact analysis model, an EHL-based friction coefficient computation formulation and a gear pair mechanical efficiency computational formulation. In their research, surface finish and lubricant temperature were regarded as very influential factors on the mechanical efficiency of hypoid gears.

2.3.4 Bearing Losses

The bearing losses should be taken into consideration as a shaft with gears is always supported by a variety of bearings, which are chosen according to loads and speeds. Table 2.3 shows the formulae for torque losses in bearings, comprising of friction torque and viscous

Table 2.3 Formulae for bearing torque losses

Bearing	Author	Formula	Application range
Ball and Cylindrical bearing	Palmgren (1959)	$M_l + M_v = f_l F d_m + 10^{-7} f_o (v_o N)^{2/3} d_m^3$ $M_l + M_v = f_l F d_m + 10^{-7} f_o d_m^3$	$v_o N \geq 2 \times 10^{-3} \left[\frac{m^2}{s \min} \right]$ $v_o N \leq 2 \times 10^{-3} \left[\frac{m^2}{s \min} \right]$
Spherical roller	SKF catalog (SKF, 2007)	$M_l + M_v = f_l F^a d_m^b + 10^{-7} f_o (v_o N)^{2/3} d_m^3$ $M_l + M_v = f_l F^a d_m^b + 10^{-7} f_o d_m^3$	$v_o N \geq 2 \times 10^{-3} \left[\frac{m^2}{s \min} \right]$ $v_o N \leq 2 \times 10^{-3} \left[\frac{m^2}{s \min} \right]$
Needle roller	Chiu and Myers (1985)	$M = d_m \left(4.5 \times 10^{-7} v_o^{0.3} N^{0.6} + 0.12 F_r^{0.41} \right)$	
Taper roller	Witte (1973)	$M = 3.35 \times 10^{-8} G (v_o N)^{1/2} (f_l F / K) d_m^3$	

torque according to the bearings types. The formulae for friction torque for a rolling element bearing losses was empirically determined by Palmgren (1959) and are composed of the viscous friction torque M_v and the applied load dependant torque M_l due to rolling and sliding.

2.4 Windage Losses

The windage power losses measured in air for spur gears are known to be relatively small at 0.1~0.2 percent (Jeon, 2007) for one of main spin power losses with churning losses. However, windage loss is dependent upon the atmospheric density, which comes from the oil vapour, lubricant droplets and the mist atmosphere contained in a real gearbox: and should be considered as an important factor affecting the losses. The mist and fine drops of lubricant suspended in the air could increase the loss more than larger drops that are centrifuged outwards to the wall of case and drain into the base. Thus, windage loss could be considered a part of churning losses, which arise from “oily air”. Windage loss for an air is more significant and considerable at high speed.

The first analysis of spin power loss such as the windage and churning losses caused by a rotating disc immersed in a fluid originated from Von Karman (1921). He first solved the problems for an axially running disc in an infinite fluid in 1921. The concept was developed a little further theoretically and experimentally by Daily and Nece (1960) and Mann and Marston (1961), from the view of windage losses from high-speed blade discs of turbo machine. However, Daily and Nece studied the power loss for a single rotating smooth plane disc enclosed within a right-cylindrical chamber theoretically and experimentally by introducing the non-dimensional torque coefficient as a function of Reynolds number, while

Mann and Marston investigated the fluid flow over the rotating bladed discs as well as the unbladed discs.

2.4.1 Windage Losses Model on Disc or Drum with Teeth

The research for windage loss began by using a rotating drum or disc with teeth around circumference because a windage drag had been a more significant loss factor in turbines, turbochargers and high-speed electrical machineries.

The early study was made by Stodola (1927), who proposed an empirical equation of power loss for blades in turbines as

$$P = [C_1 D_M^2 + C_2 (1 - \epsilon) D_M H_B^{1.5}] \frac{\eta^3 \rho g}{10^6} \quad (2.17)$$

Where D_M is mean blade diameter, ϵ is ratio of arc of admission to total circumference, H_B blade height, η is dynamic viscosity and C_1 , C_2 are constants based on experiments. Then, Mann and Marston (1961) suggested an empirical formula of windage drag for blade disc in gas turbine by using moment coefficient C_m and considering housing effect and Reynolds number in 1961 as shown below.

$$C_m = \frac{2M_t}{0.5\rho\omega^2 R_w^5} = \frac{K}{Re^\gamma} \quad (2.18)$$

where M_t is loss torque, ω is rotational speed, R_w is radius of test wheels and K , γ is an experimental coefficient. Re is Reynolds number, which is defined as $Re = \rho\omega R^2/\mu$, and moment coefficients C_m are shown in Table 2.4, where e_a is axial clearance. The value γ for turbulent flow in their experimental result varies from 0.15 to 0.21. The empirical results shown in Table 2.4 were collected by Schlichting (Schlichting & Gersten, 2000). The proposed formulations for the drag torque are associated with air circulation and can be feasibly applied to the rotation of a disc submerged in a fluid because this theoretical moment

coefficient C_m includes the Reynolds number. This indicates that the dynamic viscosity and the density of the fluid make an influence on the windage losses, which was developed into the research of churning losses.

Table 2.4 Theoretical moment coefficient C_m (Mann & Marstan, 1961; Schlichting & Gersten, 2000)

Housing geometry	Lamina flow range	Turbulent flow range
Unclosed (centrifugal circulation)	$\frac{3.87}{Re^{1/2}}$	$\frac{0.146}{Re^{1/5}}$
Enclosed, axial clearance greater than twice the boundary layer thickness	$\frac{2.67}{Re^{1/2}}$	$\frac{0.0622}{Re^{1/5}}$
Enclosed, axial clearance less than twice the boundary layer thickness	$\frac{2\pi R_w}{e_a} \left(\frac{1}{Re} \right)$	

Thereafter in 1992, Etemad et al. (Etemad, Pullen, Besant, & Baines, 1992) provided an Equation 2.19 for rotor windage losses from air mass flow and temperature change by using the turbulent flow range equation of Table 2.4 and considering effects of air jet at the rotor rim. In this experiment, the lowest windage losses were found when air was forced through the gap between rotator and stator in the direction of rim to bore; experimental result was smaller than theoretical calculation.

$$\begin{aligned}
 P &= 0.5 \left(\frac{0.0622}{(\rho\omega R^2/\mu)^{0.25}(e_a/r)^{0.25}} \right) \rho\omega^3 r^5 \\
 &= 0.0311 \left[\frac{e_a}{r} \right]^{-1/4} \mu^{1/4} \rho^{3/4} \omega^{11/4} r^{9/2}
 \end{aligned} \tag{2.19}$$

In 1997, Durkin and Schauer (1997) carried out an experimental study on windage power loss for high-speed poled rotor in electro machines and inspected the effects of a slotted stator surface, pole gap dimensions and shrouding of rotor.

2.4.2 Windage Losses Model on Gears

The research on windage losses for rotating spur gears have been conducted after the studies on the windage losses for disc. Anderson and Loewenthal (1981) suggested the following formula to predict windage losses for a pinion-gear pair; windage losses were described as a function of face width, pitch radius, viscosity and speed.

$$P_{win} = 2.82 \times 10^{-7} \left(1 + 2.3 \frac{b}{R_p}\right) N_s^{2.8} R_p^{4.6} (0.028\eta + 0.019)^{0.2} \quad (2.20)$$

where b is face width, R_p is pitch radius, N_s is RPM and η is dynamic viscosity.

Another experimental formula for the prediction of windage loss for a large high speed gear (i.e. 300 to 1160 mm diameter drum) covered with corrugated teeth, was deduced from the experiments of P.H. Dawson (1984), where it was assumed that the side plates and the toothed periphery were treated separately.

$$P_{win} = N_s^{2.9} (0.16 D_f^{3.9} + D_f^{2.9} b^{0.75} m_n^{1.15}) \times 10^{-20} \Phi \lambda \quad (2.21)$$

where D_f is root diameter of spur gear, m_n is module and the oil mixture function Φ indicates the state of atmosphere (for example, function $\Phi = 1$ means oil free atmosphere at ambient

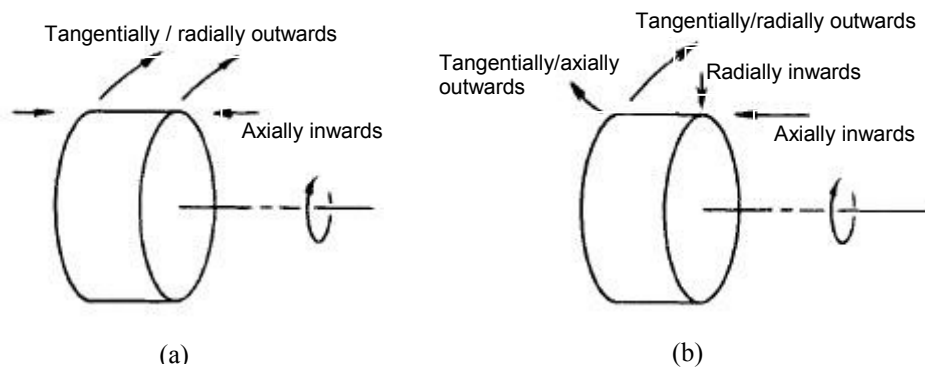


Figure 2.5 Air flow pattern (a) spur gear (b) single helical gear (Dawson, 1984)

temperature), and λ is a box space function including the effect of the gear case. The latter is set at 1.0 for free space, 0.6~0.7 for a gear in a large enclosure and 0.5~0.6 for a fitting of baffles or shrouds around the gear. Thus, windage losses are affected by module, diameter, face width, rotational speed, axial hole and the shape of enclosure. Dawson also showed the air pattern of a single spur gear and helical gear by using smoke as illustrated in Figure 2.5. Here, the teeth acted as a centrifugal fan to draw air in axially at the ends of teeth and eject it radially and tangentially from the periphery in spur gear. Additionally, the air also entered radially inwards through the tooth tips and went out tangentially/radially outwards along the helical path in helical gears. Further, in 1988, Dawson (1988) developed the empirical equation (2.22) using dimensionless parameters and considering density for an oil-mist filled gear box.

$$P_{win} = 1.12 \times 10^{-8} C' \rho N_s^{2.85} D_f^{4.7} \nu^{0.15} \lambda \quad (2.22)$$

$$C = C'(Re/(5 \times 10^5))^{-0.15} \quad (2.23)$$

Where ρ is density of atmosphere surrounding the gear, ν is kinematic viscosity of atmosphere surrounding gear, C is torque coefficient and C' is gear shape factor.

In 2004, Diab et al. (Diab, Changenet, Ville, & Velez, 2004) presented an experimental study for a single spur gear and a dimensional analysis, and compared the experimental results with the quasi-analytical power loss formula obtained from fluid flow. Windage power losses were derived from

$$P_{win} = 0.5 C_t \rho \omega^3 R_p^5 \quad (2.24)$$

where dimensionless moment coefficient C_t is defined as

$$C_t = C_f + C_l = M_g / (0.5 \rho \omega^2 R_p^5) \quad (2.25)$$

where M_g is windage resisting moment, C_f is dimensionless moment coefficient for faces of a spur gear and C_l is for teeth of a spur gear.

For front and rear face of a spur gear, dimensionless moment coefficient C_f is

$$C_f = \frac{2n_1\pi}{5-2m_1} \frac{1}{Re^{*m_1}} \left(\frac{R^*}{R}\right)^5 + \frac{2n_2\pi}{5-2m_2} \left(\frac{1}{Re^{*m_2}} - \frac{1}{Re^{*m_2}} \left(\frac{R^*}{R}\right)^5\right) \quad (2.26)$$

where n_1, m_1 are numerical coefficient for laminar flows and n_2, m_2 are numerical coefficient for turbulent flows. R^* is the critical radius separating laminar and turbulent zones and Re^* is critical Reynolds number. For side of disc or teeth of gear, the dimensionless moment coefficient C_l is given as

$$C_l = \xi \frac{Z}{4R} \left(1 + \frac{2(1+X_A)}{Z}\right)^4 \left[1 - \cos\left\{\frac{\pi}{Z} - 2(inv\alpha_p - inv\alpha_A)\right\}\right] \left[1 + \cos\left\{\frac{\pi}{Z} - 2(inv\alpha_p - inv\alpha_A)\right\}\right]^3 \quad (2.27)$$

where Z is number of teeth, ξ is reduction factor for fluid flow analysis, X_A is profile shift coefficient, α_p is pressure angle at pitch point and α_A is pressure angle at tooth tip. In 2005, Diab *et al.* (Diab, Ville, Houjoh, Sainsot, & Vexex, 2005) simulated an approximated hydrodynamic model of an air-oil mixture in high-speed rotating spur and helical gears and then compared results to show that pumping phenomenon in high speed conditions could generate significant power losses.

Recently, Al-Shibl *et al.* (Al-Shibl, Simmons, & Eastwick, 2007) carried out a two-dimensional study of windage losses for an enclosed single spur gear rotating up to 2000 rev/min in the air using the commercial computational fluid dynamics (CFD) code Fluent 6.2.16. This research showed that a small or moderate chamfer on the leading edge of gear tooth tip gave a reduction of windage losses whereas a small fillet increased these losses by 13 percent. Johnson *et al.* (Johnson, Simmons, & Foord, 2007) performed experiments on the windage loss for unmeshed spiral bevel gear within a bearing chamber of an aero-engine to investigate the shroud effect on spiral bevel gears and found that the gear shroud has the effect of decreasing windage losses.

2.5 Churning Losses in Spur Gears

If the gears rotate in an oily atmosphere as described in Section 2.4 as opposed to an air atmosphere, a correction should be applied for the increased density of the atmosphere. Al-Shibl *et al.* (2007) defined windage losses as those due to the presence of oil/air mist around the gear and churning losses as hydraulic those occurring when the gear dips into a pool or film of liquid oil or rotates through slugs of oil. Thus, the need for a better understanding of the amount of lubricant suspended in a gearbox is required, and its viscosity, its temperature, the speed of gears immersed and its sprayed way in the case of spray lubrication, can be deemed to be important factors.

Churning losses can be also defined as the loss caused by the action of the gears moving the lubricant inside the gear case. It refers in particular to the losses due to entrapment of the lubricant in the gear mesh, which is more applicable to spur gears than to helical gears as shown by Townsend (1985). As the gears submerged in the oil-filled sump rotate, the lubricant is flung off by the gear teeth in small oil droplets due to the centrifugal force acting on the lubricant. These lubricant droplets produce a fine mist inside the gear casing and turbulence, which increase the power consumption. In addition, the created oil foams modify the boundary of the submerged oil level, which consequently increases churning losses. Accordingly, churning losses in dip-lubrication can be classified into oil-pumping loss splashing the oil in the oil bath, oil-accelerating loss increasing the speed of the oil in the tooth spaces, oil-trapping loss trapping and deforming the oil between the mating gear teeth and oil-aeration loss inducing an air draft near the circumference of running gears (Ariura, Ueno, & Sunamoto, 1973). Further, another churning loss could be thought of as kind of oil

inertia loss caused by the bend and acceleration of oil flow due to rotating gear teeth if the oil impinges on the meshing zone of mating gear teeth in jet-lubrication.

As previously mentioned, the concept of fluid drag losses suggested from Von Karman (1921) has been developed to study churning losses through windage losses. Thereafter, the early study on churning losses was made Daily and Nece (1960), who developed his theory and explored the fluid drag losses for a smooth plane disc. It was followed by Ariura *et al.* (1973) who identified sources of churning losses in spur gear with jet-lubrication. The study on churning losses was made with windage losses in jet-lubricated spur gears by Akin and Mross (1975). Boness (1989) measured churning torque for the disks of varied geometries based on the study of Mann and Marston. Terekhov (1991) developed empirical equations for single and meshing gears using dimensionless moment coefficient from numerous experiments on rotating gears submerged in a fluid in 1991 although the experimental basis was not given. In 1996, Höhn *et al.* (Höhn *et al.*, 1996) came up with the formula for hydraulic loss torque of splash lubricated gear wheel in a gear case. It was followed by Luke and Olver (1999) who investigated the discrepancy among the formulae of Boness, Terekhov and Höhn by performing a number of experiments to determine churning losses in single and mating spur gears. In 2007, Changenet and Velez (2007) suggested the empirical equation for churning losses of a pair of meshing gears considering parameters such as gear module, gear diameter, gear face width, rotational speed and lubricant viscosity. Thereafter, more accurate empirical formula including the effects of flanges and deflectors was also suggested by Changenet and Velez (2008). More recently, Seetharaman and Kahraman (Seetharaman & Kahraman, 2009a, b) presented a physics-based oil churning loss model for a pair of spur gears, which described in full detail that the churning losses was caused by pocketing and squeezing as well as air-oil drag in two phase state. This model was also validated with a maximum deviation well within

0.1 kW (a relative deviation 5~25%) between predictions in this model and the measurements by experimental studies.

2.5.1 Von Karman's Fluid Flow

The fundamental aspects of the research of churning losses have a relation with the fluid mechanics of an axially symmetric, smooth-walled, right-circular rotating plane disc. The steady flow generated by a plane disc of large diameter rotating with a uniform angular velocity in contact with an incompressible viscous fluid was first investigated by Von Karman (1921), who used the form of momentum analysis for turbulent flow as a solution. Thereafter, most methods to predict or evaluate churning torque have been the empirical adaptations of the momentum integral analysis. A dimensionless moment coefficient C_m was expressed as a function of modified Reynolds number as shown previously by Equation 2.18.

2.5.2 Boness's Experimental Formula

The formula of churning torque proposed by Boness (1989), who investigated the drag torque generated by discs of varying geometry and gears rotating up to 3000 rev/min at different immersion depths in water, 15W40 oil or SAE 10W, is expressed in terms of a dimensionless torque coefficient C_m as

$$C_m = \frac{M_f}{0.5\rho\omega^2 R_p^2 A} = \frac{K}{Re^\gamma} \quad (2.28)$$

$$Re = \frac{R_p^2 \omega}{\nu} \quad (2.29)$$

where M_f is the fluid friction drag torque [Nm], ρ is the fluid density [kg/m³], A is the total wetted surface area [m²], ω is the angular velocity [rad/sec], R_p is pitch radius of gear or disc

radius [m], K and γ are experimental constants, and Re is Reynolds Number. Boness defined M as the friction torque caused by fluid and so is the same as the churning drag torque. He also determined experimental constants K and γ in equation from the direct measurement of a Lebow strain gauged torque transducer according to three distinct regimes (laminar, transition and turbulent) as shown in Table 2.5.

Table 2.5 Coefficients for Boness's empirical equation ($h/R_p \leq 0.9$)

Flow regime	K	γ
$Re < 2,000$	20	1
$2,000 < Re < 100,000$	$8.6E-4$	-1/3
$Re > 100,000$	$5.0E8$	2

2.5.3 Terekhov's Experimental Formula

Terekhov (1991) conducted a number of experiments on rotating single and meshing spur gears with high viscosity lubricant and low rotational speed. The calculation of the torque coefficient, C_m is deduced from dimensional analysis based on the flow regime and the rotating direction of a pair of gears.

If gears rotate so as to approach one another in the oil bath, churning torque C_{ch} and a dimensionless torque C_m is expressed as

$$C_{ch} = \rho \left(\frac{\pi N_s}{30} \right)^2 R_o^4 b C_m = \rho \omega^2 R_o^4 b C_m \quad (2.30)$$

$$C_m = \psi_1 \left(\frac{h}{R_o} \right)^{\psi_2} \left(\frac{V_l}{V_m} \right)^{\psi_3} \left(\frac{V_g}{V_m} \right)^{\psi_4} \left(\frac{b}{R_o} \right)^{\psi_5} Re^{\psi_6} Fr^{\psi_7} \quad (2.31)$$

$$Fr = \frac{\omega^2 R_o}{g} \quad (2.32)$$

where N_s is the number of revolutions per minute, R_o is the outer radius, b is the width of a gear, h is the immersion depth of gear into oil, V_l is the immersed volume of the gear, V_g is the total immersed volume, V_m is the volume of oil bath, and Fr is Froude number defined by Equation 2.32. The coefficients for the above Equation 2.31 are shown in Table 2.6.

Table 2.6 Exponential coefficients for Terekhov's empirical equation

Flow regime	Coefficient						
	ψ_1	ψ_2	ψ_3	ψ_4	ψ_5	ψ_6	ψ_7
$10 < Re < 2,250$ $Re^{-0.6} Fr^{-0.75} < 8.7E-3$	4.57	1.5	-0.3	-0.2	-0.4	-0.6	-0.25
$10 < Re < 2,250$ $Re^{-0.6} Fr^{-0.75} > 8.7E-3$	2.63	1.5	-0.53	-0.2	-0.4	-0.6	-0.25
$2,250 < Re < 360,000$	0.97	0.37	-0.376	-0.2	-0.124	-0.3	$-0.464 - 0.037(R_o/h)$

If a pair of mating gear rotates so as to approach one another in an oil sump, a dimensionless torque is expressed as

$$C_m = 0.506 \left(\frac{h}{R_o}\right)^{2.1} \left(\frac{V_l}{V_m}\right)^{-0.8} \left(\frac{V_g}{V_m}\right)^{-0.2} \left(\frac{b}{R_o}\right)^{-0.27} Re^{-0.32} Fr^{-0.25} \quad (2.33)$$

Equations 2.31, 2.32 and 2.33 are valid for the range of $1.6 \leq Fr \leq 1,400$, $0.025 \leq \frac{h}{R_o} \leq 2.0$,

$0.06 \leq \frac{b}{R_o} \leq 1.0$ and $0.01 \leq \frac{V_l}{V_m} \leq 0.25$. Terekhov's formula, unlike that of Boness, involves

the immersed volume and Froude number, which is found by considering the effect of gravity.

In Terekhov's analysis, however, neither the gear tooth geometry nor the immersed surface area are not considered.

2.5.4 Höhn's Experimental Formula

Höhn *et al.* (Höhn et al., 1996) took into consideration the mean oil temperature of lubricant from a thermal balance between the generated heat in a gearbox and that dissipated from the gear case surface for a pinion meshed with a gear under splash lubrication. This approach is represented in Figure 2.6.

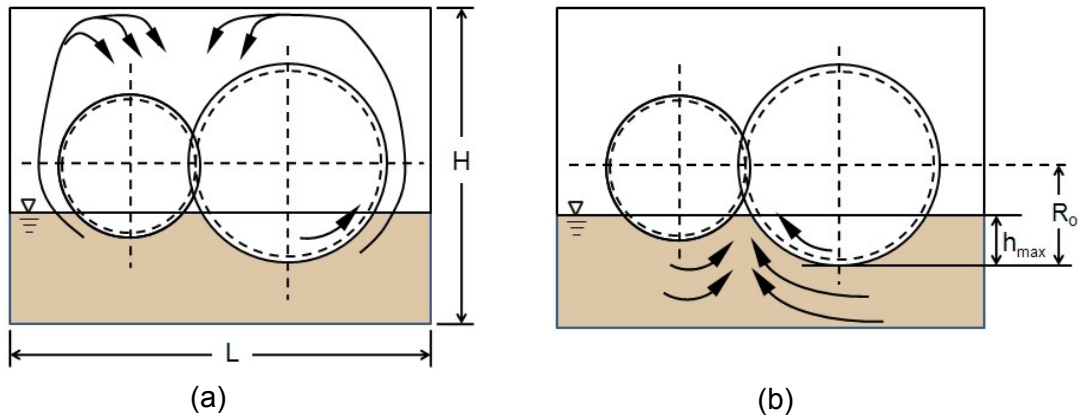


Figure 2.6 Parameters for dip lubrication of a pinion and a gear in mesh rotating in oil bath (Höhn, Michaelis, & Völlmer, 1996)

$$C_{ch} = C_{sp} C_a e^{C_b (V_t/V_{t0})} \quad (2.34)$$

where V_t is the pitch line velocity, V_{t0} is the reference pitch line speed (10 m/s), C_{sp} is the splash oil factor (given in Equation 2.35) and, C_a and C_b are defined as shown below (Equations 2.36 and 2.37).

$$C_{sp} = \left[R_{o2} / \left(\frac{L \cdot H}{L + H} \right) \right] \left(\frac{4h_2}{3R_{o2}} \right)^{1.5} \quad (2.35)$$

where L is internal width of housing, H is internal height of housing, R_{o2} is tip radius of the larger gear as R_o shown in Figure 2.6 (b) and h_2 is the larger immersion depth.

$$C_a = 0.063 \left(\frac{h_1 + h_2}{h_0} \right) + 0.0128 \left(\frac{b}{b_0} \right)^3 \quad (2.36)$$

$$C_b = \frac{h_1 + h_2}{80h_0} + 0.2 \quad (2.37)$$

where h_1 is immersion depth of pinion, h_2 is immersion depth of gear, b is face width, b_0 reference face width of 10mm and h_0 is reference immersion depth of 10mm. C_{sp} is dependent on the immersion depth h , while C_a and C_b depend on the face width b and h . This analysis shows that churning losses are influenced by the speed and immersion depth of gears, while the oil viscosity has no influence on the losses, since no lubricant viscosity factor is present in the relationship.

2.5.5 Changenet's Experimental Formula

In order to isolate the contribution of churning, Changenet and Velez (2007) measured the losses with lubricant and without lubricant. For high-speed conditions, windage losses were measured in the absence of lubricant and were subtracted from the total losses. The values of churning losses were obtained by using thermocouples to measure the ambient and the lubricant temperatures after thermal equilibrium between the two. Changenet and Velez proposed the following experimental formula for churning losses.

$$C_{ch} = 0.5\rho \left(\frac{\pi N_s}{30} \right)^2 S_m R_p^3 C_m = 0.5\rho \omega^2 S_m R_p^3 C_m \quad (2.38)$$

$$C_m = \psi_1 \left(\frac{h}{D_p} \right)^{\psi_2} \left(\frac{V_o}{D_p^{\psi_4}} \right)^{\psi_3} \left(\frac{b}{D_p} \right)^{\psi_5} Fr^{\psi_6} Re^{\psi_7} \quad (2.39)$$

where R_p is pitch radius, S_m is the immersed surface area of the pinion, V_o is oil volume. Table 2.7 represents the coefficients of the above formula of dimensionless churning torque C_m

according to Reynolds number. In the range of the transition zone between 6000 and 9000, a linear interpolation between two formulae is employed to the coefficient.

Table 2.7 Coefficients of Changenet's empirical equation

Flow Regime	Coefficients						
	ψ_1	ψ_2	ψ_3	ψ_4	ψ_5	ψ_6	ψ_7
Low (Re<6000)	1.366	0.45	0.1	3	0	-0.6	-0.21
High (Re>9000)	3.644	0.1	-0.35	3	0.85	-0.88	0

When meshed gears rotated approaching in the air in their experiments, the churning losses

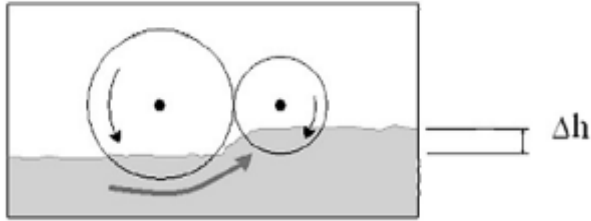


Figure 2.7 Approaching-direction rotating of a pair of spur gears in oil bath (Changenet & Velex, 2007)

were the addition of the two individual contributions. However, the variation of churning losses ΔP took place when the meshed gears approached each other under the lubricant surface (counter-clockwise rotating direction) as shown in Figure 2.7 because of the trapping of

lubricant by the meshing teeth. In this figure, it can be seen that under these conditions the oil immersion depth of the smaller gear is increased by the swell effect; this does not occur under high-speed ratios. The additional churning power loss ΔP was expressed in terms of a dimensionless variation of churning torque ΔC_m by

$$\Delta P_{churn} = 0.5\rho \left(\frac{\pi N_s}{30}\right)^3 S_m R_p^3 \Delta C_m \quad (2.40)$$

$$\Delta C_m = 17.7 Fr^{0.68} \frac{1}{u^8} \left[\left(\frac{h}{R_p}\right)_{gear} - \left(\frac{h}{R_p}\right)_{pinon} \right] = 17.7 Fr^{0.68} \frac{(u-1)}{u^8} \left[1 - \left(\frac{h}{R_p}\right)_{gear} \right] \quad (2.41)$$

where u is speed ratio of pinion to gear. In Changenet's formula of dimensionless churning torque, the forms of lubricant viscosity for temperatures and gear geometry as well as Froude number were taken into consideration. In addition, it should be noted that gear geometry the submerged surface area plays a role, while teeth number and face width are only influential at high speeds. At high speeds, churning losses are likely to be affected more by inertia force than by oil viscosity.

Changenet and Vex's subsequent study (2008) in 2008 concerned the influences of flanges and deflectors on churning losses for spur gears submerged in oil. It showed that the effect of radial clearance is smaller than that of axial clearance on churning losses for spur gears.

2.5.6 Kahraman's Physics-based Model

A physics-based analytical fluid mechanics model was proposed by Seetharaman and Kahraman (Seetharaman & Kahraman, 2009a), who divided churning losses into drag power loss P_d from the interaction of each individual gear with the surrounding medium (oil), and oil pocket power loss P_p caused by the interaction of the gear pair with oil the at gear mesh interface.

The former churning losses P_{di} were described as

$$P_{di} = P_{dpi} + P_{dfi} + P_{rfi}, \quad i=1,2 \quad (2.42)$$

where i is gear index. Power loss P_{dpi} due to drag on the periphery of a single gear was given by the product of the drag force F_{dpi} acting on the periphery of the gear and the tangential velocity at the periphery by

$$P_{dpi} = 4\mu_d b_i r_{oi}^2 \omega_i^2 \varphi_i \quad (2.43)$$

where μ_d is the dynamic viscosity, b is gear face width, r_o is the tip radius and φ is the immersion angle. Power loss P_{dfi} due to the drag on the faces of the gear was comprised of the loss $P_{dfi}^{(L)}$ in the laminar and $P_{dfi}^{(T)}$ in the turbulent regime as shown below.

$$P_{dfi}^{(L)} = \frac{0.41\rho v^{0.5}\omega_i^{2.5}r_{oi}^2 A_{dfi}}{\sqrt{\sin\varphi_i}} \quad (2.44)$$

$$P_{dfi}^{(T)} = \frac{0.025\rho v^{0.14}\omega_i^{2.86}r_{oi}^{2.72} A_{dfi}}{(\sin\varphi_i)^{0.14}} \quad (2.45)$$

where v is kinematic viscosity and A_{dfi} is wetted face area of gear. Power loss due to root filling inside one tooth cavity can be calculated from the product of rotational speed ω and the torque loss T_{rfi} due to the swirling flow of the lubricant inside the cavity

$$P_{rfi} = 2n_{ci}\mu_d A_v \omega_i (r_{oi} - r_{ri}) \left(\frac{D_{3i}}{r_{oi}r_{ri}} - D_{4i} \right) \quad (2.46)$$

where n_c is the average number of cavities, r_r is the root radius, A_v is the area of cavity between the teeth, μ_d is the dynamic viscosity, and D_3 and D_4 are coefficients determined from boundary conditions. Seetharaman and Kahraman suggested that oil pocket power loss P_p due to oil pocketing at the gear mesh interface was composed of those arising from backlash flow area and end flow areas. In order to derive the pocket power loss, the velocity of the oil squeezed out, and the force acting on the oil escaping through the backlash and end flow areas were first calculated.

2.6 Churning Losses in Other Gears

Although a large number of studies have been made on churning losses for spur gears, little is known about churning losses for other types of gears.

2.6.1 Churning Losses of Helical Gears

Heingartner and Mba (2005) suggested the mathematical model for power losses comprises of windage losses and oil churning loss as well as friction losses of helical gears. They found that the windage losses remained constant at the same speed in its experiment and were calculated using Dawson's formula. The oil churning loss for helical gears was divided into three categories as shown below based on British Standards BS ISO/TR 14179. Churning loss for smooth outside diameters was described by

$$P_{ch} = \frac{7.37 f_g \nu N_s^3 D^{4.7} L_c}{0.2 \times 10^{26}} \quad (2.47)$$

where f_g is the gear dip factor, D is the diameter, ν is the kinematic viscosity at operating temperature, N_s is the rotational speed, and L_c is the constant line length. Churning loss for gear side faces was given by

$$P_{ch} = \frac{1.474 f_g \nu N_s^3 D^{5.7}}{0.2 \times 10^{26}} \quad (2.48)$$

Churning loss for tooth surfaces was given by

$$P_{ch} = \frac{7.37 f_g \nu N_s^3 D^{4.7} (S_{rf} / \sqrt{\tan \beta})}{0.2 \times 10^{26}} \quad (2.49)$$

Where S_{rf} is roughness factor and β is helix angle. According to the above equations, the deeper the components are submerged, the higher the churning losses. For larger helix angles, the losses are lower than smaller ones or spur gears ($\beta = 0^\circ$).

2.6.2 Churning Losses in Hypoid Gears

In recent years, interest in the churning losses has grown as the demands for increased power density and fuel efficiency have become more stringent; as a result, much research on churning losses and windage losses for parallel axis gears as well as friction losses have been carried out (as previously described). However, to the author's knowledge, no studies have ever attempted to measure the churning losses from hypoid gears, although such gears are widely being applied to automotive and aerospace industry such as in hypoid axles, helicopter gear boxes and the bearing chamber of aero engines.

On the other hand, several studies into the load-carry capacity and lubricant characteristics for hypoid gears have recently been carried out. Simon (Simon & Vilmos, 2005) used numerical program to study the influence of lubricant characteristics considering the relation among pressure viscosity, EHD film thickness, load carrying capacity and power loss. It was followed by Höhn et al. (Höhn, Conrado, Michaelis, & Klein, 2007), who investigated the influence of different oil immersion depth on scuffing load capacity of hypoid gears and showed that a reduction of the sump oil level decreased scuffing load capacity. Thus, more research is required to establish a detailed relationship between the churning losses and the influential variables in hypoid gears.

2.7 Concluding Remarks

The foregoing review has shown that much experimental works on load dependent losses and spin power loss for parallel axis gears such as spur gears have been carried out. A large number of theoretical and experimental studies of gear friction have been proposed to

quantify the losses from both the rolling and slipping teeth as well as more exact spontaneous friction coefficients based on EHL models. In addition, much empirical research on churning losses for spur gears have been performed; the majority has used dimensional analysis based on the concept of Von Karman's infinite rotating disc immersed in oil. From this review, it is clear that the main factors that affect the spin power losses for spur gears are rotational speed, diameter of the rotating gear and the shrouding. Windage power losses for spur gears are approximately proportional to rotational speed raised to the third power. Additionally, the previous empirical formulae for churning loss models of spur gears were calculated through the data of simple spur gears and compared to inspect the influencing variables on their churning models, which are shown in Appendix 2.

However, the churning losses for hypoid gears have yet to be studied in order to diminish the churning losses due to their relatively complex gear geometry, and the large size of apparatus required for measurement. This is the case despite the fact that hypoid gears are widely used in industries. In Chapter 3, the experimental methodology and apparatus for measuring churning losses of hypoid gears are introduced.

CHAPTER 3

A DESCRIPTION OF TEST RIG AND EXPERIMENTAL METHODOLOGY

This Chapter presents design and construction of the extension housing and the test rig for measuring the churning losses and the experimental techniques used in the present work.

3.1 Axle Assembly and Extension Housing

Two types of housing were employed to investigate the churning losses in the hypoid axle. Firstly, an axle housing produced on the line at the factory, was provided by a motor company. Secondly, another bigger housing (the “Extension” housing) than the production axle housing was designed and fabricated in order to investigate the effect of housing clearance on churning losses. To carry out this experiment, a range of special radial and axial baffles were designed by the author and fabricated by the motor company.

3.1.1 Structure of Axle in Automobile

A hypoid axle is composed of crown wheel and pinion gear as previously described in chapter 2 and shown in Figures 2.2 and 2.4. In the experiments in this research, two variants of hypoid gears were employed to investigate the effect of gear geometry. These had crown wheels with an outer diameter of 222 and 213.2 mm respectively.

3.1.1.1 Hypoid Crown wheel Gear (Ring Gear)

The hypoid crown wheel assembly is composed of crown-shaped ring gear, two pairs of bevel side gears and a housing surrounding them. The bearings, which are inner rings press-fitted to both the end shafts of the side gear housing with the outer rings mounted in axle housing, are usually taper roller bearing as indicated in Figure 3.1 (c). For the this experiment, thin section single deep groove ball bearings were used in place of the taper roller bearings in order to reduce bearing friction losses since the friction torque of taper roller bearings is larger than that of ball bearings for the same loads. (Höhn et al., 1996)

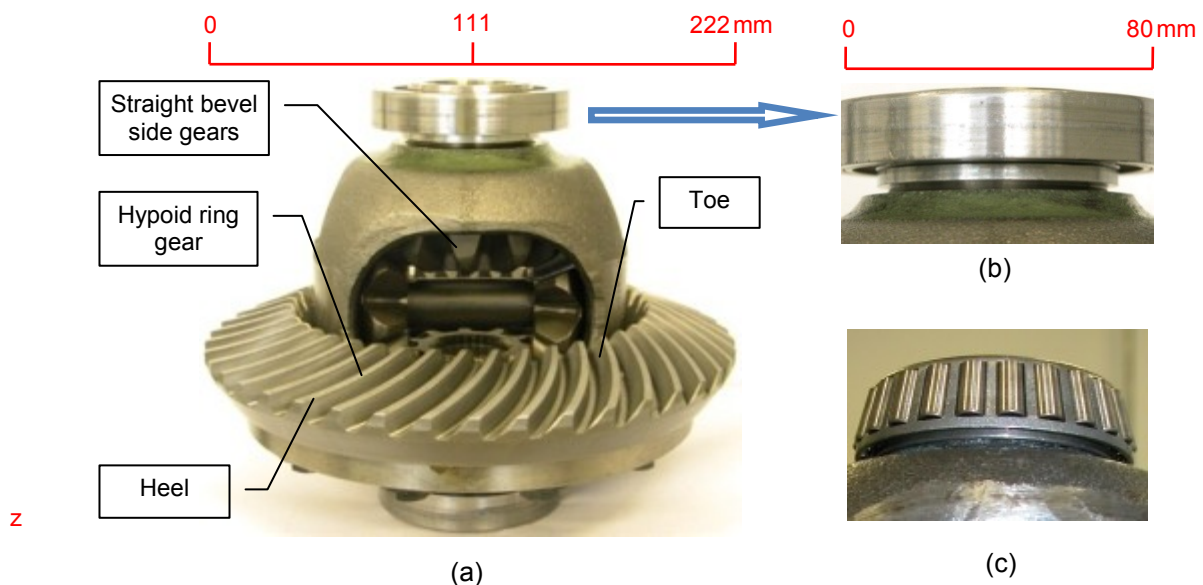


Figure 3.1 (a) A crown wheel gear of which bearings are replaced with (b) Thin section single deep groove ball bearing (c) Taper roller bearing

Accordingly, several spacers were inserted to fill the gap between the ball bearing and the end of wheel assembly due to the reduced thickness of the ball bearing compared with the taper roller bearing as shown in Figure 3.1 (b).

In the author's experiments, the straight bevel side gears were bonded to each other and to the housing with Araldite epoxy adhesive to prevent them from rotating. In other words, the differential mechanism was disabled so that both output shafts were constrained to rotate at the same speed as shown in Figure 2.2(a). The outer diameters of the two crown wheels were 213.2 and 222 mm; their nominal pitch diameters were 210 and 220 mm respectively.

3.1.1.2 Hypoid Drive Pinion Gear

The taper roller bearings of the hypoid pinion drive gear were also replaced with the thin

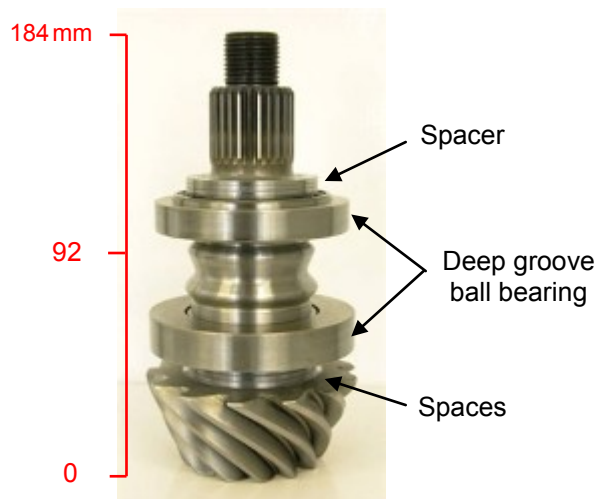


Figure 3.2 Pinion gear with deep groove ball bearings and spacers

section deep groove ball bearings as shown in Figure 3.2. The difference of thickness between taper roller bearings and thin section ball bearings were filled with spacers because the position of bearings determines the clearance of the gear mesh between the hypoid pinion and the crown wheel. The distance

between the ball bearing and the pinion gear was checked after being assembled to the axle housing together with crown wheel several times, and then the position of bearing could be adjusted exactly using a trial-error method.

3.1.1.3 Bearing Jig for Press fit

A bearing jig was fabricated for the effective press fitting of bearing to the shaft of side bevel gear housing as demonstrated in Figure 3.3. It enables the application of forces on the periphery of bearing and the pressing of the inner ring of bearing into the shaft perpendicularly, which could influence on the clearance of gear mesh between hypoid pinion and crown wheel.

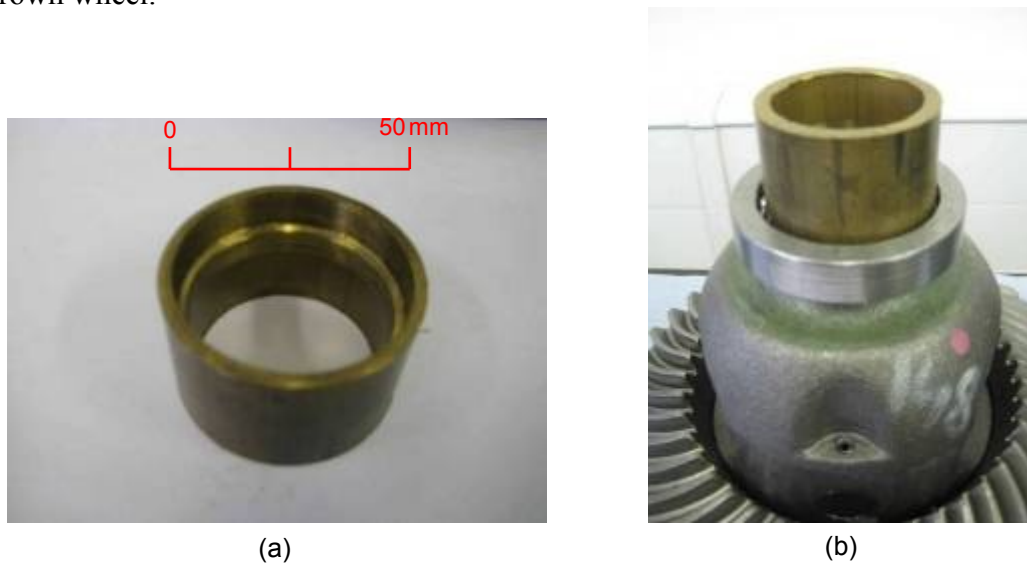


Figure 3.3 (a) Bearing jig and (b) Bearing jig pushing bearing to crown wheel

3.1.2 Extension Housing

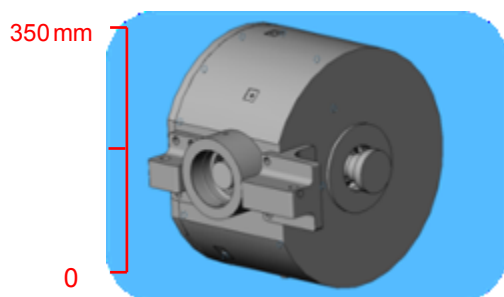


Figure 3.4 Extension housing

A cylindrical shaped housing shown in Figure 3.4 was designed and fabricated for investigating the effect of housing geometry on churning losses because there was a limit in changing the shroud conditions inside the production axle housing. This “extension” housing is assembled

of seven parts comprising the main housing, its lid, a pair of bearing housing supporting

bearings at both sides, a pinion gear bearing housing supporting bearing for the pinion at its front side and a pair of housing supporters at its front side. The main housing part was made by turning the inside of a large Aluminium cylinder after cutting one side. On this main housing having an inner depth of 180 mm and an inner diameter of 290 mm, three bearing-supported components were press-fitted and then two housing-supported frames were bolted. For the lubrication of drive pinion gear bearings, rubber tubes as an oil path were connected

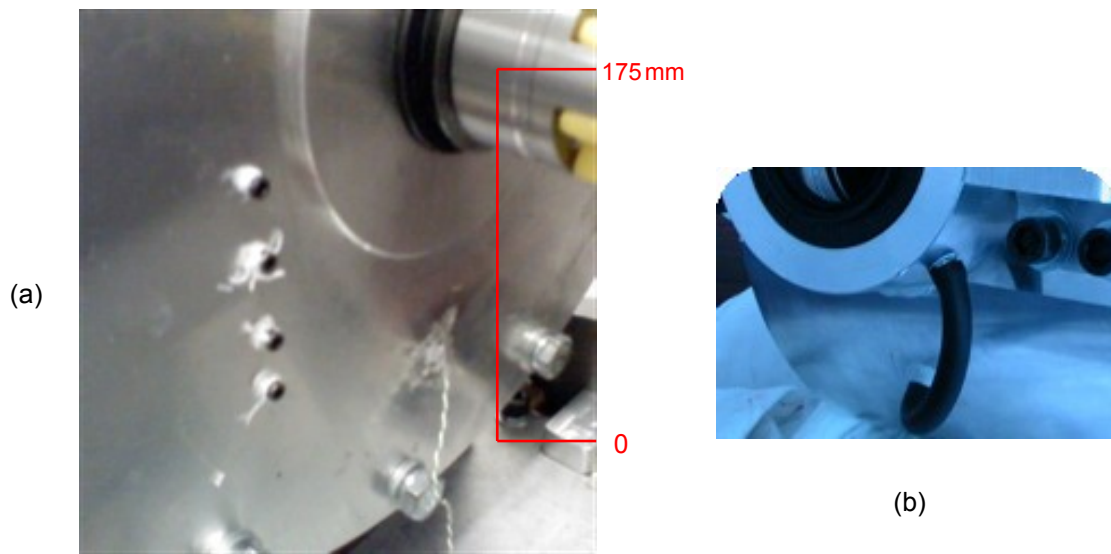


Figure 3.5 The parts of extension housing: (a) oil depth holes and (b) oil circulation hose for bearing lubrication

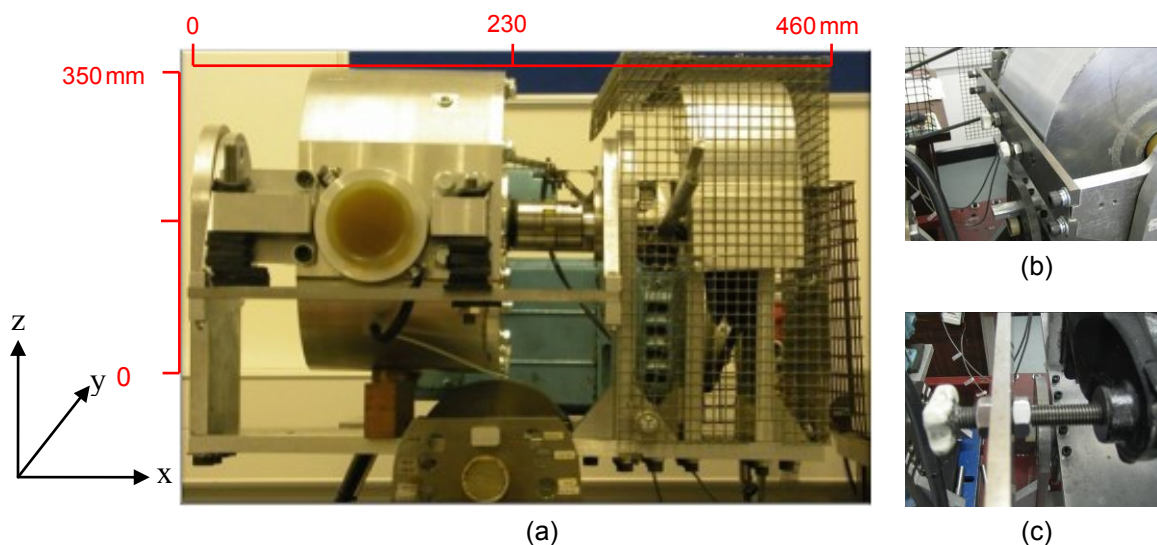


Figure 3.6 Extension housing on the test rig (a) Front view (b) Rear view of extension housing fixed in Y and Z direction (c) Rear view of the production axle housing fixed in Y and Z direction

from main housing to two holes which were made on the pinion gear bearing-supported component using the clamp as shown in Figure 3.5 (b). In addition, the small threaded holes for oil level gauge and a hole of 1 mm dia. for thermocouple wire were also made on the side of main housing as presented in Figure 3.5(a). In this figure, the oil-level check holes were located at 20, 40, 47, 64 and 80 mm distant from the rotating axis of crown wheel; the oil quantity dwindles as the distance goes bigger. Added to these holes in the main housing, an air breather hole on the top and a drain hole on the bottom of main housing were bored. Figure 3.6 shows how the extension housing was mounted on the test rig. Those mounted locations of this extension housing were fixed by a threaded bar and nut at four points on the test rig so that two front points of housing were located mainly in the x and y direction of a three dimensional Cartesian coordinate system as shown in Figure 3.6. The other two points of the back of the housing were fixed with regard to z and x direction by a threaded bar passing through a hole in the test rig frame and being tightened to the screw holes of main extension housing with nut. The design of housing mounting position is very significant because its tolerance has an effect on coaxiality between power-transferred shafts.

3.1.2.1 Baffles for Axial and Radial Shroud Enclosure

The baffle material was Nylon 6, namely Natural Ertalon 6PLA. They were newly designed, fabricated and employed in the tests in order to investigate the effects of radial and axial enclosure in the extension housing on churning losses with respect to crown wheel gear and drive pinion gear.

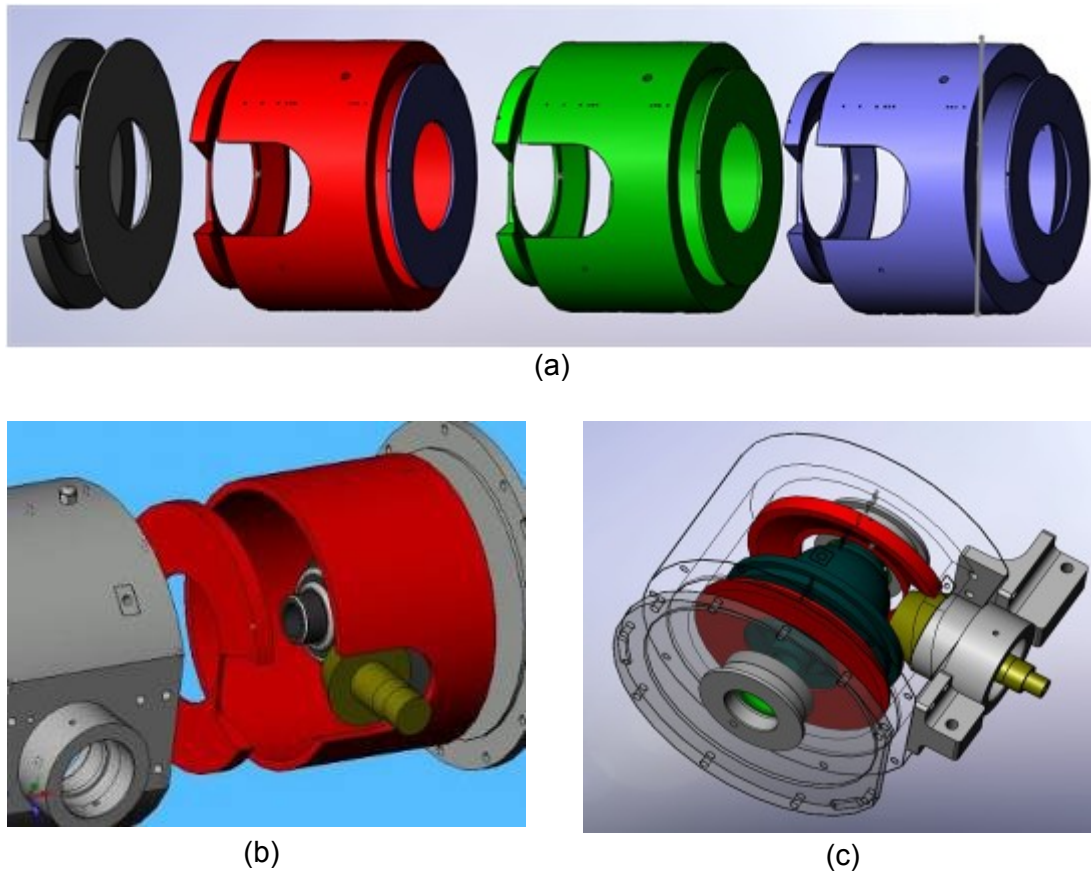


Figure 3.7 Configuration of radial and axial baffles (a) three kinds of size of baffles (b) baffles inserted in extension housing (c) location of axial baffles from downside view

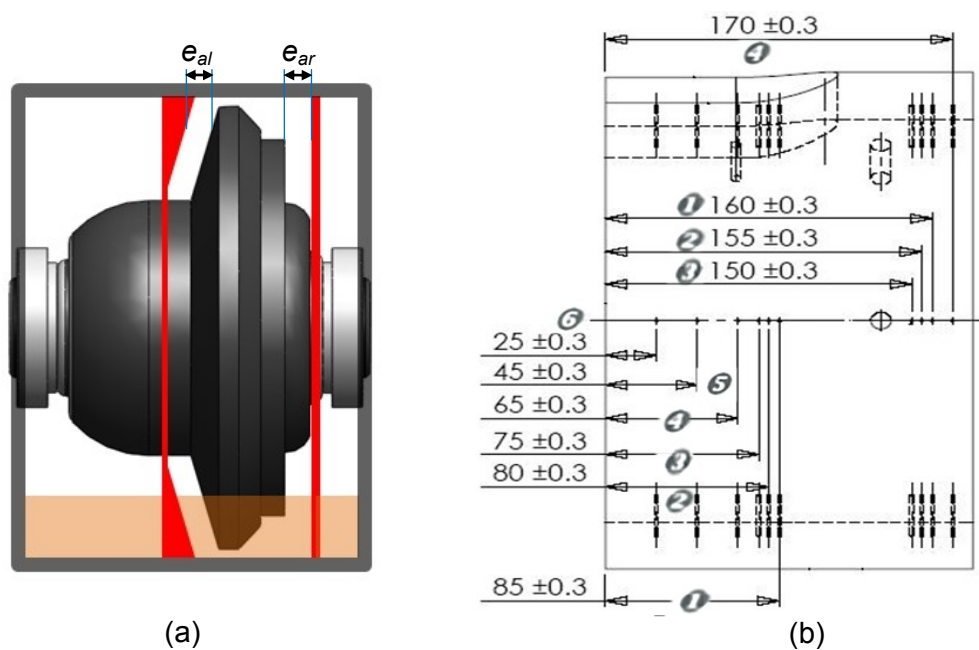


Figure 3.8 (a) Configuration of Axial clearance; simple section view of extension housing for axial baffles (b) Location of axial baffles on a radial baffle from the bottom view of radial baffle in the drawing

Three diameters 226, 234.8 and 343.6 mm of radial baffles and their corresponded axial baffles were designed as shown in Figure 3.7(a). Figures 3.7(b) and Figure 3.7(c) show the configuration of axial and radial baffles inserted in the extension housing. Two of the axial baffles could be fixed on the corresponded radial baffle respectively at both sides by small cup point set screws on three places of the periphery of the axial baffle. Axial clearances e_{al} and e_{ar} caused by axial baffles were shown in Figure 3.8 (a) and the location of axial baffles inside the radial baffle was drawn and numbered as shown in Figure 3.8 (b). This figure shows the positions of cup point set screws for fixing the axial baffles to the radial baffles. The slope of axial baffles at the left side in Figure 3.8 (a) is 67° . This is almost the same as that of the crown wheel on the toothed side. A radial baffle of diameter with 226 mm was employed in axial clearance tests of crown wheel in the extension housing.

Tables 3.1 and 3.2 indicate how the radial clearances and axial clearances of the crown wheel in the extension housing varied with different radial baffles and the positions of the axial

Table 3.1 Radial clearance e_r [mm] between baffle and crown wheel

Inner diameter of radial baffle [mm]	Crown wheel size	
	M210	M220
226	6.40	2.00
234.8	10.80	6.40
243.6	15.20	10.80
290 (housing wall with no baffle)	38.40	34.00

Table 3.2 Axial clearance e_a with the position of axial baffles shown in Figure 3.8

Axial clearances	Location number of axial baffle on radial baffle from Figure 3.8					
	①	②	③	④	⑤	⑥
e_{al} [mm]	2.5	7.5	12.5	22.5	42.5	62.5
e_{ar} [mm]	12	17	22	32	-	-

baffles. Under these conditions, experiments for the effect of housing clearances on churning losses were carried out. Radial clearances are also varied according to the size of crown wheel, and the inner diameter of the extension housing is 290 mm when no radial baffle is installed inside it as shown in Table 3.1. On the other hand, the axial clearances e_{al} and e_{ar} are changeable depending on the positions of two axial baffles and the numbers of their positions in Table 3.2 are shown in Figure 3.8 (b). The axial baffle at the toothed side of the crown wheel having a clearance e_{al} can be placed at six positions and the other axial baffle for e_{ar} four positions. Accordingly, 24 combinations of positions of axial baffles for the axial clearances e_{al} and e_{ar} can be made in the extension housing.

3.1.2.2 Transparent Extension Housing

The lid of extension housing was replaced by a transparent lid made of Plexiglas in order to visualize the lubricant flows around rotating crown wheel as shown in Figure 3.9. This transparent lid was designed and fabricated at the last stage of all the experiments and so it was only employed under the radial baffle with a diameter of 234.8 mm and the axial baffles

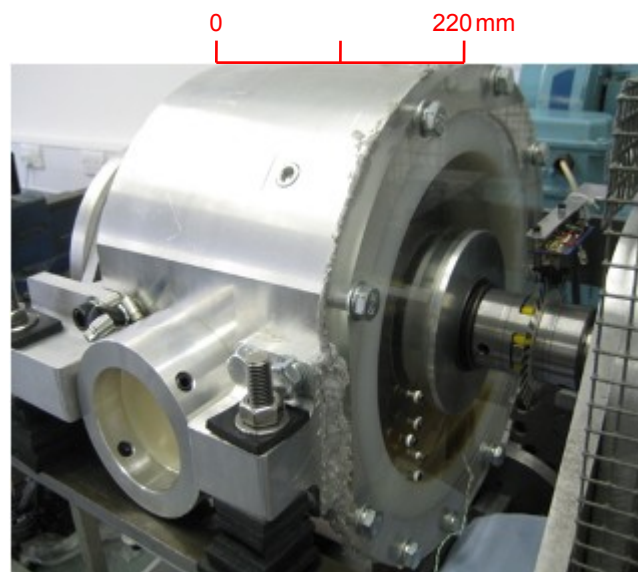


Figure 3.9 Transparent side of extension housing mounted on the test rig

of their axial clearances $e_{al}=7.5\text{ mm}$ and $e_{ar}=17\text{ mm}$ inside the extension housing in this thesis.

3.2 Test Rig Design and Manufacture

The test rig is to be used primarily to measure the churning losses for a hypoid axle. To satisfy this purpose, the test rig was designed to be able to mount the extension housing as well as the production axle housing and to measure the churning losses at a high accuracy for influencing variables.

3.2.1 Concept of Test Rig

The current approach to oil management in gear transmission systems is based mainly on direct experiment although Computational Fluid Dynamics (CFD) modelling is occasionally used. (Al-Shibl et al., 2007; Wild, Djilali, & Vickers, 1996) The reason is that CFD modelling of churning losses can prove to be extremely complex and inaccurate in the point of view of oil-air mixture and is still difficult to predict the quantitative values of churning losses with a high accuracy.

There are three possible ways to determine churning losses from gear boxes in a dip lubrication: direct torque measurement from strain gauges, heat dissipation measurement and, finally, the inertia rundown method. First, the method using torque meters is generally preferred to measure the losses in industry, but the torque meter is not accurate for measurement of churning torque because the difference between input and output torques is relatively small under very low torque conditions caused by no transmitted power. Additionally, temperature variation and vibration within the experimental setup of strain

gauge could influence on the gauges and cause disturbances to the reading values. In efficiency tests, measuring the heat dissipation to oil as well as torque at the same time are widely carried out because friction causes temperature increase and may be measured as a motion-resisting torque. All measurements of losses in heat dissipation are made in a steady state when the oil temperature is stabilized. In this method, all concerned components are assumed to be connected by thermal resistances by analogy with an electric resistance in Ohm's law, and thermal resistance is dependent on the kind of heat transfer such as conduction, free or forced convection and small radiation (Koffel, Ville, Changenet, & Vexel, 2009). Thus, a bulk temperature or corresponding temperature distribution to the concerned components box requires to be measured in an insulated condition. Namely, environment temperature should be also kept constant and time taken for a system to reach thermal equilibrium can be relatively protracted.

In this research, the author chose the method of inertia run-down due to the reasons previously described. This method requires the driving assembly speeding up and free run-down. Once up to a certain speed of a drive gear of the system from motor, the electric motor

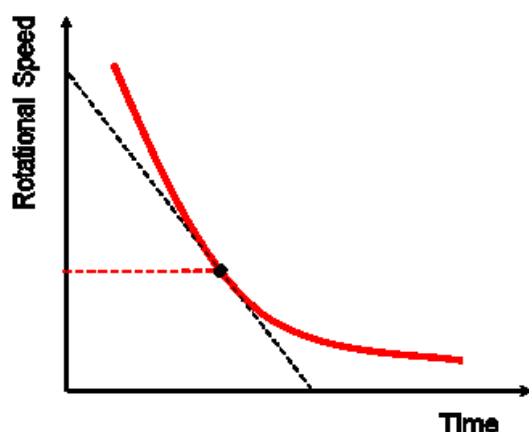


Figure 3.10 Decay graph of rotational speed versus time

and the drive are disengaged from each other and the system is allowed to decelerate freely under the dissipative mechanism. The ring gear of axle is driven up to a required speed (a bit higher than 2,000 rev/min in this research), and then the deceleration of rotating gear in axle is measured in the absence of a power supply. This rotational decreased

speed for the free run-down due to the various frictions is noted against time as shown in

Figure 3.10. The gradient at different rotational speed can be determined to give the angular deceleration of the system. The product of this deceleration and the effective moment of inertia of the system make the correspondent loss torque under the given conditions in the efficiency test rig of axle as shown Figure 3.11. Therefore, the slope of the graph can be expressed as a differential equation as presented below.

$$\ddot{\theta} = \dot{\omega} = \frac{d\omega}{dt} = \frac{T}{I_{eff}} \quad (3.1)$$

$$T_{drag} = I_{eff} \frac{d\omega}{dt} \quad (3.2)$$

$$P = \omega T_{drag} = \omega I_{eff} \frac{d\omega}{dt} \quad (3.3)$$

where $\ddot{\theta}$ is angular acceleration, $I_{eff}[\text{kg m}^2]$ is the effective moment of inertia of the system, ω is angular velocity, ω is a rotational speed, t is time, T is a total drag torque and P is a total loss. In order to separate churning drag torque from this total retarded torque, another test of decay curve under the absence of lubricant in the oil bath needs to be carried out as well as run-down with oil in the bath. Thus, churning drag torque can be calculated if the no-oil drag torque obtained from the experiment without oil is consequently subtracted from the total retarding torque under a dip-lubricated condition.

$$T_{churn} = T_{total} - T_{no\ oil} \quad (3.4)$$

To achieve high accuracy of churning drag torque in the above run-down methodology, the effective moment of inertia has to be determined with sufficient accuracy as well as measurement of the previously described decelerated value. The effective moment of inertia I_{eff} in question can be calculated from the harmonic motion that the spring having the spring coefficient k is connected to the radius R_n of cylindrical model as shown below.

$$f = \frac{\omega_n}{2\pi} = \frac{1}{2\pi} \sqrt{\frac{k}{M_n}} = \frac{1}{2\pi} \sqrt{\frac{k R_n^2}{I_{eff}}} \quad (3.5)$$

where, f is natural frequency, ω_n is angular frequency and M_n is a mass. The trifilar suspension method was chosen to determine the effective moment of inertia instead of the above free harmonic vibration because the former was simpler in experimental set-up and easier in the measurement than the latter. The trifilar suspension is explained in Section 3.3.1. Subsequently, power loss is obtained by the product of loss torque and angular velocity. The contribution of component such as pinion gear in axle can be

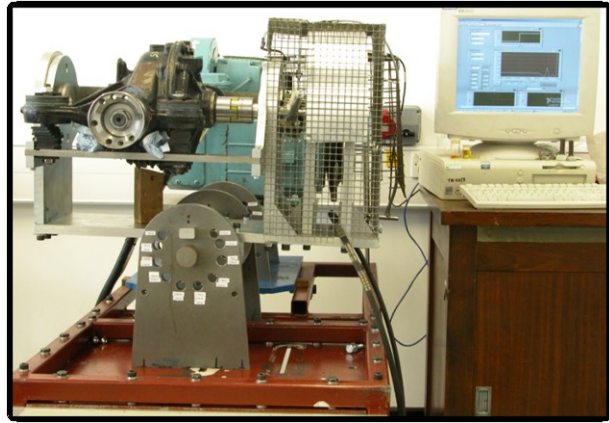


Figure 3.12 Churning losses test rig on which axle housing is mounted

determined by subtracting the power loss without having it from the total system loss. Thus, the total losses of rotating gears merged in the oil are measured prior to measurements of the losses in the absence of oil to be subtracted from the total loss in order to attain churning losses caused by lubricant.

Figure 3.11 and 3.12 show the test rig that was produced for this project for measuring the churning losses under the several environments. The test rig is able to accept not only the production axle housing shown in Figure 3.11, but also the extension housing shown in Figure 3.12. The test

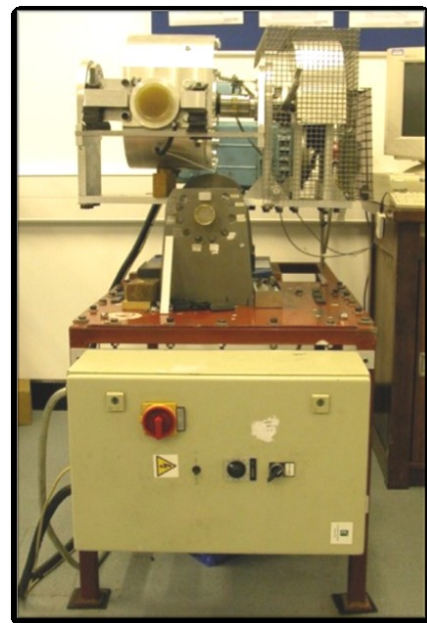


Figure 3.11 Test rig on which extension housing is mounted

rig is composed of motor, pulley, V-belt, tilting devices, encoder, shaft, bearings, thermocouple, clutch device, data logger and PC.

A 2 kW electric motor transfers the power to the pulley through a V-type rubber belt to drive the hub gear connected to pulley through a shaft at a speed of 2,000 rev/min forward or backward. Next to the hub gear connected to the pulley, there was located another hub gear connected to the axle at the opposite side through a shaft; where the hub gears and shafts are connected by tight spline fittings and fitted keys. A sleeve gear engages on both hub gears to transfer the power from the drive hub gear to the axle. When the axle reaches the required speed (up to 2,000 rev/min), the sleeve gear is shifted in the direction of pulley so as to disengage it from the opposite hub gear. This clutch device that disconnects the drive by sliding sleeve gear between the two hub gears, is shown in the dotted line box of Figure 3.13 and is magnified in Figure 3.17. After the disengagement between the hub gears, the decline speed due to the resisting moment is measured by speed sensor. In addition, the test rig can

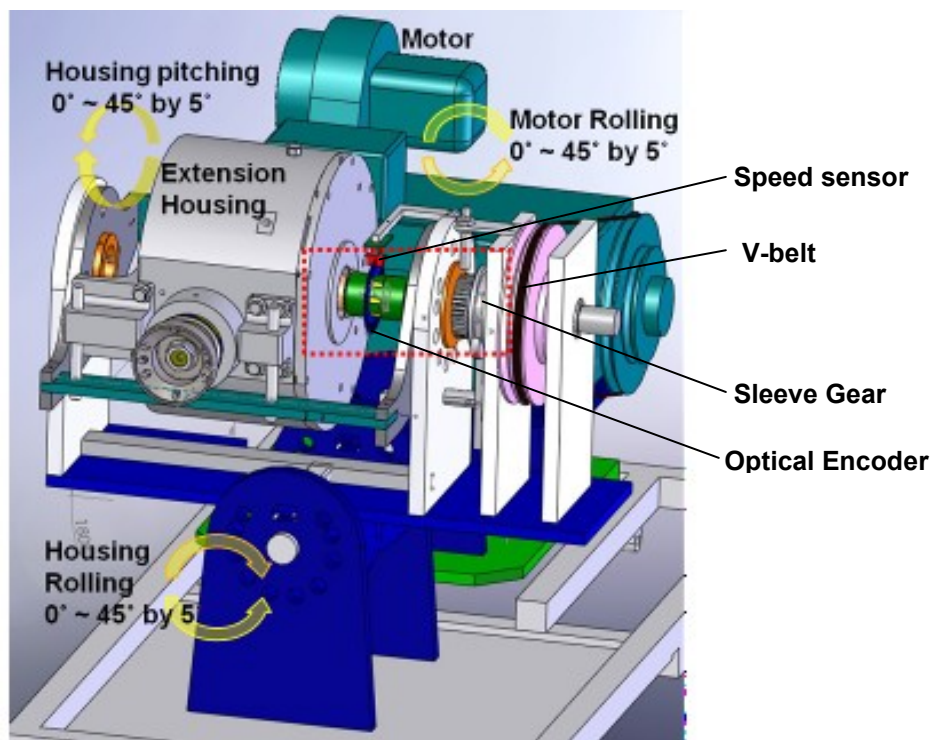


Figure 3.13 Attitude configuration of test rig

reproduce the attitude of vehicles in rolling and pitching angles from zero to 45 degrees in five degree increase as shown in Figure 3.13. Mounting plate on the test rig can be rotated as many as a required amount of angle and then fixed using an aluminium pin inserted the holes that coincides with the required degrees.

3.2.2 Test Rig Design

Most of the components in the test rig except motor and motor controller were drafted using Solid Works by the author. The test rig was designed to be able to be widely utilised for any other similar experiment only if its housing could be mounted on the rig. In addition to it, all the components were assembled by the bolt and it is feasible to change the rig into another and to repair it easily. The drawings and specifications for the rig components are described in the Appendices 4 and 6.

3.3 Apparatus for Measurement

3.3.1 Experimental Measurement of Moment of Inertia

A trifilar method was used to measure the moment of inertia for the rotating components in the test rig. It was explained how to drive the equation for the trifilar method and to measure the moment of inertia of the component in question using the trifilar method in Appendix 3. The moment of inertia, I_1 in question finally is calculated from the measured values by Equation 3.6, which was derived in Appendix 3.

$$I_1 = (I_1 + I_2) - I_2 = \frac{T_{period,1\&2}^2}{4\pi^2} \frac{(m_1 + m_2)gr^2}{L_{1\&2}} - \frac{T_{period,2}^2}{4\pi^2} \frac{m_2gr^2}{L_2} \quad (3.6)$$

where, I_1 is the moment of inertia for the component in question, I_2 is the mass moment of inertia for circular platform, m_1 the mass of the component in question, m_2 is the mass of the circular platform, $T_{period,1\&2}$ is the period of suspension of the component and the platform and $L_{1\&2}$ is the length of suspended string when the component is placed on the platform. From the above Equation 3.6, (I_1+I_2) and I_2 can be calculated by measuring the period times, $T_{period,1\&2}$ and $T_{period,2}$. The lengths of string $L_{1\&2}$ and L_2 are generally same. However, the lengths of strings get to be different due to the stretch of string and need to be measured individually if the weight of the component, m_1 is much bigger than that of platform, m_2 . When performing the measurement of those using this method, there are some important assumptions to pay an attention.

- (1) The centre of gravity of component is coincident with the axis of rotation of platform. Namely, the rotational axis of the component with regard to mass moment should be same with the axis of rotation of platform.
- (2) Only one degree of freedom is excited. In the trifilar suspension, gravity is the only external force and other damping forces due to air and environmental vibration are neglected.
- (3) A smaller angle of oscillation can make a more exact measurement although the frequency of oscillation is theoretically independent of amplitude.

In order to satisfy the previously mentioned assumptions in real measurements, the following things were carefully applied to the author's trifilar suspension.

- (1) A inserting hole on the platform was made of the same diameter as that of object in question so that it was positioned on the centre of platform. Thus, two platforms with a hole in the centre were fabricated by the author.
- (2) A tranquil place for experiments was chosen to be without any wind, noise and vibration.

(3) Timing of many consecutive oscillations was measured and then it was divided by the number of oscillations. The values of periods were measured from 9 to 10 trials and each trial involved the average period of more than 30 times oscillated.

(4) A monofilament fishing line of diameter 0.30mm was used for strings due to its negligible stiffness and low mass. Results were much different from the case of ordinary strings that had a thickness of about 5 mm diameter and gave values around 50 % higher.

Figure 3.14 shows the experimental setup for determination of mass moments of inertia using the trifilar suspension method. The input shaft components mounted on the platform and

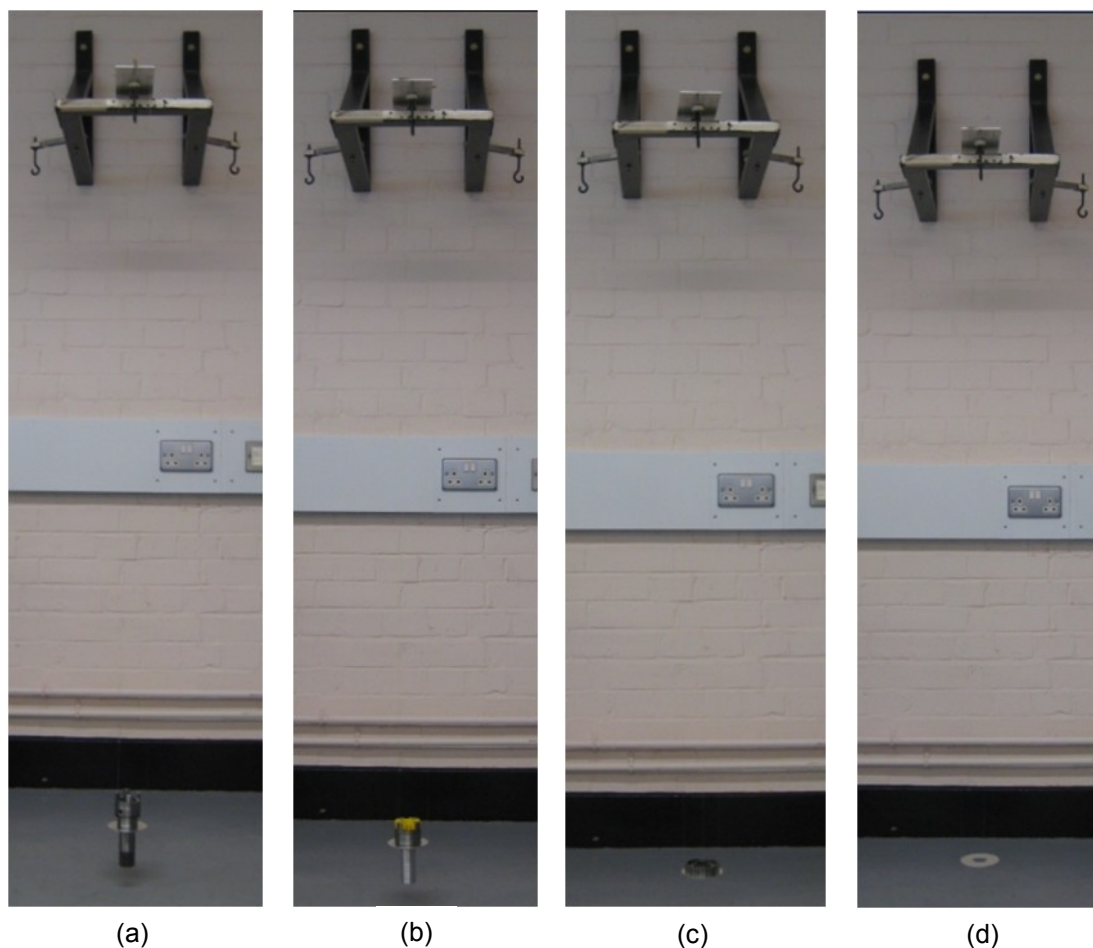


Figure 3.14 Experimental set-up of Trifilar suspension for measuring the mass moment of inertia (a) SHAFT-AXLE with HALF COUPING + Dia.35 circular plate ($I_1 + I_2$) (b) SHAFT-CLUTCH with KEY & CIRCLIP & HALF COUPING + Dia. Of 30 mm Circular plate ($I_1 + I_2$) (c) HUB-CLUTCH + Dia. of 35 mm circular plate ($I_1 + I_2$) (d) Circular Plate of Dia. of 35mm (I_2)

suspended by very thin fishing line are shown in Figure 3.14 (a), (b) and (c). The platform without any component was suspended for the measurements of the moment of inertia of it in Figure 3.14(d). For the platforms, two kinds of size of holes were made on two platforms respectively to fit the diameter 30 mm and 35 mm of shafts so that the centre line of gravity on the platform might coincide with their rotational axis exactly. In the caption of Figure 3.14, I_1 is the moment of inertia for a rotating component in question and I_2 is the mass moment for the platform having a hole of diameter 35 mm. I'_2 is the mass moment for the platform having a hole of diameter 30 mm and it can be substituted with I_2 in Equation 3.6. The thread length was measured each time, and a bit extended length was involved in it due to the increased mass on the platform when suspended.

3.3.2 Measurement System in Rig

A data acquisition device used in the test rig for measuring rotational speed is NI USB 6009 shown in Figure 3.15, which is compatible with LabVIEW and has eight analogue inputs with 14 bit and max sample rate 48kS/s. The temperature of lubricant as well as rotational speed was measured at each test.



Figure 3.15 NI USB 6009 for a data acquisition

3.3.2.1 Angular Velocity

Data acquisition device NI USB 6009, which transfers the data to LabVIEW in the computer, receives the signals from optical sensor (Transmissive Optoschmitt Sensor) shown in Figure 3.17 (b) and (c). Hall effect gear tooth sensor shown in Figure 3.16 (a) was first used, but was replaced with a kind of slotted optical switch (HOA 2001). This was because the size of the

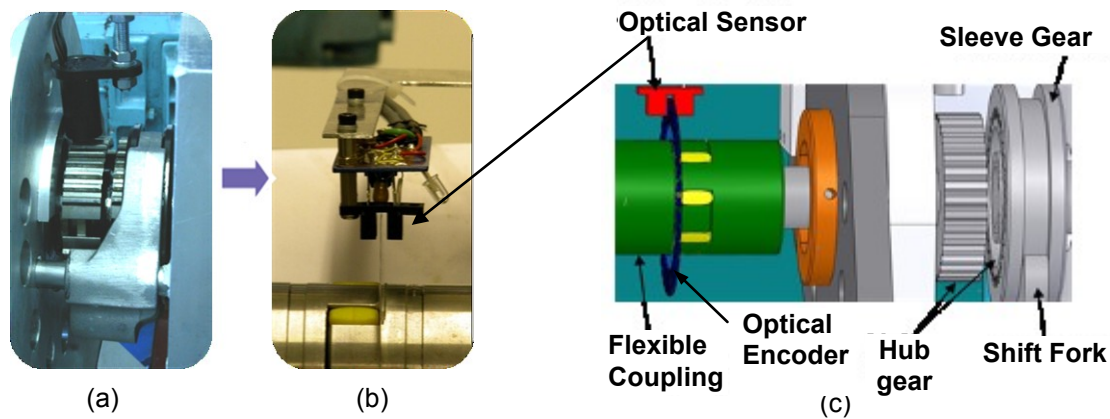


Figure 3.16 Speed acquisition apparatus (a) Gear teeth pick-up sensor (b) Optical sensor circuit unit (c) configuration of optical sensor and optical encoder mounted on the test rig

gear tooth was not enough to pick up clear signals and low speed was not detected completely even if steel poles were inserted on the three places of the periphery of hub gear for an effective magnetic pick-up. In the optical encoder for the optical sensor, the optical slots of 36 indexes were made. The schematic of the optical sensor circuit unit and the optical encoder were shown in Appendix 8.

3.3.2.2 Temperature

As previously described in Section 3.1.2, the lubricant temperature was measured by the thermocouple that was inserted to the inside of extension housing as shown in Figure 3.5 (a), and the opposite side of thermocouple was connected to a Fluke thermocouple module



Figure 3.17 A Fluke thermocouple module for an oil temperature measurement

shown in Figure 3.17. The oil temperature was checked at the start of run-down test and then measured again after the test. Not only the oil temperature but also the ambient temperature

was measured during the experiment.

3.3.3 Measurement Software

LabVIEW program was employed to acquire the data from the optical sensor circuit unit for measurement of rotational speed through the data acquisition device NI USB 6009. The diagram of LabVIEW program used in the tests is shown in Appendix 9.

3.4 Lubricant

The lubricant properties are very significant in the study of churning losses. The viscosity of lubricants is explained in this section.

3.4.1 Lubricant Viscosity (Tribology Lecture Note, 2007)

Viscosity is a measure of fluid friction as a ratio of shear stress to strain rate in the fluid. Viscosity is the single most important fluid property in lubrication since it not only determines lubricant film thickness, but also affects the churning losses. A lubricant film is used to separate two rubbing surfaces and to reduce the friction between them. The churning losses result in heat generation and the high viscosity oils may be less effective at convecting away this heat. High viscosity oil also flows slowly. Thus, the viscosity is the most important parameter in selecting gear oil, and an optimum viscosity of lubricant is required for lubrication systems.

Viscosity depends largely on temperature; and the Viscosity Index (VI) is a measure of how much the oil viscosity varies with temperatures. Mineral oils tend to have viscosity indices of

0 to 100, whereas synthetic, multigrades and severely hydrotreated mineral oils have VIs of up to 200. Viscosity is also dependent on pressure and generally increases approximately exponentially with pressure. This Barus equation is used to relate viscosity and pressure and is given by

$$\eta_p = \eta_0 e^{\alpha p} \quad (3.7)$$

$$\alpha = \frac{1}{\eta} \left(\frac{\partial \eta}{\partial p} \right)_T \quad (3.8)$$

Where η_p is the viscosity at pressure p , η_0 is the viscosity at zero pressure and α is pressure viscosity coefficient (approximately constant with pressure up to 10^8 Pa).

3.4.2 Lubricants

Two lubricants are used in this research on churning losses for splash lubricated axles, a low viscosity mineral oil and a high viscosity synthetic oil. SAE75W90, which was produced by Castrol and classified into API GL-5, was employed as the high viscosity synthetic oil, whereas the mineral oil, classified into API group I, is an Esso-produced paraffinic mineral oil. The lubricant SAE75W90 is widely applied to gear boxes such as manual transmissions as well as hypoid axles. Thus, it was chosen as a test lubricant for a high viscosity oil. In addition, the mineral oil was chosen such that its viscosity at ambient temperatures is similar to that of 75W90 at the high operating temperatures of axle housing.

3.4.3 Lubricant Viscosity Measurement

The viscosity of the lubricants employed in the test were measured using a Stabinger Viscometer as shown in Figure 3.18. This viscometer is based on the principle of a small

rotating measuring tube filled with oil inside which a cylindrical magnetic body floats. It can measure kinematic viscosity, dynamic viscosity and density simultaneously at varying temperatures (Novotny-Farks).



**Figure 3.18 SVM 3000
Stabinger Viscometer**
(<http://www.anton-paar.com>)

3.5 Test Configuration and Procedure

3.5.1 Initial Test and Calibration

Before running the hypoid axle gears, a similar run-in test was carried out for two reasons. One is that rotating components such as bearings and input shaft are required to accommodate some angular misalignment, if any exists, and the other reason is that all the components in transmission get into soaking up the lubricant. Moreover, it not only gets rid of very small burrs, but also expels tiny debris from the rotating components inside a gearbox. Thus, this similar run-in test of hypoid axle at an initial stage could make all the rotating components soaked in lubricant, self-alignment and ready for the efficient tests before starting the data acquisition. This test was once performed for around 20 minutes. For the calibration of measurement of rotational speed, a stroboscope shown in Figure 3.19 was used. Rotational speeds of 500, 1,000, 1,500 and 2,000 rev/min were calibrated with this stroboscope.



Figure 3.19 Stroboscope

3.5.2 Test Procedure

The test procedures to measure churning losses from axle housing or extension housing mounted on the test rig are as follows. The below procedures have been usually performed for all the churning tests so far. It could be changed with the awareness of safety according to its own purpose of research.

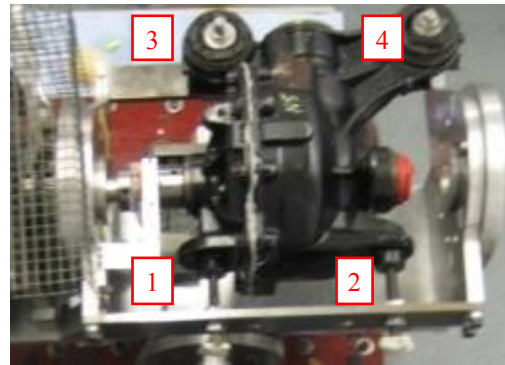


Figure 3.20 Mounting positions of Axle housing

1. Mount up axle housing on the rig carefully to prevent damage to the sensor for measurement of speed and other components in the rig while carrying it.
2. Fit the circular optical encoder plate for a speed-measurement into the half coupling connected to the input shaft at the side of clutch device.
3. Place the axle housing exactly at the positions supported by four points (1, 2, 3 and 4 in Figure 3.20) and then fasten the bolts and nuts to until the stub engaged into crown wheel by spine coupling is able to connect easily to input shaft by using the coupling.
4. Connect the coupling between the stub from axle housing and the input shaft from clutch system and then adjust the bolts of position 3 and 4 through running the shaft in order to avoid misalignment.

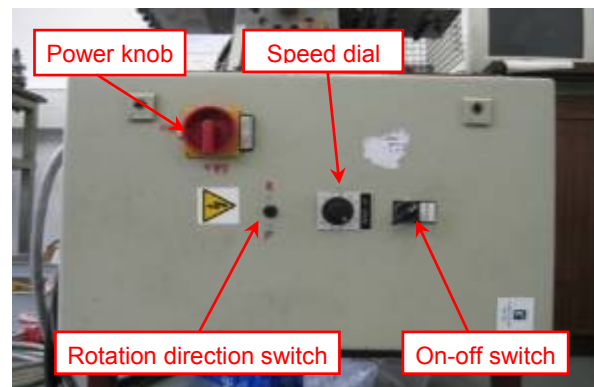


Figure 3.21 Motor controller

5. Install the speed-measurement sensor circuit unit to the appropriate position
6. Regulate the speed by rotating speed dial shown in Figure 3.21 to the required speed scale.

7. Turn on the on-off switch shown in Figure 3.21 to start the motor.
8. Run the LabVIEW program and then disconnect the power transfer sliding the sleeve gear as soon as the speed reached the required one (a bit higher than 2000 rev/min).
9. Turn off the motor as soon as the sleeve gear is disconnected.
10. Stop running the LabVIEW program after the axle slows down to a halt.
11. Change the rotational direction using a switch shown in Figure 3.21 if needed.

For churning tests in this research, more than five times of trials under a certain same condition at every test were made to get the more exact mean values of the deceleration speed slope for them.

3.5.3 Design of Experiments

A systematic and concise way to perform the experiments for measuring churning losses was designed in order to achieve maximum information with minimal experimental tries when numerous variables are involved. The variables considered in author's experiments and those conditions are as shown below.

- (1) Oil immersion depth: 31, 47, 64.25, 71 and 91 mm in the extension housing
- (2) Oil volume: 200, 500, 800, 1000 and 1200 cm³ in the production axle housing
- (3) Maximum rotational speed: run-down from up to 500, 1000 and 2000 rev/min
- (4) Viscosity of lubricant: 75W90 and mineral oil at the ambient temperature
- (5) Diameter of ring gear: tip diameter 213.2 and 222 mm
- (6) Tilting degrees (pitch): pitch $\pm 15^\circ$, $\pm 30^\circ$
- (7) Tilting degrees (rolling): rolling $\pm 15^\circ$, $\pm 30^\circ$
- (8) Radial clearances: the inner diameter of radial baffles 226, 234.8 and 243.6 mm,

the inner diameter of housing 290 mm in the extension housing

(9) Axial clearances: the teeth side of ring gear 2.5, 7.5, 12.5, 22.5, 62.5 mm

the rear side of ring gear 12, 17, 22, 32 mm

Out of the above variables, the volume of oil depends on the depth of oil for a given enclosure. In addition, the radial clearance and the immersion depth vary with the diameter of ring gear. For all the experiments, the test condition of varied immersion depths was carried out with the other conditions.

3.6 Concluding Remarks

The test rig concept for churning losses, axle housing, extension housing, experimental methodology and procedures have been presented in this Chapter. The following Chapter, Chapter 4, presents the experimental result of churning losses in the production axle housing including experimental data of moment of inertia and table of lubricant viscosity.

CHAPTER 4

EXPERIMENTAL RESULTS USING PRODUCTION AXLE HOUSING

4.1 Introduction

As reviewed in Chapter 2, a variety of factors such as oil immersion depth, viscosity of lubricant and geometric characteristics influence the losses of spur gears in the dipped lubrication systems. Therefore, a controlled test program of hypoid geared rear axles is needed to determine what losses take place in the gear housing, to assess how the churning losses occur and to determine by which factors the losses are affected and which is most dominant. This study is able to provide a qualitative and quantitative evaluation of churning losses for axles composed of hypoid gears, and also can be used as a basis for suggesting the ways of reducing the losses of gear boxes.

A test rig for the measurement of churning losses in hypoid axles was designed, manufactured

and commissioned as shown in Chapter 3. The measurement is based on the inertia-run-down method and a series of churning loss tests have been carried out on both production axle assemblies and the extended housing. Variables included ring-gear diameter, radial clearance and oil fill level.

In the present chapter, the results of measuring mass moments of inertia for the input shaft components in the test rig and the hypoid gears in axles, as well as the test results of churning losses from the production axle housings, are presented. First, two different methods for obtaining the moment of inertia were used and these were compared in order to find the best value to use to calculate the exact churning losses from the speed test data. Second, the test results from the real axle housings, which involved several factors such as oil quantity with fill level, revolution speed, wheel diameter, presence of pinion gear and vehicle attitude. Furthermore, two types of lubricants at room temperature were employed to investigate the effect of lubricant viscosity. The viscosity of the lower viscosity oil at room temperature (20°C) was the same as that of the other oil at 70°C.

For the experimental technique, as described in the last Chapter, the drag torque on the hypoid gear submerged in oil was measured five times for each condition. Then, the drag torque measured in the absence of oil at each condition was subtracted from those in order to calculate the spin power losses arising from the lubricant churning.

4.2 Moment of Inertia of Rotating Parts

The values of moment of inertia for the rotating input shaft components with regard to a rotating axis are required in order to calculate drag torque and losses using the test data of

run-down graphs. The moment of inertia of the rotating component was first calculated using the 3D modelling package Solid Works, based on the measured properties such as mass and volume per each component. Secondly, the trifilar suspension method described in Section 3.3.1 was carried out to measure the undamped period of suspended oscillation and then these were finally compared to check for good agreement between both methods.

4.2.1 Calculation from Solid Modelling

Mass moments of inertia for rotating input shaft components connected to crown wheel assembly were calculated by using Solid Works program as shown in Table 4.1. The 3D modelling for each component was made to full scale after measurement of its dimensions, mass and volume, which was followed by the verification process where the calculated density was compared with the density from published material data. Thereafter, the moment of inertia in question was calculated from the 3D solid model. The coupling was separated to two pieces for the calculation in Solid Works.

The hypoid axle is composed of a crown wheel and a pinion gear. Two different sizes of ring gears were considered in this research as described previously. The data of mass moments of inertia of the crown wheel assembly and the pinion gear assembly, shown in Table 4.2, were obtained from 3D models in Solid Works, and were almost the same as the data provided by the motor company.

Table 4.1 Moment of inertia for input shaft components calculated from the modelling

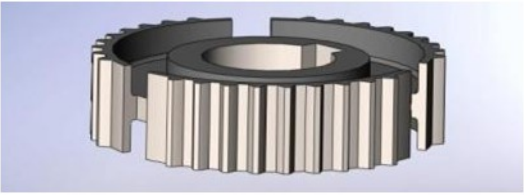
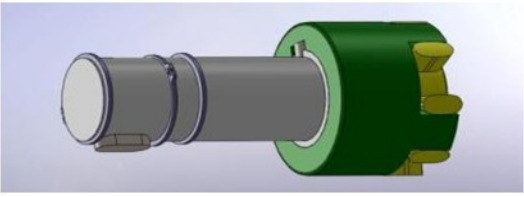
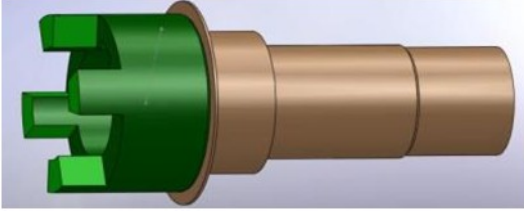
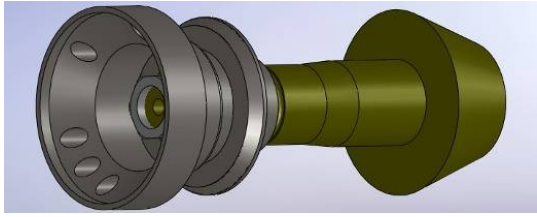
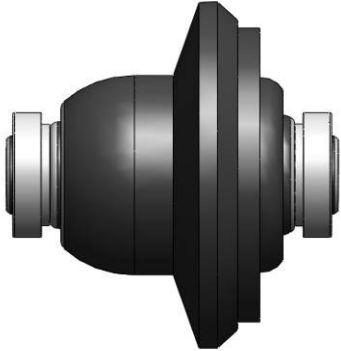
Component		Mass [g]	Volume [mm ³]	Moment of inertia [kg·m ²]
Hub gear-clutch		271.1	39,862.5	0.0002197312
Shaft-clutch with key, circlip and a half coupling		649.2	147,830.8	0.0002382486
Stub with a half coupling		1,380.3	177,391.9	0.0003811395

Table 4.2 Moment of inertia of hypoid gear assembly

	Pinion gear assembly (M220)	Ring gear assembly (M220)
Hypoid gear assembly		
Mass moment of inertia from Solid Works [kg·m ²]	0.00400675	0.0604127
Mass moment of inertia by the company [kg·m ²]	0.00396268	0.0617594
Difference	1.11 %	-2.18 %

4.2.2 Measured Results of Trifilar Suspension Method

A detailed description of the trifilar suspension method was previously made in Section 3.3.1. The measured results using the trifilar suspension method are shown in Table 4.3. In this table, I_1 represents the object component, I_2 the platform having the hole of diameter 35 mm and I'_2 the platform having the hole of diameter 30mm. The measurement of period T_{period} was carried out more than nine times and its value was calculated as the average. In each case, the mass of the component and platform or just the platform alone was measured, as well as the length of the strings when suspended. The values of moment of inertia were then calculated by using the Equation 3.6. When the moment of inertia of hub gear was measured, the real sized hub with pitch diameter 78.32 mm was replaced by one of the same shape but a smaller size of pitch diameter 69.5 mm, because the two real sized hubs were already press-fitted to the shafts in the test rig. These hub gears were supplied by motor company and all the dimensions are recognized by the author. The mass moments of inertia of same shaped hubs of different size are proportional to the mass and the square of the pitch radius of different sized one, and so the real value can be calculated as,

$$I = I' \times (m/m') \times (R_p/R'_p)^2 \quad (4.1)$$

where I , m and R_p are respectively moment of inertia, mass and pitch diameter of real sized hub in question. I' and m' are moment of inertia and mass for the smaller sized hub with pitch diameter 69.5 mm. If the measured values are input to the corresponding variables, Equation 4.1 can be calculated by

$$\begin{aligned} I &= 0.00013185 \times (271.07 / 200.1) \times (76.2 \times 0.5 / (0.5 \times 69.5))^2 \\ &= 0.0002147 \end{aligned} \quad (4.2)$$

Table 4.3 Measured data for moment of inertia

Trial		Parts				
		HUB-CLUTCH + 35 mm Dia. Circular platform ($I_1 + I_2$)	SHAFT-CLUTCH with KEY & CIRCLIP & HALF COUPING + 30 mm Dia. Circular platform ($I_1 + I'_2$)	SHAFT-AXLE with HALF COUPING + 35 mm Dia. Circular platform ($I_1 + I_2$)	Circular platform of Hole Diameter of 30 mm (I'_2)	Circular platform of Diameter of 35 mm (I_2)
Period T_{period} (sec)	1	1.6217	1.2690	1.0803	2.0060	2.0105
	2	1.6183	1.2663	1.0810	1.9585	2.0145
	3	1.6005	1.2627	1.0850	2.0150	2.0105
	4	1.6130	1.2563	1.0847	1.9545	2.0120
	5	1.6123	1.2530	1.0860	1.9505	2.0125
	6	1.6140	1.2673	1.0857	1.9885	2.0115
	7	1.6370	1.2583	1.0900	1.9665	2.0115
	8	1.6173	1.2540	1.0797	1.9680	2.0140
	9	1.6167	1.2587	1.0803	1.9760	2.0120
	10	1.6167	1.2503	1.0803	-	2.0120
	Average	1.617	1.260	1.083	1.976	2.012
Mass m (g)		219.37	639.84	1,363.61	19.50	18.47
Thread length l (mm)		1,563	1,595	1,645	1,574	1,584
Moment of inertia (kg m ²)		1.51324 E-04	2.62536 E-04	4.01361 E-04	1.99523 E-05	1.94726 E-05

Therefore, the values of moment of inertia of the rotating input components in the test rig are as shown in Table 4.4. The difference between the values obtained from 3D modelling in Solid works and the measured values is just 0.008% and they are in good agreement with each other.

Table 4.4 Comparison between calculated and measured figures of moment of inertia

Component	Moment of inertia calculated in Solid Works [kg m ²]	Moment of inertia in experiment [kg m²]	Difference of measurement to calculation
Hub gear-clutch	0.00021973	0.00021471	-2.2834 %
Shaft-clutch with key, circlip and a half coupling	0.00023825	0.00024258	+1.8198 %
Stub with a half coupling	0.00038114	0.00038189	+0.1965 %
TOTAL	0.00083912	0.00083919	-0.008 %

4.2.3 Calculation of Moment of inertia of individual Components

In this section, the total moment of inertia of the ring gear in the axle linked to the rotating shaft components in the test rig is obtained. The total mass moment is divided into two groups depending on the size of the hypoid axle gear, of which the gear ratio is, however, the same 3.30769. The tip diameters of the ring gears considered in this research are 222 mm (pitch diameter 220 mm) and 213.2 mm (pitch diameter 210 mm). In addition, all the experiments for each group were performed both on the crown wheel with engaged pinion drive gear and without the pinion gear.

Table 4.5 summarizes the calculated results of total moment of inertia of crown wheel gear assembly, meshed with pinion gear, and the crown wheel assembly alone, each of which includes the moment of inertia for the rotating input shaft components connected to the crown wheel by a spline coupling inside the housing on the test rig.

Table 4.5 Sum of mass moments of inertia for all rotating components

Part		Mass [g]	Moment of inertia [g·mm ²]	Moment of inertia [kg·m ²]
220 Dia. Hypoid Gear Ratio 3.31	Pinion shaft assembly	5,040.0	3,962,680.0	0.0039626800
	Crown wheel gear assembly	13,753.9	61,759,400.0	0.0617594000
210 Dia. Hypoid Gear Ratio 3.31	Pinion shaft assembly	5,018.0	3,910,000.0	0.0039100000
	Crown wheel gear assembly	13,460.0	57,200,000.0	0.0572000000
Rotating input shaft components		2,300.6	839,119.319	0.0008391193
Crown wheel gear assay	220 Dia. Gear 3.31	0.0625985193		
	210 Dia. Gear 3.31	0.0580391193		
Crown wheel gear assay meshed with pinion gear	220 Dia. Gear 3.31	0.1060140377		
	210 Dia. Gear 3.31	0.1008774703		

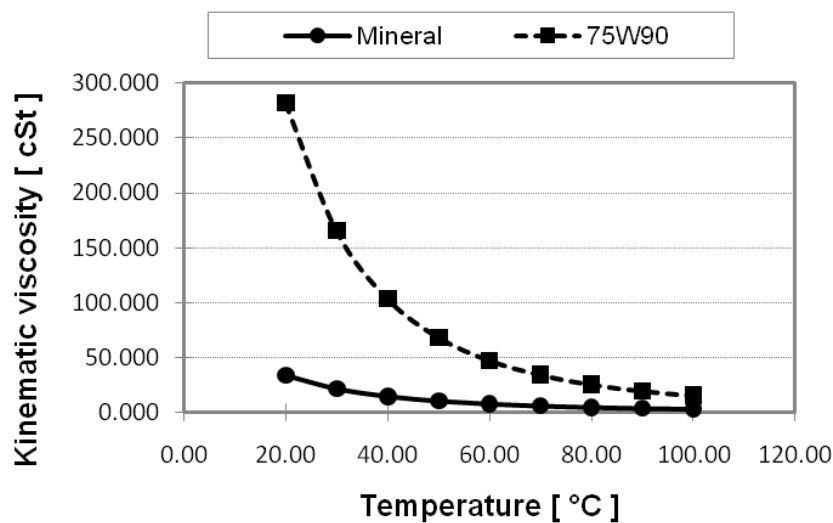
4.3 Viscosity of Lubricant

As previously mentioned in Section 3.3.2, a Stabinger viscometer was used to measure the properties of the lubricants employed in this experimental study. Table 4.6 shows the measurement results for viscosity and density at 20°C and 70°C of the two lubricants employed in this experimental study, Esso API group I paraffinic mineral oil and Castrol SAE75W90. The viscosity of 75W90 at 70°C is almost the same as that of the mineral oil at 20°C, and the working temperature range of economic driving in axle housings is typically from 70°C to 100°C, although it varies according to the environmental temperature. Thus, the churning losses of axle housings at working temperatures can be measured using the mineral

Table 4.6 Comparison of oil properties at 20°C with those at 70°C

Temperature	75W90		Mineral	
	Density [g/cm ³]	Kinematic viscosity [mm ² /s]	Density [g/cm ³]	Kinematic viscosity [mm ² /s]
20°C	0.8556	282.09	0.8522	33.882
70°C	0.8249	34.309	0.8204	6.2353

oil at ambient temperature without heating the lubricant in the housing. Figure 4.1 compares the kinematic viscosity between SAE75W90 and mineral oil over the temperature range of 20 to 100°C. From this figure, SAE75W90 is the high viscosity oil and the mineral oil is the low viscosity oil. SAE J306 viscosity grades for axle and manual transmission oils are presented in Appendix 4.

**Figure 4.1 Viscosity-temperature characteristic for lubricants 75W90 and mineral oil**

The low viscosity oil has the highest kinematic viscosity at 20°C with $\nu = 33.9 \text{ mm}^2/\text{s}$ and the lowest one at 100°C with $\nu = 3.4 \text{ mm}^2/\text{s}$. The high viscosity oil is more viscous than mineral oil at lower temperature and has the highest kinematic viscosity at 20°C, with $\nu = 282.1 \text{ mm}^2/\text{s}$, whereas it has a fivefold increasing viscosity, $\nu = 15.5 \text{ mm}^2/\text{s}$, at 100°C compared to

the low viscosity oil. The measured values of density, dynamic viscosity and kinematic viscosity for the high viscosity oil and the low viscosity oil at a range of temperature from 20 to 100°C are shown in Appendix 4. Especially, the properties between 20 and 35°C were measured at 1°C increments because all the tests were performed at ambient temperatures within this range.

4.4 Churning Test results for Geometry and Oil Viscosity

In this section, the test results for the geometry of crown wheel and the calculation of churning losses from the measured losses are presented. In addition, the test results for the effect of lubricant viscosity on churning losses are followed by test results for the meshed hypoid gears together.

4.4.1 Crown Wheel Gear with Diameter 210mm

4.4.1.1 Tests with Taper Roller Bearing for Crown Wheel

The test conditions in this section are indicated in Table 4.7. The smaller diameter of ring gear was employed without the mating drive pinion gear in the production axle housing filled with

Table 4.7 Test condition for test results of Figures 4.3 ~ 4.7

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Housing hole for pinion gear
Condition	Production axle housing	M210	Production Taper roller bearing	The low viscosity oil / 20~23.7°C	Blocked by a circular steel plate



Figure 4.2 Seal of the hole for pinion drive gear

the low viscosity oil, 200, 500, 800, 1,000 and 1,200 cm^3 respectively. The observation error of oil volume depending on the immersion depth appeared to be within $\pm 50 \text{ cm}^3$ difference over the all the tests. The hole in which the pinion gear is installed in the housing was obstructed by a steel plate at the inside with sealant from outside in order to prevent oil

from flowing out through the hole as shown in Figure 4.2. This condition was intended to run the hypoid crown wheel submerged in the lubricant without being engaged to the pinion and thus to measure the churning losses caused by crown wheel alone. Figure 4.3 indicates the decay curves of crown wheel of 210 mm dia. running down from 2,000 rev/min to zero as the graph of time versus RPM according to the condition of varied oil volumes from 0 to 1,200 cm^3 .

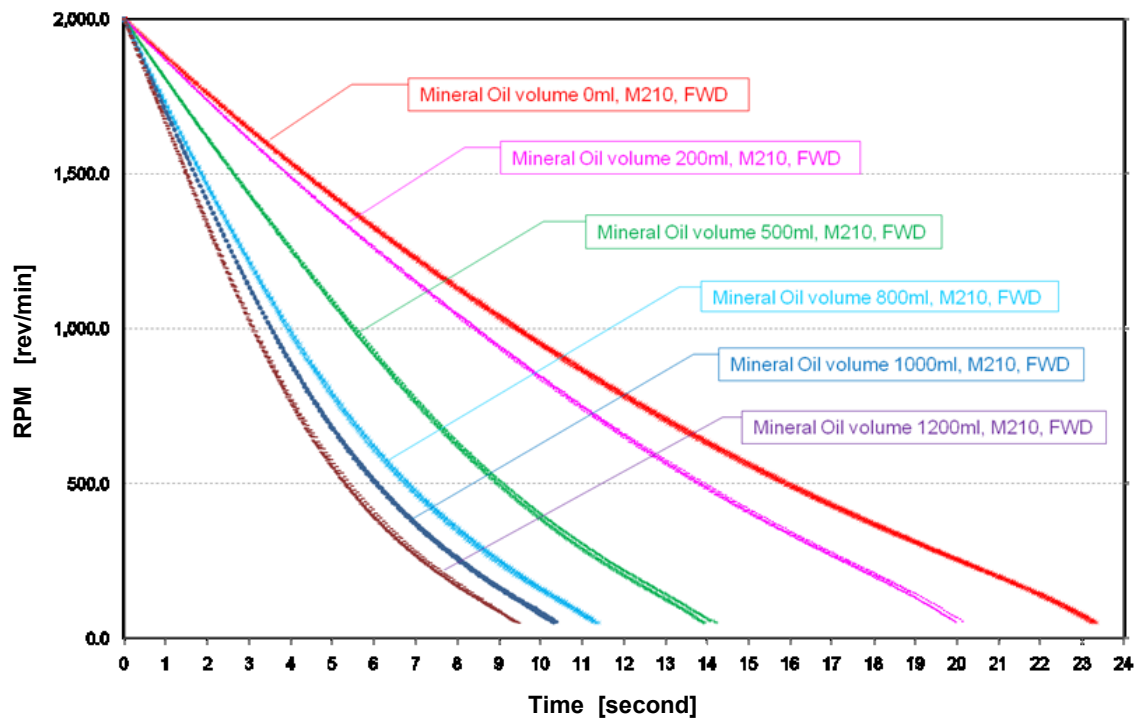


Figure 4.3 Rotational speed decay curve of the ring gear of 210 mm dia. with regard to time

Out of them, 1000cm^3 is correspondent to the normal oil quantity of axle in field service. Run-down speeds were measured more than five times at each oil quantity and five run-downs of data were selected, except the data of initial trials. This is because the data of initial trials were more affected by the bearing friction and shaft misalignment. The speed of the crown wheel submerged in the low viscosity oil decreases fast from the starting 2000rev/min of free run-down and then its deceleration decreases gradually until the speed reaches near to zero. The time of the speed decay in the range of low rotational speed is generally longer than in an equivalent range at high speed. The larger the oil quantity is, the steeper is the slope of the graph.

The data of decelerated rotational speeds were acquired at a rate of 16 per a second through LabVIEW and the decay seems likely to be logarithmic decrement without any inflection point. Thus, the 3rd order polynomial regression was applied to the data and shows a good fit of experimental result, which is explained in Section 4.6. The decelerations were obtained by differentiating this regression graph of decay data as shown in Figure 4.3 with respect to time for the different oil volumes. The drag torque was calculated by multiplying the moment of inertia shown in Section 4.2.3 with the differentiated speed with respect to time. The power loss is the product of drag torque and its corresponding speed. Furthermore, the torque was obtained from tests with no oil in the same way after draining the oil in the axle housing, and this torque is named no-oil drag torque in this thesis. Finally, the churning drag torque was calculated by subtracting the no-oil drag torque from the above total drag torque obtained at each oil level as shown in Equation 4.3.

$$T_{ch} = T_{total} - T_{no\ oil} \quad (4.3)$$

The total drag torque is comprised of the churning and windage torques caused by the interaction among rotating components, air and lubricant, and the friction torques due to the

gear, bearing and seal. Thus, the measured no oil torque $T_{no\ oil}$ of ring gear 210 mm dia. in the absence of oil in the oil sump was presented in Equation 4.4.

$$T_{no\ oil} = T_{gear} + T_{bearing} + T_{seal} + T_{windage} \quad (4.4)$$

Therefore, Figure 4.4 indicates total drag torque T_{total} and Figure 4.6 indicates total power losses P_{total} . Figure 4.5 represents the churning torque T_{ch} that was obtained by subtracting the no-oil drag torque from the total drag torque, and churning losses P_{ch} are shown in Figure 4.7. The total drag torque plotted in Figure 4.4 was found by multiplying the moment of inertia shown in Section 4.2.3 with the deceleration at that time, the differential of speed with respect to time (the slope of the regression lines in Figure 4.3 and these regression lines were 3rd order regression graphs). Then, the churning torque shown in Figure 4.5 was obtained subtracting the no-oil drag torque from the total drag torque shown in Figure 4.4. The churning losses were also obtained in the same way from the total power losses in Figure 4.6. Thus, the data of churning drag torque and churning losses were obtained to plot the regression graphs shown in Figure 4.5 and 4.7 at corresponding oil levels, using Equation 4.3 and 4.5 to isolate the contribution of churning losses.

Generally, the drag torque increased with rotational speed to the power of a number less than one, while the power losses showed a tendency of increase with rotational speed to the power of number more than one. The churning torque shown in Figure 4.5 increases with oil immersion depth according to oil volume as well as rotational speed. In this graph, the churning torque for the smallest depth and smallest volume 200 cm³ of oil remains almost constant over all the range of rotational speed, and even appears a slight decrease above about 1500 rev/min. What has to be noticed is that the churning torque for 500 cm³ as well as 200 cm³ increases with rotational speed, but begins to decrease from around 1500 rev/min even if the decrease is only 5 % up to 2000 rev/min. Accordingly, the increasing rate of churning

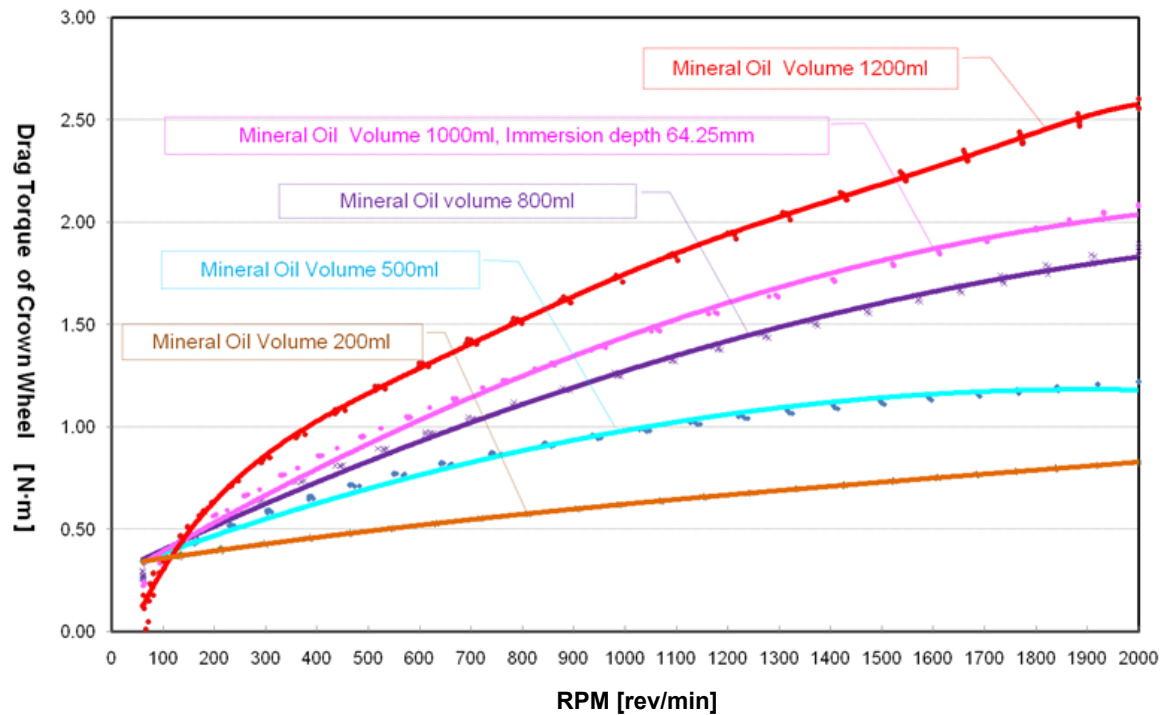


Figure 4.4 Total drag torque of the ring gear of 210 mm dia. rotating forward in axle housing and depending on the low viscosity oil volumes and levels

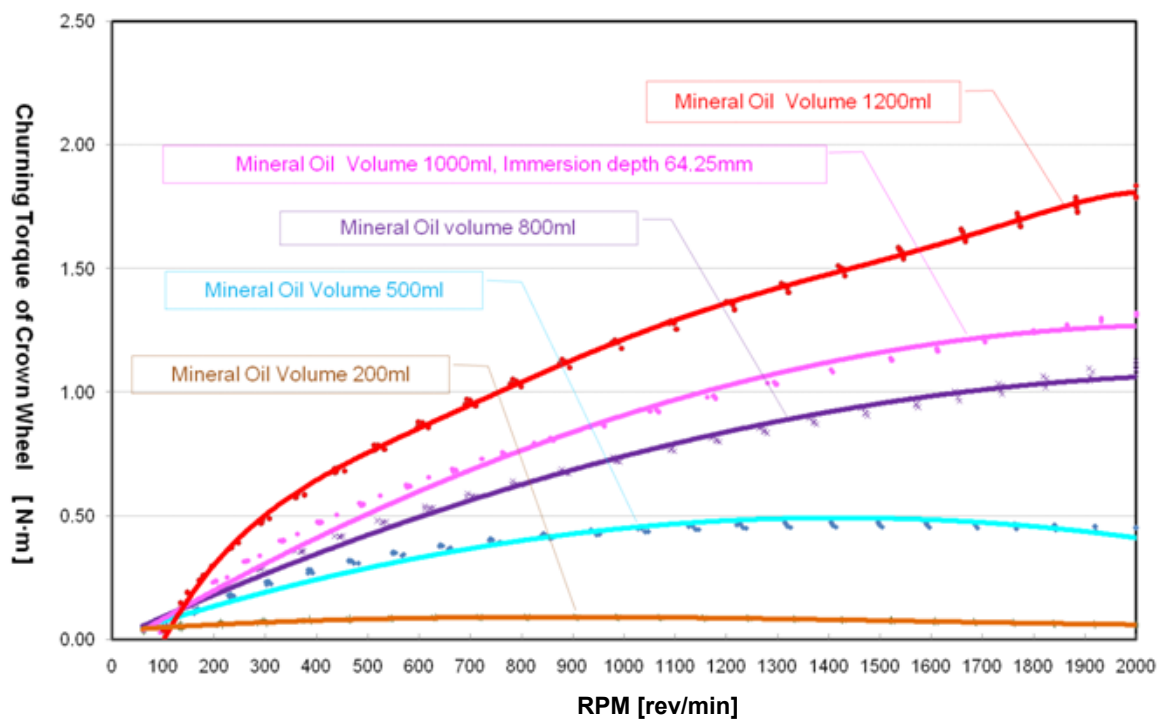


Figure 4.5 Effect of the low viscosity oil volume and oil level on the churning torque of the ring gear of 210mm dia. rotating forward in axle housing

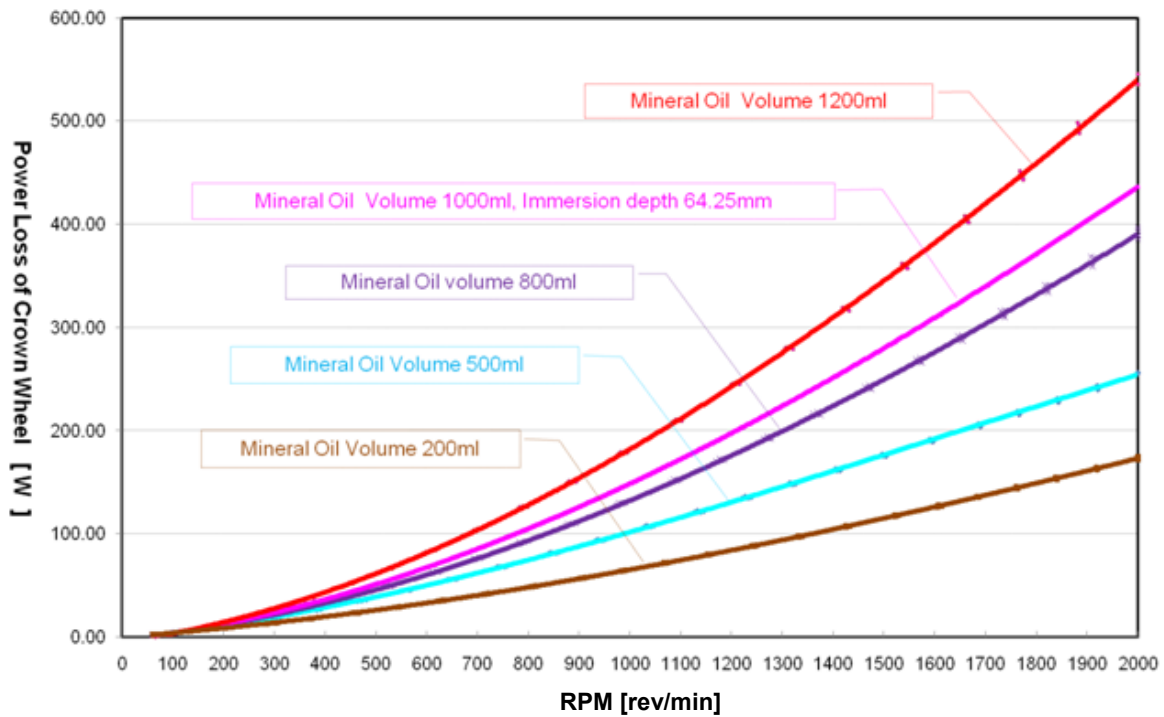


Figure 4.6 Total power losses of the ring gear of 210 mm dia. rotating forward in the axle housing and depending on the low viscosity oil volumes and levels

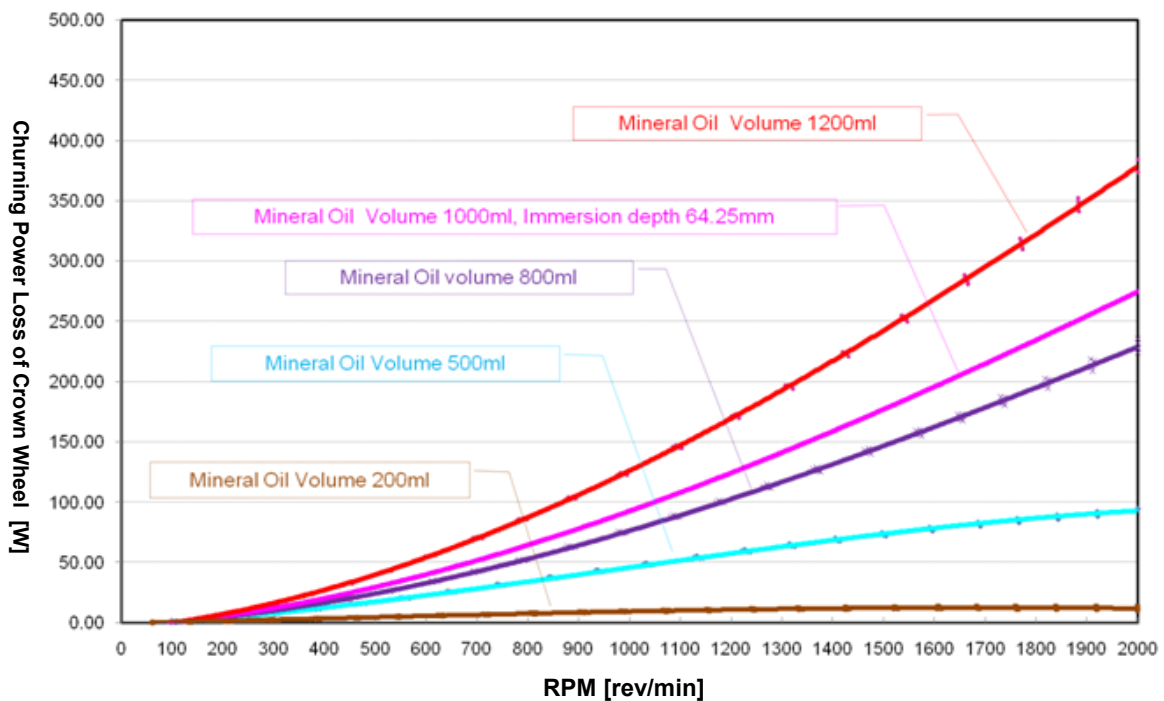


Figure 4.7 Effect of the low viscosity oil volume and oil level on the churning losses of the ring gear of 210 mm dia. rotating forward in the axle housing

losses diminished from about 1500 rev/min in Figure 4.7. Thus, it was found that the churning drag torque decreased for the condition of small immersion depth in the low viscosity oil from a certain higher speed. This is a point to which the author shall return later in other test results and in Chapter 6.

The behaviour of churning torque and churning losses for the hypoid axle filled with an oil volume of $1,000\text{cm}^3$ in Figure 4.5 and 4.7 corresponded to an oil immersion depth at 64.25 mm for the axles of vehicles in field service. The churning torques at 500, 1000, 1500 and 2000 rev/min are approximately 0.5, 0.9, 1.15 and 1.3 Nm and those corresponding churning losses are around 30, 75, 150 and 225 W respectively. The experimental error appeared to have an overall reproducibility of about $\pm 0.03\text{Nm}$ and $\pm 3\text{W}$ for torque and power loss, which is presented in Section 4.6.

4.4.1.2 Tests with Deep Groove Ball Bearing for Crown wheel

At the beginning of this experimental research, tests were performed with a different bearing type and the different way of blocking the housing hole. The results are presented in Figure 4.8~4.10, test conditions of which are shown in Table 4.8. Thin section deep groove ball bearings were employed for the crown wheel instead of taper roller bearings and the housing hole for pinion gear was blocked by a rubber bung instead of a steel plate.

Table 4.8 Test condition for the test results of Figures 4.8 ~ 4.10

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Housing hole for pinion gear
Condition	Production axle housing	M210	Thin section deep groove ball bearing	The low viscosity oil / 20~23.7°C	Blocked by a rubber bung

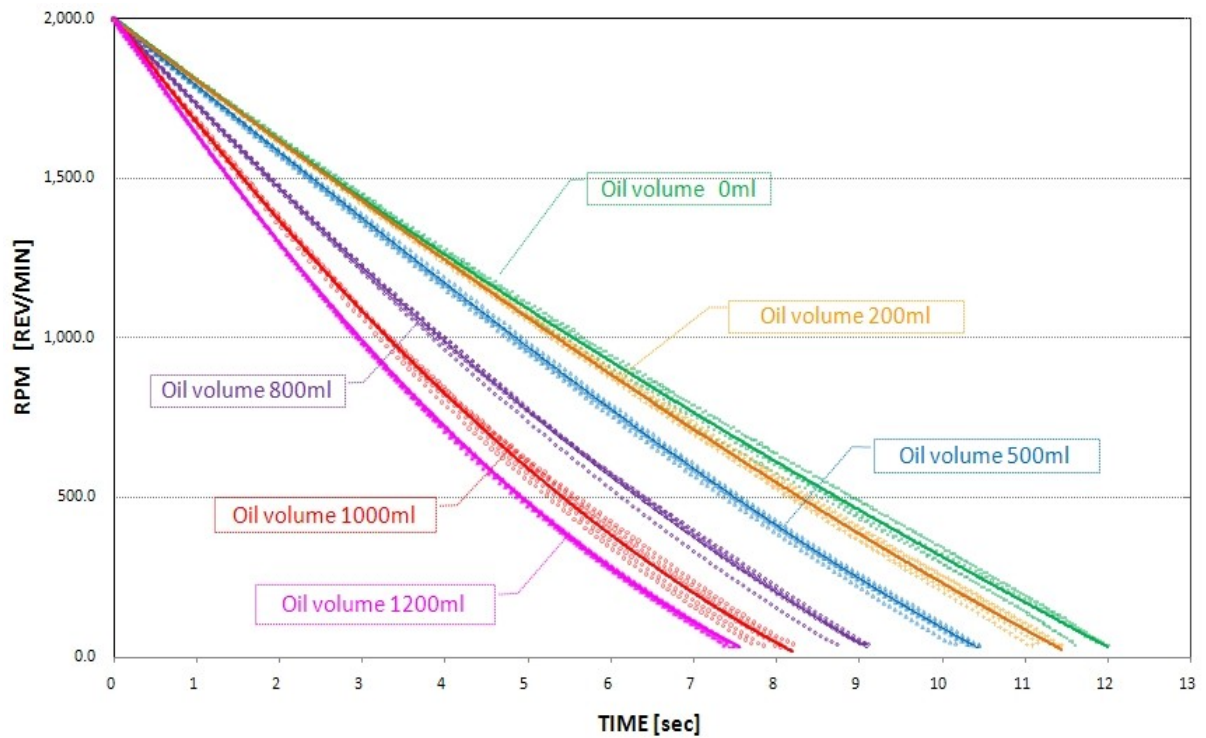


Figure 4.8 Speed decay curve of the ring gear of 210 mm dia. with regard to time

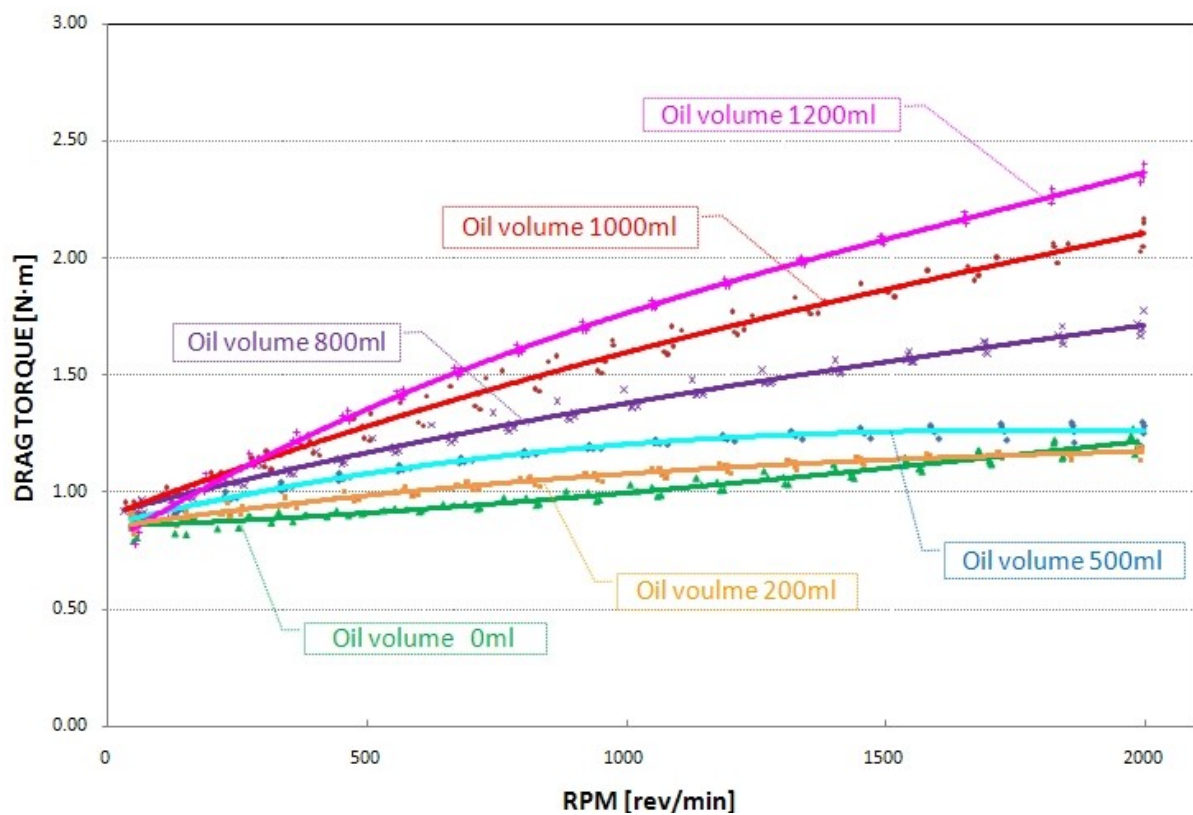


Figure 4.9 Drag torque of the ring gear of 210 mm dia. with regard to RPM

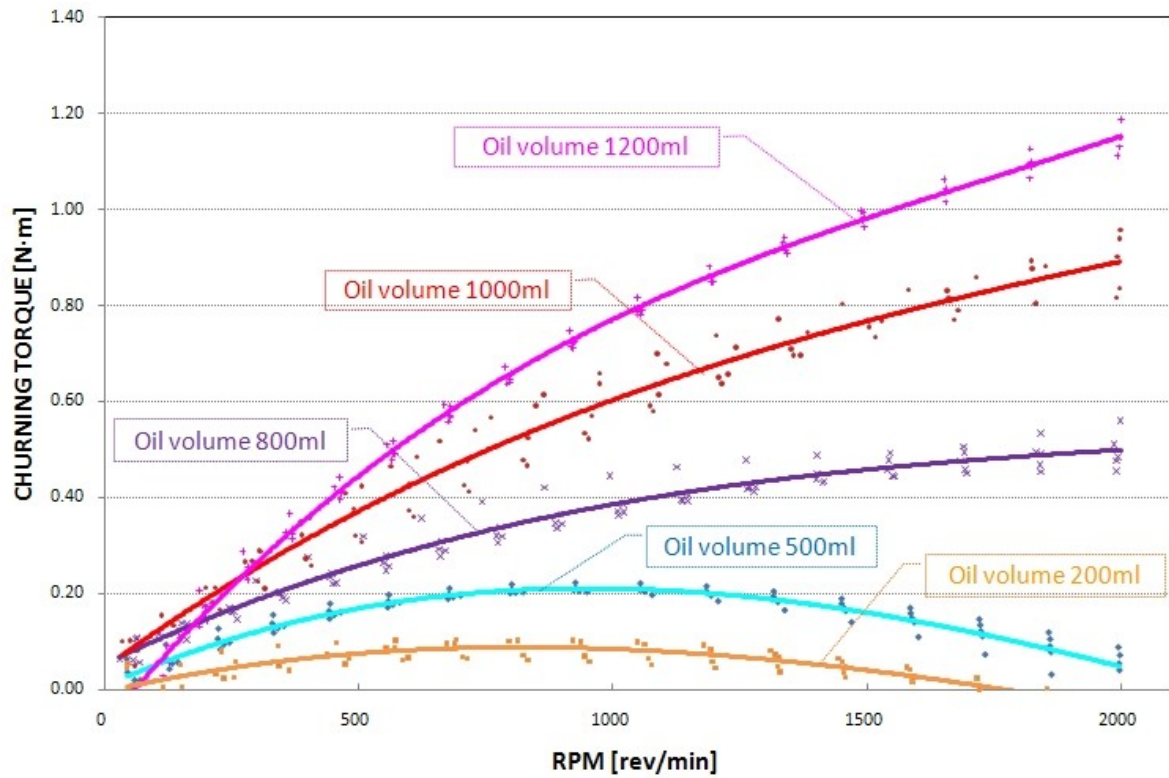


Figure 4.10 Churning torque of the ring gear of 210mm dia.

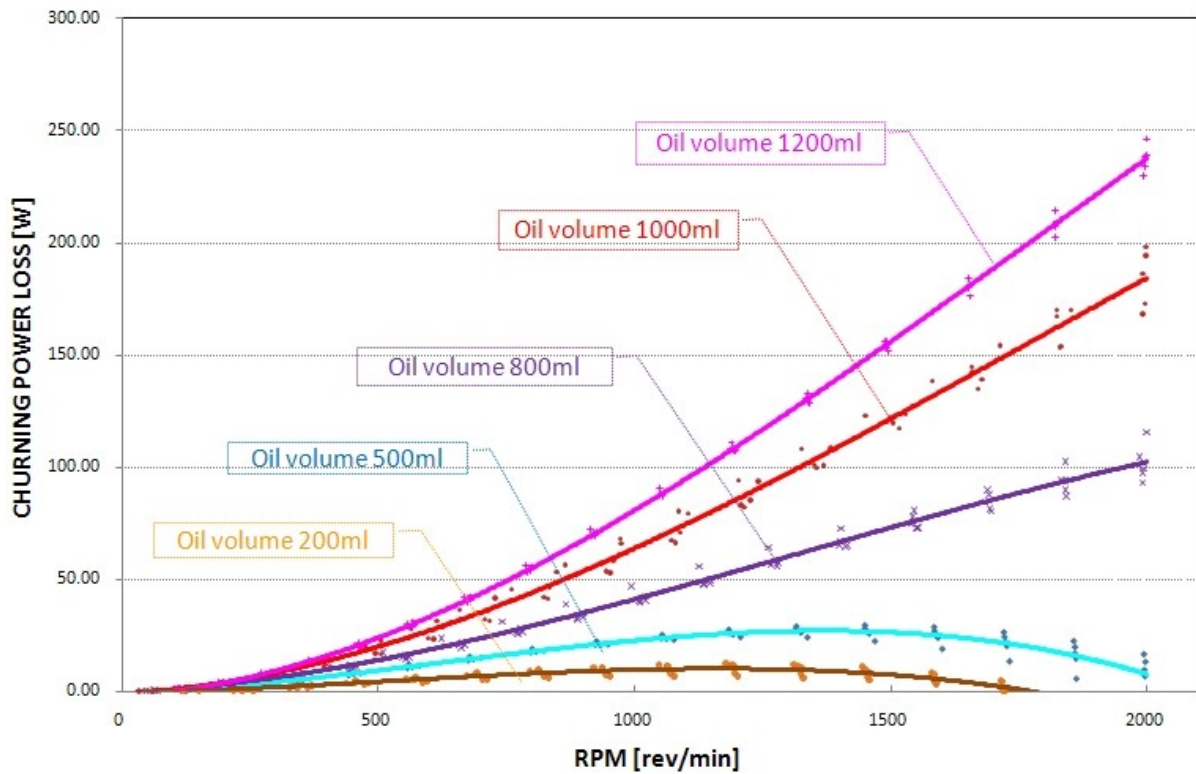


Figure 4.11 Churning power losses of the ring gear of 210 mm dia.

Although run-in tests were performed for the bearings to minimize the effects of the friction losses, bearing friction losses were measured to be higher than for taper roller bearings from a production axle. This seems to be mainly due to insufficient run-in and tight bearing inner tolerance resulting from special size of inner diameter of crown wheel housing shaft. Figure 4.8 indicates the decay curves of crown wheel with diameter of 210 mm running down from 2000 rev/min to zero as the graph of time versus rotational speed according to the condition of varied oil volumes from 0 to 1200 cm³. The differentiation of this regression graph of decay data with respect to time gives the deceleration for each oil volume respectively. By multiplying this deceleration with the moment of inertia shown in Section 4.2.3 at that time, the drag torque was plotted in Figure 4.9. Thus, the data of churning drag torque and churning losses were obtained to plot the regression graphs shown in Figure 4.10 and 4.11 at corresponding oil levels by using Equation 4.3 in order to isolate the contribution of churning torque and losses.

The brown-coloured line regression graphs with an oil volume of 1000 cm³ in Figure 4.10 and 4.11 show the behaviour of the churning drag torque and the churning losses for the hypoid axle filled with the same oil volume of 1000 cm³ as vehicles in field service, and this volume corresponds to an oil immersion depth of 64.25 mm. For a reference, the corresponding velocity of vehicles according to the rotational speed of the hypoid crown wheel was suggested in Appendix 10. The churning torques at 500, 1000, 1500 and 2000 rev/min are approximately 0.38, 0.60, 0.78 and 0.9 Nm and the corresponding churning losses are around 20, 65, 120 and 180 W respectively. These lower values compared to the previous case resulted from the higher friction torque of the bearings of the crown wheel. The inner tolerance of those bearings was tight and these tests were performed first when the test rig was built. It can be also thought that the oil flowed into the space made by a rubber bung and the oil level was reduced.

The salient point to observe is that the churning torque for 500 cm³ and 200 cm³ increases with rotational speed, but begins to decrease from around 1000 rev/min that is a lower speed than the case previously described in this section. Further discussion on this point will be presented in Chapter 6.

4.4.2 Crown Wheel Gear with Diameter 220mm

4.4.2.1 Tests with Taper Roller Bearing for Crown Wheel

As discussed above, the same procedures were applied to obtain the churning torque and churning losses for crown wheel with 220mm diameter. Test conditions in this section are also indicated in Table 4.9 and are the same conditions as those of Table 4.7 except for the size of ring gear.

Table 4.9 Test condition for test results of Figures 4.12 ~ 4.13

tem	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear
Condition	Production axle housing	M220	Production Taper roller bearing	The low viscosity oil / 19.1~24°C	Blocked by a circular steel plate with sealant

The regression graphs of the decay curves for the crown wheel of diameter 220mm were compared with the Figure 4.3. The time consumed near to zero speed, for the same oil volume, in the crown wheel of 220mm dia. is longer than that of 210mm dia. from this comparison. Figure 4.12 indicates the regression curve of crown wheel with the diameter of 220 mm as churning drag torque versus RPM and Figure 4.13 shows the regression graph of churning losses of crown wheel with 220 mm dia.

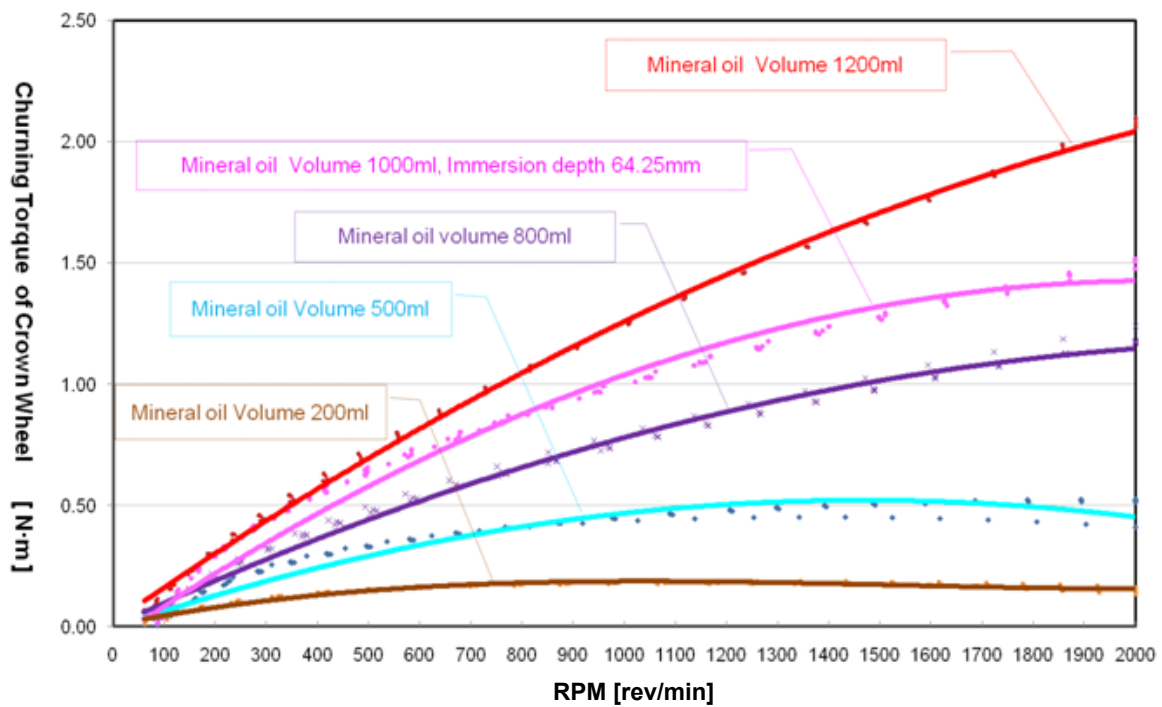


Figure 4.12 Effect of the low viscosity oil volume and oil level on the churning torque of the ring gear of 220 mm dia. rotating forward in axle housing

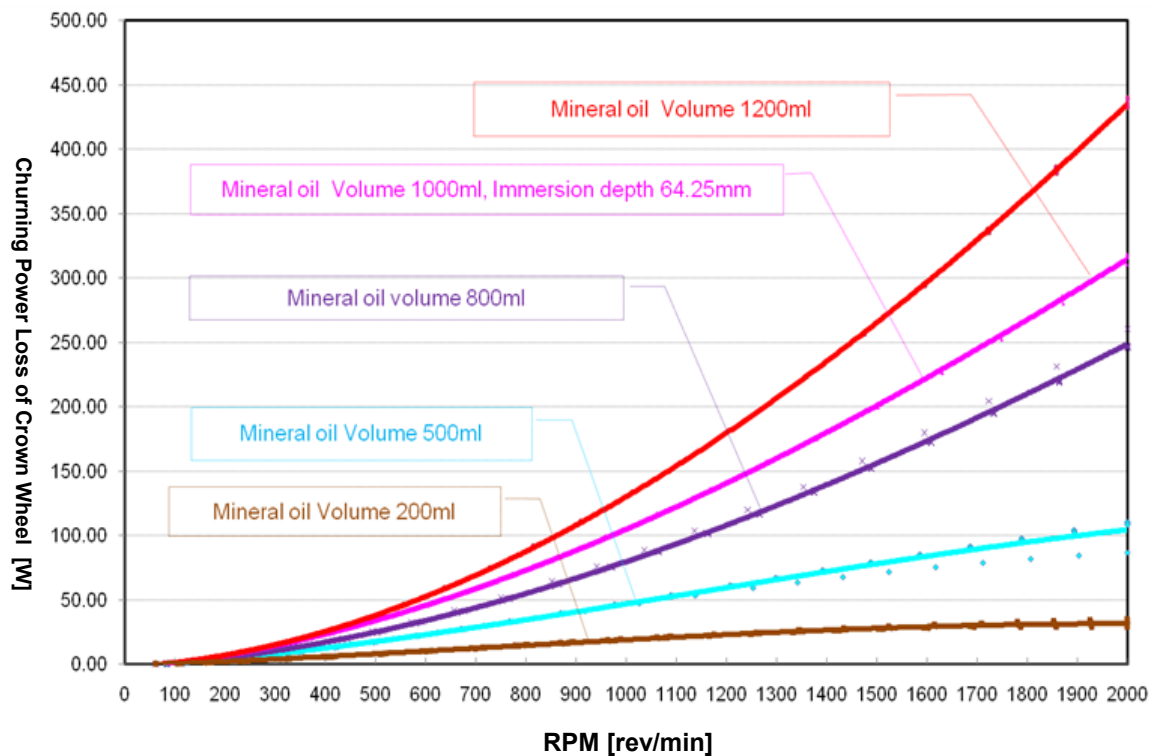


Figure 4.13 Effect of the low viscosity oil volume and oil level on the churning losses of the ring gear of 220 mm dia. rotating forward in the axle housing

The churning torque measured for an oil volume of 1000 cm^3 , which corresponds to an immersion depth of the crown wheel of 64.25 mm is shown in Figure 4.12. The values of churning torques at 500, 1000, 1500 and 2000 cm^3 are approximately 0.6, 1.05, 1.3 and 1.5 Nm and these corresponding churning losses are around 30, 105, 200 and 315 W respectively. It also shows the churning torque and the increasing rate of churning losses at the oil volume of 200 and 500 cm^3 have a shallow maximum at a rotational speed of around 1400 rev/min. The higher the rotational speed is, the bigger is the difference of the churning torque between crown wheels of diameter 210 and 220 mm.

4.4.2.2 Tests with Deep Groove Ball Bearing for Crown wheel

In this section, test results for the 220 mm dia. crown wheel supported in ball bearings are reported. The results are presented in Figure 4.14 and test conditions of which are shown in Table 4.10. Thin section angular contact bearings were employed for the crown wheel with the 220 dia. instead of taper roller bearings and the housing hole for pinion gear was blocked by rubber bung instead of steel plate.

Table 4.10 Test condition for test results of Figure 4.14

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear
Condition	Production axle housing	M220	Thin section deep groove ball bearing	The low viscosity oil / $20 \sim 23.7^\circ\text{C}$	Blocked by a rubber bung

The churning torque measured for an oil volume of 1000 cm^3 , which is shown in Figure 4.14 corresponds to an immersion depth of the crown wheel of 64.25 mm. The churning torques at 500, 1000, 1500 and 2000 rev/min are approximately 0.35, 0.65, 0.82 and 0.95 Nm. These

values are approximately 38 % lower than the corresponding churning torques in Section 4.4.2.1. The churning torque at the oil volume of 200 and 500 cm³ also have a shallow maximum at a rotational speed of around 1200 rev/min.

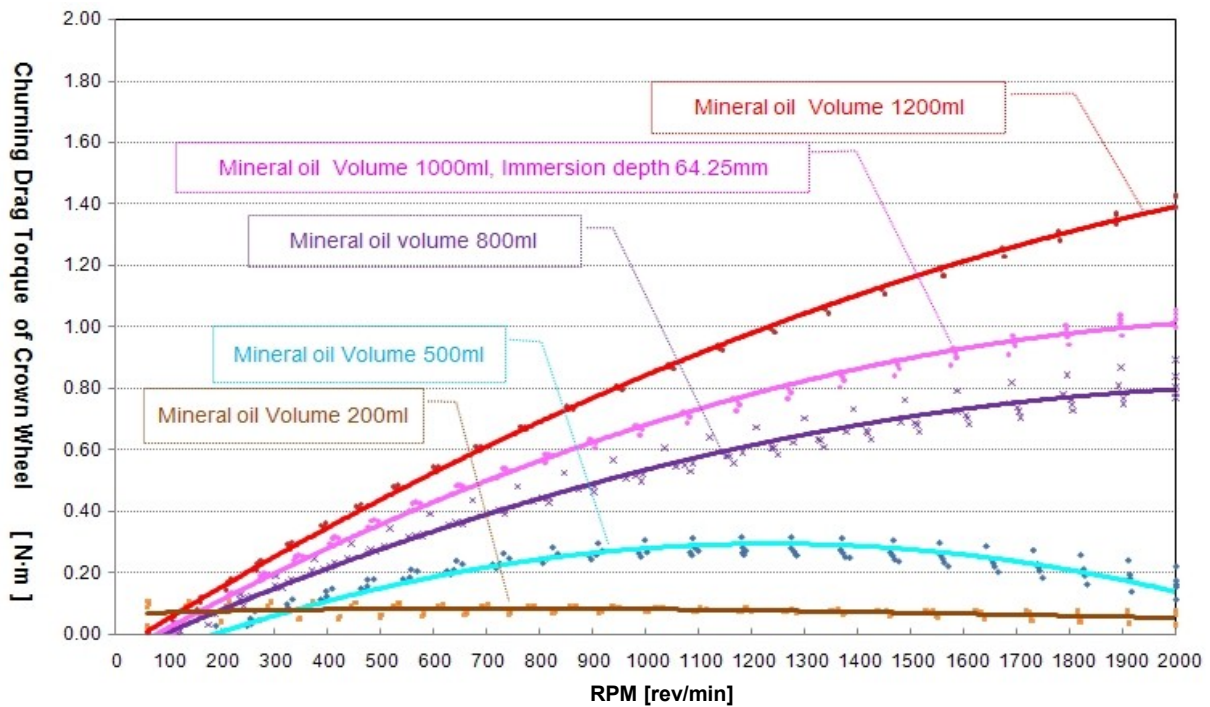


Figure 4.14 Churning torque of the ring gear of 220 mm dia.

4.4.3 Test Result for Lubricants

In the previous section, the experimental results of two types of rotating crown wheels submerged in the low viscosity oil were presented to inspect the geometry effect of crown wheel diameter on churning. This section shows the experimental results of crown wheel of 220 mm dia. dipped into the high viscosity oil in order to compare the effects of lubricant viscosity on churning.

4.4.3.1 Tests with Taper Roller Bearing in the High Viscosity Oil

Table 4.11 shows test conditions for Figure 4.15. This condition is the same as Table 4.9 except for the application of the high viscosity oil instead of the low viscosity oil. The results of measurement of churning torque for forward-rotating crown wheel submerged in the high

Table 4.11 Test condition for test results of Figures 4.15 ~ 4.16

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear
Condition	Production axle housing	M220	Production Taper roller bearing	high viscosity oil / 21.2~23.3°C	Blocked by a circular steel plate with sealant

viscosity lubricant is shown as solid regression line graph in Figure 4.15 with the test result for the low viscosity oil which represented by dotted regression line. Figure 4.16 shows the test results for the reverse rotating crown wheel submerged in the high viscosity oil compared with that in the low viscosity oil. When the churning torques measured for the high viscosity oil volume of 1000 cm^3 are compared between Figure 4.15 and 4.16, the churning torques for the crown wheel rotating forward at 500, 1000, 1500 and 2000 rev/min are approximately 0.79, 1.05, 1.26 and 1.38 Nm, whereas the churning torques for reverse rotating at the same speeds are approximately 0.79, 1.13, 1.30 and 1.25 Nm. Over all, there seems to be little discrepancy between the churning losses for forward rotation and reverse rotation, although churning torque for reverse rotation is smaller at high speeds than for forward rotation.

The churning torque for the crown wheel submerged in the high viscosity lubricant with volume below 800 cm^3 began to decrease above around 1350 rev/min as shown in Figure 4.16. The reverse-rotating churning torque at the oil volume of 1000 cm^3 also has a shallow maximum at a rotational speed of around 1400 rev/min. When comparing the regression solid

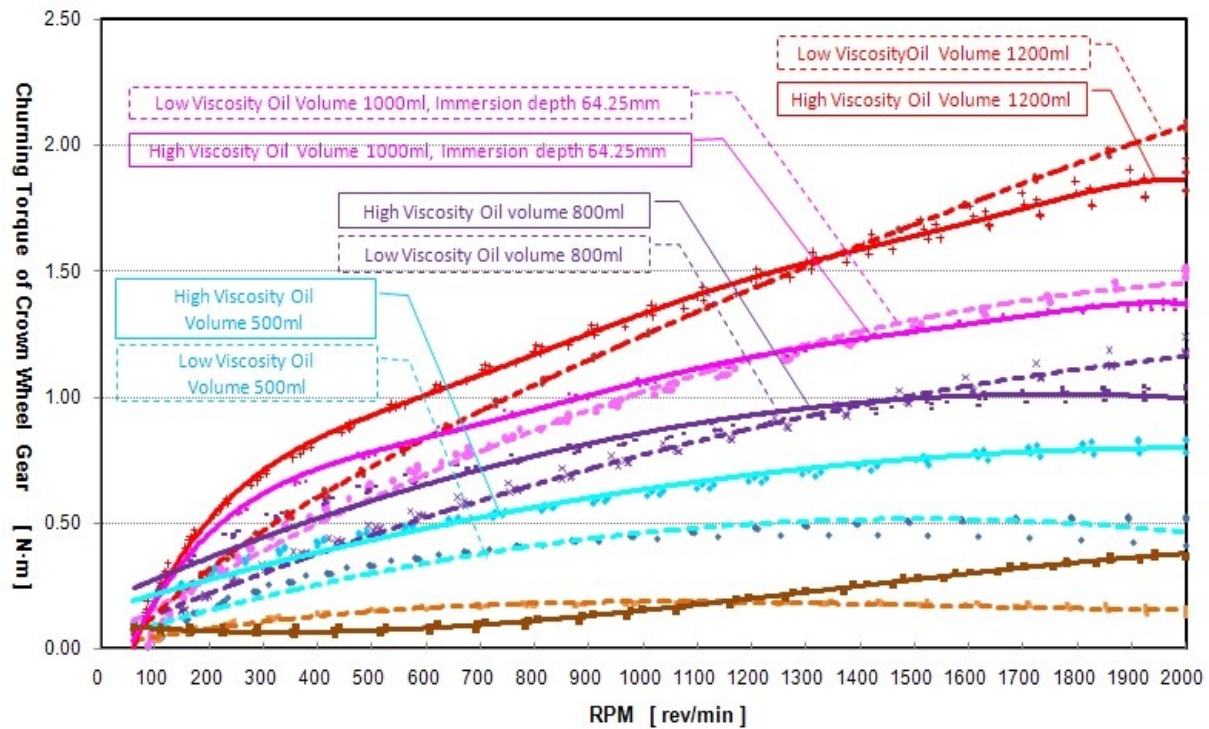


Figure 4.15 Comparison of the churning torque between the low viscosity oil and the high viscosity oil in which the forward-rotating crown wheel of 220 mm dia. is submerged

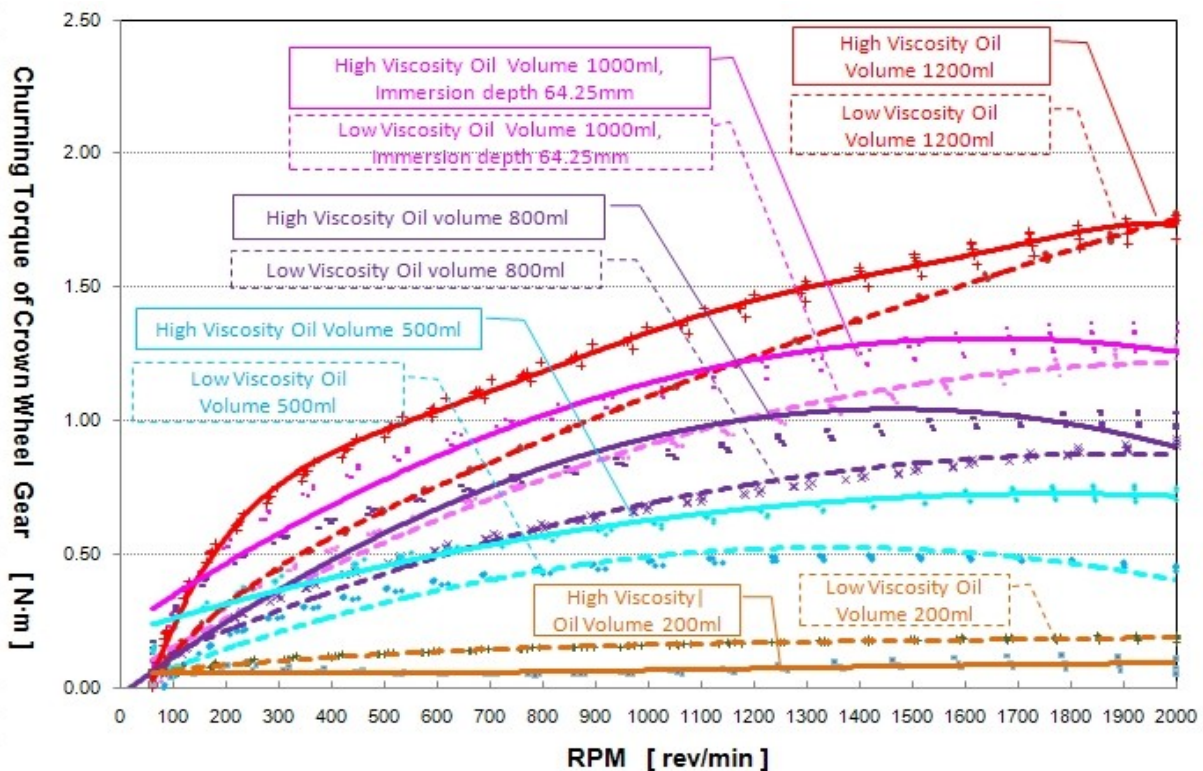


Figure 4.16 Comparison of the churning torque between the low viscosity oil and the high viscosity oil in which the reverse-rotating crown wheel of 220 mm dia. is submerged

line graphs for the high viscosity oil with dotted graphs for the low viscosity oil, the churning torque for the high viscosity oil is higher than the low viscosity oil in the case of reverse rotation. However, the churning torque for the high viscosity oil with volume above 800 cm^3 is lower than the low viscosity oil at the speeds above 1480 rev/min in the case of forward rotation.

4.4.3.2 Tests with Ball bearings in the High Viscosity Oil

The same tests as those in Section 4.4.3.1 were performed with the conditions of the different bearing type and the different blocking way of housing hole for pinion gear. The results are presented in Figure 4.17 of which test conditions are shown in Table 4.12. Thin section deep groove ball bearings were employed on the crown wheel of diameter 220 mm instead of taper roller bearings and the housing hole for pinion gear was blocked by a rubber bung instead of a steel plate. Additionally, it was carried out an early stage so that gear teeth pick-up sensor was employed to acquire the data of rotational speed in these tests instead of optical sensor. This caused an overall reproducibility of about $\pm 0.15\text{ Nm}$.

Table 4.12 Test condition for test results of Figure 4.17

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Rotational direction
Condition	Production axle housing	M220	Thin section deep groove ball bearing	High viscosity oil / about $20\sim 23^\circ\text{C}$	Blocked by a rubber bung	Reverse

Another different point from the previous tests is that the interval between the tests was very short and the repeat of tests was performed very fast. This condition would even reduce the churning torque for the high viscosity oil in the tests; which will be more discussed in later chapters. In addition, the blocking way of a rubber bung could be one of the reasons to make

the churning torque for the high viscosity less than for the low viscosity oil.

The regression solid line graph represents the measured values for the high viscosity oil and the dotted curve for the low viscosity oil was plotted together for comparison. The salient features in the tests were that the churning losses for the low viscosity oil were much higher than for the high viscosity oil, which is different from the results repeated in Section 4.4.3.1. In Figure 4.17, the rising rate of churning torque for the high viscosity oil decreased gradually and the rate at higher speeds became small. Namely, the decline rate of churning drag torque for the high viscosity oil at lower RPM is bigger than mineral oil.

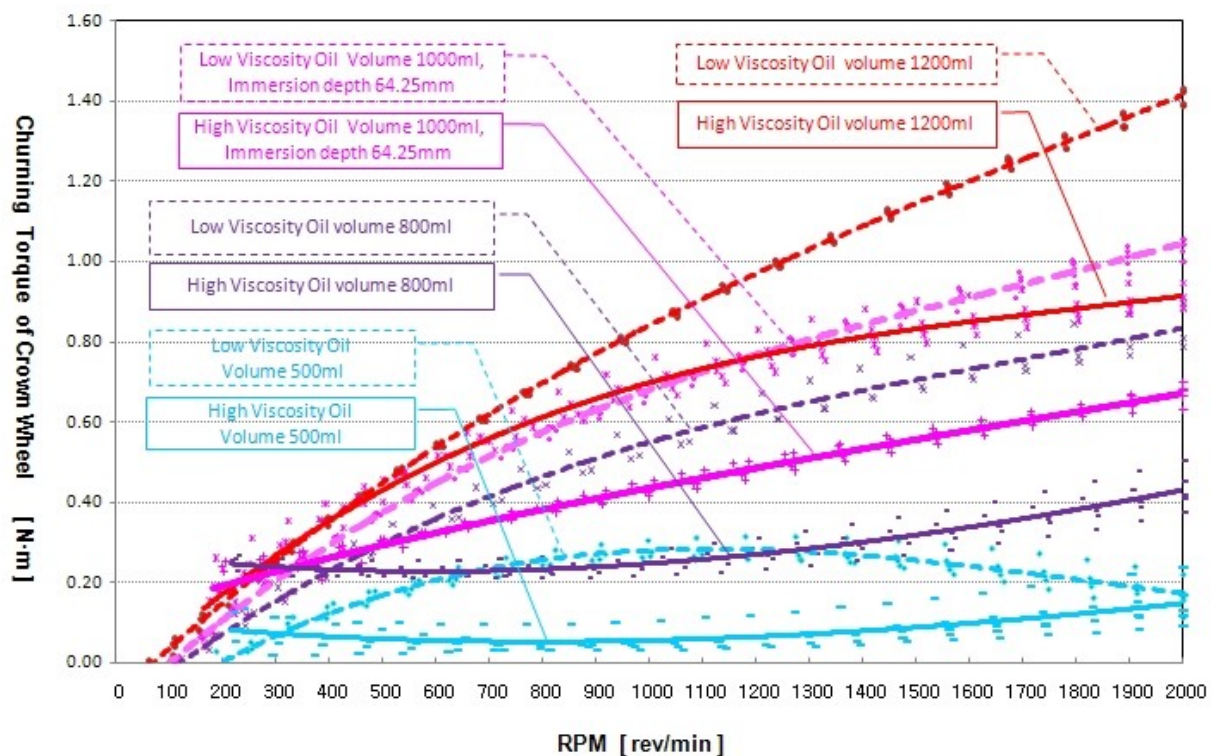


Figure 4.17 Comparison of the churning torque between the low viscosity oil and the high viscosity oil in which the reverse-rotating crown wheel of 220 mm dia. is submerged

4.4.4 Crown Wheel Gear Meshed with Pinion Gear

In the previous sections, churning losses for the crown wheel rotating without mating hypoid pinion gear were investigated. This section depicts the churning losses for meshed hypoid axle in the production axle housing.

4.4.4.1 Tests on Different Diameter of Crown Wheel

The test condition for these experiments is shown in Table 4.13. The low viscosity oil was employed to simulate the viscosity of the high viscosity oil at the temperatures of general driving conditions and thin section angular contact ball bearings were applied to the crown wheel assembly and corresponding hypoid drive pinion gear.

Table 4.13 Test condition for test results of Figure 4.18

Item	Housing	Size of ring gear	Bearing of crown wheel	Bearing of pinion gear	Lubricant / Temperature
Condition	Production axle housing	M210 M220	Thin section deep groove ball bearing	Thin section deep groove ball bearing	The low viscosity oil / 22.8~27.1℃

In the case of the oil volume of 1200 cm^3 for both hypoid gear pairs of 210 and 220 mm dia., churning drag torque is almost proportional to rotational speed and the slopes of both the regression graphs are almost constant during the rundown period. The churning torque for a gear pair of 210 mm dia. shows smaller increase at higher speeds when compared with that of diameter 220 mm. The churning torques of hypoid gear pairs of 210 mm dia. for an oil volume of 1000 cm^3 at 500, 1000, 1500 and 2000 rev/min were approximately 0.7, 1.5, 2.0 and 2.5 Nm, which are lower than those of diameter 220 mm. Thus, this shows that the churning torque of a hypoid gear pair with a ring gear of 210 mm dia. are also lower than that of a hypoid gear pair with a ring gear of 220 mm dia. like the test results for the different sizes of a

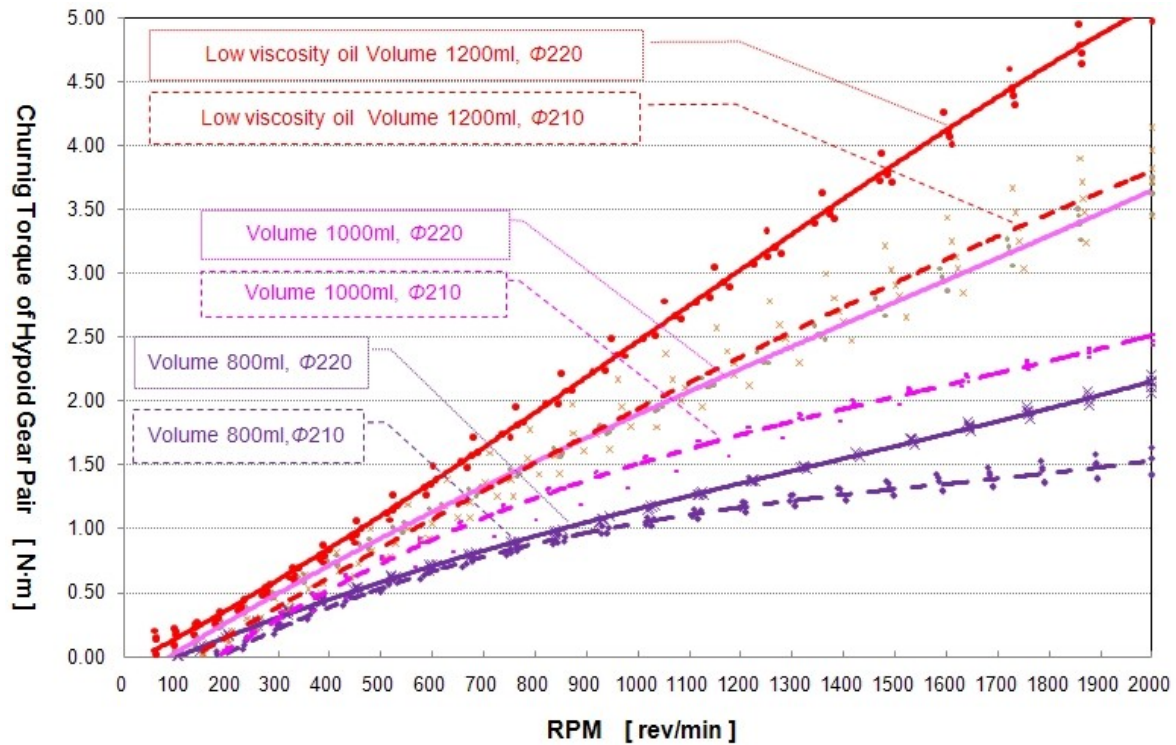


Figure 4.18 Comparison of the churning torque between dia. 220 and 210mm of a hypoid gear pair

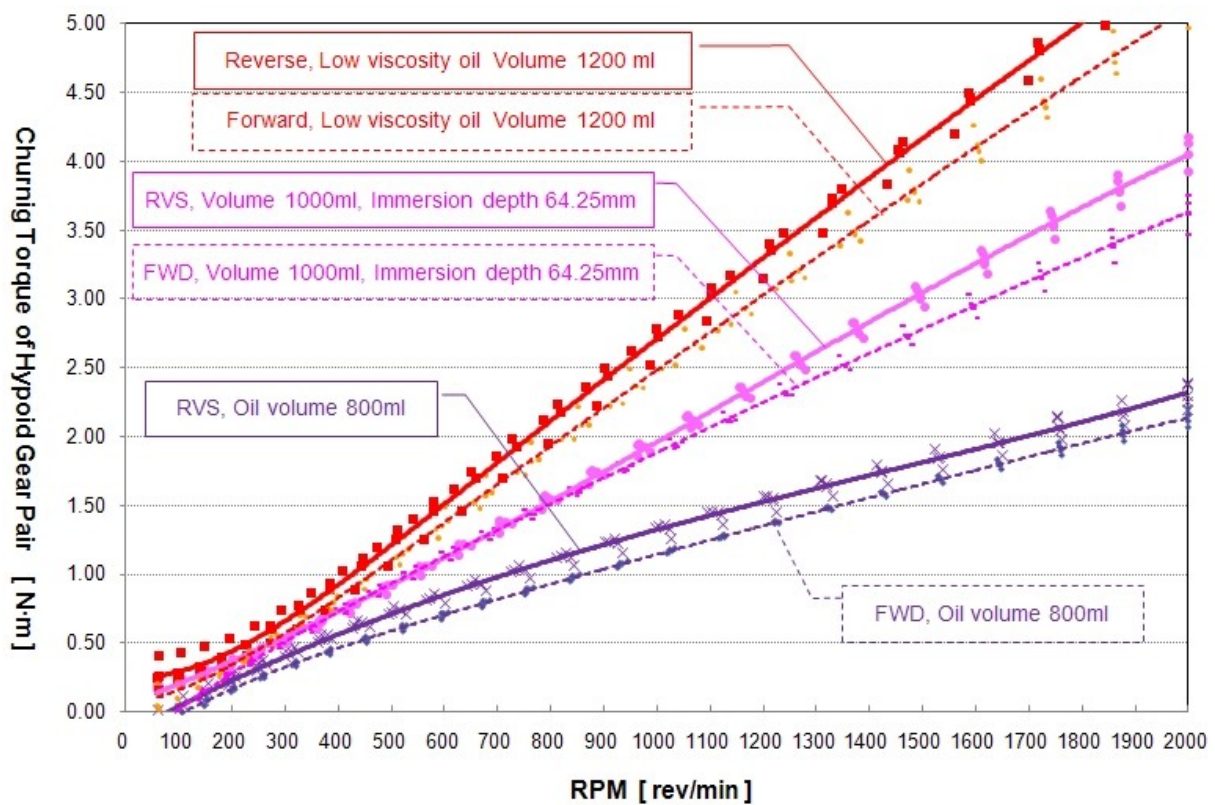


Figure 4.19 Comparison of churning torque between forward and reverse rotating of a hypoid gear pair

crown wheel alone. In addition, when compared with the test results for the crown wheel alone, the churning losses of a meshed hypoid gear pair were as much as 2.5 ~ 3.5 times larger than those of crown wheel alone. Therefore, the presence of a hypoid pinion gear could make a big contribution to the churning losses even if the pitch diameter of pinion gear is small.

4.4.4.2 Tests on Rotational Direction of Meshed Hypoid Gears

For meshed hypoid gears, the effect of rotational direction on churning losses was investigated in this section. Table 4.14 presents the test condition for Figure 4.19.

Table 4.14 Test condition for test results of Figure 4.19

Item	Housing	Size of ring gear	Bearing of crown wheel	Bearing of pinion gear	Lubricant / Temperature
Condition	Production axle housing	M220	Thin section deep groove ball bearing	Thin section deep groove ball bearing	The low viscosity oil / 20~23.6℃

Results from Figure 4.19 show the churning torques for a mating hypoid gear pair rotating forward and in reverse, submerged in the low viscosity oil. The oil immersion depth acted as an important factor dominating churning losses regardless of rotating direction and the churning torque of a mating hypoid gear pair rotating in reverse was higher than forward rotating by about 10% as can be seen from Figure 4.19. It could result from the swell effect caused by the sense of rotating direction and the rotating hypoid gear pairs drawing the oil into the meshed portion; which could contribute the churning torque in addition to the contribution of the two individual gears rotating forward.

4.5 Attitude Tests

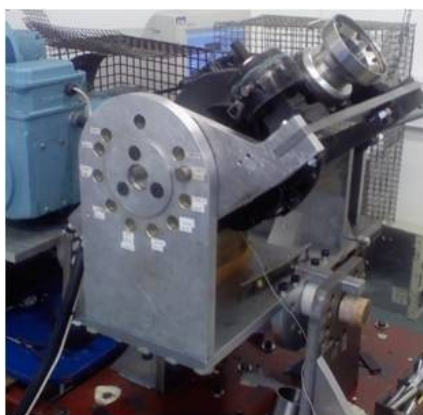
In general, vehicles do not run only on horizontal level roads, but also on inclined roads such as a hill or curved highways. These inclined road conditions were employed to the position of axle housing, and it is investigated how these factors influence on the churning losses in this section.

4.5.1 Pitch Attitude

Test conditions of the experiments on pitch position of the meshed hypoid gears are shown in Table 4.15, and Figure 4.20 represents how the pitch angle of the axle housing was realized in the test rig built by the author. A positive pitch angle means that the axle housing rotates

Table 4.15 Test condition for test results of Figure 4.21 and Figure 4.22

Item	Housing	Size of ring gear	Bearing of crown wheel	Bearing of pinion gear	Lubricant (Temperature)	Oil volume
Condition	Production axle housing	M220	Production Taper roller bearing	Thin section deep groove ball bearing	the low viscosity oil (28.0~31°C) the high viscosity oil (28.6~30°C)	1000 cm ³



(a) Pitch angle +30°



(b) Pitch angle -45°

Figure 4.20 Pictures of axle housing pitched at a certain angle on the rig

(a) [+] notation of pitch angle direction (b) [-] notation of pitch angle direction

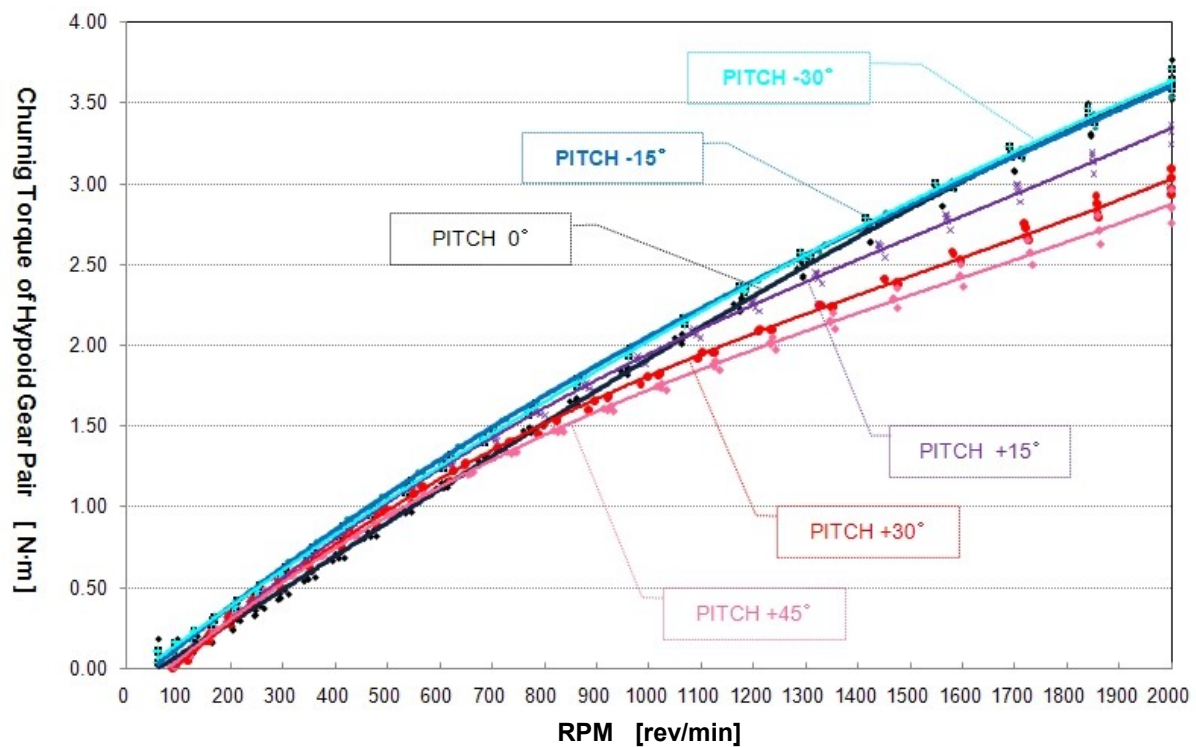


Figure 4.21 Effect of pitch angles of a meshed axle of the crown wheel with 220mm dia. on churning torque in the low viscosity oil

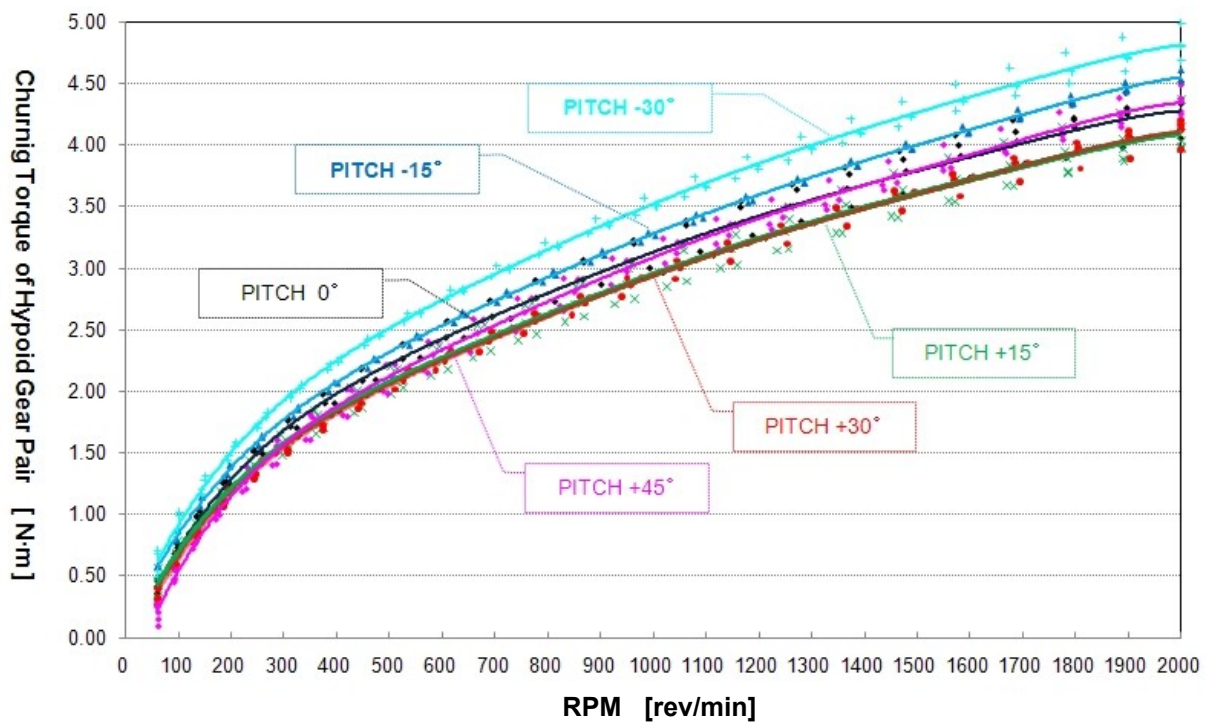


Figure 4.22 Effect of pitch angles of a meshed axle of the crown wheel with 220mm dia. on churning torque in the high viscosity oil

counter-clockwise from the view of Figure 4.20 and the companion of hypoid pinion is raised upwards. The volume of low viscosity oil in the axle housing was 1000 cm^3 , of which the corresponding immersion depth was 64.25 mm and it was kept constant over the pitch and roll experiments. Two types of lubricants were employed to investigate and compare the effect of lubricant viscosity on churning losses for meshed hypoid gears regarding pitch positions.

The results of the housing pitch angle tests for the churning torque in the low viscosity oil are shown in Figure 4.21, while the test results for the high viscosity oil are presented in Figure 4.22. The immersion depth of the pinion gets gradually larger as the pitch angle becomes more negative. There was a slight increase of churning torque for the high viscosity oil, but the values for the low viscosity oil were almost constant for all negative pitch angles. For the positive pitch angles that the pinion gear is not submerged in the oil, the churning torque for the low viscosity oil decreased at high speeds more than the high viscosity oil. From the experiments of the varied pitch angles, the immersion depth of the pinion into the low viscosity oil caused by pitch attitude could not be influential even if pinion gear was fully immersed, but the churning torque could decrease at the above 1000 rev/min if the hypoid pinion gear is located above the surface of mineral oil. On the other hand, the higher immersion depth of the pinion into the high viscosity oil gives a gradual increase in the churning torque although the changed amount is as small as less than 5% at low speeds. Overall, the pitch positions have little influence on churning losses at low rotational speeds regardless of viscosity.

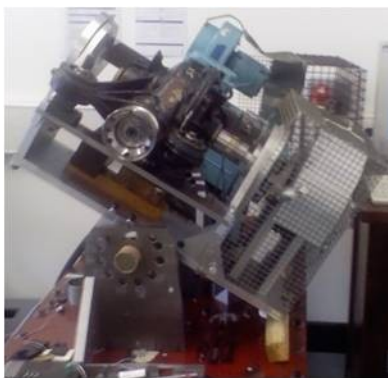
4.5.2 Roll Attitude

In this section, the results of the experiments for the effect of rolling positions of the axle housing on churning losses are presented. Test conditions of the experiments on roll attitude

are shown in Table 4.16. Taper roller bearings were employed as the bearings of the crown wheel for supporting the thrust force from the sloped axis of the driven crown wheel caused by the rolling angle. These roll attitude tests were performed using the two lubricants and the test conditions are as detailed in Table 4.16. Figure 4.23 represents the notation of rolling positions. Rolling position CW30° means that the axle housing declines to 30° clockwise from the view of the pinion side in Figure 4.23 while CCW15° means a decline of 15° counterclockwise. All the experiments in this section were carried out with an oil volume of only 1000 cm³, which is the normal oil amount in the axle.

Table 4.16 Test condition for rolling position test results of Figure 4.24 and Figure 4.25

Item	Housing	Size of ring gear	Bearing of crown wheel	Bearing of pinion gear	Lubricant (Temperature)	Oil volume
Condition	Production axle housing	M220	Production needle taper roller bearing	Thin section deep groove ball bearing	The low viscosity oil (26.2~30.3 °C) The high viscosity oil (28.8~32.4 °C)	1000 cm ³



(a) Roll angle: CW 30°



(b) Roll angle: CCW 30°

Figure 4.23 Rolling notation of axle housing at a certain angle on the rig

(a) [CW] notation of rolling direction of clockwise (b) [CCW] notation of rolling direction of counter-clockwise

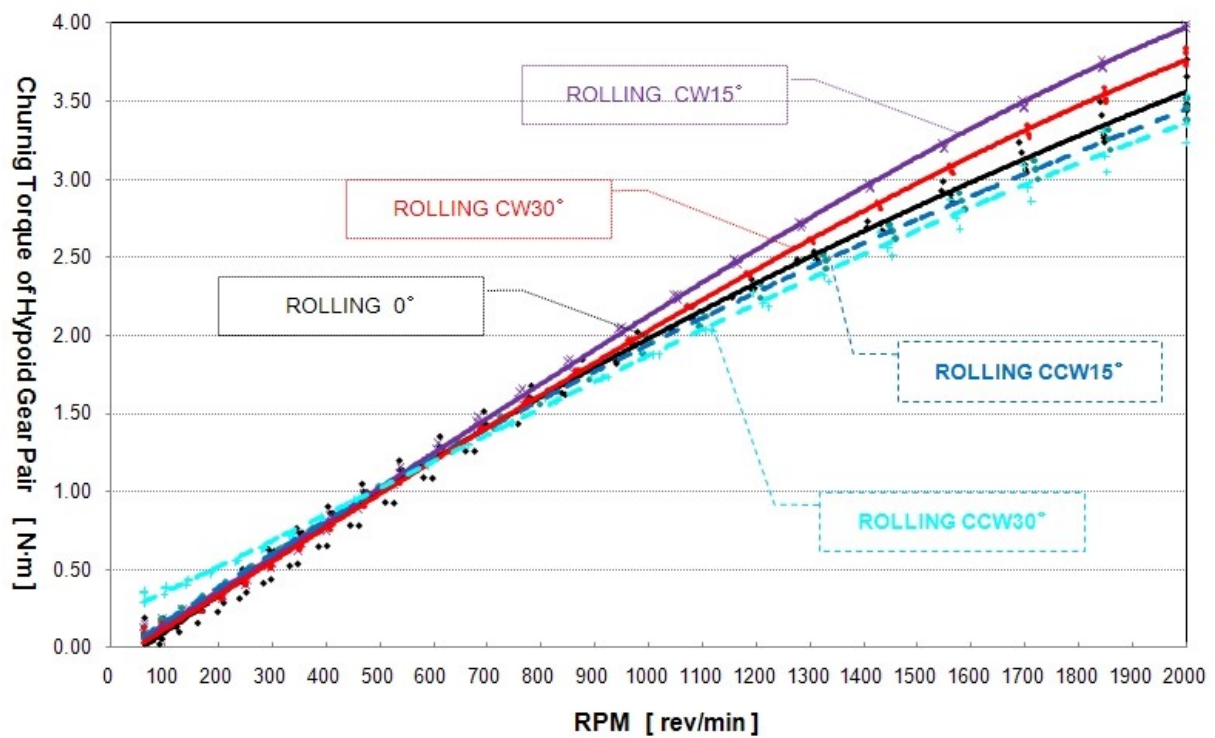


Figure 4.24 Effect of roll angles of the axle of the crown wheel with 220 mm dia. on the churning torque in the low viscosity oil

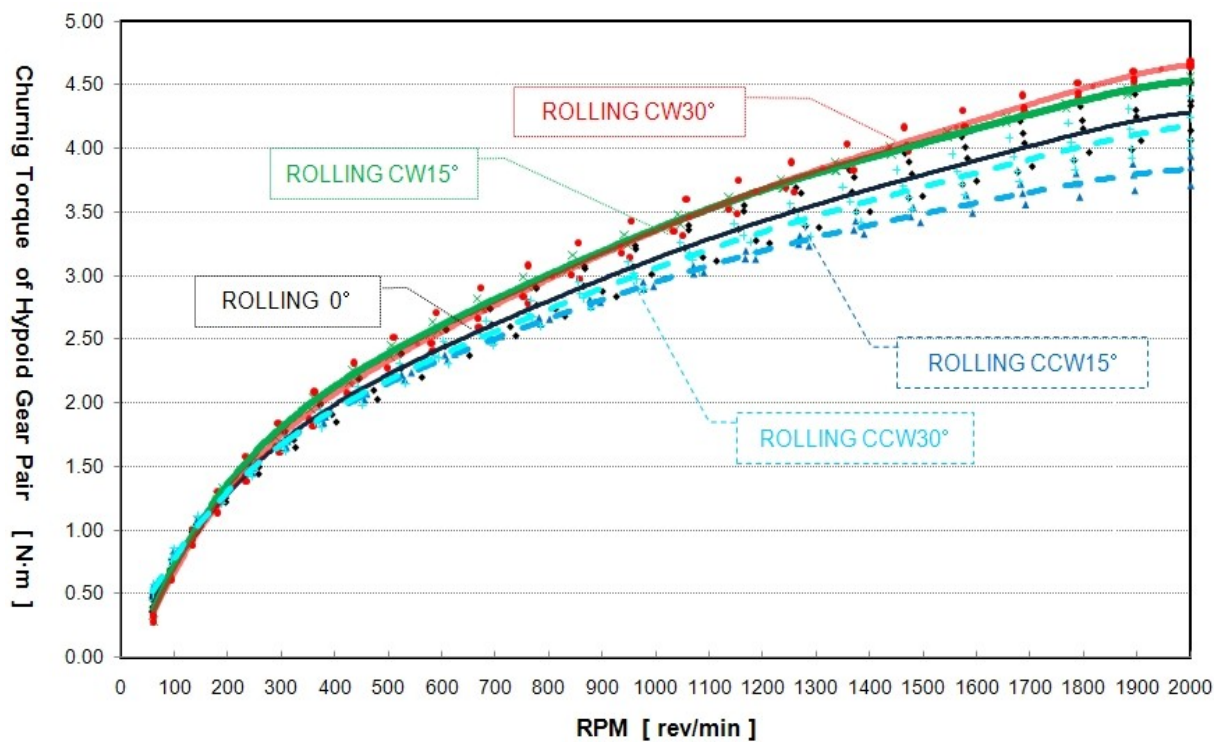


Figure 4.25 Effect of roll angles of the axle of the crown wheel with 220 mm dia. on the churning torque in the high viscosity oil

Results are shown in Figure 4.24 and Figure 4.25. From these figures, the values of the churning torque for all rolling positions are almost the same up to 900 rev/min when these regression curves are compared with the values for the normal horizontal position. For the axle housing rolled in the clockwise direction, the churning torque was slightly greater than for the horizontal position from the above 900 rev/min regardless of the oil viscosity.

On the contrary, for the rolling positions in anticlockwise, the churning torque for the low viscosity oil could be almost the same as that of horizontal position. The rolling positions in the anticlockwise for the high viscosity oil seem to decrease the churning torque compared to the horizontal position. Overall, the rolling positions in the clockwise have more influences on the churning torque than those in the anticlockwise. It could be said that the influence of the immersion depth of the crown wheel is greater than that of the pinion.

The churning torques of hypoid gear pairs of the crown wheel with 220mm dia. for the low viscosity oil volume of 1000 cm³ at 500, 1000, 1500 and 2000 rev/min were measured as approximately 1.0, 2.0, 2.8 and 3.6 Nm. These values are in a good agreement with the measured values of churning torques of the meshed hypoid gear pairs rotating under the same conditions except for the thin section deep groove ball bearings employed instead of taper roller bearings.

A salient feature is that the churning torque for a hypoid gear pair immersed in the high viscosity oil increased with rotational speed as a logarithmic growth, whereas the rise of churning torque in the low viscosity oil was close to a straight line. In addition, the overall churning torque in the high viscosity oil was bigger than in the low viscosity oil, but the difference in churning torque and losses between the high viscosity oil and the low viscosity oil became smaller at higher speeds, becoming close to the same value as each other.

4.6 Error Analysis of Experimental Data

Generally, there are two types of errors associated with an experimental result: the errors of precision and accuracy. Precision is related the random error distribution associated with a particular experiment (Pugh & Winslow, 1966). The error of precision refers to reliability, which can be defined as the consistency of a set of measurements or of a measuring instrument. It also depends on the repeatability of the data and they exist within the reasonable tolerance.

The Author's experiment was repeated five times per a test and secured the consistency and repeatability within the tolerance. First of all, the data of decelerated rotational speeds were obtained once per 0.0625 ± 0.0005 sec. and then the regression graph of decay curve was calculated using the 3rd polynomial linear regression in Excel. The values of R-squared (R^2 : the square of the sample correlation coefficient) for the regression graphs of Figure 4.3 are from 0.9997 to 1.0 as shown in Figure 4.26 and its error is very small to be neglected.

The moment of inertia can be regarded as constant and so the deviation error for the five time trials of rotational speed measurement could be the error for the drag torque. The deviation error for Figure 4.26 is relatively about ± 0.75 %. Then, the churning torque can be calculated by subtracting the no-oil torque from the total drag torque. Thus, the error for the churning torque for Fig. 4.3 is finally within ± 1.5 %. It corresponds to ± 3 Nm ($= \pm 1.5 \% \times 2.0$ Nm) for the error of the churning torque at 2000 rev/min and oil immersion depth 64.35 mm in Fig. 4.5. The values of R-squared for the regression graphs of Figure 4.8 are from 0.9999 to 1.0. But, its deviation is large and the relative error is about ± 2.5 % because this test was performed first in the test rig and the test condition such as calibration and vibration of the rig were not perfect. Thus, the error for the churning torque for Fig. 4.10 is finally within ± 5.0 %

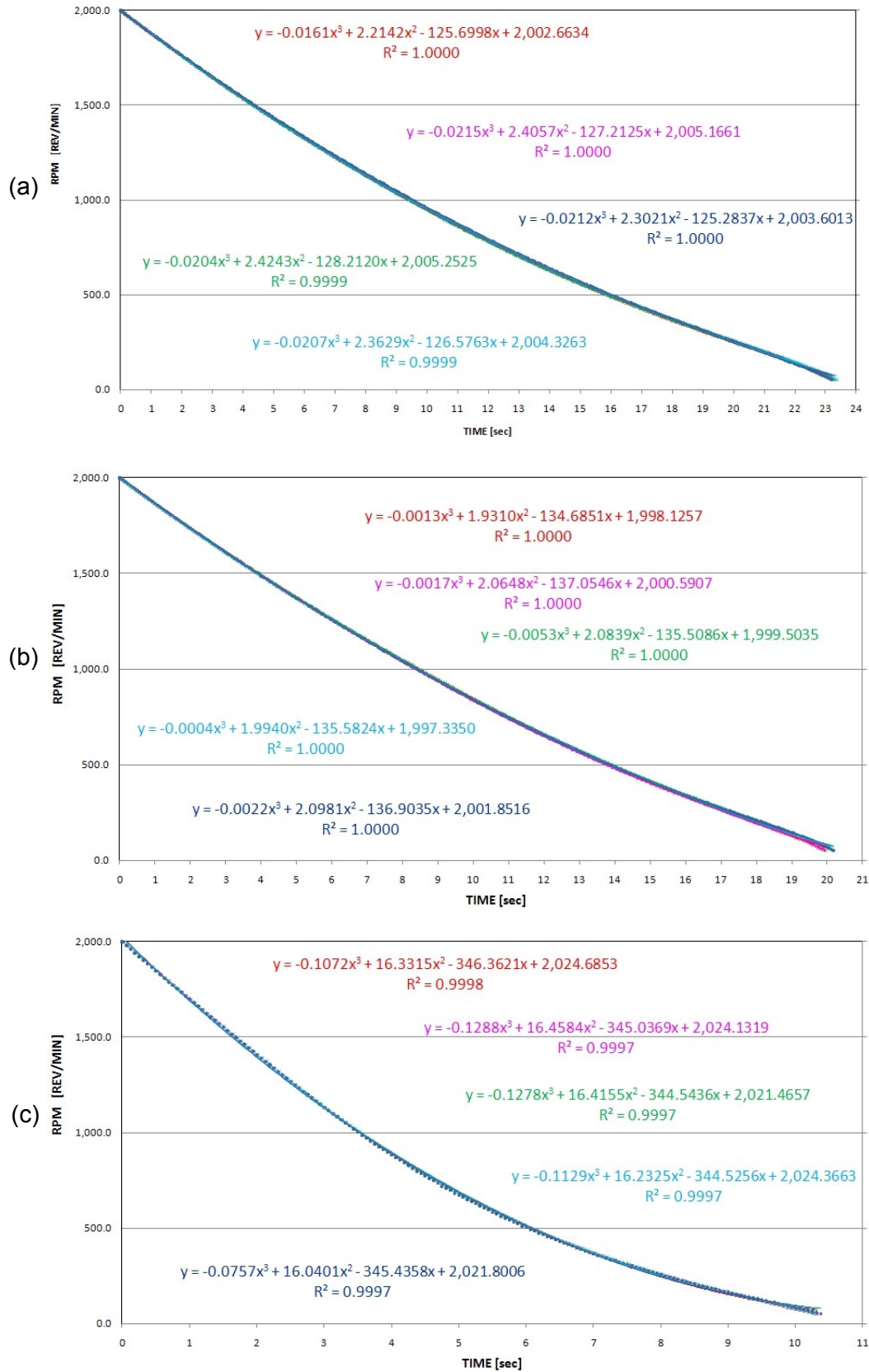


Figure 4.26 R^2 for the 3rd polynomial regression graphs of the decay data for Fig 4.3
 (a) oil volume 0 cm³ (b) oil volume 200 cm³ (c) oil volume 1000 cm³

On the other hand, accuracy is related to the existence of systematic errors and is regarding to the experimental method itself. The errors of accuracy depend on validity, which is whether a study is scientifically able to answer the question that is intended to answer. Although the error of precision is very small, the error of accuracy can be big. This error could depend on the setting of the housing on the rig, the assembled clearance of hypoid gears and the initial speed of inertia run-down in the author's experiment.

4.7 Concluding Remarks

The experimental study of churning torque and churning losses of hypoid axle gears in mass-produced axle housing was presented in this Chapter.

(1) The churning torque of hypoid axle increased with the rotational speed was found smaller for small diameter crown wheels and for lower viscosity oils. The churning torque for the crown wheel with 220 mm dia. and oil immersion depth 64.25 mm was approximately 22 % higher than for 210 mm dia. at 2000 rev/min, and was 17 % higher at 1000 rev/min.

(2) The influence of oil viscosity was not as large at high speeds as much as at low speeds.

(3) The attitude of pitch and rolling influenced little the churning losses at low speeds and its influence at high speeds showed the dependence on the immersion depth of the meshed portion. The churning torque for the meshed hypoid gear with 220 mm dia. had a 6% decrease at the attitude of pitch up in the high viscosity oil sump at high speeds. For the rolling position, the churning torque varied about 10% depending on the immersion depth of the meshed portion.

(4) The presence of the pinion had a large influence on the churning losses. The churning torque for the meshed hypoid gear was 2.5 ~ 3.5 times greater than for the crown wheel alone.

In Chapter 5, additional experimental studies in an extended housing are presented to investigate the effect of housing clearance on churning, and the lubricant flow inside the housing.

CHAPTER 5

EXPERIMENTAL RESULT USING EXTENSION HOUSING

5.1 Introduction

Further experimental studies of the churning losses phenomenon in hypoid axles are now reported. The experimental study, reported in Chapter 4, showed how much the variables such as lubricant, immersion depth, rotating speed and the gear geometry could influence the churning losses in a hypoid axle housing. In the work described in this chapter, the same rig with a new built extension housing was used to study the effects of changes in the boundaries surrounding the lubricant and its viscosity on the churning losses behaviour. In addition, one side of the Aluminium extension housing was replaced by transparent material and the oil flow inside the housing was observed. Although the experiments for the inspection of oil follow inside the extension housing was performed at the last minute, this is described firstly in Chap. 5 for the better understanding of the readers.

5.2 Oil Flow inside the Housing

The additionally designed and manufactured transparent lid and axial baffle replacing the aluminium lid of the extension housing were employed to inspect the lubricant flow inside the housing. Figures 5.1 and 5.2 show the oil flows of the low and high viscosity oils inside the extension housing with axial baffles positioned at '2 by 2' and a radial baffle of a diameter of 234.8 mm. These figures were taken while the rotational speed was raised slowly.

As shown in Figure 5.1, the lubricant was moved away along the radial wall and the oil immersion depth was comparatively lowered. At lower speeds, the lubricant flowed like a laminar flow along the radial wall of housing but it seemed to become turbulent flow at around 600rev/min. Minute bubbles occurred inside the lubricant, which soon became too murky to see through. Most of all, the salient feature observed in the oil flow of the high viscosity oil was that it took a long time to return to the initial oil level after the end of the run-down test due to the slow flowing of dispersed lubricant on the wall after stopping of the rotating crown wheel. The schematic diagrams for oil level changes of the high viscosity oil are presented in order in Figure 5.3. The oil level was decreased when the crown wheel submerged in the oil rotated and scattered it to the inside of the housing. After the rotation of the crown wheel ceased, the lubricant remained on the wall and crown wheel flowed down very slowly for a long time due to the high viscosity. Unlike the high viscosity oil, many bubbles were found to occur in the low viscosity oil, which does not include an anti-foam additive, as soon as the crown wheel started rotating. The low viscosity oil was also observed to flow along the radial wall but changed soon to turbulent flow. In addition, the low viscosity oil level returned to the initial oil level faster than the high viscosity oil as shown in Figure 5.4.

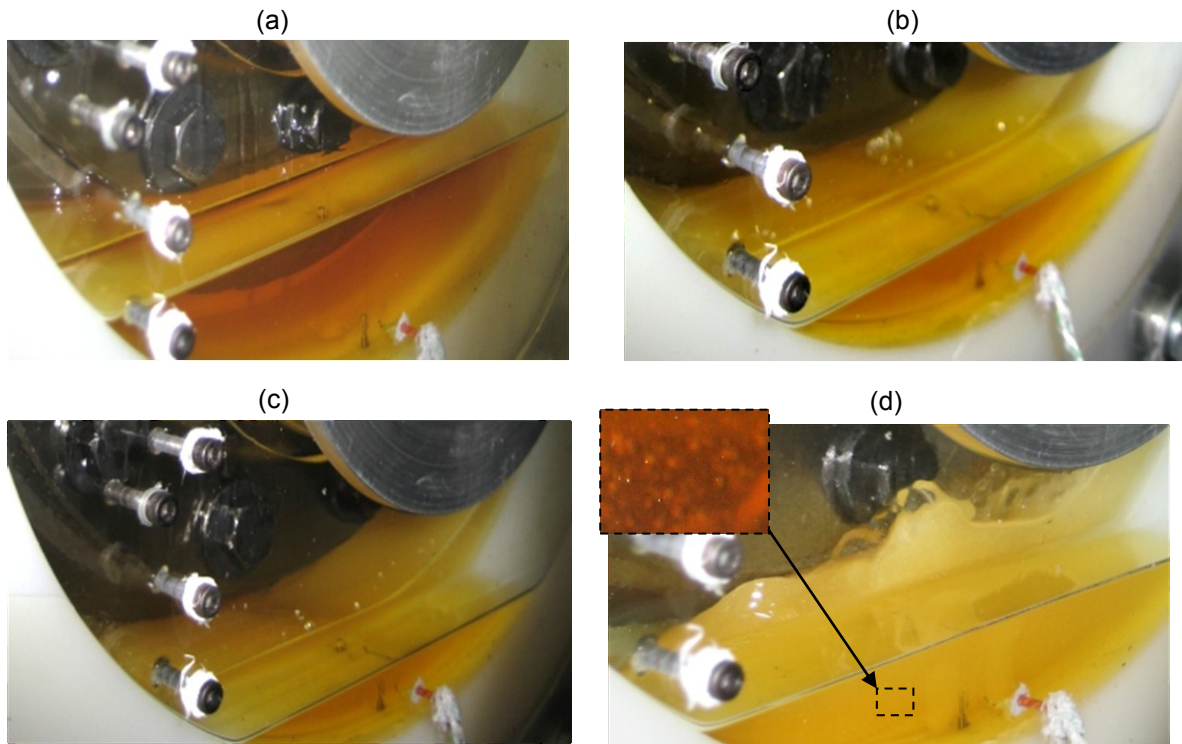


Figure 5.1 Oil flow of the high viscosity oil (synthetic lubricant 75W90) caused by rotating crown wheel of $\phi 220$ with an immersion depth 31 mm (a) before crown wheel rotates (b) rotating at about 100 rev/min (c) rotating at about 300 rev/min (d) rotating at above 600 rev/min; Minute bubbles in top left rectangular

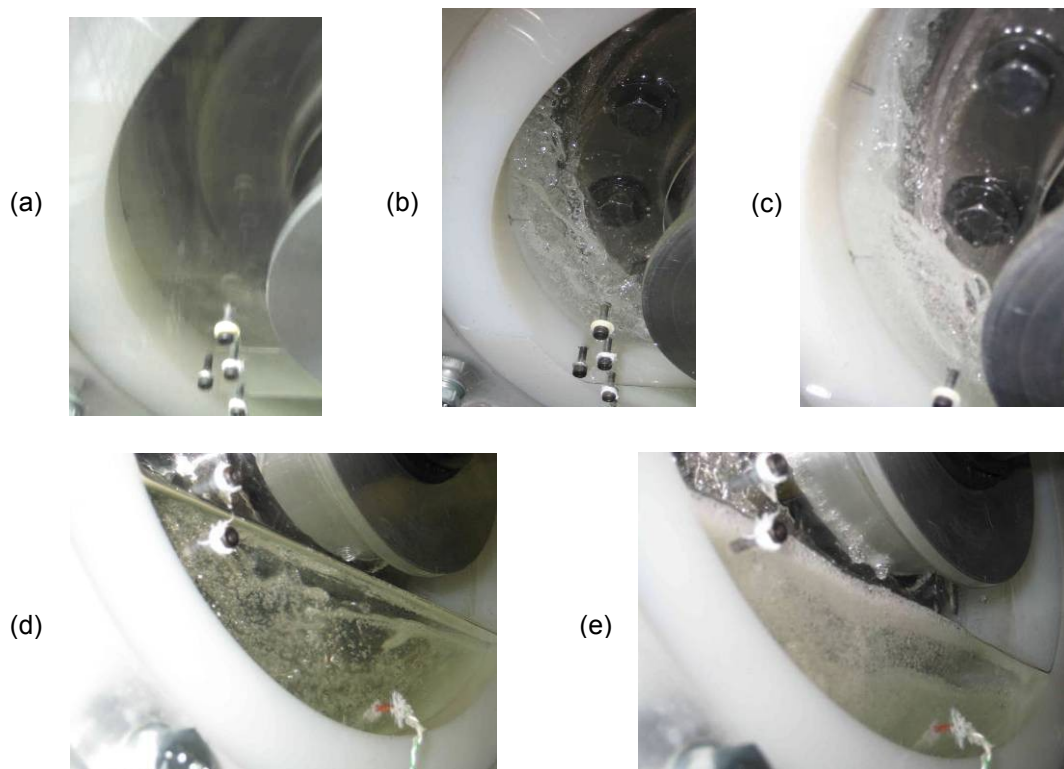


Figure 5.2 The low viscosity oil flow and bubbles caused by the rotating crown wheel of 220 mm dia. (a) laminar flow (b) turbulent flow (c) bubbles (d) foam

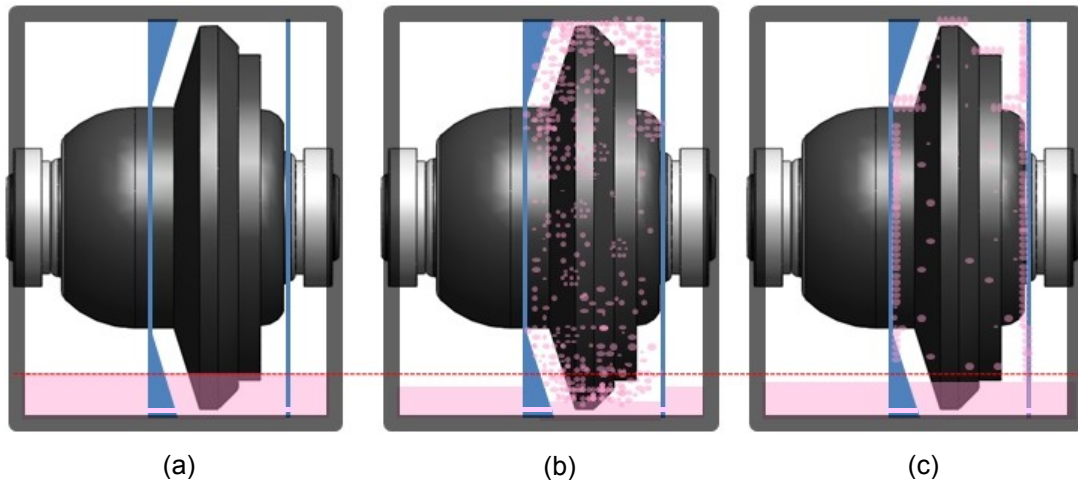


Figure 5.3 Oil level change for the high viscosity oil (a) before running (b) while running (c) 1 hour after end of run-down test

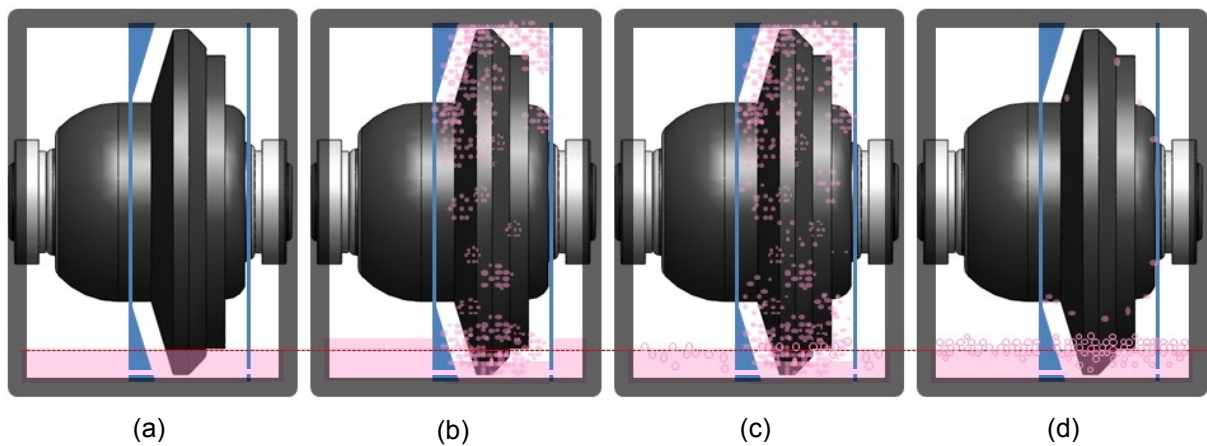


Figure 5.4 Oil level change for the low viscosity oil during the run-down test (a) before test (b) running at low speeds (c) running at higher speeds (d) after end of the run-down test

The noteworthy point is that the low viscosity oil levels were raised after the run-down test due to the foam mainly on the surface, but the measured churning losses before and after the raised oil level due to foam were always same. Thus, bubbles and foams occurred in mineral oil did not affect the churning losses in the experiments in this thesis.

5.3 Application of Baffles to Extension Housing

The extension housing as well as radial and axial baffles were used in the experimental studies of this chapter in order to investigate the effect of radial and axial clearances between the housing wall and rotating hypoid gear pairs on churning losses. As explained in Section 3.1, the extension housing and the baffles were designed by the author and precisely manufactured by the workshop of the motor company according to the author's drawings, which are presented in Appendix 6.

The details of the axial and radial baffles in the extension housing were shown in Section 3.1.2.1. The radial clearances according to radial baffles were previously described in Table 3.1 and three different radial baffles were applied only to the ring gear of diameter 220 mm. The positions of axial baffles were explained and expressed in Figure 3.8 and Table 3.2. The notation used in the graphs of this chapter to describe the axial baffle instruction is shown in Table 5.1. Axial clearance e_{al} is the distance between the inclined gear surface of the crown wheel and the housing wall and e_{ar} is the distance between the opposite side of the ring gear and the housing wall.

Table 5.1 Axial clearances for positions of axial baffles employed in the author's tests
(Refer to Figure 3.1 for the position of axial baffles in the extension housing)

Notation	① by ②	② by ②	③ by ③	④ by ③	⑤ by ④	No baffle
e_{al}	2.5 mm	7.5 mm	12.5 mm	22.5 mm	42.5 mm	87.5 mm
e_{ar}	17 mm	17 mm	22 mm	22 mm	32 mm	42 mm
e_{al} / e_{ar}	0.15	0.44	0.57	1.02	1.33	2.08

According to the author's measurements, the radial clearance of the axle housing was approximately 4.4 mm and the axial clearances of axle housing were approximately 40 and 22 mm for e_{al} and e_{ar} respectively.

5.4 Effects of Housing Clearance on Churning Losses

In this Chapter, the results for the effects of housing clearances on churning losses are shown. The effects of axial and radial baffles were separately tested to determine the effects of them clearly.

5.4.1 Effect of Radial Baffles

As previously described in Table 3.1, three different radial baffles were employed in the extension housing at different oil depths in order to inspect the effect of the radial clearances on churning losses. No axial baffle was installed inside the extension housing so that the axial clearances were the distances from the ring gear to the wall of the extension housing, as which were 87.5 and 42 mm as shown in Table 5.2. Those test results are illustrated respectively in Figures 5.5 and 5.7 to 5.9, for which the test conditions are shown in Table 5.1. Those results in Figures were obtained under the mineral oil immersion depth of 31, 47, 71 and 91 mm respectively for inner diameters of radial baffles of diameters 226, 234.8, 243.6 mm and the radial wall of housing.

Table 5.2 Test condition for the test results of Figures 5.5 and 5.7 ~ 5.9

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Axial clearance
Condition	Extension housing	M220	Thin section deep groove ball bearing	The low viscosity oil / 19.7 ~ 24.6 °C	Blocked by a circular steel plate with sealant	$e_{al} = 87.5 \text{ mm}$ $e_{ar} = 42.0 \text{ mm}$

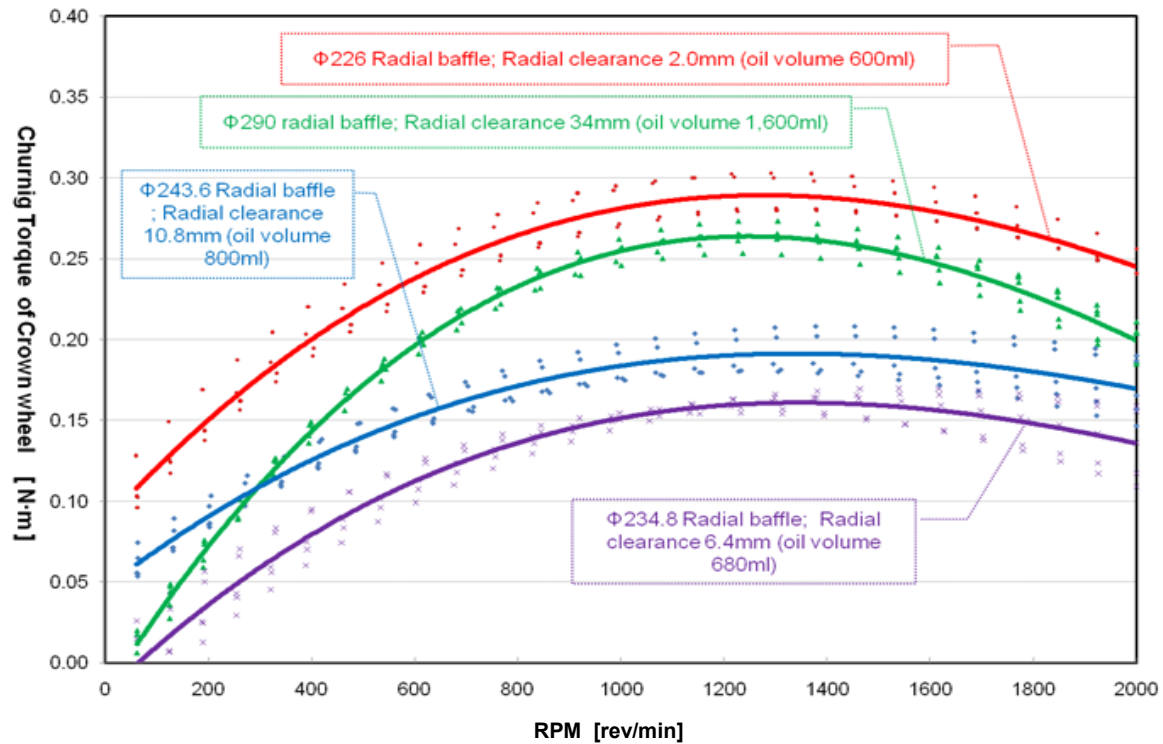


Figure 5.5 Effect of radial clearances of the crown wheel with 220 mm dia. at the oil immersion depth of 31 mm on the churning torque

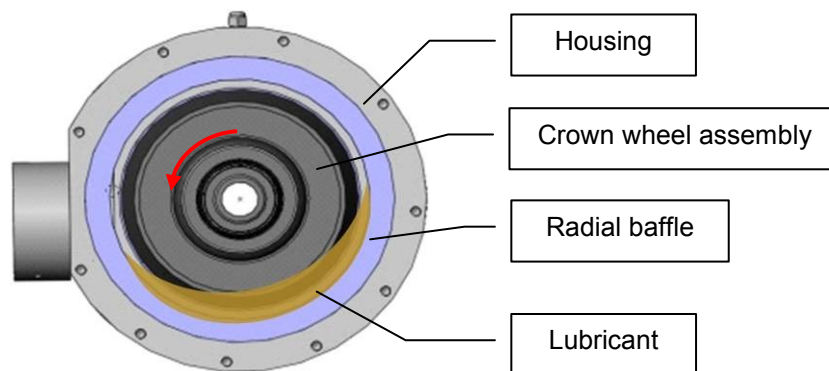


Figure 5.6 Behaviour of free surface of lubricant inside the extension housing while the ring gear was rotating forward, which was observed through the transparent lid of the extension housing

As shown in Figure 5.5, the radial clearances, in order of increasing values of the churning torque and losses, were 6.4, 10.8, 34 and 2 mm for a mineral oil immersion depth 31 mm. The differences of churning torques among the radial clearances were approximately 0.04 to 0.06 Nm. These graphs clearly show the effects of lower immersion depths with the small radial clearances compared with higher immersion depth shown in Figure 5.7, 5.8 and 5.9. The churning torque started to reduce from 1250 rev/min because the free surface of oil was moved away through the induction of air draft caused by the rotating crown wheel to make the oil immersion depth lower as shown in Figure 5.6, which could finally decrease the churning torque. More noteworthy is that the axial clearance 2.0 mm for the radial baffle with a diameter of 226 mm had bigger churning losses than other clearances although the oil volume is the smallest out of the four cases (600 cm^3).

As seen from the graphs in Figure 5.7, the radial clearances at the immersion depth 47 mm, in order of increasing values of the churning torque and losses, can be same as the test results of the oil immersion depth 31 mm. However, the churning torque for a radial clearance 34 mm became much larger than that for a radial clearance 2 mm from 800 up to 2000 rev/min. Additionally, the churning torque for a radial clearance 6.4 mm became bigger than that for a radial clearance 10.8 mm from 1000 rev/min, even if the difference was relatively small to be less than 0.05 Nm. The outstanding feature of the test results is that the different radial clearance (the different distances between the bottom tip of the ring gear submerged and the bottom of the radial wall) at a same oil immersion of the crown wheel could cause different churning torques. The sweeping-away of the free surface at high speed causing the smallest churning torque due to the lower immersion depth could be effective at smallest radial clearance 2 mm. In these immersion tests, the churning torque for a radial clearance 2.0 mm decreased from 1400 up to 2000 rev/min; this could be a result of the small volume of oil and

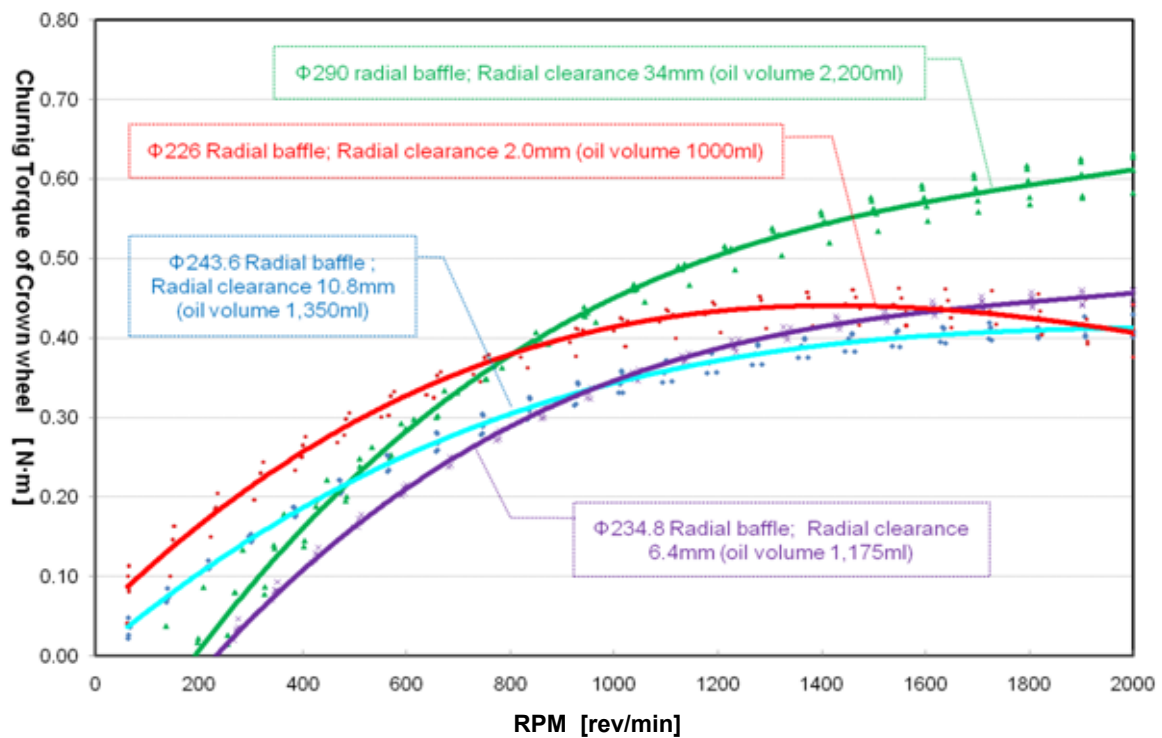


Figure 5.7 Effect of radial clearances of the crown wheel with 220 mm dia. at the oil immersion depth of 47 mm on the churning torque

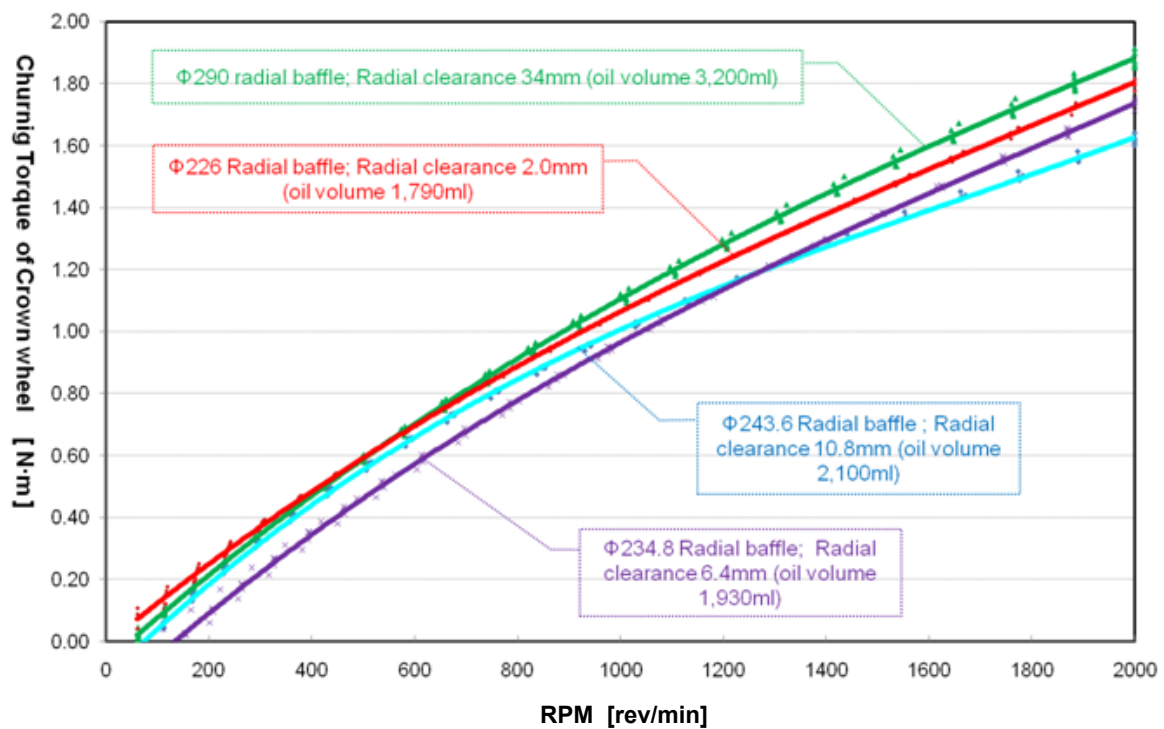


Figure 5.8 Effect of radial clearances of the crown wheel with 220 mm dia. at the oil immersion depth of 71 mm on the churning torque

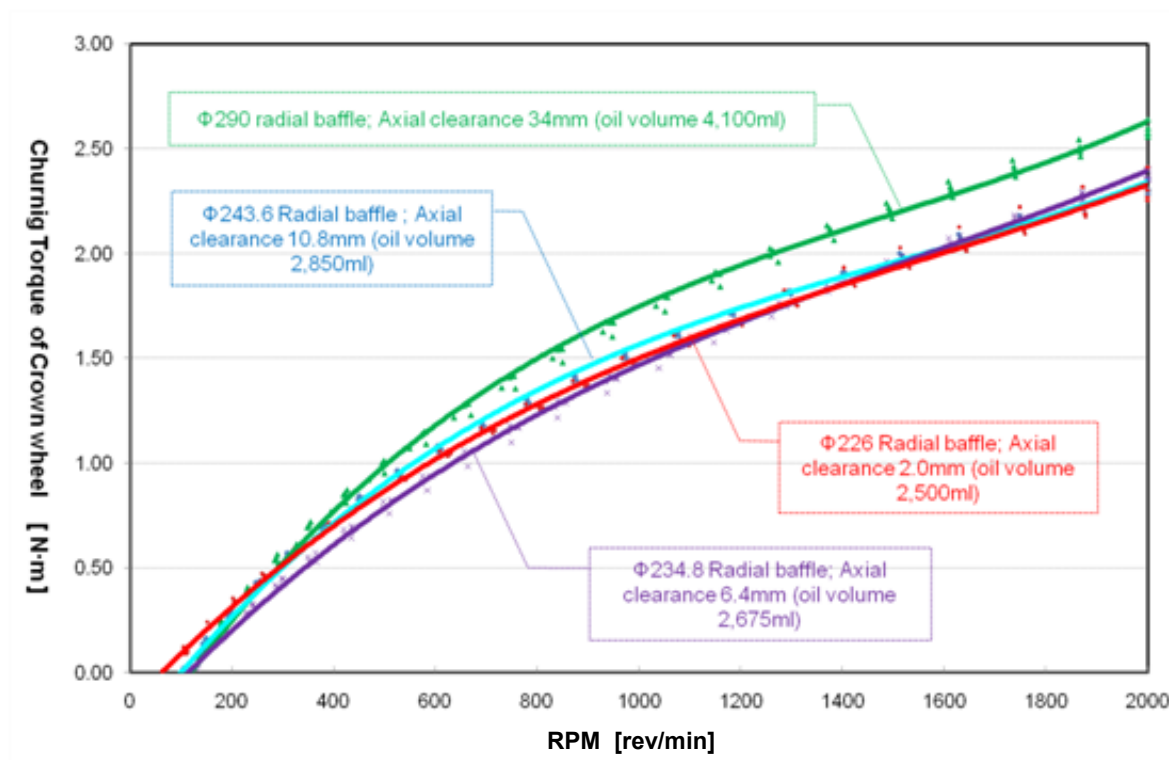


Figure 5.9 Effect of radial clearances of the crown wheel with 220 mm dia. at the oil immersion depth of 91 mm on the churning torque

small radial clearance. Thus, the churning torque and losses for radial clearance 2.0 mm were much lower than for the others at the range of more than 1600 rev/min. As seen from Figure 5.9, the order of churning torques under the mineral oil immersion depth of 71 mm was almost the same as the results for an oil immersion depth of 47 mm; the radial clearances, in order of increasing values of the churning torque, were 6.4, 10.8, 2.0 and 34 mm. The differences of churning torques among these radial clearances were approximately 0.04 to 0.07 Nm. The churning losses for a radial clearance of 6.4 mm became bigger than for a radial clearance of 10.8 mm from around 1300 rev/min.

In the test results for an oil immersion depth of 91 mm shown in Figure 5.9, the churning torques of the smaller radial clearances below 10.8 mm were the same with the difference of

less than 0.02 Nm at any rotational speed and it could be from the partially-submerged differential case. Another reason for the same churning losses for radial clearances of 6.4, 2.0 and 10.8 mm could be caused by the similar oil volumes from 2500 to 2850 cm³ in the extension housing, which showed that oil volume could be a factor influencing the churning losses. For an oil immersion depth of 91 mm, a differential case that consists of pinion, pinion gear and side gear was considerably submerged with oil and half of the hypoid drive pinion gear was immersed. It means that a differential case having pinions in its inside space could affect the oil flow as well as ring gear.

5.4.2 Effects of Axial Baffles

In this section, the experimental results for the effects of axial clearances on churning losses are presented. Table 5.3 shows the test conditions for Figure 5.10. The oil immersion depth of the tests results in this section was 64.5 mm corresponding to the oil immersion depth for an oil volume of 1000 cm³ of the production axle housing, except the test result for the axial baffles of axial clearances 7.5 and 17 mm for the positions of '2 by 2' with an immersion depth 71 mm. The radial baffle with an inner diameter of 226 mm was employed for the axial clearance to be 2.0 mm, which was in pursuit of minimising the effect of radial clearance on churning losses. As previously described, the churning losses for axial clearance 2.0 mm was relatively large when compared with other clearances.

Table 5.3 Test condition for the test result of Figure 5.10

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Radial clearance
Condition	Extension housing	M220	Thin section deep groove ball bearing	The low viscosity oil / 17.5~22.1 °C	Blocked by a circular steel plate with sealant	2.0 mm (ϕ 226 radial baffle)

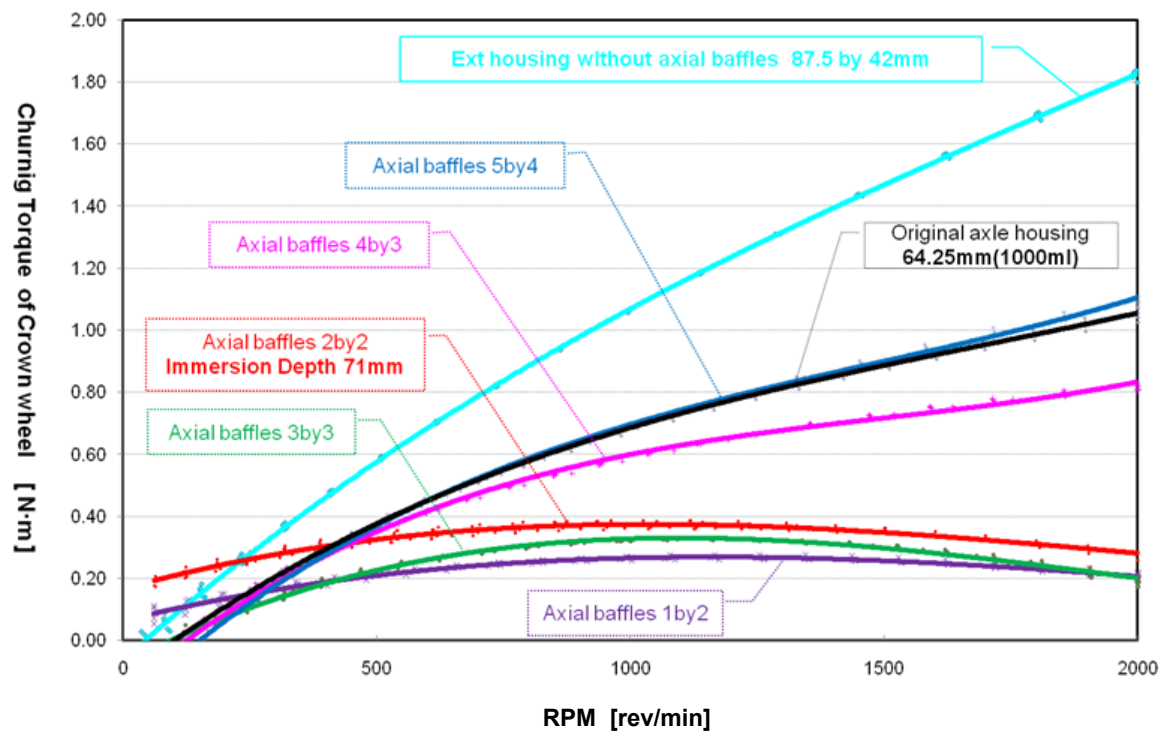


Figure 5.10 Effect of axial clearances on the churning torque of the crown wheel with 220 mm dia. at the oil immersion depth of 64.25 mm under a radial baffle of 226 mm dia.

Figure 5.10 clearly shows that the smaller axial baffles gave the smaller churning losses and the churning torques plotted as a tinted blue line for the extension housing in the absence of axial baffles were definitely larger than others. In addition, the condition of oil immersion depth 71 mm with axial baffles ‘2 by 2’ caused the churning losses to be bigger than oil immersion depth 64.25 mm with axial baffles ‘3 by 3’, which showed that oil immersion depth was more dominant on churning losses than axial clearances. It is noteworthy that the churning losses of the production axle housings were in a good agreement with those of the extension housing with axial baffles ‘5 by 4’.

5.4.3 Effect of Maximum Rotational Speed on Run-down

In order to inspect the effect of an initial speed of the inertia run-down method, which had been chosen in this research, on the measured values, three initial rotational speeds 500, 1000 and 2000 rev/min were employed on the tests in this section and their results were compared. For these tests, the conditions are shown in Table 5.4. The run-down experiments slowed down from 500, 1000 and 2000 rev/min were carried out respectively with the radial baffle of an radial clearance 2.0 mm according to the varied axial clearances. In this section, the effect of an initial speed of these run-down tests was investigated under the oil immersion depths of 64.5 and 71mm.

Table 5.4 Test condition for the test results of Figures 5.11 ~ 5.13

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Radial clearance
Condition	Extension housing	M220	Thin section deep groove ball bearing	The low viscosity oil / 17.5~22.1 °C	Blocked by a circular steel plate with sealant	2.0 mm ($\phi 226$ radial baffle)

The test results of inertia run-down from 500 and 1000 rev/min as an initial speed were in good agreement within a margin of reproducibility errors ± 0.05 N and ± 3 W as shown in Figure 5.11 although there was a bit difference at the speed of 500 rev/min. However, the test results from up to 1000 and 2000 rev/min were not in agreement with each other as shown in Figures 5.11 ~ 5.13.

Figure 5.12 represents the comparison of churning torques of the crown wheel submerged in the oil depth 64.25 mm between 1000 and 2000 rev/min as an initial speed. For the axial baffles of '1 by 2' and '3 by 3', the churning torques measured by run-down from 2000

rev/min were smaller than 1000 rev/min above 500 rev/min due to the bigger sweeping-away of free surface of oil in the run-down from 2000 rev/min. The considerable inertia force for slowing down from a higher speed, 2000 rev/min, could cause this outstanding feature. In addition, the overall lower churning torques and losses measured by the run-down from 2000 rev/min could be a result of the bigger scooping of oil from the oil sump. Accordingly, the scattering of the oil around inside the housing at higher rotational speeds reduces oil immersion volume and oil immersion depth in the oil sump at the range of rotational speeds below 2000 rev/min. Figure 5.13 shows the comparison of churning torques of the crown wheel submerged in the oil depth of 71 mm for 1000 and 2000 rev/min as an initial speed of the run-down tests. The churning torques for axial baffle ‘1 by 2’ and ‘3 by 3’ also showed a tendency of a big difference comparatively with those of the other axial baffles.

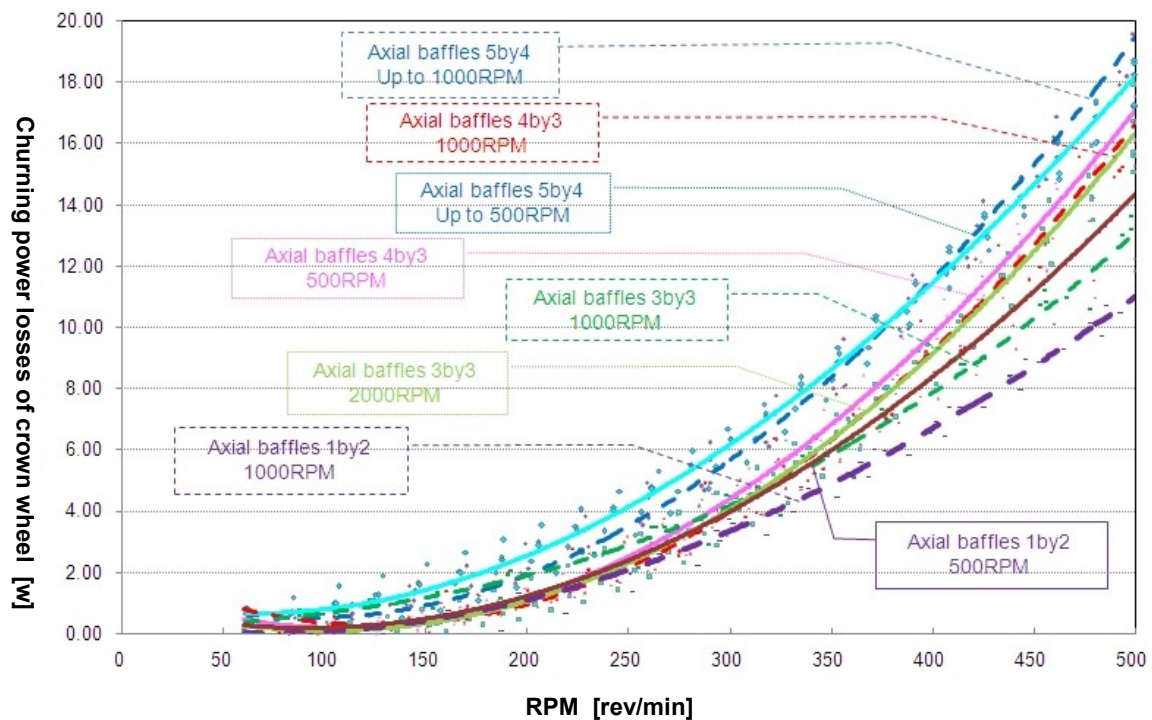


Figure 5.11 Comparison of churning losses of the crown wheel with 220 mm dia. at the low viscosity oil immersion depth of 64.25 mm under radial baffle of 220 mm dia. between 1000 and 500 rev/min as an initial speed of run-down test

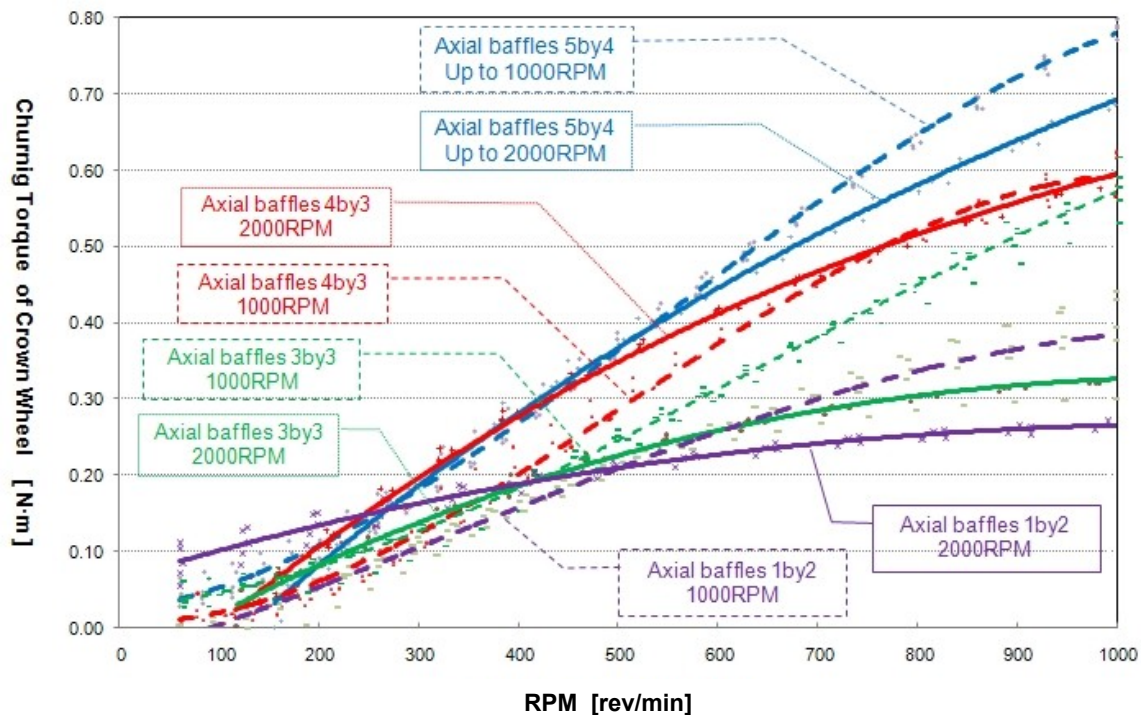


Figure 5.12 Comparison of churning torques of the crown wheel with 220 mm dia. at the oil immersion depth of 64.25 mm under radial baffle of 226 mm dia. between 1000 and 2000 rev/min as an initial speed of run-down test

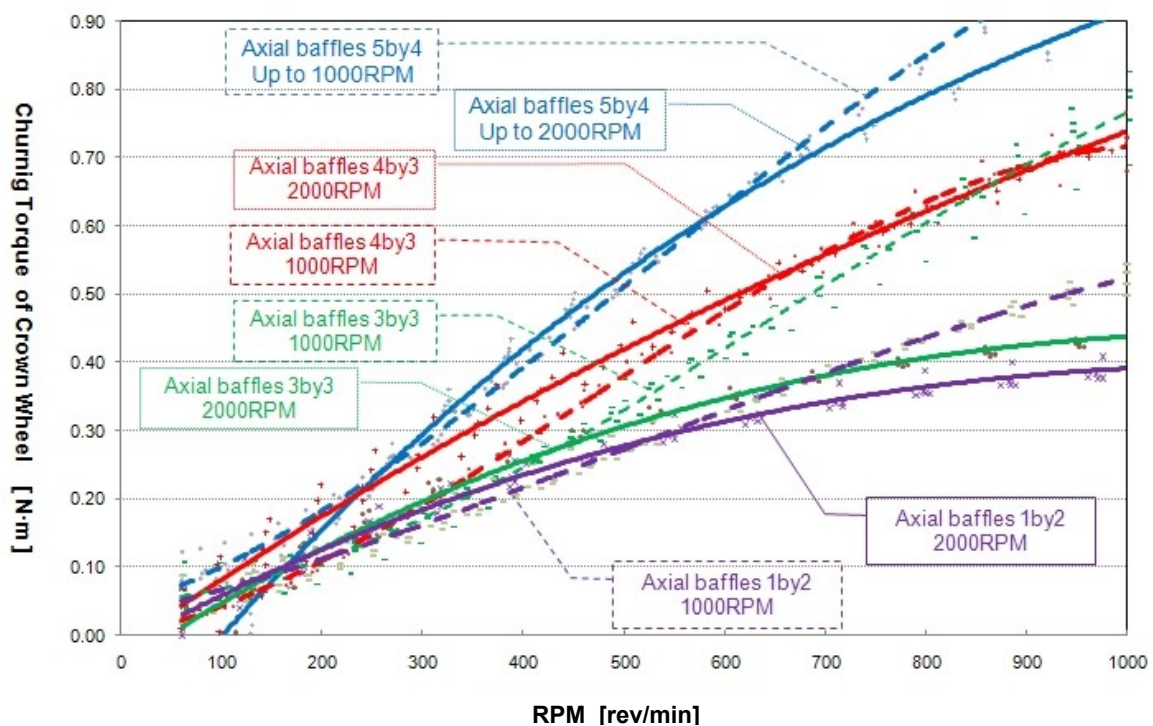


Figure 5.13 Comparison of churning torques of the crown wheel with 220 mm dia. at the oil immersion depth of 71 mm under radial baffle of 226 mm dia. between 1000 and 2000 rev/min as an initial speed of run-down test

5.5 Effect of Lubricant

In order to understand the effect of lubricant, especially the viscosity, on churning losses, two lubricants, the low viscosity oil and the high viscosity oil were also employed to the churning tests of extension housing with radial and axial baffles.

5.5.1 Low Viscosity Oil

Table 5.5 shows the test conditions for Figure 5.14. In this experimental study, the radial baffle with a diameter of 234.8 mm was chosen and the axial baffles with a position of ‘2 by 2’ were installed in the extension housing, which according to the previous test results gave the lowest churning losses. The position of axial baffles of ‘1 by 1’ with the smallest axial clearances was not employed because it could bring about the smallest oil volume in the oil sump. These enclosure conditions were aimed to inspect the effect of the viscosity of lubricants on churning losses and to make sure that the lowest churning torques in the extension housing would take place at the same time. Accordingly, the axial clearances e_{al} and e_{ar} for the position of axial baffles of ‘2 by 2’ were 7.5 and 17 mm and the radial clearance was 6.4 mm. In addition, taper roller bearings were employed on the crown wheel with a pitch diameter of 220 mm.

Table 5.5 Test conditions for the test result of Figure 5.14

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Radial / Axial clearance
Condition	Extension housing	M220	Production Taper roller bearing	Low viscosity oil / 20.4~27.6°C	Blocked by a circular steel plate with sealant	6.4 mm ($\phi 234.8$ baffle) / $e_{al} = 7.5$ mm $e_{ar} = 17$ mm (Position ‘2 by 2’)

The Figure 5.14 shows that the churning torques of the tests results with regard to rotational speeds varied depending on the oil immersion depths 31, 47, 71 and 91 mm. The values of churning torque and losses increased generally with the rotational speeds, but the churning torques below an oil immersion depth of 47 mm did not rise up any more from 900 up to 2000 rev/min and still more reduced a bit. When compared with the values for the radial clearance of 6.4 mm without any axial baffles in Section 5.4.1, the churning torque decreased by more than 30% at all the immersion depths. Furthermore, the axial baffles installed with a small clearance of radial baffle in the housing were more effective in reducing the churning losses at higher rotational speeds. The increasing rate of churning torque at an oil immersion depth of 91 mm was much bigger than the other oil immersion depths 71 and 64.25 mm at higher speeds and did not show a level out or a fall before reaching 2000 rev/min. It could indicate that the effect of axial baffles was not strong at a high oil immersion depth of a pinion housing immersed largely in the oil.

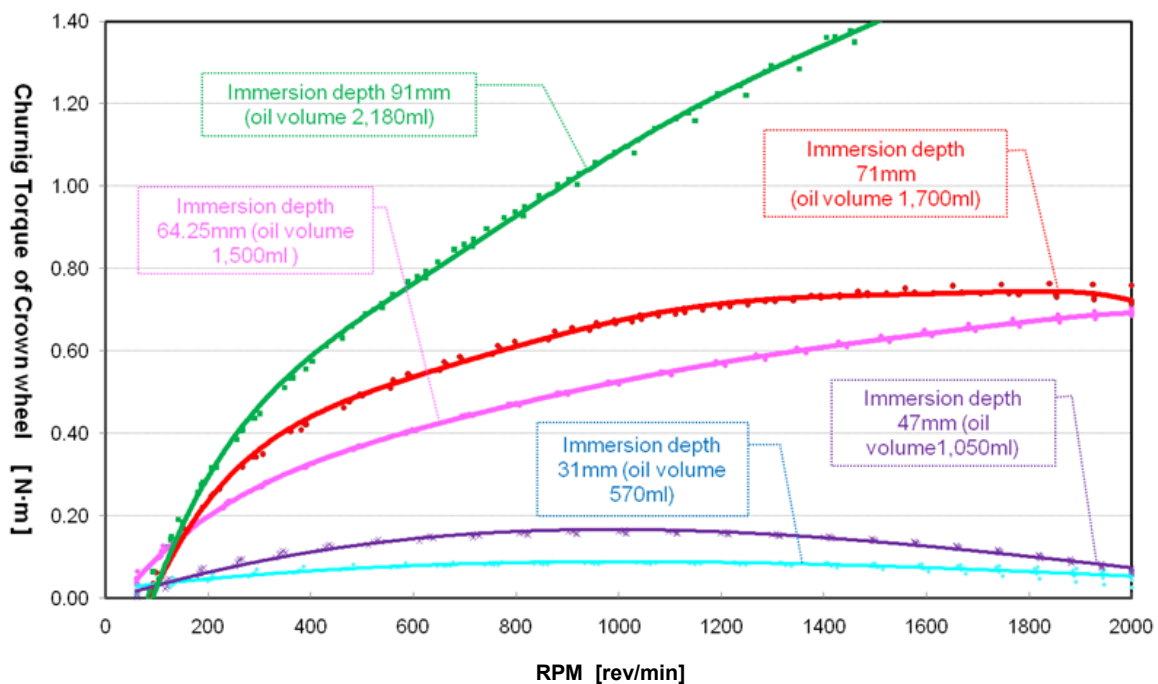


Figure 5.14 Churning torque of the crown wheel with 220 mm diameter, which is submerged in the low viscosity oil and rotating forward under a radial baffle of 234.8 mm diameter and a position of axial baffles of ‘2 by 2’

5.5.2 High Viscosity Oil

The same tests as described in Section 5.4.1 were performed again for the high viscosity lubricant to inspect the effect of high viscosity of lubricant on churning losses. The test conditions are shown in Table 5.6 and the results were plotted as churning torques against rotational speeds as shown in Figure 5.15.

Table 5.6 Test condition for the test result of Figure 5.15

Item	Housing	Size of ring gear	Bearing of crown wheel	Lubricant / Temperature	Hole for pinion gear	Radial / Axial clearance
Condition	Extension housing	M220	Production Taper roller bearing	The high viscosity oil 22.8~28.9℃	Blocked by a circular steel plate with sealant	6.4 mm (baffle dia. $\phi 234.8$) / $A_a = 7.5$ mm $B_a = 17$ mm (Position 2 by 2)

From Figure 5.15, the churning torques for the high viscosity oil were much lower than any other churning test results previously carried out and furthermore, the values of churning torque for oil immersion depths below 71 mm began to decrease before the rotational speed reached 1200 rev/min. When compared with the churning torques for the mineral oil, the values decreased approximately by 40% at the immersion depths of 64.25 and 71 mm from the rotational speed of 1000 rev/min. The inflection point of churning torques for an oil immersion depth of 71 mm was approximately at 1100 rev/min, those of oil immersion depths 64.25 mm and 47 mm were around 900 rev/min. It may have a tendency to be dependent on the oil immersion depth in this splash lubrication. The inflection point of the rotational speeds became to be very low comparatively depending on the oil immersion depth and the increasing torque rate with rotational speeds was small. The sweeping-away of the lubricant free surface from shallow oil in the round housing could be more effective in the case of high

viscosity of lubricant under the smaller axial clearances. However, in the case of the oil immersion depth 91 mm, the churning torque and losses were very large.

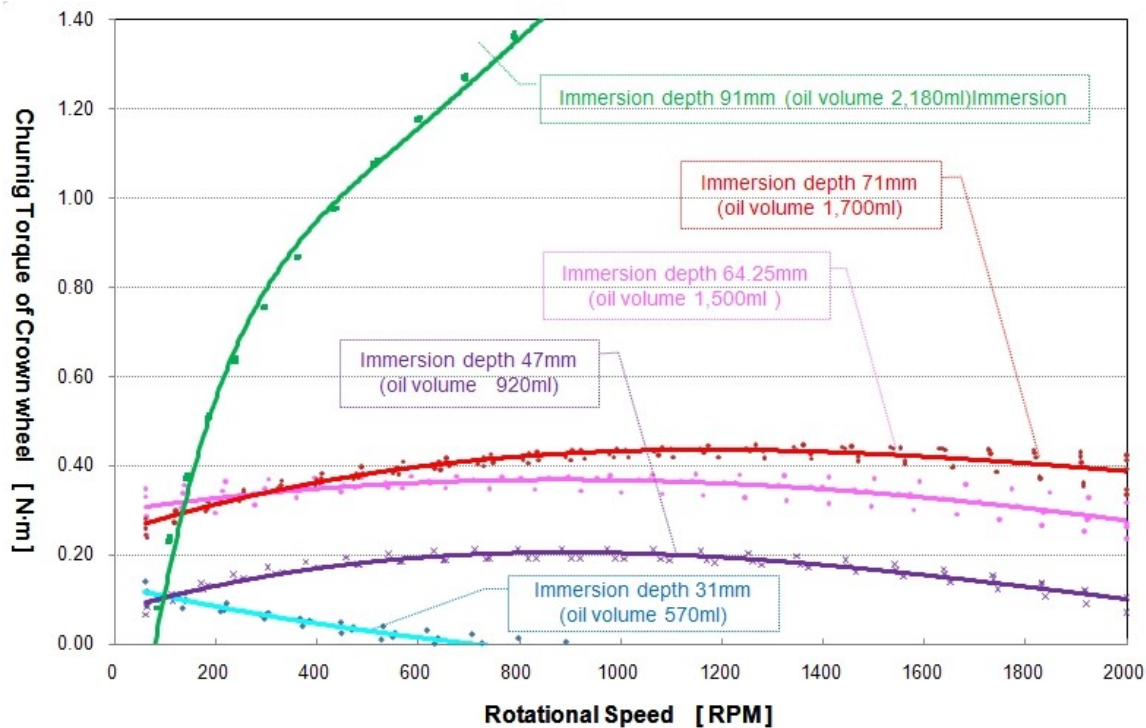


Figure 5.15 Churning torque of the crown wheel of 220 mm diameter, which is submerged in the high viscosity oil and rotating forward under a radial baffle of 234.8 mm diameter and a position of axial baffles of ‘2 by 2’

The salient feature is that the churning losses from the high viscosity oil were not always bigger than the churning losses of the low viscosity oil although the lubricant viscosity at test temperatures was as much as eight times that of the low viscosity oil. When compared between Figure 5.14 and 5.15, the churning losses of the low viscosity oil at the highest immersion depth was smaller than for the high viscosity oil, but the churning losses of the low viscosity oil for other volumes and immersion depths were almost same or larger than those of the high viscosity oil. However, the fact that churning losses depended mainly on the viscosity of the lubricant, not on the lubricant type was described by Michaelis and Höhn (Michaelis & Höhn, 1994) in 1994. Thereafter, another research on the energy efficiency of industrial oil showed that churning losses in a hydraulic system are mostly influenced by oil viscosity while

friction losses largely are dependent on oil chemical position (Bronshteyn & Kreiner, 1999). Further discussion about this point will be presented in the next chapter.

From the above test results, smaller axial clearances with a high viscosity lubricant showed a high sweeping-away tendency of the oil free surface as previously shown in Figure 5.6. Namely, high viscosity lubricant showed the sweeping-away of the free surface was much larger than the low viscosity oil at the higher speeds than 1000 rev/min. In an oil immersion depth of 64.25 mm, churning torque of the high viscosity oil is less than mineral oil at above 700 rev/min, whereas two lubricants are almost same in the immersion depth of 47 mm.

5.6 Concluding Remarks

The radial and axial baffles were employed in the newly manufactured extension housing to investigate the effect of housing clearances on churning. As expected for spur gears, axial clearances showed a bigger influence on the churning losses than radial clearances, and from the experimental observations in the present study, it was proposed that there is a critical radial clearance to be able to reduce the churning. Additionally, the oil flow was observed from the rear side of crown wheel through the transparent side of the extension housing.

Further investigations, such as the effect of the foam or bubbles at higher speeds and the quantitative measurement of bubbles and thermal balance of different temperatures caused by a rise of oil temperature due to friction and churning losses, are required to understand fully the mechanisms of churning losses for hypoid gears.

In Chapter 6, these findings are compared and discussed critically in the light of the experimental studies of Chapter 2 and the analysis of experimental data is described in Chapter 4 and 5.

CHAPTER 6

DISCUSSION

6.1 Introduction

The discussion in this chapter is divided into four main sections. In the first, the findings of the detailed investigations on the test results reported in chapters 4 and 5 are summarised and secondly, those are analyzed and compared with each other. Thirdly, the empirical equations for the churning losses in axles, are deduced. Finally, some suggestions are made for the practical exploitation of the increased understanding provided by the present work in order to reduce the churning losses and improve the efficiency in gear boxes such as axle housing.

6.2 Summary of Test Results (Chapter 4-5)

In this section, a short summary of these findings is given prior to a more detailed discussion of their significance to the churning losses, and the potential issues on the experiments are presented.

6.2.1 Summary of Chapter 4

Chapter 4 reported that the experimental study for obtaining churning torque and churning losses in production axle housing was carried out to investigate the influencing variables from the sizes of crown wheel, different viscosity of lubricants, the presence of hypoid pinion and the tilting position of axle housing.

It clearly showed that the immersion depth of the rotating crown wheel submerged in the lubricant as well as rotational speed of the crown wheel is predominant in determining the churning losses. The churning torque and losses certainly increased with the oil immersion depth and rotational speed of gear. However, the churning torque culminates at a certain higher rotational speed and then begins to fall gently. This phenomenon was due to the sweeping of free surface of oil as shown in Figure 5.6 and it was prominent in the lower immersion depth and under the high viscosity oil. From the comparison tests of the churning torque between the oil viscosities, the churning torque of the high viscosity oil was higher than that of the low viscosity oil at low speeds, but it was not at high speeds.

The bigger size of crown wheel gave rise to more churning losses over the whole range of rotational speeds except those below 500 rev/min. These experiments of the hypoid crown wheel were performed under both thin section deep groove ball bearings and production taper roller bearings from the concerns that enough running-in was not done in the deep groove ball bearings and a bit higher friction resistance existed inside the ball bearings.

In addition, the effects of the existence of hypoid pinion and housing tilting positions were also inspected on the churning torque measured in the axle housing. The existence of drive pinion gear gave rise to more churning losses than its absence, but its oil immersion depth due to roll and pitch position had little influence on the churning losses for a hypoid gear pair at certain immersion depths.

6.2.2 Summary of Chapter 5

Chapter 5 reported that the experimental study for the effects of enclosures and viscosity of lubricant in the newly manufactured extension housing was performed. The test results in Chapter 5 showed that the higher viscosity of lubricant does not always bring about higher churning losses and the axial clearances were effective in reducing the churning losses for hypoid ring gear rotating in splash lubrication. However, it appears that the reduction of churning losses was affected by the other influencing factors of immersion depths and radial clearances, which made the effect of axial clearances bigger or smaller.

In addition, the effect of the initial rotational speed in inertia free run-down method was inspected for 500, 1000 and 2000 rev/min. It showed that there could be a small difference in churning losses between acceleration and deceleration at above around 800 rev/min of the initial run-down speed and the churning torque at 1000 rev/min measured down from 2000 rev/min was smaller than that down from 1000 rev/min. Finally, the oil flow inside the housing explained why the high viscosity oil for the churning losses was more efficient than the low viscosity oil at high speeds. The high viscosity oil was much scooped away and stuck on the housing surface longer than the low viscosity oil.

6.2.3 Potential Issues in the Experiments of This Thesis

In this section, some potential issues that limited test conditions and may influence the test results in the experiments of this thesis are presented.

1. The no-oil drag torque depended on the type of bearings and this affected the total drag torque. Although the absolute values of churning torque were different as shown in Figures 4.5 and 4.10, the relative results that the churning torque varies with the oil immersion depth

were the same. The fitting clearance of the bearings for the hypoid axle was non-standard and it was difficult to find a thin section deep groove ball bearings with the proper fitting clearance to the shaft of crown wheel. In addition, there was no need to apply a thin section bearing instead of a normal ball bearing because the bearing loss is not dependent on the size of contact area or ball size in ball bearings. Another critical reason for the higher ball bearing drag torque was the starvation in rolling bearings. Starvation can be caused by the sweeping of the lubricant from the track by the rolling elements of the bearings (Olver, 2002), which led to sweep the lubricant inside ball bearings out easily rather than taper roller bearings in the empty oil sump. Thus, the data for the no-oil drag torque of the thin section ball bearings were obtained in the absence of lubricant inside them, and so friction losses in the bearings could be bigger than under normal lubricated running condition.

2. The observation of oil flow inside the extension housing was limited to the rear non-toothed side of the crown wheel. The application of a transparent wall in the housing for the inspection of oil flow was determined at the final stage of experiments and there was a limitation in employing the whole transparent housing due to the time, cost and structure strength. Thus, a transparent material was applied only to the lid of housing and only the rear side of the hypoid axle was able to be seen through it.

3. The churning losses caused by bearings as well as load-dependent losses such as gear and bearing friction losses were not isolated perfectly although load-dependent losses were not large because no load was transferred between hypoid gears in the experiments. By the way, the thermal balance method might be useful to find the evidence of churning mechanism by measuring the local temperatures with a good insulation of housing. The measurements of local temperatures should be taken around the bearings and the surface of housing as well as the lubricant.

6.3 Discussion of Churning Losses

In all the experiments of measuring churning torques detailed in this thesis, churning torque values were obtained through a decay curve of rotational speed versus time. The decay curve shown previously in Figure 4.3 indicated that a higher immersion depth in lubricant made run-down time shorter and it means that a bigger drag torque of lubricant acts on the rotating component, the crown wheel, as the immersion depth is deeper. The regression graphs obtained from the decay curve showed that the churning torque increased with rotational

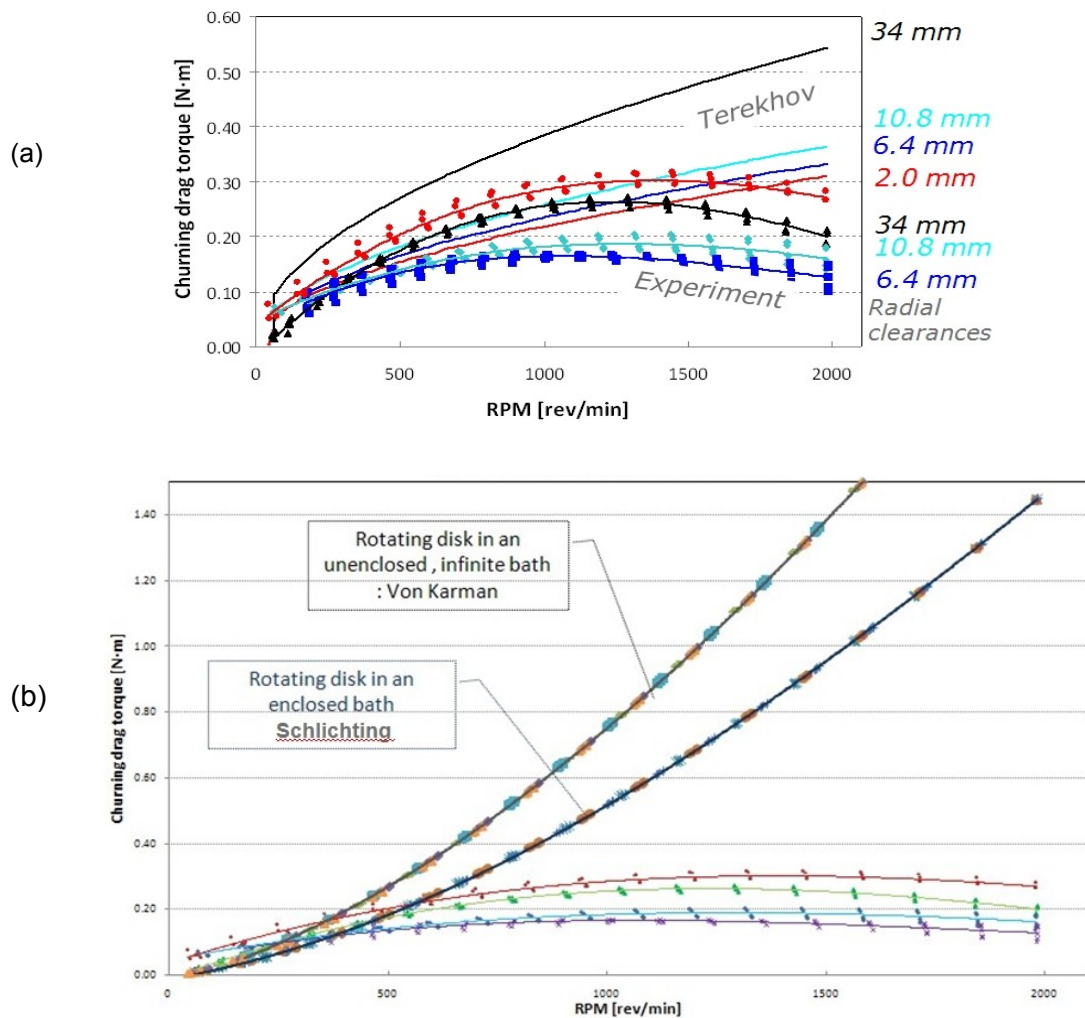


Figure 6.1 Comparison of test results in extension housing for various radial clearances with the values calculated from (a) Terekhov's empirical formula and (b) Von Karman and Schlichting, ring gear of 220 mm dia. and 31 mm oil depth

speeds to the power of a number less than 1.0. In other words, a rate of increase of the churning torque became smaller as rotational speed increased. Moreover, above a certain speed, the churning began to reduce as the speed increased unlike Terekhov's formula (Terekhov, 1991) as shown in the graph of Figure 6.1(a). Figure 6.1 shows how the measured churning torques varied with the radial baffles in the extension housing and how these compared with the previous formula for churning torque. The difference between the test data and the previous formula is large especially at higher rotational speeds. The churning torques from the formulae of Terekhov, Von Karman (1921) and Schlichting (Schlichting & Gersten, 2000) were based on a rotating spur gear in an oil sump or a disc rotating in an infinite fluid whereas the experimental data were obtained from a hypoid ring gear.

From the author's test results, it could suggest that the viscous force is more predominant on rotating gears at low speeds, but for high speeds, the internal force, such as inertia force from the rotating crown wheel, has a greater influence on churning torque of the lubricant compared to the viscosity of the lubricant. In addition, most experiments in this thesis were performed with varied immersion oil depths. These churning torques and power losses for different oil immersion depths increased as the rotational speed increased and the oil immersion depth of the hypoid ring gear was deeper. This indicated that the rotational speed is a significant factor affecting churning losses, and that the immersion depth, or oil volume, is more influential at higher speeds.

It was found from the experimental results in Section 4.4 that the churning torque of the crown wheel for production axle increased with rotational speed up to a limiting value, but then a further increase of speed lead to the drag torque decreasing for small immersion depths of the low viscosity oil. The reason for this phenomenon is thought to be the high-speed rotating ring gear induced a stream of air into the lubricant such that the oil was blown away

from the rotating gear. The blown free surface of lubricant thereby made the oil immersion depth of crown wheel smaller as previously shown in Figure 5.6. Thus, the lubricant in the housing was usually blown off by the air draft arising from the rotating crown wheel at high speeds. Luke and Olver (1999) described it as windage from the rotating gear and it was seen to displace the fluid surface so that the immersion depth on the gear was reduced. Furthermore, this phenomenon was clear and distinct in the high viscosity lubricant, SAE 75W90 from the author's experiments shown in this thesis.

On the other hand, Changenet and Velez (2007) presented a swell effect, which is where meshed spur gear teeth trap the lubricant between a pinion-gear pair rotating in opposite direction as shown in Figure 2.7. For this case, the swell effect of the meshed pinion-gear pair increased churning losses more than a sum of the independent churning losses of pinion and gear; which was from the increased immersion depth of the smaller gear. Changenet's test conditions were that the rotational speed of pinion was from 500 to 1500 rev/min and the immersion depth over pitch radius is between 0.4 and 0.7. Therefore, this swell effect occurring in a pair of meshed gears is a factor that increases the churning losses, whereas it could reduce the churning if the swell happens to the free surface of the lubricant by a high-speed rotating ring gear submerged in the oil. This swell comes to sweep the oil away to the housing as previously described.

6.3.1 Geometric Effects of Crown Wheel Gear on Churning Losses

To examine the effect of the size of crown wheel, two different diameters were employed to measure the churning torque. From the test results, the overall churning torque for the crown wheel of 220 mm dia. was higher than that of 210 mm dia. as shown in Figure 6.2. Figure 6.2

shows a comparison of the two different diameters as previously shown in Figure 4.5 and 4.12. The difference between churning torques for 220 and 210 mm increases with rotational speeds. In addition, the temperature change after run-down tests showed a tendency of the crown wheel of 220 mm dia. being a little bigger than that of 210 mm dia. although the difference was less than 0.5°C . It is likely to show that the churning losses caused by the rotation of the crown wheel of 220 mm, is higher than that of 210 mm even if the difference is very small.

Although the actual immersion depth and the radial clearance were different according to the size of crown wheels, the radial clearance could not be significant enough to influence on the churning losses when compared to the immersion depth. The difference of immersion depth and radial clearance between the crown wheels of 220 and 210 mm dia. is relatively small at just 4.4mm. The radial clearance of the crown wheel with a tip diameter of 222 mm for that of 220 mm is 4.4mm smaller, and the oil immersion depth for that of 220 mm is 4.4mm deeper than that of 210mm. Thus, the deeper immersion depth of the crown wheel of 220 mm, even if it is a very small difference, could influence on churning losses as a factor.

If there is a more influencing factor rather than the above factors on the difference between the different sizes of crown wheels, it could be the surface area of contact between the crown wheel assembly and the lubricant. It was considered as a significant variable by Changenet and Velez (Changenet & Velez, 2007, 2008), and the submerged volume was considered instead of submerged surface area by Terekhov (1991) in developing the dimensional analysis for obtaining the churning losses of spur gears rotating in splash lubrication. In addition, the submerged surface area of spur gear was also used as a physical criterion for churning losses by Seetharaman and Kahraman (Seetharaman & Kahraman, 2009a, b). Thus, the larger submerged surface area of the crown wheel of 220mm could influence the bigger churning losses.

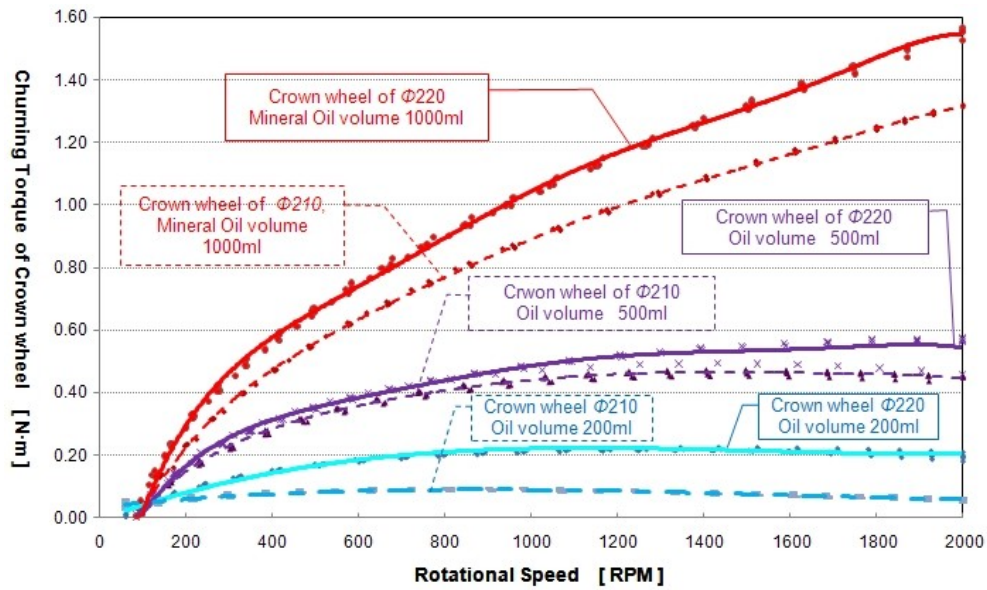


Figure 6.2 Comparison of churning torque between crown wheels with a diameter of 220 and 210 mm submerged in the low viscosity oil

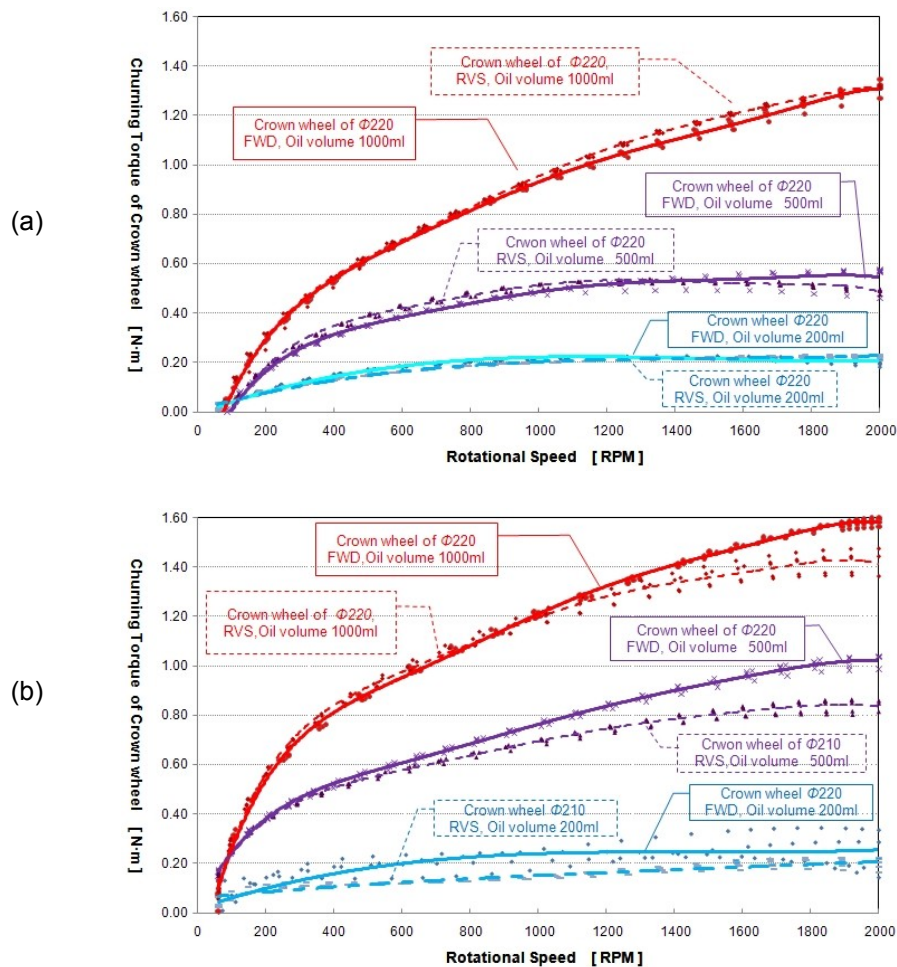


Figure 6.3 The effect of rotating direction on churning torque of the crown wheel with a diameter of 220 mm (a) submerged in the low viscosity oil (b) submerged in the high viscosity oil

The noteworthy point is the radial clearances of them are of difference as 4.4 mm caused by two different diameters of ring gears. However, it seems that churning drag was barely affected by small radial clearances because the difference of churning torque between radial clearances 10.8 and 6.4 mm was less than 0.05 Nm as previously shown in Section 5.4.1. Therefore, the factors such as oil immersion depth, oil volume and submerged surface area due to the size of gear as a gear geometry are influential on churning losses of hypoid gears.

The spiral angle of ring gear teeth could be considered, along with the rotational direction as an influencing factor in some conditions from Figure 6.3. The rotating direction of the crown wheel submerged in the low viscosity oil did not affect the churning losses. However, when submerged in the high viscosity oil, the churning torque of the reverse-rotating crown wheel was less than the case of rotating forward. It also shows that the difference between forward and reverse rotating increased as rotational speeds. It seems that the direction of the spiral angle of the teeth has an effect on the churning losses from the fact that the teeth scooped some lubricant up to a height by the concave part of the spiral shaped teeth when forward rotating, whereas, the lubricant was flung off outward by the bulged portion of the spiral shaped teeth when rotating reverse. Therefore, the spiral angle of ring gear in splash lubrication using a high viscosity lubricant could influence the churning torque. On the other hand, the spiral-angled teeth of the ring gear rotating forward could scoop and distribute the lubricant well to all the places inside the housing. As a result, this could make the best lubrication environment for a higher durability despite the inefficiency due to the churning losses of high-speed forward rotating of crown gear.

In addition, the test results for the crown wheel gear meshed with the hypoid pinion gear also revealed that churning torque of the meshed ring gear with 220 mm dia. are much larger than that of 210 mm as shown in Figure 4.18 of Section 4.4.4. The salient feature in mating hypoid

gears is that the churning losses for reverse rotating were 10% greater than for forward rotating as shown in Figure 4.19. Regardless of the spiral angle direction, this shows that the swell effect of mated gears approaching under the lubricant free surface increased churning losses, which was previously described in Section 2.5.5. This swell effect described by Changenet and Velez (2007) was limited in the test rotational speed of spur pinion and immersion depths, and disappeared at high speeds. However, from Figure 4.19 in the author's experimental result of hypoid gear pairs, the swell effect increased with rotational speeds and the differences in churning torques between forward and reverse rotating grew. This could be caused by the large ratio of ring gear to drive pinion gear and the non-intersected axis between them.

6.3.2 Effects of Pinion Gear on Churning Losses

This section considers the effect of the presence of the drive pinion gear on churning losses in the axle housing. In Chapter 4, it was reported that the presence of the hypoid pinion gear resulted in more churning losses than running the crown wheel alone. The churning torque of the meshed hypoid gear with diameter 220 mm was about 2.5 ~ 3.5 times of that of the crown wheel without the drive pinion gear when rotating in the low viscosity oil (Figure 4.14 and 4.18). In the meshed hypoid gear of crown wheel with diameter 210 mm, it rose 2.0 ~ 2.7 times when compared with the churning torque of the crown wheel without the drive pinion gear when rotating in the low viscosity oil (Figure 4.10 and 4.18). However, the difference when using taper roller bearings was smaller, being 1.5 ~ 2.4 times from the comparison between Figure 4.12 and 4.21. Thus, the churning losses in the presence of the hypoid drive pinion can be more than twice as large as that of the crown wheel without the drive pinion gear.

From the test results, the churning losses for a meshed hypoid gear pair were greater than the sum of the churning losses for the crown wheel and pinion alone although the crown wheel of which the diameter was much bigger than that of hypoid pinion could be a predominant component in determining churning losses. This can be attributed to the meshed portion. Ariura et al. (Ariura et al., 1973) classified the churning losses of spur gears into five classes based on their sources: accelerating, trapping, splashing the oil, bending the oil flow and inducing air draft. Out of the sources of churning losses defined by Ariura et al. (Ariura et al., 1973), the difference between a single crown wheel and a pair of hypoid meshed gears can be mainly caused by the oil trapping loss. This is the power loss required to trap the oil between meshing gear teeth and deforming it. Thus, the oil trapping loss made the churning losses of a pair of meshed hypoid gears larger than the individual churning losses that are just as much as splashing and accelerating losses out of churning losses.

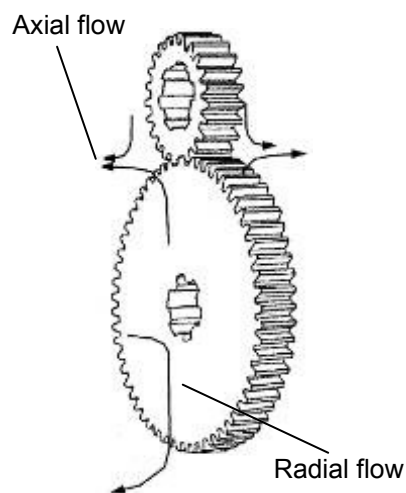


Figure 6.4 Axial and radial air flow in the vicinity of a rotating pair of spur gears (Olver, 2002)

Another reason could be due to the non-intersecting and non-parallel axes of the hypoid gears. In a rotating pair of spur gears as shown in Figure 6.4, the airflow near the gears is predominantly radial around the gear side but axial near the gear teeth, which was presented by Olver (Olver, 2002). The existence of a pinion drive gear could block the radial and an axial flow of the lubricant if the hypoid ring gear is regarded as the large spur gear.

In other words, if there is the meshed drive pinion gear with the crown wheel inside the housing, it may block the oil flow circulating along the rotating crown wheel and make its flow more complicated due to the non-intersecting and non-parallel axes. This was based on the observation of no sweeping-away effect occurring in

the experimental results of churning torque under the low depth of oil volume. As can be seen from Figures 4.18 to 4.19 for a pair of hypoid gears, there was no decline in churning torque at the high speeds. Instead of the decline, there was an increase in churning losses greater than the sum of the individual churning losses for the ring gear and pinion alone at high speeds.

6.3.3 Effects of Vehicle Positions on Churning Losses

In this section, the effect of the attitude of the axle housing on churning losses is addressed. The test results from Section 4.5 showed that the influence of pitch and roll attitude on churning losses was not significant at low speeds less than 800 rev/min for both lubricants, the low viscosity and high viscosity oils.

Depending on the pitch degrees at pitch attitude position, the immersion depths of the hypoid pinion and the crown wheel varied. The deeper immersion depth of hypoid pinion with larger negative pitch angles showed higher churning losses in the high viscosity lubricant. However, the churning losses in the low viscosity oil were not dependent on the immersion depth of the hypoid pinion but had a tendency to reduce if the hypoid gear was not immersed in the low viscosity oil. It seems likely that the immersed meshed part is more influential than the immersion depth of the hypoid pinion in the low viscosity oil. Thus, the influence of pitch attitude on churning losses was not big, but it varied depending on the immersion depth of the hypoid drive pinion at higher viscosity and high rotational speeds.

For rolling attitude position, the rolling angles in the direction of clockwise viewed from the pinion side showed the increase in churning torque, and the other direction of rolling angles presented the bigger reduction at high rotational speeds from Figures 4.24 and 4.25. It seems likely that the increasing immersion depth or immersion surface area of the crown wheel with the increasing rolling angles in the clockwise direction could have an effect on the churning

losses regardless of the immersion of hypoid pinion. Therefore, the influence of rolling attitude on churning losses depends mainly on the immersion depth of ring gear according to the rolling degrees and direction at high rotational speeds.

On the other hand, Olver (2002) pointed out that the common source of oil starvation in splash lubrication is variation in the attitude or inclination of gear boxes and so a circulating oil system with pump is necessary. Therefore, during the above attitude experiments, the taper roller bearings were employed to the crown wheel and attention was given to the supply of lubricant to the bearings in order to exclude the effect of starvation on bearings. Additionally, the running time was short and there was little effect of starvation on bearings.

6.3.4 Effects of Lubricant on Churning Losses

A number of studies (Boness, 1989; Seetharaman & Kahraman, 2009b; Taylor, Dixon, Wayne, & Gunsell, 2004; Terekhov, 1991) have generally shown that spin power losses involving churning losses are highly dependent on the viscosity of the lubricant in splash lubrication. It means that a high viscosity lubricant requires more energy to run the rotating components such as gears and bearings through the lubricant. Meanwhile, Luke and Olver (1999) concluded that the experiments showed relatively little effect of fluid viscosity between SAE20W50 and 10W. Thereafter, Changenet and Vexlex (2007) reported a weak dependence of spin power loss on lubricant viscosity, which will be discussed in more details in Section 6.4.

Test results in this thesis showed that the churning losses for the rotating hypoid ring gear submerged in the high viscosity lubricant was not always higher than those submerged in the low viscosity lubricant (comparing between SAE75W90 and API Group1 Esso-produced

mineral oil). The dependence of churning power losses on lubricant viscosity in the overall experiments of crown wheel was observed to be weak as shown in Figure 4.17 of Section 4.4.3. The churning torque of the higher viscosity were found to be a little higher than the low viscosity oil for forward rotational speeds less than 1400 rev/min, but were lower at higher speeds. From these test results, the oil viscosity is thought to be very influential on churning losses at low speeds because the viscosity determines how much resistance there is to rotation of the gear. However, the effect of viscosity of the oil is not as significant at high speeds when the inertia force of the oil becomes more dominant. Most of all, the salient point is that the churning torque from the high viscosity oil at speeds less than 300 rev/min increased sharply with rotational speed, which can expect that “cold start” of cars could be attributed to the churning torque caused by the high viscosity of lubricant at low temperature.

Furthermore, the high viscosity lubricant showed lower churning losses than the low viscosity oil from the test results as shown in Section 5.5. In these tests, the installed axial baffles were effective in reducing churning losses. The data in Figure 6.5, which was obtained from the data of the graphs in Section 5.5, show the influence of the axial baffles on the higher viscosity lubricant. The axial clearances for these tests were 7.5 and 17 mm for e_{al} and e_{ar} , and the radial clearance was 6.4 mm. The churning torque from the high viscosity lubricant did not increase for the rotational speeds above around 1000 rev/min. This could be due to the lower oil immersion depth caused by the induction of air draft and the sweeping away of the oil free surface as shown in Figure 5.6. As previously explained, the oil free surface moved along the circumferential wall of the round housing in the same direction as the ring gear rotated at high speeds. This effect seems to have been strong in the higher viscosity lubricant, and the axial baffles made it even stronger. It was because the high viscosity oil seems to remain sweeping-away of the free surface to the circumferential direction of the crown wheel along radial baffle surface longer than the low viscosity oil as shown in Figure 5.1(c). If the hypoid ring gear is

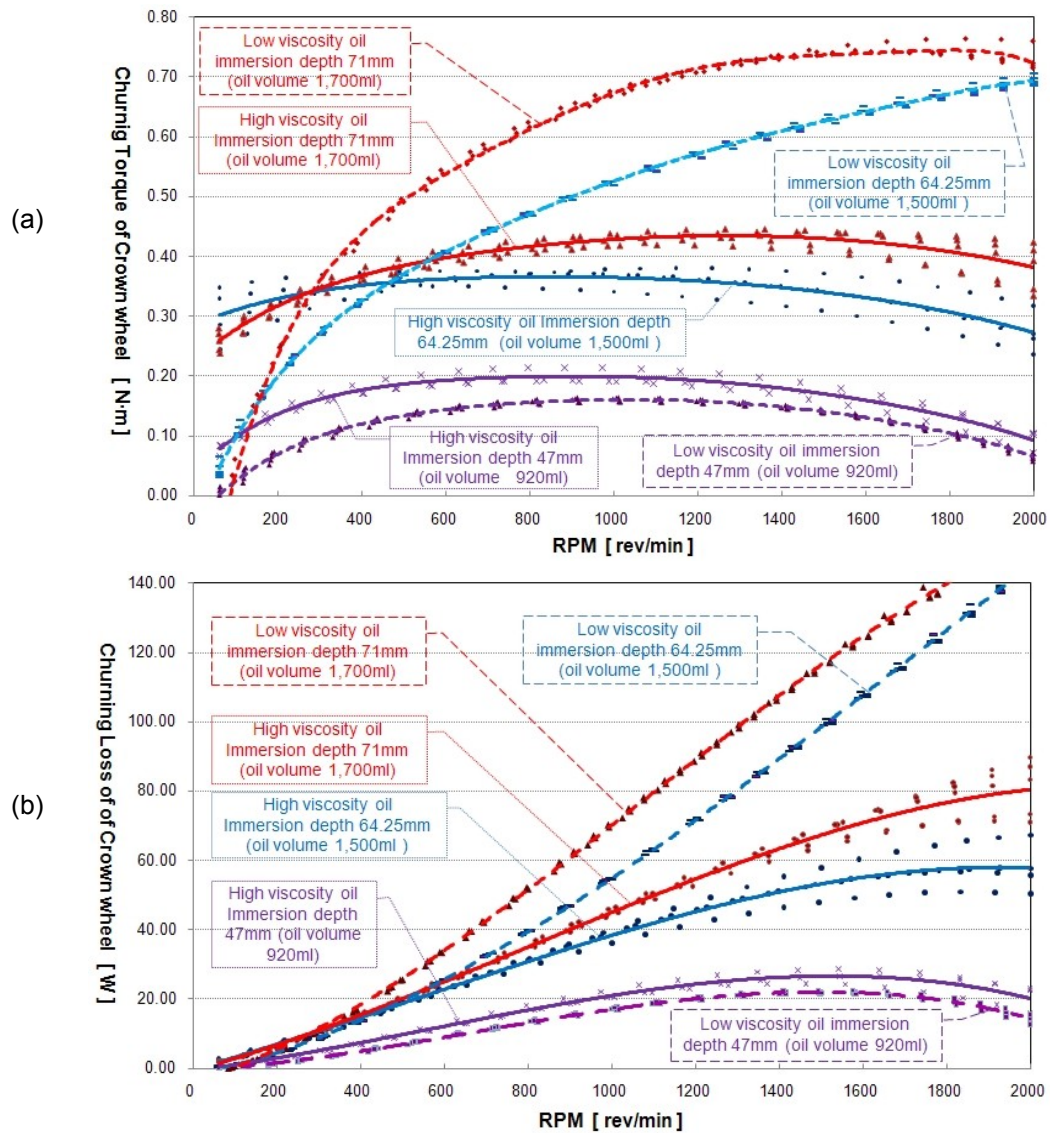


Figure 6.5 Comparison of effect of oil viscosity between the high and low viscosity oil in the extension housing (a) churning torque (b) churning losses

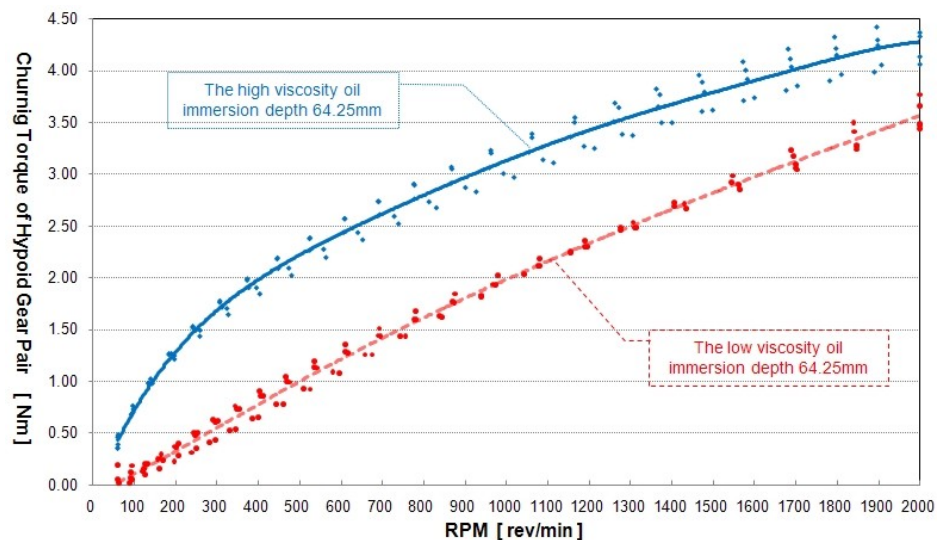


Figure 6.6 Comparison of the effect of oil viscosity for a meshed hypoid gear pair on the churning torque

regarded as a large spur gear, a viscous fluid flows in the same way as an air flow as shown in Figure 6.4. Another reason for the reduction of churning torque is shown in Figure 5.3 (c). The lubricant was flung off onto the housing wall and the rotating components. Then, the slow flow rate of the high viscosity lubricant could stick to them for longer period than the lower viscosity lubricant. Accordingly, the oil level in the sump was reduced so that the churning torque decreased due to the reduced oil immersion depth. Moreover, the presence of the axial baffles could make the oil volume in the bottom of oil sump smaller due to the ring gear's more effective pumping of the oil. A large amount of the pumped oil remained on the surfaces of enclosures and crown wheel due to the high viscosity of oil after the end of run-down of the crown wheel. This resulted in a low oil immersion depth in the sump and the churning torque decreased at higher speeds.

However, Figure 6.6 showed the distinct results that churning was more severe for the high viscosity lubricant rather than the low viscosity oil in the experiments using a pair of meshed hypoid gear. The data of Figure 6.6 came from the graphs in Section 4.5. The rate of increase of churning torque with speed for the high viscosity oil in Figure 6.6 was also higher than that of the low viscosity oil for speeds less than 500 rev/min, for an immersion depth of 64.25 mm. As described in the previous section, the difference between a single crown wheel and a pair of meshed hypoid gears can be caused by the oil trapping loss. This trapping loss for a pair of meshed hypoid gears can be strongly dependent on the viscosity from the test results. Therefore, the influence of the viscosity on the trapping loss of a pair of meshed hypoid gears was strong enough to increase the churning losses, whereas the effect of the viscosity of lubricant on churning losses of a single crown wheel was not strong. Also, it seems reasonable to suppose that the trapping loss has a majority of the churning losses. In addition, jet lubrication for a pair of meshed gears, in which no pumping loss out of churning losses

takes place, can be no less dependent on the viscosity of lubricant than dip lubrication for a pair of meshed gears.

6.3.5 Effects of Housing Clearance on Churning Losses

The effect of housing clearances on the windage losses of discs was early inspected by Mann and Marston (Mann & Marston, 1961) who found that the influence of radial ($e_a/R_p = 0.0015, 0.0035, 0.0085$) and axial ($e_r/R_p = 0.01, 0.025, 0.05$) clearances was small enough to be neglected. Later, the placing of baffles or screens in the splash lubrication was reported to be effective in the reduction of windage and churning losses by Lynwander (1983). In 1983, he described in his book that trial-and-error solutions are required for placing the baffles and it should be verified by testing. In the following year, Dawson (1984) showed in his experiments of a large high speed spur gear (760 mm dia.) that up to 66% of windage losses of a spur gear in free space were reduced by using an enclosure with an axial clearance of 27 mm and a radial clearance of 15 mm. A radial clearance of 15 mm ($e_r/R_p = 0.04$) was more effective than a radial clearance of 30 mm ($e_r/R_p = 0.08$) in Dawson's windage losses experiments. However, the dimensional effect was not revealed clear because he applied just only one axial clearance and two radial clearances of 15 mm and 30 mm for windage losses. Thereafter, Changenet and Vex (Changenet & Vex, 2008) showed that smaller axial clearances decreased churning losses through their experiments with three axial clearances from one side and two radial clearances. ($e_a/R_p = 0.013, 0.052, 0.092$; $e_r/R_p = 0.065, 0.130$)

The experimental results previously described in Section 5.5 showed that the axial clearance is more significant for decreasing churning losses compared to the radial clearances. The smaller the axial clearances, the smaller the churning losses decreased regardless of the viscosity of lubricant. The experimental data for varied axial housing clearances at the same

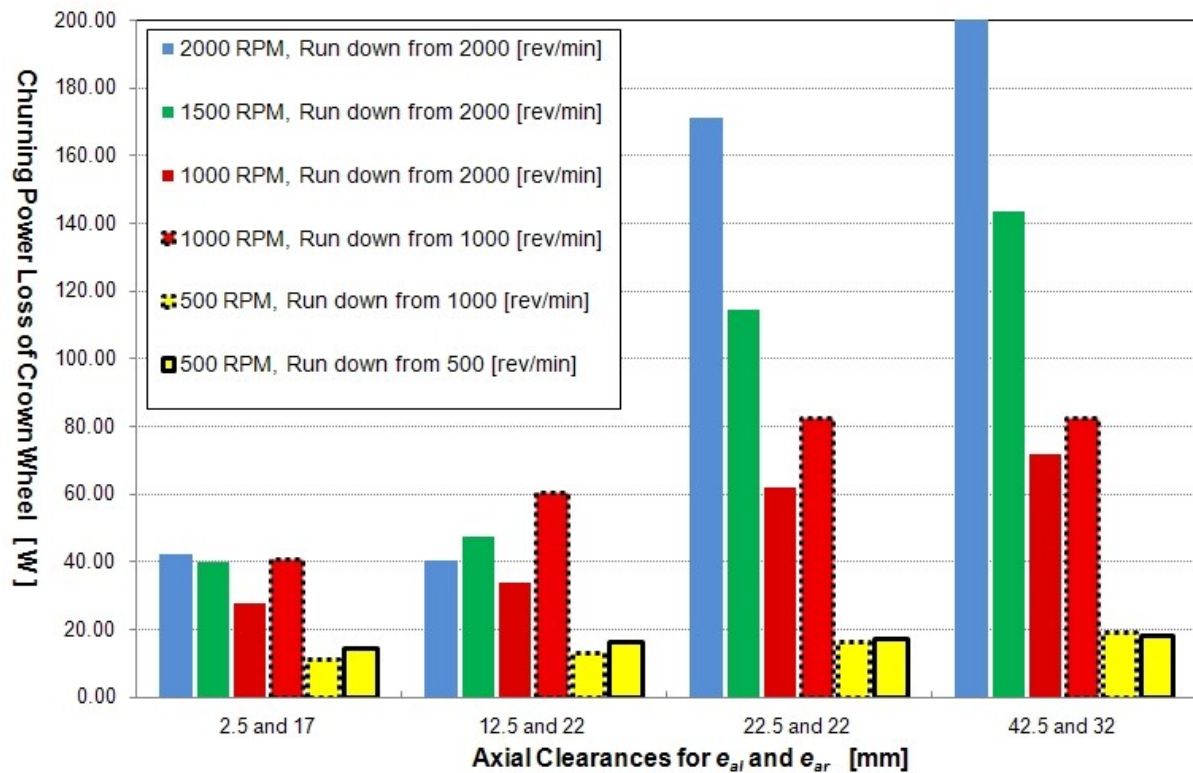


Figure 6.7 Effect of the axial clearances of a housing on the churning losses at an oil immersion depth of 64.25 mm

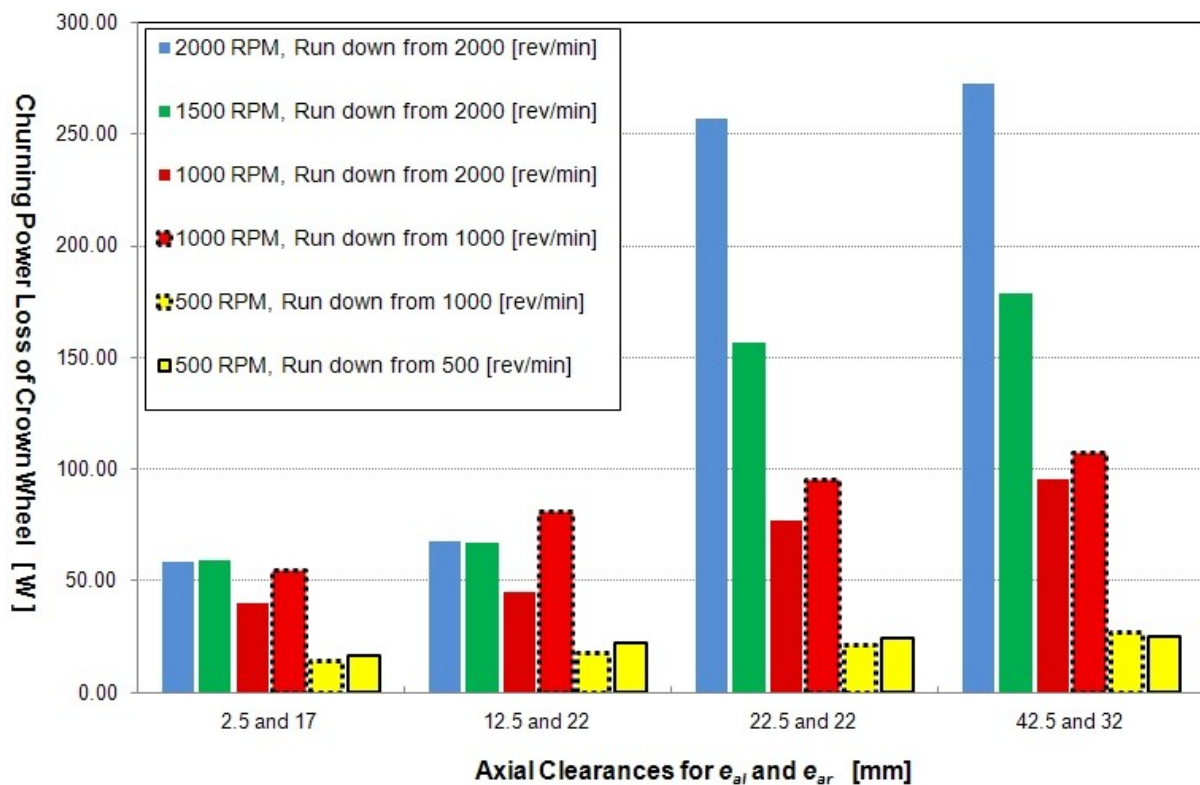


Figure 6.8 Effect of the axial clearances of a housing on the churning losses at an oil immersion depth 71 mm

immersion depth and the same radial clearance 2.0 mm, were plotted in Figures 6.7 and 6.8; which show the effect of the axial clearances on churning losses very well regardless of oil immersion depths and the effect depends on the rotational speed. At high speeds, the effect of the axial clearances was more influential and the churning losses at 2000 rev/min under the axial clearances of 12.5 and 22 mm were smaller than those at 1000 rev/min. The effect of oil volume varied according to the axial housing clearances, but it was small because the oil was supplied from the adjacent chamber to the sump through oil holes on the bottom of the axial baffles in the author's experiment. The scooping of oil from the rotating ring gear caused the different heights of free surfaces between the oil sump and the adjacent chamber, and it made the oil flow through the holes to the oil sump where the ring gear was rotating. Therefore, the effect of axial clearances was large and it reduces the churning losses.

The salient feature from the test results was that the churning torque for the axial clearances of 12.5 mm and 22 mm for e_{al} and e_{ar} respectively decreased as the speed increased from more than 1000 rev/min; this showed the strong effect of sweeping-away of the oil free surface under the small axial clearances as shown in Figure 5.6. The ratios of the smallest axial clearances e_{al} to the radius of the ring gear were 0.022, 0.068 and 0.11, which is in good agreement with the ratio of axial clearances from Changenet's experiment. Therefore, smaller axial clearances show the distinct effects to decrease the churning losses.

Unlike for the case of the axial clearances, the smallest radial clearance 2.0 mm did not show a good efficiency in the author's experiments. The lower losses for the radial clearances were from 6.4 to 10.8 mm; the churning torque for the radial clearance of 6.4 mm was lower than that for 10.8 mm for the rotational speeds below 1000 rev/min. The ratio of these radial clearances to the radius of the ring gear were as between 0.058 and 0.098; this corresponds to the values of the ratio of axial clearances for Changenet's experiment and includes the value of the radial clearance for Dawson's experiment. However, the author's test results showed

that the churning losses for the radial clearance 10.8 mm were lowest at high speeds. It seems that this could be the effect of the oil flow from the small axial clearances and the gear geometry of hypoid ring gears being different from spur gears. Therefore, it is clear that the ratio of radial clearance to the radius between 0.5 and 0.9 gives better efficiency, but the churning losses increased for the small radial clearance being no less than 1% of a diameter of ring gear.

It is interesting to note that the churning torque for an oil immersion depth of 91 mm with a radial clearance of 34 mm in Figure 5.9 was higher than for the same immersion depth with a radial clearance of 2.0 mm by 0.3 Nm. At the lower immersion depth of 71 mm, the churning torque for a radial clearance of 2.0 mm was lower than that for 34 mm, which shows the radial clearance of 2.0 mm was more effective than 34 mm. This could be caused by a larger oil volume from a larger radial clearance 34 mm, and the radial clearance 2.0 mm could cause the air sweeping-away effect in Figure 5.6. The test result for radial clearances at an oil immersion depth of 91 mm also showed that the same churning torques for radial clearances of 6.4, 2.0 and 10.8 mm was caused by the similar volume of oil.

The effect of radial clearances with a small axial clearance on the churning losses was examined through the test results from Sections 5.3 and 5.4. Figure 6.9 shows clearly the effect of radial clearances for the positions of axial clearances of 7.5 and 17 mm for e_{al} and e_{ar} on the churning losses of the crown wheel of a pitch diameter 220 mm. From the comparison between the radial clearances of 2 and 6.4 mm, the churning losses for a radial clearance of 2 mm were smaller than for 6.4 mm at most immersion depths unlike the above result. This could be caused by the small axial clearance and the different volume of oil due to the radial clearances. The oil volume is more influential on churning losses than radial clearances under the same axial clearances. Thus, it can be concluded that the influence of axial clearances is stronger than that of radial clearances and smaller axial clearances are more effective than

smaller radial clearances in the reduction of the churning losses.

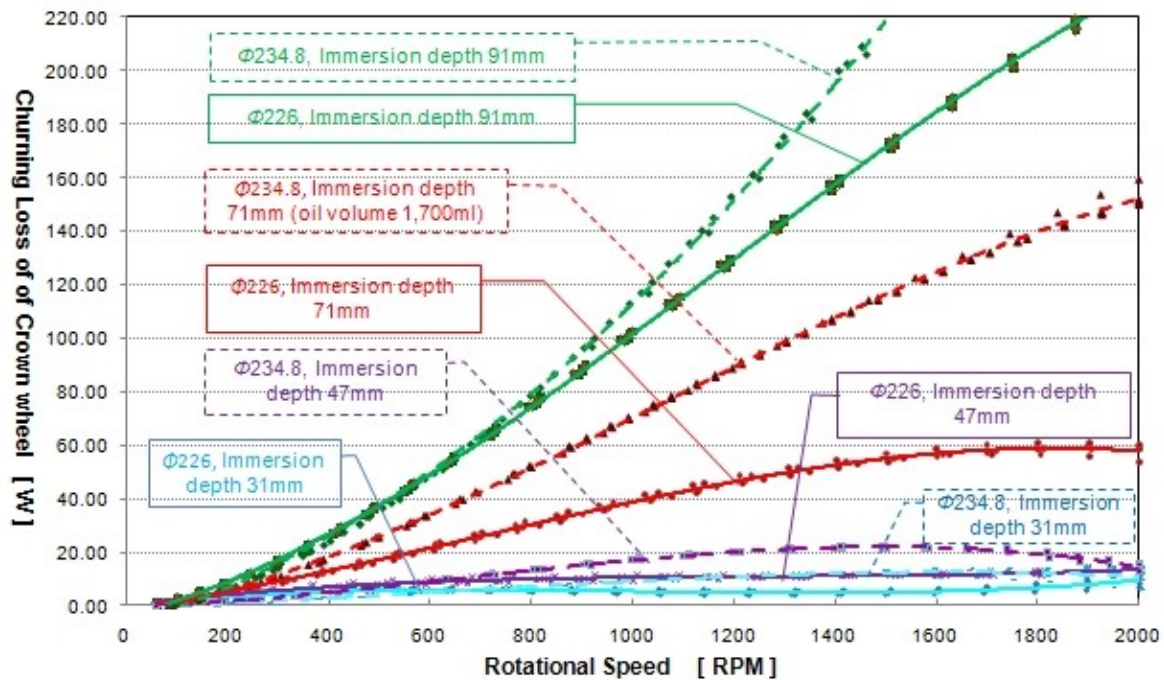


Figure 6.9 The comparison of the churning losses in the low viscosity oil between radial baffles of 226 mm dia. and 234.8 mm dia. under the position '2 by 2' of the axial baffles

6.3.6 Effects of Oil Flow on Churning Losses

The oil flow caused by the rotating hypoid gears inside the housing is thought to have an influence on the churning losses and an inspection on the oil flow seems to explain the phenomenon of the churning losses more clearly. However, the transparent glass required for this inspection was installed only on the rear side of the extension housing at the final stage of the experiments and so the author's research on the oil flow was for a limited set of conditions. As previously shown in Figure 5.2, bubbles occurred in the low viscosity oil. This indicates that the oil flow inside the housing has a close relationship with the aeration because two phases of air and oil exists while the gear submerged in the oil rotates. Accordingly, oil flow may be similar to the air flow in the vicinity of the rotating gear. As earlier described in

Section 2.4.2, Dawson (1984) stated that the air flow pattern of a single spur and helical gears was dependent on the gear geometry. On the periphery of a single spur gear, the air is induced axially and flows out tangentially and radially, but on that of a single helical gear, an air flows tangentially/radially and tangentially/axially outwards for a radial and axial inward flow. The gear teeth act as a centrifugal fan drawing and ejecting air tangentially and this air flow may be compared to the lubricant flow of the hypoid ring gear. The hypoid spiral angles of the ring gear may make the oil flow stronger circumferentially from the oil sump and so that it was observed that the lubricant flowed along the circumferential wall of the housing at the low speeds of the rotating crown wheel alone. At the high speeds, the lubricant slung off by the spiral angle of gear teeth scattered in oil drops. The larger oil drops centrifuged outwards to the wall of housing drained into the base of the oil sump. The overall oil flow was dominated by pumping from the crown wheel so it was observed that the oil was circulated from the teeth side of crown wheel to its rear side in the case of small axial clearances, as shown in Figure 5.2 (c). Therefore, the higher churning losses for the bigger diameter of crown wheel could arise partly from the greater pumping action of the longer spiral angle teeth.

The air flow pattern in meshing spur gears was previously shown in Figure 6.4 and Olver (2002) indicated that the flow of air around the meshing spur gears that is not only a source of losses in itself, but can also hinder the jet lubrication and result in starvation. Moreover, the radial flow of air around the gear side is more significant than the axial flow except in near the gear mesh where axial flow is predominant. In the meshing hypoid gears, it seems likely that the presence of a hypoid pinion gear could block the radial oil and increase the churning losses more than the arithmetic sum of a pair of gears.

The salient feature of the oil flow is that the air draught at high speeds could sweep the free surface of oil away and it leads to a drop in the oil level of sump as shown in Figure 5.6. As a result, the lower oil immersion depth could reduce the churning losses. This phenomenon

seems to be more prominent in the case of a higher viscosity of lubricant, higher rotating speeds, low oil immersion depths, and smaller radial clearances with small axial clearances.

Foam was mainly observed on the surface of the low viscosity oil after the run-down tests because the low viscosity oil does not include an anti-foam additive. Hayward (Hayward, 1962) distinguished bubble from foam, and defined bubbly oil as oil in which discrete bubbles of air or gas are entrained and are separated from each other by relatively thick films of oil, whereas the air bubbles in foam are separated by only thin films of oil or may be interconnected. Bubbles were found in both the low viscosity oil and the high viscosity oil (synthetic oil 75W90) after the run-down tests; the colour of the lubricants became opaque due to many bubbles. However, foams were only found on the surface of the low viscosity oil, which made the oil level rise. These foams and bubbles are known to have an influence on the viscosity of the oil but it is not satisfactorily established (Stepina & Vesely, 1992). However, foam seems to have no influence on churning losses based on the author's test results in the temperature range of 21~28°C for the extension housing with axial baffles, since the churning losses measured before and after the occurrence of bubbles and foam were the same even if the foam made the oil level higher. Similar experimental results were observed by Luke and Olver (1999), who found that the effect of foaming was negligible on an oil above 30°C and water in a spur gear with dip lubrication. In their experiments, considerable foaming was built up during the low-temperature tests with oil, which resulted in a change of immersion conditions and affected the churning losses.

In addition, research on the oil flow inside a housing has been made at Cranfield University by using the completely transparent production axle housing. Professor A. Olver and the author visited Cranfield University and observed the oil flow in the transparent production axle housing. The observations were described by A. Olver as follows.

1. The overall circulation is dominated by pumping from the crown wheel; oil is lifted circumferentially from the sump at the rear of the assembly and is expelled axially from the tip clearance constriction, creating a prominent vortex close to the rear housing surface.
2. At higher speeds, appreciably more oil is thrown towards the toothed side of the crown wheel (starboard) than to the bolted side (port).
3. The feed to the pinion bearings is copiously supplied with oil at low speeds but appears to become ‘choked’ with air at higher speeds perhaps leading to a reduced supply.
4. The return feed for the forward bearing assembly exits very close to the pinion.
5. As the speed increase, the level in the sump drops leading to a lower effective immersion depth on the crown wheel.
6. In contrast, the pinion is fed with an increasing supply at higher speeds (from the crown wheel and from the pinion bearing return).
7. The return feed channel from both output shaft bearings was apparently empty at all speeds.

6.3.7 Bearing Losses in the Experiments

Bearing losses were calculated from the test results by subtracting the values obtained for the run-down tests of using only input shaft components from the no-oil torque in the absence of oil. From Equation 4.4, the bearing drag torque was calculated as shown below.

$$T_{bearing} = T_{no-oil} - T_{Input\ shaft} = (T_{gear} + T_{bearing} + T_{seal}) - (T_{gear} + T_{seal}) \quad (6.1)$$

Figure 6.10 shows how the lubricant viscosity affects the bearing drag torque and that the difference of the bearing drag torques between the lubricants increased with the rotational speeds. In this figure, the logarithmic increase of the bearing torque may indicate that some churning losses took place. Accordingly, it seems likely that this bearing drag torque may

include mainly bearing friction torque and some bearing churning torque even if the run-down test was performed without any load and no oil was supplied to the rotating bearings from the sump. Two kinds of phenomena could be found. Firstly, the bearing losses are significant as

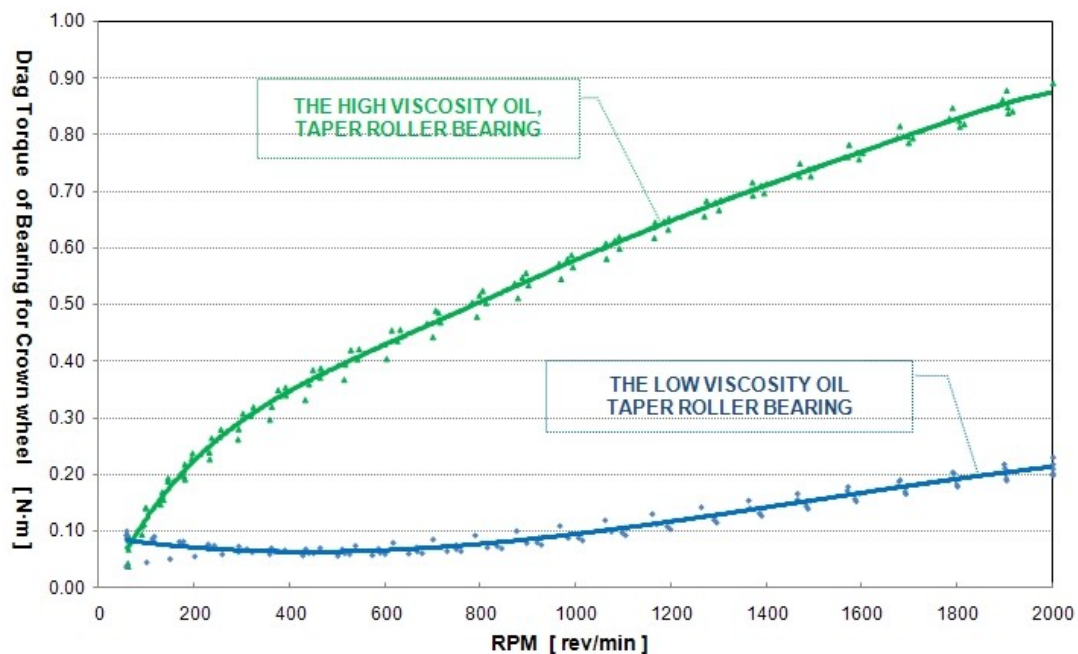


Figure 6.10 Effect of lubricants on the bearing drag torque

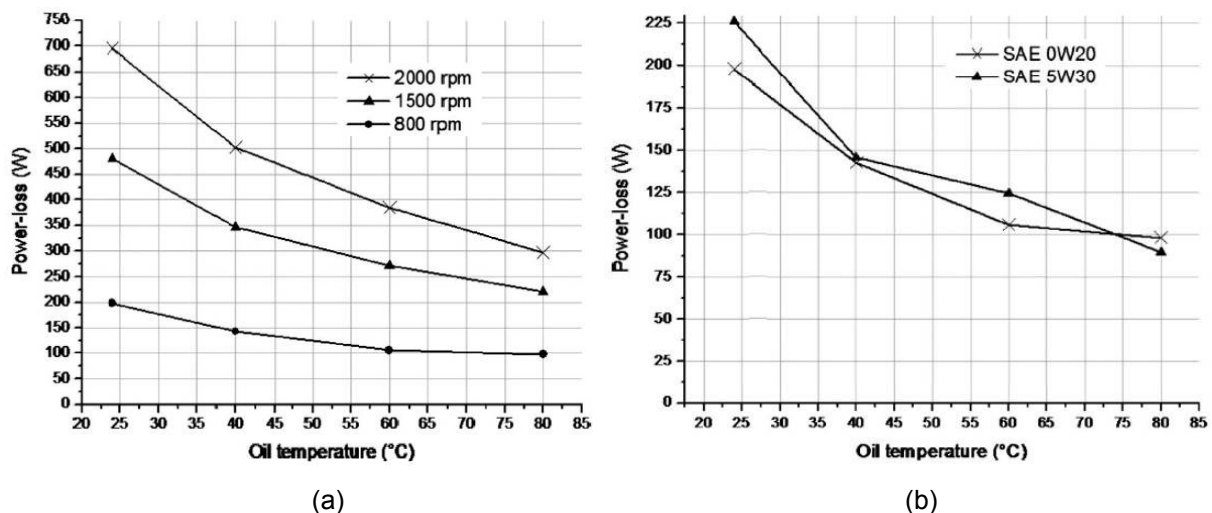


Figure 6.11 Bearing friction power loss (a) SAE0W20 at different speeds (b) comparison between SAE0W20 and SAE5W30 at 800 rev/min (Mufti & Priest, 2009)

well as churning losses and gear friction losses. Secondly, the drag losses for bearings immersed or lubricated in the high viscosity oil are bigger than for bearings in the low viscosity oil. Mufti and Martin (Mufti & Priest, 2009) recently and experimentally proved that the bearing friction loss is one of the main losses in an engine and the bearing friction power loss increases with the viscosity of the lubricant. Figure 6.11 shows that the bearing losses increased with the speed, and oil SAE 5W30, having a higher viscosity of $57.4 \text{ mm}^2/\text{s}$ at 40°C , gave a higher bearing loss than SAE 0W20 oil having a viscosity of $41.37 \text{ mm}^2/\text{s}$ at 40°C . In addition, there is a decrease in bearing power loss as the lubricant temperature increases at any given speed. Therefore, the bearing losses are also significant in gear boxes and the bearing losses are largely dependent on bearing type, lubricant viscosity and rotating speeds.

6.4 Analysis of Experimental Data for Churning Losses

The analysis has been performed mainly using dimensional analysis.

6.4.1 Dimensional Analysis

The variables considered in the dimensional analysis are shown in Table 6.1

Table 6.1 The parameters for dimensional analysis

Geometric parameters	Fluid dynamic parameters
Diameter of ring gear D_p (Radius R_p)	Oil kinematic viscosity ν_{lub}
Oil immersion depth h_{lub}	Rotational speed ω
Oil volume V_{lub}	Gravitational acceleration g
Axial clearances e_{al} , e_{ar}	Air kinematic viscosity ν_{air}
Radial clearance e_r	
Width of the ring gear b	
Immersed surface area of the crown wheel S_m	

The churning torque T_{ch} can be expressed as a function of the above parameters.

$$T_{ch} = f(D_p, e_{al}, e_{ar}, e_r, b, S_m, h_{lub}, V_{lub}, v_{lub}, \omega, g, v_{air}) \quad (6.2)$$

If the drag force which acts on the body by the fluid flowing past it, is considered, a dimensionless coefficient C_D (Schlichting & Gersten, 2000) for the drag is given by

$$C_D = D / (0.5 \rho V^2 S) \quad (6.3)$$

where, D is drag, V is velocity and S is characteristic surface of the body. Accordingly, the dimensionless churning torque coefficient can be defined from the Equation 6.3 as

$$C_m = T_{ch} / (0.5 \rho_{lub} \omega^2 R_p^2 b S_m) \quad (6.4)$$

where, the width b is used instead of R_p in this equation so that C_m is not too small and b is a considered variable on churning losses. According to the Π theorem from the dimensional analysis, the dimensionless churning torque is described by

$$C_m \equiv \psi_1 \left(\frac{e_{al}}{R_p} \right)^{\psi_2} \left(\frac{e_{ar}}{R_p} \right)^{\psi_3} \left(\frac{e_r}{R_p} \right)^{\psi_4} \left(\frac{h_{lub}}{R_p} \right)^{\psi_5} \left(\frac{V_{lub}}{R_p^3} \right)^{\psi_6} Re^{\psi_7} Fr^{\psi_8} \left(\frac{v_{lub}}{v_{air}} \right)^{\psi_9} \quad (6.5)$$

Reynolds number, Re is defined as the ratio of inertia force to viscous force by

$$Re = \frac{\omega(D_P/2)^2}{v_{lub}} \quad (6.6)$$

Froude number, Fr is defined as the ratio of the inertia force to gravitational force as shown below.

$$Fr = \frac{\omega^2(D_P/2)}{g} \quad (6.7)$$

6.4.2 Dimensionless Numbers

Through the previous discussion of the experimental results, the churning losses for the hypoid crown wheel can be influenced mainly by three types of forces: viscous force, inertia force and gravitational force of the lubricant. These forces could be expressed in the form of the dimensionless numbers, Reynolds number (Re) and Froude number (Fr).

6.4.2.1 Reynolds Number

The test results for the rotating ring gear submerged in the low viscosity oil and the high viscosity oil with an immersion depth of 64.25 mm were used in Equation 6.3 to calculate the ψ_7 exponent. All other test conditions remained the same except the viscosity of lubricant. Thus, the ψ_7 exponent can be derived from the two lubricant test results at the same rotational speed and expressed in the term of the dimensionless churning torques and the oil viscosities or the two viscosity oils. The subscript numbers indicate the high viscosity oil and the low viscosity oil respectively.

$$\psi_7 = \frac{\ln(C_{m,1}/C_{m,2})}{\ln(Re_1/Re_2)} = \frac{\ln(C_{m,1}/C_{m,2})}{\ln(\nu_2/\nu_1)} \quad (6.8)$$

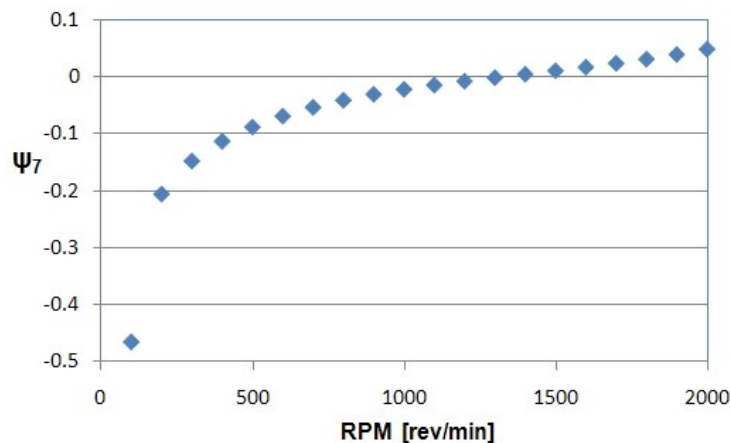


Figure 6.12 Graph of RPM versus the exponent of Re for the crown wheel alone

where 1 and 2 indicate the different viscosity oil. The ψ_7 exponent varied with the rotational speed and was plotted as shown in Figure 6.12. From the graph, the value, ψ_7 is zero at approximately 1250 rev/min

and can be divided into high speed and low speed regimes. In the lowest speed regime of the experimental data, the exponent ψ_7 ranges mostly between -0.1 to -0.5 and could be -0.4 at 100 rev/rpm. Thus, the relation between C_m and ν is

$$C_m \propto Re^{-0.4} \propto \nu^{0.4} \quad (6.9)$$

In the high-speed regime above 1250 rev/min, the ψ_7 exponent could be 0.05 at 2000 rev/min.

$$C_m \propto Re^{0.05} \propto \nu^{-0.05} \quad (6.10)$$

The value of Re^{ψ_7} at around 1250 rev/min is 1.0 and the lubricant viscosity has no effect. Moreover, the influence of lubricant viscosity on the churning losses is very small or does not exist at the speeds above 1000 rev/min, and just exists at the low speeds. This result is in good agreement with Changenet's formula that showed the exponent of Reynolds number is zero at high speeds as shown in Table 2.7.

Table 6.2 The relation between the fluid churning torque (T_{ch}) and the lubricant viscosity (ν)

Regime	Von Karman	Boness	Terekhov	Changenet
Low speed	$T_{ch} \propto \nu^{0.5}$	$T_{ch} \propto \nu^1$	$T_{ch} \propto \nu^{0.6}$	$T_{ch} \propto \nu^{0.21}$
Transition	-	$T_{ch} \propto \nu^{-1/3}$	$T_{ch} \propto \nu^{0.6}$	Interpolation
High speed	$T_{ch} \propto \nu^{0.2}$	$T_{ch} \propto \nu^2$	$T_{ch} \propto \nu^{0.3}$	$T_{ch} \propto \nu^0$

Table 6.2 shows the relationship between the churning torque and the lubricant viscosity in the formulae presented so far. In these formulae, the torque caused by the fluid was described as T_{ch} . These formulae in Table 6.2 were described previously in Sections 2.4 and 2.5 of the

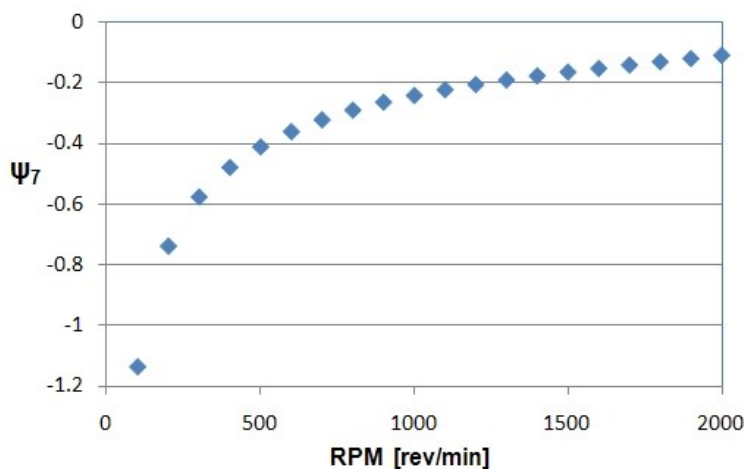


Figure 6.13 Graph of RPM versus the exponent of Re for a pair of the hypoid gears

literature review. This table explains well the reason why the predicted torque from Boness's formula increases with a great power of the speed at high speeds.

The ψ_7 exponent of Reynolds number for a pair

of the hypoid gears in the production axle housing was also calculated in the same way and plotted in Figure 6.13. From the graph, the value of ψ_7 rose sharply up to 500rev/min and converges to zero at higher speeds. This means that the viscous force is weak at higher speeds. However, the viscosity force exists over the full range of speeds when compared with Figure 6.12. This could explain well that the churning torque for a pair of the hypoid gears submerged in the high viscosity oil is higher than that in the low viscosity oil as shown in Figure 6.6. In the lowest speed regime of the experimental data, the ψ_7 exponent can be -1.0 at 100 rev/min, and seems to range mostly between -0.4 and -1.3 at below 500 rev/min.

6.4.2.2 Froude Number

The Froude number is interpreted physically as the ratio between the mean flow velocity and the speed of an elementary gravity (surface or disturbance) wave travelling over the fluid surface. The ψ_8 exponent of the Froude number can be obtained by changing the speed alone in the same viscosity lubricant from Equation 6.4 after the calculation of the Reynolds numbers for these speeds as shown below.

$$\psi_8 = \frac{\ln \left[\frac{C_{m,1}}{C_{m,2}} \left(\frac{Re_2}{Re_1} \right)^{\psi_7} \right]}{\ln(Fr_1/Fr_2)} = \frac{\ln \left[\frac{C_{m,1}}{C_{m,2}} \left(\frac{\omega_2}{\omega_1} \right)^{\psi_7} \right]}{\ln(\omega_1/\omega_2)} \quad (6.11)$$

where 1 and 2 indicate the different rotational speed. The exponent of the Froude numbers for the low viscosity oil were calculated for the rotating crown wheel submerged in the low viscosity oil and were plotted in Figure 6.14. The oil immersion depth was 64.25mm. From the graph, the value of the exponent falls steeply at speeds below 500 rev/min. Above 500 rev/min the exponent of the Froude number remains at approximately -0.7 for these speeds. For the high viscosity oil, the absolute values of the exponent are bigger than for the low viscosity oil and is approximately -0.8 at speeds above 500rev/min. Figure 6.15 shows that

the exponents of the Froude number decreases with the rotational speed but falls below zero, thus the absolute value increases with these high speeds. Above 200 rev/min the value for the high viscosity oil is lower at the same speeds than for the low viscosity oil so that the inertia force in the low viscosity oil is stronger than in the high viscosity oil. The inertia force to gravitational force has a relation with the free surface flow of the lubricant caused by an immersed source in the housing.

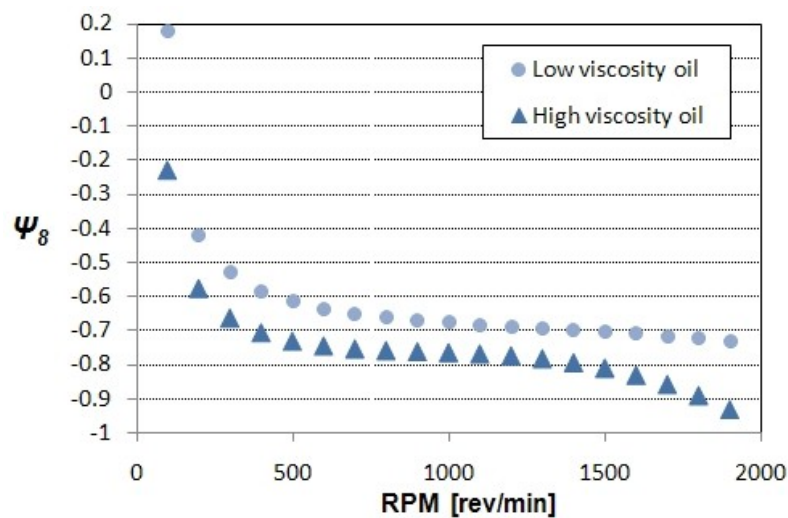


Figure 6.14 Graph for RPM versus the exponent of Fr for the crown wheel submerged with an immersion depth of 64.25 mm in the low and high viscosity oils

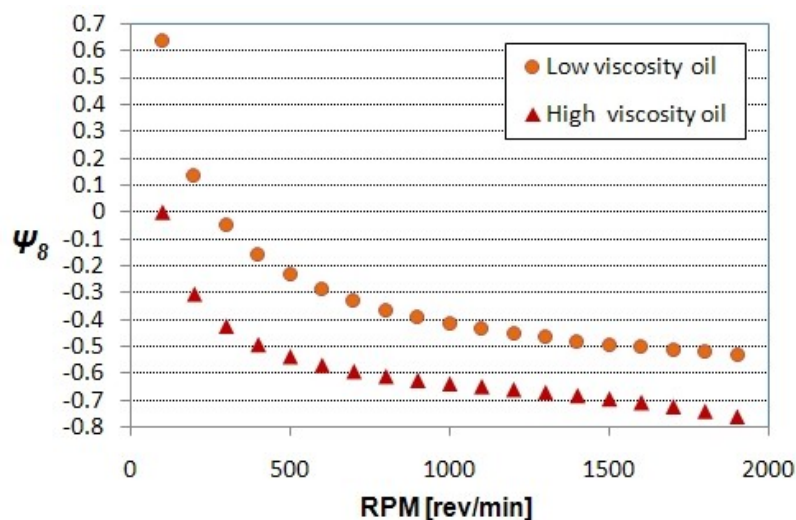


Figure 6.15 Graph for RPM versus the exponent of Fr for a pair of meshed hypoid gear with an immersion depth of 64.25 mm in the low and high viscosity oils

The free surface was studied by Vanden-Broeck (Vanden-Broeck, 1998), who considered a

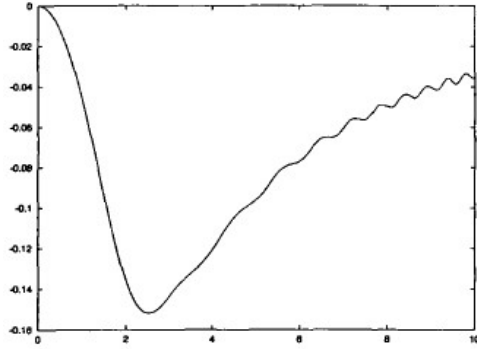


Figure 6.16 Free surface profile presented by Vanden-Broeck (Vanden-Broeck, 1998)

free surface flow due to a source submerged in a fluid of infinite depth. He calculated the free surface flow profiles

as shown in Figure 6.16 and presented that the larger values of Fr^* (Changenet & Vexel, 2008) has a train of waves on the free-surface.

$$Fr^* = \frac{v}{\sqrt{gy}} \quad (6.12)$$

where v is the mean flow velocity, y is the height of the flow and g is the acceleration of gravity field. Also, he showed that the wavelength of waves decrease with distance from the source and viscosity will ultimately damp the train of waves in the far field. These are in good agreement with the author's test results. The churning torque decreased with the lower oil depth to the bottom caused by the radial baffles in Section 5.4. In addition, the churning torque of the crown wheel alone submerged in the high viscosity oil was no more than for the low viscosity oil; which is due to the low immersion depth caused by the sweep-away effect as shown in Figure 5.6.

Another way to derive the exponent ψ_8 of Froude is to use the exponent of the rotational speed in Equation 6.3. The dimensionless torque C_m is proportional to T_{ch}/ω^2 so that T_{ch} is finally proportional to $\omega^{2(1+\psi_8)}$.

$$C_m \propto Fr^{\psi_8} \quad (6.13)$$

$$\frac{T_{ch}}{\omega^2} \propto C_m \propto (\omega^2)^{\psi_8} \quad (6.14)$$

$$T_{ch} \propto \omega^{2+2\psi_8} = \omega^{2(1+\psi_8)} \quad (6.15)$$

If the logarithm is taken of the both sides, the exponent ψ_8 of Froude number can be obtained

easily from the log graph of T_{ch} versus ω .

$$\log T_{ch} \propto \log \omega^{2(1+\psi_8)} = 2(1 + \psi_8) \log \omega \quad (6.16)$$

Table 6.3 The relation between the churning torque and the rotational speed (the exponent ψ_8)

Regime		Von Karman	Boness	Terekhov	Changenet
Low speed	ω	$T_{ch} \propto \omega^{1.5}$	$T_{ch} \propto \omega^1$	$T_{ch} \propto \omega^{0.9}$	$T_{ch} \propto \omega^{0.59}$
	ψ_8	-0.25	-0.5	-0.55	-0.705
Transition	ω	-	$T_{ch} \propto \omega^{7/3}$	$T_{ch} \propto \omega^{0.9}$	Interpolation
	ψ_8	-	1/6	-0.55	
High speed	ω	$T_{ch} \propto \omega^{1.8}$	$T_{ch} \propto \omega^0$	$T_{ch} \propto$ from $\omega^{-2.188}$ to $\omega^{0.735}$	$T_{ch} \propto \omega^{0.24}$
	ψ_8	-0.1	-1	from -0.6325 to -2.094	-0.88

Table 6.3 shows that the relation between the churning torque and the rotational speed in the the formulae presented in Chapter 2 and the exponent of the Froude number calculated from Equation 6.15. These values of the exponent ψ_8 are mostly within the range of the author's experimental values shown in Figure 6.14. The simple theoretical models for the churning including Von Karman's flow are shown in Figure 6.17 and the churning torque graphs of the author's experimental results were plotted on log scales as shown in Figure 6.18.

Couette flow is the laminar flow of a viscous fluid in the space between two parallel plates and Taylor-Couette flow is the circular flow of a viscous fluid confined in the gap between two rotating cylinders as shown in Figure 6.17(a) (Schlichting & Gersten, 2000; Wikipedia). The shear stress of Couette flow is constant. When the Reynolds number increases, the flow within the annulus transforms from Taylor-Couette flow finally to the turbulent Taylor vortex flow and the dimensionless torque becomes to be proportional to $Re^{1.5}$ rather than $Re^{1.0}$ (Lim & Tan, 2004).

The ψ_8 exponents for the crown wheel in the author's tests were the same as that for Taylor-Couette flow, -0.5 at the range between 200 and 300 rev/min as shown in Figure 6.14.

The absolute value of the exponent of the Froude number increased with the rotational speed and the higher viscosity as shown in Figure 6.18. It shows that inertia force and gravitational force are more influential at higher speeds and in the higher viscosity oil flow.

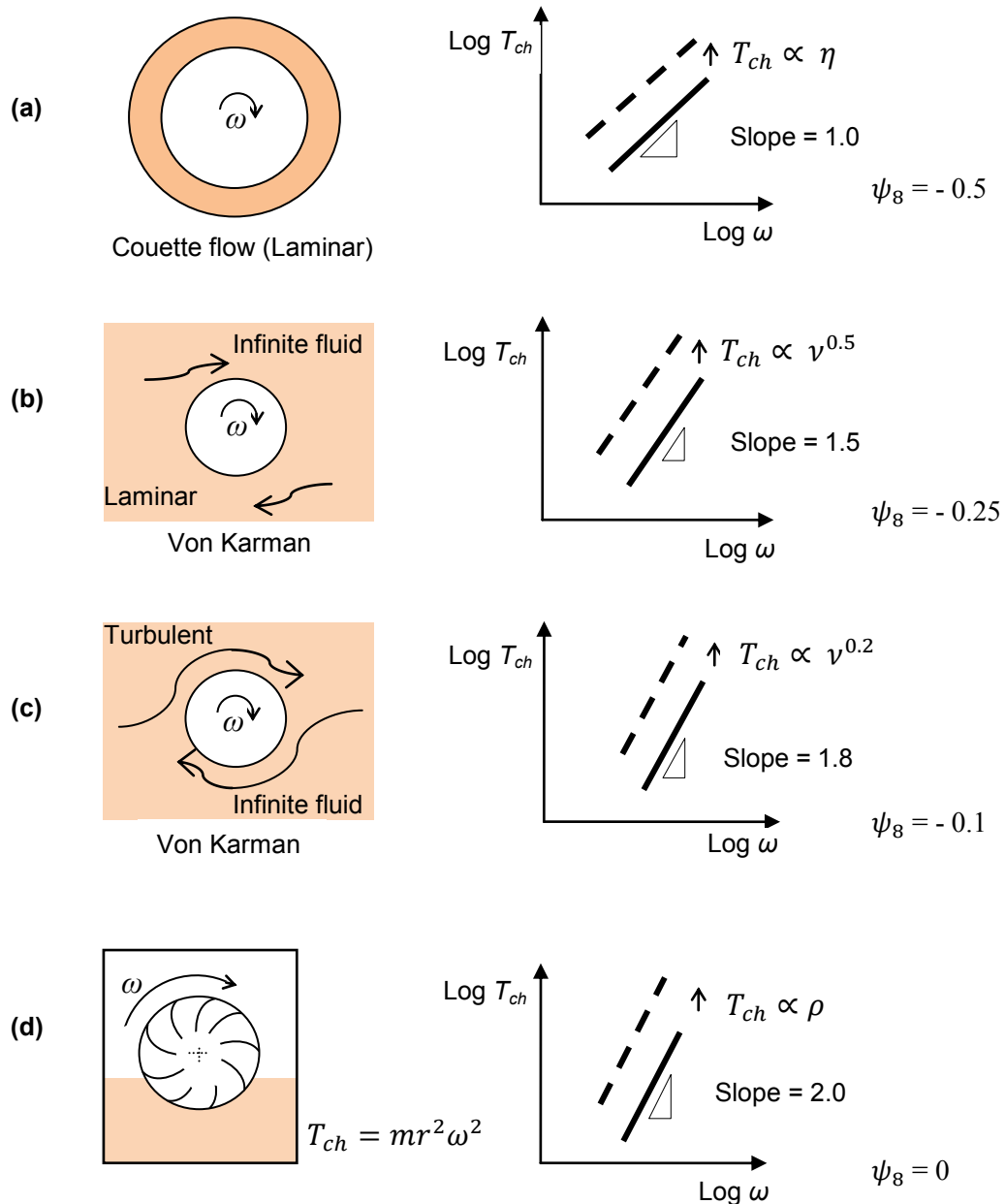


Figure 6.17 Relation between T_{ch} and ω in the simple theoretic model (a) Taylor-Couette flow (laminar) (b) Laminar flow for infinite fluid (c) Turbulent flow for infinite fluid (d) the flow by the rotating impeller submerged in water

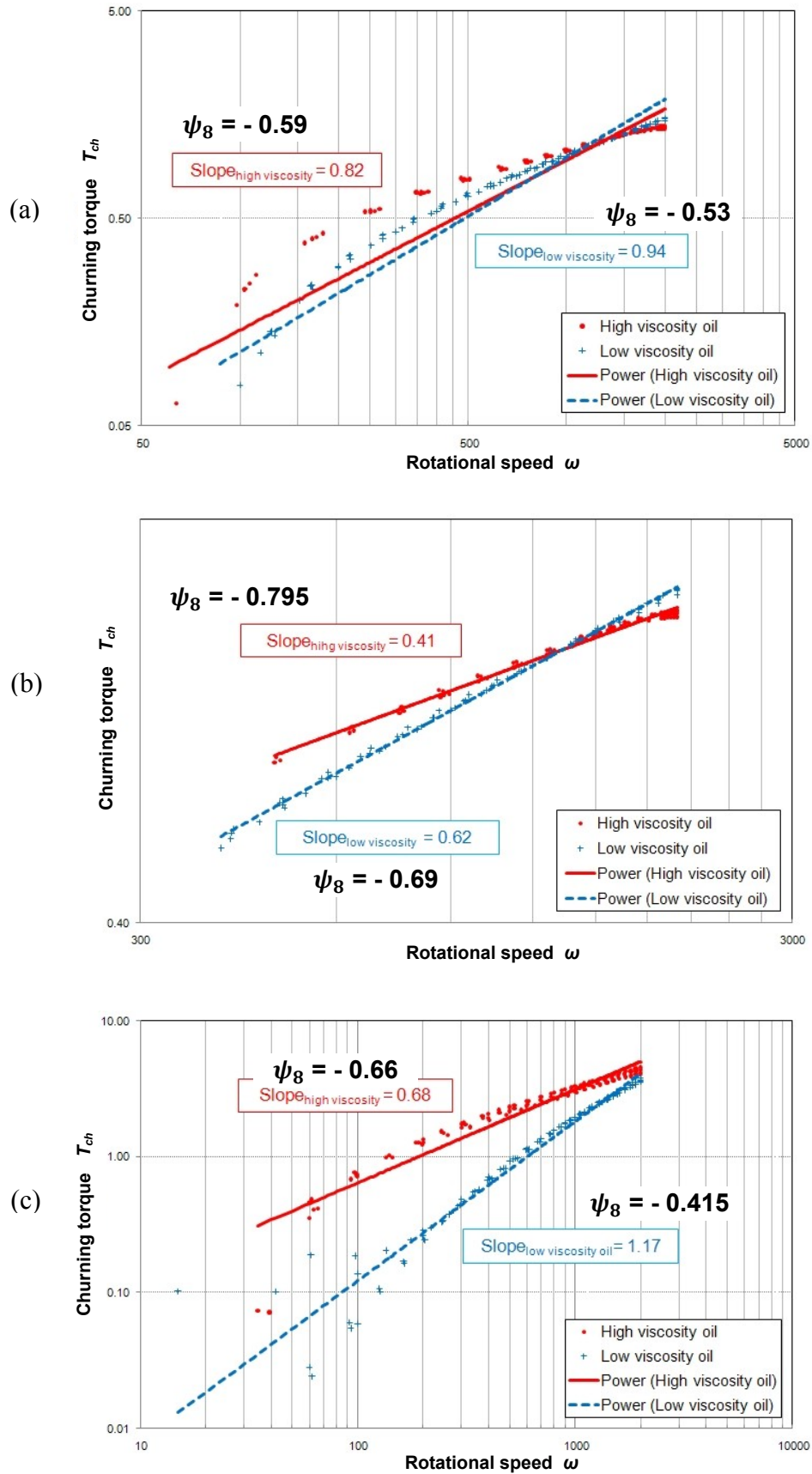


Figure 6.18 Graphs on log scales for ω versus T_{ch} (a) Crown wheel alone (b) Crown wheel alone at above 400 rev/min (c) a pair of mesh hypoid gear

6.4.3 Empirical formula from Statistical treatment of Dimensional Analysis

The statistical method was employed to determine the coefficient and the exponents of Equation 6.4. A freeware statistic program ‘R’ was used and the exponents were obtained through a nonlinear least square regression method.

6.4.3.1 Empirical formula for the production axle

The experimental data for the crown wheel alone in the production axle housing were used to obtain the exponents of Equation 6.4. The variables of oil immersion depth, oil volume, Reynolds and Froude numbers were considered and the value of the root mean square error (RMSE) was good. RMSE shows the discrepancy between the measured and calculated values as shown in Figure 6.19(a). The values at x-axis indicate the measured values of samples and those at y-axis the calculated values from the formula presented in this section.

$$C_m = 2.18599 \left(\frac{h_{lub}}{R_p^3} \right)^{0.14772} \left(\frac{V_{lub}}{R_p^3} \right)^{-0.1988} Re^{-0.25015} Fr^{-0.52985} \quad (6.17)$$

Equation 6.17 as shown below was obtained considering the ratio of the lubricant viscosity to the air viscosity when compared to Equation 6.16.

$$C_m = 10.0 \left(\frac{h_{lub}}{R_p^3} \right)^{0.1786} \left(\frac{V_{lub}}{R_p^3} \right)^{-0.2195} Re^{-0.4169} Fr^{-0.4482} \left(\frac{\nu_{lub}}{\nu_{air}} \right)^{-0.1777} \quad (6.18)$$

The RSME of Equation 6.17 was a little improved over than Equation 6.16 as shown in Figure 6.19. The values at low speeds are in good agreement, but those at high speeds are not; which suggests that the formula does not include the exact volume of air induced into the oil.

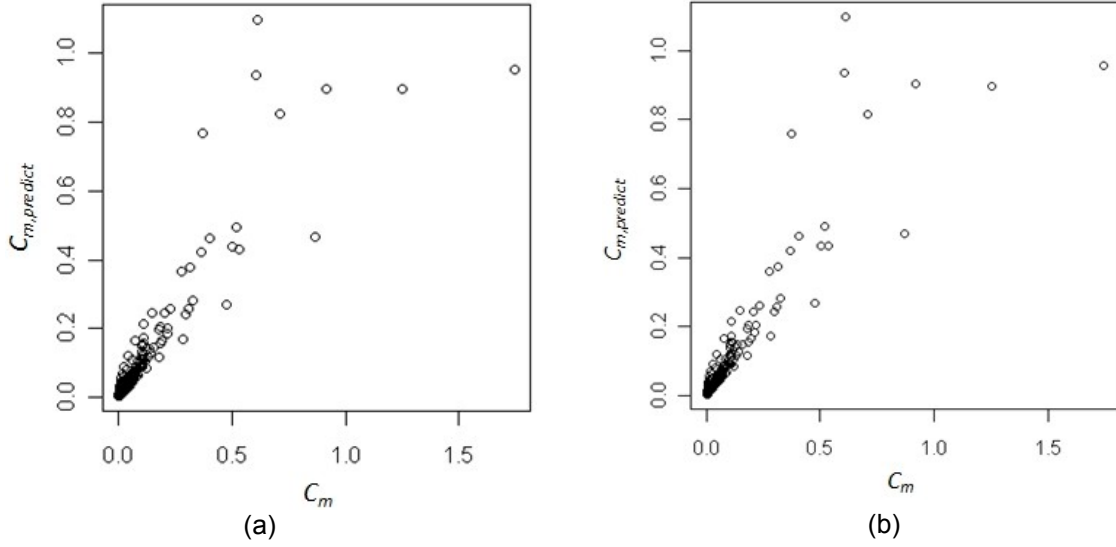


Figure 6.19 Comparison between the observed and predicted values of the dimensionless torque (a) RMSE= 0.06638176 for Equation 6.16 (b) RMSE= 0.0659472 for Equation 6.17

6.4.3.2 Empirical equation for the extension housing

The axial and radial clearances were considered to derive the empirical formulae of Equations 6.18 and 6.19. The root mean squared error (RMSE) showing the discrepancy between the measured and calculated values is shown in Figure 6.20(a).

$$C_m = 4.94 \times 10^{-7.0} \left(\frac{e_{al}}{R_p} \right)^{1.31} \left(\frac{e_{ar}}{R_p} \right)^{-8.92} \left(\frac{e_r}{R_p} \right)^{-0.151} \left(\frac{h_{lub}}{R_p} \right)^{1.38} Re^{-0.792} Fr^{-0.593} \left(\frac{\nu_{lub}}{\nu_{air}} \right)^{-0.156} \quad (6.19)$$

Although Equation 6.18 included the ratio of the lubricant viscosity to the air viscosity, the root mean squared error was bigger than Equation 6.19. Thus, the air viscosity on churning losses could not be influential. This could be because the formula does not include the exact volume of air induced into the oil in the tests, since it was difficult to measure the exact quantity of air. This seems to make the discrepancy between the observed and predicted

values bigger for the higher range of C_m .

$$C_m = 5.19 \times 10^{-7.0} \left(\frac{e_{al}}{R_p} \right)^{1.31} \left(\frac{e_{ar}}{R_p} \right)^{-8.92} \left(\frac{e_r}{R_p} \right)^{-1.51} \left(\frac{h_{lub}}{R_p} \right)^{-1.38} Re^{-0.792} Fr^{-0.593} \quad (6.20)$$

The discrepancy between the measured and calculated values and the RSME of Equation 6.20 was a little improved over than Equation 6.19 as shown in Figure 6.20.

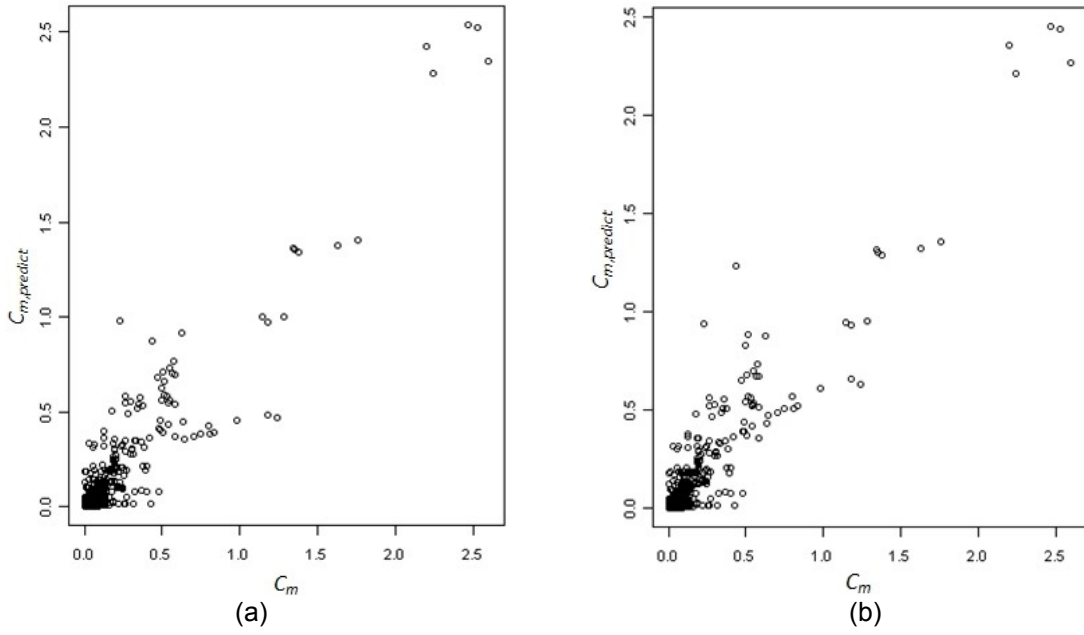


Figure 6.20 Comparison between the observed and predicted values of the dimensionless torque (a) RSME=0.04443777 for Equation 6.18 (b) RSME=0.04278817 for Equation 6.19

6.5 The New Churning Model

The following model can explain the effect of the change of the oil immersion depth on the pumping churning losses for the hypoid gears in the axle housing.

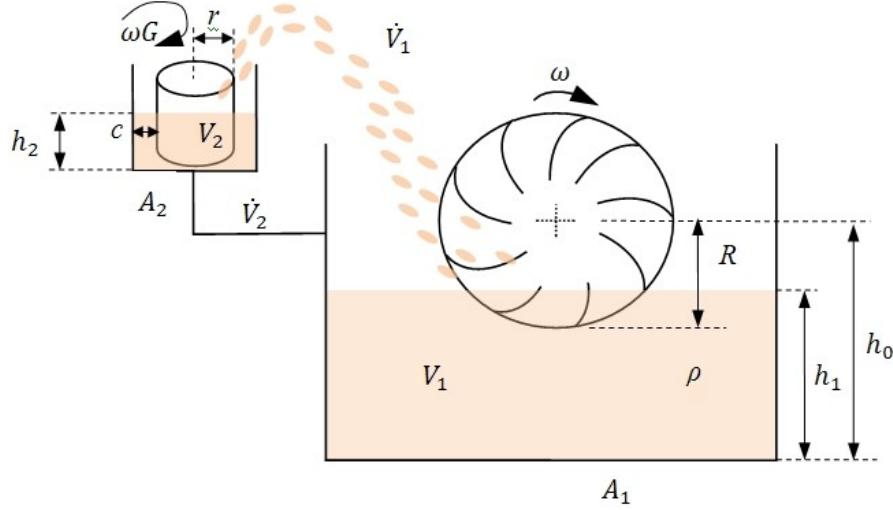


Figure 6.21 Schematic diagram of the new churning model of the crown wheel

Figure 6.21 shows the schematic diagram of new churning loss model for the hypoid axle and this model explains well the reduction of the oil immersion depth caused by the sweeping-away effect for the high viscosity oil as shown in Figure 5.6. The shape of the housing for this new model is assumed a rectangular parallelepiped and the housing is decomposed into a housing for the crown wheel and the other for the bearings of the pinion. A_1 and A_2 are the bottom area of the housings, h_1 and h_2 the oil depth to the bottom of the housings, V_1 and V_2 the oil volumes, R and r the radius of the crown wheel and the bearing, $\dot{V}_1$ the oil pumping volumetric flow rate and $\dot{V}_2$ the oil volumetric flow rate through a capillary tube.

$$V_1 = A_1 h_1 \quad (6.21)$$

$$V_2 = A_2 h_2 \quad (6.22)$$

The total oil volume V_o is the sum of the oil volumes V_1 and V_2 .

$$V_o = V_1 + V_2 = A_1 h_1 + A_2 h_2 \quad (6.23)$$

$$h_1 = \frac{V_o - A_2 h_2}{A_1} \quad (6.24)$$

The immersion depth h_1 is dependent on the total volume. If the pumping oil volumetric flow rate $\dot{V}_1$ from the rotating crown wheel is the same as the volumetric flow rate $\dot{V}_2$ sent through the capillary tube,

$$\dot{V}_1 = [R - (h_0 - h_1)]bR\omega = \dot{V}_2 = \frac{C_2 \rho g h_2}{\eta} \quad (6.25)$$

where h_0 is the distance between the centre of the crown wheel and the bottom of the oil sump, η is the dynamic viscosity of the oil, ρ is the oil density and b the width of the crown wheel.

Thus, h_2 can be driven as

$$h_2 = \frac{bR\eta}{C_2 \rho g} (R - h_0 + h_1)\omega \quad (6.26)$$

$$h_2 \frac{C_2 \rho g}{bR\eta} = (R - h_0)\omega + \frac{V_o - A_2 h_2}{A_1} \omega \quad (6.27)$$

$$h_2 = [(R - h_0) + \frac{V_o}{A_1}]\omega / \left(\frac{C_2 \rho g}{bR\eta} + \frac{A_2 \omega}{A_1} \right) \quad (6.28)$$

where C_2 is the constant for a real capillary tube flow and g is the gravity acceleration. h_2 is dependent on the rotational speed and the immersion depth h_1 . Thereafter, the pumping torque caused by crown wheel can be derived from that force F_x , which follows a circumferential path x . This sum of the forces acted on the crown wheel is the rate of change of momentum $\Delta \dot{M}$ due to oil being accelerated.

$$\sum F_x = \Delta \dot{M} = \dot{m} \Delta v = \rho \dot{V} \Delta v \quad (6.29)$$

The change of the velocity is the product of the radius and the rotational speed of the crown wheel. Therefore, the pumping torque and the power of the crown wheel is derived as

$$T_{wheel} = \rho \dot{V}_1 \omega R^2 \quad (6.30)$$

$$P_{wheel} = \rho \dot{V}_1 \omega^2 R^2 \quad (6.31)$$

The shaft torque in the oil volume V_2 is a notional torque of the oil from the pinion bearings and the flow in this oil sump can be regarded as a simple Couette flow. Thus, its shear force per unit area can be obtained from the viscosity dissipation of Couette flow.

$$\tau = \eta \frac{du}{dy} = \eta \frac{Gr\omega}{c} \quad (6.32)$$

where G is the gear ratio between the crown wheel and the pinion, and c is the bearing clearance. Therefore, the shaft torque and power loss are derived.

$$T_{shaft} = \eta \frac{\omega r}{c} G h_2 2\pi r^2 \quad (6.33)$$

$$P_{shaft} = \eta \frac{\omega^2 r}{c} G^2 h_2 2\pi r^2 \quad (6.34)$$

Table 6.4 The inputs for the variables of new churning model

Variable	Input	Variable	Input	Variable	Input
R	0.11 m	h_0	0.13 m	ρ_{low}	850.6 kg/m ³
r	0.036 m	A_1	0.0169 m ²	η_{low}	0.0262 Ns/m ²
G	3.31	A_2	0.064 m ²	ρ_{high}	856.6 kg/m ³
c	0.0005 m	V_0	0.001 m ³	η_{high}	0.21505 Ns/m ²
b	0.03 m	C_2	1.5E-06		

The inputs in Table 6.4 were used to calculate the wheel pumping torque in Equation 6.28, the shaft torque in Equation 6.30 and the total torque that is the sum of the wheel pumping torque and shaft torque. They were shown in Figure 6.23 and plotted on log scale as shown in Figure 6.22(a). From Figure 6.23, the pumping churning torque of the wheel for the high viscosity oil is lower than that for the low viscosity oil at high speeds. This is caused by the higher reduction in the oil immersion depth for the high viscosity oil than that for the low viscosity oil as shown in Figure 6.22(b). This result could be in good agreement with the author's explanation of sweeping-away of the oil free surface as shown in Figure 5.6 and the different oil flow rate due to the oil viscosity as shown in Figure 5.3 and Figure 5.4. They caused the reduction of oil immersion depth in the author's tests. The difference between the author's test result and this model is that the sweeping-away of the free oil surface was made not only by

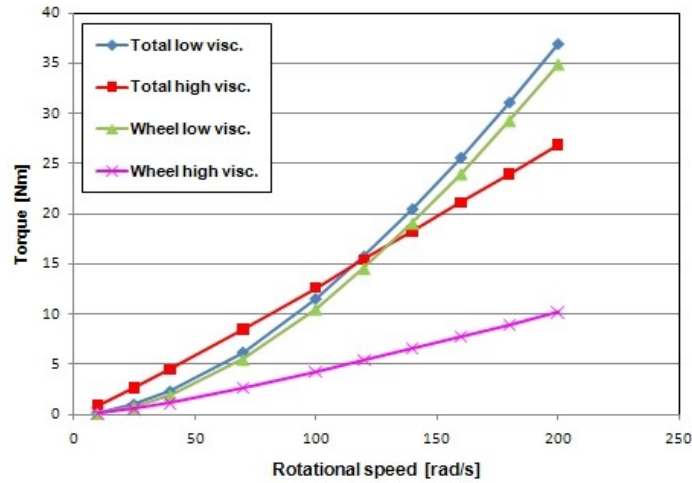


Figure 6.23 The torque of new churning model for the inputs in Table 6.4

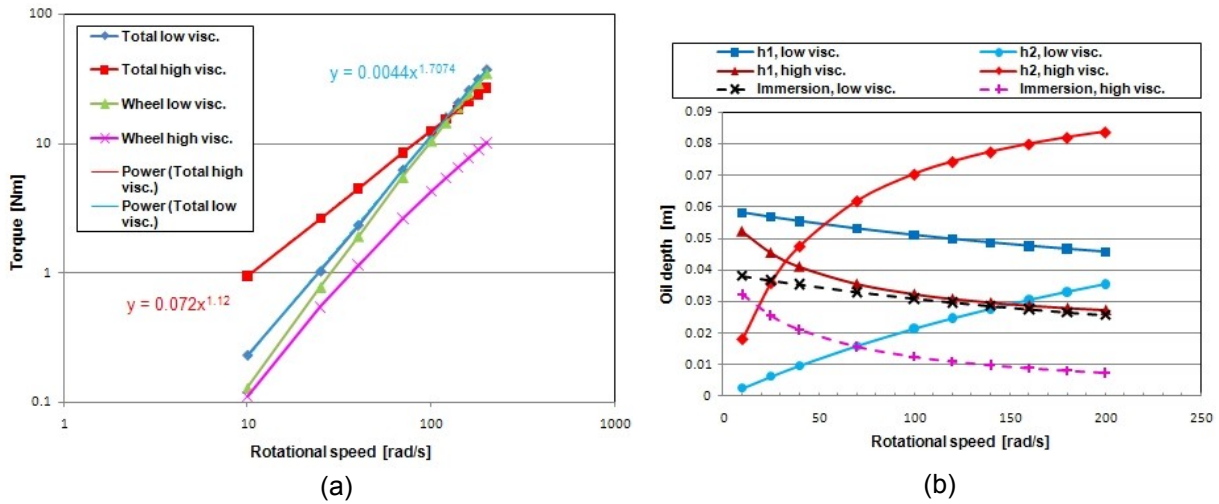


Figure 6.22 The calculated torque for the high viscosity oil and the low viscosity oil at oil immersion depth of 0.039 m according to $A_I=0.0169 \text{ m}^2$ (a) the torques on log scale (b) the oil depths of h_1 , h_2 and the immersion depths $[R-(h_0-h_1)]$ of two oils

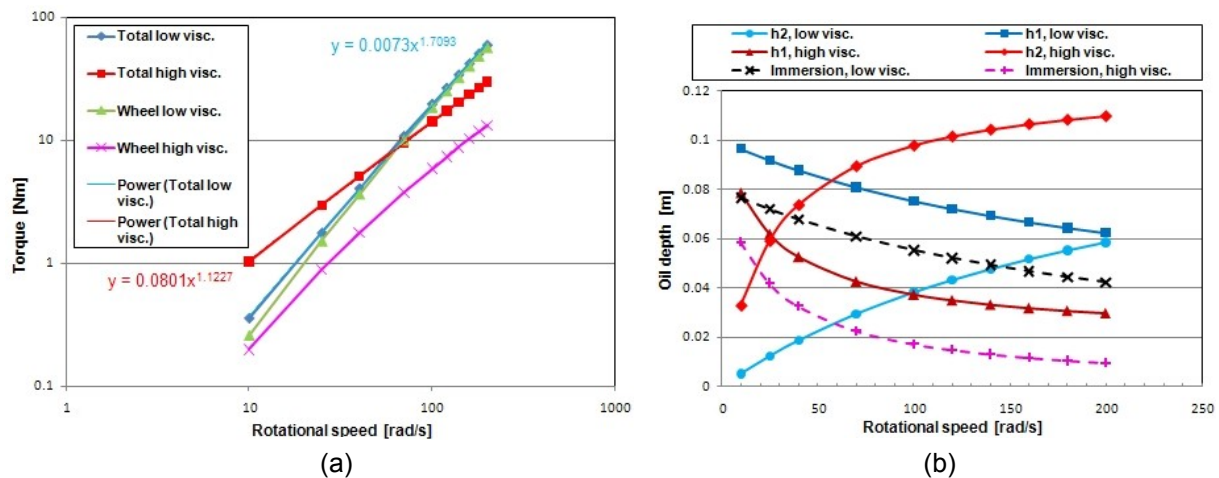


Figure 6.24 The calculated torque for the high viscosity oil and the low viscosity oil at oil immersion depth of 0.08 m according to $A_I=0.01 \text{ m}^2$ (a) the torques on log scale (b) the oil depths of h_1 , h_2 and the immersion depths $[R-(h_0-h_1)]$ of two oils

the pumping and but also mainly by the induced air from the rotating wheel. Also, some of the oil pumped by the crown fell directly and returned to the oil sump soon. Thus, the value of pumping torque in the author's test result could be different from that of this model. However, this model explains very well the author's test result as shown in Figure 4.15 that the churning torque for the high viscosity oil was a bit higher at low rotational speeds and was less than that for the low viscosity oil at high speeds.

The salient point is that the torque graphs for the high viscosity oil and the low viscosity oil plotted on log scale in Figure 6.22 and Figure 6.24 intersect and this is the same as those in Figure 6.18(a) and (b). Figure 6.24(a) shows the torques on log scale and they were calculated under the different bottom area of oil sump alone from the condition of Table 6.4. The change of the oil immersion depth for the high viscosity oil in Figure 6.24(b) is bigger than that in Figure 6.22(b). Thus, the intersection of two-viscosity graphs shows that the immersion depth for the high viscosity oil is more reduced by the pumping.

6.6 Suggestions for Improving Efficiencies

From the research described in the previous chapters and sections, the structure of housing to improve the efficiency regarding churning losses is suggested in this section.

From the experimental observations in the present study, it is proposed that a smaller size of crown wheel and an optimized oil immersion depth are beneficial in reduction of the churning losses in the dip lubrication of the axle. In addition, the proper positions of oil holes and oil guides are useful to reduce the churning losses for the high viscosity oil as well as to improve the lubrication of the system.

Low viscosity lubricants are preferable in dip lubrication, but it could cause durability problems due to a thin film thickness between gear teeth. In order to reduce the churning losses for the high viscosity oil, the housing requires small axial clearances with the ratio of the radial clearances to the diameter of the ring gear from 0.03 to 0.05 based on the tests for the axial and radial baffles in the extension housing in this thesis. However, the small oil volume capacity due to small axial clearances may cause the poor cooling from the heat generation inside the housing.

Thus, an alternative proposal is to install the small axial clearance baffles with the height less than a half diameter of the crown wheel instead of the compact housing as shown in Figure 6.25. This baffle has holes near the bottom of the housing, through which the oil could pass. This structure could minimize the pumping loss portion of the churning losses because the oil immersion depth between the baffles is reduced as the crown wheel rotates, whilst a large amount of oil is retained in the housing.

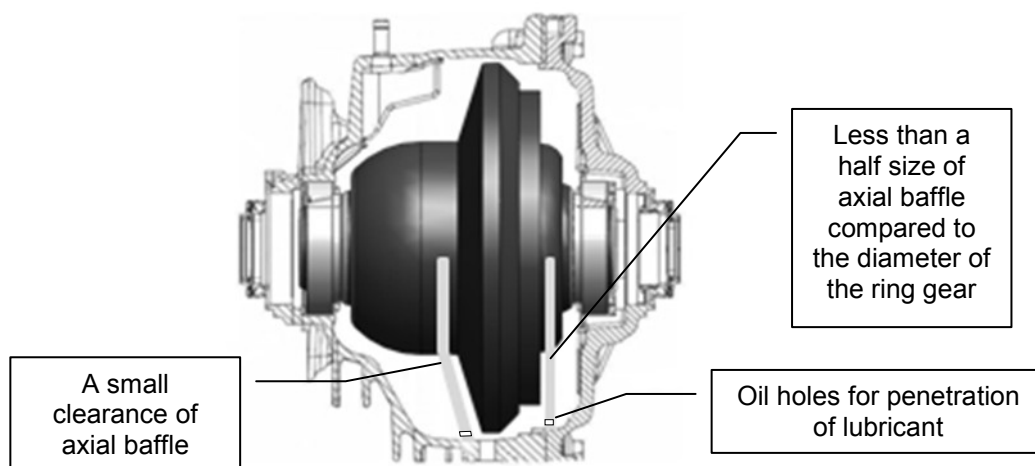


Figure 6.25 Schematic diagram for improving the churning losses in the dip lubrication

6.7 Concluding Remarks

In this section, the influencing variables of the author's experiments on the churning losses were fully discussed. In addition, the effects of dimensionless numbers were inspected through the dimensional analysis so that the dimensionless churning torque was approximately independent of Reynolds number for the lubricant at medium and high speeds. Finally, an empirical equation for the dimensionless churning torque was obtained through a non-linear regression method and a proposal to reduce the churning losses as well as the new churning model was suggested based on the test results in this thesis.

CHAPTER 7

CONCLUSIONS AND FUTURE WORK

This chapter presents conclusions that can be drawn from the research described in this thesis and suggestions for future work which can be carried out to develop the research into the churning losses for hypoid gears.

7.1 Introduction

This chapter consists of three parts. In the first, the achievements of the present research are summarized, covering the experimental finding and the theoretical advances. Secondly, the ways in which the present work may be extended are reviewed. Finally, a summary is given of the main conclusions.

7.2 Achievements of the Research

The primary requirement for detailed study of any losses in gearboxes is to be able to measure the losses occurring inside the housing in detail. Load-dependent losses and spin losses affect to each other, and so it is not easy to measure the losses separately. A study on factors affecting the churning losses has been carried out.

(a) A test rig for the measurement of churning torque in hypoid axles was designed, manufactured and commissioned. The measurement was based on the inertia-run-down method.

(b) An “extension housing” tests which allows the variation of clearances between wide limits was designed and manufactured. Additionally, a transparent lid and baffle for the extension housing were designed and made for the inspection of the oil flow inside the housing.

(c) A series of churning tests were carried out both on the production axle and on the extension housing. The experimental variables were rotational speeds, a ring-gear diameter, radial clearances, axial clearances, attitudes, oil viscosity and oil fill level. Through these experiments, influencing factors were found that decreased churning losses.

(a) Oil immersion depth was the most influential factor on the churning losses in dip lubrication.

(b) The churning torque for the crown wheel with 210mm diameter was smaller than for that with 220mm diameter.

(c) Much higher churning losses took place in the presence of the hypoid drive pinion when compared to those of the crown wheel alone.

- (d) The influence of tilting angle on churning losses was small, but a rolling position to the rear side of ring gear and a pitch down made churning losses increase by approximately 10 % at high speeds.
- (e) The influence of oil viscosity on the crown wheel alone was not large, but the high viscosity oil was more efficient than the lower viscosity oil in the continuous rotation in dip lubrication except at low speeds.
- (f) Axial clearances were more effective than radial clearances on the churning losses. An optimum radial clearance was estimated to be from $e_r/R_p = 0.055$ to 0.065 .
- (g) Oil flow and oil foam were investigated throughout the experiments using the extension housing with the transparent lid and baffle. The oil flow is an important influencing factor affecting the churning losses. However, oil foam changed the oil immersion level, but did not affect the churning losses.
- (h) The empirical equations from the experimental data were obtained and the new churning model was presented.

7.3 Suggestion for Future Work

The research forming the subject of this thesis has fallen two main areas: the experimental study of measuring churning losses and the analysis of its mechanism. In both areas, considerable further potential exists for extending the knowledge and understanding of churning losses related with load-dependent losses that has been generated by the present work. Suggestions for further work in the experimental and analytical sides are made

separately although it is recognized that both are likely to be needed for further advances to be made.

7.3.1 Experimental studies of churning losses

The study of churning torque requires measurements using a set-up equivalent to the real application, and cannot be achieved using simplified lubrication test such as using Mini Traction Machines etc. So the present experimental work based on the real set-up has established that the churning losses for hypoid gears in dip lubrication do not always become larger with higher rotational speeds as well as with the higher viscosity of lubricant. The influencing factors on the churning losses interact. For example, the enclosure housing experiments in this thesis showed that smaller axial clearances were effective to reduce the churning losses even in the high viscosity oil, but a very small radial clearance with large axial clearances would reduce the churning losses only if the oil volume was small. The influence of lubricant viscosity on churning losses has not been clearly identified because only two lubricants were compared. The present work suggests that other factors such as axial baffles combined with a high viscosity lubricant could reduce the churning losses. The present study will provide a very useful baseline for the investigation of further possibilities.

A number of possible studies may be suggested, some of which might shed considerable light on the churning losses in jet lubrication rather than dip lubrication. Jet lubrication could reduce churning losses fundamentally rather than dip lubrication because there is no immersion of the gears in jet lubrication.

The effect of the shape of housings could be considered for the future study. A rounded shape along the side of the crown wheel could be beneficial to decrease the churning losses due to

Sweeping-away effect of oil free surface, but the shape of the side has not been studied.

Pressure-viscosity as well as the variance of temperatures during the period of tests based on thermal balance may be taken into consideration for hypoid axles because a rise in temperatures caused by friction and spin losses could change the pressure inside axle housing, and affect the lubricant viscosity. In addition, the effect of viscosity and the additives on the churning losses should be investigated for various synthetic lubricants. Bubbles inside the lubricant are thought to be an important factor at low temperatures to be combined in the analysis of the churning losses although there seemed to be no influence in the author's experiments.

Finally, the effect of the presence of a hypoid pinion on churning losses for the extension housing could be further investigated under the several conditions such as the enclosures and the geometry. The experiments were not performed for the effect of the enclosures on a pair of the hypoid gears in the extension housing although the baffles for the tests were made. In addition, the effect of gear spiral angles and module on churning losses needs to be investigated through the comparison tests between hypoid gears and bevel gears.

7.3.2 Analysis of churning losses mechanisms

The work described in this thesis has entailed a detailed examination of the churning losses mechanism based on the test results and the dimensional analysis. The most of the time during this research was spent designing and manufacturing the test rig and the extension housing with the baffles in order to carry out the experiments for the measurement of the churning losses. However, there is considerable potential for a more vigorous numerical analysis such a many in principles be obtained using CFD.

Further analysis of the churning losses for the meshed hypoid gears is also required for the full development of the theory related to the load-dependent losses such as gear friction, bearing and seal losses. In addition, the new churning model introduced in Chapter 6 could be further developed for a pair of meshed hypoid gears.

It is likely that both these approaches can usefully be extended to improve the current understanding of the churning losses for the hypoid axles.

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APPENDIX 1. – HYPOID GEAR STRUCTURE

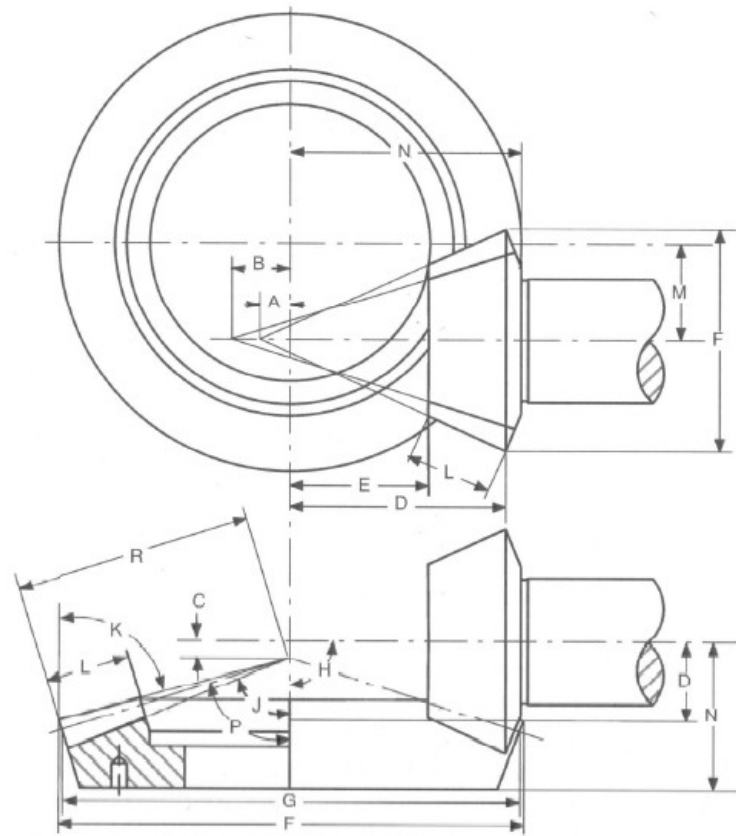


Figure A.1 Hypoid gear nomenclature

A: Face apex beyond crossing point

B: Root apex beyond crossing point

C: Pitch apex beyond crossing point

D: Crown crossing point

E: Front crown to crossing point

F: Outside diameter

G: Pitch diameter

H: Shaft angle

J: Root angle

F: Outside diameter

G: Pitch diameter

H: Shaft angle

J: Root angle

K: Face angle of blank

L: Face width

M: Pinion offset

N: Mounting distance

P: Pitch angle

R: Outer cone distance

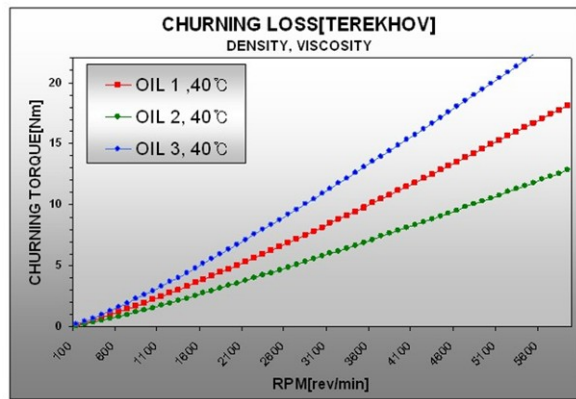
APPENDIX 2. – EFFECTS OF PARAMETERS ON CHURNING LOSSES MODEL (BONNES, TEREKHOV AND CHANGENET)

By using the formulae in Section 2.5, the influential factors were investigated.

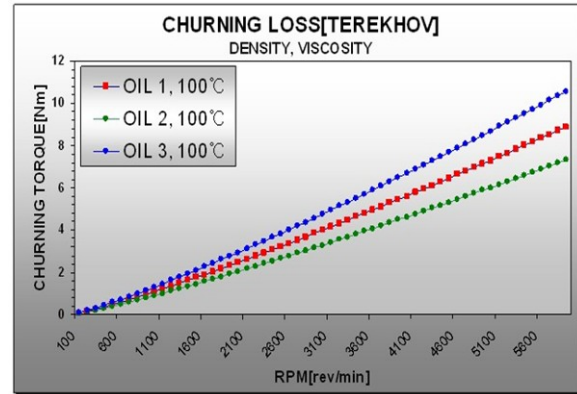
Table A2.1 Properties of lubricant for Figure 2.9

Property		OIL No.1	OIL No.2	OIL No.3
Density [kg/m^3]		865	873	897.8
Viscosity [m^2/s]	40°C	0.0001455	0.000048	0.00032
	100°C	0.0000157	0.0000083	0.000024

In order to examine the effect of lubricant viscosity and density on churning, the churning drag torques for a spur gear rotating in three different oils were plotted as Figure A 2.1 using Terekhov's empirical formula. Properties of the three oils are given in Table 2.8. The densities of three oils are similar, however the kinematic viscosity increase in the order oil: 2 < oil 1 < oil 3. Changenet's empirical formula was used to calculate the churning torque, and the influence of oil immersion depth for a spur gear is shown in Figure A2.2(a). It can be seen that large churning losses occur at large ratios of immersion depth to the radius of a spur gear especially at high speeds. In order to investigate the effect of the teeth number and the diameter of spur gears on churning, the data for three types of spur gears were put in Changenet's empirical formula. Each of three types of gears had 3mm module and 20mm face width, but the different number of teeth, 72, 56 and 40. The calculated values of churning torque were plotted in Figure A2.2(b). The churning drag torque is higher in the order: 72 teeth > 56 teeth > 40 teeth. This could be mainly due to the increased oil immersion depth of the gears, . Figure A2.3 shows the calculated results of churning losses for meshed spur gears immersed in oil using the empirical formulae of Bonnes, Terekhov and Changenet in the program coded by the author. Boness's formula is much different from the two other formulae.

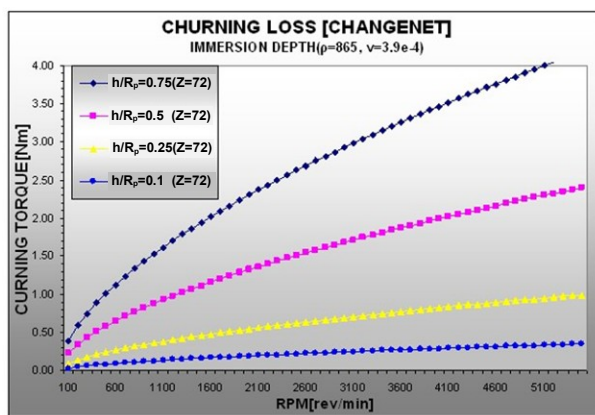


(a)

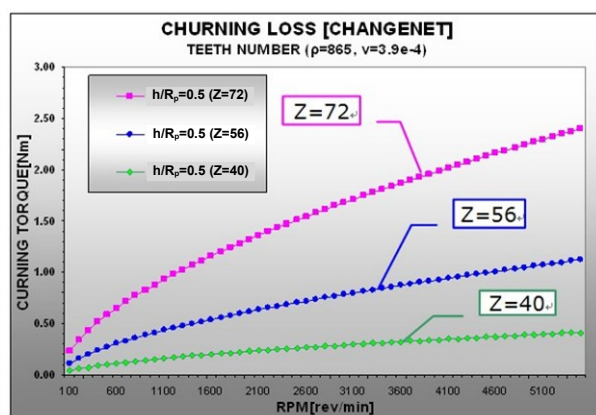


(b)

Figure A 2.1 The effect of density and viscosity on churning losses over a range of rotational speeds (a) viscosity of oil at 40°C (b) viscosity of oil at 100°C

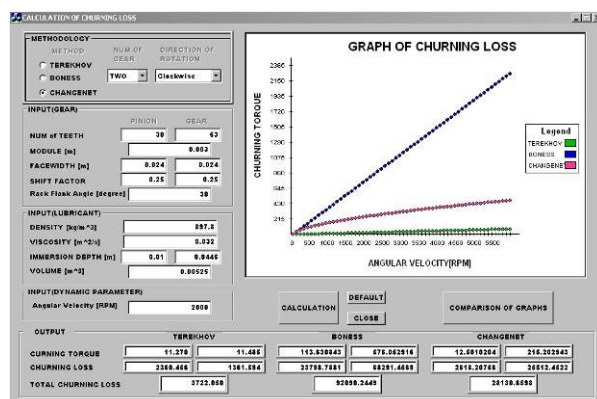


(a)

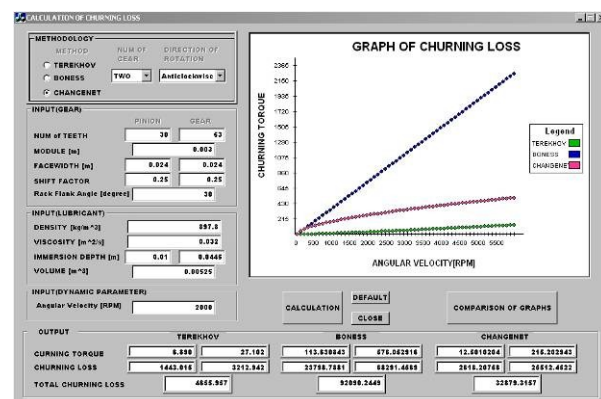


(b)

Figure A2.2 The effect of oil immersion depth on churning losses over a range of rotational speeds (a) effect of immersion depth (b) effect of number of teeth and pitch diameter



(a)



(b)

Figure A2.3 Calculation of churning losses for meshed spur gears using the formulae of Boness, Terekhov and Changenet (programmed with Visual C) (a) clockwise rotation (b) anti-clockwise rotation

APPENDIX 3. – TRIFILAR SUSPENSION METHOD

Generally, the moment of inertia of a rigid body with regard to an axis is mathematically defined by

$$I = \int r^2 dm \quad (\text{A3.1})$$

One of the effective experimental ways to determine the moment of inertia of particular shaped rotating components is to use the trifilar suspension method based on a simple vibration theory. This method is generally applied when one needs to get the moment of inertia of an irregular shaped component with respect to the rotational axis. The method uses a circular platform suspended by three parallel wires of the same length supporting the irregular component. On this platform, the object in question is centrally located at G point, the centre of the platform as shown in Figure A3.1. The mass of the empty platform is m and the platform is suspended at three points having a distance r from its centre G by three strings of

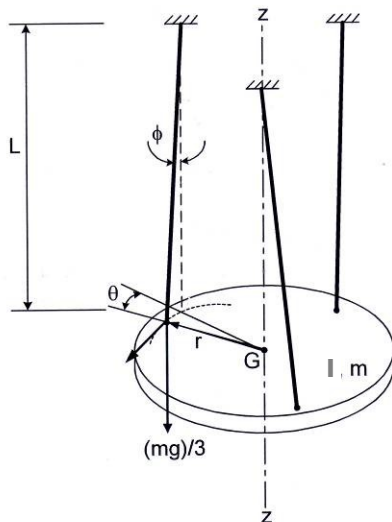


Figure A3.1 Experimental schematic of trifilar suspension method (Department of Mechanical Engineering, 2004; Lim, 2008)

length L . The angle ϕ is a very small displacement and θ is the small angle at which the platform oscillates. The moment of inertia of the platform is denoted as I , and the centre G of gravity lies on the axis z - z of rotation of platform.

The derivation of trifilar suspension equation for the platform in Figure A3.1 is as shown below. According to Newton's law and a geometric assumption that ϕ is a very small displacement, Equations A3.2 and A3.3 are

derived. The restoring force is given by

$$F = mg \sin \phi \approx mg\phi = \frac{mgr\theta}{L} \quad (\text{A3.2})$$

$$I\ddot{\theta} - \sum Fr = I\ddot{\theta} + \frac{mgr^2\theta}{L} = 0 \quad (\text{A3.3})$$

This equation is for harmonic motion without damping and so the natural frequency is

$$\omega_n = \pm \sqrt{\frac{mgr^2}{IL}} \quad (\text{A3.4})$$

$$\frac{2\pi}{\omega_n} = T_{\text{period}} = 2\pi \sqrt{\frac{IL}{mgr^2}} \quad (\text{A3.5})$$

Thereafter, I is driven as

$$I = \frac{T_{\text{period}}^2}{4\pi^2} \frac{mgr^2}{L} \quad (\text{A3.6})$$

From Equation A3.6, the moment of inertia of the object component can be expressed as

$$I_{\text{component}} = I_{\text{component+platform}} - I_{\text{platform}} \quad (\text{A3.7})$$

$$I_1 + I_2 = \frac{T_{\text{period},1\&2}^2}{4\pi^2} \frac{(m_1 + m_2)gr^2}{L_{1\&2}} \quad (\text{A3.8})$$

Where I_1 is the moment of inertia for the component in question, I_2 is the moment of inertia for circular platform, m_1 the mass of the component, m_2 is the mass of the circular platform, $T_{\text{period},1\&2}$ is the period of suspension of the component and the platform and $L_{1\&2}$ is the length of suspended string when the component is placed on the platform. Finally, it is calculated by

$$I_1 = (I_1 + I_2) - I_2 = \frac{T_{\text{period},1\&2}^2}{4\pi^2} \frac{(m_1 + m_2)gr^2}{L_{1\&2}} - \frac{T_{\text{period},2}^2}{4\pi^2} \frac{m_2gr^2}{L_2} \quad (\text{A3.9})$$

APPENDIX 4. – LUBRICANT VISCOSITY

Table A4.1 Viscosity classification of automotive gear oils by SAE J 306

SAE Viscosity grade	Maximum temperature for viscosity of 150 [Pa·s ¹]	Viscosity at 100°C [mm ² ·s ⁻¹]	
		Minimum	Maximum
70W	-55	4.1	-
75W	-40	4.1	-
80W	-26	7.0	-
85W	-12	11.0	-
90	-	13.5	24.0
140	-	24.0	41.0
250	-	41.0	-

Table A4.2 Viscosity of lubricants measured from Stabinger viscometer

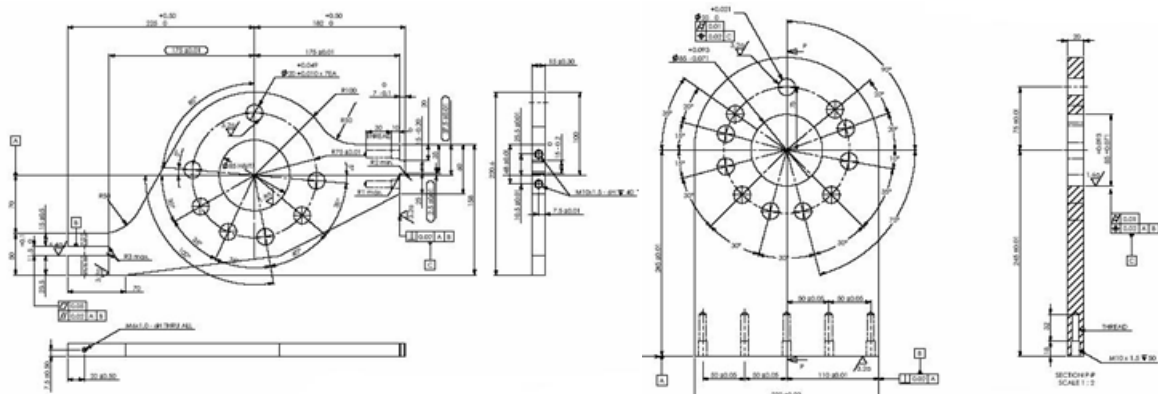
Temp. [°C]	Mineral oil			SAE 75W90		
	Density [g/cm ³]	Dynamic viscosity [mPa·s]	Kinematic viscosity [mm ² /s]	Density [g/cm ³]	Dynamic viscosity [mPa·s]	Kinematic viscosity [mm ² /s]
20.0	0.8522	28.874	33.882	0.8556	241.37	282.09
22.0	0.8506	26.204	30.807	0.8549	215.05	251.56
23.0	0.8499	25.016	29.434	0.8543	203.37	238.06
24.0	0.8493	23.904	28.145	0.8537	192.53	225.53
25.0	0.8486	22.852	26.928	0.8531	182.35	213.75
26.0	0.8480	21.859	25.777	0.8525	172.89	202.80
27.0	0.8474	20.924	24.693	0.8520	163.91	192.39
28.0	0.8467	20.043	23.671	0.8514	155.65	182.81
29.0	0.8461	19.210	22.705	0.8509	147.84	173.75
30.0	0.8455	18.453	21.818	0.8503	140.50	165.23
31.0	0.8448	17.682	20.930	0.8498	133.57	157.18
32.0	0.8442	16.980	20.114	0.8493	127.11	149.66
33.0	0.8436	16.313	19.338	0.8487	121.01	142.58
34.0	0.8430	15.684	18.606	0.8482	115.31	135.96
35.0	0.8423	15.086	17.910	0.8476	109.91	129.67
40.0	0.8393	12.534	14.933	0.8432	87.436	103.69
50.0	0.8332	8.9428	10.733	0.8370	57.390	68.564
60.0	0.8267	6.6476	8.0415	0.8309	39.496	47.532
70.0	0.8204	5.1151	6.2353	0.8249	28.301	34.309
80.0	0.8141	4.0497	4.9741	0.8186	20.967	25.615
90.0	0.8078	3.2827	4.0637	0.8123	16.006	19.704
100.0	0.8016	2.7155	3.3874	0.8062	12.532	15.544

¹ Measured according to ASTM D-2983-87 in the Brookfield viscometer

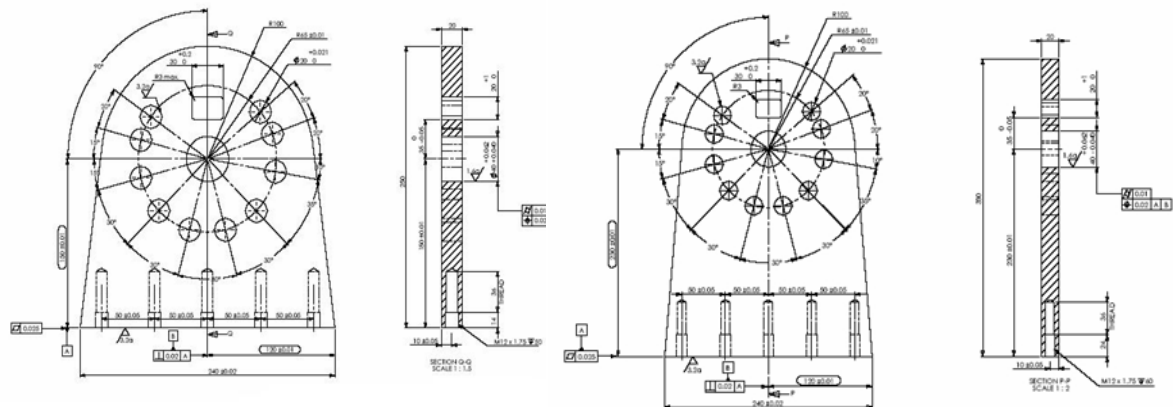
The image contains two sets of technical drawings. The left set, labeled 'Pulley support stand', includes a front view showing a circular pulley with a diameter of 80 mm and a mounting plate with four vertical slots. A side view shows the pulley's profile. A detail view shows a cross-section of the pulley with a diameter of 80 mm and a thickness of 10 mm. The right set, labeled 'Tilting side plate-rolling', includes a front view showing a semi-circular plate with a radius of 100 mm and a central pivot point. A side view shows the plate's profile. A detail view shows a cross-section of the plate with a diameter of 80 mm and a thickness of 10 mm. The drawings are labeled with dimensions and tolerances.

Pulley support stand

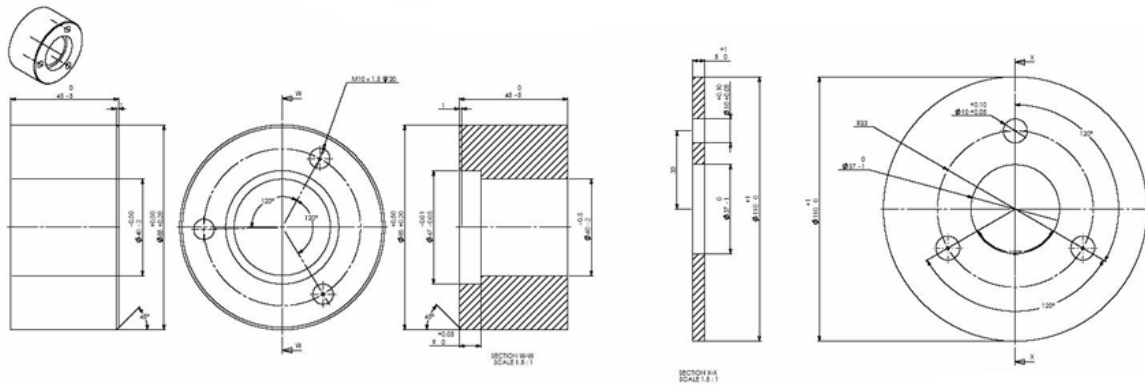
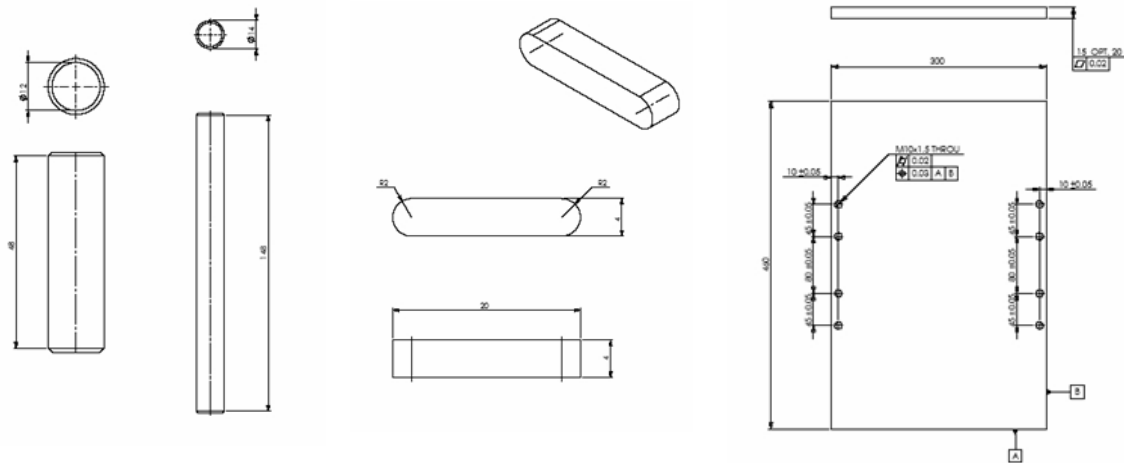
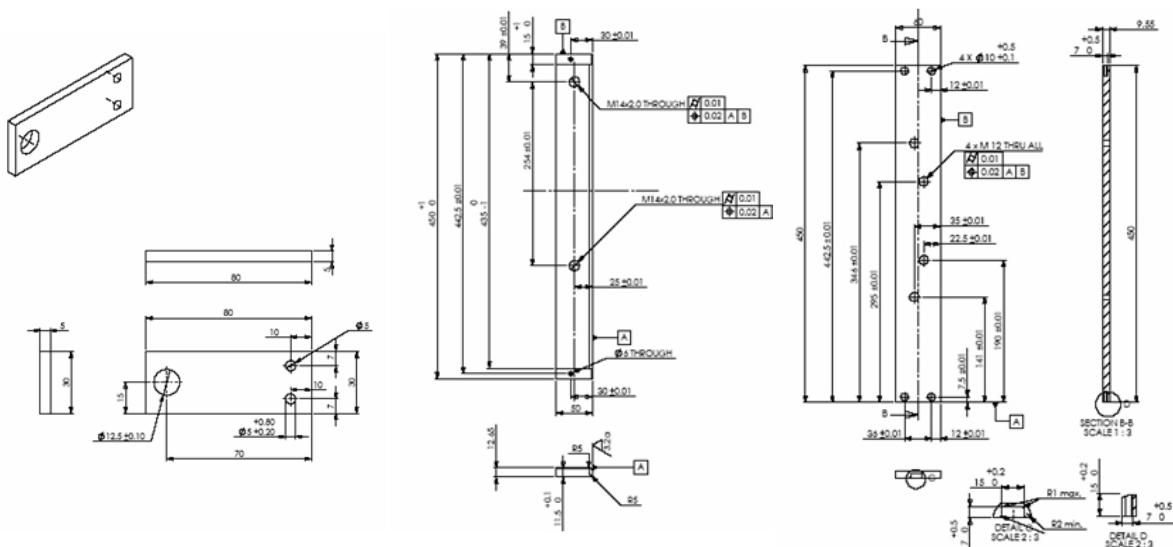
Tilting side plate-rolling

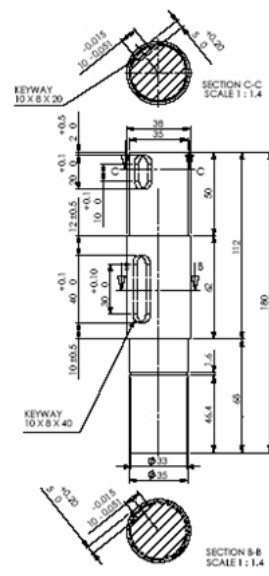
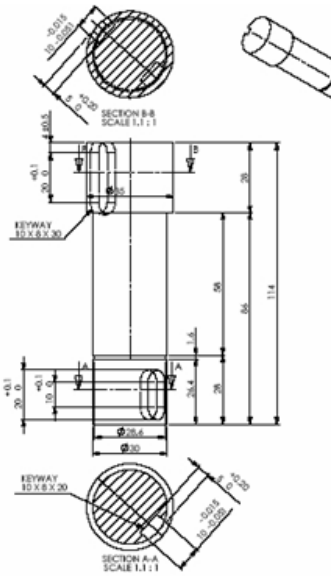
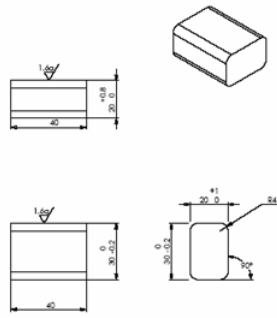


Housing tilting support stand-pitching



Housing tilting support stand-rolling

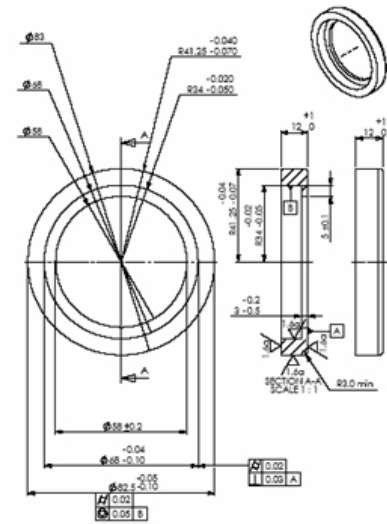
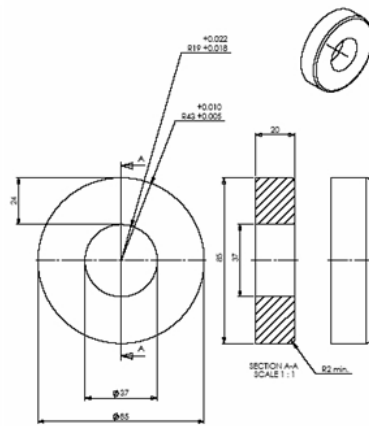
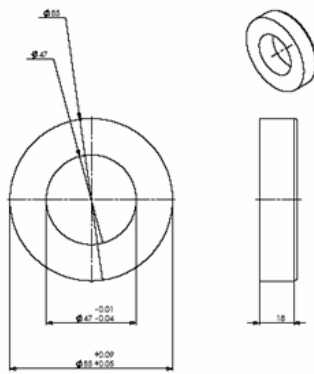
**Bearing support****Bearing support lid****Tilting plate key****Shaft key****Tilting mounted plate-rolling****Hall sensor plate****Tilting front plate-rolling****Tilting rear plate-rolling**



Rectangular key

Shaft-clutch

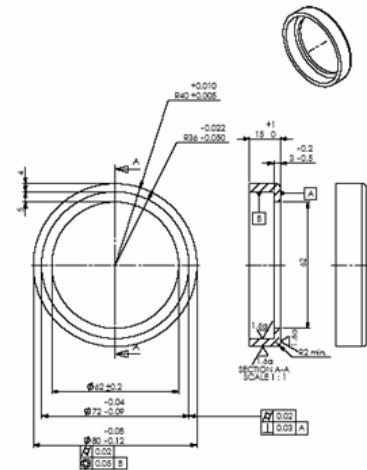
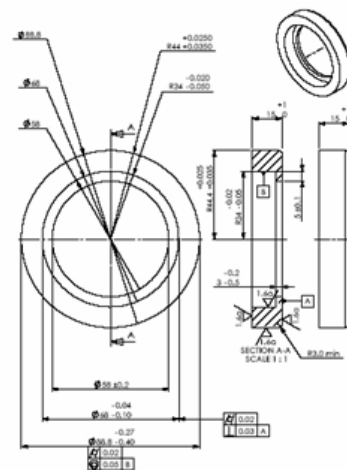
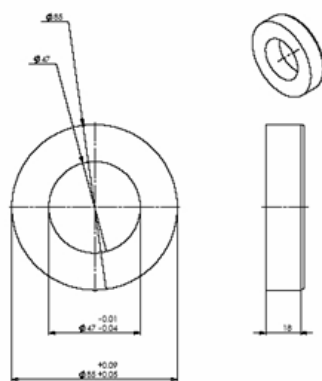
Shaft-pulley



Sleeve 68_82.5 7.25t

Sleeve 47_85 19t

Sleeve 37_85 24t



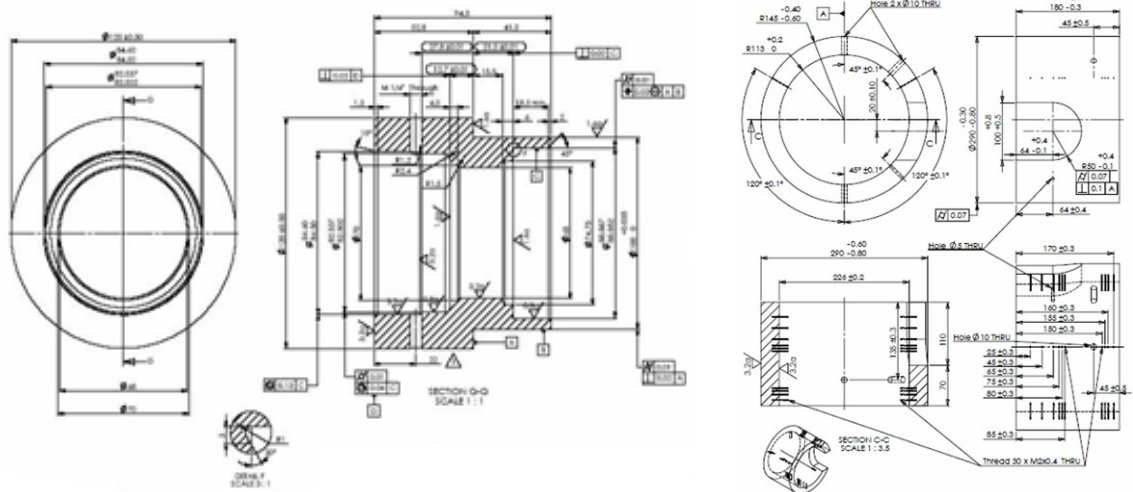
Sleeve 41.5_45 2.7t

Sleeve 68_88.9 10.45t

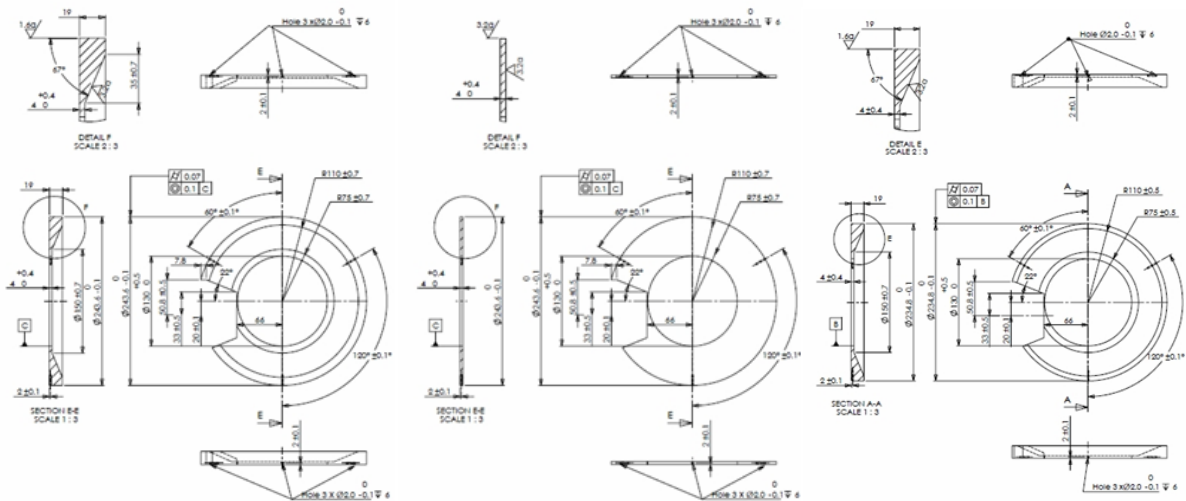
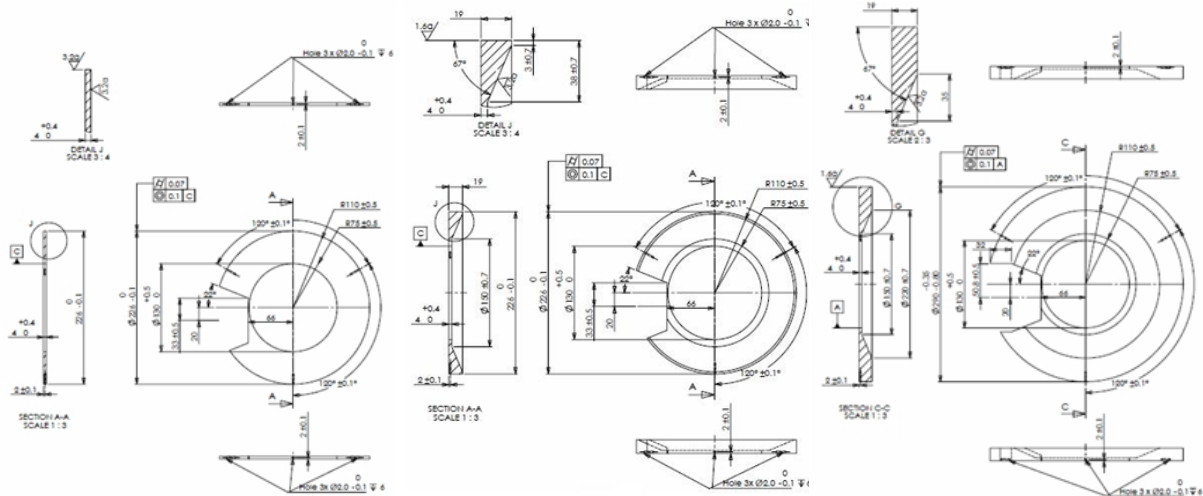
Sleeve 72_80 4t

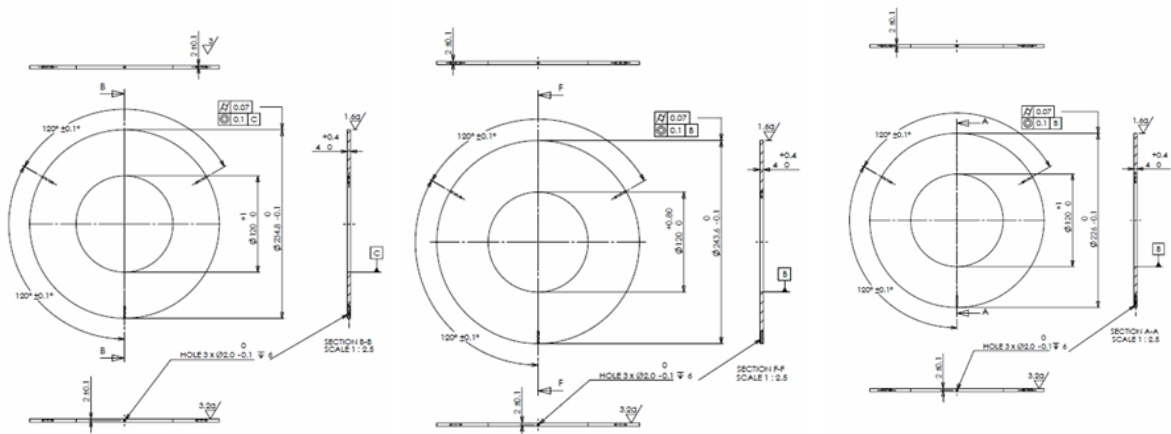
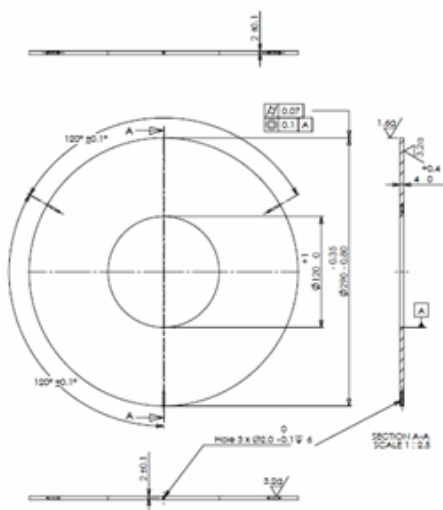
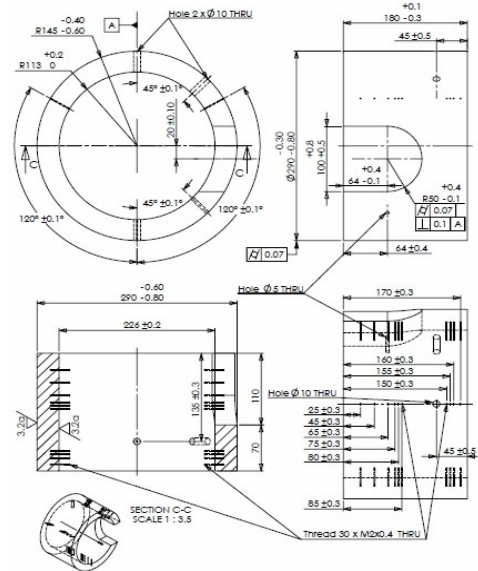
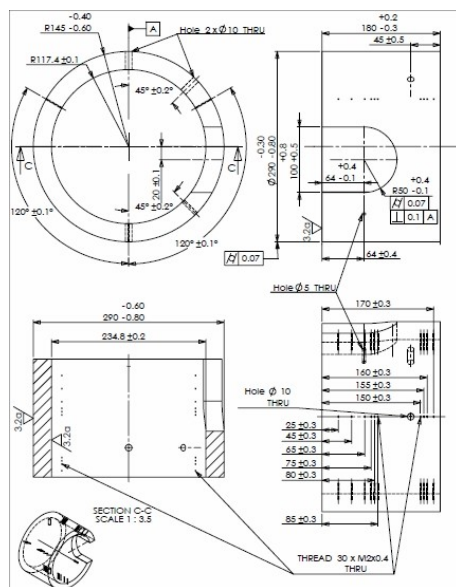
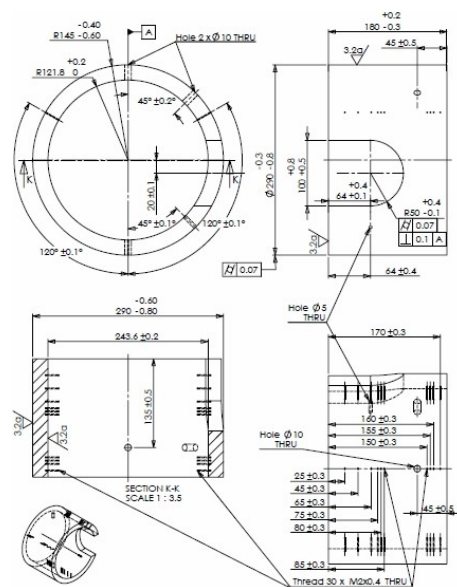
No.	NAME	SHEET	QTY
1	5x1 housing part1	1	1
2	5x1 housing part2	1	1
3	5x1 housing part3	1	2
4	5x1 housing part4	1	1
5	5x1 housing part5	1	1
6	5x1 housing part6	1	1
7	M12 x 30mm bolt	-	8EA
8	M10 x 40mm bolt	-	13EA





Ext. housing part4

Radial baffle $\Phi 226$ Axial baffle-left1 $\Phi 243$ Axial baffle-left2 $\Phi 243$ Axial baffle-left1 $\Phi 234$ Axial baffle-left1 $\Phi 226$ Axial baffle-left2 $\Phi 226$ Axial baffle left $\Phi 290$

Axial baffle-right $\Phi 234$ Axial baffle-right $\Phi 243$ Axial baffle-right $\Phi 226$ Axial baffle-right $\Phi 290$ Radial baffle $\Phi 226$ Radial baffle $\Phi 234.8$ Radial baffle $\Phi 243.6$

APPENDIX 7. – RIG COMPONENTS LIST

NO	ITEM	Q'TY	SPECIFICATION	MANUFACTURER
1	MOTOR	1	DC Motor ; 1120 143(Serial no.) 4KW, Max 3,000RPM, 170/180V, 28AMPS, Excitation volt 195/205V Enclosure DPFV ; BRGS DE 6207, NDE 6205/22	Dursley Mawdsley's
2	METAL POLE	1	Φ20X300mm	SELF MANUFACTURE
3	METAL POLE	1	Φ40X500mm	Mecheng store + SELF MODIFY
4	PLATE	1	700mm X 220mm X 20mm (Motor)	SELF MANUFACTURE
5	PLATE	1	450mm X 275mm X 20mm (Tilting Rig)	SELF MANUFACTURE
6	Tilting plate side	4	TOLWORTH TOOLS	TOLWORTH +SELF MODIFY
7	Cs_Tilting side	2	TOLWORTH TOOLS	TOLWORTH +SELF MODIFY
8	Cs_Tilting stand	2	TOLWORTH TOOLS	TOLWORTH +SELF MODIFY
9	T_support stand	2	TOLWORTH TOOLS	TOLWORTH +SELF MODIFY
10	M_support stand	2	TOLWORTH TOOLS	TOLWORTH +SELF MODIFY
11	THREAD BAR	1	SIM-14 (M14, Pitch 2.0, 1000mm) : LOT 1EA	ON DRIVES
12	HEXAGON FULL NUTS	10	SHN-14 (M14, Pitch 2.0) : LOT 1EA	ON DRIVES
13	FITTING	2	RS 807-255	RS
14	FITTING	2	RS 110-466	RS
15	PLUG	3	BRASS THREADED BLANKING PLUG G1/4" (RS 721-397)	RS
16	PLUG	2	BRASS THREADED BLANKING PLUG M5 (RS 721-375)	RS
17	Deep Groove Ball BRG	2	Φ35 X Φ62 X 14mm	ON DRIVES
18	Thin section Ball BRG	6	Φ45 X Φ68 X 12mm OPEN (6909 EZO)	SMB
19	Thin section Ball BRG	6	Φ50 X Φ72 X 12mm OPEN (6910 EZO)	SMB
20	Thin section Ball BRG	2	Φ30 X Φ47 X 9mm (6906ZZ EZO)	SMB
21	SLEEVE	2	TOLWORTH TOOLS [Φ41.5 X Φ45 X 20mm (1.75t)]	TOLWORTH +SELF MODIFY
22	SLEEVE	2	TOLWORTH TOOLS [Φ68 X Φ82.5 X 12mm (7.25t)]	TOLWORTH +SELF MODIFY
23	SLEEVE	2	TOLWORTH TOOLS [Φ68 X Φ88.9 X 12mm (10.45t)]	TOLWORTH +SELF MODIFY
24	SLEEVE	4	TOLWORTH TOOLS [Φ72 X Φ80 X 15mm (4.0t)]	TOLWORTH +SELF MODIFY
25	SLEEVE	2	TOLWORTH TOOLS [Φ85 X Φ47 / 40 X 45mm]	TOLWORTH +SELF MODIFY
26	Circular plate	2	TOLWORTH TOOLS [Φ30 X Φ110 X 5mm (4.0t)]	TOLWORTH +SELF MODIFY
27	PULLEY STAND	2	TOLWORTH TOOLS [Φ30 X Φ110 X 5mm (4.0t)]	TOLWORTH +SELF MODIFY
28	Data Logger	1	Low cost Multifunction DAQ for USB6009(779026-01)	National Instrument
29	HALL SENSOR	1	Hall effect gear tooth sensor,4.5-24Vdc (RS 235-5706)	RS
30	Stand for hall sensor	1	60 X 30 mm	SELF MANUFACTURE
31	THERMOCOUPLE	3	K PTFE min fitted plug thermocouple,1m (RS no : 363-0250)	RS
32	Steel socket Head Cap Screws	22	AM12-60 [M12 X 60(L) X 1.75 X36(B)]	ON DRIVES
33	Steel socket Head Cap Screws	11	AM12-110 [M12 X 110(L) X 1.75 X36(B)]	ON DRIVES
34	DPDT std Toggle switch	1	DPDT toggle switch w/flat lever, 15A(RS 665-685)	RS
35	CIRCLIP	4	CX30 Φ40.6 X Φ28.6 X 1.5	ON DRIVES
36	CIRCLIP	2	CX35 Φ47 X Φ33 X 1.5	ON DRIVES
37	ROUND END FEATHER KEY	3	KK-10-8-20 (10 * 8* 20mm)	ON DRIVES
38	ROUND END FEATHER KEY	2	KK-10-8-30 (10 * 8* 30mm)	ON DRIVES

39	ROUND END FEATHER KEY	1	KK-10-8-40 (10 * 8* 40mm)	ON DRIVES
40	PULLEY SHAFT	1	Φ35 / Φ38 * 153 mm (Spline, Keyway)	SELF MANUFACTURE
41	PULLEY	1	Belt Pulley (SPA 224) 224mmPCD	BEAR BOYS
42	PULLEY TAPER BUSH	1	2012 BUSH	BEAR BOYS
43	BELT	1	SPA-2000, Wedge belt, 12.7mm Top width, 2000mm	BEAR BOYS
44	BRG	2	6007-2RS C3 (Φ35 * Φ62 * 14mm)	SHOP
45	CLUTCH SHAFT	1	Φ30 / Φ35 * 114mm (Keyway)	SELF MANUFACTURE
46	STUB SHAFT	1	Φ35 * 114mm (Keyway)	SELF MANUFACTURE
47	PC	1	Tribology Lab Property	SELF SUPPLY
48	FLEXIBLE COUPLING	1	RTE-SL-24/28 Φ35 * Φ55 * 78mm	ON DRIVES
49	HUB-SYNC	2	Keyway added	SELF SUPPLY
50	SLEEVE-SYNC	1	Proper width and diameter required	SELF SUPPLY
51	SHIFT FORK	1	Modification required	SELF SUPPLY
52	POLE_SHIFT FORK	1	Modification required	SELF SUPPLY
53	STD AXLE ASS'Y	2	Supplied parts by LANDROVER (1 Ass'y + 1 Axle internals)	Motor company
54	SEAL	4	For replacement (Supplied parts by LANDROVER)	Motor company
55	Hall-Effective device	1	Hall-Effective device	RS
56	Epoxy putty	1	Magic bond Epoxy putty stick	RS
57	MAGNET	6	Switch/Sensor magnet (12.7 x 3.18 x 1.59mm)	RS
58	BRG SHIM WASHER	1	CPS45x56x0.3 (Φ45 x Φ56 x 0.3)	Ondrives
59	BRG SHIM WASHER	1	CPS45x56x2.0 (Φ45 x Φ56 x 2.0)	Ondrives
60	ARALDITE RAPID STEEL	1		SHOP
61	CHISEL COLD	1	6x1 2in	SHOP
62	ALUMINIUM TAPE	1		RS
63	RTV SILICON SEALANT	3	HA6 RTV SILICON SEALANT 3EA + CAULKING GUN	SHOP
64	SILICON STOPPERS	1	103-1790, 103-1984, 103-2081, 03-66-4721	SHOP
65	FUSE 10A	10	Anti surge T LBC min 10A 6.3x32mmm(4189618)	RS
66	FUSE 12.5A	10	Anti surge T LBC min 12.5A 6.3x32mmm(4189630)	RS
67	FUSE 32A	10	Type aM fuse 32A 500V 14x51mm(1887381)	RS
68	PTFE thread tape	1	Thread seal tape (12m L x 12mm W; 512238)	RS
69	3 JAW puller	1	150mm capacity, RS Stock No. 3497306	RS
70	PLASTIC BAR	3	BAR (natural Ertalon 6PLA) Φ290 x 220 mm	theplaticshop
71	PLASTIC THIN PLAE	4	THIN PLATE (natural Ertalon 6 PLA) 300 x 300 x 20 mm	
72	PLASTIC THIN PLAE	6	THIN PLATE (natural Ertalon 6 PLA) 300 x 300 x 4 mm	
73	Slotted optical switch	1	Slotted optical switch (HOA 2001-001)	RS
74	DC motor speed controller, 32A	1	150mm capacity, RS Stock No. 3497306	RS
75	Type aM fuse	1	50A 400V 14x51mm	Maplin
76	Anti surge T LBC min fuse	2	10A 250V 6.3x32mm	Maplin
77	PTFE thread seal tape	2	12mm L x 12mm W	Mecheng store
78	Araldite(R) epoxy adhesive	1	32mg tube	Maplin
79	SET SCREW CUP POINT	50	A2303/304 18-8(SM2-12), Thread: M2, Length: 12, pitch: 0.40, 1.0, 0.9	ON DRIVES

APPENDIX 8. – CIRCUIT OF PHOTO SENSOR CIRCUIT UNIT AND PHOTO GATE

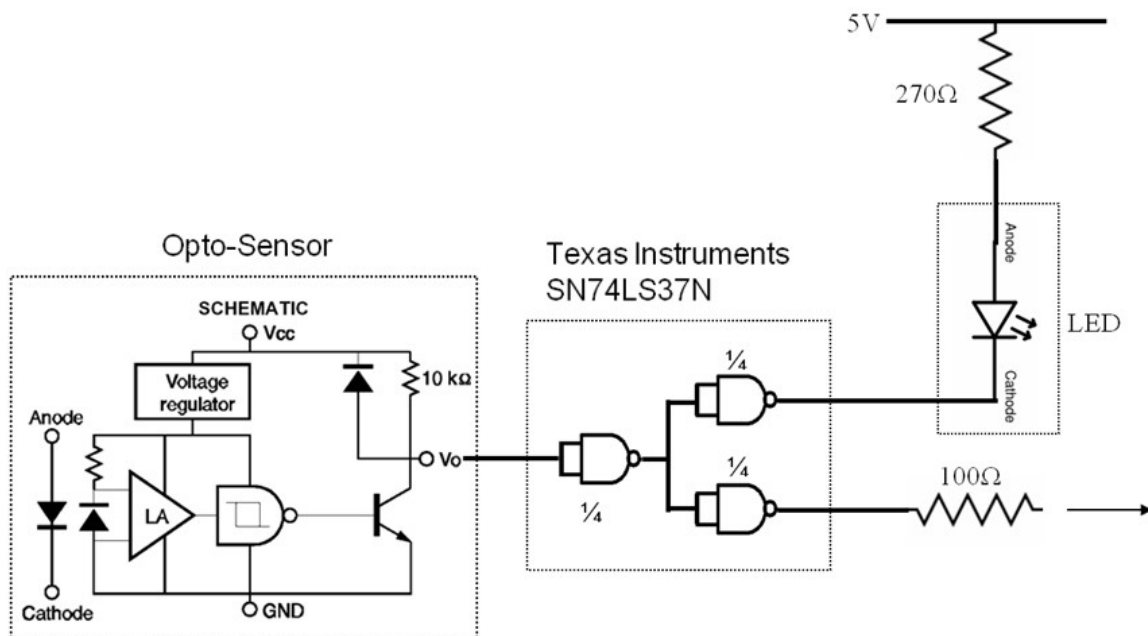


Figure A8.0.1 Circuit diagram of photo sensor

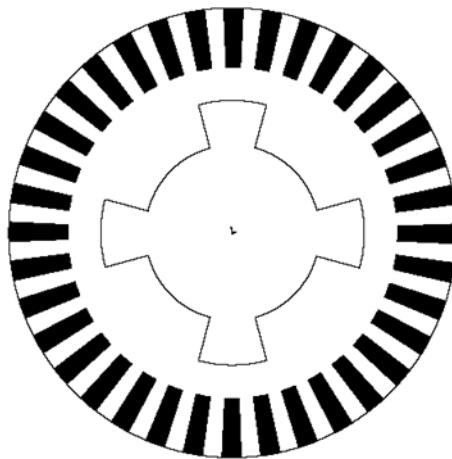


Figure A8.0.2 Optical encoder

APPENDIX 9. – LABVIEW PROGRAMMING

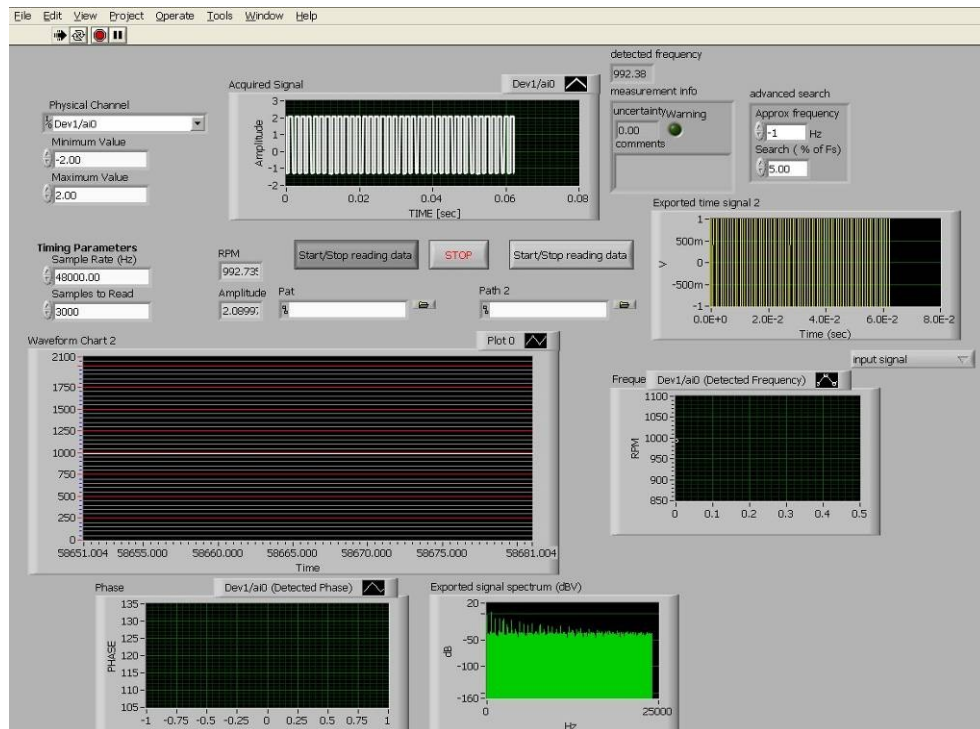


Figure A9.9.1 Screen capture of front panel of LabVIEW program at 1000RPM of hypoid axle

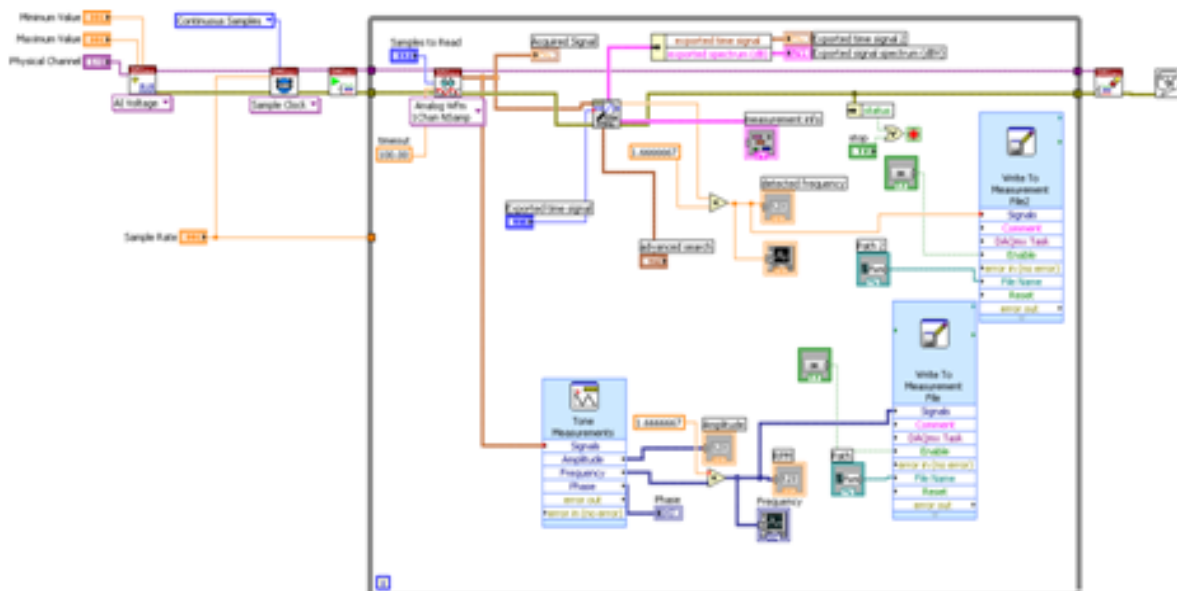


Figure A9.9.2 Block diagram of LabVIEW program used in the experiments of this thesis

APPENDIX 10. – COMPARISON BETWEEN ROTATIONAL SPEED OF CROWN WHEEL AND VEHICLE SPEED

Table A10.1 Specification of tyre

Tyre	Width [mm]	Height [mm]	Circumference [mm]
215/65R15	215	139.75	660.5
225/55R17	225	123.75	679.3
235/55R18	235	129.25	715.7

Table A10.2 Correlation between RPM of rig and vehicle speed

Tyre type		RPM of TEST RIG [RPM]				
		1210	1000	804	643	482
Vehicle speed [km/h]	215/65R15	150.65	124.5	100.1	80.05	60.01
	225/55R17	153.65	128.05	102.95	82.33	61.72
	235/55R18	161.89	134.91	108.46	86.74	65.02