AN INVESTIGATION INTO POWER LOSSES OF MANUAL TRANSMISSION SYSTEM USING DIFFERENT VISCOSITY OILS

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Submitted by
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CERTIFICATION

I, Rohit Kumar, declare that this thesis report entitled ***"An Investigation Into Power Losses Of Manual Transmission System Using Different Viscosity Oils"***, submitted towards fulfillment of the requirements for the award of Master's Degree in Thermal Engineering, in Mechanical Engineering Department of Thapar University, Patiala, is entirely my own work. This document has not been submitted for any degree in any other institution.

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This is to certify that above statement made by the candidate is correct and true to the best of my knowledge.

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ABSTRACT

In this study, experiments are carried out to investigate the influence of several operating conditions on the power losses and efficiency of an automotive manual transmission. With all the operating parameters controlled tightly, power losses are measured for different loaded and unloaded conditions in a five-speed manual transmission system. Various experimental studies have been conducted to see the influence of parameters such as input speed, load, oil temperature and oil volume on load-dependent and load–independent power losses of the transmission. The effect of viscosity on the power losses are found by using four different oils (75W90, 80W90, 20W50 and 85W140) having different viscosities (0.075, 0.0844, 0.0919 and 0.3399 Pa-s) in the gearbox. It is found that all these parameters play a very significant role in the transmission power losses. Other than oils, nanofluids are also used to check the power losses by dispersing alumina nanoparticles in gear oil (75W90). It is reported that power losses are not affected too much on addition of nanoparticles but the rise in temperature is minimized. In order to increase the overall transmission efficiency, several conclusions are drawn.

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P_m: Mechanical power losses [W]

P_T: Total power losses [W]

P_S: Spin power losses [W]

η_T: Total efficiency [%]

η_m: Mechanical efficiency [%]

V_{oil}: Volume of oil [litre]

μ_{oil}: Viscosity of oil [Pa-s]

CHAPTER 1: INTRODUCTION AND OBJECTIVES

1.1 Introduction

Gearbox is usually a power transmission system which can provide controlled application of power. Gearbox mainly consist of gears and gear trains to provide speed and torque conversions from a rotating power source to another device (Uicker et al. 2003). Automobile engine generates power by the combustion of fuel, and this power is mainly present on the engine shaft or crank shaft. Further, this power is mainly used to drive the wheels and run the vehicle. But these engines operate at a very high rotational speed, which is inappropriate for starting, stopping and slower travel of the vehicle. Here the gearbox reduces the very high engine speed to slower wheel speed and increase the torque in this process, so that the vehicle can easily start, stop and run smoothly (Paul et al. 1979). Gearbox can be used in many industries areas like Sugar, Cement, Elevator, Textile, Solvent Extraction, Paper, Plastic & leather, Rubber, Power plants, Mines and minerals, Steel industry, Waste water treatment, Food & tobacco, wind mill industries. In automobile industry gearbox is also major part of their transmission system which is used in almost every automobile industry and in every automobile vehicle to transmit power from engine to the wheels in an appropriate manner.

Today almost every country face the problem of energy crisis and environmental concerns in regards to air quality global warming which greatly emphasis on fuel economy. Commercial vehicles consume very much amount of fuel. Both emission and fuel consumption by any vehicle are largely influenced by the efficiency of power train or transmission system of the vehicle. If the power losses are minimized by enhancing the efficiency of transmission systems then utilization of fuel can be increased to maximum extent and also the adverse effects on the environment can be decreased. Gearbox is major component of the transmission system not only in terms of its contribution to power losses, but also gearbox has high potential to improving the efficiency of overall power train or transmission system. In some applications like wind mill turbine or paper industries, the shaft and gears of gearbox rotating at very high speed. Almost 40% of maintenance cost of entire plant goes for maintaining the gearbox. Also if losses in gearbox reduces, the efficiency of overall plant and transmission system may highly increases. Continuous breakdown of gearbox in these plant occur, which is mainly due to temperature increases in the gearbox just because of very high rotational speed. And due to this increase in temperature, the viscosity of oil decreases drastically which increase the mechanical power losses in gearbox, which is mainly due to friction between the gears and sometimes due to excessive friction failure of gearbox may occur. This problem can be overcome by using some material which can't let the viscosity of the oil goes down with rise in temperature i.e. increases the viscosity index of the oil and also prevents the rise in temperature of oil at high rotational speed. This problem can be minimized by using some materials in the oil which enhances its thermal conductivity and viscosity index such as nanoparticles. So for more efficient transmission system creates interest in fundamental understanding of the factors which causing power losses in the transmission system before any steps like logical and cost-effective could be taken to minimize them. It is safe to categories the power losses of gearbox in two group. The 1st group includes all the losses associated with the torque transmission. These mechanical or load-dependent losses are mainly induced due to the friction between the contacting interfaces of the gearbox. The meshing gears and the rolling bearing element creates multiple contacts, which significantly contributes to the mechanical power losses of the transmission or gearbox (Seetharaman et al. 2008). The 2nd group of power losses is mostly due to the interaction between the components like rotating gears and bearing with the surrounding medium (such as oil, air and mixture of oil and air) (Seetharaman et al. 2008). These power losses are independent of load transmitted and generally known as spin power losses. This spin losses are mainly effected mostly by some factors like rotational speed, oil volumes, transmission temperature, oil viscosity, and the geometry of the gearbox housing.

1.2 Objectives

In view of above all limitations following objectives in this study has been taken:

(1) To quantify both mechanical and spin power losses.

(2) To obtain the optimum operating condition to enhance the overall efficiency of gearbox.

(3) To study the effect of nanoparticle in addition of lubricant.

CHAPTER 2: LITERATURE REVIEW

2.1 Gearbox

Gearbox is basically a power transmission system. Transmission system (gearbox) can be defined as an assemblage of parts, including the speed-changing gears and the propeller shaft, which is responsible for transmitting the power from an engine to a live axle. Transmission actually means the whole drive train, including clutch, gearbox, propeller shaft (for rear-wheel drive), differential, and final drive shafts. But it is generally used to refer the gearbox that helps in speed and torque conversions from a rotating power source to another device, using gears and gear trains. Gearbox makes the vehicle to function smoothly by converting the engine’s high speed to the slower wheel speed, thus increasing torque. Transmission also provides the multiple gear ratios which make it easier to switch between the gears as speed increases/decreases. It can either be done manually or automatically. In motor vehicles, internal combustion engines cannot run below a specific speed, so transmission is connected to the engine crankshaft via flywheel and/or clutch and/or fluid coupling. The motor output gets transmitted to the differentials via the driveshaft, which helps in driving the wheels. Differential also helps in the gear reduction.

The output shaft of a gearbox rotates as the torque increases. Nowadays, modern gearboxes are specifically used to increase the torque and lower the speed of a prime mover output shaft. The same gearbox can also do just opposite of this by switching different gear ratios i.e. decrease the torque and increase the shaft speed. Gearboxes are used in several areas such as wind turbines, automotive, agricultural and industrial equipment etc. Cast iron and aluminum are used as a material to construct gears and transmissions used in automobiles and truck. However aluminum is generally preferred due to its lower weight.

2.2 Types of transmissions

Transmissions can be categorized into manual, automatic or semi-automatic transmission.

2.2.1 Manual Transmission

These are further divided into two types:

1. **Sliding mesh gearbox:**

   In this gearbox, straight-cut spur gears are synchronized by matching the engine's revolutions to the road speed, so that the noise and the clashing of gears can be avoided. Here, double clutching is required to switch from one gear to another as until and unless gears are shifted, pairs aren't meshed. That's why it is also called as unsynchronized/non-synchronous.

2. **Constant mesh gearbox:**

   This gearbox make use of the diagonally cut helical gears which are always in the mesh state and switching of gears is done by fastening different pairs of gears to input and output shaft. That's why they are also called synchronized/synchromesh systems. They can also be non-synchronized.

2.2.2 Automatic transmission

Automatic transmissions work mainly by using hydraulics for selecting the gears. An optimum gear ratio is chosen automatically. A fluid flywheel or torque convertor is put in between the engine and transmission, rather than a clutch. They are very simple and easy to use, though expensive.

2.2.3 Semi-automatic transmission

It’s the combination of manual and automatic transmission as sometimes it may be required to manually control the gears and an integrated control system manages the clutch. This is sometimes called a "clutch less manual", or "automated manual" transmission. Most of the transmissions has the provision of shifting the gears by using the control system as if it was an automatic transmission system.

2.3 Lubrication and Cooling

Lubrication reduces the wear and rubbing between the moving parts of the machine. It helps in minimizing the power loss which occurs due to the heat generated between the meshing parts. Therefore, a cooling system is needed to dissipate the heat generated and to provide the cooling in the engine, so that temperature can be maintained. Two types of cooling systems can be used, air cooling and liquid cooling. Generally, the liquid cooling is preferred. The space between the bearings and journals is filled with the lubricating oil. This oil helps in cushioning the load, when the load strikes against the bearings. It immensely helps in minimizing the wear. When the lubricating oil passes through the engine, it acts as a cleansing agent by mixing with the metal and carbon particles. Larger particles settle at the bottom while smaller particles get filtered.

2.4 Nanofluids

With the advent of nanotechnology, nanoparticles came into being which was said to possess superior thermal properties as compared to conventional micro particles. Nanoparticles dispersed into the base fluid are called nanofluids. Nanoparticles could be made up of various materials, such as metals (Cu, Ag, Al), metallic oxides (Al_2O_3, CuO), nitride ceramics (AlN, SiN), carbon nanotubes etc., sizes varying from 1-100 nm while the base fluids could be water, oil, ethylene glycol, gear oil etc. Choi (1995) firstly introduced the term nanofluids. Conventional particles pose a major problem of rapid settling of particles, abrasion, clogging of the micro channels. To overcome this, nanoparticles played a very significant role. Due to its higher surface area and its nano size, nanoparticles remain suspended for a long time and thus improve heat conduction properties. Other than this, nanoparticles also minimized the pumping power required to maintain its flow through the channel and the pressure drop. The dawn of nanotechnology will lead to miniaturization of the system by designing the compact and light weight heat exchangers. Nanofluids can be considered as the next generation coolants due to its enhanced heat transfer capabilities. Nanofluids help to achieve the highest possible thermal properties at the smallest possible concentrations by uniform dispersion and stable suspension of nanoparticles in the base fluids (Choi et al. 2001).

Table 2.1: Comparison between micro particles and nano particles (Das et al.2006)

	Microparticles	Nanoparticles
Stability	Settle	Stable
Conductivity	Low	High
Surface/Volume ratio	1	1000 times larger than microparticles
Erosion	Yes	No
Pumping power	High	Small
Clogging in microchannel	Yes	No
Nanoscale phenomena	No	Yes

Nanoparticles can be fabricated by two ways: Physical method and Chemical method. Physical method incorporates mechanical grinding, inert condensation etc. while chemical method includes chemical precipitation, spray pyrolysis, thermal spray, micro emulsions etc. One step and two step methods are used to produce stable nanofluids with more conductivity (Xie et al. 2005).

The first step method produces and disperses nanoparticles simultaneously into the base fluid. In this method the agglomeration of nanoparticles get reduced to an extent. But this method is not successful at commercial volume. In two step method, firstly nanoparticles are made by physical or chemical process and then disperse into the base fluid. In two step method, nanoparticles tend to agglomerate. Ultrasonic agitation or surfactant addition to the nanofluids helps in minimizing the particles agglomeration and stabilize the fluids. This method is reliable and has already been used in industrial levels (Wang et al. 2006).

2.5 Previous research work related to Power losses of gearbox and nanoparticles

A significant number of studies has been available on power losses and efficiency of gear pair and gear trains, which has been reviewed in papers by Martin (1978), Yada (1997), Li and Seireg et al., Seetharaman and Kahraman (2008), and Seetharaman, et al. (2008). A very few literature review has been available on the power losses and efficiency of manual power transmission. Heingartner et al. (2003) experimentally studied the power losses of helical gear transmissions and proposed a mathematical model for the validation. Power losses were divided into speed and load dependent losses. The speed-dependent losses were further divided into windage losses, churning losses, bearing churning losses and seal losses while the load dependent losses were categorized into sliding friction losses, rolling friction losses and bearing losses. Sliding friction losses were evaluated with the help of the instantaneous sliding velocity and the friction force, which depend on the normal tooth load and the instantaneous coefficient of friction while the rolling friction losses were found by the instantaneous rolling velocity and the instantaneous lubricant film thickness. Experiments were performed for the same at different load volumes at the constant speed. It was observed that at a constant speed, sliding losses increases with the increase in load and rolling friction losses decreases and the windage losses remained constant.

Changenet et al. (2006) performed a theoretical and experimental comparison of power losses in a six-speed manual gearbox. They separated the power losses into two categories: Load dependent losses, which take into account the friction at mating teeth and rolling element bearings and No-load losses, which take into account oil churning & oil shearing in journal bearings and synchronizer cones. Windage losses were not incorporated. Transmission efficiency was found to be higher at elevated temperatures and at higher loads, when load and temperature effects were considered.

Handschuh et al. (2003) experimentally and analytically performed the comparison of the efficiency of high-speed helical gear trains used in a tilt-rotor aircraft. In a closed-loop power circulation arrangement, two identical gearboxes were aligned. The parameters taken were load and speed. At high load and high speed state, windage losses were found to be comparable to the gear meshing losses. It was found that the power losses enhanced on increasing the gear rotational speed. On the contrary, torque which is transmitted by the gear train has no significance as compared to the effect of rotational speed.

Dongen (1982) studied manual and automatic transmissions to compute the effect of speed and load on transmission efficiency. Temperature was taken constant. At different load and speed volumes, loaded and unloaded tests were conducted. He reported that efficiency enhances with the load and maximum efficiency of 96% can be achieved by a manual transmission. In the manual transmission losses, oil churning, bearing friction and load dependent gear mesh losses contributed significantly.

Barzaghi et al. (1995) did a test where two manual transmissions were used in the same test rig. One acted as a speed reducer and the other as an overdrive. Efficiency was evaluated on the basis of the torque measurements. It was observed that at the diminished load conditions, efficiency was found to be less and higher efficiency was reported at the higher load conditions on using lubricants with anti-friction additives.

Greenbaum et al. (1994) examined the effect of the inclination angle of the transmission and the transmission oil temperature. Efficiency was found to vary from 88-99.6%. On raising the angle of inclination, efficiency decreases. Five degrees of inclination caused 1 % difference in transmission efficiency. As the oil’s temperature increased, efficiency value scaled up. As the temperature increased from 50 to 75° C, there was a 6% raise in the efficiency. Torque was considered as the major factor in deciding the transmission efficiency as compared to the speed.

Kluger et al. (1995) carried out a study to attain the improvements in the efficiency by incorporating some changes in the design of the transmission improvements under actual duty cycle requirements of passenger vehicle. A lube pump and het lubrication method was considered to reduce the oil churning losses to enhance the manual transmission efficiency. Windage losses can also be reduced by placing a synchronizer on the input shaft to disengage the layshaft when in the 4th gear. A dual layshaft transmission was also recommended where the windage losses from the nearby gears can be reduced, when not in use.

Kluger et al. (1999) studied the five speed manual transmission to calculate the average value of efficiency (96.2%). In this study, numerous methods of enhancing the efficiency were proposed. Some of them were increasing the layshaft speed which helps in reducing the torque dependent losses, reducing the casting imperfections on the inside of the transmission case to lower the windage losses and so on. To reduce the parasitic losses, steel ball bearings were replaced by the ceramic ball bearings. All these modifications were said to have increase the transmission efficiency by 0.5%.

Pedro et al. (2014) used a two stage multiplying gearbox with helical gears and four different wind turbine gear oils (a mineral, a mineral + additives, a poly alkylene glycol, a poly alpha olefin) were tested. In this a test rig was set and test was performed using these oil at different rpm (100, 200, and 400) and different torque Nm (500, 750, and 1000). All these oil have same viscosity at 40^0C. At different operating condition of different torque gives different oil sump temperature which causes the different viscosity, density, and coefficient of friction at these different temperature and due to these power losses (gear, rolling bearing, seal) in the gearbox occur. Result showed that out of these four turbine oils a poly alkylene glycol have good viscosity index, density and low coefficient of friction even at a serve operating condition of 100^0C. By using poly alkylene glycol 20 % power losses was reduced.

Pedro et al. (2013) used a two stage multiplying gearbox with helical gears and four different wind turbine gear oils (a mineral, a mineral + additives, a poly alkylene glycol, a poly alpha olefin) were tested. In this they found the difference in oil sump temperature and wall of gearbox temperature, and difference between the oil sump temperature and room temperature at different input speed (100,200, and 300-500) rpm and torque. At these speeds different heat transfer coefficient and Reynolds number are occur. At 100 rpm oil flow in sump behave like a laminar flow and have low Reynolds number and, at 300-500 rpm behave like turbulent flow and have high Reynolds number, and at 200 rpm there occur a transition zone. The poly alkylene glycol showed the lowest operating temperature and mineral oil showed highest operating temperature.

Carlos et al. (2014) studied the torque loss in C40 gears of the gear box in no load condition by using five type of turbine oil in the FZG test rig. The oil used was mineral oil (MINR), ester oil (ESTR), mineral oil + additive (MINE), poly alkylene glycol (PAGD), and poly alpha olefin (PAO). The test was conducted at different load in Nm (k1=4.95, k5=104.97, k7=198.68, k5=323.27) and at different rpm 200, 400, and 1200. The oil volumetric flow was around 3 L/min at temperature of 80^0C. The result showed that up to load k5, MINE oil generated the lower friction torque loss, no matter the rotational speed selected. MINR oil generated the highest toque loss than other oil at 200-400 rpm. At 1200 rpm PAGD oil showed the highest torque loss generated in no load condition. At k7 load the highest torque load was generated in MINR and PAGD showed highest viscosity index at 200-400 rpm so it showed lowest torque loss.

Carlos et al. (2013) studied the friction torque loss and power loss in the thurst ball bearing lubricated with different wind turbine gear oil. A modified Four-Ball machine was used to find out the friction torque loss/power loss, operating and stabilization temperature, film thickness and coefficient of sliding friction in the thrust ball bearing at axial load of about 7000 N and rotation speed in the range of 150-1500 rpm. The thrust ball bearing worked in the mixed film thickness. The total friction torque loss in the thrust ball bearing above 500 rpm decreased when operating temperature and speed increased for the lubricant (MINR, ESTR, PAGD and PAOR).

Below 1000 rpm the poly alkylene glycol oil produced higher sliding coefficient of friction than other oil and above this rpm mineral oil produced higher sliding coefficient of friction.

Greco et al. (2011) studied the bearing from fully utility-scale wind turbine gearbox. Four different bearings from gearbox was taken out of which (A,C and D) as from high speed output shaft of the gearbox which rotated at around 1600 rpm and bearing B was from main shaft of the gearbox and this rotated at 10-15 rpm. Bearing B and D was tapered roller and B was case carburized (380-575 HV). Bearing A and C was cylindrical roller and made from hardened steel (675-720 HV). All bearings was taken after the 18,000h of operation. This study showed that three bearings (A, C, and D) exhibited axial cracks on the surface of bearing raceway. Each this cross-section was etched by using 3% of nital etchant. Irregular microstructure changes in the surface looked white by applied etchant because this area was more resistant to etching process, and this area namely as white etching areas (WESs). The nano-indentation of this WEA showed increase in 45% of hardness compared to the surrounding matix. Plastic deformation also happened in the bearing due to the squeezed out of the lubricant from the contact area and due to removal of protective oxide layer.

Martins et al. (2009) studied the power loss performance of carburized and high pressure nitride steel gears in FZG test rig, when lubricated with a commercial mineral gear oil containing a special additive package and a biodegradable low-toxicity ester based fluid at different rpm and different load. The result showed that the nitride steel gears showed lower stabilization temperature as compared to carburized steel when the mineral oil was used. When ester oil was used the nitride and carburized steel gears presented similar stabilization temperature. The nitride steel gear presented higher mass loss and more damaged surface than the carburized steel gear, irrespective of the lubricant. The nitride gears could not stand so high contact pressures as the carburized gear due to its lower surface hardness. The ester biodegradable oil showed lower stabilization temperatures and lower mass loss than the mineral lubricant for both materials. The type of lubricants promoted a largest difference in mass loss than the type of material. The carburized gear combined with ester oil showed the best overall result among both the gears.

Tiago et al. (2011) studied the friction torque loss and power loss in thrust ball bearing lubricated with different greases in a modified four ball machine. The friction torque and operating temperatures in thrust ball bearing were measured during the test at axial load of 7000N and 500-2000 rpm. Six different lubricant grease were tested. These greases formulated from different base oil (mineral oil, ester oil and polyalphaolefin base oil) and thickened by using thickener (lithium, calcium and propylene). At the end of each test a grease sample was analyzed through ferrographic techniques in order to quantify and evaluate bearing wear and also found the lubricant parameter value at operating temperature. Lubricant parameter greatly influenced the viscosity of the base oil because it strongly dependent on the nature of the base oil. Results showed that the base oil with lower lubricant parameter had lower coefficient of friction and lower specific film thickness, and base oil with higher lubricant parameter had higher coefficient of friction and high specific film thickness and showed higher sliding torque.

Carlos et al. (2013) studied the torque loss in thrust ball bearings lubricated with wind turbine gear oils at constant temperature. Six different wind turbine oil were tested in modified four-ball machine in which oils were maintained at constant temperature of 80^0C at different load (700N and 7000N) and different rpm (75-1200). The operating temperature of oil must be selected by two things, the highest possible to decrease the friction torque generated and promote a film thickness that protect the surface. Result showed that the specific lubricant film thickness inside the thrust ball bearing increased when the operating speed increased. The total friction torque increased when the operating speed increased for the two different axialloads, although its values are much higher for higher axialload. Mineral oil with additives generated significantly lower friction torque than other oils mainly because of additives in its composition. Mineral oil generated highest friction torque at low speed and polyalkylene glycol oil generated highest friction torque at higher speed because of significantly change in viscosity index.

Martins et al. (2006) studied the friction coefficient in FZG gears lubricated with industrial gear oils. Mineral oil with additives and biodegradable ester oil was used to compare and determine the coefficient of friction between the gear teeth in the gearbox. Test was performed at different load (5-500Nm) and different rpm (1000, 2000 and 3000). At no-load tests, performed at very low torque (5Nm), the stabilization temperature dependent mainly on the viscosity and the specific weight of lubricating oils. Results showed that churning losses for both the lubricating oil are identical. The ester oil generated smaller stabilization temperature for the same operating conditions, suggesting it promoted smaller coefficient of friction between the gear teeth. The mineral oil operated above the 100^0C for most of the time which showed excessive amount of wear when compared with ester oil. The mass loss along test was also larger for mineral oil.

Patel et al. (2006) studied the power losses and efficiency of an automotive manual transmission was investigated experimentally at variety of operating conditions. The power losses were measured of manual transmission at loaded and unloaded condition. The operating conditions including load, oil viscosity and oil volume on load-dependent (mechanical) and load-independent (spin) power losses of the transmission. Result showed that total power loss increase with increase of load and decreased with increase gear stage. The spin power loss was constant through each stage so was independent of load and mainly depend on oil churning and oil shearing in gears and bearings. Mechanical power loss increased with the load and depends mainly on friction between matting teeth and roller bearing. Spin power loss increased with speed and higher oil volume. Spin power loss decreased with low viscosity but mechanical power loss (due to friction) decreased with low volume of oil and higher viscosity.

Madhusree et al. (2013) studied the effect of CuO nanoparticles concentration (with different volume fraction) on viscosity of the gear oil. Viscosity showed great dependence on loading of nanoparticles in the base fluid (gear oil) and on the temperature between 10^0C to 80^0C.The viscosity of the nanofluid increased by nearly 3 times of the base fluid (gear oil) with increase in the fraction of CuO nanoparticles in the gear oil. Shear thinning behavior also become prominent when volume fraction of CuO loading is increased in nanofluid. Newtonian feature of the gear oil also changed to non-Newtonian with increased volume fraction of CuO in the base fluid (gear oil). The increased viscosity of the nanofluid by the addition of nanoparticles in it was because of increased in the fluid’s internal shear stress.

Madhusree et al. (2010) studied the effect of AL_2O_3 nanoparticles (average mean size is less than 50 nm) on the thermal conductivity and viscosity of car engine coolant. These values compared at different concentration (.001, 0.004, 0.007, 0.01, 0.015, 0.025, and 0.035) and at different temperature (10^0 C-80^0C). The results indicate that the addition of 0.035 volume fraction of Al_2O_3 nanoparticles in the engine coolant enhances the thermal conductivity of the fluid. The enhancement in thermal conductivity of the nanofluid varies linearly with the volume fraction of the nanoparticles and reaches a maximum of 11.25% at 80^0Cfor the nanofluid containing 0.035 volume fraction of Al_2O_3nanoparticles. The engine oil with and without surfactant behave as a Newtonian fluid. The viscosity of engine coolant is increases with volume fraction of alumina and decreases with temperature. Newtonian behavior is observed for nanofluids with low Al_2O_3 loading (≤ 0.004) only at higher temperatures, while nanofluids with higherAl_2O_3 loading display non-Newtonian features throughout the measured temperature range.

Madhusree et al. (2011): studied the thermal conductivity of CuO–gear oil nanofluids as a function of interfacial layer and clustering. With increase in volume fraction thermal conductivity enhanced and maximum enhancement was 10.49% observed at room temperature which reaches to 11.9% at 80^0C. DLS (Dynamic Light Scattering) data of the prepared nanofluids confirms the presence of aggregates. The thermal conductivity of CuO–gear oil nanofluids at room temperature was predicted well considering the role of both nanolayer and the nanoparticles clusters. But with the temperature up to 800C the thermal conductivity enhancement was mainly due to Brownian motion of nanoparticle.

Pal et al. (2014) studied the thermo physical properties of α-and γ-Al_2O_3having size in the range of 30–50 nm in aqueous suspension. The study showed that α-AL_2O_3 was crystalline and γ-$Al_2O_3$was amorphous in nature. Result showed that 6%-10% enhancement in the thermal conductivity as compared with the de-ionized water but α-AL_2O_3 showed 4%-6% more thermal conductivity than the γ-Al_2O_3. The thermal conductivity of nanofluids enhanced with the sonication time (4-8 hours) was due to strong sonic waves break down the large cluster into smaller one. However after 10 hours of sonication the charged density of the particles might be increased and which leads to aggregation of particles due to increase force of attraction between the particles. The heat transfer rate of nanofluids also influenced by the temperature gradient. The Brownian motion and clustering of particle highly effected by the increase in temperature. With increase in temperature the Brownian motion and phonon vibration become stronger and enhance the heat transfer rate. Viscosity highly depend on temperature and volume fraction of nanoparticle. At low loading (0.05%) suspension show Newtonian behavior.

Choi et al. (2008) analyzed the preparation and heat transfer properties of Al_2O_3 nanoparticle suspended in transformer oil. He concluded that particles agglomerated due to the hydrophilic surface properties. Oleic acid was used to change the property from hydrophilic to hydrophobic surface property. But the excess amount of oleic acid results in the reduction of thermal properties of nanofluid and cause physical and chemical instability. It was found that 0.5% volume fraction of nanoparticles can increase the thermal conductivity of the oil by 8% and the overall heat transfer coefficient by 20%.

Wenyang et al. (2012) studied the tri-biological and thermal behavior of Ag–MoS_2 Nanoparticle-based multi-component lubricating system. To enhance the properties of molybdenum disulfide (MoS2), silver (Ag) nanoparticles were embedded in it by the chemo-mechanical processing method, at three different concentrations (2%, 5% and 10%). It was found that by using the multi-component nanoparticles, coefficient of friction diminished from 0.10 to 0.08 and wear scar diameter from 0.462 to 0.293 mm and at a temp of 450°C, silver molybdate possess good strength of resisting the early failure of lubricating layer.

Madhusree et al. (2013) studied the thermo physical properties of Cu nanoparticles dispersed in the gear oil. The nanofluids formed at different concentration between 0.11 and 2% are prepared with oleic acid surfactant. Presence of agglomerated Cu nanoparticles in the prepared nanofluids is confirmed from Dynamic Light Scattering (DLS) data. Thermal conductivity and viscosity properties measured at different volume fraction and different temperature (10-80^{0}C). The result showed that at 2% volume of Cu nanoparticle the thermal conductivity enhanced by 24%. The formation of interfacial layer at the nanoparticle-liquid boundary and the ballistic transport of phonons across the percolating aggregates are believed to be responsible for the observed thermal conductivity enhancement. The viscosity was enhanced by 71% at 2% volume fraction. The viscosity is highly dependent on temperature. The nanofluid show Newtonian behavior at low concentration but show non-Newtonian behavior as concentration is increased by 2%.

Murshed et al. (2005) conducted experiments with two different shaped nanoparticles: cylindrical (φ10nm x40nm) diameter by length and other one is spherical (φ 15 nm) dispersed in water (base fluid). He found that at the volume fraction of 5%, thermal conductivity was enhanced by 30% to 33%. Surfactant named CTAB (Cetyltrimethyl ammonium bromide) of concentration (0.01%-0.02%) was used to stabilize the nanoparticles in the base fluid, without affecting the thermo physical properties of nanofluid. It was also observed that cylindrical shaped particles are more thermal conductive than spherical shape particles.

Michel et al. (2009) performed the experiments on alumina nanoparticles with two different base fluids (water and ethylene glycol). He found that enhancement in thermal conductivity decreases when the particle size is below 50 nm due to the fact that particles size becomes small enough for phonon scattering at solid-liquid interface. The correlation of thermal conductivity enhancement which depend on diameter of nanoparticle is given by

$$\in= (k - k1)/k1 = \in_{max} \, (1 - e^{-0.025d}) \tag{1}$$

$$\in_{max} = 4.4134\emptyset \tag{2}$$

$$\in_{max} = 5.527\emptyset \tag{3}$$

Where k is thermal conductivity of nanofluid, k1 is thermal conductivity of base fluid, d is 15 diameter of nanoparticle. $\emptyset$ is volume fraction. $\in_{max}$ is limiting value of thermal conductivity at any volume fraction as particle size increases and equation (2) & (3) is for nanoparticles in water and ethylene glycol respectively.

Lotfizadeh et al. (2013) conducted the experiments using ethylene glycol-water (60:40). He also studied the effect of the concentration of Sodium dodeobcyl benzene sulfonate (SDBS surfactant) on viscosity and thermal conductivity of nanofluids. It was observed that at the low concentration of SDBS (<1 wt. %), thermal conductivity enhanced while at higher concentration of SDBS (>1 wt. %), conductivity decreased and viscosity increased. It was concluded that viscosity was significantly dependent on temperature.

Xuan et al. (2005) investigated the effect of the variation of copper nanoparticles concentration with different volume fractions on heat transfer enhancement. On increasing the volume fraction and fluid flow velocity, convective heat transfer coefficient increased for Reynolds number between 10,000 and 25,000. At higher concentration, viscosity of nanofluids increases which in turn reduces the heat transfer enhancement as turbulence of the flow decreases. The heat transfer coefficient of nanofluid was found to be larger than that of base fluid.

Yasutsune et al. (2008) studied about the lubricant churning losses in spur gear system. In gear system, power loss usually consists of the tooth friction, lubricant churning loss, and bearing loss. All of these losses change into heat and affect the working of lubricant. Results showed that with higher operating power and speed, the operating temperature of gears become higher and the oil film thickness between the tooth surfaces decreases as its viscosity decreases. Highly viscous and large amount of oil increase the churning loss but also reduces the friction between the mating gears as well as the operating temperature.

Dan et al. (2011) carried out a study on lipophilic Cu nanoparticles which were synthesized by surface modification method to improve their dispersion stability in hydrophobic organic media. They were treated with O-di-n-cetyldithio phosphate to modify the properties of Cu nanoparticles. These modified Cu-nanoparticles were taken at different concentrations in different oils like kerosene, toluene and dehydro naphthalene. The result showed that the viscosities and thermal conductivities of nanofluids with the surface-modified nanoparticles had higher values than that of the base fluids. Appropriate amounts of Cu-nanoparticles added into a hydrocarbon fuel can enhance its thermal oxidation stability.

Chandrasekar et al. (2010) experimentally studied the thermal conductivity and viscosity of Al2O3/water nanofluid and validated these results with theoretical models such as modified Maxwell equation for thermal conductivity and Krieger–Dougherty equation (K-D model) for viscosity. Al_2O_3/water nanofluids with various volume concentrations (0.33% to 5%) were prepared with two hours of ultra-sonication. No surfactant was used. Thermal conductivity was measured using a KD2 Pro thermal properties analyzer while viscosity was measured Brookfield cone. The result showed that thermal conductivity of nanofluid increased linearly with volume concentration and viscosity exhibited nonlinear relation at volume concentration.

Kang et al. (2004) performed experiments on a conventional grooved circular heat pipe with water-based nanofluids containing 1 to 50 ppm of 35nm silver nanoparticles and measured the temperature distribution and thermal resistance. At the same charge volume, thermal resistance of the heat pipe with nanofluids decreased by 10% to 80% as compared to DI water at an input power of 30 to 60 W. Maximum reductions in the thermal resistance of heat pipe was 50% for 10nm silver nanoparticles and 80% for 35nm silver nanoparticles.

Lee et al. (1997) conducted an experiment using two kind of nanoparticles CuO (18.6 and 23.6 nm) and Al_2O_3 (24.4 and 38.4) suspended in two different base fluids: water and ethylene glycol (EG). At 4% volume fraction of nanoparticles, CuO/EG mixture exhibited a great increase in conductivity (>20%). While at the lower volume fraction range (<0.05), thermal conductivity ratios increased almost linearly with volume fraction.

Wang et al. (2003) took Al_2O_3 as nanoparticle and water as the base fluid with 5% of concentration. Increase in viscosity was found to be of 90% and did not observe any non-newtonian effect.

Das et al. (2003) examined the viscosity of Al_2O_3-water based nanofluid and noticed a little change in viscosity with the increase in shear rate. It also showed the Newtonian behavior. It was also found that viscosity depends on temperature and the nanofluid with the highest loading of nanoparticles had the maximum viscosity.

Tzeng et al. (2005) studied the effects of nanofluid in the cooling of automatic transmission. Transmission fluid was loaded with the CuO & Al_2O_3 nanoparticles as well as antifoam agents and used in a four wheel automatic transmission. It was concluded that at high and low rotating speed, CuO nanoparticles had the lowest temperature distribution.

Tsai et al. (2004) made the first move to demonstrate the significance of using nanofluids to enhance the thermal performance of heat pipe. He used gold nanoparticles (17 nm) suspended in water in a disk shaped miniature heat pipe and found that thermal resistance of the disk shaped miniature heat pipe was reduced by nearly 40% on using nanofluids instead of de-ionized (DI) water.

Chein et al. (2006) numerically investigated the performance of nanofluids (Cu nanoparticles + water) as coolants in silicon micro channels. He concluded that the overall performance of the micro channel heat sink was greatly improved due to the increased thermal conductivity and thermal dispersion effects. It was also ascertained that nanoparticles suspended in water did not cause much pressure drop.

Wang et al. (1991) stated that dispersion technique doesn't affect the measured thermal conductivity enhancement. The addition of polymeric surfactant decreases the conductivity due to the polymer molecules coating onto the surface of nanoparticles which increases the interfacial thermal resistance.

Eastman et al. (2004) investigated the increment in thermal conductivity of CuO-water nanofluid. He reported that at 5% volume fraction of CuO nanoparticles, thermal conductivity was measured approx. 60% as compared to the 30% increase in thermal conductivity of Al_2O_3-water nanofluids at the same concentration. The enhancement in thermal conductivity was linearly proportional to the particle concentration.

Sharma et al. (2009) conducted an experiment to evaluate the heat transfer coefficient for flow in a tube with twisted tape inserts in the transition range of flow with Al_2O_3 nanofluid. Al_2O_3 nanofluid had shown the extensive augmentation in the convective heat transfer. Results confirmed that the heat transfer coefficient of nanofluid flowing in a tube with 0.1% volume concentration is 23.7% higher than that of the water flowing in a tube.

Dan et al. (2011) reported that viscosity of CuO nanofluid decreased with the rise in temperature and with the increase in the concentration of nanoparticles, viscosity increased. Viscosity was found to increase till 5% with the addition of the nanoparticles with 1% of mass fraction. It was concluded that that the preparation of nanofluids has no major effect on the viscous resistance.

Duangthongsuk et al. (2009) performed a comparison between the viscosity of nanofluid and the base fluid. TiO_2 nanoparticles were suspended in the base fluid (water) with the volume concentration of 0.2-2 %. Temperature varied from 15ºC to 35ºC. It was concluded that viscosity of nanofluids was higher than that of base fluid by approx. 4-15%. With the increase in temperature, viscosity was found to increase as well. The increase in particle concentration also leads to the increase in viscosity.

Nguyen et al. (2007) stated the dependence of viscosity of nanofluids on both temperature and concentration. He studied the dynamic viscosity for the Al_2O_3- water nanofluid by taking Al_2O_3 nanoparticles of diameter 36nm-47nm dispersed into water at different volume concentration from 1-9.5% and temperature ranging from 22 to 75^0C. It was found that viscosity of nanofluid decreases with the increase in temperature of nanofluid.

Putra et al. (2003) performed an experiment for the viscosity of the Al_2O_3- water nanofluid at different concentration (1-4 vol %) and temperature varying from 20-50ºC. Viscosity of nanofluid was measured as a function of Al_2O_3 volume fraction. He reported that at the lower concentration of nanofluids, Newtonian behavior was observed.

Daxiong et al. (2010) studied the variation in viscosity of the Al_2O_3-water nanofluid by using nanoparticles of 60 nm and water as the base fluid. Volume fraction of nanofluid was was varied from 0.33-0.5% and temperature from 10-45ºC. Viscosity of the nanofluids was measured as a function of Al_2O_3 volume fraction. It was concluded that viscosity rises linearly up to 2% volume concentration and exponentially after 2%.

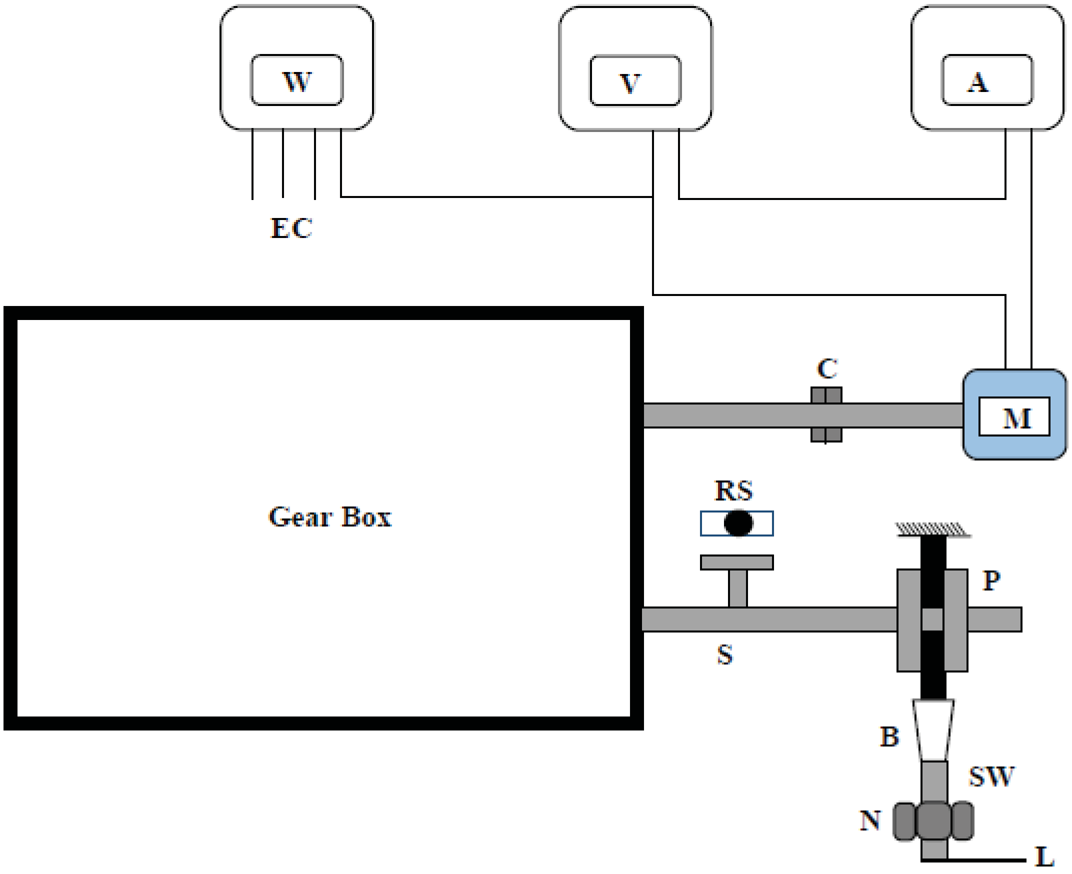
CHAPTER 3: TEST FACILITY AND EXPERIMENTAL PROCEDURES

3.1 Introduction

Gearbox is a transmission system which is used to transmit power from input shaft to output at desired output condition. Basically, gearbox consist of gears, gear train for torque and speed conversion so that the vehicle can easily start, stop, easily run at slow and high speed. Gearbox is also utilized in those application such as wind mill, where rotor is rotating at very low speed but output shaft is rotating at very high speed for generating electricity, for this purpose gearbox of high gear ratio is used like 20:1 or 50:1. Gearbox is also used in automobile industries from many decades. There occur some power losses during transmission of power from engine to the wheels or rotor to the generator shaft, and if these losses were minimized, the engine efficiency, transmission efficiency and industries or mill efficiency will also enhance. Literature survey showed that, the methods shown are not able to minimize the power losses expected in the gearbox. So to quantifying these losses in the gearbox a special-purpose test setup was fabricated for the direct measurement of power losses of an automobile manual transmission with varying load, oil volume, oils of different viscosity, and oil using nanoparticles at different concentrations.

3.2 Experimental setup

For performing the experiments for measuring the power losses in the gearbox, a test setup was developed. The schematic of test facility is shown in Figure 3.1 and the experimental setup is shown in Figure 3.2.



A: Ammeter
B: Belt
C: Coupling
L: Lever
M: Electric motor
N: Nut
P: Pulley

S: Shaft
V: Voltmeter
W: Wattmeter
EC: Electric cables
RS: RPM sensor
SW: Spring weight

Figure 3.1: Schematic diagram of experimental setup

Figure 3.2: Experimental setup

The test facility consist of a gearbox and an electric motor. The input shaft of gearbox was coupled to the shaft of electric motor. Hyundai accent gearbox (2006 model) was used in this experiment. Electric motor (Figure 3.3) has specification as follows rated power: 1 HP, rated current: 6.0 A. A pulley of diameter 13 cm was connected to the output shaft of gearbox. A nut and screw mechanism provided with spring weight having maximum load capacity of 100 kg was used to apply the load using a belt of thickness 6 mm and width 35 mm whose one end was fixed and other end was connected to spring weight. A tachometer (Figure 3.4) was used to measure the rpm of the gear box whereas a digital magnetic rpm sensor (Figure 3.6) was used to record the rpm of the output shaft of the gearbox.

Figure 3.3: Electric motor

Figure 3.4: Tachometer

Figure 3.5: Digital thermometer

Figure 3.6: Digital magnetic rpm sensor

For measuring the power supplied by the electric motor to the input shaft of the gearbox, a wattmeter was used. An ammeter and a voltmeter were also used to measure the power factor of the electric motor. A digital thermometer (Figure 3.5) was used to find out the rise in the temperature of the oil while doing the experiments at different oils and different concentration of nanoparticle based oil.

3.3 Properties of oil used

Company usually prefer to use 75W90 gear oil in the gearbox for their proper working. Viscosity and thermal conductivity are the properties of the oil which are highly effected by the temperature of the oil. Thermal conductivity was measured by thermal property analyzer i.e. KD2 Pro (Figure 3.7). It mainly work on the principle of transient hot wire method. In this there was a needle having very high length to the diameter ratio. This needle was placed in the fluid. Current was passed through the needle and according to the thermal conductivity of fluid, there was rise in the temperature of needle. If conductivity was high then there was less rise in temperature and if conductivity was low then there was more rise in the temperature of needle. According to the temperature, conductivity is shown in the display screen in W/(mk).

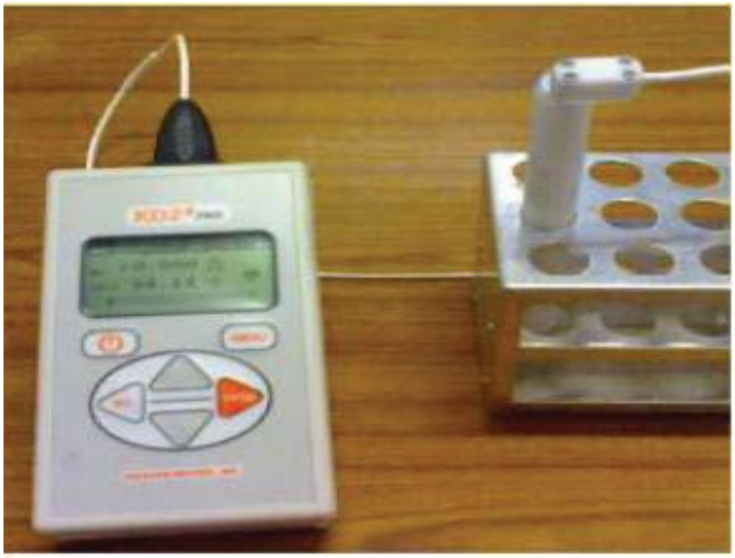


Figure 3.7: KD2 Pro

Figure 3.8: Brookfield rheometer

Viscosity is measured by the Brookfield rheometer (Figure 3.8) in which there was cup and cone arrangement. The oil was put in the cup and fit below the spindle of cone. Then the spindle was rotated. According to drag force on the spindle, it give the readings of viscosity on the display screen.

The measured values of conductivity at different concentration and at different temperature is shown in the Figure 3.10. Graph shows that thermal conductivity has increased with increase in temperature. At a particular temperature conductivity has also increased with increase in concentration of nanoparticles. The reason for increasing thermal conductivity is basically the brownian motion of the nanoparticle and the formation of nanolayer (Figure 3.9). Nanolayer is an Interfacial layer between the nanoparticle and base fluid, which is formed over the nanoparticle having conductivity higher than the base fluid but less than the nanoparticle. This nanolayer mainly helps the base fluid to enhance its conductivity.

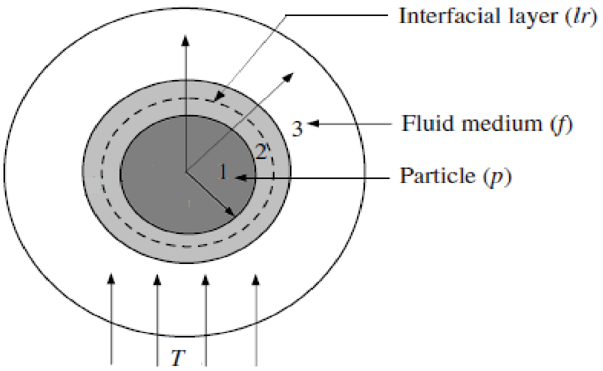


Figure 3.9: Nanolayer (Leong et al. 2006)

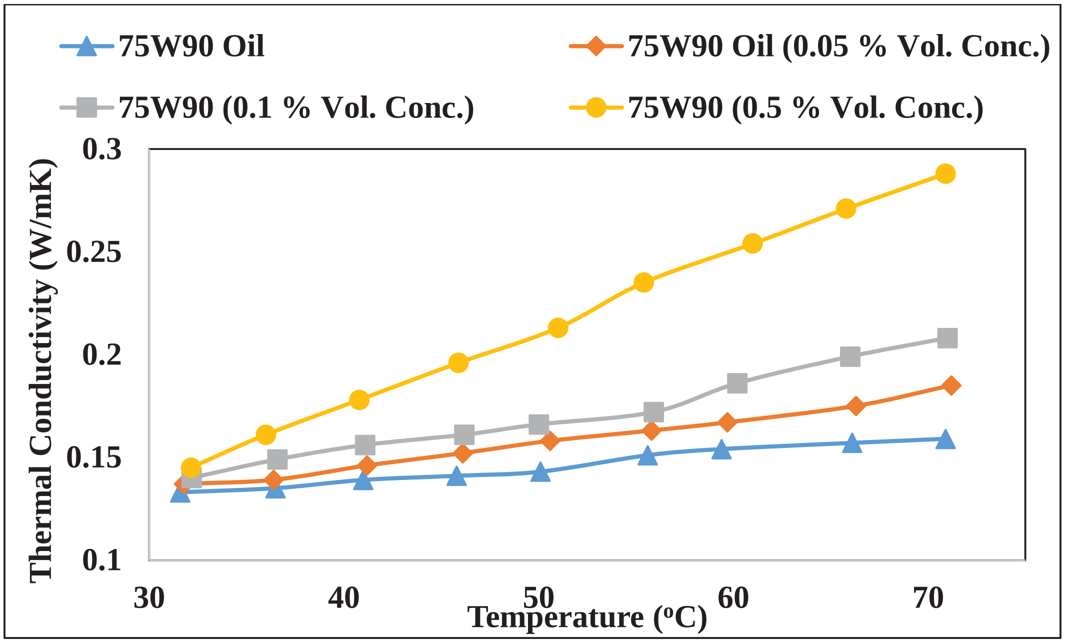


Figure 3.10: Thermal conductivity vs temperature at different concentration

Viscosity of each lubricating oil was measured using rheometer at room temperature. The measured values of viscosity are given in table 3.1 for all the oils. It can be seen from the table that 85W140 gear oil has highest viscosity, whereas lowest viscosity was obtained for 75W90 gear oil.

Table 3.1: Viscosity of oil used

Sr. No.	Oil	Viscosity (Pa-s)
1	Oil 1 (75W90)	0.075
2	Oil 2 (80W90)	0.0844
3	Oil 3 (20W50)	0.0919
4	Oil 4 (85W140)	0.3399

3.4 Calibration

A standard calibration procedure was followed to calibrate spring and magnetic rpm sensor. To calibrate the spring weight, a known weight (2, 5, 10 kg etc) was attached to it and its value is recorded from spring weight scale. This procedure was repeated periodically after performing the experiments at each oil volume. A standard calibration curve for spring weight is shown in Figure 3.11.

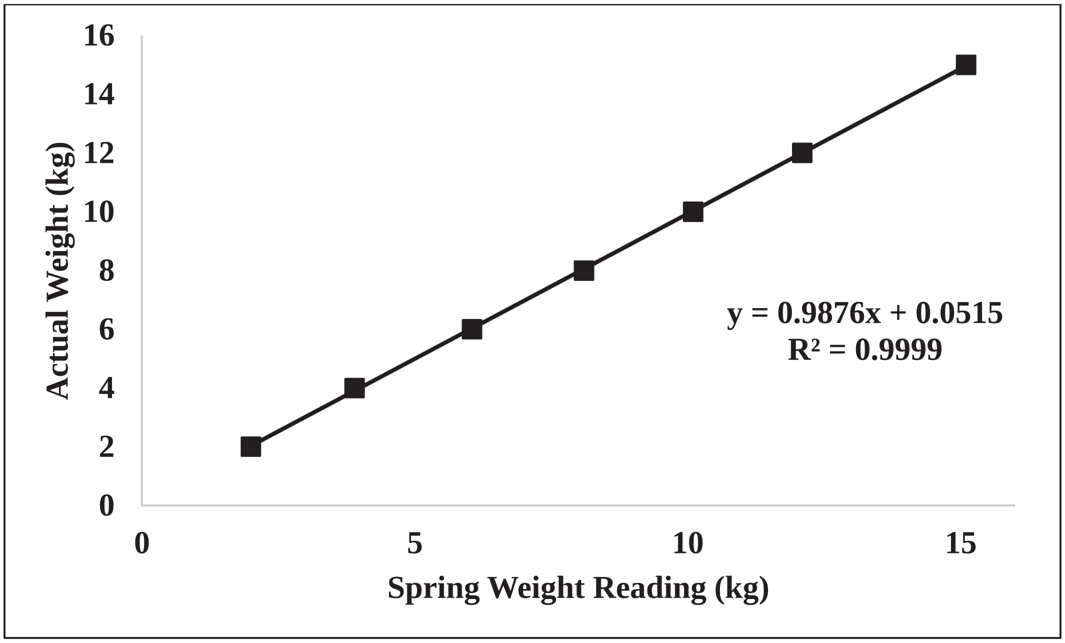


Figure 3.11: Calibration curve for spring weight

To calibrate the rpm sensor, rpm of output shaft was recorded by using tachometer (which was used as standard) and simultaneously the rpm reading of magnetic rpm sensor was also recorded. Rpm of output shaft was varied by shifting the gear ratio. Calibration curve for rpm sensor is shown in Figure 3.12.

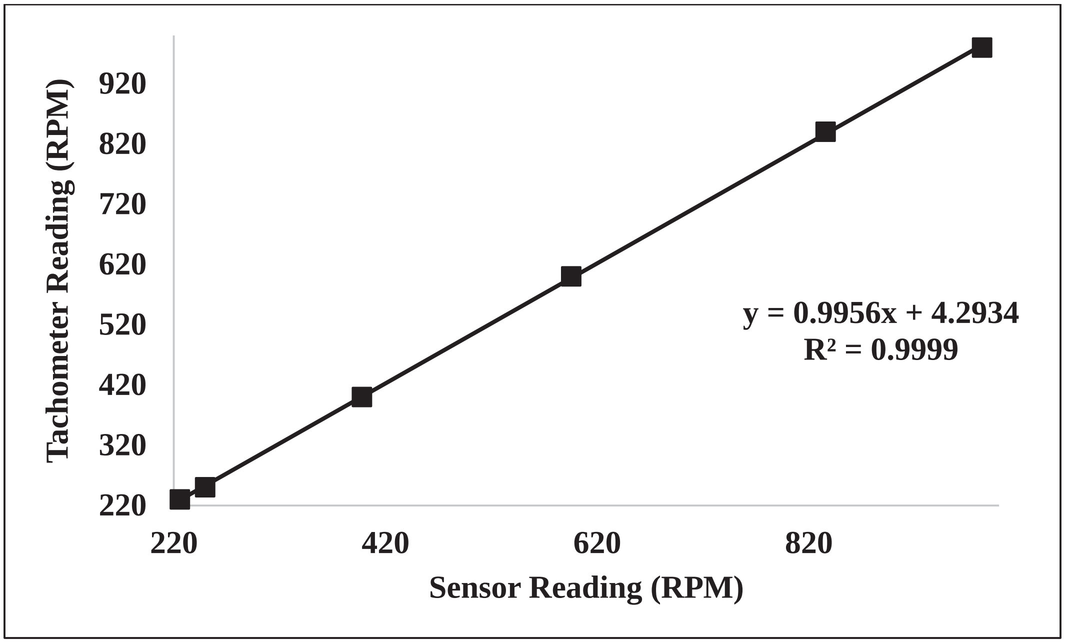


Figure 3.12: Calibration curve for rpm sensor

3.5 Operational procedure

For measuring the power losses in the manual transmission system, experiments were performed. First and foremost, gearbox was 100% filled with the 75W90 oil up to the mark by which 2.5 L of oil was used in the gearbox. Then the electric motor was started and 1st gear was pushed with the help of lever. Gearbox kept running at no load condition till the temperature reaches almost at its uniform point. Rpm of the input shaft was measured with the help of tachometer manually and rpm of the output shaft was measured with the help of digital magnetic rpm sensor. The readings from wattmeter, ammeter and voltmeter was also recorded. Next step was to increase the load from 1 kg to the point at which ammeter reaches the rated current of the electric motor. Note down all the values such as rpm of input and output shaft current, voltage, wattmeter reading, and the rise in the temperature of oil with load. As in this setup, there was no arrangement for cooling of belt and pulley arrangement, so with the rise in load, pulley becomes too hot. So before taking the next reading, pulley was cool down to room temperature and then, same experiments were performed for the next gear ratios and so on. The next readings were taken at 75%, 50% and 25% oil volume i.e. at 1.875 L, 1.25 L and 0.625 L of oil, respectively. The same procedure was followed for three other types of oil (20W50, 80W90 and 85W140 gear oil) and alumina nanoparticle (Average particle size: 20 nm)-75W90 oil nanofluid at different concentration of nanoparticles (0.05, 0.1 and 0.5%).

CHAPTER 4: SPIN AND MECHANICAL POWER LOSSES IN GEARBOX

4.1 Introduction

There are mainly two types of power losses that occur in the gearbox. First type of losses are the mechanical losses which are mainly load dependent losses and mostly occur due to the friction between the contacting surface of the gears and bearing. And second type of losses are the load independent losses and mainly known as spin losses and mostly due to the interaction between the components like rotating gears and bearing with the surrounding medium such as oil, air and mixture of both air and oil.

For this, experimental work are performed on the gearbox at different key operating parameters like load (kg), oil volume (V_{oil}), oil viscosity (μ_{oil}), nanofluids at different concentration, and four different types of oils to measure total power loss (P_T) , mechanical power loss (P_m) and spin power loss (P_S) as well as total efficiency (η_T) and mechanical efficiency (η_m) .

4.2 Effect of gear ratio on power losses

Mechanical power losses *versus* load

In Figure 4.1 the mechanical power losses P_m are plotted against the load for different gear ratio at 100% oil volume which means gearbox is having 2.5 L of oil. 75W90 gear oil is used here, which is mostly used for lubrication of gearbox. Figure 4.1 shows that the mechanical power losses decreases with increase in load for all the gears. But for instance, the measured mechanical power loss for particular load is higher for the higher gear ratio which means 5th gear has maximum power losses. This is due to the fact that at higher gear ratios, the rotational speeds of the gears are higher which increases the oil churning activity and imparts additional drag and damping to the motion of the gear. Also with increase in the load the rotational speed of gears decreases which is responsible for decreasing trend of mechanical power losses with load. The value of mechanical power losses at 3 kg load for 1st gear is 257.182 W and for 5th gear is 480.14 W (which is almost 86.6% more than the 1st gear).

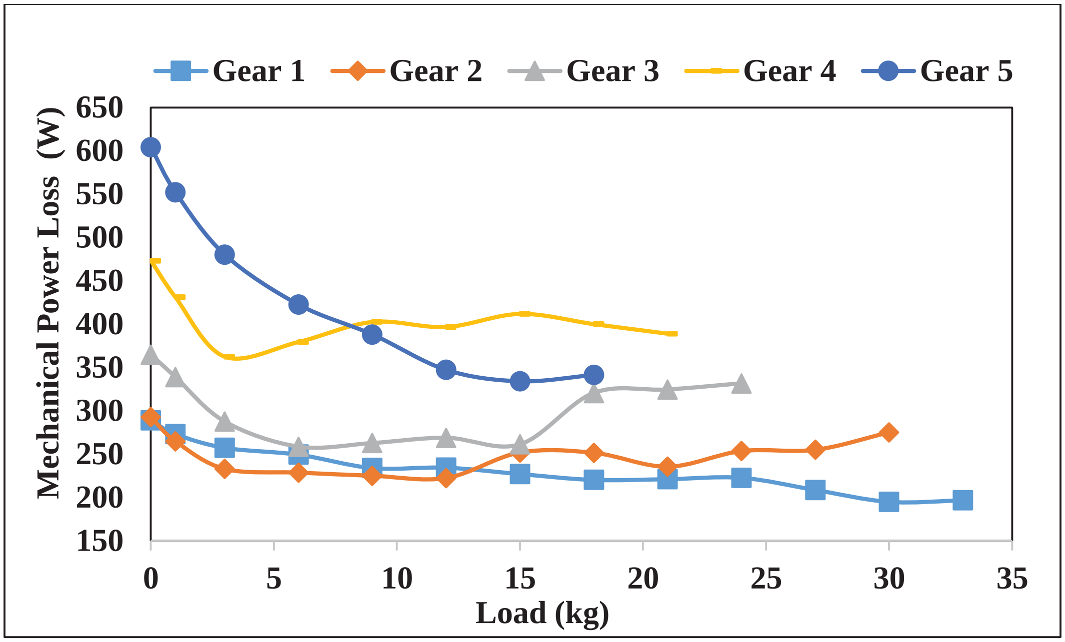


Figure 4.1: Mechanical power losses with load, for all gear ratios, 100% volume, 75W90 gear oil

At smaller gear ratios, motor has reached to its maximum amperes at higher loads, however at higher gear ratios the same point is obtained at the lower value of load. Hence the mechanical power losses are evaluated for the smaller gears upto 35 kg, whereas load values are limited to 18 kg for higher gear ratios.

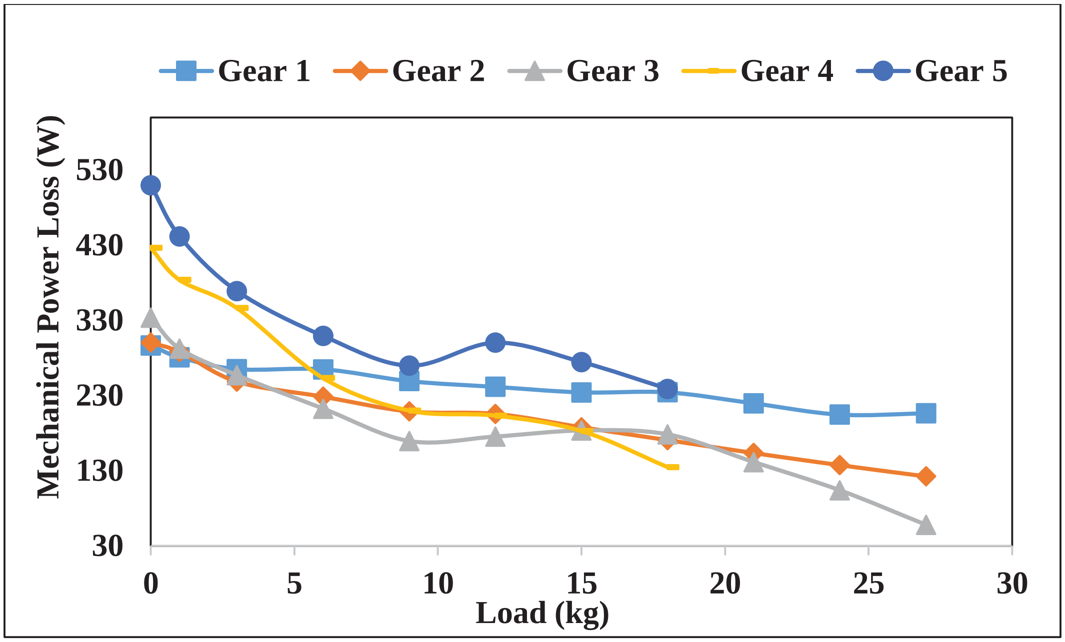


Figure 4.2: Mechanical power losses with load, for all gear ratios, 75% volume, 75W90 gear oil

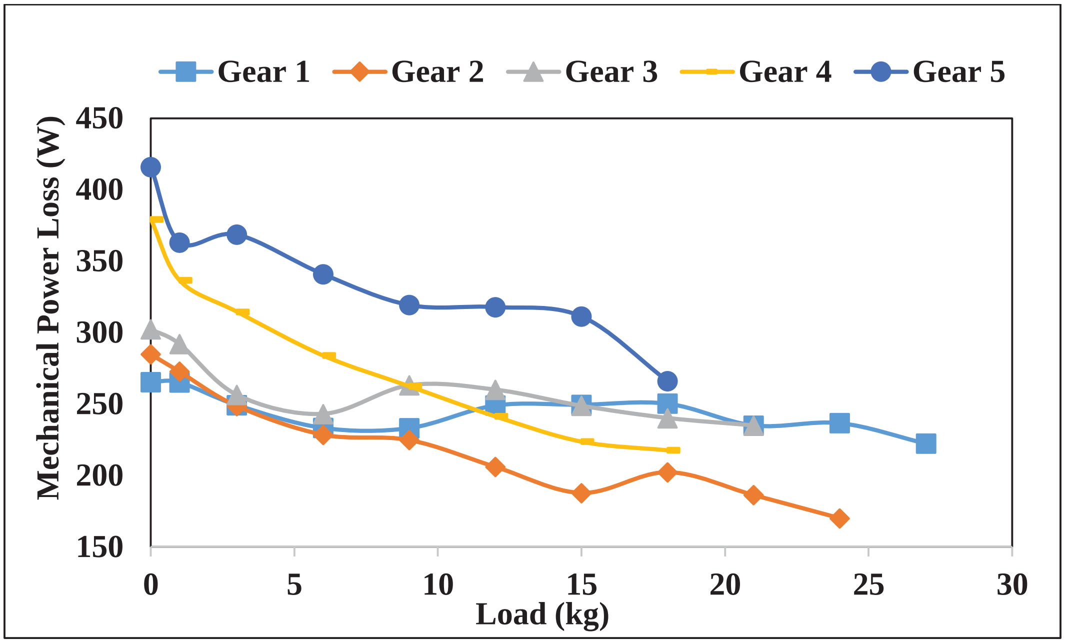


Figure 4.3: Mechanical power losses with load, for all gear ratios, 50% volume, 75W90 gear oil

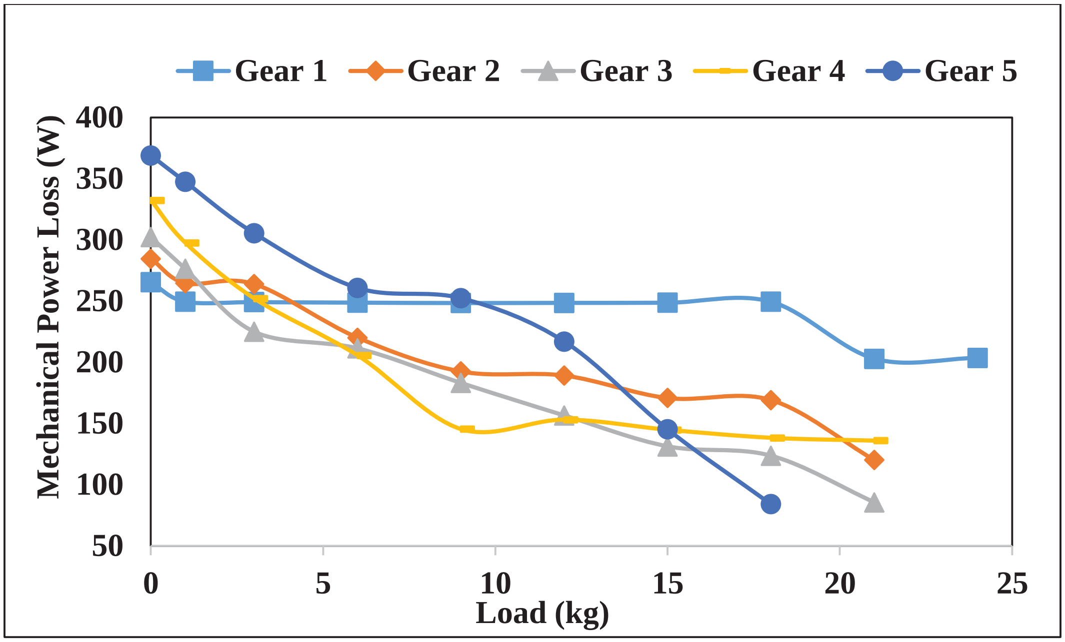


Figure 4.4: Mechanical power losses with load, for all gear ratios, 25% volume, 75W90 gear oil

Figures 4.2 to 4.4 shows mechanical power losses against the load at 75% oil volume (1.875 L), 50% oil volume (1.25 L) and 25% oil volume (0.625 L), respectively. The graphs show the same pattern i.e. mechanical power losses has decreased with increase in load. All the graph have shown that the mechanical power losses have increased with increase in load for reverse gear.

Spin power loss versus load

In Figures 4.5 and 4.6, spin power losses are plotted against the load at 100% oil volume and at 75% oil volume, respectively. The graph shows that the spin power losses has increased with increase in gear ratio. This increase in the spin power losses is mainly due to increase in the rotational speed of gears with increase in gear ratio which exerts the additional drag on the gears due to churning activity of oil.

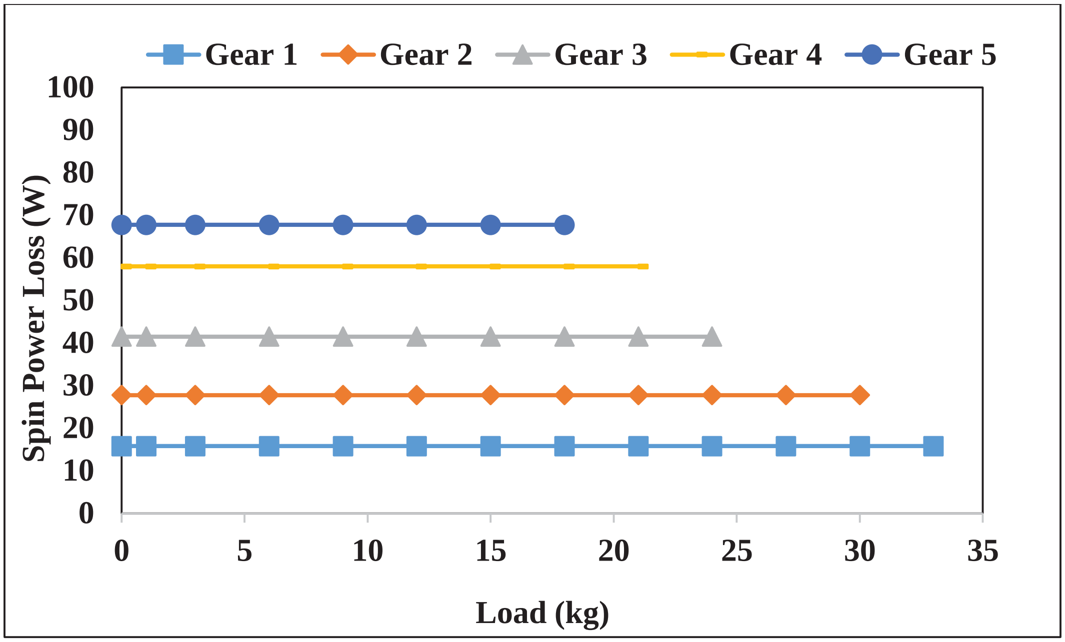


Figure 4.5: Spin power losses with load, for all gear ratios, 100% volume, 75W90 gear oil

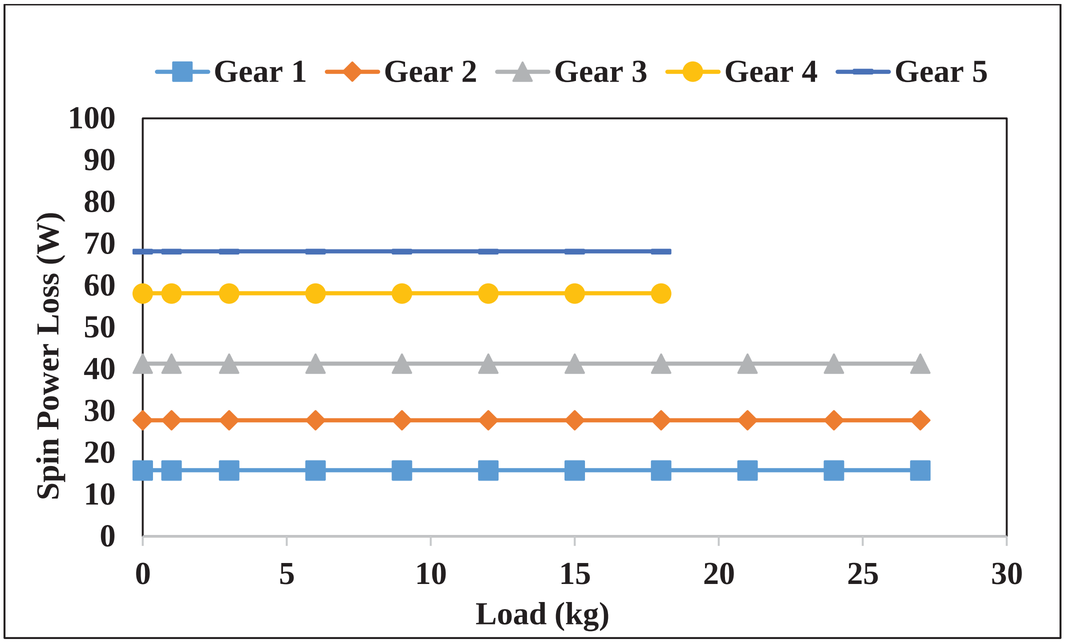


Figure 4.6: Spin power losses with load, for all gear ratios, 75% volume, 75W90 gear oil

4.3 Effect of oil volume on power losses

To evaluate the effect of oil volume on mechanical and spin power losses, experiments are performed by varying the load at a particular gear ratio and oil volume. Similar experiments are performed at all the gear ratios and oil volumes. The outcome of analyzed result is given below.

Mechanical power losses versus load

In Figures 4.7 to 4.10 the mechanical power losses are plotted against the load at different oil volume of 80W90 gear oil for different gear ratio. Graphs show that for all the gears 100% oil volume has maximum mechanical power losses and lowest mechanical power losses is observed when the gearbox is operated without lubricating oil. For smaller gears the best working condition is obtained at 50% oil volume and 75% oil volume, however minimum mechanical losses are obtained at 25% oil volume for higher gear ratios. At 100% oil volume the mechanical power losses for 1st gear ratio is 296.1 W and for 5th gear ratio is 730.7 W which is almost 146% more than the 1st gear ratio at 12 kg load. The lubricating oils are added to impart damping and reduce wear between the gears, but with increase in the volume of the lubricating oil, power losses increases. When the concerns of energy become more prominent than the durability or service life of gears, mechanical power losses can be decreased by decreasing the oil volume because with decrease in oil volume drag acting on the gear decreases.

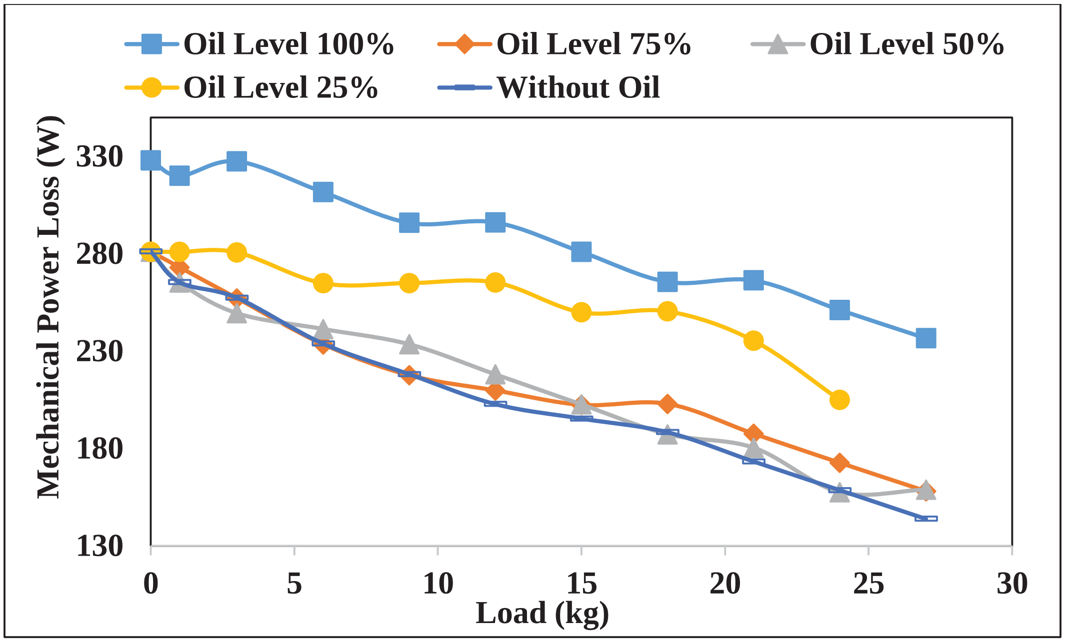


Figure 4.7: Mechanical power losses with load, for all oil volumes of 80W90 gear oil, 1st gear

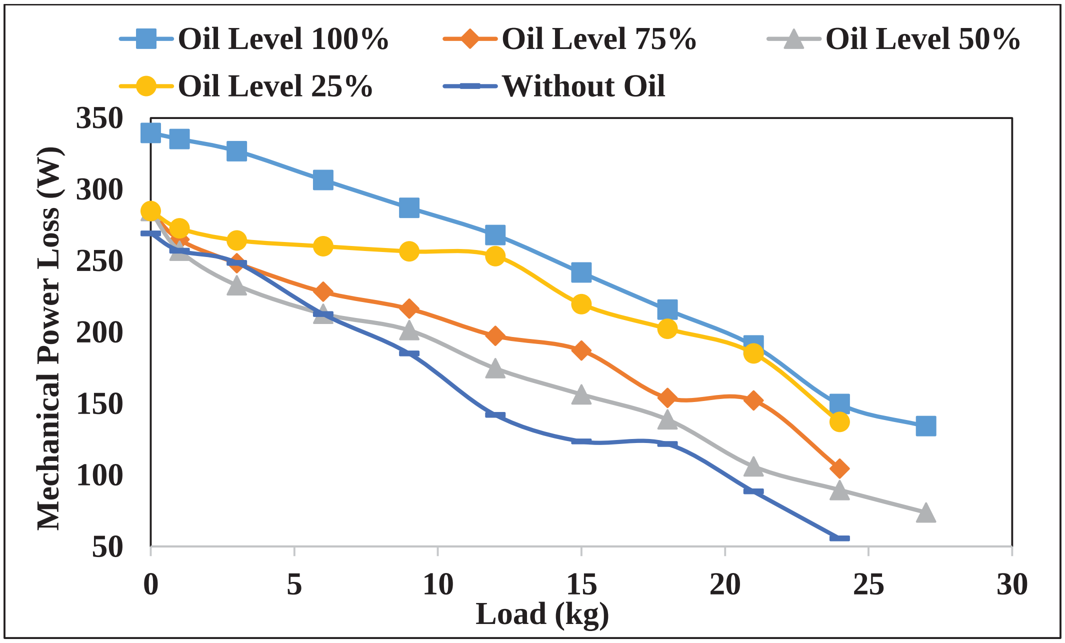


Figure 4.8: Mechanical power losses with load, for all oil volumes of 80W90 gear oil, 2nd gear

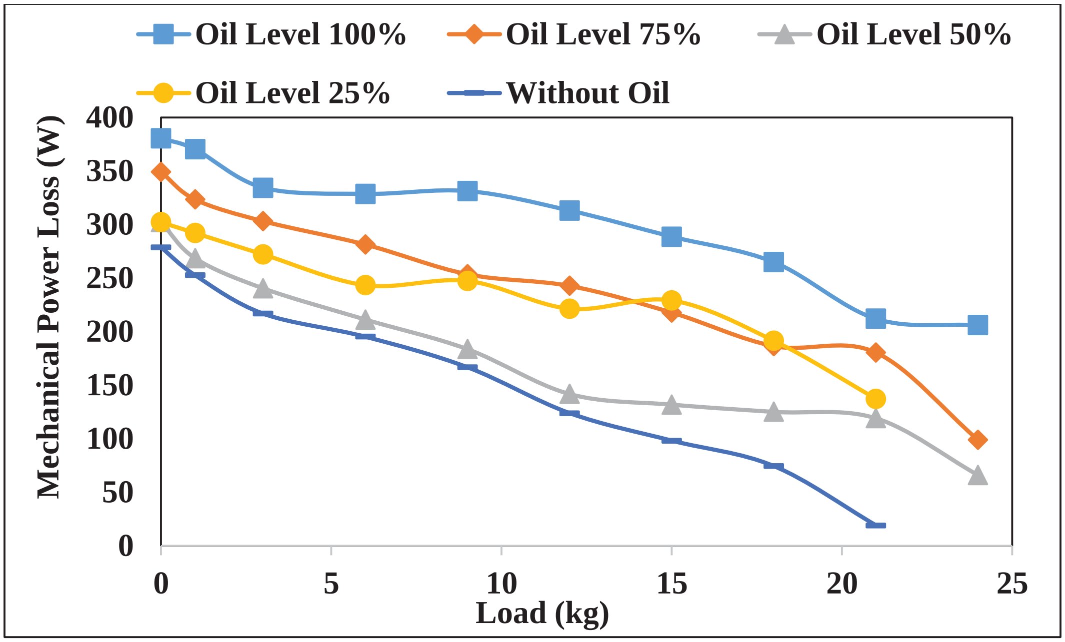


Figure 4.9: Mechanical power losses with load, for all oil volumes of 80W90 gear oil, 3rd gear

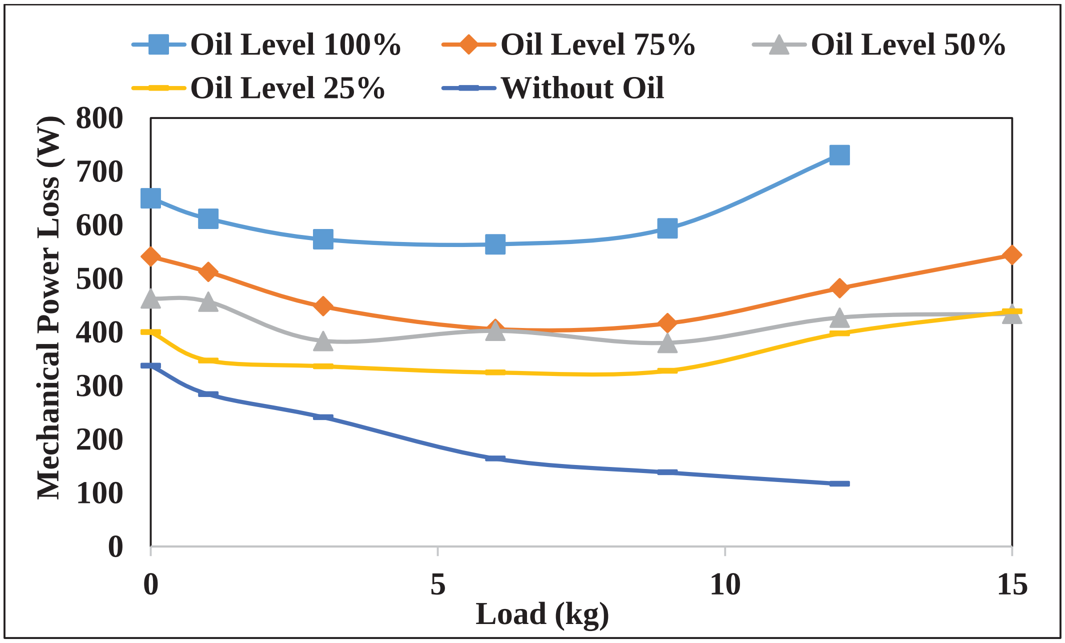


Figure 4.10: Mechanical power losses with load, for all oil volumes of 80W90 gear oil, 5th gear

Spin power losses versus load

Figure 4.11 shows spin power losses against load for gear ratio 2 at different oil volume. Graph shows that gearbox without lubricating oil has maximum spin power losses, whichis due to more rubbing of gears. Here graph is shown only for gear ratio 2 because all other gear ratios have shown similar trend. A very small increase in spin power has been observed when the oil volume is raised from 25 to 100%.

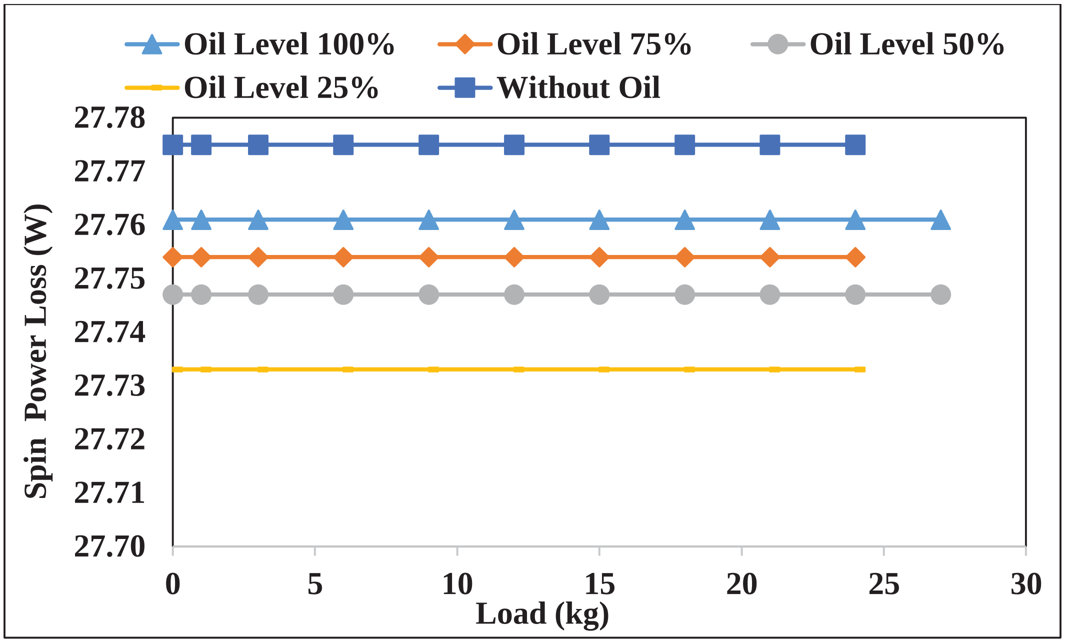


Figure 4.11: Spin power losses with load, for all oil volumes of 80W90 gear oil, 2nd gear

4.4 Effect of oil viscosity on power losses

For measuring effect of oil viscosity on the mechanical and spin power losses, four oils of different viscosity are used. Each oil is tested with increase in load at different gear ratios.

Mechanical power losses versus load:

In Figures 4.12 to 4.14, the mechanical power losses are plotted against the load for different oils having different viscosity at 100% oil volume and gear ratios 1, 3 and 5. It can be seen from the figures that 85W140 oil (viscosity: 0.339 Pa-s) has shown maximum mechanical power losses at all the gear ratios. 80W90 gear oil (viscosity: 0.0844 Pa-s) has intermediate power losses, whereas the mechanical power losses have further decreased by using lower viscosity oil such as 20W50 (viscosity: 0.0919 Pa-s) and 75W 90 (viscosity: 0.075 Pa-s). The mechanical power losses at 3 kg load are 468.784, 327.58, 257.344 and 265.233 W for oils 85W140, 80W90, 75W90 and 20W50, respectively, which means that mechanical power losses for 85W140 oil is 82% more in comparison to power loss obtained for 75W90 oil. While at 27 kg load around 188% more power losses are obtained for 85W140 gear oil when compared to 80W90 gear oil. Almost similar trend is shown by all the oils for gear ratio 3. However, 5th gear has shown significantly high values of mechanical power losses at no load condition when operated using 85W140 gear oil, which are even more than the input power capacity of the electric motor used. This is because, the high viscosity of the oil has exerted huge load on the motor. So using this oil experiments are not performed at gear ratio 5.

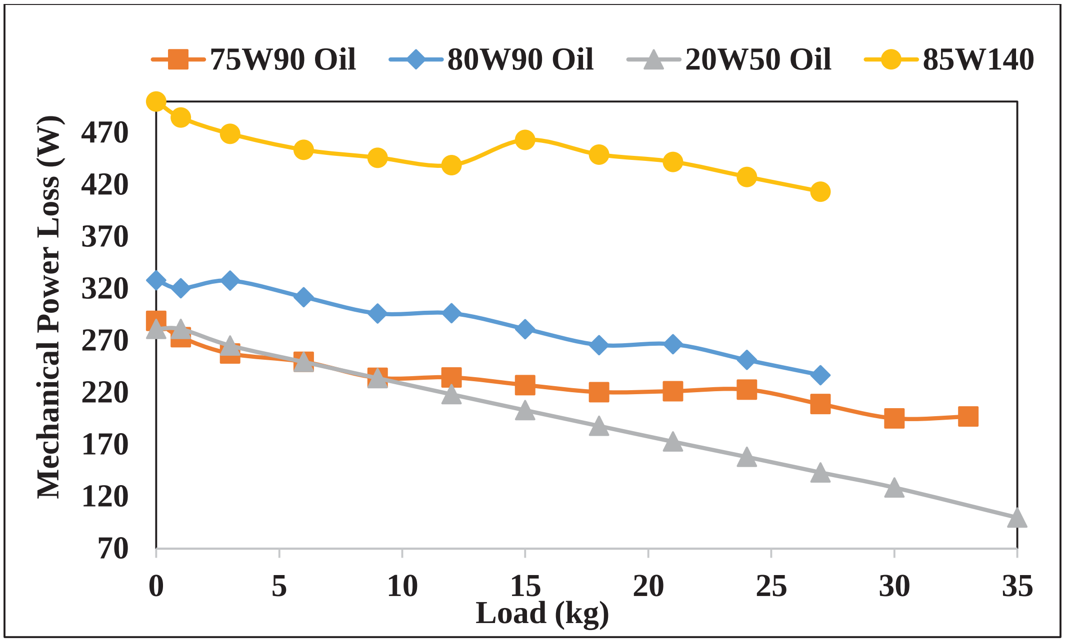


Figure 4.12: Mechanical power losses with load, for all oil volumes of oil, different viscosity 100% volume, 1st gear

However, the other oils has also shown sharp increase in the mechanical power losses for this gear ratio with increase in the load. Whereas, 75W90 oil has shown decreasing trend for mechanical power loss with increase in the load. From this it can be concluded that with increase in the viscosity of lubricating oil, the mechanical power losses increases because more viscous oil exerts more drag on the gears and other moving component of gearbox assembly.

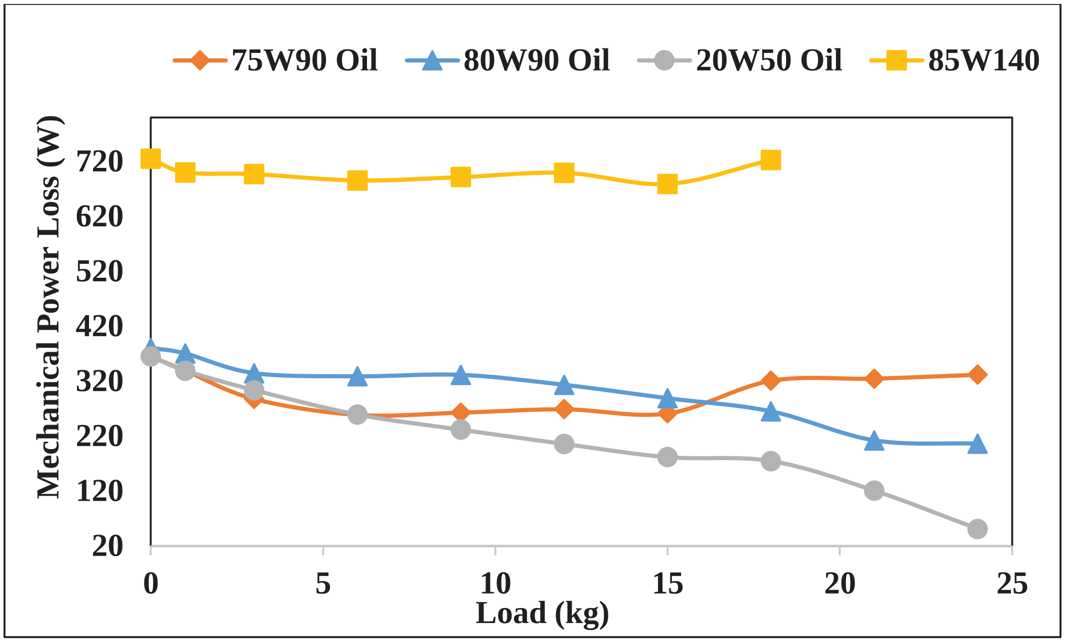


Figure 4.13: Mechanical power losses with load, for all oil volumes of oil, different viscosity, 100% volume, 3rd gear

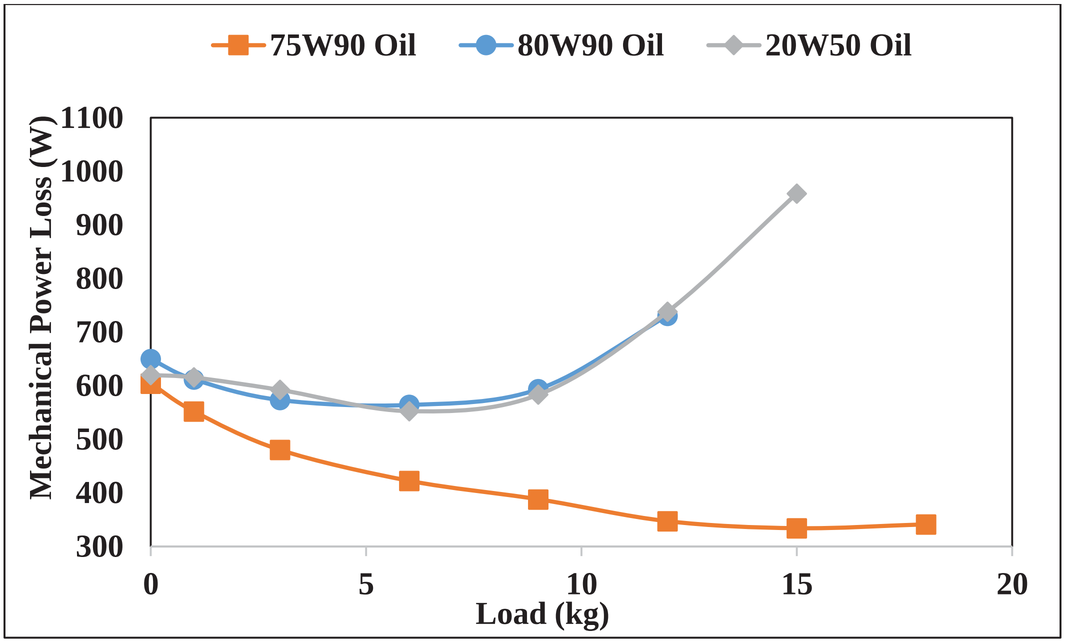


Figure 4.14: Mechanical power losses with load, for all oil volumes of oil, different viscosity, 100% volume, 5th gear

4.5 Effect of gear ratio on mechanical efficiency of gearbox

Mechanical efficiency versus load

In Figures 4.15 and 4.16, mechanical efficiency are plotted against the load for different gear ratio at 100% and 25% oil volume corresponding to the mechanical power losses in the gearbox. The graphs shows that with increase in load, the mechanical efficiency has increased. Further with increase in gear ratio mechanical efficiency has also increased. The efficiency value for different gears 1 to 5 at 18 kg load are 57.266, 67.147, 70.266, 72.173 and 77.483, which means that around 35% mechanical efficiency has increased at 18 kg load when the gear ratio is shifted from 1st to 5th gear.

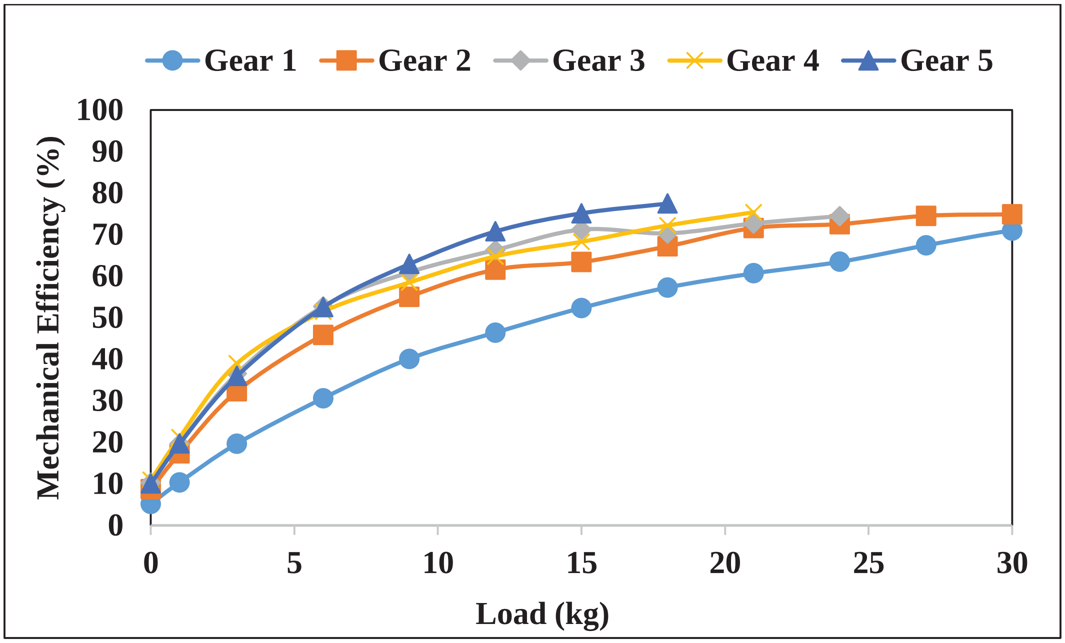


Figure 4.15: Mechanical efficiency with load, for all gear ratios, 100% volume, 75W90 gear oil

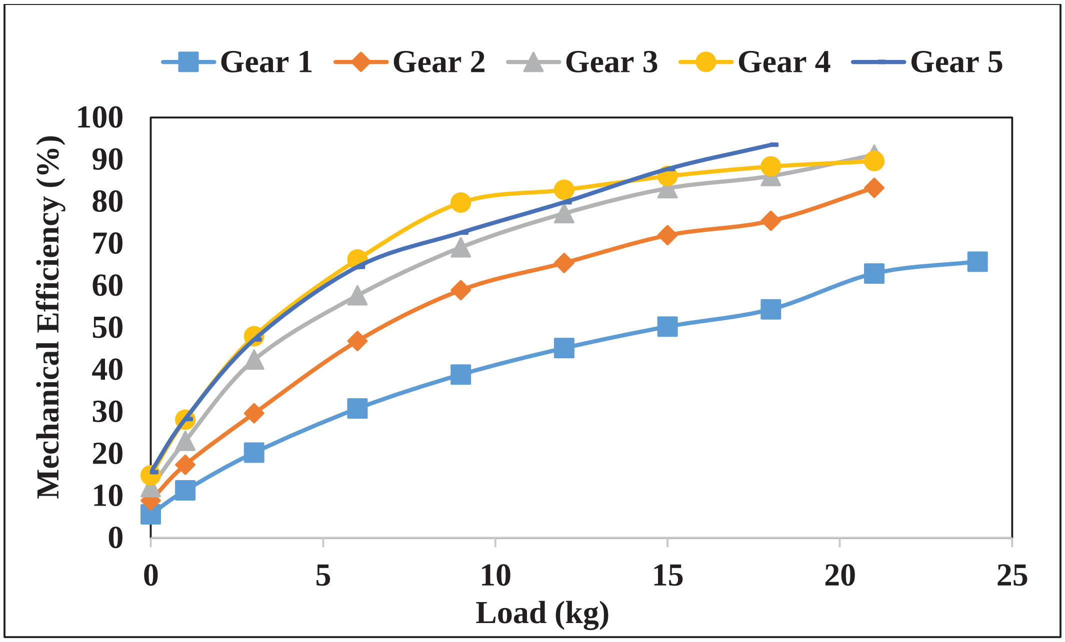


Figure 4.16: Mechanical efficiency with load, for all gear ratios, 25% volume, 75W90 gear oil

Figure 4.16 shows the similar trend for the mechanical efficiency of the gearbox for 25% oil volume but only difference is that in 25% oil by volume, the rise in the mechanical efficiency is around 72% when the gear ratio is shifted from 1st to 5th gear. This trend is obvious, as mechanical power losses decreases with increase in the load, which leads to increase in total power output. Also with decrease in the oil volume mechanical power losses decreases, hence the higher values of mechanical efficiency obtained at lower oil volumes, are justified.

4.6 Effect of oil volume on mechanical efficiency

Effect of oil volume on the mechanical efficiency are measured and corresponding graph are plotted against the load. Figure 4.17 shows the effect of oil volume on mechanical efficiency of gearbox for the lower ratios and Figure 4.18 shows the effect of oil volume on efficiency for higher gear ratios. Figure 4.17 shows that at lower gear ratio almost similar efficiency is obtained for different oil volume i.e. 100%, 75%, 50%, and 25% oil volumes. Whereas, with increase in gear ratios the mechanical efficiency has increased with increase in the load and decrease in oil volume i.e. 5th gear has shown maximum efficiency at 25% oil volume. This is because more power is required to drive the gears at larger oil volumes due to increase in the drag and churning activity of oil.

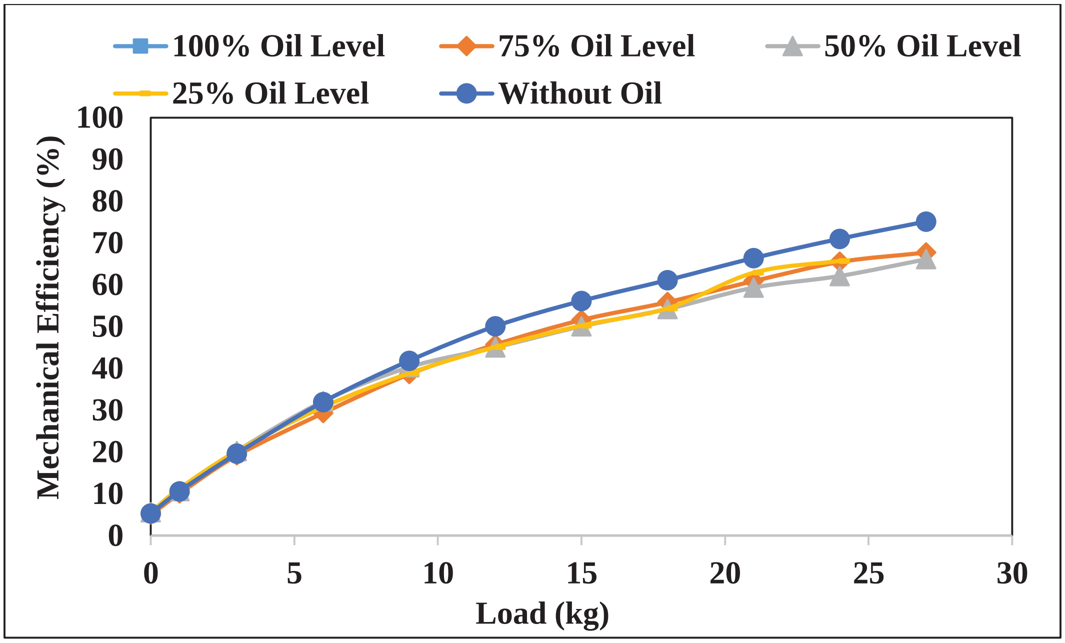


Figure 4.17: Mechanical efficiency with load, for all oil volumes, 1st gear, 75W90 gear oil

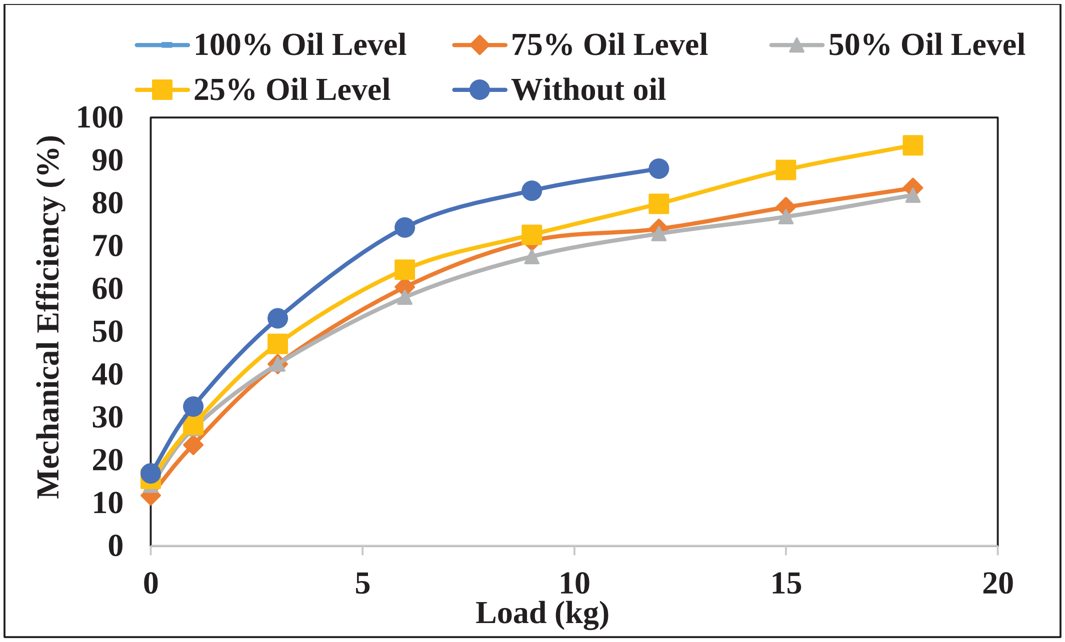


Figure 4.18: Mechanical efficiency with load, for all oil volumes, 5th gear, 75W90 gear oil

4.7 Effect of oil viscosity on mechanical efficiency

For measuring the effect of viscosity on the mechanical efficiency, different oils having different viscosity were used. Figures 4.19 to 4.21 show mechanical efficiency of gearbox using four types of oils having different viscosity at gear ratios 1, 3 and 5. Almost all the graphs shows similar trend i.e. with rise in load mechanical efficiency increases. The oil having highest viscosity (85W140) has shown minimum rise in efficiency as compared to the other oils having lower viscosity. At lower gear ratios 80W90 gear oil, 75W90 and 20W50 all the oils shows almost similar rise in efficiency with rise in load but at higher gear ratios 75W90 oil showed highest mechanical efficiency as compared to the other oils. A rise 47% rise in mechanical efficiency is obtained for 1st gear at 27 kg load when 20W50 oil is used instead of 85W140 oil.

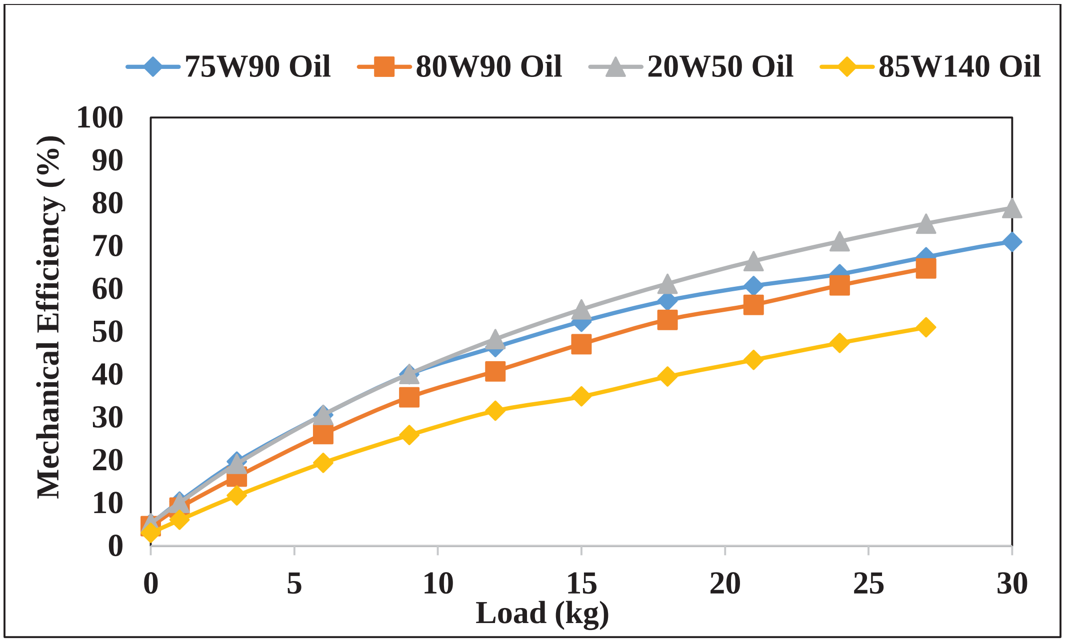


Figure 4.19: Mechanical efficiency with load, for all oil volumes, different viscosity, 100% oil volume, 1st gear

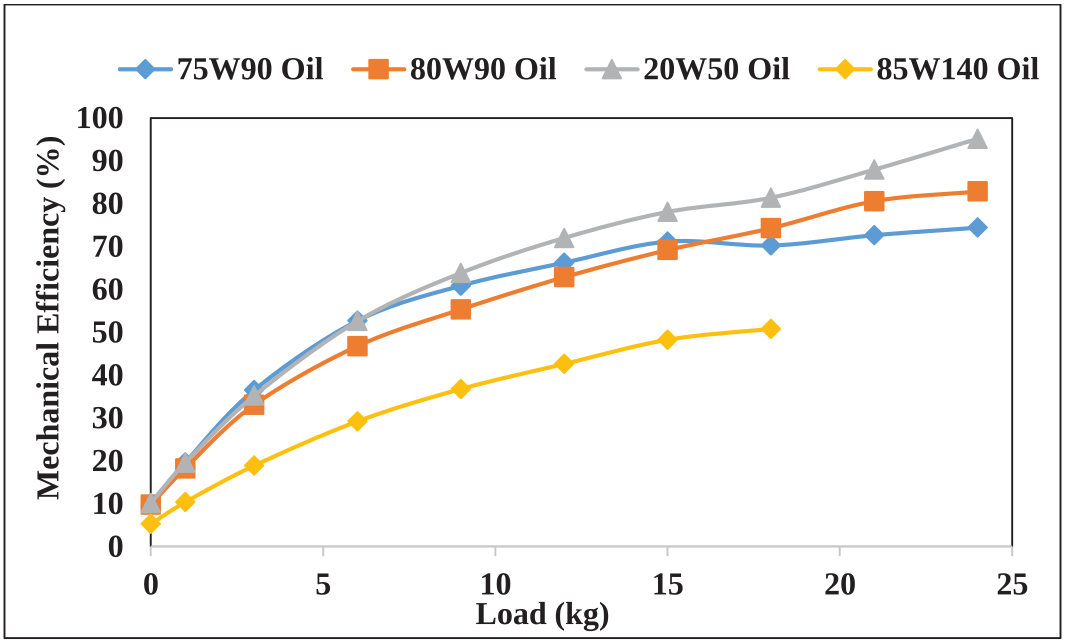


Figure 4.20: Variation of mechanical efficiency with load for different oils having different viscosity for 100% oil volume at 3rd gear

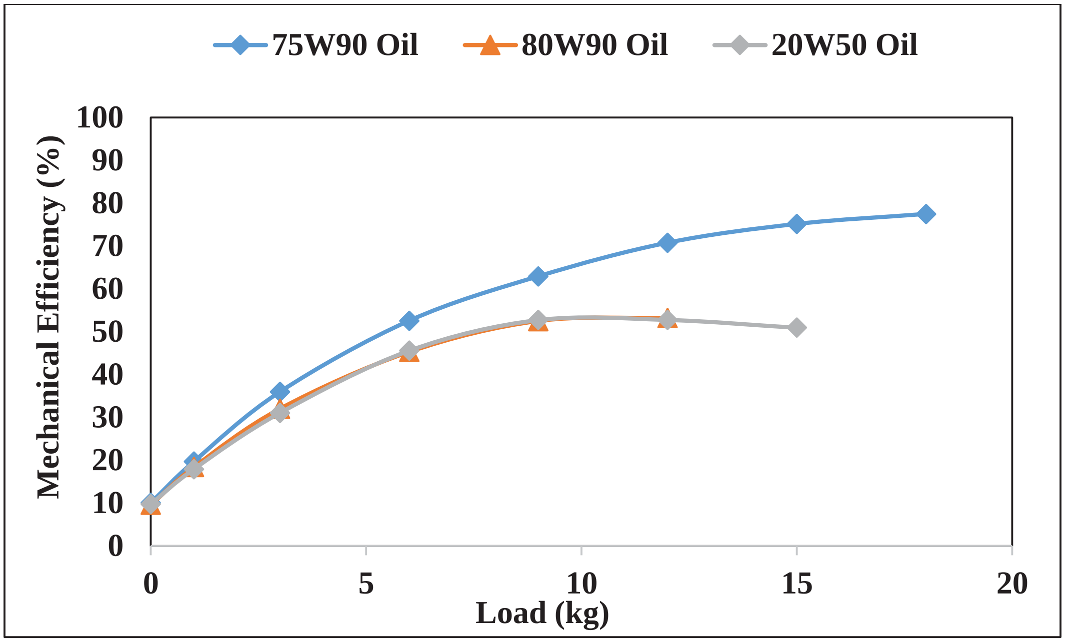


Figure 4.21: Mechanical efficiency with load, for all oil volumes, different viscosity, 100% oil volume, 5th gear

The rate of increase of mechanical efficiency is more at lower values of loads, whereas at rate of increase has decreased at higher loads. Increased viscosity of the oil is responsible for lower mechanical efficiencies.

4.8 Effect of nanoparticle concentration on power losses

For measuring the effects of nanofluid on the power losses, nanofluids are prepared at different concentration (0.05, 0.1 and 0.5% vol. conc.) and experiments are performed on these samples. Figures 4.22 and 4.23 shows the mechanical efficiency at different loads for 75W90 gear oil having three different concentrations for 1st and 5th gear ratio. These graphs shows that the mechanical power losses have not shown significant change on addition of nanoparticles at low concentrations such as 0.05 and 0.1%, whereas a small increase in the mechanical power losses is obtained at higher concentrations such as 0.5%. This increase in the mechanical power losses is due to increase in the viscosity of the base lubricating oil on addition of nanoparticles. Almost all the gear ratio shows similar result with the concentration of nanoparticle.

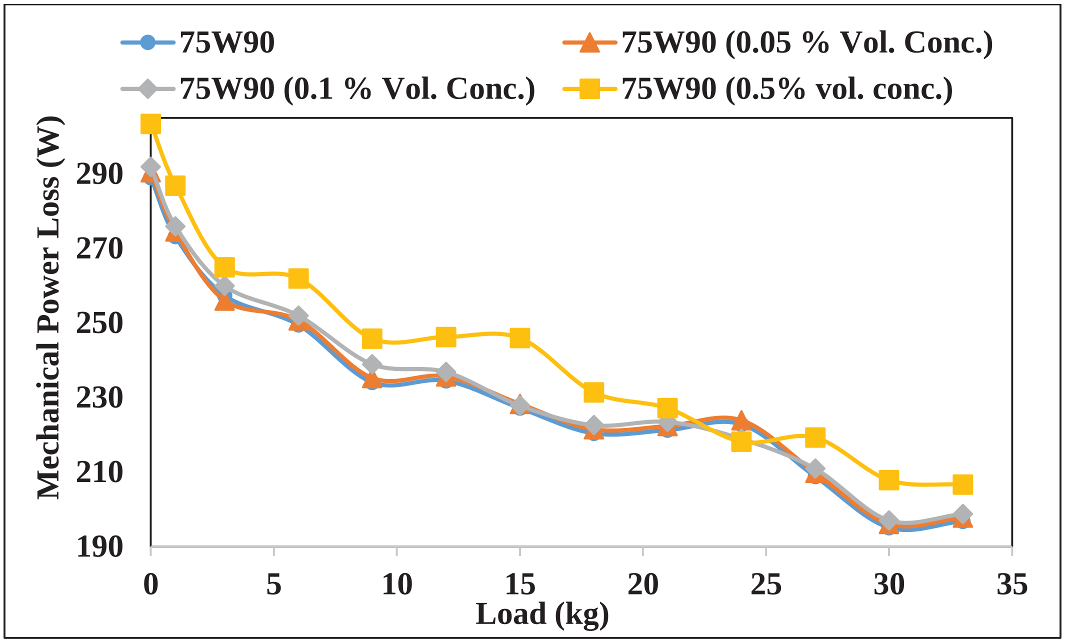


Figure 4.22: Mechanical power losses with load, at all conc. of nanoparticles, different viscosity, 100% oil volume, 1st gear

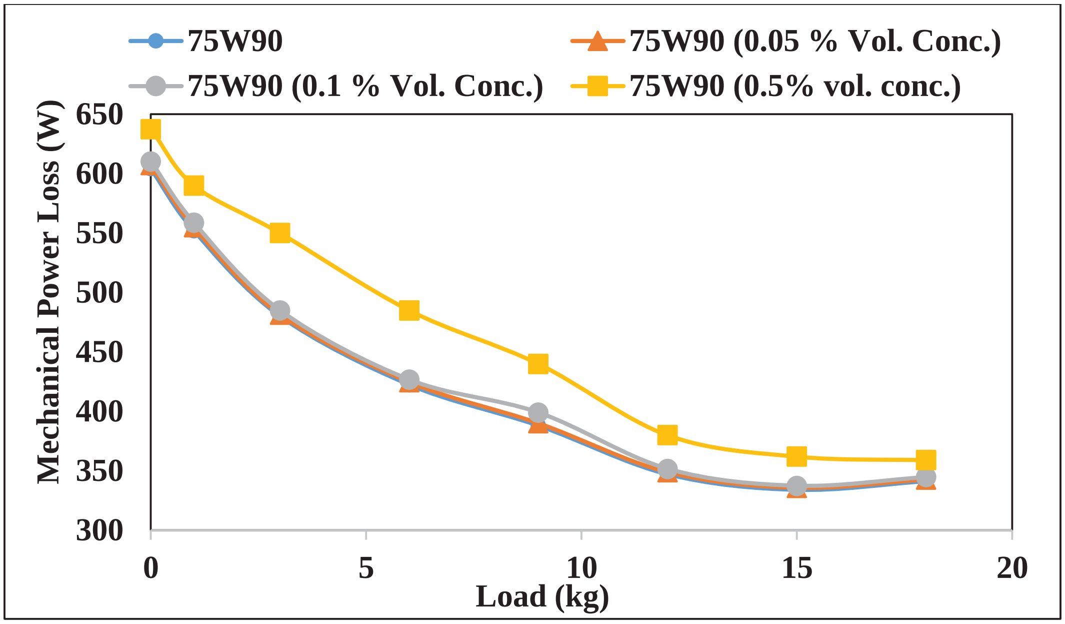


Figure 4.23: Mechanical power losses with load, at all conc. of nanoparticles, different viscosity, 100% oil volume, 5th gear

4.9 Effect of nanoparticle on the temperature of gear oil

During the operation, the temperature of gear oil rises because of friction in the meshed gear and bearings. But with the continuous operation a steady state is reached in which there is no further rise in the temperature. Further a very little amount of increase in temperature can occur with the load. Figure 4.24 shows the temperature of gear oil at different oil volume for different gear ratios. Graph shows that with rise in oil volume the temperature of gear oil rises, means that 100% oil volume shows higher rise in the temperature and 25% oil volume has least rise in the temperature. This is mainly because, with decrease in oil volume, the volume to surface area ratio goes down which leads to increase in the area of heat transfer available for oil. The temperature of gear oil also increases with increase in the gear ratio, means that maximum temperature rise occur in the 5th gear. This is due to fact that 5th gear is operating at very high rpm and maximum rubbing would occur between the meshed gears.

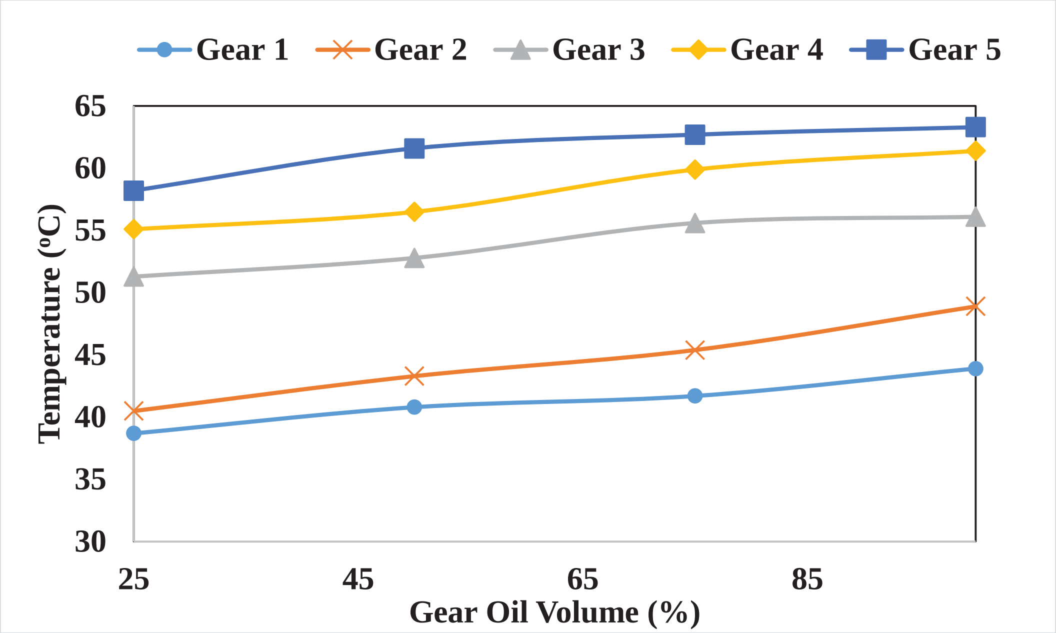


Figure 4.24: Temperature vs Gear oil volume at different gear ratios

This rise in the temperature causes decrease in the viscosity of the gear oil which further leads to the rise in mechanical and spin losses, because oil thickness between the meshed gears at high temperature goes down and metal to metal contact can occur which increases the friction and rubbing between the gears. This may also leads to failure of the meshed gears and damage the entire transmission system. This is mainly a plague in the field of transmission system. This problem would occur mainly at high rpm or at high gear ratio. To overcome this difficulty nanoparticles are used in the gear oil. Al_2O_3 nanoparticles are used at different concentration (0.05%, 0.1% and 0.5% vol. conc.) in the 75W90 gear oil.

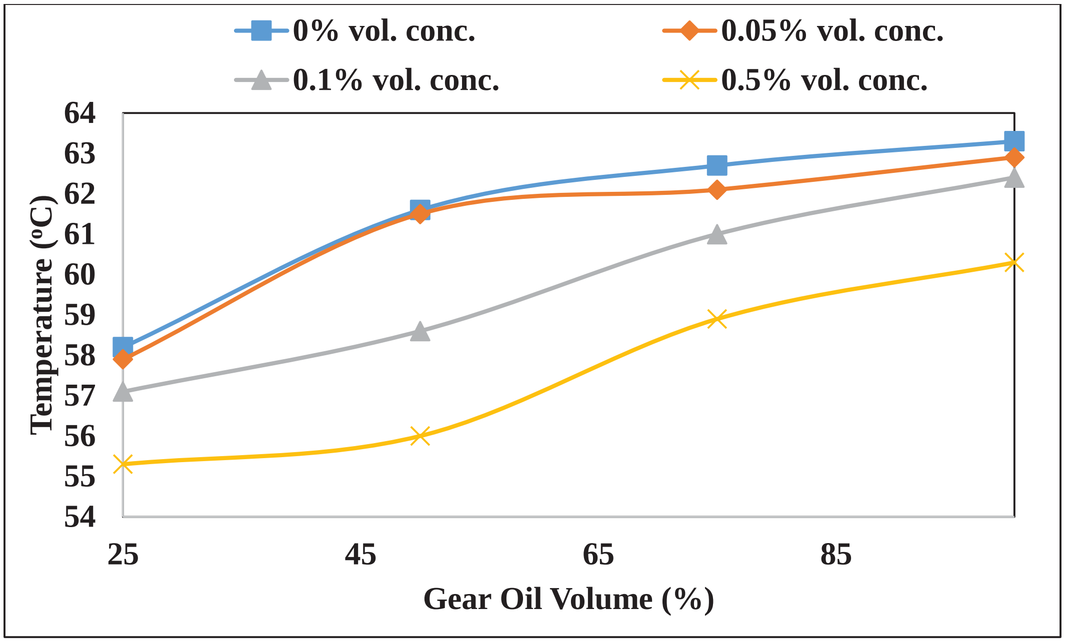


Figure 4.25: Temperature vs. Gear oil volume at different concentration of nanoparticles for gear 5

Figure 4.25 shows the temperature for different oil volume and at different concentration of nanoparticles (0.05%, 0.1% and 0.5%) for 5th gear. Graphs shows that at low concentration i.e.0.05% of nanoparticles there is little decrease in the temperature but at high concentration there is large decrease in the temperature.

CHAPTER 5: CONCLUSIONS

Conclusions

Experimentation has been carried out using oils of different viscosity to quantify spin and mechanical power losses of an automobile gearbox at different gear ratios. Al_2O_3 nanoparticles are also added to 75W90 gear oil and their effect on spin and mechanical power losses are obtained. Based on the outcomes of experimental work following conclusions can be made:

- Mechanical power losses has shown decreasing trend with the increase in load, for all the gear ratio from 1st to 5th. At a particular load, mechanical power losses has increased with the increase in gear ratio. Similar trend is obtained at all oil volumes but the values of losses has decreased with the decrease in the oil volume.
- Load independent loses i.e. spin losses has also increased with the increase in gear ratio. Higher mechanical power losses are obtained at 100% oil volume, whereas, minimum power losses are obtained when gearbox is operated without the lubricating oil, for all the gear ratios.
- Maximum spin power losses are obtained when gearbox is operated without lubricating oil followed by 100% oil volume and minimum losses are obtained with 25% volume of oil.
- Mechanical power losses shows increasing trend with the increase in oil viscosity for all the gear ratios. With increase in the oil viscosity, oil volume and gear ratio the churning activity of the oil increases, which exerts additional drag on the gears and other rotational component of gearbox assembly, hence increased power losses.
- Mechanical efficiency has shown increasing trend with increasing load at all the gear ratio. The rate of increase is more at smaller load values whereas this rate of increase has decreased

at higher load. Minimum mechanical efficiency is obtained for 1st gear whereas increases with the increase in gear ratio, it has been increased at a particular load.

- Mechanical efficiency has shown similar trend at all the oil volumes i.e. increases with load whereas the maximum value of mechanical efficiency is obtained when gearbox is operated without oil and minimum efficiency is obtained at 100% oil volume. The spread of mechanical efficiency is very small at low gear ratio with the variation in oil volume.
- Lower value of mechanical efficiency is obtained for high viscosity oil and the value of efficiency increases, as the viscosity of oil goes down. When the nanoparticles are added to the gear oil, the minor changes has been observed in the mechanical power loses at small volume concentration of the nanoparticle in the gear oil (0.05% and 0.1%).
- At the 0.5% volume concentration, the increase in mechanical power losses is more in comparison to the lower concentration. This variation is found for all other the gear ratios. It has been found that with the increase in the nanoparticle concentration, the viscosity of the oil goes up which leads to the increase in power loses.
- The benefit of using nanoparticle in the gear box is to minimize the temperature of the oil during its operation.

FUTURE SCOPE OF WORK

In this study, experimental results followed a particular trend in respect of the influence of various parameters on the operating condition on transmission power losses. There is no such specific process which can predict the trend in these results. This experimental study must be followed by a theoretical study to have an in-depth study of the behavior of measured power loss of manual transmission efficiency.

A detailed analysis of the parameters was carried out in this study, but there is still a room for certain improvements. High power motor with a very high rpm can be selected, so that rpm of the engine matches well with the rpm of motor to obtain the original operating conditions. A variable frequency derive can also be used to vary the input rpm to check the mechanical power loss at different rpm of the motor. A device can also be used to maintain the temperature of oil in the gearbox to find out the various spin losses at various steady state conditions. Nanoparticles can also be employed to provide better lubrication in the gearbox by providing cushion effect between the meshed gears, so that the rubbing and friction between the meshed gears can be reduced and power losses in the gearbox are minimized. Torque meters of higher sensitivity and accuracy can also be utilized for measuring and validating the value of torque measured manually.

REFERENCES

Barzaghi, C., Berti, F., and Gommellini, C., “Development of a Bench Test Procedure for Assessing the Effect of Lubricants on Car Manual Transmission Efficiency,” S.A.E. Transaction, 104 (1995) 668.

Chandrasekar, M., Suresh, S, and Bose, A. C., “Preparation and properties of copper-oil-based nanofluids” Experimental Thermal and Fluid Science 34 (2010) 210–216.

Chein, R., and Haung, G., “Analysis of micro channel heat sink performance using nanofluids” Applied Thermal Engineering, 25 (2006) 3104-3114.

Choi, C., Yoo, H. S., and Oh, J. M., “Preparation and heat transfer properties of nanoparticle-in-transformer oil dispersions as advanced energy-efficient coolants” Current Applied Physics 8 (2008) 710–712.

Carlos, M. C. G., Fernandes, Ramiro C. Martins, Jorge, H. O. Seabra, “Torque Loss of Type C40 FZG Gears Lubricated With Wind Turbine Gear Oils” Tribology International 70 (2014) 83–93.

Carlos, M. C. G., Fernandes, Ramiro C. Martins, Jorge, H. O. Seabra, “Friction Torque of Thrust Ball Bearings Lubricated With Wind Turbine Gear Oils” Tribology International 58 (2013) 47–54.

Carlos, M. C. G Fernandes, Pedro, M. P. Amaro, Ramiro C. Martins, Jorge, H. O. Seabra, “Torque loss in thrust ball bearings lubricated with wind turbine gear oils at constant temperature” International 66 (2013) 194–202.

Changenet, C., Oviedo-Marlot, X., and Velex, P., “Power Loss Predictions in Geared Transmissions Using Thermal Networks-Applications to a Six-Speed Manual Gearbox,” Transactions of the ASME, 128 (2006) 618-625.

Choi, S. U. S., Zhang, Z. G., Yu, W., Lockwood, F. E., and Grulke, E. A., “Anomalous thermal conductivity enhancement in nano tube suspensions,” Appl Phys Lett, 69 (2001) 2252.

Dae-Hwang, Yoo, Hong, K. S. and Ho-Soon Yang, “Study of thermal conductivity of nanofluid for the application of heat transfer fluids” ThemochimicaActa, (Elsevier), 44 (2007) 66-69.

Dan Li., Wenjie, X. and Wenjun Fang., "Preparation and properties of copper-oil-based Nanofluids" Nanoscale Research Letters, 6 (2011) 373.

Dan Li and wenjun Fang, "preparation and properties of of copper – oil based nanofluids" Nanoscale Research Letter, 6 (2011) 373-381.

Das, S. K., Putra, N. and Roetzel, W., "Pool boiling characteristics of nanofluids" International Journal of Heat and Mass Transfer, 46 (2003) 851-862.

Das, K Sarit, Choi, U. S. Stephen, Wenhua Yu and Pradeep T., "Nanofluids: Science and Technology," John Wiley & Sons, Inc. New Jersey (2006) 1-2.

Daxiong, W. and Liqui, W., "Preparation and properties of copper-oil-based nanofluids" Experimental Thermal and Fluid Science, 35 (2010) 212–218.

Duangthongsuk, W. and Wongwise S., "Measurement of temperature-dependent thermal conductivity and viscosity of TiO_2 – water nanofluids" Experimental Thermal and Fluid Science, 33 (2009) 706-714.

Eastman, J. A., Phillpot, S. U. S. and Keblinski, P., "Thermal transport in nanofluids" Annual review of Materials research, 34 (2004) 219-246.

Greco, A., Mistry, K., Sista, V., Eryilmaz, O. and Erdemir, A., "Friction and Wear Behaviour of Boron Based Surface Treatment and Nano-Particle Lubricant Additives for Wind Turbine Gearbox Applications" Wear, 271 (2011) 1754– 1760.

Greenbaum, J. J., Kluger, M. A. and Westmoreland, B. E., "Manual Transmission Efficiency Trends and Characteristics," SAE Technical Paper Series, 942274, Warrendale, PA: Society of Automotive Engineers (1994).

Van Dongen, L. A. M., "Efficiency Characteristics of Manual and Automatic Passenger Car Transaxles," SAE Technical Paper Series, 820741, Warrendale, PA: Society of Automotive Engineers (1982).

Heingartner, P. and Mba, D., “Determining Power Losses in the Helical Gear Mesh,” Gear Technology, (2005) 32-37.

Kang, S. W., Wei, W. C., Tsai, S. H. and Yang, S. Y., "Experimental Investigation of Silver Nano Fluid on Heat Pipe Thermal Performance" Applied Thermal Engineering, 26 (2004) 2377-2382.

Kluger, M. A., Greenbaum, J. J. and Mairet, D. R., “Proposed Efficiency Guidelines for Manual Transmissions for the Year 2000,” SAE Technical Paper Series, 950892, Warrendale, PA: Society of Automotive Engineers (1995).

Kluger, M. A. and Long, D. M., “An Overview of Current Automatic, Manual and Continuously Variable Transmission Efficiencies and Their Projected Future Improvements,” S.A.E. Transaction, 108 (1999) 1-6.

Lee, S., Eastman, J. A., Choi, S. U. S., Li, S., and Thompson, L. J., “Enhanced thermal conductivity through the development of nanofluids” Materials Research Society Symposium, 45 (1997) 3–11.

Leong, K.C., Yang, C. and Murshed, S.M.S., “A model for the thermal conductivity of nanofluids – the effect of interfacial layer” Journal of Nanoparticle Research, 8 (2006) 245–254.

Li, Y. and Seireg, A. A., “Predicting the Coefficient of Friction in Sliding-Rolling Contacts” Tribology Conference

Lotfizadeh Dehkordi B., Hamdi M., Ghadimi A., Metselaar H. S. C., Kazi S. N., and Sadeghinezhad E., “Investigation of Viscosity and Thermal Conductivity of Alumina Nanofluids With Addition of SDBS” Heat Mass Transfer, 49 (2013) 1109–1115.

Madhusree Kole and Dey, T. K., “Thermal conductivity and viscosity of Al2O3 nanofluid based on car engine coolant” Appl. Phys, 43 (2010) 315-501.

Madhusree Kole and Dey, T. K., “Role of interfacial layer and clustering on the effective thermal conductivity of CuO–gear oil nanofluids” Experimental Thermal and Fluid Science, 35 (2011) 1490–1495.

Madhusree Kole and Dey, T. K., “Enhanced thermophysical properties of copper nanoparticles dispersed in gear oil” Applied Thermal Engineering, 56 (2013) 45-53.

Martins, R.C., Cardoso, N. F. R., Bock, H., Igartua, A., Jorge, H. O. Seabra, “Power loss performance of high pressure nitride steel gears” Tribology International 42 (2009) 1807–1815.

Martins, R., Seabrab J., Britoc A., Seyfertd Ch., Lutherd R., Igartua A. “Friction coefficient in FZG gears lubricated with industrial gear oils: Biodegradable ester vs. mineral oil” Tribology International 39 (2006) 512–521.

Martin, K. F., “A Review of Friction Predictions in Gear Teeth,” Wear, 49 (1978) 201-238.

Murshed, S. M. S, Leong, K. C. and Yang, C., “Enhanced thermal conductivity of $Ti0_2$ water based nanofluids” International Journal of Thermal Sciences 44 (2005) 367-373.

Michael, P. Beck, Yanhui Yuan, Pramod, W. and Amyn, S. Teja, “The effect of particle size on the thermal conductivity of alumina nanofluids” J Nanopart Res, 11 (2009) 1129–1136.

Nandy Putra, and Yanuar, “Application of Nanofluids to a heat pipe liquid-block and the thermoelectric cooling of electronic equipment” Experimental Thermal and Fluid Science, 35 (2011) 1274-1281.

Nguyen, C. T. and Desgranges, F., Roy, G., “Temperature and particle-size dependent viscosity data for water-based nanofluids” International journal of Heat and Fluid Flow, 28 (2007) 1492-1506.

Pal, B., Mallick, S. S. and Bonamali, P., “Phase-dependent thermo physical properties of α-and γ -Al2O3 in aqueous suspension” Journal of Industrial and Engineering Chemistry 48 (2014) 1518-1524.

Paul, B., “Kinematics and Dynamics of Planar Machinery” Prentice Hall” (1979).

Pedro, M. T. Marques., Carlos, M. C. G., Fernandes., Ramiro C. Martins., and Jorge H. O. Seabra, “Efficiency of a Gearbox Lubricated With Wind Turbine Gear Oils” Tribology International, 71 (2014) 7–16.

Pedro, M. T. Marques., Carlos, M. C. G., Fernandes., Ramiro C. Martins., and Jorge, H. O. Seabra “Power Losses at Low Speed in a Gearbox Lubricated With Wind Turbine Gear Oils With Special Focus on Churning Losses” Tribology International 62 (2013) 186–197.

Prakash, D. P. and Patel, J. M., “An Experimental Investigation of Power Losses in Manual Transmission Gear Box” International Journal of Applied Research in Mechanical Engineering (IJARME) ISSN: (2006) 2231 –5950.

Putra, N., Roetzel, W. and Das, S. K., “Natural convection of nano-fluids” Heat and Mass Transfer 39 (2003) 775–784.

Seetharaman, S., Kahraman, A., Bednarek, G. and Rosander, P., “A Model to Predict Mechanical Power Losses of Manual Transmissions,” Automobil technische Zeitschrift, 4 (2008) 346-357.

Seetharaman, S., Kahraman A., “Load Independent Spin Power Losses of a Spur Gear Pair: Model Formulation,” (in review) Journal of Tribology (2008).

Seetharaman, S., Kahraman, A., Moorhead, M. D., and Petri-Johnson, T. T., “Load Independent Spin Power Losses of a Spur Gear Pair: Experiments and Model Validation,” (in review) Journal of Tribology (2008).

Sharma, K. V., SyamSundar, L. and Sharma, P. K., “Estimation of heat transfer coefficient and friction factor in the transition flow with low volume concentration of Al2O3 nanofluid flowing in a circular tube and with twisted tape insert” International Communications in Heat and Mass Transfer 36 (2009) 503–507

Tiago Cousseau, Beatriz Grac, Armando Campos, Jorge, H. O. Seabra “Friction torque in grease lubricated thrust ball bearings” Tribology International 44 (2011) 523–531.

Tsai, C. Y., Chien, H. T., Ding, P. P., Chan, B., Luh, T. Y. and Chen, P. H., “Effect of Structural Character of Gold Nanoparticles in Nanofluid on Heat Pipe Thermal Performance”, Materials Letters, 58 (2004) 1461–1465.

Tzeng, S., Lin, C. and Huang, K., “Heat Transfer Enhancement of Nano Fluids in Rotary Balde Coupling of Four Wheel Drive Vehicles” Act a Mechanica, 179 (2005) 11-23.

Uicker, J. J., Pennock, G. R., Shigley, J. E., ” Theory of Machines and Mechanisms” 3rd edition, Vol 1,Oxford University Press, New York, (2003).

Van Dongen, L. A. M., “Efficiency Characteristics of Manual and Automatic Passenger Car Transaxles,” SAE Technical Paper Series, 820741, Warrendale, PA: Society of Automotive Engineers (1982).

Wang, B.X., Zhou, L.P., and Peng, X. F., “A fractal model for predicting the effective thermal conductivity of liquid with suspension of nanoparticles” International journal of Heat and Mass Transfer, 46 (2003) 2665-2672.

Wang, X., Xu, X., and Choi, S. U. S., “Thermal Conductivity of Nanoparticle -Fluid Mixture” Journal of Thermophysics and Heat Transfer, 13 (1991) 474-480.

Wang, X. Q., Mujumdar A.S., “Heat transfer characteristics of nanofluids: a review,” Int J Therm Sci, 46 (2006) 1.

Wenyang Zhang, Dmytro Demydov, Muhammad P. Jahan, Kuldeep Mistry, Ali Erdemir, and Ajay P. Malshe, “Fundamental understanding of the tribological and thermal behavior of Ag–MoS_2 nanoparticle-based multi-component lubricating system” Wear, 288 (2012) 9– 16.

Xie, H., Fujii, M. and Zhang, X., “Effect of interfacial nanolayer on the effective thermal conductivity of nanoparticle-fluid mixture,” Int J Heat Mass Transfer, 48 (2005) 2926.

Xuan Yimin, and Li Qiang, “Effect of aggregation on the viscosity of copper oxide-gear oil nanofluids” International Journal of Thermal Sciences 50 (2011) 1741-1747.

Xuan Yimin, and Li Qiang, “Investigation on Convective Heat Transfer and Flow Features of Nanofluids” Heat Transfer, 125 (2005) 151-155.

Yada, T., “Review of Gear Efficiency Equation and Force Treatment,” JSME 40 (1997)1-8.

Yasutsune, A., Taku, U., Teruo S. and Shigemi, S., “The Lubricant Churning Loss in Spur Gear Systems” Wear, 16 (2008) 833-842.

Appendix: A1

Table A1.1: Conductivity values of 75W90 gear oil at different temperature

Sr. No.	Temperature (ºC)	Conductivity (W/m-K)
1	31.6	0.133
2	36.5	0.135
3	41	0.139
4	45.8	0.141
5	50.1	0.143
6	55.6	0.151
7	59.4	0.154
8	66.1	0.157
9	70.9	0.159

Table A1.2: Conductivity values of 0.05% conc. of 75W90 gear oil at different temperature

Sr. No.	Temperature (ºC)	Conductivity (W/m-K)
1	31.8	0.137
2	36.4	0.139
3	41.2	0.146
4	46.1	0.152
5	50.6	0.158
6	55.8	0.163
7	59.7	0.167
8	66.3	0.175
9	71.2	0.185

Table A1.3: Conductivity values of 0.1% conc. of 75W90 gear oil at different temperature

Sr. No.	Temperature (ºC)	Conductivity (W/m-K)
1	31.8	0.137
2	36.4	0.139
3	41.2	0.146
4	46.1	0.152
5	50.6	0.158
6	55.8	0.163
7	59.7	0.167
8	66.3	0.175
9	71.2	0.185

Table A1.4: Conductivity values of 0.5% conc. of 75W90 gear oil at different temperature

Sr. No.	Temperature (ºC)	Conductivity (W/m-K)
1	32.15	0.145
2	36	0.161
3	40.8	0.178
4	45.9	0.196
5	51	0.213
6	55.4	0.235
7	61	0.254
8	65.8	0.271
9	70.9	0.288

Appendix: A2

Table A2.1: Experimental data for gear 1 using 75W90 gear oil at 100% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1435.00	225.60	304.69	15.75	0.00	304.69	288.93	5.17
1	1430.00	225.50	304.69	15.75	15.75	288.93	273.18	10.34
3	1430.00	225.30	320.31	15.75	47.22	273.10	257.34	19.66
6	1425.00	224.70	359.38	15.75	94.18	265.19	249.44	30.59
9	1425.00	224.00	390.63	15.75	140.83	249.79	234.04	40.09
12	1410.00	223.40	437.50	15.75	187.27	250.23	234.48	46.41
15	1410.00	223.00	476.56	15.75	233.67	242.89	227.14	52.34
18	1400.00	222.30	515.63	15.75	279.52	236.10	220.35	57.27
21	1400.00	221.90	562.50	15.75	325.52	236.98	221.22	60.67
24	1405.00	221.20	609.38	15.75	370.85	238.52	222.77	63.44
27	1400.00	220.60	640.63	15.75	416.08	224.55	208.79	67.41
30	1400.00	220.00	671.88	15.75	461.05	210.82	195.07	70.97

Table A2.2: Experimental data for gear 2 using 75W90 gear oil at 100% oil volume

Gear 2								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	396.8	320.31	27.73	0.00	320.31	292.59	8.66
1	1440	396.9	320.31	27.73	27.73	292.59	264.86	17.31
3	1430	396.8	343.75	27.73	83.16	260.59	232.87	32.26
6	1420	395.0	421.88	27.73	165.56	256.32	228.59	45.82
9	1425	393.3	500.00	27.73	247.27	252.73	225.00	55.00
12	1415	391.3	578.13	27.73	328.02	250.11	222.38	61.53
15	1400	389.3	687.50	27.73	407.93	279.57	251.85	63.37
18	1400	386.8	765.63	27.73	486.37	279.26	251.53	67.15
21	1390	385.1	828.13	27.73	564.94	263.19	235.46	71.57
24	1390	382.1	921.88	27.73	640.61	281.26	253.54	72.50
27	1385	380.3	1000.00	27.73	717.29	282.71	254.98	74.50
30	1385	377.4	1093.75	27.73	790.91	302.84	275.11	74.85

Table A2.3: Experimental data for gear 3 using 75W90 gear oil at 100% oil volume

Gear 3								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	593.9	406.3	41.5	0.0	406.3	364.8	10.2
1	1440	593.4	421.9	41.5	41.5	380.4	339.0	19.7
3	1435	592.2	453.1	41.5	124.1	329.0	287.6	36.5
6	1430	588.9	546.9	41.5	246.8	300.0	258.6	52.7
9	1425	584.8	671.9	41.5	367.7	304.2	262.8	60.9
12	1410	580.4	796.9	41.5	486.5	310.3	268.9	66.3
15	1395	576	906.3	41.5	603.6	302.7	261.2	71.2
18	1380	569.5	1078.1	41.5	716.1	362.0	320.6	70.3
21	1380	560	1187.5	41.5	821.5	366.0	324.5	72.7
24	1380	551	1296.9	41.5	923.8	373.1	331.6	74.4

Table A2.4: Experimental data for gear 4 using 75W90 gear oil at 100% oil volume

Gear 4								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	829.7	531.25	57.95	0.00	531.25	473.30	10.91
1	1430	829.6	546.88	57.95	57.95	488.92	430.97	21.19
3	1425	827.5	593.75	57.95	173.42	420.33	362.38	38.97
6	1390	820.5	781.25	57.95	343.90	437.35	379.39	51.44
9	1400	808.5	968.75	57.95	508.31	460.44	402.49	58.45
12	1400	799.7	1125.00	57.95	670.37	454.63	396.68	64.74
15	1380	789.4	1296.88	57.95	827.17	469.70	411.75	68.25
18	1350	779	1437.50	57.95	979.53	457.97	400.02	72.17
21	1350	771.1	1578.13	57.95	1131.19	446.93	388.98	75.35

Table A2.5: Experimental data for gear 5 using 75W90 gear oil at 100% oil volume

Gear 5								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	970.1	671.88	67.70	0.00	671.88	604.17	10.08
1	1440	969.2	687.50	67.70	67.70	619.80	552.09	19.70
3	1430	964.6	750.00	67.70	202.15	547.85	480.14	35.98
6	1420	955.4	890.63	67.70	400.45	490.18	422.47	52.56
9	1400	940.2	1046.88	67.70	591.11	455.76	388.06	62.93
12	1380	921.7	1187.50	67.70	772.64	414.86	347.15	70.77
15	1320	899	1343.75	67.70	942.01	401.74	334.03	75.14
18	1300	880.1	1515.63	67.70	1106.65	408.97	341.27	77.48

Table A2.6: Experimental data for reverse gear using 75W90 gear oil at 100% oil volume

Reverse gear								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1420	249.3	437.50	17.42	0	437.50	420.08	3.98
1	1420	249.3	437.50	17.42	17.42	420.08	402.67	7.96
3	1430	248.4	500.00	17.42	52.06	447.94	430.53	13.89
6	1425	245.4	656.25	17.42	102.86	553.39	535.98	18.33
9	1420	244.4	765.63	17.42	153.66	611.97	594.55	22.34
12	1410	242.5	875.00	17.42	203.28	671.72	654.30	25.22
15	1410	241.6	937.50	17.42	253.16	684.34	666.92	28.86
18	1400	239.2	1062.50	17.42	300.77	761.73	744.31	29.95
21	1390	237	1109.38	17.42	347.68	761.70	744.28	32.91

Table A2.7: Experimental data for gear 1 using 80W90 gear oil at 100% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1455	225.7	343.75	15.72	0	343.75	328.03	4.57
1	1455	225.1	351.56	15.72	15.72	335.84	320.11	8.95
3	1440	225.8	390.63	15.72	47.32	343.30	327.58	16.14
6	1447	225.3	421.88	15.72	94.43	327.44	311.72	26.11
9	1450	225	453.13	15.72	141.46	311.67	295.94	34.69
12	1445	224.4	500.00	15.72	188.11	311.89	296.17	40.77
15	1430	223.8	531.25	15.72	234.51	296.74	281.02	47.10
18	1435	223.6	562.50	15.72	281.16	281.34	265.62	52.78
21	1435	223.1	609.38	15.72	327.28	282.09	266.37	56.29
24	1420	222.9	640.63	15.72	373.70	266.92	251.20	60.79
27	1420	222.4	671.88	15.72	419.47	252.40	236.68	64.77

Table A2.8: Experimental data for gear 5 using 80W90 gear oil at 100% oil volume

Gear 5								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1405	988.4	718.75	68.98	0	718.75	649.77	9.60
1	1400	987.5	750	68.98	68.98	681.02	612.03	18.40
3	1405	960.6	843.75	68.98	201.31	642.44	573.45	32.04
6	1395	949.5	1031.25	68.98	397.97	633.28	564.29	45.28
9	1400	933.8	1250	68.98	587.09	662.91	593.93	52.49
12	1340	909.9	1562.5	68.98	762.75	799.75	730.77	53.23

Table A2.9: Experimental data for gear 1 using 80W90 gear oil at 75% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1435	226.1	296.88	15.83	0	296.88	281.05	5.33
1	1440	226.6	304.69	15.83	15.83	288.86	273.03	10.39
3	1440	226.1	320.31	15.83	47.38	272.93	257.10	19.73
6	1435	225.3	343.75	15.83	94.43	249.32	233.49	32.08
9	1440	225.2	375.00	15.83	141.59	233.41	217.59	41.98
12	1435	224.8	414.06	15.83	188.44	225.62	209.79	49.33
15	1435	224.2	453.13	15.83	234.93	218.20	202.37	55.34
18	1430	223.7	500.00	15.83	281.28	218.72	202.89	59.42
21	1425	223.5	531.25	15.83	327.87	203.38	187.55	64.70
24	1410	223.1	562.50	15.83	374.04	188.46	172.63	69.31
27	1415	222.6	593.75	15.83	419.85	173.90	158.07	73.38

Table A2.10: Experimental data for gear 5 using 80W90 gear oil at 75% oil volume

Gear 5								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1400	973.5	609.38	67.81	0	609.38	541.57	11.13
1	1415	970.7	648.44	67.81	67.81	580.63	512.82	20.91
3	1410	965.9	718.75	67.81	202.42	516.33	448.52	37.60
6	1400	956.6	875.00	67.81	400.95	474.05	406.24	53.57
9	1370	943.9	1078.13	67.81	593.44	484.69	416.88	61.33
12	1360	928	1328.13	67.81	777.92	550.20	482.39	63.68
15	1340	907	1562.50	67.81	950.40	612.10	544.29	65.17

Table A2.11: Experimental data for gear 1 using 80W90 gear oil at 50% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	225.9	296.88	15.79	0	296.88	281.09	5.32
1	1430	226	296.88	15.79	15.79	281.09	265.30	10.64
3	1425	225.6	312.50	15.79	47.28	265.22	249.43	20.18
6	1435	225.3	351.56	15.79	94.43	257.13	241.34	31.35
9	1430	224.8	390.63	15.79	141.33	249.29	233.50	40.22
12	1425	224.3	421.88	15.79	188.03	233.85	218.06	48.31
15	1430	224	453.13	15.79	234.72	218.41	202.62	55.28
18	1425	223.7	484.38	15.79	281.28	203.09	187.30	61.33
21	1420	223.1	523.44	15.79	327.28	196.15	180.37	65.54
24	1420	222.8	546.88	15.79	373.54	173.34	157.55	71.19
27	1415	222.2	593.75	15.79	419.10	174.65	158.87	73.24

Table A2.12: Experimental data for gear 5 using 80W90 gear oil at 50% oil volume

Gear 5								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1440	979.8	531.25	68.33	0	531.25	462.92	12.86
1	1440	978.1	593.75	68.33	68.33	525.42	457.10	23.02
3	1435	973.6	656.25	68.33	204.04	452.21	383.89	41.50
6	1430	964.2	875.00	68.33	404.13	470.87	402.54	54.00
9	1410	951.8	1046.88	68.33	598.40	448.47	380.14	63.69
12	1390	937.1	1281.25	68.33	785.55	495.70	427.37	66.64
15	1360	922	1468.75	68.33	966.11	502.64	434.31	70.43

Table A2.13: Experimental data for gear 1 using 80W90 gear oil at 25% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1460	225.8	296.88	15.77	0	296.88	281.10	5.31
1	1460	225.8	312.50	15.77	15.77	296.73	280.95	10.10
3	1450	225.5	343.75	15.77	47.26	296.49	280.72	18.34
6	1450	225.1	375.00	15.77	94.35	280.65	264.88	29.37
9	1450	224.5	421.88	15.77	141.14	280.73	264.96	37.20
12	1445	223.9	468.75	15.77	187.69	281.06	265.29	43.41
15	1445	223.6	500.00	15.77	234.30	265.70	249.93	50.01
18	1435	223.2	546.88	15.77	280.66	266.22	250.45	54.20
21	1435	222.9	578.13	15.77	326.99	251.13	235.36	59.29
24	1430	222.5	593.75	15.77	373.03	220.72	204.94	65.48

Table A2.14: Experimental data for gear 5 using 80W90 gear oil at 25% oil volume

Gear 5								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1450	981.1	468.75	68.46	0	468.75	400.29	14.60
1	1450	980	484.38	68.46	68.46	415.92	347.46	28.27
3	1440	974.1	609.38	68.46	204.14	405.23	336.77	44.73
6	1430	962.4	796.88	68.46	403.38	393.50	325.04	59.21
9	1410	959.7	1000.00	68.46	603.37	396.63	328.17	67.18
12	1390	934.9	1250.00	68.46	783.71	466.29	397.84	68.17
15	1350	917.3	1468.75	68.46	961.19	507.56	439.10	70.10

Table A2.15: Experimental data for gear 1 using 20W50 gear oil at 100% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1450	225.2	296.88	15.74	0	296.88	281.14	5.30
1	1450	225.3	312.50	15.74	15.74	296.76	281.02	10.07
3	1440	225	328.13	15.74	47.15	280.97	265.23	19.17
6	1440	224.8	359.38	15.74	94.22	265.15	249.41	30.60
9	1435	224.4	390.63	15.74	141.08	249.54	233.80	40.15
12	1440	224.1	421.88	15.74	187.86	234.02	218.28	48.26
15	1440	223.7	453.13	15.74	234.40	218.72	202.98	55.20
18	1435	223.3	484.38	15.74	280.78	203.59	187.86	61.22
21	1435	223	515.63	15.74	327.14	188.49	172.75	66.50
24	1435	222.5	546.88	15.74	373.03	173.84	158.10	71.09
27	1435	222.3	578.13	15.74	419.29	158.84	143.10	75.25
30	1435	221.9	609.38	15.74	465.03	144.34	128.60	78.90
35	1435	221.2	656.25	15.74	540.83	115.42	99.68	84.81

Table A2.16: Experimental data for gear 1 using 85W140 gear oil at 100% oil volume

Gear 1								
Load (kg)	**Motor (rpm)**	**Pulley (rpm)**	**P_{in} (W)**	**P_s (W)**	**P_{out} (W)**	**P_t (W)**	**P_m (W)**	**η_m (%)**
0	1450	223.9	515.63	15.63	0	515.63	500.00	3.03
1	1445	223.7	515.63	15.63	15.63	500.00	484.37	6.06
3	1445	223.5	531.25	15.63	46.84	484.41	468.78	11.76
6	1435	223.1	562.50	15.63	93.51	468.99	453.36	19.40
9	1440	222.8	601.56	15.63	140.08	461.49	445.86	25.88
12	1430	222.4	640.63	15.63	186.43	454.19	438.57	31.54
15	1430	221.8	710.94	15.63	232.41	478.53	462.90	34.89
18	1420	220.9	742.19	15.63	277.76	464.42	448.80	39.53
21	1420	220.7	781.25	15.63	323.76	457.49	441.86	43.44
24	1415	220.4	812.50	15.63	369.51	442.99	427.36	47.40
27	1415	220	843.75	15.63	414.95	428.80	413.18	51.03

Table A2.17: Experimental data for gear 1 using 75W90 gear oil at 100% oil volume at different nanoparticle concentration

Gear 1				
	Mechanical power losses (W)			
Load (kg)	P_m (0% vol. conc.)	P_m (0.05% vol. conc.)	P_m (0.1% vol. conc.)	P_m (0.5% vol. conc.)
0	288.93	290.38	291.82	303.38
1	273.18	274.55	275.91	286.84
3	257.34	256.00	259.92	264.91
6	249.44	250.69	251.94	261.91
9	234.04	235.21	238.90	245.74
12	234.48	235.65	236.82	246.20
15	227.14	228.28	227.80	245.90
18	220.35	221.45	222.55	231.37
21	221.22	222.33	223.44	227.10
24	222.77	223.88	218.90	218.10
27	208.79	209.84	210.88	219.23
30	195.07	196.05	197.02	207.80
33	196.76	197.75	198.73	206.60

Table A2.18: Experimental data for gear 5 using 75W90 gear oil at 100% oil volume at different nanoparticle concentration

Gear 5				
	Mechanical power losses (W)			
Load (kg)	P_m (0% vol. conc.)	P_m (0.05% vol. conc.)	P_m (0.1% vol. conc.)	P_m (0.5% vol. conc.)
0	604.17	607.19	610.21	637.55
1	552.09	554.85	558.76	590.00
3	480.14	481.54	484.95	550.00
6	422.47	424.59	426.69	485.00
9	388.06	390.01	399.00	440.00
12	347.15	348.89	351.66	380.00
15	334.03	335.70	337.37	362.00
18	341.27	342.67	344.68	359.00