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# Planetary Gearset Lubrication Requirement Estimation Based on Heat Generation

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## Introduction

Delivering exactly the necessary amount of lubrication to each component of transmission is the ultimate goal of lubrication system design. This goal may not be attained until the right amount of lubrication is fully understood. In planetary gearsets, lubrication is needed for mainly two reasons. One is to remove heat generated inside the gear system and the other is to separate contact interfaces by hydrodynamic or elastohydrodynamic (EHD) films.

Planetary gearsets need lubrication for the gear teeth (sliding and rolling contacts), the planet needle bearing system (rolling plus sliding contact), and the planet washer system (sliding contact). Heat sources in a planetary gearset are depicted in Figure 1. In general, the EHD film thicknesses at gear, washer, and bearing contact surfaces are so thin that we may claim not much lubrication is needed for the planetary gear system. However, in order to build an adequate EHD film at contact interfaces, a proper inlet condition is required, to ensure a fully

flooded condition in the needle bearing system, which leads to non starved lubrications where Reynolds equation is applicable. In typical planetary systems, washer interfaces and meshing gears are not fully flooded, so Reynolds equation may not be applicable for those interfaces.

So not only the amount of lubrication but also effectiveness of lubricant delivery to each contact interface are important for planetary gear systems. In order to examine lubrication flow rate required for a planetary gearset at various operating conditions as torque and speed change, heat generation inside a planetary gearset was empirically investigated. Empirically obtained heat generation data was used to estimate the frictional power losses inside the system. Based on the heat generation a required quantity of lubrication can be determined. An empirical study of heat generation inside gear system is presented and a practical analysis is performed to rationalize lubrication requirement for the planetary gearset.

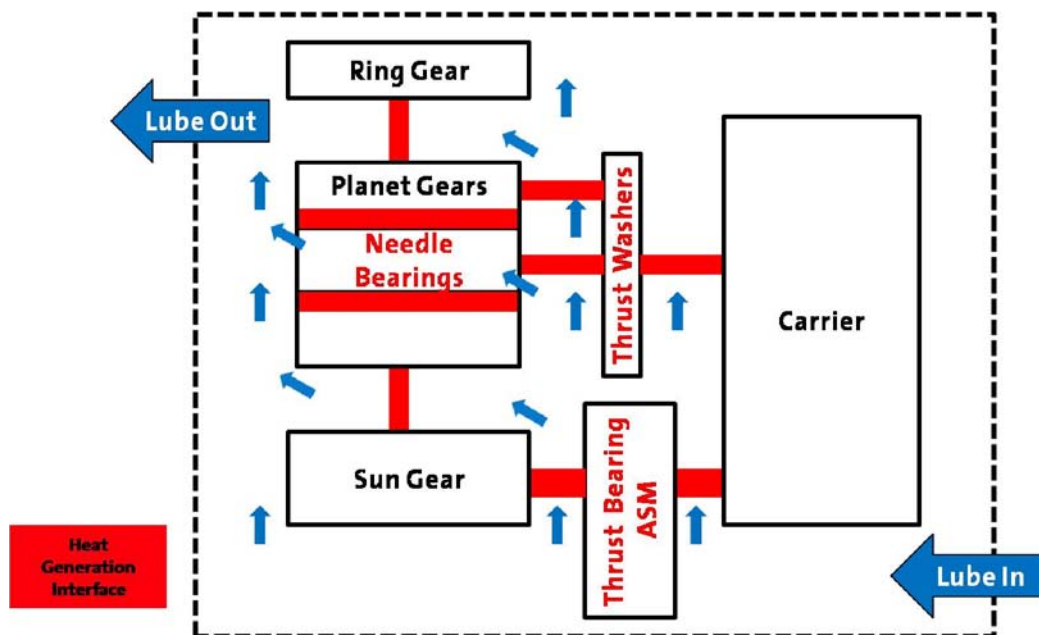


Figure 1. Illustration of heat sources in a planetary gearset

## Planetary gearset lubrication system

Planetary gearset systems can have several different types of lubrication systems. The lubrication requirements for the gearset depend on the lubrication system design – how efficiently the available lubrication is used. The systems can be categorized as follows: splash lube, directed lube, and pressurized lube.

Splash lubrication is the simplest type of lubrication in use for automotive automatic transmissions today. In this type of system the fluid is sprayed out from radial holes in a shaft near the sun gear, or through a thrust bearing, outward to the carrier. The carrier usually has some method of collecting some of this fluid, such as a machined groove in the inside diameter. From this groove, the fluid passes into localized face grooves in the carrier or planet washer, and into the planet needle bearings. The gears are lubricated by some of the excess fluid that does not get to the washers, and from some of the fluid that exits the needle system through the opposite washer interface. In this type of system it is difficult to know the precise amount of lubricant getting to the needle system, since much of the overall lubricant provided to the system does not enter the washer interface – but is splashed in the gear mesh – or flows by the carrier altogether. If a fully flooded lubrication flow is provided to the needle system, the resulting film thickness will minimize heat generation at that interface. Any additional lube flow provided to the overall system may be used to reduce heat generation in the gear mesh, or carry away more heat by convection.

Directed lubrication utilizes the fluid more efficiently – it is still sprayed from a shaft, or through a thrust bearing, but it is collected in a lube dam, and directed into the planet pins through axial holes. If the lube dam is sealed to the carrier to avoid leakage, this type of system ensures that a known amount of fluid is provided at the inlet (at a known pressure, due to the known rotational speed of the carrier). It gets to the needle bearings through one or more radial holes in the planet pins. From the needles, the fluid exits out both sides of the planet to the washer interfaces. Some of the fluid then splashes in the gear mesh to provide oil film, and remove heat. Gear meshes can also be lubricated from splashed fluid that is from other sources. If more fluid is

supplied than that required to fully fill the lube dam, it will flow by the carrier without providing any additional benefit.

The most efficient system is pressurized lubrication. This system routes pressurized fluid through passages in the carrier (usually they are drilled holes), directly to the planet pins. The pins must have holes that are sealed from exhaust to get the fluid to the needle bearings. This system also provides a known amount of fluid (at the supply pressure) at the inlet. From there it is similar to directed lubrication. Pressurized lubrication systems work even if the carrier is stationary.

Directed lubrication and pressurized lubrication systems can be analyzed using Computational Fluid Dynamics (CFD) to determine the amount of fluid getting to the needle bearing system. The amount of fluid that is available at the inlet (lube dam, or pressurized passages) can be analyzed – the less accurate factor in the analysis is the effective orifice at the exits (washer interfaces). Splash lubrication analysis is much less accurate, since it is not known if the grooves are filled with fluid, nor what percentage of the open area in the carrier or washer face grooves is filled with fluid.

The gearset tested in this project was of the splash lubrication type. It has lubrication supplied from holes in the shafts, and collected by grooves in the carrier bore ID. There are rotating washers with face grooves, and also flat round washers whose ID is greater than shaft OD to allow the fluid to enter the planet needle bearing system.

## Test method

A schematic of planetary gearsets used for the present study is shown in Figure 2. Each planetary gearset has four planet gears. A sun gear in gearset 1 is used as input and it is connected to input shaft of a back to back test fixture. Another sun gear in gearset 2 is held stationary and torque is applied to it. Thermocouples are installed onto outside diameter of each planet gear shaft (gearset 2) where thrust washers are located between carrier and planet gear. “A” and “B” in Figure 2 indicate thermocouple probe locations. Thermocouple wires are connected to four channel slip ring and the amplified signals out of the slip ring are transferred to main computer system.

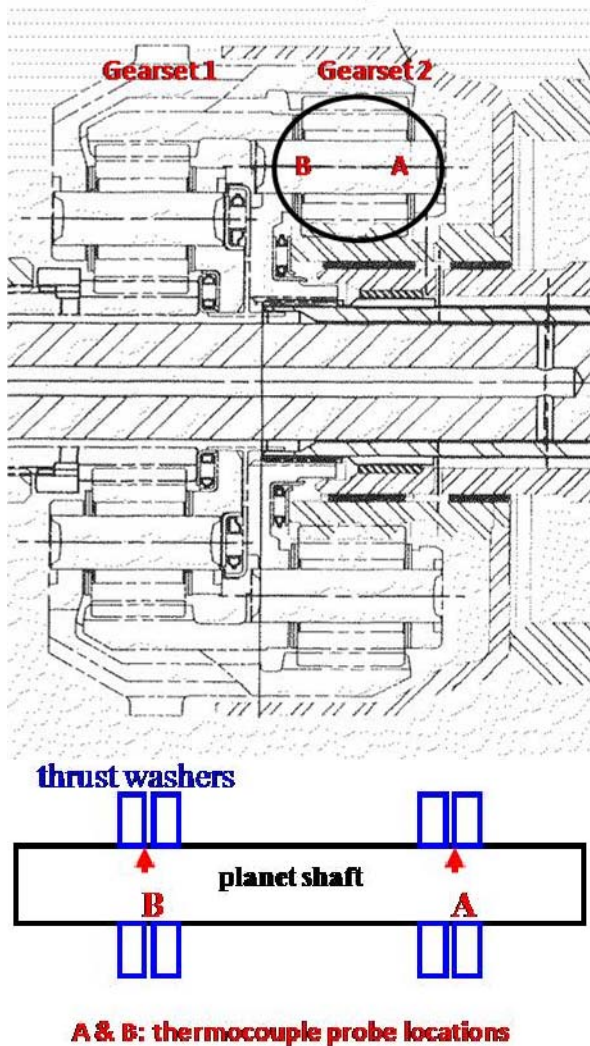


**Figure 2. Planetary gearset test setup for temperature measurements**

Flow rate is controlled at the inlet manifold. It is difficult to calculate the exact amount of flow rate delivered to each gearset because leakage is inevitable at connections and orifices, and some of the fluid in this splash lube system does not get caught by the carrier assembly. Planet gear shafts, carrier and test fixtures were modified for wiring thermocouples to a station where the slip ring was located.

In order to understand how much flow rate is needed to lubricate a planetary gearset, a variety of tests were undertaken at various conditions of torque and speed. Steady state tests and transient tests are performed to examine heat generation inside the planetary gearset. The amount of lubrication required to take away the heat can be

determined by temperature rise inside the gearset. Each steady test runs ten or thirty minutes until temperature signals become stable without significant fluctuations. For steady condition tests a thermal equilibrium was established in less than five minutes.

## Test results

$\Delta T$ 's in each table in test results section represent the average of temperature rises out of four planet gears (gearset 2) as shown in Figure 2. It is "temperature near the planet gear shaft minus sump temperature". Firstly, steady state condition test results for constant torque test, constant speed test and constant power test are represented. Each steady state test was undertaken in arbitrary order and performed twice to see repetitiveness of measurement data. Secondly, transient condition test results where torque and speed change with time are also demonstrated.

### Constant torque test

- input sun gear torque: 300 Nm;
- input sun gear speed: 1000 rpm, 2000 rpm, 3000 rpm;
- inlet lube flow rate: 5 liter/min;
- sump temperature: 90 deg C and 100 deg C;
- thermocouple probes at "A" (Figure 2);
- two trials at each sump temperature for repetitiveness.
- Gearset temperature rises out of the constant torque test are represented in Table 1 and Figure 3. A linearly proportional increase in temperature rise ( $\Delta T$ ), from 12.5 deg C to 24.5 deg C (average of 1st and 2nd) at 90 deg C sump and from 21.5 deg C to 42.5 deg C at 100 deg C, is observed as input sun gear speed changes from 1000 rpm to 2000 rpm.
- As the speed increases from 2000 rpm to 3000 rpm, the temperature rise ( $\Delta T$ ) shows a large difference between 1st and 2nd trials. The difference could be attributable to inconsistent lubrication distribution inside the gearset at the high input power (94.2 kW) condition. For 100 deg C sump at 3000 rpm an overheat ( $\Delta T = 50$ ) might occur to bearing system at the 1st trial and higher temperature rise at the 2nd trial could be associated with continuous damage to the bearing system due to the overheat.

**Table 1. Gearset temperature rises at one constant torque (300 Nm) and three different speed (1000 rpm, 2000 rpm, 3000 rpm) conditions**

| Input sun gear speed, rpm | Gearset temperature rises, $\Delta T$ , deg C |     |                |     | Input power, kW |
|---------------------------|-----------------------------------------------|-----|----------------|-----|-----------------|
|                           | 90 deg C sump                                 |     | 100 deg C sump |     |                 |
|                           | 1st                                           | 2nd | 1st            | 2nd |                 |
| 1000                      | 13                                            | 12  | 21             | 22  | 31.4            |
| 2000                      | 26                                            | 23  | 42             | 43  | 62.8            |
| 3000                      | 33                                            | 27  | 50             | 70  | 94.2            |

- As input power increases by approximately 30 kW, gearset temperature rise increases by about 10 deg C at 90 deg C sump. The increment of the gearset temperature rise due to the power increase varies with the sump temperature.
- A significant wear to thrust washer and discoloration in needle bearings are observed when temperature rise ( $\Delta T$ ) is higher than 50 deg C for longer than 5 minutes.

#### Constant speed test

- input sun gear speed: 4500 rpm;
- input sun gear torque: 50 Nm, 100 Nm, 150 Nm;
- inlet lube flow rate: 5 liter/min;
- sump temperature: 90 deg C and 100 deg C;
- thermocouple probes at "A" (Figure 2);
- two trials at each sump temperature for repetitiveness.

#### **Constant Torque Test (input sun gear torque 300 Nm, 5 liter/min)**

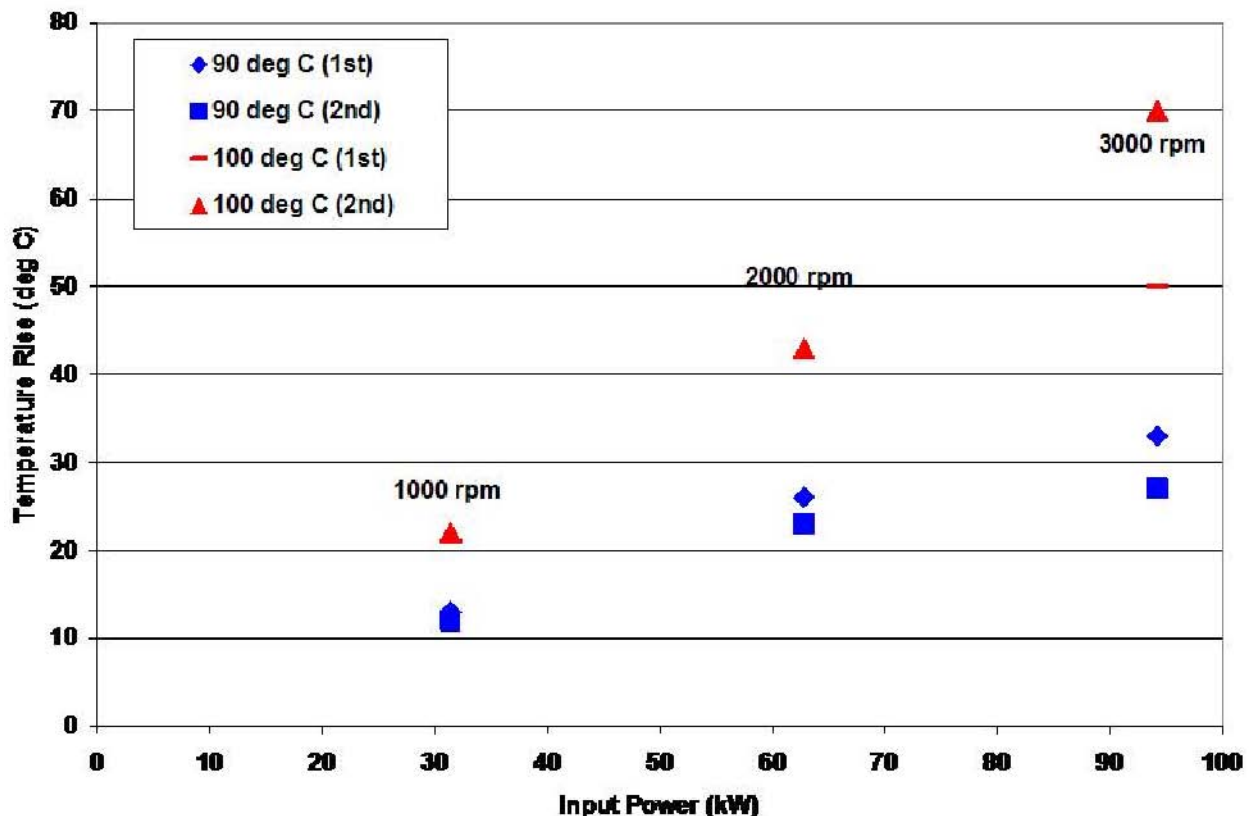


**Figure 3. Gearset temperature rises at one constant torque (300 Nm) with three different speed (1000 rpm, 2000 rpm, 3000 rpm) conditions at two sump temperatures (90 deg C and 100 deg C)**