

Separation of Lubrication and Cooling in Oil-Jet Lubricated Gears

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American Gear Manufacturers Association



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ABSTRACT:

Results from the tests on a high-speed back-to-back stand ($v_t = 70 \text{ m/sec} = 13725 \text{ ft/min}$) show the influence of the separation of lubrication and cooling oil supply on the gear temperatures (scuffing load capacity) and efficiency. The gear mesh is only given a minimum oil flow rate necessary for lubrication of the working tooth flanks. Cooling is provided by spraying oil onto the inner surfaces of the rims. This leads to a reduction of the total oil flow rate of up to 60% compared to the currently recommended flow rates. In spite of the reduced total oil flow rate the temperature level of the gears can be kept low and the efficiency slightly increases due to reduced hydraulic losses in the gear mesh.

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1 Introduction

Theoretical analyses and experimental tests [2,3] with oil-jet lubricated gears have shown that the actual delivered oil flow rate is considerably higher than necessary to maintain a load carrying, elasto-hydrodynamic oil film. Excessive oil flow rates cause a reduction in efficiency, especially in high-speed gears with a pitch line velocity of $v_t = 200$ m/sec (≈ 40000 ft/min) the load-independent losses ("hydraulic losses" due to lubricant, see [14]) rise considerably. Too low oil flow rates on the other hand cause an unacceptable temperature rise of the working tooth flanks, since the frictional heat produced in the gear mesh is insufficiently carried away by the oil.

The considerations of reducing the oil flow rate and keeping constant gear mesh temperatures lead to a separation of the oil supply into a lubricating and a cooling oil supply. In doing so, the mesh is only given a minimum oil flow rate necessary for lubrication of the working tooth flanks. The gears are cooled outside the mesh area by oil that is sprayed onto the inner surfaces of the gear rims on both sides of the gears.

The goals of the investigations in [1] consisted of determining the influences of the separation of lubricating and cooling oil supply on the pinion and wheel temperatures, on the power losses, and on the wear of a single-stage helical gearset as a function of load and pitch line velocity. It was to be shown how and to what extent the total oil supply can be reduced.

2 Background

An overview of the actual situation concerning "Separation of lubricating and cooling oil supply of high duty gears" is given by a literature research [4] as well as the explanations in [1]. It was proved that up to now no experimental investigations have been done as to this subject and that the theoretical knowledge, too, is insufficient.

Oil jet lubrication is usually used for gears with pitch line velocities of $v_t > 30$ m/sec (≈ 6000 ft/min). The oil jet lubrication variants common in research and industry basically vary in position of the nozzles and their spray direction into the gear mesh. As shown in Fig. 1 they can be classified as follows:

- Variant A: Oil-jet is directed towards the pitch point perpendicular to the center line.
- Variant B: Oil-jet is directed towards the gear center.
- Variant C: Oil-jet is directed towards the gear mesh respectively towards the face parallel to the axis of the gears.
- Variant D: Centrifugal lubrication; Fling-off cooling.

For gears for industrial usage basically variants A1, A2, A1+A2 as well as B1, B2, B1+B2 are used. In literature [5] you generally find the statement that as to the lubrication "Spraying into the engaging mesh" (A1,B1) and as to cooling "Spraying into the disengaging mesh" (A2,B2) should be preferred. However, several experimental investigations with pitch line velocities up to $v_t = 100$ m/sec (≈ 20000 ft/min) have shown, that the gear bulk temperatures are considerably higher (lower scuffing load capacity) when spraying into the disengaging mesh. On the other hand temperature measurements at velocities in the range of 100 to 200 m/sec (≈ 20000 to 40000 ft/min) support the statement that A2 is with regard to cooling more appropriate than A1.

At pitch line velocities of more than 100 m/sec (≈ 20000 ft/min) frequently both variants are combined (A1+A2,B1+B2). The oil flow sprayed into the engaging mesh has to be kept as low as possible because of hydraulic losses, so that only the lubrication is guaranteed, while much more oil is sprayed into the disengaging mesh for cooling. It is a common fear that at these velocities the oil sprayed into the disengaging mesh will be immediately rejected by the approaching tooth flanks. The oil film thickness on the tooth flanks will be reduced due to the extremely high centrifugal forces till the next tooth engagement occurs, which can lead to insufficient lubrication conditions. It is a general opinion that the cooling effect is much better when oil is sprayed into the disengaging mesh, because there is a higher temperature difference between the "hot" flanks coming out of the mesh and the oil that is sprayed in.

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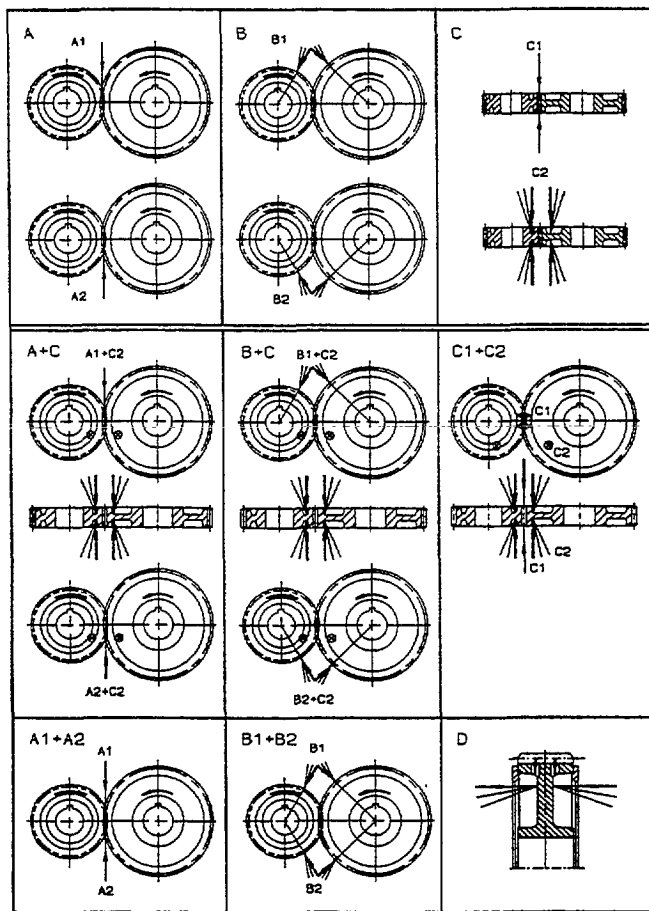


Fig. 1: Injection Variants A,B,C,D with and without separated cooling oil supply

For separate supply of lubricating and cooling oil, combinations of injection variants A1, A2 and C2 as well as B1, B2 and C2 would be well suited (see Fig. 1). For lubrication the oil is sprayed either directly into the mesh or radially onto the gears. Cooling is provided from oil sprayed onto the inner surfaces of the gear rims.

Dudley/Richardz /9/ mention in their final conclusions that spraying of cooling oil onto the faces of the gears is a procedure especially used for aviation gears with thin rims and webs (see Fig. 2).

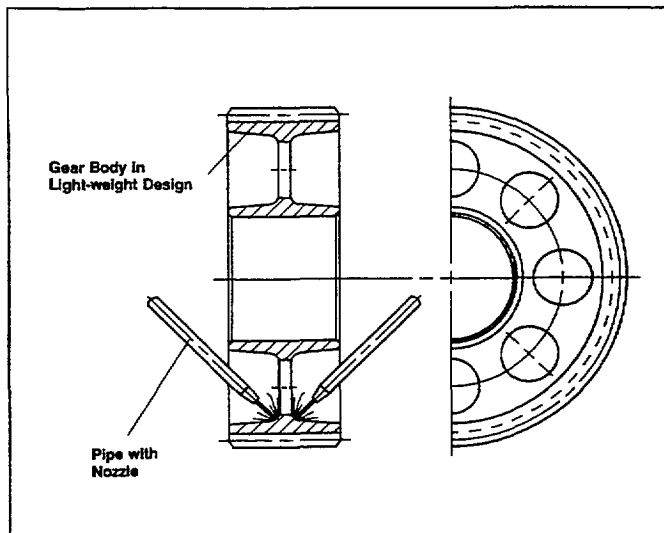


Fig. 2: Lubrication of aviation gears acc. to /9/

The extremely fast running but relatively low loaded flanks in aviation gears can (according to /9/) even be operated without "direct" lubrication of the gear mesh, because the oil mist caused by the horizontal spraying of oil on web and rim is sufficient for lubrication. This, however has to be questioned for very wide gears.

With a separate cooling oil supply the question of the minimum lubricating oil flow necessary for an operational lubrication becomes very important. In industry the determination of the lubricating oil flow rate is done by taking into consideration the expected power losses and the heat balance of the gear box /8,9,12/, the face width /8,10,11/ and the EHD-Theory /2,13/. To be able to compare the different recommendations as to the amount of the lubricating oil flow rate, the operating conditions of a single-helical gearset were set up in Table 1. Table 2 shows the calculated results.

The recommendations regarding the heat balance of the gear box show considerably higher oil flow rates. This is mainly due to the very small permitted temperature difference between oil inlet and oil outlet. A doubling of the allowed temperature difference would cause a halving of the oil flow rate and thus less hydraulic losses.

When calculating the lubricating oil flow rate by regarding the face width one is on the safe side because of experiences made with slowly running gears.

The according to /2,13/ calculated minimum oil flow rates cannot be considered as recommendations, since the frictional heat in the gear mesh can no longer be dissipated. The idea of cooling the gears by spraying oil on the faces requires the determination of the optimum of sufficient lubrication while minimizing the hydraulic losses and dissipation of the frictional heat.

3 Experimental Equipment

3.1 Back-to-Back Test Stand, Test Gears

Fig. 3 schematically shows the back-to-back test stand with the experimental arrangement for the temperature and efficiency investigations. Fig. 4 shows the operable test stand in the gear lab. Basically the test stand corresponds to a design which has been reported several times in literature /1,2/. Table 3 shows the technical data of the two different test gearsets. Profile B is designed for balanced sliding at tooth root and tip, while the sliding velocity at the tip of the pinion of profile E is approximately 16 % higher than at the wheel. Since the tests did not show significant differences in the following only the results with profile B will be discussed.

3.2 Positions of Nozzles

Fig. 5 shows the view of the test gear box with the pipework for the separate supply of lubricating and cooling oil without gear cover. At the end of the pipe of the lubricating oil supply of the gear mesh a spray-head with 3 solid stream nozzles is mounted. It distributes the oil equally along the whole width of the gear mesh. The axes of the nozzles in the spray-head are directed towards the pitch point perpendicular to the center line.

Two oil distributors outside the housing guarantee an equal distribution of the cooling oil onto the inner surfaces of the gear rims. The axes of these solid stream nozzles point on the fillet radius between the web and the rim. The two injection variants A1+C2 and A2+C2 are realized by corresponding choice of the rotation direction.

3.3 Measuring Equipment

In the back-to-back test stand two torque-meters are mounted for measuring the transmitted power and the power losses (see Fig. 3). The temperatures of the gears and the oil temperatures in the housing

Transmitted Power	P	kW	1000
Permitted temperature difference between oil inlet and oil outlet	$\Delta\theta$	K	15
Outside surface of housing	A	m ²	0.5
Number of teeth on pinion/wheel	z_1/z_2	-	31/47
Face width	b	mm	40
Pitch line velocity	v_t	m/sec	60
Helix angle	β	deg.	10
Tooth flank surface roughness	R_z	μm	2.5
Characteristic Values of Mineral Oil: Specific heat capacity $c = 2000 \text{ Wsec/(kgK)}$ Density $\rho = 850 \text{ kg/m}^3$			

Table 1: Operating conditions of single-helical gearset

Reference ; Source Reference Value	Equation; Standard Value	Recomm. for $\dot{V}_z$ in l/min	Comments
Niemann /12/ Gear Mesh Losses P_{VZ} Temperature Difference between Oil Inlet and Oil Outlet $\Delta\theta$	$\dot{V}_z = 40,8 \cdot (P_{VZ}/\Delta\theta)$ with $P_{VZ} \approx P \cdot ((0.1/z_1/\cos\beta) + 0.03/(v_t+2))$	10.2	- $P_{VZ} \approx 3.76 \text{ kW}$; $\dot{V}_z$ for full compensation of P_{VZ} by the oil
Niemann ; Winter /8/ 1. Face Width b 2. Power Losses P_V Heat Dissipation P_{KG} due to Convection	$\dot{V}_{zmin} = 0.08 \dots 0.1 \text{ l/(min}\cdot\text{mm)}$ $\dot{V}_z = (P_{VZ} - P_{KG})/(c \cdot \rho \cdot \Delta\theta)$	3.2-4.0 17.4	- Estimation - at complete dissipation of P_{VZ} by the oil $P_{KG} \approx 0.8 \dots 1.2 \text{ kW/m}^2 = 0.6 \text{ kW}$ $P_{VZ} \approx 0.8 \% \text{ of } P = 8 \text{ kW}$
Dudley ; Richardz /9/ Transmitted Power P	$\dot{V}_z = 1.27 \text{ l/min per 100 kW of } P$ $\dot{V}_z = 0.64 \text{ l/min per 100 kW of } P$ $\dot{V}_z = 0.42 \text{ l/min per 100 kW of } P$ $\dot{V}_z = 0.32 \text{ l/min per 100 kW of } P$	12.7 6.4 4.2 3.2	- generous supply $\Delta\theta$ up to 10°C - normal supply $\Delta\theta$ up to 24°C - lean supply $\Delta\theta$ up to 38°C - starvation supply $\Delta\theta > 38^\circ\text{C}$
Shell /10/ 1. Transmitted Power P 2. Face Width b	- Diagram - $\dot{V}_z = 0.55 \dots 0.8 \text{ l/(min}\cdot\text{cm)}$ $\dot{V}_z = 1.6 \text{ l/(min}\cdot\text{cm)}$	23.0 2.2-3.2 6.4	- friction coeff. $\mu_{mZ} = 0.06$; allow. temp. rise $\Delta\theta = 8^\circ\text{C}$ - for high-speed gears
Mobil /11/ Face Width b	$\dot{V}_z = 0.5 \dots 1 \text{ l/(min}\cdot\text{cm)}$	2.0-4.0	
Ohlendorf /13/ Tooth Flank Roughness	$\dot{V}_{zmin} = 2 \cdot b \cdot v_t \cdot (R_{p1} + R_{p2})/10^4$	1.44	- $R_{p1} + R_{p2} = 2 \cdot (0.6 \cdot R_z) = 3 \mu\text{m}$
Schober /2/ EHD-Theory	$\dot{V}_{zmin} = 2 \cdot b \cdot v_t \cdot h_i/10^4$	> 0.09	- $h_i > 0.18 \mu\text{m}$

Table 2: Minimum oil flow rate for sufficient lubrication

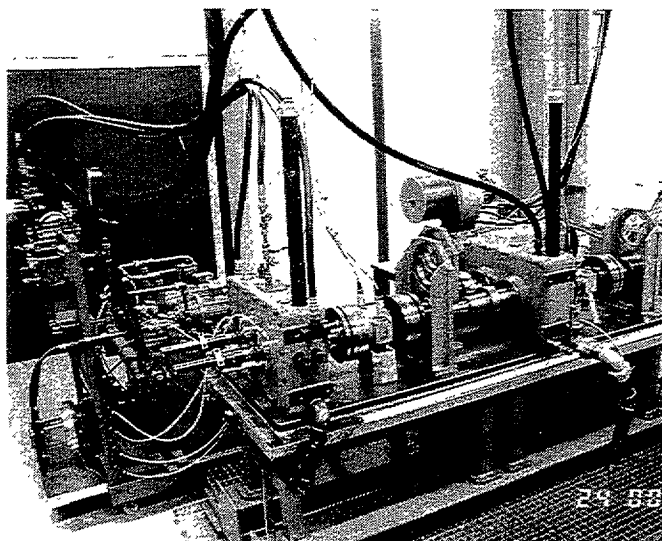
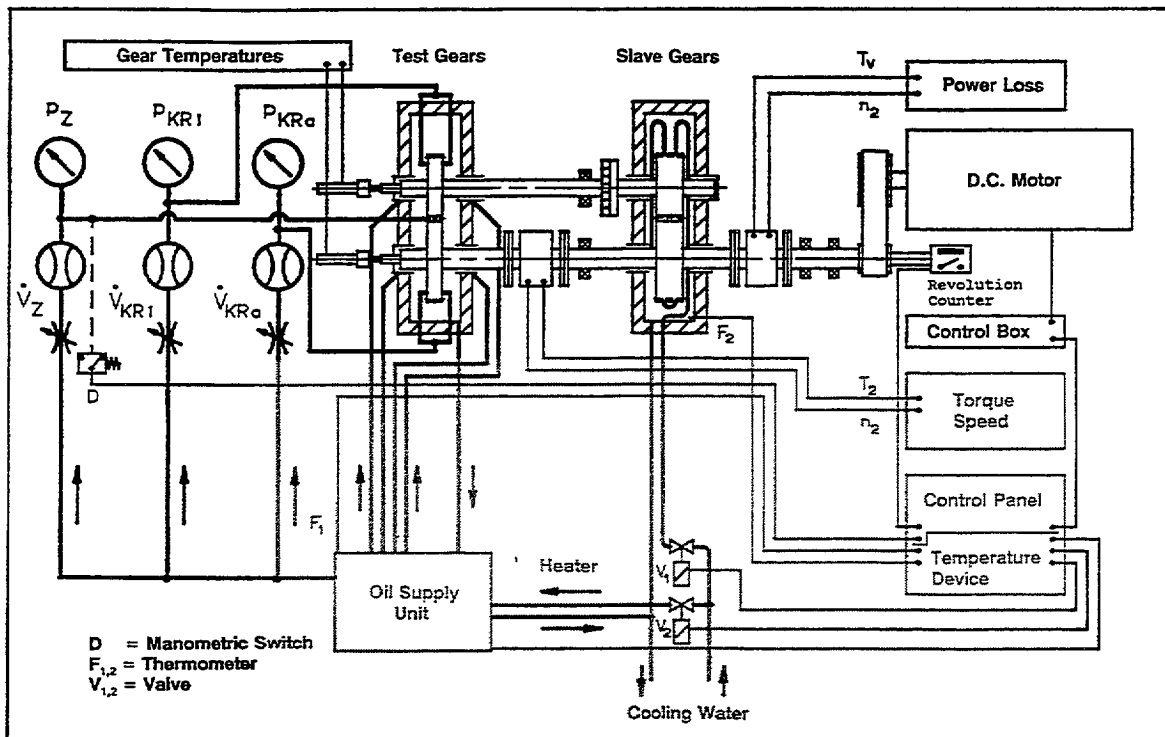


Fig. 4: Back-to-Back Test Stand in the Gear Lab

Fig. 5: Positions of nozzles at separation of lubrication and cooling oil supply (Injection variants A1,A1+C2,A2,A2+C2)

