Tribology International 71 (2014) 7–16

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Efficiency of a gearbox lubricated with wind turbine gear oils

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ARTICLE INFO

Article history:
Received 5 August 2013
Received in revised form
24 October 2013
Accepted 26 October 2013
Available online 4 November 2013

Keywords:
Gearbox efficiency
Wind turbine gear oils
Coefficient of friction

ABSTRACT

In this study a two stage multiplying gearbox with helical gears and four fully formulated wind turbine gear oils were tested, on a back-to-back gearbox test rig with recirculating power, at low input speeds (100–500 rpm) and high input torques (500–1000 Nm). The gearbox oil sump temperature was set free.

A numeric power loss model simulating all the relevant power loss mechanisms was implemented, aiming to evaluate the relative influence of each power loss component.

The experimental results have shown that each wind turbine gear oil formulation generated different power loss resulting in distinct stabilized operating temperatures.

1. Introduction

As proven source of clean and affordable energy, wind resources clearly have a vital role to play in energetic sustainability [1]. Wind turbines are one of the machines that take advantage of the wind energy to generate electrical power. The blades on a wind turbine rotate at very low speed which is not adequate for conventional electrical power generation. The blades speed is then multiplied using multiple stages gear trains before meeting the generator. These gearboxes already have good efficiency, nevertheless, any small efficiency increase will have a significant impact not only due to the high power involved, but also due to the increasing number of wind turbine farms all across the globe.

A large number of tests totaling over 300 h of experimental work were performed. The test gearbox was installed on a multiplying configuration and the oil sump temperature was set free. The tests were conducted at low speeds (100, 200, 400 rpm), and high torques (500, 750, 1000 N m). The operating temperatures and torques were recorded and their averages at stabilized operating conditions were considered. Between the selected gear oils three different base oils can be found. A Mineral, a Polyalkylene Glycol, a Polyalphaolefin and a Mineral+PAMA mixture were considered.

Accurate power loss predictions at the design stage will allow the development of more efficient and reliable designs in less time, ultimately saving resources not only at the design stage but also during operation.

According to Höhn et al. [2] the power loss in a gearbox consists of gear, bearing, seal and auxiliary losses.

Gear and bearing losses can be separated in no-load and load losses. No-load losses occur with the rotation of mechanical components, even without torque transmission. No-load losses are mainly related to lubricant viscosity and density as well as immersion depth of the components on a sump lubricated gearbox, but it also depends on operating conditions and internal casing design. No-load rolling bearing losses depend on type and size, arrangement, lubricant viscosity and immersion depth.

Load dependent losses occur in the contact of the power transmitting components. Load losses depended on the transmitted torque, coefficient of friction and sliding velocity in the contact areas of the components. Load dependent rolling bearing losses also depend on type and size, rolling and sliding conditions and lubricant type [3].

2. Gearbox test rig

The gearbox test rig (Fig. 1) follows the principle of recirculating power. This test rig allows test gearbox input speeds from 100 to 1900 rpm and input torques from 100 to 1300 N m. The gearbox oil sump temperature was set free.

In this gearbox test rig it is possible to test different gearboxes. The gearboxes must fit within the dimensional constraints of the test rig and must be reversible, since two identical gearboxes (test and slave) are used to close the kinematic loop (Fig. 1).

The gearbox test rig allows monitoring and recording several operating parameters, namely:

- input and output torque/speed of the test gearbox;
- room temperature;

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0301-679X/$ - see front matter
http://dx.doi.org/10.1016/j.triboint.2013.10.017

Nomenclature

b	tooth face width, m
d_{sh}	shaft diameter, mm
dm	mean bearing diameter, mm
f_0	coefficients for rolling bearing loss calculation, –
F_{bt}	tooth normal force, transverse section, N
l_{min}	minimum length of the line of contact, mm
m	module, m
M	total frictional torque, N mm
M_{rr}	rolling frictional torque, N mm
M_{sl}	sliding frictional torque, N mm
M_{seal}	frictional torque of the seal(s), N mm
M_{drag}	frictional torque of drag losses, churning, splashing, etc., N mm
n_{mesh}	number of gear meshes, –
P_V	power loss, W
P_{VZ0}	gears no load power loss, W
P_{VZP}	gears load power loss, W
P_{VL}	rolling bearings power loss, W
P_{VL0}	rolling bearings no load power loss, W
P_{VLP}	rolling bearings load power loss, W
P_{VD}	seals power loss, W
R_a	arithmetic average roughness of pinion and gear, μm
$r_{a1,2}$	tip radius, m
$r_{p1,2}$	pitch radius, m
T_{oil}	oil sump temperature, °C
T_{room}	room temperature, °C
u	gear ratio (z_1/z_2), –
$v_{\Sigma C}$	sum velocity at pitch point, m/s
V_t	pitch line velocity, m/s
x	addendum modification, –
X_L	lubricant factor, –
$z_{1,2}$	number of teeth, –
α_t	transverse pressure angle
β	helix angle
β_b	base helix angle
ε_α	length of path of contact, –
ε_1	length of path of addendum contact of pinion, –
ε_2	length of path of addendum contact of gear, –
η_{oil}	dynamic oil viscosity at operating oil sump temperature, mPas
ν_{oil}	kinematic oil viscosity at operating oil sump temperature, mm^2/s
μ_{mz}	mean coefficient of friction at the gear mesh, –
$\Delta T_{or} = T_{oil} - T_{room}$	
Λ	specific film thickness, –
$\Lambda_{5\%}$	critical specific film thickness for a failure probability of 5%, –
ν	kinematic viscosity, m^2/s
ρ	oil bulk density, kg/m^3
ρ_C	equivalent radius of curvature at pitch point, mm

- oil sump temperature in the test gearbox;
- external wall temperature of the housing on both test and slave gearboxes.

Industrial grade 3 wire $Pt100$ RTD's were used to monitor the aforementioned the test rig temperatures.

Table 1 shows the performance specifications of the torque transducers.

Prior to the tests with the selected gear oils, several runs (in different days), at the same operating conditions, were conducted in order to assess the stability of the test rig and the repeatability of the measurements. The temperature measurements (ΔT_{or} and ΔT_{ow}) presented a standard deviation of $\approx 0.15\%$ of the measured temperatures between tests. The torque measurements presented a standard deviation of $\approx 0.2\%$ of the measured value. The speed measurements also presented a standard deviation of $\approx 0.2\%$ of the measured speed.

3. Test gearbox

Fig. 2 shows a schematic view of the test gearbox. This gearbox has three shafts where five gears are mounted. The gears in the middle shaft (pinions 2 and 3) are keyed while the gears on the

Table 1
Performance specifications of the torque transducers.

Transducer	Capacity (N m)	Non-linearity (%)[a]	Hysteresis (%)[a]	Repeatability (%)[a]	Temperature range[b] (°C)
Input torque	5650	± 0.026	± 0.031	± 0.05	$+21$ to $+77$
Output torque	2250				

[a] % of rated output.
[b] Compensated.

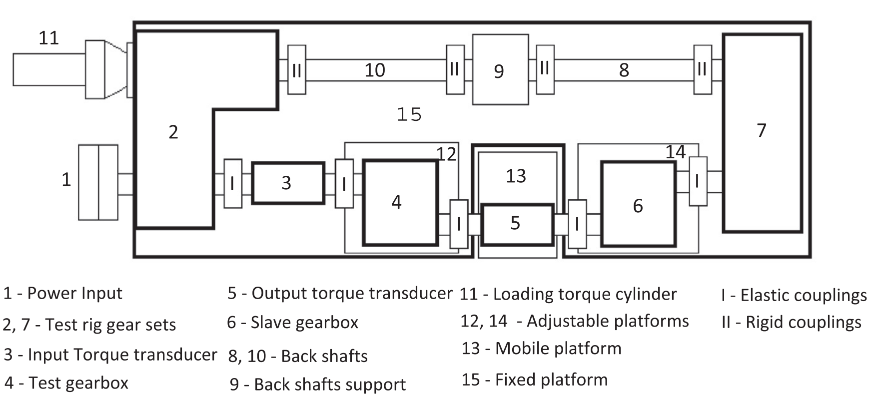

Fig. 1. Top view scheme of the gearbox test rig.