Design and Development of Torque Testing Rig for a Gearbox

Amruta Lomate[1], Suhas Mohite[2], and Rahul Shinde[3]

[1]Mechanical Department, Bharati Vidhyapeeth, College of Engineering, Pune, Maharashtra
Amu2384@yahoo.co.in
[2]Mechanical department, Govt. College of Engineering, Karad, Maharashtra
[3]Executive director, Cyclo Transmission Ltd. Satara, Maharashtra
mohitess@yahoo.com, cyclobox@rediffmail.com

***Abstract* - Gearbox is an indispensae element of power transmission drives of most mechanical systems. Therefore, it is very essential to assure the performance of gear drives before they are put to use to ensure the trouble free functioning of systems. While testing performance of the gearbox, it is importance to check the torque carrying capacity at rated speeds. This work presents the design and development of a torque testing rig for 0.5-10 kN-m capacity carried out for a gearbox having multi-plate brake system. While calibration of torque testing rig experimentally measured data is compared with the theoretical calculations and a good agreement between experimental and theoretical calculations is observed.**

***Index terms* –Gear box, design, torque, calibration**

I. INTRODUCTION

Gearbox is an indispensable element of power transmission drives of most mechanical systems such as vehicles, machinery and cranes, etc. The main function of the industrial gear drive is to reliably transmit torque and rotary motion between prime mover and driven piece of equipment, at acceptable level of noise, vibration and temperature. When one or more of the preceding operating characteristics exceeds allowable limits, it can lead to failure of the gearbox. Manufacturing companies or customers have to bear big cost and time for the failure of gear drive system during operation. Therefore, it is very essential for the manufacturer to assure the quality of gear drives before they are put to use to ensure trouble free functioning of systems.

While testing the performance of gearbox, it is importance to check the torque carrying capacity at different speeds. At present, Cyclo transmission ltd. does not have any torque testing facility. So to measure the torque; company has to rely on theoretical calculations according to ISO (International Organization for Standardization) and AGMA (American Gear Manufacturers Association) standards to specify torque ratings. Therefore, it is very necessary for them to have a test rig for gearboxes to assure quality of the gearbox. Torque testing rig is an essential facility that every gear manufacturing company must have.

To fulfil above needs of the torque testing rig, a literature survey was undertaken with the view to finalize its features and specification.

Different types of torque testing rig available in market are studied. From above literature it is found out that each method has some advantages and disadvantages. For torque testing back to back, digi-torque and multi-plate brake systems are studied “Ref. [1-3]” For back to back arrangement two similar gear boxes are required, which may not be available in some of the cases. In digi-torque method the flywheel was used to apply the torque, which is not suitable for high torque measurement. Both these methods are not suitable for this application.

Multi-plate brake torque testing system (torque applying) is suitable because it can provide wide range of torque (from 500 N-m to 10 KN-m) at different speeds. The brake plates are partially wetted to avoid wear and control the temperature of brake plates. For actuating multi-plate brake system, actuators are needed. While studying different brake actuators [4]., it was found that hydraulic actuator is suitable for applying the pressure for multi plate brake system, because it provides fast response along with smooth engagement by controlling the rate of pressure built-up with a pressure control valve. By referring to the literature in the point of torque measurement, it was found out that canister type load cell can provide high torque measurement because this load cell is suitable for high torque or load range and also eliminates the need of prior checking [5]. Finally it is decided to design and develop a hydraulically actuated multiplate brake torque testing rig for different gearbox, maximum torque of 10 kNm at rated speed.

II. METHODOLOGY

The proposed system has multi plate brake assembly as a main and core component. The assembly is mounted in between the bracket and fixed on and around the shaft.

Figure 1 Outline of the proposed gearbox test rig

Multi-plate brake system is used to apply the torque on the gearbox. For actuation of the brake, hydraulic actuator was used. Brake system is directly coupled with the gearbox with the help of coupling or gear pair. As the shaft is coupled with gearbox shaft, brake force is transmitted to the gearbox and at the same time torque arm (torque arm is fixed with brake assembly). Torque arm in turn exert the same amount of force on load cell.

To maintain normal temperature of the brake plate, proper cooling system is designed. The total system is divided into following sub systems, viz.,

A. Brake assembly
B. Actuation system
C. Cooling system
D. Torque measurement and display unit

III. DESIGN AND MANUFACTURING

To design a torque testing rig for a gearbox is an innovative idea. On the basis of the conceptual design, the main system is divided in to a number of subsystems carrying out specific functions as follows.

A. Brake assembly
B. Actuation system
C. Cooling system
D. Torque measurement and display unit

The overview of the test rig depicting different sub systems is as shown in “Fig. 2”.

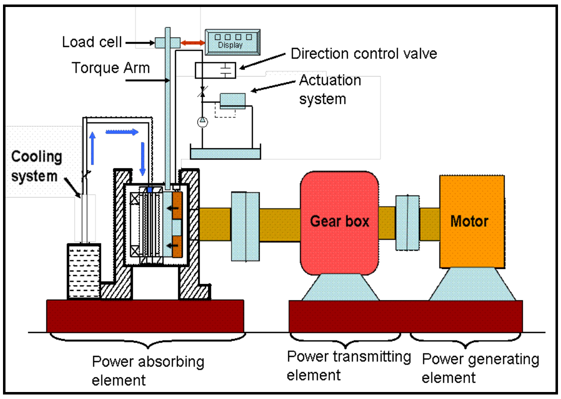

Figure 2 Overview of the designed torque testing rig

A. *Brake Assembly*

Brake assembly is the main part of test rig. During design of brake assembly, following procedure was followed. Selection of materials “Ref. [6, 7]”, design and analysis of various components like shaft, brake plate, piston, cylinder, torque arm, selection of bearings, oil seals, and piston rings etc were done. Some of the important calculations are shown below. For finding number of brake plates required we used uniform wear criteria for brake plate design. “Ref. [8, 9]”

$$M_t = \frac{\mu FZ \times (D_{1O} + d_{2i})}{4} \tag{1}$$

Where, M_t is maximum torque, F is working force, Z is number of matching surfaces, μ is coefficient of friction, D_{10} is inner diameter of outer brake plate and d_{2i} is outer diameter of inner brake plate. Total number of brake plate is 27. Shaft is basically designed for crushing, shear, torsional and bending type of loading. Maximum shear stress theory is used for determining diameter of shaft because material of the shaft is ductile

$$\tau = \frac{16}{\pi d^3}\sqrt{(k_b M_b^{\,2}) + (k_t M_t^{\,2})} \tag{2}$$

Where, τ is maximum shear stress, d is diameter of shaft, k_b and k_t are shocks and fatigue factors respectively and M_b is bending moment. The minimum diameter of the shaft is 120mm.

Actuation System

The hand operated hydraulic actuator has capacity of 20 bars is used.

C. *Thermal Calculations and Design of Cooling System*

In case of multi plate brake, heat dissipation is the main problem. As the temperature of oil increased above 90^0C, oil does not obeyed its properties, because that wear of the brake plates will be takes place so to avoid such problem water cooling jacket is provided.

Torque Measurement and Display Unit

For torque transfer from brake assembly to the load cell, torque arm of 1000 mm was used. For torque measurement canister type load cell of capacity 2000 kg is used with digital display. Resolution of ±0.1 kg-m in 100 kg-m is used.

After design manufacturing and assembly is done as per drawings. The actual view of test rig is as shown in “Fig. 3”

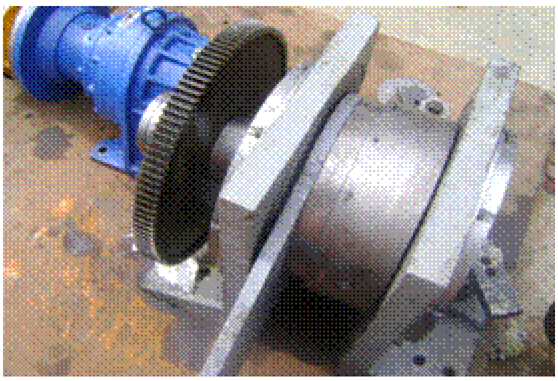

Figure 3 Torque testing rig

IV. EXPERIMENTAL WORK

In the experimental work the set-up the test rig as shown in “Fig.4” and by using the few gearboxes calibration of test rig have been done in following way. The analytically determined torques values were compared with the measured torque values for different models of the gear box. The analytical and the experimental readings are compared.

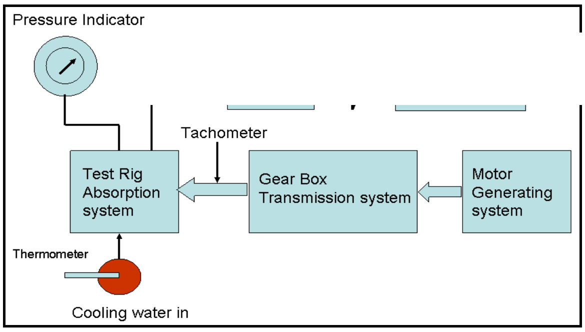

Figure 4 Schematic arrangement of the test rig for calibration

These results are given for different pressure values ranging from 2 bars to 20 bars. The calibrated measurement units (load cell, digital display and pressure gauge) are used. For finding theoretical value following equation is referred.

$$M_t = \frac{\mu \times \pi \times Z \times (D^2 - d^2) \times (D_{1O} + d_{2i})}{16} \times P_h$$

(3)

Where, P_h is hydraulic pressure, D is outer diameter of piston and d is inner diameter of piston. The test rig can also be tested for duty cycle test of some of the gear boxes.

V. RESULTS AND DISCUSSION

The testing has been done on different types of gearboxes, different types of testing is done the results are discussed below. Calibration of the test rig carried out by comparing experimental result with theoretical values. The test results of a few models are done, one is discussed below.

1. Gear box Model number 4010-4,[11].
   Specifications: Input power- 10H.P.,
   Output speed- 5.75 r.p.m,
   Output torque- 10.5 Nm.

Testing was carried out for 10 minutes at different constant pressure values and readings were observed after every 2 minutes to allow for the system to get stabilized.

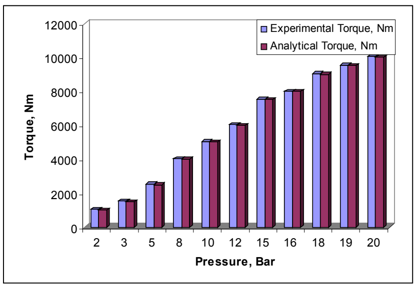

Figure 5 Comparison of analytical and experimental torque values

Comparison of analytical and experimental torque values

Sr. No.	Pressure	Experimental Torque	Analytical Torque	Absolute error	% Error
1	2	1044	1002	42	4.12
2	3	1545	1503	42	2.79
3	5	2550	2505	45	1.79
4	8	4052	4008	44	1.09
5	10	5050	5010	40	0.79
6	12	6055	6012	43	0.71
7	15	7559	7515	44	0.59
8	16	8061	8016	45	0.56
9	18	9067	9018	49	0.54
10	19	9568	9519	49	0.51
11	20	10069	10020	49	0.49

It is observed that the experimental readings for torque are higher than analytical values by an absolute margin of 40-50 Nm for the entire torque range. In terms of the percentage this accounts for approximately 5 to 0.5%. Further, higher error (5%) is observed at lower torque (and lower pressure) whereas, lower error (0.5%) is observed at higher torque values. The reasons for this error could be attributed to various causes that are not considered in theoretical calculations. The possible causes are mentioned below.

a) In analytical calculation, friction in the bearings, oil seals and other moving parts is not considered.
b) The tested gear boxes have a lower operating speed (5-10 rpm) at such speeds the stick-slip phenomenon in between the brake plates is more significant, this could be one of the sources of the error.
c) Inertia of the system.
d) When the hydraulic pressure increases the friction in the various parts increases.
e) Errors in load cell readings.
f) Errors in display unit.
g) The error in the pressure gauge also the pressure indicator is analogue display so the manual mistake may be occurred.

The obtained results were compared to the theoretically calculated and a good agreement between the experimentally obtained and theoretically calculated results were seen. The analytical and experimental values are closely matched.

A. Comparison of errors in experimental values with respect to torque

After calibration of the test rig, it was found that in the system there are three types of error, error in load cell, error in brake assembly and error in digital display. After calibration of test rig, total error was determined for each torque value, which is tabulated in the "Fig. 6"

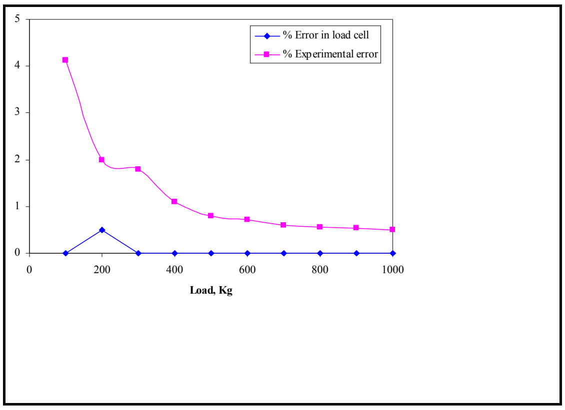

Figure 6 Error analysis of torque testing rig

While actual testing of the gearboxes, corresponding error factor should be added in the experimental readings. Gear box was tested for 10 minutes. Reading was observed for every two minutes and average torque reading was taken, which is shown in below "Table 2".

Table 2 Tests conducted on the different models (output torque)

Sr. No	Model No	Actual torque	Experimental values	Output speed
1	4005-2	556	590	88.5
2	4005-2	754	794	65.3
3	4005-2	1049	1090	47
4	4006-3	2276	2321	20.8
5	4006-3	3142	3184	15
6	4007-3	3847	3889	12.3
7	4007-3	4290	4331	11
8	4007-3	5210	5250	9.1
9	4008-3	6500	6542	7.27
10	4008-3	7280	7323	6.52
11	4007-4	8477	8517	5.35

From the above table, it was observed that all the gear boxes are successfully running for output torque at given speed.

B. Duty cycle test

For the gearbox, it is necessary to carry out the duty cycle test. One such result of the duty cycle tests conducted on gearbox is presented in "Fig.7". The gear box model 4005-2 with an output speed 52.4 rpm and output torque 940 Nm.

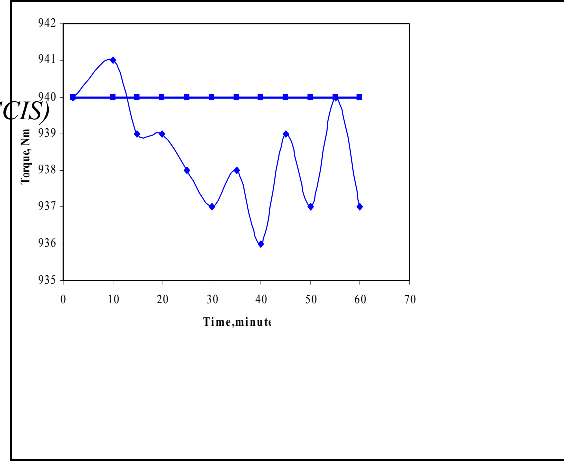

Figure 7 Comparison of analytical and experimental torque readings with time

From above graph it was observed that the torque value is fluctuating in-between 936 to 941Nm with respect to time. That variation is negligible (approx. 0.5%).

C. *Relation between power and outlet temperature*

Temperature is an important parameter while calibrating the test rig, because if the oil temperature is increased beyond 90^0 C, the lubricating oil will lose its properties and rig will not work efficiently. Cooling jacket is used for heat dissipation purpose. Water is used as a coolant. The mass flow rate of the water was taken as 9.5 lit/minutes.

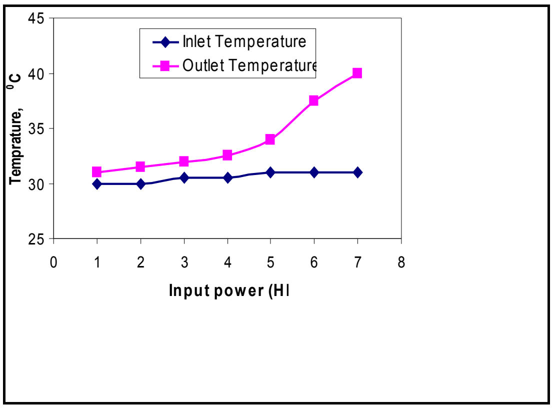

Figure 8 Comparison of temperature and input power at different models

It is observed in "Fig. 8" that as the input power increases the temperature values also increase steadily.

XI. CONCLUSION

Design and development of hydraulically actuated multi-plate brake system is carried out for measuring torque on the gearbox. System is designed for measuring torque from 0.5 kNm to 10 kNm at rated speed. The test rig is calibrated with theoretical values. During calibrating of the systems, it is found out that the maximum error in the measured torque using the test rig is about 50 Nm (carried out on few gear boxes using the test rig and the results are in good agreement with the theoretical values. The test rig can also be used for peak load test and duty cycle test of some of the gear boxes. Temperature of cooling water is measured and it is found to be well within the accepted limits. The overall performance of the test rig is found to be trouble free.

XII. ACKNOWLEDGMENTS

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