Full-Load Testing of Large Gearboxes Using Closed-Loop Power Circulation

R.B. Frost
University of New South Wales, Kensington, Australia

T.R. Cross
John Welsh Pty. Ltd, Melbourne, Australia

Summary: This method of testing large gearboxes or, indeed, any power transmission element, has numerous advantages and offers the possibility of large savings in time, energy, and plant, if the overall situation is conducive to its use. This usually requires that several such units need to be tested, and that they can be conveniently connected to each other in such a way as to form a closed-loop drive train. No power sink is required, and the drive input system has only to make up power losses. The level of circulating power is controlled by the torque, which is applied statically during rotation, and the drive speed. Principles, advantage, and limitations are described, together with recent experiences in the only known large-scale usage of this technique in Australia.

Introduction

Full-load testing of large gearboxes and other power transmission modules or components has traditionally been carried out with the testing drive train arranged in a schematically linear or open manner, as shown in Fig. 1a. This necessitates the use of a power source and a power sink, or brake, each with a capacity at least equal to that of the required test. Further, all energy which passes through the gearbox under test, which can be considerable, is wasted in the sink. This last factor is compounded if numerous units need to be tested. If the rated power of the gearbox under test is large, and the output speed is low, the torque at the sink can be inconveniently massive, unless further supplementary gearing is provided. One way of doing this, of course, is to test two gearboxes simultaneously in series, as shown in Fig. 1b. This, however, does nothing to avoid the necessity for a large source and sink.

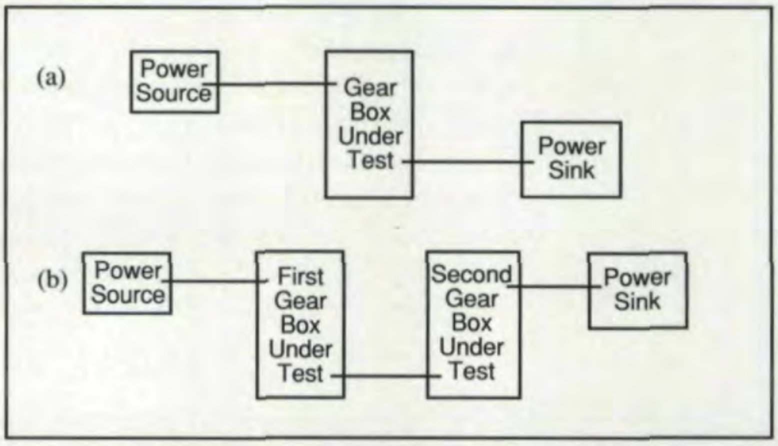

Fig. 1 - The traditional, schematically open arrangement of gearbox test rigs: (a) the simplest arrangement; (b) a tandem arrangememt which tests two gearboxes simultaneously and which avoids a massive torque at a low speed at the power sink.

The above situation can be markedly improved by the use of a closed-loop arrangement of the test drive train. Such an arrangement allows the return to the test system of the output power from each gearbox under test, so that

• no power sink is required;

• the power source only has to provide make-up power equal to the sum of system losses;

• several gearboxes are tested simultaneously.

Obviously, large savings can accrue from the avoidance of the necessity for a power sink and

R.B. Frost
is Senior Lecturer, School of Mechanical & Industrial Engineering, University of New South Wales.

T.R. Cross
is Managing Director, John Welsh Pty. Ltd. in Melbourne, Australia.

from the large reduction in the required capacity of the power source. If numerous units need to be tested, large savings in time and energy are possible.

Disadvantages or limitations on the use of such a test method are that

- non-backdriving gearboxes, such as high-ratio worm drives, cannot be tested in this manner;
- a somewhat larger floor area is required for the test setup;
- the power level in each gearbox in a particular setup will be slightly different due to losses around the drive train loop;
- the method is best suited to situations where at least as many gearboxes need to be tested as are required to make up the drive train loop;
- if the number of units to be tested is not an integral multiple of the number required to make up the loop, some will need to undergo two tests to allow formation of the final loop at the end of the series of tests.

While such multiple simultaneous testing is not uncommon in Europe, it has not been used to any great extent in Australia.

The Drive Train Loop

This consists of several of the units to be tested, coupling shafts, and three other components: a motion input drive, a torque application coupling, and a torque transducer.

Such a loop can be achieved in several convenient ways. If the gearboxes are parallel-shafted, two can be coupled as shown in Fig. 2; if they involve shafts at right angles, e.g., as with bevel input stages, then four can be coupled as shown in Fig. 3. For generality, and because of recent experience with such a setup, the remainder of this article is written as relating to a setup as shown in Fig. 3.

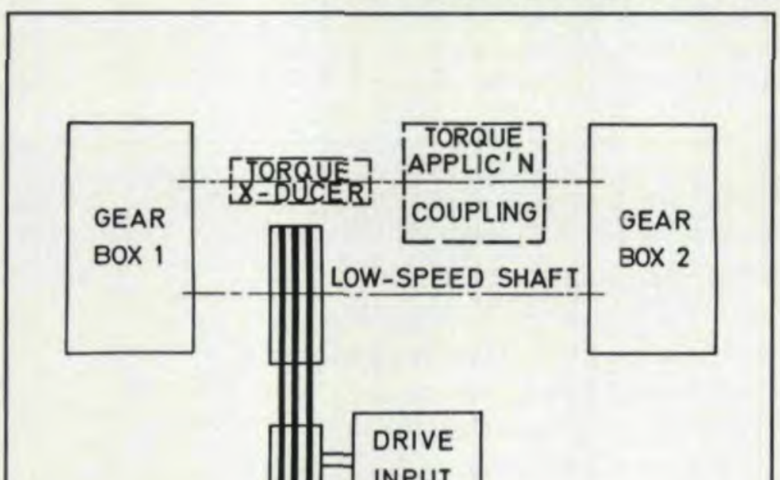

Fig. 2 - A closed-loop arrangement for testing two parallel-shafted gearboxes simultaneously and utilizing circulating power, thereby avoiding the necessity for a power sink.

The three additional components referred to above can be incorporated in any convenient manner or location in the loop. The drive input can be incorporated in a coupling shaft as a belt or chain drive or as a separate gearbox with a double-ended output shaft, or it can be attached directly to a shaft extension on one of the units under test, if one is available. Alternatively, if the motor speed corresponds to the speed of the high-speed shafts, and the motor has a double-ended shaft, it can be incorporated directly in one of the high-speed shaft lines. The torque application coupling is a device which joins a pair of adjacent shaft ends and which applies equal and opposite torques to them while they are both rotating, and in the presence of some degree of relative rotation between them. It may be thought of in practice as a hydraulic motor, supplied and drained by two lines via a rotatable hydraulic coupling, with its shaft connected to one of the two shaft ends referred to above, and its body connected to the other.

The Action of the Test Rig

This is best described by considering the rig at rest while the torque application coupling is loaded by the application of hydraulic pressure. The whole drive train loop is thereby loaded torsionally so that each section of it, being in torsional equilibrium, has equal and opposite torques at its two ends; i.e., an action and a reaction. If the motion input is now started, the whole drive train loop will rotate in the direction in which it is driven. For a particular loop section, the applied torque and the rotation will be in the same direction at one end, where power will be entering that loop section, and in opposite directions at the other end, where power will be leaving that loop section. This applies to each loop section, so that power circulates continuously around the drive train loop in a direction which is determined by the combination of the directions of the drive rotation and the torque application.

The level of circulating power is determined by the combination of the speed of the drive rotation and the level of the torque application.

With the exception of the losses due to the inefficiencies in the gearboxes, power does not leave the drive train loop; therefore, it does not need to be provided by the drive input, except for the losses, regardless of the level of the circulating power.

Power flows and losses are shown in Fig. 3. If P_1 is the input power to the gearbox on the "downstream" side of the drive input (in the power-flow sense) and P_5 is the power delivered to the drive input by the adjacent gearbox on the "upstream" side, then

$$P_5 = \eta^4 P_1 \tag{1}$$

where η is the efficiency of each individual gearbox, so that the total power loss is given by

$$P_{loss} = P_1 - P_5 = P_1(1-\eta^4) = P_{input} \tag{2}$$

The input power fraction, which is the ratio of input power required when using a closed-loop rig to that required when using a rig of the type shown in Fig. 1, where input power must equal test power, is given by

$$f_p = \frac{P_{input}}{P_4} = \frac{1 - \eta^4}{\eta^3} \tag{3}$$

where P_4, which is the lowest input power level to any gearbox in the loop, is taken to be the agreed power level for the test. This agreement could be made differently, of course, in which case, f_p will be defined differently, but this will make little difference to f_p.

The input energy fraction for testing a series of gear boxes in this way, rather than as in Fig. 1, is given by

$$f_E = \frac{f_p}{N} = \frac{1 - \eta^4}{N\eta^3} \tag{4}$$

where N is the number of gearbox units tested simultaneously. In the situation considered here, N = 4.

Similarly, the overload factor on Gearbox 1, in order to achieve P_4 into Gearbox 4, is given by

$$F_O = \frac{P_1}{P_4} = \frac{1}{\eta^3} \tag{5}$$

Fig. 4 shows values of f_p, f_E, and F_O for a range of η which is realistic for a large range of gearboxes. Clearly, large savings in installed capacity, time and energy can be realized, provided the overload on some units can be tolerated for the duration of the test. This overload is, of course, significantly reduced if P_1, P_2, or P_3 can be agreed upon as the power level for the test, rather than P_4.

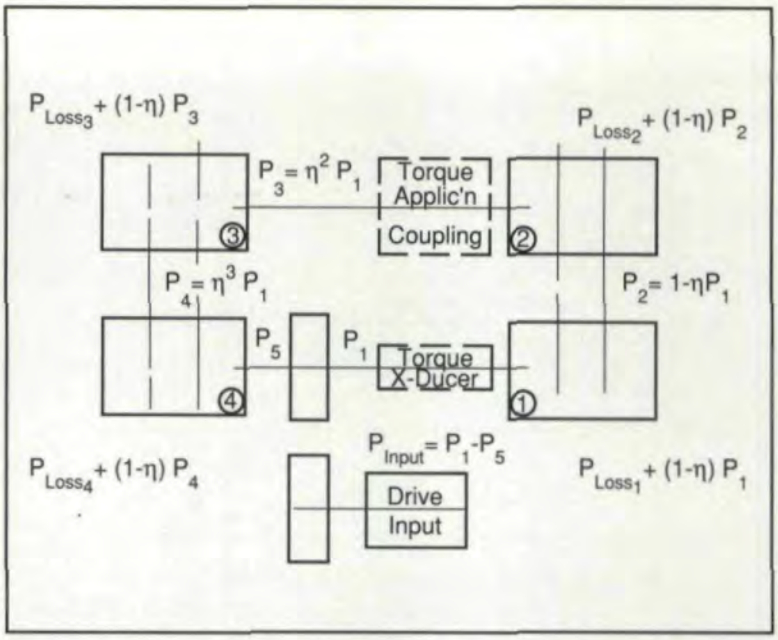

Fig. 3 - Similar to Fig. 2, but to suit four right-angle-drive gearboxes. Power flows and losses are shown.

Consideration needs to be given to the "handing" of gearboxes and the resulting compatibility of rotation around the drive train loop. Regardless of this, however, it is necessarily true that one diagonally opposite pair of boxes will be operating in the schematically forward (power-flow) sense as speed reducers, bearing on the forward flanks of the teeth, while the other diagonally opposite pair are operating in the schematically backward sense as speed increasers, bearing on the rear flanks. For spiral bevel gearing the differences in geometry and force levels between these two cases are significant. Whether this is seen as a problem, as in the necessity to prove every box in every mode, or as an opportunity whereby each of these two cases can be demonstrated simultaneously, depends on the way in which test requirements are written. At most, however, this problem would only involve a reversal of rotation and/or torque in the same setup once testing is under way.

Recent Experience With Closed-Loop Testing

John Welsh Pty. Ltd. of Melbourne recently supplied 21 gearboxes, each rated at 610 kW, 1485 RPM input speed, 17.76:1 reduction ratio with a spiral bevel input stage and a helical

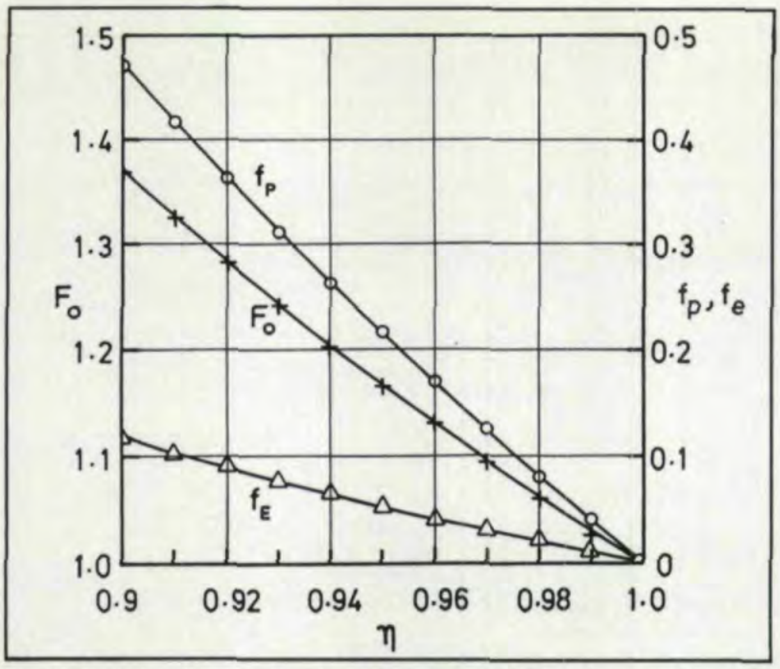

Fig. 4 - Variation of energy fraction f_E, power fraction f_p, and overload factor F_O for a range of efficiency, η, for the setup shown in Fig. 3.

output stage, for coal conveyor drives. These were tested four-at-a-time in an arrangement similar to that shown in Fig. 3, at 50% of full load and then at 100% of full load, in each case until temperature equilibrium was reached and thereafter to allow noise measurements to be taken.

Drive input was via a multiple V-belt drive of the appropriate ratio from an induction motor rated at 220 kW at 960 RPM, onto one of the high-speed shafts connecting the gearboxes. This shaft also contained the torque application coupling, which was a proprietary item, model VM2-4000-100 from Glyco in Germany. It provided two active hydraulic chambers in each direction to facilitate the production of a large torque from a compact design, and thereby provided a total relative rotation of only 100°. The elastic torsional deflection around the drive train loop at 115% of full-load torque, θ_E, was calculated to be approximately 31.5° of relative rotation at the adjacent shaft ends where the torque application coupling was to be located and the backlash in the gearing, θ_{BL}, was estimated to be approximately 5.5° at this location. For loading in both directions, this required a total relative rotation of the torque application coupling of

$$\theta_{TOTAL} = 2\theta_E + \theta_{BL} = 68.5° \qquad (6)$$

which was within its range and which left approximately 31.5° for angular assembly of the flanges of this coupling, which, it was anticipated, would be assembled last in the drive train loop with its mating flanges on the adjacent coupling shaft ends. In the event, however, it was found that the torque transducer had sufficient holes in its flanges to allow convenient final closure and connection of the drive train loop at its flanges, with the torque application coupling initially set so that it had approximately 75% of its stroke available in the required direction of testing.

The Lebow 1641-100K torque transducer was located in the other high-speed shaft.

In each instance of testing, the motor driving the rig was started while the torque application coupling was subject to a small required minimum pressure of approximately 5 bar. This gave rise to the necessity to overcome some degree of initial stiction in the lightly torsionally pre-loaded rig. This was not a real problem, however, and once the rig broke the stiction and started to move, it ran very freely. The hydraulic pressure to the torque application coupling was then progressively increased, thereby increasing the circulating power level, the losses, and the motor load, until the required reading was achieved on the torque meter. This reading, of course, had to be calculated specifically in view of the position in the drive train loop occupied by the torque transducer. The rig was then run as required for the testing procedure. In all cases the rig behaved in a stable manner and the testing proceeded without problems.

Conclusion

Simultaneous testing of multiple gearboxes using a closed-loop test rig arrangement is a very convenient and cost-effective testing method where the conditions favoring such a method, as outlined in the introduction, are satisfied. On such occasions, the authors recommend consideration of this method. ■

Acknowledgement: *This article was originally presented at The Institution of Engineers, Australia, Fifth International Conference on Manufacturing Engineering, July 11-13, 1990, Wollongong, NSW, Australia. Reprinted with permission.*