Friction
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RESEARCH ARTICLE

Experimental rigs for testing components of advanced industrial applications

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Abstract: This paper presents experimental rigs of the Research Centre for the Mechanics of Turbomachinery of the Department of Civil and Industrial Engineering of the University of Pisa. Most of them were designed and constructed to allow investigations of real machine components and to furnish more realistic results than basic tribological test rigs.

Tilting pad journal bearings, as well as gears and complete gearboxes for advanced industrial applications, can be tested using the rigs described in the paper. A novel test rig with a power rating of approximately 1 MW allows investigations of the static and dynamic characteristics of high-performance tilting pad journal bearings for turbomachinery. A twin disc machine and closed loop gear test rig are used to investigate the different kinds of wear mechanisms occurring in gears. Functional and durability tests on planetary gearboxes for new turbo-fan engines could be performed using another novel large test rig. A circulating power configuration was adopted for most of the rigs so that only the power needed to cover the friction losses has to be supplied, while the circulating power can be more than 20 times higher. All the test rigs include very complex load applications and lubrication plants, as well as dedicated control and data acquisition systems.

The rigs and related plants were designed and constructed through strong and fruitful collaborations between the university and some large and small–medium companies. Despite some limitations in the publication of the results as a result of the industrial sensitivity of the data, the synergy among these different actors was stimulating and fundamental for the realization of new advanced industrial applications.

Keywords: experimental rigs; tilting pad journal bearings; twin disc machine; gears; gearboxes

1 Introduction

Increasing the power density of new machines and decreasing the energy losses and wear to decrease the environmental impact are very important objectives today. The design of new machine components that are able to work at higher speeds and larger loads than those in the past is not possible without deep tribological investigations to obtain significant reductions in friction and wear.

Tribological studies can be performed both theoretically and experimentally. However, despite the availability of advanced computer programs today, experimental investigations are essential for new developments and software validation. Experimental tests are often performed on basic test rigs such as pin-on-disc or twin disc machines, which allow deeper investigations of the contacts. However, this only allows a simulation of the real pairs, and some aspects of their actual behavior can be missed. For a complete manifestation of the involved phenomena, tests on full-scale real components should be performed, particularly for the development of the components of new machines.

Tests using basic test rigs and real machines are useful and present both advantages and drawbacks.

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With basic test rigs, it is generally easier to measure several quantities such as the temperature, pressure, force, and lubricant film thickness, but it is not easy to simulate the real working conditions. Basic tests can be useful when comparative studies for the development of a new machine must be performed; for instance, tests on different materials can be carried out to select the best material for a specific application. There are several kinds of equipment for basic experimental tests that can simulate the different geometries, loads, and kinematic conditions that can be used for testing different materials. It is of fundamental importance that the simulated contact reproduces the conditions of the real contact as realistically as possible because otherwise, the results could be misleading. For instance, the behavior of the lubricant can change depending on the contact type; a lubricant that furnishes the best results in a rolling bearing test rig might not be the best one for gears [1]. However, there are also studies showing that similar trends can be found for the investigated quantities with different test rigs [2, 3]. In any case, some aspects of a real machine are very difficult to simulate, such as the vibration and thermal effects. On the other hand, tests on full-scale components can provide more realistic results, but it is more difficult to carry out detailed measurements. The positioning of sensors at all points of interest is sometimes impossible, particularly if tests are performed with a real machine. In addition, full-scale tests can be difficult to perform because of the dimensions of the machine and the power involved; they can also be very expensive, particularly when destructive tests are necessary. Recent examples of tests on full-scale components are shown in Ref. [4].

Test rigs for testing industrial components, particularly if large dimensions and power ratings are involved, are not easy to develop and require large quantities of time and money for their realization. Therefore, a relatively small number of test rigs have been developed over the years. Once a research group has realized a test rig, it is normally used for many years for testing different kinds of bearings or gears with minor modifications to the rig.

At the University of Pisa, tests with both basic test rigs and real components have been conducted for many years [5]. In particular, investigations on real components such as bearings and gears are performed. Two main factors contribute to the successful realization of very complex tests on real components: the recent progress in sensors and instrumentation and the cooperation between academia and industry. The essential characteristics and potentialities of the test rigs and related plants of the Research Centre for the Mechanics of Turbomachinery of the Department of Civil and Industrial Engineering of the University of Pisa are presented. The experimental systems realized are the product of fruitful collaborations between the university and large and small–medium companies.

Some typical experimental results are also shown in this paper.

2 Research Centre for Mechanics of Turbomachinery

The Research Centre for the Mechanics of Turbomachinery was recently established at the Department of Civil and Industrial Engineering of the University of Pisa. The Centre includes the pre-existing Research Centre on Advanced Technology Mechanical Transmissions and the new Laboratories for Rotordynamics and Bearings and for Mechanical Transmissions. The Centre was established through the strong cooperation between the University of Pisa, two large companies (GE Oil & Gas—Nuovo Pignone of Florence and GE Avio of Turin), and a small–medium-sized enterprise (AM Testing). This collaboration is fundamental for the design and realization of experimental rigs, as well as for managing the tests. In particular, the small–medium-sized company is an essential link between the university and the large companies because of its ability to provide quick responses when problems arise.

These activities began at the beginning of the 21st century with the realization of two test rigs for investigations on journal bearings and gears for aeronautical applications. The success of the enterprise and the recent availability of regional and European funding pushed the realization of new and more powerful test rigs with very complex plants. Because of their large dimensions (the new rigs are 4–6 m in length) and the power involved (approximately 1 MW), a new building was built in 2015 to host the experimental rigs and related plants (Fig. 1).

Fig. 1 Image of new laboratory with test rigs and lubrication plants.

The building and its plants were designed and realized with the potential to also host other rigs for testing different high-power components in the future.

The test rigs presently running in the Research Centre are described in the following paragraphs. Each experimental apparatus can produce several results of industrial interest, most of which are covered by non-disclosure agreements. However, when the results are not considered particularly sensitive by the companies, numerical data can be published after a specific approval process. In some cases, diagrams showing the trends of the most significant quantities can be presented without displaying numbers. The experimental activities on some of the rigs are just beginning.

3 Investigations on journal bearings

3.1 General aspects

The main aspects of journal bearings investigated are the influences of the lubricant flow rate, radial clearance, and materials at different speeds and loads on the bearing performances. Most studies today address the friction [6, 7] and wear [8]. Measurements of the lubricant film pressure and temperature distributions are not easy to perform, but have occasionally been conducted [9]. Investigations on water-lubricated bearings [10] and bearings with modified (textured or scratched) surfaces [11, 12] are also being conducted. In addition to these aspects, instability is a very important problem in this kind of bearing. According to the classical approach for studying instability, the hydrodynamic forces exerted by the fluid can be approximated around the equilibrium configuration by linear relationships using matrices of the stiffness and damping coefficients [13]. The rotordynamic coefficients can be determined using numerical programs. However, the simulation programs often employ empirical constants based on experimental results obtained with specific test rigs. In addition, the extension of the results to different kinds of bearings or even to bearings with the same shape but different dimensions is questionable, and experimental investigations are necessary for different bearings.

Journal bearings with adjustable/controllable geometries can be used to reduce the vibration problems [14, 15]. Controlled lubrication is also used in tilting pad journal bearings [16–18].

Tilting pad journal bearings are widely used in turbomachinery because of their stability at high rotational speeds. Because of their influence on the dynamic behavior of the rotor, their dynamic characterization is essential in the design phase of a turbomachine. The characterization is normally conducted experimentally, particularly for new models of bearings. The experimental results can also be used for the validation of the existing numerical programs and calibration of the previously mentioned constants used in the programs. The dynamic coefficients are usually determined by applying dynamic loads to the bearing and measuring the relative displacement of the rotor/stator. The identification is mainly made in the frequency domain [19].

Two different test rig configurations for tilting pad journal bearings are used by researchers: one with the test bearing floating at the mid-span of a rotor supported by rolling bearings and one with a fixed bearing housing and moving shaft. The most common configuration is the one with the floating test bearing. The bearing is loaded statically and dynamically excited by two independent actuators in two orthogonal directions. Different design solutions can be found in laboratories around the world, including the Turbomachinery Lab of Texas A & M University, USA [20];

the Research and Development Centre in Hanjung, Korea [21]; the Department of Mechanical and Aerospace Engineering of the University of Virginia, USA [22, 23]; the NRC in Ottawa and Kingsbury, Canada; the Power and Industrial System R & D Center of Toshiba Corporation, Japan [24]; Doosan Heavy Industries & Construction and Pusan National University, Korea [25]; the GE Global Research Center, USA [26]; and the Department of Mechanical Engineering and Automation of Shanghai University, China [27]. The main differences concern the load application systems. The static load is applied through one or two helical springs in the American laboratories, placed in the same direction as one of the dynamic actuators in Ref. [20] and at 45° with respect to the dynamic actuators in Ref. [23]. Air bellows are used in the Asiatic laboratories instead. Dynamic actuators are placed in the upper part in the American laboratories and in the bottom part in the Asiatic ones.

The solutions adopted by the various laboratories have quite different performances. The rotor diameter ranges from a minimum of 70 mm at the Universities of Virginia and Shanghai to a maximum of 580 mm at Toshiba. The maximum specific load is placed on the bearing at Texas A & M University (3,100 kPa), while the maximum applicable load can be found at NRC Ottawa (25,000 N). The maximum rotor speed is achieved by Kingsbury (128 m/s), the maximum driver power by Toshiba (2,400 kW), and the maximum lubricant flow rate by Toshiba (774 L/min).

An example of a configuration with a fixed bearing housing and moving shaft is found in Ref. [28].

3.2 Tilting pad journal bearing test rig

A novel experimental apparatus for testing tilting pad journal bearings was recently designed and built through collaboration between the Department of Civil and Industrial Engineering of Pisa, GE Oil & Gas, and AM Testing [29, 30]. This activity was co-financed by the Region of Tuscany as part of the Advanced Technologies for ENergy Efficiency (ATENE) project.

After an extensive analysis of the advantages and drawbacks of the different configurations, the one with the floating bearing housing, as shown in Fig. 2,

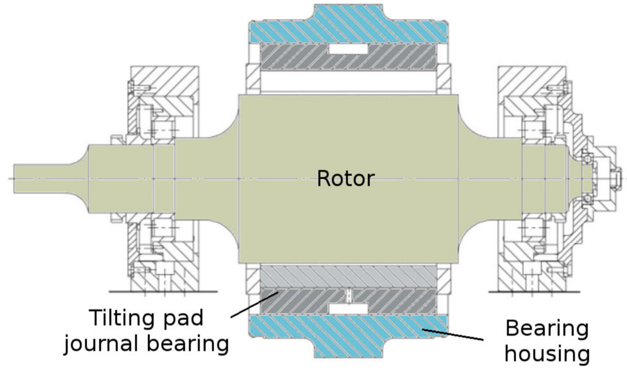

Fig. 2 Schematic of test zone with tilting pad journal bearing.

was selected, mainly because of the large size of the bearings to be tested.

The realized test rig for the static and dynamic characterization of high-performance tilting pad journal bearings can test bearings with diameters of 150–300 mm, an axial length to diameter ratio $L/D = 0.4$–1.0, peripheral speeds up to 150 m/s, and a maximum specific load of 3 MPa.

The static load is applied in the vertical direction by means of a hydraulic actuator able to apply loads of up to 300 kN (Fig. 3). The dynamic loads are applied by two other hydraulic actuators that can produce alternating forces of up to 30 kN with frequencies of up to 350 Hz. The dynamic actuators are identical and placed in mutually orthogonal positions, at 45° with respect to the vertical direction. They can

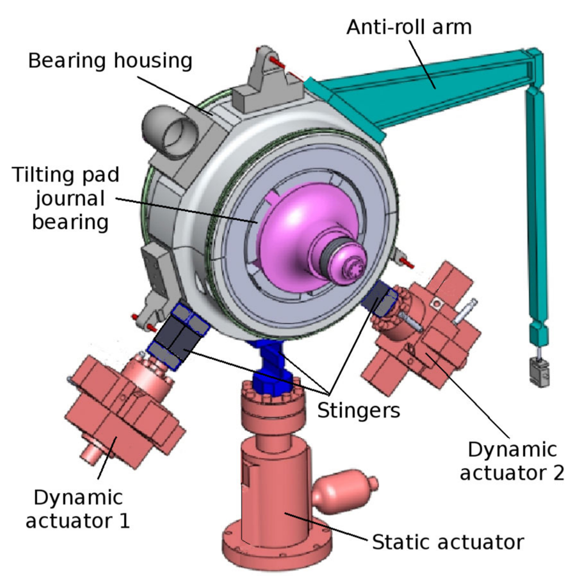

Fig. 3 3D schematic of test zone of tilting pad journal bearing rig showing load application actuators.

work independently or simultaneously in phase, to obtain a vertical force, or in anti-phase, to provide a horizontal force. Each actuator has four high dynamic servo valves in parallel to guarantee good performances up to the highest frequencies. A picture of one of the dynamic actuators is shown in Fig. 4. Stingers are interposed between the actuators and bearing housing to limit the tangential forces transmitted to the housing. Because the transverse stiffness is much lower than the longitudinal one but is not negligible, all the components of the transmitted forces are measured. The actuators are capable of developing an alternating displacement of approximately 0.1 mm when exerting the maximum force at the maximum frequency. The deformation at the base of the actuators is negligible thanks to the very stiff structure realized.

An "anti-roll arm", as shown in Fig. 3, prevents the rotation of the housing; the reaction exerted on the stator is measured by a load cell.

The rotor is driven by a 630 kW three-phase asynchronous motor, controlled by an inverter, with a maximum speed of 4,000 rpm and a nominal torque of 3,000 N·m. The motor is connected to a single-stage multiplier with a gear ratio of 6:1, with the output shaft connected to the rotor of the bearing by a torque meter with a full scale of 1 kN·m. The maximum rotational speed is 24,000 rpm.

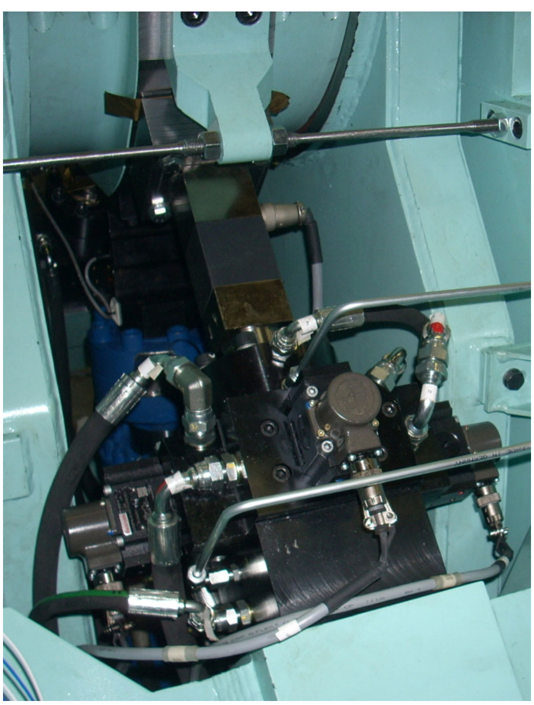

Fig. 4 Image of one of two dynamic actuators of tilting pad journal bearing test rig.

Figure 5 shows a picture of the test rig, with the test zone containing the bearing on the right, motor on the left, and multiplier gearbox in the middle.

Three independent oil plants are used for lubricating the rig: one for the tilting pad journal bearing to be tested, one for the load application system, and one for the multiplication gearbox. The main lubrication system can supply oil to the tested bearing by maintaining a constant flow rate and temperature in ranges of 125–1,100 L/min and 30–120 °C, respectively. A storage tank of approximately 4,000 L is positioned below the test rig. The oil is conveyed into the bearing by means of a tube connected to the upper part of the housing, as shown in Fig. 5. A 200 L pressurized spare tank is present for lubricating the bearing in the case of a fault in the main system. A high-pressure system supplies to the hydraulic actuators. This system includes two circuits, one with a pump with a maximum pressure of 315 bar for the application of the load to the housing and one with a pressure of 200 bar for feeding the hydrostatic bearings supporting the stems of the two dynamic actuators. An additional lubrication system is used to supply the multiplier gearbox.

A suitable oil cooling system is employed to remove the heat, which is primarily dissipated in the journal bearing.

Several sensors are installed in the test section with the journal bearing to measure the quantities needed for the identification of the static and dynamic characteristics of the tested bearing. In particular,

Fig. 5 Image of tilting pad journal bearing test rig.

four high-resolution proximity sensors are used to measure the relative displacements between the bearing housing and rotor. They are located at two cross-sections of the housing in the lateral parts. Two proximity sensors are placed in mutually orthogonal positions in each section, at 45° with respect to the vertical direction. To verify the possible deformations of the bearing, additional sensors with lower precision are also placed in each section in diametrically opposite positions with respect to the high-resolution ones. Four uni-axial acceleration sensors are placed on the housing to measure the acceleration in two planes and two orthogonal directions for indirect measurements of the inertial forces. The dynamic forces applied to the bearing housing are measured through strain gauges installed on the stingers connected to the dynamic actuators, which also measure the tangential forces. A load cell is used to measure the static load and another is located on the anti-roll arm. The rotational speed of the shaft and the applied torque are also measured. Some thermocouples are mounted on the bearing's pads. Additional sensors are located in the auxiliary systems and plants to measure the temperature, acceleration, lubricant pressure, and flow rate at specific points.

Two control and data acquisition systems are used for managing the tests by setting and controlling the working conditions. They also acquire the data of several sensors. One system acquires high-frequency signals at 100 kHz and the other low-frequency data at 1 Hz. Approximately 30 channels are acquired at high frequency and 60 at low frequency. Several computers and monitors are used, which are located in a dedicated control room of the laboratory (Fig. 6).

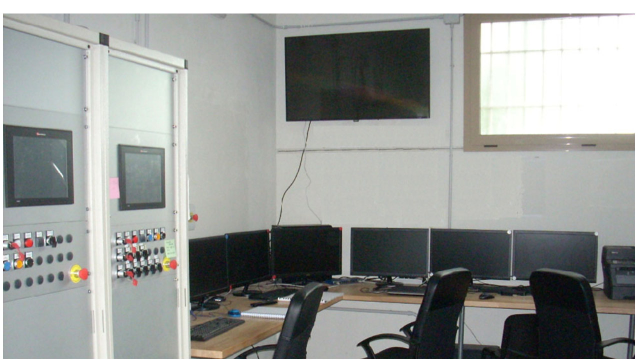

Fig. 6 Image of control room of laboratories for Rotordynamics and Bearings and for Mechanical Transmissions.

When all the systems are included, the maximum power needed to operate the test rig is approximately 1 MW.

The main parameters of the test rig are summarized in Table 1.

The experimental activity has recently started. Typical trends for some of the fundamental quantities recorded during a test carried out for the commissioning of the rig are shown in Fig. 7. These results are from a step variable speed test with a tilting pad journal bearing with four pads in the load-between-pads configuration. The static load is kept constant for almost the entire duration of the test. The torque and power increase with speed (Fig. 7(a)), along with the temperatures measured by the thermocouples mounted in the outlet zones of the pads (Fig. 7(b)). An increase in the load close to the end of the test causes an increase in the temperatures of the most-loaded pads (1 and 2) and a small decrease for the less-loaded ones (3 and 4).

Excitations at different frequencies and displacement amplitudes have been applied under working conditions with different rotational speeds, static loads, oil flow rates, and temperatures. The standard procedure for recording the data necessary for identification began once the desired steady state working conditions were reached. After that, sinusoidal forces were applied in the frequency range of interest for a short time.

Table 1 Main parameters of tilting pad journal bearing test rig.

Parameter	Value
Bearing diameter	150–300 mm
Bearing length to diameter ratio	0.4–1
Shaft rotational speed	0–24,000 rpm
Bearing peripheral speed	0–150 m/s
Static load	0–270 kN
Dynamic load	0–40 kN
Frequency of the dynamic load	0–350 Hz
Bearing oil flow rate	125–1,100 L/min
Bearing oil inlet temperature	30–120 °C
Electric motor power	630 kW
Plant maximum total power	1 MW

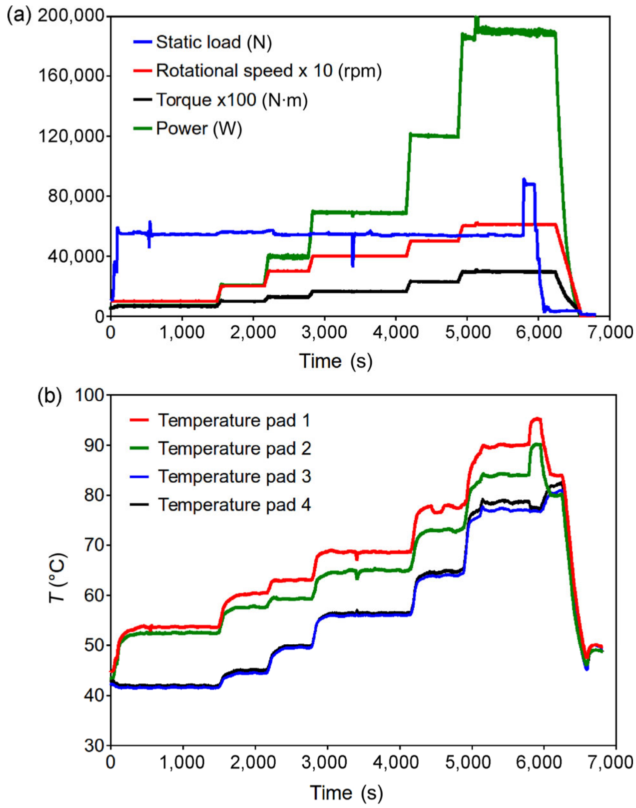

Fig. 7 Sample time trends for static load, rotational speed, torque, and power (a) and pad temperatures (b). Test with a four pad tilting pad journal bearing.

The methodology used to identify the dynamic coefficients was the one used in Refs. [26, 31], and also described in Ref. [30]. Sample results obtained using the tilting pad journal bearing with four pads at five different excitation frequencies are shown in Fig. 8. Here, Kxx and Kyy are the direct stiffness coefficients, and Kxy and Kyx are the cross ones.

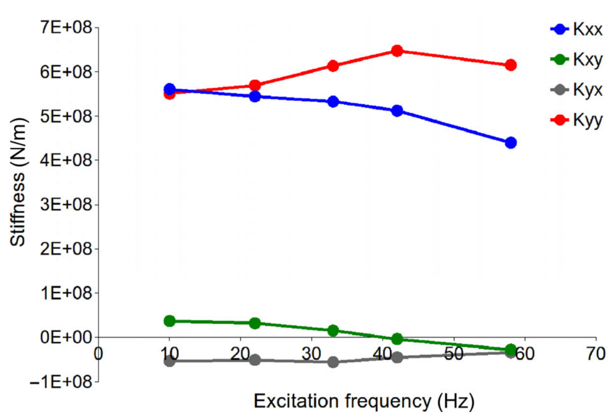

Fig. 8 Sample direct and cross stiffness coefficients as function of excitation frequency. Test with a four pad tilting pad journal bearing.

4 Investigations on gears

4.1 General aspects

Many studies on gears have been conducted to investigate their possible failures, particularly from scuffing and pitting. As is well known, the different wear mechanisms involved make it necessary to use different kinds of gears to reach the two failure conditions independently. Typically, larger teeth with a higher modulus are necessary to produce scuffing failures because this kind of adhesive wear is usually obtained close to the bottom and top of the teeth, where the maximum sliding occurs. In contrast, smaller teeth need to be used to obtain the typical fatigue wear such as pitting to avoid the scuffing conditions previously reached. Some experimental rigs have been created by researchers that are suitable for these two different investigations, including the FZG test rig, which is a standard today [32, 33] and is used by many research groups for investigations on the friction and wear of spur gears [1, 34–38]. This rig adopts today's preferred configuration for circulating power: only the power for balancing the friction losses is necessary, while the circulating power can be more than 20 times higher. Several kinds of preloading systems can be used to reach this goal. Different test rigs are sometimes designed and built for specific studies, such as the one used to quantify churning losses in [39].

Gear test rigs can also be used to measure gear losses [40], but normally only average friction evaluations can be made based on power measurements [41]. When more detailed friction investigations need to be made, disc machines are often used [42, 43]. Examples include studying the influence of specific aspects such as the materials and roughness, and investigating pitting [44]. Disc machines are also preferable for economic reasons because of the higher costs associated to the realization of gears. However, there is a potential drawback in using disc machines when it is necessary to investigate mixed lubrication conditions, which are often the working conditions of greatest interest. Under these conditions, the orientation of the surface roughness becomes very important, but the technological realization of discs with the same surface topography as the real teeth is not simple. However, discs with machining marks almost parallel to their axes have been realized and tested in some twin disc machines [42, 45]. Other aspects of the real contacts between teeth that are difficult to simulate are the continuous variations of the radius of curvature and the slide-to-roll ratio, which is the ratio of the sliding and rolling velocities of the two teeth in the contact zone during meshing. Disc machines can use discs with different diameters and different peripheral speeds to obtain the desired slide-to-roll ratios. However, each slide-to-roll ratio is usually tested under steady-state conditions. Transient conditions, including variations in the radius of curvature, load, and speed, can be obtained with different kinds of experimental rigs such as the one described in Ref. [46]. This rig simulates the contact between two teeth using a cam-follower system, and it is able to perform simultaneous measurements of the film thickness and all the components of the contact force using optical interferometry and a purposely developed dynamometer with six axes.

Both a twin disc machine and two gear test rigs are present in the Research Centre for the Mechanics of Turbomachinery.

4.2 Twin disc machine

A twin disc test machine was designed and realized by AM Testing in the framework of a wider research collaboration with the University of Pisa and GE Avio, as reported in Ref. [45]. A schematic drawing of the machine is shown in Fig. 9. A back-to-back configuration is employed with a test box, containing the discs to be tested, connected through two transmission shafts to a slave box containing a set of gears. One of the two transmission shafts is equipped with a torque meter to measure the transmitted torque and evaluate the power losses. A 35 kW electric motor is employed with a maximum torque and rotational speed of 54 N·m and 6,250 rpm, respectively.

A loading device composed of a hydraulic cylinder and mechanical leverage unit is used to push the two discs against each other to obtain the desired contact force. This system can be seen on the right in Fig. 10, which shows a picture of the test rig. The electric motor is on the left, and the slave gear box in the middle.

Two independent lubrication systems are used to lubricate the discs being tested and the auxiliaries (rolling bearings and gears). The lubricant used for the tested discs is heated or cooled by a thermoregulator capable of maintaining the oil at a constant temperature in the range of 20–180 °C. The oil flow rate can be regulated up to 20 L/min. In particular, the test box was designed to allow the separate lubrication of the test discs and shaft bearings. The shaft housing also includes a water cooling circuit for the bearings. Scavenge pumps are used to transport the oil back to the tanks. The total power of the rig is approximately 40 kW.

The test rig can simulate gear teeth mating conditions in terms of the entraining and sliding velocities, slide-to-roll ratio, contact pressure, and lubrication conditions. In particular, the lubrication systems utilized in the aeronautical and automotive industries can be replicated. The oil can be sprayed in the inlet and outlet directions and also on the front

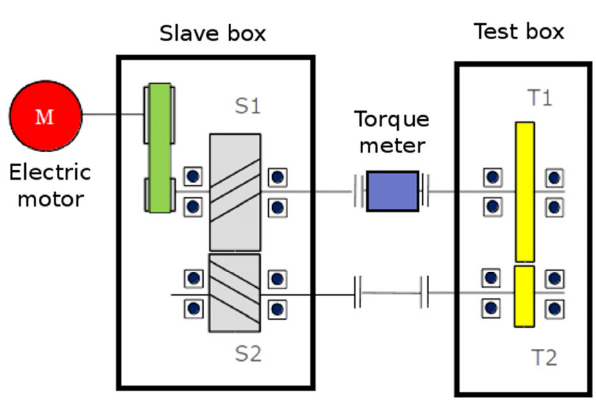

Fig. 9 Schematic of twin disc machine.

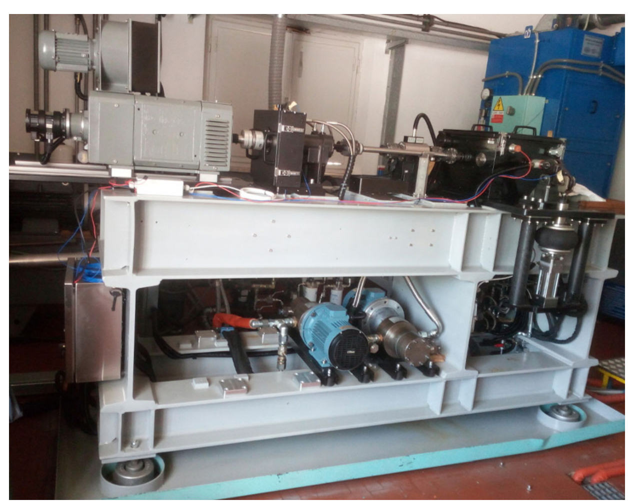

Fig. 10 Image of twin disc machine.

face of the two discs; the system can also run in an oil bath. The lubricant flow rate, temperature, and pressure are varied and monitored during tests. A picture of the test box with the discs and lubrication pipes is shown in Fig. 11.

The sub-surface and bulk disc temperatures are measured by two thermocouples fitted on one of the rotating shafts. On-board conditioning and amplifying electronics allow the transmission of the temperature signals by a telemetry system. An eddy current probe is installed to detect changes in the surface morphology so that surface and sub-surface damage can be detected during running. Several other sensors and actuators are installed in the rig to measure and control the main test parameters. A CompactRIO digital acquisition board with specific software purposely developed in the National Instruments Labview® environment is used to regulate both the load and speed and continuously monitor and acquire the signals from several transducers. A scanning rate of approximately 10 Hz is generally used.

The rig also allows the easy positioning of a portable roughness tester, an image acquisition unit, and other tools that can be used for accurate surface analyses.

Several types of tests can be carried out using different discs in the test box and gears in the slave box, including fatigue tests (pitting and micro-pitting) and scuffing tests. Discs of several diameters and diverse surface shapes, curvatures (with and without crowning), surface roughness values, and grinding mark directions have been designed and manufactured to replicate actual gear surface morphologies.

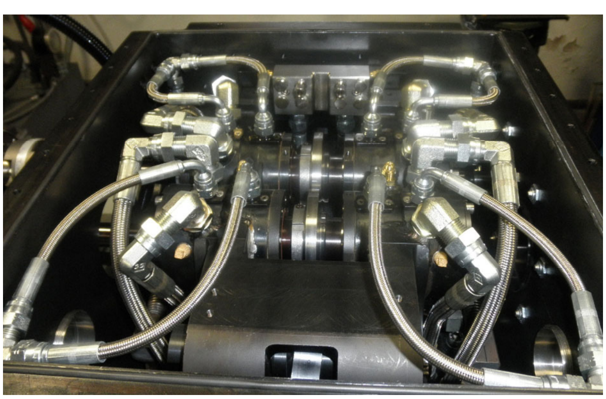

Fig. 11 Image of test box of twin disc machine.

The main parameters of the test rig are summarized in Table 2.

Some activities have been started to optimize the test rig functions and performance by spanning the test rig working range and using lubricants and discs of different sizes and surface finishes. One of the main problems to be solved has been the realization of discs with grinding lines parallel to the direction of the disc axes, similar to the machining marks of gear teeth. A purposely designed apparatus has been realized for manufacturing discs using a standard grinding machine.

Some typical results of the experimental tests carried out with this rig are curves of the friction coefficient as a function of the speed or load (both quantities can be varied continuously). Systematic tests with the twin disc machine have not yet started because of activities related to the other test rigs of the Research Centre. However, just to give an idea of the possible results, an example of the friction coefficient trend as a function of the rolling speed (the semi-sum of the peripheral speeds of the two contacting bodies) obtained with a pin-on-disc machine is shown in Fig. 12. Each test was carried out at a constant load and oil temperature. The three curves refer to three different values of the slide-to-roll ratio S. The friction coefficient increases with the slide-to-roll ratio.

The results clearly indicate a mixed lubrication regime at low speed, with a decrease in friction by increasing the speed. A further increase in the speed produces a small increase in the friction coefficient, indicating the arising of the full fluid lubrication

Table 2 Main parameters of twin disc machine.

Parameter	Value
Radial force	0–35 kN
Contact pressure	0–3.5 GPa
Entraining velocity	0–60 m/s
Slide to roll ratio	0–1
Sliding velocity	0–20 m/s
Rotational speed	0–12000 rpm
Inlet oil flow rate	0–20 L/min
Inlet oil temperature	25–150 °C
Electric motor power	35 kW

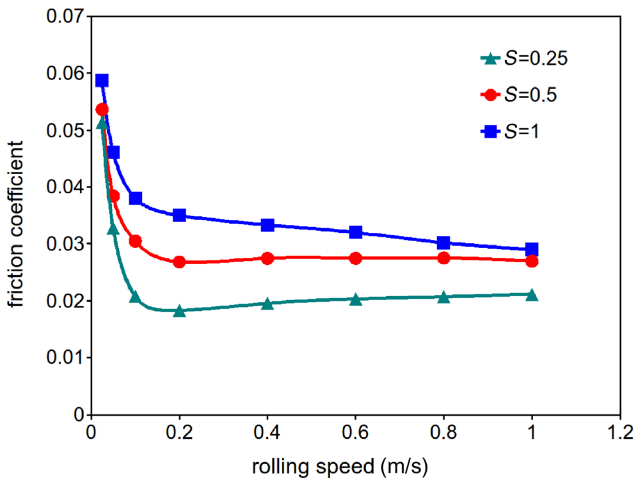

Fig. 12 Trends of friction coefficient as function of rolling speed for three values of slide-to-roll ratio *S*.

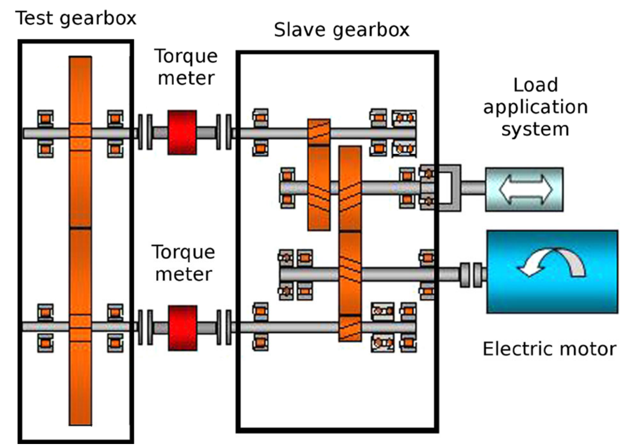

Fig. 13 Schematic of closed loop test rig.

Fig. 14 Image of closed loop test rig.

regime. The always descending trend of the friction coefficient for the highest value of the slide-to-roll ratio is due to greater thermal effects related to the higher sliding.

Other tests can be performed to produce damages on the surfaces of the discs and then analyze the defects, similar to what is done for the closed loop test rig described in the next section.

4.3 Closed loop gear test rig

An experimental apparatus for testing gears has been installed at the University of Pisa in the framework of cooperation with Avio and AM Testing. With this apparatus, tests for investigations on different damage modes such as pitting, micro-pitting, and scuffing can be done directly on gears. The test rig is a closed loop one with two gearboxes, one with the gears to be tested and a slave one, as shown in Fig. 13. Helical gears are used in the slave gearbox, which is used for both speed multiplication and loading. An electromechanical servo-actuator applies the load by varying the axial load acting on the helical gear pair. The two gearboxes are connected by two torque meters mounted on the high-speed shafts. A 50 kW electric motor is used; thanks to the closed loop configuration, the circulating power is approximately 1 MW. The maximum speed of the tested gears is 18,000 rpm, and the maximum torque is 500 N·m.

A picture of the test rig is shown in Fig. 14. The test gearbox is on the left, the motor is on the right, and the slave gearbox is in the middle.

The lubrication plants used for the twin disc machine or dedicated circuits can be used to supply the oil to the test gears and auxiliary systems.

The oil inlet and outlet temperatures are measured by thermocouples. The test torque and rotational speed are also monitored. High-frequency accelerometers are installed to monitor the vibrations and control the damage to the tested gears. Complex diagnostic techniques are used to detect failures.

The digital acquisition system used for the twin disc test rig employs a purposely developed version of the program.

The main parameters of the test rig are listed in Table 3.

Extensive experimental test campaigns have been performed for the characterization of teeth damage

Table 3 Main parameters of closed loop gear test rig.

Parameter	Value
Maximum gear rotational speed	18,000 rpm
Maximum gear torque	500 Nm
Maximum oil temperature	180 ℃
Maximum peripheral speed	135 m/s
Test gear transmission ratio	1:1
Electric motor power	50 kW
Circulating power	1 MW

at high levels of velocity and load testing for two identical gears (1:1 gear ratio) [47–49]. Spur gears with a higher number of teeth (80) have been used for surface fatigue (typically pitting and micro-pitting) tests, and those with a lower number of teeth (28) have been used in scuffing tests. The teeth damage effects of the different load, velocity, teeth geometry, material, and surface finishing conditions have been investigated.

Pitting tests are performed under constant working conditions for millions of revolution cycles. Because it is not easy to detect damage, the tests are usually stopped after a certain number of cycles to analyze the surface condition. Visual inspection is often used. After this, test surface analyses are performed with a scanning electron microscope and profilometer. An example of a tooth surface with micro-pitting damage is shown in Fig. 15.

The onset of damage is usually detected by an increase in the out-of-mesh temperature and a change in the root mean square (RMS) value of the vibration in scuffing tests, as shown in Fig. 16. The increase in the RMS value detected in this test by three accelerometers is connected to an increase in the difference between the temperature of the lubricant at the outlet of the teeth contact, T_{out}, and that at the contact inlet, T_{in}. Both temperatures are measured by conveniently located thermocouples.

Oil off conditions have also been investigated by performing some tests without lubricant [50]. An example of the surface appearance of a scuffed tooth after running for some seconds without the lubricant is shown in Fig. 17.

The tooth bulk temperature was measured using a thermocouple buried in the tooth. An on-board electronic amplifier was purposely developed for conditioning the temperature signal to be transmitted to the acquisition board through a slip ring [51].

A modification was also made to the apparatus for testing bevel gears at higher speeds than spur gears (Fig. 18). A multiplication gearbox was introduced to obtain the desired rotational speed of the tested gears,

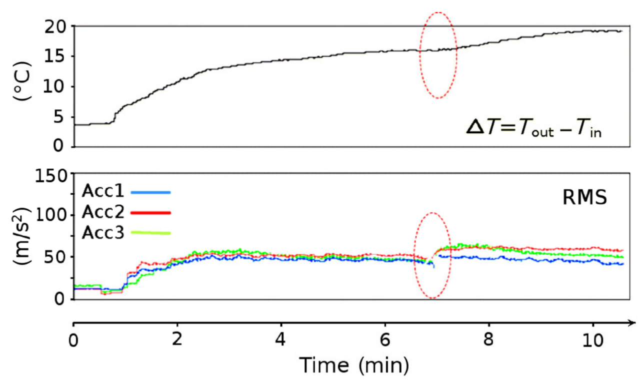

Fig. 16 Arising of scuffing failure: (a) increase in out-of-mesh temperature and (b) change in acceleration RMS value.

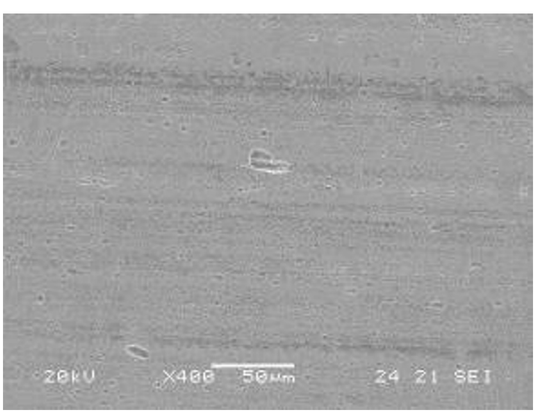

Fig. 15 Scanning electron microscope image of micro-pitted tooth.

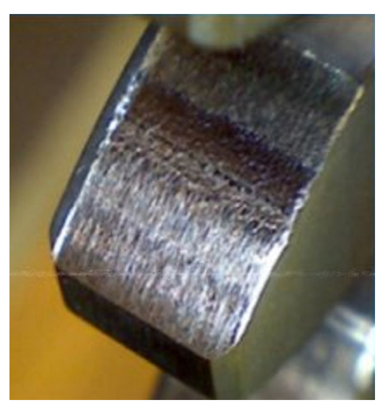

Fig. 17 Sample scuffed tooth.

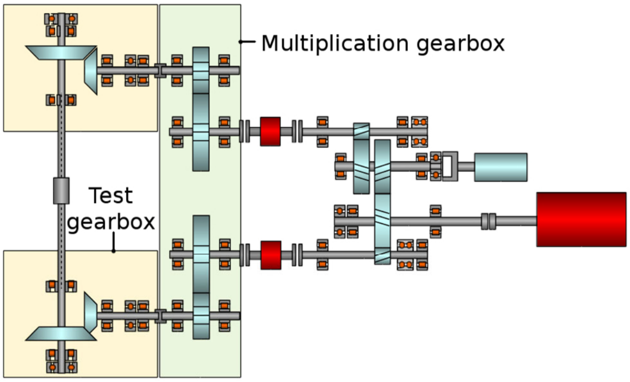

Fig. 18 Schematic of closed loop test rig in configuration for testing bevel gears.

Fig. 19 Image of planetary gear test rig.

and the back-to-back configuration was again adopted to obtain a circulating power of approximately 1 MW.

4.4 Planetary gear test rig

A novel test rig for investigating planetary gearboxes was designed and realized in the framework of the European project GeTFuTuRe funded by the Clean Sky platform [52]. A consortium among the University of Pisa, AM Testing, and Catarsi Ing. Piero & C. in cooperation with GE Avio Aero has been established for this purpose. New power transmission levels can be tested using this rig, which will be used in new turbo-fan engines with turbines rotating at higher speeds and fans at lower speeds than those currently in practice to reduce their environmental impact. Full-scale prototype gearboxes can be installed and tested under the same lubrication, speed, and load conditions as the real engines. An image of the test rig is shown in Fig. 19. The test section is on the left, the motor is on the right, and a multiplication gearbox is located in the middle.

Because of the extremely high power involved, a back-to-back configuration was selected. Tests with circulating power values of tens of megawatts can be performed using a 630 kW electric motor, which is identical to the one used for the tilting pad journal bearing rig.

The test section contains three main parts: the tested gearbox, slave box, and load application system. The two gearboxes are similar and connected through both the high- and low-speed lines. The torque necessary for loading the system is generated by the mechanical rotation of the carter connected to the planet carrier of the tested gearbox. This is obtained as a result of the kinematic constraint of the back-to-back connection with the slave gearbox. Hydraulic jacks connected to a leverage system, one of which is visible on the right in Fig. 20, produce the rotation.

The multiplication gearbox located between the motor and test section has two outgoing shafts to obtain two different multiplication ratios (3.2 and 4.8). In this way, different tests can be performed by maintaining the motor's rotational speed inside an optimal power range with a rotational speed between 2,000 and 3,000 rpm.

Two independent circuits are used to lubricate the two gearboxes, as schematically shown in Fig. 20. Flow rates up to 300 L/min are available for each gearbox at

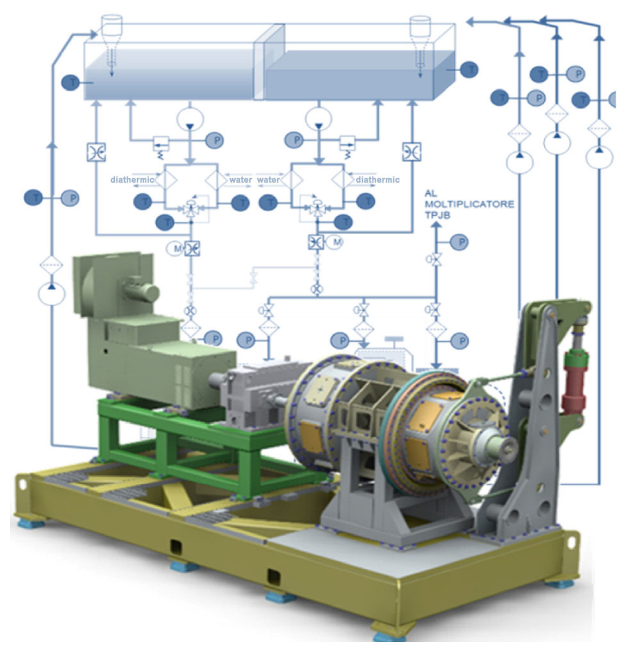

Fig. 20 Schematic of lubrication system of planetary gear test rig.

a maximum pressure of 16 bar. Two pressurized spare tanks are also present for lubricating the gearboxes in the case of a fault in the main systems. The same lubrication systems used for the tilting pad journal bearing test rig are used for the multiplication gearbox and some other auxiliaries.

Many sensors are present in the test rig and its auxiliaries: thermocouples for measuring the temperatures of the bearings supporting the rotating shafts, extensometers at several points of the tested gearbox, proximity sensors, accelerometers, microphones located close to the rig's framework, load cells for measuring the force produced by the jacks, a torque meter for measuring the torque acting on the tested gearbox, speed sensors for measuring the rotational speeds of the shafts, oil debris monitoring sensors (ODM), resistance thermometers, pressure transducers, and flow meters located at several points of the lubrication systems.

A data acquisition system suitable for managing the tests, monitoring the working conditions, and recording the data is used. Signals are acquired by two systems, one for recording at high frequency (100 kHz) and one for recording at low frequency (1 Hz). Approximately 60 channels are acquired at high frequency and 200 at low frequency. Approximately 50 MB of data are recorded each second.

Functional and durability tests can be carried out with the rig to investigate the main characteristics of the gearbox, such as its efficiency and reliability.

The first article tested was furnished directly by GE Avio Aero and no further data are presently available for publication. The test bench can be adjusted to host other gearboxes in the near future.

5 Conclusions

This paper has described the test rigs currently being operated by the Research Centre for the Mechanics of Turbomachinery of the Department of Civil and Industrial Engineering of the University of Pisa.

Real tilting pad journal bearings and gears for advanced industrial applications can be tested using the presented experimental rigs. The shape, dimensions, materials, instability problems, power loss, and damaging phenomena can be investigated. The rigs are useful tools for testing innovative design solutions for the realization of more efficient power transmission systems with lower environmental impact.

The realization and operation of these very complex test rigs and their related plants would not be possible without strong cooperation between academia and industry, such as that established at the Research Centre for the Mechanics of Turbomachinery of Pisa. The synergy among the university, large companies, and small–medium-sized enterprise is essential to obtain significant results with industrial applications. The publication of these results is not a simple matter because there are often industrially sensitive data whose dissemination is regulated by specific non-disclosure agreements. However, results that do not contain sensitive data, as well as diagrams showing the trends of scientifically significant quantities without displaying the numerical values, can usually be published after a specific publication approval process.

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