A full size rotor dynamic test rig for whole engine mechanics of aero engines

R Liebich, P Kalinowski, O van Bargen
Chair of Engineering Design and Product Reliability, Berlin Institute of Technology, Germany

ABSTRACT

At the Berlin Institute of Technology a test rig is under construction that represents the rotor dynamic behaviour of an aero-engine almost in a full size scale. This test rig offers unique opportunities for vibration measurements especially for core blade off events. In this paper some of the most important subsystems of the test rig are presented. The information given covers design issues as well as supporting simulations. In this way the abilities and specifications of the test rig are shown.

1 INTRODUCTION

The optimal design of modern rotating machinery such as jet engines or power plant turbines requires reliable information about the dynamic behaviour of the developed engine. Especially the knowledge about the vibration level is very important to predict the lifespan of components such as bearings, shafts and structures. This could be reduced severely in case of out of balance operation or engine failure events. Much work has been done in the last years and decades on the influence of blade-out events on the durability of aircraft engines and how to prevent them. The main focus of these investigations was to show the strength of engine and aircraft components during Fan-Blade-Off events for certification issues. The blade off of small size blades from compressor and turbine stages was certified by numerical analyses only but never by experiments.

For experimental investigations of those Core-Blade-Off events, a full size “Whole Engine Mechanical Simulator” has been developed at the Berlin Institute of Technology during the last years (1). This simulator is a mechanical and rotor dynamic model of a typical small to midsize aero engine for business and regional aircrafts. It includes a complete casing-structure with typical pylons for fuselage mounting and a two-shaft assembly with operational speeds up to 12,500 rpm. The chosen full size test rig avoids the typical up scaling of results known from smaller test rigs. It is therefore much more next to reality and provides the opportunity to integrate real engine parts for special investigations.

The study of the available literature sources providing the current state of the technology confirms the uniqueness of the developed whole engine mechanical simulator. Until now any such complex test rig for experimental investigation of blade off events hasn’t been documented. The only example comes from Peters et al. who built up a test rig for a supercritical rotor similar to a low pressure shaft of an aero engine aircraft (2). That test rig can be successful used in order to
investigate the vibration behaviour of the mounted rotor, but it isn’t suitable for general investigations as described above.

2 ENGINEERING DESIGN OF THE SIMULATOR

The construction of the simulator is first of all a question of engineering design. In general the simulator shall be as close to a real aero engine as possible. On the other hand there are several restrictions that complicate the use of a real engine as a test rig. Besides the price of a real engine that significantly exceeds the budget for the test rig, an aero engine has several properties that make it difficult to measure the dynamic behaviour.

Especially the casing structure of modern engines mainly consists of closed circumferential ring structures. These structures are jointed by bolted connections that reduce the accessibility of many components in the inner engine. Moreover the aerodynamically shaped parts in the airflow within the engine also cover dynamically sensitive and interesting parts such as bearings and dampers. In this way the aerodynamic design hinders direct measurement at these locations. Several of the main components of the test rig have been adapted for the reasons given. The following sections cover some of these components and the major designs.

2.1 Rotors and Propulsion

The most important parts of the test rig are the rotors. Most of the other components such as bearings, seals or the casing depend mainly on the rotor’s design. Furthermore the rotors are the most critical parts of the entire engine. The rotors have to withstand excessive stresses by turning several hundred kilograms up to speed levels of 12,500 rpm. A rig failure would have severe consequences.

For these reasons the main parts of modern aero engine rotors are made of high strength titanium alloys. These materials combine very durable properties with a comparable small weight. Unfortunately these alloys are very expensive. Therefore, cheaper steel alloys have been chosen for full size test rig. Heat-treatable steel such as 42CrMo4 have comparable strength as titanium alloys, but the density of steel is much higher. The changes in weight and strength have significant impact on the design of the test rig rotors. In order to keep the stresses under the allowable level the diameters of the stages and the revolution speeds must be reduced. This leads to further modifications of the geometry (diameters and lengths) in order to match the natural frequencies of real engines.

All these considerations have been taken into account in an extensive iterative process, supported by detailed simulations (see below). Finally the design according to Fig. 1 has been obtained. The final design of the rotors mainly consists of two shafts with discs that represent the different stages of the engine. Due to manufacturing restrictions the shafts consist of different sections jointed by flanges and bolts. Both shafts are driven by electric motors, in case of the high pressure shaft by using belt drives in addition. Elastic couplings decouple the electrical propulsion systems from the rotor lateral motions. A real combustion propulsion system would exceed the effort regarding cost, risk, gas emission and heat. The aim of this fundamental investigation is to concentrate on the structural dynamics and rotordynamics during blade of events. For this reason the rig is called “Whole Engine Mechanical Simulator”.

48
2.2 Casing
As it has already been mentioned, the casing structure of modern aero engines is not optimal for the measurement of vibrations. The structure usually is encapsulated in order to guide the air-flow and uses excessive integral construction to reduce the weight of the engine. All these characteristics limit the accessibility of the major parts. Especially the rotors and bearings that are of great interest in rotor dynamic investigations are difficult to reach.

For these reasons the casing design of the test rig is set up different to aero engines. The carrying structure consists of three mounting rings that contain the bearing-cartridges, see Fig. 2. These mounting rings are connected by steal carriers and are directly fastened to the mounting pylons. In this way a very open and accessible configuration is obtained. Additional steal carriers are used to mount data acquisition systems and monitoring devices. The entire structure so far is very flexible in the axial direction. Cover sheets made out of aluminium are used to stiffening this direction and to close the structure while tests are carried out. These cover sheets are easy to remove and allow a dynamic behaviour comparable to that of real engines.

Figure 1. Design of the whole engine mechanical simulator, rotors and assembly

Figure 2. Casing of the test rig (left) and CAD section through bearing cartridges (right)
Intensive modal tests have shown that significant noise and vibration of the cover sheets must be expected within the considered range of revolution speeds. As a corrective for this problem foam sheets have been adhesively bonded to the covers. These foam sheets damp the vibration of the cover sheets and thus reduce the noise significantly. The realised casing is shown in Fig. 2.

2.3 Bearings, Lubrication and Sealing
There are challenging boundary conditions for the application of bearing technology in the test rig. In order to stay close to aero engines rolling element bearings must be used. Otherwise the dynamic properties would be altered too much. Considering the high revolution speeds (up to 12,500 rpm) especially the high pressure shaft has a very challenging ratio of bearing diameter and circumferential velocity. Hence, genuine engine bearings are the only available solution for this shaft. The low pressure shaft has much smaller revolution speeds (6,000 rpm) and smaller diameters at the bearings. Here, some standard ball bearings can be used as cheap alternatives.

In order to achieve a sufficient lifespan of the bearings an adequate lubrication system is needed. For this application an active oil injection system with independent injection for all bearings and three injectors per bearing has been chosen. As lubricant special purpose engine oil is highly recommended. The entire assembly is sealed by high velocity lip type seals made of Teflon.

In order to keep the test rig adaptable to for future tasks the whole bearing technology is mounted in removable cartridges. In this way further devices such as squeeze film dampers, used in real aero engines, can be added.

2.4 Safety devices
The rotors of the test rig reach revolution speeds of 6,000 rpm or 12,500 rpm respectively. This means the test rig is operated beyond the first resonance speed. Considering the rotor’s weight of several hundred kilograms and the respective inertia a tremendous amount of energy stored in the rotors during operation. In order to handle this situation two kinds of precautions against damage are taken.

First of all the entire test rig is comprehensively monitored during operation. This condition monitoring includes sensors for vibrations and rotation speeds as well as sensors monitoring pressures and flow rates of the lubrication system. Other sensors monitor the state of the propulsion system. All these data are evaluated in a programmable logic control that surveys the test rig and can initiate an emergency shut down if necessary.

On the other hand the test rig is built in a steal enforced concrete bunker. In case of a disaster the bunker will contain the debris and save the surrounding structures. The operation of the test rig is done from a surveillance room separated from the test rig.

3 SIMULATION TECHNIQUES

Intensive simulations have been carried out in order to ensure the security and the suitability of the test rig for whole engine mechanics. Some of these simulations will be presented in the next chapters. Basically they consist of state of the art finite element analyses and advanced rotordynamic simulations. Some of these simulations have been discussed already in detail in (3, 4 and 5).

3.1 Natural frequencies
The identification of the critical speeds is one of the most important tasks while developing rotating machines. A reliable definition of the low-vibration rpm-range
will be guaranteed only if the natural frequencies and the corresponding modes are known.

![Figure 3. Frequency response of the low pressure shaft measured in a free-free state (left) and simulated Campbell-Diagram of the mounted low pressure shaft (right)](image)

In order to validate a simulation model an experimental modal analysis has been performed. The assembled shafts have been hung up on elastic ropes representing a free-free condition. Fig. 3 (left) shows a result of the measurement illustrating a frequency response to a hammer excitation of the low pressure shaft. The marked peaks match the bending resonances located by 25 Hz, 70 Hz and 183 Hz. This information can be used to validate a rotor dynamic simulation model which is necessary to predict dynamic rotor behaviour.

The first simulation to be performed provides the natural frequencies of the mounted shaft. As the stiffness characteristics of the casing are still not known because of the enduring modal analysis tests of the housing a stiffness of 1.3×10⁸ N/m respectively 2×10⁸ N/m has been chosen. An example of a Campbell-diagram for the low pressure shaft is given in Fig. 3 (right). The dashed black line describes the driving speed. It crosses the green line of the torsion mode as well as the drifting apart blue and purple lines indicating the forward and the backward modes of the first and second bending resonance. The intersection of the lines marks the searched natural frequencies. Table 1 gives an overview of the relevant resonances for the LP- as well as for the HP-rotor.

### Table 1. Natural frequencies of the simulated LP- and HP-rotor

<table>
<thead>
<tr>
<th></th>
<th>1&lt;sup&gt;st&lt;/sup&gt; bending resonance</th>
<th>2&lt;sup&gt;nd&lt;/sup&gt; bending resonance</th>
<th>1&lt;sup&gt;st&lt;/sup&gt; torsion resonance</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Backward Mode</td>
<td>Forward Mode</td>
<td>Backward Mode</td>
</tr>
<tr>
<td>LP-rotor</td>
<td>31 Hz</td>
<td>38 Hz</td>
<td>104 Hz</td>
</tr>
<tr>
<td>HP-rotor</td>
<td>117 Hz</td>
<td>126 Hz</td>
<td>not relevant</td>
</tr>
</tbody>
</table>

#### 3.2 Rotor dynamic behaviour

In order to predict the rotor behaviour while the out of balance running it is necessary to carry out some particular analyses as frequency response or transient simulations. The following plots show some results of the frequency response analysis (see Fig. 4). The diagrams represent the maximal displacement of the low pressure (left figure) respectively high pressure shaft (right figure) as a function of
rotational frequency and illustrate the frequency range of high vibration due to resonance. The used excitation calculated from the unbalance amounts to 7.7 kgmm² applied to one of the discs.

![Figure 4. FEA results: frequency response of the low pressure shaft (left) and the high pressure shaft (right)](image)

Based on these findings it is possible to define some operating speeds for recommendation. For the low pressure shaft the speed-band of the steady-state operations extends to an approximately range from 0 to 1,800 rpm and from 2,700 to 5,400 rpm. For the high pressure shaft an operation within the range of approximately 0 to 6,600 rpm and above the speed of 8,400 rpm is recommended.

### 3.3 Strength and durability

One of the key aspects of the development of the test rig is the durability of all components. Due to the high rotating energy level system failure can cause severe damage. Furthermore, even minor defects, e.g. bearing failures, may result in significant down time and maintenance costs despite the open design of the test rig. Hence, a significant reliability is an important requirement for the rig.

In addition the test rig must be adaptable to future tasks which could not be specified yet. Otherwise the high investment for the test rig can not be justified. For this purpose a robust structure capable to withstand high test loads is desirable.

In order to fulfil all these requirements several finite elements analyses (FEA) have been conducted in order to optimise the structure. Some results are given in Fig. 5. Further effort was needed to optimise several initial designs for manufacturing.

![Figure 5. FEA results: optimisation of disc profile (left) and rough deformations of low pressure shaft (right)](image)
4 PROTOTYPE VALIDATION

In order to investigate the preferred blade release mechanism and to carry out the validation of the calculation method for the later simulation of blade off events a prototype has been build. It consists of a shaft with 2 large discs and it is supported by 2 groove ball bearings mounted in an ordinary bearing block. The shaft, which is a real copy of the LP compressor shaft of the whole engine mechanical simulator, is driven by an asynchronous motor and a belt drive to raise the rotational speed of the rotor up to 6,500 rpm. Fig. 6 shows the rig and the containment casing.

Figure 6. Prototype of the whole engine simulator without (left) and with containment casing (right)

The idea of the blade release mechanism bases on the imbalance weights release mechanism designed and tested by (6). The mechanism built by the authors consists generally of one or more blade loss probes that are shear pins more precisely considering a certain mass for release, two plungers, each with a hammer at the head and two springs which preload the plungers, see Fig.7. The sudden imbalance is generated by activating one plunger which that kicks the blade loss probes out of the disc while the shaft is rotating.

Figure 7. “Blade release” mechanism using hammer and shear pins

4.1 Measurement of bearing forces
There are different ways to measure the rotor response to an unbalance excitation. The common practice is to use displacement sensors to identify the shaft distortion. However, when the rotor shows a rigid behaviour it is necessary to provide an alternative measuring concept. An example of such a rigid rotor is the shaft of the prototype of the whole engine simulator whose first natural frequency amounts to approximately 300 Hz. In this case, the measurement of bearing forces is a good choice.
Figure 8. Bearing force development while blade off event: measurement (left) and simulation (right)

The left plot in Fig. 8 shows a typical result of the blade off measurement carried out on the prototype test rig. The applied piezoelectric sensors are very sensitive so that the bearing force development can be measured very well. The balanced state as well as the rotor response to an unbalance excitation can be observed easily. The high dynamics during the blade off can be seen as well. The load peak levels of nearly 600 N are lined up vertically and can be put down to the hammer excitation, see Fig. 7, that occurred in this direction. A comparison of the measured with the simulated results exemplified in Fig. 8 confirms the high agreement of both ways to predict the vibration development due to sudden mass loss. Though the simulated force peaks are slightly lower than the measured, the result is very satisfying.

The simulation techniques providing the right plot in Fig. 8 are based on a fully transient simulation program that was written by the authors and documented in the precedent papers, for example in (4). The numerical algorithm enables the modelling of the rotor and considers the gyroscopic effects and necessary boundary conditions as the variable ball-bearing stiffness for example. After simulation the rotor displacement on each grid point as well as the bearing forces are given.

The highest benefit resulting from the usage of this simulation program consists in the fully transient simulation technique which doesn’t require any characteristics of the rotational speed as a function of time but only the torque specification and the initial condition. Another advantage is the low amount of parameters which is appropriate for basic research.

4.2 Measurement of displacements

As mentioned the measurement of displacement is another way to quantify the rotor vibrations due to sudden mass loss. As this solution is suitable for elastic rotors in particular it will be put into practice on the whole engine simulator equipped with eddy current sensors. Those sensors have been also integrated in the rigid prototype test rig initially. However, the measured deflections were as expected very small. It confirms that the vibration level measured on a rigid rotor prior to mass release is very low and charged with many inaccuracies in dimension of the rotor deflection e.g. with the possible out-of-roundness of the shaft. Therefore, the authors pass the use of displacement measuring for the rigid prototype.
5 CONCLUSION

The paper at hand presents some of the most important systems of a test rig developed at the Berlin Institute of Technology.

The test rigs aims at the experimental simulation of engine vibrations due to sudden mass loss (blade loss events). Focusing on smaller core blade off events the test rig is believed to be unique as it almost reaches a one to one scale. The information given covers some important design issues as well as supporting numerical and experimental investigations.

Some insights in the development of this very complex test rig for whole engine dynamics have been provided. In this way important operation specification such as revolution speeds, unbalances and bearing forces have been discussed. Although the description of the work carried out cannot be presented completely the paper provides a general understanding of the abilities and opportunities offered by a whole engine mechanical simulator for blade off events.

6 REFERENCE LIST


