Low-cycle Fatigue in Small Turbines

Low-cycle fatigue is discussed and is the basis for explaining a compressor-rotor failure which occurred during the development of a turbojet engine

by G. R. Sorensen and H. R. Clemett

Abstract—Over the years, great quantities of fatigue data have been accumulated with the major emphasis placed on establishing either a finite endurance limit or fatigue strength within the range of 10,000 to 500,000,000 cycles. In recent years, particularly in areas of engineering where hot, high-speed rotating machines have been developed, the importance of defining fatigue strength from 1/4 to 10,000 cycles has been realized. Low-cycle fatigue, often ignored in the original design concept and frequently appearing in the later phases of product development, is discussed and is the basis for explaining a compressor-rotor failure which occurred during the development of a turbojet engine. The number of centrifugal stress cycles accumulated due to routine starting and stopping of the engine during its normal life was sufficient to cause a failure of the low-cycle fatigue type. The steps taken to investigate and eventually solve this particular problem are included. Based on the above, a correlation between stress history and residual strength of the part is postulated.

Introduction
Will it fail in service? This question, while probably the only one of significance to engineers concerned with the structural reliability of a given part, is generally the most difficult to answer. It is interesting that the engineer's answer to this question depends largely upon judgment in spite of his knowledge of the theory or information in the literature. If he is to answer the question properly, a thorough cognizance of the forces acting upon the structure in service and the environment in which it will operate is necessary. A limited knowledge of what the structure will be expected to endure may result in a false measure of reliability. Even armed with all of the variables, the answer is still largely a matter of judgment, although it would seem reasonable that the engineer's ability to judge is enhanced by an awareness of the facts.

In structures which transmit as much energy per pound of material as do small turbines, an awareness of the stress or strain history and its corresponding effects on residual strength is of extreme importance. Because of its high operating speed and severe temperature characteristics, the main rotor system is often the most critical structure in the turbine engine, and is therefore normally given the most attention. An aluminum compressor rotor is frequently part of this rotating assembly and during its normal life will be subjected to many different states of stress, as well as a range of stress levels, so that its remaining strength (life) at any point will actually be a function of time. To illustrate this point graphically, Fig. 1 is plotted to show generally the types of thermal, mechanical and aerodynamic forces it will be subjected to, the stresses resulting from these forces, and the resulting effect upon residual strength of the rotor. Notice that a change of environment, such as operating in a corrosive atmosphere, might significantly affect the rotor's ability to withstand the operational forces. Notice also, that the centrifugal stresses resulting from repeated starts and stops have a more pronounced effect upon residual strength than does a low-amplitude vibration. The internal stress conditions resulting from early material processing may also play a significant role in the rotor's ability to endure operation and carry out the purpose for which it was intended.

It is not the authors' intent here to carry out a complete analysis of structural problems, but merely

![Fig. 1—Possible life history of aluminum rotor](image-url)
to point out the need for an awareness of the complex nature of some of these problems, and to discuss one of the often overlooked and underemphasized variables, *low-cycle fatigue*.

**What is Low-cycle Fatigue?**

It is difficult to distinguish between low-cycle fatigue and regular or high-cycle fatigue. Coffin\(^1\) defines it as "that class of fracture in ductile metals where, under cyclic loading, failure takes place in less than 10,000 cycles." The authors suggest that line of demarkation may in some cases be closer to 100,000 cycles and thus overlaps the range of 10,000 cycles and includes part of the range reserved for normal fatigue.\(^2\)

Regardless of exact definition, it is important to know that low-cycle fatigue results often times from a duty cycle normally considered by the design
engineer to be static. Assembly stresses, thermal and centrifugal stresses are examples. The stresses resulting from assembly, of course, are not significant in themselves as a cause of low-cycle fatigue unless a large number of disassemblies are accomplished during the life of the machine considered. (They may be significant, however, in an additive sense.) In the case of a turbine or compressor rotor, the thermal stress resulting from sharp temperature gradients during an acceleration and/or centrifugal stress are of great significance not only from the stress-rupture point of view but, in addition, fatigue from repeated start and stop cycles, as well as changes in engine speed and power during operation. The material described in this paper deals primarily with the latter, and a specific example of the approach taken to improve this condition is discussed.

Dealing with Low-cycle Fatigue

The importance of considering low-cycle fatigue in the design analysis of high-stressed structures was recognized at Continental early in the development of turbojet engines. It is safe to say, based on our own experience and verified by other published technical papers, that turbine and compressor burst problems have been a concern of all turbine-engine manufacturers.

The centrifugal compressor shown in Fig. 2 burst during engine endurance testing on a sea-level test stand, after 500 hr of operation. During this 500 hr, the compressor stress history included many stress cycles due to changes in engine speed, as well as some 1400 start and stop cycles. The type of failure shown here can cause extensive damage to structures in its path, and implies the seriousness of a rotor burst during aircraft operation.

The desired life for the compressor rotor in this particular engine was a minimum of 1000 hr. A second test rotor failed in 600 hr on the same type of testing, indicating that the first failure was not an isolated case, and that design changes were necessary if the 1000-hr objective was to be achieved.

The low-cycle fatigue-type failure very often does not reveal its existence until the latter phases of engine development because of its very nature, unless special accelerated tests are set up to ferret it out. In the case of the example being discussed, the rotor had passed all normal engine-endurance tests and was in production before enough test time, on enough samples, had been accumulated to determine the low side of the S-N curve. Because of production considerations, the freedom to make obvious design changes was not possible because of interchangeability reasons and the prohibitive cost of replacing mating parts. As seen in Fig. 3, this rotor is driven through slots on the back-face near the bore and it is immediately apparent in Fig. 2 that the fracture originated in the slot.

Although the bore size, hub length and slot location could not be altered, several methods were available to reduce the stress at the origin of fracture, improve the state of stress, or the material strength:

1. Overspinning to introduce favorable residual compressive stresses at the slot and bore.
2. Change slot geometry.
3. Introduce additional slots as relief slots.
4. Peening of slot.
5. Increase ductility.
6. In the case of used rotors, remove the damaged material at the bottom of the slot.

Several methods of evaluating these various design changes were available:

1. Analytical.
2. Engine-life tests.
5. Stress measurement in spin pit.
6. Stress measurement with static bench test.

Analytical methods of evaluation of proposed changes are not accurate enough to establish the desired improvement. Engine testing, although used in final evaluation, is better used only after design improvements can be demonstrated by other means. Therefore, several experimental methods were used simultaneously. The bench test shown in Fig. 4 was the simplest but the least accurate test. Here the compressors were cut through at one point from the outside diameter into the bore, and a pocket milled out for a hydraulic ram. Application of a load at this location approximately simulates the centrifugal stress on the opposite side of the bore. Various shapes and depths of relief slots were evaluated with Stresscoat and strain gages, as shown in Figs. 4 and 5. After a given configuration had produced a significant stress reduction by this method, the modification was then reproduced on another rotor and stress measurements were made.
with Stresscoat by spinning in an evacuated spin chamber. The modification shown in Figs. 6 and 7 demonstrated a 25 percent reduction in peak stress and was then subjected to simulated-life tests in the spin chamber. Figure 8 shows the automatic control console and the chamber used for the life tests. Rotors were fatigue tested continuously on a 24-hr basis until failure occurred at which time the operation would shut itself down. Several rotors of both the original and modified design, with and without previous engine time, were cycled to failure. Some of the fracture patterns are shown in Figs. 9 through 11.

Figure 12 is a low-cycle fatigue diagram resulting from the rotors tested. Test points (6) and (7) at 1/4 cycle are bursts of new parts of the original design which were established early in the engine-development program and prior to release of the part for production. Based on general design rules for rotors, a burst speed of 38,000 rpm and an operating speed of 22,700 rpm should have resulted in a satisfactory service life. Some sources have indicated an operating speed of 75 percent of burst speed is acceptable. By comparison, this rotor is operating at 63 percent of its burst speed. If the stresses involved are considered, it is operating at a
for tension and compression, this predicted life would be conservative. Thus, in this case, the actual engine life does not correlate with predicted life but is actually only 1,400 cycles minimum. There are several variables in the engine which tend to lower the cycles to failure that are not present in the spin chamber, such as temperatures of 200 to 350°F, and temperature gradients causing thermal stresses which change magnitude each time the engine speed is changed. Because of these factors, a spin chamber cycle of from 10,000 to 20,000 rpm was chosen for relative design evaluation in order to produce failure in a reasonable length of time. This represents a stress range of \((126^2 - (10)^2)/36)^2\) or 44 percent of the burst stress as compared with engine conditions of 0 to 22,700 rpm or 40 percent of burst stress. At this stress level (44 percent) and at room temperature, the part should go 40,000 cycles according to the notched S-N curve for the material. Fatigue tests of the original design run in the spin chamber demonstrated a minimum life expectancy of just over 2,000 cycles, as shown by test points (8) through (12), and this compares to engine life of 1,400 cycles. Test points (3), (4) and (5) show the reduction in static strength or bursting strength of rotors which had accumulated 488, 570 and 840 hr in the engine prior to bursting in the spin chamber. It is interesting to note that this curve intersects the 2,000-cycle life of rotors that were cycled to failure at 44 percent of burst stress in a manner that appears to form an S-N curve for this part. Overspinning to 33,000 rpm prior to cycling did not improve the life as shown by test points (8) and (11). The modified slot configuration with a 25-percent lower stress in the slot demonstrated what appears to be a notable increase in fatigue life. Test points (15) and (16) represent rotors that had accumulated 1,000 hr of engine time as original designs before being modified and fatigue tested. Test points (13) and (14) represent original unused rotors that were modified and fatigue tested. None of these rotors were cycled to failure but one rotor, new but modified, and one with previous engine time and modified, were burst after the accumulation of more than 10,000 cycles. The static strength of the unused modified design after 11,000 cycles is the same as the static strength of the original. If certain assumptions are made with the limited data of two test points, the curve would indicate fracture in about 20,000 cycles. At the completion of the laboratory phase of the test program, it was of interest to check the burst strength of a new, but modified, rotor to compare with the original design. A single sample shown by test point (17) burst at a speed of 39,500 rpm. This close agreement with earlier burst tests was not surprising since the ductility of the metal in the notch was felt to be sufficient to allow the nominal stress to reach ultimate even in the original design.

It is interesting to note that Coffin in his paper, "Design for Low Cycle Fatigue," demonstrates that low-cycle fatigue life can be predicted mathemati-
Fig. 12—Low-cycle fatigue diagram—original and modified designs

Fig. 13—S-N curve for 2014-T6 aluminum—plain and notched

ally if the plastic strain range is known. It is unfortunate that at the time the compressor burst program was conducted, no practical means were available for determining strain range in service. Recent advances in the state-of-the-art have resulted in an improved strain-measuring technique which makes feasible the direct measurement of the low-cycle or “static” operational strains. Current and future work at Continental, in which low-cycle fatigue is of interest, will include attempts to predict life by direct strain measurement.

Closing Remarks

The ultimate goal of the design engineer is to be able to predict the life of a structure prior to the sudden answer he often gets when the service report indicates a field failure. This goal is still off in the distance in many cases but, through the constant efforts of stress analysts, experimental mechanics and materials engineers, the gap is closing down. Structural problems, so often complicated by multistates of stress and changing environment, make it mandatory for the design engineers to utilize all the means at their disposal in gathering the facts such that the proper judgment might be made.

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