ORDER

8110.9

HANDBOOK ON VIBRATION SUBSTANTIATION AND FATIGUE EVALUATION OF HELICOPTER AND OTHER POWER TRANSMISSIONS SYSTEMS

January 20, 1975

DEPARTMENT OF TRANSPORTATION
FEDERAL AVIATION ADMINISTRATION

Distribution: WFS-3; RFS-4 | Initiated By: AFS-140

FOREWORD

1. PURPOSE. This handbook contains information on vibration substantiation and fatigue evaluation of helicopter and other power transmission systems.

2. DISTRIBUTION. This handbook is distributed to section level and above in all Regional Engineering and Manufacturing Branches in Flight Standards Division offices, to Branch level and above in the Aircraft Engineering Division in the Western Region and to Branch level and above in the Engineering and Manufacturing Division in Flight Standards Service.

Paul L. Clark
Acting Chief,
Engineering and Manufacturing Division
Flight Standards Service

T A B L E O F C O N T E N T S

CHAPTER 1 - INTRODUCTION

1. SCOPE. The procedures described in this handbook provide information on the vibration substantiation and fatigue strength evaluation of power transmissions.

2. GENERAL CONSIDERATIONS.

a. The power transmission system between the engine and the various other rotating components of an aircraft, (such as a helicopter VTOL or STOL) may have many modes of vibration, some of which may be of a hazardous nature. The power transmission could alter the vibrational characteristics of the basic engine to the extent that the vibrational data determined during engine certification may no longer be applicable.

b. Vibration in the power transmission system can be the result of engine excitation, aerodynamic excitation, and mechanical excitation. It can occur either during steady state or transient operation. When single-engine installations are involved, variations in both power and r.p.m. are important. In multi-engine or multi-rotor installations, however, not only power and r.p.m. but also phasing of the engines and rotors should be considered to determine whether maximum vibration occurs under in-phase or out-of-phase combinations.

c. Each branch of the drive system should be investigated for both torsional and bending type of vibration. In reciprocating engine installations the power transmission system vibrational spectrum may be between 0 and 1,000 cycles per second. In turbine engine installations the upper end of the vibration spectrum could be considerably higher if the excitation originates in the engine. Certain modes of vibration can be investigated and substantiated by ground testing of the complete power transmission system or components thereof. Other modes of vibration, where the excitation does not occur during ground operations, should be investigated in flight under the maneuvers that produce the vibration.

d. The drive system is responsive to engine, mechanical, and aircraft aerodynamic forces primarily because the various branches of the system possess certain natural frequencies. Resonance may occur in these branches when excitation frequencies approach or coincide with the natural frequency of the branch. When a complex drive system is in resonance, the complete system is involved, not just an individual portion. Each branch of a system becomes important when investigating the vibrational response of the system, however, some modes of vibration of a complex system may be more predominant in certain branches. Therefore, all branches of the system should be investigated.

e. Stress and/or amplitude measurements on the same branch of the system may be necessary to evaluate the vibration characteristics of the power transmission system during all normal operating conditions.

Each branch of the system should be investigated by measurement unless other means are found acceptable. Measurements for critical speeds in bending (whirl mode) made on shafting sections, such as the tail rotor or interconnecting shafts, should include determination of the zero frequency component, since the shaft metal does not experience a reversal of stress but merely an increase in steady stress. For a perfectly balanced drive shaft, it is possible to go through the critical whirling speed without detecting its location unless the shaft is desplaced from its normal undeflected position. Once deflected, however, it can remain deflected regardless of speed.

f. Vibrations encountered during transient operation of a system may be more important than steady state vibrations. Even though the transient may last for only a relatively few cycles, the stresses may be of such magnitude as to cause fatigue damage. In helicopters, such transients can occur during clutch engagement or disengagement, recovery from autorotation, starting of the engine, etc. For other drive systems such as STOL or VTOL aircraft, transition from vertical to horizontal flight or other flight maneuvers may be involved. Where multi-engine or multi-drive systems are used, phasing of the systems may be important, with in-phase excitation being grossly different than out-of-phase excitation. If the design of the system does not control such phasing by locking the system in a single relationship, all possible phasing should be examined. Where a portion of the system incorporates a belt drive, the flexibility of the belts and belt tightness are important. Belt designs incorporating interwoven steel or metal cables will have a different rigidity torquewise as compared to an all-rubber belt system. Changing from one belt type to another may alter the vibrational characteristics.

3. VIBRATION MEASUREMENT INSTRUMENTATION

a. The electrical strain gage has become the most useful tool for determining the vibrational characteristics of power transmission systems. Signals generated by these transducers attached to the part under investigation can be fed into suitable signal conditioning equipment and then recorded on a multi-channel device. This equipment can provide stress, amplitude, frequency, and phase relationship at any instant of time. In this regard, it is important to be aware of the limitations of the equipment and their possible effect on the accuracy of measurement.

b. It is desirable to locate the strain gage at the point of greatest stress to provide maximum gage response, however, in many cases this is impossible due to component geometry. It then becomes necessary to determine the stress relationship between the point where the strain gage is located and the point of maximum stress. Stress evaluation of the part using conventional techniques such as stress coat, photoelastic stress analysis, or other methods can provide this stress ratio. Stresses measured during the flight strain survey may then be adjusted by these stress ratios to provide the highest stress to which the part is exposed.

CHAPTER 2 - SYSTEM ANALYSIS

4. GENERAL.

A dynamic analysis can help to identify critical vibration modes and estimate the vibratory response under the most servere conditions that would probably occur. This analysis is useful in defining the instrumentation needed to investigate the power transmission system.

5. DRIVE SYSTEM TORSIONAL EVALUATION.

a. A power transmission system is made up of a combination of concentrated masses connected together by a number of different flexible units. The masses may be in the form of rotors, flywheels, gears, cooling fans, engine components, clutch units, etc.; while the flexible units are the shaft components, such as the main rotor drive shaft, tail rotor drive shaft, fan drive, interconnecting drive shafts, etc. Any dynamic system has as many possible natural modes of vibration as it has combinations of masses and stiffnesses. These natural modes of vibration do not result in potential problems unless the natural frequency of a vibration mode coincides with an exciting force frequency. Since the dynamics of the system can react in either a torsional or bending direction on its components, this at least doubles the number of critical vibration conditions that can result in potential dynamic problems in the transmission system. This is one of the reasons for the need to measure both bending and torsional vibratory stresses in each branch of the system. Since these natural modes of vibration can be excited by either steady state or transient operating conditions, an investigation should be made for all conditions. The transients can be due to various operating conditions such as changes from one operating regime to another, power chops, gust loading, maneuvers, etc.; whereas, the steady state operation involves stabilizied rpm, power, or airspeed.

b. With these potential problems in mind, the evaluation should begin with a detailed dynamic analysis of the engine/drive system/rotor system combination using measured or calculated masses and spring rates for each element of the system. The number of torsional modes to be analyzed would depend on the complexity of the drive system; however, for the conventional single main rotor/tail rotor helicopter, the first four modes may be all that are of interest. More complex drive systems require considerably more analytical work which can best be handled by a computer which is programmed with a mathematical model of the dynamic system. The analysis should define the natural frequency for each mode, the mode shape, and the relative torsional deflection throughout the system. A corresponding study of the engine, the rotors, couplings, universal joints, fans, accessory drives, gears, control devices, etc., is required to identify the vibration excitation sources and their frequencies, most of which vary directly with rotor or engine rotational speed. A cross plot of exciting frequencies and the resonant conditions will define the critical vibration conditions. Also, the analysis will establish the nodes and anti-nodes which, in turn, help define the type and sensitivity of the instrumentation to be used. For example, strain gages should be placed at or close to nodal points where maximum stresses occur. Torsiographs, accelerometers and velocity pickups should be placed at anti-nodal points where maximum deflection occurs. Of equal importance, an analysis highlights the probable significant vibration phenomena to be expected and provides sufficient insight into the drive system responses to enable the evaluator to quickly and accurately classify the output of the instrumentation.

c. The analysis can be extended to cover transmission gear teeth contact frequencies which must be compared with turbine blade and disc resonant frequency data (obtained from the engine manufacturer). Any coincident frequencies should be avoided or substantiated.

d. Since each engine is considered as one branch of the drive system, the vibration environment of its dynamic components, including accessory drives, will differ from that which was substantiated under FAR 33. This new vibration environment should not be damaging to the engine, and the engine manufacturer should be encouraged to investigate and substantiate this. An acceptable alternative would be for the airframe manufacturer to forward his vibration investigation data to the engine manufacturer. The results of the engine manufacturer's review could either confirm the airworthiness of the combination or define special tests to establish this objective. Advance coordination between the engine manufacturer and the airframe manufacturer is recommended to maintain some uniformity in the basic test parameters and method of measurement to assure the desired compatibility of vibration data.

6. TORSIONAL INSTABILITY.

a. Some torsional instability can be expected with governor controlled engines. This phenomena is exhibited as a low frequency torsional oscillation. It occurs when the fuel control governor system has a response rate close to the first or second drive system torsional natural frequency. In the extreme situation, this coupling of the exciting force (torque oscillations responding directly to the fuel control) and the drive system resonant frequency can become divergent and destructive. The designer should determine early in the development program the probable first and second natural or resonant frequencies of the dynamic components, including the components of the engine directly coupled to the drive system. The engine manufacturer should be able to provide data on the fuel control response rate. If frequency coupling is apparent, redesign efforts by either or both the airframe manufacturer and the engine manufacturer are required to provide a suitable combination. The redesign may consist of changes to avoid frequency coupling or improvement in the damping characteristics of the system. In either case, changes to the fuel control are usually the most productive.

b. The suitability of the final configuration should be demonstrated by test. The test program should include cyclic inputs of the collective, anti-torque, cyclic, and most importantly, the fuel or governor control available to the crew. Frequency of control input should be as close as possible to drive system resonant frequency. While the use of mechanical control inputs is desirable, reasonably good test results can be obtained by pilot inputs, although several attempts may be necessary to achieve a meaningful test. Input amplitude should be adequate to obtain a meaningful record.

c. Instrumentation requirements may vary widely with drive system complexity. Some typical examples are: torsionally oriented rotor mast strain gages, engine fuel nozzle pressure, engine torque signal, gas generator tachometer, control displacement indicators, etc. All parameters should be recorded on tape or by oscillagraph in order to determine the phase relationships of the measured parameters and to establish that the oscillatory inputs from the controls damp immediately and at an acceptable rate.

7. DRIVE SYSTEM BENDING.

a. Shaft bending in helicopter drive systems falls into two categories, one that is associated with whirl modes, which is essentially zero frequency, and the other that is associated with thrust forces at the unsupported end points of the drive system. The component of the thrust load normal to the shaft produces a bending load, which for two bladed rotors, will vary, depending on whether bending is measured in-plane with the rotor or perpendicular to the rotor. Both conditions should be evaluated. Cyclic bending of shafting also occurs with the use of belt drives, certain gear arrangements, etc., which produce loads normal to the axis of the shaft.

b. The critical (resonant) rotational speed should be calculated for each segment of shafting. Calculation methods should employ conservatism and should normally show large margins with respect to any rotational speed to be expected. Overspeeds may occur during transient or speed control failures. Exceptions to this acceptance criteria would be shafting which is designed to operate super-critically, i.e., at speeds above the resonant frequency of the whirl mode. Special precautions are in order in this case to insure accurate balancing, relatively high shaft stiffness, reliable support bearings and shaft protection from external forces or damage inducing conditions which will affect shaft balance.

8. INSTALLED ENGINE VIBRATION.

a. Airworthiness objectives in this area require that the engine be installed and mounted in such a manner as to insure that external vibratory inputs transmitted to the engine through mounts and other attach points are within the limits established by the engine manufacturer. The combination of engine mounting and airframe vibrations inherent in a rotary wing aircraft will produce an environment which justifies investigation and evaluation.

b. The engine manufacturer is expected to establish vibratory limits for all engine components and to define the basic instrumentation to dettermine compliance with these limits. Usually, the limits are defined as an allowable combination of frequency and amplitude, or, more simply as an allowable acceleration. The instrumentation would consist of various transducers mounted as prescribed by the engine manufacturer to measure motion in three planes.

c. The test program should explore the complete range of rotational speeds of the drive system, both coupled to the engine and in autorotation. Engine start and acceleration should be examined, and speed/power sweeps should be conducted over the expected operating range. In some instances, rotorcraft center of gravity will affect test results. Multi-engine rotorcraft test programs should consider combinations of inoperative engine(s) and unequal power from the engines. Preliminary review of the data should be accomplished and any unusual or marginal vibratory condition should be reexplored in detail.

9. INLET/EXHAUST DISTORTION.

a. The basic purpose of inlet distortion investigations is to establish that turbine engine compressor blades are not excited at resonant frequencies by circumferential variations in air inflow to the engine. Ideally, this would be established by strain gage measurements of compressor blades during in-flight maneuvers of the aircraft under evaluation. In practice, the engine manufacturer determines experimentally the compressor blade response to partial inlet blockage during test stand runs. Different blockage patterns are used to excite higher orders of inflow distortion and from this basic information the engine manufacturer can determine the tolerance of his compressor to inflow disturbances. The airframe manufacturer can then measure inflow disturbances which can be compared with the distortion limits established by the engine manufacturer.

b. Instrumentation for measuring the inflow disturbances should provide data compatible with the parameters established by the engine manufacturer. These parameters usually involve differential air pressure and are measured by multiple orifice pressure probes installed directly in front of the engine compressor. With this instrumentation installed, the aircraft should be flown through its complete flight envelope.

c. In some instances, the engine manufacturer may specify a particular inlet configuration which in itself insures that airflow to the compressor is not distorted beyond compressor tolerances. In these cases, no airflow distortion tests by the airframe manufacturer are required.

d. Exhaust gas flow deflections associated with severe duct section discontinuties can result in exhaust gas pressure reflections back into the turbine stages. These pressure reflections must be evaluated for possible turbine blade vibration. The engine manufacturer is the source of this information and data regarding the measurement and interpretation of this condition.

CHAPTER 3 - VIBRATION MEASUREMENT EVALUATION

10. GENERAL.

Several of the problems discussed in Chapters 1 and 2 can be evaluated by straight forward comparison of measured or calculated loads with firmly established limits. Among these are torsional instability, installed engine vibration, inlet/exhaust distortion, and transmission gear tooth excitations. Other vibratory phenomena such as drive system torsion and bending require special evaluation techniques. These techniques fall into two basic catergories, the analytical method or fatigue test method. These methods are briefly discussed below; however, use of this information obviously must be supplemented by experience and by guidance from the extensive text book and handbook data available to vibration and fatigue investigators.

11. ANALYTICAL METHOD.

a. It is recognized that, if allowable stress levels are established by acceptable means, and the stresses measured during ground and flight conditions are lower than these established levels, no fatigue testing of actual parts is necessary.

b. At this point it is appropriate to discuss fatigue substantiation of gearbox components other than critical components, such as the main rotor shaft, which should have the same degree of structural reliability as the main rotors. Components such as gears, gearshafts, and bearings do not readily lend themselves to evaluation by flight strain survey data. Fatigue margins of safety may be computed for gears and gearshafts. Analysis of this type should, where applicable, consider the simultaneous application of combined stresses involving axial bending, and torsion loads in conjunction with steady state and vibratory loads. Equations used for this purpose should be based on a theory of failure appropriate for the component under consideration and consistent with good industry practice. In this regard, acceptable standards for gear design are published by the American Gear Manufacturers Association (AGMA). Postive margins of safety computed by the above techniques indicate infinite life for the part, and therefore, no further fatigue evaluation would be required. Conversely, negative margins of safety indicate a finite life which should be established by fatigue tests of actual parts.

c. Fatigue evaluation of bearings should be based on acceptable industry standards such as those published by the Anti-Friction Bearing Manufacturers Association (AFBMA). Bearings used in non-redundant critical applications may necessitate special selection criteria with respect to tolerances and materials.

d. For other parts a method, somewhat similar to the one described for gears and gearshafts in Para. 2b, is based on the use of the modified Goodman diagram. This method is also considered acceptable for establishing an allowable stress level as follows:

(1) Estimate the mean endurance limit of the part from test results of specimens with similar stress concentrations. The test specimen material should be representative of the actual part and sufficient test data should be available to substantiate the mean endurance limit. The estimate should account for surface conditions, fabrication methods, fretting, chafing, size and shape effects, as well as differences in stress concentrations between the test specimen and the actual part. Referring to Figure 1, the mean endurance limit may be represented by a straight line drawn through the yield stress (point A on the horizontal axis) and the maximum oscillatory stress which the average specimen can withstand at a given steady stress (point B) without failure for 10^7 cycles to 5 X 10^7 cycles depending on the material. (Aluminum would require at least 5X 10^7 cycles).

(2) A factor of safety of 3 should then be applied to the mean endurance limit so that the slope of line AC would be 1/3 of line AB.

(3) If the flight strain measurements indicate that all of the operating stresses fall below the operating boundary line (AC), no fatigue testing is necessary. When the measured stresses are above the operating boundry line, however, fatigue testing of the actual parts should be conducted.

(4) Limitations of the analytical method: Caution should be exercised in the application of the method above, particularly when the following items are involved:

(a) Large parts in proportion to the laboratory specimens.

(b) Irregularly shaped parts containing numerous or superimposed fillets, holes, threads, or lugs.

(c) Parts of unique design for which no past service experience is available.

(d) Parts subject to fretting or chafing.

(e) Bolted or pinned connections.

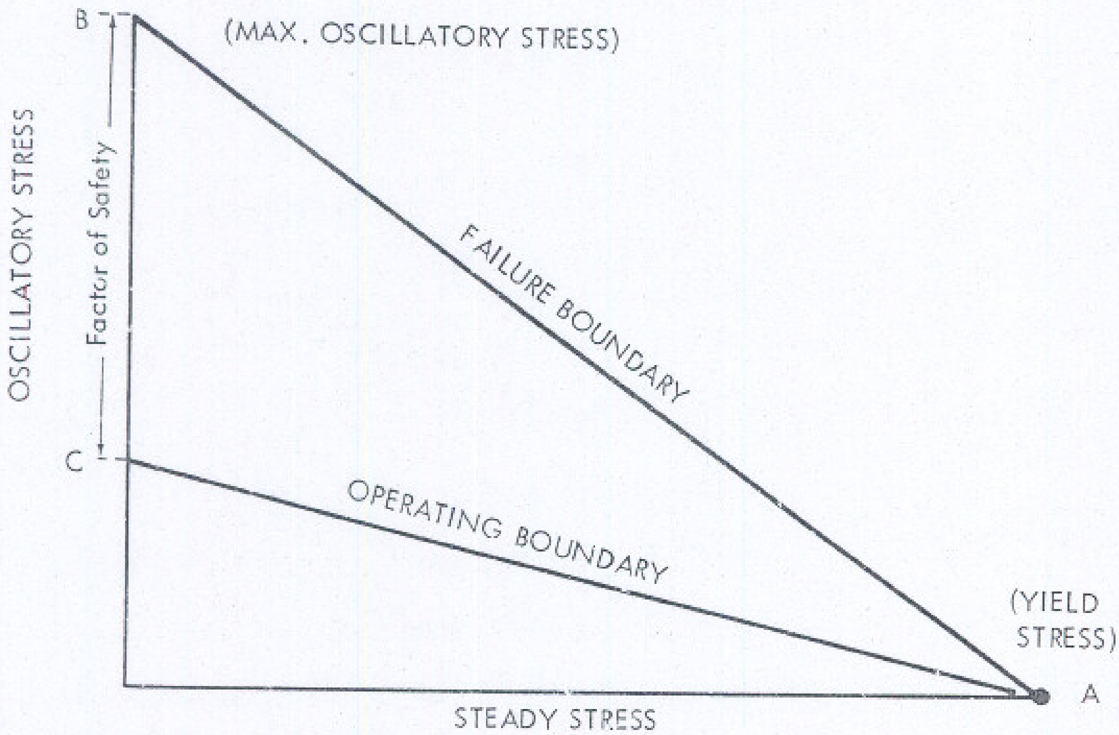

FIGURE I

12. FATIGUE TEST METHOD.

a. The fatigue strength of powerplant and rotor drive system components may be determined in appropriate labratory tests. These tests should be conducted in such a manner as to produce valid fatigue strength data for the component in the presence of both steady and oscillatory loads as they are generated by other components of the rotorcraft. Normal practice is to obtain sufficient specimen failures to enable the evaluator to establish a statistically valid S-N curve as shown in Figure II for each steady load selected.

b. The establishment of each S-N curve involves testing a sufficient number of parts at the same steady stress level and varying the oscillatory stress. Thus, in Figure II, if at a steady stress level A and an oscillatory stress of level B, the part is tested until failure, failure occuring at N_1 cycles, a point on the S-N curve for steady stress of level A is determined. Additional points on the S-N representing a steady stress of level A may be determined by choosing a different oscillatory stress level and testing the part to failure. If no failure occurs for a specific loading condition, after 10^7 to 5 X 10^7 cycles, depending on the material, the part can be considered to have infinite life at that stress level.

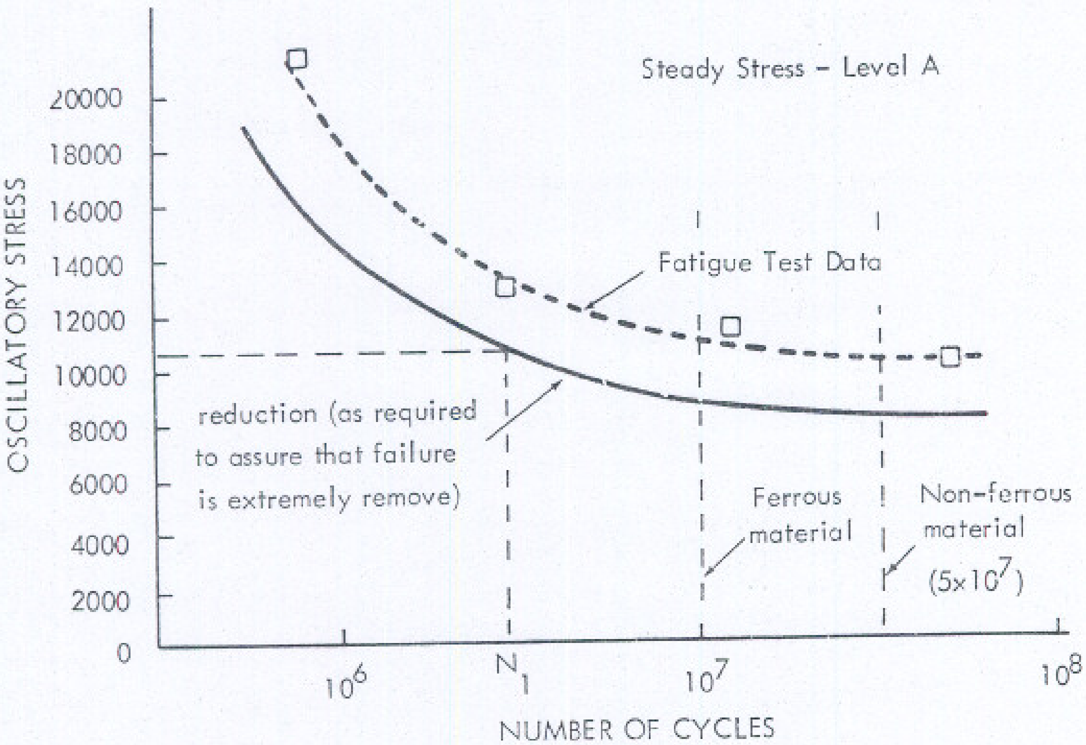

FIGURE II

c. To obtain representative results from fatigue testing, a large number of test specimens is desirable in establishing each S-N curve. However, most manufacturers cannot afford the cost and time necessary to obtain such accuracy. Therefore, a minimum of four test specimens which will establish a well defined curve over the range of oscillatory stress levels expected to occur in service is considered acceptable in establishing each S-N curve. At least one test specimen should survive the number of test cycles required to define infinite life. In order to compensate for the scatter associated with fatigue testing, the mean S-N curves should be reduced by an appropriate factor. This factor applied to the stress axis, should be based on the type of material being tested, past service experience with the material, and type of design. For critical components, whose failure may be catastrophic, the reduction factor should be selected to assure that the probability of failure is extremely remote. Statistical methods are useful in this aspect of the evaluation. The shape of the resulting reduced curve should be influenced by typical published S-N data for the material, and all of the test points should fall above the reduced curve. This curve would then represent the S-N curve for use in determining the fatigue lives. Figure II represents this method of constructing a typical S-N curve based on test specimens. A separate S-N curve should be established for each critical steady stress level determined in the flight strain measurment survey. If it is desired to limit the fatigue tests, a single S-N curve based on the highest measured steady stress may be used in the fatigue life calculations. However, if this approach tends to unduly limit the fatigue life, a family of curves may be developed from two established S-N curves by means of Goodman or similar diagrams or by rational methods.

d. The reduction factor may also be based on statistical analysis of the failure data projected to the number of cycles associated with infinite life for the particular material. The shape of the curve drawn through each data point is obtained from experimentally derived constants appropirate for the material. The mean endurance limit is the statistical average of the curves projected to 10^7 or 5×10^7 cycles. The working S-N curve would be established at 3 standard deviations below the mean value as shown in Figure III.

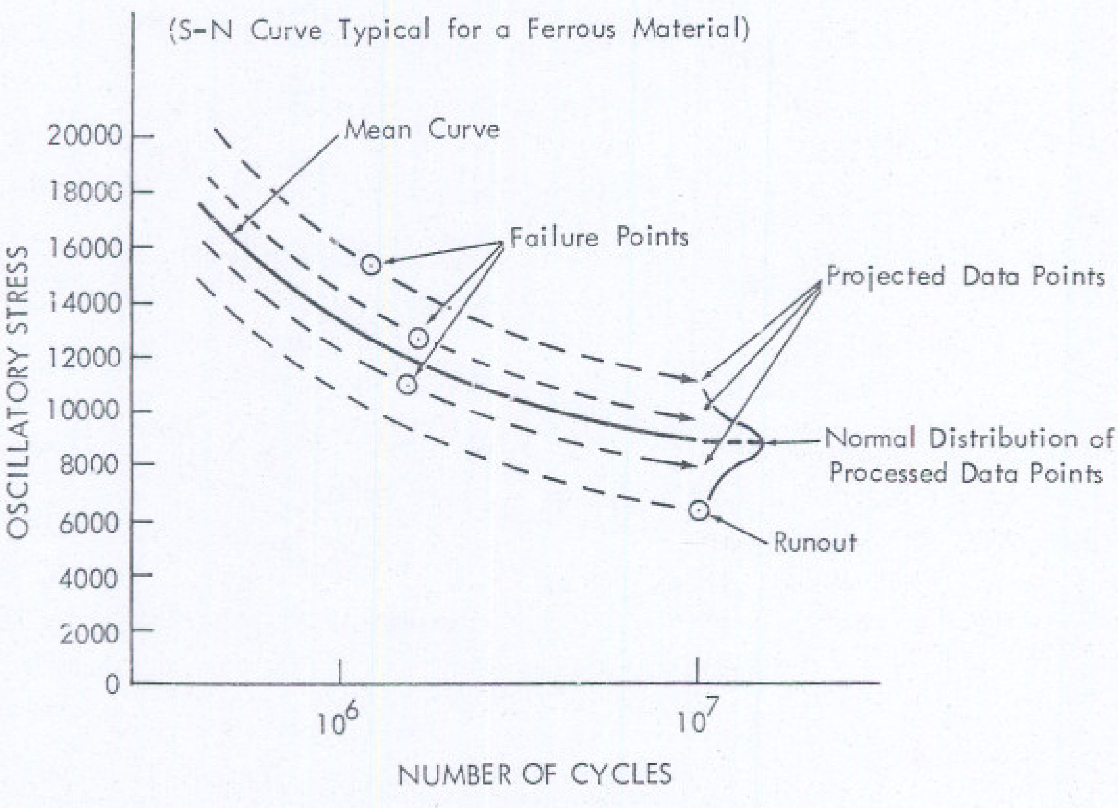

FIGURE III

e. Using the S-N data obtained above and a knowledge of stresses imposed on the component by flight and ground conditions, the fatigue life limitation, if applicable, can be established using one of the cumulative damage theories available from current literature on this subject. An example of this, using Miner's theory, is shown below. Note that flight conditions which result in stresses below the endurance limit defined by the S-N curve are considered to have no effect on the fatigue life of the component under evaluation.

<u>Sample calculation based on S-N data</u>:

The Cumulative Damage Hypothesis states that every cycle of stress above an endurance limit produces damage proportional to the ratio of cycles run at that stress to the fatigue life at that stress level. Thus, if a part is subject to random loading for n_1 cycles at a stress level of S_1, n_2 cylces at S_2, n_3 cycles at S_3, and so on, and if N_1, N_2, N_3 are the corresponding number of cycles to failure for each stress level, then failure will occur with the summation:

$$\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+\frac{n_i}{N_i}=1, \text{ or } \Sigma\frac{n_i}{N_i}=1$$

Using this expression, the calculated service life of a part subjected to random loading can be determined if the percent of life used per hour at each damaging stress level is known. The percent of life used per hour at each damaging stress level, can be expressed by:

$$l=\frac{an}{N} \qquad (1)$$

where l = percent of life used per hour at the damaging stress level;

a = percdnt of total operating time allotted to the flight condition during which the damaging stress level was recorded;

N = total number of cycles of the damaging stress level at failure; and

n = number of cycles the damaging stress level occurs per hour.

Thus, the calculated service life, L_c of a particular part of component subject to a random number of damaging stress levels, would be:

$$L_c=\frac{100}{l_1+l_2+l_3+\ldots l_i}=\frac{100}{\Sigma l_i} \qquad (2)$$

A sample calculation illustrating this method for determining the calculated service life is shown in Table II. In this example, the peak steady and vibratory stress levels associated with each maneuver have been assumed to occur for the duration of the maneuver (columns 3 and 4). In addition, the cycles of oscillatory stress per hour also has been conservatively assumed at the maximum level throughout the flight spectrum (column 5). If this procedure tends to limit the service life unduly, it is acceptable to use the actural measured stress level distributions if proper account of possible variations is provided by repeated maneuvers.

The number of cycles to failure for each damaging stress level (column 6) was determined from Figure II. As an example, consider flight condition II(c) of Table II. The percent of total operating time (a) considered at this maneuver is 0.5 percent, the damaging osciliatory stress level is 10,500 psi, the number of cycles of damaging stress per hour (n) is 23,200, and the number of cycles to failure (N) from the S-N curve (Figure II) is 3,200,000 cycles. Then by equation (1) the percent of life used per hour at this damaging stress level would be:

$$1=\frac{an}{N}=\frac{0.5 \times 23{,}000}{3{,}200{,}000}=0.00362$$

The summation of the individual percentages of life used per hour for each damaging stress level is shown in column 7 of Table II. Therefore, by equation (2), the calculated service life of this part would be:

$$L_c=\frac{100}{\Sigma 1_i}=\frac{100}{0.15289}=654 \text{ hours}$$

A summary of the measured stress and percent life used at the various flight conditions should be submitted with the fatigue evaluation program in a form similar to Table II.

TABLE I

Percent Occurrence*

I.	GROUND CONDITIONS	1.5
(a)	Rapid increase of rpm on ground to quickly engage clutch	0.5
(b)	Taxiing with full cyclic control	.5
(c)	Jump takeoff	(1)
(d)	Ground-air-ground cycle	(1)
II.	HOVERING	2.0
(a)	Steady hovering	.5
(b)	Lateral reversal	.5
(c)	Longitudinal reversal	.5
(d)	Rudder reversal	.5
III.	FORWARD FLIGHT-POWER ON	87.5
(a)	Level flight - 20% V_{NE}	1.0
(b)	Level flight - 40% V_{NE}	3.0
(c)	Level flight - 60% V_{NE}	18.0
(d)	Level flight - 80% V_{NE}	25.0
(e)	Maximum level flight (but not greater than V_{NE})	15.0
(f)	V_{NE}	3.0
(g)	111% V_{NE}	.5
(h)	Right turns - 30 60 90% V_{NE}	(2) 3.0
(i)	Left turns - 30, 60, 90% V_{NE}	(2) 3.0
(j)	Climb (Takeoff power)	2.0
(k)	Climb (Max, continuous power)	4.0
(l)	Change to autorotation from power-on-flight - 30, 60, 90% V_{NE}	1.5
(m)	Partial power descent (including condition of zero flow through rotor)	2.0
(n)	Cyclic and collective pull-up from level flight	(2)
(o)	Pushovers	(2)
(p)	Gusts	(2)
(q)	Quick stops	(2)
(r)	Flares	(2)
(s)	Lateral reversals at V_H	.5
(t)	Longitudinal reversals at V_H	.5
(u)	Rudder reversals at V_H	.5
(v)	Landing approach	3.0
(x)	Sideward flight	.5
(y)	Rearward flight	.5
IV.	AUTOROTATION - POWER OFF	9.0
(a)	Steady forward flight	2.0
(b)	Rapid power recovery from autorotational flight	.5
(c)	Right turns - 30, 60, 90% V_{NE}	(2)1.0
(d)	Left turns - 30, 60, 90% V_{NE}	(2)1.0
(e)	Lateral reversals	.5
(f)	Longitudinal reversals	.5
(g)	Rudder reversals	.5
(h)	Cyclic and collective pull-ups	(2)

(i)	Landing approach	2.0
(j)	Flares	(2)
		100.0

(1) One flight every 10 minutes with less frequent rotor stops.

(2) A vertical load factor frequency curve should be developed that is representative of the more critical types of operation. The time spent in each turn should be adjusted to give the specified per cent of occurrence.

NOTE: Extensive adjustment to this table is normally required to closely approximate the rotorcraft under evaluation.

TABLE II

DETERMINATION OF SERVICE LIFE
(Sample Calculation)

1	2	3	4	5	6	7
Flight Condition Table I	Percent Occurrence Table I	Critical Steady Stress	Critical Oscillatory Stress	Cycles Oscillatory Stress	Cycles to Failure Figure V	Percent of Life Used per hour
I(a)----	0.5	Level A	1900	23,200 Cycles/Hr.	--------	--------
(b)----	.5		2100		--------	--------
(c)----	.5		2300		--------	--------
II(a)----	.5		2800		--------	--------
(b)----	.5		9800		3,000,000	0.00232
(c)----	.5		10500		3,200,000	.00362
(d)----	.5		3400		--------	--------
III(a)----	1.0		6300		--------	--------
(b)----	3.0		3100		--------	--------
(c)----	18.0		7700		--------	--------
(d)----	25.0		8100		--------	--------
(e)----	15.0		8350		16,000,000	0.02173
(f)----	3.0		8200		8,400,000	.00522
(g)----	.5		9100		7,400,000	.00157
(h)----	3.0		11200		2,400,000	.02900
(i)----	3.0		11400		2,250,000	.03093
(j)----	2.0		10900		2,700,000	.01719
(k)----	4.0		9900		4,100,000	.02203
(l)----	1.5		7800		---------	--------
(m)----	2.0		8700		---------	--------
(n)----	1.0		9700		4,800,000	.00483
(o)----	.5		7800		---------	--------
(p)----	.5		7000		---------	--------
(q)----	.5		7300		---------	--------
(r)----	3.0		6700		---------	--------
(s)----	.5		6700		---------	--------
(t)----	.5		7900		---------	--------
IIV(a)----	2.0		7100		---------	--------
(b)----	.5		9700		4,800,000	.00242
(c)----	1.0		9300		6,300,000	.00383
(d)----	1.0		9900		4,100,000	.00586
(e)----	.5		6800		---------	--------
(f)----	.5		6100		---------	--------
(g)----	.5		5900		---------	--------
(h)----	1.0		7600		---------	--------
(j)----	2.0		7900		---------	--------
TOTALS	100.0					.15289

37612