In-Flight Computation of Helicopter Transmission Fatigue Life Expenditure

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Estimates of the safe fatigue life of critical helicopter transmission components may be made if in-service load data together with component fatigue data are available. Instrumentation has been developed to provide in-flight computation and indication of the current values of fatigue life expended for critical gears in single- or twin-engine helicopter transmission systems. In addition, basic transmission load data in the form of totalized times spent in a number of contiguous torque bands are continually updated and stored during flight. The basic load data together with values of life expenditure for critical gears for the current flight can be automatically printed out after flight. This development opens the way towards fatigue life monitoring of individual transmissions.

Nomenclature

A, B = gear fatigue curve shape constants
f = torque readings per second
G = torque bandwidth
K_1 = constant referred to hypothetical torque load spectrum 1
K_2 = constant referred to hypothetical torque load spectrum 2
N = gear endurance (used for cycles to failure and "safe" cycles)
N_i = safe cycles corresponding to torque band T_i
Q = proportion of operating time torque T is exceeded
r = gear rotational speed
t = elapsed time
T = torque corresponding to failure at N cycles
T_i = specified torque band
T_{EF} = highest torque for infinite fatigue life (corresponding to endurance limit)
T_{ES} = factored highest torque for infinite fatigue life
T_H = maximum torque transmitted during gear operation
u = life usage increment per torque reading
U = fraction of safe fatigue life expended
v_i = rationalized mean amplitude of noise signal element i
W = mean rate of fatigue life usage
X = normalized amount by which torque endurance limit is exceeded
α = proportional change in indicated rate of fatigue life usage due to measurement error
β = square wave noise amplitude factor
ϵ = fractional torque measurement error referred to T_{ES}
θ = instantaneous phase of alternating signal component
σ = maximum value of X achieved
τ = total operating time of gear
ω = radian frequency

Introduction

COMPONENTS subject to high-amplitude fluctuating loads may eventually undergo fatigue failure if a sufficiently high number of load cycles is applied before the component is replaced. For the simple arrangement of two gears in mesh, each tooth will experience one load cycle per revolution. If gears are not subject to high loads there will be no tendency for fatigue failure to occur and life will then be limited by wear considerations.

Usually the lives of certain critical components in helicopter transmission systems are limited by fatigue considerations. Prediction of the safe life of such components is of particular concern to aircraft operators, as components must be replaced before failure occurs. Valid estimates of the safe life of helicopter transmission systems, apart from being vital for operational safety, are necessary for the formulation of an economic maintenance policy.

For fatigue life estimation of transmission components, two basic sets of data are required. First, gear fatigue data in the form of the number of cycles to failure (endurance) as a function of stress level (*S-N* curve) are required together with a suitable stress or safety factor to define a "safe" curve. The *S-N* curve absorbs the effects of component geometry and materials properties. Second, load spectra (giving proportion of flying time above various torque levels) must be available for the transmission under normal operating conditions. For helicopter transmissions, computation of load cycles is simplified because gear rotational speed can be assumed to be constant.

The requisite gear fatigue data can be established by testing gears to failure. Normally such data are acquired by the transmission manufacturers. On the other hand load spectra for a given helicopter may vary significantly for different operational roles and hence it is usually necessary for each operator to measure his own spectra.

It is current practice to replace critical helicopter transmission components after a specified operating time. To establish the safe replacement interval with confidence in such instances it is necessary to measure the load spectra which apply for various types of sortie. If the sortie pattern for the whole fleet is known, statistical methods[1,2] can be used to specify a mean torque spectrum which can be used to compute the safe replacement interval.

With the ever increasing cost of ownership there is a growing impetus in the aircraft industry to look towards a maintenance policy which involves replacement of gears after service corresponding to the estimated fatigue life. The relevant time in service will vary from aircraft to aircraft according to missions flown. To implement such a policy it is necessary to monitor fatigue life expenditure of individual transmissions. In general, an increase in the average

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replacement life of critical components should be achieved, since the conservatism inherent in basing life on generalized mission spectra covering the full fleet is removed.

Recent advances in microcomputer technology have made it possible to incorporate the gear fatigue data in read-only-memory programs and to compute and indicate fatigue life expenditure of gears during flight. Third generation instrumentation[3] developed for use in fatigue life studies of transmission components exploits the new technology to compute and indicate expenditure for up to four gears in the main rotor gear box for the Sea King Mk 50 helicopter operated by the Royal Australian Navy (RAN). In addition, basic load data are stored during flight and automatically printed out after flight.

Analytical Representation of Gear Fatigue Data

The shape of the *S-N* curve (or more conveniently the *T-N* curve, where stress S is proportional to applied torque T) is established by applying curve fitting techniques to a sufficiently large sample of data derived from tests to failure (Fig. 1) of relevant gears.

For the Sea King studies the curve shape used was based on previous failure tests, conducted by the manufacturer at various constant torque levels, on a batch of 47 bevel gears of similar material for the tail gear box in the Wessex helicopter. The *T-N* curve shape in this case was established by the manufacturer by drawing a best-fit curve through the medians of the logarithms of the endurances at each test torque level. The curve was given the mathematical form

$$T=T_{EF}(1+AN^{-1/B}) \tag{1}$$

Values of A and B, and a nominal value for T_{EF} were computed using three points on this best-fit curve. Estimated values for parameters A and B, 48.9 and 2.58, respectively, are used in this paper. Using experimental values of T and N corresponding to each of the 47 failures, and the values of A and B estimated as above, the mean and standard deviation of the computed T_{EF} values were established. A coefficient of variation (ratio of standard deviation to mean) of 0.068 resulted. The distribution of T_{EF} was only moderately skewed and, within the restraints of the limited number of test points, approximated the Normal distribution ($\alpha\chi^2$ test applied to the test data shows that at 95% confidence level the data are consistent with the hypothesis that they conform to a Normal distribution). Values of T_{EF} for critical Sea King gears were estimated from a smaller number of failure tests in a similar manner to that used for the Wessex tail gears.

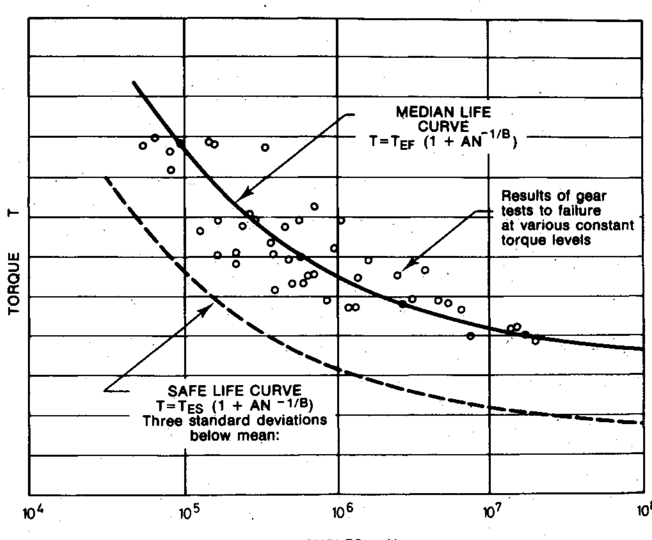

Fig. 1 Graphical presentation of gear fatigue data.

A "safe" working relationship may be derived by applying a suitable safety factor to the computed mean endurance limit. A value of endurance limit T_{ES} three standard deviations below the mean is typically used. Analytically the safe curve may be expressed as

$$T=T_{ES}(1+AN^{-1/B}) \tag{2}$$

Other variations[3] of the safe curve relationship can be applied, but only that described by Eq. (2) will be considered here.

The three-standard-deviation safe curve illustrated in Fig. 1 represents a stress or safety factor (T_{EF}/T_{ES}) of about 1.25. At this value of stress factor the probability of a premature gear failure occurring is approximately one 1:1000 (as predicted by the Normal distribution).

The endurance relationship may be expressed generally in normalized form as

$$X=T/T_{ES}-1=AN^{-1/B} \tag{3}$$

Consider the torque range to be divided into a number of narrow bands of mean torque value T_1, T_2,..., etc. If the gear is subject to n_1 cycles within the T_1 band, n_2 cycles within the T_2 band, etc., then it is normal to apply Miner's summation law[4] to quantify the cumulative damage:

$$U=\frac{n_1}{N_1}+\frac{n_2}{N_2}+\ldots+\frac{n_K}{N_K}=\sum_{i=1}^{K}\frac{n_i}{N_i} \tag{4}$$

When $U=1$ (or any other selected value if a further factor is to be applied), safe life is considered totally expended.

When load measurements are made on a sample of helicopters with the aim of predicting a safe replacement interval for critical transmission components using statistical methods, it is first necessary to determine the applicable torque load spectrum. Totalized times spent within each of a number of contiguous torque bands may be measured during flight and the results may be plotted as in Fig. 2. Although no fatigue damage occurs at torque levels below the endurance limit T_{ES}, it is necessary to measure totalized times spent in bands below that limit so that the shape of the torque spectrum curve may be established by curve fitting techniques.

The proportion of time for which torque has value between T and $T+\Delta T$ may be expressed as ΔQ. At torque T the rate of life usage is given by r/N. The fatigue life usage ΔU which results due to torque developed in the range T to $T+\Delta T$ is given by

$$\Delta U=\tau r\frac{\Delta Q}{N} \tag{5}$$

where N is the mean safe cycles within the torque interval.

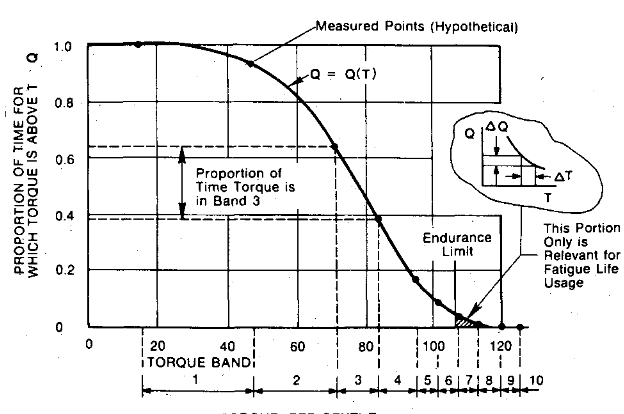

Fig. 2 Torque load spectrum.

By summing contributions to fatigue life usage resulting from torque in excess of the endurance value T_{ES}, the current value of life fraction usage U can be computed:

$$U = \tau r \sum_{T=T_{ES}}^{\infty} \frac{\Delta Q}{N} \tag{6}$$

The above summation can be performed numerically by adding contributions for specified torque increments (say 1% rated torque) above the endurance limit.

If Q is known or can be approximated analytically as a function of T, the above summation may be expressed more conveniently by the following integral:

$$U = \tau r \int_{T_{ES}}^{\infty} -\frac{dQ}{dT}\frac{1}{N}\,dT \tag{7}$$

The minus sign ahead of dQ/dT is necessary since dQ/dT is always negative (or zero).

The safe replacement interval τ_R is the value of τ for $U=1$ (or other nominated value). Thus

$$\tau_R = 1\Big/\left\{r \sum_{T=T_{ES}}^{\infty} \frac{\Delta Q}{N}\right\} \tag{8}$$

$$\tau_R = 1\Big/\left\{r \int_{T_{ES}}^{\infty} -\frac{dQ}{dT}\frac{1}{N}\,dT\right\} \tag{9}$$

Transmission Torque Measurement

Requirements for Single- and Twin-Engine Helicopters

Engine torque is a parameter of vital significance to helicopter operators and is always displayed via cockpit meters to the pilots. Usually it is feasible to develop very high levels of torque, and pilots are required to maintain torque below prescribed limits. Because torque has to be displayed to pilots, a method of torque sensing is always included as part of the helicopter transmission design. This is convenient for life estimation purposes as torque is a very difficult quantity to sense as an "afterthought." However, fatigue monitoring places high demands on torque measurement accuracy which may conflict with the less demanding requirement for pilot indication.

Normally torque is expressed in normalized form as percent rated value, 100% rated torque being a somewhat arbitrary figure which can be exceeded. For single-engine helicopters only one torque needs to be sensed and only one torque spectrum needs to be measured to estimate the safe lives of all gears in the main rotor gear box. For twin-engine helicopters, three torque spectra are of interest: 1) the spectrum of torque developed by port engine for gears at the output of that engine; 2) the spectrum of torque developed by starboard engine for gears at the output of that engine; and 3) the spectrum of total torque for gears in the main rotor gear box which transmit combined torque.

The spectra may be established by measuring output torque for each engine and summing the torque values to give total torque required for spectrum 3. To allow the constant speed property of helicopter transmission systems to be used in the fatigue calculations, it is essential that torque be sensed in the constant speed portion of the propulsion system.

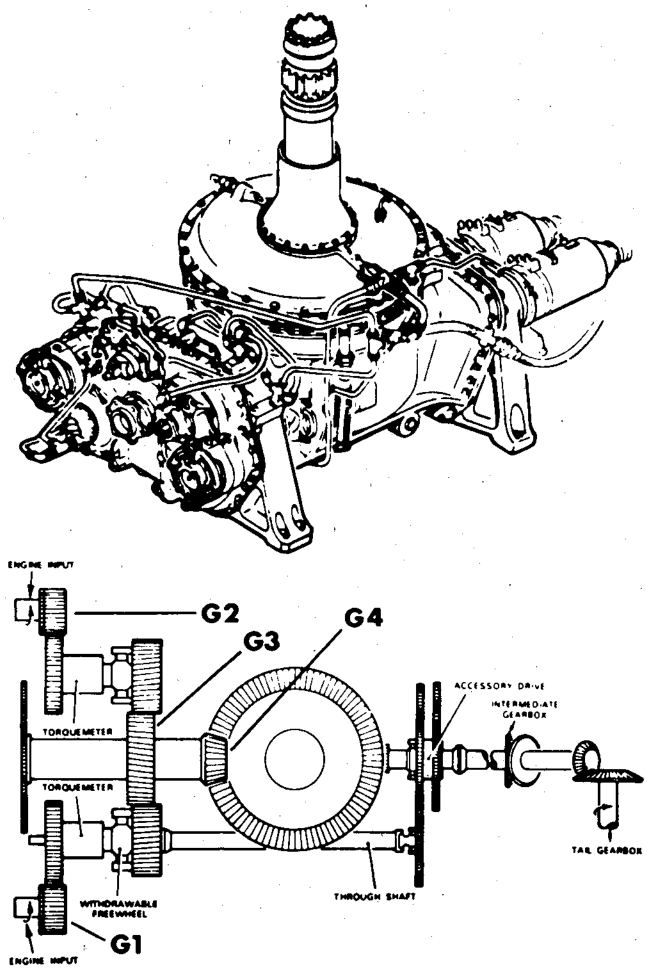

Fig. 3 Sea King transmission system.

Transmission torque can be sensed using hydraulic or other techniques. In particular the twin-engine Sea King employs hydraulic systems to sense the torque developed by each engine.

Fatigue life studies are currently proceeding in relation to the RAN Sea King helicopter transmission system (Fig. 3).

Lives of four gears as indicated in Table 1 are being monitored.

Individual engine torques are not necessarily equal, although it is normal practice to keep them fairly well balanced. Single-engine flying is essential during certain emergencies and indeed pilots are required to regularly practice single-engine flying. Maximum permissible torque for single-engine operation is different from that for twin-engine operation.

Fatigue damage will occur only at very high torque levels. For gears G1-G3, fatigue damage can occur only during single-engine operation when permissible individual engine torque is high. Fatigue damage to G4 can occur only during twin-engine operation.

Table 1 Sea King gears under study

Gear	Description	Rotational speed, 103% rated value	Torque loading
G1	Spur pinion	19,525	Port engine torque
G2	Spur pinion	19,525	Starboard engine torque
G3	Helical gear (summing)	3,291	Port/starboard engine torque[a]
G4	Spiral bevel pinion	3,291	Total torque

[a] Gear receives two load applications per revolution port and starboard torques in sequence.

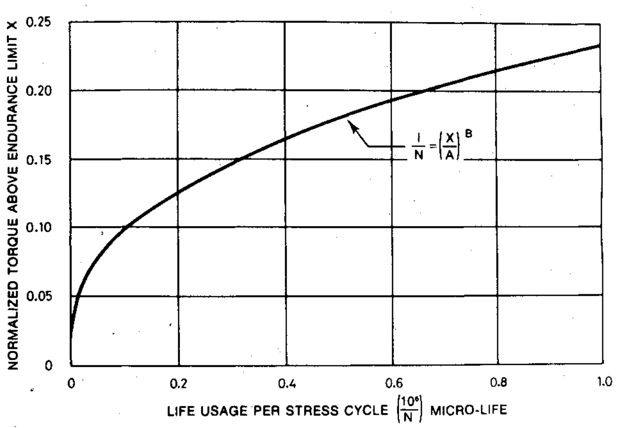

Fig. 4 Rate of life usage.

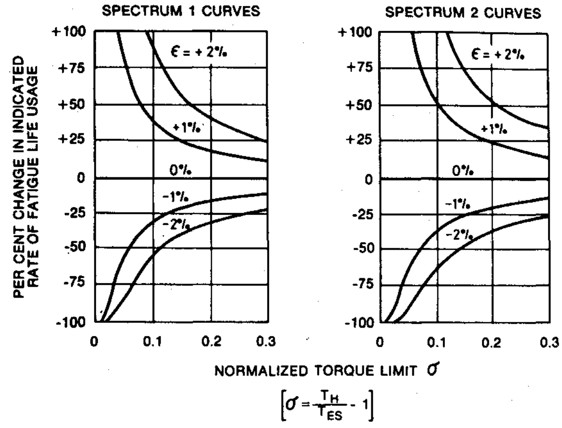

Fig. 6 Effect of absolute measurement error ϵT_{ES}.

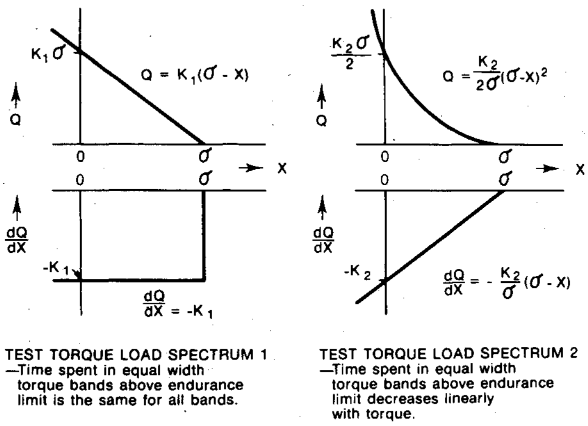

Fig. 5 Hypothetical torque load spectra.

Demands of the Measuring System

Effect of Absolute Measurement Inaccuracy

The rate of fatigue life usage increases nonlinearly as a function of applied torque in excess of the endurance limit. In Fig. 4 the life usage per stress cycle (equal to the reciprocal of life cycles N) is plotted on a linear scale as a function of applied torque for the fatigue relationship of Eq. (3). Values of constants A and B used for this graph are those used for the Sea King studies. Multiplying the life usage per stress cycle by gear rotational speed yields the instantaneous rate of fatigue life usage. It is of interest to examine the amount by which the measured rate of fatigue life usage (henceforth to be called the "indicated" rate of fatigue life usage) is changed owing to absolute or mean torque measurement inaccuracies. A simple relationship does not apply, as the difference in rate will be a function of the torque load spectrum applicable to the helicopter transmission system. However, the order of the difference to be expected can be appreciated by assessing the differences which apply when some very simple torque spectra are used as test examples.

Two simple torque spectra (Fig. 5) which can be readily represented analytically will be considered. For the first spectrum, the time spent in equal width torque bands within the torque range extending from just below the endurance limit T_{ES} to a value $(1+\sigma)\,T_{ES}$ is the same for all bands. For the second spectrum, the time spent in equal width torque bands within the torque range extending from just below the endurance limit T_{ES} to a value $(1+\sigma)T_{ES}$ decreases linearly to zero at the high torque limit.

The following relationships apply:

$$T_H = (1+\sigma)\,T_{ES} \qquad \sigma = T_H/T_{ES} - 1 \tag{10}$$

Analytically the hypothetical spectra defined above can be represented by the following relationships: spectrum 1,

$$\frac{dQ}{dX} = -K_1 \tag{11}$$

and spectrum 2,

$$\frac{dQ}{dX} = -\frac{K_2}{\sigma}(\sigma - X) \tag{12}$$

From Eq. (7) the mean rate of fatigue life usage W may be written

$$W = \frac{U}{\tau}$$

$$= r\int_0^{\sigma} -\frac{dQ}{dX}\frac{1}{N}dX \tag{13}$$

If true torque T is read as $T+\epsilon T_{ES}$ then the maximum torque will appear to shift from $T=(1+\sigma)T_{ES}$ to $T=(1+\sigma+\epsilon)\,T_{ES}$ or from $X=\sigma$ to $X=\sigma+\epsilon$.

The proportional change α in the indicated rate of fatigue life usage due to absolute measurement inaccuracy is given by

$$\alpha = (W_{\sigma+\epsilon} - W_{\sigma})/W_{\sigma} = W_{\sigma+\epsilon}/W_{\sigma} - 1 \tag{14}$$

where suffixes indicate the upper limit of integration for variable X in Eq. (13).

Using the relationships presented above, expressions for proportional change α due to absolute torque measurement error ϵ can be derived[3] for both spectra considered.

For torque spectrum 1,

$$\alpha = (1+\epsilon/\sigma)^{B+1} - 1 \tag{15}$$

and for torque spectrum 2,

$$\alpha = (1+\epsilon/\sigma)^{B+2} - 1 \tag{16}$$

The change α, in the indicated rate of fatigue life usage due to absolute measurement inaccuracy for the torque spectra considered is presented graphically in Fig. 6.

The graphs serve to indicate that small errors in torque measurement result in very significant changes in the indicated rate of fatigue life usage. For each case the resultant changes are very high for low values of σ. For any given gear train subject to high torque loading the normalized endurance limit will be lowest for the most critical gear and hence that gear will have the highest value of σ. In practical terms this means that variations in indicated rate of fatigue life usage resulting from absolute torque measurement inaccuracies will be lowest for the most critical gears which have the shortest fatigue life.

It is normal for the actual torque spectrum for a helicopter transmission system to be characterized by progressively less time being spent within any specified width torque band as the torque is increased above the endurance limit T_{ES}. Thus the absolute torque measurement error curves for torque spectrum 1 represent unrealistically favorable conditions, and in practice the errors will be closer to those given using torque spectrum 2.

For the RAN Sea King, torque excursions above the endurance limit are rare. Under worst conditions the value of σ is not likely to exceed 0.15 for the most critical gear. According to the graphs of Fig. 6 a 1% torque measurement error is likely to produce a change in excess of 25% in the indicated rate of fatigue life usage at that value of σ. Consequently there is a need for fairly precise measurement of torque so that results obtained for different gear boxes and possibly by different operators can be meaningfully compared. Furthermore, a knowledge of the accuracy of the torque measurement is required so that an adequate factor can be applied to the total life summation limit, especially when the fatigue life usage for individual transmissions is monitored.

Alternatively, if the maximum measurement inaccuracy has been established, the working endurance limit can be lowered to take account of the measurement system inaccuracy. Under such conditions higher accuracy will obviously yield higher average values of life and thus reduce maintenance costs if a policy of replacement of individual transmission components after service corresponding to the indicated fatigue life is adopted.

Significant inaccuracies in the measurement of transmission torque (after appropriate sensing via the aircraft torque measuring system) would mean, in effect, that two instruments monitoring the same torque data would be likely to indicate very different amounts of fatigue expenditure and the readings would have to be derated accordingly. However, it must be stressed that the inherent uncertainties of the estimated safe *T-N* relationship together with the inaccuracies of the hydraulic torque sensing system may have far greater bearing on the attainable gear life.

The analysis of this section can be applied[3] to assess the effect of a stress factor value change which will reflect a change in σ.

Any error in the estimation of the number of stress cycles which occur per unit time will be reflected as a change in the indicated rate of fatigue life usage. Such an error could arise from two causes: 1) an absolute error in the estimation of helicopter gear rotational speed, and 2) an absolute timing inaccuracy in the measuring system.

However such errors reflect only a linear change in the indicated rate of fatigue life usage, and hence the accuracy demands for gear rotational speed and for torque duration timing are not stringent.

Effect of Noise

Any unfiltered alternating signal components which do not reflect true torque variations will be referred to as "noise." Such components will have an effect on the indicated mean rate of fatigue life usage. Because the rate of fatigue life usage increases with increasing torque, the effect of noise (which by definition is assumed to have a mean value of zero) will always be to increase the indicated rate of fatigue life usage to a value above that which applies when no noise is present. It follows that any estimates of safe life based on torque spectra established by "noisy" measuring systems will be conservative.

The net effect of noise on the indicated mean rate of fatigue life usage will be a function of the waveshape and amplitude of the noise, and of the torque spectrum which applies. To obtain some analytical assessment of the order of change in rate to be expected as a result of the presence of noise the analysis of the previous section relating to mean torque measurement errors can be used to advantage. Hypothetical torque spectra (Fig. 5) as used previously will be considered again here.

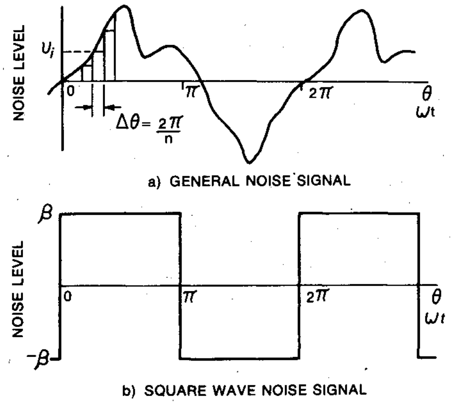

Fig. 7 Noise signal representation.

Any noise signal which is repetitive can be approximated as the sum of a number n of rectangular elements (Fig. 7a) of different amplitudes. For convenience the elements can be made of equal width $\Delta\theta$ (equal to $2\pi/n$). Consider initially just element i representing a torque deviation $v_i T_{ES}$ relative to the mean torque value. The proportion of time for which the torque deviation from the mean value falls within the range of element v_i is $(\Delta\theta)/(2\pi)$ or $1/n$. The change in the mean rate of fatigue life usage due to the element i will be $1/n$ times the change which would result if level v_i were applied continuously.

Using Eq. (14) and introducing the suffix N to indicate the change in the indicated rate of fatigue life usage due to the presence of noise, the following expression results for the element i:

$$\alpha_N = 1/n\ (W_{\sigma+v_i}/W_\sigma - 1)$$

The net effect for the composite noise signal is obtained by summing the effect of all n elements in the range $0<\theta<2\pi$.

$$\alpha_N = \frac{1}{n}\sum_{i=1}^{n}(W_{\sigma+v_i}/W_\sigma - 1) \tag{17}$$

The above relationship may be expressed in integral form:

$$\alpha_N = \frac{1}{2\pi}\int_0^{2\pi}\frac{W_{\sigma+v_i}}{W_\sigma}\,d\theta - 1 \tag{18}$$

To obtain some analytical appreciation of the effect of noise, a square wave noise signal (Fig. 7b) of amplitude βT_{ES} will be considered. Such a noise waveshape, although most unlikely in practice, represents the worst case waveshape for noise of that amplitude. Using Eq. (18), the following expression results:

$$\alpha_N = 0.5\left(\frac{W_{\sigma+\beta}+W_{\sigma-\beta}}{W_\sigma} - 1\right)$$

$$= 0.5(\alpha_{\sigma,+\beta}+\alpha_{\sigma,-\beta}) \tag{19}$$

where $\alpha_{\sigma,+\beta}$ and $\alpha_{\sigma,-\beta}$ represent the values of α according to Eqs. (15) and (16) for $\epsilon = +\beta$ and $-\beta$, respectively. $\alpha_{\sigma,+\beta}$ is always positive and $\alpha_{\sigma,-\beta}$ is always negative.

For the two spectra considered in Fig. 5 the increases in the indicated rate of fatigue life usage due to square wave noise of 2% amplitude for σ equal to 0.15 and for spectrums 1 and 2, respectively, are 8.3 and 14.7%.

Hence superimposed noise transferred through the measuring system can produce a significant increase in the indicated rate of fatigue life usage. It is therefore essential that alternating components which do not reflect true torque should be attenuated as much as possible in the measuring system.

Dynamic Response Requirements

In all helicopters there will be alternating or vibratory components of true torque which are transferred through the transmission system. Attenuation of such components, either due to limitations in the response of the measuring system or due to deliberate filtering, will cause the indicated mean rate of fatigue life usage to be lower than the true value. Quantitatively the reduction in rate can be assessed in exactly the same way that the increase in rate due to superimposed noise was assessed in the previous section.

A component of torque at blade passing frequency (equal to rotor speed multiplied by number of blades) will always be present in helicopter transmission systems. For example, the passing frequency for Sea King is about 17 Hz. Instrumentation described herein passes the fundamental component at this frequency without attenuation, but to minimize the transfer of extraneous noise through the system, low-pass filters are used to attenuate any signals above this frequency.

From the above discussion it follows that a system bandwidth typically of the order of 20 Hz is required for instrumentation used in helicopter transmission fatigue studies. It is to be emphasized that the dynamic response need arises primarily because of the nonlinear relationship between torque and rate of fatigue life usage. If a linear relationship were to apply, the changes in fatigue life usage rate due to positive and negative excursions of the alternating torque component would tend to cancel over a long period, thus yielding a value of life usage which would be the same as that which would result if the alternating component were completely removed.

To digitize the measured value of torque some form of analog to digital conversion with associated signal sampling must be used. If the alternating component amplitude at any given level of mean torque tends to remain fairly constant over the life of the gear then a sampling frequency much lower than the highest component frequency of interest can be used.

The minimum acceptable value of sampling frequency cannot be clearly defined; it will be a function of the torque spectrum applicable and of the requirement in relation to torque transient response. Generally, if much of the gear life usage occurs owing to relatively few transient excursions to high torque levels for which the usage rate becomes very high, there may be insufficient samples to obtain an accurate indication of usage if a low sampling rate is used. Clearly there is no merit in deliberately reducing the sampling frequency if a higher value can be accommodated without penalty. Broadly it may be stated that a sampling frequency of 100 Hz should accommodate any application regardless of the torque spectrum applicable and of the nature of high-level transient loads which may occur in service. On the other hand, a sampling frequency of 1 Hz, say, would place significant restrictions on the torque spectra which could be accommodated and may be too low to allow high-level transients, which may account for significant fatigue life usage, to be followed.

Considerations Relating to Measurement Resolution

To exploit the advantages of modern digital circuit technology and computing it is essential that the value of torque measured be digitized using a suitable analog-to-digital converter (ADC). If the torque corresponding to a particular ADC reading i is T_i, then the system resolution can be defined as $T_{i+1} - T_i$, where i is an integer. If an ADC reading of i is obtained, then all that can be deduced is that the true torque value T lies in the range $T_i < T < T_{i+1}$, assuming that the ADC is aligned such that a true torque input T_i results in the ADC output being just on the switching point $(i-1) \rightarrow i$. It follows that on the average the torque will be underestimated by an amount equal to $0.5(T_{i+1} - T_i)$ (i.e., half the resolution). Thus the magnitude of the mean underestimate is proportional to the system measurement resolution.

To establish the relevant torque spectra and estimate safe replacement intervals for critical transmission components using a ground based digital computer, it is essential that sufficient points be obtained on the Q-T graph (Fig. 2) so that a curve can be accurately fitted to the measurement points. Once an adequate resolution to satisfy this condition has been established, no improvement in measurement precision can be provided by increasing the resolution. However, any lack of precision in establishing the band changeover points will result in errors as discussed earlier. Hence for torque spectrum indication, high measurement precision is necessary but low resolution is quite satisfactory.

For fatigue life usage computation during flight, much higher resolution is necessary if the complex Q-T curve fitting techniques which can be readily applied in a ground based analysis are to be avoided. However, it is essential that account be taken of the mean underestimate which is a fundamental characteristic of the conversion process. For a narrow torque band, it can be assumed that, when considered over a long period, there will be equal probability that the true torque have any value within the band. Consider the band bounded by T_i and T_{i+1}, which is applicable for an ADC reading of i. A better estimate for the fatigue life usage u_i per sample is obtained by averaging as follows:

$$u_i = \frac{1}{T_{i+1} - T_i} \int_{T_i}^{T_{i+1}} u \, \mathrm{d}T$$

$$= \frac{r}{f(T_{i+1} - T_i)} \int_{T_i}^{T_{i+1}} \frac{1}{N} \, \mathrm{d}T \tag{20}$$

Because of the nonlinearity of the T-N relationship, the above provides a better estimate of fatigue life usage than a computation based on mean torque $0.5(T_i + T_{i+1})$.

For a linear torque measuring system the width of all torque bands will be equal, and in such cases Eq. (20) may be rewritten:

$$u_i = \frac{r}{fG} \int_{T_i}^{T_i+G} \frac{1}{N} \, \mathrm{d}T \tag{21}$$

To take account of an endurance limit T_{ES} which has some intermediate value between T_0 and T_1, the fatigue life usage per sample for an ADC reading corresponding to T_0 is computed as follows:

$$u_0 = \frac{r}{f(T_1 - T_0)} \int_{T_{ES}}^{T_1} \frac{1}{N} \, \mathrm{d}T \tag{22}$$

The general relationships developed in this section may be used for computing life usage from torque readings taken using a torque measuring system with finite resolution.

Instrumentation Developed for Real Time Indication of Fatigue Life Usage

General Outline

A Transmission Fatigue Life Usage Indicator (Fig. 8) capable of real time computation and indication of fatigue life usage for the four selected gears (Fig. 3) and of storing data for establishing the three requisite torque spectra for the RAN Sea King was test flown and commissioned in July 1980.

A commercially available printer slightly modified for airborne applications is used to print out basic torque

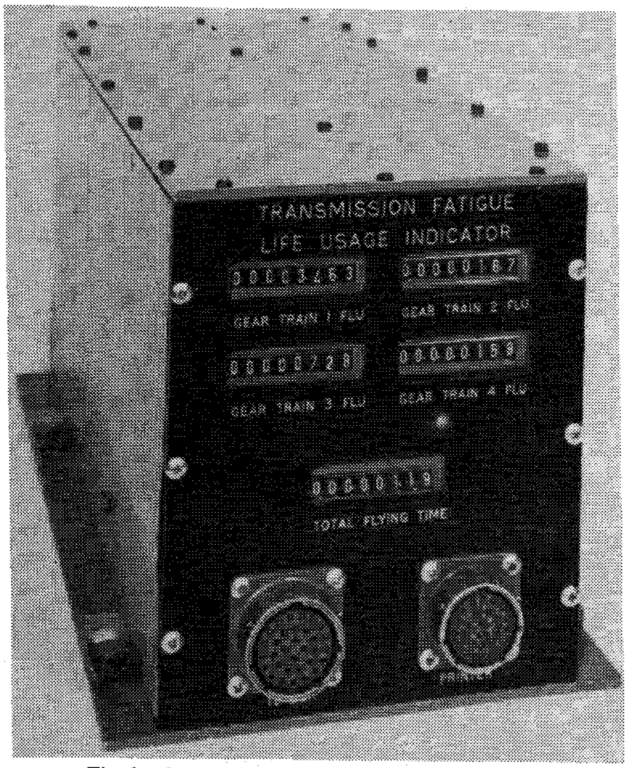

Fig. 8 Transmission fatigue life usage indicator.

SEA KING TRMLIF 01

BAND	DURN (SEC)
PORT-0	5694.09
PORT-1	3007.40
PORT-2	0396.61
PORT-3	0050.74
PORT-4	0009.33
PORT-5	0004.10
PORT-6	0000.00
PORT-7	0000.00
PORT-8	0000.00
PORT-9	0000.00
STBD-0	4956.44
STBD-1	3204.31
STBD-2	0310.77
STBD-3	0005.09
STBD-4	0000.00
STBD-5	0000.00
STBD-6	0000.00
STBD-7	0000.00
STBD-8	0000.00
STBD-9	0000.00
TOTL-0	5321.02
TOTL-1	4108.30
TOTL-2	1856.32
TOTL-3	0202.13
TOTL-4	0001.23
TOTL-5	0000.06
TOTL-6	0000.00
TOTL-7	0000.00
TOTL-8	0000.00
TOTL-9	0000.00

GEAR	FLU
G1	00012
G2	00000
G3	00017
G4	00000

TOTAL FLYING TIME

11490 SEC
3.1914 HR

AIRCRAFT NO. N16 -118
DATE: 9 - 7 - 80
TAKE-OFF TIME: 1115
SORTIE TYPE: G.F.P.

COUNTER READINGS

GEAR 1: 2876
GEAR 2: 2798
GEAR 3: 7014
GEAR 4: 1207
TFTIME: 19902

Fig. 9 Sample postflight printout.

spectrum data (totalized times spent in each of ten selected torque bands for each of the three requisite spectra). In addition, life expenditure for the four selected gears and the duration of the current flight are printed out at the end of the flight. Since printing takes place only at the termination of the flight, it is not essential that the printer be carried during flight. However the present instrumentation does not include nonvolatile memory or battery backup, so it is essential that the data be printed before power-down.

For the Sea King installation the printer is carried during flight and printouts are normally initiated automatically after the first engine is shut down. However aircrew can manually initiate a printout if desired. The sample printout (Fig. 9) provides details of the data printed and includes information written on the printout by the aircrew.

High-performance strain gage pressure transducers have been installed in the torquemeter pressure lines in the transmission compartments for the Sea King application. These transducers convert the pressure signals, which are proportional to torque developed by the respective engines, to analog voltage signals.

Significant long-term shifts have occurred for the initial set of transducers after a relatively short period of service in the adverse environment of the Sea King transmission compartment. Such shifts are of considerable concern because of the need for fairly precise torque/pressure measurement. Evaluation of further strain gage transducers is proceeding. Other transducer types such as the vibrating cylinder, vibrating wire, or the more expensive quartz resonator[5] are likely to provide better long-term stability.

Principle of Operation

A functional block schema for the Transmission Fatigue Life Usage Indicator is drawn in Fig. 10.

The low-level outputs from the pressure transducers are amplified and filtered (with low-pass filters having a cutoff

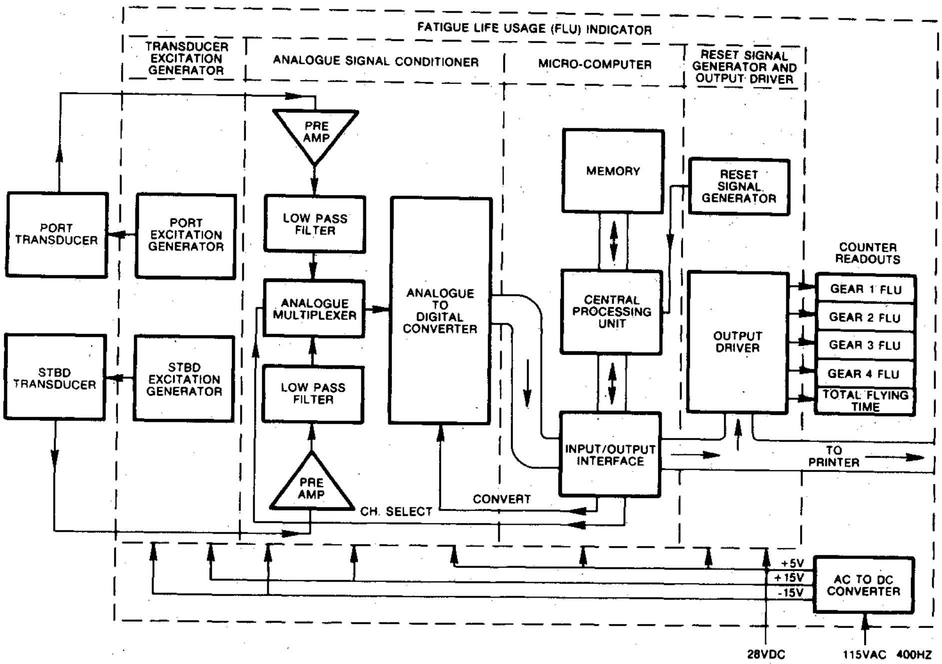

Fig. 10 Block schema of fatigue life usage indicator.

frequency of about 22 Hz) to remove high-frequency noise components. The amplifier outputs are passed to a solid state analog multiplexing switch. Conversion of the selected amplifier output to digital form is provided by an analog-to-digital converter.

System operation is controlled by the microprocessor and associated circuits which issue conversion commands and allow the two torque inputs to be selected in sequence. Operating programs are stored in programmable read-only-memories.

To enable computation of gear fatigue life usage, the gear fatigue relationships are precomputed and entered in look-up tables in the airborne program.

Both port and starboard engine torques are read 100 times per second. During the 10-ms repetition interval, the following actions are performed by the microcomputing system.

1) Fatigue life usage (if any) for each of the four gears under examination is computed and the life usage increment is added to the contents of associated summing stores.

2) The contents of each of the four summing stores are examined and if the respective values are in excess of one microlife unit of usage, the count value of the associated electromechanical counter (Fig. 8) is advanced by 1, and one microlife unit of usage is subtracted from the contents of the relevant summing store.

3) Flying time is advanced by 0.01 s and for each second of totalized time accumulated the total flying time counter reading is advanced by 1.

4) Torque bands corresponding to the current values of torque are determined for the requisite three spectra and 0.01 s is added to appropriate elements of arrays used to store the basic torque spectrum data.

When a request-to-print signal is received at the end of the flight the above operations cease and the relevant stored data, after suitable conversion, are transferred to the printer. Transfer of the data, as shown in Fig. 9, takes about 60 s.

The instantaneous rate of fatigue life usage may exceed the capability of the electromechanical counters. Ample storage allows any residual life usage to be transferred as soon as the torque level falls.

Concluding Remarks

Conservative maintenance practices which must be applied if the load history is not known lead to reduced replacement life for critical transmission components. Reduced maintenance costs can result if individual transmissions are monitored with the aid of suitable instrumentation.

Values of safe fatigue life predicted by monitoring instrumentation must be suitably factored to take account of worst-case torque measurement errors. Fortunately the factor will be closest to unity for the most critical gear with the lowest fatigue life. Unless the gear fatigue relationships can be established with confidence and reasonable torque measurement accuracy achieved, little credibility can be attached to safe life predictions.

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