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An Evaluation on Damping Property of Non-linear Shimmy Damper and Equivalent Linearization Methods

Nanchang Aircraft Manufacturing Co., Su Kaixin

Abstract

In this paper the formula of requisite linear damping for overcoming landing-gear shimmy ~~is derived~~ according to the point-contact theory. However, the damping property of the hydraulic shimmy damper is mostly non-linear. Two methods are presented to treat this case: the tangent method and the minimum equivalent linear damping method. Their correlation is also found. It is demonstrated by computation that both methods are effective and reliable.

I. Simplification of Theoretical Equation for Single Wheel Shimmy without Inclination

Based on the following assumptions that (1) the support (or the directional axis) is vertical and rigid, (2) a static

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rigidity model is used to treat tire distortion and there is pure rotation and no skidding, (3) the entire system is linear, and (4) it is moving at a uniform speed and the vertical load is also constant, then the equation of motion for shimmy is:

$$\left.\begin{aligned} & IV^2\frac{d^2\theta}{ds^2}+hV\frac{d\theta}{ds}-a\lambda t-b\varphi=0 \\ & \frac{d(\theta+\varphi)}{ds}-\alpha\lambda+\beta\varphi=0 \\ & t\frac{d\theta}{ds}+\frac{d\lambda}{ds}+(\theta+\varphi)=0 \end{aligned}\right\} \quad (1)$$

where

I - rotational moment of inertia of the shimmy part with respect to the directional axis,

t - distance of stability,

h - damping moment coefficient centered around the directional axis,

V - speed,

s - rolling distance along the direction of motion,

θ - angle between the diametral plane after the wheel yawing and that of the wheel in neutral position,

α, β - motion constants of the wheel,

a, b - lateral rigidity and torsion rigidity of the wheel,

φ, λ — variables introduced in the point-contact theory to describe torsion distortion and lateral distortion of the tire. φ is the yawing angle of the tangent at the contact center and counterclockwise is positive. λ is the lateral shift at the contact center and it is positive toward the right (see Figure 1).

Let

$$\left.\begin{aligned} \theta &= Ae^{xt} \\ \varphi &= Be^{xt} \\ \lambda &= Ce^{xt} \end{aligned}\right\} \quad (2)$$

Substitute it into equation (1) and let

$$\left.\begin{aligned} a_0 &= ab + a\beta t \\ a_1 &= abt + a\beta t^2 + hVa \\ a_2 &= hV\beta + b + at^2 + IaV^2 \\ a_3 &= IV^2\beta + hV \\ a_4 &= IV^2 \end{aligned}\right\} \quad (3)$$

We then get the following eigen function:

$$a_4x^4 + a_3x^3 + a_2x^2 + a_1x + a_0 = 0 \quad (4)$$

Based on the Routh-Hurwitz stability condition, the real part of the root of the quadratic polynominal eigen function (4) is negative when the coefficients a_0, a_1, a_2, a_3 and a_4 have the same sign as the discriminant:

$$R = a_1a_2a_3 - a_0a_3^2 - a_4a_1^2$$

In this case, the motion is stable. From equation (3) we know that a_4, a_1, a_2, a_3 and a_0 are always greater than zero. Therefore, R should also be greater than zero. To this end, the following must be satisfied:

$$a_1 a_2 a_3 > a_0 a_3^2 + a_4 a_1^2 \qquad (5)$$

By substituting (3) into (5) an inequality with the damping coefficient h as an unknown can be obtained. This inequality can be solved with respect to h. This is the formula needed to calculate the minimum damping coefficient.

$$A_3 h^3 + A_2 h^2 + A_1 h + A_0 = 0 \qquad (6)$$

where

$$\left.\begin{aligned} A_0 &= a'_1 V + a'_3 V^3 \\ A_1 &= b_0 + b_2 V^2 + b_4 V^4 \\ A_2 &= c_1 V + c_3 V^3 \\ A_3 &= d_2 V^2 \end{aligned}\right\} \qquad (7)$$

In equation (7)

$$a'_1 = -I\alpha^2 b t\left(b + \frac{\beta}{\alpha} a t\right)\left(t - \frac{\beta}{\alpha}\right)$$

$$a'_3 = I^2 \alpha^2 \beta \left(b + \frac{\beta}{\alpha} a t\right)\left(t - \frac{\beta}{\alpha}\right)$$

$$b_0 = \alpha t\left(b + \frac{\beta}{\alpha} a t\right)\left(b + a t^2\right)$$

$$b_2 = I\alpha\beta[(b+\frac{\beta}{\alpha}at)(\beta t-\frac{\beta}{\alpha}t-2)+(b+at^2)]$$
$$b_4 = I^2\alpha^2\beta$$
$$c_1 = \alpha[(b+\frac{\beta}{\alpha}at)(\beta t-1)+(at^2+b)]$$
$$c_3 = I\alpha\beta^2$$
$$d_2 = \alpha\beta \qquad (8)$$

Now, we have all the equations required to solve the linear damping coefficients which include 8 characteristic parameters, where I, h and t are structural parameters, a, b, α and β are tire parameters, and V is a rolling parameter. When the load is determined, the parameters I, t, a, b, α and β are also fixed. Hence, other parameters can be calculated from equation (8). With a specified speed, V the parameters in equation (7) can also be determined. Therefore, all the coefficients in equation (6) can be determined. The minimum damping coefficient h can be obtained by solving (6).

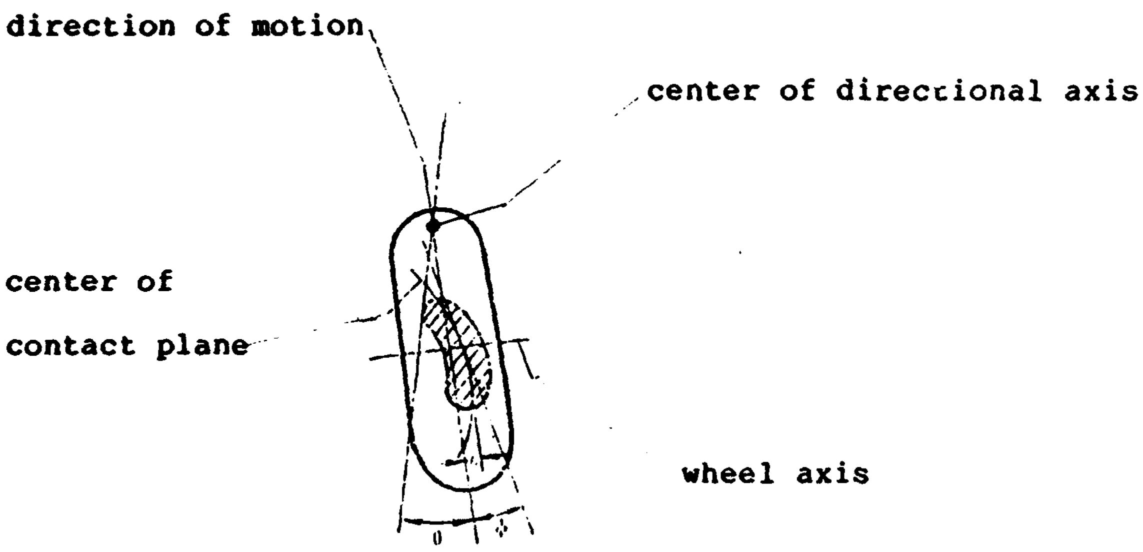


Figure 1 Tire Distortion during Wheel Shimmy

II. Assessment of Damping Characteristics of the Damper and Treatment of Non-linear Problems

The minimum damping value h_{min} obtained in the previous section is obtained under the assumption that the damper is linear, i.e. the damping moment is proportional to the rotational angular speed. In reality, most hydraulic dampers are non-linear. In general, the relation between the damping moment M and the angular speed of the driving lever $\dot{\theta}$ is

$$M=M_0+a\dot{\theta}_{ii}+b\dot{\theta}^2_{ii} \qquad (9)$$

where M_0 - *friction* moment, which is produced by the friction between the piston and the inner wall of the damper,

a, b - known constants,

$\dot{\theta}_{ii}$ - angular speed of the driving lever of the damper.

It was pointed out in reference [2] that the drag of the fluid is determined by the speed at which it flows through a channel. The drag coefficient of the fluid through a hole is determined by the shape of the jet and the velocity of the fluid in the jet. If the channel is not very long and the speed is not very fast, drag is proportional to the square of the fluid velocity. If the channel is long and flow speed is slow, then the drag is proportional to the flow speed. The damper shown in Figure 2 belongs to the former type. It satisfies the following:

$$M = M_s + b\dot{\theta}^2_1,$$

In the M $-\dot{\theta}^2$ plot, the damping coefficient b is a constant, (see Fig.3b).

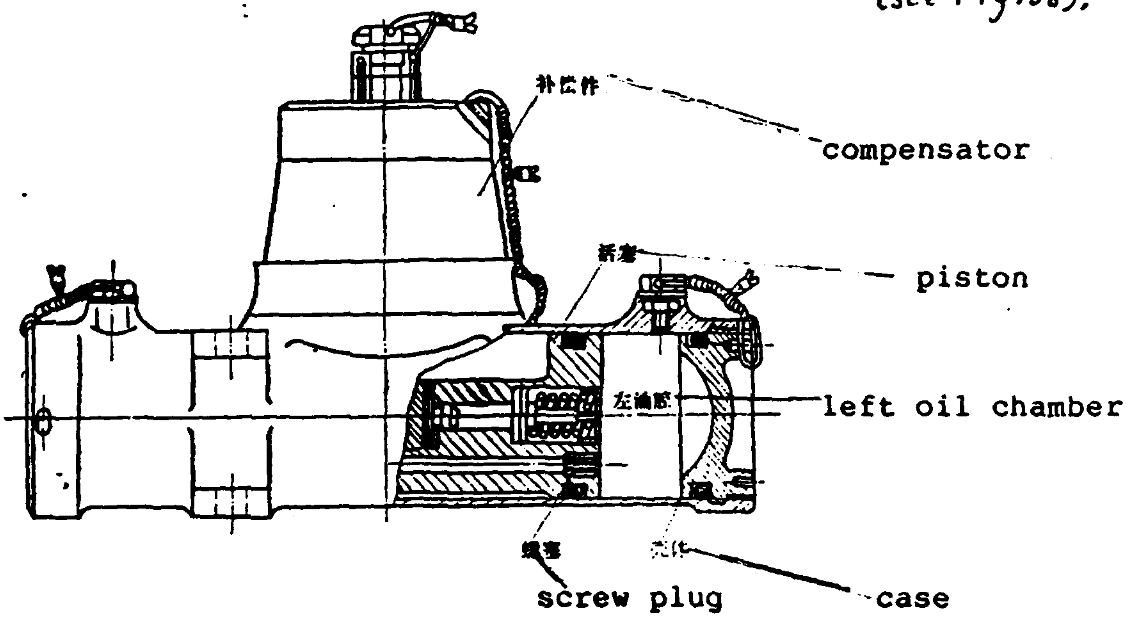


Figure 2 Structural Diagram of the JXX Damper

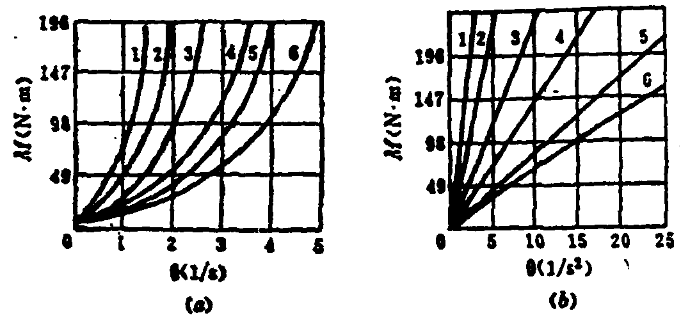
Non-linear dampers have the following advantages:

(1) It is easy to machine the flow restrictor hole on the piston. It will not be necessary to scrap the piston when the hole is drilled too deep beyond tolerance or the drill bit is broken inside and cannot be removed. This is because this kind of flow restrictor hole is shorter and the

machining is done on a special screw plug which is then screwed onto the two sides of the piston to avoid the problems described above.

(2) The non-linear damper is more effective to prevent shimmy. It increases with the angular velocity of the driving lever. The damping moment picks up faster than that produced by a linear damper. When the angular velocity is small, the damping moment is smaller than that generated by a linear damper. Therefore, the turning performance would not be affected. Hence, this type of damper is recommended to effectively prevent shimmy.

However, when this type of damper is used, how is the non-linear damping compared to the theoretical minimum damping in order to assess its ability to overcome shimmy? Two methods are introduced in this paper. Both are convenient and reliable to use in various occasions.



(a) damping moment vs. angular velocity

(b) damping moment vs. square angular velocity

damping aperture diameter 1—ϕ1.3mm 2—ϕ1.5mm 3—ϕ1.8mm 4—ϕ2.0mm 5—ϕ2.5mm 6—ϕ2.8mm

Figure 3 Damping Characteristics of the JXX Damper

1. Minimum Equivalent Linearization Method

Let the yawing angle be θ and assume it is in simple harmonic motion:

$$\theta = H\sin\omega t \tag{10}$$

where ω is the angular frequency and H is the amplitude of vibration.

(1) *energy* consumed by the linear damper per cycle

$$M = h\dot{\theta}$$

$$A = \int_0^{2\pi} M d\theta = 4\int_0^{\pi/2} (h\dot{\theta})(H\omega\cos\omega t dt) = \pi h\omega H^2 \tag{11}$$

(2) energy consumed by the non-linear damper per cycle

$$M = M_0 + a\dot{\theta} + b\dot{\theta}^2$$

$$A' = \int_0^{2\pi} M d\theta = 4\int_0^{\pi/2} (M_0 + a\dot{\theta} + b\dot{\theta}^2) d\theta$$

$$= 4M_0 H + a\pi\omega H^2 + \frac{8}{3} b\omega^2 H^3 \quad (12)$$

The equivalent damping coefficient is determined by the fact that the energy consumption per cycle is equal. From A = A' we get

$$h = \frac{4}{\pi}\left[\frac{M_0}{\omega H} + \frac{\pi a}{4} + \frac{2}{3} b\omega H\right] \quad (13)$$

Hence, h is a function of the product of ω and H. Because we are looking for the minimum equivalent damp to prevent shimmy, we have to find its extremum. In order to simplify the process, let us assume $p = \omega H$ and substitute it into (13):

$$\bar{h} = \frac{4}{\pi}\left(\frac{M_0}{p} + \frac{\pi\omega}{4} + \frac{2b}{3} p\right) \quad (13a)$$

From $\partial h/\partial p = 0$ we get

$$-\frac{M_0}{p^2} + \frac{2}{3} b = 0$$

Because ω and H are positive,

$$p = +\sqrt{\frac{3M_0}{2b}} \quad (14)$$

Also because $\partial^2 h/\partial p^2 = 8M_0/\pi p^3 > 0$, h has a minimum. Substitute (14) into (13a), we get the formula to calculate the minimum equivalent linear damping coefficient

$$h_{min} = \frac{4}{\pi}\left(\frac{M_0}{\sqrt{3M_0/2b}} + \frac{\pi a}{4} + \frac{2b}{3}\sqrt{\frac{3M_0}{2b}}\right)$$

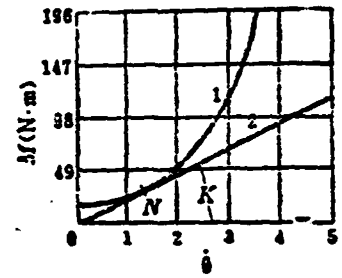
$$= \frac{8}{\pi}\sqrt{\frac{2bM_0}{3}} + a \qquad (15)$$

We can first determine the constants M_0, a and b by experimentally measuring the damping characteristics, or performing hydraulic computation, or using mathematical fit and then find the equivalent linear damping value for the non-linear damper using (15).

2. Tangent Method

This method does not require to find the constants in equation (9). Instead, the data obtained in test the damping characteristics of the damper are plotted on the $M-\dot{\theta}$ curve. M is the moment applied onto the damper and $\dot{\theta}$ is the angular velocity of the driving lever. A tangent is drawn through the origin o and the slope k is the equivalent linear damp of the curve, as shown in Figure 4.

The damping value obtained from the slope k of the tangent is very close to that calculated based on equation (15). The significance of these values is also clear, as shown in Figure 4.



1为$M=M_0+a\dot{\theta}+b\dot{\theta}^2$　2为$M=K\dot{\theta}$

Figure 4 Equivalent Linear Damp Obtained by Tangent Method

Let the equation of the curve and the equation of the tangent passing through the origin be

$$\left.\begin{array}{l} M=M_0+a\dot{\theta}+b\dot{\theta}^2 \\ M=k\dot{\theta} \end{array}\right\} \qquad (16)$$

At point N,

$$M_0+a\dot{\theta}+b\dot{\theta}^2=k\dot{\theta},$$

i.e.

$$b\dot{\theta}^2+(a-k)\dot{\theta}+M_0=0 \qquad (17)$$

Let us solve this equation and we get

$$\dot{\theta}=\frac{-(a-k)\pm\sqrt{(a-k)^2-4bM_0}}{2b} \qquad (18)$$

By analyzing (18) we know that only when

$$(a-k)^2-4bM_0=0 \tag{19}$$

and

$$a-k<0 \tag{20}$$

the straight line and the curve can intersect each other in the first quadrant. Moreover, they intersect at only one point (θ_N, M_N). From equations (19) and (20) we get

$$a-k=-\sqrt{4bM_0}$$

Therefore,

$$k=2\sqrt{bM_0}+a \tag{21}$$

Equation (21) is an expression for the slope of the tangent equation in Figure 4. The coordinate of the point of tangency is

$$\left.\begin{aligned} \theta_N &= \sqrt{\frac{M_0}{b}} \\ M_N &= 2b+a\sqrt{\frac{M_0}{b}} \end{aligned}\right\} \tag{22}$$

In order to explain the significance of the slope expressed by (21), we calculated the constant in equation (15).

$$h_{min}=\frac{8}{\pi}\sqrt{\frac{2bM_0}{3}}+a$$

$$=2.0792\sqrt{bM_0}+a \qquad (23)$$

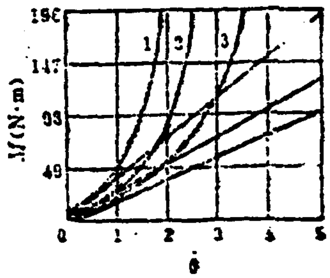
Comparing (21) to (23), we arrived at almost the same result using two different approaches. The only difference is in the coefficient. Therefore, the geometric meaning of the non-linear damper derived from equation (15) is equivalent to the tangent going through the origin. Furthermore, it is unique.

It is worthwhile pointing out that the minimum equivalent damping value obtained with either method is very conservative. However, it is necessary to prevent wheel shimmy because the minimum linear damping value computed from equation (6) is the critical value to keep the motion stable. In practice, we must consider a 个1.5~3.0 safety factor to ensure a safe margin for stability. Because the upper limit that would affect stability is very high[4], this minimum method is certainly feasible.

III. Example

The JXX front wheel shimmy damper[5] is used as an example to calculate the minimum necessary damping value to satisfy the entire range of load and speed based on equations (6~8). It is converted to h;!. = 14.1 Nms.

The non-linear damper chosen is shown in Figure 2. The flow restrictor hole was determined by referring to that in similar airplanes and by a rough estimate. It ranges from approximately $\phi 1.5$mm, $\phi 1.8$mm, $\phi 2.0$mm in diameter. Figure 5 shows the M-$\dot{\theta}$ curve based on measured damping data.



flow restrictor diameter: 1 -ϕ1.5 mm, 2 -ϕ1.8 mm, 3 -ϕ2.0 mm

Figure 5 Damping Characteristics of the JXX Damper

Which damping curve do we choose when the safety factor f = 1.8?

We first used the tangent method to find the solution. The damp value is designed to be

$$k_{;}^{'} = 1.8 \times 14.1 = 25.4\ \mathrm{Nms}$$

The tangent lines for curves 1, 2 and 3 that also pass the origin are ch_1, ch_2 and ch_3. Their slopes are shown in Table 1

Table 1 Slopes unit: Nms

	$M(N\cdot m)$	$\dot{\theta}$ (rad/s)	$K=M/\dot{\theta}$
curve ch_1	98	2.6	37.7
curve ch_2	98	3.75	26.1
curve ch_3	98	5.25	18.7

We can see that curve ch_2 meets the design requirement. The diameter of the flow restrictor should be $\phi 1.8$ mm.

Let us use the analytical method to solve the problem. The data measured with the $\phi 1.8$ mm diameter hole is used to determine the constants in (9): M = 7.35 Nm, a = 0 and b = 23 Nms^2. The result obtained from equation (21) is

$$k=2\times\sqrt{23\times 7.35}=26\,Nms$$

When using the minimum equivalent linear damping formula (15), it is

$$h_{min}=\frac{8}{\pi}\sqrt{\frac{2\times 23\times 7.35}{3}}=27\,Nms$$

The relative error between the tangent and the equivalent linearization method is

$$c=\frac{27-26}{27}\times 100\%=3.7\%$$

The shimmy experiments and long term use of the JXX landing gear have demonstrated that this assessment method is safe and reliable. Moreover, the turning ability is also very good.

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A Study on Approaches for Data Treatment of Measured Ground Loads on Airplanes

Northwestern Polytechnical University:
Lin Fujia and Zhu Depei
Flight Test Research Institute:
Wang Yuchang

Abstract

This paper discusses the requirements for measuring ground load on an aircraft. A practical data treatment method has been developed for the case that the measurement satisfy the minimum requirement. Although the method is illustrated with taxiing data in this paper, however, it is also applicable to the treatment of turning and braking load data. Chinese Translations (Jd.)

I. Introduction

Landing gear fatigue is a serious problem to consider. Among the Airworthiness Directives issued by the FAA between 1955 and 1963, 41% of the orders for piston engine airplanes and 31% of the orders for jets are related to the structure of the landing gear.[1] More accurate estimation of fatigue

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life and crack propagation will require a more accurate load spectrum. In the load spectrum, the landing and takeoff taxiing are most critical.[2,3] Therefore, the focus of discussion is centered on the treatment of taxiing load. Nevertheless, a practical data treatment method can be easily used to treat other load data.

Statistical treatment of the taxiing load data is relatively complicated. The authors did a preliminary investigation on this subject matter in reference [6]. In practice, we must consider the limitations in manpower, financial resources and flight measurement conditions and take a series of necessary engineering approximations. For example, there might be some correlations between the various parameters measured and they are very important in the force analysis of the structure. However, they are difficult to obtain and it is necessary to make some assumptions.[1] On the basis of reference [6], a practical statistical treatment of the round load data is presented in this paper.

II. Requirements for Measured Results

1. Minimum Requirements

(1) Overload record at the center of gravity of the aircraft with dual parameter counting, and the frequency at which load amplitude and mean value occur at the same time.

(2) Loading record of 3 landing gears in 3 directions with dual parameter counting, and the frequency at which load amplitude and mean value occur at the same time.

It is not necessary to record the phase correlation between the parameters measured above.

2. Satisfactory Requirements

(1) The comparative relation between the vertical overload cumulative frequency curve at the center of gravity and the vertical load cumulative frequency curve of the landing gear. Statistical data shows that the primary taxiing frequency is 3/4 Hz and could be as high as 2 ~ 4 Hz. For small aircraft with a first order wing frequency of 7 ~ 8 Hz it is considered as a rigid body. The comparison mentioned above could provide a clear correlation. For larger airplanes this correlation does not exist because of flexibility. A direct dynamic analysis is required.

(2) In the landing impact stage, the correlation between P_x (front and back) and P_y (up and down) are recorded and a joint distribution is made. If the actual data is not available, use the statistical data shown in reference [5]. For example, the landing drag coefficient for F-100 or F-104G is 0.6, the rebound coefficient is 0.5 and the lateral

coefficient (for the main landing gear) is 0.5.

(3) Record the relation between P_z (left and right) and P_y in taxiing and make a joint distribution. If there is no actual data, choose lateral overload to be $n_z = 0.05$ and combine with n_z according to certain frequency and phase as suggested in reference [5].

III. A Practical Treatment of Taxiing Data

When the data satisfies the minimum requirements, it is possible to obtain a table of joint occurrence frequency involving the load amplitudes in all directions and the mean values at the center of gravity or on the landing gear (see Table 1). In Table 1, ξ is the random load amplitude, x represents its value, η is the mean load, and y is its value. ξ and η are in unit of overload g.

Table 1 Joint Occurrence Frequency of Load Amplitude (ξ) and Mean Value (η)

η \ ξ	y_1	y_2	…	y_i	…	y_m	Σ
x_1	n_{11}	n_{12}	…	n_{1j}	…	n_{1m}	$n_{1.}$
x_2	n_{21}	n_{22}	…	n_{2j}	…	n_{2m}	$n_{2.}$
⋮	⋮	⋮		⋮		⋮	…
x_i	n_{i1}	n_{i2}	…	n_{ij}	…	n_{im}	$n_{i.}$
⋮	⋮	⋮		⋮		⋮	⋮
x_k	n_{k1}	n_{k2}	…	n_{kj}	…	n_{km}	$n_{k.}$
Σ	$n_{.1}$	$n_{.2}$	…	$n_{.j}$	…	$n_{.m}$	n^{*}

The joint probability distribution of a two-dimensional random variable (ξ, η) can be estimated based on the data shown in Table 1. On this basis, the variable mean load spectrum[8] can then be derived. However, this method requires more data and more statistical processes. Furthermore, it fails to emphasize the fact that load amplitude is a major factor. In reality, from the load data and equivalent life curves of materials, the effect of load amplitude on fatigue life is far more greater than that of mean load value. The following method takes this major factor into account and simplifies the treatment for engineering applications. The concept is to first convert all the mean values measured to normalized mean values (such as on the 1 g stress level) based on the equivalent life curve and then treat the load amplitude statistically. Finally, a simplified load spectrum with a given order is derived from the normalized damage principle.

The purpose of measuring the ground load spectrum of an aircraft is to provide the original load spectrum to the design department. This is a standard load spectrum. Therefore, we cannot use the fatigue curve (such as the S-N curve or equivalent life curve) of a specific assembly when we analyze the measured data. Instead, we should determine a more general and representative fatigue curve and then carry out the necessary simplification procedures based on the measured load data.

1. Several Assumptions

(1) It is assumed based on a lot of data[7] that the stress amplitude S_a and fatigue life N have the following relation when the mean stress S_m is a constant:

$$\lg N = a + b \lg S_a \quad (1)$$

where a and b are material constants.

(2) The equivalent life S_a - S_m curve is a straight line which passes through (0, S_{-1}) and (S_b, 0). The equation is

$$S_a = S_{-1}\left(1 - \frac{S_m}{S_b}\right) \quad (2)$$

where S_b is the stretching limit of the chose material and S_{-1} is the endurance limit of the material under a symmetric cycle (R=-1).

(3) The fatigue damage follows the Miner linear cumulative damage law.

(4) When the design load coefficient n_{max} is reached, the total stress at the ideal endangered point reaches the stretching limit S_b. Based on this, the stress level at 1 g

during taxiing is

$$S_{1g} = \frac{S_b}{n_{max}} \qquad (3)$$

2. Conversion to Equivalent Life

The data in Table 1 are converted to dual parameter stress at the endangered point.

$$\left.\begin{array}{l} S_{ai} = S_{1g} \cdot x_i, \ i = 1, 2, \cdots, k \\ S_{mj} = S_{1g} \cdot y_j, \ j = 1, 2, \cdots, m \end{array}\right\} \qquad (4)$$

The dual parameter stress (S_{ai}, S_{mj}) has the same frequency of occurrence as that of (x_i, y_i), i.e. n_{ij}.

Based on the equivalent life principle, the dual parameter stress data are simplified to single parameter stress data with a constant mean stress value. The constant mean stress is denoted as S_{mo}. If the equivalent life conversion of any pair of stress (S_{ai}, S_{mj}) is (S_{aij}, S_{mo}), then based on assumption (2) we have

$$S_{aij} = S_{ai} \frac{S_b - S_{mo}}{S_b - S_{mj}} \quad \begin{array}{l} i = 1, 2, \cdots, k \\ j = 1, 2, \cdots, m \end{array} \qquad (5)$$

The frequency of occurrence of S_{aij} is still n_{ij}.

3. Equivalent Damage Conversion

After equivalent life conversion, the order of stress amplitude is usually still too high to use. Therefore, we still have to make another conversion based on the equivalent damage principle.

(1) Direct Conversion

The order of converted stress amplitude is pre-set at M based on pre-determined requirements. Choose M = 8 for taxiing load by referring to reference [3] Denote that the stress amplitude at each order is $S_a^{(k)}$ (k = 1, 2, .., M).

S_{aij} is converted to its closest stress amplitude $S_a^{(k)}$. Let us assume that the conversion order is $n_{ij}^{(k)}$, and k_1 is the sequence of S_{aij} to be converted to the kth order of stress amplitude. N_{ij} represents the life of S_{aij} at the ideal endangered point and $N^{(k)}$ is the life of $S_a^{(k)}$ at the ideal critical point. Based on assumption (1) we have

$$N^{(k)}=10^{a}(S_a^{(k)})^{b} \quad (6)$$

$$N_{ij}=10^{a}(S_{aij})^{b} \quad (6')$$

Again, based on assumption (3)

$$\frac{n_{ij}^{(k)}}{N^{(k)}}=\frac{n_{ij}}{N_{ij}} \quad (7)$$

From equations (6), (6') and (7), it is not difficult to get

$$n_{ij}^{(k)} = n_{ij}\left(\frac{S_a^{(k)}}{S_{aij}}\right)^b \tag{8}$$

Add all the numbers simplified to the same order of stress $S_a^{(k)}$ and we have the number of times $n^{(k)}$ that stress occurs in a landing or takeoff, i.e.

$$n^{(1)} = \sum_{k_1=1}^{m_1} n_{ij}^{(1)}$$

where m_1 is the number of S_{aij} occurring in $\left(0, \frac{S_a^{(1)}+S_a^{(2)}}{2}\right)$.

$$n^{(k)} = \sum_{k_1=1}^{m_k} n_{ij}^{(k)}$$

where m_k is the number of S_{aij} occurring in $\left(\frac{S_a^{(k-1)}+S_a^{(k)}}{2}, \frac{S_a^{(k)}+S_a^{(k+1)}}{2}\right)$ and $k = 2, 3, \ldots, M-1$.

$$n^{(M)} = \sum_{k_1=1}^{m_M} n_{ij}^{(M)}$$

where m_M is the number of S_{aij} that is greater than $\frac{S_a^{(M-1)}+S_a^{M}}{2}$ (9)

A stress amplitude spectrum with a constant mean value S_{mo} is obtained (see Table 2).

The overload coefficient spectrum with a mean overload of S_{mo}/S_{1g} can be obtained by dividing stress with S_{1g}.

Table 2 Stress Amplitude Spectrum with a Constant Mean Value S_{aij}

stress amplitude	$S_a^{(1)}$	$S_a^{(2)}$	...	$S_a^{(i)}$	...	$S_a^{(N)}$
number of occurrences	$n^{(1)}$	$n^{(2)}$	...	$n^{(i)}$	...	$n^{(N)}$

(2) Statistical Smoothing Conversion Method

The equivalent life converted stress amplitude can be smoothed out by using an appropriate probability distribution function before performing the equivalent damage conversion.

(a) selection of suitable stress amplitude probability function

In order to select a suitable function from several probability distributions, an optimization check for the fit can be used. To this end, the S_{aij} obtained are rearranged from low to high (as shown in Table 3), where

$$n' = k \cdot m, \quad \sum_{i=1}^{n'} n_{(i)} = n^* \qquad (10)$$

Table 3 Table of S_{aij} in Sequence

stress amplitude	$S_{a(1)}$	$S_{a(2)}$	...	$S_{a(i)}$	...	$S_{a(n')}$
number of occurrences	$n_{(1)}$	$n_{(2)}$	...	$n_{(i)}$	...	$n_{(n')}$

There are many methods to check the fit. Here, we are using the W^2 method introduced by Simlnov:

$$n^*W^2 = n^* \int_{-\infty}^{\infty} \left[F_n(x) - F_0(x)\right]^2 dF_0(x) \tag{11}$$

where F_n and F_o are the empirical and theoretical distribution functions of S_{aij}, respectively.

n^*W^2 can be calculated based on the following equation:

$$n^*W^2 = \frac{1}{12n^*} + \sum_{i=1}^{n'} \sum_{k=1}^{n_{(i)}} G_{ik} \tag{12}$$

where

$$G_{ik} = \left\{ \begin{array}{ll} \left[F_0(S'_{aik}) - \frac{2k-1}{2n^*}\right]^2, & i=1,\ n_{(i)} \neq 0 \\ \left[F_0(S'_{aik}) - \frac{2\left(\sum_{j=1}^{i-1} n_{(j)}\right) + 2k - 1}{2n^*}\right], & i \neq 1,\ n_{(i)} \neq 0 \\ 0 & n_{(i)} = 0 \end{array} \right\} \tag{13}$$

$$S'_{aij} = \begin{cases} k \cdot \dfrac{S_{a(2)} - S_{a(1)}}{2n_{(1)}}, & i=1 \\ \dfrac{S_{a(i)} + S_{a(i-1)}}{2} + k \cdot \dfrac{S_{a(i+1)} - S_{a(i-1)}}{2n_{(i)}}, & 1<i<n' \\ \dfrac{S_{a(n')} + S_{a(n'-1)}}{2} + k \cdot \dfrac{S_{a(n')} - S_{a(n'-1)}}{n_{(n')}}, & i=n' \end{cases} \quad (14)$$

Based on the given degree of significance a, the percentage Z_a corresponding to 100 a % can be looked up from the n^*W^2 percentage table. If $n^*W^2 < Z_a$, F_o is considered as the probability distribution function of S_{aij}. If several theoretical functions pass the test, the one with the smallest n^*W^2 can usually be chosen as the distribution function for S_{aij}.

(b) Discrete Stress Amplitude

The range of S_{aij}, (d_o, d_{k2}), is divided into k_2 intervals of equal width. Based on the probability function $F_o(x)$ of S_{aij}, we can calculate the probability of occurrence of S_{aij} in each interval and use it as the probability for the stress amplitude to occur at the center of that interval. Based on the number of loadings in one takeoff or landing, the frequency for the stress amplitude to occur at the center of each interval can be calculated. The width of each interval is

$$W = \frac{d_{k2} - d_0}{k_2} \quad (15)$$

In the ith interval (d_{i-I}, d_i) $(i = 1, 2, \ldots, k_2)$, the representative stress is denoted as S_{ai}:

$$S_{ai}=d_0+(i-\frac{1}{2})W \qquad (16)$$

The probability for S_{ai} to occur is

$$p_i=F_0(d_i)-F_0(d_{i-1}) \qquad (17)$$

In one landing and takeoff, the number of occurrence of S_{ai} is

$$n_i=n^*p_i \qquad (18)$$

(c) The method used to perform equivalent damage conversion of the statistically treated data is similar to that described before in (1). S_{aij} and n_{ij} are converted to S_{ai} and n_i, respectively to get the simplified stress spectrum and overload coefficient spectrum.

No matter which method we use, after the simplified stress spectrum and overload coefficient spectrum are obtained, we can derive the cumulative frequency spectrum for stress or overload coefficient based on need.

4. Statistical Treatment of Data on Multiple Landings and Takeoffs

The taxiing load data were obtained in L landings and takeoffs under nominally identical conditions. The taxiing data, for each run can be processed to obtain the simplified overload coefficient spectrum. The results are shown in Table 4.

Table 4 Simplified Overload Coefficient Frequency Table for Multiple Landings and Takeoffs

number \ overload coefficient	$x^{(1)}$	$x^{(2)}$	$\cdots$	$x^{(i)}$	$\cdots$	$x^{(M)}$
1	$n_1^{(1)}$	$n_1^{(2)}$	$\cdots$	$n_1^{(i)}$	$\cdots$	$n_1^{(M)}$
2	$n_2^{(1)}$	$n_2^{(2)}$	$\cdots$	$n_2^{(i)}$	$\cdots$	$n_2^{(M)}$
$\vdots$	$\vdots$	$\vdots$		$\vdots$		$\vdots$
i	$n_i^{(1)}$	$n_i^{(2)}$	$\cdots$	$n_i^{(i)}$	$\cdots$	$n_i^{(M)}$
$\vdots$	$\vdots$	$\vdots$		$\vdots$		$\vdots$
L	$n_L^{(1)}$	$n_L^{(2)}$	$\cdots$	$n_L^{(i)}$	$\cdots$	$n_L^{(M)}$

(1) If for an overload coefficient $X^{(k)}$ ($k = 1, 2, \cdots, M$), the frequency of occurrence in each taxiing follows the normal distribution (which was found to be valid experimentally), it is possible to calculate the upper limit of confidence for the mean occurrences of an overload coefficient $X^{(k)}$.

$$n_k^{(i)} = \bar{n}^{(i)} + \frac{S^{(i)}}{\sqrt{L}} t_\alpha, \quad k = 1, 2, \cdots, L \qquad (19)$$

where

$$\bar{n}^{(k)} = \frac{1}{L}\sum_{i=1}^{L} n_i^{(k)}$$

$$S^{(k)} = \sqrt{\frac{1}{L-1}\sum_{i=1}^{L}\left[n_i^{(k)} - \bar{n}^{(k)}\right]^2}$$

t_a is the point corresponding to 100 α % in the t distribution which has v = L-1 degrees of freedom. It is looked up from the $1-\alpha$ distribution table based on the given confidence level.

Finally, the upper limit spectrum for each overload coefficient in a single taxiing event is obtained (see Table 5).

Table 5 Upper Limit of Mean Number of Occurrences for Each Overload Coefficient

overload coefficient	$x^{(1)}$	$x^{(2)}$	—	$x^{(k)}$	—	$x^{(K)}$
number of occurrences	$n_s^{(1)}$	$n_s^{(2)}$	—	$n_s^{(k)}$	—	$n_s^{(K)}$

(2) Based on need, it is also possible to calculate the upper confidence limit for the number of occurrences for each $X^{(k)}$ in each landing or takeoff, $n_s^{(k)}$.

$$n_s^{(k)} = \bar{n}^{(k)} + t_a S^{(k)} \sqrt{\frac{L+1}{L}} \tag{20}$$

where $\bar{n}^{(k)}$, $S^{(k)}$ and t_a are defined as before.

Thus, the upper limit spectrum for the overload coefficients to occur in one taxiing is obtained.

With the above result, the cumulative frequency spectrum of the overload coefficients in one taxiing can be conveniently derived.

It is easy to see that the above load treatment method can be easily expanded to treat turning and braking data measured on the ground.

IV. Discussion and Conclusions

In addition to the equivalent damage method, which was described earlier to convert the equivalent life data (i.e. number of occurrences for each stress amplitude of a constant mean value) to the effective number of occurrences of each simplified stress amplitude, there is another equivalent damage probability method which guarantees equal probability of stress induced damage before and after conversion. However, it can be verified[8] that when assumption (1) is valid the fatigue life follows a logarithmic normal distribution at all stress levels. The standard deviation is a constant. (It was proven experimentally that aeronautical metals can usually meet this requirement.) In this case, the equivalent damage probability method is essentially the same as the equivalent damage method. In other words, the equivalent damage method used in this work is the equivalent damage probability method under less rigorous conditions.

The treatment of aircraft ground load data is a very complex job. The method presented in this paper is a statistical treatment of the data under the premise that the minimum requirements are met. Figure 1 shows the flowchart of this method. If the data meets the satisfactory requirements in the future, new statistical methods can be developed based on the correlation of the parameters.

Statistical treatment of ground load data is only a portion of the study of ground load spectrum and the formulation of standards. The result does not represent a standardized load spectrum. Many factors such as load conditions and environmental impact are difficult to be reflected in finite measurements. They must be considered by other suitable methods (such as empirical parameters).

The methods introduced in this paper has been successfully used to treat the data obtained in 120 landings and takeoffs of the Chinese made Yun-7 transport at different airports. A landing load cumulative frequency spectrum similar to the one described in MIL-A-008866B(USAF) has been obtained. In addition, it is being used in the fatigue testing of the landing gear of that aircraft.

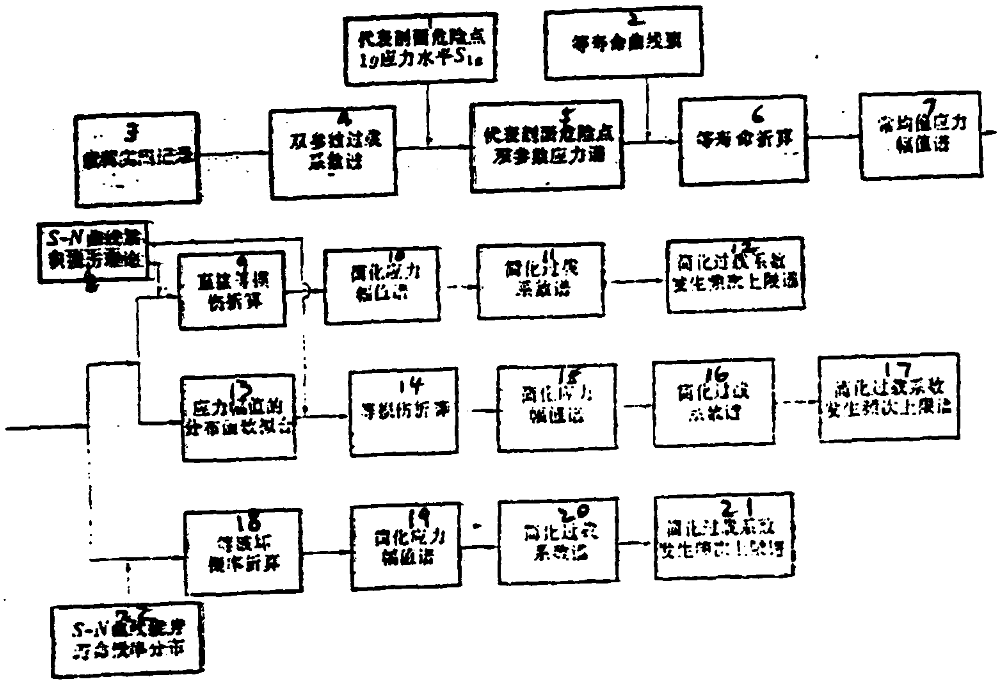

Figure 1 Flowchart for the Statistical Treatment of Data

1. 1g stress level, S_{1g}, at representative danger point in the cross-section; 2. family of equivalent life curves; 3. measured load data; 4. dual parameter overload coefficient spectrum; 5. dual parameter stress spectrum at representative danger point in the cross-section; 6. equivalent life conversion; 7. constant mean stress amplitude spectrum; 8. S-N curve cumulative damage theory; 9. direct equivalent damage conversion; 10. simplified stress amplitude spectrum; 11. simplified overload coefficient spectrum; 12. upper limit frequency spectrum for the occurrences of simplified overload coefficients; 13. fitting stress amplitude with a distribution function; 14. equivalent damage conversion; 15. simplified stress amplitude spectrum; 16. simplified overload coefficient spectrum; 17. upper limit frequency spectrum for the occurrences of simplified overload coefficients; 18. equivalent damage probability conversion; 19. simplified stress amplitude spectrum; 20. simplified overload coefficient spectrum; 21. upper limit frequency spectrum for the occurrences of simplified overload coefficients; 22. S-N curve fatigue life probability distribution

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