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THEORETICAL AND EXPERIMENTAL INVESTIGATION OF JOINT-STRUCTURAL DAMPING INTERACTION FOR AIRPLANE CONSTRUCTION

I. N. Krivošić
University in Belgrade
Yugoslavia

Abstract

In the first part of this paper the general review of joint-damping interaction was given with corresponding literature. Also some theoretical aspects and possible physical models of joint - damping interaction have been considered.

Experimental results, for typical airplane construction, are mainly treated in the second part of this paper. The experiments have been done on hipothetical wing by using the same geometrical model with different types of joint between the skin, stringers and transversal elements. The rivet joints, bolt joint and adhesive joint have been analysed, and different types of methods have been used for carrying out the same experiment. The corresponding mutual influence of the amplitude on structural damping was analysed by using the experimental results. Those results have been used for obtaining the most adequate conclusions to be applied in practice.

Nomenclature

q - coordinate (displacement)
a, a_1 - amplitude
ε - strain
Z_i - vertical displacement
φ_i, α_i - angle of twisting
m_i, J_i - mass, moment of inertia
$f_{(q)}$ - skeleton curve
δ - logarithmic decrement
k_i - viscosity coefficient
$W, \Delta W$ - energy, loss energy
c_i - coefficient
n - number degree of freedom
Q - force
$\vec{\phi}, \vec{Q}$ - upwards curve
$\overleftarrow{\phi}, \overleftarrow{Q}$ - downwards curve
$P_{i(t)}$ - external excitation forces
M_i - bending moment
σ - normal stress
t - time

I. Introduction

Damping properties of realistic systems which are manifested in converting of the mechanical energy of oscillation into another mode of energy are caused by various dissipation forces. Most important areas being: inner friction forces in the material, forces caused by structural damping and resistance forces of the enviroment.

During oscillation of structures with thin walls the major part of energy is dissipated along the system as a result of dissipation forces caused by structural damping. The nature of these forces is determined by specific characteristics of the involved structures which have a large number of elements which are connected by rivets or screws. Dissipation forces usually have a positive function which limits the oscillation amplitude and extends the stability fields, though there are cases of excess damping which has to be reduced.

There are many papers treating theory and practice concering the analysis of dissipation force influence on system behaviour. Many studies, including also a large bibliography of this specific matter can be found in the papers of general character[1,2,7], as well as in proceedings of scientific and technical conferences concerning energy dissipation during oscillation of mechanical system[3,4,5]. However there are only few papers dealing with inner structural damping of the structures with thin walls. There is an evergrowing need for research work of this particular problem owing to the significant influence of the dissipation forces on the dynamic behaviour of these systems, specially during analysis and determination of the dynamic strength and stability of modern airplanes and cosmic structures.

It is well known that joints between elements in structures with thin walls, in addition to its basic function providing of stiffness and required strength, may be also used as regulators of the vibration level. The problem of structural damping has not been studied enough up to now and there are no generally recognized solutions, even for a specific type of a structure, because dissipation of energy along the structure depends on various factor of a very complex nature which differs for various types of joints. Forming of reliable expressions for damping forces of realistic structure represents a complex problem which still has to be thoroughly researched. So far carried out experimental and theoretical researches as well as the obtained results give only certain phenomenological notions and assumptions which can be used for solving the problems in practice. In general, during analysis of mechanical energy dissipation during oscillation we are faced with two major problems:

The choice of physical dependance in order to describe the behaviour of structure during oscillation with influence of various factors (temperature, frequency, amplitude etc.) as well as a method for solving the equations obtained in that way.

The shape of curves which describe the hysteresis loop depends actually upon the mechanism of mechanical energy dissipation in the system and upon the system status defined by treatment, distortion velocity, temperature and other factors.

It is rather difficult to make a proper analysis of the structure´s dynamic behaviour without knowing well the damping properties of the system. With the purpose of a complete studying of the basic damping features of the structure, many theories are being[7,6] applied describing the relation between stress and strain (force and displacement). Two basic tendencies are obvious in such papers: the first one tries to describe exactly and completely the hysteresis loop (these expressions have as a rule a lot of parameters which are obtained by means of complex investigations) and the second one which aims at obtaining only the exact surface of loop and eventually its dependance upon certain parameters.

The attempts to solve the problem by forming an exact physical model did not give satisfactory results except for some simple cases. The problem is more complex owing to deformability of all structure elements and unreliability of parameters which define the internal damping.

This paper tries to solve the problem in a phenomenological way without considering in details the physical model. The structural damping consists of a part which depends on coordinates and part which depends on velocity. The required parameters are determined by simple testing and the obtained results may be used also for a description of a more complex dynamic behaviour of the system.

II. Formulation

Starting from the assumption that in the realistic complex system we have a large number of elements and joints in which dissipation of the mechanical energy can be effected along the structure and that by the increase of the oscillation amplitude they are being activated one after another, the appropriate stress-strain skeletion curve shall be discontinual and each new slip in the joints shall cause corresponding reducement of stiffness.

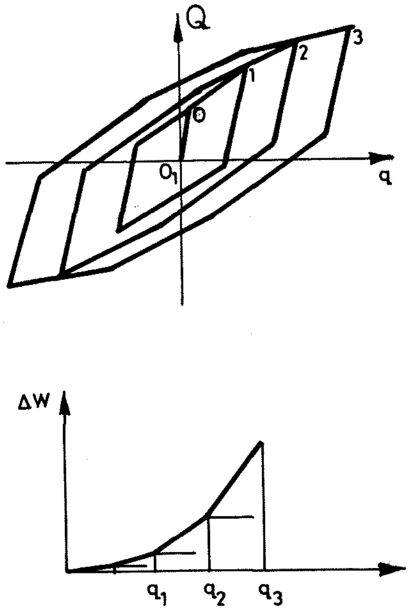

Figure 1.

On the basis of the law of Rayleigh it is possible to get a corresponding hysteresis loop as well as the function of the lost energy with the amplitude change. It is obvious that within 0_1-0 no energy dissipator is actuated which has an influence on the function $W(q)=0$ for $0<q<q_0$. Making further analysis of this dependance we shall see that one hysteresis loop corresponds to point 1, the other one to point 2 etc. Assuming very short intervals between actuation of certain dissipators in the system, the paper[8] deals with the dependance between the skeleton curve and dissipated energy in a diferential form:

$$\frac{df}{dq} = \frac{f}{q} - \frac{dW}{dq}\,\frac{1}{4q} \qquad (1)$$

or through logarithmic decrement

$$\frac{f'}{f} = \frac{4\delta' q-\delta}{(4+\delta)q} \qquad (2)$$

The expressions (1) and (2) may be used for obtaining the skeleton curve if the function $\delta(q)$ can be obtained.

In the recent time the model of amplitude dependant hysteresis is being used in the form[10]:

$$\sigma = \sigma_e(\varepsilon,\varepsilon_0) \pm \sigma_p(\varepsilon,\varepsilon_0) \; . \qquad (3)$$

We may observe the existance of a skeletòn curve from which afterwards the nonelastic component is either added or reduced depending on the velocity sign (fig. 2).

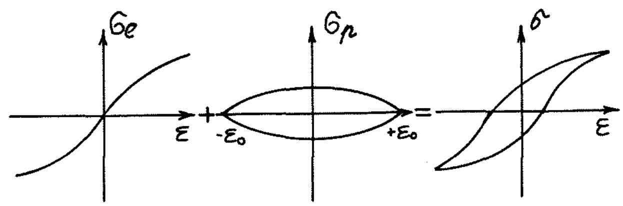

Figure 2.

Such models may be used only for analysis of system where the deformation is carried out according to harmonic or near harmonic law.

Both models assume the same form of hysteresis cycle branch during ascending and descending. However numerous investigations contradict this assumption so that various corrections of the mentioned principle were carried out by introducing unsymmetrical increasement for upwards and downwards curves which complicates extremely the calculation for which the parameter identification for realistic system is too complex and rather unreliable.

The problem is further treated through a general approach i.e. it is assumed that the ascending and descending curve of the hysteresis loop may be of an arbitrary type and can be obtained by means of a functional transformation of initial branch $F_i(q\dot{q})$ and coordinates of "frozen" points (fig. 3) in the shape of:

$$Q_i(q,\dot{q}) = \begin{cases} \vec{\phi}_1[q_A, Q_A, f(q)] & \dot{q}(t)>0 \\ \overleftarrow{\phi}_2[q_B, Q_B, f(q)] & \dot{q}(t)<0 \end{cases} \qquad (4)$$

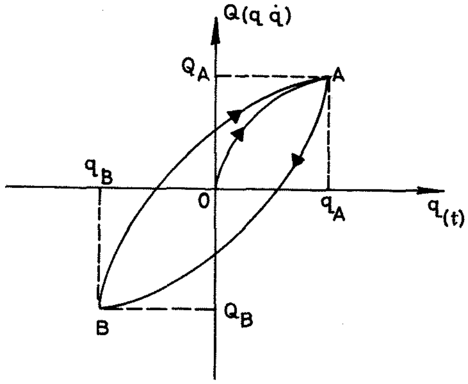

Figure 3.

It is obvious that during change of the sign $\dot{q}(t)$ the return points may be at any place on the curve and arbitraries consants C_i (i=1,2,3,4) may be considered as variables, so that dependance (4) may be formulated in the following way:

$$\overleftrightarrow{Q} = \begin{cases} \vec{\phi}\,(q, C_1, C_2) & \dot{q}>0 \\ \overleftarrow{\phi}\,(q, C_3, C_4) & \dot{q}<0 \end{cases} \qquad (5)$$

Special characteristics of so defined hysteresis loop is in a simultanious definition of frequent and decrement characteristics of the system. Equation (5) is possible to be given in form of addition of two functions of which only one depends on of the velocity

$$\overleftrightarrow{\phi}_{(q)} = f_{1(q)} \operatorname{sign} \dot{q} + f_{2(q)} , \qquad (6)$$

where

$$f_{1(q)} \operatorname{sign}\dot{q} = \frac{1}{2}\left[\vec{\phi}_{(q)} - \overleftarrow{\phi}_{(q)} \operatorname{sign} \dot{q}\right] \qquad (6a)$$

is a value which defines nonconservative damping force for one cycle of oscilations, and

$$f_{2(q)} = \frac{1}{2}\left[\vec{\phi}_{(q)} - \overleftarrow{\phi}_{(q)}\right] \qquad (6b)$$

is a value by which the force of formation of skeleton curve for one cycle is defined.

Equation $f_i(q)$ can be given in the form

$$f_{i(q)} = \bar{f}_{i(q)} + \hat{f}_{i(q)}; \quad \bar{f}_{i(q)} = -\bar{f}_{i(-q)}$$
$$\hat{f}_{i(q)} = \hat{f}_{i(-q)} \qquad (i = 1, 2) \qquad (7)$$

and their shape is given in the figure (4)

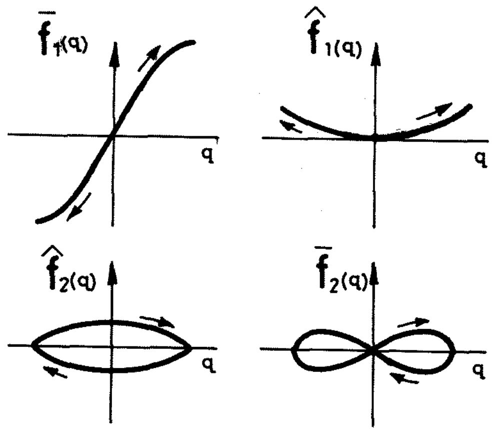

Figure 4.

and each of them describes some nonlinearity of the system. All the values (7) can be defined on the basis of relatively simple experiments, described in detail in bibliography[11]. For the purpose of simplicity, it can be assumed that only the function $f_2(q,a)\operatorname{sign}\dot{q}$ depends on oscilation amplitudes, permiting the hysteresis loop to be closed. Hence it follows that with the increasing of the amplitude from $a=a_1$ to $a=\mu a_1 (\mu>1)$ the value of the function $f_2(q,a)$, at the same phase changes $q(t)$, is increasing by μ^s timed, where $s>-1$ and function $f_2(q,a)$ satisfy the condition

$$\mu^s f_2(q,a) = f_2(\mu q, \mu a) \quad . \qquad (8)$$

Condition (8) represents Euler's functional equation, the solution of which for $a \neq 0$ according to[12] is:

$$f_2(q,a) = a^s \Upsilon(\frac{q}{a}) \tag{9}$$

where $\Upsilon(\frac{q}{a})$ is an arbitrary function of the argument $\frac{q}{a}$ which characterises the shape of the curve $f_2(q,a) \operatorname{sign}\dot{q}$. Finally for estimating $f_2(q,a)$, which will satisfy equation (9), it is necessary to determine experimentally the dependance of $\Upsilon(\frac{q}{a})$, which characterises the dissipated energy for one cycle, while the procedure for its estimation can be found in the bibliography [13]. However, experiments have shown [10] that depending on the starting conditions and according to figure 3 the functions Q-q (or σ-ε) have one family of ascending curves for $\dot{q}>0$ and one family of descending curves for $q<0$ (noncontinued lines in fig. 5).

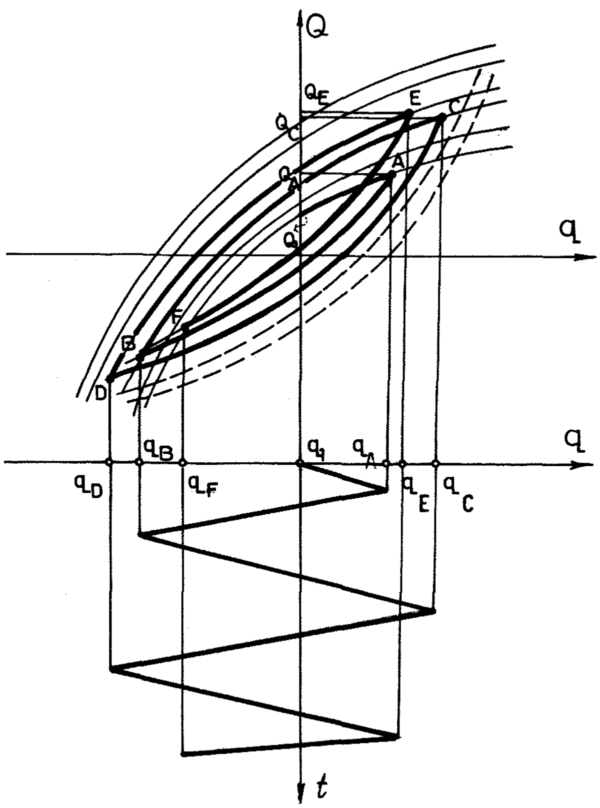

Figure 5.

In the general case the postion of both families at the observed plane at the given temperature and other external conditions is defined also by the speed of deformation.

Loading and unloading curves can be obtained by relatively simple and inexpensive experiment — fig. 6 and 7.

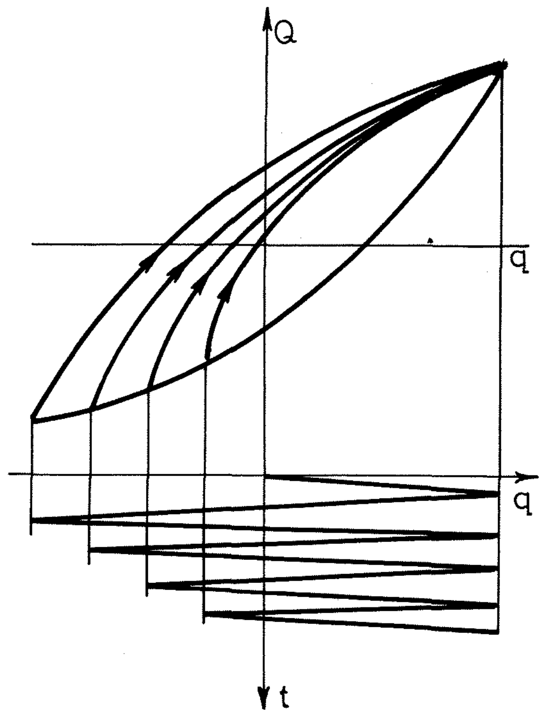

Figure 6.

At the given arbitrary law of deformation $q=q(t)$, fig. 5, there will also be an appropriate hysteresis cycle on the basis of equation 5 as follows: At the points at which the sign of speed $\dot{q}(0_1$ A B C D E) is changing, the coordinates Q and q are fixed depending on the sign of $\dot{q}$; from the curves $\vec{\phi}$ and $\overleftarrow{\phi}$ we find constants C_i and we now have a actual depedance on which the cycle continues to develop. In other words at the points where the sign of $\dot{q}$ is changed it is necessary to solve the Cauchy's problem. If the solution of the problem is monotonous, then the movement shall continue to develop along one curve which can be accomplished by choising function (4).

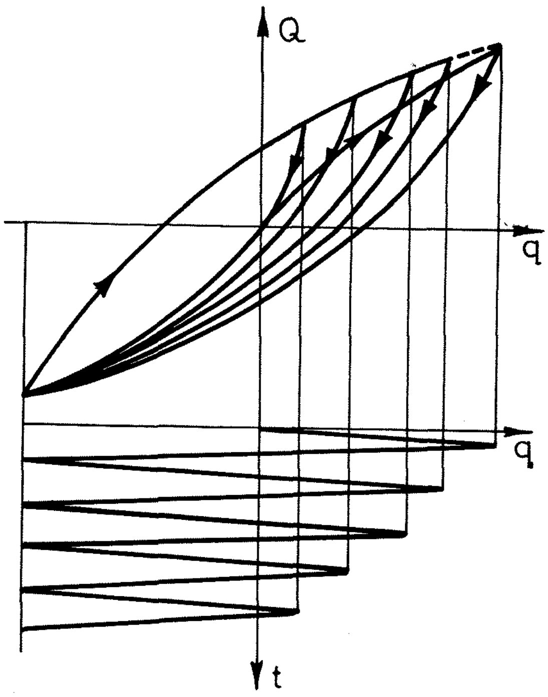

Figure 7.

Analysis of an arbitrary hysteresis system can be carried out by computer technique so that at the points where the sign of speed $\dot{q}$ is changed the transition from one family of curves to another apropriate family of curves has to be defined.

III. Application to aeroplane wing oscillation

The above mentioned method will be in the rest of the paper applied to the oscillation of an aeroplane wing with arbitrary structural damping.

Mathematically it is very difficult to solve the problem as a continuous system with infinite degrees of freedom, so that a discrete model must approximately be analysed. The number of degrees of freedom is not limited. The mechanical model of wing is taken in the form of discrete inertial elements mutually connected by nonlinear hysteresis joints, defined according to relations (5), fig. 8.

In the analysis of pure flexional oscillations with n degrees of freedom appropriate system of equations would be:

$$m_i \frac{d^2 Z_i}{dt^2} + k_i \frac{dZ_i}{dt} + F_i(Z,\dot{Z}) - F_{i+1}(Z,\dot{Z}) = P_{i(t)} \qquad (10)$$

$$(i = 1, 2, \ldots, n) \qquad F_{n+1}(Z,\dot{Z}) = 0$$

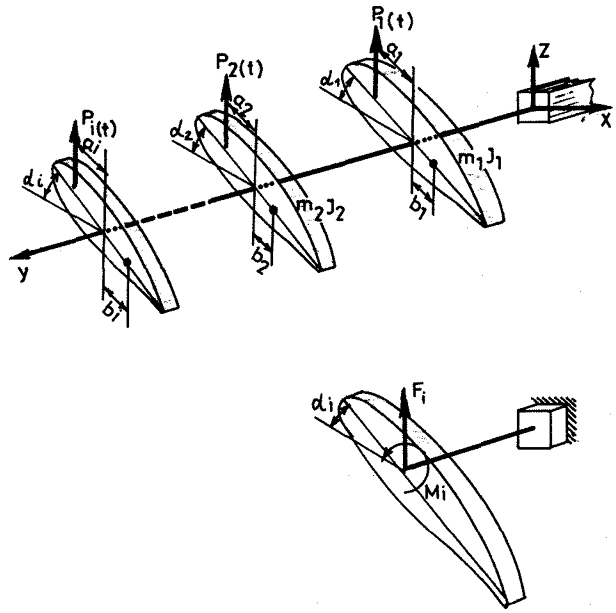

Figure 8.

It is obvious that the system of equations (10) permits the analysis of stationary and nonstationary oscillations and also the passing through a resonation zone

Equations with n degree of freedom for pure torsion oscilations can be written in the form of

$$\frac{d^2\varphi_i}{dt^2} = -\frac{a_i P_{i(t)}}{J_i} + \frac{a_{i-1} P_{i-1(t)}}{J_{i-1}} - \frac{k_i}{J_i}\frac{d\varphi_i}{dt} + \frac{M_{i-1}(\varphi_{i-1},\dot{\varphi}_{(i-1)})}{J_{i-1}} - M_i(\varphi_i,\dot{\varphi}_i)\cdot\left(\frac{1}{J_{i-1}}+\frac{1}{J_i}\right) + \frac{M_{i+1}(\varphi_{i+1},\dot{\varphi}_{i+1})}{J_{i+1}} \qquad (11)$$

$$(i = 1, 2, \ldots, n) \qquad (\varphi_i = \alpha_i - \alpha_{i-1})$$

where the accepted hysteresis characteristics of wing are given as a function of relative angular displacements of sections.

System of equations of motion (11) is written in a non-dimensional form and can be solved by modern computer techniquès

Flexional-torsinal oscillations are the most interesting and dangerous for an aero plane wing. Under the assumption that the dissipation of energy at the bending and twisting is mutually independent [14], the flexional-torsional oscillations of a wing with 2n degree of freedom can be written in the form of:

$$m_i\frac{d^2Z_i}{dt^2}+S_i\frac{d^2\alpha_i}{dt^2}+k_i\frac{dZ_i}{dt}+F_i(Z,\dot Z)-F_{i+1}(Z,\dot Z)=P_{i(t)}$$

$$J_i\frac{d^2\alpha_i}{dt^2}+S_i\frac{d^2Z_i}{dt^2}+k_\alpha\frac{d\alpha}{dt}+M_i(\Delta\alpha_i,\Delta\dot\alpha_i)-$$

$$-M_{i+1}(\Delta\alpha_{i+1},\Delta\dot\alpha_{i+1})=-a_iP_i(t) \qquad (12)$$

$$i=1,2,\dots,n \quad \Delta\alpha_i=\alpha_i-\alpha_{i+1} \quad F_{n+1}=M_{n+1}=0$$

$$S_i=m_ib_i \qquad J_i=J_{io}+m_ib_i^2$$

Taking into consideration that the families of loading and inloading curves are obtained by static tests, in the equations (10), (11) and (12) the part of viscosity damping has been introduced. The coefficient k_i is determined experimentally by taking into account the difference between damping during dynamic and static testing.

Sugested method for a dynamical analysis of the aeroplane wing permits to carry out a qualitative and quantitative analysis of the constructive damping influence of a realistic system, whose parameters have been obtained by relatively simple experiments so that the field of stability can be estimated more precisely. It can be analyzed to what extent a rivetted or other types of joints, except its basic function, can also have the function of regulator of the level of vibration on aeroplane wings. For nonlinear system the friction forces in the joint elements have influence also on own frequenceś and with their help it is possible to "remove" the construction from the resonant zone. On the other side, it has been shown that the damping parameters are variable with the age of the construction[14] and it is possible to apply the proposed method also to a simple technical analysis of the state of the construction when it has already been in exploatation.

IV. Experimental results and comparasion with numerical data

From 1980 to 1982 , the author Carried out a large number of experiments on the hypothetical airplane wing model at the laboratories of the Department of Aerospace Engineering, the Faculty of Mechanical Engineering, the University of Belgrade. All the models were the same except in the type of joints between the elements. Three types of joints were used: adhesive, rivets and bolts. At the bolted model experiments were carried out applying various torque tightenings — so that the analysis dealt also with the influence of this factor on the damping characteristics. The models were first tested statically. These static tests gave the variations of the damping characterstics with changes of the displacement amplitude at the cyclical loading by the static hysteresis loop. The families of loading and unloading curves for all the tested models were subsequently obtained as in figures 6 and 7.

After that, the models were subjected to dynamical testings for the purpose of getting damping characteristics dependance on the amplitude of oscilations. Three different methods were used for definition of damping: the spread of the resonance peak; the hysterésis dynamical loop in case of forced oscilations; the falling of the amplitude in case of free oscillations. In all three cases the influence of the amplitude

-damping forces in the system in comparison to the inertial and elastic forces are small,

-the greatest part of the damping does not depend on the velocity of oscillations(data obtained by static histeresis loop),

-it can be assumed,according to fig.12, that the ratio between logarithmic decrement for static and dynamic testing depends on the velosity of oscillatons and considering that it is a smaller part of the total damping (except for the adhesive joints for which the relation is more complex)the viscous damping forces could be treated like in equations (10),(11),(12).

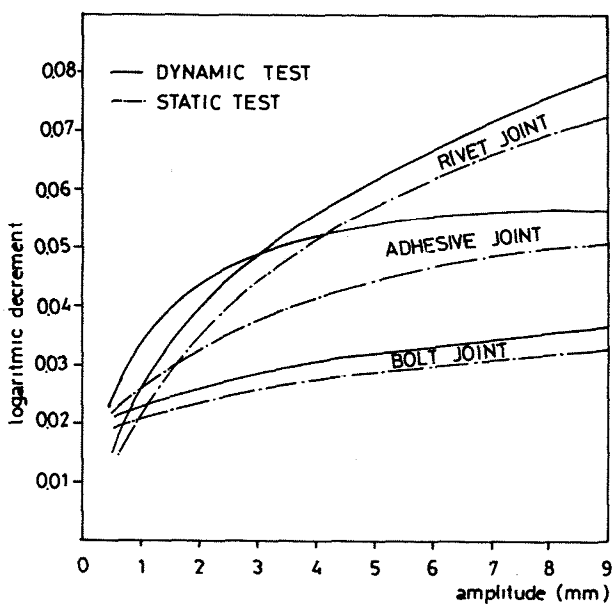

Figure 11.

The dectement oscillation dependence with amplitude is obtained in analytical form for the first form of oscillation according to equation (10),without external excitation forces,and it is shown in figure 13.In the same figure good agreement with experimental results can be seen,except for the adhesive model. The

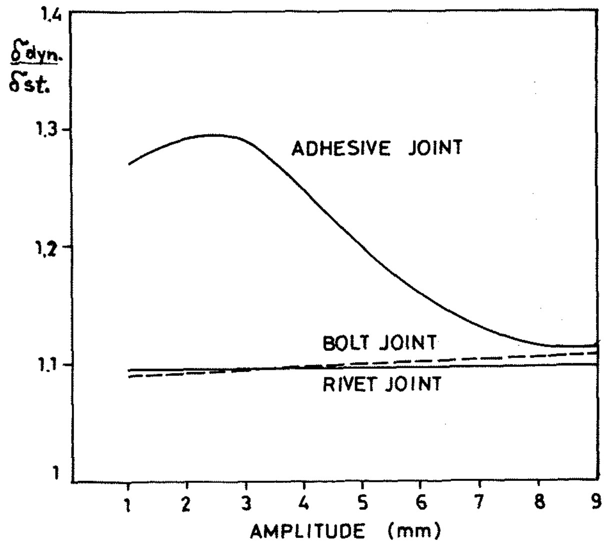

Figure 12.

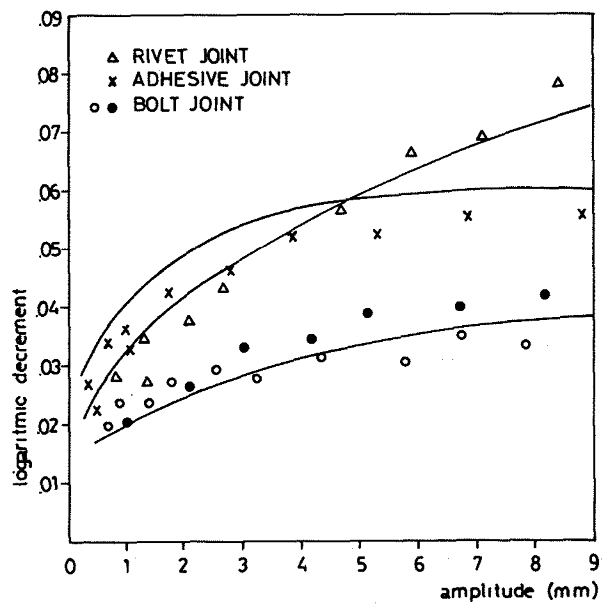

Figure 13.

disagreement for the adhesive model can be explained by a higher dependance of the ratio of logarithmic decrements for static and dynamic testing with amplitude according to fig.12.

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of oscilations on the damping characteristics were examined.

The logarithmentic decrements of free oscillations are obtained by the equation:

$$\delta = \frac{1}{n} \ln \frac{Z_k}{Z_{k+n}} \tag{13}$$

The relevant amplitudes of displacement and stress were taken as:

$$\frac{1}{2}(Z_k + Z_{k+n}) \qquad \frac{1}{2}(\sigma_k + \sigma_{k+n}) \tag{14}$$

The number of cycle changes **n** were changed fróm 1 to 50,and in the case of n=10 it was experimentaly i onfirmed as the optimal one giving the lowest distortion of the results,which is in accordance with ref.(15)

In the analysis of the forced oscillations by the method of resonance peak appropriate logarithmetic decremants were ob obtained from reference(9a,16)for the various amplitudes for the same form of oscillation.The results, obtained by various methods of dynamical testing, differ very little one from another.

The variation of the logarithmetical decremant in the function of the amplitude of oscillations for the adhesive,rivets, and bolted models is shown at figure 9.

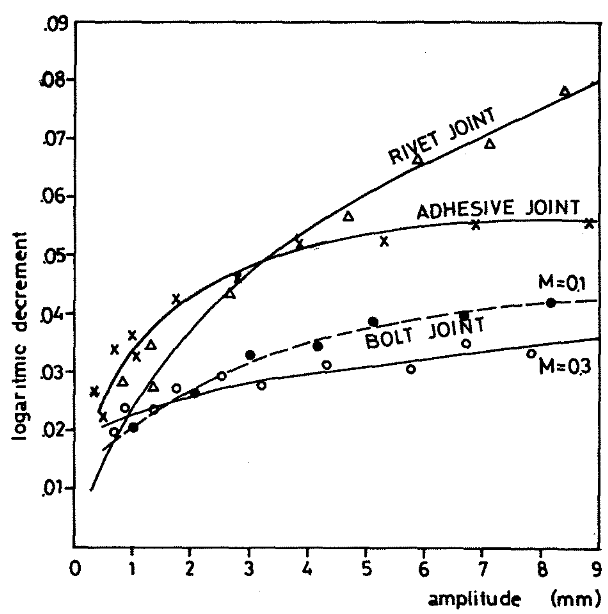

Figure 9.

A great dependance of damping in the rivet model with amplitude of oscillations is noticeable,while for the adhesive model the high gradient of logarithmic decrement is characteristic at the lower amplitudes and it is smaller at the higher amplitudes. The bolted models have the lowest decrement The logarithmic decrement dependence on bolted torque tightening with different oscillation amplitude is given in fig.10.

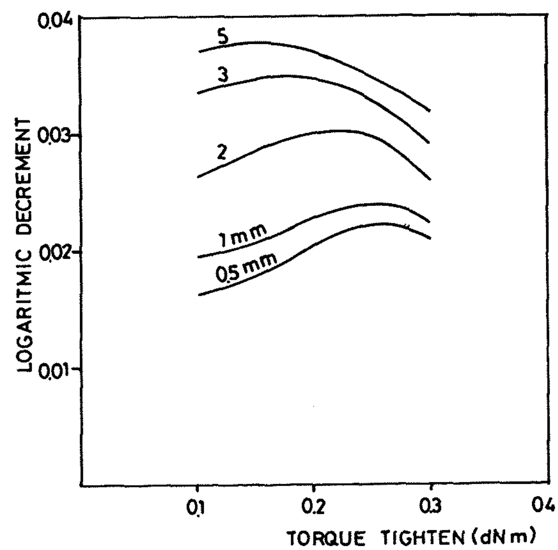

Figure 10.

For all the amplitudes an optimal torque is obvious:it gives the largest damping. It is also obvious from figure 10 that different amplitudes of oscillations correspond to different torque tightenings, which give the maximum of damping.

In figure 11,acomperatative presentation of oscillation decrement changes for static and dynamic testing is given.It is noticeable,in fig.12,that the difference ratio between logarithmic decrement which is obtained for static and dynamic testing depends very little on the amplitude for the rivet and bolted joints,while for the adhesive joints this ratio depends a great deal on the amplitude.

According to the above mentioned it can be conclued:

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