Discussion

Practical Aspects of Turbine Cylinder Joints[1]

George L. Bascome.[2] The author's remarks relative to several factors which must be considered in the design of the longitudinal joint of a steam-turbine casing, subjected to high pressure and temperature, apply as well to a bolted flanged joint for steam piping.

Some years ago the writer was engaged in the testing of various types of flanged joints under consideration for a 1500-lb 900 F installation, for the power station of the Houston Lighting and Power Company. These tests were conducted jointly by the engineers of the utility and the fabricating companies responsible for the installation of the piping. At that time little was known concerning the behavior of the steel at high temperatures, so that in conducting these tests nothing was taken for granted, and careful micrometer measurements were made of bolts and flanges before and after each test. Thermocouple attachments were made at various locations about each joint, in order that temperature changes might be noted between the several parts.

At the beginning, it was felt that a type of joint, with ground faces, without a gasket would prove most satisfactory, but within a few hours the joint leaked. A close examination showed that due to the high temperature a slight warping of ground faces had occurred. This indicated that a metal-to-metal type of joint for this class of service should be thoroughly stress-relieved prior to finishing the surfaces. It is not recalled that the author made mention of this fact, but I presume that the turbine casings are stress-relieved prior to machining, in order to prevent distortion from heat under operating conditions.

In order to determine the best bolting material and a safe working stress, a crude but satisfactory relaxation-test method was employed. The bolts were stressed to 40,000 lb per sq in., between the faces of a cylindrical sleeve, or cage, carefully micrometered, and then submerged for a week in melted lead at 900 F. At the end of the period the increase in bolt length over the initial length was noted. This information determined the bolting steel best fitted for this service and also the permissible stress which might be employed. Micrometer measurements of the bolts during erection insured that proper unit pressure was provided to the joint faces.

The author stated that a net unit pressure between joint faces of the turbine casings of three times the internal pressure was necessary to maintain a tight joint under loaded conditions. The tests on pipe joints just referred to, indicated that the net pressure need be between two and three times the internal pressure when the joint was under pressure, including calculated bending moments. It is surprising that these early tests made with limited facilities should agree so closely with the author's findings.

It might be stated that the final design of the joint for this installation included a thin gasket to care for any slight warping of the joint faces, with a calculated gasket area, such as to provide the suitable net unit pressure with a safe working stress in the bolting material. It was stated that no leakage occurred during the first two years of service in any of the joints so designed and erected.

In conclusion it seems that if due consideration is given to all factors, a bolted joint can be designed to meet the most exacting service, and such information as was disclosed by the author is of great value to the profession.

C. Richard Soderberg.[3] The relation between bolt pitch and flange dimensions which is given in the equation $h \geqq \sqrt{\left[d^2 + \left(\frac{P}{2}\right)^2\right]}$ appears somewhat arbitrary, although it is probably a safe working rule. The following presents a point of view which throws further light on this important question.

In a straight joint of the type shown in Fig. 3 of the paper, the flange may be represented as a double beam, each half of dimensions $b \times h$ held together by a series of concentrated forces P produced by the bolts. See Fig. 1 herewith. The material between the neutral axes of the beams is elastic under the influence

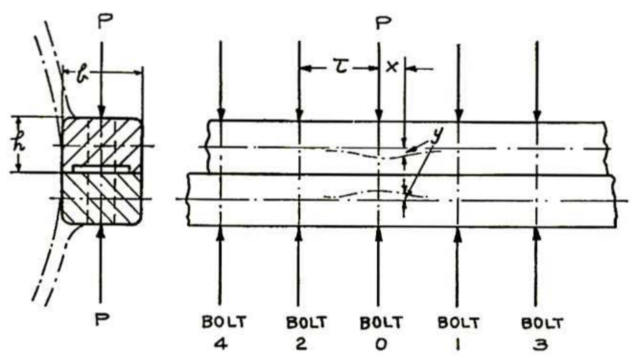

Fig. 1 Straight Joint With Concentrated Bolt Loads

of the compression forces. Considering one of these beams, therefore, it may be regarded as supported on an elastic foundation, the modulus of which is

$$k = \frac{2Eb}{h} \text{ lb per sq in.} \qquad [1]$$

The influence of relief and bolt holes will be neglected, although there are no serious difficulties in taking these into account, at least approximately. If the two beams deflect symmetrically toward each other a distance y, the resulting compression load per unit length is

$$q = ky \text{ lb per in.} \qquad [2]$$

Thus, if the deflection of the beam can be expressed as a function of x, the resulting load distribution is also obtained.

The theory for a beam on an elastic foundation is well known,[4] and the results may be used to gain an insight into the present problem. Considering first the effect of a single force P in a long beam, the results of this theory give for the load distribution

$$q = q_0\varphi \qquad [3]$$

[1] By C. B. Campbell. Published in the June, 1938, issue of the Journal of Applied Mechanics, Trans. A.S.M.E., vol. 60, 1938, p. A-49.

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[3] Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass. Mem. A.S.M.E.

[4] "Applied Elasticity," by S. Timoshenko and J. M. Lessells, Westinghouse Technical Night School Press, E. Pittsburgh, Pa., p. 132, et seq.

where φ is a function of βx.

$$\varphi = e^{-\beta x} (\cos \beta x + \sin \beta x) \ldots\ldots\ldots\ldots [4]$$

$$\beta = \sqrt[4]{\left(\frac{k}{4EI}\right)} = \frac{\sqrt[4]{6}}{h} = \frac{1.565}{h} \ldots\ldots\ldots\ldots [5]$$

and

$$q_0 = \frac{Pk}{8\beta^3 EI} = \frac{\sqrt[4]{6}}{2}\frac{P}{h} = 0.783\frac{P}{h} \ldots\ldots\ldots\ldots [6]$$

This distribution is shown in Fig. 2. At about $\beta x = 2.4$, the load becomes zero, and for larger values it is negative. Negative

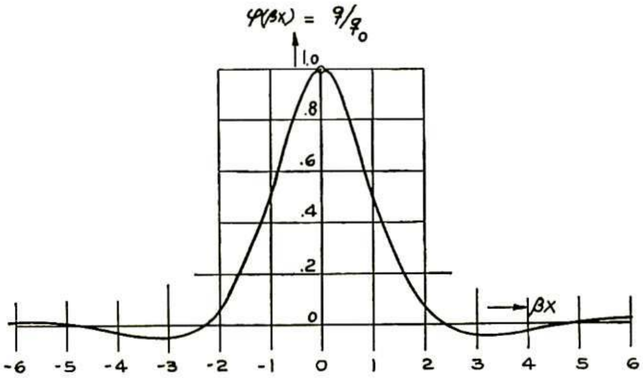

FIG. 2 LOAD DISTRIBUTION OF ONE BOLT

loads are excluded for practical reasons, but this is not serious since the case of a single bolt has only theoretical interest. The characteristic feature of the load distribution lies in the fact that it is a periodic function having a wave length

$$L = 2\pi/\beta \approx 4h \ldots\ldots\ldots\ldots [7]$$

To obtain the load distribution for a series of bolts spaced at a pitch τ, it is necessary to add a series of such functions. The convergence is rapid for the pitches of practical interest, so that only a few terms are necessary. For convenience, it is assumed that the pitch is made a certain fraction ν of the wave length, so that

$$\tau = \nu L \ldots\ldots\ldots\ldots [8]$$

Referring to Fig. 1, it is easily seen that the load distribution becomes

$$\begin{aligned} &\quad\text{Bolt 0} \qquad\quad \text{Bolts 1,3, ...} \qquad \text{Bolts 2,4, ...} \\ q &= q_0 [\varphi(\beta x) + \varphi(2\pi\nu - \beta x) + \varphi(2\pi\nu + \beta x) \\ &\quad + \varphi(4\pi\nu - \beta x) + \varphi(4\pi\nu + \beta x) + \ldots] \end{aligned} \ldots\ldots [9]$$

where q_0 represents the maximum load due to a single bolt.

Fig. 3 shows the load distributions obtained in this manner for $\nu = \tau/L = \infty$, $^3/_4$, $^1/_2$, $^3/_8$, and $^1/_4$. Fig. 4 shows the ratio of q_{min}/q_{max} plotted as a function of ν.

These results must be regarded in the light of rough approximations of the real problem, but they permit the following conclusions:

1 The longitudinal load distribution is a sensitive function of the ratio of flange thickness h to bolt pitch τ.

2 The load midway between two bolts will fall to zero if the bolt pitch is greater than $(^3/_4)L = 3h$, which corresponds to

$$h \leqq (^1/_3)\tau \ldots\ldots\ldots\ldots [10]$$

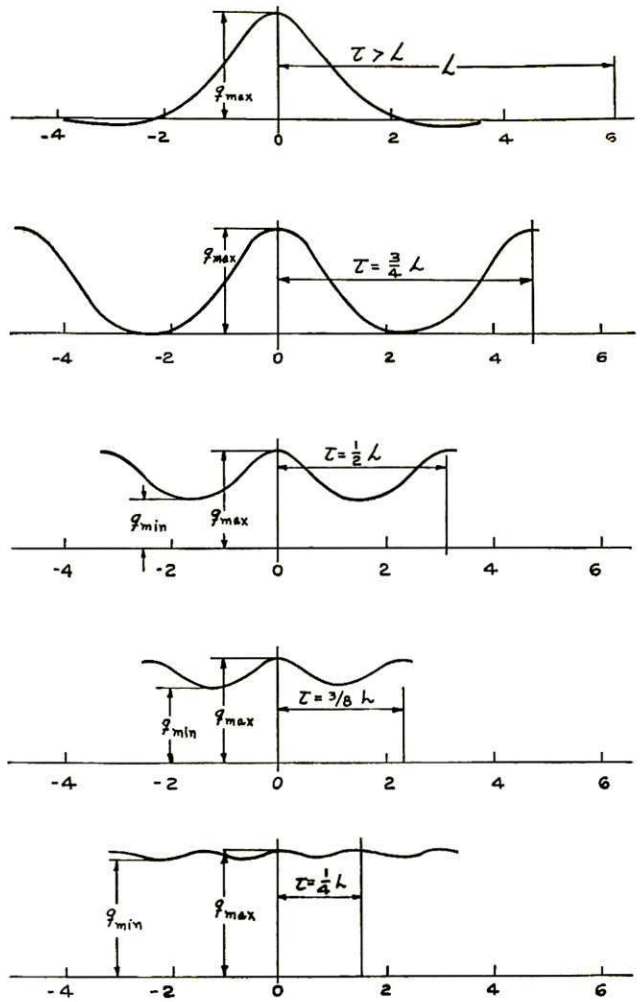

FIG. 3 LOAD DISTRIBUTION FOR VARIOUS VALUES OF $\nu = \tau/L$

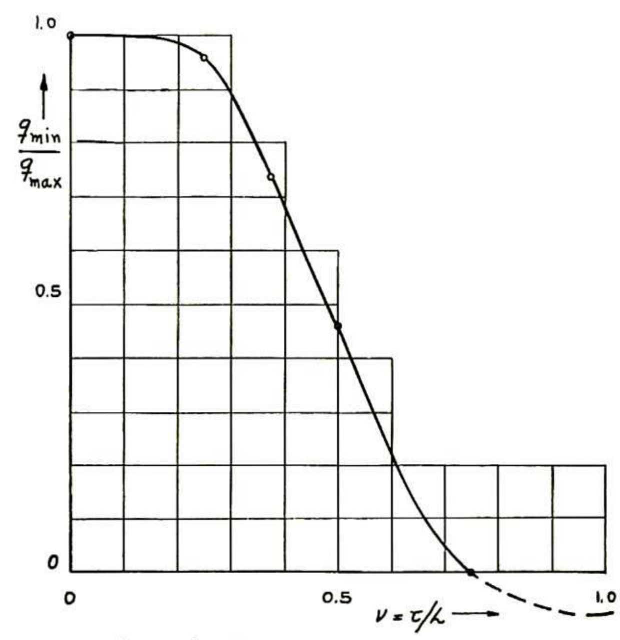

FIG. 4 RATIO q_{min}/q_{max} AS A FUNCTION OF ν

Flange thicknesses equal to or below this value must be regarded as definitely unsatisfactory.

3 In order to obtain approximately uniform load distribution along the flange, it is necessary that the bolt pitch be greater than $(^1/_4)L$, which corresponds to

$$h \geqq \tau \qquad [11]$$

A satisfactory simple rule, therefore, is to make flange thickness and bolt pitch equal.

It is important to note that the width of the flange does not influence the longitudinal load distribution. This is not true, of course, if the flange is made so wide as to have appreciable deflection over the relieved portion. The design rules proposed by the author will satisfactorily guard against this.

Author's Closure

Mr. Bascome is correct in assuming that the turbine cylinder casings are stress-relieved prior to final machining.

As stated by Mr. Soderberg, the relation between bolt pitch and flange thickness $h \geqq \sqrt{\left[d^2 + \left(\frac{P}{2}\right)^2\right]}$ is arbitrary, and is purely a rule of thumb which results in rational and well proportioned design. Combined with Mr. Soderberg's conclusion that flange thickness should be equal to or greater than the bolt pitch, we find that the distance d, Fig. 3 of the original paper, should not exceed 0.9 × bolt pitch, an entirely rational result.

Stress Model of a Complete Airship Structure[1]

L. B. Tuckerman.[2] The evident care and foresight used by the authors cannot but impress anyone who appreciates the pitfalls which may lie on the path from test to structure. It would be difficult to find any essential property of their model which has been overlooked.

The discussion of the scales N is complete, and it is particularly gratifying to find second-order effects associated with buckling considered adequately. The model girder is ingenious, with its slotted tubes so placed that advantage is taken of the slot to reduce the torsional stiffness without affecting the flexural stiffness. Finally, the spacing of the tubes in the middle solves well the problem of controlling the axial stiffness.

A nice instance of the care exercised is shown in the method used in soldering the wires and allowing for the reduced tension in the bulkhead wires due to the installation of the shear wires. In speaking of the wires, it is the hope of the writer that the authors will publish a full description of their specially designed tensometer for measuring the tension in a taut wire. The writer has seen what he believes is this instrument or one like it, and it should be better known.

The many check tests, and especially those which lead to the results shown in Figs. 14, 15, and 16 inspire confidence. It is easy to say that the precautions taken throughout are fairly evident. So they are in many cases after they have been pointed out. Unfortunately, however, much testing is done in which adequate consideration is not given to what should be evident. It is a pleasure to find an experimental investigation carried out with the attention to important detail which the authors have shown.

[1] By L. H. Donnell, E. L. Shaw, and W. C. Potthoff. Published in the June, 1938, issue of the Journal of Applied Mechanics, Trans. A.S.M.E., vol. 60, 1938, p. A-67.

[2] Assistant Chief, Division of Mechanics and Sound, National Bureau of Standards, Washington, D. C.

An Improved Method for Calculating Free Vibrations in Systems of Several Degrees of Freedom[1]

Geometric Interpretations of the Method of Successive Approximations

M. A. Biot.[2] The method of successive approximations using matrix algebra is based on the same fundamental principles as what is generally known as Vianello's or Stodola's method. Starting from an arbitrary deflection of the system, the corresponding dynamic forces are evaluated assuming a given angular frequency, for instance, $\omega = 1$. A new set of deflections due to these dynamic forces is found, and with this new set, the process can be repeated. A certain number of such processes leads toward a convergent shape of the deflection, which is the shape of the fundamental mode, and the ratio of two successive deflections is equal to the square of the fundamental angular frequency.

It will be shown hereafter that a simple geometrical interpretation exists for this method.

Consider a system of two masses m_1 and m_2 connected to each other and to a fixed base B by springs, as shown in Fig. 1. It is assumed that there are only two degrees of freedom determined by the displacements x_1 and x_2 of the masses.

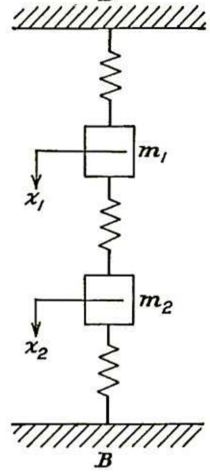

Fig. 1

Two loads P_1 and P_2 applied, respectively, to the masses m_1 m_2 produce deflections

$$\left.\begin{aligned} x_1 &= d_{11}P_1 + d_{12}P_2 \\ x_2 &= d_{21}P_2 + d_{22}P_2 \end{aligned}\right\} \qquad [1]$$

when the d's are the influence numbers. According to Maxwell's reciprocity theorem, we have

$$d_{12} = d_{21} \qquad [2]$$

The elastic potential energy under the loads P_1 P_2 is

$$W = {}^1/_2(x_1P_1 + x_2P_2) = {}^1/_2[d_{11}P_1{}^2 + 2d_{12}P_1P_2 + d_{22}P_2{}^2] \qquad [3]$$

Equations [1] of this discussion may be written as

$$\left.\begin{aligned} x_1 &= \partial W/\partial P_1 \\ x_2 &= \partial W/\partial P_2 \end{aligned}\right\} \qquad [4]$$

For the sake of simplicity let us first assume that the two masses are both equal to unity. The problem of finding the natural vibrations of our system will be solved if we know values of P_1 and P_2 such that they are proportional to the displacements x_1 and x_2 produced by these forces, that is

$$P_1/x_1 = P_2/x_2 = \omega^2 \qquad [5]$$

This follows from the fact that ω^2x_1 and ω^2x_2 are the inertia forces when the angular frequency is ω.

This problem is illustrated geometrically as follows:

Consider a coordinate system P_1 P_2. Equations [1] of this discussion define a linear transformation of the vector P of components P_1 P_2 into a vector X of components x_1 x_2. The vector X

[1] By Winston M. Dudley. Published in the June, 1938, issue of the Journal of Applied Mechanics, Trans. A.S.M.E., vol. 60, 1938, p. A-61.

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