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EXPERIMENTS IN STRESS RELIEVING CASTINGS AND WELDED STRUCTURES BY VIBRATION

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SOME EXPERIMENTS IN STRESS-RELIEVING CASTINGS AND WELDED STRUCTURES BY VIBRATION.

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INTRODUCTION.

It has become common practice in industry to stress-relieve castings and welded structures by annealing them in furnaces. If locked-up stresses exist, they relieve themselves by local stretching, upsetting, or distortion of the structure at the elevated temperatures. The equipment required for this operation is often large and expensive.

Certain structures are so large that they: cannot be accommodated in annealing furnaces locally available, or perhaps not in.

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the largest furnace available anywhere. They may be of such form and shape that the orthodox stress-relieving process would cause so much distortion as to render them useless for the purpose intended.

A possible solution of this problem is to permit them to weather or age for an extended period, during which time the natural changes in temperature and the extended "soaking" cause variations and reversals in stress, in the course of which the residual or locked-up stresses are gradually relieved.

This aging process has been extensively applied in the stress-relieving of iron castings, especially where the retention of proper shape after machining is important, as in cast-iron parts for precision instruments and machine tools. It has been more or less taken for granted by those who have used it that this aging process should continue for twelve months or more; manifestly no such delay can be tolerated in the midst of an important or urgent development or production program, and equivalent alterna-tive methods must be sought.

This article describes some experimental methods undertaken by the David W. Taylor Model Bauin and one of its contractors * in an attempt to expedite the stress-relieving and stabilizing of welded steel and cast-iron structures by vibration and by a combination of vibration and annealing.

STRESS RELIEVING BY VIBRATION.

Although there has been little opportunity to investigate the theoretical phases of the problem, and it has not been possible to uncover any published literature on this subject, there is some experimental proof that locked-up stresses in a complicated structure can be and are relieved to a considerable extent by the working which the structure receives in service.

There is, for example, the well-known experience with ship structures, immortalized by Kipling,† in which the excess stresses in certain areas are worked out and relieved by the motion of the ship in the seas which she encounters. During this series of loading cycles, in which definite working stresses are developed,

* The Kutztown Foundry and Machine Corporation, Kutztown, PA

† "The Ship that Found Herself" from "The Day's Work," by Rudyard Kipling.

these working stresses when combined with the residual stresses reach such high local values that the yield point is exceeded * and plastic flow takes place, reducing the residual stress when the working stress is removed. In the aging process, some action of this kind apparently takes place, but at a much slower rate. Whether it is the combination of the long time of action and the moderate stresses set up by temperature variation, or the slow yielding of the metal through plastic action under the influence of stresses in the elastic range applied for long periods, is not known at present. Possibly it is a combination of both.

In fact, it is considered by many that the operation of riveting in a ship that is partly welded and partly riveted goes far to relieve locked-up stresses in the welded portion of the structure, even before the ship is completed and launched. Working of the ship structure in service undoubtedly does the rest. Recent tests † have proved that in smaller structures stress relief can be accomplished by the steady application of pneumatic riveting hammers to some convenient part of the structure for more or less extended periods.

In the customary annealing process on relatively small units, the residual stresses are largely removed by lowering the yield point temporarily and permitting plastic flow to take place readily under the elevated temperatures. However, there is a reluctance among those who produce fine machinery and precision equipment which must remain stabilized for long periods to accept even the best of annealing processes as the final and complete answer to the problem of internal stabilization in metal parts.

One solution which has been proposed and on which some experiments have been made is a combination of the customary annealing process with vibration of the metal part at a relatively high frequency. If annealing cannot be used, for one reason or another, or if it is considered not adequate by itself, the vibration process is employed as a means of subjecting the part to a great number of working cycles in a short time, in an endeavor to accomplish the same end.

* This action appears to be extremely localized and has no detrimental effect upon the strength or serviceability of the structure as a whole.

† This information has been kindly furnished by Mr. E. H. Ewerts, of the Electric Boat Company of Manitowoc, Wisconsin.

GENERAL CONSIDERATIONS.

In the application of these experimental processes, involving a combination of annealing and vibration or of vibration alone, a few simple rules have been followed.

(*a*) The part has been machined or trimmed down as nearly as practicable to its finished size before the stress-relieving process was undertaken.

It was considered that some of the residual stresses, such as those in the skin of a casting, would be removed with the metal, and that the remaining metal would yield more readily to the internal forces. This is the reason for the rather general practice in industry of rough machining castings before annealing.

(*b*) The stresses in the vibrating operation were made as high as possible in an effort to produce the desired effect, but not so high as to damage the part unless a weak spot existed in it.

If a weak spot does exist, the best time to find it is before a great deal of expensive machine or hand-finishing has been done. The localized stresses in the other parts must exceed the yield point, to permit plastic flow to take place.

(*c*) The vibrating operation, and the frequency employed, was selected to suit the particular part or structure.

Setting up resonant vibration in a part will increase the effect for a given excitation, will develop higher stresses than can otherwise be produced, and will decrease the vibration time.

VIBRATION EXPERIMENTS TO STABILIZE CASTINGS.

As an example of the application of these procedures in practice, there will be described the normalizing experiments on two rather different types of iron and semi-steel castings.

The first is the group of chair castings for the support of the heavy rails for the towing carriage tracks at the David W. Taylor Model Basin, developed in 1938 and 1939. A section of one of these castings is shown in Figure 1. Alternate chair castings were 12 feet and 4 feet long. For the main rail these weighed 1695 and 530 pounds respectively; for the steady rail, 770 and 247 pounds each. They were made of high-strength iron castings, with about 30 to 40 per cent of steel scrap. Two of the chain castings installed in the tracks are illustrated in Figure 2.

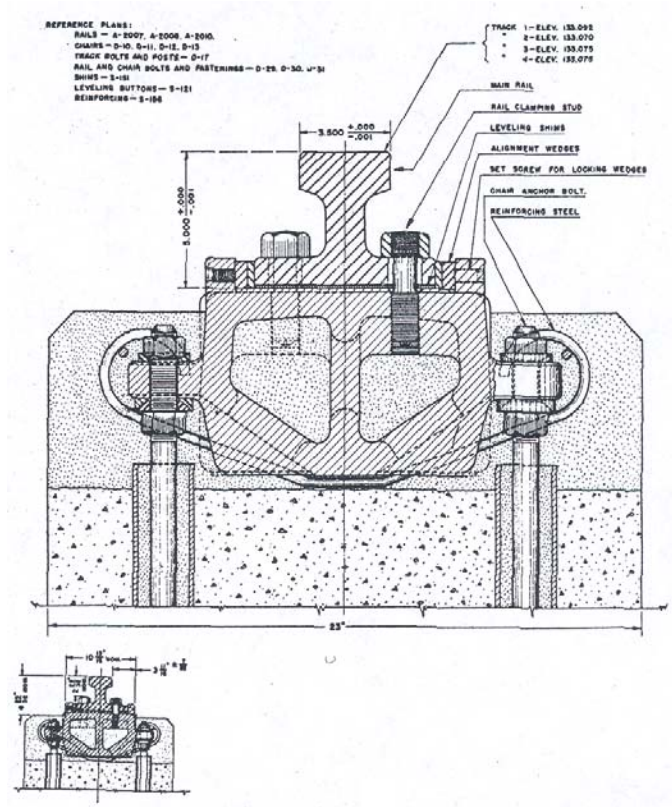

FIGURE 1.- TYPICAL TRACK SECTION THROUGH MAIN RAILS, CHAIN AND FOUNDATION FOR TOWING CARRIAGE TRACKS.

The chair is the box-shaped structure with lugs on the side. Alternate chains were 12 feet long, the remainder were 4 feet long.

The specifications for these chairs called for flat-machined surfaces on the top where the leveling shims under the rails were to bear, accurate to within 0.0015 inch in 6 feet, when measured in any direction. It was imperative that once the chair castings were machined accurately, they remain so for indefinite periods.

In an endeavor to achieve this result and to eliminate the aging procedure, the following specifications were developed:

"All chairs shall be of high-test gray iron (semi-steel), in general accordance with the requirements of Navy Department Specifications 4615b, Class B, having a minimum tensile strength of 30,000 pounds per square inch, and capable of withstanding a transverse load test of 2250 pounds with a corresponding deflection of 0.24 inch.

"It is the intent and purpose of these specifications to obtain chair castings which, when machined to a straight and true surface on top, will hold this accurate surface while being leveled and adjusted to position, and which will retain it for an indefinite time after having been grouted and ballasted in place on the concrete basin walls.

"After cleaning, the castings shall be rough milled or rough planed on the bottom of the side flanges to provide a practically continuous bearing for support during the subsequent planing operations on the top. The castings shall then be milled, planed or machine ground on the top face and on the end projections, leaving only sufficient metal to clean up to the finished dimensions after the annealing operation.

"Before annealing, the castings shall be vibrated, jarred or bumped to accelerate the removal of all residual strains due to casting. Each casting shall be subjected to at least 25 definite heavy blows or vibrations, or to vibration at high frequency for one (1) minute.* This operation may he undertaken in or by any convenient machine in the contractor's plant.

"The chair castings shall be annealed to eliminate stresses in the metal as a result of the casting operation. They shall be loaded on a car or table where they will be adequately and uniformly supported and then be placed in a suitable annealing furnace, which has arrangements for automatic temperature control and recording. The furnace shall be brought to a temperature of from 1000 degrees to 1050 degrees F. and maintained at this temperature for at least six (6) hours. The furnace shall then

*This requirement was admittedly an arbitrary one, arrived at on a basis of production requirements and the experience of a firm which had manufactured cast iron parts successfully for many years.

be allowed to cool at a rate not in excess of 1000 degrees F. in 48 hours, and the castings removed.

"After cooling, the castings shall again be vibrated, jarred or bumped to remove residual strains, as previously described."

It is to be noted that the chairs were vibrated both before and after annealing, and that all these operations took place between the rough machining and the finish machining processes. The contractor used a large molding machine for the vibrating operation. This machine was available and it was large enough to take the whole chair casting.

Before the molding machine was thought of as a vibrating appliance, it -was proposed to use one or all of the following vibration treatments on these castings:

1. Suspend them dear of the floor and strike them with a maul, sledge or ram.
2. Roll them along the shop floor by parbuckling, i.e., by winding a rope around them and then pulling on the rope.
3. Carry them around in a springless wagon over rough pavements or roads.
4. Lift them a short distance off their supports and drop them, repeating this operation a number of times.

Although, in the days of vibration generators, these methods all appear rather crude, they are still acceptable substitutes in case a special machine is not available.

Several of the experimental, chair castings which had been manufactured by the process previously described were checked carefully at intervals for more than a year, without showing any measureable change in shape. It may of course be argued that this proves nothing, that the chair castings might either have been stabilized without the vibration treatment, or that no stress relief resulting in a change of shape took place within the year in question. It appears at least reasonable in this case, however, to give the combined vibration and annealing operation the benefit of the doubt.

The second example of the application of the vibration-annealing procedure at the Taylor Model Basin was in the manufacture

of a large surface plate and a. large layout table in 1942, by the same firm that supplied the rail chairs.

The surface plate is 12 feet 4½ inches long by 8 feet 7½ inches wide by 12 inches deep, and it weighs about 9000 pounds. It is, as shown in Figure 3, a box-shaped iron casting having completely symmetrical upper and lower surfaces and two sets of lightened vertical webs, one longitudinal and the other transverse. It is, by virtue of its designs, well adapted to preserve its shape, apart from any aging or annealing process.*

It was manufactured under specifications almost identical with those for the chair castings, and it was vibrated both before and after annealing in a large molding machine. Its performance since delivery has left nothing to be desired.

In the development of this surface plate, it was at one time hoped to make it of such shape that all the interior as well as the exterior surfaces could be machined. It was believed that still greater stability could be achieved by removing all the skin from the original casting. This design was found not practicable, but an attempt has been made to achieve the same end in the large layout table shown in Figure 4 by the removal of a considerable portion of the casting skin on the interior surfaces in a pickling operation. The holes in the edges were plugged, the casting was stood up on one side, and it was filled with a sulphuric acid solution, which was left there to work for several days.

This large table is in three sections, each 10 feet 8 inches long by 5 feet 3/4 inch wide by 9 inches thick. Each section weighed 6150 pounds when finished

The combined pickling and vibrating operations on these castings have been relatively simple, inexpensive, and easy to combine with the usual machining and annealing operations. Unfortunately, neither the time, the facilities nor the personnel have been available during the war to make precision measurements on these tables over long periods, nor have there been available for comparison, any layout tables of exactly the same design which were not subjected to vibration.

*The same remarks apply to the box-shaped chair castings shown in Figure 1 and 2.

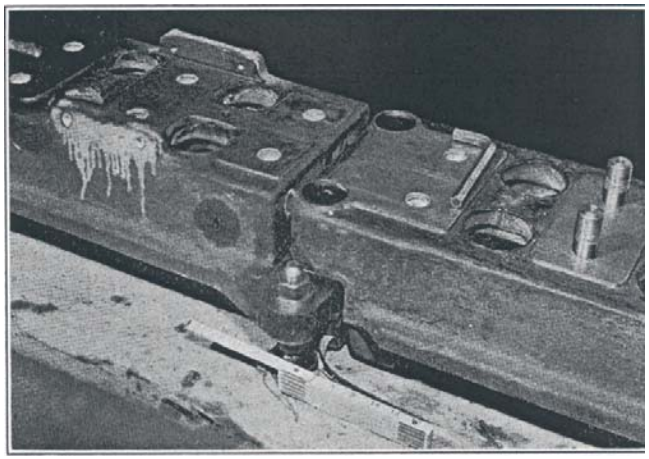

Figure 2. – Main Rail Chairs for Carriage Tracks

This is a joint between two of the main rail chairs shown in section in Figure 1. The bosses or "islands" on the top has to be machined accurately and had to remain true to hold the rails in place.

Figure 3. – Large Surface Plate.

This plate has complete upper and lower surfaces, making it symmetrical about a horizontal plane through the plate at mid-height. It is stiffened internally by a system of longitudinal and transverse lightened webs.

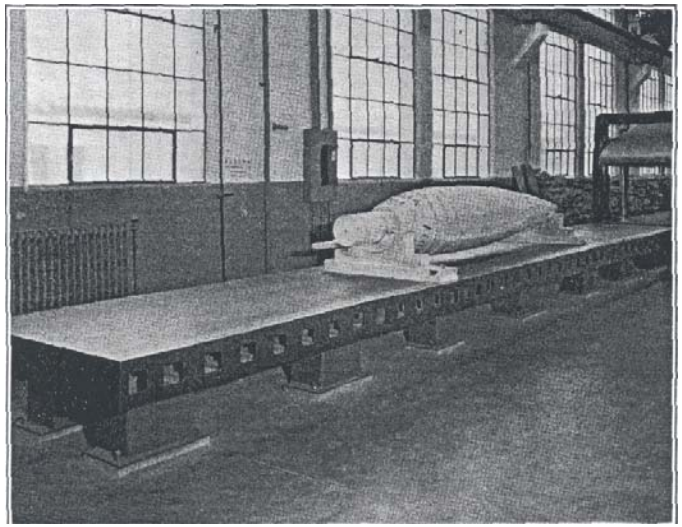

FIGURE 4. – LARGE LAYOUT TABLE, 32, FEET LONG.

This table is in three sections, bolted together. Each section is a symmetrical stiffened casting, with complete upper and lower surfaces, similar to the surface plate shown in Figure 3.

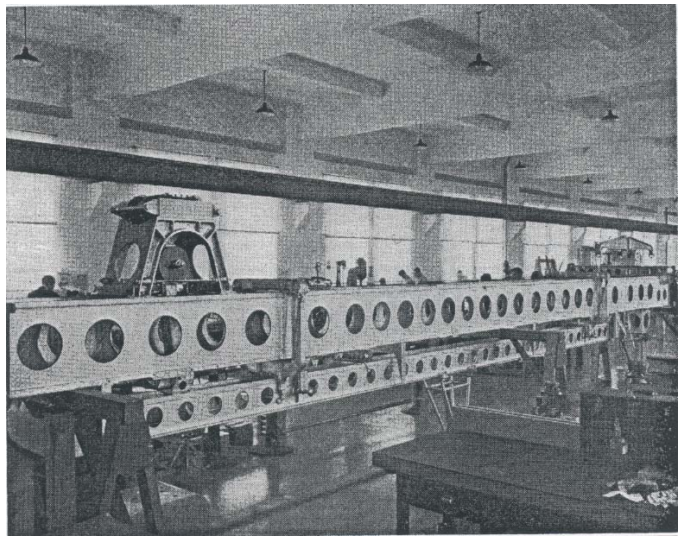

FIGURE 5. – TOWING DYNMOMETER FOR CARRIAGE I.

This upper girder, built of 1 by 1 by ⅛-inch angles and 0.109-inch sheet steel, forms the fixed foundation structure for all the precision apparatus. The lower beam built of 1 by 1 by ⅛-inch angles and 0.079-inch sheet steel is the weighing member to which the models are attached.

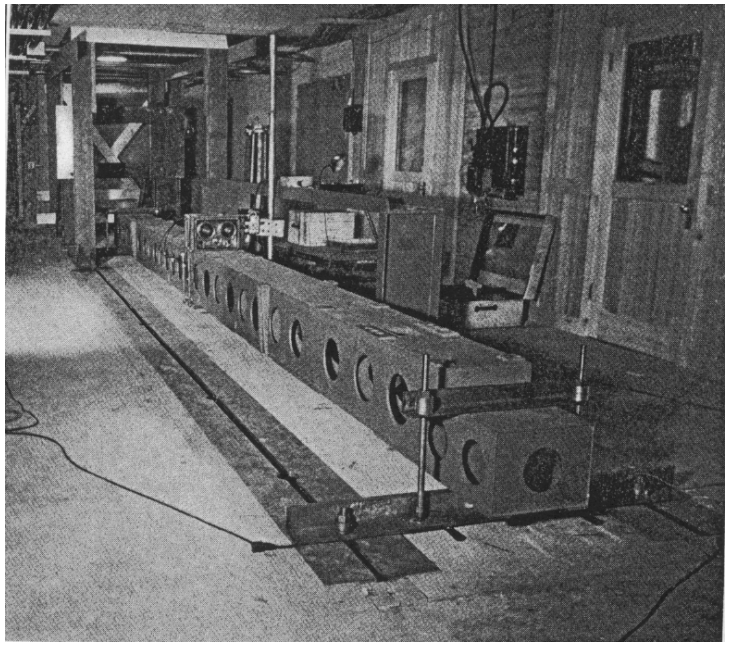

Figure 7. – Fixed Dynamometer Girder Setup for Stress Relieving Operation

The clamping arrangement simulates simple support with nodes near the ends. The maximum stress and deflection occur at the center. The vibration generator, shown mounted at the mid-length of the girder, is run at a speed slightly below resonance, which produces a steady forced vibration.

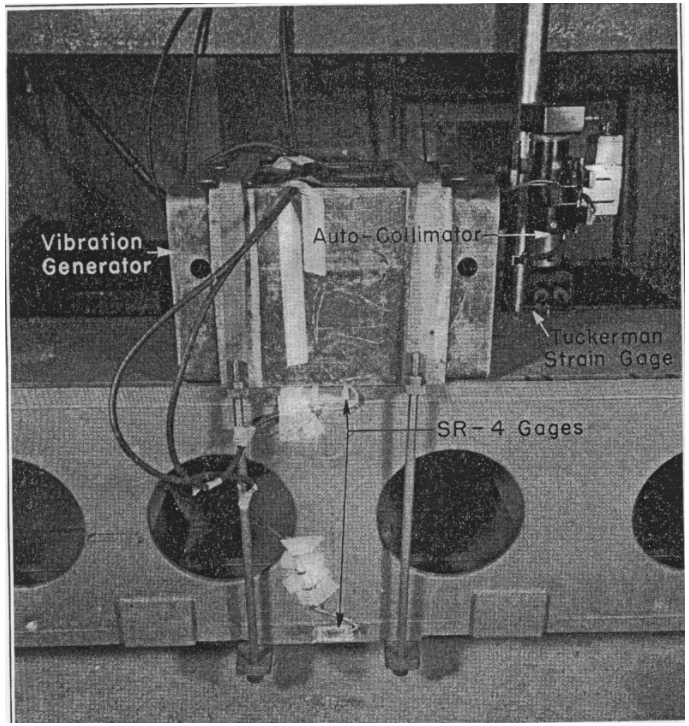

Figure 8. – Fixed Girder with Vibration Generator and Dynamic Strain Gages Mounted for Test

The vibration generator is secured to the girder by special U-clamps and tie rods. Metalectric strain pickups were cemented to the girder after the paint had been removed and the surface polished. A Tuckerman optical strain unit was secured to the top of the girder; the fasteners are not shown. The autocollimator with the photographic recording gear was attached to a stand fastened to the foundation. Note the butt weld in the side plate of the girder.

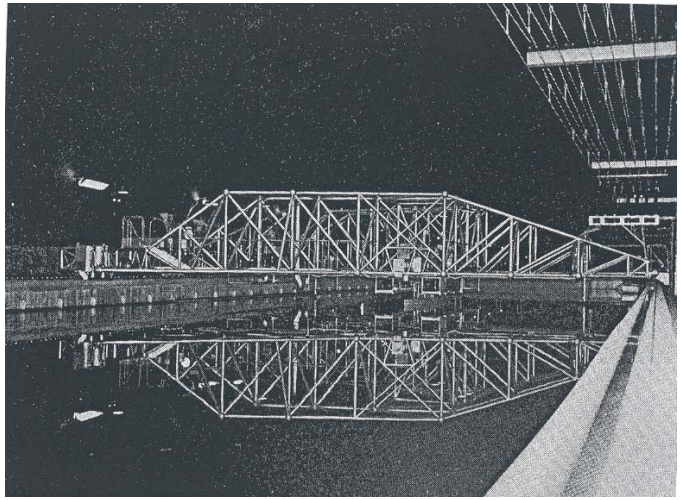

FIGURE 12.–GENERAL VIEW OF STRUCTURAL FRAME FOR TOWING CARRIAGE 1.

The height of each of the four transverse trusses is about 9 feet, and the span between the two rails if 52 feet 8 inches. The frame is of welded seamless steel tubing, with hollow cast steel spheres at the joints. The end of the fixed dynamometer girder can be seen in the center of the frame.

VIBRATION EXPERIMENTS ON WELDED STRUCTURES.

The welded tubular structure of the towing carriage designed to run on the Taylor Model Basin carriage tracks previously mentioned was too large for any annealing furnace then available. It was too important a piece of machinery to risk even a minor failure in service, yet such a failure might have been expected in so complex a structure.

This carriage frame was built to carry a large towing dynamometer made up of two large structural parts, a box-shaped foundation girder 15 inches by 16 inches in section and 29 feet 10 inches long, and a floating beam 8 inches by 8 inches by 21 feet 9½ inches long. Both of these parts had to be built with the precision of the chair castings and the surface plates previously described. Like them, they had to remain stable and true for long years of service.

A general view of the two girders assembled in the dyanometer is included in Figure 5. Transverse sections through the girders are shown in Figure 6.

The shorter of the two girders was not too long for the available annealing furnace but there was so little experience with the annealing of a structure having such thin sections, 0.079 inch, that no one was willing to do the pioneer work on this important assembly.

For the stress-relieving operation on these girders, a vibration generator was used which had been developed and built specially for work of this kind. This generator consists essentially of two parallel shafts, motor driven, geared together to run in opposite directions at the same speed and in opposite phase, and carrying eccentric weights on both ends. The small generator used for these experiments was rated at 440 pounds driving force and it had a speed range of from 250 to 3000 Rpm. Both the amplitudes of vibration and the ranges of stress were measured with modern equipment. The process is believed to be of wide general interest and it will therefore be described in some detail.

The girders were supported near the ends on two knife-edges made of structural angles which in turn were bolted down to a special vibration-testing foundation anchored to bed rock. The

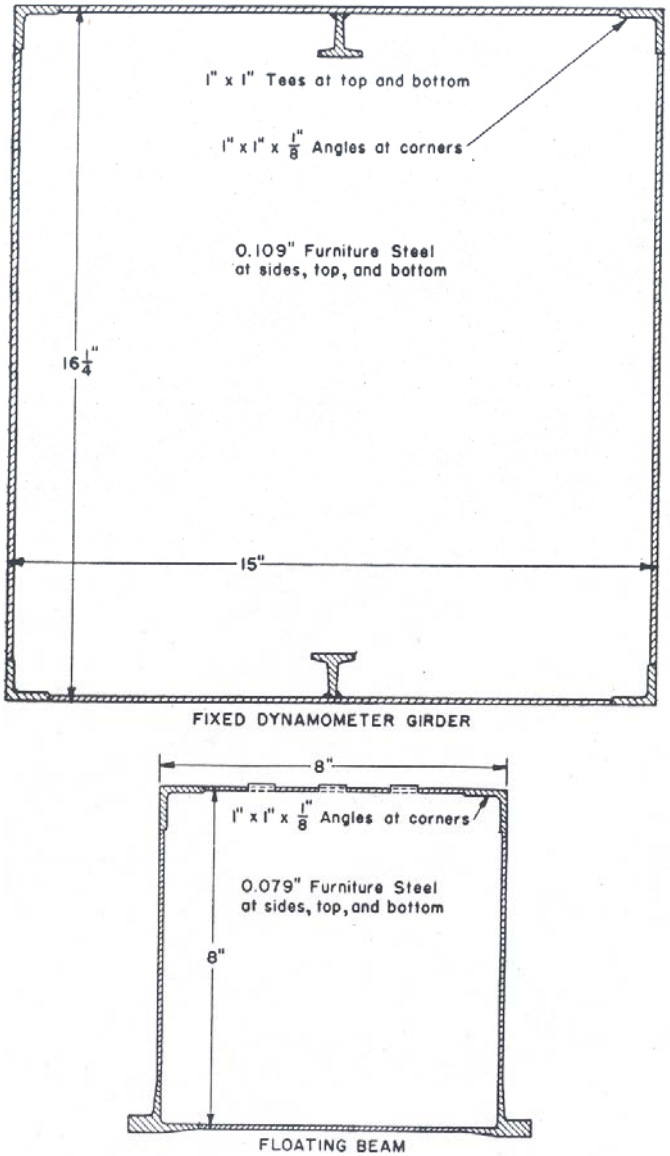

FIGURE 6.–SECTIONS OF FIXED DYNAMOMETER GIRDER AND FLOATING BEAM

The hollow box-shaped girder structures shown are reinforced at intervals with transverse bulk-heads to hold the sections in shape. The side and top and bottom members are lightened with flanged circular holes as shown in the photographs; the edges of the holes are stiffened by shallow flanges.

girders were held down against the knife-edges by smaller angles and tie rods, as shown in Figure 7. The small vibration generator was clamped on top of the girder at the center as shown in the photograph, Figure 8.

A maximum dynamic stress range* of 15,000 pounds per square inch was adopted, as a satisfactory range for stress relief, and the amplituude required to produce this stress was roughly estimated from the simple beam theory. The natural frequency of the girder in the two-noded flexural mode was also estimated on the assumption of simple support at the knife-edges with the mass of the vibration generator added at the center. An experimental check of the natural frequency was then made by applying an impulse to the girder and tuning a calibrated reed† to the frequency of the resulting vibration. A further check was made ofthe relation between static stress and deflection by strain gages and dial micrometers.

The vibration generator was then adjusted to an eccentricity sufficient to produce the desired amplitude at a frequency slightly below resonance. On the low side of resonance conditions are stable; that is, the speed fluctuations encountered here for slight changes of voltage are not as wide as those on the high side of the resonance curve. When the amplitude reached the value estimated to give the desired dynamic stress in the girder a check of the stress was made with both metalectric (wire resistance) and optical strain gages set up for dynamic measurements. The vibration generator was then left running at constant speed for a period of 8 hours.‡

VIBRATION TEST PROCEDURE ON WELDED BOX GIRDERS.

The general procedure was as follows:

First, to calculate the elastic constants of the box girders considered as simple beams.

Second, to estimate their natural frequencies when simply supported at the ends with the mass of the vibration generator added at the middle.

*The term "range" as used here covers all values between the specified stress in tension and the stress in compression.

†A Westinghouse reed vibrometer was used.

‡Again this figure was an arbitrary one, arrived at by a consideration of production requirements and some ten years of experience in vibration work.

Third, to estimate the amplitude of vibration required to produce the selected stress range.

Fourth, to check experimentally the static stress produced by a deflection equal to this amplitude.

Fifth, to check experimentally the relation between dynamic stress and amplitude.

Sixth, to determine the form of resonance response of the girder, from which its damping could be estimated.

Seventh, to vibrate the girder for 8 hours or more at the required amplitude and frequency.

The following constants necessary for stress and frequency estimates were first calculated from the dimensions of the girders; see Figure 6.

Constant	Fired Girder	Floating Bean,
Area of transverse section, square inches	7.33	3.75
Distance from center of gravity of area to bottom, inches	8.0	3.13
Moment of inertia of area with respect to center of gravity axis, inches	310.0	38.2
Radius of gyration of area about hodzontal axis through center of gravity of section, inches	6.51	3.19
Distance from center of gravity to extreme fiber, inches	8.0	4.99
Section modulus, inches^3	38.7	7.65

The natural frequency of the fixed girder supported on knife-edges over a 28-foot span with the mass of the vibration generator added at the center was estimated by the formula

$$f_1 \cong 21.7 \sqrt{\frac{EI}{\left(W + \frac{17}{35} wl\right) l^3}}$$

where f_1 is the frequency in cycles per second

E is Young's modulus in pounds per square inch

I is the moment of inertia of cross section* in inches^4

W is the weight of the vibration generator in pounds

w is the weight of the girder per unit length in pounds per inch

l is the length of the span in inches.

Substituting numerical values in this formula gives

$$f_1 \cong 21.7^2 \frac{30 \times 10^6 \times 310}{(140 + 339)\,(336)^3}$$

$$f_1 \cong 15.4 \text{ cycles per second} = 924 \text{ cycles per minute}$$

A preliminary estimate was made in the same manner of the amplitude required to produce the desired stress by assuming the girder to deform into a half-sine wave during vibration. On this assumption the maximum stress is given in terms of the deflection at the center by the formula

$$\sigma = \frac{\pi^2 Ec \Delta}{l^2}$$

where E is Young's modulus in pounds per square inch

c is the distance from the neutral axis to the extreme fiber in inches

Δ is the deflection at the center in inches

l is the length between knife-edges in inches.

Hence to obtain a stress range of 15,000 pounds per square inch or a single stress amplitude of 7500 pounds per square inch the single amplitude required at the center of the girder is

*I is the moment of inertia of the full section, because the lightening holes were near the neutral axis. Because of the stiffening effect of the flanges around them, the presence of the holes was neglected.

$$\triangle = \frac{l^2 \sigma}{Ec\pi^2} = \frac{336 \times 336 \times 7500}{\pi^2 \times 30 \times 10^6 \times 8}$$

or

$$\triangle = 0.36 \text{ inch}$$

The acceleration corresponding to a single amplitude of 0.36 inch at a frequency of 924 cycles per minute is approximately 8½ g. This value was near the upper limit for the vibration generator available but was not too high to be ruled out. Actually when the girder was clamped down with the knife-edges 28 feet apart and the vibration generator was mounted in the center the frequency was found to be 830 cycles per minute. The reduction from the calculated figure of 924 cycles per minute was probably due to one or all three causes, the flexibility of the knife-edges, the effect of the lightening holes, and the incipient instability of the side walls in compression.

When static loads were applied to the fixed girder it was found that a load increment of 554 pounds at the center produced a stress increment of 925 pounds per square inch at a point 11 inches to one side of the center, equivalent to about 990 pounds per square inch in the center, with a deflection of 0.054 inch. These stresses were measured by both Huggenberger extensometers and Tuckerman optical strain gages. If the static stress-deflection ratio held in the dynamic case this would indicate that the single amplitude required for a stress amplitude of 7500 pounds per square inch would have been 0.41 inch. When the girder was set in vibration it was found that a single amplitude of 0.375 inch gave the required stress amplitude. This very closely approximated the value of 0.36 inch estimated for a half-sine wave, which is the theoretical dynamic form where no concentrated mass is added to the beam.

The dynamic stresses were checked both by dynamic Tuckerman gages and SR-4 metalectric strain gages; Figure 8 shows both types of dynamic gages in position on the fixed girder. The Tuckerman gage consists of a strain unit with a rotatable knife-edge prism and an autocollimator for visual reading or photographic recording. The rotatable knife-edge forms an optical lever. The light from an illuminated slit in the focal plane

of the objective of the autocollimator emerges from the objective in the form of parallel rays which are reflected back into the autocollimator by the strain unit, but at an angle which varies with the unit strain. If the strain is dynamic the image of the slit will move back and forth in the focal plane of the objective, where it may be observed with an eyepiece or may be photographed. Both photographic and visual methods were used. A sample photographic record of the dynamic strain due to vibration of the girder is shown in Figure 9.

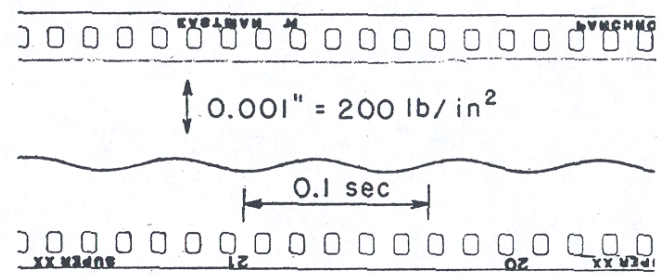

FIGURE 9. – STRAIN RECORD DURING VIBRATION TEST

This record was taken with a dynamic Tuckerman gage on the fixed dynamometer girder.

The SR-4 gages contain fine wins whose resistance varies with strain. The unit is cemented directly to the stressed member as shown in Figure 8. When current flows through these elements any change in the resistance due to changes in dynamic stress causes a change in the potential at the terminals, which may be amplified and impressed on a cathode-ray beam or measured directly with an electronic voltmeter.

Finally, the eccentricity of the vibration generator was reduced to 10 degrees and the machine was operated through the range of resonance. Amplitudes were observed with a General Radio vibration meter. This is a direct-reading instrument consisting of a piezo crystal pickup unit, an amplifier, a rectifier, and a meter on which amplitudes are given directly. The resonance curve so obtained is shown in Figure 10.

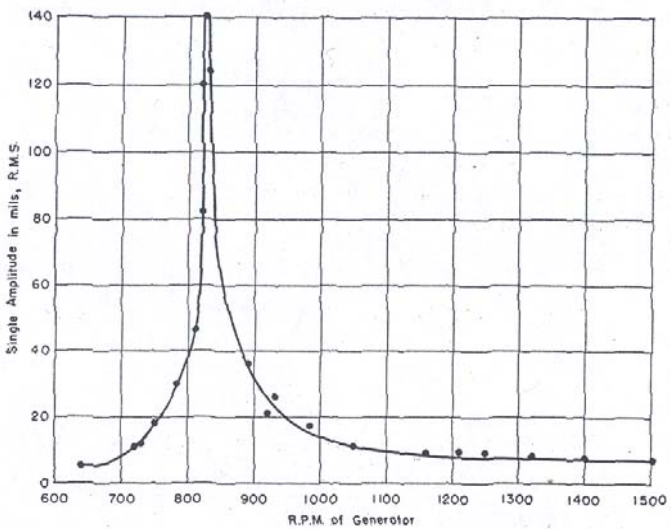

FIGURE 10.-RESONANCE CURVE TAKEN ON FIXED DYNAMOMETER GIRDER.

The amplitudes were measured at the center of the girder. The RMS amplitude = $\frac{\text{(peak single amplitude)}}{\sqrt{2}}$

The damping of a structure can be estimated very simply from the resonance curve obtained with a driving force increasing as the square of the frequency, as in the vibration generator, from the approximate formula

$$\frac{C}{C_c} = \frac{n}{4} \quad \frac{N_2^2 - N_1^2}{N^2_{max}}$$

where C/C_0 is the ratio of the actual damping constant to the critical value and n is the ratio of the amplitudes at frequencies N_1 and N_2 to the maximum amplitude occurring at N_{max}. This formula gave a value of 1.6 per cent of critical damping for the fixed girder. An ideal system of one degree of freedom with 1.6 per cent of critical damping has a resonance magnification factor of 31, which means that the single amplitude at resonance will be 31 times the deflection that would exist under the same driving force acting statically. The girder differs from the ideal

system in that its effective dynamic spring constant is not the same as the static constant, because the elastic line has somewhat different forms in the two cases. The driving force of the vibrator at resonance in this instance was 39 pounds and static loading measurements showed that this load would produce a static deflection of 0.00038 inch. The vibration amplitude was 0.14 inch, which indicated a resonance magnification of 37.

The girder was then vibrated for 8 hours at a single amplitude of 0.375 inch and a frequency slightly below resonance. The fact that no change in the natural frequency of the fixed girder was observed after 8 hours of vibration and that the internal damping remained very low was taken as an indication that no welds had failed. No cracks in the welds were observed. How much yielding of the material took place in regions of high stress could not be determined.

A similar procedure was followed with the floating dynamometer beam. This was supported on an 18-foot span. The observed natural frequencies were 1310 cycles per minute without the vibration generator mounted, and 870 cycles per minute with the vibration generator. The machine was run for 8 hours at

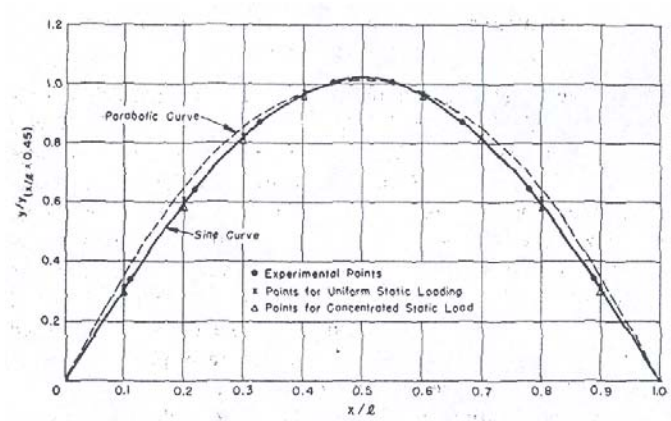

FIGURE 11.-COMPARISON OF VIBRATION PROFILE OF GIRDER WITH THEORETICAL DEFLECTION CURVES OF A SIMPLE BEAM.

The deflection at x / 1 – 0.45 is set equal to unity in each case.

840 cycles per minute with a maximum dynamic stress range of 15,000 pounds per square inch and a double amplitude of 0.4 inch. The calculated stress for a deflection of 0.4 inch, assuming that the beam deformed into a half-sine wave, is 12,300 pounds per square inch. The amplitude profile of the entire length of beam during resonance was determined experimentally by the use of a small vibrometer. In Figure 11 this form is compared with the deflection curves of a simple beam for the two most common types of static loading, and with the half-sine wave which is the ideal case for free vibration with no added mass. These curves are made to coincide at points 0.05 of the length from the center; these are the points where the deflection was actually measured. The actual deflections fall just about on the half-sine wave, which is midway between the theoretical values for uniform and concentrated static loading, so that any one of these assumptions can be safely used in estimating the relation between stress and amplitude in such a case. As the ratio of the mass of the vibration machine to that of the beam increases, the formula for concentrated load becomes the best approximation.

EXPERIMENTS ON STRESS RELIEVING OF TOWING CARRIAGE FRAME.

The largest and most complicated structures on which the vibration method of stabilization has been attempted by the Taylor Model Basin are the welded tubular frames of the two large towing carriages; one of these is shown in Figure 12. These frames are about 68 feet long, measured parallel to the tracks, 56 feet wide, and 9 feet high. While they are made of nine sections bolted together, for convenience in transportation from the contractor's plant and assembly in the Basin building, the bolted joints are very rigid and are considered the equivalent of solid material. These joints are made up in each piece of tubing *before* that piece is welded into the frame.

Before any vibrating was done on the frame, deflection measurements were taken, first with all the construction supports in place, under the whole frame, and then with the frame supported under the driving wheel and steady wheel locations only, where the

reactions would come in service. The deflection at the center was found to be about 0.1 inch, due to the weight of the structure alone. This result is approximate because it was not certain that in the first condition the frame was uniformly supported all around.

A check was then made to determine how much deflection would be caused by adding a 170-pound dead weight, about equal to the weight of the vibration generator and its clamps, on the member to which the generator was to be attached. The deflection for that load was about 0.0035 inch. On adjacent members the deflection varied from 0.0035 inch to 0.0025 inch, which indicated that the member to which the generator was attached was stiff enough at its joints to transmit most of the vibration to the structure as a whole.

Deflections in the center, in the dynamometer girder position, were measured again when a dead load of 3000 pounds was placed in the dynamometer bay to simulate the weight of the dynamometer and personnel, and of the loads the structure would carry elsewhere when in operation. This deflection at the center of the carriage frame due to increase of load was about 0.1 inch, making a total deflection of about 0.2 inch.

The carriage frame, supported at the points where the drive wheels and steady wheels would later be attached, was then vibrated with the small generator mounted in the center of the frame. A natural frequency of 600 cycles per minute was obtained and the maximum double amplitude at the center of the span was about 0.05 inch. This resulted in an increase in stress of about one-eighth of that due to the static load alone, a rather small value to be used in an effort to relieve excess locked-up stresses in the structure but all that could be obtained with the equipment then available.

The vibration generator was operated at resonance for a total period of about 15 hours, with no change in the resonant frequency.

This operation on the first carriage was repeated some three years later on the frame of Towing Carriage 2, built to the same plans. The natural frequency and double amplitude were found to be the same as for the frame of Towing Carriage 1, but be-

cause of other urgent war projects which required the use of the vibrating equipment, the carriage frame was vibrated for a total period of only about 10 hours.

On paper, the low stress range and the relatively short period of vibration, 2 days as compared to an expected life of 50 years, appear quite inadequate. Actually, however, the frame members of Carriage 1 vibrated to a greater amplitude in one cycle of the vibration test than they have been observed to move, at resonant frequency, in more than two years of hard service. There has been not the slightest sign of structural weakness in that time. Again, however, there has been no similar structure, not vibrated, available for comparison.

DISCUSSION.

On the welded structures described, dynamic stress data were taken during the vibrating operation only on the dynamometer fixed girder and on the floating beam. They were taken here merely to furnish a safe guide in applying vibratory loads, and not for the purpose of stress analysis. In general the stresses observed agree with elementary beam theory except as noted previously. However, stresses were measured only on the top and bottom angles, where maximum values are to be expected, and at one point near the center of the beam.

While the damping was very low on the girder and beam at the amplitudes encountered it would not have remained so with increasing amplitude. In view of the high resonance magnification involved, of the order of 30 or more, it might be feared that damage would occur in such a method of stress-relieving owing to accidentally running through resonance. However, there is little danger of this because the damping will increase rapidly as the average, stresses approach the elastic limit. If the endurance limit is exceeded there is the possibility of a fatigue failure, but this would occur only after a large number of stress cycles.

It was not practicable in any of the cases described here to measure the amount of stress relief actually obtained. On test specimens used to indicate and measure this stress relief the procedure calls for the more or less destructive process of drilling holes or milling slits in the specimen; this cannot be done in the

testing of parts which are to be used in working structures. Strain gage readings might have been taken at selected stations before and after vibration. This was not done, but observations were made on the general distortion of the structures after the various methods of stress-relieving were carried out;

The shaking and bumping processes described subjected the structures so treated to variations of stress beyond those which would be encountered in subsequent service. It was considered that if the parts or structures showed no large distortion due to this process they would not distort in actual subsequent operation. For instance, the carriage structures were vibrated when loaded to the full service static load and in the fundamental mode of vibration which would occur in actual operation. While the amplitude of the superimposed dynamic stress was only 13 per cent of the static stress it was more than actual operation would put into the structure. Since no observable distortion took place it was assumed that none would take place in actual operation. A similar line of reasoning holds for the dynamometer beams which were dynamically stressed far more drastically than the carriage structure without observable signs of distortion.

CONCLUSION.

The process of vibrating parts and completed structures may or may not relieve stresses, but it increases confidence in the stability of the shaken part. The evidence of achievement is negative but the feeling of safety is positive, and this feeling accumulates with compound interest as the years go by and the structures continue to carry, their burdens without distress or complaint.

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