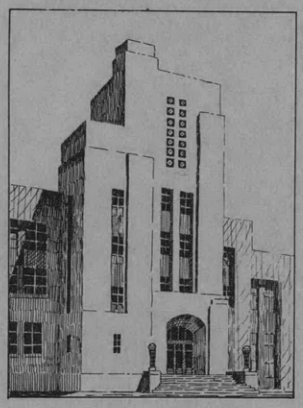
THE DAVID W. TAYLOR MODEL BASIN

UNITED STATES NAVY

DYNAMIC STRESS MEASUREMENTS

BY DR.-ING. S. BERG



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DYNAMIC STRESS MEASUREMENTS

(DYNAMISCHE SPANNUNGSMESSUNGEN)

by

Dr.-Ing. S. Berg, VDI, Kiel

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DYNAMIC STRESS MEASUREMENTS

Presented at the meeting of the Committee on Trials and Tests of Machine Parts as a Basis for Design at the VDI Trials and Tests Convention, 1 December 1936.*

In most cases, the loads on structural parts in operation can be determined only by dynamic strain measurements. The apparatus developed for this purpose by the Deutsche Werke, Kiel, makes use preponderantly of mechanical-optical means for measuring strain. The stress is then determined from the strain by Hooke's Law. A number of strain gages and their possible uses are described.

The actual loading of structural parts can be determined mathematically only in cases of simple support, and frequently tests are necessary. Two principal methods are available for this: strength tests of entire structural parts, particularly vibration tests, and stress measurements of models or completed structural parts. The vibration test indicates the point of maximum loading, but supplies no information as to whether the loads acting on other points of the structural element may be unnecessarily low, i.e., whether the quantity of material used is unnecessarily large. On the other hand, it is possible by means of stress measurement thoroughly to determine the entire zone of stress.

Every test of materials should be so carried out that the loading of the material will correspond as closely as possible to operating conditions. To this end it is necessary to know the total stress or strain condition of the completed structural part in operating conditions. In the case of complicated parts and involved action of forces, such as occur in restrained deformations, for example, it is frequently impossible to obtain this knowledge by intuition or reasoning. This gap can be closed by dynamic stress measurement. Its chief use, therefore, is found where in addition to the stresses, the forces and restrained deformations as well, are unknown as to kind, origin and magnitude.

DETERMINING STRESSES BY STRAIN MEASUREMENTS

Assuming validity for Hooke's Law, the stresses can be calculated from the elongation of the structural part at the test station, so that the measurement of stresses is equivalent to strain measurement. When the stress is constant in time, we have the case of loading at rest. Usually, however, variable loading occurs. This may vary in time either periodically or non-periodically, cyclically or transiently. In general the alternating load is superposed upon a static base load. Extensometers should therefore be designed to permit measurement of any given stress curve as far as possible.

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* See Zeitschrift des Vereines deutscher Ingenieure, vol. 80 (1936) p. 1555. The papers presented at the convention have been assembled in the book "Trials and Tests" (Prüfen und Messen), a report on the Scientific Convention of the VDI on 1 and 2 December 1936, Berlin 1937.

The first dynamic strain measurements were probably undertaken at the turn of the century in bridge building, where they likewise served less to measure actual stresses than to investigate the forces actually set up and the impact factor. Although it was not until later that mechanical engineering and other branches of technic adopted strain measurements in practice, this was because their structural forms are usually less susceptible to mathematical measurement than are those in bridge building, which are designed in accordance with simpler mathematical principles. In addition, the difficulties of carrying out tests, which are already very considerable in mechanical engineering, because of the smallness of the strains, are greatly increased in structural engineering since the base lengths of the instruments and consequently the deformations must be smaller by about one order of magnitude, for various reasons.

REQUIREMENTS OF DYNAMIC STRAIN GAGES

E. Lehr has reported on a number of dynamic strain gages preferred for tests of steel structures.* The instruments described by him, however, are inadequate because of excessively low natural frequencies, awkward dimensions or too low sensitivity in tests of machines and similar structural parts. The strain gages suitable for this must satisfy the following requirements:

The natural frequencies of the instruments must be as high as possible in order to be able to register loads of high frequencies and intermittent loads.

The base length should be kept as small as possible, since the stresses frequently vary greatly even in closely neighboring points, and a large base length would yield only a mean value of unequal stresses over the base length.

The necessity of making tests in places of difficult accessibility such as inside angles and notches, requires small over-all dimensions.

The instruments must have large magnification, so that in spite of a small base length wide deflections will be obtained.

The high accelerations demand that weight be cut as much as possible, so that the instrument can be firmly attached even to moving machine parts by the clamping devices. The clamps,, however, cannot apply unlimited forces, since the knife-edges of the instrument usually will stand only limited pressures and the clamping device itself is subject to the same acceleration as the instrument.

It is essential to the practical usefulness of an instrument, finally, that no specially trained engineers be required for its operation, that it be easy to manipulate, and that errors can be easily recognized.

According to these principles, strain gages have been developed by the Deutsche Werke, Kiel, for use in their own shops, which indicate the strain chiefly by mechanical-optical means. Tests with these instruments are limited, naturally, to

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* E. Lehr, Maschinen Bau, vol. 10 (1931) p. 711.

structural parts which can be illuminated during operation. For covered structural parts an electrical strain gage was developed.

STRAIN GAGES WITH OPTICAL LEVER READINGS

The strain gage first constructed, Figure 1, consists essentially of two main members, *a* and *b*, provided with knife-edges, which, when the base length is changed, move in contrary directions, rotating a thin rocker *c* between them while doing so. To this rocker is fastened a small mirror *d*, illuminated by an arc-lamp through two adjustable lenses, which throws a beam of light on a screen or a recording drum. The two main members maintain contact with each other by a three-point bearing formed by the two supporting ends of the rocker *c* and the ball *e* in a groove. The whole instrument is also supported on three points with respect to the object under test, these being the knife-edges, one of which is attached to the lower and two to the upper main member. The base length of the instrument is 20 mm (0.78 inch) and the rocker is 0.2 mm (0.00787 inch) thick. The magnification of the strain is governed by the ratio of twice the distance of the recording drum from the mirror to the diameter of the drum, and in this case is 10,000. A deflection of 1 cm (0.39 inch) of the light beam when the mirror is 1 m (39.37 inches) from the recording drum thus corresponds to an extension of 1μ per 20 mm, i.e., $0.5 \cdot 10^{-4}$ per 0.78 inch which in steel having an E-modulus of 20,000 kg/mm² (28,446,000 pounds per square inch) would be a load of 1 kg/mm² (1422 pounds per square inch). The weight of the instrument is only 3.4 g (0.12 ounce) and its natural frequency is above 2,000 Hertz.

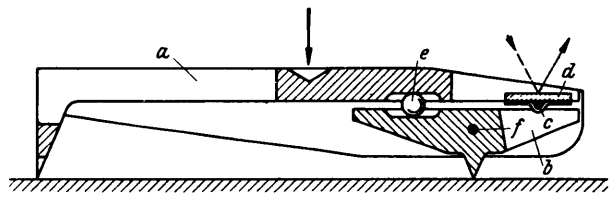


Figure 1 - Multiple Part Dynamic Strain Gage

Base length 20 mm (0.78 inch)
Overall length 25 mm (0.98 inch)
Weight 3.4 g (0.12 ounce)
Natural Frequency about 2200 Hertz

a Upper part with double knife edge
b Lower part with single knife edge
c Thin rocker
d Mirror
e Ball bearing
f Pin

With this strain gage it was possible without special preliminary tests to carry out useful measurements directly. Because of its simple structure, it was found possible to design, build and use the instrument within a period of eight days. To the advantages of this design is opposed the disadvantage that it is built up of a number of separate parts. These are held together by a pin *f*, which must be taken out after the instrument is attached, and put back in again before detaching, which necessitates some care in handling. Furthermore, it is sometimes desirable to eliminate by damping the natural vibrations of the instrument, which, when the deflections in the record are small, are superimposed on the record.

For these reasons this design was abandoned in favor of one with oil damping, Figure 2. A completely tight housing *a*, equipped with a rigid double knife-edge *b*, is closed at the bottom by a thin diaphragm *c*. Into this is inserted a rocking knife-edge *d*, which carries the bearing for the mirror rocker *e* inside the housing. The rocker is constantly pressed against the bearing by an adjustable spring *g*. Above the mirror *f* is a glass window *i* for the ingress and egress of light rays. The housing is filled with oil to damp the natural vibrations. The base length is 20 mm (0.78 inch), the over-all length 25 mm (0.98 inch). In Figure 3 is represented a similar strain gage with the extraordinarily short base length of 5.5 mm (0.22 inch), with which it is possible to carry out tests in notches that are not excessively narrow. The special arrangement of the diaphragm permitted utilization of the entire length of the gage for measurements.

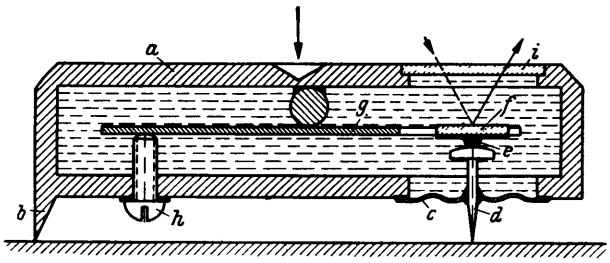
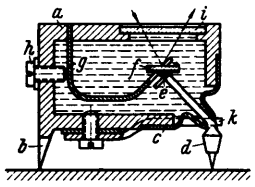


Figure 2 - Oil-damped Strain Gage

Base length 20 mm (0.78 inch)
Overall length 25 mm (0.98 inch)
Weight about 6 g (0.2 ounce)

a Housing
b Rigid double knife edge
c Diaphragm
d Rocking knife edge
e Thin rocker
f Mirror
g Flat spring
h Set screw
i Glass window

In developing the oil-damped instruments, the advantages of one-piece construction, and of damping, were attained at the cost of one disadvantage: At zero position the mirror must be parallel to the glass window, since otherwise the light will be diffused by the oil prism enclosed by the two glass surfaces, and the image will thus be fogged by colored edges. The convenience of the undamped instruments, that they permit adjustment of the ray of light on the drum by turning the mirror to the most convenient position, is thus sacrificed. Therefore the question arose whether by doing without the damping which is only rarely necessary, but retaining the one-piece structure, it would be possible to develop an instrument on the whole even more advantageous.



a Housing
b Rigid double knife edge
c Diaphragm
d Rocking knife edge
e Thin rocker
f Mirror
g Flat spring
h Set screw
i Glass window
k Thrust block

Figure 3 - Small Oil-damped Strain Gage

Base length 5.5 mm* (0.22 inch)
Overall length 6 mm (0.24 inch)
Weight 1.6 g (0.06 ounce)

* Enlargement in this figure is 1 1/2 times that of Figures 1, 2, 4, and 5.

In the instrument constructed with this thought in mind, Figure 4, the rocking knife-edge *c* is connected to the solid bridge to which the rigid knife-edges *a* are attached, by means of a coil-spring coupling *b*. The coil spring must have the greatest possible flexural strength in the direction parallel to the surface of the test body, since otherwise it will be free to deflect in phase with the stress vibrations and in the same direction, which would make the deflection of the mirror too small. This danger, it is true, would exist only at very high frequencies, namely when the frequency of the vibrating load approaches the natural frequency of the vibrating system composed of the mass of the rocking knife-edge and the spring. The base length of this instrument is 15 mm (0.59 inch). This might without difficulty be reduced to half if necessary. Obviously in that case the sensitivity would also be only half as great, the diameter of the roller *d* remaining the same. The over-all length is only immaterially greater than the base length. With a magnification of 10,000, the stress in steel corresponding to a deflection of 1 cm (0.39 inch) with the drum 1 m (39.37 inches) distant would be about 1.3 kg/mm² (1849 pounds per square inch). The weight of the strain gage is less than 2 g.

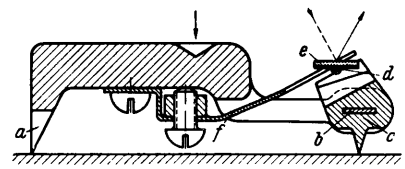


Figure 4 - Simple Undamped Strain Gage

Base length 15 mm (0.59 inch)
Overall length 16.5 mm (0.65 inch)
Weight 1.9 g (0.07 ounce)
Natural frequency about 2200 Hertz

a Rigid double knife edge
b Elastic hinge
c Rocking knife edge
d Thin rocker
e Mirror
f Tension spring

For the various purposes for which they were to be used, strain gages were developed with base lengths of from 4.8 to 40 mm (0.19 to 1.57 inch). The roller of the largest instrument is 1 mm (0.04 inch) in thickness, magnification thus being 4,000 at a distance of 1 m (39.37 inches) from the drum with a sensitivity of 1.25 kg/mm² (1777.9 pounds per square inch) when the optical lever deflects 1 cm, and the elasticity modulus is 20,000 kg/mm² (28,446,000 pounds per square inch). This instrument is used chiefly as a static strain gage in endurance tests. Since it is believed to be preferable to determine the chronological progress, type and magnitude of the external forces, not to measure the local stress increases in notches, the smallest instrument has been little used hitherto. The difficulties of making tests, moreover, are in this instance very considerable, since the instrument must be put in place with forceps.

In addition to the actual testing instrument, the optical accessories - lighting apparatus and film drum - must also be appropriately designed so as to permit tests in working conditions which are often highly inconvenient during operation. An arc lamp for alternating current and direct current is suitable as a lighting arrangement, with a bracket, bearing at its end the adjustable lenses and two movable mirrors which permit the ray of light to be adjusted in any direction. The optical arrangement is crowded together as much as possible in order to avoid as far as possible any

interference with the ray of light between the strain gage mirror and the film drum. A film spool previously described* has been found suitable as a recording device. It can be turned in any direction by means of a ball-and-socket joint, and is suitably mounted on a stand which is adjustable laterally and vertically.

ELECTRICAL STRAIN GAGES

For tests of structural parts impossible to illuminate, the mechanical-optical instruments cannot be used. They can then be suitably replaced by electrical strain gages. With the instrument developed for this purpose, the resistance of a fluid changes with the base length, Figure 5. The change in resistance can be recorded in a bridge arrangement by oscillograph. The instrument itself resembles in structure the oil-damped mechanical-optical strain gages. In place of the rocking knife-edge with the support for the mirror roller, there is introduced a tongue *c* which splits up the resistance of the liquid between two closely spaced electrodes, *a* and *b*, as governed by the extension of the test body. The natural oscillations of the instrument are greatly damped by the internal friction of the electrolyte. Its sensitivity increases as the distance between the electrodes decreases.

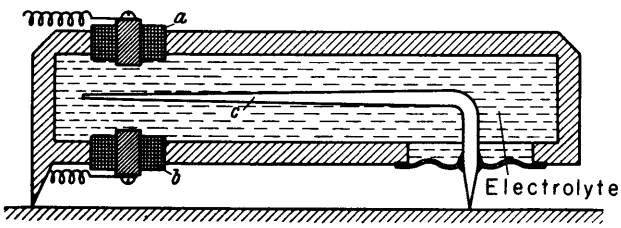


Figure 5 - Electrical Strain Gage

Base length 20 mm (0.78 inch)
Overall length 25 mm (0.98 inch)
Weight about 6 g (0.2 ounce)

a and b electrodes c tongue

ATTACHING STRAIN GAGES

Essential for the usefulness of the strain gage is the design of devices for attaching it; several examples are illustrated in Figures 6 and 7. The gripping pressure must be strong enough to overcome the restoring forces even when there are large extensions, and must be greater than the accelerating forces acting on the instrument and the gripping device, so that in tests of moving structural parts the knife-edges cannot lift or slip. For this reason the gripping device must also be built as light as possible, and because of space conditions, it must likewise be small but nevertheless handy. Depending upon local conditions, spring clamps or adjustable screw clamps are used, which can be fastened to the structural part by suction cups.

RESULTS OF STRESS MEASUREMENTS

In a six-cylinder four-cycle Diesel engine of an older model, whose housing was reinforced by seven triangular ribs opposite the mounting flange, the middle rib

* S. Berg, Zeitschrift des Vereines deutscher Ingenieure, vol. 78 (1934) p. 1295.

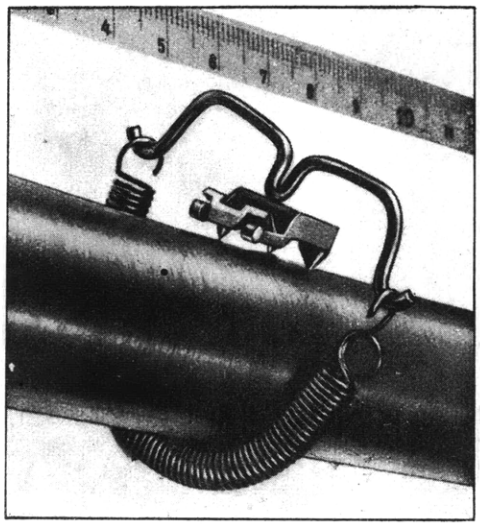


Figure 6 - Mounting Device consisting of Bracket and Coil Spring, for Dynamic Strain Gages

The design of the mounting device permits a wide range in selecting locations best suited to the optical arrangements.

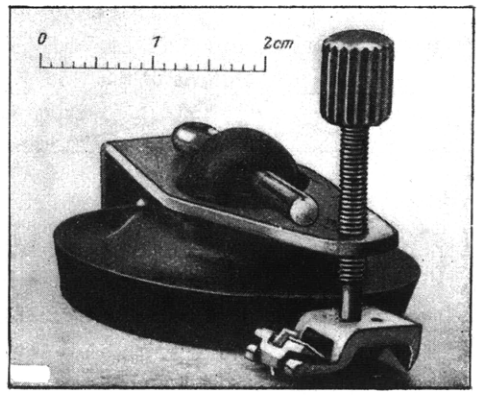


Figure 7 - Suction-Cup with Clamp for mounting Strain Gage

This is the gage shown in Figure 4, for use on nearly plane surfaces. Contact pressure is capable of fine adjustment by means of the thumbscrew.

on the operating side had cracked. It was hardly possible to decide purely by reasoning by what the stresses acting in the ribs are set up, or to make a definite statement as to their magnitude. Therefore the loads in all the ribs were measured with the engine running. This showed the stress distribution illustrated in Figure 8. Disregarding the middle rib, where due to the crack, the stress condition was disturbed, the measured stresses agree perfectly with the deflection curve of the revolving crankshaft which was not equipped with counterweights. From this fact and the circumstance that the stresses changed in phase with the RPM of the engine and not with the ignition it followed that the loading of the ribs originated not from excessively high combustion pressures, but from the gear mechanism. This knowledge was essential in adopting remedial measures.

Noteworthy stress measurements were carried out on a drive shaft of a Diesel generator, Figure 9. Distorted somewhat by the cross-oscillations of the shaft, the strains of the drive shaft mirror the indicator card of the engine.

In developing a vibration testing machine, stress curves were obtained on the clamping head of the machine and on the test rod. The testing machine produces a nearly sine-shaped vibration-stress curve (Figure 10), while according

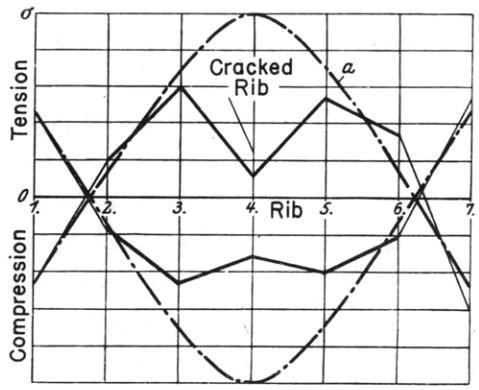


Figure 8 - Alternating Stress in the Base Ribs of a Motor Housing

a - Deflection curve of the rotating crankshaft

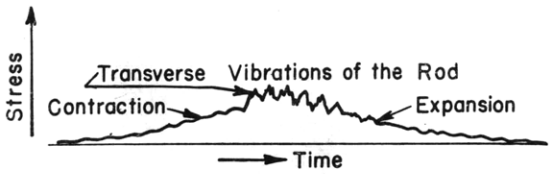
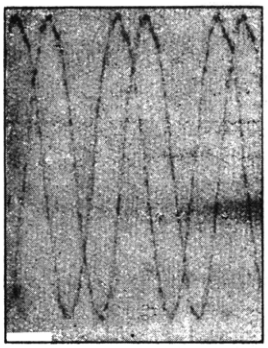
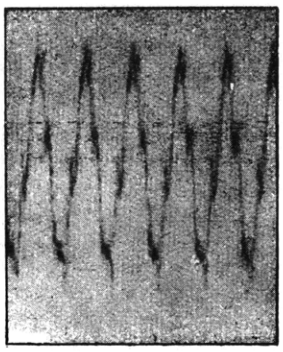


Figure 9 - Stress Distribution in a Drive Shaft of a Diesel Engine

The stress curve resembles the indicator card of the engine.





Figures 10 and 11 - Stress Oscillations in the Head of a Vibration-testing Machine (left) and in the Test Rod (right)

to Figure 11, the test rod carried in addition undesirable transverse vibrations of four times greater frequency. This information gave ideas for changing the construction or the working method of the machine.

While the tests hitherto discussed were performed with test bodies under dynamic load but standing still in space, it is necessary in tests of moving parts to take into account the individual motion of the testing instrument. Structural parts with an oscillating motion can be tested as long as the test point does not move out of the light cone of the lighting apparatus. Since parallel displacement of the instrument has no effect on the position of the path of the light, the photograph faithfully reproduces the stress conditions of the structural part. Thus, for example, Figure 12 shows the stresses in a valve push rod of a Diesel motor. The impact of the

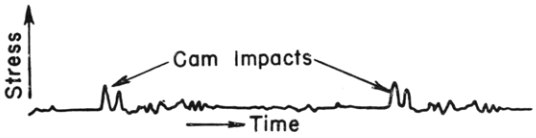


Figure 12 - Dynamic Stress Data of a Valve Push-rod

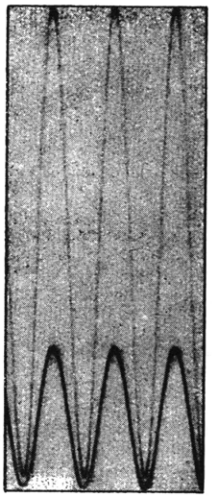


Figure 13 - Record of Stresses in the Test-bar of a Schenck Flat-bending Machine.

cam in starting and the jumping of the roller are plainly discernible. When the test body has a turning motion, a fixed zero mirror must be attached to the instrument, and the values recorded by this must be deducted from those of the pivoted mirror. Thus, stress measurements were made on the test bar of a Schenck flat bending machine with an oil-damped instrument having a base length of 20 mm, (0.78 inch), (Figure 13). The heavy curve is the tracing of the zero-mirror, which must be deducted from the fine curve in order to get the stress amplitude. The smooth, sine-shaped course of the curve is an indication of the excellence of the stress test set-up.