STRAIN PATTERNS IN PLAIN BEARINGS

BY

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This article describes the laboratory experiment carried out by the author at the Royal Naval Engineering College, Manadon, in October 1977, as part of his post-promotion course for special duties marine engineer officers. The article proposes a new mechanical engineering concept: that of using strain measurement in lieu of temperature measurement as a means of monitoring the condition of plain bearings.

Introduction

Main machinery transmission systems in H.M. ships use thermometric techniques for monitoring journal bearings, usually in the form of a thermocouple. Bearing temperature monitoring is the conventional method adopted to give the operator an indication of bearing condition during machinery operation.

In order that the bearing temperature should rise to an unacceptable level, the bearing must be subjected to an abnormal condition (overloading), either because of a breakdown in lubrication or because of a misalignment condition. Either cause will lead eventually to overheating of the bearing with a corresponding temperature increase. It would appear that monitoring a temperature rise is observing the *effect* rather than the *cause* of the original problem. It became apparent that it was possible to look closer at this cause using a monitoring system which was not temperature dependent.

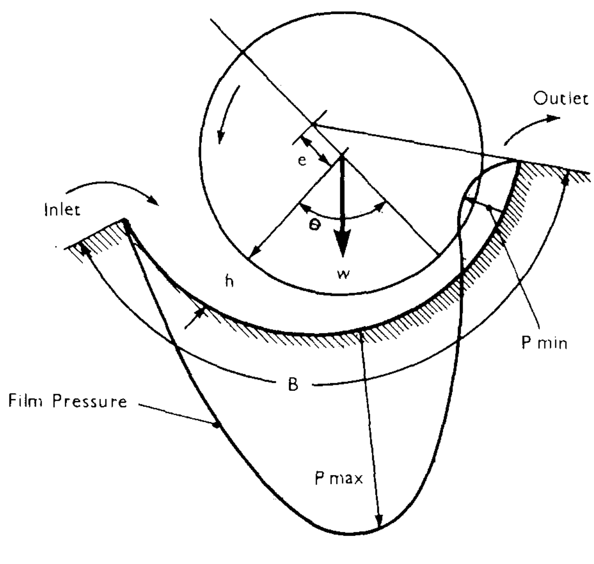

FIG. 1—OIL FILM CHARACTERISTICS
Where: e = eccentricity
W = load on journal
h = film thickness
= bearing radial clearance—e cosλ

Concept

When a plain bearing is operating under normal conditions with hydrodynamic (fluid) lubrication, it is subjected to stresses from at least two sources: thermal stresses incurred by the oil shear behaviour and pressure stresses due to the loading on the bearing and pressures generated by the fluid oil wedge. It was postulated that because of these stresses the actual bearing and possibly also the housing become deformed to some extent. Moreover, should the bearing be subjected to a different loading condition, these stresses change as do the deformation characteristics—the strain patterns.

Additionally, if the deformation can be monitored, it is reasonable to assume that, for a given load

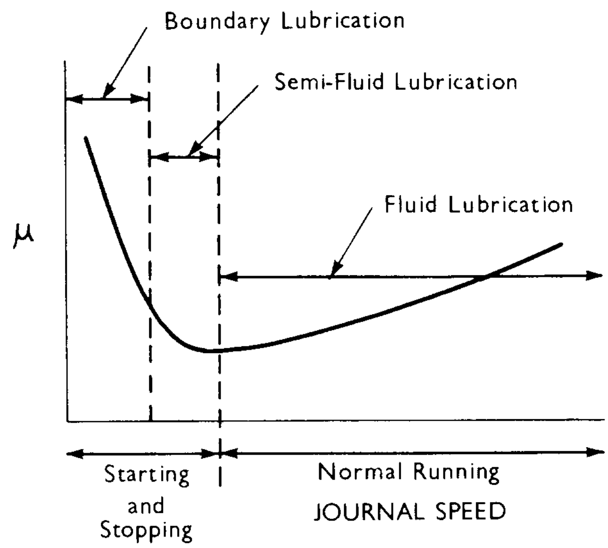

FIG. 2—RELATIONSHIP BETWEEN JOURNAL SPEED AND COEFFICIENT OF FRICTION

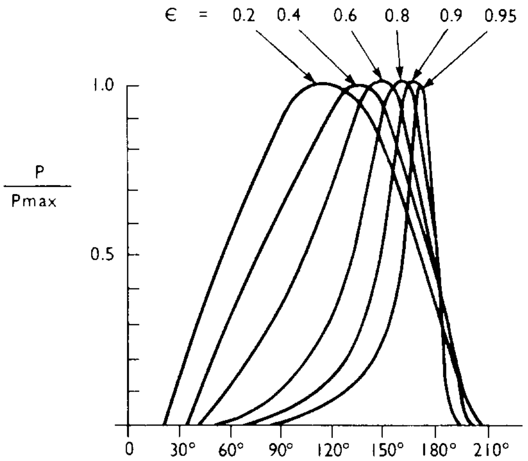

FIG. 3—PRESSURE CURVES FOR A BEARING WITH L/D RATIO OF 1

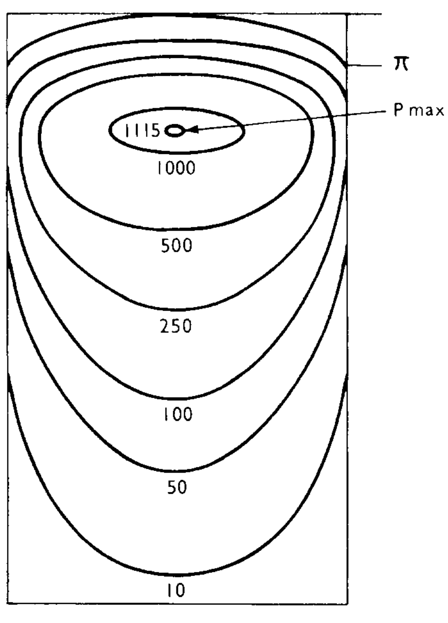

FIG. 4—ISOBARS OF A JOURNAL BEARING WITH L/D RATIO OF 1 AND ECCENTRICITY RATIO OF 0·8

change on the bearing, any corresponding deformation of the shells would be observed significantly earlier than any temperature change so induced.

Principles of Hydrodynamic Lubrication

As the principles of fluid lubrication form the foundation upon which the postulate is built, they are briefly recalled here. Hydrodynamic (or fluid) lubrication was observed as long ago as the middle of the last century. Fluid lubrication occurs when two moving surfaces are completely separated by a thick oil film. In the hydrodynamic condition, the resistance to motion becomes solely a function of the viscosity of the lubricant rather than the nature of the two surfaces involved. In the case of a plain bearing, when a journal rotates inside a lubricated bearing above certain speeds, a wedge-shaped oil film forms under the journal forcing it to rotate completely free of the bearing surface.

The fluid oil wedge formed generates uneven pressure about the circumference of the journal forcing it to rotate eccentrically, and at the same time giving the journal a considerable load carrying capacity. The load on the journal is balanced by the resultant upthrust pressure of the oil wedge. Should the load on the journal change, be it gradually or impulsively, the pressures generated in the lubrication process change correspondingly. It is generally accepted that, in the hydrodynamic condition, the difference in velocity between the journal and bearing surface is taken up wholly by the shearing of the oil, there being no slip at the surfaces of the journal and bearing. The shearing of the oil—the tearing apart of the lubricant molecules—produces heat, and the oil flow rates are adjusted to dissipate the heat thus generated. FIG. 1 indicates the oil film pressure characteristics in the hydrodynamic process.

Journal speed affects the lubrication process and also the coefficient of friction (μ). The curve (FIG. 2) shows these relationships for a given oil viscosity and journal load.

Bearing Design

FIGS. 3 and 4 indicate how the eccentricity of the journal and the bearing clearance relationship affect the point at which peak lubrication pressures occur.

$$\text{Eccentricity ratio } (\varepsilon) = \frac{\text{Eccentricity of the journal}}{\text{Bearing radial clearance}}$$

Test Rig Design

Bearing shells that have a thickness of whitemetal of less than 0·004 inches can withstand pressures up to 2500 psi. However, bearing shells with a whitemetal thickness of more than 0·004 inchess have reduced maximum pressure; this is in the region of 2000 psi. The shells that were to be used for the experiment had a whitemetal thickness of 0·010 inches and were to be subjected to pressures of up to 2400 psi at speed ranges between 500 and 1500 rev/min—load factors well in excess of the normal maximum.

The test rig was designed and manufactured at the R.N.E. College and consisted of two shells held in a housing that permitted access to the centre one-third section of the shells, for monitoring by strain gauge. The assembly was suspended on a rotating shaft that was itself held by two self-aligning ball bearings with a load carrying capacity of over 5 tonf. Loads up to 2·5 tonf were applied to the housing and the rotating shaft was driven by an electric motor. The bearing was forced lubricated.

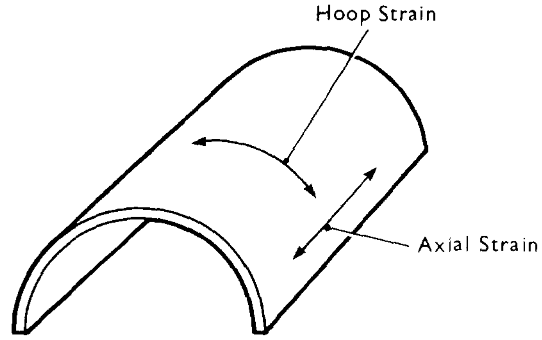

FIG. 5—HOOP AND AXIAL STRAIN

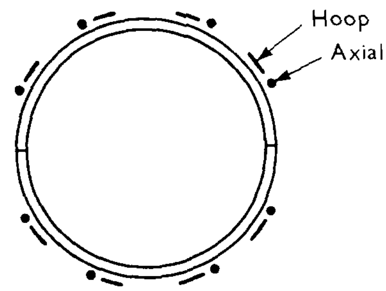

FIG. 6—ANGULAR ARRANGEMENT OF STRAIN GAUGES

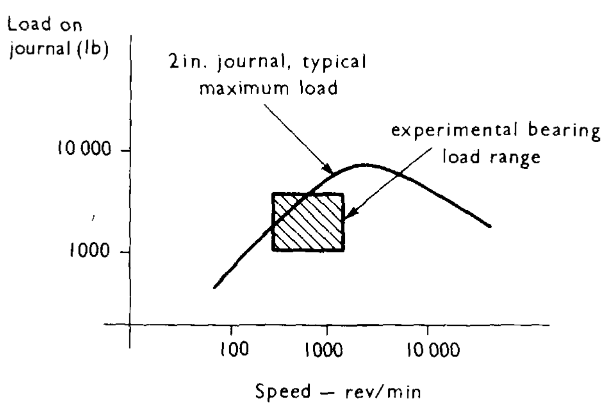

FIG. 7—BEARING LOAD RANGES

Strain gauges were fitted to a pair of thin-walled shell bearings to monitor the strain patterns developed around the section where, for an open-ended bearing, peak pressures would occur. The strain patterns were divided into two types: 'hoop' strain and 'axial' strain. FIGS. 5 and 6 describe these two types and show where the strain gauges were fitted in pairs around the back of the shells. There were sixteen gauges in all, eight measuring each type of strain.

A nominal 2-inch journal size was chosen, and it was intended to subject the bearing to the load ranges indicated in FIG. 7. The shells were manufactured to a commercial specification, the only difference being in the thickness of whitemetal.

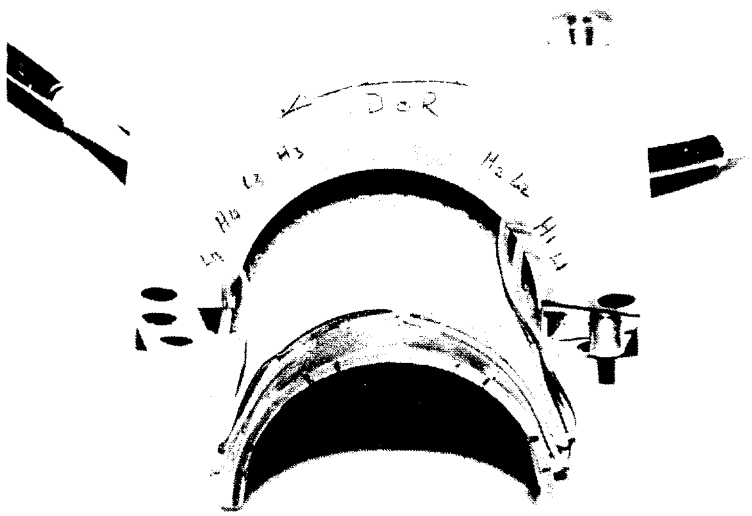

FIG. 8—BEARING HOUSING TOP HALF SHOWING TOP SHELL AND STRAIN GAUGE LEADS

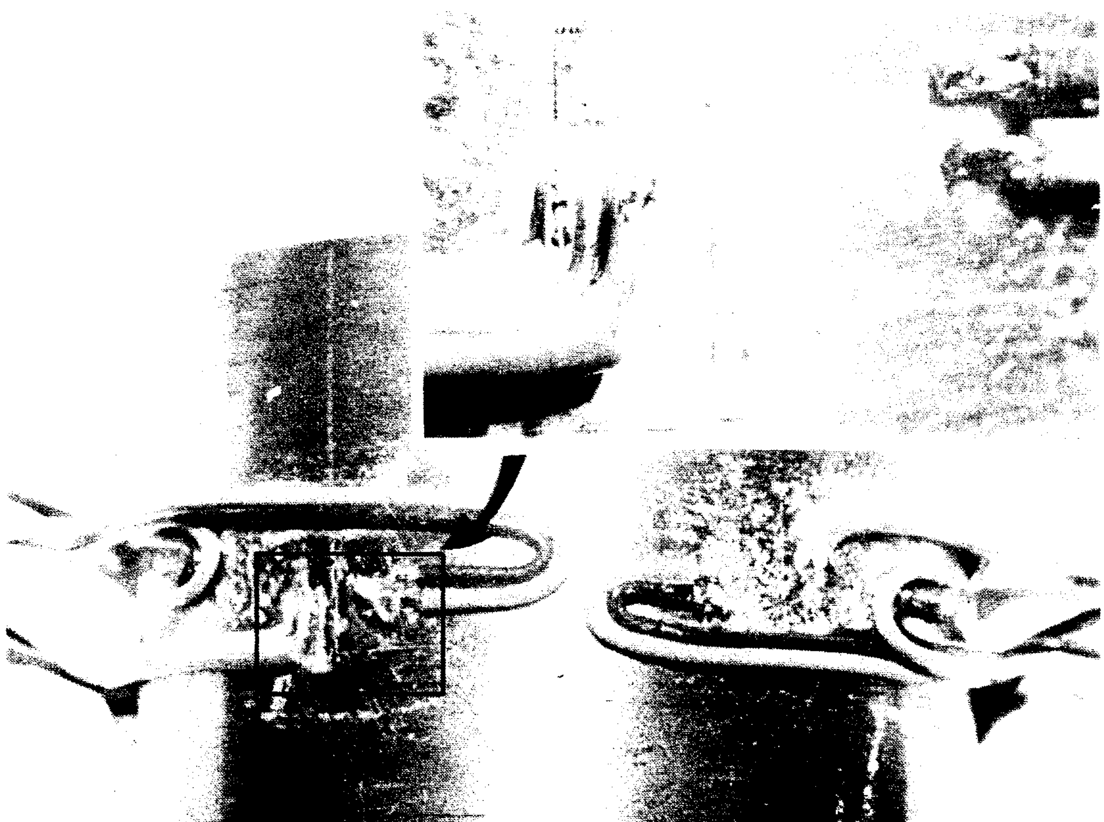

FIG. 9—BEARING SHELL WITH STRAIN GAUGES FITTED AND INSET SHOWING CLOSE-UP OF STRAIN GAUGES

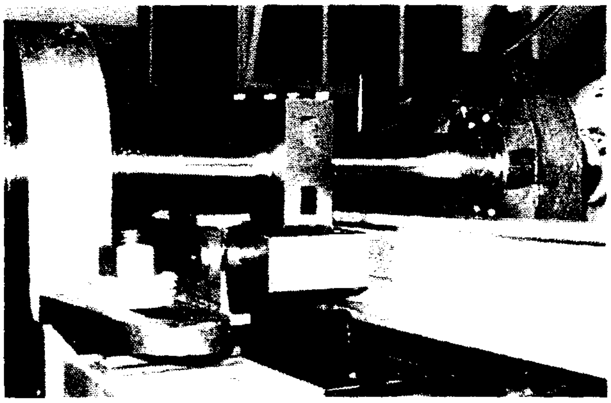

FIG. 10—BEARING HOUSING SHOWING ACCESS PORTS, LOADING LEVERS, DRIVEN SHAFT, AND SUPPORT BEARING AT EACH END

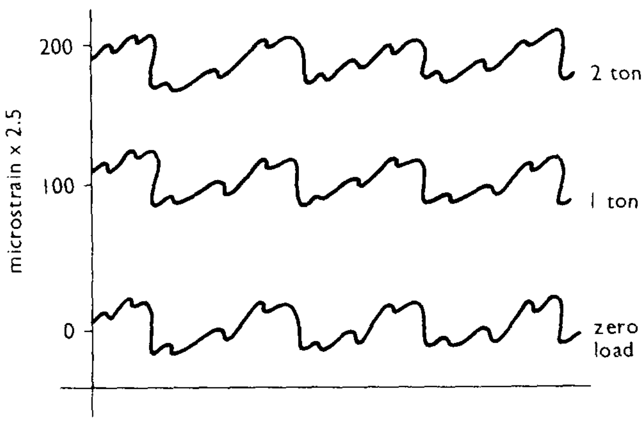

FIG. 11—OSCILLOSCOPE READINGS—FIRST EXPERIMENT

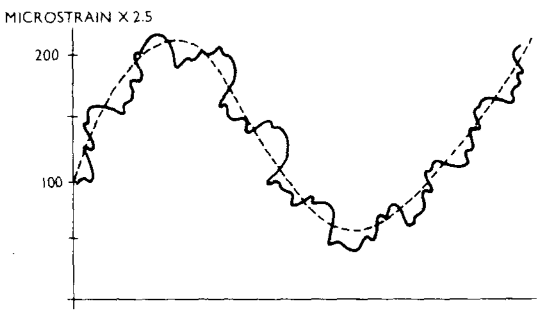

FIG. 12—OSCILLOSCOPE READING WHEN ONE LOAD LEVER WAS SET VIBRATING ACCIDENTALLY

Results

The First Experiment

Because of difficulties in setting the test rig to work and time constraints on the project, a period of one day only was available on the test rig for the actual experiment. Initially, therefore, it was decided to ascertain if indeed strain patterns were present at all. To this end, a single strain gauge on the bottom shell—in the area of highest lubrication film pressure—was connected to an oscilloscope. A test was conducted at constant shaft speed whilst varying the load on the bearing.

The saw-tooth picture (FIG. 11) was recorded at the loads indicated. The profile of this picture is only approximate, but it merely indicated vibration and cavitation. During the course of this first experiment, one of the horizontal loading levers was accidentally set vibrating (FIG. 12). The damped oscillatory mode of the lever immediately set the signal on the oscilloscope completely sinusoidal, maintaining the saw-tooth shape. The signal changed—amplitude for amplitude—with the vibrating lever until the lever came to rest.

In either loading condition, change of the strain patterns—the bearing shell deformation—was occurring instantaneously with load change. At no time was any appreciable temperature change observed.

The Second Experiment

For the second experiment, all the strain gauges were connected to a digital read-out meter; also, the loading system was modified to read this time in kilonewtons (kN). The shaft speed was set initially at 500 rev/min and the load applied in stages, a set of readings being taken from the strain gauges at every load change. This experiment was repeated for shaft speeds of 750, 1000, 1250, and 1500 rev/min.

FIG. 13 shows the set of readings taken at a shaft speed of 500 rev/min plotted on circular graph paper.

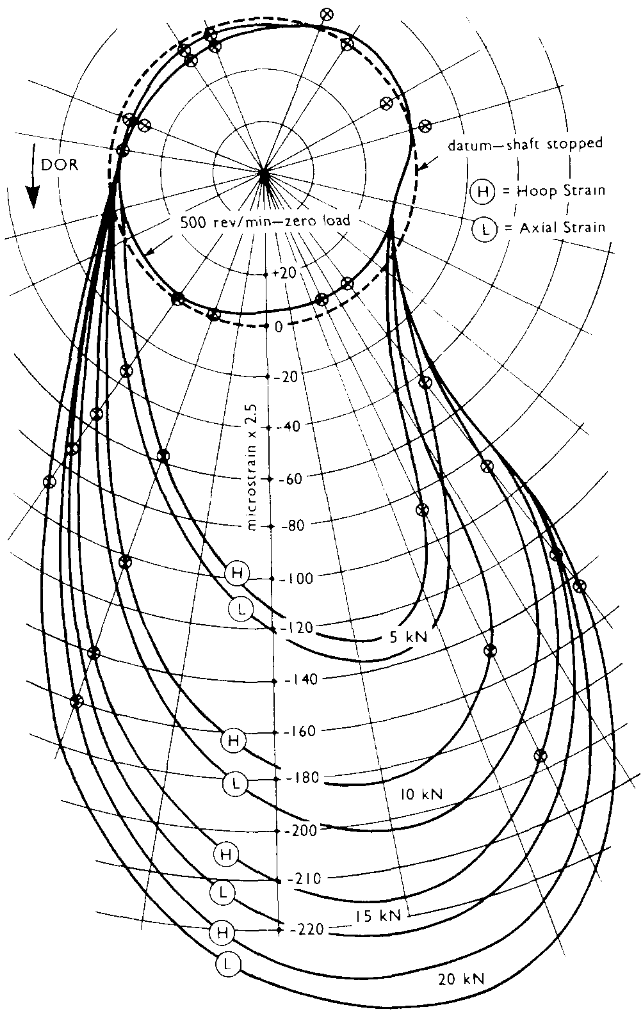

FIG. 13—STRAIN PATTERNS OF A LOADED PLAIN BEARING

Analysis

Oscilloscope Readings

The observations made during this brief experiment indicated that:

(*a*) strain deformation was present in a loaded plain bearing and this could be monitored;

(*b*) bearing shell deformation altered instantaneously with load changes, whether the load change was impulsive or gradual.

The Strain Patterns in a Plain Bearing

The graph (FIG. 13), obtained during the second experiment, shows that:

(*a*) the bearing deformation produced by loading coincides with the oil-film characteristics of hydrodynamic lubrication;

(*b*) the deformation profile is reasonably constant for different loads;

(*c*) the deformation is instantaneous with load change and represents elastic deformation of the shells within the limits of proportionality. (All strain gauge readings returned to within one or two microstrains of datum when the load was removed.)

Critique

The experiments conducted so far have provided adequate empirical evidence in support of the theory that the pressures generated in the hydrodynamic lubrication process produce predictable deformation patterns in plain bearing shells. Furthermore, the experiments have shown that this deformation occurs instantaneously with load changes on the bearing, and that it can be monitored. It is certain also that very little of the deformation is due to thermal stresses as the readings obtained in the 'cold' and 'warm' running conditions varied only slightly, and then only some time after the load had been applied.

Although every effort was made to simulate authentic operating conditions, the middle one-third section of the bearing was unsupported. In practice, this unsupported area would be much smaller—perhaps only one sixth of the bearing length. As no temperature measuring instruments were fitted to the bearing, exact temperature changes were not monitored. This aspect, however, can reasonably be ignored as no injuriously high bearing temperatures were encountered during the experiments, i.e. no temperatures above approximately 160°F at the edge of the bearing shell.

The curves of deformation shown on the circular graph (FIG. 13) are, to a large extent, extrapolated.

Only very basic data could be obtained from the test rig due to its fundamental design; and also, because the time constraints did not permit maximum use of the test rig itself, the experiments carried out were far from exhaustive.

Conclusions

As less space becomes available in H.M. ships for the main propulsion machinery systems, transmission components operate much nearer their critical conditions. There is, therefore, an increasing need for more efficient and accurate monitoring systems and improved diagnostic techniques in routine maintenance checks. To this end, the potential of a strain pattern monitoring system, if developed sufficiently, may provide a useful and efficient diagnostic aid. What has been determined in the experiments so far is only a very basic concept.

Further experimentation is necessary on an improved test rig with a temperature monitoring system and a network of strain gauges in order to provide more information about the deformation characteristics.

There is also a need to determine effects such as a breakdown in lubrication, contamination of the lubricant, or misalignment of loading on the bearing.

The possibility of monitoring the pressure generated as a result of the lubrication process by insertion of a strain gauge between bearing shell and bearing housing should be investigated.

Finally, consideration might be given to using this system in the development stages of gearbox design; not only would the strain pattern give information about the deformation of the shells but also would determine the magnitude and direction of the load creating the deformation.

Acknowledgement

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