THE SPRING RESEARCH AND MANUFACTURERS' ASSOCIATION

USE OF STRAIN GAUGES TO INVESTIGATE DYNAMIC STRESSES IN SPRINGS

Report No 415

by

M S Bayliss, B.Eng.

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SUMMARY

This report covers the work carried out at SRAMA using strain gauge techniques to investigate the dynamic performance of springs when cycled at speeds approaching their natural frequency.

The project studied the increased stresses produced at the springs' sub-harmonics (1/2, 1/3, 1/4 etc of the natural frequency), and the general increase in stress that occurs as cycling speed is increased up to and beyond the natural frequency.

The common use of conical and parallel sided variable pitch springs to reduce vibration-induced stresses was also investigated.

Results indicated that considerable increases in stress occurred at the sub-harmonic frequencies, that the simple formulae generally used to calculate dynamic stresses predict stress increases higher than those measured, and that the conical spring exhibited very good dynamic performance compared with the variable pitch and conventional compression springs.

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CONTENTS

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1. INTRODUCTION

When a spring is cycled at a speed which is equal or close to a sub-harmonic of its natural frequency, additional stresses over and above the nominal static values are induced. The dynamics of these vibration-induced stresses have previously been mathematically analysed by a considerable number of authorities, and comprehensively documented in SRAMA Research Report No 318, "Dynamic Stresses in Helical Compression Springs - A Literature Survey", May 1980.

The resulting formulae vary considerably in complexity depending on the number of controlling parameters accounted for in the analysis (eg damping, dynamics of the input etc). Meaningful application of these theories can therefore be difficult if the full practical implications of what is happening to the spring during high speed excitation are not fully understood.

Strain gauge techniques enable the dynamically induced stresses in a spring to be visually displayed in order that its performance can be assessed (see Fig 1).

2. EQUIPMENT REQUIRED TO UNDERTAKE STRAIN GAUGE TESTING

Four items of equipment are required to undertake basic strain gauge testing:

1) Strain Gauges
2) Amplifier
3) Power Supply
4) Monitoring Equipment

2.1 The strain gauges required for the majority of spring testing need to be very small in size to enable them to be fitted to springs with wire sizes down to approximately 3 mm diameter, and to be able to accommodate the directional curvature of the wire.

The gauges used during this work had an active size of 1 mm x 1 mm and an overall size of 4 mm x 2 mm. In addition to the gauge on the spring, three other gauges were required to form a balanced whetstone bridge (Fig 2) or a strain gauge bridge.

2.2 The voltage of the electrical signals from the strain gauges is very low and so requires amplifying to a level at which standard measuring equipment (eg oscilloscope) can be used. The use of an amplifier also allows for its gain (amount of amplification) to be adjusted so that its output is a direct measure of the strain applied to the spring. This is useful when absolute values of stress are required rather than proportional increases.

2.3 The power supply is required not only to supply the amplifier but also to apply a fixed voltage across the strain gauge bridge, whose output signal is proportional to both the strain applied and the voltage across the bridge. It is therefore important that the power supply is very stable.

2.4 The type of monitoring equipment used is dependent upon the format in which the information is to be used. In most cases an oscilloscope is adequate, particularly if it has a storage facility which will enable the output trace to be frozen.

Although only a modest amount of equipment is required, it is important that it is of a type capable of measuring dynamics signals accurately. Much strain gauge equipment is suitable only for measuring static or low speed dynamic stresses and its use on a high speed test will mean the magnitude of the measured stress is considerably lower than the true value present in the spring. The ability of equipment to measure dynamic stresses is defined by its band width, or half power frequency response. The equipment used should always have a band width at least 100 times greater than the maximum excitation speed of the spring under test (ie, for a maximum excitation frequency of 50Hz, the equipment should have a band width of at least 5000Hz).

The method of cycling the spring during the test programme should always be as close as possible to that found in service. This is often not possible and the use of a fatigue machine or an electro-magnetic vibrator will be required.

3. ALIGNMENT OF STRAIN GAUGES

The shape of a spring with its two directional curvature complicates the fixing of strain gauges. A strain gauge will only measure stress in one direction and so must be bonded to the spring in alignment to the direction of maximum stress. Any deviation will cause errors in the measured stress. The use of a strain gauge rosette consisting of 2 or 3 gauges orientated at 45° or 90° to each other will enable the true stress to be calculated regardless of direction of alignment. However, the disadvantage of a rosette is that it is considerably larger than a single gauge, and requires additional amplifiers and monitoring equipment, followed by mathematical calculations. Because the static stresses in most shapes of spring can be accurately calculated, the simplest solution to the alignment problem is to bond the gauge in approximately the required direction and to calibrate the test equipment in a load testing machine using the load-stress formulae for the spring type, ie for compression springs:

$$\text{Stress} = \frac{8\,P\,D}{\pi d^3} K \ (\text{N/mm}^2)$$

where:
P = Load (N)
D = Mean Coil Dia (mm)
K = Stress Correction Factor
d = Wire Dia (mm)

Stress measured by the gauge = $E \times \frac{L}{L}$

where:

E = Youngs Modulus for the material

L = Strain measured by gauge

The correction factor C found during calibration:

$$C = \frac{8PDK}{\pi d^3 \, E \, \frac{L}{L}}$$

Thus the stress measured by the strain gauge must be multiplied by 'C' to find the true maximum stress in the springs.

4. BONDING OF THE GAUGES TO THE SPRING

Strain gauges work by stretching or compressing with the specimen under test, causing the gauge resistance to change in proportion to the applied strain. It is therefore imperative that the gauge is firmly bonded to the test specimen.

Two types of glue are generally used for the affixing of strain gauges: the epoxy type of glue, consisting of resin and hardener, and the cyanoacrylate (Super glue) adhesives. The former is very good but slow, taking up to 24 hours to cure. The latter is quick drying and suitable for most spring applications.

All scale and grease must be removed from the surface of the test specimen before any attempt is made to bond the gauge. The use of an activator fluid in conjunction with the cyanoacrylate glue improves the bond considerably. Having bonded the gauge, a terminal pad is also glued to the spring. The pad enables a solder junction to be made between the gauge wires and the instrument wires.

After soldering the wires, the gauge and the terminal pad require coating with a silicon rubber solution to help protect the gauge from the ingress of water or mechanical damage.

5. EQUIPMENT USED IN TEST PROGRAMME

The equipment used for the experimental work can be seen in Figure 2. The four-gauge bridge was made by glueing three gauges onto a piece of steel and connecting it to the gauge on the test spring. The use of dummy gauges in this manner improves the temperature stability of the equipment.

The amplifier used is one made specifically for strain gauges and load cells, incorporating a very accurate bridge supply circuit. With a band width far in excess of the 30 kHz required, a small amount of filtering could be incorporated to remove some of the electrical noise (interference) and thus improve the quality of the output. This is fed directly into an oscilloscope to enable the stress patterns to be monitored.

The first of two methods used to cycle the test springs was via a SRAMA single station fatigue machine which produces a motion very close to a sine wave, with a variable cycling speed of up to a maximum of 3000 rpm (50 Hz).

The second method used an electromagnetic vibrator, enabling very high excitation speeds to be used and, by varying the control signal, producing either sine, square or saw tooth wave forms.

Although the limited power output of this machine restricts the forced deflections of the spring, the electromagnetic vibrator is very good for examining springs for resonance affects.

6. TEST PROGRAMME

The test programme was devised to demonstrate the effect of the cycling speed of a spring on the stress patterns produced.

6.1 Test One

A spring with a low theoretical natural frequency was tested on the SRAMA Single Station Fatigue Machine, the speed was slowly increased and the output waveform at speeds equal to the harmonics was noted. It became apparent that the spring had many natural frequencies and associated harmonics other than the fundamental axial resonances being investigated. This created problems in the selection of the axial resonances of interest and in attaining clear outputs on the oscilloscope. When running the fatigue machine at speeds greater than the fourth harmonic, induced vibration in the spring became very severe causing it to jump out of the machine. The stroke of the machine had to be considerably reduced before the higher test frequecies could be attempted. Typical stress patterns found in the spring during testing at speeds equal to the harmonic frequencies can be seen in Figure 3.

6.2 Test Two

This test investigated two engine valve springs specifically designed for better high speed performance than conventional compression springs. The first was a variable pitch, parallel sided compression spring; the second an even pitch conical compression spring. Both designs were cycled on the fatigue machine with a stroke of 10 mm and their performance noted. It was found that the conical spring exhibited extremely good dynamic performance with no harmonics being detected. The variable pitch spring, however, gave very poor performance, with severe vibration being induced from moderate speeds upwards.

6.3 Test Three

In an attempt to eliminate and overcome the problems experienced with non-axial vibration and resonances during Tests 1 and 2, both were repeated using the electromagnetic vibrator unit. The vibrator gives a true uni-directional motion which, in conjunction with the small stroke (0.5 mm), reduces non-axial vibrations and considerably improves the quality of the output.

The limited power of the vibrator restricted the magnitude of vibrations induced in the spring, thus enabling much higher test speeds to be used.

The re-tests confirmed the results of Tests 1 and 2 and generated the good quality oscilloscope traces shown in Figure 3. The performance of the conical valve spring was found to be extremely good even when cycled at speeds of up to 200 Hz.

6.4 Test Four

This test measured the increase in overall stress in a spring as the cycling speed increases, rather than concentrating on the peaks that occur at resonances or harmonics. The speed was gradually increased from the 1/10 harmonic up to and beyond the natural frequency, the stroke of the vibrator being constant throughout the test. The results can be seen plotted in Figure 5, and can be compared with the performance predicted by simple theory developed by Gross:-

$$\frac{\tau_{dyn}}{\tau_{stat}} = \frac{w/wn\pi}{\sin(w/wn\pi)}$$

where: τ_{dyn} = dynamic stress
τ_{stat} = static stress
w = test frequency
wn = natural frequency

$\frac{w}{wn}$ =	0	0.2	0.4	0.6	0.8	1.0
$\frac{\tau_{dyn}}{\tau_{stat}}$ =	1	1.07	1.32	1.98	4.28	

7. DISCUSSION OF THE RESULTS

Results of Tests 1 and 3 can be seen in Figure 3, showing the effect of cycling the spring at odd and even sub-harmonics. It is evident that, when running a spring at an odd harmonic, the induced stresses add to the peaks of the stress wave form, thus producing a potentially larger and more damaging stress range. However, it is apparent that the overall stress increase at a sub-harmonic is greater the closer it is to the spring's natural frequency.

The second notable feature of these tests was that the frequency of the induced vibrations was twice that of the anticipated natural frequency. Investigation into the reasons for this anomaly indicated the cause to be a combination of the positioning of the strain gauge and the movement of the individual coils of the spring. The strain gauges were bonded to the centre coil on all the test springs so that, during each cycle, they registered the presence of the dynamic stress wave twice as it passed up and down the spring body. The centre coils of the spring will therefore be subject to a dynamic stress at twice the natural frequency. In addition, it was found that the sub-harmonics of twice the natural frequency ($\frac{2wn}{3}$, $\frac{2wn}{5}$, $\frac{2wn}{7}$, $\frac{2wn}{4}$ etc) also produce major resonant vibrations (Fig 1)

The results from Test 2 were particularly interesting as the springs are both current designs used in the automotive industry. The conical spring performed extremely well and exhibited negligible resonances. The variable pitch spring, however, had very severe resonance problems even at quite slow speeds. On inspection, it was found that the pitch of the spring was not continuously variable but made up of two even pitches. During the cycling test the narrow pitch coils became solid at the same time, causing an impact loading to be induced. Impact loadings of a spring will produce severe stress patterns even at quite moderate cycling speeds.

The use of the vibrator unit in Test 3 enabled a test spring to be cycled up to four times its natural frequency. During these high speed tests, the coil movements in the spring were observed using a strobe torch. It can be seen from Figure 4 that the springs behaved exactly as predicted by general vibration theory, with the coils separating to produce nodes about which the remaining coils moved.

The results of Test 4 prove that simple theory predicts the dynamic performance quite accurately. The theory neglects any damping that occurs in the spring and, as would be expected, predicts a higher than measured stress increase. It also neglects the stress peaks which occur at the harmonic frequencies which, as previously shown, can be high.

8. CONCLUSIONS

1. The techniques and equipment used in this work provide a clear and accurate method of assessing the dynamic stresses in springs cycled at high speed.

2. The established formulae for predicting the general dynamic stress increase provide a sound basis for design, leading in general to a conservative prediction of stress levels.

3. The existence of damaging resonance effects at speeds equal to sub-multiples of the natural frequency has been proven. In addition it has been established that these effects can also occur at speeds which are sub-multiples of higher order natural frequencies.

4. The stress wave in the active coils, generated by the dynamic effect at sub-harmonic speeds, cycles at a speed of twice the natural frequency.

5. Odd sub-harmonics (1/3, 1/5, 1/7 etc) are potentially more damaging than even sub-harmonics (1/2, 1/4, 1/6 etc) due to the additive effect of the stress peaks of the former.

6. At large amplitudes of motion, torsional and lateral vibrations can become significant.

7. If variable pitch springs are used to reduce the influence of resonance effects, the pitch must be continuously variable. If the spring has only two distinct pitches, dynamic effects can be amplified due to the impact conditions generated when all the low pitch coils close at the same instant.

8. Variable rate conical springs provide a highly effective means of minimising dynamic stresses in high speed applications across a wide speed range.

9. RECOMMENDATIONS FOR FURTHER WORK

This work has practically demonstrated the damages of dynamic resonance effects on the stress levels within springs. The main concern of the designer of high speed equipment is to know how these effects may be avoided. To this end, it is recommended that, using the successful techniques established in this work, a practical programme of work should be undertaken to investigate the relative merits of variable pitch and conical springs and seek guidance for the designer on the spring geometrics which provide best performance.

Another approach which would provide valuable information would be to investigate the effect of lateral vibrations in compression springs. Many springs used in vibratory equipment are subject to lateral vibrations as their primary mode of movement, yet little is known of the dynamics of lateral vibrations.

10. APPENDIX

Spring Designs

Tests 1/3/4

Wire Diameter	: 3.6 mm
Free Length	: 85 mm
Outside Diameter	: 44.5 mm
Number of Turns	: 10.25
Theoretically Natural Frequency	: 94Hz

Tests 2/3 Parallel Sided Variable Pitch

Wire Diameter	: 3.65 mm
Free Length	: 61.75 mm
Outside Diameter	: 36 mm
Number of Turns	: 7.75
Pitch 1	: 2.65 mm
Pitch 2	: 7.90 mm
Theoretically Natural Frequency	: 215Hz

Tests 2/3 Conical Spring

Wire Diameter	: 4.1 mm
Free Length	: 46 mm
Outside Diameter - Large End	: 37.5 mm
Outside Diameter - Small End	: 33.5 mm
Number of Turns	: 5.5

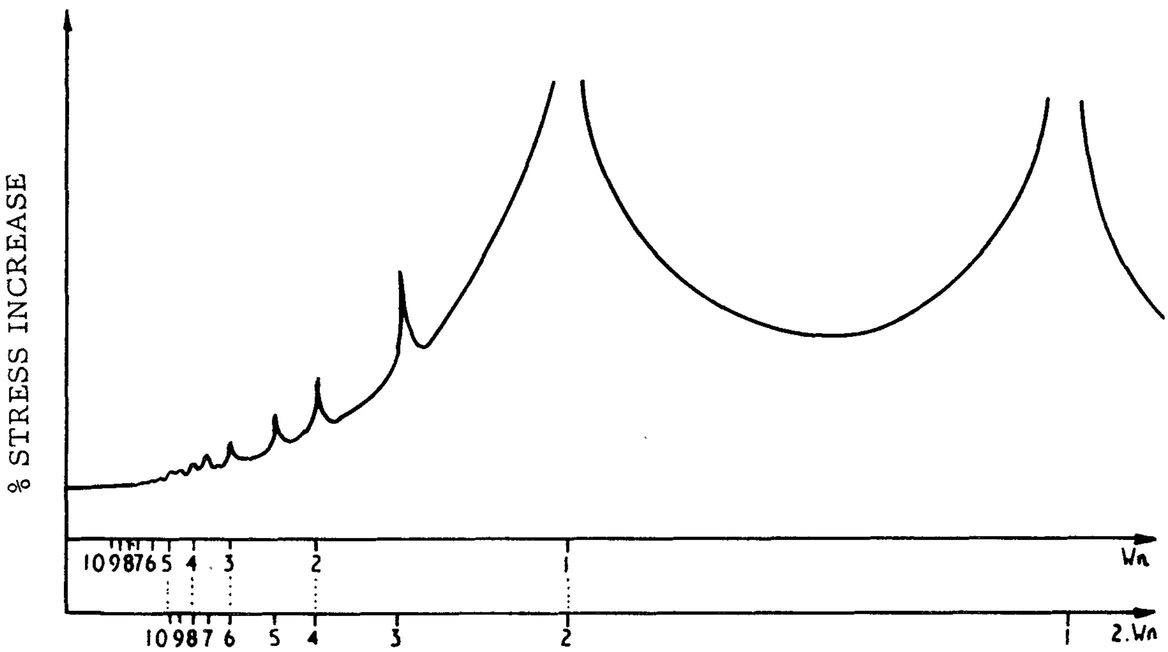

FIG 1 TYPICAL DYNAMIC RESPONSE CURVE FOR A COMPRESSION SPRING

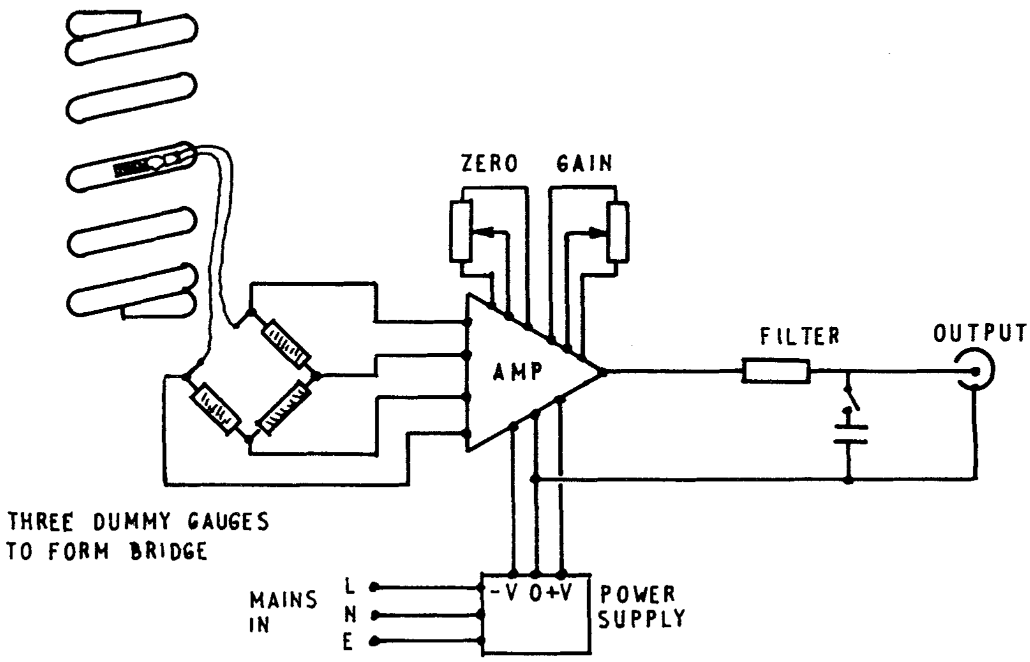

FIG 2 SCHEMATIC LAYOUT OF EQUIPMENT USED

Theoretical W_n = 94 Hz

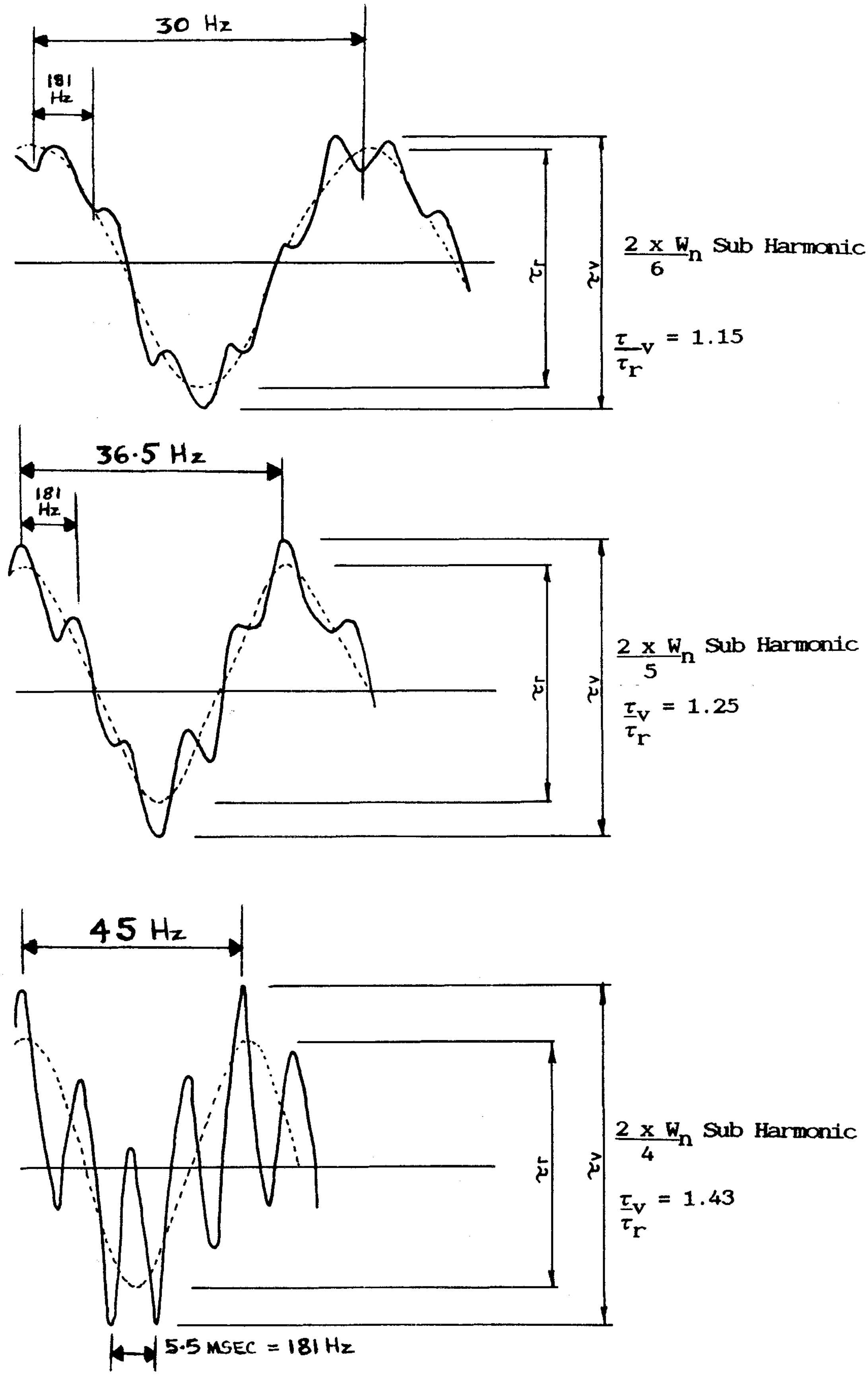

FIG 3 RESULTS FROM TEST THREE

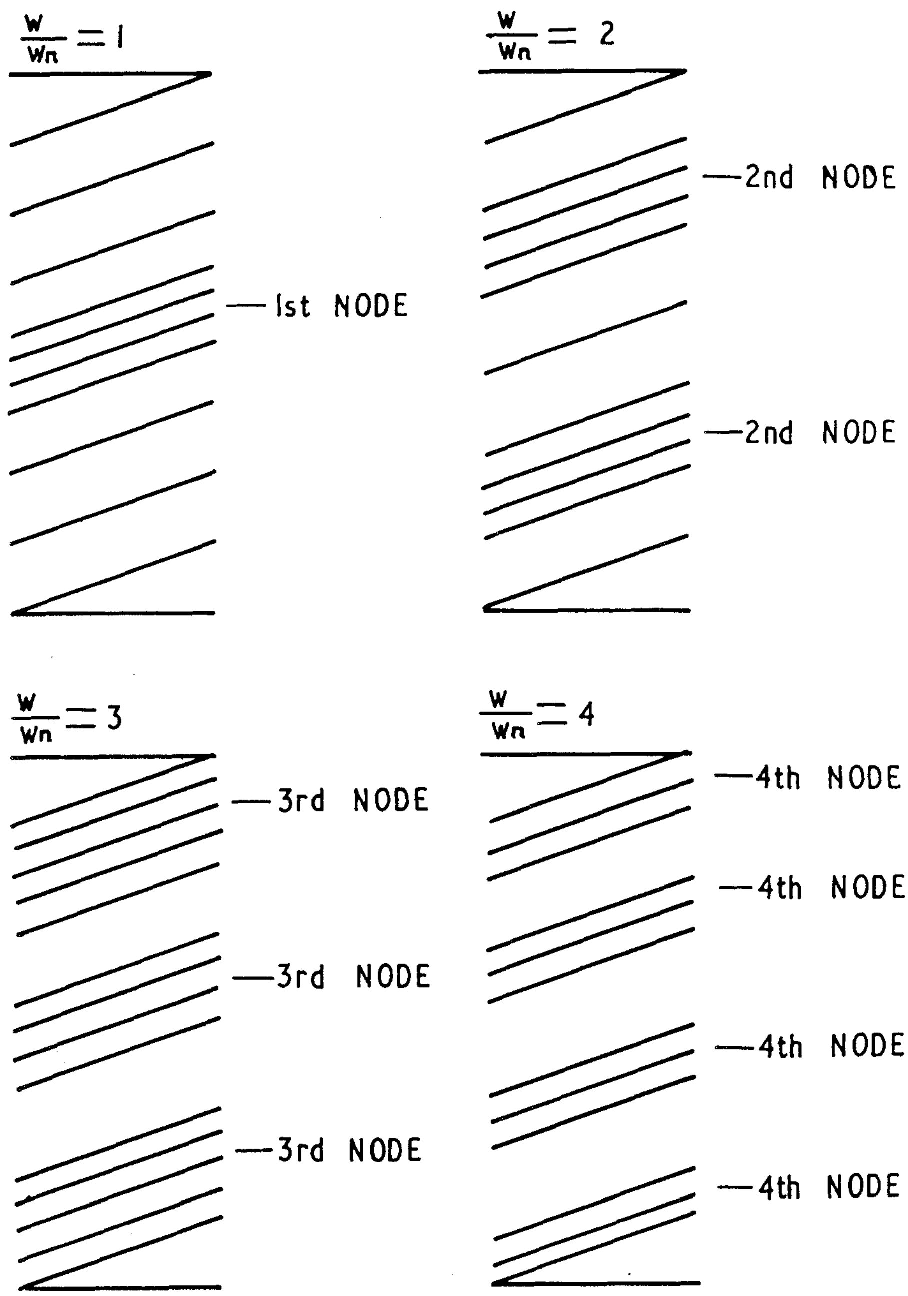

FIG 4 COIL FIGURATION WHEN CYCLING SPRING AT SPEEDS EQUAL TO AND GREATER THAN NATURAL FREQUENCY

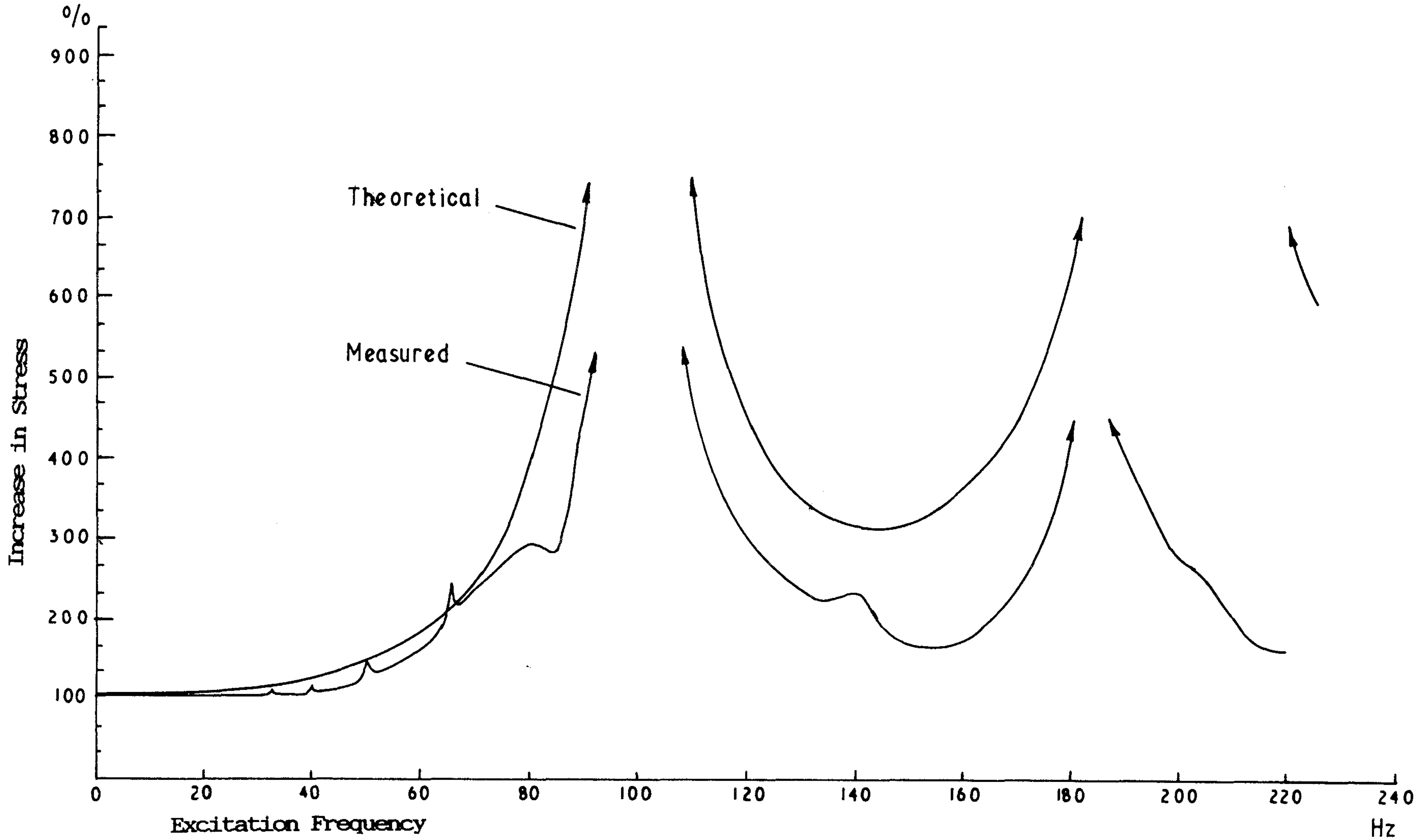

FIG 5 THEORETICAL VS MEASURED STRESS