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STATIC AND DYNAMIC CALIBRATION OF A TRIAXIAL FORCE GAGE FOR MONITORING THE STRUCTUREBORNE FORCES WITHIN A FREON COMPRESSOR

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INTRODUCTION

Reciprocating freon compressors are commonly used in air conditioning and heat pump applications. Compressor manufacturers are interested in understanding how internal structural and acoustical energy travels to become acoustical energy outside of the sealed shell. Basic understanding of this nature will allow new designs to be developed that will be quieter than present units. Three key areas for analysis are the motor / crankcase assembly, the acoustic and structural transmission paths between this assembly and the shell, and the acoustic radiation behavior of the shell among other things. The identification of the transmission paths requires measurement of the compressor assembly forces on the shell. The most dominant paths should be the support springs and the freon in the shell. The freon path is easily measured with a pressure transducer but measurement of the forces from the springs is difficult without modification of the compressor.

INSTALLATION

The Kistler 9251A triaxial force gage was chosen to measure the interaction forces between the compressor support springs and the shell. These forces, if measured, can be correlated to the acoustic output of the compressor to determine the importance of the individual paths and can also be used to drive a finite element model of the compressor. The force gage shown in Fig. 1 is 0.945 in (24 mm) square with a height of 0.394 in (10 mm). It is a charge-mode piezoelectric transducer which can measure both axial and two lateral forces passing through the support springs. This force gage has a force range of 2200 lbf (9800 N) in compression and 1100 lbf (4900 N) in shear with the standard 5600 lbf (25,000 N) preload. It can withstand the freon environment and the AC field produced by the motor during compressor operation. The main disadvantages of this force gage are the required preload, the possibility of large moments at startup cracking the quartz crystals, and the modifications in the compressor necessary for installation.

A typical compressor has two identical support springs that rest on brackets spot welded to the side of the shell (see Fig. 2) and a centering spring at the top that rests against a mount which is spot welded to the top of the shell (see Fig. 3). It was a challenge to devise the mounting assemblies necessary for the force gages without significantly stiffening or mass loading the shell. Kistler approved a 3750 lbf (16,700 N) preload for our application in order to measure the lateral forces and improve its tolerance to the high moments at compressor startup. The mating surfaces must be as smooth and plane as practical. A small boss was machined on each of these surfaces to fit within the recesses at the edge of the mounting hole in the force gage (see Fig. 1, Isometric view only) to aid in alignment during installation and to ensure the transmission of the lateral forces to the transducer. Stops were also installed to limit the compressor motion at startup to limit the startup moments.

The standard lower spring support bracket is shown in Fig. 2. The spring retaining post was removed and the assembly shown in Fig. 4 was installed in its place. The threaded force gage mount fits inside of the bracket in place of the spring retainer post and was silver soldered to the end of the bracket. The disk at the top of this mount provides the required mating surface for the force gage and rests on top of the modified bracket. The spring mount that fits on top of the transducer mimics the old post. It has straight sides up to the height of the original post minus the force gage height and platform thicknesses. It then slopes back to the center at approximately 10 degrees to avoid contact with the spring. The extra height ensures that the spring cannot hop off the mount. A 0.25 in Allen head bolt fits through the spring mount and force gage with the transducer mount serving as the nut. In order to keep the compressor in the same position relative to the bracket, three of the four dead coils at the base of the stock springs were removed to account for the added height from the force gage installation.

The top spring mount was more difficult to design. The standard top spring retainer is shown in Fig. 3. The assembly that was installed in its place is shown in Fig. 5. A 0.25 in thick threaded disk was silver soldered to the top of the shell to provide a solid base for the force gage. The spring mount was designed to be similar to the original spring retainer with enough clearance in the center to allow a 5/16 in hex-head bolt to be installed. The four holes in the side of the spring mount help reduce the weight. The top spring has no dead coils so the spring well in the motor cap was deepened to allow for the extra height of the force gage installation.

STATIC CALIBRATION

The unloaded sensitivity calibrations for the transducers were used when applying the preload. Once the transducers are preloaded these sensitivity values are no longer valid. Since the standard Kistler preload and hardware were not used in this application, a static calibration needed to be performed as described by Soom and Kubler [4].

The static calibration was performed using several weights ranging from 1 lbf (4.448 N) to 8.66 lbf (38.52 N). Limited clearance inside the shell prevented some weights from being used in all of the calibrations. The weights were either set on or hung from the force gage along one of its primary axes. The charge amplifier was set to the long time constant to allow time to read the voltage output from a digital voltmeter before significant decay. The amplifier was reset and if there was not any significant drift in the voltage readings the force gage was unloaded and the steady-state voltage value was recorded. A number of values were taken with each weight and averaged. These averaged values were used in the linear regression fit of the charge sensitivity. The lateral sensitivity values were averaged to find a single sensitivity value for both of these axes. A typical sensitivity calibration for all three axes is shown in Fig. 6.

DYNAMIC CALIBRATION

An impact test was then performed to verify the force gage dynamic calibration. This type of dynamic test requires an instrumented modal hammer which has a built-in force gage and a multi-channel fast Fourier transform (FFT) analyzer. For this type of calibration the spring mount was impacted with a modal hammer along a primary axis of the force gage. The FFT analyzer is set to trigger on the impulse from the hammer and records the time responses of both the hammer and triaxial force gage or accelerometer. These time responses are Fourier transformed into the frequency domain. The individual channel spectra and cross spectra between the channel are averaged over several impacts to improve accuracy. The cross spectrum and auto-spectrum are used to determine the frequency response function (FRF) between the hammer force and the force gage output. This FRF is the dynamic calibration of the transducer. The time responses from an impact test is shown in Fig. 7 and the averaged auto spectrums are shown in Fig. 8. The stiffness of the tip on the hammer and the compliance of the structure limit the frequency range of an impact test. The impulse in the time response narrows as the combined stiffness of the tip and structure increases. As the impulse narrows, the frequency range required to represent it increases and the quality frequency range of the data is generally the -10 dB point in the force autospectrum. The coherence between the input and output is also used as a reference to indicate FRF quality and possible bias error in the frequency response function.

The frequency response function between the hammer impact on the spring mount and the force gage response was expected to be smooth with a rise in sensitivity as the transducer assembly reached its first resonance frequency. The actual response as shown in Fig. 9 turned out to contain a number of apparent resonances beginning at approximately 800 Hz. The time response values showed an impulse for the modal hammer and the impulse followed by a ringing from the triaxial force gage indicative of its functioning as an accelerometer. Several impact tests were then performed attempting to correlate the force gage response to the accelerations of the spring mount or the support bracket. The frequency response between the force gage and an accelerometer mounted perpendicular to the shell on the end of the lower bracket under the force gage shown in Fig. 10 indicated that the distortion in the direct impact calibration was due to the effects of the force gage assembly being mounted on a light dynamic structure and in turn it acting as an accelerometer. The hammer impact was on the outside of the shell at a spot of high curvature which has a lower compliance and should yield a wider frequency range of the impact. This was the only axis and mount that exhibited a fairly clean correlation between the force gage and accelerometer. This means that the spring adapter mass was acting like an inertial mass on top of a force gage. This combination made an accelerometer. The remaining comparison for accelerometer to force for the other directions had many peaks in their frequency response functions. This did not conclusively attribute the peaky FRF from the direct impact calibration shown in Fig. 9 to the dynamic forces from the relative motion between the spring mount and the support bracket.

Another test was devised to verify that the transducer assembly was acting like an accelerometer on a light dynamic structure. A solid steel cylinder with a diameter of 4.0 in and a height of 7.5 in was machined and threaded to accept the 0.25 in bolt used in the side compression mounting spring force gage assemblies. This cylinder weighed approximately 25 lb and should serve as inertial base for the test. This cylinder was estimated to have it first transverse resonance above 12 kHz which is above our frequency range of interest. Axial modes were significantly above this frequency. One of the side force gages and its spring mount were removed and fastened to this cylinder. A direct impact test on the spring mount produced the smooth FRF shown in Fig. 11 which was originally expected. This helped validate the hypothesis that the distortion in the frequency response from a direct impact of the force gage was a result of the shell and bracket response showing up as inertial forces in the triaxial force gage. The force gage was then removed and reinstalled on the support spring bracket and recalibrated statically.

It should be apparent from the above work that the triaxial force gage installation has an acceleration sensitivity. Using the experiments with the accelerometer to force gage comparison when mounted on the flexible shell, one can determine the various force gage sensitivities to acceleration. The inertial mass above the side spring mount force gage is approximately 0.10 lbm (44 gm) which agrees with the FRF in Fig. 10. This means the force gage has an acceleration sensitivity along this axis of 0.1 lbf/g (0.44 N/g). The acceleration sensitivity should be the same for the other axes with the peaks in the other force gage to accelerometer frequency response functions the result of angular acceleration.

CONCLUSIONS

The Kistler 9251A triaxial force gage can be very useful in measuring interaction forces between structures. The modifications necessary to install it require planning to minimize local stiffening and mass loading effects. In-situ static calibration will be necessary when the standard preload and preload hardware are not used. A dynamic calibration using an instrumented hammer can be used to verify the static calibration value and to determine the acceleration sensitivity of the force transducer assembly. When the force transducer is measuring the interaction forces between two structures the acceleration sensitivities may be insignificant. However, as is the case here the acceleration-based inertia forces of the spring mounts are true forces being "felt" by the shell so that they must be accepted as a true excitation force.

RECOMMENDATIONS

The use of the Kistler 9251A triaxial force gage on light dynamic structures needs more study. The hypothesis that the peaky FRF is the result of an inertial mass on top of a force gage needs to be verified. The good correlation between the radial force measurement and the acceleration of the bracket help verify this but does not conclusively prove it. An acceleration sensitivity study in conjunction with a modal survey of the structure may identify and quantify the acceleration sensitivity and the structural vibration modes responsible for the peaks.

ACKNOWLEDGEMENTS

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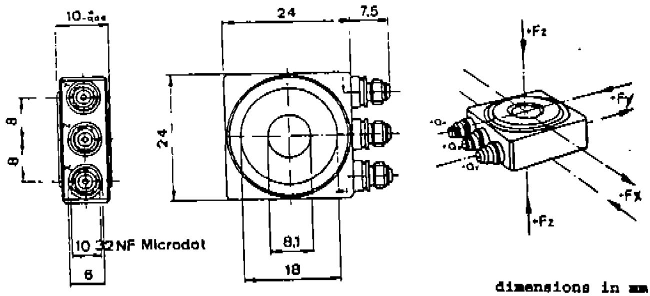

Fig. 1: Kistler 9251A triaxial force gage

(After [1])

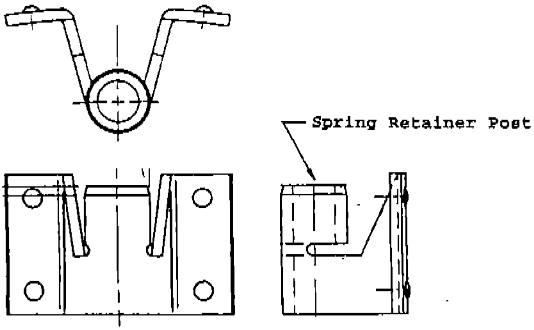

Fig. 2: Standard side spring mounting bracket
(After [2])

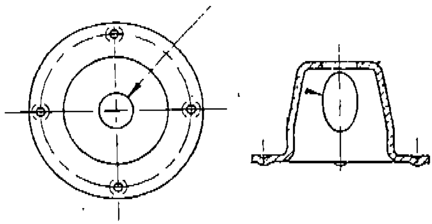

Fig. 3: Standard top spring retainer
(After [3])

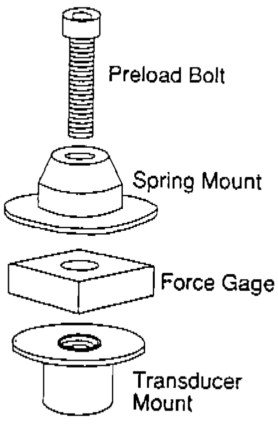

Fig. 4: Side spring force gage assembly

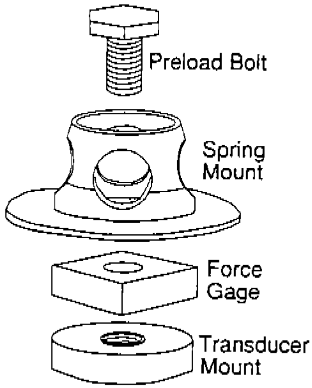

Fig. 5: Top spring force gage assembly

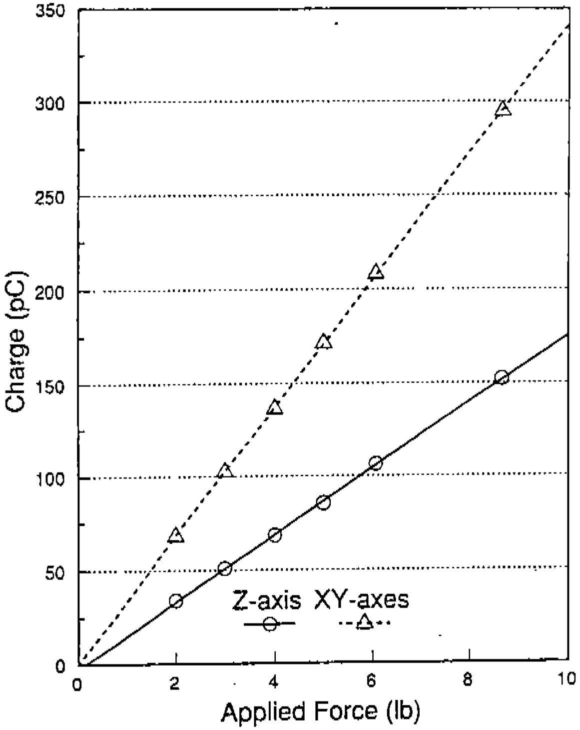

Fig. 6: Static Calibration (S/N 50090)

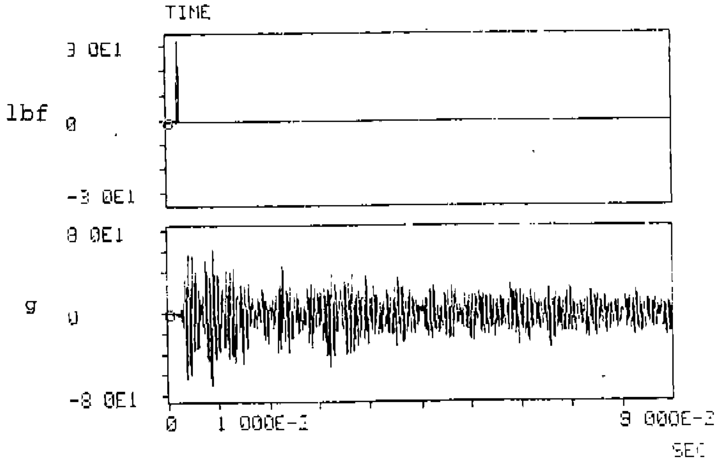

Fig. 7: Time responses from an impact test

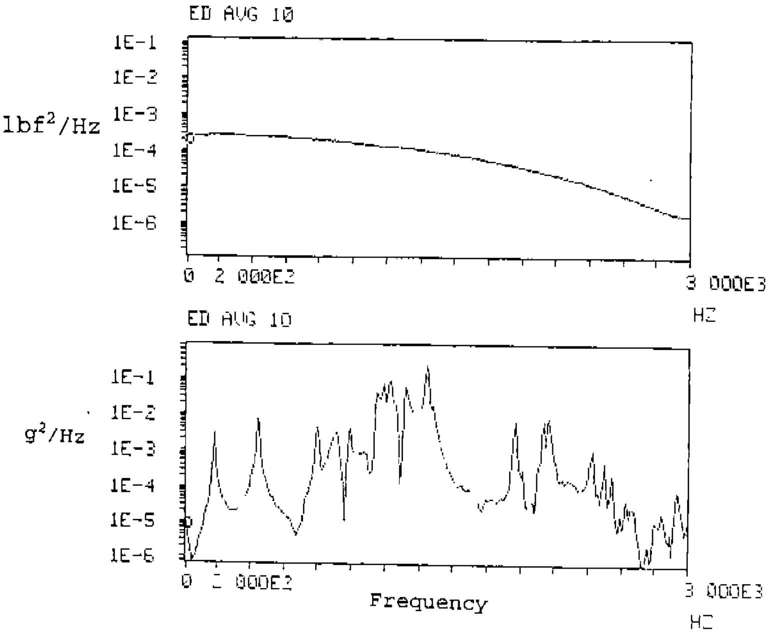

Fig. 8: Energy spectrums from an impact test

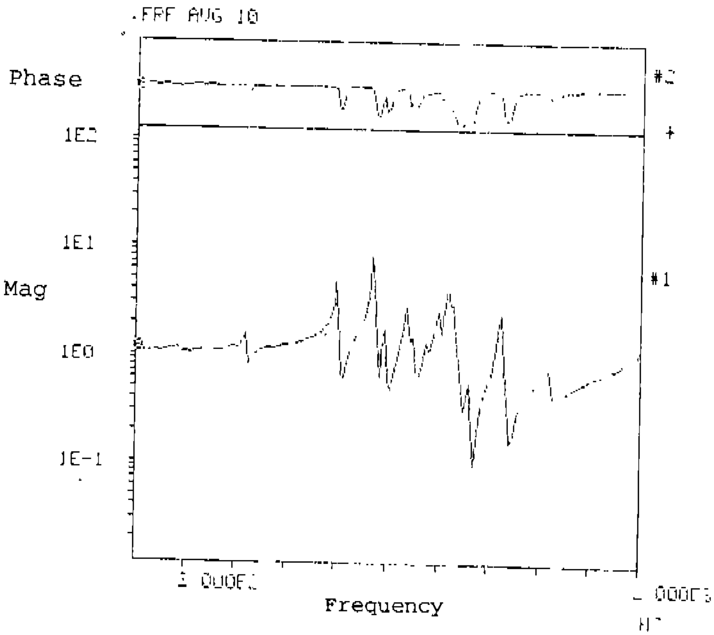

Fig. 9: Side spring mount FRF from direct impact

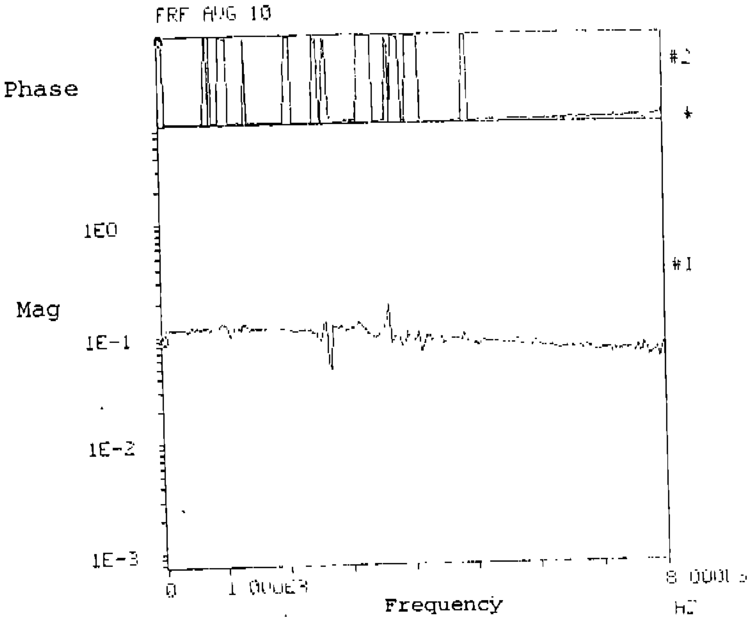

Fig. 10: Force gage to accelerometer FRF

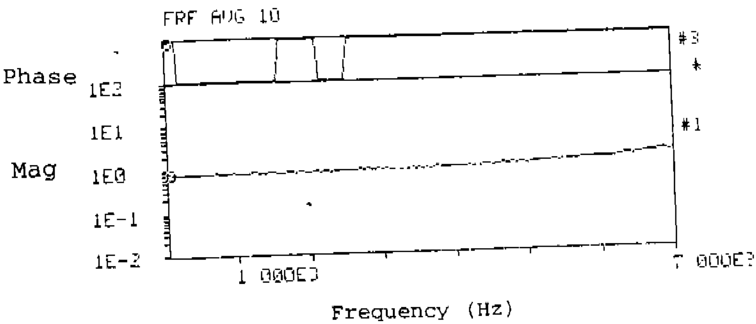

Fig. 11: Direct impact FRF from side spring force gage assembly mounted on the cylinder