Determination of the Dynamic Stiffness of a Hydrostatic Carriage of a High-Speed Milling Machine

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Abstract

Comprehensive vibration data were generated for a hydrostatic carriage system that carries workpiece of a High-Speed Milling Machine at the Machine Tool Research Center of the University of Florida. Machine tool accuracy is critical if final quality of machined parts is of importance. In high speed machining, easy movement of parts is also a desired feature and the use of a hydrostatic bearing and guides provides a way to achieve this. Data was generated using experimental modal analysis; a technique that has found wide use with vibration analysts. To obtain various stiffnesses within the broader carriage, a series of measurement was taking across the length a breadth of the hydrostatic table, the results obtain depicts the dynamic behavior of workpiece table during the high speed operations. The data shows good vibration characteristics of the hydrostatic carriage. Analysis of data in across the whole show high dynamic stiffness as expected from design. The k value obtained for pitching is 1.3386+009N/m. The stiffness of each bearing is k. Estimated bearing stiffness is 0.6693+009N/m. For the case of rolling, obtained is 1.7859+009N/m. Estimated bearing stiffness is 0.8925+009N/m. The variability in stiffness may be due to unequal tolerances in the bearing gaps of individual bearings Modal parameters of the hydrostatic table, which include all important natural frequencies, modal damping ratios and mode shapes, were observable.

Keywords: hydrostatic carriage, experimental modal analysis, dynamic stiffness

1. Introduction

Manufacturing strategies require high productivity and good quality of produced parts simultaneously. In Mechanical Manufacuturing where Milling Machines are also used, the demand for high quality means that the overall accuracy of the machine tool must be high. Productivity and quality are contradictory goals because easy movement of workpiece or tool and achieving accuracy are conflicting objectives. The challenge facing the machine tool industry is to be able to develop new technologies to meet increasing specifications and requirements from consumers.

High speed machining (HSM) is revolutionizing mechanical production especially in the area of aircraft manufacturing where high material-removal rate (MRR) is of great importance (Tlusty, 2000). The biggest challenge to HSM is to achieve good surface quality of machined parts with high material removal rate. The life span of the High Speed Milling Machine must also be comparable with a traditional or conventional type. These demands require continued improvement of existing equipment used for HSM.

Surface quality and surface integrity of machine parts are the direct results of a stable, rigid axis design and construction for which vibration effects due to the dynamics of cutting can be eliminated. When machining, it is highly desirable to cut with high forces, (i.e., large depths of cut and at high speeds with minimal errors to the workpiece). For a machine tool to handle these large forces it must be very stiff and should have good damping characteristics. The stiffness of a machine tool allows for small vibrations under heavier forces, whereas the damping helps to dissipate excess energy generated by these

forces. In short, high stiffness and damping help to reduce the errors associated with machining and allows for high metal-removal rates and longer tool life (Slocum, 1992).

Undesirable attributes of a machine tool are errors in motion and variable stiffness in different configurations. Many of the errors come from the manufacturing of the machine tool itself. Errors in form and surface finish of any of the guideways can lead to undesirable errors of motion in the machine tool and to points in the machinable space where the stiffness is less than desired.

1.1 Accuracy of Machine Tools

Machine tools are made up of linear and rotary axes combined in series with the work and cutting tool. Machine axes inherently have unintended motions, thereby introducing errors. The error motions of each of the axes combine to cause a resultant volumetric error between the work piece and the tool (Tlusty, 2000). These can be decomposed into six fundamental degrees of freedom corresponding with the chosen coordinate system. These errors are referred to as displacement, straightness, roll, pitch, and yaw for a linear axis. Displacement error is the error along the intended-motion direction of the axis. Straightness error is in the two directions perpendicular to the motion axis of interest. Rolling, pitching, and yawing are the rotational motions about the coordinate axes.

Rotational errors are due to inadequate stiffness in the entirety of a machine tool bed. In traditional design of a machine tool, bearings with mechanical contact between elements are used as intermediaries between the moving parts of the machine and the stationary base. The bearings, supporting a carriage, ride along an accurately machined way and act as springs between the moveable carriage and the stationary ways. Roller-pack, re-circulating ball, and sliding element are examples of bearings that use mechanical contact as a method of supporting as well as stiffening the machine tool. While mechanical-contact bearings can be made to allow high stiffness between the carriage and the base, they are susceptible to the before-mentioned errors in the form and surface finish of the ways they ride on.

1.2 Hydrostatic Bearings

An alternative to mechanical contact bearings is the use of hydrostatic bearings. Hydrostatic bearings use a thin film of high-pressure fluid to support a load (in this case the carriage). Hydrostatic systems on machine tools are evolving as next-generation designs to achieve this difficult task of high surface finish. They are known to have such properties as high geometric precision, high speed of response, suppression of 'stick-slip' effect at low speed, minimum friction factor, low heat intake, durability, high stiffness, high vibration damping and running accuracy (Peterson, 2000). This research determines the stiffness of a hydrostatic carriage, bearing and guideway on a High Speed Machine tool.

2. Methodology

2.1 Experimental Modal Analysis

Experimental modal analysis, basically, is a procedure of "experimental modeling". The primary purpose is to develop a dynamic model for a mechanical system using experimental data. The data generated is a set of response properties such as measured frequency response functions. Generally a modal model consisting of natural frequencies, modal damping ratios and mode shape vectors is obtained from the frequency response functions. Complex vibrating systems usually consist of components that possess distributed energy-storage and energy dissipative characteristics. In these systems, the inertial, stiffness, and damping properties vary (piecewise) continuously with respect to the spatial location. Consequently, partial differential equations with spatial coordinates, e.g. Cartesian coordinates (x, y, z) and time t as independent variables, are necessary to represent their vibration response.

A distributed (continuous) vibrating system can be approximated (modeled) by an appropriate set of lumped masses properly interconnected using discrete spring and damper elements. Such a model is called a lumped-parameter model or discrete model. An immediate advantage resulting from this lumped-parameter representation is that the system equations become ordinarily differential equations. Often, linear springs and linear viscous damping elements are used in these models. The method is based on the fact that these idealized systems (models) have preferred frequencies and geometric configurations (or natural modes), in which they tend to execute free vibration. An arbitrarily response of the system can be interpreted as a linear combination of these modal vibrations; and as a result, its analysis can be conveniently done using modal techniques, an important tool in vibration analysis, diagnosis, design, and control. In some systems, mechanical malfunction or failure can be attributed to the excitation of their preferred motion such as modal vibrations and resonances. By modal analysis, it is possible to establish the extent and location of severe vibrations in a system. For this reason, it is an important diagnostic tool. For the same reason, modal analysis is also a useful method for predicting impending malfunctions or other mechanical problems. Structural modification and substructuring are techniques of vibration analysis and design, which are based on modal analysis. By sensitivity analysis methods using a "modal" model, it is possible to determine what degrees of freedom of a mechanical system are most sensitive to addition or removal of mass and stiffness elements. In this manner, a convenient and systematic method can be established for making structural modifications to eliminate an existing vibration problem or to verify the effects of a particular modification. A large and complex system can be divided into several subsystems that can be independently analyzed. By modal analysis techniques, the dynamic characteristics of the overall system can be determined from the subsystem information. This approach has several advantages, including:

(1) subsystems can be developed by different methods such as experimentation, or other modeling techniques

and assembled to obtain the overall model;

(2) the analysis of higher order system can be reduced to several lower-order analyses; and

(3) the design of a complex system can be done by designing and developing its subsystems separately. Modal control, a technique that employs modal analysis, is quite effective in the vibration control of complex mechanical systems.

2.1 Experimental Set-Up

The experiment setup included a computer, an ICP accelerometer, two amplifiers, and an instrumented impact hammer together referred to as dynamic analyzer. The computer ran on the TXF software. The TXF 98 version developed by Manufacturing Laboratory Incorporated was used. It is specifically designed for modal testing of machine tools using impact test methods. It utilizes data acquisition and computing routines to produce a transfer function in 10, 5, 2.5 KHz ranges. TXF monitors two signals to produce transfer functions when the impact is made on the table with the hammer; a high impedance charge signal is generated by the piezoelectric sensing element. The hammer provides a simple means of exciting the table into vibration. It has a set of different tips and heads, which serve to extend the frequency and force level ranges for testing a variety of structures. Integral with the impacting end of the hammer is a force transducer, which detects the magnitude of the force, felt by the impact end, which is assumed to be equal and opposite to the force experienced by the table. The magnitude of the force applied is determined by the mass of the hammer and the velocity with which it is moving when it hits the table. The TXF 98 has an inbuilt program to reject impacts that resulted in an overload.

An accelerometer is a linear seismic transducer, which produces an electric charge proportional to the applied acceleration. A simple model of an accelerometer is shown in

A mass is supported on a piece of piezoelectric ceramic crystal which is fastened to the frame of the transducer body. Piezoelectric materials have the property that if they are compressed or sheared, they produce an electric potential between their extremities, and this electric potential is proportional to the amount of compression or shear. As the frame experiences an upward acceleration it also experiences a displacement. Because the mass is attached to the frame through the spring-like piezoelectric element, the resulting displacement it experiences is of different phase and amplitude than the displacement of the frame. This relative displacement between the frame and mass causes the piezoelectric crystal to be compressed, giving off a voltage proportional to the acceleration of the frame.

The Machine Tool Research Center has a wide range of accelerometers with varying sensitivities. Sensitivity ranges are from 5.30mV/g(1851m/s2/V) to 1086mV/g(9.03m/s2/V). High sensitivity is desired to get accurate readings of frequency response functions. The higher the sensitivity the heavier and larger the transducer and the more space required by the accelerometer. After some test using different accelerometer, an accelerometer of 101.3 mV/g (98.7m/s2/V) was selected because of its good response in the frequency range of vibration of the table.

In order to determine the dynamic characteristic of the hydrostatic bed, transfer functions were taken along and across the entirety of the bed. A transfer function is a function that represents the cause and effect relationship of a force and the vibration it creates. A transfer function can either be a direct transfer function or a cross transfer function. A direct transfer function is obtained by applying a force to a point and measuring the vibration at the same point. A cross transfer function is obtained by applying a force at one point and measuring the response at a different point. When the impact is made on the table with the hammer, a high impedance charge signal is generated by the piezoelectric sensing element. This is then converted by the built-in electronics in the ICP accelerometers into usable low impedance voltage signals that can be readily transmitted over ordinary two wire or co-axial cables. The electronics within the ICP accelerometers require excitation power from a constant current regulated, DC voltage source. So one DC voltage supply is provided to each accelerometer. The cables then run into the FFT analyzer where the computer performs a Fast Fourier Transform (FFT) on both signals. Then a double integration is performed on the accelerometer signal in order to convert from acceleration (m/s) to displacement (m). The displacement signal is divided by the hammer signal to give the transfer function across a specified frequency range. The real and imaginary part of the transfer function is plotted against frequency. The natural frequency of the system can be read directly from the transfer function plot. By reading peak points of the imaginary value the natural frequency for various modes of vibration is obtained. The difference between the frequencies Δf of the two peaks of the real part of the transfer function along with the natural frequency can be used to calculate the damping ratio (ζ) of the mode.

$$\xi = \frac{\Delta f}{2f_n}$$

where fn is the natural frequency of vibration of the mode.

The dynamic stiffness, k is found from the imaginary peak value, H, referred to as a dynamic flexibility.

$$k = -\frac{1}{2H\xi}$$

The first set of experiments captured modes of vibration along the bed of the hydrostatic table. HSM 2 hydraulic system was turned on and allowed for flow of oil into the four bearings supporting the table. The circulating pressure was recorded as 55 bar and some time was allowed for steady conditions to be achieved. The oil temperature was also recorded as 78oF.

The accelerometer was placed at one end of the table and the impact hammer was applied at five different points along the table. It includes one direct transfer function and four cross transfer functions. The location of the accelerometer was changed to the other end of the table and an impact was made at an arbitrary point to determine whether a mode has not been recorded. The explanation to this is that if the accelerometer is positioned at or very close to a node of one or more of the table's mode, it will be difficult to make an effective measurement of that particular mode.

The second set of experiments was performed across the bed of the hydrostatic table. Three impacts were made. The position of the accelerometer was placed in the line of impact to capture as much modes of vibration as possible. transfer functions were obtained for the entire table to capture coupling modes present in the vibration of the table. Each result was stored in the computer for analysis.

3. Results and Discussion

Results are in the form of a frequency response functions and show plots of imaginary and real values against frequency in the range of vibration of the table. The first three frequency response functions are transfer functions of experiment about the X-axis. Modal parameters that is stiffness and modal damping ratios are calculated for each result. These are summarized in a table below each figure. The next frequency response functions show results of experiment that investigates the transfer functions across the table in the direction of the Y-axis.

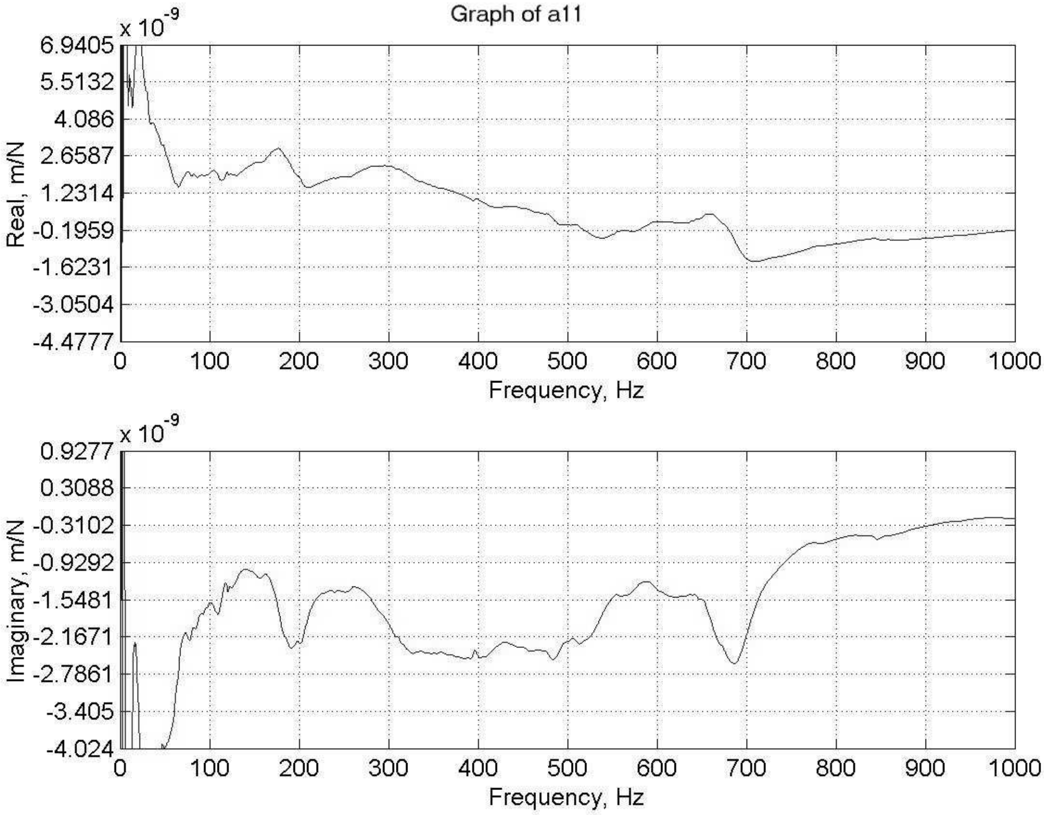

Figure 3-1 Transfer function of the hydrostatic bed along the table.

Table 3-1 Modal damping ratios and stiffness of the hydrostatic.

Real magnitude		Imaginary magnitude	Natural frequency	Damping ratio	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	f_n (Hz)	ζ	K (N/m)
208.66	176.24	-2.36E-09	190.73	0.08498922	2.50E+09
658.42	708.77	-2.59E-09	683.21	0.03684814	5.24E+09

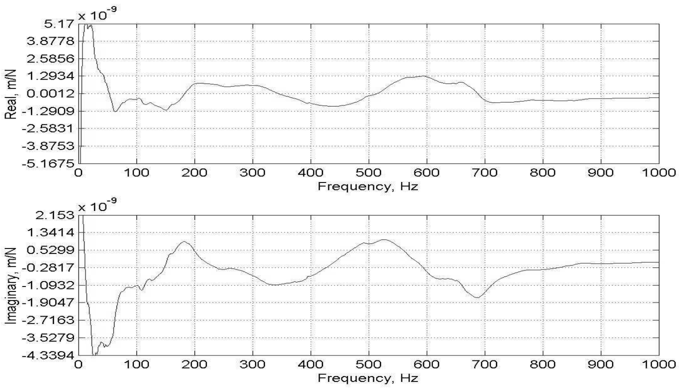

Figure 3-2 Transfer function of the hydrostatic bed along the table.

Table 3-2 Modal damping ratios and stiffness of the hydrostatic bed.

Real magnitude		Imaginary magnitude	Natural frequency	Damping ratio	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	f_n(Hz)	ζ	K (N/m)
200.27	150.30	9.26E-10	182.34	0.137024 24	3.94E+09
440.22	311.28	-1.09E- 09	340.27	0.189467 18	2.43E+09
712.59	662.23	-1.67E- 09	687.41	0.036630 25	8.17E+09
594.33	451.28	1.01E-09	526.43	0.135868 07	3.63E+09

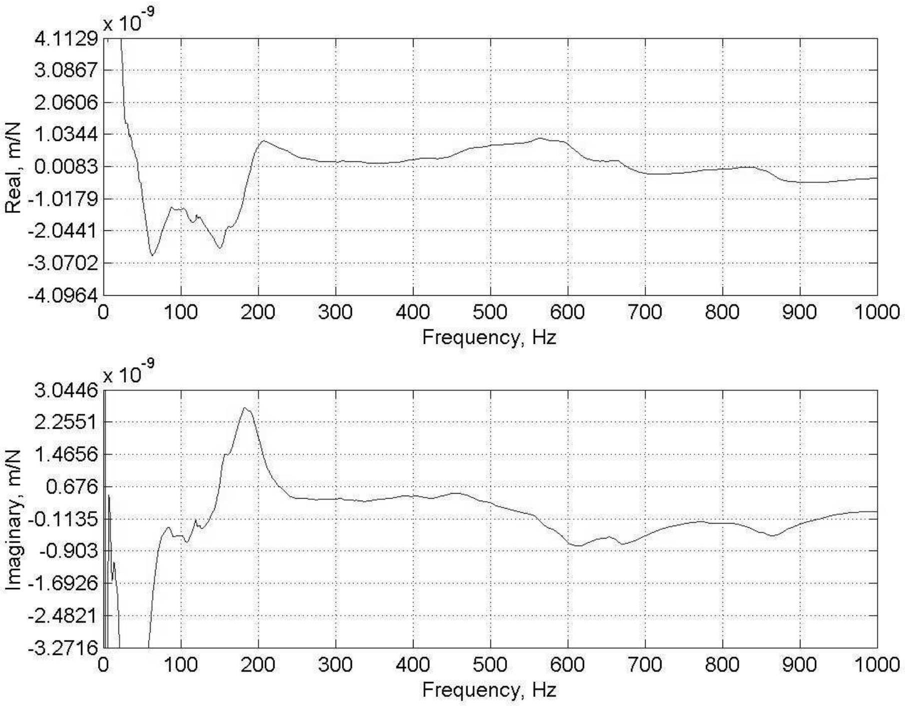

Figure 3-3 Transfer function of the hydrostatic bed at along the table.

Table 3-3 Modal damping ratios and stiffness of the hydrostatic bed.

Real magnitude		Imaginary magnitude	Natural frequency	Damping ratio	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	nf (Hz)	ζ	K (N/m)
206.38	149.54	2.57E-09	181.58	0.1565150 35	1.24E+09
713.35	663.76	-7.06E+00	677.87	0.03657781	1.94E+10
886.92	844.19	-5.17E-10	869.37	0.0245752 67	3.94E+10
1165.80	1052.90	7.76E-10	1148.2	0.0491639 09	1.31E+10

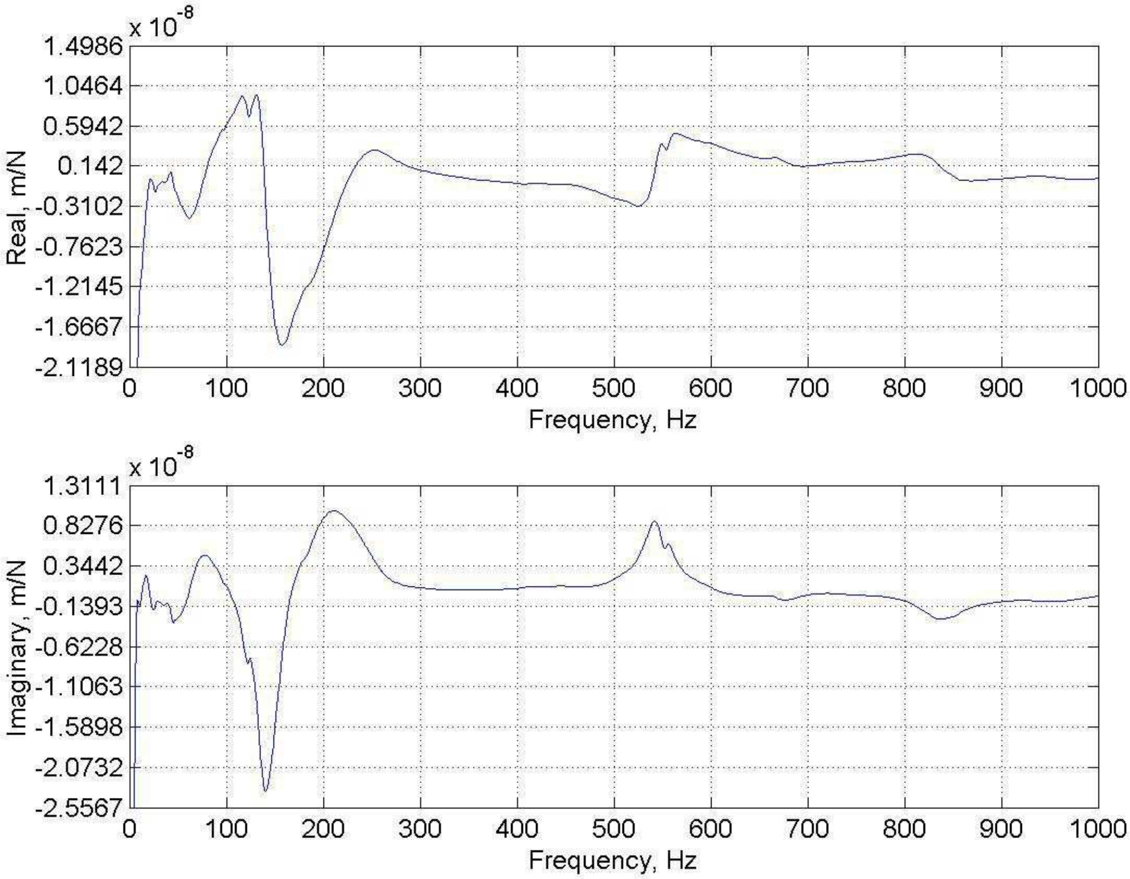

Figure 3-4 Transfer function of the hydrostatic bed across the table along Y-axis

Table 3-4 Modal damping ratios and stiffness of the hydrostatic bed along Y-axis.

Real magnitude		Imaginary Magnitude	Natural frequency nf	Damping Ratio ζ	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	(Hz)		K (N/m)
158.69	129.7	-2.36E-08	140.76	0.102976698	2.05E+08
566.48	530.24	+8.70E-09	540.92	033498484	1.72E+09
862.12	811.77	-2.97E-09	836.94	0.030079815	5.60E+09

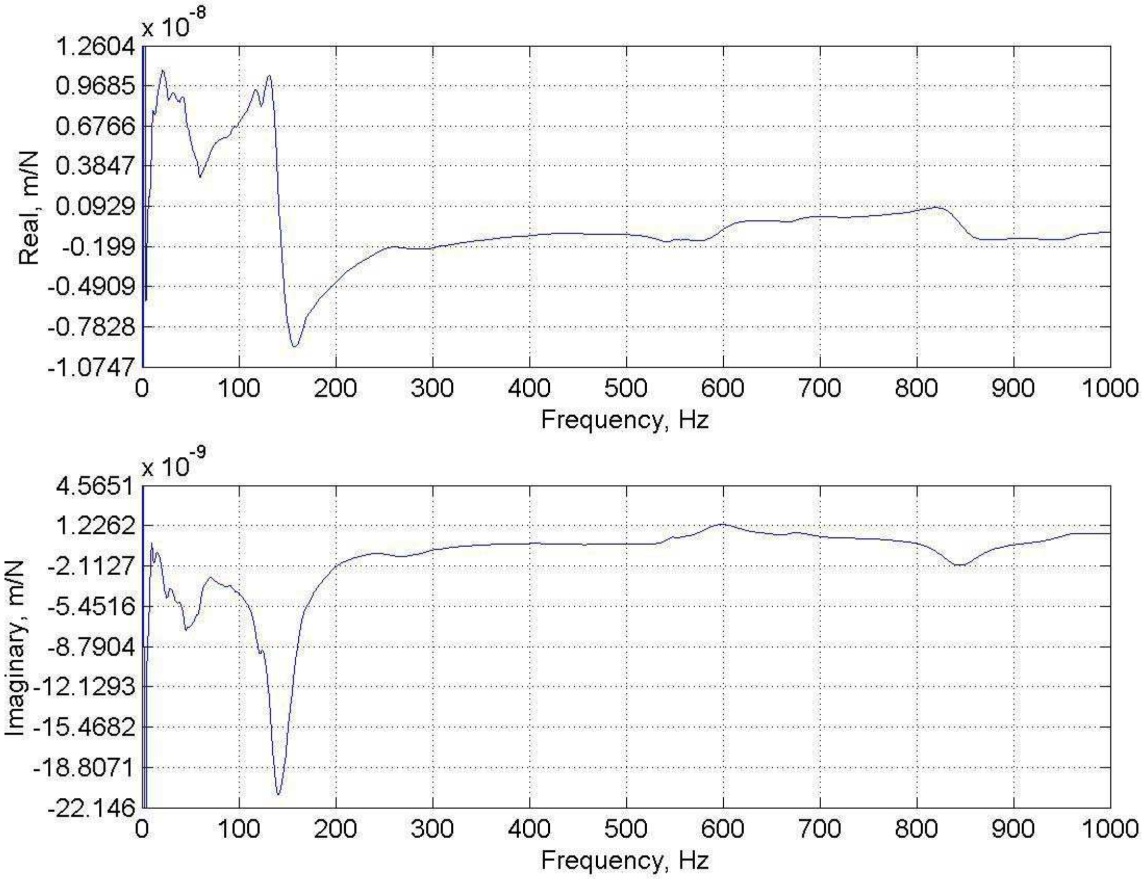

Figure 3-5 Transfer function of the hydrostatic bed across the table along Y-axis

Table 3-5 Modal damping ratios and stiffness of the hydrostatic bed along Y-axis

Real magnitude		Imaginary magnitude	Natural frequency nf	Damping ratio ζ	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	(Hz)		K (N/m)
158.31	129.70	-2.11E-08	140.38	0.10190198	2.33E+08
617.98	574.87	1.26E-09	592.8	0.036361336	1.09E+10
865.94	822.83	-2.03E-09	840.76	0.025637518	9.62E+09

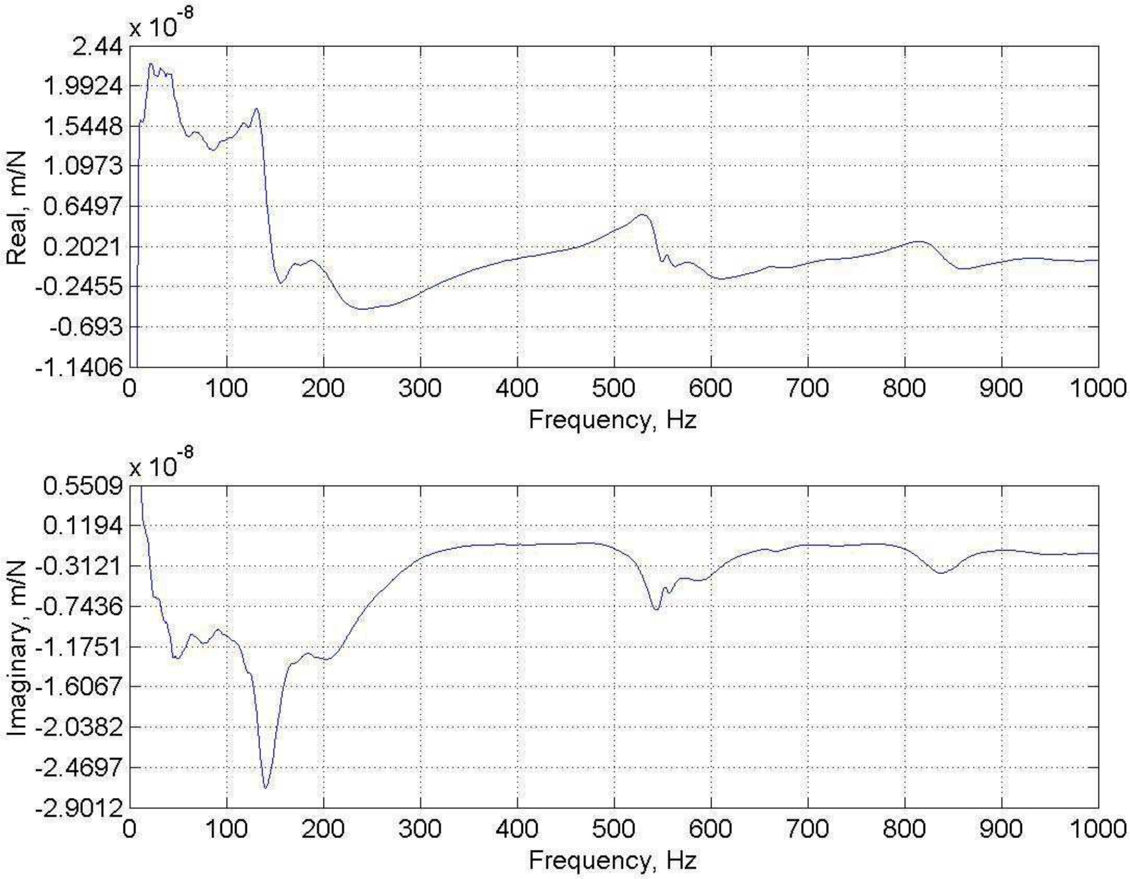

Figure 3-3 Transfer function of the hydrostatic bed across the table along Y-axis

Table 3-3 Modal damping ratios and stiffness of the hydrostatic bed along Y-axis

Real magnitude		Imaginary magnitude	Natural frequency	Damping ratio	Stiffness
Max (Hz)	Min (Hz)	H (m/N)	nf (Hz)	ζ	K (N/m)
158.69	130.80	-2.69E-08	140.76	0.099069338	1.87E+08
562.29	529.86	-7.45E-09	540.54	0.02999778	2.24E+09
853.73	817.87	-3.88E-09	839.61	0.021355153	6.03E+09

Also modal damping ratios and stiffness were calculated for these set of experiments and summarized in tables preceding figures. They were obtained when the impact hammer was applied across the bed. The piezo- electric accelerometer was placed at the end of the bed at mid position along the bed to capture as much frequency of excitations as possible. These experiments were performed to have a full data of vibration characteristics of the entire hydrostatic bed. The intent of the study is to identify the modal parameters of the hydrostatic table. These include all important natural frequencies, modal damping ratios and mode shapes. The mode shapes of hydrostatic are drawn for each natural frequency. Also the mode shape vector is determined.

A dominant mode at a frequency of 185Hz was observed along the bed in the direction of the X-axis. It is noted from the graph that a bending type of vibration is present. Modal stiffness ranges from 1.24E+09 N/m and 5.99E+09 N/m. Damping ratios are in the order of 0.15. The mode shape vector with normalization at mid- point on the table is

$$\begin{Bmatrix} -0.91 \\ 0.36 \\ 1 \\ 0.63 \\ -0.41 \end{Bmatrix}$$

A second distinct and dominant mode was found at a 685 Hz. along the bed in the direction of the X-axis.. A vibration form of the pitching type is identified in this mode of vibration. Calculated stiffness is in the range of 8.17E+09 N/m and 3.94E+10 N/m. Modal damping ratios are in the magnitude of 0.03.The modal shape vector with normalization at mid-point on the table is given as

$$\begin{Bmatrix} 3.66 \\ 2.38 \\ 1 \\ -2.31 \\ -1.95 \end{Bmatrix}$$

Three frequencies were distinct in experiment (b) where the hammer was applied across the bed in the direction of Y-axis. A dominant mode at a frequency of 140Hz was observed along the bed in this direction. It can be inferred, that a translational type of vibration is present. Stiffness ranges from 1.87E+08 N/m and 2.33E+08N/m. Damping ratios are in the magnitude of 0.1. The mode shape vector with normalization at point 2 on the table is

$$\begin{Bmatrix} 1.12 \\ 1 \\ 1.27 \end{Bmatrix}$$

The second dominant mode in the direction of the Y-axis across the bed is at 540Hz. A vibration form of the rolling type is identified in this mode of vibration about the x-axis. Calculated stiffness is in the range of 1.72E+09 N/m and 2.24E+10 N/m. Modal damping ratios are in the magnitude of 0.03. The modal shape vector with normalization at point 2 on the table is

$$\begin{Bmatrix} 6.91 \\ 1 \\ -9.91 \end{Bmatrix}$$

A third distinct and dominant mode was found at 838 Hz. along the bed in the direction of the Y-axis. Figure 6-5 shows the mode shape at this frequency. A vibration form of the translational type is identified in this mode of vibration. Calculated stiffness are in the range of 5.60E+09 N/m and 9.62E+09 N/m. Modal damping ratios are in the magnitude of 0.03.The modal shape vector with normalization at mid-point on the table is given by

$$\begin{Bmatrix} 1.46 \\ 1 \\ 1.92 \end{Bmatrix}$$

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The hydrostatic system has high damping values especially at low frequencies. In some modes a modal damping ratio of 0.15 is obtained which is about four times that obtained on a mechanical bed. Overall the hydrostatic system has a better damping characteristic than the mechanical bed.

To determine the stiffness of the bearings, the rigid body motion of the table is studied. The rigid body motions of the hydrostatic table are at frequencies 540 Hz and 685 Hz. All four hydrostatic bearings have the same size with equal volumetric flow of damping oil, implying that the total bearing stiffness of the table is equal to the sum of individual stiffness of each bearing.

The rotational plus translational motion of the table is considered. Here two rigid modes are of interest, the rigid mode due to pitching of the table and the mode due to rolling. Analysis of such motion was discussed in the experimental set up.

The equation

$$\begin{vmatrix} k_1 + k_2 - mw^2 & -(k_1 L_1 - k_2 L_2) \\ -(k_1 L_1 - k_2 L_2) & k_1 L_1^2 + k_2 L_2^2 - Jw^2 \end{vmatrix} = 0$$

Since the table is rigid and symmetrical in this equation with equal stiffness, that is

$k_1 = k_2 = k$ and $L_1 = L_2 = L/2$

Where k = measured stiffness of bearings

And L = distance between two bearing support points of the table.

A new determinant is obtained as

$$\begin{vmatrix} 2k - w^2 m & 0 \\ 0 & kl^2/2 - w^2 J \end{vmatrix} = 0$$

And that; $(2k - w^2 m)(kl^2/2 - w^2 J) = 0$

$$or\ \ 2k^2 l^2/2 - 2kw^2 J - kl^2 w^2 m/2 + w^4 Jm = 0$$

$$(Jm)w^4 - (2kJ + kl^2 m/2)w^2 + k^2 l^2 = 0$$

$J = m*(a^2 + b^2)/12$ where a= thickness of table and b = length of table.

The J calculated is the moment of inertia of table along the table and determines the natural frequency in the pitching mode. J is also determined for the case of rolling motion.

Measured pitching frequency = 680 Hz.

Measured rolling frequency = 540 Hz.

From the relation

$(2k - w^2 m)(kl^2/2 - w^2 J) = 0$, from which the stiffness of the each bearing is determined, and measured table values and calculated table properties are given below.

a = 0.0762, b = 1.016, m = 280, c = 0.7112.

$$J_1 = m*(a^2 + b^2)/12,\ \ l_1 = 0.8128,\ \ l_2 = 0.5588,\ \ J_2 = m*(a^2 + b^2)/12.$$

The stiffness k is obtained from the rotational part of the equation

$(2k - w^2 m)(kl^2 / 2 - w^2 J) = 0$, that is

$(kl^2 / 2 - w^2 J) = 0$, or

$$k = \frac{2w^2 J}{l^2}$$

The k value obtained for pitching is 1.3386+009N/m. The stiffness of each bearing is k. Estimated bearing stiffness is 0.6693+009N/m. For the case of rolling, obtained is 1.7859+009N/m. Estimated bearing stiffness is 0.8925+009N/m. The variability in stiffness may be due to unequal tolerances in the bearing gaps of individual bearings.

4. Conclusion

Results obtained with experimental modal analysis proved to be qualitatively and quantitatively excellent. The dynamic parameters of the hydrostatic table as determined are good, that is the k value obtained for pitching is 1.3386+009N/m. The stiffness of each bearing is k. Estimated bearing stiffness is 0.6693+009N/m. For the case of rolling, obtained is 1.7859+009N/m. Estimated bearing stiffness is 0.8925+009N/m. The variability in stiffness may be due to unequal tolerances in the bearing gaps of individual bearings. the information content of excitation and response signal is abundantly sufficient, the reliability of modal parameter extraction methods is accepted, the table's constraints respond to good modal analysis practice requirements. It was observed that both High speed milling machine with hydrostatic bed had high stiffness, good damping values and stable mode shapes to support production operations. The dynamic response expected for a hydrostatic system, which is high stiffness and damping has been confirmed. The results obtained from the calculation of stiffness in the hydrostatic bearing are acceptable.

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