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# ANALYSIS OF A DUAL-SPINDLE HIGH-SPEED DRILLING MACHINE HEAD

by

Narinder Singh CHANA

A Thesis
submitted to the
Faculty of Graduate Studies and Research
through the Department of
Mechanical Engineering in partial Fulfilment
of the requirements for the Degree of
Master of Applied Science
at the University of Windsor

Windsor, Ontario, Canada

1994

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# ABSTRACT


Vibration characteristics of a high-speed spindle and Dual-Spindle High-Speed Drilling Machine Head were studied to determine if the modal, free-run, and cutting characteristics allow for a stable machining system at the required operating speed of 167 Hz (10 000 RPM). A finite element analysis of the high-speed spindle model was conducted to determine the validity of the finite element method in modeling the bearing support structure of the spindle-bearing assembly.

The finite element analysis under simply applied boundary conditions does not provide valid results in terms of the displacement shape and the natural frequency of the spindle-bearing support structure. A detailed understanding of the structure's boundary interactions is required to correctly model such an assembly. Because of its scope, this problem was separated for future work.

Modal analysis indicates that the lowest natural frequency of the spindle assemblies occurs at 712 Hz (43 200 RPM). This value is beyond the required operating speed of 167 Hz (10 000 RPM), and with only two cutting edges on the drill bit, vibration problems during cutting were not expected.

Free-run characteristics indicate a fairly stable system with low vibration, and bearing thermal levels. Run-up and run-down tests were carried out at speeds up to 267 Hz (16 000 RPM).

Vibration levels during metal cutting with two different diameter drills were only slightly higher than those at the free run tests. There was no indication of any tendency to produce chatter or any other form of excessive vibration.

The vibration analysis of the dual-spindle high-speed drilling machine head demonstrates that the system is capable of operating under the required working parameters.

**Dedicated to my family**

**for understanding and coping with my idiosyncracies during this study.**

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## NOMENCLATURE

| | |
|---|---|
| ° | degrees |
| °C | degrees Celsius |
| °F | degrees Fahrenheit |
| | |
| AMRP | Advanced Machining Research Project |
| B&K | Brüel & Kjær |
| CAD/CAM | Computer-Aided Design / Computer-Aided Machining |
| CAFE | Clean Air Fuel Efficiency |
| DARPA | Defense Advanced Research Project Agency |
| DN | bearing bore diameter in millimetres times the rotational speed in revolutions per minute |
| DOF | Degree of Freedom |
| DSA | Dynamic Signal Analyzer |
| EU | Engineering Units |
| FE | Finite Element |
| FEA | Finite Element Analysis |
| FEM | Finite Element Method |
| FFT | Fast Fourier Transform |
| FRF | Frequency Response Function |
| HP | Hewlett-Packard |
| hp | horsepower |
| in | inch |
| ipr | inches per revolution |
| HSM | High-speed Machining |
| HSS | High Speed Steel |
| Hz | Hertz (cycles per second) |
| kg | kilograms |
| LAM | Laser Assisted Machining |
| lbs. | pounds |
| mm | millimetres |
| mpr | metres per revolution |
| MRR | Material (or Metal) Removal Rate |
| RMS, rms | Root Mean Square |
| RPM, rpm | Revolutions Per Minute |
| SDOF | Single Degree of Freedom |
| sfm | surface feet per minute |
| smm | surface metres per minute |
| US | United States (of America) |

# 1. INTRODUCTION

Conventional machining practices at surface speeds below 185 surface metre per minute (smm) or 600 surface feet per minute (sfm) and feed rates below 0.004 to 0.030 inches per revolution (ipr) or 102E-6 to 760E-6 metres per revolution (mpr)[1] restrict the amount of time to machine a product [1,2,3]. This increases the manufacturing cost, especially for products requiring long machining periods. In 1985, the United States of America spent approximately $115 billion (US) to perform metal cutting operations using the conventional machining technology [4].

Manufacturing costs in machining operations are dependent on the length of cutting period, loading and unloading (non-cutting) time, labour costs, and additional overhead costs. To improve productivity, reduce costs, and manufacture a product at a competitive price, it is important to analyze the factors affecting the manufacturing costs. The goal being the reduction of time spent in machining a product.

1 Note : Values given for light alloys such as aluminum for drilling. Conventional machining speeds vary for each type of metal being removed.

Current long-term trends that will influence future machining technology and tools include :

1. the replacement of many mechanical components with electronically-controlled functions;

2 a shift towards a variety of different materials including composites, plastics, and other non-metallics;

3. an increasing amount of die castings, regular castings manufactured closer to the final shape and tolerance [5];

4. international technology advancements and changing political tides (ie. economic trading blocks);

5. demands of large manufacturers to use fewer component suppliers; and

6. CAFE, recycling and other environmental requirements that affect the manufacturing sector.

The aforementioned trends indicate that machining of particular products can be reduced by as much as 25-30%, and these changes will affect the design of machine tools, spindles, cutting tools and machining processes.

The use of modern materials requires the use of new machine tools and technology to meet the current and future market trends. Conventional machining processes and technology cannot meet these growing demands. In terms of the metal removal rate (MRR), and of reduction in cutting time, high-speed machining (HSM) offers a great economic and technological gain.

High-speed machining is generally defined as machining at higher than conventional speeds. High-speed machining, in relation to conventional machining, increases the material removal rate, reduces the cycle time, lowers the cutting forces, improves the part quality, decreases manpower, and lowers the cost per machined part.

The cutting speed achievable with high-speed machining is dependent on the material to be machined, the type of machining operation, machine tool characteristics, and the part requirements. A high-speed machine requires higher spindle speeds combined with higher feed rates and spindle power. An integrated HSM system requires the use of capable high-speed tooling systems (spindle assembly, tool holders, and power drives) that must be balanced to maintain its structural integrity.

In normal machining operations, the machine spindle, structure, cutting tool and workpiece interact with each other, generating high frequency vibrations that may significantly affect the structural characteristics and machining performance.

The monitoring or preventative diagnostics in conventional machining of critical components is essential in maintaining a productive and trouble free operating machine. The monitoring of and design against critical component failures in high-speed machining becomes even more apparent.

New design problems arise that are not seen in conventional machining designs and operations. Most of these problems are mainly caused by the higher speed of operation. Centrifugal forces, unbalance, and misalignment effect increase in proportion to the square of the speed ratio and have the potential of generating excessive vibrations. In addition, high-speed machining systems require the support structure to be more stable and tolerant of the high-speed operating conditions.

It is critical, to properly implement a high-speed machining system or design, to ensure that the machining unit is capable of performing the desired machining operations at the required speeds. If the design reliability of a high-speed machining unit is not properly investigated, undesirable machining and safety concerns could arise. The practical limitations of a high-speed machining unit are the high-speed spindle design and the machining safety.

The focus of this study was to investigate the vibration characteristics of a proposed Dual-Spindle High-Speed Drilling Machine Head unit to determine if its design is capable of operating at the required operating conditions.

The investigation for this study is developed through the following objectives:

1. Compare the Finite Element (FE) modal analysis with the results of modal analysis of the spindle supported in a V-block test structure, and the actual structure (machine head) to determine the validity of the Finite Element Method (FEM) approach to this problem.

2. Determine by testing of the machining unit, that the spindle performs satisfactorily during free-run and cutting operations.

# 2. LITERATURE SURVEY

A literature review was conducted to aid in understanding some of the aspects and components affecting a high-speed machining system. More specifically, towards the drilling of aluminum workpieces at high speeds. The literature reviewed is used to gain a perception of the high-speed machining fundamentals and requirements and the analysis tools to ensure a capable operating high-speed machine design.

## 2.1 HIGH-SPEED MACHINING

High-speed machining has generally been referred to as machining at much higher cutting speeds than conventional machining practices. Although much research has been conducted into high-speed machining, there does not exist a common predetermined or an agreed upon value to distinguish when a machining process is operating either under conventional or high-speed machining processes.

The lack of a common high-speed or conventional machining value, usually given in surface feet per minute or surface metres per minute (sfm or smm), can be attributed to the variations of machining operations. Operations such as drilling, milling, and turning, involve the cutting tool geometry, the material workpiece properties, and the machining unit characteristics.

A relative lack of research conducted into high-speed drilling exists in comparison with milling and turning studies, upon which a majority of the high-speed numbers and configurations are based. This aspect is astonishing considering that 95 percent of all workpieces contain holes, 80 per cent of all chips by weight come from drilled holes, and 70 percent of these chips are produced by drills under 7/16 inch (11 mm) diameter [6].

High-speed machining currently is applicable to non-ferrous materials such as aluminum, magnesium, and plastics. The machinability characteristics, or the lower melting temperature of the non-ferrous workpiece with respect to the cutting tool melting temperature permits unlimited material removal rates to be achieved. Material removal rates for aluminum as high as 73 000 smm have been achieved, with speeds of 1 500 to 4 500 smm being used in some studies [4, 5, 7,8].

The increased cutting speeds result in a decrease in the chip thickness and correspondingly a decrease in the cutting force [9]. The local temperature around the chip rises and the thickness diminishes reducing the stresses and forces required to break the chip continuously in the shear zone [10, 11].

The material removal rate is dependent on the on the following factors [12, 13]:

- size and type of machine;
- the power available;
- the type of cutting tool;
- the material to be machined;
- the speed and depth of cut;
- the machining operation;
- part requirements; and
- chip disposal safety.

The boundary for the speed of machining is attributed to the temperature limitations of the tool material with respect to the workpiece material.

The implementation of a high-speed machining system provides the following benefits [10, 13, 14]:

- increased material removal rates;
- reduced cycle time;
- reduced tooling and setup time;
- reduced cutting forces;
- improved surface finish;
- increased utilization of the machine tool;
- generally reduced production cost based on cost per part produced.

Difficulties and challenges that face the implementation of a high-speed machining system are [13, 14] :

- the need, suitability, and limitations of application;
- risks of investment and obsolescence;
- the status of the technology (new, old, fading);
- competing technologies, and integration of technologies;
- the research and development required;
- safety concerns.

The process of high-speed machining is limited by the capability of the tooling, the machine tool spindle rigidity, chip removal, noise and vibrations, fixture stability, reliability, damping, stiffness characteristics of the spindle bearing assembly, the response mechanisms, and in-process monitoring [14]. To ensure the machining capability in terms machine-tool-spindle interactions and stability of a high-speed machining design, vibration studies involving modal, free-run, and cutting are used. Finite element analysis is currently used as a comparison and design modification tool.

The main focus of high-speed machining is the increased metal removal rate and cost efficiency. The application is most suitable for products that spend a large portion of their time under a machine tool.

### 2.1.1 **Historical Perspective**

The advent of the high-speed steel (HSS) tool in 1906 allowed machinists to increase the cutting speed by three to four times. This led to new machine tools of higher power, speeds, feeds and machine tool rigidity [15]. The intent to cut material faster continues to this day, as studied here via high-speed machining.

High-speed machining has developed historically through four periods of research and development. The following is a summary of the case studies presented in [4, 13, 16, 17].

The concept of high speed machining was initiated through the work of Dr. Carl J. Salomon from 1924 to 1931 based on his German patent number 523594 [4, 13]. The contention of Dr. Salomon's study was that as the cutting speed is increased, the cutting temperature also increases until a critical maximum temperature of the workpiece material is reached. Further increases beyond this critical cutting speed produced a decrease in the cutting temperature. If the cutting speed is increased still further, the cutting temperature is reduced to the normal operating levels. The net result of this effort is an increase in the metal removal rate (MRR). Figure 1 illustrates graphically Dr. Salomon's visualization of the high-speed machining relationships for cutting speeds and temperature.

The second period of research covers the industrial interest in high-speed machining from 1958 to 1970. The most widely acknowledged work during this period is that by R.L Vaughn [4, 13, 16].

Vaughn's investigation at Lockhead was to evaluate the machinability of high strength materials between 20 000 to 240 000 sfm, and the feasibility of ultra-high-speed machining in the aerospace industry. In this study it was stated that ultra-high-speed machining was feasible, however, the area was felt to be one without application by academia and industry.

The study by Vaughn indicated that improvement limits of 50 to 1000 times, and material removal rates of 240 times as great as conventional practice could be achieved. Other studies that were conducted during this period also confirmed the material removal advantage, however, for higher strength materials rapid wear of cutting tools and vibration of machine tools were also recognized.

Studies conducted under R.I. King at Lockhead Missles and Space Company for the U.S. Navy are used to define the third phase of high-speed machining development, and covered the time period between 1970 to 1979 [4, 13, 16].

The purpose of King's study was to determine the feasibility of using high-speed machining in a production mode. It was demonstrated that it was economically feasible to introduce high-speed machining procedures into the production environment to realize major improvements in productivity.

During this phase, F.S McGee [4, 13, 16] conducted a research project at Vought regarding the machining of aluminum alloys. It was found that the cutting temperature curves tended to peak near the melting point of the aluminum alloys. From this result, it was speculated that as the cutting edge temperature approached 1 200 °F, defined as the melting point of aluminum, at 19 600 sfm the cutting edge temperature ceased to rise. The plateauing effect was seen as being indicative of an infinitely high-cutting speed feasibility for the machining of aluminum alloys.

The fourth phase of the development of high-speed machining covers the period from 1979 to the present. In 1979 the U.S. Air Force awarded a contract to the General Electric Company for the Defense Advanced Research Project Agency (DARPA) under the Advanced Machining Research Program (AMRP) to provide a scientific base for faster metal removal via high-speed machining and laser-assisted machining (LAM) and its implications [18]. The study was conducted over a four year period involving studies conducted by many academic and industrial organizations.

Some of the results obtained from this study were :

- the tool life decreases rapidly with increasing cutting speeds, except for aluminum alloys;
- wide cyclic variations in force are accompanied by chatter; chatter avoidance control is preferred over chatter suppression despite increased control complexity;
- use of a geometry that allows a tool insert to wear without impairing its ability to cut; and
- the importance of rigidity and power are of prime importance if higher material removal rates are to be attained.

Current studies indicate that enough progress has been made where high-speed motorized spindles with the necessary periphery and monitoring systems are available for high-speed machining [5]. However industrial, non-aerospace, acceptance has been slow and in some cases it is not worthwhile for builders to approach this avenue due to the cost and the need for research and development.

Application of high-speed machining processes can also provide beneficial results, as indicated by Milicron Corporation [14] and by Applied Engineering [19]. Machining gains of three to seven times can be achieved.

The future for high-speed machining lies in broadening the industrial usage by proving the machining capabilities and the system integrity and rigidity to manufactures. Fear of the high-speed operations has been felt to be one of the drawbacks limiting the industrial development [14]. Industrial research studies indicate, through vibration monitoring, high-speed machining stability, reliability and economic feasibility may aid in improving the acceptance of high-speed machining to broader industrial markets. Further research and development areas include ultra-high-speed machining using ballistic processes, laser-assisted machining for harder materials, and improvements of and to the cutting tools.

### 2.1.2 High-Speed Machining - Design Criteria

In the area of design of high-speed spindle, or machining units, standard formats for the methodology of a design for machine system components do not exist. The efficiency of the design is based on the practical experience and skills of the designer to develop a structurally stable, rigid high-speed spindle machining unit.

The complexities involved in designing a system to operate at higher cutting speeds include safeguarding the system against the effects of centrifugal forces, unbalance, misalignment effects, and excessive vibrations. By increasing the cutting speed by one magnitude requires an increase of the table feed by a compatible level, necessitating lighter inertia tables, more powerful drive motors and responsive control systems [15, 17].

The inclusion of a high-speed spindle design in a machining unit requires a knowledge of the interactions between the spindle and machine tool structure to identify vibration modes and resonant frequencies. The high frequency modes of vibration of a machine system limits the rate of material removal (MRR) [20]. It is important to understand and study the cutting tool-spindle-machine tool interaction in a high-speed machining design to identify any structural or component faults.

Although many studies and design configurations detail various design parameters for a high-speed machining unit, the challenge is to build a precise, rigid, efficient, reliable high-speed and high horsepower spindle. The design should also consider a higher product quality, balance of cutting tools and holders, and provide a chip removal and disposal system [14].

A procedure for determining the high-speed machine tool requirements for the milling of aluminum alloys is presented in [21]. Strategies for developing tooling systems for high-speed machining are presented in [22].

#### 2.1.2.1 High-Speed Machining - Design Difficulties

Control of the machining conditions of an unbalanced tool holder, or spindle assembly during metal removal creates a sufficient amount of disturbing force to the point where the entire machine is capable of vibrating, causing chatter and reduction of the spindle-bearing and tool life. Due to the effects of centrifugal forces, unbalance and misalignment, the use of large diameter spindles is in some cases impractical. An adaptive control feedback excitation-vibration sensing system is difficult and expensive to apply for some cases due to the high operating levels [5, 23].

The stability problems encountered in machine tool dynamics are due to the dynamics of the cutting process, which produce the forcing functions, rather than the shaft whirl or gyroscopic effects [24]. This indicates that the cutting tool geometry and configuration must also be considered.

The limiting criteria and design characteristics of caution for a high-speed machining system are [14] :

- cutting tools available;
- rigidity and structural stability of the tool-machine spindle-bearing assembly;
- workpiece fixture stability;
- noise and vibrations;
- safety and reliability.

Another limiting factor is that high-speed machining has not proven itself economically feasible for some harder materials [10], and is currently cost effective only for the machining of non-ferrous metals such as aluminum.

#### 2.1.2.2 High-Speed Machining - Design Conditions

Design of a high-speed machining unit must be optimized for the required speed of rotation, the dynamic stiffness, available spindle power, and the minimization of vibration levels. The main components of concern in a HSM system are the high-speed spindle, the bearings, lubrication and cooling circuits [25]. The machining tool must also be considered in the design as it will influence the stability and influence the force vibration response characteristics during cutting operations.

High-speed spindles must be machined within very precise tolerance levels and be well balanced. The spindle material should be capable of supporting very large dynamic stresses [25]. The spindle should be capable of providing adequate spindle power, ensure rigidity, damping, structural stability and reliability, and freedom from vibrations at high operating speeds [7].

The bearings chosen to support the high-speed spindle must guarantee the highest possible precision of rotation. Selection of the bearings is dependent on the thermal stability, stiffness requirements, its lubrication capability, load carrying capacity, and the required operating speed range.

Precision angular contact ball bearings with contact angles of $15^{o}$ or $25^{o}$ are most frequently used in high-speed machining systems. The ball bearings provide a lower structural stiffness in comparison to the roller bearings, as the ball bearings maintain point contacts with the raceways. The advantage of lower heat generation provided by the use of ball bearings in high-speed operations greatly influences the useful life of the bearings [12, 26, 27].

It is necessary to optimize the bearing-spindle housing design. The spindle-bearing system when assembled in the housing can have considerable effects on the preload of the bearings after mounting [26].

Lubrication or cooling of spindle-bearing assembly is essential as it effects the life, reliability and performance of a high-speed machining system. There have been many methodologies or systems for lubrication and cooling that have been proposed. The criteria for lubrication requirements generally depend on the operating speeds required.

For lower speed ranges of high-speed machining grease lubrication can be applied. In the middle operating speed ranges of DN = 1E6 mm/min (DN= bore diameter [mm] * rotational speed [RPM]) oil lubrication may be used. For the higher operating speed ranges of DN > 1E6 mm/min an oil-air mist lubrication system is recommended [25, 26]. Generally, the complexities of the cooling system increase with complexities of the machining system and the machining requirements.

The most important function of a toolholder is to provide a rigid and accurate connection to the machining unit [28]. A capable high-speed machining tooling system (holders and bodies) with greater dynamic stiffness and stability must be balanced and be very resistant to chatter [29]. In the drilling of aluminum, high speed steel (HSS) tools are preferred and are considered as a standard. The drill should possess a high helix angle ($40^{o}$) to facilitate chip removal, and a lip clearance of 3 to 5 degrees [3]. A small diameter and short length cutting tool is preferred to facilitate good rotational balance [30]. Rigidity between the workpiece and machining unit is essential to prevent drill breakage.

The support system for high-speed machining must be designed for low vibration, have quick response machine drives, high feed rate tables, rapid braking, automated loading and unloading, and have adequate power. In terms of the power necessary, in [23] it is suggested that for each horsepower (hp.) 5 $in^3$/min of material may be removed, but it is advisable to use 0.25 hp per 1 $in^3$/min. In-process monitoring and control of all conditions affecting the tool-spindle-bearing assembly should be considered as an integral part of the design process.

## 2.2 FINITE ELEMENT MODELING

The finite element method (FEM) originated in 1906, and was popularized in 1943 as a method for stress analysis. Today it is used to solve many difficult engineering design problems, including vibrations. Finite element analysis (FEA) as a numerical procedure models a structure as an assemblage of elements of the structures geometry. The analysis approximates a complex system of the model constructed of piecewise-continuous single solutions. The results of most finite element studies are rarely exact. Through refinement and model modifications errors can be reduced to the point where they are acceptable for engineering design purposes [31].

The goal for finite element modeling is to is to take a proposed model from the analytical finite element study to the end product. Currently, the process used is to test the structural integrity of the prototype, and analyze the prototype using finite element analysis or analytical routines before the final product is manufactured. In terms of machining systems, finite element analysis is used to determine and assess if the modal or frequency response and operating conditions are predictable without modal analysis or experimental testing.

The main advantage offered by a finite element model is that it allows for pretest planning and test simulation before a prototype is built. The finite element model aids in understanding the dynamic characteristics of a structure. Finite element modeling allows for relatively quick "what if" investigations. The finite element model closely resembles the actual physical structure [31, 32, 33]. It can be used to determine the natural frequencies and mode shapes of any linear, elastic structure, and it is capable of producing more DOF's or modes than the experimental modal analysis.

The foremost disadvantage of the finite element model is that it cannot be better than the physical model. The results in some cases may not necessarily reflect the physical situation. The analysis is time-consuming and may yield poor results due to uncommon shapes, and the difficultly in modeling bcundary conditions and dynamic component behaviour [31, 34]. The analysis is limited by the size of the computer available and the desired accuracy.

The majority of finite element studies and programs, for simplification, ignore the effect of damping for modal analysis [31]. This assumption is used, primarily due to the low damping ratios of most metallic materials with a damping ratio of less than two percent.

Variables used to modify the finite element model include the number of elements, modal patterns, the material properties, the number and types of degrees of freedom (DOF), and the boundary conditions [35]. Results may suffer if the mesh refinement is poor or the applicable boundary conditions are not properly identified and applied.

The boundary conditions, which influence the behaviour of the structure, are often unknown in practice. The use of proper boundary conditions requires added attention. This further complicates the selection of proper element types that best simulate the stiffness characteristics [36, 37].

The use of line or beam elements to model straight shafts, when the plane section remains plane during bending, is invalid when the spindle has abrupt changes in diameter and configuration. The local distortion increases the overall bending flexibility [38].

Finite element modeling of many structures has gained significant attention, achieving 20 000 publications in 1986 [31]. A synopsis of the literature materials available in FEA is available in [39]. The past studies indicate many methodologies and available FEA software applied in solving various vibration and structural design problems. They also indicate the difficulties and current limitations of some applications.

A heavily damped complex space structure was modeled using bar elements to model viscoelastic struts with both static and dynamic stiffness characteristics in [40]. Rod elements and springs as elastic elements were also used. Agreeable correlation between the finite element model and modal analysis was achieved in this study.

Use of symmetry condition in three-dimensional finite element analysis using isoparemetric brick and wedge elements was modeled in [41]. The primary focus in using symmetry was to reduce the number of elements used. Difficulties that were encountered in using symmetry conditions, is the application of proper boundary conditions at the nodes that define the plane of symmetry.

Brandon and Al-Sareef [24] applied the Euler-Bernouilli beam model, neglecting the effects of shear deformation and rotary inertia, to a lathe spindle. The study found it difficult to model dissipative mechanisms, which were primarily concentrated at the bearings. The modeling of various bearing stiffness showed significant shape sensitivity. The elastic properties of ball bearings and bushings are not readily available, and this makes the modeling of most bearing-support structures very difficult [34].

A lathe spindle study was conducted by Reddy and Sharan [42] to determine the influence of bearing spacing, bearing stiffness, workpiece diameter, and location of an external damper. In terms of bearing spacing, as the spacing increased the natural frequency decreased. The diameter of the workpiece greatly influenced the stiffness and the natural damped frequency. The static study indicated that there exist different stiffness requirements for static and dynamic studies. Reddy and Sharan state that a study of the spindle dynamics under each force excitation is required, and how these forces effect the design parameters (bearing stiffness) should be studied.

Modeling of the spindle-bearing-housing system is very complex. In [43] non-linear rod elements for ball race contacts were used. The effect of the housing stiffness on the ball load distribution is quite complex, and required the use of trial and error procedures. The method applied to model the structural compliance of the spindle-bearing-housing was not applicable to high-speed bearings. This was due to the variations of contact angles in the presence of centrifugal forces and gyroscopic effects.

The use of finite element analysis in modeling of a machining assembly system or structure requires an indepth knowledge of the finite element method. In addition, the influence and proper application of the support boundary conditions must be understood and applied.

## 2.3 MODAL ANALYSIS

Modal analysis is an experimental technique and tool, which is used to obtain the modal vibration parameters and identify the resonant or critical frequencies of a structure. The accepted techniques, procedures, and equipment required to obtain the modal model parameters for mass, stiffness, and damping are well documented in [44,45,46,47,48,49,50].

Modal or system analysis as a technique to find the inherent properties of a system is garnered by stimulating the system with measurable forces and studying the response. The response, as a complex ratio of the output over the stimulus is defined as the frequency response function (FRF) [48]. The frequency response function allows for analysis of the dynamic characteristics of the model, as it contains the modal frequency, stiffness and damping parameters. Curvefitting of the acquired FRFs for the entire model produces the global parameters that describe the system and can be used to create a mathematical representation of the system.

The mathematical model formulated is of the following form [44, 45, 51]:

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{0\}$$

where : [M] = global modal mass matrix, measure of mode shape inertial scaling,
[C] = global modal damping matrix,
[K] = global modal stiffness matrix,
{X} = modal displacement vector,
{0} = force vector.

The advantages offered by modal analysis or testing are : that as an experimental technique the results obtained are more realistic than many analytical techniques such as FEA; more convenient to describe the models dynamic properties by modes of vibration; model dynamic properties can be represented graphically; modifications can be easily investigated; the modal model of the test specimen is obtained directly by curvefitting the measured FRF data; and modal analysis gives a model of the structures dynamic properties at the degree of freedom where the FRFs are measured [44, 45, 47, 50, 51].

The main disadvantage of modal analysis is that it requires a physical specimen to be constructed and used, including modified designs. The analysis is sensitive to the quality of the data collected where even the incorrect technique for applying the force stimulus and measurement can invalidate the results. Modal analysis, in general, is not capable of accounting for non-linearities, and the procedure can be quite time consuming [44, 45, 46, 51, 52].

Modal analysis requires a technique to make direct measurements of mobility properties of a test structure. There are many methodologies to acquire the data. A single point of excitation with either many fixed transducers or a single transducer that is moved to the location of response interest, can be used. Another method is to have a stationary response location, and excite the structure at various points of interest [44, 45, 50].

Important elements of modal analysis technique include [44, 52]

1. the test specimen characteristics (internal components),
2. the boundary conditions imposed by the support structure,
3. location and type of exciters and transducers to correctly excite the structure and retrieve correct response,
4. correct transduction of quantities to be measured (force input and motion response), and
5. signal processing of excitation data and analysis.

The test specimen characteristics and boundary conditions applied influence the location of the transducers and excitation points. The structure should exhibit observable modes. If the internal components, and boundary conditions are not properly considered, the experimental and analytical results can suffer [37].

The physical mass, stiffness, damping properties and the applied boundary conditions of a structure influence how it vibrates. Changes in structures boundary conditions are viewed as changes in the mass, stiffness or damping of the structure, and this influences the modal parameters. If the stiffness changes, only the modal frequency is affected. If the damping changes, modal frequency and damping change. Changes in mass influence changes in modal frequency and damping [53, 54].

Generally three methods of excitation are used. These include impact testing, random excitation using a shaker, and burst random excitation [44, 45, 55].

Burst random excitation applies a force immediately after data acquisition has begun and before it stops. This allows the force and response to be zero at start and end of time window, which helps to prevent leakage.

Random excitation provides a continuous white noise signal. This results in a random force level with energy level at all frequencies in the bandwidth, however leakage tends to be a problem.

Impact testing, using an impact hammer, requires minimal setup time, is cheaper compared to the other methods, and is the preferred technique of excitation. Potential errors include non-axial forces from poor impacts and side loads. Difficulty exists in obtaining consistent results, primarily due to the challenge of obtaining consistent impacts.

It is necessary to have a good exciter and response location for the given mode and it may be necessary to evaluate many locations of excitation or response before selecting the most appropriate location [56]. The forcing and measuring instrumentation should be placed such that large values of response are measured to obtain large signal-to noise ratios [57].

The transducer plays a very important part of the measurement system, as accurate measurements must be made for both the input to the structure and its response. The transducer should interfere with the structure or specimen as little as possible, and the performance should be adequate for the frequency range required [44]. In most analysis, the transducer used for response measurements has been the accelerometer due to its sturdiness, relatively small mass, threshold levels, and useful frequency range [45, 46, 47, 49, 58, 59].

Analysis of the acquired data requires, as with ENTEKs Easy software, a procedure to identify the resonant frequencies and to develop the dynamic model of the structure. The tests points on the model are excited and the response is collected as frequency response function for each point. Using interactive fit routine the modal model is extracted from the acquired data. The data must then be curve-fitted to extract the modal shape and parameters of the structure at the critical or resonant frequencies [44,45, 50, 51].

The deflections are represented by the modal vector that gives the relative displacement strengths, allowing for an immediate interpretation of the model. This can be used to indicate design flaws when viewed in its animated form [44, 45, 50, 51].

In a study conducted [24], it was stated that the static (modal) analysis was used to generate good approximation to the first natural frequency of the spindle-bearing system. The variation of the bearing stiffness had more significant effect on latter modes. In this study it was further stated that it is more appropriate to concentrate on the first mode as it has peak displacement at the cutting zone, and chatter is commonly dominated by the response due to a single mode.

## 2.4 SYSTEM MONITORING and DIAGNOSTICS

The design of a new machining system must ensure that its machining and operating capabilities are very reliable, especially if it is to sell in a very competitive environment. Assurance of the machining system capabilities can be monitored and analyzed using system monitoring and diagnostic techniques. Analysis to ensure the systems capabilities are quickly becoming an industry standard to ensure the machining units operational life, and machining capabilities to ensure production of a quality component [60, 61, 62].

In essence there are three types or levels of monitoring techniques. These are [63] :

1. machine runs till breakdown,
2. periodic or timed inspections, and
3. regular monitoring or condition-based monitoring.

The most common properties that are monitored include thermodynamic pressure, temperature, flow rates, power consumption, torque, force, vibration and machining component speeds [64].

To properly implement a monitoring system requires early fault detection of impending failure. The ability of a monitoring system to predict developing failures would prevent serious damage and decrease downtime, operating costs and improve quality. Monitoring systems progressively monitor machine conditions with prior trends to predict the time of failure. Thus, a monitoring system, by observing certain changes in signature patterns of the designed machining system, can assist in preventing catastrophic damage to the machining unit [65, 66, 67, 68].

The most critical component of monitoring the condition of the machining unit is the sensing medium, and the analysis procedure to provide the most consistent indicators of progressing wear and failure [66]. The sensor should respond only to the particular machine or component under test, and be sufficiently sensitive to changes in the systems performance and health [66, 68, 69].

Commonly used sensors include piezoelectric accelerometers (vibrations), strain sensors, force sensors, strain gauges, power sensors and thermocouples. To identify as many characteristics of the machining unit, vibration, as shown in Table 1 [69], emerges as the best medium for analysis. The preferred sensor is the accelerometer due to its small size, easy mounting, and potential for providing a wealth of information. The location of the sensor has a significant influence on the signal transmission received, and is best located near the cutting process [62, 66, 69].

It has been shown in [66] that a single accelerometer mounted on the spindle casing of the machine provides information on the condition of the bearings, cutting tools, drive belts and electric motors, indicates tool wear, the onset of machining chatter, and unbalance of the assembly. It is recommended that the best position is directly over the bearing. The vibration of rotating equipment is transmitted through the bearings, and serves as the strongest source and indicator of impending failures or system operational difficulties [64].

The acquisition of the vibration data must be presented in a manner that observes the behaviour of machining unit. The acquired data must be displayed in a format that can be readily understood to determine the operational characteristics of the system.

The time-domain spectrum has limited value, except for simple configurations, or where other information is absent. To detect the initial fault of the system components and the roller bearings, both frequency and magnitude information of the vibration spectrum are required. Frequency spectrum analysis is the most commonly used method for observing the machining units behaviour [70, 71]

Acceleration monitored in terms of rms is the recommended measure of amplitude as the simplest monitoring procedure based on maximum amplitude. The acceleration signal can be obtained with minimal conditioning, and is often represented in terms of Engineering Units (EU) [70, 72, 73].

The frequency spectrum and acceleration amplitude provide sufficient information as to the behaviour of the machining unit and its components. Table 2 [73, 74] indicates the pattern of the operating components of a machining system and the methods used to recognize their behaviour using the frequency spectrum. Table 3 [63, 69] indicates the components that can be recognized between the low, medium and high frequency ranges. The recommended high frequency range upto 25 kHz is used for detection and indication of bearing faults. Due to the proportionality of the output to the square of the frequency, the high frequency band gives better results [66, 72, 73, 75].

Evaluation of the acquired data for most monitoring and diagnostic systems has been through pattern recognition and formulation of increased vibration trends with respect to a base data. These are then related to components of the machining unit to detect impending failure [76, 77].

Machine diagnostics are most often performed by frequency comparison, by comparing the processed data with known frequencies that are characteristic of machines design or operating parameters. The severity of the problem is judged by the increase of vibration amplitude or by the fluctuations of the waveform [73]. Evaluation or comparative parameters defined by [73] include :

- broad characteristics as rotational frequencies, harmonics and subharmonics;
- vibration, temperature gradients, or pressures initiated by an operating component or system;
- component characteristics identified with the machine type,
- natural frequency and mode shape;
- sensitivity to instablity due to wear changes in operating conditions; and
- sensitivity to vibration due to mass imbalance and other malfunction and defect excitation.

Application of a machinery monitoring system to ensure the stability and reliability of its operating functions assists in maintaining a machining unit that will offer continuous performance, conserve downtime and minimize operating costs.

# 3. EXPERIMENTAL DETAILS

A discussion of the analytical model, equipment, instrumentation and experimental setup is given in this chapter.

## 3.1 EQUIPMENT and INSTRUMENTATION

The equipment and instruments used in this study have been classified into four separate areas, these being :

1. High-speed spindle models : spindle 1 test specimen, spindle 1 Finite Element model, spindle 2, and the dual-spindle high-speed drilling machine head.

2. Equipment for Finite Element Analysis.

3. Equipment and instruments used for acquiring the data.

4. Equipment and instruments used in the data analysis.

### 3.1.1 High-Speed Spindle Models

#### 3.1.1.1 Spindle 1 - Test Specimen

A test specimen of the spindle to be used in the high-speed machining unit was constructed to evaluate the integrity of the spindle design and the locations of the bearings and supports. Figure 2 is a diagram of the spindle used in this study.

The mass representing the gear wheel has been integrated into the design of the spindle test specimen to incorporate some of the spindle assembly unit characteristics for modelling and measurement purposes. Mass of the gear wheel is incorporated to produce modal test results closer to the actual conditions, as shown in Figure 3. Figure 4 is a photograph of the spindle test specimen used in the modal study. Masses of part of the tool holder and of the drill were not allowed for, since in practice they will vary.

The spindle test specimen was manufactured of 1020 steel, whereas the working spindle was manufactured of 8620 honed and ground steel. Appendix I details the material properties of 1020 and 8620 steel.

Subsequently the test specimen is referred to as spindle 1.

3.1.1.2 Spindle 1 V-block Fixture

The support and bearing modeling characteristics were studied by use of a V-block fixture. The fixture, as can be seen in Figure 5, consists of a base plate, two V-blocks, two A-clamps and a set of bevel supports located in line with the A-clamps. The A-clamps and bevels indicate the bearing locations used to support the spindle in the machine structure. Figure 6 is a picture of spindle 1 in the V-block fixture. The clamping and bevel supports firmly hold the spindle in place at the bearing locations, and represent approximately the bearing conditions.

The V-block fixture support structure was used primarily to conserve cost for a test rig and bearing expenditures. The model is designed to give an approximate modal model and indicate the effectiveness of the location of the structural bearing supports.

3.1.1.3 Spindle 1 Finite Element Model

The finite element model of the high-speed spindle is developed using ALGORs' Finite Element Analysis software. The model is developed from a two dimensional model, by rotation of 30 and 40 degrees about the axial length of the spindle. The spindle is analyzed using three dimensional eight (8) node brick elements. Allowance of the gear mass is considered in the finite element model. Rotational development of 30 and 40 degrees facilitates application of three point, or 120 degrees support conditions as in the V-block fixture. Fixed boundary conditions in translation and rotation are applied at the representative bearing locations of the V-block fixture. Material properties of 1020 steel were used to allow for simulation and comparison to the modal spindle 1 V-block study. Figure 7 is a model of the finite element structure analyzed in this study.

3.1.1.4 Spindle 2

A second spindle, involved in the final machine design, was not used for the initial modal analysis study, because of similar dimensions and hence vibration properties. The final design of this spindle is illustrated in Figure 8, and it is subsequently referred to as spindle 2. Spindle 2 is also manufactured from 8620 honed and ground steel, and is slightly larger than spindle 1 as can be seen in Figure 9, which presents the two spindles together.

3.1.1.5 Dual-Spindle High-Speed Drilling Machine Head

The dual-spindle high-speed drilling machine head is shown in Figure 10. The machining head is supported on a solid machine base. Initially it was supported on table test stand with a heavy base plate. The machine head contains the spindles, the spindle assembly (bearings, collet, gearing mechanism, lubrication and cooling), the cutting tools, and the casing.

An initial study of the machine head supported on table base was conducted to determine the running characteristics and capabilities of the machine head prior to its final installation on the machine base. The table base study is used to define the critical measurement points for the sensing transducer. The machine base study is used to determine the improved structural free-run and drilling vibration characteristics.

The machine head is powered by two motors controlled by an Allen-Bradley Adjustable Frequency AC Motor Drive unit. A Gould Permanent Magnet Servo Motor is used to control the feed by means of a lead screw, and a Brook Compton Inc. 1.5 hp motor is used to drive the spindle units via a timing belt drive and gearing mechanism. Figure 11 illustrates the location of the bearing supports and the assembly relationship for spindles 1 and 2.

The bearings supporting the spindles are grease packed GMN BHT C TAM series. Spindle 1 is supported by the 6003/4 series, while spindle 2 is supported by the 6005/6 series. The bearings offer simple mounting procedures due to separate installation of the inner and outer ring, and the balancing of rotating components is possible with the installed inner ring. The balls are supported in a cage manufactured from polyetherether ketone thermoplastic material with carbon fibres. High deformation stability under load is obtainable with reliable dry running properties. Geometric characteristics of the bearings allows the grease lubrication to be retained within the confines of the balls and cage. The lubricated bearings are designed to be thermally stable within the operating temperatures under 150 degrees Celsius. Properties of the bearing and grease lubrication are identified in Appendix II. The machining head has also been designed to be cooled, if necessary, using cooled forced air.

For this study, spindle 2 supports the cutting tools used to machine the workpiece. Spindle 1 has been fitted with a 3/8 inch steel dowel to prevent loose chips from flying into the collet, and to model the system as if it were supporting a drill. Two HSS (High Speed Steel) drills were used in machining the workpiece, drill 1 of 3/8 inch diameter and drill 2 of 1/4 inch diameter. Both drills had a flute angle of 40 degrees, with a point angle of approximately 120 degrees. The two drill sizes are used for comparison purposes to evaluate the vibration characteristics by variation of the drill diameters.

An aluminum ingot block of approximately 18 in. * 8 in. * 3 in. served as the workpiece. Aluminum was chosen as the workpiece material for its growing popularity in the automotive and aeronautic sector, and future sales markets. In addition, the study should provide machining characteristics of drilling aluminum specimens.

The operating speed for the machine head is referenced to spindle 2. Spindle 1 operates at approximately 1.075 times the rotational speed of spindle 2. The designed operating speed for the dual-spindle high-speed drilling machine head is 167 Hz (10 000 RPM).

### 3.1.2 Equipment for Finite Element Analysis

Finite element analysis of spindle 1 were studied using a professional computer supported by the Faculty of Engineering CAD/CAM network facility. Analysis was carried out using ALGORs Finite Element Analysis software on the 386 and 486 support systems with hard disk capacities of 32 and 52 megabytes respectively. FEA studies were stored onto floppy diskettes, and hardcopies produced via a HP Laserjet II Series printer.

### 3.1.3 Equipment and Instruments for Data Acquisition

The equipment and instruments required for data acquisition include an accelerometer for vibration pickup, an impact hammer with a force transducer to excite the modal frequencies of the system, and a method of storing the desired vibration signal.

A PCB 303A02 accelerometer with a sensitivity of 10 mV/g, and a useful range 1 to 10 000 Hz was used as the vibration transducer. The transducer was mounted using an aluminum mount with an epoxy wax for the table base studies. In terms of permanent mounting for the machine base, an industrial adhesive was used instead of a stud-screw devise. This was done to avoid damage to the machine head. To excite the system a PCB 086 B01 impulse hammer with a useful range upto 10 000 Hz with a steel tip was applied. Frequency response, and frequency spectrum signals were captured using the sensing and impact mediums. The HP 35660A Dynamic Signal Analyzer was used to condition the vibration signals and to store the data onto floppy diskettes.

### 3.1.4 Equipment and Instruments for Data Analysis

The acquired vibration signals were analyzed in the frequency domain (amplitude versus frequency). Two properties of the frequency domain signal were examined. The frequency spectrum was used to evaluate the free-run and cutting force excited vibration characteristics of the system, and the frequency response was used to analyze the systems modal behaviour.

The captured vibration signals were analyzed using the dual channel HP 35660A Dynamic Signal Analyzer (DSA). It uses the Fast Fourier Transform (FFT) algorithm to convert the analog input signal from the time-domain to the frequency domain. The analyzer samples at approximately 2.56 times the highest frequency of interest, whereby the transferred data is represented by 400 sampled values (or 400 filter locations). The DSA is capable of making one channel measurements from 488 μHz to 102.4 kHz, or two channel measurements from 244 μHz to 51.2 kHz. The DSA is capable of averaging the measurements by either RMS, peak hold, linear, or exponential. Amplitude representation can be selected as EU (Engineering Units or g's), non-dimensional (mV/g / mV/g), linear or RMS volts, and dB. The analyzer is equipped with an IEEE 488 interface, on-board memory, an internal disc and it is programmable using the HPBasic language.

The modal excitation data, which were obtained from the impact tests on spindle 1 in the V-block fixture were analyzed by means of the HP 236 Series 9000 computer, with the ENTEK Easy Analysis software. ENTEK's Easy software supports many applications such as structural analysis, machinery diagnostics, and acoustic intensity. The application employed to analyze the collected frequency response data was the structural analysis portion containing EModal for the study of modal analysis of a structure. The software allows the user to construct a model of the structure or specimen, enter the measurement data, select the natural frequencies through interactive fit routines, and curvefit the acquired data to obtain the modal shapes and the modal parameters : modal mass, stiffness, damping, and the modal or natural frequency. The data acquired and the results obtained were copied using the HP 82905B printer, and the HP 7475A XY plotter via an IEEE 488 interface.

## 3.2 EXPERIMENTAL DESIGN and PREPARATION

The data collected in any study is the weakest link in the results. This section describes the experimental setup and preparation taken to ensure that reliable data is collected for the experimental studies undertaken.

To acquire the data, an accelerometer was chosen as the vibration signal sensing medium. The utilization of an accelerometer offers many advantages over the use of displacement and velocity type transducers. The sensors operate over a wide amplitude and frequency range under adverse industrial (environmental) conditions. The low mass of the PCB 303A02, of 2.3 grams, allows the accelerometer to measure the motion of most light structures without noticeably affecting the behaviour of the test specimen. The small size allows for easier installation of the transducer. The seismic type of accelerometer generates its own reference point. The accelerometer is used as the lone sensing medium in measuring vibration signals when the machining unit is in operation, both in the free-run and drilling modes.

Acquisition of the modal characteristics of the system requires an input force. The input force is applied via an impulse hammer that contains a force sensor on the striking end, and a known mass on the other. The hammer pulse produces a constant force over a broad frequency range, and excites all resonant frequencies within the specified range. The impulse at the input location creates an input accelerance signal to the structure, and the accelerometer senses the amplitude caused by the input force, and outputs the result as the output accelerance signal. The output over the input creates the transfer function which once processed and curve fitted for the entire structure details the modal parameters for the structure.

The impulse hammer and accelerometer are powered via PCB 480D06 power units that supply constant current to the sensors over the leads. The power units assist in debiasing the output signals.

Calibration of the transducers ensures that correct vibration signals are analyzed. Standard calibration procedures were used to ensure that correct sensitivity characteristics were entered into the HP 35660A DSA. The accelerometer signal is calibrated using the B&K 4294 calibration exciter that emits a 1g RMS signal at 152.9 Hz (1 000 rad/s). The hammer is calibrated by hitting a freely suspended mass with a calibrated reference accelerometer. The transfer function behaviour as a rigid body is constant, and ratioing of the force and acceleration signals produces the impulse hammer sensitivity or calibration constant. The accelerometer and impulse hammer sensitivities are stored in the respective channels of the DSA analyzer.

RMS averaging was used as the averaging technique for all of the analysis as the RMS of FRF and frequency spectrum gives a rapid increase in values when failure is imminent, and minimizes the effects of spurious noise.

The frequency span of interest in determining the critical frequencies of the structure was a window of 3 200 Hz. In some instances, such as impacts on the machine head, a frequency span of 6 400 Hz was used to obtain clearer results of the machine structure.

To study the various components of the system during free-run and drilling, four individual frequency windows were used of 200, 400, 3 200 and 25 000 Hz. Each frequency spectrum window reveals the vibration frequencies that are most significantly associated with the operation at or near the location of the perceived signal.

Analysis of the vibration signals helps to determine the sources of the characteristic frequencies of the signal, and the modes in which the sources are excited. The 200 and 400 Hz windows are primarily used to characterize the driving mechanisms such as the spindle, gear-drive mechanisms, motor, and their frequency harmonics. The 3 200 and 25 600 Hz windows are used to define vibration characteristics and conditions of the gear-drive assembly and the bearings.

### 3.2.1 **Static Study**

Static analysis refers to the study of the system(s) with no rotational motion of the spindle, or of the spindle assemblies. The main criteria of this type of study is to determine and describe the modal characteristics of the high-speed spindle and the dual-spindle high-speed drilling machine head to ensure the reliability of the bearing locations and structural supports. In each of the successive studies, to ensure a strong signal, an amplification of 100 times was applied to the signals.

#### 3.2.1.1 Spindle 1 Finite Element Model

Assurance of the finite element model was established by the element type, the mesh density, boundary condition supports, and the material correlation to the physical test specimen.

The finite element model was constructed and meshed in two dimensions using four node quadrilateral elements. The model was then developed into a three dimensional, eight node brick element by revolving the model about its axial length at predetermined angular intervals. Two models were developed, one at 30 degree revolution, and the second at 40 degree rotations. The quadrilateral and brick elements were found to be the simplest in construction and use compared with triangular and ALGORs auto generated mesh routines. The use of quadrilateral and brick elements is based on the reasonable results of studies of simple models of a cantilever beam and fixed pipe analysis. The simple models showed results within two to 30 percent of the analytical solutions.

The limiting factor for the mesh size is the free hard disk capacity of the computer system. Modification to the mesh structure and size to obtain finer mesh densities and better results was limited by this factor. Mesh densities for the 30 degree revolution was 2 004 elements, and 1 503 elements for the 40 degree revolution.

Clamping characteristics of the V-block fixture as fixed in both translation and rotation were applied at each of the bevel and A-clamp support positions. The support conditions were applied at intervals of 120 degrees representing a three point support at each bearing location.

Model and material correlation were ensured by comparison of the mass of the spindle 1 test specimen (1.8 lbs or 0.82 kg) to that produced by the finite element model. The results from the finite element correlated very well with the actual model and can be seen in Appendix III. The finite element model study is subsequently referred to as Test 1.

3.2.1.2 Spindle 1 - V-Block Fixture

Modal characteristics for the spindle 1 test specimen were analyzed using the V-block test fixture as shown in Figure 5. The spindle was supported at three locations representing the bearings of the spindle assembly. The support attributes of the bearings are provided at each location by two bevel supports along the V-block, and by an A-clamp along the top portion of the spindle. Each support is located 120 degrees apart representing a fixed three point support.

A line was drawn along the axial direction on the surface of the spindle and points were marked at 1/4 inch intervals along its length. The 1/4 inch intervals were found to be the most suitable in terms of impact spacing and for frequency response curvefitting requirements. The co-ordinates of each point along this line, as shown in Figure 12, were entered into the ENTEK modal analysis software. A PCB 303A03 accelerometer was attached to the spindle using epoxy at point 2 near the collet and tool position. To allow for clear and proper impulse hammer impacts, the reference impact line was rotated such that it was between the A-clamp and the bevel support. Figure 6 depicts the spindle-fixture setup.

Spindle 1 was firmly clamped by positioning the clamps directly over spindle bearing locations and by tightening the bolts on the A-clamps. This was to ensure that no motion of spindle 1 in the V-block would occur.

The accelerometer location at point two, near the tool holder was selected based on the clear and reliable signal transmission that was achieved over other accelerometer location points. Such as in the centre position and near the bearing locations. In addition, the behaviour of the spindle near the tool holder and cutting point is of a prime concern.

Each point was successively impacted by means of a PCB 086B01 impact hammer with a force transducer and a steel tip. Impacts were made at each point using root-mean square (RMS) averaging. Time-domain spectrum analysis was used to ensure that clean and clear impacts were made, such that no multiple impacts were present. Real, imaginary, phase, and coherence components of the obtained frequency response function values were analyzed for their correctness to ensure that reliable data was obtained. The FRF data were recorded using the dual-channel HP 35660A Dynamic Signal Analyzer (DSA).

The frequency response spectra obtained from the impact test were then transferred to the HP 236 computer for analysis via the ENTEK modal analysis software. To allow the data to be transferred correctly, coherence and frequency response outputs for each impact location must be observed on the DSA front panel (screen, window) simultaneously in their respective window panel. The experimental setup is shown in Figure 13. The software program stores the acquired data corresponding to the impact location or point in a storage block for the test specimen. Resonant frequencies are identified using interactive fit routines, and once these frequencies are defined, the frequency response data can be curve-fitted to acquire the mode shapes and the modal parameters. The modal analysis study of spindle 1 supported in the V-block fixture is subsequently referred to as Test 2.

#### 3.2.1.3 Dual-Spindle High-Speed Drilling Machine Head

The modal characteristics of the Dual-spindle High-speed Drilling Machine Head were determined by impact tests with four predetermined accelerometer locations on the housing of the spindle assembly casing. Because the primary sources of vibration were the spindles, their selection was determined by the expected locations of greatest vibration response of the housing. Figures 14 and 15 illustrate the machine head spindle unit with the accelerometer locations. The accelerometer mounts, an example of which is shown in Figure 16, were attached to the machine with an industrial adhesive.

Mounting of the transducer with an industrial adhesive was used as opposed to the a threaded stud , so as not to damage the surface of the machine head.

Locations 1, 3 and 5 are directly over the bearing positions. Vibration signal measurement measured directly over the bearing housings provide the best indicator of the machining units component characteristics. Location 2, directly on the top of the machine head is used to gauge the vibration transmission of the machining components of the structure, in terms of structural stability. Location 2 is not directly over any major operating component of the machine head.

Using the PCB 086B01 impulse hammer, impacts were made near the accelerometer, at the collet, dowel, and near the end of the drill. The acquired frequency response measurements were recorded on the HP 35660A DSA. The approach was repeated for all four locations of interest for drill 1, drill 2, and the dowel. First with the drill freely hanging and then in contact with the workpiece to simulate drilling conditions. RMS averaging was used, as it is difficult to obtain clear and consistent impacts on the surface of the drill, collet and machine head.

Different drill sizes are used to compare the influence and change of drill diameter to the modal characteristics of the machining unit. Changes of modal frequency response and of the critical frequencies may affect the operating requirements of the machine head.

The frequency response study of the machine head with drill 1 is subsequently referred to as Test 3. Similarly, Test 4 refers to the frequency response study with drill 2, and Test 5 refers to the frequency response with drill 2 with respect to spindle 1. Spindle 1 in Tests 3, 4 and 5 supports the dowel.

The modal analysis data for Test 3 and 4 was collected after the machine had been used in operation during free-run and cutting. Test 5 was conducted during a non-operation period. The influence of cold and warm tests can influence the stiffness characteristics of the assembly, just as the stiffness values of rotational equipment can change with increased operating speeds and loads.

### 3.2.2 **Free-run Study**

Analysis of the vibration signal during machine operation reveals frequencies that are most significantly associated with components of the system. This assists in determining the sources of the characteristic frequencies of the signal.

Frequency spectrum signals of four windows, with ranges of 200, 400, 3 200, and 25 600 Hz respectively, were analyzed. The machine was allowed to initially run freely prior to vibration measurements being taken. Assurance of clear and clean signal from the accelerometer were taken by affixing the transducer directly over the bearing housing and at the transducer position being studied. The cables were ensured against excessive looseness and motion by taping the cable to the machine surface. Root-mean square averaging was used to minimize the amount of spurious noise in the collected data. To eliminate random components, the number of averages was varied for each measurement in the free-run trials until the measurement fluctuations stabilized.

#### 3.2.2.1 Table Base Study

The table base study was primarily used to assure that the machine head design operated correctly. This study was also used to minimize the critical vibration measurement locations when the machine head is supported on the machine base for monitoring purposes.

Initially thirteen locations were studied, as shown in Figure 17. The machine head configuration and shape dictates some of the appropriate locations for the vibration sensor mounting. From initial analysis and prior studies [60, 66, 72, 76], four critical accelerometer mounting positions were identified, as in Figure 14 and 15, and are used in the remainder of this study.

Free-run characteristics of the machine head were monitored at the desired operating speed of 10 000 RPM. Run-up and run-down studies were monitored between 8 000 to 16 000 RPM. The corresponding characteristics were measured at location 5. Thermal characteristics were monitored during the table base study by use of thermocouples near the bearing locations, shown as channel locations in Figure 14. The thermal study is used to indicate whether the grease lubrication is sufficient, and whether forced cool air will be necessary.

Prior to the initial measurement taken, the machine head had been in continuous free-run operation for over 200 hours. The table base study was conducted in a room isolated from the factory environment. Load applicators were attached to the collet, but were not used in this study.

#### 3.2.2.2 Machine Base Study

Assurance of the free-run analysis of the machine head supported on the machine base were similar to those of the table base study. The machine head did not support any load actuators, but instead supported the cutting tool in spindle 2, and a dowel in spindle 1. The accelerometer mounts in this study were fixed at the four predetermined measurement locations by an industrial adhesive.

The free-run study on the machine base was monitored at the operating speed of 10 000 RPM. Run-up analysis was monitored from 8 000 to 13 000 RPM. Thermal characteristics were also measured during the run-up analysis at the five thermocouple channel locations. The machine base analysis is used to compare the influence of structural support with other structures on the spindle system.

### 3.2.3 **Drilling Study**

Drilling characteristics of the dual-spindle high-speed drilling machine head were measured at the four reference accelerometer mount locations, as shown in Figure 14 and 15 of the machine head. The operating speed for drill 1 was 9 960 RPM, and 10 530 RPM for drill 2. In the drilling study, four frequency windows were used, 200, 400, 3 200, and 25 600 Hz windows.

The aluminum ingot, or the workpiece, was securely fastened to a fixture that allowed it to be moved such that parallel holes could be drilled side by side. The transducer cables were secured to the system to prevent excessive electrical noise in the output.

To ensure proper collection of data for the 200 Hz window, a feed rate of 60 ipm (inches per minute), and hole depth of 1 inch were used. This produces a cutting time of approximately two seconds between the cutting tool engagement and retraction from the workpiece. These parameters allow the DSA to capture and process the data at the lower frequency window, which requires a longer data capturing sequence.

No coolant was used during the cutting process. However, WD40 oil was used to treat the cutting tool at approximately every four to six drilled holes.

## 3.3 EXPERIMENTAL DESCRIPTORS

Descriptors used for the data acquired are selected to ensure that the data is in a format that will properly describe the system. The descriptors or parameters should provide the necessary information to correctly analyze the system.

The descriptors used to detail and analyze the acquired data are described in this section.

### 3.3.1 Coherence

Coherence measures the power in the response channel caused by the power via the input channel in a dual-channel DSA. The unitless coherence value ranges from 0 to 1. A coherence value of 1 indicates that all of the output power at that frequency is caused by the input or source. A coherence value of 0 indicates that none of the output power is caused by the input. The coherence output is obtained only from a series of averaged frequency response measurements.

Poor coherence results are in most cases attributed to leakage errors, poor signal to noise ratios, non-linearities of the system, and extraneous noise. The coherence descriptor should be used with great care, as it is a tool, and not necessarily the answer to the quality of the data collected.

### 3.3.2 Engineering Units (EU)

Engineering units correspond to the measurement of acceleration, in this case with units of "g". The user adjusts the gain of each channel of the analyzer such that the display corresponds to the physical parameter that the transducer is measuring. The DSA performs the conversion and displays the vibration spectrum in the desired engineering units.

### 3.3.3 Frequency Response

The frequency response data contains the amplitude and phase (or real and imaginary) response characteristics, which are generated by exciting the system with a force impulse. The frequency response data contains distinctive peaks, when the system is able to move more freely when subjected to input energies at certain frequencies than at other frequencies. Each of the resonant peaks corresponds to a mode of vibration of the structure. Frequency response is used to measure the modal properties of a structure.

### 3.3.4 **Frequency Spectrum**

The frequency spectrum shows the vibration components of the machine as the machine passes through specific operating ranges. The spectrum displays the energy of each frequency component at the sampled point. Each frequency component appears as a vertical line, with the height representing its amplitude.

The frequency spectrum is used to measure the vibration and operating characteristics of the machine. The absolute level is not always the prime concern, but how the amplitude and signal varies with time. Small vibrations from all other components can also be monitored by the frequency spectrum.

### 3.3.5 **Real and Imaginary Components**

The real and imaginary components are contained within the information for the frequency response characteristics. The components are generally analyzed using Nyquist plots, or during curve-fitting routines in the ENTEK Easy modal analysis software. A resonant frequency is indicated by a zero value of the real component and a peak value of the imaginary component. These components are used to assure that the resonant or critical frequency chosen is valid.

### 3.3.6 RMS Averaging

Use of rms averaging improves the estimate of the mean level of the vibration components. RMS averaging is primarily used when the component levels vary significantly, producing variance of the signal levels. Noise can also cause the spectral components to vary widely in amplitude. Amplitude averaging takes place only, as the phase component is ignored. Averaging improves the statistical accuracy, and ensures more accurate estimates of the total signal and noise levels. Reduction of random and transient components.

### 3.3.7 Windowing

Windowing is a time-domain weighting function applied to the input signal. It is used to filter out non-periodic signals, reduce leakage, and prevent signal overload. Windowing is also used to attenuate certain parts of the time record. In the modal analysis study a force (impact transducer) and exponential (accelerometer) windowing routines were used.

# 4. EXPERIMENTAL PROCEDURE

The process used to acquire the data is detailed in the step by step procedure outlined in the following sections of this chapter.

## 4.1 STATIC STUDY

### 4.1.1 Finite Element Analysis of Spindle 1

Modal characteristics of spindle 1 using FEA were collected using ALGORs FEA software with the following procedure :

1. Develop the spindle model using ALGORs Superdraw II sub-menu.
2. Transfer the model to the Decoder sub-menu and enter the spindle material properties and analysis requirements.
3. Process the model design to check finite element development integrity.
4. Using the modal analysis processor collect the modal frequency and displacement output.
5. View the modal displacement and resonant or critical frequency output of the model using the Superview submenu.
6. Produce hardcopies of the spindle displacement, and frequency output.

### 4.1.2 **Spindle 1 V-block Fixture**

The procedure used to collect the frequency response data from the spindle supported in a V-block structure consisted of the following steps :

1. Set the DSA to accept the frequency response data.
2. Set the ENTEK software to accept frequency response data from the DSA.
3. Position the spindle in the V-block fixture.
4. Position the impact position line between the A-clamp and the bevel supports.
5. Securely clamp the spindle in place.
6. Attach the accelerometer to impact point 2 on the spindle.
7. Impact point of interest with the impulse hammer approximately five to ten times using rms averaging.
8. Ensure the correctness of frequency response data.
9. Store the data to floppy disc on the DSA, transfer data to the ENTEK software, and store data in program block.
10. Repeat steps 7 to 9 for each of the impact points.
11. Identify the resonant frequencies using the interactive fit routines from ENTEK Easy software.
12. Curve-fit the data to obtain the modal parameters.
13. Produce hard copies of the frequency response data, the modal parameters and the modal displacement shapes.

### 4.1.3 **Dual-Spindle High-Speed Drilling Machine Head**

The procedure used to obtain the frequency response data for the machine head comprised of the following steps :

1. Set the DSA to accept frequency response data.
2. Attach the accelerometer to location 1.
3. Impact at the chisel end of drill 1, the collet of spindle 2, the dowel, the collet of spindle 1, and the machine head near the accelerometer mount.
4. Impacts made in line with direction of accelerometer (transverse).
5. RMS averaging for two to five impacts at each location.
6. Check the data for correctness.
7. Store the frequency response data onto floppy discs.
8. Repeat steps 3 to 6 for locations 2, 3 and 5.
9. Repeat steps 2 to 7 for drill 2, and the dowel.
10. Produce hardcopies of the frequency response data.

## 4.2 FREE-RUN STUDY

The procedure for measurement of the machine head freely running applies to both the table base and machine base studies. The procedure used is as follows :

1. Set the DSA to accept frequency spectrum data for the corresponding frequency window of interest.
2. Attach the accelerometer at the measurement location of interest.
3. Record the vibration signal using rms averaging.
4. Store the measurement onto floppy discs.
5. Repeat steps 3 to 4 for each frequency window of interest.
6. Record the thermal temperatures of bearings (lubrication) at the five specified locations.
7. Repeat steps 1 to 6 for each measurement location of interest.
8. Produce hardcopies of free-run frequency spectrum data.

## 4.3 DRILLING STUDY

The procedure used for acquiring the vibration signal during cutting was as follows:

1. Set the DSA to accept frequency spectrum data for the corresponding frequency window.

2. Attach the accelerometer at the measurement location of interest.

3. Record the vibration spectrum from the instant the cutting tool engages the workpiece, and until it is disengaged after the hole has been drilled.

4. Record the measurement onto floppy disc.

5. Adjust the workpiece for the next hole to be drilled.

6. Repeat steps 3 to 5 for each frequency window of interest.

7. Apply WD40 to cutting tool.

8. Repeat steps 2 to 8 for all measurement locations of interest.

9. Produce hardcopies of the drilling frequency spectrum data.

# 5. DISCUSSION OF RESULTS

Analysis of the Dual-Spindle High-Speed Drilling Machine Head was conducted by testing of the stability and adequacy of the bearing position for spindle 1 through modal analysis via a finite element model, and the spindle test specimen supported in a V-block structure. The machine head was analyzed for its frequency response, free-run, and drilling characteristics, with variation of the cutting tool size.

The finite element model was used to determine if it can be used as a tool to proceed from a design analysis to the final product, disregarding the prototype model development and analysis requirements.

The maximum or critical operating limit of the machining unit was determined by means of frequency response and modal analysis. The free-run and drilling analysis was used to determine if the machining unit is capable of machining at the required operating speed of 167 Hz or 10 000 RPM.

## 5.1 STATIC ANALYSIS

Modal analysis and frequency response measurements are used to determine natural frequencies, which indicate the potential of generating excessive vibrations, such as for example metal cutting chatter, that may affect the operation of the spindle and machinery components. Modal analysis via curvefitting of the FRF data is used to determine where the greatest displacement may occur of a system operating at the critical or natural frequency.

The results obtained by modal analysis via a finite element model, and by the impact method on spindle 1, for the first two natural frequencies, are presented in Table 4 for Tests 1 and 2. Table 5 presents the results for the first natural frequency of the frequency response study of the machine head from Tests 3, 4, and 5.

The results presented in Table 4 indicate a very large difference between the finite element model and spindle 1 supported in the V-block structure. This suggests that the rigid support assumption applied to the finite element model based on the support structure of the fixture is incorrect.

Differences between the frequency response of Test 3, 4 and 5 exist, as well as for the tests conducted for spindle 1 in Tests 1 and 2. Varying the drill size changes the frequency response of the system. This suggests that the drill size can influence the operating speed limit that can be used in a machine.

Frequency response Test 5 was conducted with no prior warm-up period, supporting drill 2, appears to differ in the frequency response characteristics. The lack of a warm-up period may affect the systems inherent stiffness characteristics because of incomplete expansion of components. An analogy that can be used as a comparison is the human body when it is forced to awaken from a blissful sleep, often cranky and stiff.

### 5.1.1. Spindle 1 Finite Element Model Analysis

The finite element frequency modes considered in this study are those occurring in transverse bending. The finite element software is capable of solving all other modes, such as torsional, longitudinal, and flaring. The primary concern is to model the spindle as it is supported in the V-block fixture.

The first natural frequency that is solved by ALGORs finite element package is in the bending mode. A frequency of 1 597 Hz, as shown in Table 4, was obtained for the given rigid boundary condition assumption at the bearing locations. Figure 18 represents an exaggerated displacement output of the finite element model at the first modal frequency. As can be seen in Figure 18, the spindle has the greatest displacement near the gear-mass. The remainder of the spindle between the bearing boundary supports and near the tool holder portion of the spindle model has relatively very little displacement.

The second transverse bending frequency was obtained at 4 021 Hz. The modal displacement of the spindle model is shown in Figure 19. The spindle in this instance has a predominant displacement at the tool holder portion of the spindle.

From Figures 18 and 19, the spindle behaviour suggest two separate behaviours at the two modes, with the unsupported overhangs taking up most of the dynamic energy.

When these results are compared to the modal analysis study of spindle 1, significant differences exist in respect of the frequency magnitudes and deflection shapes.

Sources of error that are possible include poor boundary condition assumption, the element type may not support the required flexibility characteristics, and the existence of non-linearities in the spindle, which are not normally solved for in modal analysis. The most likely of these is the boundary condition application, as in most instances it is the most difficult to fully understand and model.

The finite element model under the assumed boundary conditions was unable, in this study, to substantiate the frequency and modal displacement results obtained from the spindle supported in the V-block structure. Because the procedure to properly define the boundary conditions at the supports and bearings was shown by a concurrent study to be quite involved and complex, the finite element study was concluded at this stage. It is proposed to carry this study forward as a separate research project.

### 5.1.2. Spindle 1 - V-block Fixture

The test spindle was supported in a V-block fixture at three points corresponding to the locations of the actual bearings via two bevels and an A-clamp per bearing location. This type of clamping procedure does not directly exhibit boundary or stiffness characteristics inherent to bearings, but was used as an approximation. The clamps ensured that the spindle was fixed, but not completely rigid while the bearing and support network in the machine head allow for some play and less stiffness in the actual spindle assembly. With less stiffness and rigidity of mounting, the spindles in the machine head will have lower fundamental natural frequencies when compared to the clamped conditions of the spindle-fixture assembly, as shown in Tables 4 and 5.

Impact analysis along the spindle in the frequency window of 0 to 3 200 Hz, provided two distinct peaks or modal frequencies. An example of the frequency response output is shown in Figure 20, which is taken at impact location 33 near the gear-mass allowance portion of the spindle. The amplitude levels of the two distinct peaks indicate that both frequencies influence this position of the spindle. The amplitude levels of the modal frequencies vary across impact locations of the spindle. A very low amplitude or zero value is obtained for the corresponding frequency near a nodal position.

Validation of the peaks as natural frequencies was analyzed by using the real (zero) and imaginary (maximum or minimum) components of the frequency response as shown in Figure 21. Interactive fits of the two modal frequencies, 921 and 1 535 Hz, were used to curvefit the displacement characteristics of the spindle. The single degree of freedom (SDOF) curvefitting routine provided by the ENTEK Easy software was used.

An outline of the undeformed spindle model entered along its impact coordinates is shown in Figure 22. Displacement at the first modal frequency of 921 Hz is shown in Figure 23. The vectors shown indicate the impact location, the direction of displacement, and by their length, the relative amplitude of displacement. As can be seen from this figure, the largest amount of deformation occurs near the gear mass drivetrain section of the spindle. The deflection occurring near the gear-mass end is most likely due to the single bearing support, and the additional allowance for the mass of the gear. The mass of the gear system greatly influences the obtained frequency component through its increased mass inertial characteristics. A lower gear mass may increase the value of the first modal frequency, whereas a larger gear mass may lower the first modal frequency. The frequency value may also be increased by increasing the stiffness of the spindle bearing supports.

It is of interest to note that since some movement is indicated at the clamping locations, their stiffness is not high enough for a rigid clamping assumption. Deflections between the bearing supports and the tool holder end exist, whereas, these deflections were not obtained in the finite element model. The significant differences in the deflection shapes from FEA and modal tests are primarily caused by the flexibility of the supports in the latter case, and confirm further the need for proper conditions with the FEA.

The second modal displacement at 1 535 Hz is shown in Figure 24. The most significant deflection occurs near the tool-holder end of the spindle. Deflections are visible near the bearing supports and the gear-mass section of the spindle, as was the case for the first mode. The finite element model failed to substantiate the modal results in terms of shape and frequency also for the second modal frequency for the same reasons.

There exists little correlation between the spindle supported in the V-block fixture and the finite element model of the spindle, in terms of modal frequencies and mode shapes. The major cause for this difference is the significant difference in the rigidity of the supports.

The results indicate that the V-block supports are not rigid. Because the ball bearings are likely to be more flexible, an increased divergence from the FEA results is expected. Analysis of the spindle system with actual bearings support would have given a better indication, however, measurements under such conditions were not practically possible.

The results obtained from the modal analysis of the spindle test specimen indicate that the first natural (critical) frequency of 921 Hz is beyond the operating speed of 167 Hz (10 000 RPM), indicating that the bearing locations and the spindle design do not present significant problems.

### 5.1.3 **Dual-Spindle High-Speed Drilling Machine Head**

Frequency response analysis of the machine head was used to determine the behaviour of the machining unit, and the spindles with the actual bearing supports. These results were used for comparison with those obtained for spindle 1 supported in the V-block structure. The study was conducted with two drills of different sizes, with the drill freely hanging, and in contact with the workpiece. The drill in contact with the workpiece was used to simulate the system under drilling conditions.

In Tests 3 and 4 measurements were carried out for the two drills after the system was heated up by previous operations. Test 5 was conducted with drill 2 prior to any machining operation. Results from these studies, however are indicative of the machining unit rather than of individual components, and are used to determine the operating capability of the machining unit.

The frequency response and modal analysis results are used for comparison of the free-run and drilling frequency spectra to determine if any of the components may be influenced by any of the critical frequencies.

Each component of the machine head is related to the property of its adjacent component, such as the drill, collet, spindle, bearings, gear drivetrain, and housing assembly. The measured acceleration frequency response and spectrum are functions of all of the component properties. The transmitted or excited vibration influences the dynamic properties of the main components and their own resonant frequencies.

The configuration of the machine head and the location of the four predetermined accelerometer mount locations influences the transmission of the signal received, predominantly in respect of magnitude strength. Transmissibility characteristics of impacts made from various positions can cause some of the internal components not to have been excited. Impacts made near the accelerometer location, collet, on the drill and dowel may excite other modes of vibration (torsional, axial) that may influence the acquired results. It proved difficult to acquire clear, repetitive modal FRF data due to the difficulty of impacting on the collet and drill.

The results obtained from the frequency study of the machine head for the first mode are presented in Table 5. Variation between drill size, when the data is collected prior to and after operation, the location of impact, the transducer location, the housing support and design, and the signal transmission path appear to influence the acquired frequency response results.

Locations 2 and 5 provided the most consistent and reliable frequency response signals. The results from locations 2 and 5 are presented for Tests 3 and 4. To relate the relationships for spindle 1 with spindle 2 and the machining unit, the frequency response results obtained from locations 3 and 5 are presented in Test 5. Location 3 and 5 are located directly over the first bearing housing of spindle 1 and 2, respectively.

The traces presented in the following figures in this section in blue represent the impacts made when the drill is not in contact with the workpiece. The red trace indicates the response observed when the drill is in contact with the workpiece.

5.1.3.1 Test 3 Drill 1

Frequency response results for the machine head, with drill 1, are presented in Figures 25 to 30. Figures 25 and 26 represent the impacts made near the accelerometer mount locations. Figures 27 and 28 represent the impacts made on the collet of spindle 2. Figures 29 and 30 represent the results obtained with impacts on the drill when it is free from the workpiece. The purpose of showing results for the impact and accelerometer locations is to illustrate their influence on the vibration response for the machining unit.

Impacts made near the respective accelerometer locations on the housing indicate that frequency responses are dependent on the surrounding housing support characteristics. A frequency window of 3 200 Hz is used to show the responses for Figures 25 and 26. At these locations, peaks corresponding to a natural frequency of approximately 712 Hz were obtained, when the drill is in contact with the workpiece and freely hanging. The amplitude levels are fairly low in this region.

There is a slight shift in the frequency and amplitude levels when the drill is free and when it is in contact with the workpiece. Small amplitude level variations exist with the drill free experiencing slightly higher amplitude levels, indicating some damping effects when the drill is in contact with the workpiece.

Impacts made at the collet of spindle 2 produced some spurious results with amplitude and frequency response modulations, as shown in Figures 27 and 28. The drill free and in contact with the workpiece traces, respectively have similar frequency response, but with significant amplitude changes. The amplitude of the free drill, as indicated by the blue trace, is higher indicating the presence of effective damping when in contact with the workpiece. The causes of the sideband type of peaks following the 712 Hz peak are not obvious, but are likely to be caused by the ball-race contact properties of the bearings.

Frequency response spectra, taken at locations 2 and 5, from impacts made on the drill when it is free, are presented in Figures 29 and 30. A conclusive trend is not evident. The difficulty in correctly impacting the hammer tip against the end of the drill bit may cause some fluctuations in amplitude and frequency values, as well as exciting localized resonances. The transmission of the impact signal to various transducer location may also affect the overall results obtained.

Analysis of the results obtained with the machine head indicates that many factors can affect the quality and nature of the measurements, such as: the design and configuration of the machine head, the support structure for the machining unit, the component interactions, such as the drill-collet-spindle-bearing assembly, the location of the transducer or signal measuring device, and the boundary conditions of the system.

The design, configuration, and support structure of a system influences the stability, mass distribution, and stiffness of the unit that can influence the frequency response traits of the machining system. Component interaction influences the boundary conditions, stiffness, mass inertia, and damping that affect the value and strength of the transmitted signal to the transducer. However, the combination of the component interactions gives the frequency response characteristics of the system that can be used to set operating parameters and limits for the machining unit.

The boundary conditions influence the frequency characteristics such as the modes of vibration. Unless the boundary conditions are well defined and modeled, the results obtained may not be very useful, such as in the case of bearings. Fluctuation of boundary conditions occur when the drill is free, and in contact with the workpiece. The study with drill 1 showed the influence of such fluctuations.

The lowest natural frequency of the spindle is 712 Hz which is lower than that obtained with V-block fixture. The results clearly indicate that the stiffness of the ball bearings is significantly lower than that of the clamping arrangement.

#### 5.1.3.2 Test 4 Drill 2

Similarities with results from Test 3 exist between trace shapes with fluctuations in frequency and amplitude. There appears to be a consistent increase of the fundamental frequency of the spindle from Test 3, indicating the influence of the lower mass of drill 2 (3.4 times less).

Figure 31 presents the frequency response obtained with the impact made near location 2. The trace indicates a frequency corresponding to approximately 772 Hz, when the drill is free and in contact with the workpiece. The trace also indicates that the frequency response amplitude level is lower with the drill in contact with the workpiece.

Figure 32 presents the frequency response obtained with impact near location 5. The trace in the frequency range shown indicates that a distinct frequency response occurs at approximately 4 148 Hz. This may be a higher mode of the spindle or a localized resonance. Amplitude levels are higher at this location, predominantly because of frequency multiplication. This point can be shown such that, if the frequency value was increased by five times (4 148 / 772 ≈ 5), the acceleration amplitude should be 25 times higher ($5^2 = 25$) than at 772 Hz. On the other hand, the less distinct peak at 772 Hz is partly due to a doubling of the bandwidth of the spectrum analyzer at the higher frequency window. This results in a greater spread of energy and thus a reduction in height and sharpness of that peak.

Frequency response results for impacts made on the collet of spindle 2 are presented in Figures 33 and 34. Both locations appear to exhibit similar responses. The first frequency for both the drill free and in contact with the workpiece is 772 Hz. The amplitude decreases by almost four times, when in contact, indicating that the drill is damped by friction or impacts. Beyond 1 200 Hz, the frequency response level for the drill in contact with the workpiece is higher, significantly so at 1 680 Hz and may represent a resonance of another part of the system. The appearance again of side bands on the side of the fundamental frequency can be only explained by possible changes in the clearances between the balls and races in the bearings at larger amplitudes of vibration. This further emphasizes the complexity of boundary conditions resulting from ball bearing supports.

The frequency responses acquired due to impact at the end of drill 2 are shown in Figures 35 and 36. The first frequency from the two traces do not correspond, 932 Hz for location 2 and 1 060 Hz at location 5. From the trace in Figure 35 for location 2, two distinct peaks are visible. The second peak is a multiple of the first peak. Whereas, in Figure 36, the second distinguishable peak occurs at more than three times the first. Variations of this sort can be attributed to difficulty in impacting the end of the drill, as the hammer tip was wider than the drill diameter. This produces forces in other directions and, together with clearance effects of the ball bearings, excites other resonances with their harmonics.

Frequency response analysis of the machine head with drill 2 indicates that the lowest critical frequency occurs at approximately 772 Hz. This frequency value does not appear to cause any difficulty for the system to operate at the required operating speed. The obtained value does not correspond to the value obtained by modal analysis of spindle 1 by finite element or support in the V-block fixture.

#### 5.1.3.3 Test 5 Dowel

Spindle 2, which supports the cutting tool, was of prime concern in Tests 3 and 4, as it contacts the workpiece during the drilling operation in this study. The focus for this test was to compare and determine the operating characteristics for spindle 1 in relation to and with spindle 2 in the machining unit. Frequency values were obtained equivalent to Test 4 for impacts made on the spindle 2 system.

Frequency response outputs for impacts measured at locations 5 and 3 are presented in Figures 37 to 40 and 41 to 43, respectively. The impact on the collet of spindle 2, as shown in Figure 37, has similar details to the results obtained in Test 4, with the first critical frequency at 788 Hz. The amplitude level is slightly lower when the drill is in contact with the workpiece, as was the trend in the previous test.

Impact at the collet of spindle 1, as shown in Figure 38, indicates a distinct frequency value for the drill free of 884 Hz, and 932 Hz with the drill in contact with the workpiece. The contact amplitude trace level is higher only at this peak value, with respect to the amplitude levels for the drill free. A clear explanation of this is not obvious.

Impacts made on the dowel of spindle 1 produced different results, as shown in Figure 39. Three distinct peaks with the free drill including multiples of the first fundamental frequency at 908 Hz. With the drill in spindle 2 in contact with the workpiece, measurable differences in both frequency and amplitude are observed. The fluctuations due to impacts on the dowel and collet of spindle 1 suggest that the interaction of the components influences the vibration response of both spindles.

Impact on the collet of spindle 2 produces a first frequency at approximately 812 Hz with the drill free and when in contact with the workpiece, as shown in Figure 40. Beyond 1 000 Hz, spindle 1 attains higher amplitude values, with three distinguishable peaks occurring beyond the first frequency. These peaks did not exist with the drill free, suggesting the influence of other components on spindle 1, when the drill is in contact with workpiece. Changes in boundary conditions of spindle 2 are significant enough to alter the frequency response of spindle 1.

From impact on the collet of spindle 1 in Figure 41, a similar response as with Figure 40 is achieved. However, only two distinct peaks beyond the first frequency of 796 Hz are obtained.

Figure 42 presents the frequency response obtained with impacts on the dowel. Three distinct, multiple frequencies are obtained, with the first fundamental frequency for the drill free at 900 Hz, and with spindle 2 in contact with the workpiece at 884 Hz.

Changes in frequency and amplitude response are affected by the boundary conditions, the transducer and impact location, and the interrelation of the individual components (spindle 2, gear drivetrain, and spindle 1) imposed on spindle 2. Impacts on the dowel, in comparison with those on the drill tip, produced generally cleaner results. This supports the assumption of the more complex force distribution in the latter case. Although, some of the variation in the fundamental frequency is caused by the differing conditions, some allowance must also be made for the effect of the wide bandwidth.

The frequency response level corresponding to spindle 1, as studied in this section corresponds to approximately 812 Hz. The fundamental frequency value obtained for spindle 1, indicates that no problems can be expected when operating the machine head at the desired speed of 167 Hz or 10 000 RPM.

### 5.2 FREE-RUN ANALYSIS

Free-run analysis of the machine head was studied using the table base and machine base support structures. The analysis was used to characterize the running behaviour of the machine head to determine its capabilities of operation at the desired operating speeds. Modal and frequency response results are compared to determine if any critical frequencies are excited.

#### 5.2.1 Table Base

The table base study was conducted with the machine head supported on steel table stand with a heavy base. Load actuators to simulate drilling conditions were configured to the machine head, but were not active for this portion of the study. Analysis of the table base study was used to determine the critical location for vibration measurements for subsequent studies.

Initially, thirteen locations were used to determine the stability of machine head and its support structure, as shown in Figure 17. Four critical points, as in Figures 14 and 15, were selected as the reference locations for subsequent studies.

Examples of the free-run output at the operating speed of 10 000 RPM or 167 Hz, are presented in Figures 43 to 46 for the four frequency windows. In general, the amplitude levels are fairly low.

Figure 43 presents the results in the 200 Hz window which displays the amplitude level of spindle 2 at 167 Hz (10 000 RPM), spindle 1 at 183 Hz (10 750 RPM). The smaller visible peaks represent at 60 and 120 Hz the first and second harmonic operating frequency of the motor, and at 83 Hz the frequency of the timing belt gear drivetrain. The level of the motor and gear drivetrain is fairly low and does not appear to affect the running characteristics of the spindles.

With the 400 Hz window, as shown in Figure 44, a harmonic of the operating frequency appears at 354 Hz, but its value is relatively small. If it was significantly higher, misalignment or looseness of some components may have been indicated. Figures 45 and 46 do not provide any indication of faults in gears or rolling elements of bearings.

The use of the 25 600 Hz window should be used with some precaution as the accelerometer linear range is of 10 000 Hz. Effects of non-linearity may effect the results in the higher range. In addition, it should be noted that as the acceleration amplitude is a function of the square of the angular velocity, if the operating speed is increased 10 times, its level may increase respectively by 100 times for the same magnitude vibration amplitude. Thus in Figures 45 and 46 no indication of significant problems is evident.

To evaluate the operating characteristics of the machining system, a run-up and run-down test were carried out from 8 000 to 16 000 RPM (133 to 267 Hz). The run-down study was measured from 16 000 to 8 000 RPM (167 to 83 Hz). The results for the amplitude levels recorded at location 5 for the spindles 1 and 2 at their respective operating speeds are summarized in Table 6, and are plotted in Figures 47 and 48.

Concurrently, with the run-up and run-down study, thermal readings of the lubrication used in the bearings were measured. Results of the thermal readings are presented in Figures 49 and 50, measured at the five thermocouple channel locations, as shown in Figure 14.

The run-up and run-down vibration amplitude levels share a similar trace patterns, where the maximum level appears at 15 000 RPM for spindle 2 and 14 000 RPM for spindle 1. A minimum amplitude level occurs at 11 000 RPM, with a sharp rise again at 13 000 RPM. The amplitude at 10 000 RPM is relatively small and no significant problems were foreseen during the run-up and rundown study, even at speeds of 16 000 RPM.

Thermal stability of the bearings is critical, as poorly lubricated bearings can easily lead to failure of the machine head. The bearings are critical in supporting the spindles and machining components. The thermal plots indicate that the temperature of the bearings during this study increases proportionally with the increase in the operating speeds, rather than with changes in the vibration levels. This can be explained by the low level of vibrations encountered. If the vibration levels were large enough, the additional effects may cause the thermal levels to increase. The levels measured are well below the critical value of 150 $^{o}$C, and show only an increase of 10 $^{o}$C over a range of 8 000 RPM (133 Hz).

Figures 51 to 54 are presented to illustrate the changes in the frequency spectrum in comparing the machining unit operation at 8 000 and 13 000 RPM (133 and 217 Hz). In Figure 51, the dominant peak at 133 Hz represents the speed of rotation of spindle 2 and similarly in Figure 53 at 217 Hz.

Analysis of the machine head supported on the table base indicates that no major operating problems exist, and the system with respect to lubrication of the bearings is thermally stable. No large vibration levels were recognized at the critical frequencies measured in the static analysis study, indicating that no critical frequencies were excited during operation.

### 5.2.2 Machine Base Analysis

The analysis of the machine head supported on the machine base is similar to that of the table analysis. This study is used to analyze the influence of structural support in comparison to the table base. A run-up analysis is considered in this portion of the study. The machine head was not supported using load actuators but was fitted with drill 1 supported by spindle 2, and a dowel supported by spindle 1.

Figures 54 to 57 represent the frequency spectra vibration levels observed with free-run mode at the operating speed for the four frequency windows. In Figures 54 and 55, the first peak represents the operating speed of the gear drivetrain at 83 Hz. The two dominant peaks represent the operating speeds of spindles 1 and 2. The vibration amplitude levels are still fairly small. The effects of the gear drivetrain were minimized, as well as the spurious spikes seen in Figure 55. The vibration levels shown in Figures 56 and 57 are fairly low, and indicate little difficulty with the operation of the machine head on the machine base, with respect to operation of the gear drivetrain and bearings.

The vibration amplitude levels measured for spindle 1 and 2 at each reference location during the run-up of the machine head are presented in Table 7. Figure 58 presents the table results graphically. The vibration level from operation is lowest at location 2, and the levels fluctuate at the other points, with levels larger at locations about spindle 2.

Location 2 is situated such that it does not lie directly over a bearing, and thus tends to pick up a weaker signal. It can be seen that the vibration levels are fairly small, and the operation of the machine head during the free-run analysis presents no major vibration problems.

Figure 59 presents the thermal characteristics of the bearing lubrication during the run-up study. The thermal level is well below the critical value of 150 $^{0}$C, and the pattern indicates again that the thermal properties of the bearings are proportional to the operating speed, and currently due to the low level of vibrations, are not affected by the fluctuations of vibrations during operation.

Run-up frequency spectra are presented in Figures 60 to 63. Free-run, and run-up analysis of the machine head supported on the machine base indicates more noise free data, as well as fewer disturbances or influences due to the motor and other power support systems, such as on the table base. The dominant source of vibration was due to the rotation of the spindles, although the levels were not very large.

Direct correlation between the operating frequencies and the modal frequency response values were not found. No difficulties are apparent with operation of the machine head at the operating speed. The vibration response is more stable when the head is mounted on the machine base.

The proportional relationship between the operating speed and the thermal condition of the bearing lubrication suggest that temperature can be used in addition to vibration levels to determine if the system is operating satisfactorily. Spurious or large non-proportional rises in temperature can be used as measure of the system and thermal instability of the bearings.

## 5.3 DRILLING ANALYSIS

The drilling study is used to determine the capability and stability of the machine head during machining operations. Two HSS drills of different diameters were used to determine any differences that may exist during the cutting process based on the diameter of the drill. The two drills used did not affect the general behaviour of the cutting process and vibration behaviour of the machine head and the machining unit.

In general, the vibration response during drilling behaviour was consistent at all transducer locations. Locations 1 and 3 provided the least obstructed output for monitoring purposes. These are located directly over the first bearing of spindle 1 and 2, at opposite sides of the machine head. They also exhibit similar vibration transmission characteristics from the bearings.

Spindle 2 supports the cutting tool, and its behaviour is of prime concern to determine the stability of the machine head. The results presented in this section are for locations 1 for spindle 2. It should be noted from the free-run analysis that at each measuring location, the transducer is capable of sensing the vibration behaviour of the machine head components. In Figures 64 to 71, the spectra in blue denotes the free-run and the spectra in red denotes the drilling vibration characteristics.

The operating speed used for drill 1 of 0.375 inch diameter HSS, was approximately 9 960 RPM (166 Hz). A slightly higher operating speed of 10 530 RPM (168 Hz) was used for drill 2 of 0.250 inch diameter HSS.

The drilling process produced increased vibration levels above the free-run values. The increase is due to the cutting forces which produce forced vibrations. Although, levels increased, their amplitudes are moderate and do not indicate any weaknesses of the machine head structure and operational difficulties.

### 5.3.1 **Drill 1**

In the lower frequency windows, as in Figures 64 and 65, distinct, but minor frequency sidebands exist near the operating frequency of spindle 1 and 2, during the drilling process. From observations, during the drilling process, these minor peaks were formed during the drill retraction or pullout, and are predominantly caused by chip removal as the drill feed direction is reversed.

Spindle 1, which supports a dowel, exhibits similar effects during the drilling process by spindle 2. The fluctuations experienced by both spindles, although only spindle 2 is drilling, indicate the level of interaction between the spindles and the machining unit. If both spindles were drilling and one were to experience poor drilling behaviour, the other spindle would reflect this behaviour in its drilling process. Although, no problems were experienced in this study, this facet should be monitored closely.

In Figures 65 and 66 during the drilling process, a harmonic frequency is visible at the operating speed of spindle 2. It was not visible during the drilling process with drill 2. The second harmonic suggests that perhaps there was some looseness of drill 1 supported by the collet.

The amplitude levels in Figure 67 are relatively low. In the frequency range presented, this indicates that the bearings are relatively new and significant amplitude increases can be used as indicators of bearing wear. At a frequency of 4 496 Hz a spurious spike occurs. This was previously observed in some free-runs and may be due to some local effects.

### 5.3.2 **Drill 2**

The free-run and drilling characteristics of drill 2 are similar to that of drill 1. Unlike drill 1 no distinct harmonic frequencies were discernible, as shown in Figures 68 to 71.

The vibration levels are fairly low, as was the case with drill 1. However, it appears that drill 2 experiences slightly higher levels of vibration. This can be attributed to the fact that approximately the same operating speed was used for the same feed rate. This increases the drilling force required, which in turn would increase the vibration levels. As in practice, a much higher operating speed should be used with smaller diameter drills.

The low levels of vibration during free-run and drilling indicate that the operation of spindles, and the machining unit presents no major difficulties at 10 000 RPM. There is no evidence of any vibration or tendency to chatter at the fundamental natural frequencies.

Also, the obtained operating spectra levels represent a good base for future machinery monitoring and diagnostic systems.

# 6. CONCLUSIONS

Vibration analysis of the Dual-Spindle High-Speed Drilling Machine Head was studied by means of modal analysis via a high-speed spindle specimen, frequency response of the machine head, and free-run and drilling operations.

The conclusions reached from the tests performed indicate the following :

- The developed high speed machining head will perform satisfactorily in respect of vibration and bearing temperature, when drilling aluminum.
- There is no indication of the presence of vibration or tendency to chatter at the determined natural frequencies of bending vibration of the spindles.
- Significant differences existed in the magnitudes of the natural frequencies and vibration mode shapes measured with FEA, the impact modal analysis tests in the V-block fixture, and in the actual ball bearing supports. There was a progressive and significant reduction in the frequency magnitudes, which indicated that the bearings provided the most flexible supports.
- The use of rigid supports for the boundary conditions with the FEA produced unsatisfactory results. Some additional studies, which are not presented in this work, suggested that a solution of this problem was complex. Because the application of the FEA to prototype development is essential, a separate project is proposed.

- A large number of vibration sources and dynamic interaction between the system components. The essential vibration components of the tool-spindle, as excited by the metal cutting action, are sufficiently predominant to permit adequate study.

- Appropriate vibration measurements during long term operation, are capable of providing effective information for the condition monitoring of the tools and bearings.

The study, in addition provides information regarding the drilling of aluminum samples. The Dual-Spindle High-Speed Drilling Machine Head proved successful in its application and study, such that the unit was purchased by an automotive manufacturer for transfer line drilling application of aluminum engine blocks.

# 7. RECOMMENDATIONS

The study conducted indicates that the system is capable of performing adequately at the required operating speed. To improve the accuracy of the results obtained, further analysis and improvements are recommended.

Analysis of the study can be improved by :

- Modal analysis of both spindles 1 and 2 supported in bearing structures, rather than in the V-block structure that represents only the bearing locations.

- For the system studied, three frequency windows are recommended 400, 3 200, and 25 600 Hz, as the 200 Hz window was difficult to average during drilling (2 seconds per cut and average).

- Compare modal analysis and vibration levels with a dull or worn drill.

- Vary the drill sizes and cutting speeds to determine and classify useful ranges of cutting speeds and operations.

- Development of a consistent procedure for defining the boundary conditions for ball and roller bearings is essential for initial design studies utilizing FEA. Due to the point of contact, the problem with the former type of bearings is more complex and should be the subject of a separate research project.

To improve the machining system capabilities, the following are recommended :

- An increase in the helix angle of the drill to facilitate and improve the removal of chips during cutting.

- The use of a coated carbide tool to improve the thermal range between the aluminum workpiece and the cutting tool.

- An increase of the motor horsepower, a minimum of two hp, to allow for increases in feed rates, and higher operating speeds.

- The study provided base line measurements which can be used for condition monitoring with a preventative maintenance system.

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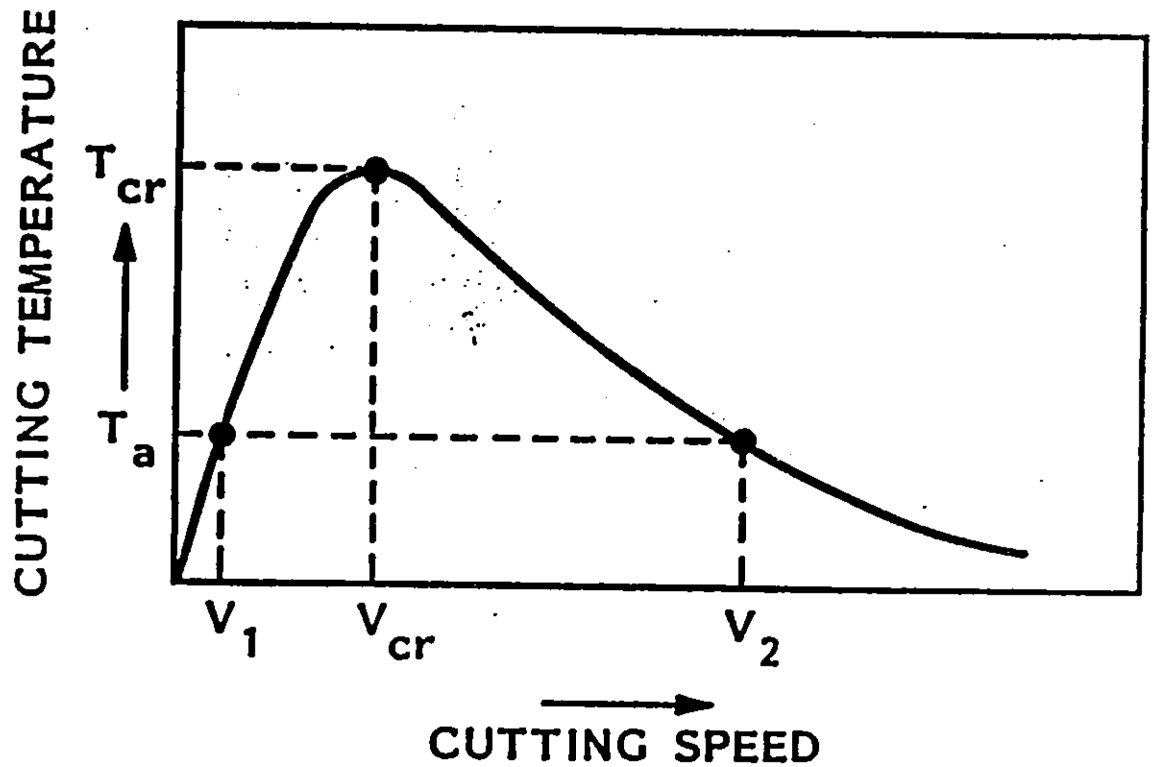



Figure 1 : Idealized cutting speed-cutting temperature plot [4].

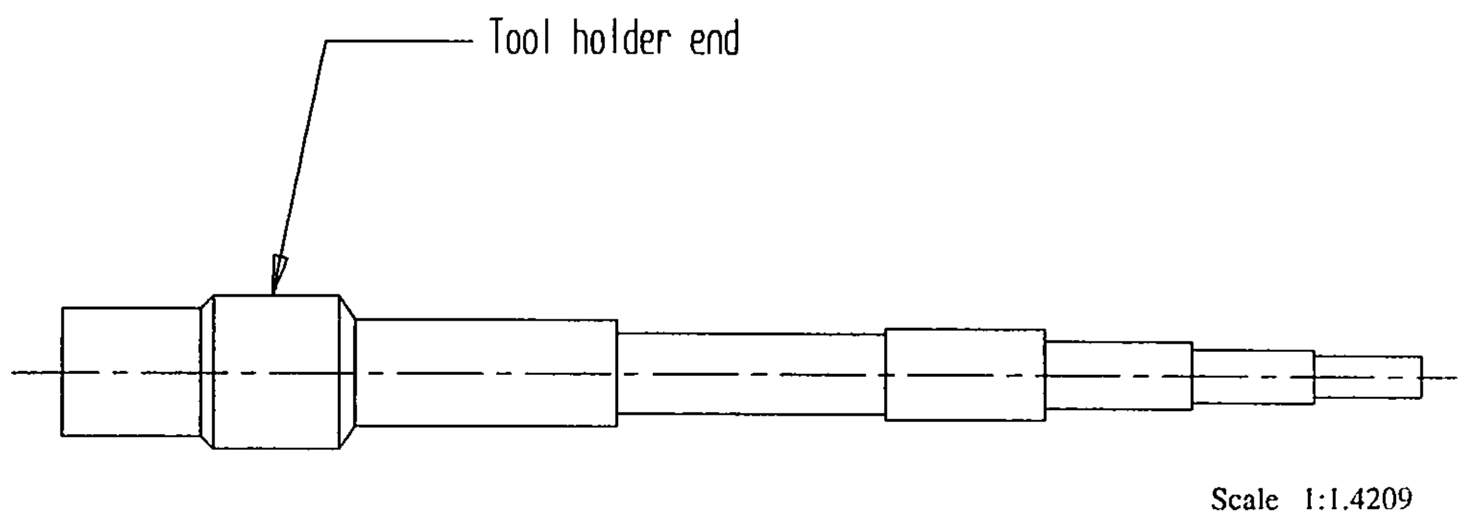



Figure 2 : Diagram of Spindle 1 design

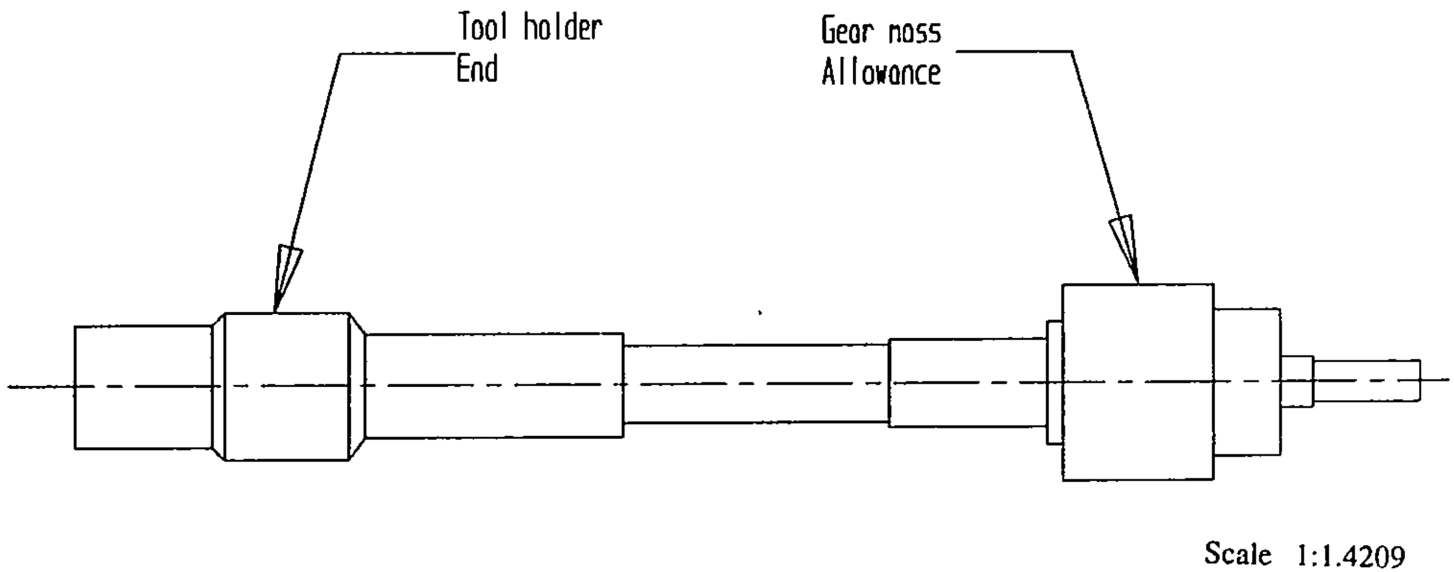



**Figure 3 : Spindle 1 test specimen design with gear mass allowance**

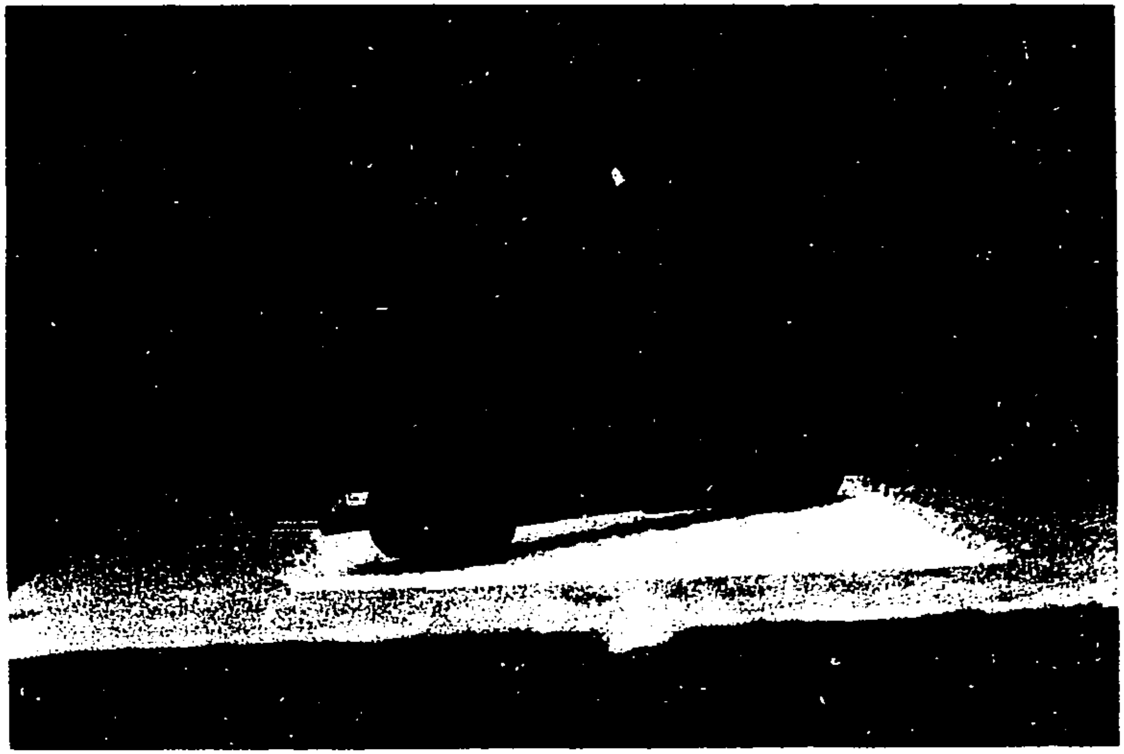


Figure 4 : Photograph of Spindle 1 with gear mass allowance

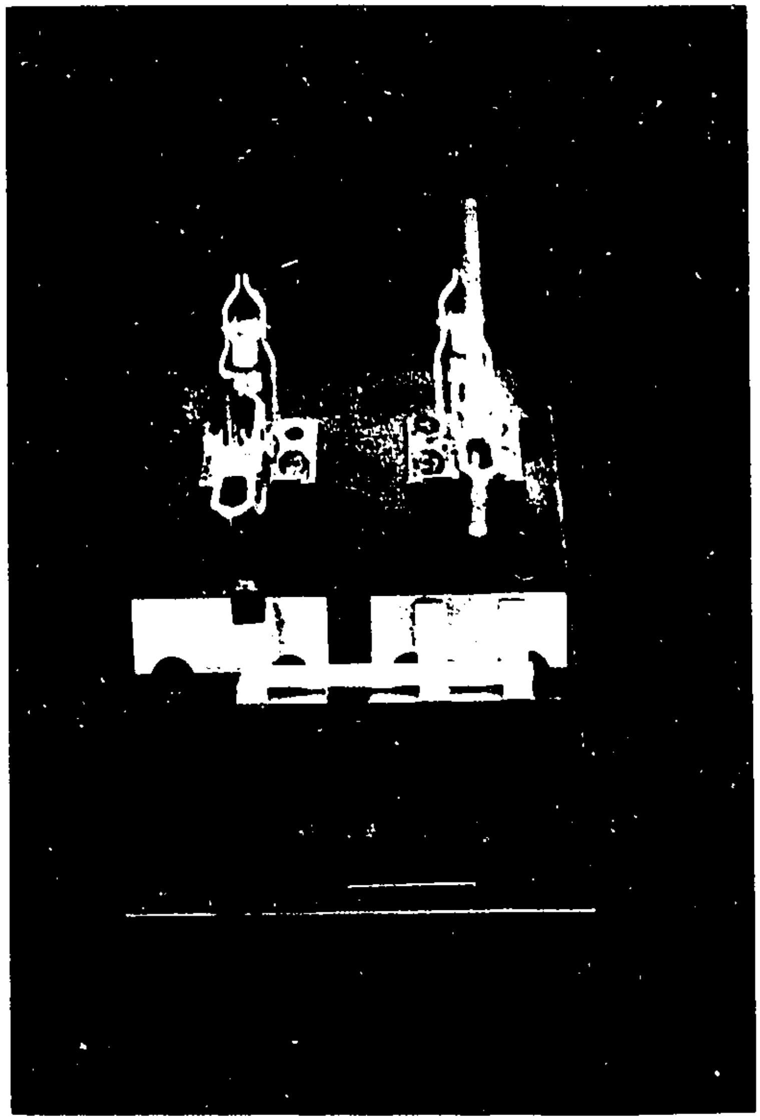


Figure 5 : Photograph of V-block fixture

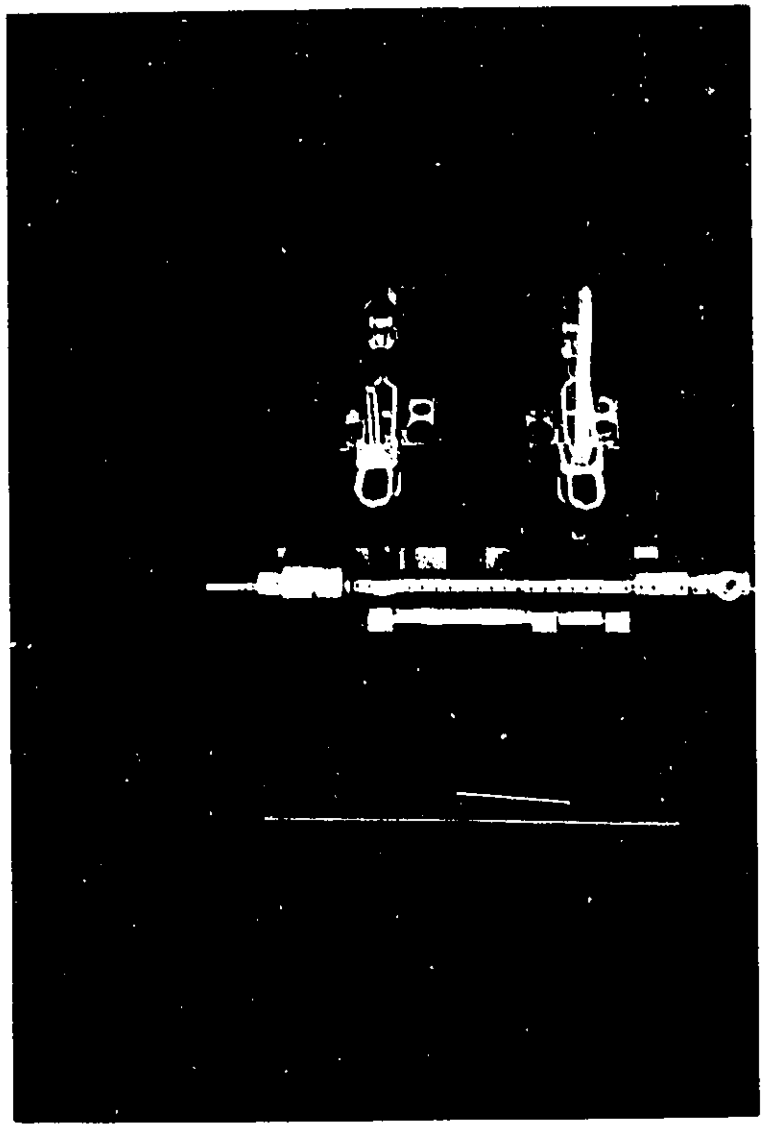


Figure 6 : Photograph of the V-block fixture with Spindle 1

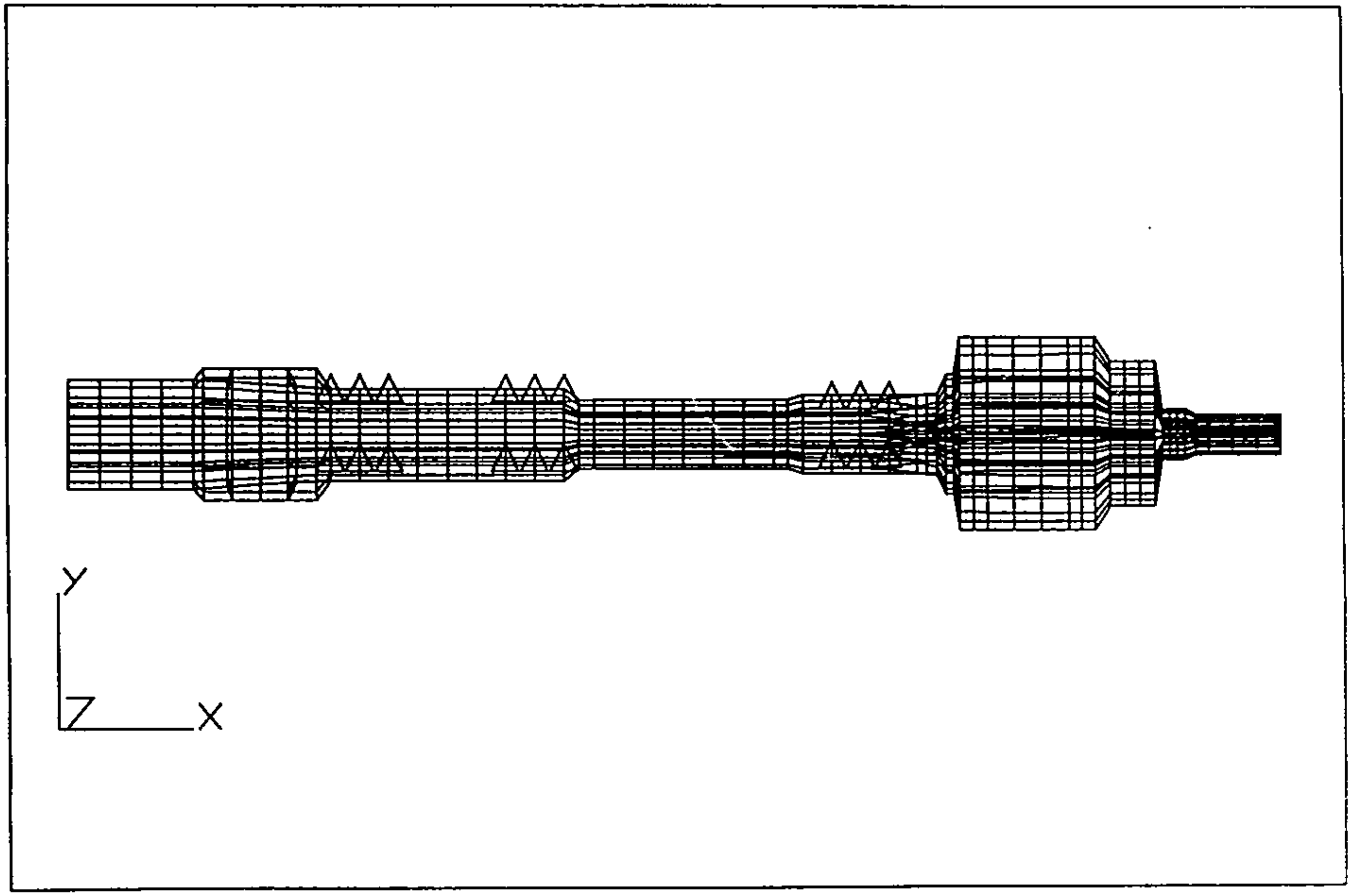



Figure 7 : Finite element model of spindle 1.

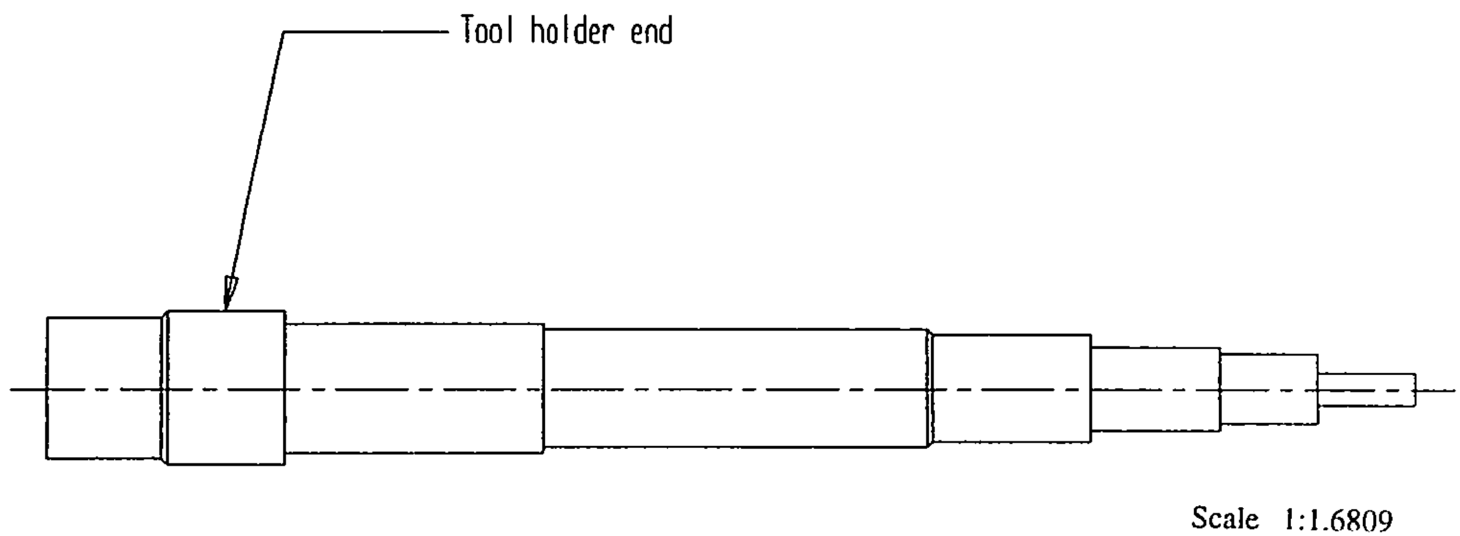



Figure 8 : Diagram of Spindle 2 design

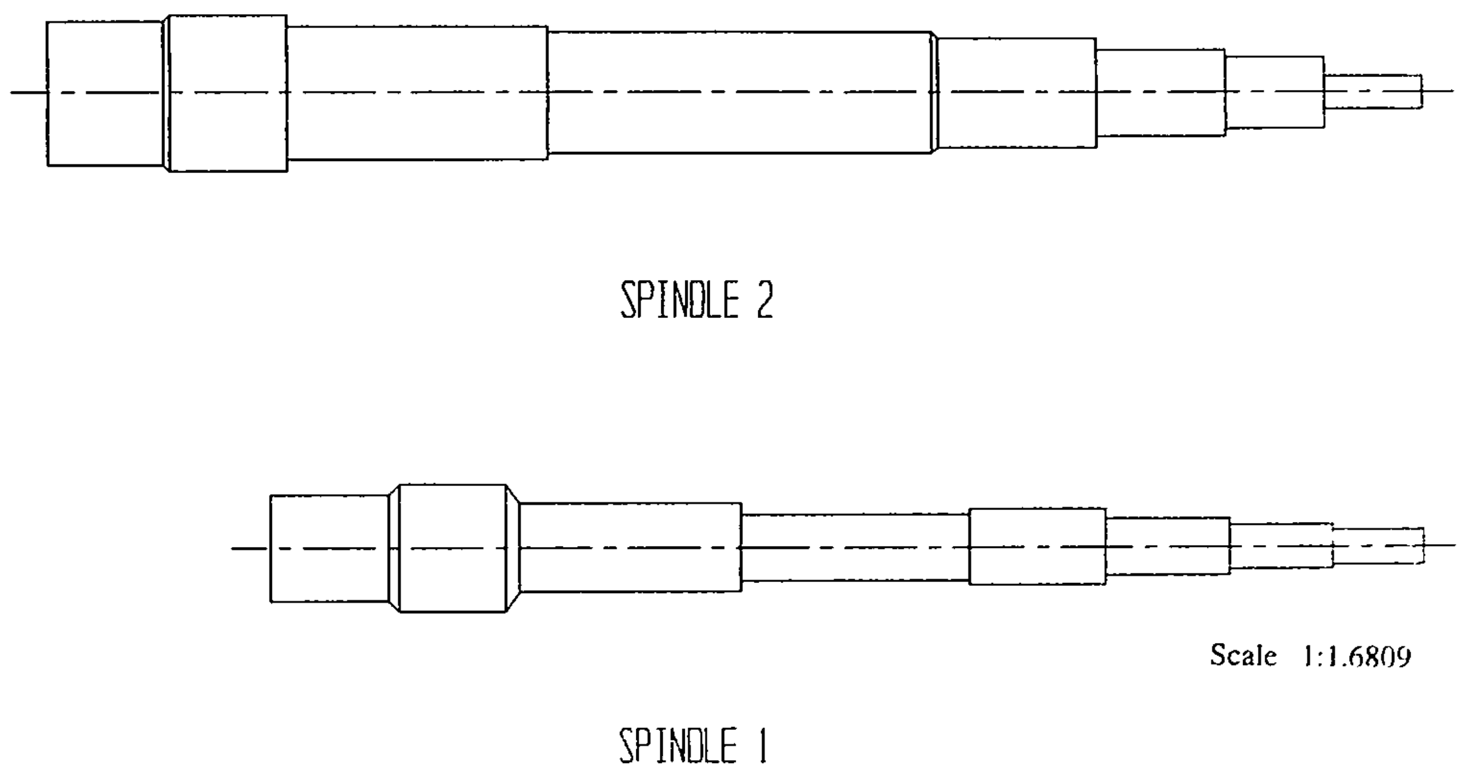



**Figure 9 : Illustration of Spindle 1 and Spindle 2**

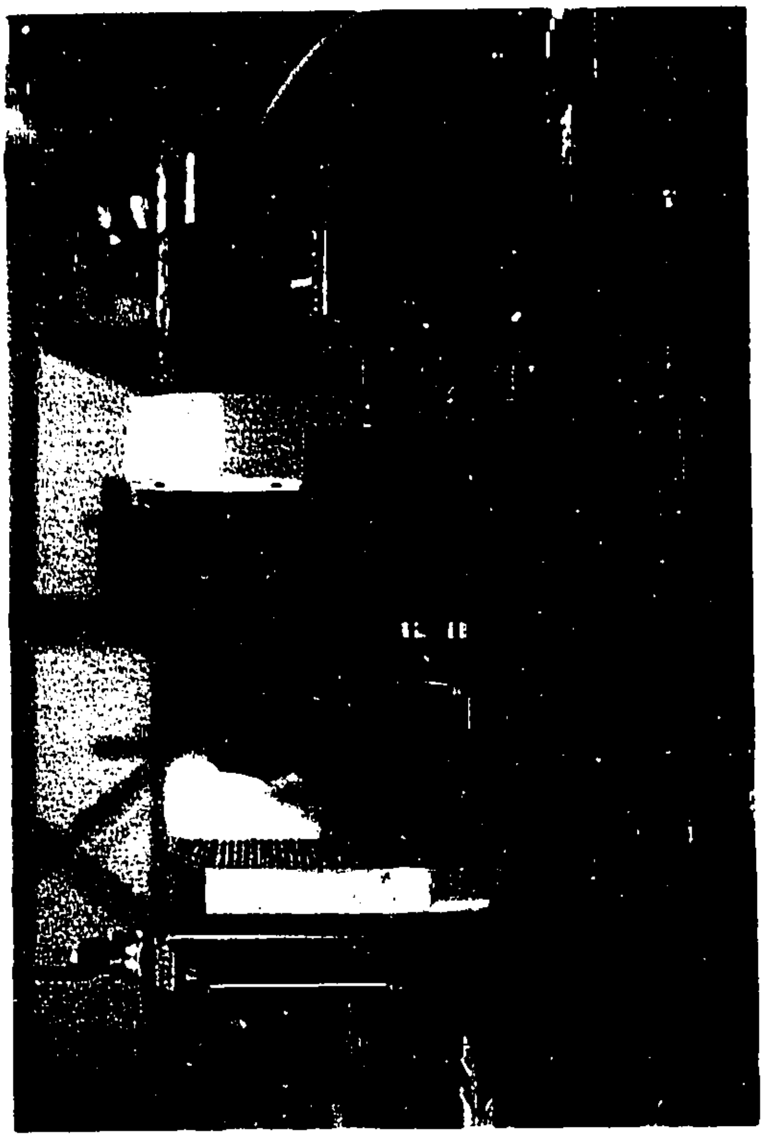


Figure 10 : Photograph of the Dual-Spindle High-Speed Drilling Machine Head

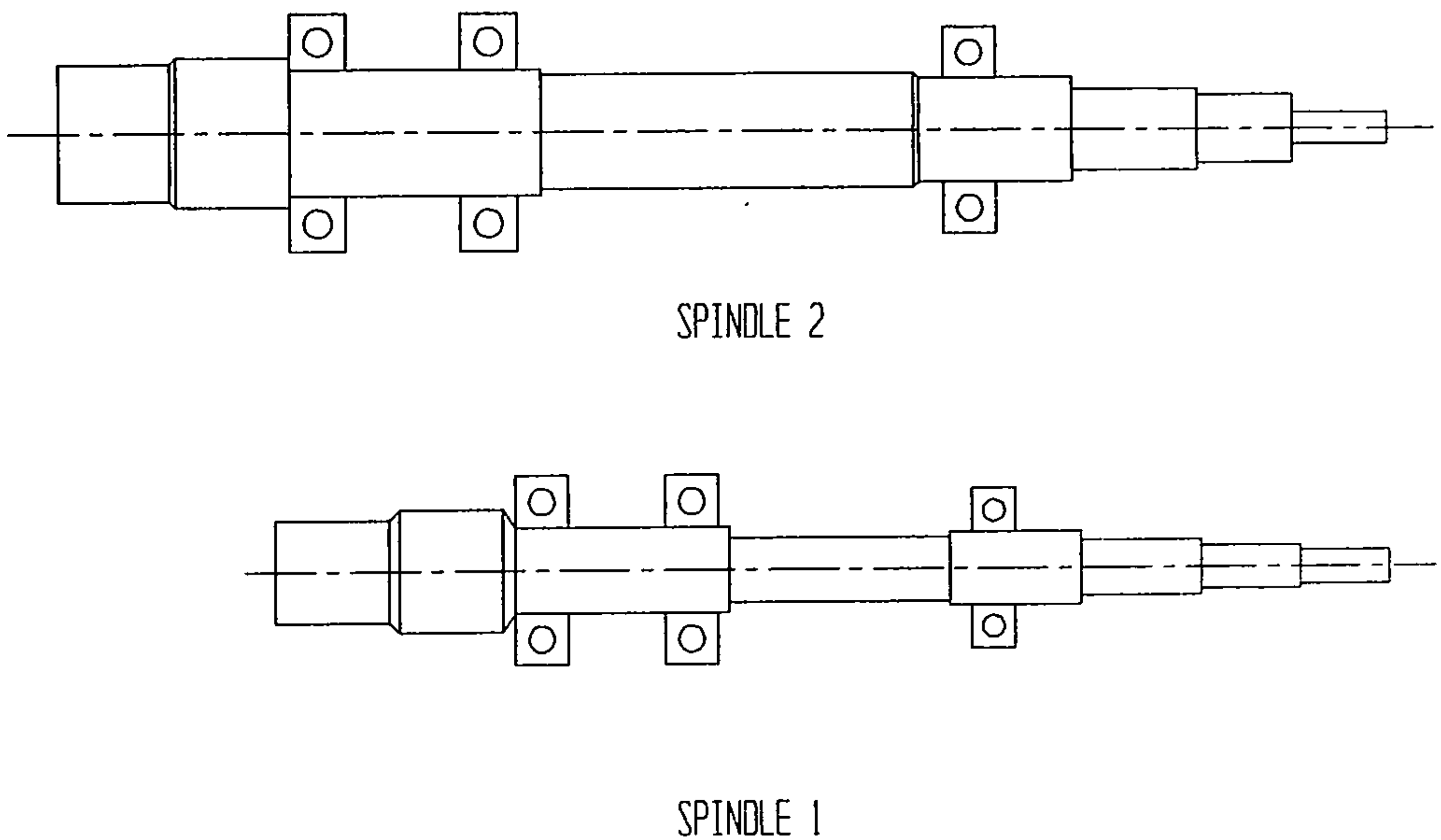



Figure 11 : Spindle 1 and spindle 2 bearing support relation

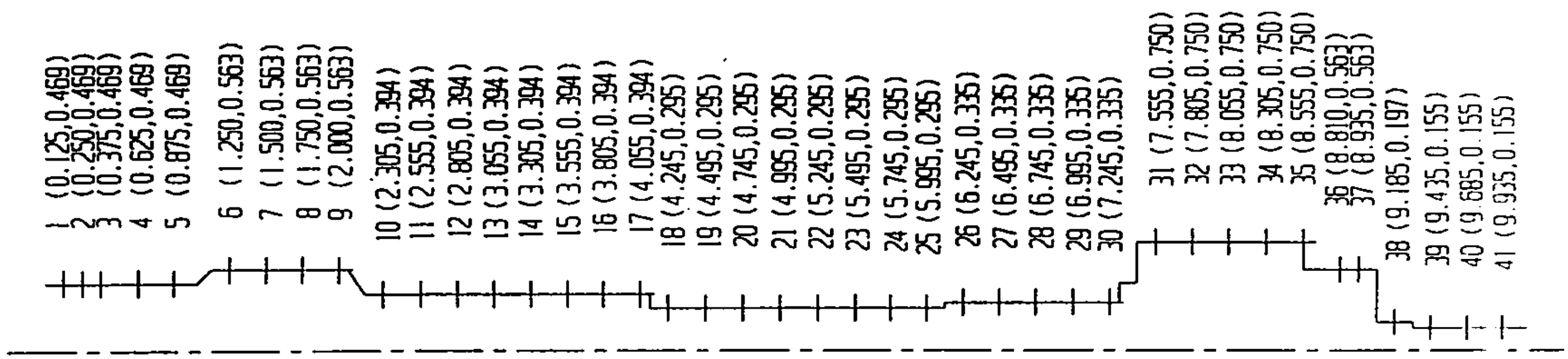



Figure 12 : Co-ordinates of impact points along Spindle 1

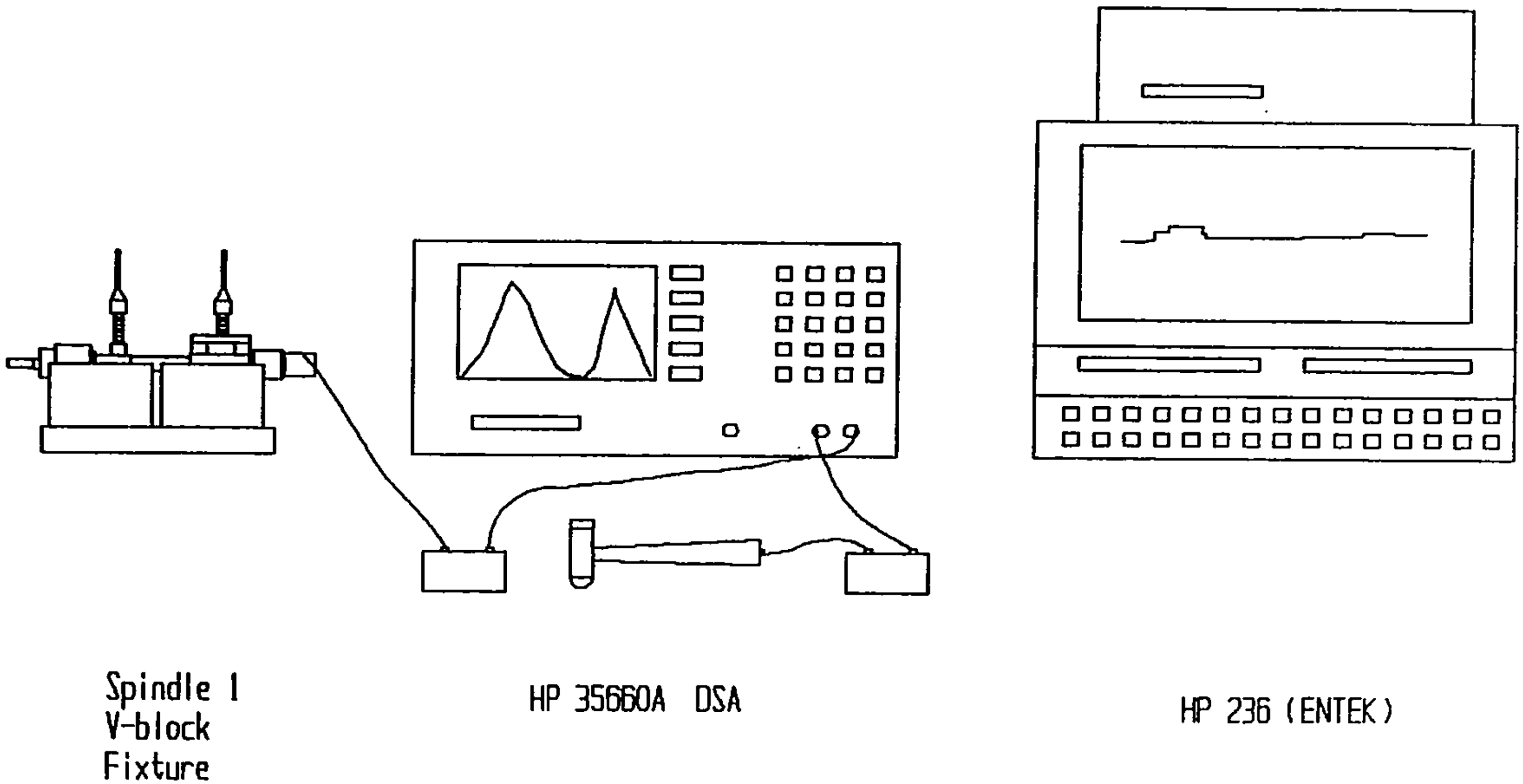



Figure 13 : Experimental modal analysis setup

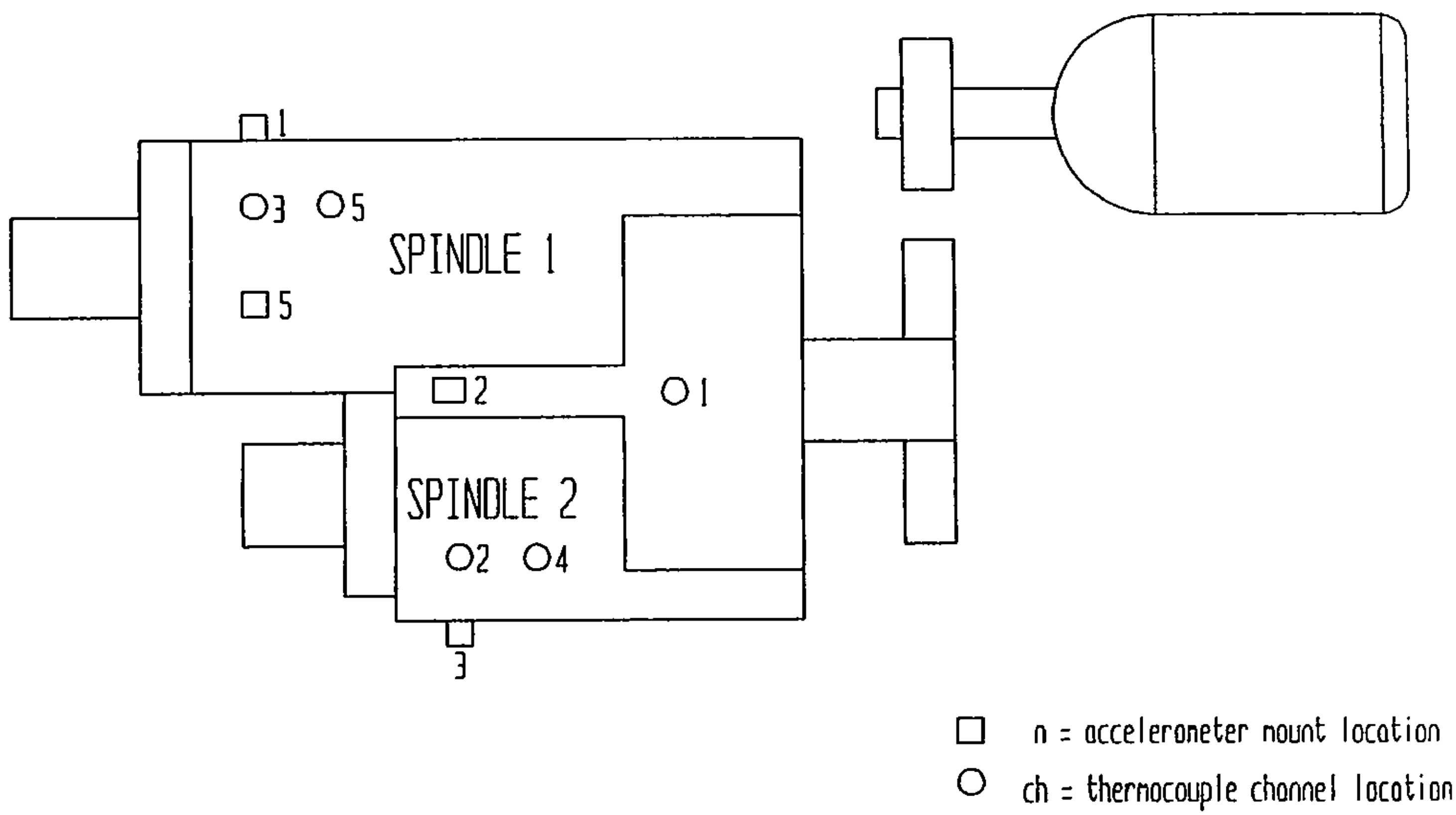



Figure 14 : Machine Head accelerometer mount locations (plan view)

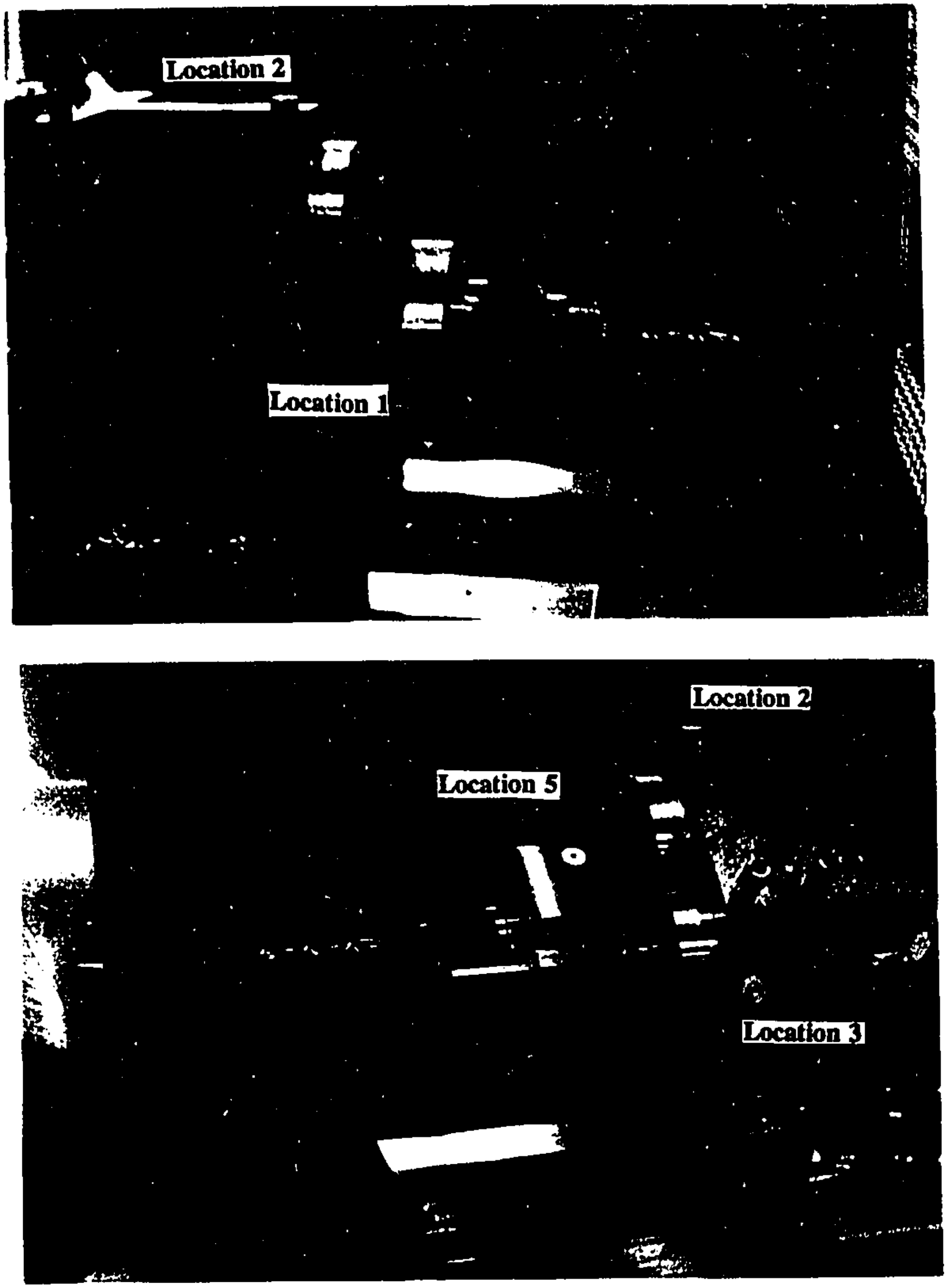



Figure 15 : Machine Head accelerometer mount locations

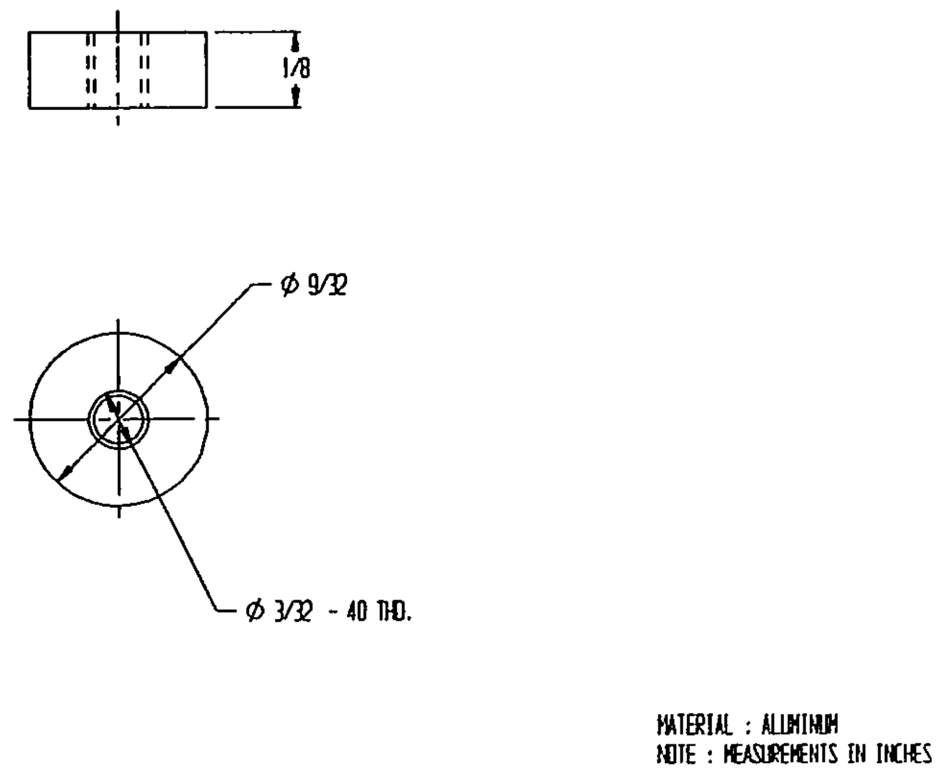



**Figure 16 : Diagram of accelerometer mount**

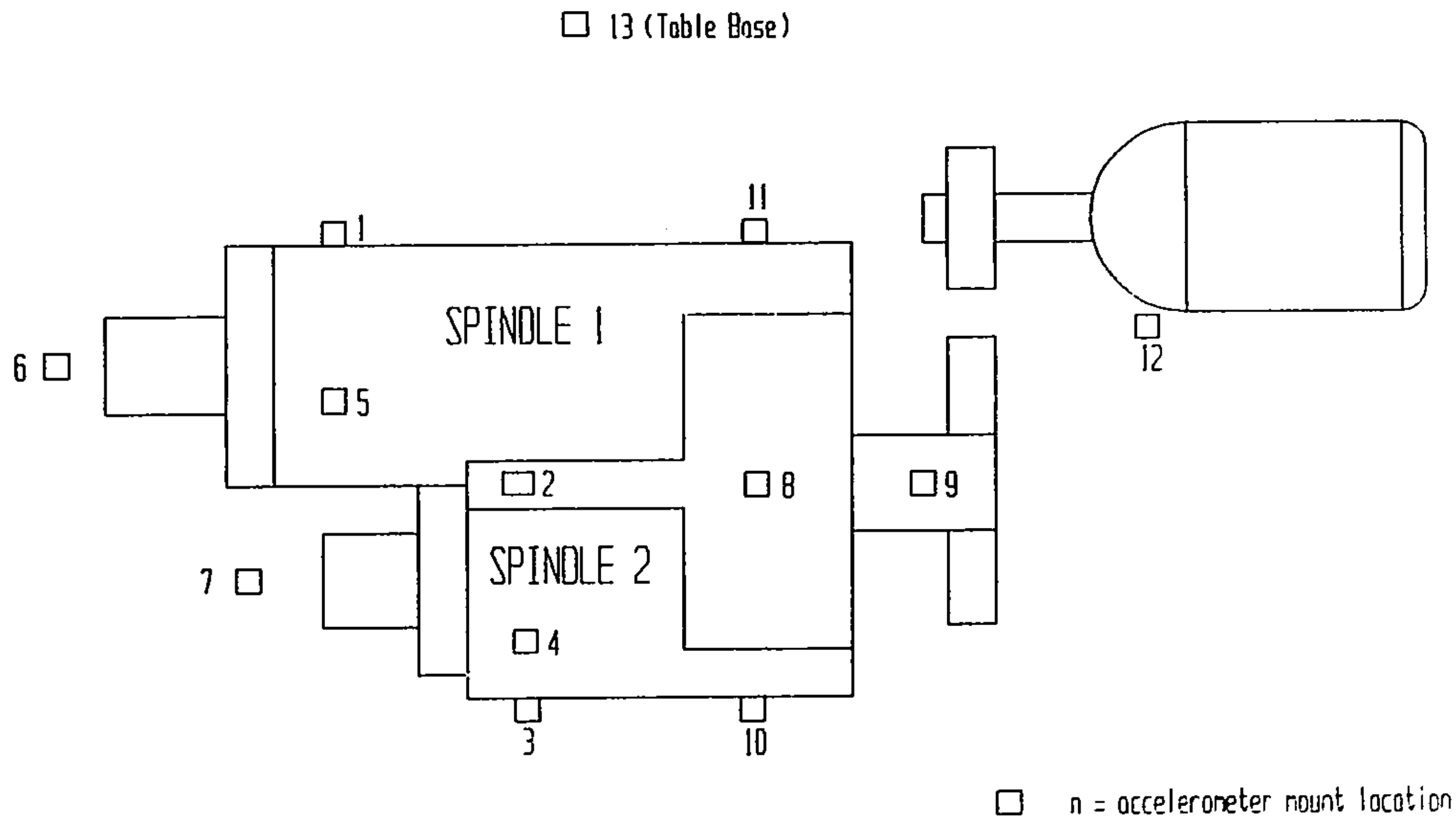



Figure 17 : Machine Head - Table base initial 13 vibration positions of study

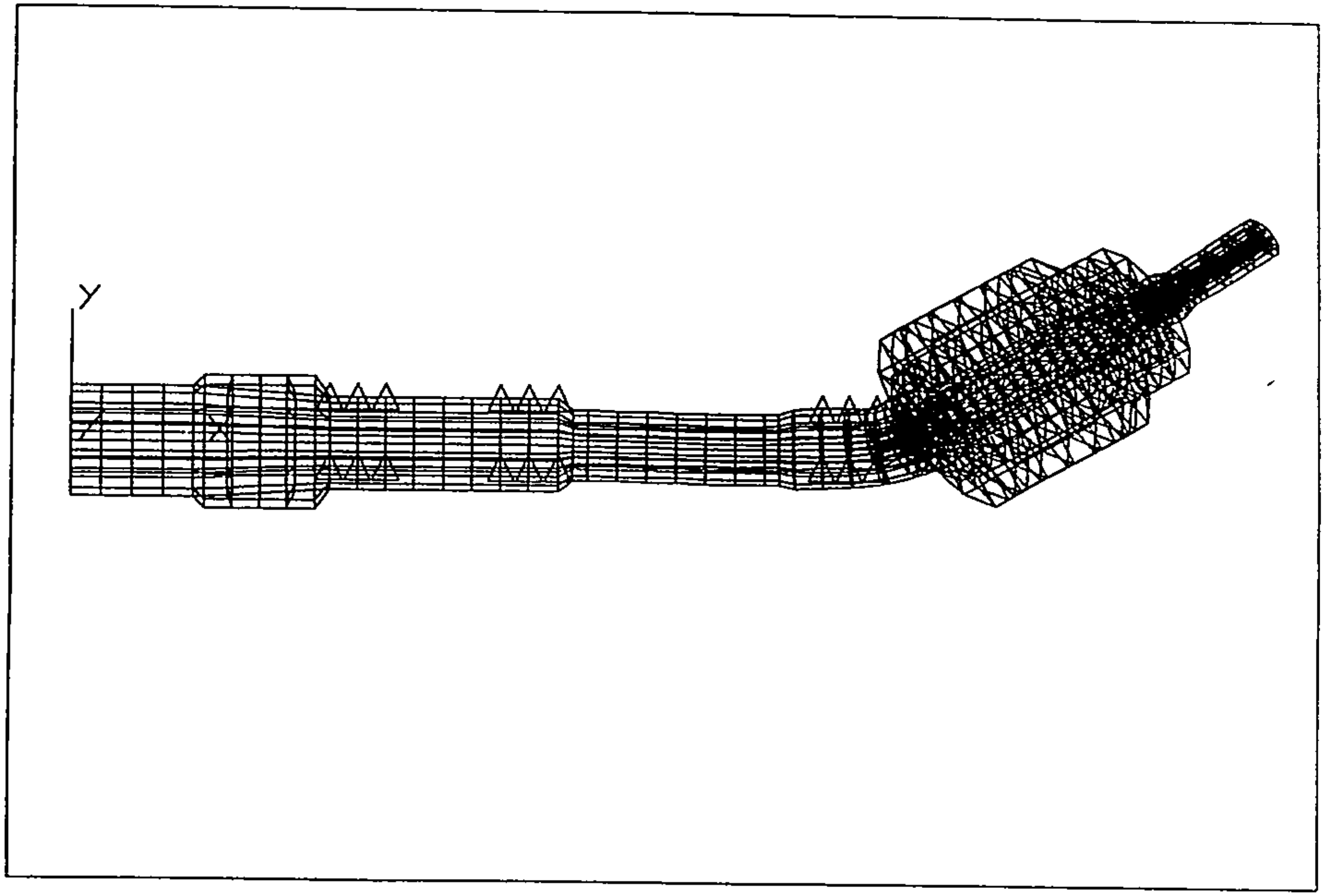



Figure 18 : Spindle 1 finite element deformation at first modal frequency.

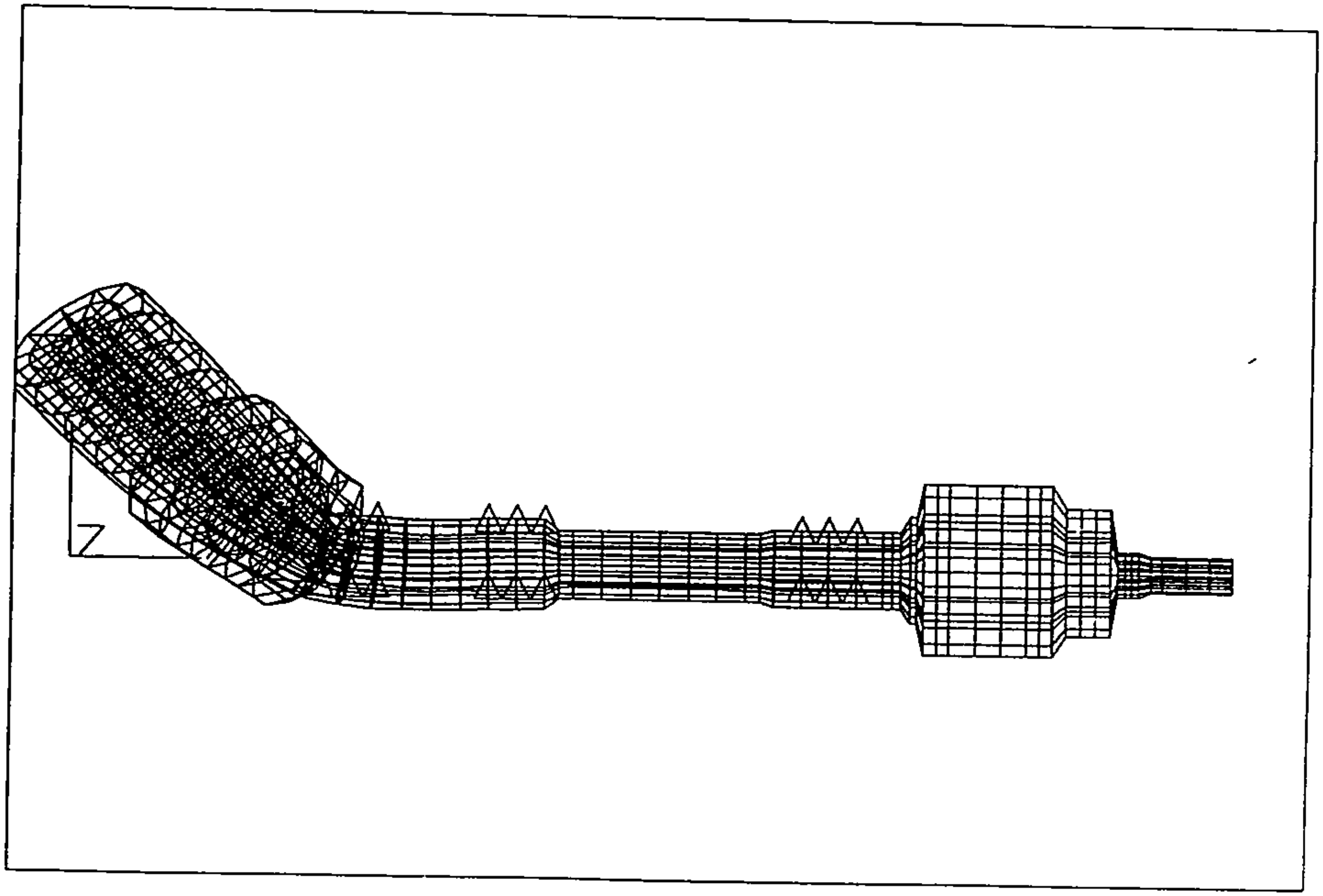


Figure 19 : Spindle 1 finite element deformation at second modal frequency.

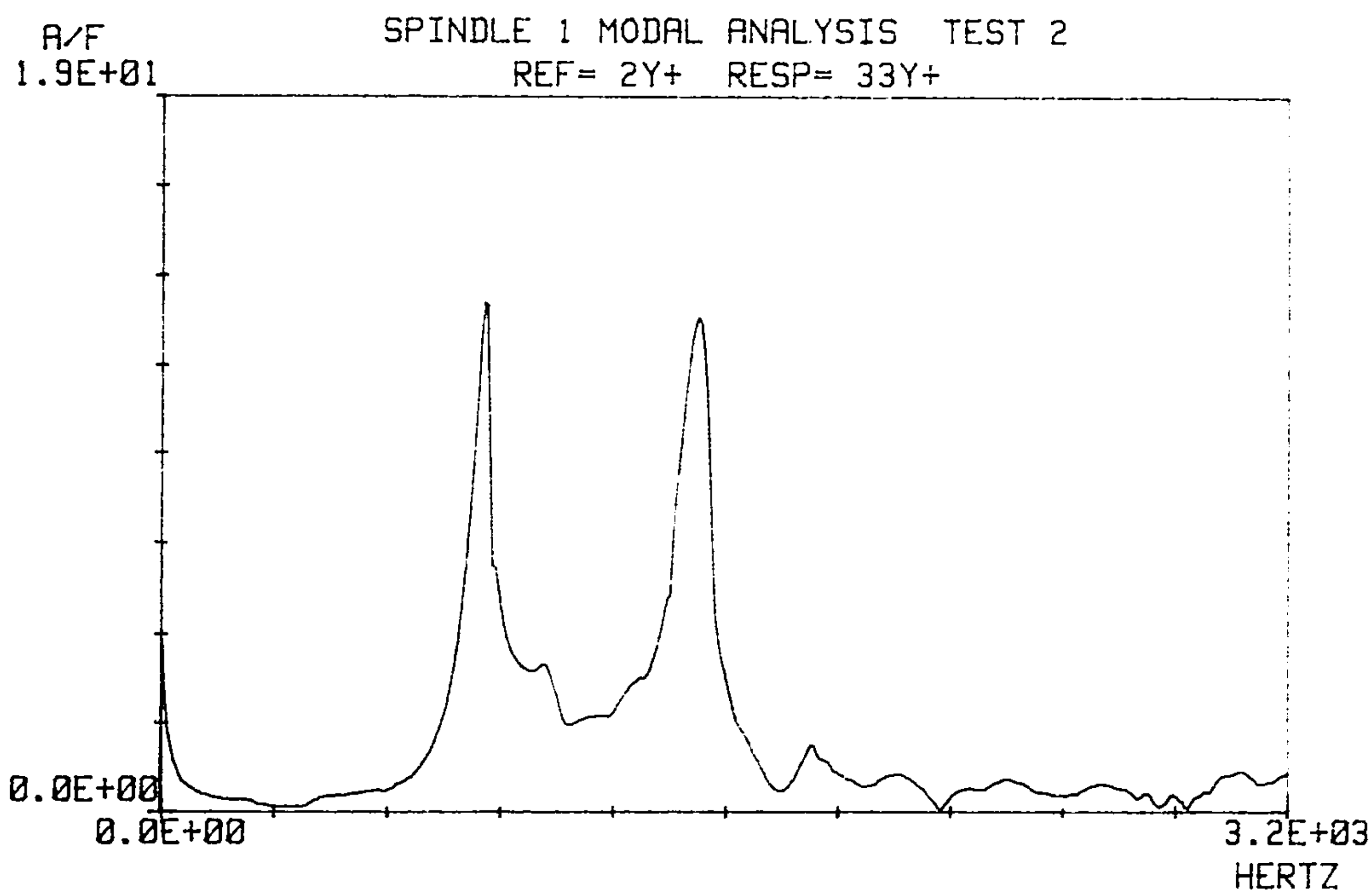



Figure 20: Frequency response curve of Spindle 1 V-block Fixture.

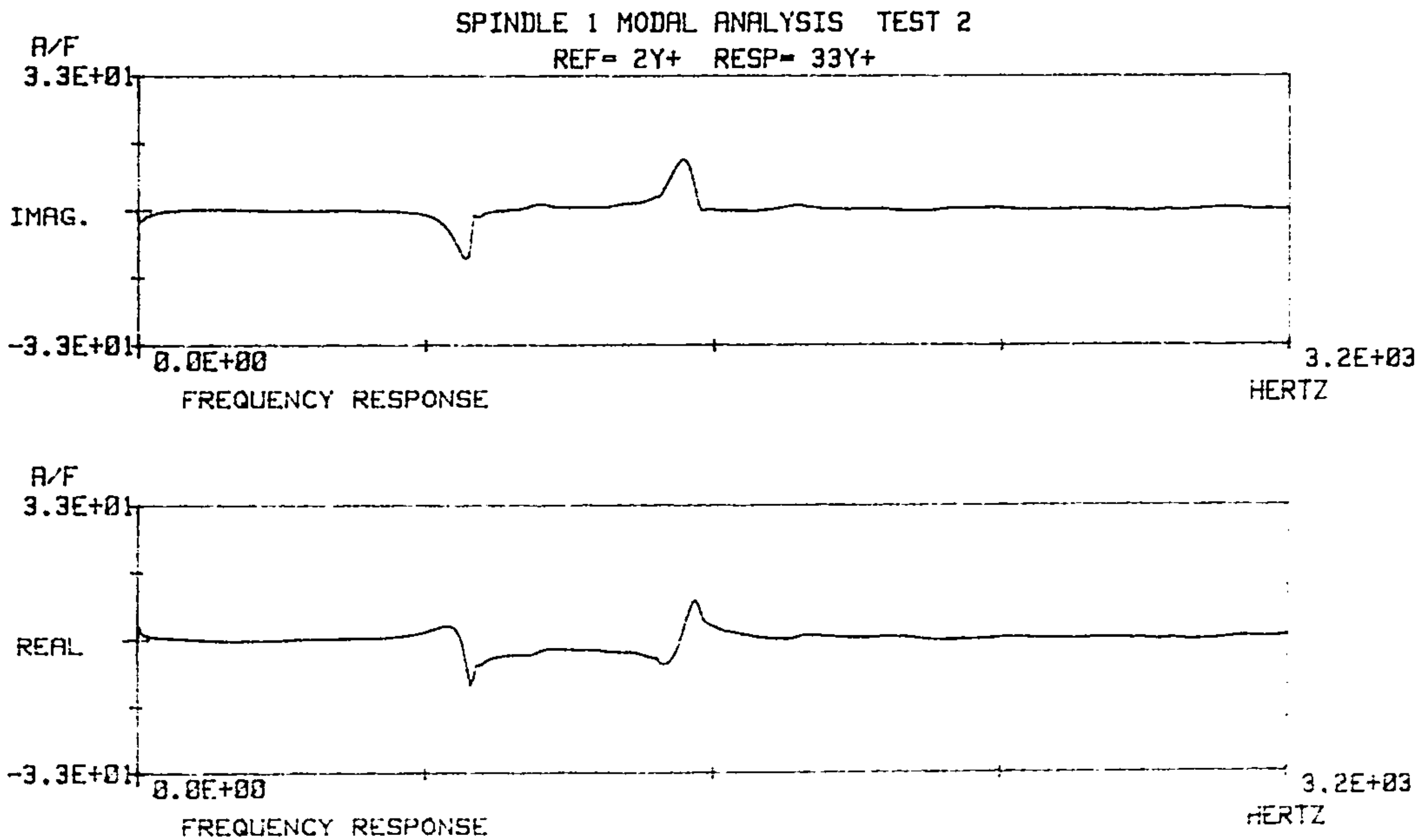



Figure 21: Real and imaginary response curves of Spindle 1 V-Block Fixture.

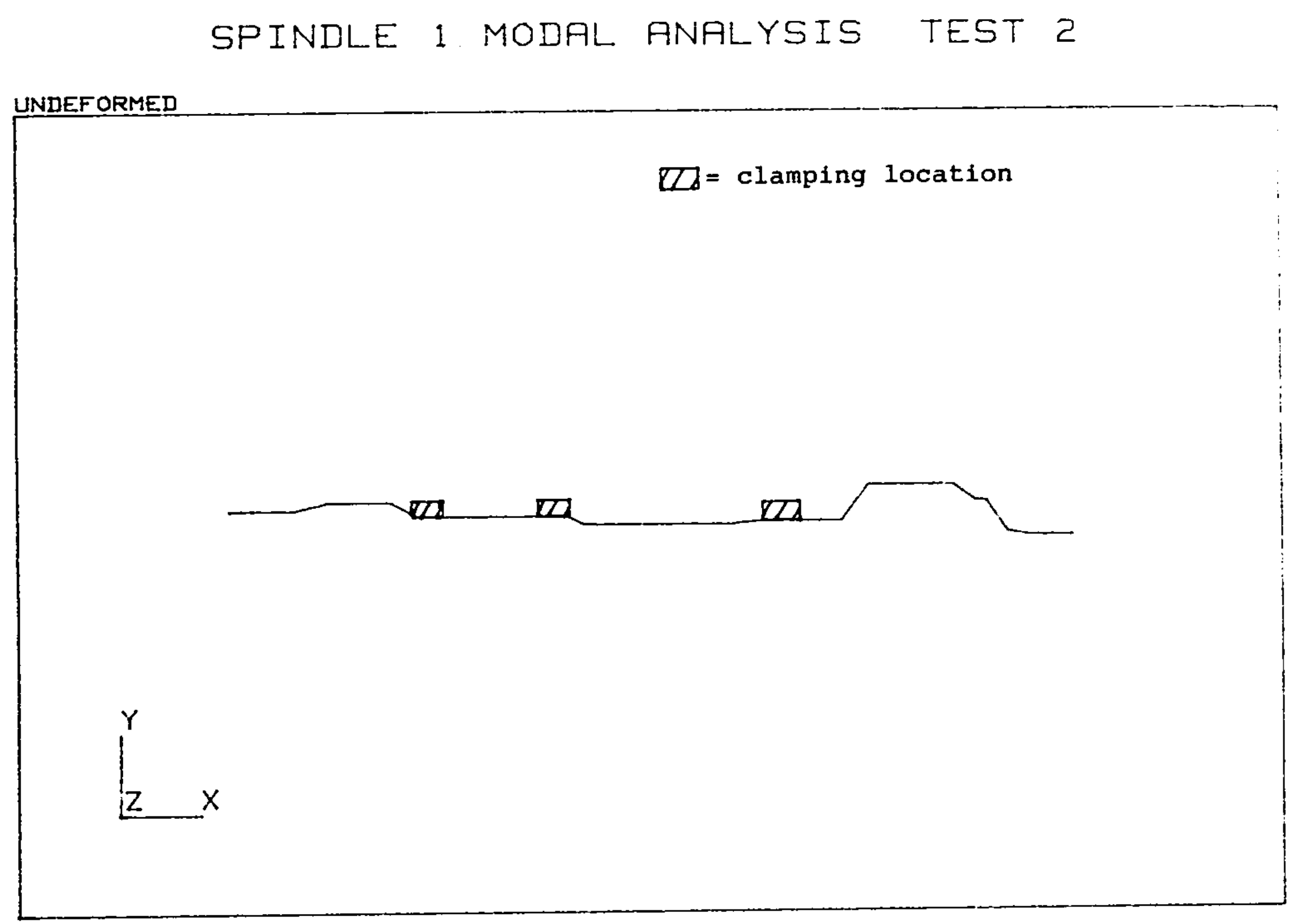



Figure 22: Undeformed outline of Spindle 1.

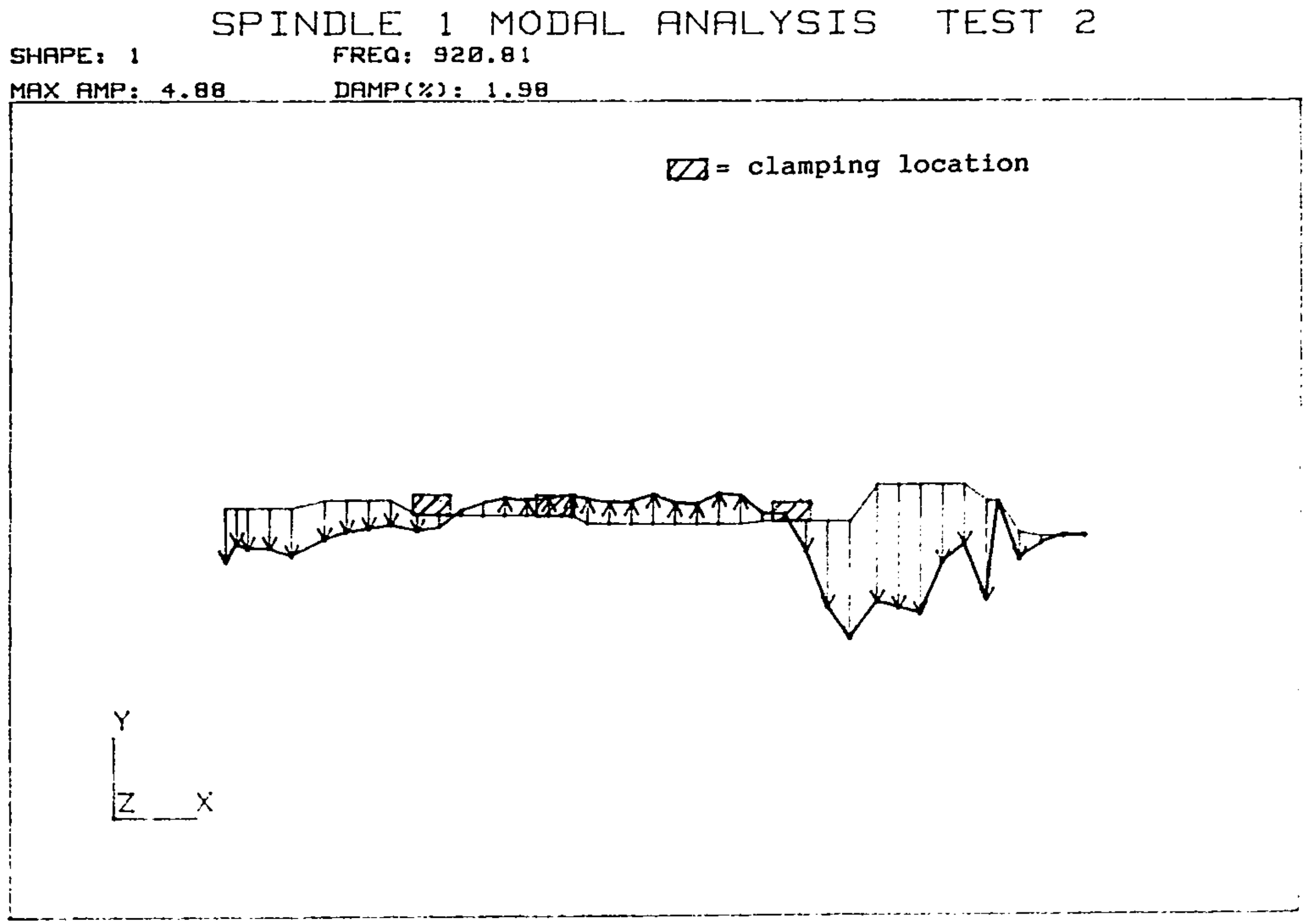



Figure 23: Spindle 1 deformation at the first modal frequency.

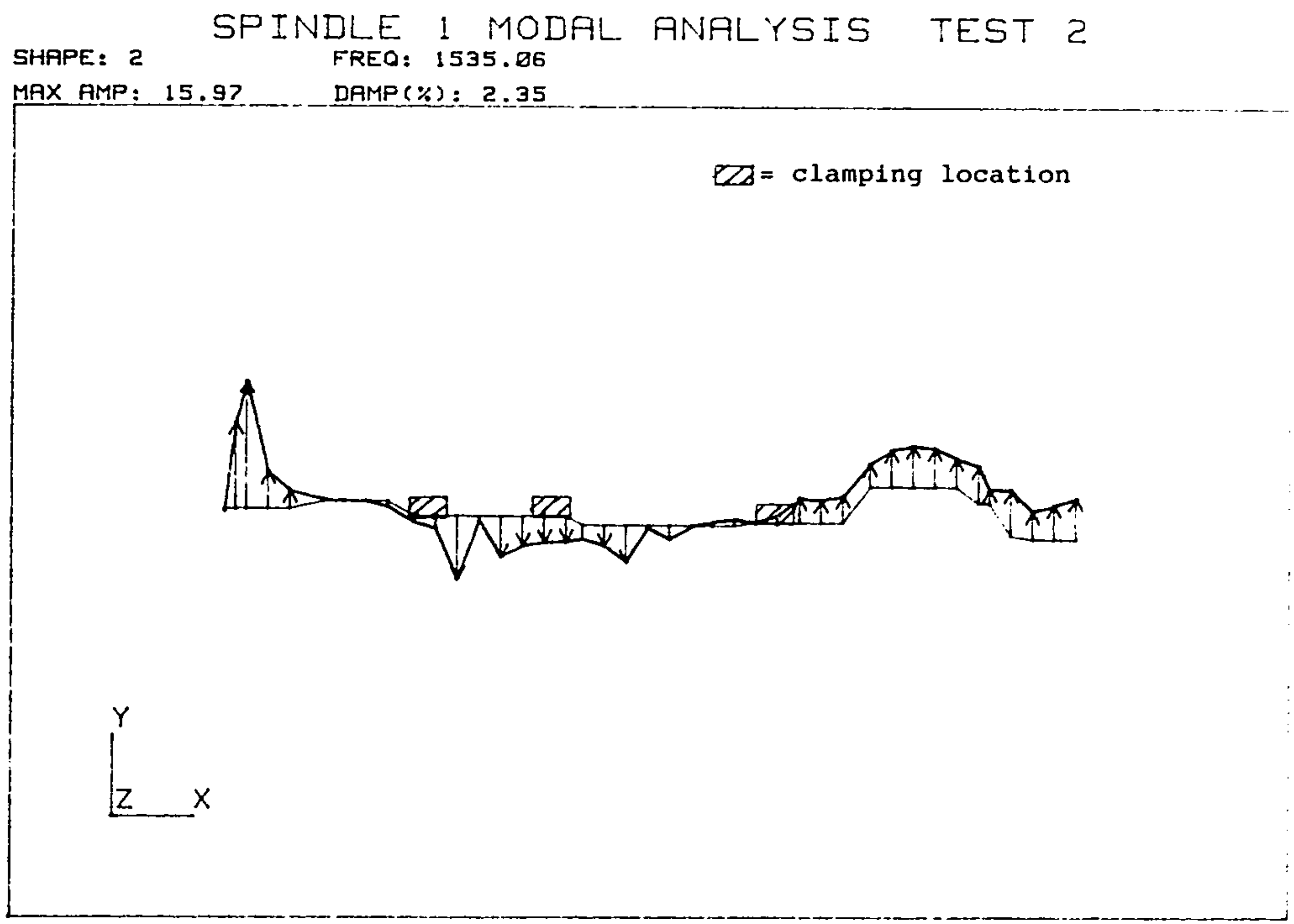



Figure 24: Spindle 1 deformation at the second modal frequency.

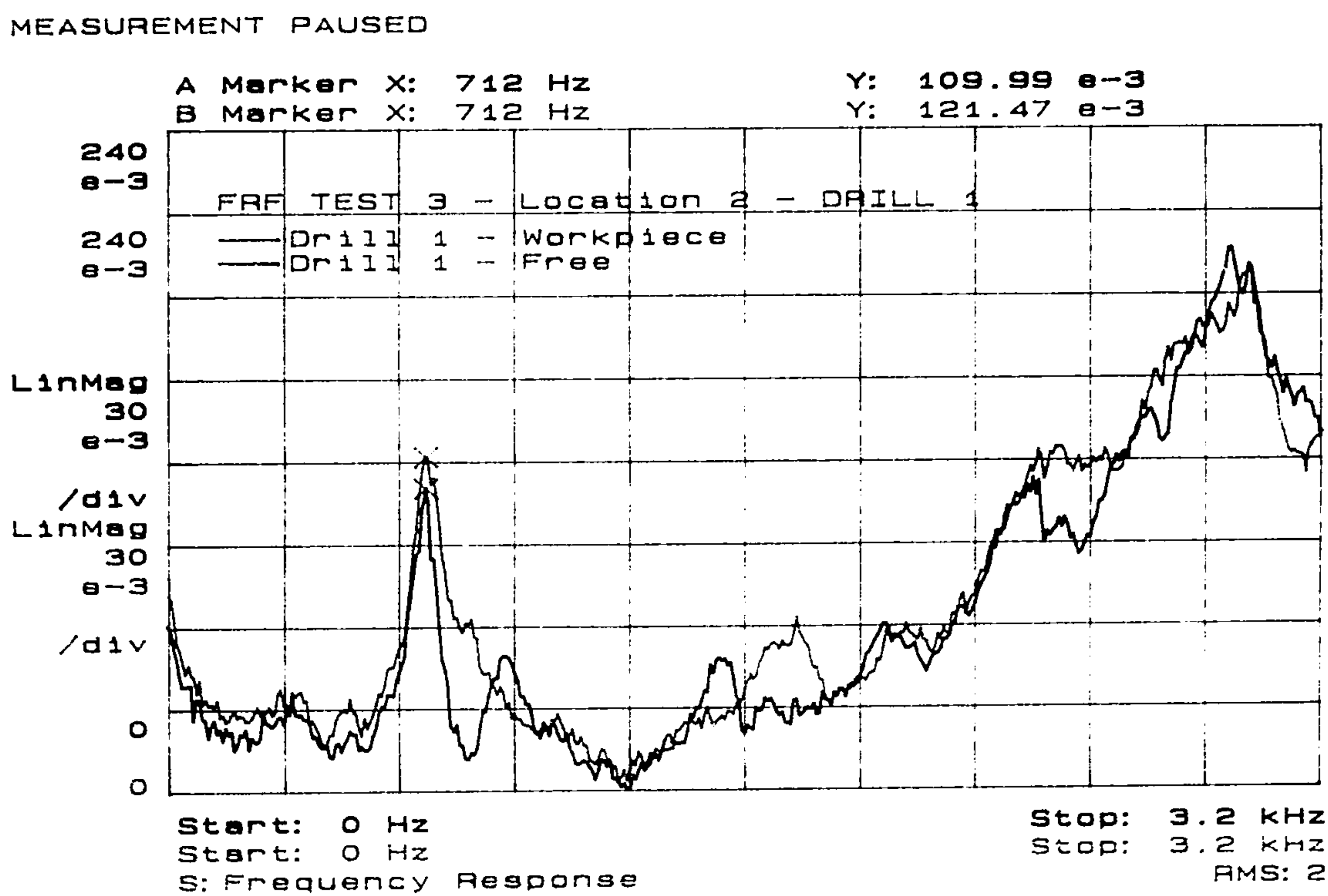



Figure 25: Machine head FRF Test 3 Accelerometer location 2- impact location 2.

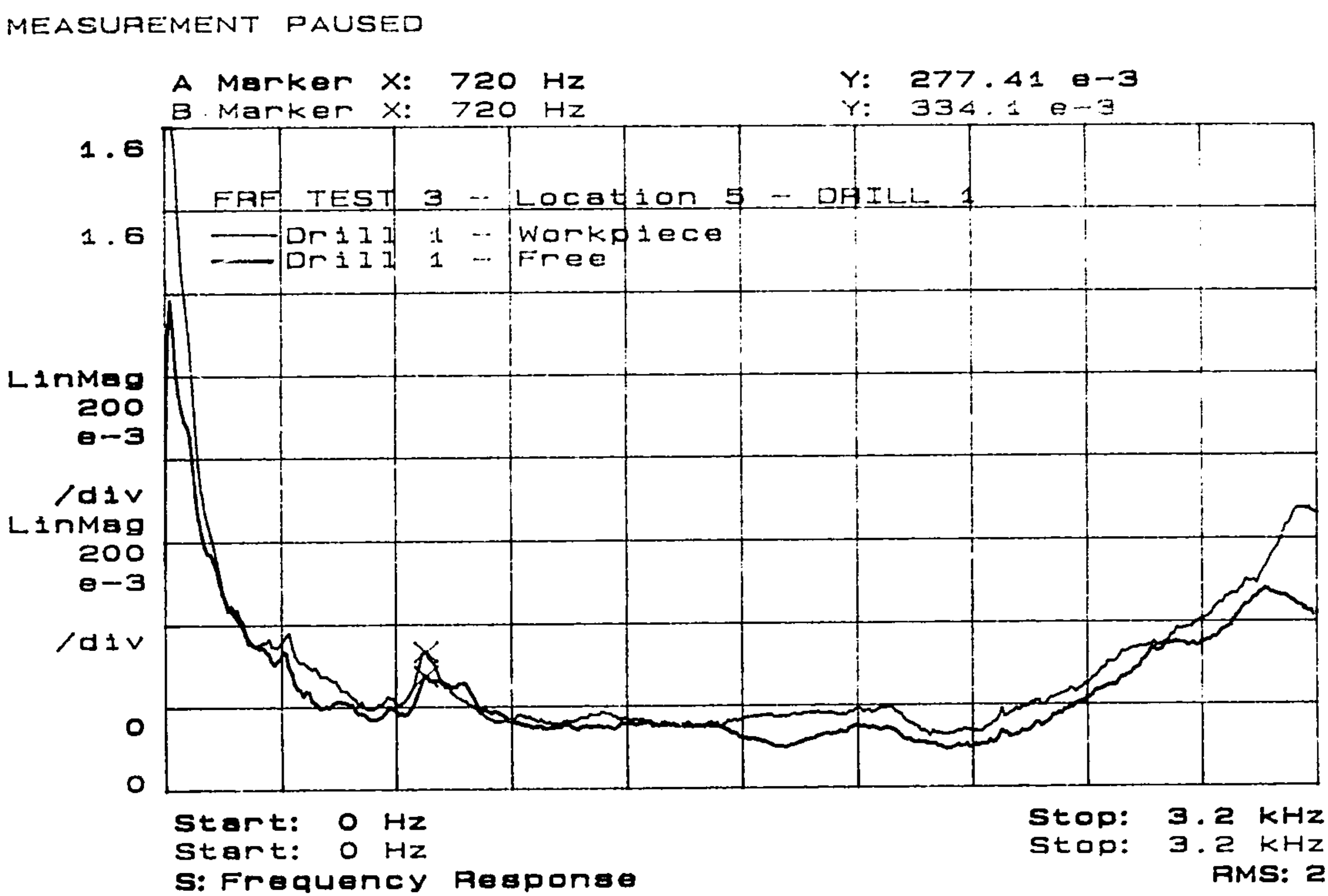



Figure 26: Machine head FRF Test 3 Accelerometer location 5- impact location 5.

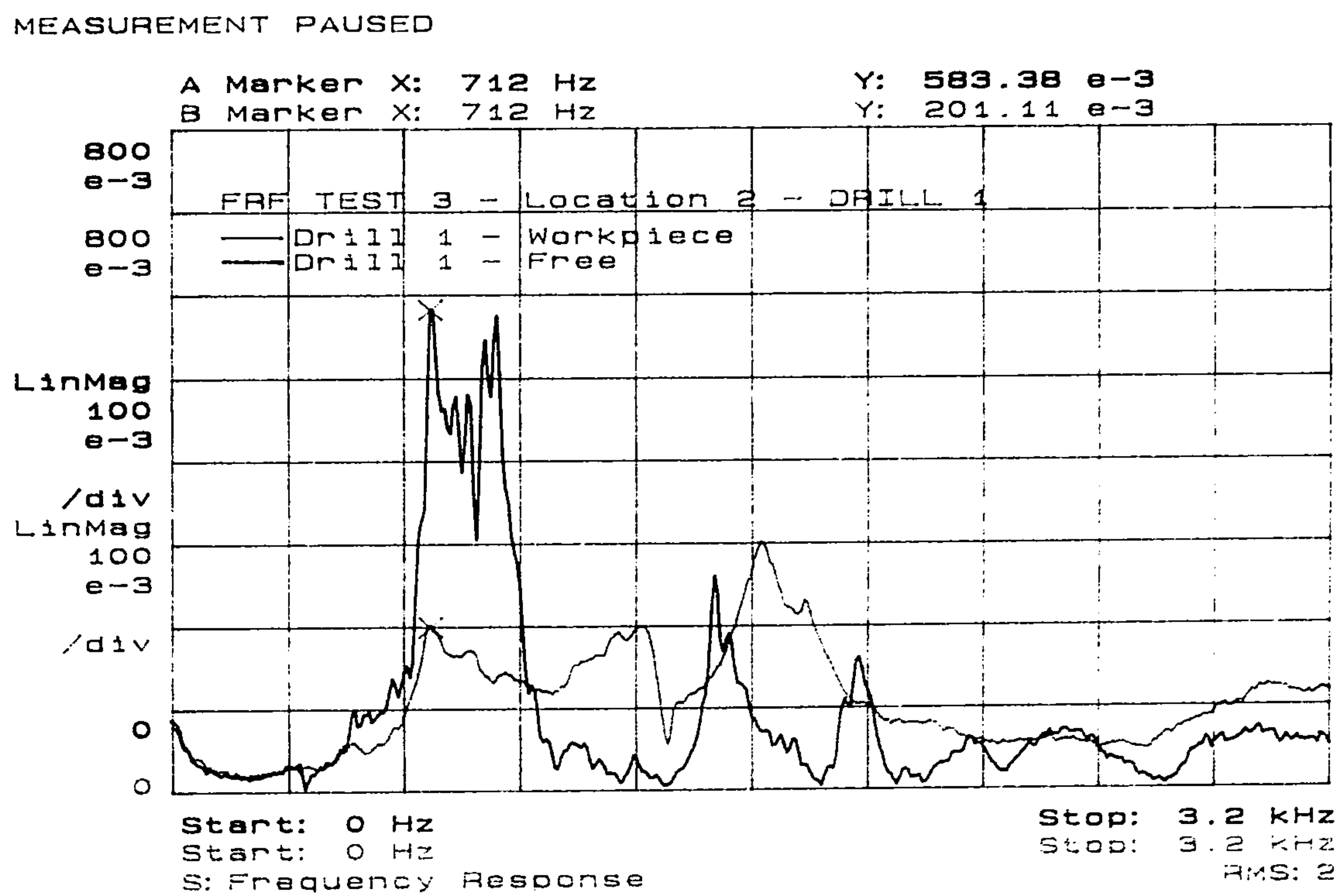



Figure 27: Machine head FRF Test 3 Accelerometer location 2- impact collet spindle.

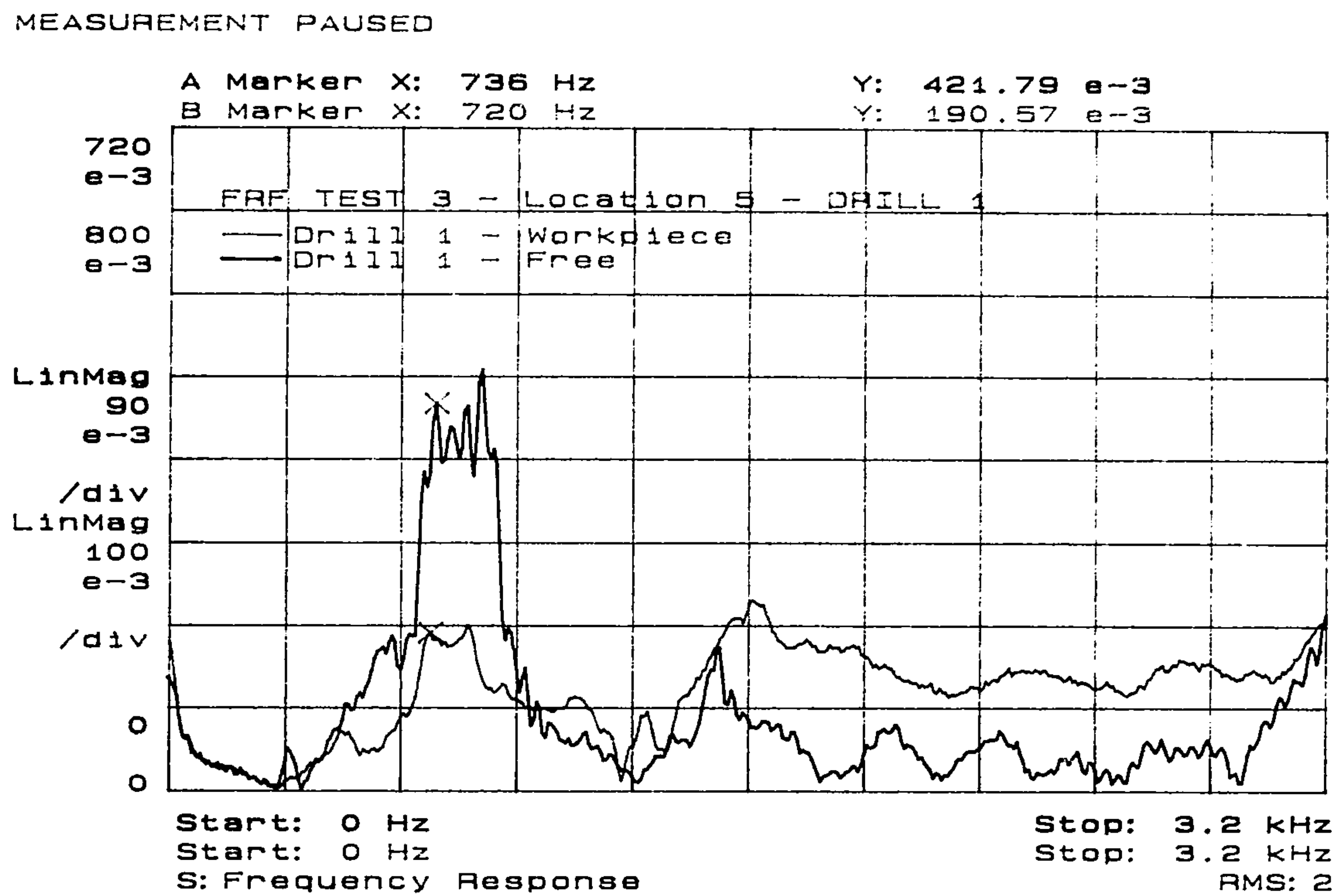



Figure 28: Machine head FRF Test 3 Accelerometer location 5-impact collet spindle 2.

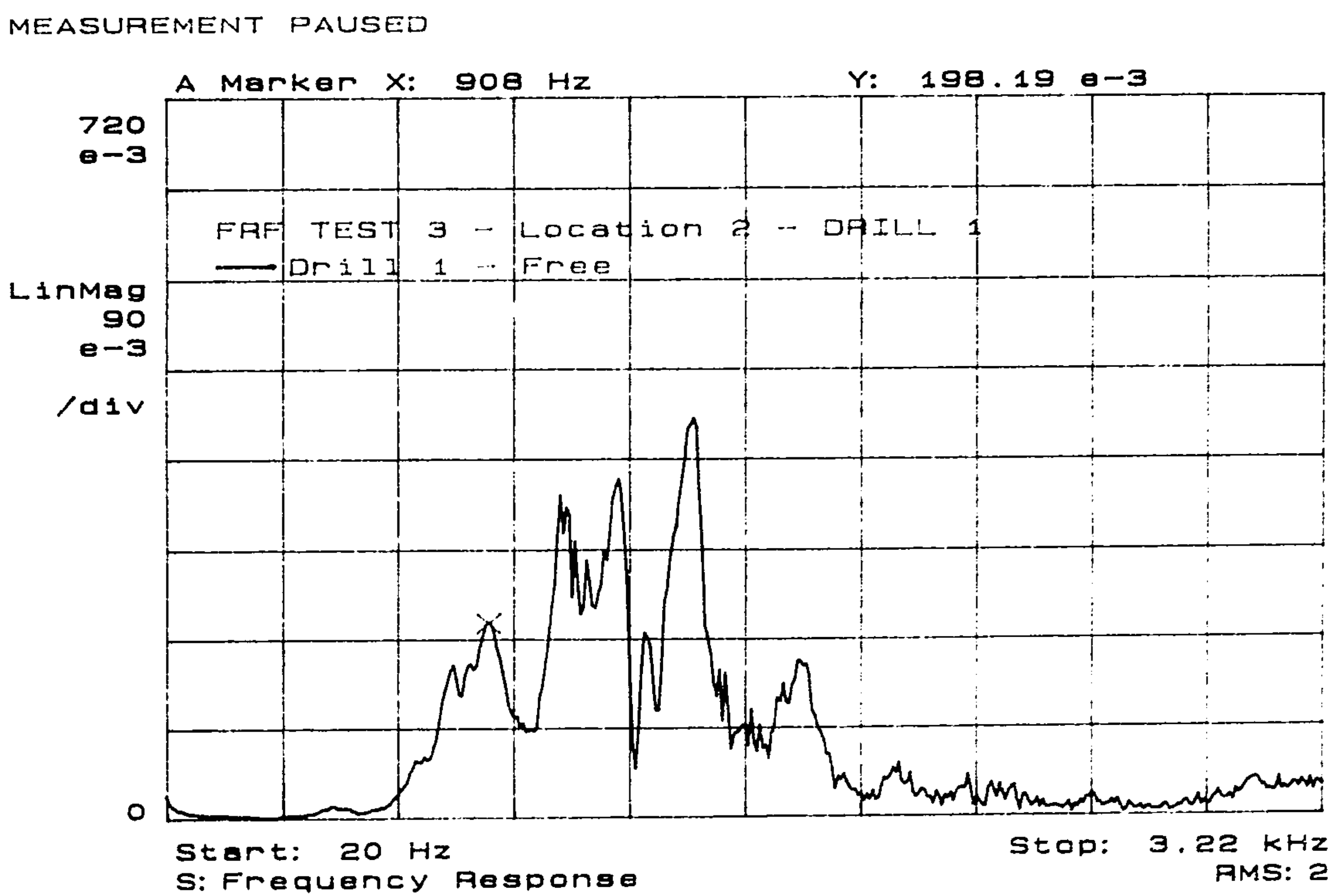



Figure 29: Machine head FRF Test 3 Accelerometer location 2- impact drill 1.

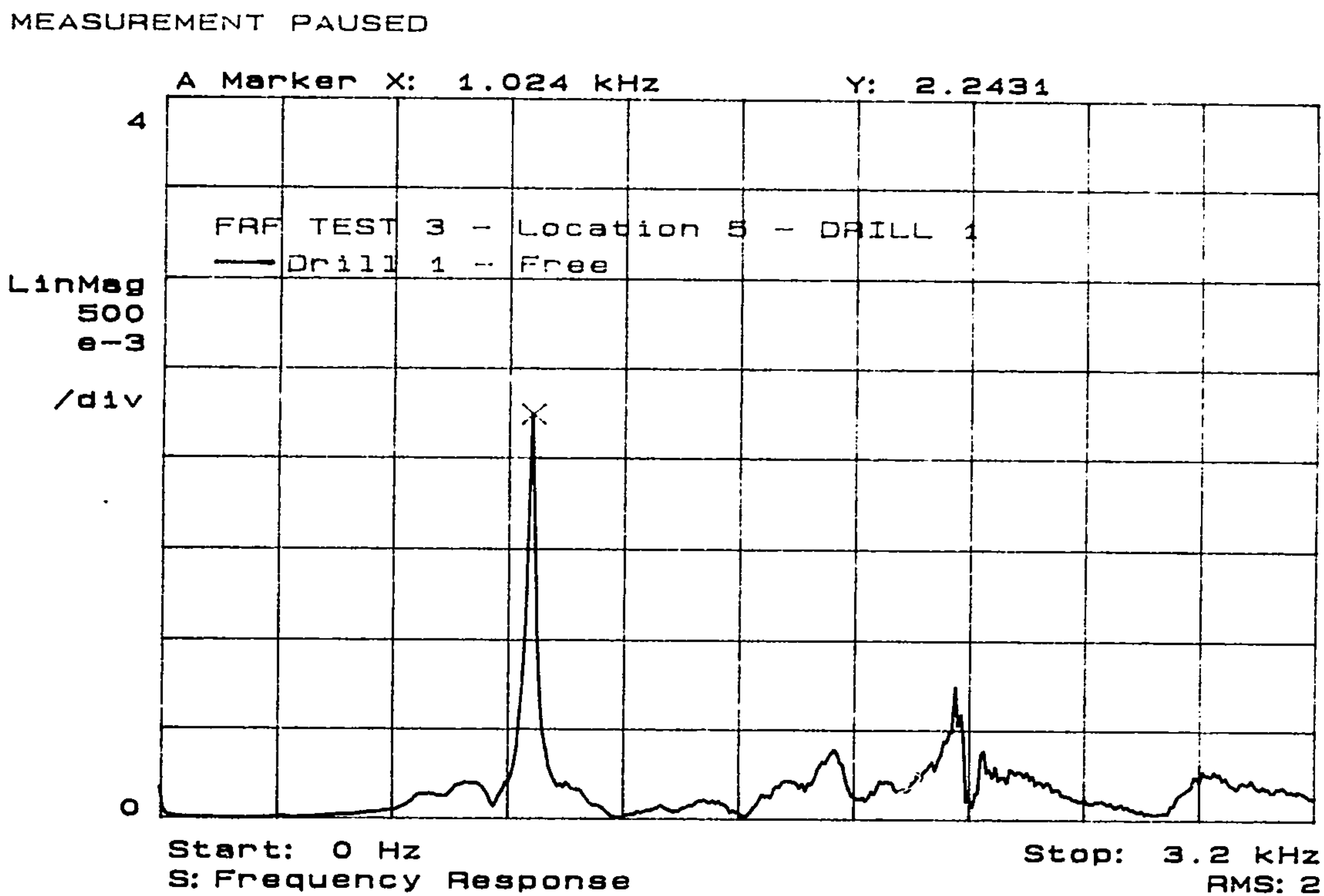



Figure 30: Machine head FRF Test 3 Accelerometer location 5- impact drill 1.

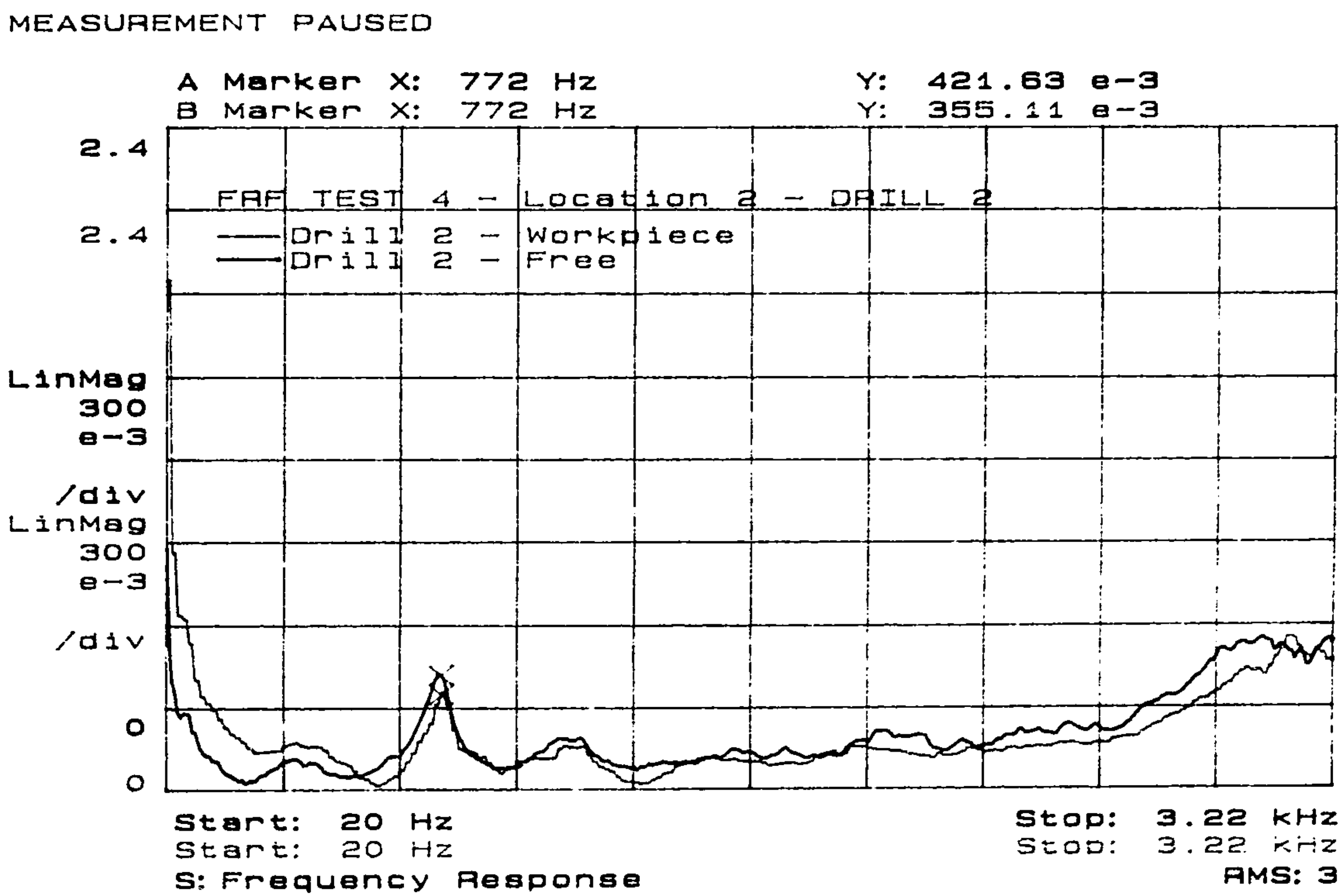



Figure 31: Machine head FRF Test 4 Accelerometer location 2- impact location 2.

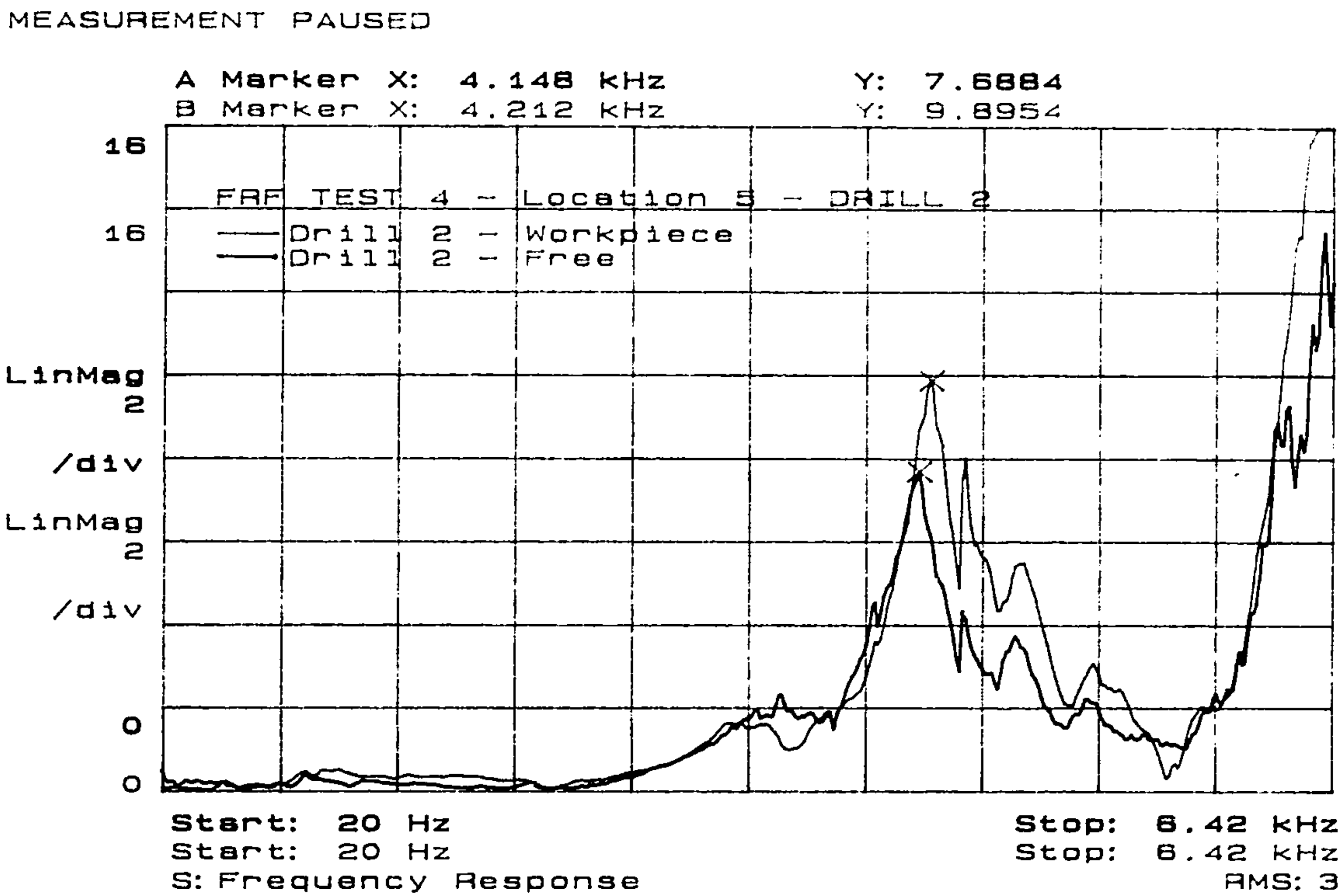



Figure 32: Machine head FRF Test 4 Accelerometer location 5- impact location 5.

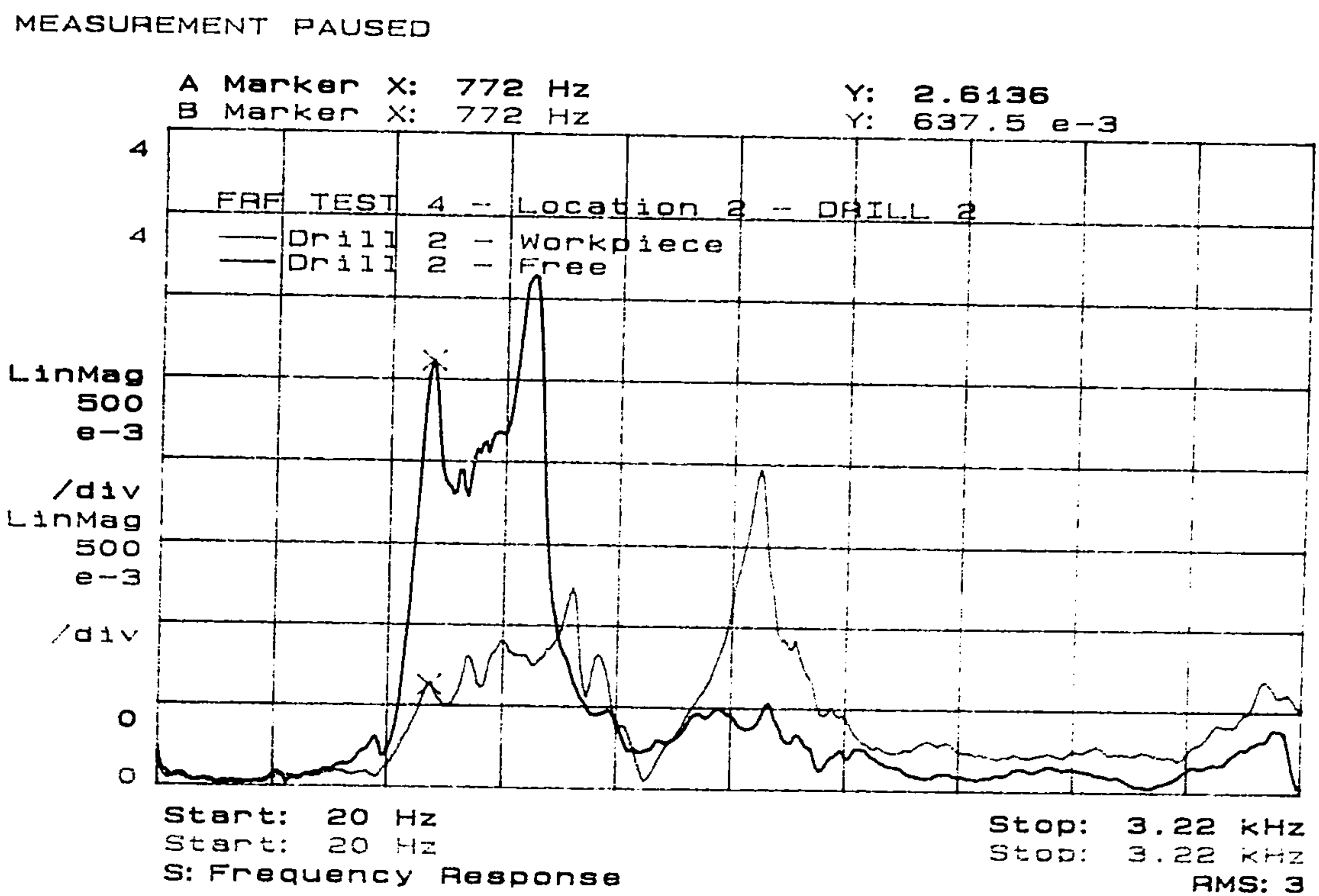



Figure 33: Machine head FRF Test 4 Accelerometer location 2-impact collet spindle 2.

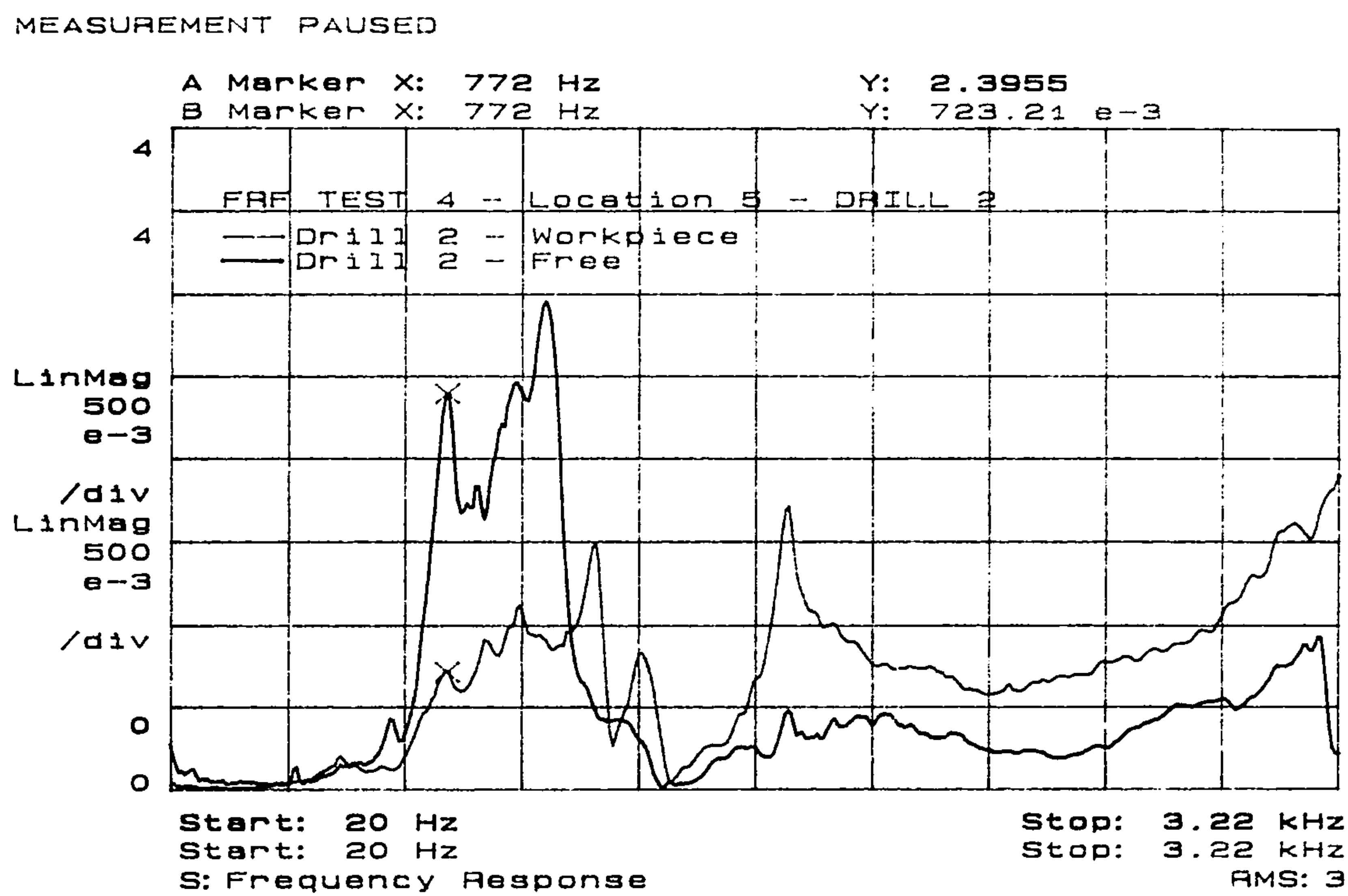



Figure 34: Machine head FRF Test 4 Accelerometer location 5-impact collet spindle 2.

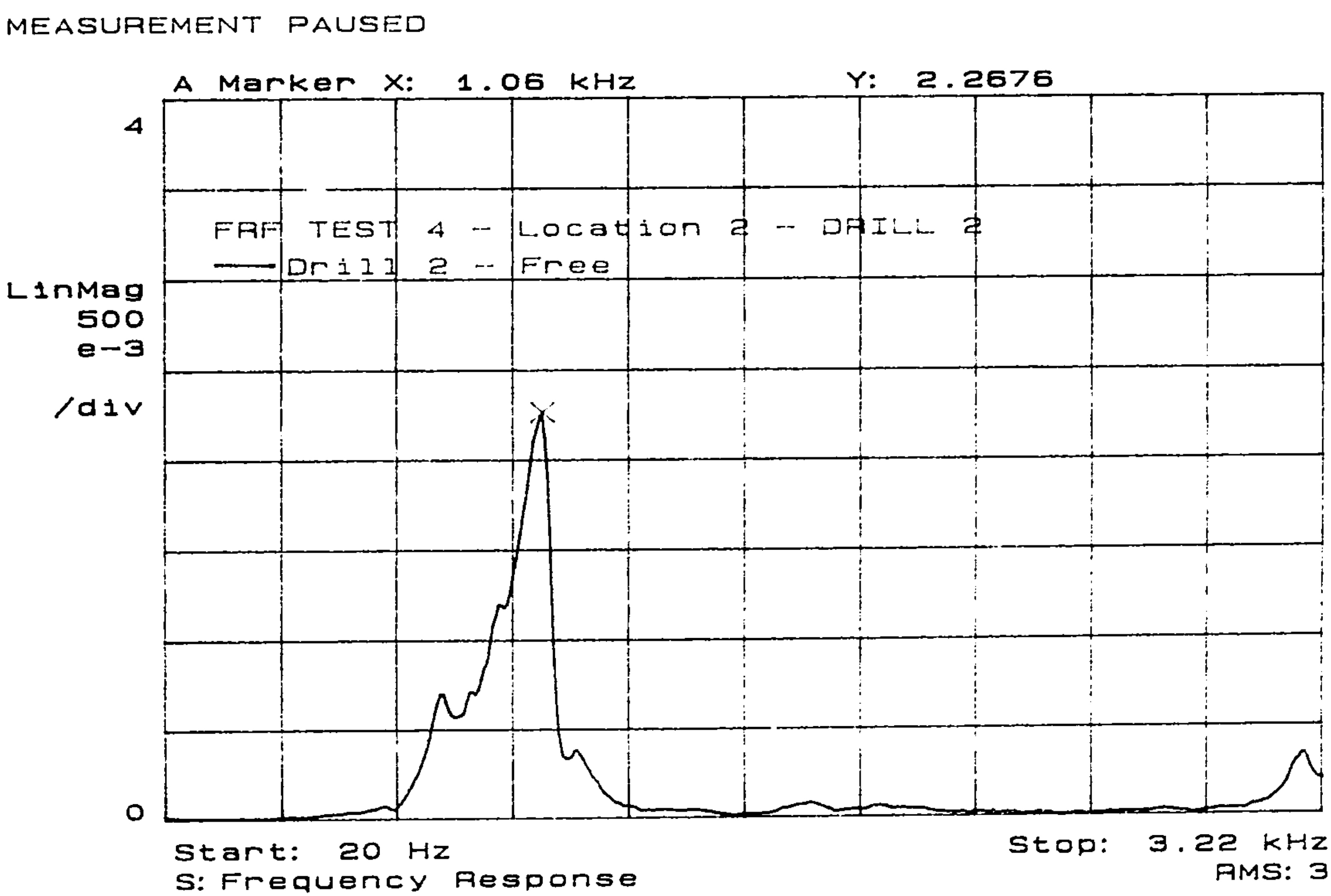



Figure 35: Machine head FRF Test 4 Accelerometer location 2- impact drill 2.

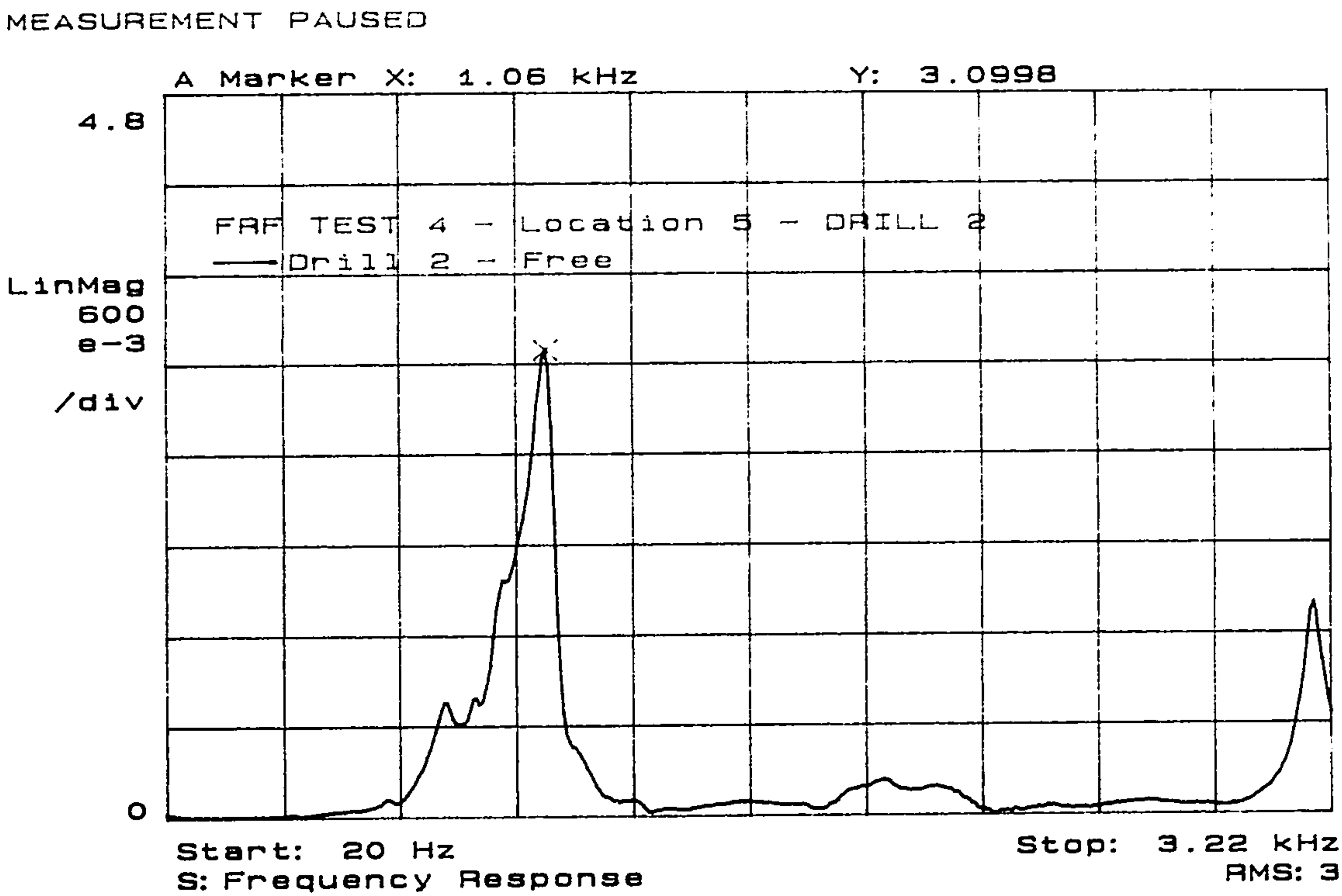



Figure 36: Machine head FRF Test 4 Accelerometer location 5- impact drill 2.

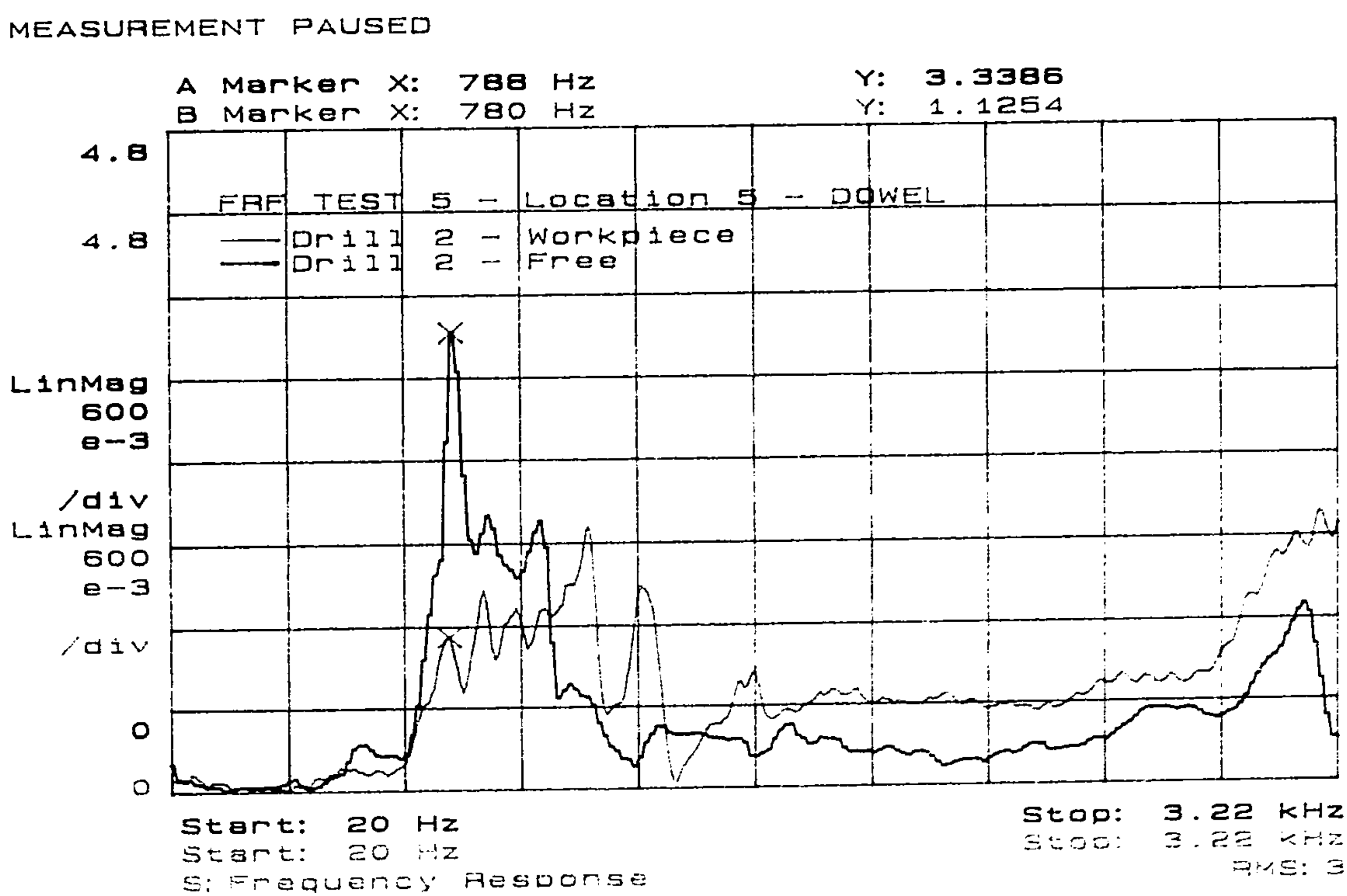



Figure 37: Machine head FRF Test 5 Accelerometer location 5-impact collet spindle 2.

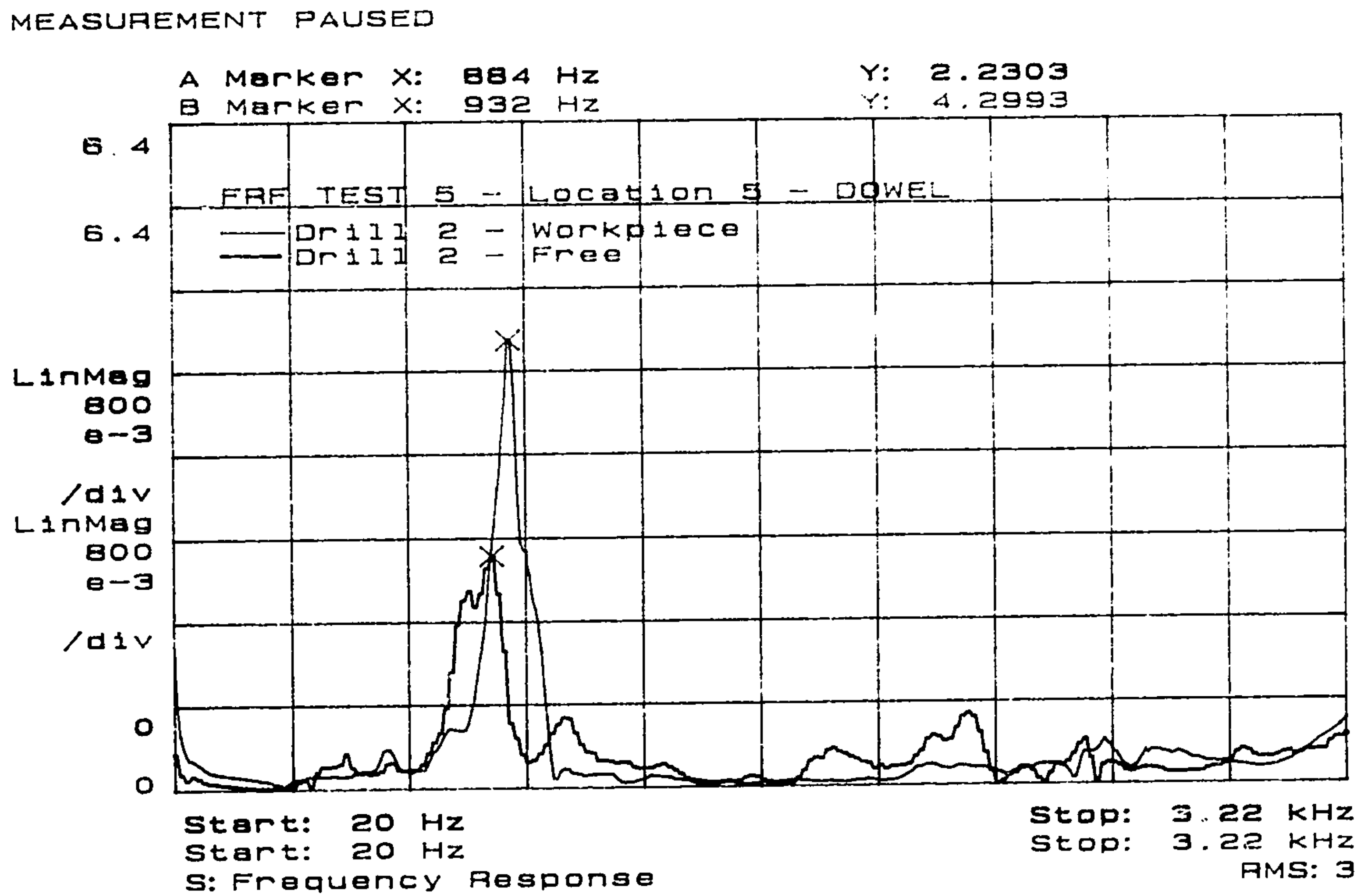



Figure 38: Machine head FRF Test 5 Accelerometer location 5-impact collet spindle 1.

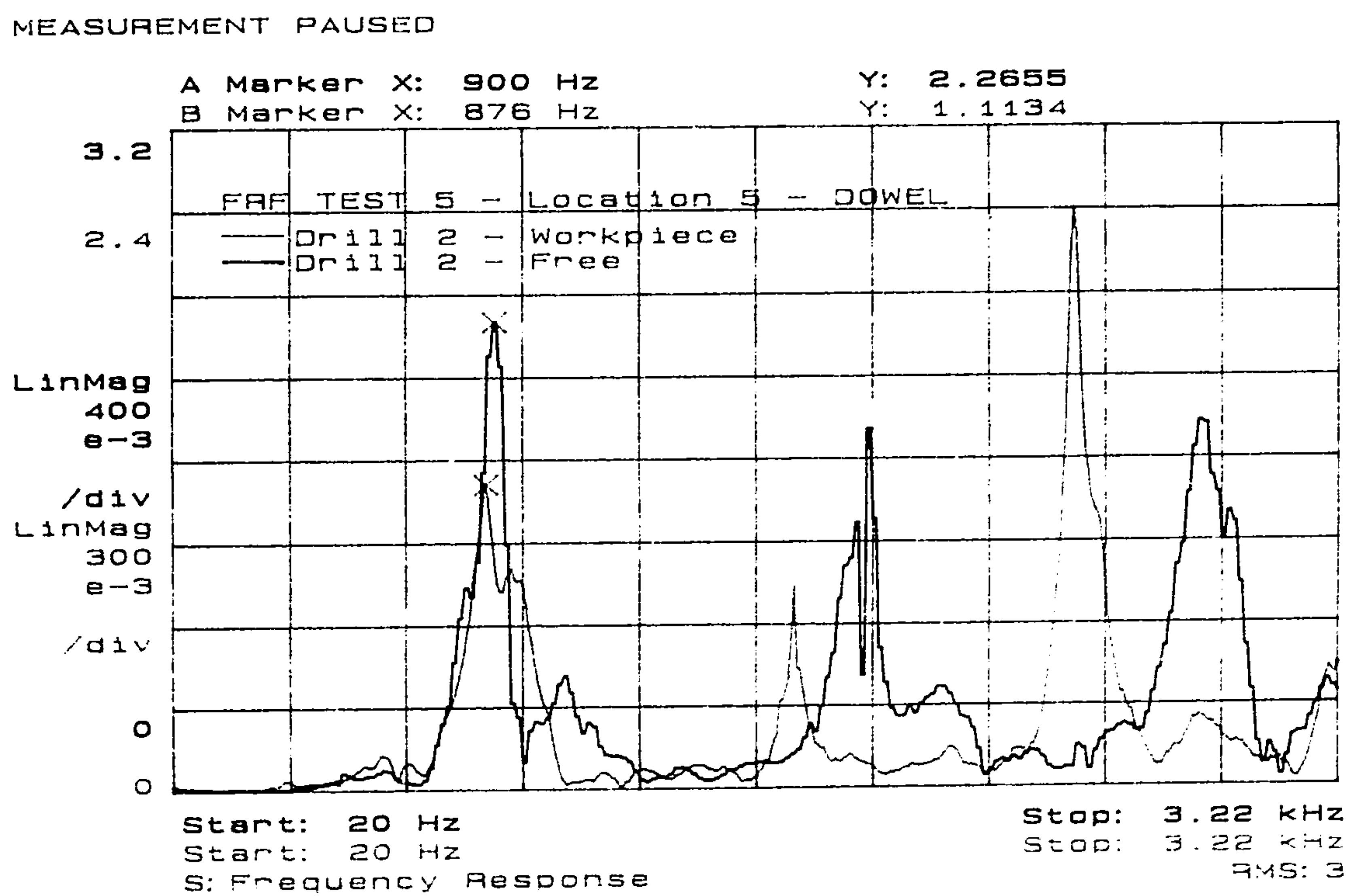



Figure 39: Machine head FRF Test 5 Accelerometer location 5- impact dowel.

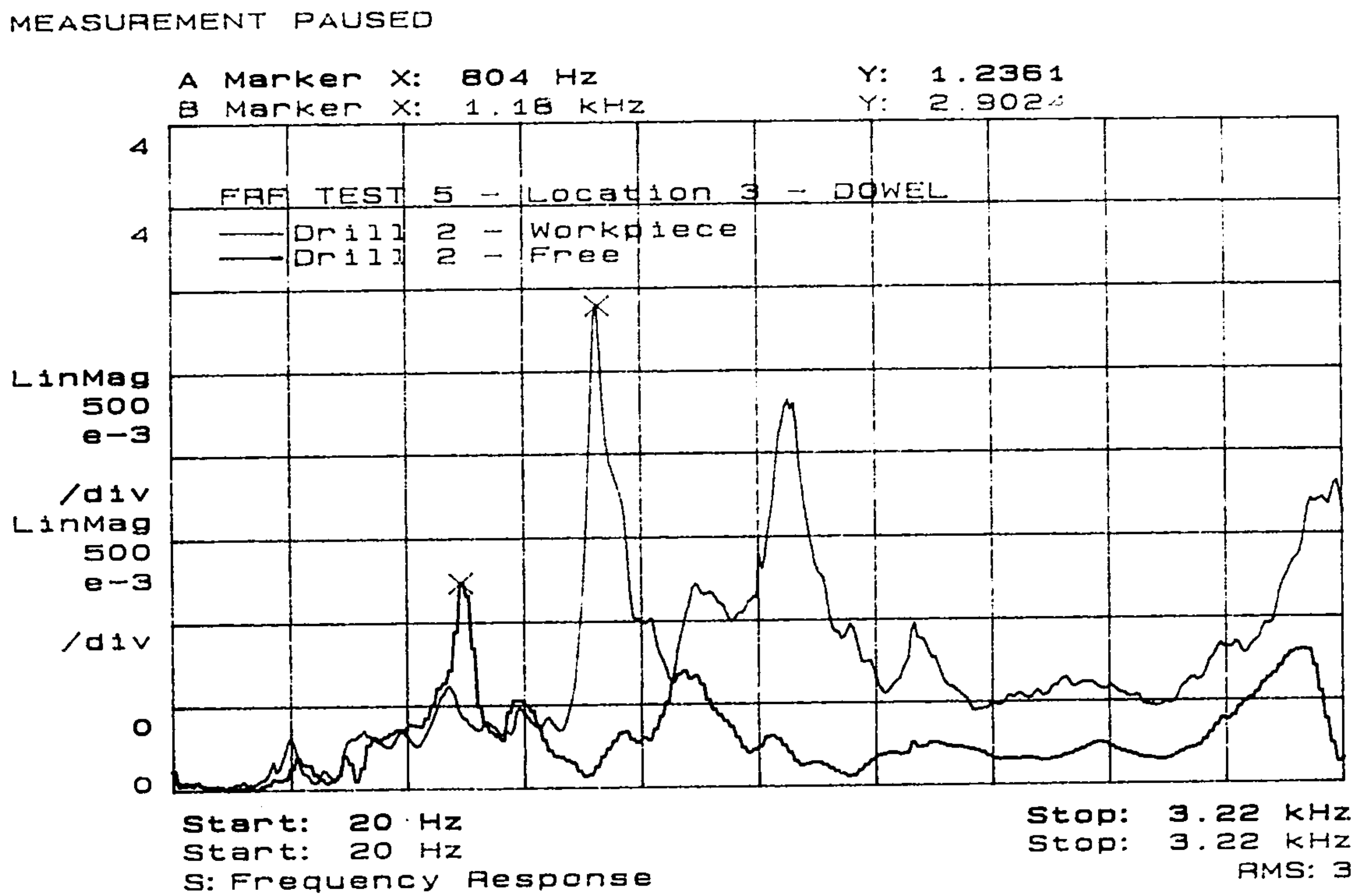



**Figure 40:** Machine head FRF Test 5 Accelerometer location 3-impact collet spindle 2.

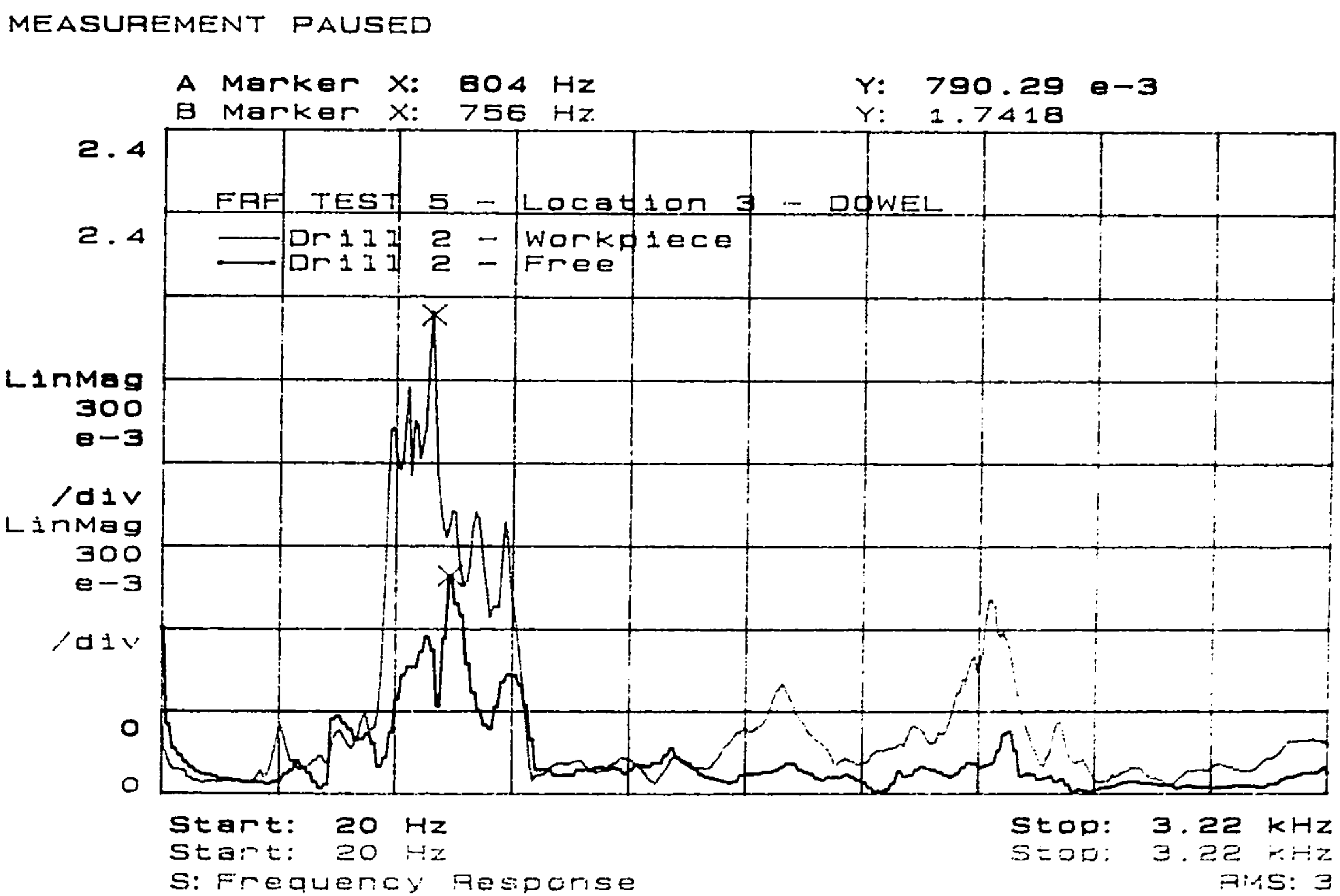



Figure 41: Machine head FRF Test 5 Accelerometer location 3-impact collet spindle 1.

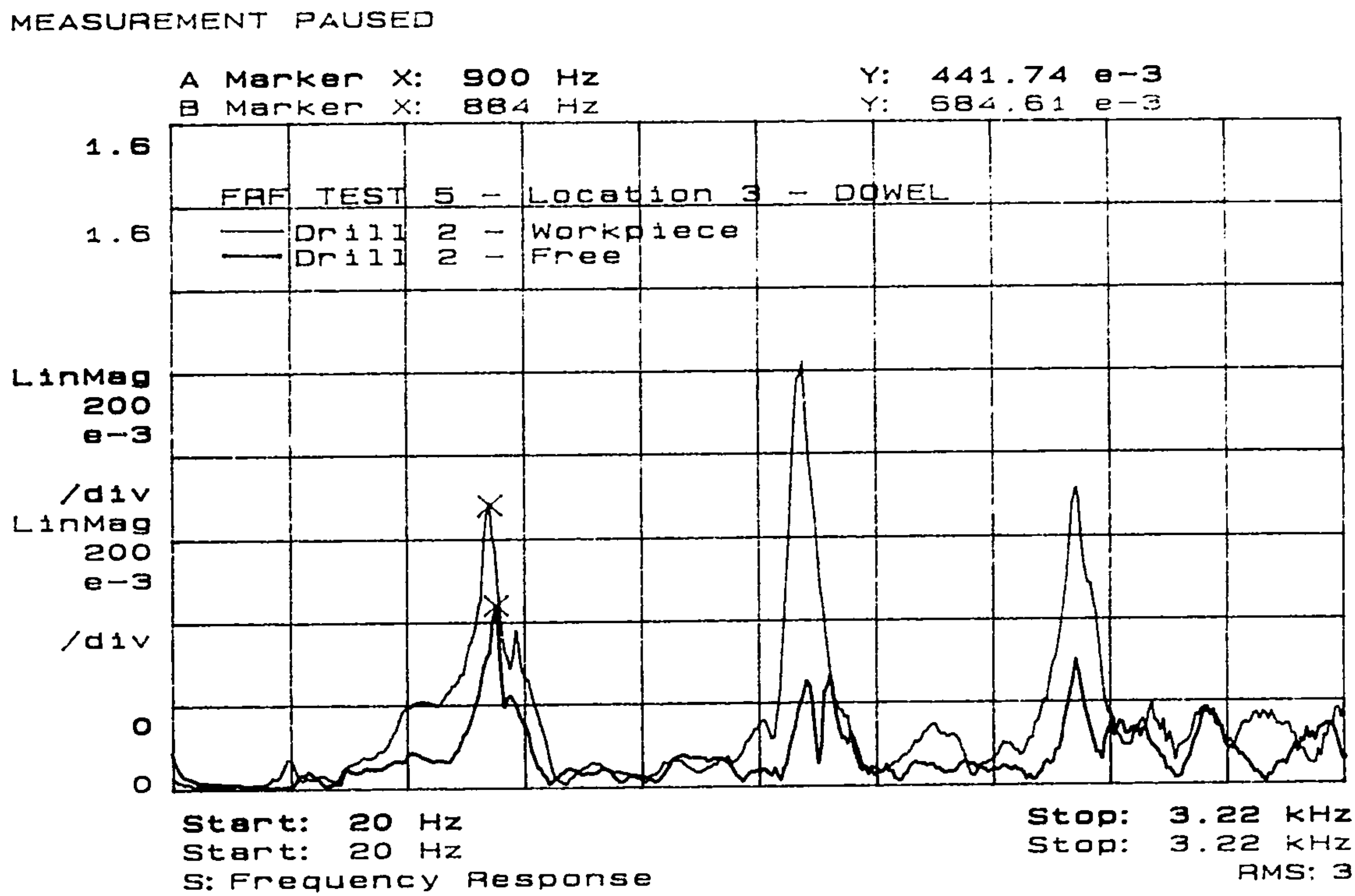



Figure 42: Machine head FRF Test 5 Accelerometer location 3-impact dowel.

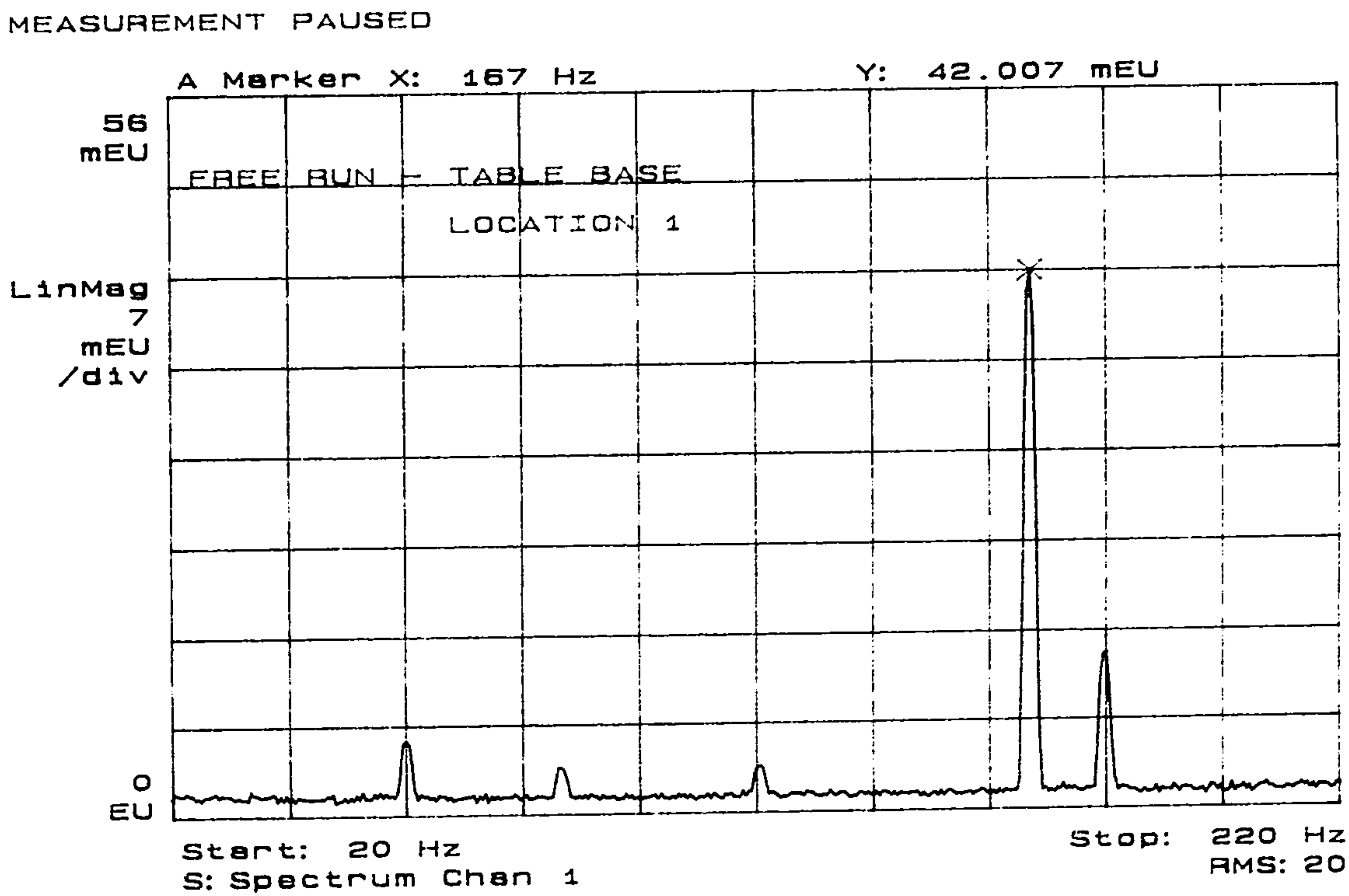



Figure 43: Machine head free-run Accelerometer location 1: Table base 200 Hz.

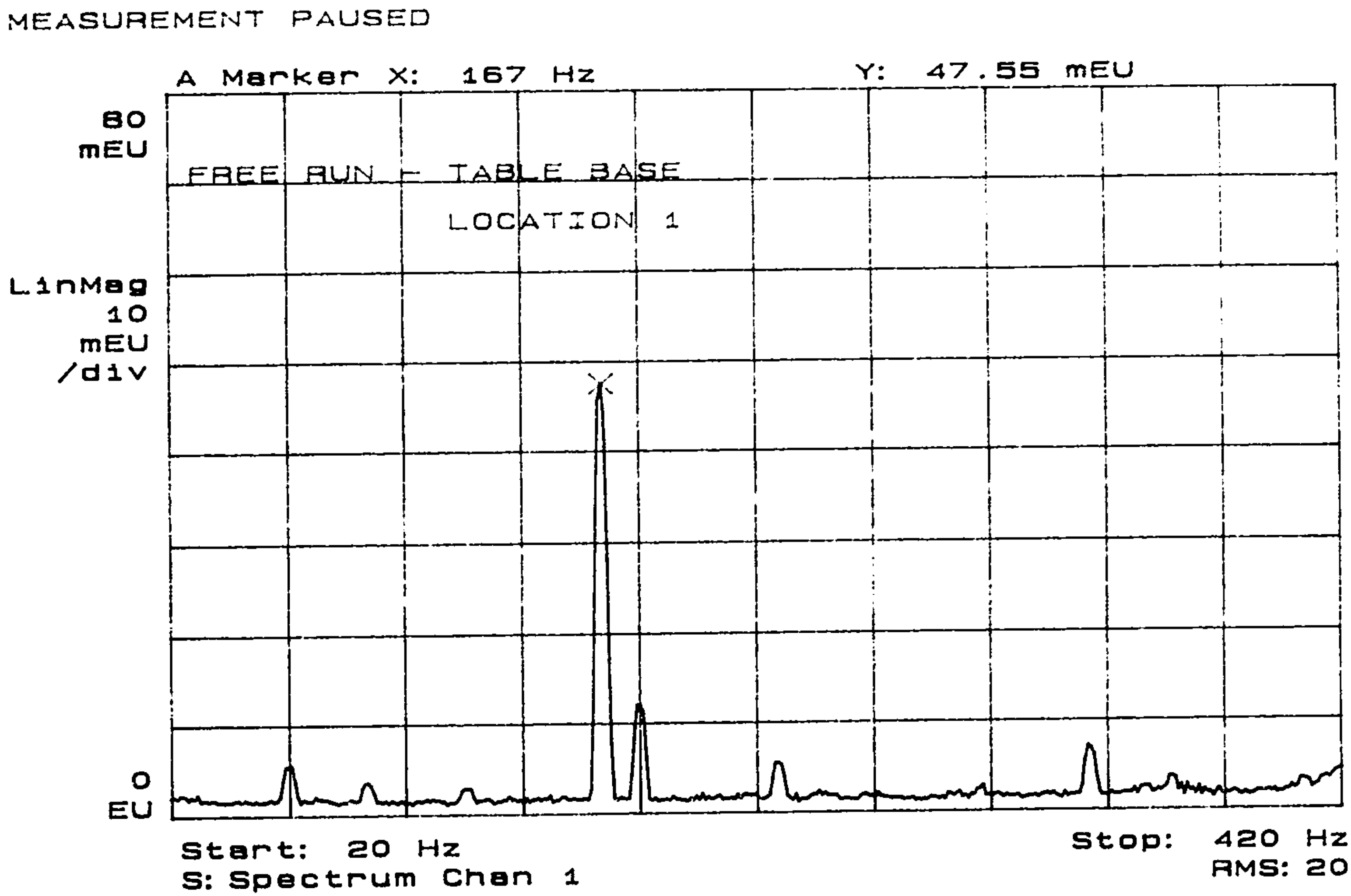



Figure 44: Machine head free-run- Accelerometer location 1: Table base 400 Hz.

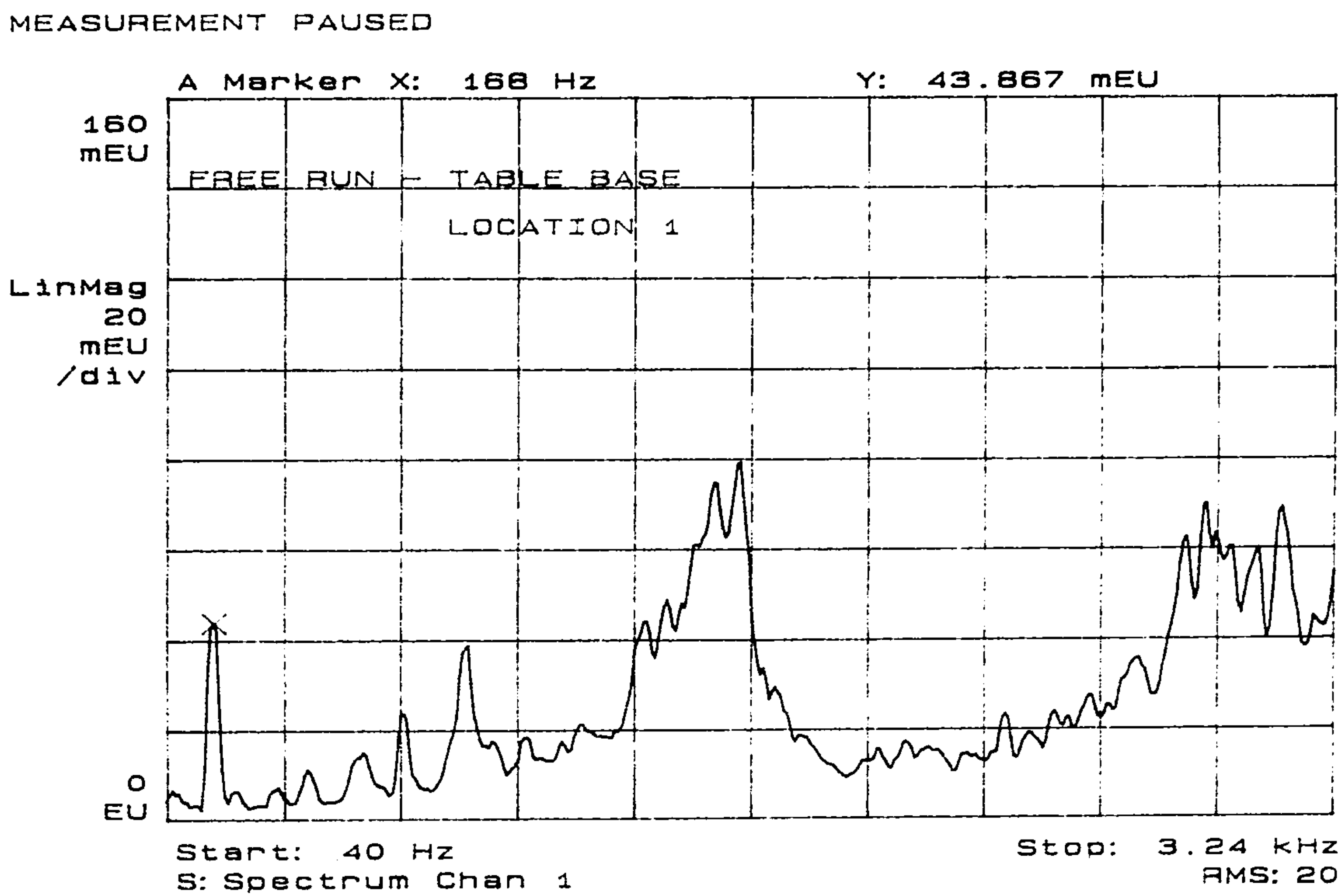



Figure 45: Machine head free-run- Accelerometer location 1: Table base 3200 Hz.

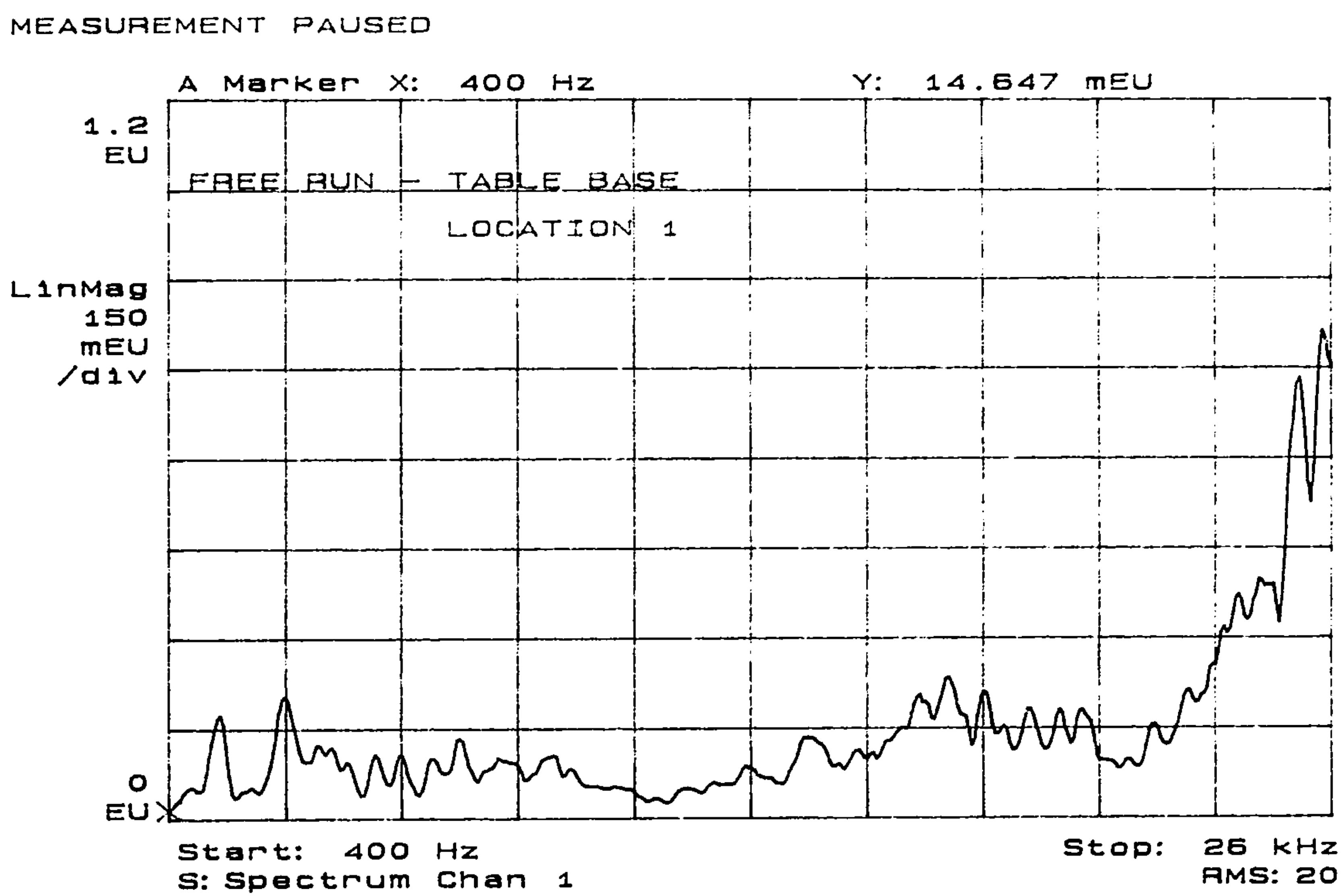



Figure 46: Machine head free-run- Accelerometer location 1: Table base 25600 Hz.

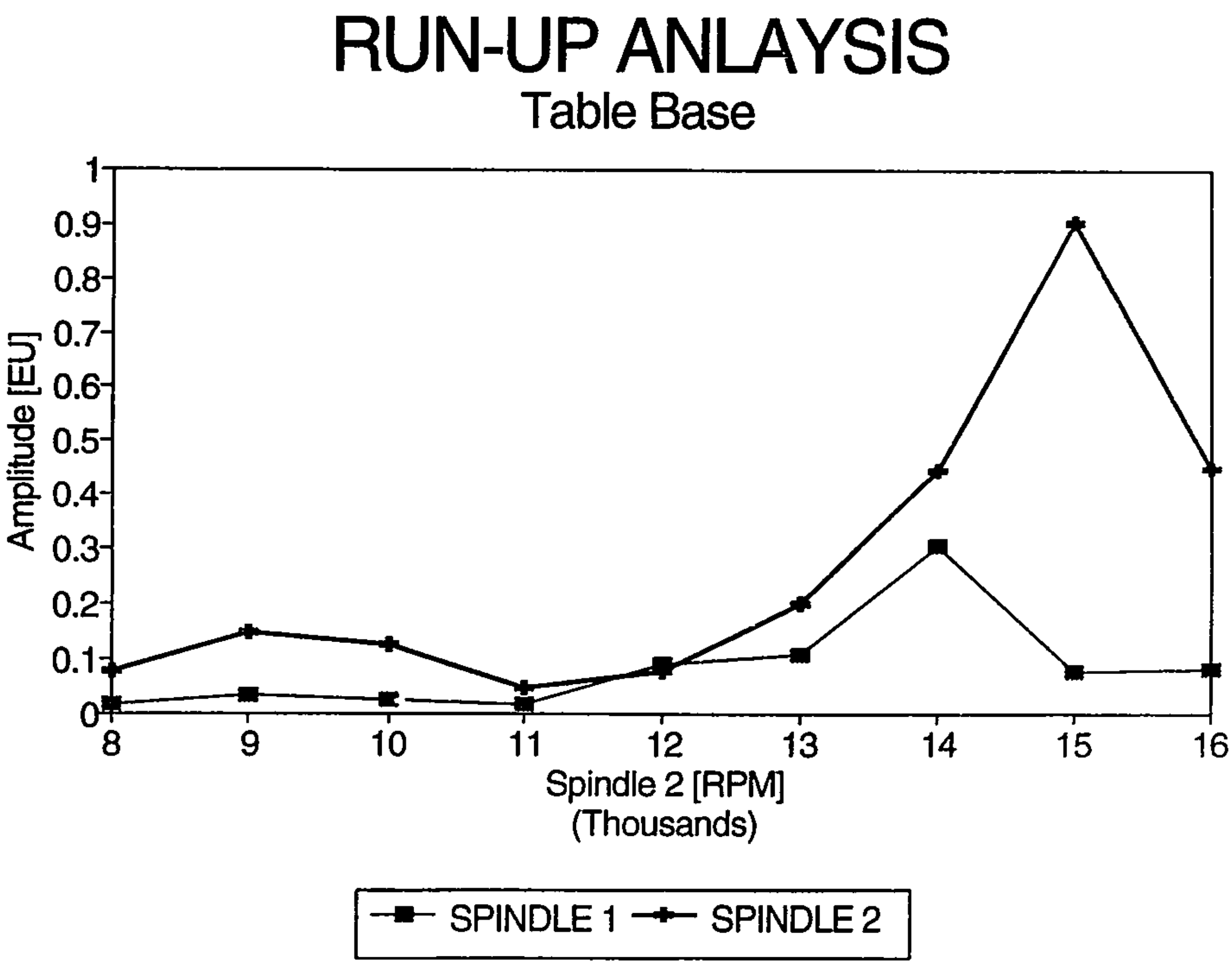



Figure 47 : Run-up analysis - table base.

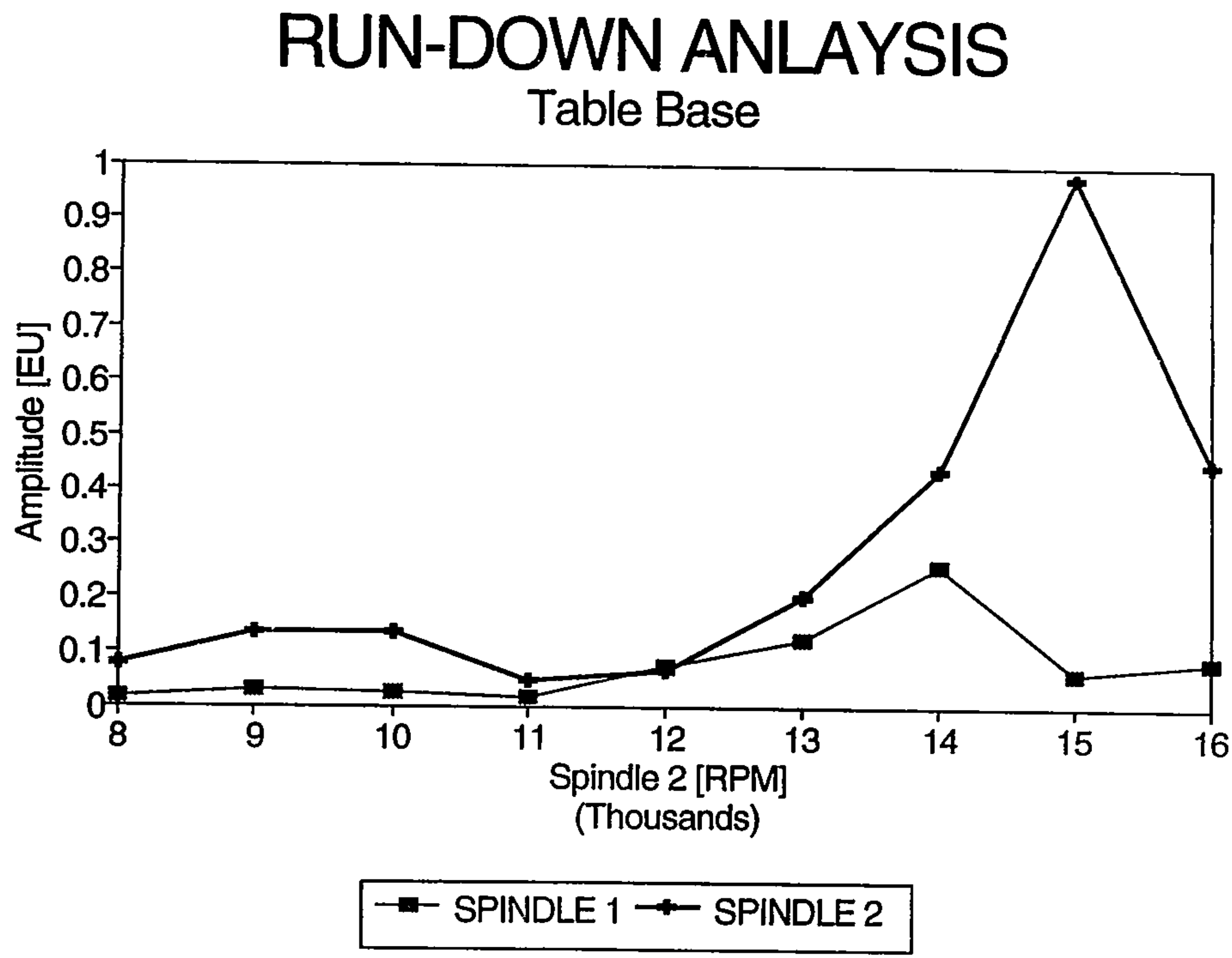



Figure 48 : Run-down analysis - table base.

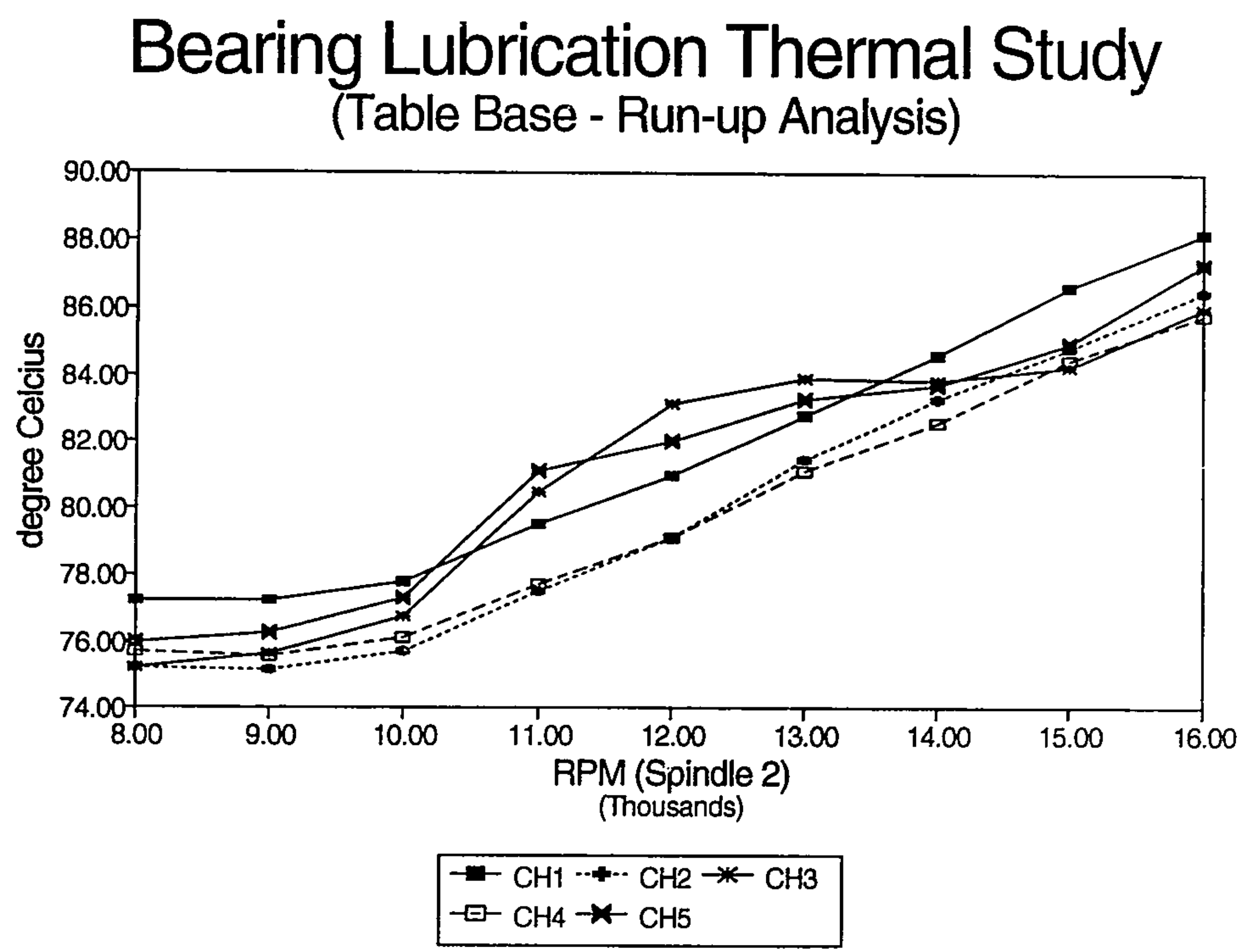



Figure 49 : Thermal properties of bearing lubrication - Table base run-up analysis.

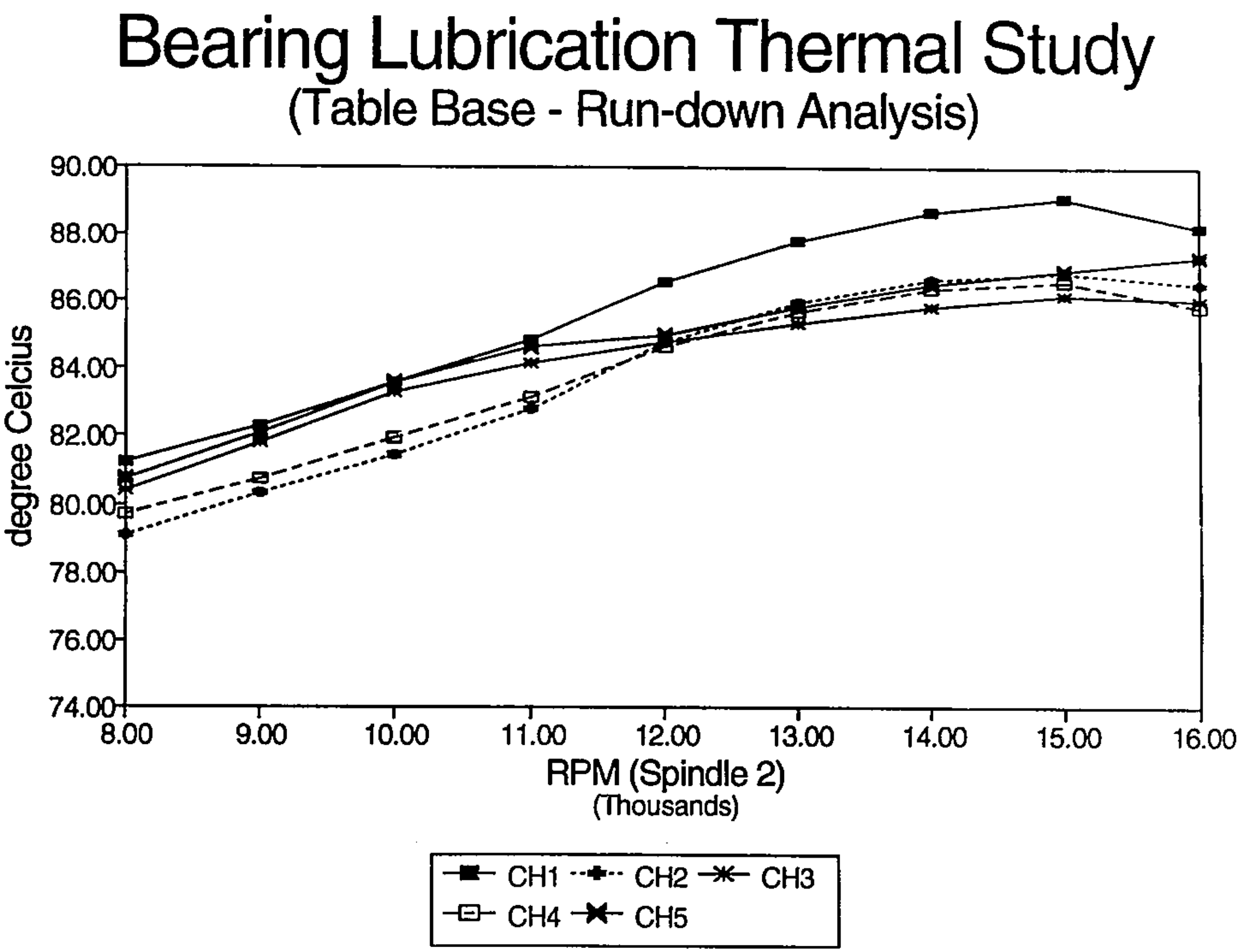



Figure 50 : Thermal properties of bearing lubrication - Table base run-down analysis.

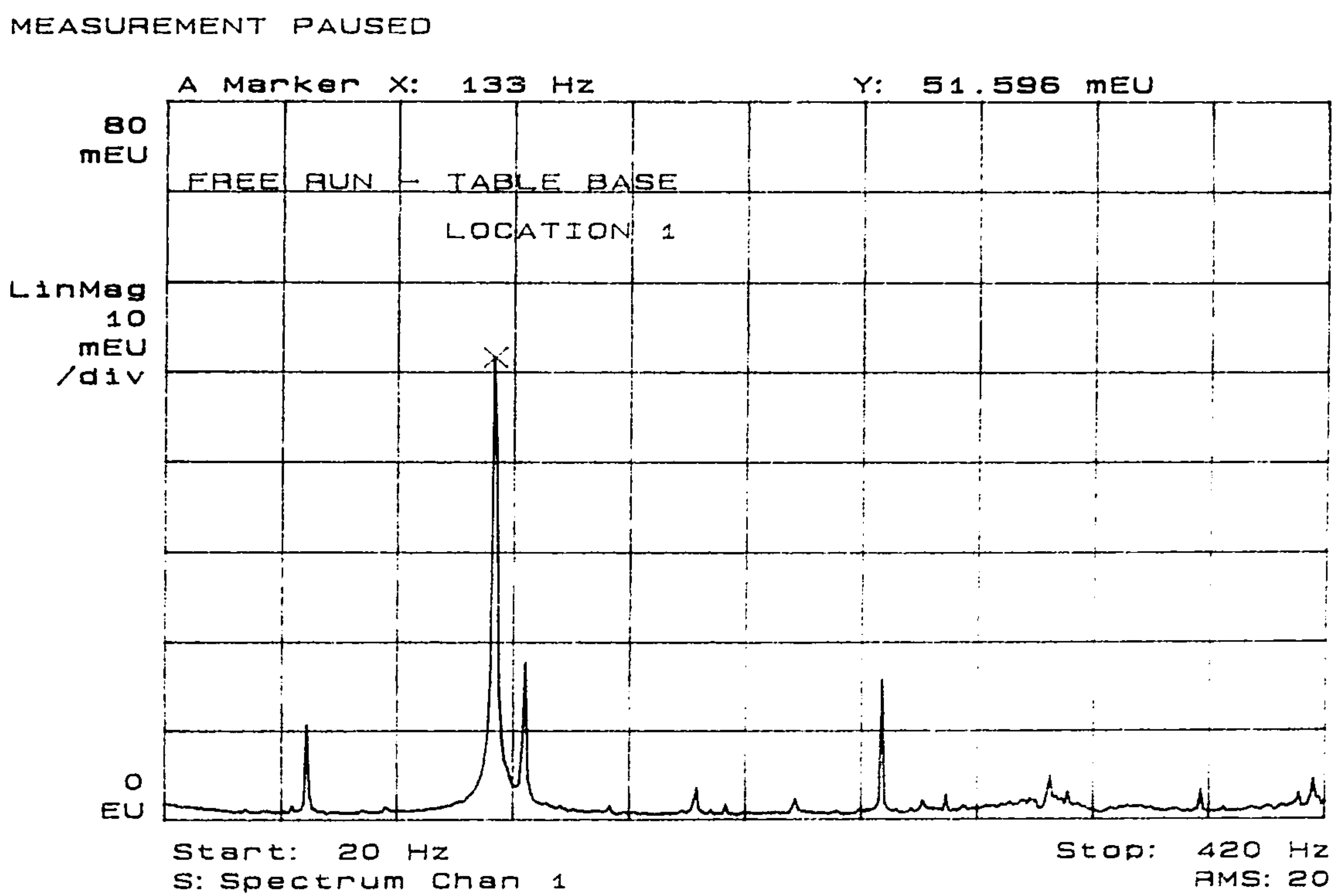



Figure 51: Machine head run-up analysis- table base (8000 RPM): 400 Hz.

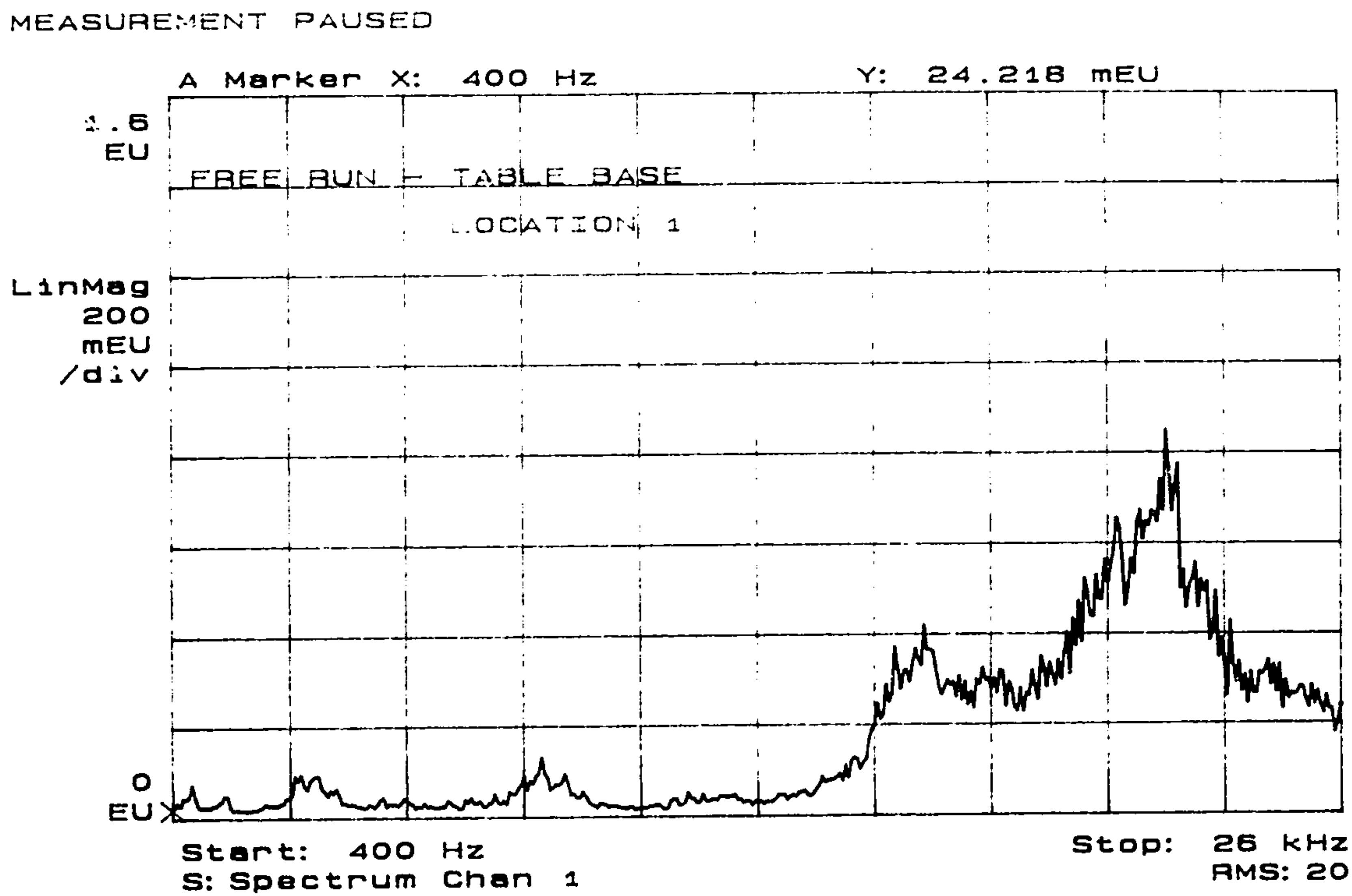



Figure 52: Machine head run-up analysis- table base (8000 RPM): 25 600 Hz.

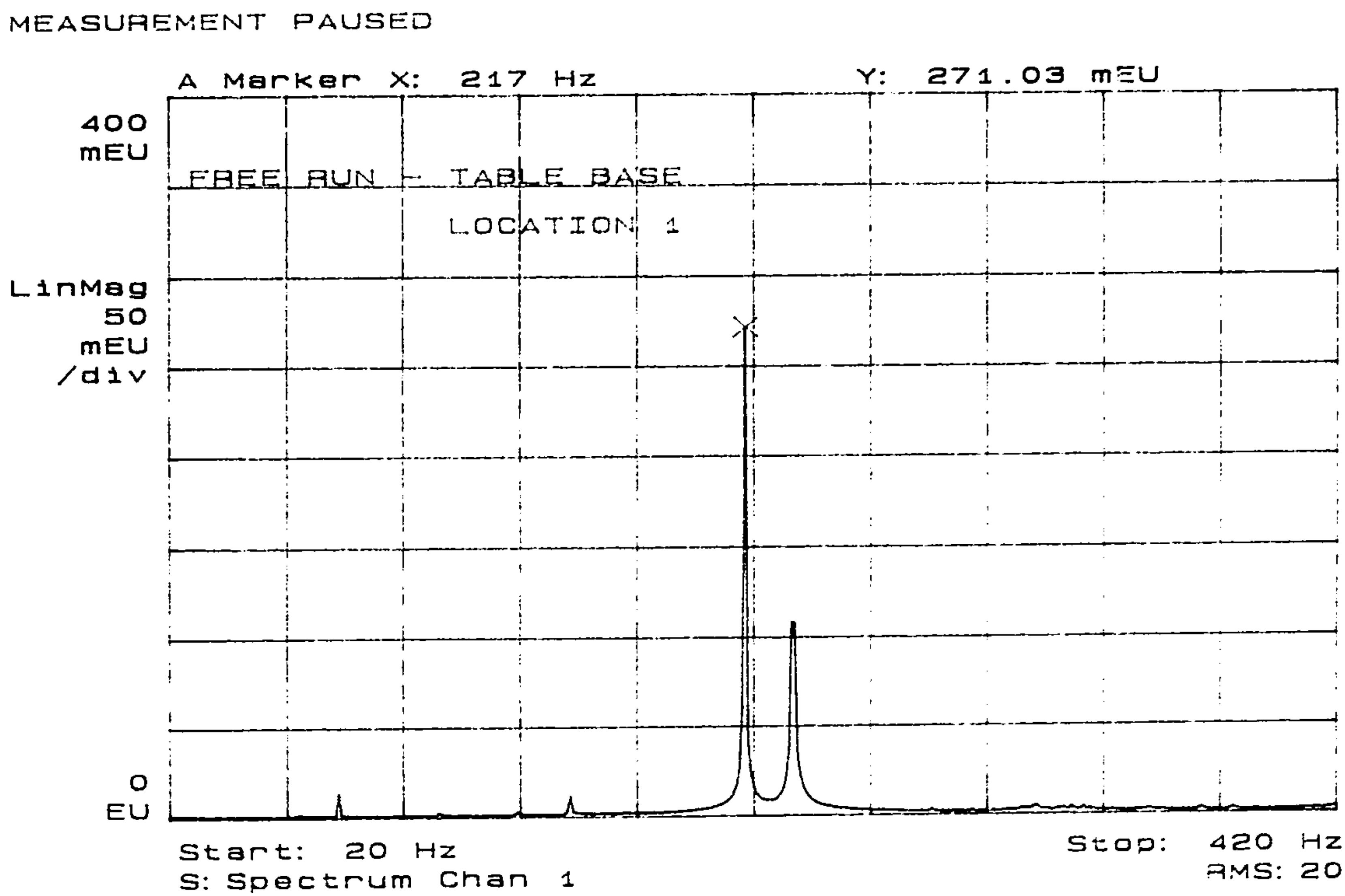



Figure 53: Machine head run-up analysis- table base (13000 RPM): 400 Hz.

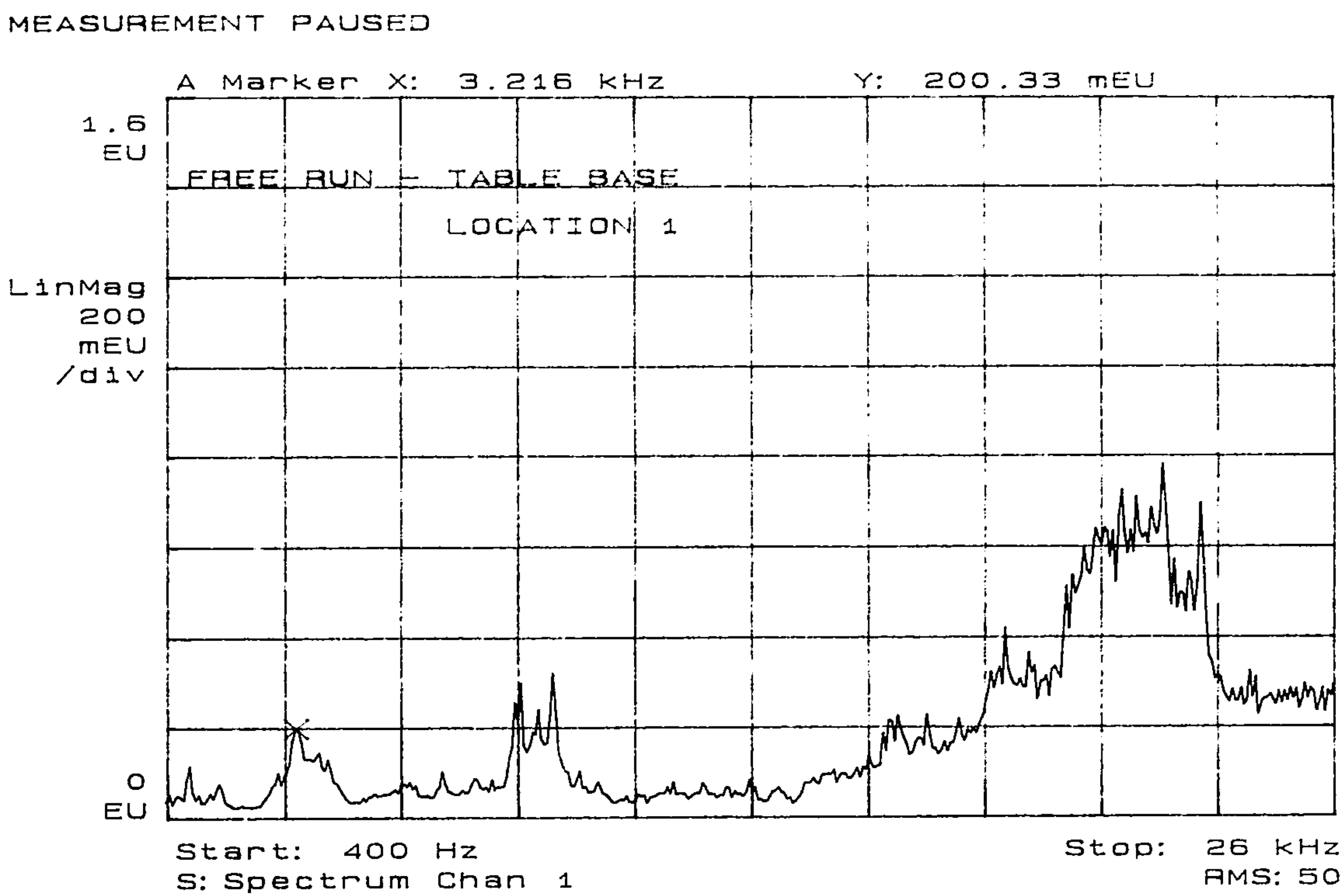



**Figure 54: Machine head run-up analysis- table base (13000 RPM):** 25 600 Hz.

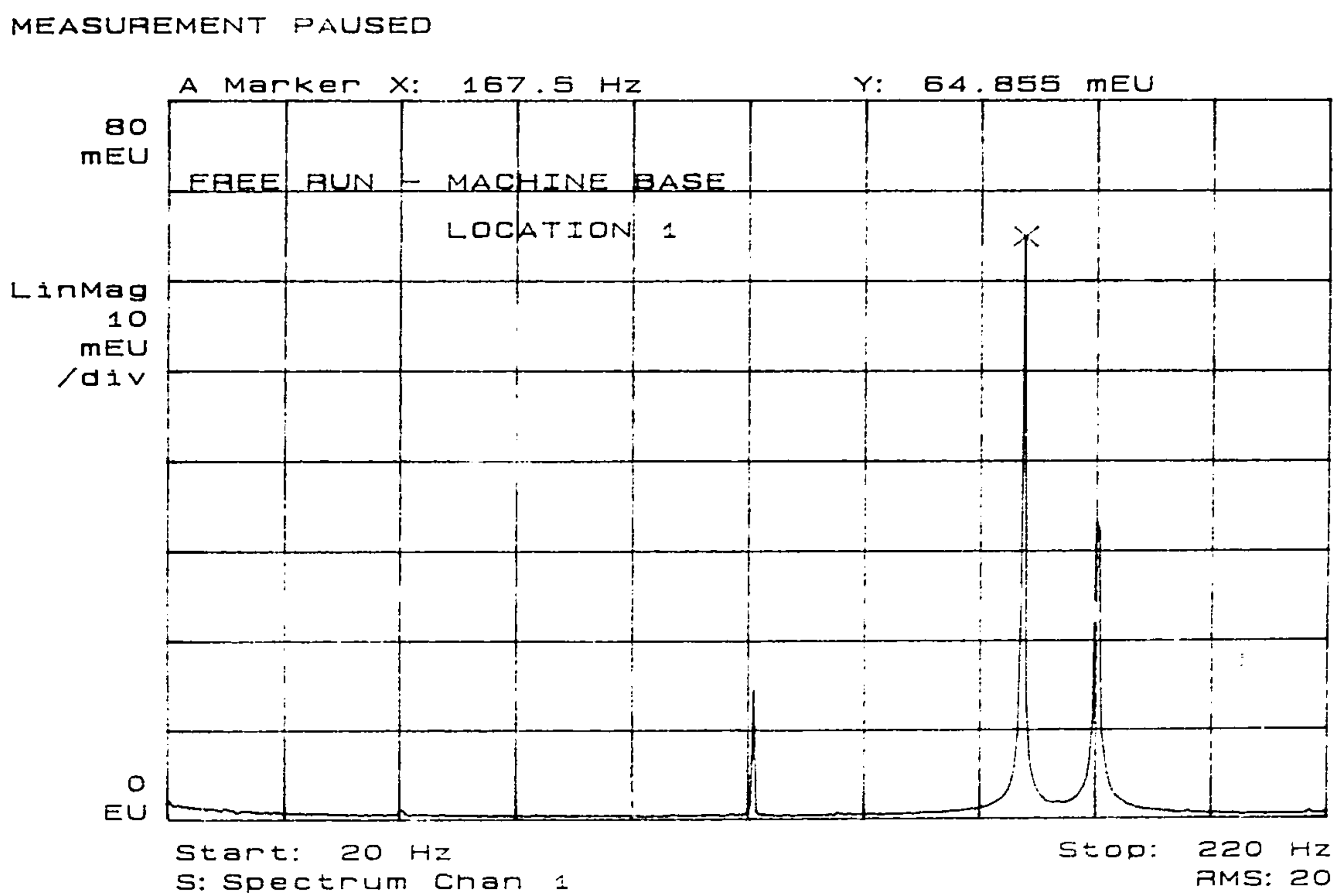



Figure 55: Machine head free-run- Accelerometer location 1: Machine base 200 Hz.

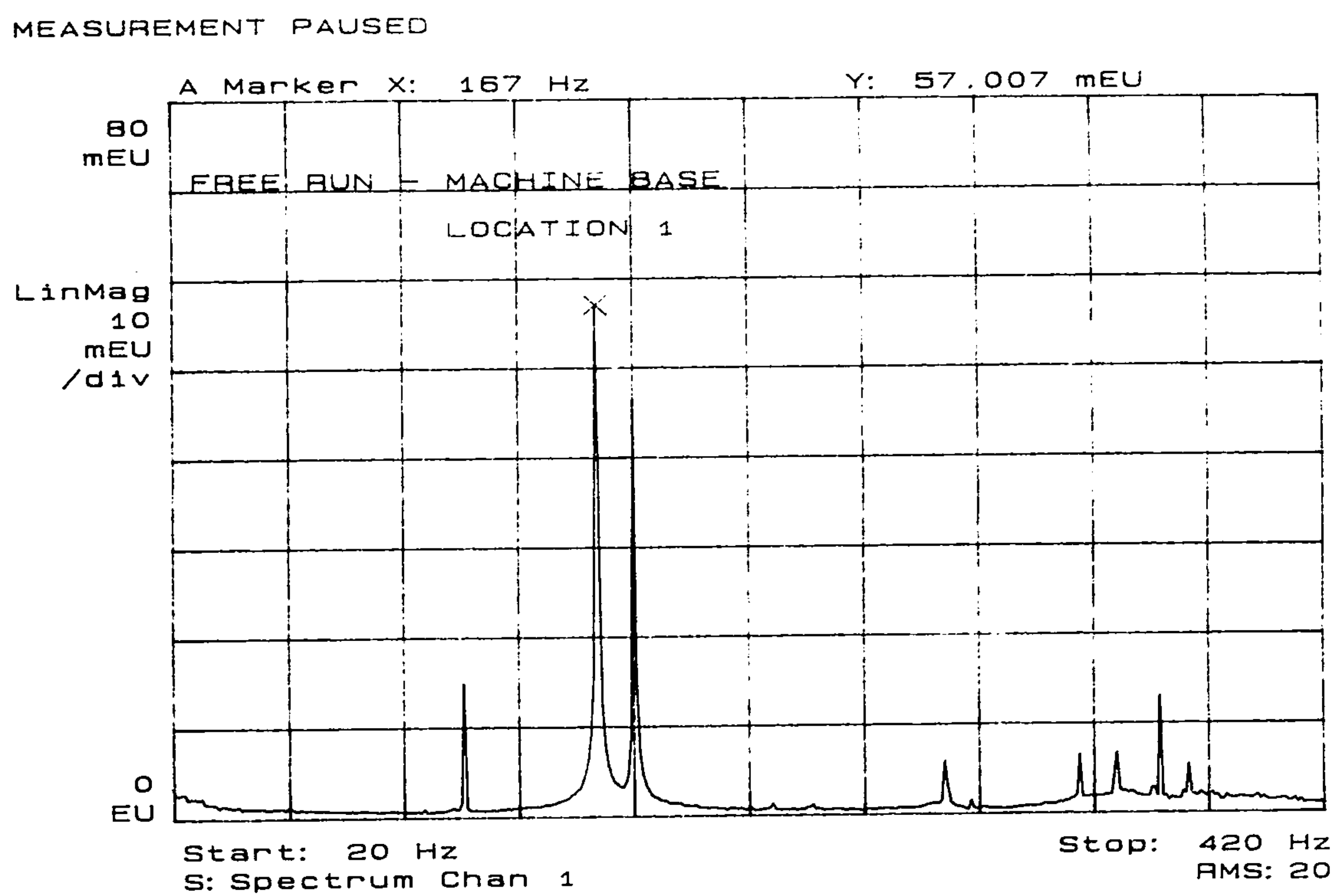



**Figure 56:** Machine head free-run- Accelerometer location 1: Machine base 400 Hz.

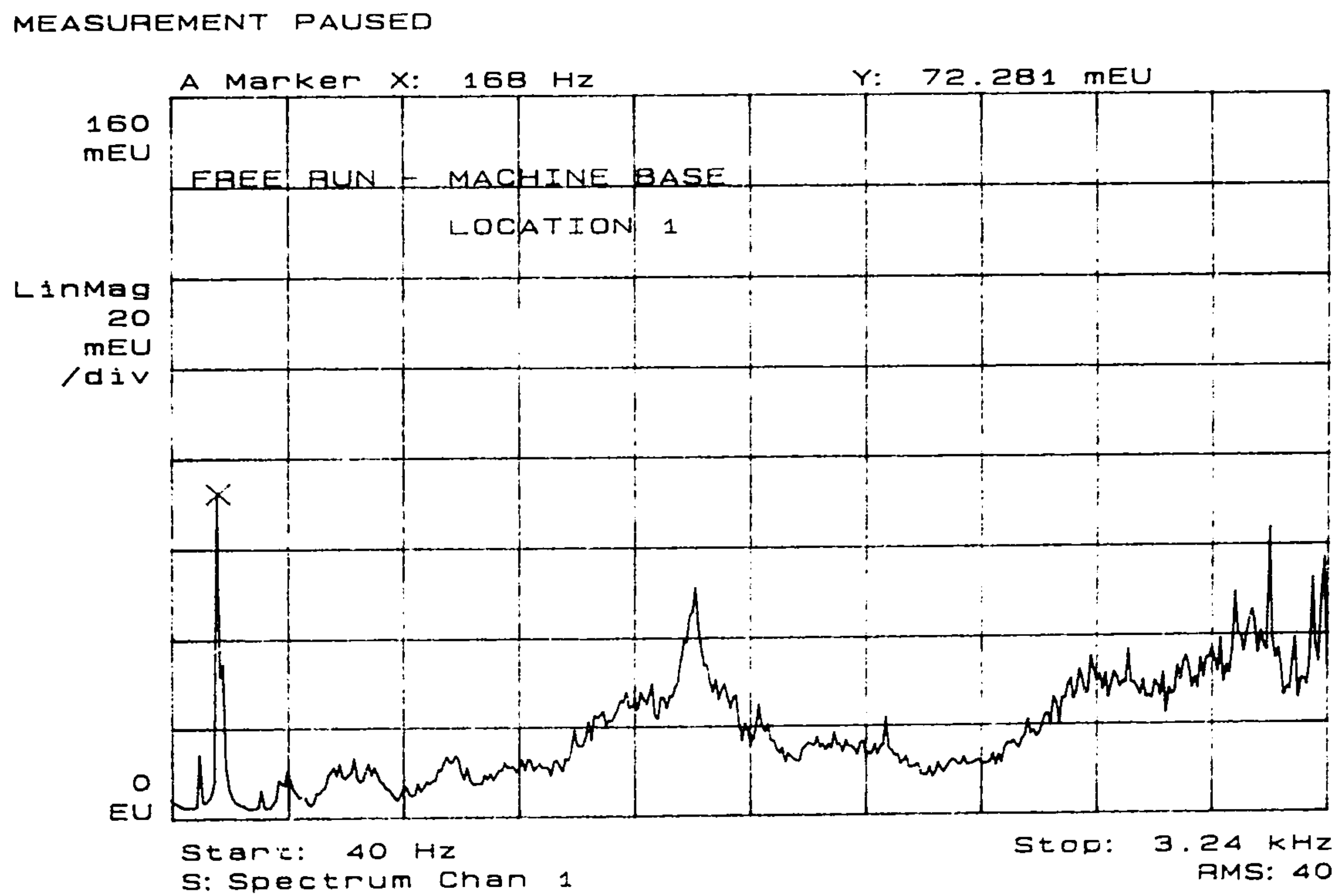



**Figure 57:** Machine head free-run- Accelerometer location 1: Machine base 3 200 Hz.

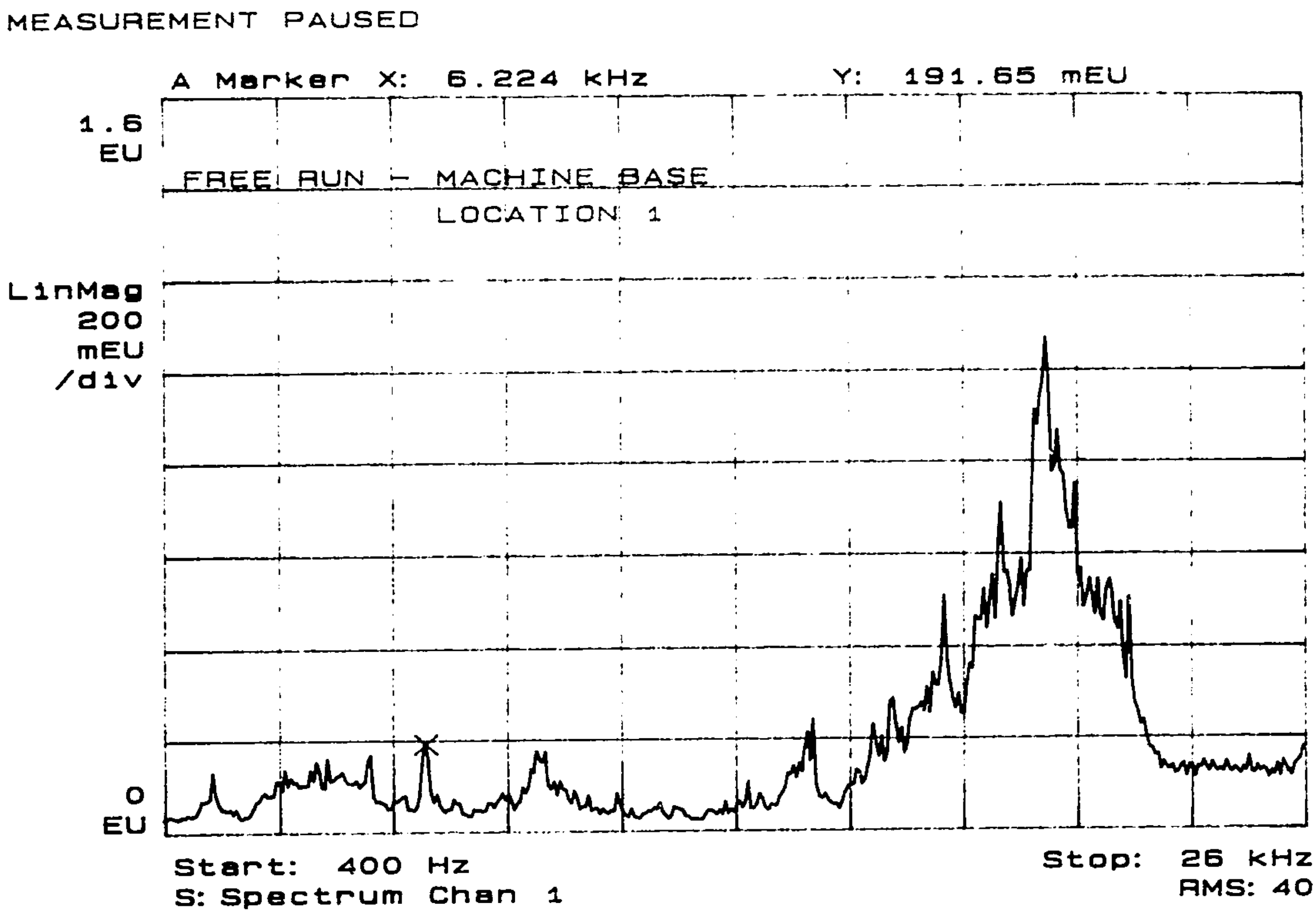



Figure 58: Machine head free-run- Accelerometer location 1: Machine base 25600 Hz.

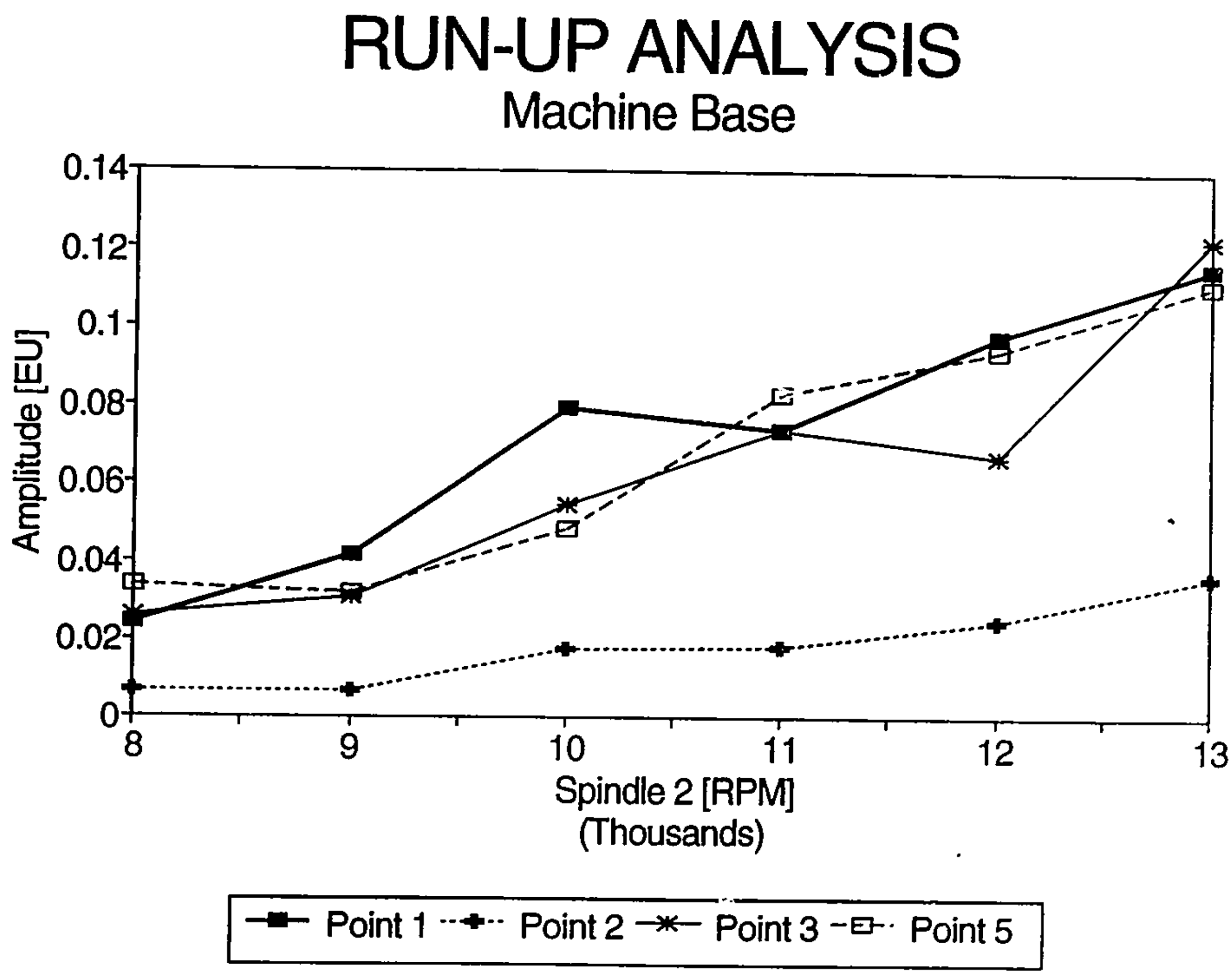



Figure 59 : Run-up analysis - machine base.

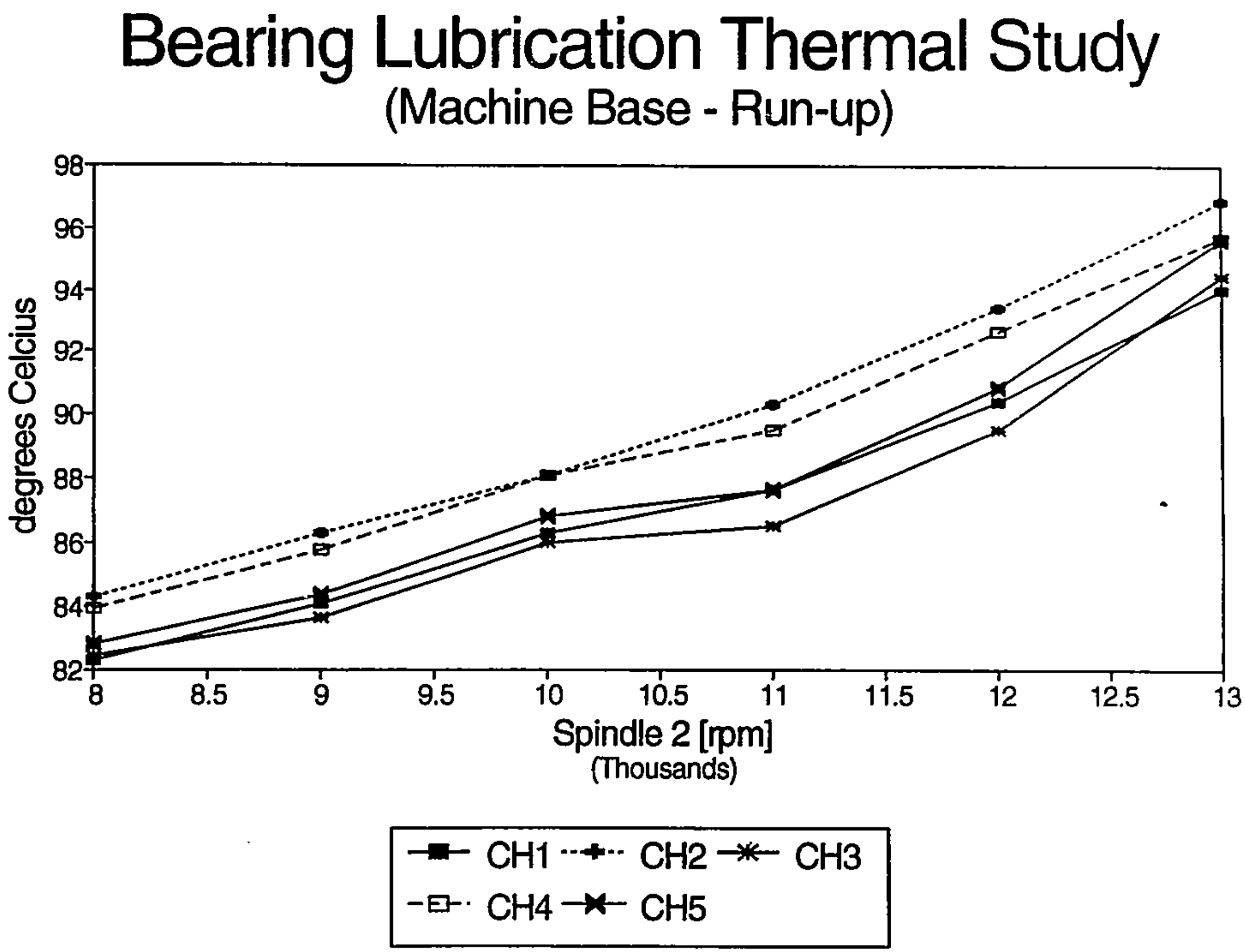



Figure 60 : Thermal properties of bearing lubrication - Machine base run-up analysis.

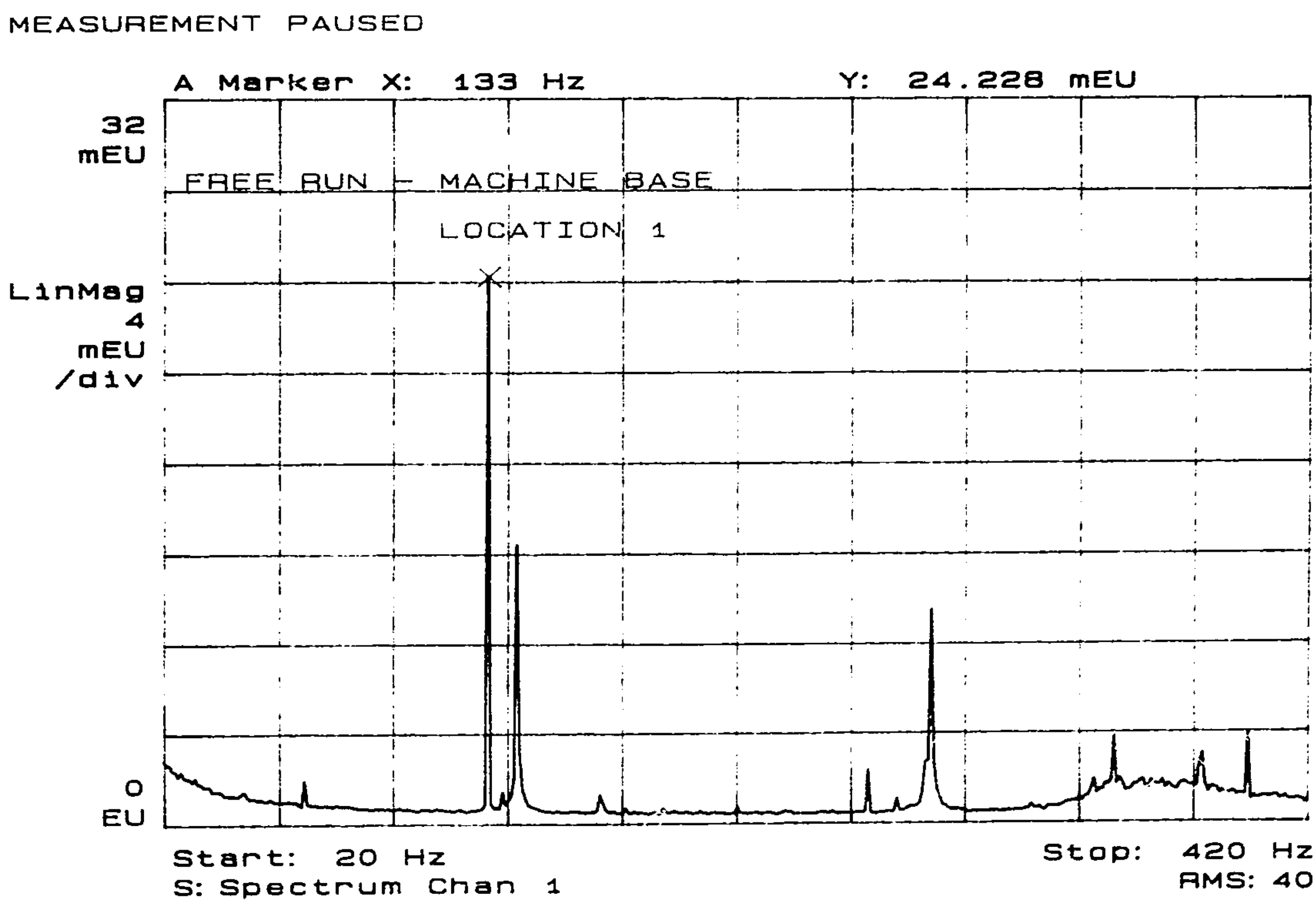



Figure 61: Machine head run-up analysis- machine base (8000 RPM): 400 Hz.

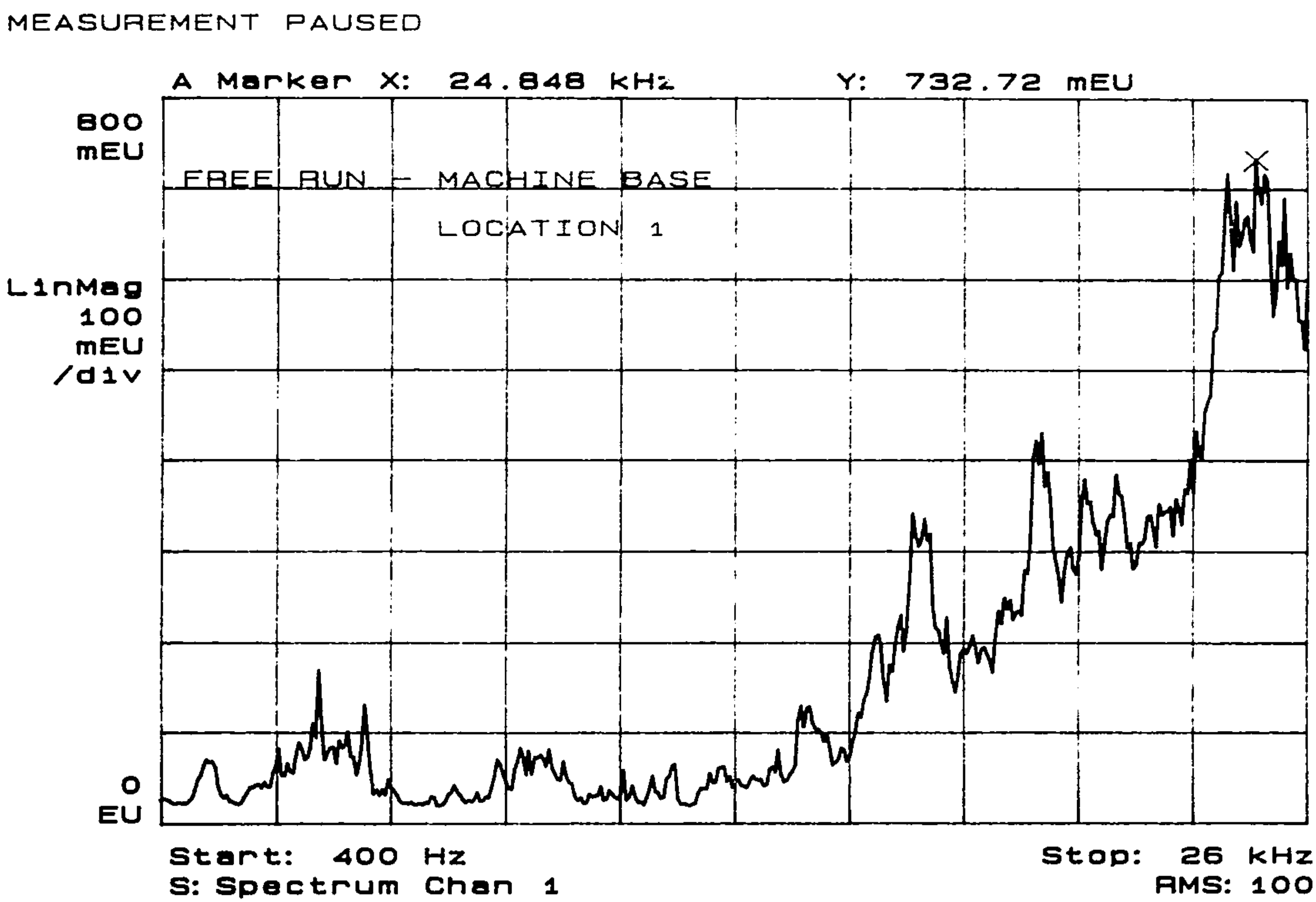



Figure 62: Machine head run-up analysis- machine base (8000 RPM): 25 600 Hz.

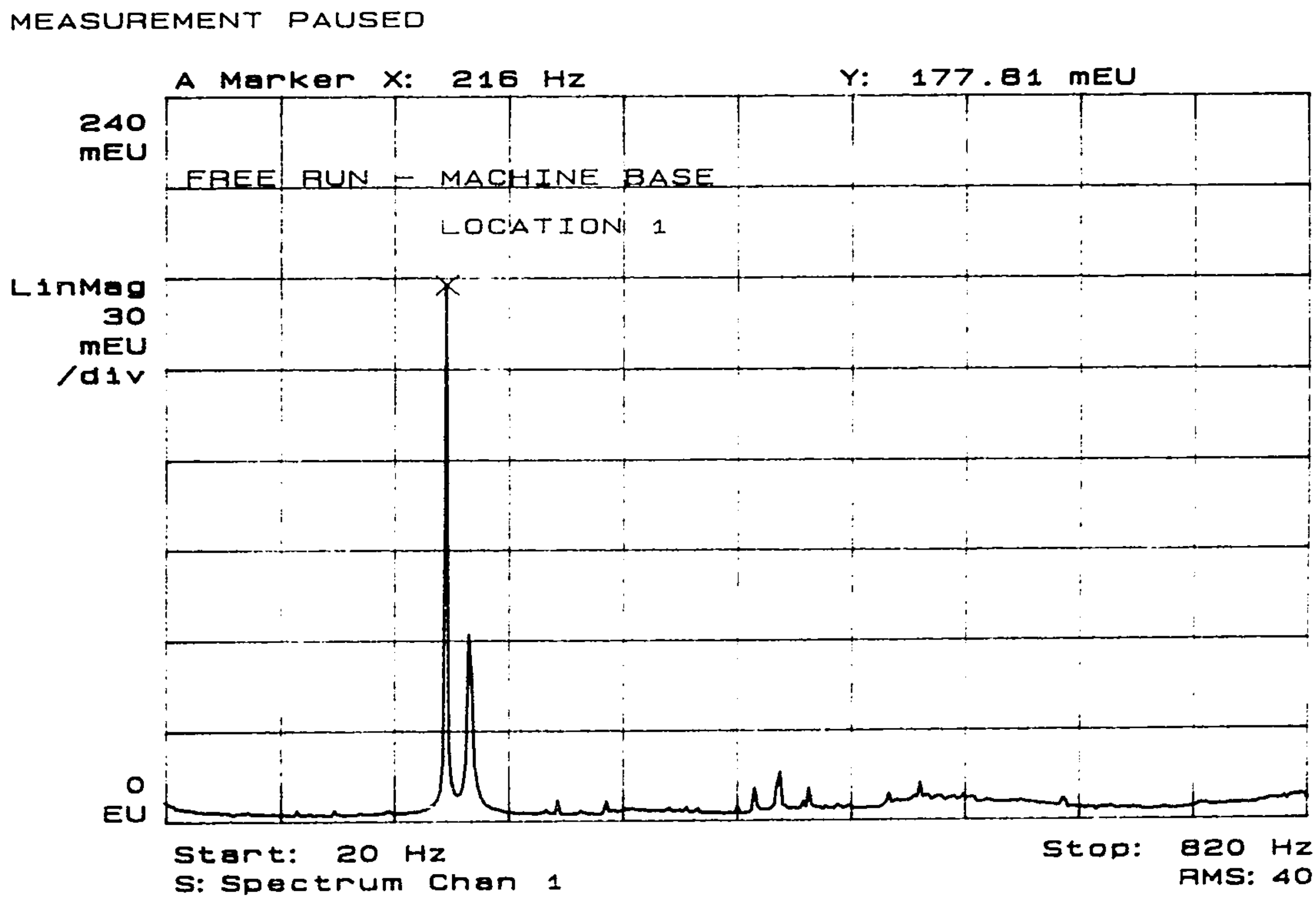



Figure 63: Machine head run-up analysis- machine base (13000 RPM): 800 Hz.

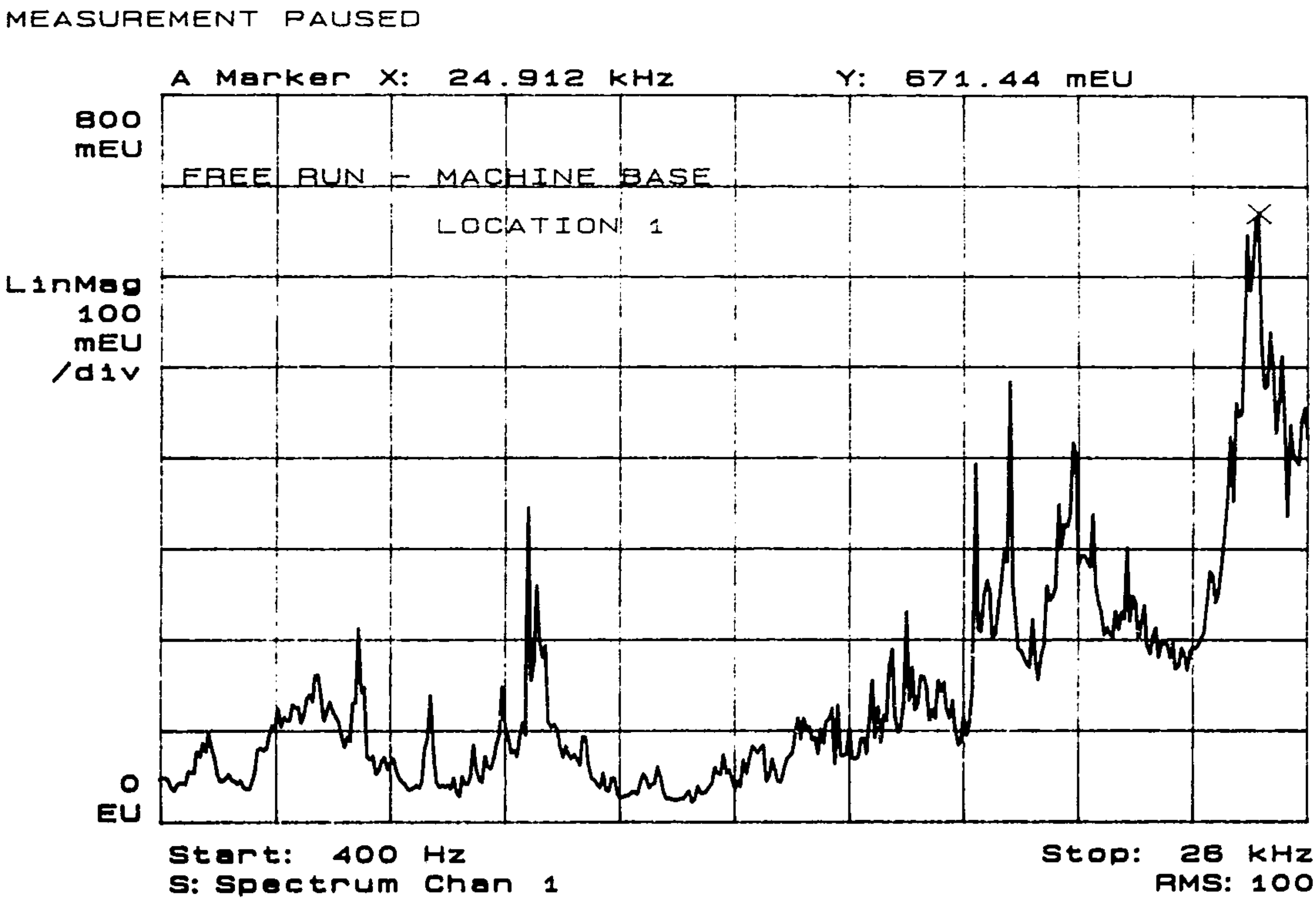



Figure 64: Machine head run-up analysis- machine base (13000 RPM): 25 600 Hz.

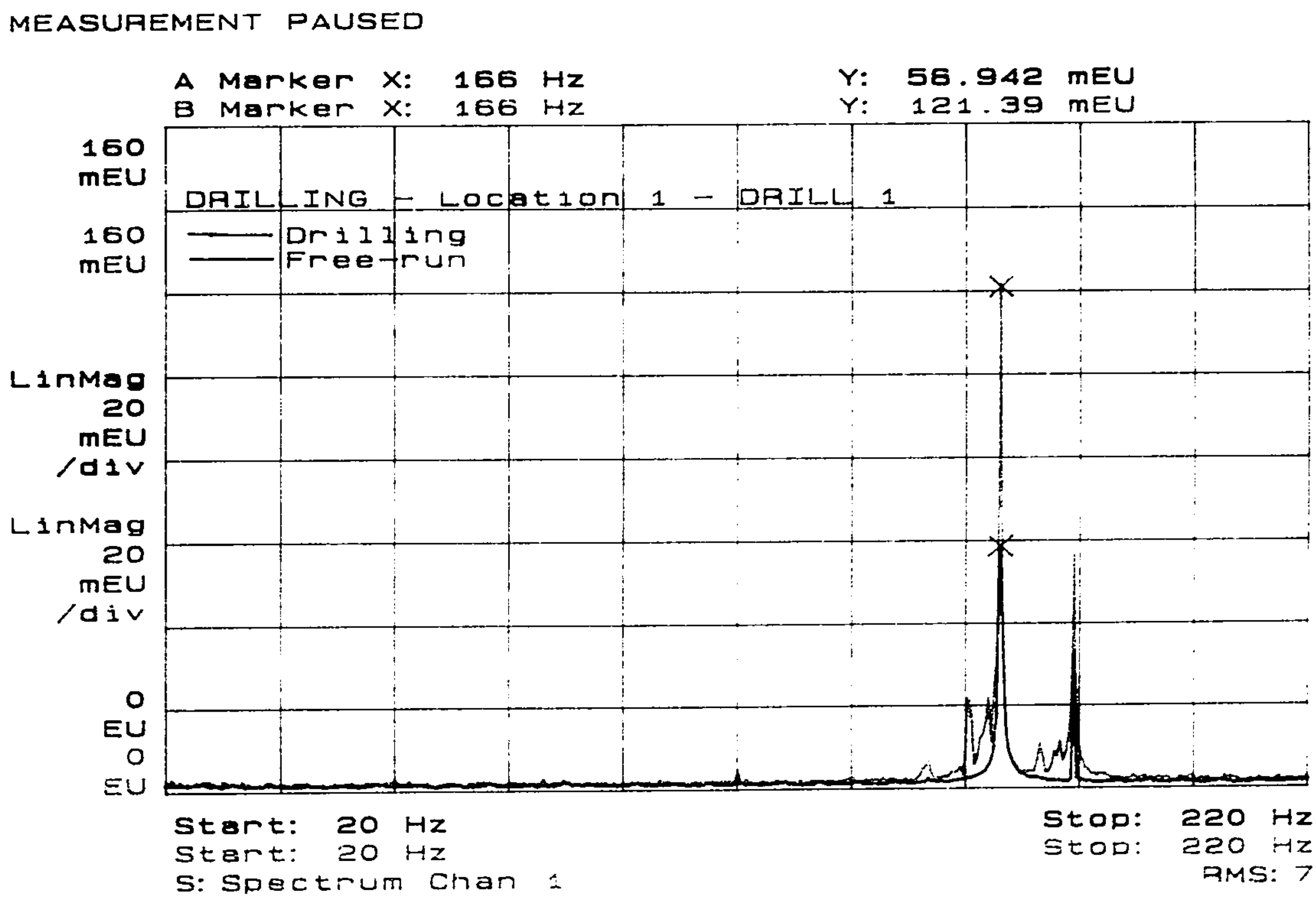



Figure 65: Machine head drilling- Accelerometer location 1: drill 1 200 Hz.

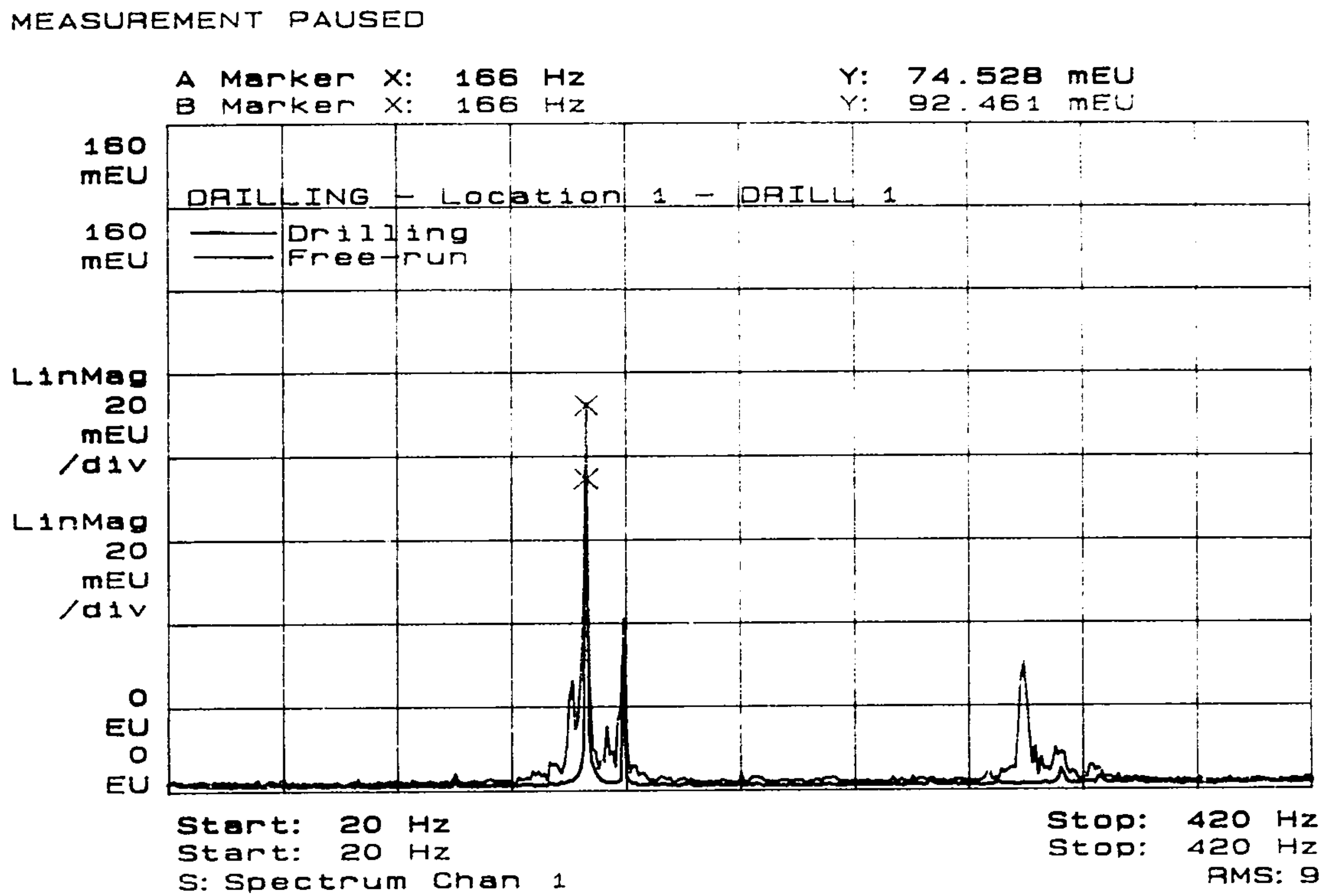



Figure 66: Machine head drilling- Accelerometer location 1: drill 1 400 Hz.

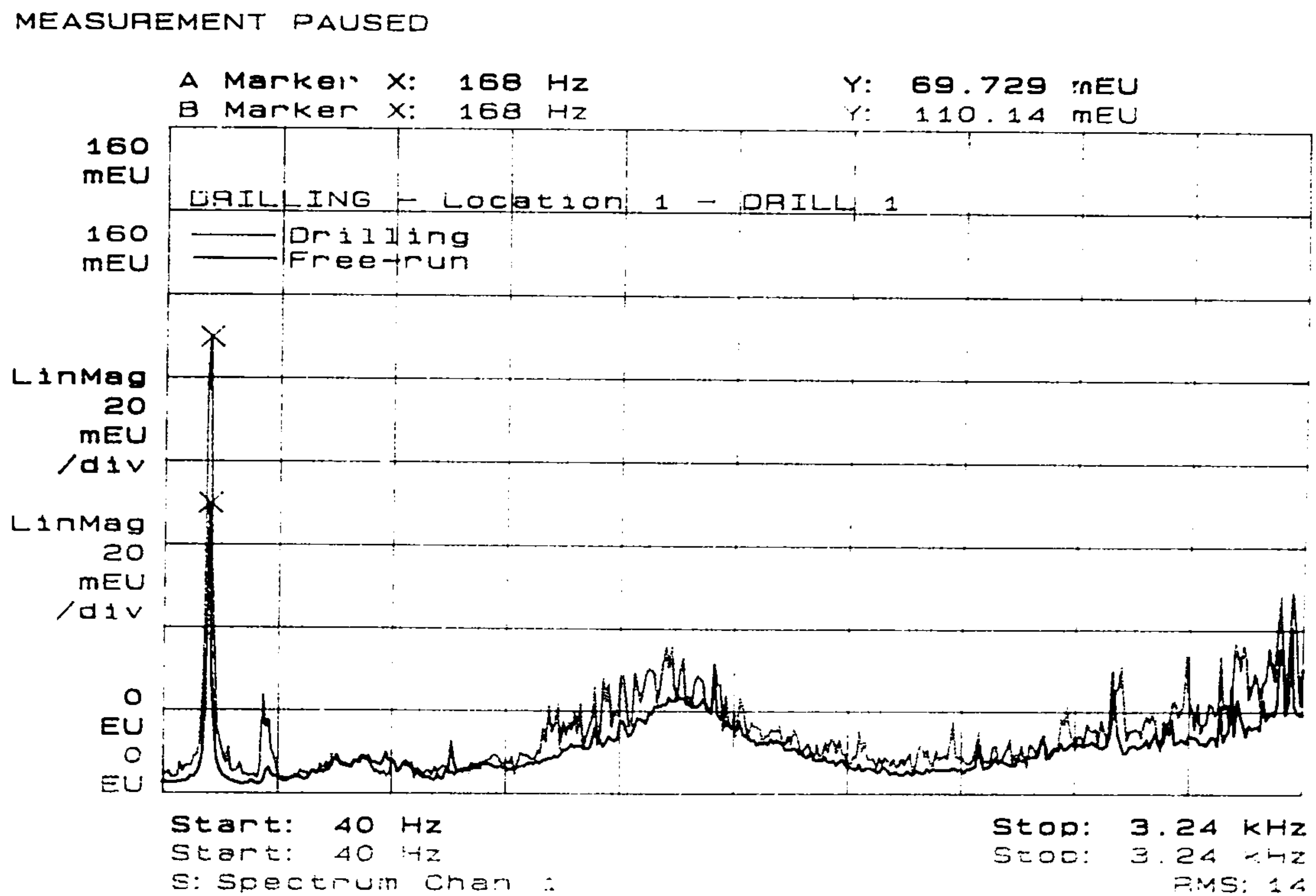



Figure 67: Machine head drilling- Accelerometer location 1 : drill 1 - 3 200 Hz.

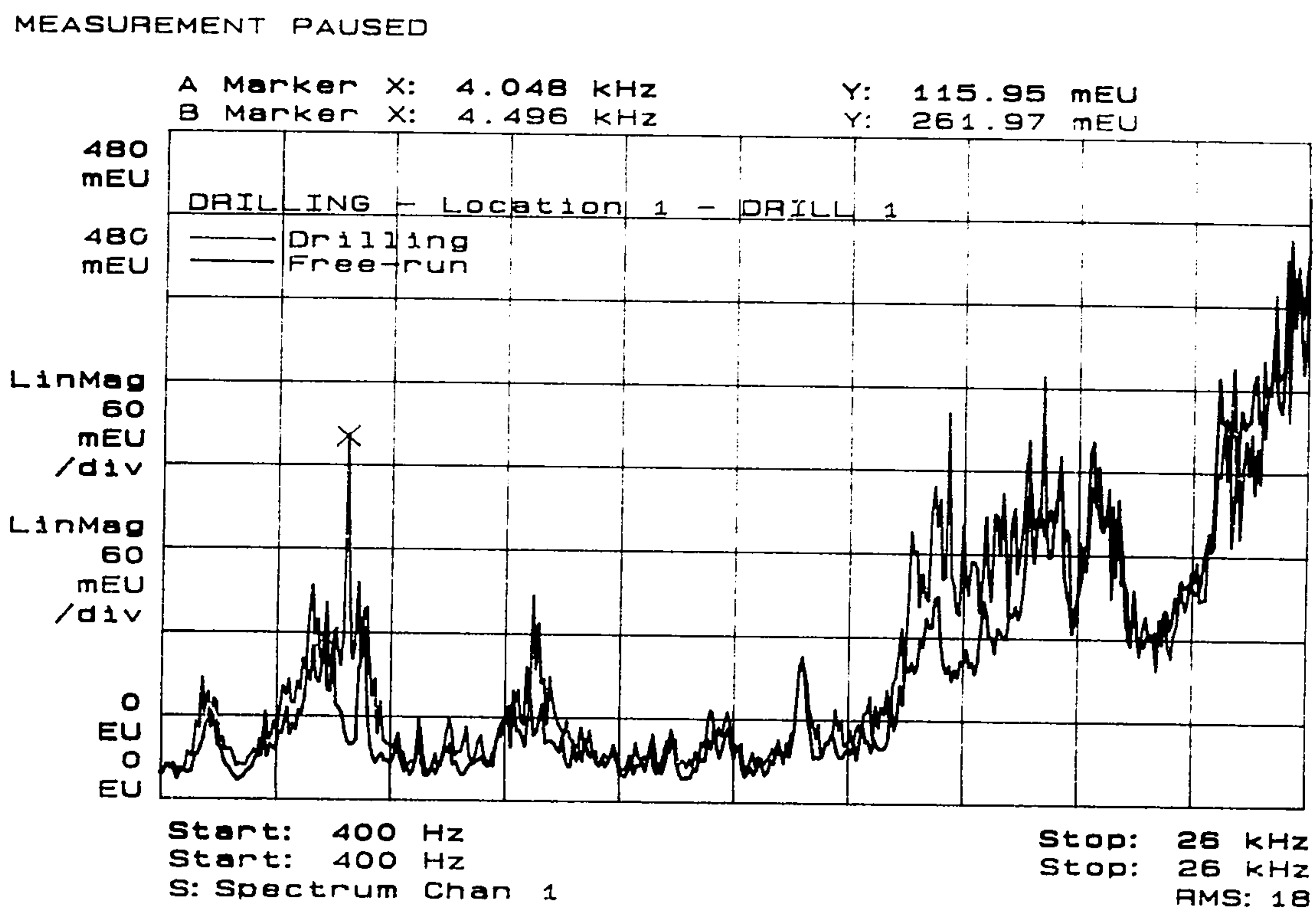



Figure 68: Machine head drilling- Accelerometer location 1 : drill 2 25 600 Hz.

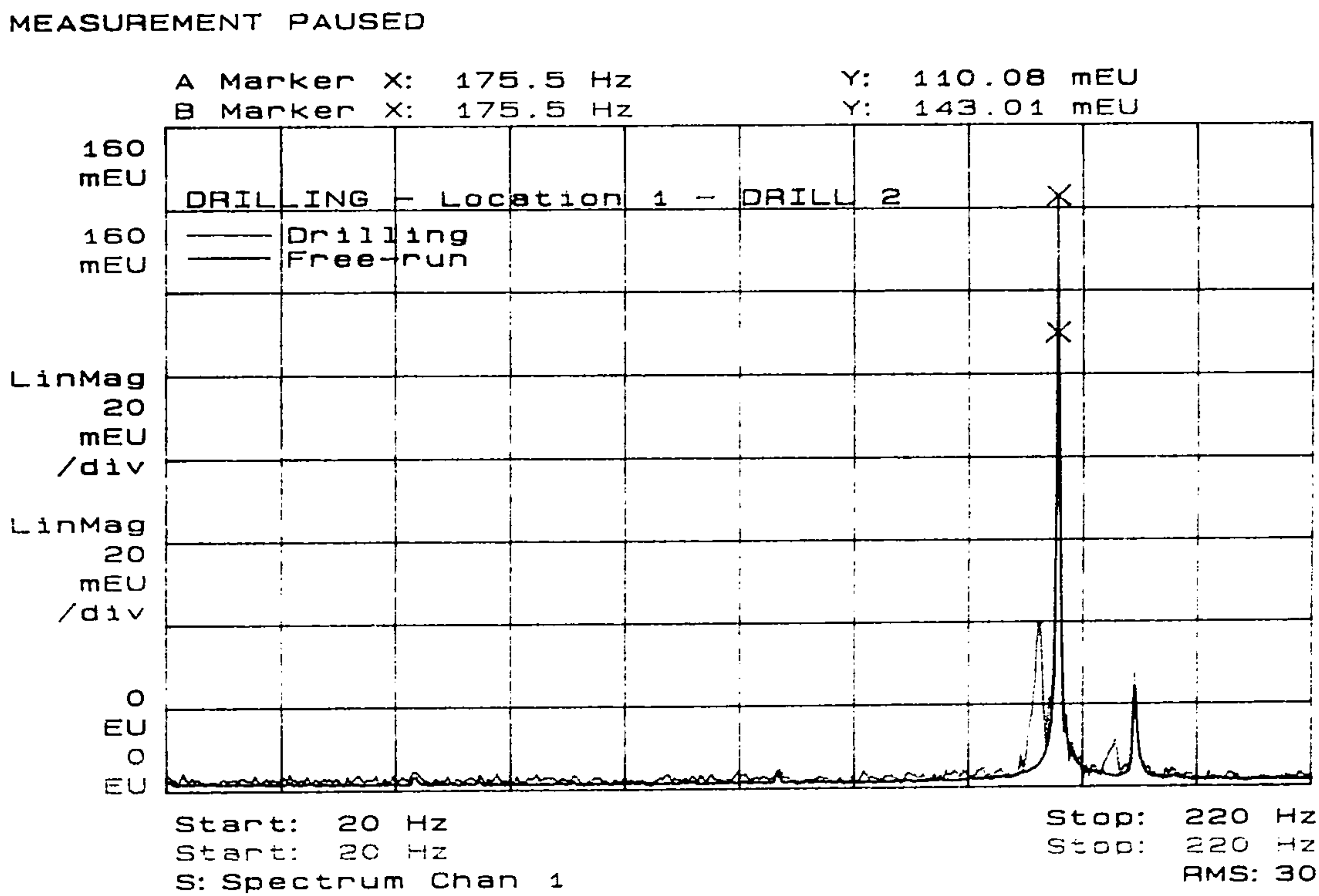



Figure 69: Machine head drilling- Accelerometer location 1 : drill 2 200 Hz.

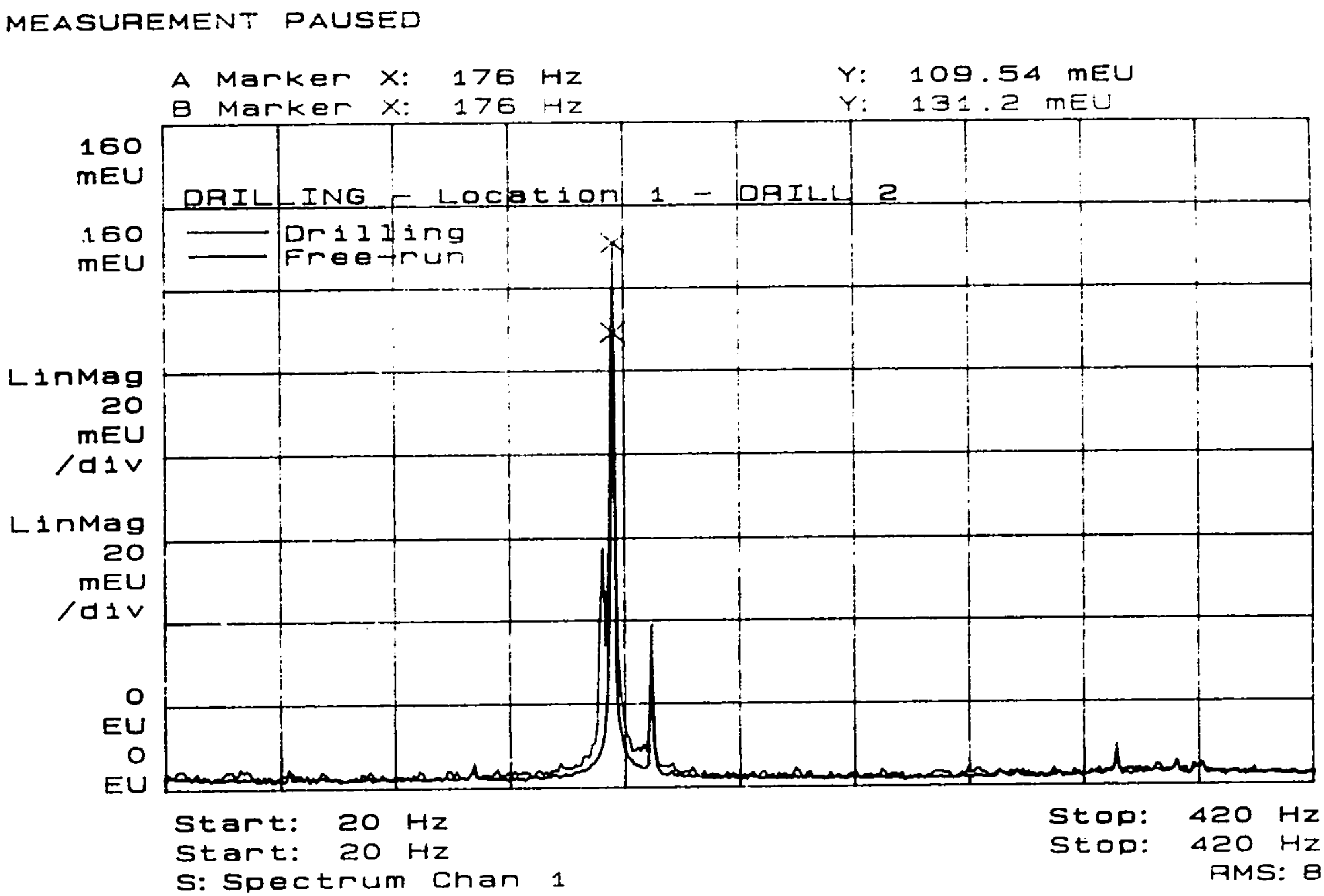



Figure 70: Machine head drilling- Accelerometer location 1: drill 2 400 Hz.

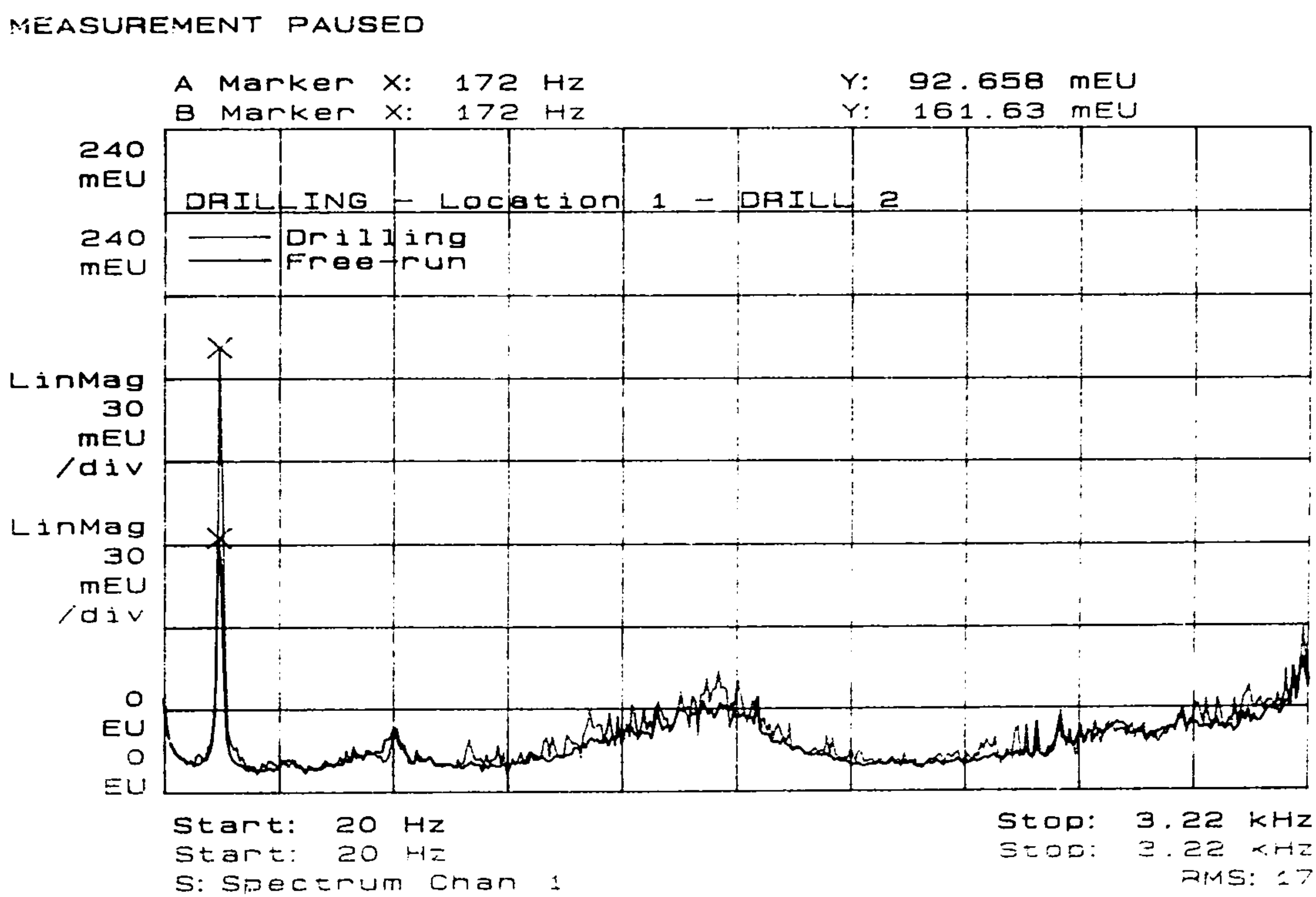



Figure 71: Machine head drilling- Accelerometer location 1: drill 2 3200 Hz.

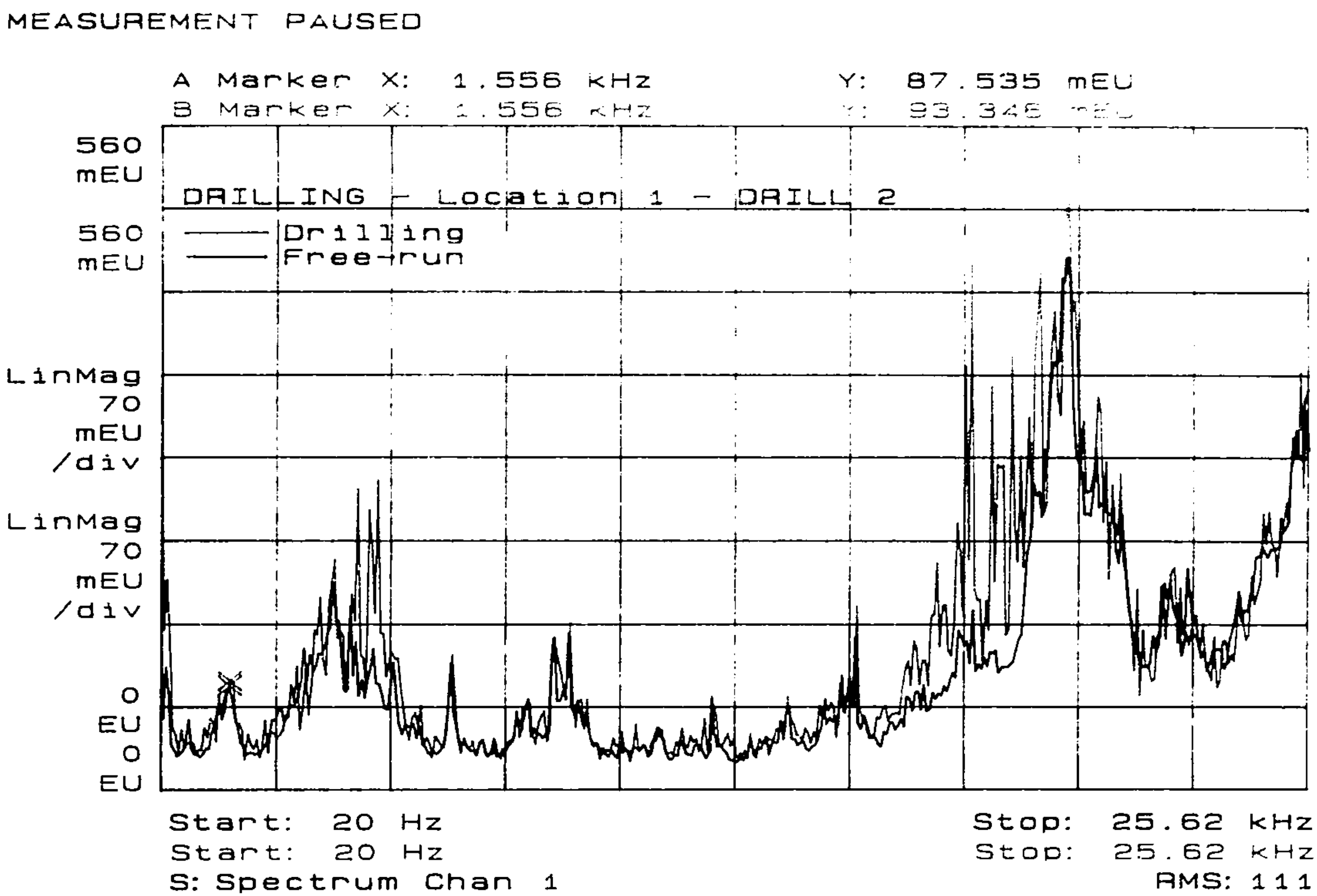



Figure 72: Machine head drilling- Accelerometer location 1: drill 2 25600 Hz.

| Parameter / Machine Fault | Temperature | Pressure | Flow | Oil Analysis | Vibration |
|---|---|---|---|---|---|
| Out of Balance | | | | | X |
| Misalignment/ Bent Shaft | X | | | | X |
| Damaged Rolling Element Bearings | | | | | X |
| Damaged Journal Bearings | X | X | X | X | X |
| Damaged or Worn Gears | | | | X | X |
| Mechanical Looseness | | | | | X |

Table 1 : Indicators for overall machine condition [69]

| Source | Frequency (Multiples of rpm) | Comparative Amplitude | Remarks |
|---|---|---|---|
| <u>Fault Induced</u><br>Mass imbalance | 1* | Proportional to imbalance; largest in radial direction | Most common cause of vibration. |
| Misalignment | 1*, and harmonics | Large in axial direction | Usually observed as severe axial vibration |
| Bent Shaft | 1* | Large in axial direction | Usually observed as severe axial vibration |
| Mechanical Looseness | 1*, and harmonics | Depends on looseness | Accompanied by imbalance or misalignment |
| Casing | 1* | | |
| <u>Design Induced</u><br>Gear Mesh (n teeth)<br>Bearings | n*<br>bearing frequency<br>0.43* to 0.47* | | |

Table 2 Vibration sources of rotating machinery [73, 74].

| Frequency Range | Low | Medium | High |
|---|---|---|---|
| Fault to be detected | Unbalance<br>Misalignment<br>Bent shaft<br>Oil whirl<br>Eccentricity | Wear<br>Faults in gears | Faults in rolling element bearings |

Table 3 Frequency range required to detect types of faults [63, 69]

| TEST # | STRUCTURE | FIRST MODAL FREQUENCY [Hz] (RPM) | SECOND MOD FREQUENCY [Hz] (RPM) |
|---|---|---|---|
| 1 | Finite Element Analysis | 1 532 (91 920) | 4 032 (241 920) |
| 2 | Spindle - V-block Fixture | 920 (55 380) | 1 597 (95 820) |

Table 4 Spindle 1 modal analysis results, required operating speed 167 Hz (10 000 RPM)

| TEST # | STRUCTURE | FIRST FREQUENCY Hz (RPM) |
|---|---|---|
| 3 | Machine Head Drill 1 | 712 (42 720) |
| 4 | Machine Head Drill 2 | 772 (46 320) |
| 5 | Machine Head Dowel | 812 (48 720) |

Table 5 Machine head frequency response analysis results, required operating speed 167 Hz (10 000 RPM)

RUN-UP ANALYSIS - Table Base

| SPINDLE 1 | | | SPINDLE 2 | | |
|---|---|---|---|---|---|
| [RPM] | [Hz] | Amplitude [EU] | [RPM] | [Hz] | Amplitude [EU] |
| 8640 | 144 | 0.016 | 8000 | 133 | 0.074 |
| 9720 | 162 | 0.034 | 9000 | 150 | 0.147 |
| 10800 | 180 | 0.021 | 10000 | 167 | 0.124 |
| 11880 | 198 | 0.014 | 11000 | 183 | 0.044 |
| 12960 | 216 | 0.086 | 12000 | 200 | 0.076 |
| 13980 | 233 | 0.108 | 13000 | 217 | 0.200 |
| 15060 | 251 | 0.304 | 14000 | 233 | 0.445 |
| 16140 | 269 | 0.074 | 15000 | 250 | 0.904 |
| 17220 | 287 | 0.083 | 16000 | 267 | 0.448 |

RUN-DOWN ANALYSIS - Table Base

| SPINDLE 1 | | | SPINDLE 2 | | |
|---|---|---|---|---|---|
| [RPM] | [Hz] | Amplitude [EU] | [RPM] | [Hz] | Amplitude [EU] |
| 17220 | 287 | 0.083 | 16000 | 267 | 0.448 |
| 16200 | 270 | 0.057 | 15000 | 250 | 0.977 |
| 15060 | 251 | 0.256 | 14000 | 233 | 0.437 |
| 13980 | 233 | 0.119 | 13000 | 217 | 0.2 |
| 12960 | 216 | 0.071 | 12000 | 200 | 0.062 |
| 11880 | 198 | 0.014 | 11000 | 183 | 0.047 |
| 10800 | 180 | 0.025 | 10000 | 167 | 0.136 |
| 9720 | 162 | 0.026 | 9000 | 150 | 0.132 |
| 8640 | 144 | 0.017 | 8000 | 133 | 0.078 |

Table 6 Run-up and run-down analysis of table base

RUN-UP ANALYSIS - Machine Base

| SPINDLE 1 [RPM] | [Hz] | AMPLITUDE LOCATION 1 [EU] | LOCATION 2 [EU] | LOCATION 3 [EU] | LOCATION 5 [EU] |
|---|---|---|---|---|---|
| 8550 | 143 | 0.010 | 0.002 | 0.160 | 0.175 |
| 9720 | 162 | 0.022 | 0.001 | 0.233 | 0.341 |
| 10860 | 181 | 0.037 | 0.008 | 0.204 | 0.430 |
| 11880 | 198 | 0.014 | 0.009 | 0.131 | 0.356 |
| 12960 | 216 | 0.047 | 0.013 | 0.049 | 0.451 |
| 13920 | 232 | 0.062 | 0.026 | 0.105 | 0.490 |

| SPINDLE 2 [RPM] | [Hz] | AMPLITUDE LOCATION 1 [EU] | LOCATION 2 [EU] | LOCATION 3 [EU] | LOCATION 5 [EU] |
|---|---|---|---|---|---|
| 8000 | 133 | 0.024 | 0.006 | 0.260 | 0.034 |
| 9000 | 150 | 0.042 | 0.006 | 0.031 | 0.032 |
| 10000 | 167 | 0.079 | 0.017 | 0.055 | 0.049 |
| 11000 | 183 | 0.074 | 0.018 | 0.074 | 0.083 |
| 12000 | 200 | 0.098 | 0.025 | 0.067 | 0.094 |
| 13000 | 217 | 0.115 | 0.036 | 0.123 | 0.111 |

Table 7 Run-up analysis on machine base

# APPENDIX I

## MATERIAL PROPERTIES OF 1020 and 8620 STEEL

## Material Characteristics of 1020 and 8620 Steel

Material Composition [%]

| | | | | |
|---|---|---|---|---|
| 1020 | C | 0.18-0.23 | Mn | 0.30-0.60 |
| | P | 0.04 | S | 0.05 |
| 8620 | C | 0.18 | Mn | 0.07-0.90 |
| | P | 0.035 | S | 0.04 |
| | Si | 0.15-0.35 | Ni | 0.40-0.70 |
| | Cr | 0.04-0.60 | Mo | 0.15-0.25 |

Material Strength [kpsi (MPa)]

| | | tensile | yield | Brinell Hardness |
|---|---|---|---|---|
| 1020 | hot rolled | 55 (380) | 30 (210) | 111 |
| | cold drawn | 61 (420) | 51 (350) | 121 |
| 8620 | normalized | 91.7 (634) | 51.7 (357) | 183 |
| | annealed | 77.8 (538) | 55.9 (386) | 143 |

# APPENDIX II

# PROPERTIES OF BEARINGS and GREASE LUBRICATION

Dual-Spindle High-Speed Machine Head
Bearing Characteristics

| | OD [mm] | ID [mm] | Width [mm] | Balls Dia. [mm] | No. Balls |
|---|---|---|---|---|---|
| Spindle 1 | | | | | |
| BHT 6003 C TA | 35 | 17 | 10 | 4.762 | 13 |
| BHT 6004 C TA | 42 | 20 | 12 | 6.350 | 12 |
| Spindle 2 | | | | | |
| BHT 6005 C TA | 47 | 25 | 12 | 6.350 | 14 |
| BHT 6006 C TA | 55 | 30 | 13 | 7.144 | 15 |

# TECHNICAL INFORMATION

Kluber Lubrication North America, Inc.
54 Wentworth Avenue
Londonderry, NH 03053
Tel. (603) 434-7704
FAX (603) 434-8046

KLÜBER LUBRICATION

## ISOFLEX® NBU 15

**Technical Data:**

| Characteristic | Typical Performance | Test Specification |
|---|---|---|
| Thickener | Barium complex | --- |
| Base Oil | Ester/Mineral | --- |
| Texture | creamy | --- |
| Color | beige | --- |
| NLGI Grade | 2 | DIN 51818 |
| Density at 20°C, (g/ml) | 1.02 | --- |
| Service Temp., (°C) | -40 to 130 | --- |
| Drop Point, (°C) | > 220 | DIN 51801/1 |
| Penetration, Worked @25°C, (0.1 mm) | 265 - 295 | DIN 51804/1 |
| Apparent Dynamic Viscosity (mPas) (Shear Rate = 300 $S^{-1}$, @25°C) | 5,000 | Rotovisco Cone/Plate |
| Speed factor | 600,000 | |
| Base Oil Viscosity ($mm^2/s$) at 40°C at 100°C | 25<br>5 | DIN 51561 |
| Water Resistance 5 hrs. @ 90°C | fully resistant | DIN 51807 |
| Water wash-out test after 1 hr/38°C after 7 hr/79°C | 3% weight loss<br>7% weight loss | ASTM D 217 |
| Corrosion against copper 24 hrs @ 100°C | no corrosion | DIN 51811 |
| Oil Separation After 7 days @ 40°C % by weight | less than 3% | DIN 51817 |
| Evaporation loss 22 hrs/100°C | < 1% | ASTM D 942 |
| SKF Mechanical Dynamic Testing | passed 120°C | DIN 51806 |

July 1991

Article No. 004026

These statements represent our present day information and are intended to describe our products and their applications. They are not, however, intended to guarantee particular properties of the products or their qualifications for specific applications. This information is also confidential to the recipient and may in no case, without special permission be transmitted to a third party, particularly competitors, either in its original form or as a copy or extract. The recipient of this data incurs the obligation to keep them confidential and to be responsible for damages resulting from infringements whereby Kluber Lubrication North America will not have to prove negligence.

# APPENDIX III

## FEM CENTRE of GRAVITY WEIGHT STUDY OF SPINDLE 1

**** ALGOR Interactive Systems, Inc. ****
SSAP9 (Wt and C.G. calculations) Rel. 17-JUL-92 Ver 10.00

Input file name: spl
Date & Time : 1993-12-14 10:24:32
GMA - 30 degree revolution

---

Prepared by DECODS 2.04

Input information:

Number of nodes (NUMNP) = 2132
Number of element types (NELTYP) = 1
Number of load cases (LL) = 0
Analysis code (NDYN) = 1
Gravitational constant (GRAV) = 386.40000

Element type = 5 (3-D Brick)

Number of elements = 2004

| No. | Ele. # | Mat. # | Mass Density | Weight Density |
|---|---|---|---|---|
| 1 | 5 | 1 | 7.33954E-04 | 2.8360000E-01 |

================ Output results ================

I. Volume, Weight and C.G. (Global coordinates):

| Volume | Weight | XC | YC | ZC |
|---|---|---|---|---|
| 6.2083E+00 | 1.7607E+00 | 5.1422E+00 | 3.7175E-09 | 3.0938E-0[illegible] |

II. Moment of inertia w.r.t. X-Y-Z axes at (XR,YR,ZR):
Reference point (XR,YR,ZR) is at ( 0.000E+00, 0.000E+00,
0.000E+00)

MASS moment of inertia

| IX | IY | IZ |
|---|---|---|
| 7.0419E-04 | 1.6342E-01 | 1.6342E-01 |

# APPENDIX IV

# DUAL-SPINDLE HIGH-SPEED DRILLING MACHINE HEAD INFORMATION

Dual-Spindle High Speed Drilling Machine Head

Drill Diameter (D)
Drill 1 = 0.375 [inch]
Drill 2 = 0.250 [inch]

RPM = 10 000 rpm
N = number of flutes = 2
R = 10 000 [rpm]
f = 60 [ipm]

| | Drill 1 | Drill 2 |
|---|---|---|
| Cutting Speed (V) = (RPM)*D/4 [sfm] | 937.5 | 625.0 |
| Feed (F) = N*R*D*0.017 [ipm] (attainable) | 127.5 | 85.0 |
| MRR(Z) = f*pi*(D^2)/4 [in^3/min] | 6.6 | 2.9 |

Horsepower at cutting tool, Chp

P = power constant factor = 0.2
for aluminum alloys
F = feed [inch/rev] = 5.988e-3 [inch/rev]

| | Drill 1 | Drill 2 |
|---|---|---|
| Chp = P*pi*D^2/4*F*R | 1.3 | 0.6 |

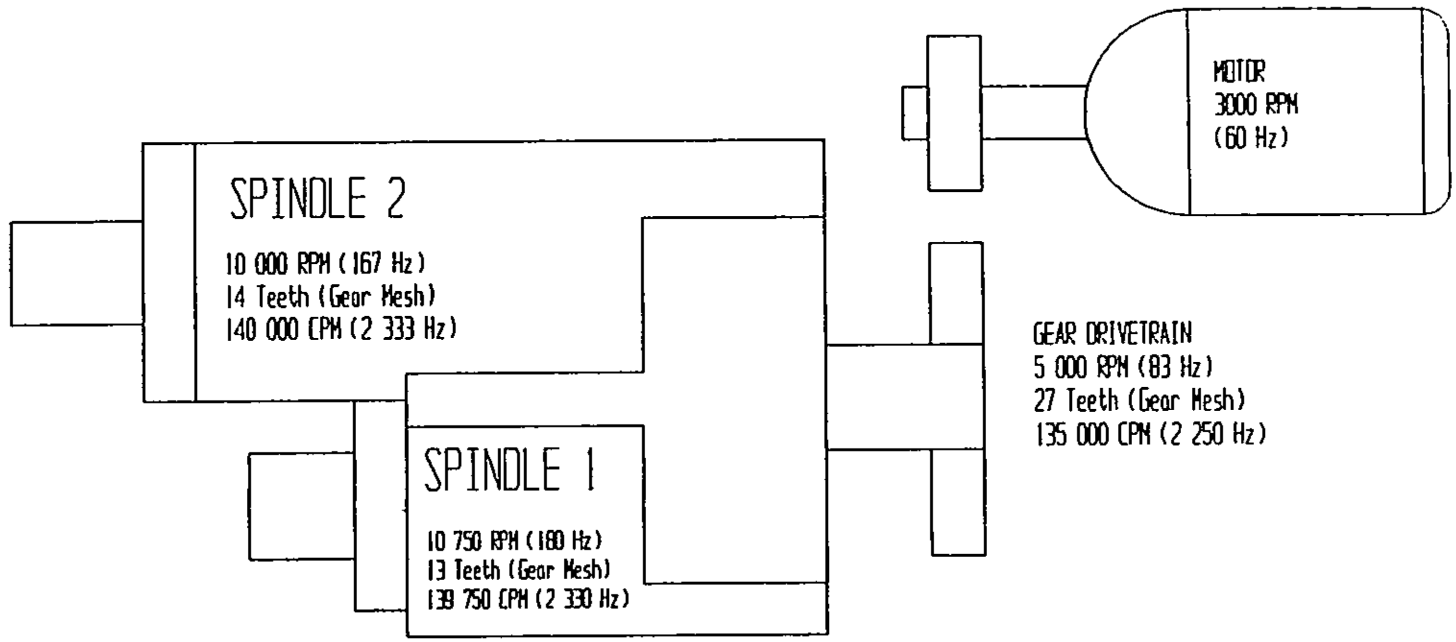
Horsepower required from motor, Mhp

e = drive motor factor = 0.9 direct belt drive

| | Drill 1 | Drill 2 |
|---|---|---|
| Mhp = Chp/e | 1.5 | 0.7 |

Thrust recquired for drilling (Tp)

| | Drill 1 | Drill 2 |
|---|---|---|
| Tp = (148 500)*PFD [lbf] | 66.5 | 44.3 |



SPINDLE 2
10 000 RPM (167 Hz)
14 Teeth (Gear Mesh)
140 000 CPM (2 333 Hz)
SPINDLE 1
10 750 RPM (180 Hz)
13 Teeth (Gear Mesh)
139 750 CPM (2 330 Hz)
MOTOR
3000 RPM
(60 Hz)
GEAR DRIVETRAIN
5 000 RPM (83 Hz)
27 Teeth (Gear Mesh)
135 000 CPM (2 250 Hz)

# VITA AUCTORIS

1967 Born in Jullunder,Punjab. India.

1987 Received OSSGHD from Vincent Massey Secondary School.

1991 Received the Degree of Bachelor of Applied Science from the University of Windsor, Windsor, Ontario, Canada.

1994 Candidate for the Degree of Master of Applied Science at the University of Windsor, Windsor, Ontario, Canada.