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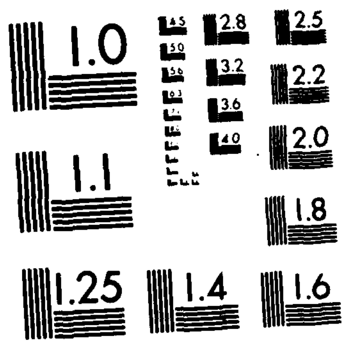

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Technical Report

Flexible Conformable Clamps for a Machining Cell with Applications to Turbine Blade Machining

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CMU-RI-TR-83-16

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May 1983

The research culminating in this technical report formed the author's Master's project for the degree of Master of Engineering in the Mechanical Engineering Department at Carnegie-Mellon University. Sections of this report are adapted from an earlier paper presented at Autofact IV, Philadelphia, November 1982.

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A flexible, conformable clamping scheme for a steam turbine blade machining cell is discussed. Experiments that have been carried out to investigate the clamping efficiency of this novel design are also described. Flexible fixtures for turbine blade machining have been developed and installed to demonstrate the reduction of human intervention for workpiece setup in the laboratory at Carnegie-Mellon University. The key features of the clamp design and the cell configuration are described. Subsequently, the theory of vibration is reviewed and the experimental results of

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Abstract

A flexible, conformable clamping scheme for a steam turbine blade machining cell is discussed. Experiments that have been carried out to investigate the clamping efficiency of this novel design are also described. Flexible fixtures for turbine blade machining have been developed and installed to demonstrate the reduction of human intervention for workpiece setup in the laboratory at Carnegie-Mellon University. The key features of the clamp design and the cell configuration are described. Subsequently, the theory of vibration is reviewed and the experimental results of damped natural frequencies of the clamped blade are presented. For future research a cell model may be established and simulated. Improvements in the clamp mechanism and an intensive study of vibration are also expected.

1. Introduction

The concept of flexible machining cells has been described by a number of investigators and the goal is to produce batches of different parts without human intervention [1, 2, 3, 4]. Although CNC machine tools, industrial robots and supervisory computers are currently available and may combined to construct a flexible machining cell, the success of a truly flexible machining cell that runs without human assistance depends on several additional developments of cell functions such as inspection, tool setup, workpiece setup, chip removal and tool monitoring, etc.

Most current flexible machining cells employ pallets [1, 5] to reduce human assistance for workpiece setup during a batch run. Ideally human intervention between batch runs should be eliminated but since there are difficulties involved in aligning and clamping parts, this is hard to achieve. With pallets, the fixturing becomes identical at each machine tool regardless of variations in shape and size between different workpieces. At some point, however, parts must be accurately aligned and clamped to the pallets. In today's pilot flexible machining cells this is done manually but it is desirable to devise an automatic clamping system.

In this paper, flexible conformable clamps for a steam turbine blade machining cell are discussed to eliminate human assistance between batch runs. There are many other factors to prevent the completion of the truly unmanned cell, however, we can approach one step closer to the ideal system by this attempt.

In a conventional steam turbine blade manufacturing plant, there are 6000 different blade designs each requiring different jigs and fixtures. This represents an enormous capital investment and provides a strong motivation for devising a flexible machining cell. The blades arrive at CNC machining centers as closed-die forgings. These taper-twisted shapes are difficult to clamp without specially designed jigs and fixtures. The important roles of the clamping are rapid fixturing, stiffness against cutting forces, good vibration characteristics and holding accuracy. In the manufacturing sequence at present[1], a low melting point alloy, *cerubin*, is cast around the airfoil of the blade to provide efficient and secure clamping (Fig.2-2). To machine out *stubs* (Fig.2-1) near the center of the blade span, a special cradle fixture is used to eliminate chatter (Fig.2-3).

There are a number of reasons for moving away from this system involving the casting of cerubin around the turbine blades and heavy cradle type fixtures. The cerubin is very expensive and labor involved in accurately fixing the blade and then pouring the alloy around the blade is expensive. At the end of the process, the alloy must also be remelted and retrieved. In addition, the blade with cast or cradle fixture is too heavy to be handled by a current industrial robot.

In order to maintain flexibility of clamping and eliminate the need for the casting process a programmable, conformable clamp has been developed [6].

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[1] as carried out in a Westinghouse Electric Co. plant at Winston-Salem N.C.

2. Steam Turbine Blade Manufacturing Processes

In order to get a better understanding of the machining process during a blade production sequence, each operation is briefly examined.

2.1. A sequence of blade production

There are several possible ways to produce steam turbine blades. A typical sequence introduced in an actual manufacturing plant is listed. This procedure is used to produce large taper-twisted turbine rotor blades for electric power generation. Simpler shaped blades are machined from extruded bar stock and more complicated blades such as gas turbine blades, are made by casting. However, the sequence below is typical for the mid-range of complexity such as steam turbine blades.

1. Raw material - Cylindrical billet
2. ***Open die forging - Swaging***[2]
3. Closed precision die forging
4. Trimming
5. Heat treatment
6. Inspection - Hardness and Dimension
7. Remedy distortion by press and repeat inspection
8. ***Machining***
9. Grinding - Airfoil surface
10. Inspection - Dimension, Weight and Vibration

2.2. Machining process

The machining process is examined in detail to design an appropriate fixture. Fig.2-1 shows each area to be machined corresponding to the following list. Typical milling conditions are also listed in accordance with Machining Data Handbook [7] for a solution heat treated precipitation hardening stainless steel of Brinell hardness 275 - 325. [d = depth of cut, v = speed, f = feed per tooth]

1. Root (Fig.2-2)
   a. Bottom surface milling - Face cutter [d = 4 mm, v = 105 m/min, f = 0.18 mm]
   b. Rough milling - Tapered side cutter [d = 4 mm, v = 70 m/min, f = 0.13 mm]

---

[2] A Flexible Swaging Cell has been developed and installed at the Turbine Components Plant, Westinghouse Electric Co. NC [4]

c. Pre-finish milling - Formed cutter [d = 4 mm, v = 14 m/min, f = 0.15 mm]

d. Finish milling - Formed cutter [d = 0.2 mm, v = 50 m/min, f = 0.15 mm]

2. Sides of Root (Width) - Face cutter [d = 4 mm, v = 105 m/min, f = 0.18 mm]

3. Side edges - End mill cutter [d = 1.5 mm, v = 64 m/min, f = 0.15 mm]

4. Tip of blade

a. Tip edge - Face cutter [d = 4 mm, v = 105 m/min, f = 0.18 mm]

b. Tenon - End mill cutter [d = 1.5 mm, v = 64 m/min, f = 0.15 mm]

5. Stub (Fig.2-3)

a. Face - Face cutter [d = 4 mm, v = 105 m/min, f = 0.18 mm]

b. Contour - Ball end mill cutter [v = 64 m/min, f = 0.15 mm]

6. Stellite brazing groove - End mill cutter [d = 3 mm, v = 14 m/min, f = 0.05 mm]

7. Locking piece hole - End mill cutter [v = 14 m/min, f = 0.05 mm]

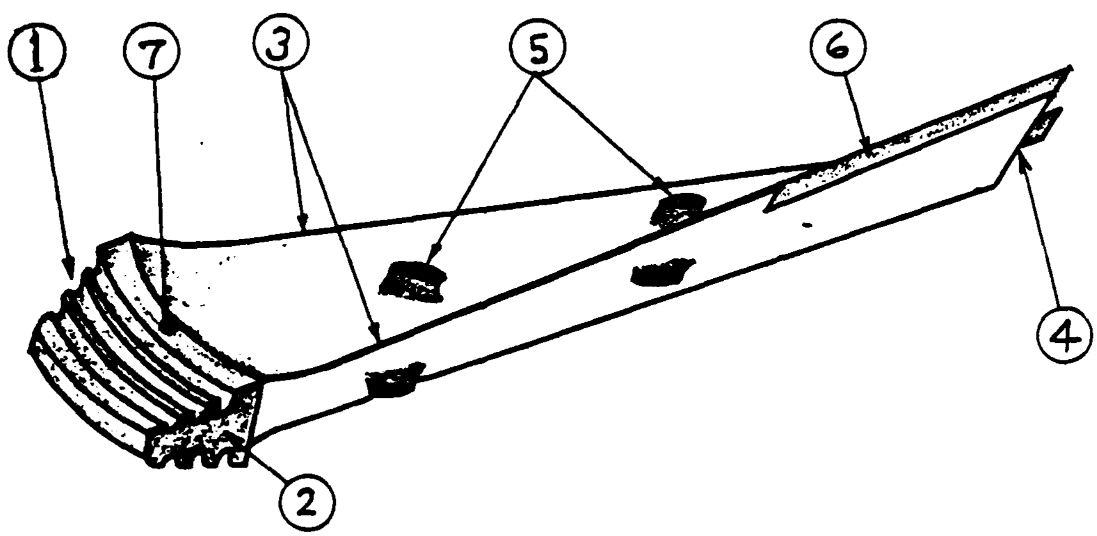

Figure 2-1: Areas to be machined for a typical turbine blade

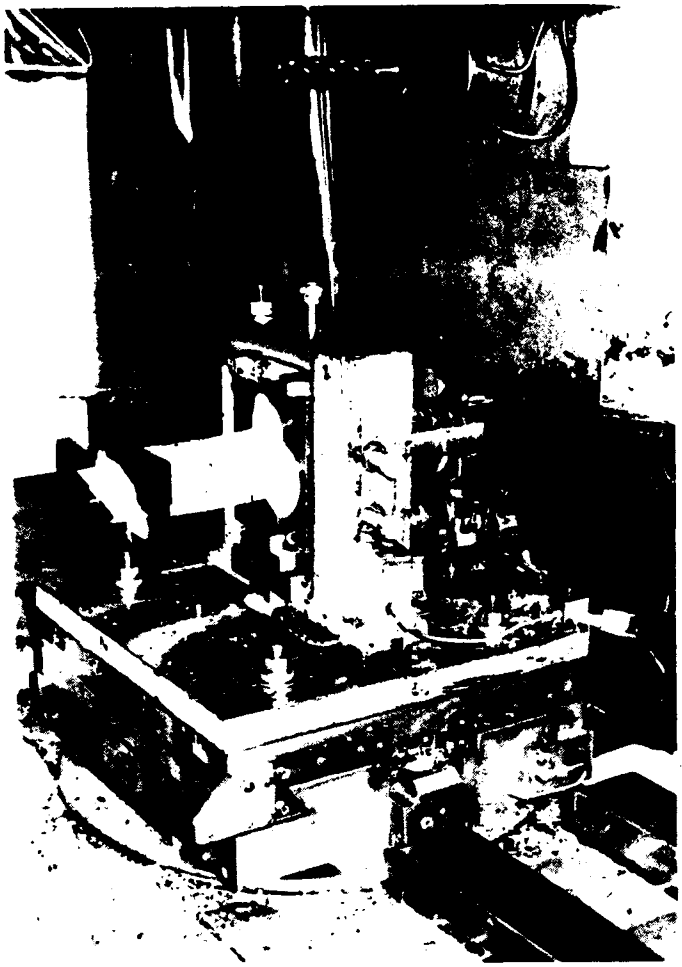

Figure 2-2: Root Machining in the Cast Fixture

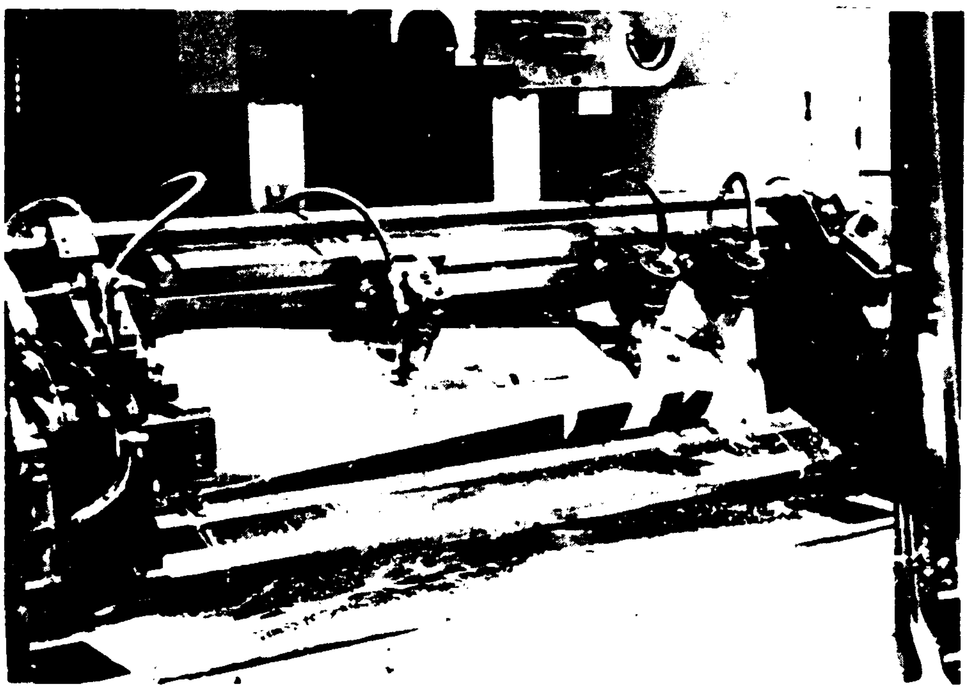

Figure 2-3: Stub Machining in the Cradle Fixture

3. Clamp Design

3.1. A flexible Clamping System

In keeping with the above considerations a flexible clamping system has been devised for a machining cell in the author's laboratory (Figs.3-1,3-2). The design is able to cope with the complex three dimensional form of the blades and the absence of obvious orientational features such as holes or square edges. Additional requirements for a successful clamping system may be listed as follows:

- The clamps should be light and compact enough to travel with the blade so that the process of alignment need be done only once.

- The clamps should handle as many as possible of the different blade styles.

- The clamping should be rigid enough against cutting force and have good vibration characteristics to prevent chatter.

- The clamps should facilitate automatic clamping and unclamping.

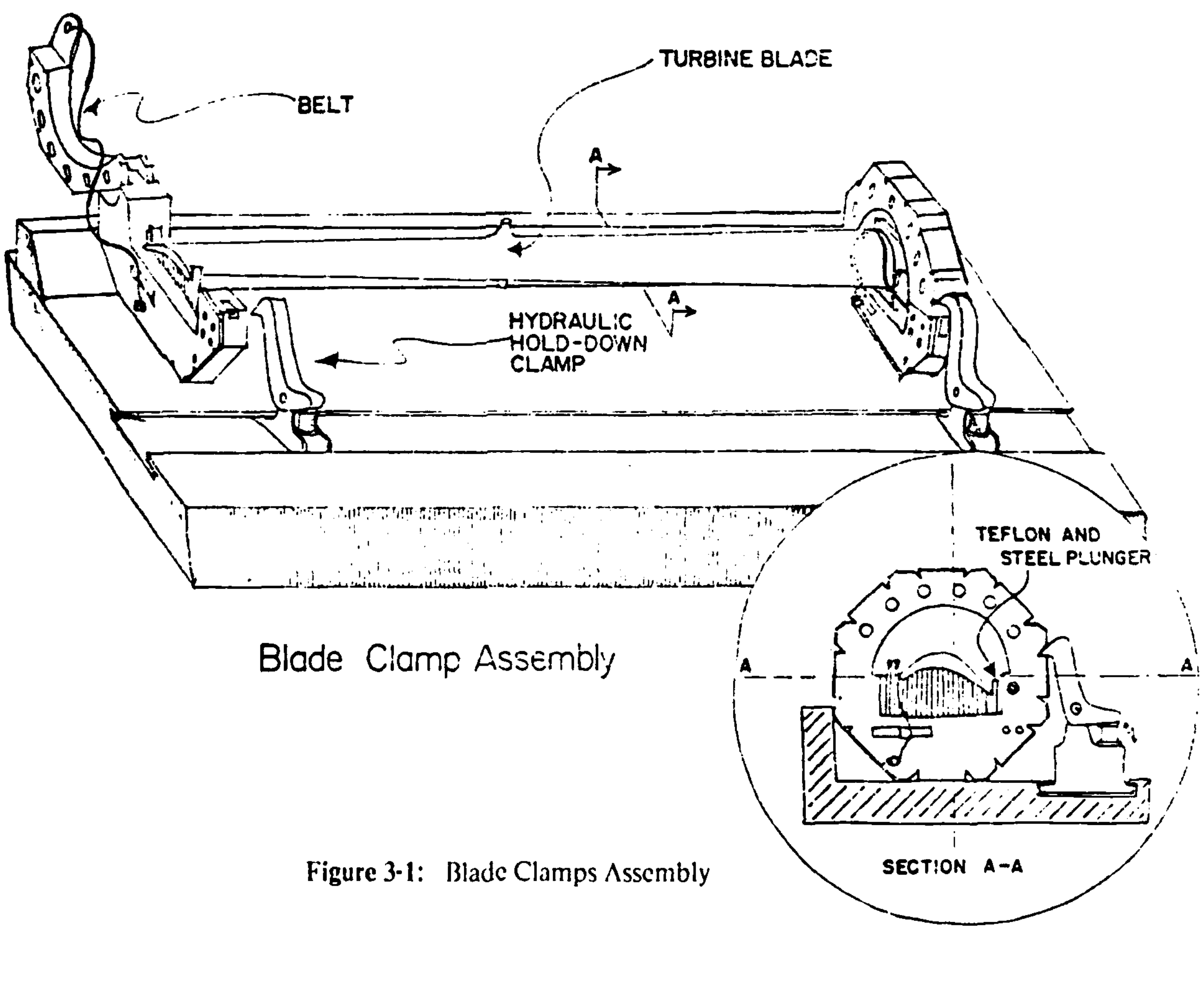

Figure 3-1: Blade Clamps Assembly

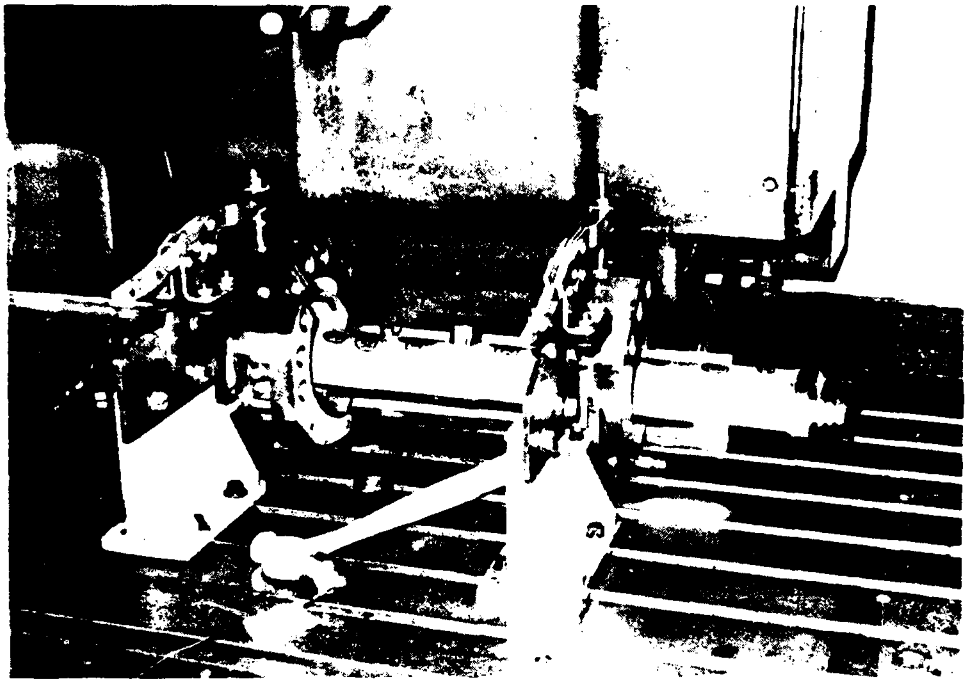

Figure 3-2: Blade Clamps Setup on the NC Machining Center

3.2. Clamp Design

The clamp design shown in Figs.3-3,3-4 eliminates the shortcomings of the clamps being used at present but retains the adaptability of the cast alloy clamp system. The clamps consist of octagonal frames that are hinged so that they may be opened to accept a blade and then closed. The clamps may be placed almost anywhere along the turbine blade. Usually two or three clamps will hold the blade rigidly. Using octagonal shapes rather than a square allows the blade-clamp assembly to assume more orientations about the blade's longitudinal axis. Since the blade cross sections twist substantially from one end of the blade to the other this is a necessary feature to obtain an appropriate clamping angle between a blade and plungers (Fig.3-2).

The lower half of each clamp employs plates or plungers that, when released, are free to conform to the profile of the turbine blade. A similar technique has been used for programmable powder metal dies and for sheet metal forming. [8, 9] A high strength belt is wrapped over the convex surface of the blade and is used to hold the blade against the plungers in the lower half of the clamp. The plungers are forced against the turbine blade using air pressure. Once a profile has been set, the plungers are mechanically locked in place and then air supply may be disconnected. The belt is then tightened and the clamps are free to travel with the blade. If the position of a clamp must be changed during the machining process it is possible to allow a new clamp to conform to the blade in a new location and then to remove the clamp from the old location so that the orientation of the turbine blade is never lost.

To experiment with different plate shapes and sealing materials of the air chamber in the lower half, a small box containing just a dozen steel plungers was first constructed. In the original design, every other steel plate was alternated with a teflon plate. The purpose of the teflon plates was to allow the steel plates to slide freely and to embed any small particle that might gall or scratch the steel plates. Because of the soft and elastic nature of the thick teflon plates (1/16 of a inch), the teflon plates have been abandoned in favor of coating the steel plates with a thin (0.001 of a inch) film of teflon to improve the dimensional accuracy and stability of the clamps. Tests with different sealing materials have shown that loss of air, while the clamps are connected to an air supply, can be kept low. The required air pressure is about 40 psi.

A number of mechanisms for locking the plungers, for locking upper and lower frames and for tightening the high strength friction belt have been investigated. The plungers are locked by a series of set screws with locking pieces inside the air chamber. For locking the upper and lower frames, a *toggle* type clamp is devised and placed at the other end of the hinge pin. The toggle clamp is easy for a robot to operate by one touch action whereas a bolt-and-thread type mechanism amounts to an assembly task for a robot. The belt is reinforced by *Kevlar*[3] cables and is tightened or released by a sliding block moved by a driving screw. Although Kevlar cables are strong, they are not resistant to abrasion. Alternatively, swivel screw clamps may be used instead of the belt for cases in which the edge of a workpiece is very sharp. These mechanisms must be compact, simple and easy to operate automatically. The last requirement is the most difficult to satisfy. In the development so far, a series of set screws and the belt tightening screw may be driven by an air tool held by a small 3 axis robot installed at the clamping station in the cell.

3.3. The Blade-Clamp Assembly

Once two or three clamps have been fastened to a turbine blade, the blade-clamp assembly becomes a pallet in the form of an octagonal prism. The outer dimensions of the assembly are well defined and do not vary from part to part. A single fixturing arrangement can be used on all the machine tools in the cell. Furthermore, there is no need to modify the fixturing when the cell changes between blade styles. Figure3-1 shows a blade held by two clamps which are mounted on a fixturing plate. Standard hydraulic hold-down clamps are used to fix the assembly in place while it is machined.

The robot which transports the blade-clamp assembly also benefits from a standard octagonal shape. The robot does not need to hold the assembly as rigidly as the machine tools, just firmly enough to keep the assembly from slipping while it is being transported. The robot can therefore use a simple gripping arrangement.

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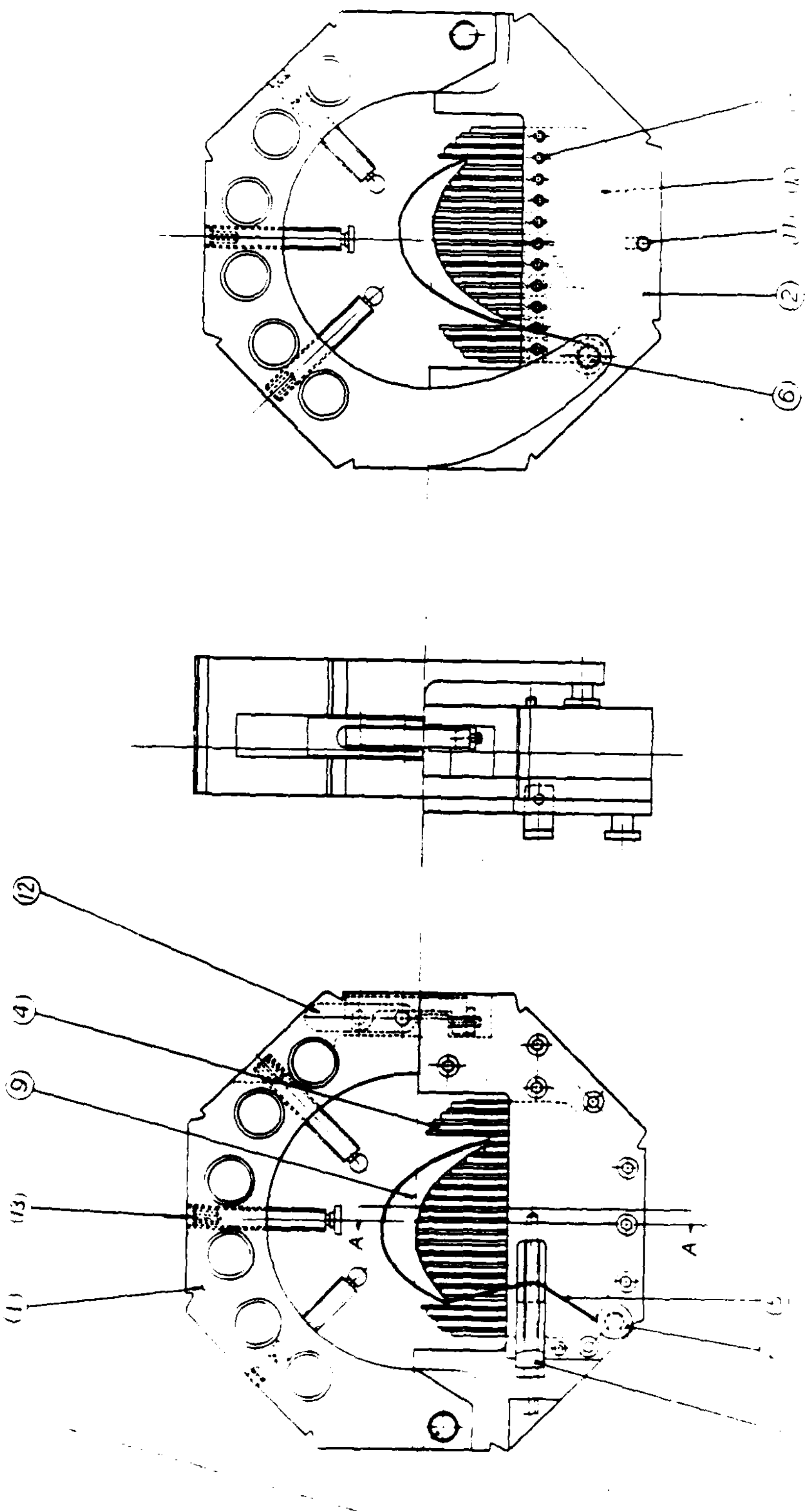
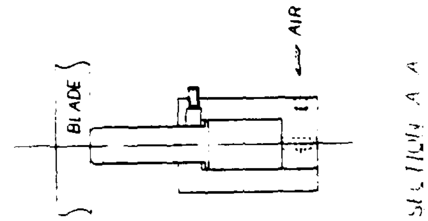
[3] Kevlar is a registered trademark of Du Pont.

Figure 3-3: Flexible Conformable Clamps - Side View



Key to Illustrations:

1. upper frame of clamp
2. lower frame of clamp
3. belt tightening mechanism
4. steel plungers (teflon coated)
5. high strength belt
6. belt termination
7. clamp hinge
8. locking screw
9. turbine blade section
10. air chamber
11. air supply connection
12. toggle locking mechanism
13. swivel clamping screw



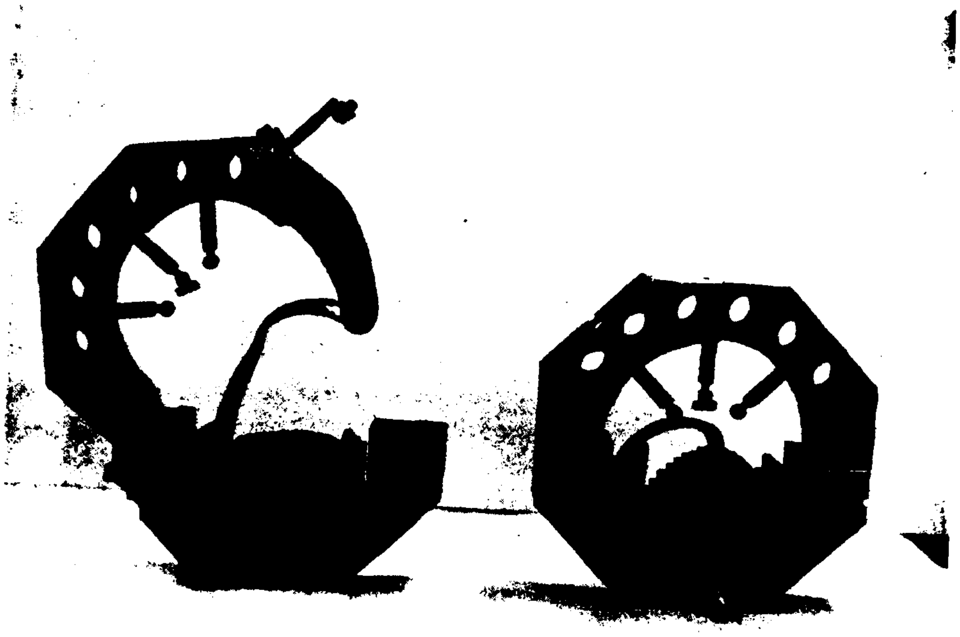


Figure 3-4: Photograph of The Flexible Conformable Clamp

3.4. Pre-configuration the Clamps

The discussion thus far has focused on the conformable nature of the clamps. Once a turbine blade is brought into alignment the clamps will conform to its profile. At other times, when starting a new batch of blades for example, it is desirable to have the clamps pre-configured to the correct profile for the new part. To give them this profile the use a "master profile" has been proposed [6]. The master profile would consist of plates or plungers that are automatically driven to a desired position using stepping motors. The desired profile would be specified by a computer which would rely upon a database containing the geometrical representation of the turbine blade. The individual conformable clamps would then be held against this master profile and adopt to its shape. Since the master profile would not travel with the parts it would not have to be compact or lightweight. The process of adjusting the master profile would not have to be a rapid one since it would not occur on line. While the cell was busy machining blades from a current batch, the master profile would be adjusted and additional clamps would be held against it to prepare them for the new part shape.

4. Vibration Characteristics

Although the main objective of this research is to develop a clamping system for an unmanned flexible machining cell, it is also important to understand the vibration characteristics of the designed clamp with a blade during metal cutting operation.

In the practical turbine blade machining process large vibrations are often encountered. These give rise to undulations on the machined surface and excessive variations of the cutting force which endanger the life of the tool and of the machine. Therefore cutting conditions are chosen lower than the desired value and additional pads or fixtures are applied to increase stiffness (Fig.2-2).

As a result of the various kinds of forces which occur during the machining operation, various specifications of requirements on stiffness may be stated within the following three categories.

1. Deformations caused by cutting forces

2. Vibrations excited by cutting forces

3. Self-excited vibrations - Chatter

(1): During the cutting operation the cutting force varies and its point of application moves. In consequence, the deformation of the workpiece and/or the frame of the machine will vary, causing deviations of the form of the machined surfaces. This effect may be limited by decreasing the cutting conditions and consequently the output of the operation. In this particular blade machining process, it is assumed that the least stiff component of the system tool-machine-workpiece is the blade, since we usually choose a machine tool having a sufficiently rigid spindle. Then the important feature is to know the dynamic stiffness of the blade-clamp system. In general, experience shows that once a cutting condition without chatter is satisfied, deformations caused by cutting forces are also within the desired value [10].

(2): Forced vibrations are now considered. This problem was studied by Tlusty [10] etc. to analyze the vibration characteristics of machine tool structures. The key feature to prevent vibrations excited by the cutting force is to choose cutting speed that do not coincide with the natural frequency of the system tool-machine-workpiece. If cutting speeds in the range 200 - 700 ft/min used for carbide cutters when milling steel and cutters with a comparatively fine pitch of 2 inches are assumed, the tooth frequency will be in the range 20 - 70 Hz. In general, the significant modes of relative vibration between tool and workpiece in milling machines have natural frequencies in the range 150 - 300 Hz. Therefore, the basic harmonic component of the cutting force acts far from resonance with the significant modes of the structure. The particular results for this turbine blade machining are discussed in the later section.

(3): Self-excited vibration - Chatter is discussed detail in the following subsection.

4.1. Theoretical overview of Chatter

4.1.1. A single-degree-of-freedom vibrating system

A single degree-of freedom system is introduced to obtain the basic idea and to define basic variables. Diagrammatically such a system may be illustrated by Fig.4-1, as a mass attached to a massless beam subject to bending. [Much of the theoretical work reviewed below has been based on the texts by Koenigsberger and Tlusty [10] and Den Hartog [11] and the reader is refered to these for fuller descriptions.]

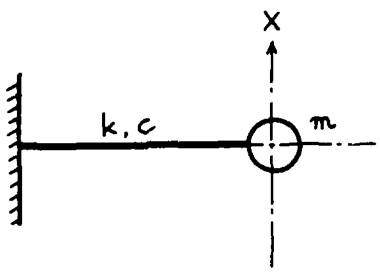


Figure 4-1: Single degree-of-freedom system

Movement of mass m, which is assumed possible only one direction denoted (X), is resisted by a force resolvable into two components, one proportional to the displacement and the other proportional to the velocity of the mass. The governing equation which describes this system is given :

$$mx + cx + kx = p \tag{1}$$

where m, c, k, p are mass, damping coefficient, stiffness and external force respectively. If deflected by X_0 and released, and if damping is small, the mass m performs natural vibrations described thus:

$$x = X_0 e^{(-\delta + j\nu)t} \tag{2}$$

The frequency of this vibration is:

$$f_n = \frac{1}{2\pi}\nu = \frac{1}{2\pi}\sqrt{\Omega^2 - \delta^2} \tag{3}$$

where $\Omega = (k/m)^{1/2}$ is the circular natural frequency of the undamped system and $\delta = c/2m$.

The concept of the relative damping

$$d = c/c_c = \delta/\Omega \tag{4}$$

is used to express the ratio of the damping coefficient c to the critical value of $c_c = 2(km)^{1/2}$ for which the movement of the system would just cease periodic.

If an input force $p = Pe^{j\omega t}$ acts on the mass in the direction (X), the system performs steady vibrations:

$x = Xe^{j\omega t}$ where $X = Ae^{j\varphi}$.

The amplitude of the output vibration is A and the phase shift between the applied force and the resulting displacement is φ, both these quantities being implicit in the factor X. Substituting x and p into the equation (1), X for harmonic excitation with frequency ω is obtained.

$$X = \frac{P}{k} \cdot \frac{\Omega^2}{\Omega^2 - \omega^2 + 2j\delta\omega} = P \cdot \Phi(\omega) \tag{5}$$

The function $\Phi(\omega)$ is called the ***receptance*** or dynamic compliance (inverse of dynamic stiffness) of the system. Its absolute value $F(\omega)$ is expressed in Fig.7-1 in units of $1/k$, for various values of d. The real part of Φ, as a function of ω, is expressed by:

$$\mathrm{Re}(\Phi) = G = \frac{1}{k} \cdot \frac{\Omega^2(\Omega^2 - \omega^2)}{(\Omega^2 - \omega^2)^2 + (2\delta\omega)^2} \tag{6}$$

The imaginary part of Φ is given by:

$$\mathrm{Im}(\Phi) = H = -\frac{2}{k} \cdot \frac{\Omega^2\delta\omega}{(\Omega^2 - \omega^2)^2 + (2\delta\omega)^2} \tag{7}$$

The phase shift φ of X with respect to P is given by:

$$\tan\varphi = \frac{H}{G} = -\frac{2\delta\omega}{\Omega^2 - \omega^2} \tag{8}$$

Expressions (6), (7) are called real and imaginary receptance respectively. The real receptance, the imaginary receptance and the phase shift are indicated in Figs. 7-2, 7-3, 7-4.

Significant points of the receptance curves appear at the following values of the frequency of the exciting force:

1. $\omega = \Omega$, phase shift is exactly $\varphi = -\pi/2$.

2. $\omega = \Omega\sqrt{1-2d^2}$, or approximately $\omega = \Omega(1-d^2)$
maximum of the absolute receptance, F_{max}.

3. $\omega = \Omega\sqrt{1-d^2} = \nu$
(frequency of natural vibrations of the system) no special significance for response curves.

4. $\omega = \Omega\sqrt{1 \pm 2d}$, or approximately $\omega = \Omega(1 \pm d)$

maximum and minimum points of the real receptance $G_{max,min}$.

$$G_{max} = \frac{1}{4d(1-d)k}, \quad G_{min} = \frac{1}{4d(1+d)k}$$

5. $\omega = \Omega \frac{1}{\sqrt{3}} \sqrt{1-2d^2+2\sqrt{1-d^2+d^4}}$, or approximately $\omega = \Omega\sqrt{1-d^2} = \nu$,

minimum of the imaginary response $H_{min} = -1/2dk$.

If a real receptance curve is given the damping ratio d is calculated by:

$$d \simeq \frac{\omega_{min} - \omega_{max}}{2\Omega} \tag{9}$$

where ω_{min} and ω_{max} are frequencies at which G attains its minimum and maximum respectively.

4.1.2. Systems with many Degrees-of-freedom

For the general case, systems with many masses, located quite generally in three-dimensional space, partly interconnected with many springs, may be assumed. A two-dimensional diagrammatical representation of one such system is shown in Fig.4-2. This system may have many natural frequencies, Ω_i, damping factors δ_i and corresponding modal shapes. To every modal shape there corresponds a particular direction (X_i) in which a particular point, say a, of the system vibrates in the given mode. If a variable excitation force $p = Pe^{j\omega t}$ acts on one point a of the system, and if a particular direction (Y) is chosen, the angle between (Y) and P being β and the angles between (Y) and individual modal directions (X_i) being α_i, then vibration of the point a in the direction (Y) will be:

$$y = Ye^{j\omega t}$$

where

$$Y = P\sum_{i=1}^{i=n} \cdot \frac{u_i}{k_i} \cdot \frac{\Omega_i^2}{\Omega_i^2 - \omega^2 + 2j\delta_i\omega} \tag{10}$$

The "directional factors u_i" are :

$$u_i = \cos\alpha_i \cos(\alpha_i - \beta) \tag{11}$$

and k_i are stiffnesses corresponding to the individual modes and to the chosen point a in the system, for the case in which the individual modes would be excited by a harmonic force acting on point a in the directions (X_i) of the individual modes respectively. Equation (10) may be written using F, G and H, as:

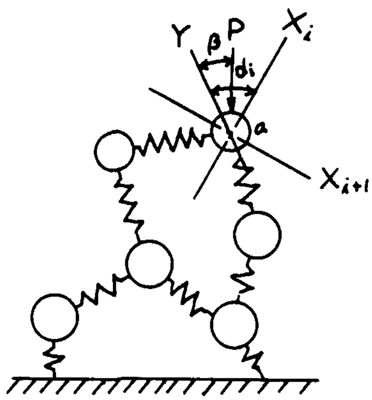


Figure 4-2: Diagram of system with many degrees-of freedom

$$Y = P\sum_{i=1}^{i=n} u_i G_i + ju_i H_i = P(G + jH) = P\Phi \tag{12}$$

where

$$G = \sum_i^n u_i G_i \text{ and } H = \sum_i^n u_i H_i \tag{13}$$

where G_i and H_i are of the forms by eqs. (6) and (7) respectively.

It is to be understood that the partial real and imaginary responses G_i and H_i of the individual modes correspond to cases where the individual modes would be excited in their corresponding directions (X_i). They are direct-receptances. Functions Φ, G and H are called *cross-receptances.*

The system tool-workpiece-machine has infinite number of degrees- of-freedom. The higher the natural frequency of the mode, the higher the value of its stiffness and also the value of damping is usually higher in higher modes. Therefore, the higher the mode the smaller is its participation in the resulting vibration. It follows that for practical significance of metal cutting vibration all but several lower modes of the system can be neglected.

4.1.3. Basic theory of Self-excited Vibration in Metal Cutting

The basic diagram of the process of self-excited vibration in metal cutting is presented schematically in Fig.4-3. It is a closed loop system including two fundamental parts, the cutting process and the vibratory system of the machine. It indicates that the vibration Y between tool and workpiece influences the cutting process so as to cause a variation P of the cutting force which, acting on the vibratory system of the machine, creates again vibration Y.

The relationship expressed in Fig.4-3 is written thus:

$$P = -RY \quad \text{or} \quad P = -brY$$

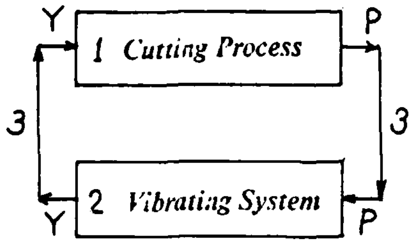


Figure 4-3: Basic diagram of chatter

(14)

The coefficient R expresses the intensity of the coupling between vibration and cuting force, or the "gain" in this part of the loop. The coefficient b is the chip width and r is a coefficient, depending on all other cutting conditions (b, r are real positive). If the amplitude of vibration in the direction of the normal to the cut surface in the i th cut was Y_0 and in the $(i + 1)$th cut it is Y, then the amplitude of chip thickness variation is $(Y - Y_0)$. This model is shown in Fig.4-4.Consequently, instead of eq.(14), eq.(15) must be used for actual metal cutting operation:

$$P = -br(Y - Y_0) \tag{15}$$

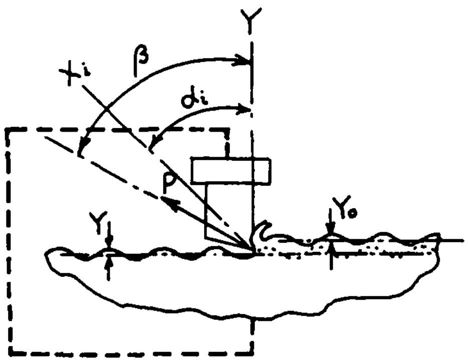


Figure 4-4: Basic diagram for the process of self-excited vibration

Equations (12) and (15) can be combined into

$$Y/\Phi = -br(Y - Y_0) = br(Y_0 - Y) \tag{16}$$

and, after modification

$$\frac{Y_0}{Y} = \frac{1/br + \Phi}{\Phi} \tag{17}$$

Equation (17) describes the closed-loop system of self-excited vibrations. If the amplitude Y of vibrations in a cut is larger than the amplitude Y_0 of vibrations in the preceding cut, the system will be unstable. The condition for the limit of stability can be expressed as:

$$|q| = \left|\frac{Y_0}{Y}\right| = \left|\frac{1/br + \Phi}{\Phi}\right| = 1 \tag{18}$$

which signifies that Y will be equal to Y_0. Equation (18) will be transcribed according to (12) to

$$|q| = \left|\frac{1/br + G + jH}{G + jH}\right| = 1 \tag{19}$$

Since values b and r are real positive, the value of $1/br$ is real and positive. Thus the vectors in the numerator and in the denominator of q in eq.(19) have the same imaginary part jH. Equation (19) requires that the moduli of the vectors in the numerator and in the denominator of q be equal. Then the absolute values of the real parts of both vectors are equal.

$$|q'| = \left|\frac{1/br + G}{G}\right| = 1 \tag{20}$$

Condition (20) can be satisfied only if:

$$1/br + G = -G \text{ or } 1/2br_{\text{lim}} = -G(\omega_{\text{lim}}) \tag{21}$$

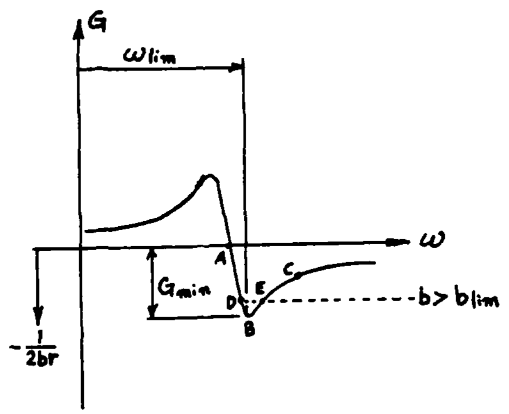


Figure 4-5: Real cross-receptance G and its minimum

Equation of (21) is the form of the condition for the limit of stability. If the real receptance curve $G(\omega)$ was given and r was constant, the maximum chip width b_{lim} for which cutting can be stable, is obtained as Fig.4-5. Assuming the larger chip width (the value of - 1/2br is larger than point B), the chatter could occur at any points between D and E.

4.2. Experiments and results

The aim of the experiments carried out so far was to find the damped natural frequencies of the clamped turbine blade. A piezoelectric accelerometer was mounted on the blade and an impulse input was given by "impulse response testing" (a sharp hammer blow). The load cell which was mounted on the hammer triggered the measurement of the output response from the turbine blade. The output waveforms in the time domain were measured and recorded by a Nicolet Digital Osciloscope which was equipped with an I/O interface with a floppy disk drive device. The stored response waveforms were directly transferred to the VAX 11/750 computer of the Mechanical Engineering Department and analyzed by the Fast Fourier Transformation (FFT) method to find the primary peak of the acceleration amplitude in the frequency domain. The frequency value at the primary peak is significant because it represents the damped natural frequency of the clamped blade system. The value of acceleration amplitude is significant only in each individual experiment because an impulse input is given by a manual hammer blow and the resultant acceleration amplitude is not normalized by an impulse input.

A blade of 22 inches length was set up on the two conformable clamps 12 inches apart. The whole blade-clamps assembly was then set on the work table of a Brown & Sharpe vertical spindle CNC machining center (Fig.3-2). The vibration amplitudes were measured for both belt clamps and swivel screw clamps. The vibration was also measured for the same blade in the cradle type fixture with rigid fixed clamps (Fig.2-3) of 12 inches clamping span and the results were compared with the new conformable clamps cases.

The results are shown in Fig.7-5, Fig.7-6 and Fig.7-7 as frequency versus acceleration amplitude plots. In the cradle type fixture with rigid clamp forms (Fig.7-7) a very sharp peak of the acceleration amplitude is observed at 490 Hz, and three other minor peaks of less than one third of the primary amplitude are observed at 320 Hz, 1530 Hz and 2670 Hz. On the other hand, in the new conformable clamps (Fig.7-5) the primary peak is obtained at 1000 Hz and minor peaks are observed at 280 Hz, 1540 Hz and 2900 Hz. In comparison with the above two results, the primary sharp peak at 490 Hz in the cradle fixture has disappeared in the newly developed conformable clamps. This fact illustrates another advantage of the new conformable clamps over the heavy cradle fixture. For the case of using swivel screws instead of belts Fig. 7-6, the primary peak is at 1590 Hz and the secondary peak is at 610 Hz. The primary peak at 490 Hz has also disappeared in this case.

Considering the typical machining conditions shown in section 2.2, frequencies of external cutting forces applied to the blade are the range from 5 Hz (pre-finish milling of the root) to 200 Hz (contour milling at the root of the stub). If a roughing end mill cutter were used a higher frequency could easily attained. For example a roughing end mill of 2 millimeters tooth pitch was selected for side edge milling and cutting speed was set as 64 m/min, the frequency of this cutting force is 533 Hz. There is a possibility that the frequency of the external cutting force could coincide with the natural frequency of the clamped blade in the cradle fixture with rigid clamp forms. We may conclude that the cradle type fixture is not appropriate for this operation.

In the experiments described so far, the clamping span was chosen to be 12 inches in order to compare the natural frequency of the conformable clamps with that of the cradle type rigid fixed clamps. For the conformable clamps, clamping positions can be easily changed to the desired span. To demonstrate the effect of changing the clamping span a span of 16 inches was tested. The result is shown in Fig.7-8. In this case, the primary peak is observed at 245 Hz which is lower than the 12 inches span case as expected.

5. Conclusion

This research is concerned with the development of flexible conformable clamps for a steam turbine blade machining cell. The design presented here has the flexibility to conform various complex shapes of workpieces by introducing plungers and high strength belts. Additional orientational flexibility is provided by the octagonal shape of the clamps. These flexibilities, together with light-weight design of the clamps, make it possible to construct an unmanned workpiece setup station. A number of interesting possibilities are occur at this point. For example, the setup station can be combined with an inspection station. A non-contact measuring device mounted on a precision industrial robot could serve for this purpose.

The turbine blade clamped by the conformable clamps has good characteristics for *forced-excited* vibrations. The basic harmonic component of the cutting force acts far from resonance. In future research, the modal analysis and displacement measurements will be repeated in order to find the performance against *self-excited* vibrations (chatter).

6. Acknowledgements

This research was conducted under the direction of Professor Paul K. Wright of the Mechanical Engineering Department, Carnegie-Mellon University. The original concept of the clamp design was sketched out by Mr. Mark R. Cutkosky and the development proceeded with his discussion. The author thanks Mr. Robert Milligan Jr. for his providing FFT programs for the vibration analysis work. The author also thanks Mr. J. R. Dillinger and Mr. D. T. McKeel for their advice and assistance in producing the clamps. The research environment was provided by Westinghouse Electric Corporation. The author's educational opportunity was given by Mitsubishi Heavy Industries, Ltd.

7. Appendix

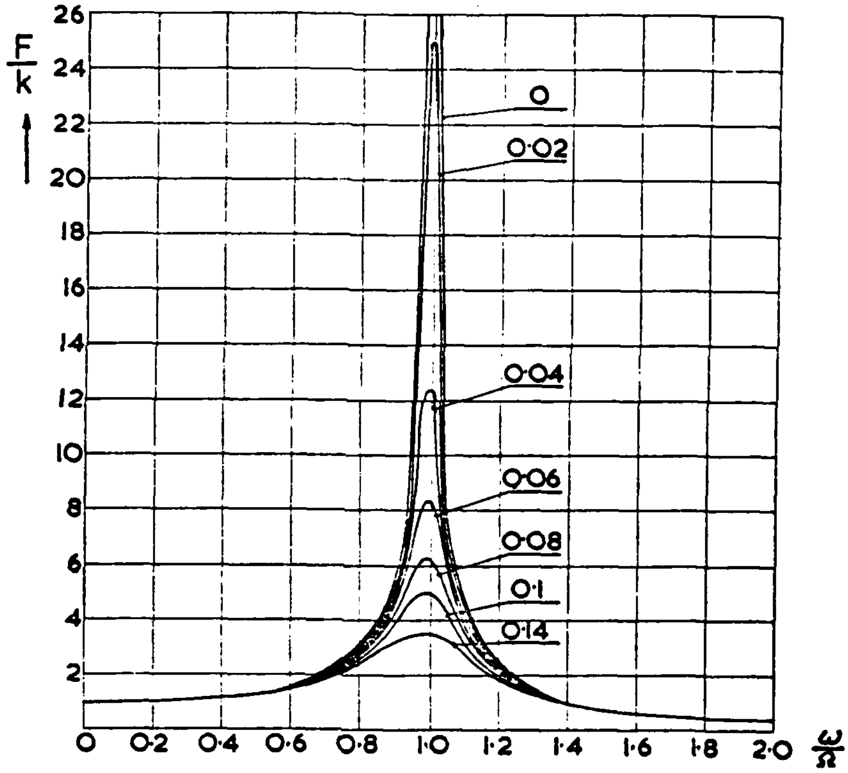


Figure 7-1: Absolute receptance - from Tlusty

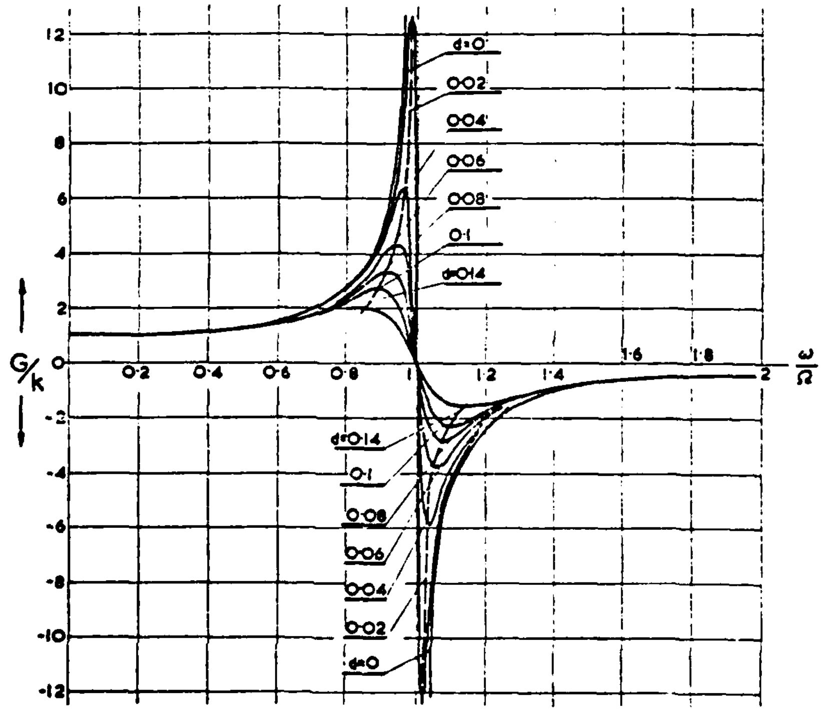


Figure 7-2: Real receptance - from Tlusty

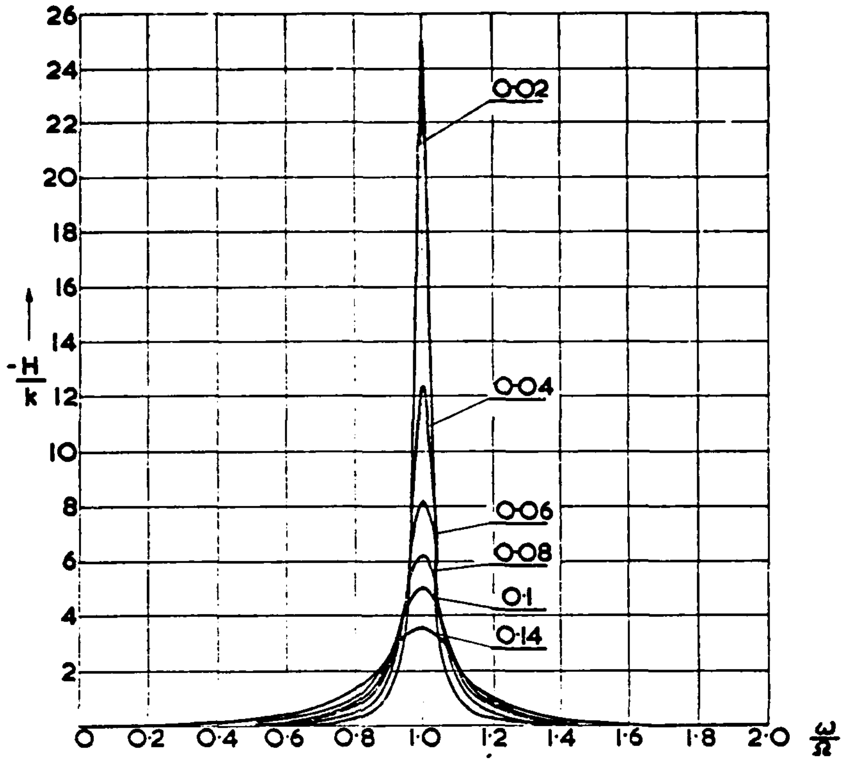


Figure 7-3: Imaginary receptance - from Tlusty

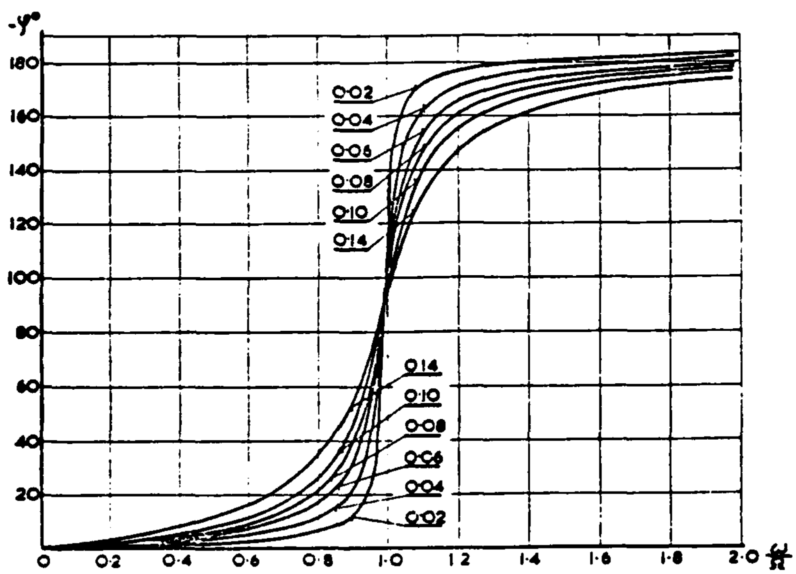


Figure 7-4: Phase shift between force and vibration -from Tlusty

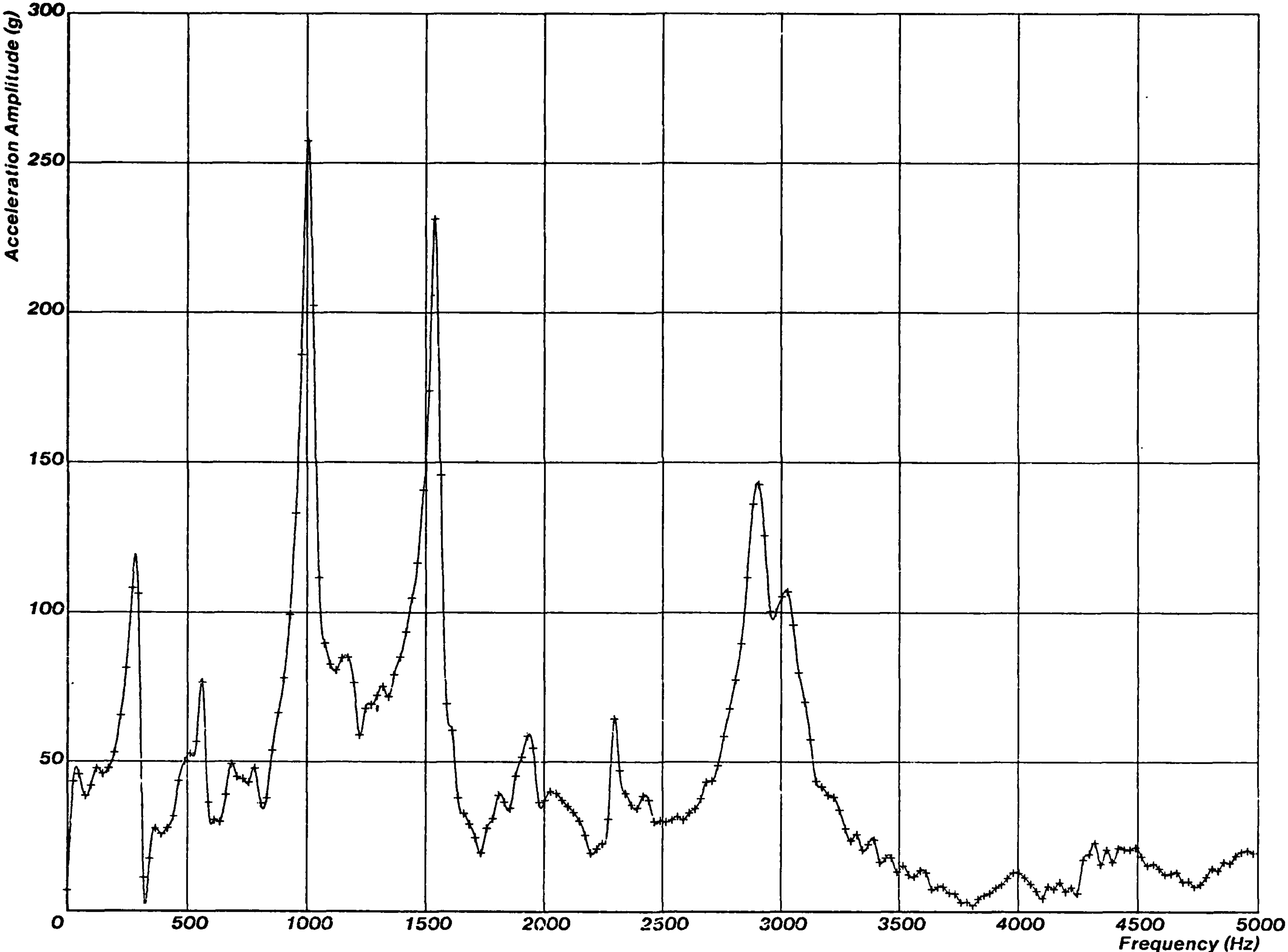


Fig. 7-5 Natural Frequency of the Blade (Conformable Clamps) - Belt Clamp

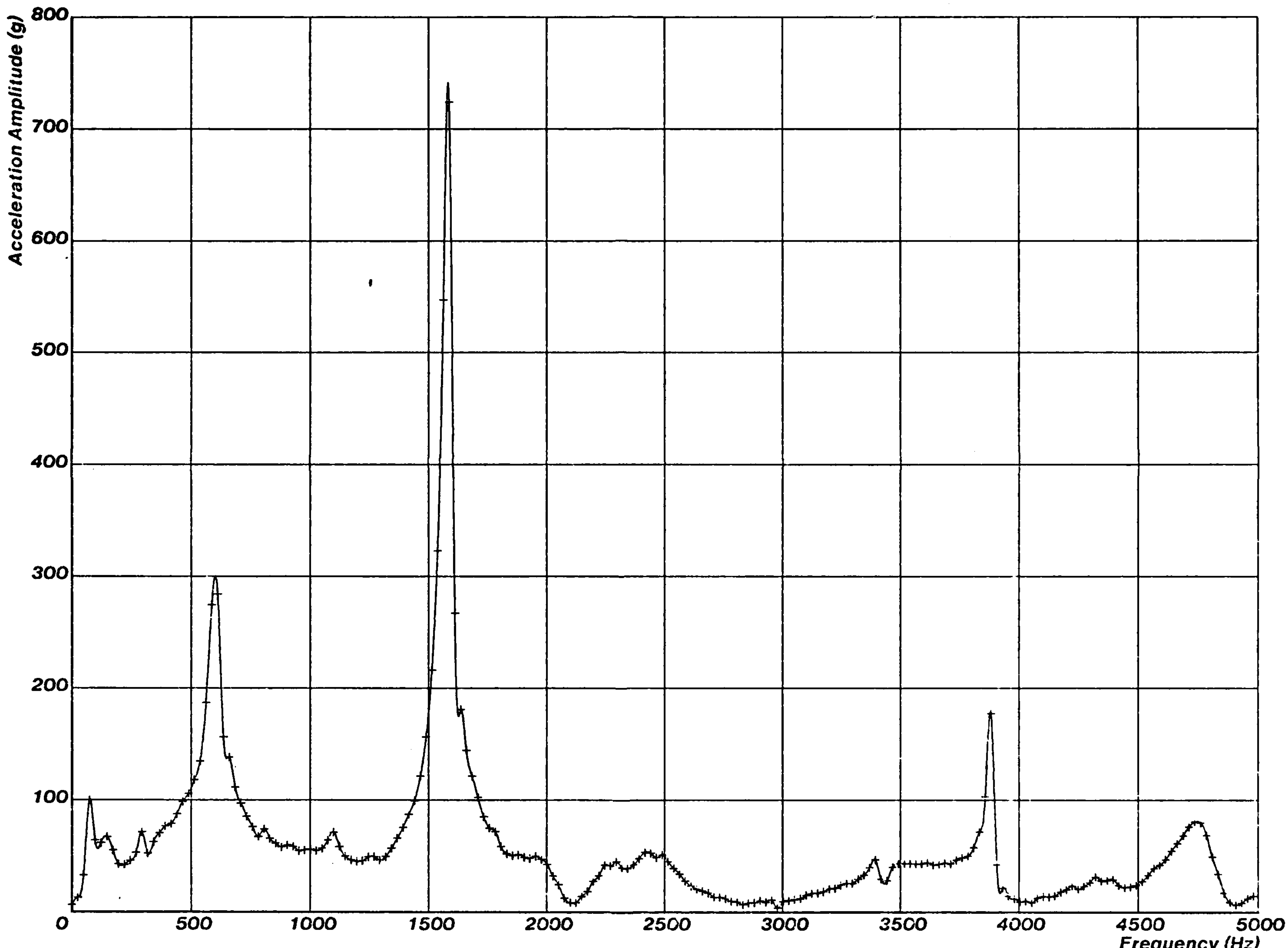


Fig. 7-6 Natural Frequency of the Blade(Conformable Clamps) - Screw Clamps

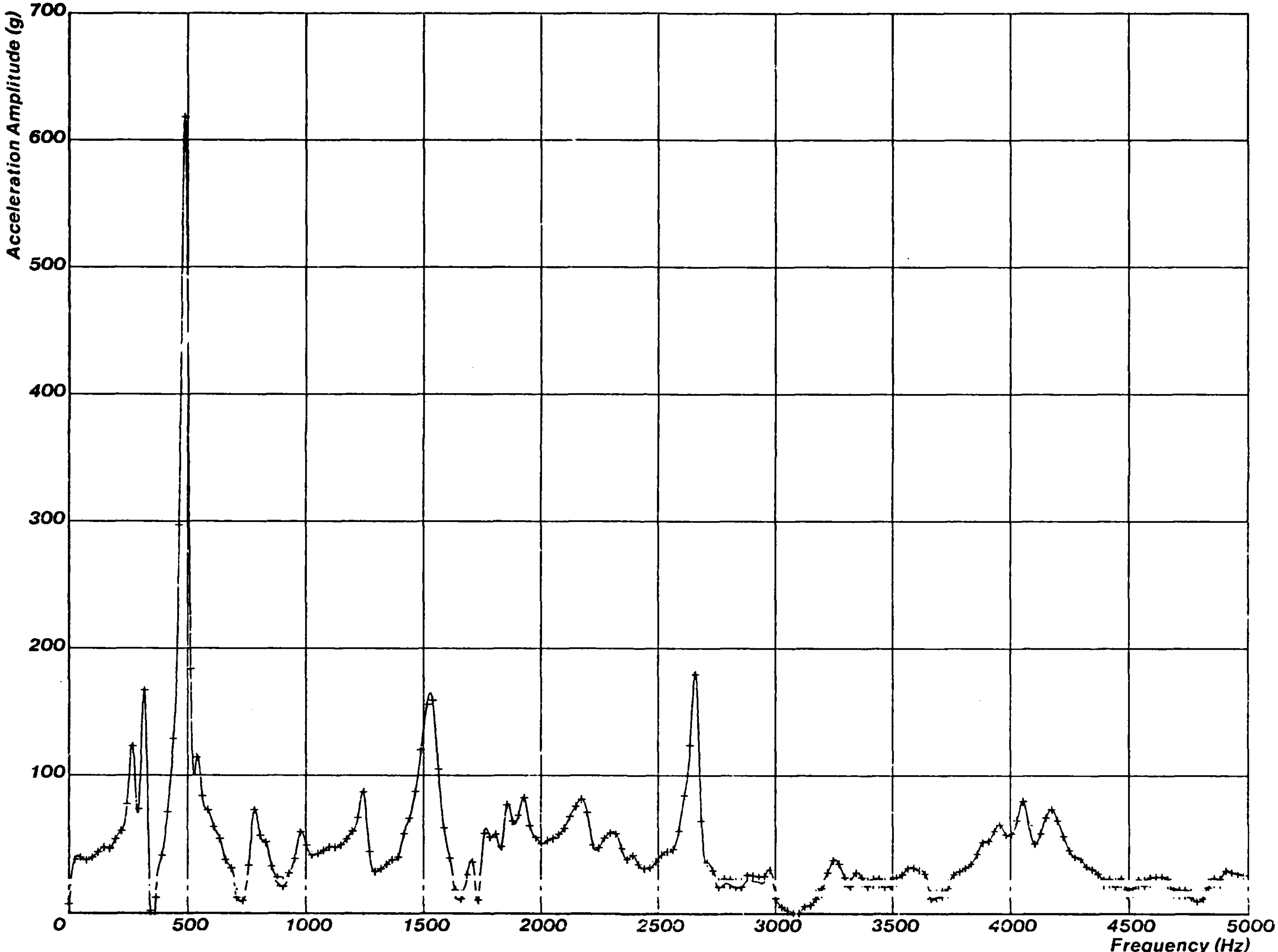


Fig. 7-7 Natural Frequency of the Blade(Cradle Clamp) - Rigid fixed profile

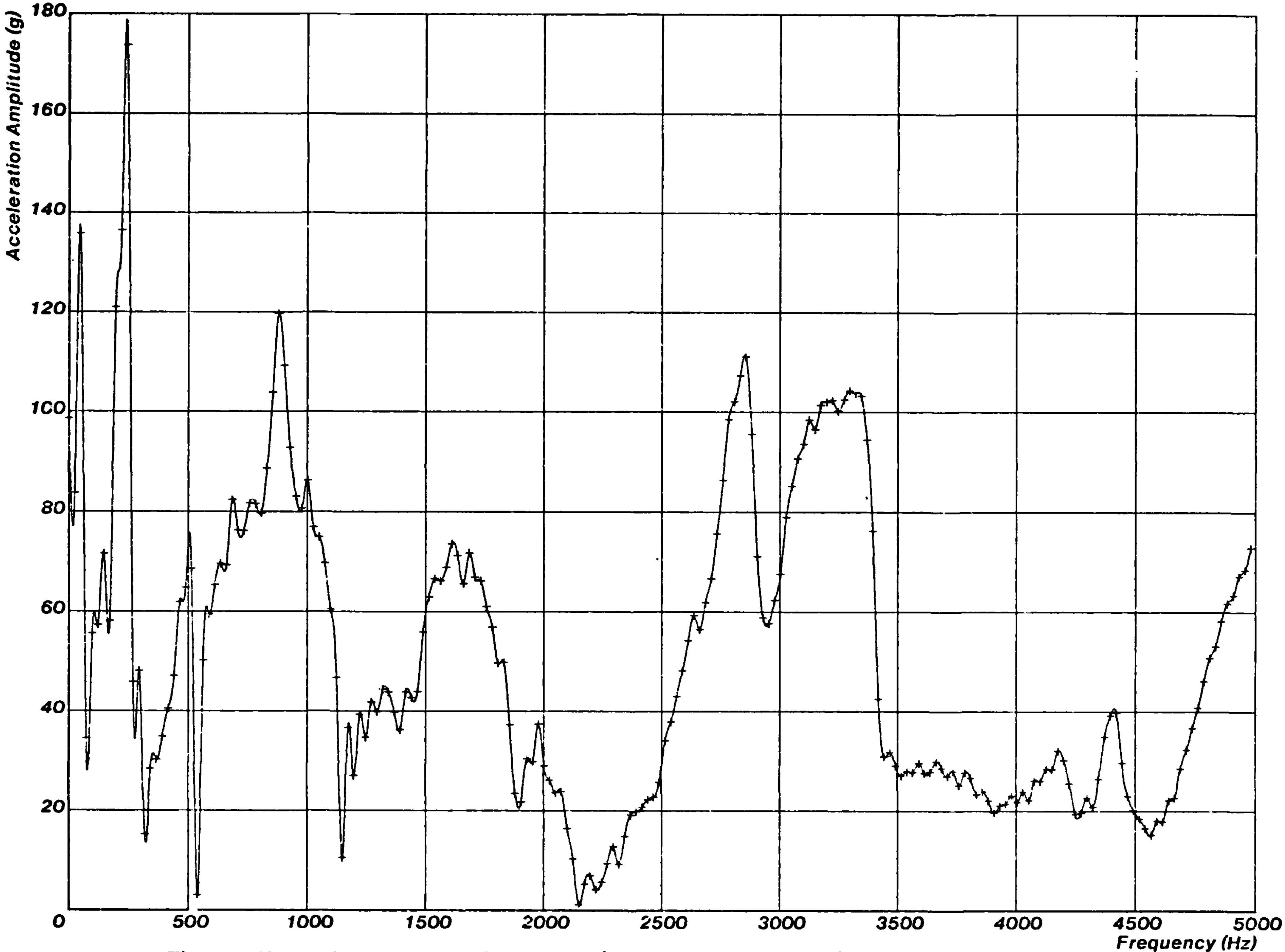
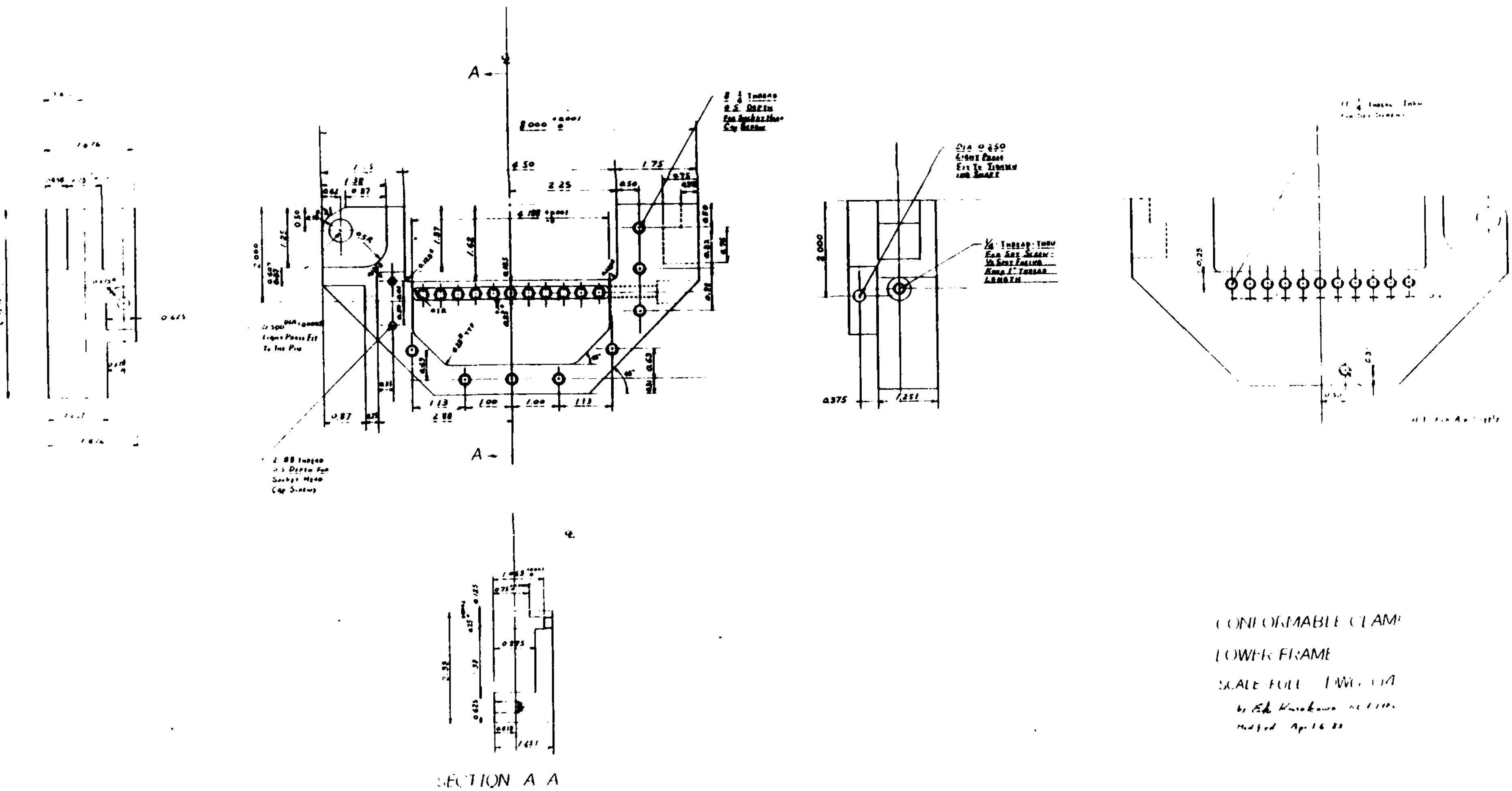


Fig. 7-8 Natural Frequency of the Blade(Conformable Clamps) - Belt Clamps-16 in. span

NOTE

1. 2 PIECES REQ'D



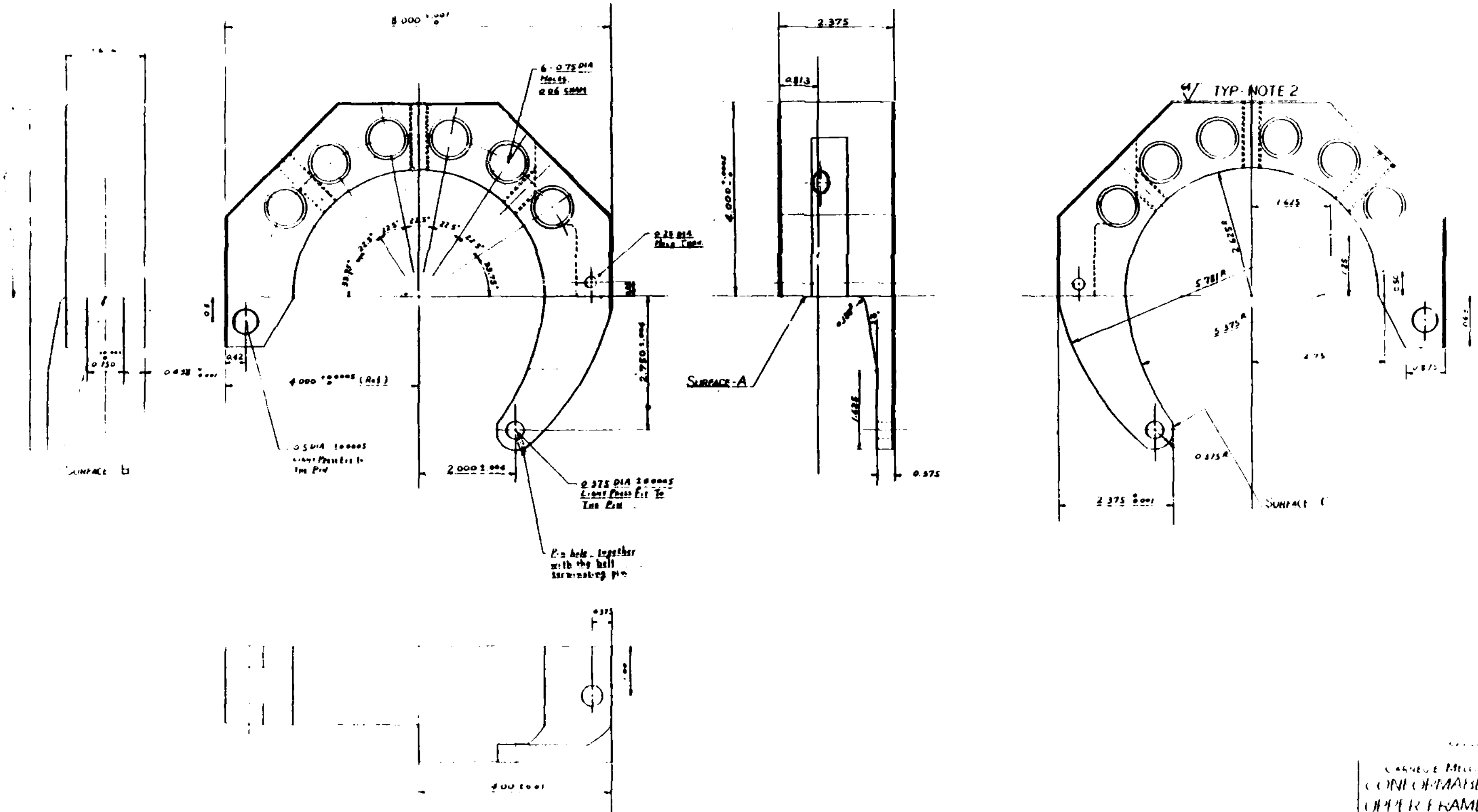
NOTES (2 PIECES REQUIRED)

1 Surface A & B should be on the same plane, which is the horizontal joint surface of upper and lower halves

2 Outer surfaces of octagon should be [illegible] remaining [illegible] stock [illegible] and [illegible] after assembling upper and lower halves [illegible] pin

3 Surface C should be [illegible] after outer surface grinding

4 2 Pieces Req'd

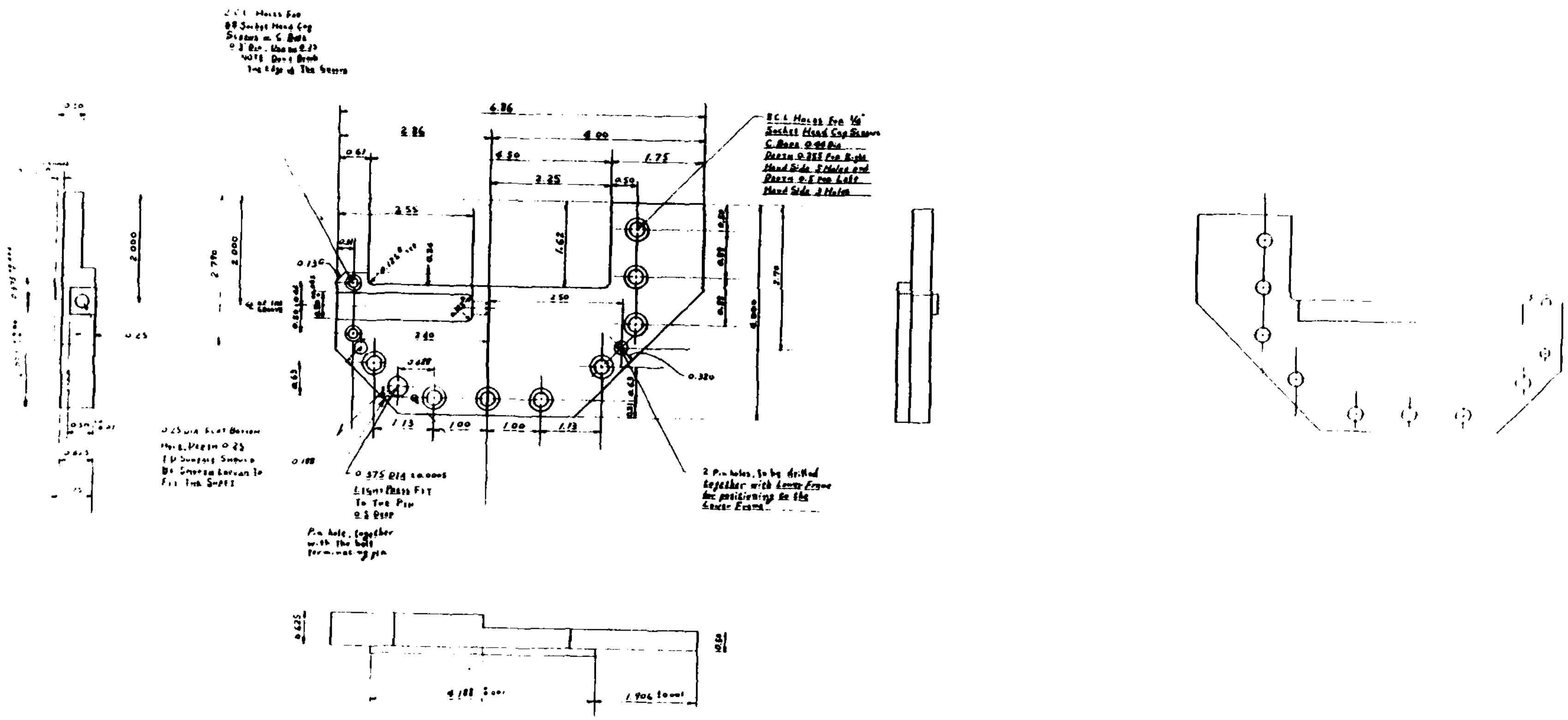


CARNEGIE MELLON UNIVERSITY
CONFORMABLE [illegible]
UPPER FRAME
SCALE FULL
[illegible]

NOTE

1 2PIECES REQ'D

2 Outer surface of octagon should be machined [illegible] ground after assembly



CONFORMABLE [illegible]

LOWER FRAME COVER

SCALE FULL DWG [illegible] 5

by [illegible]

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