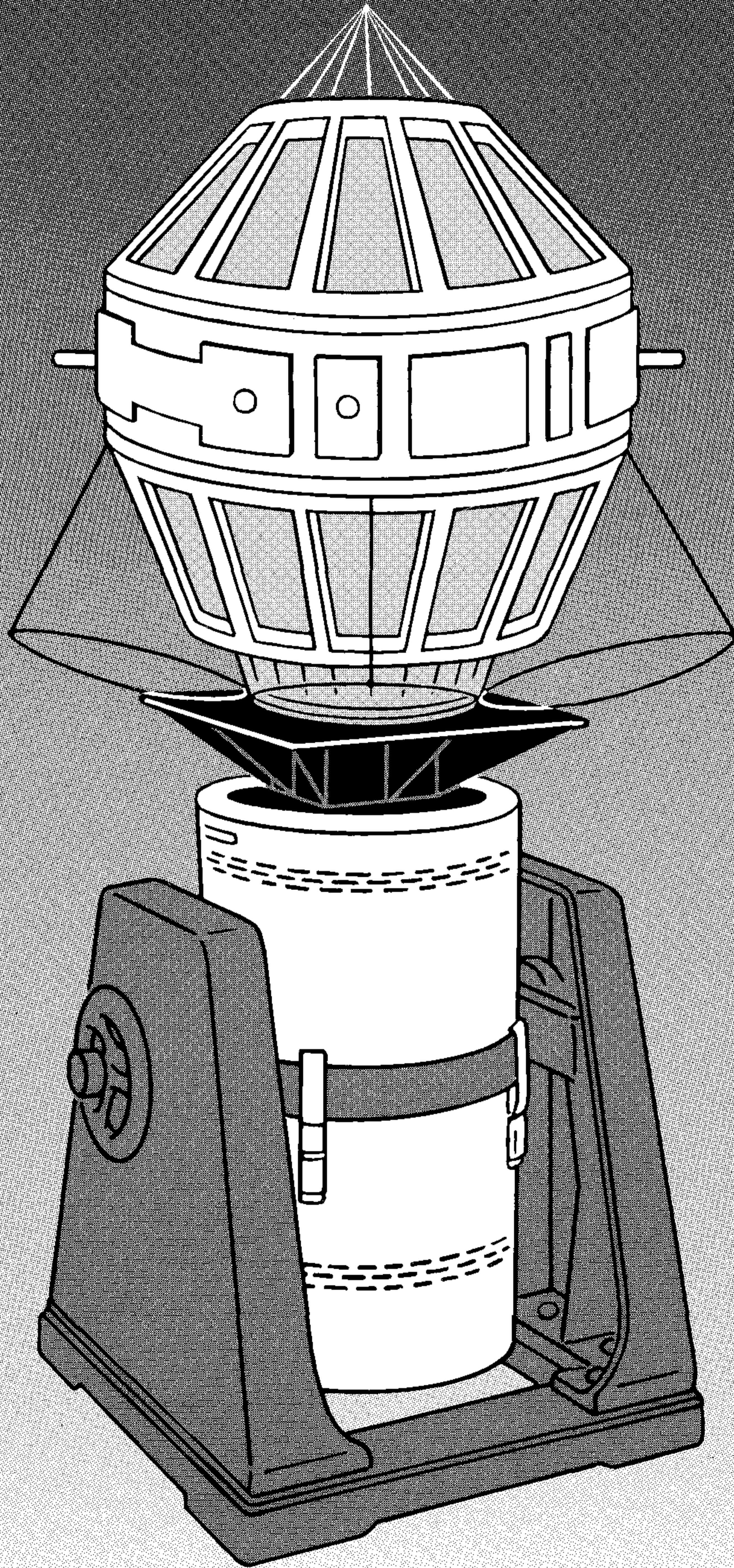


Fixtures for B&K Exciters



Brüel & Kjær 

FIXTURES FOR B & K VIBRATION EXCITERS

Revised October 1987

CONTENTS

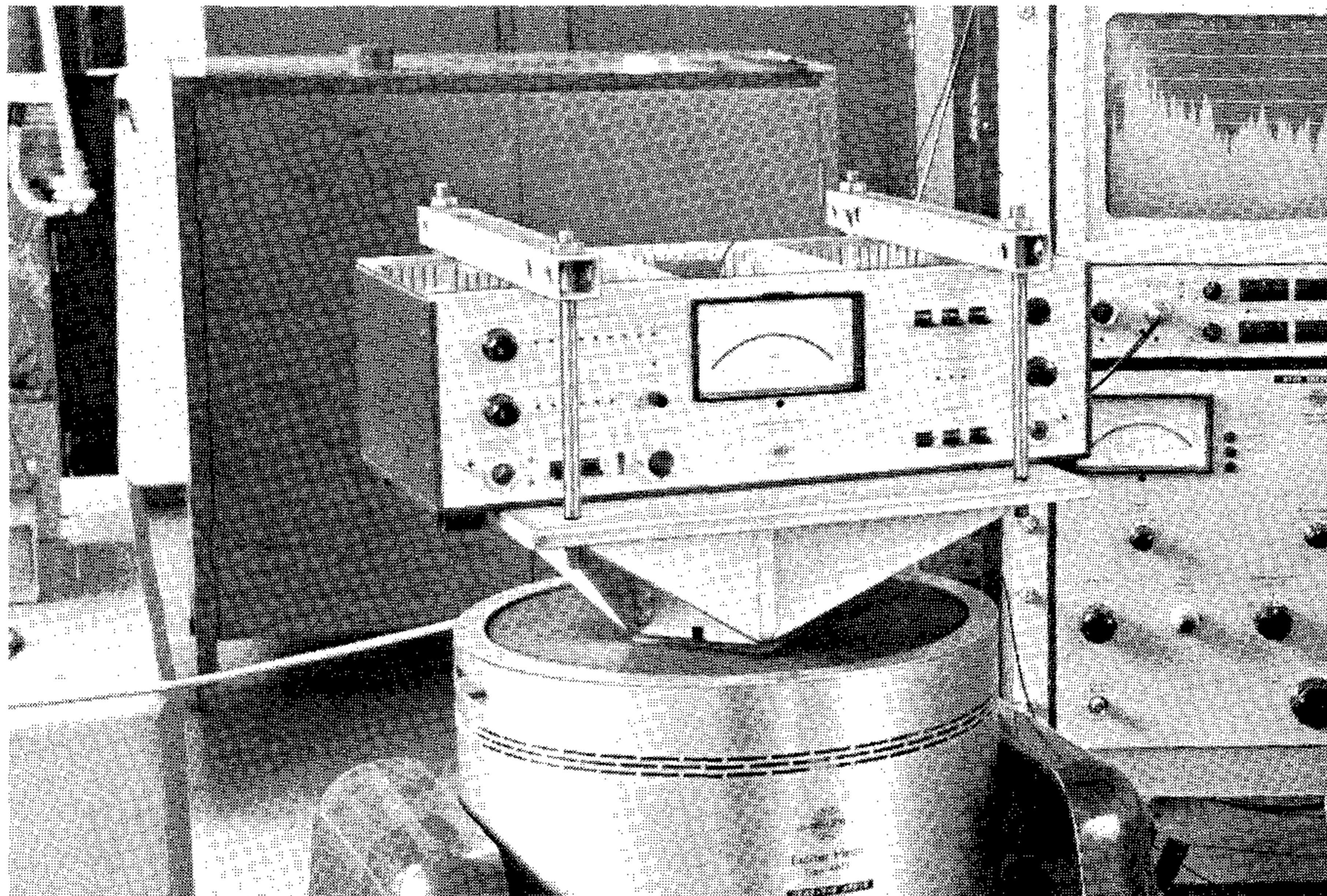
1. INTRODUCTION	1
2. DESIGN CONSIDERATIONS	2
2.1. GENERAL	2
2.2. MATERIALS AND FABRICATION	2
Some General Rules for Fixtures	4
2.3. PREDICTION OF RESONANT FREQUENCIES	4
Beams	5
Plates	5
Stretching Resonance	7
3. SAMPLE DESIGN	8
3.1. DESIGN PARAMETERS	8
3.2. CONSTRUCTION DETAILS	11
3.3. EVALUATION	11
3.4. TROUBLE — SHOOTING	13
4. CHARTS AND TABLES	15
4.1. BOLT HOLE PATTERNS FOR B & K SHAKERS	15
4.2. VALUES OF B FOR BEAMS	16
4.3. VALUES OF λ FOR PLATES	17
4.4. PLOT OF b/L VERSUS λ	18
4.5. NOMOGRAPH FOR PLATE EQUATION	19
5. BIBLIOGRAPHY OF TEST FIXTURE LITERATURE	20

1. INTRODUCTION

The design and construction of fixtures can be a very complicated undertaking, as a look at some of the literature will show. Some firms and research organizations keep a full-time staff of qualified engineers occupied with nothing but designing fixtures. Indeed, when the test involves a large airborne radar unit which must be tested at several g in the frequency range from 10 to 2000 Hz, the design of the pertinent fixture is a job for an expert.

However, we have found that many of our customers have neither the staff nor the experience for this kind of work, and quite often only need a simple general purpose fixture to handle all their testing needs.

This Application Note is intended as a practical construction guide for those customers who have purchased a B & K vibration exciter and are faced with the problem of building a vibration fixture for it. It is not meant to be a comprehensive study of fixture design, which would indeed fill quite a large volume. A bibliography of fixture and related literature is given in the last section for further study.



2. DESIGN CONSIDERATIONS

2.1. GENERAL

The basic idea with any fixture is that it should transmit uniformly the forces produced by the vibration exciter to the test object. It is generally impossible to fix a test object directly to the shaker table itself, and the fixture acts as a transition piece between the two.

In B & K vibration exciters, the force from the driver coil is transmitted via the skeleton to the four mounting holes at each corner of the table. The object to be tested usually has a number of finite points (feet, mounting brackets, screw holes) where the force is to be applied, the idea being to duplicate as nearly as possible the force input experienced in service. Between the table and the test object is the fixture. In order to get the most out of the vibration exciter, this fixture should carry the force from the four mounting holes in the table to the mounting points on the test object with a minimum of loss and distortion. In other words the fixture should be as rigid as possible. This is condition number one in fixture design.

The acceleration level possible with the vibration exciter is inversely proportional to the total mass it has to drive, therefore the fixture should be as light as possible. This is condition number two. An ideal fixture, then, would have infinite stiffness at all frequencies and zero mass, and it is this paradox which is the basis of fixture design.

Although a fixturing problem may seem very complicated, in fact many of the design parameters are determined before-hand. The weight, for instance, is limited by the weight of the test object and the force available from the exciter. The shape is determined in part by the fact that it has to fit the exciter table at its bottom surface and the test object at one of its other surfaces. Finally, the test conditions impose restrictions on the fixture. A typical test specification calls for testing to 2000 Hz with no fixture resonances below 1000 Hz and an allowable variation in vibration levels between 1 and 2 kHz of not more than 2 : 1 between any pair of points on the table.

2.2. MATERIALS AND FABRICATION

Of the materials generally considered for fixture construction, steel, aluminum and magnesium are the most common. For very small items, beryllium is sometimes used, but its very high strength and light weight are counteracted by its cost and difficulty in machining.

The ratio of Young's Modulus to density (E/ρ) is a controlling factor for the natural frequency of a construction, and since this ratio is roughly the same for steel, aluminum and magnesium, the choice of material will not greatly affect the natural frequency of the fixture. It will, however, in the case of steel, affect the weight, and in the case of magnesium, the cost. Table 2.1 compares the E , ρ and E/ρ of the three metals. Titanium, used in B & K accelerometers, is included for comparison.

There are numerous ways to fabricate a fixture; among them, machining the fixture out of one solid piece of material, casting, bolting together several pieces to form an assembly, building up an assembly by laminating strips of material together, and welding.

	Steel	Aluminum	Magnesium	Titanium
Young's Modulus (E) N/m ²	20,7 x 10 ¹⁰	6,9 x 10 ¹⁰	4,14 x 10 ¹⁰	10,7 x 10 ¹⁰
Density (ρ) kg/m ³	7840	2770	1800	4510
E/ ρ N m/kg	2,65 x 10 ⁷	2,49 x 10 ⁷	2,3 x 10 ⁷	2,38 x 10 ⁷

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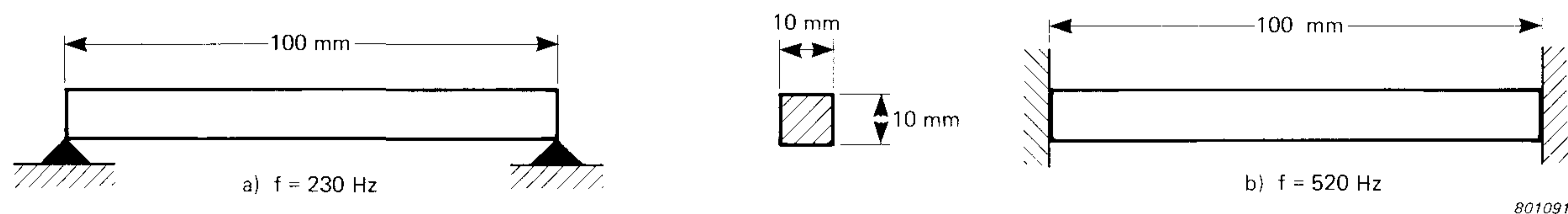
Table 2.1. Characteristics of fixture metals

Machining. Machining from solid stock usually yields a good fixture because there are no joints to work loose or bolts which act as springs at high frequencies. It is also a simple, easy method of construction and is often used for small test items such as electrical connectors or electronic components. However, for larger fixtures it would be expensive and wasteful to machine from solid stock.

Casting. A cast fixture has the advantage of homogeneous, monolithic construction, as well as very little machining work to be done. However, casting is only economically feasible when it involves making a number of units, because a master has to be made up before-hand, usually of wood.

Bolting. Bolted fixtures are not recommended. Although it is a simple way to assemble a fixture, it can produce a lot of trouble later on, with bolts working loose under vibration. Furthermore a bolted connection is not as rigid as a welded connection and is usually considered as a pin connection rather than a fixed connection when designing.

Consider the two identical aluminum beams in Fig.2.1.



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Fig.2.1. Comparison of two beams

The beam in Fig.2.1a is simply supported and that in Fig.2.1b is fixed at both ends. The ratio of the frequencies of the two beams is 2,3. This is, in effect, the advantage to be gained by welding instead of bolting.

Laminating. Something can be said for the method of laminating, whereby a fixture is built up in layers from different materials. This technique has two advantages: 1) It can be done by lab personnel with fairly simple tools, and 2) by alternating layers of rubber or plastic a highly damped fixture can be built. The disadvantage is that a considerable amount of time is used in the construction, which makes it expensive.

Welding. For most fixtures for the B & K interchangeable head exciters, welding is probably the best construction method, both from the point of view of economy and that of performance.

All seams should be fully welded on at least one side. Spot welds are no more effective than bolts and should be avoided. After welding and before machining, let the fixture stand overnight in an oven at 250°C. This will relax some of the stresses induced by welding. It will also reduce some of the warping.

To compensate for this warping, the top plate of the fixture should be made thicker than the design thickness, then machined down to the design thickness after annealing.

Wooden Fixtures. It should also be noted here that wooden fixtures have been used successfully at low frequencies and that a reasonable fixture can be made of laminated wood, which has a high damping coefficient.

2.2.1. Some General Rules for Fixtures

1. The easiest and cheapest fixture for the B & K interchangeable head exciters is that made of welded aluminum plate.
2. The mating surfaces of the fixture should be machined as flat as possible. If the surface is a plate, the fabrication thickness should be greater than the design thickness as some warping will occur during fabrication which can later be machined off.
3. All seams should be full welded on at least one side. Avoid spot welds and bolts.
4. Attachment bolts from fixture to exciter table should be counter-sunk so that the stressed length of the bolt is twice the diameter.
5. It is very difficult to eliminate resonances completely in the frequency range 1000 — 2000 Hz. What can be done, however, is to introduce some damping into the structure in order to reduce the severity of the resonance. This may be done by bonding a damping material onto the member in question or fitting rubber or wooden blocks between resonating plates. Hollow fixtures may be pumped full of polyurethane foam.
6. Make a drilling template to match the threaded holes in the shaker table. This insures proper alignment of the bolts.
7. Tighten all the bolts as hard as the limiting torque of the inserts will allow.

2.3. PREDICTION OF RESONANT FREQUENCIES

The load carrying ability of the material almost never enters into fixture design. This is because, if the structure is designed so as to avoid resonances, it is over-designed by far with respect to static or dynamic loading. The resonant frequency then becomes the critical design factor.

The prediction of natural frequencies of a structure can be very complicated. However, for the purpose of designing a fixture, only the first natural frequency is important. The first natural frequency is the most damaging and the appearance of any resonances will often disqualify a fixture for use.

A further simplification is the assumption of a single degree of freedom. Probably everyone reading this is familiar with the concept of a simple mass m supported by a pure spring of stiffness k , which when deflected, resonates at a frequency

$$f = \frac{\omega}{2\pi} = \frac{1}{2\pi} \sqrt{\frac{k}{m}} \text{ Hz}$$

By breaking the fixture down to a combination of simple elements and assuming a single degree of freedom system, this formula can be used as the basis for almost all frequency calculations in fixture design. The trick then comes in deciding what k should be.

2.3.1. Beams

As an example, take the aluminum cantilever beam shown in Fig.2.2.

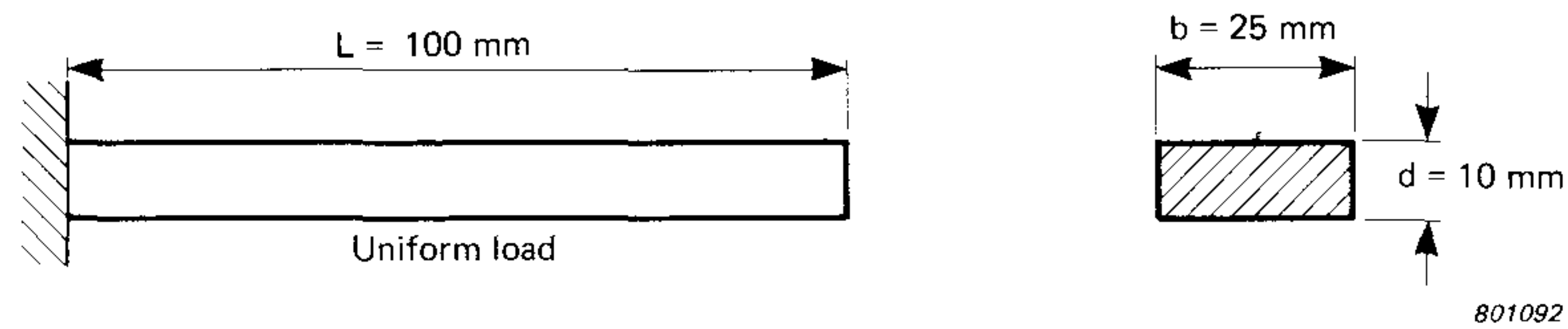


Fig.2.2. Cantilever beam

This is a fairly common type of configuration. For instance, resonant fatigue specimens are typically bars of material clamped at one end in a fixture. The k value for a beam is

$$k = \frac{BEI}{L^3} \text{ N/m}$$

where

E = elastic modulus N/m²

I = second moment of area m⁴

L = length m

B = a constant based on end conditions, given in Table 4.2.

For a rectangular section,

$$I = \frac{bd^3}{12} \text{ m}^4 \quad \text{and} \quad m = \rho bdL \text{ kg}$$

where

b = width m

d = thickness of the beam m

ρ = density kg/m³

Hence,

$$\omega = \sqrt{\frac{k}{m}} = \sqrt{\frac{BEI}{mL^3}} = \sqrt{\frac{12Ebd^3}{12\rho bdLL^3}} = \sqrt{\frac{Ed^2}{\rho L^4}} = 5 \times 10^3 \frac{d}{L^2} \text{ rad/s}$$

which shows that for a rectangular section the resonant frequency is independent of width. For the above example,

$$f = \frac{1}{2\pi} \times 5 \times 10^3 \times \frac{d}{L^2} \cong 800 \times \frac{0,010}{0,1^2} = 800 \text{ Hz}$$

2.3.2. Plates

Because of the great number of edge conditions and shapes possible there has been a great deal of work done on plate resonances. A very useful compilation of all this research has been made by Arthur Leissa and published as a NASA bulletin (see bibliography). However, anyone using this information is cautioned to read very carefully because the various authors state their results in slightly different terms and it can be misleading.

Most plate formulae are based on the work of Warburton, who used the Rayleigh-Ritz method of equating kinetic and potential energy and evolved a formula for resonant frequency.

For a plate:

$$\omega^2 = \frac{\pi^4 D}{b^4 \rho h} \lambda^2, \quad \text{where } D = \frac{E h^3}{12 (1 - \nu^2)}$$

where

h = thickness of the plate and

ν = Poisson's ratio.

and λ is a mode constant based upon edge conditions.

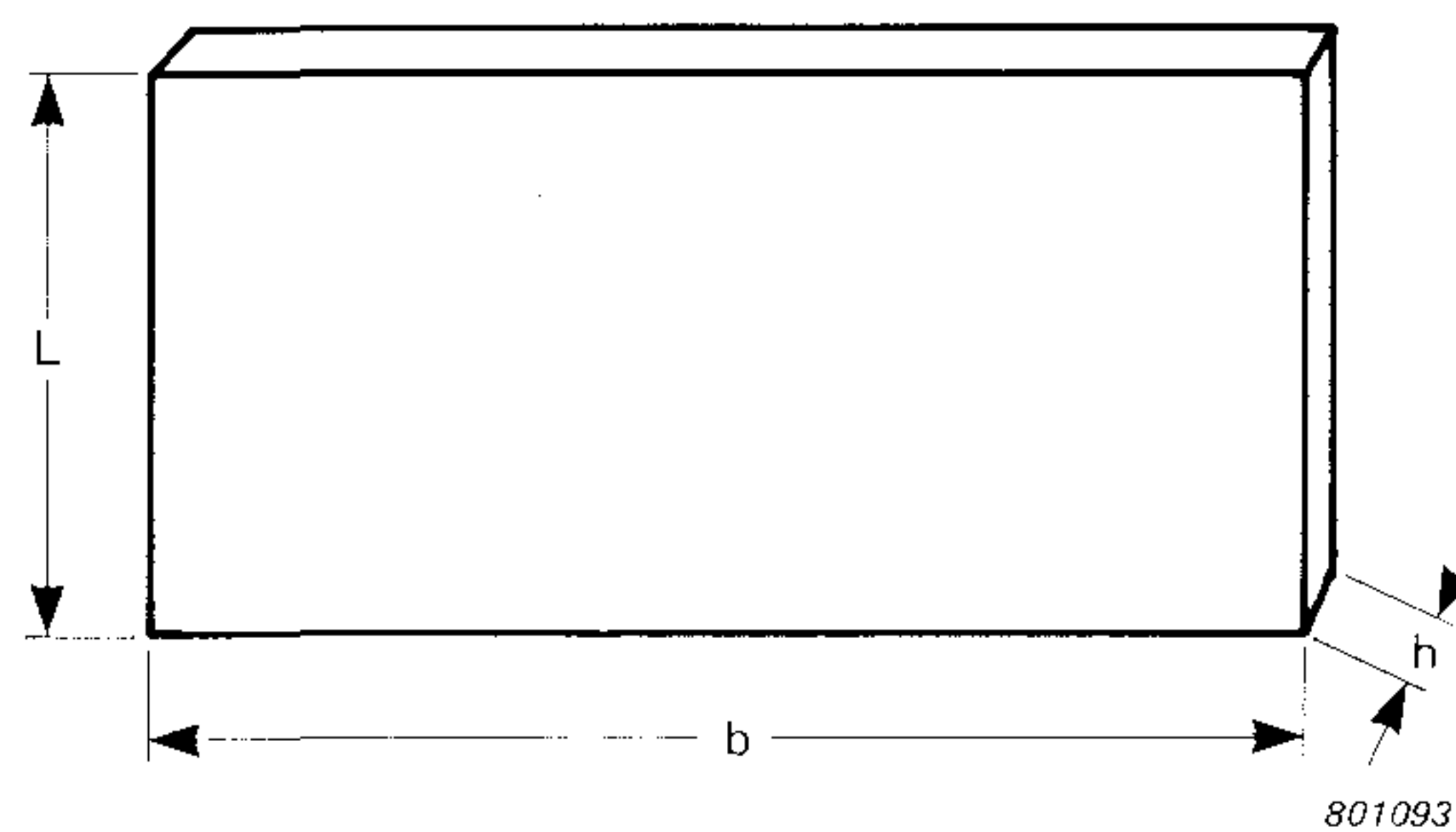


Fig.2.3. Plate resonance parameters

Since the use of steel, aluminum or magnesium is assumed for fixtures, this can be simplified to

$$f = 23,5 \times 10^4 \frac{h}{b^2} \lambda, \quad \text{where } h \text{ and } b \text{ are in centimeters} \quad \left. \vphantom{f} \right\} (1)$$

or

$$f = 23,5 \times 10^4 \frac{h}{L^2} \lambda$$

$$f_{\text{minimum}} \iff \text{Maximum } (b, L)$$

(see Fig.2.3) and λ may be found in Table 4.3. This simplification holds true as long as the following conditions are met:

1. The plate should have uniform thickness, no applied load and h less than $1/20$ of b , although this last is somewhat flexible.
2. Clamped edges are rigidly restrained. Simply supported edges allow bending at the support. Free edges have little or no restraint.
3. E/ρ is approximately 25×10^6 Nm/kg and ν (Poisson's ratio) is 0,3.

Note that the frequency thus obtained is the first mode frequency. For quick estimations and comparison between alternative designs, a graph of b/L vs. λ is given in Table 4.4 and a nomograph for computation of f is supplied in Table 4.5. Note also the relationships given in Fig.2.4, which will simplify certain calculations.

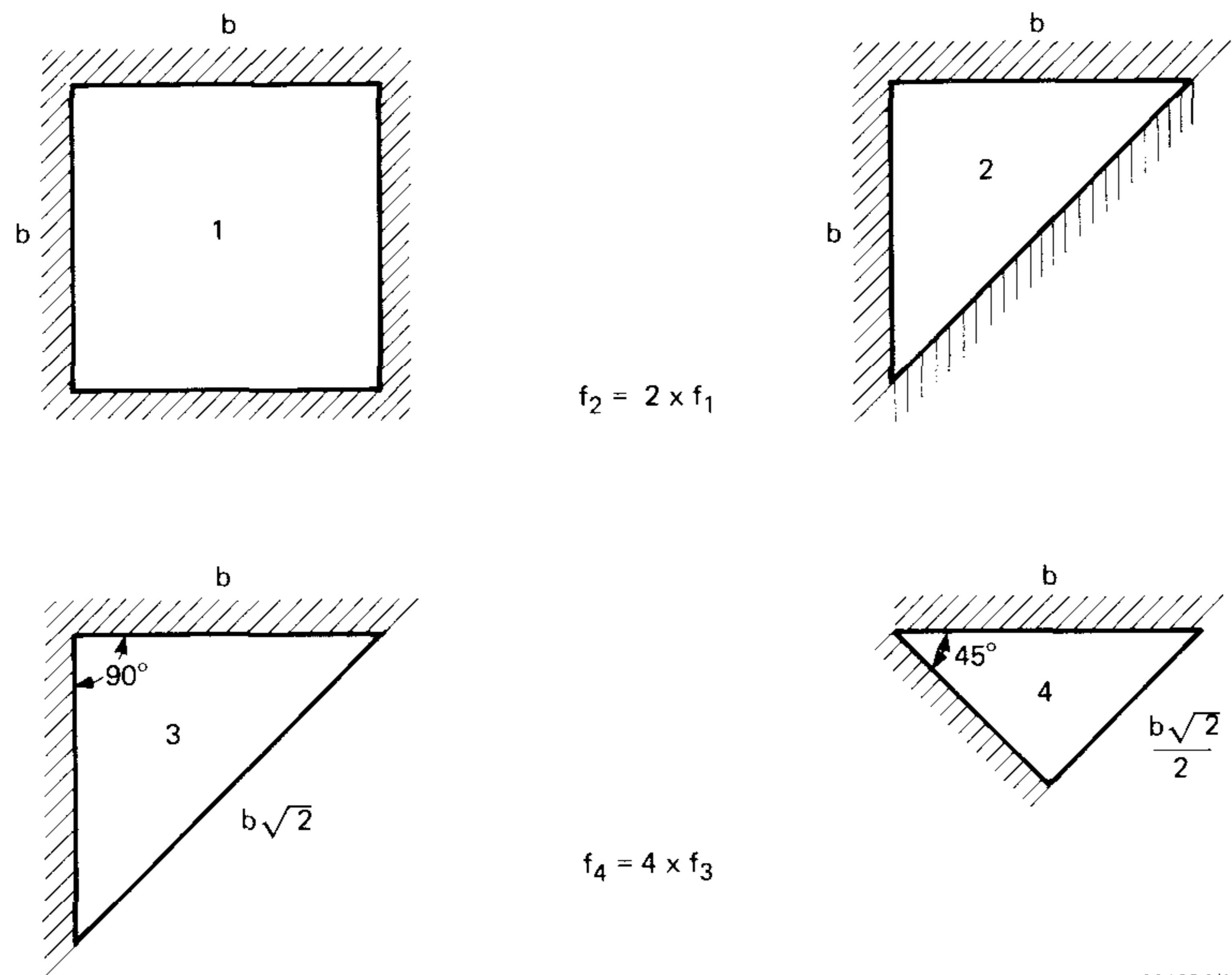


Fig.2.4. Frequency relationships of different plates

2.3.3. Stretching Resonance

One question which usually occurs to the novice fixture designer is whether to take into account the stretching of the fixture as well as the flexing modes of the plates. Generally speaking, the answer is no.

The stretching resonance frequency of any object is $f = V/4L$ where V is the velocity of sound and L is the length. Shape, width and thickness have no effect. V and fL for the three fixture metals and titanium are given in Table 2.2.

	Steel	Aluminum	Magnesium	Titanium
V m/s	5148	4990	4796	5990
fL m/s	1287	1248	1199	1497

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Table 2.2. Velocity constants for fixture metals

For an aluminum fixture to have a stretching resonance below 2000 Hz it would have to be at least 624 mm long, which eliminates the problem for most fixtures. The usual place where stretching resonances occur is in slip tables but these are a subject in themselves and won't be covered here.

3. SAMPLE DESIGN

As an illustration of the foregoing principles we will design a general purpose fixture which will fit a Type 4813 or 4817 Exciter Head. The requirements are:

- a. It must have a working surface 400 mm square.
- b. It must have no resonances below 1000 Hz.

3.1. DESIGN PARAMETERS

As in most cases, the dimensions of the fixture are pretty much predetermined. The top surface will measure 400 mm × 400 mm. The bottom surface (see Table 4.1) will measure 100 mm × 100 mm. If we simply mounted a plate on the shaker table it would have to be quite thick and therefore heavy and would obstruct the movement of the table on the downstroke. If we make the fixture very high, any off-centre weight in the test object will cause rocking. So we choose a height of 100 mm as reasonable and set the top plate atop a 100 mm cube.

Since the forces in the drive coil are delivered to the mounting points on the table we add four diagonal braces such that the forces are led from the mounting points to the far corners of the table. The layout of the fixture at this point looks like Fig.3.1.

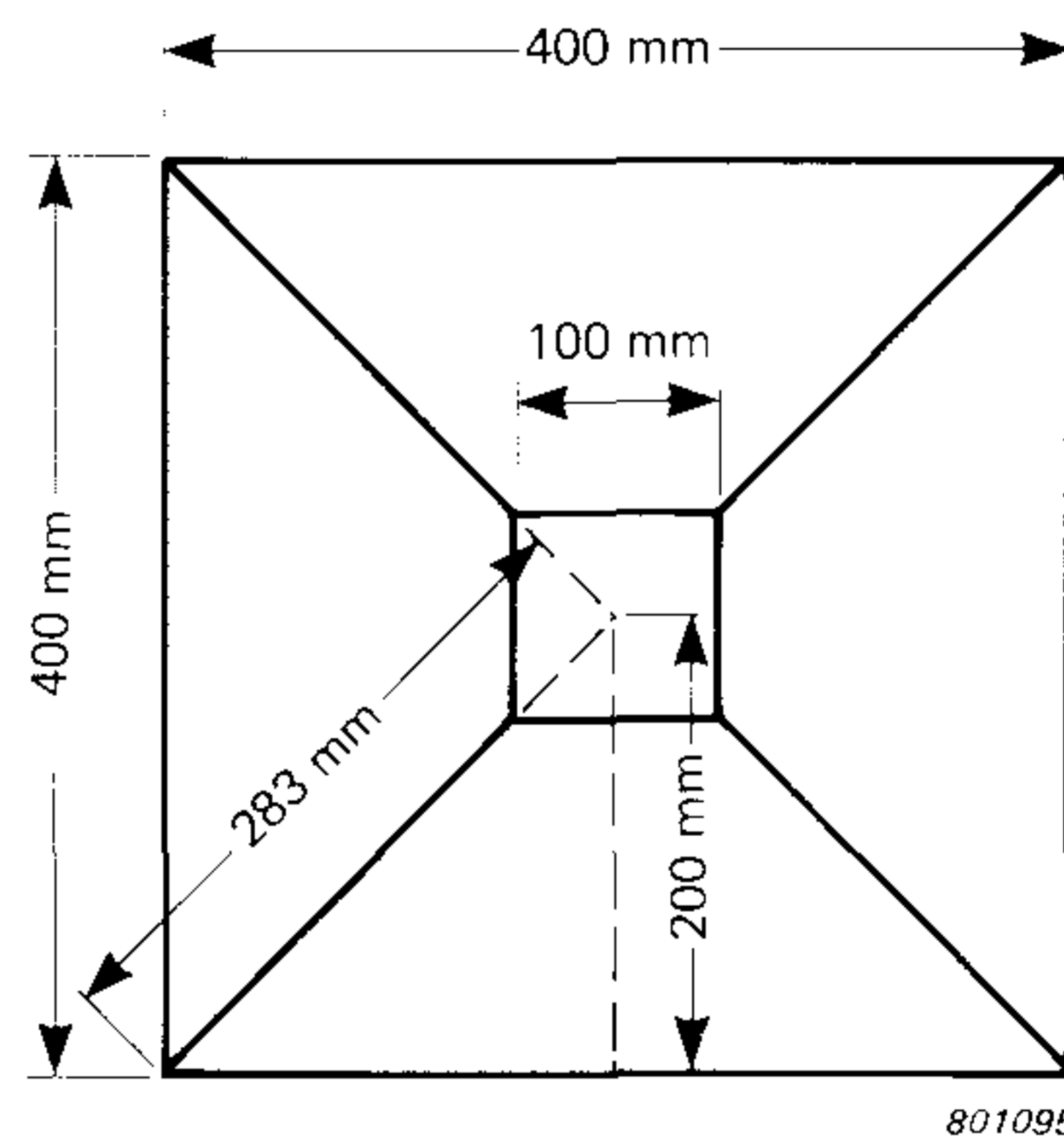


Fig.3.1. Initial layout of fixture

The top plate area bounded by the diagonals can be approximated by a right isosceles triangle with the equal edges clamped and the remaining edge free (see Fig.3.1). The clamped edges have a length of 28,3 cm. To have a resonant frequency above 1000 Hz, the required plate thickness would be given by eq. (1) using $\lambda = 1,31$ from Table 4.3.

$$h = \frac{1000 \times b^2}{23,5 \times 10^4 \times \lambda} = \frac{1000 \times 28,3^2}{23,5 \times 10^4 \times 1,31} = 2,60 \text{ cm}$$

This plate would weigh 11,52 kg by itself. In order to reduce this, we can add another

brace to bisect the angle of the plate. This would allow us to reduce the thickness to 6,5 mm but we will actually make it 12 mm thick. There are two reasons for this; 1) If all the plates are designed for exactly 1000 Hz we could get a beating effect when we near that frequency and may even get resonances around 800 — 900 Hz due to the composite effect of all the plates. It is therefore a good idea to spread the resonant frequencies a little. 2) The top plate is the best place to put any "fat" in because it is here that the test object is mounted and there will be extra stress induced at the mounting points.

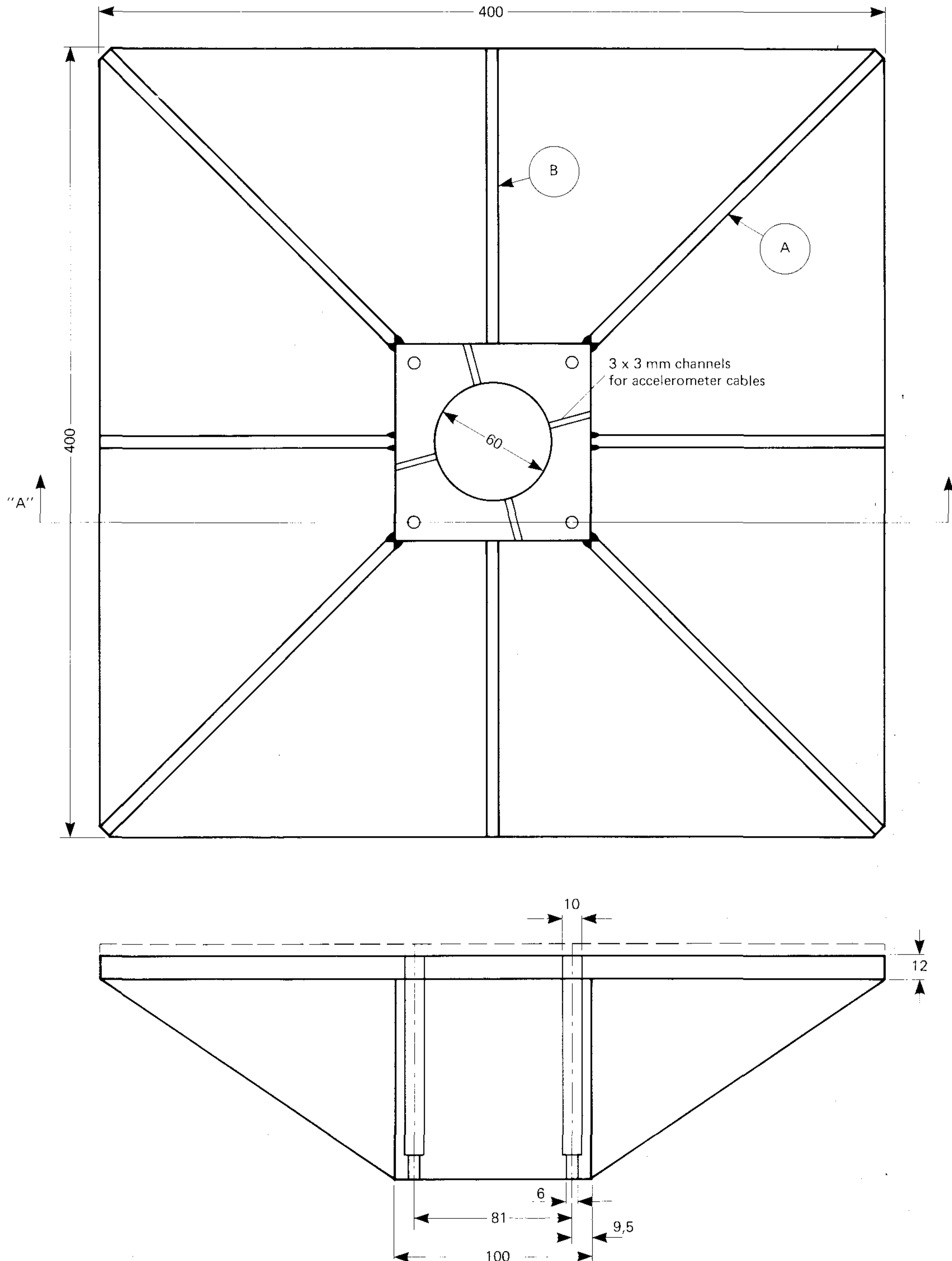


Fig.3.2. General Purpose Fixture for 4813 or 4817

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The fixture now looks like that shown in Fig.3.2. The resonant frequency of the top plate can be calculated using eq. (1) and the relation given in Fig.2.4.

$$f = 4 \times 23,5 \times 10^4 \times \frac{1,2}{28,3^2} \times 1,31 = 1845 \text{ Hz}$$

The side braces are shown in Fig.3.3. For brace (B) the calculations are as follows:

$$\frac{b}{L} = 1,5$$

From Table 4.3

$$\lambda^2 = 0,39 + 0,96 \left(\frac{b}{L}\right)^2 + 0,36 \left(\frac{b}{L}\right)^4 = 4,37$$

$$\lambda = 2,09$$

For a 6 mm thick plate the resonant frequency of brace (B) is given by:

$$f = 23,5 \times 10^4 \times \frac{0,6}{15^2} \times 2,09 = 1310 \text{ Hz}$$

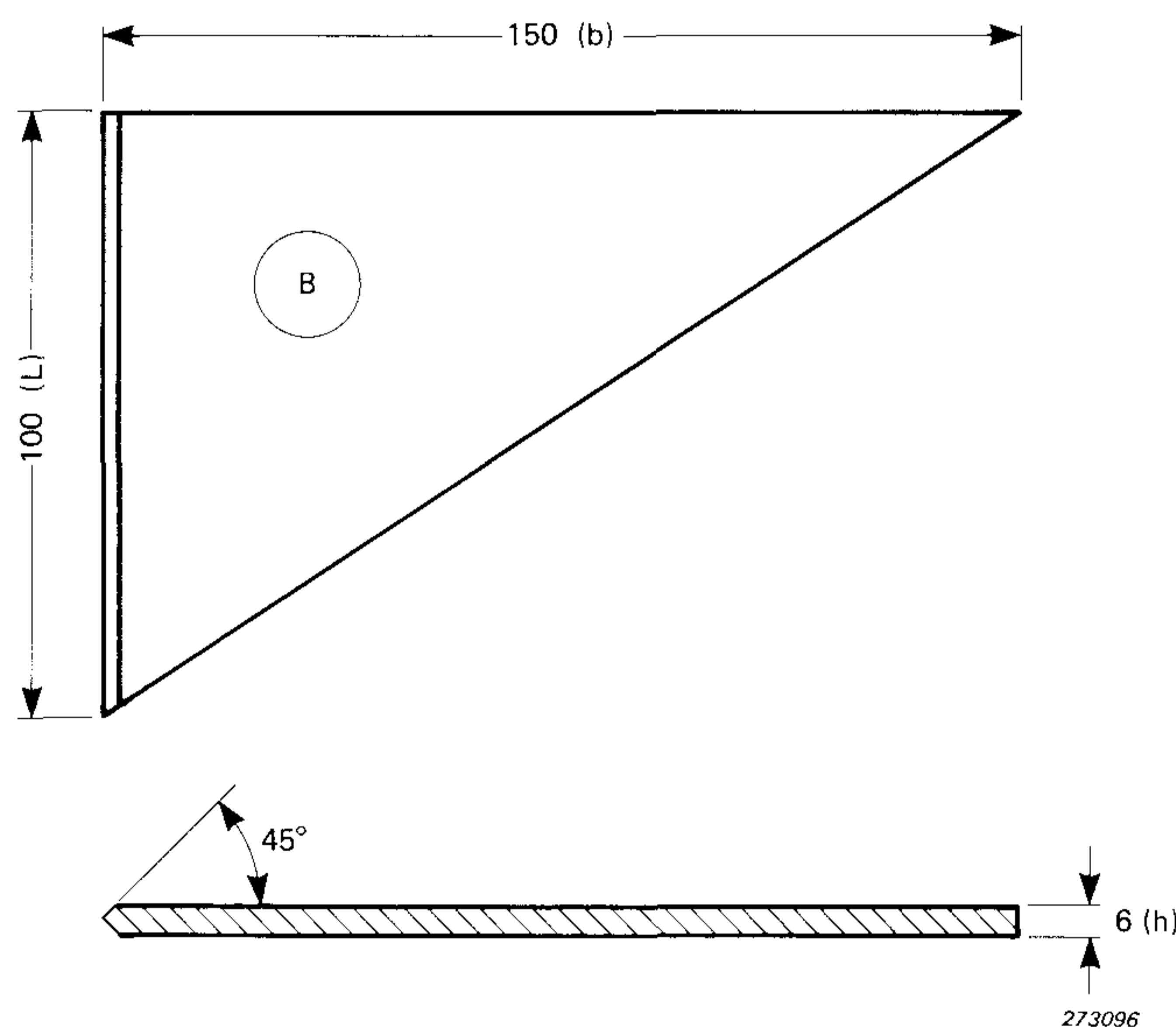


Fig.3.3. Side Braces

The plate, of course, could have been 4,5 mm thick but 6 mm was a more convenient size. Fig. 3.4 shows the diagonal brace (A). The calculations follow:

$$\frac{b}{L} = 2,09$$

$$\lambda^2 = 0,39 + 0,96 \left(\frac{b}{L}\right)^2 + 0,36 \left(\frac{b}{L}\right)^4 = 11,45$$

$$\lambda = 3,38$$

$$f = 23,5 \times 10^4 \times \frac{0,6}{20,9^2} \times 3,38 = 1091 \text{ Hz}$$

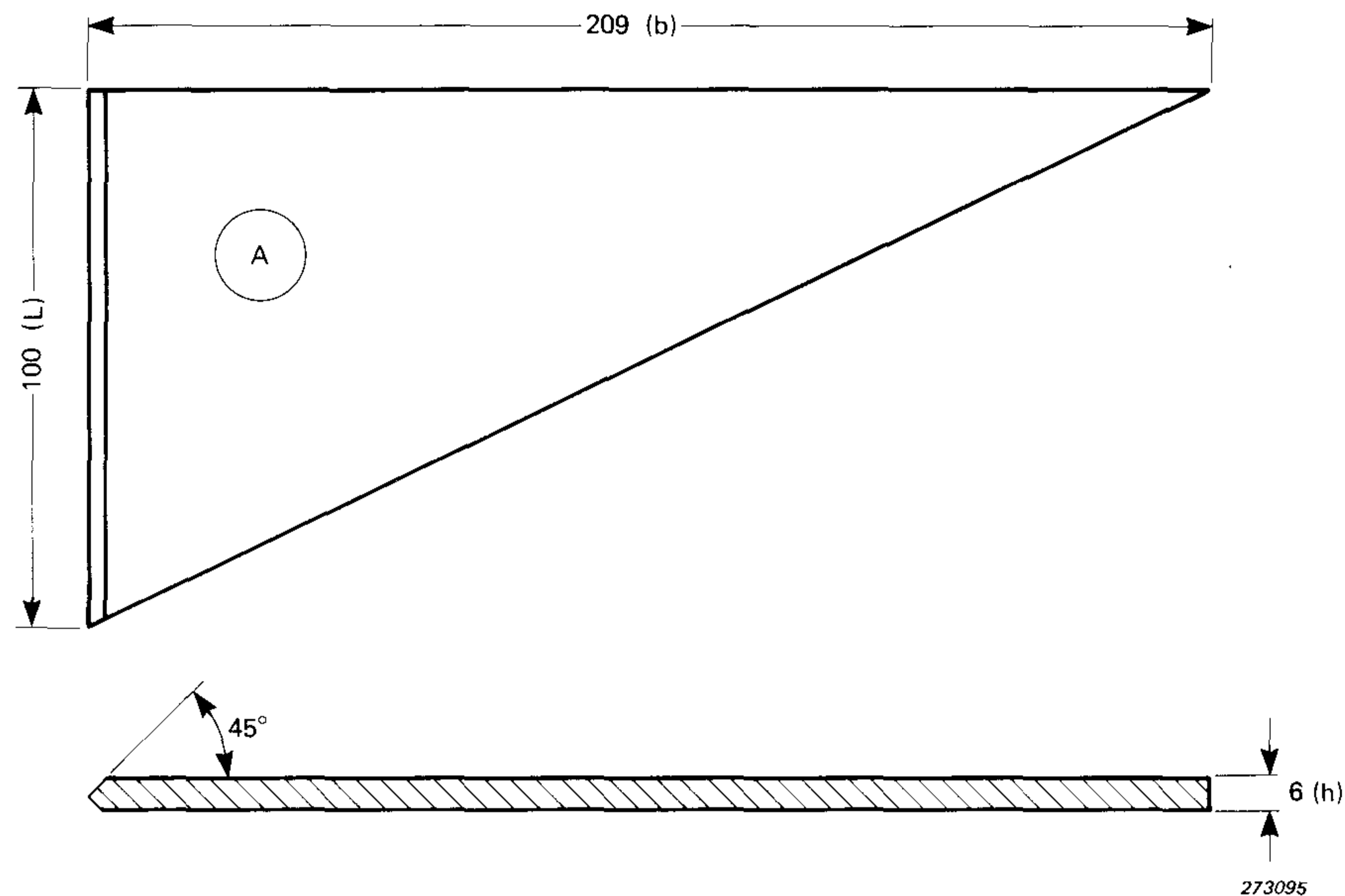


Fig.3.4. Diagonal Braces

3.2. CONSTRUCTION DETAILS

As it is not necessary to have a full 100 mm cube for a core, a 60 mm hole was milled out after the main parts were welded together. This serves two purposes:

1. It removes some of the weight, and
2. It provides a convenient mounting point for the accelerometer.

The hole is milled all the way through the core to the top plate and a 10-32 UNF threaded hole is provided in the centre. The accelerometer is mounted upside down at this point, thus putting the control as close as possible to the test object without being in the way. 4 channels, each 3 mm square in section, are milled into the base so that the accelerometer cable can be led out in whatever direction is convenient. It is a good idea to write the sensitivity on a piece of tape and stick it on the fixture. This saves looking for the calibration chart when you set up.

The top plate was originally 20 mm thick, then after welding and soaking overnight at 250°C was milled down to 12 mm. The bottom surface was ground off to obtain a smooth mating surface with the shaker table.

Mounting holes can be made in the top surface according to need. At B & K we made a series of holes around the edge at 40 mm centre-to-centre. Into each hole is fitted an M 10 helicoil. The total weight of the fixture was found to be 7,7 kg.

3.3. EVALUATION

When all the welding and machining is finished, the fixture should be evaluated before use, both as a check on the design and construction and as a way of knowing the conditions at the table surface during testing.

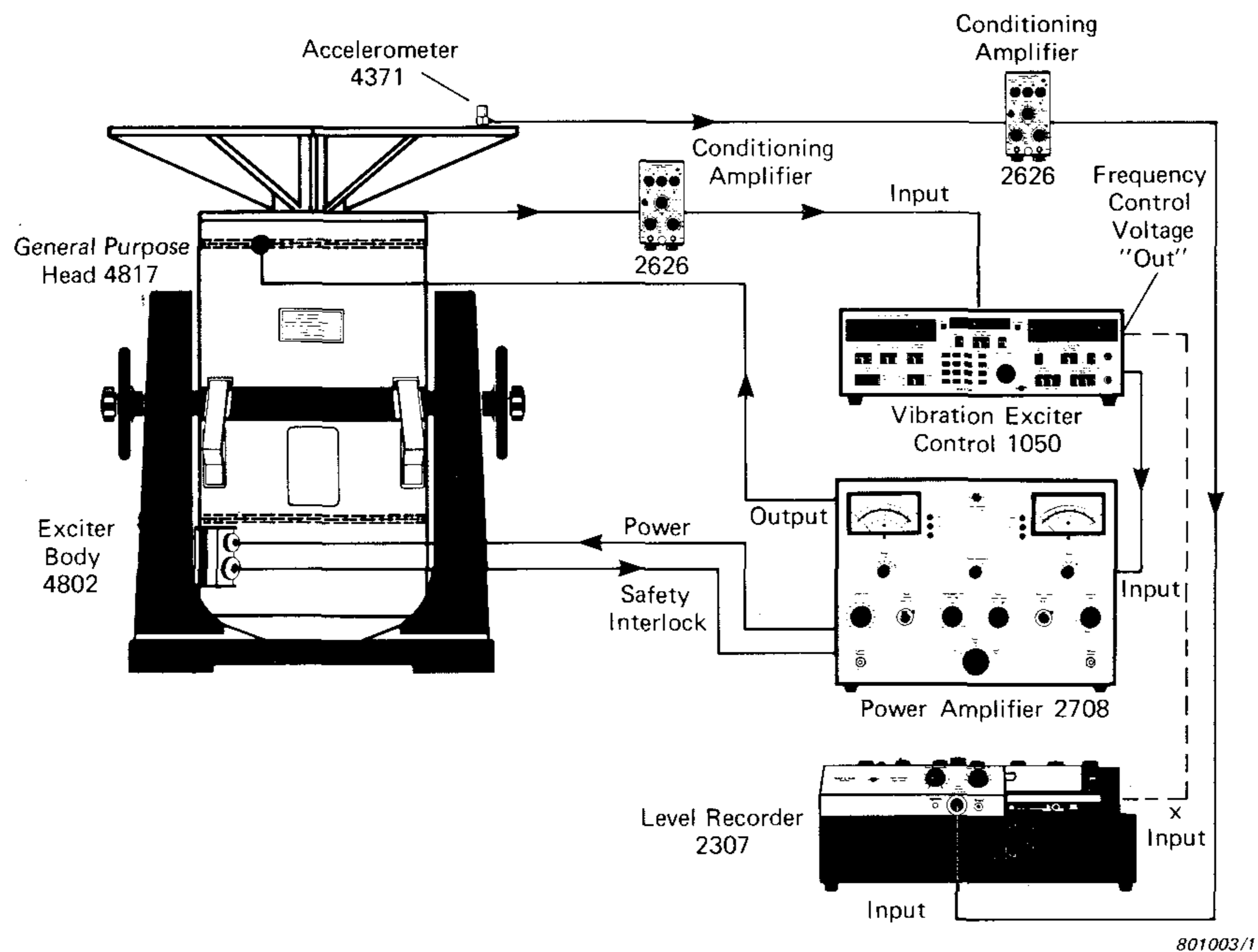


Fig.3.5. Setup for fixture evaluation

Fig. 3.5 shows a typical evaluation setup. The fixture is mounted on the Exciter with a control accelerometer mounted just under the centre of the top plate. The signal from this accelerometer is fed to the Input of the Exciter Control via the Conditioning Amplifier Type 2626. Here the vibration level is set (10 m/s^2 is a good level for evaluation) and monitored. The acceleration level is held constant at the centre by a voltage controlled amplifier in the Exciter Control.

A second accelerometer, preferably as small as possible so as not to affect the resonant frequencies, is mounted at various places on the fixture. The signal from this accelerometer is conditioned in a second 2626 preamplifier. The frequency of the Exciter Control is now scanned in search of resonances at various points on the fixture. The frequency can be adjusted manually on the Exciter Control to locate the resonance within 0,002 Hz.

It is often useful if a "signature" can be made for different parts of the fixture. This is a plot of the output of the second accelerometer versus frequency and is made by feeding the output of the second 2626 to a level recorder. Such a signature is shown in Fig.3.6 for the fixture designed earlier in this chapter.

The control accelerometer is mounted on the centre of the top plate, while the second accelerometer is mounted on the mid-span. In this case the level is the same as the control level throughout most of the frequency range. Just before the first resonance the level creeps up a little so that it is about 3 dB (40%) higher at 1000 Hz. The first peak is at 1116 Hz. The calculated first peak should be at 1091 Hz, thus there is an error of only 2,2%. A second peak appears at 1558 Hz, which does not comply with either of the next two calculated resonances, but this is to be expected after the first resonance because of interaction between resonances and anti-resonances.

What is important in evaluating a fixture is to find out what the actual levels are across the whole table so that if the control is set for a certain level one can be reasonably sure that no over-testing will occur. A vibration contour map of the fixture can also be made at the first resonance to see which parts of the fixture have the highest acceleration levels at this frequency. Then, if desired these could be damped or changed.

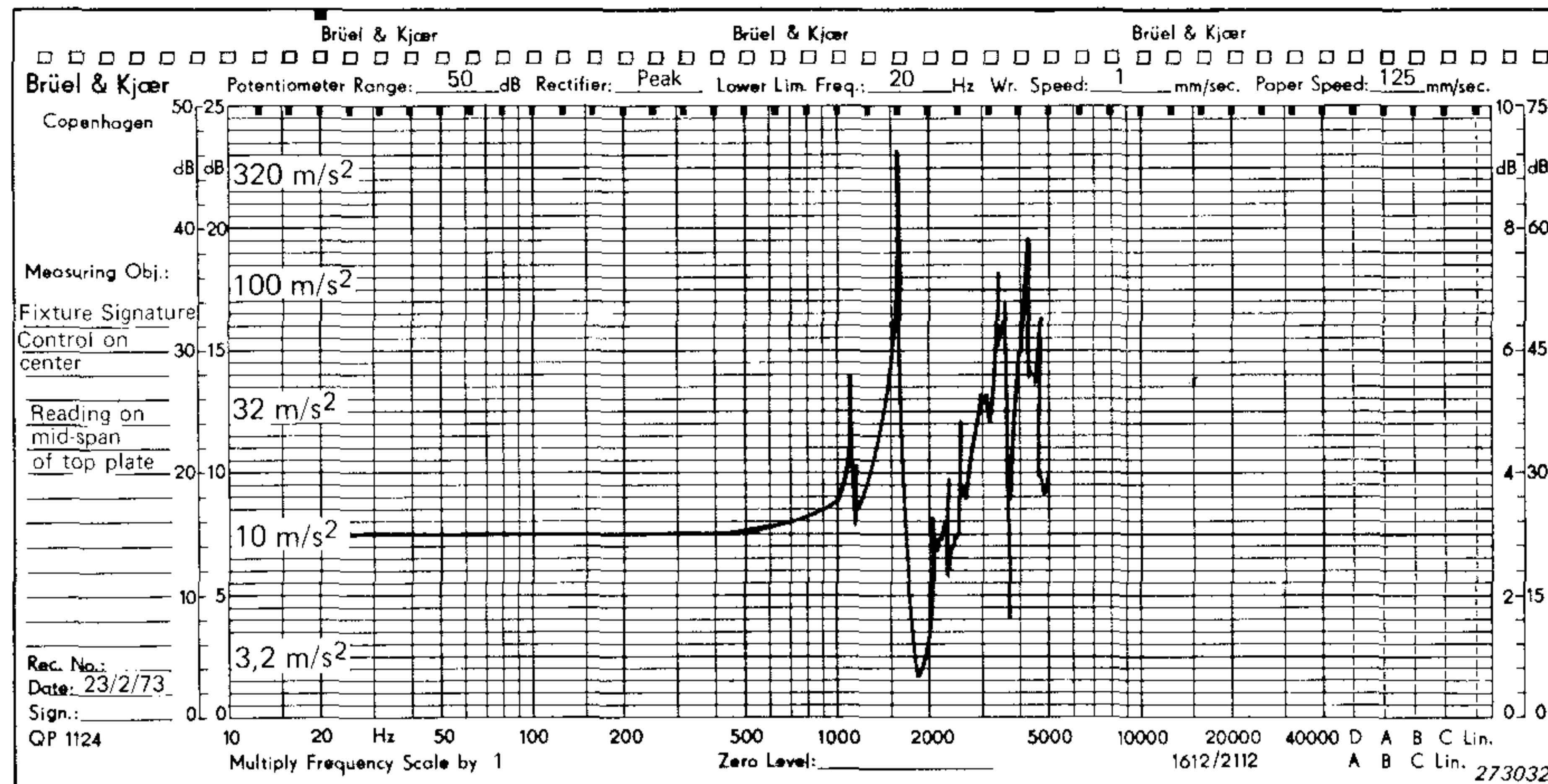


Fig.3.6. Fixture signature

3.4. TROUBLE-SHOOTING

If it happens that the fixture fails its evaluation for some reason, it is still possible to remedy it. If a resonance appears at too low a frequency for example, that frequency can be changed by bonding an extra brace on with epoxy.

If the resonance build-up occurs in such a way that the level is too high near the upper limits of the range even though there is no resonance within the limits, this can be remedied with damping. Say, for example, that the build-up shown in Fig.3.6 should be brought down to 25% over 10 m/s² instead of 40%. We could then try to damp the resonance down.

We can be reasonably sure that the peak shown corresponds to the resonance frequency of the diagonal brace. Therefore we could either add some damping material to this brace by coating it with a damping paint or paste, or we could fit wooden blocks between all the braces and epoxy them in place.

It is sometimes difficult to ascertain exactly where a particular resonance is coming from because of transmission throughout the entire structure. If this is the case, a thorough investigation can be made by impedance methods.

Fig.3.7 shows a setup for measuring the inertance of the different parts of the fixture. A small hole is drilled at the point of application and tapped with a 10-32 UNF thread. A Mini-Shaker Type 4810 and Impedance Head Type 8001 are mounted at this point. The Impedance Head measures force and acceleration simultaneously at the same point. The force signal is used as a feed back signal to hold the force constant. The acceleration signal is fed to a level recorder via a second conditioning Amplifier Type 2626. A plot of acceleration as a function of frequency is thus obtained from which the resonances can be identified.

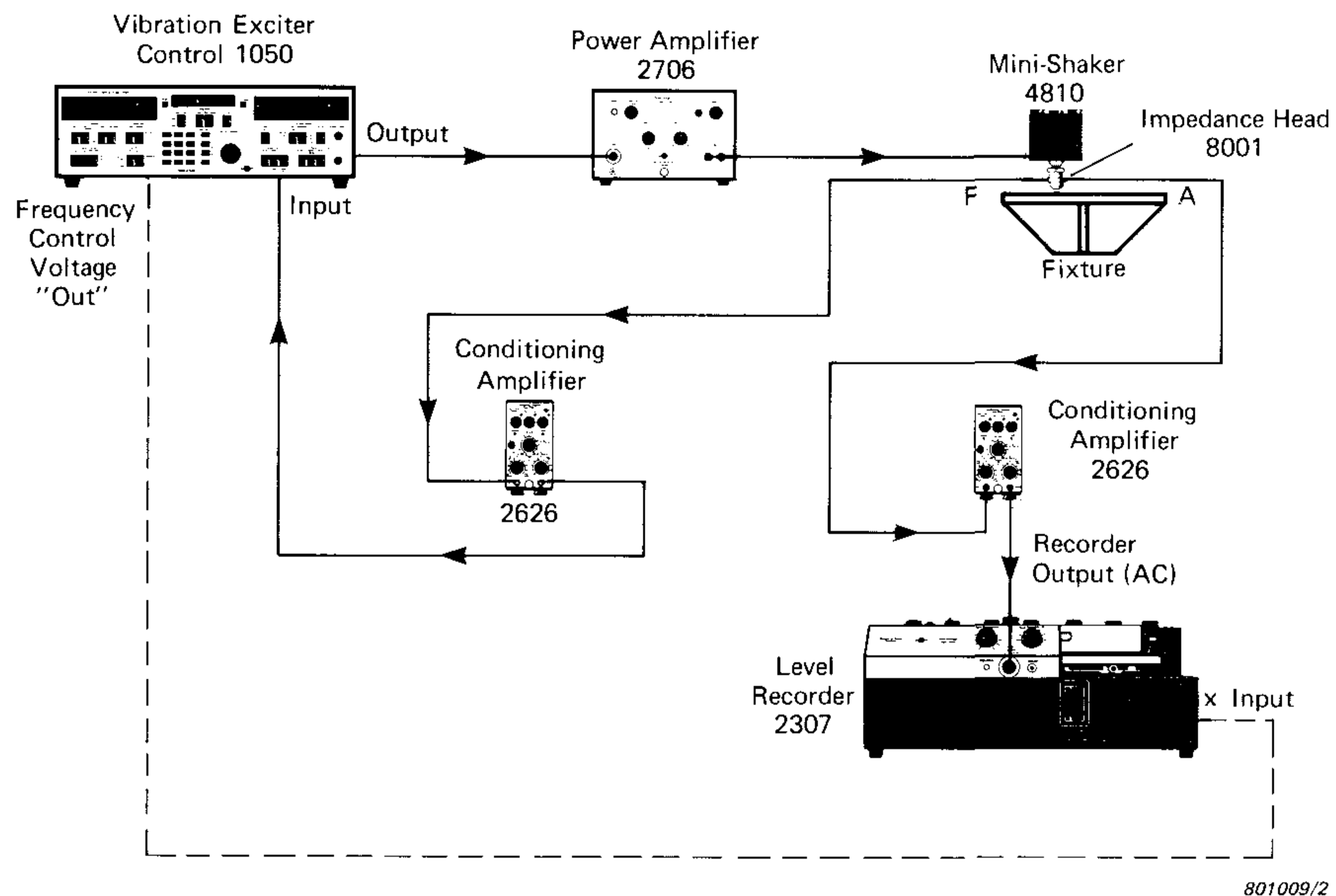


Fig.3.7. Setup for measurement of inertance

4. CHARTS AND TABLES

4.1. BOLT PATTERNS AND INSERT SIZES FOR B & K SHAKERS

Square Tables

All dimensions in mm

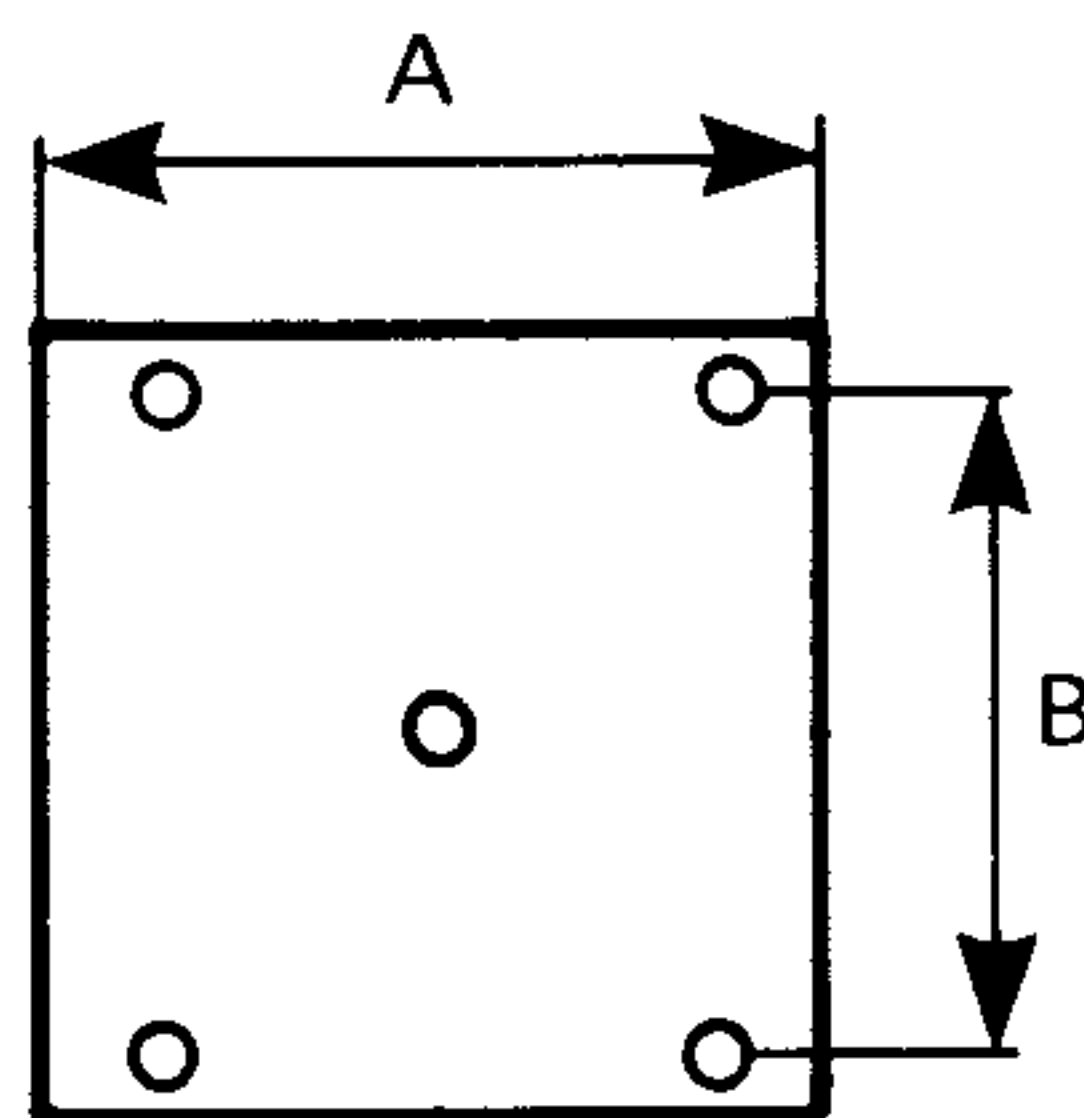


Table Dimensions Type	Table Dimensions		Depth of Hole	Thread Size	Insert Depth
	A	B			
4811	44,7	35,9	7,5	M4	5,6
4812	66,5	53,9	7,5	M5	3,9
4813	100,0	81,0	15,9	M6	4,7
4816	66,5	53,9	6,8	M5	3,9
4817	93,5	81,0	12,7	M6	4,7
4818	139,7	121,0	12,7	M8	7,1
Mode Study Head	Diameter				
4814	19,0		7,5	M5	3,9

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Table 4.1.

4.2. VALUES OF B FOR BEAMS

		Type of Load	B
1		CANTILEVER Uniform load	12
2		Point load at end	3
3		Distributed load at end	$\frac{24(L-a)}{3L - \frac{4a^3}{L^2} + \frac{a^4}{L^3}}$
4		Load at any point	$\frac{2}{\frac{a^2}{L^3} \left(L - \frac{a}{3} \right)}$
5		Point load at end plus uniform load	$\frac{24(wL + P)}{3wL + 8P}$
6		FIXED ENDS Uniform load	384
7		Point load	192
8		Off-centre load	$\frac{3(3a_1 + a_2)^2 L^4}{4a_1^3 a_2^3}$
9		Uniform load plus point load	$\frac{384(P + wL)}{2P + wL}$

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Table 4.2.

4.3. VALUES OF λ FOR PLATES

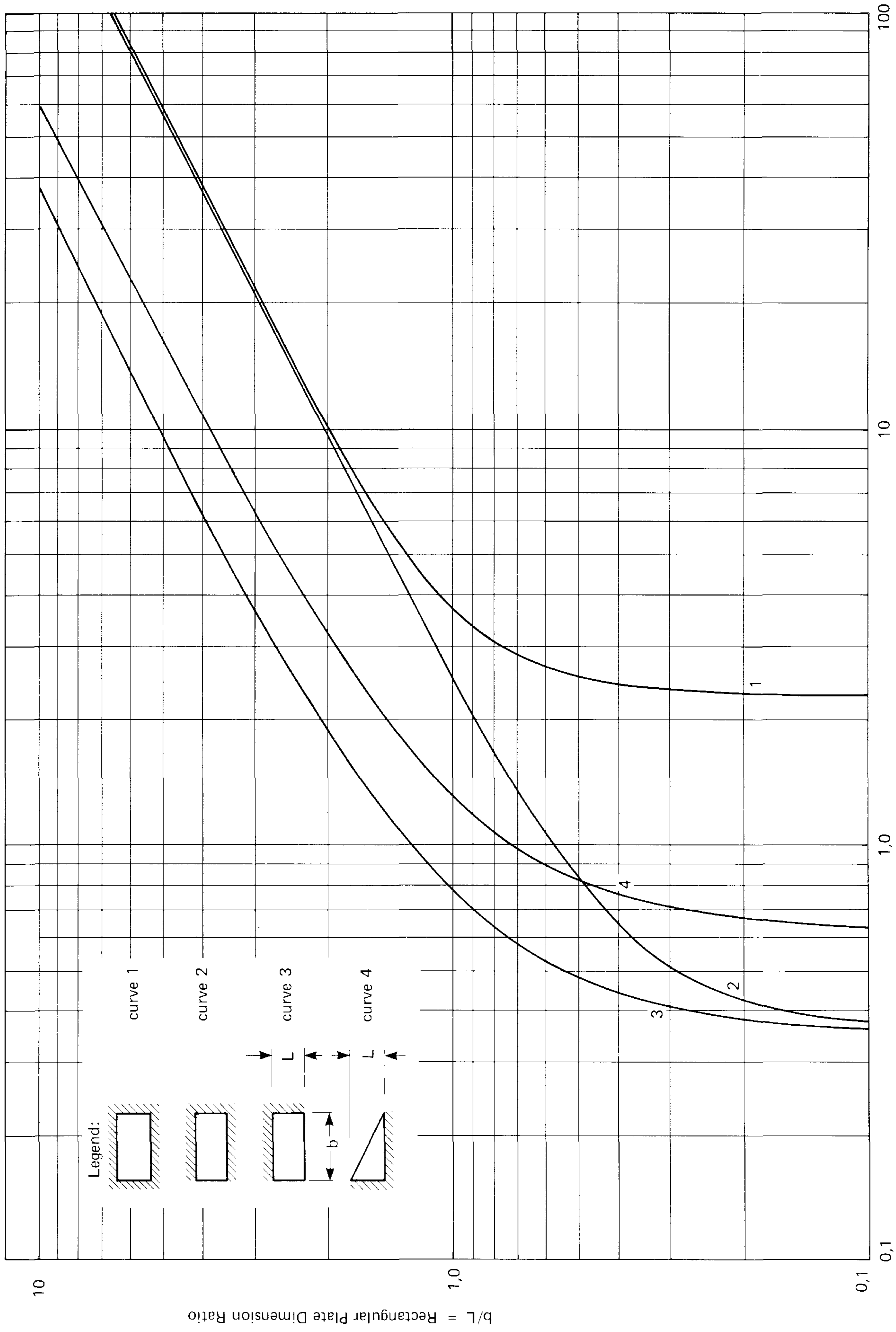
	λ^2	λ^{2*}	λ^*
	0,14	0,14	0,37
	$0,14 + 0,47 \left(\frac{b}{L}\right)^2 + 0,14 \left(\frac{4}{L}\right)^4$	0,75	0,87
	5,33	5,33	2,31
	$0,14 + \left(\frac{b}{L}\right)^2 + 5,34 \left(\frac{b}{L}\right)^4$	6,48	2,55
	$5,33 + 3,55 \left(\frac{b}{L}\right)^2 + 5,33 \left(\frac{b}{L}\right)^4$	14,21	3,77
	$0,39 + 0,96 \left(\frac{b}{L}\right)^2 + 0,36 \left(\frac{b}{L}\right)^4$	1,71	1,31

* Indicates that $b/L = 1$

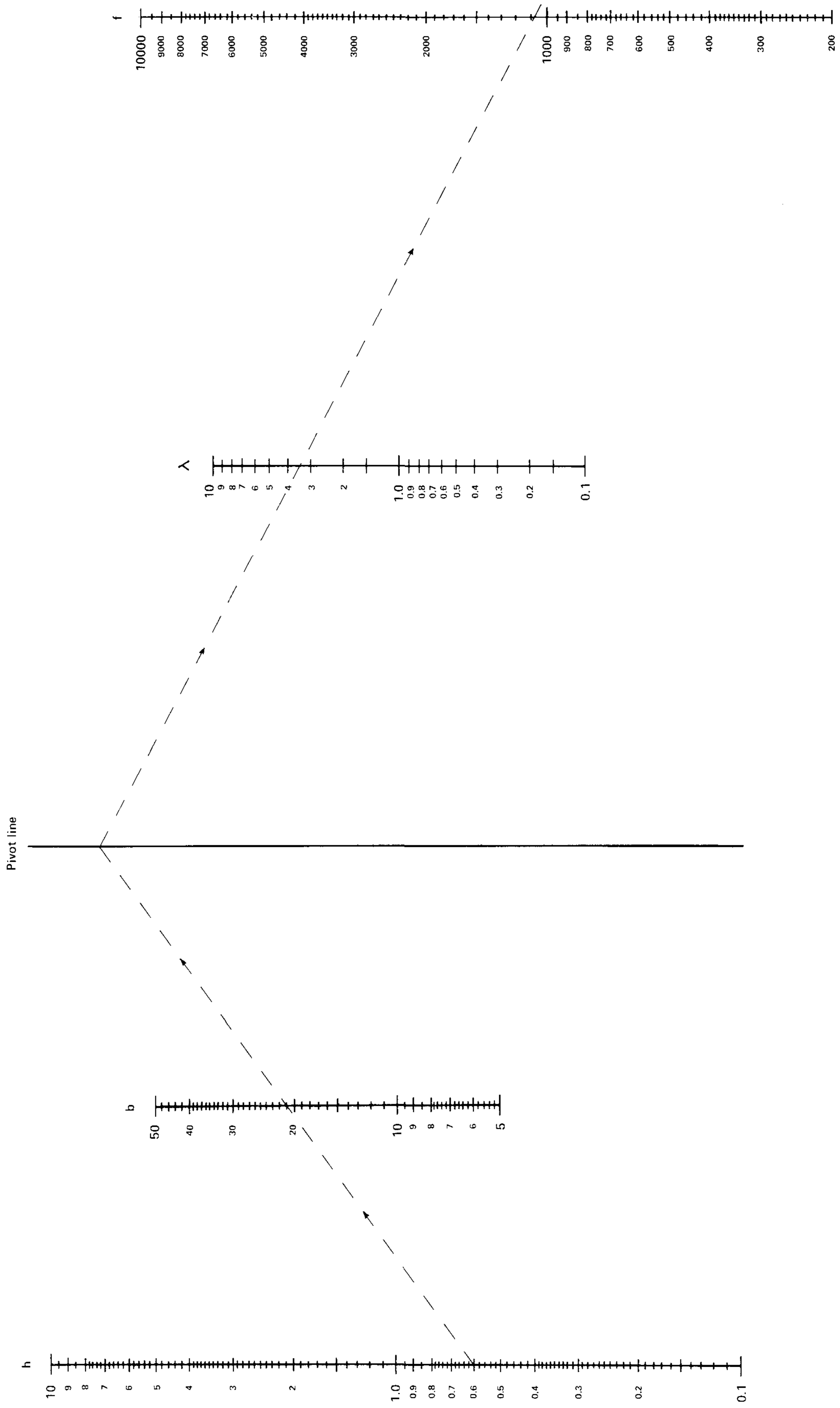
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Table 4.3.

8 4.4. PLOT OF b/L VERSUS λ



4.5. NOMOGRAPH FOR PLATE EQUATION



5. BIBLIOGRAPHY OF TEST FIXTURE LITERATURE

1. "Vibration and Shock Test Fixture Design", Tustin Institute of Technology, Santa Barbara, California, USA, 1971.
2. "The Vibration of Rectangular Plates", G.B. Warburton, Proc, of Inst. of M.E., 1953.
3. "Lectures on Shock and Vibration", Robert L. Stallard, Avco Corporation, Wilmington, Mass., USA, given March-June 1967 at Northeastern University.
4. "Vibration of Plates", A.W. Leissa, NASA Publication SP-160.
5. Handbook of Shock and Vibration, edited by Harris and Crede, McGraw-Hill.
6. Mechanical Design and Systems Handbook, Harold A. Rothbart, Editor, McGraw-Hill.
7. "Estimating Natural Frequencies", H.W. Schmitt, Machine Design magazine, Dec. 9, 1965.
8. "Speed of Sound Limits Size of Vibrating Structure", Wayne Tustin, Test Engineering and Management, Sept. 1966.
9. "Built-in Damping", R.P. Thorn, Machine Design, Nov. 25 and Dec. 9, 1965.
10. "Impedance Simulation Vibration Test Fixtures for Spacecraft Tests", Terry D. Schar-ton, Bolt Beranek and Newman Inc. Van Nuys, California, USA.
11. "Vibration of Sandwich Panels in a Vacuum", Clemans A. Powell, Jr. and David G. Stephens, NASA Langley Research Center, in Shock and Vibration Bulletin, No. 38.
12. "Formulas for Natural Frequency and Mode Shape", Robert D. Blevins. Van Nostrand Reinhold Company.

There is also a wealth of information available from:

Shock and Vibration Information Center
Code 6020,
Naval Research Laboratory
Washington, D.C. 20390 USA

Quite often an article in their publication, "Shock and Vibration Bulletin", will have information on the fixturing for a test even though the test itself is the main theme of the article.

