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An Aeroelastic Model Helicopter Rotor

by

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AN AEROELASTIC MODEL HELICOPTER ROTOR

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SUMMARY

This Report describes the design, construction and testing of a 1/5 scale model of a non-articulated helicopter main rotor. The scaling was such as to ensure correct Froude number representation.

The measured flap and lag stiffnesses of the model were close to the scaled values, although the frequencies of the fundamental flap and lag modes were lower than the design figure. The higher mode frequencies, however, agreed well with the scaled values.

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* Replaces RAE Technical Report 73042 - ARC 34836

CONTENTS

1 INTRODUCTION

In 1971 a technique was developed at RAE to measure the impedances of a spinning model helicopter rotor[1]. The preliminary rig used a rotor of 1.3m diameter, and after successful proving trials it was decided to build an improved rig capable of taking a rotor of approximately twice this size.

Although, for the purposes of these tests, it was not essential to build a scale model of an actual helicopter rotor, it was felt that it would be useful to do so as a test of the accuracy of modelling technique. The Westland Lynx was chosen as the basis for a 1/5-scale model.

This Report describes the design, construction and testing of the model, shown in Fig.1, and an assessment is made of the degree of fidelity with which the full-scale rotor is simulated.

2 DESCRIPTION OF THE WESTLAND LYNX MAIN ROTOR

The Westland Lynx has a non-articulated or hingeless rotor (Fig.2). The conventional flap and lag hinges are replaced by flexible members made of titanium, the stiffnesses of which largely determine the dynamic characteristics of the rotor in the lower order modes. Flapping freedom is provided mostly by the inboard member, known as the 'cutlet' and lagging freedom by the member outboard of the pitch change bearing, known as the 'dogbone'.

The rotor blades are of cambered aerofoil section with constant chord, and are so designed as to incorporate mass and stiffness taper. Each blade consists of a stainless steel 'D' spar bonded to a trailing edge constructed of fibreglass skins with a honeycomb interior. A mass balance weight is incorporated in the leading edge, (Fig.3). The 'D' spar has two internal webs and is built up from four sections. A number of stainless steel doublers are bonded to the upper and lower surfaces to carry the loads on the blades into the root fitting, which is attached to the outer end of the dogbone by two bolts.

3 DESIGN OF MODEL ROTOR

3.1 Scaling

As Hunt has shown[2], there are six non-dimensional parameters that determine the aerodynamic and dynamic similarity between a helicopter rotor and a geometrically similar model. These are Reynolds number, Mach number and advance ratio ($V/\Omega\ell$), and the force ratios ρ/σ, $E/\rho V^2$ and $V^2/g\ell$ (known as the Froude number).

It can be shown that it is not possible to satisfy all six conditions with a model of less than full-scale, but that it is necessary that $V/\Omega\ell$, ρ/σ and $E/\rho V^2$ should have their full-scale values to achieve dynamic similarity. Two of the remaining three parameters have to be disregarded, and since it is not generally practical to test a model rotor at full-scale Reynolds number, model rotors are usually designed to have either correct Mach number or correct Froude number.

It was intended that impedance tests should be made, at any rate initially, under still air conditions; Mach number was therefore of minor importance. However, the tests were linked to an investigation into the phenomenon of ground resonance of helicopters, and therefore it was important that all the forces acting on the rotor should bear the same relationship to one another as they do at full-scale. This condition is satisfied only if the Froude number is correct.

The model was therefore designed for this condition. The implications on the properties of the model are shown in the following table:-

Property		Scaling factor
Length	ℓ	λ
Gravitational const	g	1
Air density	ρ	1
Air viscosity	μ	1
Speed of sound	a	1
Velocity	V	$\sqrt{\lambda}$
Rotational speed	Ω	$1/\sqrt{\lambda}$
Frequency	ω	$1/\sqrt{\lambda}$
Material density	σ	1
Elastic modulus	E	λ
Mass	m	λ^3
Force		λ^3
Reynolds number	Re	$\lambda^{3/2}$
Mach number	M	$\sqrt{\lambda}$

$$\text{where } \lambda = \frac{\text{model length}}{\text{full-scale length}}$$

3.2 Blade design

Although the blades on the Lynx are cambered and twisted, it did not appear possible to produce blades of this complexity within the available time scale. Accordingly the design of the model was simplified to give an untwisted blade of symmetrical section. Apart from this modification to the full-scale geometry the general design of the model was a replica of the full-scale structure.

The fundamental problem then was the choice of material for the blade spar. The leading edge of the spar of the full-scale blade is stainless steel sheet 1.25mm thick, with a Young's modulus of about 200 GN/m^2. For a geometrically scaled replica model this part would be 0.25mm thick and made from some material of modulus 40 GN/m^2, and the same density as steel. Such a material is not readily available and skins of this thickness are not easy to handle, so thicker skins of lower modulus material were used. Fibreglass was the obvious choice, since it is easy to fabricate and its comparatively low modulus gave skins of reasonable thickness, and approximatelv correct weight.

The design was based on a Young's modulus of 11.2 GN/m^2 and the skin thicknesses are shown in Fig.4.

The rear part of the Lynx blade is of low stiffness compared to the spar, the skins being 0.46mm fibreglass enclosing an aluminium honeycomb. The model blade trailing edges were made from thin balsa sheet with chordwise cuts from trailing edge to spar every 50 mm to reduce the stiffness.

Fibreglass doublers are used at the blade root with steel attachment plates glued and bolted over the top of them.

3.3 Design of rotor hub and flexible members

The hub and inner flexible members of the full-scale rotor are a single unit in the form of a monoblock titanium forging. The outer shells of the pitch change bearings and the dogbones are also made in one piece of titanium. For ease of manufacture, the model hub is broken down into more components; four separate cutlets are bolted to a central hub, and the dogbones are bolted to a separate bearing housing (Fig.5).

Because of their complex shape and the problems of bolting fibre reinforced plastics, it was decided that the flexible hub members should be made in metal. Titanium has a modulus of about 104 GN/m^2 so for a true scale model a metal of modulus 21 GN/m^2 is needed. Since no such metal (e.g. lead) is suitable, it was necessary to use a metal of different modulus and modify the cross-section of the parts to achieve the desired scale stiffness.

A simple analysis showed that, although the parts would have a smaller cross-section if made in steel rather than aluminium alloy because of the former's higher modulus, they would in fact be stronger. The strength would be higher still if titanium itself were used, but for reasons of cost, easier working, and ready availability, S28 steel was chosen. Because the forces on the model are reduced by a factor λ^3 compared to full-scale, the model hub, although made of steel, still has a higher safety factor than the full-scale titanium hub.

Two further simplifications were made to the design. Firstly the elliptical cross-section of the cutlet was changed to rectangular for ease of manufacture, and secondly the centrifugal loads were carried through the pitch-change bearing by an angular contact ball race rather than a tension member running through the centre of the bearing as in the full-scale rotor (Fig.6).

3.4 Pitch control

For still-air tests it is not necessary to have a cyclic pitch control, so the pitch change arms in the model are linked by track rods to a fixed spider above the rotor (Fig.7). Provision is made for changing individual blade incidence by means of adjustable-length track rods; the pitch change arms can also be rotated relative to the blades. Collective pitch change is effected by moving the spider up and down relative to the rotor. The rotor shaft is carried on simple bearing housings which attach to the test rig.

Extension of the tests to the forward speed case necessitates the use of some form of cyclic pitch control. For convenience of operation in a wind tunnel this should be remotely controllable. A further condition is that the system should be interchangeable with the fixed pitch system without modification to the impedance test rig.

Such a system has been designed and built (Fig.8). It uses a swash plate mounted below the rotor and actuated by three screw jacks driven by electric motors mounted in the rotor shaft bearing housing. The position of the jacks is controlled by electrical servos. Combined operation gives collective pitch control and differential operation cyclic control.

4 CONSTRUCTION OF MODEL ROTOR

4.1 Blades

The blade spars were made from four fibreglass mouldings. They were made in moulds having both male and female parts to ensure accuracy of skin thickness (Fig.9). The glasscloth used was Marglass 116 and 101 cut on the bias, and the resin was 75% Araldite MY 753, 25% Araldite MY 750 and 22 parts of hardener MY 956 per hundred parts of resin. After trimming, the sections were assembled in jigs using packet Araldite. The last 25 mm of the root end of the complete spar was filled with Araldite.

The rear part of the blades was made from 0.8mm balsa sheet faced with 0.025mm fibreglass. The skins were stabilised by two balsa spars, and the complete unit glued to the 'D' spar to complete the blade section. At the root end a number of fibreglass doublers were bonded to the top and bottom surfaces to bring the blade up to the same thickness as the end of the dogbone, (Fig.10). The tapered steel attachment plates were hot primed with Araldite 15 (used with hardener 15) before being finally glued and bolted to the blades.

The blades were weighed at this stage. In order to bring the centre of gravity forward of the quarter-chord line, the leading-edge cavity was filled with very fine metal powder. Powder was used so that mass was added without additional stiffness. Two powders were used, one pure lead and the other a mixture of lead and copper. By using mixtures of the two it was possible to ballast the blades to the same weight; approximately 0.04 kg of powder was used on each blade bringing the weight up to approximately 0.29 kg.

The tip of each blade was finished by gluing in a lead tip weight and wooden rear tip piece (Fig.11). After the tips had been finished to a smooth profile they were painted with orange 'Dayglo' paint to improve their visibility under test conditions.

The final weight of all blades was 0.343 kg, which, although slightly below the target figure of 0.349 kg, was considered satisfactory. In all, five matched blades were made, i.e. a complete rotor set plus one spare (Fig.12). Construction time for each blade was approximately 30 man-hours, not including the time taken to make the moulds.

4.2 Hub

The flexible parts of the hub were made in S28 steel. The material was crack-detected before work started, and the parts were finished by heat-treating and grinding. The pitch-change arms and bearing housings were made in light alloy (Fig.6).

The pitch-change bearing housing has a needle-roller bearing at one end and two angular-contact ball races at the other. The roller bearing runs directly on the shaft of the cutlet to allow as large a diameter shaft as possible to be used; this is important since there should be minimal bending of the pitch-change bearing. One of the angular-contact ball races carries the centrifugal loads and the other, which is placed back-to-back, provides axial location.

The cutlets are bolted between two mild steel discs which form the central boss. These are machined so that the rotor has a 3^{o} coning angle.

5 TESTING

5.1 Tensile strength test of blade root

A 10 ton Avery testing machine was used to measure the tensile strength of the blade root. At full design speed the centrifugal load at the point where the blade is attached to the dogbone is approximately 1.5 kN, and this is obviously a potential failure area. The specimen used was a 0.5m length of blade spar complete with doublers and attachment plates. These were bolted to a plate designed to fit the testing machine. A second plate was made up with an extension which fitted inside the other end of the spar. This was Araldited in position (Fig.13).

Failure of the specimen occurred at a load of approximately 10 kN, and it was the tip fitting and not the root that failed. However since the root had shown a safety factor of at least 6 it was not felt to be worthwhile repeating the test to find its ultimate strength.

5.2 Stiffness tests of blades

The stiffnesses of the blades and their root fittings were measured in the flap, lag and torsion directions. The 'cutlet' was bolted to a steel block which itself was bolted down to a bedplate. Loads were applied to the tip of the blade by means of a small scale pan, and the deflection at the tip was measured by a linear variable differential transformer displacement transducer. The results are shown in Table 1.

Since there were only four sets of root fittings, the spare blade E was tested with the root fitting of the blade that appeared least well matched, blade C.

The flap stiffnesses are very close indeed to the design value, with blades A, B and D being well matched. Blade C is about 7% too stiff and blade E 4%.

The blades are on average about 20% too stiff in the lag plane, possibly due to the trailing edge being too stiff even with cuts in it. The lag stiffnesses of the blades are all within 3% of the average value.

The torsional stiffnesses are nearly twice the design value, with 2½% scatter in values about the mean. The values are too high because the glass-cloth used for the blade spars was cut on the bias in order to reduce the flexural stiffness of the blades; since the fibres are laid at 45° to the blade axis this results in a correspondingly high torsional stiffness. However, for the tests for which the rotor has been built the first torsion mode frequency is well above the range of interest, so the incorrect stiffness is not considered a serious defect. If this frequency had been important, it would have been necessary to lay the cloth differently and reduce the wall thickness to lower the flexural stiffness. Some additional means of bringing the spar up to the correct weight would then have been necessary.

5.3 Resonance tests of blades

Resonance tests were made to determine the dynamic characteristics of each blade and its root fitting. The cutlet was bolted to a steel block which was bolted to a bedplate (Fig. 14). The pitch horn was linked to the cruciform member used in the fixed pitch configuration which was also bolted to the steel block. Excitation was provided by a small electromagnetic shaker and the blade motion was monitored by means of a Wayne Kerr vibration meter.

The frequencies of the first four flap modes, the first two lag modes and the first torsion mode were measured for all blades (Table 2). In addition, on blade A, the mode shapes were measured (Fig. 15) and the modal dampings determined from vector plots (Table 3). Table 2 also shows for comparison a set of frequencies scaled from calculated values for the full-size rotor.

The fundamental flap and lag frequencies are both too low by an appreciable amount, although the stiffnesses are very close to the correct scale values. The probable reason for these low fundamental frequencies is that the inertias of the blades are too high, and this may have arisen because the added mass in the leading edge to compensate for the low weight of the model spar was not tapered spanwise.

There is, however, good agreement between the calculated frequencies and those of the model blades for the second, third and fourth flap modes and the second lag mode. As the stiffness tests indicated, the torsion mode frequency is too high, though not as high as might be expected from the torsional stiffness measurements.

As regards matching of the blades, none of the frequencies departs from the average value for the mode by more than 3½%.

No figures were available for direct comparison with the supplementary tests carried out on blade A. However, comparing the measured modes with a set of mode shapes calculated for a full-scale rotating blade suggests that the normalised blade tip displacements are too small; this again indicates that the inertia towards the tip is too high.

The dampings in the higher modes are all about 1/3% of critical. The first flap mode and the torsion mode are more heavily damped due to aerodynamic effects. The damping in the fundamental lag mode is very low since there is little aerodynamic effect and, due to the high edgewise stiffness of the fibreglass spar, virtually all the bending takes place in the steel dogbone.

6 CONCLUSIONS

A 1/5-scale model based on the Westland Lynx main rotor has been built and tests made to establish the accuracy of the dynamic representation.

Of the model stiffnesses, that in the flap direction was correctly scaled whilst that in the lag plane was 20% too high. Torsional stiffness was 100% too high, but could have been reduced had this been an important parameter in the tests for which the model was designed.

Fundamental flap and lag frequencies were rather low, probably due to an incorrect spanwise mass distribution. However, the higher mode frequencies agreed well with design values, and the scatter on frequencies between the blades was less than 3½% for all modes.

The requirements of Froude number scaling are such that a less efficient structure is needed for a model than for the full-scale rotor. Therefore fundamental materials problems are unlikely to arise in the design of such a model, and in fact it may prove necessary, particularly for a replica construction, artificially to increase the weight of the blades. The experience gained from the model described in this Report shows that it is possible to build a well matched, accurately scaled, model rotor. To achieve this it is essential to maintain very close weight control at all stages of manufacture since there is likely to be only limited scope for adjustment when the blades have been assembled.

Acknowledgment

The author is much indebted to Messrs. J.A. Payne and E.W. Hopgood and their colleagues in the Structures Department Aeromodelling Laboratory whose expertise and ingenuity contributed very greatly to the success of the project.

Table 1

BLADE STIFFNESSES. LOAD/UNIT DEFLECTION

Blade →	A	B	C	D	E Root C	Scaled Lynx
Flap (N/m)	29.59	29.50	32.15	29.94	31.06	29.76
Lag (N/m)	73.53	76.92	75.76	75.19	76.92	62.11
Torsion (Nm/rad)	14.88	15.60	14.99	15.15	15.50	8.40

Table 2

MODAL FREQUENCIES. (Hz)

Blade → Mode ↓	A	B	C	D	E Root C	Scaled Lynx
1st flap	2.76	2.73	2.90	2.86	2.81	3.22
2nd flap	15.47	15.17	16.05	15.40	15.40	14.96
3rd flap	38.65	38.20	38.85	39.30	38.35	38.20
4th flap	69.60	69.15	70.90	69.50	69.00	72.30
1st lag	4.07	4.09	4.13	4.06	4.14	5.01
2nd lag	40.72	40.85	42.60	40.85	42.05	41.10
1st torsion	101.0	102.0	104.2	107.0	107.2	93.0

Table 3

MODAL DAMPINGS. BLADE A

Mode	% critical damping
1st flap	0.94
2nd flap	0.36
3rd flap	0.32
4th flap	0.34
1st lag	0.14
2nd lag	0.37
1st torsion	0.83

SYMBOLS

a	velocity of sound in air
E	elastic modulus of a material
F	Froude number = $V^2/g\ell$
g	acceleration due to gravity
ℓ	a length
m	mass
M	Mach number
Re	Reynolds number
V	velocity
λ	linear scale factor (model/aircraft)
μ	viscosity of air
ρ	density of air
σ	density of structural material
ω	$2\pi \times$ frequency of vibration
Ω	angular velocity of rotor

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No.	Author	Title, etc.
1	R. Cansdale D.R. Gaukroger C.W. Skingle	A technique for measuring impedances of a spinning model rotor. RAE Technical Report 71092 (ARC 33498) (1971)
2	G.K. Hunt	Similarity requirements for aeroelastic models of helicopter rotors. RAE Technical Report 72005 (ARC 33730) (1972)

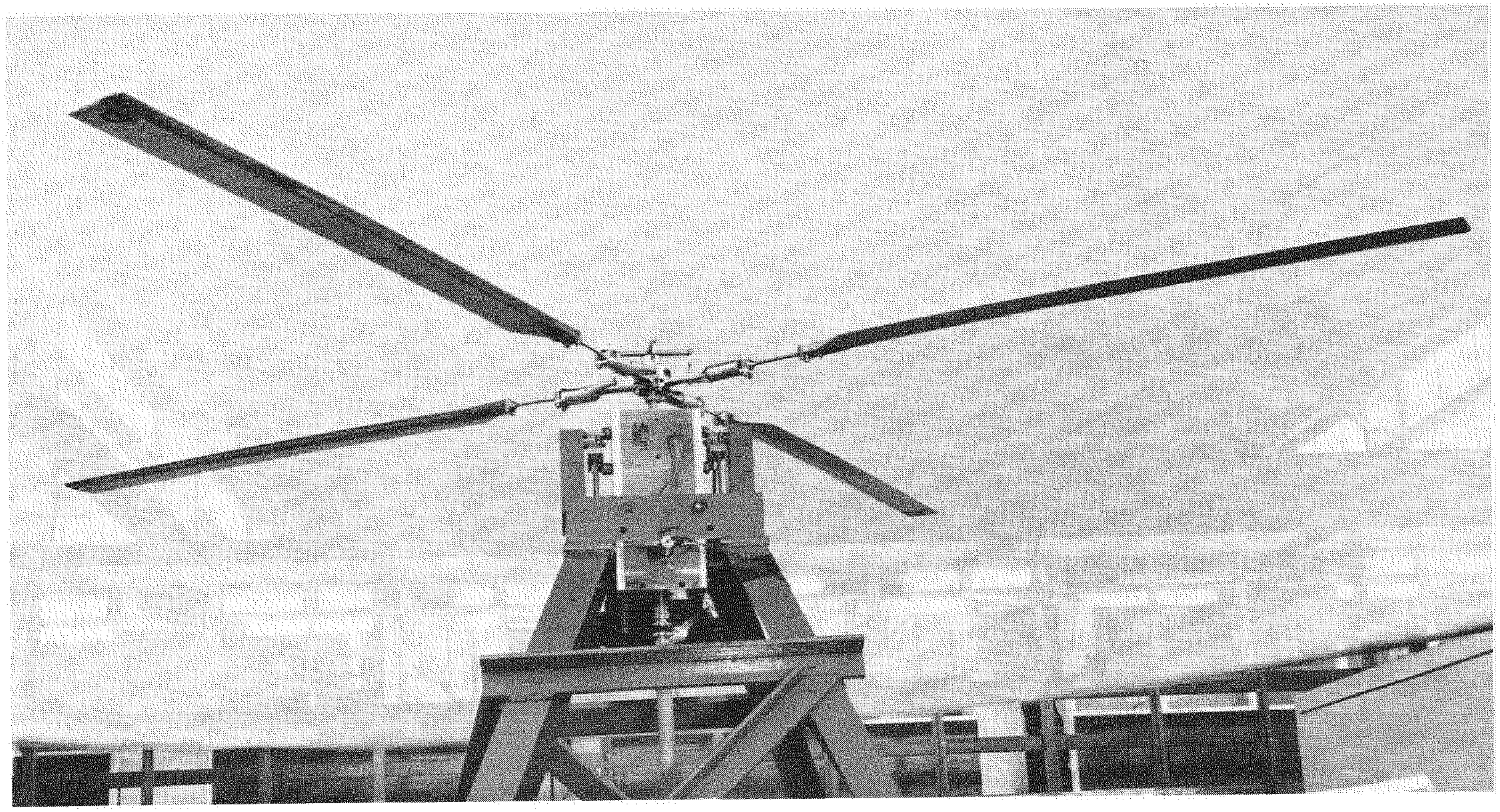

Fig.1 1/5 scale model of the Lynx rotor

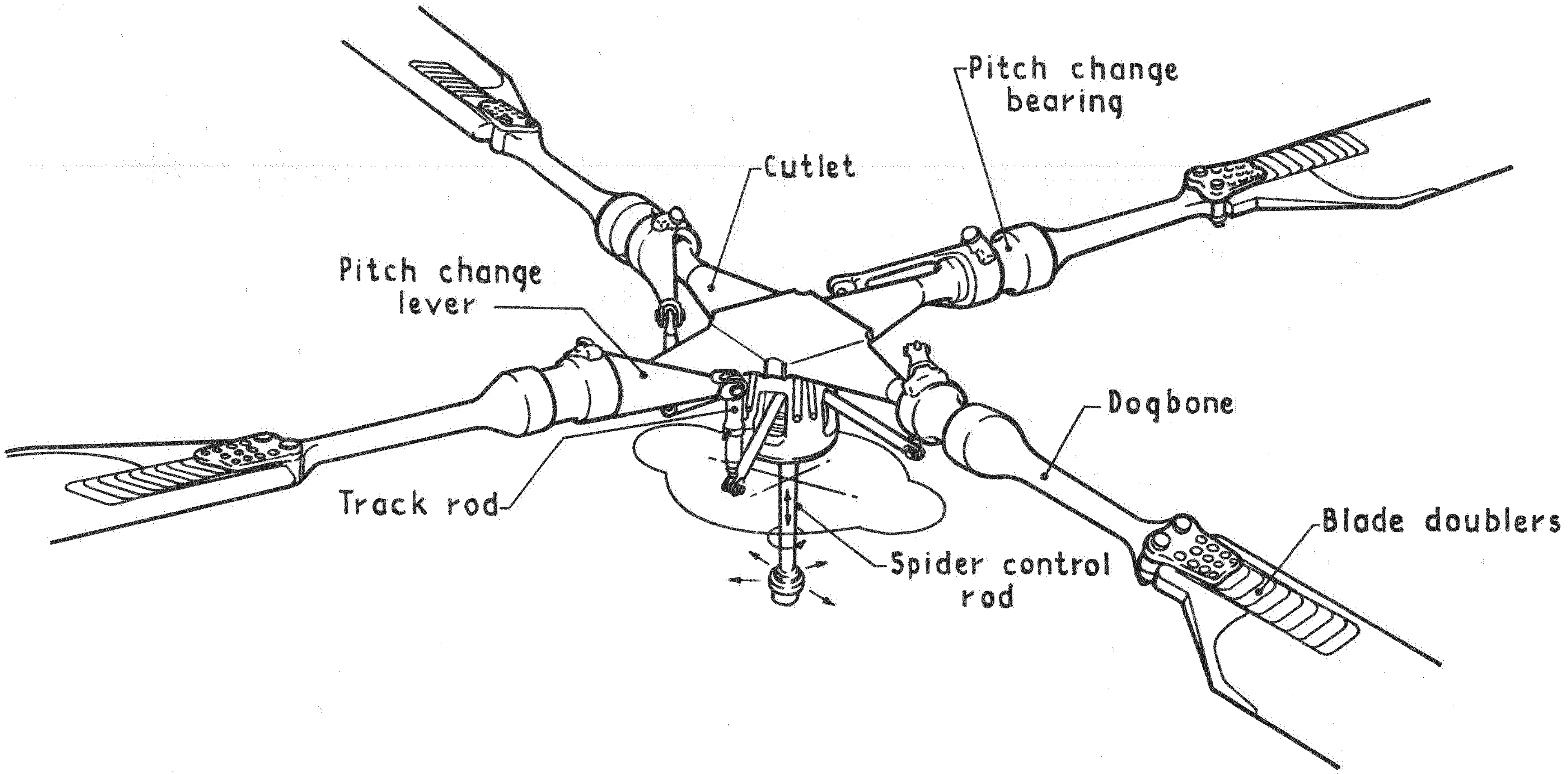

Fig. 2 Westland Lynx main rotor

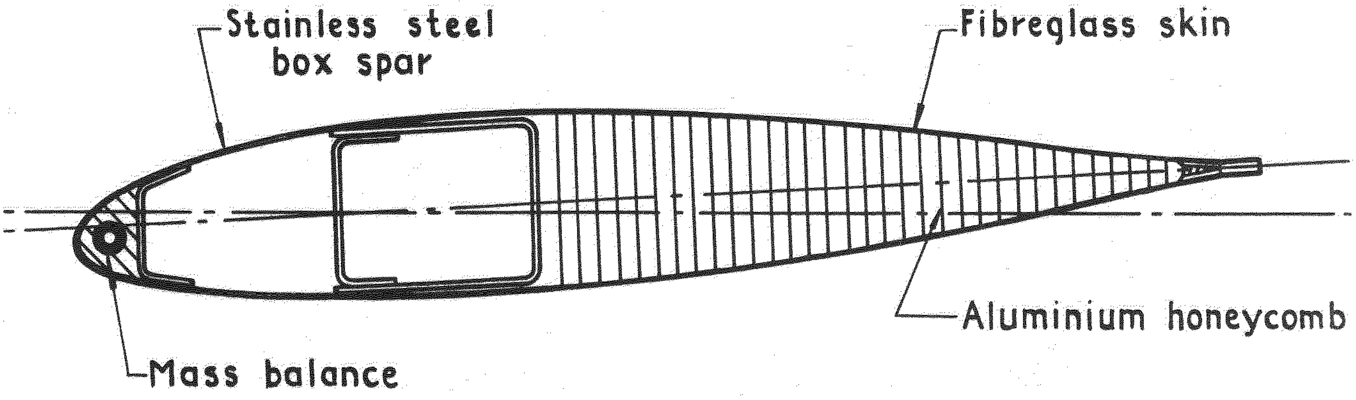

Fig. 3 Typical Lynx blade section

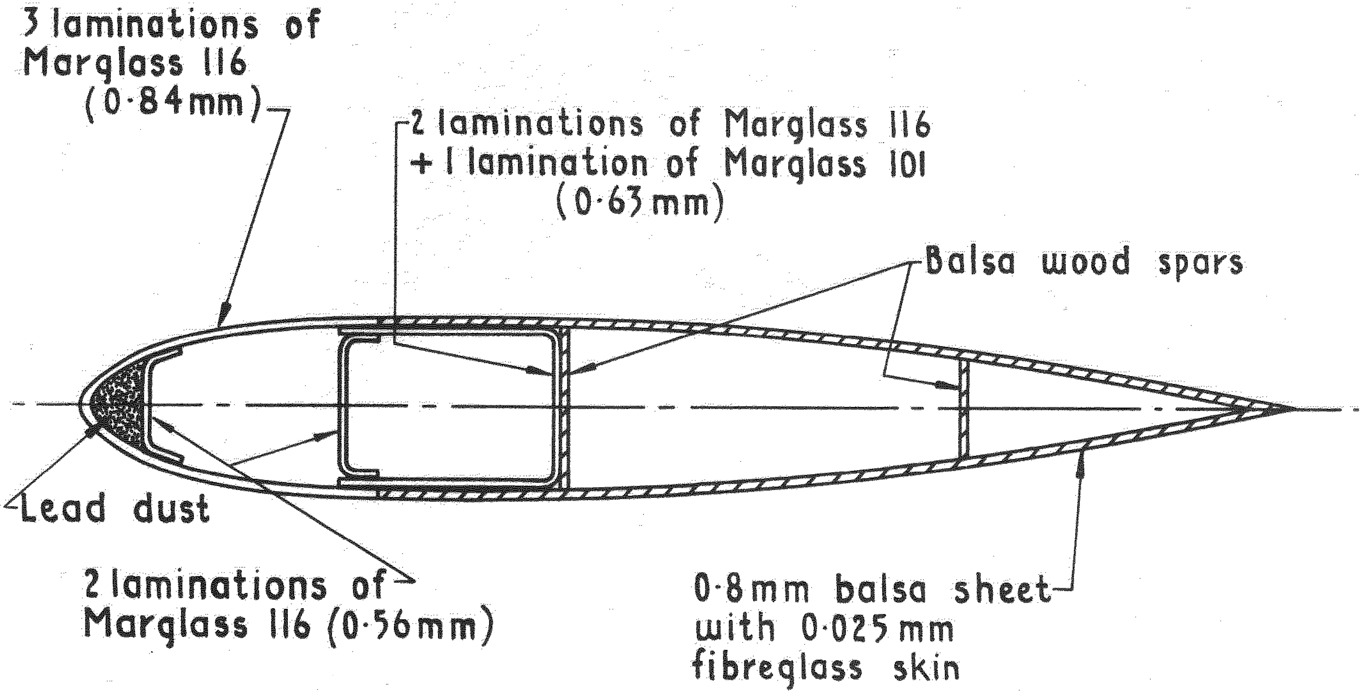

Fig. 4 Model blade section

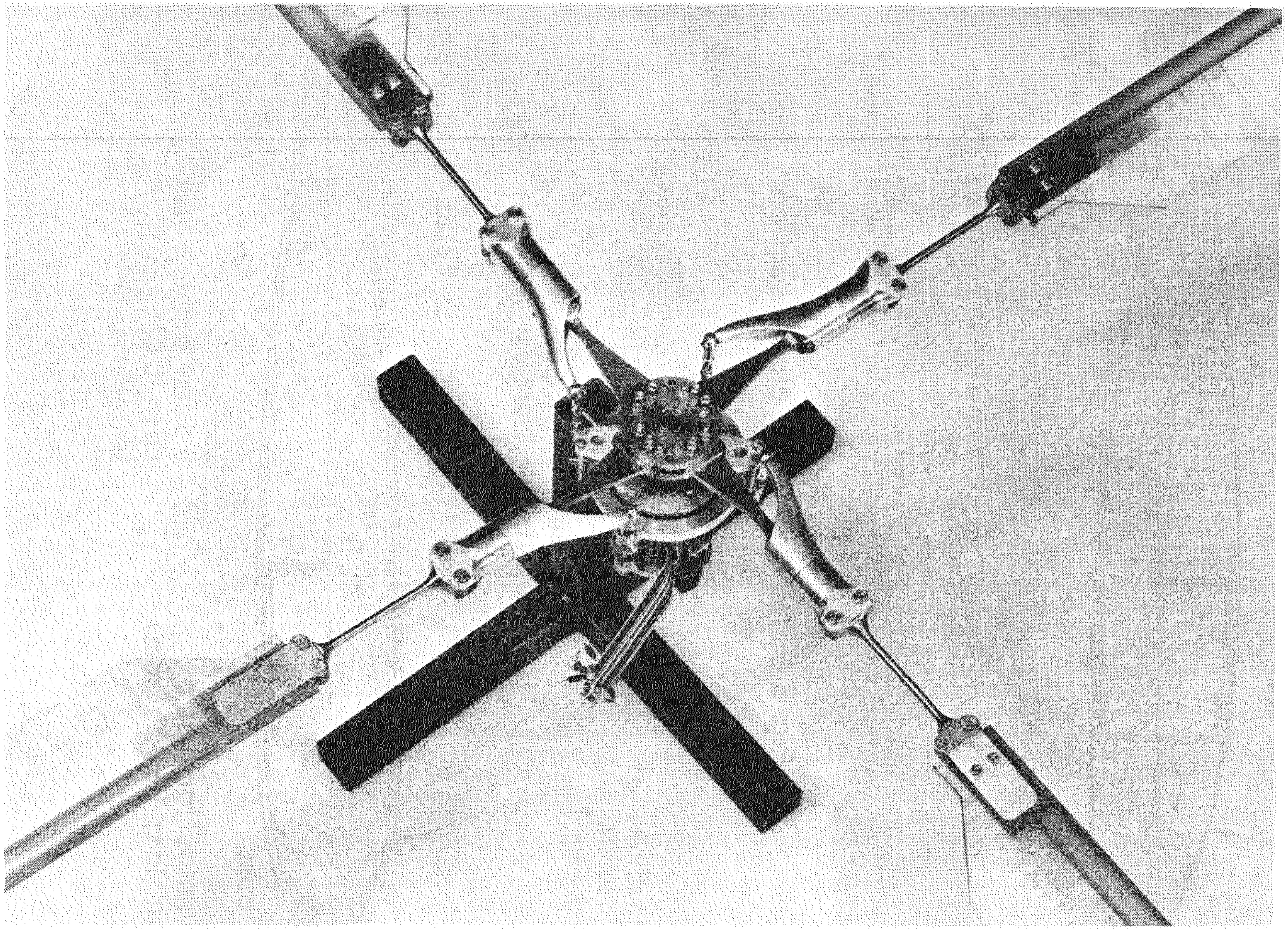

Fig.5 Model rotor head

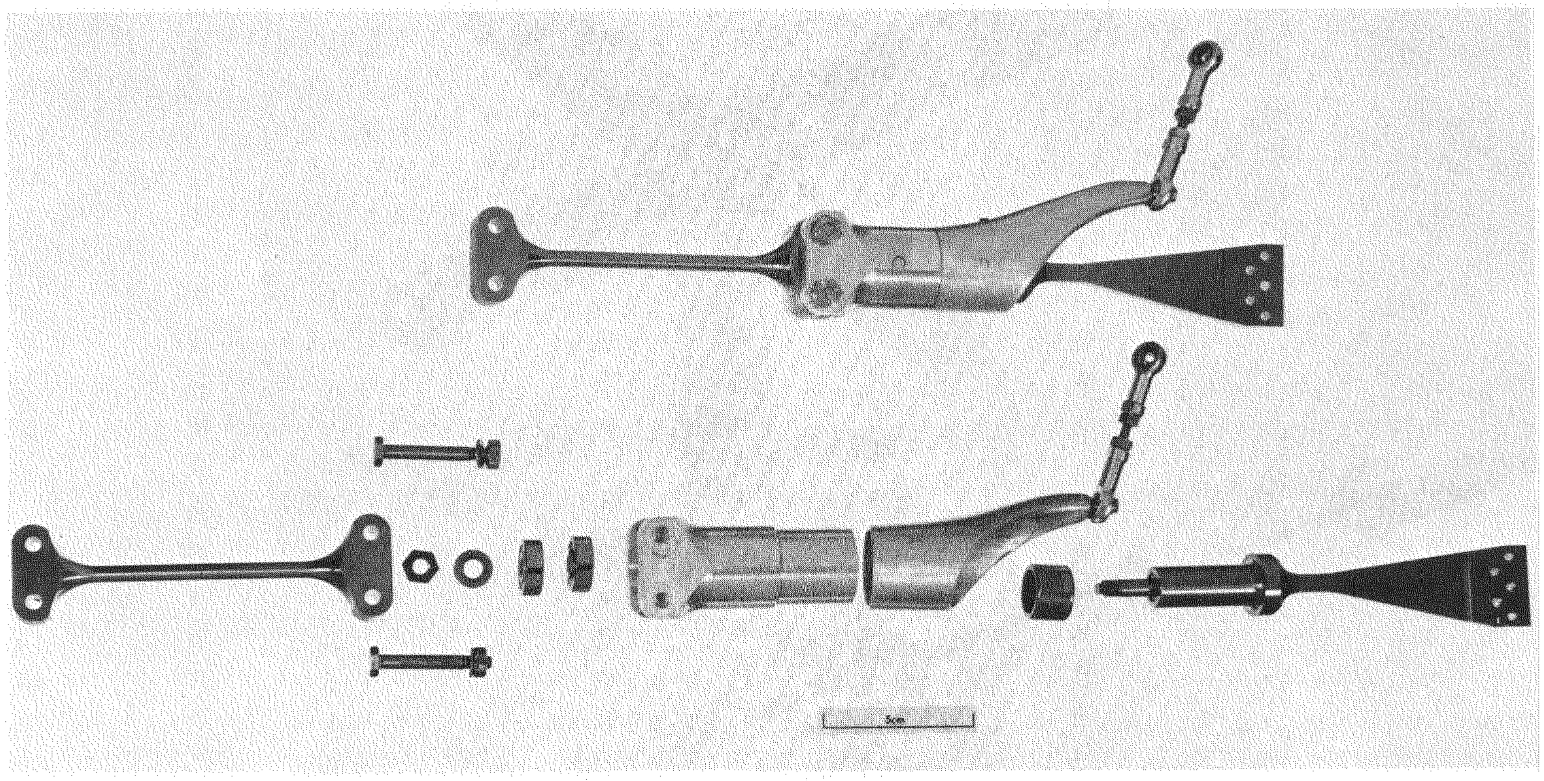

Fig.6 Rotor head components

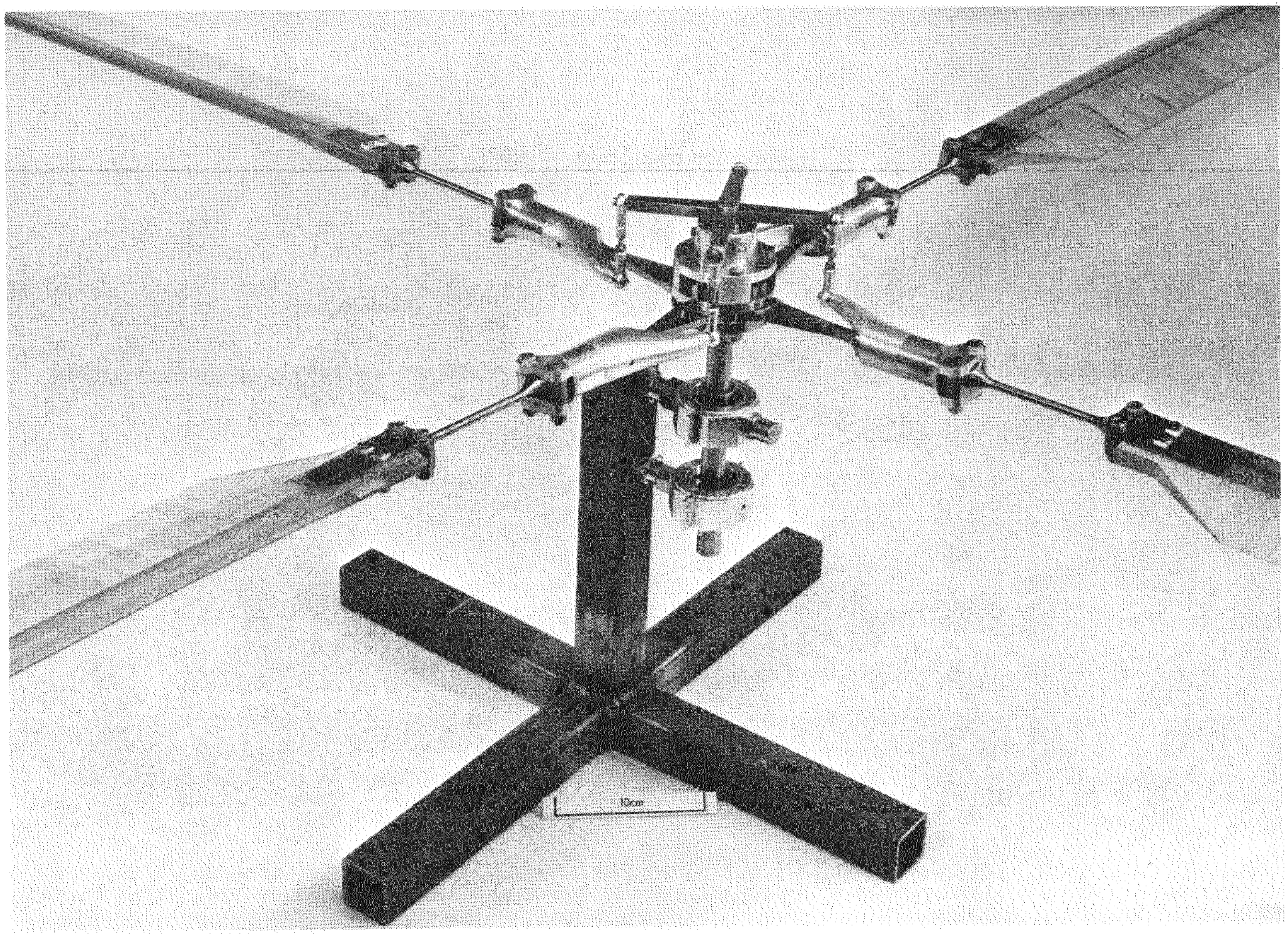

Fig.7 Fixed pitch rotor head

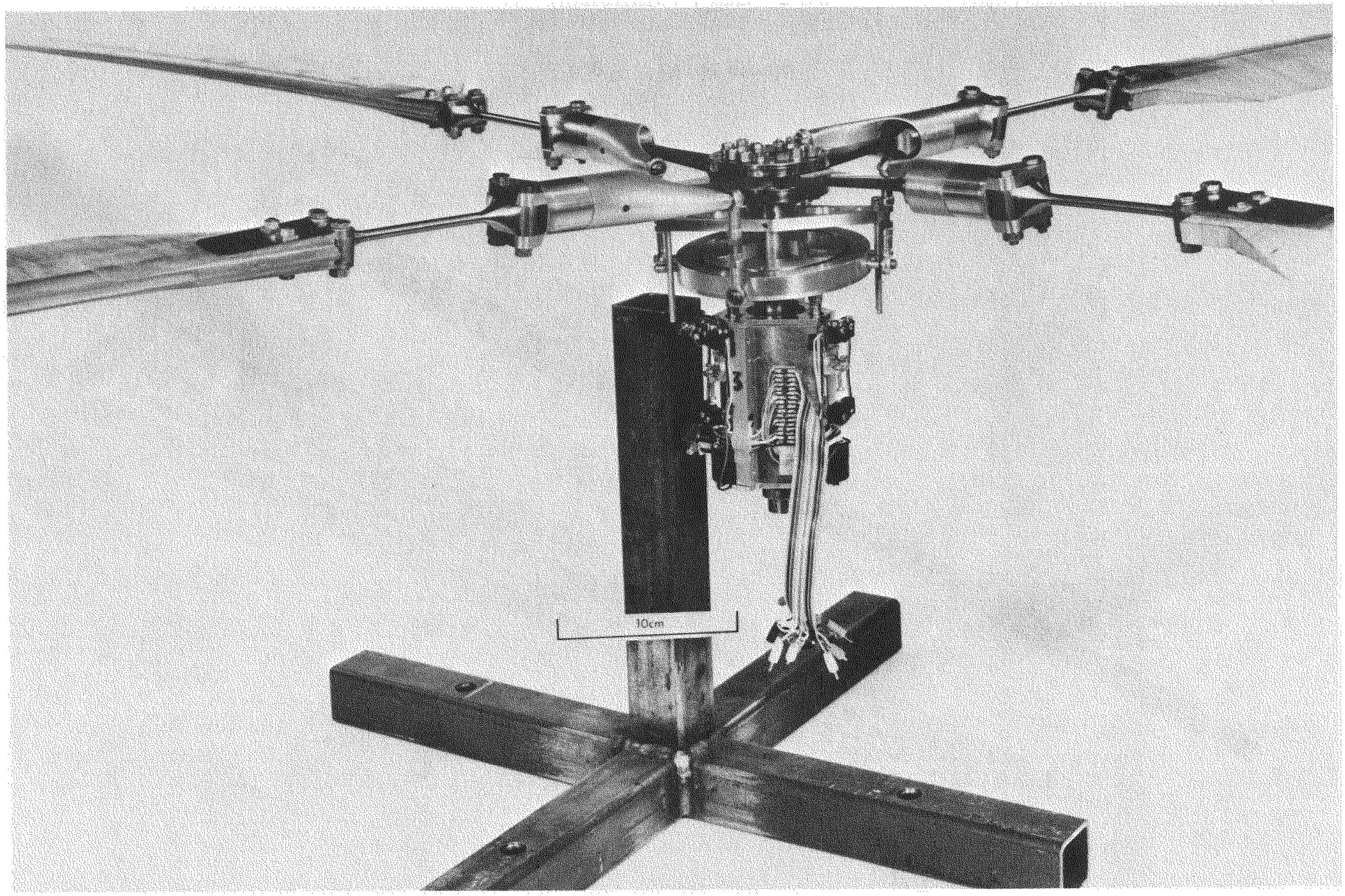

Fig.8 Variable pitch rotor head

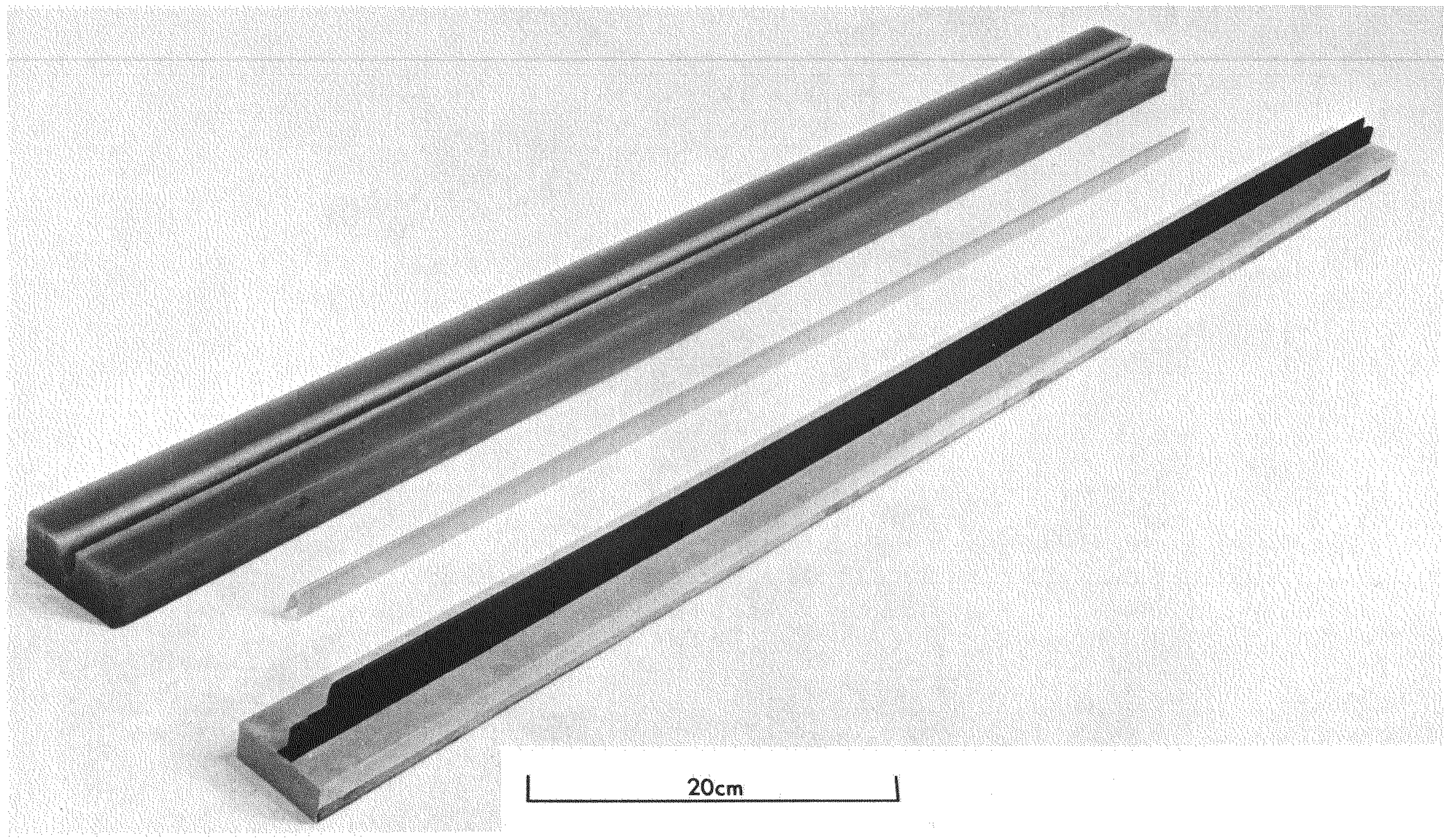

Fig.9 Blade mould

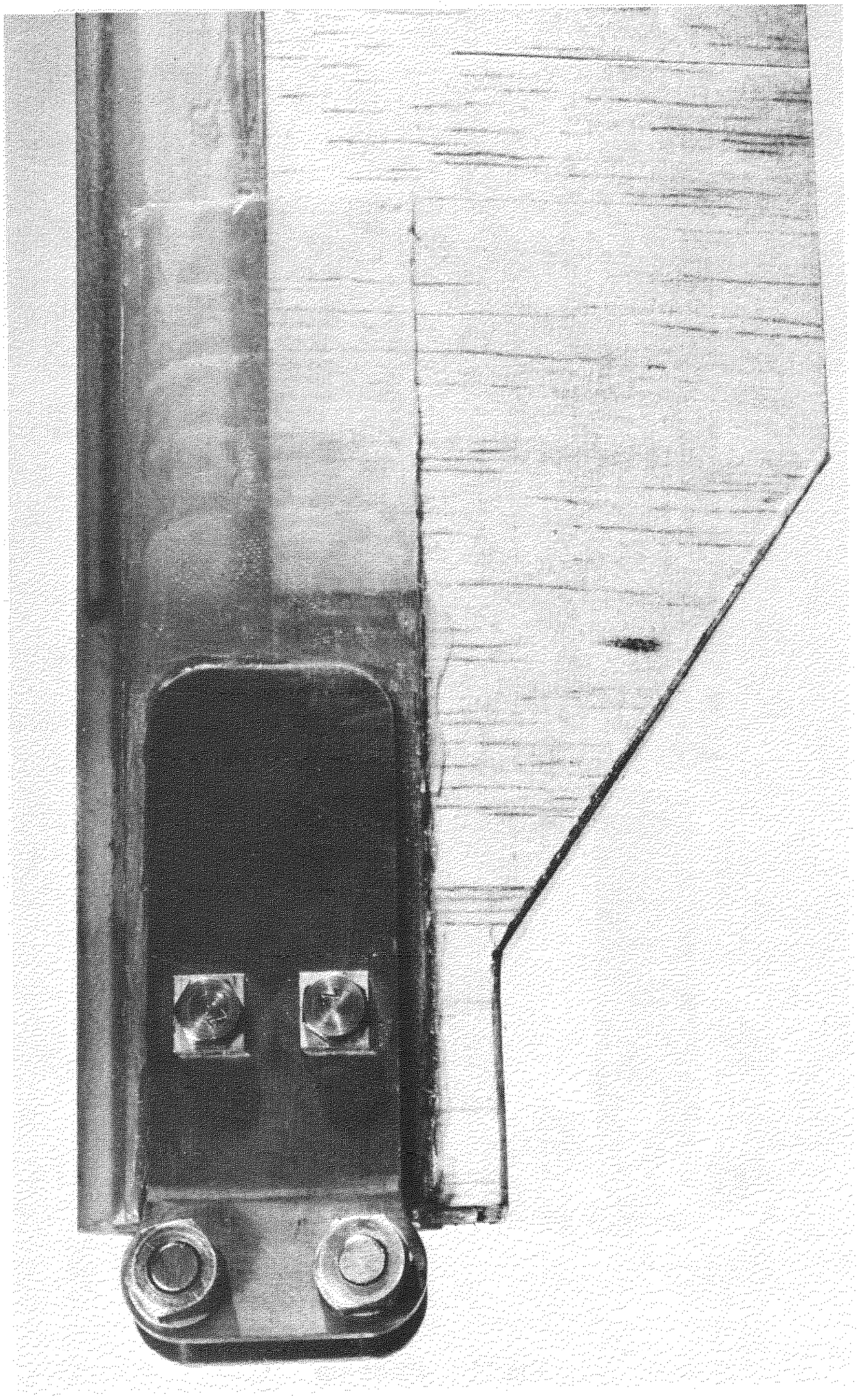

Fig. 10   Blade root

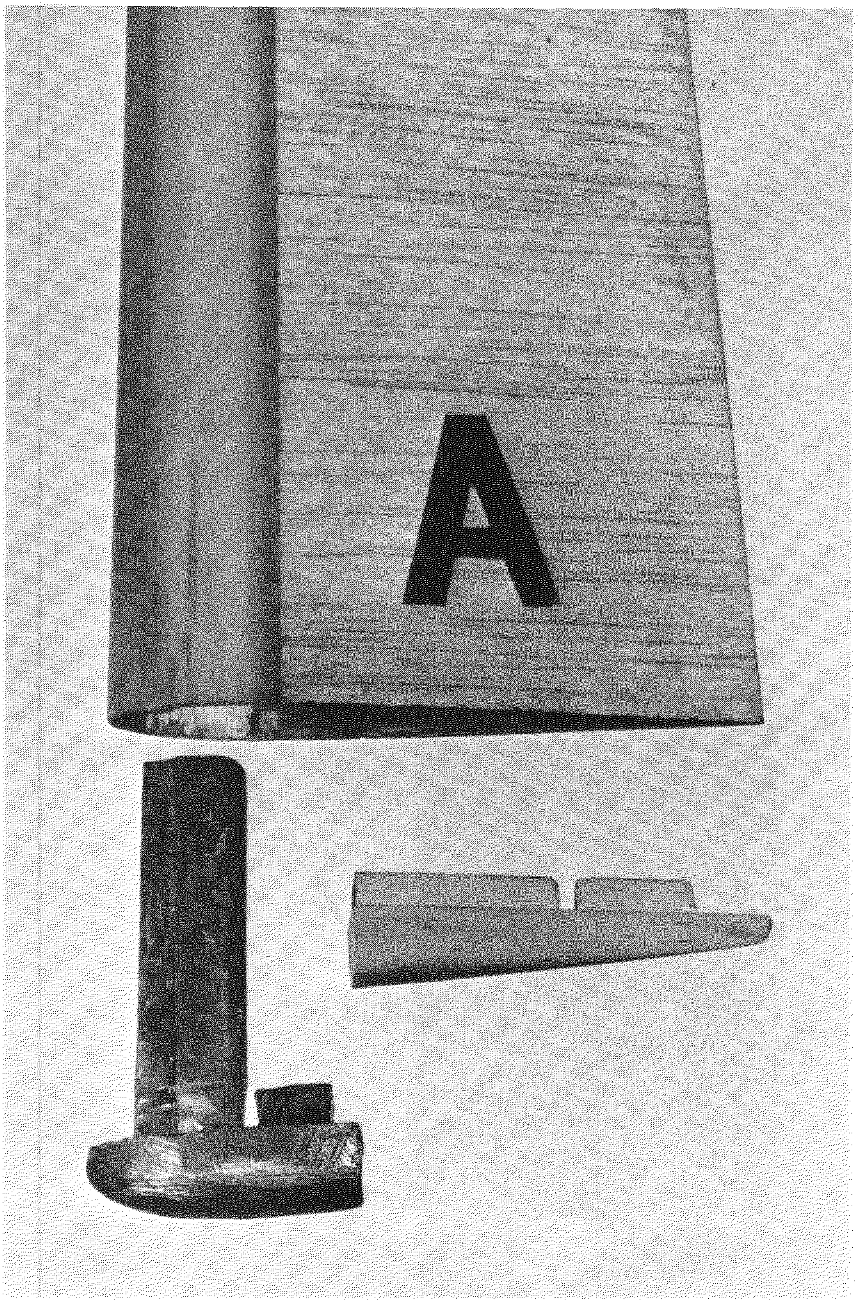

Fig.11 Blade tip

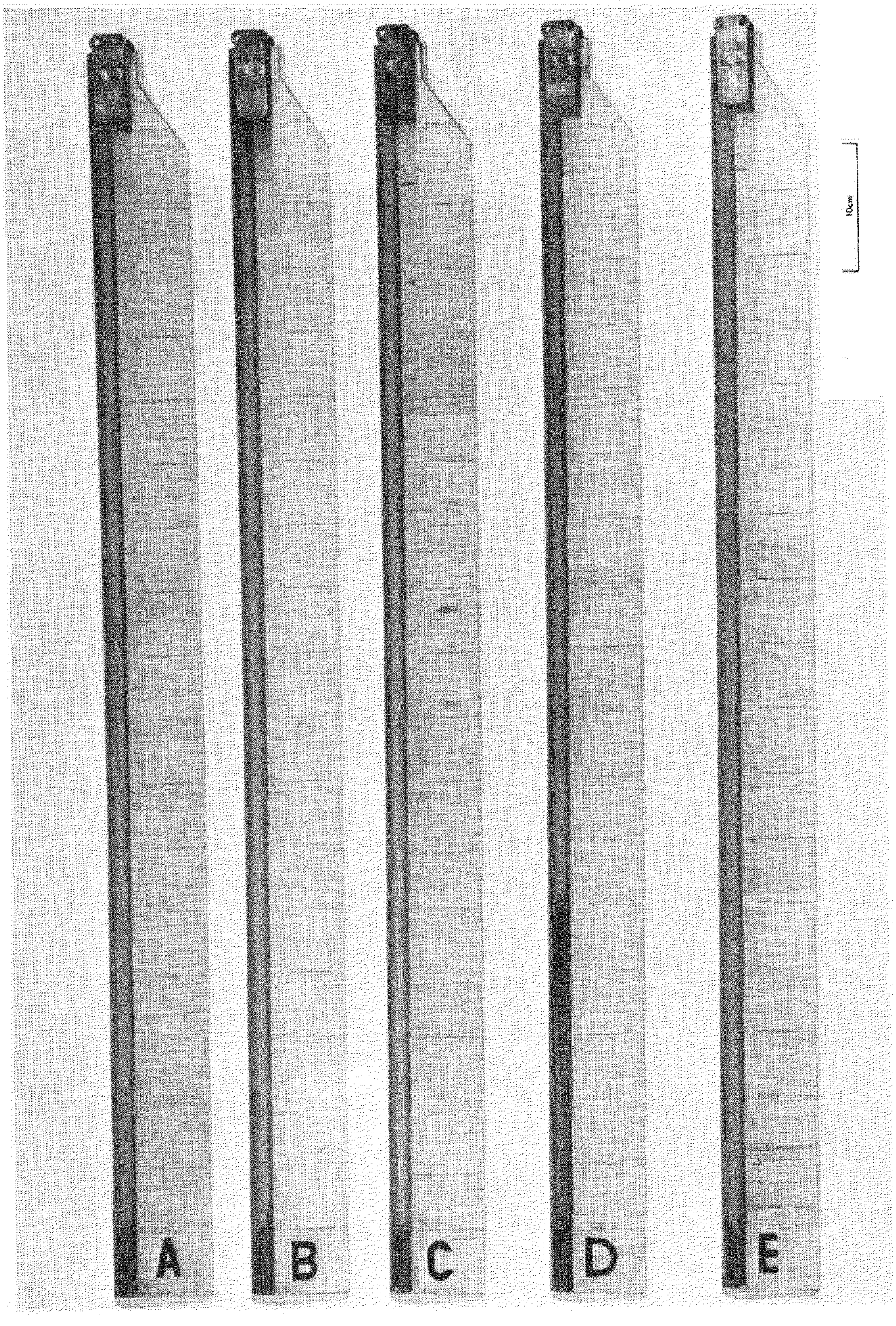

Fig.12 Finished set of blades

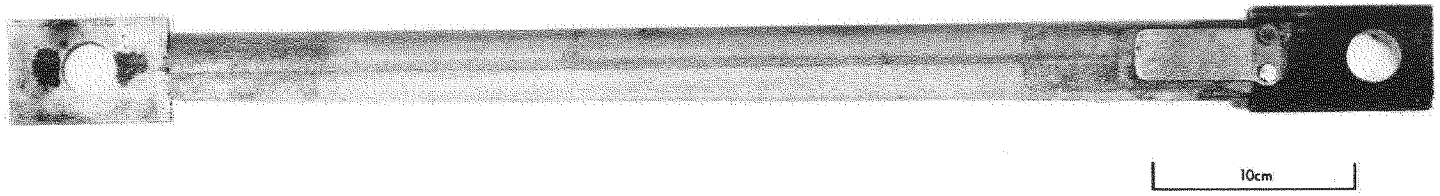

Fig. 13 Tensile test specimen

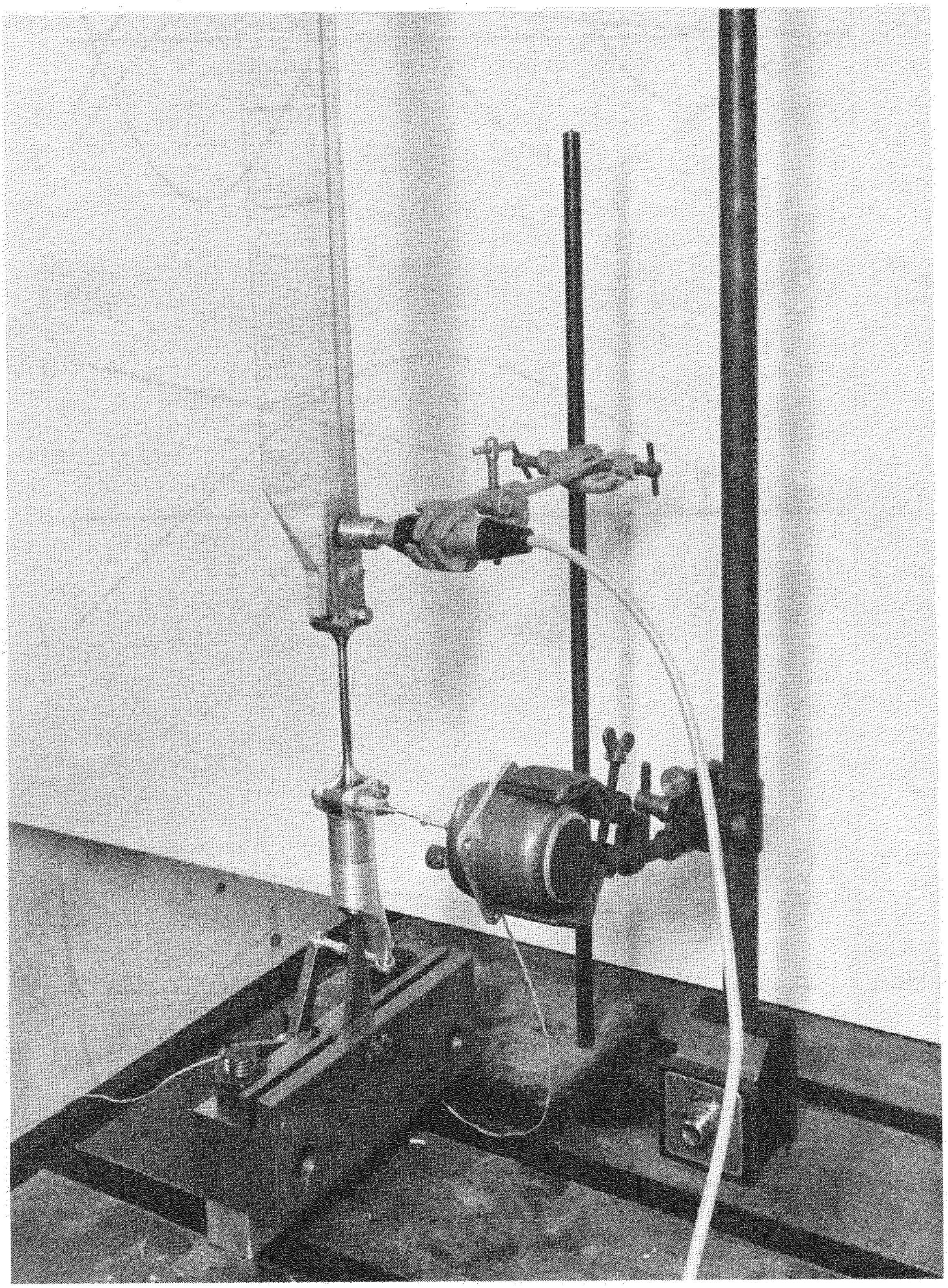

Fig.14 Blade resonance test

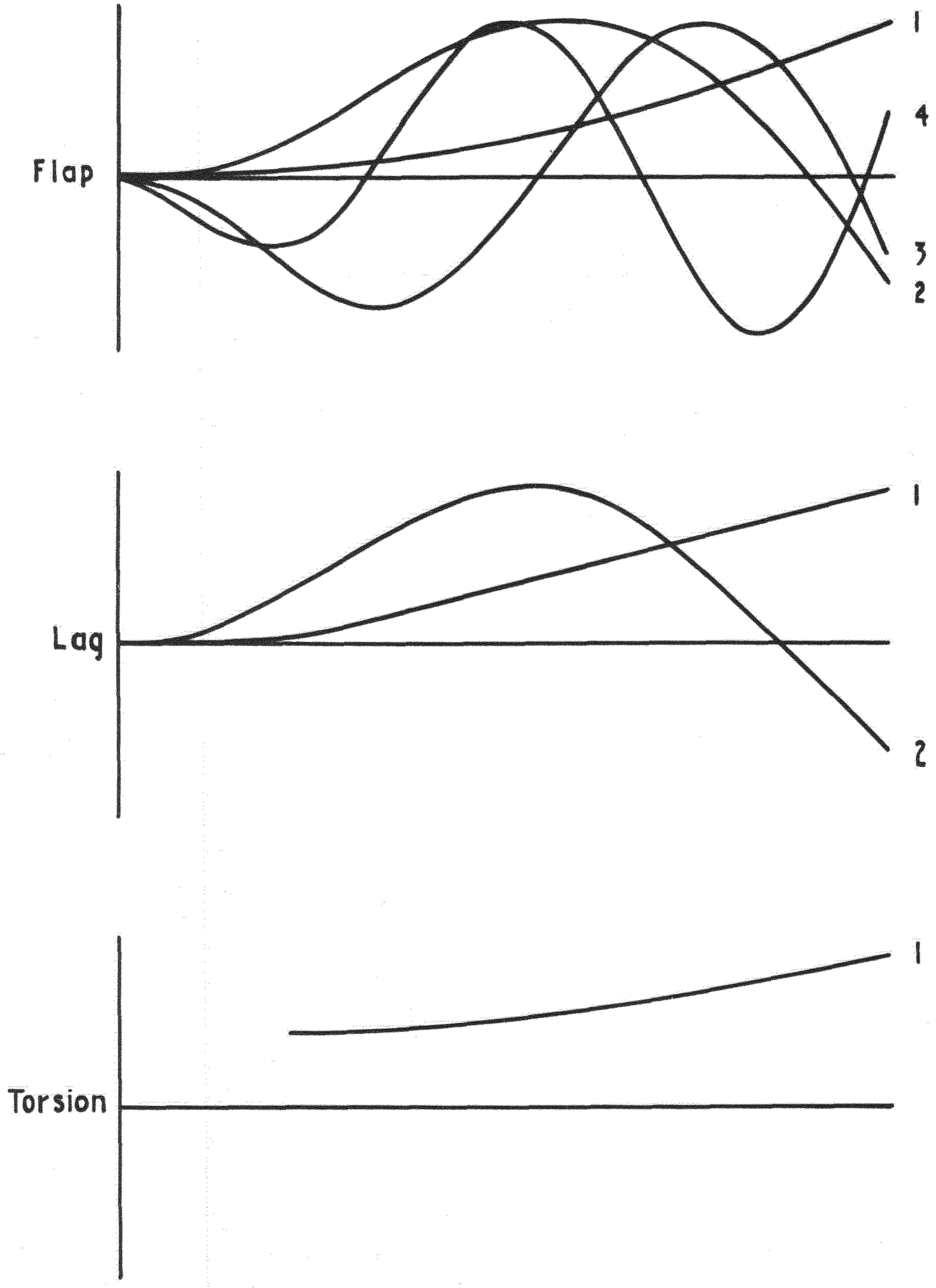

Fig. 15 Blade mode shapes

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