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DEVELOPMENT OF WIND TUNNEL FAN BLADES MADE OF COMPOSITE MATERIALS

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Abstract

To meet the increased requirements for large windtunnel-fan blades with respect to life cycles, high stiffness fibre composite materials and blade geometry open a new generation of fan design.

For obtaining acceptable stress levels and specific frequencies for the DNW (Deutsch-Niederländischer Windkanal) fan blades, carbon fibre reinforced composites have been used. The design includes a special blade attachment concept and a very detailed product assurance procedure. The design is based on the methods used in aircraft and helicopter development.

For verification of the overall static and dynamic behaviour different finite element calculations have been performed.Special attention has been given to anisotropic stress and frequency analysis and thermal calculations. Some aspects of the manufacturing process are given.

Notation

$E_{\parallel}$	[N/mm^2]	Young's modulus in fibre direction
$E_{\perp}$	[N/mm^2]	Young's modulus normal to fibre direction
$\nu_{\parallel\perp}$	[-]	Poisson's ratio
$G_{\#}$	[N/mm^2]	Interlaminar shear modulus
ϕ_F	[-]	Fibre content by volume
$\sigma_{\parallel B}$	[N/mm^2]	Tensile strength of composite in fibre direction
$\tau_{\# B}$	[N/mm^2]	Interlaminar shear strenght of composite

1. Introduction

Carbon Re-inforced Composites (CRC) have been of special interest for a long period of time in the design of aircraft and space structures. However, there are only a few primary structures containing carbon fibre re-inforcement in service, which are suitable for gaining experience about CRC in the design and service phases. In 1977 the MBB helicopter division began the development of rotor fan blades using CRC for two different wind tunnel projects. These blades are now in service and this paper gives a summary of the development work on the project.

The first project concerned the fan blade for the Deutsch-Niederländischer Windkanal (DNW) (see Fig. 1), which was set up in North East Polder in Holland. The second project was the re-installation of damaged rotor blades in the wind tunnel of the Eidgenössische Flugzeugwerke Emmen (see Fig. 2). The design of the blades is based on two different concepts; the first on the principle of a slender solid beam with hybrid-structured cross-section, the second on the principle of sandwich-shell-structures. Thus the "Emmen - blade" is comparable to the classic rotor blade of helicopters or propellers whereas the "DNW-Blade" is similar to a light-weight wing structure. Both can be of special interest to aircraft design and development personnel.

2. Project Aspects

Both projects contain detailed specifications, from which only the most pertinent requirements have been extracted as follows:

	Emmen		DNW
Power:	2 x 1430 kW		11770 kW
RPM:	390		225
Diameter:	8.5 m		12.35 m
No. of blades:	2 x 8 for two contra-rotating rotors		8
Blade length:	3.2 m		3.162 m
Chord length:	580 to 190 mm		2550 to 1250 mm
Blade mass:	27 kg		120 kg
Centrifugal load:	99500 N		270000 N
Thrust:	10100 N		36000 N
Eigenfrequency Requirements:	$15 {+1.5 \atop -0.5}$ Hz (rotating)	1. bending mode	> 34 Hz 1. bending mode (rotating)
	12 Hz (static)	1. bending mode	> 31 Hz 1. bending mode (fully coupled)
	≈ 100 Hz (rotating)	1. torsion mode	
Temperature range:	15 to 70^0 C		- 20^0 to 40^0 C

Geometry:

Given by customers model	Given by data file and tolerance specifications
1) Non-linear tilt-angle distribution along the blade length in direction of chord length and	1) Constant tilt-angle in direction of rotation 5.6^0, in direction of air flow 10.9^0
2) Non-linear twist distribution	2) Non-linear twist distribution 0 to 15.9^0
	Tolerances: Twist: $\pm 0.1^0$ Tilt angle $\pm 0.1^0$ Deviation of profile coordinates: $\pm$ 1 mm Contour deviation: $\pm$ 1 mm

3. Design Criteria

The different geometry of the blades resulted in two different concepts in design and manufacture. In order to guarantee a long service life and high safety margin, the design was based on the following criteria.

3.1 Emmen-Blade (see Fig. 3 and 4)

The expected service life of 30 years is based on:

- Large surfaces of bonded or laminated joints to reduce the stress levels in adhesives and resin.
- Distribution of uni-directional laminates over large areas to reduce stress concentrations.
- Continuous support of uni-directional laminates by foam core to prevent shape deformation and to partially transfer cross loads.
- Stress reduction by additional laminates in the area of load introduction with gradual variations in thickness.
- Prevention of local cracks by a low stress level over the blade.
- Reduction of bending moments for normal service conditions (see Fig. 5) by tilt angles.
- Elimination of secondary bonded joints by MBB rotorblade technology by means of a wet-lamination process in combination with rigid closed moulds.
- Easily changeable erosion protection strip at leading edge.

3.2 "DNW-blade"

This blade is a shell-structure, which requires a different design approach, but which maintains some aspects similar to the Emmen blade: -

- Large surfaces for bonded and laminated connections to reduce the stress levels in adhesives and resin.

- Integration of the uni-directional laminates in the shell structure over large areas to reduce stress concentrations and to obtain maximum efficiency of the inertia moment.

- Three primary spars and two secondary shorter spars are employed to improve the blades shape in the load introduction area and to transfer crossloads to the attachment structure.

- Crossloads and shear loads are transferred to the attachment structure with no load concentration points by use of a "bottom rib" clamped and bolted between the blade and the attachment disc.

- Stress reduction is achieved by additional unidirectional and cross-ply laminates to eliminate creep effects in the area of the prestressed attachment bolts.

- Minimization of maximum stresses in the load introduction region is achieved by the optimization of the tilt angles for 12 characteristic load cases.

- Easily changeable erosion protection strip at leading edge.

4. Emmen-blade Characteristics

A more detailed description of the development work for this blade is available in [1], but for this presentation it is only possible to give a short summary.
The method used in this case is the MBB wet-lamination process with rigid closed moulds. The moulds were formed from wooden blades obtained from the customer.

The foam core-structure is made out of a pre-bonded foam block and then shaped to contour by copy-milling within close tolerances.

Pre-inpregnation of uni-directional laminates by filament winding is followed by wet lay-up in the moulds together with glass-epoxy skin laminates and finally the laying up of the foam core. The mould is then closed and the blade is cured at 50^{0} C within 5 hours. To ensure a finished curing reaction the blade is tempered, after removal from mould, at 75^{0} C in an oven for 12 hours.

After milling and drilling of the blade root to fit the attachment steal disc, the blade is sealed within the fitting and finally painted and balanced.

5. DNW-Blade Development and Manufacturing

5.1 Description of Structure and Materials

The blade structure is shown in Fig. 6 and Fig. 7. With reference to the numbered parts in Fig. 6 and 7, the function of the structural parts are as follows:

1) CRC-skin has dimensions according to the torsional stiffness requirements.

2) Outer and inner integrated unidirectional CFC-laminates have dimensions according to the flexural stiffness requirements.

3) Foam core has dimensions according to the sandwich panel's flexural stiffness.

4) Sandwich spars, dimensioned to shear and cross-load transfer.

5) Joints between spars and upper shell dimensioned to shear load transfer.

6) Joints between spars and lower shell and the fittings for shell assembly are dimensioned to shear load transfer.

7) Inner and outer nose caps are dimensioned to torsional shear load transfer between upper and lower shell.

8) Trailing edge cap is dimensioned to torsional shear load transfer between upper and lower shell.

9) Sandwich tip rib dimensioned to shear and cross-loads.

10) Inner sandwich rib dimensioned to shear and cross-loads.

11) Glass Re-inforced Composite (GRC) root rib dimensioned to torsion and crossloads and according to thermal expansion requirements between blade root configuration and steel attachment disc.

12) Secondary spars dimensioned to shear and cross loads

The type of materials which were employed and their respective mechanical properties are as follows:

Fibre material:	T 300 Torayca		
Resin material:	XB 2878 A/B Ciba		
Foam:	Conticell C60 Continental		
Adhesive:	AW 106	Ciba	
Carbon fabric:	G 808	Brochier	(unidirectional laminates)
	G 803	Brochier	(skin-laminates)
	G 801	Brochier	(skin-lamiantes)

Mechanical Properties for the CRC-Laminates employed:

T 350-Fibre:	$E_{\parallel}$	=	230000	N/mm^2
	$E_{\perp}$	=	24000	N/mm^2
	$\nu_{\parallel\perp}$	=	0.28	
UD-Laminate:	ϕ_F	=	40.5	Vol-%
	$E_{\parallel}$	=	95250	N/mm^2
	$E_{\perp}$	=	6850	N/mm^2
	$G_{\#}$	=	2650	N/mm^2
	$\nu_{\parallel\perp}$	=	0,322	
	$\sigma_{\parallel B}$	= ±	1080	N/mm^2
	$\tau_{\# B}$	=	58	N/mm^2
$\pm 45^0$ Laminate:	ϕ_F	=	40.5	Vol.-%
	$E_{\parallel}$	=	9650	N/mm^2
	$E_{\perp}$	=	9650	N/mm^2
	$G_{\#}$	=	24650	N/mm^2
	$\tau_{\# B}$	=	280	N/mm^2

5.2 Test and Theoretical Investigations

During the development of the DNW Fan Blade, the high frequency requirements led to a blade stiffness design. Therefore, many theoretical investigations have been necessary in parallel to the manufacturing studies. For checking the theoretical results and the required data as weight, frequency etc. several kinds of tests have been performed in addition.

5.2.1 Dynamic Studies during Design Phase

Following the definition of the blade concept, i.e.

- outer geometry (profile, chord, thickness, span),
- type of structure (2 shells, made of fibre sandwich, 3 primary sandwich spars) and
- distribution of material,

a beam model was used to obtain the influence of structural parameters on the fundamental bending frequency. In using fibres for a construction, consideration must be given to utilizing special layers, with different fibre orientation, in order to obtain the required strength properties. In contrast to a component made from metal, a fibre component which is mainly constructed against bending moments, additional means are needed to withstand the torsional moments and shear forces. Fig. 8 shows the mass distribution of the final fan blade. The steps in the resultant line are due to the variable length of the unidirectional layers required for tensile stiffness. At the blade root are 13 layers; at the blade tip there only 1 layer exists. About 50% of the total mass is in the sandwich shells.

During the design phase, it is important to define a relatively simple model to investigate the influence of the main structural parameters. For a high stiffness structure with mass limits, a certain ratio between bending stiffness and shear stiffness, must be ensured. Therefore, the model took into account three essential parameters, the total mass M, a variable α relating to the bending stiffness, and a variable β relating to the shear stiffness.

The parameters α and β are defined in such a way, that they represent a stiffness per unit structural mass. In Figure 9 the relationship between M, α and β can be seen. Each total mass M contains the mass corresponding to unidirectional fibres for tensile stiffness, the mass corresponding to the cross ply for shear stiffness in the spars and a constant "dead" mass which represents the rest of the structure (there is the structure for torsion, the foam of the shells and spars, the paint etc.). The plot shows, that a frequency F_1 of more than 32 Hz can only be reached with a total mass greater than 102 kg. In this case the following values: $\alpha = 32$, $\beta = 8$, M = 104 kg, were chosen. These data have been the basis of the final design.

5.2.2 Stress Survey and Corresponding Tests

As well as the restriction for total mass of the fan blade and for the lowest bending frequency, there exist requirements for the factor of safety. In the case of static loads a factor of 5 was used and for dynamic loads a value of 3, corresponding to 10^6 load cycles.

As previously mentioned, the high frequency requirements led to a very stiff structure. For this reason, no problems resulted from the stress analysis especially for the blade itself. The factor of safety for the blade exeeds, for all given load conditions, a value of 20. Nevertheless, for reduction of the loads resulting from aerodynamic forces and centrifugal forces, an optimal tilt angle was defined by consideration of the elastic properties of the blade

A more detailed investigation has been performed for the blade root area. The unconventional attachment of the blade to the disc had to be examined. In order to obtain an idea of the stress distribution around the holes of the laminate, a Finite Element Method (FEM) model was prepared (see Fig. 10 and Fig. 11). Two load cases were studied. The first load condition simulated the preload of the bolts. In the second case the additional centrifugal force was applied. As expected, the higher stresses arose in the y-direction. The stress distribution for the two load conditions (see Fig. 12) differ significantly but the maximum stresses are more or less equal. The bearing pressure of 20 N/mm^2 can be neglected in comparison with the bearing strength of the cross ply laminate of about 650 N/mm^2.

In addition to this theoretical work, static and dynamic tests for the bolted area had to be performed. The test specimen is shown in Figure 13. The static test demonstrated a factor of safety of about 13. For the dynamic test, an increased load was used in order to shorten the test time. Corresponding to a number of load cycles of 10^6, a factor of safety of 3.65 was obtained.

During the design phase, a special test box, with spars and a bolted area was constructed to prove the transfer of the shear loads from the shell to the bolts. In a static test, a factor of safety of 5 was obtained. Fig. 14 shows the test box after failure. For the final design of the blade the number of spars in the blade root area were increased from 3 to 5. Therefore, a higher safety factor than 5 is ensured.

The natural frequencies of the blade are mainly influenced by the distribution of mass and stiffness and the elasticity of the attachment area. For a structure similar to the DNW-fan-blade, the bolted area can have a significant effect on them, since there exists the danger of shear deformation and the possibility of local bending of the shell. The static test of the first prototype showed that the number of spars at the blade root area had to be increased, but local bending of the shell could not be measured.

5.2.3 Dynamic Results for Final Design

The beam model used during the design phase included

- non constant distribution of mass, inertia and stiffness properties,
- coupled bending and torsion and
- elasticity of the rotor hub provided by the customers.

To obtain a better understanding of the differences between this above mentioned beam model and a more detailed investigation, a FEM-Analysis, was performed.

Figure 15 shows the idealization for the final frequency calculation. The main purpose of the investigation was to calculate the frequency difference between the two methods. Furthermore, an idea of the dynamic behaviour of the shell itself was required. The lowest local bending frequency of the shell was determined to be higher than 200 Hz.

Our contract required that we guaranteed a fully coupled frequency, in the rotating system, higher than 31 Hz. This frequency has to be achieved while considering the elasticity of the rotor hub. That means that, for the frequency qualification test (rigid mounted blade without rotation but with original attachment), a somewhat higher frequency is required. There are three parameters which influence this frequency:

- elasticity of rotor hub - ΔF 2,5 Hz
- trim mass at blade tip - ΔF 3,5 Hz
- rotating to non-rotating system - ΔF -0,8 Hz

With these considerations, a test frequency of about 36 Hz was to be expected. The final design was based on a further conservative assumption, so that a frequency of about 42 Hz for the FEM model and about 41 Hz for the beam model was anticipated.

Figure 16 compares the mode shapes of the calculated and measured fundamental frequency F_1. The great efforts that were made, to get a realistic distribution in bending and shear stiffness, were confirmed by the good correlation between theoretical and measured frequencies on the one side and the good correlation of the calculated frequencies with different mathematical models on the other side.

The higher frequencies and the corresponding mode shapes are displayed in Figure 17. The lowest torsional frequency lies at 77 Hz. That means a non-dimensional frequency of about 20 times the rotor speed. This high frequency ensured that problems relating to classical flutter would not be encountered.

5.3 Manufacturing of the DNW-blade

5.3.1 Tools

The three main tools employed in the manufacturing process are:

- a very precise mould
- a very precise milling jig
- a drilling template

The high accuracy of the mould and milling jig is defined by the male half-models of the blade. Each half of this model has a rigid plane steel fundament on which contoured plates, produced by numerically controlled milling, are fixed in position (see Fig. 18). The smooth outline was produced by finishing of the jig with GRC material and wet grinding of the surface.

The mould and the milling jig fittings are made from the models by a conventional pattern moulding process. A drilling template is made to fit the blade contour and a master template, which is also used to drill the attachment disc of the rotor. The drilling template is fixed to the blade so that hand-drilling can be carried out.

5.3.2 Manufacture of Blade

The manufacturing process has 41 work phases with appropiate references of drawings, manufacture and inspection instructions.

The bonding lines of the spars cannot be inspected after assembly, thus a fitting configuration has to be provided in order to ensure an optimal joint between the spars and the lower shell. In Fig. 19 the principle of the joint is shown.

The edge of the spar has a slit in its foam core, which allows the spar skin to bend when the fittings are joined. The flexibility of the skin defines the bonding pressure and partially eliminates tolerances. This joint was tested by means of a small box beam test for cross load and shear load transfer (see Fig. 14). After all the lamination work is completed the blade is tempered at 70^0C and then cut to length within the milling jig on a standard horizontal drilling machine.

With this process the blade position is defined:

- radius position
- tilt angles
- twist angle.

The drilling procedure is shown in Fig. 20. The last structural member is the bottom rib, which can only be fitted to the blade when the attachment plane is completed. This rib is pre-manufactured with its holes drilled from the master template and has bonded steel strips in the clamped area against fretting between the steel disc and GRC rib.

Finally the blade is primed and painted with an erosion resistant polyurethane-varnish, which is also used on helicopter rotorblades to protect against rain and dust abrasion. To protect the leading edge against the impact of small solid or liquid particles, a replaceable polyurethane film strip, thickness 0.6 mm, is bonded to the blade.

5.3.3 Quality Control (QC)

Weight:

The three requirements for the blade properties in general resulted in a great deal of effort in QC:

- high flexural stiffness at minimum weight to meet the natural frequency requirements

- total weight limited to 130 kg (including metal parts)

- high factors of safety: 3 for dynamic loads
  5 for static loads

The minimum weight demands economic application of the materials expecially of resin and adhesives in the bond lines and fittings between the following:

- foam core blocks

- foam core and laminates

- reinforcing laminates within the attachment area

- spar connections to the shell

- leading and trailing edge laminates and the shells.

The inspection of each work phase ensures that the material input corresponds to the weight and structural strength requirements.

Geometry:

The tempering of structures always is a critical exercise especially for shell structures. The deformation by the tempering process with the first prototype blade was checked by means of a theodolite with stadia lines, the blade being held in the horizontal position. It was found that within the limits of the apparatus precision no shape deformation was detected i.e.: 0.1 mm normal to the blades surface. The determination on the first prototype was carried out to demonstrate that the geometric tolerances are within the required limits. This was tested by a three-dimensional measurement including: -

- profile accuracy

- twist distribution

- accuracy of tilt angles

- position of axis.

A cross check was performed on each serial blade placed on a permanent fixed attachment plate by measuring the position of the tip profile on the blade. The results are shown in Fig. 21.

Static Moment of Inertia

There were three requirements to be satisfied:

- equal static moment for every blade

- total weight difference between any two blades has to be less than 1 kg

- interchangeability of all blades including two spare blades.

It is still not possible to meet these requirements with the possible weight tolerances of FRC's only. The deviation in the blade weights was improved during the manufacture of three blades. The requirements were verified by three trim-weights; one at the tip and two at the root.
The determination of the final blade weight and the static moment was carried out by sensitive balancing of all the blades minus trimmings and by calculation of the centre of gravity and an optimizing computer program , which defines the single trimming masses for each blade.

6. Conclusion

The two projects on the different fan rotor blades were completed within two years and this includes all stages from the design to installation and also the manufacture of all tools and appliances. Arising from the experience gained from the serial production and the service of helicopter blades in GRC, it can be said that this development work is a good basis for rotor blade design in the next generation, expecially for wind converter systems of the near future. The reasons for this conclusion are as follows:

- The material and design are understood well enough to ensure the production of structures, which are capable of being highly statically or dynamically loaded.

- The two different concepts in design, together with their combinations, offer many possibilites which can be used to determine special blade or structural requirements.

- Structures for various types of load introductions are developed and tested.

- Most of the required quality control procedures have been developed to standards for the control of manufacture and materials.

- It was possible to obtain a great deal of experience in tools and appliances, which are necessary for FRC work, however it must be noted that there are still several procedures, where development has still to be carried out.

- Finally it has been shown that the most advantageous properties of FRC-materials are still valid in large structures and thick laminates.

7. References

1) H. Bansemir, Windkanalflügel in Faserverbund-Bauweise für die Eidgen. Flugzeugwerke Emmen/Schweiz, DGLR-Symposium "Ermüdungsfestigkeit von Flugzeugen und modernen Bauweisen", Darmstadt, Sept. 1978

Figure 1: Rotor Blade of the Deutsch-Niederländischer Windkanal (DNW) Shaker Tested

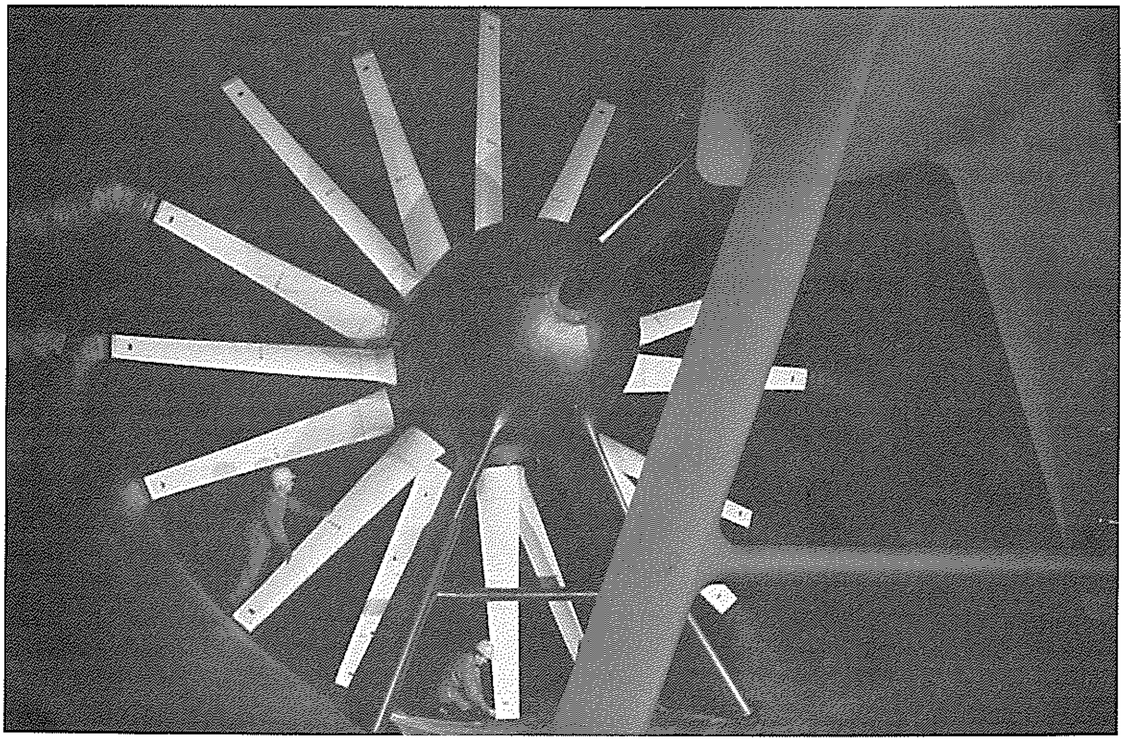

Figure 2: Fan of the Windkanal in Emmen with two Contra-Rotating Rotors

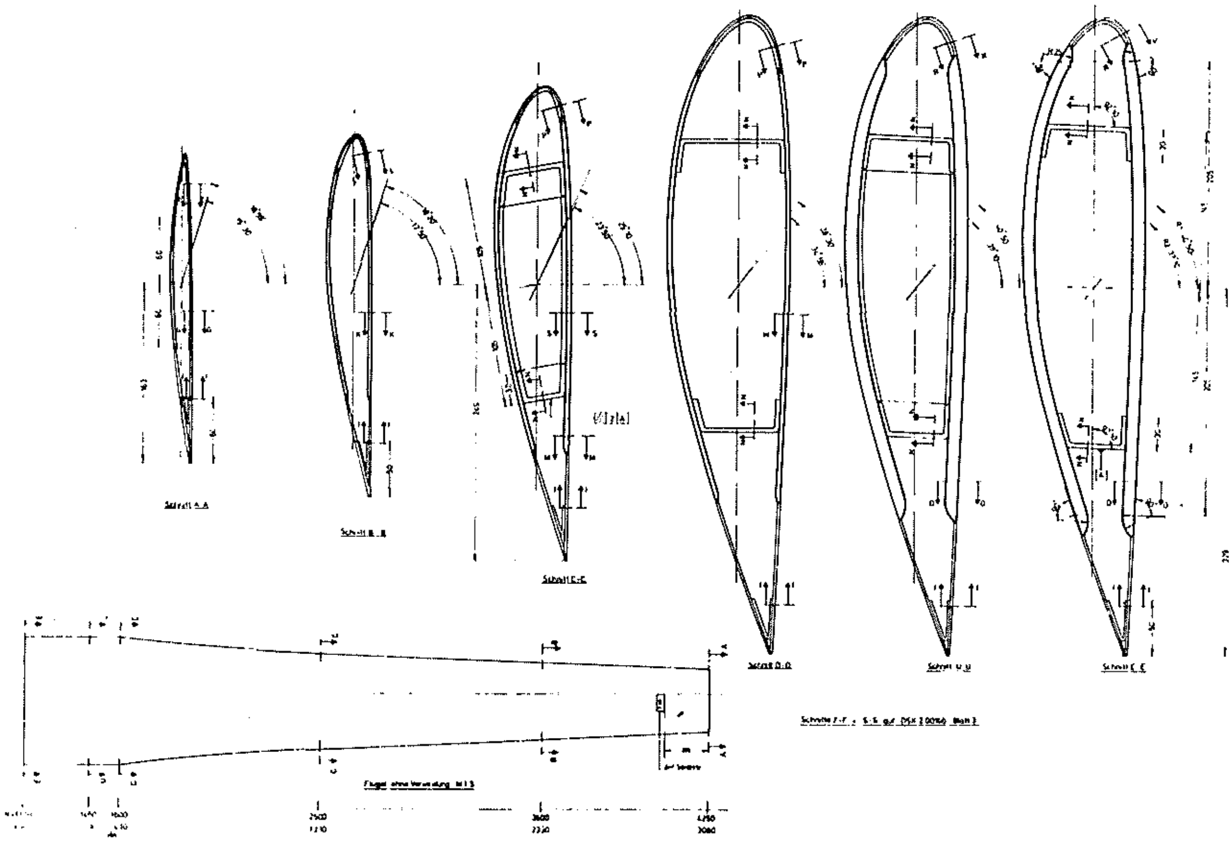

Figure 3: Rotor Blade of the Wind Tunnel in Emmen

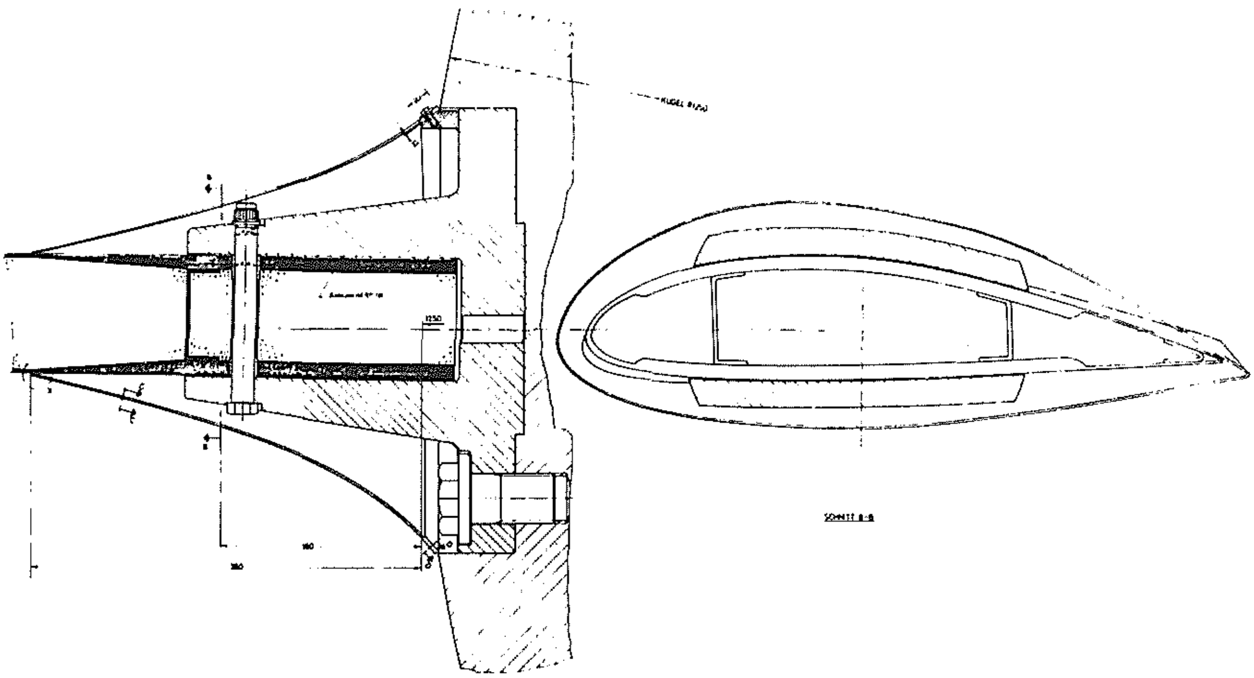

Figure 4: Rotor Blade of the Wind Tunnel in Emmen, Attachment Configuration

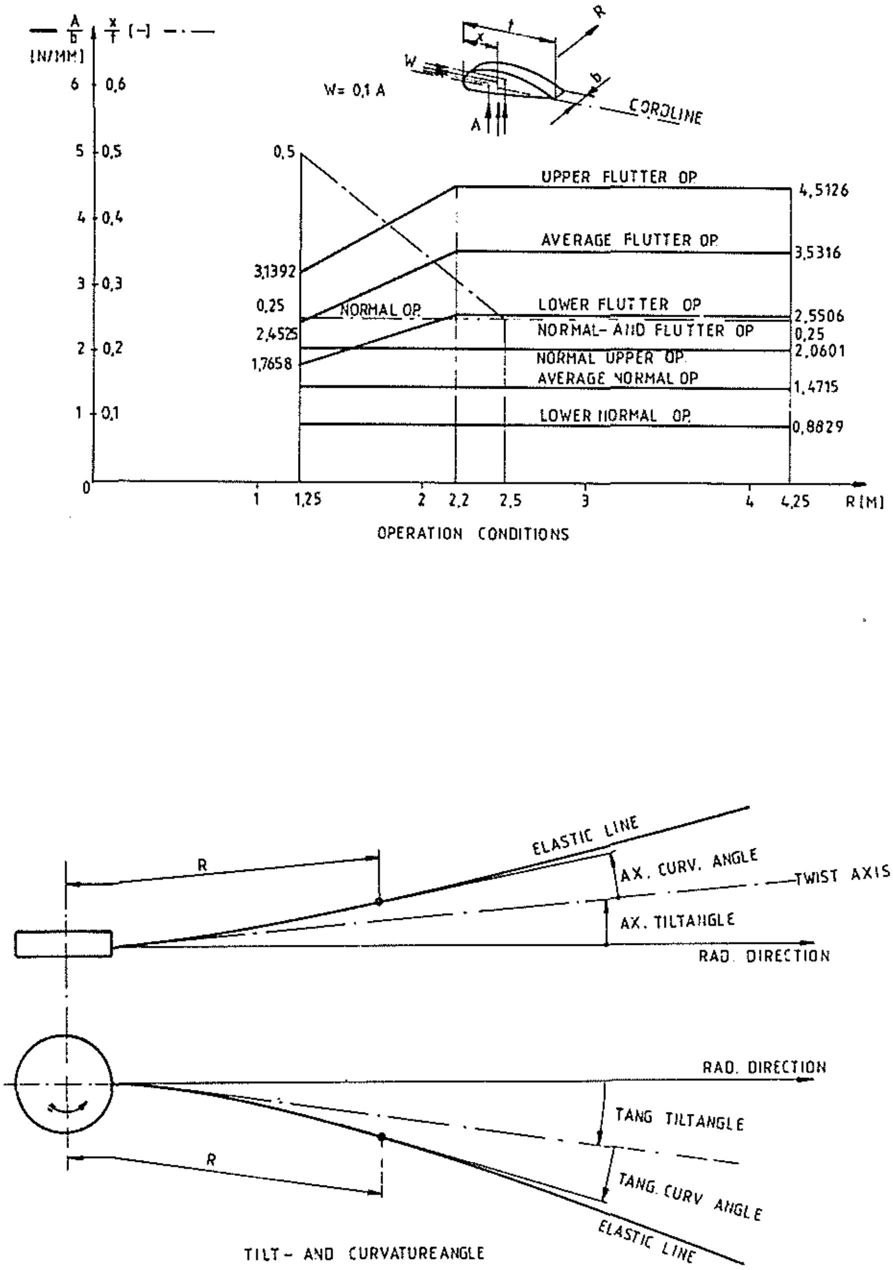

Figure 5: Tilt Angles for Reduction of Bending Moments for Normal Load Condition

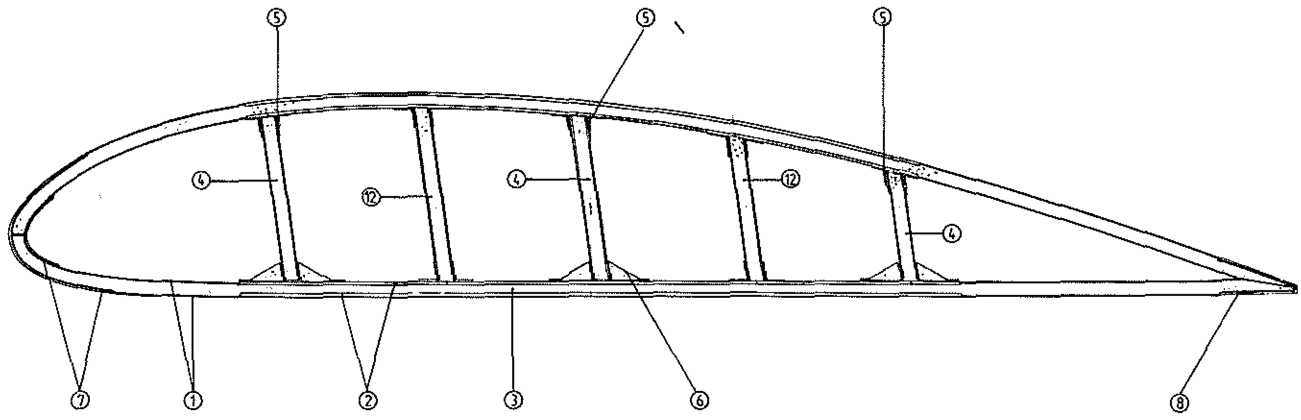

Figure 6: Cross-Section of DNW-Rotor Blade

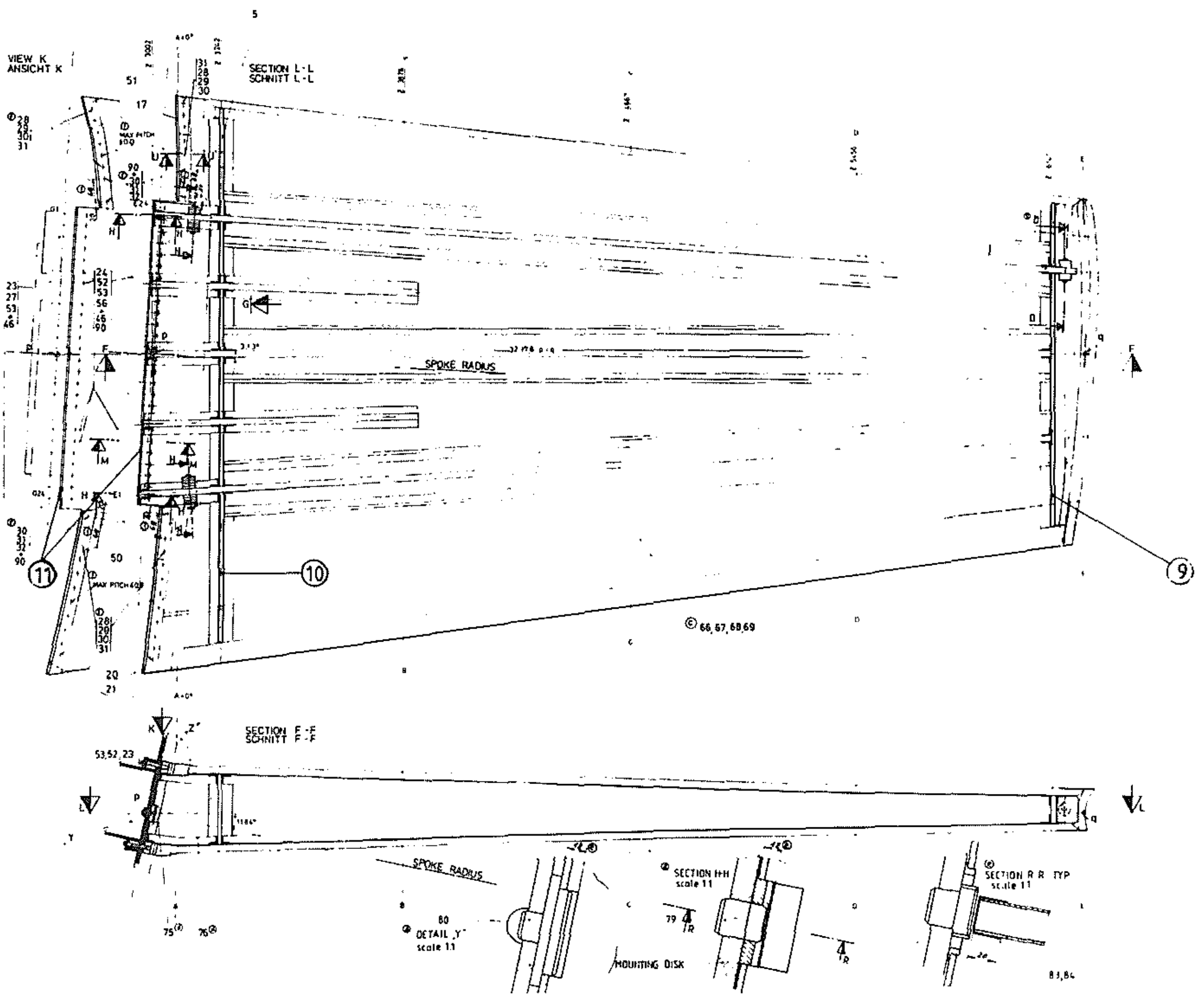

Figure 7: DNW-Rotor Blade

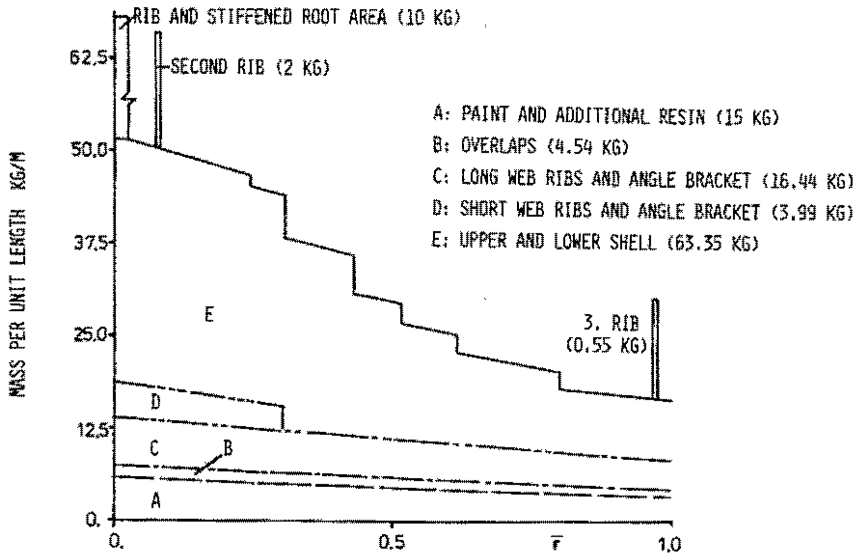

Figure 8: Fan Blade Mass Distribution

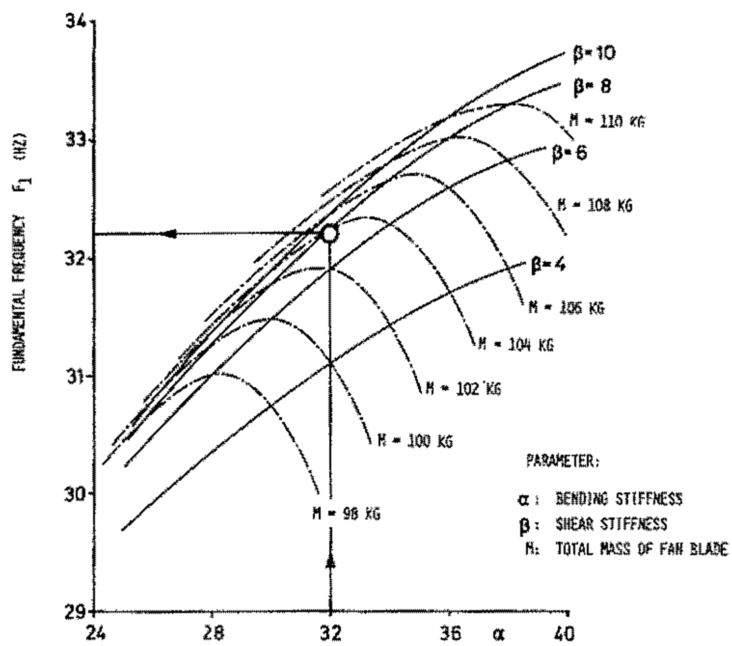

Figure 9: Selection of Structural Parameters

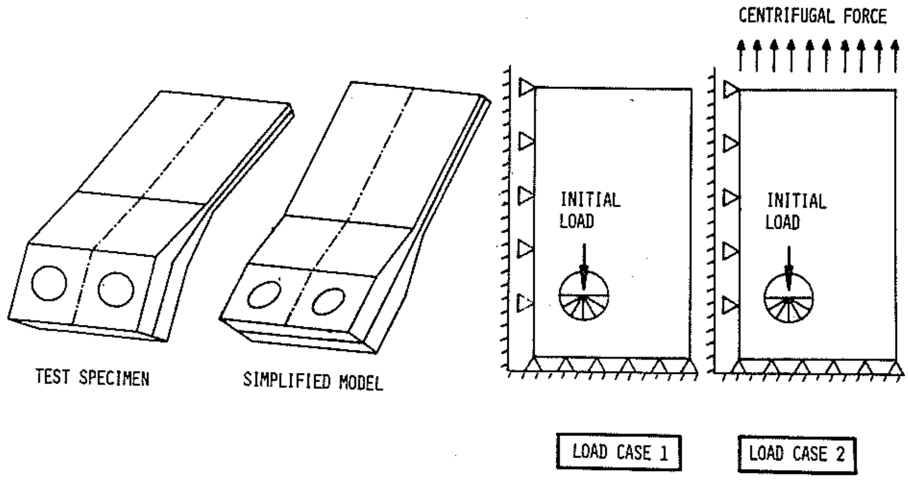

Figure 10: Bolt Area Analysis: System and Load Conditions

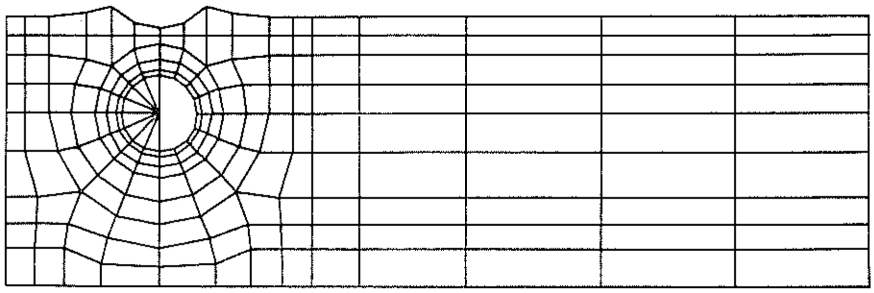

Figure 11: Bolt Area Idealisation for FEM Stress Calculation

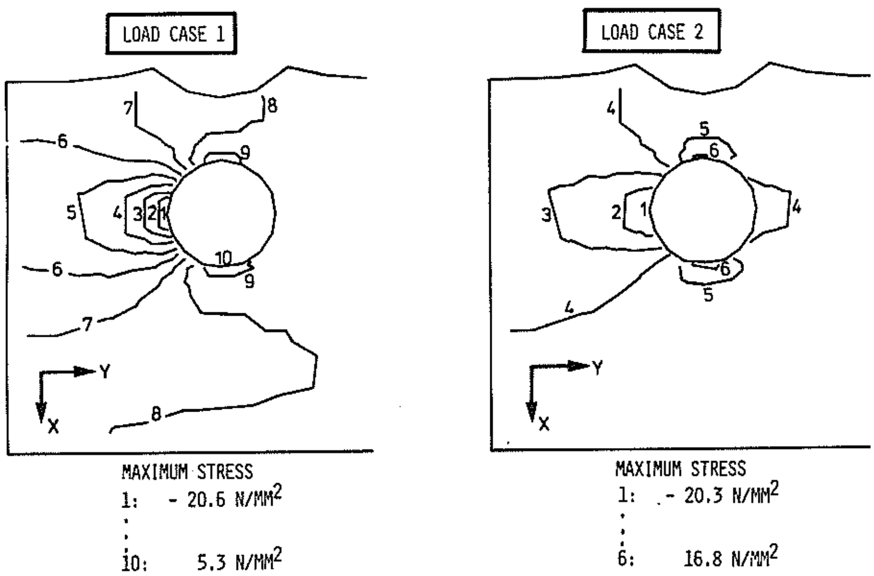

Figure 12: Stress Distribution in Bolted Area (y-Direction)

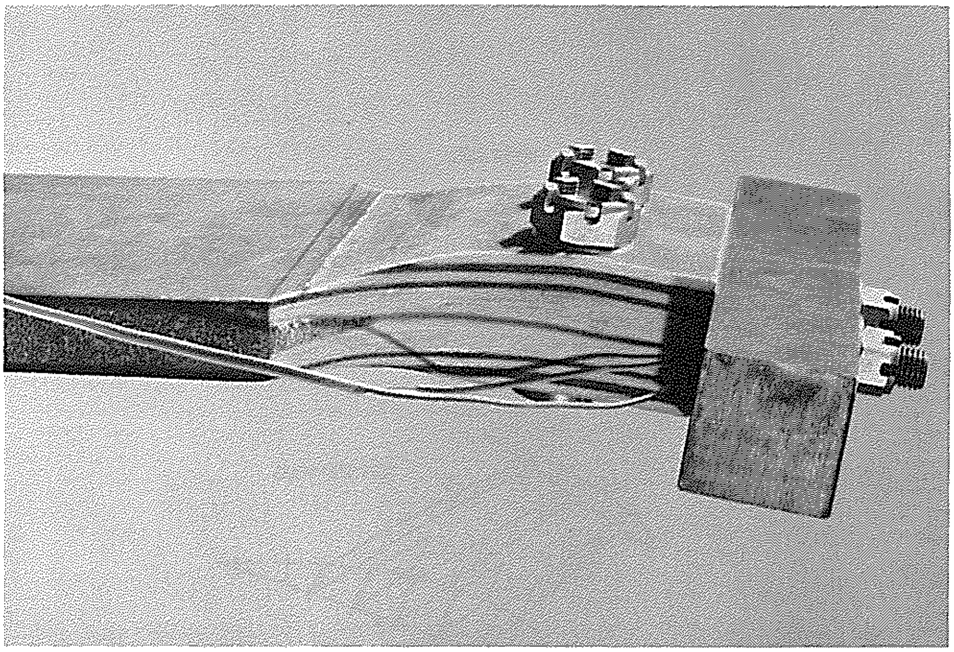

Figure 13: Specimen for Bearing Pressure Test

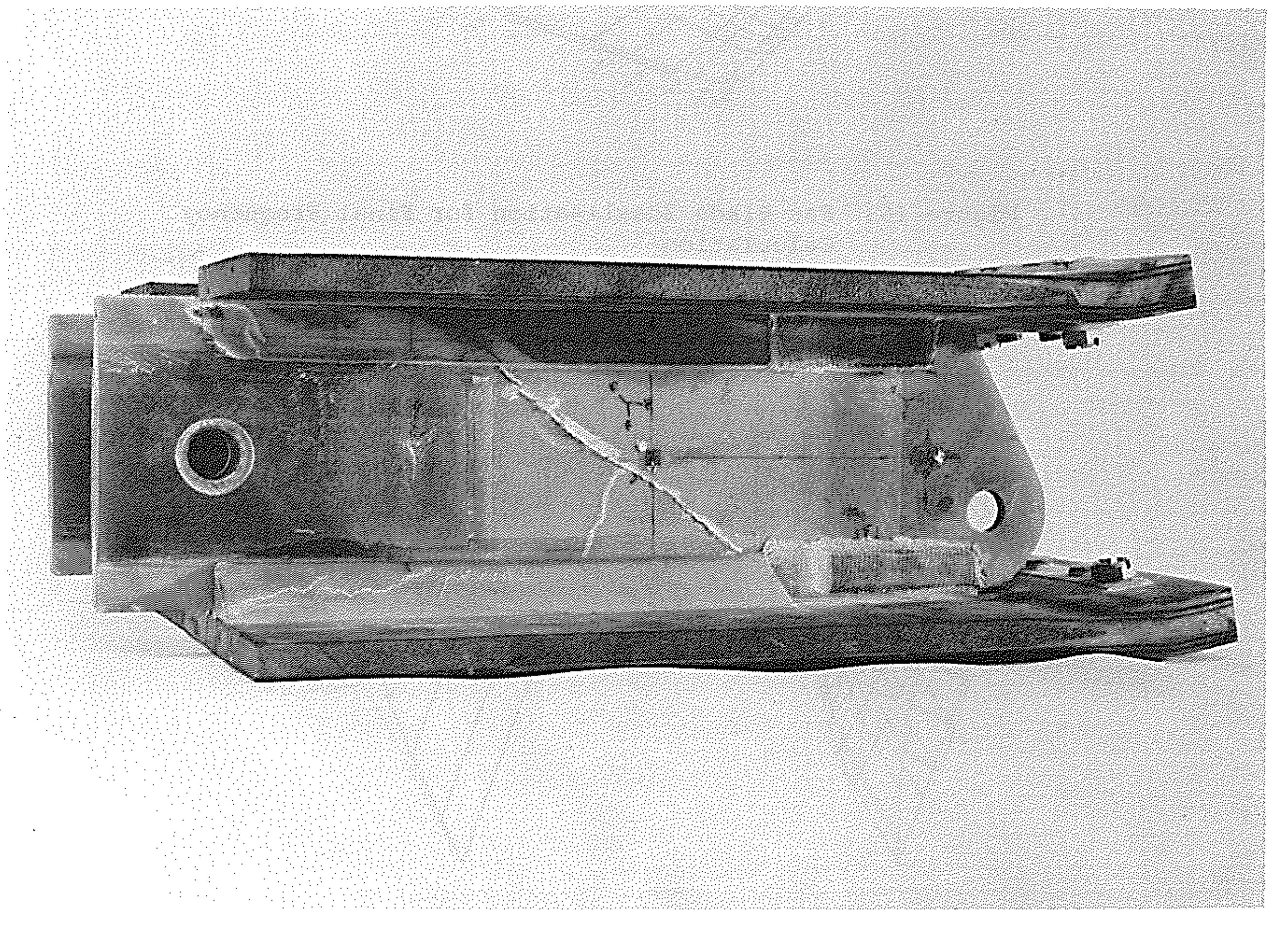

Figure 14: Failure Mode of Test Box

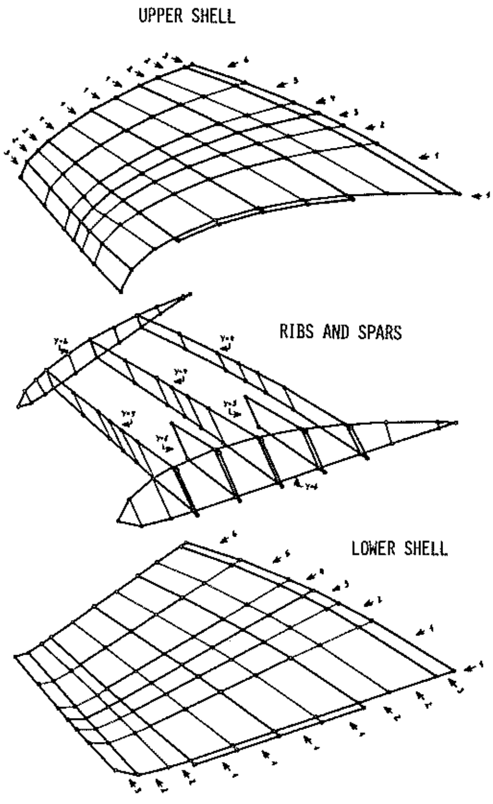

Figure 15: Fan Blade Idealisation for Final Frequency Calculation

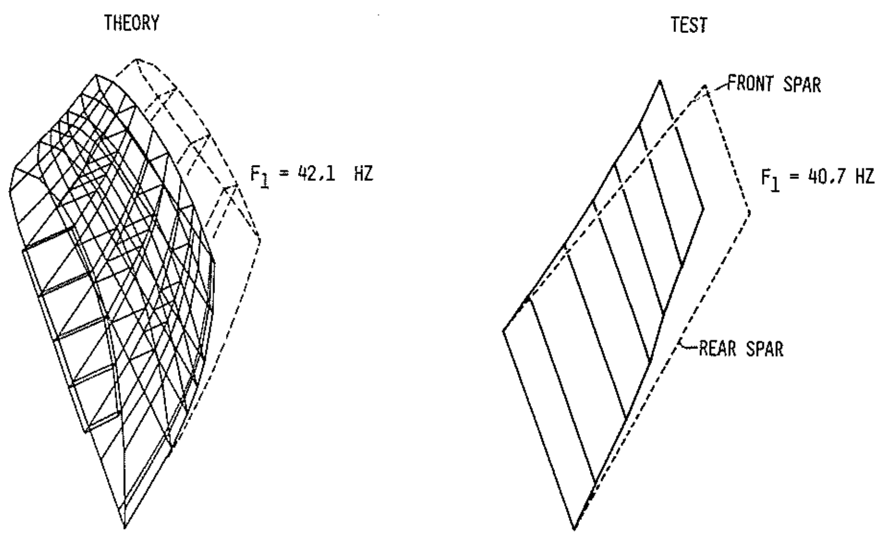

Figure 16: Fan Blade Mode Shape: Fundamental Bending

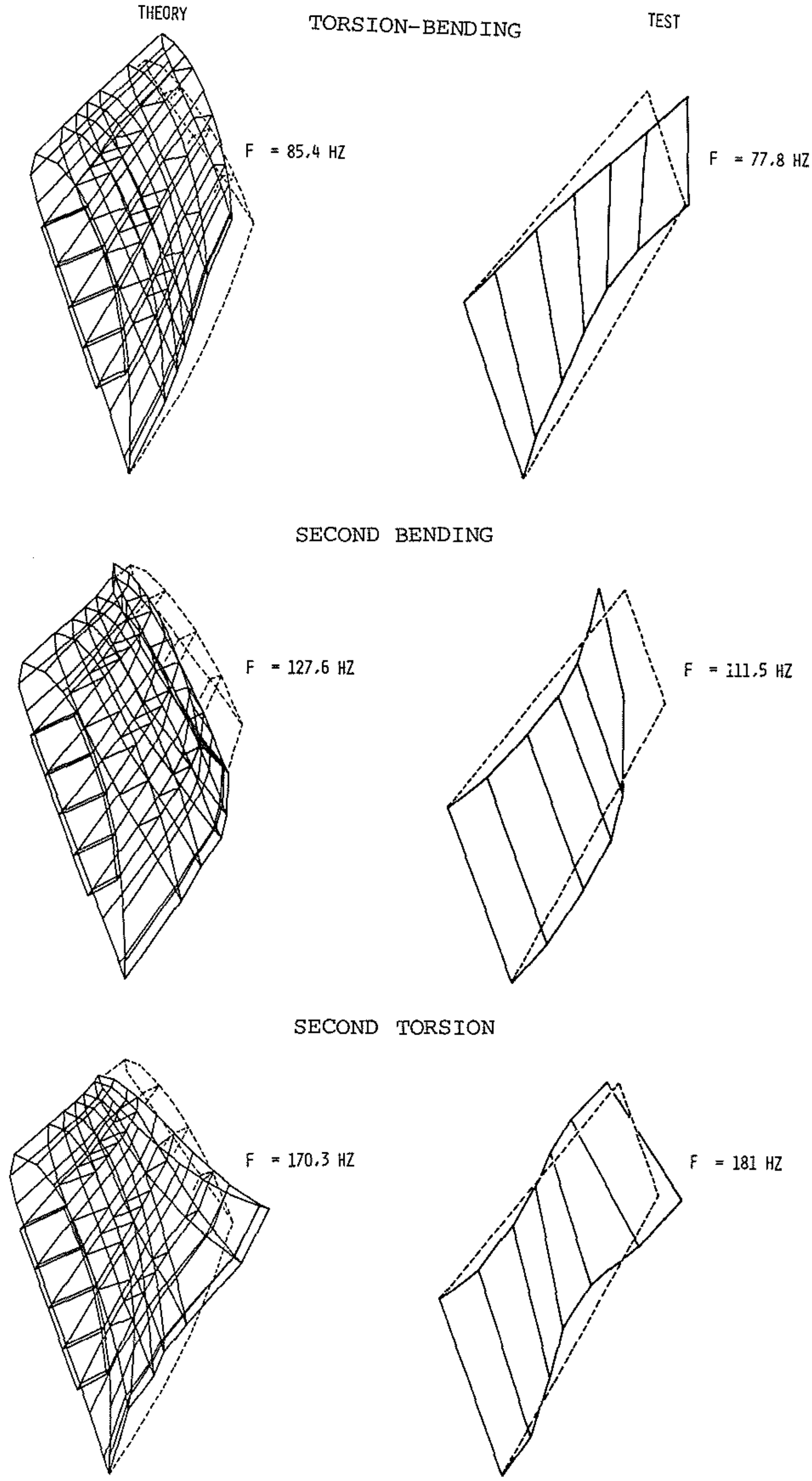

Figure 17: Fan Blade higher Mode Shapes and Frequencies

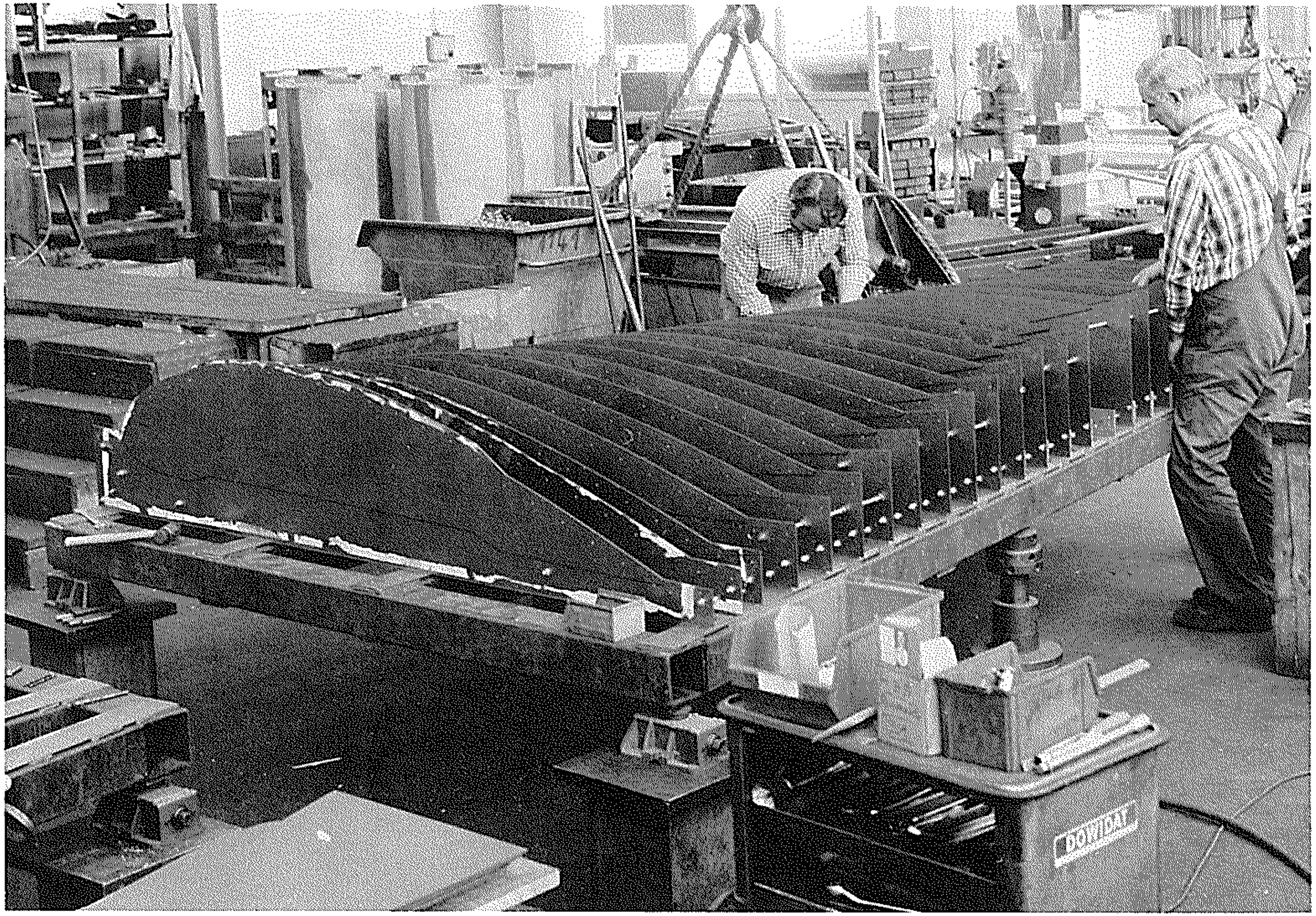

Figure 18: Male Half Model Jig

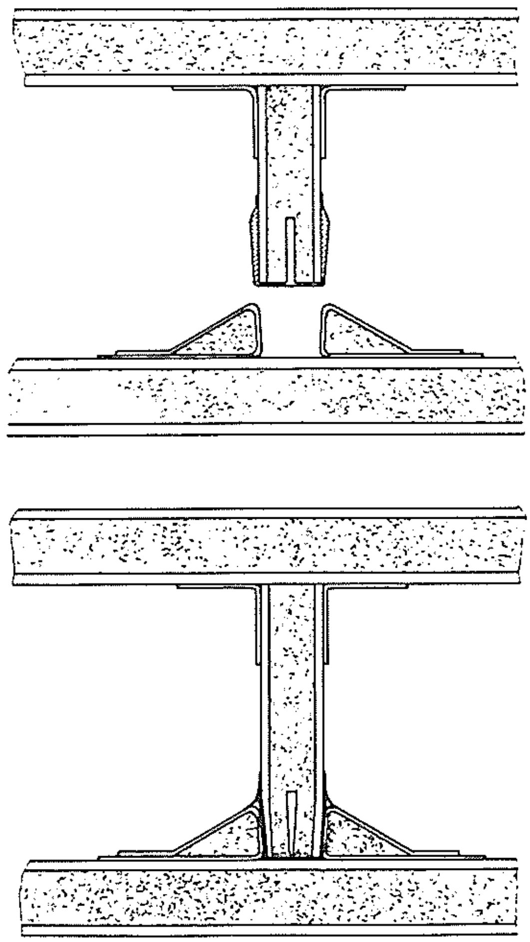

Figure 19: Principle of Joint between Spars and Lower Shell of the DNW-Blade

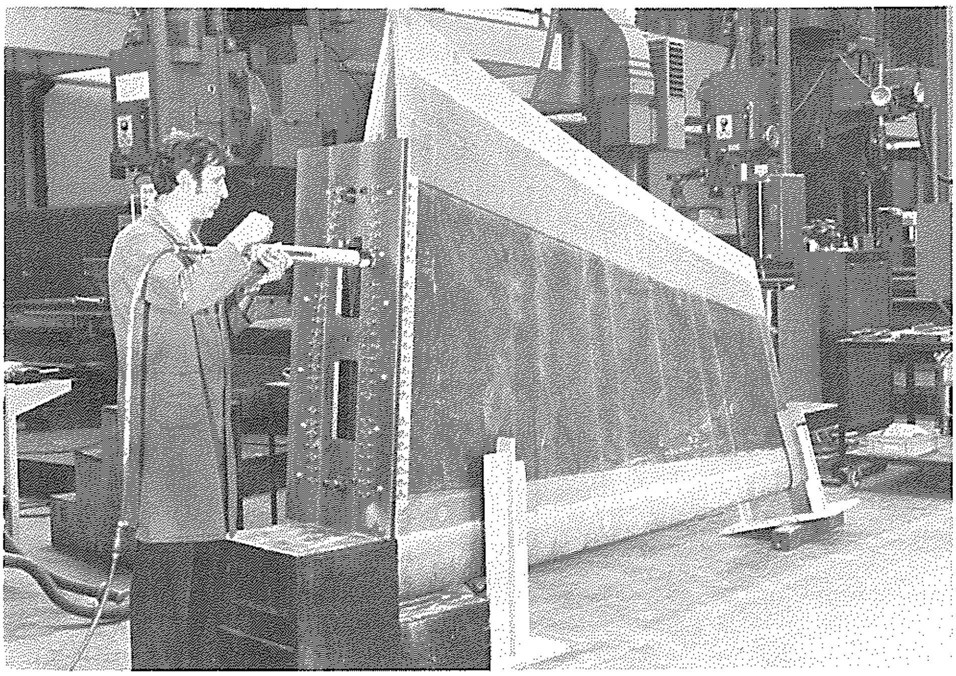

Figure 20: Drilling Process for DNW-Blade

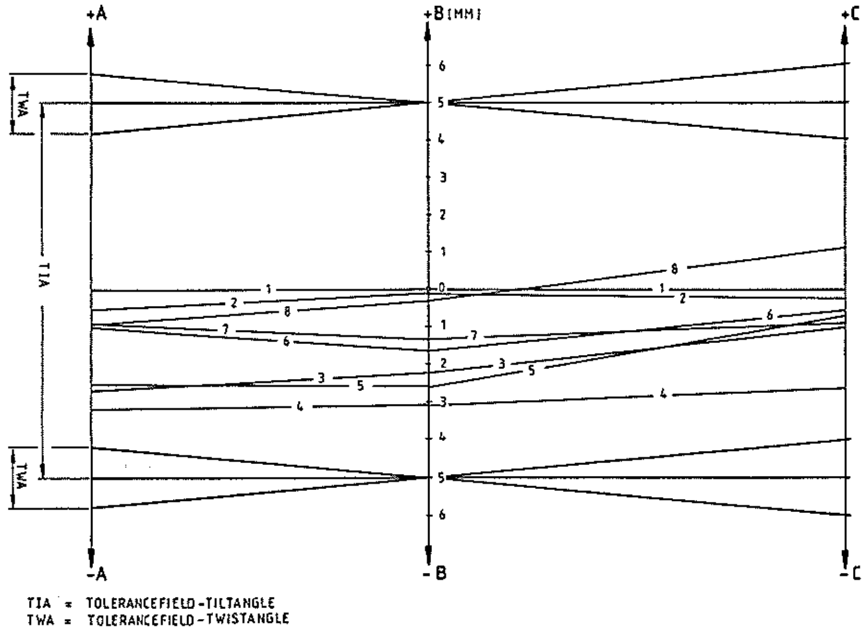

Figure 21: Accuracy of the Tip Profile Position of the DNW-Blades