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VIBRATION EXITATION OF TRANSONIC TURBINE BLADES IN A LINEAR CASCADE

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1. Introduction

Rotor blades of the last stages of low pressure turbines with profiles similar to flat plates are sensible to vibration exitation. This mainly happens at transonic and supersonic flow velocities. The self-exited vibration of an elastic structure is called flutter. Depending on the nature of the unsteady flow one distinguishes three types of flutter:

- Potentional flutter. It develops by cooperating of an unsteady fluid force and the movement of the vibrating blade. Depending on the phase angle between the blades vibrating with the same frequency damping or intensifying occurs. Boundary layer phenomena and flow separation play no role.

- Shock-flutter. It arises by the periodic oscillation of the trailing edge shock of a blade on the suction side of the adjacent blade, where it causes a periodic change of the blade force.

- Separation-flutter. A strong compression shock causes local or complete flow separation. The higher flow resistance forces the shock upstream. The pressure in front of the shock rises until the resistance of the separation can be overcome and the shock moves downstream again.(This procedure called buffeting also occurs at flat walls, e.g. in transonic diffusers.) The pressure fluctuation in the separated region is of stochastic character, the elastic blade reacts on the part near to its natural frequency and enlarges the pressure fluctuation by its own movement. The amplitudes are also stochastic.

In turbine cascades separation-flutter develops at Mach number about one, shock-flutter at higher Mach numbers. Potentional flutter can partly be estimated, the other types of flutter have to be investigated experimentally.

In our Cascade Steam Tunnel we performed investigations at rigidly fixed blades in superheated steam. During the tests we observed by means of Schlieren technique the trailing edge shock oscillating on the suction side of the adjacent blade, and recorded it with an High Speed camera. Two distinguished frequencies were found (about 350 Hz and 550 Hz), which rise with rising Mach number. In corresponding measurements in wet steam the frequencies were 300 Hz and 490 Hz. This fits quite well to the ratio of the velocities of sound of superheated and wet steam.

The Strouhal numbers computed with these frequencies are clearly lower than those typical for self-exited vibration of free-standing blades. That means, this phenomenon doesn`t affect the integrity of the blades. Nevertheless it is interesting and useful to know the reaction of blades free to vibrate on these shock movements.

2. Steam tunnel and linear cascade

About the Cascade Steam Tunnel of the flow laboratory of Siemens KWU in Mülheim was reported at a former symposion. In figure 1 the test rig can be seen in two sections. The rigid cascade firstly investigated consisted of 7 blades of a tip section of a low pressure last moving row. Figure 2 shows the position of the blades and of the dynamic pressure transducers for measuring the shock frequencies. The elastically mounted blades should have the same arrangement.

3. Similarity considerations

The similarity of geometrical dimensions is given by the scaled reduction of the turbine cascade. The aerodynamic similarity is fullfilled by equal Mach numbers. The similarity of elasticity is given by equal Strouhal numbers:

$$S = l_S \cdot f / c$$

(l_S : blade length, f : frequency, c : velocity)

or the enlarged Strouhal number

$$S_E = S \cdot (\rho_{steam} / \rho_{blade}) \cdot (l_S / t)^2.$$

(ρ : density, t : pitch)

Regarding the similarity of cascade flows, for blades vibrating in the same frequency it is necessary that the phase angle of adjacent blades is the same too.
In subcooled steam it is difficult to realise the thermodynamic similarity. Because of the scaled down profile the expansion velocity is higher and the subcooling is larger in the test rig than in the turbine. The influence of these differencies, however, is thought to be neglectible.

4. Dynamically mounted blades

Two methods of blade-suspension were roughly calculated and then judged by preliminary tests: firstly the suspension by strings, secondly by bending cantilevers.

4.1 Suspension by strings

A constant first mode natural frequency of the system blade/string needs constant tension force and constant string length. The principle of a suitable system is given in figure 3: The string is fixed at one side, at the other side it lies around a roller and a weight gives constant tension force. The first mode of the system is that of a standing wave with nodes at both ends. The influence of the blade mass on first mode and string stress has been investigated by the following model, figure 4: The blade mass was substituted by a point mass. With the given string parameters (length, diameter, density) for different values of weight and string force the first mode frequency of this "vibrating blade" was determined. The sound waves of the system were recorded by a microphone and transferred to an impulse sound level meter.

In order to gain the first mode frequency from the measured frequency spectrum this was shown on an oscilloscope and superposed by a frequency from a frequency generator. During the vibration process the amplitudes of the higher modes decrease rapidly and after a short time only the more powerful first mode is to be seen. The frequency from the generator can be variied until it is equal to the measured frequency of the first mode and so its value is found.

The measured sound waves are also given to an FFT - analyser, that delivers the natural frequencies of the spectrum. Because of the disturbancies by other noise sources this method is less certain and it is only a suppplement of the first one.

We investigated the suspension system with two different strings with a diameter of 1 and 2 mm and a length of 570 and 510 mm, respectivly. From figure 5 one can see, that with constant tension force an only small point mass results in a large decrease of the natural frequency. Extrapolating the results for the given blade mass of 250 g up to the desired 210 Hz one gets a tension force, which cannot be realised in the rig. Another disadvantage is that in the steam tunnel the acoustic measurement method is not applicable. The advantage of this suspension method is the low disturbance of the inlet flow to the cascade.

Considering the advantages and disadvantages we came to the conclusion that this method shouldn`t be realised, even not for a much lighter blade of 40 g (blade made out of resin).

4.2 Suspension by bending cantilevers

Figure 6 shows a simplified system of this principle. Its natural frequency depends to a large extend on the length of the bar-type cantilevers. With blade weight and material values given, one can change the natural frequency only by changing the areal moment of inertia, that means the profile of the cantilever. The natural frequencies of several types of cantilevers have been measured by measuring the acceleration of the vibrating system and evaluating them with an FFT - analyser. Length and cross section of the cantilever and blade mass have been varied.

The resulting frequencies are lower than those calculated in advance, figure 7. But one can see that it is possible to realise the wanted natural frequency by this way. For the given conditions of the steam tunnel - maximum and minimum length of the cantilever, space for the suspending devices - we tried to optimise the cantilever, that is here: to raise the bending stiffness. This can be done by changing the material, the profile of the cross section and a cross section profile varying with the length of the cantilever. Suitable is a material with a high modulus of elasticity and low density,e.g. titanium. Taking the cost of manufacturing into account we choose stainless steel.

The cross section should have a small area but a high areal moment of inertia. Also favourable is a cantilever becoming smaller over its length towards the blade. For both measures one has to consider manufacturing problems and flow disturbances by the cantilever. Furthermore the strength is important, especially of the connection between cantilever and blade. This connection should be designed in such a way that Schlieren technique can be applied and mainly the suction sides of the blades are free from suspension arrangements. As a compromise between wanted and possible kind of cantilever each blade is supported by two cantilevers which have rectangular cross section with variable area.

Parallel to the optimisation of the cantilever described, it was investigated whether and by what means the natural frequency of the vibrating system can be raised by reducing the blade weight, that means to employ another material than stainless steel. This material has to meet the conditions of the steam tunnel regarding temperature, wetness, and steady as well as unsteady forces. The manufacture of a blade made out of epoxy resin was investigated in more detail. Different methods of casting including reinforcement by glass fiber were tested.

The tests showed that with a proper combination of glass fiber material, structure of texture, kind of resin, and manufacturing procedure it was possible to produce blades with high strength, high stiffness, and low mass, which withstand temperature about 120 °C. Unfortunately, the procedure is expensive, and for manufacturing several identical blades expensive casting models are needed. For these reasons we decided to take stainless steel as blade material. Therefore the first mode natural frequency realised is lower than according to the model laws wanted, nevertheless the main features of the problem can be studied.

The final solution for blade suspension with two bending cantilevers is shown in figure 8, simplified for computation. The natural frequency is 170 Hz calculated with the Finite Element code ADINA. The blades with the cantilevers are assembed in a so called cascade insert, figure 9. After the assembly the natural frequencies were measured and values of 1 to 9 Hz lower than the calculated ones have been found. Reasons for this can be small differencies in connections or geometry. Also feedback via the cascade insert is possible.

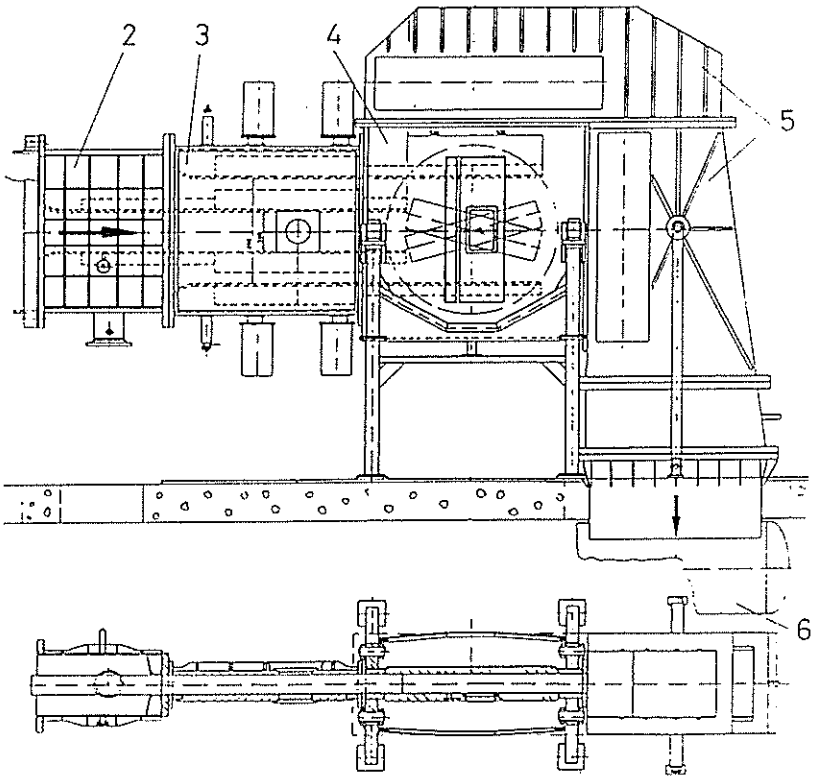
In order to measure the blade movement, strain gauges were applicated at the lower side of all left side cantilevers. To get the deflection of the measuring point from the measured elongation values, in a previous test the blade was exited electromagnetically. Simultaneously, the elongation by strain gauges and the deflection by laser-vibrometer were measured, so that elongation and deflection can be correlated. The maximum deflection of the blade was determind in another pretest. In this test the deflection line of the system was measured by the holografic method ESPI (Electronic Speckle Pattern Interferometry).

5. Example of results

Figure 10 shows amplitudes gained from tests in wet steam, above for the pressure oscillations of the wall pressure transducer no. 4, below for the elongations of the strain gauge of blade 3. The exit Mach number Ma2 was raised from about 0.8 (test number 1) to about 1.2 (test number 14). Up to test number 7 the amplitudes are very low. At test number 8, Ma2 ≈ 1.0 is reached and the amplidudes grow rapidly. At test number 10 (Ma2 ≈ 1.1) they have maximum values and go back to unimportant values at test number 14. The appropriate frequency of both amplitudes is about 159 Hz, that is somewhat lower than the natural frequency of the blade (163 Hz). In the superheated steam test for the same pressure ratios the amplitudes and the elongations are more than twice of that in wet steam.

6. Final remarks

First tests have shown that investigations on vibration phenomena in a linear cascade are reasonable. For further investigations some improvements are foreseen, e.g. pressure transducers in the blades flush with the surface, separately tuned blades, and adjustment of the phase angle.



2 Settling chamber
3 Inlet casing
4 Cascade casing with cascade support
5 Exhaust casing
6 Steam chamber

Fig.1: Cascade steam tunnel

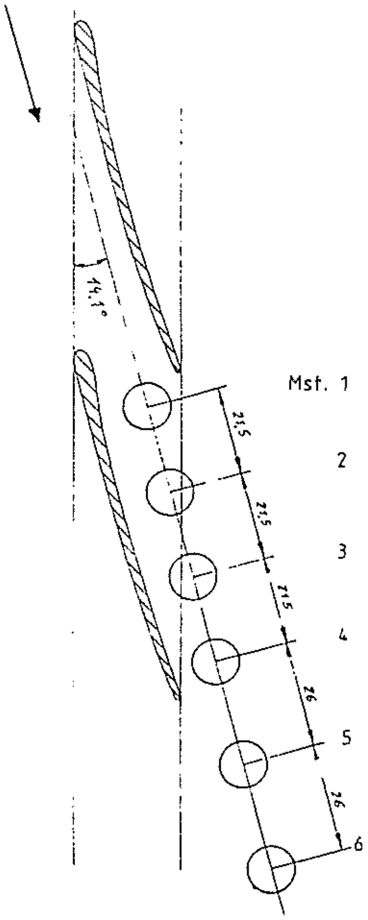
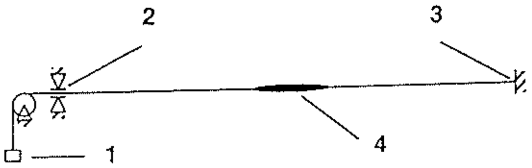
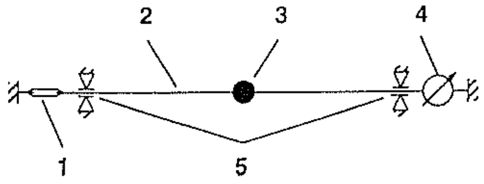


Fig.2: Blade configuration and position of dynamic pressure transducers flush with side wall



1 Weight 2 String guide 3 Fixture 4 Blade

Fig.3: Elastic blade suspension with strings



1 Tensioning device 2 String 3 Point mass
4 Force meter 5 String guide

Fig.4: Arrangement for evaluating the string tension

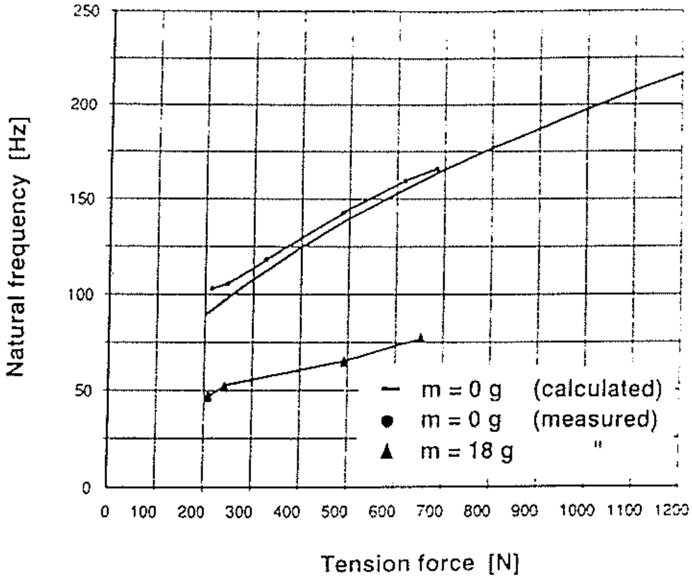


Fig.5: Natural frequency dependent on tension force (string: Φ = 2 mm, length = 510 mm)

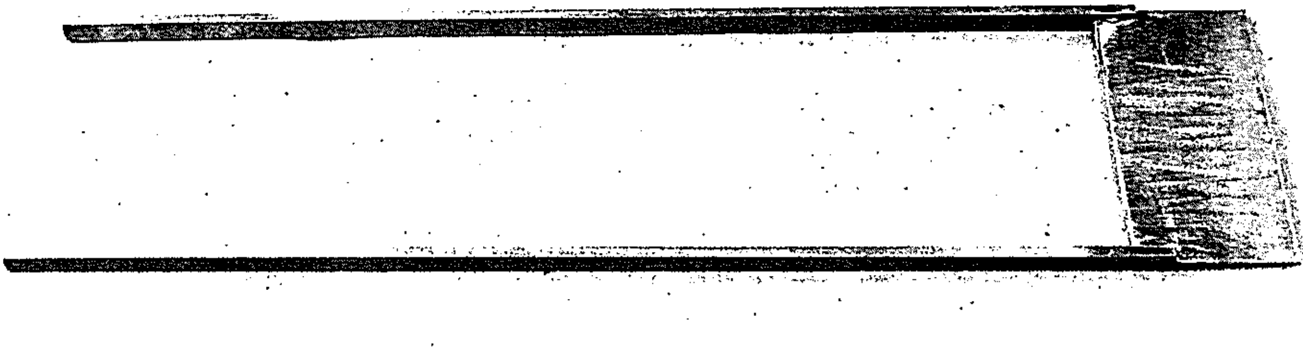


Fig.6: Simple blade/cantilever model

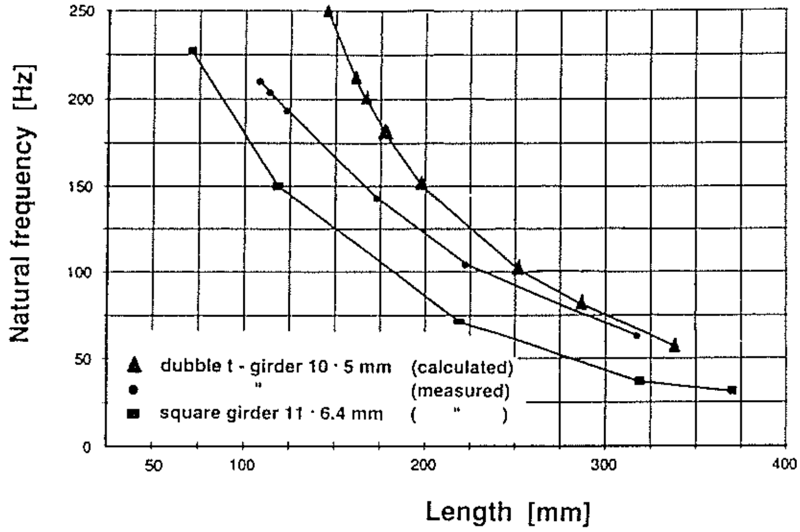


Fig.7: Natural frequency dependent on length and profile of the cantilever

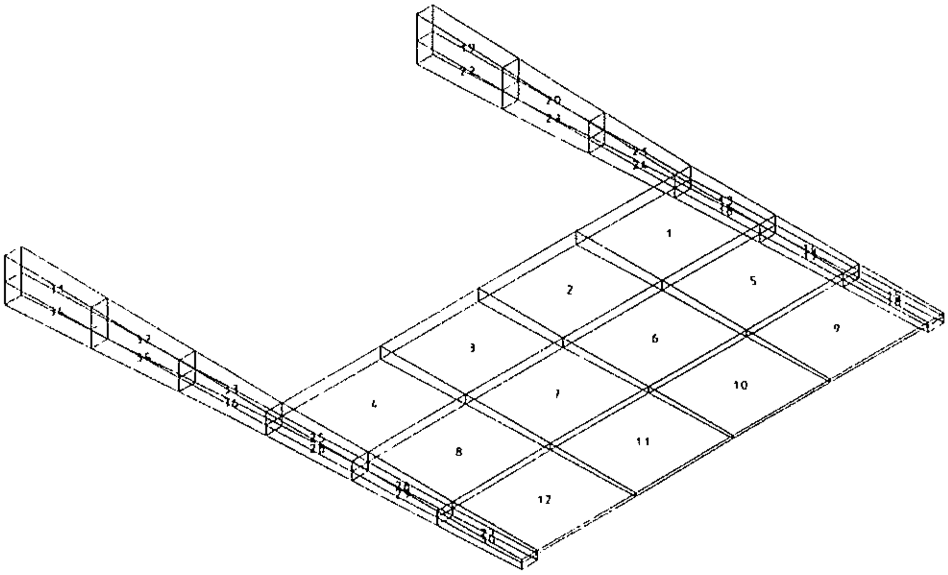


Fig.8: Elastic blade suspension with bending cantilevers
(simplified for Finite Element computation)

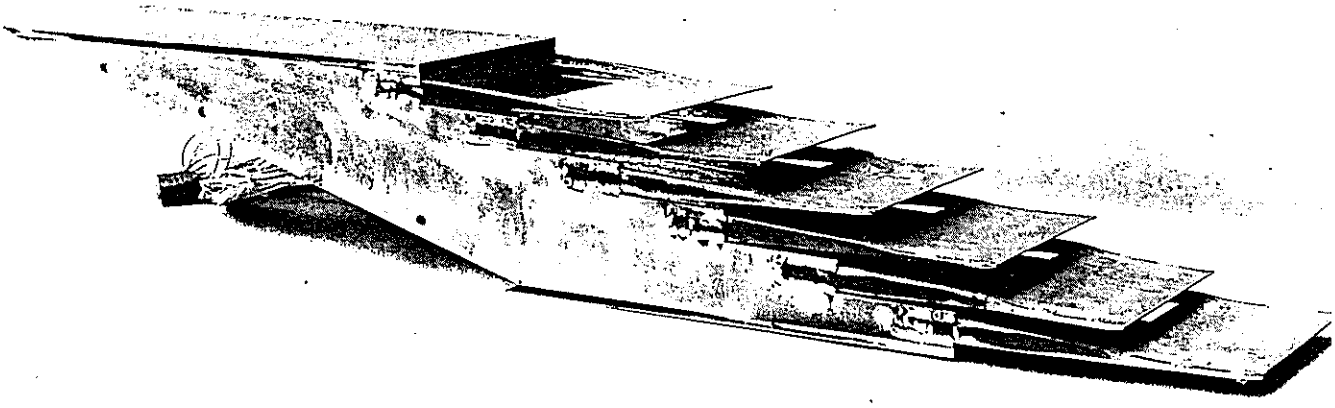


Fig.9: Cascade insert with strain gauges

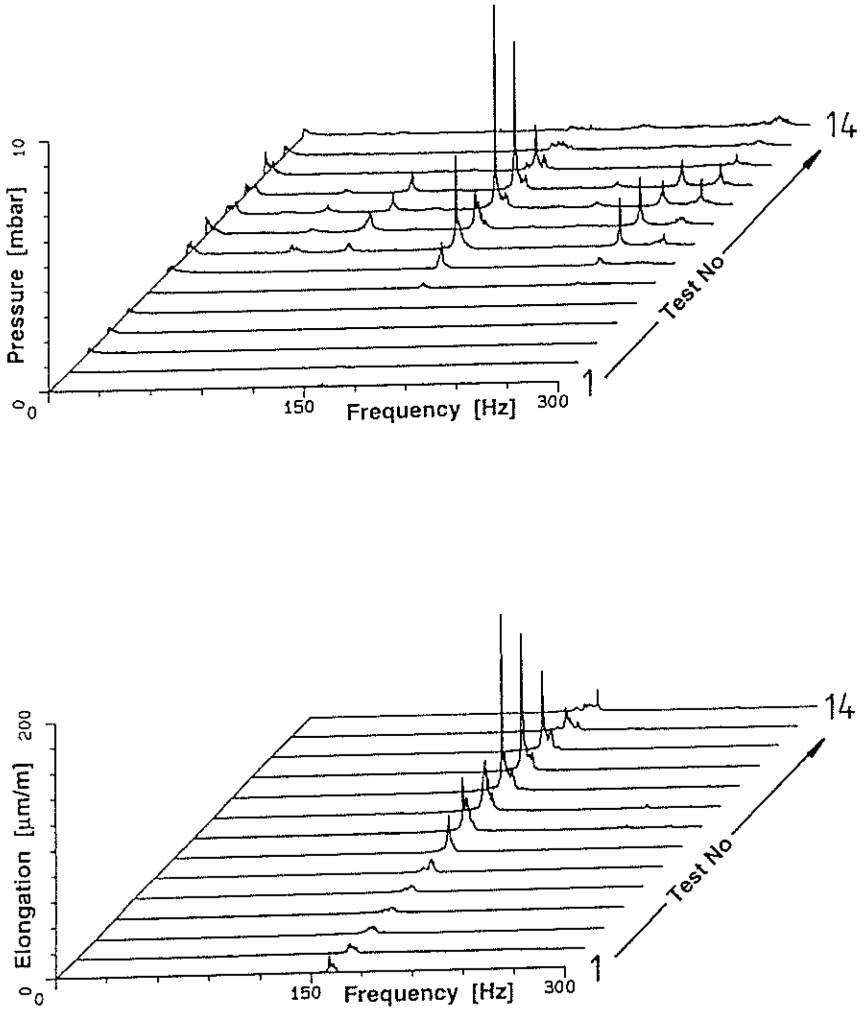


Fig.10: Wet steam test: Amplitudes of wall pressure and blade vibration