HPT-11/01

INFORMATION FOR 3D COMPUTATIONS OF THE STCF 11 TEST CASES

Markus Jöcker

STOCKHOLM
3-May-01

Technical report

Avdelningen för Kraft- och Värmeteknologi
Kungliga Tekniska Högskolan
100 44 STOCKHOLM

KUNGL TEKNISKA HÖGSKOLAN

Department of Energy Technology
Chair of Heat and Power Technology
Professor: Torsten H. Fransson

Title Information for 3D Computations of the STCF 11 test cases		
Author Markus Jöcker	**Report No** HPT-11/01	
Project STCF11	**Pages** 7	**Drawings** 4
Supervisor KTH:	**Date** 01-05-03	**Appendices**
Overall responsible at KTH: T. H. Fransson		
Approved at KTH by:	**Signature:**	
Overall responsible at industry:		
Industrial partners:		
Approved by industrial partners:	**Signature:**	
Approved for distribution:	**Restricted, to distribution list**	
	Open:	**Open for web site**

Abstract

The existing set of 10 configurations for unsteady flow through vibrating axial-flow turbomachine cascades was extended by the "Standard Configuration 11". This configuration represents a turbine blade geometry with transonic design flow conditions characterized by a normal shock impinging on 75 % real chord on the suction side. Out of a set of test cases covering all relevant flow regimes two cases were selected for publication: A subsonic, attached flow case and an off-design transonic case showing a separation bubble at about 30% real chord on the suction side. The performed tests are shown to be repeatable and suitable for code validations of numerical models predicting flutter in viscous flows. All tests were performed at the annular non-rotating test facility situated at the EPF Lausanne. The published data and geometry are given at midspan, nearly no measurements were made at other blade heights. However, the 3D shape of the blade is prismatic (which means a variable pitch-to-chord ratio over the channel height). The 3D geometry definition can be obtained by stacking the midspan plane profile in radial direction; the tip gap is 0.8 mm according to the model drawings. The present document provides detailed information on the 3D geometry and the 3D blade motion as reconstructed from the available drawings and reports.

The 2D data were published with the permission of ABB, which is gratefully acknowledged. Standard Configuration 11 was presented to the public with a publication presented at the ASME TURBO EXPO 1998, Stockholm, Sweden (paper 98-GT-490) and was published in the Journal of Turbomachinery (Volume 121, Number 1, October 1999, pp 717-725).

Distribution list					
Open for web site distribution					

Introduction

The experiments performed during 1991 had the aim to deliver 2D test results at mid span of a low-pressure turbine blade to investigate shock flutter. Unfortunately, it was regarded as sufficient at that time to only record blade displacements at mid span, not expecting the need of 3D data for future calculations. The present document gives an estimation of the 3D blade geometry and blade motion based on documentation of the design of the blade and the excitation mechanism. However, there can be some error in the proposed blade motion kinematics, because the exact design of the blade suspension, especially the specifics of the spring, is not available anymore. Furthermore, we want to make the reader aware of the fact that the blade experienced also a static displacement due to the aerodynamic load during the experiments. This displacement is not estimated here and was also neglected in previous computations. But an influence of this displacement on the blade surface pressures is possible due to the small change in relative flow conditions.

3D blade geometry of STCF 11

The 3D blade geometry can be obtained by stacking the given 2D profile on a radial line along the centers of gravity of the 2D blade sections. Fig. 2 gives an illustration. Center of gravity (see Fig. 1): (x_g,y_g) = (30.61 mm, 8.58 mm) in a co-ordinate system conform to the provided data file “STCF11.geo.dat”. The tip gap is 0.8 mm according to the model drawings.

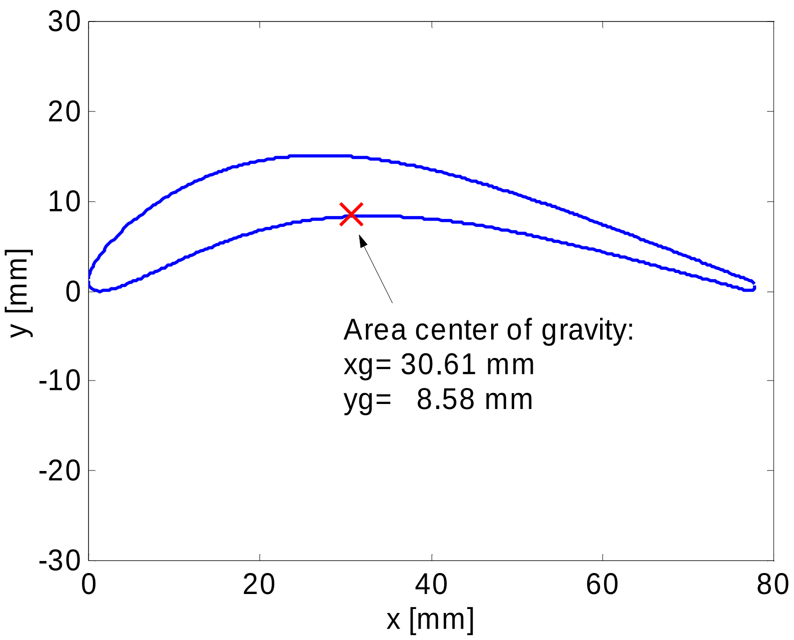


Fig. 1*: STCF 11, Area Center of Gravity*

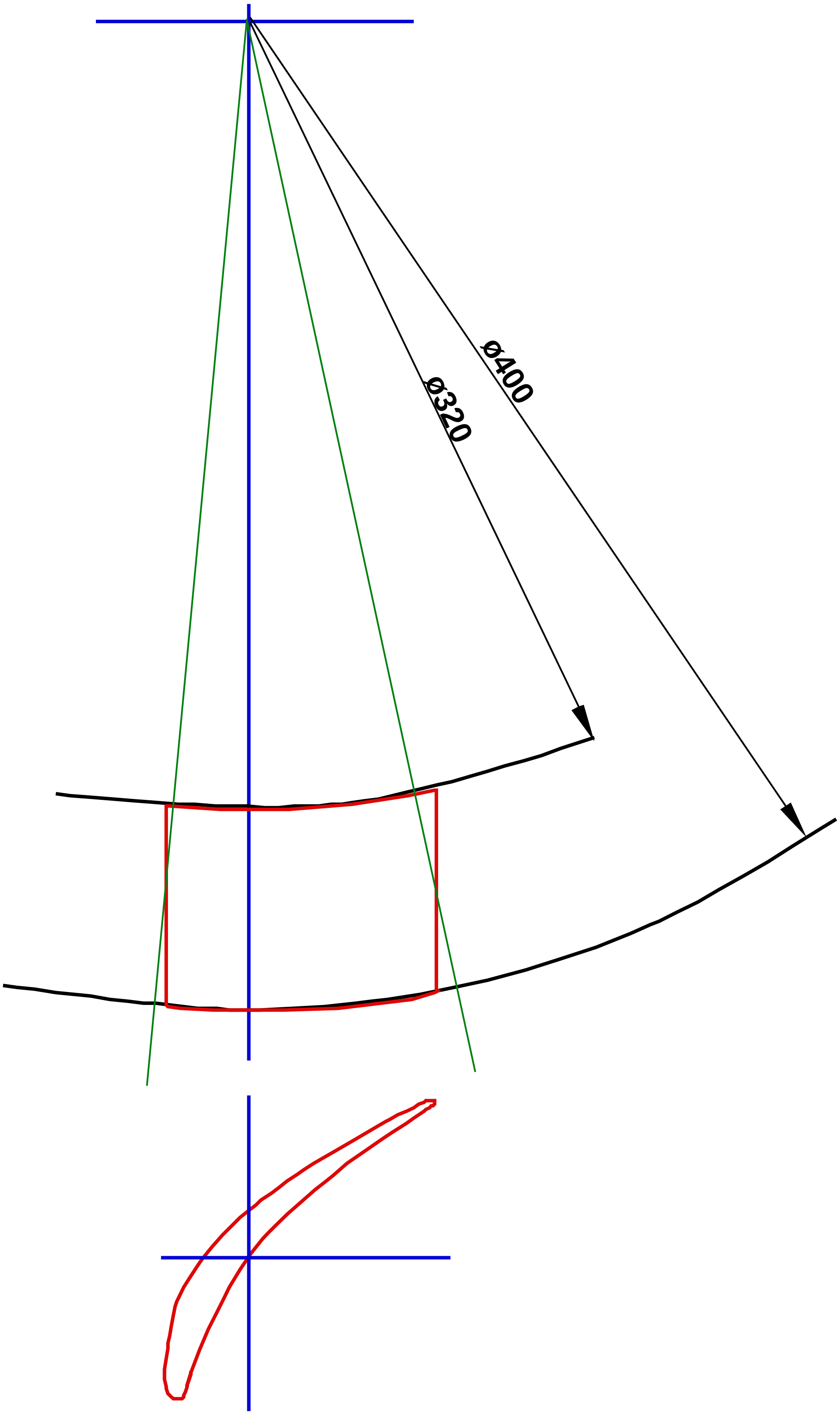


Fig. 2: *Illustration of 3D blade stacking, STCF 11 [Ott, 2001]*

Estimation of 3D blade motion of the STCF 11 test case

Fig. 3 shows a drawing detail of the blade suspension indicating the measures on which the present estimation is based on. Fig. 4 shows the model used to estimate the displacements. Neither the exact load (F_0, M_0 and blade load) nor the exact design of the spring (length l_0, height h and thickness b) is known. Only the 1st harmonic vibration amplitude at mid span of the blade (δ_{mid}) is known.

The following equations describe the kinematics of the model in Fig. 4, which assumes that only the spring deforms, the rest undergoes a rigid body motion:

Displacement of the spring end due to the force F_0:

$$\delta_F = \frac{F_0}{c_F} \qquad \text{(Eq. 1)}$$

Displacement of the spring end due to the moment M_0:

$$\delta_M = \frac{M_0}{c_M} \qquad \text{(Eq. 2)}$$

The factors c_F and c_M are the bending and torsion stiffness of the spring.

Displacement of the blade relative to the spring end:

$$\delta_B = \tan\alpha \cdot L \approx \alpha \cdot L \qquad \text{(Eq. 3)}$$

The spring angle at the end of the spring results from the torsion and the bending part to

$$\alpha = \alpha_F + \alpha_M \qquad \text{(Eq. 4)}$$

$$\text{with} \quad \alpha_F = \frac{3 \cdot \delta_F}{2 \cdot l_0}, \quad \alpha_M = \frac{2 \cdot \delta_M}{l_0} \qquad \text{(Eq. 5)}$$

resulting from the application of the kinematic of the simple bending beam

The complete displacement is

$$\delta = \delta_F + \delta_M + \delta_B \qquad \text{(Eq. 6)}$$

which would result from a static load on the model. With a dynamic load a similar modeshape but different amplitudes can be expected. Based on this model three approaches to estimate the variation of blade vibration amplitude vs. span are compared below.

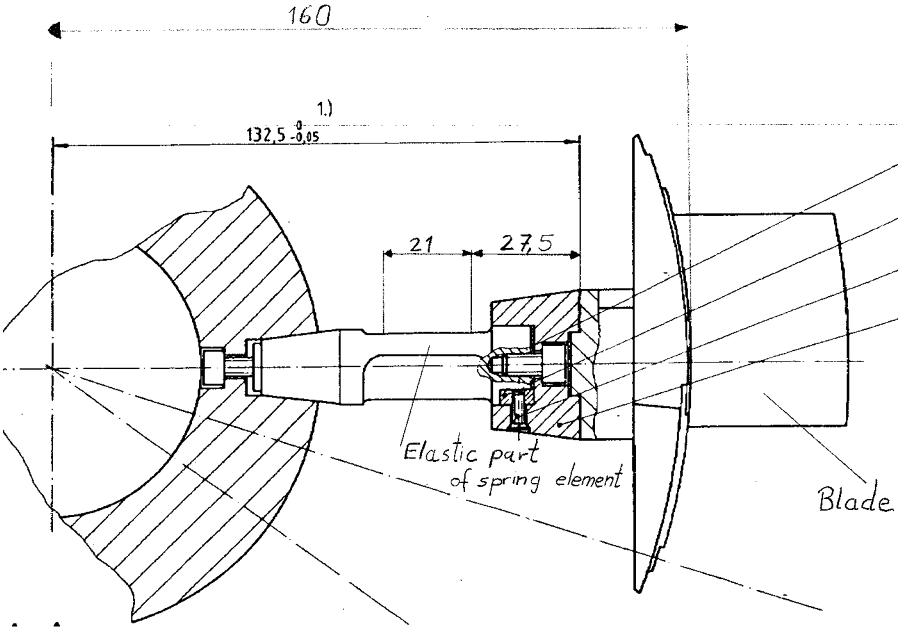
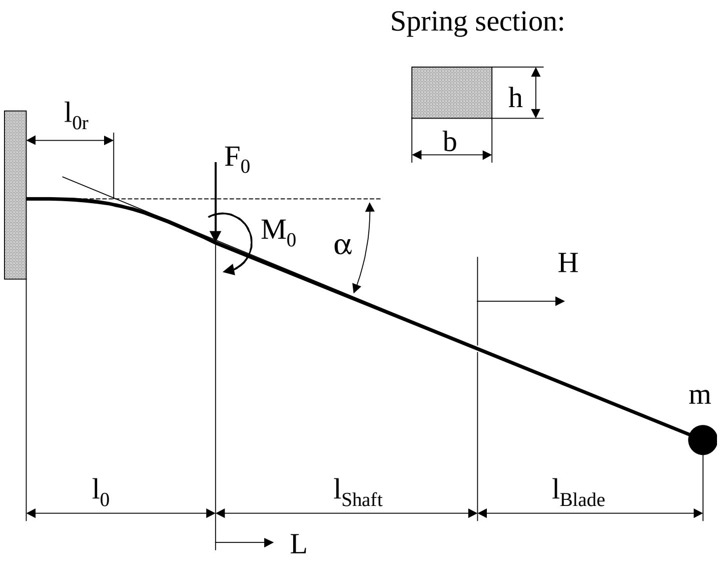


Fig. 3: *Blade suspension design (LTT/EPFL, 1985)*



l_0	21 mm
l_{Shaft}	55 mm
l_{Blade}	40 mm

Fig. 4: *Model of blade and suspension to estimate the blade motion*

1.Bending model:

Blade displacement amplitude due to pure (static) force F_0, the moment M_0 induced by the masses and a static displacement is neglected.

$$\delta = \frac{F_0}{c_F} + \alpha_F \cdot L = \frac{F_0}{c_F} \cdot (1 + \frac{3 \cdot L}{2 \cdot l_0}) \qquad \text{(Eq. 7)}$$

From the known displacement amplitude at midspan the factor F_0/c_F can be estimated. This gives a linear relation between the displacement δ and the distance L from the spring end.

2. Rotation model:

Blade vibration amplitude modeled by a pure rotation about a point with distance l_{0r} from the clamped support.

An equivalent linear relation to the “bending model” can be formulated by assuming a fictive rotation axis somewhere on the un-deformed spring. In [Ott, 86] it is cited that this fictive rotation axis has to be located in the interval

$$l_0/3 < l_{0r} < l_0/2 \qquad \text{(Eq. 8)}$$

A center of rotation located at l_{0r}=10 mm seems to give a good approximation (see Fig. 4 for nomenclature) leading to the following linear relation between blade height position H and amplitude (h/c):

$$(h/c) = \alpha \cdot (L + l_0 - l_{0r}) = \frac{(h/c)_{mid}}{H_{mid} + l_{Shaft} + l_0 - l_{0r}} \cdot (H + l_{Shaft} + l_0 - l_{0r}) \qquad \text{(Eq. 9)}$$

3. Force model:

Blade vibration amplitude modeled by assumptions of the mechanical and geometrical data of the system and an application of a static force resulting in a reasonable displacement at midspan.

The following assumptions were made based on design data found in [Ott, 1986], (see Fig. 4 for nomenclature):

b	19 mm
h	5 mm
l_0	21 mm
E	219000 N/mm^2
m	500 g

***Table 1**: Assumptions on mass-spring system characteristics*

This model regards a moment induced by the weight of the mass and the blade, where the worst case weight distribution has been chosen by placing a point weight at the tip of the blade. Also the mass is estimated with some plus to 500g. Hence, the influence of the moment displacement will be smaller in the real case. A force F_0 is chosen to give the known displacement at mid span.

All three approaches are compared in Fig. 5.

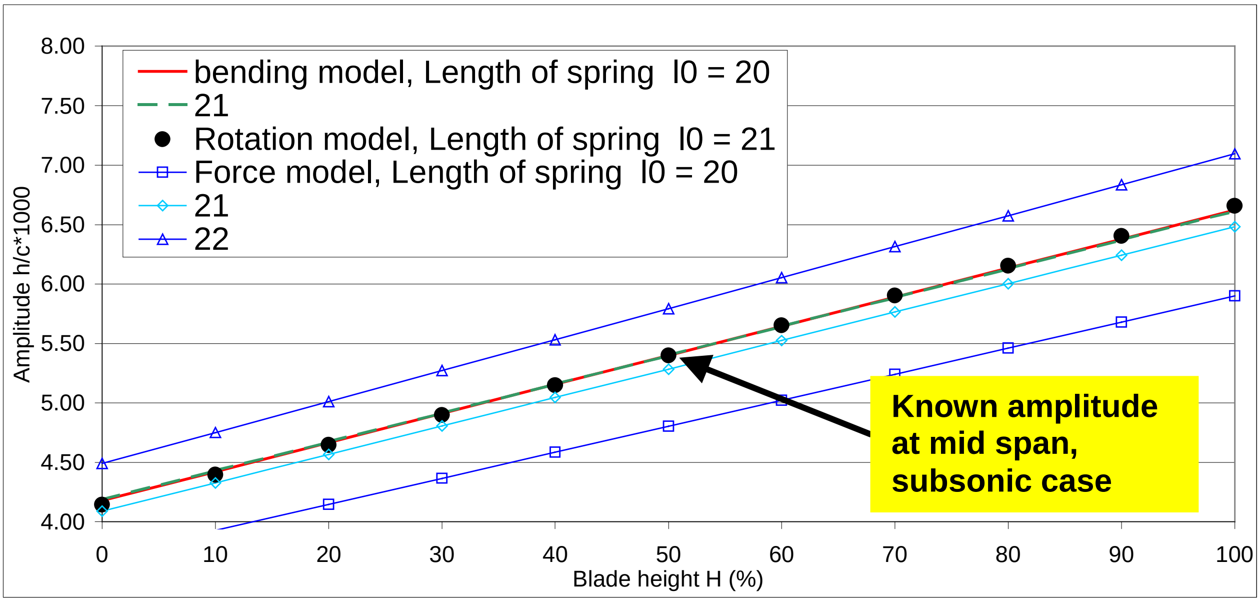


Fig. 5: *Comparison of various models to estimate the vibration amplitude vs. blade height, subsonic case*

The estimations have been made with various spring lengths, which have a significant influence on the amplitude level obtained with the "force model". However, all models show the same variation of amplitude with blade height. It is seen that the "bending model" and the "rotation model" with l_{0r}=10mm, are equivalent. The comparison to the "force model" shows that the induced moment has no significant influence on the variation of amplitude vs. span. Only the level of displacement is varied with length of the spring and magnitude of the force (not shown here).

Conclusion

The simple "rotation model" with a rotational axis given by l_{or}=10mm seems to give a good estimation of the radial distribution of blade amplitude. It is proposed to use that for modeling the 3D blade vibration. A radial distribution of blade vibration amplitude obtained with this "rotation model" is given below for the published cases 101-119 and 201-209. This is also included in the data files. The authors are open for comments and suggestions and improvements concerning this estimation.

Cases 101-119 (subsonic):

Blade Height H (%)	0	10	20	30	40	50	60	70	80	90	100
h/c*1000	4.14	4.40	4.65	4.90	5.15	5.40	5.65	5.90	6.15	6.40	6.66

Calculated with (Eq. 9) and a displacement at midspan of $(h/c)_{mid}$=0.0054 mm
h/c is the bending amplitude normalised with chord length c.

Cases 201-209 (transonic off-design):

Blade Height H (%)	0	10	20	30	40	50	60	70	80	90	100
h/c*1000	2.69	2.85	3.01	3.17	3.34	3.50	3.66	3.83	3.99	4.15	4.31

Calculated with (Eq. 9) and a displacement at midspan of $(h/c)_{mid}$=0.0035 mm
h/c is the bending amplitude normalised with chord length c.

References

P. Ott, D. Schlaefli, 1986
Ueberarbeitung des CERS-Gitters, LTT, EPF Lausanne, Bericht LTT-86-02, non-public report

LTT/EPFL 1985
Drawing No. ZX – S 12026
17.1.1985, non-public drawing of the test rig details

P. Ott, 2001
Private Communications