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Non-steady Force Measurement in an Orpheous Gas Turbine Engine Using Hydraulic Analogy

J. S. RAO, K. L. AWASTHY & P. P. S. SANDHU*
Indian Institute of Technology, New Delhi-110016

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Abstract. Determination of non-steady forces in a real turbine stage is difficult due to local flow conditions e.g. high pressure, high temperature and inaccessibility to the region etc. Experimentation in a real turbine is also prohibitive due to the costs involved. Recently, an alternative method of arriving at these non-steady forces through use of hydraulic analogy has been tried on flat plates. The paper describes the simulation of an orpheous gas turbine engine stage on the rotating water table. It discusses the modelling aspects and presents a comparison of the experimental and theoretical results obtained.

1. Introduction

It is widely recognised that failure of turbine blades is a major cause of down time in turbo-machines. Although such failures are comparatively infrequent compared with number of blades in service, their failure can be very costly in terms of loss of aircraft particularly in case of single engine fighter aircraft. Dewey and Rieger[1] made a survey of blade failures in U.S.A. and found that high cycle fatigue alone was responsible for at least 40 per cent failures in high pressure stages of steam turbines. Blade failures due to metal fatigue are predominantly vibration related and suppression of blade vibration must be a high priority item in reducing the number of blade failures.

Blade vibrations are caused by a number of different excitation mechanisms, such as nozzle passing frequency excitation, low per revolution excitation, partial admission operation and stall flutter. Any of these sources may cause resonance with some blade natural frequency, and so give rise to large dynamic stresses.

For smooth and efficient extraction of work related fluid thermodynamic energy from the gas stream, impulse and reaction principle are both used in a given machine. The relative proportioning of impulse and reaction effect may also vary along the radial length of the blade for axial machines. Therefore, the blade shape must provide efficient fluid thermodynamic requirements and reliable structural strength

*Present Address : Dte. of Engineering, Air HQ., Vayu Bhawan, New Delhi.

using a suitable material. The steps involved in estimating the blade life can be as given in Fig. 1.

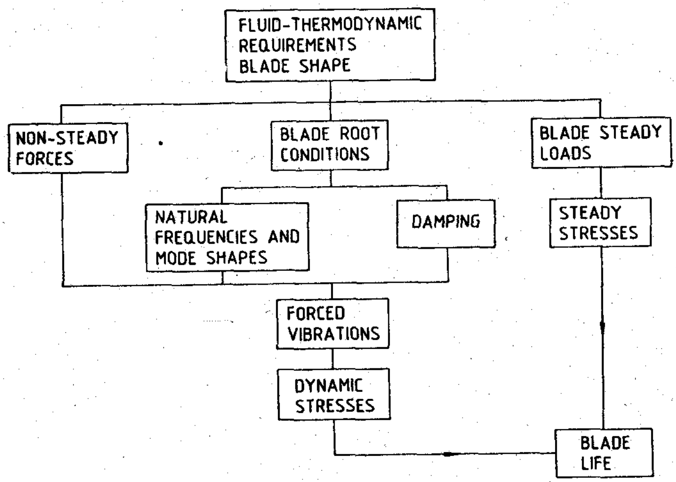

Figure 1. Design steps for a single blade.

In-service failure of aircraft and power turbo-machinery are a high cost operation item. Improved design procedures are needed to minimise blade failures. Many such failures are caused by fatigue of blade material initiated by non-steady gas forces acting on the blade under rotation and load conditions. The development of improved blade design capable of withstanding flow-induced non-steady forces, in addition to steady forces, is presently hampered by lack of knowledge of the magnitude and the nature of non-steady forces in addition to the steady forces themselves.

A major portion of the non-steady forces arises from the passage of moving blades through nozzle wakes. This gives rise to excitation at nozzle passing frequencies and at higher multiples (2x, 3x, etc.) of nozzle passing frequency (NPF). The strength of these excitation harmonics depends upon factors e.g. drop across the flow guides, ratio of number of guide blades to number of rotor blades, local Mach number, blade camber, flow incidence angle etc. The excitation occurs in tangential, axial and torsional directions. High pressure stage excitation in a turbomachine due to NPF and its harmonics are important as the blade natural frequencies fall in this high frequency excitation range.

A general computer programme which calculates the non-steady lifts and moments on stator and rotor blades has been developed by Rao[2]. In order to substantiate the results obtained through theoretical analysis, experiments were proposed to be conducted to determine the non-steady lift forces in a practical turbine stage. However, experiments for determination of non-steady forces in an actual turbo-machine stage will be hampered by local flow conditions like high velocities, high temperatures and inaccessibility to the region of flow etc. These conditions do not allow the non-steady forces to be determined in a manner convenient for their accurate prediction. Further, it is difficult to visualize the process due to high velocities; predictions have to purely depend upon certain indirect measurements that are made. Experiments in an actual turbo-machine are also prohibitive due to high costs involved in view of the sophisticated instruments that are to be specially designed and fabricated for the purpose. Hence, the method of hydraulic analogy in which the essential features of the gas flow under consideration are retained and where the complex flow phenomenon can be visualised is used to measure the non-steady forces in a turbo-machine stage.

The existence of a mathematical analogy between the frictionless flow of a liquid over horizontal free surface and the iso-entropic compressible flow of a gas with specific heat ratio $\gamma = 2$ has been known since long time[3]. The validity of such an analogy has been proved for unsteady case by Loh[4] for one dimensional flow and by Bryant[5] for two dimensional flow. The analogy has limited applications for quantitative studies because no gas having specific heat ratio equal to 2 is known to exist. For real gas situations, the analogy has been suitably modified. Rao[6] recognised that the correction of Mach number for modelling of real gases should be incorporated in the corrections of pressure and temperature also and accordingly proposed a modified analogy through a series of five reports. This modified analogy was verified by Rao *et al.*[7] using a flat water table and rocket nozzle models.

Significant investigations to determine the non-steady blade forces in a turbine stage were carried out simultaniously by two groups, one at Indian Institute of Technology (IIT), New Delhi, and the other at Rochester Institute of Technology, New York[8]. On the rotating water table designed for the purpose, water flows through a stage of accurately scaled turbine nozzle and blade profiles. Similarity was maintained between the nozzle exit conditions of the prototype and water table model in their experiments. Experiments were also conducted by Rao *et al.*[9] on the rotating water table designed at IIT, New Delhi, on the flat plate turbine stage. In the earlier experiments conducted on the water table, prespex sheet was used as a table surface which resulted in warping due to temperature effect and giving an uneven surface.

In the following sections, the modification of the water table done at IIT, New Delhi, is explained. It also gives the procedure for fabrication of the turbine stage blades. The modelling aspects of a gas turbine stage through modified analogy on water table are discussed. The turbine stage of an Orpheous gas turbine engine under consideration is simulated on the rotating water table and the non-steady lift ratios are obtained through experiments conducted on the simulated stage. The experimental results are compared with the theoretical results obtained from computer programme.

2. Modification of Water Table

The water table used by Rao *et al.*[9] employed a prespex sheet as the table surface which due to temperature effect warped resulting in depth differences in the circumference. Also the back pressure screen, employed to get the stage pressure ratio, used to get clogged resulting in the segment effects. During this experimentation, the prespex sheet was replaced with optically flat glass plate as a table surface and the back pressure mesh was also replaced with aluminium strip dam. This removed major deficiencies in the experiment viz. non-existance of a flat surface as observed with warped prespex sheet and non-uniform back pressure around the table because of clogging in front of a screen. The overall view of the water table is given in Fig. 2

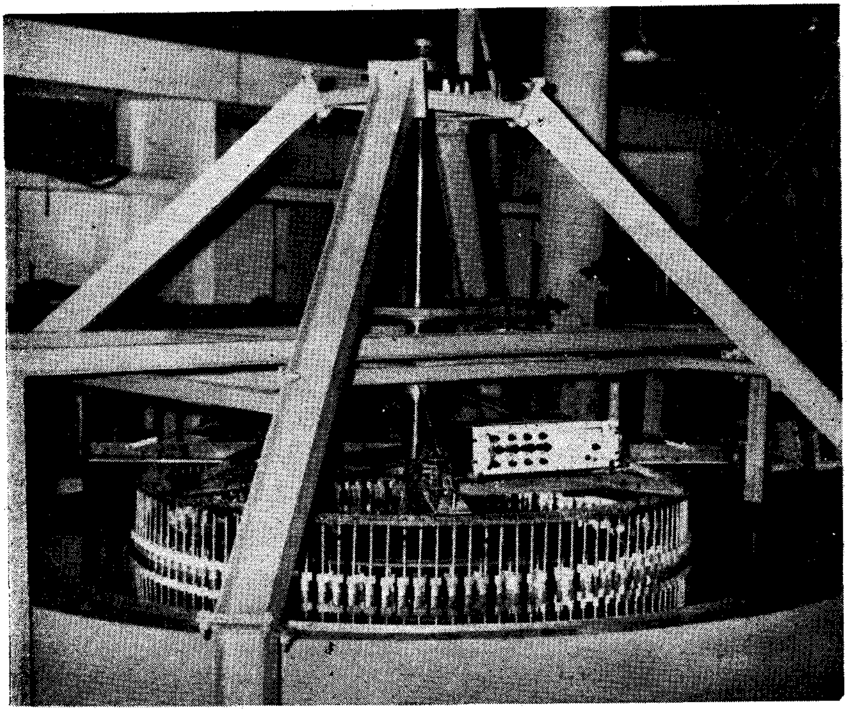

Figure 2. Overall view of the water table set up.

3. Fabrication of Turbine Blades

Nozzle guide vanes and the rotor blades fitted in an Orpheous gas turbine engine are of varying cross section, hence, the nonsteady force varies along the blade length. In order to model the turbine stage on the water table, several tests are to be conducted with blade profiles at different stations along the length. In this work, only the mid-section profile of the nozzle and rotor blade is selected for the tests. The profile drawing of nozzle and rotor blade at mid-section is shown in Fig. 3.

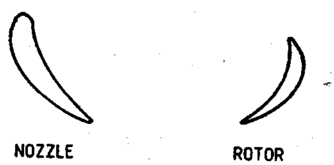

Figure 3. Nozzle and rotor blade profile at mid section.

Manufacturing operation of the nozzle and rotor blade is carried out on Beaver NC 5 milling machine provided with Plessey NC 1130 control system. The three-axis positioning is provided with two-axis linear and circular interpolation for maximum versatility. The manual part programming is done for both stator and rotor blade. The manual part programme is fed to the computer terminal for coding and onward transference into paper tape on Teletype machine. The Plessey NC 1130 control is fed with data coded into paper tape which instruct it as to the shape of the blades which the Beaver NC 5 is required to produce. The control interprets the data wtih the help of a photo-cell and feeds electrical signal to the machine tool axis which move as required. After completion of the milling operation, the blades are drilled and tapped precisely at the centroid of the airfoil section to receive the mounting rods. The foregoing procedure results in a very smooth surface finish.

4. Modelling Aspects

The orpheous gas turbine stage modelled on the water table has the following stage data :

σ_s Solidity ratio in the stator cascade	0.4094
σ_r Solidity ratio in the rotor cacade	0.5856
$\dfrac{C_r}{C_s}$ Blade chord ratio	0.7616
$\dfrac{b^l}{C_r}$ Axial gap/rotor half chord	1.90
$\dfrac{V_s}{U}$ Nozzle exit velocity/blade velocity	1.5746
α_s Stagger angle stator cascade	24°
α_r Stagger angle rotor cascade	32°
$\dfrac{d_r}{d_s}$ Blade spacing ratio	0.5324
Number of stator blades	63
Number of rotor blades	125
Pressure ratio of the stage	1.9754
Mach Number at the nozzle exit	0.798

In order to simulate the above stage on the rotating water table, as a first step the Mach number in the gas flow has to be modified to obtain corresponding Froude number on the water table, since the classical analogy is valid for $\gamma = 2$ only. The Froude number on the water table, corresponding to Mach number 0.798 at the nozzle exit in the gas flow, can be obtained from the equations given below.

$$\bar{M} = C_{\bar{M}}\, M \tag{1}$$

where

$$C_{\bar{M}} = \frac{\left(1 + \frac{\bar{M}^2}{2}\right) 1.5}{1.8371 \left[\left(\frac{2}{\gamma + 1}\right)\left(1 + \frac{\gamma - 1}{2} M^2\right)\right] \frac{\gamma + 1}{2(\gamma - 1)}} \tag{2}$$

Froude number corresponding to Mach number of 0.798 obtained from Eqn. (1) using an iterative procedure is

$$\bar{M} = 0.781$$

The flow on the water table is to be adjusted with this valve of Froude number at the exit of the nozzle blades besides simulating various geometric parameters mentioned above.

The second important parameter to be taken into consideration while simulating the flow through a turbine stage on the water table is pressure ratio of the stage. The stage pressure ratio in a turbine is given by the ratio of nozzle inlet pressure and blade exit pressure. The magnitudes of both the steady and non-steady forces will be affected by the stage pressure ratio since this parameter determines the flow through the stage. While simulating the turbine stage on water table, the pressure ratio in the prototype turbine has to be again modified to obtain the corresponding pressure ratio on water table. The modified pressure ratio can be obtained with the equation given below :

$$\frac{P_c}{P'_c} = C_p \left(\frac{P}{P'}\right) \tag{3}$$

where

$$C_p = \left[\frac{\left(1 + \frac{\bar{M}^2}{2}\right)}{\left(1 + \frac{\bar{M}'^2}{2}\right)}\right]^2 \left[\frac{1 + \frac{\gamma - 1}{2} M'^2_c}{1 + \frac{\gamma - 1}{2} M_c^2}\right]^{\gamma/(\gamma-1)} \tag{4}$$

The Froude number corresponding to Mach number of 0.254 and 0.535 at nozzle inlet and rotor exit conditions is given as 0.239 and 0.512 respectively. The corrected pressure ratio on the water table is 2.0561 and the corresponding height ratio is given by

$$\frac{P_1}{P_2} = \left(\frac{h_1}{h_2}\right)^2 = 2.0561$$

$$\frac{h_1}{h_2} = 1.4339 \tag{5}$$

The required pressure ratio on the water table is obtained by adjusting the height of the back pressure strip so that we get the required height at the rotor exit.

5. Simulation of the Stage

The actual orpheous gas turbine stage whose modelling aspects are considered in the preceding section is simulated on the rotating water table. The nozzle and rotor blades described in Section 3 have a heigh of 30 mm and a chord of 34.4 mm and 26.2 mm respectively. The solidity ratios of stator and rotor blades are 0.4094 and 0.5856 respectively. The pitch of the stator blades is 8.4025 cms and that of rotor blades is 4.4736 cms. The inlet tips of the stator blades are arranged on a circle having a diameter of 165 cms and the outlet tips of the rotor are on a circle having diameter of 180.5 cms. The stator stagger angle is 24° by keeping the blades in the radial direction on the water table. The rotor blades are arranged at an angle of 32° to the radial direction such that the condition of the stagger angle for rotor blade to be $\alpha_r = 32°$ is satisfied (Fig. 4).

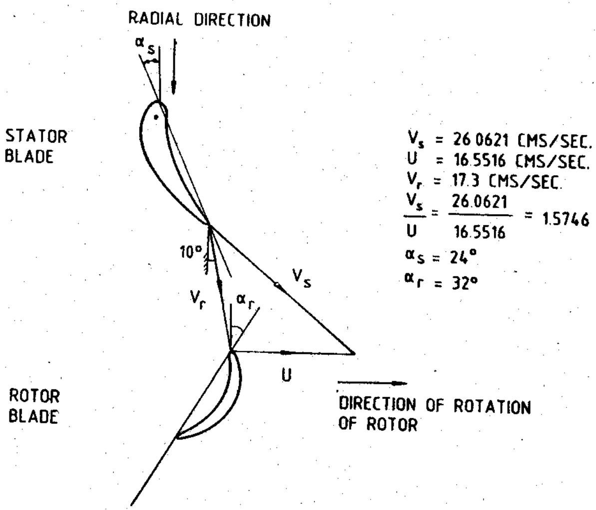

Figure 4. Velocity triangle at stator exit and rotor entry.

The nozzle exit Froude number of the value 0.781 corresponding to the Mach number of 0.798 in the gas flow is achieved on the water table by adjusting the flow to 15.846 kg./sec. (measured through orific meter). The depth obtained at stator exit with this flow is 1.1351 cms with a velocity of 26.0621 cm/sec, and corresponding depth at rotor outlet is obtained as 0.81 cms. The depth of water at stator inlet for this flow is 1.195 cms.

6. Test Procedure

Before commencing the test, the table surface is cleaned perfectly. The strain gauge bridge is balanced so that meter on the carrier amplifier shows zero reading in both the channels (axial and tangential) when no water is flowing. Then, the water flow is commenced. The flow is adjusted to get the required depth at the nozzle exit. The height of the aluminium strip is adjusted all around the periphery of the water table in order to get the required height of the water at rotor exit. The height of the water at rotor exit and nozzle exit is measured with the help of template at number of points on the water table surface. The drive motor for the rotor is switched on and is allowed to run in the established flow for some time to have steady condition. Then the exact speed of rotation is measured with the help of a stop watch. The signals are then taken on to real time analyser for axial and tangential loadings respectively. Typical signals obtained in both axial and tangential loadings are shown in Fig. 5. The bridge balance was exceptionally good during the entire experiment.

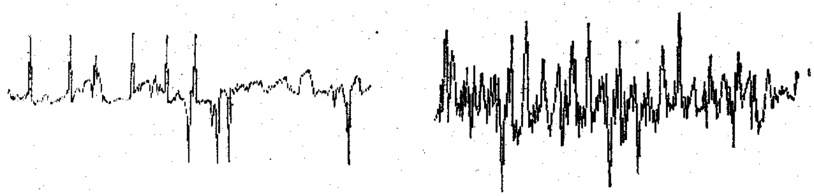

Figure 5. Sample chart recordings.

7. Analysis of the Test Data

The signals for both the direction were analysed respectively on the real time analyser to determine the magnitude of each frequency component present in the flow through out the range. Figs. 6 and 7 show the spectrum of amplitude Vs frequency obtained from such signal processing for axial and tangential loading at the velocity ratio of 1.5746. The 1st, 2nd and 3rd harmonics of the nozzle passing frequency in each spectrum are indicated by 1x, 2x and 3x respectively. The signal strengths are indicated in mV. The strength of the signal corresponds to non-steady force magnitudes in respective harmonics.

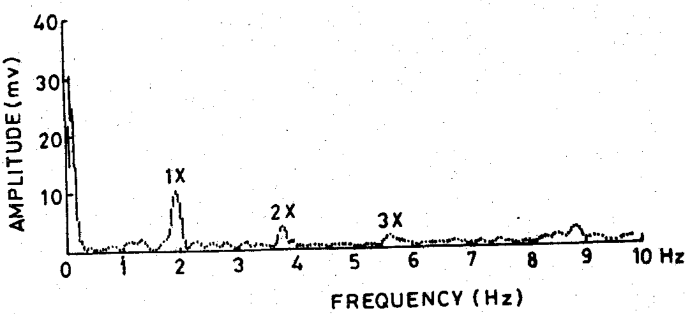

Figure 6. Frequency spectrum of the non-steady force in axial direction.

The corresponding DC level of the signals are obtained by passing the relevant signal through the storage Oscilloscope and by measurement of DC level directly from the Oscilloscope screen. The DC levels obtained in the present test are taken as

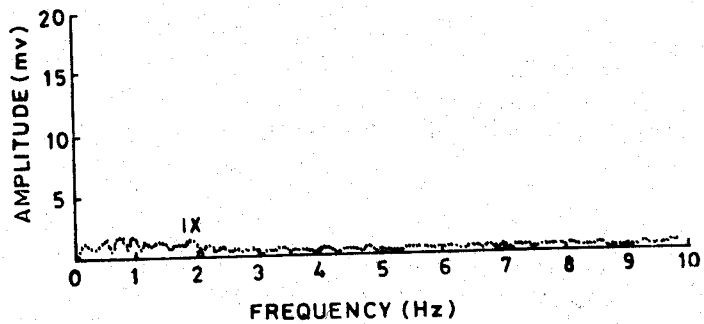

Figure 7. Frequency spectrum of the non-steady force in tangential direction.

steady bending forces acting on the moving blade. Table 1 show the values of the steady and 1st harmonic non-steady component obtained on water table.

Table1. Steady and Non-steady forces on rotor blade at Velocity ratio of 1.5746

Steady force (mV)		Non-steady forces (mV)	
Axial	Tangential	Axial 1st Harmonics	Tangential 1st Harmonics
248	53.6	10.2	1.00

The non-steady force ratios thus obtained are given in Table 2. The non-steady forces acting on the rotor blade in orpheous gas turbine stage are also obtained through the computer programme of Ramana Rao[10].

Table 2. Non-steady to Steady force ratio on rotor blade at velocity ratio of 1.5746

Axial		Tangential	
Theoretical compressible	Experimental	Theoretical compressible	Experimental
0.0475	0.041129	0.0141	0.01884

While obtaining the theoretical values for the stage under consideration, in addition to the data given in Section 4, the angle at which the flow is incident to the rotor blade is also taken into account. The incident angle is obtained through velocity diagram drawn for stage as shown in Fig. 4.

8. Conclusions

The comparison between the theoretical and experimental results is given in Table 2. The results obtained from the water table have exhibited a very close relation to the theoretical values obtained from the computer programme.

In the previous tests of Ramana Rao[10] on this water table for a flat plate stage, the agreement between the theory and the experiment has been not good though a significant improvement (attributed to modified analogy) has been shown over the tests of Rieger *et al.*[8] In the present test set up, a glass plate is used in place of prespex sheet which has shown a radical improvement in the experiment.

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