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Fast-response total pressure probe for turbomachinery application[†]

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Abstract

To evaluate the accurate performance and characteristics of turbomachinery, it is important to measure the unsteady flow phenomena downstream of the rotating blades. This paper presents the development of a fast-response total pressure probe for the measurement of the total pressure field at the exit of blades. The result of measurement in a one-stage axial turbine is also presented. The fast-response total pressure probe is fabricated by installing a fast-response pressure sensor in the cylindrical head of the probe. In terms of simplicity of the measurement system and data reduction method, this method is more competitive over established methods that use more than four sensors. The probe is applied to the one-stage axial turbine in order to measure the instantaneous total pressure downstream of rotor blades. The measured instantaneous signal is decomposed to obtain the blade-to-blade pressure distribution. The pressure distribution due to blade passing is clearly captured. Due to the loss generation in the casing region, the total pressure and its amplitude of fluctuation by the blade passing are lower in the shroud and hub region than in mid-span. The total pressure distribution at the exit of the rotor blade is found to be slightly different from blade to blade due to the geometric difference and the different relative positions of the rotor blades and stator vanes. The developed probe successfully measures the accurate total pressure distribution at the rotor exit, and allows the evaluation of the loss distribution and the accurate performance of turbomachinery.

Keywords: Axial turbine; Fast-response total pressure probe; Turbomachinery; Unsteady flow

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1. Introduction

To evaluate the accurate performance of axial turbomachinery and to investigate detailed flow field, it is important to measure the detailed flow phenomena at the exit of the blade. The flow phenomena downstream of the rotor blade show strong periodic and aperiodic, unsteady characteristics due to blade passing and rotor-stator interaction and so on. Therefore, it is necessary to measure the unsteady flow phenomena downstream of rotor blades for the accurate evaluation of performance.

A common method to measure the unsteady flow downstream of rotor is to install the fast-response pressure sensors near the head of the probe. Gossweiler et al. [1] and Humm et al. [2] developed fast-response probes by installing four pressure sensors near the head of each probe. Marathe et al. [3] developed a fast-response five-hole probe for the measurement of three-dimensional velocity and total pressure, in which five pressure sensors are installed near the head of the probe. These probes use multi-sensors (more than four sensors) to measure unsteady total pressure. Kupferschmied et al. [4, 5] developed a fast-response cylindrical probe with three miniature pressure sensors. This probe is able to measure flow yaw angle and velocity while the flow pitch angle is held negligible. They also developed a one-sensor fast-response probe that has the same function as a three-sensor probe in pseudo three-sensor mode.

Another method of measuring the flow downstream of rotor without a fast-response probe is installing a probe in the rotational frame and acquiring data through telemetry [6]. However, application of this method is limited when the flow phenomena are different from blade to blade [7, 8] because it measures flow only in one blade passage. It is also not capable of measuring the unsteady flow phenomena of rotor-stator interaction and so on.

Van Zante et al. [9] tried to measure the total pressure distribution downstream of the compressor by using an aspirating probe in which two hot wires are installed. This method gives good results in the measurement of unsteady total pressure but has intrinsic limitations in that the wires are easily broken under severe test conditions.

All the aforementioned methods use multi-sensors for the fast-response total pressure probe. These probes require bigger post-processing equipment, larger costs, and complicated data

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reduction processes. This paper presents the development of a fast-response total pressure probe with simple hardware, where only one fast-response pressure sensor is flush-mounted on the inside wall of the cylindrical head of the probe. The probe is applied to the unsteady flow downstream of a one-stage axial turbine, and the results are delineated.

2. Probe, test rig and facility

2.1 Fast-response total pressure probe

The schematic drawing of the developed fast-response total pressure probe is shown in Fig. 1. The head of the probe has a cylindrical bucket shape that stagnates the flow inside the bucket. The leading edge of the bucket is elliptic on the inside and its outside diameter is 3.175 mm. The probe is able to pass through a 6.35 mm hole. A miniature fast-response pressure sensor (Kulite XCS-062) is flush-mounted on the inside wall of the head of the probe. The diameter of the fast-response pressure sensor is 1.64 mm and the diameter of the tube is 3.175 mm. Its natural frequency is 300 kHz and its temperature limit is 120˚ C. The signals from the sensor are amplified and stored in the data acquisition system. The data sampling frequency is 100 kHz. The calibration results of the sensor are shown in Fig. 2. It is evident that the linearity of the pressure vs. voltage line is nearly perfect. The data for pressure and the rotor position are measured simultaneously to decompose the instantaneous pressure. The probe is insensitive to flow angles of up to ±29° for the yaw and pitch angles. This wide range insensitivity to flow angles makes the probe suitable for measuring the fluctuating downstream flow.

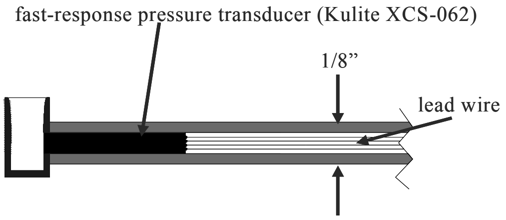


Fig. 1. Schematic drawing of fast-response total pressure probe.

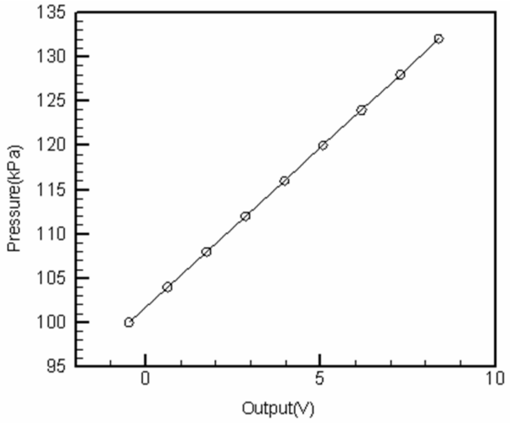


Fig. 2. Calibration curve of unsteady total pressure probe.

2.2 Test rig and facility

The conceptual drawing of the turbine test facility in Korea Aerospace Research Institute is shown in Fig. 3. The facility is designed such that the compressed air from the air compressors drives the test turbine. Table 1 shows the specifications of the test facility and test conditions of the turbine. The turbine test rig is a one-stage axial turbine. An encoder installed in the shaft measures the rotational position of the rotor makes it possible to decompose the instantaneous total pressure signals. Fig. 4 is a picture of the turbine test facility and Fig. 5 shows the cross-section of the test turbine. The probe is mounted on an auto-traverse unit that can traverse both radial and rotational directions. The measured position (X/C) is 0.31, as shown in Fig. 5. The number of measured positions in the radial direction is 52.

Table 1. Specifications of turbine test facility and test condition.

Item		Specification
Max. flow rate	Kg/s	12
Max. turbine power	kW	250
Max. turbine torque	Nm	1200
Turbine inlet total pressure	kPa	161
Turbine inlet total temperature	K	312
RPM	-	1700
Rotor tip diameter	mm	548
Rotor hub diameter	mm	484
No. of blade	-	122

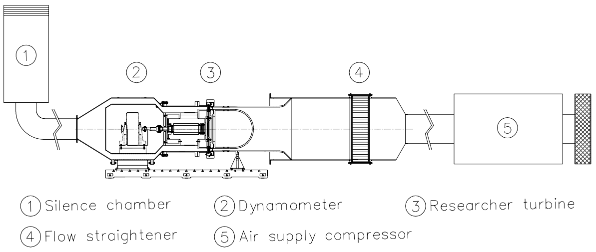


Fig. 3. One-stage turbine test facility at KARI.

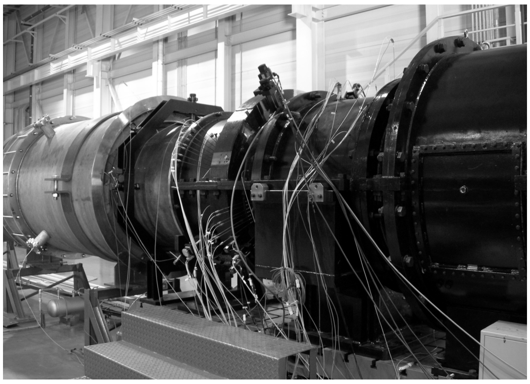


Fig. 4. Axial turbine test rig.

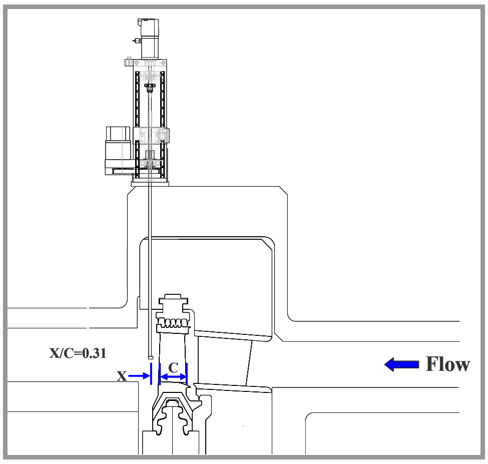


Fig. 5. Cross-section of the turbine test rig and probe traverse location.

3. Decomposition of instantaneous pressure

To analyze the measured instantaneous signal, a decomposition method suggested by Suryavamshi et al. [10, 11] and modified by Kang et al. [12] is used. Each discrete measurement of the total pressure is indexed with *i*, *j,* and *k*. The subscripts *i*, *j,* and *k* represent indices in the ensemble averaging process (*i* indicates the index of revolution, *j* the index of the blade, and *k* the index of the window in the blade passage). In this study, there are 25 windows in the blade passage. The instantaneous total pressure (Po_{ijk}) is decomposed into shaft resolved ($(Po_{jk})_s$) and unresolved (Po'_{ijk}) components.

$$Po_{ijk} = (Po_{jk})_s + Po'_{ijk} \tag{1}$$

where the shaft resolved component is obtained by ensemble averaging the instantaneous pressure ratio data set.

$$(Po_{jk})_s = \sum_{i=1}^{N_{rev}} (Po_{ijk}) / N_{rev} \tag{2}$$

When the number of data (*N*) in every window is not the same, Eq. (2) is modified into Eq. (2a) [11],

$$(Po_{jk})_s = \sum_{i=1}^{N_{rev,jk}} (Po_{ijk}) / N_{rev,jk} \tag{2a}$$

$$Po'_{ijk} = Po_{ijk} - (Po_{jk})_s \tag{3}$$

The shaft resolved component is generally called the phase-lock-average total pressure. It is further decomposed into time average ($\overline{Po}$), revolution periodic ($(P_{ojk})_{RP}$), and revolution aperiodic ($(P_{oj})_{RA}$) components.

$$(Po_{jk})_s = \overline{Po} + (Po_j)_{RA} + (Po_{jk})_{RP} \tag{4}$$

where

$$\overline{Po} = \sum_{n=1}^{N_{data}} Po_n / N_{data} \tag{5}$$

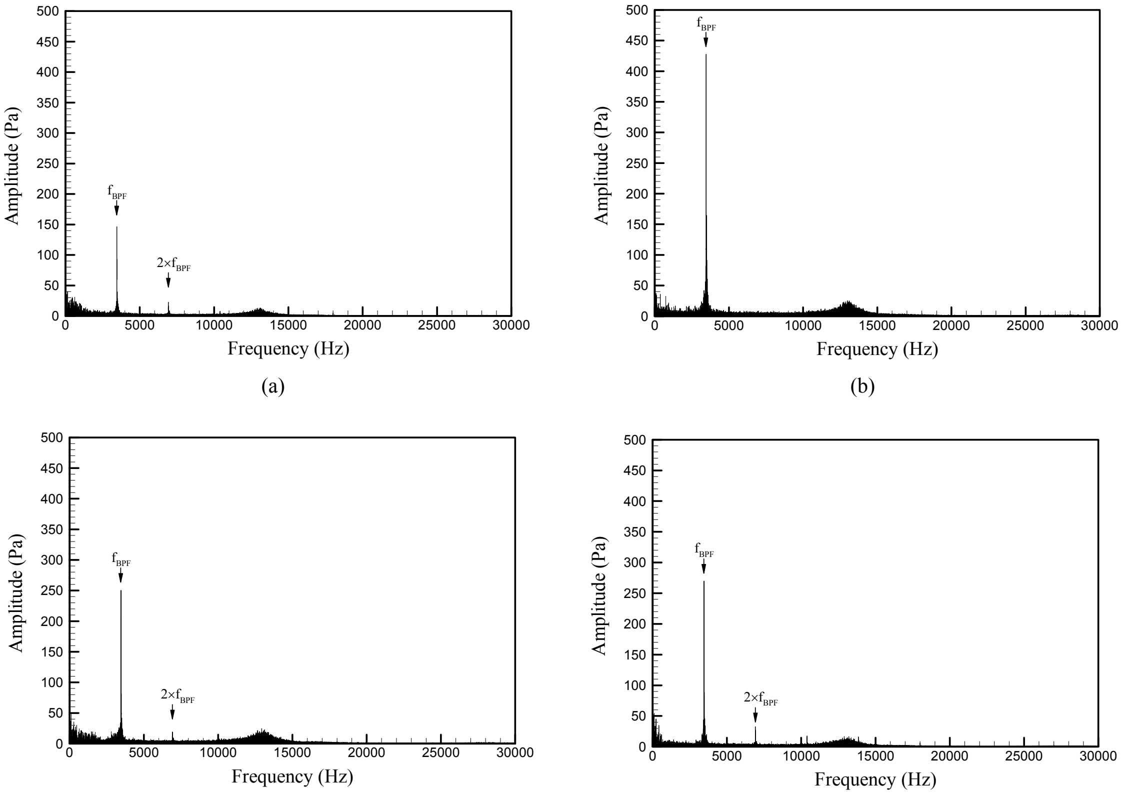
$$(Po_j)_{RA} = \left\{ \sum_{k=1}^{N_{pb}} \left[(Po_{jk})_s - \overline{Po} \right] / N_{pb} \right\}_j \tag{6}$$

$$(Po_{jk})_{RP} = (Po_{jk})_s - \overline{Po} - (Po_j)_{RA} \tag{7}$$

4. Results

The spectrum analysis of instantaneous pressure data measured at 3%, 28%, 51%, and 97% of span is shown in Fig. 6. The blade passage frequency (f_{BPF}) at 1700 rpm is 3456.7 Hz. The figures clearly show the blade passage frequency, which means the probe has enough frequency response characteristics. Maximum pressure is measured approximately at 28% and 82% of the span. The phase-lock-averaged total pressure and the revolution periodic component measured at the 28% span of the rotor blade are shown in Fig. 7. The pressure fluctuation by blade passing is clearly seen in the figure and the difference between the maximum and minimum pressure from one blade to another is approximately 1.4 kPa. The phase-lock-averaged total pressure and revolution periodic component near the shroud is shown in Fig. 8. The mean value of total pressure near the shroud is lower than that in the mid-span region, and the difference between the maximum and minimum pressure from one blade to another is approximately 0.8 kPa. The phase-lock-averaged total pressure and revolution periodic component near the hub are shown in Fig. 9. The mean value of total pressure near the hub is lower than that in the shroud region, and the difference between the maximum and minimum pressure from one blade to another is approximately 0.5 kPa. The reduced amplitude of total pressure from the mid- span region to the hub and shroud regions is believed to be due to the increased loss at the hub and shroud regions. This result is similar to that of Christopher et al. [7]. The phase-lock-averaged total pressure distribution from hub to shroud is shown in Fig. 10, where the core and wake flow regions are clearly visible as high and low pressure regions. The high-pressure core flow is found to be divided into two high-pressure regions and that the lowest total pressure region is located in the hub.

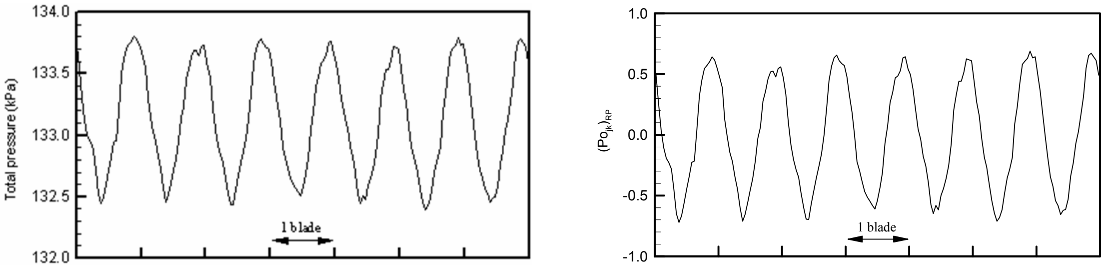
As seen in Figs. 7-10, the waveform of pressure in a blade width is slightly different from blade to blade. This is more evident in Fig. 11, where the phase-lock-averaged total pressure at mid-span during the half revolution period is shown. The observed difference is due to the different flow phenomena in all the blade passages caused by the geometric difference of the blades and the different relative positions of the rotor blades and stator vanes. This means that the flow phenomena and performance based on a blade passage may be different from the averaged data from for all the blades. The effect of the total pressure fluctuation on the efficiency of turbine is analyzed as follows. The total-to-total efficiency based on the averaged total pressure ($\overline{\eta_{TT}}$) can be calculated using Eq. (8).



(a) (b)

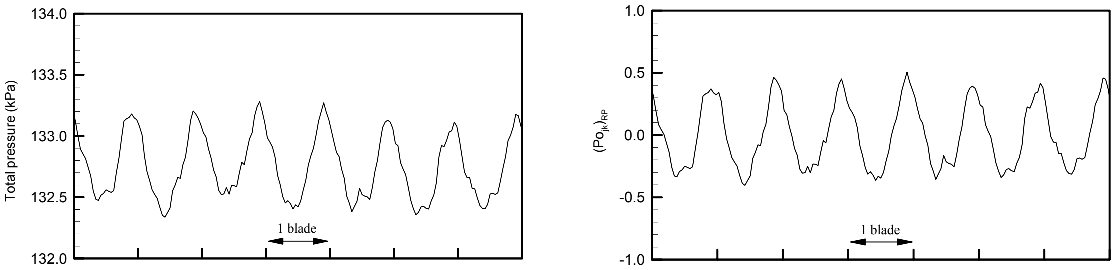
(c) (d)

Fig. 6. Spectrum analysis of instantaneous pressure data at (a) 3% (near hub), (b) 28 %, (c) 51%, and (d) 97% (near shroud) span.



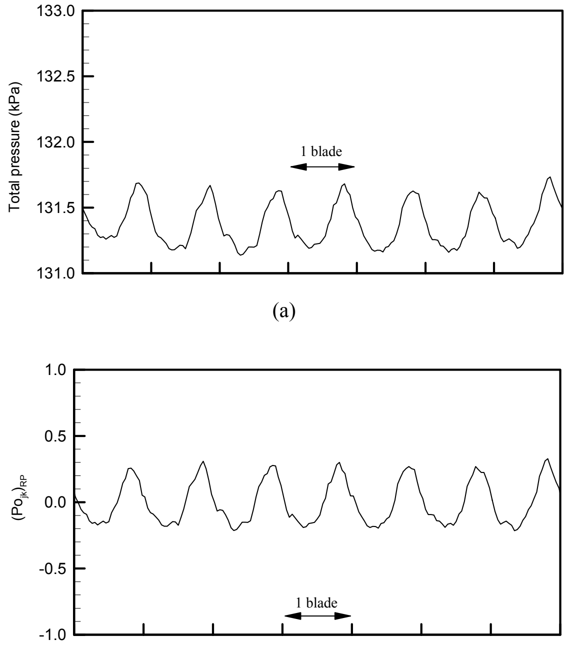
(a) (b)

Fig. 7. Measured total pressure at 28% span: (a) Phase-lock average; (b) Revolution periodic component.



(a) (b)

Fig. 8. Measured total pressure near shroud: (a) Phase-lock average; (b) Revolution periodic component.



(a)

(b)

Fig. 9. Measured total pressure near hub: (a) Phase-lock average; (b) Revolution periodic component.

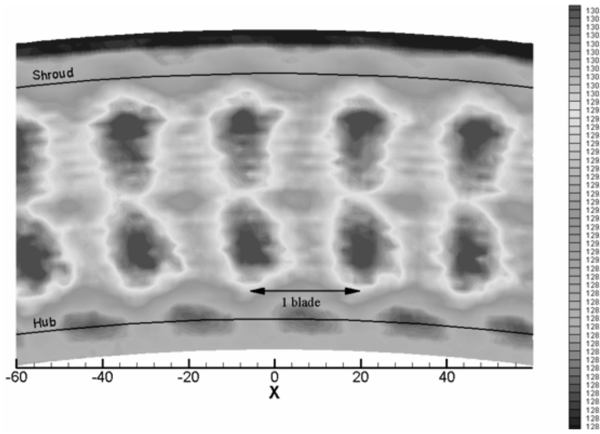


Fig. 10. Phase-lock averaged total pressure distribution downstream of rotor blades.

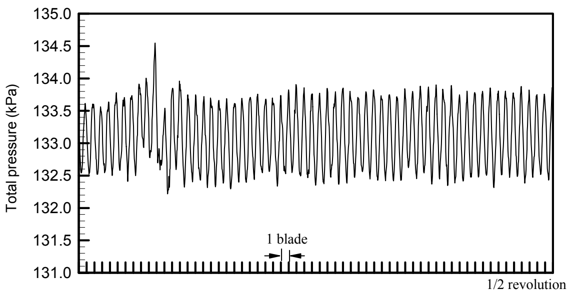


Fig. 11. Phase-lock averaged pressure at mid-span during half-revolution period.

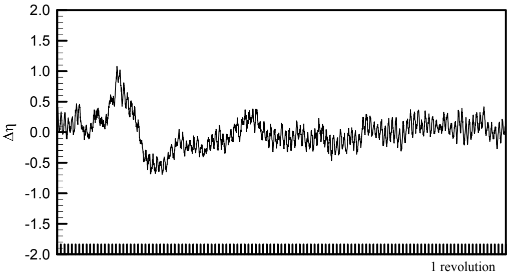


Fig. 12. Variations of non-averaged efficiency at mid-span during one revolution period.

$$\bar{\eta}_{TT} = \frac{Torque \times \omega}{\dot{m} C_p T_{O1}\left[1-\left(\frac{\bar{P}_{O2}}{P_{O1}}\right)^{\frac{\gamma-1}{\gamma}}\right]} \tag{8}$$

where C_p is the specific heat and ω is the angular velocity. The efficiency based on the total pressure of a window can be calculated through Eq. (9).

$$\eta_{TT} = \frac{Torque \times \omega}{\dot{m} C_p T_{O1}\left[1-\left(\frac{P_{O2}}{P_{O1}}\right)^{\frac{\gamma-1}{\gamma}}\right]} \tag{9}$$

The difference between these two efficiencies is calculated as Eq. (10), as shown in Fig. 12, where the results for all the blade passage are also shown.

$$\Delta\eta = \bar{\eta}_{TT} - \eta_{TT} \tag{10}$$

Because of the slight difference in total pressure in each blade, the maximum efficiency is larger than the average efficiency by 1.1%, and the minimum efficiency is lower than the averaged efficiency by 0.7%. This shows that because this fast-response total pressure probe measures flow field downstream of all the blades, it can detect damage on a specific blade.

5. Conclusion

Fast-response total pressure probe with one sensor is developed to measure the total pressure downstream of blades in a turbomachiney. The probe is applied to a one-stage axial turbine, in which it successfully measures the unsteady total pressure distribution downstream of the blades. Due to the loss generation in the hub and shroud regions, the averaged total pressure and the amplitude of fluctuation by blade passing are lower in the hub and shroud than in mid-span. In addition, the waveforms of total pressure downstream of the blades are found to be slightly different from blade to blade, which could be due to the geometric difference between the blades and the different relative positions of the rotor blades and stator vanes. Since it is possible to measure the accurate total pressure distribution downstream of the blades, it is also possible to measure the accurate performance of turbomachinery.

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