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New method for mapping radial turbines exposed to pulsating flows

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A R T I C L E I N F O

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A B S T R A C T

In this paper, a new method for mapping radial turbines will be demonstrated and shown to be superior to the standard approach typical to gas-stand laboratories. This is especially true in situations where the turbine is exposed to unsteady flow as is the case in the turbocharger. In order to generate realistic pulsating flows in a laboratory environment, a pulsation generator, including a cylinder head of three cylinders, was designed in the University of Bath.

This study demonstrated that the standard mapping approach is not able to capture the period of negative power since it is a purely transient phenomenon. Therefore, a new approach of generating maps is suggested based on extrapolating data generated from the unsteady measurement resulting from the pulse rig. This method is then tested by demonstrating its ability to recreate the real pulsating behaviour and ability to predict compressor power.

In this study, turbine instantaneous power measurements were conducted under both three-cylinder mode and two-cylinder modes. Negative turbine power was measured during the trough of a pulse, indicating that both turbine and compressor absorb energy from the rotating inertial during that period. This study found the negative turbine work can take up approximately 15% of turbine net work under 20 Hz pulses. This percentage is even more once one or more cylinders are deactivated. This paper highlighted that if not considering the negative efficiency, it will result in an error shaft speed prediction, and thereby may affect the overall performance of turbocharger even engines.

1. Introduction

One dimensional (1D) gas dynamic codes are commonly used for matching a turbocharger to an internal combustion engine (ICE) due to its reduced computational cost and reasonable accuracy. They have become an essential tool to engine development and calibration engineers as they permit a system level understanding an ability to study component matching and control. In most 1D turbocharged engine models, the solution of turbocharger performance parameters is usually based on the turbocharger performance maps which is a look-up table gathered from steady-state gas stand test. Therefore, in order to improve the turbocharger performance predictions, providing accurate and effective turbocharger maps to the 1D model is paramount. Unlike a turbocharger compressor, a turbocharger turbine is inherently subjected to pulsating flows when mounted to an ICE due to the opening and closing of the exhaust valves, and the reciprocating motion of pistons. The upstream gas fluctuations cause the turbine to operate under a wide range of working conditions, causing significant instantaneous variations in turbine blade speed ratio [1–3]. However, it is well known that the whole picture of turbine performance characteristics under pulsating flow conditions cannot be entirely captured using conventional steady-state gas stand measurements, since they are limited by the surge and choke of compressors.

In order to address the limited data on the turbine, extrapolation techniques are employed to extend the range of maps. Researchers of [4–6] extrapolated the map based on empirical relationships, where the turbine mass flow parameters and efficiencies were expressed as polynomial or exponential functions of the turbine expansion ratio, blade speed ratio or speed parameters. [7–11] used partly empirical models where the turbine was modeled as adiabatic nozzles of effective area fitted with experimental data. [12,13] used turbine mean-line models to predict the turbine efficiency, where loss terms were validated against steady-state measurement. However, those fitting methods suffered from a limited range of data, where extreme off-design conditions such as choking or impeller free-wheeling cannot be captured via standard gas stand measurement. Thus, errors are involved in the extrapolation

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process due to the lack of flow physics. This paper present a novel fitting technique to predict the swallowing capacity characteristic of the turbine by using data directly from unsteady measurements. A mean-line model was also developed which is able to predict the free-wheeling behaviour of turbine, that is, during periods of negative efficiency.

The phenomenon of negative turbine efficiency was first presented by Karamanis [14], and later confirmed by the experimental study of Szymko et al. [1] and CFD study of Palfreyman and Martinez-Botas [15]. They commented that the negative efficiency is caused by the free-wheeling of impeller occurring towards the end of the pulse where the pressure ratio is close to unity. Under such circumstance, the momentum energy of the spinning wheel and shaft is transferred (dissipated) to the gas thereby resulting in an actual power that is below zero. Therefore, the negative efficiency is a physical phenomenon that *only* occurs in the pulsating flow conditions and cannot be easily captured by standard Gas-stand measurements. As investigated by Szymko [1], 50% of the pulse cycle contains negative efficiency at 20 Hz pulses, and the negative power accounts for 4% of the available isentropic power. The present research work found the impact of not considering the role of negative energy would result in an error in the prediction of compressor power of up to 7.9%.

Only a few papers mentioned the implement of turbine unsteady map on engine performance study. Pesiridis et al. [16] construct an equivalent turbine unsteady map based on cycle averaging the unsteady data collected from experiment. However, this may not be a practical way since it requires a very large library of unsteady testing data to build the map. Chiong et al. [17] produced a cycle-averaged turbine unsteady map by integrating a 1D volute model with a mean-line rotor model. They found the cycle-averaged turbine map is not far from quasi-steady assumption. However, another drawback of using cycle-averaged approaches is that the valuable turbine performance data at off-design regions will be eliminated due to average.

Extensive studies [12,18–28] were carried out to understand the unsteady behaviour of turbine. As noted by many researchers, a clear sign of the turbine behaviour under pulsating flow conditions is turbine characteristic map forming a hysteresis loop, indicating the unbalance of mass or energy during its operation. There is a general agreement that the unsteady behaviour of a turbine results from the internal volumes of the turbine, and the largest contribution comes from the volute, and the rotor itself is deemed to operate in a broadly quasi-steady manner [2,12,23,27–29]. To quantify the intensity of unsteadiness of a turbine subjecting to pulsating flows, researchers of [23,24] used strouhal number (*St*) as an indicator; Szymko et al. [1] presented modified strouhal number (*MSt*), where the St was corrected by the fraction of effective pulse length over the entire pulse length. Copeland et al. [27] introduced a Λ criterion, as found in EQ. (1) where the strouhal number was modified by taking the pulse amplitude effects into account. where $A, f, L, \overline{P}, v$, and γ are the pulse amplitude, pulse frequency, domain characteristic length, mean turbine inlet pressure, averaged flow velocity across the turbine stage, and specific heat ratio respectively.

$$\Lambda = \Pi \cdot St = \frac{2A}{\gamma \overline{P}} \cdot \frac{fL}{v} \tag{1}$$

Cao et al. [28] argued that the quasi-steady assumption of the rotor only valid under cycle-averaged basis, and the intensity of turbine unsteadiness is varying instantaneously and depended on the pulse form. Cao et al. [28] proposed $|\varepsilon(t)|\beta_{local}(t)$ criterion, as shown in EQ. (2). Where t_f is the time of fluid particle traveling through the domain, and Δt is the data sampling time. Interestingly, it could derive Λ parameter by time averaging the $|\varepsilon(t)|\beta_{local}(t)$ over a pulse. As [28] suggested, if this parameter is smaller than 0.07, the turbine is deemed to operate in a quasi-steady manner.

$$|\varepsilon(t)|\beta_{local}(t) = \frac{|\Delta P(t)|}{\overline{P}} \frac{t_f}{\Delta t} \tag{2}$$

Based on those criteria, it could found that a practical way to reduce the measured turbine unsteadiness is to reduce the characteristic length of what is considered to be the turbine stage. A short route can simply be created by putting the inlet measurement section closer to the rotor.

In summary, while there has been considerable effort by other authors to understand the nature of flow unsteadiness in the turbine, there are limited usable proposals that aim to map the true unsteady performance in a way that is useful to engine modelers. Thus, this paper aims to propose a new method of measuring turbine performance in a gas stand that is able to represent the true unsteady nature when exposed to flow pulses. This is particularly important since the widely accepted turbine mapping technique is unable to account for periods of negative turbine work.

The structure of the paper is as follows:

- Section 2 will detail the experimental methodology including the details of gathering data from a mixed-flow turbine using a bespoke pulsating generator. It will also briefly touch on the CFD modeling utilized to complement this data set and help in demonstrating the role of the quasi-steady criterion.
- Section 3 will display and discuss the experimental data set used to quantify the unsteady performance of the turbine. The unsteady criterion is used as a method to down-select the data that is then fed into the maps in the next section.
- Section 4 will demonstrate map-fitting techniques using physics-based models of the turbine swallowing capacity and efficiency.
- summarises the overall unsteady mapping technique from data gathering and processing to final performance map generation.
- Finally, section 6 selects a number of cases studies and demonstrates the influence of the new methodology on the turbocharger performance prediction.

The most significant outcome of the work is the proposal for a new method of mapping the turbine performance that uses the pulse itself, thereby creating a single speed line using one dynamic measurement over a pulse cycle. This, to the author's knowledge, is a completely unique approach.

2. Methodologies

2.1. Experimental apparatus

Fig. 1 illustrates the schematic view of the turbocharger test facilities in the University of Bath. Air supplied to the test cell is compressed externally via two Ingersoll Rand compressors, providing a maximum pressure of 8 bar and 0.7 kg/s mass flow rate. A ball valve (Kinetrol) and a butterfly valve control the pressure and mass flow level upstream of the pulse generator. Two electrical heaters in parallel, where each can deliver up to 44 kW and typically heat air up to 750 °C. The heated air then passes through the pulse rig inlet manifold, inlet valves, cylinder, and exhaust valves, producing pulsating flow at the turbine inlet. The air outlet of the turbine is recycled by joining the flow with fresh air prior to the heater in order to reduce the electricity consumption. Two data acquisition systems Sierra CP and Dewetron Soft were used to performing slow measurement (40 Hz) and fast measurement (up