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Non-adiabatic pressure loss boundary condition for modelling turbocharger turbine pulsating flow

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ABSTRACT

This paper presents a simplified methodology of pulse flow turbine modelling, as an alternative over the meanline integrated methodology outlined in previous work, in order to make its application to engine cycle simulation codes much more straight forward. This is enabled through the development of a *bespoke non-adiabatic pressure loss* boundary to represent the turbine rotor. In this paper, turbocharger turbine pulse flow performance predictions are presented along with a comparison of computation duration against the previously established integrated meanline method. Plots of prediction deviation indicate that the mass flow rate and actual power predictions from both methods are highly comparable and are reasonably close to experimental data. However, the new boundary condition required significantly lower computational time and rotor geometrical inputs. In addition, the pressure wave propagation in this simplified unsteady turbine model at different pulse frequencies has also been found to be in agreement with data from the literature, thereby supporting the confidence in its ability to simulate the wave action encountered in turbine pulse flow operation.

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1. Introduction

Turbocharging is regarded as one of the key elements in the success of downsized internal combustion engine systems, an effective strategy towards CO_2 emissions reduction. These days, a turbocharger is no longer restricted to its conventional application, but in various other usages such as turbo-compounding [1], electrically assisted [2,3] and steam turbocharging [4], where the turbine will often operate at more extreme conditions. In all cases, the process of engine-turbocharger matching during the development stage plays a significant role towards achieving the best possible system performance, in terms of minimizing fuel consumption while maintaining good transient response. In current industry practice, engine modelling does not consider the full unsteady analysis of the turbocharger turbine, but instead treats it as a quasi-steady device. While this traditional approach can provide adequate simulations of the engine's steady state engine performance, its deficiencies become apparent when attempting to accurately predict transient response [5], especially so when the desire is to predict the benefit of turbocharger technologies such as twin scroll turbines. Numerous unsteady turbine models have been developed over the years, yet none of these models have been widely implemented into commercial one-dimensional engine cycle simulation codes, mainly due to the associated complexity.

1.1. Background study

Commercial one-dimensional engine cycle simulation software tools model the turbocharger turbine by following a quasi-steady approach (turbine inlet and outlet are considered as the points where the flow conditions are experimentally measured). However, due to the reciprocating nature of the engine, the flow entering the turbine is of a pulsating nature, the form of which varies depending on engine speed, displacement, number of cylinders, etc. Over the last fifty years, several researchers have investigated this phenomenon, showing that during each engine cycle the instantaneous mass flow measured at the turbine inlet does not follow the steady-state characteristic, but instead forms an unsteady hysteresis loop, particularly at lower pulse flow frequencies [6–17]. The unsteady characteristic was also found (experimentally [16] and analytically [18]) to vary in accordance with turbine inlet pulse frequency. However, detailed experimental flow field investigations of the turbine [19] suggest that the rotor itself does operate in quasi-steady manner (compared to the turbine

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Nomenclature

a	speed of sound (m/s)
A	dimensionless speed of sound
A_A	dimensionless entropy level
$\mathbf{C}$	source vector
C_θ	rotor tangential velocity
d	non-adiabatic term
e	specific internal energy (J/kg)
F	cross-sectional area (m^2)
$\mathbf{F}$	flux vector
G	friction source term
h	specific enthalpy (J/kg)
K	loss coefficient/pressure loss coefficient
L	loss
$\dot{m}$	mass flow rate
M	Mach number
MFP	pseudo-non-dimensional mass flow parameter
N_s	dimensionless speed parameter
NAPL	non-adiabatic pressure loss
p	pressure (Pa)
PR	pressure ratio
q	specific heat transfer rate (J/kg s)
t	temporal dimension
T	temperature (K)
u	velocity (m/s)
U	dimensionless velocity/rotor tangential velocity
$\mathbf{W}$	state vector
W	relative velocity
$\dot{W}_{act}$	work transfer (kW)
x	spatial dimension
Z	blade number

Superscript

$*$	values normalized with respect to entropy level

Greek letters

α	boundary cross-sectional area ratio
β	relative flow angle (rad)
Δ	increment/changes
κ	ratio of specific heats
λ	Riemann variable
ρ	density (kg/m^3)
σ	standard deviation
θ	absolute flow angle (rad)
μ	gas flow dynamic viscosity

Subscripts

0	total or stagnation value
1	upstream of subject station
2	downstream of subject station
blade	parameter associated to rotor blade
B	parameter associated to boundary
c	corrected form of parameter
clearance	parameter associated to clearance loss
d	derived parameter
dev	prediction deviation
disk friction	parameter associated to disk friction
entry	parameter associated to rotor entry
exp	experimental value
hub	dimension at rotor hub
in	incoming or incidence parameter
incidence	parameter associated to incidence loss
max	maximum value
n	uncorrected form of parameter
out	outgoing or transmitted parameter
passage	parameter associated to passage loss
pred	prediction value
ref	reference value
tip	dimension at rotor tip
turb	parameter associated to turbine

stage as a whole), and this has been analytically supported by Strouhal number analyses [16,20] and 3D CFD [21–23]. This leaves the turbine volute as the primary source of the observed unsteadiness.

The major deficiency of a quasi-steady turbine model is therefore the lack of a spatial dimension (or dimensions), and the consequent inability to account for the flow dynamics taking place within the turbine volute during pulsating flow operation. Ref. [23] further highlighted the deficiencies of a quasi-steady turbine model where the temperature dependent heat transfer and mechanical losses do not vary according to on-engine conditions but are restricted to the measured "apparent" steady-state characteristic [24]. Thus modelling improvements have been obtained by incorporating, as a first step, the volume of the volute in a 0D or "filling-and-emptying" approach [25], and taking it a step further by explicitly modelling the volute as a one-dimensional form to additionally capture pressure wave action effects [23,26–33].

In order to model the quasi-steady and unsteady characteristics of the turbine rotor and volute respectively, an adiabatic pressure loss (APL) boundary and a series of 1D tapered (non-constant cross-sectional area) pipe ducts have previously been considered [30,34]. Once calibrated for steady flow conditions, this type of unsteady turbine model has shown its ability to capture the wave action phenomena under pulsating flow operation, reflected in the swallowing capacity hysteresis prediction and the correct variation trend against growing pulsating flow frequency [18]. Although this model showed satisfactory unsteady mass flow prediction, prediction of instantaneous turbine power was less convincing. In an attempt to improve power prediction [35], a turbine meanline model was integrated into the 1D model. From the established flow state, the meanline model makes a performance prediction using a number of empirical loss relations, which are in turn based on knowledge of the turbine geometry and operating parameters. Validation at different pulse flow frequencies suggested good potential for the integrated 1D-meanline turbine model [36,37], but this new methodology, although innovative, has a few potential downsides.

Firstly, the meanline model required detailed rotor geometrical inputs, e.g., rotor blade angle and tip-to-shroud clearance (this will be discussed further in Section 4). However, rotor geometrical information is rarely accessible especially during engine-turbocharger matching. Secondly, the instantaneous turbine power calculation was performed from the already established pulsating turbine instantaneous flow field, sampled into a complete pulse cycle such that the phase synchronization of the instantaneous velocity components extracted at different locations in the turbine volute can be accomplished (e.g., the gas flow velocity taken at the volute tongue for instantaneous rotor upstream tangential velocity calculation via the free-swirl relation). Even though this approach showed good results in cold-flow testing data validation, such a model may fall short during transient engine modelling in the absence of continuous complete pulse cycles under constant frequency. Lastly, the pressure loss boundary used in this methodology to establish the pulsating turbine flow field was considered

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