Performance and flow-field assessment of an EGR pulse optimised asymmetric double-entry turbocharger turbine

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ABSTRACT

In this paper it will be presented a novel turbine concept specifically designed for exhausts pulse flow energy conservation and EGR control. In order to combine both these features, an asymmetrically divided double-entry turbine was developed to respond to the imbalance of mass flow due to EGR extraction from one side. The EGR side was equipped with variable geometry vanes in order to control the EGR rate and to optimize the flow entering the turbine wheel, whereas no vanes were contemplated in the other turbine entry. A detailed analysis on the design and efficiency of the asymmetric turbine is provided in this paper.

Keywords: Double-entry turbine, Variable Geometry Turbocharger, Exhaust Gas Recirculation, Pulse turbocharging, Computational Fluid Dynamics, Unequal admission

NOMENCLATURE

Π	Pressure ratio	α	Absolute angle (°)
$\dot{m}$	Mass flow (kg/s)	β	Relative angle (°)
A	Cross sectional area (mm^2)	o_{nv}	Vane throat distance (mm)
R	Radius to area centre (mm)	s_{nv}	Vane pitch (mm)
c	Absolute flow velocity (m/s)	l_{nv}	Vane chord (mm)
ρ	Density (kg/m^3)	η	Efficiency
θ	Circumferential angle (°)		
b	Inlet width (mm)		
ψ	Azimuth angle (°)		
VGT	Variable Geometry Turbocharger		
EGR	Exhaust Gas Recirculation		
SPR	Scroll Pressure Ratio		
CFD	Computational Fluid Dynamics		
NO_x	Nitro oxygen		

Subscript

0	Total/Stagnation condition	L	Large scroll
1	Volute inlet	S	Small scroll
4	Vane exit	D	Double-entry
r	Radial direction	ts	Total-to-static
θ	Circumferential direction		

1 INTRODUCTION AND PROBLEM DEFINITION

Due to the increasingly stringent emission regulations and demand for high fuel economy, turbocharger is inherently one of the most promising enabling technologies towards achieving engine design for low emissions and fuel consumption. Turbochargers are not only expected to provide efficient exhausts energy recovery, but they are also used in order to support engine operation by controlling engine back-pressure for enhanced EGR rate. In large multi-cylinder engines, single-entry VGTs are normally used due to their ability to support EGR and instantaneous boost pressure response. However, the disadvantage of single-entry VGTs is that they do not enable to maximize energy extraction out of the exhausts pulses (pulse turbocharging) since there is no pulse separation within the turbine volute (as in multiple-entry turbines). Hence, aim of this research, is that of combining the advantages of exhausts pulse energy extraction and variable geometry turbocharging in one single turbine design.

Pulse turbocharging (i.e. conserving pulse energy until the rotor entry) is a well-known technique to utilise the maximum energy of the exhausts. In order to avoid exhausts pulse interference, these are isolated with multiple-entry turbines. Two types of multiple-entry turbines are currently available in the market: meridionally divided “twin-entry” turbines and circumferentially divided “double-entry” turbines (Figure 1). The twin-entry turbine is divided meridionally and each incoming flow is fed into the entire rotor circumference. In contrast, double-entry turbine feeding area is divided circumferentially and incoming flows are guided with radially doubled scrolls.

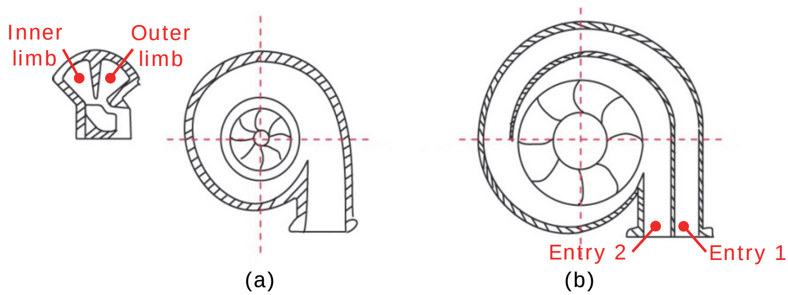

Figure 1: Comparison between multiple-entry turbine designs (1): (a) Twin-entry and (b) Double-entry turbine design

In order to merge the requirements of imbalanced mass flow and the pulse turbocharging advantage, asymmetric vaneless twin-entry turbine design has been proposed by Müller et al. (2) and lately optimized by Brinkert et al. (3). Their work showed that it is possible to achieve remarkable EGR-rates in some regions of the engine map, even though the average exhausts back-pressure is lower than the charge air pressure thus limiting the operations for this design. A different approach than that provided by (2) and (3) is proposed in the current paper, with the design of an asymmetric variable geometry double-entry turbine. The reasons for the selection of this turbine configuration are explained as follows.

In order to extract exhaust gases and recirculate it into the intake side, exhaust manifold pressure has to be higher than the intake manifold pressure. Therefore a VGT is necessary to increase exhaust manifold pressure corresponding with widely changeable transient operation. Besides supporting EGR control strategies by changing exhaust manifold pressure, VGT also offers the additional advantage of varying the inlet area to the turbine wheel thus maximizing exhausts flow energy extraction at different engine operating conditions. The choice of asymmetric double-entry geometry in place of a twin-entry one can be explained by considering that adopting symmetrically divided multiple-entry turbines in EGR engines, would lead to an imbalance of mass flow caused by EGR extraction from one side of the exhausts manifold. This flow imbalance can hardly be controlled in twin-entry turbines since the flow leaving the two entries mix together before entering the wheel. This is not the case in double-entry turbine configuration since the flows from the two entries are completely isolated and introduced into the turbine wheel separately, so that the flow controllability is believed to be more effective than in twin-entry turbine. A typical arrangement for the asymmetric variable geometry double-entry turbine is given in Figure 2. More details are provided in the next paragraph.

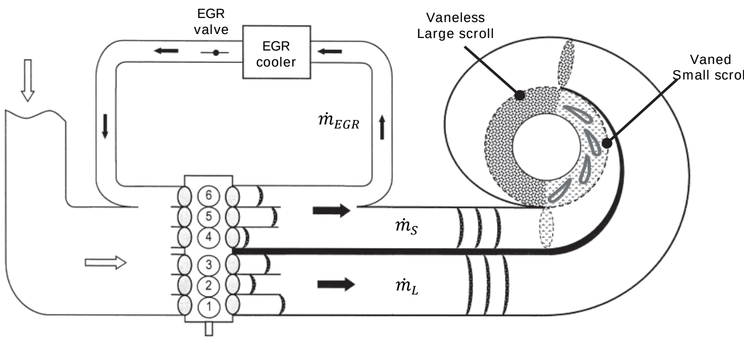

Figure 2: The system of the Asymmetric double-entry turbine with a 6-cylinders engine

2 PRELIMINARY ANALYSIS

The asymmetric double-entry turbine design started with preliminary off-design performance analysis. The two turbine inlets were treated as two independent turbine scrolls. Variable geometry vanes were only contemplated at the end of the scroll in which exhausts are extracted (small scroll in Figure 2) whereas in the other scroll no vanes were included. The main purpose of this partial vane arrangement is to minimise the losses due to the presence of vanes (e.g. vane pressure loss and vane clearance leakage loss) but still supporting EGR operations. With the asymmetric variable geometry double-entry turbine housing, the turbine wheel will mainly be driven by the large scroll flow which is optimised for higher efficiency over wide turbine operating conditions (no vanes, less losses). In the small scroll instead, the presence of variable geometry vanes is made necessary in order to support EGR operations. As the EGR rate varies (hence $\dot{m}_S$ varies) the variable geometry vanes will optimize the inlet flow area to the turbine wheel (i.e. the inlet flow angle) in order to maximize the exhausts flow energy extraction. In order to calculate the asymmetric double-entry turbine performance, the efficiency of the small and large scroll turbines were individually calculated using a mean-line model developed at Imperial College (4). Since the impact of EGR in the large scroll is negligible, the performance of the large vaneless scroll was calculated as conventional fixed geometry turbine. In contrast, the small vaned scroll was treated as single-entry VGT. The overall efficiency of the asymmetric double-entry turbine was then calculated as mass flow weighted average of the efficiencies of each turbine scroll.

$$\boldsymbol{\eta_D} = \frac{\dot{\boldsymbol{m}}_S \boldsymbol{\eta_S} + \dot{\boldsymbol{m}}_L \boldsymbol{\eta_L}}{\dot{\boldsymbol{m}}_S + \dot{\boldsymbol{m}}_L} \qquad (1)$$

where, η_S and η_L are individually calculated turbine efficiency of small and large scroll.

In order to choose the optimum circumferential division[1], the efficiency of several configurations were assessed and compared with that of an equivalent single-entry VGT with same turbine wheel. The analysis showed that the optimum circumferential division varies depending on different EGR rates and the final choice fell on a 160:200 arrangement (refer to Figure 4). The mean-line method results showed that it is possible to achieve better turbine efficiency than conventional VGTs when EGR is being operated widely.

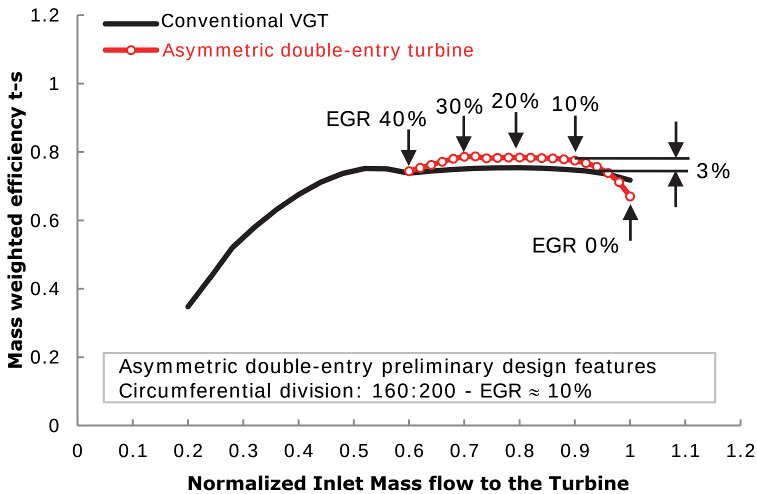

Figure 3: The turbine efficiency advantage of asymmetric double-entry turbine against conventional VGT when EGR is being operated widely
(The mass flow was normalized with the design-point mass flow rate of the single-entry VGT used for comparison)

This is shown in Figure 3 where for the selected turbine division, an improvement of ≈3% in efficiency could be found for a wide range of EGR rates (from 0% to 40%). This can be achieved by varying the vanes angle in the small scroll in order to match the best flow angle at the inlet to the turbine wheel and hence maximize exhausts flow energy extraction. After a preliminary analysis run on a typical duty cycle of an off-road engine, an EGR rate of 10% was chosen as initial design value (refer to the text box in Figure 3).

3 ASYMMETRIC DOUBLE-ENTRY DESIGN

The design of the asymmetric variable geometry turbine scroll started with the selection of the EGR rate and the circumferential division to be considered at the design-point (10% and 160:200 respectively as described in the previous paragraph). Once these two parameters were fixed, the design procedure revolved around the assumption that the incidence angle at the end of small scroll should be identical to that at the end of large scroll.

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[1] In an asymmetric double-entry turbine, there are same radii and widths of the flow path but different two circumferential flow areas for small and large scrolls (Figure 4).

From the free vortex correlation, the relationship between the centroid radius of the area and flow velocity is:

$$\boldsymbol{R_1 \cdot c_{\theta 1} = R_4 \cdot c_{\theta 4}} \tag{2}$$

The continuity equation for incompressible flow provides the relationship between radial flow velocities at the inlet and exit to the volute (station 1 and 4 respectively);

$$\boldsymbol{\rho_1 \cdot A_1 \cdot c_{\theta 1} = \rho_4 \cdot R_4 \theta b_4 \cdot c_{r4}} \tag{3}$$

By substituting equation (2) and (3) into the following absolute turbine inlet angle,

$$\boldsymbol{\cot \alpha = \frac{c_{r\psi}}{c_{\theta\psi}}} \tag{4}$$

We obtain,

$$\boldsymbol{cot\, \alpha_4 = \frac{\rho_1 \cdot A_1 \cdot \left(\frac{R_4}{R_1} c_{\theta 4}\right)}{\rho_4 \cdot R_4 \theta b_4 \cdot c_{\theta 4}}} \tag{5}$$

For incompressible flow, the equation (5) can be simplified,

$$\boldsymbol{cot\, \alpha_4 = (A_1/R_1) \cdot \frac{1}{\theta b_4}} \tag{6}$$

In case of single-entry turbine, $\theta = 2\pi$ since all the incoming flow is distributed around the circumference of the turbine wheel, whereas in a double-entry design turbine the inlet angles should be considered separately and identical to each other, ($\theta_S + \theta_L = 2\pi$).

$$\boldsymbol{cot\, \alpha_4 = (A_1/R_1)_S \cdot \frac{1}{\theta_S b_4} = (A_1/R_1)_L \cdot \frac{1}{\theta_L b_4}} \tag{7}$$

$$\boldsymbol{\frac{(A_1/R_1)_S}{(A_1/R_1)_L} = \frac{\theta_S}{\theta_L}} \tag{8}$$

From equation (8), it can be gathered that for a set EGR rate of 10%, the A/R ratio between the large and small scrolls can be expressed as in equation (9).

$$\boldsymbol{\frac{(A_1/R_1)_S}{(A_1/R_1)_L} = \frac{\theta_S}{\theta_L} = \frac{40\%}{50\%}} \tag{9}$$

Then it is now possible to separate circumference into two, and set the ratio between the two A/Rs identical to circumferential division.

In order to perform a comparison with a variable geometry single-entry turbine, a turbine wheel designed and tested at Imperial College by Abidat (5) (and lately used by Rajoo (6) for single-entry VGT design) was chosen. In order to run a one-to-one comparison, the sum of the small and large volute A/R values and their ratio against azimuth angle (Figure 4) were designed to be identical to that of the single-entry VGT designed by Rajoo (6). In other words, despite the asymmetric double-entry turbine volute is divided into two, the A/R values were set to obey to the free vortex condition following the single-entry VGT designed at Imperial College. The key features for the VGT asymmetric double-entry turbine are shown in Table 1 and Table 2.

Table 1: Design conditions

Parameter		
Pressure ratio	2.91	
Inlet temperature	344	K
Mass flow rate	0.678	kg/s
Rotational speed	60000	rpm
Target EGR	≈10	%

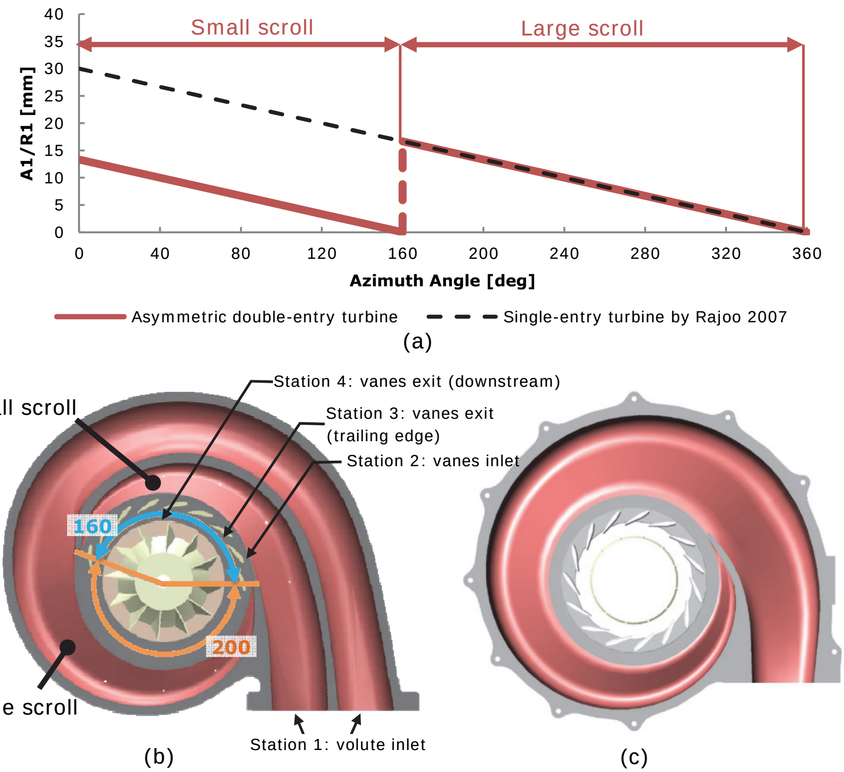

Figure 4: Asymmetric double-entry turbine volutes A/R design
(a) Comparison of A/R change along the volutes from its tongue between asymmetric double-entry turbine and single-entry turbine
(b) Asymmetirc double-entry turbine layout
(c) Single-entry turbine layout for comparison

Table 2: Geometric details of asymmetric double-entry turbine

Geometric feature			
Asymmetric circumferential division		160 : 200	
A/R	Small scroll	13.33	mm
	Large scroll	16.67	mm
Radius	Tongues	70	mm
	Turbine wheel (reference diameter)	42.07	mm
Number of blades		12	
Number of vanes		9	
Vanes angle (standard vanes angle)		67.65	°
Vane pitch angle		20	°

Unlike single-entry VGTs in which the vanes are arranged around the entire turbine wheel circumference, in a double-entry turbine configuration, the number of vanes is strictly related to the location of the two tongues. If the vanes are equally spaced around the periphery of the turbine wheel, the angle between two vanes is required to be a common divisor of the asymmetric circumferential division angles in order to have identical distance from tongue to vane. In the 160:200 circumferential division, the angle between two vanes should be a common divisor between 200° and 160° (i.e. 40°, 20° and 10°). In addition to this, the vane pitch is also dependent on the exit flow angle and the Zweifel's criterion (7). The former is given as function of vane geometries (8),

$$\boldsymbol{\alpha_4 = \cos^{-1}(o_{nv}/s_{nv})} \tag{10}$$

whereas the latter is a function of the optimum tangential lift coefficient ζ which in the Zweifel's criterion (7) is suggested to fall between 0.75 ~ 0.85,

$$\boldsymbol{\zeta = 2\left({}^{s_{nv}}/_{l_{nv}}\right) cos^2\alpha_3 |tan\alpha_2 - tan\alpha_3|} \tag{11}$$

Unlike the single-entry turbine housing designed by Rajoo (6), the vanes shape was changed from straight to curve in order to maintain the vane angle from the leading edge to the trailing edge and therefore obtain the same flow angle (in station 4) as that in the vaneless section.

4 CFD ANALYSIS

4.1 Computational analysis and discussion

In order to understand the turbine basic behaviour of the asymmetric double-entry turbine, the simulation results have been obtained with CFD analysis. The CFD analysis was conducted using commercial software ANSYS-CFX and in Table 3 it is provided the mesh characteristics of the whole turbine domain.

Table 3: Mesh characteristics of turbine domain

Region	Element type	Number of Elements	Number of Nodes
Volute	Unstructured (Tetrahedra, Wedges)	575,323	120,868
Vane section	Structured (Hexahedral) and Unstructured (Tetrahedra, Wedges)	78,180	75,684
Rotor domain	Structured (Hexahedral)	1,246,416	1,372,956
Exit ducting	Unstructured (Tetrahedra, Wedges)	39,045	8,121
Total		1,938,964	1,577,629

The turbine performance analysis was run under equal (same pressure ratio within the turbine inlets) and unequal admission (unbalance of pressure ratio between the inlets) conditions. For ease of discussion a parameter, here defined Scroll Pressure Ratio (SPR), has been introduced (equation 12) to provide the rate of imbalance between the small and large scroll;

$$\boldsymbol{SPR = \frac{\Pi_S}{\Pi_L} = \frac{P_{S0}}{P_{L0}}} \tag{12}$$

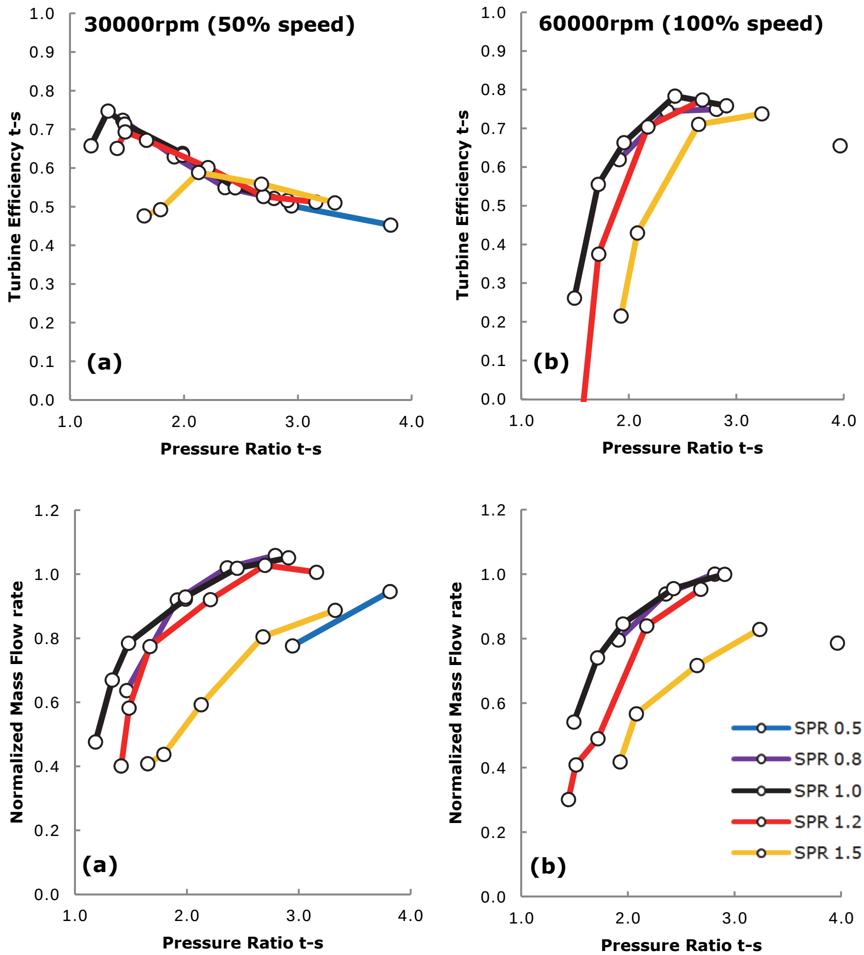

Figure 5: Turbine performance of the Asymmetric double entry turbine at (a) 50% speed and standard vane position, (b) 100% speed and standard vane position
(The mass flow was normalized with the design-point mass flow rate of the single-entry VGT used for comparison)

The analysis started by setting the vane angle at design-point (refer to Table 2), considering tow rotational speeds of 60000rpm (100% design-speed) and 30000rpm (50% design-speed, in which it is believed large amount of EGR is likely to be required), and five different values for the SPR. The simulation results are given in Figure 5 and show that the highest turbine efficiency occurs under equal admission conditions with an efficiency value of 76% at 1.3 pressure ratio for 50% speed and 78% at 2.43 pressure ratio at 100% speed. This is consistent with the initial design assumption of peak efficiency point occurring at 2.91 pressure ratio (5) (6) (refer to Table 1) and also with previous available literature showing that in multiple-entry turbines the peak efficiency point occurs under equal admission (9) (10) (11). As the rate of imbalance between the two inlets increases, the turbine performance decreases significantly, independent of which limb is flowing less mass flow (either SPR 0.5 or 1.5 present large efficiency drop).

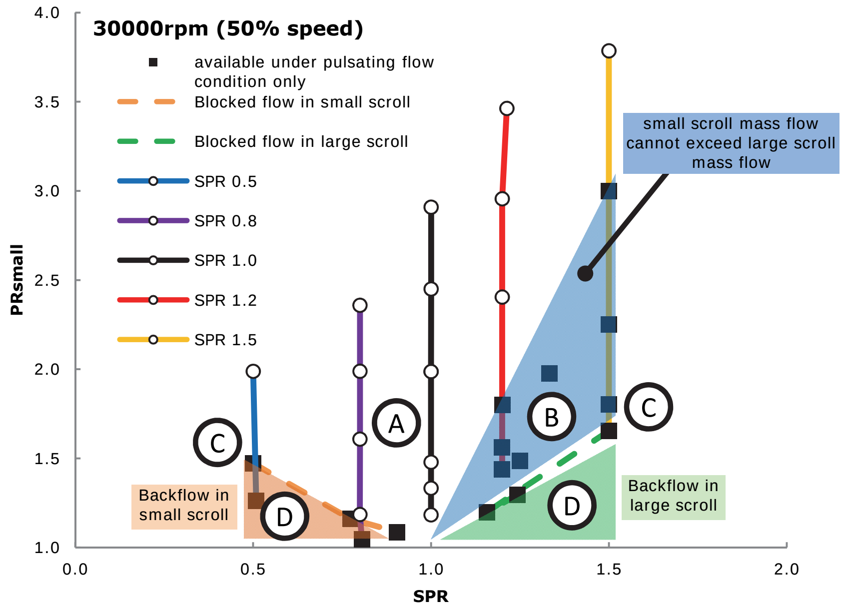

Figure 6: Small turbine volute flow interaction:
Ⓐ: Steady-state operation, Ⓑ: $\dot{m}_S > \dot{m}_L$, Ⓒ: $\dot{m}_S = 0$ or $\dot{m}_L = 0$,
Ⓓ: $\dot{m}_S < 0$ or $\dot{m}_L < 0$

In order to understand the limit of mass flows through which the asymmetric double-entry turbine can operate, typical mass flow patterns are illustrated in Figure 6 for 50% turbine speed. In the figure have been identified four flow areas, from A to B, trying to include also the effects of pulsating exhausts flow conditions. Starting with steady-state flow assumption, it can be noticed that in contrast with standard flow type A, the mass flow in the small scroll can never exceed that in the large scroll due to the cylinder distribution (3 cylinders connected to the small volute and EGR circuit, and other 3 cylinders connected to the large scroll: $\dot{m}_L = \dot{m}_S + \dot{m}_{EGR}$). Hence the flow type B (blue shaded area in Figure 6) cannot be obtained in the steady-state condition of asymmetric double-entry turbine design. As the pressure in the small scroll keeps decreasing as consequence of larger and larger EGR rates, the mass flow in the small scroll would experience a blocked flow condition (dashed line, flow type C) and in an extreme case some backflow could occur (flow type D). In the large scroll instead the mass flow can never be exceeded by that in the small scroll since in the large scroll there is not EGR flow extraction. Hence flow type B, flow type C and flow type D in the large scroll can only be obtained in an experimental lab set-up but it would not occur during standard engine operating conditions since there is no EGR flow extraction. However, it is worth noting that the results of Figure 6 are true only under the assumption of steady-state flow. This is not the case in a real engine since the exhausts flow are instantaneously pulsating. Therefore over an entire engine cycle, due to the firing order of the engine, the mass flow in large scroll can be exceeded by that in the small scroll as shown from the black marks in Figure 6. However this will not be discussed further within this paper which is currently looking at steady-state efficiency assessment for the asymmetric double-entry turbine design.

4.2 Turbine performance and optimisation with vanes angle

From the previous analysis it was found that the highest efficiency point occurs under equal admission conditions. If the vanes angles were not optimized to achieve equal admission condition, the turbine efficiency would drop due to the imbalance of mass flow between the two inlets. By changing the vanes angle when target EGR rate is changed, it is possible to keep higher turbine efficiency. In other words, the optimum EGR rate can be adjusted by varying the vanes angle position.

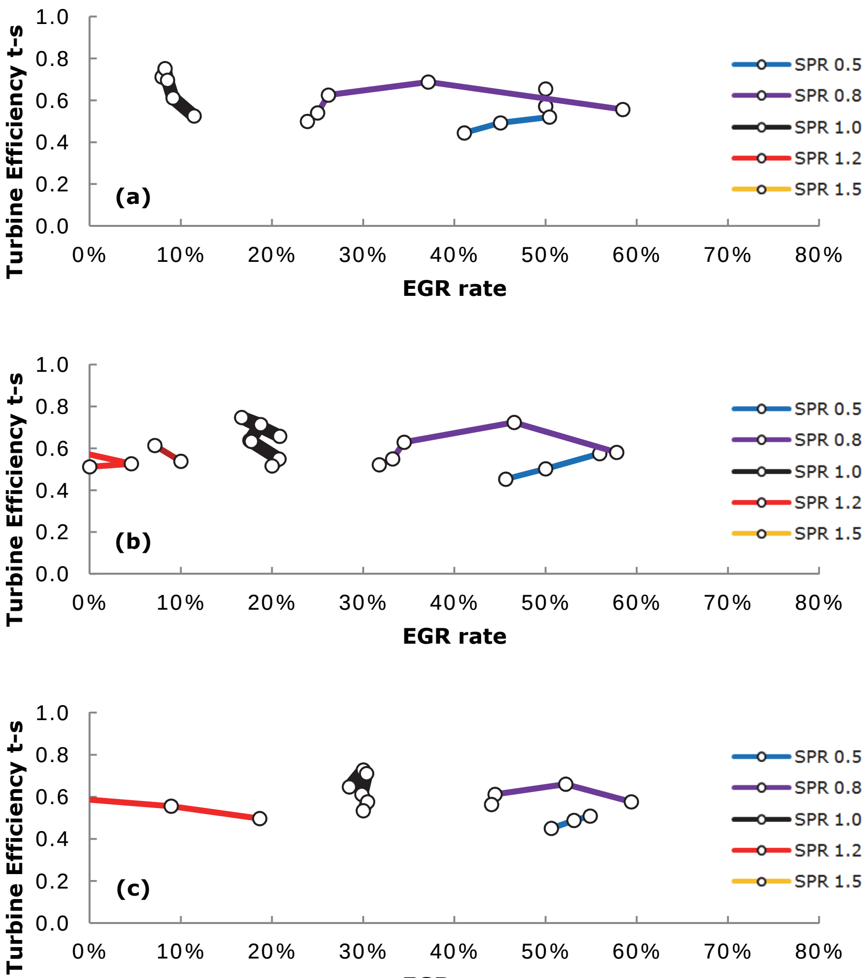

Figure 7: Comparison of optimum EGR rate:
(a) at 50% speed and -10° of vanes angle
(b) at 50% speed and standard vanes angle (design-point)
(c) at 50% speed and +10° of vanes angle

In Figure 7 it is reported a comparison between efficiencies at 50% speed for different vanes angle position, EGR rates and SPR values. EGR rate is expressed as in equation 13.

$$EGR = \frac{\dot{m}_{EGR}}{\dot{m}_L + \dot{m}_S + \dot{m}_{EGR}} = \frac{\dot{m}_L - \dot{m}_S}{2 \times \dot{m}_L} \tag{13}$$

Two additional vanes position (±10° of the design-point vanes angle) were considered in the analysis, as shown in Figures 7a and 7c. At design-point vane position (Figure 7b), the optimum EGR rate is approximately 20%[2] as shown by the black bold line. As the SPR decreases slightly (SPR 0.8) in order to maintain the same level of efficiency within the turbine, the EGR rate should increase significantly (more than 30% EGR). In some engine operating conditions, large EGR rate is not required, and therefore in order to maintain optimum turbine flow conditions (corresponding to equal admission conditions), the vanes angle need to be varied. By slightly varying the vanes position (±10° of the design-point vanes angle), an equal admission condition can be obtained for no excessive EGR rate values (10% and 30% for -10° and +10° vane position respectively). It is obvious that the equal admission condition line, which gets always maximum efficiency, is totally depending on the variable geometry vanes angle in Figure 7. However, it is notable that the equal admission condition is not always available since the intake manifold pressure and the exhaust manifold pressure are always changing on engine operation map.

5 CONCLUSION

In this paper, the design and performance assessment of a novel asymmetric double-entry variable geometry turbine was discussed. This novel turbine housing design was conceived for improving engine EGR operation and enhancing exhausts pulsating flow energy extraction in large multi-cylinder engines. The design started with preliminary mean-line analysis in order to fix the basic turbine geometrical parameters. Then the complete turbine design moved into 3-D modelling and CFD performance assessment. As a result of this process, the final turbine housing arrangement comes as asymmetric double-entry with circumferential division of 160:200 for 10% EGR rate (design-point) and variable geometry vanes only in the turbine inlet where EGR flow extraction occurs.

Simulations were run for a number of different turbine speeds (100% and 50% design-speeds), SPRs (from 0.5 to 1.5) and vanes angles (-10°, ±0°, and +10°). The simulation results showed that the peak efficiency point occurs under equal admission (SPR 1.0) with 78% turbine efficiency at 100% design speed and 20% EGR rate. Thanks to vanes position adjustment no much penalty in efficiency was observed at 50% speed, with peak efficiency value of 76%. The benefit of variable vane configuration could be appreciated for different EGR rates where the possibility to optimize the flow condition at the inlet to the turbine wheel showed that it is possible to retain an equal admission conditions (and hence optimum turbine efficiency) for a wide range of EGR rates.

As final remark about this project, it is worth saying that a prototype of the asymmetric variable geometry double-entry turbine will soon be tested at Imperial College London. The test programme will focus on the validation of the presented CFD results as well as in pulsating flow performance assessment.

[2] Despite 10% target EGR rate the optimum EGR rate calculated by CFD was 20%. This may require some further investigation and analysis for the vanes design and profile.

REFERENCE LIST

1. Watson, N. and Janota M. S., 1982, “Turbocharging the Internal Combustion Engine”, pp. 60.
2. Müller, M. et al., 2011, “The Asymmetric Twin Scroll Turbine for Exhaust Gas”, Proceedings of ASME Turbo Expo 2008, GT2008-50614.
3. Brinkert, N. et al., 2011, “Understanding the Twin Scroll Turbine-Flow Similarity”, Proceedings of ASME Turbo Expo 2011, GT2011-46820.
4. Romagnoli, A. and Martinez-Botas, R. F., 2011, “Performance Prediction of a Nozzled and Nozzleless Mixed-Flow Turbine in Steady Conditions”, International Journal of Mechanical Sciences, Vol. 53, pp. 557-574.
5. Abidat, M., 1991, “Design and Testing of a Highly Loaded Mixed Flow Turbine”, Ph.D. Thesis, Imperial College of Science, Technology and Medicine.
6. Rajoo, S., 2007, “Steady and Pulsating Performance of a Variable Geometry Mixed Flow Turbocharger Turbine”, Ph.D. Thesis, Imperial College of Science, Technology and Medicine.
7. Hiett, G. F. and Johnston, I.H, 1963, “Experiments Concerning the Aerodynamic Performance of Inward Flow Radial Turbine”, Proceedings of Institution of Mechanical Engineers, 178, Pt 31(II).
8. Japikse, D. and Baines, N., 1984, “Introduction to Turbomachinery”, pp. 6-17.
9. Copeland, C. D., Martinez-Botas, R. F. and Seiler, M., 2008, “Unsteady Performance of a Double Entry Turbocharger Turbine with a Comparison to Steady Flow Conditions”, Proceedings of ASME Turbo Expo 2008, GT2008-50827.
10. Copeland, C. D., Martinez-Botas, R. F. and Seiler, M., 2011, “Comparison Between Steady and Unsteady Double-Entry Turbine Performance Using the Quasi-Steady Assumption”, ASME Journal of Turbomachinery, Vol. 133, 031001.
11. Romagnoli, A. et al., 2012, “Comparison between the Steady Performance of Double-Entry and Twin-Entry Turbocharger Turbines”, ASME Journal of Turbomachinery.