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Experimental Investigation and Numerical Simulation of Gas Flow Through Wastegated Turbine of Gasoline Turbocharger

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ABSTRACT

Turbochargers are generally used in the combustion engine due to their capability to increase the specific power. This paper investigates the performance of the turbocharger turbine, which is mounted in a gasoline engine. Different working points, including close and open wastegate positions, are studied in a steady-state condition. The experimental test of this article has been performed on an engine test cell. The movement of the wastegate linkage in different working conditions of the engine was measured in the test cell. Furthermore, the static pressure was measured at the different positions of the turbine housing. Simulation results show that as the wastegate starts to open, maximum loading happens. However, increasing the waste-gate opening angle will decrease the force, which is caused by the passing gas. It was found that at 2 degrees opening angle of the wastegate, there is an 80 kPa pressure difference between two sides of the wastegate valve. When the wastegate has a small opening angle, the pressure distribution on the flat surface of the valve is not symmetric, which means the gas does not provide a 360° flow distribution around the valve. Moreover, streamlines show that the high-speed flow passing bypass passage disturbs the flow exiting the turbine blades. Results show that the opening of the wastegate flap can reduce the turbine's power. The 6 degrees opening of the wastegate causes a 30 percent power reduction. Moreover, the simulation results of the turbine's map show that at the constant mass flow rate, the opening of the wastegate from the closed position to 6 degrees causes the pressure ratio to decrease 26 percent.

Keywords: Gasoline engine; Pressure distribution; Turbine housing; Turbocharger; Wastegate.

NOMENCLATURE

h_0	total Enthalpy
$\dot{m}$	mass flow rate
P	power
S	sorce term
T	blade Torque
V	velocity vector
Y_plus	dimensionless wall distance
ρ	density
τ	shear stress tensor
ω	angular velocity

1. INTRODUCTION

The global trend of stringent emissions regulations makes automotive manufactures use different technologies that reduce pollution and fuel consumption (Gao *et al.* 2016). Turbocharging is an important technology for the downsizing of engines, which helps to lower emission and fuel consumption. The turbocharger improves the combustion efficiency of the engine by increasing the mass of the inlet air and proportionately higher fuel that increases the engine specific power. The exhaust gas turbochargers are using pre-compressed air and utilizing exhaust gas energy. The turbocharger consists of a compressor and a turbine. The gas expands in the turbine and generates the kinetic energy, which drives the shaft and compresses the air using the compressor. The turbine housing collects the exhaust gas and directs it to the turbine rotor. The volute is a critical component since it must set the correct inlet flow conditions to the impeller to transfer work efficiently. The turbine power is a function of the expansion ratio from the engine exhaust manifold pressure to the turbine exit, the turbine inlet temperature, and the efficiency (Baines 2005). The maximum temperature of exhaust gases in gasoline engines is higher than diesel engines. Experimental results show that the temperature of exhaust gas at the turbine inlet exceeds 900°C, which causes glowing exhaust manifold and turbine housing on the engine test cell (Alaviyoun and Ziabasharhagh 2020).

The wastegate valve, as a controller of the boost pressure, plays a crucial role in the performance of the turbine. In wastegated turbochargers, the amount of opening of the wastegate is controlled either pneumatically or electronically. In the lower speed range as well as under part-load operation, whenever the wastegate is closed, the turbine can be defined entirely by its characteristic map (Hiereth and Prenninger 2007). However, when the wastegate is opened, the exhaust gas is passed through the wastegate, and the swallowing capacity of the turbine is increased.

The exhaust gas temperature decreases considerably in the turbine. Therefore, during the warm-up time, a strategy can be used to control the opening of the wastegate bypass valve. The valve opening in the cold start condition will pass higher exhaust gases through the catalytic converter. Therefore, the efficiency of the catalyst increases rapidly, and the engine emission decreases during the cold start (Gao. *et al.* 2019a,b).

Numerous studies were focused mainly on the turbocharger's turbine using different methods, steady performance measurement to transient flow field measurement, 0-D to 3-D simulations, single passage to full blade modeling, single entry to twin entry turbine, wastegate to the variable geometry turbine. Salehi *et al.* (2013) offered a zero-dimensional model for the gas flow passing through the turbine housing and the wastegate passage. The turbine was model as an isentropic nozzle and the wastegate was modeled as an orifice. Fogarty (2013) experimentally investigated three different wastegated turbochargers of varying turbine/wastegate combination. Steady flow experiments were performed on a flow bench and cold-flow turbocharger test stand. Finally, a semi-empirical physical model has been developed in this study that is useful for 1-D engine simulation codes. In the simple one-dimensional model, energy losses have not been taken precisely. However, useful information has been accumulated in terms of empirical loss coefficients that are modeled in the one-dimensional analysis. The coefficients are empirical, which should not be applied to all classes of the turbine with any expectation of high accuracy. However, the data can be applied to develop similar types and sizes of turbines (Watson and Janota 1982). Some of the recent works have been focused on Map-based turbocharger models which are popular in one-dimensional engine modeling. Serrano *et al.* (2017) worked on a method for characterizing the discharge coefficient of a wastegate valve. The one-dimensional gas model with empirical data is correlated and validated. Finally, the map of the discharge coefficient has been drawn. One dimensional modeling often provides good results and acceptable predictions, but many of the fluid flow details like flow, pressure, and temperature distribution cannot be predicted. Also, this approach is unable to investigate the flow inside the wastegate passage and its effect on the main flow.

The fluid flow in the turbocharger is highly turbulent, and some complex flow phenomenon like backflow and vortices frequently occurs in the housing and around the wheel. Therefore, computational fluid dynamics methods were used by Benajes *et al.* (2014) to analyze the flow field in the turbine. Hajilouy *et al.* (2009) performed a flow study of a twin-entry turbine. The absolute angle and velocity at the turbine exit are measured with a five-hole probe. Numerical results show that the flow is three-dimensional and complicated in the volute tongue, and due to distortion near this region, the lowest entropy gain factor is obtained. One of the most significant areas of entropy generation in the whole turbine was associated with the shear layer formed between the flowing and the non-flowing regions at the tongue (Newton *et al.* 2012).

Siggeirsson and Gunnarsson (2015) developed a conceptual method for 3D simulation and analyzing the fluid flow in radial turbines. Two methods have been developed for the simulations: Single Blade Passage (SBP) model as well as a more detailed Full Turbine (FT) model. In the SBP method, instead of considering the whole turbine, including the volute, rotor, and stator, only the flow inside one passage of the blade is modeled. A single passage model was used with a periodic interface to account for the domain motion by (Shah *et al.* 2016). On the other hand, in the FT approach, the whole turbine is modeled completely, and the flow inside the entire turbine is investigated (Siggeirsson and Gunnarsson 2015). Therefore, the FT model is a much more realistic model than the SBP model. Tabatabaei *et al.* (2012) simulated the 3D full turbine model of a gasoline turbocharger. Results showed that in both the steady and unsteady flows, by increasing engine speed, the simulation errors were reduced. The simulation was performed under a fully closed wastegate condition, and the effect of the wastegate channel on the flow pattern was not considered.

The gas flow inside the bypass passage of the turbine housing is complex that can affect the main flow inside the turbine. Depending on the amount of opening of the wastegate, the turbine performance and gas flow pattern can be affected. Wibmer *et al.* (2015) used optical measurement techniques for the video-imaging of the wastegate system. Transient simulation of a turbine housing showed that the flow of exhaust gas creates a transient pressure distribution on the wastegate flap and lever. CFD results showed that the gas flow velocity increases to almost 800 m/s at the wastegate channel as a result of the narrowing flow channel (Getzlaff *et al.*

2010). Research on the wastegated turbine shows that the more collinear the bypass flow reintroduction to the turbine bulk flow, the higher the wastegate-open turbine stage efficiency. Therefore, an optimum flow reintroduction port would be angled such that it will turn the flow to follow the main gas flow (Hasler 2018). However, there are limitations such as packaging and manufacturing that influence the position and the direction of the wastegate valve.

Marelli and Capobianco (2011) investigated the performance behavior and isentropic efficiency of a turbine in the open wastegate position in the turbocharger test rig. The results show that the opening of the wastegate valve gives rise to a significant reduction in turbine efficiency. The efficiency drop mainly happened as some of the gas flow will be bypassed through the waste-gate and will not produce any useful work output. The apparent efficiency of the turbine was found to decrease by up to 53.7% for the most open wastegate condition of an external wastegate (Salim 2014). However, an internal wastegate completely differs from the external one which is necessary to be considered in an engine operating condition.

A review of relevant works concludes that there are limited works on turbine performance using the CFD method. However, no CFD study exists addressing the influence of the wastegate opening on the performance of the turbine in the gasoline engine condition. The engine conditions in great detail will not be obtained from the turbocharge test rig. Therefore, the characteristics of the turbine must also account for the effect of engine conditions. This scenario serves as an innovative method for research carried out in this article. This paper presents an investigation of 3D fluid flow inside a turbine under steady conditions. The engine test experiments validated the simulation results. Most of the researches measures the gas mass flow rate for the validation. However, in this article, a new approach is used to validate simulation results with pressure distribution and mass flow measurements. The static pressure is measured at different radial locations on the turbine housing to validate the model. The non-uniformity of pressure around the casing is investigated. The flow field is analyzed through the turbine housing from the inlet to the outlet of the casing. The effect of an internal waste-gate valve on the characteristics of the system is also considered in the study. Therefore, a CFD calculation is carried out to compare the flow around the wastegate channel at open and close conditions. Finally, the aim of the present article sets out to study the effect of the internal waste-gate port as a potential source of disturbance to the gas flow.

2. EXPERIMENTAL FACILITY

The experimental task of this work has been performed on a standard engine test bench. Turbocharger performance investigation needs an accurate measurement across the turbocharger and adjacent parts in the engine. Temperature and pressure sensors are installed on the inlet and the outlet pipes of the compressor and the turbine. Also, several sensors were used in the test cell to measure fluids temperatures, pressure, and mass flow. For instance, the mass flow of the fuel is measured. Furthermore, the air-fuel ratio has been measured using the lambda sensor. The turbocharger shaft speed has been measured using an eddy current device. The position of turbocharger on the engine has defined packaging limits for pressure sensors installation. Therefore, three static pressure sensors have been mounted on the turbine housing of the turbocharger.

It is not possible to measure the pressure of the hot gas directly. Therefore, a long steel pipe is used to ensure that the pressure sensor does not face any problem in a hot area. Pressure sensors were measured in hot regions (exhaust manifold and turbine housing) according to the following procedure. First of all, the location of the sensor on the surface of the part is drilled (Fig. 1). Then a steel pipe (at least 35 cm length) is welded on the drilled position. Finally, a heat resistant tube is connected to the steel pipe and the pressure sensor. All the connections were checked to ensure that there is no leakage.

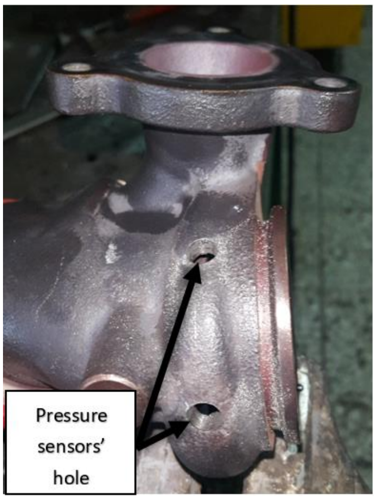


Fig. 1. Pressure sensors' position on the turbine housing.

The pressure sensors are circumferentially located at 100°, 170°, and 250° (Fig. 2). An experimental investigation has been performed by measuring and analyzing gas pressure at different points on turbine housing for different working speeds and full load conditions.

The wastegate arm opening in different working conditions of the engine was measured in the test cell. The high accuracy digital contact sensor was applied at the end of the wastegate rod to measure rod movement. Figure 3 shows the engine test bench that mounting one pressure sensor on the turbine inlet, one sensor on the turbine outlet, and three static pressure sensors on the turbine housing. Steady-state test conditions were investigated in this article. Different types of steady-state experiments have been performed at different working conditions. For instance, at engine speed 1500 rpm and the full load condition, the engine stays at least 5 minutes to stabilize the parameter. Then, the measurement of the parameter (like temperature and pressure) was started. Each test has been repeated three times to ensure the repeatability of the measurements.

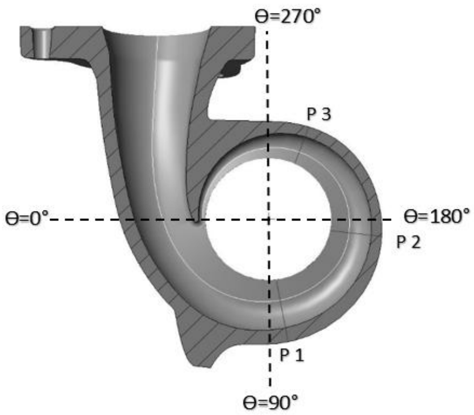


Fig. 2. Pressure sensors' angle in the section view of the turbine housing.

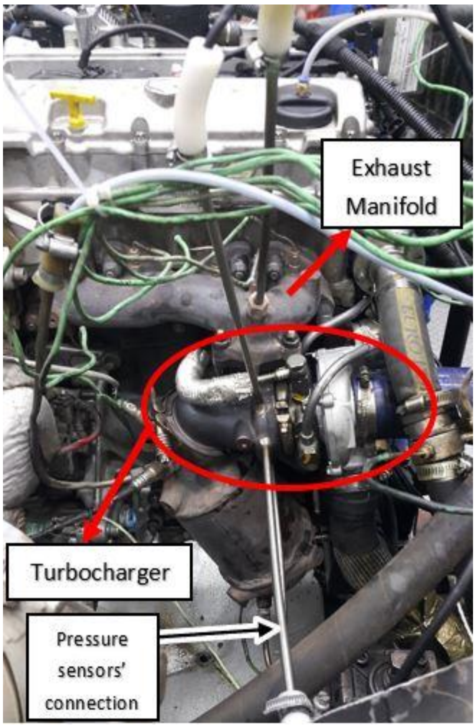


Fig. 3. Turbocharger with pressure sensors' connections in an engine test cell.

In the engine test cell, the ventilation fan is placed on the exhaust side of the engine to have the same ventilation condition as turbocharger faced in the vehicle. The air temperature is the same as the test cell temperature, and the air has the same speed as it has in the vehicle. Table 1 shows the main information about the measurement range of sensors used in the test bench. The accuracy of the temperature sensor was ±2°C, and the accuracy for the pressure sensor was ±2.5kPa. Moreover, the speed sensor and displacement sensor have 1rpm and 3μm accuracy respectively. The air to fuel ratio sensor (Lambda sensor) has ±0.1% accuracy at the stoichiometric point. The measurement precision of the fuel meter is less than 0.12%.

Table 1 Specifications of sensors

Sensor	Type	range
Temperature	Thermocouple K Type	-200°C to1200°C
Pressure	Piezoelectric	0 to 2.5bar
speed sensor	Magnetic	0 to 400000rpm
Displacement	Digital contact	0 to 32 mm
Air to fuel ratio analyzer	UEGO Sensor	0.27 to 30
Fuel flow meter	Coriolis Mass Flow sensor	0.1 to 110 kg/h

3. NUMERICAL METHOD

3.1 Turbine Geometry

The turbocharger which is investigated in this article has been installed on a 1.7-liter gasoline engine. Figure 4 shows the position of wastegate passage which bypasses part of the exhaust gas based on the position of the wastegate valve.

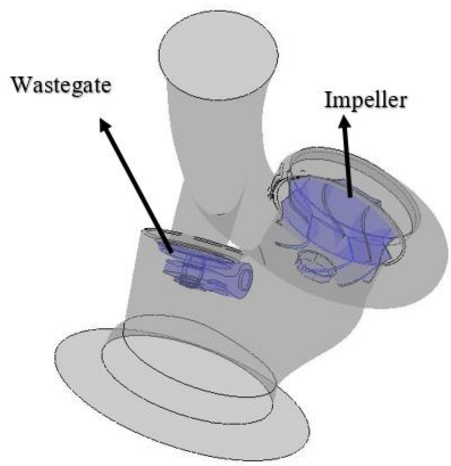


Fig. 4. Turbine housing, wastegate, and impeller model.

The turbocharger has a turbine with the specifications which are shown in Table 2. A/R describes the ratio between the area of the turbine and the radius R. The radius is the distance from the center of the impeller to a theoretical flow path by which the section is halved (Hiereth and Prenninger 2007).

Table 2 Specifications of the turbine

Geometrical feature	Value
Impeller Inducer Diameter	43.6 [mm]
Turbine housing A/R	0.5
Maximum Tip Speed	502 [m/s]
Turbine Inlet Diameter	39 mm
Turbine Outlet Diameter	60.64 mm
Wastegate Passage Diameter	20 mm
Wastegate valve diameter	27 mm
Wastegate type	Normally closed

The wastegate valve was normally closed and using a pressurized actuator system to open the wastegate.

3.2 Computational Domains

CFD simulations of radial turbines have been performed using several approaches for the boundary conditions. The first method applied the boundary conditions directly in the turbine housing's inlet and outlet surface (Shah *et al.* (2014)). Other researches imposed the boundary conditions on the turbine with long outlet ducts that ensure flow development (Capiluppi (2012)). Therefore, in this article, an outlet surface was extended ten times of impeller diameter to avoid reverse flow (Fig. 5).

Fig. 5. Turbine housing model with extended outlet.

The fluid domain of the turbine housing includes the volute, the diffuser, and the wheel. The fluid domain is constructed in two separate domains (rotating and stationary). The computational grid mainly consists of tetra mesh. Unstructured tetrahedral mesh with boundary layers is used (Fig. 6).

Multiple reference frame(MRF) is used as a method to simulate the rotating domain. This technique is also known as the frozen rotor model, which is a cost-efficient approach in analyzing rotating domains. It is a steady-state approximation that a coordinate system rotates with the rotating domain. The momentum equation of the stationary and rotating domains are shown respectively in Eqs. 1 and 2. The rotating reference frame has source terms of the Coriolis and centrifugal forces.

$$\frac{\partial \rho \vec{V}}{\partial t} + \nabla.(\rho \vec{V}\vec{V}) = -\nabla.p + \nabla.\tau + S \qquad (1)$$

$$\frac{\partial \rho \vec{V}_r}{\partial t} + \nabla.(\rho \vec{V}_r \vec{V}_r) = -\rho(2\vec{\Omega}\times\vec{V}_r + \vec{\Omega}\times\vec{\Omega}\times\vec{r})r$$
$$-\nabla.p + \nabla.\tau + S \qquad (2)$$

$\vec{V}$ is the absolute velocity, $\vec{V}_r$ is the velocity relative to the rotating frame, ρ is the fluid density, p is the pressure, τ is the shear stress, $\vec{\Omega}\times\vec{r}$ is the whirl velocity, and S refers to source terms.

The boundary conditions consist of total pressure and temperature at the inlet and static pressure at the outlet. These variables measured in an engine test cell are shown in two working points in Table 3. There is a relation between wastegate rod displacement and the wastegate flap angle. Results show that the wastegate is closed at 1500 rpm engine speed and full load condition (case 1), while the wastegate flap opening is 2 degrees at 2500 rpm and full load condition (case 2). The maximum opening of the wastegate flap is 7 degrees at 5000 rpm and full load condition. The maximum movement of wastegate in the engine operating condition is half of the allowable movement of the wastegate linkage.

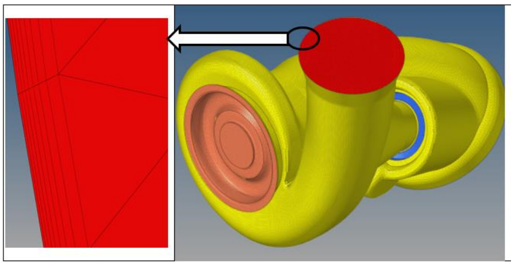


Fig. 6. Computational grids of the gas flow for turbine housing with a boundary layer on the turbine inlet surface.

Table 3 Boundary conditions

	Case 1	Case 2
Turbine rotation speed (rpm)	92000	142000
Inlet Total Pressure (kPa)	30.5	100.2
Inlet Total Temp. (K)	1087	1156
Outlet Static Pressure (kPa)	3.16	16.86

The commercial CFD code solved RANS equations. Simulations were carried out using the pressure-based solver and coupled algorithm. The gas is a burnt mixture of gasoline and air, which is defined by the polynomial approach for the thermal conductivity and also the specific heat capacity.

K-epsilon model was a reliable model for the prediction of turbine performance (Zhao *et al.* 2016). CFD analysis with Y-plus less than 10 on the rotor wall is an acceptable requirement for the turbine simulation (Cox 2015). However, in most of the literature, the K-ω SST model has been used. Galindo *et al.* (2013) described a CFD simulation method of a variable geometry turbine. The simulations were performed using a K-ω SST model for turbulence, maintaining a Y-plus equal to one. Therefore, in this research, the SST model has been used for the turbulence. The mesh has been refined until a dimensionless wall distance of Y-plus less than two has been realized.

3.3 Grid Independency

A grid study was performed to specify the grid size in which the solution is independent of the mesh size. The gas mass flow and turbine wheel torque are shown in Fig. 7. The turbine impeller torque is computed by integrating pressure and shear forces over the rotating blade surfaces relative to the position vector from the origin of the rotating reference. The blade torque based on CFD is computed from Eq. 3.

$$T = \frac{P}{\omega} = \frac{\dot{m}.\Delta h_0}{\omega} \qquad (3)$$

Where T is the torque, ω is the angular speed, P is the power, $\dot{m}$ is the mass flow rate across the turbine, and Δh_0 is the change of the total enthalpy through the turbine. Mesh independency study shows that the calculated gas mass flow and torque reach a constant value as the number of cells increases (Fig. 7). Finally, the grid with 2.3 million cells is considered to be sufficiently cost-effective and accurate.

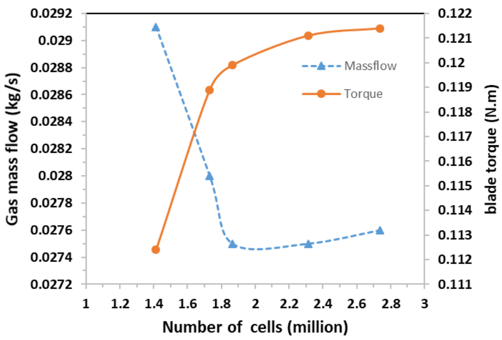


Fig. 7. Mesh independency study based on mass flow and impeller torque parameters.

4. RESULTS AND DISCUSSIONS

Experimental results show static pressure distribution on the turbine housing versus engine speed (Fig. 8). The pressure at the turbine inlet reaches 190 kPa at engine speed 4500 rpm and full load condition. As the engine speed increases, the pressure difference between sensors also increases. At engine speed 4500 rpm, pressure difference between turbine inlet and point P1 is equal to pressure decrement between P1 and P2. However, experimental results show that the pressure difference between P2 and P3 probes is small in the range of few Pascal.

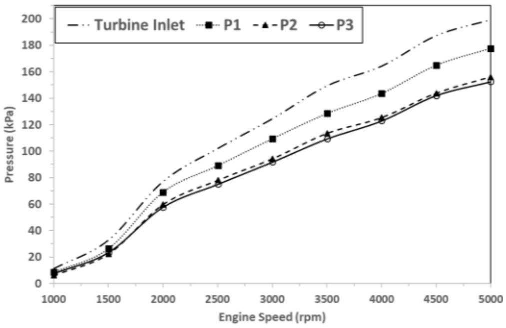


Fig. 8. Pressure distribution of different points on the turbine housing versus engine speed.

4.1 Validation of the Numerical Method

The model verification has been performed based on the engine test data. The comparison of the simulations results and experimental test results are shown in Table 4. Two working points (1500 rpm and 2500 rpm engine speed and full load condition) were chosen to represent the different performance of the turbine. The deviation of the calculated parameter from the measured results is shown in Table 4. The computational errors are limited to 5%, and the model agrees reasonably with the data measured from the engine test.

Table 4 Simulation and experimental results

	Measured	Simulation	Measured	Simulation
	Case 1		Case 2	
Pressure @ P1 (kPa)	26.23	25.07	89.19	85.43
P1 Prediction Error	4.4 %		4.2 %	
Pressure @ P3 (kPa)	23.41	22.53	75.12	79.28
P3 Prediction Error	3.7 %		5.5 %	
Mass Flow (kg/s)	2.82 e-2	2.76 e-2	5.812 e-2	6.044e-2
Mass Flow Error	2.1 %		3.3 %	

4.2 Flowfield Inside the Turbine Housing

The streamlines at 92000 rpm with the expansion ratio 1.3 is shown in Fig. 9. Cross-sectional velocity contours of the turbine housing are shown for a low

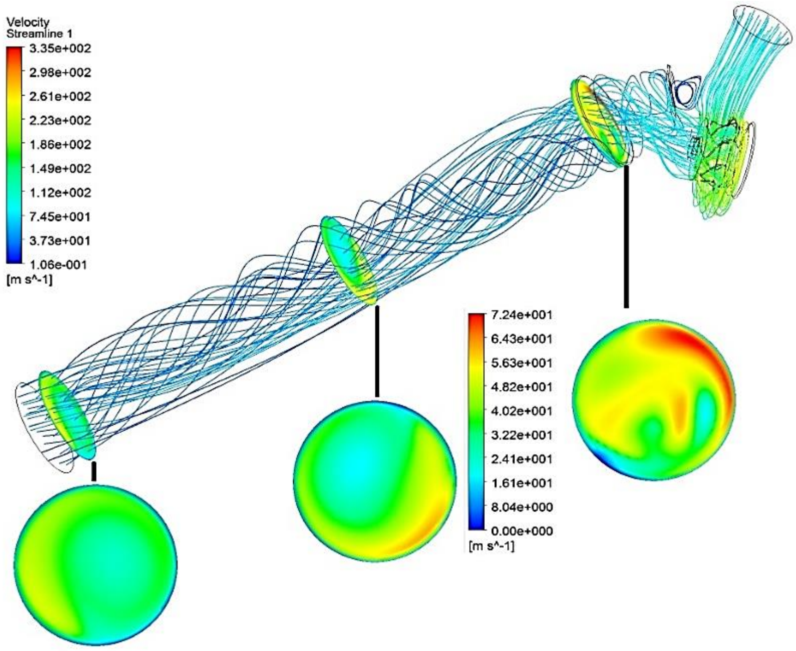


Fig. 9. Streamlines with the magnitude of the velocity and velocity contours at three locations at the turbine outlet.

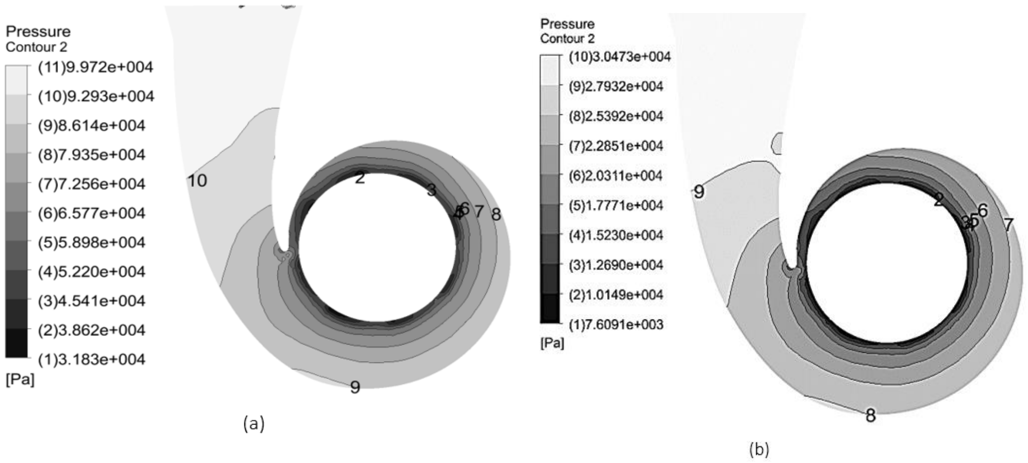


Fig. 10. **Contour of static pressure on the sectioned view of turbine housing at the rotational speed of 142000 rpm (case 2).**

mass flow rate. The streamlines show different vortices that are created downstream the wheel. The velocity of the flow after the turbine is slowed down and includes a swirl. A similar behavior appears for a case with the open wastegate. The gas flow downstream of a turbine is three dimensional, and the swirl is created after the impeller.

Figure 10 presents the results of a static pressure field in a section plane of the turbine housing for working points 1 and 2. The housing has the function of flow guidance at the impeller entrance and flow distribution. It is common to have some reduction in flow area between the inlet flange of the turbine and the tongue point (equal to $\Theta=0°$ in Fig. 2 where the scroll first admits to the turbine impeller). Therefore, the gas accelerates and the frictional losses increase due to high velocity passing flow.

4.3 Wastegate Passage

The turbocharger of passenger cars is small, and it is difficult to measure the wastegate flow inside turbine housing due to the internal position of the wastegate channel. Therefore, the bypass mass flow is calculated using numerical results. A parameter is defined as a bypass flow ratio, which is the wastegate flow divided by total turbine flow. Simulation results show that, as the engine speed increases from 2500 rpm to 5000 rpm, the bypass flow ratio increases from 16 to 32 percent.

The exhaust gas flow entering the housing would either stay on the volute or pass through the wastegate passage and exit the valve. Figure 11 shows the Mach number as the wastegate opens 2 degrees. The Mach number reaches 1.01. Therefore, the wastegate passage is choked, and the Mach number is equal (or higher) than 1 in part of the passage.

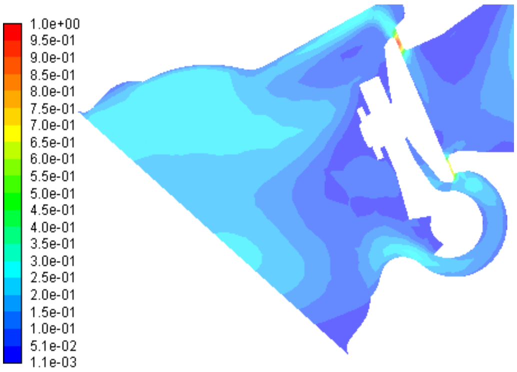


Fig. 11. Mach contour of open wastegate condition.

The pressure distribution in the narrow wastegate channel is shown in Fig. 12 at 2 degrees opening of flap.

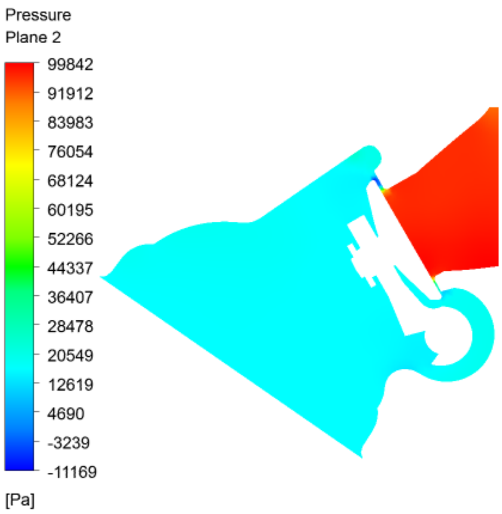


Fig. 12. Pressure contour of open wastegate condition.

As the gas flows inside the narrows passage from point A to point B (Fig. 13), the total pressure decreased. Then, the high-speed flow conflicts with the flow exiting the turbine blades, which causes stagnation at the junction where it is reintroduced into the main gas flow. Therefore, the total pressure increases to 20 kPa at point B. There is a high-pressure difference between two sides of the wastegate valve, which increases valve instabilities and makes it difficult to control the wastegate precisely. Simulation results of the torque due to the passing gas on the wastegate valve show that as the wastegate starts to open, maximum loading happens. Increasing the wastegate opening angle to 5 degrees will decrease the blade toque. The same trend is shown for the normally open wastegate (vacuum-actuated system) that the valve loading has a maximum torque when the valve starts to open (Dupuis *et al.* 2014).

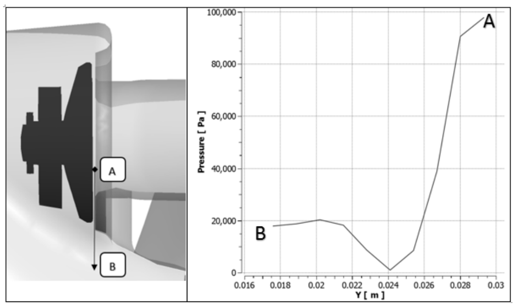


Fig. 13. Pressure distribution on the line AB in the wastegate channel at 2 degrees of opening angle.

The pressure distribution on the flat surface of the waste-gate flap is shown in Fig. 14. Results show that moving from the center of the valve to the outer side, the pressure decreases. As the wastegate opens 2 degrees, pressure on the bottom side of the circle is higher than the other sides. As the rotational axis is not on the center of the circular valve, the movement of flap around the axis of rotation causes the gas to pass the circumference of a circle non-symmetrically.

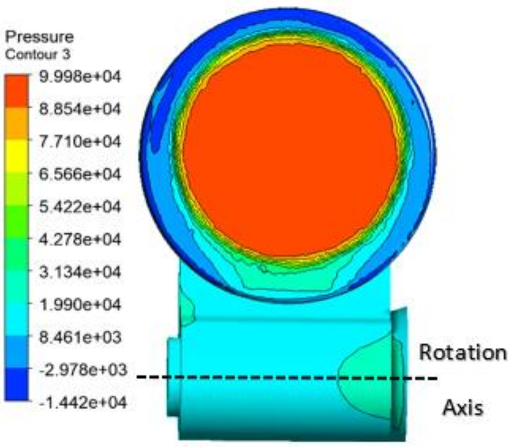
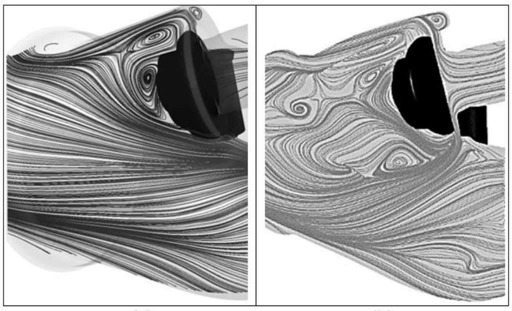


Fig. 14. Pressure distribution contour on the wastegate surface for 2 degrees opening of the wastegate.

Plane streamlines in the sectioned view of the turbine housing are shown at two different working conditions (Fig. 15). At engine speed 2500 rpm and full load condition, the wastegate opens 2 degrees, and the maximum velocity of the gas reaches 630 m/s. The streamlines on the left side of Fig. 15 shows that the flow distribution around the valve is not symmetrical. However, as the engine speed reaches 5000 rpm, the wastegate opens 7 degrees, and a 360° flow distribution around the valve happens.

A comparison of streamlines for two working points shows that further opening of the wastegate causes the bypass flow to disturbs the main flow more. The wake flow downstream of the wastegate, produces high losses, resulting in a pressure gradient and high turbine back-pressure. The former flow may block the later one which results in the turbine performance reduction. This issue is mainly due to the unparalleled direction of the wastegate passage and turbine blade centerline. The turbocharge that is investigated in this article has 25 degrees mixing angle between the bypass axis and the main fluid stream. The turbine design has the potential to decrease the mixing angle. The other factor that is important in the design of the wastegate is the bypass port position. The location of the wastegate port is acceptable, which the bypass passage branches off the straight port of the turbine. This type of the wastegate passage is favorable in comparison to the design that the bypass is located in the inside curve of the inlet port (Fogarty (2013)).



(a) (b)

Fig. 15. Fluid streamlines for 2 degrees opening of wastegate (a) and 7 degrees opening (b).

Table 5 Simulation results of the turbine's power at constant pressure ratio 3 and two different rotational speeds

	Rotational speed (rpm)	Turbine Power (kW) @		
		Closed position	2° opening	6° opening
1	100000	11	9.2	7.6
2	200000	15.9	12.8	10.4

In this article, the effect of the wastegate opening on the turbine performance was investigated under different wastegate positions. The turbine's power has a maximum value at a closed wastegate condition. As the rotational speed remained constant, the turbine's power increases linearly versus the pressure ratio. When the wastegate opens, the turbine power decreases, which the main reason is, exhaust gas bypasses the turbine and the overall enthalpy of the gas passing the turbine wheel is reduced. Results show that at maximum speed and constant pressure ratio 3, turbine power decreased from 15.9 kW to 10.4 kW as the wastegate opens 6 degrees (Table 5). Simulation results at lower rotational speed show the same trend; for instance, at 100,000 rpm speed and constant pressure ratio 3, the turbine's power decreased from 11 kW to 7.6 kW as the wastegate opens 6 degrees (Table 5).

Moreover, the simulation results of the turbine's characteristics map show that at constant mass flow rate 0.01 kg/s, the opening of the wastegate from the closed position to 6 degrees causes the pressure ratio to decrease from 1.71 to 1.26 (Table 6).

Table 6 Simulation results of the turbine's pressure ratio at two similar mass flow (kg*K^0.5/s*kPa)

	Similar Mass flow	Pressure Ratio @			
		Closed position	2° opening	4° opening	6° openi
1	0.005	1.19	1.15	1.13	1.12
2	0.01	1.71	1.42	1.32	1.26

5. CONCLUSION

The aim of the present article sets out to study and model the turbine of a gasoline engine turbocharger. The experimental results include pressure, temperature, and flow measurements in an engine test cell. The pressure distribution inside the turbine housing is investigated at different working conditions. The 3D numerical simulation of the turbine housing is validated by pressure and mass flow of exhaust gas. Moreover, this paper describes a CFD analysis of the turbine performance under open and closed wastegate positions. The conclusion can be stated as below:

1. The ratio of the wastegate flow to the turbine total flow is defined as a bypass flow ratio. Simulation results show that increasing engine speed from 2500 rpm to 5000 rpm causes the bypass flow ratio to increases from 16 to 32 percent. Therefore, at maximum speed and full load conditions, one-third of the gad flow is bypassed the turbine wheel.

2. The streamlines show that the unparalleled direction of the wastegate passage and turbine centerline causes the bypass flow to disturbs the main flow. The wake flow downstream of the wastegate produces high-pressure losses, which are not favorable. Therefore, the parameter that is necessary to be considered in the design of the turbine housing is the wastegate passage angle. It is recommended to consider the minimum angle between two passages, which can decrease the flow disturbance.

3. Simulation results show that at 2 degrees opening angle of the wastegate, there is an 80 kPa pressure difference between two sides of the valve. Hence, a significant loading caused by the passing gas which increases valve

instabilities and makes it difficult to control the valve precisely. In addition, the pressure distribution on the flat surface of the valve is not symmetric which means the gas does not provide a 360° flow distribution around the valve. The new geometrical design of the wastegate valve is one of the options that make a symmetrical flow distribution and decreases the load required to keep the valve in the position.

4. The opening of the wastegate valve reduces the turbine's power significantly. However, it is necessary to quantify the effect of the valve opening on the turbine performance. Simulation results show that 6 degrees opening of the valve will decrease turbine power around 30%. Finally, results show that at the higher mass flow rate, the opening of the valve has a significant effect on power reduction.

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