Numerical analysis of flow interaction of turbine system in two-stage turbocharger of internal combustion engine

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Abstract: To reach the goal of energy conservation and emission reduction, high intake pressure is needed to meet the demand of high power density and high EGR rate for internal combustion engine. Present power density of diesel engine has reached 90KW/L and intake pressure ratio needed is over 5. Two-stage turbocharging system is an effective way to realize high compression ratio. Because turbocharging system compression work derives from exhaust gas energy. Efficiency of exhaust gas energy influenced by design and matching of turbine system is important to performance of high supercharging engine. Conventional turbine system is assembled by single-stage turbocharger turbines and turbine matching is based on turbine MAP measured on test rig. Flow between turbine system is assumed uniform and value of outlet physical quantities of turbine are regarded as the same as ambient value. However, there are three-dimension flow field distortion and outlet physical quantities value change which will influence performance of turbine system as were demonstrated by some studies. For engine equipped with two-stage turbocharging system, optimization of turbine system design will increase efficiency of exhaust gas energy and thereby increase engine power density. However flow interaction of turbine system will change flow in turbine and influence turbine performance. To recognize the interaction characteristics between high pressure turbine and low pressure turbine, flow in turbine system is modeled and simulated numerically. The calculation results suggested that static pressure field at inlet to low pressure turbine increases back pressure of high pressure turbine, however efficiency of high pressure turbine changes little; distorted velocity field at outlet to high pressure turbine results in swirl at inlet to low pressure turbine. Clockwise swirl results in large negative angle of attack at inlet to rotor which causes flow loss in turbine impeller passages and decreases turbine efficiency. However negative angle of attack decreases when inlet swirl is anti-clockwise and efficiency of low pressure turbine can be increased by 3% compared to inlet condition of clockwise swirl. Consequently flow simulation and analysis are able to aid in figuring out interaction mechanism of turbine system and optimizing turbine system design.
1. Introduction
Energy crisis and CO2 emission problem demands engine raising power density to realize downsizing, and charging system is an effective way to raise power density [1-3]. Besides, high combustion temperature of charged internal combustion engine brings about significant formation of nitrogen oxides [4, 5]. So it is necessary to use high EGR rate to avoid the shortages above. High inlet pressure is needed to meet the demand of high power density and high EGR rate. Present power density of diesel engine has reached 90KW/L and intake pressure ratio needed is over 5[6]. Single-stage turbocharger can realize high pressure ratio, however it will result in flow range of turbocharger narrowed and reliability worsen. Therefore, how to raise engine performance in whole operational conditions becomes the key issue of high charging engine. Reaching the same total intake pressure ratio, pressure ratio of each turbocharger in two-stage turbocharger is lower than single-stage turbocharger. So two-stage turbocharger can broaden flow range and decrease rotation speed of turbocharger to ensure operational reliability.

High pressure ratio will result in compression work increased. When intake pressure ratio reaches 5, ratio of compression work to engine output power is close to 50% [7]. To turbocharging system compression work derives from exhaust gas energy. So efficiency of exhaust gas energy influenced by design and matching of turbine system is important to performance of high charging engine. Conventional turbine system is assembled by single-stage turbocharger turbines and turbine matching is based on turbine MAP measured on test rig. Flow between turbine system is assumed uniform and value of outlet physical quantities of turbine are regarded as the same as ambient value. However, there are three-dimension flow field distortion and outlet physical quantities value change which will influence performance of turbine system as were demonstrated by some studies [8-11]. But because lack of deep analysis for mechanism of interaction between turbine systems, influence caused by flow distortion cannot be considered in process of turbine system design and matching. So in the paper, mechanism of interaction between turbine systems is analyzed by building turbine system simulation model. Based on analysis, flow field matching method can be raised to increase efficiency of turbine system.

2. Models and method
The two-stage turbocharger simulated is based on a type of diesel engine designed for high power density. Because it is difficult to build turbine system model as a whole, calculation model is divided into two parts: HP module and LP module as is shown in figure 1. HP module includes turbine rotor and connection pipe to simulate flow in turbine. LP module includes volute and rotor. To both models, flow control equation uses Reynolds-Averaged Navier-Stokes Equations solved by NUMECA code; turbulence model adopts S-A model which is verified appropriate to simulate turbulent flow in turbine rotor [12]. The flow solver is based on a cell centred finite volume approach, associated with the Jameson’s center scheme as space discretization. For HP module, four parameters are given to inlet boundary condition: flow mass rate in accordance with engine operational condition; velocities in three dimensions making angle of attack -20° ; static pressure calculated according to LP module is given to outlet boundary condition. For LP module, flow mass rate and velocities in three dimensions get from HP module are given to inlet boundary condition; and outlet boundary condition of LP module is given static pressure. Rotation speeds of two turbine rotors are given according to actual values measured on two-stage test rig.
Figure 1. Simulation model of two-stage turbocharger

In process of calculation of single simulation module, boundary conditions of calculation region are fixed and flow parameters of two modules cannot be exchanged. After convergence is finished, inlet flow field of upstream module is different from outlet flow field of downstream module. So flow in two modules is different from each other which results in simulation credibility decreasing. Therefore, it is necessary to add a module between two calculation modules to transfer flow information. Based on transfer module, outlet boundary conditions of HP module and inlet boundary conditions of LP module are given. The boundary conditions can not only reflect flow interaction between parts but also meet the characteristic compatibility condition for numerical calculation. In transfer module, flow mass rate, static temperature and velocity direction are transferred from upstream to downstream; static pressure is transferred from downstream to upstream. Flow transfer module is expressed by the equations listed below:

\[ T_d = \frac{\sum T_u \times \text{den}}{\sum \text{den}} \]  
(1)

\[ \dot{G}_d = \dot{G}_u \]  
(2)

\[ \frac{V_{dx}}{|V_d|} = \frac{V_{ux}}{|V_u|} \]  
(3)

\[ \frac{V_{dy}}{|V_d|} = \frac{V_{uy}}{|V_u|} \]  
(4)

\[ \frac{V_{dz}}{|V_d|} = \frac{V_{uz}}{|V_u|} \]  
(5)

\[ |V| = \frac{\sum V \times \text{den}}{\sum \text{den}} \]  
(6)

\[ p_u = \frac{\sum p_d \times \text{den}}{\sum \text{den}} \]  
(7)

The calculation process of turbine system model is showed in figure 2. Based on boundary conditions, calculation is performed in each module. Then flow characters are exchanged between two modules. By iterating calculation, convergence can be regarded finished when

\[ p_u^{n+1} = p_d^n \]  
(8)
Results and discussion

2.1. analysis of inlet static pressure field of LP module
Because LP module is at downstream, performance of HP module is mainly influenced by static pressure field of inlet to LP module. By numeral calculation inlet static pressure field is simulated. Boundary conditions are shown in table 1. In addition, inlet flow is assumed as uniform.

Table 1. Boundary conditions of LP module

| Case number | Flow mass rate(kg/s) | Outlet static pressure(Pa) | Rotation speed(r/min) |
|-------------|----------------------|---------------------------|-----------------------|
| 1           | 0.4                  | 101300                    | 75000                 |
| 2           | 0.5                  | 101300                    | 80000                 |
| 3           | 0.6                  | 101300                    | 85000                 |

Static pressure field is similar among conditions of different inlet flow mass. So the paper analyzed pressure field when flow mass rate is 0.5kg/s. The inlet pressure field is shown in figure 3.

Figure 3. Inlet static pressure field of LP module inlet

In figures above, (a) shows distortion of static pressure field on inlet plane of LP module; (b) is the chart showing law of pressure value along line c from top to bottom in (a).

Figure 3(b) illustrates that static pressure decreases from 195KPa to 170KPa along radial direction on inlet plane. Distortion is large in area close to volute top and bottom; however in middle area pressure field is approximate to uniformity. Besides, static pressure mean value is about 180KPa far greater than ambient pressure value. So outlet pressure of HP turbine in two-stage turbocharger is different from single-stage turbine which will influence flow in rotor and cause performance of turbine changed.
In two-stage turbocharging system, there is connection pipe between HP turbine and LP turbine. Distortion changes in process of pressure field transfers from downstream to upstream. So simulation is done to get static pressure field developing process shown in figure 4.

![Figure 4. Pressure field in connection pipe](image)

According to figure 4, we can conclude that distortion of pressure field gradually disappears when closing to HP turbine. At outlet to HP turbine static pressure field is uniform. So outlet pressure field of HP module can be regarded as uniform. Moreover, pressure mean values in different planes are almost the same and larger than ambient pressure.

2.2. Analysis of outlet flow field of HP module

By simulation, flow field at outlet to HP module is investigated. Boundary conditions are shown in table 2. In addition, inlet velocities of rotor are given assumed angle of attack is -20°.

| Case number | Flow mass rate(kg/s) | Outlet static pressure(Pa) | Rotation speed(r/min) |
|-------------|----------------------|---------------------------|-----------------------|
| 1           | 0.4                  | 150000                    | 80000                 |
| 2           | 0.5                  | 170000                    | 85000                 |
| 3           | 0.6                  | 190000                    | 87000                 |

Outlet flow field includes pressure field, temperature field and velocity field. Degrees of distortion of pressure and temperature are evaluated by equation following and shown in table 3.

$$\phi = \frac{V_{\text{max}} - V_{\text{min}}}{V_{\text{mean}}}$$ (9)

| Case number | Flow mass(kg/s) | Pressure field (%) | Temperature field (%) |
|-------------|-----------------|--------------------|-----------------------|
| 1           | 0.4             | 2.1                | 1.58                  |
| 2           | 0.5             | 2.34               | 1.74                  |
| 3           | 0.6             | 2.5                | 1.97                  |

Table 3 illustrates that degrees of distortion of pressure field and temperature field are both below 2%, and pressure and temperature can be regarded uniform at inlet to LP module.

Outlet velocity triangle of turbine rotor analysis shows that outlet velocity has not only axial component but also secondary flow component in plane vertical to axial direction [13]. To turbine rotor, direction of blade curved is opposite to rotor rotation direction to raise turbine efficiency. So
relative velocity at outlet inclined and thereby absolute velocity inclined to direction opposite to rotation speed forming outlet swirl. In accordance to velocity triangle analysis and simulation result, swirls at outlet to HP module of different planes are shown in figure 5.

![Figure 5. Swirls in different planes](image)

Swirl strength is defined as

\[ \phi = \frac{v_x}{v_{r_a}} \]  

Contour of swirl strength for plane 1 in figure 5 is drawn in figure 6 in which positive sign means that direction of swirl is opposite to rotation direction. In plane 1, swirl strength increases first and decreases then from rotor hub to rotor shroud. This is because gas expands more fully at hub than at shroud which brings about magnitude of velocity is larger at hub and so does axial component of velocity \( v_{r_a} \). Meanwhile, exit angle of blade changes from \( 24^\circ \) to \( 48^\circ \) with blade height increasing, which results in tangential component of relative velocity increasing with blade height. However, with blade height increasing convected velocity increases which causes tangential component of absolute velocity \( v_x \) goes up first and down then from hub to shroud.

![Figure 6. Swirl strength at outlet to HP turbine](image)

2.3. analysis of interaction of LP module to HP module

In two-stage turbocharger, back pressure of HP turbine is higher than that of single-stage turbine because of LP turbine. Greater is expansion ratio of LP turbine, higher is back pressure of HP turbine. To investigate law and mechanism of turbine performance changing with outlet static pressure, flow in rotor is simulated based on HP module. Boundary conditions and rotation speed for calculation is listed in table 4. When flow mass is constant, rotation speed is assumed the same in condition of different back pressure.
According to calculation, characteristics of flow capacity and efficiency of turbine in different flow mass and back pressure are shown in figure 7. In figure 7(a), points represent characteristic of turbine flow capacity in condition of different back pressure are close to a line. So it can be deduced that outlet back pressure has little influence on characteristic of turbine flow capacity. Figure 7(b) illustrates that efficiency of turbine in condition of different back pressure are close to a line too. So static pressure at outlet to turbine will not change flow in turbine, and in two-stage turbine system low pressure turbine has little influence on high pressure turbine performance.

![Figure 7](image)

(a) Characteristics of flow capacity  
(b) Characteristics of efficiency

**Figure 7. Performance of turbine**

### Table 4. boundary condition of high pressure turbine model

| Case number | Flow mass(kg/s) | Outlet static pressure(KPa) | Rotation speed(r/min) |
|-------------|----------------|---------------------------|----------------------|
| 1           | 0.4            | 190                       | 75000                |
|             |                | 250                       |                      |
| 2           | 0.5            | 190                       | 82000                |
|             |                | 250                       |                      |
| 3           | 0.6            | 190                       | 85000                |
|             |                | 250                       |                      |

According to calculation, characteristics of flow capacity and efficiency of turbine in different flow mass and back pressure are shown in figure 7. In figure 7(a), points represent characteristic of turbine flow capacity in condition of different back pressure are close to a line. So it can be deduced that outlet back pressure has little influence on characteristic of turbine flow capacity. Figure 7(b) illustrates that efficiency of turbine in condition of different back pressure are close to a line too. So static pressure at outlet to turbine will not change flow in turbine, and in two-stage turbine system low pressure turbine has little influence on high pressure turbine performance.

### 2.4. Analysis of interaction of HP turbine to LP turbine
Swirl at outlet to HP module causes non-uniform velocity field at inlet to LP module. According to analysis of outlet swirl of HP turbine, operational condition, boundary condition and selecting of two turbines will influence rotation direction and strength of outlet swirl. So inlet swirl of LP module may have different direction and strength. Different inlet condition has different influence on turbine performance. To analyze interaction HP turbine on LP turbine, swirls in clockwise and anti-clockwise direction and with different strength are given to inlet to LP module. Simulation is done to study mechanism of influence inlet condition on turbine. Boundary conditions and rotation speed for calculation is listed in table 5. When flow mass rate is constant, rotation speed is assumed the same in condition of different inlet velocity field.

| Case number | Flow mass(kg/s) | Rotation speed(r/min) | Swirl strength |
|-------------|----------------|-----------------------|----------------|
| 1           | 0.4            | 50000                 | 0.173          |
|             |                |                       | 0.337          |
|             |                |                       | 0.514          |
| 2           | 0.5            | 55000                 | 0.173          |
|             |                |                       | 0.337          |
|             |                |                       | 0.514          |
| 3           | 0.6            | 65000                 | 0.173          |
|             |                |                       | 0.337          |
|             |                |                       | 0.514          |

Figure 8 illustrates performance of turbine in different operational condition and in different condition of inlet swirls.

Figure 8. Performance of turbine

(a) Characteristics of efficiency
(b) Characteristics of flow capacity
In condition of different inlet swells, total pressure and total temperature are almost the same when flow mass rate is constant. So flow mass rate can be used to reflect operation condition. In figures above 0.4, 0.5 and 0.6 refer to flow mass rate. X axis is swirl strength and Y axis refers to efficiency of turbine in figure 8(a) and expansion ratio in figure 8(b). It can be deduced that turbine efficiency decreases with swirl strength increasing when flow mass is constant and direction of swirl is clockwise; however, when direction of swirl is anti-clockwise turbine efficiency goes up slightly and down then. Besides, turbine efficiency is higher when direction of swirl is anti-clockwise. Efficiency difference between two inlet swirl directions is about 3% when swirl strength is constant. Because total pressure is almost the same in different inlet condition, expansion ratio almost keeps constant. Taking flow mass rate 0.6kg/s as example, flow in turbine is investigated to study influence inlet swirl on turbine performance.

3.4.1 Flow analysis in volute. Flow in volute influences inlet velocity field of rotor and flow in rotor further. So it is necessary to investigate inlet swirl development in volute. Swirl at inlet is kept and transferred downstream. Because flow in volute is influenced by radial centrifugal force, counter-rotation swirl is formed in volute when inlet velocity field is uniform. When inlet velocity field is non-uniform, inlet swirl will be combined with counter-rotation swirl. At sections close to volute tongue, centrifugal force is small and inlet swirl dominates flow structure. Figure 9 shows outlet angle of volute at section close to volute tongue in three different inlet conditions. Outlet angle in condition of inlet clockwise swirl is about 30° close to rotor hub while close to shroud angle is about 80°. In condition of inlet anti-clockwise swirl, outlet angle is in contrast to the inlet condition of clockwise swirl.

![Figure 9. Outlet angle of volute in different inlet conditions](image)

At sections far away from tongue, inlet swirl dissipates and flow structure mainly is counter-rotation swirl. Flow structure is almost the same even if inlet condition is different. So influence inlet swirl on turbine mainly concentrates in sections close to volute tongue.

3.4.2 Flow analysis in rotor. Small incidence angle of rotor means low tangential velocity and large radial velocity which results in great negative angle of attack as is shown in figure 10. Assuming convected velocity $u_1$ and value of absolute $c_1$ is constant, angle of attack of rotor $\theta$ is decided by incidence angle $\theta' \text{[14]}$. 
So from figure 10 it can be deduced that angle of attack is not uniform along inlet height of blade. When inlet swirl is clockwise, great negative angle of attack is close to shroud and small angle is close to hub. Outlet angle in volute is uniform relatively when inlet swirl is anti-clockwise or inlet velocity field is uniform. So angle of attack of rotor is uniform relatively too in the two inlet conditions. Distribution of angle of attack influence flow in impeller passage and causes turbine performance changed.

Based on simulation results, entropy diagram in blade to blade planes of 10%, 50% and 90% blade height are drawn in the three inlet conditions. In the paper, inlet condition when velocity field is uniform is defined IU and inlet swirl is clockwise and anti-clockwise inlet condition are defined as IC and IAC respectively. Entropy differences between IU and IC and between IU and IAC are calculated and drawn in figure 11.

From figure 11 it can be concluded that entropy difference is mainly located in five impeller passages close to volute tongue which proves that influence inlet swirl on flow in rotor is in passages close to volute tongue. Entropy difference is little close to hub and great close to 50% blade height. The difference is located at suction surface and 2/3 chord length. However, entropy difference is great in the whole impeller passage close to shroud especially at inlet close to pressure surface and outlet close to suction surface. Meanwhile, Figure 11 shows that entropy difference is greater between IU and IAC. When IC, the reason high entropy generated can be analyzed by combination of entropy and streamlines diagram shown as follows.
As is analyzed, there is large negative angle of attack at inlet close to shroud when IC. Flow at inlet is wrapped with leakage flow from former passage at inlet and forms vortex as is shown in figure 12 (a). The gas flows along pressure surface from inlet and turn from pressure surface to suction surface at 1/5 chord length of blade. The lateral flow reaches to suction surface and combined with leakage flow from later passage causing high entropy increase. Along blade height, the flow goes down and to outlet. At the zones gas flow passes by high entropy increase is generated as is shown in figure 12 (a) and (b). So negative angle of attack at inlet close to shroud caused by inlet swirl is the reason entropy difference is generated. When inlet swirl is anti-clockwise, angle of attack becomes small as is shown in figure 10. So there is smaller entropy increase which results in little entropy difference between inlet condition IU and IAC as shown in figure11(b).

3. Conclusions

With raising efficiency of exhaust gas energy usage of turbine system as the research target to realize high intake pressure ratio for hydrogen internal combustion engine, the model for flow analysis of turbine system was established and analysis of the mechanism of interaction between two turbines was achieved. The achieved concrete studies and the obtained conclusions are listed below.

1) Using Numeca code, the simulation model for flow interaction among turbine system is established. The calculation model was divided into two parts: LP module and HP module which were modeled respectively.

2) By simulation, the flow field distortion at outlet to HP turbine and inlet to LP turbine is analyzed. Swirl exists at outlet to HP turbine whose strength and rotation direction varies with operation condition and boundary condition; static pressure rises at inlet to LP turbine and distorted pressure field gradually becomes uniform when transfers to outlet to HP turbine.

3) Simulation proves that back pressure has little influence on turbine performance, so in two-stage turbine system HP turbine performance has little relationship with LP turbine selection or design.

4) Inlet swirls caused by HP turbine influence efficiency of LP turbine. To turbine in the paper, large negative angle of attack close to shroud generates strong vortexes in rotor passages and results in great flow losses when direction of inlet swirl is clockwise. On the contrary, anti-clockwise swirl helps to decrease vortex strength and raise turbine efficiency.
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# Appendix A.

## Nomenclature

| Symbol | Definition | Symbol | Definition |
|--------|------------|--------|------------|
| $\varphi$ | Degree of distortion | $\theta$ | Angle of attack of rotor |
| $\phi$ | Swirl strength | $\theta'$ | Incidence angle of rotor |
| $\nu_2$ | Tangential component of velocity | $u$ | Upstream characters |
| $\nu_d$ | Axial component of velocity | $d$ | Downstream characters |
| $c_1$ | Inlet absolute velocity of rotor | $x$ | X direction |
| $\omega_1$ | Inlet relative velocity of rotor | $y$ | Y direction |
| $u_1$ | Inlet convected velocity of rotor | $z$ | Z direction |
| $c_2$ | Outlet absolute velocity of rotor | $den$ | density |
| $\omega_2$ | Outlet relative velocity of rotor | $p$ | Static pressure |
| $u_2$ | Outlet convected velocity of rotor | $n$ | Interactions |
| $p^*$ | Inlet total pressure | $G$ | Flow mass rate |
| $\Delta \eta$ | Turbine efficiency change | $V_{\text{max}}$ | Maximum quantity value |
| $\Delta \eta_R$ | Turbine rotor efficiency change | $V_{\text{min}}$ | Minimum quantity value |
| $\Delta \eta_V$ | Transmission efficiency of volute change | $V_{\text{mean}}$ | Mean quantity value |
| $\eta_R$ | Turbine rotor efficiency | $V$ | velocity |
| $|\vec{v}|$ | Mean value of velocity | IAC | Inlet anti-clockwise swirl |
| $T$ | Static temperature | LP | Low pressure stage |
| IU | Uniform inlet velocity field | HP | High pressure stage |
| IC | Inlet clockwise swirl | EGR | Exhaust gas recirculation |