Centrifugal compressor design for electrically assisted boost

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Abstract. Electrically assisted boost is a prominent method to solve the issues of transient lag in turbocharger and remains an optimized operation condition for a compressor due to decoupling from turbine. Usually a centrifugal compressor for gasoline engine boosting is operated at high rotational speed which is beyond the ability of an electric motor in market. In this paper a centrifugal compressor with rotational speed as 120k RPM and pressure ratio as 2.0 is specially developed for electrically assisted boost. A centrifugal compressor including the impeller, vaneless diffuser and the volute is designed by meanline method followed by 3D detailed design. Then CFD method is employed to predict as well as analyse the performance of the design compressor. The results show that the pressure ratio and efficiency at design point is 2.07 and 78% specifically.

1. Introduction

Engine downsizing is one of the prominent methodologies for reducing the CO₂ emissions in internal combustion engine [1]. In order to sustain the engine power after downsizing, specific power of the engine has to be improved. Turbochargers are the key technology to allow engine downsizing by improving specific power of the engine and are widely used for boosting IC engine. The turbine recovers the energy from the exhaust gas from IC engine then drives a centrifugal compressor to boost the intake air. One of the challenges for the turbocharger prevailing in IC engine is ‘turbo-lag,’ which is mainly caused by the exhaust gas delay in overcoming turbine inertia, especially at low engine load conditions. Electrically-assisted boosting is an upcoming trend being employed in order to reduce the ‘turbo-lag’.

Electrical-assisted booster consists of an electrical motor driving a supercharger element or centrifugal compressor [2, 3]. The electrically-assisted booster with a centrifugal compressor has advantage in efficiency, response time over a displacement supercharger [4, 5]. It also helps improve engines fuel consumption due to the elimination of the parasitic losses. Electrical boosters are often employed in series with a conventional turbocharger. The principle of engaging the electrical booster when required by the engine results in a far superior transient performance compared to conventional turbocharging methods. The rotation speed of an electrical booster is limited by the motor, which is usually lower than a normal compressor operation condition. In order to achieve target efficiency and pressure ratio constrained by the maximum speed of the driving motor, a centrifugal compressor with relatively low specific speed has to be designed for this purpose.

In the paper the designing process and performance analysis of a centrifugal compressor for electrically assisted boost in a 2.0L petrol engine is presented.
2. Configuration and Design Requirements

The power-dense electric machine and its control technologies in a 48V e-supercharger (e-SC) are being developed by Aeristech Ltd. Figure 1 below shows the lay-out of the low-power assisted boosting configuration. The main turbocharger and Aeristech’s e-SC are arranged in series. One of the boosters is a conventional low-pressure turbocharger while the other is a centrifugal compressor driven by an electrical motor (i.e. e-SC). This e-SC is designed as a low-power assist device in a typical 2.0 L petrol engine across 0-2000 rpm. This electric supercharging application requires a centrifugal compressor design that is capable of delivering high-power at low engine speed in order to improve low-end torque response and increase the ability to downsize internal combustion engines.

![Figure 1. Configuration of the electrical assisted booster in an IC engine.](image)

The rotational speed of the compressor is limited by the electrical motor, which is specified as 120,000 rpm based on the high-speed electrical motor developed by Aeristech in this case. The design operating point of the centrifugal compressor required by the engine is rated at a pressure ratio of 2.0 and mass flow rate of 0.08 kg/s. Furthermore, response time of an electric booster is one of the key advantages over the conventional turbocharger. For the centrifugal compressor design, the acceleration time from 3k rpm to 12k rpm is targeted in 1 second, thus the inertia of the compressor should be designed to be kept minimal and within the optimal limits of compressor performance.

3. Compressor Design and Results Analysis

3.1. Meanline design

Firstly the specific speed is chosen to determine the design speed of the compressor. The specific speed is evaluated by the design conditions in equation 1:

\[ N_s = \omega \sqrt{Q_v/H^{0.75}} \]  

(1)

Where \( \omega \) is the rotational speed (rad/s), \( Q_v \) is the volume flow rate and \( H \) is work input.

In a centrifugal compressor design, potential maximum efficiency can be achieved when the specific speed is near 0.7~ 0.8 [6]. However, a higher value is usually chosen for the compressor design in order to reduce the size of the impeller. For the electrically assisted compressor, the freedom of specific speed chosen is limited by the speed of the electrical motor. For the compressor design in this paper, the specific speed is chosen as 0.79 and correspond design speed is 120k RPM. The drawback on the impeller size and hence the inertia will be compensated by less blades, which will be discussed following. Main geometries of the centrifugal compressor are designed based on the velocity triangles and thermodynamic analysis. The process is shown in figure 2.
The parameters including inlet conditions, maximum rotational speed, pressure ratio and mass flow rate are given as designed constrains. In this paper a set of geometrical parameters are chosen to start the design. Exit blade angle is a key parameter for the compressor performance. A trade-off has to be made among surge margin, efficiency and boosting pressure ratio. The angle is chosen as 35 degrees in the design to achieve good efficiency and operation range. Inertia of the rotating component is the parameter directly relating with the response time of the electrical booster. The diameter of the impeller is the most important parameter indicating the inertia, but there is little freedom for the parameter since it is mainly determined by the pressure ratio and rotational speed. In order to reduce the impeller inertia hence the response time, the blade number is designed to be small to reduce the impeller weight, although the performance of the compressor may be sacrificed due to large blade loading. The vaneless diffuser type is chosen for operation range consideration. Overhang type volute is chosen and the throat area is designed based on the evaluation of flow condition at diffuser exit. The geometries of meanline design are shown in table 1.

Table 1. Main geometries of the centrifugal compressor.

| Geometrical parameters          | Value       |
|--------------------------------|-------------|
| Inlet hub radius $R_{1h}$       | 6.2 mm      |
| Inlet shroud radius $R_{1s}$    | 17.2 mm     |
| Impeller exit radius $R_2$      | 29.1 mm     |
| Exit blade height $b_2$         | 3.4 mm      |
| Blade number $Z$                | 5+5         |
| Blade angle at inlet tip $\beta_{1h-shroud}$ | -62 deg.    |
| Diffuser exit radius $R_3$      | 40.75 mm    |
| Volute throat area $A_{th}$     | 397 mm$^2$  |

The performance of the designed compressor is predicted by the meanline model in which two-zone model is applied for the impeller. Table 2 shows the predicted pressure ratio as well as the total-
total efficiency at design condition. The boost pressure ratio is 2.23 and the efficiency is 77.5%, which meets with the design target.

| Performance                        | Value |
|------------------------------------|-------|
| Total-total pressure ratio         | 2.23  |
| Overall total-total efficiency     | 0.775 |
| Total-total efficiency of impeller | 0.89  |
| Diffuser loss                      | 0.07  |
| Volute loss                        | 0.045 |

3.2. Detailed Design

The main task of the detailed design is to determine the shape of the impeller blade. In the paper the middle surface of the blade is designed firstly, then the thickness distribution is designed to produce the pressure surface and suction surface. For this design method, main geometries of the impeller needed to be determined are the shroud and hub curves, blade angle distribution and blade thickness. The hub and shroud are designed to turn the flow from axial to radial direction smoothly; the blade angle distribution is designed to peak the blade load in middle part of the blade passage, thus reduce the tip clearance flow at blade inlet and slip factor at impeller exit. Thickness of the blade is designed based on the strength as well as the inertia of the impeller. The final machined impeller is shown in figure 3(a). A parallel wall at exit of the impeller is designed as the vaneless diffuser. Conventional round shape is applied for the volute section, as shown in figure 3(b). Once the throat area is determined in meanline design, a linear distributed A/R in circumferential direction is designed for the volute and the flow is assumed to follow the free vortex rule. The final geometries of the centrifugal compressor are shown in figure 3.

![Figure 3. Details geometries of the centrifugal compressor.](image)

3.3. Performance prediction and result analysis

CFD method is applied for the performance prediction after 3D detailed geometries design. A single passage of the impeller (and diffuser) is modeled for the numerical simulation. There are 1.04 million nodes in the domain and the size of the first cell near the way is 0.001mm which guarantees $y^+$ to be small enough. Reynolds averaged Navier-Stokes equations are solved by CFD code EuranssTurbo based on 3-D steady finite volume scheme. Central scheme is used for space discretion. The Spalart-Allmaras turbulence model is applied for equation closure. Total pressure and temperature, together with axial velocity direction, are used for inlet boundary condition, while the static pressure as outlet boundary condition.
Figure 4. Predicted performance of the centrifugal compressor (impeller and diffuser) by CFD.

Figure 4 shows the predicted performance of the design impeller and vaneless diffuser at 120k (design speed), 100k and 80k RPM. The pressure ratio at design point is 2.07 and the efficiency is 78%. Specifically, the efficiency of the impeller is 82.6% and the vaneless diffuser contributes to 4.6 points efficiency reduction. The slope of the pressure ratio vs. mass flow rate still remains negative at small mass flow rate side, indicating a reasonable surge margin of the design compressor. The absolute flow angle at impeller exit is about 67.2 degrees which is in the acceptable range. The average velocity at the impeller exit is 238 m/s then decelerated to 163 m/s after the diffuser before going into the volute.

Figure 5. Streamlines along the passage in meridional plane.

Figure 5 shows the streamlines along the passage in meridional plane. Near the shroud of the impeller exit there is a recirculation region extending to nearly outlet of the diffuser. This region results from the secondary flow accumulation near the shroud of the impeller exit, which is a typical flow phenomenon in a centrifugal compressor. But the extended recirculation region in the vaneless diffuser will reduce the efficiency as well as the stability. A pinch on the shroud side of the diffuser can be applied in further optimization to depress the recirculation region. Except for this region, it can be seen that there is no obvious flow separation near the shroud or hub before the impeller exit and the flow follows the passage relatively smoothly.
4. Conclusions

Electrically assisted boost provides a facility to boost intake air with little retard of conventional turbocharger. A centrifugal compressor, including impeller with small blades number, diffuser and volute, is designed in this paper for an electrically assisted boost for a 2.0L petrol engine. The pressure ratio of the design compressor (volute is not included) is predicted as 2.07 and efficiency is 78% at design point which meets the design target. Specifically, the efficiency of the impeller is 82.6%. A recirculation region appears near the shroud of the diffuser resulting from the secondary flow at impeller exit. A pinch at inlet of the vaneless diffuser will be considered for further optimization.

References

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