

Improved Performance Prediction Model for Turbocharger Compressor

Mingyang Yang, Xinqian Zheng and Yangjun Zhang
Tsinghua University

Zhigang Li
Beijing Special Vehicle Institute

Copyright © 2008 SAE International

ABSTRACT

In order to improve centrifugal compressor performance predictive capability, an improved recirculation loss model in two-zone modeling system is presented in this paper. The new loss model correlates Reynolds number of the impeller with the recirculation loss. Performance prediction by the improved model is carried out on two turbochargers with different sizes based on COMPAL mode of the code Concepts. The result shows that predictive performance by improved model is in high accordance with experimental measurement. On the other hand, compared with the larger size compressor, the small one has a performance which is more likely to be influenced by Reynolds number.

NOMENCLATURE

A	Area
T	Temperature
AR_a	Ratio of blade passage throat area and inlet area
U	Impeller periphery velocity
AR_b	Blade passage throat area and outlet area ratio
W	Relative velocity
C_p	Specific heat
C_m	Meridional velocity
C_t	Tangential velocity
D	Impeller diffusion factor
P_0	Total pressure
DR	Impeller diffusion ratio
P	Static pressure
Q	Mass flow rate
P_{recir}	No dimensional recirculation loss
h	enthalpy

Greek Symbols

α	Meridional angle of absolute velocity
β	Meridional angle of relative velocity
β_b	Blade angle
δ	Deviation angle
ε	Area fraction of secondary flow
η_a	Effectiveness of section 'a' in TEIS model
η_b	Effectiveness of section 'b' in TEIS model
η	Efficiency
λ	Volute inlet swirl parameter
ρ	Density
χ	Mass fraction of secondary flow

INTRODUCTION

A reasonable 1-D model of turbocharger compressor can reproduce reliable performance map with little computational cost. Usually the matching calculation between a turbocharger and an engine is performed on the base of their performance maps. Although map could be provided by manufactures or obtained by experiment, the range of rotational speed or mass flow in the map is limited. Mathematical interpolation can be employed to estimate performance which is not involved in the map. However, it could not be reliable especially for extrapolation because the method has no physics basis [1]. Since a 1-D performance prediction model is not limited by specific rotational speeds or mass flow, it is a power tool to obtain turbocharger compressor performance, thus could be used to perform matching calculation. On the other hand, performance prediction

of turbocharger is an important part of the whole turbocharged engine performance estimation [2]. Because the turbocharger is one of important components in the engine, reliable engine performance estimation is influenced by turbocharger performance prediction. Therefore, a reliable performance prediction model over a wide operating range is quite important for the matching calculation between turbocharger and engine as well as the whole engine performance estimation.

The recirculation loss is one of compositions of the parasitic loss in two-zone model system of the turbocharger centrifugal compressor. It happens at the impeller discharge section. When the flow is distorted severely enough, it is separated close to the shroud and locally the fluid moves inwards. Then the reversed flow is energized by the fluid in the impeller and recirculation loss is arisen [2]. It is boundary layer development, centrifugal force and coriolos force that contribute to flow distortion then recirculation loss. Therefore, the factors reflecting these factors should be involved in the loss correlations.

The recirculation loss dominates the disc friction loss and the leakage loss in the off-design conditions, thus the predicted efficiency is strongly influenced by the loss [3]. Several empirical and semi-empirical correlations of the loss are available in the published literatures. Rodgers had suggested the following equation to estimate the value of recirculation loss [4]:

$$P_{recir} = \frac{\Delta h}{U^2} = 0.032 \left(\frac{U}{C_{m1}} \right)^2 \quad (1)$$

No factor about the impeller outlet flow angle is involved in the correlation, which seems not consistent with the backward phenomenon.

On the analogy of blade loading loss in axial compressor, diffusion factor was introduced into the recirculation loss model by Dallenbach [5].

$$P_{recir} = 0.02 D^2 \sqrt{\tan \alpha_2} \quad (2)$$

Where D is impeller diffusion factor defined as:

$$D = 1 - \frac{W_2}{W_1} + \frac{0.75 \Delta h_{th} W_2}{\left(\frac{Z}{\pi} \left(1 - \frac{r_{s1}}{r_2} \right) + \frac{2r_{s1}}{r_2} \right)} \quad (3)$$

In order to consider compressor stall, Aungier [6] introduced the blade stall limit of diffusion factor into the recirculation loss correlation at low flow region.

$$P_{recir} = \left(\frac{D}{2} - 1 \right) \left[\frac{W_{u2}}{C_{m2}} - 2 \cot \beta_2 \right] \quad \text{if } D > 2 \quad (4)$$

Oh H. W. [3] used a hyperbolic function of impeller exit flow angle in the Dallenbach's model to improve the

accuracy of predicted performance according to experiment results.

Those correlations with impeller diffusion ratio are commonly used in compressor performance prediction, however, it could be inferred from them that the minimal recirculation loss at each speed is independent on the rotational speeds or sizes of the impeller, which seems not consistent with the reduced results from experiment.

Japikse developed the loss model in a quite different way [7]. He suggested an empirical recirculation loss correlation curve ("bucket" model) as a function of no-dimensional mass flowrate based on lots of experimental data. Again the rotational effect was not involved in the model.

This paper is aimed at finding an improved recirculation loss model to consider the influence of rotational speeds as well as impeller sizes. A specified Reynolds number of the impeller is used to integrate those factors. Firstly, the two-zone performance model system is presented briefly. Then an improved recirculation loss model is established on the base of "bucket" model. The performance prediction by the new model is performed on two turbocharger centrifugal compressors. One is a gasoline turbocharger compressor with small size; the other is a diesel engine turbocharger compressor with larger size.

IMPROVED RECIRCULATION LOSS MODEL

Euler work of the impeller is used to make the recirculation loss dimensionless as:

$$P_{recir} = \frac{W_{recir}}{W_{euler}} \quad (5)$$

Because flow develops well in the passage at design point, no back flow happens at the discharger of compressor impeller. Thus it is reasonable to consider that recirculation loss is quite close to zero at the design point [7]. However, in off-design conditions, the recirculation loss increases and has a significant influence on the predicted performance.

The "bucket" model is always used in two-zone model system. In this loss model, two parabola sections are adapted to describe the relationship between recirculation loss and mass flow rate at different rotational speeds:

$$P_{recir} = a \left(\frac{Q}{Q_{design}} - 1 \right)^2 + b \quad (6)$$

Where a is an empirical coefficient defined as:

$$a = \begin{cases} \alpha_1 & \frac{Q}{Q_{design}} \leq 1, \\ \alpha_2 & \frac{Q}{Q_{design}} > 1 \end{cases} \quad (7)$$



The coefficient b is recirculation loss at the design mass flow and it is given to be zero according to the assumption mentioned above.

Figure 1 and 2 separately shows the different performance maps and distribution of recirculation loss from data reduction model of COMPAL at several rotational speeds in two turbocharger centrifugal compressors with different sizes. The first one is the compressor with a large size. It is implied in the figures that the highest efficiency and minimal recirculation loss at different rotational speeds is almost at the same level. The second one is the compressor with a smaller size. In the figures the highest efficiency circle only covers few rotational speed lines and the minimal recirculation loss is quite different from each other. It can be inferred that recirculation loss is closely related to impeller rotational speed as well as impeller geometry. But this phenomenon is not implied in the referred correlations above. In this paper, a factor of Reynolds number is introduced into loss correlation to describe those influences.

It is reasonable to correct the Reynolds number with recirculation loss, especially for the compressors with small size of which Reynolds number is small and around the sensitive value. The effect of low Reynolds number on centrifugal compressor was well reported in the literatures and kinds of correlations of compressor performance parameters were derived from abundant of experiments [8]. Their evidence all showed that when impeller Reynolds number was below some critical value, performance of the impeller, especially for the efficiency, could decay sharply. The diffuser inlet Reynolds number which is in terms of absolute velocity, impeller tip width and local values of viscosity and density provides the best correlating Reynolds number parameter for centrifugal compressor. However, it is inconvenient to determine the absolute velocity at the exit of impeller as well as the local values of gas properties before compressor is determined. Wiesner proved that Reynolds number in terms of tip speed, tip width, inlet viscosity and inlet specific volume produced similar results. Therefore this substitute definition is used in this paper which is defined as:

$$Re = \frac{U_2 b_2}{\nu_1} \quad (8)$$

For most of gasoline engine turbocharger compressors, the sizes are always quite small. When those compressors are operated at a low rotational speed, it is possible that Reynolds number of the impeller is below a critical value which is always around 10^5 when impeller

discharge absolute Reynolds number is used [8]. Because the recirculation flow comes from low energy fluid including boundary layer in the impeller, once the low Reynolds number effect happens on the development of blades boundary layer, the deficit of exit flow will be enhanced and then recirculation loss will increase.

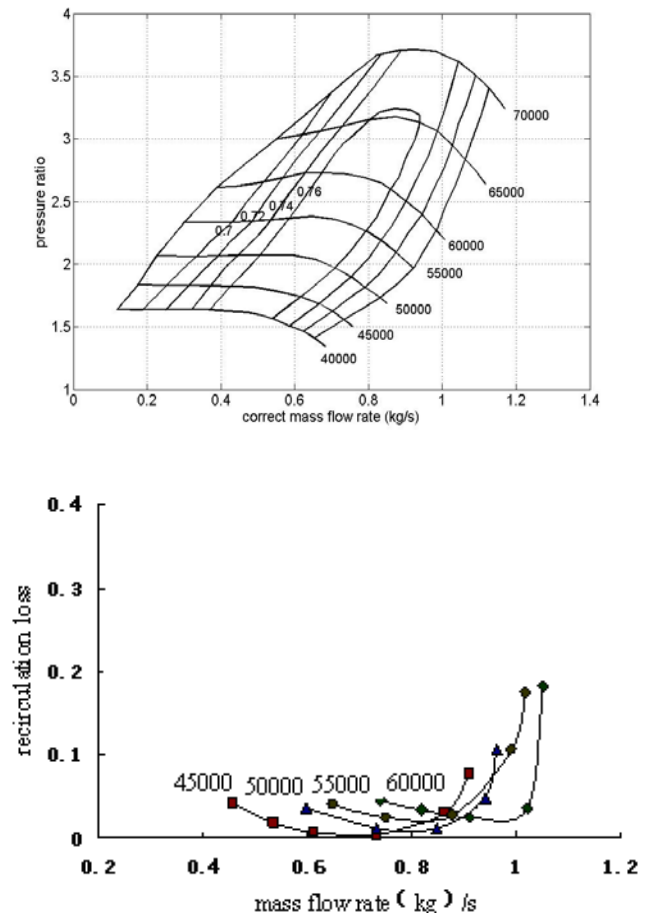


Figure 1. Compressor map and reduced recirculation loss for a compressor with large size

