Aerodynamic Optimization Design for All Condition of Centrifugal Compressor

Zhirong LIN[1], Xue-lin GAO[2], Xin YUAN[1]
[1] Key Laboratory for Thermal Science and Power Engineering of Ministry of Education, Tsinghua University, Beijing 100084, P.R. China
[2]Takasago R&D Center, Mitsubishi Heavy Industry, Takasago, Japan
linzhirong@tsinghua.edu.cn

Keywords: aerodynamic optimization, blade design, centrifugal compressor, all condition

Abstract

This paper describes an application of centrifugal compressor optimization system, in which the blade profile of impeller is represented with **NURBS**(**N**on-**U**niform **R**ational **B**-**S**pline) curve. A commercial **CFD**(**C**omputational **F**luid **D**ynamics) program named NUMECA fine/turbo was used to evaluate the performance of the whole centrifugal compressor flow passage including impeller and diffuser. The whole optimization design system was integrated based on iSIGHT, a commercial integration and optimization software, which provides a direct application of some optimization algorithms. To insure the practicability of optimization, the performance of centrifugal compressor under all condition was concerned during the optimizing process. That means a compositive object function considering the aerodynamic efficiency, pressure ratio and mass flow rate under different work condition was applied by using different weight number for different conditions. Using the optimization method described in this paper, an optimized design of the impeller blade of centrifugal compressor was obtained. Comparing to the original design, optimized design has a better performance not only under the design work condition, but also the off-design work condition including near stall and near choke condition.

Introduction

Many methods are developed to reduce the loss in turbomachinery flow, including inverse or direct design. But these methods are not easy to run automatically and the design cycles are too long to satisfy the requirement. With the development of CFD, **CAD** (**C**omputer **A**ided **D**esign) technology and optimization theory, an aerodynamic optimization method was introduced to design blade profile and reduce the flow passage's loss[1)]. In this method, the blade profile was represented with a parametric curve, such as B-spline, Bezier or NURBS curves. An optimization algorithm is used to optimize the parameters of the curves and then improve the performance of the profile, which is evaluated with CFD or some approximation method, such as **RSM** (**R**esponse **S**urface **M**ethod)[2)]. Thus, an integrated optimization system consists of three main modules: parametric modeling, optimization strategy and evaluating.

Loss of steady flow in turbomachinery can be roughly divided into two parts, cascade profiles and blade ends. The end loss is generated from the viscous friction in the end-wall boundary layer, and the secondary flow of the separation in the layer[3)].

It has long been recognized that blade bend can be used to reduce the end-wall loss with reducing the radial secondary flow by changing the pressure distribution in the span-wise direction. Before the availability of 3 dimensional CFD, "end-bend" designs based on experimental research programs were in use for some years. With introducing stacking-line into the aerodynamic optimization design system, the experimental cost was reduced greatly and the development cycle was shortened[4)].

Parametric Modeling Module Based on NURBS for Impeller/Diffuser

As Fig.1 shows, the optimization case discussed in this paper is a centrifugal compressor including an impeller and a vaned diffuser.

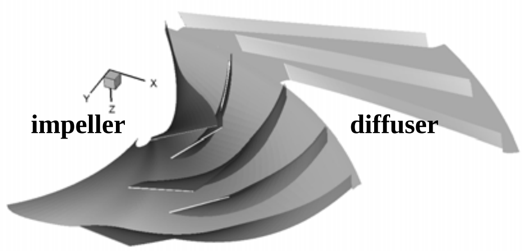

Fig.1 Sketch of the centrifugal compressor

The modeling method for centrifugal compressor blade in this paper including profiles at hub and tip represented by camber-line and its normal thickness. During the optimization process we keep the thickness distribution along the meridianal chord, thus it only needs to change the camber-lines on hub and tip profiles, namely, the distribution of camber-line slope in degrees along the meridianal chord. Hence, parametric modeling is only needed for the distribution of camber-line slope in degrees along the meridianal chord. Fig.2 shows the parametric modeling method of the impeller and the comparison between reconstructed and original slope in degrees. NURBS curves modeling the distribution of camber-line slope in degrees at hub and tip profiles are almost the same. Using 9 control points respectively reconstructed NURBS curve fits with the original camber-line slope distribution discrete points. The parametric modeling method of vaned diffuser is the same as the impeller.

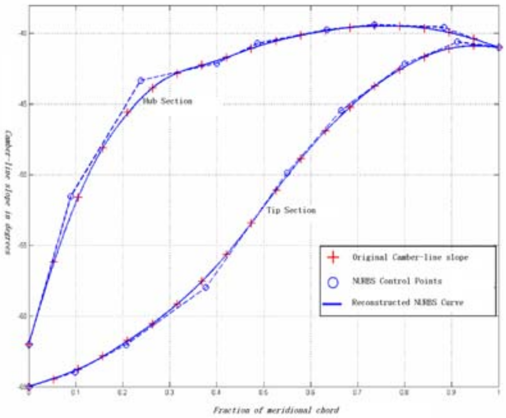

Fig.2 Parametric modeling and reconstruction for impeller camber-line slope

To parameterize the two stream lines at the impeller meridianal tip and hub passage, NURBS curves contained 8 control points were applied respectively refer to Fig.3.

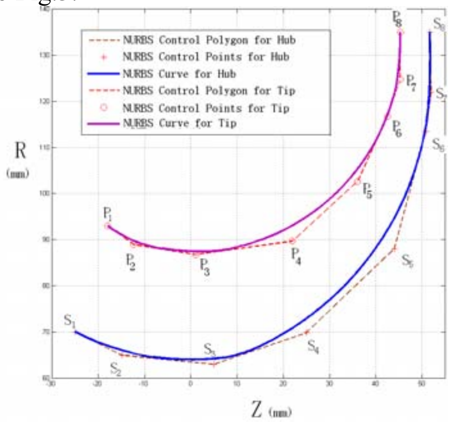

Fig.3 Parametric modeling for the stream surface

The final constructed impeller and diffuser blades are shown as Fig.4.

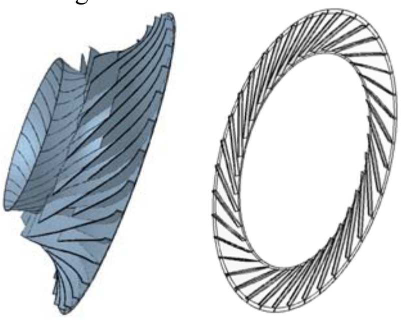

Fig.4 Sketch of impeller and diffuser modeling

Performance Evaluating Module

Computation grid and CFD solver configuration

Numerical simulation for flow field employs I-type grid generated by Autogrid/IGG shown as Fig.5. There is tip clearance around impeller and the total number of grid nodes is 251,002. The S-A turbulence model was applied according to this model has a good performance on compressor flow. The boundary conditions were set in the most general way: specify total temperature, total pressure and velocity direction at inlet; specify static pressure at outlet.

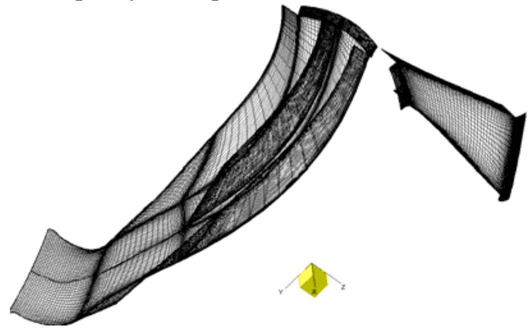

Fig.5 Computation grid for the centrifugal compressor

Eigen points for full conditions and objective functions

In order to generally evaluate the compressor's performance under full conditions and avoid the stall margin reduction when the optimization was done just under a single working condition, in this paper the aerodynamic performance optimizations under both design and off-design conditions are expected. However, the evaluation of compressor performance under full conditions generally needs calculating a series cases under different working conditions and spend too much CPU time. After overall analysis of the original design's performances under off-design conditions, we select three eigen conditions to evaluate the full condition performance. The original compressor's off-design performance is shown as Fig.6, and we select 3 eigen points in the full conditions area, near stall point, peak efficiency point and near design point. Therefore, the objective function must includes the performances under these three conditions: efficiency, mass flow rate and pressure ratio. Obviously the higher efficiency is the primary target. At the same time, prevent change of mass flow rate and pressure ratio is also necessary. So a compositive objective function considering above factor must be written as follows:

$$\begin{aligned} max\ F = & (w_1 \cdot \eta_{peak} + w_2 \cdot \eta_{design} + w_3 \cdot \eta_{stall}) + \\ & (w_4 \cdot |\Delta p_{peak}| + w_5 \cdot |\Delta p_{design}| + w_6 \cdot |\Delta p_{stall}|) + \\ & (w_7 \cdot |\Delta M_{peak}| + w_8 \cdot |\Delta M_{design}| + w_9 \cdot |\Delta M_{stall}|) \end{aligned} \quad (1)$$

Where $w_{1,2,\ldots,9}$ is the weights of the sub-objective function, η is the centrifugal compressor's efficiency, Δp is the relative variable of pressure ratio, ΔM is the relative variation of mass flow rate. Subscript peak means the performance parameter under peak efficiency condition, design means the performance parameter under near design condition and stall means the performance parameter under near stall condition.

Variable complexity model with two sets of grid

As the centrifugal compressor's calculation is hard to converge and the computation time is long in usual, we employ variable complexity model and introduce a set of coarse grid with 35266 notes which has the same topology with the fine grid introduced in above text. We find that in some case the coarse grid can predict the geometric transformation aerodynamic performance trend correctly but with much less CPU time than the fine grid. To assure this assumption in the specific case, we usually employ the Design of Experiment method before using the coarse grid and compare these two different grids' features. If the trends are consistent, coarse grid is qualified to predict the aerodynamic performance. Thus, the optimization process can be accelerated and reliable by trading off two different complexity models.

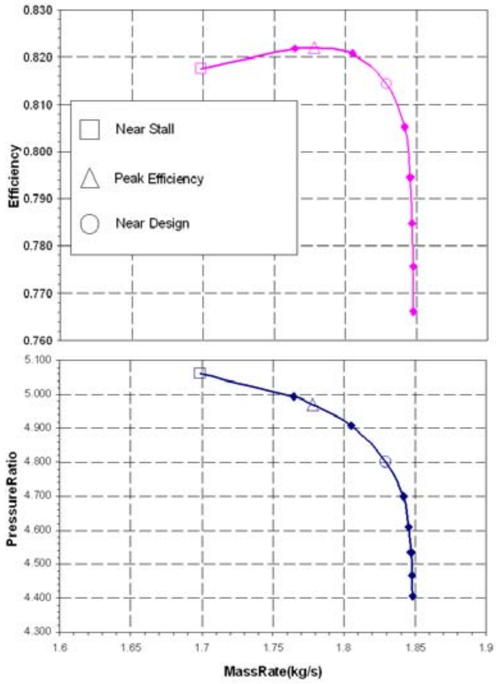

Fig.6 Compressor characteristic curve and 3 eigen points

"Latin square" method is adopted as the design of experiment method in this chapter, and comparison is taken place at 121 design points. The results of diverged or diverging design points are not taken into account in the calculating process, then the number of final effective design points comes to 73. Comparing these points we could find different grids' performance trends nearly the same as shown as in Fig.7.

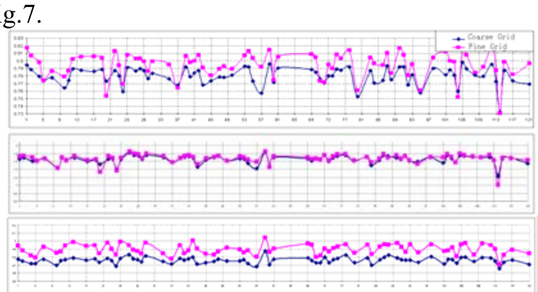

Fig.7 Comparison of coarse grid and fine grid's performance (efficiency, mass flow rate, pressure ratio)

Separate-Deformation Combined Optimization Strategy

Optimization system flowchart

With the preparative work done as above, we can optimize the centrifugal compressor's performance under full conditions with variable complexity model, and establish the optimization system according to the flow shown in Fig.8.

Besides the discussed modules, an added module could also be found in Fig.8 to judge the convergence of CFD calculation. Because of the blade shape transformation, the centrifugal compressor's calculation under the original boundary condition may not converge as usual, with incredible result of flow field calculation, and we have to deal with the abnormal results. Generally we classify the abnormal convergent conditions into three species as follows:

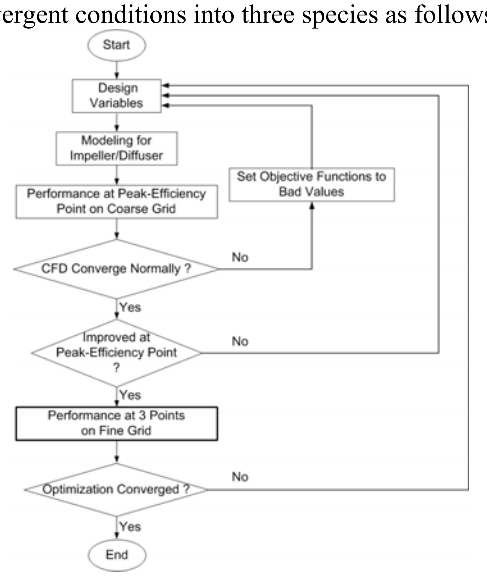

Fig.8 Optimization flowchart with variable complexity model

1)The calculating can not converge when reaching the specified iteration step number. And the result is not correct as the inlet/outlet mass-flow rates less than 0;

2)The calculating is about to diverge but not divergent when reaching the stated iteration step number as the inlet/outlet mass-flow rates drop-off obviously at the end;

3)The calculating may converge in a greater iteration step number than the specified one, but is still not convergent when reaching the specified iteration step number as the inlet/outlet mass-flow rates are quite different at the end.

For these abnormal convergence conditions, the objective function can be assigned directly in the system, and then the wrong optimization direction caused by wrong reading results would be avoided.

Design Variables

In this paper, the optimization of centrifugal compressor blade involves four aspects of design parameters: impeller hub's and tip's camber-lines, diffuser hub's and tip's camber-lines, the circumferential relation between the splitter and the main blades in impeller, and the splitter leading edge's cut position. All 37 parameters are listed in detail in Fig.9, where MB is the main blade in impeller, SB is the splitter blade in impeller, DB is the diffuser blade, hub is the hub section, tip is the tip section, beta is the control parameter of camber-line slope in degrees, ST is the circumferential starting theta angle of camber-line, bldn is the blade number. With attention, 3 diffuser outlet control points should maintain to preserve the outlet gas angle and not change too much, although there are 9 control points in both impeller and diffuser blade. That is why we only find DBhubbeta1 ~ DBhubbeta6 and DBtipbeta1 ~ DBtipbeta6 in Fig.9. We considered the variables of blade number, but they are not to be optimized in the present research work.

	Parameter	Var	Obj	Type	Lower Bound		Current Value		Upper Bound
1	MBbldn			INTEGER	12	≤	15	≤	17
2	MBhubbeta1	✓		REAL	-67.0	≤	-62.0	≤	-57.0
3	MBhubbeta2	✓		REAL	-56.5	≤	-51.525857451099	≤	-46.5
4	MBhubbeta3	✓		REAL	-48.3	≤	-43.330128979869	≤	-38.3
5	MBhubbeta4	✓		REAL	-47.1	≤	-42.118747489526	≤	-37.1
6	MBhubbeta5	✓		REAL	-45.7	≤	-40.716814007610	≤	-35.7
7	MBhubbeta6	✓		REAL	-44.7	≤	-39.727172851562	≤	-34.7
8	MBhubbeta7	✓		REAL	-44.4	≤	-39.373264312744	≤	-34.4
9	MBhubbeta8	✓		REAL	-44.6	≤	-39.572007648646	≤	-34.6
10	MBhubbeta9	✓		REAL	-46.0	≤	-41.0	≤	-36.0
11	MBtipST	✓		REAL	-8.0	≤	-2.969154574	≤	2.0
12	MBtipbeta1	✓		REAL	-70.0	≤	-65.0	≤	-60.0
13	MBtipbeta2	✓		REAL	-69.0	≤	-63.964845657340	≤	-59.0
14	MBtipbeta3	✓		REAL	-67.0	≤	-62.041631519794	≤	-57.0
15	MBtipbeta4	✓		REAL	-62.0	≤	-57.989379882812	≤	-52.0
16	MBtipbeta5	✓		REAL	-54.9	≤	-49.865162909030	≤	-44.9
17	MBtipbeta6	✓		REAL	-50.4	≤	-45.435928344726	≤	-40.4
18	MBtipbeta7	✓		REAL	-47.1	≤	-42.1640625	≤	-37.1
19	MBtipbeta8	✓		REAL	-45.6	≤	-40.606310829520	≤	-35.6
20	MBtipbeta9	✓		REAL	-46.0	≤	-41.0	≤	-36.0
21	SBhubRL	✓		REAL	0.15	≤	0.25	≤	0.35
22	SBtipRL	✓		REAL	0.2	≤	0.3	≤	0.4
23	RatS2M	✓		REAL	0.4	≤	0.5	≤	0.6
24	DBBldn			INTEGER	32	≤	37	≤	42
25	DBhubbeta1	✓		REAL	62.0	≤	67.0	≤	72.0
26	DBhubbeta2	✓		REAL	60.5	≤	65.57109	≤	70.5
27	DBhubbeta3	✓		REAL	57.4	≤	62.372408	≤	67.4
28	DBhubbeta4	✓		REAL	53.5	≤	58.534992	≤	63.5
29	DBhubbeta5	✓		REAL	48.2	≤	53.216421	≤	58.2
30	DBhubbeta6	✓		REAL	46.6	≤	51.577126	≤	56.6
31	DBtipST	✓		REAL	-6.2	≤	-1.2	≤	4.2
32	DBtipbeta1	✓		REAL	66.0	≤	71.0	≤	76.0
33	DBtipbeta2	✓		REAL	60.4	≤	65.444495	≤	70.4
34	DBtipbeta3	✓		REAL	55.0	≤	59.912499	≤	65.0
35	DBtipbeta4	✓		REAL	46.1	≤	51.142847	≤	56.1
36	DBtipbeta5	✓		REAL	42.3	≤	47.342143	≤	52.3
37	DBtipbeta6	✓		REAL	41.0	≤	46.011643	≤	51.0

Fig.9 List of design variables

Optimization Design Results

The optimization for impeller and diffuser was finished in about 25 days of CPU time with the established optimization design system. The history of weighted objective function is shown as in Fig.10 and Fig.11. The variation of objective function is shown as in Tab.1. Where "ORG" means the original design, "OPT" means optimum design, "CHG" means the difference between original and optimum design.

Tab.1 Comparison of the optimum and original design

	η_{stall}	η_{peak}	η_{design}	M_{stall}	M_{peak}	M_{design}	P_{stall}	P_{peak}	P_{design}
ORG	0.8182	0.8227	0.8148	1.70	1.78	1.83	5.062	4.973	4.801
OPT	0.8328	0.8377	0.8331	1.695	1.771	1.826	5.020	4.944	4.806
CHG	+1.8%	+1.8%	+2.2%	-0.3%	-0.5%	-0.2%	-0.8%	-0.6%	0.1%

Conclusion

In this paper, optimization of the centrifugal compressor was made. During the optimizing process, an object function considering not only efficiency but also pressure ratio and mass flow rate was introduced; to obtain a more practicable result, performance under three typical working conditions were considered synthetically. The optimum result shows that not only the performance at design condition but also at off-design condition was improved. The optimization system in this paper is adaptive for the full-condition optimization of centrifugal compressors.

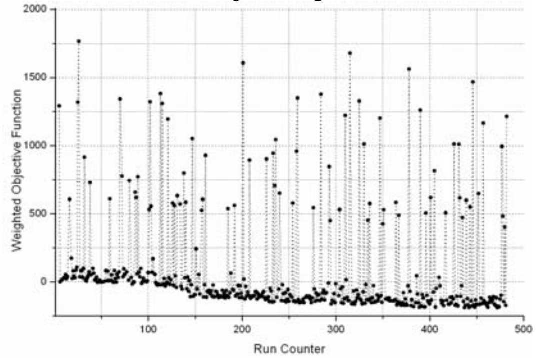

Fig.10 Optimization objective function history

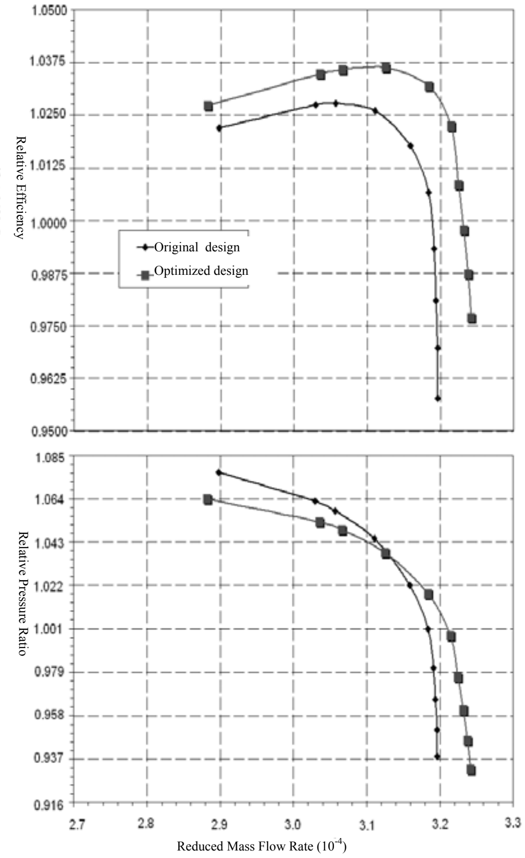

Fig.11 Comparison of original and optimized design for full condition performance curve

Acknowledgements

This work was partly supported by the National Natural Science Foundation of the People's Republic of China (Grant No. 50676043) and the Special Funds for Major State Basic Research Projects (Grant No. 2007CB210108).

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