◎ 논 문

원시험 데이터 일반화를 적용한 원심압축기 설계

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Application of Generalized Experimental Data Correlation in Centrifugal Compressor Design

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Key Words: Centrifugal Compressor(원심압축기), Gas Turbine Engine(가스터빈 엔진), Turbo-Shaft Engine(터보 샤프트 엔진), Aerodynamic Design(공력 설계)

ABSTRACT

Recently, KARI(Korea Aerospace Research Institute, Korea) and CIAM(Central Institute of Aviation Motors, Russia) have made an effort in developing a centrifugal compressor for a small gas turbine engine as part of a collaboration program. This compressor has been designed as a sub-component for an axial-centrifugal compression system for a small turbo-shaft engine aiming adiabatic efficiency higher than 0.81. The geometrical design requirement imposes restrictions to have high inlet hub-to-tip ratio and inlet swirl flow.

In this study, the compressor has been designed using the generalized experimental data established from those compressors having pressure ratio of 3.7 to 5. From this generalized empirical correlation, desirable values of design parameters could be obtained. Subsequently, quasi-3D and 3D viscous flow analyses have been performed to ensure the adopted methodology.

It is expected that the centrifugal compressor provides total pressure ratio of 4.89, corrected mass flow-rate of 1.64kg/sec, and adiabatic efficiency of 0.815 with inlet hub-to-tip ratio of 0.641. These relatively high total pressure ratio and inlet hub-to-tip ratio are the main distinctive features in this design. Besides, one of the main features of this centrifugal compressor is the adoption of a double-row bladed diffuser to effectively decelerate the transonic flow leaving the impeller. The compressor has been manufactured and will be tested in the near future.

1. Introduction

During a past few decades, a remarkable progress in the centrifugal compressor design and performance prediction has been established accompanying numerous scientific publications.[1~4] From the accumulated experimental data, designers can make various generalizations on design parameters, which can be used for the centrifugal compressor design. A curve which shows the state-of-the-art efficiency trend of centrifugal compressors has been proposed by Japikes et al.[5] (solid line on Fig. 1). The symbols shown in Fig. 1 correspond to some of the centrifugal stages recently

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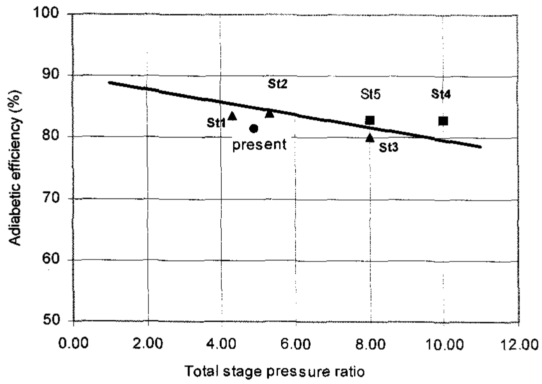

Fig. 1 State-of-the art efficiency trend of centrifugal stages

developed in Central Institute of Aviation Motors in Moscow. The symbols marked in triangles correspond to the stages which already have been tested, while the squares correspond to the stages under development.

Each of these stages has its own application and definite distinction. For instance, St1 stage has a high value of inlet hub-to-tip ratio ($d_{rel}=D_{1hub}/D_{1tip}=0.773$). Thus, it has its own design restrictions to achieve high efficiency. Nevertheless, the maximum efficiencies of St1~St5 are coincident with the state-of-the-art efficiency level. It is believed that the high efficiency of compressors can be achieved using the design principles based on experimental data generalizations, which will be described in this paper.

The present study includes the description of a centrifugal compressor design using generalized experimental data for a small gas turbine engine.

2. Experimental Generation

The aim of an one dimensional design process is to determine main geometrical and flow parameters for each section of the centrifugal stage (i.e. impeller inlet/outlet, diffuser inlet/outlet, etc.) for given total pressure ratio(π^*), corrected mass flow rate(G_{corr}), corrected rotational speed(n_{corr}), absolute inlet flow angle (α_{1avg}), specific outlet velocity(λ_6, $\lambda \equiv c/\sqrt{2k\cdot R\cdot T^*/(k+1)}$, c=absolute velocity of flow, T^*=total temperature), inlet hub diameter(D_{1hub}), and operating fluid properties. In addition to these values, a designer shall determine other geometrical parameters and blade losses according to design requirements, design experiences, and experimental data obtained from prototypes if available. In this paper some of the generalized parameters for the centrifugal compressor design are introduced.

2.1 Specific speed n_s

One of the main parameters defining centrifugal stage geometry and efficiency is the specific speed n_s which depends on volumetric flow-rate Q(m^3/s), adiabatic stage head H_{ad}(m)($=C_P T^*_1((\pi^*)^{(k-1)/k}-1)/g$, k=specific heat ratio, g=acceleration of gravity) and rotational speed n(rpm).

$$n_s = n\cdot Q^{1/2}/(H_{ad})^{3/4} \tag{1}$$

Balje[6] proposed a curve which represents efficiency trend depending upon specific speed n_s. Similarly, Fig. 2 represents the efficiency trend of radial and back-swept impellers (i.e. impeller exit blade angle, β_{2b}=90° represents radial impeller) based on experimental data from CIAM. According to Fig. 2, the optimum value of n_s is located around 40~60. As the specific speed (n_s) increases greater than optimum, the impeller loss becomes larger due to high values of inlet relative flow velocity (W_1), while as the specific speed decreases less than optimum, efficiency will drop due to the disk friction and leakage losses

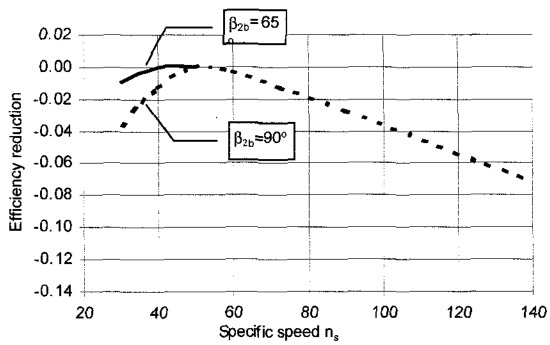

Fig. 2 Variation of impeller adiabatic efficiency vs. specific speed n_s

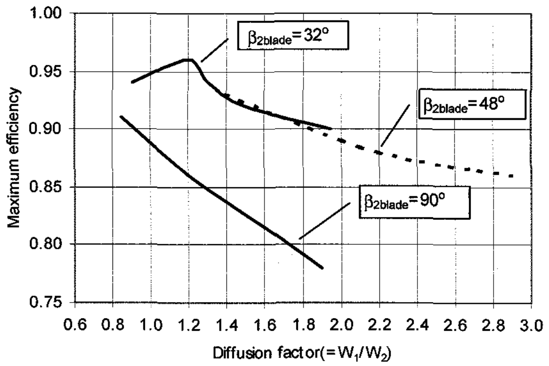

Fig. 3 Maximum adiabatic efficiency of impeller vs. diffusion factor

(α_f=impeller disk friction loss coefficient). In general, all these three values (i.e. n_s, W_1, α_f) are input data in the centrifugal stage design. From this chart, maximum efficiency can be estimated by selecting an optimum rotational speed when mass flow rate and total pressure ratio are given.

It is also useful to evaluate impeller efficiency depending on the impeller pressure ratio. Considering the fact that the impeller diffusion factor is primarily a function of the impeller pressure ratio, Fig. 3 represents a useful guide for the estimation of impeller efficiency. Obviously impeller efficiency has to be refined during further design iteration after the impeller geometry becomes substantially distinctive by geometrical parameters such as back-swept angle, inlet hub-to-tip ratio, tip clearance, blade thickness and so on.

2.2 Blockage

A designer needs to consider flow-path blockage of impeller in determining the number of impeller blades and thickness. Blockage may be specified at inlet and outlet using their initial values from experience.

Determination of blockage due to near-wall boundary layers is a complex problem. The inlet blockage factor ($K_{g1}=1-A_{eff}/A_g$, A_{eff}=effective area, A_g=geometric area) depends on the flow developed ahead of an impeller. In case of a smooth axial annular duct, K_{g1} is usually no more than 3%. If IGV exists, it may increase up to 5%. Outlet blockage (K_{g2}) is determined by a number of governing factors (i.e. impeller dimensions, blade height, diffusion factor, ratio of maximum velocities on suction and pressure sides of blades, Reynolds number, etc.).

2.3 Slip factor

The choice of a proper slip factor is very important for the calculation of impeller loads. A number of formulas have been proposed to evaluate the slip factor. A formula proposed by Lifshits(7), which is based on a relative vortex hypothesis considering blockage and viscous effects, is adopted in this study.

2.4 Vaneless diffuser

Despite its geometric simplicity, flow within a vaneless diffuser is rather complex. Thus, theoretical investigation is possible only under considerable assumptions. An experimental generalization on the loss coefficient versus equivalent divergence angle is presented in Fig. 4 The loss coefficient, ζ, is defined as follows;

$$\varsigma = H_{2\text{-}3}/(0.5\rho \cdot c_3^{\,2}) \qquad (2)$$

where, $H_{2\text{-}3}$ represents the work done by friction within vaneless diffusers. The equivalent divergence angle, θ, is defined as follows;

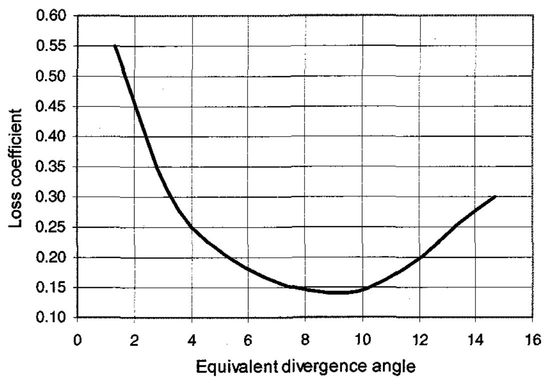

Fig. 4 Loss coefficient of equivalent diffuser

$$\tan(\theta/2) = 2\cdot\sin\alpha_{avg}\cdot (\sqrt{D_3\cdot b_3\cdot\sin\alpha_3} - \sqrt{D_2\cdot b_2\cdot\sin\alpha_2})/(D_3 - D_2) \quad (3)$$

where $\alpha_{avg} = (\alpha_3 + \alpha_2)/2$.

2.5 Bladed diffuser

The flow in bladed diffuser is influenced by a variety of factors such as diffuser channel divergence angle, inlet flow specific speed, channel length, leading and trailing edge thickness, etc.. The channel divergence angle and the flow specific speed at bladed diffuser inlet (λ_3) influence much on diffuser losses. A designer has to match well between impeller and bladed diffuser to provide required stall margin and stage efficiency (η=adiabatic efficiency).

Generalizations of a pressure recovery coefficient, $\sigma_{3\text{-}4}$ ($=P^*_3/P^*_4$), versus channel divergence, $A_{3\text{-}4}$ ($=A_4/A_3$), and flow specific speed, λ_3, are presented on Fig. 5. To provide effective flow deceleration under given value of $A_{3\text{-}4}$, the design of a bladed diffuser has to meet a number of restrictions such as throat area ratio, ($A_{th\text{-}rel}=A_{th}/A_3$, A_{th}=throat area), cascade solidity, (b/t_3), divergence angle of an equivalent diffuser, (θ_3), turning angle, ($\Delta\alpha_{3\text{-}4}$), and so on.

It should be noted that high pressure ratio centrifugal stages are frequently designed with double-row bladed diffusers to decelerate the impeller exit flow effectively.

Double-row bladed diffusers redistribute blade load by circumferentially shifting the 2nd row relative to the 1st row. Proper load redistribution brings more uniform velocity profile at the bladed diffuser outlet.

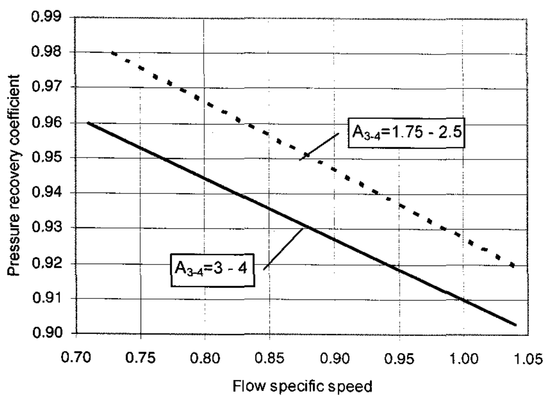

Fig. 5 Total pressure recovery vs. flow specific speed

3. Design of a Centrifugal Compressor for a Small Gas Turbine

Using the methodology of experimental generalization, a centrifugal compressor, which shall work with axial compressors in a small gas turbine engine, has been designed. Design requirements for the centrifugal compressor are presented in Table 1. As a first step of the development process, the inlet condition of centrifugal compressor was assumed uniform.

From the one dimensional streamline analysis, a geometric configuration and flow parameters of the centrifugal compressor have been obtained. Subsequently, correction and evaluation of losses, impeller profiling, axis-symmetrical flow analysis, and quasi-3D flow analysis followed by 3D Navier-Stokes calculation with refinement of blade profiling have been performed. Refinement of impeller blade profiling was achieved through the iterative process of the aerodynamics and structural analysis, because strength calculations take into account not only mass forces but also aerodynamic loading and flow temperature.

The high levels of stresses were predicted at the root of entrance part of main blades. Thus, the inlet half of the main blades were modified with thickness and leaning. Also, the main blade and the splitter have been thickened on the outlet tip from the initial distribution.

Table 1 Design requirement of a centrifugal compressor

Parameter of centrifugal compressor	Given value
Design point total pressure ratio (π^*)	4.89
Adiabatic efficiency (η)	0.8
Corrected flow-rate (G_{corr})	1.64 kg/s
Inlet flow angle (α_1)	62.2°
Flow angle at compressor outlet	90°
Corrected rotational speed (n_{corr})	36762 rpm
Hub-to-tip ratio at inlet (d_{rel})	0.641

Table 2 Final configurations of a centrifugal compressor and design parameters

Parameters	Flow-path section					
	1	2	3	4	5	6
Total pressure (kPa)	101.3	561.2	539.4	500.7	498.2	495.7
Static pressure (kPa)	87.1	285.3	308.4	486.2	483.8	490.0
Total temperature (°K)	288.1	491.9	491.9	491.9	491.9	491.9
Flow angle α (absolute motion)	62.2°	14.94°	16.42°	27.58°	39.35°	90.00°
Specific flow velocity λ	0.505	1.027	0.941	0.224	0.224	0.141
Relative λ_{IW} at impeller inlet	(λ_{IW})mean = 0.702 (λ_{IW})tip=0.815					
Other parameters	Corrected flow rate: 1.64 kg/s Adiabatic efficiency: 0.815 Total pressure ratio: 4.89 Adiabatic total head of stage: 0.61 Specific speed: 30.13 Impeller parameters: π_2=5.54, η_2=0.895, β_{2b}=51.5°					

In the design process of the bladed diffuser, the relative position of the 1st and the 2nd row in circumferential direction needs to be carefully determined since it influences on the flow non-uniformities at the bladed diffuser outlet. The best performance seems to be achieved when the 1st row of blades is located 10% pitchwise from the pressure surface of 2nd blade row according to Selnezniov et al.[8]. The final configurations of the centrifugal compressor are presented in Table 2.

The designed configuration of the centrifugal stage (impeller + vaneless diffuser + double-row vaned diffuser) has been brought into further numerical study using 3D Navier-Stokes viscous flow solver.

Fig. 6 shows a result of 3D Navier-Stokes calculation of a full viscous flow through the centrifugal stage. It shows the circumferentially averaged Mach number levels. It can be seen that the flow at the bladed diffuser inlet is subsonic and the hub-to-tip velocity distribution is reasonably uniform considering the pressure ratio and the diameter ratio of the diffuser, which represents good matching between impeller and diffuser. Besides, the flow at the bladed diffuser outlet seems relatively uniform, too. It presents good expectation of the downstream flow through radial-axial bend and axial guided vanes.

Fig. 7 demonstrates 3D viscous flow field in the mid surface of the bladed diffuser channel. It can be seen that the Mach number contours in the channel of the 1st row are approximately normal to the streamwise direction, which is desirable because it will result in low losses. The minimum Mach number on pressure surface of the 1st row profile is about 0.2, and it shows no flow separation. It means that the flow decelerates effectively in the region of the 1st row of blades.

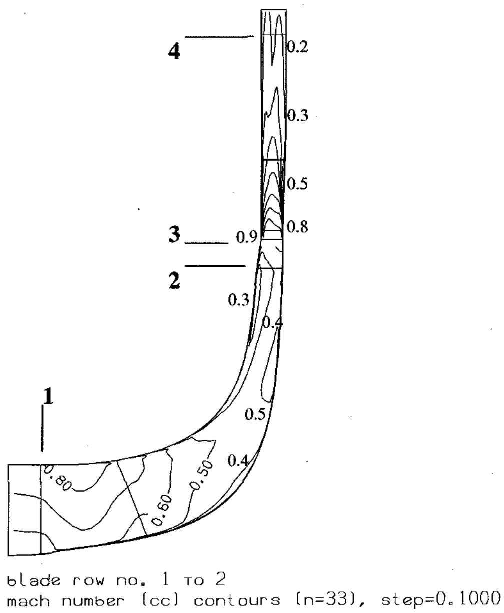

Fig. 6 3D Mach number contour in a meridional plane of a compressor stage by 3D Navier-Stokes calculation

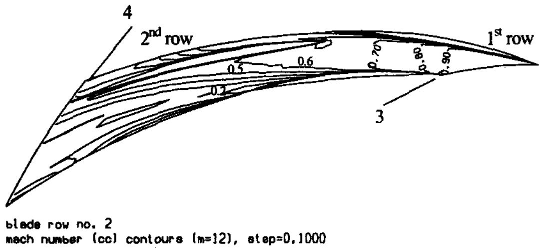

Fig. 7 3D Mach number contour in a blade-to-blade plane of a double row bladed diffuser by 3D Navier-Stokes calculation

4. Conclusion

A centrifugal compressor for a small gas turbine engine has been designed using the methodology of experimental data generalization. According to a series of calculation procedure (i.e. 3-D inviscid Euler and viscous Navier-Stokes analysis), the one dimensional design procedure using the generalized experimental data presented in this paper can be used as a useful tool in designing the up-to-date levels of centrifugal compressors. Further manufacturing and testing of the designed compressor will evaluate the predicted performance of the centrifugal compressor and the proposed design methodology.

References

(1) Oh, K. T., Kang, D. J., and Jung, P. S., 1997, "A Study on the Performance Prediction of Centrifugal Compressor for Turbocharger," Transactions of the KSME B, Vol. 21 pp. 582~588.

(2) Oh, Hyoung Woo, Yoon, Eui Soo and Chung, Myung Kyoon, 1997, "Performance Prediction and Loss Analysis of Centrifugal Compressors," Transactions of the KSME B, Vol. 21, pp. 804~812.

(3) Aungier, R. H., 1995, "Mean Streamline Aerodynamic Performance Analysis of Centrifugal Compressors," ASME Journal of Turbomachinery, Vol. 117, pp. 360~366.

(4) Galvas, M. R., 1973, Fortran Program for Predicting Off-Design Performance of Centrifugal Compressors, NASA TN D-7473.

(5) Japikse, D. and Osborn, C., 1994, The Technology of Centrifugal Compressors: Design Approach and New Prospects, Proceedings of Turbomachinery Conference, Russia, Kazan.

(6) Balje, O. E., 1981, Turbomachines, a Guide to Design, Selection and Theory, John Wiley & Sons, New York.

(7) Lifshits, S. P., 1966, Aerodynamics of Centrifugal Compressors, Mashinostroenie, Moscow-Leningrad. (in Russian).

(8) Selezniov, K P. and Galerkin, Yu. B., 1982, Centrifugal Compressors, Mashinostroenie, Leningrad. (in Russian).