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Original Paper

The Performance Analysis Method with New Pressure Loss and Leakage Flow Models of Regenerative Blower

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Abstract

For efficient design process of regenerative blower, the present study provides new generalized pressure and leakage flow loss models, which can be used in the performance analysis method of regenerative blower. The present performance analysis on designed blower is made by incorporating momentum exchange theory between impellers and side channel with mean line analysis method, and its pressure loss and leakage flow models are generalized from the related fluid mechanics correlations which can be expressed in terms of blower design variables. The present performance analysis method is applied to four existing models for verifying its prediction accuracy, and the prediction and the test results agreed well within a few percentage of relative error. Furthermore, the present performance analysis method is also applied in developing a new blower used for fuel cell application, and the newly designed blower is manufactured and tested through chamber-type test facility. The performance prediction by the present method agreed well with the test result and also with the CFD simulation results. From the comparison results, the present performance analysis method is shown to be suitable for the actual design practice of regenerative blower.

Keywords: Regenerative Blower, Performance Analysis, Pressure Loss, Leakage Flow, CFD

1. Introduction

Regenerative blowers are usually operated with high pressure rise at low flow capacity, so widely used for automotive, environmental and fuel cell applications. However, because regenerative blowers are operating with low efficiency or a lot of pressure loss resulted from the internal flow phenomena of regenerative blower[1,2], there are growing industrial needs of the reliable design method for minimizing the pressure loss as well as achieving high blower efficiency. Generally, it is known that the reliability of regenerative blower design method is strongly dependent on the prediction accuracy of the performance analysis model used in design practice.

The early theoretical researches on the performance predictions of regenerative blower and pump have been conducted to investigate the flow pattern and the energy transfer mechanism of fluid inside the machines, and showed that the energy transfer to fluid is achieved by the momentum exchange of the helical-toroidal fluid motion between rotating impeller and fixed side channel of regenerative machine [3,4]. Recent researches by Badami and Mura[5,6] have been devoted to improving the performance analysis method based on the momentum exchange theory, and their prediction results have been compared with test and CFD results. However, the most of the previous performance prediction models require model coefficients which designer should specify, so their model coefficients have been derived generally from the experiment of similar machines and then set to constant values. For this reason, it is very difficult to apply the previous prediction models in new blower design when there are no existing reference blowers. Therefore blower designers of industry call for more generalized prediction model which can reflect blower design and operation parameters in determining its model coefficients.

In the present study, a simple but more reliable performance prediction method is developed for the application to any regenerative blower design. Regenerative blower performance is predicted by incorporating the well-known momentum exchange theory between rotating impeller blades and fixed side channel of blower[5] with pressure and leakage flow loss models. The present study generalizes model coefficients for pressure loss and leakage flow predictions, which are correlated in terms of blower design variables and operation condition by using the experiments on relevant regenerative machines and flow elements.

The performance prediction accuracy of the present method is verified by comparing the prediction results with the measurement results of several actual regenerative blowers. Furthermore, with the use of the present design-analysis method, a

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new regenerative blower is designed, manufactured and tested by using chamber-type test facility, and its internal flow field is analyzed by the CFX code, a CFD code specialized for fluid machinery. The comparison between the present performance prediction, the CFD simulation and the test results of newly designed blower are made to verify the suitability and the prediction accuracy of the the present method used in actual design practice of regenerative blower.

2. Blower Design and Analysis Methods

2.1 Blower Design Method

In general, regenerative blower is composed of rotating impellers and fixed side channel and its geometry and design specifications are shown in Fig. 1[5]. As depicted in Fig.1, the main design variables of rotating impellers and fixed side channel can be defined as follows:

- Rotation speed (N)
- Tip diameter (D_2=2r)
- Channel height (h)
- Channel width (W)
- Impeller blade inlet angle (β_1)
- Impeller blade outlet angle (β_2)
- No. of impeller blades (Z)
- Impeller blade thickness (d)
- Axial clearance(c)
- Extension angle ($\boldsymbol{\theta_c}$)

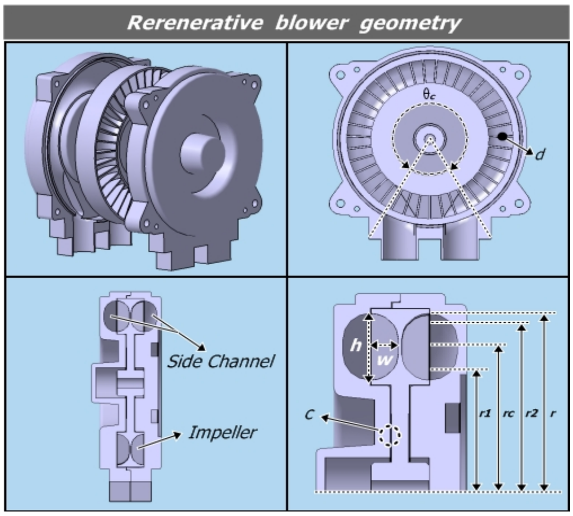
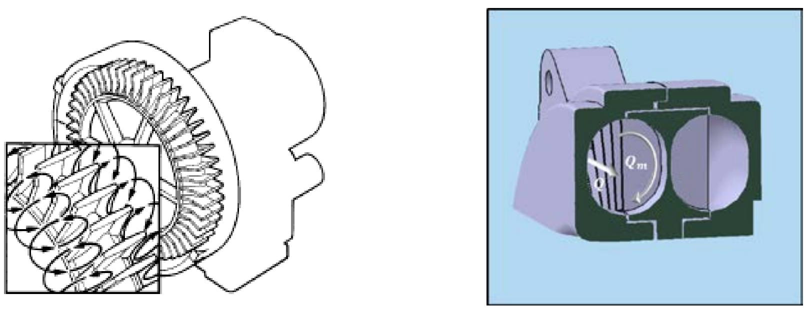


Fig. 1 Geometry and design variables of regenerative blower

Once the blower design variables are defined, 3-D blower geometry design can be easily made and then its dimensions can be used for the input data of performance analysis and CFD simulation.

2.2 Blower Performance Analysis Method

As shown in Fig. 2, the gas flow inside regenerative blower shows typically three dimensional and helical-toroidal motion where fluid rotates in and passes along the space between rotating impeller blades and fixed side channel. The present study assumes mean streamline as the representative one of the three dimensional fluid flow phenomena.



(a) Flow behavior inside regenerative blower (b) Cross section view on helical-toroidal flow

Fig. 2 Flow pattern of regenerative blower

Thus, in the present study, the performance of blower is analyzed by combining the mean line analysis method for fluid flow and the momentum exchange theory between impellers and side channel as follows:

2.2.1 Pressure Rise of Blower

Through the momentum exchange between fluid and impeller due to the flow motion in Fig.2(b), gas is gradually pressurized along tangential flow path, and its overall pressure rise(Δp_s) and pressure rise coefficient(ψ) can be calculated by

$$\psi = \frac{\Delta p_s}{\frac{1}{2}\rho u^2} = 2\frac{Q_m}{A_c u}\frac{r_1}{r_c}\left(\frac{r_2}{r_1}\frac{C_{u2}}{u} - \frac{C_{u1}}{u}\right) - K_p\varphi^2 \tag{1}$$

where

$$\frac{C_{u2}}{u} = \frac{r_2}{r}\left(1 - \frac{\Delta u_2}{u_2}\right) + \frac{A_c}{A_2}\left(\frac{Q_m}{A_c u}\right)\cot\beta_2 \qquad \frac{\Delta u_2}{u_2} = \frac{1.5 + 1.1(2 - 2\beta_2/\pi)}{Z[1 - (r_1/r_2)^2] + 1.5 + 1.1(2 - 2\beta_2/\pi)}$$

$$\frac{C_{u1}}{u} = \frac{r_1}{r}\phi \qquad \varphi = \frac{Q}{uA_c}$$

and

$$\frac{1}{2}\left[K_m + \frac{1}{\sin^2\beta_2} + \left(\frac{A_2}{A_1}\right)^2\cot^2\beta_1\right]\left(\frac{A_c}{A_2}\right)^2\left(\frac{Q_m}{A_c u}\right)^2 + \left(\frac{r_1}{r} - \frac{u_{c1}}{u}\right)\cot\beta_1\frac{A_c}{A_1}\left(\frac{Q_m}{A_c u}\right)_2 + \frac{1}{2}\left[\frac{r_1^2 - r_2^2}{r^2} + \left(\frac{u_{c2}}{u_1}\right)^2 - \left(\frac{u_{c1}}{u}\right)^2\right] = 0$$

Here ρ, u, Q_m, φ and A_c are fluid density, impeller tip speed, flow capacity, flow coefficient and side channel area, while K_p is the pressure loss coefficient related with the fluid friction on the side channel wall and K_m is that due to circulating flow loss. More detailed variable definitions and mathematical description about momentum-exchange theory are referred to Badami and Mura[4].

2.2.2 Leakage Flows of Blower

In order to calculate the actual pressure rise and efficiency characteristics, the leakage flow through the clearance between the impeller and the casing should be considered. The actual flow coefficient(φ_a) can be determined as

$$\varphi_a = \varphi - \varphi_f \tag{2}$$

where φ is the theoretical flow coefficient and φ_f is the leakage one.

The leakage flow can be categorized into two different parts. One is the leakage flow through the clearance between the impeller disk and the casing wall(Q_{f1}), which can be obtained by

$$\varphi_{f1} = \frac{Q_{f1}}{A_c u} = \frac{A_{f1}}{A_c}\frac{1}{\sqrt{K_{f1}}}\sqrt{\frac{\psi}{2}} \tag{3}$$

where A_{f1} is the leakage section area of 2(r-h)c and K_{f1} is the loss coefficient related with the leakage flow.

Another leakage flow(Q_{f2}) is produced through the stripper clearance due to the pressure difference between the inlet and the outlet of the blower, and can be expressed by the following equation:

$$\varphi_{f2} = \frac{Q_{f2}}{A_c u} = \frac{A_{f2}}{A_c}\left(\frac{1}{\sqrt{K_{f2}}}\sqrt{\psi} + \frac{1}{2}\frac{r_{mv}}{r}\right) \tag{4}$$

where A_{f2} is the leakage section area of hc, r_{mv} represents the mean radius of the impeller vanes and K_{f2} is the pressure loss coefficient due to the leakage flow.

2.2.3 Power and Efficiency of Blower

The hydraulic power transferred to the gas is given by

$$P_h = \rho Q\Delta p = \frac{1}{2}\rho A_c u^3\varphi\psi \tag{5}$$

and power input to the impellers is written by

$$P_i = T_i\omega = \frac{1}{2}\rho A_c u^3\frac{r_c}{r}(\psi + K_p\varphi^2) \tag{6}$$

Therefore, the blower efficiency can be computed from the following definition as

$$\eta = \frac{P_h}{P_i} = \frac{r}{r_c}\frac{\varphi\psi}{\psi + K_p\varphi^2}$$

(7)

2.2.4 Generalized Models for the Pressure Losses and the Leakage Flows of Blower

The four coefficients(K_i) related with pressure losses and leakage flows are represented in the equations of (1)-(7), and they have strong influence on the prediction results of overall pressure rise, leakage flow and blower efficiency. As mentioned before, the most of previous performance analysis researches set these coefficients as constants although their blower models have different design variables and operation conditions. For example, Badami and Mura simply set K_m and K_p as 1 and 4.7[6]. However, changing these coefficients more or less, the computed results on the pressure and the efficiency of the same machine could show different characteristics, which might be agreed or disagreed with the test results. For this reason, it is necessary to generalize and formulate the coefficients in terms of blower design and operation parameters.

Therefore, in the present study, new pressure loss and leakage flow models are constructed, for the general use in blower design, by formulating the correlations corresponding to pressure loss and leakage flow phenomena as follows:

(a) Circulating flow loss coefficient(K_m)

According to the experiment by Choi et al.[7], the pressure loss of the circulating flow between impeller and side channel is mainly dependent on blade height(h) and width(W). Their experimental results can be correlated as the following equation:

$$K_m = 4 - 4.583\left(\frac{2W}{h}\right) + 2.083\left(\frac{2W}{h}\right)^2 \tag{8}$$

(b) Tangential flow loss coefficient(K_p)

The pressure loss coefficient of the tangential through-flow inside blower can be approximated by the sum of the bend flow turning at (2π-θ_c) and the entry and discharge flows as follows:

$$K_p = 0.0175 f_p(\mathrm{Re}_{D_h})\theta_c \frac{r_c}{D_h}\left[0.7 + 0.35\frac{\theta_c}{90}\right]\left[\frac{0.21}{\left(r_c / D_h\right)^{0.25}}\right] + 2\left[0.95 + \frac{33.5}{180 - (\theta_c / 2)}\right] \tag{9}$$

where Re_{Dh} and r_c are the Reynolds numbers on the hydraulic diameter and the radius of curvature of the cross section of regenerative blower depicted in Fig. 2(b), and $f_p(Re_{Dh})$ can be calculated by Moody chart[8]. It is noted that the first term of equation (9) represents the pressure loss coefficient of the bend flow while the second term representing that of the entry and discharge flows.

(c) Casing-impeller leakage flow(K_{f1})

Previous CFD studies by Kang and Shim[9] showed that the pressure loss coefficient (K_{f1}) of the leakage flow through the clearance between the impeller disk and the casing wall is a function of clearance size(c) as follows:

$$K_{f1} = \frac{1.4(1 + 0.01R_v^2)}{\mathrm{Re}_{D_h}^{0.1}(r / c)^{0.25}} \tag{10}$$

where R_v is the dimensionless rotation speed defined as u/V and u, V are the rotation speed, the mean flow velocity through the clearance between impeller and casing.

(d) Stripper leakage flow(K_{f2})

According to another experimental result on regenerative fuel pump by Kang and Lim[10], the pressure loss coefficient of the leakage flow through stripper section is simply approximated as

$$K_{f2} = f_p(\mathrm{Re}_{D_h})\frac{L_c}{D_h} \tag{11}$$

where L_c is the mean tangential length of stripper section.

2.3 Blower Performance Analysis Results

Table 1 summarizes the main design variables of four actual regenerative blowers used in verifying the present performance prediction accuracy(Hwang Hae Elec.[11]: Badami and Mura[6]: Gardner Denver[12]). As shown in Table 1, the four blower models have different design specifications, so their pressure loss and leakage flow characteristics would be significantly changed due to corresponding design specifications and operation conditions.

Table 1 Design variables of four regenerative blowers

Design variables	Hawnghae Mini-H200	Hwanghae Mini-H100	Badami - H2	Gardner Denver 28H1002-AA53
Rotation speed, N[rpm]	6,673	7,052	10,000	15,000
Impeller tip diameter, 2r[m]	0.111	0.104	0.138	0.110
Side channel height, h[m]	0.016	0.015	0.024	0.014
Side channel width, w[m]	0.008	0.0075	0.012	0.007
Blade inlet angle, β1[deg]	90.0	90.0	90.0	90.0
Blade outlet angle, β2[deg]	90.0	90.0	90.0	73.5
Extension angle, θc[deg]	290	290	325	290
Axial clearance, c[m]	0.0002	0.0002	0.0002	0.0002
No. of impeller blades, Z	39	39	41	52
Working fluid	Air	Air	Hydrogen	Air

Based on the blower designs in Table 1, the pressure loss and leakage flow coefficients of each blower are computed at design flow conditions by the present method and they are compared with those of Badami's prediction model[5] in Table 2. As known from Table 2, Badami's model constant should be carefully chosen as one value within very wide range by designer, so the chosen values can severely affect the accuracy of performance prediction. However, as shown also in Table 2, the present method can give designer the coefficients specified for blower model, which are calculated automatically from the equations of (8)-(11).

Table 2 Pressure loss and leakage flow coefficients

Blower Model Coefficients	Hawnghae Mini-H200 Model	Hwanghae Mini-H100 Model	Badami – Hydrogen-Fuel Cell Model	Gardner Denver 28H1002-AA53 Model	Badami model constants[3]
K_m	1.500	1.500	1.500	1.479	0.9∼1.0
K_p	2.579	2.695	2.1631	2.744	3.5∼4.7
K_{f1}	53.40	51.71	66.57	48.96	4∼4,800
K_{f2}	34.79	17.26	37.97	15.55	26∼2,650

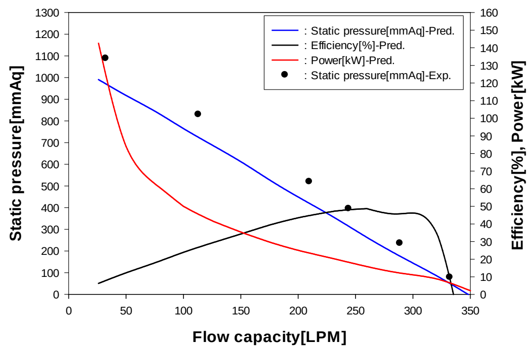


Fig. 3 Performance predictions of Mini-H200 model

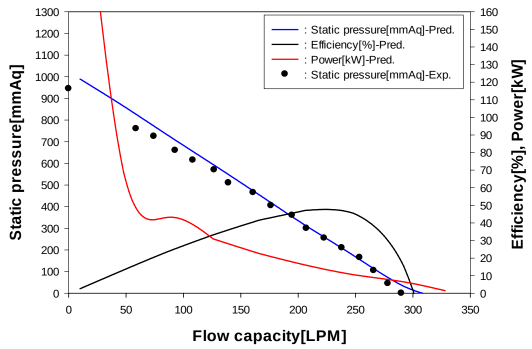


Fig. 4 Performance predictions of Mini-H100 model

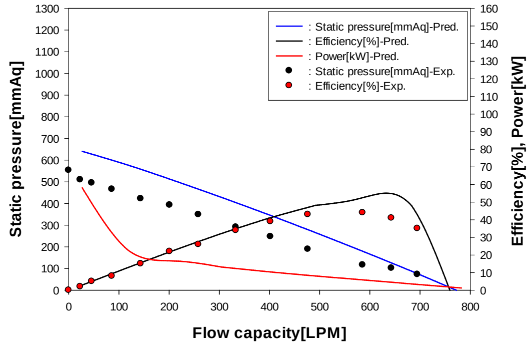


Fig. 5 Performance predictions of Badami model

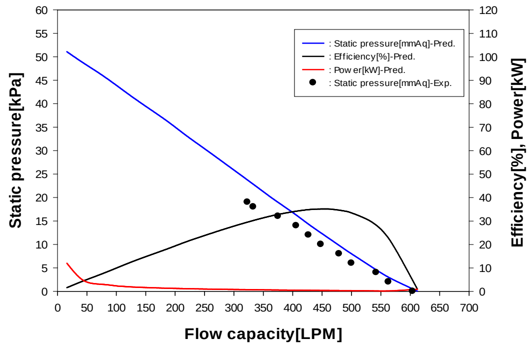


Fig. 6 Performance predictions of Gardner Denver model

Figs. 3-6 show the performance prediction results of Mini H-200, Mini H-100, Badami and Gardner Denver models by the present method, which agrees well with the measurement over entire flow capacity range. These comparison results show the present method would be very suitable for predicting the performance of any blower model in actual design practice.

3. Application of the Present Method to New Blower Development

The present design and analysis methods are also applied to develop a new regenerative blower used for fuel cell application. The design requirements and variables of new blower are summarized as follows:

- Rotation speed = 8000 rpm
- Tip diameter(2r) = 122 mm
- Side channel height(h) = 23 mm
- Side channel width(W) = 9 mm
- No. of impellers(Z) = 39

Based on the design variables listed above, new regenerative blower is designed and manufactured, and its mechanically-machined impellers and side channel are shown in Fig. 7.


Fig. 7 Manufactured model of newly designed blower

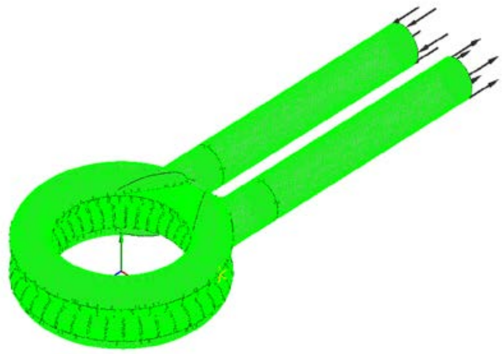


Fig. 8 Mesh system of newly designed blower

From the newly designed blower geometry, CFD modelling and simulation are also conducted to investigate the internal flow field of blower. The present study employs the CFX code, a CFD program specialized for fluid machinery analysis, where 3-D RANS(Reynolds-stress Averaged Navier Stokes equations) solver is coupled with SST(Shear Stress Transport) turbulence model. The mesh generation on rotating impellers and stationary side channel is made, and the interface between rotating and stationary flow surfaces is treated by using frozen rotor scheme[13]. Fig. 8 shows the mesh system for the CFD analysis on the internal flow between rotating impellers and fixed side channel of newly designed blower.

The CFD computation results on the fluid flow and the pressure rise through rotating impellers and side channel are depicted in Fig. 9. As shown in Fig. 9, the predicted streamlines show the fluid flow between impellers and side channel is helical-toroidal motion as expected in Fig.2, and the pressure rise of fluid passing through tangential flow path is linearly increased. Fig. 10 shows the turbulent kinetic energy inside blower, which is produced due the helical-toroidal fluid motion between impellers and side channel and is also linearly increased along tangential flow path like the pressure rise.

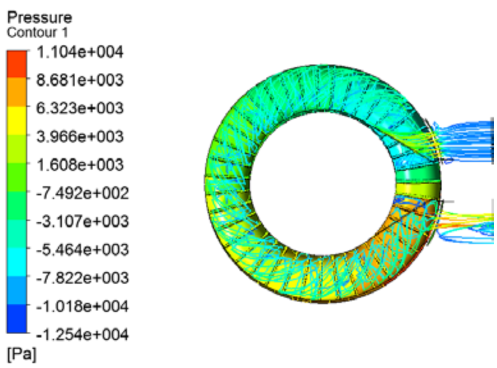


Fig. 9 Streamlines and static pressure of newly designed blower

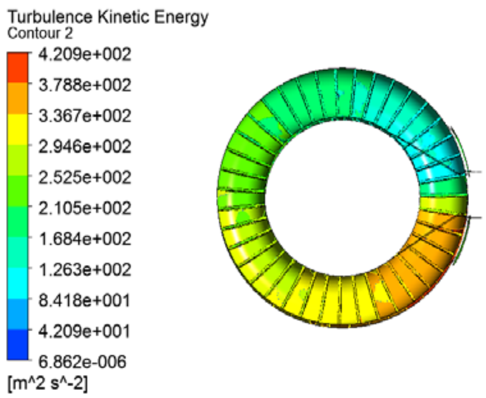


Fig. 10 Turbulent kinetic energy of newly designed blower

The performance of newly designed blower predicted by the present method is also compared with the test results obtained from chamber-type test facility(refer to Fig. 11). As shown in Fig. 11, newly designed blower model is installed at the inlet of the chamber-type performance tester, and all the performance measurements and related calibrations are made according to AMCA standard 210[14]. The comparisons between the prediction, the CFD calculation and the test results are represented in Fig. 12. As shown in Fig. 12, the predicted pressure curve is fairly agreed with the CFD result at design condition, and is also well-agreed with the test results over entire flow range except at very low flow capacity.

Fig. 11 Performance test facility and blower model

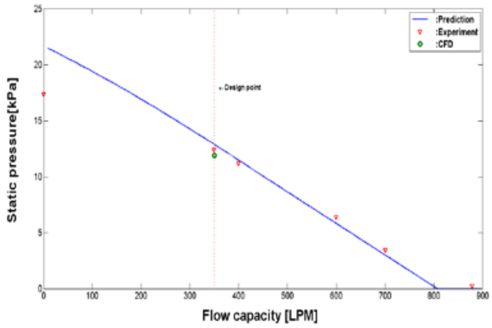


Fig. 12 Pressure curve of newly designed blower

3. Conclusion

The present study develops a design-analysis method applied for the performance prediction in the development process of regenerative blower. The present performance prediction method suggests new pressure loss and leakage flow models which are generalized and expressed in terms of the design variables and the operation conditions of regenerative blower. The present method is used in the performance predictions of four existing regenerative blowers, and then is also applied in developing new regenerative blower used in fuel cell system. The performance prediction results by the present method are well-agreed with the test results within a few percentage of relative error. Therefore the present method is expected to be the reliable design tool suitable for developing regenerative blower.

Acknowledgments

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Nomenclature

A_c	Side channel cross-section area[m^2]
A_1, A_2	Impeller input/output flow area[m^2]
C_u	Tangential absolute fluid velocity[m/s]
c	Clearance[m]
h	Channel height[m]
K	Pressure loss or leakage flow coefficient
P	Power[W]
p_s	Static pressure[Pa, mmAq]
Q	Blower flow rate[m^3/s]
Q_m	Circulatory flow rate[m^3/s]
p_1	Inlet absolute pressure[Pa]
Re_{Dh}	Reynolds number based on hydraulic diameter
r	External radius of the impeller[m]
r_c	Mean radius of the side channel[m]
r_1, r_2	Inlet/outlet mean radii of the meridian flow stream at the impeller[m]
r_{mv}	Mean radius of the impeller vane[m]
u	Impeller tangential velocity[m/s]
u_c	Tangential fluid velocity in the side channel[m/s]
W	Side channel width[m]
z	Number of impeller vanes
β_1	Impeller vane leading edge inclination angle[rad]
β_2	Impeller vane trailing edge inclination angle[rad]
ϕ	Dimensionless flow coefficient($= Q/uA_c$)
ψ	Dimensionless head coefficient($= 2\Delta p_s/\rho u^2$)
ρ	Inlet fluid density[kg/m^3]
θ_c	Angular extension angle of the side channel[rad]
η	Efficiency
ω	Angular speed[s^{-1}]

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