Preliminary Design Efficiency Optimization of 1D Multistage Axial Flow Compressor for a Fixed Distribution of Axial Velocities

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Abstract. A preliminary design efficiency optimization of a multistage axial flow compressor is studied in this paper using one-dimensional flow theory. A model for the optimum design of a multistage axial flow compressor, assuming a fixed distribution of axial velocities is presented. The absolute inlet and exit angles of the rotor of every stage are taken as the design variables. Analytical relations between the isentropic efficiency of the multistage compressor and the flow coefficient, the work coefficient, the flow angles and the degree of reaction of every stage are obtained. The results have some generalizations and can provide some guidance for the optimum design of multistage compressors.

Key words: multistage axial flow compressor; efficiency; optimization

Nomenclature

c	absolute velocity	m/s
F	through-flow area	m^2
G	Air mass flow rate	kg/s
h	enthalpy rise	J/kg
i	specific enthalpy	J/kg
k	velocity coefficient	
n	number of stage	
p	pressure	Pa
R	ideal gas constant	
s	specific entropy	J/(kgK)
u	wheel velocity	m/s
w	relative velocity	m/s

Greek symbols

α	flow angle relative to a stator, ^{0}	
α_z	reheat coefficient	
β	flow angle relative to a rotor , ^{0}	
φ	flow coefficient	
η	adiabatic efficiency	
κ	ratio of specific heats	
ρ	density	kg/m^3
Ω	degree of reaction	
ω	loss coefficient	
ψ	work coefficient	

Subscripts

a	axial direction
c	total performance of a multistage compressor
i	ith stage
j	jth station
opt	optimum
r	rotor
s	stator; isentropic process
u	tangential

Superscripts

*	stagnation condition

1. Introduction

The design of the axial flow compressor is not a precise science. The lack of accurate prediction influences the design process seriously. Until today, there are no methods currently available that guarantee to predict the absolute values of these quantities to a sufficient accuracy for a new design. Some progresses have been obtained via the application of numerical optimization techniques to single- and multi-stage axial flow compressor design, see Refs. [1-22]. Especially with the development of computational fluid dynamics (CFD), many more accuracy methods of calculating have been obtained in many references in which the techniques of computational fluid dynamics have been applied to tow-dimensional and three-dimensional optimum design of axial flow compressor, see Refs. [17-20]. However, it is still of worthwhile significance to calculate using one-dimensional flow-theory for the optimum design of compressors. Boiko [23] presented detailed mathematical model for optimum design of single- and multi-stage axial flow turbines for assuming a fixed distribution of axial velocities, and obtained the corresponding analytical results. Using the similar idea, Chen *et al*. [22] presented a mathematical model for optimum design of single-stage axial flow compressors. In this paper, the results of Ref. [22] are extended to the optimum design of multistage compressors. The model for the optimum design of multistage axial flow compressor for the fixed distribution of axial velocities is presented. Analytical relations between the isentropic efficiency of the multistage compressor and the flow coefficient, the work coefficient, the flow angles and the degree of reaction of every stage are obtained using one-dimensional flow-theory. Numerical examples are provided to illustrate the effects of various parameters on the optimum performance of the multistage compressor.

2. Fundamental equations for elemental stage compressor

Consider a n-stage axial flow compressor. Fig.1 shows the flow path of the n-stage axial flow compressor. Fig. 2 shows the specific enthalpy-specific entropy diagram of the n-stage compressor. For a n-stage axial flow compressor, there are $(2n+1)$ section stations. The stage velocity triangle of an intermediate stage (*i*th stage) is shown in Fig.3. The corresponding specific enthalpy–specific entropy diagram is shown in Fig.4. The calculation of multi-stage performance is performed by using one-dimensional flow theory. The analysis begins from the energy equation and continuity equation, and the axial flow velocities of the working fluid and wheel velocities at the different stations in the compressor are not considered as constant, that is, $u_i \neq u_j$ and $c_i \neq c_j (i \neq j)$, where i denotes *i*th stage, and j denotes *j*th station.

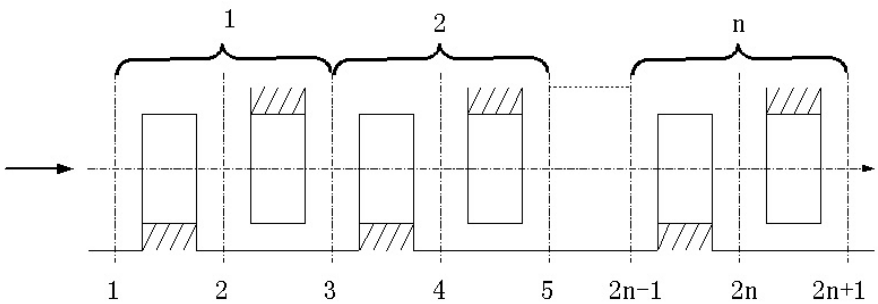

Fig.1 Flow path of a n-stage axial flow compressor

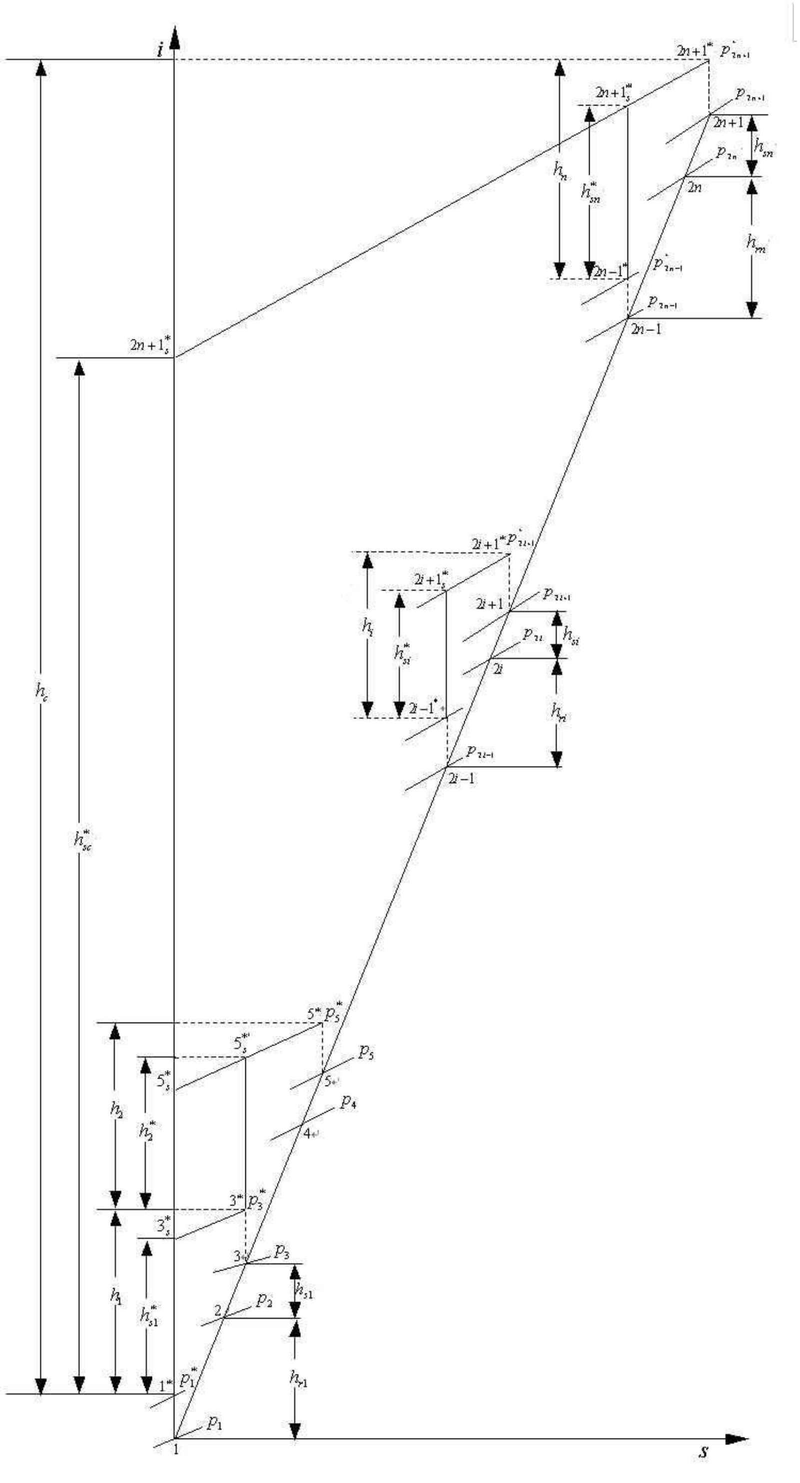

Fig. 2 Enthalpy-entropy diagram of a n-stage compressor

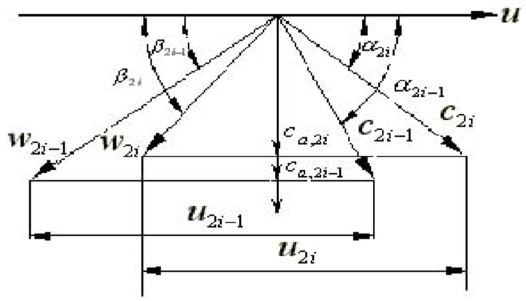

Fig. 3. Velocity triangle of intermediate stage

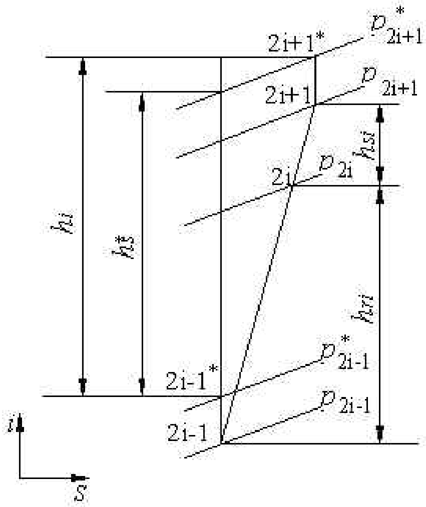

Fig. 4 Enthalpy-entropy diagram of intermediate stage

The major assumptions made in the method are as follows:

1) The working fluid flows stably relative to vanes, stators and rotors, which rotate at a fixed speed.
2) The working fluid is compressible, non-viscous and adiabatic.
3) The mass flow rate of the working fluid is constant.
4) The compression process is homogeneous in the working fluid.
5) The absolute outlet angle of the working fluid in *i*th stage is equal to the absolute inlet angle of the working fluid in $(i+1)$th stage.
6) The effects of intake and outlet piping are neglected.

Therefore, the energy equation for one-dimensional flow of *i*th stage is:

$$h_i = u_{2i} c_{a,2i} ctg\alpha_{2i} - u_{2i-1} c_{a,2i-1} ctg\alpha_{2i-1} \quad (1)$$

The continuity equation is:

$$G = \rho_{2i-1} c_{a,2i-1} F_{2i-1} = \rho_{2i} c_{a,2i} F_{2i} \quad (2)$$

The work required by the rotor is:

$$h_{r,i} = (w_{2i-1}^2 - w_{2i}^2)/2 + (u_{2i}^2 - u_{2i-1}^2)/2 \quad (3)$$

The degree of reaction of the *i*th stage compressor is defined as $\Omega_i = h_{r,i} / h_i$, hence, one has

$$\Omega_i = 1 - \frac{\varphi[k_{a,2i}^2(1+ctg^2\alpha_{2i}) - k_{a,2i-1}^2(ctg^2\alpha_{2i-1}+1)]}{2(k_{u,2i}k_{a,2i}ctg\alpha_{2i} - k_{u,2i-1}k_{a,2i-1}ctg\alpha_{2i-1})} \tag{4}$$

3. Mathematical model for multi-stage compressor

The compression work required by every stage is $h_i\,(i=1 \to n)$. The total compression work required by the multi-stage compressor is $h_c = \sum_{i=1}^{n} h_i$. The stagnation isentropic enthalpy rise of every stage is $h_{s,i}^*$. The sum of the stagnation isentropic enthalpy rise of every stage is $\sum_{i=1}^{n} h_{s,i}^*$, while the stagnation isentropic enthalpy rise of the multi-stage compressor is h_{sc}^*. One has $\sum_{i=1}^{n} h_{s,i}^* = (1+\alpha_z)h_{sc}^*$, where α_z is the reheat coefficient of the multi-stage compressor.

The stagnation isentropic efficiency of the multi-stage axial flow compressor is:

$$\eta_{sc}{}^* = h_{sc}{}^* / h_c = h_{sc}^* / \sum_{i=1}^{n} h_i \tag{5}$$

Because the flow velocities of the working fluid are not constant along the radial and axial directions, at different stations in the compressor, the velocity coefficients, $k_{u,j}, k_{a,j}\,(j=1 \to 2n)$, are introduced. They are defined as:

$$k_{a,j} = c_{a,j} / c_{a,1} = \rho_1 F_1 / \rho_j F_j\,, \quad k_{u,j} = u_j / u_1 \tag{6}$$

It should be explained that the shape of flow path, viz. the change of height of blade, is determined by $k_{a,j}$, and the shape of meridional plane, viz. the change of diameter, is determined by $k_{u,j}$. The flow coefficient is defined as $\varphi = c_{a,1}/u_1$. The isentropic work coefficient of the multi-stage is defined as $\psi_{sc} = h_{sc}^* / u_1^2$. The isentropic work coefficient of every stage is defined as $\psi_{si} = h_{si}^* / u_{2i-1}^2$.

With the help of the stage velocity triangle, it is convenient to rewrite Equation (5) in terms of velocity coefficients, flow coefficient, isentropic work coefficient and flow angles.

$$\eta_{sc}{}^* = \psi_{sc} / [\varphi \sum_{i=1}^{n} (k_{u,2i}k_{a,2i}ctg\alpha_{2i} - k_{u,2i-1}k_{a,2i-1}ctg\alpha_{2i-1})] \tag{7}$$

4. Optimization problem and its solution

The optimization problem is to find the flow angles $\alpha_j\,(j=1 \to 2n)$ to make the stagnation isentropic efficiency be maximum for the fixed $k_{u,i}, k_{a,i}$, φ and ψ_{sc} under the following constraint:

$$A \equiv i_1^* + h_c - \Delta h_r - \Delta h_s - i_{2n+1,s}^* = 0 \tag{8}$$

where Δh_r and Δh_s are the total losses of the rotors and the stators, and the other symbols are shown in Figs. 1 and 2.

The total profile losses of *i*th stage rotor and the stator are calculated as following:

$$\Delta h_{ri} = \omega_{ri} w_{2i-1}^2 / 2, \quad \Delta h_{si} = \omega_{si} c_{2i}^2 / 2 \quad (0 \le \omega_{ri}, \omega_{si} \le 1) \tag{9}$$

where ω_{ri} is the total profile loss coefficient of *i*th stage rotor blade and ω_{si} is that of *i*th stage stator blade.

The total losses of the rotors and the stators:

$$\Delta h_r = \sum_{i=1}^{n} \omega_{ri} w_{2i-1}^2 / 2, \qquad \Delta h_s = \sum_{i=1}^{n} \omega_{si} c_{2i}^2 / 2 \tag{10}$$

According to the enthalpy-entropy diagram of the multistage compressor, the constraint A can be rewritten as

$$A \equiv h_c - (1+\alpha_z) h_{sc}^* - \Delta h_r - \Delta h_s = 0 \tag{11}$$

Then, the constraint A can be rewritten in terms of velocity coefficients, flow coefficient, isentropic work coefficients, flow angles and profile loss coefficients of the blades.

$$\begin{aligned} A \equiv\ & \varphi \sum_{i=1}^{n} (k_{u,2i} k_{a,2i} ctg\alpha_{2i} - k_{u,2i-1} k_{a,2i-1} ctg\alpha_{2i-1}) - (1+\alpha_z)\psi_{sc} \\ & - \sum_{i=1}^{n} [(k_{u,2i-1} - k_{a,2i-1} \varphi ctg\alpha_{2i-1})^2 + k_{a,2i-1}^2 \varphi^2] \omega_{ri} / 2 \\ & - \sum_{i=1}^{n} k_{a,2i}^2 \varphi^2 (1 + ctg^2 \alpha_{2i}) \omega_{si} / 2 = 0 \end{aligned} \tag{12}$$

The blade profile loss coefficients ω_{ri} and ω_{si} are functions of parameters of the working fluid and blade geometry. They can be calculated using various methods. If they are considered as constants, the optimization problem can be solved using a Lagrangian function. The function $L = \eta_{sc}^* + \lambda A$ can be constructed, where λ is Lagrangian multiplier. The partial derivative equation groups can be obtained using Euler- Lagrangian equations as follows:

$$\left.\begin{aligned} & \partial L / \partial (ctg\alpha_1) = \varphi \psi_{sc} / [\varphi \sum_{i=1}^{n} (k_{u,2i} k_{a,2i} ctg\alpha_{2i} - k_{u,2i-1} k_{a,2i-1} ctg\alpha_{2i-1})]^2 + \lambda [\varphi \omega_{r1} (1 - \varphi ctg\alpha_1) - \varphi] = 0 \\ & \partial L / \partial (ctg\alpha_{2i-1}) = \varphi \psi_{sc} k_{u,2i-1} k_{a,2i-1} / [\varphi \sum_{i=1}^{n} (k_{u,2i} k_{a,2i} ctg\alpha_{2i} - k_{u,2i-1} k_{a,2i-1} ctg\alpha_{2i-1})]^2 \qquad (i = 2 \to n) \\ & \qquad + \lambda \left[-\varphi k_{u,2i-1} k_{a,2i-1} + \omega_{ri} \varphi k_{a,2i-1} (k_{u,2i-1} - \varphi k_{a,2i-1} ctg\alpha_{2i-1}) \right] = 0 \\ & \partial L / \partial (ctg\alpha_{2i}) = -\varphi \psi_{sc} k_{u,2i} k_{a,2i} / [\varphi \sum_{i=1}^{n} (k_{u,2i} k_{a,2i} ctg\alpha_{2i} - k_{u,2i-1} k_{a,2i-1} ctg\alpha_{2i-1})]^2 \\ & \qquad + \lambda \left[\varphi k_{a,2i} k_{u,2i} - \omega_{si} k_{a,2i}^2 \varphi^2 ctg\alpha_{2i} \right] = 0 \qquad (i = 1 \to n) \end{aligned}\right\} \tag{13}$$

Solving Equation (13) gives:

$$\left.\begin{aligned} ctg\alpha_{2i} &= k_{u,2i}\omega_{r1}(1-\varphi ctg\alpha_1)/(k_{a,2i}\varphi\omega_{si})(i=1\rightarrow n) \\ ctg\alpha_{2i-1} &= [k_{u,2i-1}\omega_{ri} - k_{u,2i-1}\omega_{r1}(1-\varphi ctg\alpha_1)]/(k_{a,2i-1}\varphi\omega_{ri})(i=1\rightarrow n) \end{aligned}\right\} \quad (14)$$

Substituting Equations (14) into (12) gives the quadratic equation for $ctg\alpha_1$ as follows:

$$\begin{aligned} &\varphi^2 ctg^2\alpha_1[\sum_{i=1}^{n}k_{u,2i}^2\omega_{r1}^2/\omega_{si} + \sum_{i=2}^{n}k_{u,2i-1}^2\omega_{r1}^2/\omega_{ri} + \omega_{r1}] + 2\varphi ctg\alpha_1[\sum_{i=2}^{n}k_{u,2i-1}^2\omega_{r1}/\omega_{si} \\ &+ k_{u,2i}^2\omega_{r1}/\omega_{si} - k_{u,2i-1}^2\omega_{r1}^2/\omega_{si}) + (k_{u2}^2\omega_{r1} - \omega_{s1})/\omega_{s1} - \omega_{r1} - \sum_{i=1}^{n}k_{u,2i}^2\omega_{r1}^2/\omega_{si}] \\ &+ 2(1+\alpha_z)\psi_{sc} + (\varphi^2+1)\omega_{r1} - 2k_{u2}^2\omega_{r1}/\omega_{s1} + \sum_{i=1}^{n}(k_{a,2i}^2\varphi^2\omega_{si}^2 + k_{u,2i}^2\omega_{r1}^2/\omega_{si}) \\ &- 2\sum_{i=2}^{n}[k_{u,2i}^2\omega_{r1}/\omega_{si} - k_{u,2i-1}^2(\omega_{ri}-\omega_{r1})/\omega_{ri} - k_{a,2i-1}^2\varphi^2\omega_{ri}/2 - k_{u,2i-1}^2\omega_{r1}^2/(2\omega_{ri})] \\ &= 0 \end{aligned} \quad (15)$$

The solution for $ctg\alpha_1$ from Equation (15) is the optimum value ($ctg^{opt}\alpha_1$) of $ctg\alpha_1$. Then the optimum values of $\alpha_j^{opt}(j=1\rightarrow 2n)$ can be obtained using Equations (14). The maximum of the objective function η_{sc}^*, the stagnation isentropic efficiency of the multi-stage axial flow compressor, and the optimum efficiencies, work coefficients, and degrees of reaction of *i*th stage can also be obtained.

The stagnation isentropic efficiency of *i*th stage compressor is $\eta_{s,i}^* = h_{si}^*/h_i$, then,

$$\begin{aligned} \eta_{s,i}^* &= (h_i - \Delta h_{ri} - \Delta h_{si})/h_i \\ &= 1 - [k_{a,2i-1}^2\omega_{ri}\varphi^2 + (k_{u,2i-1} - k_{a,2i-1}\varphi ctg\alpha_{2i-1})^2\omega_{ri} + k_{a,2i}^2\varphi^2(1+ctg^2\alpha_{2i})\omega_{si}] \\ &\quad /[2\varphi(k_{u,2i}k_{a,2i}ctg\alpha_{2i} - k_{u,2i-1}k_{a,2i-1}ctg\alpha_{2i-1})] \end{aligned} \quad (16)$$

The isentropic work coefficient of *i*th stage compressor is $\psi_{si} = h_{si}^*/u_{2i-1}^2$, then

$$\begin{aligned} \psi_{si} &= [2k_{u,2i}k_{a,2i}\varphi ctg\alpha_{2i} - 2k_{u,2i-1}k_{a,2i-1}\varphi ctg\alpha_{2i-1} - k_{a,2i-1}^2\omega_{ri}\varphi^2 \\ &\quad - (k_{u,2i-1} - k_{a,2i-1}\varphi ctg\alpha_{2i-1})^2\omega_{ri} - k_{a,2i}^2\varphi^2(1+ctg^2\alpha_{2i})\omega_{si}] \\ &\quad /2k_{u,2i-1}^2 \end{aligned} \quad (17)$$

According to the above analysis, for the fixed values of ω_{ri} and ω_{si}, it is easy to get the solution of the optimization problem and to obtain the obvious analytical equations.

When ω_{ri} and ω_{si} are functions of the parameters of the working fluid and blade geometry, the loss coefficients can be calculated using the method of Ref. [24], which was used and described in detail in Ref. [21]. The optimization problem can be solved using the iteration method:

First, select the original values of ω_{ri} and ω_{si}, and then calculate the parameters of the multi-stage compressor from the equations deduced above.

Secondly, calculate the values of ω_{ri} and ω_{si}, and repeat the first step until the differences between the calculated valves and the original ones are small enough.

5. Numerical examples

To find the influence of various dimensionless parameters on the optimum performance of the multi-stage stage axial flow compressor, numerical examples are provided using the universal mathematical model established earlier. The relation of the optimization objective with dimensionless parameters has been studied for the fixed values of ω_{ri}, ω_{si}, $k_{a,i}$, and $k_{u,i}$. It should be pointed out that there will be some influence on the relation of the optimization objective with these dimensionless parameters if ω_{ri}, ω_{si}, $k_{a,i}$, and $k_{u,i}$ are functions of working fluid parameters and geometry parameters of the flow path configuration. However, the relation obtained will not change qualitatively.

In the calculations, $\omega_{si} = \omega_{ri} = 0.1$, $k_{ui} = k_{ai} = 1 \quad (i = 1 \to n)$, $\alpha_z = 0.1$ and $n = 3$ are set.

The optimum stagnant isentropic efficiency vs. the work coefficient and the flow coefficient of the 3-stage compressor is shown in Fig. 5. It can be seen that the characteristic is similar to that of multi-stage turbine obtained by Boiko [23], that is, the optimum stagnant isentropic efficiency vs. the work coefficient is a parabolic-like one, there exists an optimum work coefficient corresponding to a maximum optimum (double-maximum) stagnation isentropic efficiency, and the optimum stagnant isentropic efficiency vs. the flow coefficient is a monotonic decreasing function, see Figs. 6 and 7.

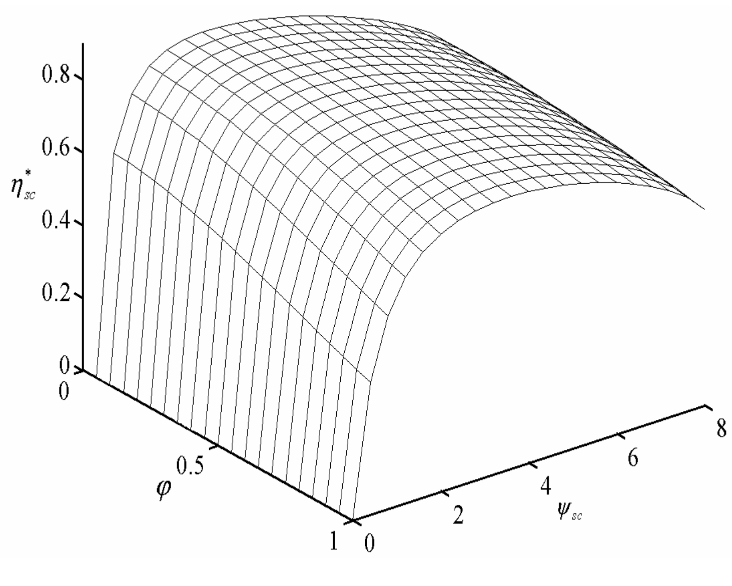

Fig.5 Optimum stagnant isentropic efficiency vs. work coefficient and flow coefficient of a 3-stage compressor

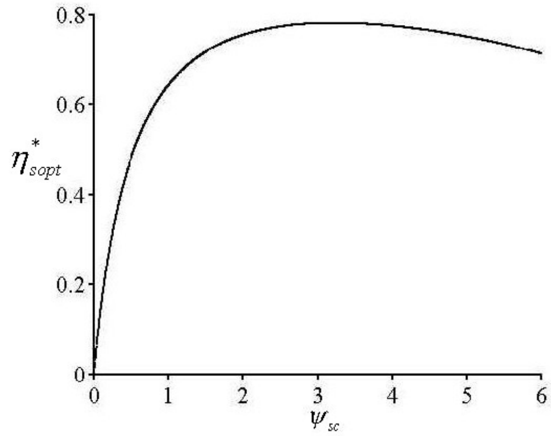

Fig. 6 Optimum stagnant isentropic efficiency vs. work coefficient of a 3-stage compressor

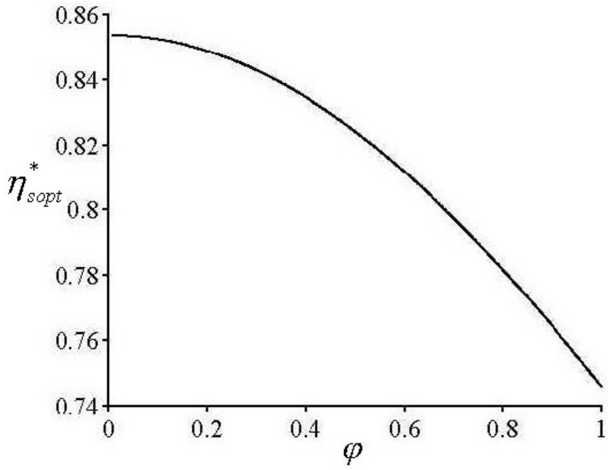

Fig. 7. Optimum stagnant isentropic efficiency vs. flow coefficient of a 3-stage compressor

The influences of the work coefficient (ψ_{sc}) on the optimum absolute inlet angle of the first rotor (α_{1opt}) versus flow coefficient (φ) are shown in Fig. 8. Figure 8 shows that the optimum absolute inlet angle is an increasing function of the flow coefficient when the work coefficient is less than some value. Otherwise it is a decreasing function of flow coefficient, and the optimum absolute inlet angle is larger than 90°. It is obviously that the optimum absolute inlet angle approaches to 90°, i. e. axial flow, when the flow coefficient increases.

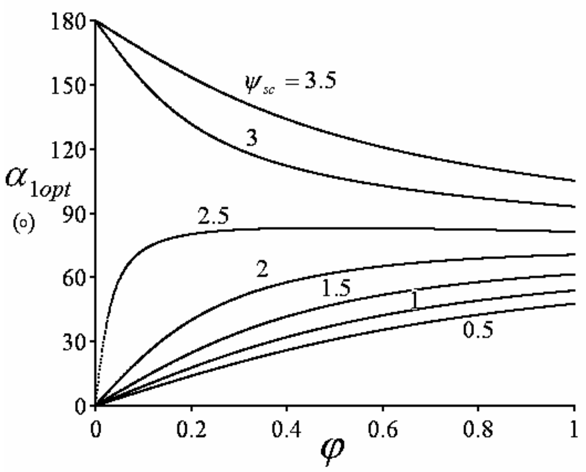

Fig.8. Effect of work coefficient on the optimum inlet airflow angle of the first rotor versus flow coefficient of a 3-stage compressor

The optimum absolute inlet angle of the first rotor (α_{1opt}) versus the work coefficient of the 3-stage compressor with $\varphi = 0.8$ is shown in Fig. 9. It can be seen that the optimum absolute inlet angle of the first rotor (α_{1opt}) is a monotonic increasing function of the work coefficient.

Using Eq. (14), one can obtain the other optimum inlet angles at all stations. For this numerical example, because the loss coefficients and the velocity coefficients at all stations are the same, the optimum absolute outlet angles of all rotors are the same, and the optimum absolute inlet angles after the second rotor are the same. Figures 10 and 11 show the optimum absolute inlet angles after the second rotor versus the work coefficient with $\varphi = 0.8$ and the optimum absolute inlet angles after the second rotor versus the flow coefficient with $\psi_{sc} = 2$. It can be seen that the optimum absolute inlet angles after the second rotor are increasing functions of the work coefficient and the flow coefficient. Figures 12 and 13 show the optimum absolute outlet angles of all rotors versus the work coefficient with $\varphi = 0.8$ and the optimum absolute outlet angles of all rotors versus the flow coefficient with $\psi_{sc} = 2$. It can be seen that the optimum absolute outlet angles of all rotors are decreasing functions of the work coefficient and increasing functions of the flow coefficient.

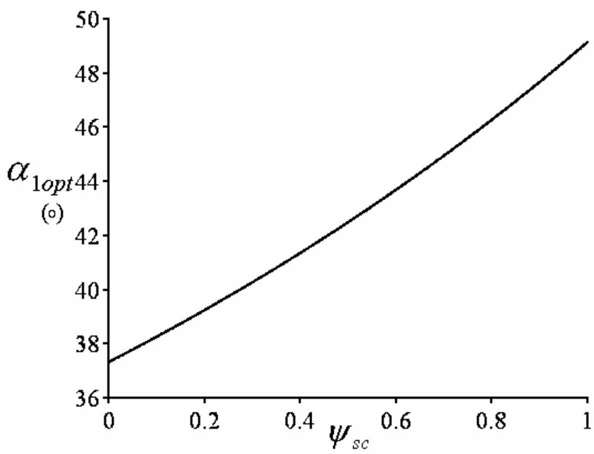

Fig.9. Optimum inlet airflow angle of the first rotor versus work coefficient of a 3-stage compressor

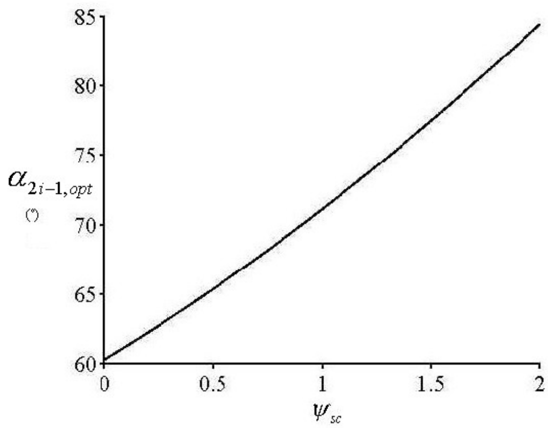

Fig.10. Optimum absolute inlet angles after the second rotor versus the work coefficient of a 3-stage compressor

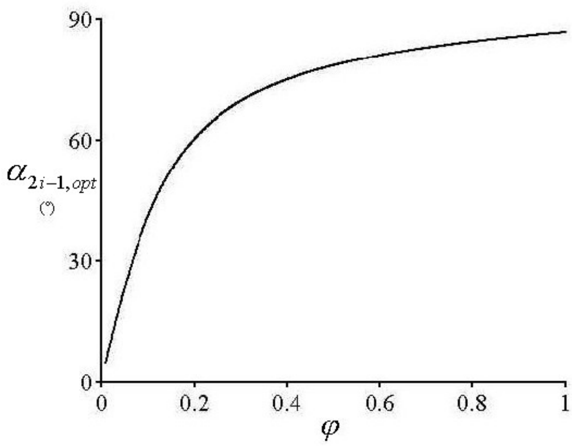

Fig.11. optimum absolute inlet angles after the second rotor versus the flow coefficient of a 3-stage compressor

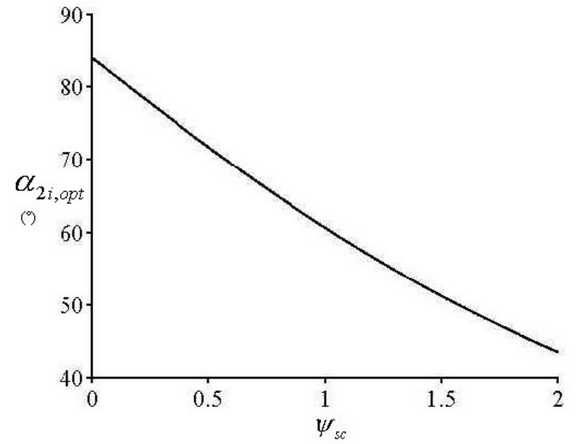

Fig. 12 Optimum absolute outlet angles of all rotors versus the work coefficient of a 3-stage compressor

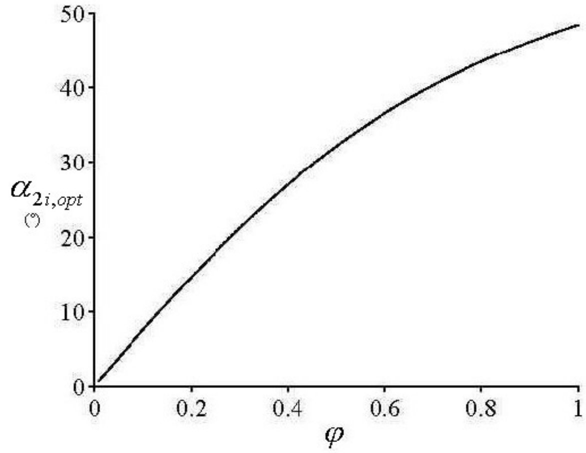

Fig. 13 Optimum absolute outlet angles of all rotors versus the flow coefficient of a 3-stage compressor

6. Conclusion

In this paper, the preliminary design efficiency optimization of a multi-stage axial flow compressor has been studied using one-dimensional flow theory. The universal characteristic relation for a multi-stage axial flow compressor is obtained. The influences of various parameters on the relations are analyzed. Numerical examples are presented. The results provide some guidance for the performance analysis and optimization of compressors. This is a preliminary study. It will be necessary to use multi-objective numerical optimization techniques [11-13, 20, 21, 25-29] and artificial neural network algorithms [10, 19, 30, 31] for practice compressor optimization.

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