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Review

Analysis of axial space between rotating blade rows and the stationary one in an axial-flow compressor

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This paper focuses on the analysis of axial space between rotating blade rows and the stationary one in an axial-flow compressor in function of the geometry and aerothermodynamic considerations. The axial distance is determined by the mathematical model obtained using design variables of rotating blade rows and the stationary blade rows or determined axial distance values. In this work, the analysis also consider the transition plane between rotating blade row and the stationary one, where outlet values for the rotating blade row is considering the same for the inlet values in the stationary blade row. Under this consideration the study was carried out for values of aerothermodynamic design, they were evaluated to identify which ones are more accurate to define a physical distance between rows. The results were obtained using the mathematical model and compared with data of axial flow compressor in one plane. There are maximum differences about a 25% between calculated distance and experimental data, for intermediate stage of axial compressor. Conclusions show that mathematical model can be applied for conditions of preliminary design of space between rotating blade rows and the stationary one in their middle zone.

Keywords: Axial space, axial flow compressor, rotating blades, turbomachine.

INTRODUCTION

Compressor is a turbomachine for increasing the pressure of a gas by mechanical decreasing of its vol-

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Symbols

A. Area: Ax. Axial; b. axial chord; c. Absolute velocity, Axial velocity; Cs. Sound Velocity;
C, Celsius; Cp, Specific heat at constant pressure; D, Diameter; F, Force; Gc, Gravitational acceleration; h, Enthalpy; Hm, Blade mean height; k, Isentropic constant; K, Kelvin; L, Axial length
$\dot{m}$, Mass flow; Mz, Force momentum; N, Blades numbers, angular velocity; P, Pressure; $\dot{Q}$, Heat transfer, volumen flow rate; R, number of blades ratio, specific gas constant; $\Re$, reaction
r, Radius; Δr, radius gradient; q, Heat; s, Entropy, blade pitch; Δs, Entropy gradient; t , Time; T, Temperature; Tq, Torque ; ΔT, Temperature gradient; U, u Tangential velocity; v, Specific volume
v, Velocity; W, w Relative velocity; W, Work transfer; z, Number of bladesα, absolute flow angle
β relative flow angle; Δ, gradient; ϕ, flow coefficient; ρ, Density; ℓ, blade length; η, Efficiency; υ, Blade stagger angle; σ, Slip factor, solidity; ω, Angular velocity; ψ, stage loading factor

Subscript

0, Inlet compressor; 1, Inlet rotor; 2, Outlet rotor; 3, Outlet stator; a, Axial component; b, Base of blade
e, External; f, Finaly; i,Inner; m, component in middle zone; m, Mean value; p, Blade tip; rr, Rotor; rs, Estator; sb, Inlet flow channel; rb, Outlet flow channel; t, Blade tip; TOT ,Total conditions; Rad.
ω, Radial velocity component; S, Stage isentropic; U, Tangential direction; °, Grade; ∞, Position to infinite distance

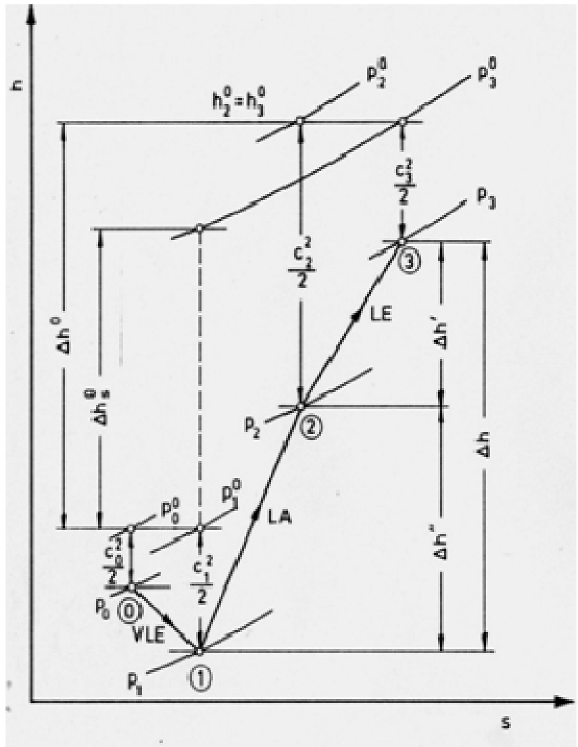

Figure 1. Enthalpy-entropy diagram compressors

ume. Air is the most frequently compressed gas but natural gas, oxygen, nitrogen, and other industrially important gases are also frequently compressed.

Compressors are used to increase the pressure of a wide variety of gases and vapors for a multitude of purposes. Applications of compressors include chemical processing, electrical generation, gas transmission, gas turbines, and aircraft.

The axial-flow compressor can achieve higher pressures at a higher level of efficiency. There are two important characteristics of the axial flow compressor—high-pressure ratios at good efficiency and thrust per unit frontal area. Although in overall appearance, axial turbines are very similar, examination of the blade cross-section will indicate a big difference.

Theoretical model

Flow normally presents a negative pressure gradient. If pressure ratio is higher, it will be more complicated the design of compressor. To achieve different pressure ratios, axial compressors are designed with different numbers of stages and rotational speeds. As a general rule-of-thumb it can assume that each stage in a given compressor has the same temperature rise. Therefore, as the entry temperature to each stage must increase progressively through the compressor, the ratio of temperature entry must decrease, thus implying a progressive reduction of the stage pressure ratio through the unit. Hence the rear stage develops a significantly lower pressure ratio than the first stage. In the axial compressor, cascades of small airfoils are mounted on a shaft that turns at a high rate of speed. Several rows, or stages, are usually used to produce a high pressure ratio, with each stage producing a small pressure increase.

Figure 1 shows enthalpy–entropy diagram (T-s), in a compressor stage. With the net heat transfer to/or from working fluid is zero and an adiabatic process, equation 1 shows the work in the entrance of compressor under these considerations:

$$W = \dot{m} \cdot c_p \left(T_{02} - T_{01} \right) \qquad (1.1)$$

Where

$W = \text{Work done} \left[W \right]$

$\dot{m} = \text{Mass flow} \left[kg / s \right]$

$c_p = \text{Specific heat in constant pressure} \left[\text{kJ} / \text{kg K} \right]$

$T = \text{Temperature} \left[\text{K} \right]$

Some parameters are important in the compressor design like temperature, pressure and density, in the inlet and outlet of rows of rotating and stationary blades.

Flow velocity and peripheral (tangential) velocity were calculated, and the velocity triangle for an axial flow compressor stage was constructed.

In the compressor, tangential velocity "U" is perpendicular to axial velocity. Air enters the rotor blade with absolute velocity C_1 at an angle α_1 measured from the axial direction.

Air leaves the rotor blade with absolute velocity C_2 at an

Table 1. Fluctuation of pressure coefficient for different axial spaces

% Vane chord length	Maximum fluctuation of pressure coefficient			
	Stator	Reduction	**Rotor**	Reduction
20	0.61		0.49	
30	0.37	0.38	0.23	0.55
40	0.21	0.43	0.16	0.27
50	0.11	0.48	0.14	0.13

angle $\square_2$. Air passes through the diverging passages formed between the rotor blades. Meanwhile work is done on the air in the rotor blades, C_2 will be larger than C_1. The rotor row has tangential velocity U.
Combining the two velocity vectors gives the relative velocity at inlet V_1 at an angle $\square_1$.

Axial Space

Axial space between rotating blade rows and the stationary ones is important to be considered in the compressor design.

The optimization of the axial space between wheels of blades is important to avoid a bad behavior due to work between rotor-stator. In this work, it is considered axial space, 20-50 % of vane chord axial, intervals of 10%. Pressure ratio, mass flow, and angular velocity are constant. Equation 1.2, define the pressure coefficient.

$$C_p = \frac{p - p_{entrada}}{(1/2)\rho_{entrada} \cdot u^2} \qquad (1.2)$$

Pressures gradients are present in the airfoil edge of stationary blade rows. This variation is higher in contrast to rotator blade rows.

The axial space was increased from 20% to 30% of the vane chord length; the maximum fluctuation of the pressure coefficient was reduced in 38% in stator and 55% in the rotor. From 30 to 40%, the fluctuation of the pressure coefficient of 43% in stator and 27% in the rotor was presented. To 50% of vane chord space, the fluctuation of pressure was of 48% for the stator and 12% for the rotor.

Since axial space is reduced in 20% of Vane chord to 50%, pressure fluctuation reduces in 82% in stator and 71% in rotor; Table 1.

A minimum axial space between rotating blade rows and the stationary ones, cause discontinuous change in the curvature of shock wave of rotor blade, the trailing edge of stator blade is cutting. The cutting shock wave changes to pressure wave over the upper surface in rotor blade.

According to pressure wave, the flow is supersonic; in this case, the entropy and losses energy rises as a consequence by shock wave, see Figure 2. These reveal a decreasing in efficiency, mass flow.

Figure 3 shows velocity triangles in the leading and trailing edge of a blade. The change in airfoil velocity is due to the boundary layer on the surface of the blade and if there is no turbulence from other blades, this wave moves downstream.

For same axial space between rotating blade rows and the stationary one, Figure 4 shows blades number versus excitation of rotor blades and Figure 5 shows for excitation rotor blades versus axial space.

MATHEMATICAL MODEL

It is important to identify variables to predict and diminish flow excitation between discs of blades, Equation 2.1. A more accurate form of the variation of axial velocity analysis is obtained with the actuator disc concept, Equation 2.2.

$$d_x = \frac{\text{axial distance stator} - \text{rotor}}{b_{rr}} \qquad (2.1)$$

$$c_x = c_{x\infty 2} + \frac{1}{2}\left(c_{x\infty 1} - c_{x\infty 2}\right)\left\{\exp\left[\frac{-\pi|x|}{H}\right] + \exp\left[\frac{-\pi|x-\delta|}{H}\right]\right\}. \qquad (2.2)$$

Where:

c_x = axial velocity

$c_{x\infty 1}$ = axial velocity far upstream.

$c_{x\infty 2}$ = axial velocity far downstream

$x - \delta$ = axial distances

H = head, blade height

Equation 2.2 indicates that the axial distance has a relation with axial velocity, blade height and axial vane chord.

Parameters that affecting space in blades discs were considered, dimensional and adimensional factors like tangential velocity, relative velocity, disc blade diameter, stage loading factor and velocity ratio.

Finally it is considered the limit distance of base to blade tip for analysis of axial space between rotating

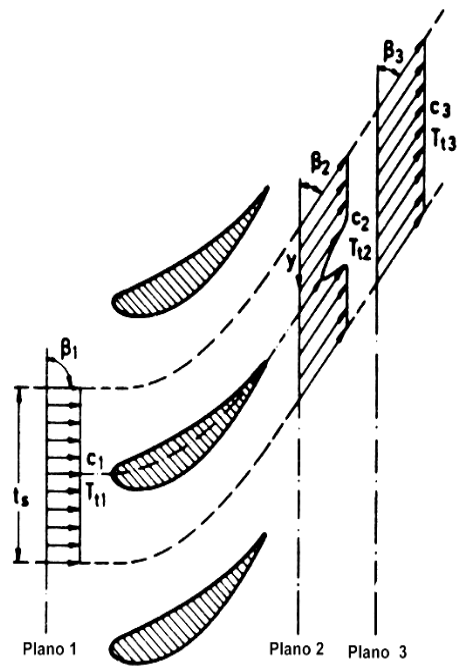

Figure 2. Shock wave for a blade.

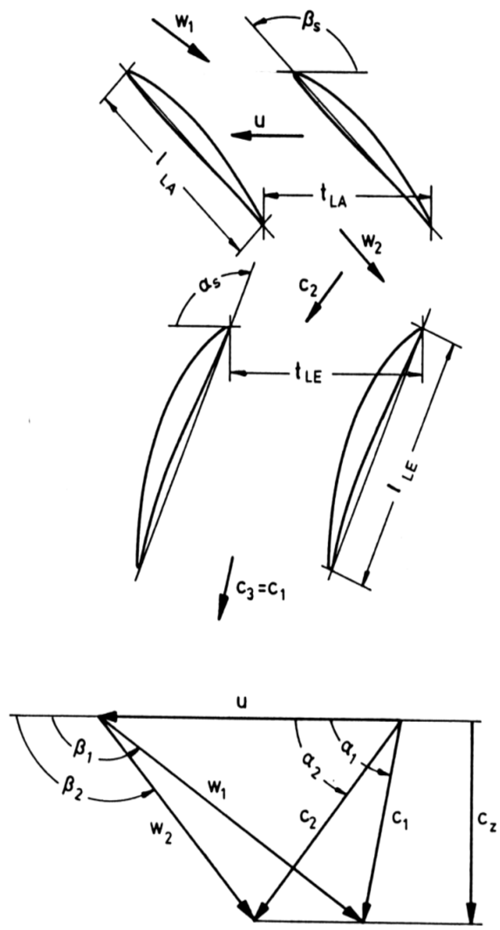

Figure 3. Velocity triangles for one stage.

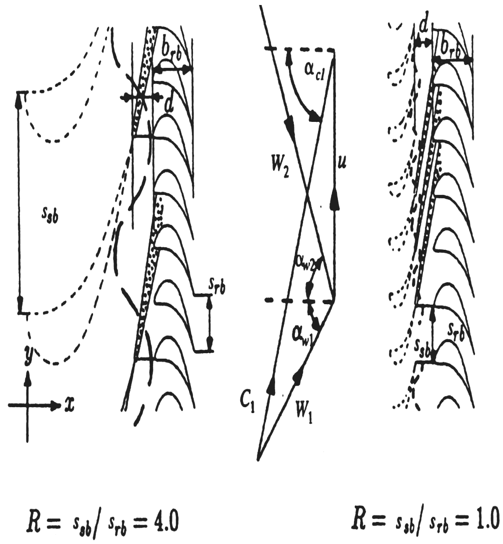

Figure 4. Rotor excitation according to blades number and stator.

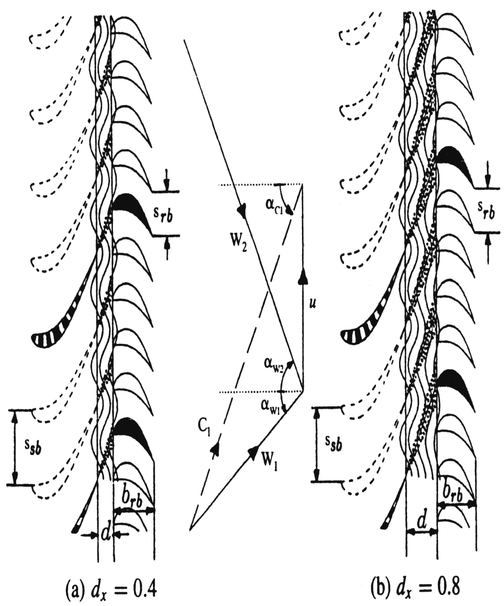

Figure 5. Rotor excitation according to axial space between rotating blade rows and the stationary one

blade rows and the stationary one, equation 2.3.

$$dx = \frac{c_a}{w_2} \cdot \frac{t}{l} \cdot \upsilon \cdot \psi \cdot h_m \qquad (2.3)$$

Where
dx = axial distance [mm]; c_a = axial velocity [m/s]
t = pitch flow channel [mm]; w_2 = outlet velocity [m/s]
l = vane chord length [mm]; □□□□ratio velocity
□□□□stage loading factor; h_m = high blade medium

CONCLUSIONS

It was obtained a mathematical model for determining space axial between rotating blade rows and stationary one with different analyses, aerothermodynamics and geometric.

Also approximations of 19% for stage number four and 16% for stage number nine of axial distance were founded.

For any axial turbomachinery, it is possible to obtain the axial distance of different stages.

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