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INVESTIGATION OF DESIGN FEATURES OF COMPRESSOR CASING TREATMENT

Casing treatment in the form of circumferential grooves over a rotor blade tips is used for improvement of an axial compressor performance. Usually, these grooves extend compressor's stall range (stable operational range) but decrease its efficiency. In the paper, there are presented main results of investigations on grooves that influence positively efficiency of compressor. There were investigated traditional (typical) and newly developed groove configurations. Certain grooves combine increase in efficiency with extension in stall range.

Nomenclature

S	Absolute blade tip clearance, mm
n	Number of grooves
h	Depth, mm
l	Width, mm
t	Distance of groove location, mm
D	Rotor outer diameter, mm
$\overline{d}$	Rotor hub/tip ratio
$\overline{h}$	Blade aspect ratio
z	Number of blades
$\overline{S}$	Relative blade tip clearance, %
$\overline{H}$	Head coefficient
$\overline{C}$	Absolute flow velocity coefficient
U	Circumferential rotor tip speed, m/s
η	Experimental rotor efficiency
$\Delta\eta$	Change of experimental rotor efficiency

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A	Experimental rotor stall range
ΔA	Change of experimental rotor stall range

Subscripts

R	rotor
g	groove
1	rotor inlet
2	rotor outlet
c	compressor; crosspiece between grooves
p	blade projection
t	blade tip; outer
T	theoretical
a	axial direction
red	reduced

1. Introduction

Application of casing treatment for improvement of axial flow compressor performance may be substantiated by some reasons. Casing treatment was traditionally used for extension of compressor stall range [1,2,3,4,5,6]. Historically, casing treatment in the form of annular grooves in compressor casing (Fig. 1) was the best known treatment. This configuration is characterized by high simplicity. Greitzer E.M. has presented [7] the main conclusions concerning the application of casing treatment (in the form of grooves and slots).

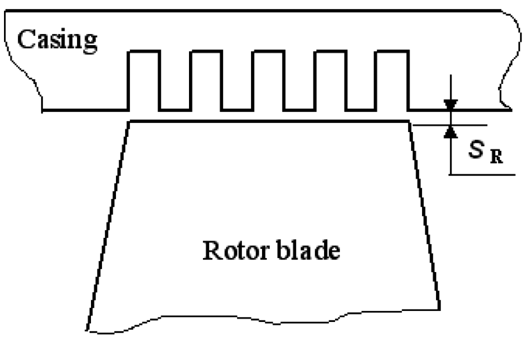

Fig. 1. Typical casing treatment in the form of annular grooves

These are the following:

– The use of a grooved casing can provide substantial improvement in compressor stall range over a smooth wall casing (baseline).

– The use of a grooved casing can lead to somewhat increased efficiency over a smooth wall casing, but in most cases these configurations cause some deterioration of efficiency.
– The configurations which give increased efficiency are not as effective in increasing the stall range.

So it is important that in many cases increase in the stall range is accompanied by a decrease in the compressor efficiency. However, positive results are possible too. For example, in the work [8] seven grooves of 9.5 mm height and 1.6 mm width were used for improvement of compressor stage performance. As a result, maximum adiabatic efficiency increased, for the uniform inlet flow, by 0.95% at relative rotor speed 1.0, by 4.76% at 0.9, and by 3.09% at 0.7 relative rotor speed.

The problem of appropriate optimization of the slot size and configuration is rather important, as well. Certain results of the experimental investigation concerning a variety of annular grooves in an axial compressor casing are presented below.

2. Results of Investigation on Traditional and New Developed Groove Configuration

The pattern of a typical annular grooves over a rotor blade (Fig. 2) may be characterized by the number of grooves n_g, one radial (h_g) and four axial (l_g, l_c, t_1, t_2) geometric dimensions (l_p is the parameter of blade chord projection). In the following considerations, it will be more convenient to use some relative geometric parameters (h_g/l_g, l_c/l_g, t_1/l_p, t_2/l_p). The analysis in the work [9] recommends, for example, the values of $h_g/l_g \approx 2.5$, $l_c \approx 0.5\ l_g$, and $n_g \leq 13$ as optimal ones.

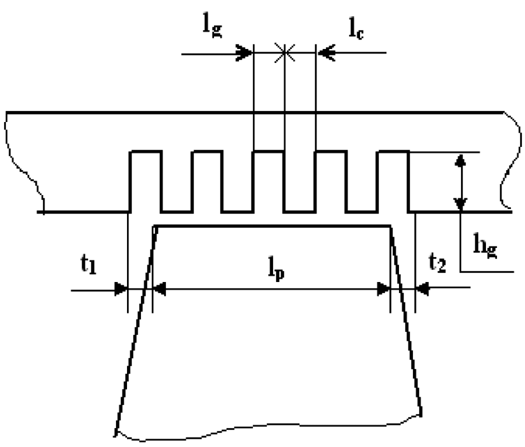

Fig. 2. Main geometric parameters of annular grooves

The main parameters of the tested compressor objects from the work [10] are presented in Table 1.

These investigations were performed at a special experimental test rig. While testing, the total pressure was measured before and after rotor with the help of five-duct radial combs installed at chosen points in the circumferential direction. Measurement of the total pressure field after guide vane was performed with the help of five-duct radial comb traversing in the circumferential direction. This procedure was accompanied by measurement data averaging along the cascade space and the blade height. Test data were obtained and processed by means of application of an automated data system. The efficiency was determined with the help of hunting stator with an accuracy of ±0.3%.

Table 1.

Main parameters of tested objects

Label	Object	Index	D_t, mm	$\overline{d}_1$	$\overline{h}_R$	z_R	S_R, mm	$\overline{S}_R$, %	$\overline{H}_T$	C_{1a}	$U_{t\,red}$, m/s
●	Stage	HEMS	393.5	0.87	1.0	57	0.3	1.0	0.32	0.46	130; 200
▲	Stage	AI-24-VI	346.5	0.80	1.0	45	0.3	1.0	0.40	0.56	185
■	Stage	AI-24-IX	346.5	0.84	0.9	39	0.4	1.43	0.34	0.52	180
◆	Block	AI-24-VIII+ R-IX	346.5	0.84	0.91	-	0.4	1.43	-	0.55	180
▼	Rotor	AI-24- R-VIII	346.5	0.84	0.91	39	0.4	1.43	0.32	0.54	180
◣	Stage	SIL	400.0	0.90	1.0	100	0.5	2.5	0.59	0.73	125; 210

The main geometric parameters of applied traditional grooves are presented in Table 2.

Table 2.

Main geometric parameters of applied traditional grooves

Index	n_g	h_g, mm	l_g, mm	l_c, mm	t_1, mm	t_2, mm	h_g/l_g	l_c/l_g	t_1/l_p	t_2/l_p
HEMS-1	6	5 15	2	2	-1	3	2.5	1	-0.05	0.15
VI-O	5	5	2	2	-6.5 0	1.5 -5	2.5	1	-0.28 0	0.07 -0.22
1-3	5	5	2	2	7	-7	2.5	1	0.39	-0.39
1-2.5	5	5	2	2	0	0	2.5	1	0	0
10	7	5	2	2	5	5	2.5	1	0.28	0.28

The compressor objects (Table 1) were tested with the traditional grooves, presented in Table 2. The main results of this investigation are shown in Table 3. Note that the changes in the experimental rotor efficiency $\Delta\eta$ and stall range ΔA were then calculated as:

$$\Delta\eta = (\eta_g - \eta)(\%) \tag{1}$$

$$\Delta A = \left(\frac{K - K_g}{K}\right) \times 100\% \tag{2}$$

where η_g and η – compressor peak efficiency with and without grooves,

K – point of a peak pressure rise (identified by a throttle coefficient proportional to the downstream flow area at a peak pressure coefficient).

Table 3.

Main results of traditional grooves investigation

Compressor object	Applied grooves	$U_{t\,red}$, m/s	h_g, mm	t_1, mm	$\Delta\eta$, %	ΔA, %
HEMS	HEMS-1	130 130 200	5 15 15	-1 -1 -1	4.87 4.87 3.55	0 0 0
AI-24-VI	VI-O	185	5	-6.5 0	1.14 2.27	2.92 1.12
AI-24-IX	1-3	180	5	7	1.07	-7.36
AI-24-VIII + R-IX	1-2.5	180	5	0	0.20	-1.5
AI-24-R-VIII	10 1-2.5	180 180	5	5 0	0.80 2.20	-1.0 -1.5

As it is seen, the obtained results have rather limited usefulness, and are not suitable for exact generalization. However, it may be assumed that the influence of grooves on compressor efficiency is closely connected both with groove geometry and the stage loading. This assumption is confirmed, to some extent, by the relationship of Fig. 3. Positive changes in efficiency take place for the stages with relatively low loading ($\bar{H}_T \approx 0.30\ldots0.40$). For comparison, effect of grooves HEMS-1 at the high-loaded stage SIL ($\bar{H}_T \approx 0.59$) has demonstrated a great decrease in efficiency (more than 10%). On the other hand, some dispersion of efficiencies in the range of $\bar{H}_T \approx 0.30\ldots0.40$ may be explained by groove dimensions difference.

Certain results of experimental investigation, concerning variety of annular grooves in the compressor casing, are presented below.

The types of investigated objects are presented in Table.4. The type A is the original one, the basic configuration for further comparison. This object, having $l_g = l_c = 2$ mm and $h_g = 5$ mm is located over the rotor blade tip (strictly along the chord axial projection, $t_1 = t_2 = 0$ mm). The differences in design between other grooves and the original configuration are as follows. The type B of grooves is intended for research on the influence of groove depth (h_g =1; 2.5; 5; 7.5; 10 mm) on the stage performance. Another one (the type C) serves for testing groove solidity ($l_g = l_c = 1$ mm in this case in contrast to $l_g = l_c = 2$ mm for the original type). The third and fourth types (D and E) are necessary for evaluation of additional groove presence before and after the original device (t_1 =4 mm and t_2 =4 mm).

Tests were performed at subsonic double-stage compressor (AI-24-VIII+IX), the grooves were located over the first rotor blading of this compressor.

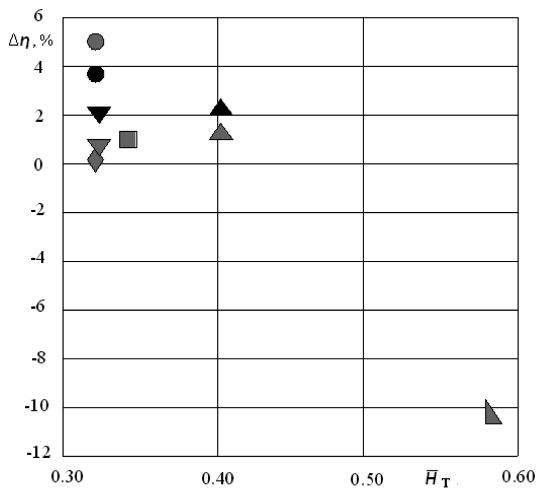

Fig. 3. Relationship between efficiency change and loading of grooved stages

Comparison of performances was carried out within the optimum domain (maximum efficiency) at 0.5% accuracy and 95% confidence level (the minimal required number of measurements varied from 9 to 14, depending on the experimental series). The overall curves were derived with the help of least squares method.

As it is seen, varieties of groove depth (from 1 to 10 mm) have practically constant influence both on efficiency and stall range (Fig. 4). Any significant influence of groove solidity change was not observed (in two times). The arrangement of additional groove before rotor blades has only slight and equal influence on the stall range. On the other hand, an additional groove before blades has rather low effect on the stage efficiency, and a groove after blades – a more negative effect.

The conclusion that follows is the necessity for optimization of existing and development of new grooves as the means for casing treatment.

Table 4.

Main geometric parameters and results of testing of devices A, B, C, D, E

Device	n_g	h_g, mm	l_g, mm	l_c, mm	t_1, mm	t_2, mm	$\Delta\eta$, %	ΔA, %
A	5	5	2	2	0	0	0	0
B	5	1 2.5 5 7.5 10	2	2	0	0	-0.4 -1.0 -1.05 -1.0 -0.8	-2.5 -2.4 -2.8 -2.6 -2.7
C	9	5	1	1	0	-1	0.44	0.02
D	6	5	2	2	4	0	-1.73	-0.59
E	6	5	2	2	0	4	-0.42	-0.59

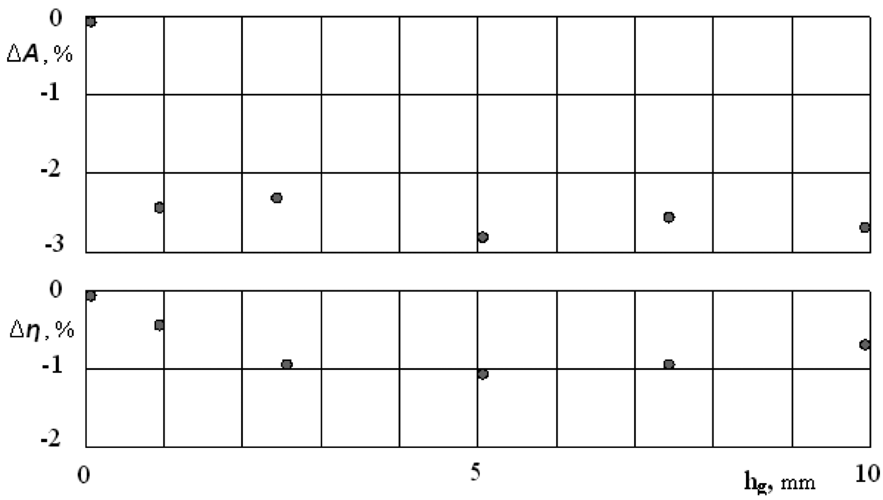

Fig. 4. Relationships between groove depth and rotor parameters changes

The principal idea was the following: the grooves with different depth along a rotor blade tip may concentrate less losses and have a higher efficiency. Correspondingly, there were proposed several new configurations of grooves (see Fig. 5 and Table 5).

The results of research on new grooves (Table 5) have confirmed the proposed approach.

There are four new devices (groove configurations) with a positive influence on compressor efficiency. The devices 2-2 and 4 are especially perspective ones, as they combine significant extension of stall range (for 4.0. . . 5.4 %) with certain efficiency increase (by 0.8. . . 1.6 %).

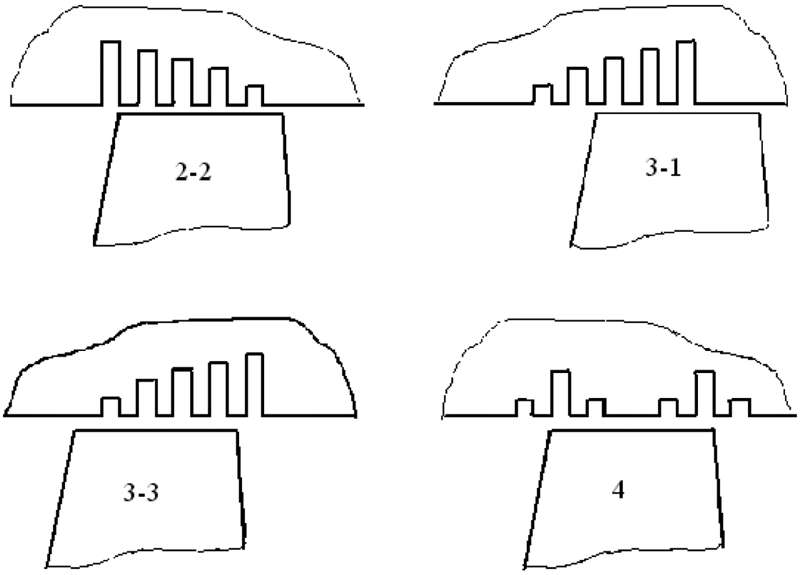

Fig. 5. Modifications of new annular groove configuration

Table 5.

Main geometric parameters and results of testing of new annular groove configurations

Device	n_g	h_g, mm (from the left to the right)	l_g =1 mm	t_1, mm	t_2, mm	$\Delta\eta$, %	ΔA, %
Baseline	-	-	-	-	-	0	0
2-2	5	5, 4.5, 4, 3-2	2	2	-2	1.55	-4.02
3-1	5	2, 3,4,4.5,5	2	7	-7	0.37	-4.80
3-3	5	2, 3,4,4.5,5	2	-3	3	0.27	-2.30
4	6	2, 5, 2,…, 2.5,2	2	4	4	0.80	-5.40

3. Conclusions

1. Annular (circumferential) grooves in an axial flow compressor casing over rotor blade tips are effective means for extension of compressor stall range.

2. Influence of grooves on compressor efficiency is connected both with grooves geometry and stage loading. The positive effect on efficiency is most probable for low-pressure compressor objects.

3. The special investigation has demonstrated that significant change in groove total depth and solidity does not affect considerably the compressor stage performance.

4. The change in groove depth distribution along rotor blade tips makes it possible to develop new casing treatment configurations, which allow for increasing efficiency and extending stall range.

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Badanie charakterystyk projektowych obróbki obudowy kompresora

S t r e s z c z e n i e

Obróbka obudowy kompresora, polegająca na wykonaniu obwodowych rowków ponad szczytem łopaty wirnika, jest stosowana dla poprawy właściwości roboczych kompresora. Zwykle rowki takie zwiększają zakres przeciągania kompresora (zakres stabilnego działania), ale pogarszają sprawność. W artykule zaprezentowano główne wyniki badań nad rowkami, które wpływają pozytywnie na sprawność kompresora. Badano zarówno tradycyjne (typowe) konfiguracje rowków, jak i konfiguracje ostatnio opracowane. Niektóre z nich zapewniają zarówno wzrost sprawności jak zakresu przeciągania.