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Experimental study of a centrifugal compressor with two successive and counter-rotating impellers

Van-Thang NGUYEN[1,2], **Amelie DANLOS**[1], **Richard PARIDAENS**[1],
Florent RAVELET[1], **Michael DELIGANT**[1], **Sofiane KHELLADI**[1],
Farid BAKIR[1]

[1]Arts et Metiers Institute of Technology, CNAM, LIFSE, HESAM University, 75013 Paris, France
[2]Faculty of Vehicle and Energy Engineering, Le Quy Don Technical University, Hanoi, Vietnam

E-mail: `amelie.danlos@lecnam.net`

Abstract. In many industries, reducing the size of the compression system represents a significant goal for designers. In fact, increasing the power density of the compressor actually means lowering production costs. The chief purpose of this study is to compare, in terms of aerodynamic performances, a centrifugal compressor consisting of an impeller and a volute, to an innovative concept based on the principle of two successive impellers and designed to work in counter-rotation. To simplify the experimental research in this study, the volute of the conventional compressor is maintained for the new design. The conventional impeller has been replaced by two successive impellers having the same meridian profile, and whose diameter of the rear impeller is identical to that of the conventional impeller. The impellers design of the counter-rotation stage was carried out granting to an approach proposed here. The numerical results underline the advantage of the innovative compressor, in terms of power density, compared to the conventional design. The numerical forecasts also agree significantly with the results of experimental tests got on a test bench exclusively designed for compressors. Experimental optimization of the rotational speed ratio of the two counter-rotating impellers has demonstrated the possibility of increasing the compression ratio and efficiency over a wide range of flow.

1. Introduction

Since the 20th century, counter-rotating turbomachines are used to improve performances with reducing the size and the weight. A counter-rotating turbomachinery consists of two impellers rotating in two opposite directions, with two different rotation speeds N_1 and N_2. This type of systems has been tested for axial-flow fans [1, 2, 3, 4, 5] and shows that it is possible to obtain a higher pressure rise, to improve the efficiency and to extend the operating range of mass flow rate. The speed ratio $\theta_i = \frac{N_1}{N_2}$ was studied in the case of counter-rotating axial compressor stage [6, 7, 8] and has shown that if the second rotor is 50% faster than the first one, the stall free range can be improved and, on the other hand, if the first rotor rotates 50% faster than the second one, the load on the first rotor is increased and the stall appears earlier in the second rotor. Counter-rotating pumps have also been investigated [9, 10] by combining a rotor-stator and a front-rear rotor and lead to a more compact system with a lower rotation speed of rotors. Counter-rotating centrifugal compressors have not been experimentally studied yet. But numerical simulations

have shown encouraging results [11]. This paper presents experimental results of the performance comparison between a Counter-Rotating Centrifugal Compressor (CRCC) and a Single Impeller Centrifugal Compressor (SICC) to show the advantages of this innovative design.

2. Design of a counter-rotating centrifugal compressor

SICC was constructed for a suction application and was chosen as a reference compressor in this study. The innovative CRCC made of two impellers rotating in opposite directions is designed based on SICC with the same point design in order to analyze the benefits of a novel compressor.

2.1. Introduction of SICC

An impeller of SICC has an external diameter D_2 equal to 286 mm. The gap between the impeller and the casing is 1.5 mm which represents 5% of the blade width $b_2 = 29.7mm$. The impeller presents 7 blades and 7 splitters with a small vanless diffuser in respect for the geometry shown obtained with the Turbo3D software developed in our laboratory (figure 1). The point design of the compressor is determined by a mass flow rate $\dot{m} = 0.73kg/s$, a pressure ratio equal to 1.31, and the impeller speed N=16000rpm. More details of SICC geometric parameter are summarized in the table 1.

Table 1. Geometric parameters of the SICC impeller

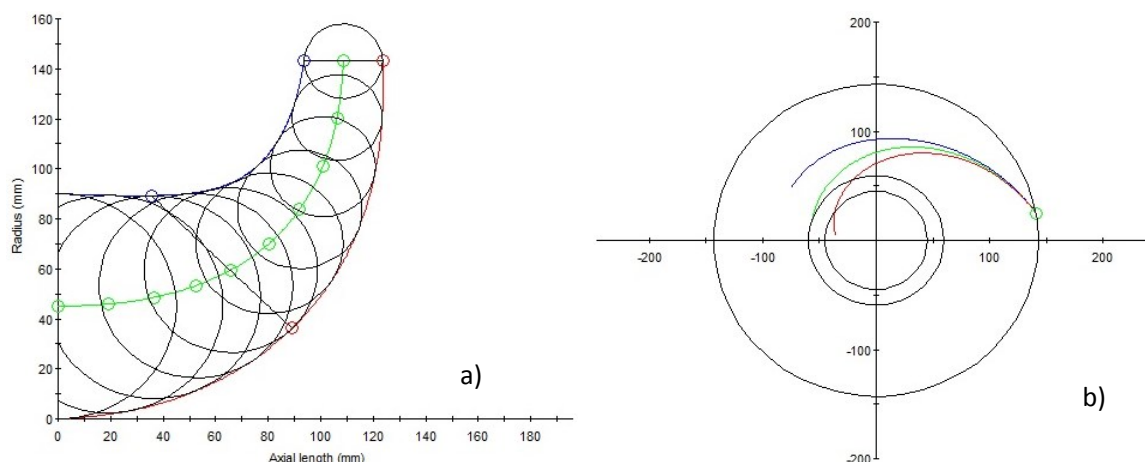| Parameter	Value	Parameter	Value
Hub diameter D_{1h}	59.5 mm	Inlet blade angle at hub β_{1h}	45.3°
Shroud diameter D_{1s}	161 mm	Inlet blade angle at Shroud β_{1sh}	69.8°
Outer diameter D_2	286 mm	Outlet blade angle β_2	64°
Blade width b_2	29.7 mm	Mass flow rate $\dot{m}$	$0.73kg/s$
Rotational speed N	16000 rpm	Pressure ratio	1.36
Number of blades Z	7+7	Power consumtion	$28kW$
Pressure ratio Π	1.31		

Figure 1. Impeller of SICC designed by Turbo3D software. Diagram of (a) meridional view and (b) front view

The global performance of SICC obtained in previous experiment is shown in figure 2. It presents
the variation of pressure ratio and polytropic efficiency according to the change of the corrected
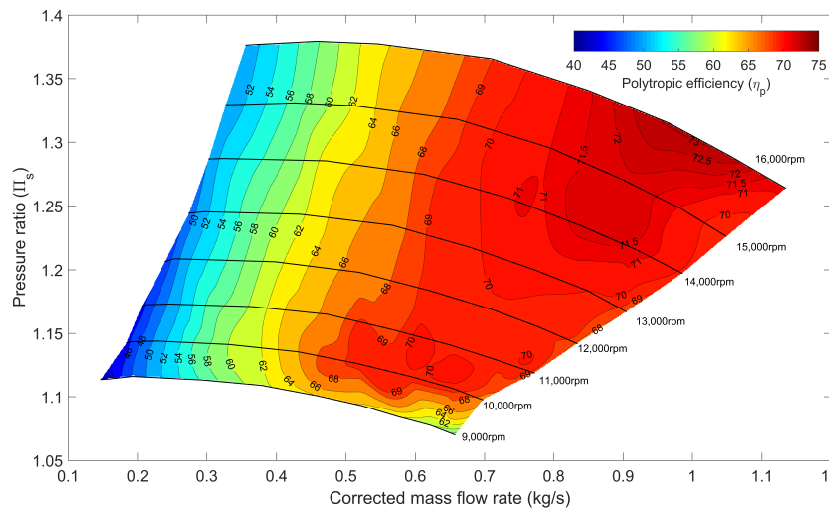mass flow rate in various rotation speeds.

Figure 2. The experimental performance map of SICC

2.2. Design of two innovative rotors

Two impellers of CRCC were designed with the same size and the meridional profile as the
SICC's impeller (reference impeller). Two compressors use the same volute for easy evaluation.
The reference impeller is then replaced by two impellers rotating in the opposite directions. The
first impeller of CRCC rotates with angular velocity ω_1 and the second one rotates with angular
velocity ω_2 in the inverse direction. At the designed point, both impellers rotate with the same
speed ($\omega_1 = \omega_2$).

The outer diameter of the first impeller, placed in front of the second impeller, is determined
with the length ratio defined by equation 1:

$$LR = \frac{L_{1st}}{L_m} \tag{1}$$

Where L_{1st} is the length of the first impeller at the hub/shroud contour and L_m the total length
of the reference impeller in meridional surface as shown in figure 3b. A previous work [11]
presented a numerical investigation of different CRCC configurations with various LR values.
The selected configuration here optimizes the efficiency and is obtained for LR=0.3. This value
imposes the outer diameter of the first impeller. The velocity triangle presented in figure 3a
gives the outlet blade angle of the first impeller. The inlet blade angle of the second impeller is
then calculated to adapt the flow in the outlet of the first impeller according to the theorem of
aerodynamic. All characteristics of these impeller are shown in table 2 and 3.

3. CRCC test bench

The test bench uses to study the performances of the CRCC is designed in a suction application
at LIFSE laboratory as shown in figure 3. Working fluid is the air at room temperature. The
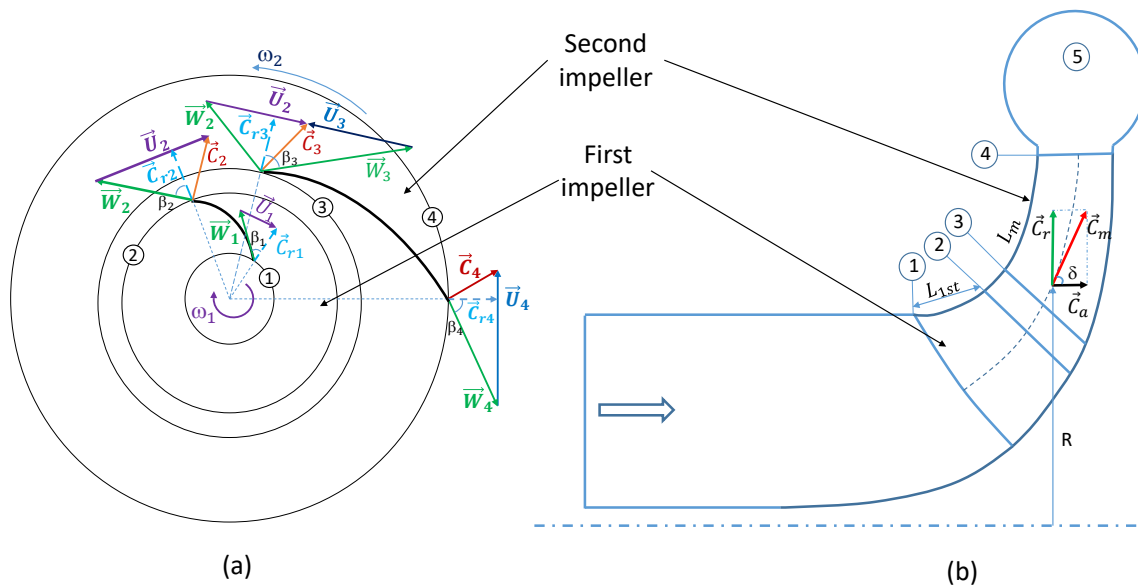
Figure 3. Diagram of (a) the velocity triangle and (b) a meridional contour of CRCC. Points 1 and 2 represent respectively the inlet and outlet of the first impeller and points 3 and 4 respectively stand for the inlet and outlet of the second impeller. Points 5 represents the volute outlet

Table 2. First impeller geometric parameters

Parameter	Value	Parameter	Value
D_{1h}	$51.5mm$	β_{1h}	$45.3°$
D_{1sh}	$161mm$	β_{1sh}	$69.8°$
D_{2h}	$118mm$	β_{2h}	$53.6°$
D_{2sh}	$176mm$	β_{2sh}	$68.4°$
N_1	$-16000rpm$	Z_1	7
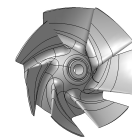
Table 3. Second impeller geometric parameters

Parameter	Value	Parameter	Value
D_{3h}	$128.5mm$	β_{3h}	$72.7°$
D_{3sh}	$183mm$	β_{3sh}	$77°$
D_4	$286mm$	β_4	$64°$
N_2	$16000rpm$	Z_2	9
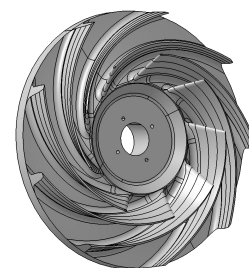
outlet pressure of the compressor is the atmospheric pressure at the time of measurement. The test bench includes the compressor (CRCC) connects to an upstream tank and a downstream muffler to reduce the noise. The tank on which a flat plate pierced 120 holes with a diameter of 8 mm allows controlling the inlet flow rate. A long pipe with a diameter of D=158,5mm connects the tank to the compressor. A honeycomb is placed upstream of the pipe to obtain a flow with minimum distortion for measuring mass flow rate and reducing the effect on the performance.

This pipe is equipped with a flow-meter, a thermocouple, and four pressure taps to measure the inlet parameters of the compressor. The flow-meter places at an upstream distance 12D is used to measure the inlet flow rate. It includes a Pitot tube connected to an FC66 manometer with an accuracy $\leq \pm 1\%$ of reading. Four pressure taps placed upstream on the circumferential inlet pipe at a distance 3D, is connected to an FC322 manometer with an accuracy 0.5% to measure the differential pressure between the inlet and the atmospheric pressure. One thermocouple Pt100 is placed upstream at a distance 20D to measure the inlet temperature. Three other ones are installed downstream on the circumferential outlet pipe at a distance 4D to measure the outlet temperature of the compressor. These thermocouples have a range of uncertainty of $0.02°C \div 0.04°C$ in the range of temperature considered in this experimental study. In the case of measurements, the CRCC is driven by two independent electric motors. The speed of the second impeller is kept constant at 9000rpm. Besides, the rotation speed of the first impeller will be adjusted to investigate the influence on the performance.
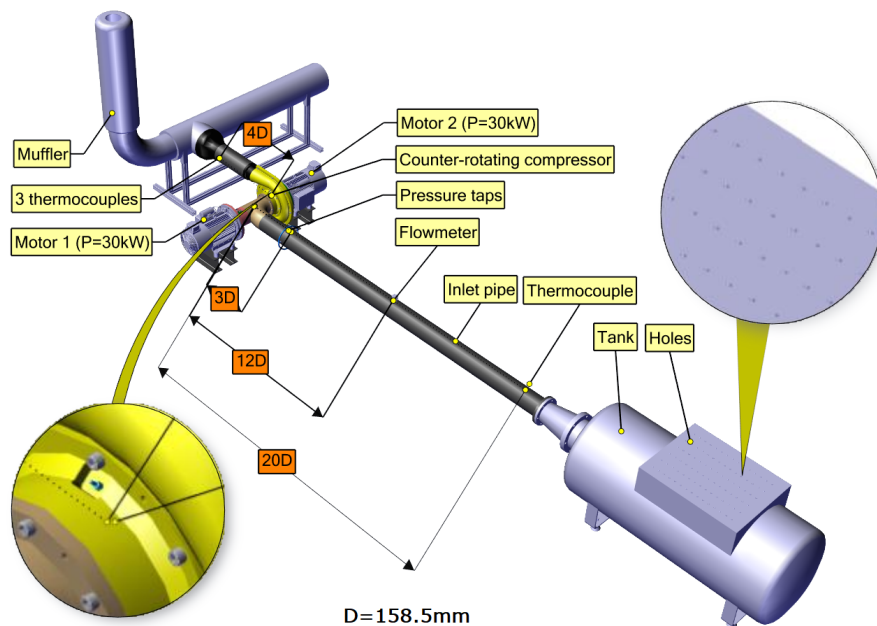

Figure 4. The test bench of CRCC

4. Experimental results

The pressure ratio is calculated by the ratio of the atmospheric pressure to the inlet pressure measured by FC322 manometer.

$$\Pi = \frac{P_{atm}}{P_{in}} \tag{2}$$

The polytropic efficiency is described by the equation 3:

$$\eta_p = \frac{\gamma - 1}{\gamma} \frac{\ln \frac{P_{out}}{P_{in}}}{\frac{T_{out}}{T_{in}}} \tag{3}$$

Where T_{out} and T_{in} are respectively the outlet and the inlet temperature of the compressor and γ the specific heat ratio.

4.1. Global performances of the innovative CRCC at the speed of 9000rpm

The innovative counter-rotating system is designed with two impellers rotate with the same speed. In this section, the experimental results is presented in case of the speed of the first impeller $N_1 = -9000rpm$ and the second impeller $N_2 = 9000rpm$. Figures 5 and 6 show the comparison between the pressure ratio (Π_s), polytropic efficiency (η_p), and power consumption of CRCC and those of SICC. The solid black line represents CRCC, while the dashed blue line describes the SICC. The red and green lines show the electric power of respectively the first and second impeller of CRCC. It is easy to see in figure 5 that the pressure ratio of CRCC is always
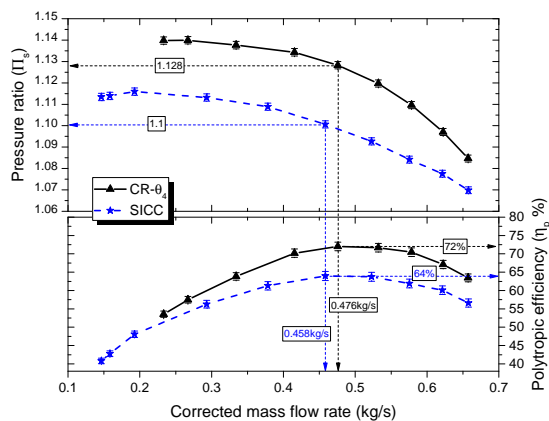

Figure 5. Comparison between CRCC's global performance (the solid black line: CR-θ_4) and SICC's one at the speed of 9000rpm

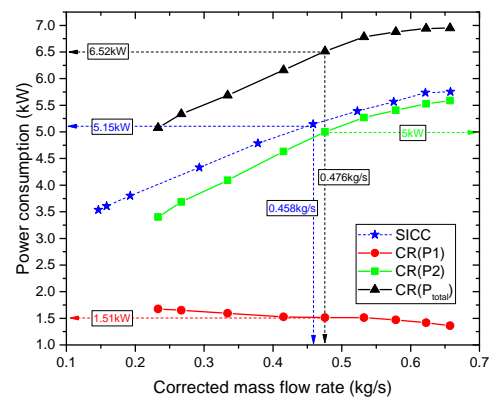**Figure 6.** Comparison between CRCC's power consumption ($CR(P_{total})$): the first rotor consumption (CR(P1)), the second rotor consumption (CR(P2)) and SICC's one

higher than that of SICC. At the operating point, the pressure ratio of CRCC gets 1.128±0.002 while that of the SICC is only 1.1±0.002. the CRCC configuration gives higher performance than the SICC. It can be said that the CRCC pressure ratio, in this case, has increased by about 2.5% in comparison to SICC's one. In addition, the efficiency curves illustrate that the CRCC's efficiency is always better than the SICC's one. The best efficiency of CRCC has been improved by about 8% compared to the efficiency of SICC. Concerning the power consumption, figure 6 shows that the total power consumption of CRCC (the solid black line) is always higher than that of SICC (the dashed blue line). At the operating point, the power consumption of the first and second impellers is 1.51kW and 5kW, respectively, while the power consumption of the SICC is 5.15kW. It can be noted that the consumption of the second impeller (the solid green line) is approximately equal to that of the SICC. Besides, the consumption of the first impeller (the solid red line) is quite small and increases slightly as the flow decreases. It looks like the characteristic of an axial turbomachine. This shows that the fluid is compressed mainly in the second impeller because it has a large variable diameter from the inlet to the outlet. Consequently, the CRCC configuration requires more energy due to higher pressure rise and overcome the additional mechanical losses of the drive system as well as the aerodynamic losses in the interactive region between two impellers.

4.2. Influence of the speed ratio
By replacing the impeller of SICC compressor with the innovative counter-rotating system, it is possible to control independently the rotation speed of each impeller. The impeller speed ratio is then defined as $\theta_i = \frac{N_1}{N_2}$. Different values of the impeller speed ratio are proposed as it can be shown in table 4. The minus sign represents two impellers rotate in the opposite direction.

Table 4. Impeller speed ratio

Speed ratio (θ)	Value	The first impeller speed (N_1)	The second impeller speed (N_2)
θ_1	-0.7	-6300	9000
θ_2	-0.8	-7200	9000
θ_3	-0.9	-8100	9000
θ_4	-1	-9000	9000
θ_5	-1.1	-9900	9000
θ_6	-1.2	-10800	9000
θ_7	-1.3	-11700	9000

For all studied impeller speed ratios, the rotation speed of the second impeller is fixed at 9000rpm. The advantages of an innovative generation compressor have been made by comparing the performance of the two machines under nearly identical operating conditions and the same peripheral velocity. Figure 7 shows the effect of the impeller speed ratio on the performance of CRCC. It is clear that the pressure ratio increase gradually with the increase of the first impeller speed. The pressure ratio has been increased from $\Pi_s = 1.107\pm0.002$ at $\theta_1 = -0.7$ to $\Pi_s = 1.145\pm0.002$ at $\theta_7 = -1.3$. It is worth noting that the pressure ratio curve of the CRCC with $\theta_7 = -1.3$ approaches that of the SICC at 11000rpm. That shows that the CRCC can achieve almost the same performance of SICC but at a lower speed of about 2000rpm. An interesting fact is that the performance of the compressor does not seem to be affected by the speed ratio. The best efficiency point is always 72%±2% comparing to 64%±2% of SICC. These points are achieved at different mass flow rates corresponding to different speed ratios. Set of the best efficiency points generates an effective operating zone of CRCC in case of change the speed of impellers. We can conclude that the impeller speed ratio in the range selected for this work greatly affects the pressure ratio, effective operating zone, and surge margin while remaining the polytropic efficiency stable of CRCC.
Nevertheless, the surge margin seems to be appeared soon in CRCC configuration. Higher speed ratio leads to the surge phenomena occurs earlier because of higher pressure ratio and higher incidence angle at the impeller inlet.

5. Conclusion
In this study, a counter-rotating centrifugal compressor (CRCC) has been experimentally studied to measure the effect of the innovative system on global performances. Results of the comparison with a single impeller centrifugal compressor (SICC) highlight the increase of the pressure rise and the polytropic efficiency. CRCC can improve by about 8% in polytropic efficiency and 2.5% in pressure ratio when the speed of two impellers is the same. Besides, the effect of the speed ratio of CRCC is conducted to find out its effect on the performance. The results show that the speed ratio has no effect on the maximum efficiency (it remains 72%±2%) but it increases significantly the pressure ratio (it increases by about 4% at $\theta_7 = -1.3$). In addition, changing
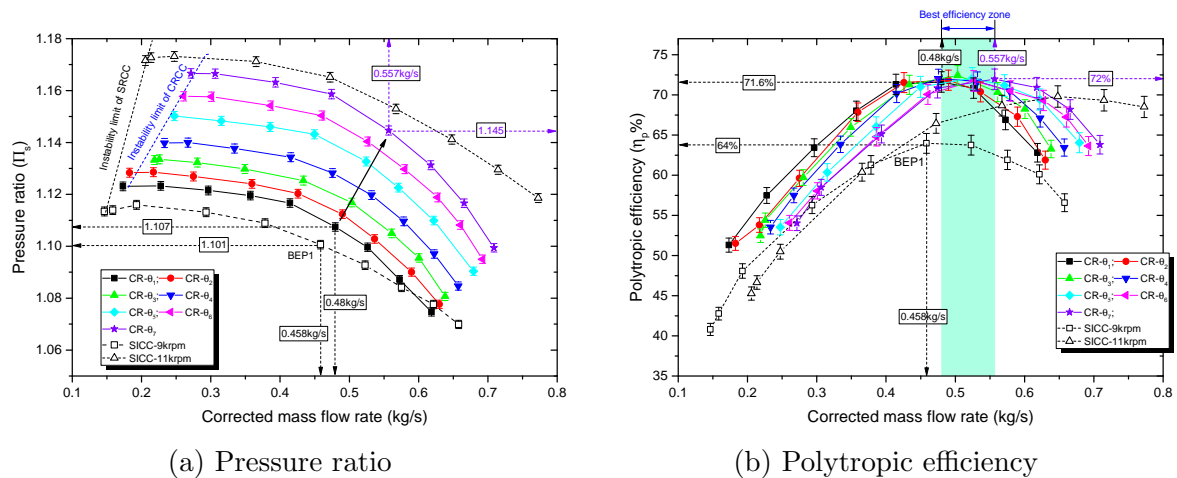
(a) Pressure ratio   (b) Polytropic efficiency

Figure 7. Effect of speed ratio θ_i on the global performances of CRCC with the second impeller rotation speed fixed at 9,000 rpm. The solid lines represent the CRCC configuration and the dash lines display the SICC.

the speed ratio can make a wider effective operating zone in comparison with the single point of SICC. However, the surge margin occurs early in CRCC configuration because of higher pressure rise and higher incidence angle at the inlet of impellers. In future work, this inconvenience of CRCC will be resolved by finding a couple of speeds of CRCC's impeller to extend the operating range and in particular to push back the surge limit to a smaller flow rate.

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