Journal of Applied Fluid Mechanics, Vol. 12, No. 6, pp. 1791-1799, 2019.
Available online at www.jafmonline.net, ISSN 1735-3572, EISSN 1735-3645.
DOI: 10.29252/jafm.12.06.30046

Improving the Performance of Centrifugal Compressors by Flow Recirculation

S. Abolfazl Moussavi Torshizi†

Niroo Research Institute, Tehran, Iran

† *Email: samoussavi@nri.ac.ir*

(Received January 19, 2019; accepted March 13, 2019)

ABSTRACT

The impeller of a centrifugal compressor is traditionally designed using some formula for only one design point which makes it less efficient in all other situations. This is especially important for compressors not experiencing a constant working condition. To improve the performance at low mass flow rates and retard the surge, an innovative concept is introduced for a centrifugal compressor. In this method pressurized air is injected at the compressor inlet to improve the flow field. With better incidence angle, related losses at off design conditions are minimized and the surge is delayed. This system is designed, modeled and adjusted for providing an optimal flow pattern at the inlet. Its implementation on a compressor has shown an increase of efficiency at low mass flow rates. It has also improved flow pattern in impeller passages and decreased the blade loading near surge condition. It is also shown that the swirl generator system can be fed up from the compressor volute or diffuser, and thus widening the compressor performance map by retarding the surge margin.

Keywords: Centrifugal compressor; Flow injection; Surge; Performance map; Incidence loss.

NOMENCLATURE

D	inducer diameter	r	local radius
d	injector diameter	Va	axial component of velocity
L	inducer length	Vc	circumferential component of velocity
R	inducer radius	Vr	radial component of velocity

1. INTRODUCTION

Centrifugal compressors have now found vast applications in different industries including aerospace, automotive and oil. Because of a long history and after evolutions, it is believed that design of the impeller of centrifugal compressors has now reached a mature state (Mojaddam, 2017). Although many papers and researches can be found in this subject, but it can be concluded that most of them are focused on different accessories and equipment which can enhance the performance of existing compressors by improving the efficiency, widening the working range and reaching the higher pressure ratio.

Also, one of the important issues in centrifugal compressors which has attracted much attention in the scientific community is the surge phenomena. In one side, understanding the mechanism of the surge phenomena is of interest (Semlitsch, 2016) and at the other side, surge control (Arnulfi, 2006) and operating stabilization (Skoch, 2003) and related prevention techniques are important concerns for operators.

Many scientific researches could be found on prediction (Dehner *et al.*, 2011), analysis (Bontempo *et al.*, 2017) and modeling (De Bellis *et al.*, 2018) of surge, however in this work we focus on different solutions which are available for widening compressor operating range by improving the surge margin. For this purpose, casing treatment as described by Jansen *et al.* (1980), Macdougal and Elder (1982) can be found. Jansen *et al.* (1980) created circumferential grooves on the shroud of a compressor, and showed its results in retarding the surge limit and extending the operating range. Also, Fisher (1988) proposed self-recirculation casing treatment, known as the ported shroud, for a turbocharger compressor and thus extended the surge margin without much stage efficiency deterioration. Hunziker *et al.* (2001) applied an internal bleed system allowing some fluid to recirculate. They improved the compressor surge line for a high pressure ratio turbocharger compressor having vaned diffuser. Different variations and optimizations of casing treatment can be found in the literature like the work of Hu *et al.* (2016). In this method, the low momentum fluid is removed from the flow path and thus passage blockage is removed and the rest of the flow is allowed to continue the diffusion process. Sivagnanasundaram *et al.* (2013) investigated various slot configurations effect on a compressor map width and explored the optimal one.

Another technique for surge control is to implement vanes in the diffuser section. Senoo (1983) proposed a low-solidity vaned diffuser to improve the compressor efficiency while maintaining a wide flow range. Also Jiang and Whitfield (1992) studied a compressor with a diffuser equipped with variable vanes and suggested that it may be used for turbocharging automotive engines to meet the requirement of a wide working range. Elder and Gill (1985) showed that a major factor for surge in vaned diffuser designs is related to the flow in the semi vane-less space.

Inlet guide vanes are also used for delaying the surge. Rodgers (1991) studied the use of variable inlet guide vanes (VIGVs) on a centrifugal compressor. He could show that the surge margin was considerably extended at high speeds but led to lower flow coefficient and some efficiency decrease.

An example of flow injection can be observed in the work of Stein *et al.* (2000). They stated that considerable experimental evidences have been reported showing that the air injection in a steady or unsteady fashion improves the overall operability of the compressor. In their work, a steady flow of air at an angle of 5^{o} from the axial direction was injected into the impeller eye. Similar work is presented by Gu *et al.* (2005, 2007) in which the flow was axially injected at the shroud section of the inlet and forced into the impeller. Skoch (2003, 2004) and also Halawa *et al.* (2015) proposed to inject the high pressure air into the space between the impeller and the vaned diffuser with some considerations. By energizing the flow at the impeller exit, they could overcome the flow stagnation in the compressor. In the mentioned works, a high pressure air source must be available to activate the system and control the compressor. In many applications, such an auxiliary system is not available.

In this paper, a combination of mentioned methods (i.e. air injection and IGV) is proposed for improving the performance of centrifugal compressors at off design points by benefitting from their advantages and avoiding their problems. To this end, it is suggested to rectify the impeller inlet flow angle by means of flow injection while variation of inlet flow angle is usually performed with IGVs. In contrast to other mentioned methods above, in this novel approach, the needed high pressure air is provided by the compressor itself and no external source or device is necessary. The final compressor behavior is similar to compressors implementing IGVs, but here there is no vane and thus no technical complexity for control and adjustment of vanes. So, it is believed that this system can be more reliable.

2. The swirl Generator

In this method, the core concept is to inject a jet of air tangentially into compressor inlet such that after mixing with the existing axial stream, the ideal swirl flow is generated at the compressor eye. Hence an air injection system which can generate a swirl in the inlet is designed and adjusted.

To have a uniform impact on the angle of attack at all spans of the impeller inlet, a forced vortex (linear distribution of circumferential velocity in the plane perpendicular to the axial direction) must be created. By adding swirl, as also could be generated using inlet guide vanes, the incidence angle of the flow can be corrected when needed.

For creating the swirl in the main duct with diameter D, four injectors are considered as shown in Fig. 1. Figure 2 shows the details of injector installations. The diameter of injectors (d) are selected such that d/D=0.1. Also they are located at a distance of 2D from the inlet. The main duct has an aspect ratio of L/D=10. Initially, the injectors are installed tangent to the main duct (ri=0.9R).

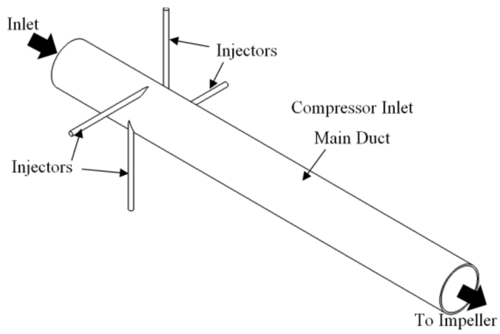


Fig. 1. Swirl Generator Configuration.

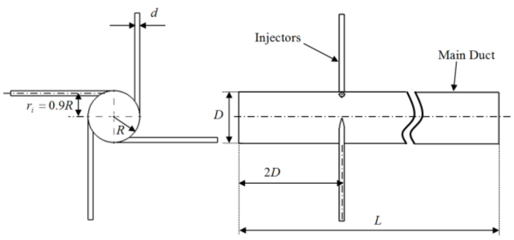


Fig. 2. Injectors and the main duct locations.

The domain is meshed with tetrahedral elements, adaptively refined in critical zones and near the walls. The boundary conditions for the main tube are total pressure at the inlet and mass flow rate at the outlet. Injectors have a predefined total pressure. Because the injected air is supplied from the compressor, it is estimated to be delivered at approximately 370K. The total inlet and injection pressure is set to 1 bar and 1.7 bar respectively. The mass flow rate of the main tube is also assumed to be 0.2 kg/s. The turbulence is taken into account using

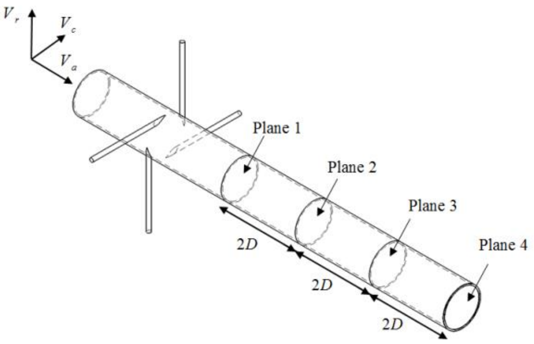


Fig. 3. Data sampling sections.

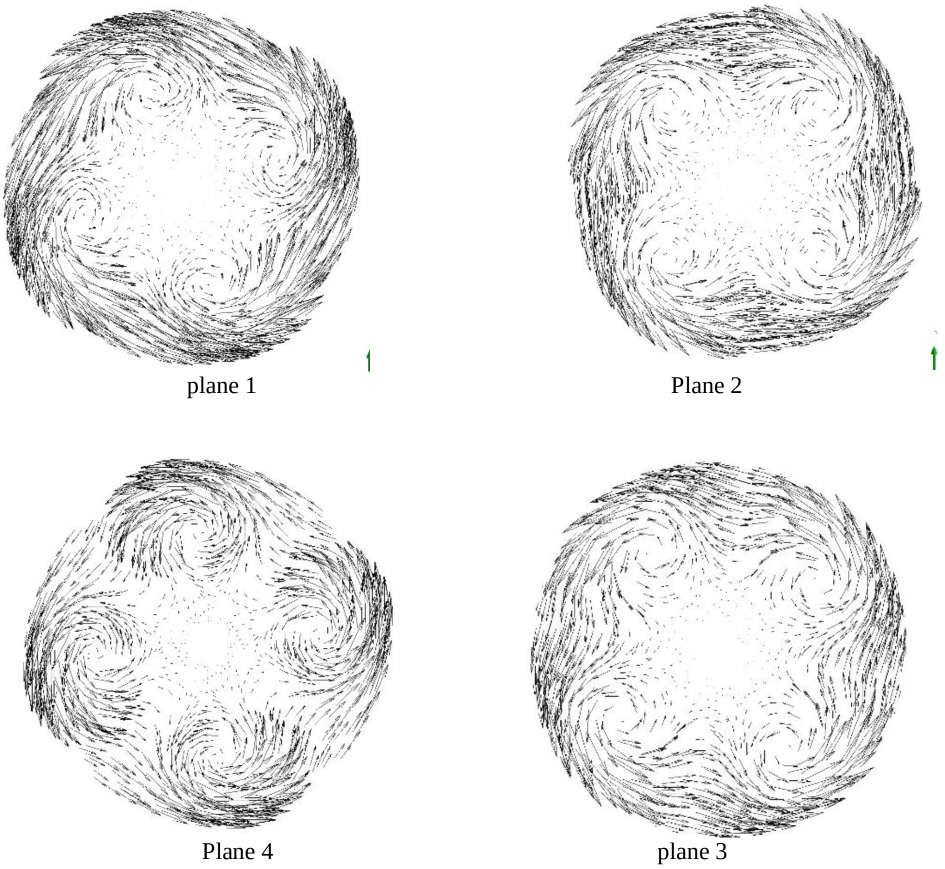


Fig. 4. Vectors of circumferential velocity along the tube for the tangent case (r_i=0.9*R*).

the k-epsilon model. The flow is simulated steadily and assumed to be compressible.

The results are shown in 4 sampling planes along the tube (Fig. 3). The plane distances are set to be 2*D*. The velocity vectors are plotted at these sampling planes, as shown in Fig. 4.

The forced vortex pattern is considered as ideal flow pattern and the deviation from this situation should be avoided. As observed, when the jets are injected tangential to walls, the flow is far from an ideal forced vortex. Instead, 4 minor vortices can be seen corresponding to each injector. It is obvious that this pattern cannot be beneficial to the compressor. Thus, an adjustment must be made to find the best injection point which will lead to the most similar flow pattern to a forced vortex. In this configuration, the swirl number at plane 4 is equal to 0.124.

To improve the flow pattern several configurations have been tested by varying the radial position of injectors (ri) as in Fig. 5. By performing several simulations, it was observed that by injecting the high speed flow tangentially at ri=0.75R, the best flow pattern is achieved as shown in Fig. 6. Here, a global vortex is observable and no sub vortices exist. Thus the achievable swirl number is increased to 0.166 which shows a 33.8% improvement over the past configuration with the same amount of air injected.

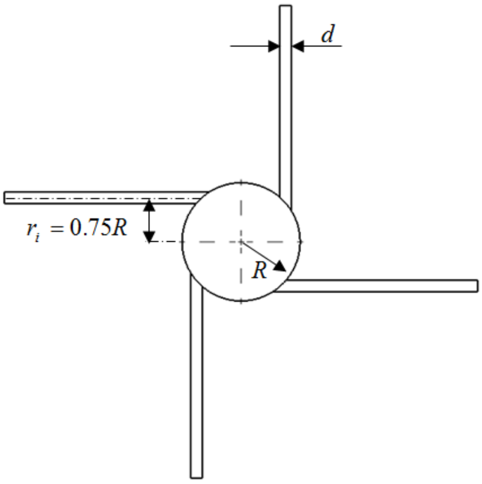
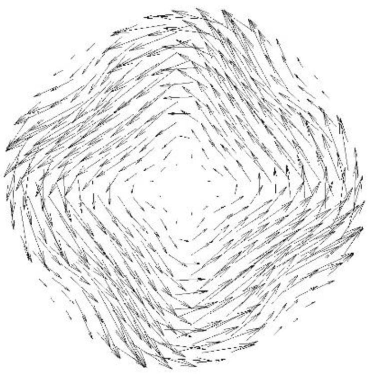
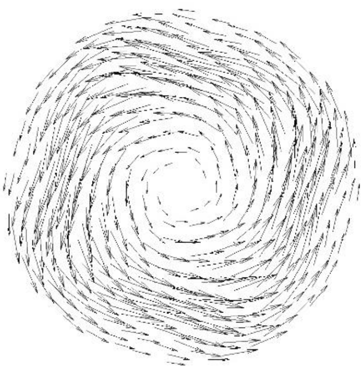


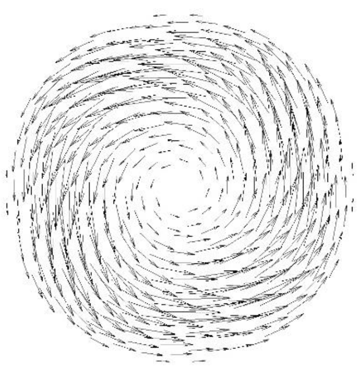
Fig. 5. Adjusted positions of the injectors.



Plane 1



Plane 2



plane 3

Plane 4

Fig. 6. Vectors of circumferential velocity along the tube for the justified case.

To better analyze the generated swirl, the distribution of circumferential velocity, V_c, normalized by axial velocity, V_a (shown in Fig. 3), is plotted against the radius in Fig. 7. It can be observed that in Plane 1, near the injection section, the circumferential velocity has an S-shaped distribution. Due to diffusion, the profile becomes smoother along the path and approaches the linear form which is the characteristic of a forced vortex. All curves present a decrease at the end which is due to wall friction. It can be concluded that almost an ideal forced vortex pattern can be generated in a tube using jet injection, similar to what we have when using inlet guide vanes.

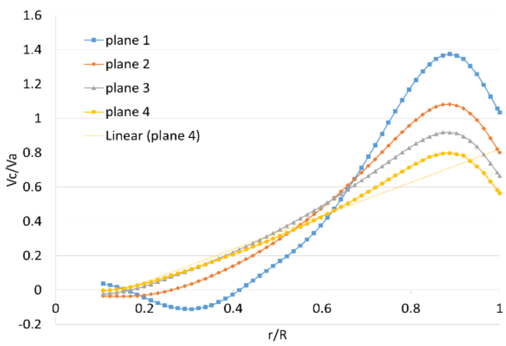


Fig. 7. Normalized circumferential velocity distribution along the tube.

The effect of injection pressure is also investigated at the same mass flow rate and plotted in Fig. 8. It is obvious that stronger jets can induce higher vorticity. At injection pressure of 170 kPa, the distribution is almost linear up to the walls ($r/R \approx 0.9$). By reducing the pressure, the strength of the swirl decreases and the peak happens at lower radii ($r/R \approx 0.65$) and thus losing its effectiveness at 40 kPa. So, it can be predicted that at low injection pressures, this system cannot be beneficial in reducing incidence loss at impeller entry.

3. IMPLEMENTATION ON A COMPRESSOR

3.1 The Initial Compressor

The mentioned system is to be added to the inducer of a centrifugal compressor. Specifications of the compressor are tabulated in Table 1. The exact geometry of the impeller is extracted using precise X-Ray scanners leading to a point cloud and finally converted into CAD model.

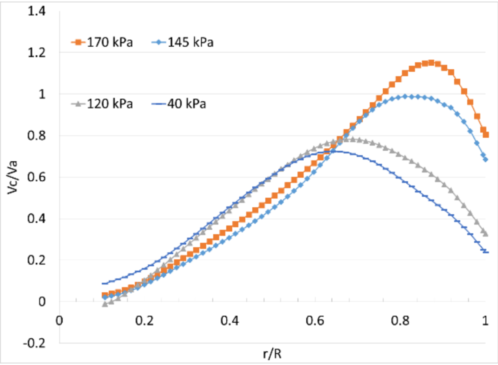


Fig. 8. Effect of injection pressure on induced vortex.

Table 1 Specifications of the investigated compressor

Nominal speed [rpm]	92000
Nominal mass flow rate [kg/s]	0.28
Nominal pressure ratio [-]	2
Inducer diameter [mm]	56
Impeller diameter [mm]	82
Number of blades [-]	12
Impeller exit width [mm]	5.5

The compressor passage is meshed using a structured hexahedral grid which is later used by a commercial CFD software to obtain the flow field inside the compressor. The grid implements near wall elements with 50μ thickness to achieve a y^+ value of less than 100. So, it will be able to catch flow details with sufficient accuracy. An O-grid scheme for leading edges and an H-grid for trailing edge of blades is considered. To save computational cost, only one passage is considered, assuming periodic boundary conditions on domain sides. A mesh independency analysis, showed that at least 480000 elements per passage are required.

Fig. 9. Structured grid of the passage.

To evaluate the accuracy of the numerical model, the compressor has been also tested experimentally at the turbomachinery lab in Sharif University (Fig. 10). Details of the laboratory equipment are described in the work of Moussavi *et al.* (2016, 2017). All results are plotted in Fig. 11 for the compressor at its nominal working point. An error analysis led to an error bar of 2% for the pressure ratio and 4.5% for the isentropic efficiency.

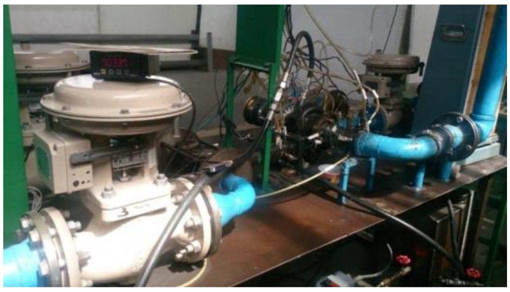


Fig. 10. Compressor test bench.

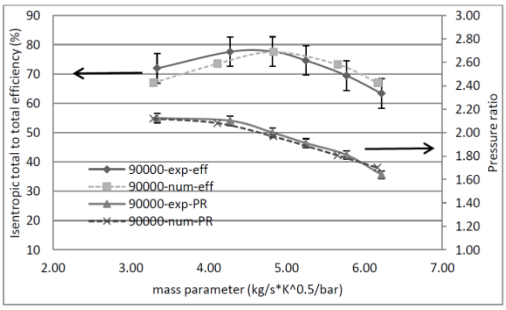


Fig. 11. Numerical and experimental performance of the compressor at 92krpm.

3.2 The Modified Compressor

An extraction port has been created on the volute (Fig. 12), intended to be used for supplying the injection system with high pressure air.

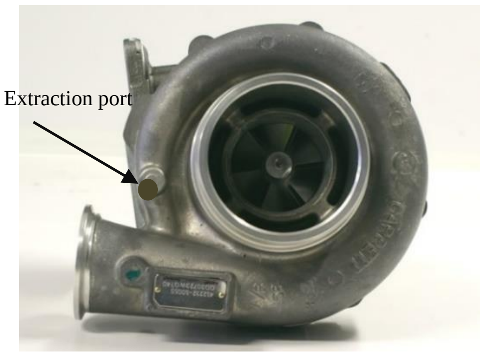


Fig. 12. Extraction point for supply of high pressure injection air.

Here, since each injector passage occupies a 90 degree span and the compressor has 6 passages with a span of 60 degree each, a direct match between them is not possible. In addition, considering the unsteady and non-symmetric nature of surge in compressors (discussed later), all passages are taken into account. So, 4 injector sections are coupled with 6 impeller passages forming the whole assembly as shown in Fig. 13.

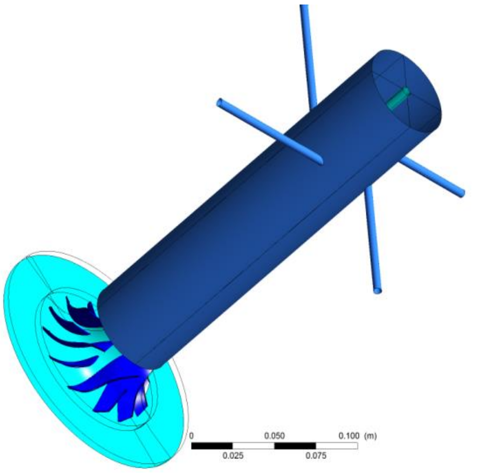


Fig. 13. Model of the compressor and the inlet assembly.

The boundary conditions are set such that at the assembly inlet the total pressure is 1 bar and total temperature is 300K. The compressor is assumed to have a specified mass flow rate at the outlet. Properties of the flow (static temperature and pressure) at the diffuser outlet are transferred to injector tubes. In this work, the k-omega SST model is used and wall heat transfer is neglected.

4. Results And Discussion

At the first step, the effect of injection pressure and flow field at the nominal point are studied and presented in Fig. 14. Injection pressures ranging from 0 to 1 bar (gauge) are investigated. In this figure the first condition (0 bar gauge) corresponds to no injection and the compressor works at its original configuration. Here, the pressure ratio is calculated with total pressure at outlet and inlet of the compressor and the isentropic efficiency being reported considering impeller borders.

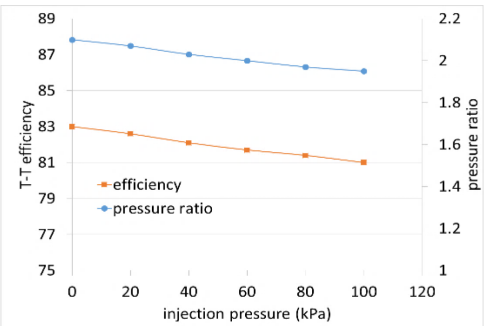


Fig. 14. Injection pressure effect on compressor performance at nominal point.

It is observed that injecting any amount of air and thus creating a free vortex at the inlet, impairs the compressor overall performance. This could be predicted earlier. Because the design point is where the compressor is expected to exhibit its highest performance and thus any deviation from that state can lead to impairment. By this injection, the incidence angle at the leading edge of blades takes a nonzero value which increases the related losses and decreases the efficiency.

Considering another state, the effect of injection pressure is investigated at low mass flow rates (0.14 kg/s) of the compressor and at the same rotational speed (92000 rpm) in Fig. 15. The maximum amount of air being injected in this case is about 0.0225 kg/s.

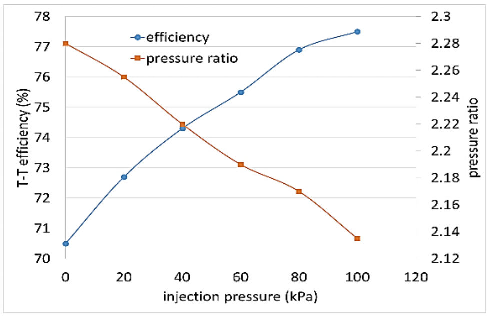


Fig. 15. Injection pressure effect on compressor performance at low mass flow rates.

In this case a change in the plot trends is observed. It can be seen that at low mass flow rate, which is far away from the design point, the injection is beneficial. With higher pressure, a stronger vortex is generated and thus the incidence angle is rectified leading to less incidence loss. So, the efficiency of the impeller has an increasing slope with injection pressure. On the other hand, the pressure ratio has conserved its negative slope. This is due to decrease of blade loading associated with decrease of incidence angle. So, in this configuration an improvement is achieved.

In the next step, injectors are directly connected to diffuser outlet. Thus the air being injected will have the same static temperature and pressure as the compressor delivery and there are no more independent. In Fig. 16 the performance of the compressor with and without the flow rectifier system are compared.

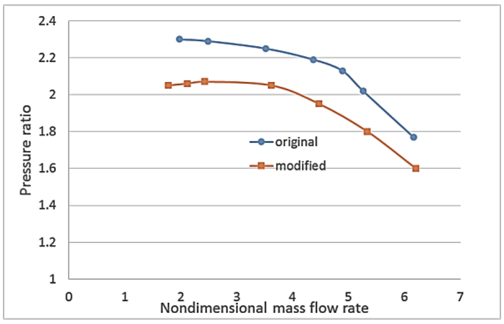


Fig. 16. Pressure ratio comparison of the original and modified compressor.

As expected, it can be observed that the compressor with inlet flow rectifier delivers the air with an overall lower pressure. This effect is already observed in compressors equipped with inlet guide vanes and this is mainly due to the reduction of blade loading caused by lower angles of attack. In Fig. 17, the isentropic efficiency of the compressor is studied.

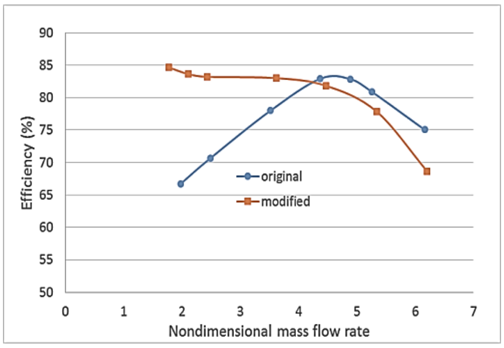


Fig. 17. Efficiency comparison of the original and modified compressor.

Here, a completely different behavior is observed. Curves are crossing each other and thus showing a relative advantage of each configuration over the other one in different conditions. Since the flow rectification system was basically intended to improve the inlet state of the flow at low mass flow rates, the compressor presents higher efficiency in this region (up to 0.18 points). As explained before, reduction of incidence angle and related losses are considered as the main cause of improvement. However, by increasing the mass flow rate and approaching the nominal working point distinguished by the highest efficiency point of the original curve, the system loses its effectiveness. At this point, any change in the entering flow state can lead to deviation from the design conditions and thus decrease of efficiency.

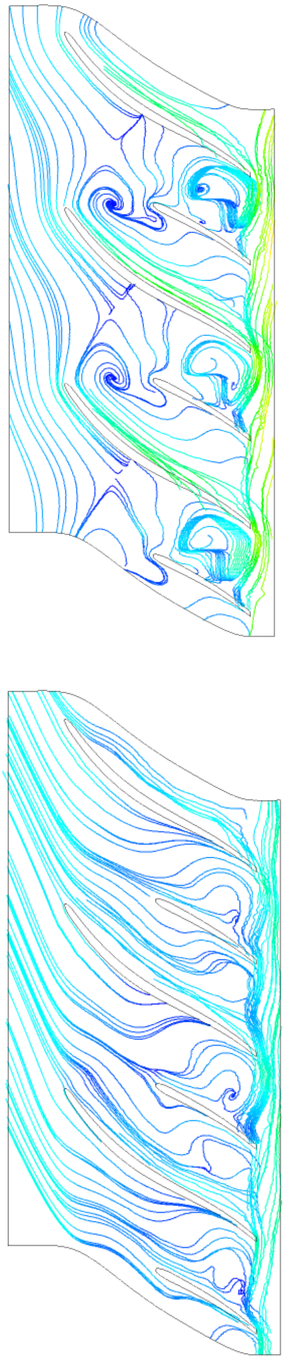
At mass flow rates higher than the nominal point, the incidence angle sign changes. Since the injectors have a fixed predefined configuration they aggravate the situation by adding to the incidence angle. So the curve shows a notable fall below the original curve. It can be concluded that this system must be turned off at high mass flow rates to preserve the original and higher performance.

Another important feature of this system is its effect on surge limit. Results show that by implementing this system, the surge mass flow rate can be decreased by 13% which is a considerable amount.

To better analyze the benefit of the proposed system, the flow field must be captured. In Fig. 18 the relative velocity streamlines at the impeller with its original configuration and with injection pressure of 1 bar at the nominal speed and very low mass flow rate (0.14 kg/s) are shown.

It can be observed that in the original compressor, the passage between two main blades and also splitters is almost totally blocked with vortices. The flow field is disturbed and a single direction for the flow cannot be distinguished. By implementing the air injection system, the flow field seems to be more organized.

The vortices are weaker and the passage is less occupied such that the main flow path from the inlet to the outlet can be followed. Another point is the notable change in the inlet flow direction. In the original compressor, main blades experience a high incidence angle in the order of 30^o which in the modified configuration is dramatically decreased.



(a)

(b)

Fig. 18. Relative streamlines at span 80% on the onset of surge for the (a) original compressor and (b) the one with swirl generator.

Such that the flow can be considered as in line with blades. This item has considerable effect on pushing back the formation of vortices and surge initiation.

In Fig. 19 the pressure loading on main blades near surge condition at (0.11 kg/s) are plotted. The decrease in incidence angle of the modified compressor is clearly seen by a decrease in pressure loading. This was also notable in decrease of pressure ratio of the modified compressor. However, the trend of pressure diagram in each configuration is strictly increasing along the blade which describes a normal situation.

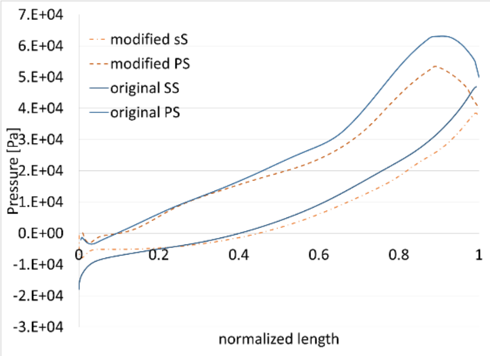


Fig. 19. Pressure loading on main blades near surge.

5. Conclusion

In this research a novel system was presented to improve the performance of a centrifugal compressor at low mass flow rates especially near its surge condition. The system consisted of a swirl generator at the compressor inlet duct. Four Injectors were implemented to generate the swirl to improve the flow incidence angle at the impeller blade leading edge. As the injector flows were supplied from the compressor itself, the system did not need any special equipment or instrument, hence it can be considered as a suitable alternative for IGVs. The advantages of this system are more simplicity, higher reliability, less pressure loss at the inlet and better control.

To generate nearly ideal forced vortex pattern, the specification of injectors was studied and optimized. The results showed that injecting the flow at location 0.75*R*, with adequate pressure results in suitable flow pattern. Implementing the system on a compressor showed that at low mass flow rates, the injection increased the efficiency by improving the incidence angle and enhancing the flow pattern in impeller passages up to 18 %. Furthermore the system improved the surge margin by 13% and decreased the blade loading near this operating condition. But the pressure ratio experienced a drop in all conditions.

References

Arnulfi G. L., F. Blanchini, P. Giannattasio, D. Micheli and P. Pinamonti (2006). Extensive study on the control of centrifugal compressor surge. *Proceedings of the Institution of Mechanical Engineers; Part A: Journal of Power and Energy* 220(3), 289-304.

Bontempo R., M. Cardone, M. Manna and G. Vorraro (2017). A statistical approach to the analysis of the surge phenomenon. *Journal of Energy* 124, 502-509.

De Bellis, V. and B. Rodolfo (2018). Development and validation of a 1D model for turbocharger compressors under deep-surge operation. *Journal of Energy* (142) 507-517.

Dehner, R., A. Selamet, P. Keller and M. Becker (2011). Prediction of surge in a turbocharger compression system vs. measurements. *SAE International Journal of Engines* 4(2011-01-1527), 2181-2192.

Elder, R. L. and M. E. Gill (1985). A discussion of the factors affecting surge in centrifugal compressors. *Journal of engineering for gas turbines and power* 107(2) 499-506.

Fisher, F. B. (1988). Application of map width enhancement devices to turbocharger compressor stages. *SAE paper 880794.*

Gu, R. and M. Yashiro (2005). Surge control for centrifugal compressor of turbocharger. *SAE Paper* 36 (2), 83–88.

Gu, R., S. Mizuki and H. Tsujita (2007). Surge control of centrifugal compressor by inducer tip injection. *Proceedings of the International Gas Turbine Congress (IGTC '07),* Tokyo, Japan.

Halawa T., M. Gadala, M. Alqaradawi and O. Badr (2015). Optimization of the Efficiency of Stall Control Using Air Injection for Centrifugal Compressors. *Journal of Engineering for Gas Turbines and Power* 137 (7) 07260401-07260410.

Hu, L., H. Sun, J. Yi, E. Curtis, J. Zhang, C. Yang and E. Krivitziky (2016). Numerical and experimental investigation of a compressor with active self-recirculation casing treatment for a wide operation range. *Proc IMechE Part D: J Automobile Engineering* 227(9) 1227–1241.

Hunziker R., H. P. Dickmann and R. Emmrich. (2001). Numerical and experimental investigation of a centrifugal compressor with an inducer casing bleed system. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy. 215(6), 783-791

Jansen, W., A. F. Carter and M. C. Swarden (1980). Improvements in surge margin for centrifugal compressors in Centrifugal Compressors, Flow Phenomenon and Performance, *AGARD-CP-282*.

Jiang, P. M. and A. Whitfield (1992). Investigation of vaned diffusers as a variable geometry device for application to turbocharger compressors. *Proc IMechE Part D: J Automobile Engineering* 206 (3) 209–220.

Macdougal, I. and R. L. Elder (1982). The improvement of operating range in a small, high speed, centrifugal compressor using casing treatment. *IMechE Conference, London ,UK.*

Mojaddam, M. and S. A. Moussavi Torshizi, (2017). Design and optimization of meridional profiles for the impeller of centrifugal compressors. *Journal of Mechanical Science and Technology* 31(10)4853-4861.

Moussavi Torshizi, S. A., A. Hajilouy Benisi and M. Durali (2017). Effect of splitter leading edge location on performance of an automotive turbocharger compressor. *Journal of Energy*

123 (2017) 511-520.

Moussavi Torshizi, S. A., A. Hajilouy Benisi, M. Durali (2016). Numerical Optimization and Manufacturing of the Impeller of a Centrifugal Compressor by Variation of Splitter Blades. *ASME Turbo EXPO; GT2016-57105*

Rodgers, C. (1991). Centrifugal compressor inlet guide vanes for increased surge margin. *Journal of Turbomachinery* 113 (4) 696–702.

Semlitsch B. and M. Mihăescu (2016). Flow phenomena leading to surge in a centrifugal compressor. *Journal of Energy* 103(2) 572-587.

Senoo, Y. (1983). Low solidity cascade diffusers for wide flow range centrifugal blowers. *ASME paper 83-GT-3.*

Sivagnanasundaram, S. (2013). An impact of various shroud bleed slot configurations and cavity vanes on compressor map width and the inducer flow field. *Journal of Turbomachinery* 135 (4) 1-10.

Skoch, J. (2003). Experimental Investigation of Centrifugal Compressor Stabilization Techniques. *ASME TurboExpo GT–2003–38524.*

Skoch, J. (2004). Experimental Investigation of Diffuser Hub Injection to Improve Centrifugal Compressor Stability. *ASME TurboExpo; GT2004-53618.*

Stein, A., S. Niazi and L. N. Sankar (2000). Computational Analysis of Stall and Separation Control in Centrifugal Compressors. *Journal of propulsion and power* 16 (1) 1732-1741.