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Review of Efficiency and Stable Operating Range Enhancements Options of Centrifugal Compressors

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Abstract

The high demand for natural gas as an energy source leads to place more focus on designing high efficiency centrifugal compressors with wider stable ranges. However, the interaction between the aerodynamic of the single compressor components makes designing the high efficient machines with the wide flow range more complicated. Thus, this paper outlines the development trend of the aerodynamic design of the centrifugal compressors components over the decades driven by the high stage efficiency and wide stable operating range. Additionally, this study will investigate the influential aspects of the unsteady interaction between the aerodynamic stage components and the consequential effect on the compressor efficiency and stable flow range. This covers the discussion of the numerical investigation findings and the experimental observation made on the centrifugal compressor performance.

Keywords: Efficiency; Centrifugal compressors; Operating range; Unsteady interaction; Design characteristics; Aerodynamic

Abbreviations: LSVD: Low Solidity Vaned Diffusers; PR: Pressure Ratio; VD: Vaned Diffuser; VLD: Vaneless Diffuser

Nomenclature

β_2	Impeller Blade Exit Angle
$\dot{m}$	Mass Flow Rate
V_r	Radial Flow Velocity
V_θ	Tangential Velocity
π	Pressure Ratio
η	Efficiency
ρ	Density
r	Diffuser Radius

Introduction

The first centrifugal compressor was used in the process industry in conjunction with the early gas turbine at the beginning of the 20th century. Today, the centrifugal compressors are used for variety of applications in oil and gas industry thus, they are considered as one of the fundamental component. Unlike the axial compressors, the flow leaves the centrifugal impeller radially due to the centrifugal and Coriolis forces within the impeller.

The centrifugal compressor consists of three basic sections which are: inlet section, impeller section and diffuser section. When the gas enters the suction nozzle, it is accelerated by reducing the flow area and then it is directed to the inlet section axially. In this section, the discharge gas from the convergence nozzle flows toward the impeller eye where it meets the leading edge of the inducer blades.

The inducer is essential to change the tangential motion of the fluid to radial direction to ensure a smooth inlet flow to the impeller section. The absence of inducers might lead to fluid separation and violent mixing near the leading edge of the impeller vane which in turn can be very noisy. In addition, the inlet section may consist of suction elbow and guide vanes prior the inducer to provide the working fluid with some degree of prerotation and to direct the flow in the desired direction to the inducer. The impeller is a very critical component to achieve a high efficiency over a wide operating range. The rotational speed of the impeller blades provides the required kinetic energy to the working fluid and a part of this energy is converted to pressure.

Several studies have been conducted with attempt to improve the machine efficiency and aerodynamic stability by optimizing the design features of the basic compressor components. A remarkable improvement in the compressor efficiency with relatively wider operating envelop has been accomplished over the years as shown in Figure 1. The plotted trend is based on the obtained efficiencies values by Sorokes et al. [1]. The slope of the efficiency change is clearly greater in the early years and it is decreasing gradually as it is approaching the asymptotic efficiency limit. This upward trend was achievable by improving the material capability and the design features of the compressor component.

However, raising both parameters is still a challenge for the manufacture since each one is attained in expenses of the other. Therefore, it is important to investigate the contribution of design characteristics on stage efficiency and operating flow range.

The present paper will review the development trend of the centrifugal compressor components aiming to achieve a higher operating efficiency and wider flow range. Moreover, it will investigate the influential aspects of the unsteady interaction between the stage components and the potential effect on the aerodynamic stability.

This paper has been structured in three main sections to investigate the potential influence of the development in the aerodynamic components design characteristics on the compressor

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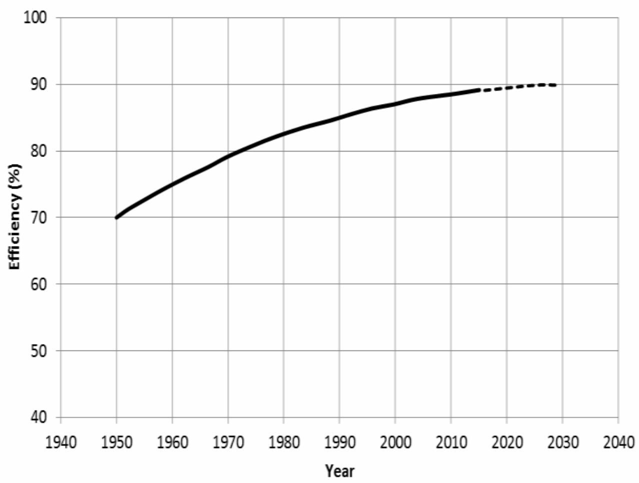

Figure 1: Derived Centrifugal Compressor Efficiency Trend over the Years with Flow Coefficient Greater than 0.08 Based on Published Data by Sopkes et al. [1].

efficiency and stable operating range. In order to achieve that, it includes the discussion of the basic aspects of the aerothermodynamic performance of centrifugal impeller, diffuser and volute to evaluate the enthalpy losses and flow unsteadiness inside the centrifugal compressor casing. This covers the experimental observation results and numerical analysis findings.

Investigation of impeller design influences on stage efficiency and flow range

The impeller is responsible to impose the whole required kinetic energy to the working fluid and to produce approximately 70% of the pressure rise in the stage [2]. Sable [3] stated that an about the half of compressor pressure rise is achieved by the impeller in practice. So, it is important to place more focus on the design and selection of the impeller geometry. Three basic characteristics will be considered regarding to impeller design which are: presence of shroud, exit blade angle and dimensional axis.

The results of the experimental studies on the impeller flow demonstrate that the velocities distribution on the blade surfaces is different from the predicted flow profile theoretically [4]. This is basically due to the created secondary flows by the pressure losses and boundary layer separation in the impeller blade passages. The viscous shearing forces in the centrifugal impeller form a boundary layer when the kinetic energy is reduced. Hence, the reduced kinetic energy should be kept higher than certain limit to avoid the stagnant and reverse flow.

The shroud on centrifugal compressor impeller has a significant effect on the internal flow of the impeller. The absence of cover allows the unshrouded impellers to operate at higher rotational speed and to generate a higher pressure ratio relative to the shrouded impellers reaching to 4:1 based on the gas conditions and impeller tip speed. Most of the shrouded impellers are capable to produce a pressure ratio of 3:1 or less. Beaty [5] specifies the generated pressure ratio between 1.5:1 to 2:1. Moreover, the rotating stall is taking place at higher flow rates in the case of shroud impellers which leads to have a narrower stable range [6]. However, based on the efficiency view, the open un-shrouded impellers are suffering from the tip leakage loss which leads to make them less efficient than the covered impellers.

Figure 2 demonstrates that the trends of isentropic efficiency and pressure ratio are increasing non-linearly with the shrouded percentage rise. So the selected shrouded percentage should be optimized to compromise between the cost and performance. Jin Tang et al. [7] investigated the effect of using partial shrouded impellers on the compressor performance. This study reported an improvement in the tested compressor performance with partial shrouded impeller than when the un-shrouded impeller is used and the flow uniformity becomes better at the exit of the impeller. The secondary flow region caused by the leaking flow is reduced at the impeller exit and closer to the shrouded impellers.

To be capable of dealing with higher flow coefficient, the three dimensional shrouded impellers have been designed. Based on the aerodynamic prospective, the impeller flow coefficient is one of the most important criteria which affect the selected impeller type. The low flow coefficient impellers are designed with 2D structure and with long and narrow passages. Besides, they are used mostly in the later compressor

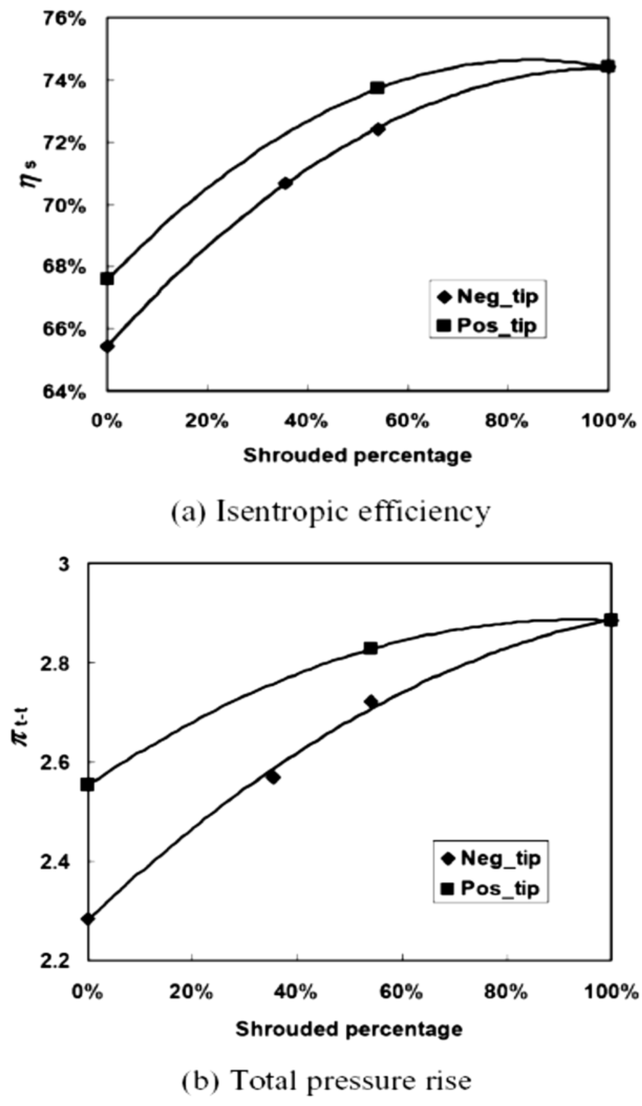

Figure 2: Effect of Shrouded Percentage on (a) isentropic efficiency (b) pressure rise [7].

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stages where the pressure ratio is higher and the volumetric flow rate is lower. Albusaidi et al. [8] demonstrated that the maximum flow coefficient of 2D backward shrouded impellers is approximately 0.09 in order to avoid the high stage efficiency drop as illustrated in Figure 3. This in fact agrees with the obtained value by Lüdtke [9]. On the other hand, the impellers with high flow coefficient have more complex structures (3D) with wider passages to accommodate the higher flow rates. Despite this complexity in the structure, the 3D-impeller delivers a higher efficiency at wide operating range with greater rotational speed comparing with the previous type. Therefore, several researches are taken place to extend the flow range of the 3D-impeller structure to cover lower flow coefficient region.

The impeller blade angle is another design factor that influences the centrifugal compressor performance as illustrated in Figure 4. The forward-leaning blades (β_2 > 90) provide a positive sloping head curve and the maximum head output since the tangential component of relative velocity (V_θ) is increasing with increasing flow. Although that the generated dynamic head by these vanes is relatively higher, the conversion from the kinetic energy to pressure is not efficiently done in

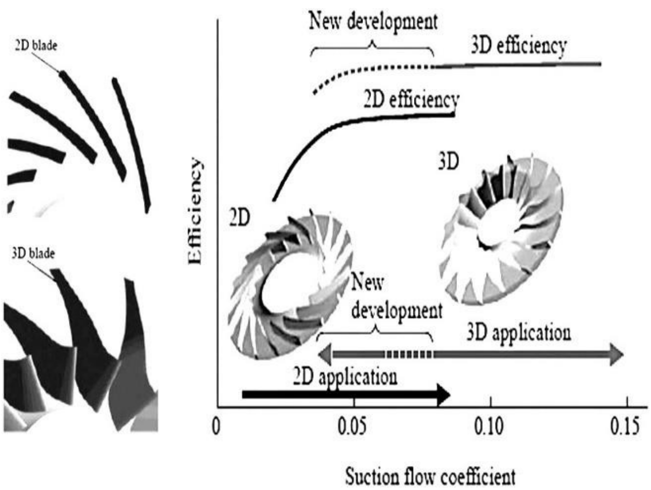

Figure 3: Selection of 2D and 3D Shrouded Impellers Based on Flow Coefficient [8].

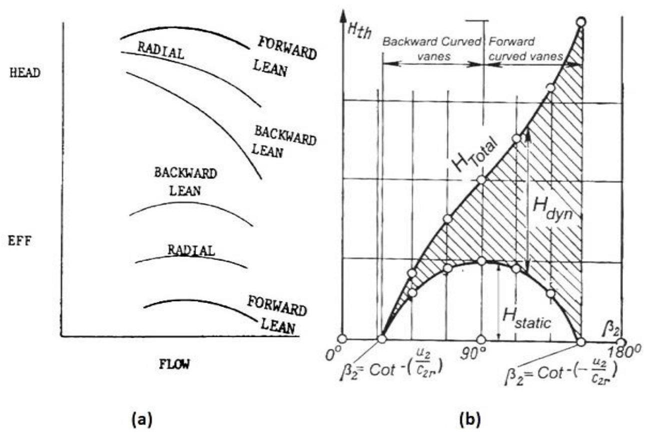

Figure 4: Effect of Impeller Blade Angle on Compressor Head and Efficiency [10].

Vanes Type	Flow Rate and Size	Pressure Ratio	Output
Forward curved vanes	Small Volume and size	High	High speed, High noise, Low efficiency
Backward curved vanes	Large volume and size	Low to high	High efficiency, Low noise
Radial vanes	Medium volume and size	Medium to high	Good efficiency

Table 1: Comparison between impeller vanes.

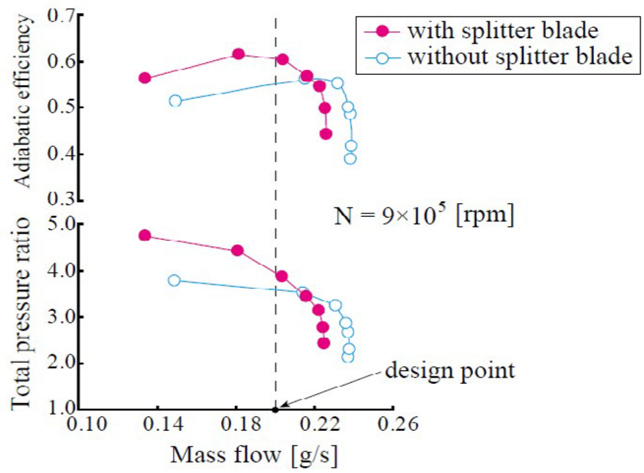

Figure 5: Comparison Between the Performance of Splitter and Conventional Blades [11].

the forward-leaning blades which results in lower overall compressor efficiency. In the straight radial blades impeller (β_2 = 90), the tangential velocity does not change with flow which leads theoretically to a constant head curve. Looking to backward-leaning blades (β_2 < 90) and as the flow decreases at constant speed, the relative velocity decreases which consequently results in a higher tangential velocity (V_θ) and greater output head.

Generally, the highest overall stage efficiency can be achieved by using backward-leaning impellers while the minimum value is expected with forward-leaning blades. Moreover, as the backward lean increases, the achieved efficiency increases but in penalty this leads to lower head as shown in Table 1. So, the blade angle should be selected to achieve the best fit with the desired head and efficiency requirements in any particular application.

To avoid the choking conditions in the inducer, many compressors incorporate splitter vanes. The adiabatic efficiency and stage pressure ratio can be improved significantly as demonstrated in Figure 5 due to the boundary layer separation and total pressure loss reduction. However, a noticeable reduction in the surge margin was observed with splitter vanes. Moreover, the use of the splitter blade is expected to rectify the flow at the impeller discharge outlet yielding to higher absolute velocity. This consequently induces a larger pressure loss by wall friction at diffuser inlet comparing with the conventional blades. Therefore, it is necessary to consider the integrated aerodynamic performance of impeller-diffuser interaction.

The traditional method to position the impellers on the shaft is by placing them one after the other and this arrangement is called by in-line impeller arrangement. The amount of the power loss as a result of flow recirculation in the balance piston is relatively high for this type

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of arrangement and it roughly fluctuates between 1% and 3.5 % based on the flow coefficient and pressure ratio. To reduce this loss in power, another method of impeller arrangement was developed by placing two sections with one common casing so that the two outlets adjacent to each other and it is called back-to-back arrangement. The advantage of this kind of configuration is that the balance piston has been replaced by a rudimentary labyrinth seal between the outlet stages of each section so that the leakage flow is re-circulated only through the latter section as shown in Figure 6. This yields to reduce the resulted power loss due to flow recirculation to about one third of that in the straight through impeller configuration which is very significant especially for high pressure ratio applications. Vijay [13] estimated the total save in horsepower by using back-to-back arrangement to approximately 4-6% lower than that of series arrangement. Figure 7 illustrates the resulted variation in the axial thrust bearing loading due to the speed change in both types of arrangement. It is clear that the encountered variation in the back- to-back arrangement is about five times less than that of series arrangement. Furthermore, this arrangement is ideal to reduce the net unbalance thrust caused by the increase in the labyrinth clearance.

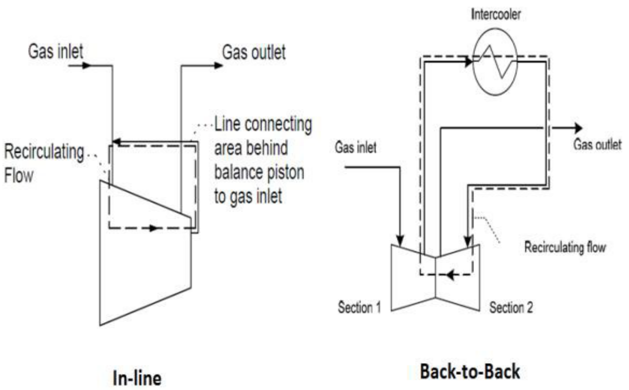

Figure 6: Types of Impeller Arrangement [12].

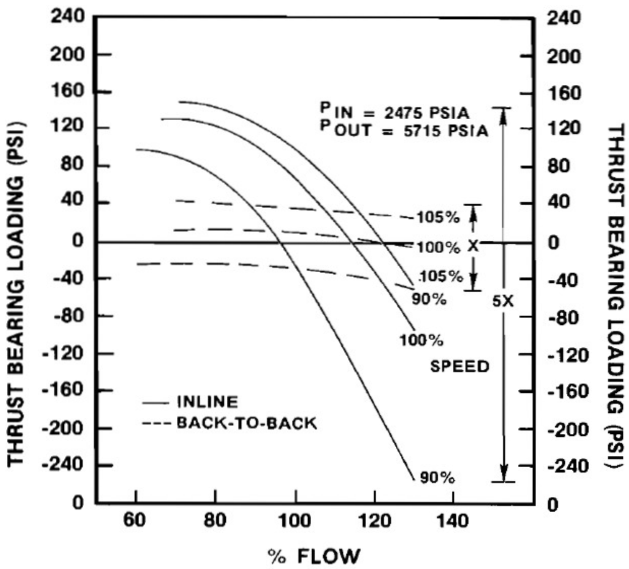

Figure 7: Effect of Impeller Arrangement on Thrust Bearing Loading Variation [13].

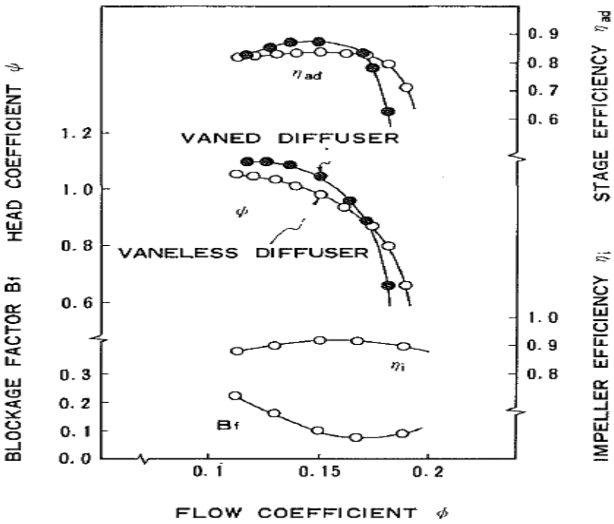

Figure 8: Performance Characteristics of Vaneless and Vaned Diffusers [15].

Studies on configurations and design characteristics of the diffuser

An efficient diffusion system at the impeller exit is essential requirement for high performance centrifugal compressors to convert the imposed kinetic energy by the impeller into the maximum static pressure recovery over a wide range of incident flow conditions. However, there are several limitations for the diffusion process in order to avoid the chance of the stall occurrence. Based on the impeller design and its matching with the diffuser component, the diffuser characteristics can greatly influence the location of stability limit of the centrifugal compressor. The conducted studies in the centrifugal compressors stall are significantly less extensive that that of axial compressors. The complexity of rotating stall in centrifugal compressor makes the postulation of any generalized correlation one of the challenges in this field.

A higher operation efficiency of centrifugal compressor can be achieved by improving the pressure recovery characteristics of the diffuser. It was generally found that a well-designed vaned diffuser has higher static pressure recovery coefficient (C_p) leading to greater stage efficiency and higher head coefficient as shown in Figure 8. This is in fact owing to the long logarithmic particle spiral path in the vaneless diffuser which leads to great frictional loss. On the other hand, the absence of a throat in vaneless diffuser makes this configuration ideal for wider operating range applications.

The vaneless diffuser has a simpler geometry which makes it easier for manufacturing and more tolerant to erosion and fouling than the vaned diffusers. However, the absence of vanes requires a large diameter ratio because of its low diffusion ratio. Brown [14] reported the largest total pressure losses at the diffuser entrance while the minimum loss was observed at the intermediate section. There are different types of vaneless diffusers which are classified based on the cross sectional shapes and wall configurations which yields to make the selection of the optimum channel diffuser is complicated. On the other hand, the vaned diffusers are classified into four main categories as illustrated in Figure 9:

a. Airfoil style diffuser which is the most common type and it is based on conventional standard cascade technology where the swirl

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velocity component is reduced by the turning vanes.

b. Channel diffuser at which the passage area is controlled by increasing the vane thickness with radius.

c. Low solidity vaned diffuser where there is no throat area.

d. Rib diffuser which is a low solidity set of diffuser vanes attached to the shroud surface so they extend over 25% to 60% of the diffuser passage.

The peak efficiency of compressors with vaned diffusers is higher by an extent of about 4% over the vaneless configuration which varies with the flow coefficient. However, there are three main disadvantages of vaned diffuser:

- The mechanical complexity of vaned diffuser raises the manufacturing cost leading to higher capital cost.
- The presence of vanes limits the available flow area and causes significant changes in the steady flow field which in turn results in a shorter operating margin.
- The significant fluctuation in the process gas molecular weight causes a high incidence angle to the entry of the vaned diffuser.

Figure 10 demonstrates the level of strain during the compressor surge of different diffuser structures with alternating blade stress of about 106 N/mm^2. The acting strain on the cambered diffuser vanes is the highest among the studied configuration so it is crucial to consider the sufficient surge margin to avoid the high fatigue and stress loads on the mechanical integrity of the blades.

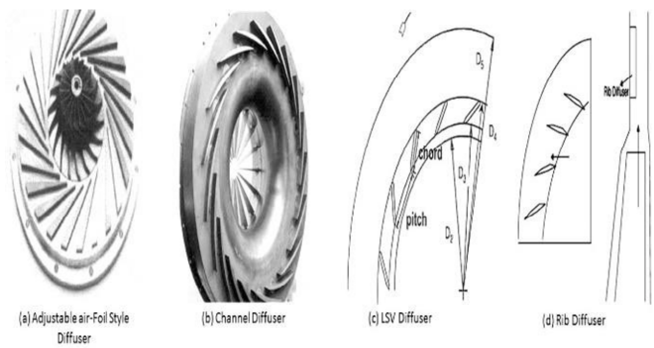

Figure 9: Types of Vaned Diffuser [4].

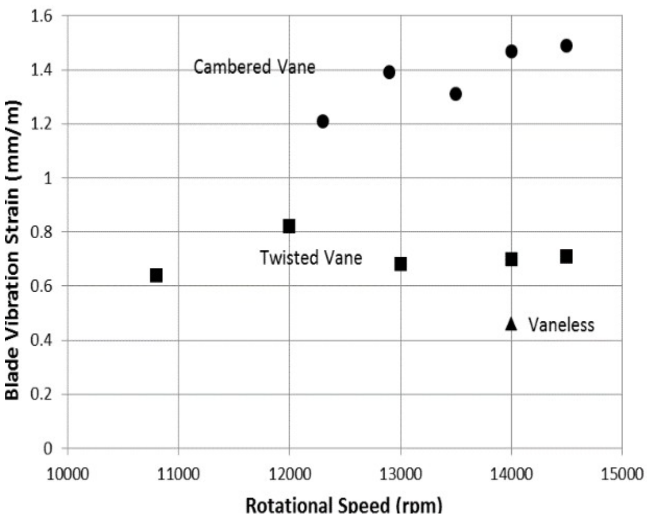

Figure 10: Blade Vibration Strain of Different Diffuser Configurations [16].

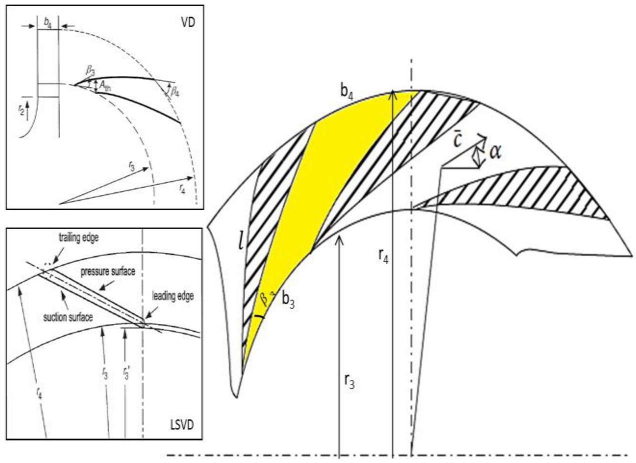

Figure 11: Diffuser Design Characteristics [17,18].

There are five major design parameters affect the overall performance of the diffuser as illustrated in Figure 11 which are the diffuser width, radius ratio, chord length, vanes number and inlet vane angle. One way to delay the stage stall is by increasing the inlet flow angle at lower flow rates which can be accomplished by several techniques. The simplest way is by reducing the axial width of the diffuser and this type of diffuser geometry is called pinched diffuser. The pinched shape of diffuser has been designed to improve the inlet flow and it might be made to the shroud inlet or hub inlet or both of them as shown in Figure 12. The conducted study by Jaatinen et al. [19] on the effect of vaneless diffuser width revealed a possible chance to improve the compressor efficiency by reducing the diffuser width. Ferrara et al. [20], Engeda [21],Cellai et al. [22] and Wu [23] reported that by reducing the diffuser width, the stall inception shifted towards lower flow rates.

Teemu et al. [24] recorded an increase in the stage pressure ratio and isentropic efficiency of the pinched vaneless diffuser at all mass flows and the largest increment was observed at high flow as illustrated in Figure 13. The stator and rotor efficiencies improved while there was a reduction in the static pressure rise coefficient of the diffuser at low and at the design flows. On the other hand, a drop was reported in the efficiency and static pressure rise coefficient of the volute. The flow field measurements showed a higher degree of flow stabilization in the pinched diffuser flow and with more uniform distribution of the flow angle at the diffuser outlet. Similar results have been obtained by Mohtar et al. [25]. Jaatinen et al. [26] concluded that the shroud pinch is much more beneficial to the efficiency comparing with hub pinch. Guofan [23] recorded a drop by about 4% in the choke flow as a result of 1 mm reduction in the axial diffuser width. However, the reduction in choke flow as a result of axial width is exceeding the surge flow drop yielding to shorter stability range. Moreover, the excessive pinching can have an adverse impact on both efficiency and operating range.

It is important to mention that the absence of vanes in the flow

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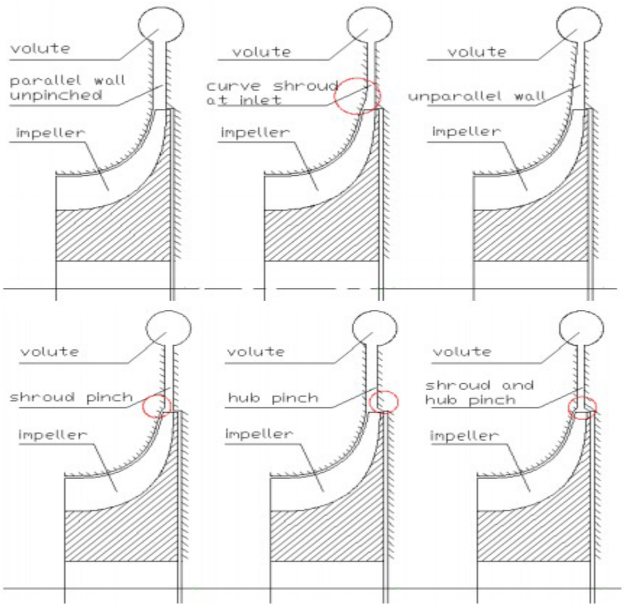
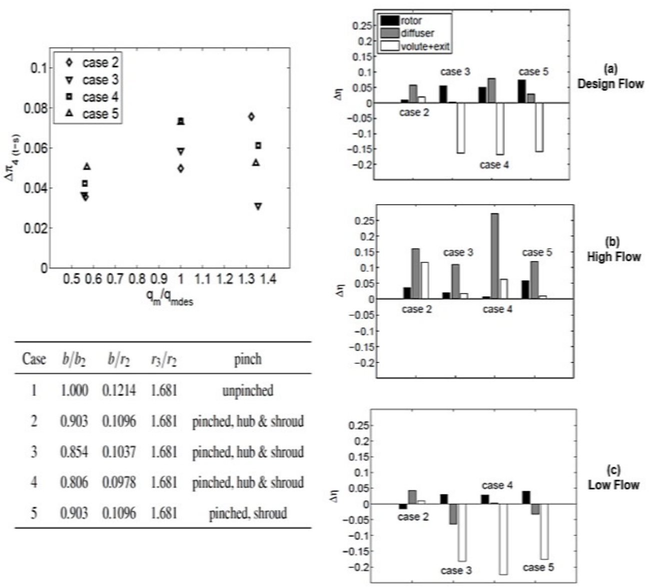


Figure 12: Vaneless Diffuser Structures [23].



Case	b/b_2	b/r_2	r_3/r_2	pinch
1	1.000	0.1214	1.681	unpinched
2	0.903	0.1096	1.681	pinched, hub & shroud
3	0.854	0.1037	1.681	pinched, hub & shroud
4	0.806	0.0978	1.681	pinched, hub & shroud
5	0.903	0.1096	1.681	pinched, shroud

Figure 13: Effect of Axial Diffuser Width on the Operating Range andIsentropic Efficiency [24].

passages of vaneless diffuser necessities the need for larger radial size to achieve the same pressure ratio as the vaned diffuser. Consequently, the frontal area of vaneless diffuser is usually larger and it influences the flow uniformity in the diffuser passage resulting in a higher pressure losses. Considering incompressible fluid and assuming constant axial width, the continuity equation can be written in the following form:

$$\dot{m} = \rho \times (2\pi r h) \times V_r = constant \quad (1)$$

From conservation of angular momentum, the absolute flow angle of the diffuser (∝) which is the angle between the flow velocity and the radial direction can be determined using equation (2). This confirms the inverse relationship between the diffuser radius and the velocity.

$$\alpha = \frac{V_\theta}{V_r} = constant \quad (2)$$

The vaneless diffusers (VLD) are typical to provide a wide operating range but the growing demand of a higher compressor efficiency pushes to use the vaned diffusers (VD). The vaned diffusers are based on changing the mean flow path radius to reduce the gas velocity. Figure 14 illustrates that vaned diffuser configuration is capable to provide a higher pressure rise and efficiency at design flow coefficient but with reduced operating range relative to the previous type as obtained by Flathers [27]. This feature makes the vaned diffusers ideal for higher pressure ratios applications especially when and there is no critical need to adjust the flow rate.

Kim et al. [17] compared between flat plate cambered and airfoil cambered diffusers and the obtained results revealed that the flat plate cambered diffuser has the highest peak efficiency but its operating flow range is narrowest. Comparing with discrete passage diffuser, the straight channel diffuser has a greater overall diffuser pressure recovery by about 10% while the variation in the pressure recovery coefficient with inlet flow angle was found similar for both types. The main geometrical difference between these two types is the position of the throat which is at 40% and 15% of non-dimensional distance along the diffuser passage canter-line of discrete passage and straight channel diffusers respectively, as shown in Figure 15. Haupt et al. [29] reported a shift in the surge line toward higher mass flow rates for the stage with the straight channel diffuser comparing with the cambered vane type. However, the stage efficiency was higher in the case of cambered vane as illustrated in Figure 16. The off- design performance of vaned diffusers is poorer than the vaneless diffuser due to the formation of a geometric throat at the diffuser vane which in turn leads to lower choke flow.

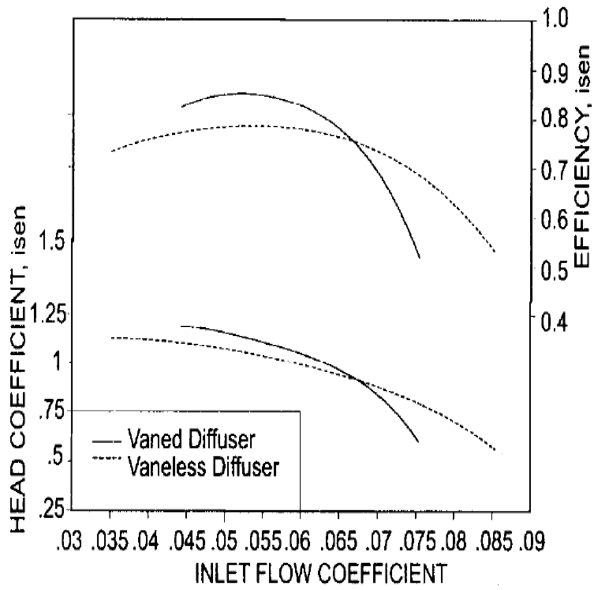


Figure 14: Comparison Between the Head Coefficient and Efficiency ofVaned and Vaneless Diffusers [27].

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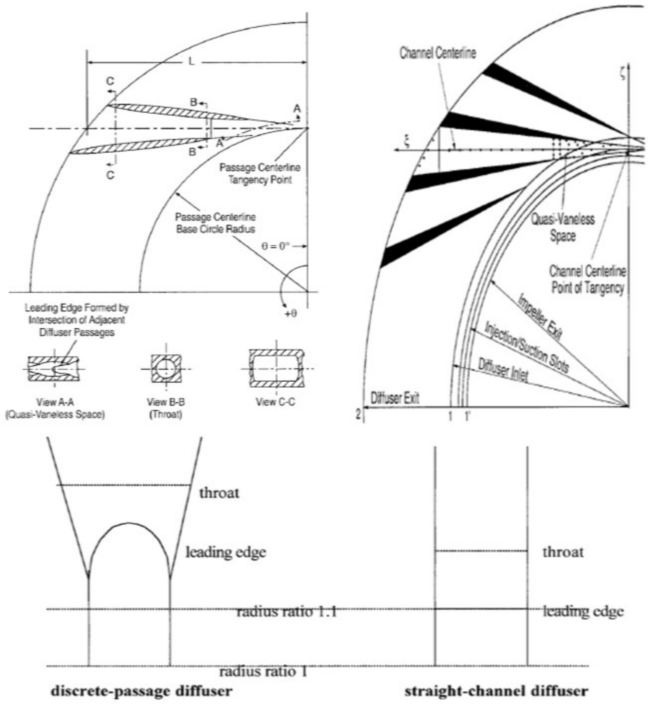

Figure 15: Comparison between discrete passage diffuser and channel diffuser [29].

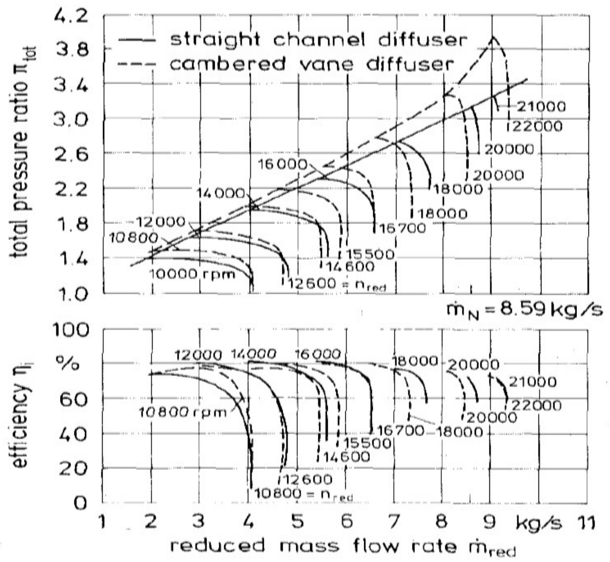

Figure 16: Performance Characteristics of Straight Channel andCambered Vane Diffuser [30].

This can be avoided by reducing the number of diffuser vanes leading to lower solidity percentage. Figure 17 plots the obtained results by Wu [23] which shows insignificant effect on the chock flow as the vanes number increases. On the other hand, there is a substantial increase in the surge flow with the vane number rise leading to reduce the flow range dramatically. More significant effect on the choke flow was observed by changing the inlet vane angle of the diffuser. The widest mass flow range was achieved at the highest angle value with an increase of 27% in the flow range relative to 7° rise in the vane inlet angle.

Based on same concept, the low solidity vaned diffusers (LSVD) are characterized by the absence of geometric throat which is the minimum channel area made by two adjacent diffuser vanes. The low-solidity vaned diffuser is used widely because of its higher efficiency compared with the vaned diffuser and its compatible operating range with the vaneless diffuser. Unlike the vaned diffuser, the choke limit is controlled by the impeller throat area as Mach number approaches unity. This allows for more significant variation in the choke limit based on the impeller geometry and according to the rotational speed value. Hohlweg et al. [30] recorded a reduction by around 2.6% in the efficiency with LSVD at high Mach number and at design flow conditions comparing with the conventional VD. Moreover, the flow range of LSVD was 30% higher than that of the VD and its efficiency was 4.9% higher than the efficiency of the vaneless configuration. Similar results have been obtained by Osborne et al. [31], Sorokes et

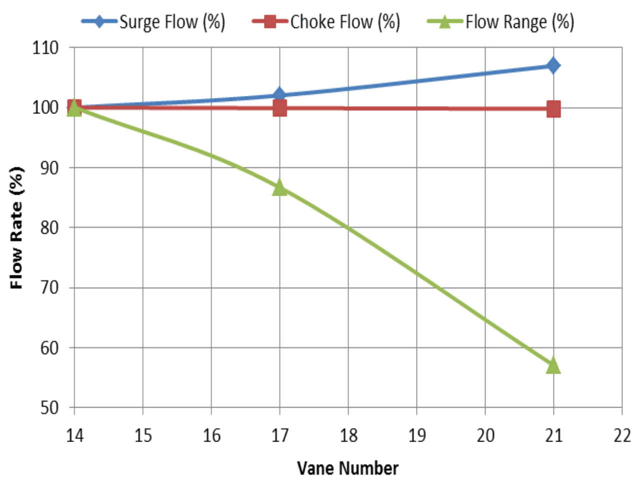

Figure 17: Impact of Vanes Number on Operating Range Considering the Diffuser with 14 Vanes As Base Point Based on Published Results by Wu [23].

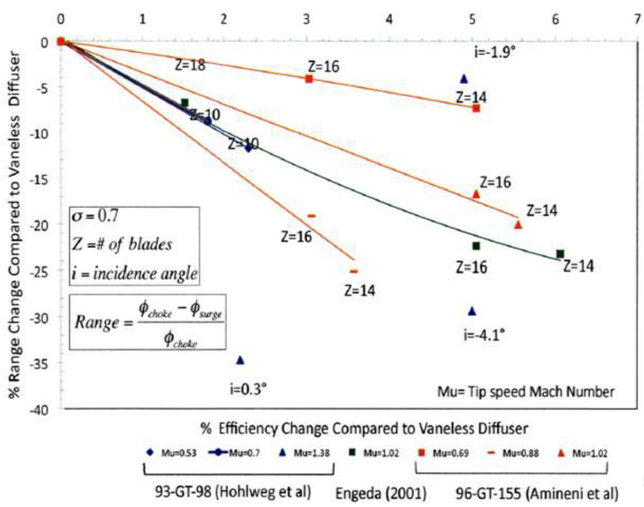

Figure 18: The characteristics of LSVD Relative to Vaneless Diffuser [41].

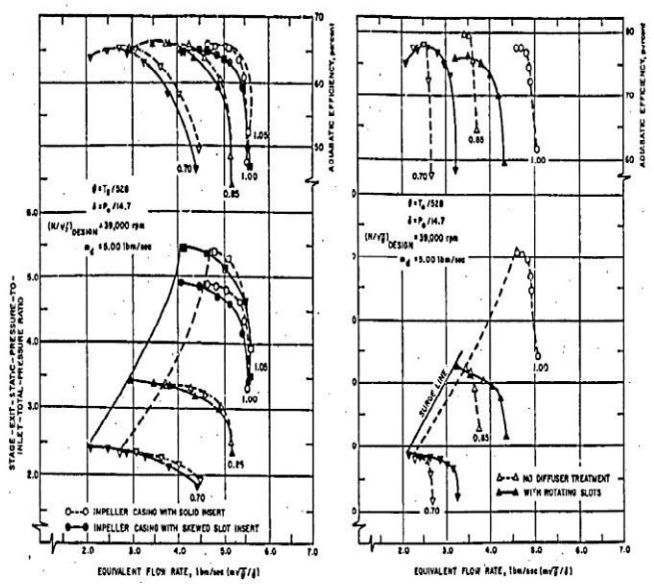

Figure 19: The Experimental Results of Impeller Wall and VanedDiffuser Treatment [42].

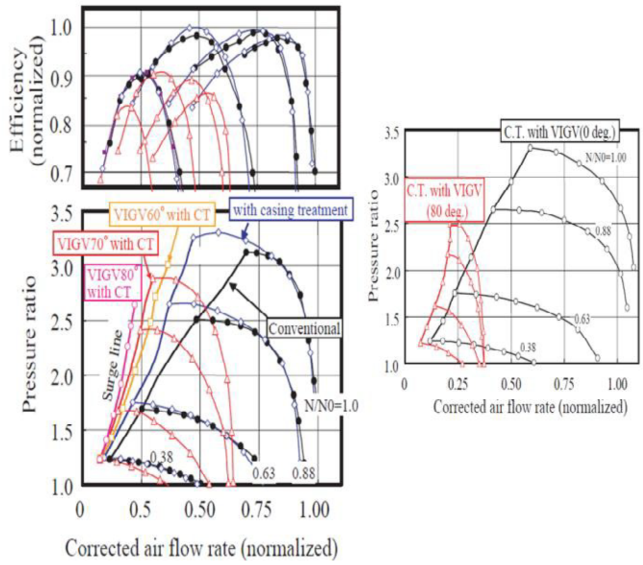

Figure 20: Effect of Casing Treatment and VIGV Setting Angle of 35º Backward Impeller at Left Side and with 20º Backward Impeller at Right Side [44].

al. [32], Amineni [33], Koumoutsos et al. [34], Siva Reddy et al. [35], Engeda [36], and Issac et al. [37].

Figure 18 shows the efficiency and operating range of LSVD relative to the VLD as a function of tip Mach number and blades number. The gain in the efficiency due to the use of LSVD is fluctuated between 2% and 6% comparing with the vaneless diffuser with penalty of flow range reduction which reaches to about 25% in some cases. However, the drop in operating range can be reduced by decreasing the incidence angle and tip Mach number. Cellai [22] studied the effects of the vane length on the operating range and the obtained results demonstrated a reduction in the surge flow coefficient when the chord length increases but with lower choke flow. Yoshinaga et al. [38] found that the optimum vanes height to achieve the greatest pressure recovery was little less than half of the diffuser width. A greater advantage was achieved when the partial vanes were fixed to both the hub and the shroud. Liu and Xu [39] concluded that the performance of the centrifugal compressor stage with shroud vane diffuser is more efficient than that with the hub vane diffuser at the same vane height.

Sorokes et al. [32] considered the influences of using an adjustable low solidity vaned diffuser so that the inlet angle of the vanes can be adjusted. It was found that the long vanes located at shorter radius ratio show better results. Besides, using the rotatable LSVD helps to enhance the surge margin by adjusting the inlet angle.

At high pressure ratio, the separation in the vaned diffuser has a significant effect on the stage performance leading to lower efficiency [40]. For that reason, the diffuser with splitter vanes can be considered as a good way to enhance the diffuser performance by reducing the flow separation and smoothing the diffuser flow. The compressor stability limit is also influenced by the diffuser inlet- to-outlet radius ratio. The results of Ljevar [41] revealed that the critical flow angle decreases with the reduction in the diffuser radius ratio so that the stability range improves. This result totally agrees with the findings of Abdelhamid et al. [42] and Tsujimoto et al. [43].

A number of conducted investigations have demonstrated several

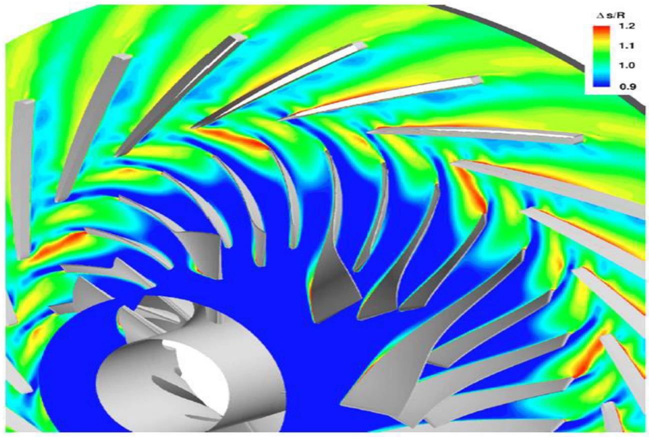

Figure 21: Entropy Contours at Mid-Span [51].

techniques to extend the stable flow range of centrifugal compressors. Jansen [44] performed an experimental test to extend the surge margin by treating the impeller wall and vaned diffuser so that the flow rate is reduced at regions where the surge always occurs.

The treatment of the impeller wall was carried by creating a series of slots adjacent the leading edges of the impeller vanes. This method achieved a remarkable improvement in the operating range both in terms of surge margin and choke flow characteristics as shown in Figure 19. On the other hand, the vaned diffuser treatment was accomplished by creating circular grooves in the wall with slotted diffuser vanes. This technique showed a less significant improvement in the stable flow range comparing with the previous method. However, an appreciable drop in the efficiency was observed in both methods. This conclusion was emphasized also by MacDougal and Elder [45]. Hiroshi Uchida [46] investigated the integrated effect of casing treatment with the VIGVs as illustrated in Figure 20. The drop in surge flow became greater when the VIGVs are associated with the casing treatment reaching to

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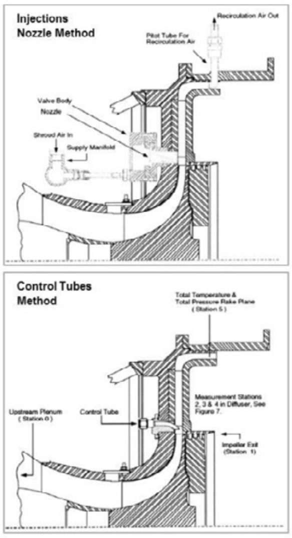

Configuration	A surge margin, percent	B Δ surge margin, percent	C Δ PR at surge, percent
Baseline	11.7		
0.9% forward-tangent	13.4	1.7	−1.0
1.7% reverse-tangent	12.7	1.0	−2.6
4.6% reverse-tangent	9.7	−2.0	−11.0
15% control-tubes	16.1	4.4	−1.4
50% control-tubes	18.2	6.5	−5.5

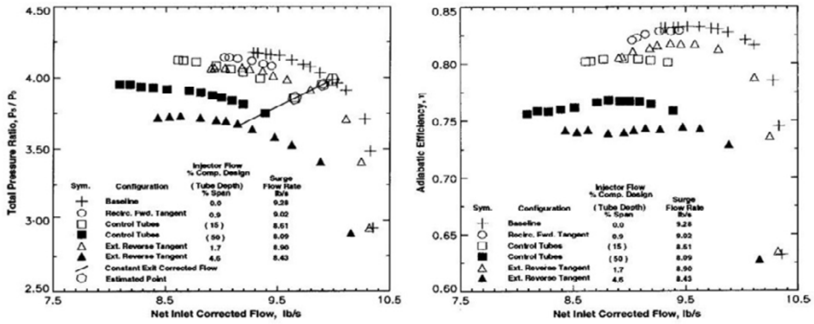

Figure 22: Effect of Injection Nozzle and Control Tubes Techniques on Stage Eficiency and Aerodynamic Stability [51].

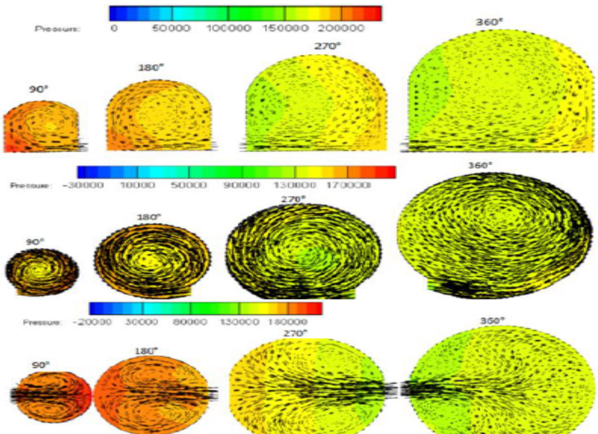

Figure 23: Static Pressure Contours and Meridional Velocity Vectors for Semi-Circular and Circular Volute with Tangential and Symmetrical Inlet [56].

about 59% comparing with only 15% and 30% when the VIGV and the casing treatment, respectively, were used separately.

Raw [47] reported an improvement in the surge margin with a conical pipe diffuser by implementing porous drillings to bleed flow from the diffuser throat region. Nelson et al. [48] used a steady flow and pulsed injection of air into the diffuser channels through slots in the suction vanes side to stabilize the axial-centrifugal compressor of a turbo-shaft engine. Stein et al. [49] found numerically that the air injection eliminated the initiated local separation by flow reversal with an improvement in the impeller stability.

The space between the impeller tip and the diffuser vanes is critical to guide the impeller flow to the diffuser and to ensure an effective diffusion process. This space serves to reduce the diffuser inlet Mach number and to settle the flow before reaching the leading edge of diffuser vane. However, this space is not a basic part since there are some compressors are designed without this gap. For better performance, it is important to consider the right space since the large space leads to increase overall radial size and the boundary layer growth which in turn yields to a flow blockage in the diffuser while the excessive short space causes a high level of noise, vibration and stresses. Figure 21 shows the instantaneous entropy contours of centrifugal compressor stage with vaned diffuser. It can be seen clearly that the high entropy is occurred at the trailing edges of the impeller vanes due to the flow separation. These wake regions are transferred with the flow through the vaneless space where its velocity is reduced slightly and then hit the diffuser vanes with high relative velocity. The impacted flow on the diffuser vanes causes a flow separation and wakes along these vanes. This in consequence, creates a high entropy at the leading edge which is reduced gradually while the gas flows into the vanes trailing edge. However, these high separation regions cause Mach number to increase yielding to low stage efficiency and shorter flow range. The induced flow unsteadiness by the upstream impact of diffuser vanes leads to larger viscous losses associated with the tip leakage. This effect was found more significant than the unsteady flow influence on diffuser performance.

Skoch [51] investigated the effect of using steadily injected air-stream and inserting control tubes into the vaneless region of a vane-island diffuser. This covered the influences on the diffuser flow stability and stage performance as shown in Figure 22. The forward-tangent injection by recirculated air produced a small pressure loss in the diffuser with little improvement in the stable flow range. On the other hand, the reverse-tangent injection generated the highest pressure losses with only a small increase in the surge margin at a low injected flow rate. However, the use of control tubes caused a significant improvement in stable flow range with a moderate pressure loss in the diffuser as illustrated in Figure 22.

Conducted researches on volute geometry

The discharge volute is a circular chamber and it used for the last stage or for a single stage compressor to collect the discharge gas from the external boundary of the diffuser and then conveys it to the discharge nozzle. Besides, the circular chamber is provided with a fin at the opposite side of the chamber from the suction nozzle to prevent the continued gas circulation around the volute which is important to avoid gas vortices.

The volute is usually designed using one-dimensional calculations method and the volute flow is assuming to be frictionless and incompressible. The exit radial and tangential velocities are calculated at the beginning and then the velocity inside the volute cross section is derived assuming constant angular momentum and circumferential static pressure. That can be acceptable at design point operation but the velocity of volute flow can't stay constant during the off-design conditions especially if the variation in the inlet mass flow was considered.

Ayder [52] specified three key geometrical parameters which have a major impact on volute design. These parameters are: circumferential variation and shape of the cross sectional area, radial position of the cross section and volute inlet and tongue geometry. Ji et al. [53] considered the connection between the outlet pipe with the volute as another important factor in design process. Braembussche [54] stated that the compactness, efficiency and absence of circumferential pressure distortion are the main targets in any volute design. The circumferential pressure distortion is occurred at off-design operation and it is really importance to be predicted since it relates to the unsteady impeller forces and noise and radial forces of the shaft. The obtained results by Hassan [55] revealed that the increase in the volute area ratio leads to raise the diffuser pressure recovery factor and the maximum flow rate and pressure rise coefficient of the compressor. The optimum area ratio of volute to achieve the maximum stable operating range was specified to be from 0.6 to 0.7. This can be improved further by increasing the clearance between the diffuser vanes and the volute casing.

However, there is still an efficiency loss inside the volute due to un-fully conversion of the kinetic energy associated with the radial component of the discharge gas velocity from the diffuser. The effect of radial flow becomes greater when a vaned diffuser is used since the vanes induce more radial flow. This creates a trade- off between increased recovery of the vaned diffuser and reduced recovery of the volute but today there is a necessary to use vaned diffusers to satisfy performance requirements. One of the main issues which should be addressed in the volute design process is the swirling flow and the non-uniformity of the inlet flow which in turn leads to unsteady interaction between the rotor and stator flows.

Figure 23 illustrates the effect of volute geometry and the inlet location on the static pressure and meridional velocity. There is only one single vortex in the volutes with tangential inlet location while two vortices are observed in the case of symmetrical inlet location. These twin vortices are generated in the small cross section and they are disappeared in the volute with large cross section due to the large diffusion. Considering the static pressure contours, the region of high pressure appears clearly on the small cross section in all tangential and symmetrical inlet locations due to the non-uniformity caused by the volute. However, the increase in the volute cross section should be measured also in term of the increase in overall size and cost. Chehhat et al. [56] concluded that the shape of the volute cross section has more significant impact on the operating range rather than the peak efficiency while the effect of volute inlet location is more substantial on the peak efficiency. The highest efficiency value was observed with tangential inlet and the widest operating range was recorded with circular cross section.

Conclusion

This paper investigated the development trend of component design of centrifugal compressors over the years driven by high efficiency and wide operating flow range. The influential aspects of the design modification on the flow aerodynamic have been discussed to determine the potential effect on the unsteady interaction between the rotor and stator. The change in the efficiency trend demonstrated a possibility of a continuous rise in the efficiency value but that will be in expenses of the stable operating range. Therefore, the design efficiency and flow range have to be optimized based on the particular application.

To improve the imposed work by the impeller, three basic characteristics have been considered: presence of shroud, exit blade angle and dimensional axis. The partial shrouded impellers were found more attractive to compromise between the high efficiency of shrouded impellers and wide operating range of unshrouded configuration. Furthermore, the use of splitter vanes and the selection of blade angle are significantly affected by the required head, surge margin and stage efficiency. However, the extension of the working flow coefficient of 3D-impellers to cover the low flow rates applications will obviously contribute to enhance the operating efficiency.

More efforts have been made to enhance the pressure recovery in the diffuser and these generally focused on five major design parameters which are: diffuser width, radius ratio, chord length, vanes number and inlet vane angle. The effect of each of these variables on the stage efficiency and surge margin has been discussed in order to identify the options for efficiency and flow range improvement. However, each diffuser type has its own impact on the performance of the volute including the losses and pressure. Besides, several methods have been investigated to extend the stage operating range including: diffuser wall treatment, creating porous drillings, using steady flow and pulsed air injection and inserting control tubes.

Moreover, the performed review on the conducted researches on the volute revealed four key geometrical parameters which have a significant influence on volute design including: circumferential variation and shape of the cross sectional area, radial position of the cross section and volute inlet, tongue geometry and connection between the outlet pipe with the volute. However, it was found that there is a trade-off between increased recovery of the vaned diffuser and reduced recovery of the volute. Furthermore, the circumferential pressure distribution at off-design conditions caused by the volute affects the diffuser flow and possibly the flow at the impeller exit.

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