

PASSIVE FLOW SEPARATION CONTROL IN A VANED RADIAL DIFFUSER

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ABSTRACT

Stability and performance are pursued by the aeronautic industry as market constraints become more compelling. The present study focuses on the implementation of a passive flow control system in the vaned radial diffuser of a centrifugal compressor. The flow control consists of holes in both the hub and the shroud, connecting interblade channels with each other. The compressor is operated at a rotational speed lower than the nominal speed. Measurements of performance on the whole machine, and both steady, unsteady pressure measurements in the radial diffuser are performed. The flow control strategy provides an increase in pressure ratio, as well as delaying instability and surge of the compressor. Analysis of pressure maps underlines uniformization of the flow due to the control, possibly stabilizing the flow. Frequential analysis of the pressure reveals the occurrence of low frequencies associated with the control.

NOMENCLATURE

Symbol	Meaning
$\dot{m}$	Mass-flow rate
T	Temperature
P	Pressure
γ	Adiabatic coefficient of air
r	Specific ideal gas constant of air
Ω	Rotational speed of the impeller
Π	Pressure ratio
η	Efficiency
Δh	Specific enthalpy variation
P	Pressure signal
$\sigma(P)$	Time-standard deviation
PSD	Power Spectral Density

$\mathcal{P}_{[f_1 : f_2]}$	Power contained in the $[f_1 ; f_2]$ bandwidth.
Subscripts	
is	Isentropic
std	Standardized quantity
ref	Reference quantity
BPF	Blade passing frequency
$IMP + IGV$	Impeller and Inlet Guide Vanes
LF	Low frequencies
RD	Radial Diffuser
0	Measured downstream of the rig
1	Measured in plane 1 see Fig. 1
2	Measured in plane 2 see Fig. 1
3	Measured in plane 3 see Fig. 1

Superscripts

s	Static
t	Total
$s - s$	Static to static
$t - s$	Total to static

1. INTRODUCTION

Environmental concerns and market demands drive engine manufacturers to enhance the versatility of helicopter turboshafts. Consequently, it is critical to extend the operating range of compressors while maintaining high efficiency.

The limiting factors of compressors operating range at low mass-flow rate, including those of centrifugal type, is the surge instability [1, 2, 3]. Surge consists of large mass-flow rate fluctuation within the system, rendering

the compressor ineffective. Surge occurs because local aerodynamic instabilities increase as the mass-flow rate diminishes and induce global fluctuations eventually triggering deep surge with a global flow reversal.

To extend the operating range of compressors, a better understanding of local instabilities causing surge is necessary. However, instabilities remain poorly understood [4] so this limits the ability to push the surge limit towards lower mass flow rates or to increase the surge margin. The instability mechanism knowledge is also a key element to develop efficient flow control strategy. Delaying the onset of surge and local instabilities to lower mass-flow rates can be achieved through technologies such as casing treatments [6, 7, 8, 9], variable inlet guide vanes (IGVs) [10, 11], and boundary layer suction [12, 13].

In the present study, the use of a passive flow control in the vaned radial diffuser of a centrifugal compressor is investigated. The flow control concept is to permit communication between adjacent interblade channels in the vaned radial diffuser via flow control holes. This passive flow control aims to increase the pressure ratio, efficiency, as well as the span of mass-flow rate at which high pressure ratio and efficiency are available.

Previous studies on the compressor pointed to the existence flow separation in the vaned radial diffuser, with experimental [14, 15] as well as numerical evidence [16]. The occurrence of the pulsating flow has been linked to the development of the flow separation zone in the radial diffuser, ultimately leading to surge. The flow control holes providing passive control have been placed to have the most impact on the separation flow pattern.

2. METHOD

2.1. Test case

The compressor TURBOCEL (Fig. 1) is a research centrifugal compressor designed and built by Safran Helicopter Engines and mounted on the test rig of Ecole Centrale de Lyon, France. The compressor stage comprises inlet guide vanes (IGV), an impeller (IMP), a vaned radial diffuser (RD) as well as an axial diffuser also known as outlet guide vanes (OGV). The impeller is driven by a 1MW electrical engine. In the present study, the compressor is operated at a constant rotational speed, lower than nominal speed. The radial diffuser includes splitter blades and has been equipped with flow control holes for flow control purposes.

More specifically, the flow control apparatus consists of flow control holes in each main inter-blade channel of the radial diffuser. Flow control holes are located near the suction side of the blades, on both hub and shroud (Fig. 2). The flow control holes have been placed near average critical points [10, 17] of the separation pattern to have

the most impact on the separation flow pattern. On the shroud, the holes are linked to a cavity which can be either closed or plugged to a tube, leading to a collector. On the hub, holes lead to an annular collector built in the machine, which is then linked to the collector outside of the machine.

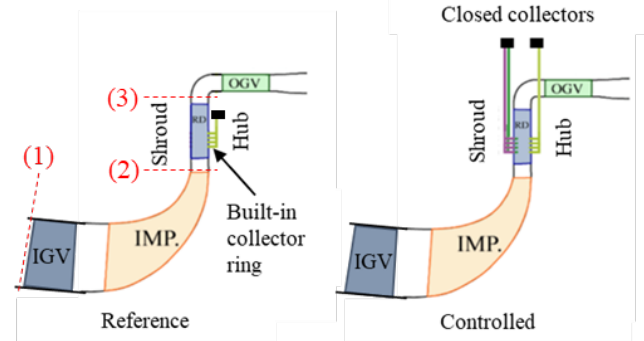


Figure 1. Meridional views of the compressor TURBOCEL, in the reference (left) and controlled case (right).

In the reference configuration (Fig. 1 left), the cavities on the shroud are all closed with screw-in caps, and the annular collector on the hub doesn't communicate with the outside. In other words, the inter-blade channels do not communicate with one another via the shroud. They do communicate via the hub because of the annular collector. No mass-flow rate is taken out of the machine through the flow control holes. In the passive controlled configuration (Fig. 1 right), collectors are connected to the cavities in the shroud, and the annular collector on the hub. Therefore, inter-blade channels communicate both on hub and shroud. No mass-flow rate is taken out of the machine through the flow control holes. The main difference between the two configurations is that in the passive controlled configuration, the inter-blade channels are communicating via the shroud.

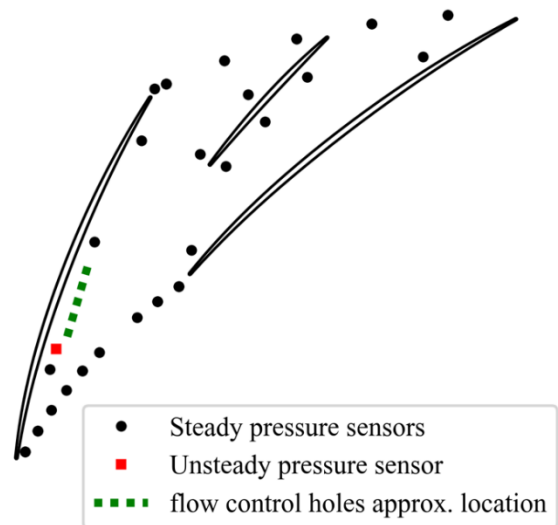


Figure 2. Hub and shroud static pressure sensors in the vaned radial diffuser.

The mass-flow rate is measured by a flowmeter upstream of the rig. The raw, measured value of the mass-flow rate $\dot{m}$ is converted to a standard value $\dot{m}_{std}$:

$$\dot{m}_{std} = \frac{\sqrt{rT_{t,0}/\gamma}}{\sqrt{r_{ref}T_{ref}/\gamma_{ref}}} \frac{P_{ref}}{P_{t,1}} \dot{m} \quad (1)$$

which takes into account humidity and temperature correction for reproducibility purposes. Relative uncertainty on $\dot{m}_{std}$ is presented in Tab. 1.

Rotational speed Ω is measured at the shaft, and is adjusted to correspond to a standard rotational speed via the relation:

$$\Omega = \frac{\sqrt{\gamma r T_{t,0}}}{\sqrt{\gamma_{ref} r_{ref} T_{ref}}} \Omega_{std} \quad (2).$$

2.2. Performance calculation

To compute the performance of the machine, steady pressure measurements are performed between the different components of the machine. Sensors are located between each component of the machine, so as to be able to measure pressure change induced by each component (Fig. 1).

The pressure ratios involved in this study are defined by the following relations:

$$\Pi^{t-s} = \frac{P_{s,3}}{P_{t,1}} \quad (3.1)$$

$$\Pi_{IMP+IGV}^{t-s} = \frac{P_{s,2}}{P_{t,1}} \quad (3.2)$$

$$\Pi_{RD}^{s-s} = \frac{P_{s,3}}{P_{s,2}} \quad (3.3).$$

From Eqs. 3.1, 3.2 and 3.3, it is deduced that:

$$\Pi^{t-s} = \Pi_{IGV+IMP}^{t-s} \cdot \Pi_{RD}^{s-s} \quad (4).$$

The compressor isentropic efficiency is defined as:

$$\eta_{is} = \frac{\Delta h_{is}}{\Delta h} = T_{t,0} \cdot \frac{\Pi^{t-s \frac{\gamma-1}{\gamma}} - 1}{T_{t,3} - T_{t,0}} \quad (5).$$

Table 1. Relative uncertainty for performance metrics, at best efficiency, nominal speed.

Performance metric	Relative uncertainty (%)
Π	± 0.06
η_{is}	± 0.38
$\dot{m}_{std}$	± 0.5

2.3. Unsteady Pressure Measurements

To better understand flow dynamics in the radial diffuser, unsteady pressure sensors are used in this study. The study presents the measurements of one sensor, located at the nearest of the flow control holes on the shroud (Fig. 2). The measurements of these sensors have been cross-checked with the other 28 unsteady pressure sensors in the radial diffuser.

The unsteady pressure sensors used are Kulite XTEH-7L series which yield a relative total uncertainty of $\pm 0.85\%$. The sampling frequency is $300kHz$. The measurement data are re-sampled via the one-per-revolution trigger, meaning that the sampling of the signal is adjusted to the variations of the rotational speed of the machine, yielding an evenly spaced, and consistent number of samples per revolution of the machine.

From the time-sampled signal, the standard deviation $\sigma(P)$ of a given signal P can be computed. Via Fourier filtering, it is possible to filter out the high frequency fluctuations of P , in order to obtain a low-frequency pressure signal, noted P_{LF} . In the study, the cut-off frequency is set at $100Hz$, to separate the high frequency fluctuations induced by the rotational speed or the blade passing frequency (both have a frequency at several hundred Hertz) versus low frequency. Computing the $\sigma(P_{LF})$ will therefore yield the fluctuations associated with the low frequency components of the pressure signal.

2.4. Aerodynamic stability of a compressor

The compressor characteristic curve is the overall pressure ratio as a function of the mass-flow rate, at a constant rotational speed. A quasi-stationary approach to the stability of the compressors consists in studying the slope of the characteristic curve of the compressor [1]. If the slope is positive, then, a perturbation reducing the mass-flow rate will induce a decrease in the overall pressure ratio. The compressor output pressure is reduced while the pressure in the downstream tank has not yet decreased, leading to a further decrease in the mass-flow rate. This will induce another decrease in pressure ratio. The behaviour of the compressor is then qualified of unstable. To the contrary, if the characteristic slope is negative, then, a fluctuation in mass-flow rate will induce an increase in the pressure ratio, compensating for the mass-flow rate fluctuation, the compressor behaviour is stable.

3. RESULTS

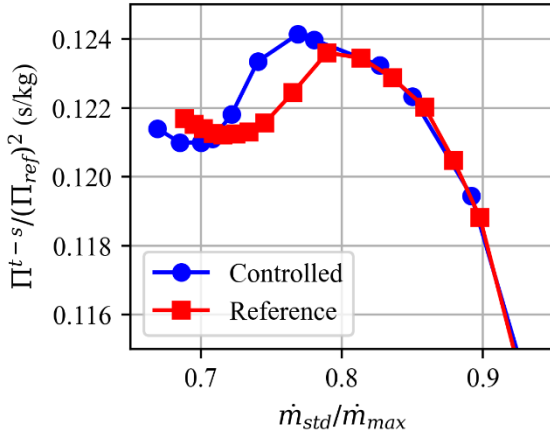


Figure 3. Compressor pressure ratio as a function of the standard mass-flow rate, for both the controlled (blue) and reference case (red).

Fig. 3 presents the pressure ratio of the compressor as a function of the mass-flow rate for a constant rotational speed. The rotational speed is lower than the nominal rotational speed. The reference and the controlled case are presented. In each curve, the operating point with the lowest mass-flow rate corresponds to the last point at which a stable measurement can be performed. Below this mass-flow rate, the compressor enters in surge, making it impossible to obtain a stabilized measurement. The strong flow fluctuations force the compressor to shut down. The standard mass-flow rate is normalised by the standard mass-flow rate at choke conditions. The mass-flow rate at choke conditions used for the normalization are adapted for each case to simplify the discussion. The two choke mass-flow rates are slightly different but the difference is lower than the incertitude level. On the reference pressure ratio curve, one can notice three regions: the decreasing region to the left at low mass-flow rate ($\dot{m}_{std}/\dot{m}_{max} < 0.73$), an increasing portion from $\dot{m}_{std}/\dot{m}_{max} = 0.73$ to $\dot{m}_{std}/\dot{m}_{max} = 0.79$; and a decreasing region with a mass-flow rate higher than $\dot{m}_{std}/\dot{m}_{max} = 0.79$. At $\dot{m}_{std}/\dot{m}_{ref} = 0.79$, the maximum pressure ratio is reached.

One noticeable feature of this characteristic curve is that it displays an unstable portion, before becoming stable again. This means that even if according to a 1D stability criterion the compressor is supposed to be unstable, the compressor keeps operating. This can be the case with centrifugal compressors, where, even if the compressor is unstable, it is still able to operate [2, 3, 7]. At low mass-flow rates, the compressor has a negative slope characteristic again, which seems to indicate that it is stable. However, the compressor goes into surge even if it appears stable in terms of characteristic.

For the controlled case, the trend is comparable, but the minimum flow rate and the point with the maximum pressure ratio are reached at lower mass-flow rate than the reference case. In the controlled case, the maximum pressure ratio is higher than the reference case. Therefore, it seems that the reference case becomes unstable at a lower mass-flow rate compared to the controlled case.

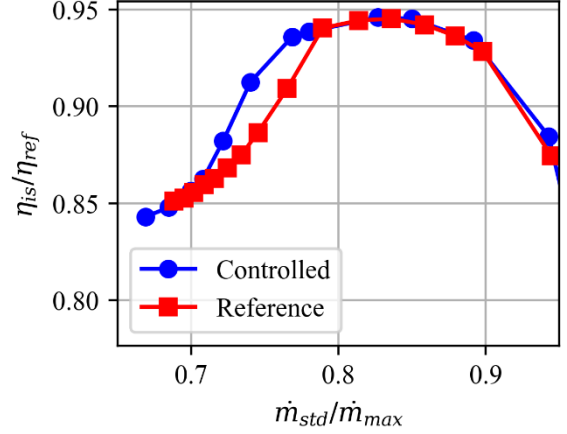


Figure 4. Compressor isentropic efficiency as a function of the standard mass-flow rate, for both the controlled (blue) and reference case (red).

Fig. 4 presents the isentropic efficiency of the compressor stage, as a function of the mass-flow rate, for both the reference and the controlled case. On the isentropic efficiency curve, in the reference case, the maximum efficiency ratio is reached at $\dot{m}_{std}/\dot{m}_{max} = 0.83$. In the increasing section of the curve (low mass-flow rate), there is a break in the slope near the peak efficiency at a mass-flow rate of 0.79. The break in slope corresponds to a loss of efficiency, correlated with the instability of the compressor when the mass-flow rate decreases. In the controlled case, the maximum value is the same as the reference case. The trend of the curve is comparable but, the break in the increasing slope occurs at $\dot{m}_{std}/\dot{m}_{max} = 0.77$.

At this point, it seems that the flow control performs as expected: the surge mass-flow rate is lower in the controlled case than in the reference case. Also, the peak pressure ratio is increased by the flow control, and peak efficiency is not decreased by the flow control. To understand better the effect of control, the stability of the different components is compared. The pressure ratio of both the radial diffuser and of the impeller are plotted respectively, Fig. 5 and Fig. 6. Comparing the whole stage pressure ratio to the pressure ratio of the radial diffuser, both curves appear very similar. Both the radial diffuser and the whole stage display the same behaviour in terms of monotony. However, it appears that the whole compressor pressure ratio is following the trend of the

radial diffuser. This is true for the reference as well as the controlled case. The impeller pressure ratio, on the other hand, appears to behave very differently from the full-stage pressure ratio. The impeller is stable on most operating points, except from $\dot{m}_{std}/\dot{m}_{max} = 0.73$ to $\dot{m}_{std}/\dot{m}_{max} = 0.82$ in the controlled case, and from $\dot{m}_{std}/\dot{m}_{max} = 0.79$ to $\dot{m}_{std}/\dot{m}_{max} = 0.81$ in the reference case.

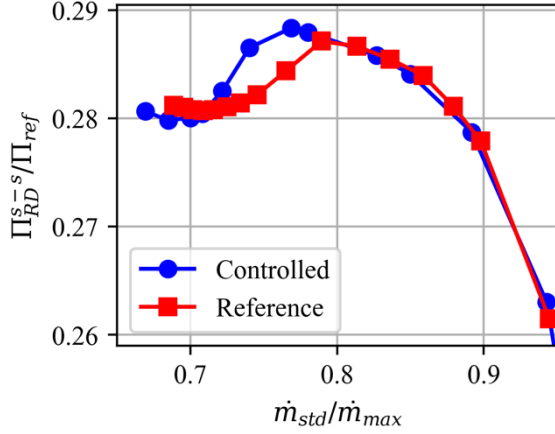


Figure 5. Radial diffuser pressure ratio as a function of the standard mass-flow rate. For both the controlled (blue) and reference case (red).

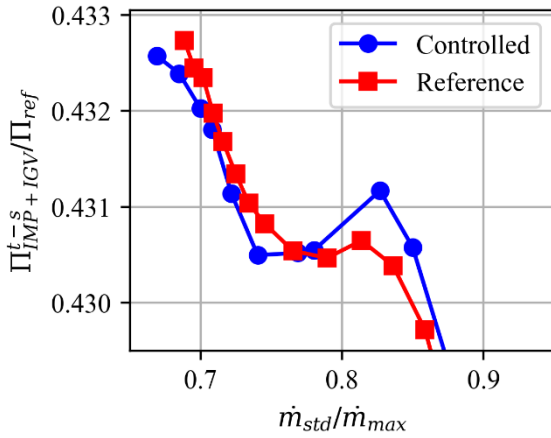


Figure 6. Impeller pressure ratio as a function of the standard mass-flow rate. For both the controlled (blue) and the reference case (red).

The impeller never actually becomes unstable and is always stable when the whole compressor is unstable. As mentioned before, it also appears that the compressor goes into surge while being stable, with both the radial diffuser and the impeller being stable as well. Therefore, the 1-dimensional stability criterion is not valid to this case. Since the overall pressure ratio is highly correlated to the radial diffuser pressure ratio, the focus is to be put

on the radial diffuser to understand the impact of control on the stability of the machine.

A first pathway to explaining why the radial diffuser is more stable in the presence of control is that the communication between the interblade channels may distribute the pressure better between the different channels. The pressure signals interpolated in the radial diffuser are presented in Fig. 7, at a relative mass-flow rate of 0.76, where a large gap in the pressure ratio exists between controlled and reference cases. Making the hypothesis that the flow pattern is uniform along the hub to shroud direction, pressures sensors located both at the hub and the shroud are interpolated together.

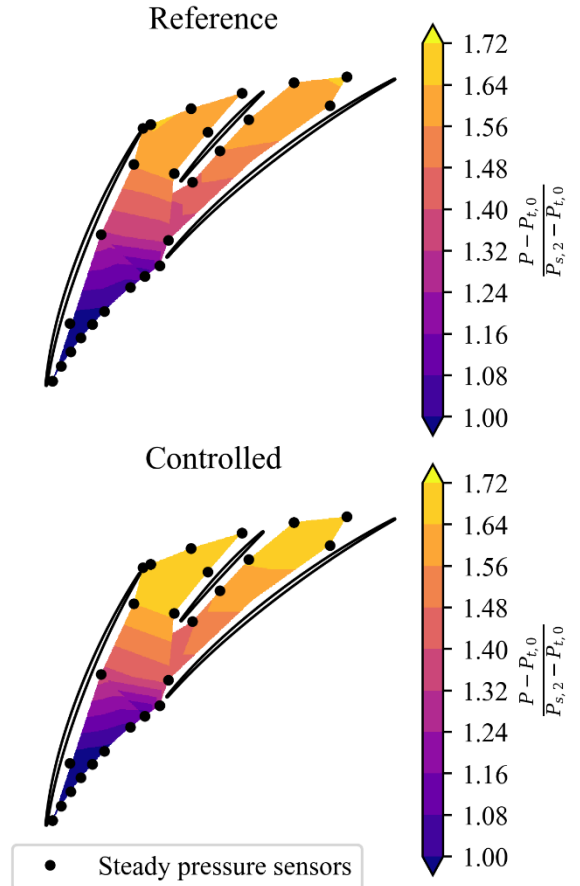


Figure 7. Interpolated pressure distributions in the radial diffuser, built with measurements from the steady pressure sensors. Sensors of both hub and shroud are interpolated together. At $\dot{m}_{std}/\dot{m}_{max} = 0.76$.

From the pressure distribution, in both the controlled and the reference case, it appears that the spatial organisation is mainly driven by the adverse pressure gradient. As presented with the radial diffuser pressure ratio, the pressure gradient is slightly more important in the controlled case. At the outlet, in both cases, the pressure distribution appears uniform along the azimuthal direction.

To explore the flow uniformity in the hub to the shroud direction, the pressure signal on the hub and the shroud, at the outlet of the radial diffuser, at $\dot{m}_{std}/\dot{m}_{max} = 0.76$, are presented on Fig. 8. With a better radial diffuser recuperation in controlled case, the pressure is higher at the outlet in this case. In addition, the flow seems globally more uniform between the hub and the shroud in the controlled case. In particular on each side of the splitter ($\theta = 0.2$ and at $\theta = 0.6$). Calculating the area between the pressure profiles of each case, the evolution from the reference to the controlled case reveals a reduction in area of around 5%.

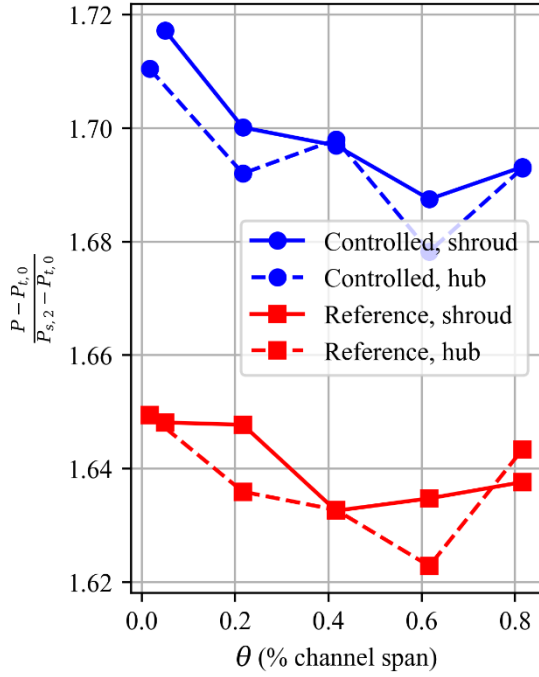


Figure 8. Pressure signal at the outlet of the radial diffuser, on the hub and the shroud, at $\dot{m}_{std}/\dot{m}_{max} = 0.76$.

Fig. 9 is the same plot, but with $\dot{m}_{std}/\dot{m}_{max} = 0.72$ (closer to stall). The description is the same as Fig. 8 but for each case, the hub to shroud disparity is more pronounced than at $\dot{m}_{std}/\dot{m}_{max} = 0.76$. It seems that the hub to shroud disparity is also more prevalent in the reference case. Calculating the area between the pressure profiles of each case, the evolution from the reference to the controlled case reveals a reduction in area of around 5% as well.

Previous numerical work, on an operating point close to the one of Fig. 9, had identified stall pattern developing on the hub, around the splitter blade, coherent with the previous observations on pressure [10]. To summarize, it appears that the presence of passive control induces a better uniformity between hub and shroud. That is interpreted as a reduction of the flow separation zone at

similar relative mass-flow rates, hence stabilizing the machine.

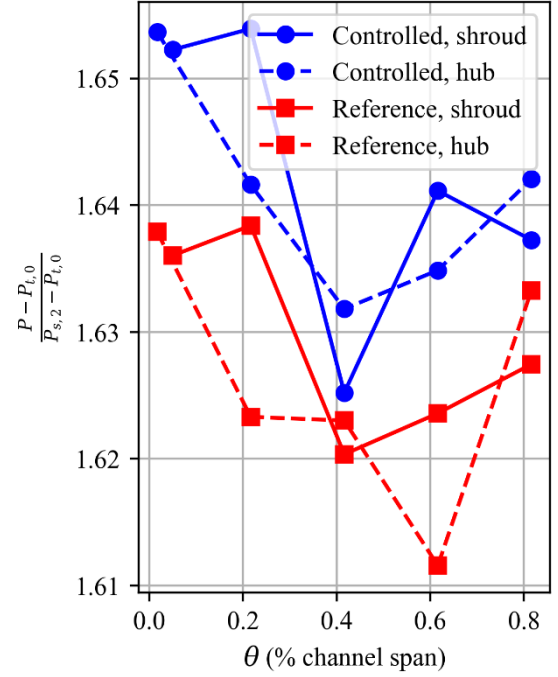


Figure 9. Pressure signal at the outlet of the radial diffuser, on the hub and the shroud, at $\dot{m}_{std}/\dot{m}_{max} = 0.72$.

The standard deviation of the signal accounts for all of the fluctuations recorded by the unsteady pressure sensor. The standard deviation of the pressure signal is therefore used as an indicator of the unsteadiness of the flow. If the pressure signal is filtered so that it only takes into account the low frequencies then, the standard deviation of the pressure signal is an indicator of the low frequency fluctuations. On Fig. 10, the ratio of the time-standard deviation of the low-frequency pressure signal and of the time-standard deviation of the whole pressure signal, of a sensor near the flow control holes on the shroud of the radial diffuser is plotted against the relative mass-flow rate.

The low frequency fluctuations in the radial diffuser increase suddenly when the mass-flow rate decreases. When the mass-flow rate decreases again, the proportion of low frequency is nearly constant around half of the total standard deviation. It has been verified that the trend in the fluctuations shown on the upcoming figures is comparable to the trend of all the unsteady sensors of the radial diffuser, meaning that the low frequency unsteadiness of the flow is dominant as the mass-flow rate decreases, in all the radial diffuser. In the reference case, the sudden increase in pressure fluctuation is at $\dot{m}_{std}/\dot{m}_{max} = 0.79$ and 0.75 for the controlled case. In both cases, this mass-flow rate correlates with the change in monotony on the pressure ratio (see Fig. 3 and 5).

Therefore, the loss in the pressure ratio, and the decrease in isentropic efficiency seems to be linked to the occurrence of low frequency content. Since the critical mass-flow rate occurs at a lower mass-flow rate in the controlled case, it seems that the delay in the emergence of the low-frequency unsteadiness of the flow, delays the onset of surge in the machine. This delay in low-frequency unsteadiness correlates with the presence of control.

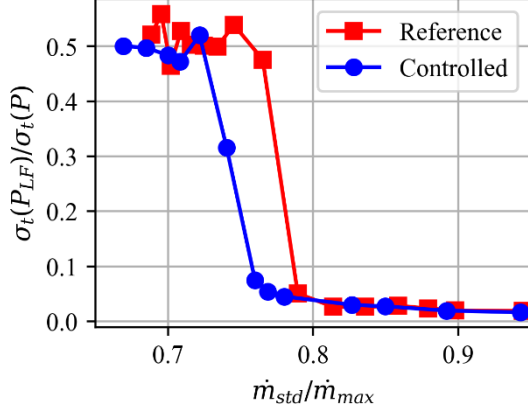


Figure 10. Ratio of low frequency fluctuations and overall fluctuations for a sensor close to the flow control holes on the shroud, as a function of the normalised standard mass-flow rate. For both the controlled (blue) and reference case (red).

Previous studies on the compressor had pointed to the existence of a flow separation in the vaned radial diffuser near surge. The flow separation, pulsating at a low frequency, locks with the Helmholtz frequency of the test rig at 11Hz [14]. The power spectral density (PSD) of the pressure signal of one sensor near the flow control holes on the shroud of the radial diffuser is presented in Fig. 11. The operating point selected is the one with the lowest mass flow rate in the reference case ($\dot{m}_{std}/\dot{m}_{max} = 0.68$). The flow separation frequency, close to 11Hz is clearly noticeable with the highest power amplitude within the 2.5Hz to 50Hz band. The amplitude of this phenomenon is comparable to the amplitude of the blade-passing frequency, which is the predominant periodic frequency in the radial diffuser inlet. The amplitude is lower for the reference than the controlled case, but the power is more dispersed around 11Hz in the reference case. Calculating the cumulative power contained in the 8Hz to 13.5Hz bandwidth at $\dot{m}_{std}/\dot{m}_{max} = 0.68$, it appears that the controlled case has around 33% less power in this bandwidth (also visible in Fig. 12). Despite what Fig. 11 might suggest, the power associated with the locking of the flow separation dynamics and the Helmholtz frequency is much greater in the reference case. The frequency dispersion of the locking occurs such that the Helmholtz frequency is well defined (narrow peak), but the flow separation dynamics has a broader

frequency signature. The flow separation frequency depends on its size and shape. When the flow separation locks with the Helmholtz frequency, the flow separation frequencies concentrate around 11Hz with a variable dispersion depending on the properties of the separation.

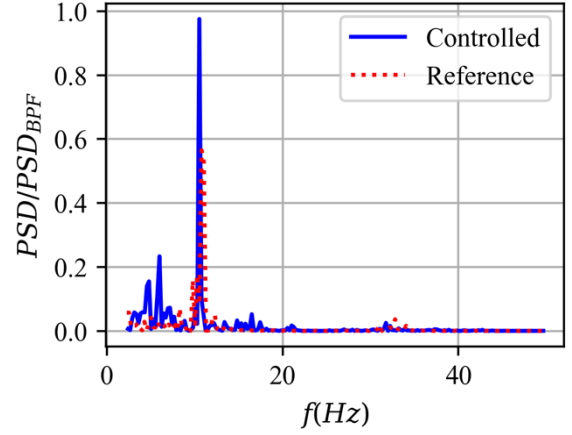


Figure 11. Power spectral density of the pressure signal for a sensor close to the flow control holes on the shroud, at an operating point located near surge at $\dot{m}_{std}/\dot{m}_{max} = 0.68$.

The other difference between the two spectres is that activity appear in the 2.5Hz to 8Hz band. This low frequency activity has been identified on other sensors in the radial diffuser but is highest near the flow control holes. This low frequency phenomenon is occurring on low mass-flow rate operating points for the controlled case only. To track the activity attributed to the flow control holes on the shroud of the radial diffuser, the cumulative power contained in the 2.5Hz to 8Hz bandwidth and in the 8Hz to 13.5Hz bandwidth are plotted against the relative mass-flow rate and presented on Fig. 12. They are plotted as a cumulated area under the curve so that the top line with bullets corresponds to the power contained in the 2.5Hz to 13.5Hz band. The 2.5Hz to 8Hz band is associated with the very low frequency below the Helmholtz frequency. For the whole 2.5Hz to 13.5Hz bandwidth, it appears that the power in this band grows as the mass-flow rate decreases. The growth of this quantity correlates with the growth of the “global fluctuations” (Fig. 10) in both the reference and the controlled case.

As for the 2.5Hz to 8Hz band, it appears that in the reference case, only close to 25% of the power is below 8Hz for operating points close to stall. In the controlled case, 50% of the power is distributed below 8Hz. These frequencies have also been observed in the reference case, on the hub (where channels communicate). Since these frequencies appear when the channels communicate, it is likely that these frequencies, below 8Hz, are produced by the communication between channels.

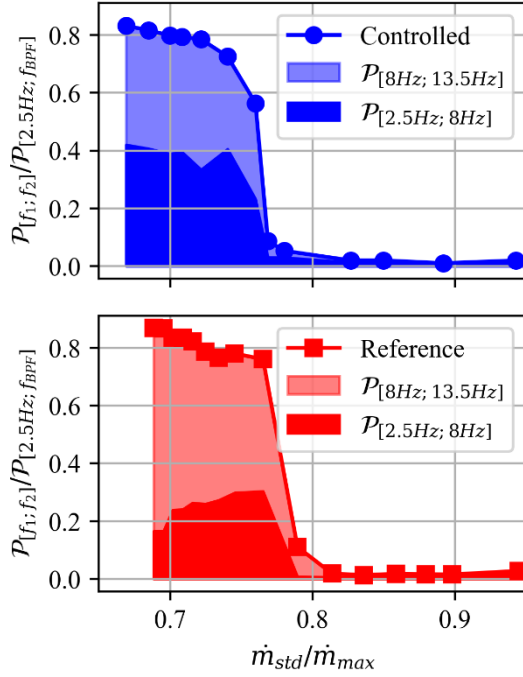


Figure 12. Colours represents power in $[2.5\text{Hz}; 8\text{Hz}]$ and $[8\text{Hz}; 13.5\text{Hz}]$. Bullets correspond to power in $[2.5\text{Hz}; 13.5\text{Hz}]$. As a function of the normalised mass-flow rate. For a sensor close to the flow control holes on the shroud.

The hypothesis to explain the mechanism that induce these low frequency contents is the dynamics of the uniformization process between the different interblade channels. In each interblade channel, the flow separation pulsates at 11Hz with probably some azimuthal disparity. The communication between channels tries to uniformize these disparities and induces the very low frequency content near the flow control holes.

4. CONCLUSION

In this paper, the effect of passive flow control applied to interblade channels in a vaned radial diffuser of a centrifugal compressor is experimentally investigated. The compressor is operated at a rotational speed lower than the nominal speed. Measurements of performance on the whole machine, and both steady and unsteady pressure measurements in the radial diffuser are performed.

The passive control technique in the radial diffuser seems to delay surge and enables operation at lower mass-flow rates compared to the reference case. The maximum isentropic efficiency of the whole stage is preserved and the maximum pressure ratio is improved by the passive flow control.

To explain this positive impact, the analysis of the pressure in the radial diffuser shows that the pressure is more homogeneous between hub and shroud in the controlled case. In the controlled case, the interblade channels are connected to each other in both the hub and the shroud (versus hub only in reference case), that induces the same flow separation control in both sides.

Furthermore, the unsteady experimental analysis points to the occurrence of the previously documented low-frequency pulsating separation. The controlled case displays additional very-low frequency activity as compared to the reference, with more power distributed in 2.5Hz to 8Hz bandwidth. These unsteady activities are associated with the uniformization process in the radial diffuser.

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5. REFERENCES

- [1] E. M. Greitzer, « The Stability of Pumping Systems—The 1980 Freeman Scholar Lecture », *J. Fluids Eng.*, vol. 103, n° 2, p. 193-242, juin 1981, doi: 10.1115/1.3241725.
- [2] D. A. Fink, N. A. Cumpsty, et E. M. Greitzer, « Surge Dynamics in a Free-Spool Centrifugal Compressor System », *J. Turbomach.*, vol. 114, n° 2, p. 321-332, avr. 1992, doi: 10.1115/1.2929146.
- [3] R. Hunziker et G. Gyarmathy, « The Operational Stability of a Centrifugal Compressor and Its Dependence on the Characteristics of the Subcomponents », *J. Turbomach.*, vol. 116, n° 2, p. 250-259, avr. 1994, doi: 10.1115/1.2928359.
- [4] I. J. Day, « Stall, Surge, and 75 Years of Research », *J. Turbomach.*, vol. 138, n° 011001, oct. 2015, doi: 10.1115/1.4031473.
- [5] A. Young, I. Day, et G. Pullan, « Stall Warning by Blade Pressure Signature Analysis », *J. Turbomach.*, vol. 135, n° 011033, oct. 2012, doi: 10.1115/1.4006426.
- [6] S. Sharma, A. Broatch, J. García-Tíscar, J. M. Allport, et A. K. Nickson, « Acoustic characterisation of a small high-speed centrifugal compressor with casing treatment: An experimental study », *Aerosp. Sci. Technol.*, vol. 95, p. 105518, déc. 2019, doi: 10.1016/j.ast.2019.105518.

- [7] H. Chen et V.-M. Lei, « Casing Treatment and Inlet Swirl of Centrifugal Compressors », *J. Turbomach.*, vol. 135, n° 041010, juin 2013, doi: 10.1115/1.4007739.
- [8] L. Ding, T. Wang, B. Yang, W. Xu, et C. Gu, « Experimental investigation of the casing treatment effects on steady and transient characteristics in an industrial centrifugal compressor », *Exp. Therm. Fluid Sci.*, vol. 45, p. 136-145, févr. 2013, doi: 10.1016/j.expthermflusci.2012.10.016.
- [9] X. He et X. Zheng, « Roles and mechanisms of casing treatment on different scales of flow instability in high pressure ratio centrifugal compressors », *Aerosp. Sci. Technol.*, vol. 84, p. 734-746, janv. 2019, doi: 10.1016/j.ast.2018.10.015.
- [10] N. Poujol, M. Buisson, P. Duquesne, et I. Trébinjac, « Unsteady Analysis of a Pulsating Alternate Flow Pattern in a Radial Vaned Diffuser », *Int. J. Turbomach. Propuls. Power*, vol. 7, juill. 2022, doi: 10.3390/ijtp7030023.
- [11] N. Poujol, I. Trébinjac, et P. Duquesne, « Effects of Inlet Guide Vanes on the Performance and Stability of an Aeronautical Centrifugal Compressor », *J. Turbomach.*, vol. 143, n° 10, mai 2021, doi: 10.1115/1.4050944.
- [12] A. Marsan, I. Trébinjac, S. Coste, et G. Leroy, « Study and Control of a Radial Vaned Diffuser Stall », *Int. J. Rotating Mach.*, vol. 2012, n° 1, p. 549048, 2012, doi: 10.1155/2012/549048.
- [13] A. Marsan, I. Trébinjac, S. Moreau, et S. Coste, « Active Flow Control in a Radial Vaned Diffuser for Surge Margin Improvement: A Multislot Suction Strategy », *Int. J. Rotating Mach.*, vol. 2017, n° 1, p. 4120862, 2017, doi: 10.1155/2017/4120862.
- [14] V. Moënné-Loccoz, I. Trébinjac, N. Poujol, et P. Duquesne, « Low frequency stall modes of a radial vaned diffuser flow », *Mech. Ind.*, vol. 20, n° 8, Art. n° 8, 2019, doi: 10.1051/meca/2020026.
- [15] V. Moënné-Loccoz, I. Trébinjac, E. Benichou, S. Goguet, B. Paoletti, et P. Laucher, « An experimental description of the flow in a centrifugal compressor from alternate stall to surge », *J. Therm. Sci.*, vol. 26, n° 4, p. 289-296, août 2017, doi: 10.1007/s11630-017-0941-8.
- [16] E. Benichou et I. Trébinjac, « Numerical Analysis of an Alternate Stall in a Radial Vaned Diffuser », in *GT2016*, Volume 2D: Turbomachinery, juin 2016. doi: 10.1115/GT2016-56485.
- [17] P. Duquesne, J. Chanéac, G. Mondin, et J. Dombard, « Topology Rule-Based Methodology for Flow Separation Analysis in Turbomachinery », *Int. J. Turbomach. Propuls. Power*, vol. 7, n° 3, Art. n° 3, sept. 2022, doi: 10.3390/ijtp7030021.