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INFLUENCE OF ROTATING STALL AND SURGE IN THE DESIGN OF A SMALL GAS TURBINE ENGINE WITH AXIAL FLOW COMPRESSOR

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Abstract. *Instabilities during operation of axial flow compressors, such as rotating stall and surge, are phenomena that reduce significantly the gas turbine performance. Mathematical and physical models that predict the transient flow in the compression system during these instabilities were developed in the past and have been improved presently. This article utilizes the nonlinear model developed by Moore and Greitzer to determine the mode of instability in compression systems. Mode and intensity of compressor instability are strongly influenced by the compressor rotational speed and the volume of the combustion chamber (plenum) because they are directly linked to the Greitzer non-dimensional B-parameter. For a given compression system, there is a critical value of B that determines the existence of rotating stall or surge. For larger values of B critical there will be surge and, conversely, for smaller values there will be rotating stall. The critical value of B and the volume of the combustion chamber are calculated as a function of the engine speeds for a 5-stage axial flow compressor. The compressor behavior, namely axial flow coefficient, pressure-rise coefficient and surge frequency, is demonstrated for chosen speeds.*

Axial Compressor, Rotating Stall, Surge, Gas Turbines

1. Introduction

At normal operating conditions the flow in axial-flow compressors is stable and nearly axisymmetric. It is subjected to aerodynamic instabilities that significantly reduce the compressor performance and, as a result, the gas turbine performance. These instabilities may give rise to small reduction in efficiency and/or increase of the noise. In more severe cases, these instabilities may cause unacceptable operating conditions. It is investigated, at early design stage of a gas turbine engine, the influence of the combustion chamber volume on some types of instabilities that may lead a 5-stage axial flow compressor, designed for such engine, to unacceptable operating condition, namely rotating stall and surge[1]. It is known that it may be easier to recuperate the stable operation from surge than from rotating stall, the latter requiring almost always engine shut-down.

This paper describes the rotating stall and surge phenomena and applies a model developed by Moore and Greitzer, 1986 for the study of these instabilities at early design stage of a gas turbine, which are influenced by the volume of the combustion chamber and compressor rotational speed.

2. Compressor Performance Maps

Multi-stage axial-flow compressor performance map, or compressor map, usually shows constant corrected speed lines ($N/\sqrt{T_{t1}}$) relating both pressure ratio (P_{t2}/P_{t1}) and isentropic efficiency (η) as functions of corrected mass flow ($\dot{m}\sqrt{T_{t1}}/P_{t1}$), where N, T_{t1}, P_{t1}, $\dot{m}$, P_{t2} and η stand for, respectively, shaft speed, compressor inlet total temperature and pressure, mass flow, compressor outlet total pressure and efficiency. An example of a compressor map is shown in Fig. (1). The instability line indicates the limit up to the compressor operating condition is stable[2]. Above that curve flow distortions may arise circumferentially, causing a non-uniform axial velocity disturbance - the rotating stall - or large oscillations of the axial flow - the surge. Both can be caused by reduction of the mass flow across the compressor due to variation in the shaft speed and by distortions of the flow entering the compressor (inlet distortion).

[1]The term instability is for characterizing either rotating stall or surge
[2]Betchov and Pearson, 1967 apud Greitzer, 1980, define stability as the system ability to overcome small perturbations

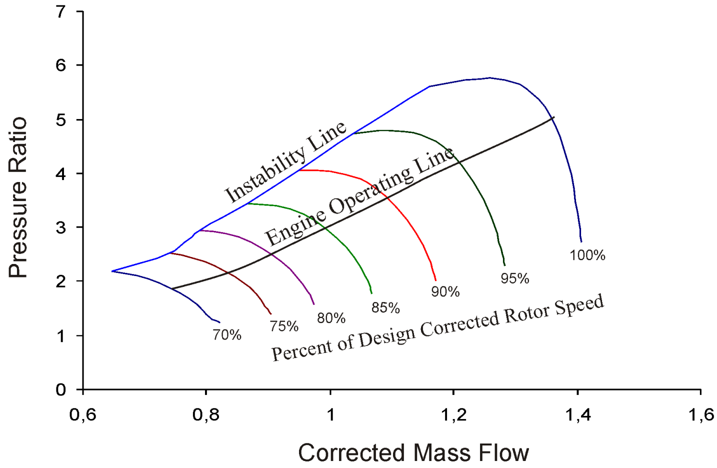


Figure 1: Example of a compressor map

3. Rotating Stall Summary

Stall is the flow separation at a single or at several rotor blades, namely stall cell. A compressor cross-section may have one or several stall cells. These cells move around in the same direction of the shaft rotation, from which the name rotating stall is originated.

Although the rotating stall causes a non-uniform axial velocity disturbance circumferentially, the average main flow remains unaltered. This non-uniform flow disturbance in a compressor of a gas turbine engine may cause excessive vibration of the blades, decrease of pressure and efficiency. A decrease of up to 20% in compressor isentropic efficiency has been observed. Reduction in mass flow may cause increase of turbine entry temperature above the accepted limits (Greitzer, 1980).

Emmons *et al.*, 1955, explain the physics of the stall, which may be expressed as the following: consider a compressor rotor cascade whose flow enters at an incidence angle (Fig. (2)), suppose that a perturbation in the flow, like a reduction of the mass-flow, causes increase of the incidence angle, making blade B entering stall. At this condition, flow separates from the blade suction surface, yielding a flow blockage or delay in the channel between blades B and C. Such a blockage moves the flow tangentially, causing increase in the incidence angle on blade C and reduction in the incidence angle on blade A, as seen in Fig. (2). Therefore, blade C is driven towards stall while stall is inhibited in blade A due to the decrease of the incidence. The stall region, or cell, may extend to several blade passages.

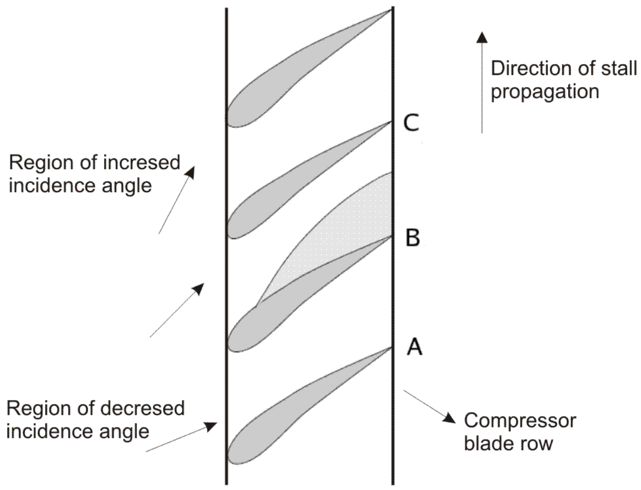


Figure 2: Stall propagation in a compressor blade row

According to Cumpsty, 1989, one may find several stall cells extending all over the blade height (*full-span stall*), or stall cells that cover part of the blade (*part-span stall*), as shown in Fig. (3). The amplitude of the stall cells gives rise to two types of rotating stall: abrupt and progressive. The abrupt stall originates from full-span stall cells. At the beginning abrupt stall causes sudden flow and pressure fall. After it has developed, it stays practically unchanged (Fig. (3a)). While the progressive stall is characterized by a gradual pressure decrease. It happens, for instance, during engine start and aircraft take-off, i.e., when the mass flow is low and the incidence is high, it requires the on-set of part-span stall cells (Fig. (3b)).

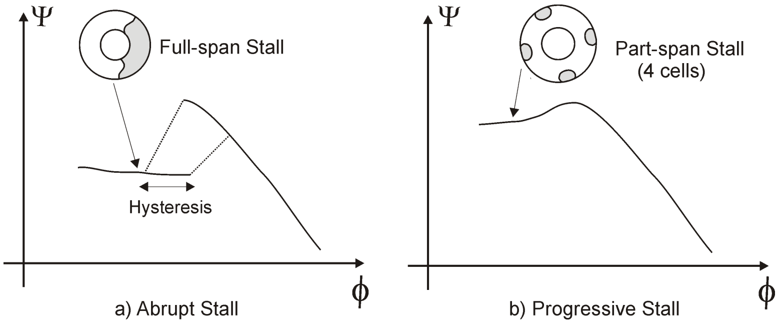


Figure 3: Stall cells types and rotating stall

4. Surge Summary

In this case both compressor average mass flow and pressure vary with time, causing the "operating point" to oscillate due to a process of cause and effect, between the stable and instable region of the compressor map. This phenomenon is known as surge. It may be so violent that the mass flow can be reverted and the hot gases from the combustion chamber emerge from the compressor intake (deep surge), as indicated in Fig. (4a). In the other hand it may be so gentle that the operating point stays close to the maximum operating pressure (classical surge), as indicated in Fig. (4b). This behavior is typical of small gas turbine compressors, prior to the settlement of more severe surge, causing reverse flow (Cumpsty, 1989).

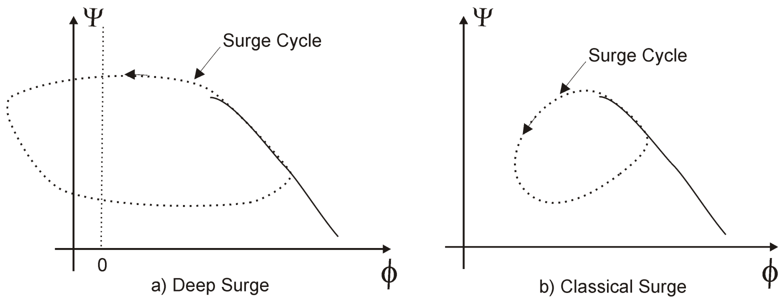


Figure 4: Deep and classical surge

The surge time scale is greater than the one for rotating stall. The frequency of the surge cycle is obtained from the number of times the "reservoir" fills up and empties. Although the surge process may be effectively axisymmetric when fully developed, in the initial transient it is not. Effectively, one of the waste surge effects in high compression ratio axial compressors is the large axial load that the rotor and casing must overcome (Cumpsty, 1989; Walsh and Fletcher, 1998).

5. Fluid Dynamic Model

Greitzer, 1976, proposed a 1-D non-linear model to predict transient responses of a compression system, when forced to operate beyond its stable limit. With this model, Greitzer defines a non-dimensional parameter B, which for a specified compressor characteristic curve, exhibits a critical value that correlates with the onset of rotating stall or surge. For larger values of B_{crit} the system would surge, while for lower values rotating stall would be present. B is defined by Eq. (1), where U is the rotor tangential velocity at blade mid height, a_s is the speed of sound, V_p is the volume of the reservoir following the compressor exit, A_C is the area of the compressor duct and L_C is the compressor length.

$$B = \frac{U}{2a_s}\left(\frac{V_P}{A_C L_C}\right)^{\frac{1}{2}} \tag{1}$$

Moore, 1984a, models the abrupt stall, developing a semi-empirical expression for the calculation of the velocity of propagation of the stall cell for small perturbation amplitudes. It is expressed as a fraction (f) of the rotor speed (U) at the blade mid height (R), applicable for abrupt stall and depending on, among others, three delay parameters associated with the flow at the inlet, outlet and compressor blades. The first two parameters can be calculated, whilst the last one must be measured. Moore, 1984b, modified his model to accommodate the oscillatory condition of the reverse and stable flows, i.e., he modeled the surge. Mainly based on these publications, the Moore-Greitzer model was developed in Moore and Greitzer, 1986.

5.1. Moore-Greitzer Model

Only the major topics of the Moore-Greitzer model are described in this paper, in order to allow the reader to follow the text. The reader is encouraged to read Moore and Greitzer, 1986 and Greitzer and Moore, 1986. The model is related to a N-stage axial compressor with admission and discharge ducts, a reservoir, a flow control valve and a duct discharging the compressor air from the reservoir to the atmosphere.

This model is a simplified model of a turbojet, where combustion chamber is represented by the reservoir and the turbine by the throttle valve, as indicated in Fig. (5).

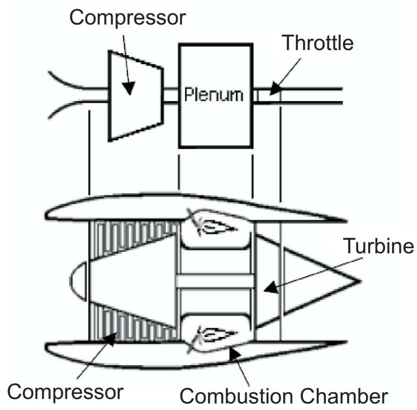


Figure 5: Turbojet and the Moore-Greitzer model schematics (Laderman *et al.*, 2003).

The compressor pumps air from an ambient at total pressure P_t to a reservoir at static pressure P_s. For the sake of simplification, only the reservoir flow is compressible although pressure gradients are taken as negligible. Suck assumptions are not really a problem because stability issues may be relatively independent of the flow Mach number in the regimes of interest (Paduano *et al.*, 2001). The air entering the compressor is considered irrotational and uniform radially. Provided the hub-tip ratio is large, as it happens in the compressor last stages, these assumptions are acceptable. This model is usually referred as low speed model due to the assumptions listed above.

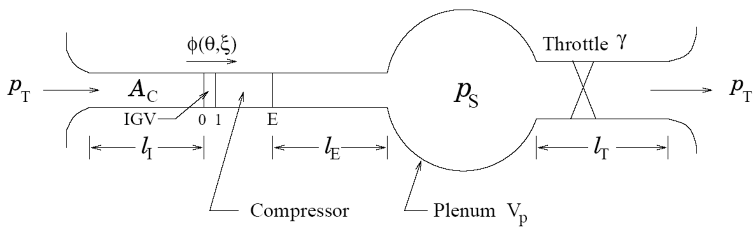


Figure 6: Schematic of the compressor installation for the model.

Figure (6) must be read taking into account that lengths were non-dimensionalised by R. It is calculated the flow at the non-dimensional time ($\xi = Ut/R$), at the tangential (θ) and axial (η) directions through the instantaneous flow (ϕ) and pressure (Ψ) non-dimensional coefficients, where

$$\phi\left(\xi,\theta,\eta\right) = \frac{C_x}{U} \;\; e \;\; \Psi\left(\xi\right) = \frac{P_s - P_t}{\rho U^2}, \tag{2}$$

where C_x is the axial velocity and ρ the air density.

Let $\widetilde{\varphi}\left(\xi,\theta,\eta\right)$ represent the flow perturbation field and $\left(\widetilde{\varphi}_{\theta}\left(\xi,\theta,\eta\right)\right)$ and $\left(\widetilde{\varphi}_{\eta}\left(\xi,\theta,\eta\right)\right)$ its tangential and axial gradients. The local average flow coefficient at station 0[3] is calculated by

$$\Phi\left(\xi\right)=\frac{1}{2\pi}\int_{0}^{2\pi}\phi\left(\xi,\theta,\eta=0\right)d\theta\ \Rightarrow\ \phi\left(\xi,\theta\right)=\Phi\left(\xi\right)+\widetilde{\varphi_{\eta}}\left(\xi,\theta,\eta=0\right). \tag{3}$$

It is possible to show that the pressure coefficient can be calculated as

$$\Psi\left(\xi\right)=\psi_{c}\left(\Phi+\widetilde{\varphi_{\eta}}\right)|_{\eta=0}-l_{c}\frac{d\Phi}{d\xi}-m\frac{\partial\widetilde{\varphi}}{\partial\xi}|_{\eta=0}-\frac{1}{2a}\left(2\frac{\partial^{2}\widetilde{\varphi}}{\partial\xi\partial\eta}+\frac{\partial^{2}\widetilde{\varphi}}{\partial\theta\partial\eta}\right)|_{\eta=0}, \tag{4}$$

where m is an empirical parameter related to the geometry of discharge duct, l_c is the aerodynamical compressor length and ducts, given by $l_c=l_I+1/a+l_E$, a is the blade passage flow lag due to inertial effects (Eq.(5)). For one cascade

$$\frac{1}{a}=k\frac{chord}{R\cos\gamma} \tag{5}$$

where γ is the stagger, k is an empirical constant that takes account of the effects of axial spacing of the cascade and ψ_c is the hypothetical compressor characteristic curve calculated for the condition of axisymmetric flow, i.e., absence of rotating stall.

The characteristic curve ψ_c has been approximated by a 3rd degree polynomial (Eq. (6)), based on parameters ψ_{c0}, H and W, as indicated in Fig. (8). Values for H and W were taken such that the inflexion point, which is also a point of symmetry, could be located.

$$\psi_{c}\left(\phi\right)=\psi_{c0}+H\left[1+\frac{3}{2}\left(\frac{\phi}{W}-1\right)-\frac{1}{2}\left(\frac{\phi}{W}-1\right)^{3}\right] \tag{6}$$

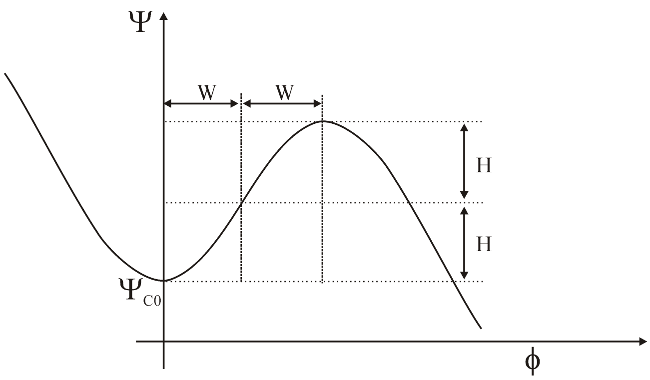


Figure 7: Compressor characteristic curve for axisymmetrical flow.

For isentropic flows, from the continuity equation one can write

$$\frac{\dot{m}_{c}-\dot{m}_{T}}{V_{p}}=\frac{d\rho}{dt}=\frac{1}{a_{s}^{2}}\frac{dP_{S}}{dt}, \tag{7}$$

where $\dot{m}_c$ is the mass flow entering the compressor, $\dot{m}_T$ the mass flow leaving the valve. From Eq. (7) it is possible to write

$$\frac{d\Psi}{d\xi}=\frac{1}{4l_{c}B^{2}}\left(\Phi-\Phi_{T}\right), \tag{8}$$

where Φ_T is the valve flow coefficient, given by

[3]Compressor inlet, from the IGV (inlet guide vane) leading edge or 1st rotor

$$\Phi_T(\Psi) = \sqrt{\frac{2\Psi}{K_T}}. \tag{9}$$

Provided that the perturbation is periodic, the first terms of the development of a Fourier series, together with the Galerkin procedure for the solution of Eq. (4), it is possible to write the simplified system of equations:

$$\frac{d\Psi}{d\xi} = \frac{W/H}{4B^2}\left(\frac{\Phi}{W} - \frac{\Phi_T}{W}\right)\frac{H}{l_c} \tag{10}$$

$$\frac{d\Phi}{d\xi} = \left[\frac{\psi_{C0} - \Psi}{H} + 1 + \frac{3}{2}\left(\frac{\Phi}{W} - 1\right)\left(1 - \frac{1}{2}J\right) - \frac{1}{2}\left(\frac{\Phi}{W} - 1\right)^3\right]\frac{H}{l_c} \tag{11}$$

$$\frac{dJ}{d\xi} = J\left[1 - \left(\frac{\Phi}{W} - 1\right)^2 - \frac{1}{4}J\right]\frac{3aH}{(1 + ma)\,W} \tag{12}$$

where J is the square of the amplitude of the angular disturbance of the axial-flow coefficient. This model is applicable for abrupt stall and deep surge.

6. Results and Discussion

Equations (10), (11) and (12) were solved numerically using a series of in-house developed softwares and the commercial software $MATLAB$. It has been studied the influence of the volume of the combustion chamber on the type of instable behavior of a 5-stage axial-flow compressor for a 1 MW gas turbine engine, that may show up when the engine is required to work near and above the stable region. Data for this study were obtained using the software developed by Tomita, 2003.

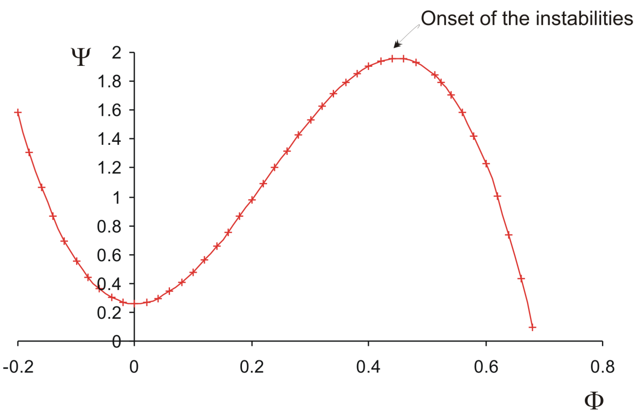


Figure 8: Axisymmetric compressor characteristic.

The nondimensionalization employed transforms the usual family of curves in the compressor map, one for each compressor speed, to one single characteristic (Gravdahl, 1998). Since this study is applicable for a compressor under design, experimental data and the methods of Day, 1994 and Koff, 1983 were used to obtain the complete axisymmetric compressor characteristic (Fig. (8)). Table (1) shows the input data used for this study based in the Moore-Greitzer method.

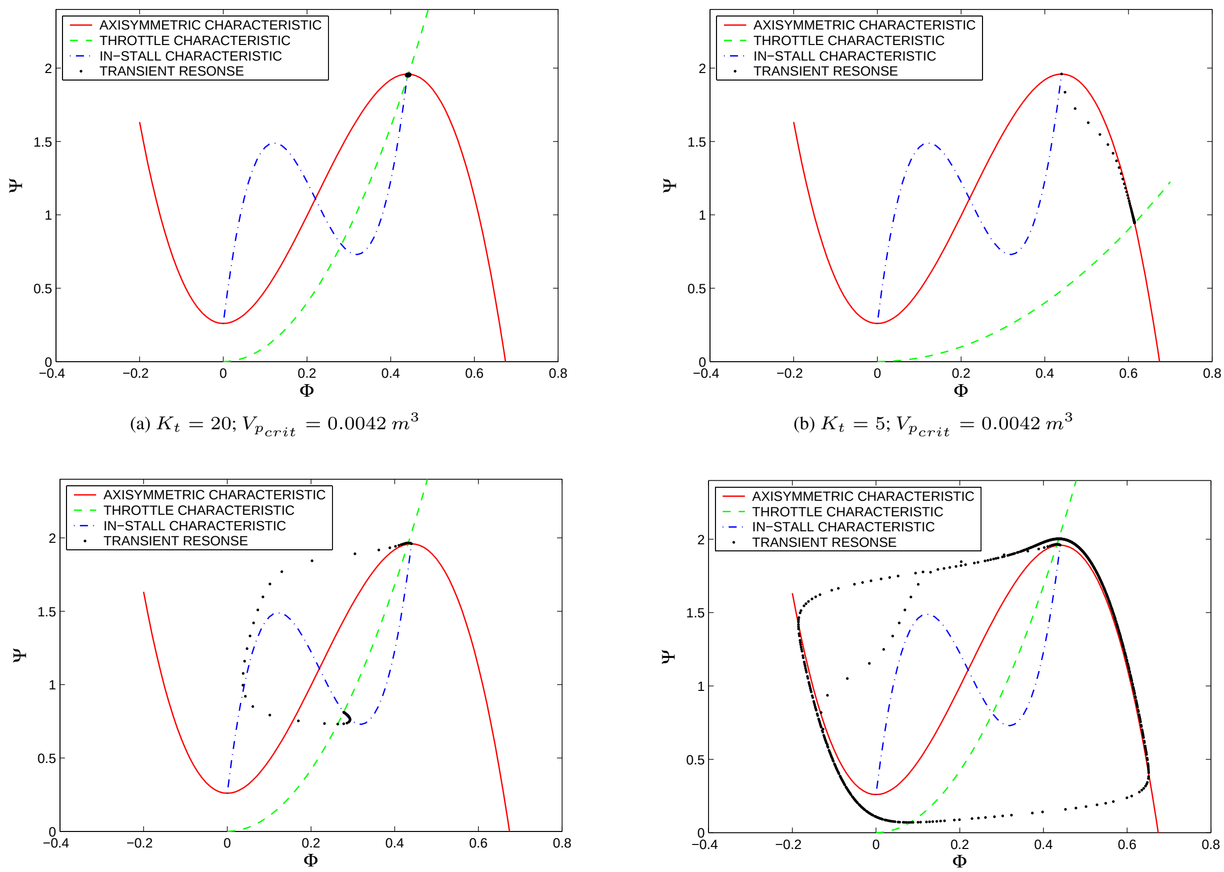
At the initial count of time ($\xi = 0$) the flow and pressure coefficient values are set as the values at the onset of the instabilities (Tab. 1) and represented by symbols Φ_{inst} and Ψ_{inst}, respectively. The initial value for J_{inst} was set to 0.01, following Greitzer and Moore, 1986.

Figure (9) shows results of the calculations for the 100% design speed. Figure (9a) shows the value of K_t that models the inception of instabilities ($K_{t_{inst}} = 20$). For $K_t \leq K_{t_{inst}}$, that is, the region of compressor stability, the new operating point will always correspond to the values of the intersection point of the characteristic curve with the system curve (throttle characteristic) curve (Fig. (9b)). For $K_t > K_{t_{inst}}$, the region above the stability limit, one may have rotating

Table 1: Compressor model parameters.

Design		Performance	
Sym		**Sym**	
Ac	0.0284 m^2	Ψ_{c0}	0.26
Lc	0.228 m	W	0.22
Le	0.1 m	H	0.85
Li	0.1 m	Φ_{inst}	0.44
R	0.112 m	Ψ_{inst}	1.96
N	5	Kt_{inst}	20
m	1.5	J_{inst}	0.01
k	3		
a	0.2		

stall (Fig. (9c)), with the new operating point determined by the intersection of system curve with in-stall characteristic curve, or surge (Fig. 9d)), without a defined operating point. For the calculation of the limiting value of the volume, when rotating stall occurs ($V_{p_{crit}}$), in the region near and above the instability line, K_t was set to $1.05K_{t_{inst}}$ and $V_{p_{crit}}$ was $0.0042\ m^3$, i.e., for a combustion chamber whose volume is greater than $0.0042\ m^3$ the compressor will enter surge.



(a) $K_t = 20$; $V_{p_{crit}} = 0.0042\ m^3$

(b) $K_t = 5$; $V_{p_{crit}} = 0.0042\ m^3$

(c) $K_t = 21$; $V_{p_{crit}} = 0.0042\ m^3$

(d) $K_t = 21$; $V_p = 0.0043\ m^3$

Figure 9: Transient responses for 100% design speed.

The critical value for the Greitzer stability parameter for this compressor is $B_{crit} = 0.36$. With this value it is possible from Eq. (1) to obtain $V_{p_{crit}}$ as a function of the rotor speed. Figure (10) shows that if the speed decreases, $V_{p_{crit}}$ increases, in other words, at low speeds it may be possible to enter rotating stall. For example, if the combustion chamber volume is about $0.012\ m^3$ and the rotational speed above $60\ \%N_{design}$, deep surge will be the type of instability that is most likely to occur.

Figure (11) shows the periodic variation of the flow coefficient Φ, pressure coefficient Ψ and squared of amplitude of disturbance J with the non-dimensional time ξ, at the surge inception. From this condition, V_p has been varied and the

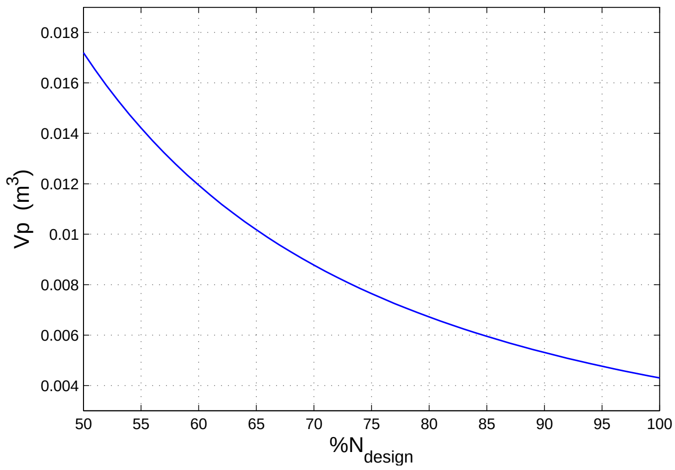


Figure 10: Combustion chamber volumes as function of the rotational speed.

value of the surge frequency, for each curve, was calculated and shown in Fig. (12). From the analysis of these figures it is possible to infer:

- For each constant speed curve the surge frequency is inversely proportional to the combustion chamber volume;
- For fixed values of the combustion chamber volume, the surge frequency decreases with engine speed;
- To avoid the abrupt rotating stall in the speed range of 70% to 100% design speed, it is required that the combustion chamber volume be larger than the critical volume determined for the 70%.

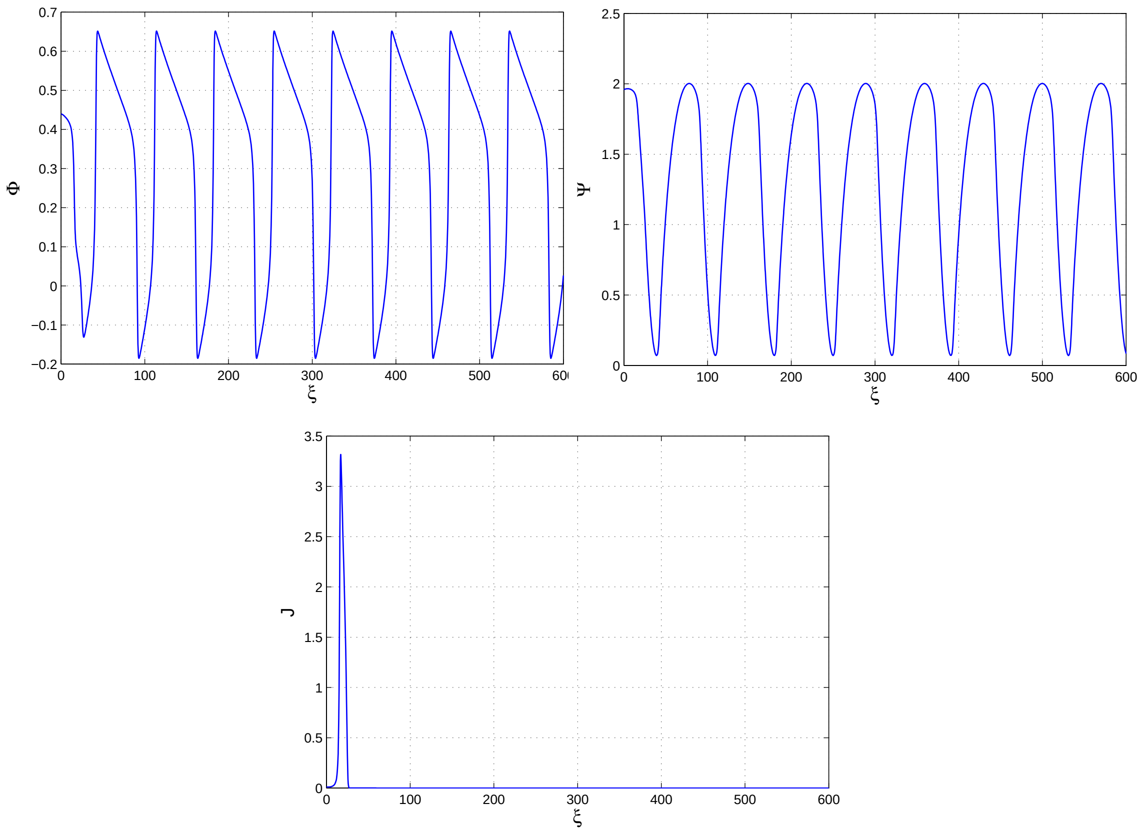


Figure 11: Φ, Ψ and J as function of time at the surge onset.

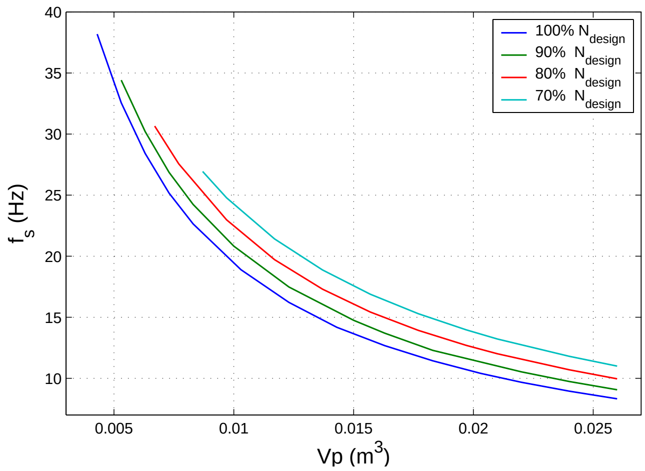


Figure 12: Surge frequencies as function of the combustion chamber volumes and engine speed.

7. Conclusion

The influence of the combustion chamber volume on the onset of surge and rotating stall has been studied giving guidance to the design of the combustor. The Moore-Greitzer model was implemented in a series of in-house computer programs that allowed the investigation of the influence of the combustion chamber volume and engine speeds on the type of the instability that can happen to the compressor under studied. The results presented in this article have shown that at low engine speed it is more likely to enter in rotating stall since the volume of the combustor is less than the critical value for low engine speeds. The surge frequencies for different volumes were also calculated, giving an estimation of the surge cycle for engine tests in the future.

8. Acknowledgements

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