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Examination of a Surge Suppression System Designed for a Centrifugal Compressor System

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Abstract

Centrifugal compressors are used in many areas of industry, for example from natural gas transportation to superchargers of internal combustion engines and compressor stages of aircraft engines to the air compression equipment of the rapidly expanding fuel cell powered systems. With regards to each field of usage of compressors, reliable operation is an important criteria for the operation of the compressor, in addition for ensuring a continuous and adequate medium supply. This process is easier to implement if the equipment is used in terrestrial applications, where these engines can be operated in closely stationary conditions, because there isn't any sudden change in environmental parameters, so these engines could be well optimized by their geometrical design for reaching its maximum efficiency. In vehicle industry, but especially in the case of aerial applications, the task is much more complicated, because the operation of the compressor is determined by the current operating states of the engine, but here at least the conditions of the air supply are almost the same. In the case of aircraft, the operation of the compressor is determined not only by the current operating state of the gas turbine engine, but also by the attitude of the aircraft. For this reason it is necessary to investigate the unstable operating conditions of centrifugal compressors and to develop equipment which is suitable for eliminating these affects.

Keywords

centrifugal compressor, surge phenomenon, surge supression system, self-recirculating bleed holes

1 Introduction

The complete cycle of the surge phenomenon can be divided into *five main phases*. Prior to the onset of the surge phenomenon, separations occur in the diffuser as a result of the continuously increasing back pressure at the compressor discharge area, due to which the effective flow cross-section in the bladed diffuser passages decreases. The flow in these narrowing channels accelerates for a while, and when the back pressure reaches a critical level, the speed of the continuous flow of the medium gradually decreases. The moment when the medium starts to move backwards due to the counter pressure in the diffuser is called *the beginning of the surge*. In accordance with that, one could distinguish two main types of instabilities. If the back pressure in the diffuser does not increase to the critical level during the formation of detachments, then the field of the flow separation starts to rotate in the same direction as the rotational direction of the impeller, but at a lower speed than the rotor speed. Thus will exhibit a *rotating separation field*. The phenomenon of rotating separation persists as long as the medium has to overcome on a back pressure greater than the nominal one. However, above the critical pressure ratio at the given rotational speed, the flow collapses and the medium will move backwards. It has to be mentioned that it is a general phenomenon for turbomachines working with compressible and incompressible fluids at nearly constant density, even under normal operating conditions (He and Zheng, 2018).

In the characteristics of the centrifugal compressor, the operating modes corresponding to the highest efficiency, called as the points of the operating curve, are not located near the highest pressure ratio on the curves at every given constant rotor speeds, because even in the case of a time-constant, quasi-stationary flow with respect to the resulting velocity vectors a certain amount of backflow could already been experienced, because there is a gap between the wall of a standing house and the tip section of the rotating impeller.

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However, the velocity vector of this backflow at the wall is modified by the tangential flow appearing at the tip edges of the impeller blades, which is created by the pressure difference occured on the two sides of the blades. The resulted separations originated from the superposition of the impeller blade tip flows and the vortices resulting from their propagation by the flowing medium are the main causes of the decrease in efficiency due to the heat generation resulting from the increased internal friction of the flowing medium. The vortices that continue from the impeller channels towards the diffuser break off when they exit the impeller channel, and by modifying the flow field, the velocity triangles of the flowing medium at the beginning section of the leading edge of the stationary blade row of the diffuser are modified. The non-optimal, swirling flow further increases the separations occurring at the leading edge of the stationary blades of the diffuser, which can be defined as the trigger points of the onset of the surge phenomenon. This results in the collapse of the complete flow (Xue and Wang, 2020).

The third phase of the surge is *the backflow stage*, when due to the collapse of the flow, the medium starts to flow backwards in the entire fluid domain. As it was mentioned above, this is caused by the high back pressure occurs at the compressor discharge area after the diffuser. This disappears immediately with the return flow, so despite of the decreasing back pressure, due to the inertia of the working medium, the backflow will stop with only a short delay, but the static pressure in the section after the diffuser will never decrease to the actual level of the ambient pressure. However, with the reduction of the backpressure, since the energy transfer is performed in the continously rotating impeller of the compressor according to the torque impulse equation, the medium starts to accelerate again in the impeller passages. This phase is called as *the point of return*. If the pressure downstream of the diffuser drops below the critical value during operation, for example with the help of a throttle valve, then the stable flow is maintained, and this phase could be defined as the *return state*. If the back pressure increases again above the critical level, the system returns to the starting phase and the sequence is repeated in accordance with the above mentioned processes until the causing effect is maintained. The surge phenomenon can be easily recognized from the rumbling noise and vibration originated from these undamped pressure fluctuation, and it yields some mechanical load resulting from these high-amplitude and high-frequency pressure oscillations that could lead to some damages or to the complete failure of the compressor.

2 The simplified compression system model

During the testing of unstable conditions of the compressor operation, E.M. Greitzer examined and developed a simplified model which describes the operation of a compression system. This describes a system-dependent parameter by which it can be described that the system is still operating in the stable range or it exhibits an undamped, unstable operational state. This system parameter is denoted by B, as behaviour-parameter and its formula is described as (Greitzer, 1976):

$$B = \frac{u}{2a}\sqrt{\frac{V_P}{A_C L_C}} = \frac{497.42}{2\times 376.8}\sqrt{\frac{0.003824}{0.00283\times 1.25}} = 0.68667 \tag{1}$$

The parameters in the formula:

- u: tangential speed of the impeller [m/s],
- a: the local speed of sound [m/s],
- V_P: Plenum Volume after the diffuser section of the compressor based on the 3D model, [m^3],
- A_C: the equalent Area of the compressor geometry [m^2],
- L_C: the equalent lenght of the compressor geometry [m].

Figs. 1 show the dimensionless mass flow and the converted dimensionless pressure at the onset of surge point at the compressor discharge section as a function of time in case of a given behaviour parameter. This is the point during the operation when the medium in the compressor starts to exhibit undamped oscillation.

In aerial applications not only the mechanical damage could be the result when the compressor of an aircraft engine starts to fall into surge, but also dangerous flight situation could be happened which could cause fatal problem. As it was mentioned above, during the surge phenomenon, the amount and the direction of the main flow are changing periodically in the turbomachine and this results in a sudden reduction of the output power of the engine in the case of turboshaft engines and in relation with the propulsion principles of thrust generation. In case of a gas turbine operation at low altitudes, and in certain flight situations, such as high pitch rolls or climbs or at low-altitude turns, the sudden loss of power or thrust can cause the aircraft to stall and crash.

But this is not the only situation when the inappropriate operation of the compressor of the engine can be a

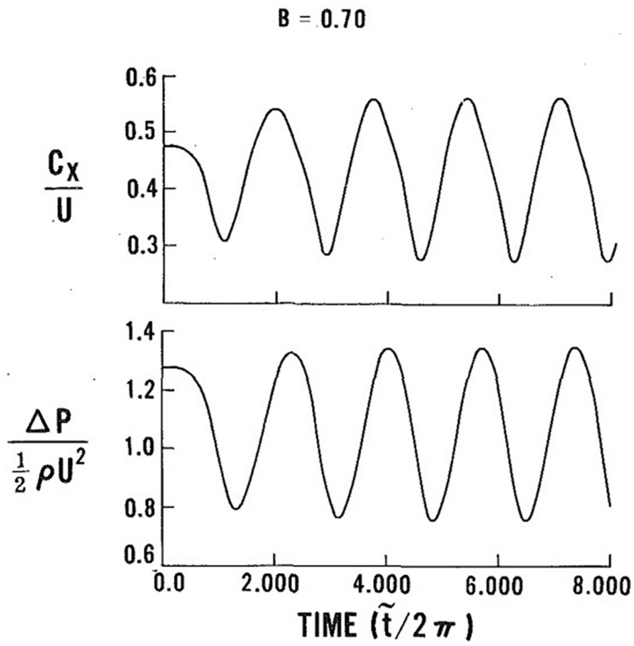

Fig. 1 The dimensionless mass flow and the converted dimensionless total pressure at the discharge section of the compressor are shown against the elapsed time at a given behaviour parameter by Greitzer, 1976

source of problem. The pressure fluctuations arising in the compressor stages can spread through the engine into the combustion chamber, where if the condition of the incoming air is changed due to the insufficient air supply, as well as a sudden change in the static pressure, these circumstances can lead to flame extinction which ultimately results in a complete shutdown of the engine. In the worst case, the combustion chamber becomes flooded with unburned fuel to such an extent that could ignite and explode uncontrollably in the combustion chamber because of the radiated heat from the inner wall surface of the combustion chamber at normal operating temperature. This could lead to the complete explosion of the engine. Then the engine can no longer be restarted and the flight must therefore be aborted.

3 Inicialization of the measurement

Before the compressor test bench was set to perform the measurements with the prototype impellers, calibration tests were performed for setting the monolithic silicon pressure sensors by using U-tube manometers. These pressure sensors provide high level, accurate analog output signals that are proportional to the applied pressures. Based on the results of this measurements the appropriate pressure sensors could be choosed in relation with the ranges of its input parameters. The measurement of the temperatures was performed by thermocouples and its calibration was done by using bar thermometers. These equipments provide voltage output signals in relation with the pressure and the temperature change. The zero point for the pressure measurement was set to the actual ambient pressure which was meaured also by a absolute pressure sensor and the zero point for the temperature signals was set to the ambient temperature manually in the digital measurement system based on a Torricelli-barometer. The mass flow rate was measured by a differential pressure flow meter and the pressure drop at the rim was measured by a center zero digital pressure sensor, because at the onset of surge conditions the direction of the main flow in the compressor could move straight and in reverse. All of the gathered datas were collected by a USB unit from National Instruments USB 6218. A LabVIEW program (National Instruments, 2021) was created and used for indicating the measured datas directly and a specialized modul was created in this Labview code for collecting all the measured data from the pressure sensors and from the thermocouples. The raw data was written into a text file from where the follow-up postprocessing were made in MS excel files after each measurements. The measuring part of the work started with the original impeller of aluminium alloy.

The general arrangement of the test bench can be found in an article previously published in relation with this research (Faltin and Beneda, 2023) the examined centrifugal compressor was the part of a Holset turbocharger rotor section. The turbine was driven by another compressor produced by Ganz.

4 Performation of the measurement

The measurement started with the original impeller of the compressor and by changing the angle of the throttle or in other words the choke valve at the discharge section of the compression system the surge phenomenon could be exhibited at almost the closed position of the valve. A picture about the original impeller and the measured surge characteristic can be found on Fig. 2.

After the measurements with the original impeller (Fig. 3) the turbocharger was moved from the test bench and it was disassembled to change the original impeller to the experimental one which contained the recirculating holes in it shub section. This stage of the work can be observed on Fig. 4.

Because of the manufacturing constrains of recirculating holes in the experimental impeller, it was manufactured by 3-dimensional technology. It was made by

Fig. 2 The original impeller of the compressor

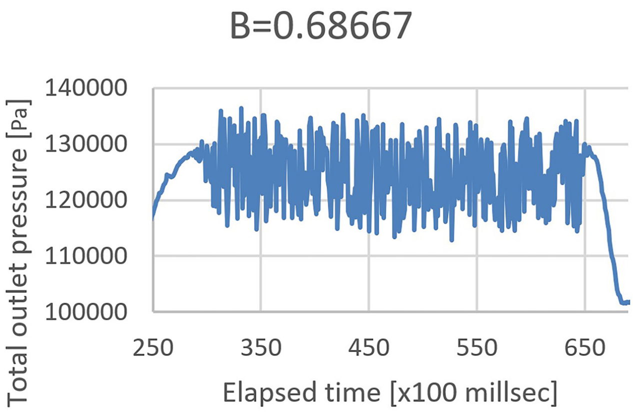

Fig. 3 The characteristic of surge of the original impeller

Fig. 4 The change of the original impeller to the 3D printed experimental impeller

polyamide powder with the help of laser sintering technology. During the testing of the impeller with 3 mm holes it was measured that the surge phenomenon exhibited a different characteristic. As it can be seen on Fig. 5 the amplitude of the pressure oscillation is reduced and the level of the oscillation was 4 kPa less than the peak presssure at a given rotor speed.

So it can be noted that by using the recirculating holes the affect of the surge phenomenon could damped.

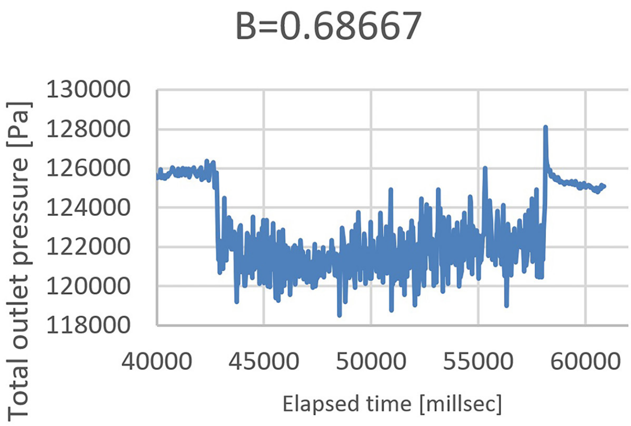

Fig. 5 The characteristic of surge of the experimental impeller with 3 mm recirculating holes in diameter

And another interesting phenomenon can be observed on the picture. This bleed air recirculating system is self-aspirated which means that it can work without using any other external energy source. The existing static pressure difference between the volute section of the compressor and in the blade passages of the rotating impeller can provide enough potential to impel the compressed air through the elements of the recirculating device. But the radial contamination which can be seen on the backplate of the impeller (Fig. 6) proved that there is a leakage flow in radial direction in the gap between the backplate of the impeller and the front side of the compressor volute casing. To improve the efficiency of the recirculating device and to minimize the loss of the amount of the recirculating air a 2-stage labyrinth sealing was designed and it was applied firstly on the next 3D printed impeller on which the increased holes was created and manufactured which were 5 mm in diameter. The number and the dimensions of the labyrinth sealing were determined by the geometrical constrains of the impeller and the possibilities of the manufacturing method.

Fig. 6 The radial contamination on the prototype impeller equipped with 3 mm recirculating holes

So it could be noted that the damping effect can be experienced by using the recirculating holes, but the efficiency of the backflow would be inappropriate because of the loss of the recirculated flow in radial direction at the backplate.

The work is continued by manufacturing and testing the impeller with increased holes in diameter. This impeller and its labyrinth sealing can be seen on Fig. 7.

The measurement data gained by using the modified impeller equipped with 5 mm holes in each blade passages has proved the tendency of the change in the characteristic of the surge phenomenon. It could be seen on Fig. 8 that further reduction was experienced in relation with the amplitude of the pressure oscillation and the peak pressure was a little bit higher than at the impeller with 3 mm holes. Further interesting experience is that the initial phase of the surge phenomenon is longer than it was observed in the previous impellers at the values of the elapsed time period from 35000 ms to 40000 ms. The continously increasing initial period of the phenomenon can be observed before the fluid is falling into deep surge. So this change was not happened in a sudden moment, but it has a processive increasing period which can be also used to predict the system state if it approximates to the deep surge condition.

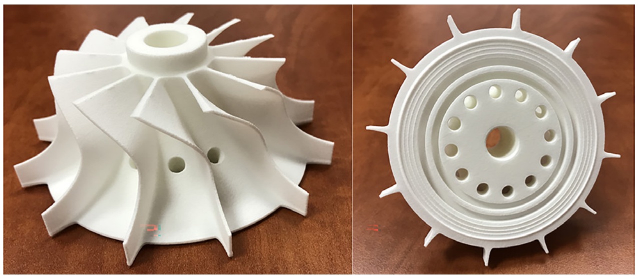

Fig. 7 The front and the aft side of the experimental impeller manufactured with 5 mm holes

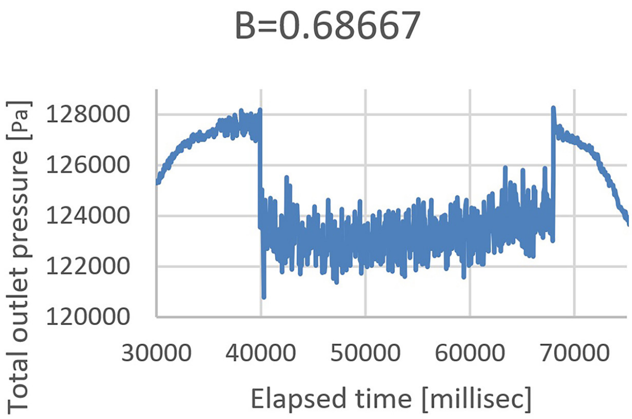

Fig. 8 The surge characteristic of the experimental impeller with 5 mm holes

5 Summary

The surge phenomenon can cause highly dangerous conditions in relation with the integration of the compressor equipment or in the operation of the compression system itself. The self-recirculating devices could operate without using any other external energy sources, because of the static pressure difference between the compressor volute and the place of usage of the backflow at the impeller blade passages. The application of the bleeding holes in the impeller hub section provides access between the volumes of the blade passages and this extra volume in the compression system acts like a damping volume which can be used as a cavity which able to work as a pressure accumulator. Based on these experiences it could be noted that by using these recirculating holes in the impeller hub the amplitude of the pressure fluctuation can be decreased which helps to improve the increase the effective lifetime of the equipment between systems checks.

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