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Identification of phenomena preceding blower surge by means of pressure spectral maps

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ABSTRACT

In this paper the concept of 'quasi-dynamic' pressure measurements is introduced and used in order to obtain compressor spectral maps describing the process of surge onset in a centrifugal blower. In the experimental rig pressure signals were recorded at several throttling valve positions and subjected to Fourier analysis. The results are shown in the form of contour maps with frequency on the horizontal and control valve opening on the vertical axis. The maps allow the frequencies dominating at consecutive stages of entering surge to be distinguished. Moreover, comparison of the maps, obtained at different pressure tappings allowed the point of instability onset to be determined. Research confirmed that the first disturbances appeared in the inlet recirculation zone much before the surge. The scale of the disturbances were considerable, amplitude increased about 10 times together with an immediate rise of its mean value. On the other hand, flow structure was not regular as no dominating frequency was detected. When the flow rate continued to decrease the instabilities propagated throughout the impeller towards the diffuser and certain frequency peaks appeared suggesting the formation of regular flow structures. At deep surge spectrum is dominated by one peak that is in good agreement with frequency of the corresponding Helmholtz resonator. Moreover, due to the high accuracy of the 'quasi-dynamic' method it was possible to recognize that damped forms of non-stable phenomena were present in stable working conditions.

1. Introduction

1.1. State of the art

1.1.1. Non-stable phenomena in centrifugal compressing units

Compressing units (compressors, blowers and fans denoted as *unit* henceforth) are known to be prone to enter a non-stable work regime at low mass flow rates. This is associated with several complex phenomena and flow structures appearing within the rotor and diffuser. An unstable work regime is regarded as very dangerous for a unit's safety not only due to the immediate drop of generated pressure head but, predominantly, because of the serious threat of permanent unit damage. Therefore, compressing units are usually equipped with systems assuring operation at a point separated from the region prone to non-stable phenomena by a so-called security margin. Nevertheless, this solution is not free from drawbacks. Provision of a satisfactory security margin reduces the compressor operating range by cutting off the region where its efficiency is the highest. On the other hand, a small security margin increases the risk of spontaneous inception of unstable phenomena which, at certain conditions, may appear even at the region regarded as stable [1].

This situation has been recognized since the 1950s and has been the subject of extensive research thereafter. Non-stable phenomena in centrifugal units were first identified and thoroughly analysed by Emmons et al. [2]. In 1976 Greitzer developed a mathematical model of instabilities [3] and confirmed it by experiment [4]. Afterwards these findings were developed to describe the shape of fully-formed transients [5,6]. Two main flow phenomena have been identified; rotating stall and surge. Rotating stall corresponds to the formation of vortex structures which are stationary in a certain rotating frame of reference (stall cells) whilst the term surge refers to fluctuations in the axial direction [5]. Greitzer described these states as different forms of natural modes of fluid oscillation. In normal working conditions they are suppressed, whereas at lower mass flow rates damping decreases and bifurcation into unstable work conditions is likely to occur [7,8]. Apart from the fact that the Greitzer model was originally created for axial compressors, it was also successfully applied to describe the operation of centrifugal units [9]. Spakovsky modified it in order to describe the impeller–diffuser interaction [10] and provided a description of pressure waves travelling forwards and backwards.

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Nomenclature

π_p	machine gross pressure ratio
A_{p-in}	cross-section area of the passage at the impeller inflow
A_{p-out}	cross-section area of the passage at the impeller outflow
c	speed of sound in the air at 23 °C
f_{BP}	blade passing frequency (2.3 kHz)
f_H	Helmholtz resonator frequency
f_{rot}	impeller rotation frequency (100 Hz)
d	blade tip clearance
f_s	frequency characteristic for phenomena present at stable operation
f_n	frequency characteristic for phenomena present at non-stable operation
L	dimensionless parameter describing gauge position along the span
L_p	average length of a passage
$\dot{m}$	machine gross mass flow rate
p_{s-in}	signal representing static pressure at the inlet
$p_{t-in1-2}$	signal representing total pressure at the inlet
p_{s-out}	signal representing static pressure at the outlet
p_{s-impX}	signal representing static pressure at the impeller shroud ($X \in \{1, 2, 3\}$)
R_{diff}	radius of the diffuser outlet
R_{in}	radius of the blade mid-span at the leading edge
R_{out}	radius of the blade mid-span at the trailing edge
R_{pin}	radius of the inlet pipe
R_{pout}	radius of the outlet pipe
S_{in}	blade span at the leading edge
S_{out}	blade span at the trailing edge
V_0	volume between impeller outlet and the control valve

van Helvoirt and de Jager have extended the model to include pipe system dynamics [11]. Due to the different character of pressure fluctuations a special model was also formed to describe stall in a centrifugal diffuser. A two dimensional model of rotating stall in vaneless diffusers was created by Jansen [12] and then developed by Frigne and van den Braembussche [13]. Extension into three dimensions was proposed by Shen et al. [14].

In fact, rotating stall can have numerous forms depending on the size, number and shape of stall cells [15]. Frigne and van den Braembussche examined flow patterns obtained in a centrifugal compressor at different positions of the inlet vanes [16]. On the basis of comparison of their experiment to previous papers the authors have distinguished three main stall patterns:

- *Diffuser rotating stall* caused by a strong interaction between boundary layer and inviscid core flow within a vaneless diffuser. The number of stall cells can vary. Their frequency is low $f \leqslant 0.2f_{rot}$.
- *Abrupt impeller rotating stall* appearing due to a strong interaction between the impeller and diffuser. The number of stall cells increases with decreasing mass flow rate. Their frequency is in the range $0.2f_{rot} \leqslant f \leqslant 0.4f_{rot}$.
- *Progressive impeller rotating stall* caused by gradual flow separation in the impeller. The number of stall cells remains constant. Their frequency is in the range $0.5f_{rot} \leqslant f \leqslant 0.8f_{rot}$.

Izmaylov [17] introduced a more complex classification resulting from examination of different diffuser shapes. However one may observe certain similarities. In both cases the diffuser modes are very slow (or even stationary if it is vaned), and may have a different number of cells (1–7), while impeller cells travel faster and do not exhibit such a wide range of possible number of stall cells. It has to be noted that the location and character of the stall depends also on impeller rotational velocity [18,9]. Cells may have different sizes and are observed as non-symmetrical pressure variation [19]. Haupt et al. noted that it is possible for different modes to occur concurrently [20].

1.1.2. Non-stabilities onset

Classic stall protection schemes rely on the detection of typical disturbances preceding its inception [21]. The process of stall initiation is very short and has a timescale comparable to the period of the impeller revolution [22]. If the pressure instability continues to propagate in the axial direction it may lead to immediate surge inception [23] which appears in the time range roughly one order of magnitude longer than the period of the stall [24]. Then, due to the stated difference in periods, stall and moments of on design compressor operation may spontaneously appear and disappear during a single cycle of the surge [5,25]. Once started surge may be stoped by adjustment of throttles or bleeds. However, due to hysteresis, the rotating stall may not diminish that easily and it may require the unit to be shut down and restarted [3].

Examination of pressure signals in the frequency domain introduced new insight into the description of rotating stall and surge. As expected, comparison of frequency spectra at the design point to those attained at surge revealed the presence of new harmonic components [26]. The number of local maxima in the spectra reflected the complexity of the non-stable flow structures. It is also possible to observe how the structures propagate throughout the unit by analysing frequency spectra at different working conditions in different locations. In this way Tamaki concluded that the stall inception takes place in the vicinity of a compressor's leading edge [23]. Horodko in his work used the continuous wavelet transform (CWT) to obtain time-dependent frequency spectra at the surge onset [27,28,25]. Study confirmed that the rotating stall initiates from the vicinity of the impeller leading edge. Similar phenomena were observed in other experiments conducted on different compressing units [29,30]. However, no complete description of possible flow structures in that region has yet been provided. Kryłłowicz suggested [31], that it may be a structure known as *inlet recirculation* which was recognized originally in centrifugal pumps [32]. Vortex structures appear at the leading edge of impeller blades. As the mass flow rate continues to decrease, vortices grow and create a torus-like vortex structure over the whole circumference. This induces a change in the angle of attack of the impeller leading edge and stall is likely to occur.

1.2. Scope of the study

Researchers have proven Fourier analysis to be a valuable tool in the analysis of centrifugal units at surge and stall. Further application of CWT introduced the possibility of distinguishing different stages of unstable phenomena. However, the application of wavelet analysis encountered difficulties arising from the limited time-frequency resolution of the method. Therefore, the authors of this paper have decided to create a concept of quasi-dynamic tests which allow the creation of continuous frequency maps. The aim of this method is to show the repeating phenomena, resulting in the averaged spectra. Additionally, the concept allows the association of flow structure to the position of the control valve rather than to time.