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Flow instability evolution in high pressure ratio centrifugal compressor with vaned diffuser

Xiao He, Xinqian Zheng*

Department of Automotive Engineering, Turbomachinery Laboratory, State Key Laboratory of Automotive Safety and Energy, Tsinghua University, Beijing 100084, China

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ABSTRACT

Compressor stability is an essential issue for developing single-stage high pressure ratio centrifugal compressors. In this paper, a comprehensive stability map of a centrifugal compressor stage with the peak pressure ratio of 6.2 has been illustrated. Fast response pressure transducers are mounted inside the casing or the back-plate wall to measure internal transient behaviors. From Fourier analysis of pressure signals, various instabilities across time and length scales are identified. At low speeds, the impeller inlet rotating instability develops in scale and evolves into the inducer stall, which eventually induces the mild surge and deep surge of the compression system. At middle speeds, the compressor successively experiences stable state, mild surge, rotating instability, and deep surge. The mild surge region coincides with the dip region of the compressor S-shape pressure rise curve, and deep surge occurs when the compressor pressure rise raises to the second peak. At high speeds, mild surge and deep surge abruptly occur without preceding stall or rotating instability. To explain the complex surge behavior, the mechanical analogy between the compression system and the mass-spring-damper system is applied. Both mild surge and deep surge transients are found to belong to the dynamic instability of the mass-spring-damper system, and the exact form of the surge state will be determined by eigenvalues of the system governing equation.

1. Introduction

Centrifugal compressors are widely applied in automobile, aviation, oil and gas industries. For reduction of manufacturing cost and engine weight, high pressure ratio centrifugal compressors are especially preferred in the aviation industry [1]. However, due to the rapid drop in stable flow range with increasing pressure ratio [2], the application of high pressure ratio centrifugal compressors is still limited. The key challenge of designing such stages is to recover its stable flow range based on insights of compressor stability.

Compressor stability is a crucial issue on the reliability of gas turbine engines, and relating researches have lasted for almost a century. To date, mainly three groups of instabilities have been identified, namely surge, rotating stall, and rotating instability in descending order of time and length scale [3].

Surge is the breakdown of the compression system that results in large oscillations of flow properties propagating through the streamwise direction. Its length scale is at the order of the streamwise length of the compression system, and its time scale is at the order of the system Helmholtz frequency. According to the oscillation intensity and the occurrence of reverse flows, the surge phenomenon can be further classified into mild surge without reverse flows and deep surge with reverse flows [4]. Mild surge usually occurs at low mass flows before the emergence of the deep surge, but it can also exist in the middle of a constant speed line in some centrifugal compressor cases [5–8]. Deep surge always emerges beyond the lowest mass flow. It often leads to drastic fluctuations in pressure, flow rate and noise and even results in catastrophic failure of the whole system. In addition to the streamwise oscillation, the surge may also present non-axisymmetric flow patterns. The occurrence of the surge could be restricted to partial spans [9]. A phase lag may exist among surge signals at different circumferential locations [10,11]. This non-axisymmetric pattern could generate a radial force that eventually leads to severe blade rubbing.

The nature of surge was unveiled by the lumped parameter model of Greitzer [12]. It considered the filling and discharging process by a plenum model and the flow inertia by a duct model, along with the small perturbation theory for linearization. The model not only successfully reproduced similar fluctuations of flow parameters to experiments but differentiated the occurrence of the surge and rotating stall by the non-dimensional B parameter. Following research aimed to improve the model accuracy by both experimental and numerical approaches: one-dimensional CFD simulations of the real piping systems

* Corresponding author.
E-mail addresses: hex15@mails.tsinghua.edu.cn (X. He), zhengxq@tsinghua.edu.cn (X. Zheng).

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Nomenclature

a	sound velocity [m/s]
A	duct cross-sectional area [m^2]
b	blade height [m]
c_p	specific heat capacity at constant pressure [J/(kg K)]
C	static pressure rise across the compressor [Pa]
F	damper constant [kg/s]
G	static pressure drop across the throttle [Pa]
K	spring constant [N/m]
L	duct length [m]
m	mass flow rate [kg/s]
M	mass [kg]
r	distance in the radial direction [m]
U	blade speed [m/s]
V	volume of the plenum [m^3]
Z	number of blades [–]
γ	heat capacity ratio [–]
π	pressure ratio [–]
ρ	density [kg/m^3]
υ	kinematic viscosity [m^2/s]
ω_n	natural frequency [Hz]
ζ	damping ratio [–]

Abbreviations

BPF	blade passing frequency
IPF	impeller passing frequency
SP	stability parameter

Definition of non-dimensional parameters

B	Greitzer's B parameter $\frac{U}{2a}\sqrt{\frac{V}{AL}}$
Mu	impeller tip Mach number $\frac{U_2}{\sqrt{\gamma R T_{t1}}}$
Re	Reynolds number $\frac{U_2 b_2}{\upsilon_1}$
φ	flow coefficient $\frac{m}{4\rho_{t1}U_2 r_2^2}$
λ	work coefficient $\frac{c_p T_{t1}\left(\pi^{\frac{\gamma-1}{\gamma}}-1\right)}{\eta U_2^2}$
N_s	specific speed $\frac{2\varphi^{1/2}}{\lambda^{3/4}}$

Subscripts

1	impeller inlet (ambient)
2	impeller exit
4	diffuser exit
6	volute exit
t	total quantity
II	transducer location at impeller exit
III	transducer location at diffuser half section
IV	transducer location at diffuser exit

were proposed to replace the duct and the plenum model [13]; special experimental procedures were designed to accurately measure the compressor pressure rise characteristic under unstable conditions [14].

Rotating stall is the circumferential propagation of single or multiple low-momentum cells (or stall cells) that limits the compressor pressure rise. Its length scale is at the order of the compressor circumference, and its time scale is on the order of impeller passing frequency (IPF). Emmons et al. [5] proposed a definite phenomenological model about the mechanism of rotating stall: when the first stall cell is formed in a blade passage, it blocks the incoming flow and causes an incidence spike on the adjacent blade, which drives the adjacent blade to stall, and thus the stall cell rotates.

Subsequent research of rotating stall focused on how the first stall cell was initiated. So far, the long-length-scale modal-type stall inception and the short-length-scale spike-type stall inception have been identified. The modal-type stall is triggered by a small full-span disturbance whose circumferential wavelength is equivalent to the annulus. It occurs near the zero slope of the compressor pressure rise characteristic, and the time from the stall inception to the fully-developed rotating stall is about tens of rotor revolutions. The long-length-scale perturbation that triggers rotating stall can be theoretically predicted. Because of its relatively low amplitude and growth rate, the perturbation can be restrained by active control methods before the rotating stall develops mature [15]. The spike-type stall is initiated by a spike-like disturbance near the blade tip whose circumferential wavelength is only two to three blade passages. It appears at a negative slope of the compressor pressure rise characteristic, and it develops from initial stage to mature stall cells within five rotor revolutions [16]. Recently, the mechanism of the spike-like disturbance has been determined as the formation of a radial vortex connecting the blade suction surface and the casing wall [17]. Although evidence in support of the mechanisms keeps accumulating in axial compressors [18–20], it is not necessarily the case in some centrifugal compressors: radial vortex structures were observed in both the impeller and the vaned diffuser, but the compressor presented modal-type stall instead of spike-type stall [21]. Thus, further experimental studies on cases of current interest are required.

Rotating instability is the pre-stall disturbance that is continually changing in amplitude and frequency. Its length scale is at the order of the compressor blade pitch, and its time scale at the order of blade passing frequency (BPF) [22]. Rotating instability usually exists in cases with large tip gaps, and its mechanism was proposed to be the circumferential movement of the radial vortex inside a blade passage [23]. Although certain similarity between the rotating instability and the spike disturbance that triggers rotating stall exists, they are not the same phenomenon as rotating instability emerges well before the stall onset and lasts for a long time. In our previous research, rotating instability was also observed in a centrifugal compressor and was referred to as high-frequency pressure oscillation [10] or high-frequency stall [24].

Among the brief review of multiple compressor unstable phenomena, fundamental understandings towards compressor stability have been achieved. However, most case studies were involved in axial compressors rather than centrifugal compressors, and differences between those two categories do exist. Because the centrifugal force provided by the impeller radial part counteracts against the streamwise adverse pressure gradient and stabilizes the whole impeller, the impeller axial part can operate under more severe unstable conditions than axial compressors do. For impellers with high inlet tip to outlet radius ratios, large areas of inlet recirculations occur when operating at low mass flows at part speeds, which leads to the impeller stall [25]. The inlet recirculation is different from the conventional rotating stall [26] but instead appears as the symmetrical blockage around all the annulus. Due to the streamline curvature and the Coriolis force in the impeller passage, the strength of its secondary flow is stronger than the axial compressor, resulting in a significant spanwise distortion of the vaned diffuser incoming flow [27]. As a consequence, the flow instability of vaned diffusers may occur at either the shroud section like the spike-type rotating stall [28] or at the hub section like the hub-suction surface corner stall [29]. Moreover, the compressor stability issue for high pressure ratio centrifugal compressors is expected to be more complex than ordinary centrifugal cases. The inlet condition of