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Control oriented modeling and analysis of centrifugal compressor working characteristic at variable altitude

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A R T I C L E I N F O

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A B S T R A C T

The centrifugal compressor converting the kinetic energy into the pressure increase has been extensively used for industry applications. Due to its high rotational speed, the volume and weight could be greatly reduced which makes it suitable for in-flight gas compression systems. Combining the high altitude circumstances such as the air density, pressure, temperature, etc., an centrifugal compressor model is first developed in this paper. This model takes the changing properties of the atmosphere, air pressure and air density into account. The working characteristics of the centrifugal compressor at different altitudes are studied. Meanwhile, the effects of the parameter variations on the compressor performance is analyzed. Moreover, dynamics of the air flow and pressure are investigated with varied altitude. A closed loop controller based on super twisting sliding mode approach is proposed to make the mass flow track the reference.

1. Introduction

The centrifugal air compressor has been widely used for the gas transport, gas compression and gas injection both in the land and aviation industries. Compared with positive displacement compressors such as scroll and piston compressors, the centrifugal compressor is more compact due to its high rotational speed which makes it more suitable for aerospace applications [1].

To effectively utilize the centrifugal compressor the working characteristic has to be appropriately designed in terms of the air flow, pressure and efficiency. Moreover, the compressor modeling is essential to predict the working performance prior to real applications. However, the accurate prediction of centrifugal compressor working performance is challenging due to the complex aerodynamics and changing environment.

Data fitting methods have been employed to obtain the compressor map based on the input and output data. A static neural network compressor model was designed based on the input–output data fitting in [2]. Surrogate modeling is anther data fitting method to predict the compressor performance [3]. These methods heavily depend on the accuracy of measured data and do not consider the variations of inlet conditions such as the temperature and pressure. In case of environment change the data fitting methods will have discrepancy results. A common approach to estimate the compressor performance is using empirical correlation method which requires details of the compressor geometry [4]. A compressor model was developed using correlation method by experimental data and constructing the map using identification technique [5]. The correlation method has been extensively applied due to its simplicity. However, the correction factors that are defined empirically strongly influence results. A polytropic model was proposed to predict the compressor characteristic by J.M. Schultz [6]. Improved method and corresponding analysis were presented in [7]. Inaccuracies of this method are introduced by neglecting the impact of Mach number and flow coefficient. Focusing on pressure fluctuations within the impeller and diffuser, unsteady Navier–Stokes equations were solved to obtain the gas flow field in the centrifugal compressor [8]. Meanline modeling technique has been used to predict the compressor performance [9]. And the impeller inlet recirculation is considered to improve the accuracy of the pressure and efficiency predictions. M. Casey and C. Robinson proposed to use four nondimensional parameters to characterize the compressor map through algebraic equations [10]. The reference shows that more uncertainty exists in the compressor surge line prediction. And those equations are influenced by the operational conditions. To predict the compressor performance at various operating conditions, an iterative method was proposed by considering

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Nomenclature

β_{1b}	Blade inlet angle	rad
β_{2b}	Blade outlet angle	rad
Δh_f	Enthalpy changes resulting from the friction losses	J
Δh_i	Enthalpy change resulting from the incidence losses	J
Δh_t	Total enthalpy increase	J
Δh_{ideal}	The ideal specific enthalpy	J
η	Compression efficiency	–
κ	Ratio of specific heats	
ω_{cp}	Compressor rotational speed	Pa
σ	Slip factor	–
τ_c	Compressor load torque	N·m
τ_m	Motor driving torque	N·m
A_{cp}	Area of the compressor impeller eye	m^2
c_p	Air specific heat at constant pressure	J/(kg·K)
c_v	Air specific heat at constant volume	J/(kg·K)
C_d	Pressure in the manifold	Pa
g_0	Gravitational acceleration constant	m/s^2
h	Altitude	m
h_0	Height at the bottom of atmospheric layer	m
h_s	Altitude at the bottom of the stratosphere	Pa
J	Inertia of the compressor	$kg{\cdot}m^2$
k_f	Fluid friction constant	$J{\cdot}(s/kg)^2$
L_m	Length of the manifold	m
M_a	Molar mass of the air	kg/mol
M_v	Molar mass of the vapor	kg/mol
m_{cp}	Compressor air mass flow	kg/s
m_{out}	Air mass flow at the outlet of the manifold	kg/s
p	Pressure in the manifold	Pa
p_0	Air pressure at sea level	Pa
p_h	Ambient air pressure	Pa
p_s	Pressure at the bottom of the stratosphere	Pa
p_{cp}	Compressor air pressure	Pa
R	Universal gas constant	J/(mol·K)
r_1	Mean inducer radius	m
r_2	Impeller radius	m
S	Opening area of the nozzle	m^2
T_0	Air temperature at sea level	K
T_h	Ambient air temperature	K
T_s	Temperature at the bottom of the stratosphere	K
V_m	Volume of the manifold	m^3
x_v	Mole fraction of vapor in the air	–
Z	Compressibility factor	–

the suction parameter impact and gas properties impact in [11] and [12], respectively. The method is applicable to various suction gas properties, such as varied temperature, pressure and gas composition. At the iteration process the determination of the parameters like the compressibility factor of different suction gases is difficult.

Another famous centrifugal compressor model was developed by Greitzer et al. [13]. Experimental validation of the model were conducted [14]. In that model the dynamics of the gas pressure and air flow in the manifold are taken into account, and the model could predict the whole operating working performance including the surge and rotating stall phenomenons. Base on this model Gravadahl et al. presented a complete compressor model using state space equations [15]. And the drive torque is a control parameter in that model which could be used for model based controller design [16]. However, variations of the gas property which strongly influence the compressor working characteristics were not taken into consideration.

Different with the references mentioned above this paper considers the altitude effect on the compressor performance. The air conditions including the temperature, air density and pressure at high altitude are very different with that at the sea level. Based on the Gravadahl's model an variable altitude compressor model is developed considering the suction air properties variations with the altitude change. Meanwhile, the effects of variations of crucial parameters on the compressor performance are studied. Moreover, dynamics of air flow and pressure of the compressor during the altitude change are presented and analyzed.

Appropriate control of the compressor/turbocharger is important for some applications, such as aero engine combustion and fuel cell [17,18]. Governor control of gas turbine was designed including a steady flow controller and transient flow controller in [19]. By linearizing of the system model predictive control was used to control the turbocharger of a diesel engine [20]. However, the variation of the inlet air condition, for example caused by the altitude variation, is not considered. Sliding mode controller is capable to deal with the nonlinearity and disturbance of the system by forcing the sliding mode surface to zero. A first order sliding mode controller was developed to control an open-ended thermoacoustic system, in which a sign control function is appropriately designed based on the sliding manifold [21]. In this paper, a second order sliding mode controller is adopted to eliminate the oscillation on the sliding manifold by adding an integral term in the controller.

The structure of this paper is arranged as follows: Section 2 presents the modeling of centrifugal compressor in high altitude, Section 3 gives the compressor performance modeling at different altitudes, with the analysis of the parameter variation effects. Meanwhile the sliding mode control performance is compared with that of a PI controller. Section 4 concludes this paper.

2. System modeling

2.1. Centrifugal compressor

The centrifugal compressor transforms the kinetic energy from high rotational speed of the air into the increase of the pressure. The characteristic of the centrifugal compressor can be described by the compressor map which gives the relationship of the air pressure p_{cp}, mass flow m_{cp} and rotational speed ω_{cp}:

$$p_{cp}(\omega_{cp}, m_{cp}) = p_h \cdot \left(1 + \frac{\eta(\omega_{cp}, m_{cp})\Delta h_{ideal}}{T_h \cdot c_p}\right)^{\frac{\kappa}{\kappa-1}} \quad (1)$$

where p_h is the ambient air pressure at the inlet of the compressor, $\eta(\omega_{cp}, m_{cp})$ is the compression efficiency affected by ω_{cp} and m_{cp}. Δh_{ideal} is the ideal specific enthalpy delivered to the fluid, T_h is the ambient air temperature, c_p is the air specific heat at constant pressure, c_v is the specific heat at constant volume, the ratio of specific heats $\kappa = c_p/c_v$. During compression the losses such as the incidence loss and friction loss result in the energy degradation. The actual enthalpy increase of the air is expressed as follows:

$$\eta(\omega, m)\Delta h_{ideal} = \Delta h_t - \Delta h_i - \Delta h_f - \Delta h_{oth} \quad (2)$$

where Δh_t is the total enthalpy increase, Δh_i and Δh_f are the enthalpy changes resulting from the incidence losses and the friction losses, respectively, Δh_{oth} is the sum of other losses, such as the clearance losses, backflow losses, leakage losses, etc. [15]. The following expressions could be used to calculate the fluid enthalpy changes [16]

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