Energy Conversion and Management 154 (2017) 380–386

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Semi-physical modeling and control of a centrifugal compressor for the air feeding of a PEM fuel cell

Dongdong Zhao[a,*], Liangcai Xu[a], Yigeng Huangfu[a], Manfeng Dou[a], Jianxing Liu[b]

[a] School of Automation, Northwestern Polytechnical University, China
[b] School of Astronautics, Harbin Institute of Technology, China

A R T I C L E I N F O

Keywords:
Compressor
Dynamic performance
Parameter identification
PEM fuel cell

A B S T R A C T

Air compressor which is to used to deliver the air (including about 21% oxygen) to the cathode channel for electrochemical reaction is a crucial component for the polymer electrolyte membrane (PEM) fuel cell. The working characteristics of the compressor greatly influences the fuel cell performance as well as the durability. In this paper a semi-physical modeling method is adopted to analyze the operating property of a centrifugal compressor which is more suitable for automotive fuel cells because of its compactness. This model includes many physical and empirical parameters which are very difficult to determine. Interior-point optimization method based on Newton iteration is used to identify those parameters. The result shows that the modeled compressor map has a good agreement with the experimental data. Meanwhile, the compressor efficiency is analyzed and compared with the measurement to further validate the developed model. This compressor is thus adapted to a validated 10 kW fuel cell model. A dynamic feedforward controller is proposed based on the load torque to control the air mass flow, eliminating the disturbance produced by the compressor load in transient. The simulation results show that this compressor could satisfy requirements of the fuel cell under dynamic load situations while keeping both the fuel cell and compressor with high efficiencies.

1. Introduction

Polymer electrolyte membrane (PEM) fuel cells utilize the electrochemical reaction of the hydrogen and oxygen to produce electricity with the water as byproduct. Because of its low temperature working condition and high efficiency the PEM fuel cell has drawn extensive studies especially for automotive applications [1,2]. Nowadays, most of the famous auto companies e.g., Ford, GM motors, Toyota, etc., are developing their fuel cell vehicles. In 2015, Toyota released the first commercialized fuel cell vehicle named *Mirai* which is a great inspiration for the researchers [3].

For the PEM fuel cell system many auxiliaries such as the air compressor, humidifier and cooler, are needed to fulfill the implementation [4]. As a key component in the fuel cell system the air compressor's characteristics greatly influence the performance and durability of the fuel cell, especially during dynamic load situations. The dynamics of the supplied mass flow and pressure by the compressor have to satisfy requirements of the fuel cell [5,6]. In addition the efficiency of the air compressor has to be as high as possible to reduce the energy consumption. Studies show that the air compressor is the biggest parasitic energy consumption component which could use up to 15% of the fuel cell generated power. Moreover the weight and volume of the compressor have to be reduced for automotive applications [7].

Many kinds of compressors such as scroll, lobe, screw and centrifugal, have been studied for the fuel cell applications [8–10]. For examples a scroll compressor driven by an induction motor was employed in [11] and reference [12] proposed to use a twin-screw compressor for high-pressure fuel cells. Compared with positive displacement compressors mentioned above the centrifugal compressor have superiorities in terms of compactness and efficiency. Moreover the centrifugal compressor transforming kinetic energy to the pressure could produce continuous air flow with a high pressure. However the centrifugal compressor has limited operating range because of the surge phenomenon with the oscillations of the pressure and mass flow. The surge phenomenon occurs as the compressor operates to the low mass flow rate with high pressure. In the previous work a reference limiter was used to limit operation to the right of the surge line thereby avoiding compressor surge [9]. By controlling the driving torque active surge control for centrifugal compressor was proposed to prohibit the surge when operating that unstable region [13–16]. To keep a high efficiency the centrifugal compressor has to operate in a high pressure without surge.

* Corresponding author.
E-mail address: zhaodong@nwpu.edu.cn (D. Zhao).

https://doi.org/10.1016/j.enconman.2017.11.030
Received 1 August 2017; Received in revised form 7 November 2017; Accepted 11 November 2017
Available online 20 November 2017
0196-8904/

The working characteristic of the centrifugal compressor is influenced by many factors such as the geometric parameters and the operating circumstance. For fuel cell applications the compressor parameters have to be determined according to the requirements regarding of the mass flow, pressure and efficiency. The insufficient mass flow rate may lead to oxygen starvation which severely reduces the life time of the stack [17,18]. While, the pressure fluctuations accelerate the aging of the membrane. Therefore, the compressor modeling and parameter identification are significant for investigating its working performance applied for the fuel cell system. In [19], the compressor was modeled using Neural network method and the mass flow and pressure were controlled by decoupled controllers for fuel cells. However, the Neural network model is like a black box without knowing the internal parameters. That model could not be used to optimize the compressor parameters. Moreover, neural network model is highly nonlinear which could not be used for the model based controller design. M. Casey et al. used a set of parameterized algebraic equations to describe the compressor characteristics with little knowledge of the geometry [20]. Whereas, the compressor losses such as the impeller losses and diffuser losses were not considered. One of the most famous centrifugal compressor model was developed by Greitzer, et al. in [21,22]. This model used the state space equations to predict the dynamics of the mass flow, pressure and compressor speed. In [23,16] the model was further developed in which many physical parameters were used. And based on aerodynamics the working characteristics of the compressor could be acquired. Whereas, to determine the values of the parameters in the model is still a difficult task because of the inaccurate measurement or the confidentiality. Using typical profile parameters reference [24] captured geometric variations of the measured blade for numerical simulation, and the parameters were identified through reconstruction. It focused only on the blade and the working property of the compressor in the whole operating range were not tested. In [25], stability parameter was determined based on an approximate realization algorithm through testing the step response of the system. That method uses cubic polynomial to approximate the compressor characteristics and it is especially used for the compressor surge characterization. Computational fluid dynamics (CFD) modeling method based on the physical model could provide the fluid flow dynamics by iteratively calculating amount of algebraic equations. CFD method is usually performed by particular software, such as Solidworks, and it is a time-consuming task. This paper focuses on developing a control oriented compressor model which needs fast calculation obtaining the characteristics of the compressor. Therefore, the semi-physical Moore-Greitzer model is employed.

In this paper the centrifugal compressor model is presented based on the geometric and empirical parameters, which shows highly nonlinearity and parameter coupling. In order to accurately characterize the centrifugal compressor interior-point optimization method is used to determine those parameters. Through properly designing the objective function this method is able to effectively solve the nonlinear optimization problem with parametric constraints. The compressor map determined by the optimized parameters is compared with the experimental data which shows a good agreement. Furthermore, the efficiency of the compressor is analyzed by considering different kinds of losses. The results show that the calculated efficiency is comparable with the measurement. Then this compressor is applied to a validated 10 kW PEM fuel cell system model with dynamic load situations. The fuel cell stack model is designed and validated for the air supply verification which is performed under varied fuel cell current. The dynamics of the air mass flow and pressure which greatly influence the fuel cell performance as well as the safety could satisfy requirements. To control the air mass flow a feedforward controller is proposed which could remove the disturbance of the load changing. Meanwhile how the air dynamics affect the fuel cell's behaviors, i.e., voltage and power are observed and analyzed.

2. Centrifugal compressor model

2.1. Dynamic model

The model of Moore-Greitzer is a classical method to model the centrifugal compressor with the plenum and throttle. In that model the dynamics of both the mass flow and pressure of the centrifugal compressor are presented. The compressor surge phenomenon which is an unstable situation could be exhibited as well. In order to take the speed dynamics into account the compressor model was further developed in [16,23]. In this paper, the centrifugal compressor is driven by a permanent synchronous motor. The driving torque could be used to regulate the compressor speed. The dynamics of the compressor model are shown as follows:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{a_0^2}{V_p}(m-k_t\sqrt{p-p_0}) \tag{1}$$

$$\frac{\mathrm{d}m}{\mathrm{d}t} = \frac{A_1}{L_c}(\Psi(\omega,m)p_0-p) \tag{2}$$

$$\frac{\mathrm{d}\omega}{\mathrm{d}t} = \frac{1}{J}(\tau_m-\tau_c) \tag{3}$$

where p is the pressure in the plenum, m is the compressor mass flow, ω is the rotational speed of the motor, a_0 is the inlet stagnation sonic velocity, V_p is the plenum volume, A_1 is the area of the impeller eye, L_c is the length of the duct, J is the total inertia of the compressor, τ_m is the torque produced by the driving motor, τ_c is the torque from the compressor. $\Psi(\omega,m)$ is the pressure ratio (compared with the inlet air pressure) at the outlet of the compressor which is related to the mass flow and compressor speed. Taking the losses into consideration $\Psi(\omega,m)$ describes the working characteristics of the compressor. The steady state relationships of the $\Psi(\omega,m)$,ω and p form the compressor map. From the above Eqs. (1)–(3) we can see that the difficulty to describe the centrifugal compressor dynamics is to obtain the mathematical expression of $\Psi(\omega,m)$.

2.2. Compressor map

The centrifugal compressor map represents the relationships of the $\Psi(\omega,m)$,ω,p. Ignoring the compressor downstream manifold the air is ideally transferred in isotropic manner from the inlet to the outlet followed by the increase of the pressure. In that compression process losses mainly including incidence losses and friction losses are inevitably produced resulting in the entropy increase as follows:

$$\Psi(\omega,m) = \left(1 + \frac{\eta(\omega,m)\Delta h_{ideal}}{T_0\cdot c_p}\right)^{\frac{\kappa}{\kappa-1}} \tag{4}$$

where $\eta(\omega,m)$ is the compression efficiency, Δh_{ideal} is the ideal specific enthalpy delivered to the fluid, T_0 is the air temperature at the inlet, c_p is the air specific heat at constant pressure, the ratio of specific heats $\kappa = c_p/c_v$,c_v is the specific heat at constant volume. Considering the incidence losses and friction losses the actual enthalpy increase of the fluid can be expressed by the following equation:

$$\eta(\omega,m)\Delta h_{ideal} = \Delta h_t-\Delta h_i-\Delta h_f-\Delta h_{oth} \tag{5}$$

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 and leakage losses, which is assumed to be 8% of the total enthalpy [23]. The following expressions could be used to calculate the fluid enthalpy changes [16]

$$\Delta h_t = \mu r_2^2\omega^2 \tag{6}$$