

Development of a Fundamental Dynamic Model for Centrifugal Compressor Surge Prediction

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1. Introduction

Recently, energy storage systems are increasingly recognized as a pivotal technology in the energy industry due to their ability to enhance grid stability and support the integration of renewable resources. Among these technologies, liquid air energy storage (LAES) has emerged as a promising option due to its potential for large-scale storage capacity and flexibility in integration with renewable energy systems.

However, the operation of LAES inherently involves frequent part-load conditions, as the charging and discharging processes are strongly coupled to the fluctuating availability of renewable energy and dynamic electricity demand. In conventional modeling approaches, turbomachinery performance under off-design operation has been represented primarily through steady-state part-load models. While such simplifications are computationally efficient, they fail to adequately capture the transient and dynamic behavior of machines subjected to rapid load fluctuations. This limitation becomes particularly critical when analyzing the turbomachinery system with compressor, as compressors are highly sensitive to instabilities at reduced mass flow rates. One of the most severe forms of such instability is surge, which manifests as large-amplitude pressure and mass flow rate oscillations. Surge not only deteriorates system performance but also poses a significant risk to the structural integrity of the compressor and the safety of the entire energy storage system.

Given these challenges, this study proposes the development of a dynamic surge model for compressors. Unlike steady-state models, the dynamic surge model is capable of resolving unsteady flow oscillations and predicting the onset of instability under varying operating conditions. The primary objective of this work is to establish a fundamental modeling approach that can be applied to centrifugal compressor under dynamic part-load condition. This research is intended to provide a theoretical foundation for future studies on the dynamic analysis of compressor including surge-condition range.

2. Research background

Part-load operation of turbomachinery introduces complex unsteady flow phenomena that are not adequately described by steady-state models. Previous

research efforts have largely focused on steady performance maps and polynomial-based corrections to approximate compressor and turbine behavior under off-design conditions. While such methods provide useful baseline estimates, they are inherently limited in addressing the transient dynamics and stability boundaries encountered during load fluctuations. This knowledge gap has motivated the exploration of dynamic models that account for time-dependent variations in mass flow, pressure ratio, and rotational dynamics.

For compressors, the risk of surge at low-flow operation is a critical factor in dynamic modeling. Surge is initiated when the compressor operates to the left of its stable operating line in compressor map, leading to alternating flow reversals and pressure oscillations throughout the compression system. The consequences include severe aerodynamic instabilities, mechanical fatigue, and potential damage to close-clearance components. From a system-level perspective, uncontrolled surge can propagate instability into the whole system, thereby undermining both reliability and efficiency. Consequently, predictive modeling of surge is essential for both design optimization and safe operation of LAES plants.

Given these research context, surge prediction become a critical challenge for systems employing compressors. Accordingly, this study aims to suggest a methodology for constructing a predictive system for compressor surge and integrating it with a dynamic compressor model, thereby providing a more accurate and reliable modeling framework.

3. Compressor dynamic surge model

Loss model analysis is a modeling approach that identifies the loss components within the compressor system and uses them to predict the overall work of the compressor. In this paper, loss model analysis is employed to determine operating parameters such as pressure ratio and efficiency under specific operating conditions.

$$\pi_c = F_{\pi(G_c, n_c)}, \quad \eta_{cs} = F_{\eta(G_c, n_c)} \quad (1)$$

$$m_c = m \frac{1.01 \times 10^5 \text{ Pa}}{P_{in}} \sqrt{\frac{T_{in}}{293.15 \text{ K}}} \quad (2)$$

$$n_c = n * \sqrt{\frac{293.15K}{T_{in}}} \quad (3)$$

π_c : Compressor pressure ratio
 η_{cs} : Compressor isentropic efficiency
 m : Mass flow rate n : Compressor RPM
 m_c : Corrected mass flow rate
 n_c : Corrected compressor rpm

In order to evaluate the performance of the compressor, the mass flow rate and rotational speed are first expressed in corrected forms to eliminate the influence of inlet temperature and pressure. This normalization allows the operating parameters to be compared under a consistent reference condition, which is essential for constructing compressor maps. Following the approach presented by Hongsheng Jiang et al. [1], the corrected mass flow rate and corrected rotational speed are employed in subsequent performance correlations (Eq. (1)~(3)).

$$\pi^* = \pi_c^{\frac{\gamma-1}{\gamma}} = 1 + \frac{(\Delta h_{imp} - \Delta h_{int})}{c_p T_1} \quad (4)$$

$$\eta^* = \frac{\left(\pi_c^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_{cs}} = \frac{(\Delta h_{imp} + \Delta h_{par})}{c_p T_1} \quad (5)$$

$$\Delta h_{imp} = \sigma u_2^2 = \sigma (\pi D_2 n)^2 \quad (6)$$

$$u_2 = \pi D_2 n \quad (7)$$

$$\Delta h_{par} = \Delta h_{df} + \Delta h_{rc} + \Delta h_{lk} \quad (8)$$

$$\Delta h_{int} = \Delta h_{inc} + \Delta h_{bl} + \Delta h_{sf} + \Delta h_{cl} + \Delta h_{mix} + \Delta h_{vld} + \Delta h_{vd} \quad (9)$$

Δh : Specific enthalpy σ : Slip factor
 γ : Specific heat ratio

df : disk friction loss

rc : Recirculation loss

lk : Leakage loss

inc : Incidence loss

bl : Blade loading loss

sf : Skin friction loss

cl : clearance loss

mix : Mixing loss

vld : Vaneless diffuser loss

vd : Vaned diffuser loss

The pressure ratio and efficiency of the compressor are determined through loss model analysis, which decomposes the total enthalpy change into various loss mechanisms such as incidence, friction, and leakage. By quantifying these loss terms, the model can predict the deviation of actual performance from the ideal case. This approach enables the reconstruction of performance parameters across a wide range of operating conditions.

3.1 Turbomachinery dynamic characteristics

$$\dot{\omega} = \left(\frac{1}{J}\right) * (\tau_s - \tau_c) \quad (10)$$

$$\Delta h_{ideal} = \frac{\dot{W}_c}{\dot{m}} = \sigma U^2 \quad (11)$$

$\dot{\omega}$: Compressor rpm time derivative

Δh_{ideal} : Ideal fluid enthalpy change

τ_s : Shaft torque τ_c : Compressor torque

J : Spool moment of inertia $\dot{W}_c$: Compressor work

U : Impeller tangential velocity

To capture the dynamic characteristics of the turbomachinery, the model incorporates the torque balance between the driving shaft and the compressor load. The governing equations express the time-dependent variation of rotor speed as a function of applied torque and compressor work. This formulation, adopted from the work of Wei Jiang et al. [2], provides the basis for analyzing unsteady transient responses of the compressor system.

3.2 System component design

<Valve>

$$\Delta p_v = \frac{\dot{m}^2}{K_v^2 \rho} \quad (12)$$

K_v : Valve coefficient ρ : air density

For the surge prediction model, the surrounding components of the compression system must also be represented. The valve model describes the relationship between mass flow rate and pressure drop across the valve, thereby linking boundary conditions to the internal compressor dynamics [3].

<Duct>

$$\left(\frac{d\dot{m}}{dt}\right) = \left(\frac{A}{L}\right) * (p_{in} - \Delta p_D - p_{out}) \quad (13)$$

$$\Delta p_D = f \left(\frac{L}{D}\right) \rho \left(\frac{V^2}{2}\right) \quad (14)$$

L : Duct length A : Duct area f : friction factor
 P : Pressure V : Flow velocity D : Duct diameter

The duct is modeled to capture the momentum dynamics of the flow between components. In particular, the governing equation relates the temporal change in mass flow rate to the pressure difference across the inlet and outlet of duct. This provides a means to evaluate flow inertia effects under varying pressure conditions.

<Plenum>

$$\left(\frac{dp_p}{dt}\right) = \left(\frac{\gamma R_g T_p}{V_p}\right) * (\dot{m}_{in} - \dot{m}_{out}) \quad (15)$$

R_g : Gas constant, T_p : Plenum temperature

V_p : Plenum volume

The plenum is treated as a control volume where pressure gradient occurs according to the imbalance between inlet and outlet mass flows. The governing relation describes how the plenum pressure rises or falls as a function of this mass flow difference. In this system,

plenum plays a role as pressure buffer component under varying pressure conditions.

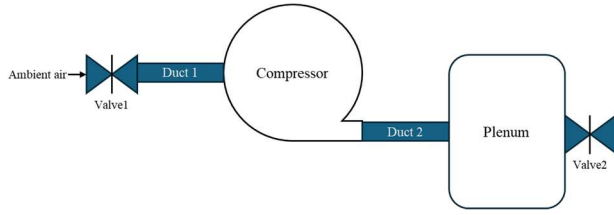


Fig. 1. Compressor surge simulation piping system

The overall system of surge prediction model integrates the compressor with its connected piping network. As illustrated in the accompanying schematic, the system is composed of valves, ducts, and a plenum arranged to reproduce the pressure–flow interactions observed in practice. This layout forms the foundation of the simulation environment used to reproduce surge cycles.

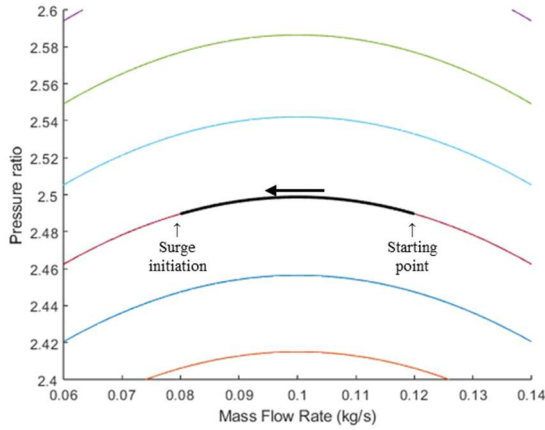


Fig. 2. Compressor performance map with mass flow rate trajectory toward surge initiation point

Figure 2 presents the compressor map generated from the loss model analysis. In the simulation, it is assumed that the operating point is gradually shifted along the characteristic curve by reducing the mass flow rate until the surge boundary is reached. This procedure enables the model to reproduce the transition from stable operation to unstable surge oscillations in a controlled manner.

4. Result and Conclusion

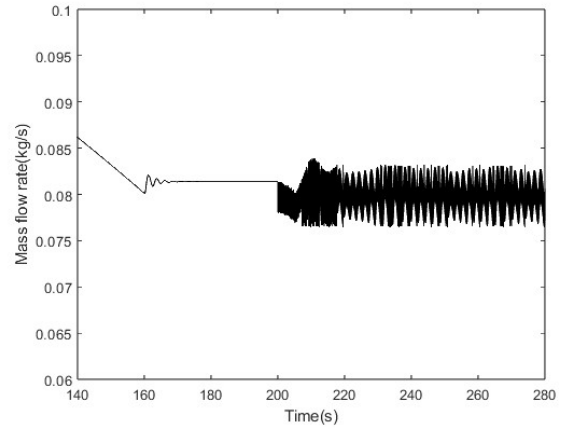


Fig. 3. Compressor mass flow rate during surge conditions

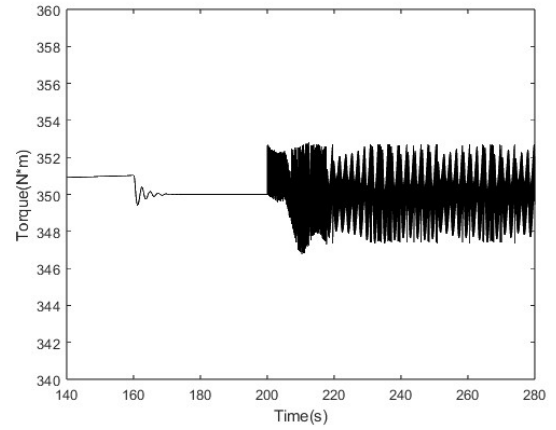


Fig. 4. Compressor torque during surge conditions

The proposed integrated model was applied to simulate the dynamic behavior of the compressor under MATLAB environment. The results demonstrate that the onset of surge can be clearly identified through both mass flow rate and torque fluctuations.

Figure 3 presents the time evolution of the mass flow rate. After approximately 200 s, the system departs from stable operation and periodic oscillations become dominant. This behavior indicates the initiation of surge, characterized by alternating cycles of flow deceleration and recovery driven by the interaction between the compressor and the downstream plenum.

Similarly, Figure 4 shows the corresponding variation of compressor torque. While the torque remains nearly constant during the stable operating period, strong oscillations appear once surge is triggered, closely matching the pattern observed in the mass flow rate. The coupled fluctuations of flow and torque confirm that the model successfully captures the unsteady fluid–mechanical dynamics associated with surge.

At the present stage, the model reproduces surge qualitatively in a simulation environment but has not yet been validated against experimental or high-fidelity numerical data. Nonetheless, the ability to capture flow

and torque oscillations provides a solid foundation for predictive analysis of surge phenomena in turbomachinery. Future work will focus on model validation and on extending the framework to support surge control and avoidance strategies in energy storage system operation.

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