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Dynamic Analysis of Compressor Surge Aggravation under Low-Flow Operating Conditions

Seok Jun Oh, Jeong Ik Lee*
Nuclear Quantum Engineering department, KAIST, Yuseong-gu, Daejeon, 34141
**Corresponding author: jeongiklee@kaist.ac.kr*

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1. Introduction

The increasing penetration of renewable energy sources has significantly altered the structure of power generation systems, as illustrated by the well-known "duck curve" phenomenon (Fig.1). As the share of renewable energy generation increases, temporal imbalances in power output have become more pronounced. To compensate for this variability, flexible energy resources are required, and nuclear power generation is being considered as one of the viable options for providing such operational flexibility.

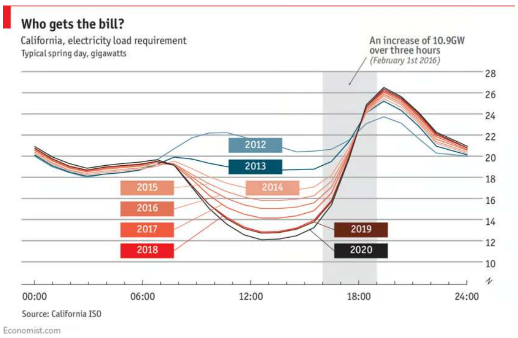

Fig. 1. Duck curve phenomenon of California (2012~2020)

In this context, energy storage systems such as Liquid Air Energy Storage (LAES) have been proposed as one viable approach to enhance the operational flexibility of nuclear power plants. By decoupling power generation from electricity supply, LAES systems enable improved load adjustment and support stable grid operation.

However, due to the inherent operational characteristics of energy storage systems, compressors and associated turbomachinery are required to operate under part-load conditions for a substantial portion of their operating time. Consequently, a detailed investigation of turbomachinery performance and stability under part-load operation is essential.

Understanding surge behavior under off-design and low-flow operating conditions is therefore essential for ensuring the stable integration of LAES with nuclear power systems. In particular, the dynamic response of the compressor as the operating point approaches the surge line requires detailed analysis based on physically consistent modeling.

This study presents the implementation of a dynamic compressor surge model and investigates the evolution of surge characteristics as the compressor mass flow rate decreases. By analyzing the progressive intensification of surge phenomena under part-load conditions, the present work aims to provide fundamental insights into compressor stability limits relevant to LAES-assisted flexible nuclear operation.

2. Compressor dynamic surge model

2.1. Compressor system design

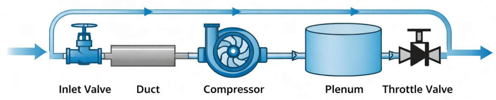

Fig. 2. Compressor system piping structure

To analyze the dynamic characteristics of the compressor, its structure was decomposed into several subdivided components. The system constructed to validate the surge behavior of the compressor is shown in Fig. 2. The inlet mass flow rate is supplied through a valve and a duct located upstream of the compressor, while the downstream flow is regulated by a throttle valve connected to a plenum installed at the compressor outlet.

The governing equations describing each component are presented as follows [1].

<Throttle Valve>

$$m_t = \frac{C_D A_T P_{out}}{\sqrt{RT}} \left(\frac{P_{in}}{P_{out}}\right)^{\frac{1}{k}} \left(\frac{2k}{k-1}\left(1-\frac{P_{in}}{P_{out}}\right)^{\frac{k-1}{k}}\right)$$

$$for \quad \frac{P_{in}}{P_{out}} > \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{1}$$

$$m_t = \frac{C_D A_T P_{out}}{\sqrt{RT}} k^{\frac{1}{2}} \left(\frac{2}{k+1}\right)^{\frac{k+1}{2(k-1)}}$$

$$for \quad \frac{P_{in}}{P_{out}} > \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{2}$$

$$k: Valve\ coefficient\ \ \rho: air\ density$$

The throttle valve is installed between the plenum and the ambient air and regulates the mass flow rate by utilizing the pressure difference between the plenum and the atmosphere.

Equation (1) describes the mass flow rate under subcritical (non-choked) conditions, whereas Equation (2) represents the mass flow rate under choked flow conditions.

By adjusting the throttle valve opening ratio, the outlet mass flow rate of the compressor system can be controlled.

<Duct>

$$\left(\frac{d\dot{m}}{dt}\right) = \left(\frac{A}{L}\right) * (p_{in} - \Delta p_D - p_{out}) \tag{3}$$

$$\Delta p_D = f\left(\frac{L}{D}\right)\rho\left(\frac{V^2}{2}\right) \tag{4}$$

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

The duct serves to attenuate abrupt variations in the inlet mass flow upstream of the valve. The governing equation determines the transient mass flow rate within the duct based on the pressure difference between the upstream and downstream sections, as well as the frictional losses along the duct. Through this formulation, flow inertia is introduced, enabling the mass flow rate to respond dynamically to pressure variations at the compressor inlet.

<Plenum>

$$\left(\frac{dp_P}{dt}\right) = \left(\frac{\gamma R_g T_P}{V_P}\right) * (\dot{m}_{in} - \dot{m}_{out}) \tag{5}$$

$$R_g: Gas\ constant, T_P: Plenum\ temperaturer$$
$$V_P: Plenum\ volume$$

The plenum is located downstream of the compressor and functions as a buffer component that prevents abrupt fluctuations in mass flow at the compressor outlet. The pressure within the plenum varies according to the difference between the inlet and outlet mass flow rates, and this pressure dynamics plays a crucial role in compressor surge behavior.

$$\dot{m}_c = \frac{A}{L}\left(P_{comp\ out} - P_P\right)\Delta t \tag{6}$$

The compressor flow dynamics are affected by the pressure interaction between the compressor discharge and the plenum. In the presence of back pressure, where the plenum pressure becomes higher than the compressor outlet pressure, the resulting adverse pressure gradient induces a reduction in compressor mass flow rate. This behavior is incorporated into the model to capture the onset and development of flow instability.

2.2. Turbomachinery dynamics

$$\pi_c = F_{\pi(G_c,n_c)},\ \ \eta_{cs} = F_{\eta(G_c,n_c)} \tag{7}$$

$$m_c = m\,\frac{1.01 * 10^5 Pa}{P_{in}}\sqrt{\frac{T_{in}}{293.15K}} \tag{8}$$

$$n_c = n * \sqrt{\frac{293.15K}{T_{in}}} \tag{9}$$

$$\pi_c: Compressor\ pressure\ ratio$$
$$\eta_{cs}: Compressor\ isentropic\ efficincy$$
$$m: Mass\ flow\ rate\ n: Compressor\ RPM$$
$$m_c: Corrected\ mass\ flow\ rate$$
$$n_c: Corrected\ compressor\ rpm$$

To assess compressor performance, the mass flow rate and rotational speed are initially converted into corrected quantities to remove the effects of inlet temperature and pressure variations. This normalization ensures that operating conditions can be evaluated on a common reference basis, which is necessary for the development of reliable compressor maps. In accordance with the methodology proposed by Jiang et al. [2], the corrected mass flow rate and corrected rotational speed are subsequently utilized in the performance correlations (Eqs. (7)–(9)).

$$\pi^* = \pi_c^{\frac{\gamma-1}{\gamma}} = 1 + \frac{(\Delta h_{imp} - \Delta h_{int})}{c_p T_1} \tag{10}$$

$$\eta^* = \frac{\left(\pi_c^{\frac{\gamma-1}{\gamma}} - 1\right)}{\eta_{cs}} = \frac{(\Delta h_{imp} + \Delta h_{par})}{c_p T_1} \tag{11}$$

$$\Delta h_{imp} = \sigma\, u_2^2 = \sigma\,(\pi\, D_2 n)^2 \tag{12}$$

$$u_2 = \pi\, D_2 n \tag{13}$$

$$\Delta h_{par} = \Delta h_{df} + \Delta h_{rc} + \Delta h_{lk} \tag{14}$$

$$\Delta h_{int} = \Delta h_{inc} + \Delta h_{bl} + \Delta h_{sf} + \Delta h_{cl} + \Delta h_{mix} + \Delta h_{vld} + \Delta h_{vd} \tag{15}$$

$$\Delta h: Specific\ enthalpy\ \ \sigma: Slip\ factor$$
$$\gamma: Specific\ heat\ ratio$$

$$df: disk\ friction\ loss$$
$$rc: Recirculation\ loss$$
$$lk: Leakage\ loss$$
$$inc: Incidence\ loss$$
$$bl: Blade\ loading\ loss$$
$$sf: Skin\ friction\ loss$$
$$cl: clearance\ loss$$
$$mix: Mixing\ loss$$
$$vld: Vaneless\ diffuser\ loss$$
$$vd: Vaned\ diffuser\ loss$$

The compressor pressure ratio and efficiency are evaluated using a loss-based analytical framework in which the overall enthalpy change is separated into individual loss components, including incidence loss, friction loss, and leakage loss. By estimating the contribution of each loss mechanism, the model captures the departure of real compressor performance from the ideal isentropic behavior. This methodology allows for the systematic reconstruction and prediction of performance characteristics over a broad range of operating conditions.

$$\dot{\omega} = \left(\frac{1}{J}\right) * (\tau_S - \tau_c) \tag{16}$$

$$\Delta h_{ideal} = \frac{\dot{W}_c}{\dot{m}} = \sigma U^2 \tag{17}$$

$\dot{\omega}$: *Compressor rpm time derivative*
Δh_{ideal}: *Ideal fluid enthalpy change*
τ_S: *Shaft torque* τ_c: *Compressor torque*
J: *Spool moment of interia* W_c: *Compressor work*
U: *Impeller tangential velocity*

To represent the dynamic behavior of the turbomachinery, the model accounts for the torque equilibrium between the driving shaft and the aerodynamic load imposed by the compressor. The governing equations describe the transient evolution of rotor speed as a function of the net torque resulting from shaft input and compressor work demand. This formulation, based on the approach proposed by Wei Jiang et al. [3], establishes the theoretical framework for evaluating the unsteady and time-dependent response of the compressor system.

3. Result and Conclusion

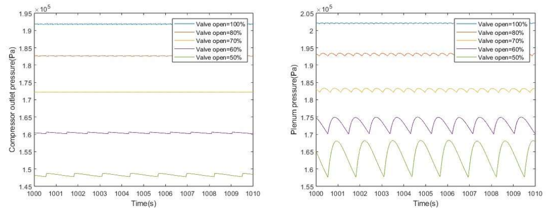

Fig. 3. Pressure response under varying valve opening rates

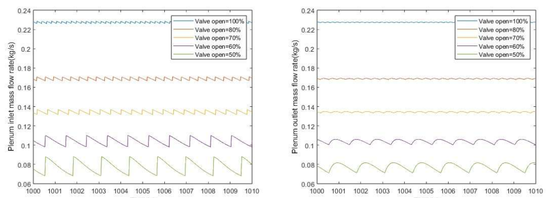

Fig. 4. Mass flow rate response under varying valve opening rates

Figures 3 and 4 illustrate the progressive intensification of surge as the compressor mass flow rate decreases. As the throttle opening is gradually reduced, the operating point shifts toward the low-flow region of the compressor characteristic curve. In this region, the slope of the compressor characteristic becomes unfavorable for stable operation, and the system becomes increasingly sensitive to small disturbances. The results show that once the operating point approaches the instability boundary, oscillations in mass flow rate and plenum pressure begin to amplify. With further reduction in mass flow, these oscillations evolve into fully developed surge cycles characterized by large-amplitude, periodic fluctuations. At 50% valve opening condition, mass flow rate at the outlet of plenum oscillates between 0.07~0.08kg/s and pressure between 158~167kPa with surge frequency of 0.8Hz.

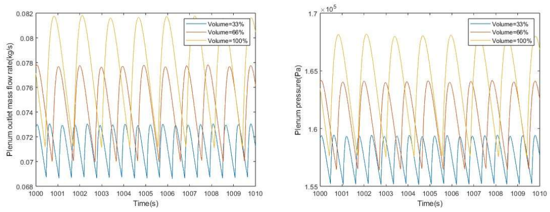

Fig. 5. Variation of surge oscillation magnitude with changing plenum volume

Figure 5 presents the influence of plenum volume on surge oscillation characteristics. By systematically increasing the plenum volume under identical operating conditions, it is observed that the magnitude of surge oscillations increases as the volume becomes larger. The larger volume allows greater pressure accumulation before flow reversal occurs due to higher energy concentration rate, leading to more pronounced excursions in both plenum pressure and compressor mass flow rate during each surge cycle. Consequently, the oscillatory behavior exhibits increased amplitude with increasing plenum volume. This result indicates that the dynamic energy storage capacity of the plenum plays a significant role in determining the severity of surge oscillations, and that larger plenum volumes can intensify the nonlinear surge response under unstable operating conditions.

In conclusion, the present results clearly demonstrate two fundamental mechanisms governing compressor surge dynamics. First, progressive reduction in mass flow shifts the operating point into an unstable regime, where disturbances grow into large-scale oscillations. Second, the plenum volume significantly influences the severity of surge by modifying the system's dynamic buffering capability. A larger plenum intensifies pressure–flow coupling and increases oscillation magnitude.

These findings emphasize that both operating conditions and system configuration parameters must be carefully considered in surge analysis and control strategies. In particular, the adjustment and optimization of plenum volume emerges as an important design variable in surge mitigation. Appropriate selection of plenum size is therefore critical for controlling surge characteristics of system.

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