Effect of Pulsating Flow on Surge Frequency of a Turbocharger Compressor

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Abstract. Turbocharger compressors operate under pulsating flow due to continuous opening and closing of engine valves placed downstream of the compressor but exhibit a hysteresis loop deviating from steady characteristics. As the flow in a compressor is throttled, both pressure and flow discharge unstably oscillate. The instability is known as surge and it affects not only the compressor itself but also its whole engine-turbocharger system. For estimating the stable operation range, partial flow tests of a turbocharger compressor under pulsating flow were conducted. Using a compressor test apparatus with a pulsating flow generator and a tank whose capacity can be changed, the flow rate was controlled by a valve. Under steady flow, surge hysteresis loops caused by oscillating pressure and flow rate were obtained in each rotational speed. By Fast Fourier Transform (FFT) of pressure signals, the occurrence of surge with different frequencies was confirmed depending on the tank capacity. Next in pulsating flow tests, a compressor rotational speed is fixed, and pulsating frequencies given by the pulsating flow generator are changed. The obtained surge frequencies were different from those in steady flow tests. Especially, it is likely to be drawn to the frequency that is equal, two times and a half rate of the pulsating frequency. Numerical computation with one-dimensional calculation is conducted to investigate such a phenomenon.

1.Introduction

Turbocharger compressors aren’t operated under steady flow but under pulsating flow due to continuous opening and closing of engine valves. The unsteady efficiency is lower than the steady one and the loss depends on the pulsating frequency [1]. The compressor operating points exhibit a hysteresis loop. As the mass flow is reduced, it becomes unstable and the flow instabilities can be two types, rotating stall and surge. Stenning discussed the properties of the rotating stall that induce large vibratory stresses in the blading of compressor and surge that is intolerable for system operation [2]. Greitzer developed a nonlinear model to predict the transient response of an axial compressor, and its response is dependent on the parameter B [3]. With a given compressor, there is a critical value of B that causes the large amplitude oscillations to happen. Surge is a system oscillation caused by the interaction of the compressor and piping system characteristics and a simplified lumped parameter model was applied to the prediction of the surge line to reflect the characteristics [4]. Especially, the main effect on the surge margin was found to be due to the presence of a volume [5]. On the other hand, some authors [6,7] demonstrated the unsteady nonlinear 1-D solver using an engine system to predict surge physics of
However, there were few investigations of surge frequency effected by the unsteady flow.

In this study, to predict the stable operation of the centrifugal compressor, partial flow tests of a turbocharger compressor under pulsating flow were performed. In addition, to simulate the behavior, a one-dimensional calculation with AMEsim was conducted.

2. Experimental Setup

2.1 Test Apparatus

The centrifugal compressor test facility is shown in Figure 1. The blade number of the impeller is 10 and that of the diffuser is 12. The impeller outlet diameter is 75.34 mm. Air flows through inlet pipe before the compressor. After it leaves the compressor, it encounters a pulsating flow generator. Then air encounters a tank whose capacity can be changed, leading to a valve at its exit. Figure 2 shows the pulsating flow generator and the tank. This device contains a rotating disk and a stationary disk. There are two types of rotating disks with different hole shapes. The pulsating frequency is set depending on the rotational speed of the rotating disk. This device can reproduce the waveform of the pulsating flow in actual engine. The parameters of the apparatus are shown in Table 1. The static pressures at the tank wall, compressor inlet and compressor outlet were measured by a pressure sensor and the velocity at compressor inlet was measured by a hot wire probe. The flow rate is calculated by the velocity and the inlet section area. All data was recorded by LabVIEW via the PXI system.
2.2 Test conditions

On the design point, the unsteady tests of the compressor are carried out with two types of the rotating disk of the pulsating flow generator to confirm the pulsating effect on the compressor.

Surge tests under steady flow are carried out. The rotational speed of the impeller was changed from 18000 rpm to 3000 rpm with every 1000 rpm interval. The flow rate is manually throttled from the design point to partial flow by a valve, and its transient response is measured.

To calculate surge frequencies by oscillating tank pressure in a transient response at a valve point, Fast Fourier Transform (FFT) is conducted. The Formula of discrete Fourier transform is defined as

\[ F(t) = \sum_{x=0}^{N-1} f(x) \exp \left( -\frac{2\pi tx}{N_s} i \right) \]  

where \( N_s \) is the sample number, which is 4096 in this experiment. The sampling frequency is 409.6 Hz, and it means one measurement is carried out for 10 seconds.

Next, the surge tests under pulsating flow are carried out. The rotational speed of the impeller is fixed to 18000 rpm at each tank size and the pulsating frequencies are changed from 1 Hz to 6 Hz with every 0.33 Hz. This is determined by matching the Strouhal number to actual engine set-ups. The Strouhal number is defined as below.

\[ St = \frac{f_{pulsating}}{(N/60)} \]  

FFT is also conducted as in steady tests.

2.3 Numerical simulation

The numerical simulation of the surge is carried out using a commercial one-dimensional modeling and simulation software AMESim. The model was made as shown in Figure 3 and its analysis conditions are in Table 2. An algebraic loop that converts flow rate given by a flowmeter into total pressure at the compressor outlet is contained instead of using a compressor model. The conversion is based on the approximate cubic curve that is obtained by a compressor steady performance test when the rotational speed is 18000 rpm. The two butterfly valves are used to simulate the channel cross-section change by the pulsating flow generator upstream and the flow control valve that throttles the flow rate downstream. The parameters of the tank and pipes are the same as the test apparatus. Pulsating frequency can be changed at the valve component in this model.

In Figure 3, each k shows a temperature and a valve angle based on the experiment. \( Q \) [m³/s] shows the flow rate. Cp is a simple pneumatic chamber for tank model. FFT is also conducted to the oscillating tank pressure obtained in the simulation.

| Table 1. Model parameters |
|---------------------------|
|                          | Tank volume \( V_p \) [L] | Pipe length \( L_c \) [m] | Acoustic velocity \( a \) [m/s] | Inlet cross-section area \( A_c \) [m²] |
| Large                    | 283.6                      | 3.303                      | 346.5                       | 0.002572                 |
| Medium                   | 212.2                      |                            |                            |                         |
| Small                    | 154                        |                            |                            |                         |

![Figure 3. AMESim model](image-url)


### Table 2

| Temperature at the inlet [K] | Rotational speed [rpm] | Condition at the outlet     |
|-----------------------------|------------------------|-----------------------------|
| 293                         | 18000                  | Atmosphere open             |

3. Result and Discussion

3.1 Influence of pulsating flow on compressor performance

The waveform of the flow rate at the compressor inlet with two different types of rotating disks of pulsating flow generator is shown in Figure 4. The pulsating frequency is 5Hz. The amplitude of the mass flow fluctuation with No.2 rotating disk is larger than that with No.1. Figure 5 shows the relation between the flow coefficient $\phi$ and the pressure coefficient $\psi$. The compressor is operated on the red points under a steady flow. However, under pulsating flow, the operating points describe the hysteresis loops. The loop with the No.2 rotating disk that causes the larger amplitude of mass flow fluctuation is wider than the other one. As a result, the effect of pulsating flow with No.2 is found to be more significant.

![Figure 4. The waveform of the flow rate](image)

![Figure 5. $\phi$ - $\psi$ curve](image)

3.2 Surge test result without pulsation

As mentioned above, Greitzer established quantitative criteria about the occurrence of surge as follows [3]. The oscillation on the compression system is modeled as that of Helmholtz resonator. The Helmholtz frequency, $f$, and B parameter are defined as

$$f = \frac{a}{2\pi} \sqrt{\frac{A_c}{V_p L_c}}$$

(3)

$$B = \frac{U}{2a} \sqrt{\frac{V_p}{A_c L_c}}$$

(4)

Where $U$ is peripheral velocity.

Transient responses of flow rate and tank pressure with a large tank are described at two rotational speed of compressor in Figure 6. These responses were shown by throttling outlet valve as much as possible. Both flow rate and pressure are oscillating after throttling valve at 18000 rpm. Similar oscillations were obtained from 18000 to 8000 rpm with every 1000 rpm interval. On the other hand, they converge to another constant value at 7000 rpm. Similar convergences were obtained below 7000 rpm. When the initial oscillation grows to large amplitude, it is the surge [3]. By the similar waveform in Figure 6 and sounds in the experiment, the occurrence of the surge was confirmed. But at 7000 rpm the surge was not confirmed. In addition, the parameter $B$ is 0.592 at 18000 rpm, and it is 0.230 at 7000 rpm. Surge is likely to occur as parameter $B$ becomes larger.
Figure 6. Transient response

Similar changes of transient responses were obtained in the 9000 rpm with the medium tank and in the 10000 rpm with the small one.

Figure 7 shows the relationship between the flow rate and the total pressure ratio of several rotational speeds of the compressor using the large tank. Surge line is defined by drawing a line on each point where the mild surge is almost likely to occur, so the flow rate doesn’t go into negative flow region. On each rotational speed, the hysteresis loop is also shown when the surge occurs. It is found that the hysteresis loop becomes wider as the rotational speed becomes larger. With other tanks, similar hysteresis loops are confirmed.

Figure 7. Surge line and hysteresis loops

Figure 8 shows the results of the FFT analysis on each tank size to calculate the surge frequency in a steady flow. The rotational speed of the compressor is 18000 rpm. For each tank, Table 3 shows the
surge frequency by the experiment and theoretical Helmholtz frequency calculated by equation 3. The dimensionless frequency \( f^* \) is also shown. The \( f^* \) is defined as below.

\[
f^* = \frac{f_{\text{surge}}}{f_{\text{surge, tank large}}}
\]

(5)

![Figure 8. FFT results under steady flow](image)

**Table 3. surge frequencies of each tank**

|                              | Experimental frequency \( f_{\text{surge}} \) [Hz] | Dimensionless frequency \( f^* \) [-] | Theoretical Helmholtz frequency \( f_{\text{surge}} \) [Hz] | Dimensionless frequency \( f^* \) [-] |
|------------------------------|-----------------------------------------------|-------------------------------------|-----------------------------------------------|-------------------------------------|
| tank_large                   | 2.3                                           | (1.0)                               | 2.9                                           | (1.0)                               |
| tank_medium                  | 2.8                                           | 1.22                                | 3.3                                           | 1.16                                |
| tank_small                   | 3.2                                           | 1.39                                | 3.9                                           | 1.36                                |

The experimental frequencies don’t correspond to Helmholtz frequencies of each tank. However, dimensionless frequencies of each tank are very close to each other and it can be said that the effect of \( V_p \) in equation 3 is enough to be seen in the experiments.

### 3.3 Surge test result with pulsation

Figure 9 shows FFT results of oscillating tank pressure of small tank under pulsating flow whose frequency is 2, 4, and 6 Hz. It shows three patterns; steady(without pulsation), with rotational disk No.1, and No.2. The surge frequencies under pulsating flow are different from those under steady flow. As shown in (a) and (b), the frequency approaches the pulsating frequency. The frequency with No.2 is more likely to approach than that with No.1, as the effect of pulsating flow with No.2 is larger than that with No.1. Especially, when the given frequency is 4 Hz, the surge frequency with No.2 is drawn into 4 Hz. By the way, the power spectrums under pulsating flow are likely to be smaller than those under steady flow. To consider this suppression effect, it seems more experiments should be done.

![Figure 9. FFT results under pulsating flow](image)
Next, plots in figure 10 show the relation between the pulsating frequency and the surge frequency of each tank. The broken lines show the surge frequencies under steady flow (without pulsation). All plots with each tank are around each steady surge frequency. The dotted lines show linear functions; $y = x, y = 2x,$ and $y = \frac{x}{2}$. For example, plots on the line of $y = x$ mean that the surge frequency coincides with the pulsating frequency. Some points of each tank are on those three lines. The surge frequency under pulsating flow is likely to be the frequency which is equal, 2 times or a half rate of the pulsating frequency. It is confirmed that the surge frequency is drawn into the pulsating frequency.

**Figure 9.** Results of FFT

**Figure 10.** The relation between the pulsating frequency and the surge frequency
3.4 Numerical simulation result

Figure 11 shows the relation between time and the tank pressure with medium tank simulated by AMESim. The oscillation of tank pressure means the occurrence of the surge. The flow rate also oscillate. The results of the FFT of each tank are shown in figure 12. It is found that surge frequency is changed in each tank the same as the results of the experiments. However, these values don’t correspond to experimental values because boundary conditions of compressor in value transmissions in AMESim is not sure to be same as the experiment.

![Figure 11. Tank pressure oscillation](image1)

![Figure 12. Results of FFT of AMESim simulation](image2)

Figure 13 shows the relation between the pulsating frequency and the surge frequency of the medium tank as in the experiments. The dotted lines mean the same as figure 10. On each line, the surge frequency is also drawn into the same as the experimental results. In addition, as the same result were obtained under pulsating flow of sine waves with the same amplitude, this phenomenon is mostly based on not the wave from but the pulsating frequency.

![Figure 13. The relation between the pulsating frequency and the surge frequency by AMESim simulation](image3)

4. Conclusion

The results of the investigation of the effect of the pulsating flow on surge frequency of a turbocharger compressor lead to the following conclusion.
Surge frequency changes depending on the tank size.
Surge frequency differs under pulsating flow from under steady flow.
Surge frequency is likely to be drawn to the frequencies that are equal, two times and a half rate of the pulsating frequency.
Surge frequency is more likely to be drawn under pulsating flow with larger amplitude.
The one-dimensional analysis simulated the surge and showed similar phenomena to the results of the experiments and it is confirmed that this phenomenon is not based on the waveform of pulsating flow.
Such lock-in phenomenon can be predicted to happen in a certain operating condition in a real engine-turbocargar system.

Nomenclature

\[
\begin{align*}
\rho & : \text{density} \quad (\text{kg/m}^3) \\
Q & : \text{flow rate} \quad (\text{m}^3/\text{s}) \\
\tau & : \text{torque} \quad (\text{Nm}) \\
f_{\text{pulsating}} & : \text{pulsating frequency} \quad (\text{Hz}) \\
P_t & : \text{total pressure} \quad (\text{Pa}) \\
U & : \text{rotational velocity} \quad (\text{m/s}) \\
N & : \text{rotational speed} \quad (\text{min}^{-1}) \\
D & : \text{impeller diameter} \quad (\text{m}) \\
b & : \text{impeller outlet height} \quad (\text{m}) \\
\omega & : \text{angular speed} \quad (\text{rad/s}) \\
\phi & : \text{flow coefficient} \\
\psi & : \text{total pressure coefficient} \\
\phi & = \frac{Q}{\pi DbU} \\
\psi & = \frac{P_{t2} - P_{t1}}{\rho U^2}
\end{align*}
\]

Subscripts:
1: inlet
2: outlet

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