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Stability Evaluation of Gas Pressure Regulator by Frequency Response Test

Tomoaki TAKEUCHI[1*], Ryoichi TORIUMI[1], Toshiharu KAGAWA[2]
[1]Pipeline Technology Center, Tokyo Gas, Yokohama, Japan
[2] Department of Mechano-Micro Engineering, Tokyo Institute of Technology, Yokohama, Japan
*Corresponding author, E-mail: take1002@tokyo-gas.co.jp

ABSTRACT

The quantitative evaluation of stability in gas pressure regulators for natural gas distribution is very important. A conventional step response test is widely used to evaluate the stability of regulators. However, the quantitative stability evaluation is often difficult and only qualitative processes based on experience are performed. We therefore propose the application of a frequency response test to gas pressure regulators. The frequency transfer function derived from the test results enables the quantitative stability evaluation of regulators by a phase margin, which is often used as an index of stability in control engineering. For the frequency response test, we fabricated the test system with a quick response servo valve unit for inputting arbitrary sinusoidal wave flow rates. To verify the practical applicability of the proposed frequency response method, we conducted several experiments using actual regulators. The results showed that the stabilities of the regulators could be quantitatively evaluated by their phase margins.

1. Natural gas distribution and gas pressure regulators

Natural gas is distributed to individual points of consumption, with the pressure decreased in stages. The devices used for controlling the pressure of the gas supplied at each pressure reduction stage are the gas pressure regulators. Any instability phenomenon, such as hunting of the regulator, can trigger serious problems. Therefore, quantitative stability evaluation of regulators is extremely important to safely supplying natural gas.

1.1 Operating principle of regulator

Figure 1 shows the schematic of unloading type pilot-operated regulator with rubber-sleeve-type main valve, which is main objective of this study. Pilot-operated regulators are controlled by two control valves. One is the pilot valve that directly senses the downstream pressure P_2 and the other is main valve that controls the flow rate in the main line, using the control pressure P_c, which is amplified in the pilot line. The pilot line output the control pressure P_c which corresponds to the pressure loss at the restriction[(1)].

Figure 2 shows a block diagram that summarizes the control of a system that combines a regulator with a downstream pipeline volume (regulator system). A disturbance is caused by the discharging flow rate from the downstream pipeline volume V_2, which is equivalent to the demand flow rate Q_2 of the regulator system. The controller $K(s)$ is a regulator that outputs send-out flow rate Q_G according to the difference between the set pressure and the downstream pressure ($P_{set} - P_2$). The controlled object $P(s)$ is the downstream volume V_2, which outputs the downstream pressure P_2 by integrating ($Q_G - Q_2$). The regulator is a self-driven control system that consists of mechanical elements in this manner.

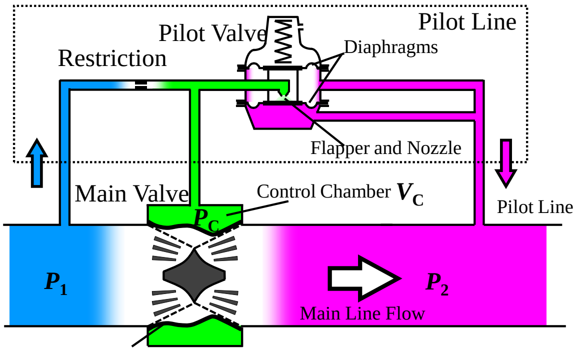

Figure 1. Schematic of pilot-operated regulator with rubber sleeve type main valve

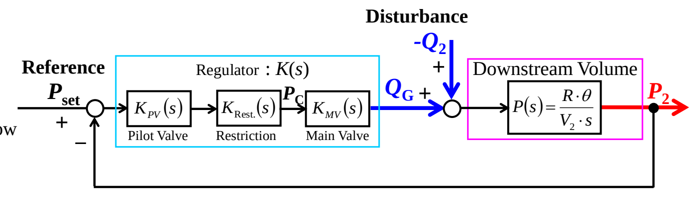

Figure 2. Block diagram of regulator system

1.2 Stability evaluation of regulator

If a transfer function can be derived from the transient response characteristics of the regulator, it will be possible to quantitatively estimate the stability through such means as the root locus method.

The transient response characteristics can be evaluated by developing a mathematical model of all the regulator elements[2–5]. However, it is not easy to build precise model owing to the variety of mechanical structures in the regulator. Moreover, there is the potential problem that parameter measurement error and modeling approximation error are accumulated.

A conventional step response test is available as a method for easily acquiring the transient response characteristics of a regulator system. However, quantitative evaluation of the stability is often difficult and only qualitative processes based on experience are performed, such as determination of the presence of any oscillation phenomenon.

Hence, a testing method that can be used for the quantitative stability evaluation of a regulator system, which is simpler to implement than building a mathematical model, is being required.

1.3 Transient response measurement using frequency response method

Therefore, the objective of this study was to quantitatively evaluate the stability of regulator systems not by building mathematical models of the various constituent components of the regulator but by applying the frequency response method to these regulator systems.

The application of the frequency response method to regulators had been proposed in the past[6], however unlike this study, previous studies did not evaluate the regulator system that includes a downstream pipeline. Furthermore, there has been no report concerning an effective validation conducted by applying the method to an actual regulator.

This study verifies whether or not it is possible to quantitatively evaluate stability of a regulator by building test equipment to measure the frequency response of a regulator and applying it to an actual regulator.

2. Test system and test equipment

The test system for the frequency response test of a regulator, which was verified by this study, is shown in Fig. 3. A servo valve unit was placed at the most downstream position to produce the downstream demand flow rate Q_2, which is the input, and a pitot-type flowmeter was placed immediately above the regulator for measuring the regulator send-out flow rate Q_G, which is the output. Compressed air was used as the test fluid.

The response characteristics were measured in terms of the gain g ($=A_O/A_I$) and the phase ϕ. The relationship of the measured input and output in this case was based on the transfer function $G_{closed}(s)$, which was a closed loop as expressed by Eq. 1.

$$Q_G = \frac{-P(s)\cdot K(s)}{1+P(s)\cdot K(s)}\cdot(-Q_2),\quad G_{Closed} = \frac{-P(s)\cdot K(s)}{1+P(s)\cdot K(s)} \quad \cdots(1)$$

The stability of the system can be quantitatively clarified based on the transfer function obtained experimentally. And the frequency response method can, in principle, be applied to regulators with any structure and any control method.

The servo valve unit and the pitot-type flowmeter used in the tests are described in further detail below.

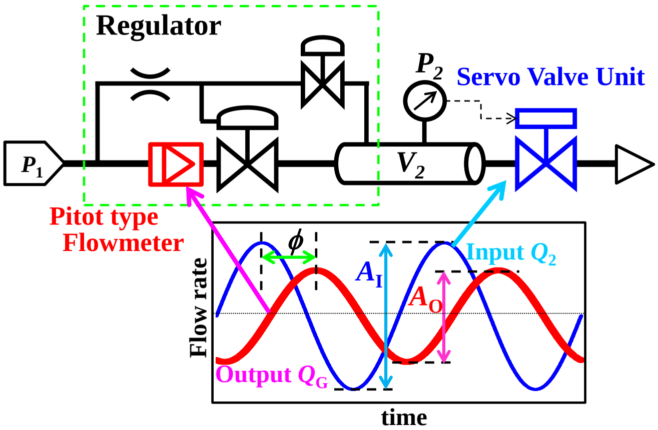

Figure 3. Schematic diagram of frequency response test

2.1 Servo valve unit and pitot-type flowmeter

A servo valve unit having 20 servo valves (MPYE-5-3/8-010-B, FESTO, Germany) connected in parallel was fabricated in order to provide the input of arbitrary sinusoidal wave flow rates for the frequency response test. A schematic diagram of a unit containing the manifold linkages is shown in Fig. 4.

The transient flow rate of the servo valve unit was controlled by the feed-forward control using the sonic conductance C of the servo valve unit, as shown in Fig. 5. The sampling rate of P_2 was 1.0 μs and the value was obtained by averaging 1000 points. The downstream side of the unit was open to the atmosphere and therefore maintained a constant atmospheric pressure. The relationship between the input voltage U of the servo valve unit and the sonic conductance C was derived by measuring the output flow rate Q_2 m^3/h(nor), the upstream and downstream pressures of the unit P_x, P_y Pa(abs) and the test fluid temperature θ K, as shown on Eq. (2). Standard condition air density ρ_{ANR}=1.185 kg/m^3, normal condition air densityρ_0=1.293 kg/m^3, and the critical pressure ratio b in Eq. 2[7]. Furthermore, the maximum response frequency of a single unit of the servo valve was 65 Hz, and the maximum flow rate at the upstream pressure of the servo valve unit of 150 kPaG and the downstream pressure equal to the atmospheric pressure was approximately 1,000 m^3/h(nor).

$$Q_2 = \begin{cases} C(U)P_x \rho_{ANR}\sqrt{\dfrac{293}{\theta}}\sqrt{1-\left(\dfrac{P_y/P_x - b}{1-b}\right)^2}\dfrac{3600}{\rho_0} \cdots \left(\dfrac{P_y}{P_x} \geq b\right) \\ C(U)P_x \rho_{ANR}\sqrt{\dfrac{293}{\theta}}\dfrac{3600}{\rho_0} \cdots \left(\dfrac{P_y}{P_x} \leq b\right) \end{cases} \quad \cdots(2)$$

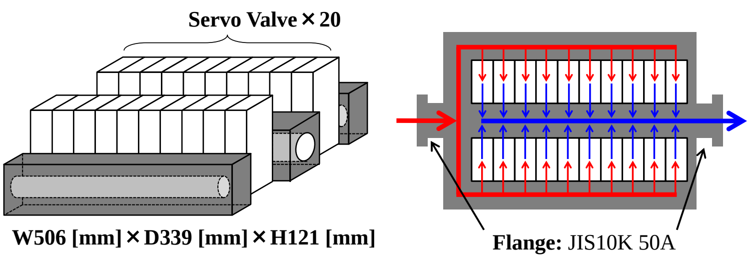

Figure 4. Schematic diagram of servo valve unit

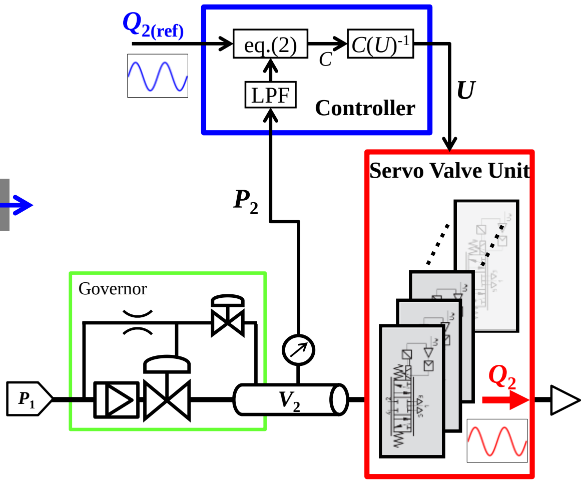

Figure 5. Block diagram of servo valve unit

A pitot-type flowmeter (200 Barflow Tube, JMS Inc., Japan) with internal diameter of 52.7 mm was used for the measurement of the output flow rate of the regulator. The pressure difference in the pitot tube was measured by pressure difference sensor (KL17, Nagano Keiki, Japan).

2.2 Response characteristics of test equipment

In order to derive the frequency response characteristics of a regulator by using the servo valve unit and a pitot-type flowmeter, the maximum measurable frequency using the relevant test equipment must be verified. Therefore, the frequency response characteristics of this equipment was examined by connecting the servo valve unit directly to a pitot-type flowmeter with a short pipe(inner diameter=52.7mm, length=300mm). The sinusoidal flow rate of 600±200 m3/h(nor) at the servo valve unit was the input, flow rate measured by the pitot-type flowmeter was the output.

The results are shown in Fig. 6. The gain was approximately 0 dB, and hardly any phase delay was observed in the low-frequency region of approximately 4 Hz or lower. The gain increased by 0.05 dB, and the phase was delayed by 1.98° at 4.0 Hz in comparison with 0.033 Hz. Once the frequency became larger than 4 Hz, slight increases in the gain and the phase delays started to be observed, and the increase in the gain and the phase delay became more extensive in the high-frequency region of 10 Hz or higher. Therefore, the maximum frequency that can be evaluated using our test equipment was determined to be 4.0 Hz.

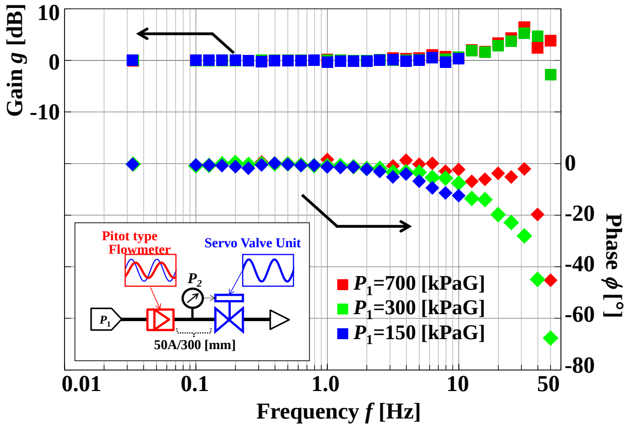

Figure 6. Bode diagram of frequency response test system

3. Stability evaluation by frequency response test

The procedure for acquiring the stability by applying the test system built for the study to actual regulator systems, through the use of the frequency response method, was verified.

The pilot-operated regulator with a rubber-sleeve-type main valve (nominal diameter: 2 inch) shown in Fig. 1 was used for the test. The diameter d at the restriction of the pilot line was 1.0 mm, and the volume of the control chamber V_c was 110.5 ml. The upstream pressure was 700 kPaG, and the set pressure for the regulator was 150 kPaG. The internal diameter of the downstream pipeline from the regulator to the servo valve unit was 304.7 mm; this pipeline had a length of 16.2 m and a piping volume V_2 of 1.18 m^3. The input flow rate Q_2 was a sinusoidal wave of 600±200 m^3/h(nor), in the standard operating flow rate region of this regulator. The measurement items were the servo valve unit transient flow rate Q_2, which was the input; the regulator send-out flow rate Q_G, which was the output; and the downstream pressure P_2.

Data were measured for ten cycles after the response behavior of the system became settled, because the response of the regulator was not stable immediately after the flow rate was input. This measurement was taken repeatedly, by changing the input frequency. Furthermore, the amplitude of the input and the output flow rate A and the phase χ were derived by a harmonic analysis of the measurement results obtained at the test frequency, using Eq. 3. The notation N in Eq. 3 represents the amount of data for ten cycles, and n represents the amount of data for one cycle.

$$\begin{cases} A = \sqrt{a_1^2 + b_1^2} \\ \tan\chi = \dfrac{b_1}{a_1} \end{cases} \quad \text{where, } a_1 = \frac{2}{N}\sum_{i=1}^{N} Q_i \cdot \cos\frac{i \cdot 2\pi}{n}, \quad b_1 = \frac{2}{N}\sum_{i=1}^{N} Q_i \cdot \sin\frac{i \cdot 2\pi}{n} \qquad \cdots(3)$$

Figure 7(a) indicates the measurement results of the output flow rate and the downstream pressure with respect to the input of a relatively low frequency, 0.316 Hz. It is evident that the regulator was tracking the gradual demand change of the 31.6 s cycle considerably well. Fig. 7(b) shows the results for 0.316 Hz. A phase delay was observed and the gain increased. Fig. 7(c) shows the measurement results for 0.794 Hz, that is the resonance peak frequency. Fig. 7(d) shows the results for 3.16 Hz, which was a higher frequency than the resonance frequency. The attenuation was significant and the phase was delayed by almost 180°. Fig. 8 shows a summary of all these in a Bode diagram.

Furthermore, the frequency transfer function for the open loop $G_{open}(j\omega) = -P \cdot K$ was derived from the gain g_{closed} and phase ϕ_{closed} on the basis of the frequency transfer function of the closed loop as expressed by Eq. 4. The phase margin could also be derived from the frequency transfer function for the open loop. Therefore, it is possible to quantitatively evaluate the stability of regulator systems as a phase margin.

$$\begin{cases} G_{\text{open}}(j\omega) = -P \cdot K = \dfrac{\alpha^2 + \alpha + \beta^2}{(1+\alpha)^2 + \beta^2} + \dfrac{\beta}{(1+\alpha)^2 + \beta^2} \cdot j \\ G_{\text{closed}}(j\omega) = \dfrac{-P \cdot K}{1 + P \cdot K} = \alpha + j\beta \end{cases} \qquad \cdots(4)$$

$$\text{where, } \alpha = g_{\text{closed}} \cdot \cos\phi_{\text{closed}}, \quad \beta = g_{\text{closed}} \cdot \sin\phi_{\text{closed}}$$

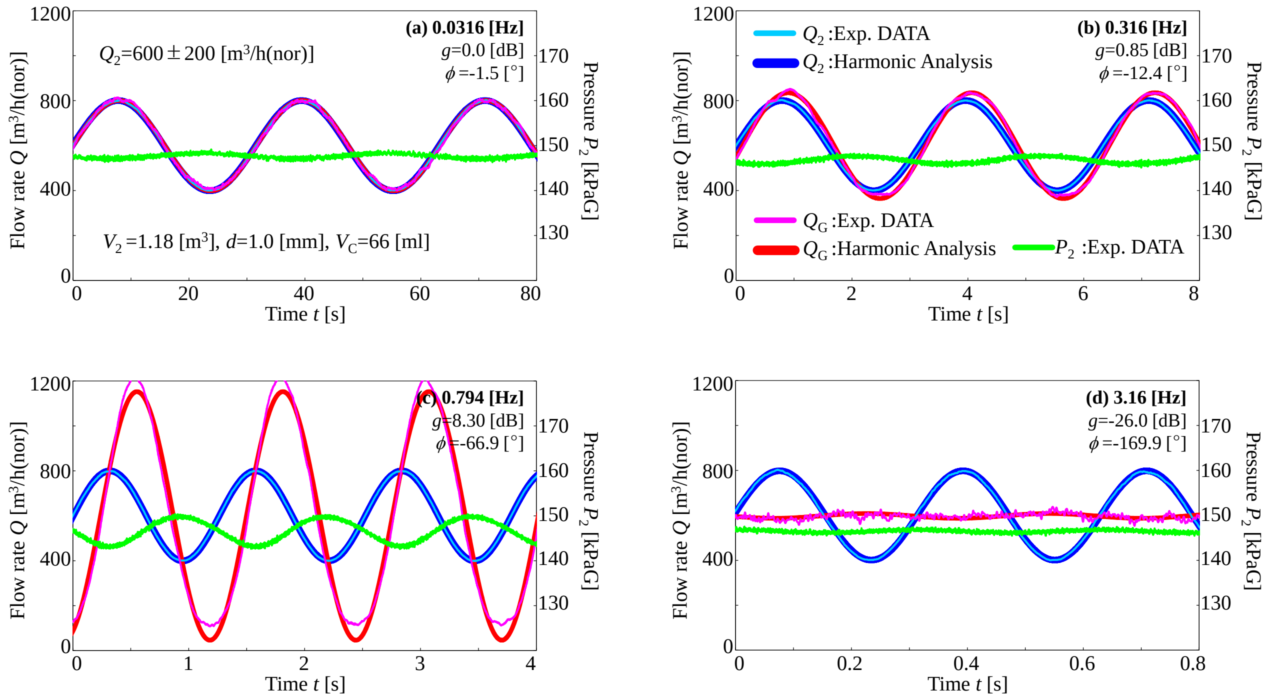

Figure 7. Input and output waves of frequency response

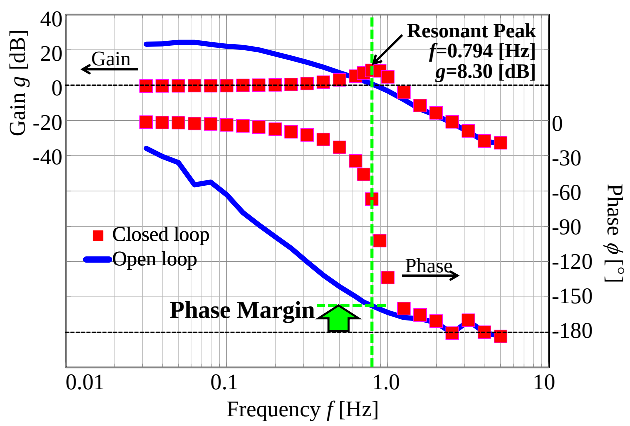

Figure 8. Bode diagram of frequency response test

4. Verification of impact of nonlinearity

An example for the application of the frequency response method to an actual regulator system was discussed in the previous section. However, in cases where the regulator system has nonlinear characteristics, the phase margin derived by the frequency response method evaluates the stability in particular regions where a linear approximation can be established. It is also necessary to select appropriate input waveforms to ensure that no nonlinear characteristics are present.

The impact of nonlinearity characteristics such as saturation or non-proportionality, as well as hysteresis, is examined to consider the limits for the evaluation of stability.

4.1 Impact of saturation

Because no backflow can occur in a regulator system, the saturation of the output flow rate occurs when the amplitude of the input flow rate is increased. For instance, in the case of the input flow rate of $Q \pm \Delta Q$= 600±100 m^3/h(nor), no saturation occurred even around the peak as shown in Fig. 9(a), but when the input amplitude was increased to 200 m^3/h(nor), an intermittent behavior of temporarily choking and then rapidly sending-out was repeated as shown in Fig. 9(b), causing distortions in the frequency response waveform. Therefore, it is necessary to select small values for the amplitude of the input flow rate, which do not cause saturation.

The gain never exceeded five times in the case of the representative regulator systems available on the market, as described in the following section. We believe that this implies that it is possible to avoid saturation, by keeping the flow rate amplitude to 1/5 or lower than the standard flow rate (ΔQ < 1/5Q).

4.2 Impact of non-proportionality

Main valves and pilot valves with proportional characteristics are used in most cases with regulators, but flow characteristics never indicate complete proportional behavior over the entire range.

The difference in response behaviors is observed in the input flow region because of such effects, as shown in Fig. 10. Therefore, it is evident that a quantitative comparison of stability using the frequency response method should be conducted within the same flow region.

4.3 Impact of hysteresis

The main valve of the regulators often has hysteresis caused by diaphragms and sleeves made of rubber or hysteresis arising from sliding resistance. Therefore, the size of the minor loop changes according to the amplitude of the input flow rate. Fig. 11 shows the gain of the flow rate output Q_G with respect to the input of the control pressure P_C at the main valve, derived from static testing as well as frequency response testing with changing the input amplitude. It is evident that a quantitative comparison of stability should be performed with the same input flow rate amplitude in order to avoid the effects of the change in the gain caused by such hysteresis.

The frequency response characteristics are changed by the input flow region or the flow rate amplitude in this manner because of the nonlinear characteristics of the regulator system. Therefore, it should be noted that the stability derived by using the frequency response method is not universally established for all flow regions, but stability is evaluated only in the vicinity of the input flow region.

This property that enables the evaluation of characteristics only in the input flow region is the same for the step response tests conducted in the past, and therefore, the frequency response test, which is capable of comparing the stability of multiple regulator systems given the same input flow region and amplitude, is still considered quite significant.

Although flow regions for the purpose of comparing stability can be set arbitrarily, the step response tests of regulator systems are in general conducted by targeting the standard flow rate region; therefore, the stability of the regulator system described in the subsequent sections, was derived on the basis of the phase margin in the standerd flow rate region.

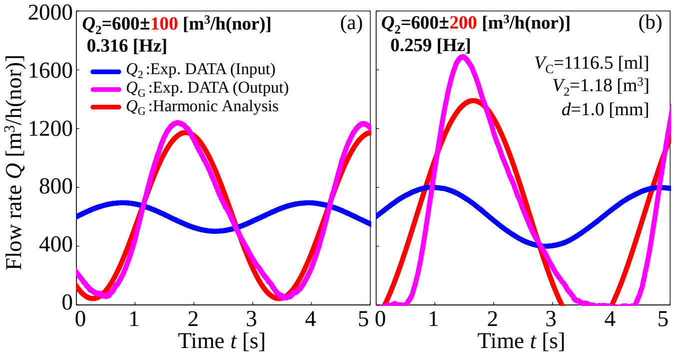

Figure 9. Change in output wave for different input flow rates

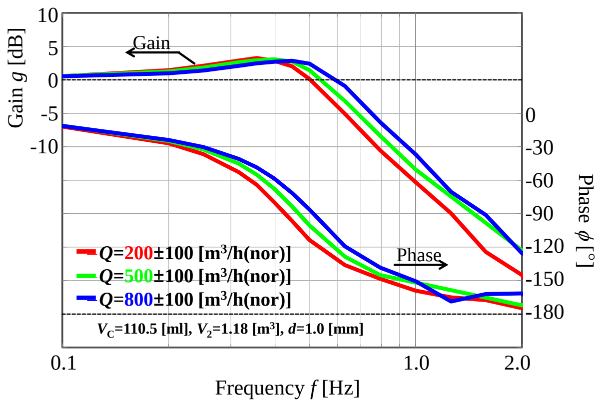

Figure 10. Bode diagram for various input flow rates

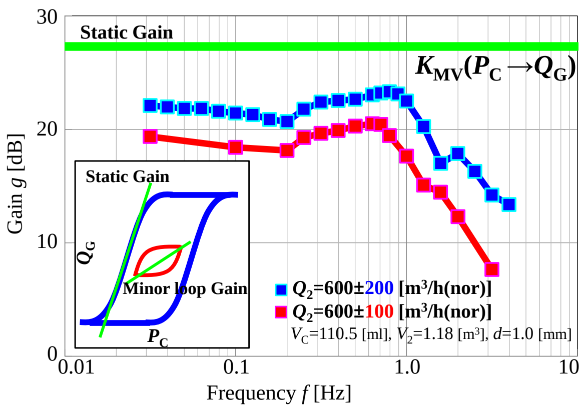

Figure 11. Frequency response gain of the main valve

5. Verification of applicability to regulators and effectiveness

In order to verify the validity of the frequency response method described thus far for practical purposes, the downstream volume V_2 and the restriction diameter d, as well as the control chamber volume V_C, which are the principal control parameters of the regulator system shown in Fig. 1, were changed as shown in Fig. 12, in order to measure the change in the frequency response and to evaluate the stability on the basis of the phase margin. The frequency response method was applied to the three representative types of regulators that are available on the market in order to evaluate the stability of regulator systems that have varying main valve structures and control systems.

5.1 Case involving change in downstream volume

It is a known fact that the system becomes more stable with increasing the downstream volume V_2 because the change in the downstream pressure P_2 decreases with respect to the change in the demand flow rate Q_2. Therefore, the stability for a case where the downstream volume was changed was examined using the frequency response method. More specifically, the pipeline length of the downstream volume, consisting of a piping with an internal diameter of 304.7 mm, was changed to 16.2, 46.1, and 122.1 m.

The Bode diagram of the test results are shown in Fig. 13(a); the resonance characteristics and the phase margin are summarized in Table 1(a). The resonance frequency shifted towards the lower frequency side as the downstream volume V_2 increased, and the peak gain decreased. Furthermore, the phase margin increased when the downstream volume increased, which evidently led to a relatively stable system.

5.2 Case involving change in diameter of restriction

The internal diameter d of the restriction in the pilot line was changed next. A larger internal diameter at the restriction is known to indicate a more stable behavior because it lowers the gain of the control pressure P_C with respect to the change in the downstream pressure P_2.

The test results are shown as a Bode diagram in Fig. 13(b) and Table 1(b). It was revealed that the relationship between the restriction diameter and stability, which has been known empirically, can be evaluated quantitatively in terms of the phase margin.

5.3 Case involving change in control chamber volume

Furthermore, a test involving a change in the volume V_C of the control chamber was performed. The change in the control pressure P_C takes longer at a larger volume of the control chamber. It is a known fact that this leads to a larger response delay of the regulator and therefore, instability. On the other hand, intentionally setting the volume V_C of the control chamber to a large value is a practice that is conducted from experience, in order to avoid hunching at higher frequencies.

The test results are shown as a Bode diagram in Fig. 13(c) and listed in Table 1(c). It is evident that an increase in V_C leads to the transition of the resonance frequency to the lower frequency side. Furthermore, it is possible to indicate the relationship between the control chamber volume and stability, which has been known from experience, in a quantitative manner on the basis of the change in the phase margin. Furthermore, in the test where the value of V_C was changed, the amplitude was set to ±100 m^3/h(nor) because saturation was observed with the output waveform when the input flow rate was set to 600±200 m^3/h(nor).

In this manner, it became evident that the impact of the change in the constituent components of a regulator on the stability of the regulator, which was known only through experience, could be evaluated quantitatively by applying the frequency response method to the regulator and without building any mathematical model.

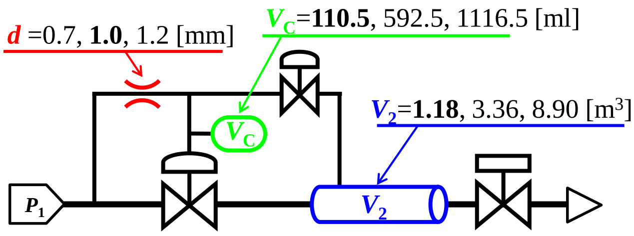

Figure 12. Schematic diagram of tested regulator systems

Table 1. Results of frequency response test in variety of V_2, d, V_C

	(a) V_2 [m^3]			(b) d [mm]			(c) V_C [ml]		
	1.18	3.36	8.90	0.7	1.0	1.2	110.5	592.5	1116.5
Phase Margin [°]	21.5	39.3	58.5	12.8	21.5	29.9	25.8	6.99	3.68
Resonant Frequency [Hz]	0.794	0.398	0.158	0.596	0.794	0.794	0.708	0.447	0.316
Resonant Peak Gain [dB]	8.30	3.46	0.588	10.9	8.30	5.67	6.86	15.98	15.21

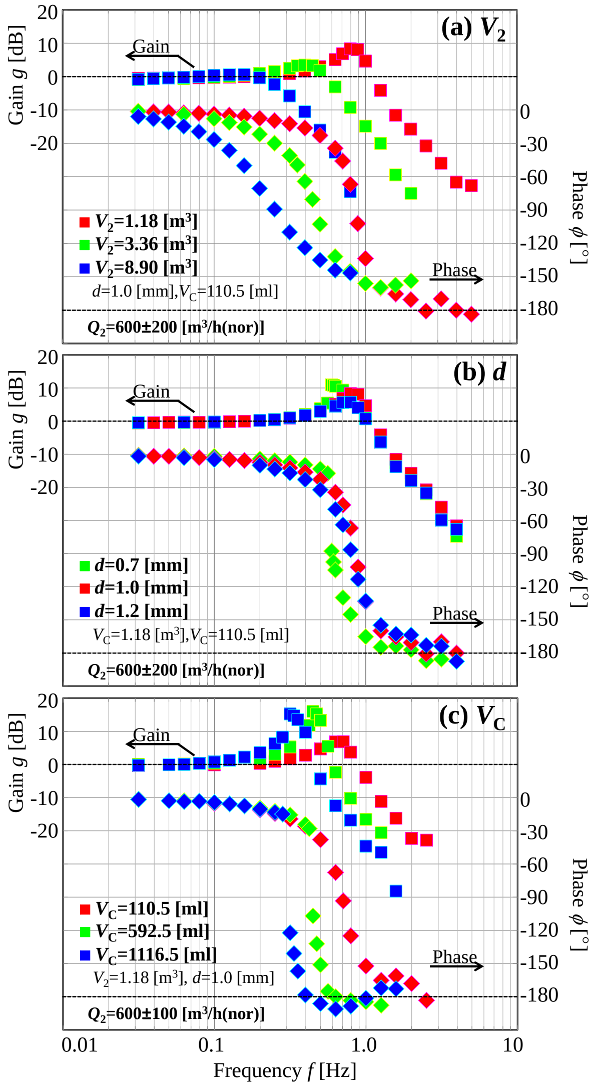

Figure 13. Bode diagram divided from frequency response test of regulators in variety of V_2, d, V_C

5.4 Regulators with different structures and control systems

In order to verify that the frequency response method presented by this study can be applied to regulators irrespective of their structures or control systems, tests were conducted on three types of regulators with representative types of main valve structures and control systems that are available on the market. The first was a regulator with a rubber-sleeve-type main valve shown in Fig. 1; the second was a regulator with a diaphragm-type main valve shown in Fig. 14(a), the third was a regulator with a plug-type main valve shown in Fig. 14(b). All these regulators had pilot valves with varying structures and different application methods for the control pressure. The regulators with rubber-sleeve-type and diaphragm-type main valves used the unloading-type method, whereas the regulator with the plug-type valve used the loading-type method. In addition, all types of regulators had a nominal diameter of 2 inch and a standard flow rate region of approximately 600 m^3/h(nor).

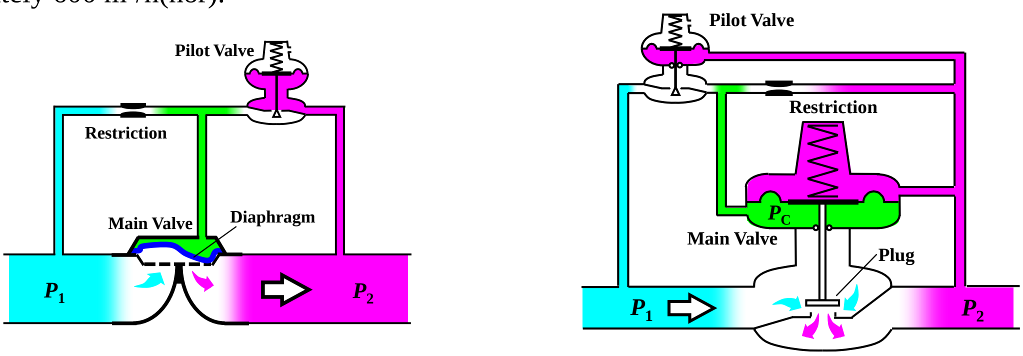

Figure 14. Schematic of loading type pilot operated regulator with diaphragm type main valve(a), unloading type pilot operated regulator with plug type main valve(b)

The results of the test are shown as a Bode diagram in Fig. 15. It was evident that stability can be compared quantitatively, even between regulator systems that have different structures and control systems, as demonstrated by the fact that the phase margin of the regulator with the plug-type valve was 15.0°, whereas that of the regulator with the rubber-sleeve-type valve was 25.8° and that of the regulator with the diaphragm-type valve was 50.3°.

Furthermore, it was verified that no significant distortion of the output waveform occurred because of non-proportionality or hysteresis, when the flow rate amplitude that allowed for the avoidance of saturation, as described in Section 4, was used.

Although regulator systems have nonlinear characteristics, given test conditions with the same input flow region and amplitude, it is still possible to quantitatively evaluate stability in applicable regions of multiple systems by using the frequency response method.

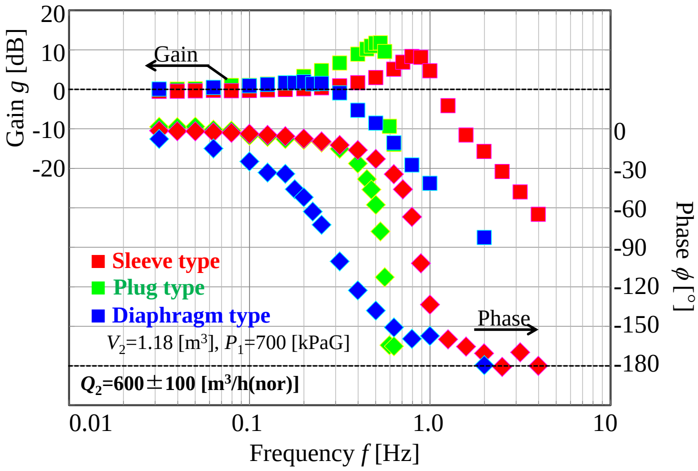

Figure 15. Bode diagram divided from frequency response test of three different types of regulators

6. Conclusion

The application of the frequency response method was considered, with the objective of conducting a quantitative stability evaluation of gas pressure regulators used for distributing natural gas. A summary of the obtained results is as follows:

1) The test equipment was built in which the regulator unit's downstream demand flow rate was input and the regulator unit's send-out flow rate was output. And it was used for verifying that the frequency response characteristics of actual regulator systems can be obtained.

2) A test involving changes in the control parameters for the constituent components of a regulator system using a rubber-sleeve-type valve and tests involving three types of regulators with varying structures were conducted; they verified that given the same input flow waveform, the frequency response method can be used for the stability evaluation of the regulators in a quantitative manner.

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