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Investigation Results for Surge Phenomena of Centrifugal Compressor related with Piping Arrangements and Test Conditions.

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ABSTRACT

The shop performance test of a hydrogen recycle gas compressor for refinery plant was conducted with the suction pressure of 50barA based on ASME PTC-10 Type-2 test.

During testing, a surge phenomenon was observed at a flow rate 15 % higher than the predicted surge flow. From the FFT results of the shaft vibration and the pressure pulsation at the discharge of the compressor, the peaks of sub-synchronous components were observed at two frequencies of 2Hz and 30Hz near surge flow. The sub-synchronous vibration at 30Hz was considered to be caused by rotating stall, but the 2Hz was too low to be attributable to rotating stall. As an assumption of the root cause, the acoustic characteristics of shop test loop were checked. The calculated and measured Helmholtz resonance frequency of the shop test loop was in agreement with the sub-synchronous frequency of 2Hz.

For further investigation, additional tests were carried out with a fully instrumented test loop. Many different test cases were carried out by changing the suction pressure and piping size, with and without installing a perforated plate near the compressor. The originating flow points of the surge were different in each case. The originating flow point of the surge was small when the suction pressure was high and/or the perforated plate was installed. From the investigation results, it was found that the mode of the acoustic characteristics of test loop was related with the originating flow point of surge. In case the compressor discharge is the node of pressure pulsation, it becomes the anti-node of flow fluctuation. In this case, if the flow approaches surge, the flow fluctuation at the compressor discharge is increased. It transmits to the compressor suction, and it would cause the surge of the compressor. In case the perforated plate was installed near the compressor discharge, the pressure pulsation at the compressor discharge was increased. It decreased the flow fluctuation of the compressor discharge, and the surge became suppressed.

INTRODUCTION

Regarding compressor performance testing, ASME

PTC-10 defines two kinds of testing, Type-1 and Type-2. Type-1 tests are conducted with the specified gas at or very near the specified operating conditions. Type-2 tests are conducted, subject to the limits of specific volume ratio, flow coefficient, machine Mach number and machine Reynolds number only. A substitute gas may be used and the compressor test speed is often different from the specified operating condition speed. The type of test to be applied is decided in advance of the actual test.

In the interest of maximizing test result accuracy, it is desirable that test conditions duplicate specified operating conditions as closely as possible. The limits in Table 1 provide maximum allowable deviations of individual parameters for Type-1 tests. The limitations of Table 2 provide maximum allowable deviations of the fundamental dimensionless parameter groupings for both types [1].

Recently, the requirement for Type-1 shop performance testing or Type-2 testing with the same pressure level as site conditions, are increasing for critical centrifugal compressors in order to mitigate any risks at site to the extent possible.

In general, test pressures are higher for Type-1 testing compared to Type-2 testing, in order to achieve the Reynolds number requirement. However, theoretically, the originating flow point of surge and rotating stall and measured performance would be the same for both the Type-1 and Type-2 performance test regardless of the test pressure. But, during the shop performance test with the different level of test pressure, a different originating flow point of the surge phenomena was found, although the measured performance was the same for all the pressure level.

The purpose of this paper is to identify the root cause of differences of the originating flow point of surge by the different test pressure.

Table 1. Permissible deviation from specified operating conditions for Type-1 tests.

Variable	Symbol	Units	Permissible Deviation
Inlet pressure	p_i	psia	5%
Inlet temperature	T_i	°R	8%
Speed	N	rpm	2%
Molecular weight	MW	lbm/lbmole	2%
Cooling temperature difference		°R	5%
Coolant flow rate		gal/min	3%
Capacity	q_i	ft³/min	4%

Table 2. Permissible deviation from specified operating parameters for Type-1 and Type-2 tests.

Parameter	Symbol	Limit of Test Values as Percent of Design Values: Min	Max
Specific volume ratio	v_i/v_d	95	105
Flow coefficient	ϕ	96	104
Machine Mach number Centrifugal compressors Axial compressors	M_m	See Fig. 3.3 See Fig. 3.4	
Machine Reynolds number Centrifugal compressors [Note (1)]	Rem	See Fig. 3.5	
Axial compressors where the Machine Reynolds number at specified conditions is below 100,000		90	105
Axial compressors where the Machine Reynolds number at specified conditions is above 100,000		10 [Note (1)]	200

NOTE:
(1) Minimum allowable test Machine Reynolds number is 90,000.

REVIEW OF SHOP PERFORMANCE TEST RESULTS

A shop performance test for a recycle gas compressor in refinery plant was conducted using a Type-2 test. The configuration of this compressor is shown on Fig. 1. This compressor has seven impellers with a very narrow flow path. Flow coefficient of the first stage is approximately 0.02.

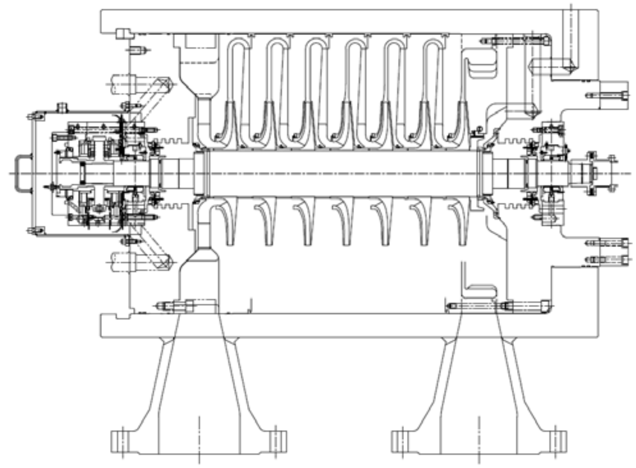


Figure 1. Configuration of recycle gas compressor.

The test conditions are shown on Table 3. For the test gas, a mixture of helium and nitrogen was used. Although the test was conducted under Type-2 conditions, the suction pressure of 50bar was almost the same as the site operating condition because the specification requested that the Reynolds number of the test condition should not deviate by more than 10% from the site operating condition.

The measured performance is shown on Fig. 2. The horizontal axis indicates the flow coefficient and the vertical axis indicates the normalized pressure coefficient. The red line indicates the expected performance and the green line indicates the measured performance during the shop test. As shown on this graph, the measured compressor performance during shop performance testing was almost the same as the expected performance, but the measured originating flow point of the surge was larger than expected by more than 15%.

Table 3. Performance test conditions for recycle gas compressors

			Test condition	Design condition	Ratio of Test/Design
Suction Temperature	T_i	(deg.C)	38.0	50.0	
Discharge Temperature 2)	T_d	(deg.C)	97.3	85.2	
Suction Pressure	p_i	(BarA)	49.0	48.0	
Discharge Pressure 2)	p_d	(BarA)	72.5	65.4	
Compression Ratio	p_d/p_i		1.479	1.363	
Suction Flow Rate	q_i	(m^3/h)	2,423	3,028	
Discharge Flow Rate 2)	q_d	(m^3/h)	1,967	2,482	
Volume Ratio	q_i/q_d		1.232	1.220	1.010 (0.95～1.05)
Molecular Weight	M		11.21	5.62	
Flow Coefficient	φ_D		0.0237	0.0237	1.000 (0.96～1.04)
Speed	N	(rpm)	7,100	8,874	
Weight Flow	W	(kg/h)	50,034	29,917	
Polytropic Head	W_p	(N・m/kg)	101,891	16,230	
Gas Horsepower	P_p	(kW)	1,768	1,652	
Machine Mach Number	M_m		0.277	0.260	0.017 (-0.21～0.22)
Machine Reynolds Number	Re		2,483,005	2,751,793	0.902 (0.10～26.11)

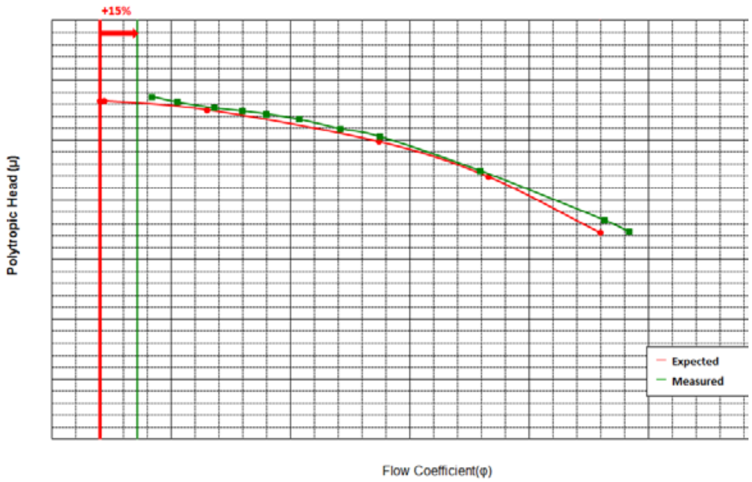


Figure 2. Normalized pressure coefficient vs. flow coefficient for recycle gas compressor.

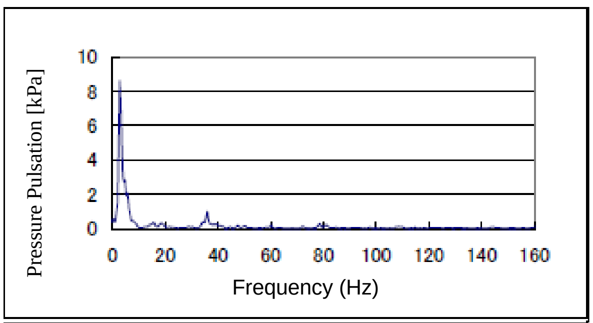


Figure 3. Pressure pulsation measured at compressor discharge piping at the flow coefficient of 0.014.

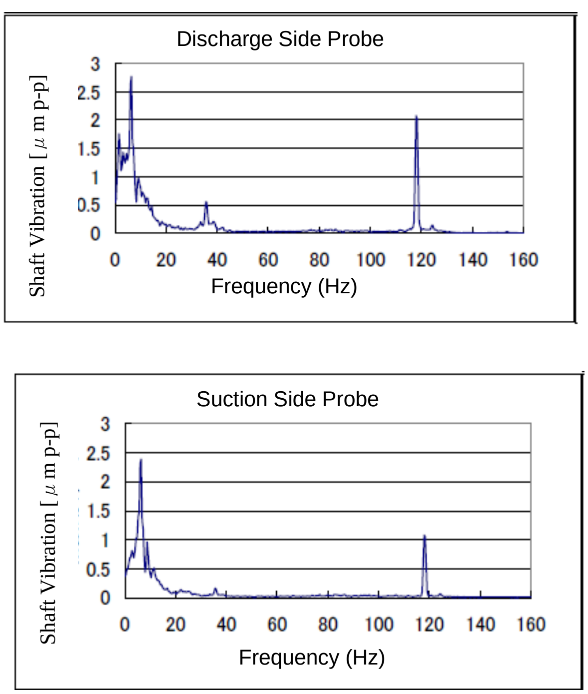


Figure 4 Shaft vibration of compressor at the flow coefficient of 0.014.

Fig. 3 and 4 shows the FFT of the discharge pressure pulsation and the compressor shaft vibration measured during the shop performance test at the flow just before the originating flow point of surge. As shown on these graphs, the sub-synchronous component of discharge pressure pulsation was observed at two frequencies of 2Hz and 30Hz. Also, the sub-synchronous shaft vibration was observed at the almost same frequencies of the discharge pressure pulsation. Therefore, these phenomena were considered to be caused by a fluid related force. The sub-synchronous component of 30Hz was considered to be caused by rotating stall because the shaft vibration measured at discharge side probes was larger than that measured at suction side probes. Rotating stall generally occurs at the discharge side impellers.

On the other hand, it seems that the observed sub-synchronous component of 2Hz is too low for rotating stall and the same level of the sub-synchronous shaft vibration at 2Hz was observed at both suction and discharge side.

In order to investigate in more detail, the suction pressure of the compressor during the performance test was changed from 50bar to 30bar or 20bar. In each of these cases, the machine mach number was the same as the original test condition and each required test conditions conformed to PTC-10 Type-2 condition. The test result is shown on Fig. 5. Although the measured compressor performance was the same across all the suction pressures, the originating flow point of surge was different. The lower the suction pressure of the compressor was, the lower the originating flow point of surge was. At the suction pressure of 20 bar, the actual surge phenomena was observed at the expected surge flow.

The measured pressure pulsation at each frequency of 2Hz and 30Hz was divided by the suction pressure of each condition and resultant normalized pressure pulsation ⊿P/Pinlet was plotted at each flow as shown on Fig. 6. The solid lines show the normalized pressure pulsation at 2Hz and the dotted lines show that at 30Hz. The normalized pressure pulsation at 30Hz did not change at any flow even when the suction pressure became larger, providing additional evidence that this sub-synchronous component was caused by rotating stall.

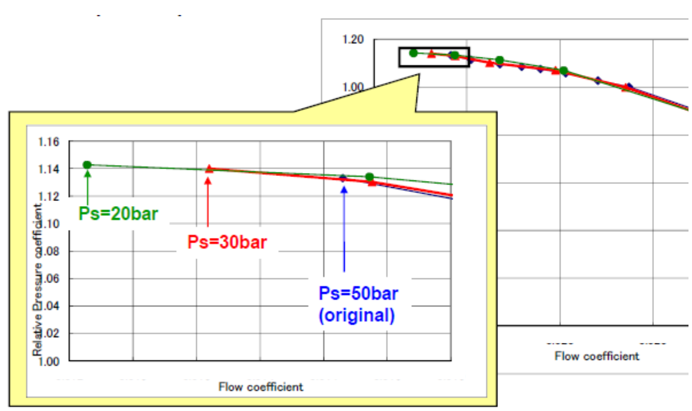


Figure 5. The effect of compressor suction pressure on originating flow point of surge.

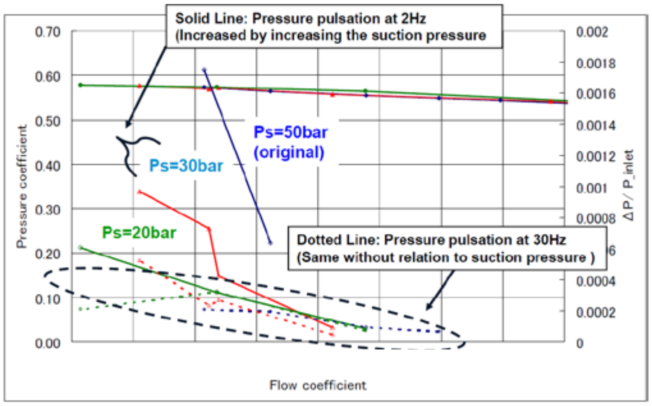


Figure 6. Normalized Pressure Pulsation for each suction pressure.

On the other hand, the normalized pressure pulsation at 2Hz increased with the suction pressure increase. The difference became larger with the reduction of the flow. From these results, it was found that the sub-synchronous vibration was not caused by the rotating stall, but by another fluid related force, and these phenomena would be related with the surge phenomena.

Since the shaft vibration level at the suction side and discharge side of the compressor was almost the same, the vibration might be caused by the fluid force related to the overall system as a test loop. Therefore, a simple model was created based on piping volume and its acoustic characteristic was roughly calculated as shown on Fig. 7. The calculated Helmholtz resonance frequency was 2.2Hz, which correlated well with the frequency of the measured pressure pulsation and the shaft vibration. For further investigation, the acoustic characteristic of the test loop was measured using the microphone as shown on Fig. 8. The microphone (M0) was installed at the volume position (cooler) and another microphone (M2) was installed at the non-volume position (piping).

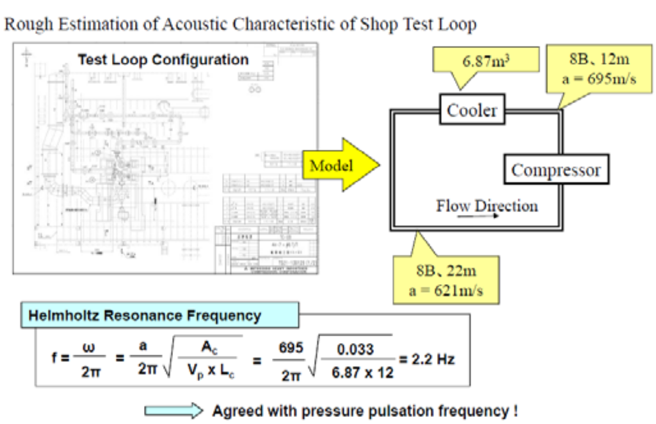


Figure 7. Rough Estimation of Acoustic Characteristic of Shop Test Loop

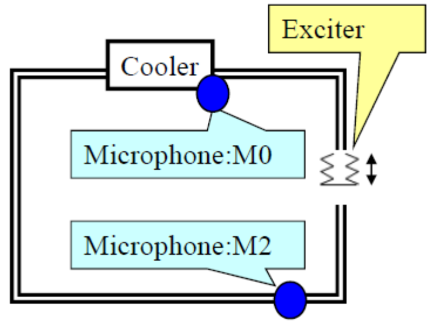


Figure 8. Measurement Procedure of Rough Acoustic Characteristic of Shop Test Loop.

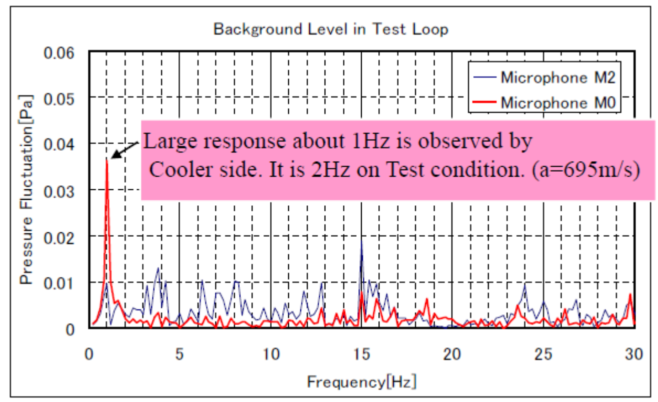


Figure 9. Measurement result of Acoustic Characteristic of Shop Test Loop.

The test loop was excited from the open portion of the test loop, where the compressor was located. The response of the microphones was shown on Fig. 9. As shown, the large response of microphone M2 was observed at approximately 1Hz. This measurement was done at atmospheric condition. Hence, in case the result is corrected to the test condition, considering the acoustic velocity of the test condition of 695 m/s, the excited frequency becomes 2Hz. From these investigation results of the acoustic characteristic of the test loop, the sub-synchronous vibration near surge condition was very likely to be related to the acoustic characteristic of the test loop.

FURTHER INVESTIGATION OF SURGE MARGIN SHORTAGE

A self-excited vibration model of coupled compressor dynamic characteristic and piping acoustic system has been investigated in the past, and Dr. Greitzer, et al reported that the possibility of the surge of the compressor can be organized by using the Greitzer's B Parameter as shown on equation 1 [2].

$$B = \frac{U}{2a}\sqrt{\frac{V_P}{A_C L_C}}$$

In cases where the Greitzer's B parameter is higher, the possibility of the surge is considered to become lower. In order to confirm the effect of Greitzer's B parameter on the originating flow point of surge, additional testing was planned as shown on Fig. 10. In addition to the usual measurement devices for Type-2 performance test, fourteen(14) pressure measurement instruments were installed all over the test loop to measure the pressure pulsation at each location and to confirm the mode of its acoustic characteristic. In addition, the hot wires were installed near the compressor suction and discharge piping to measure the flow fluctuation more accurately. Two test loops of 6 inch piping and 10 inch piping were arranged to change the Greitzer's B parameter and two types of perforated plate, with high or low pressure drops were prepared to confirm the damping effect of the acoustic resonance of the test loop. Totally, six (6) test cases were planned as shown on Table 4.

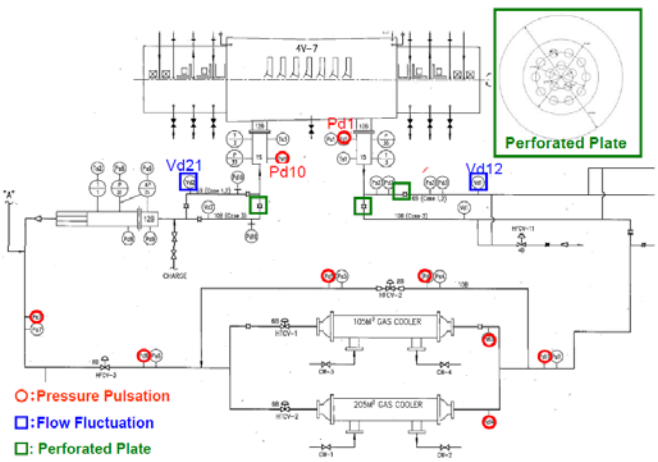


Figure 10. Shop Test Loop for Acoustic Characteristic Measurement.

Table 4. Test Cases Summary for Detail Investigation of Effect of Test Loop on Surge Margin.

Test Case	Piping Size	Perforated Plate		Greitzer's B Parameter
		Pressure Drop	Location	
Case-1	6 inch	-	-	0.185
Case-2	6 inch	Low	Downstream	0.185
Case-3	10 inch	-	-	0.114
Case-4	10 inch	Low	Downstream	0.114
Case-5	10 inch	High	Downstream	0.114
Case-6	10 inch	Low	Upstream	0.114

The purpose of case-2 is to confirm the effect of the perforated plate comparing with case-1, while the purpose of case-3 is to confirm the effect of Greitzer's B parameter comparing with case-1. The purpose of case-4 is to confirm the effect of the perforated plate comparing with case-3, and the purpose of case-5 is to confirm the effect of pressure drop of perforated plate comparing with case-4. Finally, the purpose of case-6 is to confirm the effect of the location of the perforated plate comparing with case-4.

The observed audible surge with un-measurable flow fluctuation at each case is shown on Fig. 11. The actual originating flow point of surge was changed depending on each case from 0 % to +16.7% of expected surge point although the measured compressor performance was the same. During these tests, the suction pressure of the compressor was changed from 50bar to 30 or 20bar to confirm the effect of the suction pressure on surge.

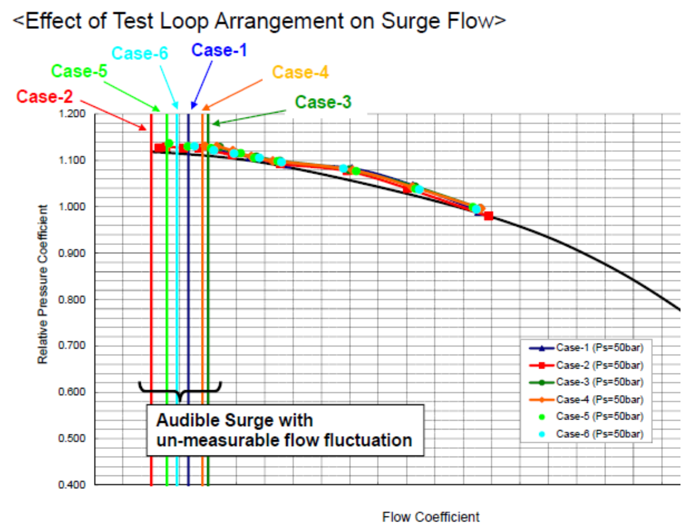


Figure 11. Observed Originating Flow Point of Surge at each Case.

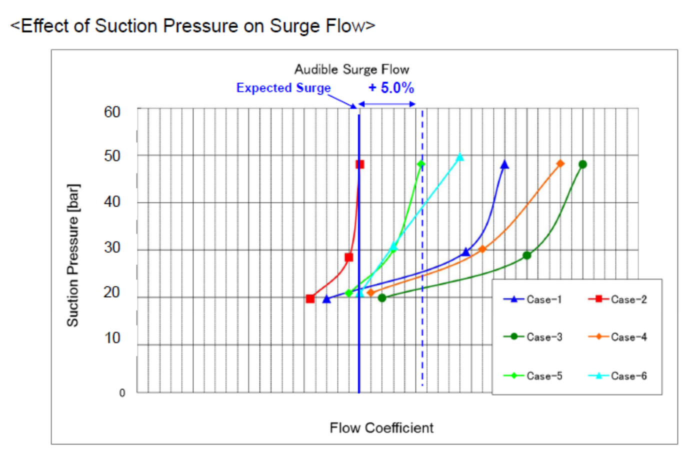


Figure 12. Effect of Compressor Suction Pressure on Originating Flow Point of Surge at each Case.

As shown on Fig. 12, when the suction pressure of the compressor was reduced, the originating flow point of surge became small for all the cases and the differences of the originating flow point of surge also became small. At a suction pressure of 20bar, for all the cases, the originating flow point of surge became less than +5% of expected surge flow. From these results, it is considered that the effect of the test loop characteristic on compressor surge phenomena becomes small and actual compressor surge characteristics may appear when the suction pressure is reduced.

For a more detail investigation of the effect of test loop on surge phenomena, the result of each test case was compared. The observed surge flow in case-1 was +10.8% of the expected surge flow. On the other hand, the actual surge flow in case-2 was observed at +0% of the expected surge flow. From this result, it was found that the perforated plate was useful in suppressing surge phenomena. Fig. 13 shows the compressor performance curves which show the relationship between the pressure ratio of the compressor and the actual suction flow. The static compressor performance was plotted as a pink line. And transient operating condition of the compressor during surge was plotted in blue using the measured flow fluctuation by the hot wires. As shown on this figures, the reverse flow was observed during the surge and the transient behavior during surge was slightly different between case-1 and 2. However, the root cause of the difference of the originating flow of surge was not found from this result.

Fig. 14 shows the discharge flow fluctuation at the audible surge measured by hot wire. During the audible surge, the flow velocity fluctuated at a certain frequency. This tendency was found in both case-1 and 2. But, in case-1, the flow fluctuation level was much higher than in case-2, by a factor of 2.5 times higher than that in case-2. From this result, the perforated plate would restrict the flow fluctuation of the compressor discharge, and it would suppress the surge phenomena.

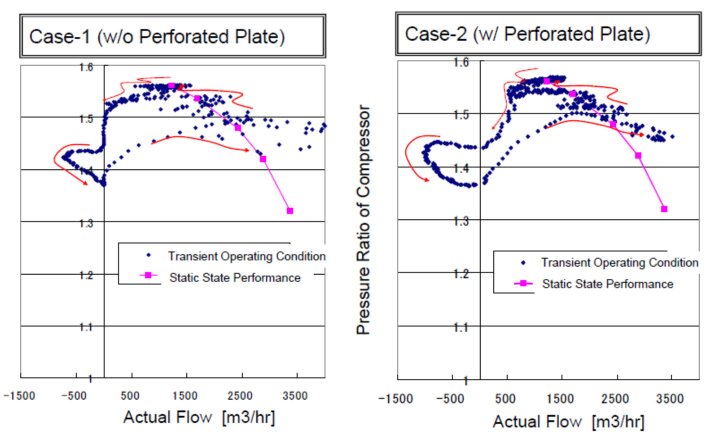
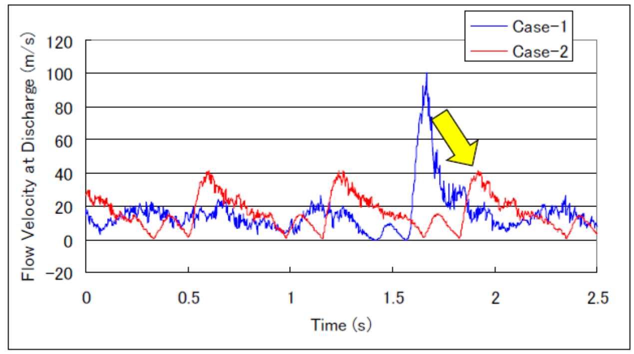


Figure 13. Transient Operating Condition during Surge with/ without Perforated Plate.



Discharge flow fluctuation was decreased by the damping effect of perforated plate.

Figure 14. Discharge Flow Fluctuation in Case-1 and 2.

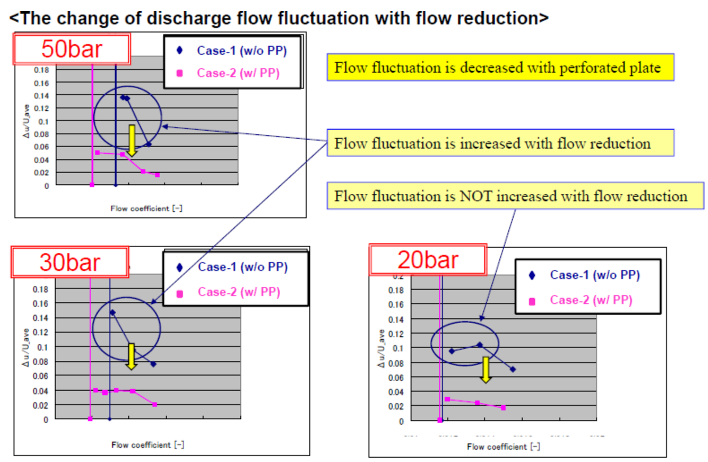


Figure 15. Change of Discharge Flow Fluctuation with Flow Reduction in Case-1 and 2.

The discharge flow fluctuation becomes larger when flow approaches to audible surge. The change of the discharge flow fluctuation with flow reduction was investigated at each suction pressure. As shown on Fig. 15, it was found that the discharge flow fluctuation decreased with the perforated plate for all suction pressure cases, and also the discharge flow fluctuation was not increased steeply even near the actual audible surge. Without perforated plate, at the suction pressure of 50bar and 30bar, the flow fluctuation steeply increased with the flow reduction. But, at the suction pressure of 20bar, the discharge flow fluctuation was not increased even near the actual audible surge condition.

Fig. 16 shows the relationship between the suction flow fluctuation and the discharge flow fluctuation. When the discharge flow fluctuation was larger, the suction flow fluctuation became also larger because the test was carried out at the closed loop. And, if the discharge flow fluctuation was decreased, the suction flow fluctuation also decreased.

From these results, it was determined that the discharge flow fluctuation was closely related with the actual surge flow.

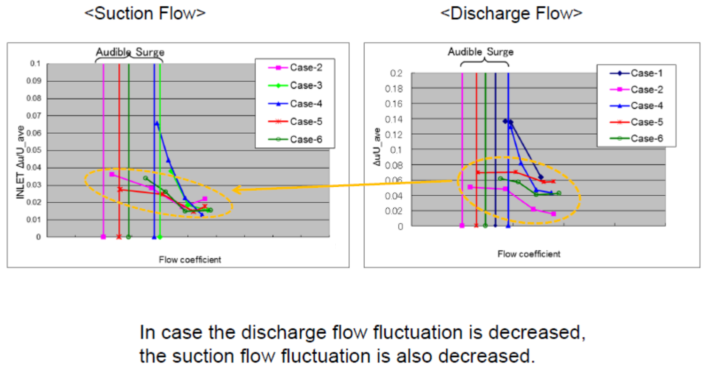


Figure 16. Relation between Discharge Flow Fluctuation and Suction Flow Fluctuation for all the cases.

The surge phenomena could occur if the flow is reduced and flow fluctuation becomes large enough. A perforated plate is useful to decrease the flow fluctuation and increase the surge margin. In case the suction pressure is low, the flow fluctuation is not increased steeply even when the flow approaches actual surge, it could increase the surge margin.

Comparing case-3 and 4, even with the perforated plate installed on case-4, the originating flow point of the surge was not changed. In case-3, the actual surge was found at +16.7% of the expected surge flow. And, in case-4, the actual surge was found at +15% of the expected surge point. The difference was only 1.7%. Therefore, further investigation was necessary to explain this phenomenon.

Fig. 17 shows the pressure pulsation FFT measured at compressor discharge (Pd1) just before the actual surge flow in case-2. The peak pressure pulsation was observed at 1.6Hz and 2.8Hz. These frequencies were considered to be the test loop acoustic resonance frequency. These pressure pulsations were measured at various points around the test loop by using fourteen (14) instruments, hence the mode of acoustic characteristic of the test loop was investigated.

<Pressure Pulsation FFT measured at Pd1(comp. discharge) >

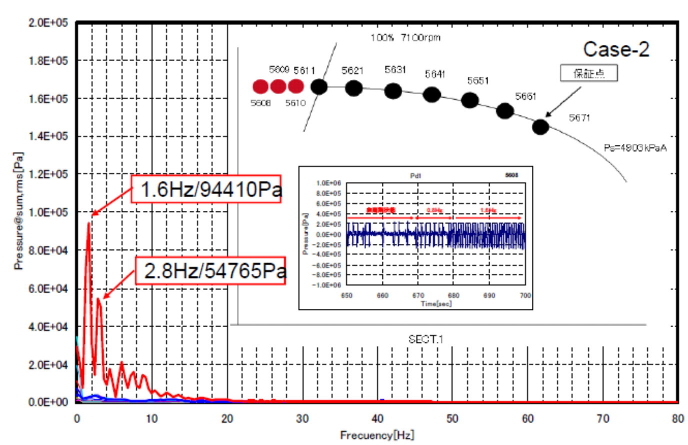


Figure 17. Pressure Pulsation FFT measured at compressor discharge in Case-2.

Fig. 18 shows the pressure pulsation mode at 1.6Hz from compressor discharge (left hand side) to suction (right hand side) for case 1 and 2. In those cases, the anti-node of pressure pulsation was observed between flow control valve and compressor suction. Comparing the acoustic mode of case 1 and 2, the difference was found only near the compressor discharge. In case 1, the node of pressure pulsation was observed at the compressor discharge. But, in case 2, it became the anti-node of the pressure pulsation by the effect of the perforated plate. Anti-node of the pressure pulsation is the node of the flow fluctuation. By adding the perforated plate near the compressor discharge, the node of the pressure pulsation became the anti-node of the pressure pulsation. Hence the anti-node of flow fluctuation near the compressor discharge became the node. Then the flow fluctuation at the compressor discharge became lower and flow fluctuation at the compressor suction also became lower. As the result, the surge phenomena were considered to be suppressed.

On the other hand, as shown on Fig. 19, there seemed to be no difference of acoustic mode of pressure pulsation between case 3 and 4.

<Pressure Pulsation at 1.6Hz at each Positions of Test Loop>

	Perforated Plate	Surge Flow %
Case-1	No	+10.8%
Case-2	Yes	+0.0%

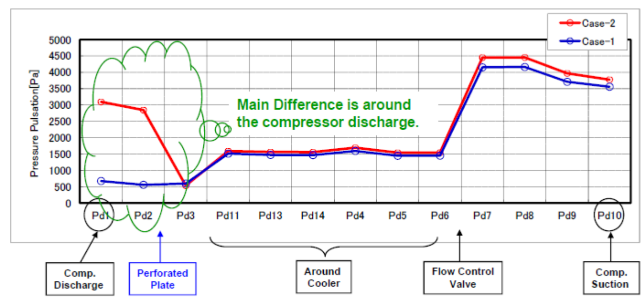


Figure 18. Pressure Pulsation Mode of Test Loop in Case 1 and 2.

<Pressure Pulsation at 1.6Hz at each Positions of Test Loop>

	Perforated Plate	Surge Flow %
Case-3	No	+16.7%
Case-4	Yes	+15.0%

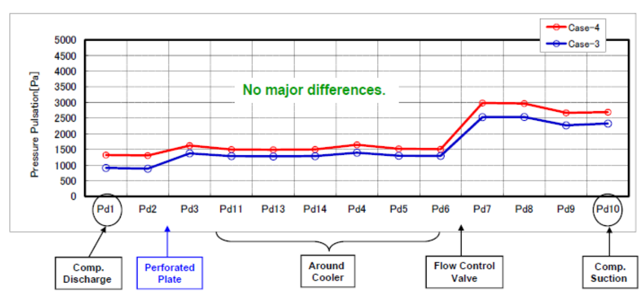


Figure 19. Pressure Pulsation Mode of Test Loop in Case 3 and 4.

Even if the perforated plate was added, the piping size of case 3 and 4 was larger than case 1 and 2. Therefore, the pressure loss was not enough to change the pressure pulsation mode around the compressor discharge. In case the pressure loss of the perforated plate was increased (case-5), the surge phenomena were suppressed as shown on the summary of the test results on Fig. 20. And also, it was found that the perforated plate was effective, even when installed upstream of the compressor, by decreasing the flow fluctuation of the compressor discharge transmitted to the suction flow. There was no major relation found between Greitzer's B parameter and the originating flow point of surge. It was considered that the Greitzer's B parameter of this test loop was too low to adopt this parameter for the judgment of the surge flow. This parameter may be useful around 0.6.

Test Case	Piping Size	Perforated Plate		Greitzer's B Parameter	Audible Suge Flow (% from Expected Surge)
		Pressure Drop	Location		
Case-1	6 inch	-	-	0.185	10.8 %
Case-2	6 inch	Low	Downstream	0.185	0.0 %
Case-3	10 inch	-	-	0.114	16.7 %
Case-4	10 inch	Low	Downstream	0.114	15.0 %
Case-5	10 inch	High	Downstream	0.114	4.6 %
Case-6	10 inch	Low	Upstream	0.114	7.5 %

Figure 20. Summary of Actual Surge Flow for Each Test Case.

CONCLUSIONS

A surge margin shortage was observed during the Type-2 shop performance test of the recycle gas compressor. The originating flow point of surge was changed when the test loop configuration and/or suction pressure were changed. The possibility of self-excited vibration coupled compressor dynamic characteristics and piping acoustic system were investigated. As a result, it was found that the lack of surge margin was caused by the large discharge flow fluctuation transmitted to the suction flow. This discharge flow fluctuation became larger in case the compressor discharge was the node of the pressure pulsation (anti-node of the flow fluctuation). If the suction pressure was low enough, the discharge flow fluctuation was not increased steeply by the reduction of flow and the surge was suppressed. When a perforated plate was installed in the test loop, where the node of the pressure pulsation could be changed to the anti-node, the flow fluctuation could be reduced and the surge phenomena could be suppressed. This coupled phenomenon of compressor dynamic characteristics and piping acoustic system would not occur at the installation site because the process piping length is so long that the flow fluctuation of the compressor discharge would not be transmitted to the compressor suction. In cases where an unexpected surge phenomenon is observed at a flow larger than the surge flow measured in shop testing, installing a perforated plate would be one countermeasure to suppress the surge. In that case, the total energy merit should be evaluated comparing the power loss by the recycle flow without perforated plate with the power loss by the pressure loss of the perforated plate.

REFERENCES

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