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ABSTRACT

Suction valves with even thinner and narrower, used in a high efficient reciprocating compressor, are required in order to reduce suction resistances. However, low flow resistance suction valves run properly under the very severe operating condition of high periodic impact stress which can cause the compressor failure. In this study, the effects of the valve design parameters on the impact fatigue strength of suction valves are investigated with Finite Element Method (FEM). Suction valve design parameters are compared and verified with compressor durability tests.

1. INTRODUCTION

Suction valves in hermetic compressors experience severe repetitive impulse stresses during operation and they are supposed to work properly during compressor lifetime. Therefore, the fatigue strength of valve system is an important factor of compressor reliability. Valve fatigue analyses can be found in several papers [1, 2], but their shapes are different from those used in our compressors. Several valves with different shapes are tested analytically and experimentally to investigate impulse stresses on the valve surface. Each valve has been made with same manufacturing process. Taguchi method is adopted for the experiment so that the number of test compressors can be minimized. Valve velocity at impact was calculated with numerical analysis. The suction pressure and cylinder pressure needed for the impact analysis were measured. Each valve surface are investigated after the 1,000 hour durability test.

2. DESIGN PARAMETERS OF SUCTION VALVE

2.1 Valve shapes

Valve thickness, width, and the minimum radius of curvature are considered as important valve design parameters. Also they can affect the compressor noise and electric power consumption. But the noise and the electric power consumption are not in our interest because the durability of the suction valve is our main consideration. Figure 2.1 shows the valve shapes and design parameters.

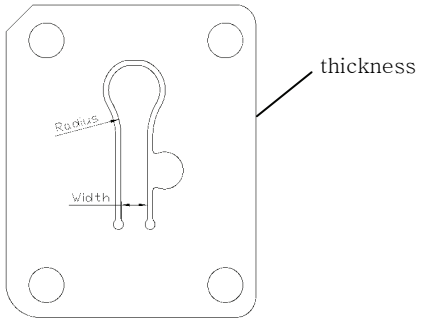

Figure 2.1: Valve design parameters

2.2 Test samples

Taguchi method is used for the experiment, which reduce the number of samples for the durability test. 4 types are selected according to L3 array at 2 levels. Table 2.1 and Table 2.2 show the properties of Sandvik 20C used in this study and the design variables.

Table 2.1: Suction valve material properties

Valve Material	Sandvik 20C
Density	7800 kg/㎥
Poisson's Ratio	0.3
Young's Modulus	210 GPa
Shear Modulus	80.77 GPa

Table 2.2: Design variables

No.	**Radius**	**Width**	**Thickness**
1	R1	2.9 mm	0.152 mm
2	R1	3.9 mm	0.203 mm
3	R10	2.9 mm	0.203 mm
4	R10	3.9 mm	0.152 mm

3. THEORITICAL ANALYSIS

Maximum impact stress on the valve surface can be defined as equation (1).

$$\sigma_{\max} = v\sqrt{\rho E} \tag{1}$$

To analyze the impact stress, the instantaneous valve velocity needs to be known when the valve collides with the stopper. The valve can be considered as 1-DOF mass spring damper system because the lateral or twisting movement of the suction valve is negligible and the higher order bending is not significant. The valve system can be modeled as equation (2). We can get the information of the valve dynamic characteristic, such as, mass, stiffness and damping, with FEM analysis.

$$\ddot{y} + 2\zeta\omega_n\dot{y} + \omega_n^2 y = \frac{A_{vp}\Delta p}{m_v} \quad , \quad (\Delta p = p_c - p_s) \tag{2}$$

Δp was obtained by measuring the pressures of the suction plenum and cylinder inside while the compressor runs in air because we assume that the discharge and suction pressure characteristics in air are similar to as in R-134a. Figure 3.1 shows how to measure the cylinder pressure and the suction plenum pressure. Table 3.1 includes 1st mode frequencies of the tested valves.

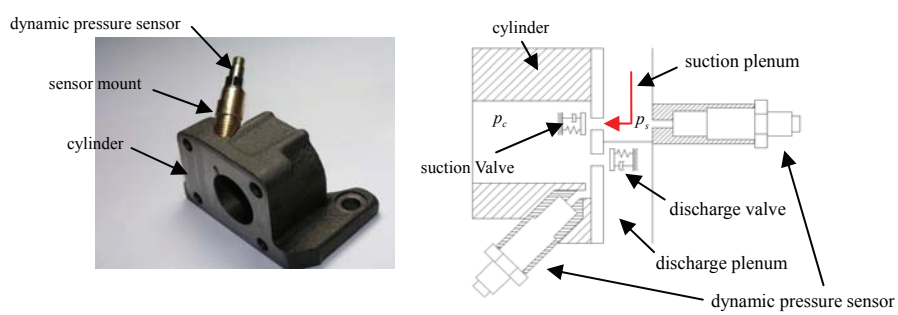

Figure 3.1: Measurement of suction plenum and cylinder inside

Table 3.1: Valve dynamic property

No.	1st mode frequency (Hz)	Mass (g)
1	231.57	0.097055
2	356.09	0.14993
3	298.54	0.13545
4	258.57	0.11552

Table 3.2: Suction valve velocity at impact

No.	Velocity (m/s)
1	1.1835
2	0.2025
3	0.3770
4	0.9498

With Runge-Kutta iteration, the valve velocity was calculated with time. The impact velocity was obtained when the valve lift becomes zero.

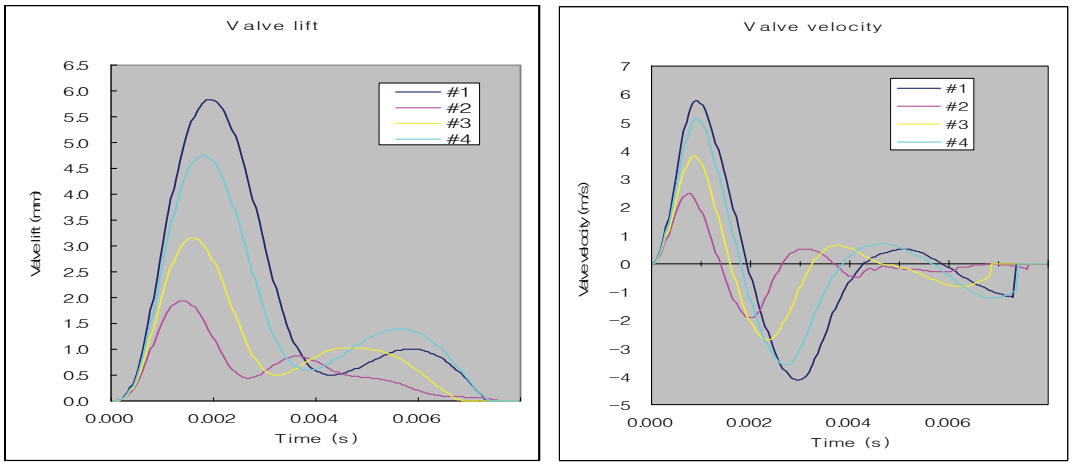

Figure 3.2: Valve lift and velocity

And the impact stress analysis was performed with FEM. The initial and boundary condition was set as shown in Figure 3.3 in order to reduce the computation time.

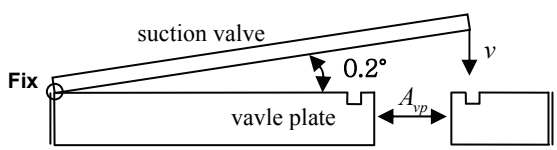

Figure 3.3: B.C. and initial condition of valve impact

The FEM results are shown in Figure 3.4. The location of the maximum level of the stress is near the curvature change and the end position of the valve. The theoretical calculation result and the FEM result are compared in Table 3.3.

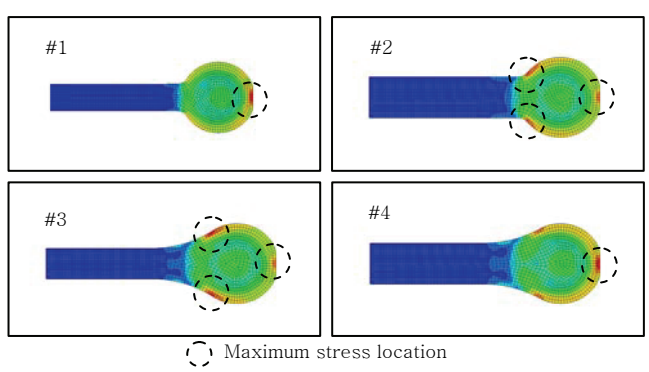

Figure 3.4: FEM results

Table 3.3: Maximum impact stress analysis results

No.	Velocity (m/s)	Maximum impact stress (MPa)	
		Calculation result	FEM result
1	1.1835	47.90	46.04
2	0.2025	8.20	5.51
3	0.3770	15.26	11.61
4	0.9498	38.44	35.83

The most important design parameter which affects the valve velocity at impact is the valve thickness and the next is the valve width. That the moment of inertia is proportional to W and t^3 can explain the analysis results. The location of maximum impact stress is the end position of the valve because the valve tip has the highest line velocity, which is shown in Figure 3.3 and 3.4. Table 3.3 shows that the calculation result is little bit higher than the FEM result. The equation (1) assumes that the whole area contacts with the stopper simultaneously so that the calculation results have higher impact energy. The velocities at impact are much lower than the endurance limit of the valve material. It is expected that the valve can operate properly during the compressor lifetime. Figure 3.5 shows the impact fatigue graph of Sandvik 20C.

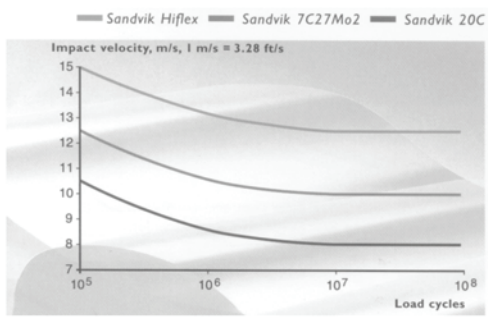

Figure 3.5: Impact fatigue of Sandvik steels

4. EXPERIMENTS

The suction valves were prepared with barreling process after wire-cut process. And the durability test bench as shown in Figure 4.1 was used for the valve impact test. 40 compressors with 4 different valves were tested under the pressure condition of 1.5kgf/cm^2 at suction side and 30kgf/cm^2 at discharge side for 1,000 hours. Any typical crack or fracture was not found on the valve surfaces after the durability test. It is well agreement with that the impact analysis results in Table 3.2 are below the velocity limits. Figure 4.2 shows the valve surface which experiences the durability test. A bit of accumulated carbon and affected area by the impact stress can be seen on the surface but they are not critical. All valves with 4 different types have the similar results.

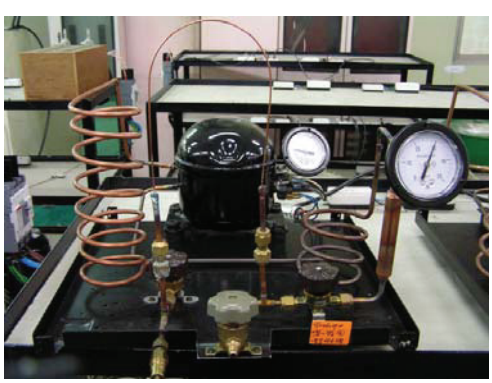

Figure 4.1: Durability test bench

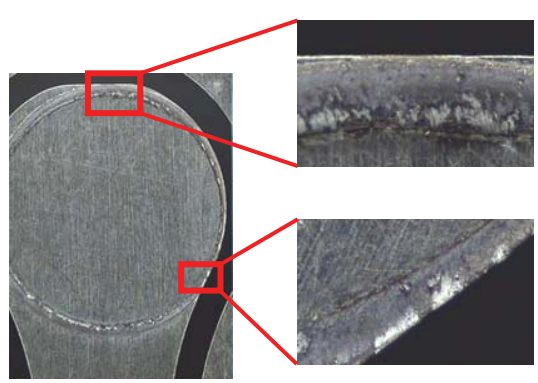

Figure4.2: Suction valve after the durability test

5. CONCLUSION

The impact stress on the suction valve surface is analyzed with FEM and its effect is investigated with the compressor durability test. The impact stress level is mainly affected by the thickness and the valve with thinner thickness has higher impact stress due to the higher impact velocity. And the computation time to get the local impact stress level could be reduced significantly by using the experiment and numerical analysis. However, the effect of the valve shape, such as valve width and minimum radius of curvature, is not clarified with FEM analysis. And the difference between the tested valves was not seen clearly after the experiment so that the relation between design parameters and valve lifetime could not be verified in this study. The test conditions and the design parameters, especially minimum radius of curvature, will be modified in order to have the relations in future work.

NOMENCLATURE

v	velocity of the valve	ω_n	natural frequency
ρ	density of the valve	ζ	damping coefficient
E	modulus of elasticity	m_v	mass of the valve
A_{vp}	area of suction port	y	valve lift
p_c	cylinder pressure	p_s	suction plenum pressure
W	width of the valve	t	thickness

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