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ABSTRACT

In this paper, Fluid Structure Interaction analysis is conducted to simulate moving mechanism of discharge valve and predict a failure in discharge valve in reciprocating compressor. Moving mechanism of discharge valve, opening and closing, shows impact phenomenon owing to a high velocity correspond to valve motion. The cylinder pressure is successfully validated before conducting impact analysis between discharge valve and other susceptible supported structure. The status of compressor model just before impact form FSI analysis is obtained. It is used as a initial condition to carry out further impact analysis. The Stress in discharge valve gives preliminary estimation about compressor reliability due to its impact phenomena.

1. INTRODUCTION

The recent extensive application of computer-aided design has enabled many engineers to conduct viable analysis in estimating the performance of many vehicles, machine, and various manufactured design. Compressor modeling and simulation give insight analysis into the compressor performance and efficiency in its several operating condition [1,2].

In this paper, we emphasize our analysis in scrutinizing discharge valve mechanism. Experimental test revealed that the failure occurred in discharge valve due to its mechanism process. Within its operating period, discharge valve failure could be a problem that makes compressor failed to operate it. Under normal operating condition, discharge valve may fail by wear, overstress, fatigue, corrosion, or any combination of these factors [3].

In order to attain long-life operating cycle and reliability of the discharge valve, investigation on its opened-closed mechanism is necessarily needed. Fluid Structure Interaction analysis is required to simulate discharge valve moving mechanism more exactly. Based on the result which is given by FSI analysis at impact moment, impact analysis is conducted to predict high-stress region around discharge valve that tend to be failed.

2. THEORETICAL BACKGROUND

ADINA is commercial finite element software which provide powerful algorithm for solving various type of fluid-structure interaction problem. In FSI analyses, the fluid force gives deformation to structural model and this deformation changes fluid domain. The ALE formulation (Arbitrary Lagrangian Eulerian) at FSI boundary condition is imposed at the adjacent between fluid and structure. The solid node and the fluid node could be not compatible each other. The mapping procedure as illustrated in Figure 1 shows that completely different elements and meshes are applicable in FSI analysis. Displacement and stress at fluid nodal at FSI boundary is determined by interpolating displacement and stress of solid node in the vicinity of corresponding fluid node [4].

There are two basic conditions for applying fluid-structure interfaces. The first condition is kinematic condition or displacement compatibility which is presented as follow:

$$\underline{d}_f = \underline{d}_s \tag{1}$$

The second condition that must to be achieved is dynamic condition or traction equilibrium:

$$n \bullet \underline{\tau}_f = n \bullet \underline{\tau}_s \tag{2}$$

where $\underline{d}_f$ and $\underline{d}_s$ are the fluid and solid displacement respectively, and $\underline{\tau}_f$ and $\underline{\tau}$s, respectively, are the fluid and solid stresses. The underlining characters denote that they are only defined in fluid-structure interfaces only.

In order to perform fluid structure interaction analysis of compressor model, the iterative fully-coupled system is performed to obtain the solution.

2.1 The Finite Element Equation of the Coupled System

The finite element equations of the coupled system can be described as:

$$F[X] = \begin{bmatrix} F_f\left[X_f, \underline{d}_s(X_s)\right] \\ F_s\left[X_s, \underline{\tau}_f(X_f)\right] \end{bmatrix} = 0 \tag{3}$$

where F_f and F_s are finite element equation corresponding to equation G_f and G_s respectively. X_f and X_s is the fluid and solid solution vectors defined at the fluid and solid nodes respectively. Thus,

$$\begin{aligned} \underline{d}_s &= \underline{d}_s(X_s) \\ \underline{\tau}_s &= \underline{\tau}_f(X_f) \end{aligned} \tag{4}$$

G_f and G_s represent the fluid equations and the solid equations respectively which are prescribed as follow:

$$\begin{aligned} G_f\left[f, \dot{f}\right] &= 0 \\ G_s\left[d, \dot{d}, \ddot{d}\right] &= 0 \end{aligned} \tag{5}$$

where the fluid variables are denoted by *f* and the solid displacements are denoted by *d*.

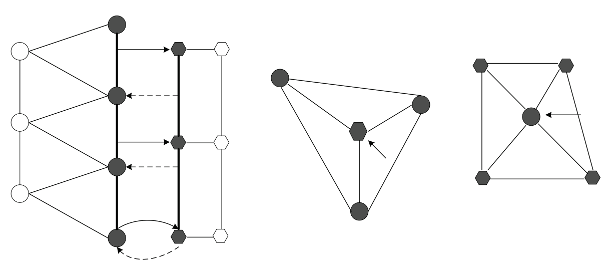


Figure 1: Mapping fluid and solid nodes in FSI analysis.

3. ANALYSIS MODEL

3.1 Compressor Model

Figure 2 shows compressor part which is taken into analysis model. Because we just emphasize the analysis in simulating discharge valve opened-closed mechanism, only cylinder compression and small distance of cylinder expansion have been considered. The initial position of cylinder is placed at maximum position of piston expansion, as shown in Figure 3.

In this analysis, compressor model is divided into 2 analysis domains, fluid domain and corresponding structural domain. Cylinder volume and discharge volume are considered as fluid domain, and discharge valve is taken as structural domain. And suction process is not included in this analysis model.

The fluid is assumed as low-speed compressible flow model. Navier-Stokes equation is used to describe the flow assumption:

$$\begin{aligned} &\frac{\partial \rho}{\partial t} + \nabla \bullet (\rho \vec{v}) = 0 \\ &\frac{\partial \rho \vec{v}}{\partial t} + \nabla \bullet (\rho \vec{v}\vec{v} - \bar{\tau}) = f^B \\ &\frac{\partial \rho E}{\partial t} + \nabla \bullet (\rho \vec{v} E - \bar{\tau} \bullet v + q) = f^B \bullet \vec{v} + q^B \end{aligned} \tag{11}$$

where t is the time, ρ is the density, $\vec{v}$ is the velocity vector, f^B is the body force vector of fluid medium, $\bar{\tau}$ is the fluid tensor, *E* is the specific total energy, and *q* is the heat flux, and q^B is the specific rate of heat generation.

The specific total energy and stress are defined as:

$$E = \frac{1}{2}\vec{v} \bullet \vec{v} + e \equiv b + e \tag{12}$$

$$\bar{\tau} = (-p + \lambda \nabla \bullet \vec{v})\mathrm{I} + 2\mu \vec{e} \tag{13}$$

where *e* is the specific internal energy, *b* is the specific kinetic energy, *p* is the pressure, μ and λ are the two coefficients of fluid viscosity, and $\vec{e}$ is the velocity strain tensor.

The heat flux is defined as follow:

$$q = -k \nabla \theta \tag{14}$$

where θ is the temperature and k is the heat conductivity coefficient.

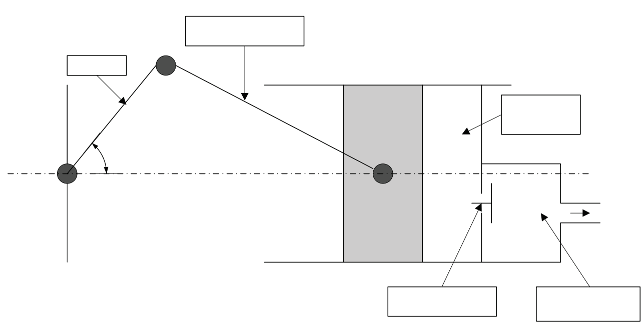


Figure 2: Schematic of compressor analysis model.

To obtain the closed system of equation, it inquires state equation which is defined as:

$$\begin{aligned} \rho &= \rho(p,\theta), \\ e &= e(p,\theta) \end{aligned} \tag{15}$$

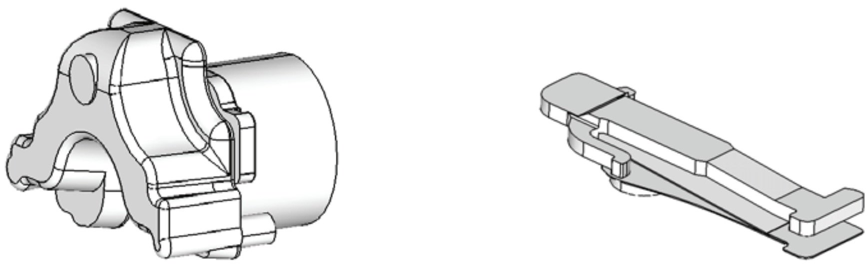


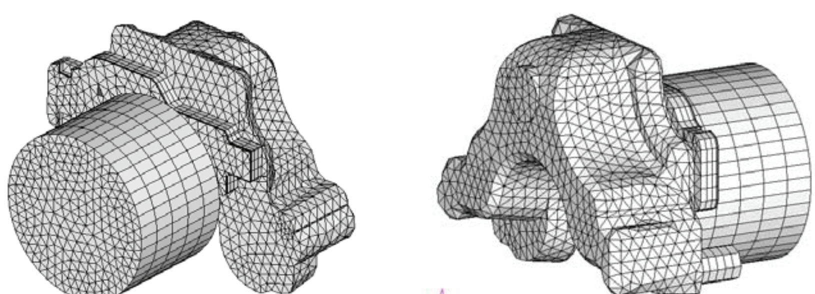
Figure 3: Compressor Analysis Model.

3.2 Finite Element Model of Compressor (Fluid domain)

Fluid domain of compressor model is divided into two primary parts, the cylinder part and the discharge volume. Between those parts, gap boundary condition is imposed to segregate high pressure region in discharge volume and low pressure region inside cylinder. Slipping boundary condition is applied on outer surface of cylinder region to maintain good mesh quality as cylinder move forward, which is described as:

$$U_x = U, U_y = 0, U_z = 0 \tag{16}$$

FSI boundary is imposed on corresponding area where discharge valve is placed in the coordinate system. Initial pressure is defined in whole fluid domain and its value is 0.093 MPa and 1.469 MPa for cylinder volume and discharge volume respectively. The fluid domain is discretized using 4-node tetrahedral and 6-node penta 3-D fluid element witch is made by Nastran format which is shown in Figure 4.



a) Back-side view b) Front side view

Figure 4: Finite element model of compressor fluid domain

The working fluid which is defined in this analysis is refrigerant gas. Material properties of fluid are shown in Table 1. In low speed compressible flow, the ideal gas law, $p = (C_p - C_v)\rho\theta$, and $e = C_v\theta$ is used to solve gas flow equation. The principal difference between incompressible and compressible flow is high dependency between pressure, temperature, and density which results in strong coupling of momentum and energy equation. This correlation obviously gives highly nonlinear behavior in analysis solution.

The material properties data is calculated by using EES Program which is based on state equation [5].

Table 1: Material properties of refrigerant

Material Constant	R134a (at P=199.23kPa, T=332.2°K)	R600a (at P=101.23 kPa, T=328.2°K)
Density (ρ)	7.5665 kg/m^3	2.1985 kg/m^3
Viscosity (μ)	1.3135e-5 kg/ms	8.2308e-6 kg/ms
Heat conductivity (k)	0.016146 W/mK	0.020152 W/mK
Coefficient of volume expansion (β)	0.003325 1/K	0.0032435 1/K
Specific heat at constant pressure (C_p)	915.25 m^2/s^{2}K	1825.5 m^2/s^{2}K
Specific heat at constant volume (C_v)	821.24 m^2/s^{2}K	1669.6 m^2/s^{2}K

3.3 Finite Element Model of Compressor (Structural domain)

Discharge valve configuration as presented in Figure 5 has important supported structures to optimize compressor performance, such as valve spring and ring-seated structure. In FSI analysis, structural model is discredited using 8-node hexahedral and 6-node penta 3-D solid element. Fixity is imposed at the root area of discharge valve as follow:

$$d_x = d_y = d_z = 0 \tag{17}$$

Valve spring that restricts discharge valve displacement has been slightly bent due to pre-stress condition induced by retainer configuration. Pre-stress valve spring shape is used as rigid contact surface model in FSI analysis as depicted in Figure 5.

Material properties used in this model is constant properties with E = 210GPa, υ = 0.29, and ρ = 7700 kg/m^3.

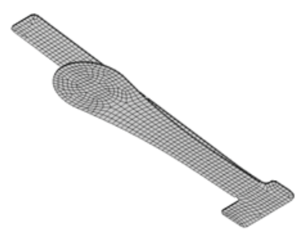


Figure 5: Finite element model of compressor structural domain

Finite element models for FSI analysis are also used in impact analysis. Time step is defined as 1.0E-8 second.

4. RESULT AND DISCUSSION

4.1 Static analysis for pre-stress

It should be considered initial stress when discharge valve, valve spring and retainer are assembled. Initial status influences valve moving mechanism for FSI analysis. Compressor Type 1 has initial stress; Figure 6 shows the result of assembly static analysis.

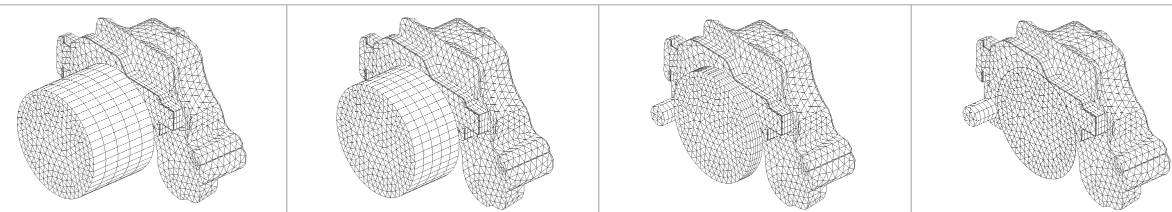
a) Valve configuration after assembly | (b) Stress contour after assembly

Figure 6: Result of structural analysis of assembly

The shape and stress after assembled are used as initial value for FSI analysis.

4.2 Mesh configuration at time

Grid movement of cylinder mesh is presented in Figure 9 and it is reached peak compressed position at time 0.00833 second.



(a) T=0.0 (b) T=0.003 (c) T=0.006 (d) T=0.00833

Figure 7: Moving grid position of cylinder at several time values [Type 1]

4.3 Pressure inside cylinder

Piston movement gives high increment of cylinder pressure. Figure 8 shows comparison graph of average pressure inside cylinder along compression process relative to time. As piston compresses refrigerant inside cylinder to its peak position (at time = 0.00833), averaged pressure in cylinder continuously increases until it reaches position at time = 0.0074. And then pressure begins to plunge. Discrepancies between numerical and computational result could be generated by adding fluid film in order to achieve higher numerical stability and by modeling dead volume in fluid model. Maximum pressure from simulation is reached at value 1.92 MPa and is very similar to experimental pressure 1.87MPa in Figure 8.

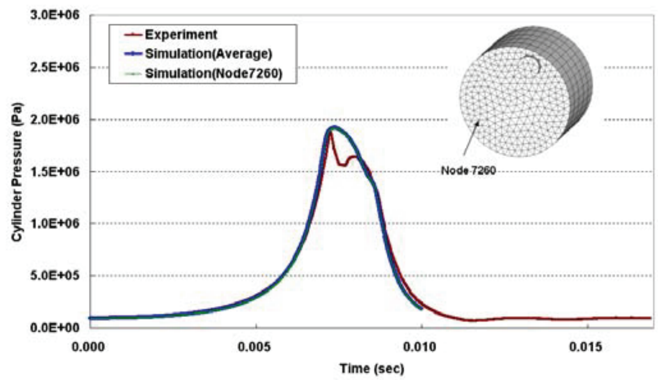
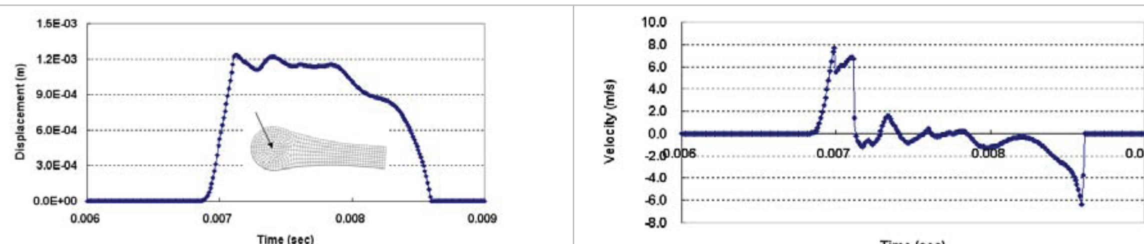


Figure 8: Pressure inside cylinder [ASHRAE]

4.4 Displacement and velocity profile of discharge valve

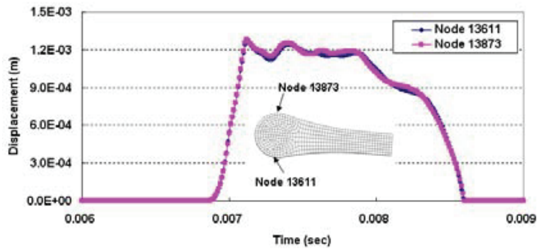
During cylinder movement at given time, discharge valve performs one cycle opened-closed mechanism. The displacement of discharge valve presented in Figure 9 shows high slope in opening and closing mechanism. The velocity graph represents sudden increment of velocity value during opened-closed mechanism as shown in Figure 10. For this type of compressor, opening velocity reaches 7.7m/s and closing velocity value is about 6.4m/s.



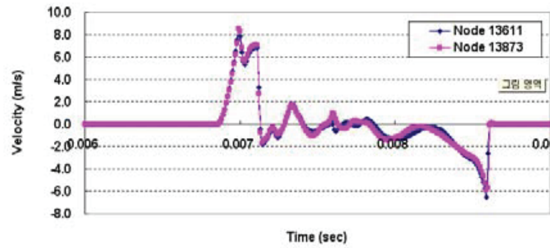
(a) Displacement of discharge valve (b) Velocity of discharge valve

Figure 9: Discharge valve motion at valve head center node [Type 1, ASHRAE]

Velocity difference on the both side edges is conducted to check twist mode which could influence the fracture of discharge. As see in Figure 10, the velocity variance is very slight at the opening and closing moment. So there could not be any problem by twist mode shape.



Displacement Comparison between both sides



Velocity Comparison between both sides

Figure 10: Valve displacement and velocity [Type 1, ASHRAE]

4.5 Displacement and velocity profile of discharge valve in discharge clogging condition

FSI analysis of compressor model has been conducted for 3 different models, Type 1, Type 2, and Type 3. Discharge valve configuration shows dissimilarity between each other which is described in Figure 11. Type 1 has one discharge port, but type 2 and type 3 has two discharge ports.

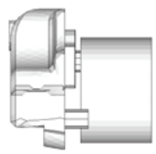
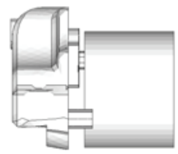
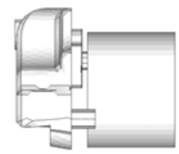
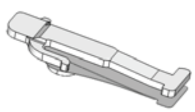
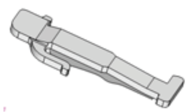
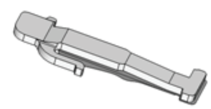
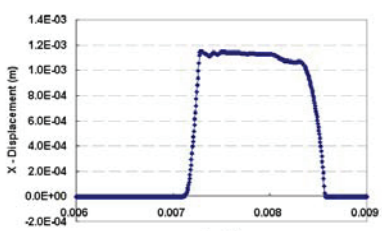
Compressor Type	Type 1	Type 2	Type 3
Fluid Model	 R134a, 5.7cc, 60Hz	 R600a, 8.2cc, 50Hz	 R600a, 8.2cc, 50Hz
Structure Model	 With long valve spring	 Without valve spring	 With short valve spring

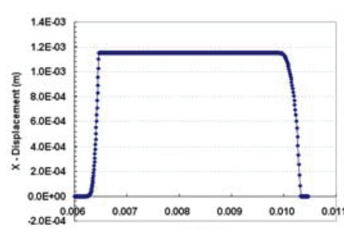
Figure 11: FSI & impact analysis models

The Figures (12 and 13) present displacement and velocity at discharge valve node in middle region of discharge hole. Velocity of type 1 reaches 12.4 m/s and 11.3 m/s, at opening and closing mechanism respectively. Opening and closing velocity for compressor Type 2 are 14.8 and 8.7 m/s, respectively. Opening and closing velocity for compressor Type 3 are 12 and 9.3 m/s, respectively. Valve spring which is placed between discharge valve and retainer in compressor Type 3 contributes in reducing opening velocity of discharge valve. In (c) type 3 of Figure 13, there are 2 peaks velocity which is illustrated in opened mechanism. It means that the contact between valve and valve spring decreases velocity magnitude and in the same time discharge valve also push forward valve spring until they reach maximum displacement of opened position.

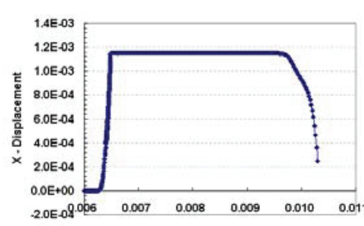
By comparing discharge valve displacement and velocity results for compressor Type 1 and Type 2, valve spring configuration as dynamic structure gives an effect to impact velocity magnitude of discharge valve in (a) Type 1 in Figure 13. Valve spring decreases the opening velocity of valve and increases the closing velocity by pushing it in closing mechanism. It gives lower velocity value of discharge valve compare to compressor Type 2 in opening mechanism.



(a) Type 1

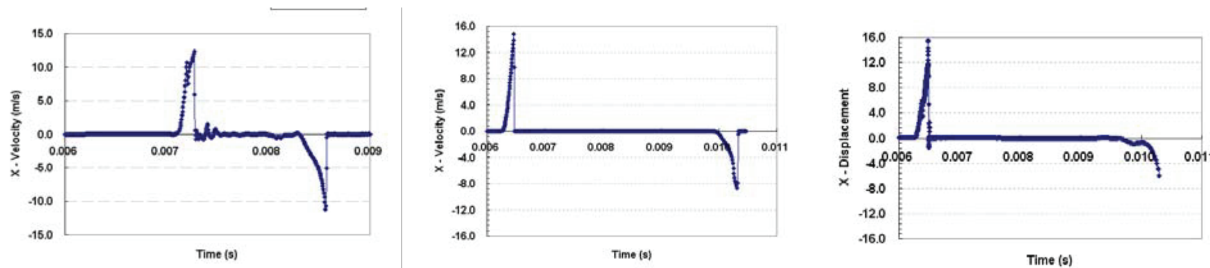


(b) Type 2



(c) Type 3

Figure 12: Displacement of discharge valve node at valve head center node



(a) Type 1 (b) Type 2 (c) Type 3

Figure 13: Local velocity of discharge valve node at valve head center node

4.6 Stress of discharge valve in discharge clogging condition

Impact analysis is carried out using the structure simulation result from FSI just before impact moment. Impact stress contour is illustrated in Figure 14 and Figure 15. In impact analysis of opening mechanism, the stress of discharge valve in Type 1 and Type 3 is smaller than that of Type 2. That is because valve spring blocks valve movement during its opening. Although the velocity of compressor type 1 is high, valve stress is low because of valve spring's support. It means that valve spring functions as a damper against impact. But in case of compressor type 2, valve directly hits against retainer with high velocity. So the stress of valve could be very high when it reaches retainer.

In impact analysis of closing mechanism, (c) position in Figure 16 places on seat later than (b) position in type 1. But the movement of type 2 is different from type 1. The stress in closing mechanism is generally low comparing that of opening mechanism.

High stress value is obtained in impact analysis for opening mechanism at compressor Type 2. As shown in type (b) of Figure 14, the value is about 1041Mpa and it is close to the fatigue limit of valve. Structural failure could be occurred at the edge of discharge valve which frequently contacts the retainer at its opening mechanism. The valve movement in compressor type 3 is similar to type 1, but opened time lasts more long. Because the reaction force of spring is less than that of type 1 and it doesn't have initial stress and its movement slightly free.

(a) Type 1 (b) Type 2 (c) Type 3

Figure 14: Stress contour of impact analysis for opening mechanism of compressor

(a) Type 1 (b) Type 2 (c) Type 3

Figure 15: Stress contour of impact analysis for closing mechanism of compressor

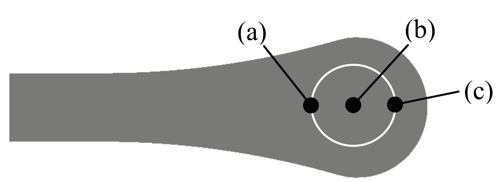


Figure 16: Position on discharge valve

5. CONCLUSION

Fluid structure interaction analysis for a compressible model gives a good comparison result with experimental test. Impact analysis corresponded to FSI velocity result data could estimate critical region of potential failure at discharge valve. Complete model analysis involving suction valve is needed to be carried out in order to simulate and analyze real physical model. Heat transfer effect needed to be considered to investigate compressor performance in several operating condition. It is also important to predict discharge valve life-cycle time based on real physical model.

6. REFERENCE

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