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Study of the Breathing Effect of Reciprocating Compressor Under Duty Cycle Regulation (DCR) Capacity Control by Simulation and Experiment

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Abstract

Duty Cycle Regulation is a new method for capacity control of reciprocating compressor. Like other suction-valve-unloaded methods, the DCR method would inevitably cause the Breathing Effect. In this article, the internal flow and heat transfer in the compressor under DCR control are analyzed using CFD simulation. The geometrical model of the breathing effect has already been worked out. The numerical analysis and experimental research have been fulfilled. The flow conditions of the breathing effect during the DCR process, the temperature field in suction chamber and cylinder after some breathing effect cycles, and the capacity regulation results using DCR method are obtained. FLUENT is used to compute temperature variation after some periods of regulation. It is found that after 20 periods of regulation the suction temperature is about 32K higher than the one in normal process of compressor. Through the numerical analysis and experiment, it could be concluded that the temperature rise resulted from the breathing effect affects the suction and discharge temperature, capacity and energy consumption. Based on the results of this work, the performance of reciprocating compressor could be improved by eliminating the influence of breathing effect.

Key words: Heat transfer; CFD; Breathing effect; Reciprocating compressor

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INTRODUCTION

Large reciprocating compressors, characterized by high efficiency throughout a wide range of output pressure and variable conditions, are important equipment in various process industries, and the energy consumption of most reciprocating compressor is tremendous. For example, compressors needed in a 1×10^7 t/a petroleum refining unit or in a 1×10^6 t/a ethylene processing unit have input power of more than 2,000 kW, and some even need more than 3,000 kW. Therefore, higher energy efficiency is the essential factor in compressor operation to save energy in the economic development. Adopting proper method of capacity control is an important means to realize energy saving. Based on this point, and considered the requirements of advanced capacity regulation proposed by process industries, Duty Cycle Regulation emerges. According to Gu et al. (2011), the DCR is based on the full-stroke suction valve opening method, and uses the duty cycle regulation on the pulse sequence instead of pressure-feedback on/off control. The core idea of DCR is: a control cycle consists of certain number of cycles, and in each cycle, a set of servo apparatus acts to open or release the suction valves and make compressor unloaded or loaded alternately; the required capacity is obtained by changing the proportion of loaded working cycles in the whole control cycle, the stability of discharge pressure is obtained by modulating arrangement of loaded working cycles or unloaded working cycles in time sequence, and varying the length of control period could ensure the accuracy and speed of the capacity regulation.

However, the DCR method induces a kind of breathing effect, a phenomenon that gas suction and reflux repeated in the flow channel of the cylinders, valves and suction chamber, which is the forced flow in the form of quasi-periodic back and forth under locomotive and complex boundary condition. The nature of the breathing effect is the thermodynamic process which combines non-steady flow with complex heat transfer. Therefore, the suction temperature rising is the first result of the breathing effect, then the discharge temperature and the power of compressor rise, and then the lubrication inside the cylinder gets worse. These consequences would decrease compressor capacity. Because of these bad influences of the breathing effect on the performance of compressor, further theoretical analysis and numerical simulation are necessary for successful implementation of the DCR method.

In this paper, in order to acquire the influence of breathing effect on reciprocating compressor, the commercial software FLUENT is used to simulate the breathing effect. By analyzing the simulation results, the flow conditions of the breathing effect during the DCR process and the temperature variation in the suction chamber and cylinder after some breathing effect cycles are obtained and then some conclusions are concluded.

The immediate consequence of the breathing effect is the rise of temperature of breathed gas. Theoretical analysis indicates that the main reasons for temperature rising are as following.

a) The overlying of gas flow losses. The energy loss resulted from the resistance of gas flow, especially the valve resistance, would be transferred into thermo-energy of gas. With the repentance of breathing effect, the thermo-energy of gas increases, making the temperature of gas and channel walls rise gradually. On the other hand, the temperature rising would result in gas viscosity rising, and which increases the flow resistance and forms vicious circle.

b) High-pressure leakage. Some cylinder arranges different pressure stage at the different sides of the piston, which inevitably causes leakage of high temperature gas from the high-pressure chamber to the lower one. If the breathing effect exists in the low-pressure chamber unfortunately, the effect of high-pressure leakage on the temperature of gas is more obvious.

c) The heating of cylinder walls. Regular work and breathing effect alternatively emerge in DCR regulation process. For a regularly-running compressor, the compressed high-temperature gas heats the cylinder walls continually. But when the compressor work cycle is regulated by DCR, the heated cylinder walls transfer heat to the gas, which is the main reason of temperature rise. Additionally, although the compressor does not compress gas when running in unloaded working cycle, heat out of the friction between cylinder wall and piston rings could not be ignored; based on Yu et al. (2011), the friction work on the piston rings usually occupies 38%-45% of entire friction work of a reciprocating compressor.

Among the three reasons above, the third reason is the most important factor that makes the gas temperature rise during the breathing effect process. According to Nie et al. (2012), the temperature rising has many influences on compressor performance, such as security, capacity, and power consumption etc.. In particular, the temperature rising in suction chamber is an important one since this temperature immediately affects the discharge temperature of the first several loaded working cycles after unloaded working cycles.

1. EQUATIONS

During the process of breathing effect, the main reason causing temperature rise of breathed gas is being heated by the cylinder. This paper only analyses the heat transfers equation of gas inside the cylinder. The heat transfer of gas in the cylinder is calculated by the first law of thermodynamics. When deducing the mathematical model of the thermodynamics process of cylinder gas, there are three assumptions: a) the gas state parameters are the same at any point in cylinder at any time; b) the potential and kinetic energies are negligible; c) air is ideal gas, so its specific heat is constant.

In this analysis the kinetic and the potential energies are assumingly negligible, so the energy of cylinder gas is the internal energy. The first law of thermodynamics of cylinder gas could be expressed as follows:

$$\frac{\mathrm{d}Q}{\mathrm{d}t}-\frac{MRT}{V}\frac{\mathrm{d}V}{\mathrm{d}t}+c_P T\frac{\mathrm{d}M_{svo}}{\mathrm{d}t}+\frac{u_{svo}^2}{2}\frac{\mathrm{d}M_{svo}}{\mathrm{d}t}-c_P T\frac{\mathrm{d}M_{dvi}}{\mathrm{d}t}=Mc_v\frac{\mathrm{d}T}{\mathrm{d}t}+c_v T\left(\frac{\mathrm{d}M_{svo}}{\mathrm{d}t}-\frac{\mathrm{d}M_{dvi}}{\mathrm{d}t}\right), \qquad (1)$$

where M_{svo} is the outlet mass of suction valve, M_{dvi} is the inlet mass of discharge valve, and Q is the heat exchanged between cylinder and surroundings.

Because of the temperature difference between gas and cylinder wall, the heat Q is exchanged by convection. In dt time, the heat transferred from wall dF to gas is calculated from

$$\mathrm{d}Q(F,t)=\alpha(F,t)\left[T_w(F,t)-T(t)\right]\mathrm{d}F\mathrm{d}t, \qquad (2)$$

where dQ is the heat transferred through area of element dF, area F is the function of piston displacement, α is the heat transfer coefficient between gas and cylinder wall, T is gas temperature, and T_w is the temperature of wall dF.

If the $\alpha(F,t)$ of each point of cylinder wall is replaced by average heat transfer coefficient $\alpha(t)$, the heat transferred from entire area F to gas in dt time could be described by

$$\mathrm{d}Q(t)=\int_{F(t)}\mathrm{d}Q(F,t)\mathrm{d}F=\alpha(t)\cdot\mathrm{d}t\sum_{i=1}^{3}\int_{F_i(t)}\left[T_s(F,t)-T(t)\right]\mathrm{d}F, \tag{3}$$

where $F_i(t)$ is the area that exchanges heat with gas, when i = 1, 2, 3, which refers to cylinder mirror area, piston surface area, and cylinder cover surface area respectively.

If integrals are calculated out, Equatioin (3) could be rewritten as

$$\mathrm{d}Q(t)=\alpha(t)\cdot\mathrm{d}t\left\{\left[\overline{T_1}(t)-T(t)\right]F_1(t)+\left[\overline{T_2}(t)-T(t)\right]F_2(t)+\left[\overline{T_3}(t)-T(t)\right]F_3(t)\right\}, \tag{4}$$

where $\overline{T_i}(t)$ (i = 1,2,3) is the average temperature of the area $F_i(t)$, and $\overline{T_i}(t)$ is obtained with

$$\overline{T_i}(t)=\frac{\int_{F_i(t)}T_s(F,t)\mathrm{d}F}{F_i(t)}. \tag{5}$$

Generally, the α in Equations (2)-(4) could be defined as a function of temperature, pressure, and average piston speed, that is

$$\alpha(t)=f\left[V_P,p_g(t),T_g(t)\right], \tag{6}$$

where $P_g(t)$ is gas pressure, $T_g(t)$ is gas temperature, and V_p is average piston speed. According to Wu (1989), the α calculation formula could be written in the following form

$$\alpha(t)=1.142(1+1.2467V_P)\left[p_g(t)^2T_g(t)\right]^{\frac{1}{3}}, \tag{7}$$

or

$$\alpha(t) = 1.639Vp[P_g(t)T_g(t)]^{\frac{1}{2}}. \tag{8}$$

2. NUMERICAL MODEL

According to the principles of DCR system, during the unloaded cycles, the suction valve is fully opened, so the gas flows through the former discharge chamber, buffer tank, suction pipe, and suction chamber, finally into the cylinder. With the piston moving reversely, the gas in the volume would be sucked into the cylinder and be exhausted out the cylinder again just like breathing. The working cycle with breathing effect is shown by solid lines in Figure 1.

2.1 Physical Modeling

In this paper, the simulation is based on a two-stage, single-acting reciprocating air compressor. The suction pressure is 1.0 atm, the discharge pressure of first-stage is 0.2 MPa, and the total discharge pressure is 1.0 MPa. The piston stroke and diameter of second-stage cylinder is 76 mm and 63 mm, respectively. The rotating speed of crank shaft is about 750 rpm.

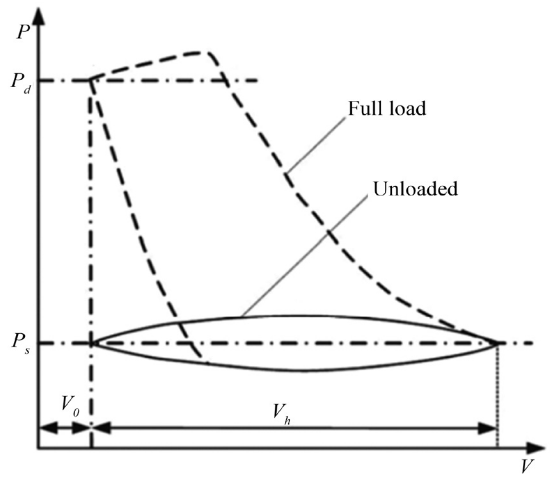

Figure 1
Working Cycle Leading To Breathing Effect

Since each flowing space is axi-symmetric, the analysis to two-dimensional section passing through the axis is reasonable, and this way would save computer resource and improve running speed (Nakano & Kinjo, 2008; Pereira et al., 2008; Subramanian et al., 2010). The physical model of breathing effect is shown in Figure 2.

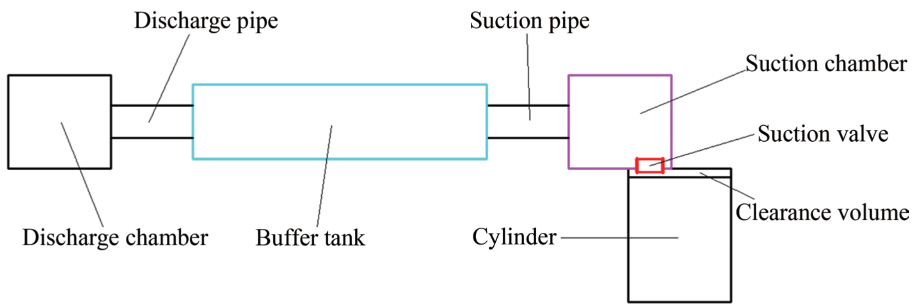

Figure 2
Physical Model of Breathing Effect

2.2 Dynamic Mesh Method

As is shown in Figure 2, the flow region of gas in the cylinder is continually changing due to the variation of crank shaft angle, so dynamic mesh is needed to establish the model. Based on the motion law of the reciprocating machinery, this paper adopts the method that combines

Spring-based Smoothing Method, Dynamic layering Method, and Local Remeshing Method, which is so-called combined dynamic meshing technique. This method not only is easy to realize, but also has good flexibility than other approaches. An important feature of this meshing method is that it could ensure mesh no dead centers and rotten surfaces in the process of stretching or compressing. Moreover, this meshing method could dispose the different arrangement of valves and cylinder and could set different time step in accordance with speed of the suction valve. Therefore, this method could meet the accuracy of simulation results for the breathing effect (Sun & Ren, 1995; Elhaj et al., 2008; Giacomelli et al., 2006).

2.3 Boundary and Initial Conditions

As mentioned above, in the model of breathing effect the gas flows in a closed volume, which has no inlet or outlet. All boundaries are set as WALL, including piston surface, cylinder walls, suction chamber walls, pipe walls, buffer tank walls, and discharge chamber walls etc.. The speed on walls meets the non-slip condition. The heat exchange between gas and walls is mainly realized through convection (Pereira et al., 2008; Peskin, 1999).

In the simulation of this paper, initial time is when the piston is at Bottom Dead Center (BDC), that is, the starting crank angle is 180°. Gas flow space connects suction chamber of second stage to discharge chamber of the first stage, and the intermediate space is the buffer tank, discharge pipe, and suction pipe. Since all of these are important heat sinks owing to air-cooling equipment of the compressor, the initial temperature field has a certain gradient, as shown in Figure 3.

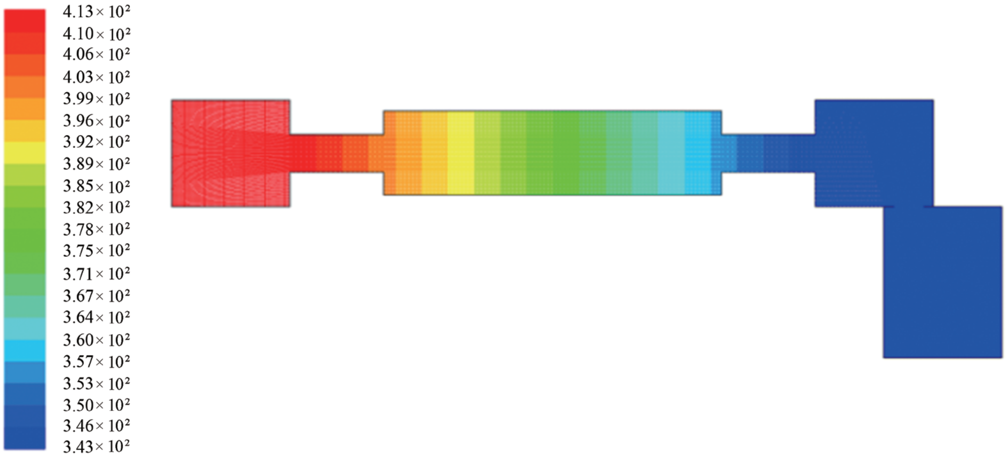

Figure 3
Initial Temperature Field

3. SIMULATION RESULTS AND DISCUSSION

This paper has computed three cycles and five cycles of the breathing effect. Figure 4 and Figure 5 shows the velocity contours after four and half cycles and five cycles, respectively. From Figure 4 and Figure 5, the situation of gas motion could be observed. The main spaces of gas flow are the cylinder, suction chamber and suction pipe. The velocity of gas in discharge chamber almost is zero during five-cycle process. Hence, the heat transfer amount in whole spaces is not so large. Despite the temperature of discharge chamber is higher than the other space, it does not immediately influence the gas temperature of the suction chamber during breathing effect process. This also illustrates why the main reasons for gas temperature rising of the suction do not include the heat transfer between flow spaces.

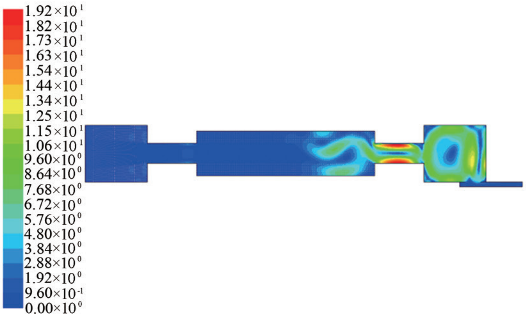

Figure 4
Velocity Contours After Four and Half Cycles

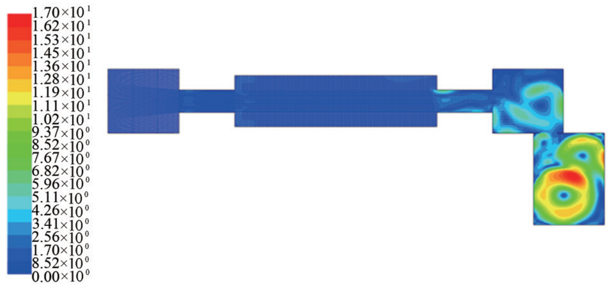

Figure 5
Velocity Contours After Five Cycles

The temperature contours after four and half cycles and five cycles is shown in Figure 6 and Figure 7. By comparing Figure 3 with Figure 6, the situation of temperature variation could be observed clearly. Figure 6 shows that the average temperature in the suction chamber has increased by about 40K after four and half cycles. This large temperature-rising is derived from several reasons. Firstly, the heat by convection from cylinder walls and piston surface to gas is considerable. Another reason is the overlying of gas flow losses, which has been explained in introduction above. Additionally, at this moment, the piston is at Top Dead Center (TDC), so the pressure in flow space is the highest during breathing effect. Accordingly, the temperature would rise with the pressure going up. Figure 8 shows the pressure contours at this moment.

Figure 7 indicates that temperature of gas in the suction chamber and cylinder is about 2-3K higher than that at initial time after five cycles. On the other hand, the gas temperature in other spaces such as suction pipe, buffer tank and discharge chamber decreases because of the air-cooling equipment. As Figure 7 shown, however, the cooling effect is not obvious during the breathing effect cycles due partly to that the heat loss through air-cooling equipment is smaller comparing with the heat obtained from heating walls, and gas flow loss etc..

As discussed above, the temperature of gas in the suction chamber is the most important factor affecting the compressor performance in first several loaded cycles after unloaded cycles. The variation of average temperature in the suction chamber is shown in Figures 9-10.

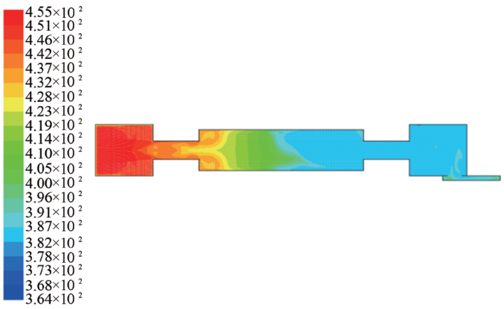

Figure 6
Temperature Contours After Four and Half Cycles

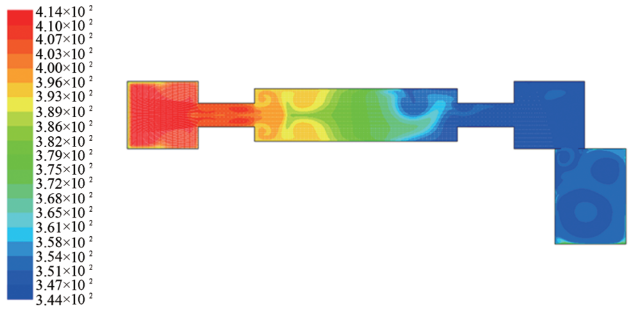

Figure 7
Temperature Contours After Five Cycles

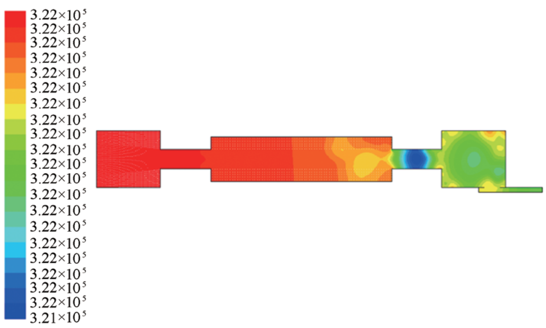

Figure 8
Pressure Contours After Four and Half Cycles

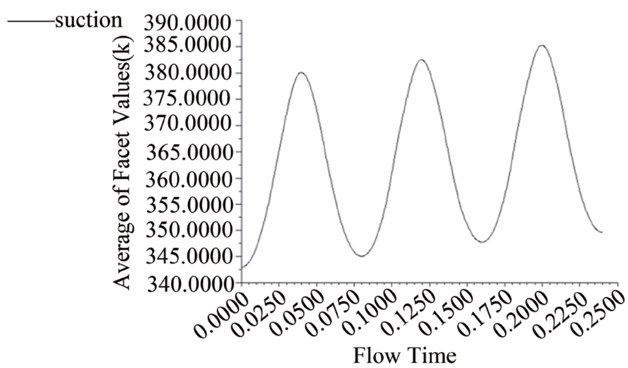

Figure 9
Average Temperature in Suction Chamber During Three Cycles

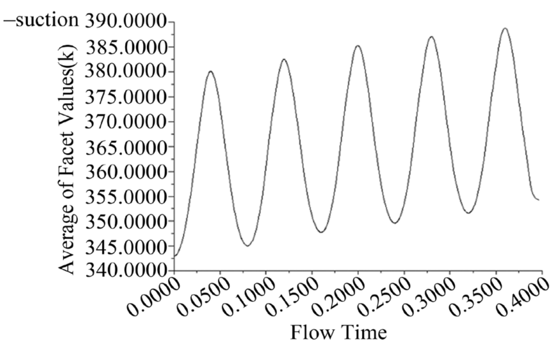

Figure 10
Average Temperature in Suction Chamber During Five Cycles

As shown in Figure 9, from unloaded to loaded after 3 breathing effect cycles, the average temperature of the gas in the suction chamber would increase 5K. This means that the average temperature of the gas in the suction chamber would increase 1.67K within one breathing effect cycle. Similarly, Figure 10 also shows that the rising magnitude of average temperature is about 1.6K within each breathing effect cycle. The average temperature has the same variation trend as shown in Figure 11. According to this result, the average temperature of gas in suction chamber would increase by about 32K during 20 cycles. If the compressor switched to full-load cycles at this moment, the suction temperature would rise by about

32K. This temperature rise's influence on compressor performance could not be neglected.

The temperature variation of valve is another crucial problem during the breathing effect cycles. Figure 12 shows temperature of the section through valve at the end of two and half cycles. As could be seen from Figure 12, the temperature distribution in this section is not uniform and the highest temperature is about 393.5K. This highest temperature would rise with the increasing of time of breathing effect cycles.

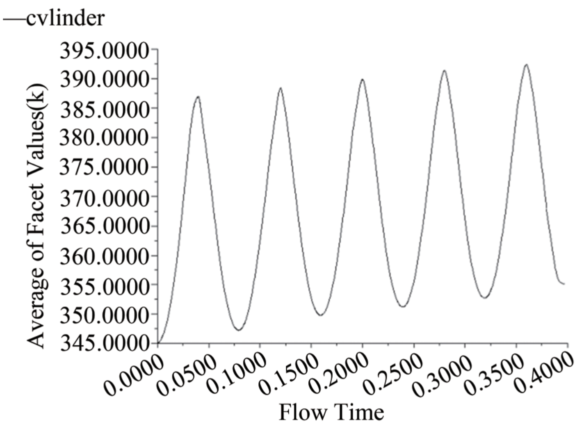

Figure 11
Average Temperature in Cylinder During Five Cycles

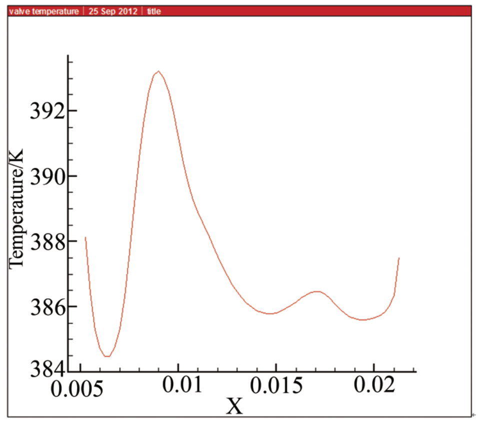

Figure 12
Temperature of the Section Through Valve (K)

4. EXPERIMENT AND ANALYSIS

Known from Chapter 4 of this paper, during the capacity regulation process by applying DCR regulation system, the existence of breathing effect would make the breathed gas temperature rise. As for the air compressor for this paper, the suction gas temperature would increase 32K after 20 regulating cycles. This study, without loss of generality, has made experimental verifications on the above simulation results when the exhaust gas pressure is 0.5 MPa and the regulation period is 30 working cycles. The experimental results of gas temperature value in the second stage suction chamber are shown in Figure 13.

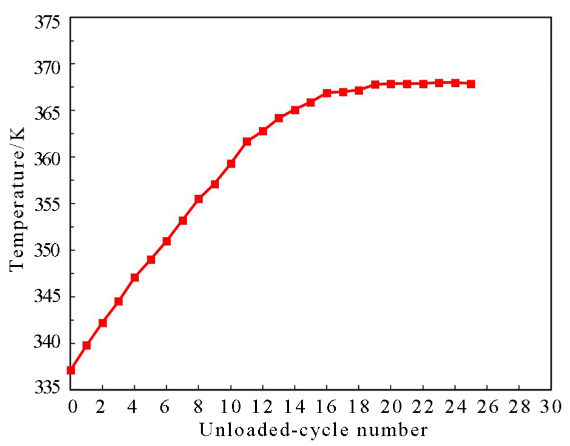

Figure 13
Temperature Rise of Gas Inside Suction Chamber During DCR Regulation

It could be seen from the Figure 13 that the gas temperature gradually increase with increasing of the unloaded-cycle number, the temperature of gas inside suction chamber would increase 30.5K after 20 cycles, which is consistent with the simulation results in this paper.

It could also be seen from Figure 13 that the speed of temperature rise decreases, which is because that the mass of the intake gas in the cylinder decreases when the temperature rises. The unloaded indicated work decreases, while the heat transfer coefficient of the passing walls increases at the same time, then the temperature rise range decreases.

After more than 20 cycles, the gas temperature would eventually become a stable temperature. However, in consideration that if the required compressed gas temperature could not exceed the upper limit of 453K when turned into the first loaded cycle after the completion of unloaded cycle, the initial gas temperature could not exceed 339K. Therefore, the number of continuous unloaded-cycle number could not be greater than 15 when adopting DCR regulation method; as well as consideration of the mechanical properties, response time and other factors of the capacity regulation system, generally, the continuous unloaded regulating working cycle number could not be less than 5. Therefore, the continuous unloaded regulating cycle number should be 5-15 when adopting DCR regulation method. When the ambient temperature is higher, in order to prevent the temperature of the suction chamber from too high, additional water cooling for suction chamber should be considered.

CONCLUSION

This paper presents the results of simulations and experiments on the breathing effect of a reciprocating compressor under Duty Cycle Regulation (DCR). The commercial CFD package Fluent was employed. The numerical model and DCR experimental test platform have been built. The flow conditions of the breathing effect during the DCR process and the temperature variation in the suction chamber and cylinder after some breathing effect cycles have been obtained. The following major conclusions could be drawn:

The average temperature would increase by about 1.6K within each breathing effect cycle. As to the reciprocating air compressor in this paper, the suction temperature would rise by 32K after 20 breathing effect cycles, and the capacity would decrease by 9%. The gas temperature gradually increases with the increase in the unloaded-cycle number, the temperature of gas inside the suction chamber would increase 30.5K after 20 cycles, which is consistent with the simulation results. The continuous unloaded regulating cycle number should be 5-15 when adopting DCR regulation method.

In order to solve the problems derived from breathing effect, for instance, suction temperature rising, setting an inter-cooler between the first stage and the second stage may be a good method. The power of unloaded cycles would change with the breathing effect going on since the gas temperature has effects on the compressor power. A further observation about the pressure change in the cylinder would help to find out the indicating work for each unloaded cycle. This should be carried on in the future studies.

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APPENDIX: NOMENCLATURE

Q	Quality of heat [J]	$\boldsymbol{T_g}$	Gas temperature [K]
T	Absolute temperature [K]	Subscripts	
M	Mass [kg]	*svo*	Suction valve outlet
F	Area [m^2]	*dvi*	Discharge valve inlet
c_p	Specific heat at constant pressure [J/(kg·K)]	w	Wall
c_v	Specific heat at constant volume [J/(kg·K)]	g	Gas
p_g	Gas pressure [Pa]	P	Piston