1994

Analysis of Enhanced Heat Exchange Caused by Pressure Pulsation in Cooler of Reciprocating Compressor

S. Sun
Xi'an Jiaotong University

J. Sun
Xi'an Jiaotong University

ANALYSIS OF ENHANCED HEAT EXCHANGE CAUSED BY PRESSURE PULSATION IN COOLER OF RECIPROCATING COMPRESSOR

Si-ying Sun and Jin-ju Sun
Department of Power Machinery, Xi'an Jiao-tong University,
Xi'an, Shaanxi Province, 710049, P. R. China

ABSTRACT

In the present paper, dynamic characteristics of the pressure pulsation and heat exchange have been analyzed in cooler of reciprocating compressor. Through the analysis of energy balance for an heat exchange element, the differential equations are established for the heat exchange element. Effect of the pressure pulsation on heat exchange has been considered in the cooler of the compressor by using a method of modified Reynolds number. The results indicate that the pressure pulsation can enhance heat exchange in the cooler.

INTRODUCTION

Many researchers at home and abroad have done the work on eliminating the gas pulsation and piping vibration in compressor system. However, the gas pulsation may have benefit in compressor system sometimes. For example, it is indicated that the gas pulsation may enhance heat exchange [1]. On the basis of establishing the energy balance equations of differential elements for heat exchange of cooler, and revising the Reynolds number, the present work shows that the pressure pulsation enhances heat exchange in cooler of compressor.

ANALYSIS FOR EFFECT OF GAS PULSATION ON HEAT EXCHANGE

It is well-known that the convection heat transfer is a combined result of fluid convection and heat conduction by the fluid own self. That is fluid has an ability of heat conduction, and the gas in cooler of compressor can not be made an exception. When the gas flow is located in the laminar state or fransitional state, the heat transferred by the macroscopic displacement is small due to the low velocity of the gas. Under the condition of gas pulsation, the heat conduction by the gas own self is largely increased. Thus, the heat exchange of cooler is enhanced. Fig. 1 is a schematic of the gas side of a water cooling cooler. When there exists no gas pulsation, the isothermal plane is O—O plane which is perpendicular to the gas flow direction. When there exists gas pulsation in pipe, displacement of the plane occurs. The plane becomes convex, and the plane is periodically vibrated. The O′-O′ is equilibrium position. The O_1-O_1 and O_2—O_2 are two exterminal position, as shown in Fig. 1. In comparison with the O—O plane, both the perpendicular temperature gradient and heat transfer plane are increased in the O′-O′ displacement plane [2]. In this case the axial heat conduction of the gas increased. When the amplitude of the gas pulsation is increased, the axial heat conduction of the gas is further enhanced. Moreover, the temperature difference of the gas is decreased at different position of the cooler, and the constituent of turbulent flow in the gas flow is enhanced. With the increase in the amplitude of the gas pulsation and the enhancement of the axial heat conduction, the gas temperature tends to be identical at different position of the cooler. If the amplitude of the gas pulsation is still further increased, the axial heat conduction of the gas can not be enhanced obviously.

COMPUTATION OF EFFECT OF GAS PULSATION ON HEAT EXCHANGE

Based on the previous analysis, it is known that the enhancement level of the pressure pulsation on the heat exchange is mainly dependent on the

Reynolds number and the amplitude of the pulsation pressure. In order to compute the effect of the pressure pulsation on the heat exchange in cooler of compressor, the above two factors are comprehensively considered, and a complementary criterion of Reynolds number is introduced. That is,

$$R_E' = A\sigma \tag{1}$$

where

$$\sigma = \frac{d}{2}\sqrt{\frac{\rho\omega}{\mu}} \quad \text{(Inertia criterion)}$$

$$A = 2P\sqrt{\frac{\rho\omega}{\mu}} \quad \text{(Amplitude criterion)}$$

and R_E' is the revised Reynolds number. d is the inner diameter of heat exchange pipe. ρ, and μ are the gas density and the viscosity, respectively. P and ω are the amplitude and the excited main frequency of the pulsation pressure, respectively.

Substituting the revised Reynolds number R_E' into R_E of heat exchange equation, effect of the pressure pulsation considered on the heat exchange in cooler can be solved. Experimental verification indicates that the revised criterion is acceptable [3]. In order to determine the amplitude P of the pulsation pressure, an acoustic wave theory is used.

HEAT EXCHANGE EQUATIONS

A counterflow shell-and-tube cooler widely used is analyzed in the present work. In addition, the energy belance equation is established for the heat exchange of differential element.

Mathematical Model

Let us consider the n th differential element of the cooler shown in Fig. 2, and the differential element enlarged is shown in Fig. 3. The energy balance analyses of the gas, cooling water, and wall of the heat exchange tube are conducted for the differential element. The following nondimentional equations are obtained:

$$N_1(T - T_g) = \frac{\partial T_g}{\partial X} \tag{2}$$

$$N_2(T - T_W) = V\frac{\partial T_w}{\partial \theta} - \frac{\partial T_w}{\partial X} \tag{3}$$

$$\frac{\partial T}{\partial \theta} = Ntu(\frac{1+R}{R})(T_g - T) + Ntu(1+R)(T_w - T) \tag{4}$$

where

$$\theta = (t - \frac{x}{u_g})\frac{m_g l_g}{mc}; \qquad X = x/l;$$

$$\frac{1}{Ntu} = m_g c_g(\frac{1}{\alpha_g f_g} + \frac{1}{\alpha_w f_w}); \qquad E = \frac{m_w c_w}{m_g c_g};$$

$$V = \frac{m_g c_g}{mc}(\frac{1}{u_g} + \frac{1}{u_w}); \qquad R = \frac{d_w f_w}{\alpha_g f_g};$$

$$N_1 = Ntu(\frac{1+R}{R}) = \frac{\alpha_g f_g}{m_g f_g};$$

$$N_2 = \frac{Ntu}{E}(1+R) = \frac{\alpha_w f_w}{m_w c_w};$$

where T is the temperature; θ is the rotary angle of crankshaft; Ntu is the dimensionless number of transfer units; R is the dimensionless conductance ratio; E is the dimensionless capacity rate ratio; X is the dimensionless tranfer function; x is the distance from gas entrance; u is the fluid velocity; t is the time; m is the mass flow rate of fluid; c is the unit heat capacity of fluid; α is the thermal conductance per unit heat transfer area; f is the heat transfer area. Moreover, the subscripts g and w represent the gas and water, respectively.

COMPUTING RESULTS

Fig. 4 shows the relation of the gas temperature at different position of the cooler with changing the rotary angle θ_c of the crankshaft. There are three kinds of working conditions. (1) No pressure pulsation; (2) Pulsation pressure is in the range of 0.3×10^4 to 0.7×10^4 Pa; (3) Pulsation pressure is 100 times the pulsation pressure of working condition (2). In the above working conditions

R_e is about 6000. If the value of R_E is not very large and the gas flow is located in the laminar state or transitional state, the results indicate that the pressure pulsation can make the gas temperature decreasing in different positions. With the increase in the amplitude of the pulsation pressure, the difference of the gas temperature in different positions is decreased. That is, the axial heat conduction is enhanced. If the pulsation pressure is further increased, the gas temperature in different positions along the axial direction tends to be identical.

CONCLUSIONS

The axial heat conduction of the gas is enhanced by the pressure pulsation in the cooler of compressor. Thus, the heat exchange of the cooler is enhanced. Moreover, the enhanced level is related to the value of the pulsation pressure and Reynolds number.

REFERENCES

[1] Y. A. Vidiakin, Pulsation and Vibration of Reciprocating Compressor, Machine Building Press, Leningrad, 1972.

[2] Y. Z. Cao, Mechanisms of inhanced heat transfer by fluid vibration in tube, Research Report, Beijing Aeronautic University, Beijing, 1988.

[3] J. J. Sun, Computer simulation and experimental studies on the heat exchange of a cooler in a reciprocating compressor, Msc Thesis, Xi'an Jiao-tong University, Xi'an, 1989.

[4] F. E. Romie, Transient response of the counterflow heat exchanger, Transactions of the ASME, Vol. 106, 1984, pp. 620—626.

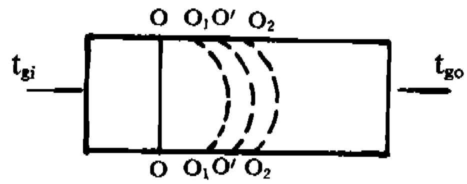

Fig. 1 Schematic of gas side

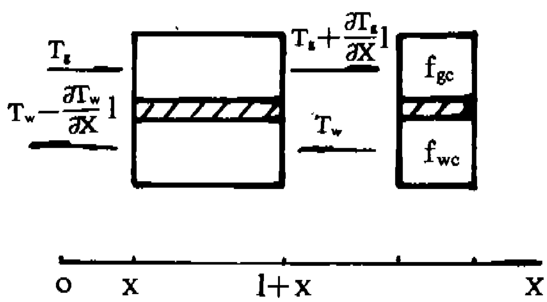

Fig. 2 Schematic of cooler

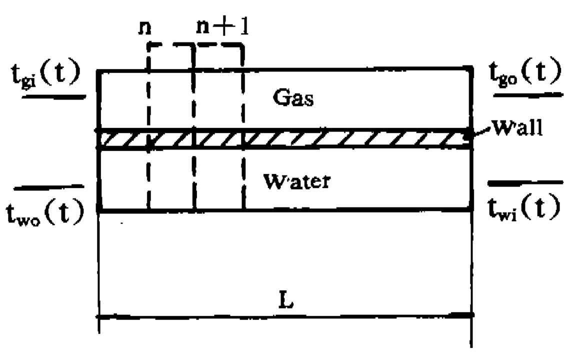

Fig. 3 Enlarged differential element

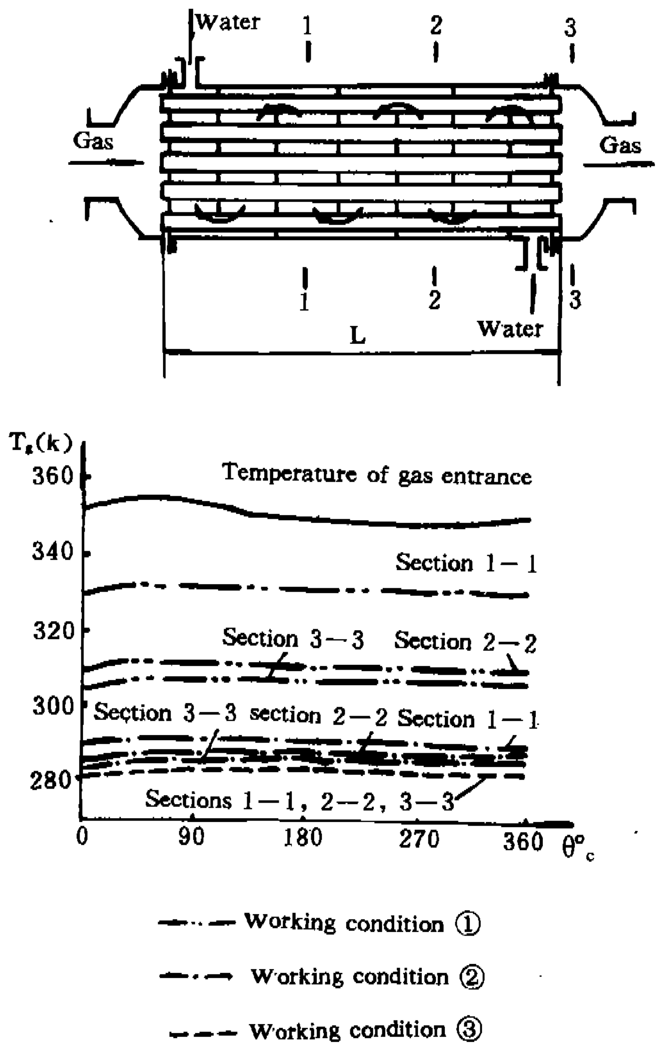

Fig. 6 Relation of gas temperature at different position of the cooler with changing the rotary angle of the crankshaft