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Investigation on non-equilibrium phase transition in wave rotor

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ABSTRACT

Wave rotor refrigeration is a new method of expansion refrigeration. Non-equilibrium phase transition is one of the key factors that influence the refrigeration performance of wave rotor. Since the process of phase transition is transient and high-frequency, there are challenges in related researches. At first, this paper traces the movement trajectory of small particles in wave rotor and confirms the existence of evaporation in wave rotor. Based on this, the numerical analysis model of wave rotor that takes into account the phase transition regarding evaporation as reverse process of condensation is built up. On this basis, the change of droplet radius, droplet number and liquefaction fraction in wave rotor is obtained. Besides, the influence of inlet pressure and humidity on refrigeration performance of wave rotor is found out. The results show that with the increase of pressure of HP inlet, the isentropic expansion efficiency increases and the increase becomes gentle. With the increase of relative humidity of HP inlet, the efficiency decreases and the decrease reaches 2.8% at maximum. The results can help optimize wave rotor performance and be used for reference to wet gas transient temperature change.

Étude sur la transition de phase en état de non-équilibre dans un rotor à ondes

Mots-clés: Rotor à ondes; Froid [artificiel]; Transition de phase; Non-équilibre; Onde de choc

1. Introduction

Parameters of the fluid that flows through strong discontinuity such as shock wave change instantly and the change process are non-equilibrium. When there is phase transition in the flow, the latent heat will complicate the flow then influence the performance (Tropea et al., 2007). However, since the phase transition is non-equilibrium and transient, it's hard to obtain accurate conclusions by means of both experiment and theory, which makes it an urgent problem to be solved. It can be seen that non-equilibrium phase transition widely appears in the fields such as thermo acoustic engine, ejector and so on (Dovgjallo and Shimanov, 2015; Yinhai and Peixue, 2018; Jakub et al., 2020).

Wave rotor is a new type of fluid equipment that uses movement of waves to realize energy transfer, as shown in Fig. 1. Wave rotor is composed of straight pressure oscillation tubes arranged around the rotational axis with two open ends. When the wave rotor rotates about the shaft, the passages connect inlets and outlets engraved on two stationary end plates in turn and resultant pressure waves are generated inside the passages. In this equipment non-equilibrium phase transition exists and keeps unsteady because of movement of waves. The work principle of wave rotor is different from that of traditional turbomachine. At first, wave rotor is used for pressurization. Since the 20th century, NASA has conducted experiments on wave rotor technology and gained significant experiences about the device design (Akbari et al., 2006; Wilson, 1993; Welch, 1996; Wilson and Paxson, 1996; Wilson, 1998). In recent years, DENG et al. from Tokyo University applied wave rotor pressurization technology in small scale devices and summarized the characteristics of internal flow and heat transfer (Deng et al., 2015). C-2 type of wave rotor pressurizer invented by WU and CHEN from the Institute of Mechanics of the Chinese Sciences Academy has been put into industrial operation and achieved good compression performance (Wen and Mingjing, 1982). Warsaw University of Technology investigated main flow features in wave rotor pressurizer using particle image

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Nomenclature

P	Pressure [MPa]
T	Temperature [K]
r_*	Critical radius of droplet [m]
r	Radius of droplet [m]
σ	Surface tension [N•m^{-1}]
ρ_l	Density of liquid phase of condensable component [kg•m^{-3}]
ρ_v	Density of gas phase of condensable component [kg•m^{-3}]
S	Supersaturation of condensable component [dimensionless]
γ	Adiabatic constant [dimensionless]
h_t	Latent heat of liquefaction [J•kg^{-1}]
φ	Non-isothermal correction factor [dimensionless]
J	Nucleation rate [m^{-3}•s^{-1}]
q_c	Coefficient of condensation [dimensionless]
M_m	Molecular mass of the condensable component [kg]
k	Boltzmann constant [J•K^{-1}]
p_{sr}	Surface saturation vapor pressure of droplet [MPa]
p_s	Saturated vapor pressure [MPa]
N	Number of droplets in a unit mass of gas [kg^{-1}]
Y	Mass of liquid in a unit mass of gas [dimensionless]
r_2	Largest radius of droplet of assumed uniform distribution [m]
η	Isentropic expansion efficiency of wave rotor [dimensionless]
T_1	Temperature of HP inlet of wave rotor [K]
T_2	Temperature of LT outlet of wave rotor [K]
p_1	Pressure of HP inlet of wave rotor [MPa]
p_2	Pressure of LT outlet of wave rotor [MPa]
H	Liquefaction fraction [dimensionless]
L	Axial coordinates in the pressure oscillation tube [mm]
ΔH_r	Real enthalpy drop
ΔH_{is}	Isentropic enthalpy drop

velocimetry (PIV) technology which helped the validation of computational fluid dynamics (CFD) calculations (Krzysztof et al., 2017). Since the pressurization wave rotor is widely used in the combustion field where the temperature of fluid rises, non-equilibrium phase transition isn't involved in the relevant investigation about the device.

As for the study of wave rotor refrigeration, in the 21st century, HU et al. proposed to use wave rotor technology to realize refrigeration and developed wave rotor refrigeration technology which can be used in areas such as dehydration of natural gas (Yuqiang, 2010; Yuqiang et al., 2009). During the process of refrigeration, the temperature of fluid declines and phase transition happens generally which influences the flow field and refrigeration performance (Dapeng et al., 2016b; 2016a; Lin et al., 2018). Besides, to avoid freeze blockage in gathering and transportation system of natural gas, antifreeze such as ethanol is injected into pipelines, this requires wave rotor refrigerator used in natural gas field able to work in two-phase flow stably (Yuqiang et al., 2010). Nevertheless, there are few investigations on phase transition in wave rotor and most of the investigations focus on theoretical analysis. Theoretical analysis from the researchers such as Zhao, Xu and so on is not systematic enough (Jiaquan, 2013; Siyuan, 2015). In experimental aspect, researchers in Eindhoven University of Technology built up an expansion shock tube experimental platform and measured the radius of droplets by means of 90 ° Mie scattering and extinction (Looijmans and Dongen, 1997; Fransen et al., 2012).

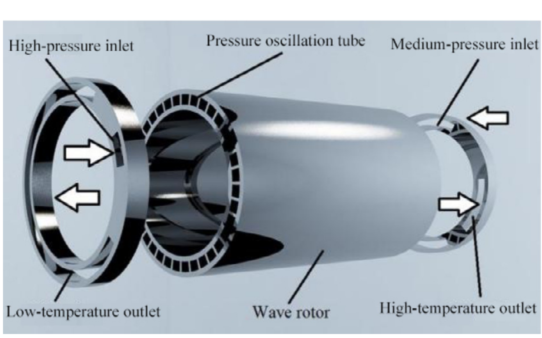

Fig. 1. Wave rotor refrigerator.

In the present study, the existence of evaporation is confirmed by investigating the movement of particles. The numerical condensation model and evaporation model using the growth of droplets reversely are established to analyze internal flow that contains phase transition in a wave rotor. Experiments and numerical simulations are conducted to find out the influences of operation parameters on the refrigeration and liquefaction performance of the wave rotor.

2. Movement analysis of particles

Fig. 2 shows a 2D ideal wave diagram and temperature–entropy diagram of the wave rotor. It could explain the working principle of the wave rotor refrigeration technology. When the wave rotor is working, pressure oscillation tubes move upward periodically. Firstly, when high-pressure (HP for short) inlet connects a tube, HP gas injects into the tube and a series of compression waves appears which converge to shock wave S1, moving to the right and compressing the original gas in the tube. At the same time, a series of expansion waves generate in the contact surface of HP gas and the original gas, which will propagate to the left end of the tube, cooling HP gas. Then, the pressure oscillation tube leaves HP inlet and a series of expansion waves E1 appear to expand the gas in the tube. The expansion waves make the pressure and temperature of the gas in pressure oscillation tube decrease. Then, pressure oscillation tube starts to connect with high-temperature (HT for short) outlet, compressed gas in the tube discharges from HT outlet. Simultaneously, S1 reaches HT outlet and reflects a series of expansion waves E2. The result is that the temperature and pressure of gas both decrease again. At the same time, due to factors such as wall reflection of shock wave, reverse compression of HT outlet, and HT backflow, the compression wave RC1 may generate and converge into reverse shock wave RS1. The gas flows out of the wave rotor through HT outlet. After exchanging heat with cool water, it flows back into the pressure oscillation tube through medium-pressure (MP for short) inlet. Finally, both sides of the pressure oscillation tube start to connect with low-temperature (LT for short) outlet and MP inlet, and the LT gas flows out through LT outlet pushed by differential pressure between LT outlet and MP inlet. So a cycle of work in the wave rotor is completed. To make sure LT gas could be completely expelled, the number and length of LT outlet and MP inlet should be increased. T-S diagram shows that HP gas expands into LT gas through expansion waves E1 and E2, while expansion work is used to compress MP gas into HT gas of next cycle. And HT gas changes into MP gas through a heat exchanger.

ANSYS Fluent 14.5 in double precision is used to perform CFD calculations in this paper. As for the geometric model of the wave rotor, calculation using the 3D model needs high computing power and extended simulation time. And the ratio of length to width of pressure oscillation tube in the wave rotor is larger than 10 and the

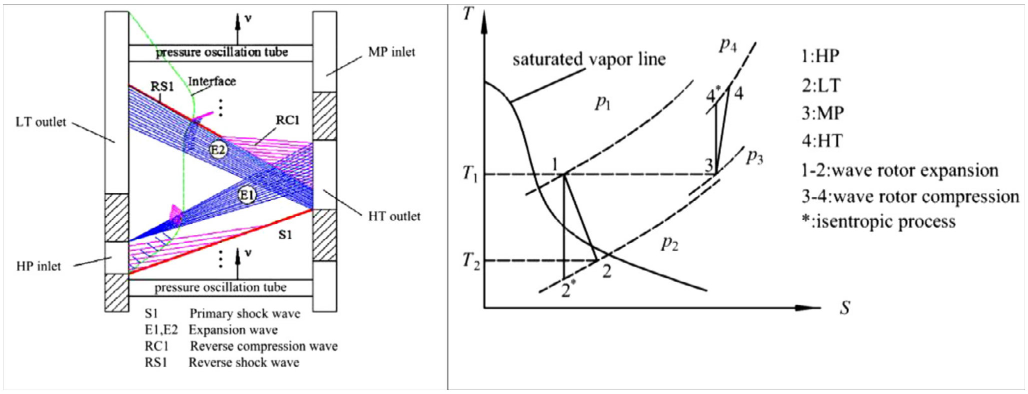

Fig. 2. Wave diagram and temperature–entropy diagram of wave rotor.

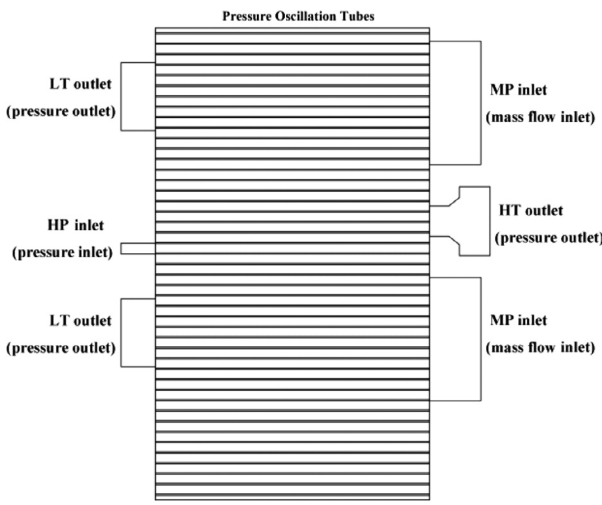

Fig. 3. 2D geometric model of the wave rotor.

Table 1
The solution strategy of the CFD simulation.

Solver type	Density-Based
Turbulence model	Realizable k-epsilon
Near-wall treatment	Standard wall function
Flux type	AUSM
Spatial discretization	Second order upwind
Transient formulation	Second order upwind
Converging criteria	0.001
Time step size	1e-6s

rotational speed of the wave rotor is relatively low, therefore, the centrifugal force and other effects caused by the three dimensional rotation of rotor can be ignored and the flow in the tube can be regarded as planar flow. Based on this, a 2D geometric model of the wave rotor, shown in Fig. 3, is used in numerical simulation. The length and width of the pressure oscillation tube are set to be 400 mm and 13 mm, respectively. The velocity of the tube is set to be 33 m•s^{-1} upward, which is corresponding to the rotational speed of 3000 rpm and the radial position of pressure oscillation tubes of 105 mm, for matching the pressure waves. Fig. 3 shows the boundary conditions of inlets and outlets.

The pressure of HT outlet and LT outlet is set constant 0.10 MPa. MP inlet is defined as the mass flow inlet to ensure its mass flow rate is equal to that of HT outlet. Flow temperature in MP inlet is defined as 298 K. The fluid is defined as ideal air, and the solution strategy is presented in Table 1. Before the simulation study, a grid independence test was conducted, as presented in Table 2.

Table 2
Grid independence test.

Grid size/mm^2	Total amount of meshes	Pressure/kPa
1 × 1	280,857	23.2
1.5 × 1.5	128,732	23.2
2 × 2	74,802	22.8
2.5 × 2.5	44,088	22.8
3 × 3	29,437	22.9
3.5 × 3.5	24,553	22.7

Pressure in the middle position of the oscillation tube connecting HP inlet of different grid size was chosen to comparatively validate. The results of 1 × 1 mm^2 grid and 1.5 × 1.5 mm^2 grid are the same. In order to save computing time, 1.5 × 1.5 mm^2 grid is confirmed feasible for numerical simulation.

The periodical boundary condition is used to simulate the periodical rotation of the wave rotor. The 2D geometric model has been confirmed valid by the previous study (Dapeng et al., 2018).

From the ideal wave diagram, when gas in HP inlet contains condensable components, condensation generally happens first when gas injects into the wave rotor. To find out the path of droplets in the wave rotor, the Discrete Phase Model can be used. In a previous study of droplets, the diameter of most condensed water droplets was found less than 10 μm (Xisheng, 2004). Therefore, the diameter of the droplets is set to be 10 μm and 1 μm respectively in this study.

The pressure and temperature of the HP inlet are 0.40 MPa and 298 K, respectively. Fig. 4 shows the calculation results including the temperature contour and the path of droplets. The area where temperature is lower than 298 K is called the LT region; otherwise, it is called the HT region. From the temperature contour, it can be seen that a contact interface separates the LT region and the HT region.

As shown in Fig. 4, there are droplets getting through the contact interface and stay in the HT region. With the increase of the diameter of droplets, the number of droplets that get through the interface rises. As for droplets with 10 μm diameter, a significant amount of droplets are expelled through HT outlet. This proves that droplets in the wave rotor will be heated and evaporate. Besides, droplets can't be fully exhausted out of the LT outlet after each work cycle of the wave rotor. There are droplets remain in the wave rotor in the next work cycle. The droplets will be heated by gas compressed by the newly generated shock waves and evaporate. Therefore, when HP inlet gas contains condensable components, there is not only condensation in the wave rotor but also evaporation.

Similar to condensation, evaporation affects the mass and energy transfer of gas in the wave rotor. The wave movement and

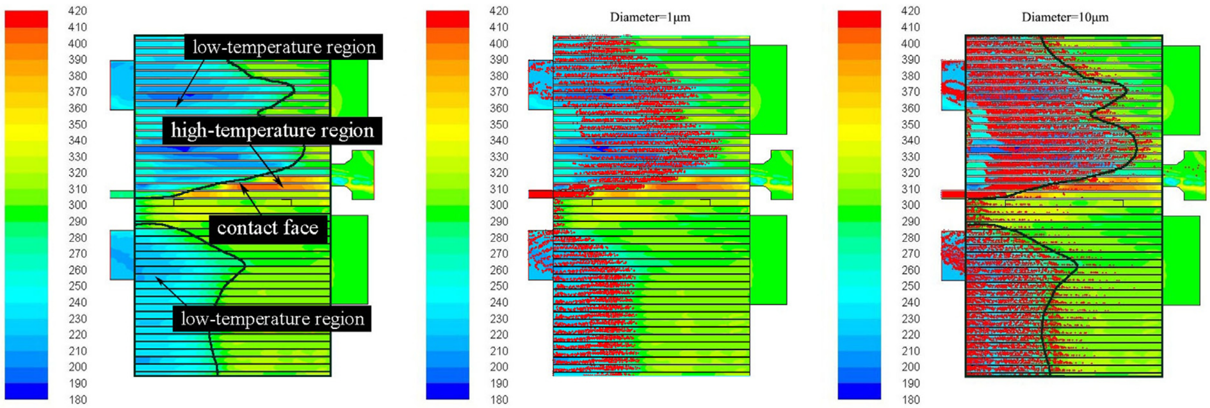

Fig. 4. Results of numerical simulation including temperature (K), particle path with diameter 1 μm and particle path with diameter 10 μm.

refrigeration performance of the wave rotor inevitably get affected. Therefore, evaporation and condensation of droplets both need to be taken into consideration while analyzing the performance of the wave rotor.

3. Experiment platform and numerical model

Because of rotation and high internal flow speed (>200 m•s^{-1}) in the wave rotor, phase transition such as condensation or evaporation in the wave rotor is difficult to observe experimentally. Besides, the previous numerical model of wave rotor (Jiaquan, 2013; Siyuan, 2015) involving phase transition did not consider evaporation. Therefore, to better investigate the effects of phase transition on the refrigeration performance of the wave rotor, a numerical phase transition model that includes condensation and evaporation, as well as the companion verification experimental platform, is hereby established.

The difficulty of establishing a numerical phase transition model is to assure the transport of mass and energy between droplets and water vapor. Besides, condensation and evaporation need to be calculated separately. Theoretically, condensation of droplets is divided into nucleation and growth of droplets and evaporation can be regarded as reverse growth of droplets.

So far, there have been three numerical methods to describe the droplet flow: Euler/Lagrange, Euler/Euler and Moment based methods. Compared with the other two methods, Euler/Euler, which defines transport equation of gas and liquid phase respectively, is more applicable for the calculation in complicated unsteady steam flow. Previous studies obtained satisfying simulation results of unsteady condensation in wet steam by using Euler/Euler (Gerber and Kermani, 2004; White and Hounslow, 2000). Therefore, in this study, Euler/Euler method is used to describe the condensation and evaporation process in the wave rotor.

As for the numerical model of phase transition, the following assumptions are made. Velocity slip between droplets and gas is neglected, volume and interaction of droplets are neglected, the internal temperature of droplets is homogeneous, and there are no foreign particles such as ionic and dust in the flow. Therefore, the nucleation process could be regarded as homogeneous nucleation which merely happens in the conditions without foreign particles. The homogenous nucleation model used in high-speed flow is the model proposed by Frenkel and modified by Feder (Young, 1982):

$$J = \frac{q_c}{1+\varphi}\left(\frac{\rho_v^2}{\rho_l}\right)\left(\frac{2\sigma}{\pi M_m^3}\right)^{1/2}\exp\left(-\frac{4\pi r_*^2\sigma}{3kT}\right) \tag{1}$$

$$r_* = \frac{2\sigma}{\rho_l RT\ln S}\frac{2(\gamma-1)}{\gamma+1}\left[\frac{h_t}{RT}-\frac{1}{2}\right]^2 \tag{2}$$

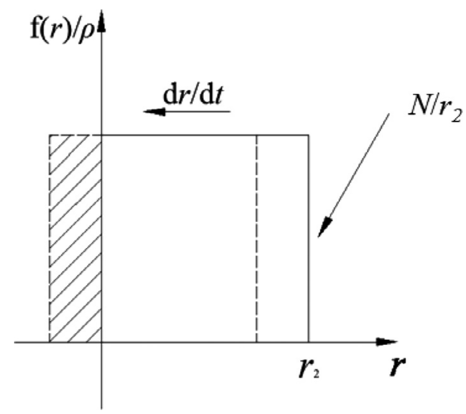

Fig. 5. Uniform distribution of droplets.

$$\varphi = \frac{2(\gamma-1)}{\gamma+1}\left[\frac{h_t}{RT}-\frac{1}{2}\right]^2 \tag{3}$$

After nucleation, when the radius of droplets is larger than critical radius, the droplets can grow by capturing vapor molecule. As for the description of the growth of droplets, Hertz-Knudsen equation has been proved valid (George, 1982; Holyst et al., 2015). The equation is expressed as:

$$\frac{dr}{dt} = \frac{p_v - p_{sr}}{\rho\sqrt{2\pi RT}} \tag{4}$$

$$p_{sr} = p_s\exp\left(\frac{2\sigma}{\rho_l RT}\right) \tag{5}$$

When the gas becomes superheated because of compression waves or heat transfer, the droplets in the gas start to evaporate. To describe the evaporation, the droplets growth model is introduced reversely to thoroughly simulate the phase transition process in the wave rotor for the first time.

Except for the droplet growth equation, calculation of evaporation also relies on the distribution of droplets. In previous studies on droplet distribution, Gaussian distribution, normal distribution, Gamma distribution, Beta distribution, and semi-normal distribution were taken into consideration (John et al., 2007). In this study, during the calculation of phase transition, only the mass of fluid and quantity of droplets could be obtained, so the distribution of droplets is assumed uniform, which is shown in Fig. 5. The distribution function is:

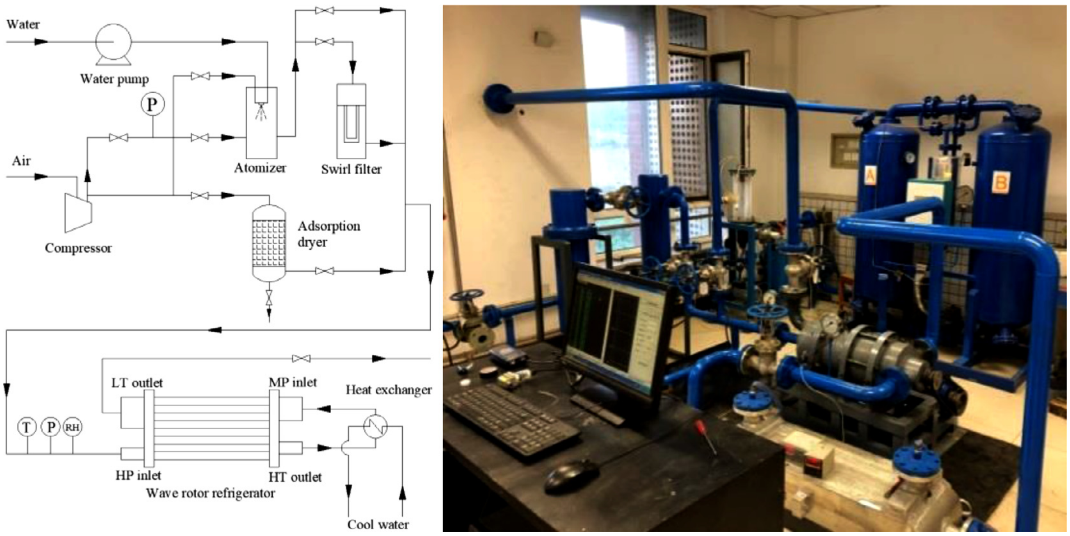

Fig. 6. The schematic and photograph of the experimental platform.

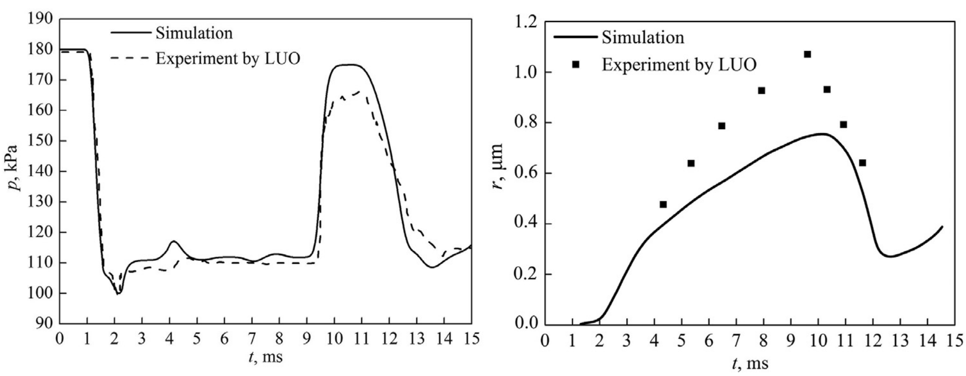

Fig. 7. Validation for evaporation and condensation model.

$$f(r) = N/r_2 \tag{6}$$

$$r_2 = \left(\frac{3Y}{N\pi\rho_l}\right)^{1/3} \tag{7}$$

When evaporation happens, the distribution function moves left at the speed of droplet growth. The size of the shadow square caused by the movement is the reduced number of droplets. The non-equilibrium phase transition process is achieved by a user-defined function (UDF) in Fluent.

The corresponding experimental platform is established in order to obtain the refrigeration performance of the wave rotor.

Fig. 6 shows the schematic and photograph of the experimental platform. The compressor compresses fresh air to obtain HP air, of which the pressure of is constant at 1 MPa. Throttle valves are used to reduce pressure in order to obtain the pressure of HP inlet of the wave rotor. Then HP air flows into two pipelines. In one pipeline, after the air gets humidified in the atomizer and removes the liquid water particles in the filter, saturated moist air is obtained. In another pipeline, the air gets dried by the adsorption dryer. By controlling and mixing the mass flow of both pipelines, the relative humidity of HP inlet gas can be adjusted. Moisture analyzer is set on HP inlet so relative humidity can be monitored. A heat exchanger is installed to make sure the gas flows out of the HT outlet could exchange heat with cold water before circulating back to MP inlet.

In pulse expansion wave tube (Looijmans and Dongen, 1997), phase transition including non-equilibrium phase transition is similar to that found in the wave rotor. So it could be used to verify the phase transition model in this study. LUO monitored the change of radius of droplets and pressure fluctuations experimentally in the pulse expansion tube (Xisheng, 2004). As shown in Fig. 7, the results of CFD simulations using condensation and evaporation model in this study are compared with experimental data by LUO. The comparison of pressure fluctuations between experiments and CFD results shows that the wave motion with phase transition could be well described by CFD calculations. The changing trend of the droplet radius in the CFD results is consistent with the experimental results, and the values of the droplet radius are found close to the measured values with the maximum error no more than 30%. The results indicate the phase transition model established in this study is feasible.

Because of the assumptions and simplification that the phase transition model bases, the calculation result deviates from experimental results in a certain degree. Since the influence of phase transition on pressure fluctuation is less than that of main moving pressure waves, the difference of droplet radius is more obvious comparing with the difference of pressure fluctuation.

In order to compare the difference between the model with evaporation and without evaporation in the pressure oscillation tube, the single condensation model was simulated (Siyuan, 2015). The model considering evaporation is referred to as WE (With Evaporation) and the model only considering condensation is referred to as OC (Only Condensation). The distributions of N in the pressure oscillation tube that totally connected with HP inlet are shown in Fig. 8. There are a large number of condensa-

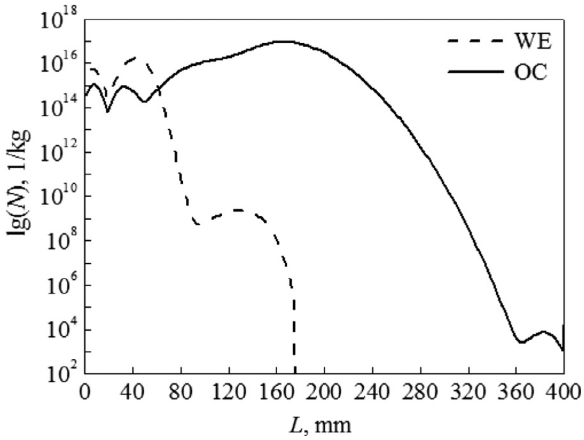

Fig. 8. The distribution of N in pressure oscillation tube of WE model and OC model.

tion droplets in OC model which peak at about 170 mm. However, fewer droplets exist in the pressure oscillation tube in WE model. The evaporation will have a great influence on the condensation inside the tube. Most of the condensation droplets will evaporate into water vapor.

4. Results and analysis

In the field of dehydration using wave rotor refrigeration technology, how to obtain good liquefaction performance by adjusting operation conditions is a problem worth exploring. Therefore, in this section, the influence of operation parameters including the temperature and humidity of HP inlet on the liquefaction performance of wave rotor is studied. The results can provide guidance for improving the liquefaction performance of wave rotor refrigeration.

In reality, due to the limit of separation technology of droplets from gas phase, not all of the droplets can be separated out from gas. The larger the size of droplets, the easier the gas-liquid separation is. Therefore, while evaluating the performance of the liquefaction in the wave rotor, the size of droplets needs to be taken into account. Two parameters, droplet radius r and liquefaction fraction H (the rate of liquid water mass to the total mass of liquid water and water vapor), are used to evaluate the liquefaction performance of wave rotor. Since these parameters are difficult to directly obtain experimentally, the liquefaction performance of wave rotor is investigated by means of CFD in this study. The nucleation process influences the subsequent process of condensation and evaporation. Therefore, the density number of droplets N which approximately represents the number of nuclei is used to help analyze the change of two parameters. A pressure oscillation tube is selected and the distribution of parameters in the tube is studied. In the pressure oscillation tube, the gas has already been affected by the main pressure waves including S1, E1, and E2.

To investigate the effect of relative humidity (RH) of HP inlet on the liquefaction performance of wave rotor, the pressure and temperature of HP inlet are set to be constant at 0.45 MPa and 288 K respectively. The distributions of N, r and H in the pressure oscillation tube are shown in Fig. 9. The distribution of N is bimodal due to expansion waves E1 and E2. According to the wave system in wave rotor, it can be found that the right peak of the distribution is caused by the expansion wave E2 and the left peak by expansion wave E1. The distribution of r is also bimodal. While at the position where N reaches the peak value, the value of r is near the relatively low value. This is because that at the position where N is high, the number of condensation nuclei is large. During the process of droplets growing, the mass of water vapor for each nucleus to grow up decreases with the increase of the number of nuclei. Therefore, the increase of N restrains the increase of radius of droplets. The peak value of r increases with relative humidity. The changing trend of the peak value of r is opposite to that of the peak value of N. The left peak value of r at RH=0.85 is 4 times larger than that at RH=0.25, which makes the droplets easier to be separated. High relative humidity makes the nucleation happen earlier and provides a longer time for droplets to grow. As for the distribution of H, with the same relative humidity, H at the position corresponding to the left peak of r is high. While at the position corresponding to the right peak of r, H is relatively low. With RH higher than 0.45, H can be maintained at a high level which is generally higher than 0.85 and the region where liquefaction happens is large. When RH is 0.25, due to small droplet radius, the droplets disappear earlier and H is generally lower than 0.7 with a small liquefaction region.

In order to obtain better liquefaction performance of wave rotor, HP inlet gas sometimes is precooled to make the temperature of LT outlet lower. Therefore, it is necessary to investigate the influence of temperature of HP inlet on liquefaction performance of wave rotor. During the investigation, the humidity and pressure of HP inlet need to be kept constant. The humidity of gas includes relative humidity and water vapor content (mass of vapor in unit mass of gas phase). Therefore, the investigation is conducted with relative humidity and water vapor content kept constant respectively.

After being compressed by the compressor, the gas that contains water vapor is usually oversaturated. Then after precooling and gas-water separation, the wet gas that flows into HP inlet becomes saturated. Therefore, in these circumstances, the water vapor content changes and the relative humidity of HP inlet remains constant, at approximately 1. With the pressure and relative humidity of HP inlet kept 0.45 MPa and 1, the influence of temperature of HP inlet on liquefaction performance is studied. Fig. 10 shows the distributions of N, r and H in the pressure oscillation tube. N increases with the decrease of the temperature. When the

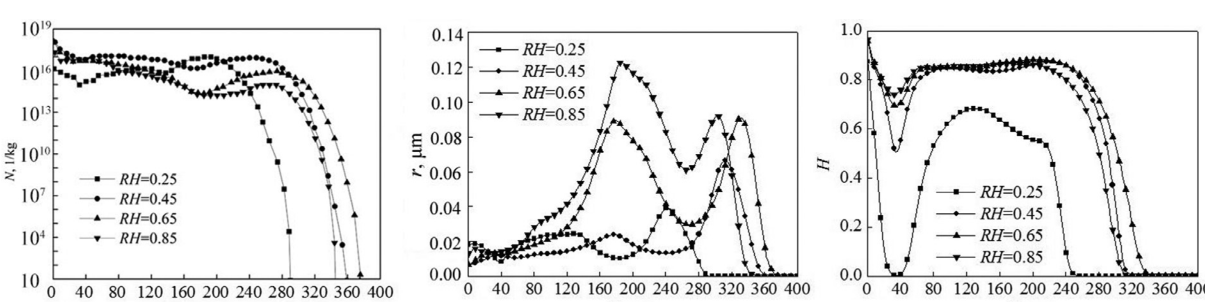

Fig. 9. Parameters distribution in pressure oscillation tube with relative humidity of HP inlet changing.

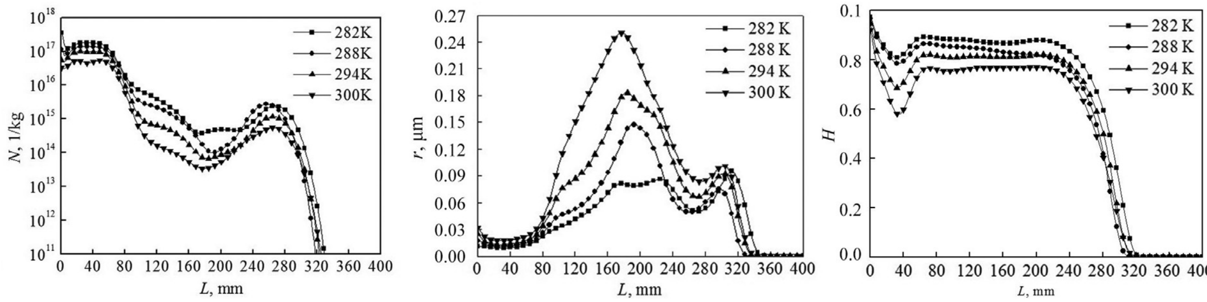

Fig. 10. With relative humidity of HP inlet constant, parameters distribution in pressure oscillation tube with temperature of HP inlet changing.

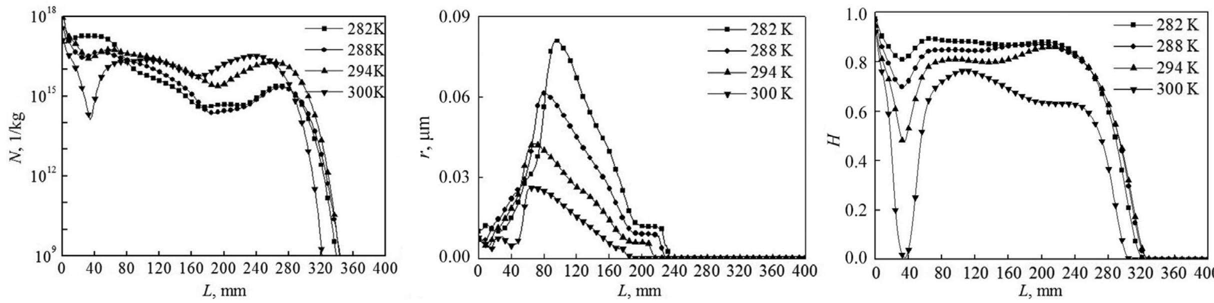

Fig. 11. With water vapor content of HP inlet constant, parameters distribution in pressure oscillation tube with temperature of HP inlet changing.

relative humidity of HP inlet is constant the partial pressure of water vapor doesn't change, the nucleation only relies on the temperature. The temperature of gas after the effect of expansion waves declines with the decrease of the temperature of HP inlet. Therefore, the nucleation is promoted by the decline of the temperature of HP inlet. Because of the increase of *N*, *r* decreases as the temperature of HP inlet decreases. When the temperature is 300 K, the peak value of *r* reaches 0.25 μm. At 282 K, the peak value of *r* is almost 1/3 of that at 300 K. As the temperature of HP inlet rises, *H* in the pressure oscillation tube decreases and the difference between 282 K and 300 K can be 0.2. Therefore, with the humidity of HP inlet kept constant as 1, when the temperature of HP inlet declines, the liquefaction in wave rotor is improved but the droplets become harder to separate.

In some circumstances, after being compressed, the wet gas is stored in a buffer tank. The internal pressure of the buffer tank is generally higher than that of HP inlet. Therefore, the wet gas in buffer tank gets depressurized and becomes unsaturated through throttle valve before being used by a wave rotor. After being precooled, the relative humidity of HP inlet changes but the water vapor content keeps constant. Therefore, it is necessary to study the influence of HP inlet temperature on liquefaction performance of wave rotor with the water vapor content kept constant. With the pressure and water vapor content of HP inlet kept constant at 0.45 MPa and 1.583 g•kg^{-1} respectively, the distributions of *N, r* and *H* in the pressure oscillation tube are shown in Fig. 11. In general, *N* increases with the temperature and this changing trend is opposite to that when the relative humidity of HP inlet remains constant. With water vapor content kept constant, when the temperature of HP inlet changes, both the partial pressure and temperature of water vapor change and influence the nucleation process. As the temperature of HP inlet increases, *r* first increases and reaches maximum at 288 K then decreases. When the temperature is 288 K, the peak value of *r* reaches 0.1 μm. *H* decreases with the increase of the temperature. With the temperature of HP inlet not higher than 294 K, the liquefaction region in pressure oscillation tube is almost the same. This is because low temperature promotes liquefaction of the vapor. When the temperature is 300 K, both *H* and *r* have low values. Therefore, the liquefaction performance at 300 K is the worst among the cases studied.

Fig. 12 shows the pressure contour in two conditions. In one condition, HP inlet contains saturated water vapor. And in the other one, dry air. There are compression waves at the LT region of wave rotor with saturated water vapor in HP inlet. These compression waves can't be found when there is no vapor in the HP inlet. This phenomenon means the compression waves are caused by condensation. The condensation releases the latent heat of water vapor and reduces the mass of water vapor. The compression waves affect the movement of subsequent expansion waves and deflect the wave system from the ideal one.

Fig. 13 shows the internal temperature distribution of the wave rotor in the situations that HP contains saturated water vapor and dry air respectively. The temperature of LT regions in the wave rotor with saturated water vapor in HP inlet is approximately 14 K higher than that with dry air in HP inlet. This result proves that heat released by condensation makes the temperature in the wave rotor rise and make the refrigeration performance of the wave rotor worse.

Experiments are implemented to measure the effect of phase transition on the refrigeration performance of the wave rotor. To study the effects of phase transition on refrigeration performance, two operating parameters, pressure and relative humidity of HP inlet, are chosen as important parameters to investigate. From the CFD results, it is known that when HP inlet contains water vapor, the isentropic expansion efficiency will decline. The equation of isentropic expansion efficiency is:

$$\eta = \frac{\Delta H_r}{\Delta H_{is}} = \frac{T_1 - T_2}{T_1\left[1 - (p_2/p_1)^{\frac{\gamma-1}{\gamma}}\right]} \tag{8}$$

It means the ratio of real enthalpy drop to the isentropic enthalpy drop from HP gas to LT gas.

Fig. 14 shows the relation between η and the pressure of HP inlet obtained by CFD and experiments with the same relative hu-

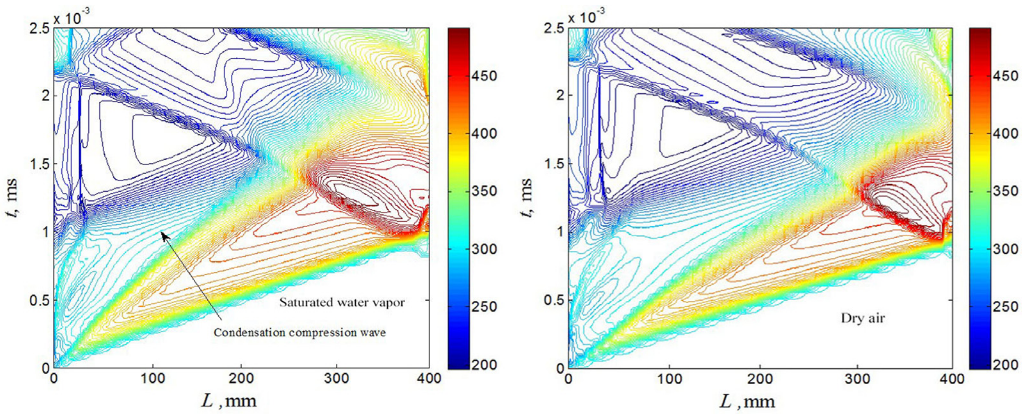

Fig. 12. Comparison of pressure contour (kPa): HP inlet contains saturated water vapor and HP inlet only contains dry air.

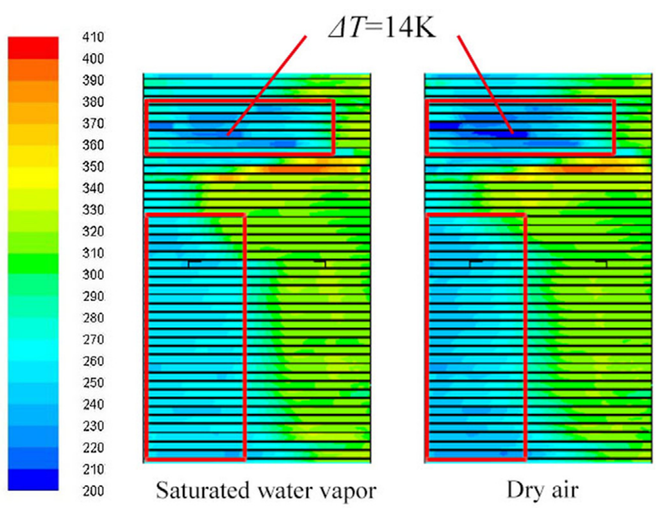

Fig. 13. Comparison of internal temperature distribution (K): HP inlet contains saturated water vapor and HP inlet only contains dry air.

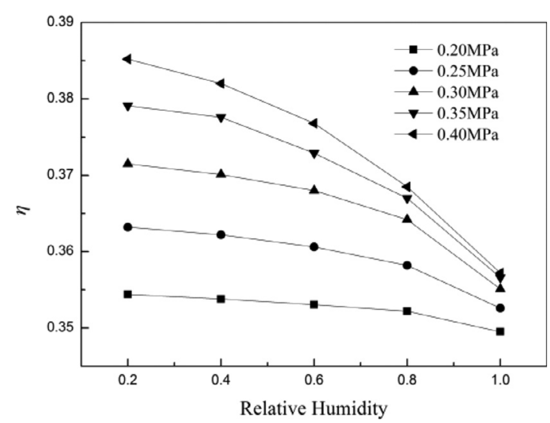

Fig. 15. Isentropic expansion efficiency to relative humidity of HP inlet with different pressure of HP inlet.

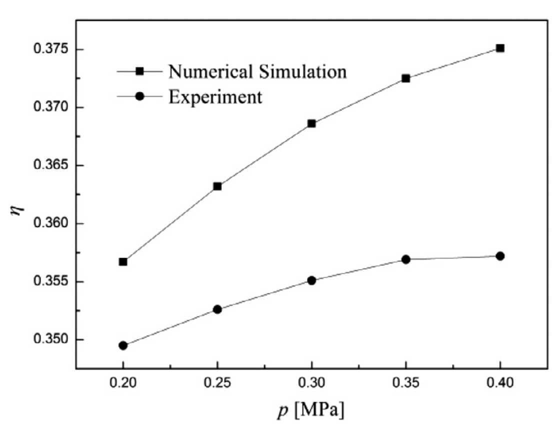

Fig. 14. Isentropic expansion efficiency to pressure of HP inlet with the same relative humidity of HP inlet of 1.

midity of HP inlet of 1. During the CFD calculation, real gas SRK model is introduced to modify density and phase equilibrium parameters to make calculation results more closed to experimental ones. With the increase of pressure of HP inlet, η increases and the increase becomes gentle. The changing trend of η from experimental results is similar to that from CFD results. The similarity further confirms the validation of phase transition numerical model. The reason for the changing of η is the ability of shock wave to do work in wave rotor. Higher pressure of HP inlet causes more powerful shock wave which does more work to promote refrigeration. At the same time, the increase of the ability of shock wave to do work decreases, so the change trend of η becomes gentle. With higher pressure in HP inlet, the real gas effect becomes more obvious in wave rotor, the deviation between numerical simulation and experiment becomes larger. The main reason for the difference between numerical and experimental results is that the numerical model does not consider the influence of the leakage and heat transfer between fluid and walls. So, the numerical value is higher than the experimental value.

Fig. 15 shows the influence of humidity of HP inlet on isentropic expansion efficiency under different pressures. With relative humidity up to 1.0 from 0.2, the efficiency declines 0.5% when pressure of HP inlet is 0.20 MPa. The decline increases with the pressure and efficiency declines 2.8% when the pressure of HP inlet is 0.40 MPa. With the same pressure of HP inlet, the partial pressure of water vapor increases with the relative humidity of HP inlet, which makes more water vapor expanded to a super-cooled state to condense, so the influence of condensation increases. Since the pressure of gathering and transportation system of natural gas is generally high, the influence of humidity on isentropic expansion efficiency will be more which needs to be taken into account by designers. Deeper analyses could help control condensation in the wave rotor which is meaningful for the wave rotor refrigeration field.

5. Conclusion

This study focuses on the non-equilibrium phase transition in the wave rotor using numerical simulations and experiments. The key results and conclusions are summarized as follows:

(1) By means of numerical simulations, it is found in the wave rotor droplets are prone to flow through the contact interface to the high-temperature region in wave rotor and then evaporate which confirms the existence of evaporation.
(2) The numerical analysis model of wave rotor that takes into account the phase transition regarding evaporation as reverse process of condensation is built up. The model is confirmed to be valid by comparing with experimental results.
(3) The radius of droplets in the wave rotor increases with the relative humidity of HP inlet and the liquefaction fraction reaches maximum with relative humidity higher than 0.45; With the relative humidity of HP inlet remains constant 1.0, with the increase of temperature of HP inlet, the radius of droplets increases while the liquefaction fraction declines; With the water vapor content of HP inlet remains constant, with the increase of temperature of HP inlet, both the radius of droplets and the liquefaction fraction decline.
(4) The change of liquid phase parameters such as droplet radius, droplet number and liquefaction fraction in wave rotor and the influence of inlet pressure and humidity on refrigeration performance of wave rotor is obtained. It's found that the isentropic expansion efficiency increases with the pressure of HP inlet and the increase becomes gentle. With the increase of relative humidity of HP inlet from 0.2 to 1.0, the isentropic expansion efficiency decreases and the decrease increases with the pressure of HP inlet. When the pressure of HP inlet is 0.20 MPa, the efficiency declines 0.5%, while declines 2.8% when the pressure is 0.40 MPa.

Declaration of Competing Interest

The authors declared that they have no conflicts of interest to this work. We declare that we do not have any commercial or associative interest that represents a conflict of interest in connection with the work submitted.

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