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Surface Tension Effect on Heat Transfer Coefficient of Condensed Gases Flow Inside Micropipe Heat Exchanger

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Abstract: Experimental and analytical investigation was carried out in this study to show the effect of surface tension on the heat transfer coefficient of gases flow inside micropipe heat exchangers during condensation process. Analytical empirical correlation was formulated for the convectional heat transfer coefficient of two phase flow inside micropipe heat exchangers. Carbon dioxide (CO_2) was used as working fluid. A comparison between experimental results and that obtained by developed correlation were carried out and a comparison between these results and literature published correlations were carried out too. The results of this research showed that the predicted heat transfer coefficient of the proposed correlation was more closely for experimental results than other correlations results, this is because of considering of the surface tension.

Key words: Surface tension, micropipe, convection heat transfer, R744, Jordan

INTRODUCTION

Experimental and analytical methods were used in literature to predict heat transfer and fluid flow inside macro tubes. Two phase flow characteristics inside micropipes required more studies to predict because of a very complicated behavior of fluid flow inside very narrow passages. Micropipes can be used in many applications; heat exchangers, condensers, evaporators, boilers, etc.

In fact, through condensation, large heat transfer rates may be achieved with small temperature differences. In addition to the latent heat, two other parameters are important in characterizing the processes, the surface tension between the liquid-vapor interface and the density difference between the two phases which is called the buoyancy force. The surface tension of a liquid, in general, decreases with temperature decrease.

Surface tension of the refrigerants influences nucleate boiling and two phase flow characteristics. A low surface tension reduces the super heat required for nucleation and growth of vapor bubbles which may positively affect heat transfer as reported by Kim *et al*. (2004). The condensation and evaporation of carbon dioxide (CO_2) inside micropipes were studied experimentally and analytically using dimensional analysis by Alshqirate (2008). He did not consider the surface tension effect in the predicted analytical model.

The dimensional analysis technique by Jokar *et al*. (2006) was used to correlate a formula for heat transfer coefficient and Nusselt number during evaporation and condensation processes of refrigerant R-134a in mini channel plate heat exchangers.

Works on CO_2 using macro scale tube heat exchangers were performed in many recent studies. Abu-Dhem (2006) considered the surface tension, as a parameter that affects the heat transfer coefficient of the refrigerant flowing inside macro-tube heat exchanger during condensation process.

Literature shows many equations and correlations formulated for two phase flow (boiling) heat transfer and validated these correlations by experimental research inside horizontal mini channels as Choi *et al*. (2007), Satish and Mark (2003) and Gungor and Winterton (1986).

Experimental and analytical studies of convection change phase heat transfer coefficient and pressure drop during flow inside tubes filled with porous media presented by Tarawneh *et al*. (2011), they were considered surface tension in formulating empirical equation to calculate heat transfer coefficient.

In this research, an empirical equation will be formulated for the convectional heat transfer coefficient of gases condensed inside micro pipe heat exchangers (two phase flow) by free convection using cold environment (chest freezer). Experimental research was carried out for the same purpose using CO_2 as gas to condense inside micro pipe heat exchangers with different diameters. The surface tension parameter will be

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considered and appear as a main parameter that affect the two phase convectional heat transfer coefficient expressed by Nusselt number (Nu).

The temperature-entropy diagram for the condensation process of carbon dioxide gas flows inside micro pipe heat exchanger was depicted in Fig. 1.

Figure 1 shows a cooling process for CO_2 from super heated conditions to saturation conditions then down to saturated liquid conditions. This process occurred as gas flows inside micro pipes. This justifies the reasonable pressure drop shown in Fig. 1. The part of condensation is the only part considered in this study. The experimental and analytical studies in this research covered a domain of independent parameters as follows:

- Micro pipe internal diameter ranged from 0.6 up to 1.6
- Inlet saturation temperature ranged from -3.19°C up to 11.74°C

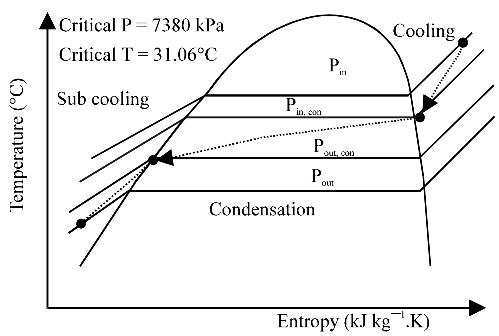

Fig. 1: Temperature-entropy diagram of CO_2 condensation process

- Mass rate of flow ranged from 2.7×10^{-5} kg sec^{-1} up to 10.6×10^{-5} kg sec^{-1}
- Pressure ranged from 32 bars up to 47 bars

This study shows an acceptable agreement between experimental results (for both heat transfer coefficient and Nusselt number) and predicted values that obtained from the resulted correlation.

MATERIALS AND METHODS

Experimental set-up: The test apparatus used in this study and its main components are shown schematically in Fig. 2. The experimental set-up consists basically of:

- The pressurized CO_2 (g) cylinder as a main source of carbon dioxide gas
- High pressure regulating valve with built-in gas cylinder pressure gauges
- Chest freezer used as environment to cool, condense and sub cool the carbon dioxide gas flowing inside the micropipe heat exchanger by free convection
- The micropipe condenser
- High pressure cutoff and isolating valves
- Pressure gauges
- Sight glasses
- The micropipe evaporator
- Data Acquisition System (DAS)
- Volume flow meter for measuring the volume flow rate of the gas

Experimental procedures: The outside wall surface temperatures of the micro pipe heat exchangers during

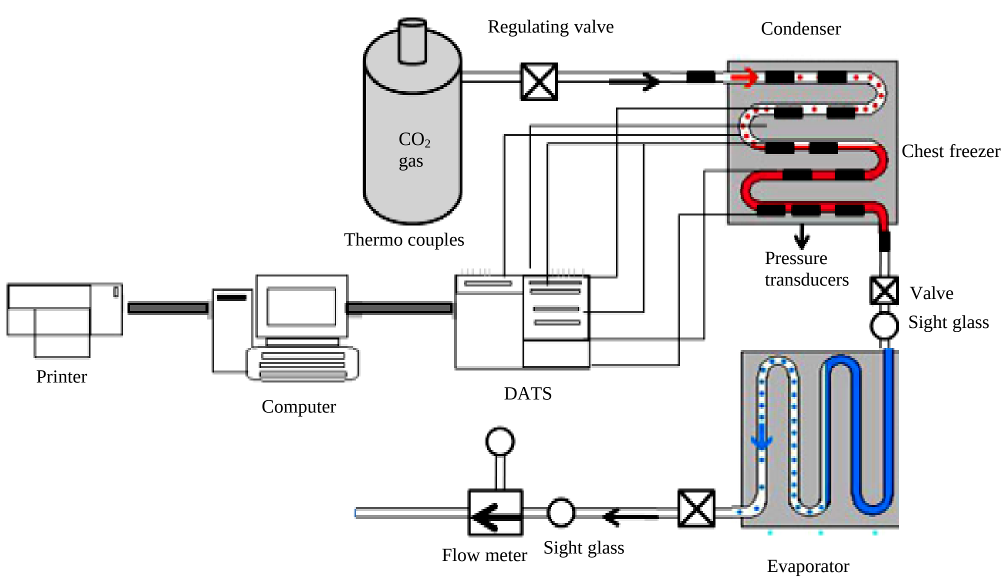

Fig. 2: Schematic diagram of the test apparatus

condensation process, the chest freezer inside air temperature and the ambient temperature were measured by means of K-type thermocouples. About 32 temperature readings, distributed along each micro pipe heat exchanger were sensed by K-type thermocouples connected to a module of 32 channels which is in turn plugged in a data acquisition system of model SCXI-1000, manufactured by National Instruments Company. The well known LAB VIEW soft ware was used for processing the signals from the thermocouples and for changing it to temperature readings on the screen of a connected personal computer. For each experimental test run, the variations of the temperature with time were monitored. The temperature readings were monitored and recorded until the steady state conditions are achieved.

The pressure was recorded in three positions at steady state conditions; they were: Before condensation, after condensation and after evaporation.

Volumetric rate of flow in m^3 sec^{-1} was read at the end outlet flow by a gas flow meter calibrated for CO_2 (g). Sight glasses were used to observe a complete achievement of condensation and evaporation processes over the whole test runs.

Experiments were conducted for different pipe diameters, at different pressures and at settings of different rate of flows.

Experimental conditions: The experimental conditions that were used in this study are listed in Table 1.

Calculations

Experimental heat transfer calculation: Condensation of gas flows inside tubes are exposure to different modes of heat transfer: The first mode is convection between the gas and the inner side of tubes, the second mode is conduction through the wall of the tubes and the third mode is convection between the outer surfaces of the tube to the surrounding.

The total heat transfer rate rejected from carbon dioxide gas in watts (W) during a complete condensation process to the surrounding air, Q_{CO_2} was calculated as follows:

$$Q_{CO_2} = m_{CO_2} \times h_{fg} \quad (1)$$

Where:

m_{CO_2} = The mass flow rate of carbon dioxide in kg sec^{-1}

h_{fg} = The latent heat of vaporization in kJ kg^{-1}

The rate of heat transfer from carbon dioxide to the inner surface of the condenser is:

$$Q_{CO_2} = h_i A_i \Delta T_{lm}$$

where h_i is the convective internal heat transfer coefficient. The logarithmic mean temperature difference calculation is:

$$\Delta T_{lm} = \left(\Delta T_{out} - \Delta T_{in}\right) / \ln\left(\Delta T_{out} / \Delta T_{in}\right) \quad (2)$$

$$\Delta T_{out} = T_{sat,out} - T_{surf} \quad (3)$$

$$\Delta T_{in} = T_{sat,in} - T_{surf} \quad (4)$$

Where:

$T_{sat,out}$ = The condenser outlet saturation temperatures at the outlet condenser pressures

$T_{sat,in}$ = The condenser inlet saturation temperatures at the inlet condenser pressures

T_{surf} = The calculated outer surface temperature of the condenser in °C

The outer surface temperature (T_{surf}) required for logarithmic mean temperature difference was calculated by Jiang *et al*. (2004) using the Eq. 5:

$$T_{surf} = \frac{\sum T_{si} \times \Delta X_J}{L}, \ J = 1, L \quad (5)$$

Where, T_{si} is the thermocouple measured surface temperature along the condensation part of test section in °C.

$$\Delta X_J = X_J - X_{J-1} \quad (6)$$

Distance along the condenser between two subsequent thermocouples (m). L is the length of the condensation part of test section in meter.

The complete condensation test section area (micropipe condenser internal surface area (A_i) in m^2) can be calculated as follows:

$$A_i = \pi D_i L \quad (7)$$

All these data were calculated and substituted in:

$$h_{exp} = Q_{CO_2} / \left(A_i \Delta T_{lm}\right) \quad (8)$$

Table 1: Experimental conditions

Variables	Values
Test sections	Three micropipe heat exchangers
Process	Condensation inside a chest freezer of -28°C
Working fluid	CO_2 (g)
Coper micropipe internal diameter (D_i, mm)	0.6, 1.0
Heat exchanger total length (m)	29.72
Test section inlet pressure (kPa)	3200, 3700, 4200, 4700
Saturation temperature (°C)	-3.19, 2.27, 7.22, 11.74
Volume flow rate (V, l min^{-1})	1-4
Mass flow rate×10^{-5} (kg sec^{-1})	2.7, 5.3, 8, 10.6

This equation was used to determine the experimental convection heat transfer coefficient, $h_{exp.}$ and the experimental Nusselt number, $\overline{Nu}_{exp.}$ of carbon dioxide gas condensed inside micro pipe heat exchangers for all test conditions.

Empirical heat transfer calculation

Dimensional analysis: The well known Buckingham π theorem and method of indices were used for finding dimensionless groups appropriate for this problem and new empirical correlations were developed using the Multiple Linear Regression Method.

The results for convection heat transfer coefficient during condensation process were presented, plotted and discussed in this research.

Alshqirate (2008) presented an empirical equation for convectional heat transfer coefficient in the form of Nusselt number as follows:

$$\overline{Nu} = 2.56\times(10)^{-5}\times\begin{bmatrix}(Re_D)^{1.27}(Pr)^{4.37}(Ga)^{-0.11}\\(Ja)^{-1.24}\left(\frac{L}{D_i}\right)^{-0.72}(Eu)^{0.21}\end{bmatrix} \quad (9)$$

The researcher neglected the effect of surface tension, as a parameter that affects condensation process (droplets formation) of any gas that internally flows.

The convection internal heat transfer coefficient (h_i) of any gas during condensation process in micro pipe heat exchanger is expected to depend on the following dimensional variables: The difference between the mean saturation temperature during condensation process and the test section surface temperature; the buoyancy force arising from the liquid-vapor density difference, the latent heat of vaporization, the surface tension, the micropipe internal diameter, the length of the micropipe condenser, the mean velocity, the pressure drop across the condenser, the mean value of the density, the mean specific heat and the mean thermal conductivity and the mean dynamic viscosity as presented by Incropera and Dewitt (2002). The mean values of the properties were defined by liquid and gas properties:

$$\text{i.e., } \rho_m = \frac{\rho_l+\rho_g}{2},\ \mu_m = \frac{\mu_l+\mu_g}{2}, \text{etc.}$$

Liquid and gas properties were obtained from CO_2 tables published by IIR. It is assumed, therefore that:

$$f\begin{bmatrix}h_i, \Delta T, g(\rho_l-\rho_g), h_{fg}, D_i, L,\\ \sigma, V_m \Delta P_{cond}, \rho_m, C_{p,m}, K_m, \mu_m\end{bmatrix} \quad (10)$$

Where, f is some function. This equation consists of 13 variables with 5 basic primary dimensions needed to express the variables, these dimensions are: Power (W) in Watt, Mass (M) in kilogram, Length (L) in meter, time (t) in second and Temperature (T) in Kelvin.

Based on the above, the number of the dimensionless variables applicable to this problem was 8. By using a step-by-step approach, the 8 prime variables are selected and another 5 repeated variables are found.

The empirical correlation equation for the carbon dioxide average convective heat transfer coefficient through the dimensionless group of the average Nusselt number can be obtained from the following dimensionless groups:

$$\text{Average Nusselt number}: \overline{Nu} = \frac{h_i D_i}{k_m} \quad (11)$$

$$\text{Reynold number}: Re_D = \frac{V_m D_i \rho_m}{\mu_m} \quad (12)$$

$$\text{Prandtl number}: Pr = \frac{C_{P,m}\mu_m}{k_m} \quad (13)$$

$$\text{Galileo number}: Ga = \frac{g(\rho_l-\rho_g)D_i^3\rho_m}{\mu_m^2} \quad (14)$$

$$\text{Jacob number}: Ja = \frac{\Delta T C_{pm}}{h_{fg}} \quad (15)$$

$$\text{Weber number} = \frac{\rho_m V_m^2 D_i}{\sigma} \quad (16)$$

$$B = \frac{L}{D_i} \quad (17)$$

$$\text{Euler number} = Eu = \frac{\Delta P_{cond}}{V_m^2 \rho_m} \quad (18)$$

By using these dimensionless groups, the averaged Nusselt number can be expressed in terms of:

$$\overline{Nu} = \text{Function}(Re_D, Pr, Ga, Ja, We, B, Eu)$$

The assumed form of this function can be suggested or expressed as a nonlinear function as the following:

$$\overline{Nu} = M(Re_D)^a(Pr)^b(Ga)^c(Ja)^d(We)^e(B)^f(Eu)^n$$

Where, M-f and n are arbitrary constants that can be obtained using the Multiple Linear Regression Method by solving for carbon dioxide data generated by the experimental research.

Thus, the final form of the empirical correlation of the average convective heat transfer coefficient in terms of the average Nusselt number during condensation process of gases flow inside micro pipe heat exchangers is:

$$\overline{\mathrm{Nu}} = 2.45\times 10^{-3}\left[\frac{(\mathrm{Re}_D)^{0.75}(\mathrm{Pr})^{21.6}(\mathrm{Ga})^{0.4}(\mathrm{Ja})^{-4.1}}{(\mathrm{We})^{22}\left(\frac{L}{D_i}\right)^{1.7}(\mathrm{Eu})^{-1.3}}\right] \quad (19)$$

To perform the correlation in terms of the independent parameters, the average Nusselt number will become:

$$h_i = 2.45\times 10^{-3}\left[\frac{(V_m^{7.75})\left((g(\rho_l-\rho_g))^{0.4}\right)(h_{fg}^{4.1})(L^{1.7})(D_i^{2.45})(\rho_m^{4.65})(C_{p,m}^{17.5})(\mu_m^{20.1})}{(\Delta T^{4.1})(\Delta P_{cond}^{1.3})(k_m^{20.6})(\sigma^{22})}\right] \quad (20)$$

The calculated values obtained from the resulted correlation were used to determine the predicted values of the average Nusselt number ($\overline{\mathrm{Nu}}_{corr.}$), then in turn the predicted values of the convection heat transfer coefficient, $h_{corr.}$.

For the purpose of validation of this correlation, a comparative study was carried out between this research correlation and literature correlations using the previous mentioned correlation, Alshqirate (2008) and well known Chato expression (Incropera and Dewitt, 2002) for the convection heat transfer coefficient of gas condensed inside tubes.

RESULTS AND DISCUSION

Figure 3 and 4 show the relation between the experimental values of heat transfer coefficient and Nusselt number with values resulted from correlated equations.

From Fig. 3 and 4, it is clear that the values of this work correlation enhanced by around 7% to be closer to the experimental values than the values predicted by Alshqirate correlation, this is because of using the surface tension as a parameter that affect the heat transfer.

Figure 5 reveals the effect of surface tension on experimental and correlated heat transfer coefficient. From Fig. 5, it is noticed that the convection heat transfer coefficient increases with increasing the surface tension. The rate of bubble growth depends on convective heat transfer through the liquid to the liquid-vapor interface required to supply the enthalpy of vaporization. The size of the bubble as it breaks away from a wetted wall (bubble growth) is an appropriate characteristic length to form a Nusselt number. This characteristic length depends mainly on the surface tension and buoyancy force.

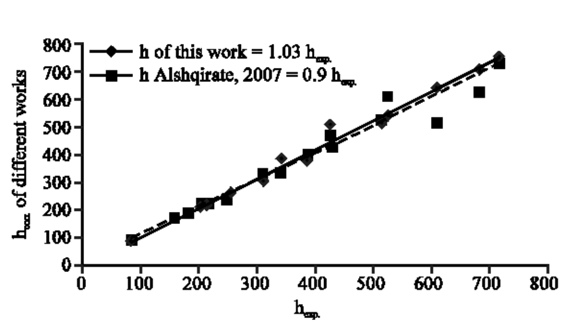

Fig. 3: Experimental heat transfer coefficient, $h_{exp.}$ vs. different correlations calculated values of heat transfer coefficient

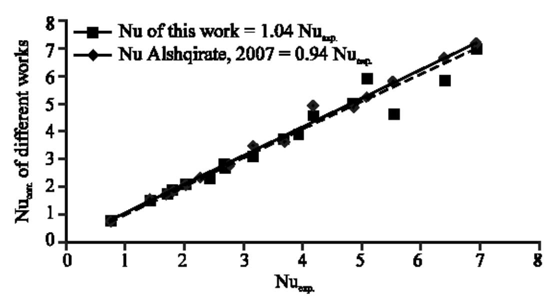

Fig. 4: Experimental Nusselt number, $Nu_{exp.}$ vs. different correlations calculated values of Nusselt number

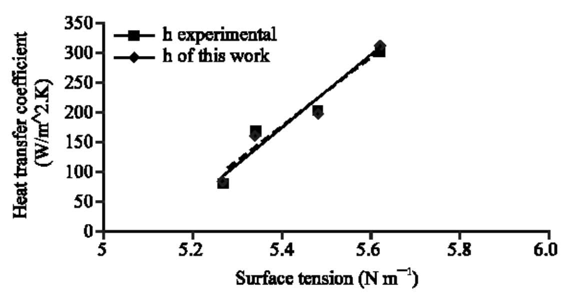

Fig. 5: Surface tension vs. heat transfer coefficient at test section inlet pressure = 3200 kPa and different volume flow rate

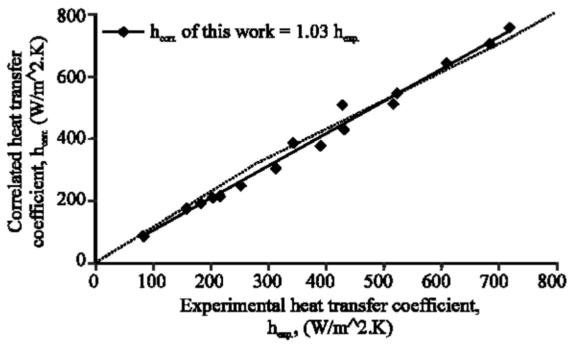

Fig. 6: Experimental heat transfer coefficient, $h_{exp.}$ vs. correlated heat transfer coefficient, h_{corr}. at different test conditions

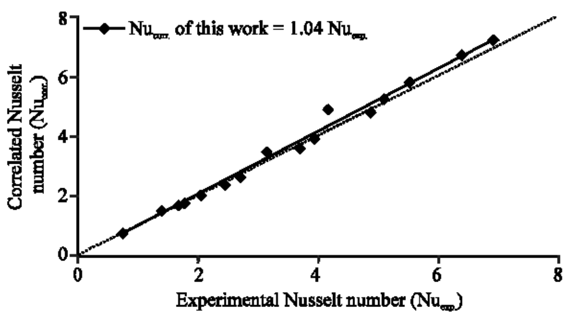

Fig. 7: Experimental Nusselt number, $Nu_{exp.}$ vs. correlated Nusselt number, $Nu_{corr.}$ at different test conditions

Figure 6 and 7 introduce the high conformity between the experimental and the predicted values for both heat transfer coefficient and Nusselt number. It is clear from Fig. 6 and 7 that the agreement of predicted values reached around 97% with the experimental results for both quantities.

Figure 8 and 9 present a comparisons between experimental heat transfer coefficient and experimental Nusselt number with those calculated using this work correlation and literature correlations of (Alshqirate correlation) and (Chato expression).

It is clear from Fig. 8 and 9 and for both h and Nu that the values of this work correlation very close to the experimental results, than those calculated using literature correlations. The agreement with experimental values was as follows: This research correlation results was over predicted with around 0.97. Alshqirate correlation under predicted with around 0.89 and for Chato expression over predicted with around 0.93.

From Fig. 8 and 9, it is noticed that the results for both h and Nu values of this work correlation agree with those calculated using literature correlation of Chato expression. The agreement reaches around 0.95 with Chato results.

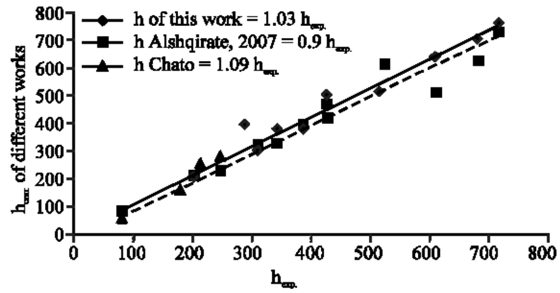

Fig. 8: Experimental heat transfer coefficient, $h_{exp.}$ vs. different correlations calculated values of heat transfer coefficient

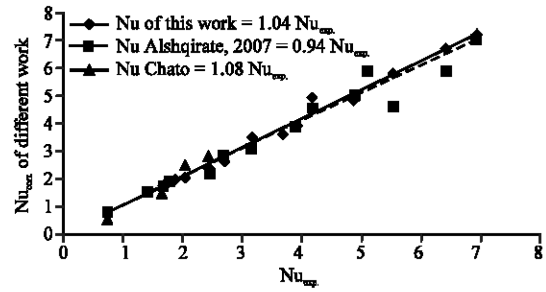

Fig. 9: Experimental Nusselt number, $Nu_{exp.}$ vs. different correlations calculated values of Nusselt number

CONCLUSION

In this research, a new correlation was obtained for calculating heat transfer coefficient for two phase (condensation process) flow inside micro pipe heat exchangers.

Using the surface tension, σ, affected both h and $\overline{Nu}$ to have values more conformal with the experimental values and in this case the predicted values of both h and $\overline{Nu}$ are closer by 70% compared to the Alshqirate correlation values where the surface tension was not considered.

By considering of the surface tension as a parameter that affect the heat transfer coefficient and enhance the heat transfer process, a high conformity between predicted and experimental results will be noticed.

Because of very close predicted values to the experimental results, the proposed model for heat transfer coefficient can be used easily instead of complicated and costly experiment installation.

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