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Transient heat convection analysis of a single rod in air cross-flow

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ABSTRACT

The aim of the paper is to supply updated air convective coefficients h_a by means of three empirical correlations set forth by Zukauskas, Churchill and Hilpert for cylindrical bodies in cross-flow. In this study, low Reynolds numbers and air velocities within the range of $v = 0.5 - 5$ m/s were considered, hence, sundry values of convective coefficients were obtained and applied in a lumped heat capacity model. Finite element analysis simulations were implemented, exhibiting good conformity based on these correlations. The findings show that among the three methodologies, Zukauskas's correlation presents minimum standard deviation $sd = 0.02$ and the maximum standard deviation is presented by Hilpert's correlation $sd = 1.92$, Churchill correlation presents a standard deviation of $sd = 0.3$. The results are reliable and can therefore be used for analyzing heat convection.

KEYWORDS

air in cross-flow, convective coefficients, lumped heat capacity model, finite element analysis simulations

1. INTRODUCTION

The attempts to investigate energy exchange by convection in circular and cylindrical bodies wherein the determination of power losses, energy rate, the thermal behavior among a system and its surroundings and the development of thermal models for specific industrial applications are of utmost relevance. Taking into consideration the importance of accurately creating thermal models, convection must be analyzed at a certain level of fidelity. It could appear that heat convection is merely obtained by multiplying an average convection coefficient h of a fluid or gas (for learning purposes can be found in literature, e.g., [1–3]) and the temperature difference between the fluid and the surface wall of a body, thereby obtaining the heat rate. Nevertheless, the determination of the average convection coefficient $\overline{h}$ presents complexity since it cannot be obtained without applying empirical correlations and considering the geometry and thermal material properties of a body under analysis. Empirical correlations were set forth in previous works by [4–6] and are widely used up to the present. Zukauskas A. [4] performed experiments of heat transfer from circular hot wires and pipes with different material, length and diameter to air cross-flow, thus obtaining an empirical correlation to determine the Nusselt number Nu for different Reynolds numbers Re. Churchill et al. [5] performed an extensive analysis of problems of heat transfer and the hydraulic resistance of single tubes, banks of tubes, and systems of tubes in cross-flow. Hilpert [6] developed a comprehensive equation for the heat rate from a circular cylinder for the entire range of Reynolds numbers. Regarding the geometry of a body in analysis, its shape, arrangement, and dimensions will influence the determination of the average

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convection coefficient and the heat performance thereof is directly influenced by the geometric parameters of the body [7]. Current studies of air flow in cross-flow have been developed by different researchers in the field of heat transfer by convection and fluid dynamics. To reference some important work, Farhana A. [8] performed numerical studies of air in cross-flow for smooth and grooved cylindrical bodies for a range of Reynolds numbers from $\mathrm{Re} = 50 - 300$, whereby effective reduction of the aerodynamic loads is obtained. Cetin C. et al. [9], performed experimental analysis of flow passing a circular cylinder with six different rectangular groove sizes revealing effects of the groove position and size over the near wake structure and turbulence. Moreover, studies reveal that by an appropriate convective analysis heat losses can be decreased by 12 % [10]. Therefore, it is not advisable for a thermal model to plainly calculate heat transfer with data from tables and references, but performing a thorough and liable analysis. It is necessary to possess reliable data that can be used in real applications, as in the case presented by Haber et al. [11], wherein heat transfer coefficients are defined for air-flow in photovoltaic modules considering air speed, geographical location of the elements and air temperature. Important information about experimental procedures regarding heated cylinders is shown by [12], wherein simultaneous measurements of velocity and temperature field of a were performed applying two different approaches, Particular Image Velocimetry (PIV) and Schlieren method for Reynold numbers Re<200.

Based on the information contained in the preceding paragraph, this article presents an analysis to determine the average convection coefficient of air h_a in cross-flow, utilizing the three empirical correlations set forth by [4–6]. A Lumped Heat Capacity (LHC) model is developed for a steel C-45 solid cylinder of length $l = 168$ mm and diameter $\varnothing = 33$ mm. In the Lumped Heat Capacity (LHC) model, the solid cylinder with an initial temperature $T_w = 60$ °C is subjected to forced convection of air at an ambient temperature $T_{fl} = 27.5$ °C. To determine a range of Reynolds numbers wherein the correlations are applicable, the air speed is varied from $0.5 - 5$ m/s . The time required for the solid cylinder to reach steady state, i.e., to cool down until the ambient temperature is determined at the maximum air velocity. Furthermore, the required energy to remove heat by convection from the cylinder is calculated. By the application of Finite Element Analysis (FEA) using ANSYS Fluent code, the results were validated and prove the applicability and reliability of the three empirical correlations in the LHC model.

2. MATERIALS AND METHODS

Explicitly, flow forced convection is generated by external means wherewith air velocity is changed. By the application of three correlations stated by [4–6] the average convection coefficient is determined when air particles are in contact with the surface wall of a steel cylindrical rod as it is shown in Fig. 1.

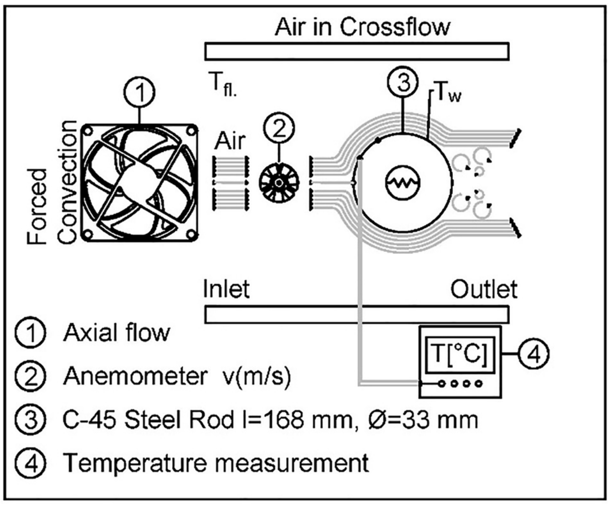

Fig. 1. Heat convection of a steel cylindrical rod in cross flow (*Source*: Author)

2.1. Heat transfer in a single tube in air cross-flow

Air in external cross-flow while in contact with a heated cylindrical surface generates heat losses. In this configuration, the characteristic parameter is the Reynolds number Re is defined in Eq. (1) [8],

$$\mathrm{Re} = \frac{v \cdot \phi}{\nu}, \tag{1}$$

where v is the stream velocity; ϕ is the diameter of the tube; ν is the kinematic viscosity of air.

Heat is determined by the flow velocity, the physical properties of the fluid, the heat flux intensity, and the geometry of the cylinder [5]. These parameters are related in a dimensionless factor called Nusselt number Nu, which is a function of the Reynolds number Re, Prandtl number Pr ,

$$\mathrm{Nu} = f(\mathrm{Re}, \mathrm{Pr}, \phi). \tag{2}$$

As it is previously mentioned, heat is influenced by several factors that are cumbersome to perform analytically. However, on account of the work by Zukauskas, Churchill et al. and Hilpert, [4–6] empirical correlations to determine the Nusselt number were established, thereby average convective coefficients can be calculated.

2.1.1. Zukauskas's correlation. By the analysis of heat transfer data of a circular tube in cross-flow with viscous fluids and gases, the following equation is recommended for practical calculations [4],

$$\mathrm{Nu}_Z = c \cdot (\mathrm{Re})^m \cdot (\mathrm{Prf})^{0.37} \cdot \left(\frac{\mathrm{Prf}}{\mathrm{Prw}}\right)^{\frac{1}{4}}, \tag{3}$$

where Prf is the Prandtl number at film temperature, Prw is the Prandtl number at wall temperature [10], c and m are constants that for different Reynolds numbers can be found in [4].

2.1.2. Churchill and Bernstein's correlation. As it is stated by Churchill et al. [5], the following overall correlation provides a reasonable approximation for all Reynolds and Prandtl numbers combinations [5],

$$Nu_C = 0.3 + \frac{0.62 \cdot (Re)^{\frac{1}{2}} \cdot (Prf)^{\frac{1}{3}}}{\left[1 + \left(\frac{0.4}{Prf}\right)^{\frac{2}{3}}\right]^{\frac{1}{4}}} \cdot \left[1 + \left(\frac{Re}{282000}\right)^{\frac{5}{8}}\right]^{\frac{4}{5}}. \quad (4)$$

2.1.3. Hilpert's correlation. From experimental procedures over hot wires and tubes with temperatures up to $T_s = 100$ °C, the following correlation to determine the Nusselt number, Nu for a wide range of Reynolds numbers was stated [6],

$$Nu_H = c \cdot (Re)^m \cdot (Prf)^{\frac{1}{3}}. \quad (5)$$

A range of Reynolds numbers were determined for air flow speeds of $v = 0.5 - 5$ m/s. The relation between the Nusselt number and Reynolds number is shown in Fig. 2.

Subsequently, the average air convection coefficient is found by

$$h_a = Nu \cdot \frac{k_a}{\varnothing}, \quad (6)$$

where k_a is the thermal conductivity of air at atmospheric pressure. The relation between the Nusselt number and the average air convection coefficient is shown in Fig. 3.

2.2. Lumped heat capacity model LHC

Originally, the surface wall of a cylinder has an initial temperature T_w. Due to the air stream in cross-flow, its internal energy is decreased until steady state condition is reached. The duration to achieve equilibrium is determined by a lumped heat capacity model. In the system shown in Fig. 1 wherein there is no input energy and no energy generation, the LHC model is expressed as,

$$-h_a \cdot A_s \cdot (T_w - T_{fl}) = \rho \cdot V \cdot c_p \cdot \frac{\partial T}{\partial t}, \quad (7)$$

where A_s is the surface area of the rod; ρ is the material density; V is the volume of the rod; c_p is the specific heat; and $\frac{\partial T}{\partial t}$ is the temperature gradient with respect to time.

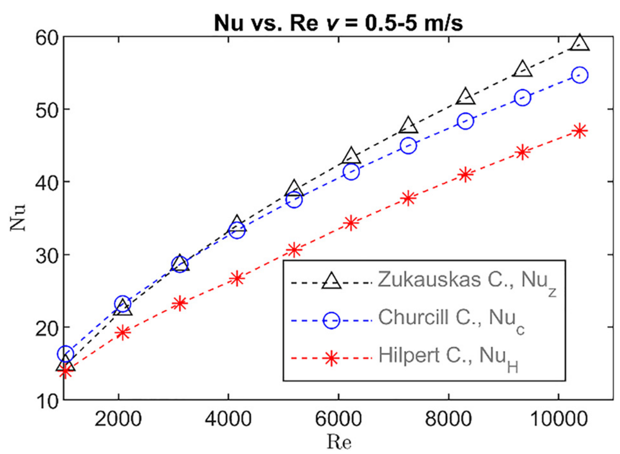

Fig. 2. Nusselt vs. Reynolds number at $v = 0.5$–5 m s^{-1}

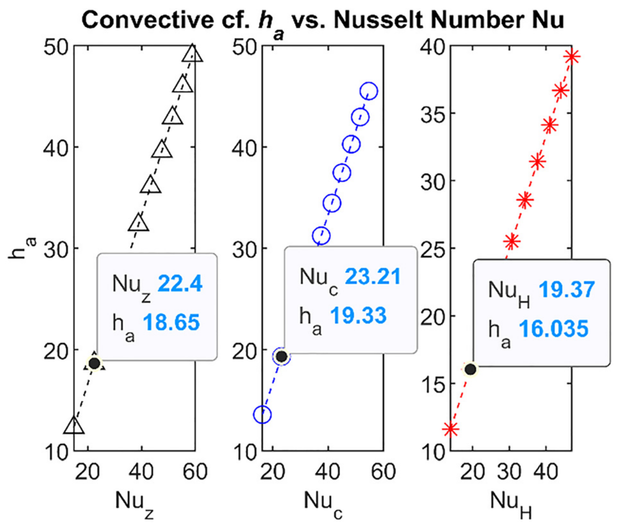

Fig. 3. Average convective coefficient h_a vs. Nu number

It is worth considering that the LHC model can be applied only if its Biot number is $Bi \leq 0.1$. The form of calculation of the Biot number can be found in Cole K. et al. [13]. Solving the differential equation, the particular solution of the LHC model is given by,

$$T(t) = T_{fl} + (T_w - T_{fl}) \cdot e^{\frac{-\overline{h_a} \cdot A_s}{\rho \cdot c_p \cdot V} \cdot t}. \quad (8)$$

3. RESULTS

The average heat convection coefficient of air h_a was calculated at different speeds in a range from $v = 0.5 - 5$ m/s. The results are shown in Table 1.

LHC analysis was performed during a period of time $t = 30$ min until steady state conditions were obtained, i.e., $T(t) = 27.5 \pm 2$ °C at max air stream velocity $v = 5$ m/s. The relation of temperature distribution and time with average air convective coefficient using Zukauskas, Churchill et al. and Hilpert correlations for a range of air velocities from $v = 0.5 - 5$ m/s are shown in Figs 4–6, respectively.

As it is illustrated in the previous figures, the steady state temperature was reached at the maximum air speed of

Table 1. Heat convection coefficient, h_a

v $\left[\frac{m}{s}\right]$	h_a $\left[\frac{W}{m^2 \cdot K}\right]$ (Zukasukas C.)	h_a $\left[\frac{W}{m^2 \cdot K}\right]$ (Churchill C.)	h_a $\left[\frac{W}{m^2 \cdot K}\right]$ (Hilpert C.)
0.5	12.31	13.57	11.60
1.0	18.65	19.32	16.03
1.5	23.79	23.85	19.37
2.0	28.28	27.75	22.23
2.5	32.33	31.24	25.52
3.0	36.07	34.45	28.56
3.5	39.56	37.44	31.42
4.0	42.86	40.26	34.12
4.5	46.00	42.95	36.70
5.0	49.01	45.52	39.17

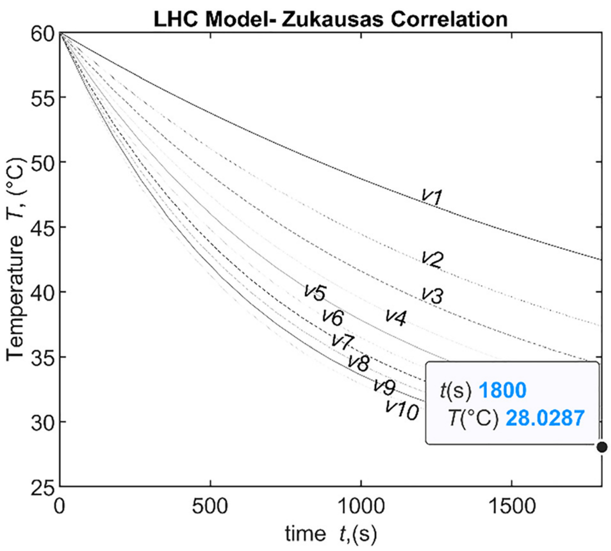

Fig. 4. Temperature distribution, Zukauskas correlation

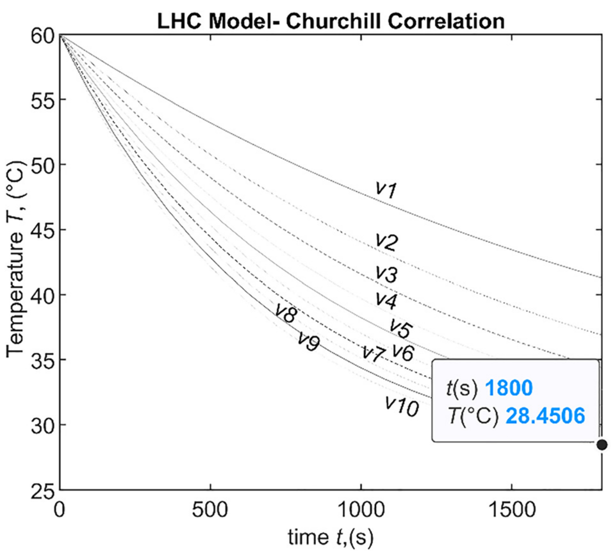

Fig. 5. Temperature distribution, Churchill correlation

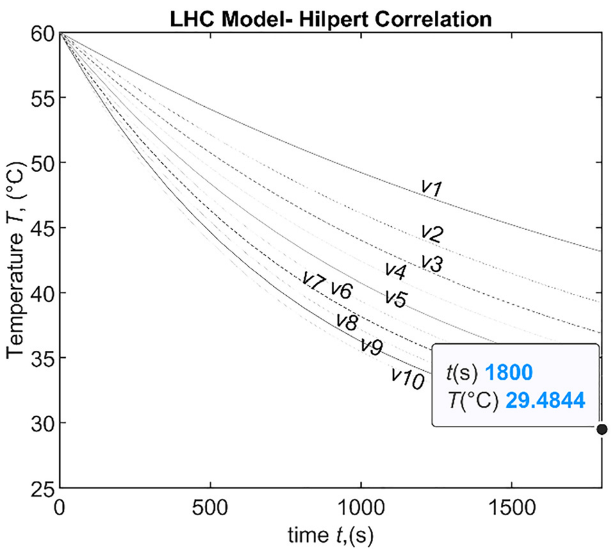

Fig. 6. Temperature distribution, Hilpert correlation

$v = 5 \text{ m s}^{-1}$. The final temperature for the three different correlations were compared and evaluated by FEA.

3.1. FEA simulation

Finite element analysis using ANSYS Fluent code was performed in a 2D geometry field considering that air velocity v flows only in one direction, thereby the processing calculations are simplified. In this model, meshing was created dividing the control volume in four parts as it is shown in Fig. 7. These are, inlet where air in cross-flow approaches the

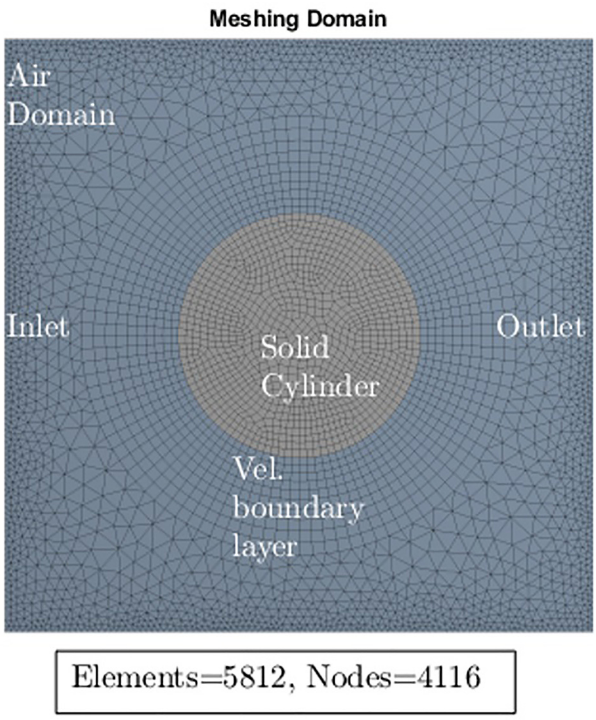

Fig. 7. Meshing. LHC model

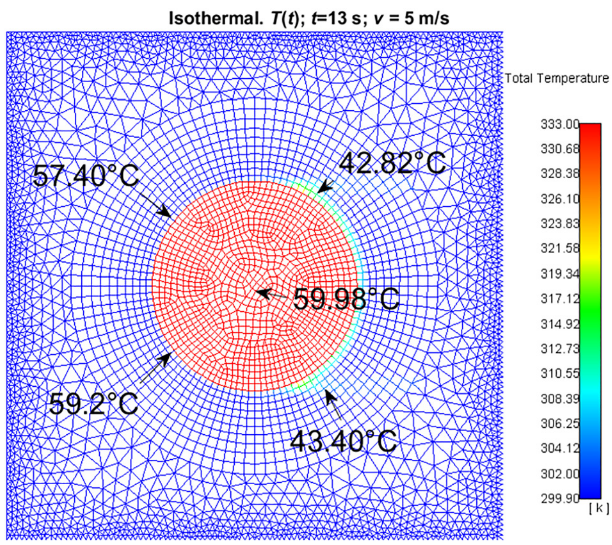

Fig. 8. Isothermal image at $t = 13$ s, $v = 5$m/s

Table 2. Analytical and FEA comparison a $t = 1800$ s

v $\left[\frac{m}{s}\right]$	$T(t)$ [°C]	$T(t)$ [°C] FEA 1) Zukauskas C.	sd	$T(t)$ [°C]	$T(t)$ [°C] FEA 2) Churchill C.	sd	$T(t)$ [°C] Hc	$T(t)$ [°C] FEA 3) Hilpert C.	sd
1.00	37.33	37.34	0.00	36.90	36.91	0.30	39.21	39.22	1.30
2.00	32.49	32.49	0.00	32.69	32.69	0.14	35.19	35.21	1.92
3.00	30.16	30.16	0.00	30.56	30.56	0.28	32.37	32.42	1.59
4.00	28.84	28.87	0.02	29.28	29.29	0.29	30.60	30.65	1.25
5.00	28.02	28.05	0.02	28.45	28.47	0.29	29.48	29.51	1.03

surface of the body, the solid cylinder wherein heat is dissipated by air convection at different speeds, outlet where air stream flow abandons the control volume and a velocity boundary layer, wherewith important factors including pressure gradients, Reynolds numbers, temperature boundary layer thickness are calculated by FEA analysis. The velocity boundary thickness must be $\delta \geq 4.95$ mm. The method to estimate the velocity boundary thickness can be found in [1]. A more accurate isothermal field around the surface of the cylindrical rod is obtained by using edge sizing feature with quadrilateral dominant method. The other part of the air domain was meshed using a triangular dominant method and its size was 0.001 mm wherewith accurate results are projected. Subsequently, 5,812 elements and 4,116 nodes were obtained.

To illustrate, the isothermal behavior at maximum air velocity $v = 5\text{m/s}$ at $t = 13$ s. was measured and shown in Fig. 8.

Analytical and FEA solutions using edge sizing feature with quadrilateral dominant method performed during a simulation time of $t = 1800$ s with $\Delta t = 0.5$ s time interval between iterations are shown in Table 2. The temperature distribution simulating the three empirical correlations at steady state conditions is shown in Fig. 9.

From Table 2 it is clear that more accurate results are obtained using the first correlation since the Standard Deviation $sd = 0.02$ is very low compared to the third

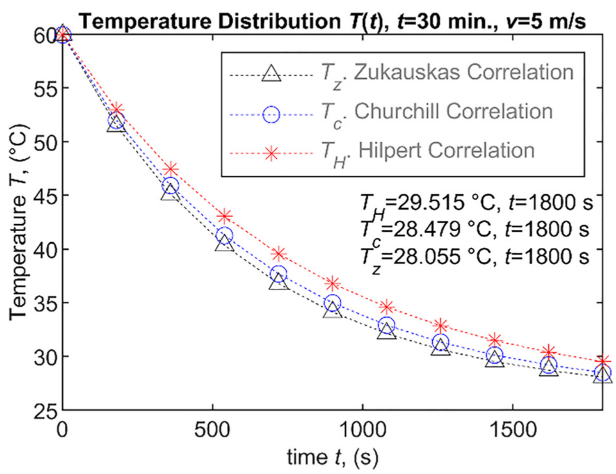

Fig. 9. Temperature distribution at steady state conditions

correlation wherein the maximum value of $sd = 1.92$. The energy dissipated by air from the surface wall of the tube was determined for each observed relationship giving $Q_1 = 17.6$ kJ, $Q_2 = 17.48$ kJ and $Q_3 = 17.3$ kJ, respectively.

4. CONCLUSIONS

The determination of sundry convective coefficients of air h_a when its particles are in contact with the surface wall of a tube due to forced convection was calculated by three different means. In order to evaluate the results a lumped heat capacity model was developed and FEA simulations were performed. In light of the results, the following conclusions were derived.

- The most accurate correlation that can be applied in phenomena with air in cross-flow over cylindrical bodies is Zukauskas correlation since the standard deviation $sd = 0.02$ is lower than the two other correlations;
- The applicability of the presented average convective coefficients of air are suitable under phenomena with Reynolds numbers in the range from $1040 \leq \text{Re} \leq 10400$. It occurs due to the characteristics of the geometry in study and the air velocity conditions;
- The findings of this study hold important implications for convective analysis in mechanical systems featuring cylindrical components. Specifically, the insights gained from the present work can be readily applied to a wide range of rotary machinery, including shafts, gears, roller bearings, seals, and other components where heat transfer occurs between the solid body and the surrounding air. This revision provides a clearer explanation of the relevance and scope of the research, and offers a more specific and engaging description of the types of mechanical elements to which the results can be applied.

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