Natural Convection in Annulus Between Two Concentric Cylinders Partially Filled with Metal Foam Distributed with New Suggested Design

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Abstract. The investigation of natural convection in an annular space between two concentric cylinders partially filled with metal foam is introduced numerically. The metal foam is inserted with a new suggested design that includes the distribution of metal foam in the annular space, not only in the radial direction, but also with the angular direction. Temperatures of inner and outer cylinders are maintained at constant value in which inner cylinder temperature is higher than the outer one. Naiver Stokes equation with Boussinesq approximation is used for fluid regime while Brinkman-Forchheimer Darcy model used for metal foam. In addition, the local thermal equilibrium condition in the energy equation of the porous media is presumed to be applicable for the present investigation. CFD ANSYS FLUENT software package (version 18.2) is used as a solver to this problem. Various parameters are examined; Rayleigh number, Darcy number, and thermal conductivity ratio to study the effect of them on fluid flow and heat transfer inside the annuli space in the suggested design of metal foam layer. current model is compared with the available published results and good agreement is noticed. Results showed that as Rayleigh number increases the dominated of convection mode increases and Nusselt increases. Also, Nusselt is larger at the higher Darcy and thermal conductivity ratio. It was found that at Rayleigh of 10⁶ and thermal conductivity ratio of 10⁴ Nusselt reach its higher value which is 6.69 for Darcy of 0.1 and 6.77 for Darcy of 0.001. A comparison between this design and the traditional design was established for Darcy 0.001 and thermal conductivity ratio 10², and its showed a good enhancement in Nusselt number and the greatest enhancement percentage was 44% at Rayleigh equal 5*10⁴ while the lowest percentage is 6% for Rayleigh equal10⁶.

Keywords: natural convection, metal foam, concentric cylinders, local thermal equilibrium, numerical investigation.

Nomenclature

| Symbol | Description |
|--------|-------------|
| c      | specific heat capacity (J · kg⁻¹ · K⁻¹) |
| Da     | Darcy number |
| μ      | dynamic viscosity (kg · m⁻¹ · s⁻¹) |
| k      | thermal conductivity (W · m⁻¹ · K⁻¹) |
| Nu     | Nusselt number |
| ϕ      | heat W |
| p      | pressure (N · m⁻²) |
| ε      | porosity |
| θ      | dimensionless temperature |
| υ      | kinematic viscosity (m² · s⁻¹) |
| K      | permeability (m²) |
| ρ      | density (kg · m⁻³) |
| q      | heat flux (W · m⁻²) |
| δ      | width of annular gap (m) |
R: dimensionless r coordinate  \( T: \) temperature (K)  
Ra: Rayleigh number  \( \omega: \) pore density [PPI (pores per inch)]  
u, v: velocity components (m · s\(^{-1}\))

**Subscripts**
- e: effective  
- f, s: fluid, solid  
- i: inner  
- o: outer  
- r: radial position (m), ratio  
- p: porous

**Greek Symbols**
- \( \beta: \) volume expansion coefficient (K\(^{-1}\))  
- w: wall

# Introduction

Natural convection in a concentric annulus has a wide variety of practical applications such as gas-insulated electrical transmission systems, double-pipe heat exchangers, solar collectors, cooling of electronic components, thermal storage, pressurized and water reactors. In addition, for more than a century, researchers have been studying porous media and have discovered numerous facts. The study of porous media has piqued the interest of many scientists due to its wide range of applications, and heat transfer has been one of the most popular. (Caltagirone 1976)[1] performed two researches for theoretical models and experimental tests with a fixed radius horizontal annulus, \( R=2 \) for natural convection two-dimension flow. (Hamad and Khan 1998)[2] ducted an experiment on natural convection in horizontal and inclined annuli to investigate the effects of inclination angle and diameter ratio on heat transfer. They compared their experimental data with numerical results predicted using the FLUENT CFD package. (Hirose et al. 2001) [3] A numerical and experimental investigation of natural convection heat transfer between a heated outer tube and a cooled inner tube in eccentric horizontal annuli with varying ordinated was performed. (Aldoss et al. 2004) [4] performed natural convection with steady laminar flow in a pipe in pipe system. The annular gap was partially filled with porous media. The porous media was introduced as an insulation media. The influences of physical parameters, Darcy and Grashof numbers and diameter of porous layer on average and local Nusselt number were examined. Comparisons with case of annular filled with porous media and with case of annular free of porous layer were presented. It was found that the relation of \( \text{Nu} \) is direct with \( \text{Gr} \) and reversed with porous layer diameter. They believed that existence of this porous media would reduce heat transfer for all \( \text{Gr} \) and \( \text{Da} \) values on a double-pipe heat exchanger's cold annular channel. two different porous rib configurations were utilized by (Targui and Kahalerras 2008)[5]. They found that employing thicker porous ribs on both the inner and outer walls of the annulus, with smaller spacing, might deliver a higher heat transfer rate than using porous ribs on only the inner wall of the annulus. (Mansour 2008)[6] studied the effects of constant temperature variation on the outside and adiabatic conditions on the inner borders with constant volumetric heat flow on two-dimensional natural convection in a thin Horizontal Cylindrical annulus filled with porous material. (Xu, Qu, and Tao 2011)[7] introduced numerically fully developed force convection in a circular tube partially filled with metal foam located aligned to the inner wall of the tube. They used Brinkman flow model, local thermal non-equilibrium to represent heat transfer equation and interfacial coupling conditions for temperature and velocity at foam-fluid interface. They concluded that the fluid and solid temperatures are not affected by difference of pore density in foam region but fluid temperature is higher in case of 20 ppi than that of 5 ppi in the hollow region. Porosity, on the other hand, has no effect on temperature distribution but does have a slight impact on total thermal resistance. They also studied the influence of thermal conductivity ratio \( K_f/K_s \) and they found that decreasing this ratio would yield an increasing in \( \text{Nu} \) follow that an increment in heat transfer as a result of the deteriorate of heat conduction resistance. Theoretical and experimental study had been conducted by (Abeed M. Ali 2012) [8] on two dimensions, laminar and steady state then transient natural convection in the annular between two concentric cylinders filled with porous media. Two situations for the cylinders were
considered; horizontal and inclined by an angle of 45°. He adopted Brinkman-extended Darcy equation and local thermal non-equilibrium model as governing equations and Fluent program was used to simulate his problem. Theoretical results showed that dissipation ability is a function of $Ra$ and $Nu$ depends monotonically on $Ra$ in the steady state case, but inversely in the transient case. Experimental outcomes conducted that nature of heat dissipation depends on relation between $Nu$ and inclined angle. Also; (Qu et al. 2013) [9] studied numerically natured convection in an annular partially filled with metallic foam. They adopted two equations model for non-equilibrium heat transfer, model I had the porous layer on the inner wall, while model II had it on the outside wall. Forchheimer and Brinkman model and local thermal non-equilibrium model were used to describe momentum and energy equations. They conducted that increasing $Ra$ and $Pr$ would strength convective performance and lead to an increment in $Nu$. By comparing $Nu$ for the two models, they noticed that $Nu$ of model I is rather larger than $Nu$ of model II. (Sheremet and Trifonova 2014)[10] considered their numerical results as standard guidelines for the use of the Darcy and Brinkman extended Darcy models in the case of transient conjugate natural convection in a vertical cylinder partially filled with a porous media and have heat-conducting solid walls of finite thickness in conditions of convective heat exchange with the environment. (Senthilkumar and Palanisamy 2015)[11] presented experimental investigation of force convection in an annular gap filled with porous media. The flow was turbulent with steady state conditions. Four different porous media particles were examined in their study cast iron, mild steel, copper, and aluminum They obtained the heat transfer coefficient and conducted to a Nusselt number correlations for the four tested porous media particles for $Re>8500$. (Alhusseny et al. 2017)[12] used the metal foam beside the rotation to enhance the heat transfer of a double-pipe exchanger. They studied the effectiveness, pressure drop, and the overall system performance and found that the thermal effectiveness could be enhanced, but extra caution should be taken to prevent the costs associated with an increase in the pressure losses. Numerical study of natural convection for unsteady, laminar and two dimensional flow in a semicircular cavity filled with saturated fluid porous media was presented by (Chamkha et al. 2018)[13]. Boussinesq approximation assumed and local thermal equilibrium were applied. Finite deference method was used to solve the governing partial differential equations. They analyzed the effect of Rayleigh number, Darcy number, dimensionless time($\tau$), and thermal conductivity ratio on fluid flow and average $Nu$. They conducted that an increment in $Ra$ or in $Da$ yield an increase in $Nu$ and fluid flow. (Roy 2019)[14] investigate the flow characteristics of natural convection in an annulus surrounded by two wavy wall cylinders generated by exothermic reaction., their numerical results illustrate that the Frank-Kamenetskii number, the Rayleigh number for thermal diffusion, the buoyancy force parameter, the amplitudes of inner and outer wavy wall cylinders, and the Lewis number all have an effect on the streamlines, isotherms, and isolines of concentration. (Chen et al. 2020)[15] investigated a numerical study for force convection in a double pipe heat exchanger filled with metal foam. The theoretical model was done using ANSYS FLUENT. This study was divided into two parts; uniform configuration and graded configuration. Results showed that for uniform configuration an increasing in the effectiveness occur with increasing thermal conductivity till a peck point then it reverses into decreasing. They noticed that effectiveness and pressure drop increase with increasing of pore density and decreasing of porosity. For graded configuration the best performance was holed for the case in which lower porosity attached to both sides of inner pipe wall and at a small pore density nearby the inner pipe wall.

It can be noted from the literature review that there is a lack of natural convection studies within an annular space filled with metal foam, especially partially filled with metal foam. In addition, we note from previous studies that researchers who have been exposed to the issue of partially adding metal foam within an annular space have focused on the distribution of metal foam partially in the redial direction, and there are no studies of distribution the metal foam partially in the angular direction. In this paper, a numerical study of natural convection in an annular space between two concentric horizontal cylinders partially filled with metal foam was introduced. metal foam was partially distributed with both redial direction and angular direction. The metal foam was arranged in the form of three pieces distributed on the surface of the inner tube, allowing
heat to be transferred from the inner tube to the fluid directly on one side and through the metal foam on the other. These pieces are designed in a way that allows the convection currents to move between them freely and smoothly. All of this contributed to an increase in heat transfer in addition to reducing the amount of metal foam used and thus reducing the cost. Commercial code ANSYS FLUNT 18 was used to simulate this problem.

2. Problem description

The geometry of the problem under investigation is shown in figure 1. It consists of two cylinders; inner cylinder maintained at temperature higher than outer cylinder temperature. Metal foam is inserted as three pieces distributed with equal distances in the annular space aligned to the inner cylinder. The geometry is similar to the radioactive sign in which metal foam surface area increase gradually with the radius toward the outer cylinder. This design leads to increase contact area between the fluid and the metal foam. Diameters of inner and outer cylinders are 10 cm and 5 cm respectively. The flow is assumed to be steady state, two-dimensional, laminar and incompressible with constant thermophysical properties except for fluid density which follows the Boussinesq approximation. metal foam is isotropic, homogeneous, and saturated with porosity of 0.9 and pore density of 10 ppi. Governing equations used for fluid and metal foam are:

Liquid region:[16]

\[ \rho_l \left[ \frac{1}{r} \frac{\partial (ru)}{\partial \phi} + \frac{1}{r} \frac{\partial (rv)}{\partial r} \right] = -\frac{1}{r} \frac{\partial p}{\partial \phi} + \mu_l \times \left[ \frac{1}{r^2} \frac{\partial^2 u}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right) \right] + \mu_l \left[ -\frac{u}{r^2} - \frac{2}{r^2} \frac{\partial v}{\partial \phi} - \rho_l \frac{u v}{r} - \rho_l g \beta (T_i - T_o) \sin \phi \right] \]

\[ \rho_l \left[ \frac{1}{r} \frac{\partial (rv)}{\partial \phi} + \frac{1}{r} \frac{\partial (ru)}{\partial r} \right] = -\frac{1}{r} \frac{\partial p}{\partial r} + \mu_l \times \left[ \frac{1}{r^2} \frac{\partial^2 v}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial v}{\partial r} \right) \right] + \mu_l \left[ -\frac{v}{r^2} - \frac{2}{r^2} \frac{\partial u}{\partial \phi} + \rho_l \frac{u}{r^2} + \rho_l g \beta (T_i - T_o) \cos \phi \right] \]

E.E:

\[ \rho_c f \left[ \frac{1}{r} \frac{\partial (uT_f)}{\partial \phi} + \frac{1}{r} \frac{\partial (vT_f)}{\partial r} \right] = k_l \left[ \frac{1}{r^2} \frac{\partial^2 T_f}{\partial \phi^2} + \frac{1}{r^2} \frac{\partial^2 T_f}{\partial r^2} \right] \]

Foam region:[9]

\[ \rho_f \left[ \frac{1}{r} \frac{\partial (uu)}{\partial \phi} + \frac{1}{r} \frac{\partial (rv)}{\partial r} \right] = -\frac{1}{r} \frac{\partial p}{\partial \phi} + \mu_f \times \left[ \frac{1}{r^2} \frac{\partial^2 u}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right) + \mu_f \left[ -\frac{u}{r^2} - \frac{2}{r^2} \frac{\partial v}{\partial \phi} - \frac{\rho_f u v}{\varepsilon^2 r} - \frac{\mu_f C_z}{\sqrt{K}} u - \frac{\rho_f C_z}{\sqrt{K}} \frac{\sqrt{u^2 + v^2}}{u^2} \right] \right] \]

\[ \rho_f \left[ \frac{1}{r} \frac{\partial (rv)}{\partial \phi} + \frac{1}{r} \frac{\partial (ru)}{\partial r} \right] = -\frac{1}{r} \frac{\partial p}{\partial r} + \mu_f \times \left[ \frac{1}{r^2} \frac{\partial^2 v}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial v}{\partial r} \right) + \mu_f \left[ -\frac{v}{r^2} - \frac{2}{r^2} \frac{\partial u}{\partial \phi} + \frac{\rho_f u v}{\varepsilon^2 r^2 - K} - \frac{\mu_f C_z}{\sqrt{K}} \frac{\sqrt{u^2 + v^2}}{u^2} \right] \right] \]

Fluid energy:
\[
\rho_ic_\varepsilon \left[ \frac{1}{r} \frac{\partial (\nu T_i)}{\partial \phi} + \frac{1}{r} \frac{\partial (r \nu T_i)}{\partial r} \right] = k_{se} \times \left[ \frac{1}{r^2} \frac{\partial^2 T_i}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_i}{\partial r} \right) \right]
\]

Solid energy:

\[
k_{se} \left[ \frac{1}{r^2} \frac{\partial^2 T_s}{\partial \phi^2} + \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial T_s}{\partial r} \right) \right] = 0
\]

Figure 1: Physical geometry of the present problem.

3. Materials and methods

3.1. Numerical method

The numerical analysis is presented for steady state, two-dimensional, laminar and natural convection heat transfer. CFD ANSYS FLUENT software package (version 18.2), a computer package that uses a finite volume method to model fluid flow and how to transfer the heat in simple and collector geometric shapes that are difficult to solve in other languages[17]. The ANSYS FLUENT provided the ability to model, mesh, give appropriate boundary conditions and simulated the governing equations of the present case. A grid independent test was made to choose the suitable mesh which contain 39245 elements. local thermal equilibrium model was considered for metal foam and air was used as a working fluid. Bringman-Forchheimer Darcy model used to describe momentum equation in metal foam. Thus it is necessary to define both the reciprocal of permeability C1, and inertial loss coefficient C2, where:(ANSYS FLUENT 18.0 User’s Guide):

\[
C2 = 3.5 \frac{(1-\varepsilon)}{\mu_p \varepsilon^2}
\]
3.2. Parameters definition

To predict the improvement of heat transfer using the new design of metal foam for natural convection in annular space several parameters were adopted. Rayleigh Ra and Darcy Da numbers are the most effected parameters and they defined as follows:

\[ Ra = \frac{g \beta \Delta T \delta^3}{\nu \alpha}, \quad Da = \frac{K}{\delta^2} \]

Where \( \delta \) is the distance between the two cylinders; \( \delta = r_o - r_i \), \( K \) is the permeability which defined as follows [9]:

\[ K = \frac{d^2 \epsilon^2}{150(1-\epsilon)^2} \]

The Nusselt number (Nu) for the inner and outer tubes can be expressed as follows [19]:

\[ Nu_i = \frac{h_i r_i}{k} \ln \frac{r_o}{r_i} \]
\[ Nu_o = \frac{h_o r_o}{k} \ln \frac{r_o}{r_i} \]

It is worth mentioning that for steady state conditions \( Nu_i = Nu_o \).

3.3. Validation

To ensure the validity of numerical work, a comparison was made with the results presented by [20] for \( Ra = 7.5 \times 10^5, \quad r_{porous}/r_i = 1.5 \quad Da = 10^{-3} \quad \text{and} \quad k_s/k_f = 100 \). Figure 2 show results of the comparison for isotherms contours.

Another compression was done with the numerical results obtained by [19] at \( Ra = 7.12 \times 10^2 \), diametric ratio=2. Figure 3 show results of the comparison for streamline contours and isotherms patterns. Table 1 shows the results of these comparisons for Nusselt number. These comparisons showed good agreements between the current results and other researchers’ work.

![Figure 2: Comparison with [20] for isotherms contours.](image-url)
4. Results and Discussion

The considered problem has been analyzed at different values for the dependent parameters; in which \( Ra = (10^4, 5 \times 10^4, 10^5, 5 \times 10^5 & 10^6), Da = (10^{-1} & 10^{-3}), k_r = (10.10^2, 10^3 & 10^4). \) The aim was to investigate the effect of these parameters on fluid flow and heat transfer inside the annuli space in present of the suggested design of metal foam layer. In addition, a comparison has been established between the results of the presented design and results of traditional design to estimate the effect of using this design for metal foam layer in annular cavities.

Figures (4 & 5) show the influence of the Rayleigh number on the isotherms and streamlines in fluid and metal foam regions at \( Da=0.1 \) and \( k_r = 100. \) Figure 4 illustrates that for \( Ra = 10^4 \) the isotherms in both regimes appear like straight lines which indicates pseudo-conductive regions for metal foam and fluid layers; meaning that the conduction is the dominant mode of heat transfer. By increasing \( Ra, \) it can be noted that convection becomes more effective and gradually increasing. On the other hand, figure 5 showed that the fluid motion driven by the buoyancy force is slow and the increase in \( Ra \) strengthens the convective performance of the flow that is belong to the increase in the temperature difference which lead to an enhancement in the bouncy force.

Influence of Rayleigh number on Nusselt number at \( k_r = 10.100&1000 \) illustrated at figures (6 and 7) for \( Da= 0.1 & 0.001 \) respectively. An increase in Rayleigh number lead to an increase in Nusselt number for all \( k_r \) values, we can observe that the increment of \( Nu \) decreases for higher Rayleigh values. they also showed that \( Nu \) number increase by the increment of \( k_r; \) an increase in thermal conductivity ratio means an increase in thermal conductivity of solid part of the metal foam which lead to a greater temperature difference between the solid matrix and saturated fluid, that boosts the convection currents and so the \( Nu \) number.
Figure 4: effect of Rayleigh number on isotherms contours at Da=0.1 and $k_F = 100$. 

$Ra = 10^4$  

$Ra = 5 \times 10^4$  

$Ra = 10^5$  

$Ra = 5 \times 10^5$  

$Ra = 10^6$  

$Ra = 5 \times 10^4$
Figure 5: effect of Rayleigh number on streamline contours at Da=0.1 and \( k_r = 100 \).

Figure 6: effect of Rayleigh number on Nusselt number at Da=0.1 for different values of \( k_r \).
Figure 7: effect of Rayleigh number on Nusselt number at Da=0.001 for different values of $k_\tau$.

Figure 8: effect of thermal conductivity ratio $k_\tau$ on isotherms and streamline contours at $Ra = 10^5$ Da=0.001.
The effect of changing the conductivity ratio on streamlines and isotherms is shown in Figure 8. As the conductivity ratio rises, it is clear that the convection is higher in metal foam than the fluid layer. As a result of the large fluid temperature difference, the level of circulation activity in the fluid layer increases. In contrast, when the conductivity ratio rises, the convective flow in the metal foam becomes weaker, leaving the porous layer practically isothermal, as seen in Figure 8. It's worth mentioning that when the conductivity ratio rises, the distance between the fluid's isothersms decreases. Figure 9 shows the influence of thermal conductivity ratio $k_r$ on Nusselt number at $Da= 0.1$ for $Ra = 10^4, 10^5 \& 10^6$. It appears an obvious increment in Nu till $Ra = 10^3$ then these increment decrease by increasing $k_r$ specially at smaller values of Ra.

![Figure 9](image-url)

**Figure 9**: effect of thermal conductivity ratio $k_r$ on Nusselt number at $Da= 0.1$ for different values of Ra.

Figure 10 shows the effect of the Darcy number on streamlines and isotherms. When the Darcy numbers are small, the fluid encounters a lot of resistance as it passes through the porous matrix, and the flow in the metal foam stops. As the Darcy number lowers, the metal foam is regarded less permeable to fluid penetration, and convective actions in the metal foam are reduced. The fluid layer's flow activities are also restricted as a result of this. The streamline contours are totally limited in the fluid layer small Darcy numbers, as shown in figure 10. In the same time, the isotherm patterns reflect a pure conduction region for the considered values of Darcy numbers.

In figure 11 it can be observed that an increasing in Darcy number $Da$ effect Nusselt number for all values of Rayleigh. It’s clear that no significant effect appear at $Ra<10^5$. The difference in the Nusselt value increases with the increase in the Rayleigh for these two Darcy values. The fact that the Nusselt is larger at the higher Darcy is due to the increase in permeability, which gives more space for the convective currents and thus increases the convection heat transfer.
Figure 10: effect of Da number on isotherms and streamline contours at $Ra = 10^4$, $k_r = 1000$
Figure.11: effect of Rayleigh number $Ra$ on Nusselt number at $k_r=100$ for $Da=0.1$ and $0.001$.

Figure.12 presented a comparison between Nusselt number of the suggested design and that of ring layer of the same metal foam volume. Both curves showed an increment in Nu number by increasing Ra number. It’s clear from this comparison that Nu is better for the suggested case especially for Ra less than $10^5$ where the bouncy force is weaker and any enhancement in convection currents seems clearer.

Figure.12: comparison between Nusselt number for the suggested design and traditional design.
5. Conclusion

Natural convection heat transfers in annular space between two concentric cylinders filled partially with metal foam inserted with new configuration was studied in this paper. The new suggested configuration includes the distribution of metal foam in the annular space, not only in the radial direction, but also with the angular direction. Metal foam is inserted as three pieces distributed with equal distances in the annular space aligned to the inner cylinder. That would allow heat to be transferred from the inner tube to the fluid directly on one side and through the metal foam on the other. These pieces are designed in a way that allows the convection currents to move between them freely and smoothly. All of this contributed to an increase in heat transfer in addition to reduce the amount of metal foam used and thus reducing the cost.

Two-dimension laminar flow, steady state conditions with constant properties were assumed. Boussinesq approximation used for fluid density. The two cylinders were kept at constant temperatures in which Ti>T0.

ANSYS FLUENT software used to solve this problem. Results showed the impact of varying Ra (104 to 106), Da (0.1 and 0.001), and thermal conductivity ratio (10 to 104) on isotherms patterns and streamline contours. The convection mode appears for Ra larger than $10^4$ and increase by increasing thermal conductivity ratios and Da number. It can be found that Nu number increase by increasing Ra and thermal conductivity ratio. Also, increasing Da results in increasing Nu for $Ra > 5 \times 10^4$. It is noteworthy that the highest value of Nusselt number for Darcy of 0.1 was 6.69 at Rayleigh equal 106 and thermal conductivity ratio equal 104 and it was 6.77 for Darcy of 0.001 at the same Rayleigh and thermal conductivity ratio. The comparison between this design and the traditional design appeared that Nusselt number get a good enhancement which reach to 44% at Rayleigh equal $5 \times 10^4$. The lowest percentage enhancement is 6% for Rayleigh of 106.

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