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Numerical Investigation of Increasing-Decreasing Stepped Micro Pin Fin Heat Sink Having Various Arrangements

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Abstract

The ongoing trend of miniaturization of electronic devices, including computer processors, high-speed servers and micro-electro-mechanical system devices, should go hand in hand with their improved performance. However, managing heat remains a major challenge for these devices. In the present study, a numerical investigation was done on a micro-channel heat sink with an open-stepped micro-pin fin heat sink with various arrangements through ANSYS software. Pin fin was varied in a fashion of increasing and decreasing. The working fluid opted for was water in a single phase. The analysis takes into account varying thermo-physical properties of water. The operating parameters, i.e. the Reynolds number was taken as 100–350 and heat flux as 500 kW/m^2. Arrangements selected were staggered and inline. Observations revealed that the staggered 2 arrangement has shown better thermal performance than other arrangements within the entire investigated range of Reynolds numbers because of the effective mixing of fluids. Furthermore, the inline configuration of micro pin fin heat sink has the worst performance. It is interesting to note that a very small difference was observed in the heat transfer capability of both staggered configurations, while the pressure drop in the staggered 2 arrangement has shown an elevated value at a higher Reynold number value compared to the staggered 1 arrangement.

Keywords: Heat transfer augmentation; Thermo-hydraulic performance; Stepped micro-channel heat sink; Open micro-channel; Inline arrangement; Staggered arrangement

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1. Introduction

Heat sinks, as per classical thermodynamics, are devices designed to dissipate heat without experiencing a temperature increase. Unfortunately, it's not feasible to construct such systems. In industrial and commercial settings, heat sinks function based on a common principle of thermal contact, they absorb excess heat generated during the operation of various systems. A significant decrease in the overall size of the component can be noticed in the industry of electronics because of the research and

Nomenclature

A_{bw} – surface area of the bottom wall, mm^2
A_{cw} – surface area of solid / liquid contact, mm^2
D_h – hydraulic diameter, mm
h – heat transfer coefficient, W/K
H_b – height of bottom substrate, mm
H_c – height of channel, mm
H_f – height of pin fin, mm
k_l – thermal conductivity of liquid, W/(m·K)
k_s – thermal conductivity of the substrate, W/(m·K)
L – length, mm
L_f – length of individual pin fin, mm
$\overline{\mathrm{Nu}}$ – average Nusselt number
ΔP – pressure drop, Pa
q – heat flux applied, W/m^2
q_{eff} – heat flux effective, W/m^2
Re – Reynolds number
u_{in} – fluid inlet velocity, m/s
$T_{avg,cw}$ – average temperature of the solid-liquid interface, K
$T_{bulk,l}$ – bulk temperature of the liquid, K
W – width, mm
W_f – width of individual pin fin, mm
W_{sw} – width of side wall, mm

Greek symbols

μ – dynamic viscosity, Pa·s
ρ – density, kg/m^3

Subscripts and Superscripts

bw – bottom wall
cw – constant wall
f – footprint
$h.fx$ – flux
in – inlet
l – liquid

Abbreviations and Acronyms

CFD – computational fluid dynamics
FEM – finite element method
HTC – heat transfer coefficient
OSMPFHS – open stepped micro pin fin heat sink
M.C.H.S. – micro channel heat sink
P.V.T. – photovoltaic trough

development taking place in the field of transistors since 1948. The removal of heat from the electronic devices is an ongoing concern having connections with the current flow via any electrical component, leading to a notable surge in heat generation per unit volume in these devices. Shortly, it is highly likely to reach a power density of nearly 1000 W/cm² [1]. Excessive heat can reduce overall efficiency and possibly cause irreversible damage to equipment if heat dissipation is not carefully controlled. As stated by the author [2], inadequate thermal management accounts for over 50% of failures in electronic VLSI circuits. Therefore, it is imperative to develop an effective and long-lasting cooling system to address this issue. Traditional cooling methods that utilize natural or forced air with fans and fins are inadequate for dissipating heat in such devices, as they are incapable of handling the significant heat generated [3]. Therefore, there is a need for innovative solutions to efficiently disperse a substantial quantity of heat from the zones which are very much confined. Additionally, these traditional systems are often bulky, noisy and unstable.

Air has a limited thermal conductivity when compared to liquids, resulting in poor heat dissipation. According to the study, liquid cooling achieves around 4–10 times higher heat flux than air cooling.

Microchannels have garnered a lot of attention in the scientific community among the various innovative technologies for dispersing heat, e.g., cooling by jet impingement, heat pipe in size of microns, Carbon nanotubes, cooling by spraying, and microchannels [4–5]. This is because it has a high surface-to-volume ratio [6] and is simple to use. For example, in [7] the authors started using this microchannel for heat dispersion, emphasizing its use in electronic cooling. Since then, a lot of work has gone into improving the architecture of the device as well as its performance based on heat dissipation to improve its utility.

2. Literature review and objective

The literature outlines various strategies to enhance heat transfer through micro-channels, which can be categorized as active and passive methods. Active approaches involve integrating microchannels with additional techniques like vibration and electrostatic forces to boost heat transfer [8]. On the other hand, passive methods focus on altering the fundamental properties of microchannel heat sinks, applying techniques such as altering the microchannel structure [9,10], using different working fluids [11,12] such as nanofluids, and so on, and adjusting the coolant operating conditions [13]. Pin fin variations for microchannel heat sinks have been included in the design, like a higher area of convection, fluid mixing with an improved strategy, secondary flow, and disturbances of laminar flow, which are all benefits of micro pin-fin topologies [14,15].

To enhance the microchannel pin fin heat sink (MPFHS) performance, several studies with experimental details with varying fin height [16], tip clearance [17,18], pin fin forms, sizes, alignments and densities have been reported in the past few years [19,20]. It was found that the performance of thermal characteristics was improved whereas the pressure was minimized in the microchannel heat sink. Figure 1 illustrates the sequential utilization of pin fins to enhance the effectiveness of open microchannel configurations. Bhandari and Prajapati first studied the influence of stepped pin fin in open micro pin fin heat sinks [21]. They stated that out of all the variants they looked at, the pin fin with rising height performed better. Increased fluid mixing, which is also seen in the 1 mm channel height [22], is the cause of this behaviour. Variable tip clearance enhances the three-dimensionality of fluid flow, which aids in thermal augmentation. Variable tip clearance along channel length and width was added to the work [23]. According to them, bidirectional steepness outperformed unidirectional steepness in terms of thermal performance factor value. The group has performed various numerical studies on open-stepped micro pin fin heat sinks. Some focused on the effect of pin fin arrangement [24], fluid flow and thermal transfer argumentation [25,26], and thermo-hydraulic performance of open-stepped micro pin fin heat sinks [27,28].

The design modifications in microchannel heat sinks are also motivated by other sectors like solar, space, automobile industries, etc. These sectors face challenges in heat transfer and heat absorption from diverse sources. Many researchers worked on specific designs and developed highly efficient and effective air heaters for their applications [29–33]. Modified designs utilized solar energy more during the day for diverse applications. The same literature presented the importance of rough shapes and perforation. It demonstrated the positive effects of roughness shapes and perforation utilized over absorber plates [29,30] and conical inserts installed inside circular heat pipes [31]. Some focused only on fins/inserts for heat transfer and absorption enhancement [32,33]. The working of solar air heaters is quite similar to heat sinks, so researchers can be inspired by it [34,36]. Some researchers have been encouraged by mini channels [37,38], microchannels [39,40], spiral tube concentric exchangers [41–43], exchangers using perforated and diverse shapes of inserts [44–46], and exchangers utilizing semi-hollow cylindrical-macro inserts [47]. In this study, we conduct simulations to assess how various arrangements impact the thermal and hydraulic performance of a heat sink with microchannels. We consider increasing and decreasing arrays of pin fins in a stepped configuration.

3. Novelty and objective

In previous works, researchers have performed analysis on pin fin heat sink having pin fin height variation throughout its length. The pattern is either in increasing style or decreasing style. However, in our works, we have considered increasing and decreasing arrays of pin fins in a stepped configuration. Further, the same configurations of pin fins were repeated throughout the length of the heat sink. Another novelty of the present geometry lies in a change in pin fin orientation (inline and staggered) along the channel width. The objective of these design alterations is to maximize the heat transfer rate in the heat sink at a minimum pressure drop penalty. In the later part, detailed dimensions of the heat sink model used in this study have been explained. The choice of the stepped configuration was made because it has demonstrated superior performance compared to uniform arrangements. This improvement is attributed to enhanced mixing of fluids, enhanced stability of 3-D fluid and distraction of thermal and hydraulic boundary layers. Based on these factors, a numerical study was conducted to compare three different arrangements containing the same fluid of single phase flowing in open microchannel pin fin heat sinks.

4. Geometry and numerical modeling

Figure 1 displays an isometric diagram of the micro pin fin heat sink (MPFHS). In Fig. 2(a), we can observe that the pin fins are shorter compared to the channel height. The study numerically analyzed the heat sink's three-dimensional geometry for various substrate materials. The overall dimensions of the computational domain are 22.50 × 12.50 × 2.00 mm (length × width × height).

Fig. 1. Isometric view of micro pin fin heat sink.

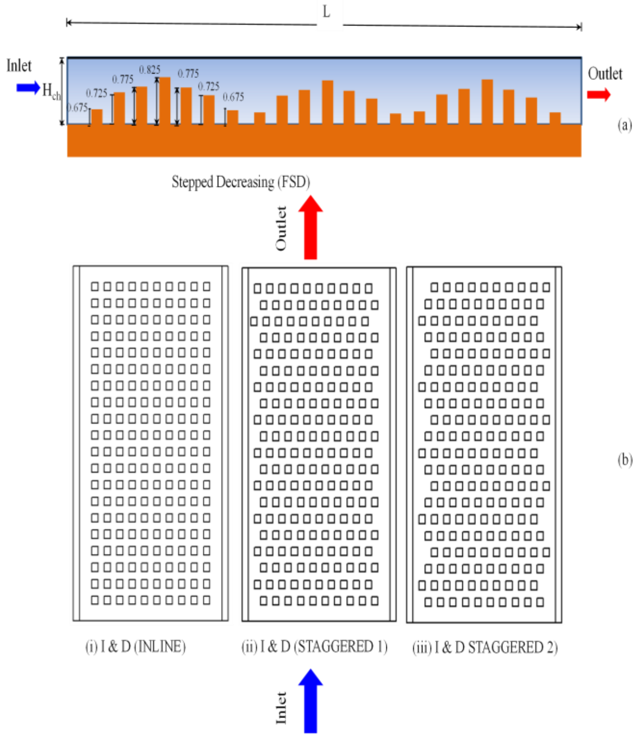

Fig. 2. (a) Side view of increasing and decreasing stepped micro pin fin heat sink.; (b) Top view of different cases considered (i) inline arrangement, (ii) staggered 1 arrangement, (iii) staggered 2 arrangement.

The heat sink consists of rows of pin fins with varying heights, with a total of 21 rows, each containing 10 fins. This results in a total of 210 pin fins of the same size, each with a footprint area of 0.5 × 0.5 mm², arranged at a pitch distance of 1.0 mm.

Two 0.5 mm thick and 1 mm high side walls are located on those sides of the heat sink which are opposite to each other to promote fluid passage between them. The intake and outflow plenums are 1 mm long to guarantee a smooth and controlled flow of the fluid, which in this case is water. Detailed dimensional information of the current study is given in Table 1.

For the current investigation, an open microchannel heat sink (MCHS) with a stepped design, specifically with increasing and decreasing heights for the fins, as shown in Fig. 3(a), is used. The pin fin height variations in a unit array along the length in the heat sink are as follows: 0.675 mm, 0.725 mm, 0.775 mm, 0.825 mm, 0.775 mm, 0.725 mm, 0.675 mm. The various investigated configurations are illustrated in Fig. 2(b). Two different staggered configurations have been opted, one having alteration in consecutive fin rows in one direction while the other having fin alteration in both directions.

Table 1. The heat sink's dimensional parameters.

No.	Parameters	Range of value
1	Heat sink length (L)	22.50 mm
2	Heat sink width (W)	12.50 mm
3	Heat sink pin fin height (H_f)	0.6–0.9 mm
4	Width of side wall (W_{sw})	0.5 mm
5	Heat sink total height ($H = H_c+H_b$)	2 mm
6	Thickness of bottom wall (H_b)	1 mm
7	Dimension of fin footprint ($L_f×W_f$)	0.5 × 0.5 mm
8	Gap between the fins (longitudinal and transverse)	0.50 mm
9	Total number of fins	210

5. Governing equations

To address the current problem with water as a single-phase working medium, the conjugate technique was employed. To streamline the analysis, certain simplifying assumptions were made, listed as follows:

a) fluid follows the law of Newtonian fluid and it is incompressible,
b) flow is assumed to be laminar at every region of analysis,
c) no slip condition was applied at the boundary,
d) the effect of thermal changes due to radiation was ignored,
e) surfaces are adiabatic except for the wall at the bottom.

The governing equations can be written as

$$\nabla \cdot (\rho_l \vec{V}) = 0, \tag{1}$$

$$\nabla \cdot (\rho_l \vec{V}\vec{V}) = -\nabla p + \nabla \cdot \mu_l[(\nabla\vec{V} + \nabla\vec{V}^t) - 2/3 I \nabla \cdot \vec{V}] + \rho_l \vec{g}, \tag{2}$$

$$\nabla \cdot (\rho_l c_{p,l} \vec{V} T) = \nabla \cdot (k_l \nabla T). \tag{3}$$

Here, $\vec{V}$ is the velocity, t is used as a subscript representing the transpose of the matrix, while the numeric l resembles the liquid, I is the unit matrix. In the case of solid substrate, the energy equation becomes:

$$k_S \nabla^2 T = 0. \tag{4}$$

To simulate the current problem, ANSYS 18.0 was implemented with the scheme of SIMPLE being used with the criteria for convergence fixed to 10^{-4} in the case of the continuity equation, 10^{-6} for the momentum equation and 10^{-7} for the energy equation.

To predict the results, we utilized polynomial functions of temperature to represent the thermal and physical properties, as referenced by Bhandari and Prajapati [16]. These relationships are applicable within the temperature range of 5–95°C, which aligns with our assumed working conditions. Fluent considered these relationships while defining the characteristics of coolant. In this study, we calculated the Reynolds number:

$$Re = \frac{\rho_l u_{in} D_h}{\mu_l}, \tag{5}$$

where u_{in} is the uniform velocity at the inlet plenums. The hydraulic diameter (D_h) is 0.5 mm, and this value remains consistent for all three configurations. We determined the heat transfer coefficients using Eq. (6):

$$h = \frac{q_{eff}}{(T_{avg,\,cw} - T_{bulk,l})}. \tag{6}$$

The average Nusselt number is calculated as:

$$\overline{Nu} = \frac{hD_h}{k_l} = \frac{q_{eff} D_h}{(T_{avg,\,cw} - T_{bulk,l}) k_l}, \tag{7}$$

where $T_{bulk,l}$ represents the bulk fluid temperature and $T_{avg,cw}$ depicts the average temperature of the solid-liquid interface.

We calculate the area-weighted average temperature as "$T_{avg,cw}$" and estimate the volume-averaged temperature for the fluid domain "$T_{bulk,l}$". The heat flux (effective), denoted as "q_{eff}", is calculated with the help of Eq. (8) as follows:

$$q_{eff} = qA_{bw}/A_{cw}. \tag{8}$$

The heat sink's bottom wall surface area is denoted as A_{bw} and the surface area where the solid contacts the liquid is A_{cw}. It is important to note that the bottom wall surface area remains constant at 281.25 mm² for all configurations. Moreover, the convective surface area is also the same for all cases, i.e. 614.25 mm².

6. Validation

To validate the present work, the work of Mei et al. [48] was replicated. Mei et al. [48] considered the tip clearance of 0.5 mm and 1.00 mm in their studies. A comparison of the average Nusselt number obtained from the present work and that of Mei et al. work [48] is shown in Fig. 3. It is observed that the results remain in good agreement with the present model. The deviation between the studies may be due to several fins considered and shape variation.

7. Results and Discussion

To understand how pin fin arrangements affect the thermal and hydraulic characteristics, we selected three different configurations. Specifically, we simulated the increasing and decreasing stepped configurations for both inline and staggered arrangements, all at a heat flux of 500 kW/m². The variations in heat transfer coefficients for these different arrangements are illustrated in Fig. 4.

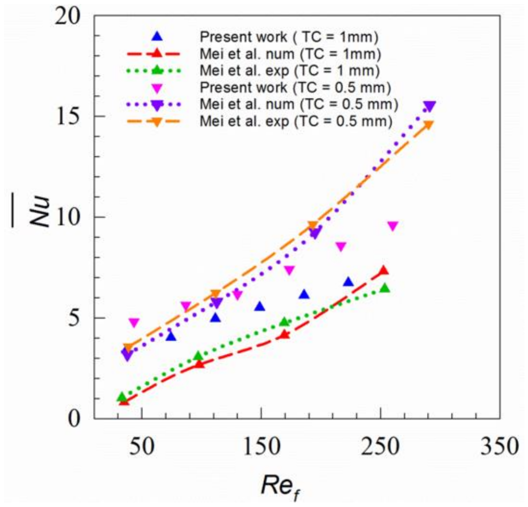

Fig. 3. Variation of average Nusselt number with fin Reynold number for different configurations.

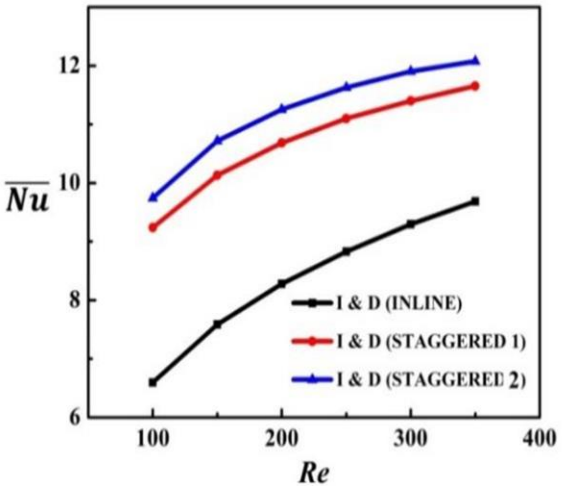

Fig. 5. Variation of $\overline{\mathrm{Nu}}$ with Re for different configurations at heat flux of 500 kW/m^2.

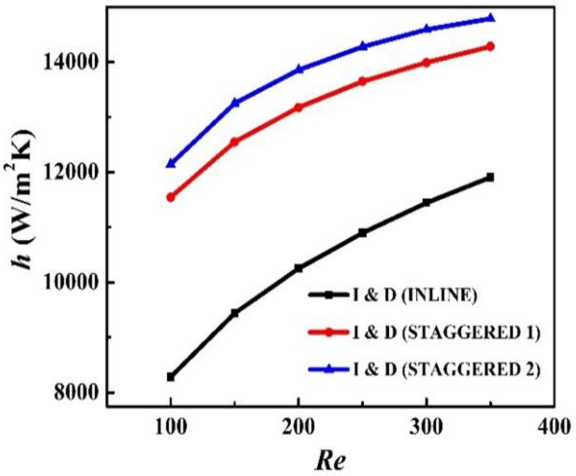

Fig. 4. Variation of heat transfer coefficients with Re for different configurations at heat flux of 500 kW/m^2.

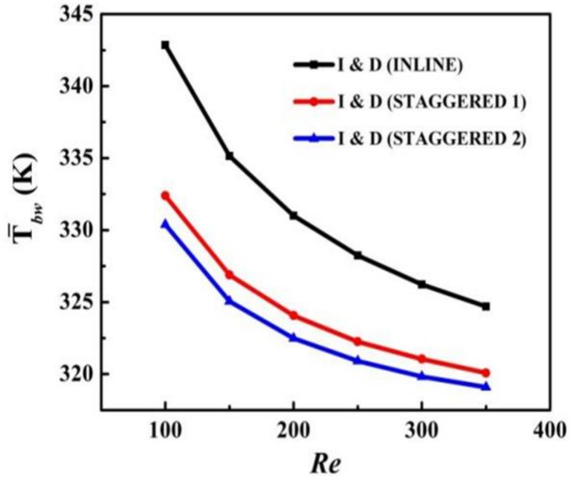

Fig. 6. Variation of $\overline{\mathrm{T_{bw}}}$ with Re for different configurations at heat flux of 500 kW/m^2.

It is observed that increasing the Reynolds number results in higher heat transfer coefficients. It is obvious since at higher Re, fluid flow rate increases subsequently; more heat is carried out by the coolant. Furthermore, it is reported that the staggered layout has higher heat transfer coefficients than the inline layout. It is interesting to note that with an increase in Re value, the heat transfer coefficient slope decreases, and at a higher Re value, it is almost flat.

A similar trend was also observed for the average Nusselt number as revealed in Fig. 5. It is observed that orientation changes from inline to staggered one showed a substantial increase in average Nu. Moreover, with further modification in the staggered arrangement, the average Nusselt number increases due to augmented flow mixing.

The variation of $\overline{T_{bw}}$ for different configurations is depicted in Fig. 6. $\overline{T_{bw}}$ was calculated as the area-weighted average temperature at the bottom wall of the heat sink where heat flux has been applied. It is noticed that significant augmentation is achieved when the arrangement changes from inline to staggered. The minimum bottom wall temperature is observed for the staggered 2 configuration due to the higher value of the average Nusselt number. It is interesting to note that increasing Re resulted in high coolant velocity and mass flow rate causing more heat transfer from the heat sink. A smaller value of $\overline{T_{bw}}$ implies higher heat dissipation at any operating condition.

Pressure difference occurs at the outlet and the inlet was examined to acquire the actual pressure drop which has been illustrated in Fig. 7. Figure 7 depicts a variation of pressure drop with change in Re for all the arrangements that are made in the current study. Among all the configurations, the maximum pressure drop was obtained for the staggered 2nd case while the lowest pressure drop was obtained in the inline case. The obstruction that occurs in the flow of fluid at the staggered 2nd case could be the reason behind this phenomenon. Furthermore, it is observed that with an increase in Re, pressure drop increases in all the configurations of the heat sink.

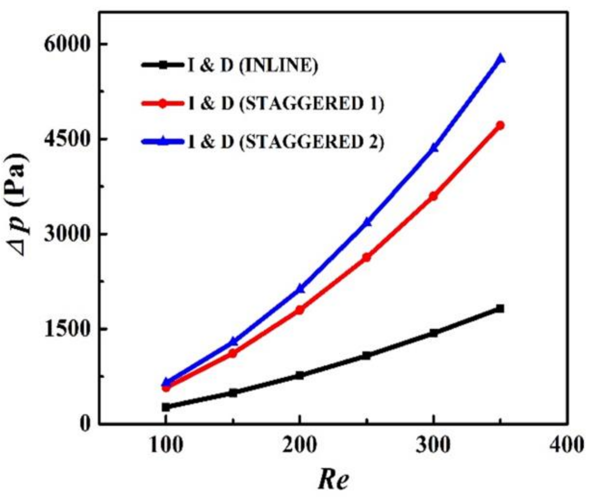

Fig. 7. Variation of pressure drop with Re for different arrangements at heat flux of 500 kW/m^2.

This is due to an increase in fluid velocity at higher Re which causes extensive flow resistance so higher pumping power is needed. From Fig. 7, it is observed that the pressure drop curve is not linear and is much steeper at higher values of Re explicitly in stepped staggered heat sinks.

It can be concluded that at low Re values, the pressure drop is low but the heat transfer capability is also low, while at high Re values, the heat transfer coefficient does not increase as much as the pressure drop. So, it is recommended to use an Increasing-Decreasing Stepped Micro Pin Fin Heat Sink for mid-range Re values.

8. Conclusions

In this study, we conducted simulations of a stepped microchannel heat sink having increasing and decreasing fin heights in various arrangements. We compared these arrangements using single-phase water as the working fluid, considering both inline and staggered configurations. The staggered arrangements included one with alterations in one direction only and another with changes in both directions. The working conditions involved a heat flux of 500 kW/m² and a Reynolds number range from 100 to 350.

Based on our findings, it can be concluded that the staggered arrangements, especially the staggered 2 arrangement, can significantly enhance the performance of the stepped microchannel configuration. This improvement is attributed to the three-dimensionality effect and improved fluid mixing.

At low Reynolds numbers, both the pressure drop and the heat transfer capability are minimal. At higher Reynolds numbers, the increase in the heat transfer coefficient involves a large rise in pressure drop. Therefore, it is advisable to use an Increasing-Decreasing Stepped Micro Pin Fin Heat Sink within a mid-range of Reynolds numbers.

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