

CFD-Based Comparative Analysis of Hydronic Piping Layouts for Radiant Cooling System

Samiullah Qureshi ^{a,*}, Afaq Ahmed Bhutto ^b, Intizar Ali Tunio ^a, Rizwan Ahmed Memon ^a, Abdul Ghafoor Memon ^a

^a Department of Mechanical Engineering, Mehran University of Engineering & Technology, Jamshoro.

^b Department of Mathematics, The University of Larkana, Larkana, 77150, Pakistan

*Corresponding author: samiullah@faculty.muet.edu.pk

Received: 11 December 2025, Revised: 02 March 2026, Accepted: 31 March 2026, Published: 01 April 2026

KEYWORDS

Energy Consumption
Radiant Cooling System
Hydronic Piping Layouts
Pressure Drop
Pumping Power
Cost Optimization

ABSTRACT

The radiant cooling (RC) system has been reported as an energy-efficient and cost effective alternative to conventional air conditioning. However, limited research has been conducted to evaluate the impact of different piping layouts on the thermal performance of the RC system. It is also noted that no study has evaluated the impact of different hydronic piping on thermal, energy and cost simultaneously. To address this gap, CFD based simulations have been carried out to evaluate the impact of serpentine, canopy-to-canopy, and tree-shaped piping layouts on the thermal, energy, and cost performance simultaneously. In this regard, piping layouts are modelled and simulated for similar operational conditions using Ansys Fluent. To ensure reliability, the model is first validated against published literature data. The steady-state, two-dimensional $k-\epsilon$ turbulence model is used to evaluate temperature distribution, pressure drop, and cooling capacity of the system. The results of the analysis show that the tree-shaped layout provides the best temperature distribution and gives the best capacity to pumping ratio with a value of 1.61. Conversely, the ratio for serpentine and canopy-to-canopy layout was significantly lower at 0.055 and 1.43, respectively. The results further show that the pressure drop for the serpentine layout was 14 % higher than the canopy-to-canopy configuration and 97 % higher than the tree-shaped layout. For the canopy-to-canopy layout, cooling capacity was slightly higher while the temperature distribution was non-uniform. In cost analysis, pipe cost, installation labor cost, and pumping cost based on electricity consumption are considered. The analysis shows that the cost of piping in the tree-shaped layout is approximately 16% lower. Furthermore, the tree-shaped layout also requires less power for circulating the water, lowering costs by 14% and 97% relative to canopy-to-canopy and serpentine layouts, respectively. Overall, the tree-shaped layout offers better performance in terms of temperature uniformity, energy consumption, and cost.

1. Introduction

Radiant cooling (RC) Systems are gradually being recognized as energy-efficient and thermally comfortable alternatives to conventional air-conditioning systems. Compared to the conventional air-conditioning system, the RC system is dependent on circulating chilled water through embedded hydronic pipes in building surfaces [1]. The

performance of RC system strongly depends on the layout of the hydronic piping network. The piping layouts control the homogeneity of the heat exchange, pressure drop, and energy consumption, which eventually define the thermal efficiency and cost effectiveness of the systems. Serpentine piping layout is usually used in radiant cooling systems [2]. Although this layout is easy to install, it comprises of a long continuous flow path. It, therefore, has huge

temperature gradients over the surface and pressure drop [3]. Conversely, the canopy-to-canopy layout has potential to overcome the problems of temperature gradient and pressure drop due to reduced flow path and parallel pipe branches [4]. Nevertheless, this layout can have problems in terms of uneven distribution of flows within branches. The tree-based layout, which is based on the natural branches system, is intended to maximize both the uniform stream distribution and pressure drop by utilizing a hierarchical network that splits the mainstream gradually [5]. The effectiveness of these layouts must be evaluated not only in terms of thermal performance but also in terms of energy and cost. Over the last few years, there has been a trend of increasing numbers of researchers adopting the Constructable theory, introduced by Bejan [6], as a framework for optimizing stream architectures in thermal systems. The constructed law states that for a stream system to persist and evolve, it must transform in a manner that facilitates easier access to currents that stream through it. This principle has been used in the design of canopy-to-canopy or tree-like stream paths in RC panels to distribute the coolant more evenly with the least hydraulic losses. These structures are architecturally similar to a river basin, a bronchial leaf or a vascular network in nature, where hierarchical branching allows even distribution over a great surface area.

Based on this theoretical foundation, Mosa et al. [7] developed a canopy-to-canopy piping layout and compared it with serpentine layout. Their findings suggest that multi-diameter piping layouts can substantially enhance cooling performance and decrease the energy required for fluid circulation. In another study [8], they compared the serpentine and other branching piping layouts, such as canopy-to-canopy and tree-shaped layouts. Notably, in all layouts, similar cooling capacity is observed. However, the presence of parallel branches reduces the effective flow path, that leads to a decrease pressure drop in canopy-to-canopy and tree-shaped layouts as compared to serpentine layout [8]. Based on the results, they concluded that the canopy-to-canopy layout is the best-performing one layout among all. However, comparative studies on the simultaneous thermal, hydraulic, and economic trade-off of these layouts are limited.

This study aims to contribute to literature through a comparative analysis of serpentine, canopy-to-canopy, and tree-shaped piping layouts for the RC system using Ansys Fluent. The objective is to systematically analyze and quantify the thermal uniformity, pumping power requirements, and cost associated with each configuration. Through

numerical modeling and mesh sensitivity analysis, the most effective design is to be determined from both energy consumption and feasibility perspectives. This research aims to contribute to the ongoing advancement of RC system design and supports informed decision-making to achieve an energy-efficient, high-performance thermal comfort environment for buildings.

2. Methodology

The overall research question for this study is how the use of different piping layouts impacts the thermal and cost performance of an RC system. In this regard, the following steps need to be taken to address the research question.

1. To develop a 2-D computational domain representing the active layer of the radiant cooling system in ANSYS SpaceClaim with three layouts that are serpentine, canopy-to-canopy, and tree-shaped.
2. To carry out meshing of the system in ANSYS Meshing Client, followed by the mesh sensitivity analysis using the Grid Convergence Index (GCI) method.
3. To simulate the models in ANSYS Fluent under steady-state conditions.
4. To determine, compare, and analyze results such as temperature and pressure distributions for all three selected types of layouts.
5. To compare the overall performance of each layout in terms of Cooling capacity ($\dot{Q}_{cool}$), pumping power ($\dot{W}_p$), cooling capacity-to-pumping power and piping costs (labour and purchasing costs), and find the best layout.

2.1 Model Description

In this study, canopy-to-canopy, tree-shaped, and serpentine layouts are all modelled and simulated with same inlet temperature and mass flow rate, as listed in Table 2. Fig. 1 shows a 2D computational domain of three layouts developed in ANSYS SpaceClaim. While geometric configuration differs, the material construction is the same for all layouts. The domain features embedded cross-linked polyethylene (PEX) pipes in a lime plaster. The working fluid (i.e., water) circulates through the pipe with a fixed flow rate. The cold water carries away heat absorbed by the surface from the environment.

To maintain consistent comparative assessment, total flow volume, surface area, and supply water temperature are kept the same across all layouts. The serpentine layout is characterized by a single, continuous meandering flow path. In contrast, there

are 29 parallel pipes connected to two headers in the canopy-to-canopy layout, as shown in Fig. 1(b). It is noted that the flow splits among parallel branches of the canopy-to-canopy layout, resulting in reduced flow path length compared to the serpentine layout. Finally, a hierarchical branching structure is employed for tree-shaped layout as shown in Fig. 1(c). In this layout, the inlet pipe bifurcates into progressively smaller pipes to ensure uniform flow distribution across the surface.

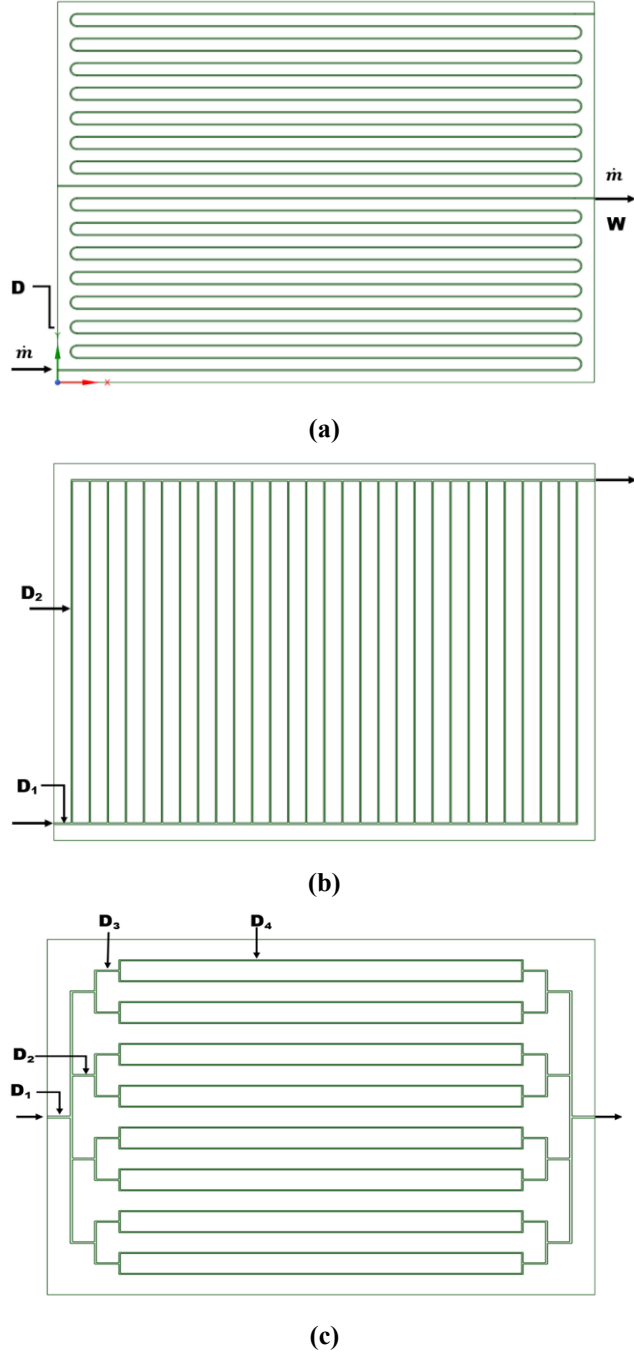


Fig. 1. 2D model of (a) serpentine, (b) canopy-to-canopy, (c) tree-shaped layouts for RC System

Based on a previous optimization study [9] and to ensure consistency, a 0.75" CTS pipe is selected for the serpentine layout, parallel branches of the canopy-to-canopy, and the terminal branches of the tree-shaped layout. The selected pipe size is represented as

D , D_2 , and D_4 in serpentine, canopy-to-canopy, and tree-shaped, respectively. To determine other pipe sizes for tree-shaped and canopy-to-canopy layouts, the condition of minimum pressure loss is considered. The pressure drop (ΔP) in a hydronic circuit is strongly influenced by pipe diameter and mass flow rate and is proportional to $\dot{m}/D^4$ [10]. Based on this, the pressure drop in the canopy-to-canopy layout can be represented as given below

$$\Delta P \sim \frac{(2\dot{m} + \sum_{i=1}^{28} \dot{m}_i \times \dot{m}/29) M}{D_1^4} + \frac{\dot{m}(W - 2M)}{29 D_2^4} \quad (1)$$

Where $\dot{m}$, W , M and D represent the mass flow rate [kg/sec], width [m], pipe spacing [m] and diameter [m] respectively.

Since all layouts were designed to circulate the same total flow volume, the canopy-to-canopy layout requires:

$$V_{flow} \sim 2(L - M)D_1^2 + 29(W - 2M)D_2^2 \quad (2)$$

For a serpentine layout, it is calculated as follows:

$$V_{flow} = \frac{\pi}{4} D^2 L_{Path} \quad (3)$$

Equating the two expressions (10) and (11) ensures that both layouts maintain identical flow volumes.

$$\frac{\pi}{4} D^2 L_{Path} \sim 2(L - M)D_1^2 + 29(W - 2M)D_2^2 \quad (4)$$

To minimize pressure losses under this constraint, the Lagrange multiplier method is applied, which gives a diameter ratio for the canopy-to-canopy layout as

$$D_r = \frac{D_1}{D_2} = 1.61 \quad (5)$$

Following the same method for the tree-shaped layout, the following relation is obtained.

$$D_r = \frac{D_i}{D_{i+1}} = 1.26 \quad (6)$$

2.2 Governing Equation

This work considers steady-state 2-D analysis of an RC system. The governing equations of the fluid flow are modified according to the simplified case. The

time-dependent and third-axis parameters are dropped from the equations. The resulting equations are.

Conservation of Mass (Continuity):

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (7)$$

The conservation of mass, also known as the continuity equation, asserts that the total mass within a fluid system remains constant. In this context, the continuity equation is applied to ensure that the total water flow rate remains constant throughout the piping layouts. Momentum:

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \nu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad (8)$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \nu \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) \quad (9)$$

Where u , v , and ρ represents velocity in x-direction [m/sec], velocity in y-direction [m/sec] and density [kg/m³].

Energy equation:

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p \Delta(uT) = k_{th} \Delta^2 T + \Phi \quad (10)$$

Where c_p , T , k_{th} and Φ represents the specific heat [J/kg.°C], temperature [°C], thermal conductivity [W/m.°C], and heat dissipation [W/m³], respectively.

The diameter used for serpentine, canopy-to-canopy and tree-shaped layouts was 0.75", 1", 1.25" and 1.5" CTS with a mass flow rate of 0.198 kg/sec. It is calculated that this size of piping with given mass flowrate results in a Reynolds number as high as 6000. It is therefore concluded that a turbulence model is required to comprehend the flow effect. There are several turbulence models available in the ANSYS Fluent. One of the most widely applied and promising approaches for analyzing fluid flow and heat transfer in engineering systems is the k-ε turbulence model [11]. It is due to its computational efficiency and accuracy in predicting turbulent flow behavior. In this model, the equations for k (turbulent kinetic energy) and ε (rate of dissipation) address the rate of change of each variable, together with convective transport. This, in turn, depends on the combined effects of diffusive transport, production, and destruction terms as given below:

$$\begin{aligned} \overline{u_j} \frac{\partial k}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] \\ - \left(\overline{\rho u_i' u_j'} \right) \frac{\partial u_i}{\partial x_j} - \varepsilon \end{aligned} \quad (11)$$

$$\begin{aligned} \overline{u_j} \frac{\partial \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] \\ - C_{\varepsilon 1} \frac{\varepsilon}{k} \left(\overline{u_i' u_j'} \right) \frac{\partial u_i}{\partial x_j} \\ - C_{\varepsilon 2} \varepsilon \frac{\varepsilon^2}{k} \end{aligned} \quad (12)$$

The turbulent viscosity μ_t , is then expressed in terms of k and ε as follows.

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (13)$$

Equation 13 models turbulent viscosity as proportional to the ratio of turbulent kinetic energy squared to its dissipation. The Reynolds stress term $-\overline{\rho u_i' u_j'}$, which models the effect of turbulence on momentum transfer, is represented using the Boussinesq approximation as:

$$-\overline{\rho u_i' u_j'} = \mu_t \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right] \quad (14)$$

Equation 14 expresses the turbulent shear stresses in terms of the mean velocity gradients and turbulent viscosity, allowing closure of the RANS (Reynolds-Averaged Navier-Stokes) equations. In the above equations, the values of the constant $C_{\varepsilon 1}$, $C_{\varepsilon 2}$, C_μ , σ_k , and σ_ε are 1.44, 1.92, 0.09, 1, and 1.3 [11], respectively.

2.3 Numerical Methods and Boundary Conditions

The finite volume numerical method is used to solve above referred equations (11-17), with a second-order upwind scheme for spatial discretization. The boundary conditions as given in Table 2, include a fixed inlet temperature and mass flow rate to ensure consistency in analysis. The steady-state simulation is performed, and for pressure-velocity coupling, the SIMPLEC algorithm is selected, as it offers better stability and efficiency. A convergence criterion of 10^{-6} for all residuals was adopted to ensure stable and accurate solutions. In CFD simulations, flow equations often stabilize quickly, and a loose criterion (like 10^{-3}) may be sufficient for velocity fields. However, the energy equation converges more slowly. If a loose criterion is used for energy, the global heat

balance (heat in versus heat out) may not be achieved, leading to significant errors in temperature distribution.

Table 1

Boundary conditions			
Domain	Boundary	BC type	
Plaster	Sides	Adiabatic	-
Water	Inlet	Mass flow	0.198 kg/sec
		Temperature	8.5 °C
	Outlet	Pressure	0

2.4 Mesh Generation and Sensitivity Analysis

The accuracy of the simulation depends on the quality of the mesh. The quadrilateral face mesh is generated in ANSYS Meshing Client, which is deemed suitable for two-dimensional analysis. Fig. 2 shows the structured mesh for the serpentine layout. This meshing approach provides a uniform distribution of cells over geometry, enabling accurate analysis of fluid flow and heat transfer.

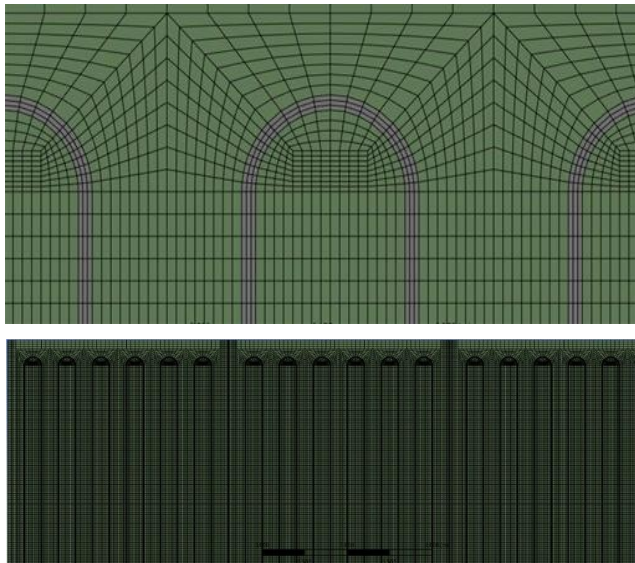


Fig. 2. Structured Mesh of serpentine layout

The sensitivity analysis of the mesh is performed using the Grid Convergence Index (GCI) method to confirm that the simulation result is independent of the mesh [11]. In this method, variations in key parameters, such as outlet temperature, are examined with successive refinements of the mesh. The GCI quantifies how these variables change as the grid density increases. The three distinct meshes (coarse, medium, and fine) are generated, representing two levels of refinement from the base mesh. The following equation is used to calculate the GCI value for the second refinement.

$$GCI_{21} = \frac{1.25e_a^{21}}{r_{21}^p - 1} \quad (15)$$

Where p is the order of convergence and represents the rate at which the solution converges as the mesh is refined. The value of p is calculated from the following equation:

$$p = \frac{1}{\ln r_{21}} \left| \ln \left| \frac{\varepsilon_{32}}{\varepsilon_{21}} \right| \right| \quad (16)$$

Where $\varepsilon_{32} = \varphi_3 - \varphi_2$ and $\varepsilon_{21} = \varphi_2 - \varphi_1$. φ_1, φ_2 and φ_3 represent the key parameter (outlet temperature) obtained from the solution for the fine, medium, or coarse mesh. In eq. 14 and 15, r is the refinement ratio and is defined as $r_i = h_{i+1} / h_i$. Where h is the cell size related to the grid and calculated as follows

$$h_i = \left[\frac{1}{n} \sum_{i=1}^n (\Delta A_i) \right]^{\frac{1}{2}} \quad (17)$$

Here n denotes the total number of cells in the mesh, and ΔA represents the area of the mesh. In Eq. 14, e_a^{21} is an approximate relative error and can be calculated as:

$$e_a^{21} = \frac{\varphi_1 - \varphi_2}{\varphi_1} \quad (18)$$

Due to the lower value of GCI as exhibited in Table 2, Mesh (M3) is considered.

Table 2

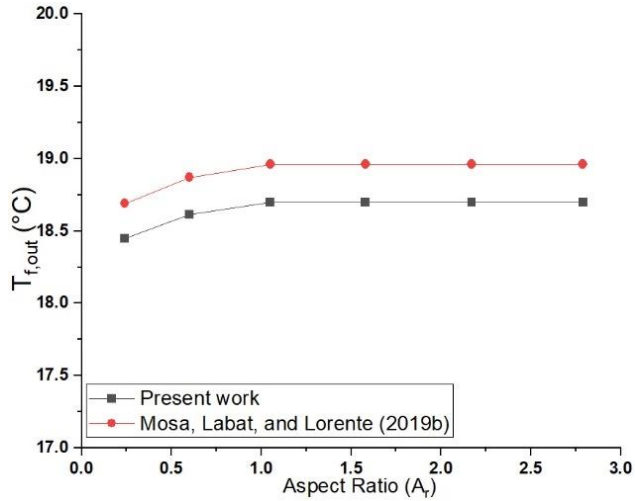
Mesh Sensitivity

Mesh	Number of Cells	Temperature (°C)	GCI (%)
Fine (M3)	496000	17.01	0.6
Medium (M2)	410000	16.99	0.9
Coarse (M1)	300000	16.96	-

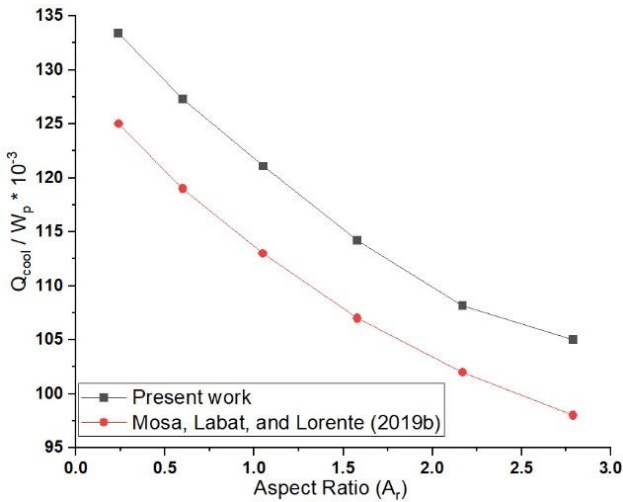
2.5 Model Validation

The model in this study is validated using similar results published in the literature [7]. The validation is carried out using two parameters, the cooling capacity-to-pump work ratio and outlet temperature. The results for these parameters are evaluated for different aspect ratios of the serpentine arrangements, as shown in Fig. 3. The Figure shows that the trends for both cooling capacity-to-pump work ratio and outlet temperature versus aspect ratio. It is clear that the cooling capacity-to-pump work ratio decreases, and the outlet temperature increases with an increase in aspect ratio. Notably, similar trends were also reported in other

published work [7]. Additionally, the error is very low as it remains around 1.3% for the outlet temperature and 6.7% for the cooling capacity-to-pump work ratio. These small errors seem to be a result of using simplified 2-D model. However, using 2-D model is worthwhile as it saves a lot of computation resources and simulated error appears to be very small. Nevertheless, 3-D CFD simulations can be carried out for future studies where highly accurate results are required.



(a)



(b)

Fig.3. Model validation results: (a) Outlet temperature, (b) Cooling capacity-to-pump power ratio

2.6 Performance Evaluation

The performance of selected RC layouts is evaluated by comparing the cooling capacity ($\dot{Q}_{cool}$), pumping power ($\dot{W}_p$), pumping cost, and piping costs (labor and purchase costs). Cooling capacity ($\dot{Q}_{cool}$) is the rate of sensible heat absorbed by the cooling surface, and is calculated as: [12]

$$\dot{q} = 8.92 (T_{space} - T_{sur})^{1.1} \quad (19)$$

The pumping power ($\dot{W}_p$) is determined by:

$$\dot{W}_p = \frac{\Delta P \dot{m}}{\rho \eta} \quad (20)$$

Table 3 shows the cost of pipes, which are used in three layouts. The labor cost is estimated as PKR 3000/per day/100 foot pipe length [13].

Table 3

Pipe cost [14]

	Serpentine	Canopy to canopy	Tree shaped
Cost of	844,	844,	844,
Pipe	Size of D	Size of D ₂	Size of D ₄
(PKR per			1628,
meter)			Size of D ₃
		3766,	3766,
		Size of D ₁	Size of D ₂
			4025,
			Size of D ₁

The pumping cost is calculated from the energy use and electricity cost, i.e., 36 PKR/kWh. The values related to performance parameters were normalized except for the ratio $\dot{Q}_{cool}/\dot{W}_p$, for comparison purposes. The maximum value of each parameter was taken as a reference.

$$\tilde{W}_p = \frac{\dot{W}_p}{\dot{W}_{p,max}} \quad (18)$$

$$\tilde{Q}_{cool} = \frac{\dot{Q}_{cool}}{\dot{Q}_{cool,max}} \quad (19)$$

$$\tilde{C}_{pumping} = \frac{C_p}{C_{p,max}} \quad (20)$$

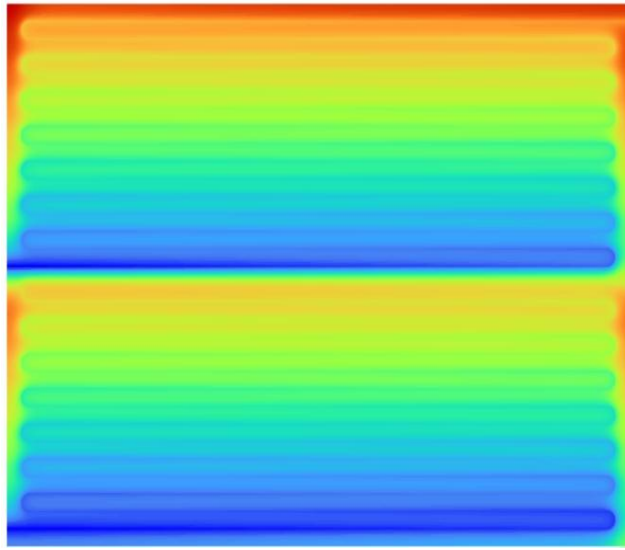
$$\tilde{C}_{piping} = \frac{Pipe\ cost + labor\ cost}{(Pipe\ cost + labor\ cost)_{max}} \quad (21)$$

3. Results and Discussion

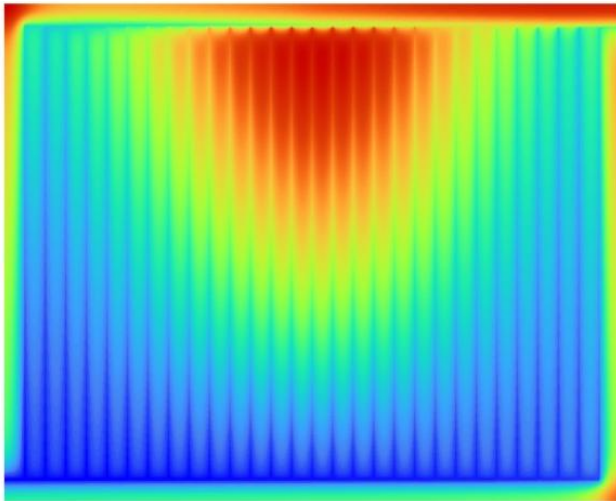
3.1 Temperature Distribution

Figure 4 shows temperature contours for the serpentine, canopy-to-canopy, and tree-shaped piping layouts. Figure 4(a) exhibits an apparent temperature gradient for the serpentine layout, with the lowest temperature occurring in the lower portion of the radiant surface; the temperature increases as water flows through the central and upper portions. This is because the water absorbs heat from the surface as it flows along the path, resulting in a non-uniformity of

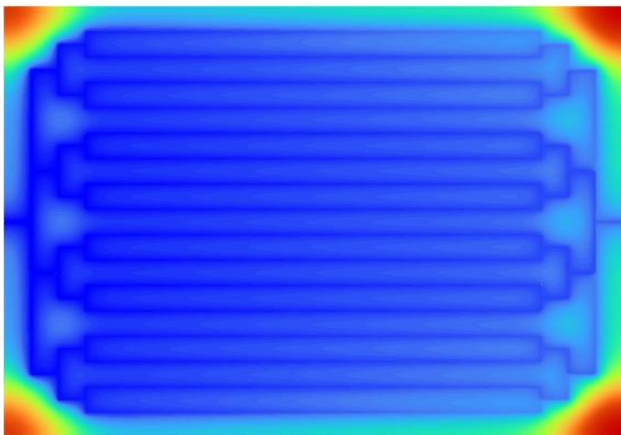
temperature distribution. This is a significant drawback of the serpentine arrangement, as it can cause uneven cooling and localized thermal discomfort. Notably, similar findings were also reported in published work [3].



(a)



(b)



(c)



Fig. 4. Temperature contours in cooling surface with (a) Serpentine, (b) Canopy-to-Canopy, (c) Tree-Shaped piping layout

Fig. 4(b) shows the temperature variation on a radiant surface embedded with a canopy-to-canopy network, which exhibits hot spots at the top and middle of the surface, primarily due to a lower mass flow rate of water. This flow imbalance results in non-uniform heat removal across the surface, limiting the effectiveness of the cooling process. In the case of a tree-shaped layout, the temperature distribution is consistent compared to other layout arrangements, as shown in Fig. 4 (c). This is due to the hierarchical branching of this layout that allows the chilling water to flow at the same rate, causing the surface to experience nearly the same temperature. However, all four corners of the surface experience hotspots as those surface areas do not encompass the tubes.

3.2 Pressure Distribution

Fig. 5 shows the pressure distribution across the radiant surface. Figure 5 (a) exhibits a pressure drop of nearly 7 kPa for the serpentine layout. This pressure drop is observed due to considerable frictional losses since water travels a longer distance from the inlet to the outlet. These losses directly increase energy consumption, as the pump must do more work to overcome the friction and maintain the flow rate. On the other hand, water in the canopy-to-canopy layout experiences smaller pressure drops, as shown in Fig. 5 (b). This is because water flows through branches, creating shorter paths. Due to the same reason, the flowing water in a tree-shaped layout also experiences smaller pressure drops compared to the serpentine layout. The results shown in Fig. 5 indicate that the tree-shaped layouts experience a 16% lower pressure drop compared to the canopy-to-canopy, despite having the same flow path length. The lowest pressure drop is primarily due to the hierarchical branching of the tree-shaped layout, which minimizes pipe losses due to friction and enhances flow uniformity.

3.3 Overall Performance

The overall performance of each of the three layouts in terms of cooling capacity, pumping power, capacity-to-pumping ratio, and piping costs is given in Table 4. It is worth noting that the cooling capacity of the serpentine layout is equivalent to that of the tree-shaped layout. Nonetheless, it has the worst $\dot{Q}_{cool}/\dot{W}_p$ – ratio. This drastic reduction is because the pressure drop in serpentine layouts is significantly higher, making the pumping power requirement higher. This causes the cost for pumping water to be the highest compared to other layouts. The canopy-to-canopy layout is much better than the serpentine one. It has a small pressure drop, and this reduces the amount of pumping power needed, and

this results in a 97 % cut in the cost of pumping when compared to serpentine. It also provides a relatively higher cooling capacity. Non-uniformity of temperature is, however, a disadvantage of this. This is because the flow of fluid in the branches is imbalanced.

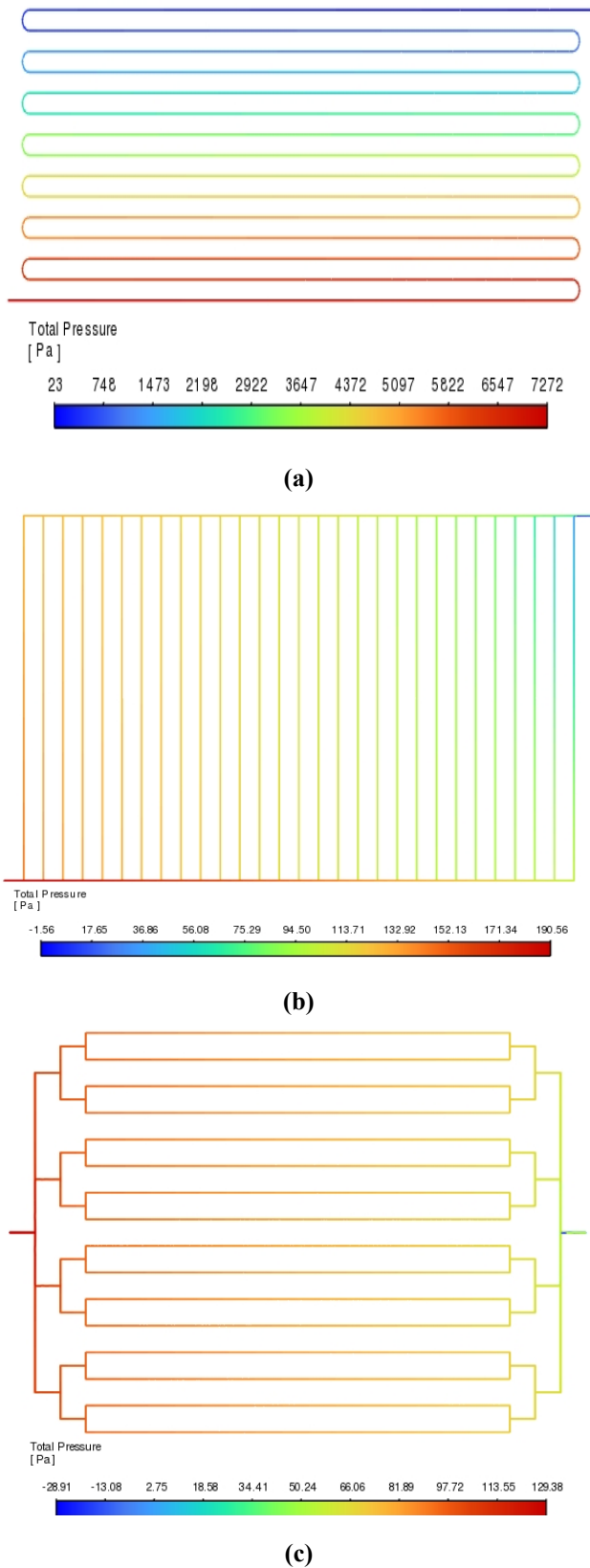


Fig. 5. Pressure distribution in flow domain for (a) Serpentine, (b) Canopy-to-Canopy, (c) Tree-Shaped piping layout

Table 4

Summary of overall performance

	Serpentine	Canopy to canopy	Tree shaped
$\bar{W}_p$ [-]	1	0.042	0.036
$\tilde{Q}_{cool}$ [-]	0.93	1	0.94
$\frac{\tilde{Q}_{cool}}{\bar{W}_p} \times 10^{-5}$ [-]	0.055	1.43	1.61
$\tilde{C}_{piping}$ [-]	1	0.99	0.84
$\tilde{C}_{pumping}$ [-]	1	0.042	0.0360

The tree-shaped layout achieves the highest $\tilde{Q}_{cool}/\bar{W}_p$ – ratio, even though its cooling capacity is nearly the same as that of serpentine. This is because the progressive branching structure minimizes frictional losses, leading to the lowest pumping power among all layouts. Additionally, the cost of piping is approximately 16% less than the cost of the other two layouts. This layout also requires the least pumping power, with a cost reduction of 14% as compared to that of the canopy-to-canopy layout and almost 97% lower as compared to the serpentine layout. This makes it the most cost-effective design, as it offers a balance between thermal performance, energy use, and cost. These results contrast with those of Mosa, Labat, and Lorente [4], who reported that the canopy-to-canopy layout is better. In fact, the layout in the study conducted by Mosa et al. [4] maintained a constant Reynolds number, whereas this study simulated layouts with a constant mass flow rate.

4. Conclusion

In this paper, numerical-based thermal, energy, and cost analyses of three pipe layouts, namely, serpentine, canopy-to-canopy, and tree-shaped, are employed in an RC system. The Ansys Fluent software was used to conduct two-dimensional steady-state simulations. It is found that the tree-shaped layout offers the best temperature distribution with the highest capacity-to-pumping ratio. In the case of the canopy-to-canopy layout, a slight increase in the cooling capacity was found compared to the other two layouts, whereas the temperature distribution was non-uniform due to flow imbalance across the branches. In terms of hydraulic performance, the serpentine layout exhibited the highest pressure drop, followed by the canopy-to-canopy layout and tree-shaped layout demonstrating the lowest drop. From the cost analysis, it is concluded that the tree-shaped layout offers the least piping cost, followed by canopy-to-canopy and serpentine,

whereas the pumping cost is highest in serpentine, intermediate in canopy-to-canopy, and lowest in tree-shaped. It is therefore concluded that the tree-shaped layout is cost-effective and provides better performance if used in an RC system.

5. References

- [1] S. Qureshi, R. A. Memon, A. G. Memon, and S. Kumar, "Energy and Cost-Saving Potential of the Radiant Cooling System for a City in Pakistan," in *International Conference on Innovation, Sustainability, and Applied Sciences*, C. Pon Selvan, N. Sehgal, S. Ruhela, and N. U. Rizvi, Eds., Cham: Springer Nature Switzerland, 2025, pp. 61–67.
- [2] K. Dharmasastha, S. M. S. Nagendra, and M. P. Maiya, "Hydronic flow path analysis on the thermal comfort performance of radiant cooling system," *Thermal Science and Engineering Progress*, vol. 39, p. 101727, Mar. 2023, doi: 10.1016/J.TSEP.2023.101727.
- [3] L. Ye, G. Lv, and Y. Ding, "Structure optimization of radiant cooling panel: A literature review and assessment," *Energy Build.*, vol. 310, p. 114097, May 2024, doi: 10.1016/J.ENBUILD.2024.114097.
- [4] N. Aste, C. del Pero, and F. Leonforte, "Water flat plate PV–thermal collectors: A review," *Solar Energy*, vol. 102, pp. 98–115, Apr. 2014, doi: 10.1016/J.SOLENER.2014.01.025.
- [5] A. F. Miguel and L. A. O. Rocha, *Tree-shaped fluid flow and heat transfer*. Springer, 2018.
- [6] A. Bejan, "Street network theory of organization in nature," *J. Adv. Transp.*, vol. 30, no. 2, pp. 85–107, 1996.
- [7] M. Mosa, M. Labat, and S. Lorente, "Role of flow architectures on the design of radiant cooling panels, a constructal approach," *Appl. Therm. Eng.*, vol. 150, pp. 1345–1352, 2019.
- [8] M. Mosa, M. Labat, and S. Lorente, "Constructal design of flow channels for radiant cooling panels," *International Journal of Thermal Sciences*, vol. 145, p. 106052, 2019.
- [9] Samiullah, "Modelling and Simulation of Energy-Efficient and Cost-Effective Radiant Cooling System for Compatible Climatic Conditions," *Mehran University of Engineering & Technology*, Jamshoro, Jamshoro, 2025.
- [10] H. Ghaedamini, M. R. Salimpour, and A. S. Mujumdar, "The effect of svelteness on the bifurcation angles role in pressure drop and flow uniformity of tree-shaped microchannels," *Appl. Therm. Eng.*, vol. 31, no. 5, pp. 708–716, 2011, doi: <https://doi.org/10.1016/j.applthermaleng.2010.10.005>.
- [11] S. Kumar et al., "Numerical investigation of the velocity and spreading characteristics of non-circular turbulent jets at different Reynolds numbers," *Transactions of the Canadian Society for Mechanical Engineering*, vol. 48, no. 1, pp. 84–93, Nov. 2023, doi: 10.1139/tcsme-2023-0028.
- [12] K.-N. Rhee, B. W. Olesen, and K. W. Kim, "Ten questions about radiant heating and cooling systems," *Build. Environ.*, vol. 112, pp. 367–381, 2017.
- [13] M. Umar, "CoolMax Group of Companies," *Personal communication*, 2023.
- [14] Supply House, "Supply House - Online Store." Accessed: Apr. 15, 2024. [Online]. Available: <https://www.supplyhouse.com/Wirsbo-Uponor-F1040750-3-4-AQUAPEX-100-ft-coil-2174000-p>