



Original Article

Analysis of heat-loss mechanisms with various gases associated with the surface emissivity of a metal containment vessel in a water-cooled small modular reactor

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ABSTRACT

In various small modular reactor (SMR) designs currently under development, the conventional concrete containment building has been replaced by a metal containment vessel (MCV). In these systems, the gap between the MCV and the reactor pressure vessel is filled with gas or vacuumed weakly, effectively suppressing conduction and convection heat transfer. However, thermal radiation remains the major mode of heat transfer during normal operation. The objective of this study was to investigate the heat-transfer mechanisms in integral pressurized water reactor (IPWR)-type SMRs under various gas-filled conditions using computational fluid dynamics. The use of thermal radiation shielding (TRS) with a much lower emissivity material than the MCV surface was also evaluated. The results showed that thermal radiation was always the dominant contributor to heat loss (48–97%), while the conjugated effects of the gas candidates on natural convection and thermal radiation varied depending on their thermal and radiative properties, including absorption coefficient. The TRS showed an excellent insulation performance, with a reduction in the total heat loss of 56–70% under the relatively low temperatures of the IPWR system, except for carbon dioxide (13%). Consequently, TRS can be utilized to enhance the thermal efficiency of SMR designs by suppressing the heat loss through the MCV.

1. Introduction

The development of small modular reactors (SMRs) has recently attracted significant global interest as a promising solution for climate change [1]. The modularized design of SMRs [2] offers greater passive safety, flexible electricity production, and synergetic application with renewable energy sources [3–5], while the standardized and modularized SMR design allows multiple modules to be manufactured continuously using the same process facilities [5,6]. In addition, because SMRs are transportable by ship or truck from off-site factories, they can be assembled and installed on-site. Consequently, because the construction period can be reduced, long-term economic competitiveness is more likely to be achieved.

Many SMRs employ a design in which a metal containment vessel (MCV) surrounds the reactor pressure vessel (RPV) (Fig. 1). Acting as a substitute for a concrete containment building, which can be up to 1 m thick, this manufacturable MCV improves the modularity of the system and can withstand the pressure of steam released during an accident [7,

8,62]. In addition, the long-term safety of SMR modules has been strengthened by utilizing the MCV as part of a passive safety system [7–11,62]. For example, in the emergency core cooling system of the NuScale Power Module (NPM), which is a representative integral pressurized water reactor (IPWR)-type SMR, the MCV is designed to act as a condenser and a recirculation sump during a potential accident [7,8, 10–12,62]. Due to the high thermal conductivity of the MCV, core cooling is maintained via coolant recirculation as long as sufficient external cooling is maintained.

The gap between the RPV and MCV is typically filled with air [16, 18], inert gas [13,14,17], or a weak vacuum [12,15,19]. The filling gas can be selected from numerous potential candidates based on its flammability [63,64] and condensation during an accident (Table 1) but, during normal operation, gas-filled or weak-vacuum gaps act as efficient insulators, effectively suppressing conductive and convective heat transfer.

Although the gap suppresses heat transfer via conduction and convection during normal operation, radiative heat transfer is more dominant. Radiative heat transfer has primarily been considered significant

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Abbreviation			
2d	Two Dimensional	N_θ	The number of angular discretization in the polar angle [–]
CFD	Computational Fluid Dynamics	N_{θ_p}	Polar angle pixel number [–]
DCMS	Direct Current Magnetron Sputtering	N_ϕ	The number of angular discretization in the azimuthal angle [–]
DO	Discrete Ordinates	N_{ϕ_p}	Azimuthal angle pixel number [–]
DOM	Discrete Ordinates Model	Nu	Nusselt number [–]
DTRM	Discrete Transfer Radiation Model	Nu_{nc}	Natural convection Nusselt number [–]
HTC	Heat Transfer Coefficient	n	Refractive index [–]
HTGR	High-Temperature Gas-Cooled Reactor	$\vec{n}_s$	Normal vector [–]
IPWR	Integrated Pressurized Water Reactor	p	Pressure [Pa]
MCV	Metal Containment Vessel	Pr	Prandtl number [–]
MSR	Molten Salt Reactor	Pr_t	Turbulent Prandtl number [–]
MC	Monte Carlo	Q_R	Radiative heat transfer rate [W]
NPM	NuScale Power Module	q_{in}	Incident radiative heat flux [W/m ²]
PMAC	Planck Mean Absorption Coefficient	q_{rside}	Radiative heat flux on the sidewall [W/m ²]
PWR	Pressurized Water Reactor	Ra	Rayleigh number [–]
RCCS	Reactor Cavity Cooling System	$\vec{r}$	Position vector [–]
RCP	Reactor Coolant Pump	s	Path length [m]
RCS	Reactor Coolant System	$\vec{s}$	Direction vector [–]
RPV	Reactor Pressure Vessel	$\vec{s}'$	Scattering direction vector [–]
RTE	Radiative Transport Equation	$\vec{s}_{out}$	Outgoing direction vector [–]
S2S	Surface-to-Surface	S_h	Volumetric heat source [W/m ³]
SMR	Small Modular Reactor	T	Temperature of the medium [°C]
SST	Shear Stress Transport	ΔT	Temperature difference [°C]
SG	Steam Generator	T_{ref}	Reference temperature [°C]
y+	Y plus	T_{SG}	Temperature of the steam generator component [°C]
TRS	Thermal Radiation Shielding	T_{surr}	Surrounding temperature [°C]
Nomenclature		T_{surf}	Surface temperature [°C]
A	Heat transfer area [m ²]	$T_{surf,PRV}$	Surface temperature of the reactor pressure vessel [°C]
a	Absorption coefficient [m ^{−1}]	$T_{surf,MCV}$	Surface temperature of the metal containment vessel [°C]
c_p	Specific heat capacity [J/kg•°C]	$\vec{v}$	Velocity vector [m/s]
E	Total energy [J]	Greek letter	
Gr	Grashof number [–]	β	Expansion coefficient [°C ^{−1}]
g	Gravitational acceleration [m/s ²]	ε_s	Surface emissivity [–]
h	Heat transfer coefficient [W/m ² •°C]	η	Wavenumber [cm ^{−1}]
h_{nc}	Natural convection heat transfer coefficient [W/m ² •°C]	ϕ	Azimuthal angle [steradian; sr]
h_{air}	Heat transfer coefficient of air [W/m ² •°C]	θ	Polar angle [sr]
h_{cand}	Heat transfer coefficient of the candidate gas [W/m ² •°C]	κ	Turbulence kinetic energy [m ² /s ²]
h_{rel}	Relative heat transfer coefficient [W/m ² •°C]	μ	Dynamic viscosity [Pa•s]
I	Radiation intensity [W/m ² •sr]	μ_t	Turbulent viscosity [Pa•s]
I_s	Radiation intensity on the surface [W/m ² •sr]	ρ	Density [kg/m ³]
I_b	Total blackbody radiation intensity [W/m ² •sr]	σ	Stefan-Boltzmann constant [5.67 × 10 ^{−8} W/m ² •K ⁴]
k	Thermal conductivity [W/m•°C]	σ_s	Scattering coefficient [m ^{−1}]
k_{eff}	Effective thermal conductivity [W/m•°C]	τ_{ij}	Deviatoric stress tensor [Pa]
L_c	Characteristic length [m]	φ	Phase function [–]
N_r	The number of cell division numbers in the radial direction [–]	Ω	Solid angle [sr]
N_x	The number of cell division number in the axial direction [–]	ω	Specific dissipation rate [s ^{−1}]

only for high-temperature applications exceeding 600 °C due to the proportional relationship with the fourth power of the surface temperature. For example, Liu et al. [20] and Chen et al. [21] investigated the radiative heat transfer in a passive residual heat removal system for molten salt reactors (MSRs), whose operating temperature exceeds 600 °C, while Zhang et al. [22] developed a radiative heat transfer model for molten salts and confirmed the contribution of thermal radiation over a temperature range of 500–1000 °C. For high-temperature gas-cooled reactors (HTGRs), which operate at a maximum temperature of 1000 °C, Frisani [23], Takamatsu [24], and Zhao et al. [25] conducted numerical evaluations of the heat-removal performance of

reactor cavity cooling systems (RCCSs). Furthermore, Oh et al. [26] evaluated the heat-removal performance of an improved RCCS for an HTGR both numerically and experimentally. These studies demonstrated that, although radiative heat transfer is insignificant within the molten salt itself, it plays a significant role in gas-filled systems under high operating temperatures, accounting for up to 91.64% of the total heat transfer.

However, radiative heat transfer in pressurized water reactors (PWRs), which operate at temperatures lower than 330 °C, has often been overlooked [27,28], even though several studies have reported that radiative heat transfer is still dominant even at lower temperatures.

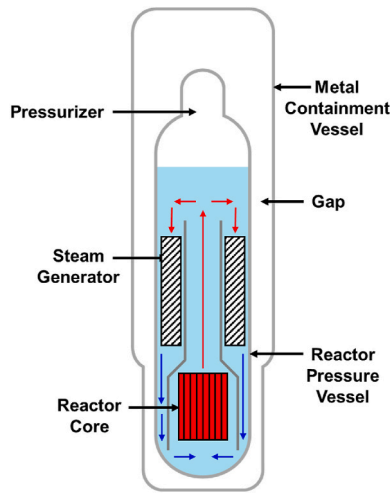


Fig. 1. A schematic of a metal containment vessel (MCV) with an integrated design consisting of small modular reactors.

Table 1

Gap filling gas and operating pressure for modern SMRs.

Reactor	Country	Gap conditions	
		Material	Pressure (under normal operation)
NuScale [12]	US	Weak vacuum	0.067 bar
BWRX-300 [13]		Nitrogen gas	1 bar
IRIS [14]		Nitrogen gas	1 bar
Westinghouse SMR [15]		Vacuum	Unknown
mPower [16]		Air	1 bar
FLEXBLUE [17]	France	Inert gas	1 bar
NUWARD [18]		Air	1.5 bar
Innovative SMR [19]	Republic of Korea	Vacuum	To be determined

Kumar et al. [29] reported that radiative heat transfer accounts for 62.3% of the total heat transfer even at a temperature of 120 °C, while Yoo et al. [30] confirmed that radiation accounts for over 60% of the total heat transfer at 1.5 atm and 225 °C in a spent fuel storage system filled with inert gas. These studies suggest that the effect of thermal radiation in conjugated heat transfer is non-negligible even at temperatures lower than 330 °C. Thus, when designing an MCV with sufficient thermal insulation, radiative heat transfer has to be considered.

The effects of various system parameters, including radiation absorption, internal pressure, and emissivity, on radiative heat transfer have been investigated in previous studies. Kumar et al. [31] and Dehbi et al. [32] have confirmed that the radiation absorption effect reduces the contribution of radiative heat transfer to total heat loss but enhances heat transfer via natural convection. In addition, Barforoush et al. [33] examined the impact of internal pressure, the emissivity of metal shields, and the heater temperature using experimental and numerical analysis, reporting that, to decrease heat loss due to thermal radiation and natural convection, the emissivity and internal pressure need to be lower. While previous studies have provided valuable insights into heat loss due to conjugated heat transfer, including thermal radiation [20–26,29–33], operating conditions such as the internal pressure, operating temperature, gas type, and geometry strongly affect heat-transfer mechanisms. Therefore, to evaluate the intrinsic heat-loss mechanisms for IPWR-type SMRs that employ MCVs, their specific geometry and numeric modeling have to be included for analysis.

The objective of this study was to analyze using computational fluid

dynamics (CFD) the conjugated heat-transfer mechanisms, including radiation and natural convection, associated with the MCV in an IPWR-type SMR. Inspired by one of the previous studies [33], the effectiveness of thermal radiation shielding (TRS) in terms of reducing heat loss was also investigated. To identify the contribution of radiative heat transfer within the gap between the RPV and MCV, the heat-transfer mechanisms were investigated using ANSYS FLUENT 21.2 software. Various gap conditions, including the gas type, internal pressure, and surface emissivity, were considered in this analysis. In particular, air, argon, xenon, nitrogen, and carbon dioxide were tested as the filling gas and compared with the use of a vacuum. To achieve the objective of this study, the methodology section (Section 2) offers an outline of the reference reactor module and described anticipated heat transfer mechanisms, identifying candidates expected to influence them. In addition, the boundary conditions and model for CFD analysis are delineated. Among the radiation models, the discrete ordinates (DO) radiation model is validated and optimized through the benchmark problem because of its importance in this study. In the results section (Section 3), a mesh is chosen based on the mesh sensitivity analysis. In addition, the heat loss mechanisms in the MCV are analyzed, discussing the effectiveness of TRS from the insulation perspective. Finally, the findings of this study are summarized in the conclusion section (Section 4).

2. Methodology for numerical analysis

2.1. Description of reference reactor: innovative small modular reactor

An innovative SMR (i-SMR), an IPWR-type SMR, currently under development in South Korea, was employed as the reference reactor in the present study to investigate the heat-transfer mechanisms. In the i-SMR design, the major components of the nuclear steam supply system, such as the 2.4-m core, top-mounted in-core instrument, pressurizer, four canned motor reactor coolant pumps (RCPs), and a once-through helical steam generator (SG), are modularized and integrated with the RPV. During normal operation, the operating temperature of the reactor coolant system (RCS) is 295.7–323.0 °C. To maintain a sufficiently subcooled state within this operating temperature range, the RCS pressure was designed to be 15.8 MPa. The thermal and electric power of the i-SMR was 540 MWt and 170 MWe, respectively. In order to replace old thermoelectric power plants and large commercial reactors whose electric power generation exceeds 600–1000 MWe, four to six reactor modules are planned to be installed within a single building.

Fig. 2 presents the i-SMR reactor module installed in an open-top reactor module cavity. The cavity has been designed to allow air circulation, with a height of 28.7 m. The MCV of the reactor module contains an RPV. The base materials of the RPV and MCV are stainless steel and carbon steel,

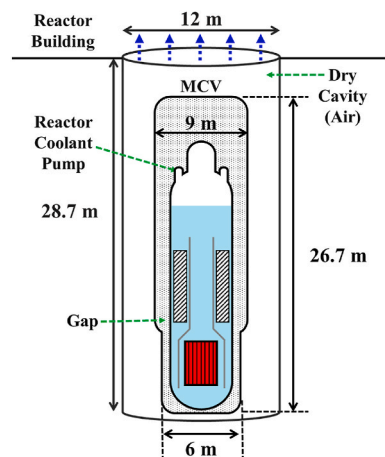


Fig. 2. Schematic of the i-SMR module in an open-top reactor module cavity.

respectively, while their heights are 22.8 m and 26.7 m, respectively. The diameter of the MCV is 6 m at the bottom and 9 m at the top to promote the rapid accumulation of condensate to maintain a sufficient level of water during an accident. The diameter of the cavity is 12 m, meaning that the MCV is installed with a radial distance margin of 1.5 m. External insulation is not employed for the outer walls of each vessel because the MCV is cooled using an emergency coolant in an accident. Instead of insulation materials, a gap between the MCV and RPV is used. The important design parameters for the i-SMR are summarized in Table 2.

2.2. Heat-loss mechanisms within the metal containment vessel during normal operation

Fig. 3 illustrates the expected heat-transfer mechanisms within the MCV in accordance with the strategy used to fill the gap to the RPV. Conduction, natural convection, and radiation are depicted using green, red, and black-dotted arrows, respectively. The direction of each arrow indicates the heat transfer direction. Conduction occurs radially within solid structures such as the RPV and MCV. On the surface of the RPV and MCV, thermal radiation and natural convection occur.

As shown in Fig. 3(a), in the gas-filled gap, both natural convection and radiation occur, but conduction can be ignored due to intrinsically low thermal conductivity. In contrast, as illustrated in Fig. 3(b), the vacuum allows only for radiative heat transfer because natural convection cannot occur in the absence of a heat transfer medium. Both natural convection and thermal radiation within the MCV are largely dependent on the wall temperature of the RPV and MCV. Accordingly, the conjugated heat transfer becomes more complicated because natural convection and radiation are expected to affect each other.

To identify the effect of the gas species employed to fill the gap on natural convection, the dimensionless Nusselt number (Nu), Grashof number (Gr), Prandtl number (Pr), and Rayleigh number (Ra), which are often employed to evaluate convective heat transfer, were calculated. The Nusselt number (Eq. (1)) represents the characteristics of convective heat transfer compared to conductive heat transfer. For natural convection, the Nusselt number can be expressed as a function of the Rayleigh number (Eq. (2)), where the constants C and m are determined by specific conditions such as the geometry, surface roughness, and flow regime of the interest area [34]. By reducing the Rayleigh number, a lower Nusselt number is achieved, indicating lower natural convection. Therefore, the thermal expansion coefficient (β), temperature difference (ΔT), density (ρ), specific heat capacity (c_p), characteristic length (L_c), dynamic viscosity (μ), and thermal conductivity (k) were compared to

predict heat transfer through natural convection.

$$Nu_{nc} = \frac{\text{Convection}}{\text{Conduction}} = \frac{h_{nc} L_c}{k} = C Ra^m \quad (1)$$

$$Ra = Pr Gr = \frac{g \beta c_p \rho^2 \Delta T L^3}{\mu k} \quad (2)$$

For the same geometry, the characteristic length and gravitational acceleration (g) are constants. The temperature difference is determined by the result of the convective heat transfer, while the thermal expansion coefficient is determined by the inverse of the absolute temperature, and this did not differ between the candidate gases at the same temperature. Accordingly, to predict the natural convection depending on the filling gas, the change of the Rayleigh number according to the specific heat capacity, density, viscosity, and thermal conductivity of the candidates was estimated.

To compare natural convection effects, a weak vacuum, air, and nitrogen were selected as gap candidates because they have been employed in several state-of-the-art SMRs (Table 1). In terms of flammability control, carbon dioxide, a representative diluent protecting against hydrogen combustion [35], was also tested. In the present study, the weak vacuum was simulated using the thermal properties of air at a pressure of 0.067 bar, which is referenced in the NPM design [12], while the properties of the other gases were obtained under an atmospheric pressure of 1 bar. The thermal properties of the gases were referenced from the NIST database [36]. However, as shown in Fig. 4, except for the density of the weak vacuum, there was not much variation in the thermal properties between air, carbon dioxide, and nitrogen. Therefore, xenon and argon were also considered as inert gas candidates. Xenon has distinct thermal properties, such as a higher density and viscosity and a lower heat capacity and thermal conductivity, while argon exhibits thermal properties that are between xenon and the other gas candidates.

The radiative heat transfer rate (Q_R) can be calculated using Eq. (3) based on the surface (T_{surf}) and surrounding (T_{surr}) temperature, the heat transfer area affected by the geometry (A), and the Boltzmann constant (σ). Thus, to reduce radiative heat transfer without changing the design parameters, reducing the surface emissivity (ϵ_s) is important [34]. The emissivity of an object, which indicates how much radiation is emitted compared to the surface of the black body, is determined by its material and surface properties. A lower emissivity is indicative of lower radiative heat transfer.

$$Q_R = \epsilon_s \sigma A (T_{surf}^4 - T_{surr}^4) \quad (3)$$

For the i-SMR, the emissivity of the carbon steel RPV was confirmed to be 0.7, while the emissivity of the stainless steel MCV was found to be 0.4 [37]. To reduce radiative heat transfer, the TRS of the present study assumed the arrangement of a replaceable-thin aluminum layer with an emissivity of 0.04 coated using the direct current magnetron sputtering (DCMS) [38]. As described in experimental studies [39,40], the aluminum coating layer deposited through the DCMS can maintain the intrinsic low emissivity in the thermal radiation range as well as has various advantages such as precise thickness control, material homogeneity, scalability over large areas, and reliable adherence strength to substances compared to the evaporation process. Because the RPV had the higher surface emissivity than the MCV, the stronger reduction in heat loss was expected when the TRS was applied. However, due to the more complicated structure of the outer wall surface of the RPV compared with the MCV, it was difficult to position the TRS on the RPV. Accordingly, the TRS was applied to the inner wall of the MCV. Emissivity can also be influenced by the wavelength, but the emissivity of the vessel materials used in the present study remains relatively stable and exhibit little change depending on wavelength [37]. Therefore, in this study, the emissivity values of vessel surfaces were set as constants to simplify the simulation. Furthermore, the additional thermal resistance caused by the aluminum layer was neglected under the assumption that

Table 2
Design parameters of the i-SMR.

	Parameter	Value
Reactor Core	Thermal power	540 MWt
	Electrical power	170 MWe
	Inlet/outlet temperature	295.7 °C/323 °C
	Pressure	15.8 MPa
	Fuel	UO ₂
Reactor Cavity	Bulk temperature	Room temperature
	Inner diameter	12 m
	Cavity height	28.7 m
Metal Containment Vessel	Outer diameter	9 m (upper)/6 m (lower)
	Height	26.7 m
	Thickness	0.1 m
	Material	Stainless steel 316
Reactor Pressure Vessel	Outer diameter	2.26 m (pressurizer region)/ 4.7 m (steam generator region)/ 3.8 m (downcomer region)
	Height	22.88 m
	Thickness	0.23 m
	Material	Carbon steel (SA508)

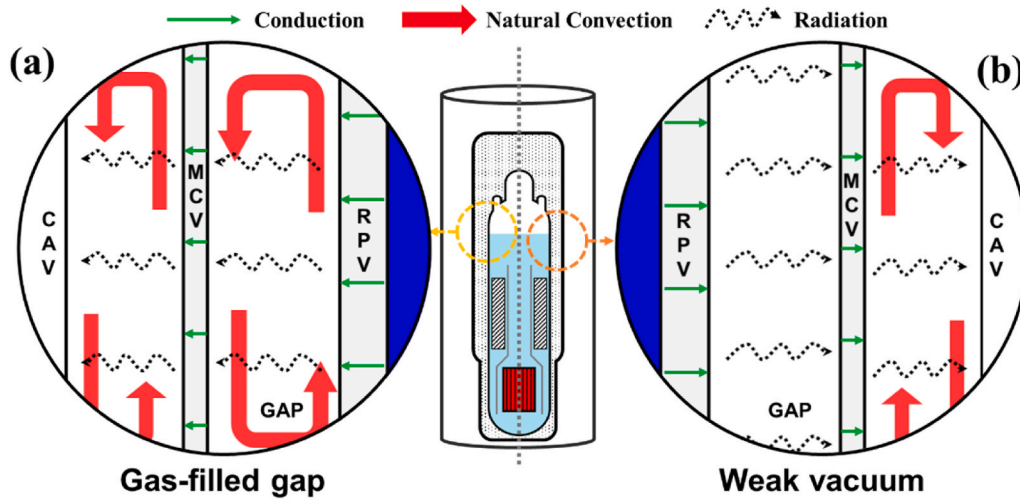


Fig. 3. Predicted heat-transfer mechanisms within the gap between the MCV and RPV when filled with (a) a gas and (b) a vacuum.

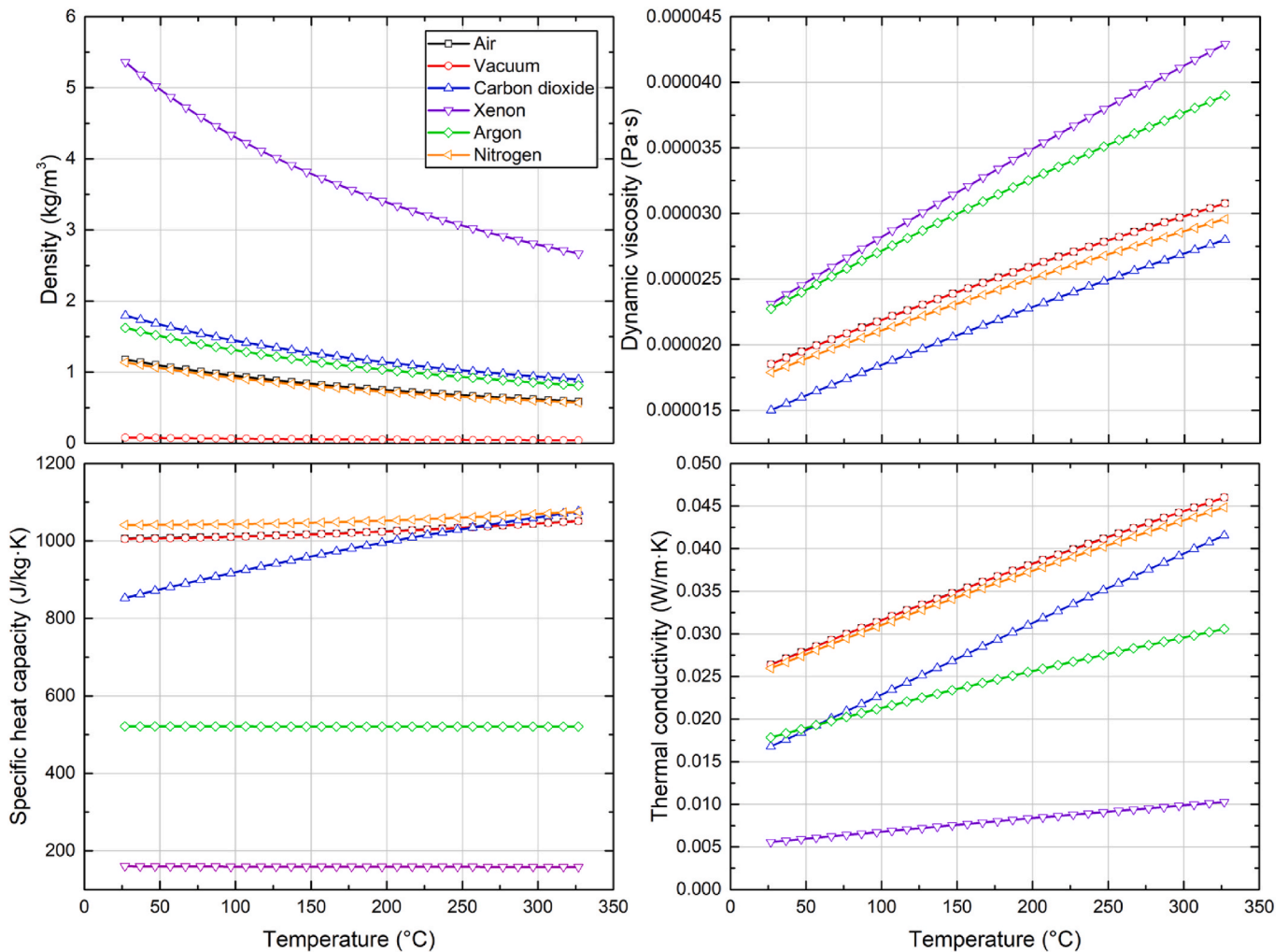


Fig. 4. Thermal properties of gap-filling gas candidates as a function of temperature.

it was sufficiently thin.

The radiative transport equation (RTE) presented in Eq. (5), which is derived from Eq. (4), was used to calculate the radiative heat transfer for the CFD analysis [41]. The RTE takes into account the variation in the radiation intensity (I) due to absorption, emission, and scattering at a

specific position ($\vec{r}$) in each direction ($\vec{s}$) for a unit path length (ds) (Fig. 5). The left-hand side of the RTE ($\frac{dI(\vec{r}, \vec{s})}{ds}$) represents the rate of change in the radiation intensity per unit path length, while the right-hand side considers the attenuation ($-(a + \sigma_s)I(\vec{r}, \vec{s})$) in

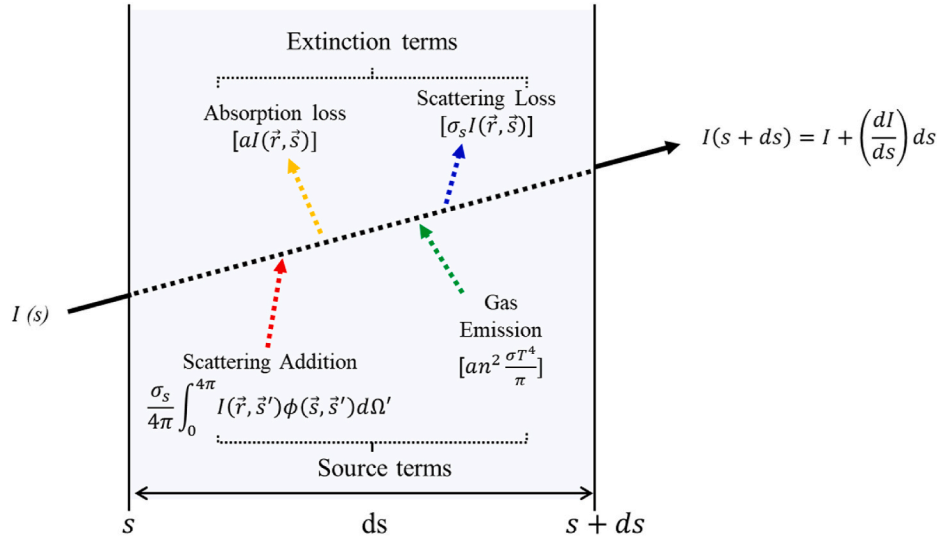


Fig. 5. Variation in the radiation intensity along a unit path length.

radiation intensity due to absorption (a) and scattering (σ_s), the radiation absorbed by the medium ($an^2\frac{\sigma T^4}{\pi}$), and the incident radiation present at position $\vec{r}$ due to background scattering ($\frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}') \phi(\vec{s}, \vec{s}') d\Omega'$). As illustrated by the RTE, the absorption coefficient (a) of the target medium influences the radiative heat transfer via absorption. For this reason, carbon dioxide, a polar molecule, was expected to exhibit distinguished not only bulk temperature but also performance of convective heat transfer compared to the other filling gas candidates. On the other hand, the absorption effect of the small amount of vapor steam in the air was ignored by assuming that the air used in this analysis was dry.

$$dI = I(s+ds) - I(s) \quad (4)$$

$$\frac{dI(\vec{r}, \vec{s})}{ds} = -(a + \sigma_s)I(\vec{r}, \vec{s}) + an^2\frac{\sigma T^4}{\pi} + \frac{\sigma_s}{4\pi} \int_0^{4\pi} I(\vec{r}, \vec{s}') \phi(\vec{s}, \vec{s}') d\Omega' \quad (5)$$

2.3. Computational fluid dynamics analysis

To conduct accurate CFD simulations, it is important to generate a reasonable geometry and establish appropriate boundary conditions. Therefore, this study generated a simplified i-SMR using the vessel geometry described in Table 2. To reduce the computational burden, a two-dimensional axisymmetric geometry was generated using ANSYS SpaceClaim 2021.R2 (Fig. 6). While radiation transfer fundamentally in three dimensions, some studies have demonstrated that in simple-symmetrical geometries it can be uniformly observed along with the azimuth angle [42] and accurately predicted as in three-dimensional simulations [42,43]. Therefore, adopting a 2d-axisymmetric geometry to simulate the simplified i-SMR could be an efficient way to ensure accuracy while reducing computational costs. To ensure stable computational convergence, the circulation of the water inside the RPV was not simulated, and the inner wall of the RPV was discretized to reflect the temperature of specific regions, such as the pressurizer, steam generator, and downcomer. Protruding components such as valves and pumps were not considered in the geometric model.

The boundary conditions for the CFD simulations are presented in Fig. 7. Temperatures of 323.0 °C (red), 295.7–323.0 °C (yellow), and 295.7 °C (blue) were assigned from top to bottom, respectively, representing the temperature of the descending coolant in direct contact with the RPV during normal operation. The coolant temperature of the soluble boron-free i-SMR may exhibit small fluctuations of approximately

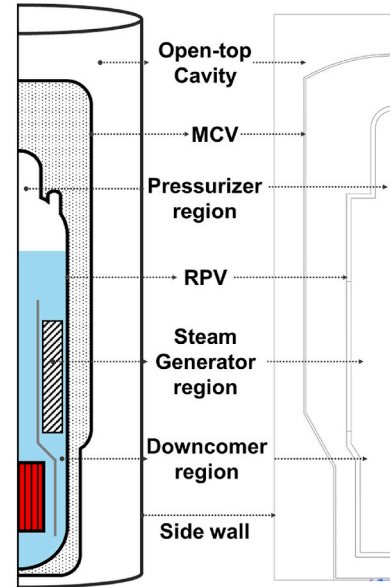


Fig. 6. 2-D axisymmetric geometry of the i-SMR.

10 °C, even though there is a 50% power output variation during the load follow-up operation [44]. Therefore, setting the boundary conditions at constant temperatures is reasonable. The temperature of the steam generator region was set to increase linearly as the height increased (Eq. (6)), while the temperatures of the other regions were fixed. The x-axis (gray) was set as the center line for the axisymmetric model in accordance with the FLUENT code manual [41]. In addition, the top was modeled as a pressure outlet (violet) to reflect an open cavity. Because the i-SMR module is initially employed at room temperature, the cavity wall was assumed to be 26.85 °C (green). In addition, heat was not transferred through the lower wall, which was defined as adiabatic (black).

$$T_{SG} (^\circ\text{C}) = 295.7 + 3.592 \cdot (x - 7.55) \quad (6)$$

Heat transfer analysis relies on the accurate input of relevant parameters. In general, thermophysical properties, including density, specific heat capacity, and thermal conductivity are temperature-dependent variables. In the present study, the properties of the vessel materials were calculated using constants and polynomial functions of

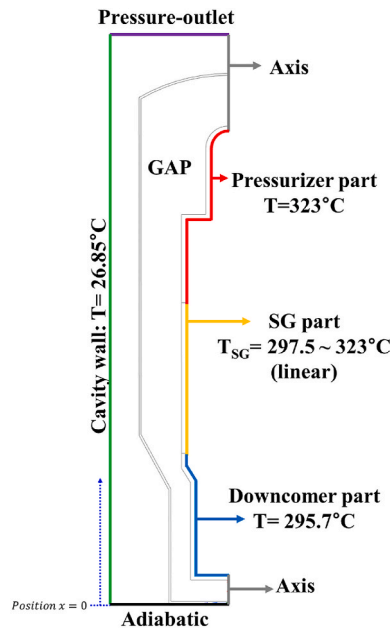


Fig. 7. Boundary conditions for the CFD simulations.

temperature (Table 3). For metal density, constant values were employed at 310.05 °C and 100 °C, which were the average temperature of the boundary conditions of the RPV inner wall and the anticipated temperature of MCV, respectively. On the other hand, specific heat capacity and thermal conductivity were set to vary with temperature as illustrated in Fig. 8 [45,46]. Concerning thermal radiation, the entire solid surfaces were set as an opaque condition to prevent the transmission of thermal radiation.

For gas materials, thermal properties, including density, were evaluated using the polynomial functions (Table 4). Concerning thermal radiation within gas media, while the scattering of radiation was ignored in all cases to simplify the CFD analysis, the absorption was considered in a specific medium. At low temperatures, i.e., target temperature range of this study, nitrogen, argon, and xenon are inert gases with chemical stability. In addition, dry air has been deemed a non-absorption medium in the infrared region because nitrogen and oxygen (99.9% of the dry air) feature a single fundamental band and no permanent dipole moment [47]. Therefore, this study assumed the four candidates as transparent media [41,48]. However, carbon dioxide exhibits a distinct characteristic related to radiation absorption. Although the FLUENT database provided an absorption coefficient of 0.43 (m⁻¹) for carbon dioxide [41], the reliability of this constant value may be questioned because it is wavelength-dependent within the infrared range [49]. Therefore, the Planck mean absorption coefficient (PMAC), defined as Eq. (7), was utilized for the carbon dioxide. The correlation of PMAC, provided by RADCAL [50,51], is described in Table 5 and plotted in Fig. 9. This model is generally recognized to be accurate up to a temperature of 600 K [52].

Table 3
Properties of the vessel materials.

	RPV (carbon steel) [45]	MCV (stainless steel 316) [46]
ρ (kg/m ³)	7760 (T = 310.05 °C)	7920 (T = 100 °C)
c_p (J/kg·K)	$49.8497 + 2.1722(T) - 0.00355(T^2) + 2.3E-6(T^3)$	$458.9848 + 0.1328(T)$
k (W/m·K)	$12.19836 + 0.15383(T) - 0.000241(T^2) + 1.08E-7(T^3)$	$9.248 + 1.57E-2(T)$
ϵ_s (—)	0.7	0.4 (reference)/0.04 (shielding)

$$PMAC = \frac{\int_0^\infty a_\eta I_{b\eta} d\eta}{\int_0^\infty I_{b\eta} d\eta} = \frac{1}{I_b} \int_0^\infty a_\eta I_{b\eta} d\eta = \frac{1}{I_b} \sum_i^N \int_0^\infty a_{i\eta} I_{b\eta} d\eta \quad (7)$$

where I_b , η , i , and N are the total blackbody intensity, wavenumber, specific number of the spectral line band, and total band number, respectively.

The solver settings are summarized in Table 6. Because the target scenario in the present study is normal operation, where the inlet and outlet temperatures do not change, we conducted the simulations under steady-state conditions. To simulate the natural convection flow generated near the walls, the k- ω shear stress transport (SST) model was adopted as the turbulence model because it has been widely used in many previous studies [29,31,32,53,54]. The k- ω SST model was developed by Menter [55], in which the standard k- ω model and the k- ϵ model are combined using a blending function. The standard k- ϵ model and the standard k- ω model have exhibited good performance in the freestream region and the near-wall region, respectively. Therefore, this study ensured accuracy and reliability in predicting natural convection flows by utilizing the k- ω SST model, which integrates the strengths of both models. The discrete ordinates model (DOM) was utilized to calculate the RTE due to its moderate computational costs and the availability of all-optical thickness ranges [29,32,54]. A benchmark problem to ensure efficiency and accuracy, along with a brief description of the DOM, was addressed in Section 2.3.1. For spatial discretization, the body force weighted scheme was employed as the pressure scheme to simulate buoyancy-driven flow where the body force is dominant in the momentum equation. This method evaluates the face pressure with an assumption of a constant normal gradient of the difference between pressure and body force. In addition, the second-order upwind scheme was used for momentum, energy, and DO intensity to evaluate them with high-order accuracy through a Taylor series expansion of the cell-centered solution. Finally, pressure-velocity coupling was conducted using a coupled algorithm because it represents a robust and efficient single-phase implementation for steady-state flows. In this study, all model constants related to the models were set to default values, as detailed in the Fluent theory guide [41].

2.3.1. The radiation model: discrete ordinates model

In the FLUENT, five radiation models were provided for solving the RTE: P-1, Rosseland, Discrete Transfer Radiation Model (DTRM), Discrete Ordinates Model (DOM), Surface-to-Surface (S2S), and Monte Carlo (MC) model [41]. Choosing the most suitable radiation model for a given analysis is essential. In this section, the reason was described based on the distinctive features of each model. The analyzed conditions and the corresponding unsuitable models are described below.

First, a wide range of optical thicknesses from transparent media to 3 or more was covered in the present study because various gases, including carbon dioxide, were used as a gap filler. Therefore, the P-1 and Rosseland models, known to be valid only in optically thick media, were considered inadequate. Conversely, the S2S model was deemed unsuitable for analysis that included absorption effects, as its fundamental assumption is the neglect of all interactions within the medium. Second, the MC model, only provided for three-dimensional geometry in FLUENT, was unsuitable for this study using a two-dimensional axisymmetric geometry. Finally, the DTRM has a low computational efficiency associated with the limitations of parallel calculation. Consequently, the DOM was adopted as the only appropriate model to conduct more accurate and efficient calculations for this analysis.

The DOM solves the RTE by discretizing a finite number of non-overlapped solid angles based on the global Cartesian system (x, y, z). For each solid angle associated with the discrete direction vector ($\vec{s}$), the RTE of DOM, represented as Eq. (8) due to the absence of radiation scattering, was calculated. In the DOM, each octant was subdivided into $N_\theta N_\phi$ control angles, N_θ and N_ϕ are the number of polar and azimuthal

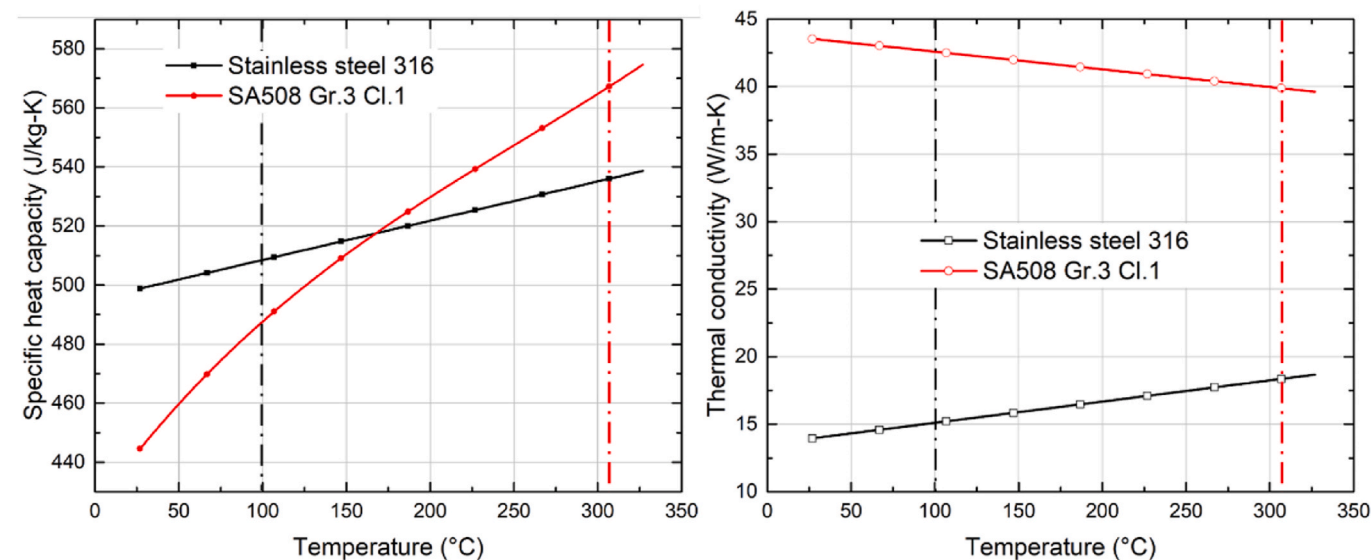


Fig. 8. Specific heat capacity (Left) and thermal conductivity (Right) of stainless steel 316 (black line) and carbon steel (red line).

Table 4
Thermal properties of the gas candidates expressed as polynomial functions.

	ρ (T) [kg/m ³]	c_p (T) [J/kg·K]	k (T) [W/m·K]	μ (T) [kg/m·s]	a (m ⁻¹)
Air	$2.50385 - 0.00577(T) + 4.32036 \times 10^{-6}(T^2)$	$955.29798 + 0.15243(T)$	$0.00722 + 6.52146 \times 10^{-5}(T)$	$6.68893 \times 10^{-6} + 4.05979 \times 10^{-8}(T)$	0
Vacuum	$0.17281 - 3.97733 \times 10^{-4}(T) + 2.97726 \times 10^{-7}(T^2)$	$953.00806 + 0.15603(T)$	$0.00717 + 6.5265 \times 10^{-5}(T)$	$6.67086 \times 10^{-6} + 4.06154 \times 10^{-8}(T)$	0
Carbondioxide	$3.83869 - 0.00889(T) + 6.6789 \times 10^{-6}(T^2)$	$640.53806 + 0.74252(T)$	$-0.00806 + 8.29598 \times 10^{-5}(T)$	$2.30234 \times 10^{-6} + 4.32413 \times 10^{-8}(T)$	PMAC
Xenon	$11.45532 - 0.02652(T) + 1.99277 \times 10^{-5}(T^2)$	$160.89063 - 0.00418(T)$	$9.3991 \times 10^{-4} + 1.56429 \times 10^{-5}(T)$	$3.53329 \times 10^{-6} + 6.6087 \times 10^{-8}(T)$	0
Argon	$3.45492 - 0.00796(T) + 5.96338 \times 10^{-6}(T^2)$	$522.18085 - 0.00289(T)$	$0.00547 + 4.23091 \times 10^{-5}(T)$	$6.94653 \times 10^{-6} + 5.40 \times 10^{-8}(T)$	0
Nitrogen	$2.42107 - 0.00558(T) + 4.17704 \times 10^{-6}(T^2)$	$1001.67309 + 0.11324(T)$	$0.00759 + 6.26869 \times 10^{-5}(T)$	$6.58123 \times 10^{-6} + 3.87872 \times 10^{-8}(T)$	0

Table 5
Polynomial fit of the Planck Mean Absorption Coefficient for carbon dioxide (Pressure: 1 [atm]) [50,51].

Planck mean absorption coefficient (PMAC)[m ⁻¹]	
Format	$C_0 + C_1 \left(\frac{1000}{T}\right) + C_2 \left(\frac{1000}{T}\right)^2 + C_3 \left(\frac{1000}{T}\right)^3 + C_4 \left(\frac{1000}{T}\right)^4 + C_5 \left(\frac{1000}{T}\right)^5$
Coefficient	$C_0 = 18.741, C_1 = -121.31, C_2 = 273.5, C_3 = -194.05, C_4 = 56.31, C_5 = -5.8169$

angles division, respectively. In two-dimensional geometry, the symmetric geometry of the azimuthal angle reduces the total number of octants from eight to four, making the total number of control angle $4N_{\theta}N_{\phi}$. When solving the RTE in the unstructured meshed, the control angles may not align with the control volume face, causing the angular overhang problem where partially incoming and partially outgoing radiations. To solve this, the control angle should be subdivided more into $N_{\theta_p}N_{\phi_p}$ pixels, having smaller partitions in each overhang control angle.

$$\nabla \cdot (I(\vec{r}, \vec{s}) \vec{s}) = -a(T)I(\vec{r}, \vec{s}) + a(T)n^2 \frac{\sigma T^4}{\pi} \quad (8)$$

The radiation behavior on the surface is determined by the boundary conditions. The boundary conditions on the diffusion-gray opaque wall are expressed through Eq. (9), showing each interaction in Fig. 10. The

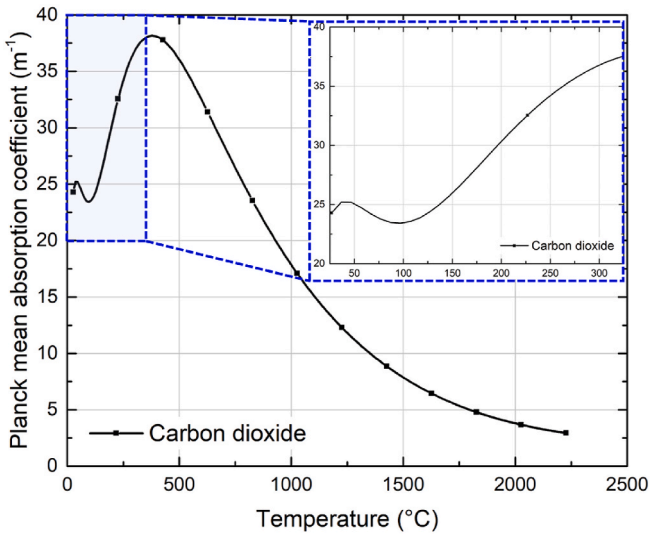
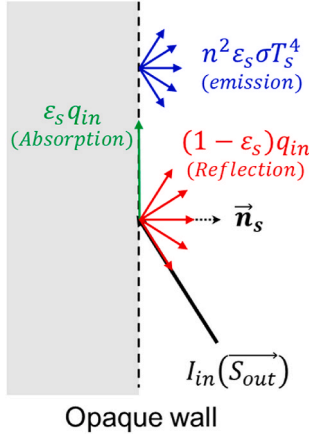


Fig. 9. Planck mean absorption coefficient of carbon dioxide as a function of temperature.

Table 6

Solver and model settings.

Turbulent Model	k-omega SST
Radiation Model (Gray rad.)	Discrete ordinates model
Density Model (For gases)	Expressed using polynomial functions
Pressure–Velocity Coupling	Coupled
Spatial Discretization	Least-squares cell based
	Pressure Body force weighted
	Momentum 2nd order upwind
	Energy 2nd order upwind
	DO intensity 2nd order upwind

**Fig. 10.** Boundary conditions on the opaque wall.

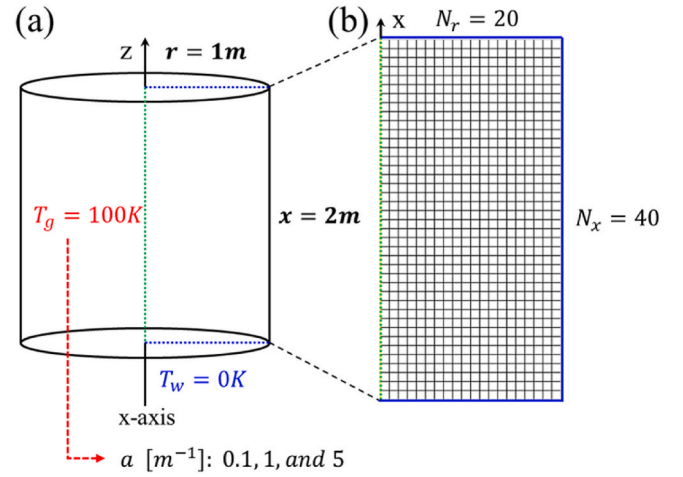
incident radiation (indicated by the black arrow) was partially absorbed according to the surface emissivity ($q_{in}\epsilon_s$), while the remaining was diffusely reflected from the wall ($\frac{(1-\epsilon_s)q_{in}}{\pi}$). Reflection and absorption were indicated by the red and green arrows, respectively. In addition, the emission of radiation was generated (blue arrows) from the wall based on its temperature ($\frac{n^2 \epsilon_s \sigma T_s^4}{\pi}$).

$$I_{s,out}(\vec{\Omega}) = \frac{n^2 \epsilon_s \sigma T_s^4}{\pi} - \frac{(1-\epsilon_s)}{\pi} \int I_{s,in} \vec{s}_{out} \cdot \vec{n}_s d\Omega', \text{ when } \vec{s}_{out} \cdot \vec{n}_s < 0 \quad (9)$$

where, $\vec{s}_{out}$ and $\vec{n}_s$ are the outgoing direction vector and normal vector, respectively.

As described above, the total solid angle is subdivided into the number of $4N_\theta N_\phi$ control angles. An increase in the control angle is essential for an accurate evaluation of radiative heat transfer, but the increase in computational cost should be considered. Therefore, the validation by the benchmark problem was conducted to determine appropriate discretization strategies of the DOM in FLUENT 21.2. The representative benchmark problem provided an exact solution of the normalized radiative heat fluxes ($q_{r,side}/\sigma T_{ref}^4$) on the side wall for a cylindrical enclosure under the constant absorption coefficients (0.1, 1, 5 m^{-1}) [56]. The boundary conditions and geometry for the benchmark problem are illustrated in Fig. 11(a). All enclosure surfaces were set black bodies ($\epsilon_s = 1$) and cold walls (0 K), and the cylinder was filled with the isothermal air of 100 K [57]. In addition, as shown in Fig. 11(b), the 2d axisymmetric cylinder geometry was discretized into the number of cell divisions in the radial ($= N_r$) and axial direction ($= N_x$), 20 and 40, respectively. The strategies of angular discretization and pixelation are summarized in Table 7. In the angle discretization conditions, pixelation was set equally to 10 by 10, while the angle discretization conditions were set equally to 10 by 10 in the pixelation conditions.

According to the approach of some studies [58], considering the discretization for azimuthal angles in 2d-axisymmetric geometry was not imperative in theory. However, as shown in Fig. 12, considerable under-rate heat fluxes were observed in Case C. Thus, Case C was

**Fig. 11.** (a) Cylinder geometry [56] and adopted boundary conditions [57] of the benchmark problem, and (b) a generated 2d axisymmetric mesh for benchmark simulation.**Table 7**

Angular discretization and pixelation conditions.

	N_θ	N_ϕ	N_{θ_p}	N_{ϕ_p}
Angular discretization				
Case A	5	5	10	10
Case B	10	10		
Case C	100	1		
Case D	20	20		
Pixelation				
Case B1	10	10	10	10
Case B2			10	1
Case B3			1	10
Case B4			1	1

unsuitable for this problem compared with Case B, even with the identical number of control angles. In addition, Case A was also judged to be inappropriate due to its error of over 1%. Furthermore, even though Case D demonstrated a minimal error among the cases, the increasing computational cost surpassed the improvement in accuracy. Consequently, Case B was selected as a suitable angular discretization among the strategies of this study.

The pixelation effect was also confirmed for the selected angular discretization. As depicted in Fig. 13, there was no pixelation effect because the global coordinate aligns with the control volume faces in the well-structured rectangular grid, resulting in the absence of the overhang problem. However, in this study, the global coordinate may be non-aligned with the control volume faces due to the unstructured meshes. Therefore, pixelation was set to 3×3 according to the recommendation of the FLUENT manual for the symmetry condition [41].

In the present study, to ensure sufficiently reliable CFD analysis, the convergence of the residual values was confirmed. The convergence criteria for the residuals were set at 10^{-4} for the continuity, turbulence kinetic energy (k), and specific dissipation rate (ω), and at 10^{-6} for the energy and discrete ordinate intensity.

After conducting the simulations under the established conditions, the results for the axial velocity, temperature, and density were extracted at a height of 22 m (Fig. 14; black line) and at a radius of 2.7 m (Fig. 14, red line). In the steady-state calculations, the heat transfer rate for the inner and outer walls of the MCV was equal. Therefore, to analyze the heat-transfer mechanisms within the gap while confirming the final heat loss, heat transfer along the inner wall of the MCV (Fig. 14; Blue line) was analyzed.

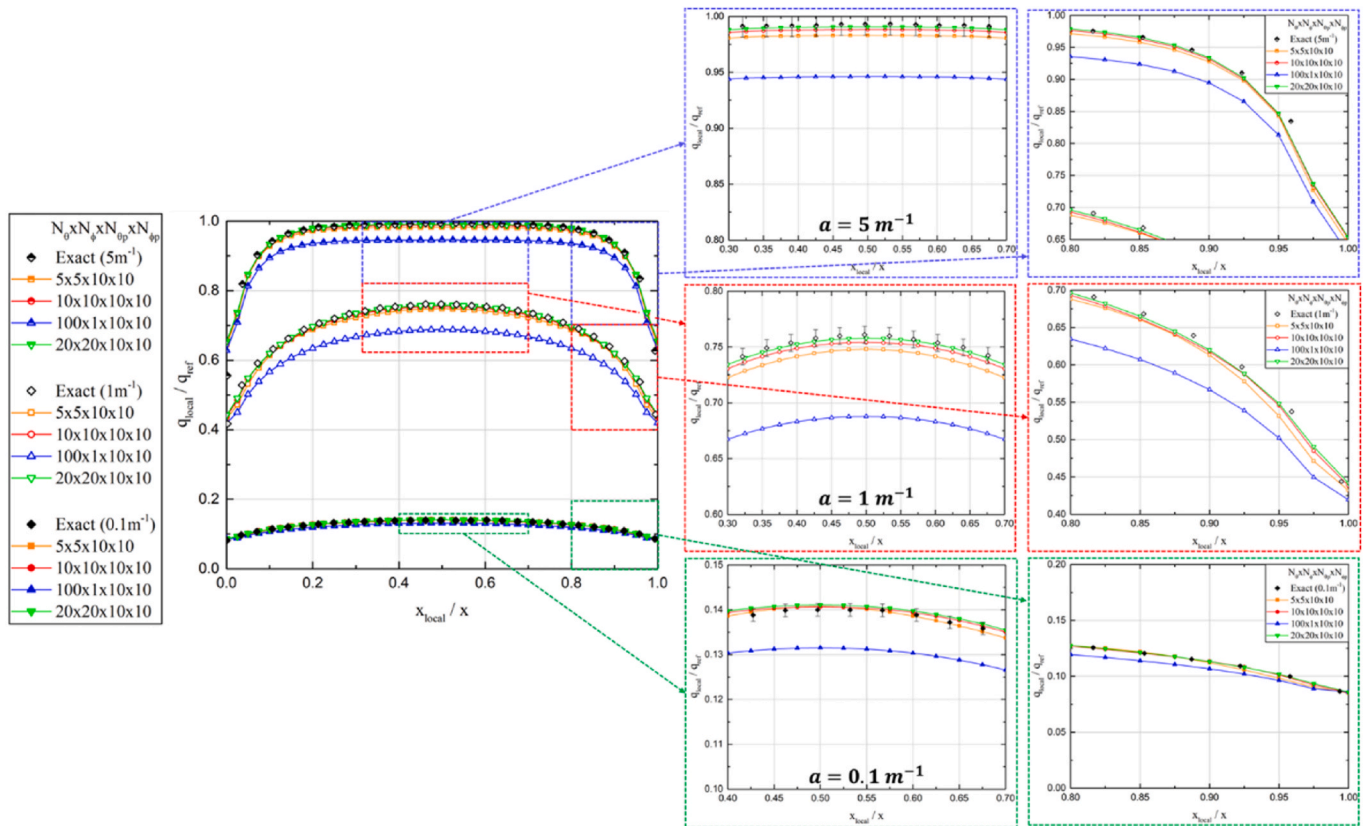


Fig. 12. Comparison of dimensionless heat flux on the sidewall between the strategies of angular discretization with different absorption coefficients: 5 m^{-1} (blue box), 1 m^{-1} (red box), and 0.1 m^{-1} (green box), respectively.

3. Results and discussion

3.1. Mesh sensitivity analysis

To ensure a sufficient mesh quality while minimizing the computational burden, five different mesh sizes were investigated. The meshes were generated using the ANSYS mesh tool. The mesh generation options and total element number are summarized in Table 8. The distributions of temperature and axial velocity at a height of 22 m, the total heat transfer rate on the MCV, and the contribution of radiative heat to total heat loss were compared between the meshes in a system with xenon-filled gap characterized by the highest mass flow rate and y^+ value (Fig. 15). The top three meshes having a larger element number exhibited sufficient mesh independence by producing a minor difference of less than 1%. Thus, for the CFD simulations, the medium-sized mesh with 416,995 elements was adopted (Fig. 16). This mesh design ensured the accurate evaluation of the flow and heat transfer rates near the wall, including the viscous sub-layer ($y^+ < 5$) where the large gradients of velocities were observed, by ensuring a high resolution indicated that the y^+ of the first layer near the wall remained below 2.0 with a xenon-filled gap.

3.2. Analysis of heat-transfer mechanisms within the gap

3.2.1. Effects of the radiation model on CFD analysis

The effects of the radiative heat transfer model on the CFD analysis were confirmed by comparing the results such as the total heat transfer rate and radial temperature distribution with and without the model. Fig. 17 shows the convective heat transfer rates for the candidate gases without the radiation model, while Fig. 18 illustrates the conjugated heat transfer rate including radiation and natural convection. The radiative heat transfer rate and convective heat transfer rate are denoted

by the red and black bars, respectively. At least 4.3 times increases in the total heat loss were observed when the radiative model was considered due to the two times higher radiative heat (86–97% of total heat) compared with convective heat while, in carbon dioxide, only three times increase in total heat loss was observed with a contribution of 48%. Despite the same boundary conditions, except for carbon dioxide (increase rate: 56%), the natural convective heat transfer rates tended to go half (reduction rate: 40–57%) when the radiation model was considered. Within the enclosure, the temperature difference between the vessels, a parameter of the Rayleigh number [59,60]. Therefore, the smaller temperature differences observed as the temperature of the RPV wall decreases along with the increase of the MCV wall temperature due to additional heat transferred by radiation have strong effects on the reduction of convective heat transfer. The insulation performance of the candidate gases changed from an order of (from best to worst) the vacuum, xenon, argon, nitrogen, air, and carbon dioxide in the absence of radiative heat to an order of carbon dioxide, vacuum, xenon, argon, air, and nitrogen in the presence of radiative heat.

Fig. 19 shows that there was also a considerable difference in the temperature distribution with and without the radiative heat transfer model. In both cases, the overall bulk temperature within the gap was intermediate between the RPV and MCV temperatures. However, with radiative heat transfer included, more heat was transferred to the MCV, leading to a notable increase in the bulk temperature, indicating that the fundamental reason for the greater heat loss was radiation. Considering that such a high bulk temperature can cause material integrity problems with thermal fatigue such as the blue brittleness in metal components in accident situations, it suggests that heat loss should be effectively reduced for the safety of RPV. In addition, except for carbon dioxide, the smaller discrepancy in the bulk temperature between the candidates when radiative heat was considered led to similar heat losses, while distinct differences were observed in the absence of radiation, i.e., when

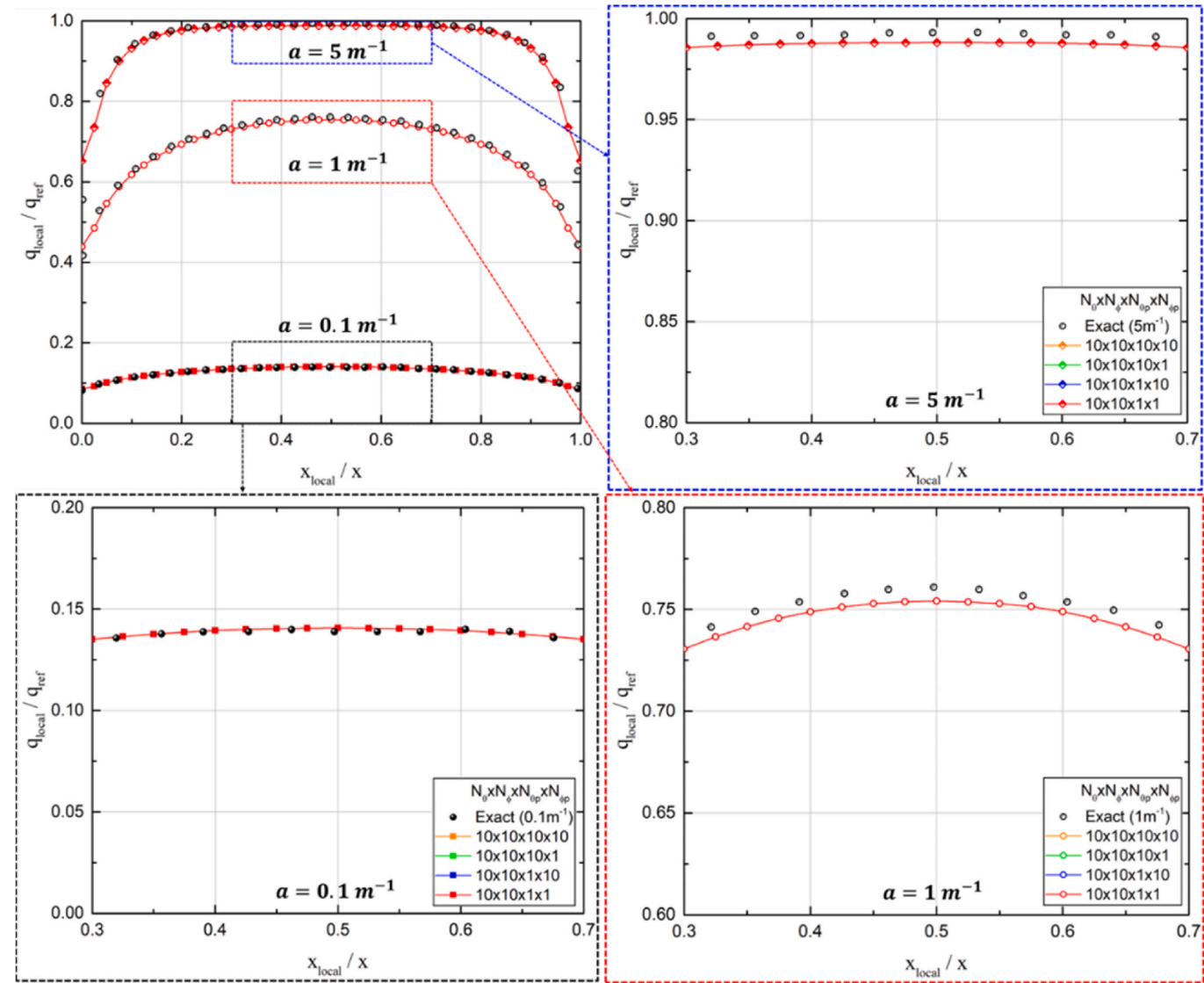


Fig. 13. Comparison of the dimensionless heat flux on the sidewall between the strategies of pixelation with different absorption coefficients: 5 m^{-1} (blue box), 1 m^{-1} (red box), and 0.1 m^{-1} (black box), respectively.

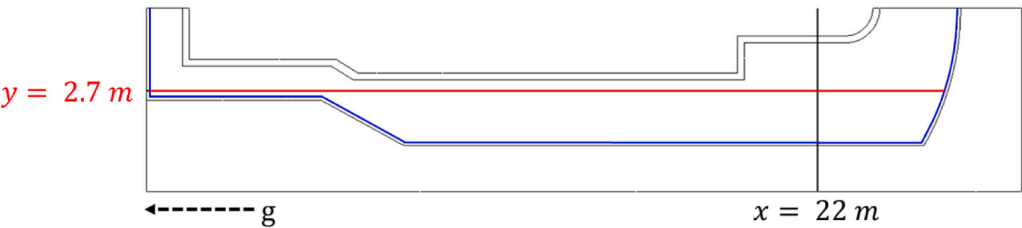


Fig. 14. Positions of the extraction line for the results.

Table 8
Mesh generation conditions.

Default face size (m)	Gap face size (m)	Inflation	Element number
0.03	0.05	First layer thickness:	152,910
	0.03	0.0003 m	240,972
	0.02	The number of layers: 10	416,995
	0.015		631,255
	0.01		1,288,072

heat was only transferred through natural convection. However, carbon dioxide was unique in that the lowest heat loss was observed even when the bulk temperature increased. In addition, as depicted in Fig. 20, in the case of carbon dioxide, dissimilar to the tendency in transparent media, a notable axial temperature gradient was observed even when considering the radiation model. It was attributed to a high absorption coefficient, reaching 37 m^{-1} , acting as a shield of thermal radiation, causing insufficient heating of the MCV. Collectively, these results suggest that thermal radiation plays a dominant role in the heat-loss mechanisms within the reactor module and significantly affects the temperature

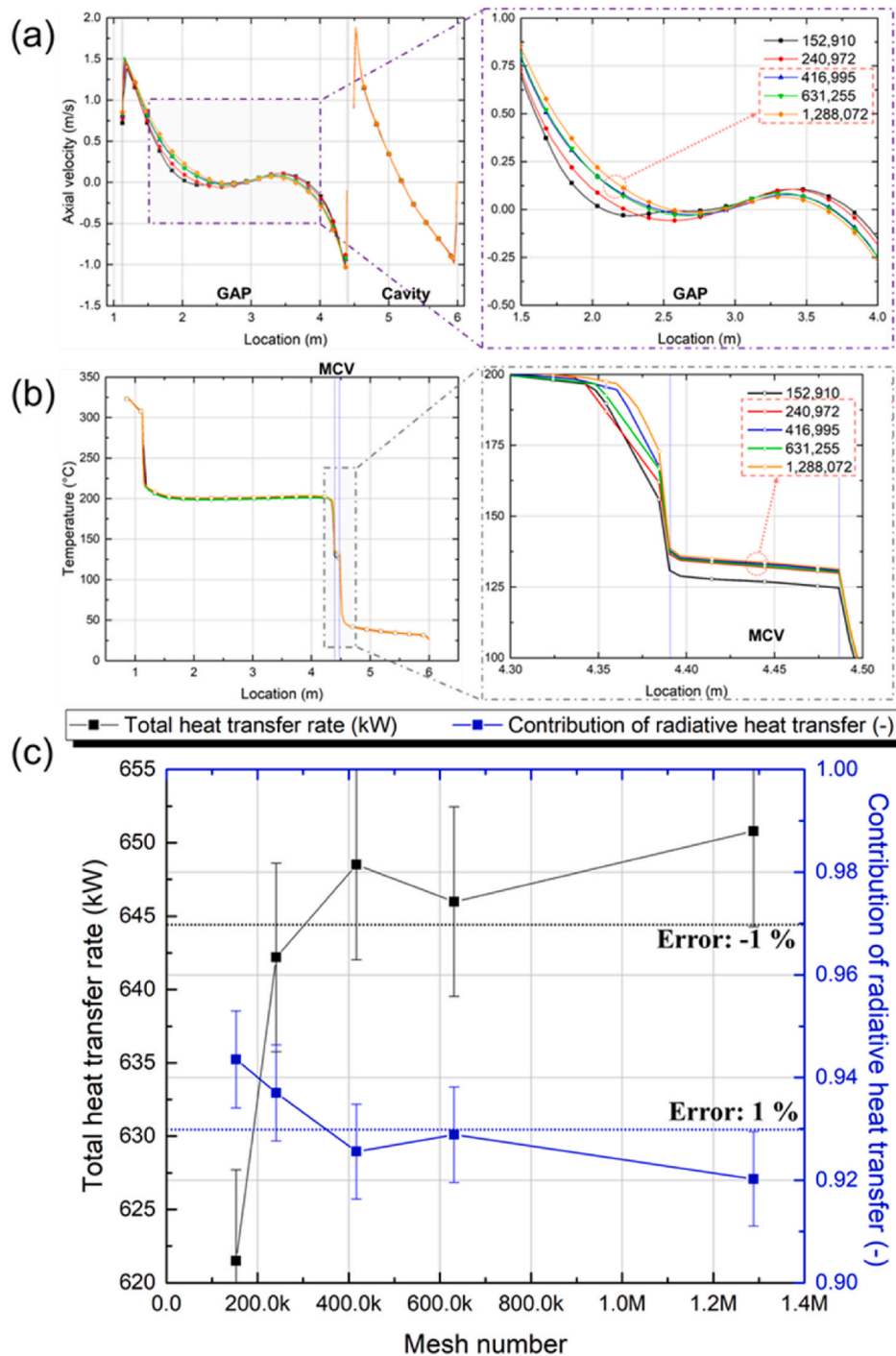


Fig. 15. Mesh sensitivity with a xenon-filled gap: (a) axial velocity and (b) temperature distribution in the gap and cavity at a height of 22 m and (c) total heat transfer rate and the contribution of radiative heat transfer to total heat transfer with 1% error bars.

distribution. Therefore, it is essential to consider conjugated heat transfer in order to more precisely evaluate heat loss patterns.

3.2.2. Convective heat transfer within the metal containment vessel

The observed overall flow patterns were consistent with those described in Section 2.2. Additionally, a weak flow in the vicinity of the RCP region due to restrictions on the gas inflow caused by its vertical shape was also observed for all candidate gases. Fig. 21 displays the streamlines of axial velocity in fluid regions. Despite the similarity in the flow pattern near the wall, a distinct difference was observed in the vacuum and carbon dioxide cases where the relatively strong convection

flow velocity. In these cases, the flow behavior in the bulk region of the gap consisted of a single large flow field, while the flows were separated into two flow fields in the rest of the cases.

In addition to the differences in flow behavior, as illustrated in Fig. 18, the convective heat transfer differed depending on the thermal properties of the candidate gases. To verify the calculated convective heat transfer, the heat transfer coefficients (HTCs) of the candidate gases were compared based on their thermal properties. To express the HTC (h) as a function of thermal properties, this study utilized a well-known correlation (Eq. (10)) [34]. In the same geometry, different from the Nusselt number, the HTC reaches a higher value when the thermal

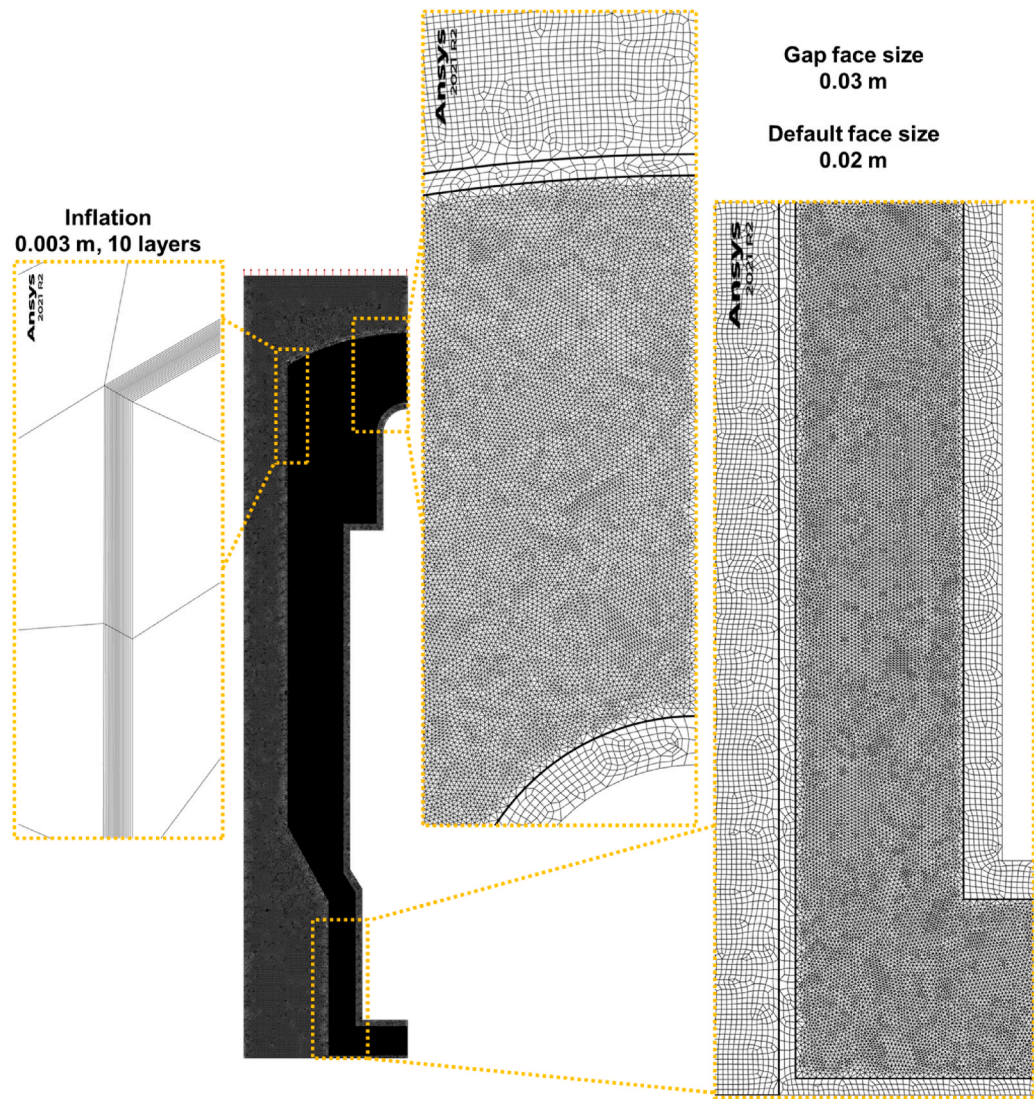


Fig. 16. Selected mesh based on the mesh sensitivity analysis (416,995 elements).

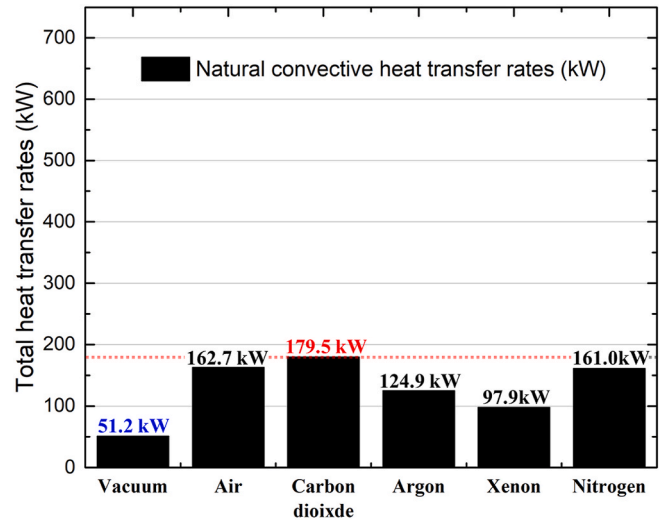


Fig. 17. Comparison of the convective heat transfer rate on the inner wall of the MCV without the thermal radiation model in CFD analysis for the candidate gases.

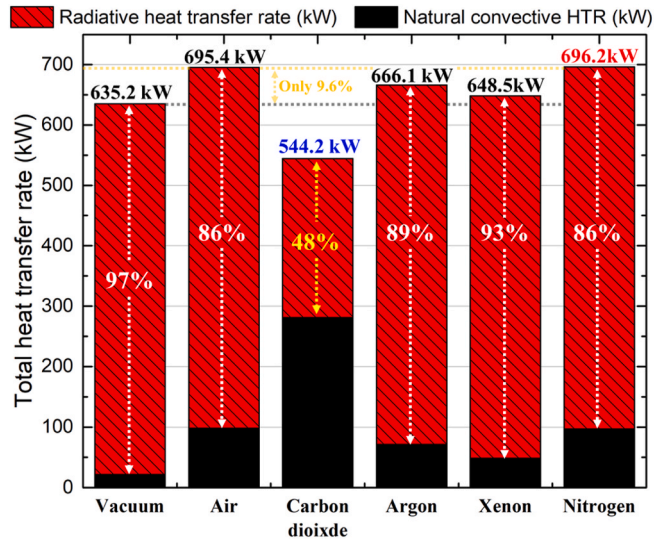


Fig. 18. Comparison of the conjugated heat transfer rate on the inner wall of the MCV ($\epsilon = 0.4$) for the candidate gases.

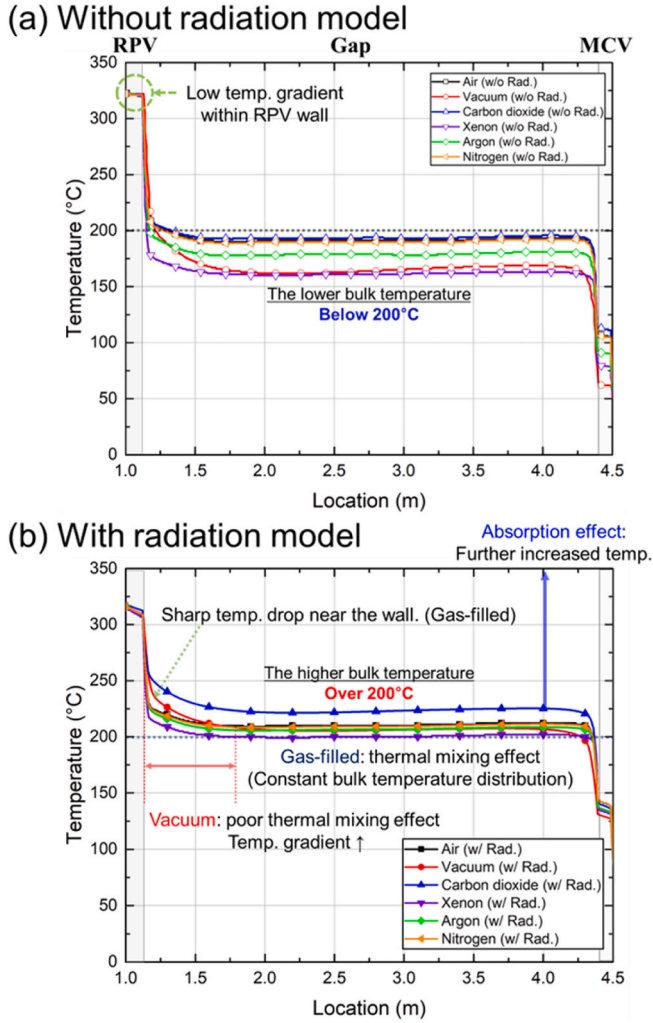


Fig. 19. Comparison of the temperature distribution in the (a) presence and (b) absence of the radiation model.

conductivity is higher. In addition, an extremely low density plays a crucial role in reducing the convective performance. Using Eq. (11), the relative HTC, defined as the ratio of the convective heat transfer rate of the candidate gas to that of air, was evaluated. For most candidates, except for carbon dioxide, properties exhibited a similar trend with air, resulting in slightly decreased slopes for the relative HTCs. However, as summarized in Table 4, the slopes of thermal conductivity and specific heat capacity of carbon dioxide are as high as 5 and 1.3 times than those of air, respectively. These higher slopes caused a significant enhancement in the relative HTC of carbon dioxide. Consequently, within the temperature range of interest, the order of convective heat transfer was consistent with the order of the relative HTC (Fig. 22). In addition, the calculated convective heat transfer rates corresponded well to the relative HTC values, except for carbon dioxide. Because the relative HTC did not consider radiation parameters, the performance of convective heat especially for carbon dioxide did not match exactly. Nevertheless, the consistency of order and specific values between the calculated natural convective heat transfer rates and the relative HTC implied that the simulation results for convective heat transfer were reasonable.

$$h_{nc} = Nu_{nc} \frac{k}{L_c} \propto Ra^{\frac{1}{3}} \left(\frac{k}{L_c} \right) \propto \left(\frac{\frac{1}{c_p \rho^{\frac{2}{3}} k^{\frac{2}{3}}}}{\mu^{\frac{1}{3}}} \right) \quad (10)$$

$$h_{rel} = \frac{h_{cand}}{h_{air}} \quad (11)$$

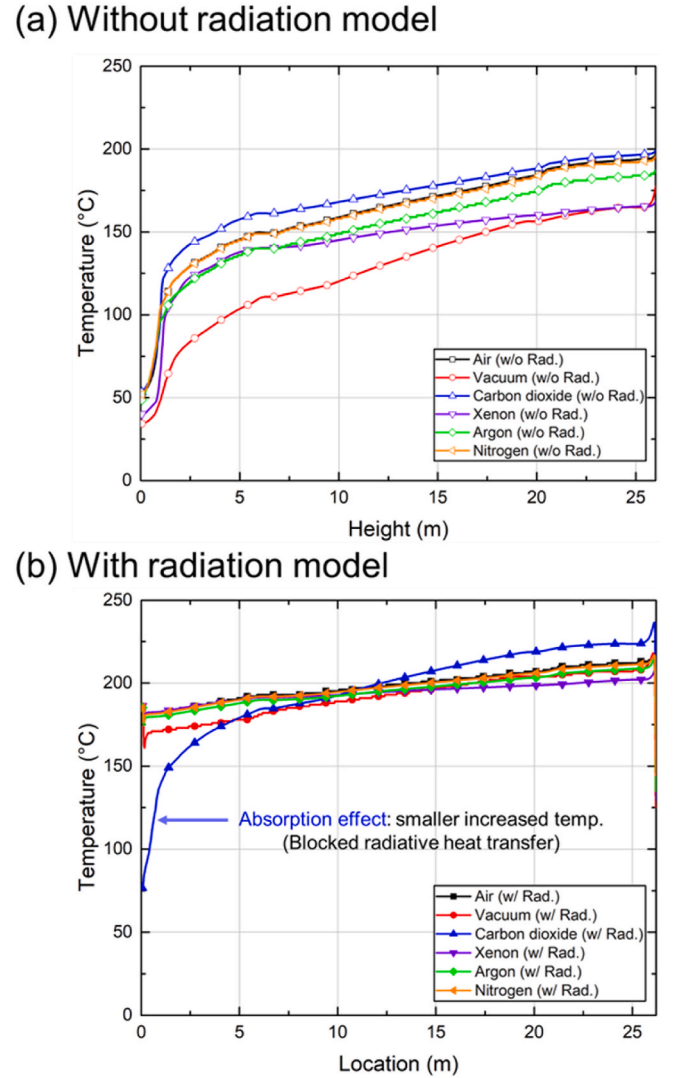


Fig. 20. Comparison of the axial temperature distribution ($y = 2.7$ m) in the (a) absence and (b) presence of the radiation model.

The possible reasons for the difference in the convective heat transfer performance were also investigated. Fig. 23 present the temperature distribution in the gap for the candidate gases. In all cases, high-temperature gases were present in the top region, and relatively low-temperature gases were located at the bottom of the gap due to natural convection. Within the vacuum, the gap was thermally stratified near the upper wall of the RPV, which was ascribed to the inferior natural convective heat transfer of a vacuum. In the vacuum, the convective heat loss (21.8 kW) was only approximately half of that for xenon, which was the most effective convection insulator among the candidate gases. Except for its density, the other thermal properties affect to the HTC are very similar to air, but the convective heat transfer rate was much lower than air. Therefore, little convective heat transfer appeared to be mainly due to the low mass flow rate, resulting from the extremely low density of the vacuum.

Fig. 24 illustrates the axial velocity and effective thermal conductivity. From the convective heat transfer equation in the k-epsilon model (Eq. (12)), we tried to select a variable that can represent convective heat transfer performance. E , k_{eff} , τ_{ij} , and S_h are the total energy, the effective thermal conductivity, the deviatoric stress tensor, and the volumetric heat sources, respectively. Among the variables, the effective thermal conductivity, defined in Eq. (2), includes turbulent viscosity (Eq. (3)), μ_t , a variable that sufficiently considers the density and

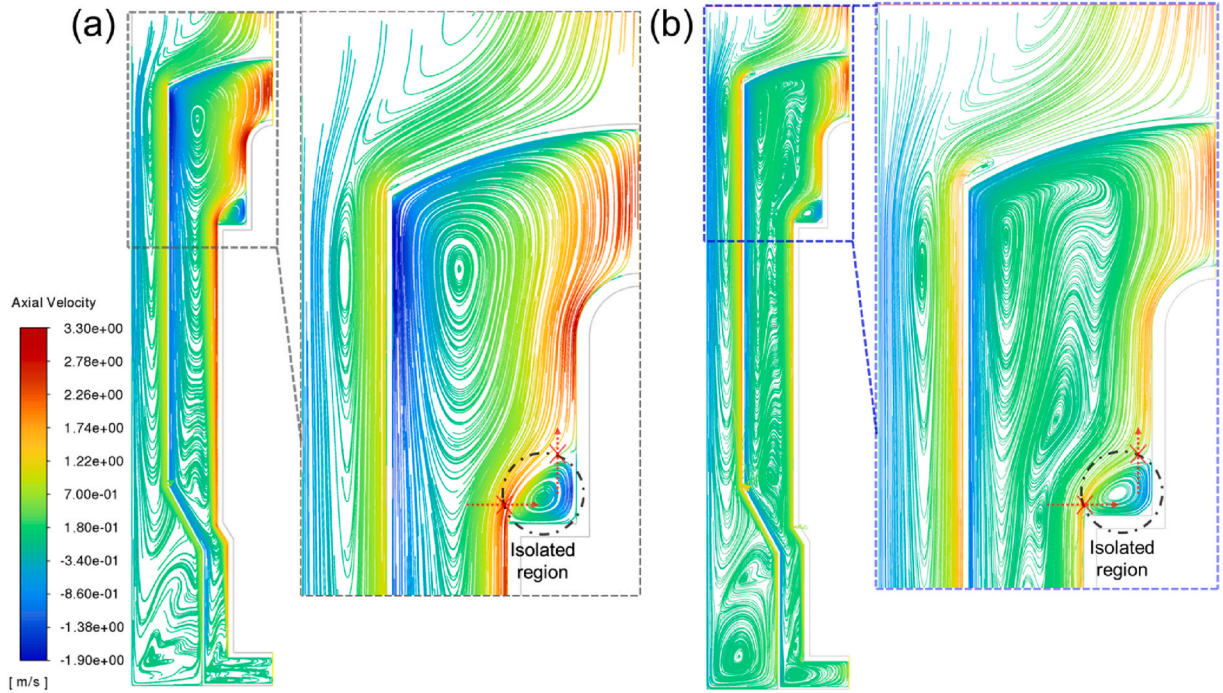


Fig. 21. Streamline of the axial velocity within the fluid region when (a) the flow velocity is relatively strong (vacuum and carbon dioxide) (b) or not (the rest cases).

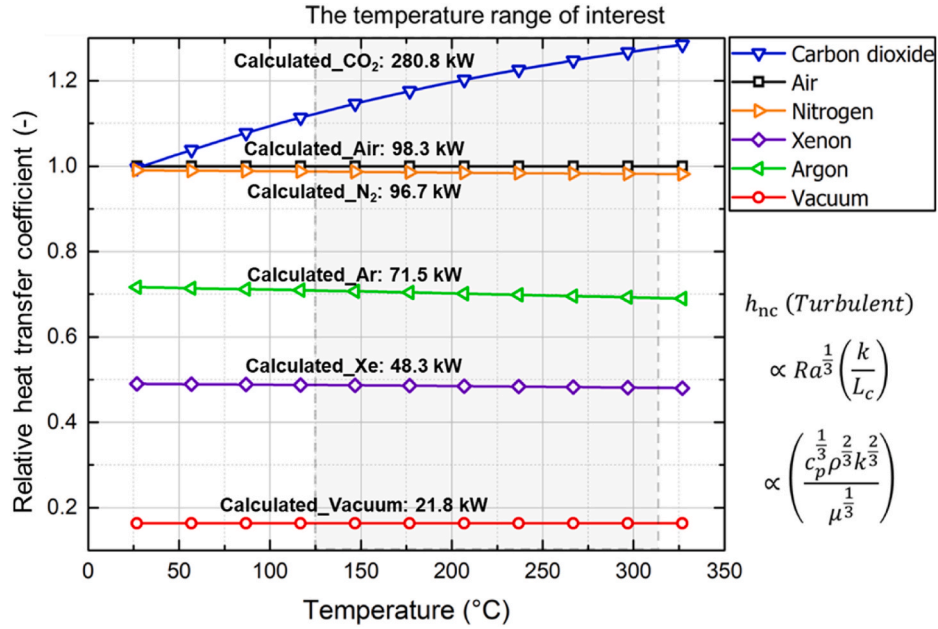


Fig. 22. Theoretical calculations of the relative heat transfer coefficients of the candidate filling gases (lines with various colors) and calculated convective heat transfer rates (bold letters).

turbulence effect. In addition, since the effective thermal conductivity also includes specific heat capacity and thermal conductivity of medium, we selected it as a representative parameter for the natural convective heat transfer performance and compared it. As shown by Fig. 24(a) and (b), although the axial velocity was the highest under a vacuum, the effective thermal conductivity was the lowest due to the extremely low density. Consequently, due to the lack of a mixing effect due to the absence of a sufficient medium, the radial temperature distribution continuously decreased instead of experiencing a sharp drop near the wall and became more stratified.

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot [\bar{\mathbf{v}}(\rho E + p)] = \nabla \cdot \left[\mathbf{k}_{\text{eff}} \frac{\partial T}{\partial x_j} + (\bar{\mathbf{v}} \cdot \boldsymbol{\tau}_{ij}) \right] + S_h \quad (12)$$

$$k_{\text{eff}} = k + \frac{c_p \mu_t}{Pr_t} \quad (13)$$

$$\mu_t = \frac{\rho k_t}{\omega} \frac{1}{\max \left[\frac{1}{a^*}, \frac{SF_2}{a_1 \omega} \right]} \quad (14)$$

where, S and F_2 are the strain rate magnitude and blending function,

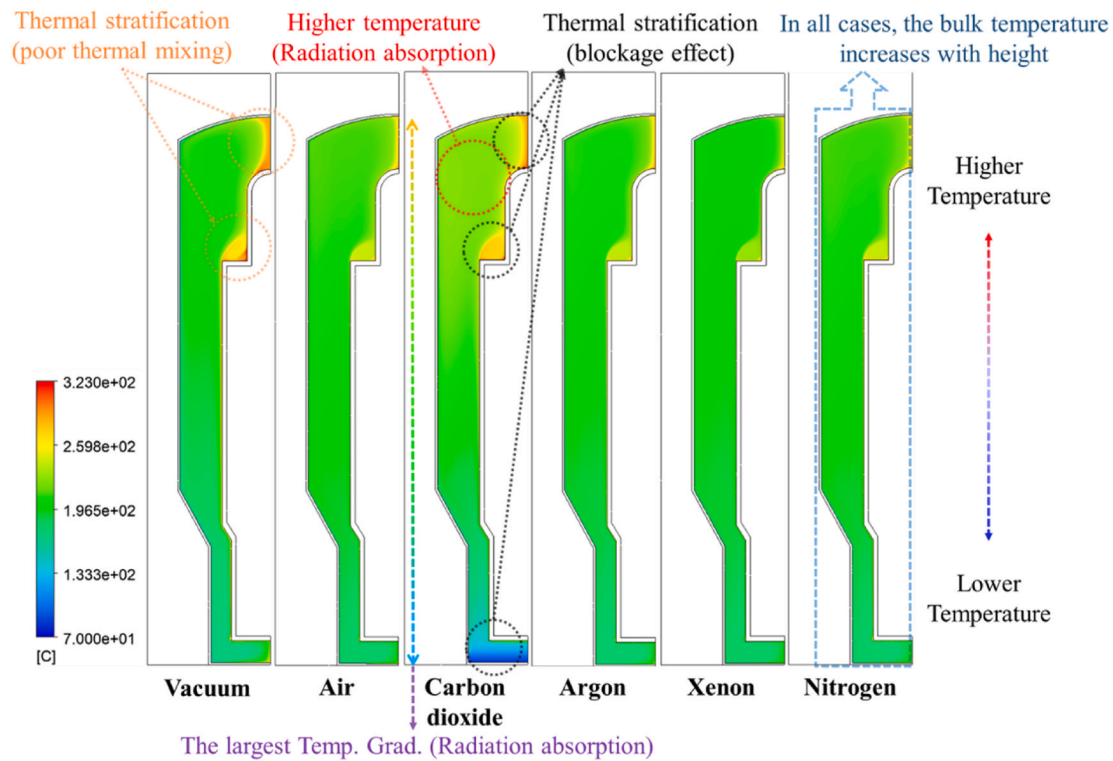


Fig. 23. Temperature distribution within the gap for different filling conditions.

respectively.

In addition, model constants a^* and a are 1 and 0.31, respectively.

Relatively large effective thermal conductivity than argon was observed in xenon due to its approximately three to four times higher density and similar axial velocity compared with the other candidates, excluding from vacuum (Fig. 24(b)). However, the thermal conductivity and specific heat capacity were only about one-fourth and one-sixth of the other candidate gases respectively. The very low thermal conductivity suppressed the transfer of heat from the RPV to the xenon, resulting in the sharpest temperature drop near the RPV wall (Fig. 19 (b)). Because the specific heat capacity is directly related to enthalpy (i.e., the internal energy at a constant volume), the temperature was maintained at its lowest level due to the lowest internal energy. Fig. 25 presents the distribution of the static enthalpy within the gap, which was significantly lower when xenon was used. Consequently, xenon exhibited the lowest convective heat transfer (48.2 kW) due to its low thermal conductivity and because it had the lowest internal energy of the gas-filled candidates due to its low specific heat capacity.

Because approximately 70% of air is composed of nitrogen, the total and convective heat transfer rate (air: 98.3 kW; nitrogen: 96.7 kW), axial velocity, and bulk temperature in the gap were very similar between nitrogen and air, though air was slightly higher. In addition, carbon dioxide, which had the highest bulk temperature, produced the highest convective heat transfer (280.8 kW), representing 2.9 times of air. However, considering the relative HTC (Fig. 22) between the candidates, the amount of difference in the natural convection heat transfer rate between carbon dioxide and air was too large. In this study, this excessive convective heat transfer rate of carbon dioxide was determined to be the result of an increase in the internal energy and the effective thermal conductivity due to the effect of the absorption coefficient [31]. Argon exhibited an intermediate bulk temperature and convective heat transfer (71.5 kW) due to its intermediate thermal properties, especially the thermal conductivity and specific heat capacity.

3.2.3. Radiative heat transfer between the metal containment vessel and the reactor pressure vessel

As shown in Eq. (3), except for the participating medium, in an equilibrium state with the same emissivity and area, the amount of radiative heat transfer is solely determined by the temperatures of the RPV and MCV, which themselves were influenced by the continuous interaction between natural convection and radiation. In all non-absorbing candidates except argon, superior convective heat transfer led to a higher surface temperature for the MCV, resulting in a relatively small temperature difference between it and the RPV. Thus, the order of the candidates in terms of performance of radiative heat transfer was the inverse of that of convective heat transfer. Due to the complementary effect of these mechanisms, there was only a slight difference of 9.6% in the total heat loss between the vacuum and non-absorbing medium, i.e., nitrogen (Fig. 18). In carbon dioxide, characterized by the absorption coefficient, even 14% lower total heat loss than in vacuum was observed, showing the best insulation performance. The observation of a comparable level of total heat loss when a vacuum was used represented a noteworthy finding. Consequently, despite the conventional assumption that a vacuum has excellent insulation advantages, these findings suggest that gas, i.e., carbon dioxide, can be used as a replacement for a vacuum.

Table 9 presents the surface temperature of the RPV and MCV at a height of 22 m under a steady state. A similarly high RPV temperature of over 306 °C among the gas candidates made it possible to predict a significant contribution of radiative heat transfer within the gap. In addition, the low temperature of the MCV strengthened the radiative heat transfer. As a result, radiative heat transfer led to larger heat loss in the order of the vacuum (613.39 kW, contribution of 97%), xenon (600.24 kW, 93%), nitrogen (599.70 kW, 86%), air (597.10 kW, 86%), argon (594.6 kW, 89%) and carbon dioxide (263.4 kW, 48%). In the case of argon, it is confirmed that the amount of radiative heat transfer rates has a small value that deviated from the tendency (from 599 kW to 600 kW). However, the contribution of radiative heat transfer rate to the total heat loss, which means the reduction potential in Section 3.3, was 89%, showing a consistent tendency. Overall, more than 48% of the total

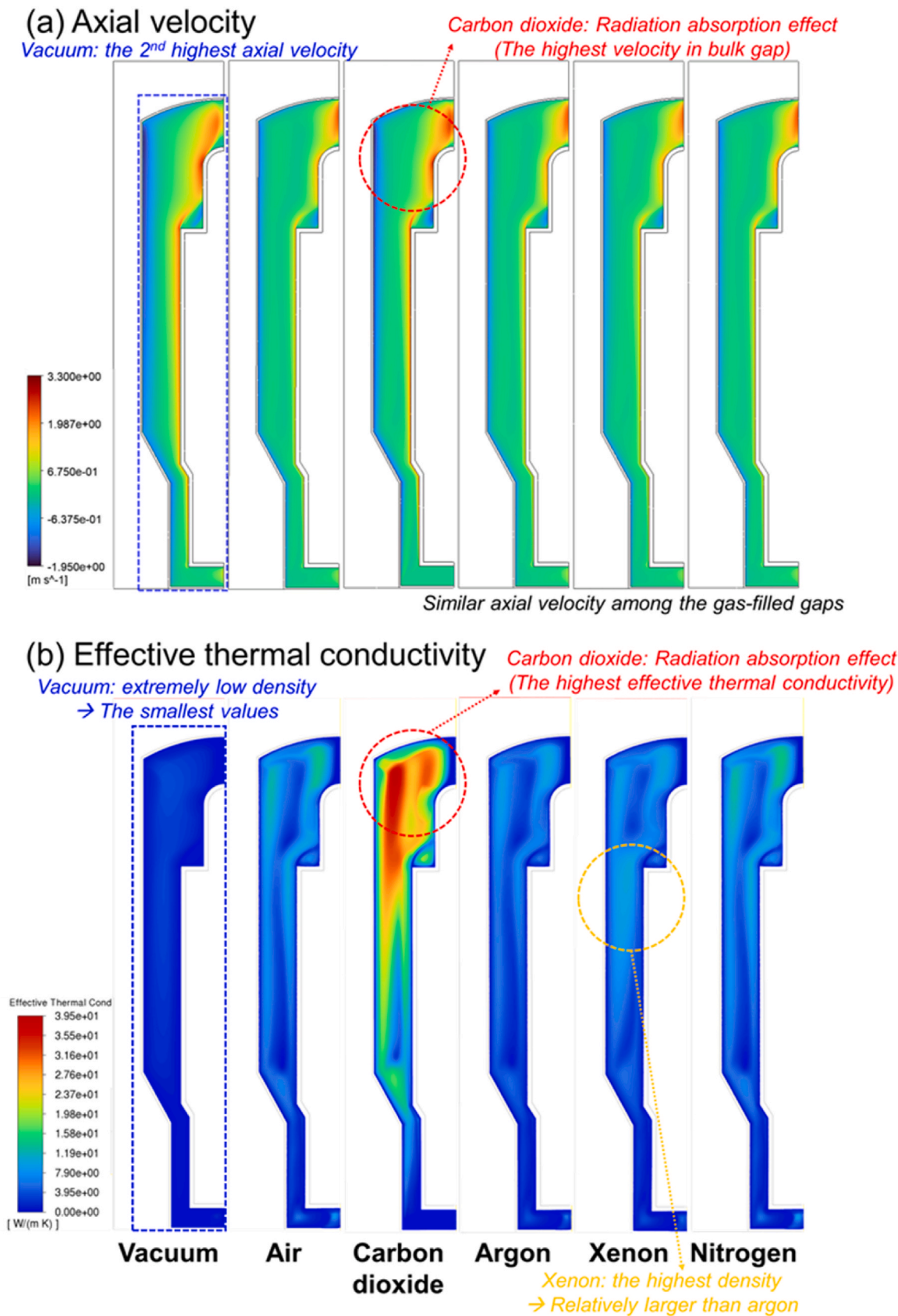


Fig. 24. (a) Contours of the axial velocity distribution across the gap and module cavity and (b) contours of the effective thermal conductivity distribution within the gap.

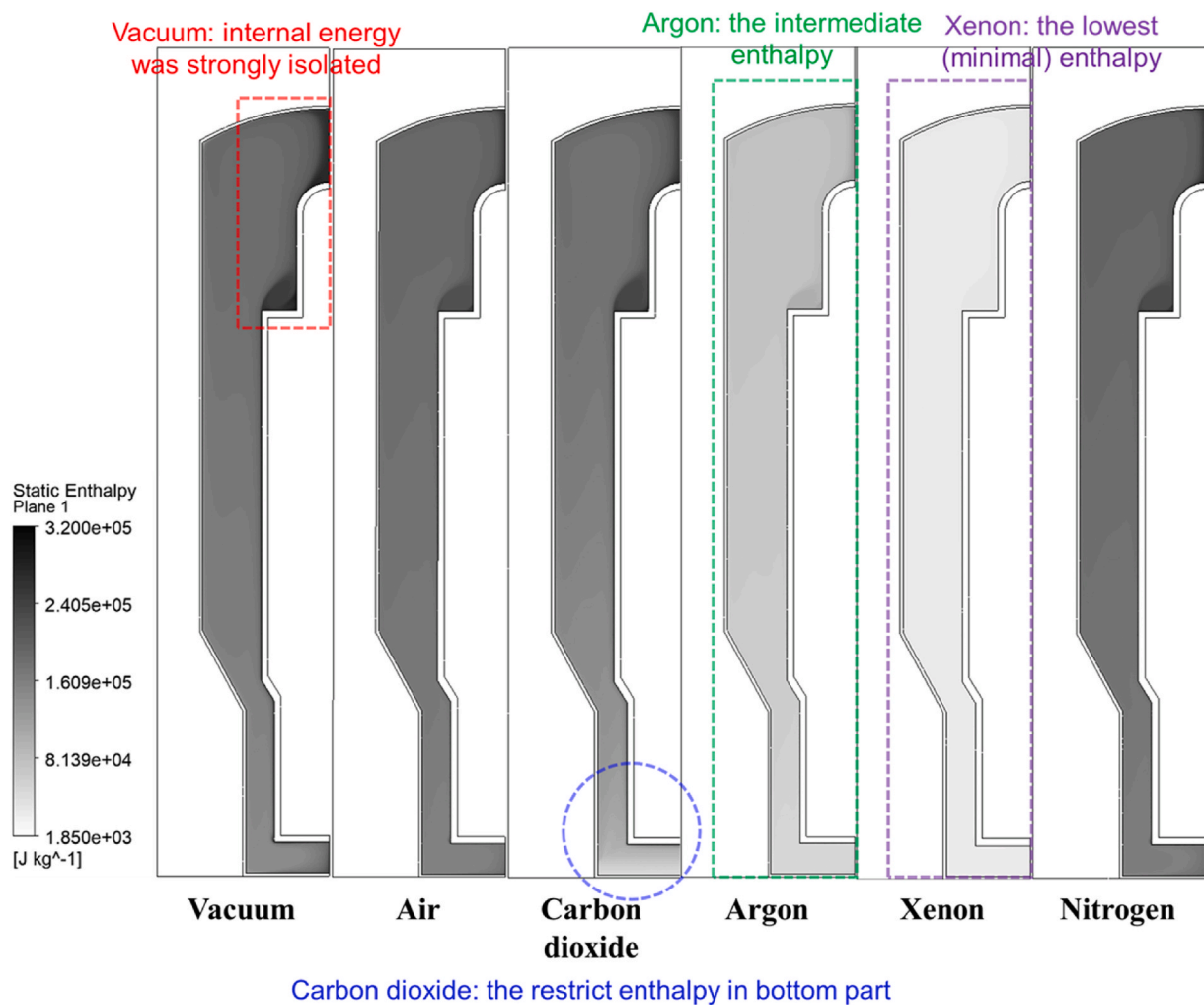


Fig. 25. Contours of the static enthalpy within the gap for the various candidate gases.

Table 9
Surface temperature (°C) of the vessels and the radiative and total heat transfer (kW) for the various gas candidates.

	Air	Vacuum	Carbon dioxide	Xenon	Argon	Nitrogen
$T_{surf,RPV}(^{\circ}C)$	308.0	309.1	312.4	306.2	308.5	308.5
$T_{surf,MCV}(^{\circ}C)$	145.0	131.4	143.9	137.9	138.4	144.5
$Q_{rad}(kW)$	597.1	613.4	263.4	600.2	594.6	599.5
$Q_{total}(kW)$	695.4	635.2	544.2	648.5	666.1	696.2

heat loss was due to radiation across all candidates, highlighting the dominant role of radiative heat transfer. These results strongly suggest that reducing the radiative heat transfer is a promising approach for enhancing insulation performance. To confirm this, the present study investigated the effectiveness of TRS using a layer of aluminum foil on the inner wall of the MCV (see Section 3.3).

Due to its high absorption coefficient, the contribution of radiation was significantly lower when carbon dioxide was used as the filling gas than with the other gas candidates, even though the RPV temperature was the highest. The carbon dioxide absorbed a portion of the thermal radiation emitted by the outer wall of the RPV and the inner wall of the MCV. The remaining thermal radiation reaching the wall on the other side was either reflected back or absorbed and re-emitted. In this process, the high absorption coefficient of carbon dioxide strongly limits heat transfer by radiation as it absorbs a large portion of incident

radiation with only a short path length [31], as shown in Fig. 26. Fig. 26 (a) and (b) compare the radiation-absorbing medium with a representative transparent medium, i.e., vacuum, for the incident radiation heat flux. In addition, the distribution of the absorption coefficient in carbon dioxide is illustrated, as shown in Fig. 26(c). The incident radiative heat flux in the carbon dioxide exhibited significant dependence on the absorption coefficient, showing further sharp decreases near the MCV walls, dissimilar to the vacuum. In particular, the contour lines of the radiative heat flux near the RPV wall were seen in a vertical shape rather than a lateral direction observed in a vacuum and seemed to be a convective flow. These results suggest that most of the radiative heat flux was absorbed as soon as it occurred on the MCV wall and then released from the gas. Overall, thermal radiation reaching RPV was judged to be radiative heat emitted from carbon dioxide at relatively low temperatures near RPV rather than the surface of MCV at higher temperatures, so much less radiant heat was transferred throughout the inner wall of MCV (Fig. 26(b)). This radiative behavior by the absorption coefficient is called a blocking effect in the present study. Consequently, due to the blocking effect, carbon dioxide was exhibited as the best insulator despite the highest natural convection heat transfer performance.

3.3. Effect of thermal radiation shielding on the heat-transfer mechanisms

A layer of aluminum foil with an emissivity of 0.04 was employed as TRS on the inner wall of the MCV to reduce the radiative heat transfer.

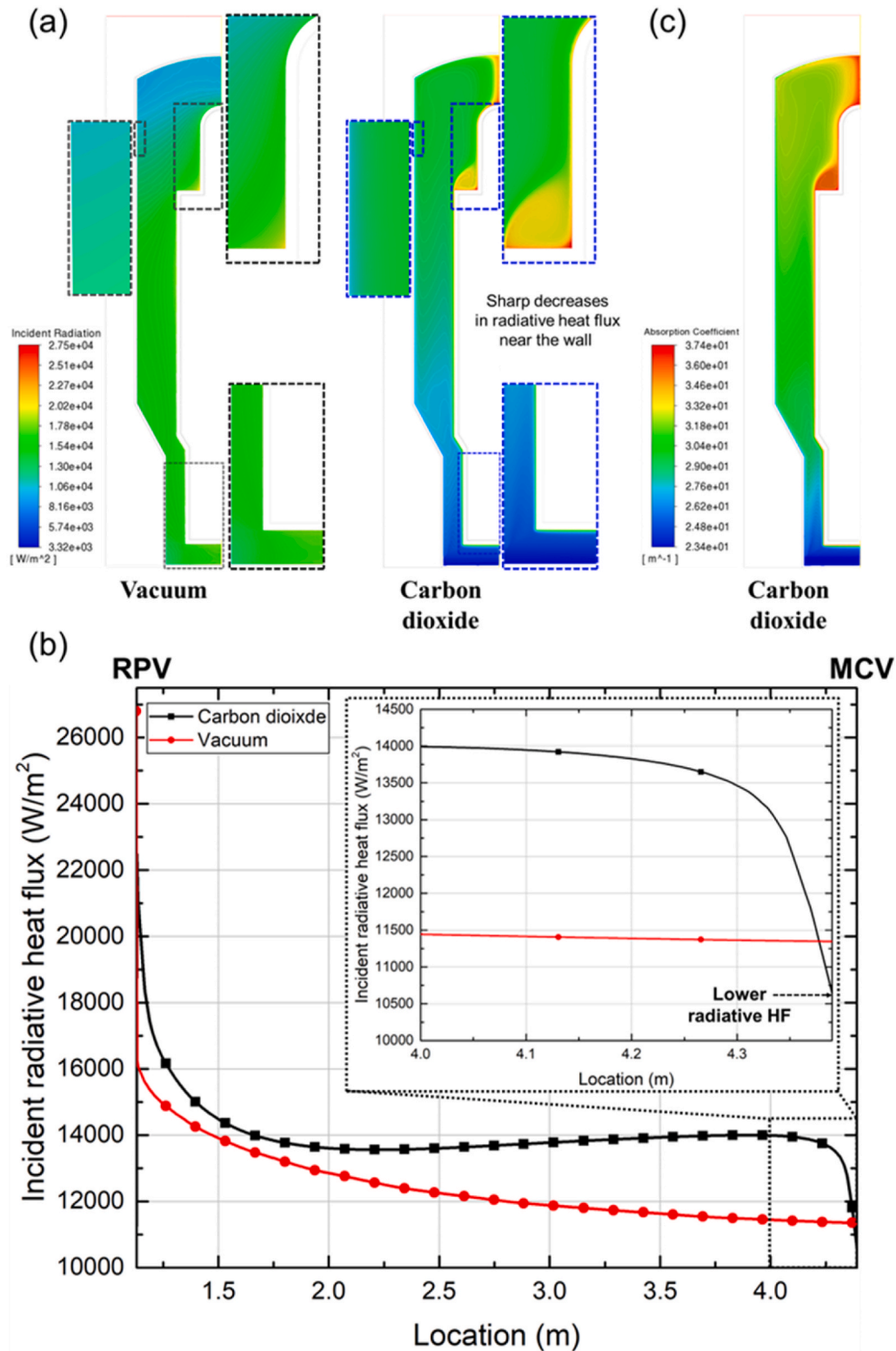


Fig. 26. (a) Comparison of radiative heat flux on the gap between (Left) the vacuum condition and (Right) carbon dioxide, (b) radial distribution of that at the height of 22 m, and (c) the calculated absorption coefficient distribution of the carbon dioxide-fill.

The effect of this TRS was assessed by comparing the total heat transfer both without ($\epsilon_s = 0.4$) and with the TRS ($\epsilon_s = 0.04$) (Fig. 27).

The use of TRS reduced the heat loss by 70% for the vacuum, 64% for xenon, 63% for argon, 56% for air, 56% for nitrogen, and 13% for carbon dioxide. This order was primarily determined by the reduction potential, which was indicated by the contribution of radiative heat

transfer in the absence of TRS. However, the specific reduction rates were determined by the interaction between natural convection and radiation.

Because the emissivity was reduced from 0.4 to 0.04, only one-tenth of the radiation was absorbed compared with the previous case, while the remaining portion was reflected back, despite the radiation reaching

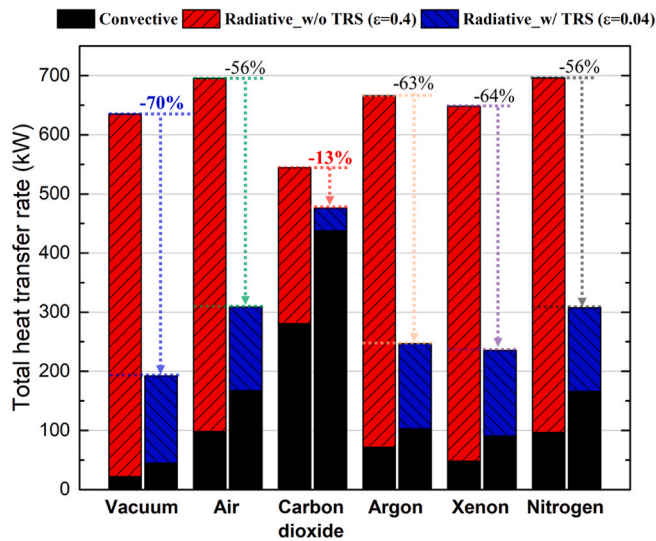


Fig. 27. Comparison of the heat loss with and without thermal radiation shielding (TRS).

the MCV remaining the same. Consequently, the MCV was not sufficiently heated, resulting in a fall of the average bulk temperature to the intermediate level between that of with radiation and without radiation, except when carbon dioxide was used (Fig. 28). In the case of carbon dioxide, the bulk temperature increased rather than having a median value because it more actively absorbed not only the radiation emitted from RPV but also the radiation reflected more on the inner wall of MCV. The increased temperature when using the TRS may impose further thermal fatigue on the metal structures than without TRS. It underscored that the design direction used carbon dioxide may not be suitable for achieving the desired structure integrity and insulation performance.

In terms of the radiative heat transfer, a significant reduction was observed when using the TRS. In most cases, the radiation heat transfer decreased by about 76%, except for carbon dioxide, for which the reduction was even higher (86%). The difference in the reduction rate with carbon dioxide was possibly due to the radiation absorption effect. Although the reduction rate was higher than the other gas candidates, the actual amount of radiative heat reduced was the lowest (225.3 kW) because carbon dioxide had the lowest reduction potential. On the other

hand, the largest reduction in the radiative heat transfer occurred when a vacuum was used (466.0 kW) because it had the highest reduction potential.

The extent of the increase in the convective heat transfer when TRS was applied differed depending on the filling gas used. The convective heat transfer increased by 157.2 kW for carbon dioxide, 69.3 kW for nitrogen, 69.2 kW for air, 42.6 kW for xenon, 31.6 kW for argon, and 23.0 kW for the vacuum. This large increase for carbon dioxide can be attributed to the radiation absorption effect. Figs. 28 and 29 show that, when applying TRS, the temperature within the gap only increased for carbon dioxide, while it decreased for the other candidates. When the TRS was applied to the inner wall of the MCV, more radiation was reflected, which led to increased radiation absorption by carbon dioxide. As a result, an increase in convective heat transfer was observed based on the increased enthalpy (Fig. 30). Along with the further-higher temperature distribution, carbon dioxide is thus not recommended as a gap-filling gas because of the possibility that the effect of TRS may be limited.

As presented in Table 1, some SMR designs fill the gap between the RPV and MCV with a gas such as air or nitrogen. Therefore, demonstrating that higher insulation performance was achieved compared to a vacuum when TRS was employed highlights the importance of TRS in these designs. To verify this, we compared the results for the use of a low internal pressure with those for TRS. The gas-filled gaps with TRS demonstrated superior insulation performance compared to the low-pressure vacuum case without TRS, even when using air and nitrogen. In these cases, air and nitrogen exhibited low heat losses of up to 48.6% and 48.4%, respectively, in comparison to the vacuum. This suggests that suppressing the radiative heat transfer using a low-emissivity surface is more effective for improving the insulation performance compared with suppressing natural convection using a low internal pressure.

4. Summary and conclusion

This study aimed to analyze the heat-transfer mechanisms within an MCV to reduce heat loss by evaluating the effect of various parameters, including thermal properties, radiation absorption, and surface emissivity. This analysis was conducted using CFD simulations, with the i-SMR used as the reference reactor. Based on the results, the following conclusions could be drawn:

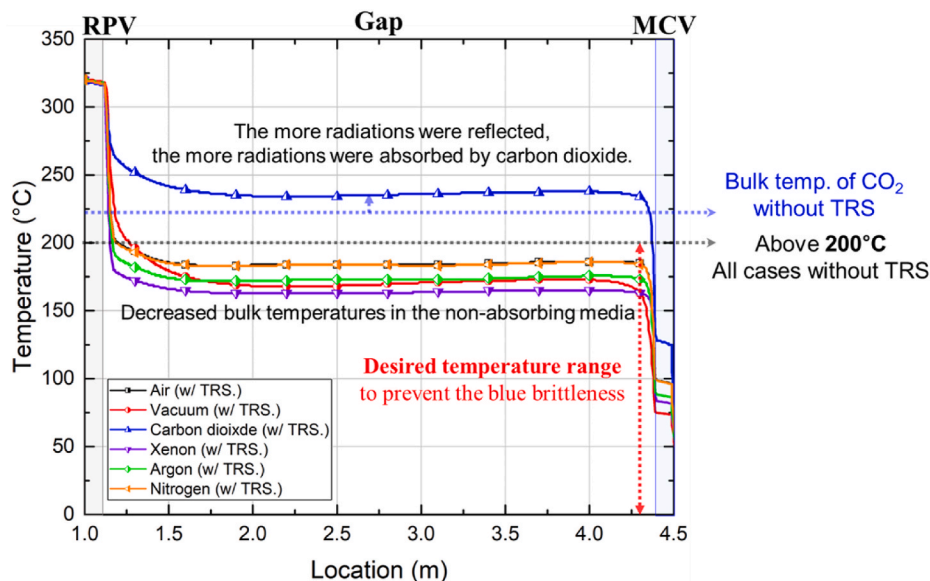


Fig. 28. Temperature distribution when TRS is applied.

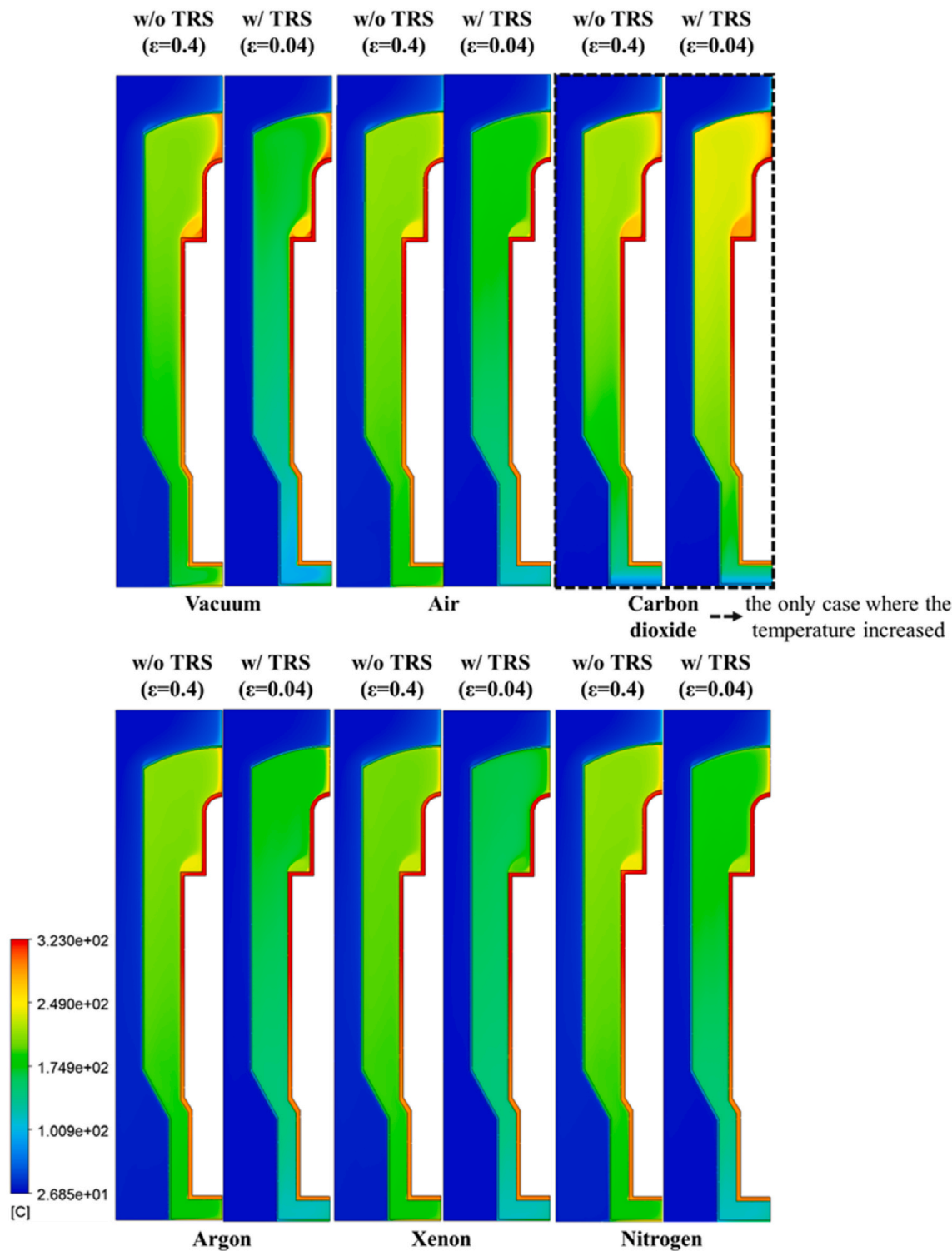


Fig. 29. Temperature contours for the candidate gases when TRS is (right) and is not (left) applied.

- A mesh sensitivity analysis was conducted on five different meshes with varying sizes within the gap between the MCV and the RPV. To ensure suitable mesh independence, the threshold for differences in major results such as temperature distribution, total heat transfer rate, the contribution of radiative heat transfer was not to exceed 1%. The analysis led to the selection of a mesh with 416,995 elements for the CFD simulations.
- The results for the natural convective heat transfer differed between the cases when radiation was and was not considered, suggesting

that there are complex interactions between the heat-transfer mechanisms. As a result, a more precise evaluation of the heat-loss mechanisms in CFD analysis requires the inclusion of a radiation model.

- Radiative heat transfer, which accounted for 48–97% of the total heat loss, was the driver for significant total heat loss under all gap conditions. The insulation performance was highest for carbon dioxide, followed in order by vacuum, xenon, argon, air, and nitrogen. However, due to the complex complementary effect between natural

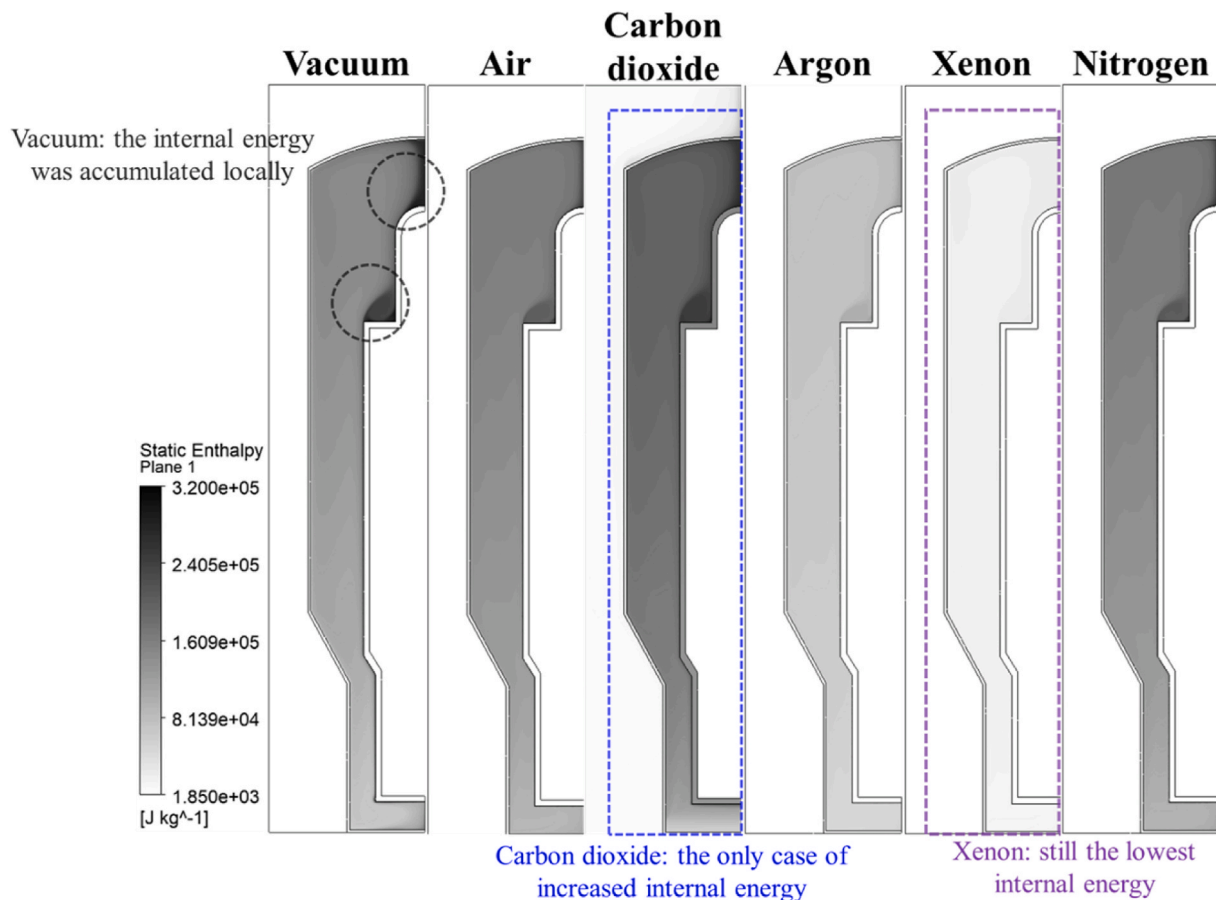


Fig. 30. Contours of the static enthalpy for the candidates when TRS is applied.

convection and radiation, the differences between the non-absorbing candidates were less than 9.6%. In addition, carbon dioxide, an absorption medium, was found to have a 14% lower total heat loss than vacuum, the best insulator among non-absorbent media.

- Since at least 48% of the heat loss was due to radiation, TRS was considered to have significant potential for improving the insulation performance of IPWR-type SMRs. This was confirmed by the significant reduction in the total heat loss with the use of TRS, ranging from 56% for air to 70% for a vacuum. However, only a 13% improvement in carbon dioxide was confirmed due to the increase in natural convection heat transfer by absorption coefficient. Therefore, carbon dioxide is not recommended as the gap condition in the TRS design.

By confirming that radiative heat transfer was the dominant mechanism within the operating temperature range for IPWRs, this study is able to recommend strategies that are most likely to be effective in reducing heat loss. By demonstrating that heat loss can be effectively mitigated with the use of a low-emissivity coating such as an aluminum layer, it is possible that other options rather than a vacuum can be considered to ensure an acceptable insulation performance, assuming that fire hazards can be prevented. In particular, we suggest that the vacuum could be replaced with the gases at a pressure of 1 bar, thus avoiding the additional maintenance costs associated with a vacuum. In addition, to avoid the oxidation of the TRS, the gap should be filled with inert gases because the emissivity of aluminum can be significantly increased by generating an oxide layer on the surface. For this purpose, we suggest carbon dioxide with an absorption coefficient and inert gas with low heat transfer characteristics, i.e., argon, as a practical alternative for IPWR design with and without TRS, respectively. Xenon,

which showed the lowest natural convective performance, was excluded from the practical alternative candidate due to its high cost. Although condensation performance remains a limitation in the event of an accident, they may also secure sufficient cooling performance through the hydrophobic coating on the TRS with high thermal conductivity [61]. In the future, the dominant role of radiative heat transfer confirmed in this study will be precisely demonstrated using more advanced computational analysis and a scaling experiment.

Declaration of competing interest

We declare that the submitted work has no conflict of interest with any relevant research institutes, academics, or industries.

Acknowledgments

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