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Effect of Radiative Filling Gas in Compound Parabolic Solar Energy Collectors

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ABSTRACT

In the present paper, the use of radiating gas instead of air inside the cavity of compound parabolic collectors (CPSs) is suggested and verified by numerical analysis. The collector under study has a simple cone shape with flat absorber which is filled with a participating gas such as carbon dioxide instead of air for the purpose of increasing the thermal performance. In numerical simulation, the continuity, momentum and energy equations for the steady natural convection laminar gas flow in the CPC's cavity and the conduction equation for glass cover and absorber plate were solved by the finite element method (FEM) using the COMSOL multi-physics. Because of the radiative term in the gas energy equation, the intensity of radiation in participating gas flow should be computed. Toward this end, the radiative transfer equation (RTE) was solved by the discrete ordinate method (DOM), considering both diffuse and collimated radiations. The S_4 approximation was employed in calculation of the diffuse part of radiation. It was observed that the gas radiation causes high temperature with more uniform distribution inside the cavity of collector. Also, numerical results reveal more than 3% increase in the rate of heat transfer from absorber surface into working fluid and hence a desired performance for the collector because of the gas radiation effect. Comparison between the present numerical results with theoretical and experimental data reported in the literature showed good consistency.

NOMENCLATURE

c_p	Specific heat (kJ/kg K)	d	Diffuse
g	Gravitational acceleration (m.s^{-2})	b	Black body
h	Convection coefficient (W/m^{2}K)	w	Wall
k	Thermal conductivity (Wm^{-1}K^{-1})	**Greek symbols**	
L	Length (m)	α_{glass}	Glass absorptivity (-)
I	Radiation intensity (W/m^2)	β	Volumetric thermal expansion (1/K)
q	Heat flux (W/m^2)	ρ_{glass}	Glass reflectivity (-)
T	Temperature (K)	τ_{glass}	Glass transmissivity (-)
(u, v)	x- and y- velocity components (m/s)	δ	Thickness (m)
p	Pressure (Pa)	μ	Viscosity (Pa. s)
r	Position vector (m)	ρ	Density (kg/m^3)
s	Radiation beam direction	σ_a	Absorption coefficient (1/m)
(x, y)	Horizontal and vertical coordinates respectively (mm)	ϵ	Surface radiation emissivity
Subscript		Ω	Solid angle
abs	Absorber	τ_0	Optical thickness
c	Collimated	ϑ	Kinematic viscosity (m^2/s)

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INTRODUCTION

Today, using renewable energies has attracted a great number of researchers; because the prediction for global population and their energy demand is important in near future. Even with the current rate of energy consumption, there is a deep gap between total demand and provided renewable energy sources. On this subject, the use of solar energy for many engineering applications such as air and water heaters has a major role. In many types of solar heaters, the incoming solar irradiation is concentrated on a receiver plate or absorber for having much more heat flux; where the solar collector is the key component of these thermal systems. Among the different types of solar concentrators, Compound Parabolic Concentrators (CPCs) have attracted a large attention. This type of solar collectors has naturally a low heat loss characteristic property. This property makes it more efficient than other common collectors; so it is a good design for medium and high temperature applications. Since CPCs are designed as stationary collectors and are capable to provide a wide range of temperature from (T < 100 ℃) to (T > 250℃); therefore, they are cost effective and capable of being used in many thermal systems such as solar desalination [1].

The schematic diagram of CPC solar collectors with different geometrical shapes is shown in Figure 1. These collectors are non-imaging concentrators with the capability of intercepting and reflecting the incident solar radiation to the absorbing surface over a wide-angle range. With a proper choice of orientation and inclination, it is possible to avoid expansive tracking systems. An additional reason of the interest is the capability of accepting diffuse solar radiation.
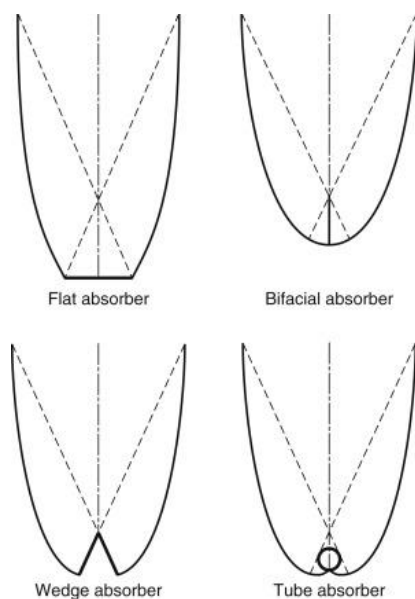

Figure 1. Schematics of CPCs with different geometrical shapes

Solar thermal applications with medium temperature can be used in many systems, such as water heater and desalination unit [1], drying process [2] and air conditioning (space heating) [3]. Among all of the solar collectors, flat plate collectors and CPCs are commonly used in these applications. Although, many research works were conducted for designing high performance collectors [4]. However, there is a deep gap between the existing collectors available in the market with the optimum and high efficient ones. A large review of the solar thermal applications of CPCs can be found in the literature [5]. A series of alternative methods for enhancing the rate of heat transfer and converting much more solar radiation into enthalpy of working fluid in CPCs have been proposed by Francesconi et al. [6] and Reichl et al. [7].

A detailed parametric numerical analysis of fluid flow and heat transfer in CPC solar collectors has been performed by Eames and Norton [8]. In that work, collectors with different acceptance angles with two sets of thermal boundary conditions were investigated. The stream function vorticity approach was employed in numerical solution of flow equations using tri angular elements. Numerical results presented in the forms of isotherm and streamline plots and Nusselt number distribution. A correlation for the Nusselt number variation with the Grashof number, incorporating an angular dependence was also reported. The laminar natural convection in a CPC cavity was simulated using the finite element method by Chew et al. [9]. Numerical findings were presented for representative CPC collectors with tubular absorbers of concentration ratio 2. The effects of Grashof number, truncation and inclination angle on the thermal behavior of collector were thoroughly explored. It was also reported that higher rates of heat transfer between the tubular absorber and the flat cover plate of the cavity are associated with high Grashof numbers and shallower cavities.

A novel concentrated solar thermal collector with evacuated tube was introduced and studied by Bhusal et al. [10]. In that work, an attempt was made for optimization of cost and performance with respect to commercially available medium temperature collectors using thermal and economic index of heat generation system. The levelized cost of heat was optimized with material cost reduction for maintaining high thermal performance using economical aluminum fin, a glass cover and highly truncated aluminum reflector. All simulations were performed by the COMSOL Multi physics, and the numerical findings were validated against experimental data. All of the theoretical and experimental findings revealed that the proposed system can be considered as an efficient CPC solar collector.

In a recent study, the development of a novel CPC evacuated tube solar collector, using a medium-temperature selective coating was described by Ma et al. [11]. The medium-temperature selective coating was

obtained by co-sputtering titanium and aluminum targets. The optimization of collector was done, using a manifold of copper U tube with anti-oxidized coating and aluminum fin. The experimental results showed that the proposed solar collector can operate continuously at temperatures over 373 K. Such that on sunny days, the maximum outlet temperature of the collector reaches 413–453 K, and the system can produce the water steam at high temperatures in the range from 381 to 418 K.

Although there are a great number of published papers about the CPCs, but after a careful review of the pertinent literature by the authors, any study was found about the exploitation of the inherent merit of radiating gases to participate and contribute with the heat transfer characteristics in improving CPC performance. Actually, for this special application, any renewable solar-thermal system, employing radiative filling gas was not still recognized as a method of heat transfer enhancement for any type of compound solar collectors. The lack of literature on this subject motivates the authors to conduct a theoretical investigation on radiative gas potential to change the destiny of this type of renewable solar-thermal system. On the subject of convective flow of radiating gases, the interaction between convection and radiation heat transfer was studied by the first author in his previous work [12]. Also, the effect of gas radiation in enhancing heat transfer in plane solar air heaters was also investigated [13, 14]. To materialize this aim in CPCs, the current study introduces the use of radiating filling gas such as carbon dioxide, water vapor and N_2O instead of air inside the cavity of compound parabolic collector. It is clear that in order to prevent gas leakage, the boundary surfaces of the chamber must be completely sealed. In numerical simulation, the set of governing equations including the conservations of mass, momentum and energy for the buoyant gas flow in the cavity and the conduction equation for the glass cover and absorber plat were solved, simultaneously. Because of the gas radiation, and the existence of radiative term in the gas energy equation, the radiative transfer equation (RTE) was also solved for calculation of radiant intensity distribution. The set of coupled partial differential equations were solved by the FVM using the COMSOL Multi-physics. The effects of gas absorption coefficient on the flow and temperature pattern and also on heat flux distribution along the absorber plate are discussed in detail. The author hopes that the obtained numerical findings would assist the scientific community in further developing efficient methods by using radiative gas instead of air in the cavity of compound parabolic collectors.

MODEL DESCRIPTION

Schematic of the computational domain used in this study with details is shown in Figure 2. The radiating gas flows
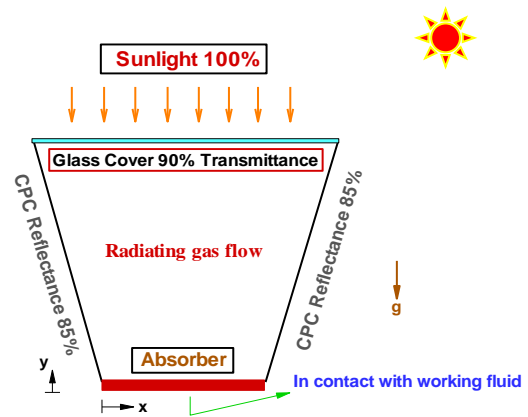

Figure 2. Geometry of a cone shaped CPC

inside the gap between the glass cover, absorber and reflecting surfaces of a cone shape collector, because of the buoyancy effect. Other geometrical parameters for different parts of the heater including the glass and absorber plate are depicted in Table 1. The Grashof number defined as $Gr = g\beta\Delta T L^3/\nu\alpha$ is kept below 10^9 for laminar flow condition. Most of the solar heat flux transmits from the glass cover and equal to $\alpha_{glass}.q_{Sun}$ of the incident radiative energy is absorbed by glass sheet. So, for this element, the conduction equation with heat generation term is solved for temperature calculation. The solar radiation is considered as collimated beam which is in normal direction respect to the glass cover. The lower absorber surface which is in contact with the working fluid to be heated by concentrated solar beam is considered isothermal at 340 K. The boundary surfaces of the glass cover and reflecting walls are assumed in convection heat transfer with surrounding. All thermo-physical parameters of the absorbing and emitting gas inside the CPC's cavity are supposed to be constant, with the exception of the medium density, as its change with temperature is determined by the Boussinesq approximation.

Table 1. Values of parameters for the CPC under study

Parameter	Value	Parameter	Value
δ_{glass}	3 mm	δ_{abs}	20 mm
ϵ_{glass}	0.9	ϵ_{abs}	0.95
α_{glass}	0.05	ρ_{glass}	0.05
τ_{glass}	0.9	T_{amb}	293 K
k_{glass}	0.78 W/m K	L_{abs}	50 mm
k_{abs}	300 W/m K	L_{glass}	100 mm
Cavity height	100 mm	q_{Sun}	**1000 W/m²**
ρ_{wall}	0.85	T_{abs}	340 K

GOVERNING EQUATIONS

In numerical simulation, the following equations including, conservations of mass, momentum and energy for the buoyant gas flow inside the cavity of collector and the conduction equation for the absorber and glass cover were solved.

i) Flow equations

Continuity:
$$\nabla.\vec{V} = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

x-momentum:
$$\vec{V}.\nabla u = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \nu \nabla^2 u \tag{2}$$

y-momentum:
$$\vec{V}.\nabla v = -\frac{1}{\rho}\frac{\partial p}{\partial y} + \nu \nabla^2 v + g\beta(T - T_{amb}) \tag{3}$$

Energy:
$$\vec{V}.\nabla T = \alpha \nabla^2 T - \frac{1}{\rho c_P}\nabla \cdot \vec{q}_r \tag{4}$$

ii) Glass cover conduction equation

$$\nabla^2 T_{glass} + \frac{\alpha_{glass}.q_{sun}}{\delta_{glass}.k_{glass}} = 0 \tag{5}$$

iii) Conduction equation for absorber

$$\nabla^2 T_{abs} = 0 \tag{6}$$

Since, the momentum equations are temperature dependent because of the buoyancy terms, all of the flow equations including conservation of mass, momentum and energy for the gas flow and conduction equations for solid parts should be solved, simultaneously.

Radiation computations

The divergence of radiative heat flux presented in the gas energy equation depends to temperature and radiant intensity distributions inside the participating medium as follows [15]:

$$\nabla.\vec{q}_r = \sigma_a.(4\pi\sigma T^4(\vec{r}) - \int_{4\pi} I(\vec{r},\hat{s})d\Omega) \tag{8}$$

Solar irradiation that enters the collector cavity via the glass cover is a collimated beam. In order to consider the effects of solar beam in radiation computations, the radiant intensity within the cavity is separated into two parts: (1) the remnant of the directional or collimated short wave beam after partial extinction along its path, and (2) a fairly diffuse part, which is the result of long wave emission from the boundaries and the radiating gas that can be varied along its path by emission and absorption [15]. Thus, we set:

$$I(r,s) = I_c(r,s) + I_d(r,s) \tag{7}$$

Such that the variation of collimated beam along its path obeys the equation of transfer;

$$(s.\nabla)I_c(r,s)] = -\sigma_a I_c(r,s) \tag{8}$$

subject to the boundary condition on the glass-gas interface.

$$I_c(r_w,s) = \tau_{glass}q_{Sun}.\delta\,[s - s_c(r_w)] \tag{9}$$

Equation (8) with its boundary condition are readily solved as

$$I_c(r,s) = \tau_{glass}q_{Sun}.\delta\,[s - s_c(r_w)]\,e^{\tau_c} \tag{10}$$

where, δ is the Dirac-delta function and $\tau_c = \int_0^s \sigma_a ds$ is the optical thickness. For the diffusion part of radiant intensity, the equation of transfer becomes:

$$(s.\nabla)I_d(r,s) = \sigma_a[-I_b(r) + I_d(r,s)] \tag{11}$$

The radiative boundary condition of the above equation depends on whether the surfaces are opaque or semitransparent. For the opaque ones, it will be as follows:

$$I_d(r_w,s) = \epsilon_w I_b(r_w) + \frac{(1-\epsilon_w)}{\pi}\int_{\vec{n}_w.\vec{s}<0} I_d(r_w,s)\,|\vec{n}_w.\vec{s}|d\hat{\Omega} \qquad \vec{n}_w.\vec{s} > 0 \tag{12}$$

and for semitransparent surface which is the glass cover in the case, the boundary condition becomes:

$$I_d(r_w,s) = \epsilon_w I_b(r_w) + \frac{(1-\epsilon_w)}{\pi}\Big[H_c(r_w) + \int_{\vec{n}_w.\vec{s}<0} I_d(r_w,s)\,|\vec{n}_w.\vec{s}|d\hat{\Omega}\Big] \qquad \vec{n}_w.\vec{s} > 0 \tag{13}$$

where

$$H_c(r_w) = \int_{\vec{n}_w.\vec{s}<0} I_c(r_w,s)\,|\vec{n}_w.\vec{s}|d\hat{\Omega}\,] \quad \vec{n}_w.\vec{s} > 0 \tag{14}$$

According to the DOM, RTE is solved for a set of n discretized directions $\vec{s}_i$, i=1,2,3,...,n, while integrals over solid angle are replaced by the numerical quadrature based on the following relation:

$$\int_{4\pi} f(\vec{s})\,d\Omega = \sum_{i=1}^{n} w_i f(\vec{s}_i) \tag{15}$$

For 2D computational domain, as in the present case, the S4 approximation is used that obliged discretized directions to be n=12. Moreover, the black body radiation intensity $I_b(r)$ can be computed as follows:

$$I_b(r) = \frac{\sigma}{\pi}T^4(r) \tag{16}$$

Since, the solar radiation is collimated and the side walls of CPC are specular and refractory surfaces, analysis of concentrating collectors is commonly done by ray-trace methods. For this, the ray trace starts with the assembly of rays of solar incident beam on the aperture and determines the intensity and distribution of those beams on the absorber plate. Ray tracing in solar collector systems is usually done with vectors as it is in the COMSOL multi-physics. Such that for a reflecting surface, the direction and point of intersection of an incident ray with the reflecting surface are determined. For this purpose, the normal vector to the surface is

computed from its shape, and the direction of the reflected ray is determined base on this rule that the angles of reflection and incidence are equal. This subject can be found in the literature [16]. It should be mentioned that the above procedure was done in the present work only for collimated beams. In the computations of diffuse part of radiation, the RTE was solved with the DOM using the S_4 approximation. The method of solution by the discrete ordinate method which its description is omitted here for space saving was described in the previous work by the first author where the radiating gas flow in a single flow solar gas heater was studied [14].

Boundary conditions

In numerical solution of governing equations for analysis of collector, appropriate boundary condition should be applied.

• No slip condition for all solid walls is employed in velocity computation.

• The incoming solar beam corresponds to q_{Sun}=1000 W/m^2 normal to the glass cover is considered as the source term of heat transfer.

• The continuity of temperature and heat flux at the gas-solid interface are also imposed, where in the radiating gas, the total heat flux is sum of conductive and radiative terms. Therefore, the following boundary condition according to the continuity of heat flux is imposed on the interfaces of gas flow with glass cover and absorber.

$$\left[-k_{th}\frac{\partial T}{\partial n}\right]_{solid} = \left[-k_{th}\frac{\partial T}{\partial n}\right]_{gas} + \epsilon_w \left(\pi I_b(\vec{r}_w) - \sum_{\vec{n}_w.\vec{s}<0} w_i |\vec{n}_w.\vec{s}| I(\vec{r}_w)\right) \quad (17)$$

Also, the temperature of absorber on its lower surface which is in contact with working fluid flow is considered to be 340 K. This temperature is considered in simulation because its value is in the normal range for many engineering applications, such as in heating water for domestic usage.

• On the top surface of the glass cover and refractory side walls adjacent to surrounding, third kinds of boundary conditions with h=5 W/m² and $T_{amb} = 293\,K$ are imposed (Figure 2). It should be noted that an extra equivalent convection coefficient for the glass cover because of the surface radiation is added to its convective counterpart. The details of convection coefficients computations which are needed in imposing the boundary conditions were reported in the literature [14] and are not explained here for space saving.

Gris size in computational domain

In order to find an optimum grid size that makes numerical results more accurate and independent from the spatial discretization and simultaneously confers a computationally cost-effective mesh, the most sensitive parameter to the grid size which is the maximum temperature inside the computational domain was selected for grid independency test. The value of this parameter against different grid sizes was investigated and the results are presented in Table 2. The precision of presented results in prediction of maximum temperature was set equal to 0.5% and 4800 node number was selected, accordingly.

The unstructured triangular grid generation method was used for the 2D geometry of CPC cavity, Figure 3, such that near the walls, absorber and glass cover, mesh was refined to precisely capture the high gradient of velocity and temperature in those regions. It should be mentioned that the same grid size and discretized domain was used for both flow and radiation computations.

VALIDATION

The result of present numerical simulation was validated first with the numerical findings of Foruzan Nia et al. [13]. The laminar forced convection of working gas in a single-pass solar gas heater with a simple geometry was solved by finite volume methods. The gas flow was considered as a participating medium which consequently radiative transfer equation coupled with the rest of governing equations have to be solved. Their exact geometry considering all corresponding boundary conditions is reproduced in COMSOL Multi-physics and solved by the finite element method and a triangular unstructured mesh. The temperature profile across the channel at two different locations, namely $x = \frac{L}{4}$ and $x = L$, is computed and plotted in Figure 4 at the optical thickness equal to 2. Due to the significant difference

Table 2. Effect of grid size on the converged solution

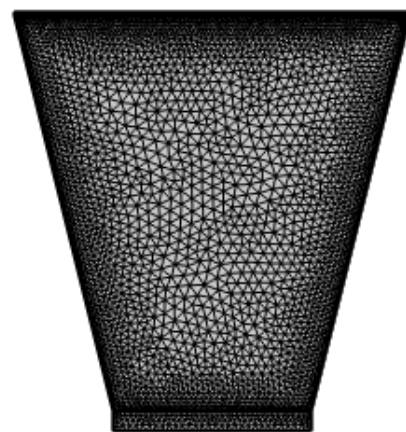| Number of grids	2400	3300	4200	4800	5400
T_{max} (°C)	333.2	349.8	355.3	357.1	358.9
Error respect to previous step	-	5%	1.5%	0.5%	0.5%

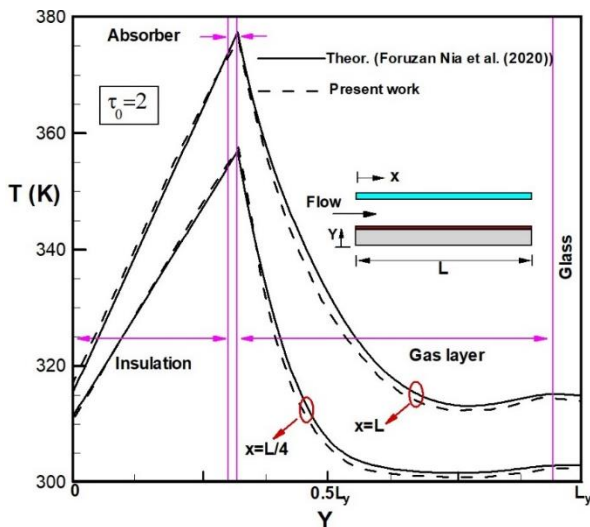**Figure 3.** The unstructured triangular mesh

Figure 4. Temperature profiles across the channel of planar solar gas heater at two different locations, comparison with results of numerical FVM simulation from Frouzan Nia et al. [13]

Figure 5. Comparison of the present numerical findings with experimental data

between the thermal conductivity of gas and absorber plate, temperature profile breaks at their interface while still keeps its continuity. As shown, there is a near consistency and a strong agreement between the current simulation and numerical results reported in the literature [13].

For further verifying the accuracy of present analysis, the plane solar air heater which was investigated experimentally in the literature [17] is numerically simulated and the findings are compared with experimental data. The geometry of this heater is similar to the one which was shown in Figure 3. At different air mass flow rate, the SAH efficiency is calculated and its distribution as a function of air mass flow rate are plotted and compared with experimental data illustrated in Figure 5. This figure illustrates that the thermal efficiency increases with increasing in air mass flow rate up to its maximum value. However, it is seen that there is a good consistency between the present numerical findings with experiment.

RESULTS AND DISCUSSION

After the strongly validated numerical procedure, from grid generation to model adoption and convergence, results of numerical simulations in a comparative form for comprehensive thermohydrodynamic analysis of laminar natural convection in a cone shaped CPC cavity are presented in this section. In the beginning, the contours of velocity magnitude in free convection air flow are illustrated in Figure 6 for non-radiating case and also for participating gases with different absorption coefficients ($\sigma_a = 0.1, 0.5 \ m^{-1}$). As seen, the flow has
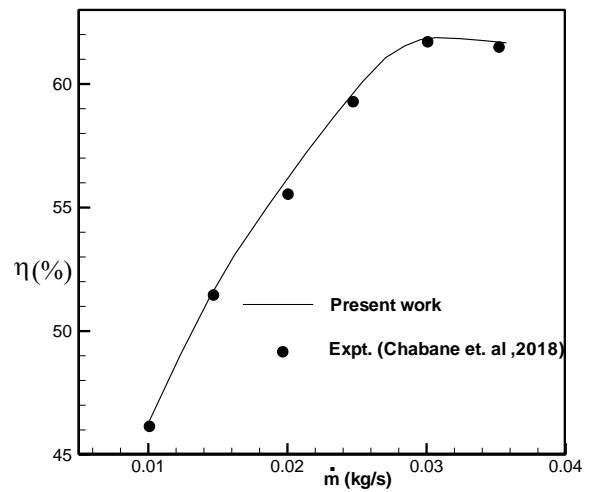
two recirculated cells lying symmetrically about the axis of cavity. The buoyant force near to the heated absorber surface pushes the working gas up and there are two air streams closed to the surrounding walls that move upward through a bi cellular flow pattern and then with downward streams near the cavity axis (Figure 7). The same trend is also seen for the free convection of radiating gases. If one compares the velocity magnitude contours for non radiating and radiating gas flows, it is seen that the gas radiation damps the convective velocity and flow vortices. It is due to this fact that in the case of radiating working gas, the radiative heat transfer in addition to its convective counterpart of heat transfer causes more uniform temperature inside the flow domain that leads to a decrease in the value of gas density gradient and then in buoyancy force. The corresponding stream line plots for these test cases are drawn in Figure 8. The two recirculated zones inside the left and right sides of the CPC cavity can easily be seen in this figure, such that lower rate of flow vortices takes place because of the gas radiation, such that in the vicinity of the absorber, the buoyant flow occurs with very low velocity such that there is almost as a stagnant zone.

A series of isotherm plots at different gas absorption coefficients and also for non-radiating gas are shown in Figure 9. In non radiating case, the central zone of the CPC cavity is in relatively low temperature and increasing in gas temperature can be seen as we moves toward the absorber and side walls which are under the incidence of Sun heat flux. In the cases of using radiating gas, the temperature field becomes more uniform due to radiative heat transfer combined with convection, but still the maximum temperature takes place in the vicinity of side walls. In the condition of very high gas absorption, say, $\sigma_a = 0.5 \ m^{-1}$, the gas temperature in central zone increases considerably, which is very greater that the absorber surface. This is due to this fact that, rather than
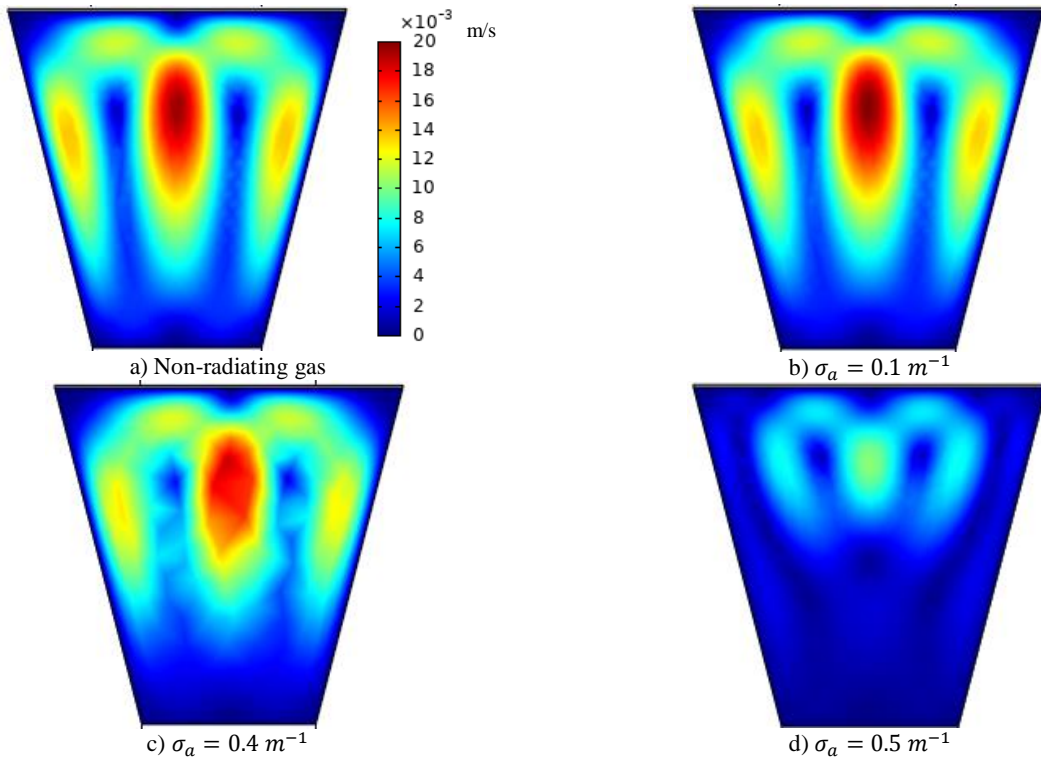
a) Non-radiating gas

b) $\sigma_a = 0.1 \ m^{-1}$

c) $\sigma_a = 0.4 \ m^{-1}$

d) $\sigma_a = 0.5 \ m^{-1}$

Figure 6. Contours of velocity magnitude

the diffuse long wave radiation from the boundary surfaces, the radiating gas can also absorb directly the incoming Sun radiative beams transmitted across the glass cover. It should be noted that when the cavity is charged with participating gas, the mechanisms of heat transfer to the absorber are both radiation and convection and while the gas temperature near to the absorber becomes greater than T_{abs}, much more thermal energy can be transferred into this element which is desired and causes an improvement in the CPC performance. This phenomenon can be studied more detail shown in Figure 10.
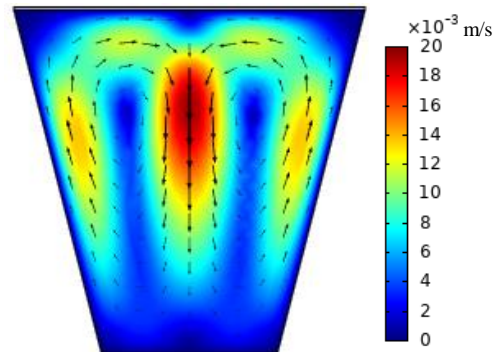

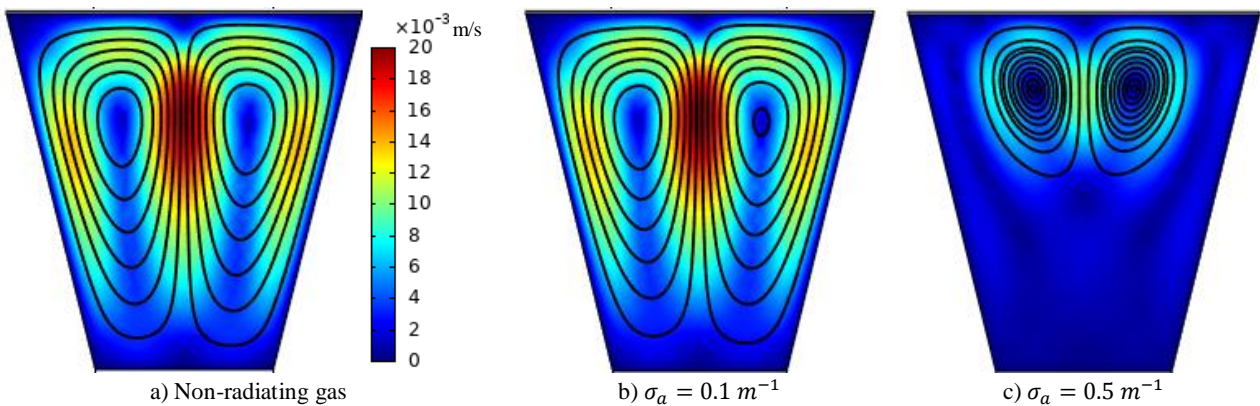**Figure 7.** Velocity vectors (Non-radiating gas)

a) Non-radiating gas

b) $\sigma_a = 0.1 \ m^{-1}$

c) $\sigma_a = 0.5 \ m^{-1}$

Figure 8. Stream line plots

a) Non radiating case   b) $\sigma_a = 0.1\ m^{-1}$ c) $\sigma_a = 0.2 m^{-1}$

d) $\sigma_a = 0.3\ m^{-1}$ e) $\sigma_a = 0.4\ m^{-1}$ f) $\sigma_a = 0.5\ m^{-1}$
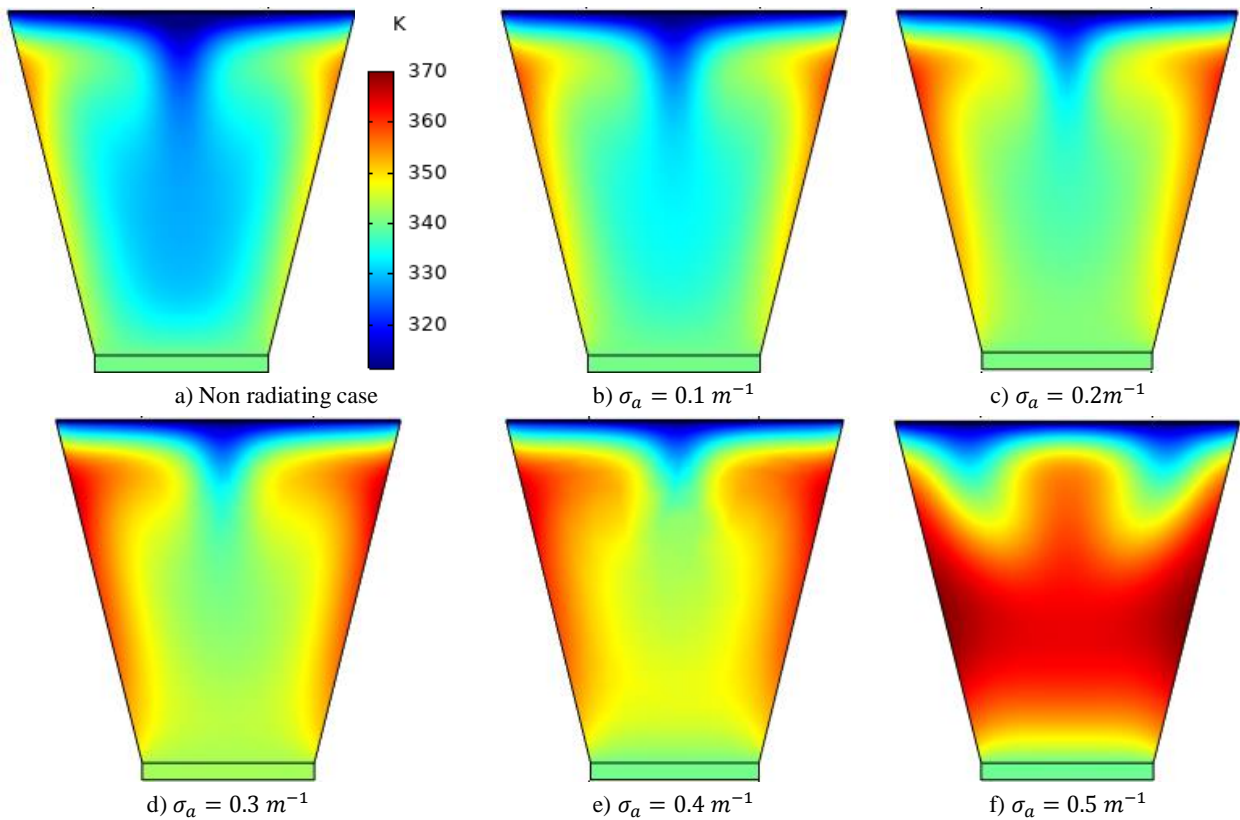
Figure 9. Isotherm plots

As shown in Figure 10, the temperature distributions along the vertical mid-line of the cavity, including absorber, gas flow and glass cover are plotted at different values of the gas absorption coefficient and also for non radiating case. This figure shows almost uniform temperature across the absorber because of high thermal conductivity and then a decreasing trend for gas temperature and also across the glass cover in the cases of non radiating gas and also for participating working gases
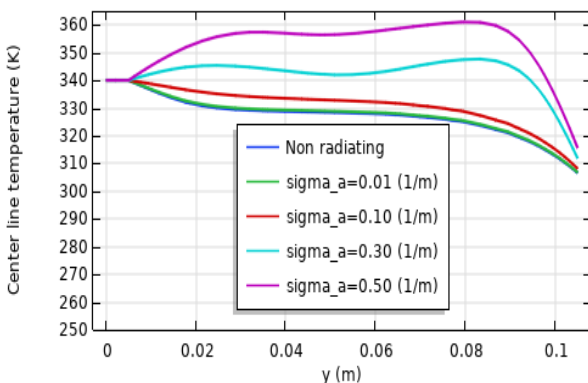

Figure 10. Temperature distribution along the vertical mid line of the CPC

with low absorption coefficient. But for high radiating gas, there is a temperature increase in the buoyant gas flow adjacent to absorber, and then almost uniform temperature distribution in central zone and finally the temperature distribution tends to a decreasing trend. So, the gas radiation has dual effect on the rate of heat transfer into the absorber, such that direct absorption of incoming Sun collimated beam by the participating gas causes less incident radiative flux on the absorber surface and less absorbed radiative heat flux, and on the other hand gas radiation effect causes convection heat transfer from hot gas into the absorber surface. Therefore, an optimization technique is needed for having the best performance for the collector under study. For this, the total heat flux distribution on the absorber which is the sum of radiative and convective parts are plotted for both radiating and non radiating gases in Figure 11. This figure demonstrates a bell shape distribution for all cases with this rule that the gas radiation causes an increase in the value of absorber heat flux. If one compares the heat flux distributions for non radiating gas with radiating one at $\sigma_a = 0.5\ m^{-1}$, about 30% increase in the value of absorber average heat flux can be computed.

As it was noted before, the gas radiation has both positive and negative effects on thermal behavior of collector. Figure 12 illustrates data which has
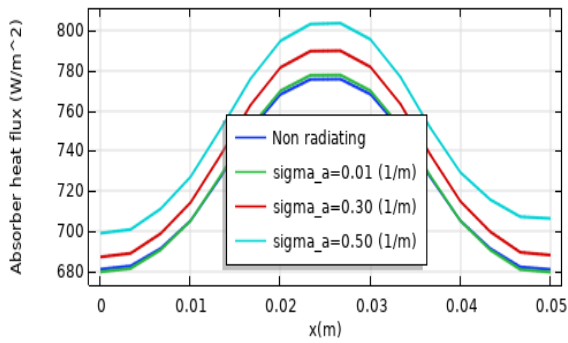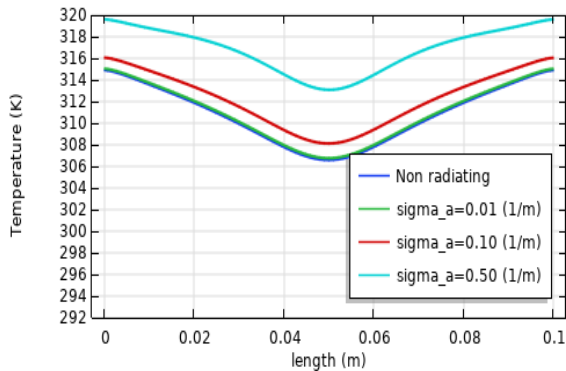
Figure 11. Total heat flux distribution along the absorber

Figure 12. Glass cover temperature distribution along its length

demonstrated the former that leads to higher rate of heat losses. As shown in this figure, the glass covers surface temperature along its length at different gas absorption coefficient and also for non-radiating gas are depicted. A considerable glass temperature increase is shown because of the gas radiation, such that the average glass temperature equal to 311 K for non radiating gas becomes 316 K at $\sigma_a = 0.5 \, m^{-1}$, corresponds to 1.6% increase. It is evident that higher glass temperature will cause more energy loss from the glass cover into the surrounding both by convection and surface radiation. As it was mentioned before that the gas radiation leads to direct absorption of the incoming solar beam from the glass cover, it is investigated in the next figure. The radiation intensity field $G = \int_0^{4\pi} I d\Omega$ inside the filling gas as a participating medium is depicted in Figure 13 for both weak and strong radiating gases. It is shown that at small values of gas absorption, $\sigma_a = 0.01 m^{-1}$, the solar beam which is transmitted across the glass cover penetrates more into the gas flow, but for high radiating gas, a major part of this incoming radiation is damped near to its entrance section, i.e. the glass cover. This behavior causes an increase in the gas temperature and finally in the value of entering heat flux towards the absorber plate as it was shown before in Figure 10.

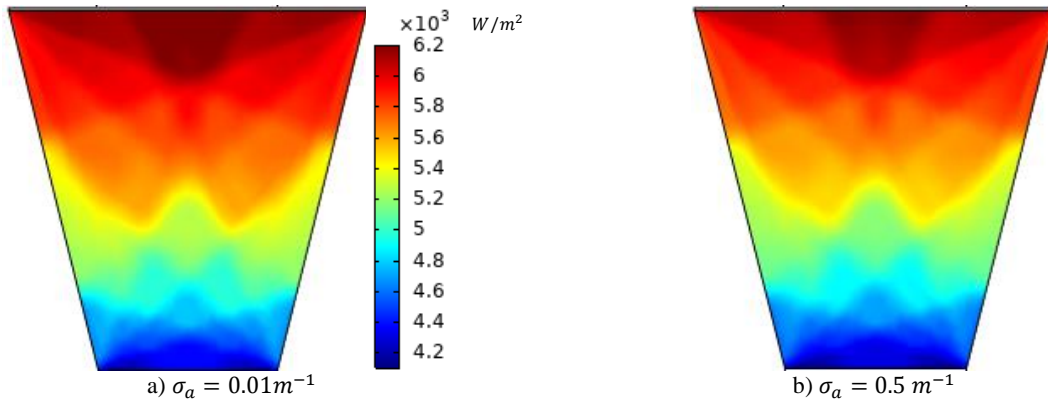a) $\sigma_a = 0.01 m^{-1}$ b) $\sigma_a = 0.5 \, m^{-1}$

Figure 13. Contour of radiant intensity

CONCLUSION

This work was dedicated to a sustainable solution for the exploitation of renewable energy facility through the thermal performance enhancement of CPC solar collector by proposing exploitation of radiative gas such as carbon dioxide. A comparison between the present numerical results with most recently published research paper showed very good consistency and accuracy. After detail thermo-hydrodynamic analysis of radiating gas laminar free convection inside the cavity of a cone shaped collector and conduction inside solid elements, peculiar

and special characteristics of radiating gas are revealed. The set of governing equations including continuity, momentum and energy for gas buoyant flow and conduction equations inside the absorber and glass cover were solved by the finite element method using the COMSOL multi-physics. Radiation computations were based on numerical solution of the RTE with the DOM. Numerical findings can be summarized as follows:

- The free convection gas flow inside the cavity of CPC has two recirculated cells lying symmetrically about the axis of cavity, while the rat rates of flow vortices decrease because of the gas radiation.

- The gas radiation causes higher temperature inside the central zone of CPC's cavity with more uniform distribution.
- Using radiative gases improves the performance of collector, such that for the studied test cases, more than 3% increase in the rate of heat transfer into the absorber of collector was seen because of the gas radiation effect.

Therefore, based on using renewable energies and avoiding fuel consumption, this study aims to motivate and pursue wide audiences for mitigation of greenhouse gas and human footprint by employing more efficient solar collectors.

DECLARATION OF COMPETING INTEREST

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

REFERENCES

1. Rojas, G. B., Rondon, R. L. A. and Gurrola, A. C. M. 2018. "Mechanical Engineering Design Theory Framework for Solar Desalination Processes: A Review and Meta-Analysis." *Iranian (Iranica) Journal of Energy and Environment*, 9(2), pp.137–145. https://doi.org/10.5829/IJEE.2018.09.02.09

2. Premkumar, S., Ramanarasimha, K. and Prakash, E. S. 2018. "Design and Development of Solar Crop Dryer Integrated with Oil Bath." *Iranian (Iranica) Journal of Energy and Environment*, 9(4), pp.277–283. https://doi.org/10.5829/IJEE.2018.09.04.08

3. Junfeng, L. and Runqing, H. 2005. "Solar thermal in China." *Refocus*, 6(5), pp.25–27. https://doi.org/10.1016/S1471-0846(05)70454-6

4. Nnamchi, S. N., Nnamchi, O. A., Sangotayo, E. O., Ismael, S. A., Nkurunziza, O. K. and Gabriel, V. 2020. "Design and Simulation of Air-Solar Preheating Unit: An Improved Design of a Flat Plate Solar Collector." *Iranian (Iranica) Journal of Energy and Environment*, 11(2), pp.97–108. https://doi.org/10.5829/IJEE.2020.11.02.02

5. Xu, D. and Qu, M. 2013. "Compound Parabolic Concentrators in Solar Thermal Applications: A Review." In *ASME 2013 7th International Conference on Energy Sustainability*. American Society of Mechanical Engineers. https://doi.org/10.1115/ES2013-18409

6. Francesconi, M., Caposciutti, G. and Antonelli, M. 2018. "CFD optimization of CPC solar collectors." *Energy Procedia*, 148, pp.551–558. https://doi.org/10.1016/j.egypro.2018.08.138

7. Reichl, C., Hengstberger, F. and Zauner, C. 2013. "Heat transfer mechanisms in a compound parabolic concentrator: Comparison of computational fluid dynamics simulations to particle image velocimetry and local temperature measurements." *Solar Energy*, 97, pp.436–446. https://doi.org/10.1016/j.solener.2013.09.003

8. Eames, P. C. and Norton, B. 1993. "Detailed parametric analyses of heat transfer in CPC solar energy collectors." *Solar Energy*, 50(4), pp.321–338. https://doi.org/10.1016/0038-092X(93)90027-L

9. Chew, T. C., Tay, A. O. and Wijeysundera, N. E. 1989. "A Numerical Study of the Natural Convection in CPC Solar Collector Cavities with Tubular Absorbers." *Journal of Solar Energy Engineering*, 111(1), pp.16–23. https://doi.org/10.1115/1.3268281

10. Bhusal, Y., Hassanzadeh, A., Jiang, L. and Winston, R. 2020. "Technical and economic analysis of a novel low-cost concentrated medium-temperature solar collector." *Renewable Energy*, 146, pp.968–985. https://doi.org/10.1016/j.renene.2019.07.032

11. Ma, G., Yin, Z., Liu, X., Qi, J. and Dai, Y. 2021. "Developments of CPC solar evacuated glass tube collector with a novel selective coating." *Solar Energy*, 220, pp.1120–1129. https://doi.org/10.1016/j.solener.2020.08.052

12. Atashafrooz, M., Gandjalikhan Nassab, S. A. and Lari, K. 2016. "Numerical analysis of interaction between non-gray radiation and forced convection flow over a recess using the full-spectrum k-distribution method." *Heat and Mass Transfer*, 52(2), pp.361–377. https://doi.org/10.1007/s00231-015-1561-z

13. Foruzan Nia, M., Gandjalikhan Nassab, S. A. and Ansari, A. B. 2020. "Numerical Simulation of Flow and Thermal Behavior of Radiating Gas Flow in Plane Solar Heaters." *Journal of Thermal Science and Engineering Applications*, 12(3). https://doi.org/10.1115/1.4044756

14. Dehghani Rayeni, A. and Gandjalikhan Nassab, S. A. 2020. "Effects of Gas Radiation on Thermal Performances of Single and Double Flow Plane Solar Heaters." *International Journal of Engineering - Transactions C: Aspects*, 33(6), pp.1156–1166. https://doi.org/10.5829/ije.2020.33.06c.14

15. Modest, M. 2003. Radiative Heat Transfer, 2nd Edition. Academic Press.

16. Terrón-Hernández, M., Peña-Cruz, M., Carrillo, J., Diego-Ayala, U. and Flores, V. 2018. "Solar Ray Tracing Analysis to Determine Energy Availability in a CPC Designed for Use as a Residential Water Heater." *Energies*, 11(2), pp.291. https://doi.org/10.3390/en11020291

17. Chabane, F., Moummi, N. and Brima, A. 2018. "Experimental study of thermal efficiency of a solar air heater with an irregularity element on absorber plate." *International Journal of Heat and Technology*, 36(3), pp.855–860. https://doi.org/10.18280/ijht.360311

Persian Abstract

چکیده

در مقاله حاضر، استفاده از گاز تشعشعی به جای هوا در داخل حفره جمع‌کننده‌های سهموی مرکب (CPS) پیشنهاد و با تجزیه و تحلیل عددی مورد بررسی قرار گرفته است. کلکتور مورد بررسی دارای یک شکل مخروطی ساده با جاذب مسطح است که به منظور افزایش عملکرد حرارتی، به جای هوا با گاز تابشی پر می‌شود. در شبیه‌سازی عددی، معادلات پیوستگی، مومنتوم و انرژی برای جریان گاز آرام لایه‌ای همرفت ثابت در حفره CPC و معادله هدایت برای پوشش شیشه‌ای و صفحه جاذب با استفاده از نرم‌افزار کامسول با روش المان محدود (FEM) حل شد. به دلیل وجود ترم تابش در معادله انرژی گاز، شدت تابش در جریان گاز تابشی باید محاسبه شود. به همین منظور، معادله انتقال تابشی (RTE) با در نظر گرفتن تابش‌های پخشی و جهتی با استفاده از روش مختصات گسسته (DOM) حل شد. از تقریب S_4 در محاسبه تابش پخشی استفاده شد. مشاهده شد که تابش گاز باعث ایجاد دمای بالا با توزیع یکنواخت‌تر در داخل حفره کلکتور می‌شود. همچنین، نتایج عددی بیش از ۳ درصد افزایش در شدت انتقال حرارت از سطح جاذب به سیال کار را نشان می‌دهد و از این رو عملکرد مطلوبی برای جمع‌کننده به دلیل اثر تابش گاز نشان می‌دهد. مقایسه نتایج عددی حاضر با داده‌های نظری و تجربی گزارش شده در مراجع، سازگاری خوبی را نشان داد.