

Transient Heat Transfer Characteristics of a Receiver for a Dish-Type Solar Thermal System

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ABSTRACT: A numerical and experimental studies are carried out to investigate the transient heat transfer characteristics of 5kWth dish-type solar air receiver. Measured solar radiation and temperatures at several different locations are used as boundary conditions for numerical simulation. Many parameters' effects (reflectivity of the reflector, the thermal conductivity of the receiver body, transmissivity of the quartz window, etc.) on the thermal performance are investigated. Discrete Transfer Method is used to calculate the radiation heat exchange in the receiver. A transient heat transfer model is developed and the rate of radiation, convection and conduction heat transfer are calculated. Comparing the experimental and numerical results, good agreement is obtained. Using the numerical model, the transient heat transfer characteristics of volumetric air receiver for dish-type solar thermal systems are known and the transient thermal performance of the receiver can be estimated.

Keywords : dish-type solar receiver system, computational transient analysis.

1. INTRODUCTION

Among renewable energy sources, it has been known that solar energy is one of the most prospective one to substitute for fossil fuel. As the concentrating technology develops, the high temperature solar energy becomes available, and the electricity can be generated using high temperature solar energy. Therefore the role of a receiver becomes very important. There are a few high temperature solar thermal systems such as dish-type solar system and solar power tower. In the present study, a receiver for a dish-type solar thermal system is considered.

Experimental and numerical investigations on natural convection heat transfer in receiver and enclosures have carried out by earlier researchers [1–3]. Singh and

Venkateshan [4] discussed a numerical study of combined laminar natural convection and surface radiation heat transfer in a 2D side vented receiver for different aspect ratios, side vented ratios and surface emissivities using air as the working fluid. Based on numerical data, separate Nusselt number correlations have been developed for convective and radiative heat transfer. A simple model for the convective heat transfer from a solar receiver based on the results of experimental studies of cubical receivers was presented by Siebers and Kraabel [5]. Subsequently McDonald [6] modified the model for a cylindrical shaped frustum receiver incorporating aperture size, surface temperature and receiver angle. Lage et al. [7] numerically analyzed natural convection and radiation in a 2D receiver

with open-end and extended simulations to an ash hopper. Dehghan and Behnia [8] analyzed numerically and experimentally the combined natural convection, conduction and radiation heat transfer in a discretely heated receiver. A comparison of the numerical and experimental observations showed that the accurate prediction of the flow and thermal field influenced by radiation heat transfer.

The previous studies were focused only on simple geometries and steady-state of receivers. However, the receiver geometry became complicate in order to improve their performances, and the solar insolation is not constant, and it strongly depends on the weather condition, which are not steady. Therefore, it is difficult to have constant solar insolation, so that the state of the receiver inside is not steady. Unfortunately, there is little information for these kinds of situations.

In the present study, a 3-D numerical model which includes transient heat transfer, natural convection, radiation heat transfer is developed to analyze the transient thermal performance of a volumetric air receiver, in which porous material is inserted. In order to verify the accuracy of the numerical model, experimental results are compared to the numerical results.

2. EXPERIMENTAL SETUP

2.1 Dish type Solar Thermal System

Fig. 1 is a schematic of the dish-type solar thermal system used for this study. The capacity of this collector, which consists of 10 pieces of parabolic reflectors are about 5kWth.

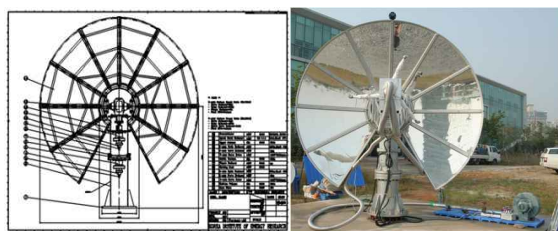


Figure 1: Dish type solar thermal system

The specification of this system is shown in Table I.

Table I: Specification of system

Specification	Size
Reflectivity of reflector	Above 95%
Diameter of reflector	3.2 m
Ofocal length	2 m
Total area of reflectors	5.90 m ²
Rim angle	43.85°

2.2 Air Receiver

The cross section of the receiver is shown in Fig. 2. In order to stand the high operating temperature, this receiver is made of 306 SS.

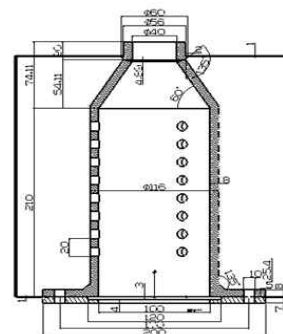


Figure 2: Cross section of the air receiver

As shown in Fig. 3(a), the air flows from the inside of the receiver through the three entrances (location2) and then flows out to the outside of the receiver through the exit (location1). The three entrances are attached in circumferential direction so that the flow naturally becomes vortex inside the receiver. Fig.3(b) is the photo of the working receiver.

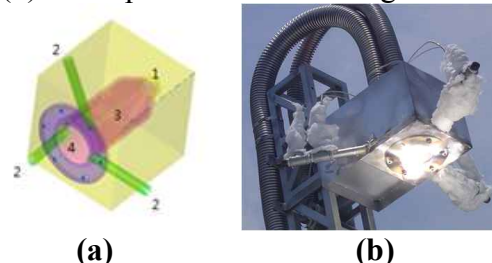


Figure 3: (a) Configuration of the receiver
(b) Photo of the working receiver

In order to increase the heat transfer area in the receiver, porous materials (nickel alloy) shown in Fig. 4 are inserted into the receiver. A quartz glass is installed at a location 4 to make a closed-loop.



Figure 4: Quartz glass, Porous material (nickel alloy)

2.3 Flux Mapping

The radiation heat flux distribution on the focal plane is measured. It is important information to determine the aperture size of a receiver. If the aperture size is too big, the heat loss increases. On the other hand, if the aperture size is too small, the spillage loss increases. Therefore, the optimal aperture size should be determined based on the radiation heat flux distribution. The aperture size of the receiver in this study is determined 10cm to capture 90% of the incoming solar radiation. Figure 5 (a) shows the target. And, CCD camera is used to take the image on the target.

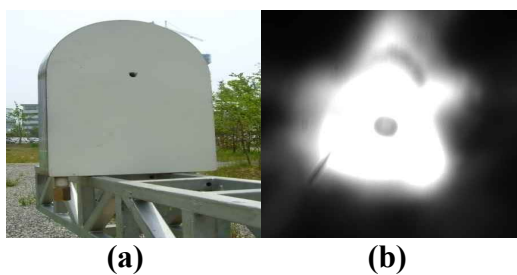


Figure 5: (a) Target with cooling system
(b) The image of radiation flux distribution

Fig. 5 (b) is a picture of the focal plane. The super thermo paint is applied on the target surface. A cooling system is installed to prevent the paint melting. A radiometer is installed at the center of the target to measure the radiation heat flux at the center of the target.

3. COMPUTATIONAL METHOD

The CFD analysis is conducted using the commercial CFD software, STAR-CCM+ Version 3.02. The flow and heat transfer simulation in the receiver is based on the simultaneous solution of the governing equations. The governing equations in the receiver are given as follows:

Continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial}(\rho u_j) = s_m$$

Momentum equation

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j u_i - \tau_{ij}) = -\frac{\partial p}{\partial x_i} + s_i$$

Energy equation

$$\frac{\partial \rho H}{\partial t} + \frac{\partial}{\partial x_i}(\rho H u_j + F_{h,j} - u_i \tau_{ij}) = \frac{\partial p}{\partial t} + s_i + s_h$$

Where,

$$H = \frac{1}{2} u_i u_i + h$$

Numerical analysis for radiation on solid surface is carried out with Discrete Transfer Method. The Discrete Transfer Method involves the tracing of representative rays from one surface to another through the domain of interest. [9] The Discrete Ordinates method is used to solve the radiative transfer equation for porous material which is treated as a participating medium.

The transient analysis is done by the implicit unsteady approach and experimentally measured unsteady heat flux is used as a boundary condition. K- ϵ turbulence model is used.

4. RESULTS AND DISCUSSION

The transient heat transfer characteristics of the volumetric air receiver are

numerically analyzed. The effects of the receiver geometry and operating conditions of the receiver on the thermal performance are investigated based on the numerical results.

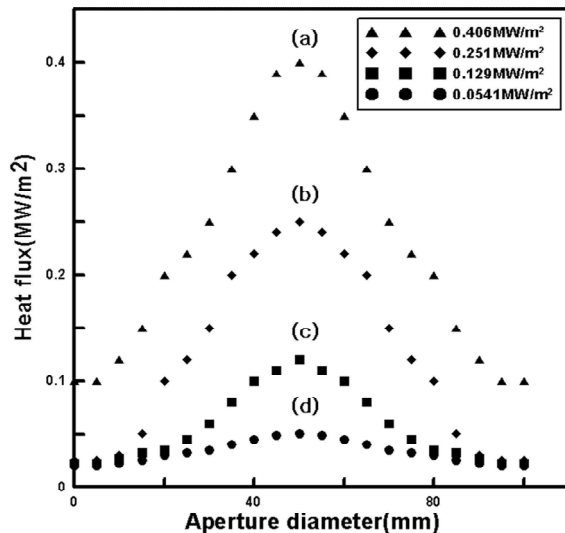


Figure 6: Distribution of the incoming concentrated heat flux on the aperture surface (a) max. heat flux: 0.406 MW/m^2 (b) max. heat flux: 0.251 MW/m^2 (c) max. heat flux: 0.129 MW/m^2 (d) max. heat flux: 0.0541 MW/m^2

Fig. 6 shows the experimentally measured incoming concentrated heat flux distribution on the aperture surface. The incoming concentrated heat flux is the Gaussian distribution ranging from 0.406 MW/m^2 at the center to 0.1 MW/m^2 on the circumference of the aperture. Also, in the theoretical and experimental results it is shown that the incoming concentrated heat flux is Gaussian distribution. Comparing the experimental and theoretical results, good agreement is obtained. On the basis of these results, the transient heat transfer is calculated.

In order to improve the thermal performance, the porous material is inserted inside the receiver.

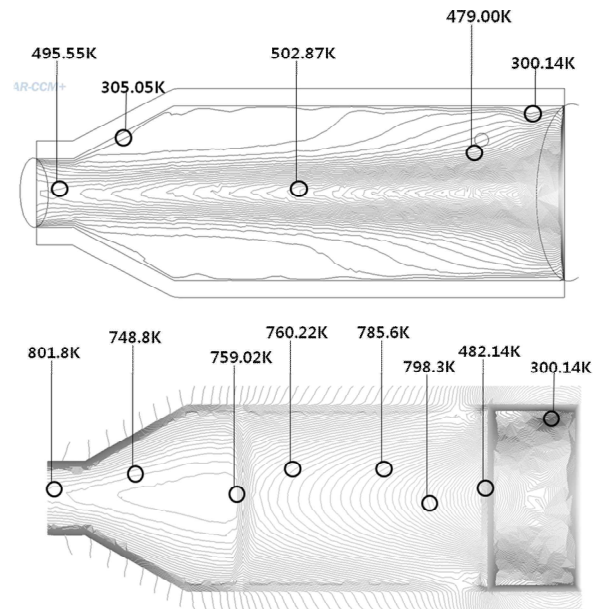


Figure 7: Temperature comparison of the different receivers (a) Receiver without porous material. (b) Receiver with porous material

In Fig 7, the temperature distributions inside the receiver are compared. Fig. 7(a) shows the temperature distribution of the receiver without porous material. Fig. 7(b) shows the temperature distribution of the receiver with porous material. This is steady results. On the exactly same conditions, the outlet temperature of the receiver with porous material is much higher than that of the receiver without porous material. When there is no porous material inside receiver, the radiation heats up the inner surface of the receiver. The air flowing through the receiver cannot have sufficient contact to the surface so that there is no sufficient heat transfer. On the other hand, when there is the porous material, radiation heats up the porous material and the air flows through the porous material. Therefore, sufficient heat transfer occurs.

In order to improve the thermal performance, the characteristics of the inserts should be known. Honeycomb structure, laminated mesh, porous material, etc. are used as the insert. From the heat transfer point of view, porous material is the best among three inserts [10].

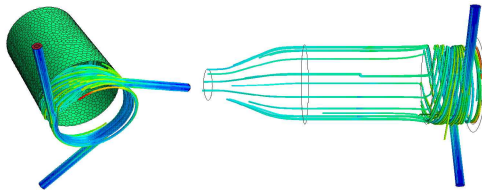


Figure 8: Flow streamline of receiver.

The internal flow streamlines of the receiver is shown in Fig. 8. The three inlets are installed in circumferential direction, and the flow becomes vortex inside the receiver. It is intentionally designed in order to increase the heat transfer rate on the aperture area which is the hottest spot of the receiver. Also, the air flow becomes well mixed and the temperature becomes uniform before entering the porous material.

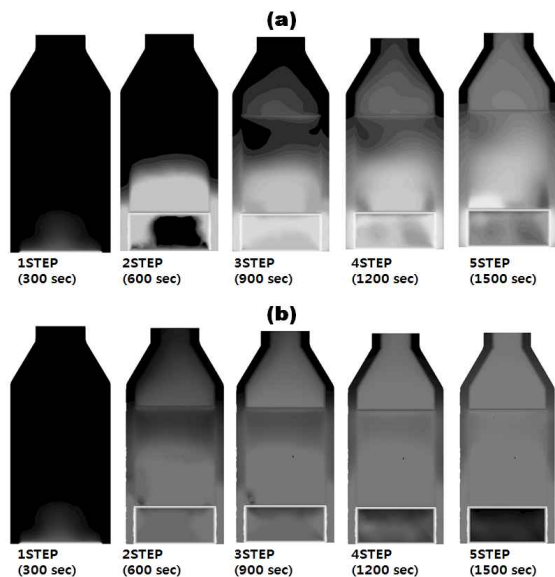


Figure 9: Transient temperature contour inside air receiver (time step = 300sec) (a) max. heat flux: 0.406 MW/m^2 (b) max. heat flux: 0.0541 MW/m^2

Fig. 9 is cross-sectional view of the transient temperature contour inside the air receiver. Measured heat flux input is used as a boundary condition at the aperture. The maximum heat flux is 0.406 MW/m^2 and the minimum heat flux is 0.0541 MW/m^2 . There are the distribution (a) and (b) of Fig. 6, respectively. The temperature contour is shown in time step 300sec. For the maximum heat flux, it takes more than

1500sec to heat the whole receiver up and continuously get the hot air from the outlet. On the other hand, for the minimum heat flux input, it takes less 1200sec to heat the whole receiver up.

The heat flux rate is in Fig. 10. Fig. 10 shows the heat flux which is variation of reflected radiation from the dish type solar concentrator to the receiver aperture as the sun moves. Decline of the heat flux is caused by declining of the sun.

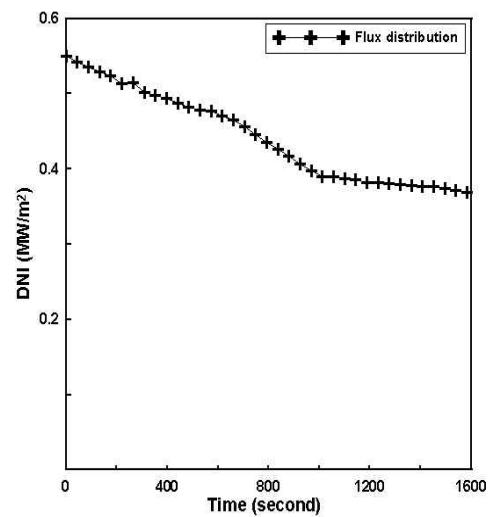


Figure 10: Heat flux rate of receiver aperture (location: latitude - $37^\circ, 23' 0.62'' \text{ N}$, longitude - $126^\circ, 39' 21.87'' \text{ E}$, time: 13:00 ~ 13:26)

Table II Transient analysis results of air receiver.

Time (second)	Max. heat flux (MW/m^2)	Outlet Temp. (K)	Inlet Temp. (K)
0	0.406	305.5	300
300	0.406	390.9	300
600	0.406	508.2	300
900	0.406	622.6	300
1200	0.251	600.1	300
1500	0.251	598.3	300
1800	0.251	558.0	300
2100	0.251	544.9	300
2400	0	508.6	300
2700	0	399.6	300
3000	0	368.6	300
3300	0	350.3	300
3600	0.0541	371.0	300
3900	0.0541	405.3	300

4200	0.0541	448.6	300
4500	0.0541	490.9	300
4800	0.129	499.6	300
5100	0.129	510.6	300
5400	0.129	539.8	300
5700	0.129	580.2	300
6000	0.406	600.9	300
6300	0.406	644.5	300
6600	0.406	690.0	300
6900	0.406	720.2	300
7200	0.406	771.5	300
7500	0.406	772.6	300
7800	0.406	777.1	300
8100	0.406	775.4	300
8400	0.406	774.9	300

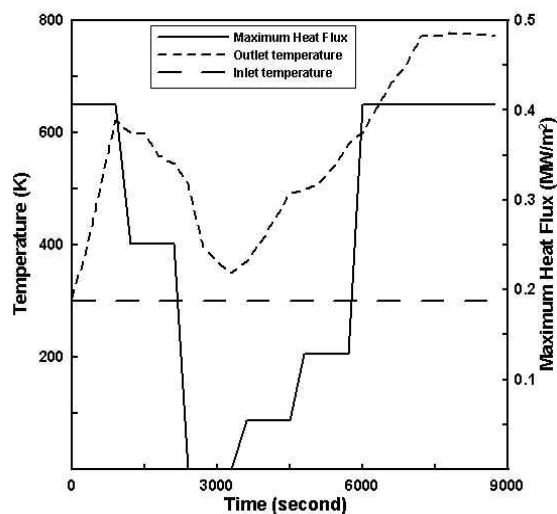


Figure 11: The outlet temperature variation depending upon the heat input variation

Table II is the change of the receiver outlet temperature. Fig. 11 shows the outlet temperature variation depending upon the input heat flux variation, which varies from 0 to 0.41 MW/m². From Fig. 11, it takes about 2200sec to get steady outlet temperature after the heat input becomes steady.

Table III Temperatures at the different radial locations along the axial direction of the receiver

L (mm)	P-1 (K)	P-2 (K)	P-3 (K)	P-4 (K)	P-5 (K)	P-6 (K)	P-7 (K)	P-8 (K)	P-9 (K)	P-10 (K)
Distance of center	0	5	10	15	20	25	30	35	40	45
(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)
7.5	911.1	1001.9	1001.9	1077.1	1222.1	1295.7	1569.9	1569.9	1197.3	1197.3
15.0	841.3	841.3	921.5	1002.3	1089.0	1089.0	1271.7	1422.6	950.3	672.4
22.5	970.4	970.4	999.2	995.1	995.1	1271.7	1271.7	1206.4	728.7	741.8
30.0	917.0	1025.4	1025.4	1071.1	1071.1	1112.7	1227.2	1217.7	926.6	926.6
37.5	705.0	929.5	1079.8	1122.6	1122.6	1174.7	1217.2	1173.7	1173.7	957.2

Fig. 12 shows the temperature distributions of different radial locations along the axial of direction of the receiver. Table III shows the numeric value of the temperature at different radial locations along the axial of direction of the receiver. The temperature decreases rapidly near the quartz window and increases rapidly when the air contacts the porous material. After that, the air temperature decreases gradually flowing through the porous material.

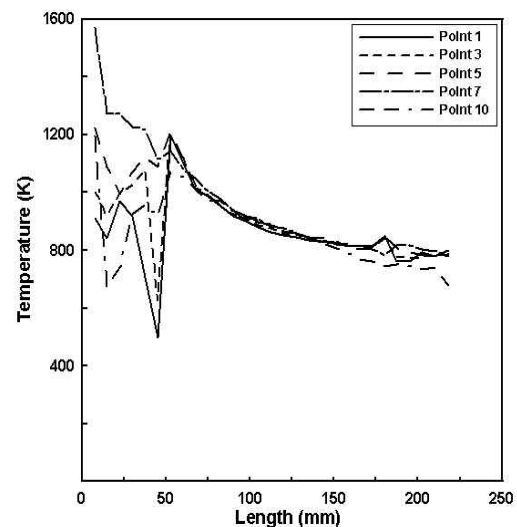


Figure 12 Temperature distribution on longitudinal axis of receiver

In the Table IV the experimental and numerical the outlet temperatures are compared when DNI is 705.6 MW/m². The minimum and maximum errors are 1.98% and 3.52%, respectively. Based on this comparison, it is shown that the numerical model can accurately estimate the temperature of the receiver.

45.0	498.4	486.6	625.9	783.3	1087.3	1087.3	1116.1	1046.3	954.8	916.3
52.5	1199.2	1197.8	1200.1	1203.3	1205.9	1206.2	1142.5	1102.3	1068.0	1071.0
60.0	1108.5	1069.0	1121.0	1097.6	1124.1	1124.9	1083.0	1095.4	1061.5	1053.9
67.5	1012.5	1026.2	1015.2	1045.2	1023.2	1045.4	1050.2	1045.3	1029.7	1010.7
75.0	988.0	988.0	992.6	993.5	991.8	995.8	1002.6	995.5	1010.1	975.9
82.5	954.2	946.2	952.2	945.5	954.4	971.3	978.7	975.6	968.4	968.4
90.0	916.8	916.4	924.5	924.1	940.4	938.5	937.7	954.6	941.5	936.5
97.5	900.6	901.4	906.9	900.7	901.6	915.7	918.1	918.1	927.5	914.2
105.0	878.7	878.1	886.8	891.9	898.2	896.4	907.5	904.2	902.5	902.3
112.5	862.2	863.7	875.2	875.2	886.8	879.8	884.9	884.6	888.3	886.0
120.0	851.7	853.1	858.3	864.1	865.7	867.6	874.9	869.9	873.1	870.4
127.5	845.7	848.8	854.5	855.6	859.3	859.9	860.3	861.3	866.2	857.6
135.0	834.2	834.1	840.2	846.4	843.0	855.1	845.6	850.6	845.9	842.7
142.5	832.4	832.4	835.6	835.6	841.2	834.1	828.1	840.2	834.3	823.7
150.0	823.2	823.6	823.6	821.6	835.2	829.7	828.1	831.0	821.2	810.5
157.5	815.8	815.9	815.9	821.6	814.4	821.0	821.0	813.8	797.9	791.5
165.0	814.1	813.4	813.4	813.4	814.4	814.4	804.0	805.7	785.8	770.6
172.5	814.1	818.9	811.0	811.0	811.8	804.0	804.0	793.2	777.6	762.0
180.0	847.5	832.5	838.9	838.9	850.2	813.0	785.4	764.3	764.3	746.2
187.5	760.8	760.8	777.5	777.5	808.7	815.8	815.8	780.6	780.6	752.1
195.0	760.8	760.8	773.0	773.0	793.1	805.9	815.8	784.6	771.6	743.5
202.5	792.8	765.7	776.0	781.8	781.8	804.1	804.1	781.0	758.0	735.6
210.0	777.7	785.8	785.8	775.5	791.9	791.9	796.5	779.3	754.4	739.6
217.5	799.4	791.4	782.7	779.5	779.5	791.0	785.2	778.8	753.4	683.4

Table IV Comparison of measured and calculated temperature

Time (second)	6000	6600	7200	7800	8400
DNI (W/m^2)	705.6	705.6	705.6	705.6	705.6
Outlet Temperature (Calculated, K)	600.9	690.0	771.5	777.1	774.9
Outlet Temperature (Measured, K)	581.8	715.2	754.4	761.7	759.5
Error rate (%)	3.17	3.52	2.26	1.98	1.98

5. CONCLUSION

A 3-D numerical model which includes transient heat transfer, natural convection, radiation heat transfer, and conduction is developed to analyze the transient thermal performance of a volumetric air receiver, in which porous material is inserted.

In conclusion, these results could be obtained.

1. When there is the porous material inside the receiver, the heat transfer rate between the air and the receiver is much greater than that of the receiver without the porous material.

2. Comparing the maximum heat flux and the minimum heat flux, the maximum heat flux takes more than time to heat the whole receiver up and continuously get the hot air from the outlet.
3. It takes about 2200sec to get steady outlet temperature after the heat input becomes steady.
4. The temperature decreases rapidly near the quartz window and increases rapidly when the air contacts the porous material. After that, the air temperature decreases gradually flowing through the porous material.
5. The experimental and numerical the outlet temperatures are compared when DNI is 705.6 MW/m^2 . The

minimum and maximum errors are 1.98% and 3.52%.

NOMENCLATOR

t	Time
x_i	Cartesian coordinate ($i=1, 2, 3$)
u_i	Absolute fluid velocity component in direction x_i
p	Piezo metric pressure
ρ	Density
τ_{ij}	Stress tensor components
s_m	Mass source
s_i	Momentum source components
$F_{h,j}$	Diffusional energy flux in direction x_j
s_h	Energy source
H	Total enthalpy
h	Static enthalpy

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