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Research Article

Novel Approach for Heat Transfer Characterization in EOR Steam Injection Wells

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Abstract: Steam injection into hydrocarbon reservoirs involves significant heat exchange between the wellbore fluid and its surroundings. During injection, the hot fluid loses heat to the cold surroundings, continuously as it moves down the borehole. The heat transfer process impacts well-integrity and, in turn, the ability of the well to perform its required function effectively and efficiently with regard to safety and environmental factors. During the design phase of a steam injection well, it is necessary to avoid risks and uncertainties and accurately plan the life cycle of wellbore. The present study aims to investigate the nature and predict the natural convection heat transfer coefficient in the annulus. The approach to model the natural convection heat transfer in this study is by analytical and numerical techniques. The annular space between the tubing and the casing was treated as a finite space bounded by walls and filled with fluid media (enclosures). Correlations for vertical enclosures were employed in the work. The flow field was modeled and simulated for numerical analysis, using ANSYS-FLUENT software package. Some boundary parameters have been defined by the user and fed to the software. The predicted values of Nusselt numbers from both analytical and numerical approaches were compared with those of previous experimental investigations. The results of the present study can be used for preliminary design calculations of steam injection wells to estimate rate of heat transfer from wells. This study also provides a novel baseline assessment for temperature related well-integrity problems in steam injection wells.

Keywords: Enhanced oil recovery, heat transfer, steam injection, well integrity

INTRODUCTION

Steam injection into hydrocarbon reservoirs through wells involves heat transfer processes. The injection well is a composite cylindrical wall. Radial heat transfer between wellbore fluid and the formation surrounding the well occurs by overcoming various resistances in series due to layers of different materials (Hasan and Kabir, 2002), as demonstrated in Fig. 1.

The major resistance is within the annular space between casing and tubing as a result of natural convection heat transfer inside the annular space between casing and tubing. Willhite (1967) proposed a method for the estimation of natural convective heat transfer in the annulus between the tubing and casing. As shown in Fig. 2, he treated the aforementioned space as a rectangular cavity, assuming that the effect of annulus curvature is negligible.

In his classic work, Will hite had adapted the correlation proposed by Dropkin and Sommerscales (1965). This is a modified version of the correlation first introduced by Globe and Dropkin (1959). The architecture of the system is a horizontal cavity heated from below as presented in Fig. 3 (Globe and Dropkin, 1959; Dropkin and Sommerscales, 1965). The correlations hold when the horizontal layer is sufficiently wide so that the effect of short vertical sides is minimal (Bejan, 2009). The steam injection well has a vertical geometry which does not satisfy the condition of Dropkin's system; therefore inclusion of the full value of the mentioned correlations often results in significant underestimation of wellbore fluid temperature (Hasan and Kabir, 2002). Natural convection heat transfer in enclosed spaces has been the subject of many experimental and numerical studies and numerous correlations for the Nusselt number exist. Simple power-law type relations in the form of $\mathrm{NU} = \mathrm{CR}_{a_\delta}{}^{n}$, where, C and n are constants and are sufficiently accurate, but they are usually applicable to a narrow range of Prandtl and Rayleigh numbers and small aspect ratios (Yunus, 2008). The relations that are more comprehensive are naturally more complex. In the case of steam injection well that is a tall vertical enclosure with high aspect ratio, accurate investigation through available correlations is imperative.

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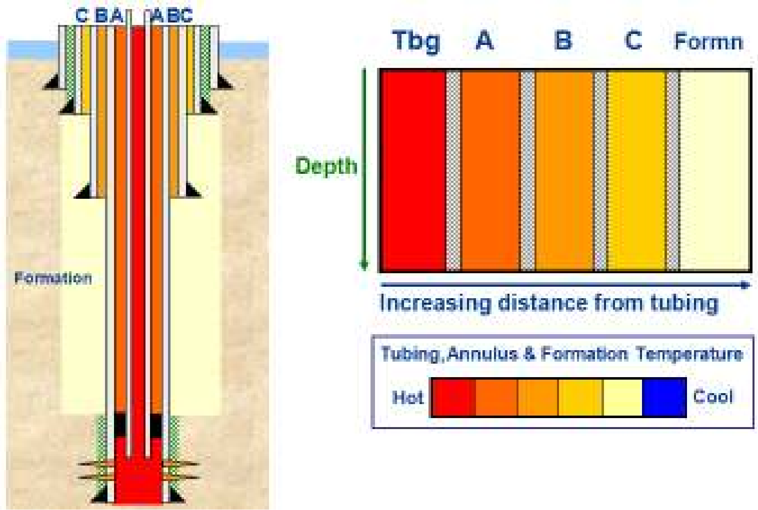

Fig. 1: Steam injection well architecture as a composite cylindrical wall

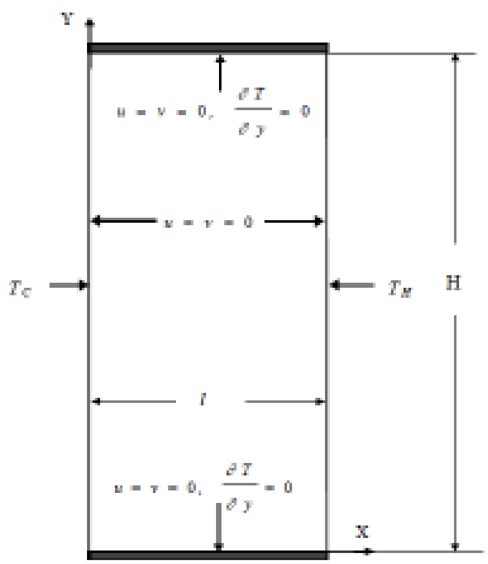

Fig. 2: Annular space between tubing and casing as a rectangular cavity

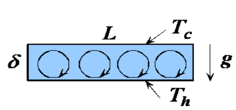

Fig. 3: Enclosure heated form below

In this study, the approach to model natural convection heat transfer is based on treating the annular space between the tubing and the casing as a finite space bounded by walls and filled with fluid media, i.e., an enclosure. Natural convection in such enclosures occurs as a result of buoyancy caused by a body force field with density variations within the field; convection in an enclosure is the result of the complex interaction between finite size fluid systems in thermal communication with all the walls that confine the enclosure. Correlations for inclined rectangular enclosures developed by Elsherbini *et al*. (1982) will be employed in the present study. This requires the assumption that the effect of curvature (cylinders) be negligible, based on the approach of Willhite (1967). In order to verify the results, the average Nusselt numbers as a function of Rayleigh number for the present study are compared with predictions using the FLUENT CFD package under similar geometrical conditions.

The aforementioned literature survey indicates that previous works have used inappropriate correlations for prediction of the system's behavior. In the present study, numerical and analytical results have been presented for a long vertical annulus, assumed as an eight meter long enclosure. The results from the available correlations for vertical rectangular enclosures have been compared with the CFD results to check the reliability and applicability of the correlations.

The intent of this study is to present a modified approach to modeling natural convection phenomena in the annular space between the tubing and casing in steam injection wells. This study shows that the proposed approach better captures the physics of heat transfer process for the examined conditions.

ANALYTICAL APPROACH

The phenomenon of natural convection heat transfer in an enclosure is dependent on the geometry and orientation of enclosure. Judging from the number of potential petroleum engineering applications, the enclosure phenomena can be divided into two large categories: enclosures heated from the side as demonstrated in Fig. 4 and enclosures heated from below, as shown in Fig. 3 (Bejan, 2009).

The fundamental difference between enclosures heated from the side, e.g., vertical wells and enclosures heated from below, e.g., horizontal or highly deviated wells, is that in the first one, a buoyancy-driven flow is present as soon as a very small temperature difference

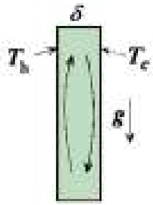

Fig. 4: Enclosure heated from the side

$(T_h - T_c)$ is imposed between the two sidewalls. By contrast, in enclosures heated from below, the imposed temperature difference must exceed a finite critical value before the first signs of fluid motion and convective heat transfer are detected. The dimensionless numbers that measure natural convection in an enclosure are Rayleigh and Prandtl numbers. The Rayleigh number is defined in Eq. (1):

$$R_{a_\delta} = \frac{\beta g\,(T_h - T_c)\,\delta^3}{\nu\,\alpha} \tag{1}$$

where,

- β = The thermal expansion coefficient of fluid inside the annulus
- g = Acceleration due to gravity
- T_h = Temperature on the hot side of the enclosure
- T_c = Temperature on the cold side of the enclosure
- δ = Distance in meters between two opposing walls of enclosures
- α = The thermal diffusivity of fluid inside the enclosure

The Prandtl number is as Eq. (2):

$$\mathrm{Pr} = \frac{\nu}{\alpha} \tag{2}$$

where,

- ν = The momentum diffusivity of fluid inside the enclosure
- α = The thermal diffusivity of the fluid

The Prandtl number provides a measure of the relative effectiveness of momentum and energy transport by diffusion in the velocity and thermal boundary layers, respectively. Willhite (1967) and Hasan and Kabir (1994) employed the correlation proposed by Dropkin and Sommerscales (1959) for natural convection heat transfer coefficient for fluids confined by two parallel plates. Their correlation for well geometry as a composite cylindrical system is:

$$h_c = \frac{0.049\,(R_{a_\delta})^{0.333}\ \mathrm{Pr}^{0.074}\ k_a}{r_{to}\ \ln(r_{ci}/r_{to})} \tag{3}$$

where, the term $0.049\,(R_{a_\delta})^{0.333}\ \mathrm{Pr}^{0.074}$ is the estimated Nusselt number for the annular space, k_a is thermal conductivity of the annular fluid at the average temperature and pressure of the annulus, r_{to} is the outside radius of tubing and r_{ci} is the inside radius of casing. According to the statement by Hasan and Kabir (2002), "inclusion of the full value of h_c calculated from Eq. (3) often leads to significant underestimation of wellbore fluid temperature". Equation (3) holds for the horizontal systems or when the horizontal layer is sufficiently wide so that the effect of the short vertical sides is minimal. Because of those conditions the values calculated from Eq. (3) will not capture the physics of field data. Elsherbini *et al*. (1982) developed correlations for the large-aspect-ratio vertical rectangular enclosures, as cited by Mills (2009):

$$\mathrm{Nu}_{L90} = \max\{\mathrm{Nu}_1, \mathrm{N}_{u2}, \mathrm{N}_{u3}\} \tag{4}$$

where,

$$\mathrm{N}_{u1} = 0.0605\ \mathrm{Ra}_\delta^{1/3} \tag{5}$$

$$\mathrm{N}_{u2} = \left\{1 + \left[\frac{0.104\ \mathrm{Ra}_\delta^{0.293}}{1+\left(\frac{6310}{\mathrm{Ra}_\delta}\right)^{1.36}}\right]^3\right\}^{1/3} \tag{6}$$

$$\mathrm{N}_{u3} = 0.242\left(\frac{\mathrm{Ra}_\delta}{H/\delta}\right)^{0.272} \tag{7}$$

is valid for $10^3 < \mathrm{Ra}_\delta < 10^7$ while for $\mathrm{Ra}_\delta \leq 10^3$, $\mathrm{Nu}_{L90} \cong 1$.

NUMERICAL SIMULATION

The concentric vertical annulus with outside tubing radius of 0.037 m, inside casing radius of 0.079 m and total well depth of 1,632 m was modeled for two-dimensional flow using GAMBIT software. The well was segmented into NL segments; each segment has a height of 8.0 m. The surface temperatures of each segment were considered as an average of the temperature variation along ΔL. The temperature variation along the depth, L was adopted from the study of Sagar *et al*. (1991). Both the flow and thermal behavior within the fluid field were solved numerically by computational simulation using FLUENT software. The buoyancy driven flow in the annulus was simulated as laminar since Rayleigh number is less than 109, where the applicable correlations are limited to this Rayleigh number range. The Boussinesq approximation for steady laminar flow was employed in the simulation (Gray and Giorgini, 1976). In the simulation, the conservation and the state equations were solved numerically for continuity, momentum and energy.

The following assumptions are totally true and applicable for the large aspect ratio annular flow:

- The variation in the peripheral direction is negligible, $\frac{\partial}{\partial\theta} = 0$

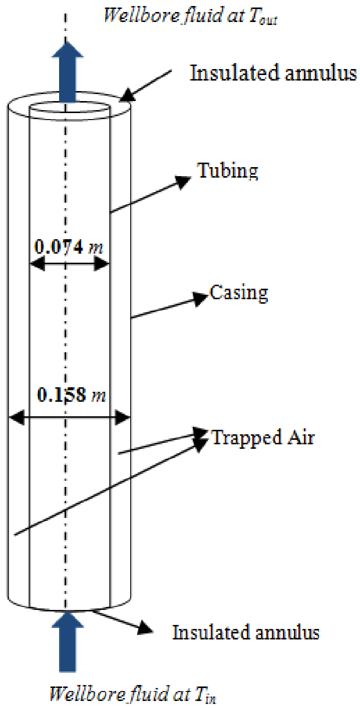

Fig. 5: Model diagram of the concentric annulus under study

Table 1: Boundary conditions of the wellbore

Component	Boundary type	Value
Tubing	Wall	As in Table 2 for each segment
Casing	Wall	As in Table 2 for each segment
Top insulator	Wall	$q = 0$ W/m^2
Bottom insulator	Wall	$q = 0$ W/m^2
Fluid	Air	Boussinesq approximation (varying density)

- There is no velocity component in the peripheral direction, $V\theta = 0$
- The flow is steady, $\frac{\partial}{\partial t} = 0$; Newtonian with constant viscosity and compressible

Hence, the governing equations in z and r directions are:
Continuity equation:

$$\frac{1}{r}\frac{\partial}{\partial r}(\rho r v_r) + \frac{\partial}{\partial z}(\rho v_z) = 0 \tag{8}$$

Momentum equation in r and z directions:
For z-direction:

$$V_r \frac{\partial}{\partial r}(\rho V_z) + V_z \frac{\partial}{\partial z}(\rho V_z) = -\frac{\partial p}{\partial z} + \mu\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r V_z)\right) + \frac{\partial^2 V_z}{\partial z^2}\right] + \rho g_z \tag{9}$$

For r-direction:

$$v_r \frac{\partial}{\partial r}(\rho v_r) + v_z \frac{\partial}{\partial z}(\rho v_r) = -\frac{\partial p}{\partial r} + \mu\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r v_r)\right) + \frac{\partial^2 v_r}{\partial z^2}\right] + \rho g_r \tag{10}$$

Energy equation in r and z direction:

$$C_p\left[v_r \frac{\partial}{\partial r}(T\rho) + v_z \frac{\partial}{\partial z}(T\rho)\right] = \frac{1}{r}\frac{\partial}{\partial r}\left(r k \frac{\partial T}{\partial r}\right) + \frac{\partial\left(k\frac{\partial T}{\partial z}\right)}{\partial z} + \varphi_{vis} \tag{11}$$

where, the viscous term, φ_{vis} is:

$$2\mu\left[\left(\frac{\partial V_r}{\partial r}\right)^2 + \left(\frac{V_r}{r}\right)^2 + \left(\frac{\partial V_z}{\partial z}\right)^2\right] + \mu\left(\frac{\partial V_z}{\partial r} + \frac{\partial V_r}{\partial z}\right)^2 \tag{12}$$

Since the flow is compressible, the state equation must be adopted in the model to relate the density change to the pressure and temperature:

$$\rho = \frac{p}{RT} \tag{13}$$

The system can be shown diagrammatically as in Fig. 5 and the conditions for the modeling are shown in Table 1.

The FLUENT 5/6 V6.3.26 package was used to solve the governing equations using SIMPLEC algorithm. Different 8 m segments with different temperatures were studied. The mesh generated for each 8 m segment is 80,000 structured cells with the boundary conditions shown in Table 1 and a no-slip condition for velocity and temperature on the walls. All values are constant over the respective component of the boundary and the system is assumed to be under isothermal steady-state condition.

RESULTS AND DISCUSSION

Data reported by Sagar *et al.* (1991) from a 1632-m-deep flowing well were used to validate the Nusselt number calculation procedure. Table 2 reports the tubing outside and casing inside temperature at different depths.

Results of the analytical model: Figure 6 was drawn based on data in Table 2 and 3. It is shown that Nusselt number started at 2.30 on the surface of the wellbore and dropped gradually throughout the wellbore.

Figure 7 was drawn based on data in Table 2 and selection of min $\{N_{u1}, N_{u2}, N_{u3}\}$ as the average Nusselt number, i.e., the novel method proposed by the present study. It is illustrated that Nusselt number started at 1.06 on the surface of the wellbore and dropped gradually throughout the wellbore; the Nusselt number is much smaller at the bottom of wellbore than at the wellhead. The marked variation in Nusselt number with well depth is a direct result of the small temperature

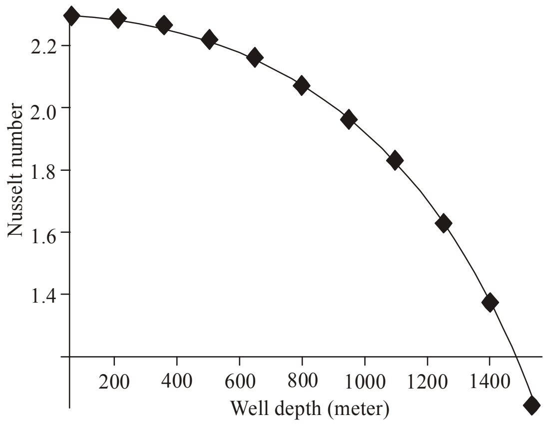

Fig. 6: Nusselt number values based on data in Table 2 and 3

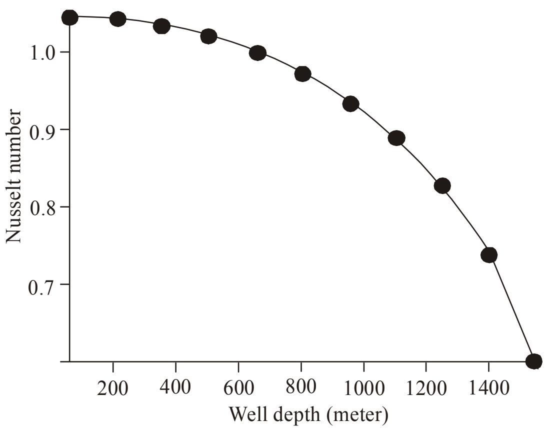

Fig. 7: Annulus nusselt number vs. well depth based on the present study

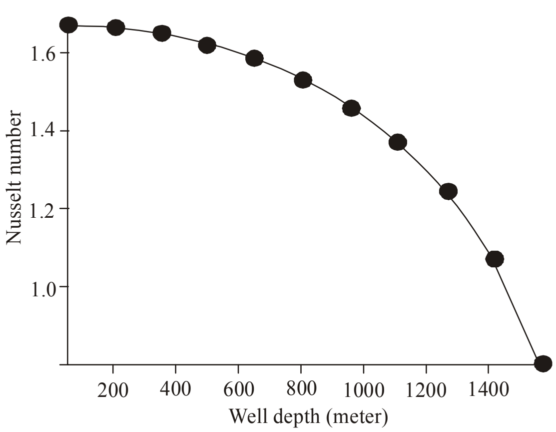

Fig. 8: Annulus nusselt number vs. well depth using classic correlations in petroleum industry Eq. (3)

difference between the tubing outside temperature and the casing inside temperature near the bottom hole. This leads to a smaller value of Nusselt number at the bottom of well.

Figure 8 was drawn based on Eq. (3), i.e., the classic correlations which are used in the petroleum industry and data in Table 2.

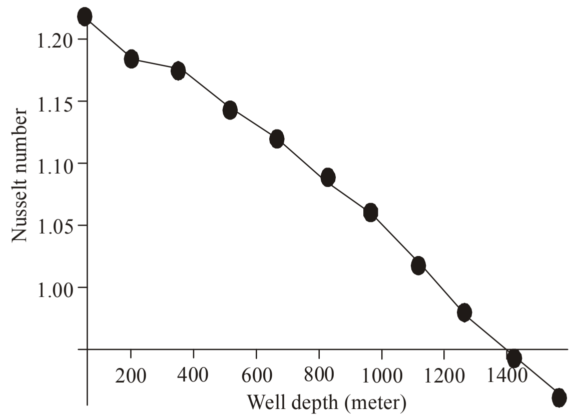

Fig. 9: Predicted nusselt values of annulus fluid, by CFD simulation

Table 2: Tubing outside and casing inside temperatures (Sagar *et al*., 1991)

Well depth (m)	Average tubing temperature (K)	Average casing temperature (K)	Average annulus air temp. (K)
0.0	304.111	297.444	300.778
152.4	306.889	299.111	303.000
304.8	308.556	300.778	304.667
457.2	309.667	302.444	306.056
609.6	310.778	304.111	307.444
762.0	311.889	305.778	308.833
914.4	313.000	307.444	310.222
1066.8	313.556	309.111	311.333
1219.2	314.111	310.778	312.444
1371.6	314.667	312.444	313.556
1524.0	315.222	314.111	314.667

Table 4: Nusselt number results obtained from the analytical approach

Well depth (m)	Nu, as max $\{Nu_1, N_{u2}, N_{u3}\}$	Nu, as Min $\{Nu_1, N_{u2}, N_{u3}\}$	Nu from Eq. (3)
0.0	2.289	1.062	1.671
152.4	2.283	1.060	1.667
304.8	2.259	1.052	1.651
457.2	2.218	1.037	1.623
609.6	2.159	1.016	1.583
762.0	2.077	0.988	1.529
914.4	1.969	0.951	1.459
1066.8	1.826	0.902	1.369
1219.2	1.629	0.837	1.249
1371.6	1.379	0.744	1.081
1524.0	1.040	0.580	0.797

Table 3: Nusselt number results obtained from the CFD simulation

Well depth (m)	Average annulus air temp. (K)	Heat transfer across annulus (W/m^2)	Nusselt number, Nu
0.0	300.778	1.793	1.232
152.4	303.000	2.047	1.196
304.8	304.667	2.041	1.189
457.2	306.056	1.852	1.158
609.6	307.444	1.675	1.129
762.0	308.833	1.493	1.094
914.4	310.222	1.330	1.067
1066.8	311.333	1.022	1.022
1219.2	312.444	0.738	0.982
1371.6	313.556	0.475	0.946
1524.0	314.667	0.230	0.911

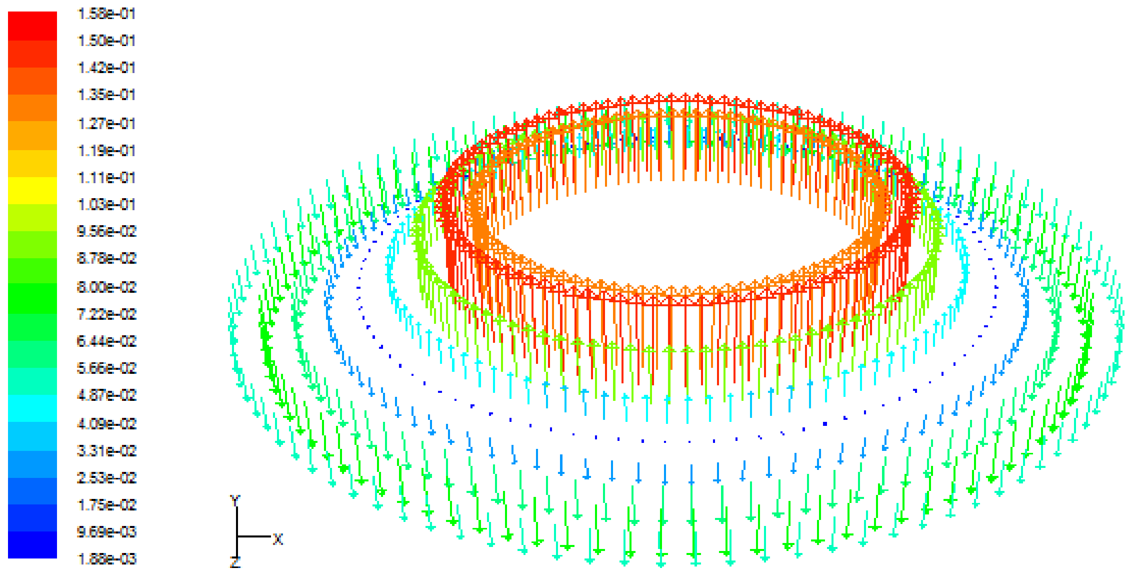

Fig. 10: Velocity vector showing the flow direction along the depth of the well

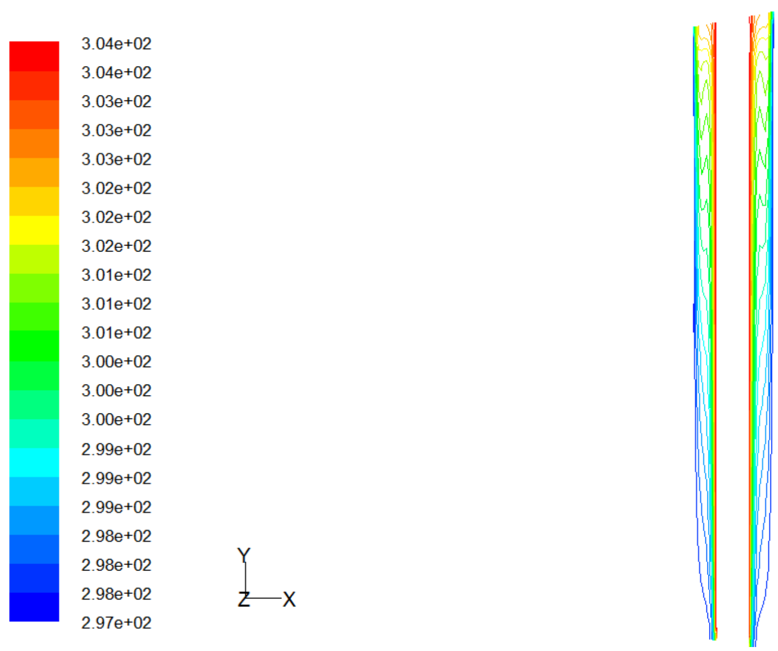

Fig. 11: Isotherm contours of a vertical annulus

Here Nusselt number started at around 1.67 on the surface of the wellbore and dropped throughout the wellbore.

Table 4 summarizes the results of the analytical approach. The first set of Nusselt numbers is based on Eq. (4), the second set is based on selecting the min $\{Nu_1, N_{u2}, N_{u3}\}$ and the final set on the right is based on Eq. (3), i.e., the classic approach in the petroleum industry.

Results of the simulation: The FLUENT software package was used to determine heat transfer across annulus, provide information on the streamline and isotherm contours. The Nusselt number, Nu, of the air at different segments of the system was calculated at the segment average temperature; Results are shown below.

Figure 9 was drawn based on data in Table 3. Here Nusselt number started at around 1.23 on the surface of the wellbore and dropped throughout the wellbore.

Predicted velocity vector field for an 8.0 m wellbore segment is presented in Fig. 10. These results show that the flow around the annulus is symmetrical. The movement of the fluid inside the annulus is due to

Table 5: Summary of nusselt results obtained from analytical and numerical analysis

Well depth (m)	Average annulus air temp. (K)	Nusselt number, max $\{N_{u1}, N_{u2}, N_{u3}\}$	Nusselt number, min $\{N_{u1}, N_{u2}, N_{u3}\}$	Nusselt number, petroleum engineering Eq. (3)	Nusselt number, CFD results
0.0	300.778	2.289	1.062	1.671	1.232
152.4	303.000	2.283	1.060	1.667	1.196
304.8	304.667	2.259	1.052	1.651	1.189
457.2	306.056	2.218	1.037	1.623	1.158
609.6	307.444	2.159	1.016	1.583	1.129
762.0	308.833	2.077	0.988	1.529	1.094
914.4	310.222	1.969	0.951	1.459	1.067
1066.8	311.333	1.826	0.902	1.369	1.022
1219.2	312.444	1.629	0.837	1.249	0.982
1371.6	313.556	1.379	0.744	1.081	0.946
1524.0	314.667	1.040	0.580	0.797	0.911

the temperature gradient. The fluid close to the inner hot surface (tubing) has lower density than that near the outer cold surface, i.e., casing. Thus, the fluid near the inner surface moves upward while the relatively heavy fluid near the casing moves downward.

Figure 11 presents isotherm contours for the wellbore segment. Isotherms indicate that the heat transfer regime is convection.

Discussion: Correlation equations Eq. (3), (5), (6) and (7) have been used to perform the analytical approach. The FLUENT CFD package was used for predicting the Nusselt number at the same condition as the analytical work and to provide information on streamline contours and isotherm contours, which were not obtain analytically. This additional detailed information is imperative to understand and explain the natural convection phenomena in the annulus. In order to verify the analytical results, the Nusselt numbers obtained analytically have been compared with CFD FLUENT results and this shows a reasonable agreement with the present study, i.e., $\min\{N_{u1}, N_{u2}, N_{u3}\}$ as Nusselt number.

Table 5 shows the Nusselt numbers calculated from different approaches.

CONCLUSION

Natural convection heat transfer in tubing-casing annulus of vertical steam injection wells has been investigated.

Natural convection in such a system, i.e., enclosures, occurs as a result of buoyancy caused by a body force field with density variations within the annulus field. Results are presented for 8.0 m high wellbore segments from different intervals, i.e., different temperatures. Analytical results of the Nusselt number for a vertical wellbore segment are in close agreement with results from the FLUENT CFD package. The main conclusions from the present study can be summarized as follows:

- The average Nusselt number decreased with increasing well depth.
- The temperature difference between the tubing outside and casing inside has a significant effect on the Nusselt number.
- The proposed approach of this study, i.e., selection of $\min\{N_{u1}, N_{u2}, N_{u3}\}$ as the Nusselt number, better captures the physics of the heat transfer.

The above analytical and numerical work has resulted in new functional correlation equations which can be used for natural convection heat transfer calculations in the well tubing-casing annulus. The functional correlations cover a wide range of well architecture, i.e., different tubing and casing size. In terms of combined accuracy and continuity, these correlations offer advantages in certain applications over those previously employed. Moreover; these correlations can be used for preliminary design calculation of HPHT wells to calculate the rate of natural convection heat transfer across the annular space between the tubing and casing.

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