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Developing Novel Wellbore Heat Transfer Strategies for HPHT Wells

Mohd Amin Shoushtari, SPE, Universiti Teknologi PETRONAS (UTP); Hussain H. Al-Kayiem, UTP; Sonny Irawan, UTP; Ismail B. Mohd Saaid, UTP

Abstract

Oil and Gas production from HTHP reservoirs involves significant heat exchange between the wellbore fluid and its surroundings. During production, the hot fluid loses heat to the cold surroundings, continuously as it moves up 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.

One often overlooked heat transfer effect is the increase in annulus pressure resulting from annular fluid thermal expansion, which can result in tubing collapse or casing burst. Damage due to high annular pressure can also dramatically affect well-integrity.

During the design phase of a HPHT well, it is necessary to avoid risks and uncertainties and accurately plan the life cycle.

The present work 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 work 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 study. The flow field was modeled and simulated for numerical analysis, using ANSYS-FLUENT – 12 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 work can be used for preliminary design calculations of HPHT wells to estimate rate of heat transfer from HPHT wells. This work provides a novel baseline assessment for temperature related well-integrity problems in HPHT wells.

Introduction

Oil and gas production from hydrocarbon reservoirs through wells involves heat transfer processes. The oil well is a composite cylindrical wall. Radial heat transfer between wellbore fluid and the formation surrounding the oil well occurs by overcoming various resistances in series [1] due to layers of different materials, as demonstrated in Fig.1.
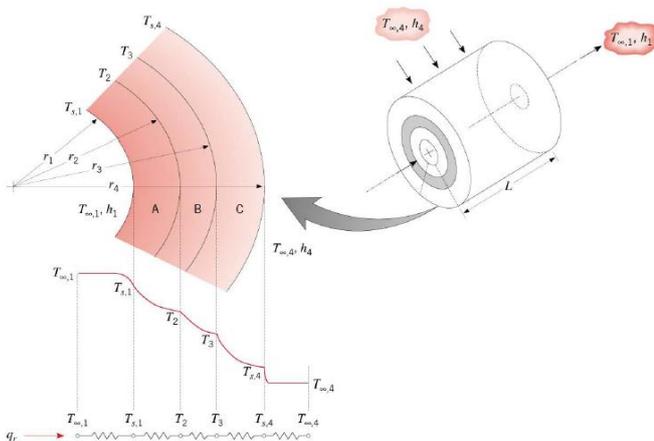

Figure1.Oil well architecture as a composite cylindrical wall

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. In 1967, Willhite [2] 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, Willhite [2] had adapted the correlation proposed by Dropkin and Sommerscales [3] which is a modified version of the correlation first introduced by Globe and Dropkin in 1959[4]. The architecture of the system in Ref. [3] and Ref. [4] is a horizontal cavity heated from below as presented in Fig.4. The correlations hold when the horizontal layer is sufficiently wide so that the effect of short vertical sides is minimal [5].
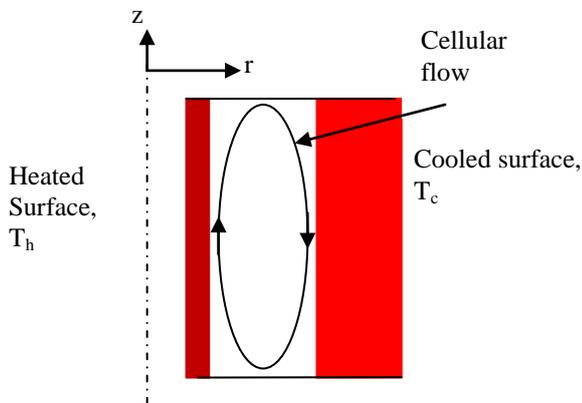

Figure 2. Annular space between tubing and casing as a rectangular cavity

The oil 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 [1]. 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 $NU = C Ra_{\delta}^{n}$, 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 [6]. The relations that are more comprehensive are naturally more complex. In the case of oil well that is a tall vertical enclosure with high aspect ratio, accurate investigation through available correlations is imperative.

In this work, 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 ElSherbiny et al. [7] 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 [2]. In order to verify the results, the average Nusselt numbers as a function of Rayleigh number for the present work 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 HPHT wells. This work 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 [5]**:** enclosures heated from the side as demonstrated in Fig.3, and enclosures heated from below, as
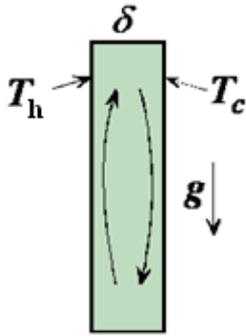
shown in Fig.4.

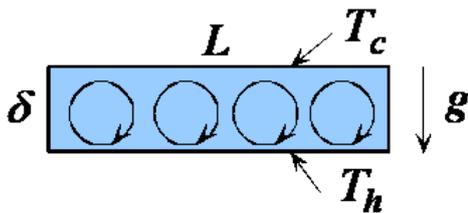**Figure 3. Enclosure heated from the side**

Figure 4. Enclosure heated form below

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

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

Where β is the thermal expansion coefficient of fluid inside the annulus, g is acceleration due to gravity, T_h is temperature on the hot side of the enclosure, T_c is temperature on the cold side of the enclosure, δ is distance in meters between two opposing walls of enclosures, and α is the thermal diffusivity of fluid inside the enclosure.
The Prandtl number is

$$\text{Pr} = \frac{v}{\alpha} \tag{2}$$

Where v is the momentum diffusivity of fluid inside the enclosure, and α is 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.

Wilhite [2] and Hassan and Kabir [9] employed the correlation proposed by Dropkin and Sommerscales [3] for natural convection heat transfer coefficient for fluids confined by two parallel plates. Their correlation for oil well geometry as a composite cylindrical system is:

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

Where the term $0.049 \, (R_{a_\delta})^{0.333} \, Pr^{0.074}$ 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 Hassan and Kabir [1], " inclusion of the full value of h_c calculated from (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 (3) will not capture the physics of field data.

In 1982 ElSherbiny et al. [7] developed correlations for the large-aspect-ratio vertical rectangular enclosures, as cited by Mills [10]:

$$Nu_{L90} = max\{Nu_1, N_{u2}, N_{u3}\} \tag{4}$$

Where

$$N_{u1} = 0.0605 \, Ra_\delta^{1/3} \tag{5}$$

$$N_{u2} = \left\{ 1 + \left[\frac{0.104 \, Ra_\delta^{0.293}}{1 + (\frac{6310}{Ra_\delta})^{1.36}} \right]^3 \right\}^{1/3} \tag{6}$$

$$N_{u3} = 0.242 \left(\frac{Ra_\delta}{H/\delta} \right)^{0.272} \tag{7}$$

Valid for $10^3 < Ra_\delta < 10^7$; while for $Ra_\delta \leq 10^3$, $Nu_{L90} \cong 1$.

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 [8]. 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 [11] for steady laminar flow was employed in the simulation. 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, $\frac{\partial}{\partial \theta} = 0$

- 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:

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

Momentum equation in r and z directions**:**

For *z* direction,

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

For *r* direction,

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

Energy equation in r and z directions:

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

Where, the viscous dissipation term, φ_{vis} is:

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

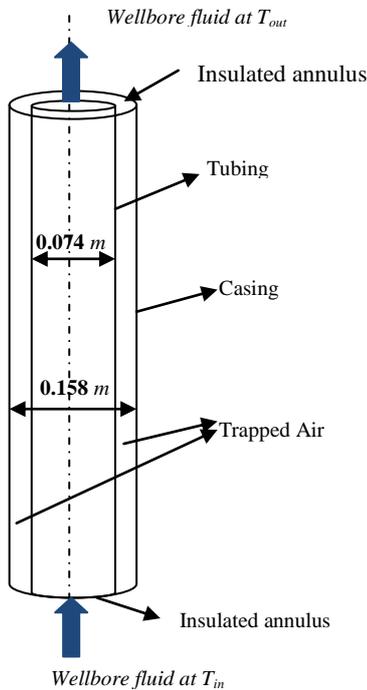The system can be shown diagrammatically as in Fig. 5, and the conditions for the modeling are shown in Table 1.

Figure 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)

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 Discussions

Data reported by Sagar et al. [8] from a 1632-meter-deep flowing oil well were used to validate the Nusselt number calculation procedure. Table2 reports the tubing outside and casing inside temperature at different depths.

TABLE 2. TUBING OUTSIDE AND CASING INSIDE TEMPERATURES [8]

Well Depth (m)	Average Tubing Temperature (K)	Average Casing Temperature (K)	Average Annulus Air Temp. (K)
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	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	315.222	314.111	314.667

i. ***Results of the Analytical Model***

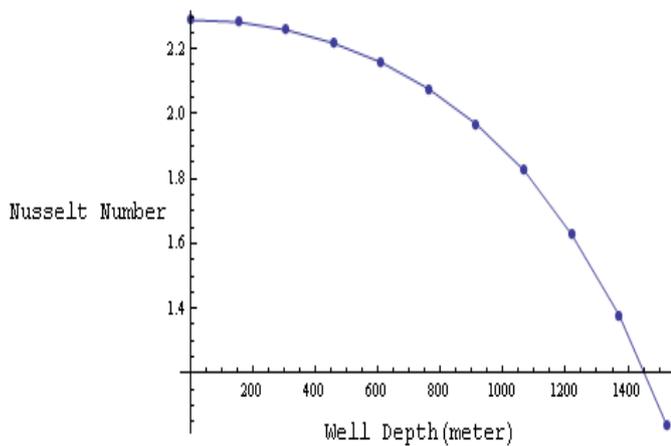Fig. 6 was drawn based on data in Table 2 and (4).

Figure.6 Nu values based on data in table 2 and (4)

Fig.6 shows that Nusselt number started at 2.30 on the surface of the wellbore and dropped gradually throughout the wellbore.

Fig.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 work.
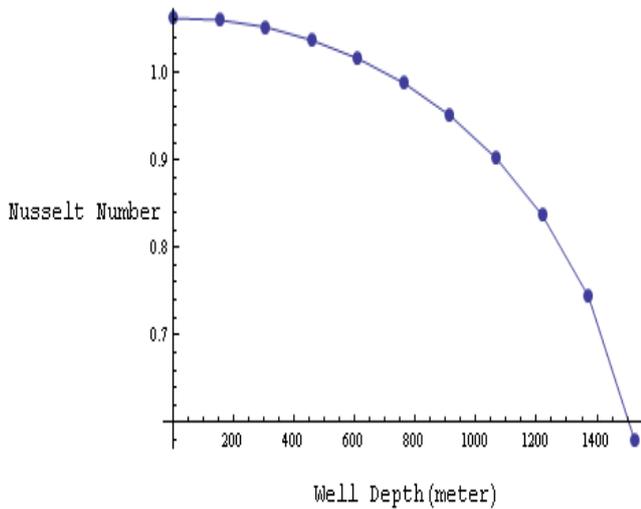

Figure 7. Annulus Nusselt number vs. well depth based on the present work

Fig.7 illustrates 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 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 oil well.

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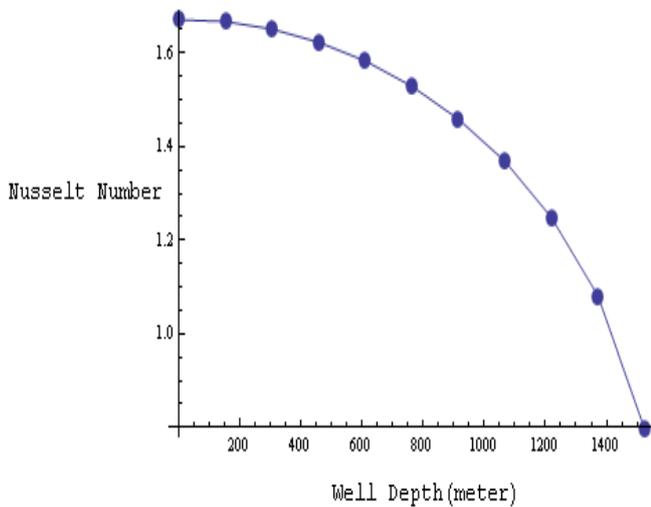

Figure 8. Annulus Nusselt number vs. well depth using classic correlations in petroleum industry (3)

Here Nusselt number started at around 1.67 on the surface of the wellbore and dropped throughout the wellbore.
Table 3 summarizes the results of the analytical approach. The first set of Nusselt numbers is based on (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 (3), i.e. the classic approach in the petroleum industry.

TABLE 3. NU RESULTS OBTAINED FROM THE ANALYTICAL APPROACH

Well Depth (m)	Nu, as max $\{Nu_1, Nu_2, Nu_3\}$	Nu, as min $\{Nu_1, Nu_2, Nu_3\}$	Nu from (3)
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	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	1.040	0.580	0.797

ii.   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.

TABLE 4. NU RESULTS OBTAINED FROM THE CFD SIMULATION

Well Depth (m)	Average Annulus Air Temp. (K)	Heat transfer across annulus (w/m²)	Nusselt Number, Nu
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	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	314.667	0.230	0.911

Fig.9 was drawn based on data in Table 4. Here Nusselt number started at around 1.23 on the surface of the wellbore and dropped throughout the wellbore.
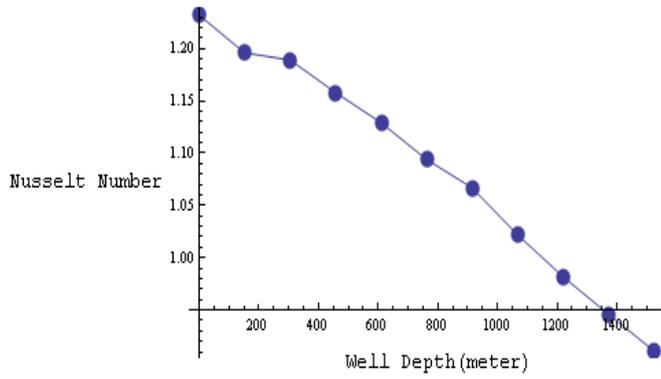
Figure 9. Predicted Nusselt values of annulus fluid, by CFD simulation

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 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.
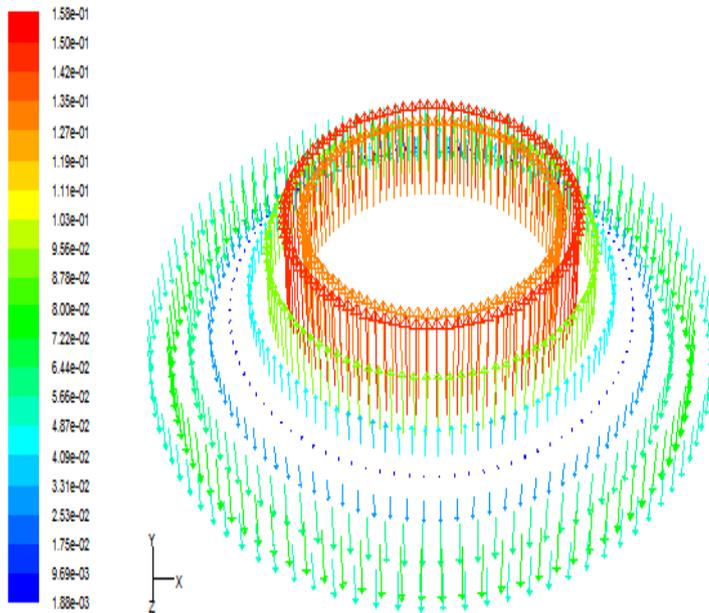

Figure 10. Velocity vector showing the flow direction along the depth of the well

Fig. 11 presents isotherm contours for the wellbore segment. Isotherms indicate that the heat transfer regime is convection.
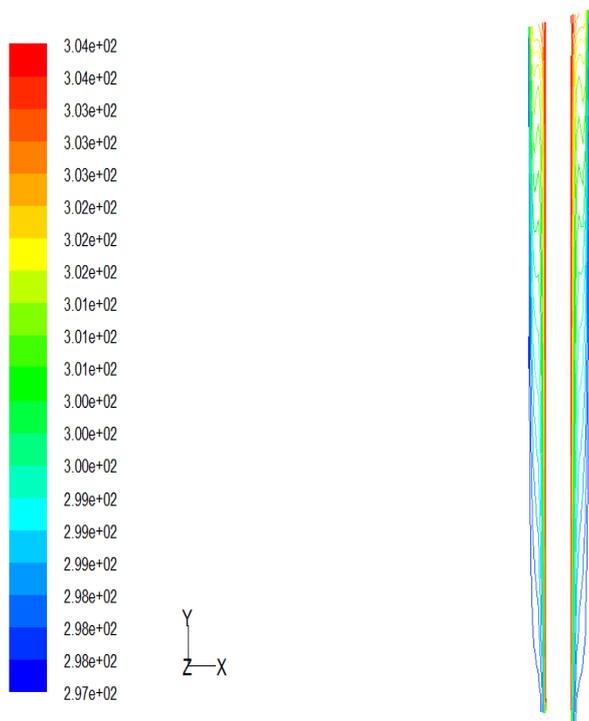
Figure 11. Isotherm contours of a vertical annulus

iii.   *Discussions*

Correlation equations (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 work, i.e. $\min\{N_{u1}, N_{u2}, N_{u3}\}$ as Nusselt number.

Table 5 shows the Nusselt numbers calculated from different approaches.

TABLE 5. SUMMARY OF NU 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 (3)	Nusselt Number, CFD Results
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	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	314.667	1.040	0.580	0.797	0.911

Conclusions

Natural convection heat transfer in tubing-casing annulus of vertical HPHT 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 work can be summarized as follows:

1. The average Nusselt number decreased with increasing well depth.
2. The temperature difference between the tubing outside and casing inside has a significant effect on the Nusselt number.
3. The proposed approach of this work, 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 oil well tubing-casing annulus. The functional correlations cover a wide range of oil 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.

Acknowledgement

References

[1] A.R. Hasan and C.S. Kabir, Fluid Flow and Heat Transfer in Wellbores, SPE, Richardson, Texas, 2002, pp. 151.

[2] Willhite, G.P., "Over-all heat transfer coefficients in steam and hot water injection wells" Paper SPE 1449-PA, Journal of Petroleum Technology, Vol. 19, NO. 5, pp. 607-615(May 1967).

[3] D. Dropkin and E. Sommerscales, "Heat Transfer by Natural Convection in Liquids Confined by Two Parallel Plates Inclined at Various Angles with Respect to the Horizontal", J. Heat Transfer; Trans., ASME, Series C (Feb., 1965) 87, 77-84.

[4] S. Globe and D. Dropkin, Natural Convection Heat Transfer in Liquids Confined by Two Horizontal Plates and Heated from Below, J. Heat Transfer, Vol. 81, 1959, pp. 24-28.

[5] A. Bejan, Convection Heat Transfer, 3rd edition, Wiely, USA, 2004, PP.243.

[6] A. Yunus, Heat and Mass Transfer, 3rd edition, Mc Graw Hill, USA, 2006, PP.522-523.

[7] Elsherbini, S. M., Raithby, G. D., and Hollands, K. G. T., "Heat transfer by natural convection across vertical and inclined air layers," J. Heat Transfer , 104, 96-102(1982).

[8] R.K.Sagar, D.R. Doty,and Z. Schmidt, "Predicting Temperature Profiles in a Flowing Well," SPEPE(Nov. 1991),pp.441.

[9] A.R.Hasan and C.S. Kabir, "Aspects of Heat Transfer During Two-Phase Flow in Wellbores," SPEPF (August 1994) pp.211.

[10] A.F. Mills, Heat Transfer, 3rd edition, Prentice Hall, USA, 1999, PP.335.

[11] Gray, D.D. and Giorgini, A., The Validity of the Boussinesq Approximation for Liquids and Gases, Int. J. Mass Transfer, 1976, 1.