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Heat Transfer in Tubing-Casing Annulus during Production Process of Geothermal Systems

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ABSTRACT: In geothermal systems, the temperature distribution of heat flow in the wellbore is dependent on the well structure and the geological conditions of the surrounding formation. Understanding of heat transfer in the tubing-casing annulus can reduce the heat losses of wellbore fluid during the production process. The present study discusses the possible means of heat transfer in the annulus, and develops a piecewise equation for estimating the convective heat transfer coefficient with a wider valid condition of $0<Ra<7.17\times10^8$. By converting the radiation and natural convection into equivalent thermal conduction, their sum is defined as a total thermal conductivity to describe the heat transfer in the annulus. The results indicate that the annulus filled with gas can be utilized as a good thermal barrier for the fluid in the wellbore. Additionally, the contribution of radiation will increase to occupy a majority proportion in the total thermal conductivity when the annular size increases and the materials have high emissivity. Otherwise, thermal radiation is just the second factor.
KEY WORDS: heat transfer, tubing-casing annulus, Rayleigh number, natural convection, geothermal system, thermal conductivity, radiation.

0 INTRODUCTION

With increases in depth, technological issues are encountered in geothermal exploration, drilling, production and injection owing to the high-temperature, high-pressure, and deep geological and hydrogeological conditions. During production, the wellbore fluid continues to lose heat to the increasingly cold surroundings, as it ascends the borehole over such a long distance (Hasan and Kabir, 2002). For example, Kanev et al. (1997) described a deep geothermal well with a depth of 3 382 m in which the fluid temperature decreased from 301 °C at the bottom to 255 °C at the wellhead after 100 days production when the mass flow was 150 t/h; Tóth (2006) measured a well with a depth of 2 930 m where the wellhead temperature was 136 °C, which was 5 °C lower than the bottom temperature when the fluid was extracted after 100 days with a flow rate of 50 kg/s. The temperature losses play as an important role on the wellbore system (Akpan, 2014; Gorman et al., 2014; Wu et al., 2014; Zhou and Zhang, 2013; Tekin and Akin, 2011). Tubing is always set down the borehole and under the dynamic water level for piping fluid. Afterwards, induced methods, such as pumping, N_2 injection, air compression or foaming agent are used to propel the fluid blowing out from the tubing (Grant and Bixley, 2011). Therefore, an annular space is used between the tubing and casing from the wellhead to the water level. The annulus affects the fluid temperature distribution during the production process in geothermal systems including the hydrothermal system (HS) and enhanced geothermal system (EGS) (Gallup, 2009). Occasionally, the annulus is sealed with a packer at the end of the tubing to separate the different reservoirs. Figure 1 shows a universal geothermal wellbore structure.

In the process of fluid rising up to the surface, the fluid diffuses heat to the cold surrounding formation through the annulus between the tubing and casing. Heat losses of the fluid may influence significantly the design of surface facilities to generate electricity, as well as for other direct uses (Kanev et al., 1997). Heat transfer in the annulus depends on the annular size and the filling materials in the space. Generally, the space

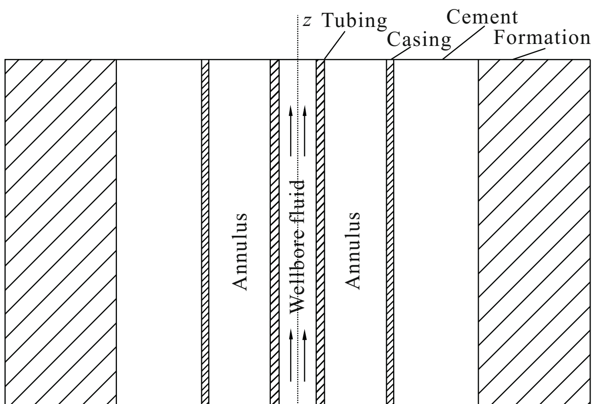

Figure 1. Wellbore structure of geothermal systems.

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is filled with gas, liquid, mud (Durucan and Olcenoglu, 1970), or foamy cement (Tóth, 2006). Vacuum is possible under special circumstances. Means of heat transfer may include thermal conduction, thermal radiation, natural convection, or the combinations. Natural convection in the annulus is generally difficult to analyze. Unfortunately, investigations of the heat transfer in the annulus have seldom been done. Fishenden and Saunders (1950) presented a correlation for the annulus Nusselt number (Nu) in a graphical form. Ramey (1962) established a mathematical model for the heat transfer in the wellbore, and defined an overall heat transfer coefficient including the heat resistance of the annulus. Dropkin and Somercales (1965) proposed an equation for the heat transfer coefficient with the Grashof number (Gr) for the natural convection of fluid between two vertical plates and gave a valid scale of Rayleigh number (Ra). Sheriff (1966) expressed the Nu number utilizing a laboratory experiment of CO_2 flowing in a short vertical annulus. Willhite (1967) and Willhite et al. (1967) suggested simplified calculation procedures for determining the overall heat transfer coefficient. However, Hasan and Kabir (2002) pointed out that the direct application of either the Fishenden and Saunders correlation or the Dropkin and Somercales correlation developed from short tubes in the laboratories was doubtful for wells thousands of meters in depth. Tóth and Bobok (2008) developed the theory of Kays and Leung (1963) and applied it in the annulus of a deep borehole heat exchange system. Yang et al. (2008) considered the influence of convection and used the compensation factor method to guarantee continuous temperature in the annulus. Tang et al. (2010) considered the relationships between natural convection and the overall heat transfer coefficient with depth in offshore high-temperature oil wells, and proposed a method to evaluate the heat transfer coefficient of the annulus at various depths by an iteration method.

Although these studies have contributed to the research on the heat transfer in the annulus, the underlying heat transfer mechanism is still not clear and in need of further investigation. The purpose of this paper is to highlight the impact of radiation and natural convection on the heat transfer in the annulus, where the heat losses from fluid may be reduced feasibly and effectively. Additionally, this paper tries to look for another method of better describing the heat transfer in the annulus, which may simplify the analysis and calculation.

1 HEAT TRANSFER IN THE ANNULUS

1.1 Thermal Conduction

Thermal conduction necessarily occurs in the annulus, whatever it is filled with, unless there is no temperature difference between the tubing outside and the casing inside. In a cylindrical coordinate, the differential equation of thermal conduction can be expressed by Fourier's Law (Yang and Tao, 1998)

$$\rho c_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}(\lambda r \frac{\partial T}{\partial r}) + \frac{1}{r^2}\frac{\partial}{\partial \varphi}(\lambda \frac{\partial T}{\partial \varphi}) + \frac{\partial}{\partial z}(\lambda \frac{\partial T}{\partial z}) + S \quad (1)$$

The left item is the increment of internal energy (unstable item); ρ, c_p, T, and t denote the density, [kg/m^3], heat capacity, [kJ/(kg·K)], temperature, [°C], and time, [s], respectively. The sum of the first 3 items on the right is the energy increment (diffused item) in the direction of r, φ and z by thermal conduction; λ is the thermal conductivity, [W/(m·K)]. The last is the source item.

For the structure of a tubing-casing annulus, Eq. 1 can be changed to a one-dimensional steady state

$$\frac{\mathrm{d}}{\mathrm{d}r}(r\frac{\mathrm{d}T}{\mathrm{d}r}) = 0 \quad (2)$$

The boundary conditions are

$$T\big|_{r=r_{to}} = T_{to} \quad (3)$$

$$T\big|_{r=r_{ci}} = T_{ci} \quad (4)$$

r_{to} and r_{ci} are the radii of the tubing outside and casing inside, [m]; T_{to} and T_{ci} are the temperatures of the tubing outside and casing inside, [°C]. Therefore, the solution of Eq. 2 is

$$\frac{\mathrm{d}T}{\mathrm{d}r} = \frac{1}{r}\frac{T_{ci} - T_{to}}{\ln(r_{to}/r_{ci})} \quad (5)$$

According to Fourier's Law, the fluid density, q, [W/m^2], through the annulus is

$$q = -\lambda_a \frac{\partial T}{\partial r} = \frac{\lambda}{r}\frac{T_{to} - T_{ci}}{\ln(r_{ci}/r_{to})} \quad (6)$$

where λ_a is the thermal conductivity of annular filled material, [W/(m·K)]. Equation 5 implies that the fluid density is inversely proportional to the radius, but the fluid through the annulus is constant.

1.2 Thermal Radiation

Because the thermal conduction and convection is dominant in case of liquid filling in the annulus, thermal radiation can be ignored. Therefore, radiation is regard to occur only when the annulus is filled with gas or vacuum in this paper. Vacuum ought to be employed under special circumstances that are hard to control, such as in exploration wells, testing wells, and wells with special functions and requirements. In these conditions, the heat transfer is simple, and only radiation exists in the annulus.

Radiation is the only way that transfers heat without a medium. The influence of radiation is dependent on the heat diffusion of the external tubing and the heat absorption of the inner casing. The radiation is attributed to the oscillations and transitions of the elementary particles that compose the materials.

In drilling engineering, it is preferable to convert the radiation energy into the radiative heat transfer coefficient, h_r, [W/(m^2·K)] (Hasan and Kabir, 2002). The expression satisfies the Stefan-Boltzmann Law

$$h_r = \frac{\sigma\left(T_{to}^{*2} + T_{ci}^{*2}\right)\left(T_{to}^{*} + T_{ci}^{*}\right)}{\frac{1}{\varepsilon_{to}} + \frac{r_{to}}{r_{ci}}\left(\frac{1}{\varepsilon_{ci}} - 1\right)} \quad (7)$$

where the asterisks denote absolute temperatures, [K]; σ represents the Stefan-Boltzmann constant with a value of 5.67×10^{-8} W/($m^2\cdot K^4$), ε is the emissivity, dimensionless.

It is difficult to acquire an accurate value of the emissivity, which depends on the temperature as well as the finish and view factors of the surface (Incropera et al., 2001). As well as these two factors, the sizes of the tubing and casing, which are classified into many series of diameters according to the API (American Petroleum Institute) standard (Gabolde and Nguyen, 1991), also affect the radiative heat transfer coefficient.

1.3 Natural Convection

Gas and liquid are materials commonly used to fill the annulus. Gases include air, N_2, CO_2 (Schulz, 2008), and liquids are water or drilling fluid (Tóth, 2006). Natural convection in the annulus is caused by fluid motion resulting from the variation of density with temperature (Willhite, 1967). Convection is usually used to describe the combined effects of heat conduction within the fluid (diffusion) and heat transference by bulk fluid flow streaming. Therefore, means of heat transfer include natural convection and radiation when the annulus is filled with gas. The proportion of the natural convection in the total heat transfer is dependent on the temperature, size of the annulus and the properties of the materials filling it (Miyauchi et al., 2014; Deguen, 2013).

Referring to natural convection, several dimensionless parameters need to be introduced: the Prandtl number (Pr), the ratio of kinematic viscosity and thermal diffusivity, $\mathrm{Pr}=c_p\eta/\lambda_a$, where η is the dynamic viscosity, [kg/(m·s)]; the Grashof number (Gr), the ratio of buoyancy and viscosity, $\mathrm{Gr}=(r_{ci}-r_{to})^3 g\rho^2\beta(T_{to}-T_{ci})/\eta^2$, where g and β are the acceleration owing to gravity, [m/s^2], and the thermal expansion coefficient, [K^{-1}], respectively. In the case of an ideal gas, β can be described as the reciprocal of the absolute value of the mean annular temperature (Willhite, 1967), T_{av}^*, [K].

$$\beta=1/T_{av}^* \tag{8}$$

The change from laminar to turbulent flow in the case of natural convection is expressed by the Rayleigh number, Ra that is defined as the product of Gr and Pr (Rohsenow et al., 1998)

$$\mathrm{Ra}=\mathrm{Gr}\cdot\mathrm{Pr}=\frac{g\beta\left(T_{to}-T_{ci}\right)\left(r_{ci}-r_{to}\right)^3}{\nu a} \tag{9}$$

where v is the kinematic viscosity, [m^2/s], and a is the thermal diffusivity, [m^2/s]. The change from laminar to turbulent flow takes place at $\mathrm{Ra}=10^9$ (Schulz, 2008). The properties of dry air and saturated water with temperature are provided by (Yang and Tao, 1998).

Three methods are used to estimate the convective heat transfer coefficient, h_c, [W/($m^2\cdot K$)] in a cylindrical coordinate. The first is the Dropkin and Somercales method (Dropkin and Somercales, 1965)

$$h_c=\frac{0.049\mathrm{Ra}^{1/3}\mathrm{Pr}^{0.074}\lambda_a}{r_{to}\ln\left(r_{ci}/r_{to}\right)} \tag{10}$$

Equation 10 is valid when $5\times10^4 < \mathrm{Ra} < 7.17\times10^8$.

The second is the Holman (1981) method

$$h_c=\begin{cases}\dfrac{\lambda_a}{r_{to}\ln(r_{ci}/r_{to})} & \left(\mathrm{Ra}\le 6\,000\right)\\ \dfrac{0.13\lambda_a\mathrm{Ra}^{0.25}}{r_{to}\ln(r_{ci}/r_{to})} & \left(6\,000<\mathrm{Ra}\le 2\times10^5\right)\\ \dfrac{0.048\lambda_a\mathrm{Ra}^{0.333}}{r_{to}\ln(r_{ci}/r_{to})} & \left(2\times10^5<\mathrm{Ra}<1.1\times10^7\right)\end{cases} \tag{11}$$

The third is the Fishenden and Saunders (1950) method. According to the annular Nu that is defined by the diameter of the tubing outside, d_{to}, m, and the convective heat transfer coefficient is

$$h_c=0.1\cdot\left(r_{ci}/r_{to}\right)^{0.15}d_{to}^{-0.1}\lambda_a\left(r_{ci}-r_{to}\right)^{-0.9}\mathrm{Ra}^{0.3} \tag{12}$$

2 RESULTS AND DISCUSSION

2.1 Radiative Heat Transfer Coefficient

The present study assumes an example of a geothermal wellbore, including a tubing-casing annulus with a depth of 2 000 m. The tubing's outside diameter (OD) is 4-1/2 inch (r_{to}=57.15 mm) and the available diameters of the casing OD are 13-3/8 inch (r_{ci}=157.65 mm), 10-3/4 inch (r_{ci}=123.95 mm), 9-5/8 inch (r_{ci}=108.4 mm), 8-5/8 inch (r_{ci}=98.1 mm), and 7 inch (r_{ci}=75.2 mm). The temperature of the tubing outside is 140 °C in stable state, and the temperature of the casing inside equals the formation temperature with a geothermal gradient of 3.5 °C/100 m. The emissivities of light metal and heavy metal are 0.15 and 0.68, respectively.

Figure 2 exhibits the radiative heat transfer coefficient in the annulus with various annular sizes when the tubing and casing are made of light metal or heavy metal, and the material filling the annulus is air with 1 bar pressure. The physical properties of air and water involved in this paper are provided by Yang and Tao (1998).

Figure 2 indicates that h_r is mostly dependent on the materials of the tubing and casing, and the ranges are 0.9–1.5 and 5.4–7.8 W/($m^2\cdot K$) for light and heavy metal, respectively. If an insulating layer is coated outside the tubing, the temperature difference across the annulus will be small, and therefore h_r is usually very small (Hasan and Kabir, 2002). However, h_r changes with the size and temperature of the annulus.

2.2 Rayleigh Number

The Rayleigh number Ra is dependent on the size and temperature of the annulus besides the properties of the material filling the annulus. Figures 3 and 4 show the trend of Ra when air (solid lines) or water (dotted lines) fills in the annulus with various sizes and temperatures. The sizes of the tubing and casing are the same as those shown in Fig. 2, and the pressure of air is set to 1 bar. The annular temperature difference is selected 40 °C due to a real wellbore condition and the satisfied results in Fig. 3, and the OD of the tubing and casing in Fig. 4 are 4-1/2 and 9-5/8 inch. The Ra of air and water are 10^3–10^6 and 10^6–10^{10} referring to these two figures. Therefore, it is desirable to fill the annulus with as much gas as possible to reduce the heat losses in the annulus.

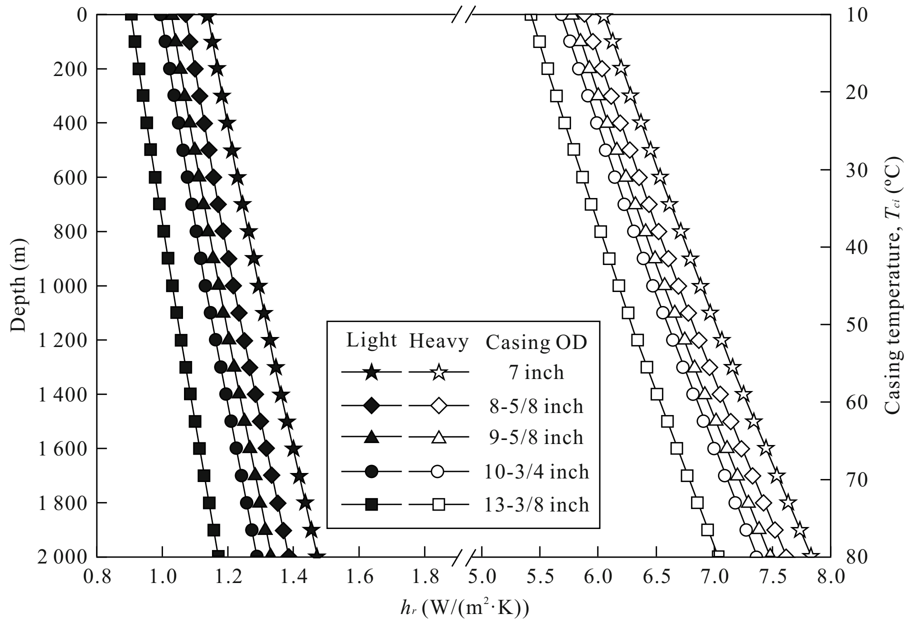

Figure 2. Radiative heat transfer coefficient in the annulus.

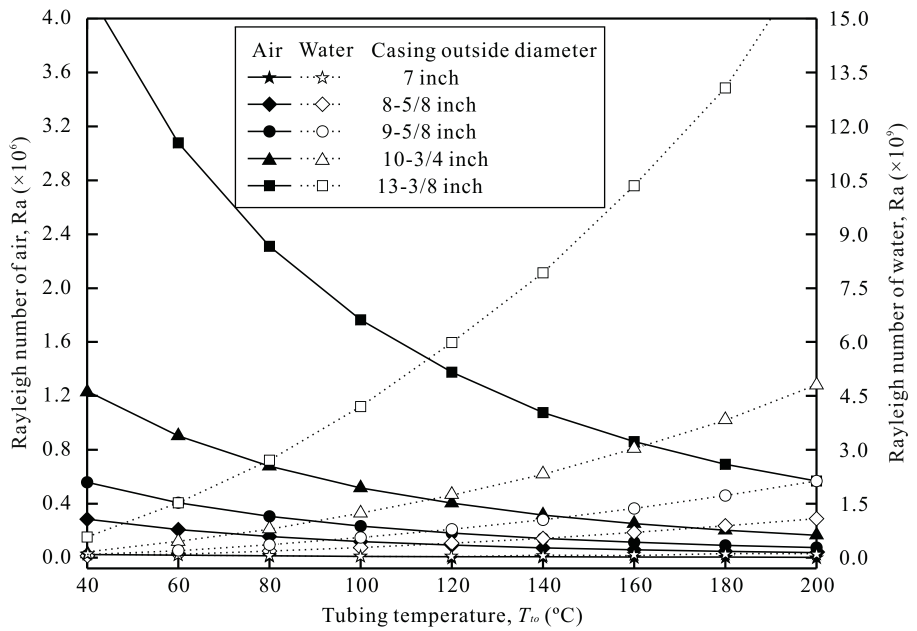

Figure 3. Ra of air and water filling the annulus with various sizes.

2.3 Convective Heat Transfer Coefficient

Owing to the significantly lower Ra for a gas filling medium, this paper extensively investigates the gas filling in the annulus for insulation. According to the above analysis of natural convection, three methods can be employed to estimate the convective heat transfer coefficient in the annulus. The Dropkin and Somercales method expressed as Eq. 10 is only valid while $5\times10^4<\text{Ra}<7.17\times10^8$, which does not cover the whole range of Ra when the annulus is filled with air. Therefore, this paper attempts to compare these three methods to develop a new one.

Within the valid range of the Dropkin and Somercales method, Figs. 5 and 6 display the results of the convective heat transfer coefficient of the three methods with low and high Rayleigh numbers, and the corresponding values of Ra are marked on the right of the figures. It is assumed that the tubing OD is 4-1/2 inch; the casing ODs are 8-5/8 inch for low Ra and 13-3/8 inch for high Ra; the annular length as well as the temperatures of the tubing outside and casing inside are the same as those exhibited in Fig. 2.

These two figures indicate that the results of the three methods are quite similar at low Ra. However, with increasing Ra, the results of the Fishenden and Saunders method deviate from those of the Dropkin and Somercales method and Holman method. The reason may be that Fishenden and Saunders defined the Nu and Gr in terms of the tubing diameter, d_{to}. Therefore, it would be better to use the Holman method when $\text{Ra}\leq5\times10^4$, and use the Dropkin and Somercales method when

Ra>5×10^4. Additionally, Hasan and Kabir suggest that 25% of h_c from Eqs. 10 and 12 is proper in real wellbore condition (Hasan and Kabir, 2002). However, this opinion is not mentioned by Holman (1981), and the results from Eq. 11 march those from Eq. 10 perfectly. Therefore, we think the suggestion of Hasan and Kabir may be applicable under certain condition. When the annulus is filled with gas, an improved piecewise equation of convective heat transfer coefficient for the whole range of Ra can be expressed as

$$h_c = \begin{cases} \dfrac{\lambda_a}{r_{to}\ln(r_{ci}/r_{to})} & \text{Ra} \le 6\,000 \\ \dfrac{0.13\lambda_a \text{Ra}^{0.25}}{r_{to}\ln(r_{ci}/r_{to})} & 6\,000 < \text{Ra} \le 5\times10^4 \\ \dfrac{0.049\lambda_a \text{Ra}^{1/3}\text{Pr}^{0.074}}{r_{to}\ln(r_{ci}/r_{to})} & 5\times10^4 < \text{Ra} < 7.17\times10^8 \end{cases} \quad (13)$$

2.4 Total Thermal Conductivity

In order to allow comparison with the thermal conductivity, h_c can be converted to an equivalent thermal conductivity, λ_{ec}, [W/(m·K)] (Dropkin and Somercales, 1965).

$$\lambda_{ec} = h_c \cdot r_{to} \cdot \ln(r_{ci}/r_{to}) \quad (14)$$

Therefore, Eq. 13 can be transformed as

$$\lambda_{ec} = \begin{cases} \lambda_a & \text{Ra} \le 6\,000 \\ 0.13\lambda_a \text{Ra}^{0.25} & 6\,000 < \text{Ra} \le 5\times10^4 \\ 0.049\lambda_a \text{Ra}^{1/3}\text{Pr}^{0.074} & 5\times10^4 < \text{Ra} < 7.17\times10^8 \end{cases} \quad (15)$$

Figure 7 shows the equivalent thermal conductivity of natural convection with various casing ODs. The tubing OD and the temperature conditions were the same as those shown in Fig. 2.

Figure 7 suggests the equivalent thermal conductivity of natural convection ranges from 0.04–0.26 W/(m·K), and it increased slowly with the difference of size and temperature in the annulus.

Similar to the equivalent thermal conductivity of natural convection, the radiative heat transfer coefficient can also be transformed into an equivalent thermal conductivity, λ_{er}, [W/(m·K)].

$$\lambda_{er} = h_r \cdot r_{to} \cdot \ln(r_{ci}/r_{to}) \quad (16)$$

Afterwards, the effects of radiation and natural convection can be compared with the same dimension in a total thermal conductivity, λ_{total}, [W/(m·K)].

$$\lambda_{\text{total}} = \lambda_{er} + \lambda_{ec} \quad (17)$$

Figures 8 and 9 exhibit the total thermal conductivity in the annulus filled with air when the tubing and casing are made of light and heavy metals. The tubing and casing OD as well as the temperature conditions resemble those shown in Fig. 2.

Implied from these two figures, the total thermal conductivities range from 0.06–0.32 W/(m·K) and 0.14–0.60 W/(m·K). In Fig. 8, λ_{er} contributes only 16%–36% to λ_{total} in case of light metals. However, the contribution will increase to 54%–76% in case of heavy metals. Therefore, several methods are proposed in the present paper to reduce the heat losses from tubing to casing: (1) filling the annulus with gas instead of liquid to decrease the Rayleigh number; (2) using low emissivity materials for the tubing and casing to reduce the radiation; (3) decreasing the annular space to reduce natural convection; and (4) coating the tubing with an insulating layer to reduce the temperature.

3 CONCLUSIONS

The paper analyzed three possible heat transfer approaches in the annulus during the production process of geothermal systems. By using realistic physical parameters for the size and temperature conditions in the annulus, the characteristics of radiation, the Rayleigh number, and natural convection have been discussed. A total thermal conductivity is proposed to describe the heat transfer in the annulus, which transformed

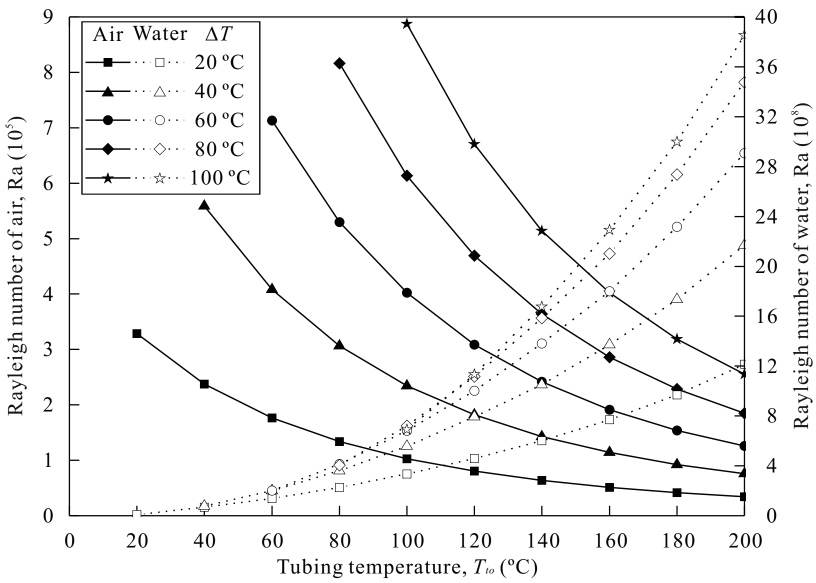

Figure 4. Ra of air and water filling the annulus with various temperatures.

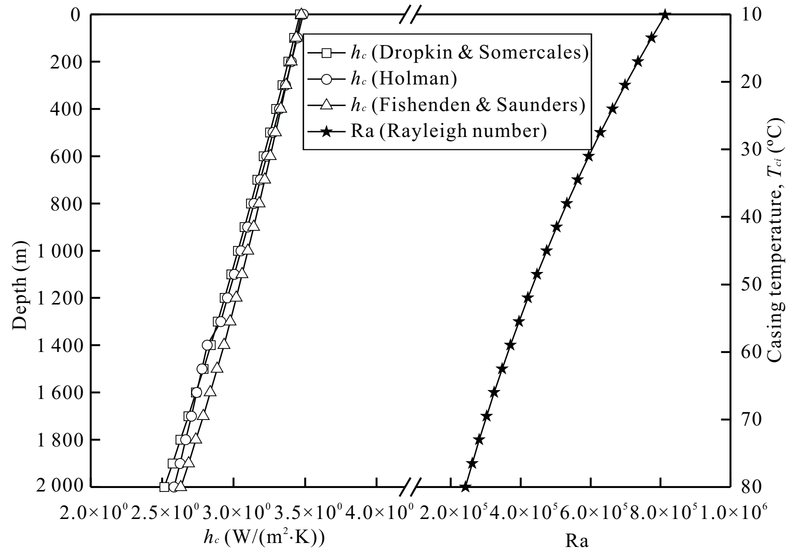

Figure 5. Convective heat transfer coefficient of three methods with low Ra.

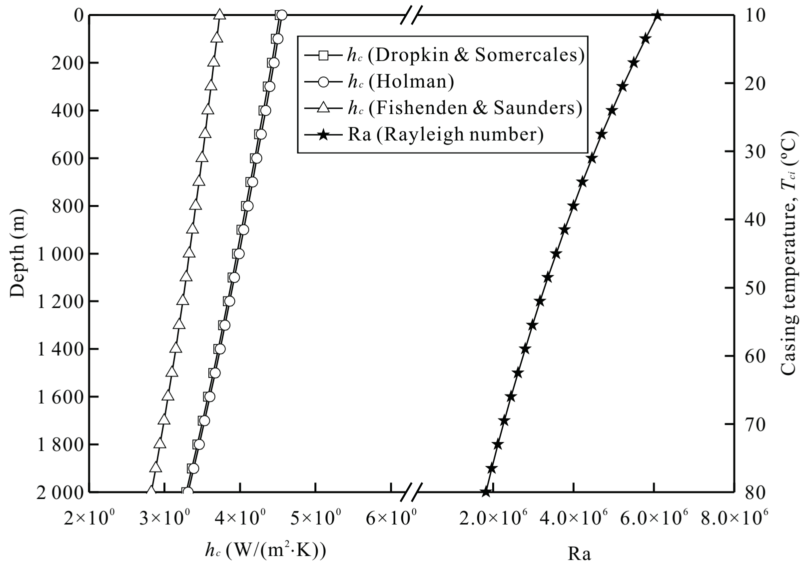

Figure 6. Convective heat transfer coefficient of three methods with high Ra.

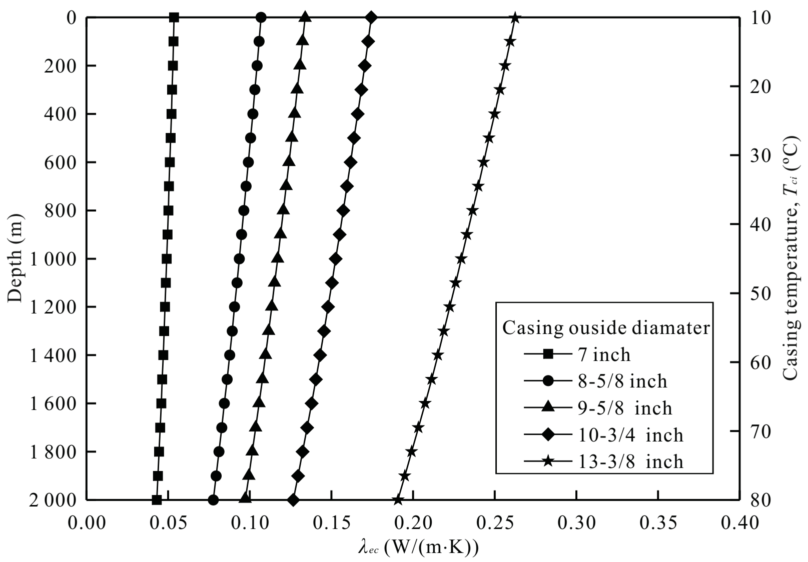

Figure 7. Equivalent thermal conductivity of natural convection when air fills the annulus.

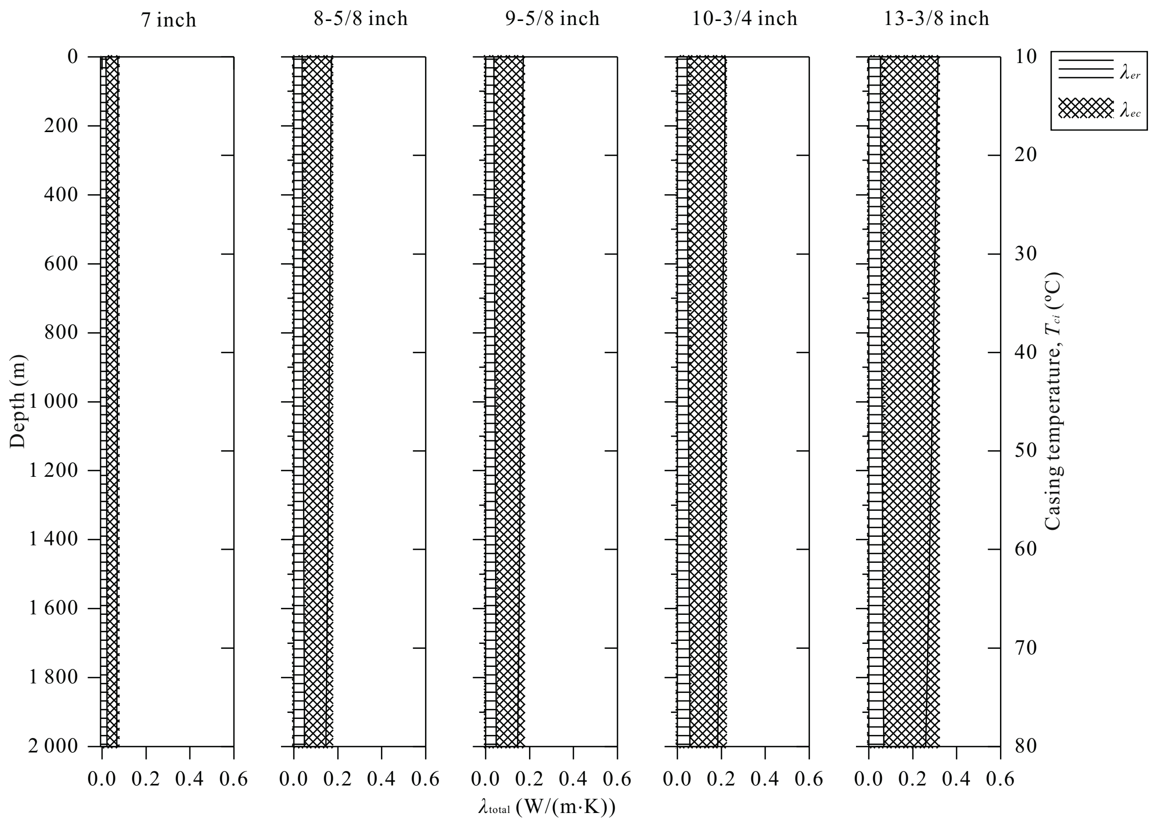

Figure 8. λ_{total} of the annulus filled with air when the tubing and casing are made of light metal.

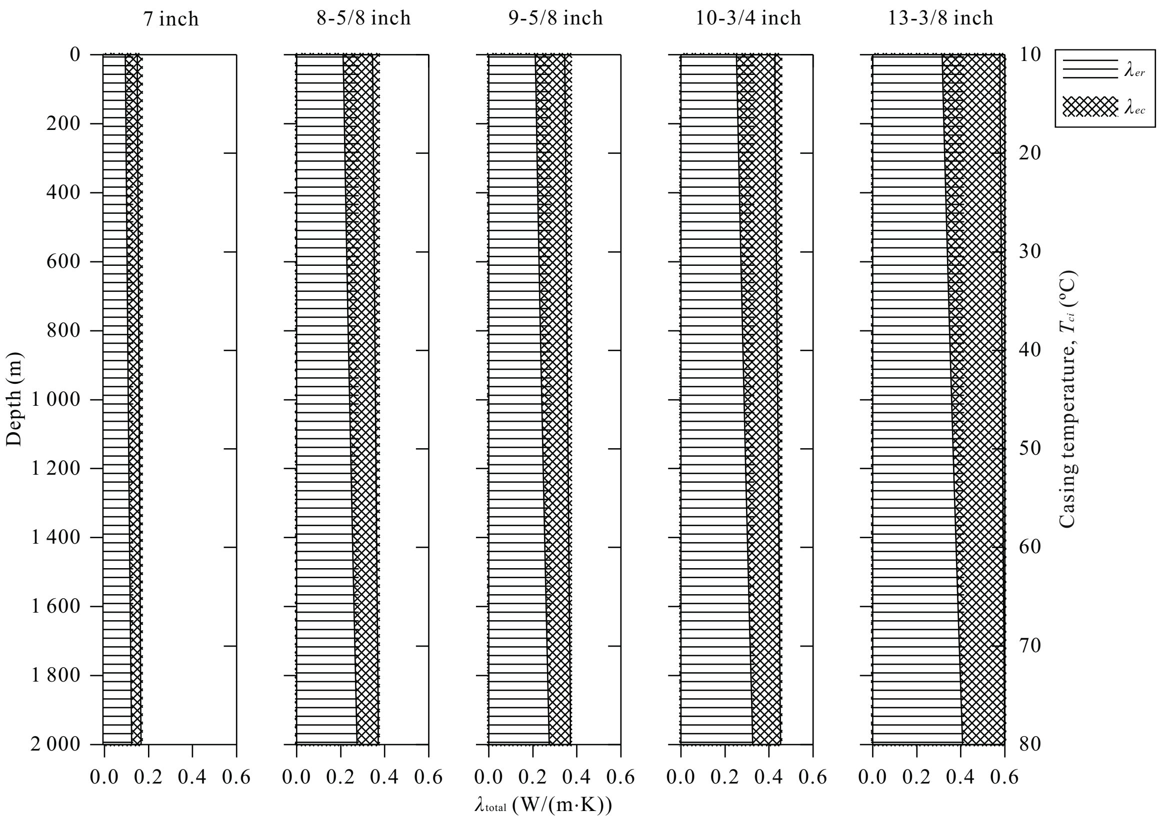

Figure 9. λ_{total} of the annulus filled with air when the tubing and casing are made of heavy metal.

the radiative and convective heat transfer coefficients into the corresponding equivalent thermal conductivities. In summary, several conclusions are obtained as follows.

(1) For the assumed size and temperature in the annulus,

the ranges of the Rayleigh number in the annulus filled with air and water are 10^3–10^6 and 10^6–10^{10}, respectively.

(2) From the comparison of three common methods, an improved correlation for estimating the convective heat transfer coefficient in the annulus is developed with a valid condition of $0<Ra<7.17\times10^8$. However, this method is suitable for natural convection when the annulus is filled with gas rather than turbulent flow when the annulus is filled with water.

(3) The total thermal conductivity in the annulus filled with air, which includes radiation and natural convection, ranges from 0.06–0.60 W/(m·K). The radiation is the second factor when the tubing and casing are made of low emissivity materials. However, the contribution increases to occupy a greater proportion when the annular size increases, and the materials have high emissivity.

By using an effective total thermal conductivity, complicated heat transfer in the annulus can be reduced to the simple problem of conduction permitting calculation of an analytical or numerical solution.

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