Numerical analysis of heat transfer characteristics for deposit formation shapes around single cylinder

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Fouling effect on heat transfer around a cylinder in cross-flow was investigated numerically by ANSYS software programme using finite element method. Calculations were made with variable local heat transfer coefficients, constant free-stream temperature and constant clean tube surface temperature. Heat transfer rates were presented for different cases with temperature field. Deposit thickness formed around the cylinder was fixed as follows: i) Non-uniform thickness of fouling shape was calculated with homogenous condition; ii) Non-uniform and non-homogenous fouling shape was considered; and iii) Effect of eccentricity was calculated for non-uniform and non-homogeneous cases. Numerical predictions were made as temperature contours through thickness of fouling and $Q_{\text{fouling}}/Q_{\text{clean}}$ was plotted against the position of fouling.

Keywords: ANSYS, Cross-flow, Deposit formation, Heat exchanger, Numerical analysis

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Introduction

Designing a heat exchanger involves factors such as heat transfer rate, fouling, power consumption, etc. in power generation industry. One of the major categories of heat exchanger fouling involves the suspended particulate matter that is encountered in many industrial fluid steams and accumulation on the surface of particles from a fluid stream on heat exchanger surface. A large number of investigations, which assessed effect of fouling on heat exchanger performance, include particle concentration, size of particle, velocity$^{1-3}$, thermal conductivity of ash$^4$, different tube geometries of tube bank$^5$, and theoretical approach of fouling$^6-10$. Buyruk$^{11}$ carried out a study to measure the effects of fouling on heat transfer characteristics of tubular heat exchanger.

Present study focuses on non-uniform thickness geometry, with special reference to fouling on a plain tube in cross-flow. Since complete heat transfer involves conjugated convection and conduction, the geometry of fouling must also provide for a convective boundary condition.

Materials and Methods

It is shown that the deposit having built up on the leading edge; it may also be build up in such a way that it is thicker at the rear stagnation point than at the forward and it may also build up different angle of the tube. In practice, deposit will not be perfectly circular but this geometry enables the situation to be explored theoretically and the general trends observed will hold regardless of the actual cross-sectional profile of the deposit. The tube is assumed to be thin and to have a constant inside surface temperature. The variations of heat transfer rate and temperature contours are obtained for different shape of fouling and for varying thermal conductivity within the non-uniform fouling over the entire cylinder by using ANSYS software program that uses finite element method (FEM).

Finite Element Method (FEM)

FEM formulations for heat flow equations and matrices under steady state and/or transient heat transfer in a three dimensional solid $\Omega$ bounded by a surface $\Gamma$ have been derived as

$$\left(\frac{\partial q_x}{\partial x} + \frac{\partial q_y}{\partial y} + \frac{\partial q_z}{\partial z}\right) + Q = \rho C \frac{\partial T}{\partial t} \quad \ldots(1)$$

If solution domain $\Omega$ is divided into $M$ elements of $r$ nodes each, finite element formulation for linear steady state problem is identified$^{12}$ as
\[ [K_e] + [K_h] \cdot \{ T \} = \{ R_Q \} + \{ R_q \} + \{ R_h \} \] ... (2)

\([K_e]\) and \([K_h]\) are element conductance matrices and relate to conduction and convection, respectively. \([R_Q], [R_q], [R_h]\) are heat load vectors arising from internal heat generation, specified surface heating, and surface convection, respectively. \([T]\) is the vector of element nodal temperatures.

\[ [K_e] = \int [B]^T \cdot [k] \cdot [B] \cdot d\Omega \]
\[ [K_h] = \int h \cdot [N] \cdot [N] \cdot d\Gamma \] ... (3)

where \([B]\) is temperature gradient interpolation matrix and \([N]\) is temperature interpolation matrix. Heat load vectors are:

\[ R_Q = \int Q \cdot [N] \cdot d\Omega \]
\[ R_q = \int q_s \cdot [N] \cdot d\Gamma \]
\[ R_h = \int h \cdot T_e \cdot [N] \cdot d\Gamma \] ... (4)

Heat Transfer Coefficient (HTC)

In present study, Reynolds number was fixed as the value of Re = 4400 based on the clean tube diameter (Re from Owen6). For a given fluid and stream velocity, local external HTC on a cylinder in cross-flow is dependent only upon angular position \(\theta\) and external diameter \(d_e\). For Reynolds number (4000-40000), average Nusselt number reported13 for airflow across the cylinder is given as

\[ \text{Nu} = 0.174 \times \text{Re}^{0.618} \] ... (5)

Since the distribution of local HTC is well defined in this range, HTC on the surface of foulant \(h_f\) can be related to HTC on the clean tube \(h_c\) as

\[ \frac{h_f d_f}{k} = 0.174 \left( \frac{U_w d_f}{\nu} \right)^{0.618} \]
\[ \frac{h_c d_c}{k} = 0.174 \left( \frac{U_w d_c}{\nu} \right)^{0.618} \] ... (6)

This relationship provides a systematic change in HTC on the external boundary as diameter increases due to fouling. HTC on the external surface is dependent on the diameter, \(d_e\), irrespective of the disposition of the foulant over the surface of clean tube.

Fouling Shapes

Buyruk11 carried out a study to measure the effects of fouling on heat transfer characteristics of tubular heat exchanger. In the present study, shape of first row fouling distribution was modelled for a single cylinder. Therefore, eccentric annulus was chosen for requirement. External surface is circular and will be unchanged even when eccentricity and hence thickness geometry is altered. Centre, bottom and upper tube of first row were taken into consideration for possible formation shape of deposit. Nature of fouling on the surface is not smooth and thickness of deposition does not have the same geometrical features. But as useful indication for engineering use, a theoretical model has been considered using the finite element analysis.

Tube diameters were as follows: clean tube, \(d_c = 0.016 \text{ m} \) (\(d_{\text{clean}}\); and fouled tube, \(d_f = 0.022 \text{ m} \) (\(d_{\text{foul}}\)). Clean tube thickness was assumed to be thin. If temperature drop across the wall is less than 1% of external temperature drop than Biot number must be less than 0.01. Clean tube surface temperature was fixed and chosen as \(T_s = 283 \text{ K}\) and free-stream temperature of air was chosen \(T_a = 373 \text{ K}\). Typical values4 of deposit material thermal conductivity (\(k_f = 0.2\) W/mK, \(k_f = 2\) W/mK) were used for calculation. In non-uniform thickness, homogenous and non-homogeneous fouling shapes (Fig. 1), non-uniform thickness and homogenous fouling shapes (Fig. 1a) had thermal conductivity of deposition as 0.2 W/mK. In this fouling shape, disposition of deposit was changed. First deposit formation was modelled, as it is thinner on front stagnation point than the rear stagnation point of the cylinder (centre tube of first row). Thinner section of deposit was moved (0-270°) gradually (possible formation of deposit of bottom and upper tube of tube bundle first row).

Secondly, deposit thermal conductivity variation is through the thickness of deposit. For this, two different deposit materials were formed as radially (Fig. 1b) and
Results and Discussion

Fig. 3 shows the temperature contours of non-uniform and homogenous fouling case. For a given fluid velocity, local external HTC on a cylinder in cross-flow is dependent only upon the angular position \( \theta \) and the external diameter \( d_e \). In those geometries, outside diameter (deposit diameter) is not changed and same HTC values were used for all calculations. Origin of clean tube location was changed to obtain different geometry of non-uniform fouling cases.

When deposit thickness is smaller on front stagnation point, temperature drop is very low due to higher convective resistance and lower conduction resistance comparing with other situations. When thickness of deposit is large then temperature gradient is higher for all geometries of non-uniform homogenous fouling cases (Fig. 3). Front surface stagnation point temperature is higher than the rear side (Fig. 3c). In this case, conduction resistance is high in the front region and resulting of high resistance, temperature drop becomes higher comparing the other cases. Similar observation can be seen in other geometries.

Heat transfer rate \( \left[ \frac{Q_{\text{foul}}}{Q_{\text{clean}}} \right] \) has been found high (Fig. 4) when fouling is concentrated on rear side (\( \theta = 180^\circ \)). Lowest heat transfer rate was observed when fouling is concentrated at front stagnation point region. This is accountable to greater asymmetry of conjugated convective and conductive resistance, as the accumulation is deposited in the region of smaller HTC.

Under temperature contours of non-homogenous (circumferential and radial) and non-uniform fouling cases (Fig. 5), in circumferentially varying thermal conductivity case (Figs 5a, 5b, 5c), first small \( k \) material replaced the front region of tube and area of \( k \) material increased gradually (45°, 90°, 120° from front stagnation point). Effect of low conductivity can be seen from the temperature contours. Low conductivity material causes the higher conduction resistance to the flow and resulting higher temperature gradient on front region.
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Fig. 5 — Temperature distributions of different non-homogeneous circumferential (a-b-c) and radial (d-e-f) fouling shapes
Fig. 6 — Temperature distributions on non-homogeneous circumferential (a-b-c) and radial (d-e-f) fouling shapes for different eccentricities.
Under radially variable (eccentricity, 0.0001 m) thermal conductivity case (Figs 5d, 5e, 5f), inside and outside tube diameters were taken as 0.016 m and 0.022 m respectively, and adjacent tube origin was moved from left to right. Small $k_1$ material was placed on the clean tube and large $k_2$ material was replaced on it. To investigate effect of different thickness of deposits, middle tubes’ centre was moved 0.0006 m (Fig. 5d), 0.0012 m (Fig. 5e) and 0.0015 m (Fig. 5f) from centre of clean tube. Temperature gradient is low at the front region in Fig. 5d (since large $k_2$ material has larger thickness) comparing with Fig. 5f. Thus, lower conductive resistance results in lower temperature drop through the thickness. This observation is seen on the rear side of the tubes for all modeled cases.

Temperature distributions on non-homogeneous circumferential and radial fouling models for different eccentricities (Fig. 6) indicate that area of small $k_1$ material was fixed for all geometries. Eccentricity values (measured from front section of tube) are changed from $7 \times 10^{-4}$ m to $12 \times 10^{-4}$ m. Small $k$ is more effective with thicker deposition on the front region of tube. When deposition thickness is small on the front region, conduction resistance becomes less effective. Temperature gradient is higher in low conductivity deposition area, resulting heat transfer rates lower in these cases. Heat transfer rates for non-homogeneous fouling shapes (Fig. 7) for both circumferentially and radially variable thermal conductivity cases decrease due to increments of lower conductivity region.

With eccentricity increase, $Q/Q_c$ decreases for radial non-homogeneous fouling case (Fig. 8). However, $Q/Q_c$ increases for circumferential non-homogeneous fouling case with increasing eccentricity, because low conductivity material causes higher conduction resistance but thinner deposition cause lower conduction resistance. Conductive resistance is more effective in circumferentially non-homogeneous fouling case for all eccentricity cases as compared with radially non-homogeneous fouling case. This is result of conjugated convective and conductive resistance.

**Conclusions**

A numerical study is presented for effect of deposit formation on a single tube of cross-flow heat exchanger. Calculations were carried out by using ANSYS software program that uses finite element program. Heat transfer efficiency of the tube was calculated using thermal resistance approach for clean and fouled tube. It was found that the deposit formation leads to reduction in the heat transfer rate strongly depends on the shape of deposit formation and thermal properties of the deposit material, and also depending on covering deposit materials’ area around clean tube. Higher temperature gradient of the deposit was obtained when large formation thickness was formed in upstream side, resulting of this heat transfer rate was found lower in this case. Effect of eccentricity on the heat transfer rate is mostly effective for circumferentially non-homogeneous case when keeping the small thermal
conductivity material area constant for radial and circumferential non-homogeneous cases.

Nomenclature

- $C$: Specific heat
- $d_c$, $d_f$: Diameter of clean and fouled tube respectively
- $e$: Eccentricity
- $h_c$, $h_f$: Heat transfer coefficient for clean tube and fouled tube respectively
- $k$: Thermal conductivity
- $N$: Element shape functions
- $Q$: Heat transfer rate
- $q_r$: Heat flux
- $R$: Heat load
- $T_a$: Air temperature
- $t$: Time
- $\rho$: Density
- $x, y, z$: Coordinates

References