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Heat Exchange at Agitation in Apparatus with Upright Heat-Exchange Devices

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Abstract—The heat exchange is examined in apparatus with agitation devices and installed heat exchange elements using semi-empirical theory of a turbulent transfer. We determined hydrodynamical parameters near a heat transfer surface.

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An amount of a removed heat in the heat exchangers of various applications including apparatus with agitators mainly determines their output. The noticeable decline in a heat exchange efficiency is due to a deposit formation caused by a heat carrier that is mostly a loose water representing dispersed medium with a large amount of admixtures. Analyzing the chemical composition of the loose water of Volgograd petroleum refinery it was found (mg l^{-1}): Al 1.61, Fe 0.39, Cu 0.22, Ca 23.6, Mg 18.2, mineral oil 108.4, organic substances 26.4, suspended mechanical solids 410. The deposits of the loose water represent heterogeneous colloid where particles of clay, silt and sand of 0.1 to 30 μm in diameter are the main components. The deposit formation on the heat transfer surfaces of the heat exchangers can lead to five times decrease of their efficiency and it means the necessity of either elimination of the formed scales or replacement of the heat-exchange devices. Thus long breaks in the operation and the output lowering are inevitable.

We assume an increase in the heat-carrier flow is a principal way of the deposit elimination that can be attained by two techniques.

Firstly we can use high-capacity pumps placed at a heat exchanger inlet. Then running costs and a consumption of the heat carrier sharply increase. The targets cannot be achieved due to engineering restrictions especially in the case of use of a vapor as the heat carrier in processes of evaporation and condensation.

There exists of great number of such processes in the petroleum refinery.

If the heat carrier is fed into pipes a problem of a uniform distribution arises especially in the case of large flows. An irregularity in the distributions impedes the efficiency growth in a whole and promotes some pipes clogging.

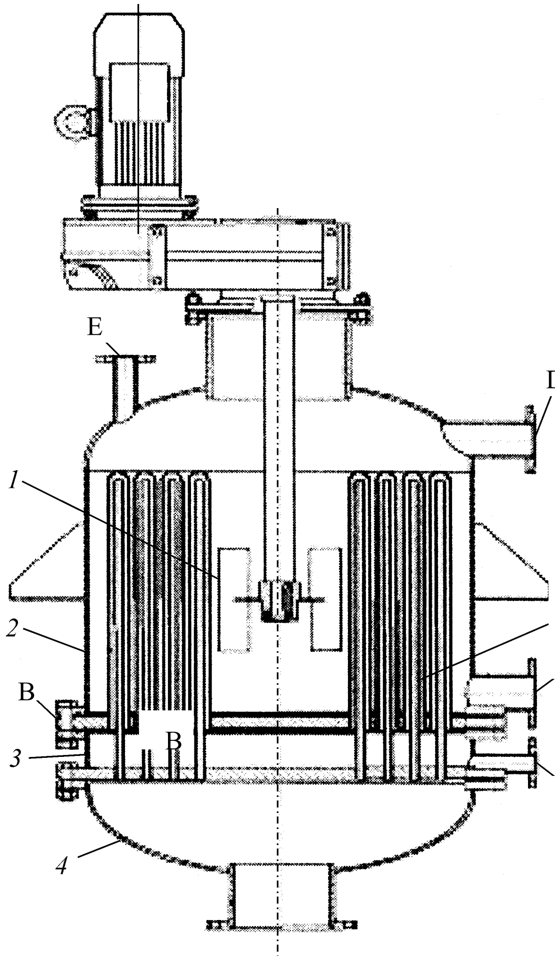
Secondly we can place a mechanical agitator berween the pipes of the heat exchanger that will provide high flow of the heat carrier near the heat exchange surface.

In this case overall efficiency of the heat exchange increases without the deposit formation on the surfaces.

It is noted the following principal disadvantages of the known constructions of the heat exchangers with the mechanical agitators: a non-efficient hydrodynamic situation and possibility of dead area formation, a difficulty of an isolation and rapid removal outside the apparatus of gaseous decomposition products which cause failures in the operational conditions of the agitators and in the rational use of the operational volume of the apparatus, and also low efficiency of the heat exchange, especially if vapor is supplied in a coil, a laboriousness of an inspection, cleaning, fixing and replacement of the heat exchange devices.

The apparatus with the upright heat exchange elements equipped by agitators enable elimination of the listed disadvantages. Obviously a hardware designing of the condensation stage requires taking into account the features of the condensation itself.

In the case of the treatment of a non-aggressive media when a life cycle of the heat exchange devices is about equal to the life cycle of the body, the pipes can be



The heat exchanger with upright pipes. (*1*) agitator, (*2*) body, (*3*) distributor chamber, (*4*) bottom, (*5*) a bundle of pipes, (A) vapor inlet, (B) vapor outlet пара; (C) water inlet; (D) water outlet, (E) thermocouples.

installed directly on the bottom (or on the cover). The pipes of the bundle can be assembled either in series or in parallel [1]. These bundles also can be installed in any order. The order of their mounting depends on a given value of a driving force of the heat transfer process, of consumption of a cooling agent and of characteristics of pumps used for traveling the cooling agent. In the case of the parallel assembling of the pipes the bundle construction is dismountable that enable regularly cleaning the inner surface of the pipes.

Apparatus presented on the figure possesses simultaneously advantages of apparatus with free liquid level and completely filled. The free liquid level enables removal of vapor and forming gases. Moreover a friction pair: a driving shaft and a shaft seal, does not contact with the medium.

At the same time negative feature inherent in the apparatus with free liquid level are absent in the operation of the suggested apparatus. The cover is a natural hindrance preventing liquid lifting upward over the wall that makes possible agitation of the liquid at any value of a speed of rotation avoiding formation of a funnel. A value of Nusselt number is in a range of 150–180 at flowing around the chess bundle of the pipes by water (Re = 10^4). Additional power consumption for an overcoming the hydraulic resistance and an increase in liquid speed up to 10 m s^{-1} results in Nusselt number growth to 800. In the apparatus with the upright heat exchange elements the funnel formation and an air absorption over the liquid surface are absent. This increases possibility of an introduction in the mixed medium of a mechanical energy promoting maintenance of the high liquid speed relative to the pipe surface which is especially important for a performance of high exothermal processes.

Convective heat transfer in the apparatus with the agitators and the installed heat exchange elements as a rule is performed under turbulent regime. However nowadays the theory of the heat exchange for the most known cases (flow in pipes, flowing around plates) under the laminar and turbulent regimes of the flow was fairly good developed but a complexity of hydrodynamics of real objects impedes employment of solutions of known problems for computation or generalization of data on the heat exchange. Experimental data on the heat irradiation usually are generalized as equations relating Nusselt number, an agitation intensity, and properties of a medium. An obtaining of such the equations requires large amount of experimental data. This problem can be essentially simplified using the semiempirical theory of the turbulent transfer at description of the heat exchange.

In this case the problem of the heat exchange computation is reduced to determination of hydrodynamic parameters near the heat transfer surface.

Let us assume that a turbulent component of thermal conductivity changes proportionally to the forth power of a distance from the wall. Then we get the following equation for computation of the heat radiation coefficient [2]

$$\alpha = 0.267\rho c\left(\varepsilon_0 \frac{\mu}{\rho}\right)^{0.25} / \mathrm{Pr}^{0.66}. \quad (1)$$

A value of the power dissipation per mass unit of the mixed medium can be assumed equal $\varepsilon = N/(\rho V)$.

Deriving equation (1) we suggested that the power dissipation near the heat exchange surface is about an average value along the apparatus. Equation (1) can be applied also to calculation of the heat radiation of the internal heat exchange devices. An area of their application is restricted by cases when the construction elements do not hinder the flowing around the pipes. Otherwise the coefficient of heat radiation diminishes as a result of the irregular distribution of an introduced energy over the whole volume of the apparatus. The coefficients of heat radiation can diminish also due to an increase in the heat heat exchange surface. In a case of additional installation of internal devices per unity of the heat exchange surface the introduced energy is lesser.

The accounting for the change in α with the increase in the overall heat exchange surface we performed by a product of ε and $F_0/(F + F_0)$ value, where F is the surface of the internal devices, F_0, the surface of the cylindrical section of the body. Since $F_0/(F + F_0)$ value is included in equation (1) to the 0.25 power, the coefficient of radiation with growth of the overall heat exchange surface decreases relatively slow. A noticeable decrease in α has to by observed at $F >> F_0$, that is confirmed experimentally.

It should be expected that an effect of a non-uniformity of the velocity field manifests itselfonly in the case of the presence in the apparatus of areas where the liquid speeds in horizontal planes significantly differ. In this case equation (1) is valid only for those sections of the volume where the hydrodynamic picture can be considered as unchangeable. A gradient of the liquid speed in respect with φ equals zero due to symmetry of the flow. Hence volume unit limited by vertical cylindrical surfaces with a radius r and $r + \Delta r$ and by horizontal planes orthogonal to the apparatus axe can be assumed as the sections with approximately uniform hydrodynamic picture. Let the main part of the energy is used at the longitudinal flowing around the installed heat exchange devices. The power required for overcoming the hydraulic resistance by an elementary unit of pipe Δh we calculate by the following equation

$$\Delta P(h, r) = \xi(h, r)\frac{\rho w^3(h, r)}{2} d_p \Delta h \qquad (2)$$

where $\xi(h, r)$ is a coefficient of resistance of the unit section of pipes; $w(h, r)$, the liquid speed, m s^{-1}, d_p, the pipe diameter, m.

The energy dissipated in the mass unit of the liquid in a trace after the unit section of the pipe is estimated by equation

$$\varepsilon(h, r) = \frac{\Delta P(h, r)}{\rho V} = \frac{\Delta P(h, r)}{\rho r \Delta\varphi \, \Delta r \Delta h}.$$

Let a sampling interval be in as φ ($\Delta\varphi = d_p/r$), then we obtain

$$\varepsilon(h, r) = \frac{\xi(h, r) w^3(h, r)}{2\Delta}.$$

The average value of $\varepsilon(h, r)$ along the pipe

$$\varepsilon_p(r) = \frac{1}{H}\int_0^H \varepsilon(h, r)\,dh = \frac{1}{H}\int_0^H \frac{\xi(h, r) w^3(h, r)}{2\Delta r}\,dh.$$

In the case of the uniform distribution of the energy along the pipe height the coefficient of heat radiation is computed by equation

$$\alpha_p(r)_u = 0.3\rho c\left[\frac{1}{H}\int_0^H \frac{\xi(h, r) w^3(h, r)}{2\Delta r}\,dh\right]^{0.25} \times \left[\frac{F_0 \mu}{(F + F_0)\rho}\right]^{0.25} / \mathrm{Pr}^{0.66}. \qquad (3)$$

Actually the energy was distributed non-uniformly and some sections of the pipe were of different efficiency. The coefficient of heat radiation of the unit section of the pipe dh is

$$\alpha(h, r) = 0.3\rho c\left[\varepsilon(h, r)\frac{F_0 \mu}{(F + F_0)\rho}\right] / \mathrm{Pr}^{0.66}.$$

An average value of the coefficient of heat radiation of the pipe can be get from equation

$$\alpha_p(r) = \frac{1}{H}\int_0^H \left\{0.3\rho c\left[\frac{\xi(h, r) w^3(h, r)}{2\Delta r} \times \frac{F_0 \mu}{(F + F\ )\rho}\right]^{0.25} / \mathrm{Pr}^{0.66}\right\} dh.$$

After division of right and left terms of given equation into right and left terms of equation (3), respectively we obtained

$$K_h = \frac{\alpha_p(r)}{\alpha_p(r)_u} = \frac{\frac{1}{H}\int_0^H [\xi^{1/4}(h, r) w^{3/4}(h, r)]\,dh}{\left[\frac{1}{H}\int_0^H \xi(h, r) w^3(h, r)\,dh\right]^{1/4}}. \qquad (4)$$

Value K_h expresses change of the coefficient of heat radiation as a result of the non-uniform power dissipation along the pipe height. Moreover an intensity of the power dissipation near some pipes can be larger or smaller than some average value that is estimated by equation (5)

$$\varepsilon_{\text{p}_{\text{av}}} = \sum_{i=1}^{J} n_{\text{p}_i} \int_0^H [\xi(h, r) w^3(h, r) \mathrm{d}h], \tag{5}$$

where n_{p_i} is number of the pipes fixed in the distance l ($r_i \le r_i + \Delta r$); J, number of the sampling intervals over the radius.

The non-uniform energy distribution between the pipes placed on a different distance from the apparatus axis is taken into account using a value of $\varepsilon_0[(\varepsilon_{\text{p}}(r)]/\varepsilon_{\text{pav}}$ instead of ε value. Finally we get dependence for the coefficient of irradiation in view of equations (4) and (5) under the conditions of the non-uniform velocity field.

$$\alpha_{\text{p}}(r) = 0.3\,\rho\, c K_r K_h \left[\frac{\varepsilon_0 F_0\, \mu}{(F + F_0)\rho}\right] / \mathrm{Pr}^{0.66}, \tag{6}$$

where K_r is coefficient expressing the change of the pipe efficiency in dependence on the distance to the apparatus axis determined by equation (7)

$$K_r = \left[\frac{\varepsilon_{\text{p}}(r)}{\varepsilon_{\text{p}_{\text{av}}}}\right]^{0.25} = \frac{\dfrac{1}{H}\displaystyle\int_0^H \xi(h, r) w^3(h, r)\mathrm{d}h \sum_i^J n_{\text{p}_i}}{\displaystyle\sum_i^J \left\{ n_{\text{p}_i} \int_0^H [\xi(h, l) w^3(h, l)\mathrm{d}h]\mathrm{d}h \right\}}. \tag{7}$$

Equation (7) can be reduced to equation (8)

$$\mathrm{Nu} = 0.3 K_r K_h \frac{d_{\text{p}}\rho}{\mu} \left[\frac{F_0 \varepsilon_0\, \mu}{(F + F_0)\rho}\right]^{1/4} / \mathrm{Pr}^{1/3} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_{\text{av}}}\right)^{1/4}. \tag{8}$$

Correction $(\mathrm{Pr}/\mathrm{Pr}_{\text{av}})^{1/4}$ is introduced to account for a direction of the heat flow.

The coefficients of heat radiation for pipes placed on the different distance from the apparatus axis are computed by equations (6) and (4). As follows from equation (4) and (7) the liquid speeds in the various horizontal planes should be known. Determination of these speeds is a laborious problem. However in the presence of a graph dependences of K_r and K_h on a value F/V or on a relative part of the body volume occupied by the pipes for the various types of the agitators the computation of the coefficient of heat radiation can be conducted significantly simpler.

$$\varphi_{\text{p}} = \frac{n_{\text{p}} d_{\text{p}}^2}{D}.$$

The velocity field can be determined for several values of number of the installed pipes n_{p}. Assuming that K_r and K_h are continuous functions $j_{\text{p}} \approx F/V$ we depicted the graph dependences $K_r(F/V)$ and $K_h(F/V)$ that further can be used for calculations of the coefficient of heat radiation. The construction of the agitator with value of a coefficient of non-uniformity K_{n} averaged in respect with all pipes is the most effective relative to the heat exchange.

$$K_{\text{n}} = \sum_{i=1}^{J} K_h(r) K_r n_i / \sum_{i=1}^{J} n_i,$$

where n_i is the number of the pipes in the sampling interval over the radius I, J, number of the sampling intervals over the radius.

The main factors influencing the heat radiation: the agitation intensity ε, the medium properties, and also indirect factors effecting through K_h, K_r, F, F_0 (the constructions of the agitator and heat exchange devices) are taken into account in equation (8). An applicability of equation (8) for coefficient of the heat radiation in a wide range of the constructive parameters and the mixed medium properties was verified experimentally [3]. A divergence of the experimental data and the calculation results were not higher than ±10%.

CONCLUSIONS

The technique for estimation of the influence of some constructive factors on the heat irradiation efficiency at the agitation in the apparatus with the installed heat exchange surface were suggested.

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