Thermal calculation of the installation for the moisture evaporation from petroleum products

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Abstract. The article presents data on the verification thermal calculation of the existing installation for moisture evaporation from petroleum products. The measures to increase its performance and reduce specific fuel consumption are proposed.

Keywords: Sludge separation unit, evaporation plant, sludge, thermal efficiency

1. Relevance of the work
Currently there are a large number of various petroleum products containing moisture, not suitable for direct use [1-10]. Their processing in distillation towers at refineries is not economically feasible due to their relatively low yield, therefore, recently there appear installations for evaporation of moisture from petroleum products called oil sludge separation units (abbreviated OSSU). Oil sludge is usually an emulsion of mineral oil and water, sometimes containing solid impurities.

In the scientific and technical literature, there are no methods for thermal calculation of OSS units, since they appeared relatively recently and are not efficient.

2. Description of the installation
The object of the study was the OSSU-3 installation, in which, with a volume of loaded sludge of ~ 3 m³ with an initial moisture content of 30%, the evaporation process up to a 1%-moisture content lasts about a day. The heat spent on the process of heating the sludge and evaporation of moisture from it, is supplied from the boiler with a nominal heat capacity of 95 kW. The boiler automation system does not allow water to heat to a temperature above 95°, while during the evaporation mode, as the operating experience has shown, the boiler works with long interruptions. From the heat balance it follows that if the nominal heat output of the boiler were used continuously, the evaporation process would last for about 6 hours, i.e. 4 times less than in the existing installation, while the productivity of the device would have increased by the same amount. This does not happen because, under the existing conditions, the heat exchange surface of the evaporator heater is insufficient to transfer the nominal heat output of the boiler to the sludge during the evaporation process. In addition, the heat of the steam leaving the
apparatus is not used at all and is lost in the heater, although numerically (with exception of losses) it is equal to the heat supplied to the oil sludge by heating water from the boiler.

The schematic diagram of the installation is shown in Figure 1.

![Figure 1](image1.png)

**Figure 1.** The schematic diagram of the OSSU installation.

First, the oil sludge is heated from a loading temperature of \( \sim 20^\circ C \) to a temperature of \( 70-80^\circ C \), at which the process of evaporation of moisture from the oil sludge begins. Evaporation takes place under vacuum created in the evaporator tank by a vacuum pump (VP). Thus, at water evaporation temperature of 75 \( ^\circ C \), the absolute pressure inside the apparatus case should be 0.039 MPa (vacuum is about 65%). The heated sludge is taken from the bottom of the evaporator, and the circulation pump (CP) is fed to the shelves of the evaporator, located in the upper part of the apparatus, serving to increase the mass transfer surface of the sludge. With partial evaporation of water on the shelves of the evaporator, the sludge is cooled by several degrees and flows back to the lower part of the apparatus.

The water vapor produced in the apparatus is pumped out with a vacuum pump, in which it mixes with the cooling water and condenses. Heated water and non-condensable gases leave the vacuum pump and enter the separator (S) connected to the atmosphere. In the separator, gases are removed into the atmosphere, and the heated water enters the air heater (H), where it is cooled with atmospheric air, and is supplied by pump P2 to a vacuum pump. The resulting condensate is periodically removed from the system. When the process of evaporation of moisture from the sludge ends, the finished oil is poured out of the tank, and a new portion of sludge is poured into the tank. Thus, the evaporation process in the OSSU installation is periodic.

3. **Scientific methodology for verification calculation**

We consider the method of verification calculation of the OSSU-3 installation in the evaporation mode. The heating surface of the evaporator consists of two registers of smooth steel pipes Dy-40 mm, length \( l=4m \), consisting of \( n=9 \) tubes each, connected in series. The temperature of the heating water at the inlet to the apparatus is \( t'_{\text{liq}}=95^\circ C \), at the outlet it is \( t''_{\text{liq}} \approx 90^\circ C \). The average temperature of oil sludge in the evaporation mode is \( t_{\text{liq}2}=75^\circ C \).

Operating experience has shown that in the mode of evaporation of oil sludge with a moisture content of not more than 30%, the consumption of steam leaving the evaporator remains almost the same throughout the entire evaporation process, which means that at constant temperatures of oil sludge and heating water, the heat flux supplied to the oil sludge, remains constant. So, the heat transfer coefficient from the heating water to the oil sludge during evaporation also remains almost unchanged. This is possible if the thermophysical properties of oil sludge that affect heat transfer are practically independent on moisture content (if it does not exceed 30% by weight).
The surface area of the heat exchange evaporator, determined by the outer diameter of pipes, is \( F = 11.5 \, m^2 \). The heat transfer coefficient from the heating water to the sludge was determined by the expression, as that for a flat wall [11]:

\[
k = \left( \frac{1}{\alpha_1} + \frac{\delta_e}{\lambda_e} + \frac{1}{\alpha_2} \right)^{-1},
\]

where \( \alpha_1 \) is heat transfer coefficient from the heating water to the inner surface of the pipes; \( \alpha_2 \) is heat transfer coefficient from the outer surface of pipes to the oil sludge, \( W/(m^2-K) \); \( \lambda_e \) is coefficient of thermal conductivity of the pipe wall; \( \delta_e \approx 0.003 \) is the pipe wall thickness, m.

For steel pipes, we took in our calculations \( \lambda_e = 51.5 \, W/(m-K) \), as for carbon steel 20 in the temperature range 20-100°C [12]. We should note that the thermal resistance of the pipe wall \( \delta_e/\lambda_e \) is many times less than the thermal resistance of heat transfer from the outer surface of pipe \( 1/\alpha_2 \), therefore some inaccuracy in setting the values of \( \delta_e \) and \( \lambda_e \) practically does not affect the accuracy of calculating the heat transfer coefficient \( k \). In addition, the thermal resistance of heat transfer on the inner surface of a pipe \( 1/\alpha_1 \) is many times less than the thermal resistance of heat transfer \( 1/\alpha_2 \) on the outer surface, since \( \alpha_1 >> \alpha_2 \).

Therefore, in the first approximation, we can take \( k \approx \alpha_2 \). At the same time, the average temperatures on the inner and outer surfaces of pipes in the first approximation are equal to: \( t_{c1} \approx t_{c2} \approx t_{sl} = (t'_{liq} + t''_{liq})/2 = (95 + 90)/2 = 92.5°C \). Further, the values of \( k, t_{c1} \) and \( t_{c2} \) are defined more accurately, since \( k < \alpha_2, t_{c2} < t_{c1} < t_{sl} \).

The heat transfer coefficient \( \alpha_1 \) from the heating water to the inner surface of pipes was determined by the well-known formula for the turbulent flow of liquid inside the pipes [13]:

\[
Nu_{l|q1} = 0.021 \cdot Re_{l|q1}^{0.8} \cdot Pr_{l|q1}^{0.43} \left( \frac{Pr_{liq}}{Pr_{c1}} \right)^{0.25};
\]

\[
Nu_{l|l} = \alpha_1 \cdot \frac{\alpha_w}{\lambda_{liq}},
\]

\[
Re_{l|l} = w \cdot \frac{d_{in}}{\nu_{liq}},
\]

where \( Pr_{liq}, Pr_{c1} \) are Prandtl numbers for water at temperatures \( t_{liq} \) and \( t_{c1} \); \( w \) is water velocity inside pipes, m/s; \( d_{in} \) is internal diameter of pipes, m; \( \lambda_{liq}, \nu_{liq} \) are coefficients of thermal conductivity and kinematic viscosity of water at temperature \( t_{liq} \).

According to the estimate, the water velocity in the heater tubes was 1.1 m/s, and the heat transfer coefficient from the heating water, calculated by the formula (2), is \( \alpha_1 = 6245 \, W/(m^2-K) \).

Further we calculate the heat transfer coefficient \( \alpha_2 \) on the outer surface of pipes. Since the cross-sectional area of the tank is large enough, the forced movement of oil sludge near heating pipes has almost no effect on convective heat transfer. The heat transfer coefficient to oil sludge \( \alpha_2 \) is determined by free convection, and it can be calculated using the well-known formula for calculating heat transfer during free convection of liquid near horizontal pipes [13]:

\[
Nu_{l|q2} = 0.5 \cdot \left( Gr_{l|q} \cdot Pr_{l|q2} \right)^{0.25} \left( \frac{Pr_{liq}}{Pr_{c2}} \right)^{0.25};
\]

\[
Nu_{l|l} = \alpha_2 \cdot \frac{d_{out}}{\lambda_{liq}}
\]

\[
Gr_{l|q2} = \gamma \cdot \beta_{liq} \cdot \left( t_{c2} - t_{liq} \right) \cdot d_{out}^3 \cdot \nu_{liq},
\]

where \( d_{out} \) is outer diameter of pipes, m; \( Pr_{liq}, Pr_{c2} \) are Prandtl numbers for oil sludge at temperatures \( t_{liq} \) and \( t_{c2} \); \( \lambda_{liq}, \beta_{liq}, \nu_{liq} \) are coefficients of heat transfer, thermal temperature volumetric expansion, kinematic viscosity of oil sludge at temperature \( t_{liq} \).

Since oil sludge is an oil-water emulsion in which water droplets are contained within the oil phase, the thermal sludge parameters affecting convective heat transfer are close to the properties of oil. The calculation will be carried out under the condition that the water content in the sludge is absent, which roughly corresponds to the end of the evaporation process. As an oil product, we choose transformer oil, its thermophysical properties were determined from tabular data [12].
At \( t_{\text{liq}2} = 75^\circ \text{C} \) and \( t_c = 92.5^\circ \text{C} \) heat transfer coefficient \( \alpha_2 \), calculated using the formula (5), is \( \alpha_2 = 100.9 \text{ W/(m}^2\text{K)} \).

Total thermal resistance of heat transfer from water to oil sludge is \( R = 1/\alpha_1 + \delta_c/\lambda_c + 1/\alpha_2 = 0.01 \text{ m}^2\text{K}/\text{W} \). The analysis shows that thermal resistance of heat transfer from water to the inner pipe surface \( 1/\alpha_1 \) is 1.6% of the total thermal resistance \( R \); thermal resistance of the pipe wall \( \delta_c/\lambda_c \) is 0.75% of the total thermal resistance \( R \); thermal resistance of heat transfer from outer wall of pipe to oil sludge \( 1/\alpha_2 \) is 98.6% of the total thermal resistance \( R \). Consequently, heat transfer from water to oil sludge is determined by the practical heat transfer from the outer pipe surface.

The heat transfer coefficient calculated using the formula (1), in the first approximation, is equal to: \( k = 99.5 \text{ W/(m}^2\text{K)} \). We specify the values \( t_c \), \( \alpha_2 \) and \( k \). Heat flux density through the pipe wall: \( q = k \cdot (t_{\text{liq}1} - t_{\text{liq}2}) = 1742 \text{ W/m}^2 \). The temperature of the outer surface of the pipe: \( t_c = t_{\text{liq}1} + q/\alpha_2 = 92.25^\circ \text{C} \). At \( t_c = 92.25^\circ \text{C} \), the heat transfer coefficient on the outer surface of the pipe is \( \alpha_2 \) = 100.6 W/(m\(^2\)K), and the heat transfer coefficient \( k = 99.2 \text{ W/(m}^2\text{K)} \), which is slightly different from originally obtained values.

We determine the heat flux transferred from the heating water to the oil sludge through the heating surface according to the equation of heat transfer [11]:

\[
Q = k \cdot (t_{\text{liq}1} - t_{\text{liq}2}) \cdot F = 19990 \text{ W}=20.0 \text{ kW}.
\]

Thus, in the mode of evaporation of transformer oil, the thermal power transmitted through the existing heating surface is 19.96 kW, which is 4.7 times less than the nominal heat output of the boiler (95 kW). This means that the heat output of the boiler in the evaporation mode is largely unused.

We will check it using the heat balance method. The heat power supplied with the heating water in the evaporation mode is spent on steam formation (if you do not take into account the small heat loss from the outer surface of the apparatus). The evaporation process of 2.8 m\(^3\) of oil sludge with an initial water content by weight of 30% to a content of 1% lasts approximately \( \tau \approx 24 \) h (according to experimental data). With the oil density of 880 kg/m\(^3\) and the water density of 998 kg/m\(^3\) (at 20 °C), the average density of oil sludge with a water content of 0.3 is 912 kg/m\(^3\), while the mass of 2.8 m\(^3\) of loaded sludge is 2553 kg. During the process of evaporation, this amount of moisture is removed from the oil sludge: \( M = (0.3-0.01) \cdot 2553 = 740.3 \) kg. The evaporation of 1 kg of water at 75 °C consumes the heat of vaporization \( r = 2320 \text{ kJ/kg} \).

The average heat power consumed for evaporation of water is: \( Q = M \cdot r/(3600 \tau) = 740.3 \cdot 2320/(3600 \cdot 24) = 19.87 \text{ kW} \).

Thus, the thermal power of the apparatus during evaporation, determined by the method of heat balance (19.87 kW), is slightly less than that from the heat transfer equation (20.0 kW). This suggests that the calculation was carried out quite correctly, and the assumptions made in the calculation of heat transfer to the oil sludge are correct.

The amount of moisture evaporated per unit of time in the evaporation mode, kg/s, can be determined from the heat balance equation:

\[
G_n = \eta \cdot \frac{Q}{r} \quad \text{(8)}
\]

where \( \eta \leq 1 \) is efficiency of evaporator; \( Q \) is heat flow supplied to the oil sludge in the evaporation mode, kW; \( r \) is the heat of vaporization, kJ/kg.

So, at \( Q = 19.9 \text{ kW} \), \( r = 2320 \text{ kJ/kg} \) and \( \eta = 0.98 \) the consumption of the produced steam is: \( G_p = 0.0086 \text{ kg/s} = 30.9 \text{ kg/h} = 741 \text{ kg/day} \). If the heat output of the boiler of 95 kW were used continuously, then the productivity would be: \( G_n = 0.041 \text{ kg/s} = 147.4 \text{ kg/h} = 3537 \text{ kg/day} \).

We will carry out a verification thermal calculation of the existing evaporator for the regime of heating of oil sludge. When cold sludge is loaded into the apparatus with a temperature of \( t_{\text{liq}20} = 20\pm25^\circ \text{C} \), it takes some time to warm it up to a temperature of 75°C, before the process of evaporation of moisture begins. Let us calculate the heat flux transferred from the heating water to the oil sludge in the heating mode using the considered method.
4. The results of the verification calculation

At the initial oil sludge temperature \( t_{\text{liq}20} = 20 ^\circ \text{C} \), the temperature pressure is \( (t_{\text{liq}1}-t_{\text{liq}20}) = 92.5-25=67.5 ^\circ \text{C} \), which is 3.85 times more than the temperature pressure in the evaporation mode \( (t_{\text{liq}1}-t_{\text{liq}2}) = 92.5-75 = 17.5 ^\circ \text{C} \). The heat transfer coefficient on the outer surface at \( t_{\text{liq}20} = 25 ^\circ \text{C} \) is \( \alpha_2 = 122.5 \) \( \text{W/(m}^2\text{-K)} \), and the heat transfer coefficient \( k = 120.5 \) \( \text{W/(m}^2\text{-K)} \). The heat flux transferred from the water to the oil sludge at the heating surface \( F = 11.5 \) \( \text{m}^2 \) will be \( Q = 93.6 \) kW, which is close to the nominal power of boiler.

During the process of heating the sludge, its average temperature \( t_{\text{liq}2} \) increases, which leads to a decrease in the temperature head \( \Delta t = t_{\text{liq}1}-t_{\text{liq}2} \), as a result of which the heat flux \( Q \) from water to oil sludge decreases, according to the expression: \( Q = k \cdot \Delta t \cdot F \).

Table 1 shows the results of calculating \( \Delta t \), \( \alpha_2 \), \( k \) and \( Q \) at various temperatures of oil sludge \( t_{\text{liq}2} \). The average temperature of heating water in tubes for calculation is assumed to be \( t_{\text{liq}1} = 92.5 ^\circ \text{C} \); the existing heating surface area \( F = 11.5 \) \( \text{m}^2 \).

| \( t_{\text{liq}2}, ^\circ \text{C} \) | \( \Delta t, ^\circ \text{C} \) | \( \alpha_2, \text{W/(m}^2\text{-K)} \) | \( k, \text{W/(m}^2\text{-K)} \) | \( Q, \text{kW} \) |
|---|---|---|---|---|
| 25 | 67.5 | 122.5 | 120.5 | 93.6 |
| 30 | 62.5 | 121.7 | 119.6 | 90.0 |
| 40 | 52.5 | 128.1 | 125.9 | 76.0 |
| 50 | 42.5 | 122.6 | 120.6 | 58.9 |
| 60 | 32.5 | 116.2 | 114.3 | 42.7 |
| 70 | 22.5 | 107.0 | 105.5 | 27.3 |
| 75 | 17.5 | 101.0 | 99.7 | 20.0 |

As it is seen from Table 1, the boiler nominal capacity (95kW) is completely used in the existing evaporator unit only at the beginning of the sludge heating process, when the temperature difference between the heating water and sludge is maximum, after which the heat flux decreases continuously with increasing sludge temperature. At the oil sludge temperature \( t_{\text{liq}2} = 75 ^\circ \text{C} \), the heat flux is 20 kW.

We calculate the duration of the process of heating the sludge. Heat flow supplied to oil sludge is used to increase its temperature according to the equation:

\[
Q = c_{\text{liq}2} \cdot G_{\text{liq}2} \cdot \frac{dt_{\text{liq}2}}{dr}
\]  

(9)

Where \( c_{\text{liq}2} \) is isobaric heat capacity of oil sludge, \( \text{kJ/(kg-K)} \), at temperature \( t_{\text{liq}2} \); \( G_{\text{liq}2} \) is mass of the loaded oil sludge, kg; \( r \) is time, s.

In expression (9), the values of \( Q \) and \( c_{\text{liq}2} \) depend on the oil sludge temperature \( t_{\text{liq}2} \), therefore, this equation is solved numerically. The duration of the heating of the oil sludge from the temperature \( t_{\text{liq}20} \) to \( t_{\text{liq}2} \) is determined as a result of the integration of the latter expression:

\[
\tau_{\text{heat}} = G_{\text{liq}2} \cdot t_{\text{liq}20} \cdot c_{\text{liq}2} \cdot \frac{dt_{\text{liq}2}}{Q(t_{\text{liq}2})}
\]  

(10)

The relationship between the heat capacity of oil sludge and the temperature \( t_{\text{liq}2} \) in the range of 20-75°C with a mass fraction of moisture of 30% can be described by the expression: \( c_{\text{liq}2} = 2.38 + 0.00364 \cdot t_{\text{liq}2} \), kJ/(kg-K).

The heat capacity of oil was selected according to the tables for transformer oil [12]. The heat capacity of water was assumed to be constant, equal to 4.19 \( \text{kJ/(kg-K)} \). The heat capacity of the oil sludge was determined taking into account the mass fractions of water and oil in the sludge.

The duration of heating of oil sludge of 2.8 \( \text{m}^3 \) volume and a mass \( G_{\text{liq}2} = 2553 \) kg with initial water content of 30% from a temperature of 20 °C to 75 °C, calculated by the formula (10), taking into account the data in Table 1, will be \( \tau_{\text{heat}} = 6990 \) s = 116.6 min. After the end of the heating mode, the evaporation mode begins.
5. Conclusions
The verification thermal calculation of the OSSU-3 installation showed that the boiler thermal power is used inefficiently, because only at the beginning of the oil sludge heating process all the nominal boiler power is transferred to the sludge, and during evaporation the heat flow from the heating water to the sludge decreases several times. To increase the capacity of the evaporator, it is necessary to increase the heater surface for several times, in addition, at the same time it is necessary to increase the cooling capacity of the heater for condensation of the increased steam consumption. In addition, the heat of the produced steam can be disposed in an additional evaporating stage, operating using the principle of the main apparatus, only instead of heating water, the additional stage will be heated with steam coming out of the main stage.

A method for thermal calculation of an evaporator for an oil sludge separation unit is proposed. Verification thermal calculation of the OSSU-3 evaporator was carried out. General recommendations for improving the thermal efficiency of this installation are given.

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