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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machines

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JAN MADEJSKI[1]

Boiling and 2-Phase Flow in Vertical Tubes of Evaporators with Natural Circulation [2]

Title problem has been considered for four types of evaporators, as used in steam boilers, refrigeration, sugar industry, and chemical plants. At first the flow in the downcomer tube has been analyzed, and it has been found that this tube, if heated, may be the source of vibrations. Conditions to avoid this phenomenon have been formulated. Furthermore a method of thermal design of evaporators has been outlined, and especially the calculation of circulation velocity, which has been illustrated by numerical examples.

Nomenclature

Symbols recommended by the International Centre for Heat and Mass Transfer are used throughout in this report. Especially the properties of saturated liquid are denoted with primes, those of vapour with double-primes. Special symbols are listed below:

C_1, C_2 – constants, (31) and (32),
d – inside diameter,
d_e – outside diameter,
f_k – frequency,
Δh – latent heat of vaporisation,
H – tube length,
i, j, k, m, n – integers,
k_d, k_g, k_u – overall heat transfer coefficients,
l – characteristic dimension, eq. (22),
n, N – number of riser tubes per one downcomer,
R – thermal resistance of the tube wall,
S – slip ratio,
T_S – saturation temperature,
T_{IB} – incipient boiling temperature,
w – velocity,
z – co-ordinate,
α – heat transfer coefficient,
β – coefficient of thermal expansion,
φ – void fraction,
ϕ_{ltt} – Martinelli-Nelson-Lockhart factor,
ω_k – angular frequency,

Subscripts

DB – Dittus-Boelter formula,
g – gas as heating fluid,
i – feed liquid,
k – condensing steam as heating fluid,
o – downcomer tube,
PB – pool boiling,
IB – incipient boiling,

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[2] Praca wykonana w projekcie badawczym IMP PAN nr 126/3/91

1. Introduction

A selection of evaporators in question is shown in Figures 1, 2 and 3. The basic design of a drum boiler is shown in Fig. 1. The downcomer tube is outside the flame zone, and the flow of flue gases is perpendicular to the plane of drawing. Therefore the downcomer tube can be treated as adiabatic, and the heat transfer coefficient from gas to the surface of riser tubes, α_u, is constant over all the surface. Another type, shown in Fig. 2, is used in sugar industry as a vacuum pan, and as heating fluid condensing steam is usually used; therefore the heat transfer coefficient on riser tubes, α_k is constant. In this case, however, the downcomer tube is also heated, which has some undesired consequencies. This type can be used in unsteady action.

Fig. 3 shows a reboiler used e.g. in petroleum industry as a part of distillation column. In this case the downcomer tube is adiabatic, and the heat transfer coefficient is constant only when heating occurs by condensing steam. If as heating fluid flue gases are used, the heat transfer coefficient is variable, as well as the local temperature of the gas. When baffles are used, as shown in the picture, the reboiler consists of several sections of cross flow, and the design philosophy applied to the type shown in Fig. 1 can be utilized although the procedure become very complicated.

In Fig. 4 a type of drum boiler heated in cross flow by flue gases from the reheater of a gas turbine plant is shown. It consists of a drum and a number of loops in which the front legs serve as risers, and the rear legs as downcomers working in the flame zone with somewhat diminished heat transfer coefficient, $\alpha_d < \alpha_u$, and also diminished gas temperature. It has been shown previously [1] that in downward flow of liquid the bubbles generated on a heated wall are not able to break off from the wall. Therefore when the wall temperature exceeds that of nucleation, $T_{IB} = T_S + \delta T_{IB}$, the bubbles cannot depart, and coalesce to form a vapour layer instead. Thus the liquid column is separated from the wall by the vapour layer which cannot be stable. As a consequence the liquid column will vibrate according to the case of Helmholtz instability. The angular frequencies can be calculated from the formula [2]

$$\omega_k^2 = k(k^2-1)\cdot\frac{8\sigma}{\varrho' d_0^3}, \quad k = 1, 2, \cdots \tag{1}$$

When the circular form of cross-section is preserved ($k = 1$) the liquid column may be subject to longitudinal vibrations which cause the column to break into droplets. This is possible in a free liquid jet. In the discussed case, however, the vapour layer is very thin, and this prevents the drop formation. Therefore it must be $k > 1$, and this means such changes of cross-section that liquid will touch the wall in k places. The forms of radial vibrations are shown in Fig. 5.

Applying the formula (1) to water under pressure of 1.003 MPa, i.e. for $T_S = 180^0C$, $\varrho' = 886.52\,kg/m^3$, $\sigma = 0.04228\,N/m$, flowing in tube of inside diameter $d_0 = 0.02\,m$, we obtain $\omega_2 = 16.87\,rad/s$, $\omega_3 = 33.73\,rad/s$, $\omega_4 = 53.34\,rad/s$.

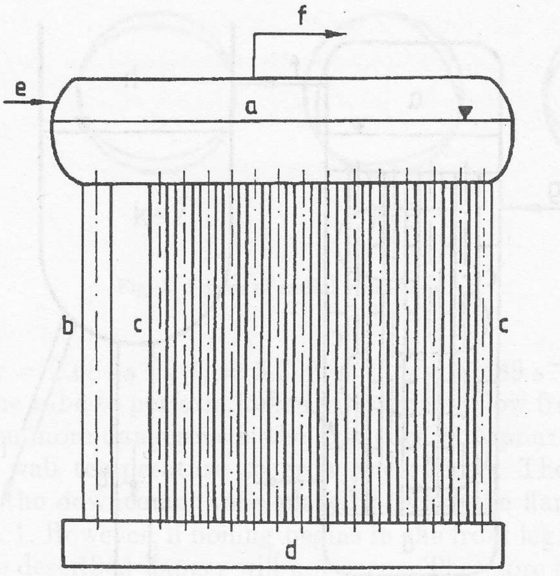

Fig. 1. a – drum, b – downcomer tube, c – riser tubes, d – collector, e – feed, f – vapour outlet

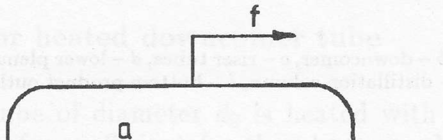

Fig. 2. a – upper plenum, b – downcomer tube, c – riser tubes, d – lower plenum, e – thin juice inlet, f – steam outlet, e' – thick juice outlet

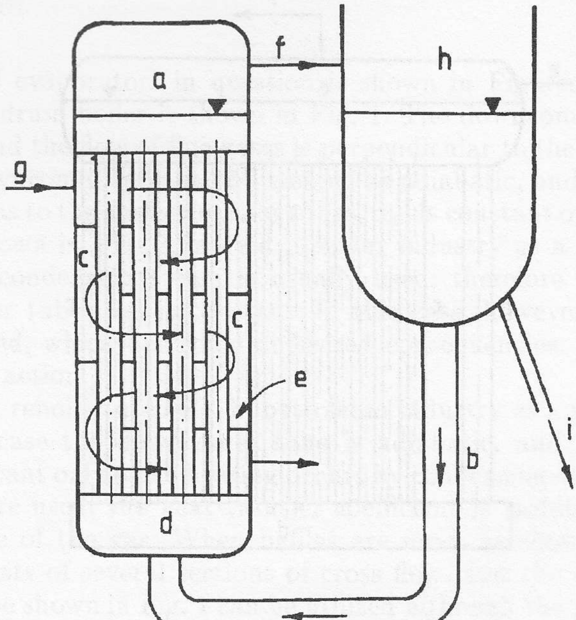

Fig. 3. a – upper plenum, b – downcomer, c – riser tubes, d – lower plenum, e – baffles, f – vapour outlet, g – heating fluid inlet, h – distillation column, i – bottom product outlet

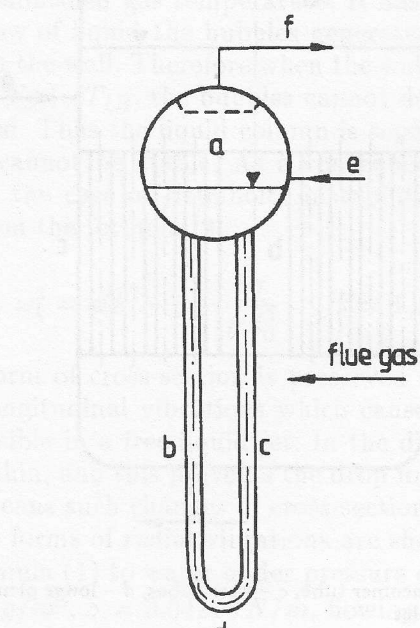

Fig. 4. a – drum, b – downcomer, c – riser, d – bend, e – feed, f – steam outlet

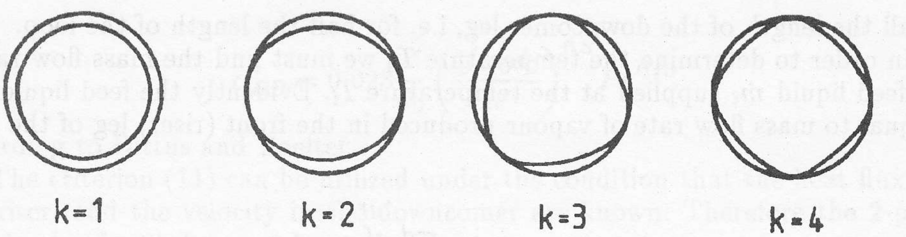

Fig. 5. Forms of radial vibrations

Hence $f_2 = \omega_2/2\pi = 2.684\,s^{-1}$, $f_3 = 5.369\,s^{-1}$, $f_4 = 8.489\,s^{-1}$. Thus the liquid column compels the tube to perform forced vibrations of low frequency which can be dangerous. Even more dangerous is the existence of vapour larger as isolation which causes the wall temperature to grow accordingly. Therefore the proper solution is to use the downcomer tube placed beyond the flame zone, as in the type shown in Fig. 1. However, if boiling begins in the front leg, even immediately after the bend, the described danger will not occur. Therefore it is possible to use such a boiler under the condition which must be mathematically formulated as follows.

2. Safety condition for heated downcomer tube

When a downcomer tube of diameter d_0 is heated with gas of temperature T_g at an overall heat transfer coefficient k_d, then heat supplied at a elementary distance dz equalling

$$d\dot{Q} = k_d(T_g - T)\pi d_0 \cdot dz \tag{2}$$

causes the temperature increase of liquid dT, where also

$$d\dot{Q} = c_p'\varrho' w_0 \frac{\pi}{4} d_0^2 dT; \tag{3}$$

w_0 is the liquid velocity. Elliminating $d\dot{Q}$ and solving the differential equation at the initial condition $T = T_0$ for $z = 0$, i.e. for the inlet from the drum to the tube, we obtain

$$T = T_g - (T_g - T_0)exp\left(-\frac{4k_d z}{c_p'\varrho' w_0 d_0}\right). \tag{4}$$

As mentioned before the temperature must not exceed the temperature T_{IB} of the begin of nucleation. Therefore the condition is

$$T_g - (T_g - T_0)exp\left(-\frac{4k_d z}{c_p'\varrho' w_0 d_0}\right) < T_{IB} \tag{5}$$

for all the length of the downcomer leg, i.e. for half the length of the loop.

In order to determine the temperature T_0 we must find the mass flow rate of the feed liquid $\dot{m}_i$ supplied at the temperature T_i. Evidently the feed liquid rate is equal to mass flow rate of vapour produced in the front (riser) leg of the loop, viz.

$$\dot{m}_i = \dot{m}'' = \frac{\pi d}{\Delta h}\int_0^H q dz, \tag{6}$$

where H is the boiling length, q local heat flux, Δh latent heat of vaporization, and d the inside diameter of riser tube; in the case considered it is $d = d_0$. In the drum we have to do with mixing of feed liquid (given temperature T_i, mass flow rate $\dot{m}_i$) and boiling liquid from riser tubes delivered at the saturation temperature T_S and mass flow rate

$$\dot{m}_0 = \frac{\pi}{4} d_0^2 \varrho' w_0 \tag{7}$$

including $\dot{m}''$. The enthalpy balance of mixing process is thus

$$(\dot{m}_0 - \dot{m}_i)c_p' T_S + \dot{m}_i c_p' T_i = \dot{m}_0 c_p' T_0 \tag{8}$$

whence

$$T_0 = \frac{\dot{m}_i}{\dot{m}_0} T_i + \left(1 - \frac{\dot{m}_i}{\dot{m}_0}\right) T_S \tag{9}$$

and finally, with use of (6) and (7), we obtain

$$T_0 = T_S - \frac{4(T_S - T_i)d}{\Delta h \varrho' w_0 d_0^2}\int_0^H q dz. \tag{10}$$

Substitution in (5) yields the condition

$$\frac{4k_d z}{c_p' \varrho' w_0 d_0} < ln\left\{\frac{1}{T_g - T_{1B}} \cdot \left[T_g - T_S + \frac{4(T_S - T_i)d}{\Delta h \varrho' w_0 d_0^2} \cdot \int_0^H q dz\right]\right\} \tag{11}$$

which should be fulfilled for the whole length of the downcomer part of tube. The overall heat transfer coefficient is to be calculated from the formula

$$k_d = \left(\frac{1}{\alpha_d} + R + \frac{1}{\alpha_{DB}^*}\right)^{-1} \tag{12}$$

where R denotes the thermal resistance of tube wall, α_d is the heat transfer coefficient from gas to wall, and

$$\alpha_{DB}^{*} = 0.023 \frac{\lambda'}{d_0} \left(\frac{\varrho' w_0 d_0}{\mu'} \right)^{0.8} Pr'^{1/3} \tag{13}$$

according to Dittus and Boelter.

The criterion (11) can be utilized under the condition that the heat flux q in the riser, and the velocity in the downcomer are known. Therefore the 2-phase flow in the riser tube must be analysed.

3. Boiling in flow throught the riser tube

The analysis is based on assumption that the frictional pressure drop in the loop as a whole, $\Sigma\Delta p$, is equilibrated by gravity force. Thus we have

$$\Sigma\Delta p = gH\varrho'\overline{\varphi}, \tag{14}$$

where φ is the void fraction in the riser, and

$$\overline{\varphi} = \frac{1}{H} \int_0^H \varphi dz. \tag{15}$$

From mass balances in cross-section z

$$\left.\begin{aligned} \dot{m} &= \tfrac{\pi}{4} d^2 \varrho' w, \\ \dot{m}'' &= \varphi \cdot \tfrac{\pi}{4} d^2 \varrho'' S w' = \frac{\pi d}{\Delta h} \int_0^z q dz = x\dot{m}, \\ \dot{m}' &= (1-\varphi) \tfrac{\pi}{4} d^2 \varrho' w' = \dot{m} - \dot{m}'' = (1-x)\dot{m} \end{aligned}\right\} \tag{16}$$

where S denotes the slip ratio, we obtain

$$\frac{w'}{w} = 1 + \frac{4}{\varrho' \Delta h w d} \cdot \left(\frac{\varrho'}{\varrho'' S} - 1 \right) \int_0^z q dz, \tag{17}$$

or

$$\frac{w'}{w} = 1 + \left(\frac{\varrho'}{\varrho'' S} - 1 \right) \cdot x, \tag{18}$$

where

$$x = \frac{4}{\varrho' \Delta h w d} \int_0^z q dz \tag{19}$$

is vapour quality. The slip ratio S is given by definition

$$S = \frac{w''}{w'} = 1 + \frac{w'' - w'}{w'} \tag{20}$$

and according to Wallis in upward bubbly flow it is

$$w'' - w' = w_\infty(1 - \varphi) \tag{21}$$

where

$$w_\infty = 1.53\sqrt{\frac{\varrho' - \varrho''}{\varrho'} gl}\,, \quad l = \sqrt{\frac{\sigma}{g(\varrho' - \varrho'')}} \tag{22}$$

and w_∞ is the velocity of a single bubble in motionless liquid according to Peebles and Garber with changed constant from 1.18 to 1.53 according to Harmathy. Thus

$$S = 1 + \frac{\omega}{w'/w} \cdot (1 - \varphi) \tag{23}$$

where

$$\omega = 1.53\sqrt{\frac{\varrho' - \varrho''}{\varrho'} \cdot \frac{gl}{w^2}} \tag{24}$$

From (16) we have

$$\varphi = \frac{\varrho'}{\varrho''} \cdot \frac{x}{Sw'/w} \tag{25}$$

so that

$$S = 1 + \frac{\omega}{w'/w} \cdot \left(1 - \frac{\varrho'}{\varrho''} \cdot \frac{x}{Sw'/w}\right). \tag{26}$$

Solving for S we obtain

$$S^2 - S\left(1 + \frac{\omega}{w'/w}\right) + \frac{\varrho'}{\varrho''} \cdot \frac{\omega x}{(w'/w)^2} = 0. \tag{27}$$

From (18) we have

$$S = \frac{\varrho'}{\varrho''} \cdot \frac{x}{(w'/w) - 1 + x} \tag{28}$$

Substituting (28) in (27) and rearranging we obtain a cubic equation

$$\left(\frac{w'}{w}\right)^3 - \left(1 - x + \frac{\varrho'}{\varrho''}x\right) \cdot \left(\frac{w'}{w}\right)^2 + \omega(1 - x) \cdot \frac{w'}{w} - \omega(1 - x)^2 = 0 \tag{29}$$

from which the ratio w'/w can be found. Evidently for $x = 0$ the solution is $w'/w = 1$. Knowing w'/w we can calculate S from (28), and φ from (25).

In order to determine the heat flux it is possible to use the method of Rohsenow's according to which the heat transfer coefficient in flow boiling is given by the sum

$$\alpha = \alpha_{PB} + \alpha_{DB} \tag{30}$$

where the first term is the heat transfer coefficient in pool boiling, and may be calculated from author's formula [3], viz.

$$\alpha_{PB} = C_1 q^{2/3},$$

$$C_1 = 0.118\frac{\lambda'}{l}\cdot\left(\frac{l\varrho'}{\varrho''\Delta h\mu'}\right)^{2/3}\cdot\left(\frac{c_p'\varrho'^2 gl}{p_s'\varrho''\Delta h}\right)^{-1/3}\cdot Pr'^{4/9}, \tag{31}$$

$$p_s' = (dp/dT)_{T=T_S}$$

and the second term is the modified forced convection coefficient according to Dittus and Boelter

$$\alpha_{DB} = C_2\left(\frac{w'}{w}\right)^{0.8}, \quad C_2 = 0.019\frac{\lambda'}{d}\cdot\left(\frac{wd\varrho'}{\mu'}\right)^{0.8} Pr'^{1/3}, \tag{32}$$

where w' is the average liquid velocity on the level z, and when $z = 0$, that is at the begin of boiling, it is $w' = w$. Thus

$$\alpha = C_1 q^{2/3} + C_2\left(\frac{w'}{w}\right)^{0.8}. \tag{33}$$

It the film coefficient on the gas side is α_u, and the thermal resistance of tube wall is represented by the term R, then the overall heat transfer coefficient equals

$$k_u = \frac{q}{T_g - T_S} = \left(\frac{1}{\alpha_u} + R + \frac{1}{\alpha}\right)^{-1}, \tag{34}$$

whence we have

$$\alpha = \frac{q}{T_g - Ts - (\frac{1}{\alpha_u} + R)\cdot q} \tag{35}$$

Combining (33) and (35) we obtain finally

$$q = \left[T_g - T_S - \left(\frac{1}{\alpha_u} + R\right)q\right]\cdot\left[C_1 q^{2/3} + C_2\left(\frac{w'}{w}\right)^{0.8}\right]. \tag{36}$$

Thus we have two equations, namely (29) and (36), from which q and w'/w can be found with x given by (19). Evidently, the only procedure possible is the numerical one.

The tube is divided into sections of length Δz. On the level $z = 0$ the heat flux equals q_0, and the liquid velocity is $w_0' = w$. On the level $z_i = i\Delta z$ there is q_i and w_i'. The integral in (19) can be rewritten thus

$$\int_0^{zi} q dz \approx \sum_{j=0}^{j=i-1} \overline{q}_{j,j+1} \Delta z, \tag{37}$$

where

$$\overline{q}_{j,j+1} = \frac{1}{2}(q_j + q_{j+1}), \tag{38}$$

so that

$$\sum_{j=0}^{j=i-1} \overline{q}_{j,j+1} \Delta z = \Delta z \cdot \left(\frac{q_0}{2} + \sum_{j=1}^{j=i-1} q_j + \frac{q_i}{2} \right), \tag{39}$$

and

$$x = \frac{4\Delta z}{\varrho' \Delta h w d} \left(\frac{q_0}{2} + \sum_{j=1}^{j=i-1} q_j + \frac{q_i}{2} \right) = x_i \tag{40}$$

which substituted in (29) enables the calculation of $w' = w_i'$. Putting $w' = w_i'$ and $q = q_i$ in (36) we can thus find q_i. By use of (28) and (25) we can also calculate the slip ratio S_i and the void fraction φ_i. The average void fraction is therefore

$$\overline{\varphi} = \frac{1}{H} \int_0^H \varphi dz \approx \frac{\Delta z}{H} \cdot \left(\frac{\varphi_0}{2} + \sum_{j=1}^{j=k-1} \varphi_j + \frac{\varphi_k}{2} \right), \tag{41}$$

where $\varphi_0 = 0$, and φ_k is the void fraction at the exit from the riser tube.

In order to find the total frictional pressure drop we first calculate the one-phase pressure drop on the length H_0. Neglecting the additional local pressure drop on the bend we have from the Colburn analogy

$$\Delta p_0 = 8 \cdot 0.023 \left(\frac{w_0 d_0 \varrho'}{\mu'} \right)^{-0.2} \cdot \frac{\varrho' w_0^2}{2} \cdot \frac{H_0}{d_0}, \tag{42}$$

and, since in this case it is $w_0 = w$, $d_0 = d$ we obtain

$$\Delta p_0 = 0.092 \left(\frac{w d \varrho'}{\mu'} \right)^{-0.2} \cdot \varrho' w^2 \frac{H_0}{d}. \tag{43}$$

In two-phase flow the frictional pressure gradient will be

$$\frac{dp}{dz} = 0.092 \left(\frac{w' d \varrho'}{\mu'} \right)^{-0.2} \cdot \varrho' w'^2 \phi_{ltt}^2 \cdot \frac{1}{d}, \tag{44}$$

where ϕ^2_{ltt} appears in the theory of Martinelli, Nelson, and Lockhart. In bubbly flow this quantity differs very little from unity, therefore the assumption $\phi^2_{ltt} = 1$ is substantiated. Hence we have in the boiling section of length H

$$\Delta p = 0.092 \left(\frac{wd\varrho'}{\mu'}\right)^{-0.2} \frac{\varrho' w^2}{d} \cdot \int_0^H \left(\frac{w'}{w}\right)^{1.8} dz \tag{45}$$

or

$$\Delta p = 0.092 \left(\frac{wd\varrho'}{\mu'}\right)^{-0.2} \cdot \frac{\varrho' w^2}{d} \cdot \Delta z \cdot \left[\frac{1}{2} + \sum_{j=1}^{j=k-1} \left(\frac{w'_j}{w}\right)^{1.8} + \frac{1}{2}\left(\frac{w'_k}{w}\right)^{1.8}\right]. \tag{46}$$

Thus we obtain

$$\sum \Delta p = \Delta p_0 + \Delta p = 0.092 \left(\frac{wd\varrho'}{\mu'}\right)^{0.2} \cdot \frac{\varrho' w^2}{d} \left\{ H_0 + \int_o^H \left(\frac{w'}{w}\right)^{1.8} dz \right\}, \tag{47}$$

which can be substituted in (14) along with (41).

To illustrate the problem we will discuss a case of flow boiling of water at $p = 1.003 MPa$, $T_S = 180^0 C$. Let the gas temperature be $T_g = 300^0 C$, $\alpha_u = 150 W/m^2 K$, $R = 0.00005$. The tube ID is $d = 38mm$. Using the tables of properties we obtain $C_1 = 6.102$, $C_2 = 6.336.57 w^{0.8}$, $\omega = 0.2246/w$, $\varrho'/\varrho'' = 172.1$, $4/(\varrho' \Delta h w d) = 5.893 \cdot 10^{-8}/w$. We assume $H = H_0 = 3m$ and $\Delta z = 0.2m$. The circulation velocity w is not known, and it must be found from the equation (14). It has been assumed $w = 0.2 m/s$ and $w = 2 m/s$. The results of calculations are given in Table 1. In both cases the heat flux has been nearly constant which has facilitated the calculations. At low velocity there are greater slip ratios and void fractions, and hence the buoyancy term is great. At the same time, due to low velocity, the frictional pressure drop is small. There is therefore a tendency to increase the circulation velocity. In the case of higher velocity ($w = 2m/s$) the slip ratios and void fractions are small, therefore the buoyancy term is much smaller that in the other case, but the frictional pressure drop prevails, therefore there is a tendency to decrease the circulation velocity. Equilibrium will be met at approximately $w = 1.99 m/s$. In the case of $w = 0.2 m/s$ the maximum void fraction is $\varphi = 0.71$. According to Wallis [10] at $\varphi \approx 0.7$ there is a transition to slug flow. Therefore in both cases there is the bubbly flow regime.

An important conclusion to be drawn is that in both discussed cases the heat flux is nearly constant along the tube, and therefore the steam output practically does not depend upon circulation velocity. For if we put $w = 0$, and hence $C_2 = 0$ in (36), and solve for q, the result is $q = 1.7227 \cdot 10^4 W/m^2$ and is smaller by 1.1% compared with the result obtained at $w = 0.2 m/s$, and by 2.6% at $w = 2m/s$.

Obviously the circulation velocity may be interesting which regards the noise produced, and vibrations of the riser tubes.

Table 1

Results of exemplary calculations for water $p = 1.003\,MPa, d = 38\,mm$, $H = 3\,m$

$w, m/s$	0.2				2.0			
z, m	x	w'/w	S	φ	x	w'/w	S	φ
0	0	1	2.1230	0	0	1	1.1123	0
0.2	0.001026	1.0898	1.9441	0.0833	0.000104	1.0160	1.1114	0.0159
0.4	0.002053	1.1960	1.7836	0.1656	0.000208	1.0321	1.1080	0.0313
0.6	0.003078	1.3194	1.6429	0.2444	0.000312	1.0485	1.1011	0.465
0.8	0.004104	1.4589	1.5255	0.3174	0.000416	1.0647	1.0995	0.0612
1.0	0.005130	1.6123	1.4299	0.3830	0.000520	1.0812	1.0951	0.0756
1.2	0.006156	1.7764	1.3538	0.4405	0.000624	1.0978	1.0915	0.0896
1.4	0.007182	1.9480	1.2939	0.4904	0.000728	1.1141	1.0908	0.1031
1.6	0.008208	2.1247	1.2469	0.5332	0.000832	1.1308	1.0877	0.1164
1.8	0.009234	2.3048	1.2094	0.5701	0.000936	1.1474	1.0860	0.1293
2.0	0.01026	2.4865	1.1797	0.6019	0.001040	1.1641	1.0835	0.1419
2.2	0.01129	2.6700	1.1557	0.6293	0.001144	1.1810	1.0809	0.1542
2.4	0.01231	2.8518	1.1365	0.6537	0.001248	1.1978	1.0789	0.1662
2.6	0.01334	3.0363	1.1201	0.6750	0.001352	1.2150	1.0756	0.1780
2.8	0.01436	3.2183	1.1069	0.6937	0.001456	1.2318	1.0743	0.1894
3.0	0.01539	3.4021	1.0956	0.7106	0.001560	1.2488	1.0717	0.2006
$\overline{\varphi}$	0.4558				0.1066			
$\int_0^3 (w'/w)^{1.8} dz$	12.3518				3.8083			
$gH\varrho'\overline{\varphi}$	11887.8				2780.2			
$\sum \Delta p$	155.3				4345.4			
$q, W/m^2$	17423 + 0.27%				17684 + 0.13%			

4. Circulation in a drum boiler

We discuss now the case illustrated by Fig. 1. The downcomer tube is usually adiabatic therefore the temperature in the chamber d, T_0, is defined by equation (10) corrected for the number n of riser tubes per one downcomer, i.e.

$$T_0 = T_S - n\frac{4(T_S - T_i)d}{\varrho' \Delta h w_0 d_0^2}\int_0^H q dz. \tag{48}$$

Since $T_0 < T_S$, the lower portion of a riser tube is subcooled, and on the initial length ΔH liquid must be heated to the temperature of incipient boiling $T_{IB} = T_S + \delta T_{IB}$ on the wall. The average temperature of liquid will be somewhat lower than T_S when subcooled boiling starts. Therefore we assume that the developed boiling starts at the place where the average temperature is that of saturation, and neglect subcooled boiling whose contribution to the vapour output is small. Thus we have

$$\Delta H = \frac{c_p^{'}\varrho^{'} wd}{4k_u} ln \frac{T_g - T_0}{T_g - T_S}, \tag{49}$$

where

$$k_u = \left(\frac{1}{\alpha_u} + R + \frac{1}{\alpha_{DB}^*}\right)^{-1}, \quad \alpha_{DB}^* = 0.023\frac{\lambda'}{d}\left(\frac{\varrho' wd}{\mu'}\right)^{0.8} Pr'^{1/3}. \tag{50}$$

For the example in the preceding section there was $n = 1$ and $d = d_0$. Assuming $T_i = 160^oC$ we have obtained $\Delta H = 0.138m$ for $w = 0.2$ and $\Delta H = 0.130m$ for $w = 2m/s$. Hence it would be $H_0 = H + \Delta H = 3.138m$ or 3.130 m.

The calculation of pressure drops is more complicated than in the previously discussed case of a simple loop, because local flow resistances must be taken into account. Also the problem of maldistribution of flow arises, because the mass flow rates in riser tubes which are nearer to the downcomer are greater than those in tubes which are far from the downcomer. It is assumed that although the maldistribution exists either by this reason or another, it is very small, especially when tubes are long and of small diameter. Obviously the problem could be considered, but it complicates and obscures the analysis, so that, to cut a long story short, it will be assumed that inlet velocities to each riser tube are equal. Denoting these by w as before, and that in the downcomer by w_o, we have for n riser tubes

$$w_0 = nw(d/d_0)^2. \tag{51}$$

The pressure drop along the downcomer is given by (42). Taking into account the local resistances at the inlet, and outlet, we must add the quantity $1.35\varrho' w_o^2$. The pressure drop along the boiling section is given by (45). It must be enlarged by adding of the pressure drop along the preheating section, which equals $\Delta p_0 \cdot \Delta H/H_0$, and of the local resistances at inlet and outlet. The former equals $0.85\varrho' w^2$, the latter $0.5\varrho' w_{exit}^2$, where $w_{exit} = w_k'$ is the outlet velocity of liquid.

Summing up the frictional pressure drops we obtain

$$\sum \Delta p = 0.092\left(\frac{w_0 d_0 \varrho'}{\mu'}\right)^{-0.2} \cdot \frac{\varrho' w_0^2}{d_0}(H + \Delta H) +$$

$$1.35\varrho' w_0^2 + 0.092\left(\frac{wd\varrho'}{\mu'}\right)^{-0.2} \cdot \frac{\varrho' w^2}{d}\left[\Delta H + \int_0^H (w'/w)^{1.8} dz\right] +$$

$$0.85\varrho' w^2 + 0.5\varrho' w^2 (w_k'/w)^2. \tag{52}$$

Substitution in (14) yields the equation whose solution is the circulation velocity w.

In the case when, due to maldistribution of flows, the inlet velocities are not equal then

$$w_0 = \sum_{m=1}^{m=n} w_m (d/d_0)^2 \tag{53}$$

and substitution of w_m instead of w in (52) gives n equations of that type. In such a case the mean void fraction in each tube may be different, so that instead of (14) we have

$$\left(\sum \Delta p\right)_m = gH\varrho'\overline{\varphi}_m. \tag{54}$$

5. Circulation in an evaporator with central downcomer tube

The basic design is shown in Fig. 2. The difference in comparison with the previously analyzed boiler from Fig. 1 is that the downcomer tube is heated with the same condensing steam as the riser tubes. This fact has undesired consequences which were mentioned previously, namely, vibrations of liquid column in the downcomer, and increase of frictional pressure drop. In order to avoid these shortcomings during boiling it is proposed to insert a liner of thin sheet in the downcomer tube as shown in Fig. 6.

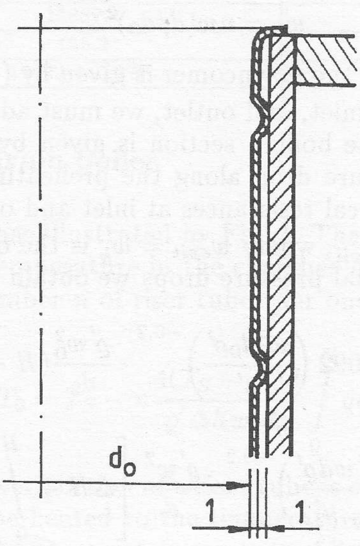

Fig. 6. Isolating liner in downcomer tube

A thin layer of gas or vapour acts as isolation so that the downcomer tube may be treated as adiabatic in the final phase of boiler operation. In such a case, during unsteady operation with only removal of vapour as it is used in sugar industry, the inlet temperature is equal to that of saturation, $T_O = T_S$, so that $\Delta H = 0$. The same procedure as in the preceding section may be applied for short periods. At the end of each period the mass balance must be done, and hence the new value of sugar content can be found. This is very important because the properties of solution depend upon the sugar content. During boiling the level of liquid is sinking, and therefore the static pressure in lower plenum diminishes also. This fact has a slight influence on properties which usually can be neglected.

At the begin of operation the temperature of liquid is well below saturation and the initial phase consists in heating of the liquid. Circulation in this case is due to thermal expansion.

The initial period of operation is thus governed by non-stationary natural convection. If tubes are heated with condensing steam the film coefficient α_k is the same for the downcomer as for the risers. Taking into account the thermal resistance of wall, R, we can use the corrected heat transfer coefficient α_k'

$$\alpha_k' = \left(\frac{1}{\alpha_k} + R\right)^{-1} \tag{55}$$

The film coefficient inside the tubes, α, is the same for the downcomer tube as for the risers. Therefore heat supplied to the liquid in time Δt equals

$$Q = \frac{T_k - T_i}{\dfrac{1}{\alpha_k'} + \dfrac{1}{\alpha}} \cdot \pi d H_0 \Delta t, \tag{56}$$

where T_k is temperature of condensing steam, and T_i the initial temperature of liquid. The final temperature $T = T_i + \Delta T$ can be found from the enthalpy balance

$$Q = \frac{\pi}{4} d^2 H_0 \varrho_i c_{pi} \Delta T \tag{57}$$

Hence

$$\Delta T = \frac{4\Delta t}{\varrho_i c_{pi} d} \cdot \frac{T_k - T_i}{\dfrac{1}{\alpha_k'} + \dfrac{1}{\alpha}} = T - T_i \tag{58}$$

The indices i refer to the initial temperature T_i.

The temperature increase in the downcomer tube can be found analogously, viz.

$$\Delta T_O = \frac{4\Delta t}{\varrho_i c_{pi} d_O} \cdot \frac{T_k - T_i}{\dfrac{1}{\alpha_k'} + \dfrac{1}{\alpha}} = T_O - T_i \tag{59}$$

If the initial density is ϱ_i, the densities after a time Δt will be different in riser tubes and in the downcomer tube. Namely

$$\varrho = \varrho_i(1-\beta\Delta T), \quad \varrho_0 = \varrho_i(1-\beta\Delta T_0), \tag{60}$$

where β is the coefficient of thermal (cubical) expansion. The thus created buoyance force $gH_0(\varrho_0 - \varrho)$ is equilibrated by the frictional pressure drop, so that we have

$$\sum \Delta p = gH_0\varrho_i\beta(\Delta T - \Delta T_0). \tag{61}$$

Substituting (58) and (59) we obtain

$$\sum \Delta p = \frac{4g\beta}{c_{pi}} \cdot H_0 \frac{T_k - T_i}{\frac{1}{\alpha_k^{'}} + \frac{1}{\alpha}} \cdot \left(\frac{1}{d} - \frac{1}{d_0}\right) \Delta t \,. \tag{62}$$

The frictional pressure drop is to be calculated from (52) on putting $\varrho^{'} = \varrho_i$, $w^{'} = w$, $H + \Delta H = H_0$. The result is

$$\sum \Delta p = 0.092 \left(\frac{w_0 d_0 \varrho_i}{\mu'}\right)^{-0.2} \cdot \frac{\varrho_i w_0^2}{d_0} H_0 +$$

$$1.35\varrho_i w_0^2 + 0.092\left(\frac{wd\varrho_i}{\mu^{'}}\right)^{-0.2} \cdot \frac{\varrho_i w^2}{d} H_0 + 1.35\varrho_i w^2 \tag{63}$$

Substituting (51) we obtain

$$\sum \Delta p = \left\{ 0.092\left(\frac{wd\varrho_i}{\mu^{'}}\right)^{-0.2} \cdot \frac{H_0}{d}\left[1 + n^{1.8}\left(\frac{d}{d_0}\right)^{4.8}\right] + \right.$$

$$\left. 1.35\left[1 + n^2\left(\frac{d}{d_0}\right)^4\right]\right\} \varrho_i w^2 \,, \tag{64}$$

which can be substituted in (62). Solving for w we obtain

$$w = \left\{ \frac{4g\beta(T_k - T_i)H_0}{\varrho_i c_{pi}\left(\frac{1}{\alpha_k^{'}} + \frac{1}{\alpha}\right)} * \right.$$

$$\left. \frac{\left(\frac{1}{d} - \frac{1}{d_0}\right)\Delta t}{0.092\left(\frac{wd\varrho_i}{\mu^{'}}\right)^{-0.2}\frac{H_0}{d}\left[1 + n^{1.8}\left(\frac{d}{d_0}\right)^{4.8}\right] + 1.35\left[1 + n^2\left(\frac{d}{d_0}\right)^4\right]} \right\}^{1/2} . \tag{65}$$

This formula should be used in the following way. All the coefficients and material properties are taken for the initial temperature T_i. The time step Δt should be so chosen that the values of properties for the final temperature T, calculated from (58), do not differ essentially from the initial values. Thus, after a number of iterations, the velocity w can be found. Note that during the initial time step we have to do with natural convection and the coefficient α must be found for the case of vertical plate of height H_0. In the second time step, and in all next steps, there is combined convection with co-current in riser tubes, and countercurrent in the downcomer. The former phenomenon is pretty well elaborated and there are working formulae at hand. The latter, however, is less studied from the heat transfer point of view as well as that of the pressure drop.

If the downcomer is isolated e.g. with help of the device shown in Fig.6 the situation becomes clear. There is $\Delta T_0 = 0$, whence the term $(1/d_0)$ in (62) and (65) can be cancelled.

The calculations step by step with help of (58) and (65) end when $T = T_S$. From this moment another method must be applied, namely that outlined in the preceding section. It seems that this period is relatively shorter than that of initial preheating therefore any augmenting of circulation by means of a propeller situated in the downcomer or injection of air or steam into riser tubes, is especially useful before boiling begins. If the former method is applied we have to do with forced convection followed by forced convection boiling in which natural circulation plays a secondary role. As to the latter method we can calculate its performance using modified procedure from the preceding section in which the vapour mass flow rate is replaced by the sum of the latter plus the additional rate of driving medium (air or steam).

6. Gas heated reboiler

This type is shown in Fig. 3. It is convenient to assume the distance Δz equal to H/k when there are $(k-1)$ baffles. In every section between the baffles

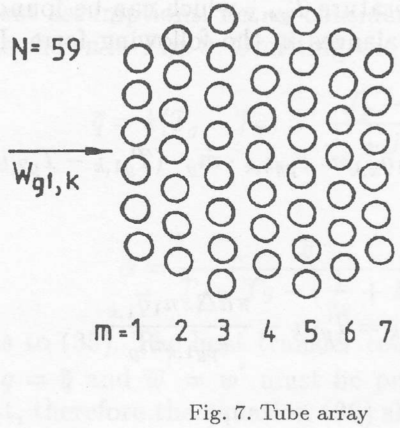

Fig. 7. Tube array

cross-flow can be assumed. For each row of tubes as shown in Fig. 7 different gas temperatures $T_{gm,i}$, gas velocities $w_{gm,i}$ and film coefficients $\alpha_{gm,i}$ appear which must be calculated separately.

Usually a staggered array of tubes is used as it is seen on Fig.7. The rows of tubes, m, are numbered off from the left, n_m being the number of tubes in the row. The total number of tubes is thus

$$N = \sum m n_m. \tag{66}$$

In the case of countercurrent the entering hot gas attacks to first row of tubes in the upper section k with velocity $w_{g1,k}$ which can be calculated from the gas mass flow rate $\dot{m}_g$ knowing the inlet gas temperature $T_{g1,k}$. Taking into account that $\varrho_g = p_g/(R_g T_g)$, where R_g is the individual gas constant, and that the flow area is proportional to (n_m+1) we have

$$\dot{m}_g \sim (n_m+1)\cdot\frac{p_g}{R_g}\cdot\frac{w_{gm,i}}{T_{gm,i}} \sim (n_m+1)\frac{w_{gm,i}}{T_{gm,i}} = constant, \tag{67}$$

whence

$$(n_m+1)\frac{w_{gm,i}}{T_{gm,i}} = (n_1+1)\frac{w_{g1,k}}{T_{g1,k}}. \tag{68}$$

The heat transfer coefficient from the gas to the surface of tube of outside diameter d_e can be calculated from the formula

$$\alpha_{gm,i} = C_g\frac{\lambda_{gm,i}}{d_e}\left(\frac{w_{gm,i}\cdot d_e\varrho_{gm,i}}{\mu_{gm,i}}\right)^{0.6}, \tag{69}$$

where the constant C_g can be found in references e.g. [4], and $\lambda_{gm,i}$, $\varrho_{gm,i}$, and $\mu_{gm,i}$ depend only upon temperature $T_{gm,i}$ which can be found for every row and every section from enthalpy balances in the following form. For the first row of upper section number k

$$\dot{Q} = \pi d\Delta z n_1\overline{q}_{k,k} = c_{pgk,k}\cdot\dot{m}_g\cdot(T_{g1,k}-T_{g2,k}), \tag{70}$$

whence

$$T_{g2,k} = T_{g1,k} - \frac{\pi d\Delta z n_1\overline{q}_{1,k}}{c_{pg1,k}\dot{m}_g} \tag{71}$$

Similarily we have

$$\left.\begin{aligned}
T_{g3,k} &= T_{g2,k} - \frac{\pi d\Delta z \cdot n_2\overline{q}_{2,k}}{c_{pg2,k}\dot{m}_g},\\
T_{g4,k} &= T_{g3,k} - \frac{\pi d \cdot \Delta z n_3\overline{q}_{3,k}}{c_{pg3,k}\cdot \dot{m}_g},\\
&\vdots\\
T_{gm,k} &= T_{g(m-1),k} - \frac{\pi d \cdot \Delta z n_{m-1}\overline{q}_{m-1,k}}{c_{pg(m-1),k}\cdot \dot{m}_g},\\
T_{gm,k-1} &= T_{gm,k} - \frac{\pi d\Delta z n_m\overline{q}_{m,k}}{c_{pgm,k}\dot{m}_g},\\
&\vdots
\end{aligned}\right\} \tag{72}$$

$$\left.\begin{aligned}
T_{g1,1} &= T_{g1,2} - \frac{\pi d\Delta z n_1\overline{q}_{1,2}}{c_{pg1,2}\dot{m}_g},\\
T_{g2,1} &= T_{g1,1} - \frac{\pi d\Delta z n_1\overline{q}_{1,1}}{c_{pg1,1}\dot{m}_g},\\
&\vdots\\
T_{gm,1} &= T_{g(m-1),1} - \frac{\pi d\Delta z n_{m-1}\overline{q}_{m-1,1}}{c_{pg(m-1),1}\dot{m}_g},
\end{aligned}\right\} \tag{73}$$

for even number of baffles, or odd k, and

$$\left.\begin{aligned}
T_{gm,1} &= T_{gm,2} - \frac{\pi d\Delta z n_m\overline{q}_{m,2}}{c_{pgm,2}\dot{m}_g},\\
T_{g(m-1),1} &= T_{gm,1} - \frac{\pi d\Delta z n_{m,1}\overline{q}_{m,1}}{c_{pgm,1}\dot{m}_g},\\
&\vdots\\
T_{g1,1} &= T_{g2,1} - \frac{\pi d\Delta z n_{2,1}\overline{q}_{2,1}}{c_{pg2,1}\dot{m}_g},
\end{aligned}\right\} \tag{74}$$

for odd number of baffles, or even k. The quantities $\overline{q}_{m,k}$ are average heat fluxes.

To calculate the latter we use the method outlined in section 2 slightly modified to fit the present assumptions. Let us consider a tube element in row number m on the level k as shown in Fig. 8. Omitting the indices we write down the heat balance

$$\overline{q} = k(T_g - T_S) = \frac{T_g - T_S}{\frac{1}{\alpha_g} + R + \frac{1}{\alpha}} \tag{75}$$

whence we have

$$\alpha = \frac{\overline{q}}{T_g - T_S - (\frac{1}{\alpha_g} + R)\overline{q}}, \tag{76}$$

which is analogous to (35). The heat transfer coefficient in flow boiling is given by (33) in which $q = \overline{q}$ and $\overline{w}' = w'$ must be put. In cach row the inlet liquid velocity is different, therefore the equation (36) should have the following form

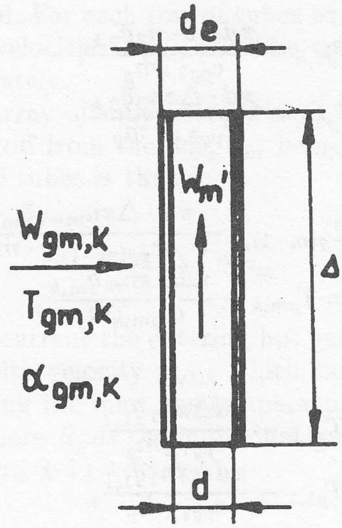

Fig. 8. Element m, k

$$\overline{q}_{m,k} = \left[T_{gm,k} - T_S - \left(\frac{1}{\alpha_{gm,k}} + R\right) \cdot \overline{q}_{m,k}\right] * \left[C_1 \overline{q}_{m,k}^{-2/3} + C_{2,m}\left(\frac{\overline{w}'_{m,k}}{w_m}\right)^{0.8}\right], \tag{77}$$

in which formula C_1 is constant since it depends upon pressure only. The liquid velocities $\overline{w}'_{m,k}$ and void fractions $\overline{\varphi}_{m,k}$ can be calculated by methods developed previously. Afterwards the average void fraction over the tube length can be calculated, viz.

$$\overline{\overline{\varphi}}_m = \frac{\Delta z}{H} \sum_k \overline{\varphi}_{m,k}. \tag{78}$$

The circulation velocity in a row, w_m, should be calculated from the modified equation (14), namely

$$\sum \Delta p = \Delta p_0 + \Delta p_m = g H \varrho' \overline{\overline{\varphi}}_m. \tag{79}$$

In the present case the downcomer tube length may differ from that of the riser tubes. The pressure drop in the vessel itself may be neglected, therefore only the frictional pressure drop along the downcomer length H_0 plus local flow resistances of the inlet and outlet must be taken into account. We have thus

$$\Delta p_0 = 0.092(w_0 d_0 \varrho'/\mu')^{-0.2} \cdot \frac{\varrho' w_0^2}{d_0} H_0 + 0.65\varrho' w_0^2 \tag{80}$$

where

$$w_0 = \sum_m n_m w_m (d/d_0)^2. \tag{81}$$

The formula (80) is simplified in comparison with (52) because there are no changes of flow direction.

The frictional pressure drop in the riser tube is equal to

$$\Delta p_m = 0.092(w_m d\varrho'/\mu')^{-0.2} \cdot \frac{\varrho' w_m^2}{d} \sum_k (\overline{w}'_{m,k}/w_m)^{1.8} +$$

$$+0.15\varrho' w_m^2 + 0.5\varrho' w_m^2 \cdot (\overline{w}'_{m,exit}/w_m)^2 \tag{82}$$

Also here the local flow resistances due to changes of flow direction do not appear.

Thus the problem is formulated, but its solution is very complex. It is seen that analytical procedure is difficult, because the governing equations are not linear, although algebraic all of them. The number of unknowns is pretty great, so that numerical procedures must be applied.

There are $mk - 1$ unknown gas temperatures $T_{gm,k}$, and the same number of equations (71), (72), (73) or (74). The properties $\varrho_{gm,k}{}'\mu_{gm,k}{}'$ and $\lambda_{gm,k}$ depend upon $T_{gm,k}$ only, and can be found with help of tables or formulae, therefore they are not treated as unknowns, being represented by $T_{gm,k}$. Furthermore there are $mk - 1$ unknown heat transfer coefficients $\alpha_{gm,k}$, and the same number of unknown gas velocities $w_{gm,k}$; they are both defined by $2(mk - 1)$ equations (68) and (69). There are also mk unknown heat fluxes $\overline{q}_{m,k}$, and corresponding mk equations (77). Also mk unknowns $\overline{w}'_{m,k}/\overline{w}_m$ defined by special procedures, along with $S_{m,k}$ and $\overline{\varphi}_{m,k}$. There are m unknowns $\overline{\overline{\varphi}}_m$ and w_m, also m equations (82) for pressure drop. Furthermore there is unknown Δp_0, and unknown w_0. At hand are m closing equations (79). Thus the total number of uknowns is equal to that of equations, namely $6mk + 4m - 1$. E.g. the design shown in Figs 3 and 7 has 59 tubes in 7 rows, and 4 baffles, so that $m = 7$, $k = 5$, and consequently the number of equations is 237.

7. Conclusions

Design of evaporators with natural circulation and calculation of their performance is at first sight a very complex problem. On closer inspection it may be concluded that at proper maintenance and safe operation conditions boiling occurs in bubbly flow as a rule. Bubbly flow offers a possibility of simplifications in characteristics of general 2-phase flow and boiling theory. First in bubbly flow the vapour velocities relative to liquid are known, and this enables the analytical calculation of slip ratios and void fractions. Secondly, the frictional pressure drop can be calculated on putting the Martinelli-Nelson-Lockhart factor $\phi_{ltt} = 1$. Thus the essential basis for calculation becomes only the formula for heat transfer coefficient in flow boiling.

In the presented analysis the well established formula of Rohsenow has been used. Also after Rohsenow the heat transfer coefficient in pool boiling has been put proportional to the two-thirds power of heat flux. The working formula, however, is taken from the author's report [3]. That formula, based on experiments of Cichelli and Bonilla for water, has been applied [5] with satisfactory results to refrigerants also. Therefore the presented method can be recommended for design practice.

It is thus seen that the presented theory is based on the set of equations (30), (31) and (32) determining the heat transfer coefficient, and (44) determining the two-phase frictional pressure drop. The rest are balances and mathematical considerations.

As known, there are numerous proposals according to the form of those four formulae, to mention by name McAdams, Kutateladze [6], Bergles and Rohsenow [7], Mikielewicz [8] among other autors. Some of those relationships have been elaborated for particular fluid, or group of fluids, as a very accurate formula of Mikielewicz, Iwicki and Mikielewicz [9] based on more than 2000 points of experiments on refrigerants R11, R12 and R22. It is therefore evident that the used here basic formulae can be replaced by any other compatible set if one is convinced that it is appropriate for the case , or simply if one got accustomed to use it. The rest can be left unchanged except of particular mathematical tricks.

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Wrzenie i przepływ dwufazowy w pionowych rurach parowników z naturalną cyrkulacją

Streszczenie

Problem tytułowy został rozpatrzony dla czterech typów parowników, stosowanych w kotłach parowych, chłodnictwie, przemyśle cukrowniczym i chemicznym. Na wstępie przeanalizowano przepływ w rurze opadowej i stwierdzono, że gdy jest ona ogrzewana, to może być źródłem wibracji. Podano warunki uniknięcia tego zjawiska. W dalszych rozważaniach podano metodę obliczania cieplnego parowników, a zwłaszcza prędkości cyrkulacji, co zostało zilustrowane przykładami liczbowymi.