IWM/CDS
2213

"STEAM LOCOMOTIVE BOILER WATER CIRCULATION - A FIRST SEMI-QUANTITATIVE APPROACH."


by.

L. D. PORTA, C.Eng,
F. Inst. F.; M.I. Mech. E.; CAI,
Chief, Dept. Thermodynamics, INTI
ARGENTINA

Adress: Azara 1557
Banfield 1828
ARGENTINA



Issued: Buenos Aires, April 1979.
Revised, Buenos Aires, June 1981.

Summary.

While water circulation is a most important issue in water tube boilers (where it is quantitatively mastered), it has been a matter of myth, vagaries and conjecture in the case of locomotive and other firetube [boilers.

Some measurements carried out during water treatment tests for the FCGB (Argentina), coupled to elementary reasoning and knowledge of boiling heat transfer phenomena now available, have permited to scheme out a reasonable picture of what life is in the boiler inside.

TGS (Advanced, Third Generation, Steam locomotive technology), aiming to keep to the STEPHENSONian boiler working at 60 ate pressure, impose the need to clear up circulation parameters. It is believed that the present paper represents the first step of a long (happy) ladder leading to better future boilers as required to operate in an energy starving world.

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## Figures.

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1 Introduction. (*)

Stationary water tube boiler water circulation is a most important subject in water tube boiler technology, and the technical literature offers a long list of references on it. This contrasts with the vagaries told about in the field of fire tube boilers, the locomotive type being no exeption. No water tube boiler designer will today rely on guess work on this area, yet thousands of succesful firetube boilers have been heated according to thumb rules supposedly based on experience, apparently with too many ~~not~~ troubles even in poor water treatment days.

The time has come to start investigting the subject, particularly in view of the needs imposed by advanced design. The writter believes that the STEPHENSONian boiler has merits on its own to be proposed for pressures at present aiming 60 atc (850 psig); but, in order to achieve that goal, vagaries and thumb rules must be substituted by better engineering; and conviction and beliefs replaced by information, calculations and checks against well established engineering science.

These matters are closely allied to water side heat transfer and two-phase flow, two areas which are presently receiving considerable attention because of the exacting requirements imposed by nuclear reactor design. The appalling stream of knowledge coming on boiling heat transfer must also be incorporated to steam locomotive technology in order to eradicate years of myths and vagaries too. Efforts in that direction is currently being carried out (1979) (PORTA (1)).

(*). Numbers in a square $\boxed{00}$ refers to additional comment made in Appendix A1.

This paper points towards being useful in coming boiler design; although use is made of knowledge referring to past practice, full advantage is taken from the now available water treatment technology (2), namely: (i) scale free heating surfaces can now be guaranteed, and (ii) very pure steam is produced under all circumstances thanks to the now available antifoams. These markedly influence circulation because (a) fewer steam bubbles of larger diameter are produced at the heating surfaces; (b) these bubbles coalesce and grow into larger ones on its way to the steam space, and (c) practically no foam lays on the top of "solid" water.

Condition (i) means that water passages areas are available in full and not reduced by scale, and that its circulation hindering roughness has been swept away from the picture; condition (ii) means that the steam space is no longer a low velocity moisture settling chamber or a froth containing volume.

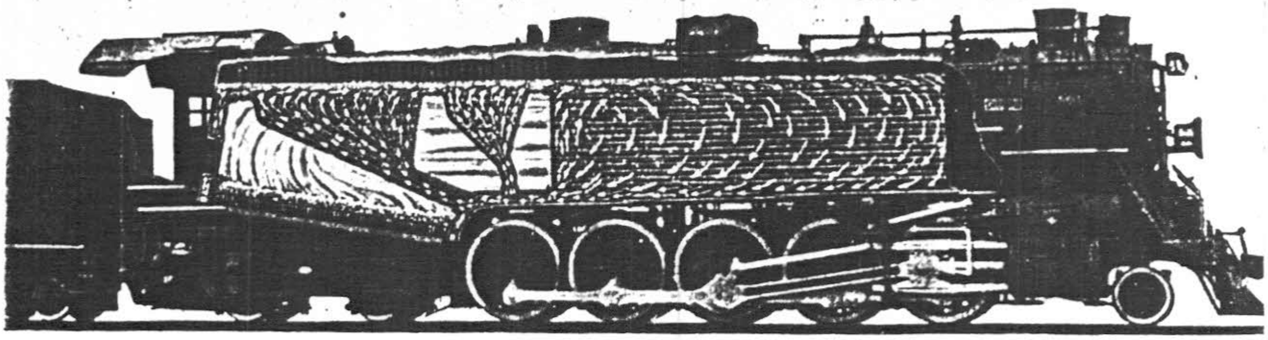

Fig. A—Diagram of water circulation in a boiler equipped with Nicholson Thermic Syphons

Boiler circulation in a locomotive boiler provided with NICHOLSON thermic syphons

Apparently, this phantom view was drawn as a result from observations made in a transparent boiler model operating at atmospheric pressure (reported by CHAPELON, Ref (5)). The understanding is that the syphons superimpose their action on general circulation as described in this paper. Circulation in the back sheet and crown are exactly as described.

2 <u>A qualitative description of the flow pattern in a typical boiler.</u>

What follows is the best image one can have before some quantitative definition can be achieved. Referring to Fig. 1, the boiler can be considered divided in two parts: (i) the back, including the first .. metre of tube length, in which 90% of the steam is produced, and (ii) the front part, in which the remaining 10% steam is produced, (*) therefore configurating a rather calm section in which steam bubbles contribute to little swelling. This is enhanced by the location of the clack valve because relatively cold feed water, in being mixed with the general flow, leads to subcooled boiling over a large part of the small tubes as it will be shown later.

This 90 - 10 % distribution sets a clear distinction: the back is a turmoil region in which most of water swelling occurs because most of the steam bubbles are produced occupying some 25% of the volume there (average figure). This low density water - steam mixture determines a strong (generally upwards) circulation stream on the firebox walls and near the tube plate, sucking a denser mixture from the bottom of the barrel and originating a strong forward stream over the top of the firebox which extends forwards over the whole evaporation plane. Circulation is, therefore, fairly definite, as shown in the familiar phantom view produced by NICHOLSON thermic syphon makers (however that flow pattern is not due to the syphon, although it contributes to enhance it).

(*) These unexpected figures will be justified later.

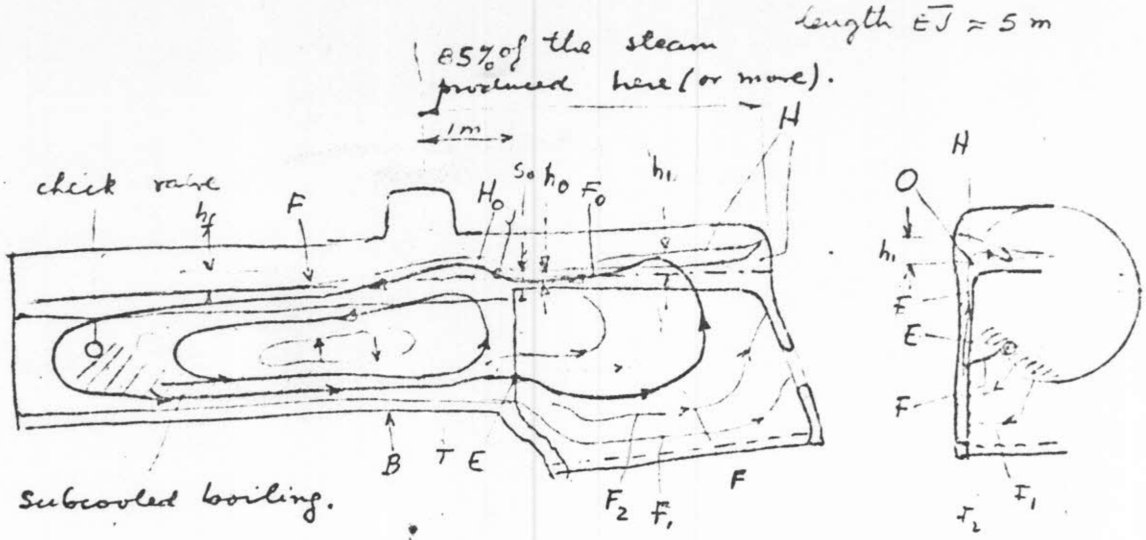


Fig. 1 Circulation flow pattern.

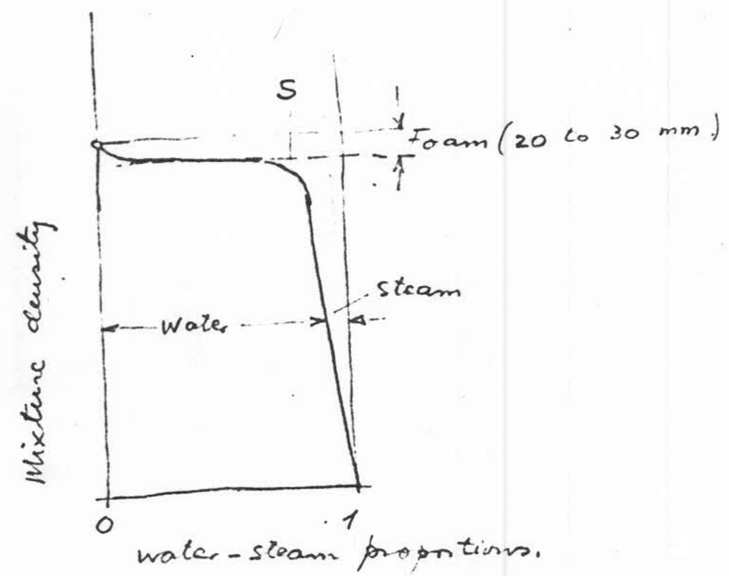


Fig. 2 Schematic view of the variation of steam-water proportions with height.



5,1

Fig 1 shows the two most important (or average) streamlines: F corresponds to the firebox and T to the tubes. The back wall is fed by streamlines $F_1$ and $F_2$ sweeping backwards the space above the foundation ring. Ordinarily, most of firebox evaporation, both with coal or oil, is produced at the bottom part because heat liberation, flame emmissivity and temperature are highest there. When the flow enters the firebox area at E, it contains relatively few bubbles; when discharged at the top of the legs at O it consists of a relatively light mixture surging upwards at high velocity, therefore leading to a hump H extending laterally all along the sides of the (BELPAIRE) steam space, and also crosswise against the outer back plate. 3 At full steaming rate, the volume dryness fraction, which on average is some 25% over the whole firebox, is probably 5 to 10% at the bottom over the ring, and perhaps 50% at the water space discharge outlet O. This figure, however, is far from the critical limit concerning adequate boiling cooling (1)(4).

Therefore the region above the firebox crown plate is fed by three sides by a strong water surge, which precipitates forwards under the gravity potential resulting from the height $h_1 > h_0$. If $h_1 - h_0$ is about 0,2 m, the potential velocity $\sqrt{2gh}$ is somewhere around 2 m $s^{-1}$! Incidentally, the jump existing against the back plate leads to false water level cock readings as demonstrated in the early American tests which resulted in the official American water gauge arrangements (3)(11), (See Appendix A2).

The forward going stream $F_o$ hits the upward stream, thereby leading to a hump $H_o$, wherefrom the combined stream proceeds forwards under the available head $h_f$. This combined flow probably has a velocity of about 1 m s' under the available head $h_f$.

So far, we have been speaking about a well defined water surface, which is not the case. Referring to Fig. 2 showing qualitatively how the water-steam proportion goes with height, that surface would correspond to that of the plane S. Above it, there is a layer of foam which experiments show to amount to some 20 to 30 mm $\boxed{1}$ when antifoams are used. This foam layer tends to even out the various jumps and the turmoil occurring below, as per the familiar image of the household washing machine.

The forward going stream F resulting from the stream $F_o$ coming from the firebox, and the stream T coming from the tube bundle, has its backward counterpart in the stream B running along the bottom of the barrel. B is assumed to be decomposed in T and F. Although most of the heat transfer obtaining in the tube bundle occurs at the small tubes located below, this does not corresponds to the evaporation, since a large part of that heat transferred in the small tubes goes to heat up the relatively cooler water getting in through the clack valve. Besides, not very much heat comes out from the gases in the second half of tube length because of the reduced temperature differential; hence, one can say that the front half and the bottom part of the barrel contain a high density water-steam mixture. This is very sound since it acts as "downcomer" to the back part... ///

///... of the boiler. Therefore, most of the swelling and bubbles exist at the back end.

Concerning the throat plate, it is probably that, in spite of the intense heat transfer obtaining there, the flow is downwards as shown in Fig. 1. It may be that under certain circumstances, relatively cold water reaches this place, therefore leading to severe thermal stresses.

The pressure at the bottom of the firebox is greater than at the top because of a static head amounting to nearly 2 m, hence ≈ 0,15 at. This means that if water is at saturation temperature at the bottom, part of it flashes on coming up, adding to an increase in bubble diameter (see [2]). Conversely, in downward flow regions, bubbles collapse. Since the total boiler flow consisting in streams F, B, etc. is hundred fold the steam production of the boiler, the steam quantities involved in flashing and collapsing are quite important in spite of the small quantity of heat per kg brought into play. This means that probably a sizable part of the steam evolved at the heating surfaces of the large tubes will never be released at the front part of the evaporation plane; on being entrained downwards, bubbles produced there will collapse, yet this steam will reappear at the back region. One can say, therefore, that perhaps the whole of the steam is released at the back part of the boiler, and certainly more than 50% at the firebox even if the heat transfer from the flames corresponds to a smaller proportion. This increases the volume dryness fraction at O (Fig. 1).

Although the velocity of bubbles relative to water is not negligible (approximately $0{,}25\ m\ s^{-1}$), the flow is probably so turbulent that in a first analysis steam bubble streamlines may be taken as coincident with water streamlines. Of course, this is not so at the separation plane and some idea about such release can be obtained when comparing the above upward velocity with a horizontal velocity of, say, 1 to 2 $m\ s^{-1}$; this results in an angle of 15 to 7°.

One may expect that the local foam layer thickness bears some relationship with the local bubble rising velocity. This foam layer is constantly pushed forwards, and in the case where anti-foams are used, it is probably that little, if any, foam obtains at the front end where no steam bubbles are evolved.

The preceding description leads to the conclusion that in the steam space only the back half of it is used. Since it no longer serves as settling chamber for entrained drops (PORTA (2)), a sort of low velocity "wind" blows from the back towards the base of the dome (*) The nominal residence time of the steam in the chamber is some 5 s; but since only the back part is utilized, it comes to be some 2.5 s; If the swelled level and the foam volume is accounted for, the above figure reduces to something around 1.5 s. or perhaps to 0,5 s when the boiler is operated with a higher mark in the glass. Under such circumstances the "wind" velocity above the hump $H_0$ may be some $4\ m\ s^{-1}$, and one may wonder whether no foam particles can be entrained as per the action of natural wind over the foam over sea waves.

(*) Velocity of the order of $2\ m\ s^{-1}$.

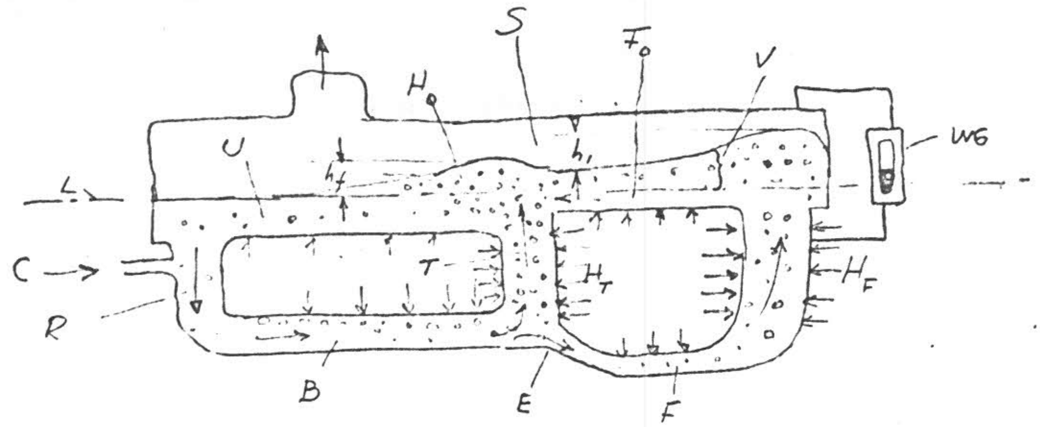


Fig. 3. A simplified scheme of water circulation.

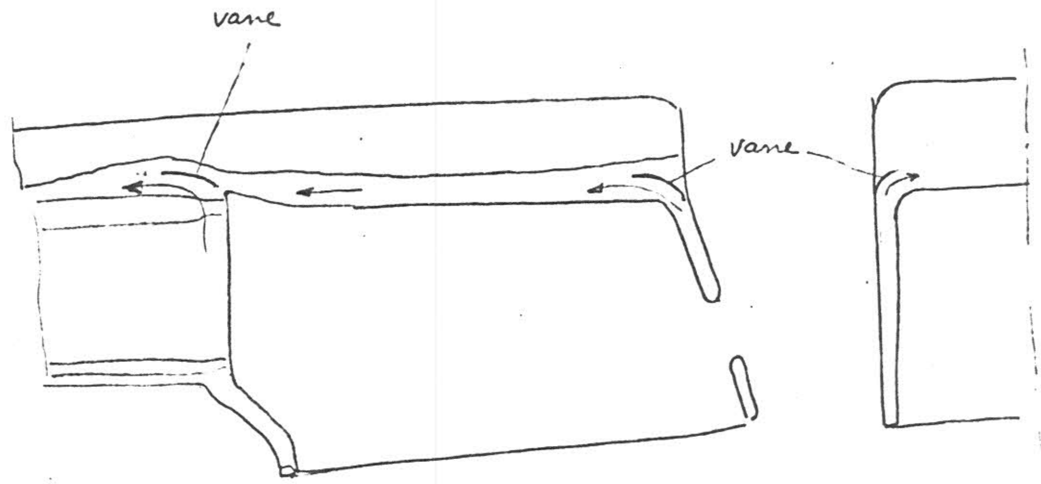


Fig 4 Guiding vanes placed to streamline circulation flow.

The whole water circuit is run by the water flow in matter of a few seconds. This has been roughly checked during antifoam testing: a foamy water condition in the boiler could be checked in such a short time upon the injection of the antifoaming compound into the boiler, this being the time taken by the antifoam to travel from the check valve to every part of the boiler.

An oversimplification of the circulation circuit is represented in Fig. 3. L represents the nominal water level plane obtaining when no evaporation takes place. Relatively cold feed water enters at C thereby inducing a subcooled boiling region at the bottom channel flow B. This flow splits in two at E: part T of it goes upwards through the upward channel where there is a strong heat input $H_T$; part F goes backwards first and upwards later, receiving a strong heat input $H_F$. Both flows T and H discharge in an open trough $H_a F_o$ where steam bubbles are disengaged at the surface. Water flows fordwards because of the heads $h_t$ and $h_f$ which are made possible because of the lower density of the fluid at T and at the upwards portion of the conduit F as compared to denser fluid present at R and U.

Current boiler circulation theories can be applied considering the various resistances and heat inputs. The objects of the exercise are:

(i) To define the various flows so as to guarantee with adequate margin of safety...

...///

///... that no water starvation occurs at any part of the circuit. This safety margin can be reduced (if neccessary or convenient) when the whole flow pattern can be more accurately predicted (hence mastered).

(ii) To define the swelling volume V and the shape of the surface in the open trough $H_0F_0$ tending to occupy the steam space S, which should have suitable height and shape, and also conveniently arranged steam outlets.

(iii) To forecast the indication of the water gauge WG and its interpretation concerning the safety of the firebox crown and proper engine handling.

(iv) To allow the designer to play with the resistances of the various parts of the circuit (say reducing water passages) so that boiler design can be mastered in this respect.

(v) To know what influence will have a great pressure rise (up to 60 ate) aimed at in future designs.

3. <u>Some practical proposals</u>.

Qualitative conclusions are already possible. If the resistance of the circuit approaches zero (i.e extra large water passages), circulation flow becomes very large and therefore the residence time of the bubbles tends to zero, hence leading to zero swelling volume i, i.e. the water surface on the top of the trough $H_0$-$F_0$ coincides with the water level L obtaining at no evaporation. If circulation is so restricted that volume dryness at the outlet of T and F is unity, and assuming that it develops linearly along the flow path, the swelling ...///

111... volume would be roughly one half the volume of the conduits F and T where most of the steam bubbles lie. This extreme is not incompatible with a still acceptable swelling volume equivalent to, say, 0,15 m rise in the water level (assumed uniform over the whole boiler length). If that volume is supposed to be concentrated at the back, it would amount to, say, 2 · 0,15 m = 0,30 m; which again is not an undue figure. Actual circulation obviously plays between both extremes. If the maximum volume dryness fraction of 0,8 is accepted, the last figure becomes 0,8 · 0,3 m ≈ 0,25 m. Such volume dryness has been given (Ref (4), p. 9-3.) (as indicative only) in water tube boiler practice; it rests upon the very good heat transfer resulting in boiling water even if the surface is contacted by a surprisingly low water-containing mixture. This question will later be dealt in more detail (1).

So far, as a consequence of the need to increase as much as possible the power-to-weight ratio, boiler design tends to pack as much as possible the required steam production into the smallest overall volume. This translates into a tube bundle cramped into the smallest possible space (hence with tubes as near as possible); firebox water spaces as small as possible (there are now stay designs offering adequate flexibility to possibilitate such narrowing); the smallest space between tubes and barrel (now no fear of scaling and sludge-chocking, (2)); the smallest possible steam space, etc.

The author has applied these recommendations successfully (although with a limited kilometrage run) in his experimental 4-8-0 locomotive, yet with no other help than an (audacious) engineering horse sense.

High pressures have always been associated with circulation difficulties in water tube boiler engineering. This stems from the fact that the circulation driving force (i.e. steam-water specific volume difference) reduces and becomes zero at the critical point. In the case of 60 ate pressure, steam specific volume becomes reduced by a factor of three when compared to the current 14-20 ate figures. It is interesting to speculate what it would mean to accept the 0,8 volume dryness fraction in the case of a firebox of current size developing 5 000 kg h⁻¹ of steam:

| | | | | |
|---|---|---|---|---|
| Steam pressure, = | 1 | ata | 14 | 60 |
| Firebox evaporation, = | 2 | $kg\ h^{-1}$ | 5000 | 5 000 |
| Saturated steam specific volume, = | 3 | $m^3\ kg^{-1}$ | 0,1435 | 0,0331 |
| Steam volume produced per sec, = | 4 | $m^3\ s^{-1}$ | 0,20 | 0,046 |
| Indicative, accepted, volume dryness fraction limit, = | 5 | – | 0.8 | 0,8 |
| Water volume per second, = (1 – 5) · 4 ÷ 5 = | 6 | $m^3\ s^{-1}$ | 0,05 | 0,011 |

These figures are so small, that they would call for very unlikably small water passages. This points towards stating that water starvation is a very unlikable contingency in the STEPHENSON can boiler; yet this has to be proven quantitatively.

There seems to be no harm in having a vigorous circulation although there is not much merit in it except in reducing the swelling volume. As is shown in Ref. (1), boiling heat transfer improves somewhat with circulation velocity (both in normal and burnout heat fluxes); but boiling heat transfer coefficients are so high that metal temperatures are affected by a few degrees only. However, a reduction of liquid superheat against the wall will be welcomed from the standpoint of the physicochemistry of the solute-concentrating (hence, tending to scaling!) boundary layer contacting it. So far, these phenomena are just starting to be studied, and a provisional conclusion could be that enhacing and facilitating circulation is to be aimed at.

In order to avoid wasting the available circulation head in shocks, bends, obstructions, etc., once the flow pattern is known, at least some qualitative effort can be done; suitable vanes (Fig. 4) can be placed so as to ease-off shocks and turns.

An important consequence of the present conception of water side phenomena is that boiling heat transfer cannot any longer be considered as "pool boiling", but as "forced flow boiling". In firebox walls, the cross section of each "tube" is of course not constant according to the varying separation of streamlines; flow T rising up in front of the tube plate can be considered as a boiling cross flow, while longitudinal flows U and B can be assimilated to that obtaining in water cooled atomic reactor bundles. The author hurries himself to state that this interest in boiling heat transfer is not due to any fear of it being on the short side (which use to be the case in water tube boiler design) but because one should master it ---///

III... so as to guarantee adequate margins when coming, advanced, designs will impose higher pressures and reduced water passages.

Looking to Fig. 1, one wonders if the location of the steam take up (the dome) located, in American designs, about one metre ahead of the tube-plate over the upward water surge coming from the tubes there, was the best position. The English or Continental location by the middle of the barrel seems a better proposal, while positioning it well ahead where no bubbles obtain and where any foam coming from the back had time to die off deserves consideration too. This even when accelerating the train backwards is a current occurrence (*)

It is probably that the height of the water near the front tube plate remains unchanged when the boiler steams, i.e. no swelling obtains there because most of it is concentrated at the back end. This should be carefully considered in the case of author's boiler design in which the tubes are just covered by some 30 to 50 mm of water only. The danger of burning because of water starvation there is not very great, but an inequality in the circumferencial distribution of heat may lead to a collapse as pointed out by TROSS (10). This danger is of course greater at higher pressures.

(*) Experience with rebuilt engine Nº 3477 (FCGR, Argentina), having very small steam space and able to reach evaporations up to 140 kg m⁻² h⁻¹ in "bunker first" accelerations, support this contention.

## 4 A typical boiler behaviour.

The present analysis refers to a typical boiler of American design which is fully known to the author. It corresponds to a metre gauge, 4-8-2 engine, class C16, FC6B (Argentina). The concerning particulars are as follows:

| | | | |
|---|---|---|---|
| Builder, | 1 | – | BALDWIN |
| Year built | 2 | year | 1948 |
| Steam pressure, nominal, = | 3 | ate | 14 |
| Current working steam pressure, = | 4 | " | 13,5 |
| Fuel, | 5 | – | oil |
| Grate area, = | 6 | $m^2$ | 4,1 |
| Firebox heating surface (including two syphons, | 7 | " | 17. |
| Large tubes heating surface, fire side, = | 8 | " | 67 |
| Ditto, small tubes | 9 | " | 140 |
| Total evaporative heating surface, fire side, = 7 + 8 + 9 = | 10 | " | 224 |
| Superheater surface, fire side, = | 11 | " | 68 |
| Water feed system, | 12 | – | injector, side clack valve. |
| Firebox volume, = | 13 | $m^3$ | ≈ 7,5 |
| Maximum evaporation in current service, = | 14 | $kg\ h^{-1}$ | 12 000 |
| Ditto reached in test, (Engine 1802, RS burner) | 15 | " | 16 000 |

| | | | |
|---|---|---|---|
| Water volume at lowest glass indication (measured), = | 16 | $m^3$ | 6 |
| Steam chamber volume at lowest water level (measured), = | 17 | " | 3 |
| Steam temperature at maximum current steam production 14, = (measured) | 18 | °C | 380 |
| Smokebox temperature (at 14), = (measured) | 19 | " | 380 |
| Heat to steam at pressure 4, = | 20 | Gcal $h^{-1}$ | 9,0 |
| Heat 20 is made up as follows: | | | |
| Heat to warm up the incoming water from 15°C to saturation, = | 21 | " | 2,2<br>% 24 |
| Heat to evaporate the water, = | 22 | " | 5,6 |
| Heat to superheat, = | 23 | " | 1,2 |
| Heat 20 is transferred as follows in different heating surfaces: | | | |
| To firebox (HUDSON-ORROK), = | 24<br>25 | Gcal $h^{-1}$<br>% | 3,5<br>% 39 |
| To superheater, = | 26<br>27 | "<br>" | 1,2<br>% 13 |
| To small tubes, = | 28<br>29 | "<br>" | 2,9<br>% 32 |
| To large tubes, = | 30<br>31 | "<br>" | 1,4<br>% 16 |
| Heat liberated in the furnace, = | 32 | Gcal $h^{-1}$ | 11,0 |

| | | | |
|---|---|---|---|
| (Evaporation heat) firebox, % of total, = | 33 | % | 45 |
| small tubes, = | 34 | " | 37 |
| large tubes, = | 35 | " | 18 |
| Heating surface load at evaporation 14, = 14 ÷ 10 = | 36 | $kg\ m^{-2}\ h^{-1}$ | 54 |
| The water volume 16 is split as follows: on the firebox, = | 37 | $m^3$ | 2 |
| | 38 | % | 33 |
| barrel, = | 39 | $m^3$ | 4 |
| | 40 | % | 67 |
| Temperature of the feed on account to the injector, = (live steam injector) | 41 | °C | 60 |
| Heat to feed given by injector steam, = (41* - 15 kcal $kg^{-1}$) · 14 = | 42 | $Gcal\ h^{-1}$ | 0,54 |
| Saturated steam supplied to the injector, = 42 ÷ (666,3* - 60*) = | 43 | $kg\ h^{-1}$ | 900 |
| Total saturated steam produced by the boiler, = 14 + 43 = | 44 | " <br> $kg\ s^{-1}$ | 12 900 <br> 3.6 |
| Therefore, part of the heat otherwise delivered from the small (bottom) tubes to evaporate water is used to heat the incoming feed, as follows | | | |
| Heat transferred in small tubes, = 28 = | 45 | $Gcal\ h^{-1}$ | 2,9 |
| Heat to warm up water, = 28 - 42 = | 46 | " | 1,7 |
| Heat to make steam in small tubes, = 45 - 46 = | 47 | " | 1.2 |

3

| | | | |
|---|---|---|---|
| Saturated steam enthalpy, | 48 | kcal $kg^{-1}$ | 666 |
| Enthalpy of water at temp. 41, = | 49 | " | 60 |
| Heat to evaporate 1 kg. of water, = | 50 | " | 606 |
| Total steam produced (saturated), = (20 - 23) ÷ 50 = 44 | 51 | kg $h^{-1}$ | 12900 |
| Steam produced by the firebox, = 24 ÷ 50 = | 52 | " % | 6500 50 |
| Steam produced by the large tubes, = 30 ÷ 50 = | 53 | " % | 2300 18 |
| Steam produced by the small tubes, = 47 ÷ 50 = | 54 | " % | 4100 32 |
| Total = 44 = | 55 | " % | 12 900 100 |
| Specific volume of saturated steam = | 56 | $m^3$ $kg^{-1}$ | 0,139 |
| Total volume of steam per hour, = 56 . 44 = | 57 | $m^3$ $h^{-1}$ | 1793 |
| Ditto per second, = 57 ÷ 3600* = | 58 | $m^3$ $s^{-1}$ | 0,5 |

4.1 <u>Flow quantities: example of a typical boiler</u>.

The purpose of the following exercise is to have some first idea about the amount of the various quantities and factors involved in circulation. So far, the author has not traced any figure, even if roughly approximate, in the last 80 years' literature.

The concerned boilers (C 16, FCGB, Argentina) are normally operated with a mass of water corresponding to what, under standing conditions, would just be showing in the glass. At normal steaming rate of 12 000 kg h⁻¹, the water level in the glass will read 80 mm, which, after correction for grade, acceleration, etc. becomes 70 mm ("three fingers"). The top of the foam layer, as recorded by electrodes located somewhat back of the dome, will read further 100 mm higher; if one takes a 20 mm foam thickness, the net raise of the water level will be (70 +100 -20) mm = 150 mm. As it has been already said, the top of the water surface is higher at the back as compared to the front end. But since the electrodes are located amidships, one can, as a first approximation, take that the swelling volume (i.e. the volume of bubbles entrapped in the water - steam mass) is that corresponding to an average rise of 150 mm, namely 1,6 $m^3$.

Hence

| | | | |
|---|---|---|---|
| Volume of steam bubbles, = | 70 | $m^3$ | 1,6 |
| Life of steam bubbles in the water mass, = 70 ÷ 58 = (average over the whole boiler) | 71 | s | 3,1 |
| mass of steam bubbles, = 70 ÷ 56 = | 72 | kg | 1,15 |
| specific volume of water, = | 73 | $m^3\ kg^{-1}$ | 0,00115 |
| Mass of water in the boiler, = 16 ÷ 73 = | 74 | kg | 5200 |
| A limit hypothesis is that water velocity is equal to that of bubbles (i.e. there is no slip between bubbles and water). Hence the total water flow is 44 $\cdot \frac{74}{72}$ = | 75 | $kg\ s^{-1}$ | 1630 |
| This corresponds to a volumetric water flow of 75 · 73 = | 76 | $m^3\ s^{-1}$ | 1,9 |
| Water mass / bubble mass, = 74 ÷ 72 = | 77 | – | 450 |
| Water volume / bubble volume, = 16 ÷ 70 = | 78 | – | 3,7 |
| Volume dryness fraction, average value referred to the whole boiler, = 1/(1 + 78) = | 79 | – | 0,21 |

However, the upward bubble velocity (about 0,25 $m\ s^{-1}$) is not negligible compared to water velocity. Therefore a correction factor is to be incorporated.

| | | | |
|---|---|---|---|
| Slip correction factor adopted, = | 80 | – | 0,8 |
| Hence water flow is $44 \cdot \frac{74}{72} \cdot 80$ = | 81 | $kg\ s^{-1}$ | 1300 |
| This corresponds to a volumetric water flow of $81 \cdot 73$ = (Flow T + F, Fig. 3) | 82 | $m^3 s^{-1}$ | 1,5 |
| Water mass / bubble mass, = $74 \div 72$ = | 83 | – | 450 |
| Water volume / bubble volume, = $16 \div 70$ = | 84 | – | 3,7 |
| Volume dryness fraction, average referred to the whole boiler, = $1/(1 + 78)$ = | 85 | – | 0,21 |
| Average time for water to run the whole loop (Fig. 3), = $(70 + 16)/82$ = | 86 | s | 5 |

On travelling upwards, both flows T and F flash part of their mass because the static pressure becomes lower. Bubbles are, on the average, generated about one metre below the surface; hence the pressure drop is roughly 0,08 at. Therefore

| | | | |
|---|---|---|---|
| Pressure in the steam space, = 4 + 1 at = | 87 | ata | 14,5 |
| Pressure at 1 m below the surface, = 87 + 0,08 at = | 88 | " | 14,58 |
| Water enthalpy change between pressures 88 and 87, = | 89 | $kcal\ kg^{-1}$ | 0,272 |
| Total heat brought into play, = $81 \cdot 89$ = | 90 | $kcal\ s^{-1}$ | 353,6 |
| Heat of evaporation, = | 91 | $kcal\ kg^{-1}$ | 467 |

| | | | |
|---|---|---|---|
| Steam produced by flash, per second, = (90) ÷ (91) = | (92) | $kg\ s^{-1}$ | 0,757 |
| Ditto per hour, = which is a far from negligibly amount. | (93) | $kg\ h^{-1}$ | 2725 |
| The heat involved in flash, per hour, is (90) · 3600* = | (94) | $Gcal\ h^{-1}$ | 1,27 |

Since the reverse effect obtains in the downward part of the flow (R, Fig. 3), bubbles present at U will collapse, and water will reach and run through the bottom part of the boiler in a subcooled condition. Were it not because of this, the heat available to evaporate steam in the small tubes is (47) = 1.2 $Gcal\ h^{-1}$. This figure is roughly equal to (94), hence it can be concluded that no bubbles will be produced at the bottom of the barrel in the region occupied by the small tubes: heat transferred there will go to warm up the cooler inlet feed and rise water temperature from the subcooled condition. Nevertheless, heat transfer will be still high under a <u>subcooled boiling</u> régime.

As a result of the above reasoning, very little steam is produced on the first metre of tube length since this comes from large tubes heating surface, which is comparatively small and has a rather low convection heat transfer because its length / diameter ratio is small too (because superheater bends are far from the tube plate). This leads to the conclusion that perhaps 90 to 95 % of the bubbles born in the firebox area, a surprising result since only 39% = (25) of the heat is transferred there.

The fact that most of the heat transferred in the tube bundle is carried out in the "subcooled boiling régime" implies that the phisycochemistry of the boundary layer contacting the heating surface of those tubes behaves differently than what is supposed concerning scaling and suspended solid baking. This is outside the scope of the present work, but must be recorded here. A parallel effect obtains at the firebox end: a large proportion of the heat goes to steam bubbles growth *after* they have left the heating surface - since they grow up because of heat transferred from the superheated liquid phase. This superheat comes both from contacting the firebox heating surface and from the pressure reduction associated with the upward flow.

It is also

| | | | |
|---|---|---|---|
| Total steam produced by the boiler, = (44) = | (96) | $kg\ s^{-1}$ | 3,6 |
| Steam volume, = (96) · (56) = (57) = (58) = | (97) | $m^3\ s^{-1}$ | 0,5 |
| Fraction assumed to be produced at the back end, = | (98) | — | 0,95 |
| Water flow = (82) = | (99) | $m^3\ s^{-1}$ | 1,5 |
| "Volume dryness" fraction at back end, = (97) · (98) / ((97) + (99)) = | (100) | — | 0,24 |
| "Weight (mass) dryness" fraction at back end (title of steam), = (96) · (98) / ((96) + (81)) = | (101) | — | 0,0026 |

This figure is exceedingly small as compared to what is usual in higher pressure water tube boilers.

| | | | |
|---|---|---|---|
| The surface of the evaporation plane at the back end, considering the reduction resulting from swelling, is ≈ | 102 | $m^2$ | 4,5 |
| Hence the steam velocity in traversing that plane is 97 ÷ 102 = | 103 | $m\,s^{-1}$ | 0,11 |
| | | $ft\,s^{-1}$ | 0,36 |

An interesting parameter concerning boiling heat transfer and bubble growth is the average life time of the bubbles.

| | | | |
|---|---|---|---|
| Assuming that back end volume is equal to (volume around the firebox + swelling volume), = 37 + 70 = | 104 | $m^3$ | 3,6 |
| The average life time of bubbles at the back end is approximately $\left[104 / (82 + 58)\right] \cdot 80 =$ | 105 | s | 1,4 |

This time is considerably greater than the usual 0,1 s considered in boiling heat transfer experiments. Therefore the amount of heat transferred to bubbles is considerable, hence requiring a revision of current heat transfer theory (1).

Some idea of the average velocity of the stream in firebox water passages can be estimated considering that the path of the mean fillet is 5 m long. (Fig 1). Hence

| | | | |
|---|---|---|---|
| Length of the path, = | 106 | m | 5 |
| Stream velocity, = (106 ÷ 105) · 80 = | 107 | $m\,s^{-1}$ | 2,9 |

This velocity is high enough to influence favourably the boiling heat transfer and the burnout critical flux. It corresponds to a mass velocity of

| | | | |
|---|---|---|---|
| 107 ÷ 73 = | 108 | $kg\,m^{-2}s^{-1}$ | 2500 |

| | | | |
|---|---|---|---|
| Ditto in $kg\,m^{-2}\,h^{-1}$, = | 109 | $\frac{kg \cdot 10^6}{m^2\,h}$ | 9,1 |
| In English units, = | 110 | $\frac{lb \cdot 10^6}{ft^2\,h}$ | 1,9 |

# "STEAM LOCOMOTIVE BOILER WATER CIRCULATION – A FIRST SEMI-QUANTITATIVE APPROACH."


by.

L.D. PORTA, C.Eng,

F. Inst. F; M.I. Mech. E.; CAI,

Chief, Dept. Thermodynamics, INTI

ARGENTINA



Issued: Buenos Aires, April 1979.

Revised, Buenos Aires, June 1981.



Adress: Azara 1557

Banfield 1828

ARGENTINA

Summary.

While water circulation is a most important issue in water tube boilers (where it is quantitatively mastered), it has been a matter of myth, vagaries and conjecture in the case of locomotive and other firetube [boilers.

Some measurements carried out during water treatment tests for the FCGB (Argentina), coupled to elementary reasoning and knowledge of boiling heat transfer phenomena now available, have permited to scheme out a reasonable picture of what life is in the boiler inside.

TGS (Advanced, Third Generation, Steam locomotive technology), aiming to keep to the STEPHENSONian boiler working at 60 ate pressure, impose the need to clear up circulation parameters. It is believed that the present paper represents the first step of a long (happy) ladder leading to better future boilers as required to operate in an energy starving world.




1.1

## Index



## Appendices



## Figures.

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1 Introduction. (*)

Stationary water tube boiler water circulation is a most important subject in water tube boiler technology, and the technical literature offers a long list of references on it. This contrasts with the vagaries told about in the field of fire tube boilers, the locomotive type being no exeption. No water tube boiler designer will today rely on guess work on this area, yet thousands of succesful firetube boilers have been heated according to thumb rules supposedly based on experience, apparently with too many ~~not~~ troubles even in poor water treatment days.

The time has come to start investigating the subject, particularly in view of the needs imposed by advanced design. The writter believes that the STEPHENSONian boiler has merits on its own to be proposed for pressures at present aiming 60 atc (850 psig); but, in order to achieve that goal, vagaries and thumb rules must be substituted by better engineering; and conviction and beliefs replaced by information, calculations and checks against well established engineering science.

These matters are closely allied to water side heat transfer and two-phase flow, two areas which are presently receiving considerable attention because of the exacting requirements imposed by nuclear reactor design. The appalling stream of knowledge coming on boiling heat transfer must also be incorporated to steam locomotive technology in order to eradicate years of myths and vagaries too. Efforts in that direction is currently being carried out (1979) (PORTA (1)).

(*) Numbers in a square $\boxed{00}$ refers to additional comment made in Appendix A1.

This paper points towards being useful in coming boiler design; although use is made of knowledge referring to past practice, full advantage is taken from the now available water treatment technology (2), namely: (i) scale free heating surfaces can now be guaranteed, and (ii) very pure steam is produced under all circumstances thanks to the now available antifoams. These markedly influence circulation because (a) fewer steam bubbles of larger diameter are produced at the heating surfaces; (b) these bubbles coalesce and grow into larger ones on its way to the steam space, and (c) practically no foam lays on the top of "solid" water.

Condition (i) means that water passages areas are available in full and not reduced by scale, and that its circulation hindering roughness has been swept away from the picture; condition (ii) means that the steam space is no longer a low velocity moisture settling chamber or a froth containing volume.

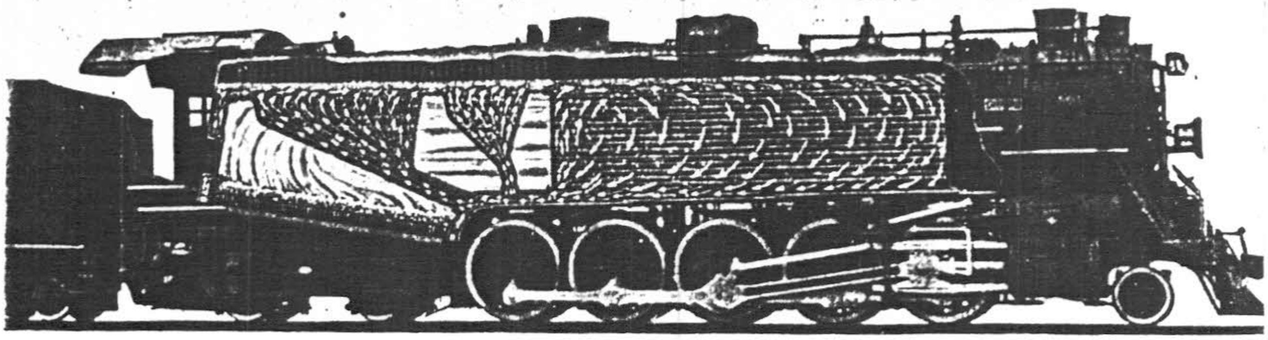

Fig. A—Diagram of water circulation in a boiler equipped with Nicholson Thermic Syphons

Boiler circulation in a locomotive boiler provided with NICHOLSON thermic syphons

Apparently, this phantom view was drawn as a result from observations made in a transparent boiler model operating at atmospheric pressure (reported by CHAPELON, Ref (5)) The understanding is that the syphons superimpose their action on general circulation as described in this paper. Circulation in the back sheet and crown are exactly as described.

## 2 A qualitative description of the flow pattern in a typical boiler.

What follows is the best image one can have before some quantitative definition can be achieved. Referring to Fig. 1, the boiler can be considered divided in two parts: (i) the back, including the first .. metre of tube length, in which 90% of the steam is produced, and (ii) the front part, in which the remaining 10% steam is produced, (✱) therefore configurating a rather calm section in which steam bubbles contribute to little swelling. This is enhanced by the location of the clack valve because relatively cold feed water, in being mixed with the general flow, leads to subcooled boiling over a large part of the small tubes as it will be shown later.

This 90 - 10% distribution sets a clear distinction: the back is a turmoil region in which most of water swelling occurs because most of the steam bubbles are produced occupying some 25% of the volume there (average figure). This low density water - steam mixture determines a strong (generally upwards) circulation stream on the firebox walls and near the tube plate, sucking a denser mixture from the bottom of the barrel and originating a strong forward stream over the top of the firebox which extends forwards over the whole evaporation plane. Circulation is, therefore, fairly definite, as shown in the familiar phantom view produced by NICHOLSON thermic syphon makers (however that flow pattern is not due to the syphon, authough it contributes to enhance it).

(✱) These unexpected figures will be justified later.

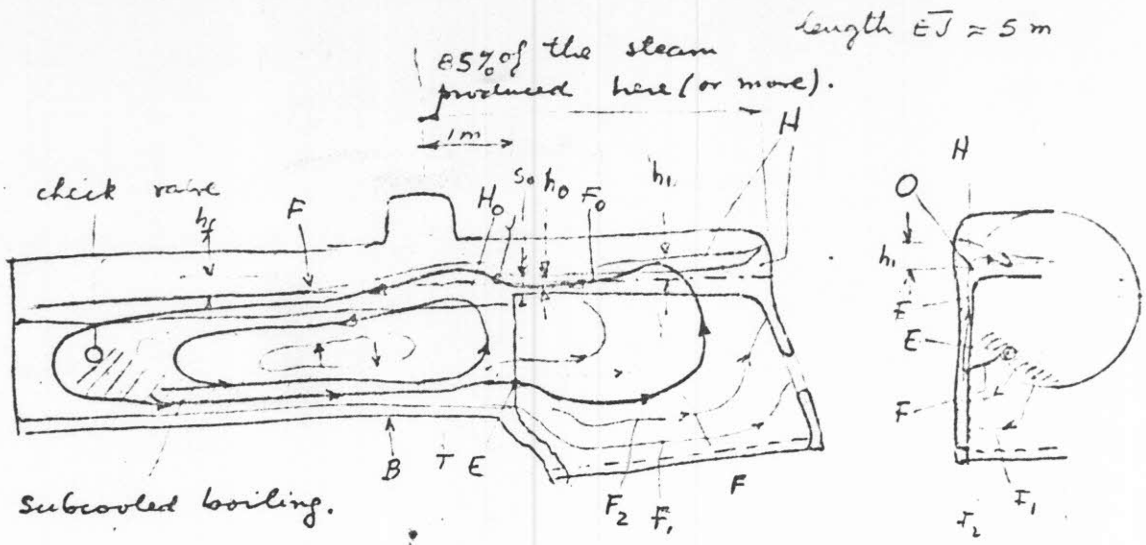


Fig. 1 Circulation flow pattern.

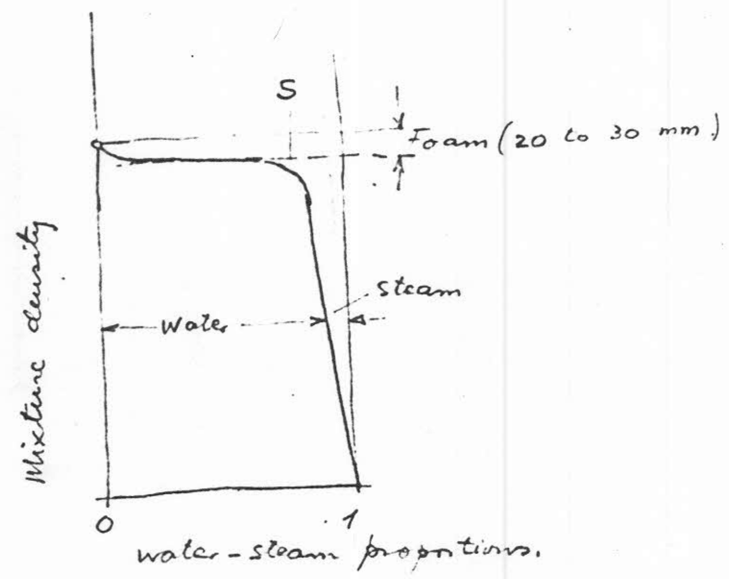


Fig. 2 Schematic view of the variation of steam-water proportions with height.



5,1

Fig 1 shows the two most important (or average) streamlines: F corresponds to the firebox and T to the tubes. The back wall is fed by streamlines $F_1$ and $F_2$ sweeping backwards the space above the foundation ring. Ordinarily, most of firebox evaporation, both with coal or oil, is produced at the bottom part because heat liberation, flame emmissivity and temperature are highest there. When the flow enters the firebox area at E, it contains relatively few bubbles; when discharged at the top of the legs at O it consists of a relatively light mixture surging upwards at high velocity, therefore leading to a hump H extending laterally all along the sides of the (BELPAIRE) steam space, and also crosswise against the outer back plate. 3 At full steaming rate, the volume dryness fraction, which on average is some 25% over the whole firebox, is probably 5 to 10% at the bottom over the ring, and perhaps 50% at the water space discharge outlet O. This figure, however, is far from the critical limit concerning adequate boiling cooling (1)(4).

Therefore the region above the firebox crown plate is fed by three sides by a strong water surge, which precipitates forwards under the gravity potential resulting from the height $h_1 > h_0$. If $h_1 - h_0$ is about 0,2 m, the potential velocity $\sqrt{2gh}$ is somewhere around 2 m $s^{-1}$! Incidentally, the jump existing against the back plate leads to false water level cock readings as demonstrated in the early American tests which resulted in the official American water gauge arrangements (3)(11), (See Appendix A2).

The forward going stream $F_o$ hits the upward stream, thereby leading to a hump $H_o$, wherefrom the combined stream proceeds forwards under the available head $h_f$. This combined flow probably has a velocity of about 1 m s$^{-1}$ under the available head $h_f$.

So far, we have been speaking about a well defined water surface, which is not the case. Referring to Fig. 2 showing qualitatively how the water-steam proportion goes with height, that surface would correspond to that of the plane S. Above it, there is a layer of foam which experiments show to amount to some 20 to 30 mm [1] when antifoams are used. This foam layer tends to even out the various jumps and the turmoil occurring below, as per the familiar image of the household washing machine.

The forward going stream F resulting from the stream $F_o$ coming from the firebox, and the stream T coming from the tube bundle, has its backward counterpart in the stream B running along the bottom of the barrel. B is assumed to be decomposed in T and F. Although most of the heat transfer obtaining in the tube bundle occurs at the small tubes located below, this does not correspond to the evaporation, since a large part of that heat transferred in the small tubes goes to heat up the relatively cooler water getting in through the clack valve. Besides, not very much heat comes out from the gases in the second half of tube length because of the reduced temperature differential; hence, one can say that the front half and the bottom part of the barrel contain a high density water-steam mixture. This is very sound since it acts as "downcomer" to the back part...///

111... of the boiler. Therefore, most of the swelling and bubbles exist at the back end.

Concerning the throat plate, it is probably that, in spite of the intense heat transfer obtaining there, the flow is downwards as shown in Fig. 1. It may be that under certain circumstances, relatively cold water reaches this place, therefore leading to severe thermal stresses.

The pressure at the bottom of the firebox is greater than at the top because of a static head amounting to nearly 2 m, hence ≈ 0,15 at. This means that if water is at saturation temperature at the bottom, part of it flashes on coming up, adding to an increase in bubble diameter (see [2]). Conversely, in downward flow regions, bubbles collapse. Since the total boiler flow consisting in streams F, B, etc. is hundred fold the steam production of the boiler, the steam quantities involved in flashing and collapsing are quite important in spite of the small quantity of heat per kg brought into play. This means that probably a sizable part of the steam evolved at the heating surfaces of the large tubes will never be released at the front part of the evaporation plane; on being entrained downwards, bubbles produced there will collapse, yet this steam will reappear at the back region. One can say, therefore, that perhaps the whole of the steam is released at the back part of the boiler, and certainly more than 50% at the firebox even if the heat transfer from the flames corresponds to a smaller proportion. This increases the volume dryness fraction at O (Fig. 1).

Although the velocity of bubbles relative to water is not negligible (approximately $0{,}25\ m\ s^{-1}$), the flow is probably so turbulent that in a first analysis steam bubble streamlines may be taken as coincident with water streamlines. Of course, this is not so at the separation plane and some idea about such release can be obtained when comparing the above upward velocity with a horizontal velocity of, say, 1 to 2 $m\ s^{-1}$; this results in an angle of 15 to 7°.

One may expect that the local foam layer thickness bears some relationship with the local bubble rising velocity. This foam layer is constantly pushed forwards, and in the case where anti-foams are used, it is probably that little, if any, foam obtains at the front end where no steam bubbles are evolved.

The preceding description leads to the conclusion that in the steam space only the back half of it is used. Since it no longer serves as settling chamber for entrained drops (PORTA (2)), a sort of low velocity "wind" blows from the back towards the base of the dome(*) The nominal residence time of the steam in the chamber is some 5 s; but since only the back part is utilized, it comes to be some 2,5 s; If the swelled level and the foam volume is accounted for, the above figure reduces to something around 1,5 s, or perhaps to 0,5 s when the boiler is operated with a higher mark in the glass. Under such circumstances the "wind" velocity above the hump $H_o$ may be some $4\ m\ s^{-1}$, and one may wonder whether no foam particles can be entrained as per the action of natural wind over the foam over sea waves.

(*) Velocity of the order of $2\ m\ s^{-1}$.

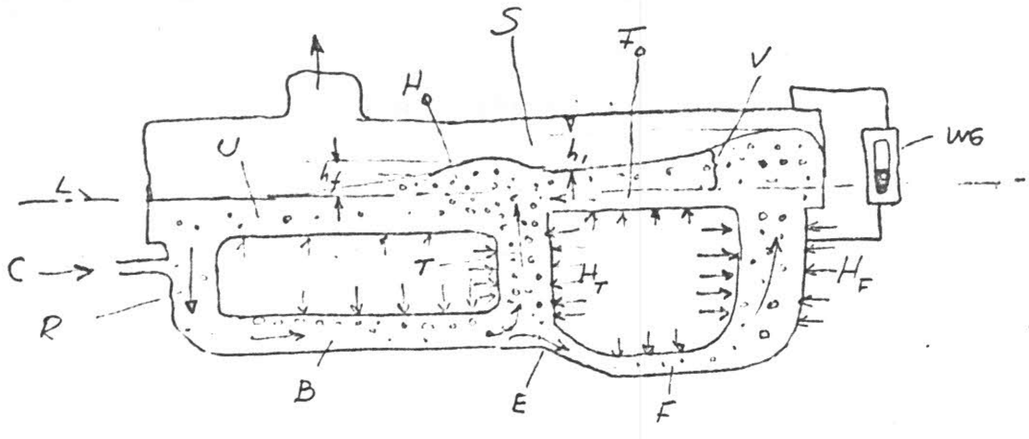


Fig. 3. A simplified scheme of water circulation.

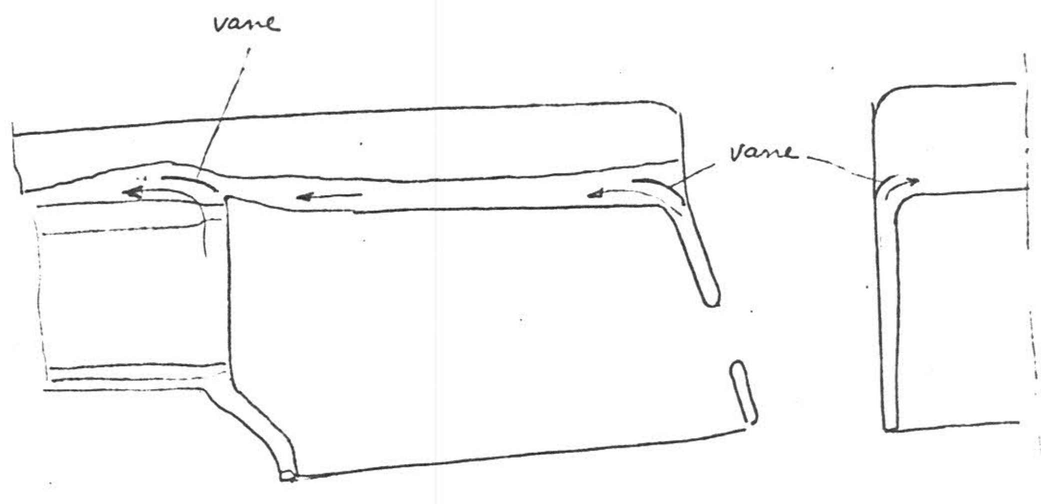


Fig 4 Guiding vanes placed to streamline circulation flow.

The whole water circuit is run by the water flow in matter of a few seconds. This has been roughly checked during antifoam testing: a foamy water condition in the boiler could be checked in such a short time upon the injection of the antiframing compound into the boiler, this being the time taken by the antifoam to travel from the check valve to every part of the boiler.

An oversimplification of the circulation circuit is represented in Fig. 3. L represents the nominal water level plane obtaining when no evaporation takes place. Relatively cold feed water enters at C thereby inducing a subcooled boiling region at the bottom channel flow B. This flow splits in two at E: part T of it goes upwards through the upward channel where there is a strong heat input $H_T$; part F goes backwards first and upwards later, receiving a strong heat input $H_F$. Both flows T and H discharge in an open trough $H_a F_o$ where steam bubbles are disengaged at the surface. Water flows fordwards because of the heads $h_1$ and $h_f$ which are made possible because of the lower density of the fluid at T and at the upwards portion of the conduit F as compared to denser fluid present at R and U.

Current boiler circulation theories can be applied considering the various resistances and heat inputs. The objects of the exercise are:

(i) To define the various flows so as to guarantee with adequate margin of safety...

.../11

///... that no water starvation occurs at any part of the circuit. This safety margin can be reduced (if neccessary or convenient) when the whole flow pattern can be more accurately predicted (hence mastered).

(ii) To define the swelling volume V and the shape of the surface in the open trough $H_oF_o$ tending to occupy the steam space S, which should have suitable height and shape, and also conveniently arranged steam outlets.

(iii) To forecast the indication of the water gauge WG and its interpetation concerning the safety of the firebox crown and proper engine handling.

(iv) To allow the designer to play with the resistances of the various parts of the circuit (say reducing water passages) so that boiler design can be mastered in this respect.

(v) To know what influence will have a great pressure rise (up to 60 ate) aimed at in future designs.

3. Some practical proposals.

Qualitative conclusions are already possible. If the resistance of the circuit approaches zero (i.e extra large water passages), circulation flow becomes very large and therefore the residence time of the bubbles tends to zero, hence leading to zero swelling volume i, i.e. the water surface on the top of the trough $H_o - F_o$ coincides with the water level L obtaining at no evaporation. If circulation is so restricted that volume dryness at the outlet of T and F is unity, and assuming that it develops linearly along the flow path, the swelling ...///

/// ... volume would be roughly one half the volume of the conduits F and T where most of the steam bubbles lie. This extreme is not incompatible with a still acceptable swelling volume equivalent to, ~~say~~, 0,15 m rise in the water level (assumed uniform over the whole boiler length). If that volume is supposed to be concentrated at the back, it would amount to, say, 2 · 0,15 m = 0,30 m ; which again is not an undue figure. Actual circulation obviously plays between both extremes. If the maximum volume dryness fraction of 0,8 is accepted, the last figure becomes 0,8 · 0,3 m ≈ 0,25 m. Such volume dryness has been given (Ref (4), p. 9-3.) (as indicative only) in water tube boiler practice ; it rests upon the very good heat transfer resulting in boiling water even if the surface is contacted by a surprisingly low water-containing mixture. This question will later be dealt in more detail (1).

So far, as a consequence of the need to increase as much as possible the power-to-weight ratio, boiler design tends to pack as much as possible the required steam production into the smallest overall volume. This translates into a tube bundle cramped into the smallest possible space (hence with tubes as near as possible); firebox water spaces as small as possible (there are now stay designs offering adequate flexibility to possibilitate such narrowing); the smallest space between tubes and barrel (now no fear of scaling and sludge-chocking, (2)); the smallest possible steam space, etc.

The author has succesfully applied these recommendations (although with a limited kilometrage run) in his experimental 4-8-0 locomotive, yet with no other help than an (audacious) engineering horse sense.

High pressures have always been associated with circulation difficulties in water tube boiler engineering. This stems from the fact that the circulation driving force (i.e. steam-water specific volume difference) reduces and becomes zero at the critical point. In the case of 60 ate pressure, steam specific volume becomes reduced by a factor of three when compared to the current 14-20 ate figures. It is interesting to speculate what it would mean to accept the 0,8 volume dryness fraction in the case of a firebox of current size developing 5 000 kg $h^{-1}$ of steam:

| | | | | |
|---|---|---|---|---|
| Steam pressure, = | 1 | ata | 14 | 60 |
| Firebox evaporation, = | 2 | kg $h^{-1}$ | 5000 | 5 000 |
| Saturated steam specific volume, = | 3 | $m^3$ $kg^{-1}$ | 0,1435 | 0,0331 |
| Steam volume produced per sec, = | 4 | $m^3$ $s^{-1}$ | 0,20 | 0,046 |
| Indicative, accepted, volume dryness fraction limit, = | 5 | – | 0.8 | 0,8 |
| Water volume per second, = (1 − 5) · 4 ÷ 5 = | 6 | $m^3$ $s^{-1}$ | 0,05 | 0,011 |

These figures are so small, that they would call for very unlikably small water passages. This points towards stating that water starvation is a very unlikable contingency in the STEPHENSON ian boiler; yet this has to be proven quantitatively.

There seems to be no harm in having a vigorous circulation although there is not much merit in it except in reducing the swelling volume. As is shown in Ref. (1), boiling heat transfer improves somewhat with circulation velocity (both in normal and burnout heat fluxes); but boiling heat transfer coefficients are so high that metal temperatures are affected by a few degrees only. However, a reduction of liquid superheat against the wall will be welcomed from the standpoint of the physicochemistry of the solute-concentrating (hence, tending to scaling!) boundary layer contacting it. So far, these phenomena are just starting to be studied, and a provisional conclusion could be that enhacing and facilitating circulation is to be aimed at.

In order to avoid wasting the available circulation head in shocks, bends, obstructions, etc., once the flow pattern is known, at least some qualitative effort can be done; suitable vanes (Fig. 4) can be placed so as to ease-off shocks and turns.

An important consequence of the present conception of water side phenomena is that boiling heat transfer cannot any longer be considered as "pool boiling", but as "forced flow boiling". In firebox walls, the cross section of each "tube" is of course not constant according to the varying separation of streamlines; flow T rising up in front of the tube plate can be considered as a boiling cross flow, while longitudinal flows U and B can be assimilated to that obtaining in water cooled atomic reactor bundles. The author hurries himself up to state that this interest in boiling heat transfer is not due to any fear of it being on the short side (which use to be the case in water tube boiler design) but because one should master it

---///

III... so as to guarantee adequate margins when coming, advanced, designs will impose higher pressures and reduced water passages.

Looking to Fig. 1, one wonders if the location of the steam take up (the dome) located, in American designs, about one metre ahead of the tube-plate over the upward water surge coming from the tubes there, was the best position. The English or Continental location by the middle of the barrel seems a better proposal, while positioning it well ahead where no bubbles obtain and where any foam coming from the back had time to die off deserves consideration too. This even when accelerating the train backwards is a current occurrence (*)

It is probably that the height of the water near the front tube plate remains unchanged when the boiler steams, i.e. no swelling obtains there because most of it is concentrated at the back end. This should be carefully considered in the case of author's boiler design in which the tubes are just covered by some 30 to 50 mm of water only. The danger of burning because of water starvation there is not very great, but an inequality in the circumferencial distribution of heat may lead to a collapse as pointed out by TROSS (10). This danger is of course greater at higher pressures.

(*) Experience with rebuilt engine Nº 3477 (FCGR, Argentina), having very small steam space and able to reach evaporations up to 140 $kg\ m^{2}\ h$ in "bunker first" accelerations, support this contention.

4 A typical boiler behaviour.

The present analysis refers to a typical boiler of American design which is fully known to the author. It corresponds to a metre gauge, 4-8-2 engine, class C16, FC6B (Argentina). The concerning particulars are as follows:

| | | | |
|---|---|---|---|
| Builder, | 1 | – | BALDWIN |
| Year built | 2 | year | 1948 |
| Steam pressure, nominal, = | 3 | ate | 14 |
| Current working steam pressure, = | 4 | " | 13,5 |
| Fuel, | 5 | – | oil |
| Grate area, = | 6 | $m^2$ | 4,1 |
| Firebox heating surface (including two syphons, | 7 | " | 17. |
| Large tubes heating surface, fire side, = | 8 | " | 67 |
| Ditto, small tubes | 9 | " | 140 |
| Total evaporative heating surface, fire side, = 7 + 8 + 9 = | 10 | " | 224 |
| Superheater surface, fire side, = | 11 | " | 68 |
| Water feed system, | 12 | – | injector, side clack valve. |
| Firebox volume, = | 13 | $m^3$ | ≈ 7,5 |
| Maximum evaporation in current service, = | 14 | $kg\ h^{-1}$ | 12 000 |
| Ditto reached in test, (Engine 1802, RS burner) | 15 | " | 16 000 |

| | | | |
|---|---|---|---|
| Water volume at lowest glass indication (measured), = | 16 | $m^3$ | 6 |
| Steam chamber volume at lowest water level (measured), = | 17 | " | 3 |
| Steam temperature at maximum current steam production (14), = (measured) | 18 | °C | 350 |
| Smokebox temperature (at (14)), = (measured) | 19 | " | 380 |
| Heat to steam at pressure (4), = | 20 | Gcal $h^{-1}$ | 9,0 |
| Heat (20) is made up as follows: | | | |
| Heat to warm up the incoming water from 15°C to saturation, = | 21 | " | 2,2<br>% 24 |
| Heat to evaporate the water, = | 22 | " | 5,6 |
| Heat to superheat, = | 23 | " | 1,2 |
| Heat (20) is transferred as follows in different heating surfaces: | | | |
| To firebox (HUDSON-ORROK), = | 24<br>25 | Gcal $h^{-1}$<br>% | 3,5<br>% 39 |
| To superheater, = | 26<br>27 | "<br>" | 1,2<br>% 13 |
| To small tubes, = | 28<br>29 | "<br>" | 2,9<br>% 32 |
| To large tubes, = | 30<br>31 | "<br>" | 1,4<br>% 16 |
| Heat liberated in the furnace, = | 32 | Gcal $h^{-1}$ | 11,0 |

| | | | |
|---|---|---|---|
| (Evaporation heat), firebox, % of total, = | 33 | % | 45 |
| small tubes, = | 34 | " | 37 |
| large tubes, = | 35 | " | 18 |
| Heating surface load at evaporation 14, = 14 ÷ 10 = | 36 | $kg\,m^{-2}\,h^{-1}$ | 54 |
| The water volume 16 is split as follows: on the firebox, = | 37 | $m^3$ | 2 |
| | 38 | % | 33 |
| barrel, = | 39 | $m^3$ | 4 |
| | 40 | % | 67 |
| Temperature of the feed on account to the injector, = (live steam injector) | 41 | °C | 60 |
| Heat to feed given by injector steam, = (41* − 15 $kcal\,kg^{-1}$) · 14 = | 42 | $Gcal\,h^{-1}$ | 0,54 |
| Saturated steam supplied to the injector, = 42 ÷ (666,3* − 60*) = | 43 | $kg\,h^{-1}$ | 900 |
| Total saturated steam produced by the boiler, = 14 + 43 = | 44 | " <br> $kg\,s^{-1}$ | 12 900 <br> 3.6 |
| Therefore, part of the heat otherwise delivered from the small (bottom) tubes to evaporate water is used to heat the incoming feed, as follows | | | 3 |
| Heat transferred in small tubes, = 28 = | 45 | $Gcal\,h^{-1}$ | 2,9 |
| Heat to warm up water, = 28 − 42 = | 46 | " | 1,7 |
| Heat to make steam in small tubes, = 45 − 46 = | 47 | " | 1.2 |

| | | | |
|---|---|---|---|
| Saturated steam enthalpy, | 48 | kcal $kg^{-1}$ | 666 |
| Enthalpy of water at temp. 41, = | 49 | " | 60 |
| Heat to evaporate 1 kg. of water, = | 50 | " | 606 |
| Total steam produced (saturated), = (20 - 23) ÷ 50 = 44 | 51 | kg $h^{-1}$ | 12900 |
| Steam produced by the firebox, = 24 ÷ 50 = | 52 | " <br> % | 6500 <br> 50 |
| Steam produced by the large tubes, = 30 ÷ 50 = | 53 | " <br> % | 2300 <br> 18 |
| Steam produced by the small tubes, = 47 ÷ 50 = | 54 | " <br> % | 4100 <br> 32 |
| Total = 44 = | 55 | " <br> % | 12 900 <br> 100 |
| Specific volume of saturated steam = | 56 | $m^3\ kg^{-1}$ | 0,139 |
| Total volume of steam per hour, = 56 . 44 = | 57 | $m^3\ h^{-1}$ | 1793 |
| Ditto per second, = 57 ÷ 3600* = | 58 | $m^3\ s^{-1}$ | 0,5 |

4.1 <u>Flow quantities: example of a typical boiler</u>.

The purpose of the following exercise is to have some first idea about the amount of the various quantities and factors involved in circulation. So far, the author has not traced any figure, even if roughly approximate, in the last 80 years' literature.

The concerned boilers (C16, FCGB, Argentina) are normally operated with a mass of water corresponding to what, under standing conditions, would just be showing in the glass. At normal steaming rate of 12 000 kg h⁻¹, the water level in the glass will read 80 mm, which, after correction for grade, acceleration, etc. becomes 70 mm ("three fingers"). The top of the foam layer, as recorded by electrodes located somewhat back of the dome, will read further 100 mm higher; if one takes a 20 mm foam thickness, the net raise of the water level will be (70 +100 -20) mm = 150 mm. As it has been already said, the top of the water surface is higher at the back as compared to the front end. But since the electrodes are located amidships, one can, as a first approximation, take that the swelling volume (i.e. the volume of bubbles entrapped in the water-steam mass) is that corresponding to an average rise of 150 mm, namely 1,6 m³.

Hence

| | | | |
|---|---|---|---|
| Volume of steam bubbles, = | 70 | $m^3$ | 1,6 |
| Life of steam bubbles in the water mass, = 70 ÷ 58 = (average over the whole boiler) | 71 | s | 3,1 |
| mass of steam bubbles, = 70 ÷ 56 = | 72 | kg | 1,15 |
| specific volume of water, = | 73 | $m^3\ kg^{-1}$ | 0,00115 |
| Mass of water in the boiler, = 16 ÷ 73 = | 74 | kg | 5200 |
| A limit hypothesis is that water velocity is equal to that of bubbles (i.e. there is no slip between bubbles and water). Hence the total water flow is 44 · $\frac{74}{72}$ = | 75 | $kg\ s^{-1}$ | 1630 |
| This corresponds to a volumetric water flow of 75 · 73 = | 76 | $m^3\ s^{-1}$ | 1,9 |
| Water mass / bubble mass, = 74 ÷ 72 = | 77 | – | 450 |
| Water volume / bubble volume, = 16 ÷ 70 = | 78 | – | 3,7 |
| Volume dryness fraction, average value referred to the whole boiler, = 1/(1 + 78) = | 79 | – | 0,21 |

However, the upward bubble velocity (about 0,25 $m\ s^{-1}$) is not negligible compared to water velocity. Therefore a correction factor is to be incorporated.

| | | | |
|---|---|---|---|
| Slip correction factor adopted, = | 80 | – | 0,8 |
| Hence water flow is $(44) \cdot \frac{(74)}{(72)} \cdot (80) =$ | 81 | $kg\ s^{-1}$ | 1300 |
| This corresponds to a volumetric water flow of $(81) \cdot (73) =$ (Flow T + F, Fig. 3) | 82 | $m^3 s^{-1}$ | 1,5 |
| Water mass / bubble mass, = $(74) \div (72) =$ | 83 | – | 450 |
| Water volume / bubble volume, = $(16) \div (70) =$ | 84 | – | 3,7 |
| Volume dryness fraction, average referred to the whole boiler, = $1/(1 + (78)) =$ | 85 | – | 0,21 |
| Average time for water to run the whole loop (Fig. 3), = $((70) + (16))/(82) =$ | 86 | s | 5 |

On travelling upwards, both flows T and F flash part of their mass because the static pressure becomes lower. Bubbles are, on the average, generated about one metre below the surface; hence the pressure drop is roughly 0,08 at. Therefore

| | | | |
|---|---|---|---|
| Pressure in the steam space, = (4) + 1 at = | 87 | ata | 14,5 |
| Pressure at 1 m below the surface, = (87) + 0,08 at = | 88 | " | 14,58 |
| Water enthalpy change between pressures (88) and (87), = | 89 | $kcal\ kg^{-1}$ | 0,272 |
| Total heat brought into play, = $(81) \cdot (89) =$ | 90 | $kcal\ s^{-1}$ | 353,6 |
| Heat of evaporation, = | 91 | $kcal\ kg^{-1}$ | 467 |

| | | | |
|---|---|---|---|
| Steam produced by flash, per second, = 90 ÷ 91 = | 92 | $kg\ s^{-1}$ | 0,757 |
| Ditto per hour, = which is a far from negligibly amount. | 93 | $kg\ h^{-1}$ | 2725 |
| The heat involved in flash, per hour, is 90 · 3600* = | 94 | $Gcal\ h^{-1}$ | 1,27 |

Since the reverse effect obtains in the downward part of the flow (R, Fig. 3), bubbles present at U will collapse, and water will reach and run through the bottom part of the boiler in a subcooled condition. Were it not because of this, the heat available to evaporate steam in the small tubes is 47 = 1.2 $Gcal\ h^{-1}$. This figure is roughly equal to 94, hence it can be concluded that no bubbles will be produced at the bottom of the barrel in the region occupied by the small tubes: heat transferred there will go to warm up the cooler inlet feed and rise water temperature from the subcooled condition. Nevertheless, heat transfer will be still high under a <u>subcooled boiling</u> régime.

As a result of the above reasoning, very little steam is produced on the first metre of tube length since this comes from large tubes heating surface, which is comparatively small and has a rather low convection heat transfer because its length / diameter ratio is small too (because superheater bends are far from the tube plate). This leads to the conclusion that perhaps 90 to 95 % of the bubbles born in the firebox area, a surprising result since only 39% = 25 of the heat is transferred there.

The fact that most of the heat transferred in the tube bundle is carried out in the "subcooled boiling régime" implies that the phisyco-chemistry of the boundary layer contacting the heating surface of those tubes behaves differently than what is supposed concerning scaling and suspended solid baking. This is outside the scope of the present work, but must be recorded here. A parallel effect obtains at the firebox end: a large proportion of the heat goes to steam bubbles growth after they have left the heating surface - since they grow up because of heat transferred from the superheated liquid phase. This superheat comes both from contacting the firebox heating surface and from the pressure reduction associated with the upward flow.

It is also

| | | | |
|---|---|---|---|
| Total steam produced by the boiler, = (44) = | (96) | $kg\ s^{-1}$ | 3,6 |
| Steam volume, = (96) · (56) = (57) = (58) = | (97) | $m^3\ s^{-1}$ | 0,5 |
| Fraction assumed to be produced at the back end, = | (98) | – | 0,95 |
| Water flow = (82) = | (99) | $m^3\ s^{-1}$ | 1,5 |
| "Volume dryness" fraction, at back end, = (97) · (98) / ((97) + (99)) = | (100) | – | 0,24 |
| "Weight (mass) dryness" fraction at back end (title of steam), = (96) · (98) / ((96) + (81)) = | (101) | – | 0,0026 |

This figure is exceedingly small as compared to what is usual in higher pressure water tube boilers.

| | | | |
|---|---|---|---|
| The surface of the evaporation plane at the back end, considering the reduction resulting from swelling, is ≈ | 102 | $m^2$ | 4,5 |
| Hence the steam velocity in traversing that plane is 97 ÷ 102 = | 103 | $m\ s^{-1}$<br>$ft\ s^{-1}$ | 0,11<br>0,36 |

An interesting parameter concerning boiling heat transfer and bubble growth is the average life time of the bubbles.

| | | | |
|---|---|---|---|
| Assuming that back end volume is equal to (volume around the firebox + swelling volume), = 37 + 70 = | 104 | $m^3$ | 3,6 |
| The average life time of bubbles at the back end is approximately [104 / (82 + 58)] · 80 = | 105 | s | 1,4 |

This time is considerably greater than the usual 0,1 s considered in boiling heat transfer experiments. Therefore the amount of heat transferred to bubbles is considerable, hence requiring a revision of current heat transfer theory (1).

Some idea of the average velocity of the stream in firebox water passages can be estimated considering that the path of the mean fillet is 5 m long. (Fig 1). Hence

| | | | |
|---|---|---|---|
| Length of the path, = | 106 | m | 5 |
| Stream velocity, = (106 ÷ 105) · 80 = | 107 | $m\ s^{-1}$ | 2,9 |

This velocity is high enough to influence favourably the boiling heat transfer and the burnout critical flux. It corresponds to a mass velocity of

| | | | |
|---|---|---|---|
| 107 ÷ 73 = | 108 | $kg\ m^{-2}\ s^{-1}$ | 2500 |

| | | | |
|---|---|---|---|
| Ditto in $kg\ m^{-2}\ h^{-1}$, = | 109 | $\frac{kg \cdot 10^6}{m^2\ h}$ | 9,1 |
| In English units, = | 110 | $\frac{lb \cdot 10^6}{ft^2\ h}$ | 1,9 |