GESTRA Steam Systems

GESTRA Information A 1.4

Trapping of Steam Tracers with Elevated Steam Traps

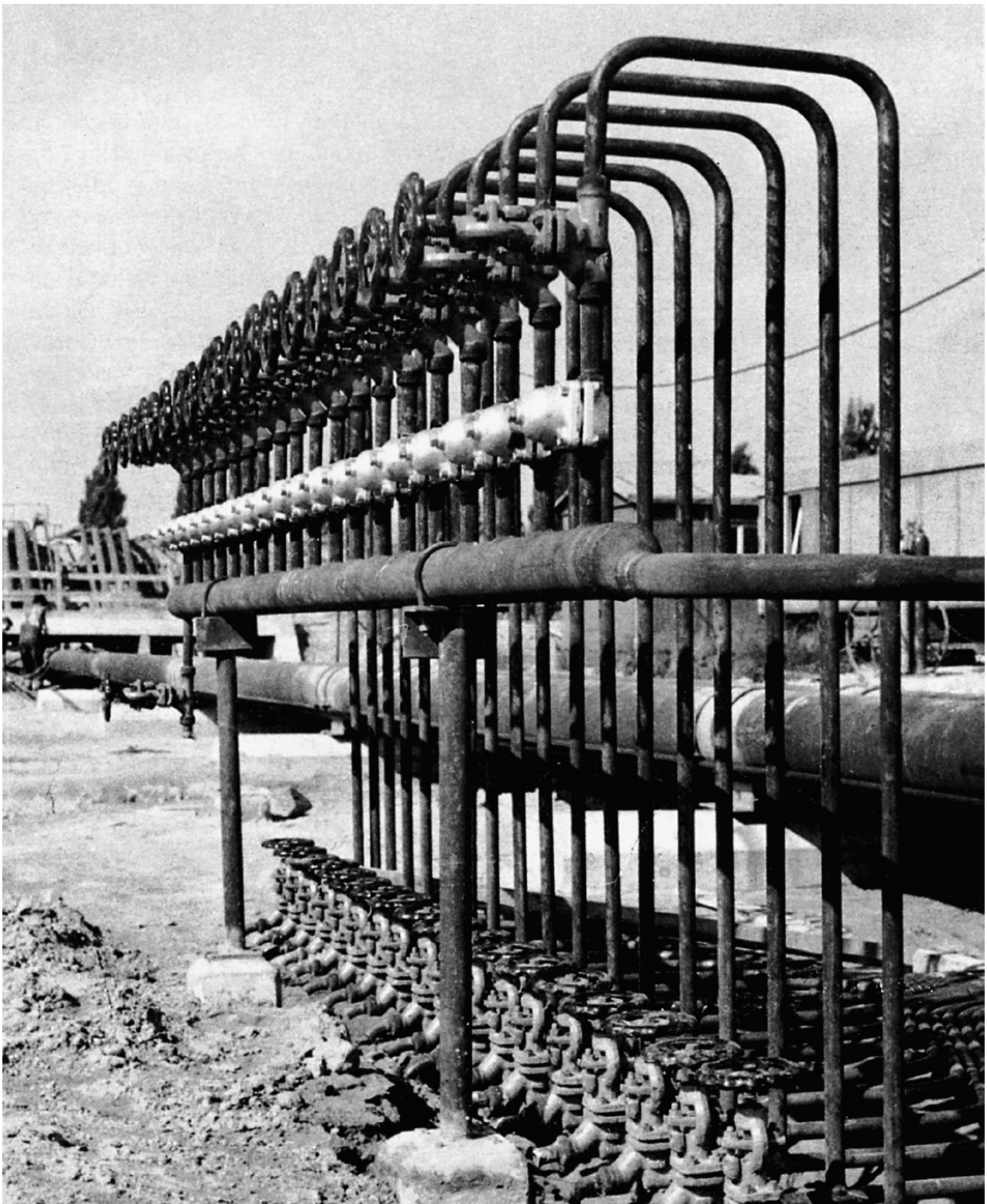


Fig. 1

It has, in the past, been normal practice in refineries to arrange the steam traps of steam tracers just above the ground, on the ground or even in trenches below ground level. Each of these arrangements results in disadvantages, which it would be desirable to eliminate.

Steam traps installed at a low level **(Fig. 2)** are generally subject to considerable external contamination caused by the opening of the free drainage valves or by earth, sand and sludge being thrown up during heavy rain. In either case water accumulating on the ground might even result in flooding of the traps. These external influences inevitably affect the proper operation and service life of the traps. In winter, there is the additional danger that the traps may freeze up. For the same reasons locating damage becomes more difficult. Moreover, accidents can easily occur as a result of poor accessibility for servicing or repairs.

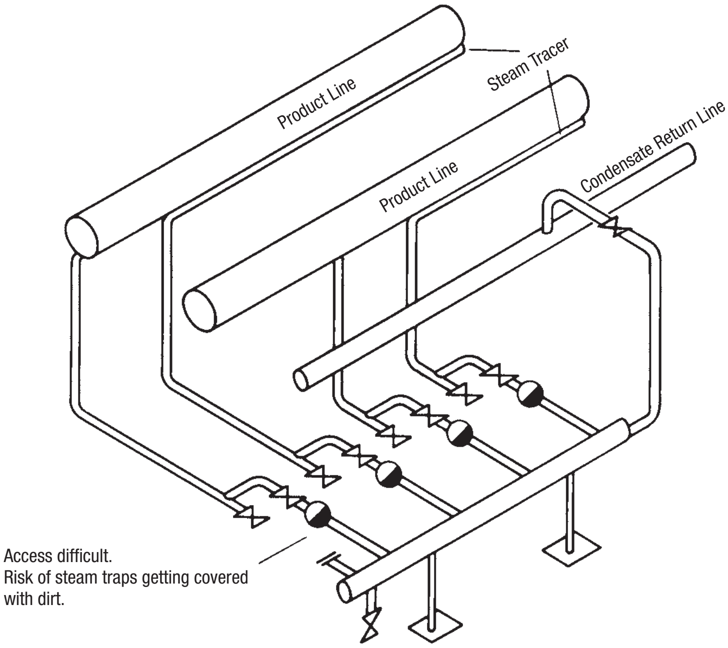


Fig. 2

In view of the disadvantages of low level traps an obvious solution is to lift the condensate to a higher level upstream of the traps and to arrange the latter in groups at a readily-accessible elevated level.

Recently, numerous tracers have been constructed with the traps arranged in the suggested manner **(Fig. 1** and **3)**.

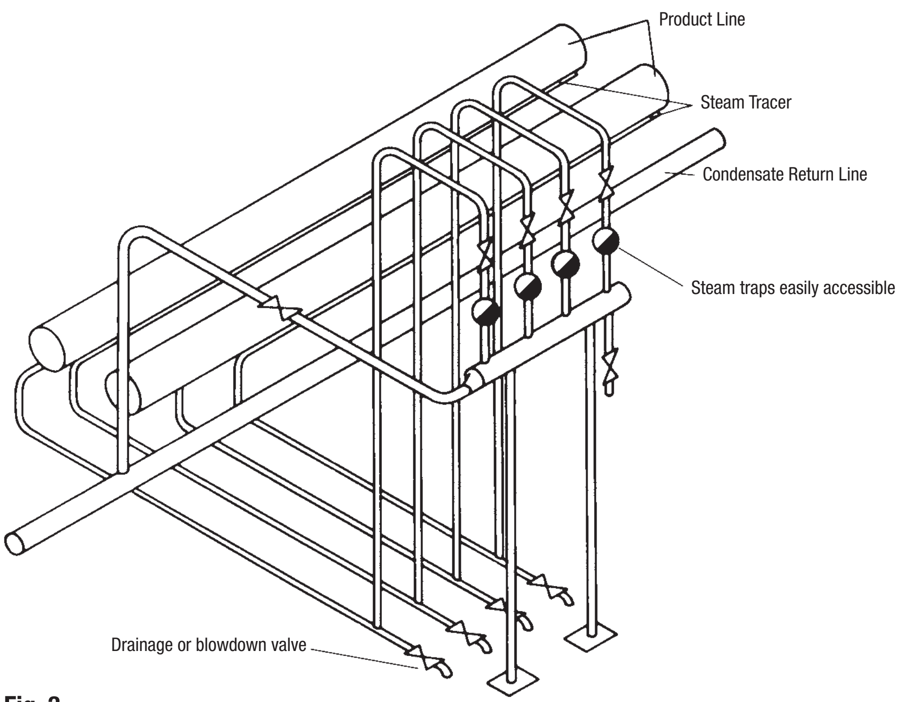


Fig. 3

The design engineers were apprehensive about lifting the condensate before it reached the traps, as two-phase current might occur in the ascending line; the steam portion of which might have a closing effect on the trap and hinder condensate discharge.

Our suggestion, to lift the condensate upstream of the trap, is based on the results of exhaustive tests.

Theoretical bases

Since in the case of elevated steam traps a two-phase current has to be reckoned with, the following valves are essential to obtain an accurate layout of the pipelines:

1. Steam drift
2. Phase friction
3. Pressure drop
4. Heat transfer

Today, it is known that the steam drift in a vertical pipe section is calculated as follows [1]:

$$S \equiv \varepsilon^* - \varepsilon = 0.71 \cdot \varepsilon^* \cdot (1 - \varepsilon^*)^{0.5} \cdot Fr_{LM}^{-0.045} \cdot (1 - p^*)$$

ε^* ≙ Volume proportion of steam related to flow cross-section

ε ≙ Volume proportion of steam related to distance under consideration

Fr ≙ Froude's number

p^* ≙ reduced pressure; $p^* = \frac{p}{p_{kr}}$

With a single-phase current $\varepsilon^* \equiv \varepsilon$. Consequently, the steam drift S disappears, since with a single-phase current the volumetric and the mass steam proportion will be zero or one. In the above equation ε^* and p^* are unknown. Both, Froude's number – here related to a liquid with the mass flow density of the two-phase mixture – and the reduced pressure can easily be calculated.

The following equation [2] applies to the still unknown ε^*:

$$\varepsilon^* = \frac{V_G^*}{V_G^* + V_L^*} = \frac{v_G \cdot \varepsilon A}{v_G \cdot \varepsilon A + v_L \cdot (1 - \varepsilon) A}$$

V_G^* ≙ Volume of steam related to flow cross-section

V_L^* ≙ Volume of liquid related to flow cross-section

A ≙ Flow cross-section ≙ $\frac{\pi D^2}{4}$

V_L ≙ Flow velocity of liquid ≙ $\frac{V_L^*}{(1 - \varepsilon) A}$

V_G ≙ Flow velocity of steam, according to Nicklin [3, 4]:

$$V_G = 1.2 \, \frac{V_G^* + V_L^*}{A} + 0.35 \cdot \sqrt{g \cdot D}$$

The term $\frac{V_G^* + V_L^*}{A}$ designates the velocity of the mixture.

The factor 1.2 is established by the flow velocity of the rising bubbles, since they do not ascend at the average flow velocity of the mixture, but are carried upward at a velocity almost the same as the maximum velocity existing on the pipe axis. The second term allows for the drift and is indicated by the velocity of the rising steam bubbles in still water.

The frictional losses can be determined either by the Martinelli method [5] or to a better approximation (according to **Nicklin** [3, 4]) using the equation:

$$\left(\frac{\Delta p}{\Delta l}\right)_f = \left(1 - \varepsilon\right) \left(\frac{\Delta p}{\Delta l}\right)_L = \frac{4\tau}{D}$$

τ ≙ Shearing stresses

$\left(\frac{\Delta p}{\Delta l}\right)_L$ ≙ Pressure gradient that would result if only water at the velocity of the mixture would flow through.

By an impulse balance in the upper part of the ascending pipe we find [2]:

$$\frac{\Delta p}{\Delta l} = \rho \cdot g \cdot \sin\alpha + \left(\frac{\Delta p}{\Delta l}\right)_f + \frac{1}{\Delta l} \cdot \left\{ \left[\rho_G \cdot \varepsilon \cdot v_G^2 + \rho_L \cdot (1 - \varepsilon) \cdot v_L^2\right]_1 - \left[\rho_G \cdot \varepsilon \cdot v_G^2 + \rho_L \cdot (1 - \varepsilon) \cdot v_L^2\right]_2 \right\}$$

In this case 1 and 2 indicate the control selection of the impuls observation and sin α the inclination of the ascending pipe section (for a vertical pipe sin α = 1, α = 90°).

In the equation $\frac{\Delta p}{\Delta l} = \ldots$,
the terms indicate the following:

$\rho \cdot g \cdot \sin\alpha$ ≙
Consideration of gravity, where

$\rho = \varepsilon \cdot \rho_G + (1 - \varepsilon) \cdot \rho_L$ ≙ the mean density of the mixture

$g = 9.81 \text{ m/s}^2$ ≙ acceleration due to gravity

The terms in brackets indicate the acceleration at the points 1 and 2 and, therefore, also the difference in the volume proportion ε. In general $\varepsilon_2 > \varepsilon_1$, for the following reasons:

The steam expands as the pressure decreases at a gradient $\frac{\Delta p}{\Delta l}$.

Consequently the boiling point of the mixture is lowered so that the flash evaporation occurs which is, however, compensated for by heat losses.

It is, therefore, possible to calculate resonably closely the ascending heigh Δl of the condensate. Another factor to be considered is that the heat losses cause some condensation of the rising steam, so that an additional condensate discharge is brought about by the pressure difference resulting from the lower pressure.

Heat transfer effects have not yet been considered and no standard formulae have been evolved.

The flow pattern which is of no great importance in this study can be taken from **Bakers'** Diagram [2, 6].

Laboratory tests

The problem of the two-phase flow in pipelines has been under investigation for a number of years and several articles have already been published on this subject.

We have also studied the practical aspects of this subject. A glass test stand was constructed to enable the phenomena to be observed and assessed.

The test stand **(Fig. 4)** is almost the same size as the actual steam-tracer drain stands. Actual plant conditions were reproduced during the tests wherever this was possible.

In the horizontal section of the steam tracer the steam flows through the line at a gradually decreasing velocity. During this process the amount of steam decreases and the amount of condensate increases through further condensation. The steam flow pushes the condensate in the direction of the ascending pipe and the phase friction causes a wave formation as can be seen in **Fig. 5** and **Fig. 6**.

During transition from the horizontal to the vertical part of the line the condensate hits the 90° elbow and is deflected so that turbulence results.

When lifting the condensate there are various forms of flow which mainly depend on the amount of condensate [2].

According to tests carried out by **Kowalczewski** [7] this is of no importance since the pressure drop, the steam drift, and the phase friction remain practically unchanged during the transition from one form of flow to another.

If the amount of condensate is small, the rising steam will at first entrain the condensate. There is a continuous thin stream of steam in the line which in its lower part is surrounded by condensate. Initially, the amount of condensate is not sufficient to form a water plug. With decreasing steam velocity during upward flow the friction also decreases. The water runs down on the inner pipe walls until sufficient condensate collects to form a water plug. When this has formed it blocks the flow to the upper part of the line, causing a faster rate of condensation and correspondingly greater pressure reduction, with the result that the plug is suddenly carried upwards.

This discharge of water plugs is repeated continuously at regular intervals.

Fig. 5 shows the transition from the horizontal to the vertical part of the test stand during operation at Δp of 23.5 psi (1.5 Atm.) with very little condensate. The intense turbulence in the elbow and the condensate which is entrained by the steam can be clearly seen.

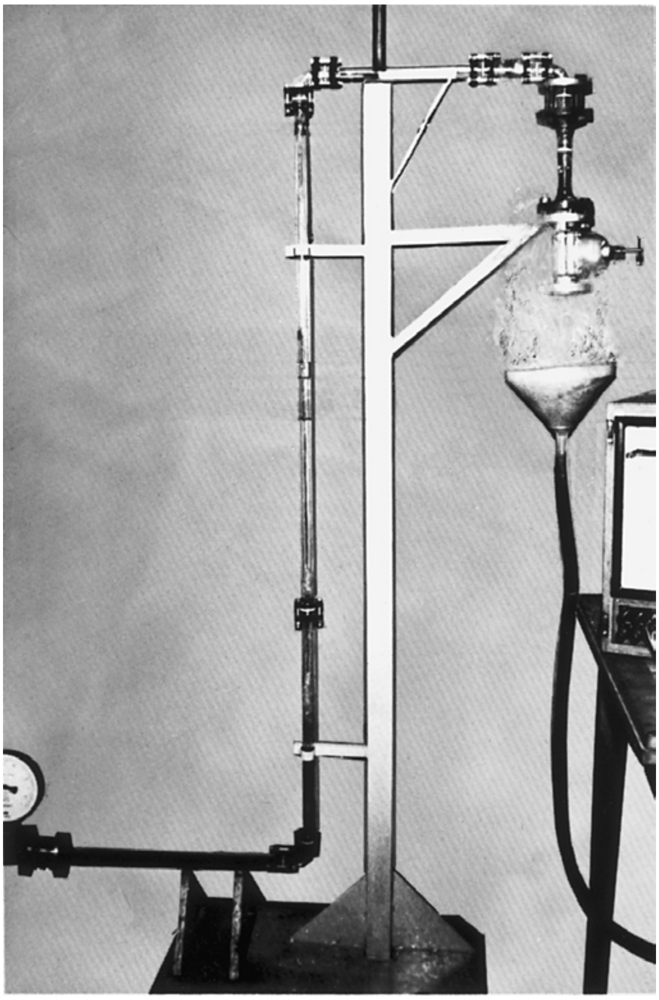


Fig. 4

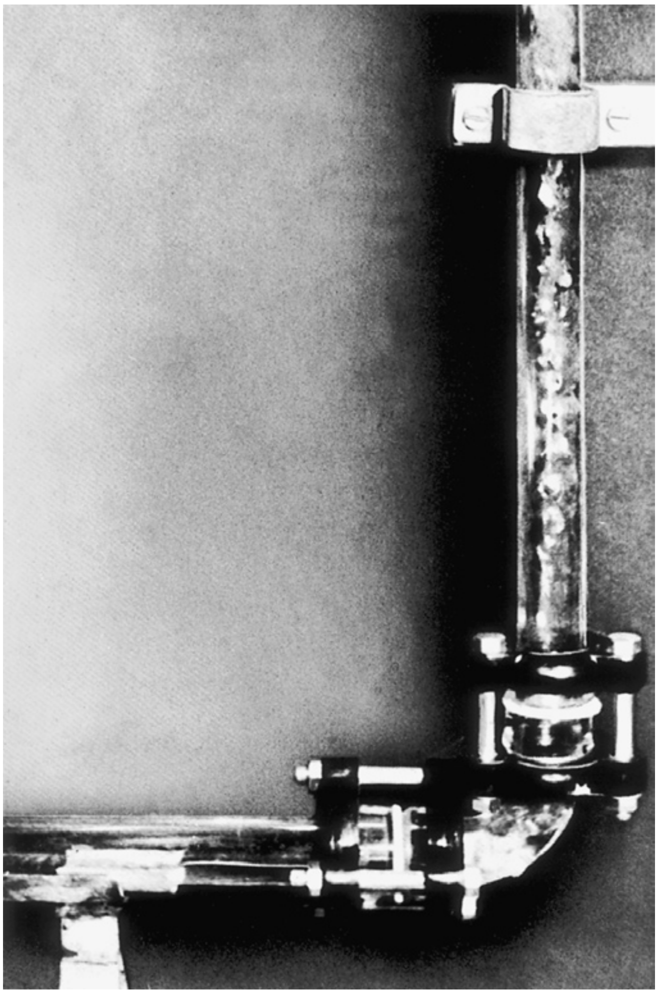


Fig. 5

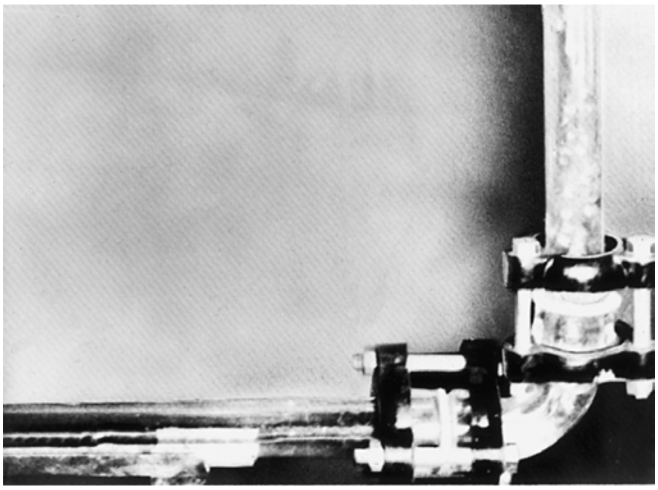


Fig. 6

Fig. 6 shows the same part of the test stand, but with a larger quantity of condensate. The turbulence in the elbow is less now. The thin stream of steam surrounded by condensate is a little shorter. The formation and discharge of the water plug occurs more frequently.

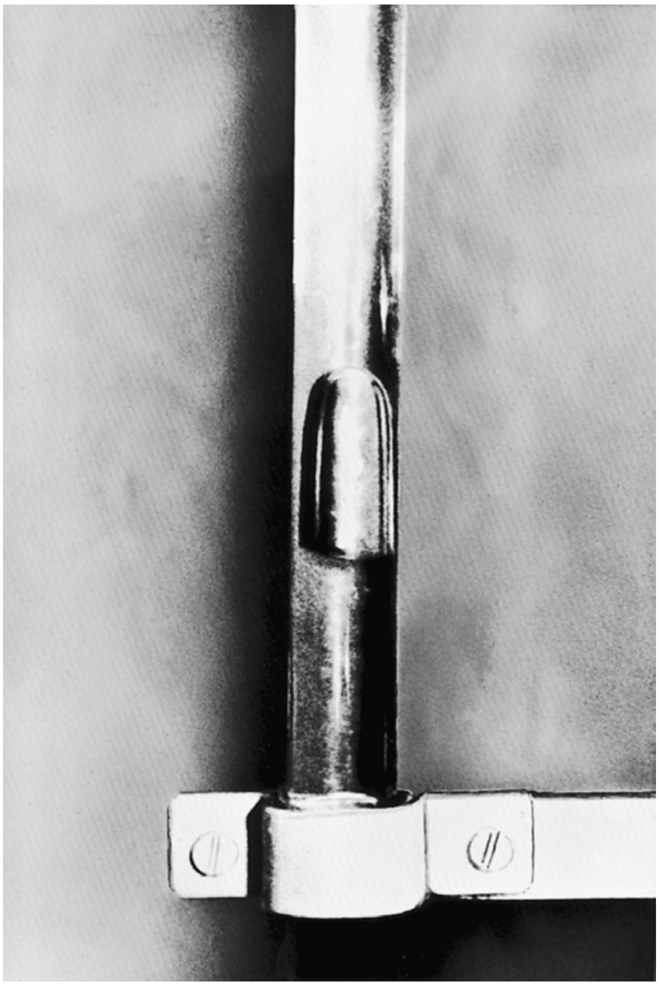


Fig. 7

In **Fig. 7** the amount of condensate formed is such that the water plugs are formed immediately at the point of deflection. The water is carried upwards by ascending steam bubbles, and this process is supported by the pressure reduction resulting from the condensation of the steam bubbles in the horizontal pipe section which follows the ascending pipe (**Fig. 4**). A return flow of water drops can no longer be observed. The steam bubbles ascending in the pipe more or less maintain their size, because the previous reduction in size of the bubbles on account of heat losses is now compensated by the increase in volume during flashing. There is only a minor formation of flash steam. The condensate formed is in most cases colder than the boiling temperature at the applied pressure. **Kirschbaum** [8] proved that the condensate may be colder than the steam even if both phases are in one space. In this case, the condensate temperature depends, among other things, on the thickness of the water wall.

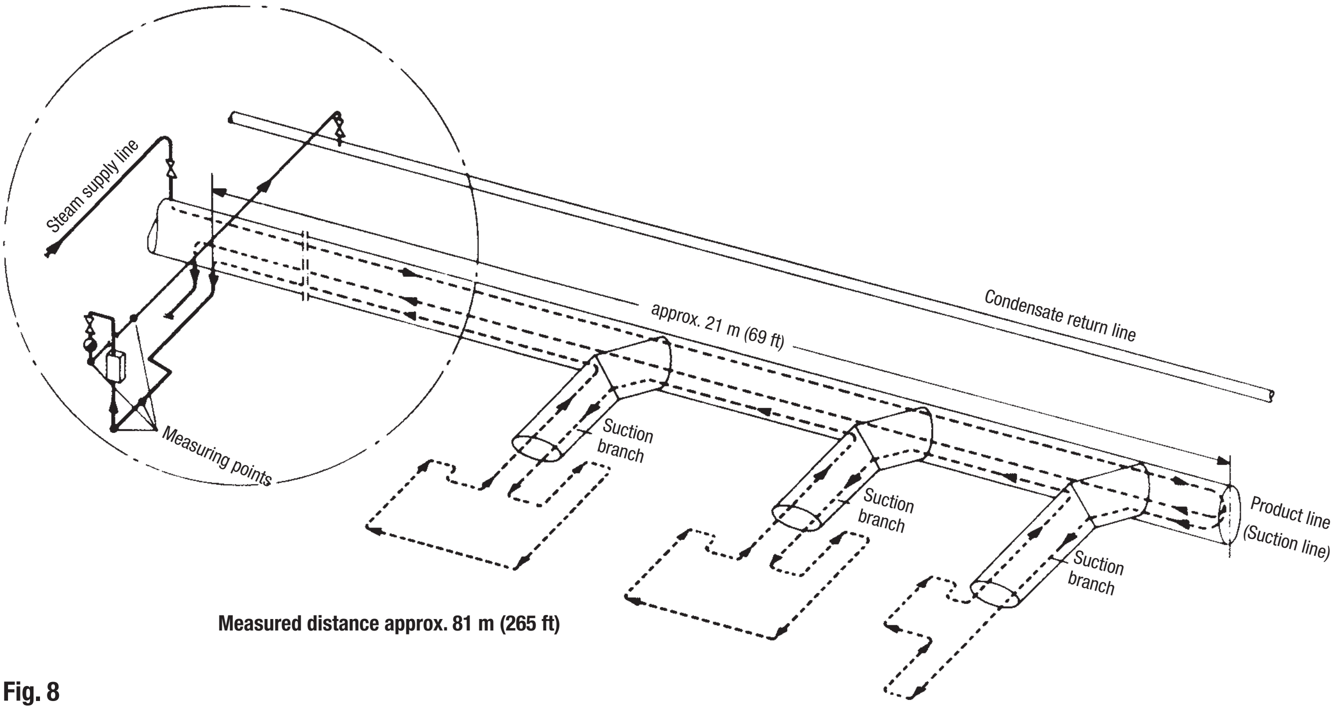
If the amount of condensate formed is so high that the condensate can no longer properly be discharged because the trap has reached its capacity, only condensate will be found in the ascending pipe.

A water lock will already have formed in the horizontal line. The discharge of the condensate downstream of the water lock is then effected by pressure difference only.

The same basic setting of the traps was maintained throughout the tests.

Application in practice

The satisfactory operation of plants already in use is explained by these observations. In all part load conditions the condensate in the steam tracer drains is satisfactorily discharged using the arragement shown here.



Measured distance approx. 81 m (265 ft)

Fig. 8

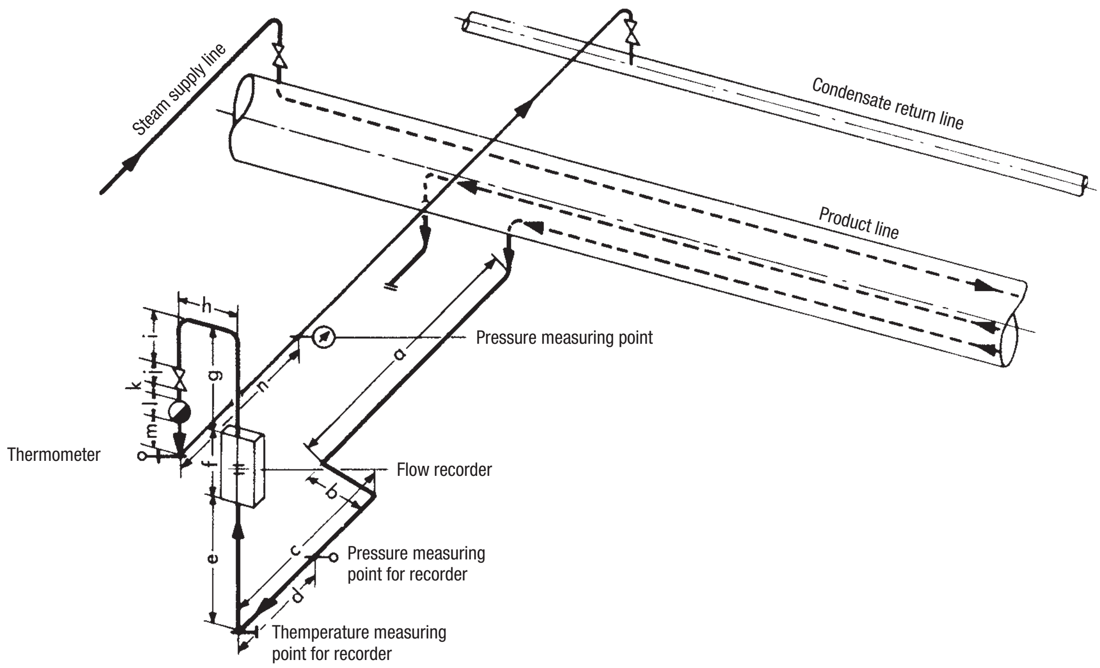
In one refinery some of the steam tracers were equipped with elevated steam traps. One of these steam tracers was selected to monitor that the traps did operate satisfactorily with this arragement. **(Fig. 8)**

The section measured serves to heat the common suction line of 3 circulating pumps and the associated suction branches. The suction line of size DN 350 (14") has to be heated by three pipes. Tracer inlet and tracer outlet are close together, approx. 1 m (3¼ ft). The tracer is led along the full length of the suction line (approx. 21 m = 69 ft). It loops at the end and divides into two individually drained return lines one of which returns in a straight run to its outlet point, while the other traces the suction branches, gate valves and pumps besides the product line. Our measurements were concerned with these supply and return tracers heating the suction branches and gate valves and having a total length of approx. 81 m (265 ft). The level difference between the centre of the suction line and the centre of the suction branch is approx. 0.6 m (2 ft). No free drainage point is provided on this measuring section.

The arrangement of the measuring points is shown in the diagrammatic representation of **Fig. 9**. A Withof triple-type recorder having a measuring range from 0 – 25 bar (0 – 350 psig) was used for measuring the upstream and back pressures. The temperature upstream of the trap was measured with an electric Siemens recorder at the base of the ascending pipe.

A Krohne rotameter with a pneumatic transmitter and an electric Siemens recorder for recording the flow were installed in the ascending pipe and a thermometer 0 to 250 °C, was arranged downstream of the trap. Condensate discharge is effected by means of Duo Steam Traps, type BK 15, which are suitable for all operating conditions up to 300 psig / 400 °C (750 °F), and which were also used for the tests **(Fig. 10)**.

a = 0.85 m (33")
b = 0.55 m (22")
c = 0.75 m (30")
d = 0.50 m (20")
e = 0.60 m (24")
f = 0.32 m (13")
g = 0.48 m (19")
h = 0.28 m (11")
i = 0.23 m (9")
j = 0.10 m (4")
k = 0.05 m (2")
l = 0.10 m (4")
m = 0.16 m (6")
n = 0.80 m (31")



Measured distance approx. 81 m (265 ft)

Fig. 9

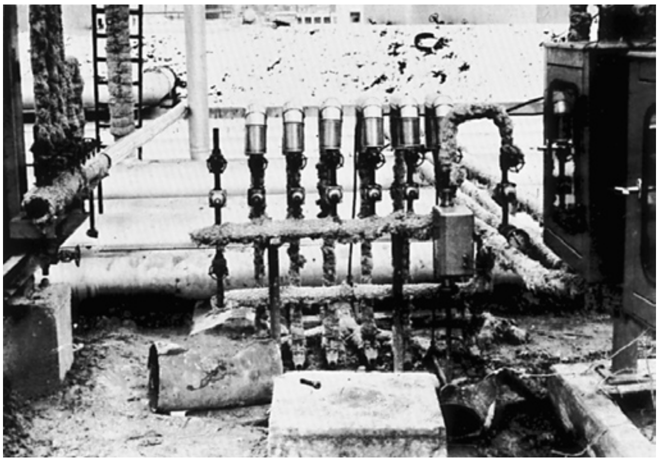


Fig. 10

The arrangement with the flow meter in the ascending pipe is rather unfavourable because, with a normally adjusted trap, the ascending steam bubbles and condensate plugs caused a deflection between 0 and 100 %. The flow meter was calibrated for water of 95 °C (γ = 0.961 kg/dm^3). In practice, however, the installation at this point was inevitable since the flow had to be upward. Consequently, only the results of the measurements obtained with the traps adjusted to allow banking-up of condensate could be evaluated.

Two tests each lasting 16 hours were carried out. In each case the measurements started at the same time, the values were recorded by measuring instruments.

The first test and measurements were undertaken with a normally adjusted trap and the diagrams showed the following results:

Pressure upstream of trap:
3.5 bar (49.8 psig) (t_S = 147.2 °C [297 °F])

Temperature upstream of trap:
142 °C – 147 °C (287.6 °F – 296.6 °F) (recorder)

Pressure downstream of trap:
2.0 bar (28 psig) (t_S = 132.8 °C [271 °F])

Temperature downstream of trap:
approx. 120 °C (248 °F) (thermometer)

A temperature of 125 °C (257 °F) was measured immediately upstream of the trap with temperature measuring powders.

During the whole period the flow meter fluctuated.

The temperature characteristic showed that the condensate temperature at the measuring point upstream of the trap was approx. 5 deg C lower during the night hours – with frost down to –5 °C (23 °F) (according to the weather station) – than during the hours of daylight.

During the second test the trap was adjusted to a narrower setting, i.e. the capacity of the trap was reduced. During this test, the condensate amount was correctly indicated by the flow meter. The following values resulted during the measuring period:

Pressure upstream of trap:
3.5 bar (49.8 psig)

Temperature upstream of trap:
varying between 123 °C – 135 °C (253 °F – 275 °F) (recorder)

One fluctuation lasted approx.
1 – 1½ hours

Pressure downstream of trap:
2.0 bar (28 psig)

Temperature downstream of trap:
98 °C (208 ° F) (thermometer)

A temperature of 105 °C (221 °F) was measured immediately upstream of the trap with temperature measuring powders.

The volume recorder recorded a quantity varying between 25 and 40 % ≙ 18 to 28 l/h (3.9 to 6.2 imp. gal.).

The test carried out with the normally adjusted trap clearly shows that no condensate was banked up to the measuring point at the base of the ascending line. Since, at this point, the temperature feeler was partly in contact with the condensate in the lower section of the pipe and since the condensate temperature must be lower than the steam temperature the difference between the measured temperature and the boiling point temperature results. The temperature difference between the recorder measuring point and the trap is explained by the lifting of the condensate which caused an expansion and, consequently, a temperature drop; the heat losses caused by a partly non-insulated pipeline resulted in a further temperature drop. Through heat loss the temperature difference upstream and downstream of the trap is higher than that resulting from pressure drop alone. The condensate was discharged by the differential pressure only.

With regard to this test, it can be said that condensate discharge was satisfactory with a normally adjusted trap. With a narrower setting of the trap the condensate was also drained without any difficulty, but with a corresponding amount of undercooling.

The practical tests and measurements with a glass test stand and an actual steam tracer system have shown that condensate will be properly drained, if the steam traps are arranged in an elevated position.

Consequently the tracing lines are satisfactorily drained irrespective of the condensate quantity as long as it is within the maximum capacity of the traps.

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