

Feb. 23, 1965

J. SCOTT, JR., ET AL

3,170,477

STEAM TRAP

Filed Aug. 2, 1961

4 Sheets-Sheet 1

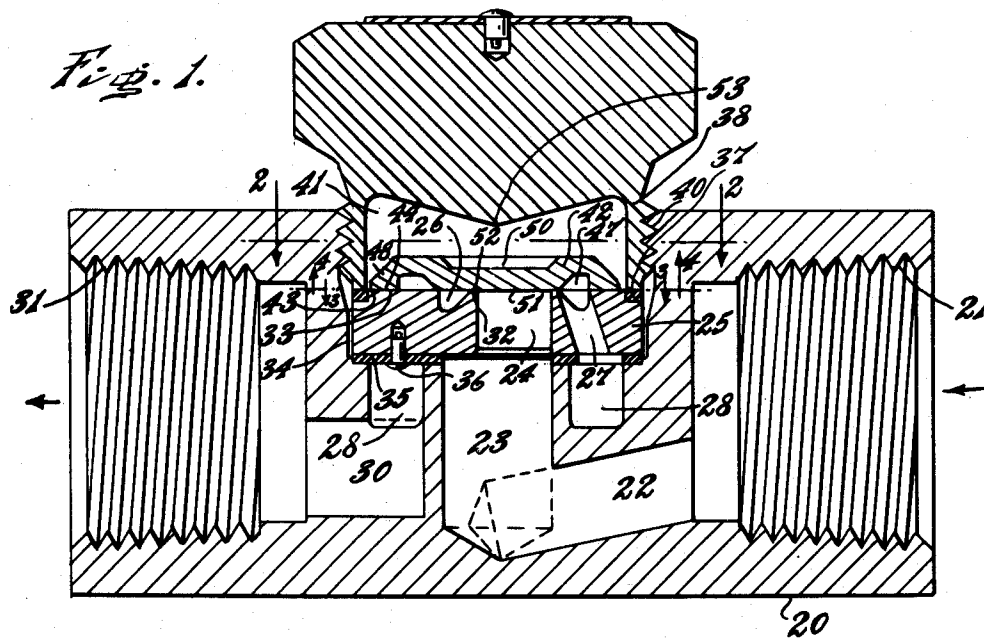


Fig. 2.

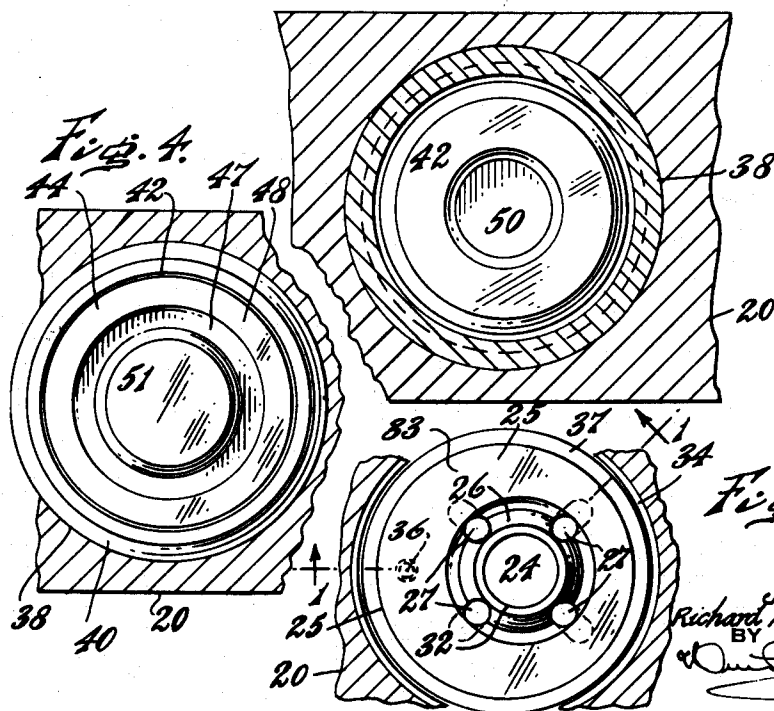


Fig. 3.

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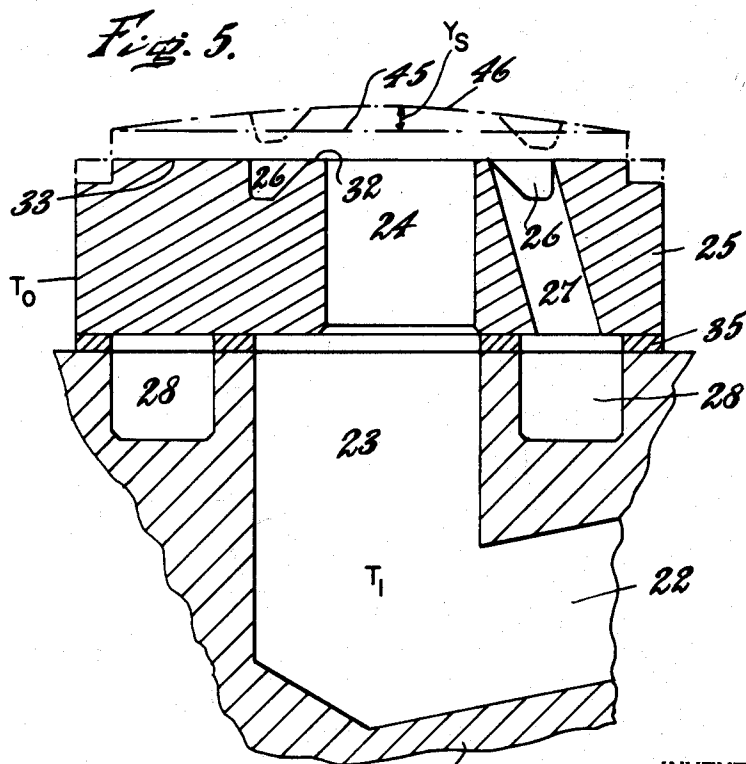
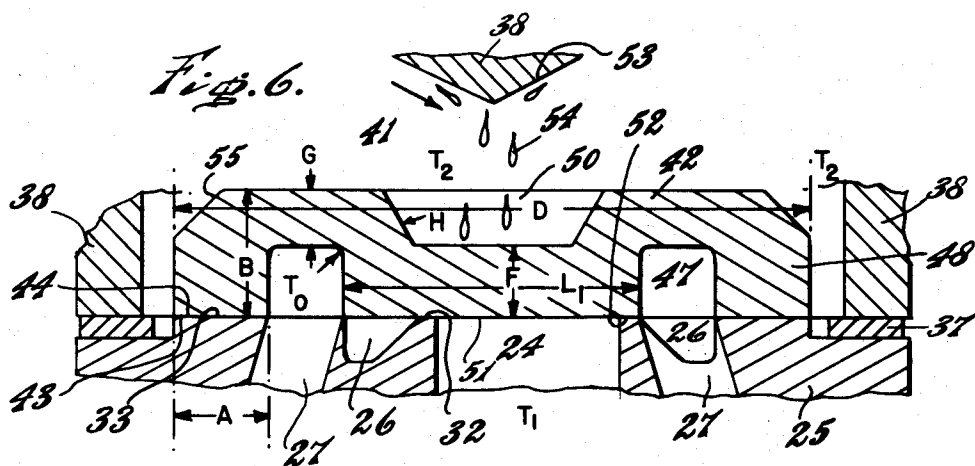
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STEAM TRAP

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STEAM TRAP

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Fig. 7.

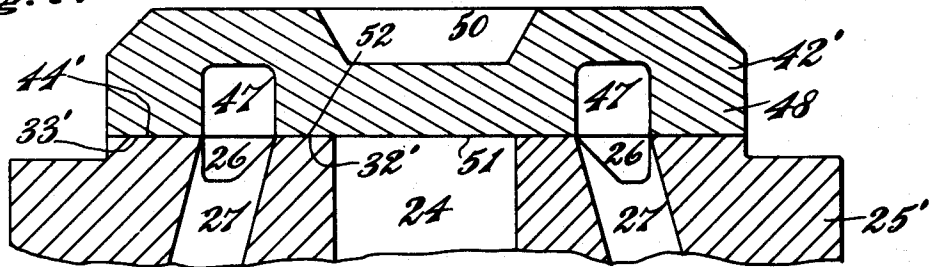


Fig. 8.

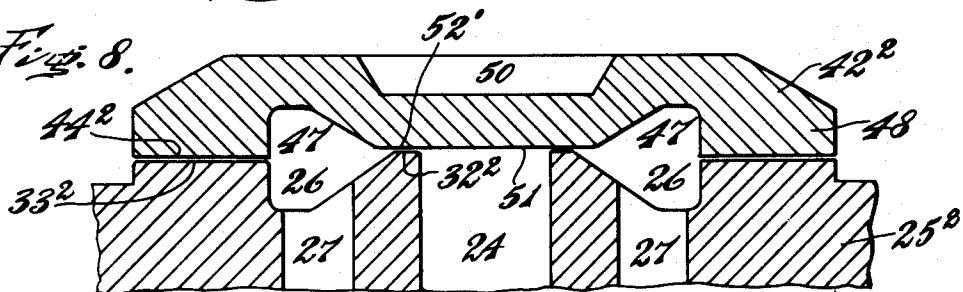


Fig. 8a.

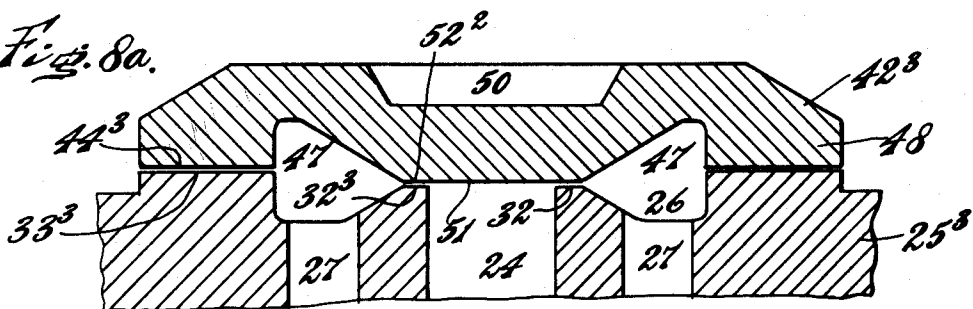
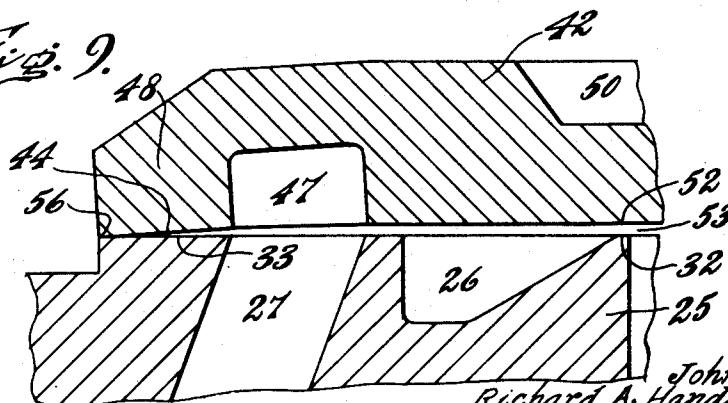


Fig. 9.



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Fig. 10.

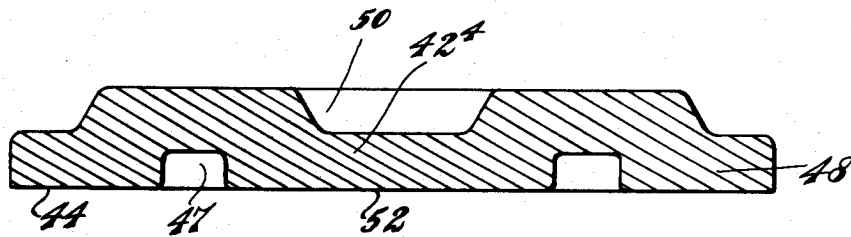
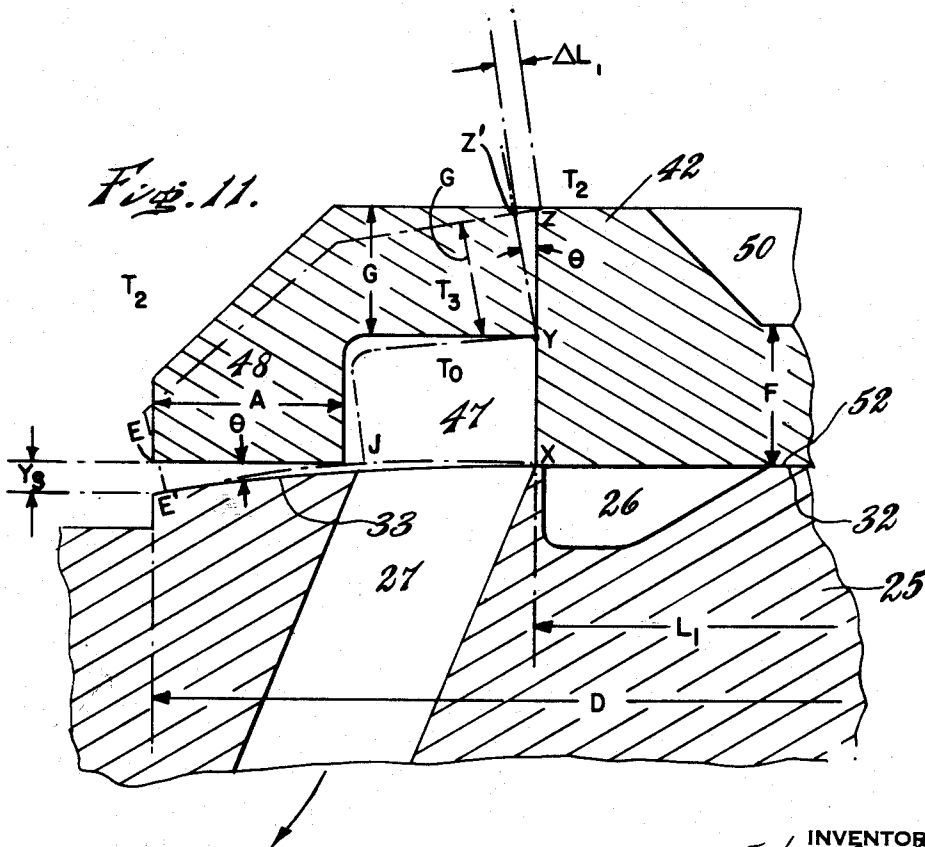


Fig. 11.



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STEAM TRAP

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Filed Aug. 2, 1961, Ser. No. 128,821

18 Claims. (Cl. 137—183)

The present invention relates to steam traps of the type which are useful to remove condensate from steam spaces such as steam lines.

A purpose of the invention is to permit a steam trap of the character which responds to control chamber pressure to operate with relatively little change in opening cycle over a wide variation in operating pressure.

A further purpose is to render a steam trap of the type which has a valve plate or disc operating in response to control chamber pressure less sensitive to the harmful deflection of the valve seat due to temperature differentials.

A further purpose is to provide a cup recess near the middle of the top of the valve plate and suitably collect condensate in said cup recess.

A further purpose is to direct condensate into the cup recess by a downwardly converging wall from the top of the control chamber.

A further purpose is to provide a relatively thin wall of the valve plate opposite the cup recess as compared with the thickness of the valve plate toward the outside of the valve plate.

A further purpose is to provide a relatively thick outer ring or rim on the valve plate and a relatively thin middle portion so as to compensate for differential temperature effects which would otherwise tend to change the cycle time of opening and closing of the valve plate.

A further purpose is to provide a relatively thin cross section of the valve plate between the central cup-like portion and the outer rim or ring so as to cause thermal restriction.

A further purpose is to provide a groove on the side of the valve plate toward the inlet and outlet and opposite the point of thermal restriction.

A further purpose is to make the thickness of the valve plate opposite the cup-like recess and also desirably at the thermal restriction in the range between $\frac{1}{8}$ and $\frac{1}{20}$ and preferably about $\frac{1}{10}$ the outside diameter or dimension of the valve plate.

A further purpose is to proportion the outer rim or ring with a relation of width to thickness which is between 2 to 1 and 1 to 2.

Further purposes appear in the specification and in the claims.

In the drawings we have chosen to illustrate a few only of the numerous embodiments in which the invention may appear, selecting the forms shown from the standpoints of convenience in illustration, satisfactory operation and clear demonstration of the principles involved.

FIGURE 1 is a vertical section of the preferred embodiment of steam trap of the invention, on the line 1—1 of FIGURE 3.

FIGURE 2 is a horizontal section on the line 2—2 of FIGURE 1.

FIGURE 3 is a horizontal section on the line 3—3 of FIGURE 1.

FIGURE 4 is a horizontal section on the line 4—4 of FIGURE 1.

FIGURE 5 is an enlarged section corresponding to a portion of FIGURE 1, illustrating diagrammatically the effect of forces on the valve seat due to temperature differentials.

FIGURE 6 is a diagrammatic section corresponding

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to a portion of FIGURE 1, showing the action of the bonnet and showing various dimensions and temperatures used in the computations.

FIGURE 7 is a diagrammatic fragmentary vertical section showing a variation in the steam trap valve and seat of the invention.

FIGURES 8 and 8a are views similar to FIGURE 7 showing further variations in the steam trap valve and seat.

FIGURE 9 is a fragmentary central vertical section showing the effect of differential temperature forces on the valve plate.

FIGURE 10 is a central vertical section of a modified valve plate according to the invention.

FIGURE 11 is a fragmentary axial section of a valve plate and seat useful in explaining the invention.

Describing in illustration but not in limitation and referring to the drawings:

The present application relates to improvements in a steam trap and particularly in the valve plate, and seat, and control chamber of a steam trap of the character of Yarnall and Kinderman United States Patent No. 2,951,496, granted September 6, 1960, which is incorporated herein by reference and made a part hereof.

The cyclic operation of a steam trap of the character under discussion which has a valve plate operating in a control chamber usually approaches maximum mean effectiveness when the period of closure on dry steam is about 30 seconds. As this steam trap has been constructed in the past, the period of the operating cycle can be established for one operating pressure, but it is subject to reduction when the operating pressure increases.

The cycle time of the steam trap is determined by the rate of pressure drop of steam which is contained in the control chamber of the trap, from the pressure required for closure which is approximately 80% of the inlet pressure to the pressure permitting reopening of the steam trap valve which is approximately 20% of the inlet pressure. As the initial pressure is increased, the steam temperature also increases and the increased temperature gradient with respect to the ambient temperature, together with a slight reduction in the latent heat of the contained steam, tends to reduce the cycle period. This feature will be apparent when it is realized that faster heat loss is associated with the higher temperature gradient if we assume a constant latent heat of the content of steam at all pressures. This effect is in fact the larger effect, but it is slightly aggravated by an actual drop in the latent heat of the contained steam at the higher pressures.

The improvements of the present invention will be better understood by reference to the steam trap of FIGURES 1 to 4 which will first be described and then explained more in detail.

A steam trap body 20 has a threaded inlet fitting 21 at one end which communicates through inlet connection 22 with an inlet opening 23 which extends suitably upwardly to communicate with an inlet opening 24 in valve seat 25. The valve seat has surrounding and in spaced relation to the inlet opening 24 an annular outlet flow passage 26 which communicates into an outlet opening 27 which in turn connects with an outlet passage 28 which discharges through connection 30 to threaded outlet fitting 31.

The valve seat 25 in the form of FIGURES 1 to 4 at the top has an initially flat surface which has between inlet opening 24 and outlet flow passage 26 a flat inlet valve seat 32 and which has at the space around the outlet flow passage 26 a flat outer valve seat 33.

The valve seat 25 is sealed at the bottom in a seat recess 34 by a gasket 35 positioned by an indexing pin 36 entering the bottom of the valve seat. The valve seat

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is sealed around the top by gasket 37 which is compressed by bonnet 38 threaded into the steam trap body at 40. The bonnet forms a control chamber 41 which receives a freely movable valve plate 42 which is capable of opening and closing, and, because of the fact that the inlet opening 24 is not central, will tend to fulcrum or tilt around the point 43.

The flat surface 44 of the valve 42 which engages the outer valve seat 33 or the surface of the seat 43 or both are provided with striations or roughening extending across the surface or radially, and this will assure continuous leakage to the outlet from the control chamber as described in detail in Patent 2,951,496 above referred to. When the valve plate is closed, a machine finish not below 4 microinches and preferably in the range between 4 and 80 microinches on at least one of the valve plate and valve seat will accomplish this result, but if desired actual scratches or scribe marks can be applied for example running radially of the valve plate or the valve seat or both at the cooperating surfaces. In other words, one of the valve plate and valve seat should be rougher than 4 microinches.

The parts of the steam trap other than the gaskets are preferably made of stainless steel or bronze, the valve plate and valve seat desirably being relatively hard. The gaskets will preferably be of Monel.

The question of whether the inlet valve seat fully closes or still permits slight leakage through the trap when the valve plate closes is merely a matter of preferred trap design, and the trap will operate using either form of construction.

The question of whether or not the seat is made replaceable is not critical in the present invention, although a replaceable seat is desirable in larger sizes.

The improvements of the present invention are directed particularly toward maintaining a normal constant cycle time over the entire operating pressure range of the steam trap, with the object of establishing operation with equal effectiveness at any operating pressure.

By referring to FIGURE 5, it will be noted that the higher temperature of the incoming steam at the center of the body in passage 23 and slightly offset in the direction away from the fulcrum 43 from the center of the body by inlet passage 24, will cause a relatively larger expansion of the material of the valve seat at the center than the lower temperature lower pressure steam at discharge conditions around the periphery of the seat.

The higher temperature is indicated as T_1 and the lower temperature is indicated as T_0 . The result of the excess temperature at the center is to cause the steam trap seat to expand in thickness as suggested in an exaggerated way by line 45 and to actually tend to form a convex surface as suggested in an exaggerated way by line 46.

The result of this temperature differential, if the valve itself does not undergo any change from a flat plate or disc form, is that the clearance of the valve from the seat around the outside is increased and thus releases steam from the control chamber 41 to the outlet at a faster rate, shortening the operating cycle. Normally in the case of a flat plate or disc valve the temperature differential will also contribute an effect which will tend to slightly aggravate the leakage from the control chamber, thus contributing further to shortening the cycle.

The present invention is concerned particularly with offsetting the factors which tend to decrease the cycle period with increased operating pressure and thus to develop an essentially constant cycle time over the entire operating range.

FIGURE 6 illustrates a comparatively thin valve plate or disc 42 which is modified over prior practice to include additional features. It has a flow passage 47 concentric of the valve plate on the undersurface communicating with the outlets 27 as in prior practice and separating the center portion from an outer ring 48. Unlike prior steam trap valve plates, near the middle of the top there is a cup

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recess 50 which cooperates with the flow passage 47 to make the cross sectional thickness of the valve plate more uniform, so that it could be manufactured either as a machined part or as a stamping as later explained.

With the valve plate 42 of FIGURE 6 in closed position, incoming steam entering the inlets 23 and 24 will contact the lower face of the valve plate at 51 near the middle at temperature T_1 corresponding to the temperature for saturation at the existing incoming pressure. The control chamber pressure at 41 and the corresponding saturated steam temperature for the control chamber combined with other significant temperatures cause the valve disc or plate 42 to compensate for the deflection of the seat.

It will be noted that in FIGURE 6 the inlet is offset from the center away from the fulcrum 43 to induce tilting action of the valve plate or disc 42. It should not however be considered that this is an essential feature, and FIGURE 7 shows a valve seat 25' and a valve plate 42' which cooperate and have the inlet port 24 located at the center of the valve seat and the valve plate so that the area 51 heated by the incoming steam is located at the center. In this case the inlet seat 32' and the outer seat 33' are in one plane and the inlet valve surface 52 on the bottom of the valve and the outlet valve surface 44' on the bottom of the valve are in one plane except as the valve tends to deflect slightly due to the temperature differential.

FIGURE 8 shows a slightly different form from that of FIGURE 7 in which the valve seat 25² has the inlet 24 centrally located but the inlet seat 32² is in a higher plane than the outer seat 33² and correspondingly the valve plate 42² is so formed that the inlet valve surface 52' cooperating with the inlet valve seat 32² is in a higher plane than the outlet valve surface 44² which cooperates with outer seat surface 33².

FIGURE 8a is a view similar to FIGURE 8 in which the valve seat 25³ has the inlet 24 centrally located but the inlet seat 32³ is in a lower plane than the outer seat 33³ and the valve plate 42³ is so formed that the inlet valve surface 52² cooperating with the inlet valve seat 32³ is in a lower plane than the outlet valve surface 44³ which cooperates with the outer seat surface 33³.

Heat loss due to the higher differential between the control chamber temperature and the ambient temperature together with a slight diminution of the total heat content with increasing pressure is partially compensated by the formation of the cup recess 50 on the upper side of the valve plate in cooperation with the ceiling 53 of the bonnet 38 which converges in a downwardly directed peak or point above the cup recess 50, causing condensate drops 54 to drain to the center peak and drip down into the valve cup recess 50 above the hot spot 51 above the inlet which is exposed to high pressure steam. In consequence the drops of condensate 54 are re-evaporated from the cup recess 50 and the pressure of the control chamber 41 is restored by an effect which offsets radiation loss. By comparison, near the radial outside edge of the valve plate or disc where cooler temperatures prevail in response to discharge temperature, there is little or no re-evaporation of any condensate which is present.

If during trap operation cooler condensate rather than steam reaches the valve plate surface 51 from inlet 24, re-evaporation of any condensate in the cup recess 50 will cease or will be diminished and the operating cycle will be shortened to bring about faster response of the trap to accomplish drainage of accumulated condensate. It should also be noted that the same cooling by incoming condensate which tends to retard or stop re-evaporation in the control chamber 41 also tends to cause contraction of the valve seat 25 and simultaneously tends to reduce the compensating response of the valve to maintain essential balance of the control function, as shown in FIGURE 9.

If the valve seat responds slightly faster than the valve plate or disc in the sequence of cooling, the effect will be to cause faster response by promotion of some initial leakage across the inlet seal between the inlet seat sur-

face 32 and the inlet valve surface 52, which tends to expedite cooling of the control chamber of the middle portion of the valve plate or disc. This effect is illustrated in FIGURE 9 where it will be noted that the valve plate or disc 42 is somewhat convex so that inlet valve element 52 has relatively considerable clearance from the inlet valve seat 32 as suggested at 53 and there is also increased clearance extending for a distance toward the outer edge of the valve element.

In some cases, as shown in FIGURE 10, the valve element 42⁴ may be a stamping or formed product, the accuracy of the contact relations being produced either by coining or by subsequent finishing of valve surfaces 52 and 44 as by grinding to meet the requirements, including the desired continuous leakage. Particularly for the higher pressures, a suitable erosion resistant material should be used as type 410 stainless steel which contains 11.5 to 13.5% chromium, 0.15% carbon maximum, 1% manganese maximum, 1% silicon maximum, balance iron, or type 440 A stainless steel which contains 16 to 18% chromium, 0.60 to 0.75% carbon, 1% manganese maximum, 1% silicon maximum and 0.75% molybdenum maximum, balance iron. The finishing operation, if any, should be performed after a heat treatment which will maintain accuracy and operation.

It will be evident that the designer may adjust the proportions to give him the character of compensating effect he desires, and the figures show examples of variations which may be employed.

The inlet sealing surface 52 of the valve should remain flat to preserve uniform sealing contact with the inlet sealing surface 32 of the seat. This indicates the importance of maintaining the thickness F of the middle portion of the valve relatively thin and relatively uniform. This thickness at the bottom of the cup recess 51 which is designated F in FIGURE 6 should be in the range between 1/5 and 1/20 of the outer diameter or dimension of the valve plate or disc which is designated D in FIGURE 6 and preferably about 1/10. There is also a great advantage in having a restricted cross section between the central cup recess and the relatively heavy outer ring or rim 48 of the valve plate or disc 42. This is accomplished by introducing the flow passage 47, which reduces the cross section at G and also at H corresponding to the top and the side of the flow passage 47 to a value in each case between 1/5 and 1/20 of the outside diameter D and preferably about 1/10. This thinness at G and H tends to retard heat flow radially. The outer ring or rim 48 as shown in FIGURE 6 has a width A and a thickness B which bear the relations to one another of between 1 to 2 and 2 to 1, with permissive beveling of the corner at 55.

The functioning of the valve plate can be understood in terms of reaction between the relatively rigid center cup section with thin walls but adequate thickness for rigidity which expands outwardly against the upper part of the section of the outer ring or rim 48 to impart a uniform radial twisting moment to the outer ring in relation to the existing temperature differentials. This moves the outer sealing edge 56 of FIGURE 9 down in compensating relation to the seat deformation with change in operating temperatures and pressures.

This will be better understood by a calculation for the purposes of example.

Referring to FIGURE 5, line 45 shows the exaggerated dimensional change which would take place in the valve seat where uniformly heated to 212° F. While line 46 shows in an exaggerated way the deflection due to introduction at the inlet near the center of 600 p.s.i. steam, it will be noted that the outer edge is restrained by the mounting in the steam trap body and regardless of inlet temperature the outer edge of the seat will not change in position significantly. Let us assume an inlet temperature of 486° F. for 600 p.s.i.a., and an outlet temperature of 212° F. Then the difference between the

inlet and outlet temperature is 274° F., assuming that the ambient temperature is 70° F.

ΔT for the temperature change=204° F., which will cause distortion.

The general equation for seat deflection is:

$$Y_s = \Sigma(Y)(C)\Delta T$$

where

Y_s is the metal deflection between line 45 and line 46 of FIGURE 5 which is a function of the outside diameter of the valve plate as explained

D is the outside diameter of the valve plate

C, the coefficient of thermal expansion in inches per inch per degrees F., is as follows:

For type 416 stainless steel for the body 5.6×10^{-6} .

For type 440 F stainless steel for the seat 6.2×10^{-6} .

For the Monel gasket 8.5×10^{-6} .

For compositions of the stainless steels, see Metals Handbook (1961) 409.

Combining the calculation for the body, the gasket and the seat:

$$\begin{aligned} Y_s &= (.125D)(5.6)(10^{-6})(204) + (.032D)(8.5)(10^{-6}) \\ &\quad (204) + (.338D)(6.2)(10^{-6})(204) \\ &= (204)(10^{-6})[(.125D)(5.6) + (.032D)(8.5) \\ &\quad + (.338D)(6.2)] \\ &= (625)(10^{-6})(D) \\ &= .000625D \end{aligned}$$

Let us now consider the compensating effect exerted by the valve plate or disc. For simplicity it is assumed that the temperature gradients are uniform. It is further assumed that the bending of the disc occurs at a specific point as illustrated in FIGURE 11, which may be X or Y.

Careful analysis will show that the question of which axis of bending is used in FIGURE 11 will impart very little error since the total deformation is very small and the length of the arc between two points will substantially equal the length of the chord. The computations have been checked experimentally and appear to agree with the experimental values.

Referring now to FIGURE 6, the middle of the valve plate or disc 42 is subjected from below when the valve is closed to a temperature of T_1 , while the top and outer edge which are in the control chamber 41 are subjected to temperature T_2 and the flow passage 47 which is in contact with the outlet is subjected to T_0 when the valve is closed. The length L_1 below the center of the valve plate or disc is subjected to a higher average temperature than the length A at the rim or ring of the valve plate or disc since T_1 is greater than T_2 , and T_2 is greater than T_0 . L_1 therefore expands a greater amount per unit length than does A which causes the valve disc or plate 42 as shown in FIGURE 11 to assume the convex form with the concave side downwardly directed as in FIGURE 11.

If we assume in FIGURE 11 that the middle portion L_1 of the valve plate or disc 42 expands uniformly over its length, then points X, Y and Z do not move significantly further due to thermal expansion. Point Z, however, which is above points X and Y at the top is exposed to a higher temperature T_2 than the temperature at points X and Y which are in contact with the outlet, and therefore point Z is expanded and translated to Z'. While surface films will be present, the actual temperatures at Y and Z are difficult to determine. It may be assumed that for average conditions point Z is subjected to temperature T_3 , which is equal to the average of T_2 and T_0 . The deflection ZZ' can be expressed as ΔL_1 since L_1 at the upper surface has expanded by this amount. In effect rotation about axis Y through angle θ has occurred. Since single θ is a small angle, the arc and the chord at EE' are substantially the same and equal the seat distortion Y_s . Likewise the arc and the chord at ZZ' are substantially the same.

Then where C is the coefficient of thermal expansion

and ΔT is the average temperature between points Y and Z the expansion

$$\Delta L_1 = \frac{L_1 \Delta T C}{2}$$

The thickness G is subjected to an average temperature which is the average of T_2 and T_0 . The point Y, however, is subjected to the lower temperature T_0 and the point Z is subjected to the higher temperature T_2 . The average temperature represents the temperature differential between points Y and Z.

Then

$$\Delta L_1 = \frac{(L_1)(C)}{2} \frac{(T_2 + T_0)}{2}$$

$$\Delta L_1 = \frac{(L_1)(C)(T_2 + T_0)}{4}$$

From the triangle YZZ'

$$\tan \theta = \frac{\Delta L_1}{G}$$

From the triangle JEE'

$$\tan \theta = \frac{Y_s}{\frac{D - L_1}{2}}$$

Equating these expressions:

$$\frac{2Y_s}{D - L_1} = \frac{\Delta L_1}{G}$$

$$G = \frac{(\Delta L_1)(D - L_1)}{2Y_s}$$

$$G = \frac{(L_1)(C)(T_2 + T_0)(D - L_1)}{8Y_s}$$

If L_1 is a function of D which is designated bD

$$G = \frac{(C)(T_2 + T_0)(b)(1 - b)(D^2)}{8Y_s}$$

This gives a general equation for the thickness G of the metal above the flow passage 47 in terms of valve plate or disc diameter D and required deflection Y_s .

If there is no flow passage 47 the valve plate or disc will still deflect, although the flow passage 47 has a tendency to concentrate the deflection toward the outer diameter. This leaves the entire portion of the valve plate or disc 42 relatively flat and better able to perform its sealing function at the inner sealing surface.

It will be evident that in the present invention, the valve plate or disc when subjected to thermal differentials of steam trap operation with change in initial pressure, will cause valve deformation to compensate for thermal dislocations transmitted to the valve seat. The increased radiation loss from the bonnet at the higher pressures and temperatures is compensated for by accumulating the condensate drops so as to cause them to engage the cup recess and be confined at that location opposite the hottest portion of the valve plate or disc immediately over the inlet with resulting restoration of the heat content of the steam in the chamber and corresponding extension of the cycle time of closure.

These two effects, although separately discussed, are combined in one valve plate or disc configuration.

An equally important feature of the invention is that it accomplishes the foregoing with a simultaneous increase in sensitivity of response as condensate reaches the trap and in this case cooling just as effectively shortens the cycle and increases the efficiency of performance. A further benefit observed in performance is that the steam trap has the ability to bring the steam line to steam temperature faster and maintain this temperature more uniformly.

The net result, therefore, is to enhance the trap func-

tioning not only from the standpoint of closure on steam for a longer period but also to discharge accumulated condensate more rapidly.

In this way it contributes not only to extend the pressure range for essentially constant closure cycle performance, but it also improves the general performance by an appreciable margin over that which would be realized for a specific pressure condition.

Aside from the features which are discussed herein, the device of the invention operates in the manner described in Yarnall and Kinderman U.S. Patent No. 2,951,496 above referred to.

In view of our invention and disclosure, variations and modifications to meet individual whim or particular need will doubtless become evident to others skilled in the art, to obtain all or part of the benefits of our invention without copying the structure shown, and we, therefore, claim all such insofar as they fall within the reasonable spirit and scope of our claims.

Having thus described our invention what we claim as new and desire to secure by Letters Patent is:

1. A steam trap having a body forming a control chamber, walls forming an inlet passage adjacent the middle of one side of said control chamber, walls forming an outlet passage to one side of the control chamber at a location more remote from said middle than said inlet passage, a valve plate freely movable in said control chamber, said valve plate having a seated position at which said inlet passage is at least nearly closed and the portion of the control chamber on the side of the valve plate remote from the inlet and outlet passages is nearly closed off from the outlet, and having an open position in which the inlet passage and the outlet passage communicate through the control chamber and with the portion of the control chamber on the side of the valve plate remote from the inlet and outlet passages, said valve plate having a relatively reduced cross section toward the middle thereof forming a depression on the top side of said plate and having a relatively thicker cross section outwardly of the middle, a first valve seat means cooperating with the valve plate in seated position of the valve plate to at least greatly reduce flow from the inlet passage to the control chamber and to the outlet passage, and a second valve seat means cooperating with the valve plate in seated position of the valve plate to restrict flow from the portion of the control chamber remote from the inlet and outlet passages to the outlet passage, there being continuous leakage means between the portion of the control chamber remote from the inlet and outlet passages to the outlet even when the valve is seated.

2. A steam trap of claim 1, in which the valve plate is imperforate.

3. A steam trap of claim 1, in which the valve plate is a disc, the thickness at the center being between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc.

4. A steam trap of claim 1, in which the valve plate is a disc, the thickness at the center being approximately $\frac{1}{10}$ the outside diameter of the disc and the valve plate being imperforate.

5. A steam trap of claim 1, in which the valve plate is a disc and has a relatively thick outer ring portion, and a groove on the side of the valve plate adjoining said inlet and outlet, which groove is located toward the middle of the disc from the ring portion, said groove forming a restricted cross section and said restricted cross section having a thickness which is between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc.

6. A steam trap of claim 5, in which the center of the disc has a thickness of between $\frac{1}{8}$ and $\frac{1}{10}$ the outside diameter of the disc.

7. A steam trap of claim 1, which has a cup-like depression near the middle of the valve plate.

8. A steam trap of claim 1, which has a cup-like depression near the middle of the valve plate, the valve plate thickness opposite the cup-like depression being

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relatively thin compared to the valve plate thickness adjacent the outer edge of the valve plate.

9. A steam trap having a body forming a control chamber which has a bottom, walls forming an inlet passage adjacent the middle of the bottom of the control chamber, walls forming an outlet passage to said bottom of the control chamber at a location more remote from said middle than said inlet passage, a valve plate freely movable in said control chamber, said valve plate having a seated position at which said inlet passage is at least nearly closed and the portion of the control chamber above the valve plate is nearly closed off from the outlet, and having an open position in which the inlet passage and the outlet passage communicate through the control chamber and with the portion of the control chamber above the valve plate, the upper side of the valve plate having a cup adjacent the middle of the valve plate and the wall of the valve plate beneath the bottom of the cup being relatively thinner than portions of the valve plate located outwardly of said cup, a first valve seat means cooperating with the valve plate in seated position of the valve plate to at least greatly reduce flow from the inlet passage to the control chamber and to the outlet passage and a second valve seat means cooperating with the valve plate in seated position of the valve plate to restrict flow from the portion of the control chamber above the valve plate to the outlet passage, there being continuous leakage means between the portion of the control chamber above the valve plate and the outlet even when the valve is seated.

10. A steam trap of claim 9, in which the control chamber has a downwardly converging portion above the cup adapted to discharge condensate into the cup.

11. A steam trap of claim 9, in which the valve plate is of disc form, having a ring portion at the outside and a generally angular groove in the underside of the valve plate inwardly of the ring portion but outward of the cup.

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12. A steam trap of claim 9, in which the valve plate is of disc form, the valve plate having a ring portion at the outside of the disc and having a restricted portion integral with and extending annularly between the ring portion and the cup which in cross section has a thickness less than that of the ring portion in cross section.

13. A steam trap of claim 12, in which the restricted portion has a thickness of between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc.

14. A steam trap of claim 13, in which the bottom of the cup has a thickness of between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc.

15. A steam trap of claim 12, in which the restricted portion has a thickness of between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc and the bottom of the cup has a thickness of between $\frac{1}{8}$ and $\frac{1}{20}$ the outside diameter of the disc.

16. A steam trap of claim 12, in which the restricted portion and also the bottom of the cup have a thickness of approximately $\frac{1}{40}$ the outside diameter of the disc.

17. A steam trap of claim 12, in which the outer ring has a width which is between $\frac{1}{2}$ and twice the depth.

18. A steam trap of claim 9, in which the valve plate is imperforate.

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