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[45] **July 17, 1973**

[54] **RINGLESS PISTON**

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[22] Filed: **June 9, 1971**

[21] Appl. No.: **151,268**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 800,956, Feb. 20, 1969, abandoned.

[52] **U.S. Cl.** **92/192,** 92/243, 92/246

[51] **Int. Cl.** **F16j 1/04**

[58] **Field of Search** 92/172, 240, 192, 92/243, 246

[56] **References Cited**

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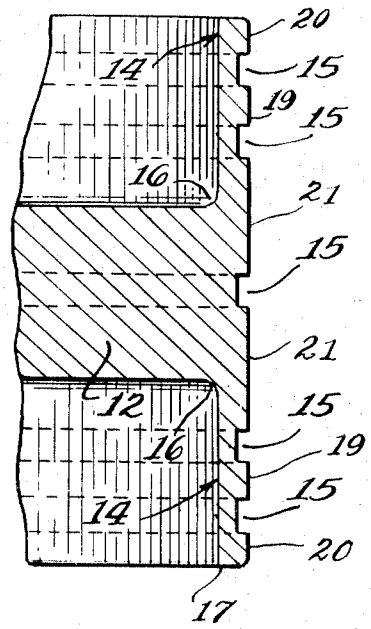
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[57] **ABSTRACT**

A ringless piston in which a sealing flange at one or both ends of the piston has a thickness such that it expands under working pressure and in accordance with bursting tendencies of the flange. The piston and flange in the absence of pressure within the cylinder have a normal running clearance with the cylinder wall.

6 Claims, 11 Drawing Figures



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SHEET 1 OF 2

Fig 1.

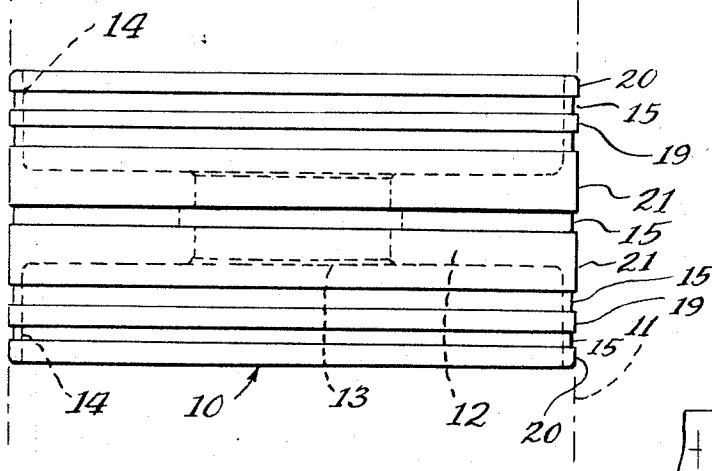
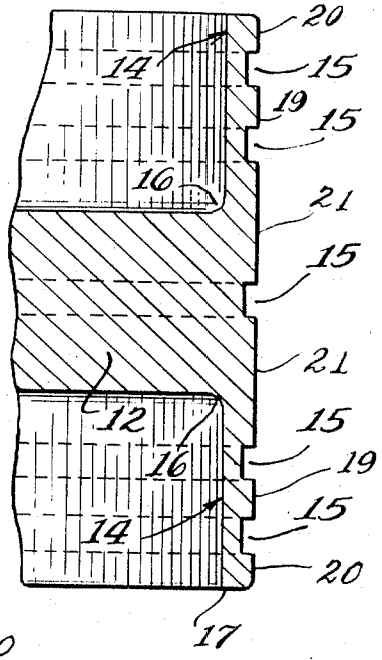


Fig. 2.



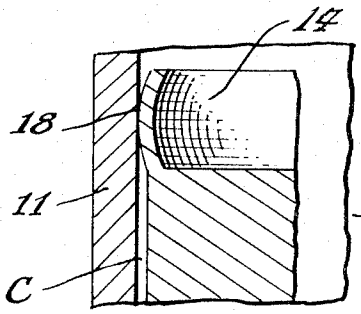


Fig. 3.

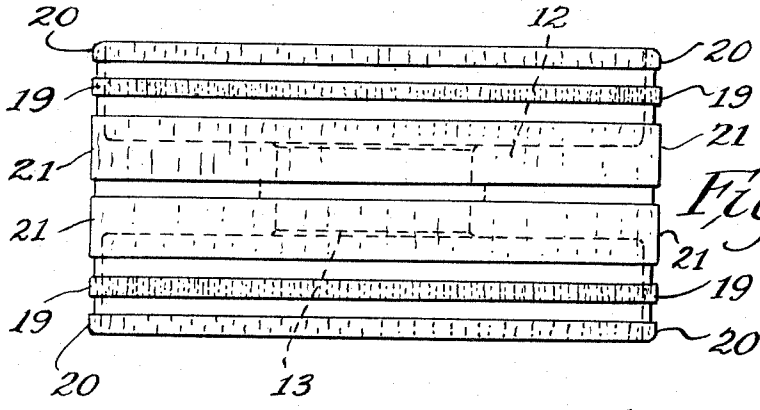


Fig. 4.

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Fig. 5.

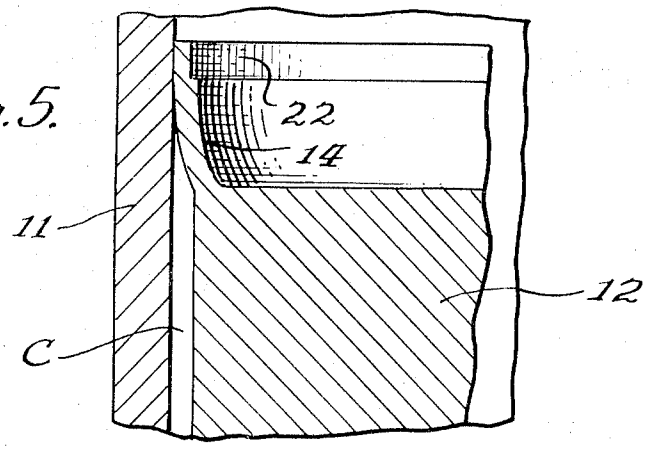


Fig. 6.

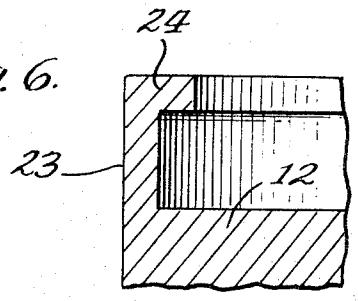


Fig. 8.

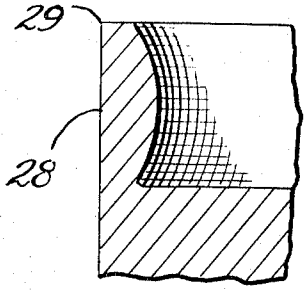


Fig. 7.

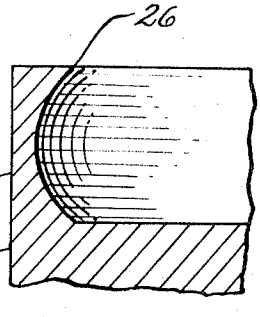
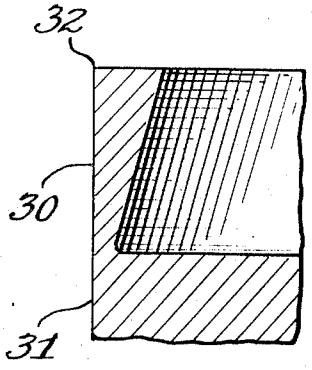


Fig. 9.



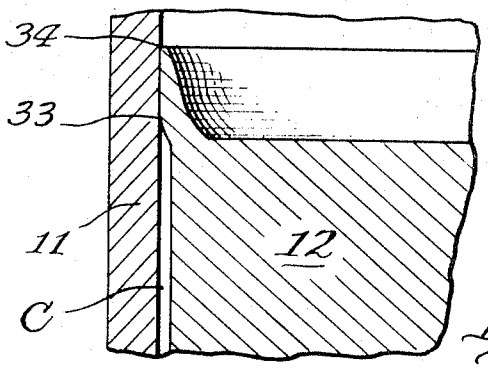


Fig. 10.

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RINGLESS PISTON

This application is a continuation-in-part of my co-pending application Ser. No. 800,956, filed Feb. 20, 1969, and now abandoned.

The present invention is directed to new and useful improvements in piston and cylinder assemblies of the type used as pumps, engines and other apparatus wherein pressure is developed between the piston and opposed end wall of the cylinder during reciprocation of the piston.

For years it has been more or less customary to use piston rings of various types on the outer walls of the pistons in pressure developing piston and cylinder assemblies. In such cases the piston has a running or working clearance with the opposed wall of the cylinder which may be on the order of one thousandth of an inch for every inch of piston diameter. In such cases, separate piston rings are carried in grooves on the outer periphery of the piston and are fitted so as to provide some sealing contact with the opposed wall of the cylinder during reciprocation of the piston. In assemblies of this type, the rings wear out and must be replaced and the use of pistons of this type necessitates assembly and machining time to provide the rings for the piston. Various proposals have been made from time to time in the past to form ringless pistons. In some of these cases, pistons are formed with skirt-like flanges on one or both ends of the piston and pressure rings are inserted within the flanges to more or less bend the flanges outwardly so that the extreme end of the flange makes more or less of a line contact with the wall of the cylinder. In other cases, proposals have been made to eliminate the use of rings by forming the piston with a diameter which is at least as great as the inside diameter of the cylinder so as to provide a form of press fit between the piston and cylinder. For some reason or another, these proposals have not gone into widespread use and the ring-type piston construction is still the most widely used construction as of the present time, possibly because of the wear that occurs when a line sealing contact is established by a piston flange and because of the difficulty holding manufacturing tolerances such that a piston can be press-fitted within the cylinder and at the same time provide an operable machine without large frictional losses or excessive costs.

With the foregoing in mind, the primary purposes of the present invention are to create a ringless piston and cylinder construction which, when manufactured, has the normal operating or running clearances or tolerances between the piston and the cylinder and which, when subjected to the normal operating pressure developed within the piston and cylinder, causes an expansion of specially-formed wall surfaces of the piston outwardly into a sealing contact with the wall of the cylinder to provide a relatively large area of wall-to-wall contact while at the same time holding frictional losses developed during reciprocation of the piston to a minimum.

These and other objects of the invention will become more apparent in the course of the ensuing specification and claims when taken with the accompanying drawings, in which:

FIG. 1 is a side elevation in section of a typical piston formed in accordance with the principles of the present invention;

FIG. 2 is an enlarged detail view of a portion of the piston illustrated in FIG. 1;

FIG. 3 is a diagrammatic sectional view illustrating, in schematic form, the manner in which the piston of the present invention operates;

FIG. 4 is a schematic view of a piston formed in accordance with the invention and illustrating wear marks on the piston;

FIG. 5 is a sectional illustration of a modified form of the invention;

FIG. 6 is a sectional view of another modification of the invention;

FIG. 7 is a sectional view of another modified form of the invention;

FIG. 8 is a sectional view of another modified form of the invention;

FIG. 9 is a sectional view of another modified form of the invention;

FIG. 10 is a sectional view of another modified form of the invention; and

FIG. 11 is a sectional view of another modified form of the invention.

Like elements are designated by like characters throughout the specification and drawings.

With specific reference now to the drawings, and in the first instance to FIG. 1, the numeral 10 generally represents a piston formed in accordance with the present invention. The piston is mounted for reciprocation in a cylinder which is diagrammatically represented by the broken lines 11 in FIG. 1. Cylinder 11 may take any one of a number of known forms. The piston and cylinder assembly thus defined may be used in a pump, an hydraulic or air cylinder, a shock absorber or accumulator.

The main portion 12 of the piston may, if used as a part of an hydraulic ram assembly, be provided with a central opening 13 for reception of an operating piston rod or ram in a manner known to the art. On the other hand, the central opening 13 may be omitted and suitable facilities may be provided with the main body portion 12 to enable connection to a connecting rod. Since such piston actuating members and connections are known to the art, and since they form no part of the present invention, they are not illustrated in the drawings.

In accordance with the invention, one or both ends of the piston body is formed with a generally cylindrical skirt-like flange 14. As illustrated in the drawings, a flange 14 is provided on opposite sides of the main body of the piston. In situations where only one side of the piston is subjected to pressure, only one flange need be provided, and on the side opposed to the pressurized space within the cylinder.

As shown in FIG. 2, the skirt-like flange 14 has a length several times its width. The length should be approximately equal to the piston external diamete divided by four. The exterior diameter of the skirt-like flange or flanges 14 is machined or formed so as to be the same as the exterior diameter of the main body portion of the piston. Both diameters are such as to provide a normal running clearance with the opposed cylinder wall 11. This clearance may be 0.001 inch for each inch of piston diameter. The exterior surface of the flanges 14 may carry relatively shallow oil entrapping grooves 15 which are spaced axially of the piston. Shallow oil entrapping groove 15 may also be provided medially of the piston. These oil entrapping grooves

should have depths considerably less than one-half of the radial width of the flanges and preferably no more than one-fifth of the overall wall thickness of the flange, when the overall wall thickness is 0.025 inch or less. When the wall thickness is 0.050 inch or more, the groove depth should not exceed 0.010 inch.

It is within the concept of this invention to form pistons which do not have oil entrapping grooves. Such a construction is illustrated in FIG. **11.** The piston and opposed skirt-like flanges are formed in accordance with the criteria set forth herein. With double-acting cylinders, the action of flanges on opposite sides of the piston naturally tends to work lubricant back and forth within the cylinder without use of oil entrapping grooves.

It is also within the concept of this invention to use an oil wiping ring of Teflon or the like at one portion of the piston as, for example, in the medial groove **15.** An O-ring of this type does not function as a normal piston ring but functions merely to confine flow and positioning of lubricating oil. In some cases this is particularly advantageous, as in the case of double-acting pistons of hydraulic cylinders when the pressurized condition is changed from one end of the cylinder to the other. As the changeover is being made, a ring of this type will tend to prevent oil "blow-by" before the pressurized flange is expanded.

The flanges **14** are formed so as to expand in more or less barrel-like fashion as illustrated in the exaggerated showing of FIG. **3.** This is done by proper selection of radial widths of the flanges. For example, the wall thickness of the flange at one end or both ends of the piston may be calculated for any given piston and cylinder construction. For example, for a cylinder with a 4.250 inch bore and a reciprocable piston with a 4.245 inch outside diameter, a typical running clearance C between the piston and cylinder wall of 0.005 inch exists. The cross-sectional clearance area between such a piston and cylinder may be calculated as 0.03336 square inch. The pressure within the pressurized space of the cylinder may then be calculated or approximated. For example, if the pressure is supplied by an hydraulic pump supplying a volume of oil at 40 gallons per minute, the pressure in the pressurized space may be calculated by the formula: Gallons per minute = 24.12 × the leakage area between the piston and cylinder in square inches × the square root of the final pressure in the pressurized space. With the assumed figures, the formula then becomes: 40 = 24.12 × 0.03336 × the square root of the final pressure.

From this, final pressure may be calculated as 2471 p.s.i. in the pressurized space of the cylinder. The actual working pressure within the cylinder may be below this calculated pressure and a pressure considerably below the calculated and actual working pressure may then be utilized to determine a flange wall thickness which will cause the flange to expand and seal against the opposed wall of the cylinder. For example, a pressure of 800 p.s.i. may be assumed to be adequate for insuring a sealing relation under the calculated pressure condition of 2471 p.s.i. The tensile strength of the piston material must be known before the wall thickness can be calculated. The tensile strength of most, if not all, piston materials is readily available from engineering handbooks or data supplied by manufacturers of piston material. As a specific example, if the piston material is made from SAE Class 30 cast iron, the tensile strength of the material is 30,000 p.s.i.

Then Barlow's formula for the burst pressure of a cylinder under internal pressure may be used to calculate the necessary wall thickness of the flange or flanges on the piston to insure sealing of the flange against the cylinder wall under the assumed pressure condition. Barlow's formula is: pressure p.s.i. = 2 × wall thickness in inches × the tensile strength of the material divided by the diameter of the cylinder.

By substituting values then, 800 p.s.i. = 2 × the wall thickness × 30,000 (tensile strength) divided by 4.245 (diameter of the cylinder). The wall thickness is then equal to 0.0566 inch. This is the wall thickness of a cylinder of the assumed diameter which would eventually burst under the pressure of 800 p.s.i. if the flange were not in some way supported to prevent bursting. The cylinder wall, however, supports the flange and prevents bursting of the flange, and the flange is simply expanded out to fill the 0.005 inch clearance space and seal against the opposed wall of the cylinder. This occurs at pressures of 800 p.s.i. and greater pressures.

The wall thickness is also such that the expansive forces caused by the pressure are nonetheless within the elastic limits of the material used in the piston so that the flange expands when the pressure is applied and then recedes towards its original diameter when the pressure forces are removed, thus again providing a normal running clearance between the flange and the cylinder wall.

It should be understood that the foregoing is intended only as a specific example of the manner in which the wall thickness of the flange can be calculated. The foregoing calculations and assumptions may be used for any type of piston material, and any type of pressurized piston and cylinder assembly in which the working pressures within the pressurized space of the cylinder can either be assumed or calculated.

While the flange expansion is formulated as indicated, this may be accompanied by some incidental thermal expansion of the flanges due to increased temperatures of the flanges. The cylinder may also expand with temperature. Effects of temperature are to be neglected when forming the flanges as set forth herein. It is not intended that the invention be used with high operating temperatures as are found in the cylinders of internal combustion engines during the time that a fuel mixture explodes within the cylinder. Temperatures within the cylinders using the invention may vary considerably and still be well below that found in the cylinders of internal combustion engines.

While the wall thickness may thus be calculated, it is usually advantageous to utilize a somewhat lesser wall thickness. In this regard, the presence of oil entrapping grooves lessens the effective wall thickness to some extent. The wall thickness may, for example, with the assumed values of piston diameter and cylinder diameter, be reduced to 0.026 inch (without a reduction by the grooves).

The interior corners of the flanges may be given a slight radius as indicated at **16** and **17.**

When pistons are formed, as disclosed herein, and with the skirt-like flange opposed to the pressurized space within the cylinder, the skirt-like flange bulges outwardly as schematically indicated in FIG. **3.** In FIG. **3** the amount of expansion and the clearance space between the piston and cylinder is exaggerated for pur-

poses of illustration. It may be noted that the skirt tends to assume a more or less barrel shape with the heaviest contact occurring at areas intermediate the ends of the skirt as indicated at **18.** Wear masks on a piston formed with the chatacteristic skirt-like flanges will appear as indicated schematically in FIG. **4.** The heaviest wear marks will appear in the regions designated **19** which are areas intermediate the ends of the skirt-like flanges and lighter scratch marks will appear at the ends of the skirt-like flange as indicated at **20.** Relatively light scratch or wear marks will appear on the piston intermediate of the skirt-like flanges as indicated at **21.**

It is important that the skirt-like flanges have shapes such that they may expand in more or less barrel-like fashion. From a machining cost standpoint, a flange of essentially uniform wall thickness may be most desirable. The flanges may, however, have reduced thicknesses at their outer end portions to provide a stepped configuration. Such a reduced thickness is indicated at **22** in FIG. **5.** FIG. **5** again shows an exaggerated clearance space between the piston and cylinder and an exaggerated expansion of the skirt-like flange to illustrate the principle involved. When the outermost portion of the skirt-like flange is given a stepped configuration as indicated in FIG. **5,** the skirt-like flange has a relatively heavier sealing contact at the outer areas of the flanges than is indicated at **20** in FIG. **4** without the stepped configuration. The stepped configuration thus provides a relatively larger area of sealing contact with the opposed cylinder wall.

The skirt-like flanges may have other shapes. FIGS. **6, 7, 8** and **9** illustrate other shapes which may produce the characteristic sealing engagement. In FIG. **6,** for example, the skirt-like flange **23** has the same calculable thickness as desribed. The flange, however, has an inturned lip **24** at the outer end thereof which tends to strengthen the outer end portion of the skirt-like flange and produce a more pronounced outward expanding effect in the medial portion of the skirt-like flange to produce sealing action in the medial area of the flange.

In FIG. **7** the inner surface of the skirt-like flange **25** is rounded so as to produce regions of greatest thickness at the outer end of the flange as at **26** and at the inner end of the flange as at **27.** The effect again, as in the structure of FIG. **7,** is to increase the outward expanding tendency in the medial portion of the skirt-like flange.

In FIG. **8** the inner surface of the skirt-like flange **28** is given a curvature opposite to that illustrated in FIG. **6.** This tends to reduce the outward expansion in the medial portion of the flange and increase the outward expansion at the outer end of the skirt-like flange as at **29.**

In FIG. **9** the skirt-like flange **30** has an expanding width from the inner portion **31** thereof to the outer end **32** thereof. Again, the configuration in FIG. **9** is such as to produce the greatest amount of expansion in the medial region of the skirt-like flange with lesser amounts of expansion and sealing contact at the outer end of the flange.

In FIG. **10** the skirt-like flange **33** is tapered so that the width thereof diminishes toward the outer end **34** of the skirt-like flange.

It should be understood that the variant forms of skirt-like flanges illustrated in FIGS. **6–10** are exaggerated in terms of variant thicknesses somewhat. These figures are intended to demonstrate that some variation from a uniform or substantially uniform wall thickness is possible in the skirt-like flanges while at the same time producing the outer bulging effect and the area-to-area sealing contact in medial portions of the skirt-like flanges. For example, if the degree of taper of the skirt-like flange **33** in FIG **10** is great enough, outward bulging of the complete flange does not take place, although some outward deflection of the flange may take place so that the outermost corner of the skirt-like flange makes essentially a line contact with the cylinder wall as opposed to the outward bulging effect and relatively large area of contact produced through the principles of the present invention.

In all forms of the invention, the wall thickness of the skirt-like flanges is calculated so as to expand outwardly and make sealing contact with the cylinder wall at pressures below the normal working pressures. This provides good sealing of the piston and cylinder without the need for piston rings. At the same time, the pistons are formed so that they have a normal running clearance with the cylinder wall in the unpressurized condition. This facilitates assembly and reduces costs. The outward bulging of the flanges, as defined herein, is to be distinguished from a simple bending movement of a flange which may occur by reason of a more or less pivotal movement about the main juncture of the flange and main body of the piston, in the case of flanges not calculated as herein described.

I claim:

1. A piston and cylinder assembly of the type wherein a piston is reciprocally mounted within a cylinder to produce variant pressure conditions on opposite sides of the piston during reciprocation thereof, the improvement comprising a cylinder and a piston reciprocally mounted therein, to develop a pressurized space in said cylinder, said piston having an outside wall surface providing a normal running clearance with the opposed wall of said cylinder and with a clearance space between the piston wall surface and the opposed cylinder wall, said piston having an extended skirt-like flange on the end thereof opposed to a closed end of the cylinder, said flange having a continuous circular wall, the normal outside diameter of said skirt-like flange being such as to provide a normal running clearance with a space between the wall of the flange and the opposed wall of the cylinder when the piston is positioned within the cylinder in an unpressurized condition, said skirt-like flange having a radial width and an axial length such, with relation to the outside diameter of said piston and flange, with relation to the particular material of which the piston and skirt-like flange are made, and with relation to the normal operating pressure developed within the cylinder against the end of the piston carrying said skirt-like flange, that said skirt-like flange is expanded outwardly because of the hoop-like bursting stresses developed therein during normal operating pressures such that the outer wall of said skirt-like flange is forced into wiping and sealing contact with the opposed wall of the cylinder during reciprocation of said piston, said piston and cylinder being further characterized by the absence of sealing rings or expansion rings, the wall thickness of said flange being such that the stresses developed on said flange are within the elastic limits of the material of the flange whereby the flange resiliently recedes toward its original condition after removal of said pressurized condition.

2. The structure of claim 1 wherein the thickness and length of the flanges are such, with relation to the specific material of the piston and flange and with respect to the normal operating pressures of the assembly, that the wiping and sealing pressure exerted by said skirt-like flange on the opposed wall of said cylinder is greatest at the areas intermediate the ends of said flange.

3. The structure of claim 1 wherein the wall thickness of said flange is calculated to be that wall thickness which will expand and burst at a pressure below the normal operating pressure of the piston and cylinder assembly unless the flange is supported.

4. The structure of claim 1 wherein the length of the flange is approximately equal to one-quarter of the piston external diameter.

5. The structure of claim 1 wherein said piston has a similarly formed flange, as previously defined, on each end of said piston.

6. The structure of claim 5 wherein each said flange has at least one oil entrapping groove formed in the exterior wall thereof.

* * * * *