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(54) ANNULAR DUAL PISTON BRAKE ARRANGEMENT AND COOLING CIRCUIT THEREFOR

(71) We, CATERPILLAR TRACTOR CO., a corporation organised and existing under the Laws of the State of California, United States of America, of 100 N. E. Adams Street, Peoria, Illinois 61629, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

Fluid operated brake arrangements for large material hauling vehicles and the like are being continually improved in an attempt to achieve safe and effective braking under severe operating conditions. Representatives of these braking arrangements are described in United States Patent Specifications Nos. 2,928,504; 3,547,233; 3,547,234; 3,613,839; and 3,770,085.

However, as far as is known, such brake arrangements have not heretofore effectively integrated service braking with emergency braking and/or parking braking in a single annularly compact and economical package. For example, it is substantially imperative that these brake arrangements be small in size, while having a relatively high multi-purpose capacity, in order for them to fit within a wheel rim where space is at a premium. An additional desirable characteristic for such annular brake arrangements is the ability to install or remove them easily as a modular package.

While the aforementioned Patent Specifications disclose the cooperating brake combination of a spring retracted, fluid engaged service brake piston with a normally fluid retracted, spring engaged parking brake piston, they are either limited in capacity by their spot type, caliper disc brake construction requiring a multiplicity

of the units to be arranged around the member to be braked, or otherwise have extended axial or radial dimensions that are generally unacceptable for use in a relatively limited space.

Some of the other deficiencies of prior art brake arrangements include the lack of cooling oil supply to the energy-absorbing surfaces, and a slower than desired rate of response in fully applying the brakes because of the extended travel distance of the actuating pistons associated therewith.

Because space is not available radially outwardly of these brake arrangements, certain sizing and constructional limitations are imposed on its members in order to permit a sufficient flow of cooling fluid therethrough and particularly the effective egress thereof. For example, the maximum radial dimensions of the service brake piston and its associated actuation chamber and also the maximum radius of the pitch circle for the compression spring members used therewith is limited in order to provide a path for such fluid egress. Consequently, a fewer number of compression springs are disposable equally around the pitch circle so that these radial limitations ultimately tend to reduce the maximum capacity of the brake arrangement for a particular maximum external diameter, and undesirably add to the weight and cost of the package.

According to the invention, an annular dual piston brake arrangement for selective holding engagement of a rotatable member of a drive train comprises a support structure; an annular service brake piston axially reciprocably and nonrotatably mounted within the support structure coaxially with the rotatable member; fluid actuating means for biasing the service brake piston toward a brake holding

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position in use with respect to such rotatable member; an annular secondary brake piston axially reciprocally and nonrotatably mounted within the support structure in axially telescoping, substantially radially aligned relation with respect to the service brake piston; fluid retracting means for holding the secondary brake piston in a disengaged condition axially spaced from the service brake piston; and spring engagement means for biasing the secondary brake piston directly axially against the service brake piston for subsequent brake holding engagement of the rotatable member in the absence of a predetermined pressure in the fluid retraction means.

This new construction may provide an annular vehicle brake arrangement engagable for individual service braking as well as for emergency braking and/or parking braking in a relatively safe and responsive manner, and the brake arrangement may be both radially and axially compact so that it can be disposed annularly within a wheel rim of a vehicle.

Preferably, the brake arrangements includes seal means disposed between the support structure and the rotatable member to thereby define an internal chamber therebetween, and fluid circulating means for delivering fluid in a fluid path radially outwardly through the internal chamber in order to cool the annular brake arrangement around its full periphery, the fluid also serving to aid in retracting the service brake piston. The service brake piston may then include fluid passage means therethrough which is disposed in communication with the fluid path in order to provide an egress path for the fluid upon substantially fully peripherally cooling the annular brake arrangement.

The cooling circuit then provides improved axial egress of the cooling fluid in order to avoid diametrical restrictions on certain members of the brake arrangement, for example to enable the brake arrangement to have a relatively large diameter service braking piston actuation chamber and also a relatively large diameter pitch circle for accommodating a greater number of compression springs for retracting the service brake piston; to effect a weight and cost savings for a predetermined braking capacity.

Two examples of brake arrangements constructed in accordance with the invention are illustrated in the accompanying drawings, in which:—

Figure 1 is a fragmentary section of one example taken through the axis of a vehicle axle housing and showing the lower half thereof including a portion of a wheel rim arrangement associated therewith; and

Figure 2 is a similar section through the second example.

Description of the first example

With reference to Fig. 1, the annular oil-cooled dual piston brake arrangement 10 of the present invention is axially mounted outboard of a radially extending flange 12 of a conically tapered axle housing 14 such as is found on a relatively large earthmoving truck. A vehicle wheel hub 22 is rotatably supported on the axle housing to the outside, or to the left when viewing the drawing, of the annular brake arrangement by an anti-friction bearing 24. The wheel hub has an axially inwardly extending flange 26 with an external spline 27 thereon, and a radially outwardly extending flange 28 to which is secured a wheel rim arrangement generally identified by the reference numeral 34 by a plurality of circumferentially spaced lug bolts 30 and associated nuts 32. A tire 36 is sealingly mounted on the wheel rim arrangement and upon screwthreadably removing the nuts, the tire and wheel rim arrangement may be pulled outwardly as a unit from the lug bolts and the axle housing for servicing purposes in the usual manner.

The annular dual piston brake arrangement 10 includes an inner end plate 38 removably secured to the flange 12 of the axle housing 14 by a plurality of threaded studs 42 anchored to the end plate and extended axially inwardly through the flange for screwthreaded receipt of a plurality of retaining nuts 46 thereon. The mounting flange 12 has an inlet passage 48 therethrough, and the end plate has an outlet passage 50 therethrough which are respectively in communication with an inlet conduit 52 and an outlet conduit 54 for the purposes of normally continuously circulating a cooling fluid through the brake arrangement. Further, the end plate has an outer counterbore 66 and an inner counterbore 68 disposed in axially outwardly piston-receiving relation, and a plurality of circumferentially spaced spring-receiving cavities 70 opening on the inner counterbore.

The brake arrangement 10 also includes a centrally disposed cylindrical shell 72 having a plurality of internal teeth 74. The shell extends in spanning relation between the inner end plate 38 and an outer end plate 78, and in order to secure the shell and outer end plate to the inner end plate a plurality of elongated bolts 82 are inserted axially inwardly therethrough and are adapted to screwthreadably receive a corresponding plurality of retaining nuts 84 thereon. Thus, the outer end plate provides a relatively fixed inner thrust surface 86 for clamping engagement of the brake arrangement.

The outer end plate 78 has a circular aperture 88 defined centrally therethrough so that it may be disposed in relatively radially closely spaced relation to the external spline 27 of the wheel hub 22. However, a cylindrical connecting drum 90 still has enough clearance to extend axially outwardly through this aperture so that an internal spline 92 thereon is engaged with the external spline. Consequently, the connecting drum rotates with vehicle movement and is provided with a plurality of external teeth 94 on which are keyed a plurality of brake discs 96. These discs are interleaved with a plurality of brake plates 98 which are similarly keyed to the internal teeth 74 of the shell 72. A pair of damping plates 100 are also arranged at the opposite ends of the interleaved discs and plates in keyed engagement with the internal teeth 74 for the general purpose of minimizing braking squall in certain modes of operation of the braking arrangement 10.

Since the dual piston brake arrangement 10 is fully peripherally cooled by the circulation of fluid from the inlet conduit 52 to an internal chamber 102, and thence radially outwardly through a plurality of passages 104 in the connecting drum 90 and the interleaved brake discs 96 and brake plates 98, an annular internal seal arrangement 105 and an annular external seal arrangement 106 are required. The internal seal arrangement is generally disposed between an adapter member 108 secured to the axle housing 14 and the rotatable connecting drum 90, while the external seal arrangement is mounted between the outer end plate 78 and another adapter member 110 secured to the wheel hub 22. Such seal arrangements are generally disclosed in the aforementioned U.S. Patent Specification No. 3,547,234.

The brake arrangement 10 also includes a parking brake housing 112 which is secured tightly against the inner end plate 38 and thus to the axle housing 14 by a plurality of retaining bolts 116 inserted therethrough for screwthreaded engagement with the end plate. The parking brake housing has an inwardly facing annular step 118 formed radially outwardly thereof with a piston retaining and aligning dowel 120 rigidly and axially inwardly extending therefrom.

More particularly, the actuation of the dual piston brake arrangement 10 of the present invention is accomplished by axial reciprocating movement of an annular service brake piston 122 and an annular secondary or parking brake piston 124 telescopically nested radially within it. The parking brake piston is normally disengaged, but is selectively or automatically actuated for either parking or emergency braking purposes as will hereinafter be described. The parking brake piston is axially

reciprocally and nonrotatably secured to the parking brake housing 112 by the guiding and aligning dowel 120, and includes a plurality of spring-receiving cavities 126 opening inwardly thereof and a plurality of guide-receiving bores 128. The parking brake piston also includes an axially outwardly facing counterbore 130 which is disposed in axial slidable cooperating relation with the annular step 118 of the parking brake housing 112 to define a fluid retractor or sealed parking brake actuation chamber 132 therebetween. Fluid pressure is delivered to this chamber from a brake control system 134 through an interconnecting conduit 135 and an internal passage 136 to normally move the parking brake piston 124 axially inwardly, or to the right when viewing the drawing, against the opposition of a plurality of relatively high capacity compression springs 138 seated intermediate the cavities 70 and 126.

As mentioned generally above, the service brake piston 122 is arranged in telescoping, compact and radially aligned relation annularly outwardly of the parking brake piston 124. More particularly, it includes an axially inward open counterbore 140 adapted to closely receive the parking brake piston reciprocally therein. The service brake piston includes an inwardly facing annular step 142 which is disposed in axial slidable cooperating relation with the counterbores 66 and 68 of the inner end plate 38 so that a fluid actuator or sealed service brake actuation chamber 144 is defined therebetween. Fluid pressure is selectively delivered to, and limitedly relieved from, this actuating chamber through a communicating internal passage 145 and a conduit 146 leading to the brake control system 134. With pressurization of the service brake actuation chamber the service brake piston moves axially outwardly to compress the brake discs 96, brake plates 98, and damping plates 100 against the thrust surface 86 and frictionally against each other for selective holding engagement of the rotatable wheel hub 22. A plurality of shouldered guide pins 148 are removably secured to the service brake piston 122 and individually extend freely slidably through the bores 128 in the parking brake piston 124 to permit axial movement, while preventing relative rotation, therebetween. Each of these pins also guidably entraps a relatively light capacity retraction spring 150 thereon for the purpose of retracting the service piston.

Operation of the first example

While the operation of the present invention is believed clearly apparent from the foregoing description, further amplification will subsequently be made in the following brief summary of such operation.

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5 The annular dual piston brake arrangement 10 is shown with no fluid pressure present in either of the internal passages 136 or 145 leading to the service brake and parking brake actuation chambers 132 and 144, respectively. This corresponds to either a vehicle parking or an emergency braking condition, and in either of these conditions the control system 134 has operated selectively or automatically to relieve pressure from these chambers. Under such relieved circumstances the compression springs 138 apply a relatively substantial force through the axially outwardly extended parking brake piston 124 directly against the counterbore 140 of the service brake piston 122. This forces the service brake piston against the stack of interleaved brake discs 96, brake plates 98 and damping plates 100 to compress them as shown against the thrust surface 86 of the outer end plate 78. In this way the discs are frictionally clamped to the axle housing 14, thereby also tending to hold the connecting drum 90, wheel hub 22 and the wheel rim arrangement 34 fixed. Any opposing force due to the relatively few and lightweight retraction springs 150 is easily overcome, resulting in a clamping force in the instant example of approximately 28,000 pounds. This is sufficient to hold a vehicle incorporating the present invention on an approximate 16% grade even though it is heavily loaded.

35 With vehicle movement, however, fluid pressure at approximately 400 psi is communicated from the control system 134 to the parking brake actuation chamber 132 to fully retract the parking brake piston 124 and thereby overcome the compression springs 138. In the absence of operating pressure in the service brake actuation chamber 144, the axially inward movement of the parking brake piston also serves to retract the service brake piston through the additionally axially compressed retraction springs 150. Thus, the two pistons are placed in a released condition.

50 Service braking is thereafter achieved through the selective delivery of fluid from the control system 134, the conduit 146, and the passage 145 to the service brake actuation chamber 144 at a pressure of up to approximately 600 psi in response to operator demand. This positions the service brake piston axially outwardly away from the retracted parking brake piston 124, and achieves a very effective maximum clamping force in the instant example of approximately 36,000 pounds.

60 The control system 134 allows only a predetermined volume of fluid to be exhausted from the service brake actuation chamber 144 upon the selective release of service braking. This permits limited retracting travel of the service brake piston

122 away from the previously tightly clamped brake discs 96, brake plates 98 and damping plates 100, and serves to automatically adjust any slack to beneficially improve service braking response upon the re-engagement thereof. For example, the volume of fluid exhausted from the service brake actuation chamber is limited to an amount sufficient to provide approximately 0.005 inch clearance between each brake disc 96 and brake plate 98. This minimizes the drag and horsepower loss therebetween, while simultaneously limiting the maximum amount of service brake piston travel independently of any wear of the brake discs 96 and brake plates 98 so that rapid response can occur. Under these conditions the service piston is axially separated from the retracted parking brake piston.

85 Further, as set forth previously, fluid is continually circulated through the conduits 52 and 54 and substantially radially outwardly through the full periphery of the annular brake arrangement 10. This circulation of cooling fluid occurs at a rate of approximately 75 gpm, and is sufficient to establish a 20 psi pressure level in the internal chamber 102. This pressure acts on the service brake piston 122 and results in a substantial retracting force which supplements that of the retraction springs 150. Thus, the presence of this pressurized fluid beneficially results in decreasing the number of retraction springs 150 required.

100 In an emergency situation, the fluid pressure to the parking brake actuation chamber 132 may be selectively or automatically released by the control system 134 so that the compression springs 138 will bias the parking brake piston 124 abuttingly against the service brake piston 122 to compress the brake discs 96 and brake plates 98 against the thrust surface 86. Under this condition, pressure to the service brake actuation chamber 144 is normally exhausted, but if desired, pressure could be simultaneously supplied thereto through the passage 145 and the control system to supplement the spring force and to apply an increased clamping force for even more rapid braking of the vehicle. This could result in a maximized clamping force of approximately 64,000 pounds.

120 Thus, the annular dual piston brake arrangement 10 of the present invention can achieve service braking at a predetermined force corresponding to a maximum pressure in the service brake actuation chamber 144, parking braking or emergency braking at a force of approximately 75% of this predetermined force, or emergency braking force of up to 175% of this predetermined force.

130 Because of the compact modular construction of the dual piston brake arrange-

ment 10, it is apparent that it may beneficially be preassembled before installation on the vehicle, and may be easily removed for servicing as a unit. For example, upon removing the wheel hub 22 and associated wheel rim arrangement 34 in the usual manner, it is thereafter only necessary to remove the adapter member 108 and the nuts 46 in order for the brake arrangement to be pulled axially outwardly from the flange 12 of the axle housing 14 as a unitary package. At this point the intimately engaged inner end plate 38, shell 72 and outer end plate 78 serve to contain the overlapping pistons 122 and 124, the interleaved brake discs 96 and brake plate 98, and the connecting drum 90 therewith, while the compression springs 138 simultaneously hold these components in an axially aligned clamped relation.

Therefore, the annular dual piston brake arrangements 10 of the present invention presents an extremely compact and effective multi-capacity, multi-purpose oil-cooled brake for a vehicle. Maximum use of available space by telescopingly and overlappingly relating the annular service brake piston 122 and the annular secondary brake piston 124 is coupled with substantially axially coextending compression springs for selective high capacity braking. Such annular arrangement also affords a unitized package which is easy to install and service.

Fluid cooling of the annular dual piston brake arrangement 10 beneficially results in aiding retraction of the service brake piston 122 so that fewer retraction springs 150 are required. This generally reduces the force that must be overcome by the actuation pressure in the service brake actuation chamber 144 to provide a specific degree of service braking.

Additionally, the limited exhausting of fluid from the service brake actuation chamber 144 upon disengagement of the service brake piston 122 limits its maximum travel to improve the rate of service braking response upon its subsequent re-engagement.

Description of the second example

With reference to the alternate embodiment of FIG. 2, an oil-cooled dual piston brake arrangement 10' is axially mounted outboard of a radially extending annular mounting flange 12', and with the flange integrally associated with a conically tapered axle housing 14'. A vehicle drive train 15' including a wheel hub 16' is rotatably supported on the axle housing axially outwardly of the brake arrangement by an anti-friction bearing 18'. The wheel hub includes an axially inwardly extending flange 20' with an external spline 22' thereon, and a radially outwardly extending

flange 24' to which is secured a wheel rim arrangement generally identified by the reference numeral 26'. Like the basic embodiment, this wheel rim arrangement is secured to flange 24' by a plurality of circumferentially spaced lug bolts 28' and associated nuts 30', and a tire 32' is sealingly mounted thereon.

The alternate embodiment dual piston brake arrangement 10' includes an anchoring inner end plate 34 which is removably secured to the mounting flange 12' of the axle housing 14' by a plurality of threaded studs 36' anchored to the end plate and extended axially inwardly through the flange for screw threaded receipt of a plurality of retaining nuts 38' thereon. Formed through the end plate are a cooling fluid inlet passage 40' and an outlet passage 42' which are respectively in communication with an inlet conduit 44' and an outlet conduit 46' for the purpose of normally continuously allowing the circulation of a cooling fluid through the brake arrangement by way of a fluid circulating apparatus 48'. Further, the end plate has formed therein an outer counterbore 50' and an inner counterbore 52' disposed in axially outwardly facing piston-receiving relation, and a plurality of circumferentially spaced spring-receiving cavities 54' opening endwardly on the inner counterbore and in open communication with the outlet passage 42'.

The alternate embodiment dual piston brake arrangement 10' also includes a centrally disposed cylindrical shell 56' having a plurality of internal teeth 58' formed therein which are disposed in substantial axial alignment with the outer counterbore 50' of the end plate 34'. The shell extends axially outwardly in spanning relation between the inner end plate and an outer end plate 60', and in order to secure them together a plurality of elongated bolts 62' are inserted axially inwardly therethrough which are adapted to screw threadably receive a corresponding plurality of retaining nuts 64' thereon. In this manner the outer end plate is suitably anchored, to thereby provide an annular inwardly facing thrust surface 66' against which the brake arrangement is clamped.

The outer end plate 60' is also provided with a circular aperture 68' centrally therethrough so that it may be spaced relatively radially closely to the external spline 22' of the wheel hub 16'. However, a cylindrical connecting drum 70' having a plurality of cooling fluid access passages 71' radially therethrough still has sufficient clearance to extend axially inwardly through this aperture from a position wherein an internal spline 72' thereon is engaged with the external spline of the hub. Consequently, the connecting drum rotates with movement of

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the vehicle and has a plurality of external teeth 74' about the periphery thereof on which are keyed a plurality of brake discs 76'. These discs are interleaved with a plurality of brake plates 78' which are similarly keyed to the internal teeth 58' of the shell 56'.

The dual piston-brake arrangement 10' of FIG. 2 also includes a parking brake housing 82' which is secured tightly against the inner end plate 34', and thereby effectively to the axle housing 14', by a plurality of retaining bolts 84' inserted therethrough for screw threaded engagement with the end plate. An inwardly facing, annular step 86' is formed about the periphery thereof, and a piston retaining and aligning dowel 88' is rigidly mounted for axially inward extension therefrom.

Actuation of the dual piston brake arrangement 10' is accomplished by axial reciprocating movement of an annular service brake piston 90' and an annular secondary or parking brake piston 92' telescopically nested radially within it, essentially like the basic embodiment. The parking brake piston is axially reciprocally and nonrotatably secured to the parking brake housing 82' by the guiding and aligning dowel 88', and includes a plurality of spring-receiving cavities 94' opening axially inwardly thereof and a plurality of guide-receiving bores 96'. Further, it includes an axially outwardly facing counterbore 98' which cooperates with the annular step 86' of the parking brake housing 82' in such a manner as to define a fluid retractor or sealed parking brake actuation chamber 100' therebetween. A pair of seal ring assemblies 102' define the radially outer and inner limits of this actuating chamber, and fluid pressure is delivered thereto from a brake control system 104' by way of an interconnecting conduit 106' and an internal passage 108'. Pressurization of the chamber normally positions the parking brake piston 92' axially to the right when viewing FIG. 2 against the opposition of a plurality of relatively high capacity compression springs 110. These compression springs are substantially equally circumferentially spaced around the periphery of the brake arrangement, and are seated intermediate the cavities 54' and 94' on a common spring pitch circle 112' having a relatively large radius.

More particularly, the service brake piston 90' includes an inner counterbore 114' and a seal ring groove 115' opening thereon, and an outer counterbore 116' adapted to axially receive the secondary brake piston reciprocally therein. It further has around its periphery an inwardly facing, annular step 118' which effectively co-operates with the counterbores 50' and 52'

of the inner end plate 34' to define a fluid actuator or sealed service brake actuation chamber 120' therebetween with the aid of a pair of seal ring assemblies 122'. Such seal ring assemblies define the radially outer and inner limits of this actuation chamber, and it is to be noted that the outer limit thereof is substantially axially aligned with the internal teeth 58' of the shell 56'. Fluid pressure is selectively delivered to this actuation chamber through an internal passage 124' and an interconnected conduit 126' leading to the brake control system 104'. With pressurization thereof, the service brake piston moves axially outwardly or to the left when viewing the drawing to compress the brake discs 76' and the brake plates 78' against the thrust surface 66'.

As is clearly shown in FIG. 2, a plurality of shouldered guide pins 128' are removably secured to the service brake piston 90' so that they individually extend freely slidably through the bores 96' in the secondary brake piston 92'. In this manner, relative axial movement of the two pistons is permitted, while the relative rotation thereof is prevented. Each of these pins also guidably entraps a relatively light capacity retraction spring 130' thereon for the purpose of retracting the service brake piston in the absence of pressure fluid in the actuation chamber 120'.

The alternate embodiment dual piston brake arrangement 10' is fully peripherally cooled by the circulation of cooling fluid from the fluid circulating apparatus 48', the inlet conduit 44', and the inlet passage 40' communicating with a centrally disposed internal chamber 132'. Because of the relative rotary motion between the wheel hub 16' and the axle housing 14', and members secured thereto, an annular internal seal arrangement 134' and an annular external seal arrangement 136' are required. The internal seal arrangement is generally disposed between an adapter member 138' secured to the axle housing 14' and the rotatable connecting drum 70', while the external seal arrangement is sealingly disposed between the outer end plate 60' and another adapter member 140' secured to the wheel hub 16'.

The alternate embodiment brake arrangement 10' provides an annular lip-type seal ring 142' of glass fiber filled polytetrafluoroethylene or comparable resilient material which is mounted in the groove 115' of the service brake piston 90' in radially sealing relation against the parking brake piston 92'. In this way the clearance between the service brake piston and the parking brake piston is effectively sealed to prevent the aforementioned cooling fluid from passing radially outwardly therebetween in bypassing relation with respect to

the brake discs 76' and brake plates 78'. Such lip-type seal ring has a relatively low coefficient of friction and the ability to remain resilient over a wide range of operating temperatures. Further, it is oil resistant and adaptable to a relatively wide range of diametrical dimensions and eccentricity tolerances which may normally occur in the manufacturing of the pistons.

For the purpose of allowing cooling fluid egress, the alternate embodiment construction of FIG. 2 further provides a plurality of equally circumferentially spaced, flow directing passages 144' through the periphery of the service brake piston 90'. These radially inwardly inclined or convergingly oriented passages communicate with an annular spring chamber 146' through a radial gap 148' defined radially between the periphery of the parking brake piston 92' and the counterbore 116' of the service brake piston 90'. Subsequently, fluid egress from the chamber is by way of the outlet passage 42', the outlet conduit 46', and the fluid circulating apparatus 48'.

Operation of the second example

As is clearly apparent in FIG. 2, the service brake piston 90' and the parking brake piston 92' are disposed to the left when no fluid pressure is present in either of the service brake or parking brake actuation chambers 120' or 100', respectively. This condition corresponds to either a vehicle parking or an emergency braking condition, wherein the compression springs 110' apply a substantial load axially outwardly against the parking brake piston to urge it abuttingly against the service brake piston. In turn, the leftward force of the service brake piston serves to compress the interleaved brake discs 76' and brake plate 78' against the thrust surface 66' of the outer end plate 60', and to thereby effectively secure the connecting drum 70', the wheel hub 16' and wheel rim arrangement 26' to the axle housing 14' for braking thereof.

To allow vehicle movement, however, pressurized fluid is communicated from the control system 104' to the parking brake actuation chamber 100' via the conduit 106' and the passage 108' for fully retracting the parking brake piston 92' and thereby overcoming the loading of the compression springs 110'. In the absence of any operating pressure in the service brake actuation chamber 120', the axially inward or rightward movement of the parking brake piston also serves to retract the service brake piston because of the action of the additionally axially compressed retraction springs 130'.

In accordance with operator demand, service braking is thereafter achieved by selectively modulating fluid delivery from the brake control system 104', the conduit

126', and the passage 124' to the service brake actuation chamber 120'. This moves the service brake piston 90 axially outwardly, in longitudinally spaced relation away from the retracted parking brake piston 92', to selectively clamp the interleaved brake discs 76' and brake plates 78' together with the desired force. In this way a holding force is applied to the wheel hub 16' and wheel rim arrangement 26' to retard or stop the vehicle.

Cooling fluid is continually circulated at a rate of approximately 75 gpm (4.7 liters per sec.) by the fluid circulating apparatus 48' in a generally counterclockwise manner when viewing the drawing as shown by the arrows. Cooling fluid ingress is by way of the conduit 44' and the passage 40' to the internal chamber 132'. From this collecting chamber the fluid at a relatively low effective pressure of approximately 20 psi (1.4 Kg per sq. cm.) passes radially outwardly through the passages 71' in the connecting drum 70' to impinge upon and cool the brake discs 76' and the brake plates 78' as well as the service brake piston 90'. Although not shown, it should be understood that these discs and/or plates are provided with a plurality of conventional surface grooves which permit cooling fluid flow there-through even though they are tightly pressed together. Beneficially, the lip-type seal ring 142' between the service brake piston and parking brake piston 92' prevents this cooling fluid from passing radially outwardly between them and assures flow solely through the interleaved brake discs and plates.

In keeping with one of the features of the invention, an egress path for this cooling fluid is provided by allowing it to flow axially and radially obliquely inwardly through the flow-directing passages 144' in the service brake piston 90'. Cooling fluid egress thereafter continues substantially axially through the annular gap 148', the spring chamber 146', the outlet passage 42', and the outlet conduit 46' to return to the fluid circulating apparatus 48' to complete the egress path routing thereof. It is important to note that fluid egress by way of the inclined passages 144' in the service brake piston permits the service brake actuation chamber 120' to be disposed radially outwardly a relatively maximum permissible amount. In this way the radially outer and inner limits thereof, corresponding to the disposition of the seal ring assemblies 122', are a maximum so that less pressure is needed in the actuation chamber to achieve a particular force on the piston. A further advantage of this radially outward chamber location is that the spring pitch circle 112', can also be at a larger radius. This enables a greater number of springs to be positioned equally around the pitch circle, so that to



provide a predetermined maximum force on the parking brake piston 92' a savings in axial length and/or spring size can be effected.

- 5 Therefore, it is apparent that the alternate embodiment dual piston brake arrangement 10' provides a telescopically related arrangement of the annular service brake piston 90' and the annular parking brake piston 92' with an improved cooling circuit therefor which permits the service braking piston actuation chamber 120' and the greater number of compression springs 110' to be disposed radially outwardly at a maximum radius for improved compactness. Maximum use of available space within the vehicle wheel rim arrangement 26' is coupled with the substantial axial alignment of these springs with the brake pistons for high capacity engagement thereof.

WHAT WE CLAIM IS:—

1. An annular dual piston brake arrangement for selective holding engagement of a rotatable member of a drive train, the arrangement comprising a support structure; an annular service brake piston axially reciprocably and nonrotatably mounted within the support structure coaxially with the rotatable member; fluid actuating means for biasing the service brake piston toward a brake holding position in use with respect to such rotatable member; an annular secondary brake piston axially reciprocably and nonrotatably mounted within the support structure in axially telescoping, substantially radially aligned relation with respect to the service brake piston; fluid retracting means for holding the secondary brake piston in a disengaged condition axially spaced from the service brake piston; and spring engagement means for biasing the secondary brake piston directly axially against the service brake piston for subsequent brake holding engagement of the rotatable member in the absence of a predetermined pressure in the fluid retraction means.

2. A brake arrangement according to Claim 1, including seal means disposed between the support structure and the rotatable member to thereby define an internal chamber therebetween, and fluid circulating means for delivering fluid in a fluid path radially outwardly through the internal chamber in order to cool the annular brake arrangement around its full periphery, the fluid also serving to aid in retracting the service brake piston.

3. A brake arrangement according to claim 1 or claim 2, wherein the support structure includes a thrust surface thereon and a plurality of radially internal teeth on which are keyed a plurality of annular plates, and the rotatable member includes a

plurality of radially external teeth on which are keyed a plurality of annular discs interleaved with the plates, the discs and plates being biased together and against the thrust surface upon axial movement of the service brake piston toward the brake holding position and being disposed in the fluid path.

4. A brake arrangement according to any one of the preceding claims, including retracting spring means for axially disengaging the service brake piston from the rotatable member in the absence of a predetermined pressure in the fluid actuating means.

5. A brake arrangement according to claim 4, wherein the fluid actuating means includes a service brake actuation chamber defined between the support structure and the service brake piston, and a control system therefor for permitting a predetermined volume of fluid to be exhausted from the chamber upon the selective release of the service brake piston away from a brake holding position to thereby limit its maximum amount of travel and improve its reengagement response.

6. A brake arrangement according to claim 4 or claim 5, including alignment means for establishing an axial free sliding nonrotating relation between the support structure and the secondary brake piston, and wherein the support structure has a plurality of cavities therein and the secondary brake piston has a corresponding plurality of opposing cavities therein, and the spring engagement means includes a plurality of compression springs individually seated within each of the aligned cavities.

7. A brake arrangement according to claim 6, including guide pin means intermediate the secondary brake piston and the service brake piston for establishing axial free sliding and nonrotating cooperation therebetween.

8. A brake arrangement according to claim 7, wherein the secondary brake piston includes a plurality of axial bores there-through concentric with a limited number of the cavities, and the guide pin means includes a plurality of shouldered guide pins secured to the service brake piston and extending through the bores, and the retracting spring means includes a plurality of retracting compression springs individually mounted on the shouldered guide pins.

9. A brake arrangement according to claim 8, wherein the service brake piston includes an axially inwardly open counterbore adapted to closely receive the secondary brake piston reciprocably therein, the secondary brake piston being arranged for limited axial movement between an abutting position with the counterbore to a retracted position axially spaced away from the counterbore and serving to



provide a predetermined maximum force on the parking brake piston 92' a savings in axial length and/or spring size can be effected.

5 Therefore, it is apparent that the alternate embodiment dual piston brake arrangement 10' provides a telescopically related arrangement of the annular service brake piston 90' and the annular parking brake piston 92' with an improved cooling circuit therefor which permits the service braking piston actuation chamber 120' and the greater number of compression springs 110' to be disposed radially outwardly at a maximum radius for improved compactness. 10
15 Maximum use of available space within the vehicle wheel rim arrangement 26' is coupled with the substantial axial alignment of these springs with the brake pistons for high capacity engagement thereof. 20

WHAT WE CLAIM IS:—

1. An annular dual piston brake arrangement for selective holding engagement of a rotatable member of a drive train, the arrangement comprising a support structure; an annular service brake piston axially reciprocably and nonrotatably mounted within the support structure coaxially with the rotatable member; fluid actuating means for biasing the service brake piston toward a brake holding position in use with respect to such rotatable member; an annular secondary brake piston axially reciprocably and nonrotatably mounted within the support structure in axially telescoping, substantially radially aligned relation with respect to the service brake piston; fluid retracting means for holding the secondary brake piston in a disengaged condition axially spaced from the service brake piston; and spring engagement means for biasing the secondary brake piston directly axially against the service brake piston for subsequent brake holding engagement of the rotatable member in the absence of a predetermined pressure in the fluid retraction means. 25
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2. A brake arrangement according to Claim 1, including seal means disposed between the support structure and the rotatable member to thereby define an internal chamber therebetween, and fluid circulating means for delivering fluid in a fluid path radially outwardly through the internal chamber in order to cool the annular brake arrangement around its full periphery, the fluid also serving to aid in retracting the service brake piston. 50
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3. A brake arrangement according to claim 1 or claim 2, wherein the support structure includes a thrust surface thereon and a plurality of radially internal teeth on which are keyed a plurality of annular plates, and the rotatable member includes a 60

plurality of radially external teeth on which are keyed a plurality of annular discs interleaved with the plates, the discs and plates being biased together and against the thrust surface upon axial movement of the service brake piston toward the brake holding position and being disposed in the fluid path. 65
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4. A brake arrangement according to any one of the preceding claims, including retracting spring means for axially disengaging the service brake piston from the rotatable member in the absence of a predetermined pressure in the fluid actuating means. 75

5. A brake arrangement according to claim 4, wherein the fluid actuating means includes a service brake actuation chamber defined between the support structure and the service brake piston, and a control system therefor for permitting a predetermined volume of fluid to be exhausted from the chamber upon the selective release of the service brake piston away from a brake holding position to thereby limit its maximum amount of travel and improve its re-engagement response. 80
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6. A brake arrangement according to claim 4 or claim 5, including alignment means for establishing an axial free sliding nonrotating relation between the support structure and the secondary brake piston, and wherein the support structure has a plurality of cavities therein and the secondary brake piston has a corresponding plurality of opposing cavities therein, and the spring engagement means includes a plurality of compression springs individually seated within each of the aligned cavities. 95
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7. A brake arrangement according to claim 6, including guide pin means intermediate the secondary brake piston and the service brake piston for establishing axial free sliding and nonrotating cooperation therebetween. 105

8. A brake arrangement according to claim 7, wherein the secondary brake piston includes a plurality of axial bores there-through concentric with a limited number of the cavities, and the guide pin means includes a plurality of shouldered guide pins secured to the service brake piston and extending through the bores, and the retracting spring means includes a plurality of retracting compression springs individually mounted on the shouldered guide pins. 110
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9. A brake arrangement according to claim 8, wherein the service brake piston includes an axially inwardly open counterbore adapted to closely receive the secondary brake piston reciprocably therein, the secondary brake piston being arranged for limited axial movement between an abutting position with the counterbore to a retracted position axially spaced away from the counterbore and serving to 120
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additionally axially compress the retracting compression springs.

10. A brake arrangement according to claim 2 or to any one of the claims 3 to 9 when dependent on claim 2, wherein the service brake piston includes fluid passage means therethrough which is disposed in communication with the fluid path in order to provide an egress path for the fluid upon peripheral cooling the annular brake arrangement.

11. A brake arrangement according to claim 10, including seal means disposed between the service brake piston and the secondary brake piston for preventing the radially outwardly flowing cooling fluid from escaping therebetween and bypassing cooling of the rotatable member.

12. A brake arrangement according to claim 11, wherein the service brake piston includes an axially inwardly open counterbore, and the secondary brake piston, is positioned axially within the counterbore, and the seal means includes an annular seal ring cooperatively disposed therebetween.

13. A brake arrangement according to claim 12, wherein the service brake piston further defines an internal groove opening radially inwardly on the counterbore and the annular seal ring is a lip-type seal ring disposed in the groove for facing engagement against the peripheral surface of the secondary brake piston.

14. A brake arrangement according to claim 13, wherein the lip-type seal ring is made of glass fibre filled polytetrafluoroethylene.

15. A brake arrangement according to claim 1, substantially as described with reference to Figure 1 of the accompanying drawings.

16. A brake arrangement according to claim 10, substantially as described with reference to Figure 2 of the accompanying drawings.

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