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A calliper-type brake gear designed to operate with oil-controlled spring units for even distribution of pressure and wear.

High-pressure oil-controlled brakes for winders

by Grant Ray Sutherland, 1958

The mechanical brake must often assist the driver in bringing the conveyance to rest at the correct point. It must act as a clamp and provide emergency retardation in the event of overspeeding or overwinding. It must therefore be precise, robust, nick in emergency response and fail to safety in the event of loss of oil or air pressure.

All these points are not always complementary to one another. For instance, for a brake to be robust it must usually be heavily built, and if it is heavily built, it is not necessarily going to be quick to action or precise in its action. Such a brake may use dead weights to fail to safety, which involves so much mass that, if required to operate in a hurry, large and dangerous urges of braking force will be produced.

South African winders

Brakes designed to South African brake regulations, particularly for double-clutched drum winders, usually involve dead weights which move through considerable distances to apply the braking force. This braking force must be sufficient to hold the normal motor torque plus the full out-of-balance static torque of the loaded cage on one brake, with one drum declutched.

Such brakes can build up high retardation rates and high stresses in brake gear components due to the physical proportions of some winders. These undesirable features called for a means to limit the speed of application of the brake and the amount of braking force which can be built up before the drum comes to rest from full speed.

In some cases, a time lag of up to 13 sec is introduced deliberately by trial and error setting, after allowing the brake shoes to make contact with the path, to prevent dangerous retardations from building up. This would result in severe stress concentrations in the ropes depending from the drum. Constant and careful checking of this slugged brake performance must made to compensate for wear of brake linings.

If these same brakes were adjusted in the manner described and called upon to give the protection required on British winders, they would not stand up to the test. On some powerful steam winders capable of accelerating at 12 ft/sec/sec (3,6 m/sec/sec), the brakes must prevent a gain in speed (under full acceleration, with an out-of-balance load descending) not exceeding 1,5 ft/sec (0,5 m/sec) after the operation of the safety device. The time lag between controller trip and full neutralising retardation must not exceed 0,125 sec.

High-pressure oil brakes

A patented high-pressure, oil-controlled, spring-powered brake meets the requirements regarding brake holding of single unbalanced loads of the South African mines department. Designed for strength and rigidity, with almost no moving parts, they can be brought into full operation safely at any rope speed bwteen 0,05 and 0,1 sec. Adjustment of these brakes is not associated with brake lining wear. The release of energy when applied rapidly is extremely small and braking forces are metered out with extreme accuracy. Conditions do, however, exist where large movement or

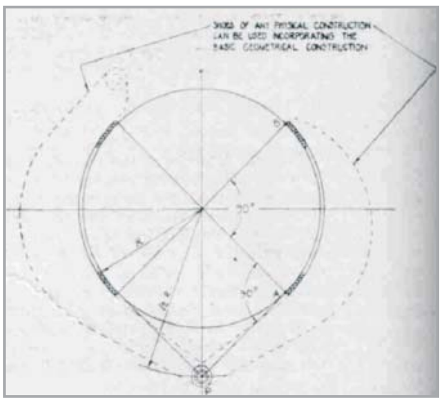

Fig. 1a: Geometrical construction of brake shoes.

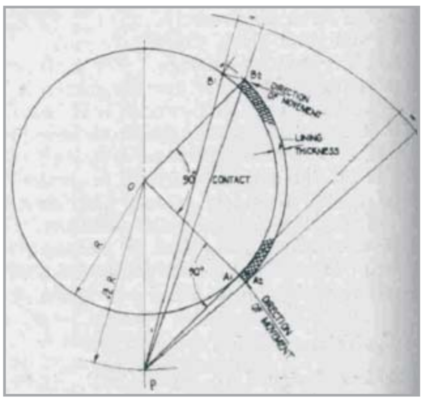

Fig. 1b: Analysis of movement of brake shoes.

inefficient transmission and distribution of forces may exist and these must be watched at the design stage.

Brake gear design

The conventional, centre-suspended, floating-type brake does not altogether take advantage of all the advantages of high-pressure oil because, with this type of construction, there is inevitably some deflection in the lever arrangement. This results in lost motion and in the difficulty of maintaining a uniform clearance between the shoes and the path.

The brake discussed here uses only three to six pivot joints instead of 16 as employed in centre-suspended brakes. Minimum even working clearance is ensured by constrained movement. Figs. 1 and 2 show the geometrical construction of the brake shoes. The single pivot shoe whose pivot pin is situated at P where OP is $\sqrt{2}$ time the radius of the path on which it operates, results in the radical movement at A. The bottom-end of the lining is exactly the same as the radial movement at B, the top end of the lining. Also, the pressure will be distributed evenly on either side of a centrally-disposed centre of pressure.

Point A, at the bottom end of the lining, is at the point of contact of a tangent drawn from P to the brake circle, and OB, the top end of the lining, is at 90° to OA from the centre of the brake circle. The only other essential feature is that the shoes, adopting this relative position of lining to pivot, shall be made very stiff so that no deflection takes place under load. Fig. 3 shows the distribution of pressure, which is equal on either side of a centrally-disposed centre of pressure, Figs. 4, 5 and 6 show three different constructions which are in general use. Constructions shown in Figs. 4 and 6 are the two most generally adopted.

The construction in Fig. 4 has a closed system of applied forces. The only force, the torque reaction at the pivots of the shoes, comes from outside the brake gear itself.

The working stroke for rigid brake shoes is ¾ in (1,9 cm), assuming a working clearance of 1/16 in (0,1 cm) at the centre of pressure of each shoe, and using a bell crank lever with a three-to-one leverage.

The working stroke of the spring thrusters in Fig. 4 is only 1/16 in (0,1 cm) for a working shoe clearance of 1/16 in (0,1 cm). With working strokes of this order, the momentum of moving parts on quick application is very small and their impact reaction on the brake path will consequently be negligible.

The impact velocity of the shoes themselves when "fired" on the path in 0,05 s is only 0,208 ft/sec (6,3 cm/sec). The entire braking force is supplied from pre-compressed springs designed to release only a very small portion of the total energy of the spring between the brakes "off" and brakes "on".

If the expansion and contraction of the springs are small compared with their total compression, the variation of strain will also be very small. The change in strain will be so small that ultimate fatigue of the spring may be ruled out for the life of the winder.

The use of springs

In England, springs in this form have been used constantly since 1947. They have been used purely as emergency forces since 1927 without a single case of fracture.

By using springs in this way, mass inertia in the brake system is eliminated almost completely. The impact and momentary crushing which normally follow quick application with deadweight brakes (at high speed) may be ignored as the

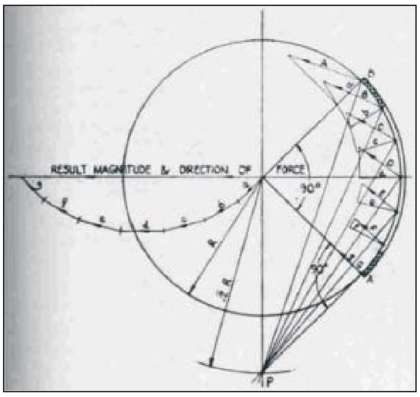

Fig. 1c: A.B.C – G – magnitude and direction of applied forces a.b.c – g – magnitude of resultant radial forces.

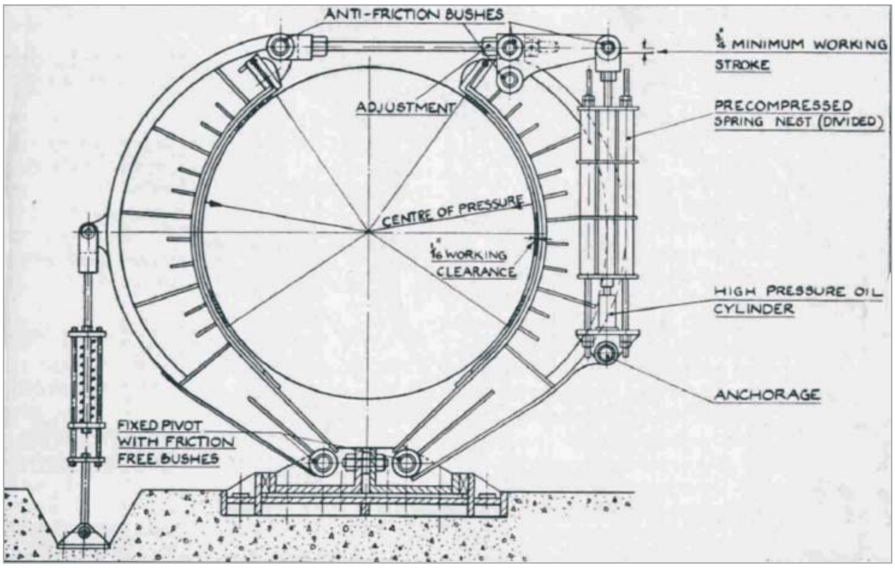

Fig. 2: Mechanical linkage between brake shoes and thruster.

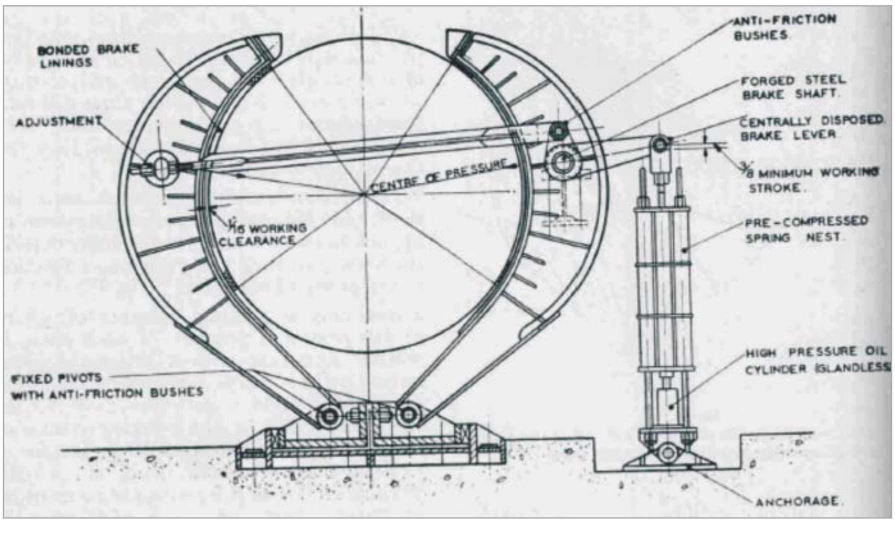

Fig. 3: Alternative arrangement of mechanical linkage.

braking force will not rise above the force the springs are set for.

The retardation set up in the winder will never exceed that for which the brake is designed and, providing the retardation at the rope does not exceed 10 ft/sec/sec (3 m/sec/sec) (preferably 8 ft./sec/sec or 2,4 m/sec/sec), the effect on riders in the cage will not be uncomfortable.

Rate of change of retardation

One could argue that, because the braking force is built up in under 0,05 s, the rate of change of retardation would cause discomfort to riders in the cage or induce high momentary stress in the winding ropes, as in the instant a retarded cage comes to rest at constant retardation. There is, however, a subtle difference between the two conditions.

In the case of a cage coming to rest at constant retardation, the rate of change of retardation is infinite, but in the case of a brake building up to say 10 ft/sec/sec (3 m/sec/sec) in 0,05 s at 40 ft/sec (12,2 m/sec), the rate of change of retardation will be 200 ft/sec^3, (12,2 m/sec^3). This change takes place while 2 ft. (61 cm) of rope is paid out. The rate of change of retardation is not infinite and the braking force from the spring nest settles to its constant value.

In Britain, winders with maximum midshaft speeds of 100 ft/sec (30,4 m/sec) (6000 ft/min or 1,8 km/min) are fitted with high-speed spring braking and the brakes described here. Anyone riding in the cage at the time of an emergency trip, when accelerated into the trip at full midshaft speed, will experience a smooth pull-up to stop.

Records taken with telemetering decelerometers would reveal dangerous stress conditions when conventional brakes are applied at points near the end of the wind. A record with such an instrument was taken and, although the average rates of retardation were set to 12 – 14 ft/sec/sec (3,6 – 4,3 m/sec/sec), the maximum recorded rope stress with brakes applied at various speeds was not much higher than the natural induced stress due to the retardation.

Winder protection

The equipment is designed on a very simple formula, elimination of item lag a simple centrifugal governor, a floating lever, trip mechanism, and spring-applied brake.

It is possible, with this combination, to reduce to 0,5 – 1 ft/sec (15 – 30,5 cm/sec) the total gain in winder speed when tripped under full on steam

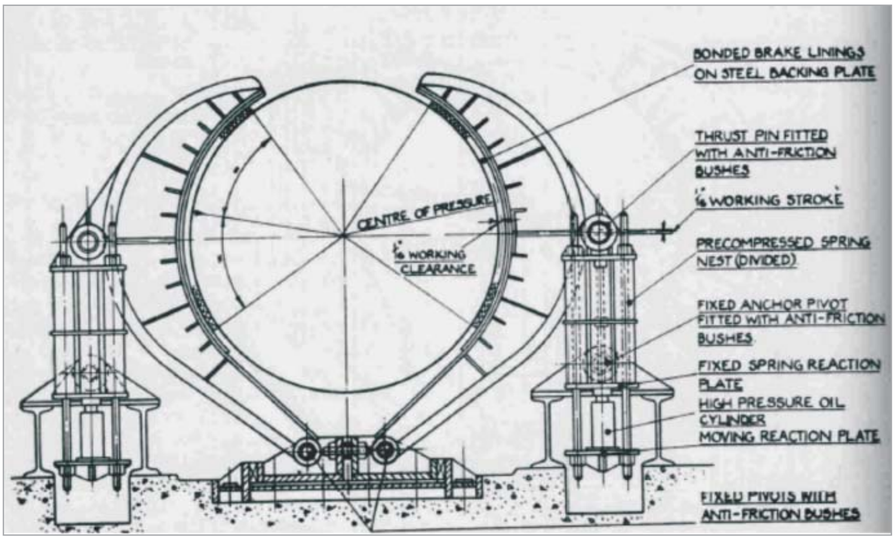

Fig. 4: Brake shoes individually operated. Linkage between shoes is entirely in the hydraulic circuit.

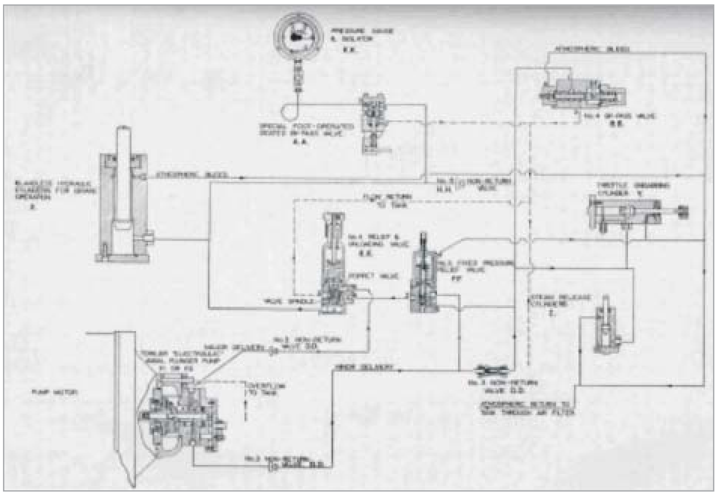

Fig. 5: High presure oil circuit for manually controlled brake.

winders and between 0,1 and 0,2 ft/sec (3,6 – 7,6 cm/sec) on electric winders.

There are various ways of damping or slugging the brake by introducing additional valves according to special requirements.

High-pressure oil brakes on friction winders

In a recent experience with four rope friction winders, the inertia of the suspended masses were considerably greater than that of the drum and oscillation between the drum and the suspended masses is induced very easily by any change in speed. The smaller the winder, the greater this difference in inertia. On one AC winder, the inertia of the suspended masses (including the rope) is ten times that of the drum. The inertia of the motor rotor referred back to the drum through the gears is just over twice that of the drum.

The brake on the winder was adjusted to induce retardation at the cage or skip of 5,5 ft/sec/sec (1,7 m/sec/sec) to enable safety and resistance to slip. It was found that, if this brake was applied in 0,05 s, oscillation would be induced. This could cause the ropes to "leap-frog" over the drum. Similar oscillations could be induced by the applying dynamic braking.

The winder was fitted with high pressure oil-controlled spring brakes, and required only an adjustable relief valve and a non-return valve in the feed pipe to the brake control cylinders for rapid build-up of braking force to 1,5 ft/sec/sec (0,5 m/sec/sec) and a further build-up over 0,3 s to 5,5 ft/sec/sec (1,7 m/sec/sec). This valve is fitted into the circuit as shown in Fig. 6.

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