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⑤④ **SAFETY MECHANISM FOR HOISTING DRUMS.**

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## Description

The invention relates to a safety system according to the preamble part of claim 1.

A safety-system as known from US—A—4 177 973 comprises a mechanical out-of-sync detector having two coaxially arranged discs, each being driven by one component of the hoist device. Both discs are provided with a peripheral recess and rotate with the drum and drive motor respectively and with the same speed. During normal operation of the hoist system both recesses are not aligned with each other. A cam follower of a rotatably mounted switch lever rides on the peripheries of both discs simultaneously. The lever cooperates with an electrical triggering switch for the actuation of the safety brake. In case of a failure in the main drive train between the drive motor and the drum the discs rotate with different speeds until both recesses are aligned. The cam follower then falls into the aligned recesses and actuates the trigger switch for the safety brake. The out-of-sync detector works mechanically but can actuate the safety brake only with the help of electrical power supply. One disadvantage of said embodiment of the safety system is its strict dependency on a correct electrical power supply which can easily be interrupted under rough working conditions. A further disadvantage is that in such hoist systems in practice the gear ratios in both drive trains from the motor to the output of the detector cannot be matched precisely enough so that there will be a constant out-of-sync rotation which accumulates and leads to an inadvertent safety-brake-setting (nuisance-brake-setting). Furthermore, such hoist systems conventionally contain a torque-limiting device or a slip clutch within the drive trains, an inherent disadvantage of which is a certain unavoidable slippage. The mechanical out-of-sync detector cannot differentiate between uncritical differences in speed and direction of the two supervised drive train branches and serious critical differences due to initial drive train failures but, triggers the actuator of the safety brake in both cases.

However, US—A—4 177 973 discloses a further embodiment of an electropneumatically working safety system for a hoist assembly, said safety system comprising an electrical out-of-sync detector with counters and comparators and an electrically working error correcting means. The counters and comparators count and compare the number of pulses derived from the motor and the drum shaft rotation respectively via encoders. As soon as a predetermined difference between the pulses counted is exceeded a solenoid valve in the supply line for a release mechanism of the actuator of the safety brake is actuated in order to set the safety brake. Said electrical error correcting means is in the form of a sensitivity adjustment and is associated with the comparators in order to compensate speed differences between both supervised drive train branches or to take up constant mechanical gear lash in the drive train. Said

electrical error correcting means influences the safety system independently from the working speed of the hoist device. Said embodiment of the safety system is strictly dependent on external power sources. i.e., pneumatical and electrical power, the constant supply of which cannot be ensured under rough working conditions. In practice the safety system cannot completely avoid nuisance-brake-settings. Furthermore, the electrical and pneumatical components of the safety system strongly suffer from vibrations, dirt and bad weather conditions.

It is a task of the present invention to achieve a reliable safety-system of the mechanical type which can withstand rough operating conditions, which avoids nuisance-brake-settings of the safety brake, and which is independent from electrical power supply.

This task can be achieved with the feature combination in the characterizing part of claim 1.

The safety-system according to the invention is together with its error correcting means completely mechanical and thus operates reliably even under rough working conditions. Its safety function does not depend on external power sources because the mechanical drive power for the error correcting means can be directly derived from the drive trains or from an internal mechanical power source. The undesirable influence of creep or unprecisely matched gear ratios is compensated with the help of an intentionally produced correcting motion which nullifies the brake setting output resulting, e.g. from creep. The sensitivity of the safety system and particularly the error correcting means adapts itself to the working speed of the hoist system in order to ensure an actuation of the safety brake in critical and dangerous situations and independently from the working speed.

Preferred embodiments are contained in the depending claims.

The drawings:

Fig. 1 is a schematic plan view of a hoisting mechanism employing the safety system of this invention.

Fig. 2 is a section along the line 2—2 of Fig. 1.

Fig. 3 is a fragmentary section taken along line 3—3 of Fig. 1.

Fig. 3A is a schematic fragmentary section of another embodiment of detection device employed in one form of safety system.

Fig. 4 is a side elevation of the hoisting device and safety system of Fig. 1.

Fig. 5 is a fragmentary section of an overspeed clutch.

Fig. 6 is a load-sensitive control for the overspeed clutch of Fig. 5.

Fig. 7 is a schematic elevation of another embodiment of brake-setting apparatus.

Fig. 8 illustrates a mechanical brake-setting apparatus.

Fig. 9 is a schematic plan of a safety system.

Fig. 10 is a fragmentary enlarged section of a portion of a brake-setting trigger employed in the apparatus of Fig. 9.

Fig. 11 is a schematic taken generally along the line 11—11 of Fig. 12, illustrating a differential assembly embodiment of the system shown in Fig. 9.

Fig. 12 is a side elevation of a portion of the apparatus shown in Fig. 9.

Fig. 13 is an isometric of a portion of the system.

Fig. 14 is a section taken generally along the line 14—14 of Fig. 11.

Fig. 15 is a section taken generally along the line 15—15 of Fig. 12.

Fig. 16 is a schematic fragmentary side elevation of another embodiment of a detector.

Fig. 17 is a fragmentary plan of the device shown in Fig. 16.

As shown in Fig. 1, a hoist system includes an operating brake 2 coupled to a motor shaft 2a which is powered by a motor 3. A coupling 4 couples the motor to a conventional gear reduction unit 5, such as a 500:1 reduction, which has an output shaft 7 rotatably carried in a pillow block 8. A drum pinion 9 meshes with the drum gear 10.

A drum 11 is rotated by the drum gear on a shaft 11a which is rotatably supported in a pair of spaced pillow blocks 13. The hoist system is provided with a second brake, such as a band brake 14, wrapped on a brake drum 12. A brake-applying assembly or brake actuator 15 will set the brake in response to a detected failure or other hazard condition.

A torque limiter assembly 6 is provided to limit the torque which would be imposed from high-speed rotational kinetic energy of the motor and high-speed drive elements of the gear reduction and motor drive if a load hang-up, overload or two-blocking occurs.

A mechanical differential, out-of-sync detector 20 is provided for detecting the failure or hazard condition. In one embodiment, the detector also provides the mechanical force for applying the band brake 14. In preferred embodiments, the detector merely signals the out-of-sync detection and a separate brake actuator sets the brake, as in Figs. 7 and 8. The out-of-sync detector 20 includes a first input shaft 30 which is coupled to the motor shaft 2a by a right angle drive 19 having a gear reduction equivalent to that of the total gear reduction between the motor and the drum. A right angle drive 18 also couples the drum shaft 11a to a second input shaft 31 to the detector 20. The purpose of the gear reduction in right angle drive 19 is to bring the two input shafts entering the detector to approximately the same speed. Other forms of speed reduction can also be provided.

Each of the input shafts 30, 31 (Fig. 3), is keyed to a drive gear which meshes with a side gear 26. The side gears 26 are keyed to pinion gears 27 that mesh with a planetary gear 27a of a planet carrier 28. Equal and opposite rotational velocities of the input shafts 30 and 31 will cause the pinion gears 27 to rotate the gear 27a about a planet carrier post 28a that is fixed by a pin 46 to an

output shaft 29 carried in bearings 25. Should the rotational velocity on either of the input shafts vary, an angular velocity will be created in the output shaft 29, the speed of which will depend on the relative variation between the velocities of the two input shafts. Thus, for example, if one of the input shafts should stop completely, the rotational output of the output shaft 29 will immediately reach a maximum speed. It is this rotation of the output shaft which triggers the brake actuation and, in one embodiment, creates or generates the force necessary for applying the brake on the brake drum 12. It is possible to place this brake on a downstream pinion shaft in close proximity to the drum rather than directly on the drum. The purpose of this brake is to apply a stopping force on the drum as close to the drum as is practicable so that no substantial risk occurs from a failure of some drive element between the brake and the drum. Furthermore, the input shaft 31 to the detector could also be from any location in the drive train between the motor and drum, which location is at the desired point to be monitored. Preferably, this location, however, will be at or close to the drum.

A brake actuator mechanism 15 in Fig. 1 includes a lever 16a keyed to an extension 29a of shaft 29. The lever is coupled to the free end of the brake band 14 such that rotation of the lever 16a in either direction will set the band brake. The lever 16a is provided with a notch 16b in which is inserted a spring-centered cog 17. A clutch 32 couples shaft 29 to its extension 29a. A clutch throw out bar 33 follows a cam 34 on the drum to decouple the shaft extension 29a once each rotation of the drum. If the shaft 29 rotates through some predetermined maximum angle, as, for example, because of differences in the speed reductions between the motor and drum and the speed reductions between the motor and drum and detector 20 or because of creep in the drive train, lever 16a will rotate toward the maximum angle, but the clutch 32 will decouple the lever 16a before it rotates far enough for the cog to leave the notch 16b. Thus each time the clutch is decoupled at each complete revolution of the drum, the lever is returned to its centered position by the cog 17. In the event of a failure or load hang-up, etc., however, the shaft 29 will rotate at a high velocity and the drum will be stopped before one or perhaps at the most two more revolutions of the drum are possible.

The torque-limiting device 6 generally has a driven gear 40 which is driven by one of the upper stages of the gear reduction in the gear case 5. The gear 40 is provided with clutch facing 39 which is splined, as at 41, to a shaft 42. A spring 37 pushes a pressure plate 38 against the clutch facing, thus releasably holding the gear 40 in driving engagement with the shaft 42. A pinion gear 44 is fixed to the shaft 42 by a key 45. By adjusting the position of the spring holder 36, a desired torque can be carried between the clutch facing and the driven gear 40. If an overload occurs, such as by excessive load, load hang-up,

two-blocking, a jamming in the drive train, or the like, the high-speed kinetic energy upstream of the driven gear 40 will be dissipated as heat in the clutch facing and the downstream drive components from pinion gear 44 to the drum will be stopped. Because of this safety feature of the torque-limiting device, however, there may be a certain minimum amount of creep or relative rotational movement between the gear 40 and the shaft 36 so that there may be slight variations between the input shaft 30 and the drum input shaft 31 in the differential detector 20.

In one embodiment, a centrifugal clutch 47 (Fig. 5) is provided to decouple input shaft 30 from motor shaft 2a when the motor shaft rotates at an overspeed above some predetermined percentage of its normal driven speed. That is, if some failure occurs which causes the motor to rotate beyond its set speed, the shaft 30 will become decoupled and stop, thus providing a variation between the relative velocities of shaft 30 and shaft 31 to provide a rotational output to output shaft 29 and set the brake. It is important in the differential assembly that the differential rotation can not drive the input shafts backwards, and thus there are provided drag mechanisms on each of the input shafts to assure that only the output shaft is rotated when one of the input shafts changes its velocity relative to the other. Continuous friction drags could be provided on each of the shafts for this purpose, or the inputs could be through worm gear drives; but in the preferred embodiment, the shafts are broken into two sections, namely, an external section 31e and an internal section 31i and an external section 30e and an internal section 30i. The internal and external sections of each shaft are joined by a conventional one-way clutch or "NO BAK" clutching device 21. These devices are positioned so that the external section 30e, when driving in either direction, will freely rotate the internal section 30i. Likewise, the clutch on the drum input shaft is positioned so that when external section 31e is providing a driving input, the internal section 31i will freely rotate. The converse is not true, however. That is, if at any time, one of the internal sections tries to drive the external section of that input shaft, the clutch will lock up so that the internal section cannot rotate. This provides a unique and more positive clutching or drag device for the input shafts to assure that when there is a change in velocity relative to the other input shaft, this change cannot be transmitted backwards to rotation of the other input shaft, but rather must be converted immediately and in all cases to a rotational output of the output shaft 29.

An overspeed clutch 47 is provided in a preferred embodiment. It is advantageous to employ a mechanical overspeed clutch having a clutch friction plate 48 keyed to shaft 2a and an opposed friction and pressure plate 49 keyed to a separate stub shaft 50 which drives the right angle gear box 19. A spring 52 is compressed by centrifugal governor weights 54 to hold the friction plates in driving engagement. When shaft 2a rotates

rapidly, as in an overspeed condition, the weights 54 swing outwardly and spring 52 is released, thereby allowing plates 48 and 49 to slip relative to one another. Once shaft 50 is released from shaft 2a, the detector signals the out-of-sync condition and the brake 14 is set.

Control systems for high-performance hoists are sometimes designed to sense the lifted load and to command motor 3 to operate at higher than full-load rated speed when handling a lighter load. This may be as high as 300 percent when operating under such no-load condition. Conventional overspeed drives are generally set in this no-load case to cut out at more than 300 percent full-load speed. This reduces the safety when handling a full load.

Thus, if desired, the clutch 47 can also be made load-sensitive. To release the friction discs 48 and 49 sooner or at a lower speed, a bell crank 56 is threaded in a nut 58 such that when screwed away from spring 52, the weights 54 will have less pressure on them and will open to release the discs 48 and 49 sooner or at a lower overspeed. If the bell crank is screwed in the opposite direction, higher overspeed can occur before the clutch plates are separated.

Motion of the bell crank 56 is provided by a line 60 coupled to a pivoted arm 62 that is balanced by a calibrated spring 64. The drum line 70 is reaved about a travelling block 72 and thence to a sheave 73 on arm 62. As the load is increased, arm 62 is lowered, thus moving bell crank 56.

While the detector 20 can signal an electrical shutoff or brake-setting device, it advantageously preferably signals or triggers a mechanical brake actuator. In the embodiment of Fig. 1, the detector can itself apply the brake. Two forms of triggering devices for setting the brake 14 are illustrated in Figs. 7 and 8. It is common to both these triggering devices that a large spring force can be applied to set the brake, but a small trigger release force is all that is necessary to release the spring.

In Fig. 7, the brake band 14 is set by a spring 74 having a large spring force, as is necessary for high-load capacity drums. A lever 75 is engaged by a trigger 76 which holds the spring in a cocked position. The trigger 76 is locked by a conventional trigger release cam 78. A solenoid 80 having an extendible arm 81 pivotally mounts one end of the cam 78. The cam is also pivoted at 83 and has an end 84 that abuts the trigger 76. A spring 89 urges the cam 78 into the phantom-line position to disengage from the trigger 76. When solenoid 80 is energized, the trigger release cam is in the solid-line position. Thus it is apparent that the small, easily controlled spring 89 is all that must be overcome to hold the large spring 74 in the cocked position.

To reset the trigger and spring 74, a relatively slow-speed rotary screw drive 90 moves the trigger, solenoid, and trigger release to the left. The trigger strikes a cam 92 that rotates the trigger counterclockwise, and the solenoid is energized to again hold the trigger in the cocked

position. Movement of the screw to shift the trigger to the right then reengages the lever 75 and recompresses the spring 74. Since the spring can be compressed slowly, the highly leveraged screw drive is easily able to overcome very large spring forces.

Fig. 8 illustrates a mechanical trigger release. In this embodiment, the cam can be electrically de-energized without having to set the brake 14, which is a disadvantage in the embodiment of Fig. 7. In this preferred embodiment, the lever 16a (Fig. 1), rather than being coupled directly to the brake band 14, is coupled to an elongated cable 94 that is connected to the trigger release cam 78 by a lost-motion slot 95. As the lever 16a rotates in a out-of-sync condition, the cable 94 is pulled, pivoting trigger release cam 78 into the phantom-line position to release trigger 76 in the same manner as in Fig. 7. Resetting of the spring 74, trigger 76, and cam 78 is similar to the above description of Fig. 7.

Fig. 3A illustrates a modification of the detector 20 capable of providing a signal for setting a brake actuator. In this embodiment, the detector output shaft 29 is provided with a flyball governor 97 that meshes with a rack 98 slidably mounted in the shaft 29. As the ball lever swing out from an out-of-sync condition, teeth on the levers meshing with the rack extend the rack. The rack engages a normally closed switch 99 to open the switch and de-energize solenoid 80, for example, to set the brake.

If desired, a normally energized electric clutch 100 can be added to any of the embodiments to decouple the motor shaft from the detector for setting the brake automatically if an electrical power failure occurs. Furthermore, this clutch or the overspeed clutch could also be placed on the drum side of the input to the differential detector.

The operation of the various embodiments of the safety system will now be described. During normal operation, such as with the motor shaft 2a being rotated at approximately 1200  $\mu$ /min the drum speed will be reduced to approximately 2.4  $\mu$ /min at the drum shaft 11a. The motor shaft at its 1200  $\mu$ /min is then coupled through the centrifugal clutch 47 and right angle/gear reducer drive 19 to the differential detector assembly 20 via the input shaft 30. Similarly, the 2.4  $\mu$ /min rotation of the drum shaft is coupled via right angle drive 18 to provide the same  $\mu$ /min input to the input shaft 31. It should be understood that these gear reductions do not have to be exact so long as they are proportionate, and the gear reduction, which is approximately 2:1 within the differential drive assembly, is sized accordingly. The desired result is that shaft 31 and shaft 30, when the hoist is operating either in the lowering or hoisting mode, provide substantially the same angular velocities to the differential gear 27a so that output shaft 29 rotates not at all or perhaps rotates one way or another at a very low rate, depending on the amount of slippage, variance in gear reductions, or creep in the system. If there is an overload, a load hang-up, a two-blocking or any failure in a drive

component such that the drum tries to stop, the torque-limiting device will dissipate the high-speed kinetic energy in the motor and upstream drive components, and the input shaft 31 will slow down or stop immediately, thus providing a variation between the angular velocities of the input shaft 31 and the input shaft 30. The motor input shaft will then cause the output shaft 29 to rotate rapidly; for example, at about 600  $\mu$ /min, since no slippage or back rotation can occur due to the clutch or drag 21 on the drum input shaft. Rotation of the output shaft 29 will immediately rotate the ball governor 97 or rotate the lever 16a, and the force applied by this rotation will be used either as a signaling device, as in Figs. 7 and 8, to set the brake, or, in a totally mechanical system, as in Fig. 1, to directly tighten the band brake. Should the motor shaft 2a rotate above its rated speed, such as where the controller may fail and allow the motor to drive the hoist too rapidly, the clutch 47 will disengage the motor shaft from the detector, stopping the input shaft 30 and providing an out-of-sync rotation of shaft 29. Similarly, if either the input shaft 30 or the input shaft 31 of the differential assembly should fail or any component in these inputs to the differential assembly should fail, the shaft 29 again will be rotated to set the brake.

In Fig. 9, in a preferred form of the invention a motor M is drivingly coupled to a drum D by a conventional power transmission main drive train including a gear reduction unit 210 and a passive energy absorption device 212.

The opposite end of the motor M is connected through a secondary drive train to an input shaft 214 of an out-of-sync detector 216 (Fig. 11). The motor M is joined to the input shaft 214 by an electric clutch 218 and right-angle drive element 219. The motor M is provided with an electrically controlled operational brake 220, which is set when electrically de-energized, i.e., in the absence of electricity. The electrical clutch 218 is employed between the motor M and the motor input shaft 214 for overspeed protection. For this purpose, the clutch 218 is drivingly coupled between the input shaft 214 and the motor when the electric motor is being energized, but becomes decoupled when the electricity is removed from the motor or during a total electrical blackout. The clutch also will become decoupled when the drum's rotational velocity exceeds a predetermined value, as in a hazard condition of the type in which the motor controller directs the motor to run at an unsafe overspeed condition. "Overspeed," as used herein, is a well known term in the art to signify a condition when, for the particular hoisting system, the motor is running at an excessive speed, for example, 150 and 200 percent of its normal operating speed.

A second input shaft 222 to the detector 216 is coupled with the drum D. The drum is provided with a safety brake 224. The brake is set by a brake actuator 226 which receives a signal from the detector 216 of a failure or hazard condition, which signal is transmitted via a mechanical cable 227.

The detector 216 is provided with a gear reduction 228 which couples the drum shaft 222 to a differential gear assembly 230 via a NO-BAK coupling 229 which will drivingly couple or transmit motion from the input shaft 222 into the differential 230 in either rotational direction, but will lock up and not transmit motion in the reverse direction, that is, from the differential assembly 230 back to the input shaft 222.

The differential assembly 230 contains a set of bevel gears 236—239 which are freely rotatable on spindles that are keyed to a housing 240. The housing is keyed to a spindle 242, whereas gears 236 and 238 are freely rotatable on spindle 242. Clockwise rotation in the direction of arrow 243 of the gear 236, with the housing held stationary, will result in counterclockwise rotation of gear 238 in the direction of arrow 244. If the entire housing rotates in a clockwise direction, as shown by arrow 245, then gear 238 will rotate in a clockwise direction. (The rotational directions for purposes of this description will be viewed arbitrarily as looking in the direction of arrows 14—14 of Fig. 11, and also are arbitrarily shown as in the load lowering mode).

Gear 238 is keyed to an output shaft 249 on which is mounted to a one-way sprag clutch 250. The shaft 249 will freewheel in the counterclockwise direction of rotation in the sprag clutch 250, but in the opposite direction will lock up with the clutch, causing rotation of the clutch.

The input to the differential housing 240 is through the NO-BAK clutch 229. The second input to the differential assembly is through worm gear 251, driven by a screw shaft 252. Worm gear 251 is keyed to gear 236. Shaft 214 from motor M is connected to shaft 252 through a slip correction device 254 (Fig. 15). The slip correction device 254 includes a first set of gears 256 and 257 having diametrical ratios which produce a slower rotational velocity in shaft 252 than is in shaft 214. Gear 257 is coupled to shaft 252 through a friction slip clutch 258. A second set of gears 260 and 261 result in a rotational velocity of shaft 252 which is greater than the rotational velocity of shaft 214. Gear 260 is coupled to shaft 214 through a one-way sprag clutch 263. The result of this gearing and clutching arrangement is that when shaft 214 is turning counterclockwise (looking left in Fig. 15), for example, the drive will be through gears 256 and 257 to reduce the velocity of shaft 252. Gear 260 will freewheel on shaft 214. When shaft 214 is rotated clockwise, sprag clutch 263 locks gear 260, resulting in shaft 252 being rotated at a higher velocity than shaft 214, with gear 257 slipping in clutch 258. The purpose of this slip correction device is to provide an input direction and velocity to the differential 230 which are different than those from the input from the NO-BAK 229 from the drum (assuming that the gear reductions between the drum and its differential input and the motor and the differential input through gear 251 are at or about the same reductions) such that gear 238 will always result in a net unidirectional rotation output in the

counterclockwise direction. With the slip correction device, the counterclockwise component added to the gear 238 allows the output shaft 249 to rotate in a freewheeling condition; and should limited slip occur between the motor and the drum due to a passive energy absorption device, such as device 212, then the slip will be less than that required to trig the brake and the added component of velocity added to the gear 238 and will null out the slip during operation of the hoist. As a further example, assume that the motor is rotated in a direction to rotate the drum in a raising mode. For this description, assume that when raising, shaft 252 will be made to turn slower than shaft 214. Also assume the teeth angles between screw 252 and gear 251 are such that the input to the differential through worm gear 251 will be faster but in the same direction as the input to the differential through NO-BAK 229. Assume these directions are counterclockwise. The net output will result in rotation of shaft 249 in the counterclockwise direction.

When lowering the load, sprag clutch 263 becomes engaged and shaft 252 is rotated at a greater speed than shaft 214. This results in an input from worm gear 251 to the differential assembly, which is greater than the input from NO-BAK 229. Since the direction of rotation of NO-BAK 229 is now in the opposite or clockwise direction, as shown by arrow 243, however, the net result will again produce an output rotation of shaft 249 again in the counterclockwise, freewheeling direction.

Various examples of the operation for detection of a hazard or failure will now be given. As a first example, consider the drive between the motor and the drum via gear reduction 210 is broken while the drum is being rotated in a raising mode. The drum will immediately reverse to a lowering mode, which will reverse the direction of the NO-BAK 229 but will not reverse the motor input direction. This will reverse the direction of the housing 240, immediately causing a clockwise rotational direction to shaft 249 to produce a brake-set signal.

As a second example, consider two-blocking or load hang-up while the load is being raised. In this mode, the drum stops rotating; this stops rotation of the differential housing 240. During normal operation in the raising direction, worm gear 251 was being rotated counterclockwise at a speed slower than the housing 240, resulting in a net counterclockwise rotation of shaft 249. However, when the NO-BAK 229 becomes stopped, then gear 238 is driven clockwise, which causes shaft 249 to engage with clutch 250 and produce a brake-set output.

As another example, consider a failure in the detector itself, for example, between the NO-BAK 229 and the input shaft 222 while raising a load. Since the NO-BAK transmits motion only when powered by the shaft 222, the NO-BAK will lock up upon failure of that drive. The gear 238 will be moved in a clockwise direction, locking up clutch 250 to produce a brake-set.

As another type of failure, consider a failure in the detector itself when raising the load, such as by a failure of shaft 252. As is well known, a worm gear cannot backdrive the screw shaft 252. This locks up the input from the motor side of the differential, and the gear 238 will be driven in a clockwise direction to produce a brake-set signal as soon as the drum is reversed to the lowering mode.

In the case of overspeed in the down or lowering direction the clutch 218 will decouple, again stopping the worm gear 251. The drum input will be the direction of arrow 245, causing a clockwise direction of shaft 249, and will set the brake.

The signal or clockwise movement of sprag clutch 250 is provided to trigger release of the brake in a manner which allows the trigger to be reset without manual intervention. This is an advantage when testing an installed system as well as to reset the brake during inadvertent trips or when the brake has been set intentionally by a failure or by the operator of the hoist. This brake-actuating mechanism is best shown in Figs. 9, 10 and 11. In the embodiment illustrated, the band brake 224 is connected at its free end to a bell crank 270. This bell crank is connected to a trigger mechanism 272 of a type similar to that shown in Figs. 7 and 8. The trigger mechanism 272 employs a catch 274 having a cam surface 276 and a lower cam surface 278. A latch 280 has an abutment end 282. The brake is set by a large force-applying compression spring, shown schematically as 284. The brake is retracted or reset by a conventional air bag or cylinder and piston 286. The air bag is such that it is energized to compress the spring 284 and loosen the brake band. The end 282 of the latch 280 engages the cam surface 276. The latch is raised by engagement of a boss 288 on the backside of the catch. The arrangement of the cams 276 and 278 are such that they will try to rotate latch 280 clockwise in a brake-releasing condition, but so long as the latch 280 is held in the upright condition, the catch 274 cannot move. Thus, once latch 280 is held in the raised position, catch 274 cannot move and the air can be vented from the air bags 286. As a typical example, the force by the spring 284 will cause approximately 0,068 tons of force to be required to hold the latch 280 in the raised position. This force is to be contrasted with the cocking force of several thousand pounds (2,72 tons, for example) that will be placed in the spring 284 by the air bags as is commonly necessary for setting this type of band brake. The advantage, of course, as with the embodiment of Figs. 1—8, is that a very quick acting, small restraining force can be used to trigger or release a much larger brake-applying force.

The upper end of latch 280 is coupled by the cable 227 to a slide 292. The slide is held in a raised position by a smaller force compression spring 294 of perhaps 10 to 15 Kp force. The slide 292 is carried in a track 295. The slide is held up by a roller 296 which is carried on a wedge ring 298.

The wedge ring is rotated counterclockwise by a reset cable 300, which is coupled to a tension spring 302. Due to the downward pull by cable 227, wedges 306 on the wedge ring lock into a wedge carrier 307 that is fixed to the sprag clutch 250 in the position shown in solid lines in Fig. 14 (the operational position). In this condition, of course, the output shaft 249 from the differential will either be stationary or rotating in the counterclockwise direction so that the wedge ring freely rotates on the shaft 249.

When an output signal to set the brake is produced by clockwise rotation of the output shaft 249, the sprag clutch 250 locks up and rotates clockwise. This clockwise rotation of the sprag clutch rotates the wedge ring 298 in the clockwise direction into the phantom line position of Fig. 14. This removes roller 296 from under the slide 292, allowing the force in cable 290 to be released. Latch 280 is rotated clockwise, releasing catch 274, allowing spring 284 to set the brake.

To reset the brake, air is applied to the air bags 286, moving the catch 274 to the left. The compression spring 294 raises slide 292. The reset spring 302 pulls wedge ring 298 counterclockwise to reset the roller beneath the slide. Wedge ring 298 is slightly oversized on its shaft such that its wedges 306 release from the wedge carrier 307 when the downward load, caused by cable 227, is released from the wedge ring. This allows the wedge ring to be rotated into its reset condition or a counterclockwise direction. The resetting of the slide 292 and the wedge ring occurs prior to the time the air bags have retracted catch 274 to its further left reset position. Thus the wedge ring remains free on clutch 250 until the latch 280 is caused to rotate slightly clockwise by the spring 284 after the air bags have been vented. Then the small force is again applied to the slide to hold the wedge ring tight against the carrier and thus the sprag clutch 250.

Figs. 16 and 17 illustrate a simplified embodiment usable with slip correction device 254. The differential detector of Fig. 16 employs a motor input shaft 252, from the motor and a hollow input shaft 242 from the drum. A collar 320 is pivotally connected to a bell crank 321. A spring 322 holds the bell crank in a counterclockwise direction and is equivalent to the spring 294 of Fig. 14. Cable 227 is attached to the right-hand end of the bell crank. The collar 320 is keyed within an axial slot 324 in the shaft 252. A roller 326 is rotatably mounted on the collar. The roller is held against a cam surface 330 which forms part of a cam block 332. The cam block is joined to shaft 242 by a friction clutch 333 and a one way clutch 338, positioned by needle thrust bearings 334 and radial bearings 337. The cam block freewheels on the shaft 242 in one direction of rotation, but is joined to the shaft 242 in locked arrangement by the one-way sprag clutch 338 and friction clutch 333. The cam block moves with the shaft 242 and is held in the direction of the arrow 340 by a coil spring 342. The cable 227 produces approximately a 0,068 tons pull on the collar,

urging it axially to the right in Fig. 16, energizing the friction clutch 333. As the cam block 332 is moved in the direction opposite to the arrow 340, the roller moves off the cam surface 330 and is pulled by the force in cable 227 into a slot 344. This allows the bell crank to rotate clockwise, releasing the trigger to set the brake in the manner shown for the embodiment of Figs. 9—15. The friction clutch is also de-energized, allowing cam block 332 to freely rotate with shaft 252. Resetting of the brake is essentially the same as in the preferred embodiment, with the spring 322 moving the collar back to the left. This movement withdraws the roller 326 from the slot 344 and allows spring 342 to reset the cam block in the direction of the arrow 340. The roller 326 then is precluded from moving to the right in Fig. 16, which then holds the bell crank in a fixed position when the 0,068 tons load is again placed on the cable 227 by the trigger mechanism.

The device senses a differential movement between the shaft 242 and the shaft 252 in a predetermined direction to cause the sprag clutch 338 to rotate clockwise in the embodiment illustrated to rotate the cam block and release the collar 320. The normal operating rotations will produce either a zero differential rotation between shafts 252 and 242, if there are perfectly matched gear reductions between the motor and drum and the detector and if there is no relative slippage in the drive trains. If these conditions do not exist, then a unidirectional rotational differential exists. In either case, a failure or hazard condition will cause an output rotation in a predetermined direction and opposite said unidirectional rotation, if such is in effect, to set the safety brake.

Various examples of operation of the embodiment of Figures 16 and 17 will now be explained. In a first example, assuming the motor input shaft 252 is turning clockwise and the drum input shaft 242 is also turning clockwise, but because of the slip correction, the shaft 242 will be traveling clockwise slower than will shaft 252. In this condition, the sprag clutch is freewheeling. The shaft 242 runs slower because the slip correction device is employed and thus the shaft is equivalent to shaft 252 of Fig. 15. There are also provided a NO-BAK 350 locking shaft 242 to the frame and a NO-BAK 360 locking shaft 252 to the frame. Thus NO-BAK 350 will allow shaft 242 to drive clockwise or counterclockwise, but will not allow shaft 242 to be driven by shaft 252. Similarly, NO-BAK 360 will allow shaft 252 to be driven from the motor but will not allow shaft 252 to backdrive through NO-BAK 360. Now assume that the motor shaft fails, creating a failure condition. NO-BAK 360 locks shaft 252, shaft 242 thus turns clockwise faster than shaft 252, clutch 338 engages, and the cam block 332 gets rotated clockwise, causing roller 326 to enter slot 344 and set the brake.

As a second example, consider raising the load, with the motor input shaft 252 turning counterclockwise and the drum shaft 242 turning counter-

clockwise. The slip correction device, however, will cause the shaft 242 to rotate faster than shaft 252 in the counterclockwise direction such that the sprag clutch still freewheels. Again assume that there is a discontinuity in the main drive between the motor and the drum. Shaft 252 will lock up. The drum, since it was raising a load until the drive failure, will immediately reverse direction and begin to lower the load. Thus the shaft 242 will reverse to a clockwise direction, causing the cam block 332 to move clockwise and set the brake.

A further advantage is that whenever the operational brake is set, any slippage or failure in the operational brake will also be detected and set the safety brake. This overcomes a major problem in hoists because the brakes can otherwise wear severely without detection by the operator and allow the load to slip.

As another example, assume a two-blocking hazard during raising. Shaft 252 will be turning counterclockwise; shaft 242 will be turning counterclockwise. Shaft 242 will stop when the cable sheaves become two-blocked ("two-blocking" occurs when the traveling blocks engage the stationary blocks). Since shaft 242 is stopped and the shaft 252 continues to turn counterclockwise, the roller 326 moves off the cam surface 330 into the slot and sets the brake.

### Claims

1. A safety system in a load-carrying hoist in which there is defined a last upstream and load-carrying component and a last downstream load-carrying component and an input motor (3,M) with a motor shaft, a power transmission main drive (5, 210, 212) operatively coupled to the motor (3,M), a drum (11,D) and a safety brake (14, 224) drivingly coupled to the drum, a safety brake actuator (15, 226) for applying said safety brake responsible for a rotational output from an out-of-sync detector (20, 216), said mechanical out-of-sync detector comprising a monitoring secondary drive train having a first input shaft (30, 214) drivingly coupled to an upstream load-carrying component, a second input shaft (31, 222) drivingly coupled to a downstream load-carrying component, said out-of-sync detector being provided for detecting a predetermined variation in relative speed or direction between said two input shafts (30, 31, 214, 222) and producing a safety brake-setting rotational output to set said safety brake, said hoist- and safety-system having an error between the relative rotations of said components and/or within the gear ratios of the hoist and the safety-system, characterized in that said safety-system comprises mechanically driven mechanical error correcting means (16b, 17, 32, 254) for producing a predetermined limited rotational movement in a rotational direction opposite to the safety-brake-setting-rotation of said out-of-sync detector (20, 216), said correcting means being coupled to an output drive train between the out-of-sync detector and the brake

actuator (15, 226) to superimpose the limited rotational movement to the safety-brake-setting-rotational output in order to compensate for the safety-brake-setting-rotational output occurring from said error.

2. The safety system of claim 1, characterized in that said out-of-sync detector (216) for detecting a predetermined variation in relative speed or direction between said components and the input shafts (214, 222, 242, 252) contains means for producing a first unidirectional rotational output to set said safety brake, said error correcting means (254) producing a second predetermined unidirectional rotational output in a direction opposite to and of a velocity greater than said first unidirectional rotational output, so as to compensate that part of said unidirectional brake-setting rotational output which results from said error in said hoist and safety system.

3. The safety system of claim 2, characterized by said out-of-sync detector (216) including a differential assembly (230) having meshing differential gears (256—239) and a differential carrier (240), a safety brake-actuating output shaft (239) being drivingly coupled to said differential assembly (230); and one-way clutch means (250) drivingly coupling said safety brake-actuating output shaft (249) to said safety brake actuator only when said output shaft (249) rotates in said safety brake-setting direction.

4. The safety system of claim 3, characterized by said detector including coaxially aligned first and second input shafts (252, 242), a collar (320) on said first input shaft (252) axially movable between a safety brake-set position and a safety brake-off position, linkage means (321, 227) coupling said collar (320) to said safety brake actuator, stop means (330) on said second input shaft (242) and operative in one unidirectional differential rotation between said input shaft to hold said collar (320) in said brake-off position but in the opposite differential direction of rotation, releasing said collar (320) to move into said safety brake-setting position and to cause the linkage means (321, 227) to produce said unidirectional safety brake-setting rotational output.

5. The safety system of any of claims 2 to 4, characterized by said brake actuator including a unidirectional, rotation transmitting, one-way clutch (250) on said output shaft (249), a wedge ring (298) releasably, drivingly coupled to said one-way clutch (250), means biasing said wedge ring (298) towards a safety brake-hold position, means on said wedge ring for holding the safety brake, said safety brake including trigger means (272) having a latch (280) and a catch (274), said catch (274) having a large force (284) biasing the catch (274) into brake-set position, said latch (280) having sufficient leverage to hold said catch (274) against said large force by a small force when in a brake-off position but movable upon release of said small force to move into a safety brake-set position, releasing said catch to move into its large force safety brake-set position, and linkage means (227, 292, 295, 296) coupling said latch to

said wedge ring (298), brake holding means for holding said latch in said brake-off position, whereby said unidirectional safety brake-setting output of said output shaft (249) rotates said wedge ring (298) to release said linkage means (227, 292, 295, 296) and set the safety brake.

6. The safety system of claim 2, characterized by said error being the result of variations in speed reductions in the main drive train between the motor shaft (214) and the drum D and the secondary drive train that also couples the motor shaft (214), drum (D) and detector (216), said correcting means (254) including means for producing said second unidirectional rotation opposite the first unidirectional safety brake-setting rotational output direction in excess of such speed reduction variations.

7. The safety system of claim 2, characterized by said main drive train having a member (212) which will cause rotational slippage error between drum and the motor shaft.

8. The safety system of claim 1, characterized by said out-of-sync detector (20) and said safety brake actuator (15) being combined for tightening the safety brake (14) on said drum (11), or drum-operating element, and the rotational output of said output shaft (29) caused by said variation in relative direction or speed of said input shafts providing the force for moving said brake actuator (15) for tightening the safety brake (14).

9. The safety system of claims 1 or 8, characterized by said first and second input shafts (30, 31) having minor and major relative angular velocity differences and including said correcting means (16b, 17, 32) for periodically restoring the output shaft (29) to an in-sync condition to compensate for minor relative angular velocity differences between said input shafts (30, 31).

10. The safety system of claim 9, characterized by said output shaft (29) having inner and outer ends (29a), said correcting means (16b, 17, 32) including a disconnect clutch (32) on said output shaft (29) operable to disconnect the outer end (29a) from said output shaft inner end, means (16a, 16b) for centering said outer end (29a) of said output shaft (29) when disconnected from said inner end, and means (33) responsible to rotation of said driving element (11) for periodically disconnecting said clutch (32).

11. The safety system of claim 1, characterized by an overspeed detection clutch (47) for disconnecting one of said input shafts (30, 31) during an excessive threshold overspeed condition to initiate said relative angular velocity difference between said first and second input shafts (30, 31).

12. The safety system of claim 11, characterized by including load magnitude responsive means (56) to correlate load magnitude to motor speed for varying the threshold excessive overspeed condition dependent upon load magnitude.

13. The system of claim 1, characterized by including an electrically activated driving clutch (100) operable when de-energized to disconnect an input shaft (30) from its respective last

upstream or last downstream load-carrying component (2a) to initiate said relative angular velocity difference between said first and second input shafts (30, 31) when an electrical failure occurs.

14. The system of claim 1, characterized by said safety brake actuator including a high-force holding device and a low-force anchor (76) releasably holding said high-force holding device in a cocked condition, said safety brake actuator including a large force spring (74) urging said safety brake in an engaged position, means coupled to said high-force holding device for holding said spring (74) in said cocked condition against movement of the safety brake (14) into an engaged condition, means (80) responsive to a hazard condition for moving said low-force anchor (76) to release said high-force holding device for setting said safety brake, and means (90) for restoring said spring (74), low-force anchor and high-force holding device to their initial cocked condition and release said safety brake (14).

15. The system of claim 1, characterized by each of said input shafts (30, 31, 222, 214) including means (21, 229) for restricting movement of said input shaft when attempted to be back-driven by said means for detecting said variation in relative speed or direction between said two input shafts, and thus the output is always caused to rotate during differential angular velocities between said input shafts rather than back-driving an input shaft.

#### Patentansprüche

1. Sicherheitssystem in einer Lasthebevorrichtung, wobei die Lasthebevorrichtung einen Antriebsmotor (3, M) mit einer Motorwelle, ein mit dem Motor (3, M) in Arbeitsverbindung stehendes Hauptgetriebe (5, 210, 212) zur Kraftübertragung, eine Windentrommel (11, D), eine mit der Windentrommel in Antriebsverbindung stehende Sicherheitsbremse (14, 224), eine Betätigungsvorrichtung (15, 226) zum Einrücken der Sicherheitsbremse (14, 224) und einen Drehzahl-Differenzdetektor (20, 216) aufweist, auf dessen Ausgangsdrehbewegung die Betätigungsvorrichtung (15, 226) anspricht, wobei ferner im Kraftübertragungsweg innerhalb der Lasthebevorrichtung eine letzte stromauf liegende, lasttragende Komponente und eine letzte strom abliegende, lasttragende Komponente vorgesehen sind, von denen die stromauf liegende Komponente mit einer ersten Eingangswelle (30, 214) eines sekundären Steuerantriebsstranges des Drehzahl-Differenzdetektors und die stromab liegende Komponente mit einer zweiten Eingangswelle (31, 222) des sekundären Steuerantriebsstranges des Drehzahl-Differenzdetektors bewegungsübertragend gekuppelt ist, wobei der Drehzahl-Differenzdetektor zum Erfassen einer vorbestimmten Abweichung der relativen Drehzahlen oder Drehrichtungen der beiden Eingangswellen (30, 31, 214, 222) ausgelegt ist, um bei Auftreten einer Abweichung eine die Sicherheitsbremse einrückende Aus-

gangsdrehbewegung zu erzeugen, und wobei in der Lasthebevorrichtung und im Sicherheitssystem zwischen den Drehzahlen der Komponenten und/oder innerhalb der Zahnradübersetzungen der Lasthebevorrichtung und des Sicherheitssystems ein Drehzahlübertragungsfehler vorliegt, dadurch gekennzeichnet, daß das Sicherheitssystem mechanisch angetriebene, mechanische Fehler-Korrigier-Mittel (16b, 17, 32, 254) zum Erzeugen einer vorbestimmten, begrenzten Drehbewegung aufweist, die in ihrer Drehrichtung der Ausgangsdrehbewegung des Drehzahl-Differenzdetektors (20, 216) zum Einrücken der Sicherheitsbremse (14, 224) entgegengesetzt ist, und daß die Fehler-Korrigier-Mittel mit einem Ausgangsantriebsstrang gekuppelt sind, der sich zwischen dem Drehzahl-Differenzdetektor und der Bremsbetätigungsvorrichtung (15, 226) erstreckt, um die begrenzte Drehbewegung der Korrigier-Mittel der Ausgangsdrehbewegung zum Einrücken der Sicherheitsbremse zu überlagern und die Ausgangsdrehbewegung des Drehzahl-Differenzdetektors, die auf den Drehzahlübertragungsfehler in der Lasthebevorrichtung und dem Sicherheitssystem zurückzuführen ist, zu kompensieren.

2. Sicherheitssystem nach Anspruch 1, dadurch gekennzeichnet, daß der Drehzahl-Differenzdetektor (216), mit dem eine vorbestimmte Abweichung in den Relativgeschwindigkeiten oder Drehrichtungen der Komponenten und der Eingangswellen (214, 222, 242, 252) feststellbar ist, Mittel enthält, mit denen eine erste in ausschließlich einer Drehrichtung verlaufende Ausgangsdrehbewegung erzeugbar ist, um die Sicherheitsbremse einzurücken, und daß mit den Fehler-Korrigier-Mitteln (254) eine zweite, vorbestimmte und ausschließlich in einer Drehrichtung gerichtete Ausgangsdrehbewegung erzeugbar ist, deren Drehrichtung der der ersten Ausgangsdrehbewegung entgegengesetzt ist und deren Geschwindigkeit größer als die der ersten Ausgangsdrehbewegung ist, so daß der Teil der die Sicherheitsbremse einrückenden Ausgangsdrehbewegung kompensiert wird, der von dem Fehler in der Lasthebevorrichtung und in dem Sicherheitssystem herrührt.

3. Sicherheitssystem nach Anspruch 2, dadurch gekennzeichnet, daß der Drehzahl-Differenzdetektor (216) eine Differential-Anordnung (230) mit kämmenden Differentialzahnradern (256—239) und einen Differentialträger (240) enthält, daß eine Ausgangswelle (249) zum Einrücken der Sicherheitsbremse mit der Differentialanordnung (230) bewegungsübertragend gekuppelt ist, und daß die Ausgangswelle (249) mit der Sicherheitsbremsen-Betätigungsvorrichtung über in nur einer Drehrichtung kuppelnde Kupplungsmittel (250) verbindbar ist, derart, daß die Sicherheitsbremsen-Betätigungsvorrichtung nur betätigt wird, wenn die Ausgangswelle (249) in der die Sicherheitsbremse einrückenden Drehrichtung gedreht wird.

4. Sicherheitssystem nach Anspruch 3, dadurch gekennzeichnet, daß der Drehzahl-Differenzdetektor koaxial ausgerichtete erste und zweite Ein-

gangswellen (252, 242) aufweist, daß auf der ersten Eingangswelle (252) eine axial zwischen einer Sicherheitsbremsen-Einrückstellung und einer Sicherheitsbremsen-Ausrückstellung hin- und herbewegliche Hülse (320) gelagert ist, daß die Hülse (320) mit der Sicherheitsbremsen-Betätigungsverrichtung durch Lenkermittel (321, 227) gekuppelt ist, daß auf der zweiten Eingangswelle (242) Anschlagmittel (330) vorgesehen sind, die in ausschließlich einer Drehrichtung bei einer Drehzahl-Differenz zwischen den beiden Eingangswellen betätigbar ist, um die Hülse (320) in der Sicherheitsbremsen-Ausrückstellung zu halten, und um bei entgegengesetzter Differenzdrehrichtung die Hülse (320) freizugeben und diese in ihre Sicherheitsbremsen-Einrückstellung zu bewegen, und über die Lenkermittel (321, 227) veranlaßt werden, die in ausschließlich einer Richtung wirkende Ausgangsdrehbewegung zum Einrücken der Sicherheitsbremse zu erzeugen.

5. Sicherheitssystem nach einem der Ansprüche 2 bis 4, dadurch gekennzeichnet, daß die Bremsbetätigungsverrichtung eine drehbewegungsübertragende und ausschließlich in einer Drehrichtung greifende Kupplung (250) auf der Ausgangswelle (249) aufweist, daß mit der Kupplung (250) ein Keilring (298) lösbar bewegungsübertragend gekuppelt ist, daß den Keilring (298) in Richtung auf eine die Sicherheitsbremse ausgerückt haltende Stellung beaufschlagende Mittel vorgesehen sind, daß an dem Keilring Mittel zum Halten der Sicherheitsbremse in ausgerücktem Zustand vorgesehen sind, und daß die Sicherheitsbremse Einrückmittel (272) mit einer Klinke (280) und einer Klinkenfangvorrichtung (274) aufweist, von denen die Klinkenfangvorrichtung (274) mit einer großen Kraft (284) beaufschlagt ist, die die Klinkenfangvorrichtung in Richtung auf die Einrückstellung der Sicherheitsbremse beaufschlagt, während die Klinke (280) mit einer ausreichenden Hebelwirkung versehen ist, um die Klinkenfangvorrichtung (274) gegen die große Kraft mittels einer kleinen Kraft abzufangen, solange sie die Sicherheitsbremsen-Ausrückstellung einnimmt, daß die Klinke jedoch nach Aufheben der kleinen Kraft in die Sicherheitsbremseneinrückstellung bewegbar ist und dabei die Klinkenfangvorrichtung freigibt, bis diese unter der großen Kraft in die Sicherheitsbremsen-Einrückstellung gelangt, daß Lenkermittel (227, 292, 295, 296) vorgesehen sind, die die Klinke mit dem Keilring (298) kuppeln, und daß Haltemittel zum Halten der Klinke in der Sicherheitsbremsen-Ausrückstellung vorgesehen sind, wobei mittels der Ausgangsdrehbewegung der Ausgangswelle (249) der Keilring (298) verdrehbar ist, um die Lenkermittel (227, 292, 295, 296) freizugeben und die Sicherheitsbremse einrücken zu lassen.

6. Sicherheitssystem nach Anspruch 2, dadurch gekennzeichnet, daß der Fehler das Resultat von Abweichungen in der Geschwindigkeitsuntersetzung im Hauptantriebsstrang zwischen der Motorwelle (214) und der Windentrommel (D) sowie dem sekundären Antriebsstrang ist, der die Motorwelle (214) und die Windentrommel (D) mit

dem Drehzahl-Differenzdetektor (216) verbindet, und daß die Korrigier-Mittel (254) Mittel enthalten, mit denen die zweite und nur in einer Drehrichtung gerichtete Drehbewegung entgegengesetzt zur ersten und nur in einer Drehrichtung auftretenden Ausgangsdrehbewegung zum Einrücken der Sicherheitsbremse erzeugbar ist, wobei die zweite Drehbewegung größer ist als die erste, auf die Abweichungen in der Geschwindigkeitsuntersetzung zurückzuführende Ausgangsdrehbewegung.

7. Sicherheitssystem nach Anspruch 2, dadurch gekennzeichnet, daß der Hauptantriebsstrang ein Glied (212) enthält, das zwischen der Windentrommel und der Motorwelle einen Drehschlupffehler erzeugt.

8. Sicherheitssystem nach Anspruch 1, dadurch gekennzeichnet, daß der Drehzahl-Differenzdetektor (20) und die Sicherheitsbremsen-Betätigungsverrichtung (15) miteinander baulich vereinigt sind, um die Sicherheitsbremse (14) an die Windentrommel (D) oder ein die Windentrommel antreibendes Element anzupressen, und daß die Ausgangsdrehbewegung der Ausgangswelle (29), die durch die Abweichung der Drehzahlen oder der Drehrichtungen der beiden Eingangswellen hervorgerufen wird, zum Erzeugen der Kraft der Betätigungsverrichtung (15) beim Anziehen der Sicherheitsbremse (14) benutzt wird.

9. Sicherheitssystem nach den Ansprüchen 1 oder 8, dadurch gekennzeichnet, daß die ersten und zweiten Eingangswellen (30, 31), minimale und maximale relative Winkelgeschwindigkeitsunterschiede aufweisen, und daß die Korrigier-Mittel (16b, 17, 32) zum periodischen Zurückstellen der Ausgangswelle (29) in eine synchronisierte Grundstellung dienen, um die minimalen relativen Winkelgeschwindigkeitsunterschiede zwischen den Eingangswellen (30, 31) zu kompensieren.

10. Sicherheitssystem nach Anspruch 9, dadurch gekennzeichnet, daß die Ausgangswelle (29) innere und äußere Enden (29a) aufweist, daß die Korrigier-Mittel (16b, 17, 32) eine Trennkupplung (32) auf der Ausgangswelle (29) enthalten, mit der das äußere Ende (29a) der Ausgangswelle von deren inneren Ende trennbar ist, daß Mittel (16a, 16b) vorgesehen sind, um das äußere Ende (29a) der Ausgangswelle (29) in einer zentrierten Position zu halten sobald das äußere Ende vom inneren Ende abgetrennt ist, und daß Mittel (33) vorgesehen sind, die auf die Windentrommeldrehung ansprechen, um die Kupplung (32) periodisch auszurücken.

11. Sicherheitssystem nach Anspruch 1, dadurch gekennzeichnet, daß eine Kupplung (47) zum Feststellen einer überhöhten Drehgeschwindigkeit vorgesehen ist, mit der eine der Eingangswellen (30, 31) während einer überhöhten Drehzahlbedingung abtrennbar ist, um die relative Winkelgeschwindigkeitsdifferenz zwischen der ersten und der zweiten Eingangswelle (30, 31) zu initiieren.

12. Sicherheitssystem nach Anspruch 11, dadurch gekennzeichnet, daß auf das Ausmaß

der Last ansprechende Mittel (56) vorgesehen sind, um das Ausmaß der Last mit der Motordrehzahl in Übereinstimmung zu bringen, und eine Ansprechschwelle des Sicherheitssystems in Abhängigkeit von einem Zustand überhöhter Geschwindigkeit in Abhängigkeit vom Ausmaß der Last zu verändern.

13. Sicherheitssystem nach Anspruch 1, gekennzeichnet durch eine elektrisch betätigbare Antriebskupplung (100), die in stromlosem Zustand eine Antriebswelle (30) von der letzten stromauf oder stromab liegenden, lasttragenden Komponente (2a) abtrennt, um die relative Winkelgeschwindigkeitsdifferenz zwischen den beiden Eingangswellen (30, 31) des Drehzahl-Differenzdetektors zu initiieren, sobald ein Stromausfall eintritt.

14. Sicherheitssystem nach Anspruch 1, dadurch gekennzeichnet, daß die Betätigungsvorrichtung für die Sicherheitsbremse eine mit großer Kraft beaufschlagte Haltevorrichtung und eine mit geringer Kraft beaufschlagte Ankervorrichtung (76) enthält, mit der die mit hoher Kraft beaufschlagte Haltevorrichtung lösbar in einer verriegelten Stellung haltbar ist, daß die Betätigungsvorrichtung eine mit großer Kraft ausgelegte Feder (74) enthält, die die Sicherheitsbremse in ausgerückter Stellung hält, daß mit der mit großer Kraft beaufschlagten Haltevorrichtung Mittel gekuppelt sind, die die Feder (74) in der verriegelten Stellung entgegengesetzt zur Bewegung der Sicherheitsbremse (14) in die eingerückte Stellung halten, daß Mittel (80) vorgesehen sind, die auf eine Gefahren-Situation zum Bewegen der Verankerungsvorrichtung (76) ansprechen, um die mit großer Kraft beaufschlagte Haltevorrichtung zu lösen und die Sicherheitsbremse einzurücken, und daß Mittel (90) vorgesehen sind, um die Feder (74), die Ankervorrichtung und die Haltevorrichtung in ihre verriegelten Stellungen zurückzuführen und dabei die Sicherheitsbremse zu lösen.

15. Sicherheitsvorrichtung nach Anspruch 1, dadurch gekennzeichnet, daß jede Eingangswelle (30, 31, 222, 214) Mittel (21, 229) enthält, mit denen eine Bewegung der Eingangswellen unterdrückbar ist, sobald die Mittel zum Feststellen der Abweichung der relativen Drehgeschwindigkeiten oder Drehrichtungen der beiden Eingangswellen eine Rückdrehbewegung erzeugen, derart, daß die Ausgangsdrehbewegung zum Einrücken der Sicherheitsbremse ausschließlich bei Differenzen der Winkelgeschwindigkeiten der beiden Eingangswellen erzeugt wird, und nicht durch Zurückdrehen wenigstens einer Eingangswelle.

## Revendications

1. Système de sécurité pour engin de levage de charge dans lequel sont prévus un dernier organe d'amont porteur de charge et un dernier organe d'aval porteur de charge et un moteur d'entrée (3, M) à arbre moteur, une transmission principale de transmission d'énergie (5, 210, 212)

accouplée au moteur (3, M), un tambour (11) et un frein de sécurité (14, 224) accouplé au tambour, un organe de commande de frein de sécurité (15, 226) pour serrer ledit frein de sécurité en réponse à un signal de sortie rotatoire émanant d'un détecteur de désynchronisation (20, 216), ledit détecteur de désynchronisation mécanique comprenant un train de transmission secondaire de surveillance comportant un premier arbre d'entrée (30, 214) accouplé à un organe porteur de charge d'amont, un second arbre d'entrée (31, 222) accouplé à un organe porteur de charge d'aval, ledit détecteur de désynchronisation étant prévu pour détecter une variation déterminée de vitesse ou de direction relative entre les deux susdits arbres d'entrée (30, 31, 214, 222) et engendrer un signal de sortie rotatoire de serrage de frein de sécurité pour serrer ledit frein de sécurité, ledit ensemble engin de levage-système de sécurité, comportant un écart entre les rotations relatives desdits organes et/ou dans les rapports d'engrenage de l'engin de levage et du système de sécurité, caractérisé en ce que ledit système de sécurité comprend des moyens de correction d'écart mécanique entraînés mécaniquement (16b, 17, 32, 254) pour provoquer un mouvement de rotation limité déterminé dans un sens de rotation opposé à la rotation de serrage de frein de sécurité dudit détecteur de désynchronisation (20, 216) lesdits moyens de correction étant accouplés à un train de transmission de sortie interposé entre le détecteur de désynchronisation et l'organe de commande de frein (15, 226) pour superposer le mouvement de rotation limité au signal de sortie rotatoire de serrage de frein de sécurité afin de compenser le signal rotatoire de serrage de frein de sécurité découplant dudit écart.

2. Système de sécurité selon la revendication 1, caractérisé en ce que ledit détecteur de désynchronisation (216) pour la détection d'une variation déterminée de vitesse ou de direction relative entre lesdits organes et les arbres d'entrée (214, 222, 242, 252) comporte des moyens pour engendrer un premier signal de sortie rotatoire unidirectionnel afin de serrer ledit frein de sécurité, lesdits moyens de correction d'écart (254) engendrant un second signal de sortie rotatoire déterminé opposé par le sens et supérieur par la vitesse audit premier signal de sortie rotatoire unidirectionnel, du façon à compenser la partie dudit signal de sortie rotatoire de serrage de frein unidirectionnel qui découle dudit écart entre lesdits engins de levage et système de sécurité.

3. Système de sécurité selon la revendication 2, caractérisé en ce que ledit détecteur de désynchronisation (216) comporte un ensemble différentiel (230) présentant des éléments d'engrenage différentiel en prise (256—239) et un porte-satellites (240), un arbre de sortie de commande de frein de sécurité (239) étant accouplé audit ensemble différentiel (230); et un moyen d'embrayage unidirectionnel (250) accouplant ledit arbre de sortie de commande de frein de sécurité

(249) audit organe de commande de frein de sécurité seulement quand ledit arbre de sortie (249) tourne dans ledit sens de serrage de frein de sécurité.

4. Système de sécurité selon la revendication 3, caractérisé en ce que ledit détecteur comporte un premier et un second arbres d'entrée alignés coaxialement (252, 242), un collier (320) porté par ledit premier arbre d'entrée (252) mobile axialement entre une position de serrage de frein de sécurité et une position de desserrage de frein de sécurité, un tringlage (321, 227) accouplant ledit collier (320) audit organe de commande de frein de sécurité, des moyens de butée (330) prévus sur ledit second arbre d'entrée (242) et agissant lors d'une première rotation différentielle unidirectionnelle entre lesdits arbres d'entrée pour maintenir ledit collier (320) dans ladite position de desserrage de frein mais libérant, lors d'une rotation différentielle de sens opposé, ledit collier (320) pour qu'il puisse prendre ladite position de serrage de frein et faire engendrer par ledit tringlage (321, 227) ledit signal de sortie rotatoire unidirectionnel de serrage de frein de sécurité.

5. Système de sécurité selon l'une quelconque des revendications 2 à 4, caractérisé en ce que ledit organe de commande de frein comporte un embrayage unidirectionnel, de transmission de rotation (250) prévu sur ledit arbre de sortie (249), un anneau de calage (298) accouplé de manière amovible audit embrayage unidirectionnel (250), des moyens sollicitant ledit anneau de calage (298) vers une position de maintien de frein de sécurité, des moyens prévus sur ledit anneau de calage pour maintenir le frein de sécurité, ledit frein de sécurité comportant des moyens de déclenchement (272) présentant un loquet (280) et un arrêtoir (274), ledit arrêtoir (274) présentant une grande force (284) le sollicitant vers la position de serrage du frein, ledit loquet (280) exerçant une force de levier suffisante pour maintenir ledit arrêtoir (274) à l'encontre de ladite grande force par une petite force quand il occupe la position de desserrage du frein mais pouvant se déplacer quand ladite petite force cesse de s'exercer pour prendre une position de serrage de frein de sécurité, et un tringlage (227, 292, 295, 296) accouplant ledit loquet audit anneau de calage (298), des moyens de maintien de frein pour le maintien dudit loquet dans ladite position de desserrage de frein, de sorte que ledit signal de sortie unidirectionnel de serrage de frein dudit arbre de sortie (249) fait tourner ledit anneau de calage (298) pour interrompre l'action dudit tringlage (227, 292, 295, 296) et serrer le frein de sécurité.

6. Système de sécurité selon la revendication 2, caractérisé en ce que ledit écart est le résultat de variations de réductions de vitesse apparues dans le train transmission principal entre l'arbre de moteur (214) et le tambour D et le train de transmission secondaire qui accouple aussi l'arbre de moteur (214), le tambour (D) et le détecteur (216), lesdits moyens de correction (254) comportant des moyens pour engendrer ladite seconde

rotation unidirectionnelle à l'opposé du sens dudit premier signal de sortie rotatoire unidirectionnel de serrage de frein de sécurité en dépassement par rapport auxdites variations de réduction de vitesse.

7. Système de sécurité selon la revendication 2, caractérisé en ce que ledit train de transmission principal comporte un élément (212) de nature à provoquer un écart de rotation par patinage entre le tambour et l'arbre de moteur.

8. Système de sécurité selon la revendication 1, caractérisé en ce que ledit détecteur de désynchronisation (20) et ledit organe de commande de frein de sécurité (115) sont combinés pour serrer le frein de sécurité (14) sur ledit tambour (11), ou élément de commande de tambour, et le signal de sortie rotatoire dudit arbre de sortie (29) dû à ladite variation de direction ou de vitesse relative entre lesdits arbres d'entrée fournit la source assurant le déplacement dudit organe de commande de frein (15) pour serrer le frein de sécurité (14).

9. Système de sécurité selon les revendications 1 ou 8, caractérisé en ce que lesdits premier et second arbres d'entrée (30, 31) présentent des différences de vitesses angulaires relatives mineures et majeures et comportent lesdits moyens de correction (16b, 17, 32) pour remettre périodiquement l'arbre de sortie (29) en synchronisme afin de compenser des différences de vitesses angulaires relatives mineures entre lesdits arbres d'entrée (30, 31).

10. Système de sécurité selon la revendication 9, caractérisé en ce que ledit arbre de sortie (29) présente des extrémité intérieure et extérieure (29a), lesdits moyens de correction (16b, 17, 32) comportant un embrayage de désaccouplement (32) prévu sur ledit arbre de sortie (29) agissant pour désaccoupler l'extrémité extérieure (29a) de l'extrémité intérieure dudit arbre de sortie, des moyens (16a, 16b) pour centrer ladite extrémité extérieure (29a) dudit arbre de sortie (29) lorsqu'elle est désaccouplée de ladite extrémité intérieure, et des moyens (33) assurant en réponse à la rotation dudit élément menant (11) un désaccouplement périodique dudit embrayage (32).

11. Système de sécurité selon la revendication 1, caractérisé par un embrayage de détection de survitesse (47) pour désaccoupler l'un desdits arbres d'entrée (30, 31) en état de survitesse de seuil excessive afin d'amorcer ladite différence de vitesses angulaires relatives entre lesdits premier et second arbres d'entrée (30, 31).

12. Système de sécurité selon la revendication 11, caractérisé en ce qu'il comporte des moyens agissant en réponse à la grandeur de la charge (56) pour corrélérer la grandeur de la charge avec la vitesse du moteur afin de faire varier l'état de survitesse de seuil excessive en fonction de la grandeur de la charge.

13. Système de sécurité selon la revendication 1, caractérisé en ce qu'il comporte un embrayage menant actionné électriquement (100) assurant, lorsqu'il est désexcité, le désaccouplement d'un arbre d'entrée (30) d'avec son dernier organe

d'amont ou dernier organe d'aval porteur de charge (2a) respectif pour amorcer ladite différence de vitesses angulaires entre lesdits premier et second arbres d'entrée (30, 31) lorsqu'apparaît une panne d'électricité.

14. Système de sécurité selon la revendication 1, caractérisé en ce que l'organe de commande de frein de sécurité comporte un dispositif de maintien à force élevée et un élément d'ancrage à force faible (76) maintenant de manière amovible ledit dispositif de maintien à force élevée à l'armé, ledit organe de commande de frein de sécurité comportant un ressort à grande force (74) sollicitant ledit frein de sécurité vers la position de serrage, des moyens accouplés audit dispositif de maintien à force élevée pour maintenir ledit ressort (74) dans ledit état d'armé à l'encontre de passage du frein de sécurité (14) en position de serrage, des moyens (80) agissant en réponse à un état hasardeux pour déplacer l'élément d'ancrage à

force faible (76) afin de rendre ledit dispositif de maintien à force élevée libre de serrer ledit frein de sécurité, et des moyens (90) pour ramener lesdits ressorts (74), élément d'ancrage à force faible et dispositif de maintien à force élevée dans leur état d'armé initial et desserrer ledit frein de sécurité (14).

15. Système de sécurité selon la revendication 1, caractérisé en ce que chacun desdits arbres d'entrée (30, 31, 222, 214) comporte des moyens (21, 229) pour restreindre le mouvement décrit par ledit arbre d'entrée lorsqu'il tend à être entraîné en sens rétrograde par lesdits moyens de détection de ladite variation de vitesse ou de direction relative entre les deux susdits arbres d'entrée, de sorte que l'élément de sortie est toujours amené à tourner lors de différences de vitesses angulaires entre lesdits arbres d'entrée au lieu qu'il y ait entraînement rétrograde d'un arbre d'entrée.

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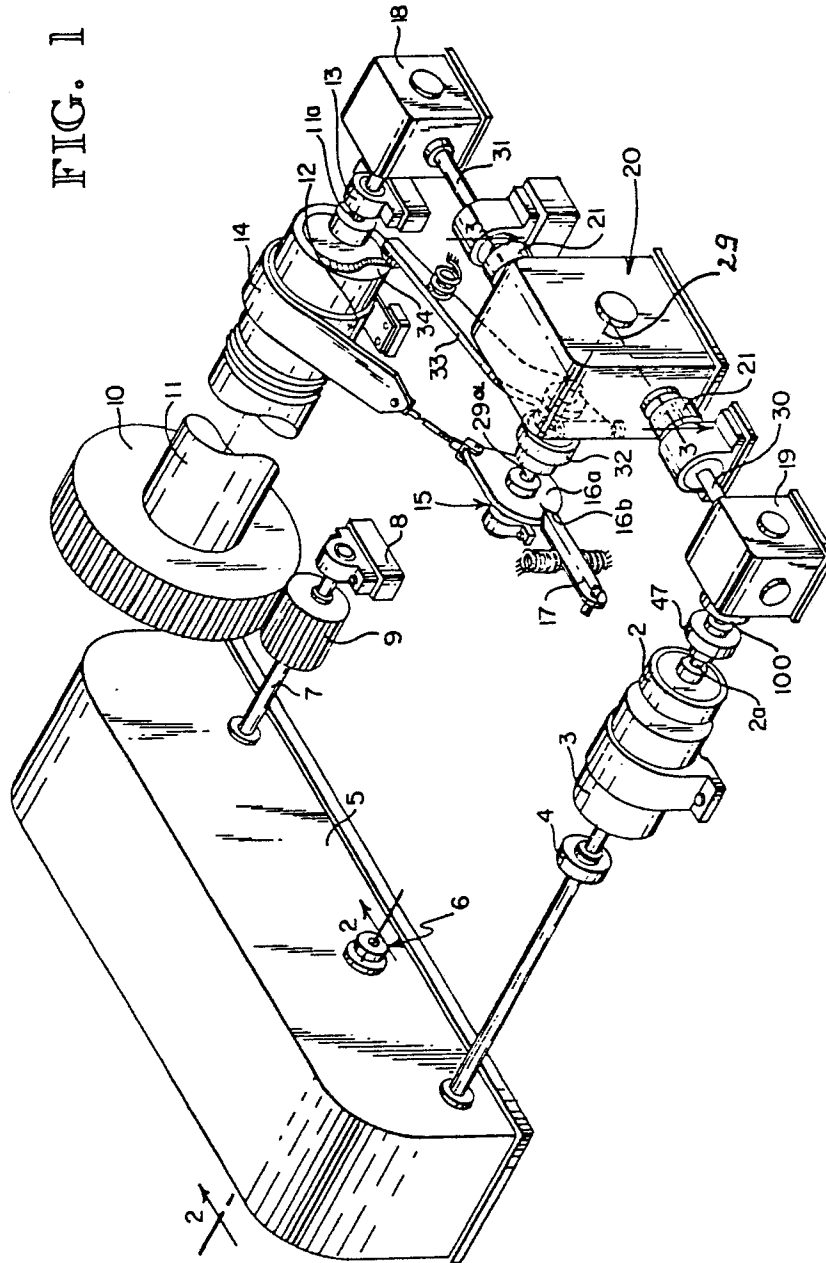
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FIG. 1



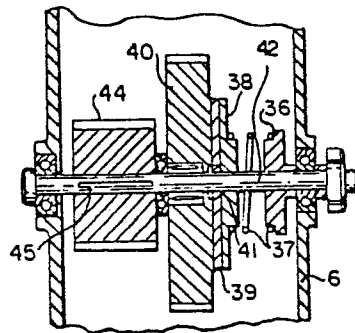


FIG. 2

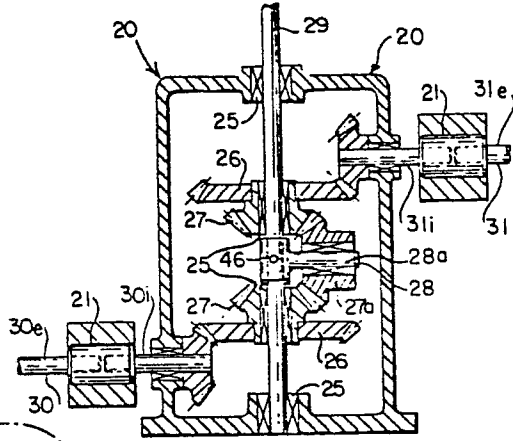


FIG. 3

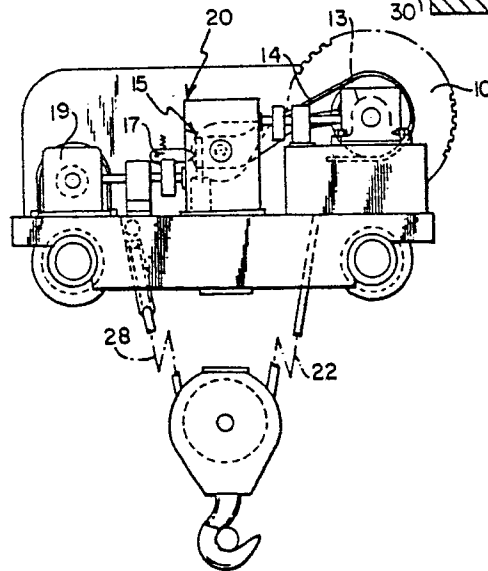


FIG. 4

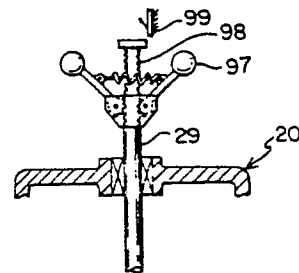


FIG. 3A

FIG. 5

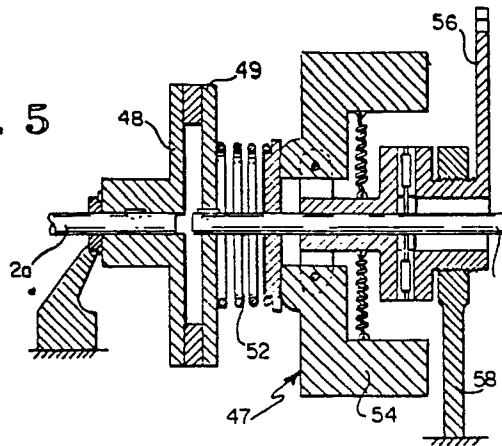


FIG. 6

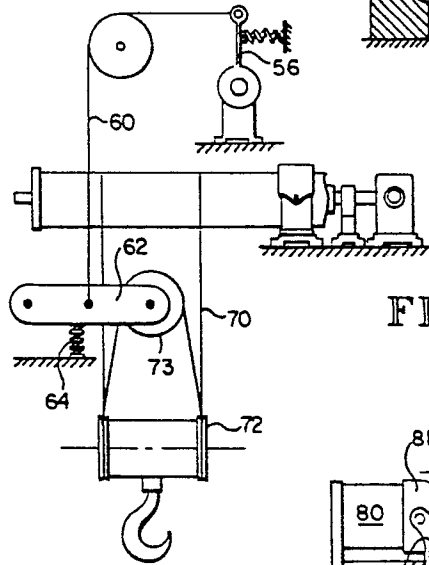


FIG. 7

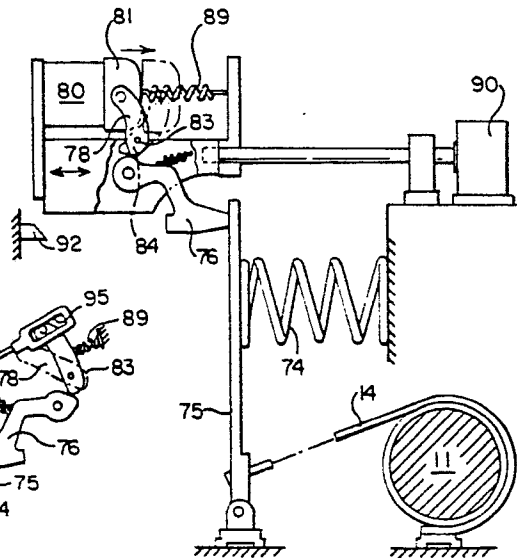
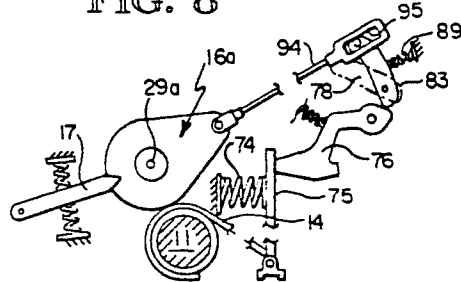


FIG. 8



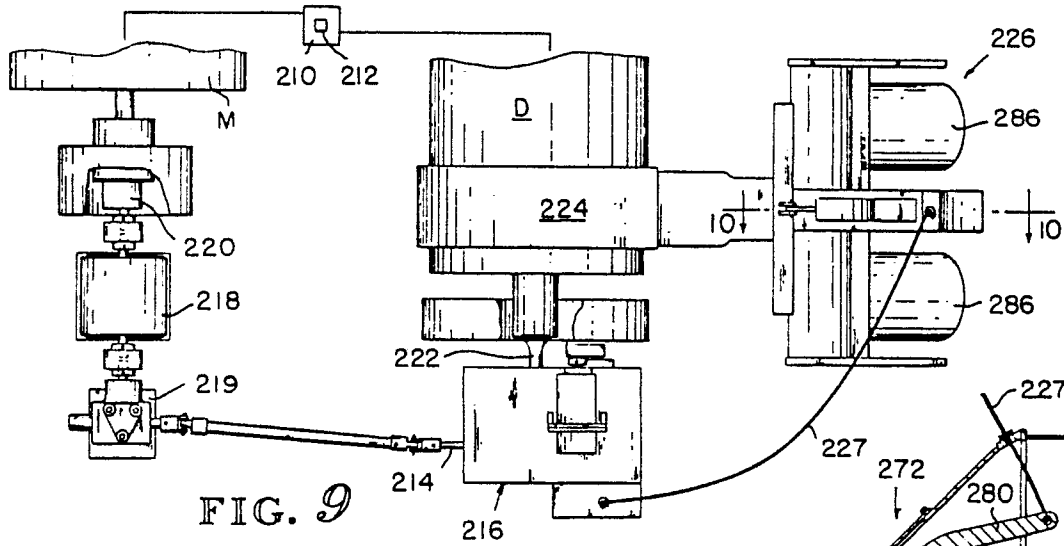
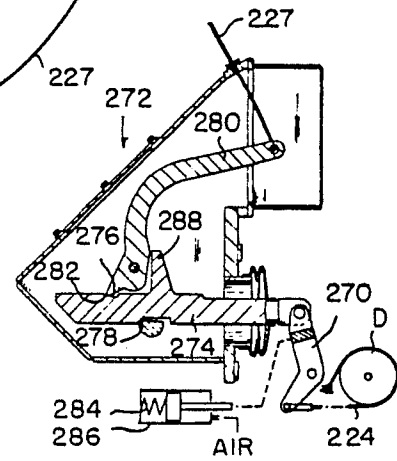
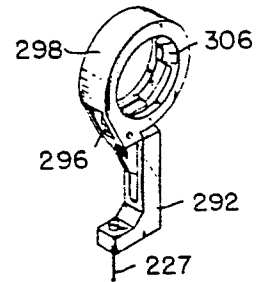
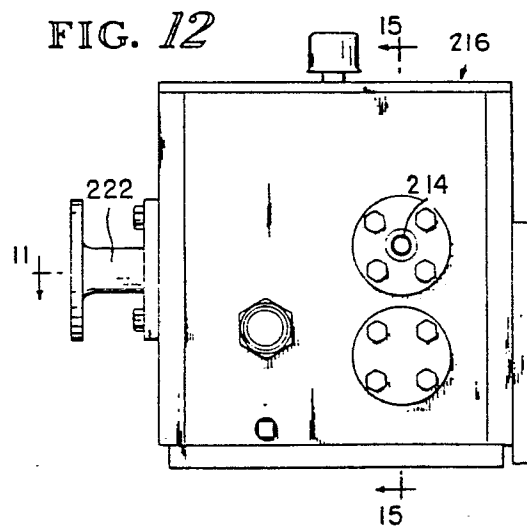
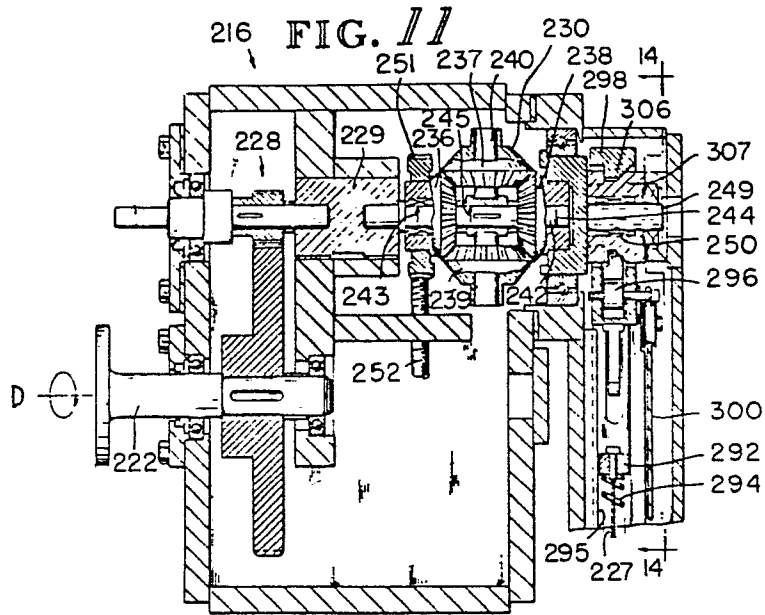


FIG. 10





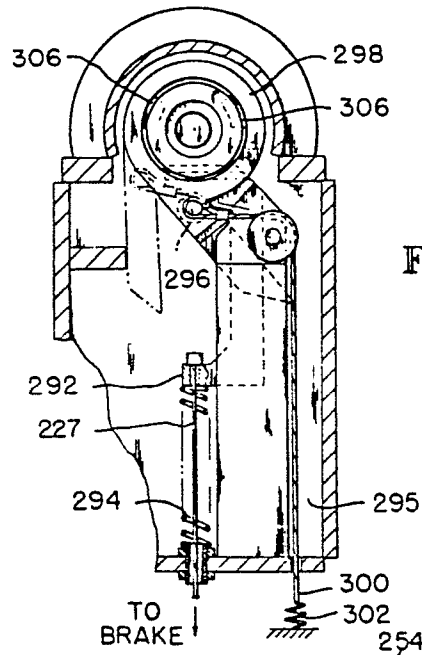


FIG. 14

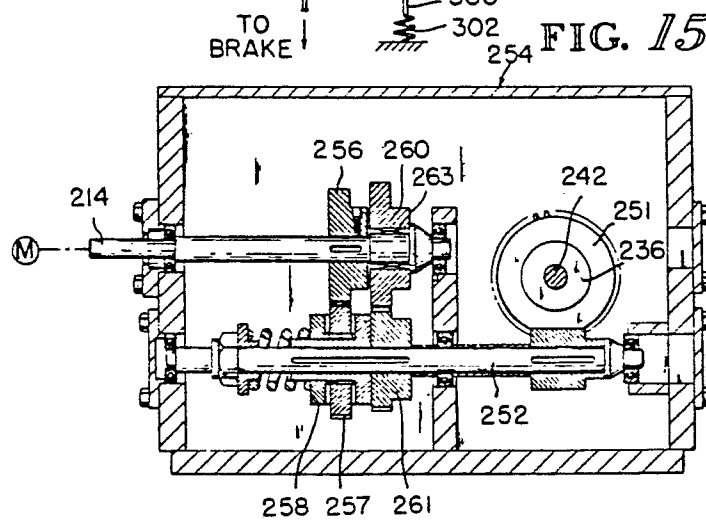


FIG. 15

