

M. A. EGGENBERGER
HYDRAULIC SERVO-MECHANISM FOR A STEAM
TURBINE INTERCEPT VALVE

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FIG. 2.

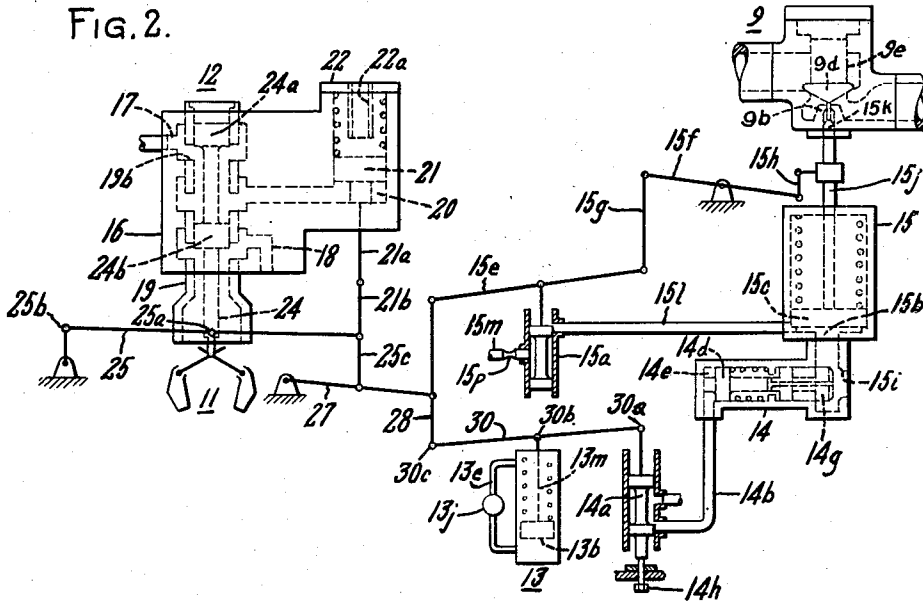
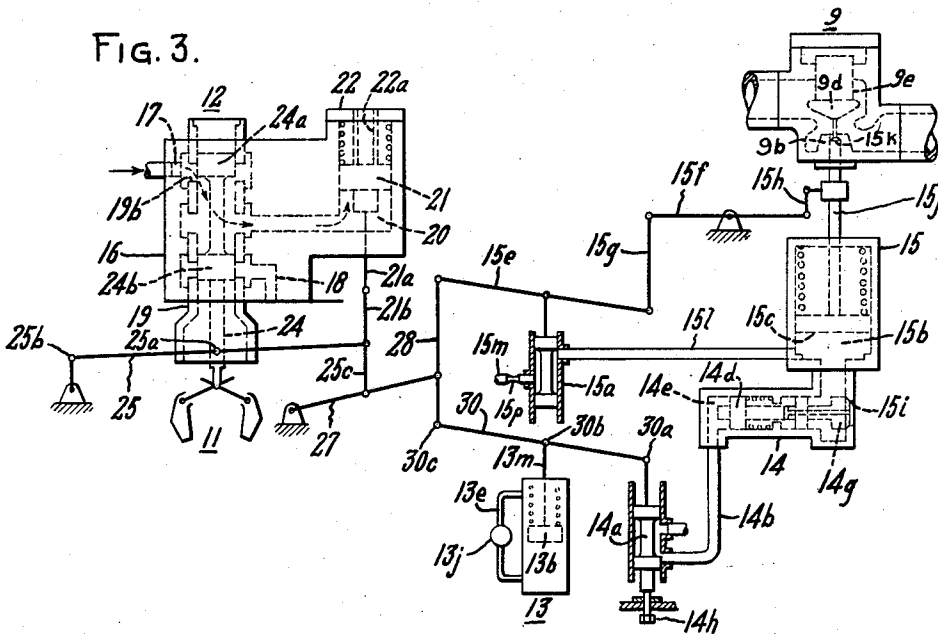


FIG. 3.



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FIG. 4

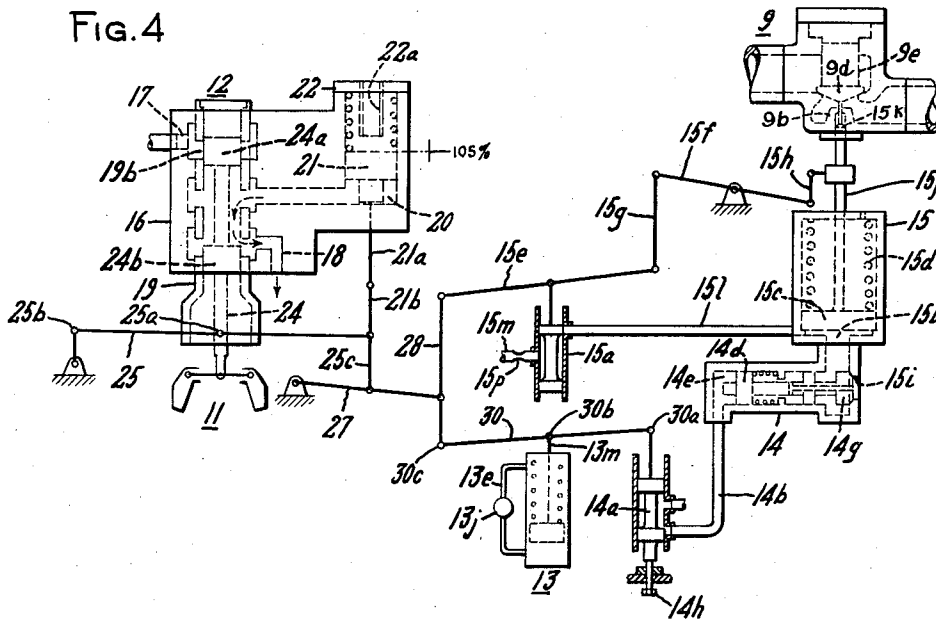
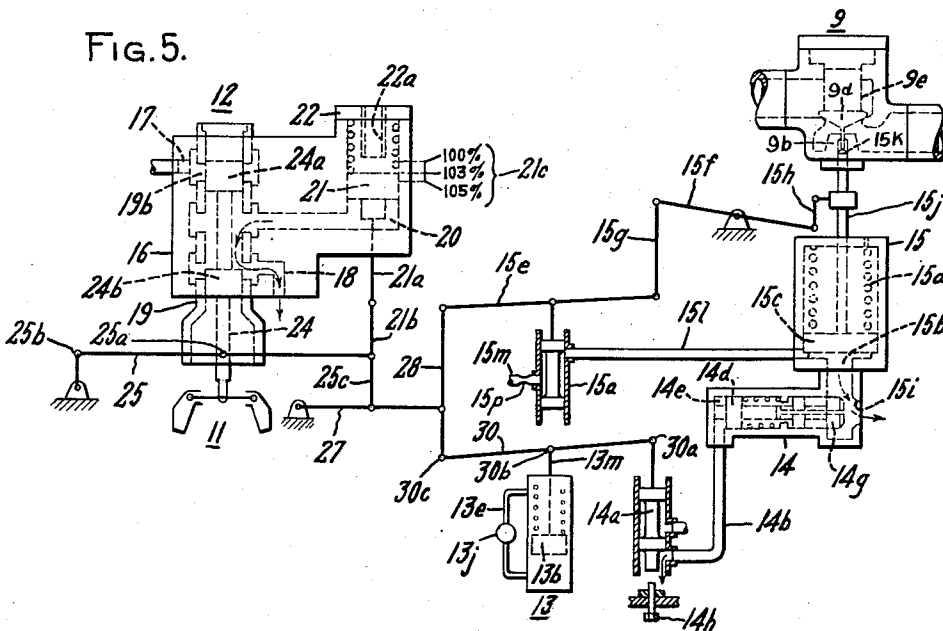


FIG. 5.



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HYDRAULIC SERVO-MECHANISM FOR A STEAM TURBINE INTERCEPT VALVE

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6 Claims. (Cl. 60—73)

This invention relates to governing mechanism for large "reheat" type steam turbines, and particularly to a hydraulic servo-mechanism for positioning the "intercept valves" in the event of abnormal overspeed conditions.

In a conventional reheat turbine, the control system consists, in addition to the usual speed control, of a "pre-emergency control system" which functions to temporarily control the speed after the load has been lost on the generator. This pre-emergency control system includes an auxiliary governing mechanism which in response to the overspeed of the turbine positions a valve to intercept the flow of steam from the "reheater" to the lower pressure turbines. The throttling and eventual closing of these "intercept valves" will, in combination with the main governor action, limit the overspeed of the unit to an acceptable value. In addition, the turbine is provided with an emergency governor which controls "main" and "re-heat" stop valves to shut off the flow of steam to the turbines in the event the unit reaches the "trip speed" of the emergency governor. (This trip speed is usually set at 110% of rated speed.) The main and reheat stop valves are either fully open or fully closed, and when closed, shut off the steam supply to the turbines and consequently, the unit will not be able to supply any station auxiliary power which may be required to sustain the operation of the plant.

The pre-emergency control system may be arranged so that upon slow acceleration, the intercept valves start closing at 101% of rated speed and fully close if the speed reaches 105% of rated speed. This insures that the intercept valves stay fully open during normal operation and when the generator is connected to the system. On a sudden loss of load, when the unit is accelerating at a high rate, the control system must quickly respond to prevent the turbine from reaching the trip speed of the emergency governor.

When a turbine-generator unit loses full load on the generator, it will instantly start accelerating at its maximum rate and will reach a peak speed as determined by the amount of energy supplied to the rotor after the load was lost. The overspeed results from (1) the energy fed to the high-pressure section of the turbine due to the time lag between the loss of load and the closing of the main control valves; (2) the energy fed to the intermediate and low-pressure section before the intercept valve closes; and (3) the energy stored in the steam conduits between the control valves and the downstream rotors. The energy fed to the high-pressure section during the closing of the control valve would only accelerate the rotor about 1%, and the energy of the "stored" steam would produce approximately 3% overspeed. It can be appreciated that little can be done about the effect of the stored steam since it can only be influenced to a very limited extent by the turbine and station layout. However, since about 75% of the turbine power output is produced by rotors downstream from the intercept valves, the time that it takes to close the intercept valves is very critical. In a

reheat turbine having an average rate of acceleration of about 12% per second upon loss of full load, and utilizing a conventional pre-emergency control system, the steam energy flowing through the intercept valves after the load is lost and before the intercept valves are closed would result in about a 5% overspeed of the turbine. Thus, such a conventional reheat turbine will not trip on overspeed after loss of full load and continues to supply necessary station auxiliary load.

However, in turbines having a rate of acceleration substantially over 12% per second, it is necessary to close the intercept valves in an appreciably shorter time than that accomplished by a conventional control system, since the conventional control system may be unable to hold the peak speed below 110% of rated speed upon loss of full load and the unit would trip the overspeed governor.

Accordingly, an object of the present invention is to provide an improved hydraulic-mechanical servo-mechanism for actuating the intercept valves of a fast-accelerating large reheat type compound steam turbine in order to obtain an acceptable overspeed performance upon loss of full load.

A specific object is to provide an improved governing mechanism which acts to close the intercept valve proportional to speed at a gradual increase in speed, and to quickly close the intercept valve when the rate of acceleration exceeds a predetermined value.

A still further object is to provide an improved hydraulic servo-mechanism for positioning an intercept valve which is constructed of simple mechanism having the utmost reliability.

Other objects and advantages will become apparent from the following description taken in connection with the accompanying drawings, in which—

Fig. 1 is a diagrammatic representation of a compound steam turbine powerplant having shown in detail an intercept valve with improved servo-mechanism incorporating the invention, illustrating the position taken during normal steady-state operation;

Figs. 2 and 3 represent the positions of the governing mechanism when the turbine is at standstill and below rated speed respectively;

Fig. 4 represents the position of the governing mechanism when the increase in turbine speed has been gradual; and

Fig. 5 represents the condition of the governing mechanism when the rate of acceleration exceeded the pre-selected value.

Generally stated, the invention is practiced by providing a pre-emergency control system having a pre-emergency governor positioning the intercept valve through a first linkage means upon gradual increase of speed, and a second linkage means for causing faster closing of the intercept valve; with special means for modifying the action of the second linkage so that the second linkage is ineffective during gradual increase in speed but acts to cause faster reaction of the intercept valve in the event of a high rate of acceleration.

Referring now more particularly to Fig. 1, the invention is illustrated as applied to a compound steam turbine-generator plant having a high pressure turbine element 1 and at least one lower pressure turbine element 2. These may be on different axes, or may be coupled together on a common axis, as shown in the drawing, and drive an electric generator 3. The turbine 2 may exhaust into still lower pressure turbines or into the condenser 4, from which condensate is returned by boiler feedpump 5 to the steam generator indicated at 6 as having primary steam generating coils 6a and a reheater section 6b.

It will be seen in Fig. 1 that the path of the turbine motive fluid is from the boiler feedpump 5 to the steam

generator 6a, then through the main stop valve 1b and main control valve gear 1a, to the high pressure turbine 1, and back to the reheater 6b. Steam discharged from the reheater passes through the reheat stop valve 10 and the intercept valve 9, and then to the inlet of the lower pressure turbine 2. This simple diagrammatic showing, of course, does not include many conventional elements of steam powerplants, such as feedwater heaters, lubricating system, packing control system and numerous minor details of the turbine control system, the arrangement of which will be understood by those familiar with the steam powerplant design.

The main steam controlling mechanism includes the main control valves represented at 1a and controlled by the main speed governor 7, and the main stop valve 1b which is controlled by the emergency speed governor 8, and used as a second line of defense against turbine overspeed. Excessive turbine speed could be caused by failure of the main control valves to close off the steam supply when the load is suddenly lost on the generator.

The reheat steam controlling mechanism comprises the pre-emergency control system with the pre-emergency speed governor 11 driven by the turbine shaft which controls the intercept valve 9 by means of special mechanism incorporating the invention. As a further precaution against overspeed, which could be caused by reheat steam in case of failure of the intercept valve to close if load was lost on the generator, a reheat stop valve 10, controlled by the emergency speed governor 8, is used. The operation of reheat stop valve 10 will be discussed more fully hereinafter.

The main speed governor 7 controls the unit under normal starting and running conditions. When the unit loses load and the speed increases, the pre-emergency governor 11 takes the control away from the main governor, and attempts to control the steady-state speed of the unit between 105% and 101% of rated speed as long as there is steam available from the reheater. If the transient speed following loss of load exceeds 110%, approximately, the emergency speed governor 8 will trip and shut the turbine down. This is a necessary safety feature but it is undesirable to have it shut the turbine down as long as all control elements are in proper working condition.

The pre-emergency servo-mechanism comprises the pre-emergency speed governor 11, a primary speed relay 12 connected through a first linkage arrangement to a servo pilot 15a which controls the servo-motor 15 to position the intercept valve 9, and a second relay means connected to the primary speed relay to cause rapid closing of the intercept valve. The second relay means includes a dashpot means which effectuates operation of the second relay means when the rate of acceleration of the turbine exceeds a preselected amount.

The primary speed relay 12 comprises a housing 16 having an oil inlet port 17 and a drain port 18, a first bore in which is slidably disposed a bushing member 19, and a second bore defining a chamber 20 in which is disposed the output piston 21. The upper end of chamber 20 is closed by a cover member 22 having formed integral therewith a central tubular member 22a which serves as a stop member defining the uppermost position of piston 21. A biasing spring 23 is disposed around the stop member 22a and engages piston 21 to cause it to move to the lowermost position in its range of movement when the oil pressure in chamber 20 is released. Actuating oil at suitable pressure is admitted to the chamber 20 by the primary pilot valve 24, which is connected to be positioned by the flyball speed governor 11. Pilot 24 is slidably disposed in the bore 19a of the slidable bushing 19. The restoring means for the primary pilot comprises a lever member 25 connected by the pivot indicated in dotted lines at 25a to the lower end of bushing 19, and is supported on a fixed fulcrum

25b. The opposite end of lever 25 is connected by link 25 to the piston rod 21a of the pre-emergency speed relay piston 21, in a manner which will be obvious from the drawing.

The combination of the flyball governor 11, bushing 19, pilot 24, and speed signal piston 21 is well known in the art. In the normal operating position, shown in Fig. 1, upper land 24a of pilot 24 limits the flow of actuating oil from inlet port 17 through port 19b of bushing 19 to the speed relay chamber 20 to that necessary to make up for leakage of oil out of chamber 20. Lower land 24b covers drain port 18 to prevent oil flow out of chamber 20 during normal operation. If the speed of flyball governor 11 should increase, pilot 24 will descend so that land 24b uncovers drain port 18a thus reducing the oil pressure in chamber 20 and causing piston 21 to descend slightly. This downward movement causes lever 25 to pivot clockwise about fulcrum 25b, so as to lower bushing 19 and return pilot 24 and bushing 19 to the same relative position they had during normal operation to retain piston 21 in the position corresponding to the increased speed condition. Thus, the position of piston 21 is a measure of the rotational speed of the turbine, and a scale can be appended, as indicated at 21c, from which the speed can be read. When the speed drops, pilot 24 rises to admit more oil to chamber 20 and move piston 21 up against the stop member 22a, which position may correspond to a speed of 99.6% of rated speed.

The speed relay 12 is connected to position servo motor 15 by the operation of servo pilot 15a through piston rod 21a, links 21b, 25c, lever 27, link 28 and lever 15e. Thus, upward movement of speed relay piston 21 results in corresponding movement of servo pilot 15a to admit operating liquid to conduit 151 and chamber 15b of servo motor 15 to move piston 15c upwardly against the bias of spring 15d. Operating oil is supplied to pilot 15a through inlet conduit 15m in which is located orifice 15p for reasons which will appear hereinafter. Pilot 15a is restored by lever 15f and links 15g, 15h, the last-mentioned of which is connected to move with piston rod 15j, in a manner which will be obvious from the drawing. In normal operation, pilot 15a will be raised by the distance indicated as 15n above the port to admit full pressure to chamber 15b and maintain piston 15c in its uppermost position, defined by engagement of an annular valve stem sealing shoulder identified 15k with the cooperating seat in a bushing 9b. The overtravel 15n corresponds to a speed change of 1% on the relay 21 so that the pilot 15a would be in the on-port position at 101% steady-state speed.

The intercept valve 9 is illustrated diagrammatically as comprising a housing 9a supporting the bushing assembly 9b, in which is slidably disposed the valve stem 9c. Because it must perform accurate throttling functions, the valve 9 is of a pressure-balanced type, the flow control disk having a piston portion 9e, the upper surface of which is exposed to the downstream pressure communicated to chamber 9f by a pressure balancing conduit 9g.

The arrangement of the servo motor 15 is such that in normal operation at rated speed, with the governor 7 controlling the main valves 1a, the valve disk 9d will be in its uppermost position shown in Fig. 1, with pilot 15a open by the distance 15n as described above. When pilot 15a is lowered beyond the distance 15n by piston rod 21a through the action of lever 27, link 28, and lever 15e, liquid is drained from chamber 15b through conduit 151 past the upper end surface of pilot 15a, permitting the piston 15c to descend and close intercept valve 9.

Servo motor 15 is also provided with a drain port 15i which is normally closed by a valve 14g which latter is controlled by an auxiliary hydraulic motor 14 having a piston 14d connected to the valve 14g and engaged by a compression spring 14c. Drain opening 15i is much larger than conduit 151 so that chamber 15b may be

drained very rapidly in the event servo motor 15 is still being supplied with operating oil through pilot 15a when valve 14g is open. The auxiliary servo motor 14 is controlled by a servo pilot 14a which functions to admit operating liquid to chamber 14e to move piston 14d against the bias of spring 14c to close drain port 15i or to drain chamber 14e to permit spring 14c to open drain port 15i.

In normal operation, with the main governor 7 in control, the pilot valve 14a is in its lowermost position against stop 14h to admit operating oil to chamber 14e to maintain valve 14g closed. The essence of the invention lies in the special linkage means which connects the primary speed relay 12 with the servo pilot 14a. Servo pilot 14a is operatively connected at 30a to the right-hand end of lever 30, and the extreme left-hand end 30c of lever 30 is connected to link 28. Intermediate the ends of lever 30 is operatively connected a dashpot mechanism 13. The dashpot 13 comprises a dashpot cylinder 13a that is at all times kept filled with oil and which contains a piston 13b formed integral with a piston rod 13m connected at 30b to lever 30. Piston 13b is biased downwardly by coil spring 13c. Spring 13c also biases lever 30 in a downward direction to move pilot 14a against stop 14h to admit operating fluid to chamber 14e to maintain valve 14g closed. Stop 14h is set to obtain a predetermined opening 14n of pilot 14a to obtain full oil pressure in chamber 14e. The above and below piston chambers of the dashpot are in communication by way of the tube 13e, in which a valve 13j is mounted, which permits continuous adjustment of the flow restriction effected by tube 13e. As will be appreciated by those skilled in the art, a dashpot of the described design will, if the restriction of stop 14h is removed, descend with a certain, for practical purposes, constant speed. Since the rate of descent of the speed relay piston 21 is a measure of the acceleration of the unit, it can be said that the rate of descent of the dashpot is a measure of a reference acceleration which, by the arrangement of the linkages is effectively compared with the actual acceleration of the unit. If we, arbitrarily, say that the dashpot spring 13c and the opening of needle valve 13j shall be matched to correspond to an acceleration of the unit of 1% per second, it can be seen that at accelerations up to 1% per second the dashpot will follow the speed relay motion and the pilot valve 14a will remain on its stop 14h. If the acceleration exceeds 1% per second, the dashpot will not be able to follow and therefore the speed relay motion will pivot lever 30 about pivot 30b to cause the pilot valve 14a to lift off its stop and actuate motor 14 to release oil from chamber 15b. The more the acceleration is above 1%, the faster valve 14a opens and, therefore, the quicker intercept valve 9 closes.

In normal operation, with the main governor in control, the pre-emergency governor 11 will maintain speed relay piston 21 at the "100%" position (Fig. 1), and the pilot valve 15a slightly open to admit operating fluid to servo motor 15 to admit full hydraulic fluid pressure to chamber 15b, and to effectuate a predetermined travel 15n of pilot 15a before chamber 15b is drained to permit closing movement of intercept valve 9. Travel 15n prevents closing of the intercept valve until the unit reaches 101% of rated speed during slow speed changes. The intercept valve 9 is thus maintained fully open by the liquid under pressure in chamber 15b. Also, piston 21 locates the dashpot piston 13b in the intermediate position and spring 13c biases lever 30 downwardly to maintain pilot 14a against stop 14h. Operating fluid is supplied through pilot 14a and conduit 14b to chamber 14e to maintain valve 14g closed.

Now it will be seen that if the speed relay piston 21 rises slightly due to a decrease in speed, pilot 15a opens wider and dashpot piston 13b moves upwardly against spring 13c without altering the position of pilot 14a.

Similarly, if speed relay piston 21 moves downward slowly due to a gradual increase in turbine speed, pilot 15a moves downwardly and as soon as it has traveled through the distance 15n it will start to drain chamber 15b to permit spring 15d to bias intercept valve 9 in the closing direction. The system can be adjusted, for instance, so the intercept valve will start closing at 101% of rated speed and be closed at 105% of rated speed when the speed increases slowly. During this gradual increase of speed, lever 30 pivots about connection 30a and dashpot piston 13b moves downwardly a corresponding amount, during which spring 13c is permitted to extend slightly, oil flows upwardly through bypass 13e into the space above the piston 13b, and pilot 14a remains against stop 14h. It remains to note, that in the event piston 13b is caused to move downwardly at a rate in excess of the capacity of the bypass 13e to transfer oil from below the piston to the space above, the dashpot becomes a temporary fulcrum and lever 30 pivots about connection 30b to position pilot 14 to rapidly actuate valve 9 towards closed position.

The above method of operation of the servo-mechanism may be seen from the following comparison of Figs. 2, 3, 4, and 5 with the normal operating condition represented by Fig. 1.

Fig. 2 represents the condition of the mechanism in the completely shut down condition, with no operating liquid supplied to the inlet port 17 or pilots 15a, 14a. When the turbine is at rest, the pre-emergency speed governor 11 is in the fully collapsed position, pilot 24 is in its extreme upper position, and piston 21 and bushing 19 are in their lowermost position. Thus, when actuating liquid is supplied to inlet port 17, it will have free access past land 24b to the chamber 20 so as to cause the speed relay piston 21 to rise. It will be observed that the linkages are so designed that in this shut down condition, pilot 15a which had been lowered to drain chamber 15b is just slightly below the "on-port" position through the action of the restoring links 15h, 15f, 15g and 15e, dashpot piston 13b is in its lowest position, and pilot 14a is against stop 14h. Similarly, with no oil being supplied to chamber 14e, valve 14g is maintained open by spring 14c. Thus, servo motor piston 15c is in its lowermost position, pulling the valve disk 9d firmly onto its seat.

The process of starting the turbine will be accomplished by the following action of the governing mechanism, beginning with the condition shown in Fig. 2. When the auxiliary oil supply pump (not shown) is started, in order to supply lubricating oil to the bearings and actuating liquid to the hydraulic mechanism, the pressure liquid supplied to port 17 will enter chamber 20 and cause piston 21 to rise to its uppermost position against stop 22a. Similarly, the pressure liquid will flow through pilot 14a to chamber 14e to close valve 14g. The resulting condition of the mechanism is shown in Fig. 3. It will be seen that upward movement of piston 21 has pulled pilot 15a upward to supply operating liquid to chamber 15b to move servo motor piston 15c to the "open" position of valve 9d, as defined by the engagement of annular shoulder 15k with the sealing seat in the bushing assembly 9b. Furthermore, the dashpot piston 13b is elevated somewhat above the position occupied in the normal steady-state operating condition of Fig. 1 while the pilot 14a remains against stop 14h. Thus, the intercept valve 9 is in position to permit the flow of steam to the low pressure turbine 2.

The main stop valve 1b may now be opened and steam admitted to the high pressure turbine 1 by actuation of the main valve gear 1a, under control of a "load limit" device (not shown) associated with the main speed governor 7. As the turbine speed increases and the speed governor 7 takes over control, the flyweights 11a of the pre-emergency speed governor move outwardly so as to pull the pilot 24 downwardly and decrease the

supply of actuating liquid to chamber 20. Thus, when the speed goes above 99.6% of rated speed, piston 21 moves downwardly away from stop 22a and approaches the 100% speed condition shown in Fig. 1. This causes pilot 15a to descend within the distance 15n from the edge of the port so that oil at full pressure is still admitted to chamber 15b. The dashpot piston 13b also descends to the position shown in Fig. 1, somewhat relieving the compression on spring 13c, without changing the position of pilot 14a which remains in contact with the stop 14h. Thus, there is no change in the condition of valve 14g and the servo motor 15. The turbine is in normal operation with the pre-emergency governing mechanism in the "stand-by" condition. The intercept valve 9 remains in the fully open position so long as the speed remains steady and below 101% of rated speed, at which speed the pre-emergency governor is set to start operating the intercept valves upon a "slow" further increase in speed.

To illustrate the action of the pre-emergency governor in the event of a gradual rise in speed, the following operation may be noted. The design of the linkage mechanism is such that after the speed relay piston 21 reaches the 101% position, the pilot 15a has moved downwardly enough to drain servo motor chamber 15b to effect closing movement of valve 9. Valve 9 begins to throttle the flow of motive fluid to the low pressure turbine 2 and, therefore, control the speed of the machine. If the turbine reaches 105% of rated speed, the pre-emergency governor will have moved the intercept valves 9 to the fully closed position. The resulting condition of the mechanism is shown in Fig. 4. During this gradual increase in speed (below that for which the dashpot is set), the bypass 13e around piston 13d bypasses sufficient oil to permit piston 13b to move downwardly and pivot lever 30 around pivot 30a. Pilot 14a remains against stop 14h and valve 14g remains closed. Thus, when the rate of acceleration is below a preselected amount, the intercept valve is controlled in a conventional manner by the pilot 15a. Conversely, when the speed is over 105%, and the turbine is slowing down gradually, the intercept valves will start opening at about 105% of rated speed. When the speed returns to 100%, the pilot valve 15a will be returned to the position shown in Fig. 1.

In the event of a high rate of acceleration which may occur on sudden loss of load, my novel governing mechanism operates in the following manner. Assume that the governing mechanism is in the normal condition shown in Fig. 1 when the turbine suddenly loses load and begins to accelerate at a rate in excess of a predetermined rate, which may for instance be on the order of 10% to 20% of rated speed per second. When this occurs, speed relay piston 21 moves downward rapidly, exceeding the speed with which the dashpot can travel downward because of the flow restriction in valve 13j. The resulting condition of the mechanism is shown in Fig. 5. The oil trapped below piston 13b causes piston 13b to move downward with a predetermined slow speed while connection point 30c is moving down rapidly which causes the lever 30 to pivot about the fulcrum 30b to raise pilot 14a to drain operating fluid from chamber 14e. With the pressure relieved in chamber 14e, spring 14c biases the valve 14g open to drain operating liquid from chamber 15b to permit spring 15d to effect rapid closing of valve 9. The foregoing operation takes place almost instantaneously and before the speed relay piston reaches the position which would be necessary to close the intercept valves with the pilot valve 15a. Thus, pilot 15a is located in the position shown where oil is still being supplied to servo motor 15. However, fast closing action of valve 9 is still obtained due to orifice 15p in pilot valve inlet conduit 15m which limits the flow of operating fluid into chamber 15b. Thus, in the event of a rate of ac-

celeration, higher than a preselected rate, the pre-emergency governor 11 starts controlling almost immediately and possibly before the speed reaches 101% of rated speed, and it also causes the valves to close substantially before the turbine speed reaches 105% of rated speed.

The improvement obtained may be seen from the following. On a unit equipped with a conventional control and having a maximum acceleration rate of 14% per second following loss of full load, the intercept valves would effectively start closing at about 103% speed and reach the closed position at 107% speed. The lag of 2% compared to the corresponding values for slow acceleration (101-105%) is caused by lags which appear at high rate of acceleration, because of the inherent characteristics of hydraulic controls. This overspeed combined with the energy in the entrained steam would result in actuation of the emergency governor and shutting down the unit. However, a similar unit but equipped with my invention will, under the same conditions, start closing the intercept valves at about 101.5% and reach closed position at about 103% of rated speed. Because of this improvement it is possible to limit the peak speed of faster accelerating units well below 110% of rated speed which would be impossible with a conventional control system.

While not important to the present invention, it remains to note that in the event the emergency governor 8 is actuated, the centrifugally operated ring of the governor 8 flies out and closes trip valves 8a, 8b. The closing of valve 8a shuts off the supply of operating oil to reheat stop valve 10 and inlet 17 of pre-emergency speed relay 12 to close valve 10 and drain the oil from relay 21. As also shown in Fig. 1, when the emergency governor trips valve 8b, the supply of operating fluid to the main stop valve 1b is also discontinued and it is immediately closed. It will further be seen that the supply of operating liquid for the main control valve gear 1a is also closed off by valve 8a and thus it will close regardless of the position of the speed governor 7.

It will be appreciated that the structure described specifically herein is intended to be illustrative only, and actual embodiments of the invention may take various forms. For example, the mechanical construction of the servo motor 15 and primary speed relay 12 may take various forms.

It is, of course, intended to cover by the appended claims all such modifications as fall within the true spirit and scope of the invention.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. In a reheat turbine powerplant having in series at least one high pressure turbine, reheater means, and one lower pressure turbine, inlet valve means admitting motive fluid to the high pressure turbine, emergency overspeed governing means connected to the inlet valve for closing the inlet valve upon occurrence of a preselected overspeed condition, an intercept valve located between the reheater and lower pressure turbine and adapted to throttle the flow of motive fluid from the reheater to the low pressure turbine, pre-emergency governor means connected to the intercept valve and adapted to close the intercept valve upon occurrence of a pre-emergency speed condition below the preselected emergency overspeed condition, the combination of first motor means controlled by first pilot means connected to the pre-emergency governor means to effect movement of the intercept valve in the closing direction upon occurrence of the pre-emergency speed condition, second relay means connected to the pre-emergency governor means and constructed and arranged to effect movement of the first pilot means to move the intercept valve in the closing direction independently of the operation of the first pilot means, the second relay means including dashpot means for preventing actuation of the second relay during movement of the pre-emergency gov-

ernor means at rates below a preselected value corresponding to a preselected rate of turbine acceleration, the dashpot being actuated by the pre-emergency governing means to move the second relay means to effect closing movement of the intercept valve upon movement of the pre-emergency governing means at a rate corresponding to a rate of turbine acceleration exceeding the preselected rate.

2. In a reheat turbine powerplant having in series at least one high pressure turbine, reheater, and one lower pressure turbine, first valve means controlling the flow of motive fluid to the high pressure turbine, emergency governing means connected to the first valve means and adapted to close the first valve means in the event the rotor speed rises to a preselected overspeed condition, an intercept valve located between the reheater and lower pressure turbine and adapted to throttle the flow of motive fluid from the reheater to the low pressure turbine, pre-emergency governor means connected to the intercept valve and adapted to close it upon occurrence of a pre-emergency speed condition below the preselected emergency overspeed condition, the combination of first motor means for positioning the intercept valve and a first pilot means controlling the flow of operating fluid to the first motor means, second valve means controlling the flow of operating fluid from the first motor means to effect rapid closing of the intercept valve, second pilot means connected to position the second valve means, first linkage means connecting the pre-emergency governor to the first pilot means, second linkage means connecting the pre-emergency governor to the second pilot means, dashpot means operatively connected to the second linkage means to prevent actuation of the second pilot means during movement of the second linkage means at rates below a preselected value corresponding to a preselected rate of acceleration of the turbine, the dashpot being effective to cause the second linkage means to actuate the second pilot means to effect opening movement of the second valve and to effect closing of the intercept valve upon movement of the second linkage means at a rate corresponding to a rate of acceleration of the turbine exceeding said preselected rate.

3. In a reheat steam turbine powerplant having in series at least one high pressure turbine, reheater means, and one lower pressure turbine, inlet valve means controlling the admission of motive fluid to the high pressure turbine, and intercept valve means adapted to throttle the supply of motive fluid from the reheater to the lower pressure turbine, the combination of servo mechanism for positioning the intercept valve comprising a speed relay responsive to turbine rotor speed and having an output member positioned as a function of speed, first servo motor means for positioning the intercept valve and a first servo pilot controlling the flow of operating fluid to the first servo motor means, second valve means controlling the flow of operating fluid from the first motor means to effect rapid closing of the intercept valve, a second servo pilot connected to position the second valve means, stop means limiting movement of the second servo pilot in a direction to maintain the second valve means closed, first linkage means connecting the speed relay output member to the first servo pilot to control the movement thereof, second linkage means connecting the speed relay output member to the second servo pilot, dashpot means operatively connected to the second linkage means comprising a piston and cylinder with restricted fluid bypass means for communicating fluid from one side of the piston to the other, and spring means biasing the dashpot piston relative to the cylinder in a direction to bias the second servo pilot into engagement with said stop means, whereby upon movement of the speed relay output member at a rate below a preselected value corresponding to the capacity of the fluid bypass to communicate fluid from one side of the dashpot piston to the other the second linkage means moves freely relative to the second servo pilot, and in the event the rate of movement of the speed relay output member exceeds the preselected

rate, the second linkage means moves the second pilot away from the stop means to effect opening movement of the second valve means to effect movement of the servo output member to close the intercept valve independently of the position of the first servo pilot.

4. In hydraulic servo mechanism for positioning an output member by actuating means including a housing with an output piston adapted to be moved to an active position by the pressure of hydraulic fluid supplied thereto, first servo pilot means controlling the admission of operating fluid to the housing, valve means controlling the drainage of operating fluid from the housing, second servo pilot means connected to position the second valve means, a first relay including a signal input member connected to a signal pilot valve for positioning a signal output member, the combination of first linkage means connecting the signal output member with the first servo pilot valve, second linkage means connecting the signal output member with the second servo pilot valve, and dashpot means operatively connected to the second linkage means, whereby the second linkage means is free to move without effecting movement of the second servo pilot during movement of the signal output member in at least one direction at a rate below a preselected value, and to actuate the second servo pilot in the event of movement of the signal output member in the same direction at a rate greater than the preselected value to effect opening movement of the second valve means to drain operating fluid from the housing to effect movement of the output piston to an inactive position.

5. Hydraulic servo-mechanism for positioning an output member by actuating means including a first cylinder with an output piston adapted to be moved to an active position by the pressure of hydraulic fluid supplied thereto, first servo pilot means controlling the admission of operating fluid to the first cylinder, valve means controlling the drainage of operating fluid from the first cylinder, second servo pilot means controlling the operation of said valve means, stop means limiting the movement of the second servo pilot in the direction to maintain the valve means closed, a hydraulic relay including a signal input member connected to a signal pilot valve for positioning a signal piston, first linkage means connecting the signal piston to the first servo pilot, second linkage means connecting the signal piston to the second servo pilot, dashpot means operatively connected to the second link means and comprising a piston member slidably disposed in a dashpot cylinder and having a stem member connected to the second linkage means, means resiliently biasing the dashpot piston relative to the dashpot cylinder in a direction to effect movement of the second servo pilot to engage said stop means, restricted bypass means communicating fluid from one side of the dashpot piston to the other, whereby upon movement of the signal piston at a rate below a preselected value corresponding to the capacity of said fluid bypass to communicate fluid from one side of the dashpot piston to the other the second linkage means moves freely relative to the second servo pilot to keep the second pilot on the stop means, and in the event the rate of movement of the signal piston exceeds the preselected rate, the connection between the stem member and the second linkage means becomes a fulcrum about which the second linkage means pivots to move the second servo pilot away from the stop means to effect opening movement of the second valve means to move the output piston to an inactive position.

6. Servo-mechanism for positioning a first motive fluid control valve means of an elastic fluid turbine comprising a servo motor having a first servo pilot member and an output member connected to said first valve means, the output member being located in a cylinder and adapted to be moved to the valve open position by fluid supplied thereto, a second valve means controlling the flow of fluid from said cylinder, second servo pilot means con-

trolling the operation of said second valve means, a speed relay including a speed responsive member connected to position; a speed signal member as a function of turbine rotor speed, the rate of position change of the speed signal member being a measure of the acceleration of the turbine rotor; first linkage means connecting the speed signal member to the first servo pilot member, second linkage means connecting the speed signal member to the second servo pilot member, dashpot means including a piston and cylinder with a restricted fluid bypass around the piston for communicating fluid from one side of the piston to the other, means operatively connecting the piston to the second linkage means, whereby the dashpot means acts to move the second linkage means without ef-

fecting movement of the second servo pilot during relatively slow acceleration of the turbine rotor and moves the second servo pilot to effect movement of the second valve means to effect rapid closing of the first valve means in the event of acceleration of the turbine rotor at a rate greater than a preselected value corresponding to the capacity of the bypass to communicate fluid from one side to the other of the dashpot piston.

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