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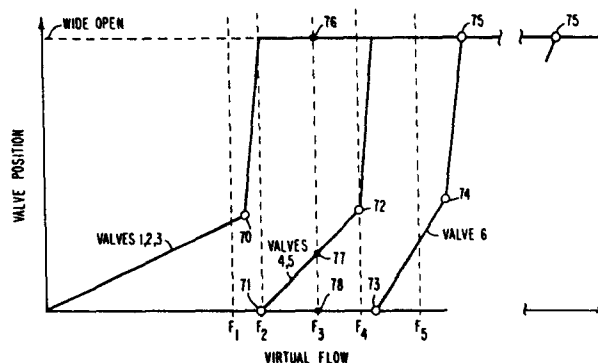
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54 **System for minimizing valve throttling losses in a steam turbine power plant.**

57 This invention relates to a system for reducing steam throttling losses which occur across turbine governor valves, by adjusting boiler throttle pressure so as to obtain optimum governor valve setting thereby providing for maximum power production efficiency. In a typical valve grouping sequential positioning pattern (Fig. 8), the flow rate is assumed to be F3 where valves 1, 2 and 3 are fully open causing substantially no throttling losses; however, valves 4 and 5 are partially open causing some losses. An optimum valve position program (56) is operative to determine optimum flow demands corresponding to optimum valve positions (72 and 71) which are greater and less than the present flow demand. Then, the closest optimum flow demand, e.g. (72) is selected. Based on this optimum flow demand, a throttle pressure control program (34) adjusts throttle pressure setpoint so that the governor valve (4, 5) positions change to (72). A major advantage of this invention is fuel savings due to maximum time spent in the optimum valve position conditions.



SYSTEM FOR MINIMIZING VALVE THROTTLING
LOSSES IN A STEAM TURBINE POWER PLANT

The present invention relates broadly to the field of boiler-turbine controlled operations, and more particularly to an efficient valve position controller for governing the regulation of boiler throttle pressure to render the steam turbine admission valves in a selected one of a plurality of predetermined valve position states which correspond to valve operating points effecting minimum throttling losses.

It has been known for some time that the efficiency of a steam turbine power plant is degraded by the throttling losses that occur during the time when the steam admission valves of the steam turbine are governing steam flow in the partially opened state. It is understood that any improvement in efficiency of plant performance by reduction of these throttling losses will substantially reduce fuel consumption and provide a significant economic savings in the process of energy production. Various methods, such as: (1) constant throttle pressure-sequential valve operation; (2) throttling control-single valve operation; (3) sliding pressure; and (4) bypassing, have been utilized by some of the utilities to effect a reduction in valve throttling losses. For a more detailed description of these methods and how they compare to each other, refer to the paper entitled "A Review of Sliding Throttle Pressure For Fossil Fueled Steam-Turbine Generators" authored by G. S. Silvestri et al. which was presented at the American Power Conference, April 18-20, 1972. Conclusions of this paper

indicate that "hybrid" type turbine designs which combine sequential valve and sliding throttle pressure operation, particularly the 50% admission "hybrid" units, have been shown to offer more efficient performance characteristics overall. The word "hybrid" was used in the Silvestri paper to describe boiler-turbine units that utilize constant throttle pressure-sequential valve operation down to some valve point, say 50% admission, at which time the valve position (admission arc) is held constant and the throttle pressure is reduced to attain lower flows. The Silvestri paper did not consider any method other than the "hybrid" method to further increase plant efficiency.

A similar "hybrid" type boiler-turbine plant operation has also been disclosed in U.S. Patent 3,262,431 issued to F. J. Hanzalek on July 26, 1966. The Hanzalek patent is directed to an operation of sliding boiler pressure and sequential valve operation utilizing a particular boiler control configuration. It appears that Hanzalek's operation pertains to sliding boiler pressure during turbine start-up and initial loading to a value where optimum temperature and pressure conditions exist in the boiler and thereafter, increases in turbine steam flow are controlled by normal sequential valve movement at constant boiler pressure until another optimum boiler condition point is desired. In neither, the paper by Silvestri et al. nor the patent 3,262,431, is there described or even suggested any control system or method of improving plant efficiency by reducing throttling losses during the sequential valve mode steam flow governing operation periods.

Recently, improvements have been directed towards sequential valve control operation of turbine power plants by calculating a set of sequential valve position ranges which relate to minimizing throttling losses and providing an indication to the power plant operators when the steam admission valves have been sequentially positioned in one of these ranges. For a more detailed description reference is made to U.S. Patent 4,088,875. This improvement, of course, allows the power plant operator to select steam turbine

operational points which correspond to minimizing throttling losses and provide a more efficient plant operation. On the other hand, this improvement normally consists of about 5 or 6 sequential valve position ranges of which each constitutes only approximately 3% or less of the steam flow; therefore, it is understood that the majority of sequential valve positioning is conducted at operational points which do not offer this minimizing effect with regard to throttling losses.

While there is a general awareness of the poor response with respect to operating turbine steam admission valves wide open and regulating boiler throttle pressure to govern load which is more commonly referred to as "sliding pressure" plant operation, some control system designers have continued to pursue this sliding pressure mode of operation by providing further improvement to the response thereof. One such control system is described in U.S. Patent 3,802,189 issued April 9, 1974 to T. W. Jenkins, Jr. Jenkins' system appears to provide a single point desired set point for a turbine control valve at a value preferably corresponding to a valve position near wide open. A rapid response to any increase in power generation demand is achieved by controlling the turbine control valve away from its steady state desired set point setting to a new position closer to wide open by a conventional turbine governor. As the actual valve position deviates from the desired set point value, the boiler throttle pressure set point is adjusted as a function of the position deviation to increase the boiler throttle pressure causing the power generation to increase beyond that demanded. Concurrently, the conventional turbine governor repositions the control valve until conditions exist which satisfy the requirements of the power generation being that demanded and the valve position being at the desired set point value. It appears that Jenkins' system controls power generation by sliding pressure in a boiler follow mode of operation permitting a faster response to power generation demand deviations as compared to a turbine follow mode of operation. However, it is understood

that in order to achieve this improvement in response, Jenkins must relinquish some efficiency by steady state positioning the control valve away from a wide open position such that the turbine governor may be capable of responding quickly to power generation demand increases by modulating the control valve temporarily closer to a wide open position until the boiler throttle pressure can be readjusted. Thus, in Jenkins' system, it is believed that the control valve is inefficiently positioned during the majority of plant operation.

From the foregoing discussion, it appears that further improvements to boiler-turbine load control operations may be achieved in the areas of minimizing the throttling losses of the steam admission valves over a greater portion of the governing load range while at the same time maintaining an acceptable responsiveness of the steam turbine governor to changes in power generation demand.

It is, therefore, an object of this invention to provide an improved system for minimizing power plant energy losses caused by steam flow valve throttling with a view to overcoming the deficiencies of the prior art.

The invention relates to a system for minimizing power plant energy losses substantially caused by steam flow valve throttling while maintaining said power plant at a desired power generation level, said power plant including a boiler for generating steam at a boiler throttle pressure which is governed by a pressure set point; a steam turbine having a plurality of steam flow valves for regulating the amount of generated steam therethrough; a valve control means governed by a reference signal corresponding to the desired power generation level to position said plurality of steam flow valves in a state according to a predetermined valve positioning pattern based on the value of said reference signal; and an electrical generator driven by said steam turbine to generate electrical energy, characterized in that said system comprises first means for adjusting said pressure set point based on a function of a selected one of a plurality of predetermined values of said reference sig-

nal, said predetermined values substantially corresponding to a minimum of valve throttling losses; and second means governed by said pressure set point adjustment to modulate said reference signal substantially to the selected value, 5 whereby the steam flow valves are positioned to regulate steam admission corresponding to the desired power generation level while providing a minimum of valve throttling losses.

10 In accordance with a preferred embodiment of the present invention, an efficient valve position controller is adapted for use in a steam turbine power plant for efficiently positioning a plurality of steam admission valves of the steam turbine to substantially effect a desired power generation level of the power plant. The power plant in- 15 cludes a boiler for generating steam to the steam turbine at a boiler throttle pressure that is governed by a pressure set point; a valve control means which is governed by a reference signal corresponding to the desired power generation level to position the plurality of steam admission 20 valves in a state according to a predetermined valve positioning pattern based on the value of a reference signal; and an electrical generator driven by the steam turbine to generate electrical energy. More specifically, a plurality of values of the reference signal are predetermined as being 25 related to efficient valve position states for regulating steam admission to the turbine. Accordingly, the controller adjusts the pressure set point based on a function of a selected one of the plurality of predetermined values of the reference signal and modulates the reference signal substan- 30 tially to the selected value as governed by the pressure set point adjustment. In particular, first and second predetermined values are segregated from the plurality of predetermined values based on their relationship to a present value of the reference signal and one of the first and second 35 predetermined values is selected based on the amount of pressure set point adjustment needed to govern the modulation of the reference signal from the present value substantially to the one predetermined value. The pressure set

point is adjusted in a direction as determined by the selected predetermined value until the reference signal is modulated to within a preset value of the selected predetermined value. In another aspect, the controller becomes
5 operative to adjust the throttle pressure set point to a predetermined value at times when the reference signal obtains a value indicative of the steam admission valves being substantially wide open.

The invention will become readily apparent from
10 the following description of exemplary embodiments thereof when taken in conjunction with the accompanying drawings, in which:

Figure 1 is a block diagram schematic of a steam turbine power plant incorporating the system according to
15 the present invention;

Figure 2 is a graph exemplifying heat rate losses with respect to power generation level (MW) substantially resulting from valve throttling losses in accordance with a predetermined valve grouping sequential positioning pattern
20 of the steam admission valves;

Figure 3 is a graph illustrating a typical boiler throttle pressure adjustment profile with respect to power generation level as determined by a plurality of predetermined valve position states;

Figure 4 is a block diagram schematic of a programmed digital computer embodiment suitable for use in the power plant of Figure 1;
25

Figure 5 is a graph illustrating the flow coefficient for various percentages of flow utilized in the programmed digital computer embodiment of Figure 4;
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Figure 6 is a graph illustrating valve lift as a function of steam flow for various total steam flow requirements utilized in the programmed digital computer embodiment of Figure 4;

Figure 7 is a graph relating boiler throttle pressure adjustment to a steam flow corresponding to the desired power generation level;
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Figure 8 is a simplified graphical illustration of

a typical predetermined valve grouping sequential positioning pattern based on a plurality of predetermined valve positioned states suitable for use in the embodiment of Figure 4;

5 Figure 9 is a flow chart characterizing the operation of a programmed digital computer according to one embodiment of the invention; and

10 Figure 10 is a functional block diagram schematic of an alternative embodiment of the invention suitable for use in the power plant depicted in Figure 1.

Referring to Fig. 1, there is shown a steam turbine power plant 10 which produces electrical energy at some desired power level to a system load 12. As part of the operation of the power plant 10, a conventional steam boiler system 14 provides steam at some regulated boiler throttle pressure, PTH, to a conventional steam turbine system 16 which is mechanically coupled to drive an electrical generator 18. The amount of steam conducted through the steam turbine system 16 is, at times, controlled by a plurality of governor valves GV1,...,GV8 which may be disposed in any number of conventional arrangements so as to permit either single valve or sequential valve arc admission operation. In the normal operation of the power plant 10, a conventional turbine controller 20 positions the plurality of governor valves GV1,...,GV8 for the purposes of admitting steam to the turbine 16 to increase the speed of the turbine 16 from turning gear to a speed which is synchronous to the system load 12, utilizing an actual speed measurement signal provided to the turbine controller 20 from a standard speed transducer 22. The governor valves GV1,...,GV8 are generally modulated to establish a state of synchronization between the generated electrical signal over power lines 24 and the electrical system load 12.

At synchronization, a set of main breakers 26 are closed to connect the output of the generator 18 with the system load 12 utilizing the power lines 24. Thereafter, the turbine controller 20 governs the electrical power generation of the generator 18 by positioning the plurality

of governor valves GV1,...,GV8 preferably in accordance with a function of a desired power generation value and a signal representative of the actual power generation level as measured from electrical power lines 24 and provided to the turbine controller by a conventional megawatt transducer 28. It is preferred for the purposes of this embodiment that the positioning of the governor valves GV1,...,GV8 be transferred to a sequential valve mode operation beyond a predetermined desired power generation level, say 37% for example, in order to reduce throttling losses resulting from the single valve mode of operation wherein all of the steam admission valves may be positioned partially opened. Concurrent to the turbine speed and load control as described hereabove, the boiler throttle pressure P_{TH} is controlled in either a boiler follow mode or a coordinated plant control mode by a conventional boiler pressure controller 30. A measurement of the pressure P_{TH} is provided to both controllers 20 and 30 from a typical pressure transducer 32 and is utilized thereby for purposes of trim correction and feedback control which will be described in greater detail hereinbelow.

While conventional load governing operation in the sequential valve mode offers a reduction in throttling losses over that of single valve mode operation, there still remains room for further reduction to minimize the throttling losses during the periods of load governing operation when each of the segregated value groups of the sequential valve pattern are exclusively operated in the partially opened position. A typical example of the heat rate losses which may occur during a sequential valve pattern is shown in the graph of Figure 2 for a 490 MW turbine-generator (2400 VSIG/1000°F./1000°F./2.5 in Hg) having 8 control valves and 5 sequential value points specified at 37.5%, 50%, 62.5%, 75% and 100% of load reference. For a better understanding of the details of operating a power plant such as that denoted by 10 as shown in Figure 1 in a sequential valve mode reference is made to the U.S. Patent 3,878,401 issued April 15, 1975 to Uri G. Ronnen. In the broadest

aspect of the preferred embodiment as shown in Figure 1, an efficient valve positioning unit 34 is coupled to both the turbine and boiler pressure controllers 20 and 30, respectively and is functionally operative to substantially reduce
5 the typical heat rate losses generally associated with sequential valve mode of operation.

According to one embodiment, the unit 34 may communicate with the turbine controller 20 over signal lines 33 to access therefrom information pertaining to a set of
10 predetermined sequential valve position ranges which have been determined to provide a minimum of throttling losses in the conventional load governing operation in the sequential valve mode. These valve position ranges may be similar to the optimum sequential valve position ranges determined by
15 the system described in the copending application, Serial No. 628,629, referenced to hereinabove. In addition, both the boiler pressure controller 30 and turbine controller 20 provide the efficient valve positioning unit 34 with their present operational status over signal lines 35 and 33,
20 respectively.

In accordance with this operational status, the efficient valve positioning unit 34 selects one of a plurality of predetermined sequential valve position ranges in which it desires the sequential valve position to operate
25 within and proceeds to adjust a boiler throttle pressure set point 36 which governs the boiler throttle pressure control within the boiler pressure controller 30 to render the control valves GV1,...,GV8 positioned within the selected predetermined sequential valve position range. This process
30 which is functionally provided by unit 34 may be repeated for each desired power generation operating point asserted by either the power plant operator locally or the automatic dispatching system remotely. An example of a resulting boiler throttle pressure profile with respect to load refer-
35 ence is shown in the graph of Figure 3. The turbine system used for plotting Figure 3 is similar in capacity and operating conditions as that used for illustration in Figure 2, and therefore, it is proposed that the heat rate losses

shown in Figure 2 as one example may be substantially eliminated through the operation of the effective valve positioning unit 34 in coordinating the control of both the boiler and turbine controllers 30 and 20, respectively. A
5 more detailed description of the efficient valve positioning unit 34 is provided hereinbelow.

In some installations, the conventional turbine controls 20 of the embodiment described in connection with Figure 1 may comprise a digital electro-hydraulic (DEH)
10 turbine control system for governing the load of the turbine power plant in a sequential valve mode. The operation of the DEH system includes the execution of a number of task oriented subroutines in accordance with a real time priority structure within a programmed digital computer to monitor
15 the status of the turbine and boiler systems 16 and 14, respectively, and control the turbine system 16 as a function of the monitored status. Accordingly, it was found suitable for this embodiment to incorporate the efficient valve positioning function 34 (see Figure 1) in a programmed
20 digital computer similar to the typical DEH as a programmed subroutine being executed in coordination with other essential subroutines as directed by the real time operating system of a DEH type controller. A simplified functional block diagram of a DEH type turbine controller 20 is depicted
25 in Figure 4 interfacing with the turbine control valves GV1,...,GV8, the boiler system 14 and boiler controls 30 using conventional digital-to-analog (D/A) and analog-to-digital (A/D) input/output (I/O) units.

Referring to Figure 4, the plurality of governor
30 valves GV1 through GV8 are controlled by an analog signal, which is applied from its associated digital-to-analog output device referred to at 40. A digital electrohydraulic turbine control system of the type described in U.S. Patent 3,878,401 is referred to generally at 42. Briefly, however,
35 the system 42 in its preferred form includes a programmed digital computer with a conventional analog input system such as that referred to at 44 and 46 to interface the system analog signals such as P_{TH} and MW, respectively, with

the computer at its input. Computer output signals are interfaced with external control devices such as the control valves GV1,...,GV8 and the boiler pressure controller 30 utilizing the digital-to-analog output devices 40 and 47
5 respectively. The system 42 also includes a conventional interrupt system to signal the computer when a computer input is to be executed, or when a computer output has been executed. An operator panel such as 43 provides for operator control, monitoring, testing and maintenance functions
10 of the turbine generator system. Signals from the panel 43 are applied to the computer through the contact closure input system; and computer display outputs are applied to the panel 43 through the contact closure and direct digital output systems. The input signals are applied to the com-
15 puter from various relay contacts in the turbine generator system through the contact closure input system. In addition, the digital electrohydraulic control system 42 not only receives signals from electric power, steam pressure, and speed detectors, but also from steam valve position
20 detectors and other miscellaneous detectors which are interfaced with the computer (see Figure 1). The contact closure outputs from the computer of the system 42 operate various system contacts, a data logger such as an electric typewriter, and various displays, lights and other devices
25 associated with the operator panel 43.

The program system for the computer is preferably organized to operate the control system 42 as a sample data system in providing turbine and plant monitoring and continuous turbine and plant control. The program system also
30 includes a standard executive or monitor program to provide scheduling control over the running of programs in the computer as well as control over the flow of computer inputs and outputs through the previously mentioned input/output systems. Generally, each program is assigned to a task
35 level in a priority system, and bids are processed to run the bidding program with the highest priority. Interrupts may bid programs, and all interrupts are processed with the priority higher than any task level. A more detailed expla-

nation of the program system as well as the digital electro-hydraulic turbine control system is disclosed in U.S. Patent No. 3,878,401, issued April 15, 1975, entitled "System and Method For Operating a Turbine Powered Electrical Generating Plant In A Sequential Mode", which patent is incorporated
5 herein by reference for a more detailed understanding thereof.

This system functions in general such that, when an operator panel signal is generated, external circuitry
10 decodes the panel input, and an interrupt is generated to cause a panel interrupt program to place a bid for the execution of a panel program which provides a response to the panel request. The panel program can itself carry out the necessary response or it can place a bid for a logic task
15 program to perform the response; or it can bid a visual display program to carry out the response. In turn, any of the above-mentioned programs may operate the contact closure outputs to produce the responsive panel display, such as the display for optimum valve position referred to at 56.
20 Periodic programs are scheduled by an auxiliary synchronizer program which in turn is bid periodically by the executive program. An analog scan program is bid periodically to select analog inputs for updating through an executive analog input handler. After scanning, the analog scan
25 program converts the inputs to engineering units, performs limit checks and makes certain logical decisions.

The system 42 generally includes a control program, a portion of which being referred to at 46, which functions to compute the positions of the control valves
30 GV1,...,GV8 to satisfy load demands during operator or remote automatic operation (ADS) and tracking valve position during manual operation. Generally, the control program shown as 46 is organized as a series of relatively short subprograms which are sequentially executed.

35 A load reference 48 is generated at a controlled or selected rate within the system 42 to meet the defined load demand. The control function denoted at 46 provides for positioning the control valves GV1,...GV8 so as to

satisfy the existing load reference with substantially optimum dynamic and steady-state response. The load reference value computed by the operating mode selection function, for example, is compensated for frequency participation by a proportional feedback trim factor (not shown) and for megawatt error by a second feedback trim factor shown at 46. The frequency and megawatt corrected load reference operates as a flow demand 50 for a valve management program 52. The output 50 of the speed and megawatt corrected load reference, functions as a governor valve set point which is converted into a percent flow prior to application to the valve management program 52.

With the utilization of the valve management system as described in the U.S. Patent 3,878,401, which is incorporated by reference herein, the governor valve control function provides for holding the governor valves closed during a turbine trip, holding the governor valves wide open during start-up and under throttle valve control (not shown), driving the governor valves closed during transfer from throttle to governor valve operation during start-up, reopening the governor valves under position control after brief closure during throttle/governor valve transfer and thereafter during subsequent load control.

During automatic computer control, the valve management program 52 develops the governor valve position demands needed to satisfy steam flow demand and ultimately the load reference; and do so in either the sequential or the single valve mode of governor valve operation or during transfer between these modes. Since changes in boiler throttle pressure P_{TH} can cause actual steam flow changes in any given turbine inlet valve position, the governor valve position demands may be corrected as a function of boiler throttle pressure P_{TH} variation. Governor valve position is calculated from a linearizing characterization in the form of a curve of valve position (or lift) versus steam flow. A curve valid for rated pressure operation is stored for use by the valve management program 52, and the curve employed for control calculations is attained by correcting the

stored curve for changes in load or flow demand, and preferably for changes in actual throttle pressure. Another stored curve of flow coefficient versus steam flow demand is used to determine the applicable flow coefficient to be used
5 in correcting the stored low-load position demand curve for load or flow changes. Preferably, the valve position demand curve is also corrected for the number of nozzles downstream from each governor valve. A more detailed explanation of such valve position versus steam flow, and flow coefficient
10 curve is provided in U.S. Patent 3,878,401.

In the sequential valve mode, which is represented by block 54 of Figure 4, the governor valve sequence is used, in determining from the corrected position demand 50, which governor valve or group or governor valves is fully
15 open, and which governor valve or group of governor valves is to be placed under position control to meet load reference changes. Position demands are determined for the individual governor valves; and individual sequential valve analog voltages 40 are generated to correspond to the calculated valve position demands.
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Referring to Figure 5, data representing flow coefficients is contained in the computer memory of the control system 42 based on the flow demand 50 computed by the digital electrohydraulic control system. The flow
25 demand value is shown on the abscissa of the curve and the flow coefficient is calculated along the ordinate. The flow coefficient is the ratio of actual flow at a flow demand over the theoretical flow if the orifice coefficient were equal to one. Once the ordinate for a particular flow
30 demand is calculated by use of the data in the computer memory, the stage flow coefficient is calculated, which is used to calculate the curve of Figure 6.

In Figure 6, the flow demand for each valve is represented as a percentage of total flow on the abscissa; and the lift of the steam inlet or governor valve is shown
35 on the ordinate, whereby the lift of the valve for a predetermined flow demand can be calculated. A curve 60 represents a dynamic characterization of operation of a control

or governor valve from its closed position to its fully open position to pass its proportionate share at approximately 64% of total steam flow. The corrected stage flow coefficient for critical flow (see Figure 5) is essentially equal to one for the typical installation described where flow demands are less than 64% of total flow. The exact transition point may vary between 60 and 70%, for example, from installation to installation depending upon the design of the governor valve. If the total flow demand is greater than that having a corrected flow coefficient of one, a different curve, such as that referred to at 61 for a total steam flow of 90%; and another curve referred to at 62 for a 100% total steam flow demand is calculated. Each curve, such as 60, 61 or 62, is composed preferably of five linear segments in order to facilitate ease of calculation and economy of memory space in the computer. The curves are calculated by multiplying the abscissa and the ordinate of each of the curves by the stage flow coefficient of Figure 5. The curves such as 60, 61, and 62 may be either calculated by the computer in accordance with the total steam flow demand or there may be a plurality of such curves stored in the computer with the appropriate curve being selected for particular steam flows. The curves of 60, 61, and 62 may also be modified dynamically for variations in the throttle pressure and also for variations in the number of nozzles under each valve, as described in the referenced patent 3,878,401. For each of the curves an FC flow point is calculated, above which a very high associated gain is required in order to maintain and linearize any action of the actuator for the control valve. Between such FC point and the fully opened position only approximately five to ten percent of the flow for that valve is controlled. Between such FC point and the fully closed position, the efficiency of the plant is reduced because of steam losses due to throttling. In calculating the FC point, the maximum steam flow that the valve is capable of admitting is calculated in accordance with the total steam flow demand. A predetermined percentage of such maximum flow, such as 92%, for

example, is the FC point.

The DEH control system 42 additionally includes a system 56 for indicating an optimum set of sequential valve position ranges during the sequential valve operating mode of the turbine power plant for the purposes of determining valve position settings offering minimum throttling losses. The system 56 operates by checking each of the steam inlet or governor valves GV1,...,GV8 in the sequence in which such valves are controlled to admit varying levels of steam flow to the turbine. In determining the fully open and fully closed positions for each of the valves, the system 56 utilizes the position demand 50 plus in some cases a small tolerance or deadband. In determining the position of the valve intermediate the fully open or fully closed position, the system 56 utilizes the flow demand for each valve Q which is calculated in accordance with a valve lift versus steam flow curve (see Figure 6). This is compared with a calculated electrical representation of an FC point for each valve, which point represents a percentage of maximum flow adjacent the end of the linear range of the valve prior to the valve going into the so-called high slope region of relatively unstable control. The FC point is calculated in accordance with a percentage GC1 of the maximum possible flow of the valve. The maximum possible flow for each such valve is determined in accordance with the steam flow versus valve lift curve (see Figure 6). The FC point also has a tolerance or deadband.

Each time the system 56 operates, it first effectively eliminates all flags which would indicate that the valves were in an optimum position. Then the system checks the operating mode to determine that the system is operating in the sequential valve mode. It then checks for each valve, as to whether or not the valve is within a fully opened deadband range; and if such is the case, the "valve open" flag is set and the program goes to the next valve in the sequence. If it is not fully opened, the system then checks to determine if the steam flow demand for the valve is greater than the calculated FC point. If such is the

case, the program 56 exists and starts from the beginning to check the complete sequence of valves. If the flow demand is not greater than the FC point, the system then checks to determine if the valve is within an FC point deadband range.

5 If such is the case, the "valve open" flag is set and the system goes on to check the next valve. If the valve is not in such range, the system then checks to determine whether or not the valve is in a fully closed position within the deadband range associated therewith. If such is the case,

10 the program then checks to determine if the "valve open" flag has been set by a previous valve; then the system continues with checking the next valve in the sequence. However, if the valve is neither in the closed position or the "valve open" flag has not been set, then the program

15 exits. Thus, each time a valve is determined not to be in one of the optimum positions, the program starts over again and eliminates all indications that any of the valves were in such optimum position. For a more detailed description of a typical optimum valve position system functioning in a

20 DEH turbine control system reference is made to the copending application Serial No. 628,629, which is incorporated by reference herein.

The efficient valve positioning system 34, as indicated above in accordance with a DEH control system

25 embodiment is implemented as a program subroutine within the DEH controller 42. The system 34 functions to coordinate the activities of the control program 46, the valve management program 52 and the optimum valve position program 56 with the boiler pressure controller 30 to provide an integrated mode of control therebetween. Under normal operation, the valve management program 52 provides information to the positioning system 34 in the form of a throttle pressure correction factor, valve flow characteristics and flow demand, for example. In addition, the optimum valve

30 position detection system 52 may provide to the positioning system 34 conditions relating to the optimum valve position status. Certain plant status such as single/sequential valve mode status, megawatt controller status and load

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change in progress status are also made available to the positioning system 34 as a result of the normal periodic execution of the logic program within the DEH system 42. To effect an in service condition of the positioning system 34, a pushbutton 59 located on the control panel 43 may be depressed. The status of the pushbutton 59 is detected by the DEH system 42, utilizing the standard panel interface and associated program supplied therewith, and is additionally made available to the positioning system 34.

The structure and operation of the efficient valve positioning system 34 may sufficiently be described by assuming a typical initial operating state of the steam turbine plant 10 which illustrates the sequential positions of the groupings of the control valves GV1,...,GV8 as a result of a recently enacted desired load change. Referring to the graph of Figure 7, the point denoted by 69 indicates the initial operating state of the turbine wherein the steam flow is denoted by F_3 and the boiler throttle pressure is denoted by P_3 . Because the control unit 46 (see Figure 4) remains operative during the functioning of the efficient valve positioning system 34, the control valves are positioned to keep steam flow substantially constant during any change in boiler throttle pressure. For this example then, the operation of the power plant 10 is maintained substantially along the vertical line of the graph of Figure 7 which intersects the abscissa at a steam flow F_3 . Therefore, any adjustment to boiler throttle pressure results in a new plant operating point along the vertical line denoted by the fixed steam flow F_3 .

Referring to the graph of Figure 8, a set of valve groups are presented in a predetermined sequential valve position opening pattern exemplifying the calculations performed by the valve management program 52 as described hereinabove. The encircled portions 70 through 75 of the graph are exemplary of a set of sequential optimum valve position ranges which may be predetermined from the operation of the optimum valve positioning detector 56. It is understood from the description provided above, that when

all of the valves are positioned in one of these predetermined ranges, a state of minimum throttling losses is anticipated. In the present assumed operating state (P_3 , F_3), the corresponding sequential valve positions are fixed by the interaction of flow line F_3 with the predetermined sequential valve position opening pattern and are denoted by the points 76, 77 and 78 wherein control valves GV1, GV2 and GV3 are wide open; GV4 and GV5 are partially opened at 77; and GV6, GV7 and GV8 are fully closed. The present valve positions at 76, 77 and 78 are not in a predetermined optimum valve position range. The closest optimum valve position ranges appear to be the encircled ranges at 71 and 72.

It is one purpose then of the efficient valve positioning system 34 to cause the valves to be repositioned in a selected one of the optimum valve position ranges by adjusting the boiler throttle pressure set point which is output from the DEH system 42 through the interface unit 40 over line 36 to a conventional steam pressure set point controller 80 located in the boiler control system 30 (see Figure 4). In turn, the controller 80 adjusts a conventional boiler firing control unit 82 to alter the conditions of the boiler 14 to cause the actual boiler throttle pressure P_{TH} as measured by the transducer 32 to converge to the adjusted value of the boiler throttle pressure set point 36. Consequently, any change in boiler throttle pressure affects the electrical power output of the plant which is reflected to the load controller 46 of the DEH system 42 via megawatt transducer 28 and A/D interface 46 (see Figure 4). Accordingly, the control valves GV1,...,GV8 are governed to maintain a fixed load by the control unit 46. Control unit 46 repositions the control valves according to the sequential valve patterns of the valve management program 52 until the efficient valve positioning unit 34 terminates its adjustment of the boiler throttle pressure set point 36 as a result of detecting that the sequential valve positioning pattern is in one of the optimum valve position ranges.

For a more detailed understanding of the efficient valve positioning program 34, a flowchart pertaining to its

sequential execution of operations is shown in Figure 9. The flowchart of Figure 9 will be described below in conjunction with the graphs of Figures 7 and 8 using the exemplary initial plant operating state (P_3 , F_3). Referring to

5 the flowchart of Figure 9, the efficient valve positioning program 34 begins with a plurality of logical decision making blocks 100, 102,..., 112, 114 to determine if a set of valid permissives for proper operation are satisfied. These conditions include, in respective correspondence to

10 the decision block 100, 102,..., 114, the following:

- (a) an optimum valve position condition;
- (b) not in sequential valve mode;
- (c) efficient valve positioning system not in service;
- 15 (d) megawatt controller not in service;
- (e) P_{TH} correction in service;
- (f) load change in progress; and
- (g) present actual throttle pressure value-set point value exceeds limit.

20 If the status of any of the aforementioned conditions are logically true indicating that an invalid condition exists, the efficient valve positioning program 34 may be prohibited from being executed during the present execution period. On the other hand, if the status of all the aforementioned con-

25 ditions are logically false indicating that a permissive state exists, then program execution is permitted to continue at block 116.

The calculations to select one of the optimum valve position ranges, which may be at 71 or 72 (see Figure

30 8) for the above described example, begins at block 116. Block 116 in cooperation with the valve management program 52 calculates a virtual flow value F_4 corresponding to the optimum valve position range which offers a greater virtual flow than the present flow demand, which is for the case at

35 hand at 72. For this calculation, the valve management program 52 may be requested to determine the throttle pressure P_4 (see Figure 7) based on the valve position settings of range 72 and the actual steam flow F_3 . Once P_4 is de-

terminated, the pressure correction portion of the valve management program 52 may be performed using the ratio of the pressure value P_4 and a predetermined value of rated throttle pressure to calculate a new flow demand value which is used as the virtual flow value F_4 . In the next block 118, the valve management program 52 is similarly requested to first calculate the pressure value P_2 corresponding to the optimum valve position range which offers a lower virtual flow than the present flow demand, which is for the case at hand at 71, and then calculate the virtual flow F_2 using the operating point (P_2, F_3) in its processing of pressure correction.

Before continuing, it should be explained that the adjustment of the boiler throttle pressure set point is limited by upper and lower pressure set point values, P_1 and P_5 , respectively, which may be conventionally entered into the DEH system 42 through the control panel 42 (see Figure 4). The values P_1 and P_5 are made available to the efficient valve positioning program 34 from the DEH system memory upon request. Thus, in the next program execution block 120, the minimum virtual flow F_1 is calculated using the pressure correction portion of the valve management program 52 based on the upper limit operating point (P_1, F_3) . The following block 122 results in the calculation of maximum virtual flow F_5 with similar use of the valve management program 52 given the lower limit operating point (P_5, F_3) .

Equipped with the complement of virtual flow values F_1, F_2, F_4, F_5 , the program execution continues at block 124 to begin the selection of one of the optimum valve position ranges. In block 124, it is decided which of the virtual flow values F_2 or F_4 is closer to the present flow value F_3 . If F_4 is closest to F_3 , execution continues at block 126 where it is decided whether F_4 is greater or less than the maximum limit flow value F_5 . If F_4 is less than F_5 , block 128 decrements the throttle pressure set point valve by a predetermined amount ΔP_D . The rate at which the throttle pressure is decreased is generally dependent on the

frequency at which the program 34 is executed and the predetermined amount ΔP_D . In the execution of blocks 124, 126 and 128; the program 34 has selected optimum range 72 and with each program execution decrements the boiler throttle pressure set point to affect the throttle pressure through the boiler controls 30 to cause the load controller 46 to react and position the valves within the optimum valve position range 72, for example. The program continues executing blocks 124, 126 and 128 to decrease the boiler throttle set point at the desired rate until the valve positions are within the range at 72. This condition, detected at the initial block of programming at 100, terminates the execution of program 34 by the DEH system 42 preventing any further decrease in set point 36 until the next desired load change is performed which will displace the valves outside an optimum valve position range.

In the event that either the value of F_4 is found to be greater than the maximum limit value F_5 , which is an unallowable and invalid state, or the value of F_2 is closest to the present flow value F_3 as detected by blocks 126 or 124, respectively, the program execution continues at block 130 wherein it is determined whether F_2 is greater or less in value than the minimum limit F_1 . If F_2 is greater in value than F_1 , the program 34 increments the throttle pressure set point by another predetermined amount ΔP_u using block 132. The increase rate of the throttle pressure set point is set by the value selected for ΔP_u and the frequency of execution of block 132. In the execution of blocks 124, 130 and 132, the program 34 has selected optimum valve position range 71, for example, and with each program execution increments the boiler throttle pressure set point at the desired rate to similarly cause the valves to be positioned within the optimum valve range 71. This condition is detected at block 100 to direct program execution to bypass further adjustment of throttle pressure set point which will remain at its last incremented value until another desired load change is performed which causes the valve positions to be displaced outside of an optimum valve position range.

In the event that the value of F_2 is found to be closest to the present flow value F_3 (124), but the value of F_2 is further found to be less than the minimum flow value F_1 , which is also an unallowable and invalid state (130),
5 then the program execution continues at block 134 wherein it is determined whether F_4 is less than or greater than the maximum limit flow value of F_5 . If F_4 is less than F_5 , then the throttle pressure set point will be similarly decreased at the desired rate to bring the valves into the optimum
10 range 72. Otherwise, the program 34 is exited and the pressure set point remains unchanged.

It is understood that the exemplary initial operating point (P_3, F_3), chosen to describe the embodiment shown in Figures 4 through 9 may be any practical value within the
15 operating limitations of the power plant 10 which may exist after a desired load change and that the efficient valve positioning unit 34 will operate automatically as described hereinabove to select one of the predetermined optimum value position ranges which offer a minimization to throttling
20 losses and adjust the throttle pressure set point to render a sequential valve position setting within the selected optimum valve position range. It is further understood that the flowcharts of Figure 9 are provided in the present specification merely to illustrate one way in which the
25 efficient valve positioning system 34 may be programmed in a DEH system embodiment and should not be considered as limiting to the scope of applicant's invention.

In other power plant installations, the conventional turbine controls 20 (see Figure 1) are embodied with
30 analog electronics in lieu of a programmed digital computer. An alternate embodiment for use in these installations is shown in Figure 10. Generally, these analog type turbine valve controllers comprise a conventional turbine master manual/automatic (M/A) stations 200 which normally receives
35 a total steam flow demand signal 202 generated from either a load demand computer or a plant master unit (neither shown). In automatic mode, the M/A station 200 may control the operation of a conventional turbine load reference motor 204

utilizing a set of increase and decrease signals 206 and 208, respectively, in accordance with the value of the steam flow demand signal 202. In manual mode, the M/A station 200 permits an operator to manually operate the increase and decrease signals 206 and 208 using pushbuttons located on a control panel (not shown), for example. The load reference motor 204 may be mechanically coupled to drive an analog signal generating device 210, such as a motor driven potentiometer, to produce a signal 212 which is representative of the total steam flow reference from the turbine unit 16 (see Figure 1). A conventional servo amplifier 214 may be coupled to each control valve GV1,...,GV8 to control the positions thereof. The servo amplifiers 214 may be offset adjusted to provide a desired sequential valve control pattern and may be characterized by a predetermined set of gains which are automatically adjusted to yield the steam flow vs. valve position transformation required to control valve position in accordance with the desired sequential value control pattern. To correct for possible inaccuracies in the open loop characterization of the servo amplifiers 214, a megawatt feedback trim correction 215 is provided, in some cases, to compensate a turbine load demand signal 216 generated from a plant master or load demand computer unit, for example. The megawatt feed trim corrector 215 is normally a proportional plus integral controller having as inputs the turbine load demand signal 216 and an actual load signal as measured by the megawatt transducer 28. The trim corrector 215 generates a trim signal 218 which increases or decreases the plant load demand signal 216 utilizing a summer function 220.

In relation to this alternate embodiment, the efficient valve positioning unit 34 (see Figure 1) comprises a plurality of deviation detectors of which three deviation detectors are shown at 224, 226, and 228 each having associated therewith a predetermined efficient valve position setting 230, 232 and 234, respectively, as one input. The total steam flow reference signal 212 is coupled to the other input of each of the deviations detectors 224, 226 and

228 and the respective output signals thereof 236, 238 and 240 are coupled to both a function 242 which determines the closest efficient valve point above a present value of the steam turbine flow reference signal 212 and a function 244
5 which determines the closest efficient valve point below the present value of the steam turbine flow signal 212. An output signal 246 of the function 242 is coupled as one input to a difference function 248 and to a comparator circuit 250 which is operative to detect that the valves are
10 positioned at one of the predetermined efficient valve position settings. An output signal 252 of the function 244 is coupled as one input to another difference function 254 and to a comparator circuit 256 which is operative to detect that the control valves GV1,...,GV8 are positioned at one of
15 the predetermined efficient valve position settings. A digital output signal 258 provided from comparator circuit 250 is supplied to one input of an OR function 260 and an inverted state of the digital signal 258 is provided to one input of an AND function 262. Likewise, a digital output
20 signal 264 from the comparator circuit 256 is supplied to the other input to the OR function 260 and an inverted state of the signal 264 is coupled to one input of an AND function 266.

Within the positioning unit 34 is included an
25 arrangement of logical gating functions to determine a permissive operational status based on logical variables 33 indicating the status of the turbine controller 20. Digital inputs to an AND gate function 268 include the following:

- (a) load feedback in service (269);
- 30 (b) MW controller in service (270);
- (c) pressure not ramping (271); and
- (d) turbine control in auto mode (272).

The output of gate 268 may be used as one input of an AND gate function 274 and in the inverted state used as one
35 input of an OR gate function 276. The other input 278 to the AND gate function 274 may be applied from a pushbutton (operator set) generally located on an operator's control panel (not shown). Similarly, the other input 280 may be

provided from another pushbutton (operator reset) which may also be located on an operator's control panel. The outputs of gates 274 and 276 provides the set and reset inputs of a conventional flip-flop 282, the output of which is connected to one input of an AND gate function 284. The other input 286 to the AND gate 284 may come from a plant load demand generator and is indicative of the status of load change in progress. The output signal 288 provides an in service permissive signal to another input of both AND gates 262 and 266.

During most of the steam flow range, the outputs of the AND gate functions 262 and 266 control the incrementing and decrementing of the boiler throttle pressure set point through OR gates 290 and 292 and over signal line outputs 294 and 296, respectively. The signals 294 and 296 are input to a pressure set point adjuster 298 which in the preferred embodiment may be an integrating type function with a selectable rate. A pressure set point adjustment signal 300 from the adjuster 298 is supplied to a window comparator function 302 and compared with predetermined maximum and minimum pressure set point values, P_{MAX} and P_{MIN} , respectively. Signals 304 and 306 are indicative of maximum and minimum limiting conditions and are provided to the adjuster 298 to prohibit further adjustment of the boiler throttle pressure set point. The maximum P_{MAX} and minimum P_{MIN} set point values are additionally provided to one input of the difference functions 308 and 309, respectively. The other input to the difference functions 308 and 309 is the generated pressure set point 300. The output signals 310 and 312 of the difference functions 308 and 309 correspond to the amount of pressure set point signal remaining before the maximum or minimum limiting conditions are reached. These signals 310 and 312 are coupled to the other input to the difference functions 248 and 254, respectively. A window comparator 314 with adjustable dead-band ranges receives the outputs from the difference functions 248 and 254 and decides if a pressure set point increment or decrement is required by either setting a signal to

one input of gate 262 true or setting a signal to one input of gate 266 true, respectively.

In this alternative embodiment, a predetermined plant normal boiler throttle set point value is provided to one input of a summator 316 from a signal line designated by 35. The pressure set point adjustment value 300 derived from the adjuster 298 is added to the plant normal pressure set point 35 in the summer 316 to generate a composite boiler throttle pressure set point 36 which is supplied to the conventional boiler control system 30 as shown in Figure 1. In addition, the set point adjustment value 300 is operated on by a function at 318 which may be comprised of at least one gain and may include phase compensation as related to the plant dynamics. The functional circuit 318 yields a signal 320 which is used to preferably multiply (324) the compensated plant load demand signal 322 to yield a turbine steam flow demand signal 202 which is corrected for the deviation 300 in pressure at point 36 from the predetermined plant normal pressure set point 35.

In addition to the above described structure, the alternative embodiment additionally includes a full load detector function comprising a comparator function 326 which compares the total steam flow reference signal 212 with a predetermined threshold value 327, say 95%, for example. The comparator output signal 328 is supplied to one input of a set of AND gate functions 330 and 332 and an inverted signal 328 is provided as the fourth input to the AND gate functions 262 and 266. The second inputs of the AND gates 330 and 332 are derived from a window comparator function 334 which compares the boiler pressure adjustment set point signal 300 with another predetermined value 335, preferably close to 0%. The outputs of the AND gates 330 and 332 are supplied to the other inputs of the OR gate functions 290 and 292, respectively.

In describing the operation of this alternative embodiment, it is assumed that a plant operating point initially exists which suggests a total steam reference value 212 which is not at one of the at least three effi-

cient valve point settings 230, 232 and 234. The deviation detectors 224, 226 and 228, which may be conventional differential amplifier configurations, compute the differences between the present value of total steam reference 212, which is representative of the present valve point setting, and each of the efficient valve point settings. These calculated differences 236, 238 and 240 may be scaled in such a manner as to be representative of the pressure set point adjustments required to move the valves to the correspondingly associated efficient valve set point setting. The smallest amplitude of the positive difference signals, which may be indicative of the adjustment in boiler throttle pressure set point required to reach the closest efficient valve point above the present valve point setting, is selected using function 242 and the smallest amplitude of the negative difference signals, which may be indicative of the adjustment in throttle pressure set point required to reach the closest efficient valve point below the present valve point setting, is selected by function 244. Functions 242 and 244 may be commonly implemented with an arrangement of limiters, absolute and low-select circuits which are of a conventional design. The smallest positive difference amplitude (246) is subtracted in 248 from the signal 310 which is representative of the amount of adjustment pressure set point increase allowed before reaching the preset max. limit P_{MAX} . The smallest negative difference amplitude (252) is subtracted in 254 from the signal 312 which is representative of the amount of adjustment pressure set point decrease allowed before reaching the preset minimum limit P_{MIN} . The window comparator 314 determines which of the two difference circuits 248 and 254 has computed the smaller positive amplitude and enables the correspondingly associated AND gate 262 or 266 to increase or decrease the pressure set point adjustment signal 300 accordingly. For example, if the status of operation exists that an in service operation is permitted (288) and a valve efficient point has not been reached (258 and 264) and the steam flow reference signal is not close to full load, then when the

output signal of the difference function 248 has a smaller positive amplitude than the output signal of the difference function 254, a request to increase the pressure set point adjustment 300 is conducted through AND gate 262, OR gate 5 290 and over signal line 294 to the integrating function 298. Likewise, if the output of 254 has the smaller positive difference, the comparator 314 requests a decrease in the pressure set point adjustment signal 300 conducted through AND gate 266, OR gate 292 and over signal line 296 10 assuming the same permissive status conditions exist as described above.

The difference functions 248 and 254 essentially compares the amount of pressure set point adjustment remaining for an allowable pressure set point state against the 15 amount required to achieve the closest predetermined efficient valve point setting and allows a pressure set point adjustment for reaching the closest efficient valve point setting to occur if that adjustment is within allowable limits (positive signal amplitude). If both pressure set 20 point adjustments are allowable as may be indicated by positive amplitude signals resulting from both difference functions 248 and 254, then window comparator 314 selects the lowest positive amplitude signal to determine the direction in which to adjust the pressure set point. Otherwise, 25 the window comparator 314 only accepts the positive amplitude signal and directs the adjustment of the pressure set point accordingly.

The pressure set point adjuster 298 modifies the set point adjustment signal 300 as directed by the increment and decrement status of the signal lines 294 and 296, respectively. The change in the signal 300 is reflected in 30 the composite throttle pressure set point 36 which directs the boiler controls 30 to alter the firing conditions of the boiler 14 to converge the boiler pressure P_{TH} as measured by 35 transducer 32 to the set point 36 (see Figures 1 and 4). In addition, the change in the set point adjustment signal 300 which is representative of the deviation of the plant normal pressure set point 35 governs the modulation of the com-

pensated load demand signal 322 in accordance with the function designated at 318 and the multiplication performed at 324 to compute the new position settings for the turbine control valves required to achieve efficient valve point setting. It appears that this feedforward type control does not rely on an interaction in the boiler-turbine-generator process to cause movement of the control valves and for this reason, it is believed that it minimizes process errors in the megawatt generation and the need to disrupt the boiler 14 by temporarily over or underfiring the fuel for purposes of changing its stored energy. In this preferred embodiment, then, the multiplier 324 operates to change the proportionality relationship between the compensated plant load demand signal 322 and the reference signal 212 in accordance with a deviation in pressure set point from the normal plant pressure set point 35. As an example of this control operation, suppose the gain of the multiplier 324 is set at one for the case in which there is no pressure set point deviation 300 from the normal plant pressure set point 35, now as the pressure set point 36 is adjusted above normal, the gain as characterized by multiplier 324 is decreased based on the signal 320 representative of the function of the deviation of the pressure set point above the normal plant set point. Therefore, as the pressure set point is adjusted to increase as described hereinabove, the total steam flow demand 202 and correspondingly the reference signal 212 are corrected concurrently therewith to cause the turbine control valves GV1,...,GV8 to close a proportional amount in a direction towards the selected efficient valve point setting.

As the control valves are positioned by the steam flow reference signal 212 at an efficient valve point setting, the comparators 250 and 256 detect substantially zero difference signals at 246 and 252, respectively. The output signals 258 and 264 of the comparators are indicative of the valves being positioned at an efficient valve point setting and may affect the output of the OR gate 260 to light a lamp 400 which may be disposed on the operator's control panel to provide the plant operator with this valve status. In

addition, the inverted signals 258 and 264 disable AND gates 262 and 266 from supplying increase and decrease adjustment signals to the pressure set point adjuster 298. The pressure set point adjustment 300 remains at its present value
5 until another desired load change is enacted resulting in repositioning the control valves outside of an efficient valve point setting.

This alternative embodiment has the additional feature of disabling the efficient valve point positioning
10 control as the turbine steam flow reference 212 attains a value substantially close to 100% which is an indication that all of the control valves are near a wide open state. More specifically, the reference signal 212 is compared with the predetermined set point 327 in comparator 326. As the
15 reference signal 212 becomes greater than the set point 327, the signal 328 enables AND gates 330 and 332 and disables AND gates 262 and 266. In this state, the adjustment of the throttle pressure set point is controlled by the window comparator 334 rather than the window comparator 314. The
20 pressure set point 36 is adjusted toward the plant normal pressure set point 35 by reducing the pressure set point adjustment signal 300 to substantially zero (i.e. set point 335). Therefore, as the control valves are positioned substantially close to a wide open condition, the boiler
25 throttle pressure is controlled to the plant normal operating state to optimize overall plant performance.

While the functional block schematic diagram of Figure 10 has been described in connection with electronic hardware such as amplifiers, limiters, absolute and low
30 limit select and logic circuits, it is understood that these functions may be performed equally as well in a programmed microprocessor or a combination of both.

What we claim is:

1. A system for minimizing power plant energy losses substantially caused by steam flow valve throttling while maintaining said power plant at a desired power generation level, said power plant including a boiler for generating steam at a boiler throttle pressure which is governed by a pressure set point; a steam turbine having a plurality of steam flow valves for regulating the amount of generated steam therethrough; a valve control means governed by a reference signal corresponding to the desired power generation level to position said plurality of steam flow valves in a state according to a predetermined valve positioning pattern based on the value of said reference signal; and an electrical generator driven by said steam turbine to generate electrical energy, characterized in that said system comprises first means for adjusting said pressure set point based on a function of a selected one of a plurality of predetermined values of said reference signal, said predetermined values substantially corresponding to a minimum of valve throttling losses; and second means governed by said pressure set point adjustment to modulate said reference signal substantially to the selected value, whereby the steam flow valves are positioned to regulate steam admission corresponding to the desired power generation level while providing a minimum of valve throttling losses.

2. A system in accordance with claim 1 characterized in that the first means includes: means for segregating a first and a second predetermined value from the plurality of predetermined values based on their relationship to a

present value of the reference signal which is other than one of the predetermined values; and means for selecting one of said first and second predetermined values based on the amount of pressure set point adjustment substantially sufficient to govern the modulation of the reference signal from said present value to said one predetermined value.

3. A system in accordance with claim 2 characterized in that the first and second predetermined values are the closest of the plurality of predetermined values above and below the present value of the reference signal, respectively.

4. A system in accordance with claim 2 characterized in that the one of the first and second predetermined values having the lower amount of pressure set point adjustment substantially sufficient to govern the modulation of the reference signal from the present value thereto becomes the selected value if the pressure set adjustment associated therewith is within predetermined pressure set point adjustment limitations, said other of the first and second predetermined values becoming the selected value otherwise.

5. A system in accordance with claim 1 characterized in that the reference signal is proportionately related to the desired power generation of the plant; said proportionality relationship is governed by the pressure set point adjustment; the pressure set point adjustment constitutes a deviation in value from a preset pressure set point value; and said pressure set point value deviation governs said proportionality relationship between the reference signal and the desired power generation level.

6. A system in accordance with claim 1 characterized in that the adjustment of the pressure set point by the first means is inhibited by an indication that the reference signal is above a threshold value which is indicative of the steam admission valves being substantially wide open.

7. A system in accordance with claim 2, 3 or 4, characterized in that the steam flow valves are organized to regulate flow in predetermined valve groupings operative according to a sequential pattern based on the value of said

reference signal; and wherein the first and second predetermined values are determined in relation to said valve grouping sequential pattern of operation.

- 5 8. A system for minimizing power plant energy bases substantially as hereinbefore described with reference to, and as shown in the accompanying drawings.

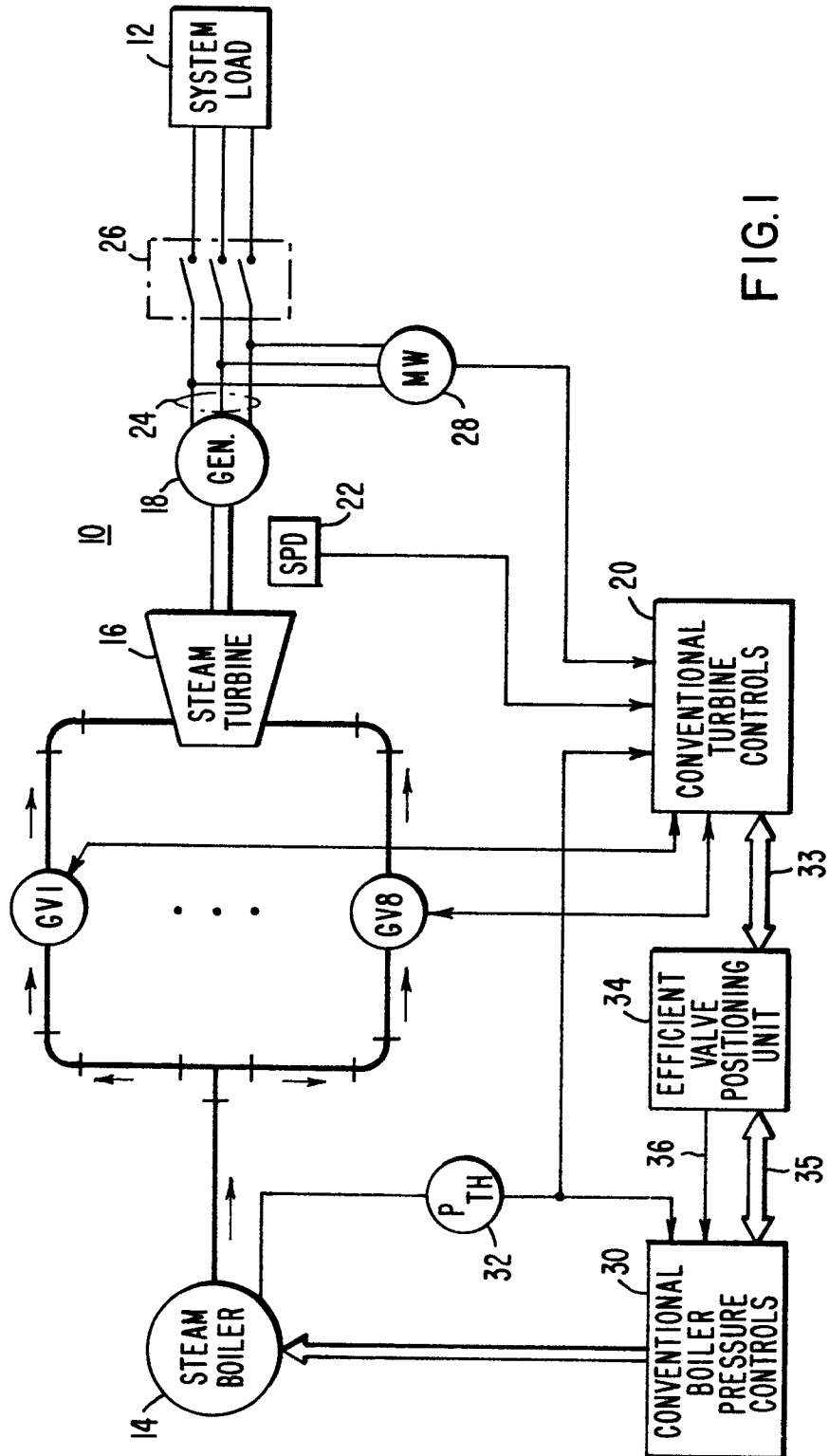


FIG. 1

FIG.2

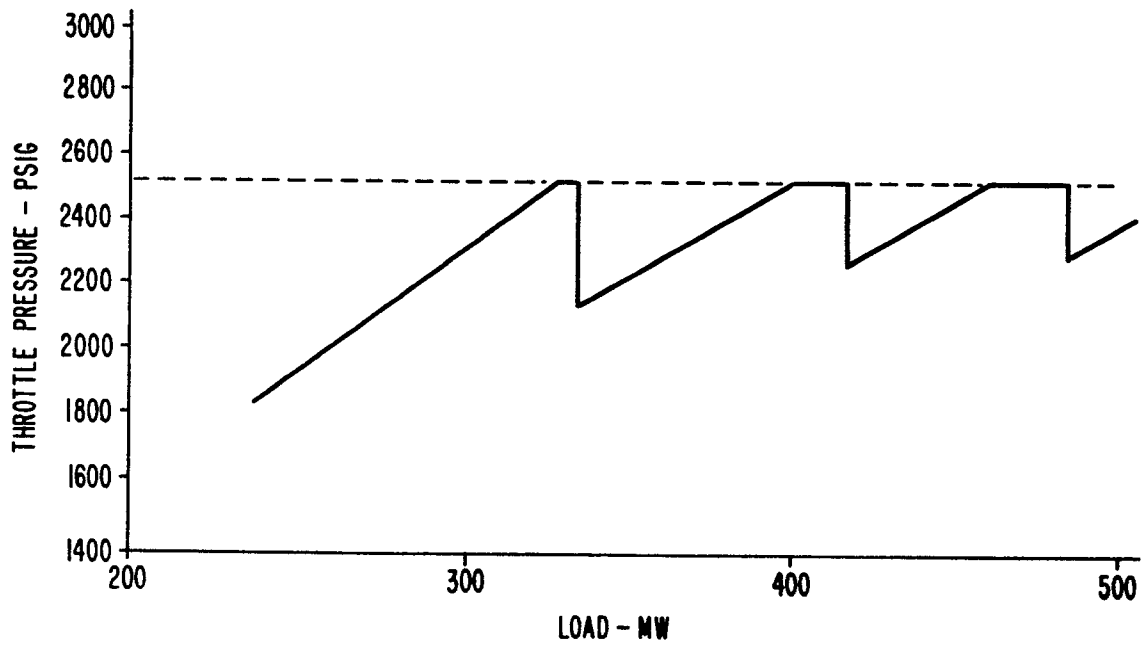
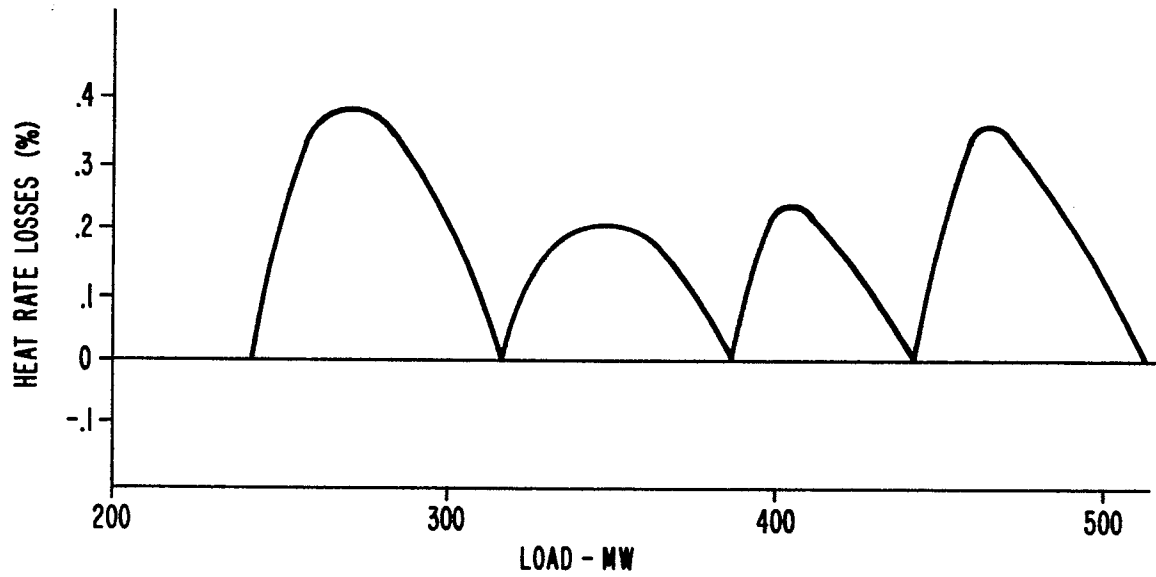


FIG.3

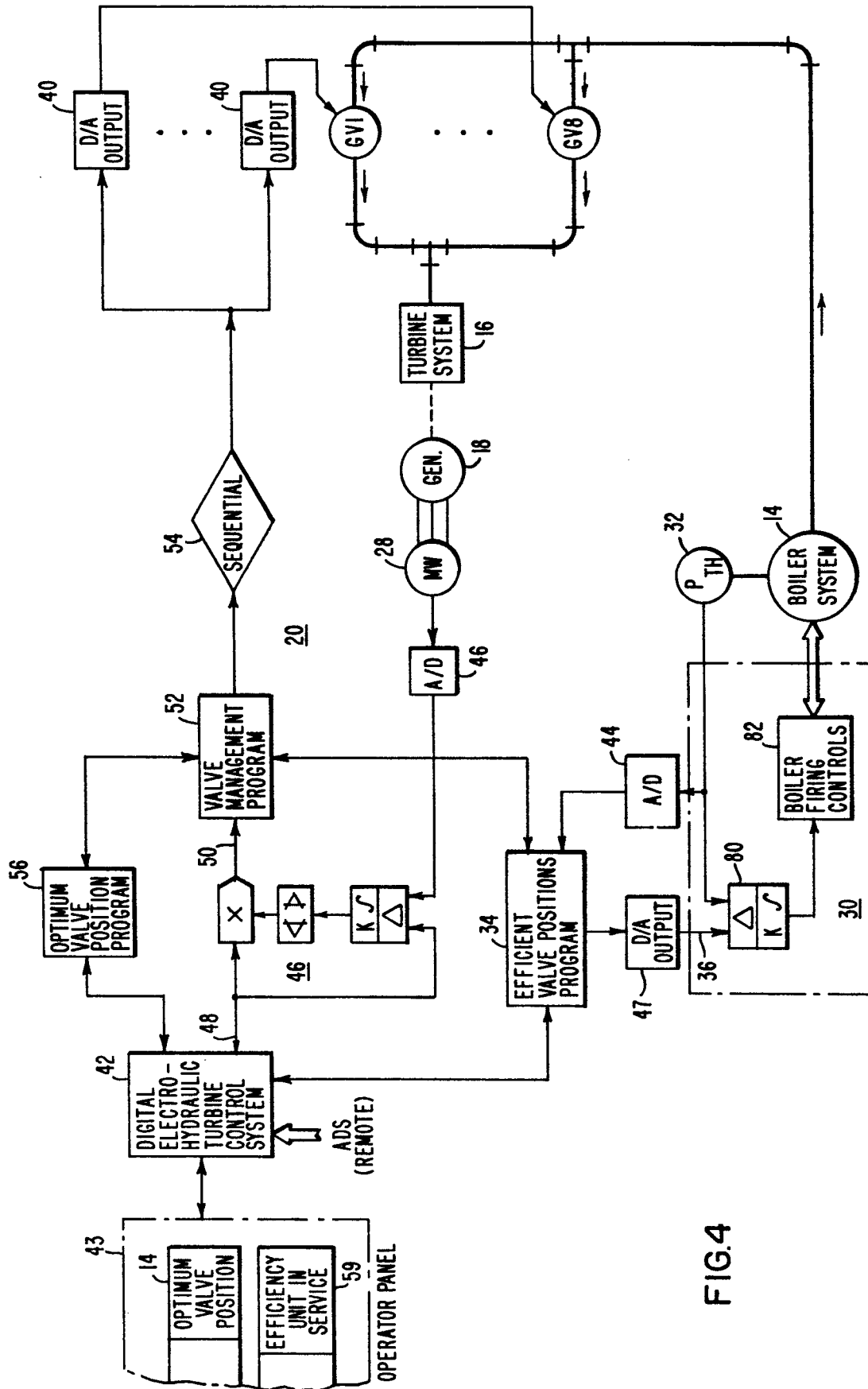


FIG. 4

FIG.5

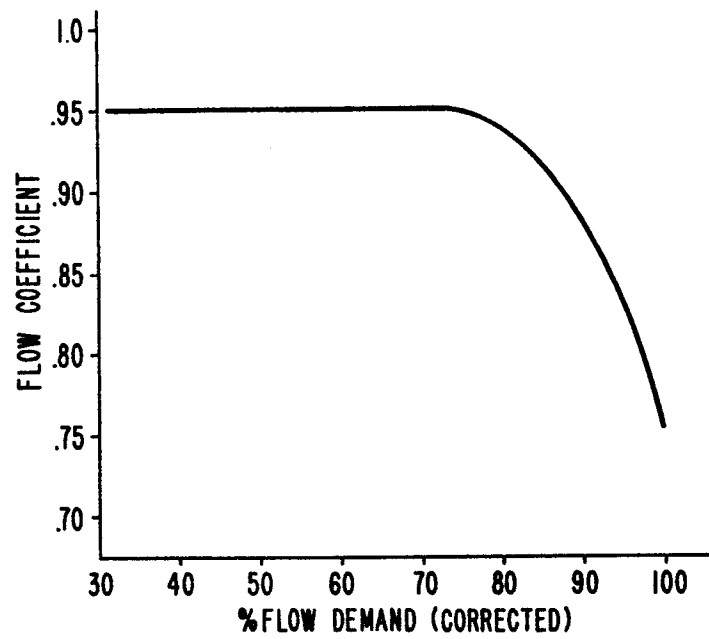
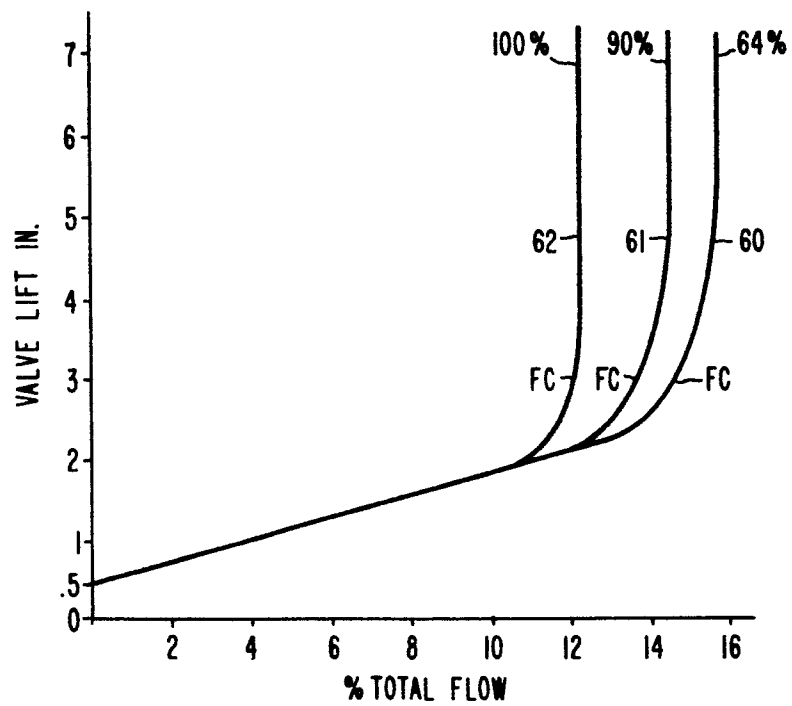


FIG.6



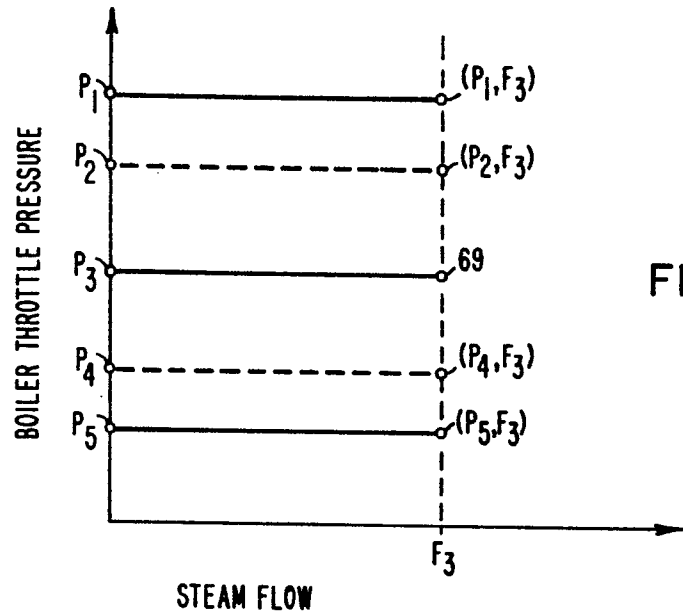


FIG. 7

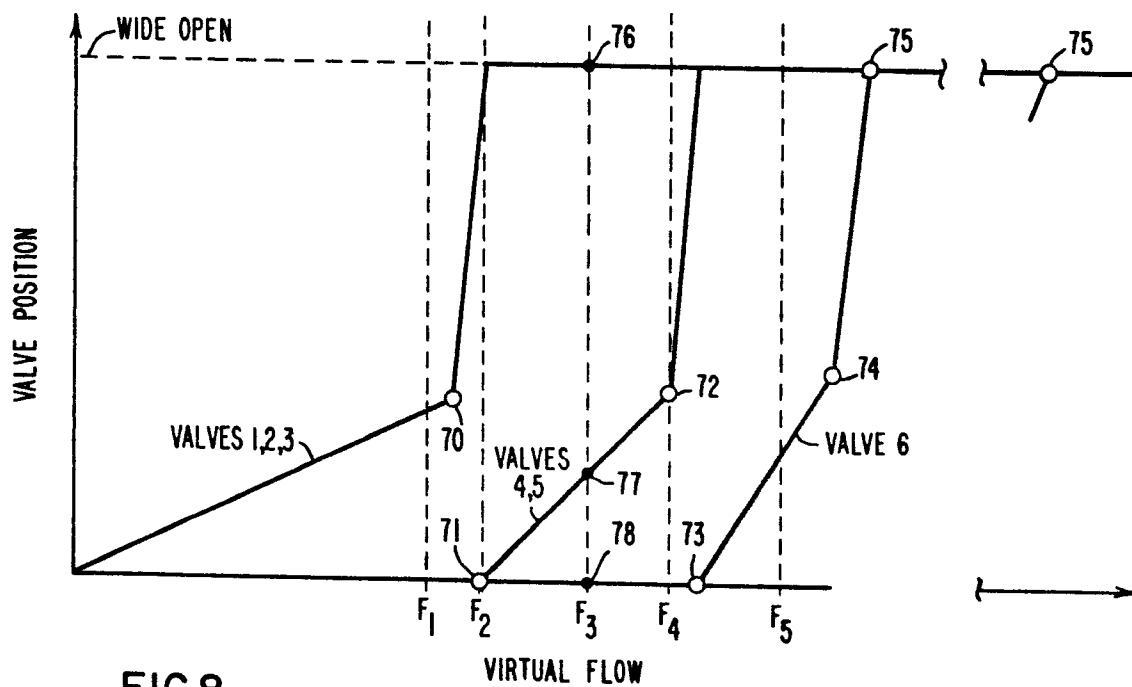
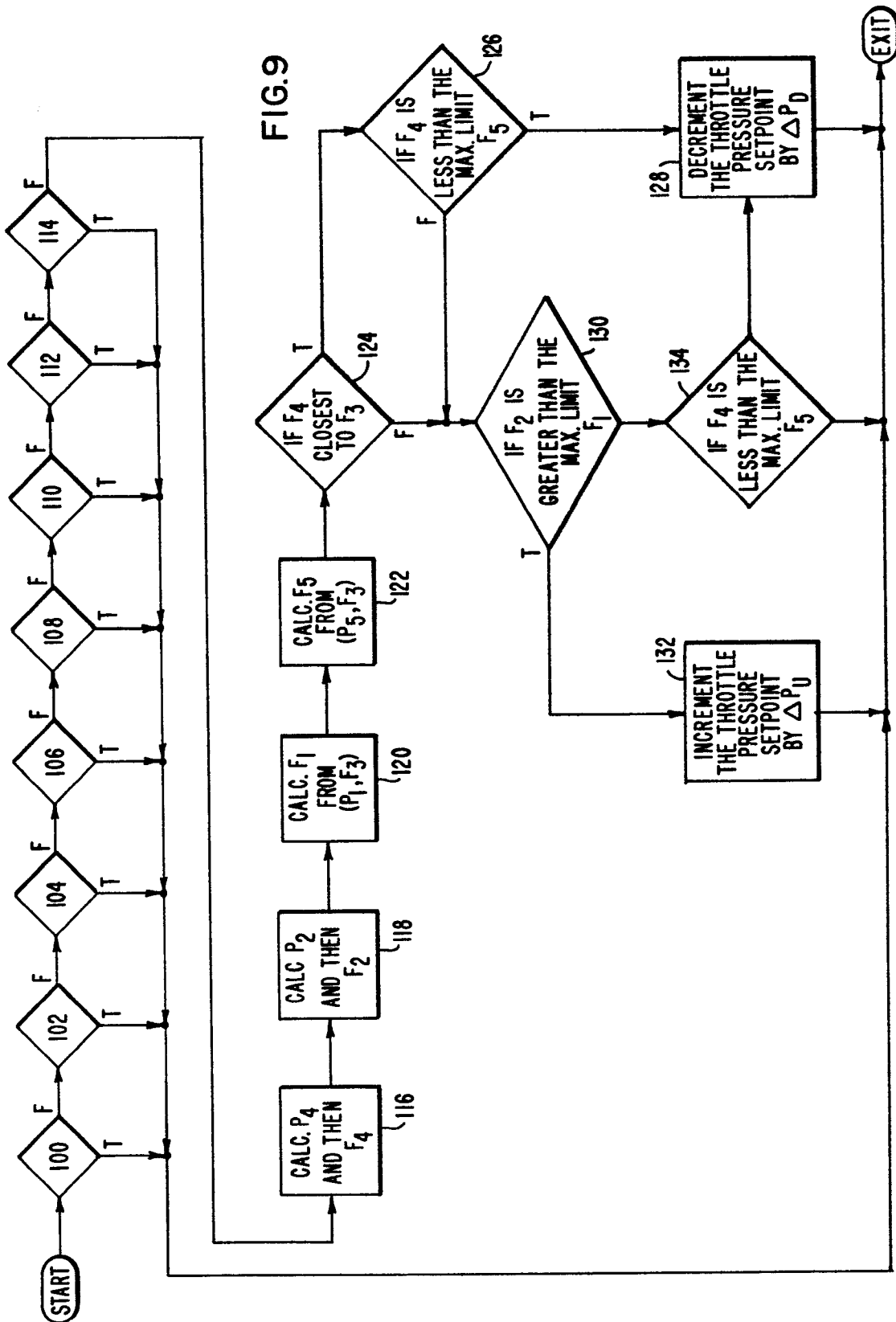


FIG. 8





DOCUMENTS CONSIDERED TO BE RELEVANT			CLASSIFICATION OF THE APPLICATION (Int. Cl.)
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	
E	<u>FR - A - 2 381 172 (HITACHI)</u> * Figures 1,16,3,4; page 9, line 18 - page 10, line 25 * --	1,5	F 01 D 17/18 F 01 K 13/02
AP.	<u>US - A - 4 088 875 (WESTINGHOUSE)</u> * Figures 1,4,5; column 3, lines 10-33; column 7, line 48 - column 8, line 36 * ----	1	TECHNICAL FIELDS SEARCHED (Int.Cl.) F 01 K F 01 D F 22 B
			CATEGORY OF CITED DOCUMENTS X: particularly relevant A: technological background O: non-written disclosure P: intermediate document T: theory or principle underlying the invention E: conflicting application D: document cited in the application L: citation for other reasons
☒ The present search report has been drawn up for all claims			&: member of the same patent family, corresponding document
Place of search The Hague		Date of completion of the search 19-06-1979	Examiner KOVAC