United States Patent

[11] 3,588,265

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[54] **SYSTEM AND METHOD FOR PROVIDING STEAM TURBINE OPERATION WITH IMPROVED DYNAMICS**
25 Claims, 12 Drawing Figs.

[52] **U.S. Cl.** **415/1,** 415/17
[51] **Int. Cl.** **F01d 21/14**
[50] **Field of Search** .... 60/105; 253/39 (Inquired); 137/16, 20, 26, 29, 30, 17, (Inquired)

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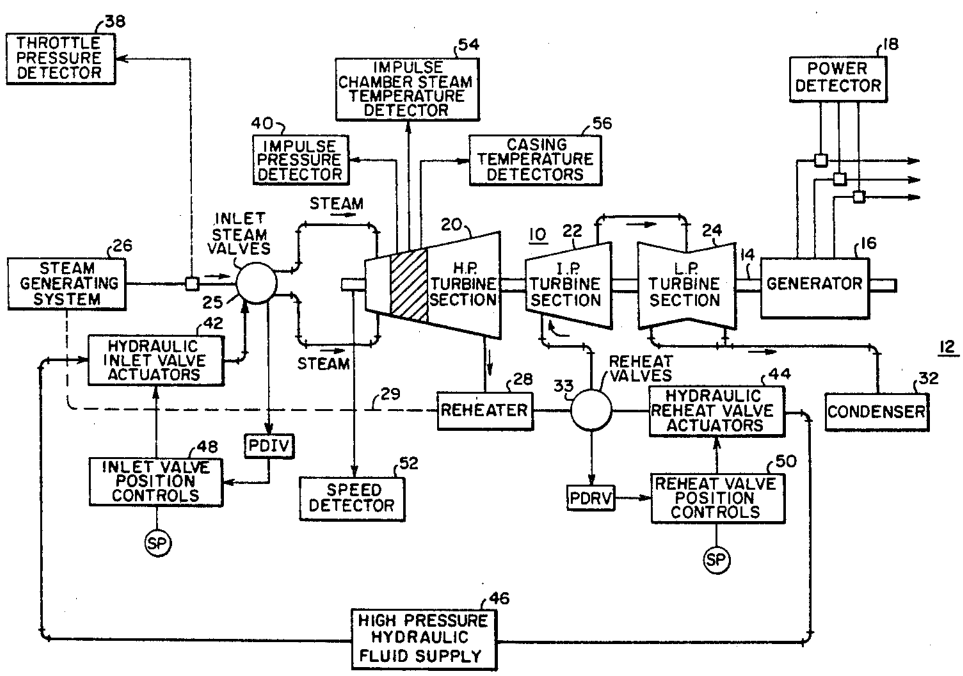
ABSTRACT: A programmed digital computer control system for an electric power plant steam turbine responds to turbine impulse chamber steam temperature and other input variables to control turbine inlet steam flow within constraint limits that prevent excessive rotor loading and allow controlled accumulation of turbine rotor plastic strain fatigue according to predetermined fatigue accumulation standards.



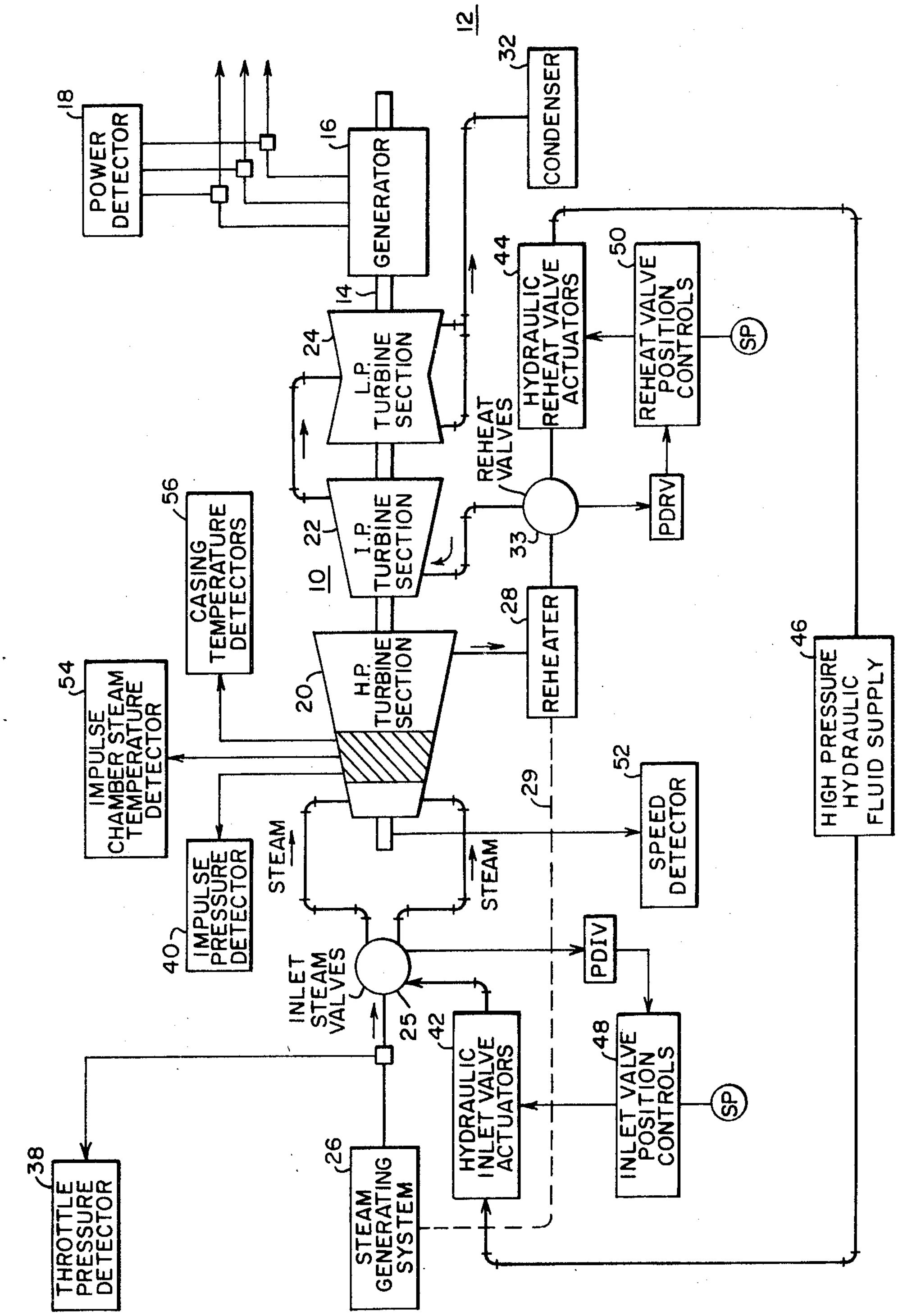


FIG. 1.

WITNESSES:
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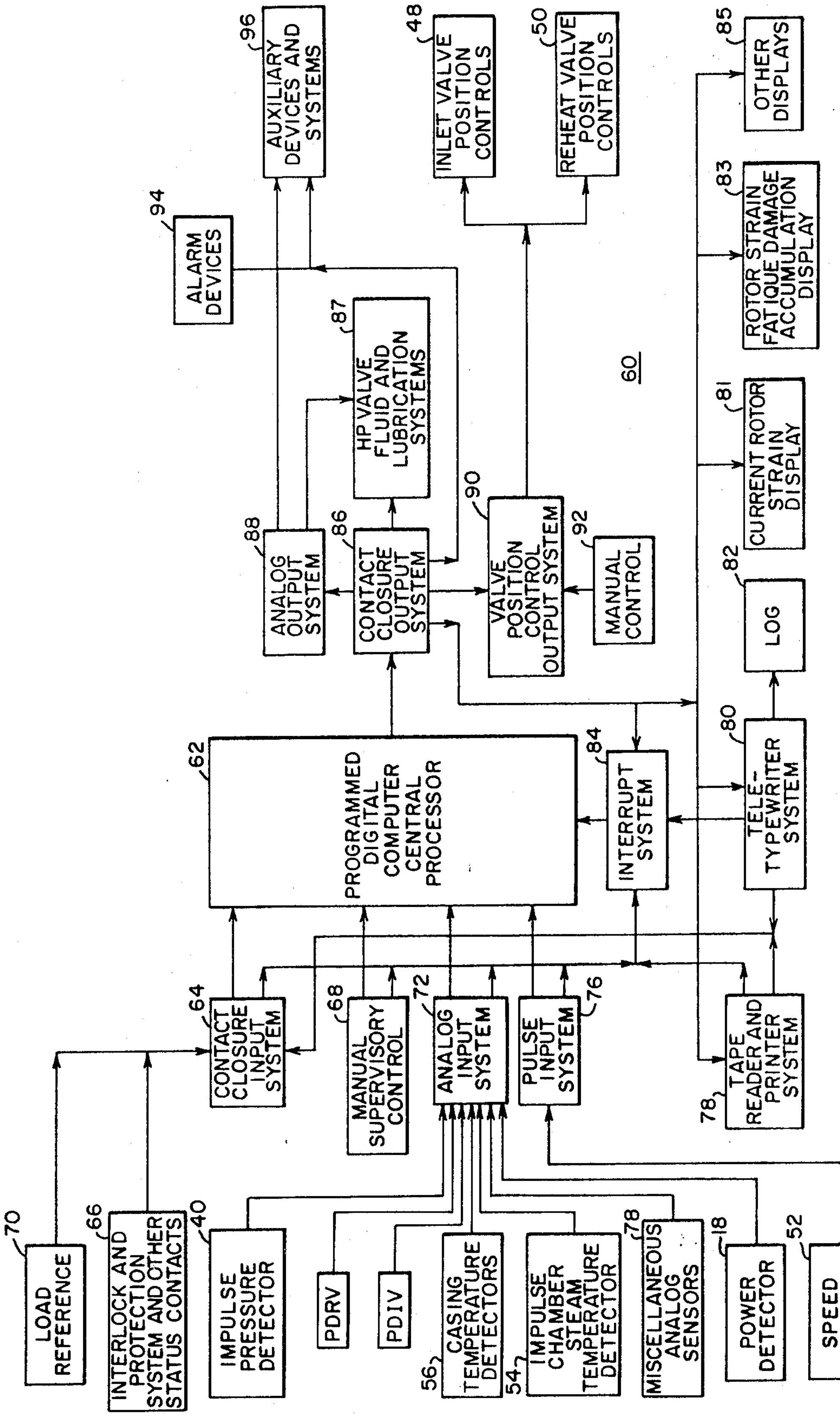
AUXILIARY DEVICES AND SYSTEMS
96
INLET VALVE POSITION CONTROLS
48
REHEAT VALVE POSITION CONTROLS
50
OTHER DISPLAYS
85
ALARM DEVICES
94
HP VALVE FLUID AND LUBRICATION SYSTEMS
87
ROTOR STRAIN FATIQUE DAMAGE ACCUMULATION DISPLAY
83
60
CURRENT ROTOR STRAIN DISPLAY
81
ANALOG OUTPUT SYSTEM
88
CONTACT CLOSURE OUTPUT SYSTEM
86
VALVE POSITION CONTROL OUTPUT SYSTEM
90
MANUAL CONTROL
92
LOG
82
PROGRAMMED DIGITAL COMPUTER CENTRAL PROCESSOR
62
INTERRUPT SYSTEM
84
TELE-TYPEWRITER SYSTEM
80
CONTACT CLOSURE INPUT SYSTEM
64
MANUAL SUPERVISORY CONTROL
68
ANALOG INPUT SYSTEM
72
PULSE INPUT SYSTEM
76
TAPE READER AND PRINTER SYSTEM
78
LOAD REFERENCE
70
INTERLOCK AND PROTECTION SYSTEM AND OTHER STATUS CONTACTS
66
IMPULSE PRESSURE DETECTOR
40
PDRV
PDIV
CASING TEMPERATURE DETECTORS
56
IMPULSE CHAMBER STEAM TEMPERATURE DETECTOR
54
MISCELLANEOUS ANALOG SENSORS
78
POWER DETECTOR
18
SPEED DETECTOR
52

FIG. 2.

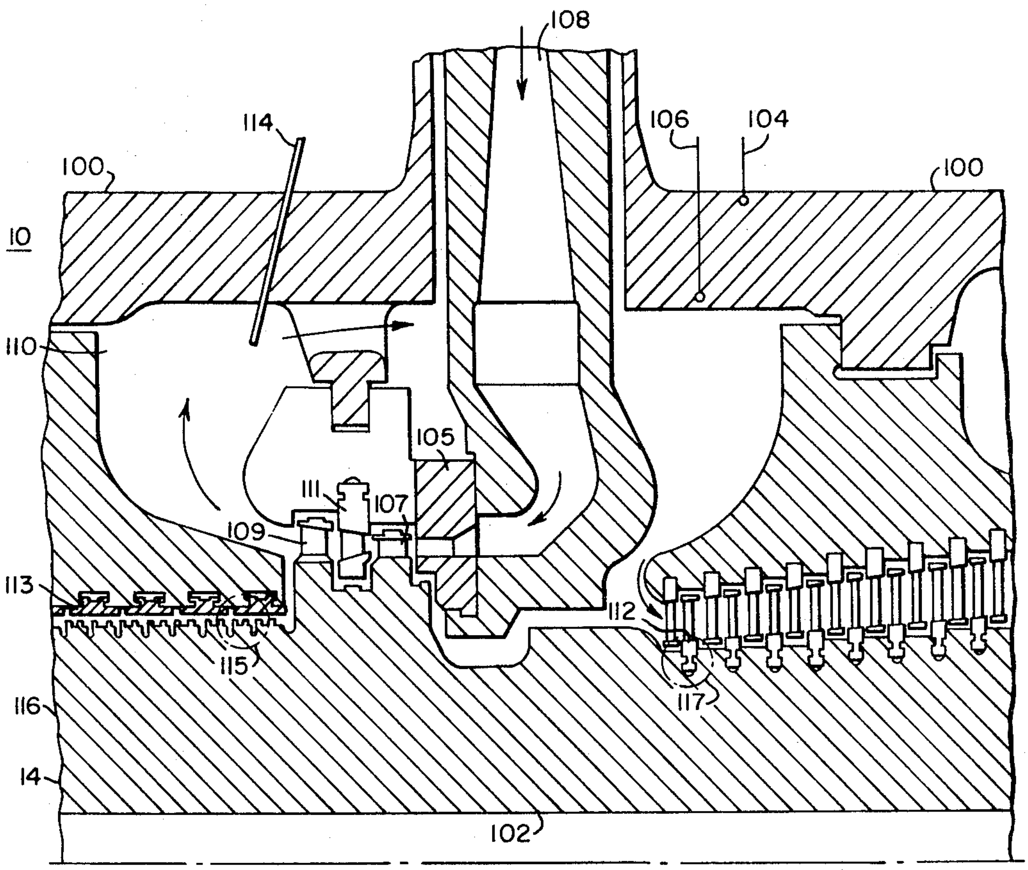

FIG. 3.

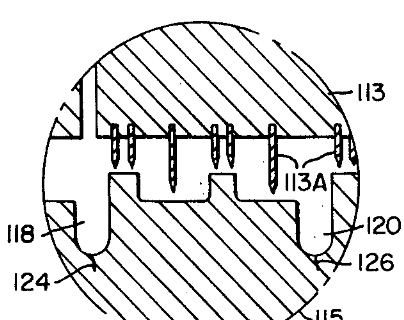

FIG. 4A.

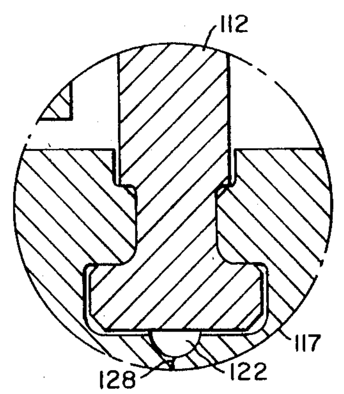

FIG. 4B.

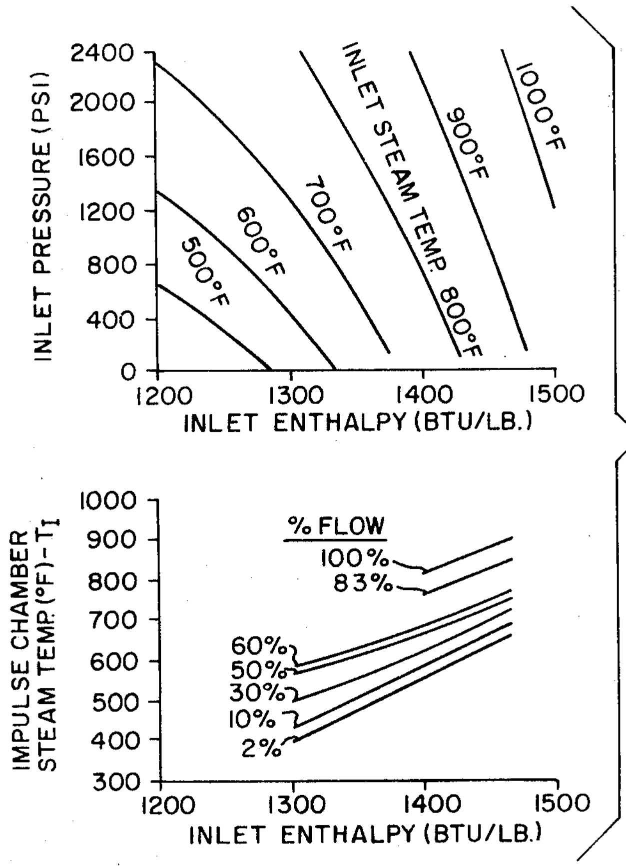
2400
2000
1600
1200
800
400
0
INLET PRESSURE (PSI)
500°F
600°F
700°F
INLET STEAM TEMP. 800°F
900°F
1000°F
1200
1300
1400
1500
INLET ENTHALPY (BTU/LB.)
1000
900
800
700
600
500
400
300
IMPULSE CHAMBER STEAM TEMP. (°F) - T_I
% FLOW
100%
83%
60%
50%
30%
10%
2%
1200
1300
1400
1500
INLET ENTHALPY (BTU/LB.)

FIG. 5.

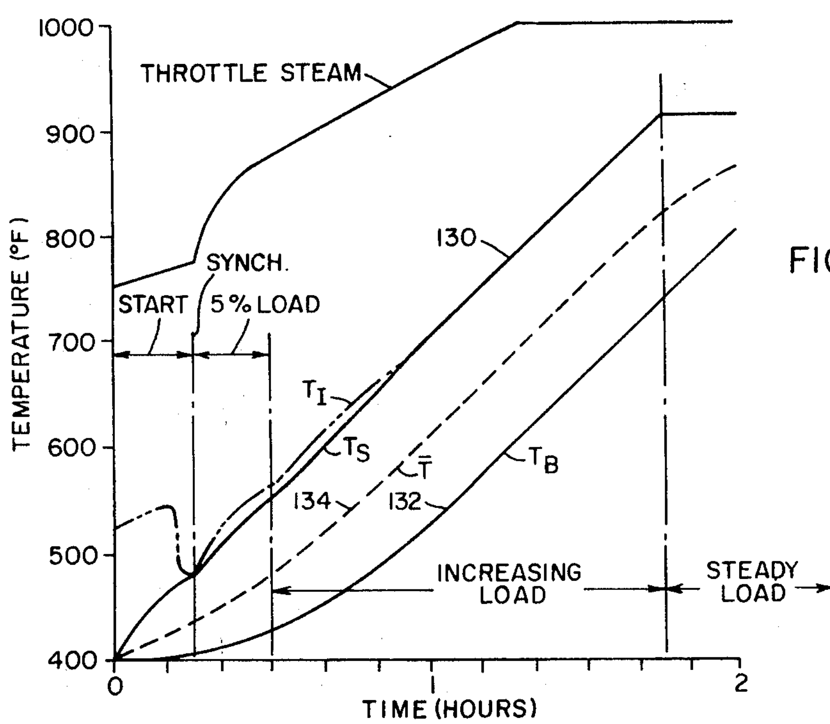
1000
900
800
700
600
500
400
TEMPERATURE (°F)
THROTTLE STEAM
SYNCH.
START
5% LOAD
130
T_I
T_S
$\bar{T}$
T_B
134
132
INCREASING LOAD
STEADY LOAD
0
1
2
TIME (HOURS)

FIG. 6.

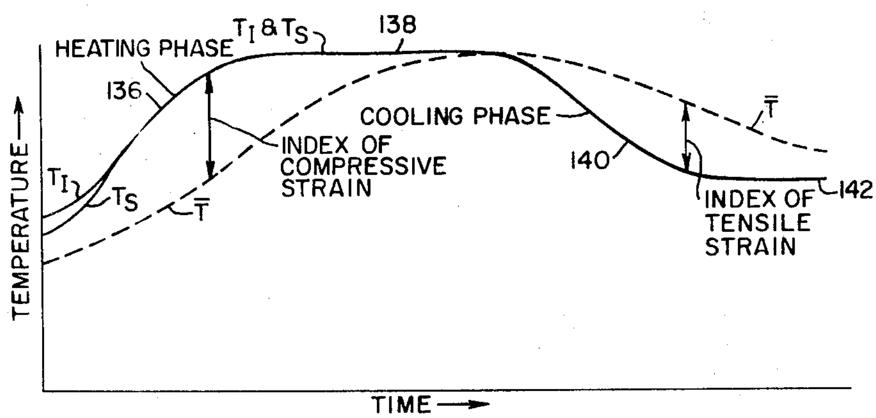

FIG.7.

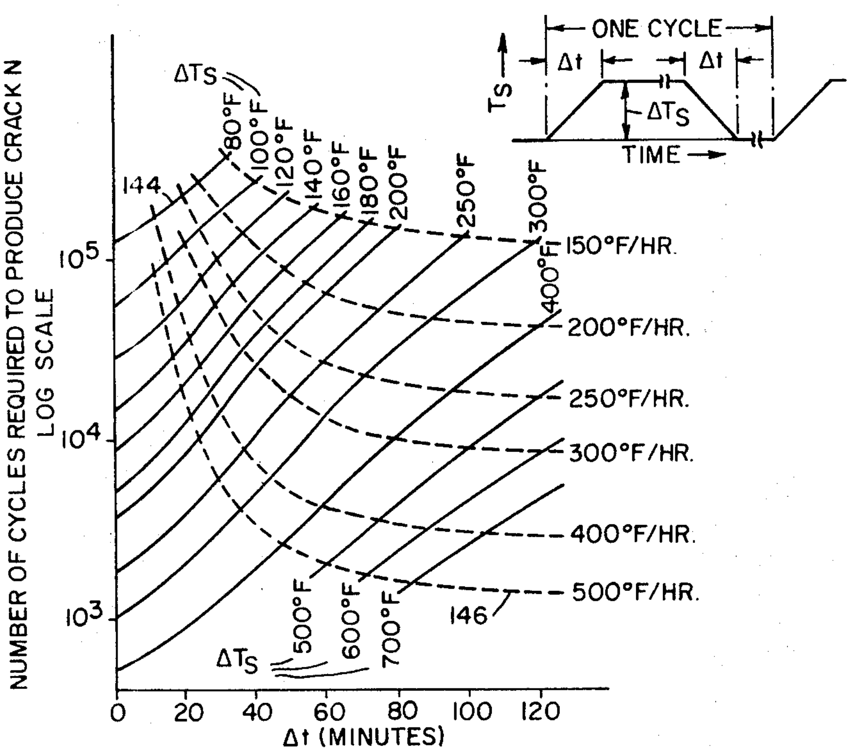

FIG.8.

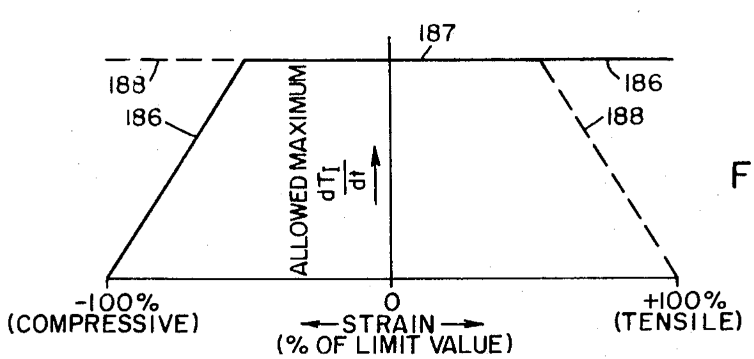

FIG.11.

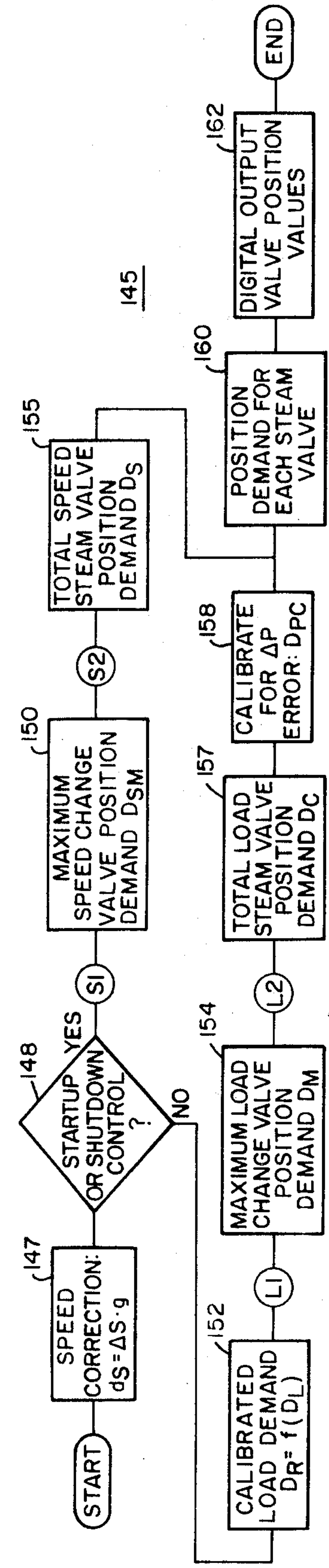
145
START
147
SPEED CORRECTION: $d_S = \Delta S \cdot g$
148
STARTUP OR SHUTDOWN CONTROL ?
YES
NO
S1
150
MAXIMUM SPEED CHANGE VALVE POSITION DEMAND D_{SM}
S2
155
TOTAL SPEED STEAM VALVE POSITION DEMAND D_S
152
CALIBRATED LOAD DEMAND $D_R = f(D_L)$
L1
154
MAXIMUM LOAD CHANGE VALVE POSITION DEMAND D_M
L2
157
TOTAL LOAD STEAM VALVE POSITION DEMAND D_C
158
CALIBRATE FOR ΔP ERROR: D_{PC}
160
POSITION DEMAND FOR EACH STEAM VALVE
162
DIGITAL OUTPUT VALVE POSITION VALUES
END

FIG. 9.

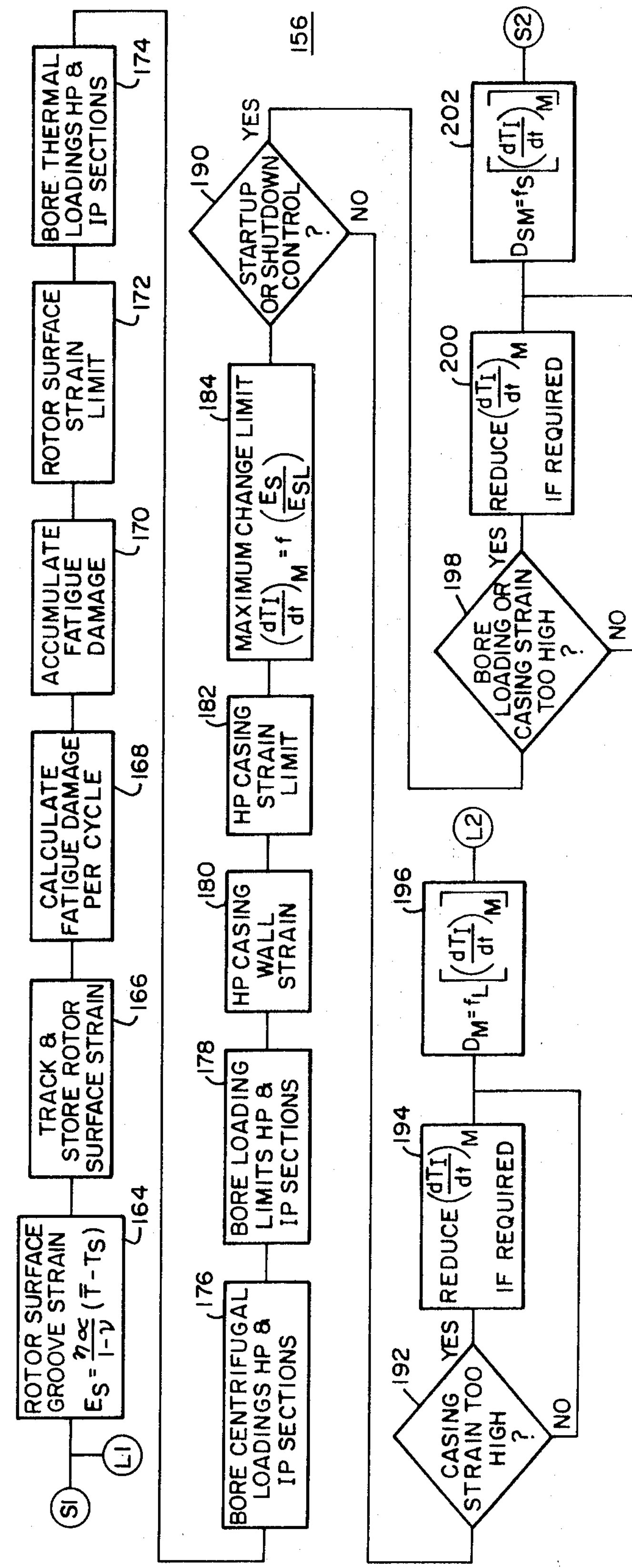
156
S1
L1
164
ROTOR SURFACE GROOVE STRAIN $E_S = \frac{\eta\alpha}{1-\nu}(\bar{T}-T_S)$
166
TRACK & STORE ROTOR SURFACE STRAIN
168
CALCULATE FATIGUE DAMAGE PER CYCLE
170
ACCUMULATE FATIGUE DAMAGE
172
ROTOR SURFACE STRAIN LIMIT
174
BORE THERMAL LOADINGS HP & IP SECTIONS
176
BORE CENTRIFUGAL LOADINGS HP & IP SECTIONS
178
BORE LOADING LIMITS HP & IP SECTIONS
180
HP CASING WALL STRAIN
182
HP CASING STRAIN LIMIT
184
MAXIMUM CHANGE LIMIT $\left(\frac{dT_I}{dt}\right)_M = f\left(\frac{E_S}{E_{SL}}\right)$
190
STARTUP OR SHUTDOWN CONTROL ?
YES
NO
192
CASING STRAIN TOO HIGH ?
YES
NO
194
REDUCE $\left(\frac{dT_I}{dt}\right)_M$ IF REQUIRED
196
$D_M = f_L\left[\left(\frac{dT_I}{dt}\right)_M\right]$
L2
198
BORE LOADING OR CASING STRAIN TOO HIGH ?
YES
NO
200
REDUCE $\left(\frac{dT_I}{dt}\right)_M$ IF REQUIRED
202
$D_{SM} = f_S\left[\left(\frac{dT_I}{dt}\right)_M\right]$
S2

FIG. 10.

SYSTEM AND METHOD FOR PROVIDING STEAM TURBINE OPERATION WITH IMPROVED DYNAMICS

CROSS-REFERENCE TO RELATED APPLICATIONS

Ser. No. 722,779 entitled Improved System and Method for Operating a Steam Turbine And An Electric Power Generating Plant, filed by M. Birnbaum and T. Giras on Apr. 19, 1968 and assigned to the present assignee.

BACKGROUND OF THE INVENTION

The present invention relates to elastic fluid turbines and more particularly to systems and methods for determining the dynamic operation of steam turbines.

Operation of each of the various types of steam turbines generally involves the application of limits to the rate at which turbine steam flow and/or inlet steam enthalpy can be changed for speed, load or other end variable control because of thermal and mechanical turbine response considerations. The turbine casing design is determined largely by steam operating pressure and temperature, and the turbine rotor design is determined largely by rated steady state centrifugal loading forces at maximum speed and rated full power torque with appropriate consideration for other factors including lateral stiffness and critical speeds of rotation. The turbine casing and rotor also must perform under transient and cyclic changes in steam conditions such as those involved in effecting a turbine speed change and/or a turbine load change. The rotor must also perform under transient centrifugal force loading conditions such as those involved in cold startups and the like. Rate or dynamic limits placed on turbine operation largely reflect considerations of thermal expansion and contraction loading associated with inlet steam condition changes and considerations of centrifugal force loading associated with transient speed changes. By "steam condition," it is meant herein to refer to steam temperature, pressure and other steam characteristics including steam flow.

Thermal loading limits have particular significance because rates of change of turbine operating level nearly always or at least quite frequently result in temperature gradients which cause portions of the turbine metal parts to expand and contract in excess of their elastic limit in most applications across the steam turbine art. As a result, the turbine structural material, and particularly the turbine rotor structural material, undergoes a developing history of transient and cyclic plastic strain with continued turbine operation. The combined considerations of turbine operating characteristics required to meet even the minimum needs of turbine users, operating steam thermodynamic properties, turbine size, and turbine structural material thermal and mechanical properties work together to make plastic strain of turbine materials a virtually unavoidable consequence of turbine operation.

Plastic strain resulting particularly from temperature cycling is significant to supervised or controlled turbine operation because of fatigue damage and eventual fatigue cracking which determines the turbine operating life. Fatigue damage accumulates most rapidly at locations most exposed to the widest and most frequent steam temperature variation. It is potential fatigue damage to turbine material at these locations which ordinarily figures most prominently in placing dynamic constraints on turbine operation.

In large electric power plant steam turbines, for example, steam temperature changes in the impulse chamber zone cause rotor temperature gradients which in turn cause substantial rotor plastic strain cycling, particularly on and near the turbine rotor surface adjacent to the impulse chamber zone and more particularly at the base of rotor blade, labyrinth seal, or other grooves or in the vicinity of other structural features such as step configurations near the impulse chamber zone as a result of thermal stress concentration produced by the groove or other special geometry. With respect to large steam turbines actually employed in power plant operation, rotor plastic strain fatigue cracking has been observed in a large number of inspected turbines having an

operating age of 7 years or more. The cracks mostly ranged in depth up to about one quarter inch although one crack was 5 inches deep.

Generally, large steam turbines will not develop rotor cracks until they have operated under cyclic load duty for 5 or more years. Typically, 1800 r.p.m. rotors develop more numerous and deeper cracks because of their larger diameter and associated higher temperature gradients and higher thermal stresses. Casing fatigue damage may occur in the large electric power plant steam turbines but these turbines now typically have separate steam chests and individual separate nozzle chambers, and casing cracking is therefore generally not being experienced by plant operators. Rotor fatigue damage is an increasingly important consideration in turbine operation because of a general trend to larger steam turbine sizes.

Transient centrifugal force loading is typically of greatest concern at turbine rotor locations of highest steady state centrifugal loading such as in the intermediate pressure section of a large steam turbine system. In the latter case; steady state centrifugal force loading eventually causes creep cracks and these are to be distinguished from plastic strain cracks.

Appropriate dynamic constraints are usually desirable to limit transient centrifugal force loading particularly in its combination with thermal loading caused by rotor temperature gradients. For example, during startup of a large steam turbine from its cold state, the total stress at the bore of the high pressure section or the intermediate pressure section can become excessive unless limited. Such loading has been known on occasion to be so great during turbine startup as to cause brittle fracture and turbine rotor blowup.

In prior art turbine plants it has been common practice to limit transient or dynamic turbine operation in the interest of operating safety and prolonged turbine life by appropriate supervisory control or by the use of a programmed controller. Both of these cases involve preawareness of the relationship between various steam flow and/or steam enthalpy change rates and the amount and kind of damage produced by corresponding thermal and/or mechanical operating stress on the turbine parts. Operating practices are devised and implemented through operator supervision or fixed program control to limit turbine dynamic operation in a general way intended to reflect the determined relationship.

Thermal and stress analysis studies have been used in conjunction with knowledge of the specific turbine design to determine the steam change rate—turbine damage relations. A prior paper entitled "Prevention Of Cylic Thermal-Stress Cracking In Steam Turbine Rotors" by W. R. Berry and I. Johnsson and published in the Journal Of Engineering For Power, Transactions ASME in July 1964 presents relatively detailed information insofar as thermal stress analysis and consequent rotor strain fatigue damage are concerned. Another prior published paper entitled "Electrohydraulic Control For Improved Availability And Operation Of Large Steam Turbines" and presented by M. Birnbaum and E. G. Noyes to the ASME-IEEE National Power Conference At Albany, New York during Sept. 19—23, 1965 shows a digital reference system employed to limit turbine speed and load change rates by fixed program control.

Other prior art techniques include that presented in a paper entitled "Automatic Electronic Control Of Steam Turbines According To A Fixed Programme" and published in the Brown-Boveri Review, Volume 51, Number 3, in Mar. 1964. In addition to fixed program control, the automatic turbine control described therein involves casing temperature measurement and feedback control directed to dynamic loading and/or speed changing without exceeding allowable casing stress conditions.

One deficiency of the prior turbine dynamic operating and control art, and perhaps the greatest deficiency because of the forefront needs for turbine rotor protection and economic and efficient turbine operation in meeting end variable control requirements, has stemmed from lack of specific knowledge of actual rotor thermal loading and actual rotor plastic strain and actual rotor fatigue damage accumulation as a basis upon which to produce limits on turbine dynamic operation accurately directed to limiting turbine damage effects including rotor fatigue damage accumulation. Thus, previous turbine supervisory operating procedures and controls have been characterized with only approximate dynamic constraint operation involving both overly conservative and overly liberal operation under different turbine operating conditions. Turbine life and efficiency and economy of turbine operation have thus been adversely affected by the limited prior art capability for dynamic turbine supervision and control with respect to large electric power plant turbines as well as turbines across the turbine art as a whole. In the case of electric power plants, limited steam turbine dynamic capability has resulted in limited plant capability to meet changing electrical load level demands.

SUMMARY OF THE INVENTION

In accordance with the broad principles of the invention, there is provided means for determining the temperature of turbine steam in at least one predetermined steam flow region associated in direct heat transfer relation with a predetermined turbine rotor portion. In an instrumentation package form of the invention, the steam temperature determining means is combined with means for determining a predetermined thermally caused rotor condition and preferably for determining rotor surface stress or strain in the predetermined rotor portion as a function of the determined steam temperature. Preferably, the monitored steam flow region is that where steam temperature variation is the greatest, such as the impulse chamber. By "rotor surface strain," it is herein intended to refer to strain occurring at or near the rotor surface including at or near rotor surface structural features such as at or near the base of blade or other rotor grooves. The term "rotor surface stress" is correspondingly defined. By "function of a certain variable," it is herein intended to refer to a function in which that variable is a significant determining variable but not necessarily the only determining variable in the function.

In another form of the invention, the steam temperature is determined in the predetermined region, the rotor thermal condition is determined, and a limit is placed on the rate at which the turbine operating level is changed in order to limit predetermined turbine conditions including the extent of the rotor thermal condition while meeting end controlled variable demand. In apparatus form, the control system including the necessary determining means provides automatic end variable control under automatic closed loop dynamic constraint control. In electric power plant turbines, load and speed changes are constrained within the rotor thermal condition limits. More accurate, more efficient and more economic turbine operation is realized by application of the invention.

In its preferred form, the control system includes a digital computer system. It is programmed to determine end variable control actions during transient and steady state operation with dynamic constraints computed as a function of steam temperature in the predetermined steam flow region.

It is therefore an object of the invention to provide a novel method and system for operating a steam turbine with improved performance characteristics.

Another object of the invention is to provide a novel method and system for operating a steam turbine with greater accuracy, efficiency and economy.

A further object of the invention is to provide a novel method and system for operating a steam turbine with both extended turbine operating life and better satisfaction of turbine operating level demand.

An additional object of the invention is to provide a novel method and system for operating a steam turbine with better control over turbine speed and load change rates.

It is another object of the invention to provide a novel method and system for operating an electric power plant steam turbine with better turbine control resulting in more responsive electric power generation control.

It is a further object of the invention to provide a novel method and system for operating a steam turbine with real time rotor stress or strain control to provide for turbine operation more closely suited to desired operating practices.

It is an additional object of the invention to provide a novel method and system for operating a steam turbine with real time rotor stress or strain control enabling continuous recordkeeping of accumulated rotor strain fatigue damage and a resulting improved information base upon which to make turbine and plant operating judgments in electric power plant and other applications.

Another object of the invention is to provide a novel turbine plant instrumentation package capable of determining real time turbine rotor stress or strain and thereby enabling turbine operation in both existing and new turbine installations to be supervised more efficiently, more economically and more accurately within dynamic constraints.

These and other objects of the invention will become more apparent upon consideration of the following detailed description along with the attached drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 shows a schematic diagram of a large electric power plant steam turbine supplied with steam by a steam generating system and operated in association with certain sensor and control devices in accordance with the principles of the invention;

FIG. 2 shows a schematic diagram of a programmed digital computer control system operable with the steam turbine and its associated devices shown in FIG. 1 in accordance with the principles of the invention;

FIG. 3 shows an enlarged portion of a longitudinal section through a high pressure section of the steam turbine of FIG. 1 and certain sensor devices placed therein;

FIGS. 4A and 4B show further enlarged portions of the section view of FIG. 3 which illustrate typical locations at which rotor plastic strain fatigue cracking might develop;

FIG. 5 illustrates the manner in which turbine inlet steam pressure and temperature and impulse chamber steam temperature are related with the use of typical specific electric power plant steam turbine data;

FIG. 6 shows typical temperature transients for various locations in a large electric plant steam turbine as the turbine is brought up to the synchronous speed value and placed under steady state load operation;

FIG. 7 shows a typical rotor surface temperature cycle for an electric power plant steam turbine;

FIG. 8 shows a typical thermal plastic stress-strain cyclic capacity chart corresponding to cyclic operation like that of FIG. 7 for a 3600 r.p.m. electric power plant steam turbine rotor having a geometric design of high thermal duty cycle capacity;

FIG. 9 shows a supervisory and control logic flow diagram employed in part of a programming system which operates the computer system of FIG. 2 in accordance with the principles of the invention;

FIG. 10 shows a turbine dynamics supervisory and control portion of the diagram of FIG. 9 in greater detail; and

FIG. 11 graphically illustrates one manner in which turbine steam enthalpy and/or flow changes and in turn impulse chamber steam temperature can be controllably constrained as a function of rotor strain.

DESCRIPTION OF THE PREFERRED EMBODIMENT

More specifically, there is shown in FIG. 1 a large single reheat steam turbine 10 constructed in a well-known manner and operated and controlled in accordance with the principles of the invention as part of a fossil fuel fired electric power plant 12. Other types of steam turbines, such as extraction turbines, reactor turbines, back pressure turbines, etc. can also be controlled in accordance with the principles of the invention.

The turbine 10 is provided with a single output shaft 14 which drives a conventional large alternating current generator 16 to produce three phase (or other phase) electric power as measured by a conventional power detector 18. Typically, the generator 16 is connected (not shown) through one or more breakers (not shown) per phase to a large electric power network and when so connected causes the turbogenerator arrangement to operate at synchronous speed under steady state conditions. Under transient electric load change conditions, system frequency may be affected and conforming turbogenerator speed changes would result. At synchronism, power contribution of the generator 16 to the network is normally determined by the turbine steam flow which in this instance is supplied to the turbine 10 at substantially constant throttle pressure.

In this case, the turbine 10 is of the multistage axial flow type and includes a high pressure section 20, an intermediate pressure section 22 and a low pressure section 24. Each of these turbine sections may include a plurality of expansion stages provided by stationary vanes and an interacting bladed rotor connected to the shaft 14. In other applications, turbines operated in accordance with the present invention can have other forms with more or fewer sections tandemly connected to one shaft or compoundly coupled to more than one shaft.

The constant throttle pressure steam for driving the turbine 10 is developed by a steam generating system 26 which is provided in the form of a conventional drum type boiler operated by fossil fuel such as pulverized coal or natural gas. From a generalized standpoint, the present invention can also be applied to steam turbines associated with other types of steam generating systems such as nuclear reactor and once through boiler systems.

The turbine 10 in this instance is further of the double ended steam chest type, and turbine inlet steam flow is directed through a plurality of throttle valves and a plurality of governor valves designated as inlet valves 25. Generally, the double ended steam chest type and other steam chest types such as the single ended steam chest type or the end bar lift type may involve varying numbers and/or arrangements of throttle valves. More detailed description on a particular throttle and governor valve arrangement is presented in the aforementioned Birnbaum and Giras copending application.

The preferred turbine startup method is to (1) raise the turbine speed from the turning gear speed of about 2 r.p.m. to about 80 percent of the synchronous speed under throttle valve control and then (2) transfer to governor valve control and raise the turbine speed to the synchronous value, close the power system breaker(s) and meet the load demand. On shutdown, similar but reverse practices are involved. Other transfer practices can be employed, but it is unlikely that transfer would ever be made at a loading point above 40 percent rated loading because of throttling efficiency considerations.

After the steam has coursed past the first stage impulse blading to the last stage reaction blading of the high pressure section 20, it is directed to a reheater system 28 which is associated with the boiler 26. In practice, the reheater system 28 might typically include a pair of parallel connected reheaters coupled to the boiler 26 in heat transfer relation as indicated by the reference character 29 and associated with opposite sides of the turbine casing.

With a raised enthalpy level, the reheated steam flows from the reheater system 28 through the intermediate pressure turbine section 22 and the low pressure turbine section 24. From the latter, the vitiated steam is exhausted to a condenser 32 from which water flow is directed (not indicated) back to the boiler 26. To control the flow of reheat steam, reheat valves 33 are provided and these include one or more normally open check or stop valves and one or more intercept valves operable to provide reheat steam flow cutback modulation under turbine overspeed conditions.

In the typical fossil fuel drum type boiler steam generating system, the boiler control system controls boiler operations so that steam throttle pressure is held substantially constant. A throttle pressure detector 38 of suitable conventional design measures the steam throttle pressure to provide assurance of substantially constant throttle pressure supply, and, if desired as a programmed computer protective system override control function, turbine control action can be directed to throttle pressure control as well as or in place of speed and/or load control if the throttle pressure falls outside predetermined constraining safety and turbine condensation protection limits. An impulse chamber steam pressure detector 40 develops signals for use in programmed computer control of turbine load and ultimately power plant electrical load.

Respective hydraulically operated valve actuators indicated by the reference character 42 are provided for the throttle and governor inlet valves 25. Hydraulically operated actuators indicated by the reference characters 44 are also provided for the reheat stop and intercept valves 33. A computer sequenced and monitored high pressure fluid supply 46 provides the controlling fluid for actuator operation of the valves 25 and 33. A computer supervised lubricating oil system (not shown) is separately provided for turbine plant lubricating requirements.

The respective actuators 42 and 44 are of conventional construction, and the actuators 42 and the actuators 44 associated with the intercept valves are operated by respective stabilizing position controls indicated by the reference characters 48 and 50. These controls each include a conventional position error feedback operated analog controller (not indicated) which drives a suitable known actuator servo valve (not indicated) in the well-known manner. Reheat intercept valve position control is imposed typically only when reheat steam flow cutback modulation is required. Stop valve operation requires no feedback position control and instead is manually or computer directed with conventional trip or other suitable emergency operation.

Since turbine power is proportional to steam flow under the assumed controlled condition of substantially constant steam throttle pressure, steam valving position is controlled to produce control over steam flow as an intermediate variable and over turbine speed and/or load as an end controlled variable(s). Actuator operation provides the steam valve positioning, and respective valve position detectors PDIV and PDRV are provided to generate respective valve position feedback signals for developing position error signals to be applied to the respective position controls 48 and 50. The position detectors are provided in suitable conventional form, for example they can make conventional use of linear variable differential transformer operation in generating negative position feedback signals for algebraic summing with respective position setpoint signals SP in developing the respective input position error signals.

The combined position control, hydraulic actuator, valve position detector element and other miscellaneous devices (not shown) form a local hydraulic-electrical analog valve position control loop for each throttle and governor inlet steam valve. The position setpoints SP are computer determined and supplied to the respective local loops and updated on a periodic basis. Setpoints SP are also determined for the intercept valve controls. A more complete general background description of electrohydraulic steam valve positioning and hydraulic fluid supply systems for valve actuation is presented in the aforementioned Birnbaum and Noyes paper.

A speed detector 52 is provided to determine the turbine shaft speed for speed control, for centrifugal stress determination and turbine constraint operation, for frequency participation control purposes, and preferably also for rotor surface heat transfer conductance computation associated with rotor thermal strain control. The speed detector 52 can for example be in the form of a reluctance pickup (not shown) magnetically coupled to a notched wheel (not shown) on the turbogenerator shaft 14. The process sensor equipment further includes an impulse chamber steam temperature detector 54 and casing temperature detectors 56 all of which are employed in programmed computer loading and thermal strain determination as subsequently described more fully. Analog and/or pulse signals produced by the speed detector 52, the power detector 18, the pressure detectors 38 and 40, the temperature detectors 54 and 56, the valve position detectors PDIV and PDRV and other sensors (not specifically shown) and status contacts (not specifically shown) are all applied to a digital computer control system 60 (FIG. 2) which provides turbine steady state and transient operation control on an on line real time basis and further provides system monitoring, sequencing, supervising, alarming, display and logging functions.

The programmed digital computer control system 60 operates the turbine 10 with improved dynamic performance characteristics, and can include conventional hardware in the form of a central processor 62 and associated input/output interfacing equipment such as that sold by Westinghouse Electric Corporation under the trade name Prodac 50 (P50). In other cases such as when the turbine 10 as well as other plant equipment units such as the steam generating system 26 are all placed under computer control, use can be made of a larger computer system such as that sold by Westinghouse Electric Corporation and known as the Prodac 250 or separate computers such as P50 computers can be employed for the respective controlled plant units. In the latter case, control process interaction is achieved by tying the separate computers together through data links and/or other means.

Generally, the P250 typically uses an integral magnetic core 16,000 word (16 bit plus parity) memory with 900 nanosecond cycle time, an external magnetic core 12,000 word or more (16 bit plus parity) memory with 1.1 microsecond cycle time and a mass 375,000 word or more (16 bit plus parity) random access disc memory unit. The P50 processor typically uses an integral magnetic core 12,000 word (14 bit) memory with 4.5 microsecond cycle time.

The interfacing equipment for the computer processor 62 includes a conventional contact closure input system 64 which scans contact or other similar signals representing the status of various plant and equipment conditions. Such contacts are generally indicated by the reference character 66 and might typically be contacts of mercury wetted relays (not shown) which are operated by energization circuits (not shown) capable of sensing the predetermined conditions associated with the various system devices. Status contact data is used in interlock logic functioning in control or other programs, protection and alarm system functioning, programmed monitoring and logging and demand logging, functioning of a computer executed manual supervisory control 68, etc.

The contact closure input system 64 also accepts digital load reference signals as indicated by the reference character 70. The load reference 70 can be manually set or it can be automatically supplied as by an economic dispatch computer (not shown). In the load control mode of operation, the load reference 70 defines the desired megawatt generating level and the computer control system 60 operates the turbine 10 to supply the power generation demand.

Input interfacing is also provided by a conventional analog input system 72 which samples analog signals from the plant 12 at a predetermined rate such as 15 points per second for each analog channel input and converts the signal samples to digital values for computer entry. The analog signals are generated by the power detector 18, the impulse pressure detector 40, the valve position detectors PDIV and PDRV, the temperature detectors 54 and 56, and miscellaneous analog sensors 74 such as the throttle pressure detector 38 (not specifically shown in FIG. 2), various steam flow detectors, other steam temperature detectors, miscellaneous equipment operating temperature detectors, generator hydrogen coolant pressure and temperature detectors, etc. A conventional pulse input system 76 provides for computer entry of pulse type detector signals such as those generated by the speed detector 52. The computer counterparts of the analog and pulse input signals are used in control program execution, protection and alarm system functioning, programmed and demand logging, etc.

Information input and output devices provide for computer entry and output of coded and noncoded information. These devices include a conventional tape reader and printer system 78 which is used for various purposes including for example program entry into the central processor core memory. A conventional teletypewriter system 80 is also provided and it is used for purposes including for example logging printouts as indicated by the reference character 82. Alphanumeric and/or other types of displays 81, 83 and 85 are used to communicate current rotor strain, accumulated rotor strain fatigue, and other information.

A conventional interrupt system 84 is provided with suitable hardware and circuitry for controlling the input and output transfer of information between the computer processor 62 and the slower input/output equipment. Thus, an interrupt signal is applied to the processor 62 when an input is ready for entry or when an output transfer has been completed. In general, the central processor 62 acts on interrupts in accordance with a conventional executive program. In some cases, particular interrupts are acknowledged and operated upon without executive priority limitations.

Output interfacing is provided for the computer by means of a conventional contact closure output system 86 which operates in conjunction with a conventional analog output system 88 and with a valve position control output system 90. A manual control 92 is coupled to the valve position control output system and is operable therewith to provide manual turbine control during computer shutdown and other desired time periods.

Certain computer digital outputs are applied directly in effecting program determined and contact controlled control actions of equipment including the high pressure valve fluid and lubrication systems as indicated by the reference character 87, alarm devices 94 such as buzzers and displays, and predetermined plant auxiliary devices and systems 96 such as the generator hydrogen coolant system. Computer digital information outputs are similarly applied directly to the tape printer and the teletypewriter system 80 and the display devices 81, 83 and 85.

Other computer digital output signals are first converted to analog signals through functioning of the analog output system 88 and the valve position control output system 90. The analog signals are then applied to the auxiliary devices and systems 96, the fluid and lubrication systems 87 and the valve controls 48 and 50 in effecting program determined control actions. The respective signals applied to the steam valve controls 48 and 50 are the valve position setpoint signals SP to which reference has previously been made.

Temperature detection is employed in the determination of the plastic strain of turbine material, and reference is made to FIG. 3 for a more detailed showing of the more significant portions of an illustrative structural arrangement for the turbine high pressure section 20 and for the preferred turbine temperature sensor arrangement associated therewith. The turbine high pressure section 20 includes a casing or cylinder wall 100 within which a rotor 102 is supported for rotation. Casing strain at predetermined casing locations is based on conventional outer and inner wall temperature thermocouple probes 104 and 106 which form a part of the casing temperature detectors 56.

A suitable steam temperature sensor (not specifically shown but included as a part of the analog sensors 78) can also be employed in the intermediate pressure section 22, such as in the inlet steam pipe but preferably in the IP inlet steam chamber (not shown). IP steam temperature data is used in the computation of rotor bore thermal stress in the intermediate pressure section 22.

Steam enters the turbine **10** through a plurality of peripherally disposed inlets **108** and associated nozzle blocks **105**, and the steam is directed through a velocity compounded impulse control stage including two rows of rotor impulse blades **107** and **109** and one row of stationary blades **111** into an impulse chamber **110**. As indicated by the flow arrows, the steam then reverses its flow direction and passes through reaction blading **112** in the successive stages of the high pressure section. A conventional thermocouple probe **114** is appropriately supported by the casing **100** to measure the impulse chamber steam temperature. A dummy section **116** includes spring backed seal rings **113** and seal strips **113A** (FIG. 4A) which provide sealing action against excessive axial steam escape through the interface between the rotating rotor shaft and the surrounding stationary turbine structure.

In FIGS. 4A and 4B, further enlarged views of areas **115** and **117** of FIG. 3 show circumferential labyrinth seal grooves **118** and **120** and a circumferential rotor blade groove **122** and typical respective fatigue cracks **124**, **126** and **128** which may develop at the base of the grooves **118**, **120** and **128** after prolonged thermal plastic strain cycling caused by impulse chamber steam temperature variation. In other cases, the blade support grooves could be axial rather than circumferential. In any case, the significance of the rotor grooves or other similar structural features at or near the rotor surface is their stress concentrating effect, i.e. greater thermal stress at these locations is associated with greater likelihood of plastic strain fatigue cracking at these locations.

Generally, fatigue cracks develop in the groove direction because stress concentration occurs more significantly in the cross groove direction. One manner in which concentrated stress can be determined with reasonably good approximation at a groove base is described in detail in the aforementioned Berry and Johnsson paper.

Before considering the programmed computer operation, a more detailed explanation of rotor thermal stress and plastic strain analysis and its connection with turbine dynamics is desirable. The thermal stress at the rotor surface and particularly at rotor surface grooves or the like is significant because it is at this location that rotor temperature gradients caused by changes in surrounding steam temperature cause the maximum rotor thermal stress and the maximum rotor thermal plastic strain. Bore thermal stress has significance primarily during cold startup and like conditions when combined centrifugal and transient thermal bore loading can become excessive unless turbine operation is constrained.

Rotor surface thermal stress and plastic thermal strain are most significant near the impulse chamber region in the high pressure section **20** because this is where they are the largest as a result of the widest ambient steam temperature variations. To determine the HP rotor surface stress and/or strain, it is necessary to determine the rotor surface temperature near the impulse chamber region. The HP rotor surface temperature is determined as subsequently described from the impulse chamber steam temperature T_I and the heat transfer conductance at the HP rotor surface. Because the heat transfer conductance between the ambient impulse chamber steam and the rotating rotor surface is very high for elevated turbine speeds, the rotor surface temperature T_S is substantially equal to the impulse chamber steam temperature T_I except during startup and shutdown under conditions of relatively low steam flow and pressure and low rotative speed. Impulse chamber steam temperature T_I in turn can vary widely with inlet steam flow changes even if inlet steam enthalpy is held constant.

With respect to the intermediate pressure section **22** of the turbine **10**, the heat transfer conductance at the rotor surface is less than that in the high pressure section **20** at various turbine speeds because of reduced steam density and pressure. For bore thermal stress calculations, determination of the rotor surface temperature in the inlet steam region of the intermediate pressure section **22** is based on the measured IP inlet chamber steam temperature and the variable and lower heat transfer conductance at the IP rotor surface. The IP heat transfer conductance is a predetermined function of the turbine speed similar to the function subsequently indicated for the HP heat transfer conductance at the HP rotor surface. In addition, the IP heat transfer conductance is preferably further determined as a predetermined function of the IP steam flow and the IP steam density or pressure, that is $(K_{IS})_{IP}=f(W_S\ \mathrm{SF},\ P_{IP})$ where W_S = actual turbine speed, SF=IP steam flow and P_{IP}=IP steam pressure. Suitable steam flow and steam pressure sensor devices (not specifically shown) are employed in block **78** to provide IP flow and pressure data needed for the computation of the IP rotor surface heat transfer conductance K_{IS}.

Generally, the relationship between the impulse chamber steam temperature T_I and the inlet steam conditions is illustrated in FIG. **5** for the high pressure section of a large steam turbine designed for 2400 p.s.i. inlet steam pressure and 1000° F. inlet steam temperature. Once the inlet steam enthalpy is determined from the inlet steam pressure and temperature, the impulse chamber steam temperature T_I is read from the intersection of the steam enthalpy and steam flow values.

Rotor surface stress can be determined with reasonably good approximation as set forth in the aforementioned Berry and Johnsson paper. Briefly, the step transient heating and cooling functions and the linear transient heating and cooling functions of ambient steam temperature change are of primary interest since these are the most commonly encountered ones and since principles of superposition can be used to construct special transients from these. Further, the surface thermal stress is proportional to the difference between the rotor surface temperature T_S and the rotor volume-average temperature $\bar{T}$. As subsequently described more fully, programmed computer control preferably involves computation of rotor surface strain E_S and it is likewise proportional to the difference between T_S and $\bar{T}$.

Generally, with a step heating or cooling transient maximum rotor surface stress occurs at or near zero time and it is proportional to the magnitude of the ambient temperature change. Surface stress decays from the peak zero time value as rotor interior temperatures approach the rotor surface temperature at rates depending on the dimensions and geometry of the generally cylindrical rotor and its thermal properties. For the linear heating or cooling transient, surface thermal stress is initially zero and it increases with widening difference between T_S and $\bar{T}$ as the interior temperature change rate lags the surface temperature change rate. As the linear transient process continues, a state is ultimately reached in which the ambient surface and interior temperatures change at the same rate. This is referred to as a quasi-steady state with constant difference between T_S and $\bar{T}$ and constant rotor surface thermal stress dependent on the rate of change of ambient temperature.

In a typical case of electric power plant steam turbine load changing, the linear transient process terminates at a final level of steady ambient steam temperature when the desired load change has been accomplished. Usually, termination of the transient occurs before the quasi-steady state is reached. Thus, the difference between T_S and $\bar{T}$ is greatest at the usual liner transient terminating point to define the maximum surface thermal stress for the transient. Maximum stress depends on both the magnitude and the rate of change of the ambient steam temperature, and for very rapid rates of change the limiting case of the step transient is approached.

To provide a better understanding of the cyclic behavior of thermal stress and thermal plastic strain at the rotor surface, there is shown in FIG. 6 a time plot of the throttle steam temperature, the impulse chamber steam temperature T_I, the rotor surface temperature T_S, the rotor bore temperature T_B and the rotor volume-average temperature $\bar{T}$ covering the starting and loading of a typical large 23 inch diameter 3600 r.p.m. electric power plant steam turbine. The rotor is initially at 400° F. with 750° F. throttle steam made available at the starting time. After throttling and expansion, the inlet steam reaches the impulse chamber at 525° F., and the rotor is subjected to a step heating transient with a followup linear heating transient during the startup period. During the startup period, rotor surface temperature response indicated by curve 130 is slow because of low surface conductance at the low velocities and subatmospheric pressure in the impulse chamber. Rotor bore temperature response indicated by curve 132 is even slower. The response of the volume-average temperature $\bar{T}$ is shown as curve 134.

Just prior to reaching synchronous speed, the impulse chamber steam temperature T_I drops because of a changeover to partial admission as control is transferred from throttle valves to governor valves. If control of synchronization and initial loading is by turbine throttle bypass valve, the steam temperature drop occurs later with more pronounced effect on rotor surface temperature at changeover to partial admission.

After synchronization, throttle steam temperature may rise rapidly with increased steam flow and firing rate as five percent load is applied to the turbine unit. The rotor surface temperature T_S at this time nearly equals the impulse chamber steam temperature T_I with the increased surface conductance associated with higher steam flow and pressure and higher rotor speed. In this instance, 5 percent load is held until the rise in throttle steam temperature stabilizes and load is then applied at approximately a uniform rate as throttle steam is brought to rated temperature. The load increase accordingly applies a linear heating transient to the turbine rotor. In this case, the rotor temperature response reaches the guasi-steady state condition during the loading period.

The primary significance of rotor thermal stress-plastic strain cycling during turbine speed and/or loading changes or more generally during turbine inlet steam flow and/or enthalpy changes is the resultant cumulative fatigue damage to the rotor. In FIG. 7, there is shown a typical daily "flat top" cycle of the impulse chamber steam temperature T_I, the rotor surface temperature T_S and the rotor volume-average temperature $\bar{T}$ in a large electric power plant steam turbine as a result of turbine load changing such as that caused by full load day operation and fractional load night operation.

Because of the cyclic rotor surface temperature variation produced by the heating and cooling transients applied to the turbine rotor, the rotor surface fibers undergo stress-strain hysteresis cycling dependent in part on the properties of the rotor material which may be for example the conventional Cr-Mo-V alloy steel rotor material. Thus, compressive thermal stressing occurs in the plastic strain range during heating cycle portion 136, residual thermal stressing occurs during cycle portion 138, tensile thermal stressing in the plastic strain range occurs during cooling cycle portion 140 and new residual thermal stressing occurs during cycle portion 142. The width of the stress-strain hysteresis loop so formed represents the plastic strain for the cycle. With some small long term inlet steam flow and/or steam enthalpy changes, the material elastic stress limit is likely not caused to be exceeded by steam temperature variation and rotor surface plastic strain then probably nearly vanishes. However, most turbine operation level cycling does involve plastic strain.

Other patterns of cyclic thermal rotor surface temperature cause similar hysteresis looping and corresponding cyclic rotor surface plastic strain. In electric power plant turbines, another typical cyclic rotor surface temperature pattern is a substantially sinusoidal pattern caused by sinusoidally applied frequency control action.

In FIG. 8, there is shown a cyclic fatigue capacity chart for flat top rotor surface temperature cycles like that of FIG. 7. This chart shows the number of plastic strain cycles required under varying duty characteristics to produce groove cracking in a 3600 r.p.m. and 23 inch diameter rotor of high thermal duty cycle capacity geometric design. Larger diameter rotors involve larger thermal inertia and therefore involve a more extended time scale than that shown in FIG. 8.

The flat top rotor surface temperature cycle pattern associated with the illustrated fatigue capacity chart is represented by the idealized one that is shown in the upper part of FIG. 8. Generally, linear heating occurs over a Δt period from a first steady state rotor surface temperature, equalizing and holding occurs at a second steady state rotor surface temperature equal to the beginning steady state rotor surface temperature plus ΔT_S, linear cooling occurs over a time period Δt, and equalizing and holding occurs at the first steady state rotor surface temperature. Heating and cooling amounts and rates are made equal for simplicity although in the general invention application case these quantities can be variable. In using the chart of FIG. 8, the rotor surface temperature change ΔT_S and the heating and cooling intervals Δt are determined and the number of cycles N required to produce rotor fatigue cracking is then read from curves 144. Similarly, N can be determined from locus curves 146 when the rotor surface temperature change rate is known and applied over the interval Δt.

Derivation of the chart of FIG. 8 from strength and other properties of the rotor material and from cyclic rotor stress determination is described in the aforementioned Berry and Johnsson paper. Similar charts can be derived for the cyclic fatigue capacity characteristic to other sources of fatigue damage, i.e. other rotor surface temperature cycle patterns such as sinusoidal patterns.

From various cycle fatigue capacity charts, fatigue damage is determined by first identifying the type of cycle, i.e. flat top, sinusoidal, etc., and then determining the cycle fatigue capacity N from corresponding cycle fatigue capacity charts on the basis of particular values of rotor surface temperature change and change rate and temperature transient duration or their equivalents. Next, the damage per cycle 1/N is determined. Cumulative fatigue damage equals the sum of the values for 1/N through the course of turbine use. When the cumulative value reaches the value one, rotor surface fatigue cracking is theoretically expected to occur. Fatigue cracking thus might result from relatively few large fatigue damage cycles, from a large number of small fatigue damage cycles, or from any of various combinations of varied fatigue damage cycles.

Typical prior art practice involves determining the desired turbine plastic strain fatigue life and determining some fixed supervisory or control program restriction on cyclic turbine operation such that resultant cumulative rotor fatigue damage as predictively calculated from cycle fatigue capacity charts and the planned restricted turbine cycling would likely conform to the desired turbine operating life to be met. As already indicated, that procedure has been too approximate with resulting shortcomings in accuracy, efficiency and economy of turbine operation.

With reference now to programmed computer operation, a programming system is employed to operate the computer system 60. It includes control and related programs as well as certain conventional housekeeping programs directed to internal control of the functioning of the computer system itself. The latter include the following:

1. Priority Executive Program

Controls the use of the processor circuitry. In general, it does so on the basis of priority classification of all of the control and housekeeping programs and some of the various kinds of interrupts. The highest bidding program or interrupt routine is determined and allowed to run when a change is to be made in the programmed instructions undergoing execution. Some interrupt routines run outside the priority structure as already indicated, particularly where safety and/or expensive equipment protection are involved.

2. Analog Scan

Periodic execution for the entry of predetermined analog inputs which have been converted by the analog input system 72 and stored in the analog input system buffer register.

3. Status Contact Scan

Periodic execution for the entry of predetermined status contact inputs.

4. Programmers Entry Program

Demand execution allows the computer operator to enter information into the computer memory.

5. Diagnostic Routine

Executed upon computer system malfunction interrupt.

The programming system control and related programs include the following:

1. Data Logging

Periodic or demand execution for printout of predetermined events and parameter values.

2. Alarm

Periodic and process interrupt execution for operating the alarm devices 94 and other system devices and for supervising and/or disabling the valve position and other control programs.

3. Display

Periodic and demand execution for visual display (alphanumeric or graphic) of predetermined parameter values and/or trends.

4. HP Valve Fluid Program

Periodic execution for supervisory control.

5. Lubrication System Program

Periodic execution for supervisory control.

6. Auxiliary Devices and Systems Programs

Periodic execution for supervisory control.

7. Inlet Steam Valve Position Control Program

Periodic execution for control purposes.

8. Turbine Rotor Loading and Thermal Strain Constraint Subprogram

Operates as part of the Inlet Steam Valve Position Control Program to constrain the rate of change of inlet steam flow for safer more prolonged and generally better turbine operation.

9. Reheat Valve Position Control Program

Execution after and during overspeed alarm demand.

The present invention primary involves the functioning of the turbine rotor loading and strain constraint subprogram and further specific programming system description herein will accordingly be limited to the valve position control program and the included rotor constraint subprogram. Flow charts including certain algorithms are shown in FIGS. 9 and 10 as a representation of the basic logic content of the steam valve position control program indicated by the reference character 145 and the constraint subprogram indicated by the reference character 156. Actual programs entered into the computer system 60 are coded in machine language from more detailed flow charts which are in turn derived from the illustrated flow charts.

Prior to startup, the turbine 10 is motor driven at the turning gear speed of about 2 r.p.m. to minimize "breakaway torque" and to maintain shaft straightness. To start the turbine 10, a start signal is applied to the computer 62 as by operation of the manual control 68. Startup is allowed by programming system operation if the predetermined interlock logic permissives are satisfied including for example steam generating system functioning normally, steam throttle pressure at required value, power breakers open, turbine steam valves in starting positions, high pressure fluid system functioning normally, etc.

After startup clearance, the steam valve position control program 145 is periodically executed such as at the rate of once per second to develop steam valve positioning actions directed first to bringing the turbine 10 to the synchronous speed and then to controlling the turbine load. Generally, for electric power plant turbine applications, the program 145 is preferably like that described in the aforementioned Birnbaum and Giras application, and program description more detailed than the description to be presented here can be obtained by reference to that application.

As indicated by block 147, feedback turbine speed correction d_S is determined from the product of gain g and ΔS which is the difference between a reference speed w_R and the actual turbine speed w_S. In this instance, the speed reference w_R approximately provides for turbine speed changing within predetermined dynamic limits, and it is determined from a computer stored startup (or shutdown) ramp curve of turbine speed versus time. The gain g corresponds to the speed regulation desired for the system. The speed regulation g might for example be 3 percent, i.e. 3 percent overspeed at full turbine load results in full closure of the turbine inlet steam valves 25. The numerical form of the speed correction d_S thus is in percentage form, and it provides advantages in the load control operating mode as more fully described in the Birnbaum and Giras application.

With the computer control 60 in the startup or shutdown operating mode, program block 148 directs the program execution to block 150 which determines a maximum speed change valve position demand D_{SM} which dynamically characterizes the control system with a constraint on the rate at which turbine inlet steam flow can be changed for turbine speed control. In applying limit action to the turbine speed change rate, the constraint demand D_{SM} effectively acts as a feedback trim on the speed ramp w_R which involves feedforward but only approximate dynamic constraint. If desired, speed constraint operation can be imposed only on startup and the conventional coastdown procedure can be used on shutdown. The program 145 is appropriately modified when coastdown operation is selected.

If the turbine 10 is already operating at synchronous speed, the program block 148 directs the program execution to block 152 where speed calibrated load demand is determined from the load reference 70 or D_L. Block 154 next determines a maximum load change valve position demand D_M which dynamically characterizes the control system with a constraint on the rate at which turbine inlet steam flow can be changed for turbine load control. In applying limit action to the turbine load change rate, the constraint demand D_M effectively acts as a feedback dynamic constraint on the load control loop which in its preferred form is a feedforward control loop. Blocks 150 and 154 contain some common execution steps and form the turbine rotor loading and strain constraint subprogram 156 which is shown in greater block detail in FIG. 10.

In the startup and shutdown speed control operating mode, total steam valve position demand D_S is determined in a closed speed feedback loop and it is made equal to a predetermined function of d_S or it is constrained by D_{SM} under predetermined rotor loading or thermal strain conditions. As indicated in the Birnbaum and Giras application, D_{SM} may or may not be a numeric variable, and in this case it preferably either allows ramping of w_R or disallows such ramping when speed change constraint is to be imposed. The steam valve movement is then determined by speed error based on a fixed reference speed value until the constraint action is released. A determination that D_{SM} is to constrain D_S in effect means making D_S equal to the predetermined function of d_S with w_R held constant.

Similarly, in the load control operating mode, total steam valve position demand D_C is determined in block 157 from D_R or it is constrained by D_M under predetermined rotor changing or thermal strain conditions, specifically when D_R is greater than D_M during or less than D_M during unloading when D_M in effect acts as a minimum constraint. Determination of D_C is preferably made from a static characterization in the feedforward load control loop and it is then load calibrated to D_{PC} on the basis of impulse chamber pressure error in block 158 as more fully explained in the Birnbaum and Giras application. Generally, the static characterization in the block 157 defines the total steam valve positioning required to satisfy the demand D_L or D_M if D_L is under constraint. The block 158 corrects for any minor characterization or other error by its trim action.

Total valve position demand D_S or D_{PC} is distributed among the inlet steam valves 25 in block 160 according to a predetermined schedule. The respective digital inlet valve position setpoint values are then determined in block 162 just before the program run is ended.

When the dynamic constraint subprogram 156 is being executed to determine maximum load change or speed change steam valve position demands for the block **150** or **154**, the rotor surface thermal strain E_S is first determined in block **164**, (FIG. **10**) according to the equation indicated therein. It is preferred that surface thermal strain E_S be the determined HP rotor thermal condition upon which supervisory or control action is to be taken in the turbine operation because this is the fundamental variable involved in cumulative fatigue damage. If desired, other HP rotor thermal conditions such as the rotor thermal stress can be determined and processed in computing supervisory or control data.

The HP rotor surface temperature T_S can be made equal to the detected impulse chamber steam temperature T_I under certain justifying operating conditions, i.e. when the heat transfer conductance K_{IS} at the HP rotor surface has an adequately high value. This can be the case for most of the typical electric power plant steam turbine cycles such as that shown in FIG. **7** and for most of other like cycles.

For those operating periods in which the steam-rotor heat transfer conductance is too low to justify making T_S and T_I equal and high accuracy is desired, such as during startup and shutdown of the turbine **10** or during normal operating periods in other turbine applications involving operating speed variation, the HP rotor surface temperature T_S can be automatically determined as a function of variables including the impulse chamber steam temperature and the steam ambient to rotor heat transfer conductance (K_{IS}) HP. Preferably, the HP rotor surface heat transfer conductance is determined as a predetermined function of the rotor speed, i.e. $(K_{IS})_{HP}=f(w_S)$, and the rotor surface temperature is calculated from the measured ambient steam temperature T_I and from the computed value for $(K_{IS})_{HP}$ in the manner subsequently described. In the load control operating mode, $(K_{IS})_{HP}$ as computed from $f(w_S)$ would normally be substantially constant and high enough to make $T_S=T_I$. Generally, the surface heat transfer conductance depends primarily on turbine speed and steam flow and secondarily on steam pressure and/or density and other thermodynamic quantities. It is from this generalization that the preferred functions $(K_{IS})_{HP}$ and $(K_{IS})_{IP}$ are derived.

The block **164** provides for determining the rotor volume-average temperature $\bar{T}$. This operation is based on standard cylinder thermal gradient transient analysis as set forth in a text by G. M. Dusenberry entitled "Numerical Analysis of Heat Flow" and published in 1949 by McGraw-Hill. Generally, the rotor is mathematically divided into a preselected number of successive rings having equal radial extent and being numbered radially inwardly. The respective rings have respective heat capacities C_1 ... Cn and interring heat flow transfer conductances K_{12}, K_{23}, ... K(n118 1) (n) associated with them.

Equations involving the ring heat capacities are set up for heat flow between the ambient steam and the rotor surface ring through the surface (film) thermal conductance $(K_{IS})_{HP}$ and between the first and second and successive ingoing rotor ring pairs through the respective interring conductances in terms of the steam and ring temperatures at time t_o and the ring temperatures after a time interval Δt or at $(t_o+\Delta t)$. In each of these equations, the ring temperature at the present time (t_o+t) is solved in terms of the other equation quantities. The computed value for T_S is the value used for T_S in block **164**. Present rotor volume-average temperature $\bar{T}$ is computed from present ring temperature values as follows:

$$\bar{T}=\frac{C_1T_1+\ .\ .\ .\ +C_nT_n}{C_1+\ .\ .\ .\ +C_n}$$

where:

C_1 = surface ring heat capacity

T_1 = present surface ring temperature $= T_S = f[(K_{IS})_{HP}, T_I]$

C_n = heat capacity of ring n

T_n = present temperature of ring n.

With $\bar{T}$ determined, T_S is subtracted from $\bar{T}$ and the difference is multiplied by the thermal expansion coefficient α and a concentration factor η which makes the computation applicable to the groove base where thermal strain and stress are concentrated. The resulting quantity is next divided by $(1-\nu$ where ν is Poisson's ration to give the HP rotor surface strain E_S. If surface thermal stress S_S rather than surface thermal strain E_S is the computed quantity, the equation in block **164** is modified by including the modulus of elasticity E as an additional multiplier of the difference quantity $(\bar{T}-T_S)$. In addition, the concentration factor η is modified to reflect the fact that stress rather than strain is being calculated.

The determined HP rotor surface strain E_S is stored as indicated by block **166**, and the successive strain values E_S from successive program runs are tracked to determine strain cycle activity. Preferably, only significant strain cycles are identified from the tracking operation, and block **168** calculates rotor plastic strain fatigue damage associated with each identified cycle. The damage calculation is made by determining N (previously defined) from stored cycle fatigue capacity chart data once the type of rotor strain cycle and the cycle duty characteristics are ascertained.

Block **170** acts as a HP rotor surface fatigue damage accounting system since it adds successively determined damage values to provide a running rotor fatigue damage total. The display devices **81** and **83** can show the current computed HP rotor surface strain E_S and the accumulated rotor surface fatigue damage by programmed scheduling or by operator demand. Generally, the effectiveness of the fatigue accumulation procedure in following actual rotor fatigue damage depends on the standards employed in identifying strain cycles, i.e. the kind of screening used in determining which cycles are to be damage counted and which cycles are not to be damage counted. For example, in electric power plant steam turbines, identified cycles could be as few as one per day with very coarse screening and in these and similar applications blocks **168** and **170** would function infrequently.

If desired, a closed constraint control loop (not shown) can be applied to the basic control in a long term supervisory sense based on the accumulated HP rotor strain fatigue damage of block **170**. For example, the subsequently considered strain limit E_{SL} of the block **184** might be modified with time passage as a function of the computed damage total.

The impulse chamber steam temperature detector **54** and the arrangement for determining HP rotor surface strain E_S and if desired for recording and accumulating cycle fatigue damage can be provided separately in the form of an instrumentation system or package useful for supervisory turbine operation where closed loop dynamic strain constraint control is not desired. In such event, the impulse chamber steam temperature sensor output and the turbine speed detector output are coupled to suitable computing means capable of performing the programmed operation described for block **164** and if desired blocks **166**, **168** and **170**. The instrumentation computer can be a special purpose analog, digital-analog, or digital computer. The instrumentation combination is especially useful for older installed turbines where improved operating supervision can be realized by operator knowledge of actual rotor strain operating conditions and actual rotor strain fatigue damage. Similar instrumentation packages can be employed for determining thermal stress and/or strain conditions in intermediate pressure sections of large turbines as well as in turbines other than large electric power plant turbines.

In the present case, closed loop dynamic rotor surface strain constraint operation is provided by execution of the program **145** and the subprogram **156**. Thus, after determination of the actual strain E_S, a HP rotor surface strain limit E_{SL} is determined in block **172**. This can be, and for simplicity in this case it is, a fixed value based on design and operating life considerations. However, in general practice, the strain limit E_{SL} can be a value determined by a predetermined function or it can be a value selected from a table of values with each table value corresponding to some predetermined computer determinable set of conditions. In the vase of variable strain limit action, rotor strain fatigue damage can be automatically escalated for various operating conditions for which it is prejudged that the overall gain from faster turbine adjustment is worth the cost of increased rotor fatigue damage.

In blocks 174 and 176, the bore thermal and centrifugal loadings are determined for the high pressure and intermediate pressure turbine sections 20 and 22. In each case, bore thermal stress is determined from the rotor temperature gradients caused by transient turbine operation and calculated from measured steam temperature and other data in a manner similar to the procedure of block 164, and centrifugal stress is determined from turbine speed data from the speed detector 52. Block 178 supplies a combined loading limit which is a predetermined fixed value, or as in the block 172, a table of hierarchical values corresponding to allowed loading under different computer determinable sets of operating conditions.

Similarly, high pressure casing wall strain is computed in block 180 on the basis of temperature readings from the casing temperature detectors 56. A fixed or other casing strain limit is provided by block 182.

Since changes in the impulse chamber steam temperature T_I are the primary cause of imposed HP rotor temperature gradients and these in turn produce rotor surface strain and stress, an allowed maximum rate of change of T_I is determined in block 184 as a predetermined function of the surface strain E_S computed in block 164. To effect a limit on the rate of change of T_I, the inlet steam enthalpy variation rate and/or, as in this case the inlet steam flow change rate is ultimately limited. In this instance, the allowed maximum change rate $(dT_I/dt)_M$ is made equal to a function of the percent ratio of the actual surface strain E_S to the limit strain E_{SL} from block 172.

Generally, if as is usually the case the turbine manufacturer has on an experience basis assigned or recommended fixed highest allowed rates of change of T_I corresponding to maximum heating and cooling rates, maximum cooling may be permissible if the rotor surface is undergoing compressive strain from a previous heating transient and likewise maximum heating may be permissible if the rotor surface is undergoing tensile strain from a previous cooling transient. If the rotor surface is in compression, the extent of that compression determines whether the maximum heating rate of change in T_I or some lesser rate is required. If the rotor surface is in tension, the extent of that tension determines whether the maximum cooling rate of change in T_I or some lesser rate is required.

These considerations are reflected in FIG. 11 which shows a relatively simple and approximate yet effective function for determining $(dT_I/dt)_M$ in block 184. Thus, solid curve 186 allows the highest value of $(dT_I/dt)_M$ for heating or increasing values of T_I for all tensile strain percent ratios and up to the 50 percent compressive strain ratio. Between 50 percent and 100 percent compressive strain ratio, the allowed increase rate of change of T_1 drops linearly to zero. Similarly, dashed curve 188 allows the highest value of $(dT_I/dt)_M$ for cooling or decreasing values of T_I for all compressive strain percent ratios and up to 50 percent tensile strain ratio. Between 50 percent and 100 percent tensile strain ratio, the allowed decrease ratio of change of T_I drops linearly to zero. In this case, curves 186 and 188 overlap in region 187. The linear proportional portions of the curves 186 and 188 in effect involve a prediction in a generalized sense that existing rotor strain level, although not yet excessive, probably requires a cutback in steam temperature change rate.

Other functions with or without prediction characteristics can of course be employed in determining the allowed rate of change of impulse chamber steam temperature T_I. Such functions may or may not have a flat top like that at 187 and further may or may not have a fixed highest rate of change value for T_I like that at 187, i.e. the highest allowed rate might be made to have different values under different sets of operating conditions.

If the control system is in the load operating mode as determined in block 190 and if after comparing the results of blocks 180 and 182 the casing strain is determined to be too high in block 192, the maximum value $(dT_I/dt)_M$ determined in block 184 is reduced a predetermined amount in block 194. The actual present change rate of T_I is then determined from sensed temperature and time data and it is compared to the computed allowed maximum value in block 196. If the actual value of dT_I/dt is equal to or greater than the allowed maximum value, the block 196 computes the maximum steam valve load position demand D_M. If casing strain is within limits, program execution goes directly from block 192 to block 196.

When the control system is in the speed control operating mode, a determination is made in block 198 whether the casing strain is too high in a manner similar to that described for block 192 and also whether the bore loading in either the high pressure turbine section or the intermediate pressure turbine section is too high on the basis of combining the loading determinations of blocks 174 and 176 and comparing the combinations to the limit determinations of block 178. Bore loading computation is included in the speed control operating mode because it has special significance during cold startups and the like. If either bore loading or casing strain is excessive, the maximum value $(dT_I/dt)_M$ determined in block 184 is reduced a predetermined amount in block 200.

As in the case of load constraint, the actual present change rate of T_I is computed and compared to the computed allowed maximum value $(dT_I/dt)_M$ in block 202. If the actual temperature change rate is equal to or greater than the allowed maximum value, the block 202 determines D_{SM}, i.e. in this case stops the ramping of w_R in block 147. When casing strain and bore loadings are within limits, program execution goes directly from block 198 to block 202. Further, if it is desired to use the conventional startup procedure of "rolling" the turbine at say two thirds of the synchronous speed value a predetermined and prefixed time period such as three or four hours which conservatively assures the development of safe rotor bore temperature rise, the on line bore loading determination in the block 198 can be eliminated altogether and the program 145 is appropriately adapted to the use of this prior art startup technique.

In the determination of D_M and in the general case for D_{SM}, the maximum steam temperature change rate $(dT_I/dt)_M$ is converted into a maximum steam valve position demand which will cause the steam valves 25 to be positioned from the present positions at a rate which causes steam flow change at a rate no greater than that corresponding to the maximum value $(dT_I/T)_M$. That is, with inlet steam temperature and pressure and therefore inlet steam enthalpy held substantially constant in this application, the steam valve positioning rate is limited to prevent the rate of change of steam flow from exceeding any value which would cause the rate of change of impulse chamber steam temperature T_I to exceed maximum value $(dT_I/dt)_M$. The inlet steam valves 25 are desirably positioned to setpoint values with slightly over-critical gain as described more fully in the Birnbaum and Giras application, and for this and other reasons it is preferable that impulse chamber steam temperature change rate of dT_I/dt be limited by limited the position demand levels D_S or D_C made on the positioning control loops as the present case has already been indicated to be. However, if it is desired to employ specific functions other than those described for the blocks 196 and 202, loop gain control can, as one example, be employed in the local analog valve position control loops or in direct digital computer valve position control loops (not shown) for the purpose of imposing direct dynamic constraints on steam valve positioning and in turn on turbine speed changing and/or loading when the temperature change rate dT_I/dt is to be constrained.

As a result of application of the invention, steam turbine operation is generally improved and more specifically it is made more accurate, more efficient and more economic. When the invention is embodied in closed control loop form, the degree of improvement is extended. The preferred programmed digital computer control system economically provides the capability needed for efficient control system functioning.

Extended turbine life is made possible by more accurate rotor plastic strain fatigue supervision and/or control. This economy is realized along with improved efficiency in turbine operating control directed to meeting load, speed or other end controlled variable demands. Dynamic constraint operation is better tailored to allowing optimum or near optimum turbine dynamic operation particularly in electric power plant and like applications where it is likely that the significant turbine operating level changes will often require constraint application. Achieved optimization is tied in a relative sense to operator standards for accumulation of rotor plastic strain fatigue. Bore loading and casing strain constraints are compatibly combined with the rotor strain constraint in the dynamic constraint control.

The foregoing description has been presented only to illustrate the principles of the invention. Accordingly, it is desired that the invention not be limited by the embodiment described, but, rather, that it be accorded an interpretation consistent with the scope and spirit of its broad principles.

I claim:

1. A system for operating a steam turbine comprising means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, means for determining a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation and the stress concentration of grooves or similar rotor surface structural features on the preselected rotor portion, means for controlling the turbine steam conditions in the predetermined turbine region, and means for operating said steam condition controlling means as a predetermined function of the thermal condition representation.

2. A system for operating a steam turbine comprising means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, means for determining a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation, means for determining cyclic rotor thermal plastic strain fatigue damage in accordance with a predetermined function of the one rotor thermal condition representation, means for controlling the turbine steam conditions in the predetermined turbine region, and means for operating said steam condition controlling means as another predetermined function of the thermal condition representation.

3. A steam turbine operating system as set forth in claim 1 wherein there is further provided means for determining respective representations of thermal loading and centrifugal loading of the rotor bore at least for one predetermined rotor portion, means are provided for determining a representation of rotor bore loading from the thermal and centrifugal loadings, and said operating means operates said steam condition controlling means as a predetermined function of the first mentioned thermal condition representation and the bore loading representation.

4. A method for operating a steam turbine, the steps of said method including determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, determining a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation, determining cyclic rotor thermal plastic strain fatigue damage in accordance with a predetermined function of the one rotor thermal condition representation, and using the thermal condition representation and the fatigue determination in determining the steam turbine operation.

5. A steam turbine operating method as set forth in claim 4 wherein the turbine is a large electric power plant steam turbine, the steam temperature is determined in the turbine impulse chamber, and the thermal condition representation is further determined as a function of the stress concentration of grooves or similar rotor surface structural features on the predetermined rotor portion.

6. A control system for a steam turbine having steam valve means for determining the flow of steam through at least one section of the turbine, said system comprising means for determining a representation of steam temperature in a predetermined region in heat transfer relation with a preselected turbine rotor portion, means for determining a representation of at least one predetermined rotor thermal condition of the rotor portion as a predetermined function of the steam temperature representation and the turbine speed, and means for position controlling the steam valve means to determine the turbine operating level as a predetermined function of the thermal condition representation.

7. A steam turbine control system as set froth in claim 6 wherein means are provided for determining rotor cyclic thermal plastic strain fatigue damage in accordance with another predetermined function of the rotor surface thermal condition.

8. An instrumentation system for making determinations useful in operating a steam turbine, said system comprising means for detecting steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, and means for determining a representation of a predetermined rotor thermal condition of the rotor portion as a predetermined function of the detected steam temperature and the stress concentration of grooves or similar rotor surface structural features on the selected rotor portion.

9. An instrumentation system as set forth in claim 8 wherein said determining means includes means for determining a representation of turbine speed and means for determining rotor surface temperatures as a function of the turbine speed and the detected steam temperature.

10. An instrumentation system for making determinations useful in operating a steam turbine, said system comprising means for detecting steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, means for determining a representation of a predetermined rotor thermal condition of the rotor portion as a predetermined function of the detected steam temperature, and means for determining cyclic plastic strain fatigue damage as a predetermined function of the rotor thermal condition representation.

11. An instrumentation system as set forth in claim 10 wherein said determining means includes computer means, the predetermined turbine region is the impulse chamber, the represented rotor thermal condition is selected from the rotor surface stress and strain conditions, and the rotor surface thermal condition is determined from the quantity

$$\frac{\eta\alpha}{1-\nu}(\overline{T}-T_S)$$

where T_S is the rotor surface temperature, $\overline{T}$ is the rotor volume-average temperature, α is the thermal expansion coefficient, ν is Poisson's ratio nd η is a concentration factor.

12. A digital computer control system for a steam turbine having steam valve means for determining the flow of steam through at least one section of the turbine, said system comprising a digital computer system including means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, and means for determining a representation of at least one predetermined rotor thermal condition of the rotor portion as a predetermined function of the steam temperature representation, means including said digital computer system for position controlling the steam valve means to determine the turbine operating level in a predetermined manner, and said digital computer system further including means for determining constraint action for application against the steam valve positioning control as a predetermined function of the thermal condition representation.

13. A digital computer control system for a steam turbine having steam valve means for determining the flow of steam through at least one section of the turbine, said system comprising a digital computer system including means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, and means for determining a representation of a rotor thermal condition selected from the rotor surface stress and strain conditions as a predetermined function of the steam temperature representation and the turbine speed and the stress concentration of grooves or similar rotor surface structural features on the selected rotor portion, and means including said digital computer system for position controlling the steam valve means to determine the turbine operating level as a predetermined function of the thermal condition representation.

14. A digital computer control system for a steam turbine having steam valve means for determining the flow of steam through at least one section of the turbine, said system comprising a digital computer system including means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, and means for determining a representation of at lest one predetermined rotor thermal condition of the rotor portion as a predetermined function of the steam temperature representation, means including said digital computer system for position controlling the steam valve means to determine the turbine operating level as a predetermined function of the thermal condition representation, the latter function being a function of the ratio of the representation of the rotor surface thermal condition to a predetermined limit representation of the rotor surface thermal condition and further defining a constant maximum turbine heat change rate over a predetermined range between compressive and tensile values of the ratio and defining a maximum turbine heat change rate inversely proportional to the magnitude of the rotor surface thermal condition ratio for greater compressive and tensile values than those defining the constant function range of the ratio.

15. A digital computer control system for a steam turbine having steam valve means for determining the flow of steam through at least one section of the turbine, said system comprising a digital computer system including means for determining a representation of steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, and means for determining a representation of at least one predetermined rotor thermal condition of the rotor portion as a predetermined function of the steam temperature representation, means including said digital computer system for position controlling the steam valve means to determine the turbine operating level as a predetermined function of the thermal condition representation, and said digital computer system further including means for determining cyclic rotor thermal plastic strain fatigue damage in accordance with another predetermined function of the rotor surface thermal condition representation and for accumulating the determined cyclic rotor plastic strain fatigue damage.

16. A digital computer control system as set froth in claim 15 wherein the steam turbine is a large electric power plant turbine and wherein said digital computer system further includes means for determining speed valve position demand for said position controlling means in a speed control operating mode, and means for determining load valve position demand for said valve position controlling means in a load control operating mode.

17. A digital computer control system as set forth in claim 16 wherein the selected rotor surface thermal condition is determined from the quantity

$$\frac{\eta\alpha}{1-\nu}(\bar{T}-T_S)$$

where T_S is the rotor surface temperature, $\bar{T}$ is the rotor volume-average temperature, α is the thermal expansion coefficient, ν is Poisson's ratio and η is a concentration factor, and wherein the quantity T_S is determined as a predetermined function of the turbine speed and the impulse chamber steam temperature representation.

18. A digital computer control system as set forth in claim 13 wherein a representation of a maximum rate of change of impulse chamber steam temperature is determined in accordance with another predetermined function of the rotor surface thermal condition representation, the first mentioned thermal condition function includes the other thermal condition function just mentioned, the steam valve positioning is determined as a predetermined function of the determined maximum rate of change of impulse chamber steam temperature, and the first mentioned thermal condition function also includes the valve positioning-steam temperature rate of change function.

19. A method for operating a steam turbine digital computer control system, the steps of said method comprising operating the computer to determine a representation of measured steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, operating the computer to determine a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation and the stress concentration of rotor grooves or similar rotor surfaces structural features on the predetermined rotor portion, and operating the computer to determine a turbine valve control action as a function of the rotor thermal condition representation.

20. A method for operating a steam turbine digital computer control system, the steps of said method comprising operating the computer to determine a representation of measured steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, operating the computer to determine a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation and a representation of measured turbine speed, and operating the computer to determine a turbine valve control action as a function of the rotor thermal condition representation.

21. A method for operating a steam turbine digital computer control system, the steps of said method comprising operating the computer to determine a representation of measured steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, operating the computer to determine a representation of at least one predetermined thermal condition of the rotor portion as a predetermined function of the steam temperature representation, operating the computer to determine cyclic rotor thermal plastic strain fatigue damage in accordance with a predetermined function of the rotor thermal condition representation, and operating the computer to determine a turbine valve control action as another function of the rotor thermal condition representation.

22. A method for operating a steam turbine digital computer control system as set froth in claim 21 wherein the rotor thermal condition representation is further determined as a function of the stress concentration of rotor grooves or similar rotor surface structural features on the predetermined rotor portion.

23. A method for operating a stress turbine digital computer control system as set forth in claim 22 wherein the rotor thermal condition representation is further determined as a function of a representation of measured turbine speed.

24. A method for operating a steam turbine digital computer control system, the steps of said method comprising operating the computer to determine a representation of measures steam temperature in a predetermined turbine region in heat transfer relation with a preselected turbine rotor portion, operating the computer to determine a representation of at

least one predetermined condition of the rotor portion as a predetermined function of the steam temperature representation and a rotor volume-average temperature, and operating the computer to determine a turbine valve control action as a function of the rotor thermal condition representation.

25. A method for operating a steam turbine digital computer control system as set forth in claim **24** wherein the rotor surface thermal condition is determined from the quantity

$$\frac{\eta\alpha}{1-\nu}(\bar{T}-T_S)$$

where T_S is the rotor surface temperature, $\bar{T}$ is the rotor volume-average temperature, α is the thermal expansion coefficient, ν is Poisson's ratio and η is a concentration factor.