# United States Patent [19]

**Ronnen**

[11] **3,878,401**

[45] **Apr. 15, 1975**


[54] **SYSTEM AND METHOD FOR OPERATING A TURBINE-POWERED ELECTRICAL GENERATING PLANT IN A SEQUENTIAL MODE**


[75] Inventor: **Uri G. Ronnen,** Monroeville, Pa.

[73] Assignee: **Westinghouse Electric Corporation,** Pittsburgh, Pa.



[22] Filed: **June 10, 1974**

[21] Appl. No.: **478,815**

**Related U.S. Application Data**

[63] Continuation of Ser. No. 306,979, Nov. 15, 1972, abandoned.

[52] **U.S. Cl.** ..... **290/40;** 235/151.21; 444/1; 60/646

[51] **Int. Cl.**..... **G06j 1/00;** G06f 15/06; G06f 15/56

[58] **Field of Search**..... 235/151.21, 151.34, 151.3, 235/151; 415/1,13–14, 17; 60/646; 290/40 A–40 C, 40 F; 444/1; 340/172.5



[56] **References Cited**

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| 3,588,265 | 6/1971 | Berry | 415/17 X |

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Monitoring and Automatic Control in Steam Power Stations by Process Computer; E. Doetsch & G. Hirschberg, Siemens Review XXXV(1968), No. 12; pp. 671–576.



*Primary Examiner*—Eugene G. Botz
*Assistant Examiner*—Edward J. Wise
*Attorney, Agent, or Firm*—H. W. Patterson

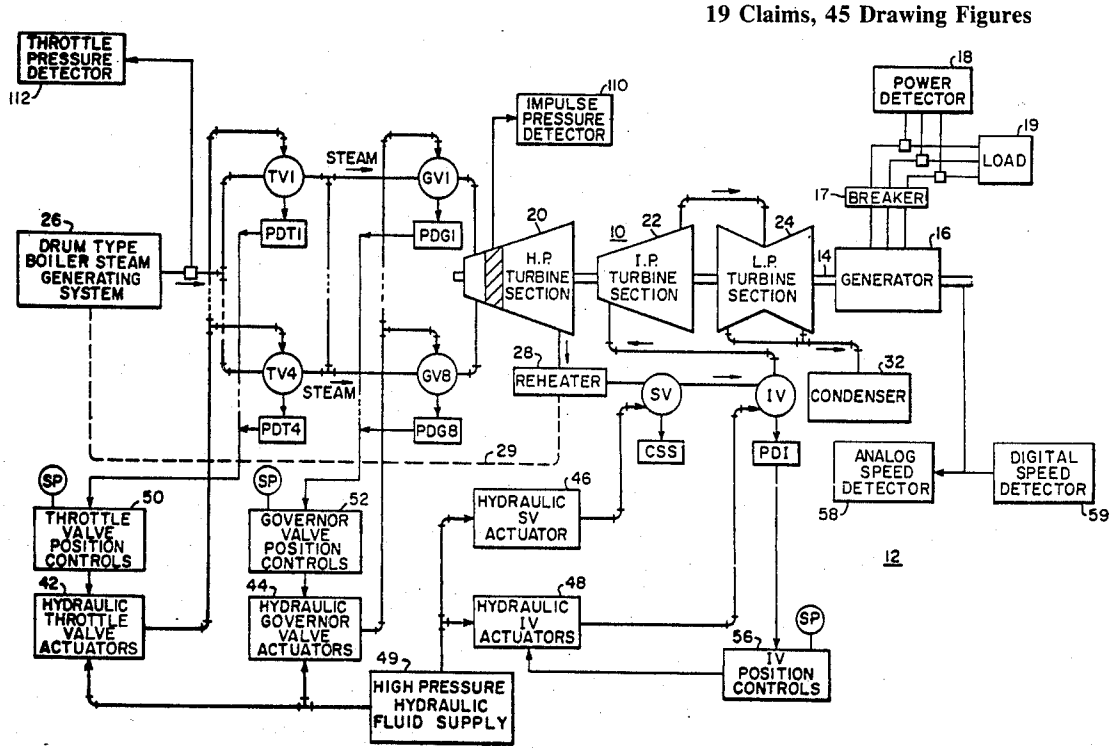


[57] **ABSTRACT**

A system and method for operating a turbine power plant wherein a feedforward steam flow demand signal governs the partial-arc or sequential operation of the steam-turbine inlet valves, is disclosed. Each valve control group is governed sequentially according to the individual valve percentage of total steam flow that such valve is required to admit. The total steam flow demand signal is dynamically calculated in accordance with a demand percentage of total flow capacity of the unit, the steam throttle pressure, the number of nozzles associated with each valve, and the effect of impulse chamber pressure.

The steam inlet valves control the steam flow in the sequential mode throughout the range of steam flow demand by incrementally varying the valve lift positions within a predetermined operating range of steam flow. The valves of one group are fully open or closed only when such position permits the steam flow demand signal to control the sequentially adjacent valve group within the predetermined operating range. The extent to which the flow demand signal varies beyond the operating range of one valve group and below the predetermined operating range of the sequentially adjacent group determines the simultaneous direction of valve travel of adjacent valve groups, and which of the adjacent valve groups are controlling in the predetermined operating range. This range is varied with respect to the actual valve position in accordance with the total flow demand signal and the predetermined operating range is varied depending on the direction of steam flow demand change.


**19 Claims, 45 Drawing Figures**



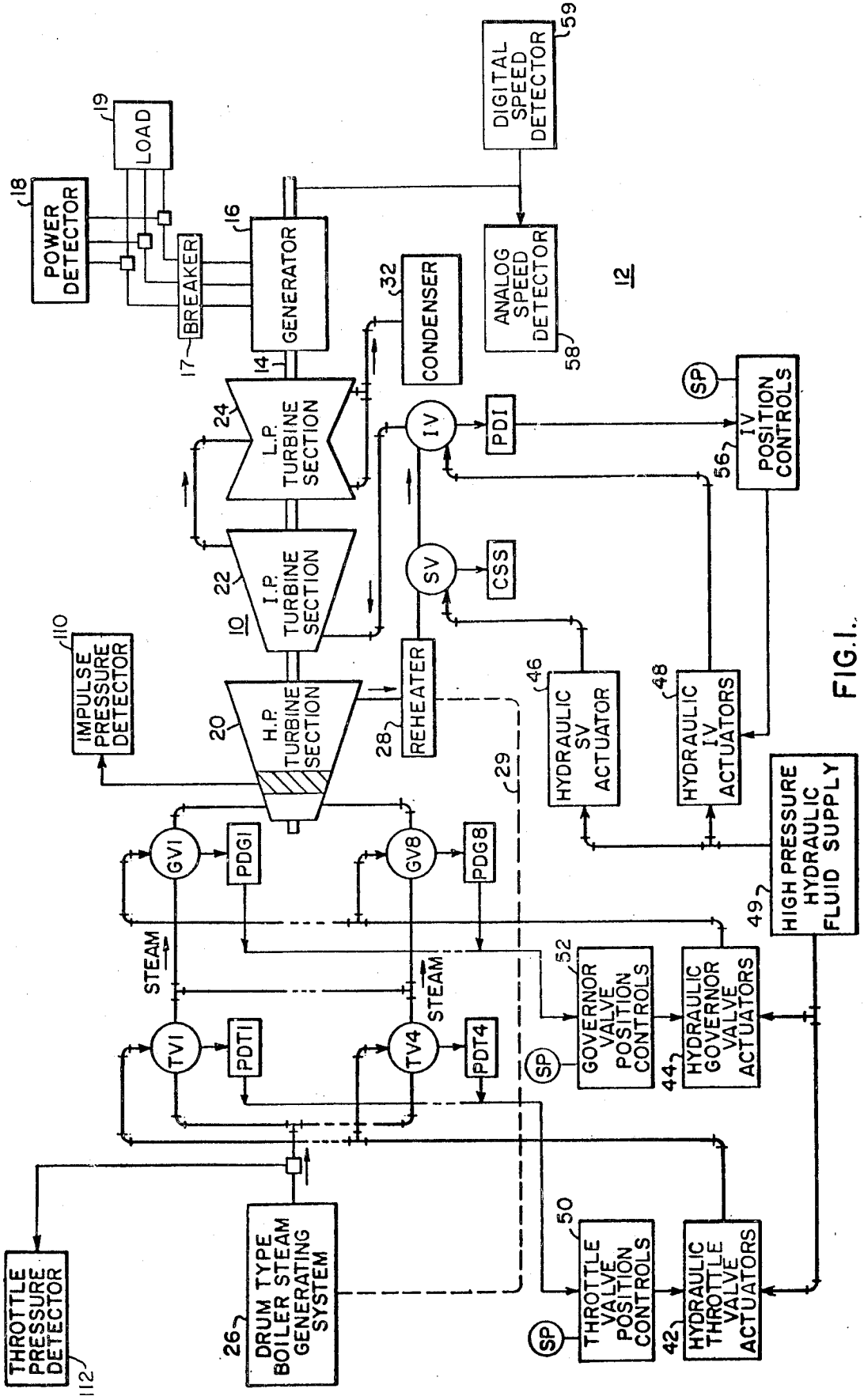

DIGITAL SPEED DETECTOR
59
LOAD
19
POWER DETECTOR
18
BREAKER
17
GENERATOR
16
CONDENSER
32
ANALOG SPEED DETECTOR
58
12
L.P. TURBINE SECTION
24
I.P. TURBINE SECTION
22
10
H.P. TURBINE SECTION
20
IV
PDI
SV
CSS
SP
IV POSITION CONTROLS
56
IMPULSE PRESSURE DETECTOR
110
REHEATER
28
29
HYDRAULIC SV ACTUATOR
46
HYDRAULIC IV ACTUATORS
48
HIGH PRESSURE HYDRAULIC FLUID SUPPLY
49
GV1
PDG1
GV8
PDG8
STEAM
TV1
PDT1
TV4
PDT4
GOVERNOR VALVE POSITION CONTROLS
52
HYDRAULIC GOVERNOR VALVE ACTUATORS
44
THROTTLE VALVE POSITION CONTROLS
50
HYDRAULIC THROTTLE VALVE ACTUATORS
42
THROTTLE PRESSURE DETECTOR
112
DRUM TYPE BOILER STEAM GENERATING SYSTEM
26

FIG. 1.

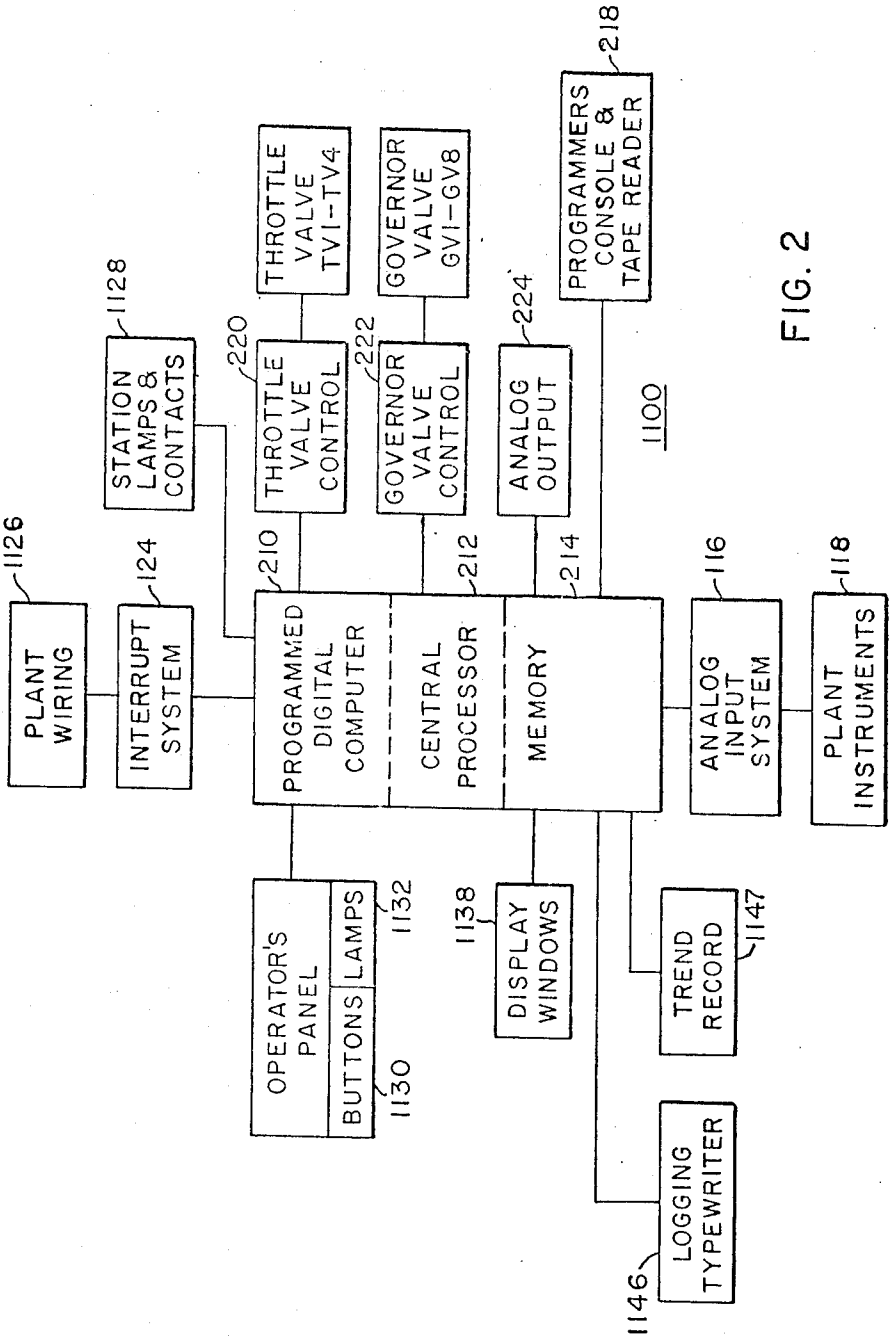

PLANT WIRING
1126
INTERRUPT SYSTEM
124
PROGRAMMED DIGITAL COMPUTER
210
CENTRAL PROCESSOR
212
MEMORY
214
STATION LAMPS & CONTACTS
1128
THROTTLE VALVE CONTROL
220
GOVERNOR VALVE CONTROL
222
ANALOG OUTPUT
224
THROTTLE VALVE TV1-TV4
GOVERNOR VALVE GV1-GV8
PROGRAMMERS CONSOLE & TAPE READER
218
1100
ANALOG INPUT SYSTEM
116
PLANT INSTRUMENTS
118
OPERATOR'S PANEL
BUTTONS
LAMPS
1132
1130
DISPLAY WINDOWS
1138
TREND RECORD
1147
LOGGING TYPEWRITER
1146

FIG. 2

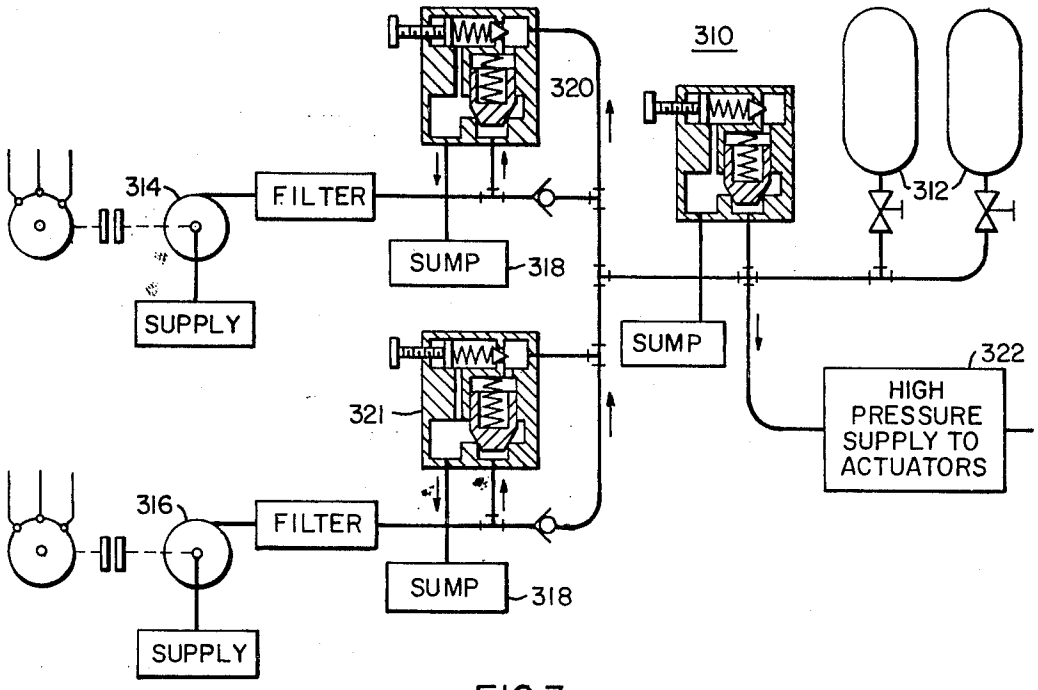

310
320
314
FILTER
SUMP
318
SUPPLY
312
SUMP
322
HIGH PRESSURE SUPPLY TO ACTUATORS
321
316
FILTER
SUMP
318
SUPPLY

FIG.3

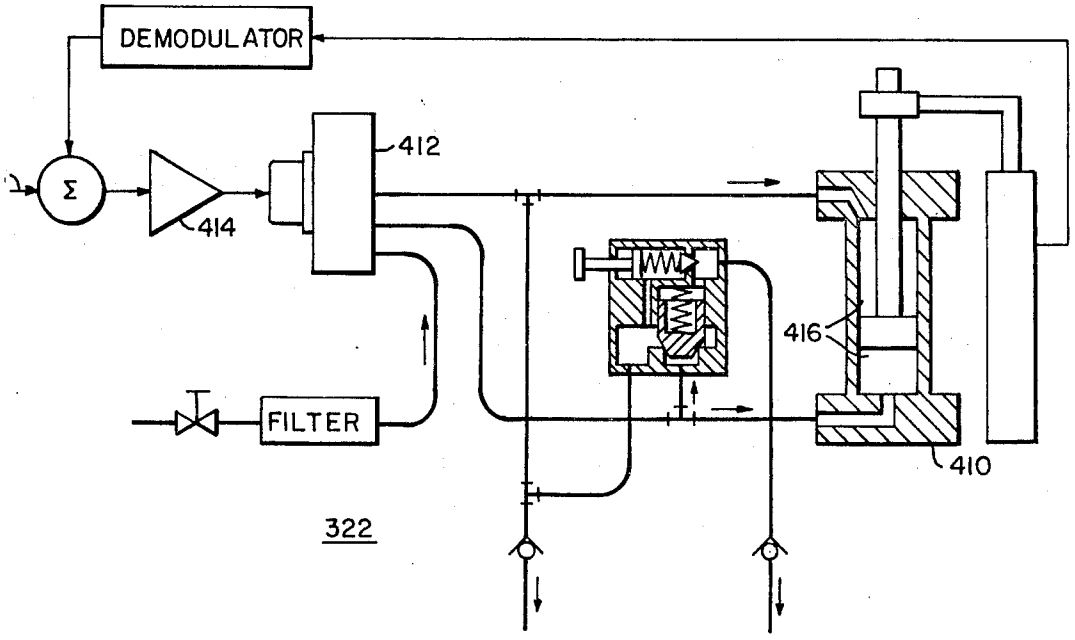

DEMODULATOR
412
Σ
414
416
FILTER
410
322

FIG.4

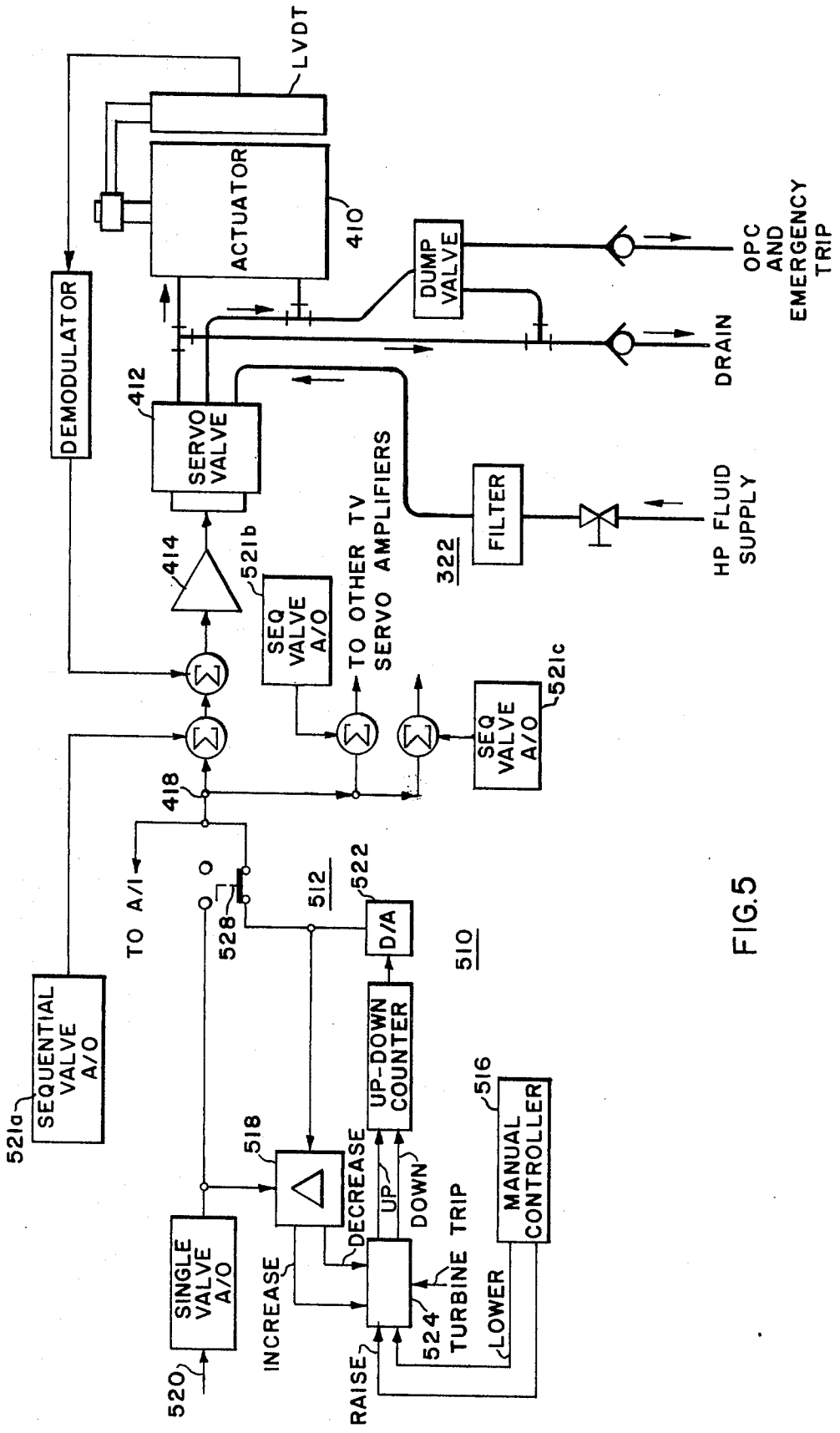

LVDT
ACTUATOR
410
DUMP VALVE
OPC AND EMERGENCY TRIP
DRAIN
DEMODULATOR
412
SERVO VALVE
414
521b
SEQ VALVE A/O
TO OTHER TV SERVO AMPLIFIERS
322
FILTER
HP FLUID SUPPLY
521c
SEQ VALVE A/O
418
TO A/I
528
512
522
D/A
510
SEQUENTIAL VALVE A/O
521a
UP-DOWN COUNTER
516
MANUAL CONTROLLER
518
DECREASE
UP
DOWN
TURBINE TRIP
SINGLE VALVE A/O
INCREASE
LOWER
520
524
RAISE

FIG.5

FIG. 6

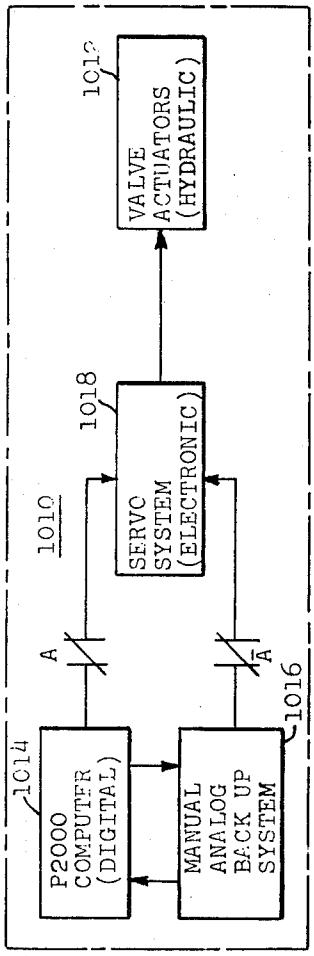
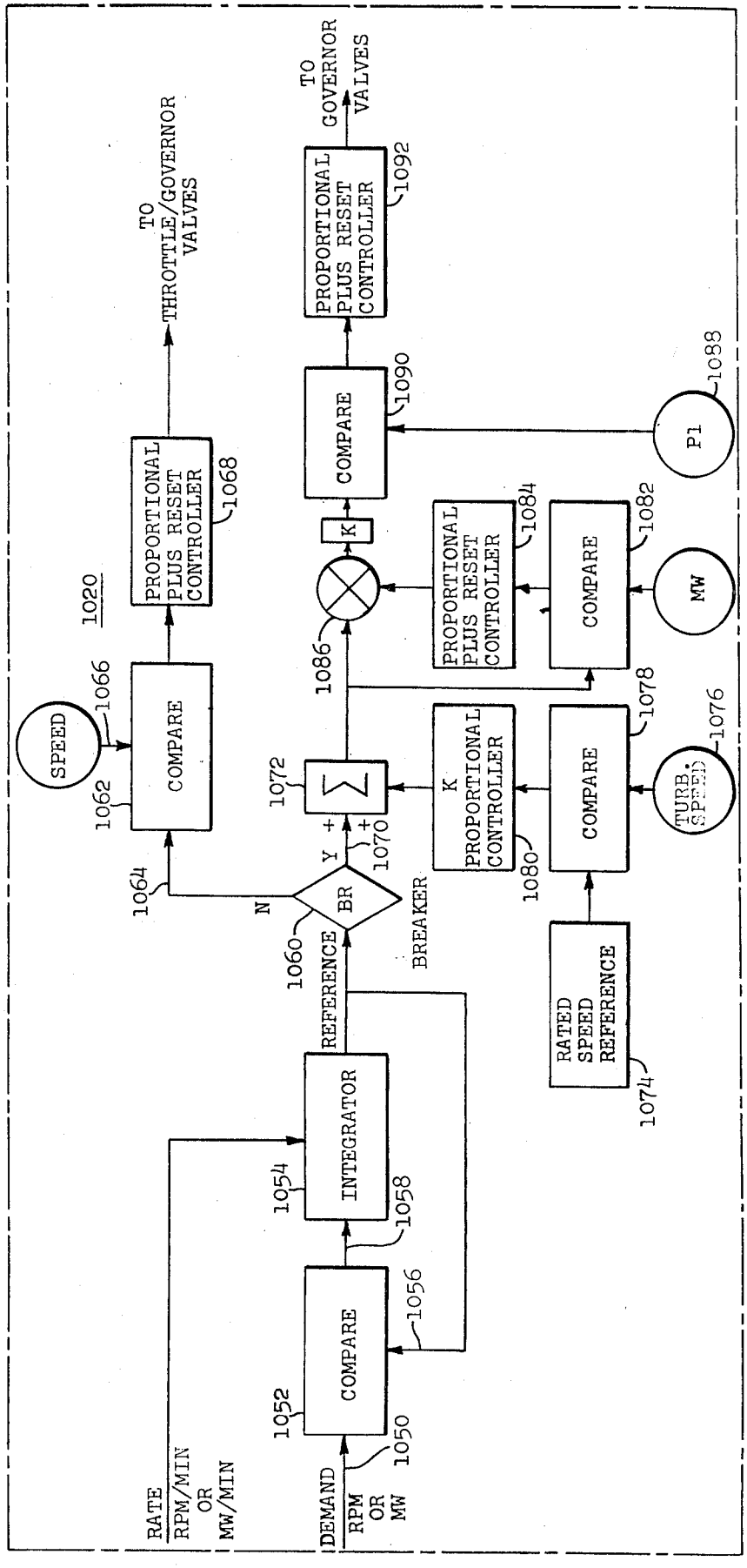

P2000 COMPUTER (DIGITAL)
MANUAL ANALOG BACK UP SYSTEM
1016
1014
A
A
1010
SERVO SYSTEM (ELECTRONIC)
1018
VALVE ACTUATORS (HYDRAULIC)

FIG. 7


RATE RPM/MIN OR MW/MIN
DEMAND RPM OR MW
1050
1052
COMPARE
1056
1058
1054
INTEGRATOR
REFERENCE
1060
N
BR
Y
1064
1062
COMPARE
SPEED
1066
1020
PROPORTIONAL PLUS RESET CONTROLLER
1068
TO THROTTLE/GOVERNOR VALVES
BREAKER
1070
1072
K PROPORTIONAL CONTROLLER
1080
COMPARE
1078
TURB. SPEED
1076
RATED SPEED REFERENCE
1074
1086
K
PROPORTIONAL PLUS RESET CONTROLLER
1084
COMPARE
1082
MW
COMPARE
1090
PROPORTIONAL PLUS RESET CONTROLLER
1092
TO GOVERNOR VALVES
P1
1088

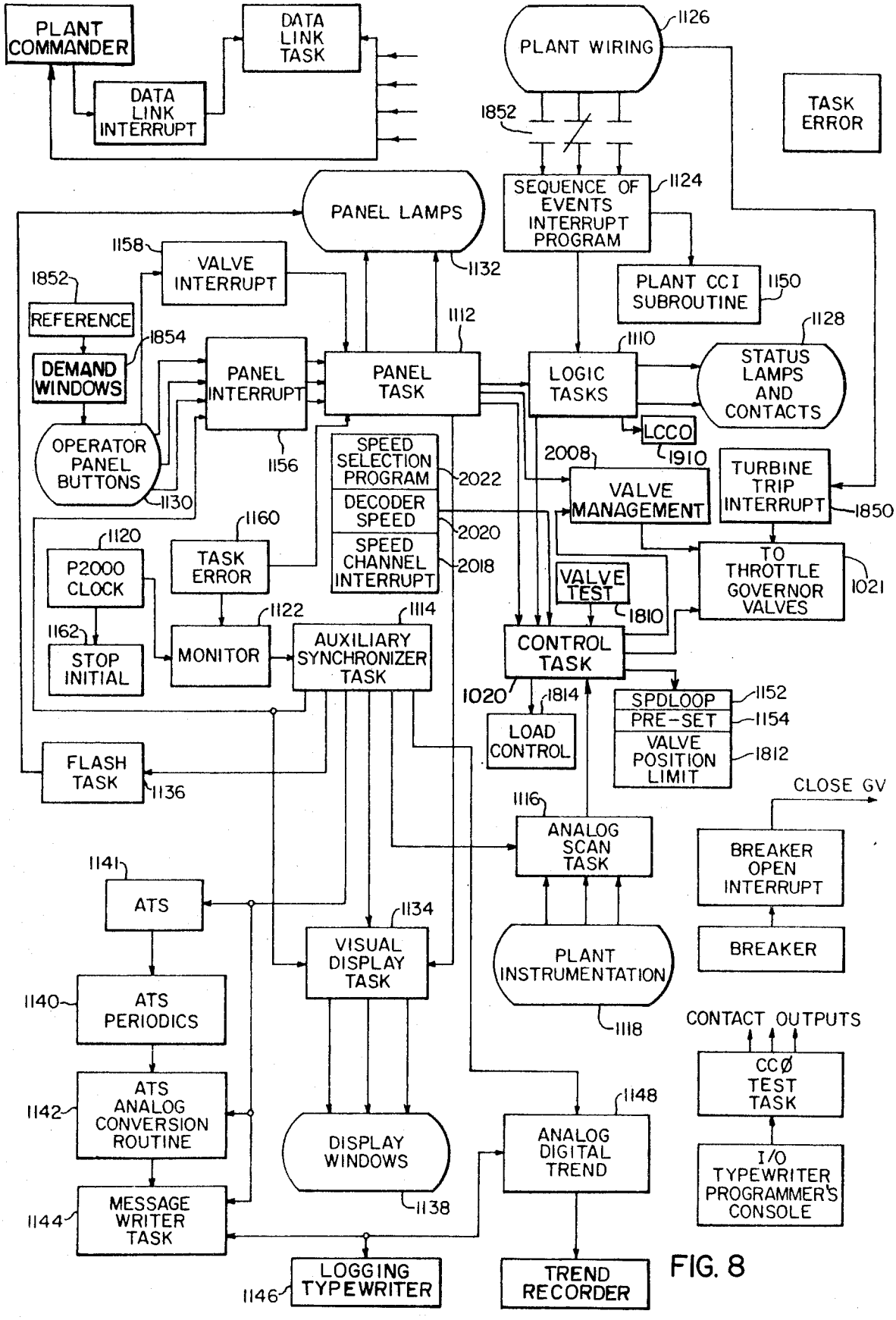

PLANT COMMANDER
DATA LINK TASK
DATA LINK INTERRUPT
PLANT WIRING
1126
TASK ERROR
1852
SEQUENCE OF EVENTS INTERRUPT PROGRAM
1124
PANEL LAMPS
1132
1158
VALVE INTERRUPT
1852
REFERENCE
1854
DEMAND WINDOWS
PANEL INTERRUPT
1156
PANEL TASK
1112
PLANT CCI SUBROUTINE
1150
1128
STATUS LAMPS AND CONTACTS
1110
LOGIC TASKS
LCCO
1910
OPERATOR PANEL BUTTONS
1130
SPEED SELECTION PROGRAM
2022
DECODER SPEED
2020
SPEED CHANNEL INTERRUPT
2018
2008
VALVE MANAGEMENT
TURBINE TRIP INTERRUPT
1850
1160
TASK ERROR
1120
P2000 CLOCK
1162
STOP INITIAL
1122
MONITOR
1114
AUXILIARY SYNCHRONIZER TASK
VALVE TEST
1810
TO THROTTLE GOVERNOR VALVES
1021
CONTROL TASK
1020
1814
LOAD CONTROL
SPDLOOP
PRE-SET
VALVE POSITION LIMIT
1152
1154
1812
FLASH TASK
1136
CLOSE GV
1116
ANALOG SCAN TASK
BREAKER OPEN INTERRUPT
BREAKER
1141
ATS
1134
VISUAL DISPLAY TASK
PLANT INSTRUMENTATION
1118
1140
ATS PERIODICS
CONTACT OUTPUTS
CCØ TEST TASK
1142
ATS ANALOG CONVERSION ROUTINE
1148
DISPLAY WINDOWS
1138
ANALOG DIGITAL TREND
I/O TYPEWRITER PROGRAMMER'S CONSOLE
1144
MESSAGE WRITER TASK
1146
LOGGING TYPEWRITER
TREND RECORDER

FIG. 8

TABLE 1-1. TASK PRIORITY ASSIGNMENT

FIG. 9

| Level | Function | Frequency | Core Location |
|---|---|---|---|
| F | STOP/INITIALIZE | ON DEMAND | 2F40 |
| E | AUXILIARY SYNCHRONIZER | 0.1 SEC | 14BD |
| D | CONTROL | 1.0 SEC | 2730 |
| C | OPERATOR'S PANEL | ON DEMAND | 21B0 |
| B | ANALOG SCAN | 0.5 SEC | 16D0 |
| A | ATS-PERIODICS | 1.0 SEC | 4420 |
| 9 | LOGIC | ON DEMAND | 1962 |
| 8 | VISUAL DISPLAY | 1.0 SEC | 1E60 |
| 7 | DATA LINK | ON DEMAND | 3D10 |
| 6 | ATS-ANALOG CONVERSIONS | 5.0 SEC | 6960 |
| 5 | FLASH | 0.5 SEC | 15A0 |
| 4 | PROGRAMMER'S CONSOLE | ON DEMAND | 3000 |
| 3 | ATS-MESSAGE WRITER | 5.0 SEC | 6CA0 |
| 2 | ANALOG/DIGITAL TREND | 1.0 SEC | 3E70 |
| 1 | CCO TEST* | ON DEMAND | 0E80 |
| 0 | BATCH PROCESSORS** | ON DEMAND | 4000 |

*The CCO test task may be used only during maintenance and debugging periods, since this program overlays the data link program area.

**The batch processors may be used only on manual control and with the sync disabled; also, the sequence of events interrupt must be disabled since the batch processor programs overlay the ATS program area.

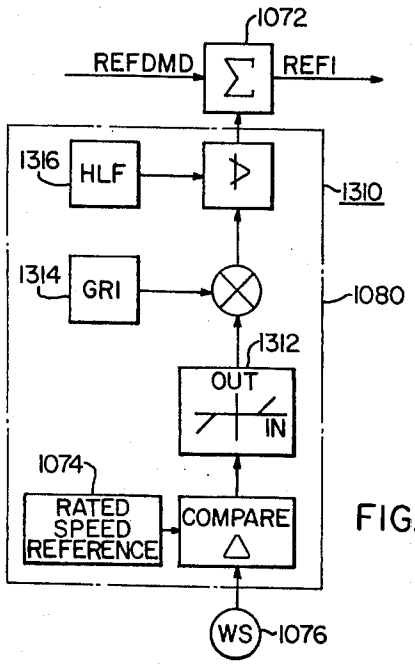

1072
REFDMD
Σ
REFI
1316
HLF
1310
1314
GRI
1080
1312
OUT
IN
1074
RATED SPEED REFERENCE
COMPARE
WS
1076

FIG. 10

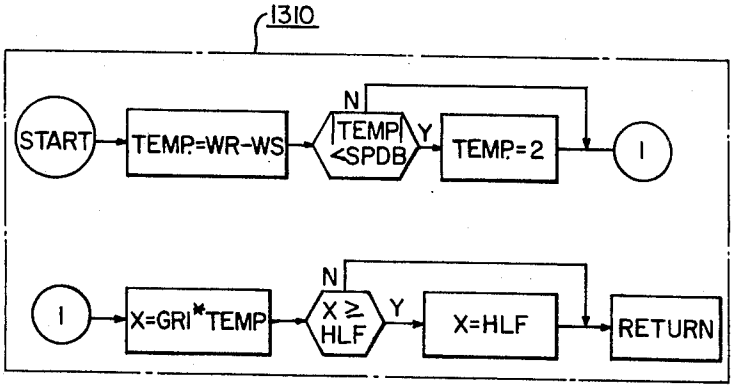

1310
START
TEMP=WR-WS
N
TEMP <SPDB
Y
TEMP=2
1
1
X=GRI*TEMP
N
X ≥ HLF
Y
X=HLF
RETURN

FIG. 11

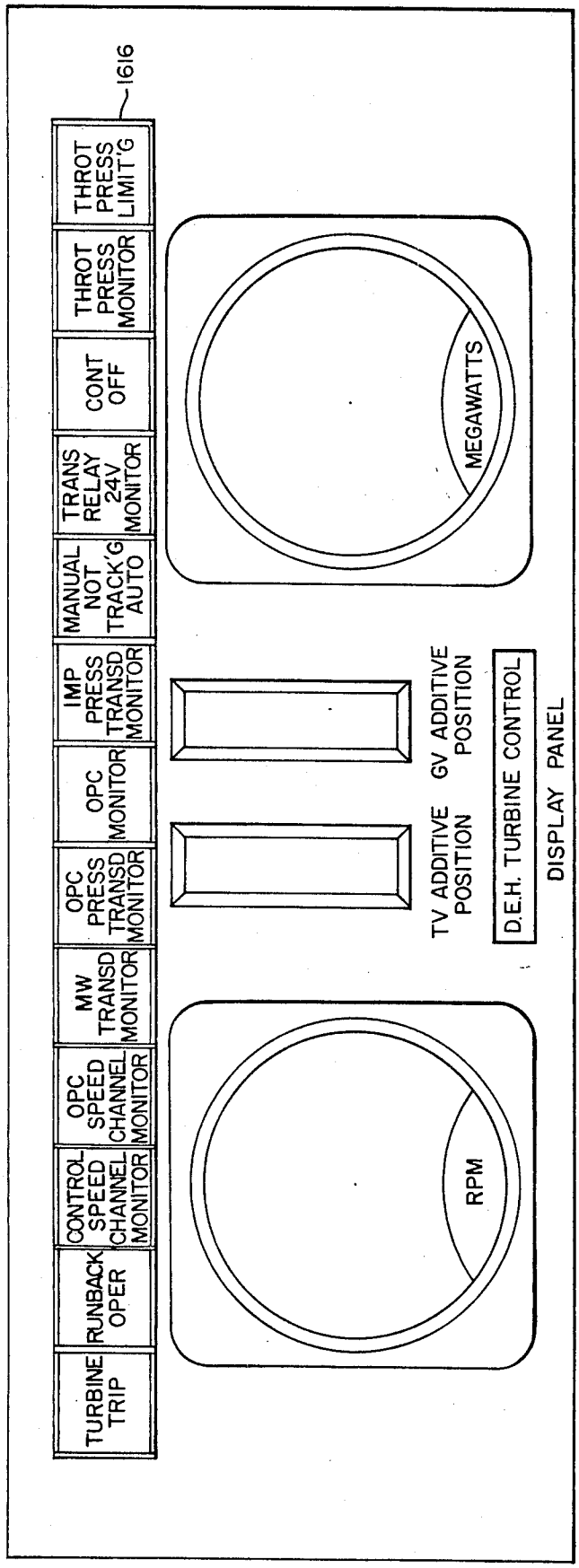

1616
TURBINE TRIP
RUNBACK OPER
CONTROL SPEED CHANNEL MONITOR
OPC SPEED CHANNEL MONITOR
MW TRANSD MONITOR
OPC PRESS TRANSD MONITOR
OPC MONITOR
IMP PRESS TRANSD MONITOR
MANUAL NOT TRACK'G AUTO
TRANS RELAY 24V MONITOR
CONT OFF
THROT PRESS MONITOR
THROT PRESS LIMIT'G
RPM
MEGAWATTS
TV ADDITIVE POSITION
GV ADDITIVE POSITION
D.E.H. TURBINE CONTROL
DISPLAY PANEL

FIG. 12

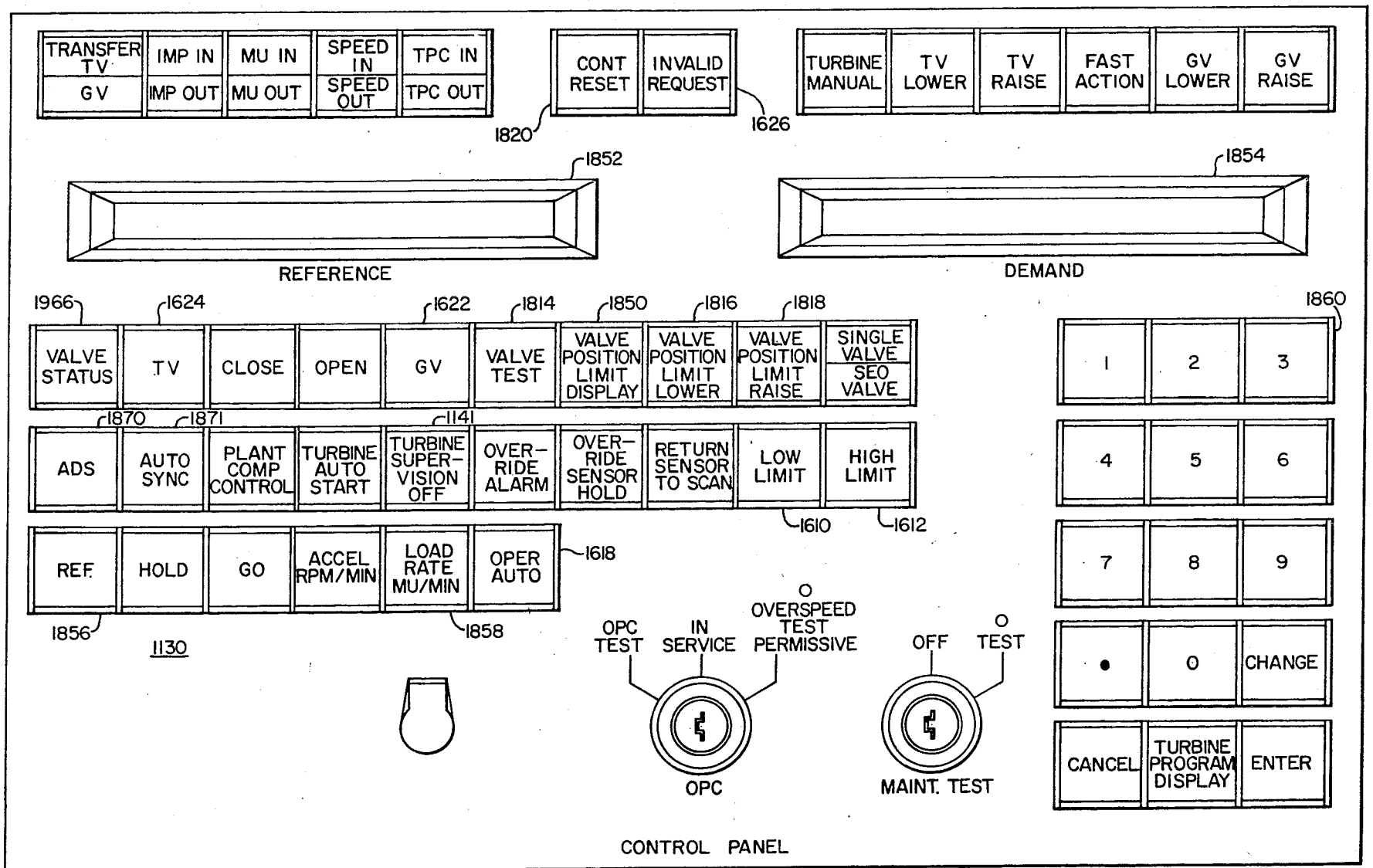

TRANSFER TV
GV
IMP IN
IMP OUT
MU IN
MU OUT
SPEED IN
SPEED OUT
TPC IN
TPC OUT
CONT RESET
INVALID REQUEST
1820
1626
TURBINE MANUAL
TV LOWER
TV RAISE
FAST ACTION
GV LOWER
GV RAISE
1852
REFERENCE
1854
DEMAND
1966
1624
1622
1814
1850
1816
1818
1860
VALVE STATUS
TV
CLOSE
OPEN
GV
VALVE TEST
VALVE POSITION LIMIT DISPLAY
VALVE POSITION LIMIT LOWER
VALVE POSITION LIMIT RAISE
SINGLE VALVE
SEQ VALVE
1870
1871
1141
ADS
AUTO SYNC
PLANT COMP CONTROL
TURBINE AUTO START
TURBINE SUPER-VISION OFF
OVER-RIDE ALARM
OVER-RIDE SENSOR HOLD
RETURN SENSOR TO SCAN
LOW LIMIT
HIGH LIMIT
1610
1612
REF
HOLD
GO
ACCEL RPM/MIN
LOAD RATE MU/MIN
OPER AUTO
1618
1856
1858
1130
1
2
3
4
5
6
7
8
9
•
0
CHANGE
CANCEL
TURBINE PROGRAM DISPLAY
ENTER
OPC TEST
IN SERVICE
OVERSPEED TEST PERMISSIVE
OPC
OFF
TEST
MAINT. TEST
CONTROL PANEL

FIG. 13

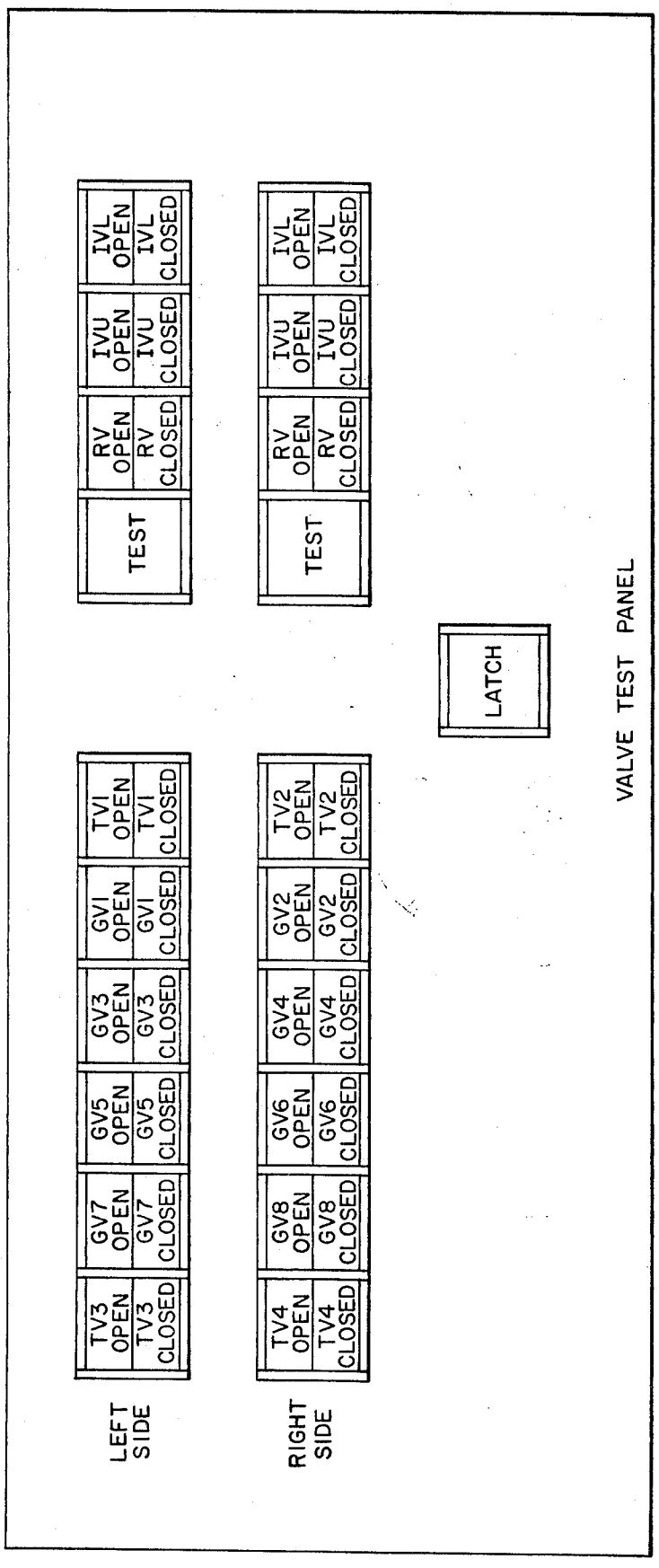

VALVE TEST PANEL
TEST
RV OPEN
RV CLOSED
IVU OPEN
IVU CLOSED
IVL OPEN
IVL CLOSED
TEST
RV OPEN
RV CLOSED
IVU OPEN
IVU CLOSED
IVL OPEN
IVL CLOSED
LATCH
LEFT SIDE
TV3 OPEN
TV3 CLOSED
GV7 OPEN
GV7 CLOSED
GV5 OPEN
GV5 CLOSED
GV3 OPEN
GV3 CLOSED
GVI OPEN
GVI CLOSED
TVI OPEN
TVI CLOSED
RIGHT SIDE
TV4 OPEN
TV4 CLOSED
GV8 OPEN
GV8 CLOSED
GV6 OPEN
GV6 CLOSED
GV4 OPEN
GV4 CLOSED
GV2 OPEN
GV2 CLOSED
TV2 OPEN
TV2 CLOSED

FIG. 14

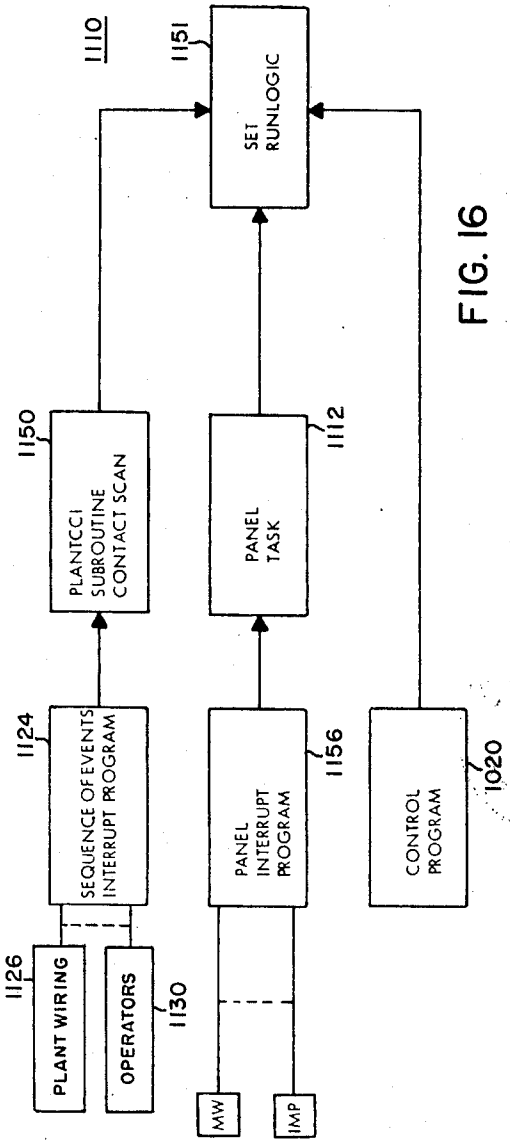

1110
1151
SET RUNLOGIC
1150
PLANTCCI SUBROUTINE CONTACT SCAN
1112
PANEL TASK
1124
SEQUENCE OF EVENTS INTERRUPT PROGRAM
1156
PANEL INTERRUPT PROGRAM
1020
CONTROL PROGRAM
1126
PLANT WIRING
1130
OPERATORS
MW
IMP

FIG. 16

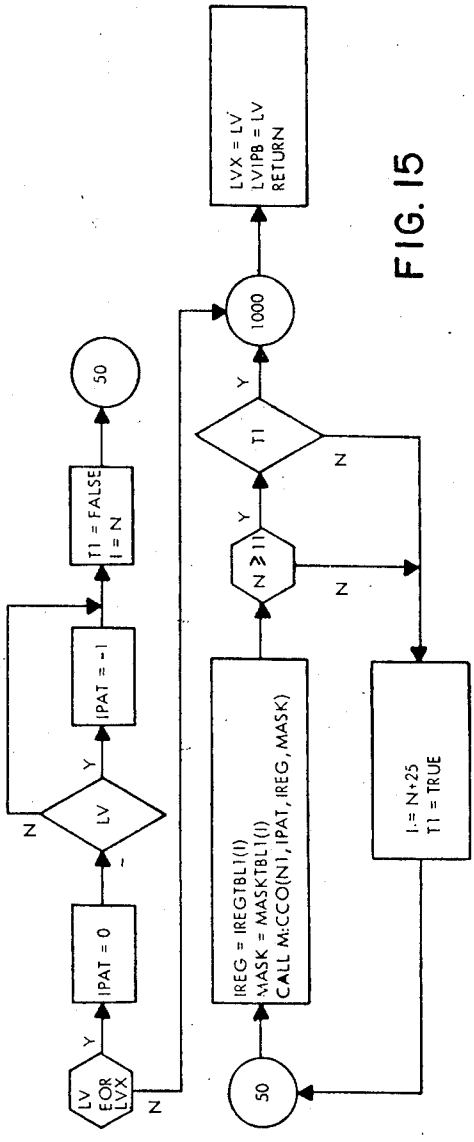

LV EOR LVX
Y
N
IPAT = 0
LV
Y
N
IPAT = -1
T1 = FALSE
I = N
50
LVX = LV
LVIPB = LV
RETURN
1000
T1
Y
N
N ≥ 11
Y
N
IREG = IREGTBL1(I)
MASK = MASKTBL1(I)
CALL MCCO(N1, IPAT, IREG, MASK)
I = N+25
T1 = TRUE
50

FIG. 15

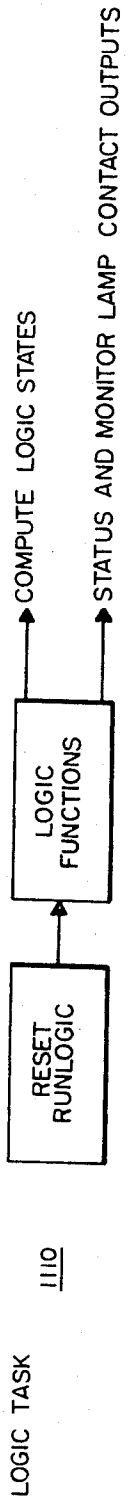

LOGIC TASK
1110
RESET RUNLOGIC
LOGIC FUNCTIONS
COMPUTE LOGIC STATES
STATUS AND MONITOR LAMP CONTACT OUTPUTS

FIG. 17

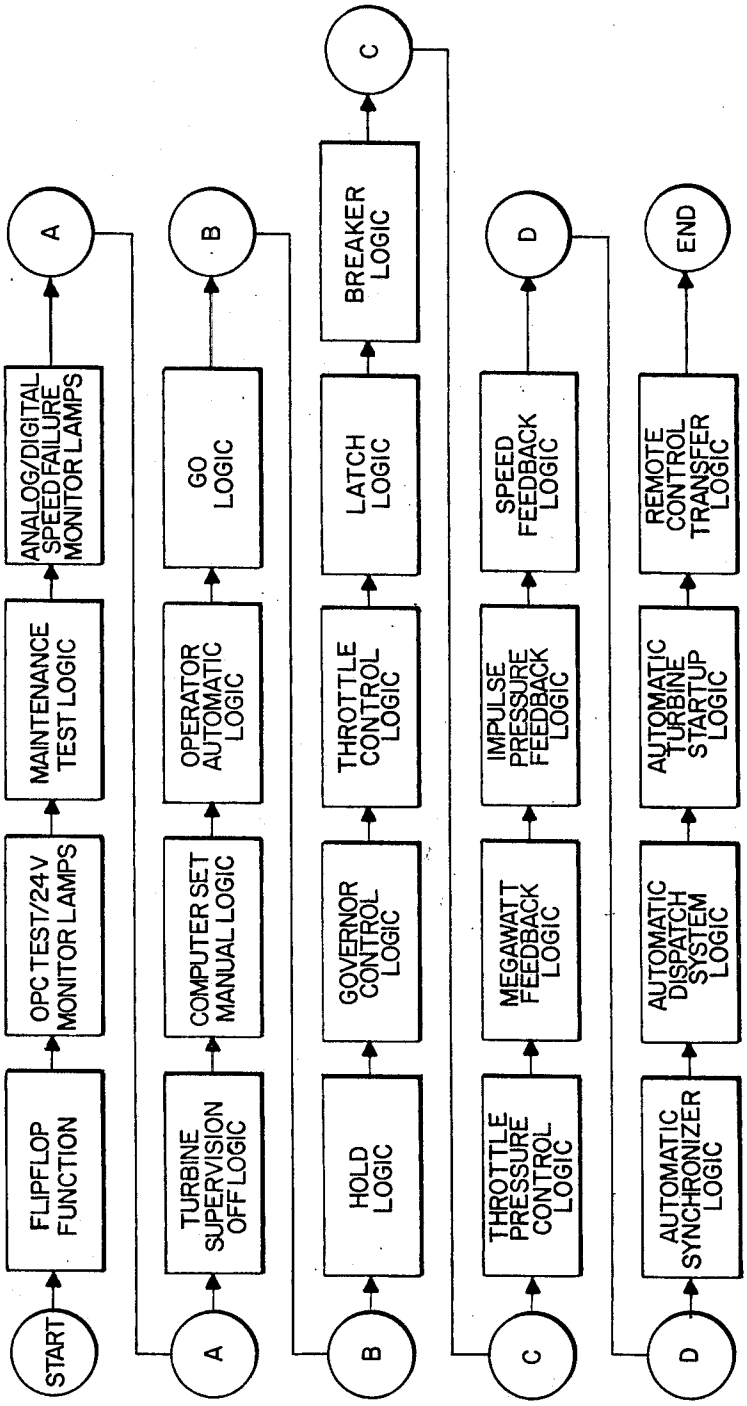

START
FLIPFLOP FUNCTION
OPC TEST/24 V MONITOR LAMPS
MAINTENANCE TEST LOGIC
ANALOG/DIGITAL SPEED FAILURE MONITOR LAMPS
A
A
TURBINE SUPERVISION OFF LOGIC
COMPUTER SET MANUAL LOGIC
OPERATOR AUTOMATIC LOGIC
GO LOGIC
B
B
HOLD LOGIC
GOVERNOR CONTROL LOGIC
THROTTLE CONTROL LOGIC
LATCH LOGIC
BREAKER LOGIC
C
C
THROTTLE PRESSURE CONTROL LOGIC
MEGAWATT FEEDBACK LOGIC
IMPULSE PRESSURE FEEDBACK LOGIC
SPEED FEEDBACK LOGIC
D
D
AUTOMATIC SYNCHRONIZER LOGIC
AUTOMATIC DISPATCH SYSTEM LOGIC
AUTOMATIC TURBINE STARTUP LOGIC
REMOTE CONTROL TRANSFER LOGIC
END

FIG. 18

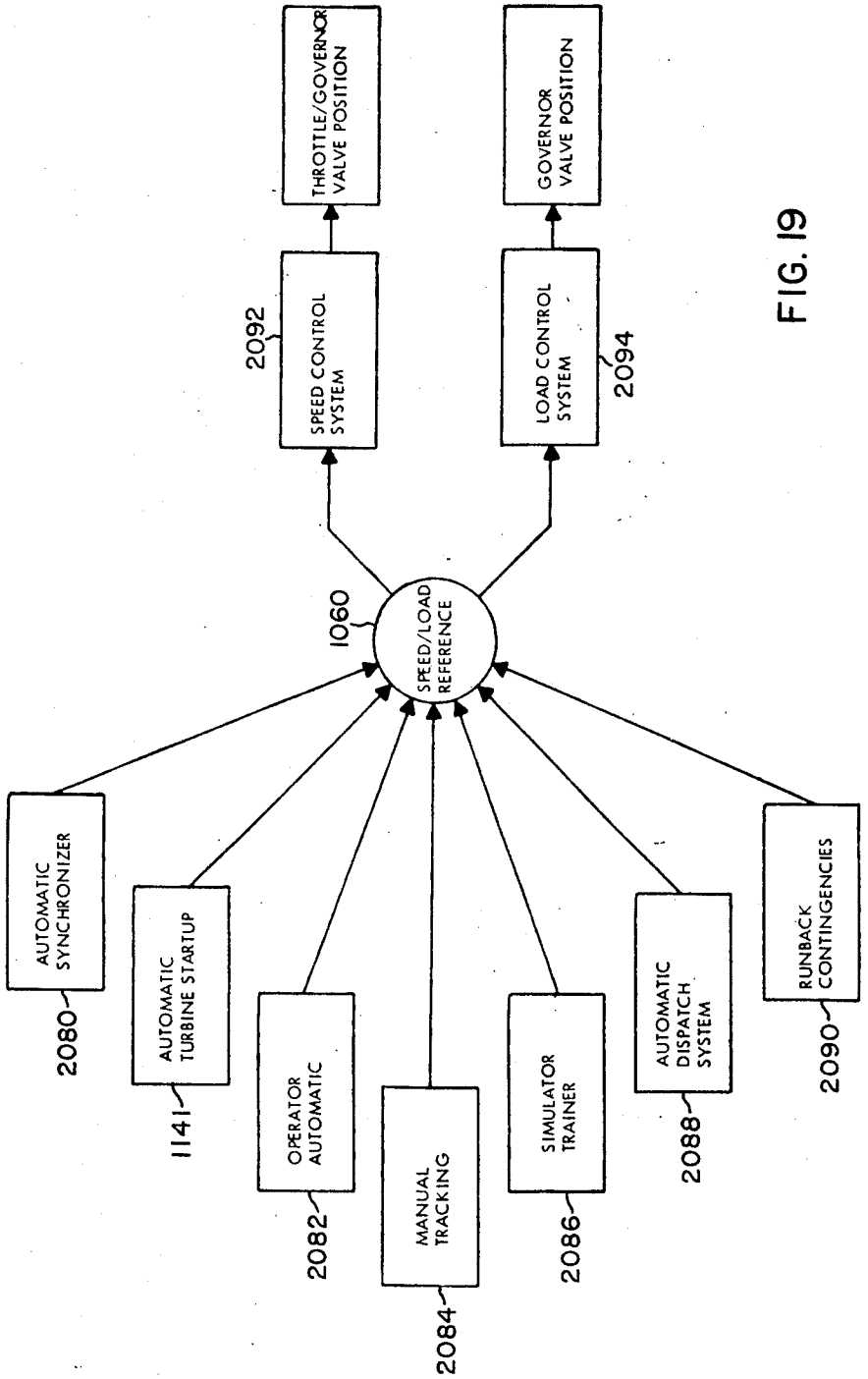
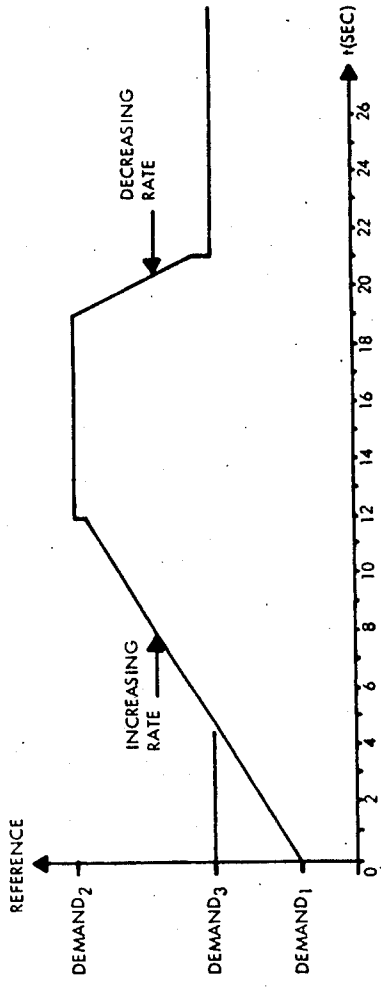

THROTTLE/GOVERNOR VALVE POSITION
GOVERNOR VALVE POSITION
2092
SPEED CONTROL SYSTEM
LOAD CONTROL SYSTEM
2094
1060
SPEED/LOAD REFERENCE
AUTOMATIC SYNCHRONIZER
2080
AUTOMATIC TURBINE STARTUP
1141
OPERATOR AUTOMATIC
2082
MANUAL TRACKING
2084
SIMULATOR TRAINER
2086
AUTOMATIC DISPATCH SYSTEM
2088
RUNBACK CONTINGENCIES
2090

FIG. 19

FIG. 20


DECREASING RATE
INCREASING RATE
REFERENCE
DEMAND2
DEMAND3
DEMAND1
t(SEC)
0
2
4
6
8
10
12
14
16
18
20
22
24
26

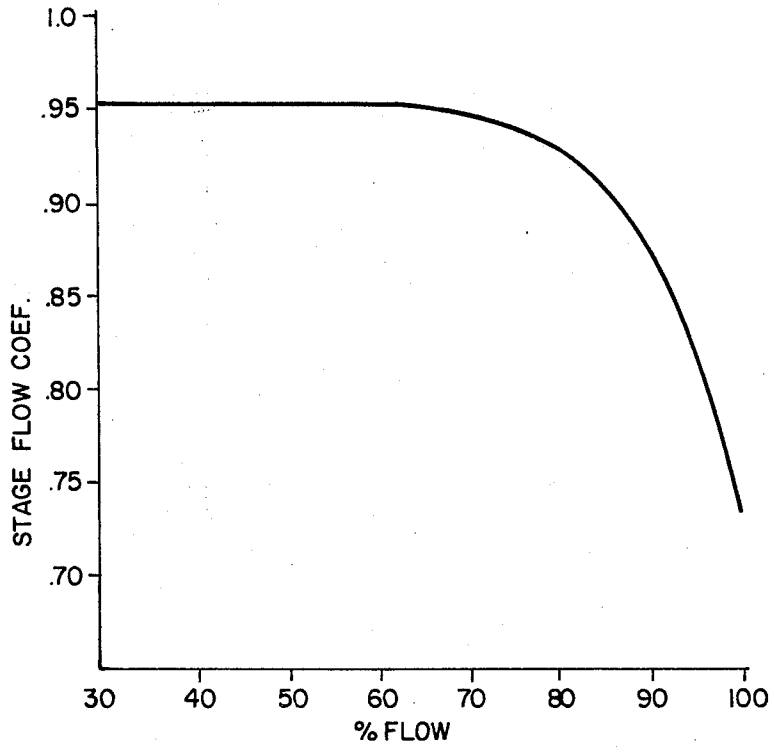

1.0
.95
.90
.85
.80
.75
.70
STAGE FLOW COEF.
30
40
50
60
70
80
90
100
% FLOW

FIG.21

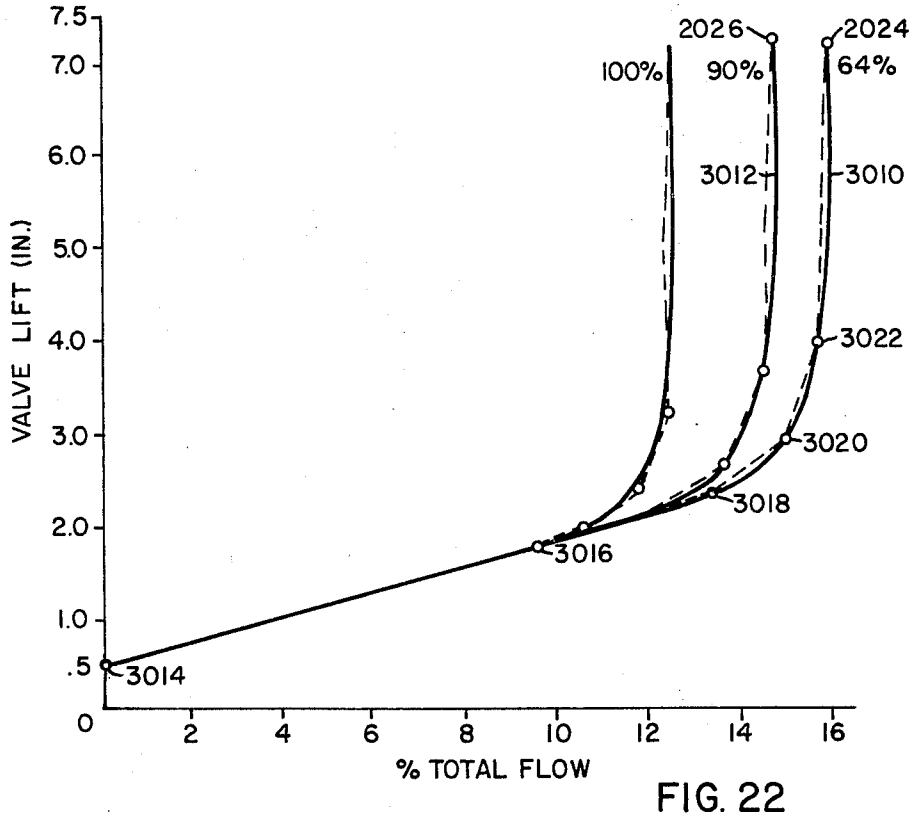

7.5
7.0
6.0
5.0
4.0
3.0
2.0
1.0
.5
0
VALVE LIFT (IN.)
2
4
6
8
10
12
14
16
% TOTAL FLOW
2026
2024
100%
90%
64%
3012
3010
3022
3020
3018
3016
3014

FIG. 22

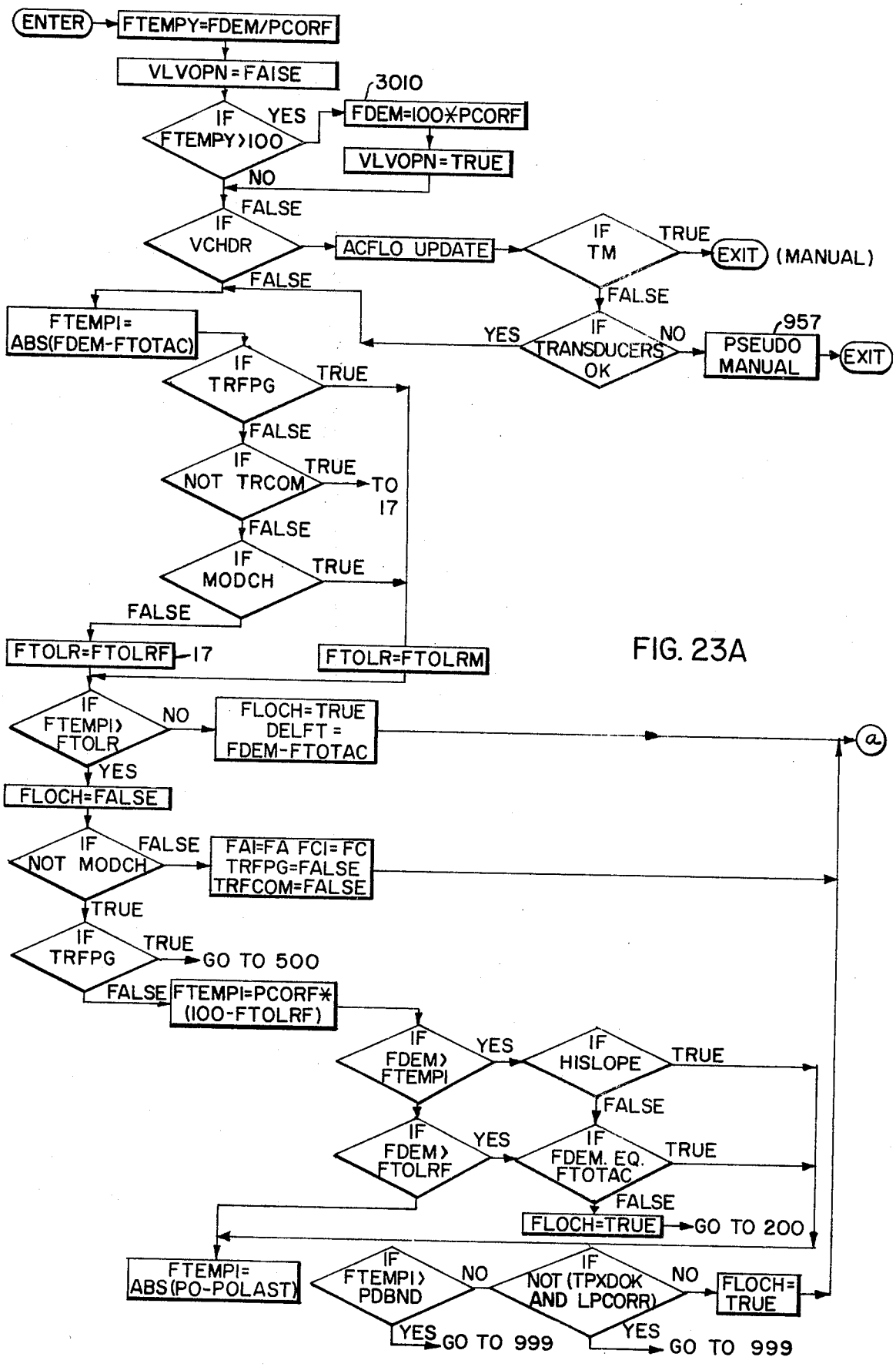

ENTER
FTEMPY=FDEM/PCORF
VLVOPN=FAISE
3010
IF FTEMPY>100
YES
FDEM=100*PCORF
VLVOPN=TRUE
NO
FALSE
IF VCHDR
ACFLO UPDATE
IF TM
TRUE
EXIT
(MANUAL)
FALSE
FALSE
FTEMPI= ABS(FDEM-FTOTAC)
YES
IF TRANSDUCERS OK
NO
957
PSEUDO MANUAL
EXIT
IF TRFPG
TRUE
FALSE
IF NOT TRCOM
TRUE
TO 17
FALSE
IF MODCH
TRUE
FALSE
FTOLR=FTOLRF
17
FTOLR=FTOLRM
IF FTEMPI> FTOLR
NO
FLOCH=TRUE DELFT = FDEM-FTOTAC
YES
FLOCH=FALSE
IF NOT MODCH
FALSE
FAI=FA FCI=FC TRFPG=FALSE TRFCOM=FALSE
TRUE
IF TRFPG
TRUE
GO TO 500
FALSE
FTEMPI=PCORF*(100-FTOLRF)
IF FDEM> FTEMPI
YES
IF HISLOPE
TRUE
FALSE
IF FDEM> FTOLRF
YES
IF FDEM. EQ. FTOTAC
TRUE
FALSE
FLOCH=TRUE
GO TO 200
FTEMPI= ABS(PO-POLAST)
IF FTEMPI> PDBND
NO
IF NOT (TPXDOK AND LPCORR)
NO
FLOCH= TRUE
YES
GO TO 999
YES
GO TO 999
a

FIG. 23A

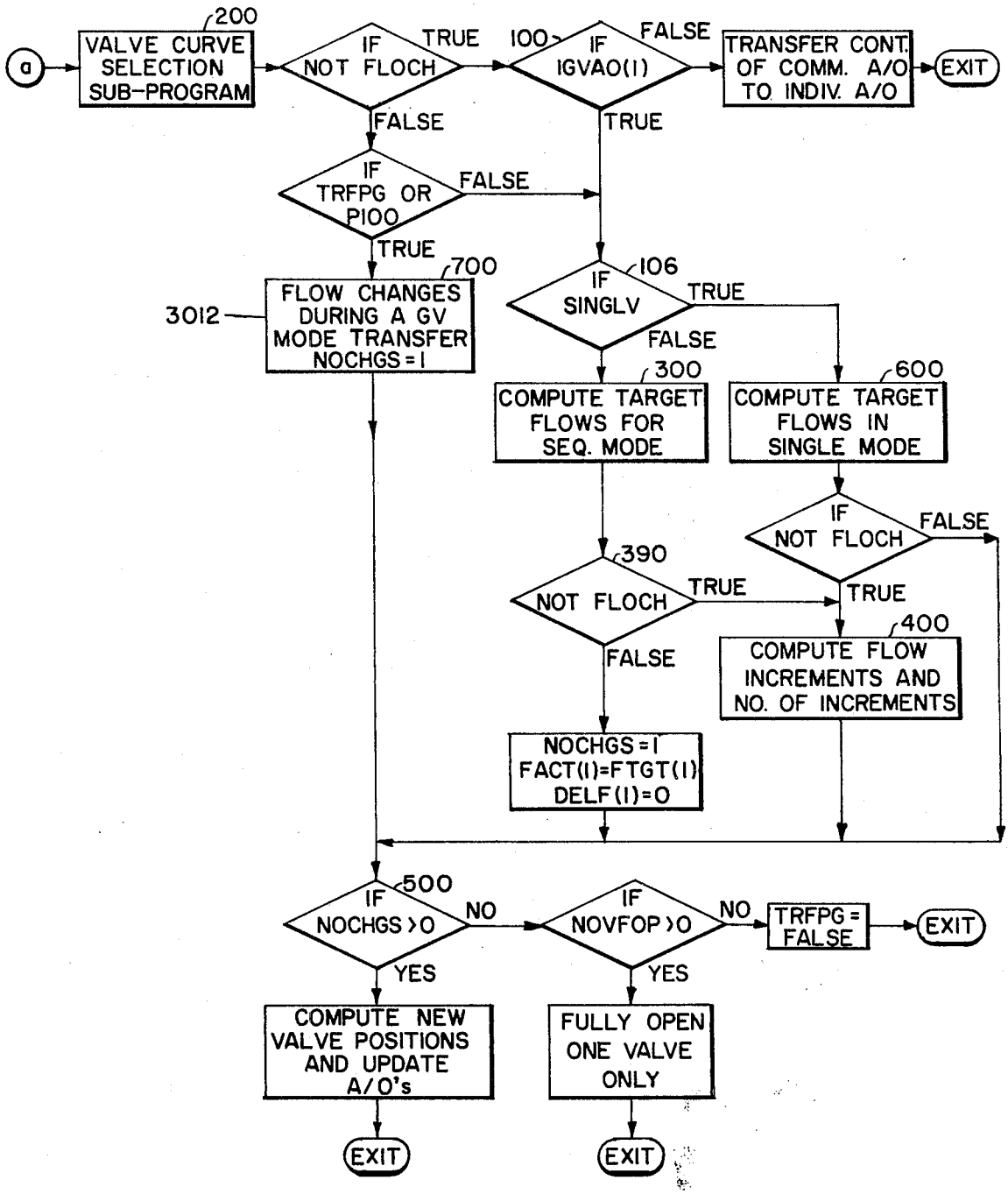

a
200
VALVE CURVE SELECTION SUB-PROGRAM
IF NOT FLOCH
TRUE
FALSE
100
IF IGVAO(I)
FALSE
TRUE
TRANSFER CONT. OF COMM. A/O TO INDIV. A/O
EXIT
IF TRFPG OR PIOO
FALSE
TRUE
700
3012
FLOW CHANGES DURING A GV MODE TRANSFER NOCHGS=1
106
IF SINGLV
TRUE
FALSE
300
COMPUTE TARGET FLOWS FOR SEQ. MODE
600
COMPUTE TARGET FLOWS IN SINGLE MODE
IF NOT FLOCH
FALSE
TRUE
390
NOT FLOCH
TRUE
FALSE
400
COMPUTE FLOW INCREMENTS AND NO. OF INCREMENTS
NOCHGS=1
FACT(I)=FTGT(I)
DELF(I)=0
500
IF NOCHGS>0
NO
YES
IF NOVFOP>0
NO
YES
TRFPG=FALSE
EXIT
COMPUTE NEW VALVE POSITIONS AND UPDATE A/O's
FULLY OPEN ONE VALVE ONLY
EXIT
EXIT

FIG. 23B

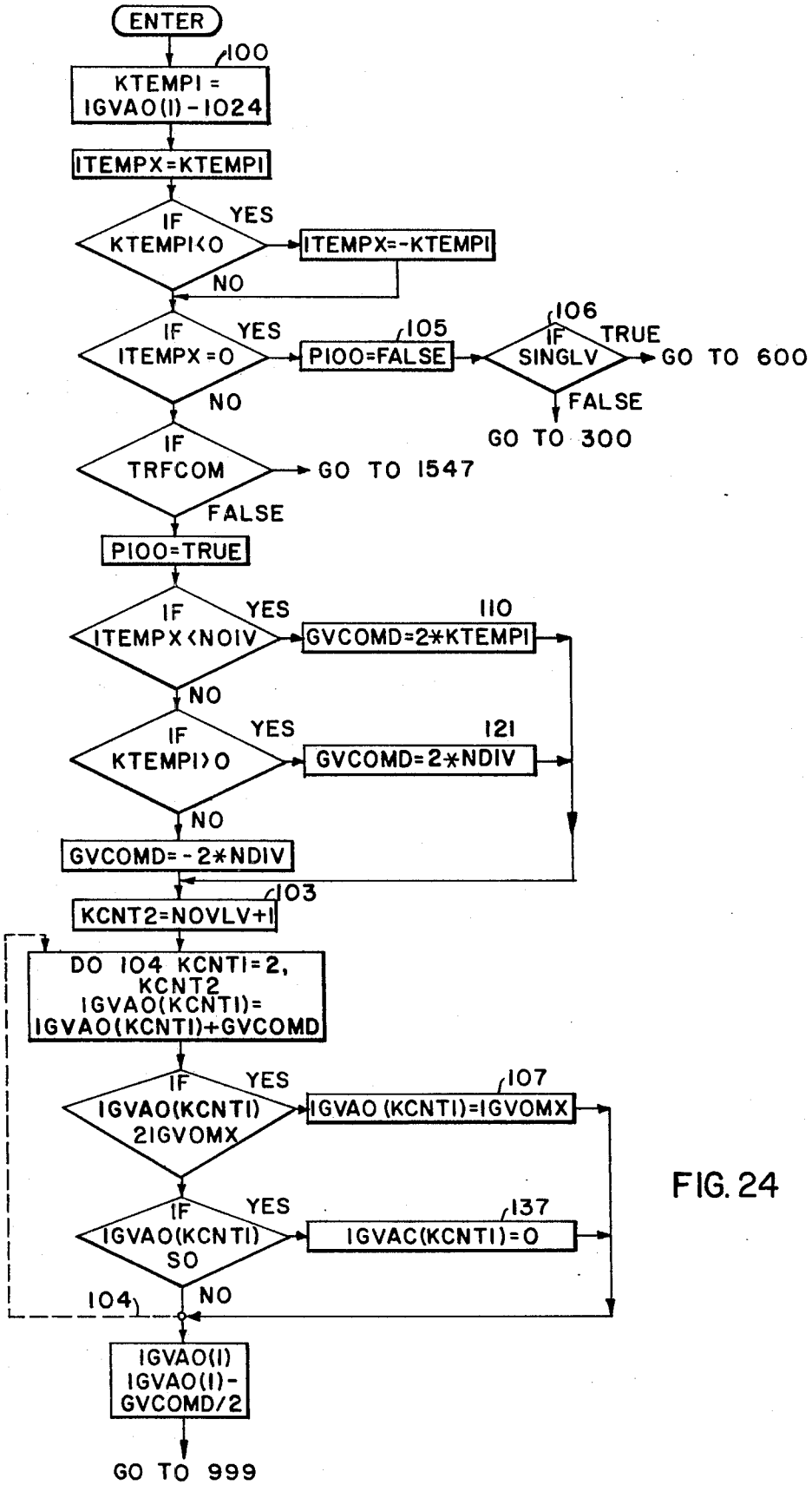

ENTER
100
KTEMPI = IGVAO(I) - 1024
ITEMPX=KTEMPI
IF KTEMPI<0
YES
ITEMPX=-KTEMPI
NO
IF ITEMPX=0
YES
105
PIOO=FALSE
106
IF SINGLV
TRUE
GO TO 600
FALSE
GO TO 300
NO
IF TRFCOM
GO TO 1547
FALSE
PIOO=TRUE
IF ITEMPX<NOIV
YES
110
GVCOMD=2*KTEMPI
NO
IF KTEMPI>0
YES
121
GVCOMD=2*NDIV
NO
GVCOMD=-2*NDIV
103
KCNT2=NOVLV+1
DO 104 KCNTI=2, KCNT2
IGVAO(KCNTI)= IGVAO(KCNTI)+GVCOMD
IF IGVAO(KCNTI) ≥IGVOMX
YES
107
IGVAO(KCNTI)=IGVOMX
IF IGVAO(KCNTI) ≤0
YES
137
IGVAC(KCNTI)=0
NO
104
IGVAO(I) IGVAO(I)- GVCOMD/2
GO TO 999

FIG. 24

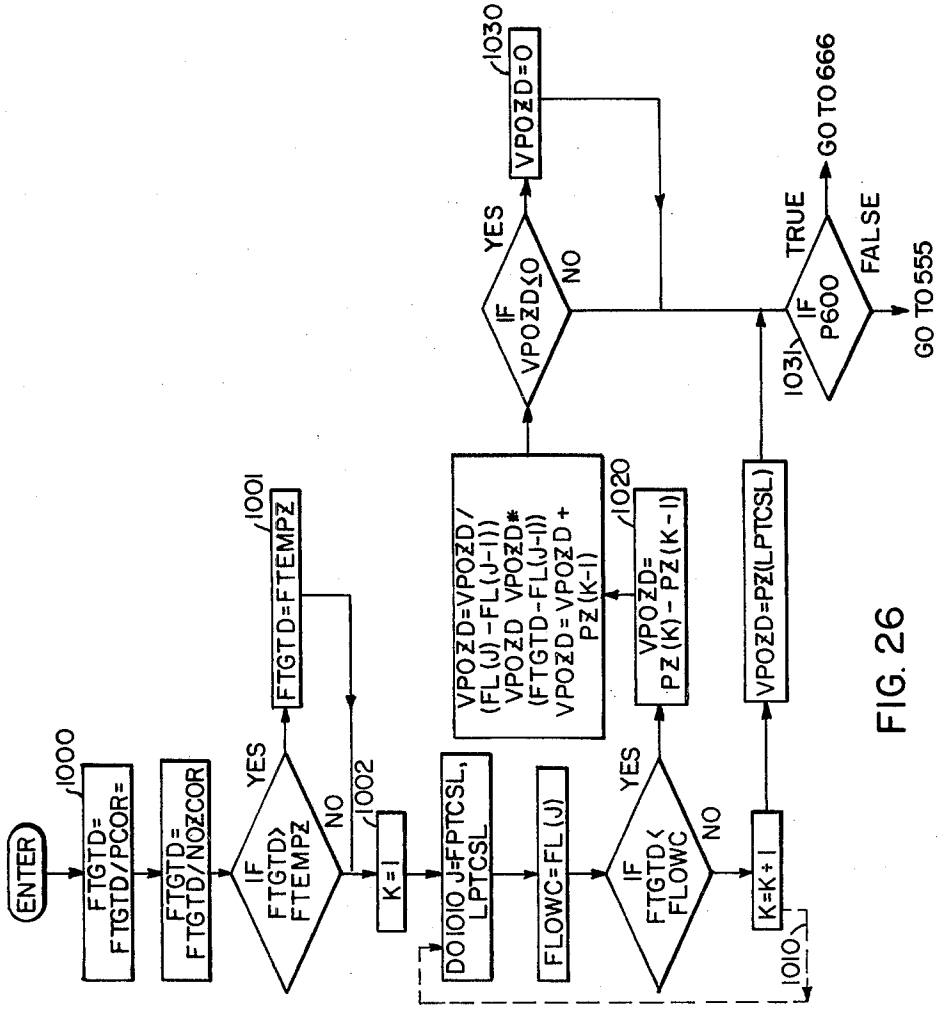

ENTER
1000
FTGTD= FTGTD/PCOR=
FTGTD= FTGTD/NOZCOR
IF FTGTD> FTEMPZ
YES
NO
1001
FTGTD=FTEMPZ
1002
K=1
DO 1010 J=FPTCSL, LPTCSL
FLOWC=FL(J)
IF FTGTD< FLOWC
YES
NO
VPOZD=VPOZD/ (FL(J)-FL(J-1)) VPOZD VPOZD* (FTGTD-FL(J-1)) VPOZD=VPOZD+ PZ(K-1)
1020
VPOZD= PZ(K)-PZ(K-1)
K=K+1
1010
VPOZD=PZ(LPTCSL)
IF VPOZD≤0
YES
NO
1030
VPOZD=0
IF P600
TRUE
FALSE
1031
GO TO 666
GO TO 555

FIG. 26

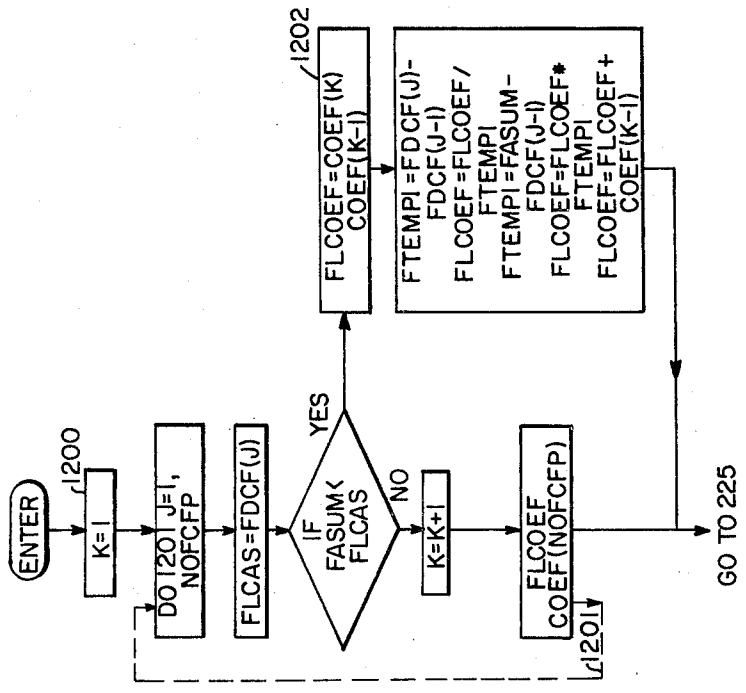

ENTER
1200
K=1
DO 1201 J=1, NOFCFP
FLCAS=FDCF(J)
IF FASUM< FLCAS
YES
NO
K=K+1
1201
FLCOEF COEF(NOFCFP)
1202
FLCOEF=COEF(K) COEF(K-1)
FTEMPI=FDCF(J)- FDCF(J-1)
FLCOEF=FLCOEF/ FTEMPI
FTEMPI=FASUM- FDCF(J-1)
FLCOEF=FLCOEF* FTEMPI
FLCOEF=FLCOEF+ COEF(K-1)
GO TO 225

FIG. 25

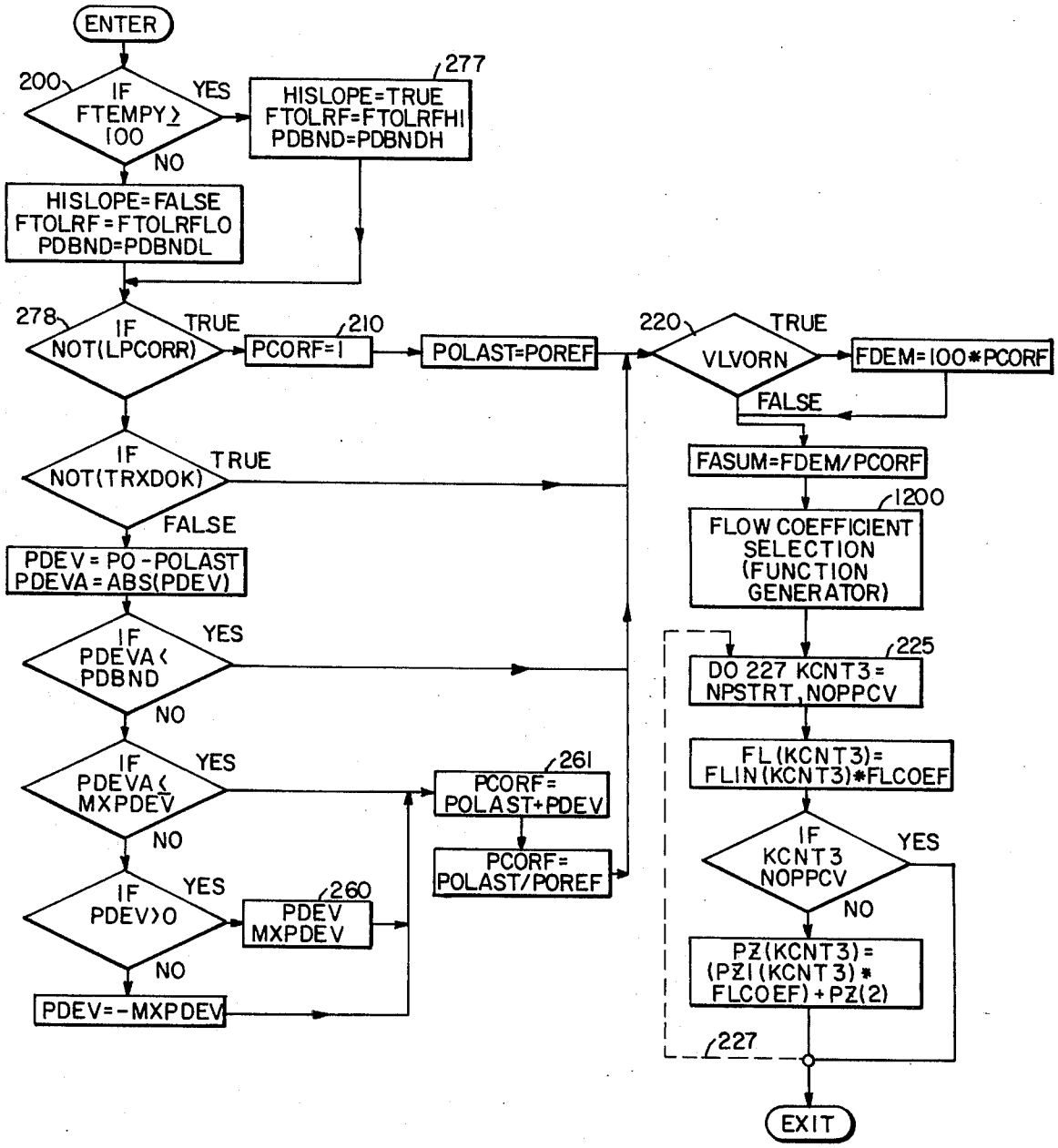

ENTER
200
IF FTEMPY ≥ 100
YES
NO
277
HISLOPE=TRUE
FTOLRF=FTOLRFHI
PDBND=PDBNDH
HISLOPE=FALSE
FTOLRF=FTOLRFLO
PDBND=PDBNDL
278
IF NOT(LPCORR)
TRUE
PCORF=1
210
POLAST=POREF
220
VLVORN
TRUE
FDEM=100*PCORF
FALSE
IF NOT(TRXDOK)
TRUE
FALSE
FASUM=FDEM/PCORF
1200
FLOW COEFFICIENT SELECTION (FUNCTION GENERATOR)
PDEV=PO-POLAST
PDEVA=ABS(PDEV)
IF PDEVA < PDBND
YES
NO
225
DO 227 KCNT3= NPSTRT,NOPPCV
IF PDEVA < MXPDEV
YES
NO
261
PCORF= POLAST+PDEV
PCORF= POLAST/POREF
FL(KCNT3)= FLIN(KCNT3)*FLCOEF
IF KCNT3 NOPPCV
YES
NO
IF PDEV>0
YES
NO
260
PDEV MXPDEV
PDEV=-MXPDEV
PZ(KCNT3)= (PZI(KCNT3)* FLCOEF)+PZ(2)
227
EXIT

FIG. 27

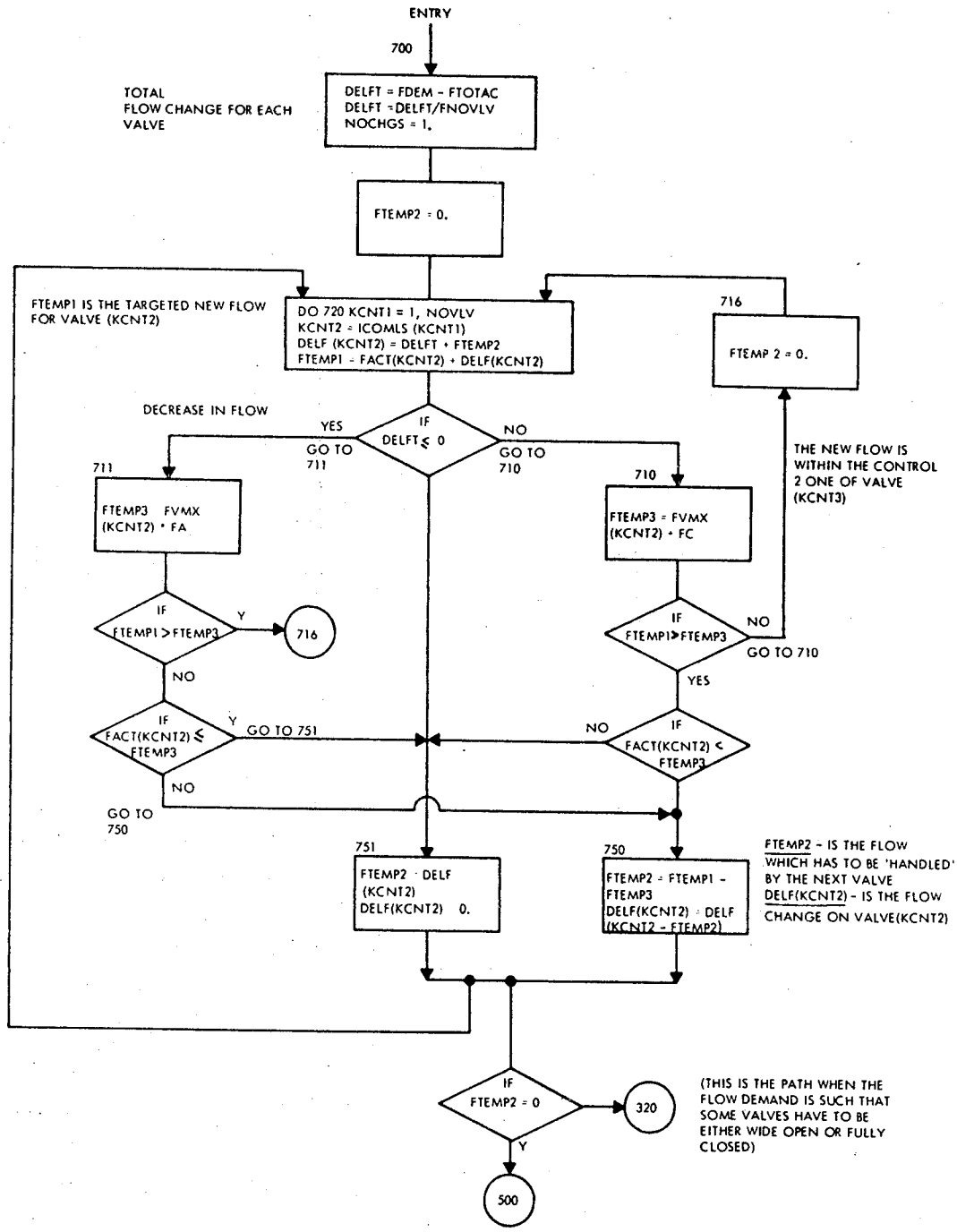

ENTRY
700
TOTAL FLOW CHANGE FOR EACH VALVE
DELFT = FDEM - FTOTAC
DELFT = DELFT/FNOVLV
NOCHGS = 1.
FTEMP2 = 0.
FTEMP1 IS THE TARGETED NEW FLOW FOR VALVE (KCNT2)
DO 720 KCNT1 = 1, NOVLV
KCNT2 = ICOMLS (KCNT1)
DELF (KCNT2) = DELFT + FTEMP2
FTEMP1 = FACT(KCNT2) + DELF(KCNT2)
716
FTEMP 2 = 0.
DECREASE IN FLOW
IF DELFT ≤ 0
YES
GO TO 711
NO
GO TO 710
711
FTEMP3 = FVMX (KCNT2) * FA
710
FTEMP3 = FVMX (KCNT2) * FC
THE NEW FLOW IS WITHIN THE CONTROL 2 ONE OF VALVE (KCNT3)
IF FTEMP1 > FTEMP3
Y
716
NO
IF FTEMP1 ≥ FTEMP3
NO
GO TO 710
YES
IF FACT(KCNT2) ≤ FTEMP3
Y
GO TO 751
NO
GO TO 750
IF FACT(KCNT2) < FTEMP3
NO
751
FTEMP2 = DELF (KCNT2)
DELF(KCNT2) = 0.
750
FTEMP2 = FTEMP1 - FTEMP3
DELF(KCNT2) = DELF (KCNT2 - FTEMP2)
FTEMP2 - IS THE FLOW WHICH HAS TO BE 'HANDLED' BY THE NEXT VALVE
DELF(KCNT2) - IS THE FLOW CHANGE ON VALVE(KCNT2)
IF FTEMP2 = 0
320
Y
(THIS IS THE PATH WHEN THE FLOW DEMAND IS SUCH THAT SOME VALVES HAVE TO BE EITHER WIDE OPEN OR FULLY CLOSED)
500

FIG. 28

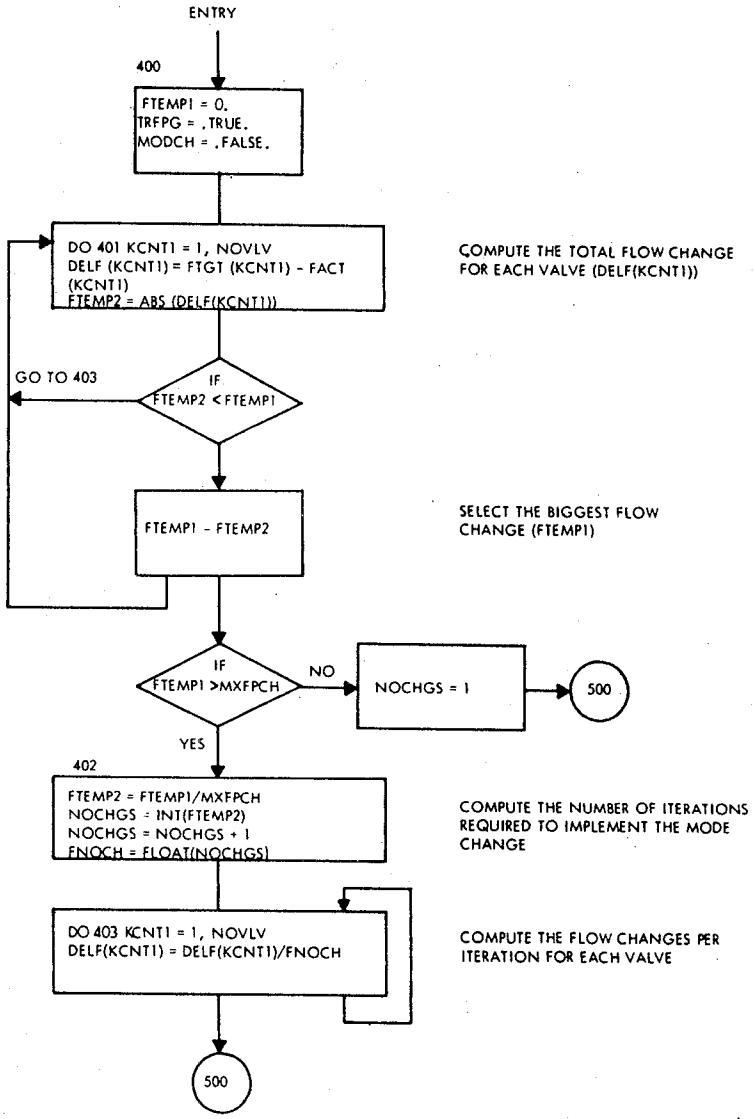

ENTRY
400
FTEMP1 = 0.
TRFPG = .TRUE.
MODCH = .FALSE.
DO 401 KCNT1 = 1, NOVLV
DELF (KCNT1) = FTGT (KCNT1) - FACT (KCNT1)
FTEMP2 = ABS (DELF(KCNT1))
COMPUTE THE TOTAL FLOW CHANGE FOR EACH VALVE (DELF(KCNT1))
GO TO 403
IF FTEMP2 < FTEMP1
FTEMP1 = FTEMP2
SELECT THE BIGGEST FLOW CHANGE (FTEMP1)
IF FTEMP1 >MXFPCH
NO
NOCHGS = 1
500
YES
402
FTEMP2 = FTEMP1/MXFPCH
NOCHGS = INT(FTEMP2)
NOCHGS = NOCHGS + 1
FNOCH = FLOAT(NOCHGS)
COMPUTE THE NUMBER OF ITERATIONS REQUIRED TO IMPLEMENT THE MODE CHANGE
DO 403 KCNT1 = 1, NOVLV
DELF(KCNT1) = DELF(KCNT1)/FNOCH
COMPUTE THE FLOW CHANGES PER ITERATION FOR EACH VALVE
500

FIG. 29

FIG. 30

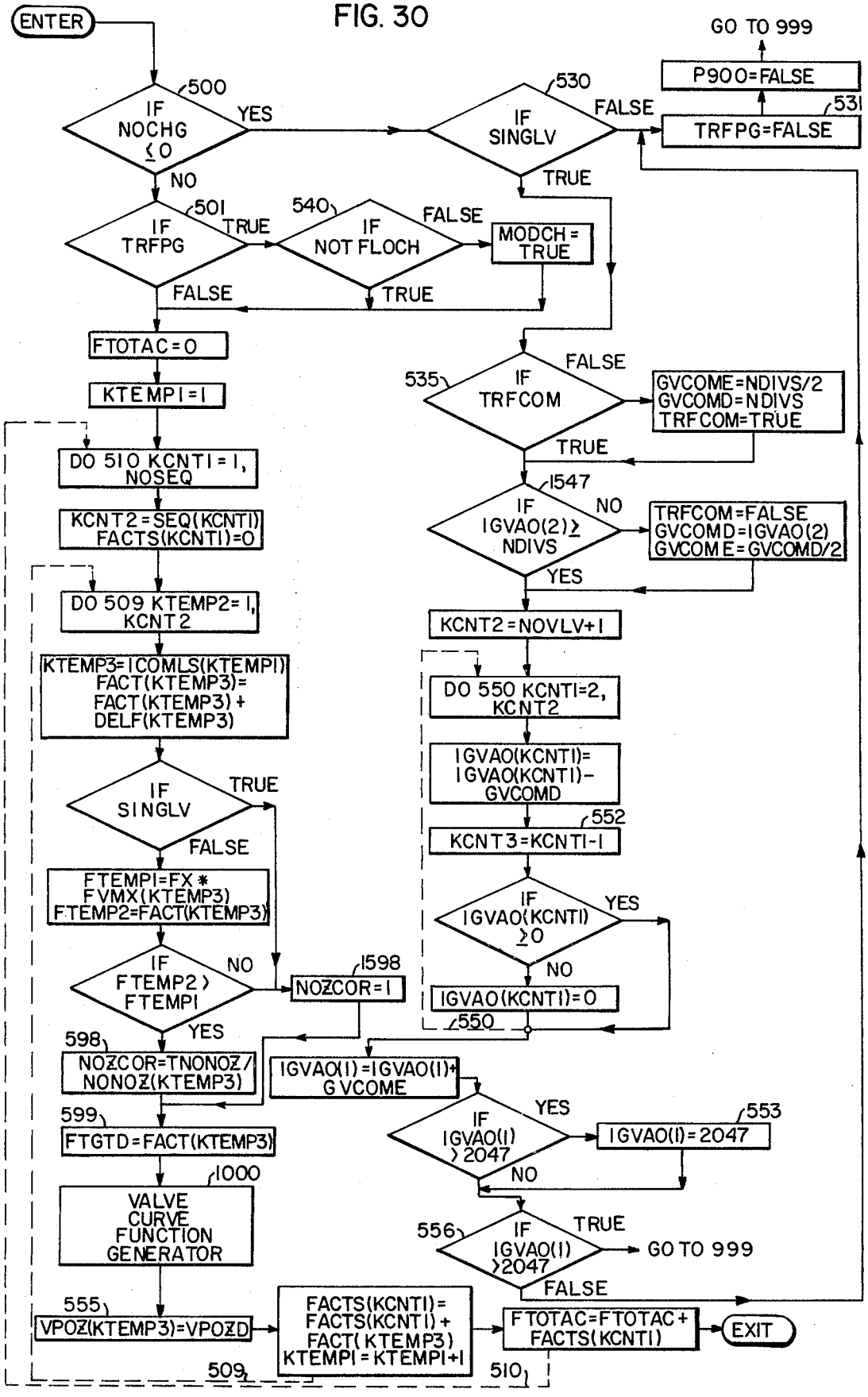
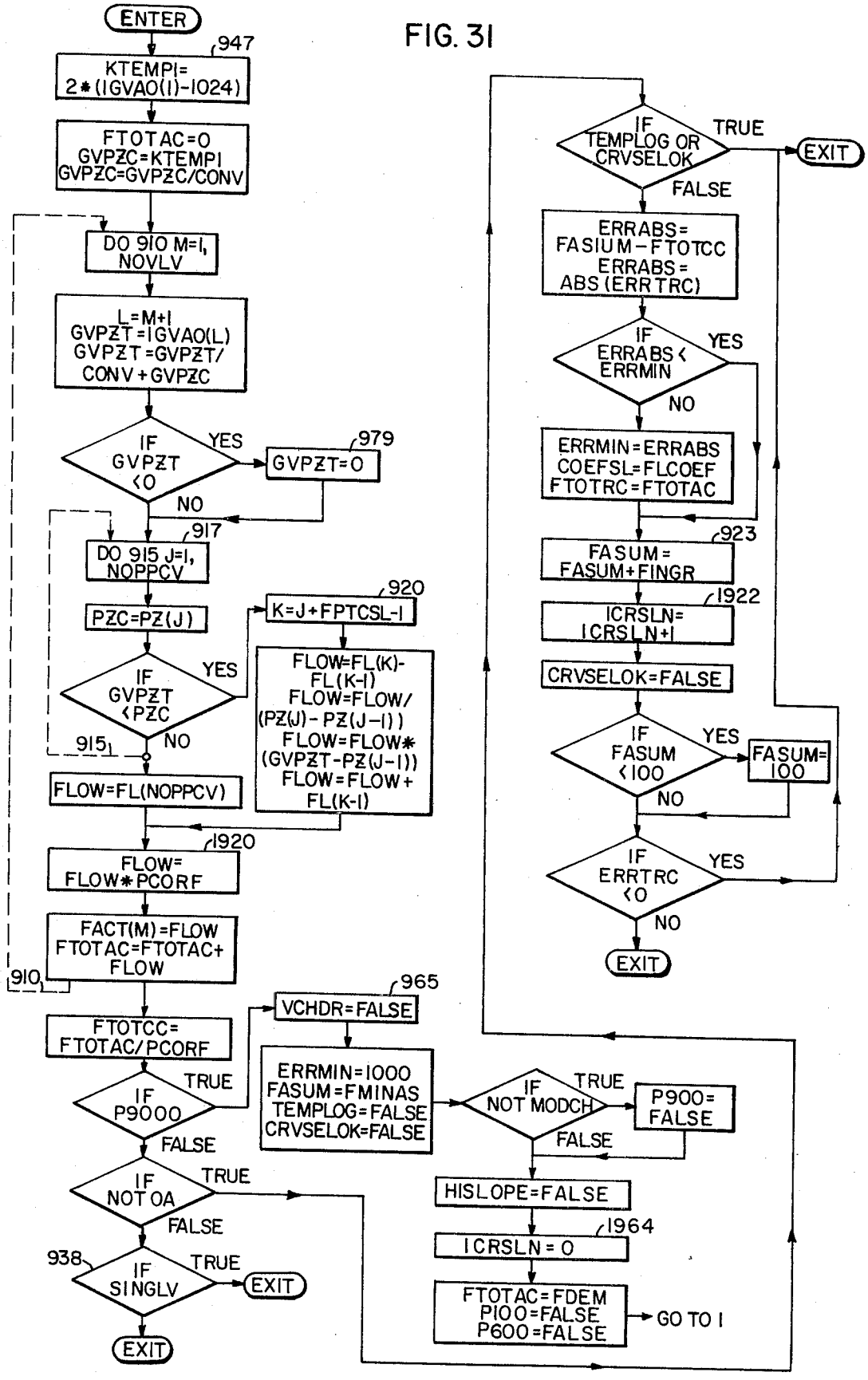

ENTER
500
IF NOCHG ≤ 0
YES
NO
530
IF SINGLV
FALSE
TRUE
GO TO 999
P900=FALSE
531
TRFPG=FALSE
501
IF TRFPG
TRUE
FALSE
540
IF NOT FLOCH
FALSE
TRUE
MODCH = TRUE
FTOTAC=0
KTEMPI=1
535
IF TRFCOM
FALSE
TRUE
GVCOME=NDIVS/2
GVCOMD=NDIVS
TRFCOM=TRUE
DO 510 KCNTI=1, NOSEQ
1547
IF IGVAO(2) ≥ NDIVS
NO
YES
TRFCOM=FALSE
GVCOMD=IGVAO(2)
GVCOME=GVCOMD/2
KCNT2=SEQ(KCNTI)
FACTS(KCNTI)=0
DO 509 KTEMP2=1, KCNT2
KCNT2=NOVLV+1
KTEMP3=ICOMLS(KTEMPI)
FACT(KTEMP3)=
FACT(KTEMP3) +
DELF(KTEMP3)
DO 550 KCNTI=2, KCNT2
IF SINGLV
TRUE
FALSE
IGVAO(KCNTI)=
IGVAO(KCNTI)-
GVCOMD
552
KCNT3=KCNTI-1
FTEMPI=FX *
FVMX(KTEMP3)
FTEMP2=FACT(KTEMP3)
IF IGVAO(KCNTI) ≥ 0
YES
NO
IF FTEMP2 > FTEMPI
NO
YES
1598
NOZCOR=1
IGVAO(KCNTI)=0
550
598
NOZCOR=TNONOZ/
NONOZ(KTEMP3)
IGVAO(I)=IGVAO(I)+
GVCOME
599
FTGTD=FACT(KTEMP3)
IF IGVAO(I) > 2047
YES
NO
553
IGVAO(I)=2047
1000
VALVE CURVE FUNCTION GENERATOR
556
IF IGVAO(I) > 2047
TRUE
FALSE
GO TO 999
555
VPOZ(KTEMP3)=VPOZD
FACTS(KCNTI)=
FACTS(KCNTI) +
FACT(KTEMP3)
KTEMPI=KTEMPI+1
FTOTAC=FTOTAC+
FACTS(KCNTI)
EXIT
509
510

FIG. 31


ENTER
947
KTEMPI=
2*(IGVAO(I)-1024)
FTOTAC=0
GVPZC=KTEMPI
GVPZC=GVPZC/CONV
DO 910 M=1,
NOVLV
L=M+1
GVPZT=IGVAO(L)
GVPZT=GVPZT/
CONV+GVPZC
IF
GVPZT
<0
YES
979
GVPZT=0
NO
917
DO 915 J=1,
NOPPCV
PZC=PZ(J)
920
K=J+FPTCSL-1
IF
GVPZT
<PZC
YES
NO
FLOW=FL(K)-
FL(K-1)
FLOW=FLOW/
(PZ(J)-PZ(J-1))
FLOW=FLOW*
(GVPZT-PZ(J-1))
FLOW=FLOW+
FL(K-1)
915
FLOW=FL(NOPPCV)
1920
FLOW=
FLOW*PCORF
FACT(M)=FLOW
FTOTAC=FTOTAC+
FLOW
910
FTOTCC=
FTOTAC/PCORF
IF
P9000
TRUE
FALSE
965
VCHDR=FALSE
ERRMIN=1000
FASUM=FMINAS
TEMPLOG=FALSE
CRVSELOK=FALSE
IF
NOT MODCH
TRUE
FALSE
P900=
FALSE
IF
NOT OA
TRUE
FALSE
HISLOPE=FALSE
1964
ICRSLN=0
938
IF
SINGLV
TRUE
EXIT
FTOTAC=FDEM
PI00=FALSE
P600=FALSE
GO TO 1
EXIT
IF
TEMPLOG OR
CRVSELOK
TRUE
EXIT
FALSE
ERRABS=
FASIUM-FTOTCC
ERRABS=
ABS(ERRTRC)
IF
ERRABS<
ERRMIN
YES
NO
ERRMIN=ERRABS
COEFSL=FLCOEF
FTOTRC=FTOTAC
923
FASUM=
FASUM+FINGR
1922
ICRSLN=
ICRSLN+1
CRVSELOK=FALSE
IF
FASUM
<100
YES
FASUM=
100
NO
IF
ERRTRC
<0
YES
NO
EXIT

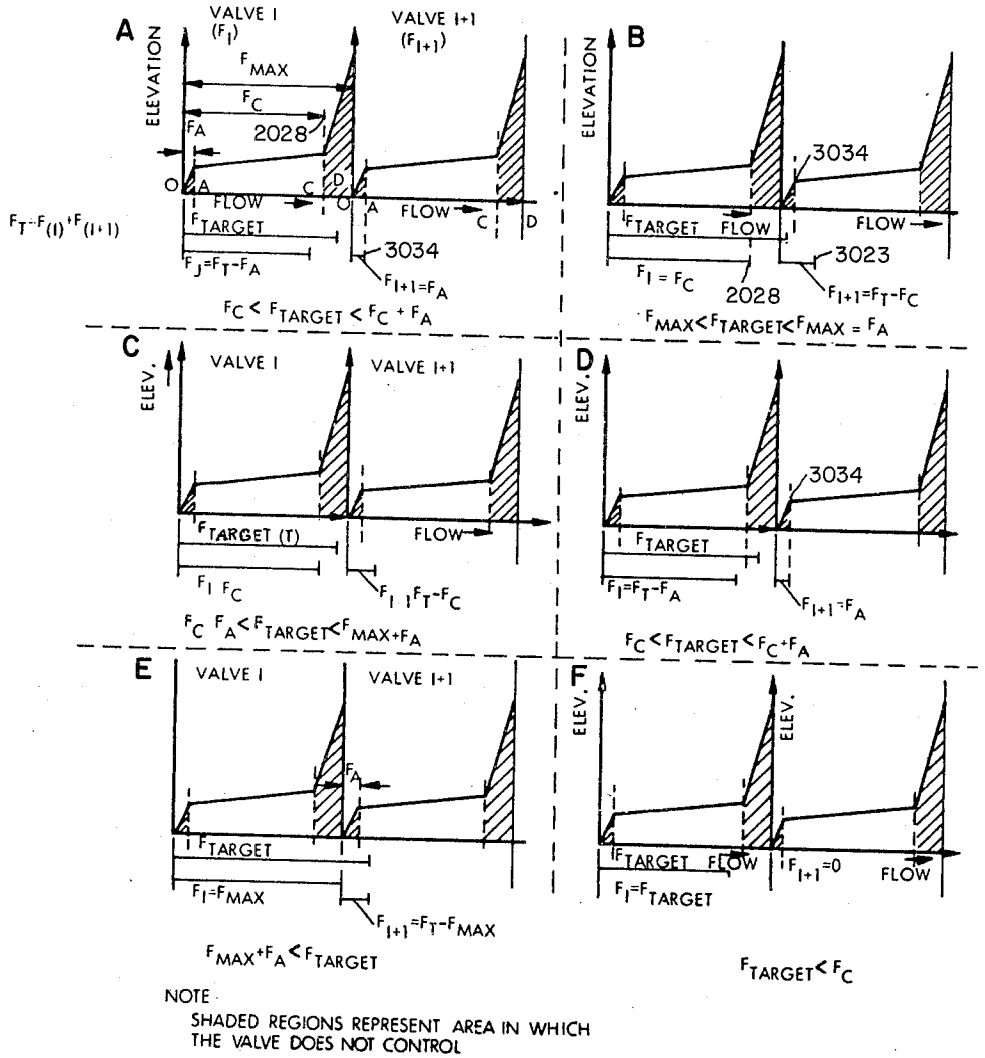

A
VALVE I
($F_I$)
VALVE I+1
($F_{I+1}$)
ELEVATION
$F_{MAX}$
$F_C$
$F_A$
2028
O
A
C
D
FLOW
$F_T=F_{(I)}+F_{(I+1)}$
$F_{TARGET}$
$F_J=F_T-F_A$
3034
$F_{I+1}=F_A$
$F_C < F_{TARGET} < F_C + F_A$
B
ELEVATION
3034
$F_{TARGET}$
FLOW
$F_I = F_C$
2028
3023
$F_{I+1}=F_T-F_C$
$F_{MAX}<F_{TARGET}<F_{MAX}=F_A$
C
ELEV.
VALVE I
VALVE I+1
$F_{TARGET}$ (T)
FLOW
$F_I$ $F_C$
$F_{I+1}$ $F_T-F_C$
$F_C$ $F_A$ $< F_{TARGET} < F_{MAX}+F_A$
D
ELEV.
3034
$F_{TARGET}$
$F_I=F_T-F_A$
$F_{I+1}=F_A$
$F_C < F_{TARGET} < F_C+F_A$
E
VALVE I
VALVE I+1
$F_A$
$F_{TARGET}$
$F_I=F_{MAX}$
$F_{I+1}=F_T-F_{MAX}$
$F_{MAX}+F_A < F_{TARGET}$
F
ELEV.
ELEV.
$F_{TARGET}$
FLOW
$F_{I+1}=0$
FLOW
$F_I=F_{TARGET}$
$F_{TARGET} < F_C$
NOTE
SHADED REGIONS REPRESENT AREA IN WHICH THE VALVE DOES NOT CONTROL

FIG. 32

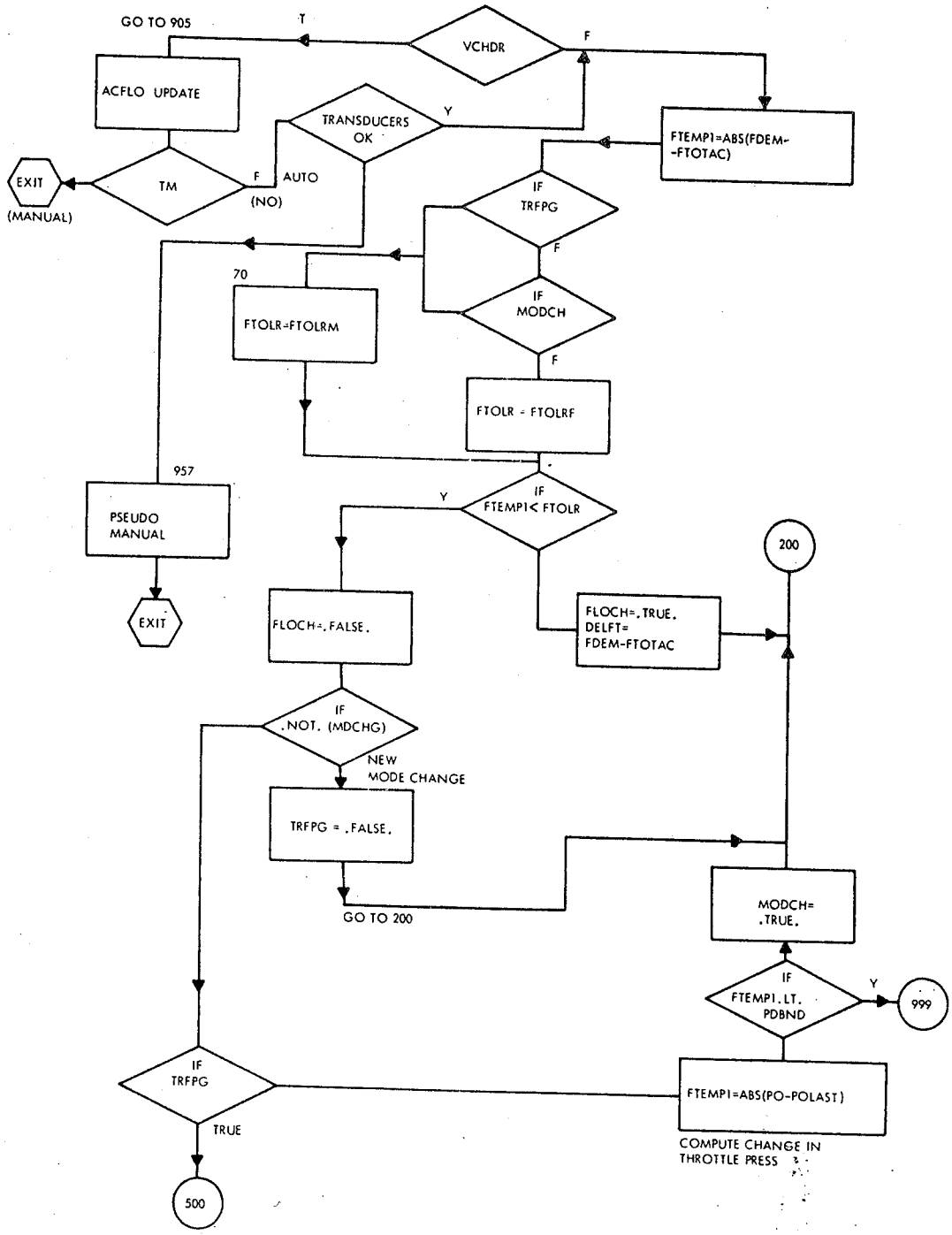

GO TO 905
T
VCHDR
F
ACFLO UPDATE
TRANSDUCERS OK
Y
FTEMP1=ABS(FDEM--FTOTAC)
EXIT
(MANUAL)
TM
F
AUTO
(NO)
IF TRFPG
F
70
IF MODCH
FTOLR=FTOLRM
F
FTOLR = FTOLRF
957
PSEUDO MANUAL
Y
IF FTEMP1< FTOLR
200
EXIT
FLOCH=.FALSE.
FLOCH=.TRUE. DELFT= FDEM-FTOTAC
IF .NOT. (MDCHG)
NEW MODE CHANGE
TRFPG = .FALSE.
GO TO 200
MODCH= .TRUE.
IF FTEMP1.LT. PDBND
Y
999
IF TRFPG
FTEMP1=ABS(PO-POLAST)
TRUE
COMPUTE CHANGE IN THROTTLE PRESS
500

FIG. 33

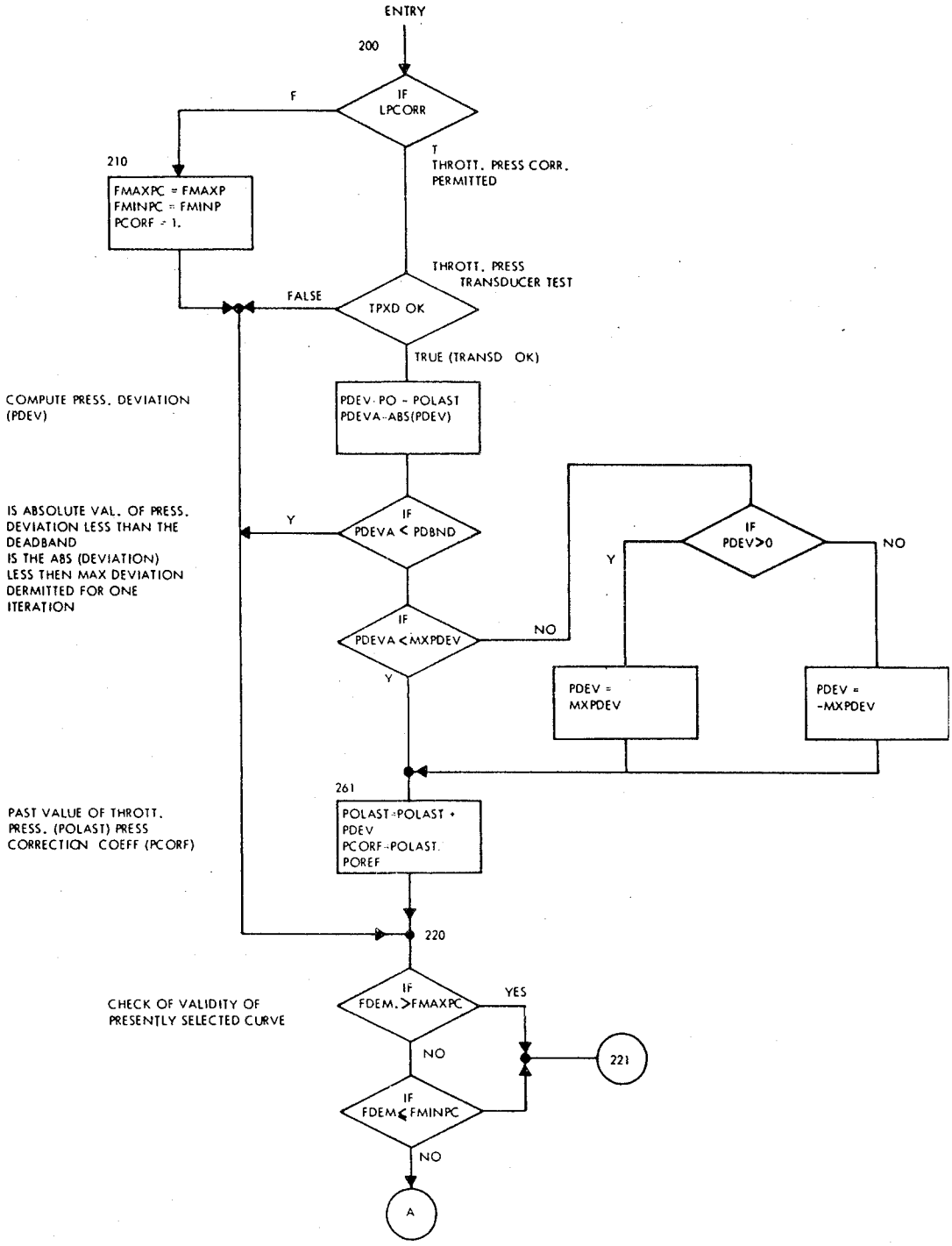

ENTRY
200
IF LPCORR
F
T
THROTT. PRESS CORR. PERMITTED
210
FMAXPC = FMAXP
FMINPC = FMINP
PCORF = 1.
THROTT. PRESS TRANSDUCER TEST
FALSE
TPXD OK
TRUE (TRANSD OK)
COMPUTE PRESS. DEVIATION (PDEV)
PDEV=PO - POLAST
PDEVA=ABS(PDEV)
IS ABSOLUTE VAL. OF PRESS. DEVIATION LESS THAN THE DEADBAND
IS THE ABS (DEVIATION) LESS THEN MAX DEVIATION DERMITTED FOR ONE ITERATION
IF PDEVA < PDBND
Y
IF PDEV>0
Y
NO
IF PDEVA <MXPDEV
NO
Y
PDEV = MXPDEV
PDEV = -MXPDEV
261
PAST VALUE OF THROTT. PRESS. (POLAST) PRESS CORRECTION COEFF (PCORF)
POLAST=POLAST + PDEV
PCORF=POLAST. POREF
220
CHECK OF VALIDITY OF PRESENTLY SELECTED CURVE
IF FDEM.>FMAXPC
YES
NO
221
IF FDEM<FMINPC
NO
A

FIG. 34

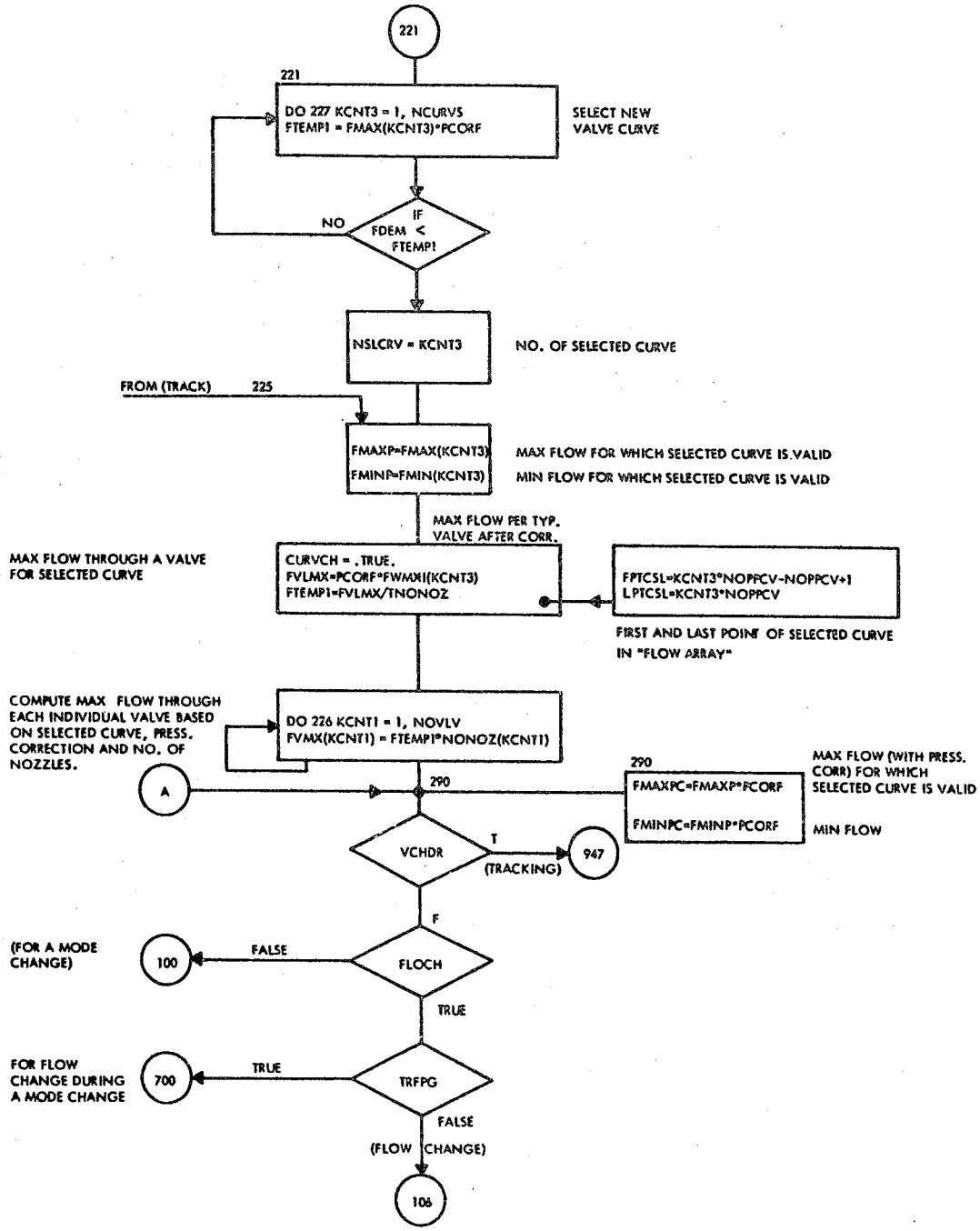

221
221
DO 227 KCNT3 = 1, NCURVS
FTEMP1 = FMAX(KCNT3)*PCORF
SELECT NEW VALVE CURVE
NO
IF FDEM < FTEMP1
NSLCRV = KCNT3
NO. OF SELECTED CURVE
FROM (TRACK) 225
FMAXP=FMAX(KCNT3)
FMINP=FMIN(KCNT3)
MAX FLOW FOR WHICH SELECTED CURVE IS VALID
MIN FLOW FOR WHICH SELECTED CURVE IS VALID
MAX FLOW PER TYP. VALVE AFTER CORR.
MAX FLOW THROUGH A VALVE FOR SELECTED CURVE
CURVCH = .TRUE.
FVLMX=PCORF*FWMXI(KCNT3)
FTEMP1=FVLMX/TNONOZ
FPTCSL=KCNT3*NOPPCV-NOPPCV+1
LPTCSL=KCNT3*NOPPCV
FIRST AND LAST POINT OF SELECTED CURVE IN "FLOW ARRAY"
COMPUTE MAX FLOW THROUGH EACH INDIVIDUAL VALVE BASED ON SELECTED CURVE, PRESS. CORRECTION AND NO. OF NOZZLES.
DO 226 KCNT1 = 1, NOVLV
FVMX(KCNT1) = FTEMP1*NONOZ(KCNT1)
A
290
290
FMAXPC=FMAXP*FCORF
FMINPC=FMINP*PCORF
MAX FLOW (WITH PRESS. CORR) FOR WHICH SELECTED CURVE IS VALID
MIN FLOW
VCHDR
T
(TRACKING)
947
F
(FOR A MODE CHANGE)
100
FALSE
FLOCH
TRUE
FOR FLOW CHANGE DURING A MODE CHANGE
700
TRUE
TRFPG
FALSE
(FLOW CHANGE)
106

FIG. 35

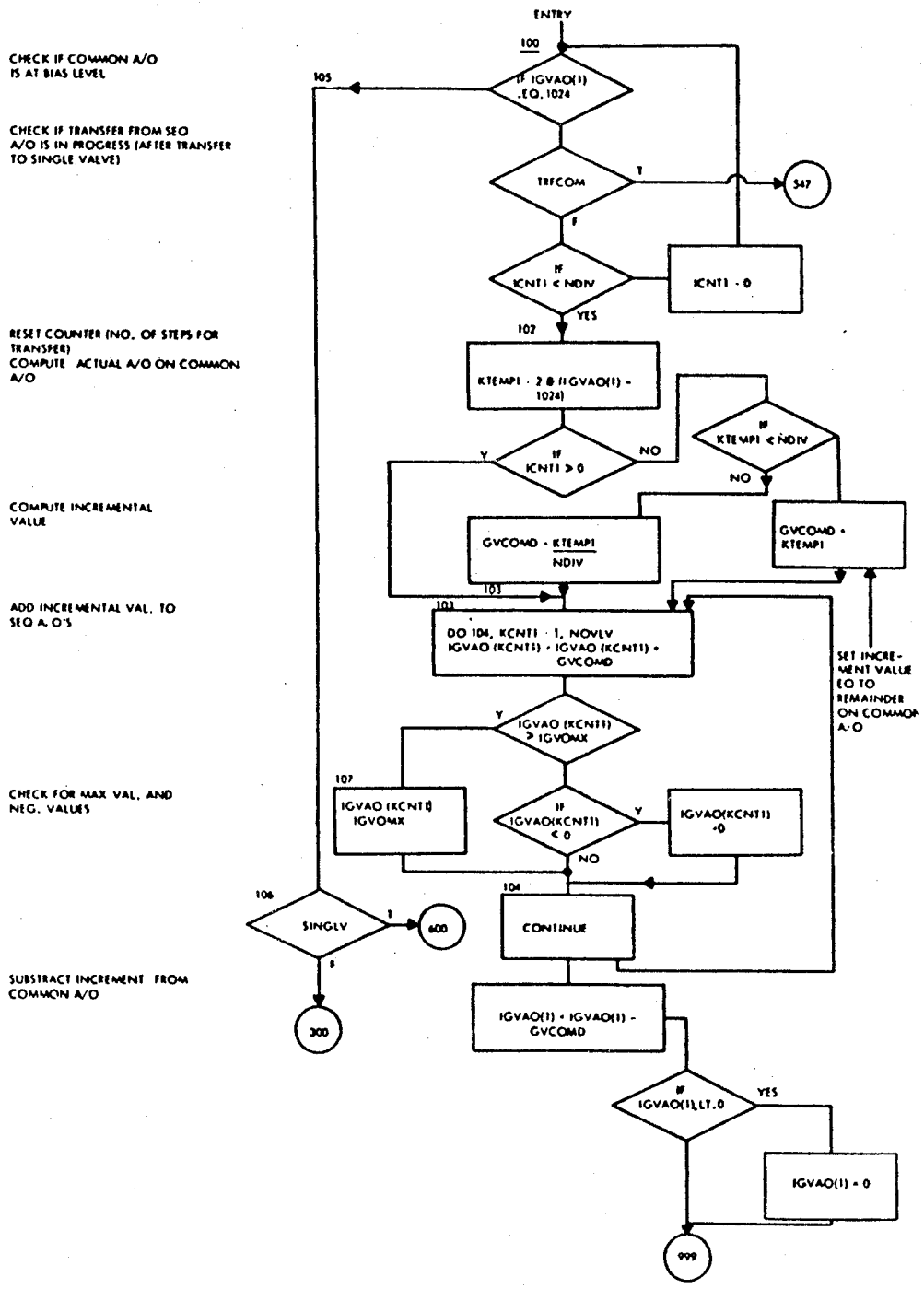

CHECK IF COMMON A/O IS AT BIAS LEVEL
CHECK IF TRANSFER FROM SEQ A/O IS IN PROGRESS (AFTER TRANSFER TO SINGLE VALVE)
RESET COUNTER (NO. OF STEPS FOR TRANSFER)
COMPUTE ACTUAL A/O ON COMMON A/O
COMPUTE INCREMENTAL VALUE
ADD INCREMENTAL VAL. TO SEQ A.O'S
CHECK FOR MAX VAL. AND NEG. VALUES
SUBSTRACT INCREMENT FROM COMMON A/O
ENTRY
100
105
IF IGVAO(1) .EQ. 1024
TRFCOM
T
547
F
IF ICNT1 < NDIV
ICNT1 = 0
YES
102
KTEMP1 = 2.0 (IGVAO(1) - 1024)
IF KTEMP1 < NDIV
IF ICNT1 > 0
Y
NO
NO
GVCOMD = KTEMP1 / NDIV
GVCOMD = KTEMP1
103
DO 104, KCNT1 = 1, NOVLV
IGVAO (KCNT1) = IGVAO (KCNT1) + GVCOMD
SET INCREMENT VALUE EQ TO REMAINDER ON COMMON A.O
Y
IGVAO (KCNT1) > IGVOMX
107
IGVAO (KCNT1) = IGVOMX
IF IGVAO(KCNT1) < 0
Y
IGVAO(KCNT1) = 0
NO
104
CONTINUE
106
SINGLV
T
600
F
300
IGVAO(1) = IGVAO(1) - GVCOMD
IF IGVAO(1) LT. 0
YES
IGVAO(1) = 0
999

FIG. 36

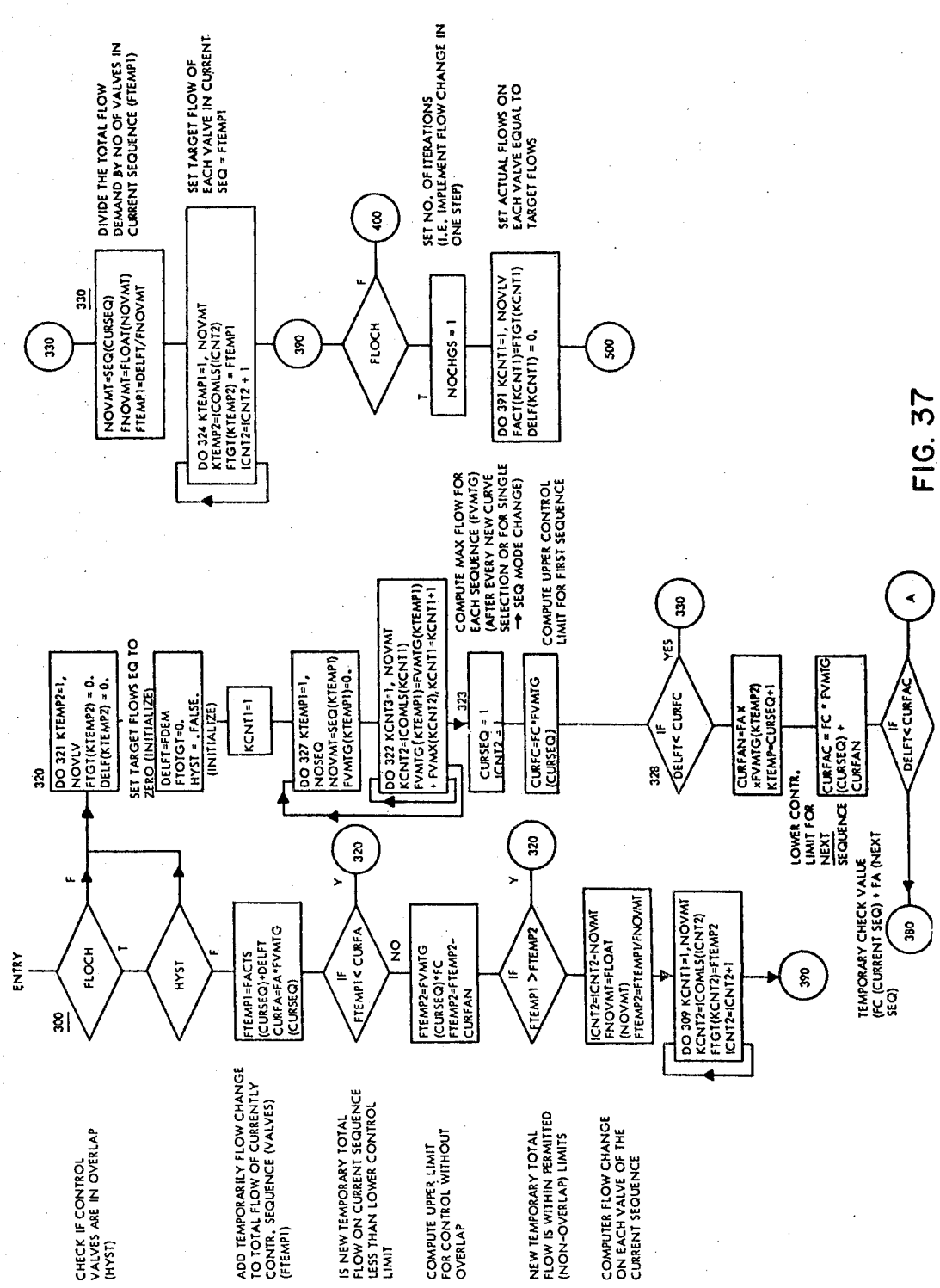

ENTRY
300
FLOCH
T
F
HYST
F
T
CHECK IF CONTROL VALVES ARE IN OVERLAP (HYST)
320
DO 321 KTEMP2=1, NOVLV
FTGT(KTEMP2) = 0.
DELF(KTEMP2) = 0.
SET TARGET FLOWS EQ TO ZERO (INITIALIZE)
DELFT=FDEM
FTOTGT=0.
HYST = .FALSE.
(INITIALIZE)
KCNT1=1
FTEMP1=FACTS (CURSEQ)+DELFT
CURFA=FA*FVMTG (CURSEQ)
ADD TEMPORARILY FLOW CHANGE TO TOTAL FLOW OF CURRENTLY CONTR. SEQUENCE (VALVES) (FTEMP1)
IF
FTEMP1< CURFA
Y
NO
320
IS NEW TEMPORARY TOTAL FLOW ON CURRENT SEQUENCE LESS THAN LOWER CONTROL LIMIT
FTEMP2=FVMTG (CURSEQ)*FC
FTEMP2=FTEMP2-CURFAN
COMPUTE UPPER LIMIT FOR CONTROL WITHOUT OVERLAP
IF
FTEMP1 > FTEMP2
Y
320
NEW TEMPORARY TOTAL FLOW IS WITHIN PERMITTED (NON-OVERLAP) LIMITS
ICNT2=ICNT2-NOVMT
FNOVMT=FLOAT (NOVMT)
FTEMP2=FTEMP1/FNOVMT
COMPUTER FLOW CHANGE ON EACH VALVE OF THE CURRENT SEQUENCE
DO 309 KCNT1=1,NOVMT
KCNT2=ICOML5(ICNT2)
FTGT(KCNT2)=FTEMP2
ICNT2=ICNT2+1
390
DO 327 KTEMP1=1, NOSEQ
NOVMT=SEQ(KTEMP1)
FVMTG(KTEMP1)=0.
DO 322 KCNT3=1, NOVMT
KCNT2=ICOML5(KCNT1)
FVMTG(KTEMP1)=FVMTG(KTEMP1)
+ FVMAX(KCNT2),KCNT1=KCNT1+1
323
COMPUTE MAX FLOW FOR EACH SEQUENCE (FVMTG) (AFTER EVERY NEW CURVE SELECTION OR FOR SINGLE → SEQ MODE CHANGE)
CURSEQ = 1
ICNT2 = 1
CURFC=FC*FVMTG (CURSEQ)
COMPUTE UPPER CONTROL LIMIT FOR FIRST SEQUENCE
328
IF
DELFT< CURFC
YES
330
CURFAN=FA* FVMTG(KTEMP2) KTEMP=CURSEQ+1
LOWER CONTR. LIMIT FOR NEXT SEQUENCE
CURFAC = FC * FVMTG (CURSEQ) + CURFAN
IF
DELFT< CURFAC
A
380
TEMPORARY CHECK VALUE (FC (CURRENT SEQ) + FA (NEXT SEQ)
330
NOVMT=SEQ(CURSEQ)
FNOVMT=FLOAT(NOVMT)
FTEMP1=DELFT/FNOVMT
DIVIDE THE TOTAL FLOW DEMAND BY NO OF VALVES IN CURRENT SEQUENCE (FTEMP1)
DO 324 KTEMP1=1, NOVMT
KTEMP2=ICOML5(ICNT2)
FTGT(KTEMP2) = FTEMP1
ICNT2=ICNT2 + 1
SET TARGET FLOW OF EACH VALVE IN CURRENT SEQ = FTEMP1
390
FLOCH
F
T
400
NOCHGS = 1
SET NO. OF ITERATIONS (I.E. IMPLEMENT FLOW CHANGE IN ONE STEP)
DO 391 KCNT1=1, NOVLV
FACT(KCNT1)=FTGT(KCNT1)
DELF(KCNT1) = 0.
SET ACTUAL FLOWS ON EACH VALVE EQUAL TO TARGET FLOWS
500

FIG. 37

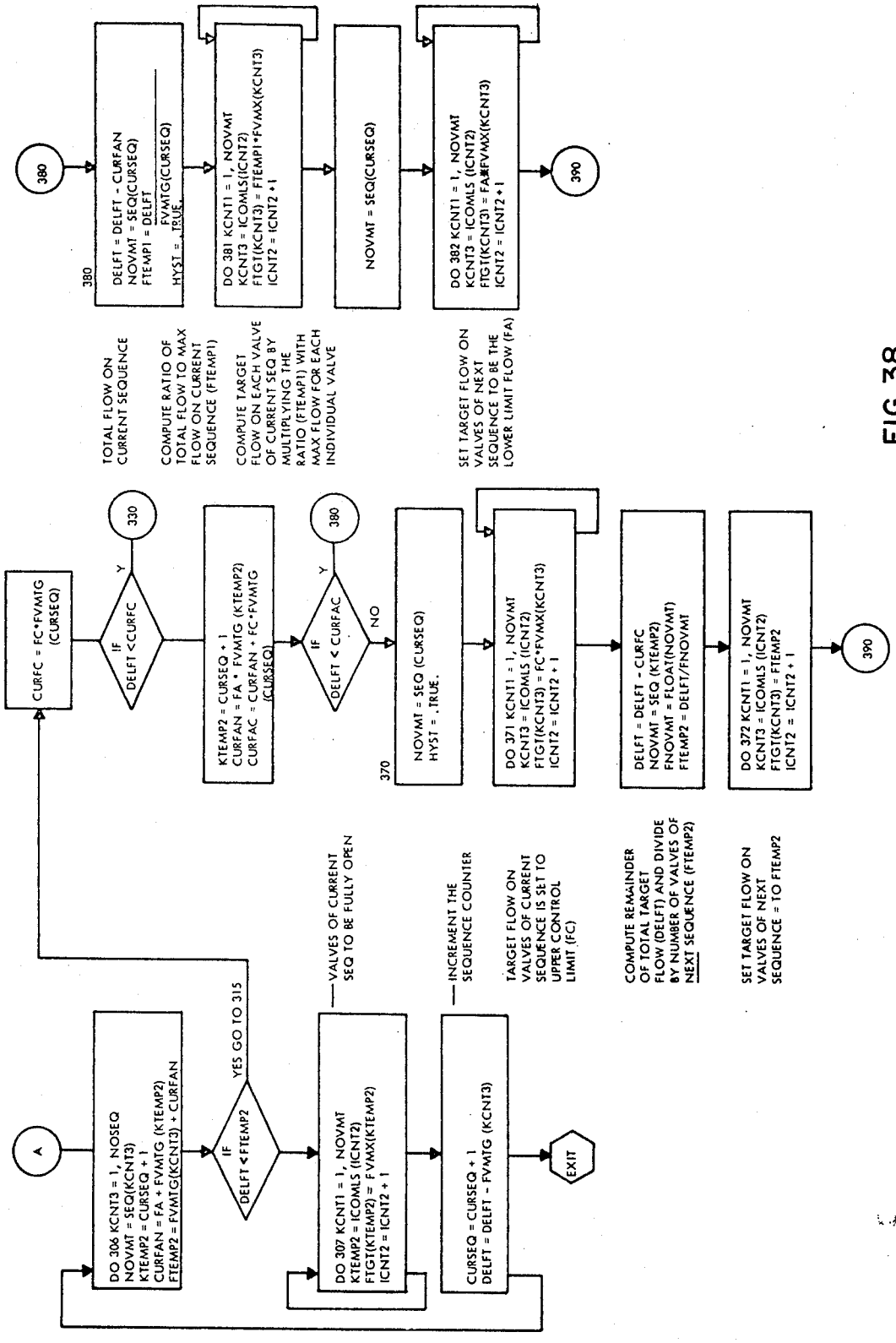

A
DO 306 KCNT3 = 1, NOSEQ
NOVMT = SEQ(KCNT3)
KTEMP2 = CURSEQ + 1
CURFAN = FA * FVMTG (KTEMP2)
FTEMP2 = FVMTG(KCNT3) + CURFAN
IF DELFT < FTEMP2
YES GO TO 315
DO 307 KCNT1 = 1, NOVMT
KTEMP2 = ICOMLS (ICNT2)
FTGT(KTEMP2) = FVMAX(KTEMP2)
ICNT2 = ICNT2 + 1
VALVES OF CURRENT SEQ TO BE FULLY OPEN
CURSEQ = CURSEQ + 1
DELFT = DELFT - FVMTG (KCNT3)
INCREMENT THE SEQUENCE COUNTER
TARGET FLOW ON VALVES OF CURRENT SEQUENCE IS SET TO UPPER CONTROL LIMIT (FC)
EXIT
CURFC = FC*FVMTG (CURSEQ)
IF DELFT < CURFC
Y
330
KTEMP2 = CURSEQ + 1
CURFAN = FA * FVMTG (KTEMP2)
CURFAC = CURFAN + FC*FVMTG (CURSEQ)
IF DELFT < CURFAC
Y
380
NO
370
NOVMT = SEQ (CURSEQ)
HYST = .TRUE.
DO 371 KCNT1 = 1, NOVMT
KCNT3 = ICOMLS (ICNT2)
FTGT(KCNT3) = FC*FVMAX(KCNT3)
ICNT2 = ICNT2 + 1
COMPUTE REMAINDER OF TOTAL TARGET FLOW (DELFT) AND DIVIDE BY NUMBER OF VALVES OF NEXT SEQUENCE (FTEMP2)
DELFT = DELFT - CURFC
NOVMT = SEQ (KTEMP2)
FNOVMT = FLOAT(NOVMT)
FTEMP2 = DELFT/FNOVMT
SET TARGET FLOW ON VALVES OF NEXT SEQUENCE = TO FTEMP2
DO 372 KCNT1 = 1, NOVMT
KCNT3 = ICOMLS (ICNT2)
FTGT(KCNT3) = FTEMP2
ICNT2 = ICNT2 + 1
390
380
TOTAL FLOW ON CURRENT SEQUENCE
380
DELFT = DELFT - CURFAN
NOVMT = SEQ(CURSEQ)
FTEMP1 = DELFT / FVMTG(CURSEQ)
HYST = .TRUE.
COMPUTE RATIO OF TOTAL FLOW TO MAX FLOW ON CURRENT SEQUENCE (FTEMP1)
DO 381 KCNT1 = 1, NOVMT
KCNT3 = ICOMLS(ICNT2)
FTGT(KCNT3) = FTEMP1*FVMAX(KCNT3)
ICNT2 = ICNT2 + 1
COMPUTE TARGET FLOW ON EACH VALVE OF CURRENT SEQ BY MULTIPLYING THE RATIO (FTEMP1) WITH MAX FLOW FOR EACH INDIVIDUAL VALVE
NOVMT = SEQ(CURSEQ)
SET TARGET FLOW ON VALVES OF NEXT SEQUENCE TO BE THE LOWER LIMIT FLOW (FA)
DO 382 KCNT1 = 1, NOVMT
KCNT3 = ICOMLS (ICNT2)
FTGT(KCNT3) = FA*FVMAX(KCNT3)
ICNT2 = ICNT2 + 1
390

FIG. 38

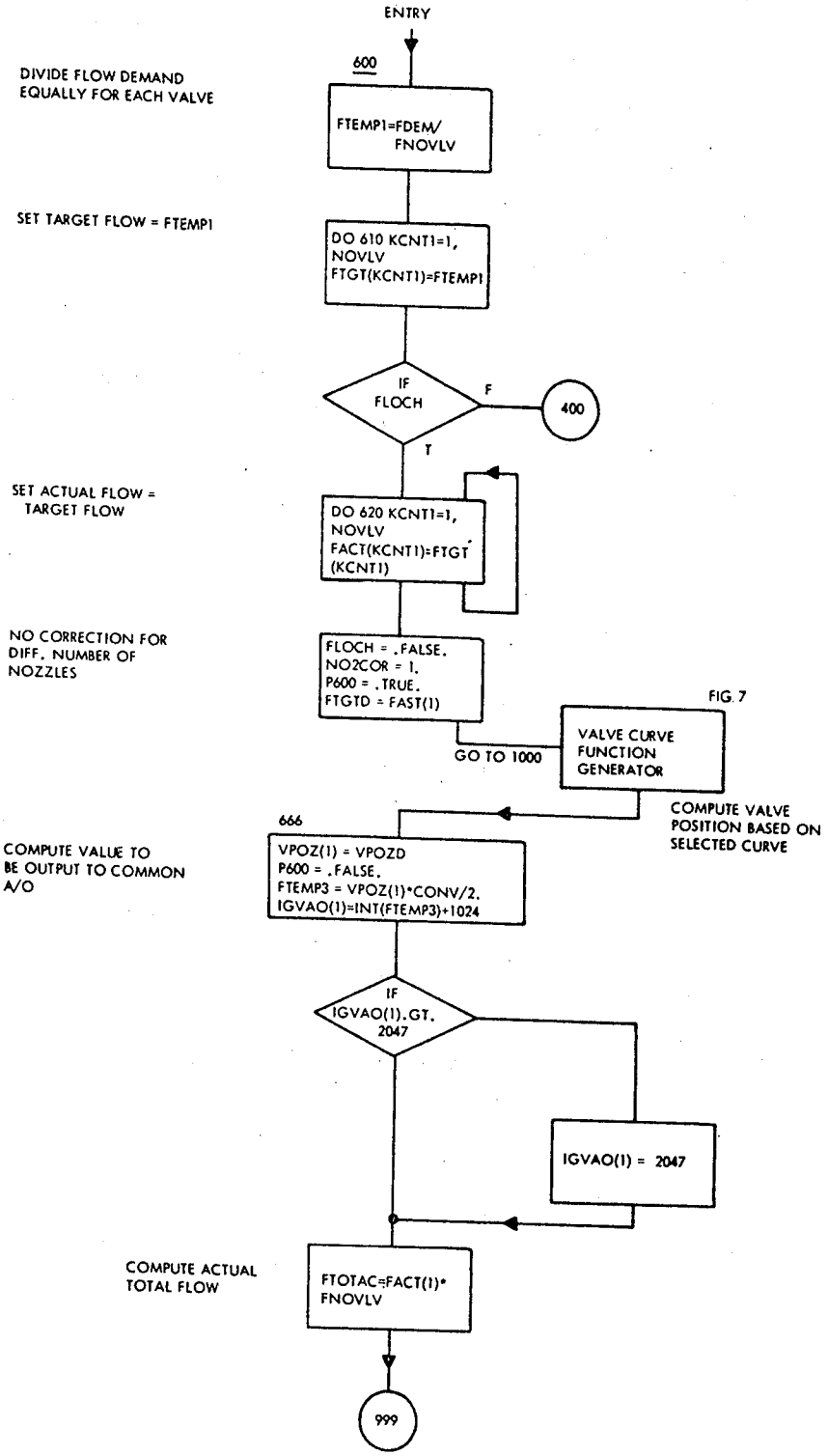

ENTRY
600
DIVIDE FLOW DEMAND EQUALLY FOR EACH VALVE
FTEMP1=FDEM/ FNOVLV
SET TARGET FLOW = FTEMP1
DO 610 KCNT1=1, NOVLV
FTGT(KCNT1)=FTEMP1
IF FLOCH
F
400
T
SET ACTUAL FLOW = TARGET FLOW
DO 620 KCNT1=1, NOVLV
FACT(KCNT1)=FTGT (KCNT1)
NO CORRECTION FOR DIFF. NUMBER OF NOZZLES
FLOCH = .FALSE.
NO2COR = 1.
P600 = .TRUE.
FTGTD = FAST(1)
GO TO 1000
VALVE CURVE FUNCTION GENERATOR
FIG. 7
COMPUTE VALVE POSITION BASED ON SELECTED CURVE
666
COMPUTE VALUE TO BE OUTPUT TO COMMON A/O
VPOZ(1) = VPOZD
P600 = .FALSE.
FTEMP3 = VPOZ(1)*CONV/2.
IGVAO(1)=INT(FTEMP3)+1024
IF IGVAO(1).GT. 2047
IGVAO(1) = 2047
COMPUTE ACTUAL TOTAL FLOW
FTOTAC=FACT(1)* FNOVLV
999

FIG. 39

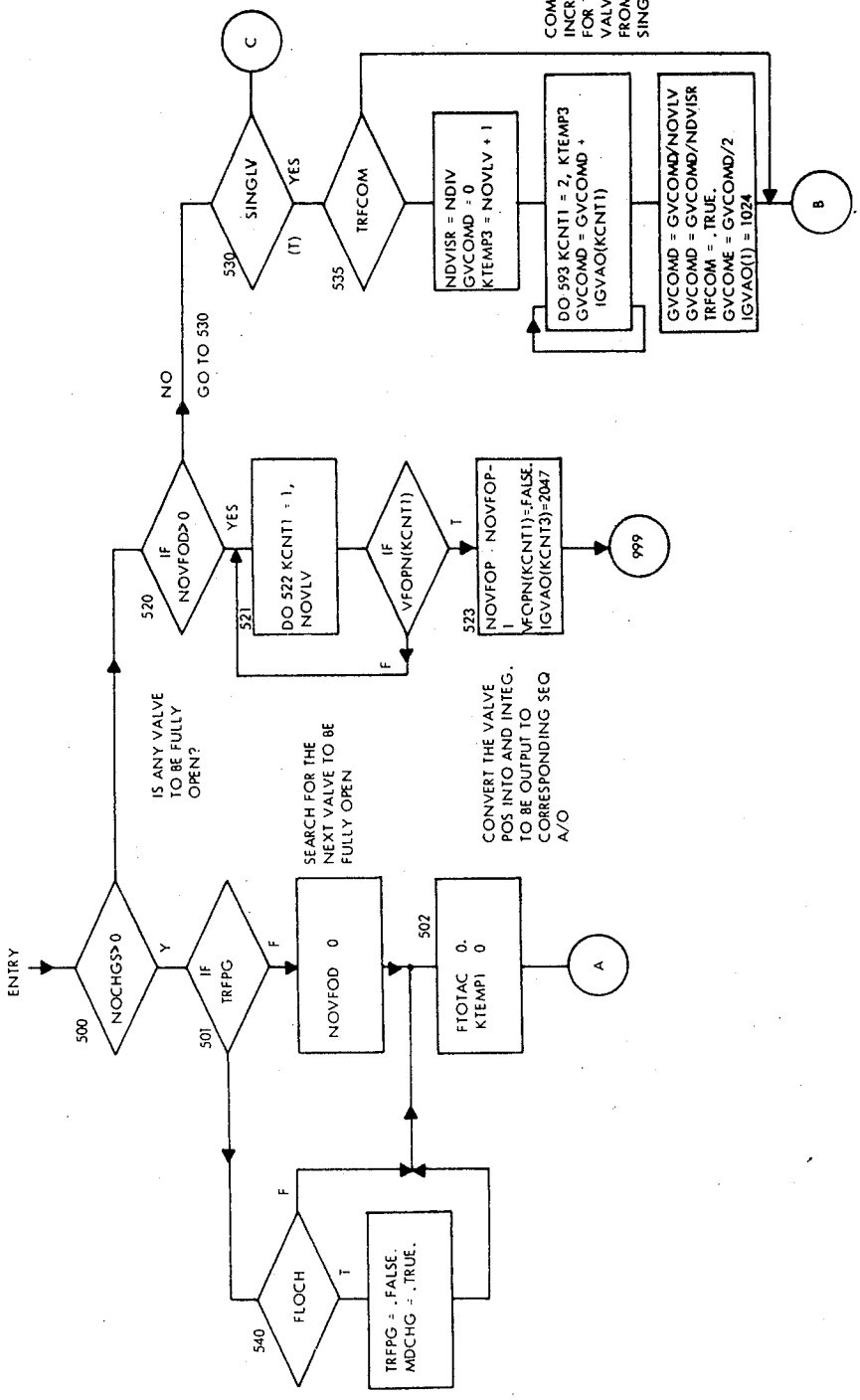

ENTRY
NOCHGS>0
500
Y
IF TRFPG
501
F
NOVFOD 0
502
FTOTAC 0.
KTEMP1 0
A
IS ANY VALVE TO BE FULLY OPEN?
SEARCH FOR THE NEXT VALVE TO BE FULLY OPEN
CONVERT THE VALVE POS INTO AND INTEG. TO BE OUTPUT TO CORRESPONDING SEQ A/O
IF NOVFOD>0
520
YES
NO
GO TO 530
DO 522 KCNT1 = 1, NOVLV
521
IF VFOPN(KCNT1)
T
F
NOVFOP = NOVFOP - 1
VFOPN(KCNT1)=.FALSE.
IGVAO(KCNT3)=2047
523
999
C
SINGLV
530
(T)
YES
TRFCOM
535
NDVISR = NDIV
GVCOMD = 0
KTEMP3 = NOVLV + 1
DO 593 KCNT1 = 2, KTEMP3
GVCOMD = GVCOMD + IGVAO(KCNT1)
GVCOMD = GVCOMD/NOVLV
GVCOMD = GVCOMD/NDVISR
TRFCOM = .TRUE.
GVCOME = GVCOMD/2
IGVAO(1) = 1024
B
COMPUTE THE INCREMENTAL VALUES FOR TRANSFER OF VALVE POSITION DATA FROM SEQ A/O TO SINGLE A/O
FLOCH
540
F
T
TRFPG = .FALSE.
MDCHG = .TRUE.

FIG. 40

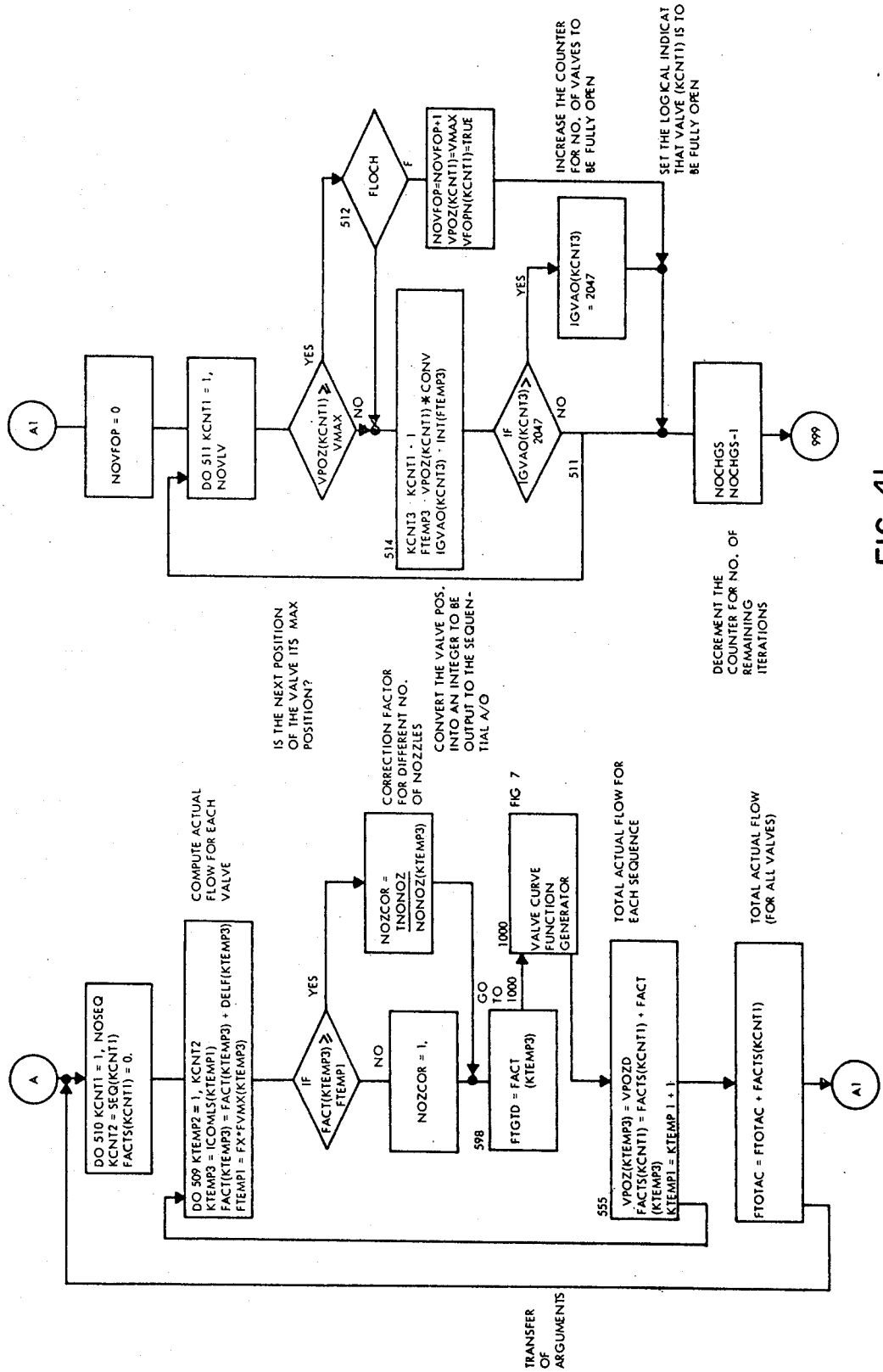

A1
NOVFOP = 0
DO 511 KCNT1 = 1, NOVLV
VPOZ(KCNT1) ≥ VMAX
YES
NO
512
FLOCH
F
NOVFOP=NOVFOP+1
VPOZ(KCNT1)=VMAX
VFOPN(KCNT1)=TRUE
INCREASE THE COUNTER FOR NO. OF VALVES TO BE FULLY OPEN
SET THE LOGICAL INDICAT THAT VALVE (KCNT1) IS TO BE FULLY OPEN
514
KCNT3 · KCNT1 · 1
FTEMP3 · VPOZ(KCNT1) * CONV
IGVAO(KCNT3) · INT(FTEMP3)
IF IGVAO(KCNT3) ≥ 2047
YES
NO
IGVAO(KCNT3) = 2047
511
NOCHGS
NOCHGS-1
999
DECREMENT THE COUNTER FOR NO. OF REMAINING ITERATIONS
IS THE NEXT POSITION OF THE VALVE ITS MAX POSITION?
CORRECTION FACTOR FOR DIFFERENT NO. OF NOZZLES
CONVERT THE VALVE POS. INTO AN INTEGER TO BE OUTPUT TO THE SEQUENTIAL A/O
FIG 7
A
DO 510 KCNT1 = 1, NOSEQ
KCNT2 = SEQ(KCNT1)
FACTS(KCNT1) = 0.
COMPUTE ACTUAL FLOW FOR EACH VALVE
DO 509 KTEMP2 = 1, KCNT2
KTEMP3 = ICOMLS(KTEMP1)
FACT(KTEMP3) = FACT(KTEMP3) + DELF(KTEMP3)
FTEMP1 = FX*FVMAX(KTEMP3)
IF FACT(KTEMP3) ≥ FTEMP1
YES
NO
NOZCOR = INONOZ NONOZ(KTEMP3)
NOZCOR = 1.
598
FTGTD = FACT (KTEMP3)
GO TO 1000
1000
VALVE CURVE FUNCTION GENERATOR
TOTAL ACTUAL FLOW FOR EACH SEQUENCE
555
VPOZ(KTEMP3) = VPOZD
FACTS(KCNT1) = FACTS(KCNT1) + FACT (KTEMP3)
KTEMP1 = KTEMP1 + 1
TOTAL ACTUAL FLOW (FOR ALL VALVES)
FTOTAC = FTOTAC + FACTS(KCNT1)
A1
TRANSFER OF ARGUMENTS

FIG. 41

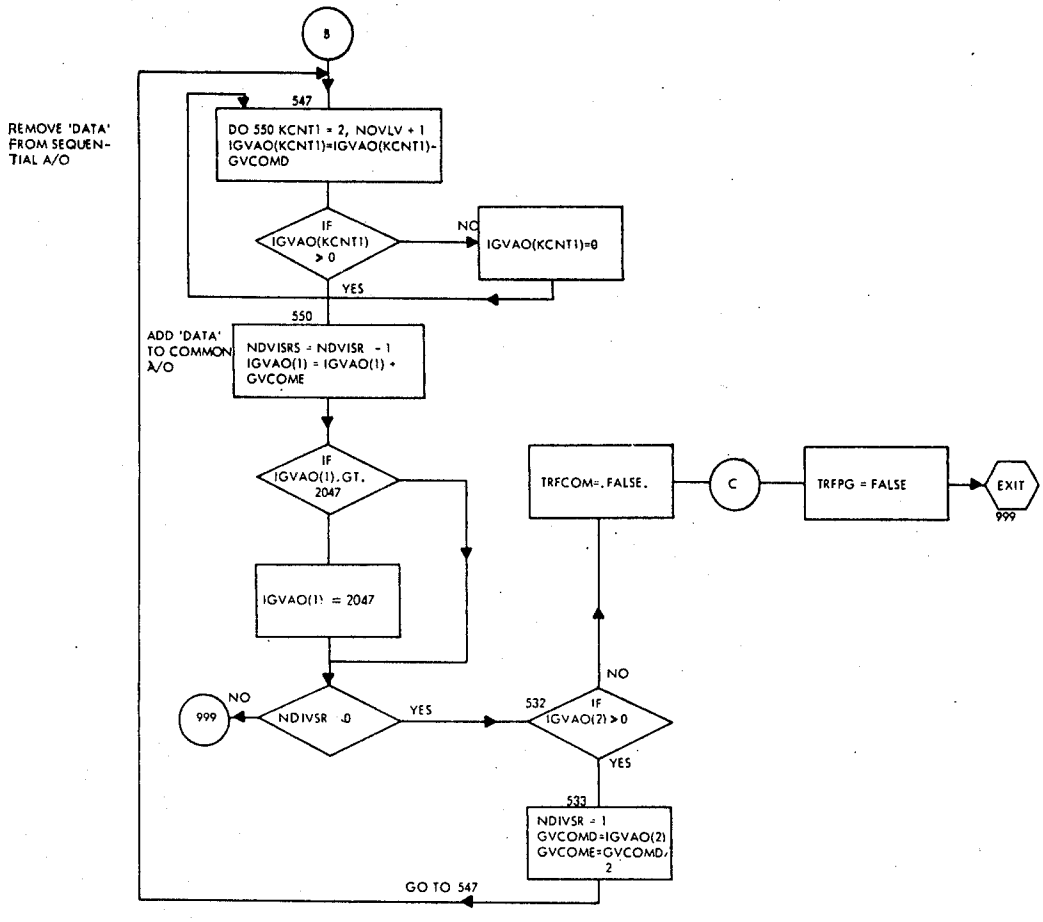

B
547
REMOVE 'DATA' FROM SEQUENTIAL A/O
DO 550 KCNT1 = 2, NOVLV + 1
IGVAO(KCNT1)=IGVAO(KCNT1)-GVCOMD
IF IGVAO(KCNT1) > 0
NO
IGVAO(KCNT1)=0
YES
550
ADD 'DATA' TO COMMON A/O
NDVISRS = NDVISR + 1
IGVAO(1) = IGVAO(1) + GVCOME
IF IGVAO(1).GT. 2047
IGVAO(1) = 2047
TRFCOM=.FALSE.
C
TRFPG = FALSE
EXIT
999
NO
999
NDIVSR .0
YES
532
IF IGVAO(2) > 0
NO
YES
533
NDIVSR = 1
GVCOMD=IGVAO(2)
GVCOME=GVCOMD/2
GO TO 547

FIG. 42

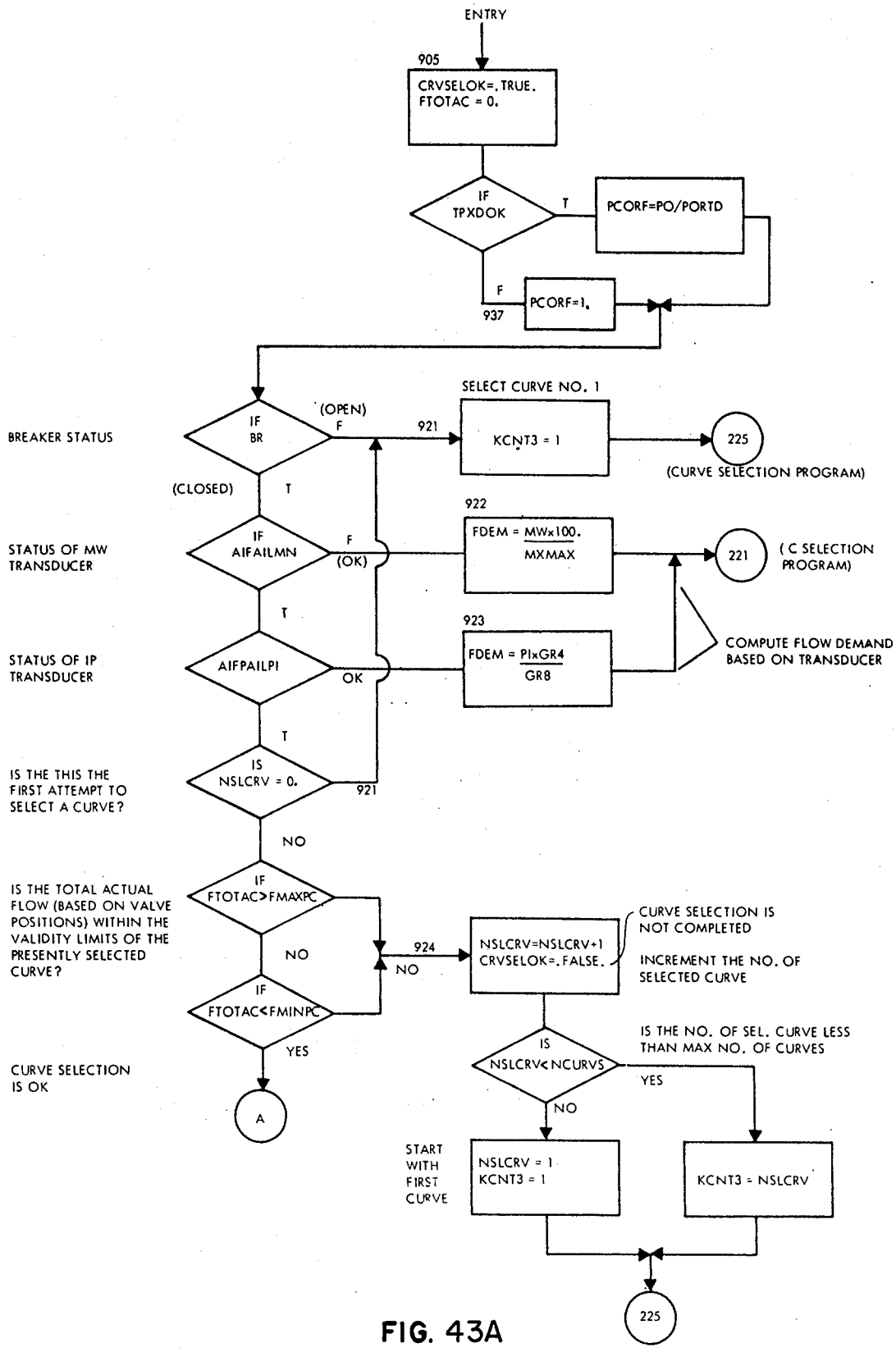

ENTRY
905
CRVSELOK=.TRUE.
FTOTAC = 0.
IF TPXDOK
T
PCORF=PO/PORTD
F
937
PCORF=1.
SELECT CURVE NO. 1
BREAKER STATUS
IF BR
(OPEN)
F
921
KCNT3 = 1
225
(CURVE SELECTION PROGRAM)
(CLOSED)
T
922
STATUS OF MW TRANSDUCER
IF AIFAILMN
F
(OK)
FDEM = MWx100. / MXMAX
221
( C SELECTION PROGRAM)
T
923
STATUS OF IP TRANSDUCER
AIFPAILPI
OK
FDEM = PIxGR4 / GR8
COMPUTE FLOW DEMAND BASED ON TRANSDUCER
T
IS THE THIS THE FIRST ATTEMPT TO SELECT A CURVE?
IS NSLCRV = 0.
921
NO
IS THE TOTAL ACTUAL FLOW (BASED ON VALVE POSITIONS) WITHIN THE VALIDITY LIMITS OF THE PRESENTLY SELECTED CURVE?
IF FTOTAC>FMAXPC
NO
924
NO
NSLCRV=NSLCRV+1
CRVSELOK=.FALSE.
CURVE SELECTION IS NOT COMPLETED
INCREMENT THE NO. OF SELECTED CURVE
IF FTOTAC<FMINPC
YES
IS NSLCRV<NCURVS
IS THE NO. OF SEL. CURVE LESS THAN MAX NO. OF CURVES
YES
NO
CURVE SELECTION IS OK
A
START WITH FIRST CURVE
NSLCRV = 1
KCNT3 = 1
KCNT3 = NSLCRV
225

FIG. 43A

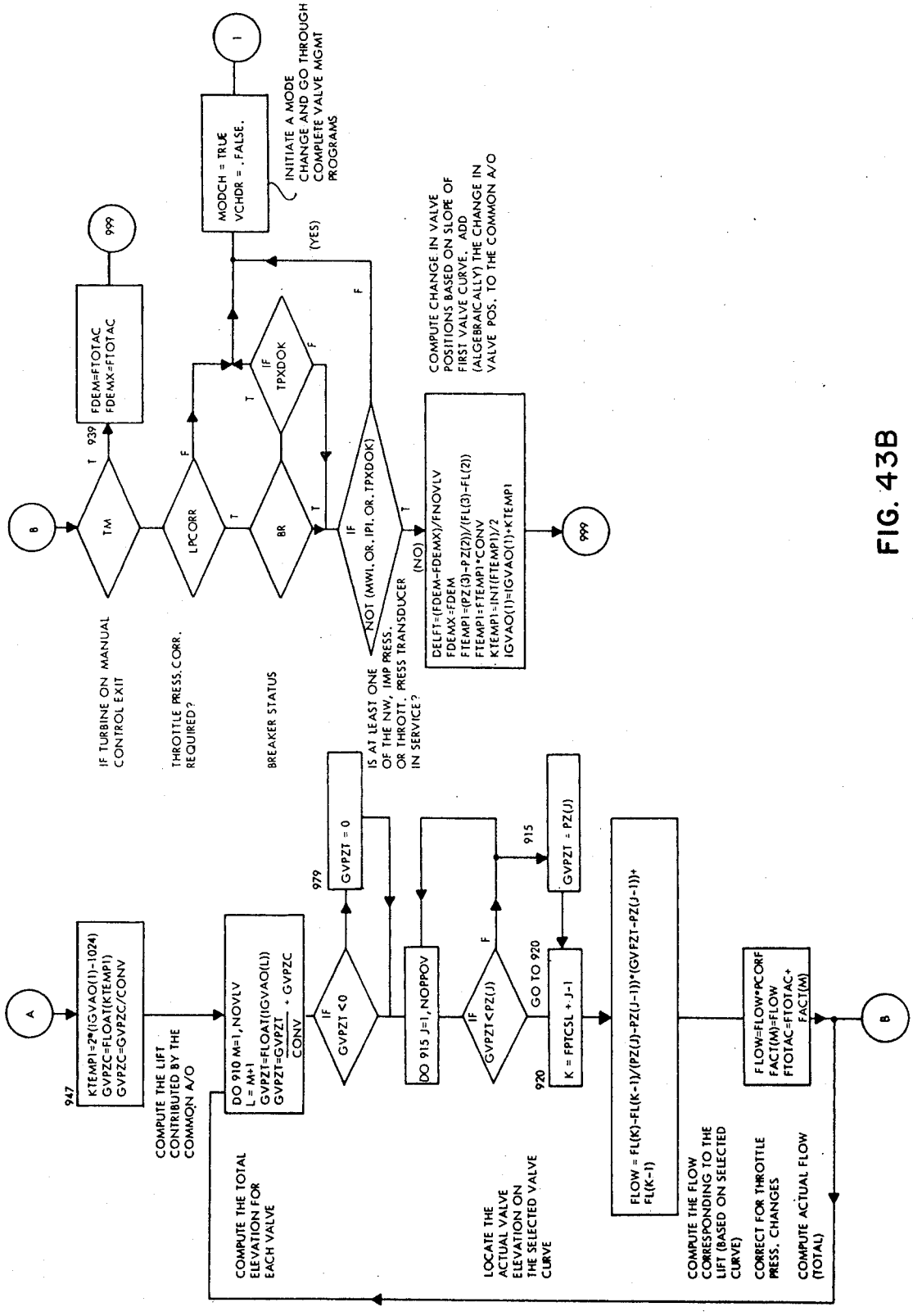

A
947
KTEMP1=2*(IGVAO(1)-1024)
GVPZC=FLOAT(KTEMP1)
GVPZC=GVPZC/CONV
COMPUTE THE LIFT CONTRIBUTED BY THE COMMON A/O
DO 910 M=1,NOVLV
L = M+1
GVPZT=FLOAT(IGVAO(L))
GVPZT=GVPZT/CONV + GVPZC
COMPUTE THE TOTAL ELEVATION FOR EACH VALVE
IF GVPZT <0
979
GVPZT = 0
DO 915 J=1,NOPPOV
IF GVPZT<PZ(J)
915
GO TO 920
920
K = FPTCSL + J-1
GVPZT = PZ(J)
LOCATE THE ACTUAL VALVE ELEVATION ON THE SELECTED VALVE CURVE
FLOW = FL(K)-FL(K-1)/(PZ(J)-PZ(J-1))*(GVPZT-PZ(J-1))+FL(K-1)
COMPUTE THE FLOW CORRESPONDING TO THE LIFT (BASED ON SELECTED CURVE)
FLOW=FLOW*PCORF
FACT(M)=FLOW
FTOTAC=FTOTAC+FACT(M)
CORRECT FOR THROTTLE PRESS. CHANGES
COMPUTE ACTUAL FLOW (TOTAL)
B
IF TURBINE ON MANUAL CONTROL EXIT
TM
T
939
FDEM=FTOTAC
FDEMX=FTOTAC
999
THROTTLE PRESS. CORR. REQUIRED?
LPCORR
F
T
BREAKER STATUS
BR
T
IF TPXDOK
T
F
IS AT LEAST ONE OF THE NW, IMP PRESS. OR THROTT. PRESS TRANSDUCER IN SERVICE?
IF NOT (MW1. OR. IP1. OR. TPXDOK)
F
(NO)
T
DELFT=(FDEM-FDEMX)/FNOVLV
FDEMX=FDEM
FTEMP1=(PZ(3)-PZ(2))/(FL(3)-FL(2))
FTEMP1=FTEMP1*CONV
KTEMP1=INT(FTEMP1)/2
IGVAO(1)=IGVAO(1)+KTEMP1
COMPUTE CHANGE IN VALVE POSITIONS BASED ON SLOPE OF FIRST VALVE CURVE. ADD (ALGEBRAICALLY) THE CHANGE IN VALVE POS. TO THE COMMON A/O
999
(YES)
MODCH = TRUE
VCHDR = .FALSE.
INITIATE A MODE CHANGE AND GO THROUGH COMPLETE VALVE MGMT PROGRAMS
1

FIG. 43B

# SYSTEM AND METHOD FOR OPERATING A TURBINE-POWERED ELECTRICAL GENERATING PLANT IN A SEQUENTIAL MODE

This is a continuation of Ser. No. 306,979, filed Nov. 15, 1972 in the U.S. Pat. Office, and now abandoned.

## CROSS-REFERENCE TO RELATED APPLICATIONS

Ser. No. 408,972, which is a continuation of Ser. No. 247,877, now abandoned, which is a continuation-in-part of Ser. No. 247,440, now abandoned, which is a continuation-in-part of Ser. No. 246,900, now abandoned, and all entitled "General System And Method For Starting, Synchronizing And Operating A Steam Turbine With Digital Computer Control," all filed by Theodore C. Giras and Robert Uram and assigned to the present assignee. Said original application being filed on Apr. 24, 1972.

Ser. No. 306,752, entitled "Improved System And Method Of Controlling Steam Inlet Valves For A Turbine Power Plant," filed by Leamon Podolsky and Theodore C. Giras on Nov. 15, 1972 and assigned to the present assignee.

Ser. No. 306,789, entitled "System and Method For Transferring The Operation Of A Turbine Power Plant Between Single And Sequential Modes Of Turbine Valve Operation," filed by Leaman Podolsky and Uri-George Ronnen on Nov. 15, 1972, and assigned to the present assignee.

Ser. No. 306,943, entitled "System and Method For Transferring Operation Of A Turbine Power Plant Between Manual And Automatic Turbine Valve Operation" filed by Uri George Ronnen and Gerald E. Waldron on Nov. 15, 1972 and assigned to the present assignee.

Ser. No. 306,942, entitled "Improved System and Method For Controlling A Turbine Power Plant In The Single And Sequential Valve Modes With Valve Dynamic Function Generation" filed by Leaman Podolsky on Nov. 15, 1972 and assigned to the present assignee.

## BACKGROUND OF THE INVENTION

In an electrical generating plant powered by a large steam turbine, the high pressure turbine is constructed to receive steam through a plurality of arcuately spaced nozzles adjacent the turbine first stage or impulse blading. The steam then flows through the impulse blading to an impulse chamber and through the remaining rows of high pressure reaction blades. The nozzles are segregated into individual groups about the circumference of the impulse blading; and an individual governor valve controls the steam flow through each nozzle group. There are eight governor valves in a typical fossil turbine generating plant, and four governor valves in a typical nuclear turbine generating plant. In turn, steam is directed to the governor valves through one or more throttle or stop valves from the steam source.

When starting up the turbine, it is common practice to operate all the governor valves in a single valve mode to admit the steam in a full 360° arc through the nozzles to the impulse blading. This practice, which is termed single valve, or full arc, operation permits the heating of the rotor and rotor blades evenly which minimizes thermal shock. However, when the turbine is "hot" and all the governor valves are admitting the required steam in a partially open position, the efficiency



of the plant is considerably reduced because of the pressure drop or throttling action across all the partially open valves. In this situation, the efficiency of the turbine can be increased by admitting the steam through a partial arc of fully-open governor valves with the steam flow variations being controlled by one or more of the remaining valves in a sequential manner.

For such sequential or partial arc operation, the governor valves, for control purposes, are segregated into adjacent groups which may be operated in sequence. For example, in a typical fossil fuel plant, there may be four valves operated as a single valve group to control the steam flow at a low demand, and a single valve in each of a second and third sequential groups operated individually to control steam flow at successively higher steam flow demands; and finally the remaining two valves may be operated as a single valve group to control the flow of steam near the maximum steam flow capacity at the plant.

When a valve is controlling in its non-linear range, such as from zero to 2 percent and from 95 to 100 percent of its maximum flow capacity for example, the valve lift position varies substantially for small increments of steam flow demand. For flow demands in a near linear operating range, such as between 2 percent and 95 percent, for example, small changes in flow demand result in correspondingly small changes in valve lift position. Because such changes are controlled only by one valve or group of valves in the sequential mode, when operating in the non-linear operating range, slight pressure variations, which affect steam flow, or slight changes in steam flow demand, caused by minor load changes, or even electrical noise results in inordinate valve movement or oscillations. Also, such variations can result in an inordinate operation of sequentially adjacent valve groups as the variations cause a simultaneous opening and closing of valves as control is transferred back and forth.

This inordinate operation results in valve wear and perturbations in turbine operation. The elimination of such unnecessary valve operation results in smoother turbine generator operation, even where extreme accuracy is provided by feedforward characterization of steam flow demand.

The prior art, insofar as is known, does not solve this problem, inherent in the sequential mode of operation of turbine power plants.

## SUMMARY OF THE INVENTION

The present invention broadly relates to a system and method of controlling the operation of turbine power plant in the sequential or partial arc mode wherein a plurality of steam inlet valve means are operated successively in a predetermined sequence to effect variations in steam flow to the turbine. A representation of turbine steam flow demand is used to allocate successively portions of the total turbine demand to respective ones of the plurality of valve means so that the allocated steam flows substantially equal to the total turbine flow demand. The steam flow is allocated to sequentially adjacent valve means such that both sequentially adjacent valve means are operated to admit a portion of their maximum steam flow capacity at times when the unallocated portion of the steam flow for such valves is in excess of a predetermined operating value for one valve means and below a predetermined

operating value for the sequentially adjacent valve means.

In a more specific aspect the present invention relates to such a system and method further that includes means to store a physical representation of predetermined operating values for each valve means corresponding to steam flow. The unallocated portion of the total turbine steam flow is compared with the maximum flow each valve means is capable of admitting and the stored operating values. When such unallocated portion of total turbine steam flow is above the predetermined operating value for the one valve means and below the predetermined operating value for the next sequential valve means, then the steam flow is allocated so that both valve means are within a predetermined operating range.

The allocated flow for each valve means is then converted to a valve lift position representation corresponding to a dynamic steam flow versus valve-lift position curve to operate the valve means. Control of incremental changes in total turbine steam flow demand in this manner is confined to the near linear range of valve operation. To further reduce valve movement the stored operating values are different for different directions of total turbine steam flow change.

## BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** shows a schematic diagram of an electric power plant including a large steam turbine and a fossile fuel fired drum type boiler and control devices in which the principles of the present invention are utilized;

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**;

FIG. **3** shows a hydraulic system for supplying hydraulic fluid to valve actuators of the steam turbine;

FIG. **4** shows a schematic diagram of a servo system connected to the valve actuators;

FIG. **5** shows a schematic diagram of a hybrid interface between a manual backup system and the digital computer connected with the servo system controlling the valve actuators;

FIG. **6** shows a simplified block diagram of the digital Electro Hydraulic Control System in which the present invention can be utilized;

FIG. **7** shows a block diagram of a control program used with the present invention;

FIG. **8** shows a block diagram of one embodiment of the programs and subroutines of the digital Electro Hydaulic system used with the present invention;

FIG. **9** shows a typical table of program or task priority assignments of one embodiment of the digital Electro Hydraulic system; FIG. **10** shows a block diagram of a proportional controller function with dead band for a system of the present invention;

FIG. **11** shows a flow chart of a speed loop (SPDLOOP) subroutine for a system of the present invention;

FIGS. **12, 13,** and **14** illustrate portions of a typical operator's control panel used with the system of the invention;

FIG. **15** is a flow chart of a logic contact closure output subroutine for a system utilizing the present invention;

FIG. **16** is a block diagram of conditions which cause initiation of a logic program for a system utilizing the present invention;

FIG. **17** is a simplified block diagram of a portion of the logic function for a system utilizing the present invention;

FIG. **18** is a block diagram of the logic program for a system utilizing the present invention;

FIG. **19** shows a symbolic diagram of the use of a speed/load reference function for a system utilizing the present invention;

FIG. **20** shows a speed/load reference graph of a system for the present invention;

FIG. **21** shows a diagram of a flow demand vs. stage flow coefficient in accordance with the present invention;

FIG. **22** shows a diagram of the total flow demand vs. valve lift in accordance with the present invention;

FIGS. **23A** and **23B** is a general flow chart for control of the governor valves in accordance with the present invention;

FIG. **24** shows a flow chart of a program for transfer between common analog outputs to individual analog outputs to initiate a transfer from single to sequential valve operation in accordance with the present invention;

FIG. **25** includes a flow chart calculation of first stage flow coefficient vs. percentage of total flow in accordance with the present invention;

FIG. **26** shows a flow chart of a subroutine for calculation of the functions of valve curves in accordance with the present invention;

FIG. **27** shows a flow chart of the first state flow coefficient function generator program in accordance with the present invention;

FIG. **28** shows a flow chart for flow change calculations in the sequential mode program in accordance with the present invention;

FIG. **29** shows a calculation for determining a number of incremental changes of flow and valve lift position in accordancd with the present invention;

FIG. **30** shows a flow chart of a program for computing incrementally changing valve lift positions during a mode change in accordance with the principles of the invention;

FIG. **31** shows a flow chart of a subroutine for computing actual flow values after a manual or emergency condition whereupon a connected curve of FIG. **22** is determined in accordance with the principles of the invention;

FIG. **32** show diagrams of sequential mode valve operation in accordance with the principles of the invention;

FIG. **33** shows a flow chart of a modification of a program for the operation of the governor valves which can be substituted for FIG. **23A** when viewed with FIG. **23B**, in accordance with the principles of the invention;

FIGS. **34** and **35** show a flow chart of a program for valve curve selection in accordance with the principles of the invention;

FIG. **36** shows a subroutine for the transfer of the contents of the common A/O to individual A/O's in accordance with the principles of the invention;

FIGS. **37** and **38** shows flow charts of a subroutine for the computation of of target flow changes in the sequential operating mode in accordance with the principles of the invention;

FIG. 39 shows a program for the computation of target flow for transfer from sequential to the single valve operating mode in accordance with the principles of the invention;

FIGS. 40, 41 and 42 show a subroutine for computation of governor valve flow changes and position in accordance with the principles of the invention;

FIGS. 43A and 43B shows a flow chart of the program to compute actual valve flows after a manual or emergency condition in accordance with the principles of the invention;

## DESCRIPTION OF THE PREFERRED EMBODIMENT

### A. POWER PLANT

More specifically, there is shown in FIG. 1 a large single reheat steam turbine constructed in a well known manner and operated and controlled in an electric power plant 12 in accordance with the principles of the invention. As will become more evident through this description, other types of steam turbines can also be controlled in accordance with the principles of the invention and particularly in accordance with th broader aspects of the invention. The generalized electric power plant shown in FIG. 1 and the more general aspects of the computer control system to be described in connection with FIG. 2 are like those disclosed in the aforementioned Giras and Birnbaum U.S. Pat. application Ser. No. 319,115. As already indicated, the present application is directed to general improvements in turbine operation and control as well as more specific improvements related to digital computer operation and control of turbines.

The turbine 10 is provided with a single output shaft 14 which drives a conventional large alternating current generator 16 to produce three-phase electric power (or any other phase electric power) as measured by a conventional power detector 18 which measures the rate of flow of electric energy. Typically, the generator 16 is connected through one or more breakers 17 per phase to a large electric power network and when so connected causes the turbo-generator arrangement to operate at synchronous speed under steady state conditions. Under transient electric load change conditions, system frequency may be affected and conforming turbo-generator 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 operating in accordance with the present invention may 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 or once through boiler systems.

The turbine 10 in this instance is of the plural inlet front end type, and steam flow is accordingly directed to the turbine steam chest (not specifically indicated) through four throttle inlet valves TV1–TV4. Generally, the plural inlet type and other front end turbine types such as the single ended type or the end bar lift type may involve different numbers and/or arrangements of valves.

Steam is directed from the admission steam chest to the first high pressure section expansion stage through eight governor inlet valves GV1–GV8 which are arranged to supply steam to inlets arcuately spaced about the turbine high pressure casing to constitute a somewhat typical governor valving arrangement for large fossil fuel turbines. Nuclear turbines might on the other hand typically utilize only four governor valves.

During start-up, the governor valves GV1–GV8 are typically all fully opened and steam flow control is provided by a full arc throttle valve operation. At some point in the start-up process, transfer is made from full arc throttle valve control to full arc governor valve control because of throttling energy losses and/or throttling control capability. Upon transfer the throttle valves TV1–TV4 are fully opened, and the governor valves GV1-GV8 are normally operated in the single valve mode. Subsequently, the governor valves may be individually operated in a predetermined sequence usually directed to achieving thermal balance on the rotor and reduced rotor blade stressing while producing the desired turbine speed and/or load operating level. For example, in a typical governor valve control mode, governor valves GV5–GV8 may be initially closed as the governor valves GV1–GV4 are jointly operated from time to time to define positions producing the desired corresponding total steam flows. After the governor valves GV1–GV4 have reached the end of their control region, i.e., upon being fully opened, or at some overlap point prior to reaching their fully opened position, the remaining governor valves GV5–GV8 are sequentially placed in operation in numerical order to produce continued steam flow control at higher steam flow levels. This governor valve sequence of operation is based on the assumption that the governor valve controlled inlets are arcuately spaced about the 360° periphery of the turbine high pressure casing and that they are numbered consecutively around the periphery so that the inlets corresponding to the governor valves GV1 and GV8 are arcuately adjacent to each other.

The preferred turbine start-up method is to raise the turbine speed from the turning gear speed of about 2 rpm to about 80% of the synchronous speed under throttle valve control and then transfer to governor valve control and raise the turbine speed to the synchronous speed, then close the power system breakers and meet the load demand. On shutdown, similar but reverse practices or simple coast down may be employed. Other transfer practice may be employed, but it is unlikely that transfer would be made at a loading point above 40% rated load because of throttling efficiency considerations.

After the steam has crossed past the first stage impulse blading to the first stage reaction blading of the high pressure section, it is directed to a reheater system 28 which is associated with a boiler or steam generating

system **26.** In practice, the reheater system **28** may 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 reheated 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, a stop valve SV including one or more check valves is normally open and closed only when the turbine is tripped. Interceptor valves IV (only one indicated), are also provided in the reheat steam flow path, and they are normally open and if desired they may be operated over a range of position control to provide reheat steam flow cutback modulation under turbine overspeed conditions. Further description of an appropriate overspeed protection system is presented in the U.S. Pat. No. 3,643,437 issued to M. Birnbaum, A. Braytenbah and A. Richardson and assigned to the present assignee.

In the typical fossil fuel drum type boiler steam generating system, the boiler system controls boiler operations so that steam throttle pressure is held substantially constant. In the present description, it is therefore assumed as previously indicated that throttle pressure is an externally controlled variable upon which the turbine operation can be based. A throttle pressure detector **38** of suitable conventional design measures the 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.

In general, the steady state power or load developed by a steam turbine supplied with substantially constant throttle pressure steam is determined as follows: Equation (1)

$$\text{power or load} = K_P\,(P_i/P_0) = K_F\,S_F$$

where
- $P_i$ = first stage impulse pressure
- $P_0$ = throttle pressure
- $K_P$ = constant of proportionality
- $S_F$ = steam flow
- $K_F$ = constant of proportionality

Where the throttle pressure is held substantially constant by external control as in the present case, the turbine load is thus proportional to the first stage impulse pressure $P_i$. The ratio $P_i/P_0$ may be used for control purposes, for example to obtain better anticipatory control of $P_i$ (i.e. turbine load) as the boiler control throttle pressure $P_0$ undergoes some variation within protective constraint limit values. However, it is preferred in the present case that the impulse pressure $P_i$ be used for feedback signalling in load control operation as subsequently more fully described, and a conventional pressure detector **40** is employed to determine the pressure $P_i$ for the assigned control usage.

Within its broad field of applicability, the invention can also be applied in nuclear reactor and other applications involving steam generating systems which produce steam without placement of relatively close steam generator control on the constancy of the turbine throttle pressure. In such cases, throttle control and operating philosophies are embodied in a form preferred for and tailored to the type of plant and turbine involved. In cases of unregulated throttle pressure supply, turbine operation may be directed with top priority to throttle pressure control or constraint and with lower priority to turbine load and/or speed control.

Respective hydraulically operated throttle valve actuators indicated by the reference character **42** are provided for the four throttle valves TV1–TV4. Similarly, respective hydraulically operated governor valve actuators indicated by the reference character **44** are provided for the eight governor valves GV1–GV8. Hydrauically operated actuators indicated by the reference characters **46** and **48** are provided for the reheat stop and interceptor valves SV and IV. A computer monitored high pressure fluid supply **50** provides the controlling fluid for actuator operation of the valves TV1–TV4, GV1–GV8, SV and IV. A computer supervised lubricating oil system (not shown) is separately provided for turbine plant lubricating requirements.

The respective actuators **42, 44, 46** and **48** are of conventional construction, and the inlet valve actuators **42** and **44** are operated by respective stabilizing position controls indicated by the reference characters **50** and **52.** If desired, the interceptor valve actuators **48** can also be operated by a position control **56** although such control is not employed in the present detailed embodiment of the invention. Each position control includes a conventional analog controller (not shown in FIG. **1**) which drives a suitably known actuator servo valve (not indicated) in the well known manner. The reheat stop valve actuators **46** are fully open unless the conventional trip system or other operating means causes them to close and stops the reheat steam flow.

Since the turbine power is proportional to steam flow under the assumed control condition of substantially constant throttle pressure, steam valve positions are controlled to produce control over steam flow as an intermediate variable and over turbine speed and/or load as an end control variable or variables. Actuator operation provides the steam valve positioning, and respective valve position detectors PDT1–PDT4, PDG1–PDG8 and PDI are provided to generate respective valve position feedback signals for developing position error signals to be applied to the respective position controls **50, 52** and **56.** One or more contact sensors CSS provides status data for the stop valving SV. The position detectors are provided in suitable conventional form, for example, they may make conventional use of linear variable differential transformer operation in generating negative position feedback signals for algebraic summing with respect to position setpoint signals SP in developing the respective input error signals. Position controlled operation of the interceptor valving IV would typically be provided only under a reheat steam flow cutback requirement.

The combined position control, hydraulic actuator, valve position detector element and other miscellaneous devices (not shown) form a local hydraulic electric analog valve position control for each throttle or 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 may also be computed for the interceptor valve controls when the latter are employed. 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.

In the present case, the described hybrid arrangement including local loop analog electrohydraulic position control is preferred primarily because of the combined effects of control computer operating speed capabilities and computer hardware economics, i.e., the cost of manual backup analog controls is less than that for backup computer capacity at present control computer operating speeds for particular applications so far developed. Further consideration of the hybrid aspects of the turbine control system is presented subsequently herein. However, economic and fast operating backup control computer capability is expected and direct digital computer control of the hydraulic valve actuators will then likely be preferred over the digital control of local analog controls described herein.

A speed detector **58** is provided for determining the turbine shaft speed for speed control and for frequency participation control purposes. The speed detector **58** can for example be in the form of a reluctance pickup (not shown) magnetically coupled to a notched wheel (not shown) on the turbo-generator shaft **14.** In the detailed embodiment subsequently described herein, a plurality of sensors are employed for speed detection. Analog and/or pulse signals produced by the speed detector **58,** the electric power detector **18,** the pressure detectors **38** and **40,** the valve position detectors PDT1–PDT4, PDG1–PDG8 and PDI, the status contact or contacts CSS, and other sensors (not shown) and status contacts (not shown) are employed in programmed computer operation of the turbine **10** for various purposes including controlling turbine performance on an on-line real time basis and further including monitoring, sequencing, supervising, alarming, displaying and logging.

## B. DEH-COMPUTER CONTROL SYSTEM

As generally illustrated in FIG. **2,** a Digital Electro-Hydraulic control system (DEH) **1100** includes a programmed digital computer **210** to operate the turbine **10** and the plant **12** with improved performance and operating characteristics. The computer **210** can include conventional hardware including a central processor **212** and a memory **214.** The digital computer **210** and its associated input/output interfacing equipment is a suitable digital computer system such as that sold by Westinghouse Electric Corporation under the trade name of *P*2000. In cases when the steam generating system **26** as well as the turbine **10** are placed under computer control, use can be made of one or more 9,2000 computers or alternatively a larger computer system such as that sold by Xerox Data Systems and known as the Sigma 5. Separate computers, such as *P*2000 computers, can be employed for the respective steam generation and turbine control functions in the controlled plant unit and interaction is achieved by interconnecting the separate computers together through data links or other means.

The digital computer used in the DEH control system **1100** is a *P*2000 computer which is designed for real time process control applications. The *P*2000 typically uses a 16 bit word length with 2's complement, a single address and fixed word length operated in a parallel mode. All the basic DEH system functions are performed with a 16,000 word (16K), 3 microsecond magnetic core memory. The integral magnetic core memory can be expanded to 65,000 words (65K).

The equipment interfacing with the computer **210** includes a contact interrupt system **124** which scans contacts representing the status of various plant and equipment conditions in plant wiring **1126.** The status contacts might typically be contacts of mercury wetted relays (not shown) which operate by energization circuits (not shown) capable of sensing the predetermined conditions associated with the various system devices. Data from status contacts is used in interlock logic functioning and control for other programs, protection analog system functioning, programmed monitoring and logging and demand logging, etc.

Operator's panel buttons **1130** transmit digital information to the computer **2010.** The operator's panel buttons **1130** can set a load reference, a pulse pressure, megawatt output, speed, etc.

In addition, interfacing with plant instrumentation **1118** is provided by an analog input system **1116.** The analog input system **1116** samples analog signals at a predetermined rate from predetermined input channels and converts the signals sampled to digital values for entry into the computer **210.** The analog signals sensed in the plant instrumentation **1118** parameters including the impulse chamber pressure, the megawatt power, the valve positions of the throttle valves TV1 through TV4 and the governor valves GV1 through GV8 and the interceptor valve IV, throttle pressure, steam flow, various steam temperatures, miscellaneous equipment operating temperature, generator hydrogen cooling pressure and temperature, etc. Such parameters include process parameters which are sensed or controlled in the process (turbine or plant) and other variables which are defined for the use in the programmed computer operation. Interfacing from external systems such as an automatic dispatch system is controlled through the operator's panel buttons **1130.**

A conventional programmer's console and tape reader **218** is provided for various purposes including program entry into the central processor **212** and the memory **214** thereof. A logging typewriter **1146** is provided for logging printouts of various monitored parameters as well as alarms generated by an automatic turbine startup system (ATS) which includes program system blocks **1140, 1142, 1144** (FIG. **8**) in the DEH control system **1100.** A trend recorder **1147** continuously records predetermined parameters of the system. An interrupt system **124** is provided for controlling the input and output transfer of information between the digital computer **210** and the input/output equipment. The digital computer **210** acts on interrupt from the interrupt system **124** in accordance with an executive program. Interrupt signals from the interrupt system **124** stop the digital computer **210** by interrupting a program in operation. The interrupt signals are serviced immediately.

Output interfacing is provided by contacts **1128** for the computer **210.** The contacts **1128** operate status display lamps, and they operate in conjunction with a conventional analog/output system and a valve position control output system comprising a throttle valve control system **220** and a governor valve control system **222.** A manual control system, is coupled to the valve position control output system **220** and is operable therewith to provide manual turbine control during

computer shut-down. The throttle and governor valve control systems **220** and **222** correspond to the valve position controls **50** and **52** and the actuators **42** and **44** in FIG. **1.** Generally, the manual control system is similar to those disclosed in prior U.S. Pat. No. 3,552,872 by T. Giras et al. and U.S. Pat. No. 3,741,246 by A. Braytenbah, both assigned to the present assignee.

Digital output data from the computer **210** is first converted to analog signals in the analog output system **224** and then transmitted to the valve control system **220** and **222.** Analog signals are also applied to auxiliary devices and systems, not shown, and interceptor valve systems, not shown.

## C. SUBSYSTEMS EXTERNAL TO THE DEH COMPUTER

At this point in the description, further consideration of certain subsystems external to the DEH computer will aid in reaching an understanding of the invention. Making reference now to FIG. **3,** a high pressure HP fluid supply system **310** for use in controlled actuation of the governor valves GV1 through GV**8,** the throttle valves TV1 through TV**4** and associated valves is shown. The high pressure fluid supply system **310** corresponds to the supply system **49** in FIG. **1** and it uses a synthetic, fire retardant phosphate ester-based fluid and operates in the range of 1,500 and 1,800 psi. Nitrogen charged piston type accumulators **312** maintain a flow of fluid to the actuators for the governor valves GV1–GV**8,** the throttle valves TV1–TV**4,** etc. when pumps **314** and **316** are discharging to a reservoir **318** through unloader valves **320** and **321.** In addition, the accumulators **312** provide additional transient flow capacity for rapid valve movements.

Referring now to FIG. **4,** a typical electrohydraulic valve actuation system **322** is shown in greater detail for positioning a modulating type valve actuator **410** against the closing force of a large coil spring. A servo-valve **412** which is driven by a servo-amplifier **414** controls the flow of fluid therethrough. The servo-valve **412** controls the flow of fluid entering or leaving the valve actuater cylinder **416** relative to the HP fluid supply system **310.** A linear voltage differential transformer LVTD generates a valve position indicating transducer voltage which is summed with a valve position demand voltage at connection **418.** The summation of the two previously mentioned voltages produces a valve position error input signal to the servo-amplifier **414.** The linear voltage differential transformer LVTD has a linear voltage characteristic with respect to displacement thereof in the preferred embodiment. Therefore, the position of the valve actuator **410** is made proportional to the valve position demand voltage at connection **418.**

Making reference now to FIG. **5,** a hardwired digital-/analog system **512** forms a part of the DEH control system **1100** (FIG. **2**). Structurally, it embraces elements which are included in the blocks **50, 52, 42** and **44** of FIG. **1** as well as additional elements. A hybrid interface **510** is included as a part of the hardwired system **512.** The hybrid interface **510** is connected to actuator system servo-amplifiers **414** for the various steam valves which in turn are connected to a manual controller **516,** an overspeed protection controller, not shown, and redundant DC power supplies, not shown.

A controller shown in FIG. **5** is employed for throttle valve TV1–TV4 control in the TV control system **50** of FIG. **1.** The governor valves GV1–GV**8** are controlled in an analogous fashion by the GV control system **52.**

While the steam turbine is controlled by the digital computer **210,** the hardwired system **511** tracks signal valve analog outputs **520** from the digital computer **210.** A comparator **518** compares a signal from a digital-to-analog converter **522** of the manual system with the signal **520** from the digital computer **210.** A signal from the comparator **518** controls a logic system **524** such that the logic system **524** runs an up-down counter **526** to the point where the output of the converter **522** is equal to the output signal **520** from the digital computer **210.** Should the hardwired system **511** fail to track the signal **520** from the digital computer **210** a monitor light will flash on the operator's panel. A sequential valve A/O **521***a, b, c,* etc., controls the governor valve GV1–GV**8** servo amplifiers in partial arc admission mode.

When the DEH control system reverts to the control of the backup manual controller **516** as a result of an operator selection or due to a contingency condition, such as loss of power on the automatic digital computer **210,** or a stoppage of a function in the digital computer **210,** or a loss of a speed channel in the wide range speed control all as described in greater detail infra, the input of the valve actuation system **322** (FIG. **4**) is switched by switches **528** from the automatic controllers in the blocks **50, 52** (FIG. **1**) or **220, 222** (FIG. **2**) to the control of the manual controller **516.** Bumpless transfer is thereby accomplished between the digital computer **210** and the manual controller **516.**

Similarly, tracking is provided in the computer **210** for switching bumplessly from manual to automatic turbine control. As previously indicated, the presently disclosed hybrid structural arrangement of software and hardware elements is the preferred arrangement for the provision of improved turbine and plant operation and control wth backup capability. However, other hybrid arrangements can be implemented within the field of application of the invention.

## D. DEH PROGRAM SYSTEM

### DEH Program System Organization, DEH Control Loops And Control Task Program

With reference now to FIG. **6,** an overall generalized control system of this invention is shown in block diagram form. The digital electrohydraulic (DEH) control system **1100** operates valve actuators **1012** for the turbine **10.** The digital electrohydraulic control system **1100** comprises a digital computer **1014,** corresponding to the digital computer **210** in FIG. **2,** and it is interconnected with a hardwired analog backup control system **1016.** The digital computer **1014** and the backup control system **1,016** are connected to an electronic servo system **1018** corresponding to blocks **220** and **222,** in FIG. **2.** The digital computer control system **1014** and the analog backup system **1016** track each other during turbine operations in the event it becomes necessary or desirable to make a bumpless transfer of control from a digital computer controlled automatic mode of operation to a manual analog backup mode or from the manual mode to the digital automatic mode.

In order to provide plant and turbine monitor and control functions and to provide operator interface functions, the DEH computer **1,014** is programmed

with a system of task and task support programs. The program system is organized efficiently and economically to achieve the end operating functions. Control functions are achieved by control loops which structurally include both hardware and software elements, with the software elements being included in the computer program system. Elements of the program system are considered herein to a level of detail sufficient to reach an understanding of the invention.

As previously discussed, a primary function of the digital electrohydraulic (DEH) system **1100** is to automatically position the turbine throttle valves TV1 through TV4 and the governor valves GV1 through GV8 at all times to maintain turbine speed and/or load. A special periodically executed program designated the CONTROL task is utilized by the P2000 computer along with other programs to be described in greater detail subsequently herein.

With reference now to FIG. 7, a functional control loop diagram in its preferred form includes the CONTROL task or program **1020** which is executed in the computer **1010.** Inputs representing demand and rate provide the desired turbine operating setpoints. The demand is typically either the target speed in specified revolutions per minute of the turbine systems during startup or shutdown operations or the target load in megawatts of electrical output to be produced by the generating system **16** during load opeations. The demand enters the block diagram configuration of FIG. 7 at the input **1,050** of a compare block **1,052.**

The rate input either in specified RPM per minute or specified megawatts per minute, depending upon which input is to be used in the demand function, is applied to an integrator block **1054.** The rate inputs in RPM and megawatts of loading per minute are established to limit the buildup of stresses in the rotor of the turbine-generator **10.** An error output of the compare block **1052** is applied to the integrator block **1054.** In generating the error output the demand value is compared with a reference corresponding to the present turbine operating setpoint in the compare block **1,052.** The reference value is representative of the setpoint RPM applied to the turbine system or the setpoint generator megawatts output, depending upon whether the turbine generating system is in the speed mode of operation or the load mode of operation. The error output is applied to the integrator **1,054** so that a negative error drives the integrator **1,054** in one sense and a positive error drives it in the opposite sense. The polarity error normally drives the integrator **1,054** until the reference and the demand are equal or if desired until they bear some other predetermined relationship with each other. The rate input to the integrator **1,054** varies the rate of integration, i.e. the rate at which the reference or the turbine operating setpoint moves toward the entered demand.

Demand and rate input signals can be entered by a human operator from a keyboard. Inputs for rate and demand can also be generated or selected by automatic synchronizing equipment, by automatic dispatching system equipment external to the computer, by another computer automatic turbine startup program or by a boiler control system. The inputs for demand and rate in automatic synchronizing and boiler control modes are preferably discrete pulses. However, time control pulse widths or continuous analog input signals may also be utilized. In the automatic startup mode, the turbine acceleration is controlled as a function of detected turbine operating conditions including rotor thermal stress. Similarly, loading rate can be controlled as a function of detected turbine operating conditions.

The output from the integrator **1,054** is applied to a breaker decision block **1,060.** The breaker decision block **1,060** checks the state of the main generator circuit breaker **17** and whether speed control or load control is to be used. The breaker block **1,060** then makes a decision as to the use of the reference value. The decision made by the breaker block **1,060** is placed at the earliest possible point in the control task **1,020** thereby reducing computational time and subsequently the duty cycle required by the control task **1,020.** If the main generator circuit breaker **17** is open whereby the turbine system is in wide range speed control the reference is applied to the compare block **1,062** and compared with the actual turbine generator speed in a feedback type control loop. A speed error value from the compare block **1,062** is fed to a proportional plus reset controller block **1,068,** to be described in greater detail later herein. The proportional plus reset controller **1,068** provides and integrating function in the control task **1,060** which reduces the speed error signal to zero. In the prior art, speed control systems limited to proportional controllers are unable to reduce a speed error signal to zero. During manual operation an offset in the required setpoint is no longer required in order to maintain the turbine speed at a predetermined value. Great accuracy and precision of turbine speed whereby the turbine speed is held within one RPM over tens of minutes is also accomplished. The accuracy of speed is so high that the turbine **10** can be manually synchronized to the power line without an external synchronizer typically required. An output from the proportional plus reset controller block **1,068** is then processed for external actuation and positioning of the appropriate throttle and/or governor valves.

If the main generator circuit breaker **17** is closed, the CONTROL task **1,020** advances from the breaker block **1,060** to a summer **1,072** where the REFERENCE acts as a feedforward set point in a combined feedforward-feedback load control system. If the main generator circuit breaker **17** is closed, the turbine generator system **10** is being loaded by the electrical network connected thereto.

In the control task **1,020** of the DEH system **1100** utilizes the summer **1,072** to compare the reference value with the output of speed loop **1,310** in order to keep the speed correction independent of load. A multiplier function has a sensitivity to varying load which is objectionable in the speed loop **1,310.**

During the load mode of operation the DEMAND represents the specified loading MW of the generator **16,** which is to be held at a predetermined value by the DEH system **1,100.** However, the actual load will be modified by any deviations in system frequency in accordance with a predetermined regulation value. To provide for frequency participation, a rated speed value in box **1,074** is compared in box **1078** with a "two signal" speed value represented by box **1,076.** The two signal speed system provides high turbine operating reliability to be described infra herein. An output from the compare function **1078** is fed through a function **1080** which is similar to a proportional controller which converts the speed error value in accordance with the regulation value. The speed error from the

proportional controller **1080** is combined with the feedforward megawatt reference, i.e., the speed error and the megawatt reference are summed in summation function or box **1,072** to generate a combined speed compensated reference signal.

The speed compensated load reference is compared with actual megawatts in a compare box or function **1,082.** The resultant error is then run through a proportional plus reset controller represented by program box **1,084** to generate a feedback megawatt trim.

The feedforward speed compensated reference is trimmed by the megawatt feedback error multiplicatively to correct load mismatch, i.e. they are multiplied together in the feedforward turbine reference path by multiplication function **1,086.** Multiplication is utilized as a safety feature such that if one signal e.g. MW should fail a large value would not result which could cause an overspeed condition but instead the DEH system **1100** would switch to a manual mode. The resulting speed compensated and megawatt trimmed reference serves as an impulse pressure setpoint in an impulse pressure controller and it is compared with a feedback impulse chamber pressure representation from input **1,088.** The difference between the feedforward reference and the impulse pressure is developed by a comparator function **1,090,** and the error output therefrom functions in a feedback impulse pressure control loop. Thus, the impulse pressure error is applied to a proportional plus reset controller function **1,092.**

During load control the megawatt loop comprising in part blocks **1,082** and **1,084** may be switched out of service leaving the speed loop **1,310** and an impulse pressure loop operative in the DEH system **1,100.**

Impulse pressure responds very quickly to changes of load and steam flow and therefore provides a signal with minimum lag which smooths the output response of the turbine generator **10** because the lag dynamics and subsequent transient response is minimized. The impulse pressure input may be switched in and out from the compare function **1,090.** An alternative embodiment embracing feedforward control with impulse pressure feedback trim is applicable.

Between block **1,092** and the governor valves GV1–GV8 a valve characterization function for the purpose of linearizing the response of the valves is interposed. The valve characterization function is utilized in both automatic modes and manual modes of operation of the DEH system **1,100.** The output of the proportional plus reset controller function **1,092** is then ultimately coupled to the governor valves GV1–GV8 through electrohydraulic position control loops implemented by equipment considered elsewhere herein. The proportional plus reset controller output **1,092** causes positioning of the governor valves GV1–GV8 in load control to achieve the desired megawatt demand while compensation is made for speed, megawatt and impulse pressure deviations from desired setpoints.

Making reference to FIG. **8,** the control program **1,020** is shown with interconnections to other programs in the program system employed in the Digital Electro Hydraulic (DEH) system **1,100.** The periodically executed program **1,020** receives data from a logic task **1,110** where mode and other decisions which affect the control program are made, a panel task **1,112** where operator inputs may be determined to affect the control program, an auxiliary synchronizer program **1,114** and an analog scan program **1,116** which processes input process data. The analog scan task **1,116** receives data from plant instrumentation **1,118** external to the computer as considered elsewhere herein, in the form of pressures, temperatures, speeds, etc and converts such data to proper form for use by other programs. Generally, the auxiliary synchronizer program **1,114** measures time for certain important events and it bids the control and other programs. An extremely accurate clock function **1,120** operates through a monitor program **1,122** to run the auxiliary synchronizer program **1,114.**

The monitor program or exectuve package **1,122** also provides for controlling certain input/output operations of the computer and, more generally, it schedules the use of the computer to the various programs in accordance with assigned priorities.

The logic task **1,110** is fed from outputs of a contact interrupt or sequence of events program **1,124** which monitors contact variables in the power plant **1,126.** The contact parameters include those which represent breaker state, turbine auto stop, tripped/latched state interrogation data states, etc. Bids from the interrupt program **1,124** are registered with and queued for execution by the executive program **1,111.** The control program **1,110** also receives data from the panel task **1,112** and transmits data to status lamps and output contacts **1,128.** The panel task **1,112** receives data instruction based on supervision signals from the operator panel buttons **1,130** and transmits data to panel lamps **1,132** and to the control program **1,020.** The auxiliary synchronizer program **1,114** synchronizes through the executive program **1,111** the bidding of the control program **1,020,** the analog scan program **1,116,** a visual display task **1,134** and a flash task **1,136.** The visual display task transmits data to display windows **1,138.**

The control program **1,020** receives numerical quantities representing process variables from the analog scan program **1,116.** As already generally considered, the control program **1,020** utilizes the values of the various feedback variables including turbine speed, impulse pressure and megawatt output to calculate the position of the throttle valves TV1–TV4 and governor valves GV1–GV8 in the turbine system **10,** thereby controlling the megawatt load and the speed of the turbine **10.**

To interface the control and logic programs efficiently, the sequence of events program **1,124** normally provides for the logic task **1,110** contact status updating on demand rather than periodically. The logic task **1,110** computes all logical states according to predetermined conditions and transmits this data to the control program **1,020** where this information is utilized in determining the positioning control action for the throttle valves TV1–TV4, and the governor valves GV1–GV8. The logic task **1,110** also controls the state of various lamps and relay type contact outputs in a predetermined manner.

## E. TASK PRIORITY ASSIGNMENTS

With reference now to FIG. **9,** a table of program priority assignments is shown as employed in the executive monitor. A program with the highest priority is run first under executive control if two or more programs are ready to run. The stop/initializer program function

has top priority and is run on startup of the computer or after the computer has been shut down momentarily and is being restarted. The control program **1,020** is next in order of priority. The operator's panel program **1,130,** which generates control data, follows the control task **1,020** in priority. The analog scan program **1,116** also provides information to the control task **1,020** and operates at a level of priority below that of the operator's panel **1,130.** The automatic turbine starting (ATS) periodic program **1,140** is next in the priority list. ATS stands for automatic turbine startup and monitoring program, and is shown as a major task program **1,140** of FIG. **8** for the operation of the DEH system **1,100.** The ATS-periodic program **1,140** monitors the various temperatures, pressures, breaker states, rotational velocity, etc. during start-up and during load operation of the turbine system.

The logic task **1,110,** which generates control and operating mode data, follows in order of operating priority. The visual display task program **1,134** follows the logic task program **1,110** and makes use of outputs from the latter. A data link program for transmitting data from the DEH system to an external computer follows. An ATS-analog conversion task program **1,142** for converting the parameters provided by the ATS-periodic program **1,142** to usable computer data follows in order of priority. The flash task program **1,136** is next, and it is followed by a programmer's console program which is used for maintenance testing and initial loading of data tapes. The next program is an ATS-message writer **1,144** which provides for printout of information from the ATS analog conversion program **1,142** on a suitable typewriter **1,146.** The next program in the priority list is an analog/digital trend which monitors parameters in the turbine system **10** and prints or plots them out for operator perusal. The remaining two programs are for debugging and special applications.

In the preferred embodiment, the stop/initialize program is given the highest priority in the table of FIG. **9** because certain initializing functions must be completed before the DEH system **1,100** can run. The auxiliary synchronizer program **1,114** provides timing for all programs other than the stop/initialize program while the DEH system **1,100** is running. Therefore, the auxiliary synchronizer task program **1,400** has the second order of priority of the programs listed. The control program **1,020** follows at the third descending order of priority since the governor valves GV1 through GV8 and the throttle valves TV1 through TV4 must be controlled at all times while the DEH system **1,100** is in operation.

The operator's panel program **1,130** is given the next order of priority in order to enable an operator to exercise direct and instantaneous control of the DEH system **1,100.** The analog scan program **1,116** provides input data for the control program **1,020** and, therefore, is subordinate only to the initialize synchronizer control and operator functions.

The logic task **1,110** which control the operations of some of the functions of the control task program **1,020** is next in order of priority. The visual display task **1,134** follows in order of priority in order to provide an operator with a visual indication of the operation of the DEH program **1,100.** The visual display program **1,134** is placed in the relatively low eighth descending order of priority since the physical response of an operator is limited in speed to to 0.2 to 0.5 sec. as to a visual signal. The rest of the programs are in essentially descending order of importance in the preferred embodiment. In alternative embodiments of the inventions, alternate priority assignments can be employed for the described or similar programs, but the general priority listing described is preferred for the various reasons presented.

A series of interrupt programs interrupt the action of the computer and function outside the task priority assignments to process interrupts. One such program in FIG. **8** is the sequence events or contact interrupt program **1124** which suspends the operation of the computer for a very short period of time to process an interrupt. Between the operator panel buttons **1,130** and the panel task program **1,112** a panel interrupt program **1,156** is utilized for signalling any changes in the operator's panel buttons **1,130.** A valve interrupt program **1,158** is connected directly between the operator's panel buttons **1,130** and the panel task program **1,112** for operation during a valve test or in case of valve contingency situations.

Proportional plus reset controller subroutine **1,1154** FIG. **11** is called by the control task program **1020** of FIG. **7** as previously described when the turbine control system is in the speed mode of control and also for computer use efficiency, when the turbine **10** is in the load mode of control with the megawatt and impulse pressure feedback loops in service. Utilizing the proportional plus reset function **1068** during speed control provides very accurate control of the angular velocity of the turbine system.

In addition to previously described functions, the auxiliary synchronizer program **1,114** is connected to and triggers the ATS periodic program **1,140,** the ATS analog conversion routine **1,142** and the message writer **1,144.** The ATS program **1140** monitors a series of temperature, vibration, pressures, speed, etc. in the turbine system and also contains a routine for automatically starting the turbine system **10.** The ATS analog conversion routine **1,142** converts the digital computer signals from the ATS periodic program **1,140** to analog or digital or hybrid form which can be typed out through the message writer task **1,144** to the logging typewriter **1146** or a similar recorder.

The auxiliary synchronizer program **1,114** also controls an analog/digital trend program **1,148.** The analog digital trend program **1,148** records a set of variables in addition to the variables of the ATS periodic program **1,140.**

Ancillary to a series of other programs is a plant CCI subroutine **1,150** where CCI stands for contact closure inputs. The plant CCI subroutine **1150** responds to changes in the state of the plant contacts as transmitted over the plant wiring **1,126.** Generally, the plant contacts are monitored by the CCI subroutine **1,150** only when a change in contact state is detected. This scheme conserves computer duty cycle as compared to periodic CCI monitoring. However, other triggers including operator demand can be employed for a CCI scan.

As shown in FIG. **8,** the control task **1020** calls ancillary thereto a speed loop task **1,152** and the preset or proportional plus reset controller program **1,154.** Ancillary to the executive monitoring program **1,122** is a task error program **1160.** In conjunction with the clock program **1,120** a stop/initialize program **1,162** is used. Various other functions in FIG. **8** are described in greater detail infra.

## 2. SPEED LOOP SUBROUTINE

Making reference now to FIG. **13**, a speed loop program **1,310** which functionally is part of the arrangement shown in FIG. **7** is shown in greater detail. The speed loop (SPDLOOP) program **1,310** normally computes data required in the functioning of the speed feedback loop in the load control comprising as shown in FIG. **7** the rated speed reference **1,074**, the actual turbine speed **1,076**, the compare function **1,078**, the proportional controller **1,080** and the summing function **1,072**. The speed loop subroutine **1,310** is called upon to perform speed control loop functions by the control program **1,020**. In FIG. **13**, the functioning of the proportional controller **1,080** is shown in detail. The error output from the compare function **1,078** is fed through a deadband function **1,312**. A proportionality constant (GR1) **1,314** and a high limit function(HLF) **1,316** and included in the computation.

The speed loop (SPDLOOP) subroutine is called by the CONTROL TASK during the load control mode and when switching occurs between actual speed signals. Subroutine form reduces the requirement for memory storage space thereby reducing the computer expense required for operation of the DEH system **1,100**.

The deadband function **1,312** provides for by passing small noise variations in the speed error generated by the compare function **1,078** so as to prevent turbine speed changes which would otherwise occur. Systems without a deadband continuously respond to small variations which are random in nature resulting in undue stress in the turbine **10** and unnecessary, time and duty cycle consuming operation of the control system. A continuous hunting about the rated speed due to the gain of the system would occur without the deadband **1,312**. The speed regulation gain GR1 at **1,314** is set to yield rated megawatt output power speed correction for a predetermined turbine speed error. The high limit function HLS at **1,316**, provides for a maximum speed correction factor.

The turbine speed **1,076** is derived from three transducers. The turbine digital speed transducer arrangement is that disclosed in greater element and system implementation detail in the aforementioned Reuther Application Ser. No. 412,513. Briefly, in the preferred embodiment for determining the speed of the turbine, the system comprises three independent speed signals. These speed signals consist of a very accurate digital signal generated by special electronic circuitry from a magnetic pickup, and accurate analog signal generated by a second independent magnetic pickup, and a supervisory analog instrument signal from a third independent pickup. The DEH system compares these signals and through logical decisions selects the proper signal to use for speed control or speed compensated load control. This selection process switches the signal used by the DEH control system **1,100** from the digital channel signal to the accurate analog channel signal or vice versa under predetermined dynamic conditions. In order to hold the governor valves at a fixed position during this speed signal switching the control program **1,020** uses the speed loop subroutine **1,310** and performs a computation to maintain a bumpless speed signal transfer.

Making referenceto FIG. **14**, the speed loop (SPDLOOP) subroutine flow chart **1,310** is shown in greater detail. Two FORTRAN statements signify the operations of the speed loop subroutine program flow chart **1310**. These statements are:

CALL SPDLOOP

REF1 = REFDMD + X

Variables in the flow chart **1,310** are defined as follows:

| FORTRAN VARIABLES | ENGLISH LANGUAGE EQUIVALENT |
|---|---|
| REFMD | Load reference |
| WR | The turbine rated speed |
| REF1 | Corrected load reference |
| WS | The actual turbine speed |
| TEMP | Temporary storage location variable |
| SPDB | The speed deadband |
| GR1 | The speed regulation gain (normally set to yield rated megawatt speed correction for a 180 rpm speed error) |
| X | Speed correction factor |
| HLF | The high limit function |

## LOGIC TASK

The logic task **1,110**, as shown in FIG. **8** selects proper operating states status lamps and contacts **1,128**, control functions **1,020**, go logic, throttle pressure logic, breaker logic, interface logic, etc. in the DEH system. Referring now to FIGS. **36** and **37**, a block diagram representing the operation of the logic task **1,100** is shown. A contact input from the plant wiring **1,126** triggers the sequence of events or interrupt program **1,124** which calls upon the plant contact closure input subroutine **1,150** which in turn requests that the logic program **1,110** be executed by the settling of a flag called RUNLOGIC **1,151** in the logic program **1,100**. The logic program **1,110** is also run by the panel interrupt program **1,156** which calls upon the panel task program **1,112** to run the logic program **1,110** in response to panel button operations. The control task program **1,020** in performing its various computations and decisions will sometimes request the logic program **1,110** to run in order to update conditions in the control system. In FIG. **38**, the functioning of the logic program **1,110** is shown. FIG. **39** shows a more explicit block diagram of the logic program **1,110**.

The logic program **1,110** controls a series of tests which determine the readiness and operability of the DEH system **1,100**. One of these tests is that for the overspeed protection controller which is part of the analog backup portion of the hardwired system **1,016** shown in FIG. **6**. Generally, the logic program **1,100** is structured from a plurality of subroutines which provide the varying logic functions for other programs in the DEH program system, and the various logic subroutines are all sequentially executed each time the logic program is run.

## SELECT OPERATING MODE FUNCTION

Input demand values of speed, load, rate of change of speed, and rate of change of load are fed to the DEH control system **1,100** from various sources and transferred bumplessly from one source to another. Each of these sources has its own independent mode of operation and provides a demand or rate signal to the control program **1,020**. The control task **1,020** responds to the input demand signals and generates outputs which ulti-

mately move the throttle valves TV1 through TV4 and/or the governor valves GV1 through GV8.

With the breaker 17 open and the turbine 10 in speed control, the following modes of operation may be selected:

1. Automatic synchronizer mode — pulse type contact input for adjusting the turbine speed reference and speed demand and moving the turbine 10 to synchronizing speed and phase.
2. Automatic turbine startup program mode — provides turbine speed demand and rate.
3. Operator automatic mode — speed, demand and rate of change of speed entered from the keyboard 1,860 on the operator's panel 1,130 shown in FIG. 18.
4. Maintenance test mode — speed demand and rate of change of speed are entered by an operator from the keyboard 1,860 on the operator's control panel 1,130 of FIG. 18 while the DEH system 1,100 is being used as a simulator or trainer.
5. Manual tracking mode — the speed demand and rate of change of speed are internally computed by the DEH system 1,100 and set to track the manual analog back-up system 1,016 as shown in FIG. 6 in preparation for a bumpless transfer to the operator automatic mode of control.

With the breaker 17 closed and the turbine 10 in the level mode control, the following modes of operation may be selected:

1. Throttle pressure limiting mode — a contingent mode in which the turbine load reference is run back or decreased at a predetermined rate to a predetermined minimum value as long as a predetermined condition exists.
2. Run-back mode — a contingency mode in which the load reference is run back or decreased at a predetermined rate as long as a predetermined condition exists.
3. Automatic dispatch system mode — pulse type contact inputs are supplied from an automatic dispatch system to adjust turbine load reference and demand when the automatic dispatch system button 1,870 on the operator's panel 1,130 is depressed.
4. Operator automatic mode — the load demand and the load rate are entered from the keyboard 1,830 on the control panel 1,130 in FIG. 18.
5. Maintenance test mode — load demand and load rate are entered from the keyboard 1,860 of the control panel 1,130 in FIG. 18 while the DEH system 1,100 is being used as a simulator or trainer.
6. Manual tracking mode — the load demand and rate are internally computed by the DEH system 1,100 and set to track the manual analog back-up system 1,016 preparatory to a bumpless transfer to the operator automatic mode of control.

## SPEED/LOAD REFERENCE FUNCTION

Referring now to FIG. 62, a block diagram of the operation of the speed/load reference function is shown. The decision breaker function 1,060, of FIG. 7, is identical to the speed/load reference function 1,060, of FIG. 62. A software speed control subsystem 2,092 of FIG. 62, corresponds to the compare function 1,062, the speed reference 1,066 and the proportional plus reset controller function 1,068, of FIG. 7. The software load control subsystem 1,094, of FIG. 62, corresponds to the rated speed reference 1,074, the turbine speed 1,076, the compare function 1,078, the proportional controller 1,080, the summing function 1,972, the compare function 1,082, the proportional plus reset controller function 1,084, the multiplication function 1,086, the compare function 1,090, the impulse pressure transducer 1,088 and the proportional plus reset controller 1,092, of FIG. 7. The speed/load reference 1,060 is controlled by, depending upon the mode, and automatic synchronizer 1,080, the automatic turbine starter program 1,141, and operator automatic mode 1,082, a manual tracking mode 2,084, a simulator/trainer 2,086, an automatic dispatch 2,088, or a run-back contingency load 2,090. Each of these modes increments the speed/load reference function 1,060 at a selected rate to meet a selected demand. A typical demand/reference rate is shown in FIG. 63 drawn as a function of time.

## DESCRIPTION OF THE EMBODIMENTS OF VALVE MANAGEMENT

Referring to FIG. 8, a diagram which shows the hardware organization of the DEH system includes a portion 2008 in which is incorporated the valve management system. The basic additional hardware required for the preferred embodiment of the valve management system and are sequential analog outputs and transfer means between sequential and single valve operation for governor values referred at 1,021.

The valve management program, which is shown generally in FIGS. 23A and 23B, and FIG. 33, which is in some respects a modification of FIG. 23A includes: Valve curve selection, computing valve curve for preferred embodiment and selecting curve for alternate embodiment hereinafter described, transfer from single analog output to sequential analog output, computation of new valve flows for a transfer from single to sequential valve mode, computation of new valve flows for a transfer from sequential to single valve mode operation, computation of new valve flows for a flow demand change during a valve mode transfer, computation of the number of iterations required to complete a new valve mode change, computation of valve positions, and computation of actual flows through valves after a manaual mode change.

In accordanace with the preferred embodiment of the present invention, a steam flow demand is calculated by the DEH control system 1,100, (see FIG. 2). Data representing a flow demand versus stage coefficient as shown in FIG. 21 is contained in computer memory based on the flow demand computed by the DEH control system. The flow value is shown on the abscissa and the stage flow coefficient is calculated along the ordinate. The stage flow coefficient is the ratio of actual flow at a flow demand over the theoretical flow if the orifice coefficient were equal to one. As the valve flow increases a range of critical flow is passed through. The resultant super critical flow exists in a range where the orifice coefficient is decreased sharply. Once the ordinate for a particular flow demand is calculated by use of the data in the computer 1,100, the stage coefficient is calculated, which is used to calculate control valve positions. It should be noted that the stage flow coefficient is corrected such that the maximum flow coefficient of the orifice is normalized at 1 for simplicity of calculation.

In FIG. 22 the flow demand is represented as a percentage of total flow on the abscissa and lift of the control or governor valve is shown on the ordinate whereby the lift of the valves for a specified flow demand can be calculated. In FIG. 21, curve **3,010** is shown which represents a dynamic characteristic of operation of a single control valve from its closed position to its fully open position at approximately 64% of total stream flow. The corrected stage flow coefficient for critical flow is essentially equal to one for the illustrated embodiment in FIG. 21 for flow demands of less than 64 percent total flow. The flow demand representing the transition from critical to super critical flow is determined by the design of the valves. In the example used in the present embodiment the transition occurs at 64% of total flow. However, such transition point is other than 64%. If the flow demand is greater than that having a corrected stage flow coefficient of one, a new curve, for example **3012** is calculated, because the curve for critical flow can only be used for flows up to 64%. The curve **3,010** which represents a corrected stage flow coefficient of one is composed of five linear segments in order to facilitate ease of calculation and economy of memory space. In order to calculate a curve with a corrected stage flow coefficient less than one, the abscissas of the curve **3,010** are multiplied by the corrected stage flow coefficient of FIG. **21.** Similarly, the ordinates of the curve **3,010** are corrected, and then multiplied by the corrected stage flow coefficient from FIG. **21.** The ordinates of the curve **3,010** are corrected by subtracting the ordinate intersection point **3,014** from the points **3,016, 3018,** and **3,022.**

Therefore, as the corrected stage coefficient varies due to changes in flow demand, a new dynamic curve is generated. The accuracy of representation and precision of operation is only limited by the resolution of the computer and the data representing the valve characteristics. Any desired degree of accuracy in developing dynamic curves can thus be achieved by increasing the resolution of the computer. In practice, five data points for each curve in FIG. **21** and **22** have been sufficient to give an accuracy of better than 2% between flow demand and actual flow.

In an alternative embodiment of the invention, the valve management program selects one of a series of curves represented by data of valve flow demand versus control valve lift stored in the computer memory. a selection of a proper curve is predicated on the total flow through the turbine **10.** Thereafter, the relationship between the total flow, the valve flow demand and the control valve lift is not affected by the mode of operation of the turbine **10.** The data representing valve flow versus control valve lift is displayed in FIG. **22.**

The transfer between single valve and sequential valve modes, and other described features in the operation of the control or governor valves GV1 through GV8 is accomplished in this embodiment by the selection of the data corresponding to an appropriate valve flow versus lift curve. As in the preferred embodiment, where the valve flow curve is computed, first a target flow or desired flow through each control or governor valve GV1 through GV8 is computed for the mode to be transferred into. Second, the flow changes, the differences between the initial flow and the desired flow to each valve are computed. Next, a number of iterations is determined by dividing the largest flow change of any of the governor valves GV1 through GV8 by a predetermined maximum allowable change of flow through a valve thereby determining a number of iterations for the transfer of mode. Flow changes for each of the control or governor valves GV1 through GV8 already computed are then divided by the number of iterations required for the mode transfer. Each iterative step does not affect the total flow of fluid through the turbine **10** since the sum of the incremental flow changes is equal to zero.

During sequential valve or partial arc operation, one valve is usually partially opened and the other valves are usually either fully opened or fully closed. Since the stage flow coefficient is dependent upon flow demand or total flow, the number of valves and their positions contributing to the flow do not affect the calculation as performed on th appropriate curve of FIG. **22.** The stage flow coefficient is the same if all the control valves are partially open, if some of the control valves are fully open, or if some are closed, and one control valve partially open. The fully open control valves contribute the percent of total flow as shown by their end point or corrected end point **2,024** or **2,026** (see FIG. **22**). The partially open control valve makes up the remainder of the total flow demand in accordance with a function whereby the percent of total flow demand for the partially opened valve is entered in the abscissa in FIG. **22** and the actuator lift is shown on the ordinate.

Thus, the valve management program dynamically calculates data which represents control valve demand or flow as a function of the valve lift of a control valve while compensating for the pressure variation and the corrected first stage flow coefficent. The calculation of a dynamic flow demand versus lift characteristic is dependent upon the total flow of fluid through the turbine. The stage flow coefficient is constant regardless of the mode operations of the turbine whether it be single valve or sequential valve. In addition, as hereinafter described the valve demand versus lift characteristic data is modified dynamically for variations in the throttle pressure and also for the variation in the number of nozzles under each valve.

Referring again to FIG. **22** from point **3,020** to point **3024,** on curve **3010,** a very high associated gain is required in order to maintain and linearize any action of the actuator for the control valves in this region.

## SEQUENTIAL VALVE MODE

In the sequential mode the control valves open in succession thereby allowing all but one of the opened valves to operate fully open, and thus have a minimum throttling loss. The valve which is not opened completely is the only valve which generally controls the flow of steam througoh the turbine. Because of the areas at the beginning and the end of a valve stroke where control is very poor methods of overlap or what may be termed asymmetric hysteresis have been developed which avoids these areas in control. During the use of asymmetric hysteresis, two valves are partially opened; however, only one is controlling the flow. Therefore the problem, which could occur when the control valves operate in sequential mode is prevented. When a valve controls flow in a saturation zone, a small flow change requires a considerable change in valve position. The present embodiment of the valve management program is arranged to avoid flow in the saturation zone by use of the valve overlap or the so-called

asymmetric hysteresis approach whereby the associated high gain requirement and associated danger of instability is avoided.

Referring now to FIG. 32 it is shown that the governor or control valves are not controlled in the shaded OA and CD regions. A very large stroke change in these regions produces a very small flow change, which therefore requires the computer to have a very high equivalent gain. As is well known in control art a high gain can cause poles in the right-hand half complex plane of the transfer function to migrate into the right-hand half plane and therefore produce instability. The regions of steep slope CD and OA at the end of the travel and at the opening of each valve, respectively are avoided, in sequential control of the control valves. Valves I and I + 1, are illustrated in FIG.32 as sequentially operable valves. For example, during an increase in flow demand $F_T$, stops momentarily at point **2,028** as shown in portion A of FIG. **32** during its opening; stroke and closes to point **2,030** at the same time as the next sequential, valve I + 1, opens to point **3,034.** Valve I and valve I + 1 are moving in opposite directions at such rates that the total flow $F_T$ contributed by the two valves I and I + 1 at points **2,030** and **3,034** respectively would be equal to the flow contributed by valve I in the shaded region. When valves I and I + 1 are moving in opposite directions the forward loop gain associated with the movement is essentially zero and therefore very stable.

The action of the valves during the opening and closing sequence is different thereby generating the hysteresis action. For example, if valve I has just opened fully, as illustrated in portion E of FIG. **32** a decrease in flow demand $F_T$ does not cause a movement of valve I until valve I + 1 has closed considerably, such as shown in portion D of FIG. **32.** Thus, the sequence where a varying flow demand requires the full, opening of value I, and then a small decrease in flow demand signals requires its closing through the upper portion of its stroke where the control is poor and associated gain very high and an oscillation may result, is avoid. Any system is prone to oscillation if small input signals produce large changes in output quantities.

If the control of the turbine system is transferred between valves I and I + 1 because of noise in the demand signal, for example, a rapid transfer may not be able to be effected because of the frequency response of the DEH system **1,300.**

Also, when the point in the operating characteristic of flow demand versus actuator lift represented by valve I + 1 closes to a point represented by **3,034** as shown in FIG. **32** before valve I + 1 closes completely in the closing sequence, valve I closes to a point represented by **2,028,** and valve I + 1 opens to a point such as **3,023** shown in FIG. **32.** Thereby, a transition through the area of high slope at the top of the valve stroke is avoided for a series of small changes in the flow demand.

Specifically, in the preferred embodiment, as shown in portion B of FIG. **32** the flow at the point **3,023** of valve I + 1 is assumed to be at least twice the flow as that at a point **3,034,** which point is equal to the flow in the uncontrollable range of the valve I + 1; and the flow at point **3,023** is greater than the flow contributed by the shaded portion of the valve I. As the flow increases, from that shown in portion A to portion B of FIG. **32** valve I opens again to point **2,028** instead of opening fully to its maximum valve stroke and valve I + 1 opens to the point such as **3,023** so that its flow is greater than that represented by **3,034.** During a decreasing flow with valve I being fully opened and valve I + 1 being partially open as illustrated in portion E of FIG. **32,** the flow in valve I + 1 decreases to the point **3,034** (see portion B of FIG. **32**) whereupon valve I closes to point **2,028** and valve I + 1 opens to point **3,023** to compensate for the decrease of flow through valve I. Upon a further decrease of flow as shown in portion D of FIG. **32** such that valve I + 1 decreases its flow again to the point **3,034**; and then as shown in portion F of FIG. **32** valve I + 1 then closes completely, and valve I moves to a point which is less than **2,028.** By the above method changes in flow demand caused by noise fluctuations in the signals within the system will not cause a repeated opening and closing operation to occur around a particular flow demand. Once the flow has been switched from one valve to the other it will not be returned to the initial valve until a flow change greater than the shaded areas has occurred. In the present invention because of the control valves GV1–GV8 characteristics, a hysteresis or deadband which is equal to twice the flow of valve I + 1 at point **3,034** is utilized.

Reference is made to FIG. **32** for other examples of the sequential operation of the control value as the target flow $F_T$ increase and decreases.

## PRESSURE CORRECTION

Referring now to FIG. **23A**, which together with FIG. **23B** is a flow chart of the valve management system in general. A pressure correction factor calculation referred to as block **3,010** corrects for any changes in the temperature and pressure of the incoming steam. The pressure correction calculations also include a deadband calculation which provide a safety measure; and also acts as a filter for noise. Without a deadband, small changes in the data due to noise typically from transducers, etc. would change the position of the valve and thereby cause unnecessary roughness in the operation of the turbine system and the output power therefrom. In addition, the governor or control valves would wear at a faster rate thereby requiring earlier maintenance.

In addition, the pressure correction factor calculations **3,010** include a limitation of rate of change. Therefore, should the throttle pressure or any other pressure, which is being utilized, change very rapidly, the correction for such a pressure change would be limited to a predetermined maximum rate of change. If the throttle flow correction transducer, not shown, and referred to in decision block TPXD OK is inoperative, the throttle pressure correction may be completely ignored and a normal throttle pressure correction factor of one as shown in block **210** of FIG. **34** is used. Under certain conditions of operations, stability is increased by the use of a throttle pressure correction factor of one. Under normal operation, however, the throttle correction factor forms a way of reducing errors in the flow demand signal through the turbine system. When the throttle pressure transducer is in service, a correction factor is computed as shown by the blocks associated with the appropriate descriptive legend of FIG. **27** or FIG. **34.** The deviation in throttle pressure PDEVA which is considered in each iteration is limited to a maximum (MXPDEV) in order to prevent sudden changes in the valve positions through calculations and

large changes along the control or governor valve curve data. As an alternative, if the throttle pressure transducer is out of service, the pressure correction factor PCORF, is kept at its last computed value POLAST as shown in FIG. **27.** When the throttle pressure transducer is put back in service, the signals therefrom are only used if one of the feedback loops is in service, which provides the necessary computation for the effect of valve curve correction. FIGS. **35** and **43B** are referenced for a more detailed understanding.

In computing desired valve flows in the single valve or full arc admission mode, the target flows are computed by dividing the total flow demand by the number of control or governor valves GV**1** through GV**8.** The valves are positioned according to individual valve flow demand. In the sequential valve mode, the flow demand is divided by the maximum flow of each control or governor valves GV**1**–GV**8** for the total flow demand whereby a whole number and a fraction result. The whole number represents the number of valves fully open in sequential valve operation; and the left-over fraction determines the flow demand of the valve which is partially open and controlling the flow of the fluid through the turbine.

Reference is made to FIGS. **24** and **36** a program for the transfer of the contents of the common analog outputs to the individual analog outputs, FIG. **5,** which shows the electronic circuitry and system of the analog outputs. Transfer between the single valve mode and the sequential valve mode is made from the common analog output system **520,** as shown in FIG. **5** to the individual analog outputs such as **521***a*, **521***b*, and **521***c*. When all the valves work together as in the single valve mode, a common analog output regulates the position of all of the governor or control valves GV**1**–GV**8.** Before a transfer can be initiated to sequential valve operation, the individual analog outputs **521***a*, **521***b*, and **521***c*, connected to each respective governor or control valves GV**1**-GV**8** are adjusted to a valve equivalent to the value in the common analog output **520.** After the transfer of the contents of the common analog output **520** to the individual analog outputs **521***a*, **521***b*, and **521***c*, the valve management program is ready to initiate a transfer to the sequential valve mode. The mechanics of the transfer between the common analog output and the individual analog output is included in detail in FIG. **5** but forms no part of the present invention and in the Braytenbah application cited supra. The subroutine as shown in FIG. **24** or for the transfer of contents of the common analog outputs to the individual analog outputs first checks whether there is any data in the common analog outputs **520** other than a pre-set bias value. If there is data in the common analog outputs **520,** the value therein is transferred from the common analog output to the individual analog outputs in a predetermined number of steps (MDIV) without any change in the total analog output settings for the control valves. The subroutine for the transfer of the contents checks the contents of the individual analog output to assure that the maximum value is equal to the maximum allowed for the digital to analog converters described in detail in U.S. Pat. 3,741,246 Ser. No. 80,170, W.E. 42,839.

Referring now to FIG. **25,** a subroutine for calculation of the flow coefficient function includes the mathematical representation of the stage flow coefficient versus percentage flow graph shown in FIG. **21.**

Reference is made to the subroutine for the calculation of the functions of the valve curves used in the governor or control valve GV**1**–GV**8** movements shown in FIG. **26,** which includes the mathematical representation for the calculation of the data represented by the valve lift versus total flow demand family of curves shown in FIG. **22.** This valve curve GV function generator subroutine generates data representing the characteristics of the control or governor valves GV**1**–GV**8** as functions of the total flow demand.

In FIG. **27** the flow coefficient function generator of FIG. **25** and the valve curve function generator subroutine of FIG. **26** are incorporated in a valve curve selection subroutine. The valve curve selection subroutine of FIG. **26 67-9** checks the throttle pressure and compensates for any changes above or below a standard pressure. In addition, the selection subroutine selects the proper flow coefficients from the flow coefficient subroutine of FIG. **25 67-7** and takes the interpolation of the valve curve function generator subroutine of FIG. **26** and computes the valve positions taking into account the actual conditions in the steam turbine system whereby a total flow can be determined.

## FLOW CHANGES

Referring to FIG. **23B,** a flow change logic calculation referred to as block **3,012** insures that any changes in flow demand are executed even during a mode transfer. Therefore, if the DEH System **1,100** request a change in flow demand during a mode transfer, the mode transfer is interrupted momentarily and the valves are changed in accordance with the flow change.

Referring to FIGS. **37** and **38,** the subroutine for computation of target flow changes in the sequential mode is entered into when a mode change from single valve operation to sequential valve operation is initiated, or when a flow change is requested while operating in the sequential valve mode; and includes valve overlap or hysteresis heretofore described. The points $F_c$ and $F_a$ are those referred to in FIG. **32** where $F_a$ is the lower limit of the control zone and $F_c$ is the upper limit of the control zone of the governor or control valve. Both $F_a$ and $F_c$ may be expressed as "per unit" or other convenient valves such as percentage of maximum flow.

In the computation of a flow change during a mode change, there are several paths which can be taken. A total flow demand which is less than the $F_c$ point as indicated in FIG. **32** of the first sequence referred to as valve I, in which case the total flow is divided equally among the valves I of the first sequence (see **330** of FIGS. **37** and **38**). An alternative would be that the total flow demand being greater than that at the $F_c$ of the current sequence plus $F_a$ point of the next sequence referred to as valve I + 1 in FIG. **32** in which case the target flow on the current sequence is fixed at the $F_c$ point, and the valve of the next sequence becomes the controlling valve (see **370** of FIG. **38**). The third possibility would be that the total flow demand is less than the $F_c$ point of the current sequence plus the $F_c$ point of the next sequence. In this case the total flow on this sequence is fixed at the $F_a$ point and the valve of the current sequence remain the controlling valve (see **380** of FIGS. **37** and **38**). Yet another path would be when the flow demand is greater than the maximum flow of the current sequence plus the $F_a$ point of the next sequence, then the target flow of the current sequence is

equal to the maximum flow and the next sequence becomes the controlling sequence.

Referring now to FIGS. **28** one of the basic requirements of the valve management program is seen, that is, to allow the response to a flow change during a mode change. In a mode change the control valves are in the middle of a transfer. The flow change demand is divided among the valves so that the steam flow through the turbine is held relatively constant. The only changes in the flow will be caused by tolerance errors in the data generating the curves. The flow change is divided among the valves with the restriction that none of the valves should be positioned in the overlapping or non-control zones. When one of the valves does not meet this requirement, the flow change on that valve is reduced and the difference is added to the flow change computed for the next valve and so on. etc. If a valve is in this condition the program jumps to .**320** in FIG. **28** and all the target flows are recomputed whereupon the change in flow is implemented.

The change in flow through the governor or control valves is implemented in the control valve or valves which are currently in control in the sequential mode. A program sequence as shown in FIG. **37** is followed which provides for the following conditions to be met, i.e., the control valves are not in an overlap mode HYST = FALSE, or the flow change DELFT will not cause the valves to go into an overlap mode. If one of these conditions are not met, the program paths to be executed are either **330, 370,** or **380** as shown in FIGS. **37** and **38,** which are those used during a mode change as previously described. Referring to FIG. **39,** a subroutine for computation of target flows for a transfer from sequential to single mode is shown. This routine has two purposes, i.e. to compute the new target flows FTGT for the transfer from the sequential to the single valve mode; and to compute the actual valve position for a flow change after the transfer to the single valve mode has been completed. The computation of the target flows FTGT for a mode change is done by dividing the total flow demand FDEM by the number of control valves FNOVLV. If a flow change is demanded when the valves are already in the single mode, the target flow becomes the actual flow since all the valves move together and there is no problem with either overlap or hysteresis. The valve position is computed using the function generator program of FIG. **26** for a selected curve thereby producing a new actual flow for each and every valve. The new position VPOZD is output to the common analog output GVAO(**1**) after a necessary conversion of units and scale.

FIG. **29** shows the calculation for determining and executing a number of incremental changes in flow and associated valve position for each governor valve GV during a mode change. In a mode change between single and and sequential valve operation, the flow of steam through the turbine system should preferably remain essentially constant. In this subroutine the total flow change for each control valve is calculated, and then the greatest flow change FTEMP1 for any of the governor valves is determined. The maximum flow change FTEMP1 thus determined is then divided by a predetermined maximum allowable flow change MXFPCH which then determines the number of incremental flow changes NOCHGS required for a mode transfer. The flow change DELF for each governor valve is divided by the determined number of incremental flow changes FNOCH. Thereupon, during a mode change, each governor valve GV moves through an equal number of incremental flow change steps. This subroutine for the selection of the incremental flow change steps for each governor valve is also utilized during a change from the manual or emergency mode, where the governor valves GV**1**–GV**8** may be in any random series of locations, to either the single valve mode or the sequential valve mode. It is within the teaching of this invention that the subroutine can be used within any mode changes. By using this method any flow variations which may occur during a mode change are given priority. This method for computation of flow changes during a mode change also insures compliance with the requirements that no flow change may occur as a result of the mode change.

The incremental flow changes DELF calculated in FIG. **29** are translated into governor valve positions in the subroutine as shown in FIG. **30.**

Where a number of valves operate in a single mode or the sequential mode in low load ranges, there is a possibility that the governor or control valves opening fully at the same time could cause a relatively large incremental change, in load. In order to minimize any shock that would occur the valve management program has logic operations which delay the command for full opening of each valve as shown in FIG. **41.** The time delays for each of the governor or control valves are different therefore staggering their full opening and preventing a simultaneous full opening. In FIGS. **34** and **35,** the value curve selection program which is initiated every time of a mode change or a flow change is indicated by the respective function (MODCH is equal to .TRUE.) Or (FLOWCH equals .TRUE.). The (see FIG. **23A**) valve curve selection subroutine **200** performs the function of correcting the data representing the valve curves shown in FIG. **22** to conform with changes in throttle pressure, and selecting a valve curve in accordance with an alternate embodiment based on the total flow demand through the turbine and computing the maximum flow for each valve based on the selected curve and corrected for the number of nozzles. A total flow demand FDEM is compared with (FMAXPC and FMINPC) as shown in FIG. **34** for selecting the new curve (see **221** of FIG. **35**).

If the total flow demand exceeds the validity limits of the presently selected valve curve, (FMAXPC and FMINPC), a new curve is selected by the valve curve selection subroutine. Based on the selection of the new data representing the valve curve the following are calculated for each value, the maximum flow (SVMX), the first and last point for selecting the present curve in the flow array (FPTCSL and LPTCSL), the correction for the number of nozzles NONOZ, and the correction for the throttle pressure setting PCORF. A subroutine for computation of valve positions is shown in consolidated form of FIG. **30** and in separate modified form in FIGS. **40, 41,** and **42.** The actual flow of steam through each control or governor valve and the corresponding valve position or lift is herein computed. If during the manual operation of the turbine, the control or governor valves have been left in a variety of seemingly random positions, the subroutine for the computation of valve positions insures that a reinitialization of the mode change subroutines will be completed. Also, as during the case of a flow change during a mode change between either the manual, single valve or se-

quential valve modes, the initialing computed values of valve position or lift become invalid. Therefore, new valve positions must be computed. Initially, the new actual flow is computed by adding the incremental flow change to the old or initial actual flow. The new valve positions are computed either selecting or computing a proper curve through the function generator subprogram as previously described. A correction fact for variation in the number of nozzles under each valve is then introduced. For flows under a preselected value of maximum flow per valve, the correction factor is set to one. For flows above this ratio the nozzles are considered to be in critical flow; and another ratio is computed between the typical number of nozzles and the actual number of nozzles for the respective control valves. The actual flow is computed by adding the actual flow through each valve with the initial flow. The new output valus thus obtained is outputted to the new position data in an analog output table where the new position of the valve is determined. In addition, stagger paths are included in order to insure that only not one valve will open fully during any one iteration. During the iterations following a mode change a search is made for the next valve to be fully opened and the analog output data corresponding to a fully open position. The stagger path is executed as many times as is required in order to open all the required valves in a particular sequence.

After a transfer from sequential to single valve mode the data contained in the sequential analog output are transferred to the common analog output which is the opposite sequence of what occurs during a transfer from the single valve mode to the sequential valve mode.

In FIG. **31**, a subroutine for computing actual flow valves after a manual or emergency condition is shown. During a manual or emergency condition the governor valves GV1–GV8 may be left in randon positions by an operator and therefore require repositioning. The subroutine of FIG. **31** calculates the desired valve positions dependent upon the flow demand. Just as in the case of the other mode changes, a maximum flow change is calculated for each valve, then this maximum flow change is divided by a maximum allowable flow change, thereby obtaining the number of iterations required for a change from the manual or emergency conditions to one of the automatic operating modes. As in the other mode changes supra any change in flow demand will take precedence over a mode change.

Referring now to FIGS. **43A** and **43B** which show the principle of the tracking scheme in connection with the actual curve selection of the alternate embodiment to make the reference demand speed equal to the actual speed. This equalization is accomplished by back calculating the controller output and all converted values on the basis of a back calculation of the flow demand made by the value management program from the actual valve positions as shown in FIG. **43B**. The tracking function is accomplished by this back calculation which is made to calculated the flow demand if the program is on governor valve control. The valve management program using the manual turbine logic restricts itself to this back calculation. In either the preferred embodiment where the flow vs. valve curves are calculated instead of selected as previously described; or in the embodiment where the valve curve is selected the slightly modified program of FIG. **31** and the programs of FIGS. **43A** and **43B** can be utilized.

In order to reduce the number of cycles of computer time required by the valve management program to computer or select the proper valve curve of the alternate embodiment, the control program gives the valve management program a total flow demand equal to the actual megawatt demand over the minimum number of megawatts capable of being produced. During governor control valve operation of the turbine system, the ratio of the actual megawatt to the maximum megawatts is utilized. In throttle valve operation, however, the flow demand is set equal to the throttle valve analog output signal. The analog output signal for the throttle valve operation as well as the analog output signals for the governor valve operation have been set equal to the manual counterparts by the control programs.

The following is a list of definitions of the symbols used in the various flow charts which are provided to give a detailed understanding of the steps of the programs referred to herein. The reference numerals associated with the various paths and blocks of the flow charts are included in such flow charts as well as legends so that the various functions can be followed within the chart or from one chart to another.

DEH COMMON DICTIONARY

| NAME | TYPE | CORE LOC. | KEYBOARD ADR. | FUNCTION |
|---|---|---|---|---|
| | | KAPPA VALVE MANAGEMENT VARIABLES | | |
| SING | L | 1450 | 1721 | SINGLE VALVE OPERATION |
| SINGX | L | 1451 | 1722 | PREVIOUS STATE OF SING |
| SEQ | L | 1452 | 1723 | SEQUENTIAL VALVE OPERATION |
| SEQX | L | 1453 | 1724 | PREVIOUS STATE OF SEQ |
| MEDCH | L | 1454 | 1725 | MODE CHANGE |
| MODCHX | L | 1455 | 1726 | PREVIOUS STATE OF MODCH |
| TRFPG | L | 1456 | 1727 | TRANSFER IN PROGRESS |
| TRFPGX | L | 1457 | 1728 | PREVIOUS STATE OF TRFPS |
| TPXDBK | L | 1458 | 1729 | PRESSURE TRANSDUCER O.K. |
| LPCORR | L | 1459 | 1730 | LOGICAL VALUE OF FLPCORR |
| VCHDR | L | 145A | 1731 | FLAG INDICATES AUTO TRACKS MANUAL |
| SINGLV | L | 145B | 1732 | SINGLE VALVE MODE |
| VT | L | 145C | 1733 | VALVE TEST |
| CRVSELOK | L | 145D | 1734 | CORRECT CURVE SELECTED IN TRACKING |
| TEMPLOG | L | 145E | 1735 | TRACKING COMPLETED |
| HISLOPE | L | 145F | 1736 | FLAGS ALL VALVES ARE FULLY OPEN |
| HYST | L | 1460 | 1737 | FLAGS CURRENT SEQUENCE IS IN OVERLAP STATE |
| NUCLININ | L* | 1461 | 1738 | NUCLEAR IN LINE |
| SINGEND | L* | 1462 | 1739 | SINGLE ENDED STEAM CHEST |
| STOPVLVS | L* | 1463 | 1740 | STOP VALVES |

DEH COMMON DICTIONARY—Continued

| NAME | TYPE | CORE LOC. | KEYBOARD ADR. | FUNCTION |
|---|---|---|---|---|
| | | | | KAPPA VALVE MANAGEMENT VARIABLES |
| VTTRACK | L | 1464 | 1741 | VALVE TEST TRACK |
| REFAO | L* | 1465 | 1742 | REFERENCE ANALOG OUTPUT OPTION |
| FRGAO | L* | 1466 | 1743 | FREQUENCY BIAS ANALOG OUTPUT OPTION |
| FIXTPSP | L* | 1467 | 1744 | FIXED THROTTLE PRESSURE SETPOINT |
| | L | 1468 | 1745 | SPARE |
| ICRSLN | I | 1469 | 2746 | NO. OF ITERATIONS PRIOR TO TRACKING COMPLETE |
| NSLCRV | I | 146A | 2747 | NO. OF SELECTED CURVES |
| IGVAO(1) | I | 146B | 2748 | GV COMMON A/O |
| IGVAO(2) | I | 146C | 2749 | GV1 SEQ A/O |
| IGVAO(3) | I | 146D | 2750 | GV2 SEQ A/O |
| IGVAO(4) | I | 146E | 2751 | GV3 SEQ A/O |
| IGVAO(5) | I | 146F | 2752 | GV4 SEQ A/O |
| IGVAO(6) | I | 1470 | 2753 | GV5 SEQ A/O |
| IGVAO(7) | I | 1471 | 2754 | GV6 SEQ A/O |
| IGVAO(8) | I | 1472 | 2755 | GV7 SEQ A/O |
| IGVAO(9) | I | 1473 | 2756 | GV8 SEQ A/O |
| IGVAO(10) | I | 1474 | 2757 | IV COMMON A/O |
| NOVLV | I* | 1475 | 2758 | NO. OF GV's |
| NTV | I* | 1476 | 2759 | NO. OF TV's |
| NDIVS | I* | 1477 | 2760 | MAX. DECREMENT FROM SEQ. A/O TO COMMON A/O |
| NDIV | I* | 1478 | 2761 | MAX. DECREMENT FROM COMMON A/O TO SEQ. A/O |
| ICOMLS(1) | I* | 1479 | 2762 | GV OPENING 1ST |
| ICOMLS(2) | I* | 147A | 2763 | GV OPENING 2ND |
| ICOMLS(3) | I* | 147B | 2764 | GV OPENING 3RD |
| ICOMLS(4) | I* | 147C | 2765 | GV OPENING 4TH |
| ICOMLS(5) | I* | 147D | 2766 | GV OPENING 5TH |
| ICOMLS(6) | I* | 147E | 2767 | GV OPENING 6TH |
| ICOMLS(7) | I* | 147F | 2768 | GV OPENING 7TH |
| ICOMLS(8) | I* | 1480 | 2769 | GV OPENING 8TH |
| NOPPCV | I* | 1481 | 2770 | NO. OF POINTS PER GOV. VALVE CURVE |
| FPTCSL | I* | 1482 | 2771 | FIRST POINT OF SELECTED CURVE |
| LPTCSL | I* | 1483 | 2772 | LAST POINT OF SELECTED CURVE |
| NOSEQ | I* | 1484 | 2773 | NO. OF SEQUENCES |
| SEGT(1) | I* | 1485 | 2774 | NO. OF VALVES IN SEQ. 1 |
| SEQT(2) | I* | 1486 | 2775 | NO. OF VALVES IN SEQ. 2 |
| SEQT(3) | I* | 1487 | 2776 | NO. OF VALVES IN SEQ. 3 |
| SEQT(4) | I* | 1488 | 2777 | NO. OF VALVES IN SEQ. 4 |
| SEQT(5) | I* | 1489 | 2778 | NO. OF VALVES IN SEQ. 5 |
| SEQT(6) | I* | 148A | 2779 | NO. OF VALVES IN SEQ. 6 |
| ICOTLR | I* | 148B | 2780 | TOLERANCE FOR TRANSFER SINGLE TO SEQ. A/O |
| NPSTRT | I* | 148C | 2781 | NO. OF FIRST POINT ON VLV CURVE FOR FLCOEF COR |
| NSFCFP | I* | 148D | 2782 | NO. OF POINTS FOR FLOW COEF. CURVE |
| INTRIES | I* | 148E | 2783 | NO. OF ITERATIONS PER TRACKING |
| ITYPTEST | I* | 148F | 2784 | TYPE OF VALVETEST |
| ITESTPAT(1) | I* | 1490 | 2785 | BIT ON CCO WORD FOR TV1 OR SV1 |
| ITESTPAT(2) | I* | 1491 | 2786 | BIT ON CCO WORD FOR TV2 OR SV2 |
| ITESTPAT(3) | I* | 1492 | 2787 | BIT ON CCO WORD FOR TV3 OR SV3 |
| ITESTPAT(4) | I* | 1493 | 2788 | BIT ON CCO WORD FOR TV4 OR SV4 |
| ITESTPAT(5) | I* | 1494 | 2789 | BIT ON CCO WORD FOR GV1 |
| ITESTPAT(6) | I* | 1495 | 2790 | BIT ON CCO WORD FOR GV2 |
| ITESTPAT(7) | I* | 1496 | 2791 | BIT ON CCO WORD FOR GV3 |
| ITESTPAT(8) | I* | 1497 | 2792 | BIT ON CCO WORD FOR GV4 |
| ITESTPAT(9) | I* | 1498 | 2793 | BIT ON CCO WORD FOR GV5 OR BV1 |
| ITESTPAT(10) | I* | 1499 | 2794 | BIT ON CCO WORD GV6 OR BV2 |
| ITESTPAT(11) | I* | 149A | 2795 | BIT ON CCO WORD FOR GV7 OR BV3 |
| ITESTPAT(12) | I* | 149B | 2796 | BIT ON CCO WORD FOR GV8 |
| ITESTPAT(13) | I* | 149C | 2797 | BIT ON CCO WORD FOR GV1 + GV3 + GV5 + GV7 |
| ITESTPAT(14) | I* | 149D | 2798 | BIT ON CCO WORD FOR GV2 + GV4 + GV6 + GV8 |
| ITVDB | I* | 149E | 2799 | THROTTLE VALVE DEADBAND FOR CONTINGENCY CHECK |
| IGVDB | I* | 149F | 2800 | GOVERNOR VALVE DEADBAND FOR CONTINGENCY CHECK |
| NOCONV | I* | 14A0 | 2801 | NO. OF A/I BIT PATTERNS CONVERTED TO ENG. UNITS |
| SPREF | I* | 14A1 | 2802 | VARIABLE USED IN THE SPEED CHAIN INT NO. 2 PROGRAM |
| SPREF2 | I* | 14A2 | 2803 | RESET VALUE FOR RK REGISTER CARD |
| NWR | I* | 14A3 | 2804 | =8913(WR=3600); =8533(WR=1800) |
| IGVDB1 | I* | 14A4 | 2805 | GOV VALVE DEADBAND FOR CONTINGENCY CHECK |
| | I | 14A5 | 2806 | SPARE |
| | I | 14A6 | 2807 | SPARE |
| | I | 14A7 | 2808 | SPARE |
| PDBND | R* | 14A8 | 3405 | PRESSURE DEADBAND |
| MXPDEV | R* | 14AA | 3406 | MAX-THROTT-PRESS-CORRECTION PER ITERATION |
| FTOLRM | R* | 14AC | 3407 | FLOWCHANGE DEADBAND DURING MODE CHANGE |
| FTOLRF | R* | 14AE | 3408 | FLOWCHANGE DEADBAND IF NO-MODE CHANGE |
| MXFPCH | R* | 14B0 | 3409 | MAX FLOW CHANGE PER ITERATION |
| FC | R* | 14B2 | 3410 | UPPER BREAK POINT OF VALVE CURVE |
| FA | R* | 14B4 | 3411 | LOWER BREAK POINT OF VALVE CURVE |
| FDEM | R | 14B6 | 4412 | FLOW DEMAND |
| FASUM | R | 14B8 | 4413 | FLOW ASSUMED, STARTING VALUE IN TM |
| FRRMIN | R | 14BA | 4414 | MINIMUM ERROR |
| PDBNDL | R* | 14BC | 3415 | PRESSURE DEADBAND FOR PRESSURE CORRECTION LOW |
| PDBNDR | R* | 14BE | 3416 | PRESSURE DEADBAND FOR PRESSURE CORRECTION HIGH |
| POLAST | R | 14C0 | 4417 | LAST VALUE OF THROTTLE PRESSURE |
| XSP | R* | 14C2 | 3418 | =1. (WR=3600); =2.(WR=1800) |
| TNONSZ | R* | 14C4 | 3419 | TYP. NO OF NOZZLES PER GV |
| FNOVLV | R* | 14C6 | 3420 | NO. OF GOV VALVES |
| FTOLR | R | 14C8 | 3421 | DEADBAND FOR FLOW CHANGES |
| VMAX | R* | 14CA | 3422 | MAXIMUM VALVE POSITION |
| FX | R* | 14CC | 3423 | CORRECT CURVES FOR INEQUAL NO. OF NOZZLES |
| CONV | R* | 14CE | 3424 | CONVERSION FACTOR BINARY A/O TO INCHES |

DEH COMMON DICTIONARY – Continued

| NAME | TYPE | CORE LOC. | KEYBOARD ADR. | FUNCTION |
|---|---|---|---|---|
| | | | | KAPPA VALVE MANAGEMENT VARIABLES |
| FL(1) | R | 14D0 | 3425 | FLOW TABLE |
| FL(2) | R | 14D2 | 3426 | FLOW TABLE |
| FL(3) | R | 14D4 | 3427 | FLOW TABLE |
| FL(4) | R | 14D6 | 3428 | FLOW TABLE |
| FL(5) | R | 14D8 | 3429 | FLOW TABLE |
| FL(6) | R | 14DA | 3430 | FLOW TABLE |
| FL(7) | R | 14DC | 3431 | FLOW TABLE |
| PZ(1) | R* | 14DE | 3432 | POSITION TABLE |
| PZ(2) | R* | 14E0 | 3433 | POSITION TABLE |
| PZ(3) | R* | 14E2 | 3434 | POSITION TABLE |
| PZ(4) | R* | 14E4 | 3435 | POSITION TABLE |
| PZ(5) | R* | 14E6 | 3436 | POSITION TABLE |
| PZ(6) | R* | 14E8 | 3437 | POSITION TABLE |
| PZ(7) | R* | 14EA | 3438 | POSITION TABLE |
| PCORF | R | 14EC | 3439 | THROTTLE PRESSURE CORRECTION FACTOR |
| FACTS(1) | R | 14EE | 3440 | ACTUAL FLOW FOR SEQ. NO. 1 |
| FACTS(2) | R | 14F0 | 3441 | ACTUAL FLOW FOR SEQ. NO. 2 |
| FACTS(3) | R | 14F2 | 3442 | ACTUAL FLOW FOR SEQ. NO. 3 |
| FACTS(4) | R | 14F4 | 3443 | ACTUAL FLOW FOR SEQ. NO. 4 |
| FACTS(5) | R | 14F6 | 3444 | ACTUAL FLOW FOR SEQ. NO. 5 |
| FACTS(6) | R | 14F8 | 3445 | ACTUAL FLOW FOR SEQ. NO. 6 |
| FACTS(7) | R | 14FA | 3446 | ACTUAL FLOW FOR SEQ. NO. 7 |
| FACTS(8) | R | 14FC | 3447 | ACTUAL FLOW FOR SEQ. NO. 8 |
| NONOZ(1) | R* | 14FE | 3448 | NO. OF NOZZLE FOR GV 1 |
| NONOZ(2) | R* | 1500 | 3449 | NO. OF NOZZLE FOR GV 2 |
| NONOZ(3) | R* | 1502 | 3450 | NO. OF NOZZLE FOR GV 3 |
| NONOZ(4) | R* | 1504 | 3451 | NO. OF NOZZLE FOR GV 4 |
| NONOZ(5) | R* | 1506 | 3452 | NO. OF NOZZLE FOR GV 5 |
| NONOZ(6) | R* | 1508 | 3453 | NO. OF NOZZLE FOR GV 6 |
| NONOZ(7) | R* | 150A | 3454 | NO. OF NOZZLE FOR GV 7 |
| NONOZ(8) | R* | 150C | 3455 | NO. OF NOZZLE FOR GV 8 |
| FDCF(1) | R* | 150E | 3456 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(2) | R* | 1510 | 3457 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(3) | R* | 1512 | 3458 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(4) | R* | 1514 | 3459 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(5) | R* | 1516 | 3460 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(6) | R* | 1518 | 3461 | FLOW TABLE FOR FLOW COEFFICIENT |
| FDCF(7) | R* | 151A | 3462 | FLOW TABLE FOR FLOW COEFFICIENT |
| FTOLRFLO | R* | 151C | 3463 | LOW FLOW TOLERANCE |
| FTOLRFHI | R* | 151E | 3464 | HIGH FLOW TOLERANCE |
| FWMXB | R* | 1520 | 3465 | MAX POSSIBLE FLOW THRU ONE GV |
| FMINAS | R* | 1522 | 3466 | USED TO INITIALIZE FASUM AFTER COMP. RESTART |
| FINCR | R* | 1524 | 3467 | FLOW INCREMENT FOR TRACKING |
| TRCTOLR | R* | 1526 | 3468 | FLOW TOLERANCE FOR TRACKING |
| COEF(1) | R* | 1528 | 3469 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(2) | R* | 152A | 3470 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(3) | R* | 152C | 3471 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(4) | R* | 152E | 3472 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(5) | R* | 1530 | 3473 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(6) | R* | 1532 | 3474 | TABLE FOR FLOW COEFFICIENT CURVE |
| COEF(7) | R* | 1534 | 3475 | TABLE FOR FLOW COEFFICIENT CURVE |
| FLIN(1) | R* | 1536 | 3476 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(2) | R* | 1538 | 3477 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(3) | R* | 153A | 3478 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(4) | R* | 153C | 3479 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(5) | R* | 153E | 3480 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(6) | R* | 1540 | 3481 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| FLIN(7) | R* | 1542 | 3482 | VALVE CURVE FOR FLOW-COEF=MAX (FLOW) |
| PZI(1) | R* | 1544 | 3483 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(2) | R* | 1546 | 3484 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(3) | R* | 1548 | 3485 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(4) | R* | 154A | 3486 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(5) | R* | 154C | 3487 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(6) | R* | 154E | 3488 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| PZI(7) | R* | 1550 | 3489 | VALVE CURVE FOR FLOW-COEF.=MAX. (POSITION) |
| FLOWCHNG | R* | 1552 | 3490 | FLOW DEMAND CHANGE FOR CONTINGENCY CHECK |
| | R | 1554 | 3491 | SPARE |
| | R | 1556 | 3492 | SPARE |
| | R | 1558 | 3493 | SPARE |
| | R | 155A | 3494 | SPARE |
| | R | 155C | 3495 | SPARE |
| | R | 155E | 3496 | SPARE |
| | R | 1560 | 3497 | SPARE |
| | R | 1562 | 3498 | SPARE |
| | R | 1564 | 3499 | SPARE |
| ATSSCAN | L* | 1566 | | ATS SCAN TRUE=YES FALSE=NO |
| ADSPULSE | L* | 1667 | | ADS INPUT PULSE TYPE TRUE=DISCRETE; FALSE=CONTIN |

* INDICATES THAT THIS VARIABLE MUST BE INITIALIZED

FEED FORWARD

In summary the preferred embodiment of the present invention includes the infinitely variable dynamic function generation of flow demand versus actuator lift characteristics, which allow the use of a feedforward system without the need of feedback.

Continuous dynamic function generation provides a virtually exact, within the resolution of the calculation, prediction of the operating characteristics of the governor valves at any load and flow of steam, and continuously updates and calculates the conditions of flow and load for any value and change in both the single and sequential valve modes and during transfer between the single, squential and manual modes.

Transfers between the single valve mode, the sequential valve mode and manual mode are accomplished by dynamic calculation of the control valve curve for a desired total flow through the turbine. First, a total flow demand is computed by the DEH program. Second, a corrected stage flow coefficient is determined for the flow demand. Third, data is generated which can be represented in curve form as total flow demand versus control valve actuator lift utilizing the corrected stage flow coefficient. Fourth, the difference between the calculated total flow demand or target flow and the initial flow demand is calculated. Fifth, the number of variations or iterations required to implement the change of positions from the actual initial flow to the target flow is computed by dividing the greatest flow change by a maximum allowable flow change per sampling period. Sixth, the flow changes for each valve are then divided by the number of iterations required to perform the change from initial to target flow. During all mode transfers the same approach is used.

In the sequential mode, the digital computer generates a system of asymmetric overlap load characteristics in order to provide stable operation. The asymmetric overlap system is provided with the continuous dynamic function generation.

The asymmetric overlap system provides for stability of operation in the upper end of the valve strokes and the opening strokes where the valve characteristics are very nonlinear, and in addition, prevents the ooccurrence of nonlinear oscillations by decreasing their frequency well below the response frequency of the computer system.

Because of the continuous dynamic function generation of the valve characteristics, the valve management system, which is a feedforward system can operate without the use of feedback transducers from the impulse pressure, speed and load.

The valve management system may operate with feedback functions. However, one of its greatest advantages is in the capability to operate as a feedforward system without the delay which is characteristic of feedback signals which are delayed by being fed back and compared with an input signal or quantity.

Also, the present invention includes the capability for correcting for the number of nozzles connected under each control valve. The number of nozzles emitting steam into the low pressure turbine may vary because of defects which inevitably do occur in the foundry casting into which the nozzles are machined. In addition, tracking is provided with the digital computer bumplessly transferring from a manual mode to the automatic mode at limiting flow changes by iterative means thereby reducing thermal and mechanical shakes in the turbine. In the pressure correction mode, a deadband and limitation of rate change similar to integration is provided thereby reducing thermal and mechanical shakes to and accuracy and precision of the system. Upon the monitoring of specific malfunctions, the digital computer automatically transfers from the sequential valve mode to the single valve mode and other malfunctions to a manual mode. Bumpless transfer is also provided between the squential analog output hardware and the single analog output hardware.

The functions of the digital computer, supra, could be performed on an analog computer, using operational amplifiers, diode function generators, etc. even the changes of parameters could be performed on a special potentiometer operator's panel where an operator changes the parameters of the system by operating potentiometers and/or switches, for example.

The entire contents of Ser. No. **306,752** entitled "Improved System and Method of Controlling Steam Inlet Valves for a Turbine Power Plant," filed by Leaman Podolsky and Theodore C. Giras on Nov. 15, 1972 and assigned to the present assignee is incorporated by reference herein.

What we claim is:

**1.** A system for operating a turbine power plant during a mode wherein a plurality of steam inlet valve means are operable successively in a predetermined sequence to effect variations in steam flow to the turbine comprising a steam turbine, a plurality of steam inlet valve means, each of which includes at least one steam inlet valve, control means for each valve means governed by a generated representation of valve lift position demand, means to generate a physical representation of total steam flow demand for the turbine, means to allocate successively portions of the total steam flow demand representation to respective ones of the plurality of valve means in the predetermined sequence so that allocated flows substantially equal the total steam flow demand, means governed by the total steam flow demand representation and the allocating means to generate for each valve means a representation of an allocated valve steam flow demand, said valve steam flow demand generating means being operative to govern the valve steam flow representations to correspond to a portion of the maximum flow capacity for sequentially operable valve means at times when the total turbine steam flow demand yet to be allocated is between predetermined flow values, and means governed by the generated allocated valve steam flow representations to generate respective valve lift position representations for the valve control means.

**2.** A system according to claim **1** wherein one of the said predetermined flow values of the total turbine steam flow demand yet to be allocated is above a predetermined portion of the maximum flow capacity of one valve means and the other said predetermined flow value is below a predetermined portion of the maximum flow capacity for the sequentially operable adjacent valve means.

**3.** A system according to claim **1** wherein said valve steam flow generating means is operative to generate a representation corresponding to a maximum flow capacity for one valve means at times only when the total turbine steam flow demand yet to be allocated is above a predetermined portion of the maximum flow value for the next sequentially operable valve means.

4. A system according to claim 1 wherein the predetermined flow values of the total turbine steam flow demand yet to be allocated differs in accordance with the direction of total turbine steam flow change.

5. A system for operating a turbine power plant during a mode wherein a plurality of steam inlet valve means are operated successively in a predetermined sequence to effect variations in steam flow, comprising: a steam turbine, a plurality of inlet valve means each operable to control steam flow to the turbine in accordance with their respective valve lift positions, control means for each valve means governed by a physical representation of valve lift position, means to generate a physical representation of turbine steam flow demand, means governed by the turbine steam flow demand representation to generate a physical representation of maximum steam flow each valve means is capable of admitting when fully open, means to store a physical representation of a predetermined flow value range of steam flow for each valve means, means governed by the difference between the turbine steam flow representation and generated allocated steam flow representations for each sequentially preceding valve means to generate a physical representation of unallocated turbine steam flow, allocating means governed by the stored physical representation and the representation of unallocated turbine steam flow and the representation of maximum valve flow to generate a physical representation of allocated valve steam flow for each of the valve means in succession, said physical representation of allocated valve steam flow for each valve means corresponding to either said maximum steam flow representation of said flow value range, and means governed by each of the allocated valve steam flow representations to generate said representations of valve lift position for each respective valve control means.

6. A system according to claim 5 wherein the stored flow value operating range for each valve means is limited at one end by a low portion of maximum valve flow and at the other end by a high portion of maximum valve flow.

7. A system according to claim 6 wherein the allocating means includes means to generate a first physical representation of an allocated valve steam flow for one valve means that corresponds to the difference between the unallocated steam flow representation for said one valve means and the low portion of maximum steam flow for the sequentially adjacent valve means, and to generate a second physical representation of allocated valve steam flow for the sequentially adjacent valve means that corresponds to said low portion of maximum flow at times when the unallocated steam flow for the one valve means is greater than said high portion of said one valve means and less than the sum of said high and low portions of maximum steam flow.

8. A system according to claim 6 wherein the allocating means includes means to generate a first physical representation of allocated valve steam flow for one valve means that corresponds to the high portion and a second physical representation of allocated valve steam flow for a sequentially adjacent valve means that corresponds to the difference between the unallocated steam flow and the predetermined high portion for such one valve means at times when the unallocated steam flow is greater than the sum of the high portion for the one valve means and the low portion of the sequentially adjacent valve means but less than the maximum steam flow for the one valve means.

9. A method of operating a steam turbine during a mode wherein a plurality of steam inlet valve means are operated successively in a predetermined sequence to effect variations in steam flow comprising: generating a representation of turbine steam flow demand, computing a maximum steam flow representation for each of a plurality of valve means in accordance with the turbine steam flow demand representation, computing an unallocated turbine steam flow representation for each of subsequent sequentially adjacent valve means in accordance with the turbine steam flow and a representation of target flow for each of the preceding sequentially operable adjacent valve means, comparing the representations of unallocated turbine steam flow and maximum steam flow and a representation of a predetermined operating range for each valve means to generate the representation of target flow, confining the target flow representations to a maximum flow or within predetermined operating range for predetermined ones of the plurality of valve means, generating a representation of valve lift position for each of the valve means in accordance with a respective target flow and turbine steam flow demand representation, and operating the valve means to a corresponding position in accordance with each respective valve lift position representation.

10. A method according to claim 9 wherein the predetermined operating range for each valve means is limited by a low portion of maximum flow and a high portion of maximum flow, and further comprising generating a representation of target flow for each valve of two sequentially operable adjacent valve means corresponding to a portion of maximum flow within the limits of the operating range at times when the unallocated flow operable valve means is greater than the high portion of the predetermined range for one sequentially operable valve means and greater than the low portion of the other sequentially operable valve means.

11. A method of operating a steam turbine generator system including a digital computer so as to operate a plurality of steam inlet valve means successively in a predetermined sequence in response to a steam flow demand representation, including a steam turbine, an electrical generator rotated by said turbine, a steam generating source for supplying steam to the turbine, a plurality of steam inlet valve means, each of which includes at least one inlet valve for controlling the flow of steam to the turbine, comprising the steps of: storing in the computer a physical representation of flow values defining the limits of a predetermined operating range for each valve means, operating the computer to generate a physical representation of the maximum flow capacity for each valve means in accordance with steam flow demand, operating the computer to allocate a portion of the total turbine steam flow representation to each one of the valve means in sequence until it substantially equals the total turbine steam flow demand representation, in accordance with the total turbine steam flow and a target flow representation for each preceding sequentially operable valve means, operating the computer to compare the maximum valve flow representation and flow value limit representations and the total turbine steam flow yet to be allocated to generate the target flow representation for each valve means, operating the computer to generate a representation of

target flow for one valve means corresponding to the operating range at times when the unallocated portion of total turbine flow is greater than one stored flow value limit and less than the other stored flow value limit, and operating the plurality of valve means in accordance with the respective target values.

**12.** A method according to claim **11**, further comprising operating the computer to generate a representation of target flow for one valve means corresponding to one stored flow value limit at times when the total turbine steam flow yet to be allocated is greater than the sum of the stored flow value limits and less than the sum of the maximum flow capacity and the stored flow value limit of the lower end of the operating range.

**13.** A method according to claim **12** further comprising operating the computer to generate a representation of target flow for one preceding sequentially operable means that is intermediate the flow value limits at times when the total turbine steam flow yet to be allocated is greater than the one stored limit value and less than the sum of both stored limit values.

**14.** A system for operating a turbine powered generating plant during a mode when a plurality of steam inlet valves are operable in a predetermined sequence to effect variations in steam flow, comprising: an electric generator for producing an electrical load, a steam turbine for driving the generator, a steam generator for supplying steam under pressure to the turbine, a plurality of valve control means operable to control the flow of steam from the steam generator to the turbine in accordance with their respective valve lift positions, means for generating a signal representative of load demand for the generator, means governed by the load demand signal for generating a representation of total turbine steam flow demand, a digital computer having a plurality of processing means for storing and calculating in a sequential manner, means in the computer to generate a physical representation of an allocation of a portion of total steam flow demand to respective ones of the plurality of valve means in the predetermined sequence so that allocated flows substantially equal the total turbine steam flow demand, said valve steam flow demand allocating means being operative to govern the valve steam flow representations to correspond to a portion of the maximum flow capacity for sequentially operable valve means at times when the total turbine steam flow demand yet to be allocated is between predetermined flow values, and means governed by the generated allocated valve steam flow representations to generate respective valve lift position representations for the valve control means.

**15.** A system according to claim **14** wherein one of the said predetermined flow values of the total turbine steam flow demand yet to be allocated is above a predetermined portion of the maximum flow capacity of one valve means and the other said predetermined flow value is below a predetermined portion of the maximum flow capacity for the sequentially operable adjacent valve means.

**16.** A system according to claim **15** wherein said valve steam flow generating means is operative to generate a representation corresponding to a maximum flow capacity for one valve means at times only when the total turbine steam flow demand yet to be allocated is above a predetermined portion of the maximum flow value for the next sequentially operable valve means.

**17.** A system for controlling the operation of a turbine power plant wherein a plurality of steam inlet valve means are operable successively in a predetermined sequence in response to variations in steam flow demand, said system comprising, a steam turbine, an electric generator driven by the turbine, valve means for controlling the flow of steam to the turbine, valve control means for operating the valve means to control lift position in response to a physical representation, means for generating a representation corresponding to turbine steam flow demand, means responsive to the steam flow demand representation to allocate portions of the turbine steam flow representations to respective valve means in a predetermined sequence, said last named means being operative for two sequentially operative adjacent valve means to allocate a valve steam flow representation to correspond to portions of the maximum flow capacity of the valve means at times when the portion of the total steam flow demand for such valve means is sufficient to operate one valve means beyond a predetermined operating value and insufficient to operate the other valve means at a predetermined operating value when such one valve means is operated beyond such predetermined operating level, and means for applying the allocated steam flow physical values to respective valve control means.

**18.** A system according to claim **17** wherein said last named means is operative to allocate a valve steam flow to one valve means that corresponds to a maximum flow level only at times when the steam flow demand portion for the next sequential valve means is above a predetermined operating level when the one valve is at maximum operating level.

**19.** A system according to claim **17** wherein said last named means is operative to select one of two sequentially adjacent valve means to respond to a change in total steam flow when both such valve means are at an operating level corresponding to a portion of maximum flow in accordance with a total steam flow change that varies between a value above a predetermined operating value of one valve means and below a predetermined operating value of the other valve means.

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