DESIGN OF A STEAM CONVERSION SYSTEM FOR CONVERTING NUCLEAR RADIOACTIVE STEAM INTO HOUSE HEATING STEAM

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ABSTRACT

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A Secondary Steam Generating System (SSGS) can be designed as an addition to an existing Candu (BLW) Nuclear power Station. The high pressure primary radioactive steam condenses in evaporator tubes after heat is transmitted to the water in the evaporator shell, secondary (non-radioactive) steam can be generated for domestic and industrial heating.

A detailed design is carried out for a Secondary Steam Generating System. Design criteria are presented together with detailed calculations for a system load of 2288 X 10^6 Btu/hr.

The system consists of four subsystems that are operated in parallel. Each subsystem can be run at 30% overload so that the full load can be handled by three units while one is shut down for maintenance.

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NOMENCLATURE

A	Heat transfer surface area, sq. ft.
A_a	Area, sq. ft.
A_c	Clean surface, sq.ft.
A_{ea}	Heating surface required each unit, sq. ft.
A_1	Heating surface required at condition I. sq. ft.
A_2	Heating surface required at condition. II. sq. ft.
A^*	Heating surface required at maximum design condition, sq. ft.
B	Evaporator blowdown, lb/hr.
C	Corrosion allowance, in.
D	Shell diameter, ft, or in.
D_s	Inside diameter of shell ft. or in.
De	Equivalent dia. for heat transfer and pressure drop, ft.
D_o	Estimated shell diameter, in or ft.
d_o	Tube outside diameter, in.
d	Pipe inside diameter, in.
d_v	Inside diameter of vent pipe, in.
E	Joint Efficiency factor
F_A	Actual blowdown flashed at the specified pipe diameter, lb/hr.
F	Tube sheet factor at wall thickness / I.D. ratio
F_B	Allowable blowdown flashed, lb/hr. in or dia.
F_D	Water discharge, lb/hr.

F_T	Total allowable flow, lb/hr.
f	Frictional factor, dimensionless, for ΔP in psi, ft^2/in^2
G	Mean dia. of gasket at stationery tube sheet, inches
GTTD	Greatest terminal temperature difference, ^oF
G_{hb}	Gallons hot water cooled by 1 gallon cold water to yeild a mixture at ^oF
G_s	Mass velocity, $lb/(hr)(ft^2)$
g	Acceleration of gravity., ft/hr^2
g'	Acceleration of gravity, ft/sec^2
h	Enthalpy, btu/lb.
h_v	Vaporization coefficient
h, h_i, h_o	Heat transfer coefficient in general, for inside fluid, outside fluid respectively, $Btu/(hr)(ft^2)(^oF)$
h_{io}	Value of h_i when referred to the tube outside diameter, $Btu/(hr)(ft^2)(^oF)$
h_1	Condensate inlet enthalpy, Btu/lb.
h_2, h_2'	Condensate outlet enthalpy, Btu/lb.
h_3	Feed water inlet enthalpy, Btu/lb.
h_4, h_4'	Feed water outlet enthalpy, Btu/lb.

L	Length or height, ft
LH	Latent heat in steam, Btu/lb.
LMTD	Log mean temperature difference, ^oF
$LMTD_c$	Corrected log mean temperature difference, ^oF
LTTD	Least terminal temperature difference, ^oF
N	Number of shell side baffles
N_{pT}	No. of tubes at pitch
N_t	No. of tube per path
n	No. of passes
P	Pressure, psf, psi or psia
Q	Total heat, Btu/hr
Q_1	Heat required on primary side, Btu/hr
Q_2	Heat required on secondary side, Btu/hr
Q_a	Quantity flowing, cu, ft. per sec.
Q_{in}	Heat in, Btu/hr
Qout	Heat out, Btu/hr

$\frac{Q}{A}$	Heat flux, Btu/(hr)(ft^2)
Q_I	Heat to be transferred at condition I, Btu/hr
Q_{II}	Heat to be transferred at condition II, Btu/hr
Q_N	Net heat transferred, Btu/hr.
R_a	Radius, in. or ft.
R_d	Dirt factor, (Hr)(ft^2)(^oF)/Btu
R_g	Temperature group, dimensionless
r_W	Wall resistance in heat, (Hr)(ft^2)(^oF)/Btu
$(h)_S$, S	Pegging steam required, lb/hr
S_s	Maximum allowable material stress, psi
S_E	Max. allowable stress with joint efficiency incl'd, psi
S_g	Temperature group dimensionless.
S_B	Pegging steam required, lb/hr. after blowdown heat exchanger is added.
SH	Sensible heat in condensate at full pressure, Btu/lb.
S_G	Density at reduced pressure, lb / gal.
s	Specific gravity
SL	Sensible heat in condensate at reduced pressure, Btu/lb.
S_F%	Flash steam, %
T	Temperature absolute, ^oR
T_1, T_2	Shell side fluid temperature, inlet and outlet, ^oF

t	Temperature , ^oF
t_{mb}	Temperature of mixture, ^oF
t_1	Condensate inlet temperature, ^oF
t_{cb}	Temperature of cold water , ^oF
t_2, t_2'	Condensate outlet temperature, ^oF
t_{hb}	Temperature of hot water, ^oF
t_3	Feed water inlet temperature, ^oF
t_m	Minimum pipe wall thickness required, in.
t_4, t_4'	Feed water outlet temperature, ^oF
t_{act}	Actual pipe wall thickness, in.
t_s	Shell thickness, in
t_{mt}	Manufacturers' tolerance, %
t_{sH}	Shell head thickness, in
t_c	Channel thickness, in
t_{cH}	Channel head thickness, in
t_t	Tube sheet thickness , in
U	Heat transfer coefficient , Btu/(hr)(ft^2)(^oF)
U_D	Designed heat transfer coefficient, Btu/(hr)(ft^2)(^oF)
U_C	Clean heat transfer coefficient, Btu/(hr)(ft^2)(^oF)

V	Volume, cu.ft
V_s	Specific volume cu.ft/lb.
V_1	Specific volume at full pressure, ft^3/lb.
V_2	Specific volume at reduced pressure, ft^3/lb.
V_B	Volume of flow water, cu.ft.
V_H	Holding volume from bottom of tank to outlet, cu.ft.
V_W	Total water volume, cu.ft.
v	Velocity, ft./sec.
W	Primary steam or condensate , lb / hr.
W_B	Weight of blowdown water, lb / hr.
W_F	Weight of water flashed into steam, lb / hr.
W_N	Net weight of blowdown water , lb / hr.
W_p	Condensate return from original circuit, lb/hr.
W_s	Condensate return from SSGS, lb/hr.
Wm	Mixed condensate, lb/hr

W_1	Treated water inlet, lb/hr.
W_2	Treated feed water outlet, lb/hr.
w	Flow, lb per sec.
X %	Percent of blowdown
y	Pressure temperature correction factor, for ferritic steel
Δh	Enthalpy difference, Btu/lb.hr.
Δh_1	Enthalpy in primary side give-up to secondary side, btu / lb.hr.
ΔP	Pressure drop, psf or psi.
ΔP_s	Shell side pressure drop. psf or psi
ΔP_t	Tube side pressure drop, psf , or psi
ΔP_r	U-tube pressure drop, psi
ΔP_T	Total tube side pressure drop, psi
Δt	Temperature difference, ^oF
$(\Delta t)_w$	Temperature difference between the tube wall and the boiling temperature.
$(\Delta t)_a$	Apparent temperature difference between the primary steam and the secondary steam.
Δt_1	Temperature difference at condition 1
Δt_2	Temperature difference at condition 2

#	lb/hr
ϕ'	Diameter, ft, or in
ϕ_t, ϕ_s	Viscosity ratio for tube and shell side respectively
ρ	Density, lb/cu.ft
*	At maximum design condition
$\theta m, \theta m1, \theta m2$	Heat head, ^{o}F at condition 1 and 2
" , '	Inches and feet respectively

Abbreviations:

FPS	Foot per second
GPM	Gallon per minute
Dia.	Diameter

Subscripts:

S; W_p, W_s, W_m; W_1, W_1', W_2; B_1, B_2; I, II, represents pegging steam; Primary, secondary primary returned condensate from SSGS and mixed condensate ; water inlet and outlet; blowdown inlet and outlet; and at condition I, or II respectively.

CHAPTER 1

INTRODUCTION

A Candu reactor is a type of nuclear reactor which uses either ordinary fresh water or heavy water as a moderator.

Most Candu reactors use a pressurized heavy water instead of a light water heat transfer system in the reactor fuel channels. One nuclear reactor at Gentilly 1 nuclear power generating station in Quebec uses light water as the heat transfer medium for the power generation. This plant started operation in 1971.

Gentilly 1 nuclear power plant has an electrical power generating capacity of 250 MW_e. It is the only light water cooled power reactor which allows the light water to boil within the reactor fuel channels. The mixture of steam and hot water in the reactor fuel channels is accumulated in the reactor steam drum where separated saturated steam is drawn off to provide the thermal power to the generator turbine (see Fig. 1 and Fig. 2).

The Candu Boiling Light Water (BLW) nuclear power generating process is said that to have reduced capital cost through elimination of heavy water. A Candu reactor of the Pressurized Heavy Water (PHW) type would inevitably have leaks in the turbine and its auxiliary equipment. Such leaks make a direct cycle prohibitively expensive.

Other advantages of using light water as a coolant in the reactor channels are:

1. A small increment of thermal efficiency which yields a saving.
2. Reduction in operating cost from the absence of loss of heavy water coolant in the system.

There are many practical difficulties in using light water as a coolant. One of the difficulty is that the light water will capture more neutrons than the heavy water when natural uranium is used in the reactor fuel channels. Because of the need for much more expensive enriched uranium fuel, this cost may offset the gains from the above advantages.

Since a sophisticated and complicated control system is designed for the Gentilly 1 nuclear power reactor, natural uranium is used as a fuel in the reactor channels. Light water is used as a coolant while it is boiling off within the reactor fuel channels. The mean density of the water is reduced while it is boiling and this reduces its neutron capturing ability making thermal power generation possible.

In the Candu Pressurized Heavy Water (PHW) power reactor, there is only a simple control system required since the heavy water is a moderate neutron absorber. However, since light water is more effective as a neutron absorber than as a moderator, a decrease in coolant density causes a further increase in reactor power and a positive feed back which must be countered by a more sophisticated control system.

It is however not the intention of this paper to go into details of the nuclear power generation system. The topic of this paper is limited to describing and discussing how the nuclear thermal power from Gentilly 1 can be efficiently utilized as a possible source of steam for a central heating system by designing an addition to the existing Candu Boiling Light Water (BLW) nuclear power process. This system will be referred to as a Secondary Steam Generating System (SSGS).

CHAPTER 2

DESIGN CONCEPT

2.1 GENERAL :

The Secondary Steam Generating System is an addition to the existing Gentilly 1 Candu Boiling Light Water (BLW) nuclear power generating system. In abbreviation, this addition will be referred to as SSGS. The system is designed to convert the nuclear reactor primary steam into secondary steam. Since the primary steam is a direct cycle product of the Candu BLW nuclear reactor, the primary steam is a radioative steam which is therefore not a desirable direct source of thermal power. It is neither suitable for direct industrial use nor for direct central heating use until a satisfactory heat transformer is developed. The SSGS is proposed to accomplish the desired heat transformation.

The object of the SSGS is to utilize the heat source in the primary steam through sets of heat exchangers to generate secondary steam and still keep the primary radioactive steam in the closed circuit.

In the petrochemical industry, heat exchangers which are used to generate vapor or to produce a low pressure steam are called vaporizer, reboiler, or evaporator. In this case the purpose is different and the reboiler of the heat exchanger will be referred to as secondary steam generator which is actually an evaporator.

The enthalpy in the primary steam is used to produce the secondary conventional steam. The primary steam is then condensed and collected in a vessel called a primary condensate collector from which the primary condensate is returned to the existing condensate system. The combined condensate is cooled through sets of heat exchangers to the required temperature and pressure. Feed water to the evaporator is preheated to the required temperature before it enters the evaporator. Since the heat exchanger serves a cooling and preheating purpose, it is referred to as a cooler / preheater.

Details of the proposed system are shown in Fig. 3 and Fig. 4 .

In this paper, only the SSGS conceptual design is presented and discussed. It is not intended to cover all details such as shelters, water treatment system, steam transmission pipe lines, or complete mechanical and instrumentation details.

From many practical examples, the minimum rating of most Candu reactors is 2.5 million pounds per hour or an equivalent electric power of 250 MW_e. The Gentilly 1 nuclear reactor power station is therefore a typical, good size example for use as a case study.

2.2. DESIGN REQUIREMENTS AND CONSTRAINTS AND TARGET

The following specific requirements and constraints shall be satisfied and cleared before any commencement of engineering study and discussion:

1. The SSGS system shall be designed to supply the secondary steam requirement at a maximum of 1.9×10^6 LB/HR at a required steam quality of 380 PSIA minimum to 500 PSIA as maximum with an maximum allowable moisture content of 1% wet.
2. The primary cycle shall be maintained in an enclosed circuit , and the primary condensate is therefore has to return to the original cycle without loss.
3. No leakage on the primary side is allowed due to it is radioactive fluids.
4. The system shall be designed to with stand enough pressure as well as vacuum.
5. Enough pressure relief device are required to protect system from burst or breakage due to over pressure.
6. Enough provisions of environmemtal safety devices are required.
7. Enough provisions for the steam and water quality control are required.
8. The system shall be provided with all necessary means of venting and drainage associated with the start-up, shutting down, operation and maintenance of the system.

9. The system shall be designed to be capable of start-up, shutting down, operation and maintenance under any climate condition.

10. The system shall be very reliable, safe, and maintainable in every aspect.

11. The condition of primary condensate returning to the original circuit shall meet the condition and requirements of the original cycle.

12. Drains, vents, and other necessary connections shall be provided at strategic locations in an enclosed special designed system.

2.3. SPECIAL FEATURES AND CHARACTERISTICS

2.3. 1. The SSG system can be divided into several modules. A module can be defined as one of the continous loops in the SSG system that may be isolated completely if there is such a situation arises or for emergency purposes.

2.3. 2. Since there is plenty pressure available in the primary circuit and the feed water is supplied from the water treatment plant; therefore as far as the SSG system is concerned, except the control valves, there will be no rotating equipment or moving parts involved in the whole SSG system.

2.3. 3. Each module in the SSG system is identical to each other and each consists of one secondary steam generator (evaporator), one primary condensate collector, and two cooler/preheaters are connected in series. Each module including equipment, piping, fittings and relevant control instrumentation and control devices are also identical and interchangeable. This special feature may tremendously reduces the maintenance and operational cost due to each module is either identical in arrangements, or in characteristics and operational functions.

2.3. 4. Each module may operates on it's own independently without any interference to the others, or it may be isolated from others; or it can be operated in parallel with all the remainder of modules.

CHAPTER 3

SYSTEM DESCRIPTION

In order to minimize the size of the secondary steam generator (evaporator), and as well as for simple and easy installation and better heat transfer efficiency, the U-tube bundle kettle reboiler type will be the most suitable one to be used for the purpose. In addition to the above reasons, the horizontal kettle reboiler type evaporator can be equally divided into two parallel streams as shown in Fig. 5 . In this way, a smooth control on flow rate control and an evenly distributed heat transfer may be achieved because each path would only require a small size of piping and control valve. The evaporator shall be designed for three-element control, namely, pressure, water level, and flow.

The primary steam to the SSGS is extracted from the main steam line which is originally designed for the electrical generator turbine use. In order to provide a ~~small~~ amount of primary steam to the turbine to maintain the minimun station load, a modification to the main turbine controller is required. This modification is simply an addition of a small by-pass to the existing system to permit a small steam flow to the turbine. The turbine will act as an absorber for the secondary steam generating system to maintain the reactor steam drum pressure stable.

The primary steam admitted to the tube side of each evaporator is controlled by the control valve throtting the steam to the appropriate secondary steam pressure and flow requirements. The control valve is operated from signals derived from flow and pressure transducers in the secondary steam line header.

The primary steam after condensing in the evaporator tubes will flow into the condensate collector. A small connecting line shall be provided from the up-stream of the evaporator inlet to the condensate collector for the purpose of pressure balance between evaporator inlet and condensate collector. An isolation valve works on the low level switch is provided to prevent any steam to the cooler/preheater.

The condensate from the condensate collector is then directed through the cooler/preheater with only a relatively small pressure drop. Since the condensate has to satisfy the existing Gentilly 1 deaerator operating pressure, a pressure reducing valve is therefore needed to achieve the desired pressure in the original closed circuit. The existing level controller on the existing deaerator shall take a complete override to control the flow control valve which controls the primary condensate returning flow rate from the SSG system.

In order to alleviate any problems like condensate flashing in the return system, and problem as not to overload the deaerator etc., the primary condensate return temperature has always to be slightly less than the existing deaerator operating saturation temperature which is the restricted condition while design the cooler/preheater.

The water which feeds to the evaporator has to be treated water which is beyond the topic. It is assumed that the water feeds to evaporator has already been deaerated and treated. The treated water is preheated to a temperature slightly below saturation temperature which is generally a good practice in preheater design. Pressure losses must be taken into considerations such as losses of pressure drop across the cooler/preheater, the pressure and friction losses in the piping system or any hydraulic head losses as well as any other pressure losses across any valves, instruments, measurement devices or elements or fittings.

For most deaerators, it is a common practice to design a deaerator operating pressure at 4 to 8 PSIG above the atmospheric pressure. The feed water enters the SSGS at 223^{o} F will be diverted to each module by individual control valves deriving their signals from individual level controllers on the evaporators. The constant water level in each evaporator will thus be maintained.

The secondary steam produced in the evaporator will pass through the integrated moisture separator to a steam line header then to the secondary steam reserve tank at the delivery boundary from which a number of steam delivery pipe lines can be designed to convey the secondary steam to the location where central heating or heat source is required.

CHAPTER 4

PROBLEM DISCUSSION

4.1 PRIMARY STEAM CONDENSATE RETURNING TEMPERATURE

The first aim of the SSGS is to generate the secondary conventional steam, and the second aim of the SSGS is to keep the original cycle in a closed circuit. In order to accomplish these two aims, it is necessary to restrict the primary condensate returning temperature and pressure to the operating condition of the existing deaerator in the primary circuit. The pressure is controlled by pressure control valve. The cooler / preheater is the device which serves the temperature control purpose by means of the heat transfer process. However, under one condition that assumes the cooler / preheater inlet temperature is fixed at 223°F (which is specified at 18 psia), the evaporator feed water inlet temperature shall not be at saturation condition. It is a general practice to keep the evaporator or boiler feed water temperature always below saturation temperature. This temperature shall be several degrees subcooled (below saturation temperature at that particular equivalent pressure), otherwise bubbling problem may occur in the system due to immediate flashing. But it is not economical to design a too low feed water temperature to the evaporator because the evaporator would demand a more heat source, thus, the primary steam. It is therefore assumed that the evaporator feed water inlet temperature shall be at a fixed sub-cooled temperature, say from 5 to 10 degrees below the equivalent saturation temperature at the evaporator operating pressure.

From the above description, it is known then that the evaporator feed water temperature across the cooler/preheater both at the inlet and the outlet are then being fixed. Based on the amount of primary condensate flow rate at different condition, the primary condensate returning temperature to the original cycle is therefore able to be calculated out.

In the calculations, based on the different flow rate and the fixed water temperature and the alternative locations and varied conditions, the primary condensate returning temperatures are luckily not exceeding the original existing deaerator operating temperature. Thus, there will be no flashing problems occuring at the existing deaerator. Results of calculations are shown in table 4.1.1.

TABLE 4.1.1

RESULTS OF CONDENSATE RETURN TEMPERATURES

	CASE 1		CASE 2	
	1. P 2X24"Φ =63psi 2. evap. ope. p= 486 psia	1. P 3x24"Φ =35psi 2. evap. ope. p =458psia	1. P 2x24"Φ= 63psi 2. evap. ope. p=486 psia	1. P 3x24"Φ = 35psi 2. evap. ope. p= 458 psia
1. PRIMARY CONDENSATE FLOW RATE lb/hr	606,230	651,939	603,470	613,158
2. TREATED EVAPORATOR FEEDWATER FLOW RATE, lb/hr	577,102	577,102	577,102	577,102
3. PRIMARY CONDENSATE OUTLET TEMP. OF COOLER/PREHEATER, °F	268.52	280.63	263.93	276.16
4. EXIS. DEAERATOR OPE TEMP @68% LOAD & 67psia- °F	300	300	300	300
5. PRIMARY CONDENSATE RETURN BELOWEXIS.DEA OPE TEMP- °F	31.48	19.37	28.40	36.46
6. SATURATION TEMP AT PRESS CONDITION °F	464	458	464	450
7. EVAPORATOR FEED WATER INLET FIXED TEMP. °F	458	450	458	8
8. EVAPORATOR FEEDWATER SUB-COOLED BELOW SAT- °F	6	8	6	8

4.2. DIFFERENT SCHEMES OF PROCESS ARRANGEMENT

Different alternative arrangements for the cooler/preheater and the condensate collectors have been studied as well as the possibility of eliminating the condensate collectors.

It is assumed that the configuration and type of the vessels and the number of units have been selected. The following are the main arrangements which are under discussion:

4.2. 1. ALTERNATIVE NO. 1 (Refer to Fig. 11)

Evaporator and cooler/preheater only, without the provision of the condensate collecting tank in the system

In this scheme of arrangement, there is no condensate collecting tank to collect the condensate. The complete system will flow only depending upon the pressure difference between the pressure inlet of the evaporator and the operating pressure at the existing deaerator. Since it is a continuous once-through passage. If there is any intermittent blockage on the down stream of the cooler/preheater outlet condensate return line due to a higher level or some other reasons, the system starting from the evaporator inlet control valve up to the existing deaerator inlet control valve would be completely flooded. Full of condensate will occur including the whole U-tubes which are normally submerged in the evaporator. The primary system which originally is aiming for the steam condensation reboiling would end up with a condensate flooded operation, which may not be a desirable efficient heat transfer process. Another additional disadvantage is due to lack of a surge capacity. The operation would be unstable in a practical point of view.

It is therefore concluded that the SSG system is definitely requiring a surge capacity which would be a vessel to accommodate the surge capacity and be able to smooth out the system operation. In the mean time, it may be a help of providing the better steam condensation heat transfer process.

4.2. 2. ALTERNATIVE NO. 2 (Refer to Fig. 12)

Evaporator with immeciate cooler/preheater connection and condensate collector at the end

In this arrangement, it is assumed that the condensate collecting tank is located immediately after the cooler/preheater. Now it is obvious that there is a surge capacity in the system but the problem arises is that in the manner of achieving the condensation heating process to produce the required secondary steam; as it has been stated in the ALTERNATIVE NO.1, due to the flooding problem in the system, it is impossible to achieve the better condensation heating process in the U-tube. In this arrangement, the flow is depending on the pressure difference between the evaporator inlet pressure and the condensate collecting tank inlet pressure. But the problem is that any shut off on the downstream of the cooler/preheater outlet, the intermittent flow would cause the same flooding problem of the U-tube in the evaporator for the next start-up. In this type of arrangement, it is still impossible to achieve a continuous operation scheme without any interruption on the continuous flow. Another reason which may cause the same problem is due to the high pressure drops while in U-tube of the evaporator is connected to the cooler/preheater in series.

It is assumed that the hydrostatic head could not overcome the pressure drop in reasonable condition. But there will be enough condensate in the system from the U-tube to the cooler/preheater section and outlet section inclusive, and the condensate would cause a momentarily instantaneous blockage of the evaporator steam inlet, or a sudden condensate level increase in the condensate collecting tank immediately after any intermittent shut off of the control valve. The problems described above would cause unstable operation. It is therefore also not recommended for this type of arrangement.

4.2. 3. ALTERNATIVE NO 3

Evaporator with immediate condensate collector and cooler/preheater at the end. (Refer to Fig. 13)

In this arrangement, while having the condensate collector located immediately after the evaporator, nevertheless, the pressure drop in the evaporator U-tube, even though pressure is equalized by one small line which connects the evaporator inlet and the condensate collector. Since the condensate collector has to be located below the evaporator in order to provide enough hydrostatic head for the continuous condensate gravity flow; in any event, or due to occasional intermittent shut down either for the complete system or due to the downstream control valve; in this arrangement there would be absolutely no condensate remaining in the U-tube or cause any blockage for the next start-up due to any condensate occuring during any shut down. The condensate would normally find two ways of draining by gravity into the condensate collector. One way is from the small equalized line, and the other is from its main flow route.

It is therefore possible to establish and maintain a continuous primary steam flow to the evaporator U-tube, but an intermittent shut off either on primary steam side or condensate side would not create any problem or effects on any scheme of the complete system operation.

4.2. 4. OTHER DIFFERENT ALTERNATIVES

Other different alternatives are mainly sub-divided from the above main alternatives. There are not much choice for the arrangement of alternative no. 1. However, investigation has been made for the possible sub-alternatives from alternative No. 2. and similar for alternative No. 3

4.2.4. 1. OTHER ALTERNATIVE NO 2-A (Refer to Fig. 14)

In this arrangement, two cooler/preheaters are connected in parallel. Each cooler/preheater equally share the load from the separate flow of both the primary condensate from the evaporator and the evaporator feed water supply. The primary steam after it has given up its heat to the water in the evaporator shell side, will condense into condensate in the lower part of the U-tube and continuously flow into the tube side of the two separated cooler/preheaters. The condensate is rejoined together into a condensate collector via a pressure reducing valve. The feed water is fed to the evaporator via the shell side of the two cooler/preheaters.

In this arrangement, although the load has been equally shared by the two cooler/preheaters, yet it has not been taken the advantage of the available high primary pressure drop. The pressure control valve has to be fairly big and accurately control of the pressure at the condensate collector otherwise the problem would be extended to the existing deaerator.

4.2.4. 2. SUBALTERNATIVE NO. 2-B (Refer to Fig. 15)

In this arrangement, the two cooler/perheaters are connected is series, but with the first one immediately located after the evaporator. The primary condensate from the evaporator U-tube is entering the shell side of the first cooler/preheater then passing through a pressure control valve and a drainer then into the tube side of the second cooler/preheater. The evaporater feed water is on the reversed flow. It enters the shell side of the second cooler/preheater and then the tube side of the first cooler/preheater.

According to calculation, the primary condensate temperature outlet of second cooler/preheater returning to the original cycle is within the limit of the existing deaerator condition. The provision for a drainer is designed to be installed after the pressure reducing valve. But the drainer may require frequent maintenance or repairing work which are not very desirable.

4.2.4. 3. SUB-ALTERNATIVE NO. 2-C (Refer to Fig. 16)

In this arrangement, the two cooler/preheaters are connected in series. In order to take the advantage of a greater pressure drop, the primary steam condensate is placed on the tube side of the two series connected cooler/preheaters. While the evaporator feed water is placed on the shell side of the cooler/preheaters, primary steam condensate is collected at the condensate collector after passing through a pressure reducing valve. The condensate collector is therefore normally should operating at a pressure just a little higher enough than the existing deaerator operating pressure to overcome the friction loss and hydrostatic head in order to make the fluid flowing to the original cycle.

However, the problem that can be foreseen would be the flooding problem which has been described in 4.2. 1.

4.2.4. 4. SUB-ALTERNATIVE NO.2-D (Refer to Fig. 17)

In this arrangement, the two sub-coolers are connected also in series, but with the fluid alternatively reversed as in 4.2.4. 3 . Due to the hot primary steam condensate is placed on the shell side, thus thermal expansion provision has to be considered on the cooler/preheater shell, which would possibly result in an ~~un~~economical higher manufacturing cost. On the other hand, a detailed calculation has been made and it is appeared that the evaporator feed water placed on the tube side would require a higher pumping power for the feed water whereas the available high pressure drop on the primary condensate has not been utilized.

4.3 EFFECTS OF DIFFERENT BOUNDARY PRESSURES

From calculations based on the different boundary pressures at assumed constant primary conditions, and at a fixed U-factor, a comparison between the higher boundary pressure and lower boundary pressure has been made as in table 4.3.1

By a glance of the table 4.3.1, it is likely to design a evaporator at a lower boundary operating steam pressure for the sake of operation, control as well as economic point of view. But, there is a very important factor that should be taken into serious consideration before the design of the evaporator. Let us assume that there will be three 24"ϕ steam delivery lines proposed for the system and the secondary steam requirement as a constant. If the boundary pressure of the secondary steam is not at constant state, that is while there are three 24"ϕ steam delivery lines in service, it would result in a lower operating pressure at the evaporator due to a lower steam flow rate passing through each line at lower pressure drop. Whereas the evaporator would result in a higher operating pressure at the evaporator due to the steam flow rate passing through each line at a higher pressure drop when there are only two 24"ϕ steam delivery lines are in service (one steam delivery line is assumed interrupted for maintenance or repairing purposes).

As it has been described in the other problem discussions, the heat head is one of the important function that would govern the operation of the evaporator. The heat head is defined as the temperature difference between the primary steam and the secondary output steam. The heat head is higher while the primary steam condition is constant and the boundary secondary steam pressure is lower and vice versa. As a result, the heat head is a direct function that would affect the heat transfer efficiency of the evaporator. That is a higher heat head would result a higher heat transfer of the evaporator and vice versa, thus, the less primary steam requirement and vice versa. From the above discussion, although, it seems to be the evaporator that should be desinged at a lowest boundary secondary steam operating pressure for the best heat transfer efficiency, yet it is not true because if the evaporator is designed at a lower pressure, in the event of a sudden rise of pressure in the system due to an interruption of one 24"ϕ steam line would result an immediate high pressure in the system which was originally designed at a lower pressure. It is therefore not a logical way to design an evaporator at a lower boundary pressure and operates at a higher pressure. On the contrary, for the most reliable and safe design point of view, the evaporator should be designed based on the maximum boundary operating pressure.

Theoretically, it would be to the most benefit of the system that the evaporator should be designed at the maximum boundary operating pressure and operated at the minimum boundary operating pressure. In this way, since the heat head would be theoretically the maximum, therefore the maximum heat transfer can be achieved, thus, the minimum heating surface would be required. However, in practical operation, this theoretical aim is very difficult to achieve. The reason is that while the evaporator has been designed according to its maximum boundary operating pressure, but the heat head is the minimum, and the heating surface of the evaporator is automatically sized at its maximum. When in operation, the evaporator at any pressure lower than that of maximum designed pressure, the heat head is higher; and therefore, the requirement of heating surface is less during that moment of low pressure while the primary steam condition is assumed fixed. If the normal operating pressure is always less than the maximum design pressure, that means the heating surface of the evaporator has been oversized. Once the heating surface is sized, it is non-variable, thus, there will be a surplus of heating surface which exceeds the normal requirements. For this reason, some of the heating surface (tube bundle) may exposed to the steam which is really not a desirable phenomenon.

From the above discussion, it can be concluded that based on the conditions as described, there are two following methods to achieve the evaporator operation efficiency: (1) Efficiently and accurately control the flow of the primary steam to the evaporator. The boundary steam pressure should be a direct function as well as a secondary steam flow of the primary steam flow rate to the evaporator. (2) A direct control of U-factor or achieve some kind of control on a variable heating surface. (This method seems very difficult to achieve). (3) Install a pressure control valve to control the boundary steam pressure(set and fixed the boundary steam pressure) at the designed condition all the time.

It is therefore concluded here that for the best operational and control point of view (not the best heat transfer in operation), the evaporator should be designed at the most possible operating boundary pressure at which the maximum pressure drop at worst condition of the secondary steam delivery lines should be allowed. Or the evaporator can be designed and operated at its maximum condition by the assistance of the addition of a pressure control valve and always maintain the set boundary pressure to the evaporator designed condition.

TABLE - 4.3.1

EFFECTS OF DIFFERENT BOUNDARY PRESSURES (Based on constant primary steam condition and at fixed U-factor)

	DESCRIPTION	HIGHER BOUNDARY PRESSURE	LOWER BOUNDARY PRESSURE
1	HEAT TRANSFER EFFICIENCY	LOWER	HIGHER
2	HEAT HEAD	LOWER	HIGHER
3	HEAT TRANSFER RATE BTU'S/FT^2 OF HEATING SURFACE	LOWER	HIGHER
4	HEATING SURFACE REQUIRED	GREATER	LESS
5	ESTIMATED REQUIRED SHELL DIAMETER	LARGER	SMALLER

4.4 EFFECTS ON NUMBER OF EVAPORATORS REQUIRED FOR THE SYSTEM

This problem is related to the problem discussed on 4.3 " Effects of boundary pressures".

The problem encountered are very similar to those that have been discussed in the "Effects of boundary pressure", except one thing that shall be pointed out here that is the difference of the evaporator shell diameters. The example has been chosen for 4-evaporator scheme and 3-evaporator sheme. From the calculated estimated shell sizes, it has been indicated that for the 4-evaporator scheme normally would require only 3 evaporators with one for stand-by at its full normal operating designed condition; and for 3-evaporator scheme normally would require only 2 evaporators with one for stand-by.

The 4-evaporator scheme is chosen because of economic reasons such as less operation and maintenance cost as well as fabrication cost. However, no matter which scheme is chosen, since it has been described in the problem discussion of "Effects of boundary pressure" that the evaporator should operate at its designed condition, that is the number of evaporators which are required for producing the heat duty to generate the amount of required secondary steam shall be designed at its maximum operating conditions and operating at its minimum operating conditions with the allowance of heat duty is designed in such a way that one additional evaporator has the fully required heat surface to take over the full load operation at any time in the event of any one of the evaporator is forced to be isolated.

In this kind of design arrangement, each evaporator would serve the best performance in operation. On the other hand, since there is an immediately available stand-by evaporator with the full load designed capacity, so that the whole module is more reliable to ensure the continuous operation without interruption; which is superior than operating all the evaporators at only a fractional load on each but with a low efficiency of evaporator preformance.

Another advantage of having all the evaporators sized and operated to the full load designed condition would be the full control scheme may be achieved . That is each control valve would be able to function properly and accurately according to its fully sized loading and capacity. As a result,the complete system performance will be improved.

For a conclusion, the evaporator should be sized in such a way that based on the designed and operational condition as described above. At a smaller shell dia. for example use 4-evaporator scheme instead of 3-evaporator scheme to achieve the purpose, but not exceeding the total number of 6 evaporators. This is determined by the economic cost factor of fabrication.

For results of calculations, refer to tables 4.4.1 through 4.4.3

TABLE 4.4.1

OPERATING SCHEME BASED ON 420 PSIA AT PLANT BOUNDARY

ITEM NUMBER	SYMBOLS AND NOTES	4-EVAP. SCHEME		3-EVAP. SCHEME	
	# = LB/HR. * : AT CONDITION WHERE ONE EVAP. IS OUT OF SERVICE ‡ : TUBE BUNDLE SHELL DIA. WITH .75" O.D. TUBING ON 1" □ PITCH AT 30' LONG NOTE: UNLESS OTHERWISE SPECIFIED, ALL VALUES ARE FOR ONE EVAPORATOR.	WITH 3 EVAP. NORMALLY IN SERVICE, 1 EVAP. STAND-BY		WITH 2 EVAP. NORMALLY IN SERVICE, 1 EVAP. STAND-BY.	
		CONDITION (I)	CONDITION (II)	CONDITION (I)	CONDITION (II)
		1. $\Delta P_{2 \times 24'' \phi} = 63$ PSI 2. EVAP. DES. OPG. P = 486 PSIA	1. $\Delta P_{3 \times 24'' \phi} = 35$ PSI 2. EVAP. DES. OPG. P = 458 PSIA	1. $\Delta P_{2 \times 24'' \phi} = 63$ PSI 2. EVAP. DES. OPG. P = 486 PSIA	1. $\Delta P_{3 \times 24'' \phi} = 35$ PSI 2. EVAP. DES. OPG. P = 458 PSIA
1	TOTAL SECONDARY STEAM REQ'D, #	2,285,553	2,285,553	2,285,553	2,285,553
2	TOTAL HEAT TRANSF. REQ'D, BTU - Q_T	1,753,422,592	1,781,260,772	1,753,422,592	1,781,260,792
3	HEAT TRANSF. REQ'D, BTU - Q^*	584,474,197*	593,753,590*	876,711,296*	890,630,386*
4	U - FACTOR, BTU/(HR)(FT²)(°F)	540	540	540	540
5	IF USE HEATING SURF. AS DESIGNED, FT², - A^*	41,951*	41,951*	62,928*	62,928*
6	HEAT HEAD, °F - θ_m	25.80	31.68	25.80	31.68
7	HEAT TRANSF'D, BTU/# OF PRIM. STEAM	723.085	722.985	723.085	722.985
8	BTU/# OF SEC. STEAM	767.177	779.357	767.177	779.357
9	BTU/FT²	13,932	17,107	13,932	17,107
10	PRIM. STEAM REQ'D # = $\frac{3}{7}$	808,306*	821,252*	1,212,460*	1,231,879*
11	ADDITIONAL PRIM. STEAM REQ'D, #	—	12,946	—	19,419
12	SEC. STEAM PROD'D, #	761,851*	761,851*	1,142,776	1,142,776
13	HEATING SURFACE USED, FT² = $\frac{3}{4 \times 6}$	41,951*	34,708*	62,928*	52,061*
14	EST'D TUBE BUNDLE SHELL I.D., ϕ" ‡ *	70*	64*	90*	80*
15	TOTAL HEATING SURFACE REQ'D, FT² = $\frac{2}{4 \times 6}$	125,855	104,123	125,855	104,123
16	TOTAL HEAT'G SURF. OF 3 EVAP., FT²	125,855*	125,855*	—	—
17	OF 2 EVAP., FT²	83,902	83,902	125,855	125,855
18	SURPLUS HEAT'G SURFACE, FT² = 5-13	—	7,243	—	10,867
19	TOTAL SURPLUS HEAT'G SURFACE, FT² = 16-15	—	21,732	—	21,732
20	TOTAL PRIM. STEAM REQ'D, #	2,424,920	2,463,758	2,424,920	2,463,758
21	HEAT'G SURF., FT²/# OF PRIM. STEAM = $\frac{13}{10}$	0.051900	0.042262	0.051900	0.042262
22	IF PRIM. STEAM REMAINED AS DESIGNED = 10	808,306*	808,306*	1,212,460*	1,212,460*
23	ADDITIONAL HEAT'G SURF. REQ'D = 11×21	—	547	—	821
24	HEAT'G SURFACE REQ'D = 13+23	41,951	35,255	62,928	52,882
25	SURPLUS OF HEAT'G SURF., FT² = 5-24	—	6,696	—	10,046

TABLE 4.4.2.

OPERATING SCHEME BASED ON 380 PSIA AT PLANT BOUNDARY

ITEM NUMBER	SYMBOLS AND NOTES	4 - EVAPORATOR SCHEME			
	# : LB/HR. * : AT CONDITION WHERE ONE EVAP. IS OUT OF SERVICE ‡ : TUBE BUNDLE SHELL DIA. WITH .75" O.D. TUBING ON 1" □ PITCH AT 30' LONG NOTE: UNLESS OTHERWISE SPECIFIED, ALL VALUES ARE FOR ONE EVAPORATOR.	WITH 3 EVAPORATORS NORMALLY IN SERVICE, 1 EVAPORATOR STAND-BY			
		CONDITION (I)	CONDITION (II)	CONDITION (III)	
		1. $\Delta P_{2\times 2\times 4}$ = 63 PSI 2. EVAP. DES. OPE. P = 486 PSIA (420+63+3) DES. COND'N	1. $\Delta P_{2\times 2\times 4}$ = 68 PSI 2. EVAP. DES. OPE. P = 451 PSIA (380+68+3)	1. $\Delta P_{3\times 3\times 4}$ = 33 PSI 2. EVAP. DES. OPE. P = 416 PSIA (38+33+3)	
1	TOTAL SECONDARY STEAM REQ'D, #	2,185,553	2,285,553	2,285,553	
2	TOTAL HEAT TRANSF. REQ'D, BTU - Q_T	1,753,422,592	1,937,369,485	1,777,885,988	
3	HEAT TRANSF. REQ'D, BTU - Q^*	584,474,197*	645,789,828*	592,628,662*	
4	U - FACTOR, BTU/(HR)(FT²)(°F)	540	540	540	
5	IF USE HEATING SURF. AS DESIGNED, FT², - A^*	41,951*	41,951*	41,951*	
6	HEAT HEAD, °F - θ_m	25.80	33.24	41.29	
7	HEAT TRANSF'D, BTU/# OF PRIM. STEAM	723.085	722.985	722.985	
8	BTU/# OF SEC. STEAM	767.177	847.658	777.880	
9	BTU/FT²	13,932	17,950	22,297	
10	PRIM. STEAM REQ'D # = 3/7	808,306*	893,226*	819,696*	
11	ADDITIONAL PRIM. STEAM REQ'D, #	—	84,920	11,390	
12	SEC. STEAM PROD'D, #	761,851*	761,851*	761,851*	
13	HEATING SURFACE USED, FT² = 3/(4×6)	41,951*	35,977*	26,579*	
14	EST'D TUBE BUNDLE SHELL I.D., φ" ‡ *	70*	64*	56*	
15	TOTAL HEATING SURFACE REQ'D, FT² = 2/(4×6)	125,855	107,933	79,737	
16	TOTAL HEAT'G SURF. OF 3 EVAP., FT²	125,855*	125,855*	125,855*	
17	OF 2 EVAP., FT²	83,902	83,902	83,902	
18	SURPLUS HEAT'G SURFACE, FT² = 5-13	—	5,974	15,372	
19	TOTAL SURPLUS HEAT'G SURFACE, FT² = 16-15	—	17,922	46,118	
20	TOTAL PRIM. STEAM REQ'D, #	2,424,920	2,679,680	2,459,090	
21	HEAT'G SURF., FT²/# OF PRIM STEAM = 13/10	0.051900	0.40277	0.032425	
22	IF PRIM STEAM REMAINED AS DESIGNED = 10	808,306	808,306	808,306	
23	ADDITIONAL HEAT'G SURF. REQ'D = 11×21	—	3,420	369	
24	HEAT'G SURFACE REQ'D = 13+23	41,951*	39,397*	26,948*	
25	SURPLUS OF HEAT'G SURF., FT² = 5-24	—	2,554	15,003	

TABLE 4.4.3.

OPERATING SCHEME BASED ON 380 PSIA AT PLANT BOUNDARY

ITEM NUMBER	SYMBOLS AND NOTES	3 - EVAPORATOR SCHEME			
	# : LB/HR. * : AT CONDITION WHERE ONE EVAP. IS OUT OF SERVICE ‡ : TUBE BUNDLE SHELL DIA. WITH .75" O.D. TUBING ON 1" D PITCH AT 30° LONG	WITH 2 EVAPORATORS NORMALLY IN SERVICE, 1 EVAPORATOR STAND-BY			
		CONDITION (I)	CONDITION (II)	CONDITION (III)	
	NOTE: UNLESS OTHERWISE SPECIFIED, ALL VALUES ARE FOR ONE EVAPORATOR.	1. $\Delta P_{2 \times 2A\phi}$ = 63 psi 2. EVAP. DES. OPE. P = 486 PSIA (420+63+3) DES. COND'N	1. $\Delta P_{2 \times 2A\phi}$ = 68 psi 2. EVAP. DES. OPE. P = 451 PSIA (380+68+3)	1. $\Delta P_{3 \times 3A\phi}$ = 33 psi 2. EVAP. DES. OPE. P = 416 PSIA (38+33+3)	
1	TOTAL SECONDARY STEAM REQ'D, #	2,285,553	2,285,553	2,285,553	
2	TOTAL HEAT TRANSF. REQ'D, BTU - Q_T	1,753,422,592	1,937,369,485	1,977,885,988	
3	HEAT TRANSF. REQ'D, BTU - Q^*	876,711,296*	968,684,742*	888,942,994*	
4	U - FACTOR, BTU / (HR) (FT²) (°F)	540	540	540	
5	IF USE HEATING SURF. AS DESIGNED, FT², - A^*	62,928*	62,928*	62,928*	
6	HEAT HEAD, °F - θ_m	25.8	33.24	41.29	
7	HEAT TRANSF'D, BTU/# OF PRIM. STEAM	723.085	722.985	722.985	
8	BTU/# OF SEC. STEAM	767.177	847.658	777.880	
9	BTU/FT²	13,932	17,950	22,297	
10	PRIM. STEAM REQ'D # = $\frac{3}{7}$	1,212,460*	1,339,844*	1,229,545*	
11	ADDITIONAL PRIM. STEAM REQ'D, #	—	127,384	17,085	
12	SEC. STEAM PROD'D, #	1,142,776*	1,142,776*	1,142,776*	
13	HEATING SURFACE USED, FT² = $\frac{3}{4 \times 6}$	62,928	53,966	39,868	
14	EST'D TUBE BUNDLE SHELL I.D., ϕ"‡ *	70*	82*	68*	
15	TOTAL HEATING SURFACE REQ'D, FT² = $\frac{2}{4 \times 6}$	125,855	10[illegible],933	79,737	
16	TOTAL HEAT'G SURF. OF 3 EVAP., FT²	125,855*	125,855*	125,855*	
17	OF 2 EVAP., FT²	83,902	83,902	83,902	
18	SURPLUS HEAT'G SURFACE, FT² = 5 - 13	—	8,962	23,060	
19	TOTAL SURPLUS HEAT'G SURFACE, FT² = 16 - 15	—	17,922	46,118	
20	TOTAL PRIM. STEAM REQ'D, #	2,424,920	2,679,680	2,459,090	
21	HEAT'G SURF., FT² / # OF PRIM STEAM = $\frac{13}{10}$	0.051900	0.040277	0.032425	
22	IF PRIM STEAM REMAINED AS DESIGNED = 10	1,212,460*	1,212,460*	1,212,460*	
23	ADDITIONAL HEAT'G SURF. REQ'D = 11 × 21	—	5,131	554	
24	HEAT'G SURFACE REQ'D = 13 + 23	62,928	59,097	40,422	
25	SURPLUS OF HEAT'G SURF., FT² = 5 - 24	—	3,831	22,506	

4.5 EFFECTS OF THE EVAPORATOR ON THE NUMBER OF STEAM DELIVERY PIPE LINES

If we look into the basic heat equation again where $Q = U_D A \theta_m$ and it is assumed that the boundary pressure range is from 380 psia to 420 psia. If it is postulated that U_D is a constant as well as "A" where due to the evaporator is fabricated to the maximum heating surface requirement at its maximum operating conditions. By analysing the cushion pressure available before shut down procedure starts, if based on the designed heat head is $\theta_m = 25.8^oF$, then the effects of the evaporator on heat head due to variable boundary pressure would be as shown in the table 4.5.1. By interpretation the results from this table, it is obvious that there would be a definite surplus of heating surface available in the evaporator at any boundary pressure lower than the designed pressure.

It is therefore concluded that 3 x 24" ø steam delivery lines would provide a definite advantage to the operation of the evaporator than 2 x 24" ø steam delivery lines, particularly when one unit is considered out of service. With 3 steam delivery lines, the 4 evaporator scheme could be operated at a lower pressure since the pressure drop has been calculated at 35 psi while it would increase to 63 psi with only two steam delivery lines. Refer to Fig, 18

The quantity of primary steam required in both cases have also been calculated and shown in tables 4.4.1 through 4.4.3 as in problem discussion of " Effects on number of evaporators required for the system. "

The economic of establishing the total heating surface by taking into account of two and three steam delivery pipe lines has been totally laied upon heating surface which is solely affected by the heat head as shown in the table 4.5.1

TABLE 4.5. 1

EFFECTS OF EVAPORATOR PRESSURE ON HEAT HEAD AND NUMBER OF STEAM DELIVERY PIPE LINES

NOTE: ΔP = 63 psi for 2x24"ϕ steam line; ΔP = 35 psi for 2x24"ϕ steam line; ΔP = 3 psi for separator min. boundary press. 380. psia.	DESIGN COND'N AT: 2x24"ϕ lines	OPERATING CONDITION AT 3x24"ϕ STEAM LINES			
		WITH 40psi	WITH 20psi	WITH 10psi	WITH 0 psi
BUNDARY PRESS., PSIA	420	420	400	390	380
PRIMARY STEAM INLET PRESS. PSIA	620	620	620	620	620
PRIMARY STEAM INLET TEMP. ^{o}F	489.8	489.74	489.74	489.74	489.74
EVAPORATOR PRESS. PSIA	486	458	438	428	418
EVAPORATOR TEMP., ^{o}F	464	458.06	453.57	451.27	448.92
HEAT HEAD θ_m, ^{o}F	25.8	31.68	36.17	38.47	40.82
HEAT TRANSFER CAPACITY % (Based on the designed condition of θ_m= 25.8^{o}F).	1.000%	1.228%	1.402%	1.491%	1.582%

4.6. EFFECTS ON DIFFERENT LOCATIONS OF EVAPORATORS

While a evaporator is located close to the primary steam source, the distance is shorter, therefore the pressure drop in the primary steam line to the evaporator is less. Based on the boundary minimum required pressure is fixed, then the primary steam pressure to the evaporator would be higher, thus, the saturation temperature nevertheless the enthalpy almost remain the same. Due to the difference of the primary steam saturation pressure and temperature to the evaporator is different, at different location while the secondary steam pressure remains unchanged, therefore the heat head is different (Where the heat head is defined as the temperature difference between the primary steam and the secondary steam). The heat head is higher at higher primary steam pressure inlet to the evaporator. That is saying if the secondary boundary pressure is fixed, while the evaporator is located as close as to the Nuclear power station, will result a higher heat head ~~for~~ the evaporator. From the basic heat equation $Q = U_D A \theta_m$ it can be seen that heat head θ_m is one of the determination function for the evaporator heating surface area requirement. If U-factor is a constant, the higher the heat head for the evaporator would result a more economic size of evaporator for the same amount of heat transfer capacity. From the following calculated table 4. 6. 1, it shows that the amount of heating surface can be saved for the evaporator by raising each degree of the evaporator heat head from that particular heat head starting point of $25.8^{o}F$. From the calculated table, based on the $25.8^{o}F$ of heat head, every degree of increment in heat head would save a certain amount of heating surface area. It is plotted as in the Fig. 4.6.1 which is a linear curve. However, it is obvious that from the calculation, starting from $10^{o}F$ difference on top of the $45.8^{o}F$ there would be not much saving in heating surface per degree of heat head increment.

If we assume that there is no pressure drop on the primary steam supply line without any fittings or control devices etc., the maximum primary steam pressure ~~to the~~ evaporator would be 750 psia while secondary ~~side~~ boundary pressure is fixed at 420 psia. Then the maximum heat head can be achieved is $46.84^{o}F$. But practically, there will be actual pressure drop in piping, as well as in all fittings and other control instruments. Therefore the maximum heat head of $46.84^{o}F$ could never be achieved. However, if the pressure drop in the primary steam supply pipe line can be minimized to a minumum, this will eventually save a certain amount of evaporator heating surface.

In order to save the evaporator heating surface, it is necessary to raise the heat head and thus the higher primary steam pressure inlet to the evaporator. However, a side effect would occur, that is while the enthalpy content in the steam has no significant difference from a pressure range within

50 psi in between 750 psia and 650 psia with a small difference of only about 2 BTU per pound of steam, but would have a temperature difference of about $4^{o}F$ for that pressure range at that equivalent saturation pressure. At higher pressure steam would have an approximately 2 BTU per pound less enthalpy content than the lower pressure steam. For this reason, the evaporator at a lower primary steam inlet pressure due to with a higher enthalpy content would require less amount of primary steam to generate the same amount of secondary steam at the same amount of available heating surface.

Another side effect would also occur, that is while the evaporator has a higher pressure primary steam inlet and at a higher flow rate would result a higher temperature of the primary condensate return at the condition of a constant rate of evaporator feed water. It has been calculated that a difference of $6^{o}F$ has been resulted due to the different flow rate of the primary condensate to the cooler/preheater at different pressure. However, the primary condensate returning temperature to the exiting deaerator is still below the existing deaerator operating temperature, therefore, it will not encount any flashing problem.

TABLE 4.6.1

EFFECTS OF HEAT HEAD ON HEATING SURFACE

HEAT HEAD θ_m-°F	HEATING SURFACE REQUIRED SQ. FT.	HEATING SURFACE SAVED SQ. FT.	DIFF. OF FT^2/θ_m °F	HEAT'G SURF. SAVED CUM. SQ.FT.
25.8	31,463	1,174		
26.8	30,289	1,103	71	2,277
27.8	29,200	1,014	89	3,291
28.8	28,186	946	68	4,237
29.8	27,240	884	62	5,121
30.8	26,356	829	55	5,950
31.8	25,527	778	51	6,728
32.8	24,749	733	45	7,461
33.8	24,016	690	43	8,151
34.8	23,326	651	39	8,802
35.8	22,675	617	34	9,419
36.8	22,058	583	34	10,002
37.8	21,475	554	29	10,556
38.8	20,921	525	29	11,081
39.8	20,396	500	25	11,581
40.8	19,896	476	24	12,057
41.8	19,420	454	22	12,511
42.8	18,966	433	21	12,944
43.8	18,533	414	19	13,358
44.8	18,119	395	19	13,753
45.8	17,724	379	16	14,132

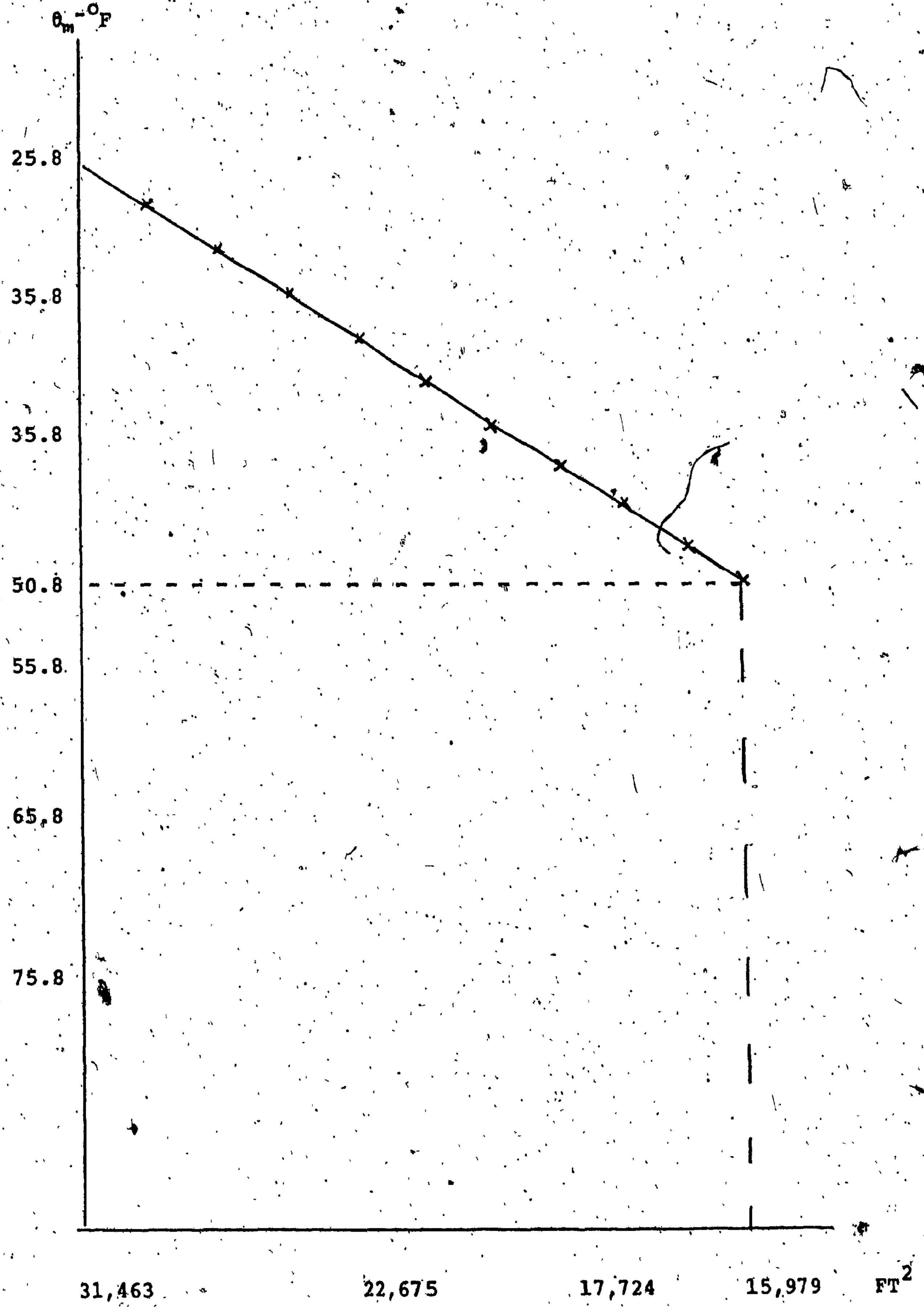
θ_m-°F
25.8
35.8
35.8
50.8
55.8
65.8
75.8
31,463
22,675
17,724
15,979
FT²

FIG. 4.6.1

4.7 SURGE STUDY:

4.7.1. If steam to turbine is interrupted either on long or short duration, the level in the main condenser is dropping and this would call the extraction pump to stop while the deaerator control valve would be on call for (demand) wide open. The water in the reserved tank can only flow to the deaerator at a higher pressure than the deaerator. However, the condenser would have a positive supply from the reserve tank during emergency by gravity. But the extraction pump probably would be too big for the reduced flow.

4. 7. 2 While the SSGS system is added in the original circuit, the primary condensate flow would be only approx. 1/10 of the original condensate flow to the existing deaerator. While the control valve had been sized at normally full flow and it is now only operating at 1/10 of the original capacity. This means the control valve has to be completely resized.

4. 7. 3 The main source of condensate supply to the deaerator is from the SSGS which has 9/10 of the total condensate flow. It should have a control valve on the supply line to the deaerator, otherwise the deaerator would be flooded whereas there is only one 2"ϕ line to the condenser is available which is not automatically controlled.

4. 7. 4. It is therefore suggested that while the control valve to the deaerator is shut, or the level in the deaerator is high, a by-pass to the main condenser should be installed. In this case, the condenser would eventually serve as a surge capacity in complete with the reserved surge tank which is original designed for.

4. 7. 5. As far as the condensate collectors in the SSGS are concern, they are simply adding additional surge capacity to the existing system.

4. 8 SYSTEM DISCUSSIONS

4. 8. 1 Take case 1 as an example, it is assumed that there will be a ΔP = 10 psi across the U-tube. If the evaporator inlet pressure is at 620 psia and the condensate outlet is at 610 psi. If the condensate is not sub-cooled in the condensate collector , the condensate would flash into vapor at that reduced pressure. Since flashing is not desirable in the system, therefore, the condensate have to be sub-cooled at a minimum requirment of 1.8^{o} F in the condensate collector.

An alternative provision may be applied to the flashing problem may occur at the condensate collector is by introducing the primary steam into the condensate collector via a small by-pass line from the evaporator primary steam inlet.

4.8.2. In order to alleviate the condensate flashing problem in the system, a pressure control valve is therefore required at the condensate collector vent outlet. The condensate collector in the system serves as the purpose of a combination of a drainer and a flash tank which are commonly used in power plants. An automatic stop valve works on low level signal has to be installed donwstream of the condensate collector to permit continuous flow of condensate but no steam. For this kind of arrangement, the two downstream cooler -/preheaters which are originally designed for condensate/water heat exchanger are being protected by this valve.

4.8.3. Since there is no pump in the system, it is therefore no provision is required for a big surge tank which is normally provided for pump suction.

4.8.4. Since the condensate pressure is lower than the steam pressure at the temperature, any objectionable gases are preliminary removed off the top of the condensate collector. The most objectionable gases are those being carried along with the primary steam including oxygen which can cause severe corrosion in the steam drum and piping.

4.8.5. As water is evaporated in the evaporator, most of the impurities from the feed water remain in the evaporator. Therefore the evaporator water concentration will increase gradually by the increasing of operation duration. The only way to reduce the concentration is by discharging some of the evaporator water to the blowdown system to waste drainage. Both continuous and intermittent blowdown are needed. Since water at high temperature is reduced in pressure, part of the water will flash into steam. A blowdown tank is therefore required to accommodate all the intermittent blowdowns from the entire system while the continuous blowdown is directed to the open drainage.

4.8.6. A heat exchanger called blowdown heat exchanger may be suggested for the continuous blowdown system in order to recover the heat energy loss. Feed water flows inside the tubes of the heat exchanger, and evaporator blowdowns flow outside of tubes of the heat exchanger. Thus, most of the heat energy in the blowdown is recovered. Refer to Appendix X.

4.8.7. Since SSGS is a system that is branched out from the main turbine supply line. In the original circuit, the main condenser is originally designed with a capacity of accomodate the full turbine load in the event of turbine trips. It is therefore no such additional needs of surge capacity which has already been automatically taken care of for the SSGS system in the entire circuit.

4.8.8. During the continuous operation, a vent system is provided to remove any entrained gases at the top of the condensate collector to the main condenser. Another dumping system is required that is an automatic control system which will open whenever ~~the main~~ condensate line to the existing deaerator is blocked off while a situation of high level is sensed and occured in the existing deaerator. In order to maintain a continuous flow of the system in the event of an high level is sensed at the existing deaerator while the main condensate return line is instaneous interrupted, the automatic control system would create an override signal to open the valve and dump the condensate to the existing main condenser continuously and maintain the entire system remain in operation.

4.8.9. During the short and long duration shut down, two manual reset type drain systems are required for the entire system. One is between the condensate collector and the cooler/preheater No. 1 and the other is at the downstream of the cooler/preheater No.2. All drains are dumped into the main condenser as well. In this way, the entire system can be emptied partially and completely depends on operation needs.

4.9. POSSIBLE HEAT RECOVERY FROM EVAPORATOR BLOWDOWN:

Since blowdown is required for the saturation water in the evaporator, and the saturated water has a great amount of sensible heat content without any provision of heat recovery otherwise a heat energy loss would be culmulated to a great amount. Blowdown is a waste but a necessary to maintain the water quality. But heat energy can be recovered by adding an heat exchanger to utilize the available heat in the contionuous blowdown otherwise would be a loss, to preheat the treated water to the deaerator. By this provivision, a great amount of steam can be saved.

Fefer to FIG. 10. and heat balance tables in relation to the calculations in Appendix X.

CHAPTER 5

HEAT BALANCE AND PIPING DESIGN

5.1. HEAT BALANCE.

The method employed for calculating heat balance in this dissertation is carried out by the application of law of conservation of energy which states that the energy entering into an equipment or a system must equal to the energy leaving the equipment or system. Heat is the energy in this case.

The primary steam from the reactor steam drum enters the evaporator, and the primary condensate collector, then pass through the series of heat exchangers and is finally returned to the original circuit. The heat energy will be always in a balance state in the complete cycle as well as for each individual equipment. This has been the basic theorey used to calculate the secondary steam production, primary steam requirement and condensate return, as well as treated water requirement based upon assumptions made on evaporator blowdowns, and heat loss as as well as calculated treated water supply deaeration pegging steam requirement.

In the evaporator, the total energy input from the primary steam is equal to the total heat energy leaving, such as enthalpy gained in the evaporator feed water then vaporizes into steam and carries away heat energy ; on the water side, some heat loss is encounted in the blowdowns, samples and some other small minor heat losses.

The enthalpy contained in the primary condensate is assumed no heat loss involved in the primary steam condensate collector inlet and outlet.

Great sensible heat is exchanged at the two heat exchangers. Heat is given up from the primary steam condensate to the treated water.

It is assumed all equipment and piping are well insulated and in an enclosed building. Heat loss through insulation by radiation and conduction or even possible convection through small distances are assumed negligible.

Four heat balance tables have been made available. Details can be referred to tables 5.1 through 5.4 in relation to FIG. 5 Heat balance calculations have been based upon the heat energy conservation law and depending on the two following variances as study cases:

1. Number of secondary steam delivery lines in service.
2. Location of the SSG system.

The following are the general assumptions and information sources which have been made as basic references for the calculation of the four attached tables:

1. Babcock and wilcox steam table-1969.
2. ASME steam table -1967.
3. 1% blowdown is allowed on secondary side.
4. ½% heat loss is allowed on primary steam
5. Heat loss through pipe line is estimated.
6. Boundary pressure is based on a range of 380 to 420 PSIA.
7. The following pressure drops have been estimated based on industrial practice:

ΔP 30 in. dia. pipe line = 7 psi
(for Table 5.1 and 5.2)

ΔP 30 in. dia. pipe line = 24 psi
(for Table 5.3 and 5.4)

ΔP evaporator U-tubes = 10 psi

ΔP cooler / preheater shell side = 16 psi

ΔP steam separator = 3 psi

ΔP control valve on treated feed water inlet = 40 psi

For details of heat balance, references can be made to the calculations as evaporator, pegging steam for the deaerators in Appendix V, VII, VIII, IX, and X

5.2. PIPING DESIGN

Main piping systems have been prelimary sized according to the maximum flow rate as shown in the heat balance tables. From annual cost curve for piping FIG-1-4 and table 2-6 of power plant theory ref.(27), the design velocity is restricted to a maximum of 150 FPS for steam and 12 FPS for condensate and water for economic reason. Pressure drops for steam and water through various size of pipe diameter at different flow rates have been calculated on the basis of 100ft by the application of Darcy formula. Turbulent flows are all verified by calculating of Reynolds numbers. Results have been shown in the heat balance table for convenient reference. Nuclear power piping code ANSI B31.7 has been used for the calculation of pipe wall thickness for both the primary and secondary side. A corrosion allowance of 0.0625 inch, and a manufacturers' allowance of 0.125 inch of tolerance have been allowed in the piping calculation. Detailed calculations for piping can be refered to piping calculations in the appendix V. In general, A 106 GR.B seamless carbon steel piping and A155 KC-70 class 1 or 2 carbon steel piping are recommended here for the piping of 24 inches diameter and smaller, and 26 inches diameter piping and greater respectively.

TABLE 5. 1

HEAT BALANCE AND PIPING DESIGN

CASE 1 : LOCATION 1 WHILE 2 X 24'' ϕ STEAM DELIVERY LINE IS ALLOWED ΔP = 63 psi.

◇ NO.	FLOW LB/HR.	PRESS PSIA	TEMP. °F	ENTHALPY BTU/LB.	PIPE DESIGN NOM. SIZE INCH	VELOCITY FPS	$\Delta P_{100 FT}$ PSI
1	1,418,800	770	513.8	1199.72	24	92.08	0.676
2	206,340	750	510.8	1199.72	24	13.77	0.015
3	1,212,460	660	496.6	1199.72	20	133.78	1.460
4	606,230 808,306*	653	495.4	1199.72	18	80.73 107.64	0.859 1.527
5	606,230 808,306*	653	495.4	1199.72	18	80.73 107.64	0.859 1.527
6	303,115 404,153*	620	489.8	1199.72	12	84.98 113.30	1.330 2.365
7	303,115 404,153*	610	488.0	473.7	8	5.36 7.14	0.341 0.606
8	606,230 808,306*	610	488.0	473.7	10	6.55 8.73	0.470 0.836
9	606,230 808,306*	594	268.5	237.3	10	5.57 7.42	0.325 0.577
10	606,230 808,306*	112	268.5	237.3	10	5.57 7.42	0.325 0.577
11	2,424,920	67	268.5	237.3	18	7.78	0.296
12	216,671	67	345.0	1199.72	12	55.31	0.658
13	98,005	750	510.8	1199.72	2x12	23.50	0.093
14	196,009	—	—	—	—	—	—
15	196,009	67	200	167.99	12	1.28	0.013
16	2,837,600	795	333.9	306.0	16	12.69	0.705
17	571,388 761,851*	486	464.0	1204.7	20	78.52 104.69	0.533 0.948
18	2,285,552	484	464.0	1204.7	30	138.15	1.002
19	640,000 960,000**	420	223.0	1202.3	24	60.81 91.22	0.277 0.623
20	2,308,408	564	223.0	191.16	16	8.77	0.55
21	577,102 769,469*	530	458.0	191.16	8	7.75 10.33	0.812 1.444
22	577,102 769,469*	486	464.0	439.3	8	7.75 10.33	0.947 1.684
23	366,295	484	464	1204.7	12	121.74	2.134
24	366,295	18	464	1204.7	—	—	—
25	5,714 7,618*	486	464	446.04	$1\frac{1}{2}$	—	—
26	4,645 6,967*	484	464	1204.7	—	—	—
27	13,934	484	464	1204.7	—	—	—
28	1,942,113	18	32	—	—	—	—

* 3 UNIT IN OPERATION

** 2 x 24" ϕ STEAM DELIVERY PIPE IN OPERATION

TABLE 5. 2

HEAT BALANCE AND PIPING DESIGN

CASE 1 : LOCATION 1 WHILE 3 X 24'' ϕ STEAM DELIVERY LINE IS ALLOWED. ΔP = 35 psi

◇ NO.	FLOW LB/HR.	PRESS PSIA	TEMP. °F	ENTHALPY BTU/LB.	PIPE DESIGN NOM. SIZE INCH	VELOCITY FPS	$\Delta P_{100\,FT}$ PSI
1	1,418,800	770	513.8	1199.72	24	92.08	0.676
2	186,921	750	510.8	1199.72	24	12.47	0.012
3	1,231,879	660	496.6	1199.72	20	135.92	1.508
4	615,939 821,252*	653	495.4	1199.72	18	80.01 109.35	0.887 1.576
5	615,939 821,252*	653	495.4	1199.72	18	80.01 109.35	0.887 1.576
6	307,969 410,626*	620	484.74	1199.72	12	86.35 115.13	1.373 2.441
7	307,969 410,626*	610	487.98	473.8	8	5.43 7.24	0.352 0.626
8	615,939 821,252*	610	487.98	473.8	10	6.64 8.85	0.485 0.863
9	615,939 821,252*	594	280.63	250.87	10	5.69 7.58	0.335 0.596
10	615,939 821,252*	112	280.63	250.87	10	5.70 7.60	0.335 0.596
11	2,463,758	67	280.63	250.87	18	7.96	0.306
12	181,657	67	345.0	1199.72	12	54.30 95.39	0.636 1.963
13	96,093	750	510.8	1199.72	2 X 12	23.04	0.089
14	192,185	—	—	—	—	—	—
15	192,185	67	200.	167.99	12	1.26	0.013
16	2,837,600	795	333.9	306.0	16	12.69	0.705
17	571,388 761,851*	458	458.06	1204.8	20	83.35 111.13	0.56 1.007
18	2,285,552	455	457.39	1204.8	30	147.62	1.071
19	640,000	420	449.40	1204.7	24	70.41	0.277
20	2,308,408	536	223	192.39	16	8.76	0.515
21	577,102 769,469*	502	223	192.32	8	7.74 10.32	0.812 1.444
22	577,102 769,469*	458	450	430.25	8	8.97 11.96	0.940 1.671
23	366,264	455	457.39	1204.8	12	130.84	2.268
24	366,264	18	457.39	1204.8	—	—	—
25	5,714 7,618*	459	458.06	439.2	$1\frac{1}{2}$	—	—
26	4,590 6,885*	455	457.39	1204.8	—	—	—
27	13,769	455	457.39	1204.8	—	—	—
28	1,942,144	18	32	—	—	—	—

* 3 UNIT IN OPERATION

** 2 x 24" ϕ STEAM DELIVERY PIPE IN OPERATION

TABLE 5.3

HEAT BALANCE AND PIPING DESIGN

CASE 2 : LOCATION 2 WHILE 2 X 24'' ϕ STEAM DELIVERY LINE IS ALLOWED. ΔP = 63 psi

◇ NO.	FLOW LB/HR.	PRESS. PSIA	TEMP. °F	ENTHALPY BTU/LB.	PIPE DESIGN NOM. SIZE INCH	VELOCITY FPS	$\Delta P_{100 FT}$ PSI
1	1,418,800	770	513.8	1199.72	24	92.08	0.676
2	211,856	750	510.8	1199.72	24	14.14	0.016
3	1,206,944	660	496.6	1199.72	20	133.17	1.448
4	603,470 804,628	636	492.5	1199.72	18	81.81 109.08	0.870 1.547
5	603,470 804,628	636	492.5	1199.72	18	81.81 109.08	0.870 1.547
6	301,735 402,314	603	486.7	1199.72	12	87.08 116.10	1.357 2.412
7	301,735 804,628	593	484.9	470.1	8	5.31 7.08	0.339 0.602
8	663,470 804,628	593	484.9	470.1	10	6.50 8.66	0.468 0.832
9	603,470 804,628	577	263.9	232.7	10	5.53 7.37	0.321 0.570
10	603,470 804,628	112	263.9	232.7	10	5.53 7.37	0.321 0.570
11	2,413,887	67	263.9	232.7	18	7.72	0.293
12	228,175	67	345.0	1199.72	12	55.31 95.37	0.659 1.960
13	97,769	750	510.8	1199.72	2X12	23.45	0.092
14	195,538	—	—	—	—	—	—
15	195,538	67	200.	167.99	12	1.28	0.013
16	2,837,600	795	333.9	306.0	16	12.69	0.705
17	571,388 761,851	486	464.0	1204.7	20	78.52 104.69	0.533 0.948
18	2,285,552	484	464.0	1204.7	30	138.15	1.002
19	640,000 960,000**	420	449.0	1202.3	24	60.81 91.22	0.277 0.623
20	2,308,408	564	223.0	191.16	16	8.77	0.515
21	577,102 769,469	530	223.0	191.16	8	7.75 10.33	0.812 1.444
22	577,102 769,469	486	458.0	439.3	8	7.75 10.33	0.947 1.684
23	366,264	484	464.0	1204.7	12	155.5	2.134
24	366,264	18	464.0	1204.7	—	—	—
25	5,714 7,618	486	464.0	446.04	1½	—	—
26	4,645 6,967	484	464.0	1204.7	—	—	—
27	13,934	484	464.0	1204.7	—	—	—
28	1,942,43	18	32	—	—	—	—

* 3 UNIT IN OPERATION

** 2 x 24" ϕ STEAM DELIVERY PIPE IN OPERATION

TABLE 5. 4

HEAT BALANCE AND PIPING DESIGN

CASE 2 : LOCATION 2 WHILE 3 X 24"ϕ STEAM DELIVERY LINE IS ALLOWED. $\Delta P = 35$ psi

◇ NO.	FLOW LB/HR.	PRESS PSIA	TEMP. °F	ENTHALPY BTU/LB.	PIPE DESIGN NOM. SIZE INCH	VELOCITY FPS	ΔP_{PIPE} PSI
1	1,418,800	770	513.8	1199.72	24	92.08	0.676
2	192,483	750	510.8	1199.72	24	12.85	0.013
3	1,226,317	660	496.6	1199.72	20	135.31	1.495
4	613,158 817,544*	636	492.5	1199.72	18	83.12 110.83	0.898 1.597
5	613,158 817,544*	636	492.5	1199.72	18	83.12 110.83	0.898 1.597
6	306,579 408,772*	603	486.73	1199.72	12	88.47 117.96	1.401 2.490
7	306,579 408,772*	593	484.94	470.23	8	5.39 7.18	0.350 0.682
8	613,158 817,544*	593	484.94	470.23	10	6.59 8.78	0.483 0.859
9	613,158 817,544*	577	276.16	246.29	10	5.66 7.54	0.331 0.589
10	613,158 817,544*	112	276.16	246.29	10	5.66 7.54	0.331 0.589
11	2,452,633	67	276.13	246.29	18	7.90	0.303
12	193,438	67	345.0	1199.72	12	54.40 95.39	0.637 1.959
13	95,765	750	510.8	1199.72	12	22.96	0.088
14	191,529	—	—	—	—	—	—
15	191,529	67	200	167.99		1.25	0.013
16	2,837,600	795	333.9	306.0		12.69	0.705
17	571,388 761,851*	458	458.06	1204.8	20	83.35 111.13	0.566 1.007
18	2,285,552	455	457.39	1204.8	30	147.62	1.071
19	640,000	420	449.40	1204.7	24	70.41	0.277
20	2,308,408	536	223.0	192.39	16	8.76	0.515
21	577,102 769,469*	562	223.0	192.32	8	7.74 10.32	0.812 1.444
22	577,102 769,469*	458	450.0	430.25	8	8.97 11.96	0.947 1.684
23	366,264	455	457.39	1204.80	12	130.84	2.268
24	366,264	18	457.39	1204.80	—	-	-
25	5,714 7,618 *	458	458.06	439.2	$1\frac{1}{2}$	-	-
26	4,590 6,885 *	455	457.39	1204.80	—	—	—
27	13,769	455	457.39	1204.80	—	—	—
28	1,942,144	18	32	—	—	—	—

* 3 UNIT IN OPERATION

** 2 x 24" ϕ STEAM DELIVERY PIPE IN OPERATION

CHAPTER 6

EQUIPMENT DESCRIPTION AND EQUIPMENT DESIGN

6.1 EQUIPMENT DESCRIPTION

6.1.1 SECONDARY STEAM GENERATOR (EVAPORATOR)

The secondary steam generator could be either vertical or horizontal, shell and tube heat exchanger. It has been experienced that in most power plants and petrochemical plants, the horizontal kettle type reboiler is the most economical and common evaporator in use. As far as the tube arrangement is concerned, in order to have a more rigid and safe construction to meet the requirements of nuclear vessel code, all welded construction is enforced. For simple construction, and consideration of better heat transfer efficiency, U-tube bundle with welded tube sheet is suggested.

According to ASME codes and specifications, the evaporator will be of all welded carbon steel construction with incoloy 800 tubes built to the appropriate ASME nuclear code. Access manways will be provided in the tube channels and shell side to permit the routine checking and cleaning or even for plugging of defective tubes which should have very little chance to occur due to extremely careful and stringent design criteria is followed in accordance with the nuclear codes.

Inside the closed kettle, two bundles of U- tubes which are mounted axially in the lower section of each end of the evaporator in which high pressure steam is condensing and the water in the shell is evaporating as a product of secondary steam.

Venting system may be provided for the primary steam and condensate, but has to be designed in an enclosed circuit due to radioactive preventative requirements.

6.1.2 PRIMARY CONDENSATE COLLECTOR

For the main safety reasons, in order to minimize the energy release in the event of catastrophic failure; a small collector would have less total energy release than a bigger one, in addition to this, transportation and construction would be generally simpler. It is proposed one small condensate collector rather than a bigger one. However this vessel would serve as a combination of a drainer and a flash tank of the condensate system. Small line connections such as vent, equalizing and dumping lines may be required in relation to operation.

This vessel will be all welded carbon steel construction to ASME nuclear codes and material specifications.

In normal operation, the operating level is set at middle of the tank. The other half of the tank volume is providing a sufficient space for gas disengagement, i.e. NH_3, Air, N_2, and H_2.

6.1.3 COOLER/PREHEATER

Cooler/preheater is actually a typical shell and tube heat exchanger with primary condensate returning to the original cycle in the tube side, and with treated feed water to the evaporator in the shell side.

There are five main following reasons to place the primary condensate on the tube side and treated water on the shell side:

1. Higher pressure drop has been calculated on the tube side.
2. Original cycle demands lower primary condensate return pressure.
3. To save evaporator feed water pumping power.
4. To the best utilization of maximum tube side pressure drop.
5. Do not require expansion joint on shell side.

The cooler/preheater will be all welded carbon steel construction with incoloy -800 tubes and shall construct to ASME nuclear codes and ASME specifications. Manways or manholes shall also provided for the access of tube checking cleaning or plugging for leakage.

6.1.4 IN-LINE SEPARATORS

In order to maintain the secondary steam quality while transporting in a long distance during severe cold winter, moisture must be removed on the steam delivery pipe lines to prevent from freeze off. It is therefore required in-line separators along the pipe line. All moisture shall be collected in the in-line separators and gathered in a condensate vessel. From which the condensate may be pumped back to the secondary system. This problem would require more additional detailed pipe line engineering studies. It is not the intention of this paper to go into any details in this regards.

6.1.5 CONDENSATE MIXING TANK

The condensate mixing tank is designed to mix two streams of primary condensate at different temperatures before they enter into the exising deaerator. This provision would provide a more uniform or constant inlet condensate temperature to the existing deaerator.

The condensate mixing tank will be a very small vessel, it could be only a two-size greater in diameter than the bigger inlet stream line size and with a length of not more than six feet. The tank of course should be all welded carbon steel and constructed to the ASME nuclear code and material specifications.

6.1.6 PIPING SYSTEM

On the primary side, while the secondary steam generating station may be located at a certain distance from the original nuclear reactor power generating station. The primary piping shall consist the steam line to the nuclear power station and the primary condensate returned from the secondary steam generating system.

On the secondary side, the piping shall consist of the secondary steam outlet lines and the feed water supply lines plus some drain or dumping lines as well as some chemical injection lines and miscellaneous piping in the system where it does not contain any radioactive fluid.

As far as the complete piping system is concerned, it is again not the intention of this paper to go into details. However it can be specified that the primary piping shall be designed and constructed in accordance to ANSI nuclear power piping code; and the secondary piping shall be designed to the ANSI B 31.1 power piping code. The design pressure and temperature of the primary side shall be the same as for the nuclear steam drum condition. Whereas design pressure and temperature of the secondary steam piping shall be the same as the evaporator; and the feed water piping shall use the pressure of the feed water pump discharge pressure. Calculations and results can be referred to Appendix V. 2

6.2. EQUIPMENT DESIGN:

In this section, the equipment design are concentrated on wall thickness calculation. Formulaes have been used are from both ASME pressure vessel code section VIII division 1 ref. (30) and standards of tubular exchanger manufacturers association ref. (30) have been used. Material have been selected as SA 515 grade 70 for high temperature steam and water services. Maximum allowable stress has been used according to the material selected.

Sizes have been based on the calculation made in appendix VI, VII, and VIII.

Design pressure and temperature has been based on the primary steam condition same as the nuclear steam drum for primary side and the possible maximum operating pressure and temperature for the secondary side. refer to table 6.2.1

Details and calculations can be referred to appendix VI.

TABLE 6. 2. 1
EQUIPMENT DESIGN PRESSURE

110% OF THE MAXIMUM OPERATING PRESSURE AS BASIS.

Primary Side	Evaporator channel tube tube sheet	850 PSIG
	Cooler/Preheater channel tube tube sheet	
	Condensate Collector	
Secondary Side	Evaporator shell Cooler/Preheater Shell Blowdown Tank	625 PSIG

CHAPTER 7

STEAM GENERATOR (EVAPORATOR) DISCUSSION

The determination of overall heat transfer coefficient U_D has been a problem. By examination of the general heat equation

$$Q = U_D A \Delta t = U_D A \theta_m \qquad (7.1)$$

in which, area A and heat transfer coefficient U_D are both the unknown determinative functions in the equation. An effort of searching for U_D is therefore required. In the evaporator calculations, kettle type reboiler with two U-tube bundles is used because of economical reason. Heat transfer coefficient is calculated as $U_D = 540$. Details of calculation can be referred to Appendix VII.

Isothermal boiling is taking place in the heat process. When vaporizing liquids from pools, extremely high maximum flux may be obtained. For Water, a theoretical maximum flux of 400,000 BTU / (HR) (FT^2) (^oF) may be obtained at critical temperature difference for perfect clean heating surface in laboratory test. Maximum flux only occurs at critical temperature difference and it is a limitation of maximum heat transfer coefficient. Beyond the critical temperature difference, both the coefficient and flux decrease.

The decrease being due to formation of a layer of vapor blanketing. Therefore it is necessary to restrict the critical temperature difference and flux to an allowable safe value. Details of calculations can be referred to Appendix VII and results of calcutions are shown in Table VII.1, VII. 2 and VII. 3. in Appendix VII.

In approaching for the evaporator calcutions, references (21) (27) have been consulted and the following procedures have been used:

7.1 Determination of heat transfer coefficient U_D :
Based on the designed process flow conditions, and constraints. Nemerous trial and error caculations such as preheat, vaporization, LMTD, maximum FLUX, overall clean heat transfer coefficient, and pressure drops etc.,

7.2 Determination of heating surface required:
Requirement of the heating surface is varied while at various operating conditions. However the heating surface requirement for the evaporator is sized at it's maximum designed conditions.

7.3 Determination of steam relieving velocity and evaporator diameter:
The size of the evaporator is governed by the steam relieving velocity. It has been claimed by the industry

that a restricted maximum steam relieving velocity is 0.25 fps. Beyond this limit, the quality of the steam would be significantly affected by the carry over.

Relieving rate is determined by the steam produced at given water surface area, and the water surface area is maximum, at the centre line of the evaporator. Therefore the steam relieving rate is the basic determinative function of the evaporator size.

While the pressure is varying, the fluctuation of pressure will cause the fluction of the water level, thus, the steam relieving rate. Therefore while sizing an evaporator, many considerations on allowance shall be made.

Evaporator steam relieving velocity should be lower at high vapor pressure because of the greater vapor density at high vapor pressures. Evaporators have given much trouble in high pressures than lower pressures when shell pressures fluctuate from practical industrial experiences.

CHAPTER 8

COOLER / PREHEATER DISCUSSION

Sizing of cooler / preheater has been the same problem as sizing of evaporator because of two unknown determinative functions U_D and A. Fortunately, it is recommended by the Power Plant Theorey and Design that a heat transfer coefficient of 350 BTU / (HR) (FT^2) (^oF) are generally used for designing of drain cooler. In the calculation, method of trial-and-error is used . U_D = 300 has been tried, but the final result is U_D = 350.

One unit has been tried out as base calculation, however two units connected in series in a counter flow arrangement is recommended because of its simplicity in fabrication and economical reasons.

Calculation has been made for two cases of locations - case 1 and case 2. Finally, case 1 with a evaporator pressure of 486 psia is determined as the basis of designing condition.

Details of calculation can be referred to cooler / preheater calculation in Appendix VIII. Results are shown in Tables 8.1 and 8.2

TABLE 8. 1

SUMMARY OF THE RESULTS

	UNIT NO. 1	UNIT NO. 2	TOTAL (average)
h OUTSIDE	h_{io} = 899 h_o = 2447	h_{io} = 918 h_o = 2906	
U_c CALCULATED	657	698	[redacted]78)
U_D USED	350	350	350
R_d CALCULATED	0.000864	0.000953	(0.000909)
R_d REQUIRED	0.000500	0.000500	0.000500
r_w REQUIRED	0.000471	0.000471	(0.000471)
ΔP CALCULATED	SHELL SIDE 0.382 psi TUBE SIDE 7.752	SHELL SIDE 0.420 psi TUBE SIDE 11.391 psi	SHELL SIDE 0.802 psi TUBE SIDE 19.143psi
ΔP ALLOWED	SHELL SIDE 5 psi TUBE SIDE no limit	SHELL SIDE 5 psi TUBE SIDE no limit	

TABLE 8. 2

SUMMARY OF COOLER / PREHEATER

SURFACE AREA REQUIRED	UNIT NO. 1	UNIT NO. 2
SQ.FT. PER UNIT	10,251	7316
NO. OF PASSES SHELL SIDE	3	3
NO. OF PASSES TUBE SIDE	6	6
SHELL I. D.	44	37
U_D DETERMINED	350	350

CHAPTER 9

PROCESS INSTRUMENTATION CONSIDERATION

The process instrumentation for the best control could be either pneumatic or electronic. The advantages and disadvantages between these two types are compensating each other. The main feature between these two types that I could think of are as following:

(1) For pneumatic instrument, it generates a signal from a signal source to the transducer, then send the signal to the controller; and from the controller, then convert into a greater output to operate the control valve or to the indicator. It may take a longer time in transporting of a signal compare to the electronic device. This is saying that the response of the pneumatic instrument would not as fast as the electronic instrument.

(2) The cost of the pneumatic instrument is much less than the cost for the electronic instrument.

(3) No frozen off problem in case of long term shut down that may happen for the electronic instrument.

(4) Maintenance cost for the pneumatic instrument would be higher than the electronic instrument. But it is much simpler to repair a pneumatic device than electronic ones.

(5) More reliable on electric device than pneumatic.

From the above, it is considered that as far as the safety consideration for any nuclear plant is concerned, it would be better to choose the use of electronic rather than pneumatic. But on the other hand, from the point view of economic, pneumatic instrumentation would have its advantages.

No matter which instruments are chosen. For any kind of control system, all selected instruments or devices shall meet the recognized standards and special operating requirements according to nuclear codes and specifications. All appropriate controls and apparatus for start-up and shut-down shall also be provided as well as other types of continuously operational instruments and displays are employed to continuously monitor the performance and controlling of the system. Complete visual indicators, set points, alarms, and other monitoring device shall also be employed in the system to warn of abnormal conditions.

Several points are made in the following for further consideration :

(1) Pressure reducing valves should be installed on the condensate return line .

(2) Control valves shall not be oversized. Also the valves should not be expected to "Kill Off" excessive pressure. This could result in cavitation and physical damage.

(3) Level controllers shall be adjusted to operate over a reasonable range to allow good valve modulation. Quick opening tends to collapse the steam or cause hammering and vibration with resultant damage to internals.

(4) Pressure controllers shall be operated as possible to operating pressure.

(5) Sensing elements and transmitting devices and their attachments shall be stainless and reliable.

Two recommendations may be made to the existing cycle :

(1) To modify the existing controlling device.

(2) Add an additional small control valve with small load capacity in parallel to the existing cycle because there will be a tremendous flow or pressure reduction in the existing cycle while the main flow is diverted to the SSG System.

For control scheme refer to Fig. 6 and Fig.7 and it is described as below :

While the SSGS is in operation, there will be a need of a great amount of the primary steam, this is means that there will be only a small limited amount of steam available to the turbine simply to maintain the station load. At this condition, the original big control valve would be too big for the small amount of flow to the turbine. Two methods may be suggested as in (1) and (2). In (1) the complicated control system for big valve and it's relevant instruments such as servos oil system etc., are required for modification. It is therefore much complicated and costful. However in (2), there is only a small control valve to be added in parallel with the original big control valve and utilize the original control system. It is therefore more practical and economical to adopt the (2) rather than (1). The reason for adding the small valve is that the original control valve becomes oversized and an unstable operation will be resulted while the load is reduced from original full flow to almost a minimum load. Since the SSG system is now on demand of the primary steam rather than turbine, the turbine is taking whatever primary steam is left. In normal operation, the small control valve would eventually become the continuous control of the turbine, but in an event such as a reduced primary steam demand from the SSG system, then, the flow rate exceeds the capacity of the small control valve. It will then automatically switch to the original big control valve to control the flow to the turbine and shut off the small valve at meantime. In this case, the orignial turbine cycle would serve as part of the energy absorber of SSGS, and the main condenser which is originally designed for the full load dumping is now become the surge reservoir of the whole system. It is expected that in case of a sudden shut down or trip off either happened at turbine or SSG system, the by-pass reducing valve would open to dump the excessive steam to the main condenser. But during the normal operation, an override control may also required to dump any excessive steam to the main condenser while the flow exceeding the capacity of the small control valve and before the big control valve take's over or in order to maintain the constant stable small station load. The small control valve may be identical in design and duty and it's relevant control sources to the existing one. But it shall operate at a small load at the similar condition of the big one.

It has been learned that the output pressure of the reactor steam drum steam has to be very stable, otherwise the reactor trip-out would occur because of pressure unstability problems. It is therefore emphasized here that the reliablity of pressure control system on the primary side is very important. Any defects in pressure transient would lead to poison injection.

A detailed-proposed mechanical and instrumentation flow diagram has been developed as shown in Fig. 6 and Fig. 7

CHAPTER 10

SYSTEM OPERATION AND DISCUSSION

PRIMARY SIDE:

Although the flow and pressure of the secondary steam is controlled by the secondary steam generator primary steam inlet control valve. It is saying that the secondary steam pressure and flow rate is actually the determinative function of the flow rate and pressure of the primary steam. The control valve throttling of the primary steam will directly respond to the variation of the secondary steam requirement. While there is a reduction in the secondary steam demand, the control valve with throttle tends to close, therefore, reduces the mass flow of the primary steam on secondary steam generator tube side. It is assumed that while the pressure is raised due to this kind of situation, the turbine generator in the original cycle would automatically absorb the additional load exerted by the SSGS. In other words, the original tubine or the main condenser would have to be able to absorb the variation flow in order to maintain a constant pressure in the reactor steam drum.

It is also assumed that in the event of sudden trip or due to emergency shut down of the SSGS, the designed amount of primary steam in the SSGS would be able to dump into the main condenser without affecting the normal operation of the original turbine cycle.

Normally a substantial increase in the secondary steam would be due to predictable cold weather. It is expected that the amount of steam produced by the reactor would have been raised and the original turbine generator cycle have been adjusted to produce the secondary steam peak demand. In case of an sudden increase on the secondary steam demand, an override signal will be transmitted to the secondary steam generator throttle valves which restrict the flow to maintian the appropriate pressure in the reactor steam drum.

The condensate collector has its own independent level control which will ensure that it will only operate at designed condensate level. However, if there is such a need, it could be emptied completely by pressure without affecting the level control of the deaerator.

Pressure equalizing lines are provided which connects the steam space of the condensate collector with the channel chamber of each U-tube bundle inlet of the evaporator. This provision will provide the pressure equalization between the secondary steam generator and the condensate collector. This also provides the balanced duty on both U-tube bundles, and prevents the condensate from flashing into steam.

It is assumed that the level controller on the main condenser will function properly and would not have effects due to the addition of the SSGS.

The reserved storage tank is also assumed existing in the primary circuit. In the event of high level occured in the existing deaerator due to occasionally high primary steam demand in the operation, the condensate will be automatically dumping into the reserved storage tank or into the main condenser. Or it will supply water by gravity to the condenser while there is a low level is occured in the condenser.

This overall control concept will thus ensure that the inventory of the condensate is still within the closed circuit without any loss among the SSGS. In addition, the condensate collectors in the SSGS will serve as an additional extra surge capcity to the complete system including original circuit and SSGS as a whole.

SECONDARY SIDE:

On the secondary side, in the event of water supplying pumps are working against closed isloation valves on the pump discharge side, the water could be recirculated back to the deaerator in the water treatment plant. This will be a typical condition of start-up procedure.

As secondary steam production commences, the control valve in the SSGS will gradually open and the water from the pumps will be diverted from the recirculation loop to the SSG system.

In normal operation, the control valves on the primary steam line to the secondary steam generator is about 75% open which is designed for full load operation. It is assumed that the secondary steam pressure and quantity demand is stable in the winter time; however due to the weather change is sometimes unpredictable, therefore the actual operation is sometimes in fluctuation rather than stable during a short period.

As the secondary steam demand varies, the pressure of the steam delivered to the steam line header will vary in accordance with the varying pressure drops in the transmission piping, fluction is occured. In order to smooth out the operation for the whole SSG system, an additional horizontal vessel is suggested to substitute the use of a common header. Any surge or pulsation or instant demand for the steam would therefore be evened out, since a minimum pressure required at the maximum steam demand is set at the secondary steam supply reserve tank. A greater energy storage is therefore available with this method of operation. Since it is rather difficult to maintain a constant pressure at down stream of the steam supply reserve tank; therefore the secondary steam supply shall be controlled and maintained constant at upstream of the steam supply reserve tank. This is saying that the pressure controlling point should be at somewhere between the SSG system and the secondary steam supply reserve tank.

The steam generator may require some kind of water conditioning treatment in the steam generator. Although the water feed to the secondary steam generator is assumed has already been treated, however still the chemical injection provision to the secondary steam generator water is required as well as the blow down system. As far as the control of these two operations is concerned, they are only a kind of conventional routine procedure. The purpose is to control the P H of the water in the steam generator.

CHAPTER 11

CONCLUSIONS

This dissertation presents a prosposed method of converting radioactive steam into conventional steam.

In conclusion, the following points can be drawn:

1. Radioactive steam can be converted into conventional steam as a safe heating supply source by using sets of heat exchangers.
2. Existing radioactive primary circuit can still be maintained in a closed circuit.
3. Energy saving on conventional fuels can be achieved by the addition of the SSGS to the existing nuclear power generating station.
4. High pressure conventional steam can be produced with an addition of SSGS and there is no firing equipment required in this system.
5. Location of the SSGS has to be selected as close as possible to the nuclear power station.

6. The best heat transfer efficiency for the evaporator shall occur at it's designed operating condition while the heat head is greatest.
7. Provision for dumping is a necessity in order to ensure continuous operation of SSGS.
8. In the case study of four modules, each module contains one evaporator, one condensate collector, and two heat exchangers connected in series, in complete with three secondary conventional steam delivery lines will be the scheme.
9. Full three-element control system has to be introduced for the secondary steam pressure and flow, and feed water level in the evaporator in conjunction with primary steam control.
10. The SSG System can be completed and built for an estimated price of approximately 40 million dollars. for the capacity described in this paper at the time of December of 1974.

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APPENDIX 1

MAINTENANCE

There have been allowed no provision for the routine maintenance since all the design and construction have been in accordance with the stringent nuclear codes and regulations and surveillance. However, it is expected that all equipments and apparatus which were designed for the SSGS shall be very reliable and will require no significant maintenance during operation. Major maintenance may be carried out during the major or annual shut down or in the summer season.

It is assumed that planned maintenance can be performed on the SSG system units will coincide with the planned maintenance carried out on the nuclear unit. However, due to there is a break in the summer time, normally the heating source is only required in the winter time, starting middle of October until end of March the next year in Canada. It is therefore recommended here that all the necessary maintenance and repairing shall be carried out and completed during those six months idle period.

APPENDIX II

SAFETY CONSIDERATION

II. 1. General

In many respects, nuclear plants contain a number of hazards to operating personnel. These hazards can be limited by careful compliance with international, federal and federal and provincial codes and regulations in matters of design, fabrication, construction and operation. Unlike many other chemical plants, it is radioactive and thermally reactive, and thus a large amount of energy may not allowed to be released accidently by misapplication of controls. Hence destruction of the plant by internal explosion is very improbable.

The principle concern regarding public safety is the radioactive of steam and condensate. Contained within the primary circuit. All welded construction is a typical leak proof of this from all process equipment considering plant design, operational and safety.

The standards, specifications and controls while will give firm assurance of the integrity of the envelope containing the radioactive fluids. Operational and sensible detection and maintenance procedures supported by adequate instrumentation and control will ensure discovery and correction of any possible leaks before they reach significant proportions. Extensive and strict monitoring system as nuclear plant shall be enforced.

Because of the radiological safety problems, it is necessary to ensure even a minor release of the primary steam from the SSG system into the surrounding environment are controlled and large releases prevented and collected and gathered into a vent system in order not to have a harmful effect.

The only way to achieve this is by strict attention to safety considerations in design, commissioning and operation.

Vent release into air via an existing stack is currently strictly limited. The radioactive concentration is reduced by filtering system. Release of the primary circuit to the atmosphere directly is strictly prohibited.

II. 2. SITE AND OTHER CONSIDERATIONS:

1. The SSGS site should be chosen closed or beside the existing nuclear power complex.
2. Supply of feed water for both process and cooling shall use the same water as for the existing nuclear complex.
3. Effluent into atmosphere should be controlled according to the regulations of appropriate international, federal and provincial authorities.
4. All other nuclear definitions and regulations shall be followed.

II. 3. SAFETY PHILOSOPHY:

II. 3. 1. GENERAL :

The construction of many large nuclear complex has led to in - depth study of safety standards which must be applied to the SSGS as well. A very comprehensive safety philosophy has been evolved which evident through all phases of design, manufacture, construction and operation of the nuclear complex and attachments.

II. 3. 2. DESIGN:

Conceptual designs are reviewed to ensure adequate safety standards, and ease of maintainablity, elimination of potential sources of leakage or failure in the containment is a key factor in minimizing nuisance leaks and serious releases, hence care is taken in detailed design of welded joints or areas susceptible to corrosion, and materials and components are chosen with view to minimizing inherent defects.

II. 3. 3. MANUFACTURE:

Wherever possible compnents are selected which have been proved in nuclear service. To ensure manufacture entirely in accordance with specification, a quality assurance program should be deviced which covers all stage of manufacture.

II. 3. 4. CONSTRUCTION:

Continuity of the quality assurance program through field installation of equipment and piping is ensured by appointment of qualified surveillance team in the field.

II. 3. 5. OPERATION:

Because of process malfunctions can lead to radioactive steam or great thermal energy release, every effort should be made to improve the efficiency with which operators can operate and control the SSGS within the nuclear complex. Operator training is vital, as well as knowledge of steam and hot condensate other than radioactive harzards by the aid of operation and design manuals.

In spite of all precautions, the possibility of primary steam and energy release can not be entirely eliminated, therefore a certain number of back--up safety system such as safety relief valves etc., shall be provided.

II. 3. 6. EMISSIONS CONTROL:

Since the release of the radioactive fluids would affect the environment if they are released in sufficient concentration or quantity. The principal release are primary steam and condensate. The regulatory requirements and guidelines should be followed in controlling the emissions.

II. 4. SAFETY FEATURES OF THE SYSTEM:

As it has been mentioned in the special features and characteristics that the SSG system is divided into several modules which is actually a kind of safety precaution; since in any event of emergency, each module can be isolated completely from others which may remain in operation.

Automatic and remote control motorized isolation valves are provided on both primary and secondary side for complete isolation of each module or as a whole the complete SSG system. Radioation detectors are installed at every equipment and at every critical strategical location and point . Once the radiation detectors sense any radioactive element, it will call for an instant isolation for the complete loop or the whole SSG system.

Safety relief valves are provided in every case on both primary and secondary side to ensure safety of the overall system. It is assumed that the primary steam already had steam relief system; but for condensate, since it is radioactive and in order not to waste any primary condensate which is normally return to the original circuit, all primary waste or start-up or shut down drains are designed to dump into the reserved storage tank and the main condenser of the original circuit as normal routine operational dumping.

For the secondary side, since it is not radioactive, therefore it is safe to have the steam and condensate relieved to the atmosphere in any case.

Although the automatic nuclear motorized isolation valves are very expensive, yet for the sake of most positive safety provision, there is such a necessity to have automatic nuclear motorized isolation valves installed on every inlet and outlet of each module and SSG system and its outlet up to the point of the secondary steam supply reserve tank.

The complete SSG system including each module is designed in such a way that in any circumstances, and in any time; either on primary side or secondary side of the whole system or it's single component become interrupted or failed, it is assumed that the faulty component or module or even the system can be isolated completely; and the interruption of primary steam supplying to the SSG system would be able to switch to the original turbine cycle to produce electricity and the nuclear reactor is uninterfered and still remain on stable condition. It has been studied that there is a provision for the emergency dumping of primary steam to the main condenser and condensate reserved surge tank in the original circuit.

In order to maintain an complete isolation condition, in the SSG system, there would be necessary to provide double motorized isolation valves and double check valves on both primary side and secondary side of each module and the whole system.

As it can be seen that it has been even specified in the design requirements and constraints that have governed and narrowed the safety provision to an almost maximum.

Double safety features has been applied such as a pressure relief valve is used upstream of a safety valve to ensure the excessive overpressure of the system.

Double check valves and double motorized valves also applied for complete isolation of the components or module or system to allow any repairing works or maintenance or do not interfer others which remain on operation.

Primary vent system has been rerouted and returned to main condenser in the original circuit without affecting the continuous operation of the systems; and drainage waste is collected in an additional waste collecting tank in order not to damage the environment and surroundings.

APPENDIX V

HEAT BALANCE CALCULATION AND PIPING DESIGN

V. 1 HEAT BALANCE CALCULATION

The material and heat balance have been calculated in the following manner, and results are shown in table 5. 1 through table 5. 4 at different operating and design conditions. The numbers specified here are in corresponding to those in Fig. 5 and in tables 5.1 through table 5.4.

PRIMARY STEAM REQUIREMENT CALCULATION

Secondary side:

Total heat energy required: Q_2

Q_2 = (1) Enthalpy in the pegging steam required for the water treating deaeration system

+

(2) Heat loss on steam delivery pipe lines

+

(3) Secondary steam enthalpy

+

(4) Blowdown heat loss

Primary side:

Total heat energy input required: Q_1

$$Q_1 = W \; (\Delta h_1)$$

and $Q_1 = Q_2$ OR $W = \dfrac{Q_1}{\Delta h_1}$ (V. 1)

Allow 0.5 % of heat loss on primary side

$$W_{actual} = 1.005 \left(\frac{Q_1}{\Delta h_1} \right) \qquad \text{(V.2)}$$

$\langle 1 \rangle = \frac{\text{available primary steam}}{2}$

$\langle 2 \rangle = (2\langle 1 \rangle - 2\langle 3 \rangle)/2$

$\langle 3 \rangle =$ See primary steam requirement calculation

$\langle 4 \rangle = \frac{1}{4} \times 2\langle 3 \rangle$ OR $\frac{1}{3} \times 2\langle 3 \rangle$ *

$\langle 5 \rangle = \langle 4 \rangle$

$\langle 6 \rangle = \frac{1}{2}\langle 5 \rangle$

$\langle 7 \rangle = \langle 6 \rangle$

$\langle 8 \rangle = \langle 4 \rangle$

$\langle 9 \rangle = \langle 8 \rangle$

$\langle 10 \rangle = \langle 9 \rangle$

$\langle 11 \rangle = 2\langle 3 \rangle$

$\langle 12 \rangle =$ See pegging steam calculations in appendix IX

$\langle 13 \rangle = \left(2\langle 2 \rangle - 12\right)/2$

$\langle 14 \rangle = 2\langle 13 \rangle$

$\langle 15 \rangle = \langle 14 \rangle$

$\langle 16 \rangle = \langle 15 \rangle + \langle 11 \rangle + \langle 12 \rangle$

$\langle 17 \rangle = \frac{1}{4}\left[(1) + (2) + (3)\right]$ OR $\frac{1}{3}\left[(1) + (2) + (3)\right]$ *

$\langle 18 \rangle = \langle 17 \rangle$

19 = 1,920,000 lb/hr Secondary steam required

20 = 4 17 + Blowdown

21 = $\frac{20}{4}$ OR $\frac{20}{3}$ *

22 = 21

23 = See pegging steam calculations in appendix IX

24 = 23

25 = $\frac{\text{total blowdown}}{4}$ OR $\frac{\text{total blowdown}}{3}$ *

26 = $\frac{\text{weight of equivalent heat loss}}{2 \text{ No. of pipe lines}}$

27 = Total weight of equivalent heat loss

28 = 20 - 24

V. 2 PIPING DESIGN

References : (7), (11), (18), (26), (32)

In design of the piping system, the pressure, temperature, velocity, and selection of material are the main functions to determine the wall thickness of the pipe and the pressure drop through the piping system.

In order to have the SSGS designed and operated in a safe manner, the ANSI piping codes and ASTM material standards are to be strictly followed.

The following formulaes are used in the calculations:

Design pressure = 110 % (Maximum operating pressure), psi

Design temperature of 650°F has been used in this case.

Manufacturers' tolerance of $t_{mt} \doteq 12.5$ % is allowed.

For pipe wall thickness calculations :

$$t_m = \frac{P\, d_o}{2\,(S_E + P\, y)} + C \qquad (\text{V. 2. 1})$$

(for secondary side)

$$t_m = \frac{P\, d_o}{2\,(S_s + y\, P\,)} + C \qquad (V.\ 2.\ 2\ )$$

(for primary side).

$$t_{act.} = \frac{t_m}{100\ \% - t_{mt}} \qquad (V.\ 2.\ 3\ )$$

$$A = W \cdot V_s \cdot \frac{1}{v} \qquad (V.\ 2.\ 4\ )$$

For pressure drop calculations :

$$P_{100'} = 0.000336\,\frac{f\, W^2\, V_s}{d^5} = 0.000336\,\frac{f\ \ W^2}{d^5\, \varrho}$$

(V. 2. 5)
(for steam)

$$P_{100'} = 0.1294\,\frac{f\ \varrho\ v^2}{d} = 0.000336\,\frac{f\ W^2}{d^5 \varrho}$$

$$= 4350\,\frac{f\,\varrho\, Q_a}{d^5}$$

(V. 2. 6)
(for water)

APPENDIX VI

EQUIPMENT DESIGN CALCULATIONS

REFERENCES : (3) ASME boiler & pressure vessel code Sect. VIII div. 1 - 1968

(30) Standard of tubular exchanger manufacturers' Association 5 th Edition, 1968

FORMULAES IN THE CODE HAVE BEEN USED:

SHELL THICKNESS t_s :

$$t_s = \frac{PR_a}{S_3E - 0.6P} + C \qquad (VI. 1)$$

SHELL HEAD THICKNESS t_{sh}:

$$t_{sh} = \frac{P\ D}{2S_3E - 0.2P} + C \qquad (VI. 2)$$

CHANNEL THICKNESS t_c :

$$t_c = \frac{P\ R_a}{S_3E - 0.6P} + C \qquad (VI. 3)$$

CHANNEL HEAD THICKNESS t_{ch}:

$$t_{ch} = \frac{P\ D}{2S_3E - 0.2P} + C \qquad (VI. 4)$$

TUBE SHEET THICKNESS t_t :

$$t_t = \frac{FG}{2}\sqrt{\frac{P}{S_s}} \qquad (\text{VI. } 5)$$

TOTAL VOLUME :

$$V = (L \text{ or } H) \times \frac{1}{4}\pi D^2 + \frac{\pi}{6} D^3 \qquad (\text{VI. } 6)$$

RESULT OF CALCULATIONS ARE SHOWN IN TABLE VI.1. 1

TABLE VI. 1. 1

RESULTS OF EQUIPMENT DESIGN CALCULATIONS

	CONDENSATE COLLECTOR	EVAPORATOR	COOLER/PREHEATERS	BLOWDOWN TANK
SHELL DIAMETER, FT.	5.0	13.0	44" 37"	8.0
SHELL LENGTH OR HEIGHT, FT.	10.0	60.0	30 30	22.0
SHELL THICKNESS, IN.	1.75	3.00	1.00	2.0
SHELL HEAD THICKNESS, IN.	1.75	3.00	1.00	2.0
CHANNEL THICKNESS, IN.	—	1.75	1.25	—
CHANNEL HD.THK'S, IN.	—	1.75	1.125	—
TUBE SHEET THICKNESS, IN.	—	8.25	6.25	—
TOTAL VOLUME CU.FT.	262	—	—	1374 cu.ft.

APPENDIX VII

EVAPORATOR CALCULATIONS

VII.1 Determination of the heat transfer coefficient U_D

In determination of the designed overall heat transfer coefficient U_D, the following constraints and formulaes have been applied:

(1). $\frac{Q}{A} \neq h_v (\Delta t)_w$ (VII.1.1)

(2). $(\frac{Q}{A})max. = U_D (\Delta t)_a = 30,000$ btu/(hr)(ft^2)

(VII.1.2)

(3). Maximum water vaporization film coefficient at standard atmospheric conditions 14.7 psia and 60^{o}F

$h_o = 1000$ btu/~~(hr)~~(ft^2)(^{o}F) (VII.1.3)

(4). Maximum condensing steam coefficient

$h_{io} = 1500$ btu/(hr)(ft^2)(^{o}F) (VII.1.4)

(5). Dirt factor $R_d = \frac{1}{U_D} - \frac{1}{U_c} = \frac{U_c - U_D}{U_D \cdot U_c}$ (VII.1.5)

(Large dirt factor is not essential to the continued operation but only as a prevention of vapor blanketing).

(6). $Q = U_D A (\Delta t)_a$ (VII.1.6)

(7). $U_c \overset{\text{must}}{\gg} U_D$ and $A \overset{\text{must}}{\gg} A_c$ (VII . 1 . 7)

(8). N_t = No. of tube per Path

$$= \frac{A}{0.1963 \times 30 \times 2}$$

Where A is determined based on $U_D = 540$

$\theta_m = 24$

0.75" o.d. U-tube x 60 ft. long is assumed

(9). Pressure drop equations in the following have been used:

Shell side:

$$\Delta P_s = \frac{f\, G_s^2\, D_s\,(N+1)}{2g\, \rho\, D_e\, \phi_s}, \text{ psf} \qquad (\text{VII . 1 . 8})$$

Tube side :

$$\Delta P_t = \frac{f\, G_t^2\, L\, n}{5.22\; 10^{10} D_e\, s\, \phi_t}, \text{ Psf} \qquad (\text{VII . 1 . 9})$$

U - Return:

$$\Delta P_r = \frac{4n}{s}\;\frac{v^2}{2g}, \text{ Psi} \qquad (\text{VII . 1 . 10})$$

Total tube side pressure drop

$$\Delta P_T = \Delta P_t + \Delta P_r, \text{ PSi} \qquad (\text{VII . 1 . 11})$$

The final results from the calculations are shown in table VII. 1.

TABLE VII. 1

SUMMARY OF HEAT TRANSFER CALCULATIONS

TUBE SIDE		SHELL SIDE
h_{io} = 1500	h outside	h_o = 1792
U_c	calculated	817
U_d	determined	540
R_d	calculated	0.0005119
R_d	required	0.000500
2,412	calculated ΔP	negligible
10	allowable ΔP	negligible
A	calculated	33,749
$\Theta_m = \Delta t$	used	24
$\frac{Q}{A}$	calculated	8,543
$\frac{Q}{A}$	allowed	30,000

VII.2 Determination of heating surface required.

Heating surface of the evaporator can be calculated from the general heat equation.

$$Q = U_D \, A \Delta t$$
$$= U_D \, A \, \theta_m \qquad \text{(VII.2.1)}$$
$$\text{or} \qquad A = \frac{Q}{U_D \, \theta_m},$$

At different operating conditions, different heating surface is required at that particular condition. The following formula is developed for the heating surface area calculation at different operating conditions:

$$\text{Since} \quad Q_1 = Q_2$$
$$Q_1 = U \, A_1 \, \Delta t_1 = U_D \, A_1 \, \theta_{m1}$$
$$Q_2 = U \, A_2 \, \Delta t_2 = U_D \, A_2 \, \theta_{m2}$$
$$U_D \, A_1 \, \Delta t_1 = U_D \, A_2 \, \Delta t_2$$
$$\text{or} \quad U_D \, A_1 \, \theta_{m1} = U_D \, A_2 \, \theta_{m2}$$
$$A_2 = \frac{\Delta t_2}{\Delta t_1} \, A_1$$
$$\text{or} \quad A_2 = \frac{\theta_{m2}}{\theta_{m1}} \, A_1 \qquad \text{(VII.2.2)}$$

TABLE VII. 2

REQUIRED HEATING SURFACE AREA AT DIFFERENT BOUNDARY OPERATING PRESSURE

HEATING SURFACE AREA REQ'D, FT^2	486 PSIA	458 PSIA
FOR 4 - EVAPORATOR	31,463	26,030
FOR 3 - EVAPORATOR	41,951	34,708

VII. 3 DETERMINATION OF STEAM RELIEVING VELOCITY AND EVAPORATOR DIAMETER.

From steam table and the following equations:

$$\rho = \frac{1}{V_s} \qquad \text{(VII.3.1)}$$

$$v = \frac{Q}{A_a} = \frac{w \; V_s}{A_a} \qquad \text{(VII.3.2)}$$

Or: $$A_a = \frac{w \; V_s}{v} \qquad \text{(VII.3.3)}$$

And : $$A_a = D \times L \qquad \text{(VII.3.4)}$$

Where L can be assumed at any convenient length .

Thus: $$D = \frac{A}{L} \qquad \text{(VII.3.5)}$$

And : $$D_o = 1.75 \;\; d_o \; (n \; N_{PT})^{0.47} \qquad \text{Ref. (6)}$$

Results of calculation are shown in table VII.2 .

TABLE VII. 3.

RESULTS OF EVAPORATOR SIZING CALCULATION

MAX. Steam relieving velocity allowed	0.25 FPS ref (27)
CALC. Steam relieving velocity	0.218FPS
LENGTH (Between enclosed water surface area)	60 FT
CALCULATED STEAM RELIEVING AREA	726 SQ. FT
CALCULATED DIAMETER	13.0 FT

APPENDIX VIII

COOLER / PREHEATER CALCULATIONS

In accordance with the references (21,27,6) and the specified flow rate and flow conditions, numerous calculations have been made for one heat exchanger and two heat exchangers in parallel and in series. The following formulaes have been used:

$$LMTD = \frac{GTTD - LTTD}{LOG_e \dfrac{GTTD}{LTTD}} \qquad (VIII.\ 1)$$

$$R_q = \frac{T_1 - T_2}{t_2 - t_1} \quad \text{AND} \quad S_q = \frac{t_2 - t_1}{T_1 - t_1} \qquad (VIII.\ 2)$$

$$A = \frac{Q}{U_D \cdot LTMD_C} \qquad (VIII.\ 3)$$

$$D_o = 1.75\ d_o\ (\ n\ N_{PT}\ )^{0.47} \qquad (VIII.\ 4)$$

TABLE VIII. 1

COOLER / PREHEATER CALCUTIONS

SYMBOLS	CONDITION 1			CONDITION 2		
# : LB/HR * : FOR 5 UNITS ‡ : WITH .75" O.D. TUBING AT 1" □ PITCH 30' LONG X : TWO HEAT EXCHANGERS CONNECTED IN SERIES.	1. $\Delta P_{3x24"+PIPE}$ = 63 PSI 2. EVAPORATOR OPE PRESS. = 486 PSIA			1. $\Delta P_{3x24"+PIPE}$ = 35 PSI 2. EVAPORATOR OPE. PRESS. = 458 PSIA		
DETAILS	CALCU'D HEAT EXCH.			CALCU'D HEAT EXCH.		
	ONE *	TWO		ONE	TWO	
		NO. 1 X	NO. 2 X		NO. 1	NO. 2
TUBE SIDE:						
FLOW RATE, #	606,230	606,230	606,230	615,939	615,939	615,939
TEMP. INLET, °F	488	488	378	487.98	487.98	395
TEMP. OUT, °F	268.52	378	268.52	280.63	395	280.63
SHELL SIDE						
FLOW RATE, #	577,102	577,102	577,102	577,102	577,102	577,102
TEMP. INLET, °F	223	339.63	223	223	337.53	223
TEMP. OUT, °F	458	458	339.63	450	450	337.53
HEAT TRANSF, BTU	143,294,585	74,190,427	69,104,157	137,311,281	69,792,048	67,519,233
CORR. LMTD	31.64	27.57	39.98	40.05	40.47	50.64
SURF. AREA REQ'D. A.* FT²	17,253	10,251	7,316	13,060	6,569	5,079
No. OF PASSES, SHELL	6	3	3	6	3	3
No. OF PASSES, TUBE	12	6	6	12	6	6
ESTD SHELL I.D.‡ " (U_o=300)	60	47	40	53	38	34
(U_o=350)	56	44	37	49	36	32

TABLE VIII. 2

COOLER / PREHEATER CALCUTIONS

SYMBOLS	CONDITION 1			CONDITION 2		
# : LB/HR * : FOR 5 UNITS ‡ : WITH .75" O.D. TUBING AT 1" □ PITCH 30' LONG X : TWO HEAT EXCHANGERS CONNECTED IN SERIES.	1. $\Delta P_{8\times24''\phi\ PIPE} = 63$ PSI 2. EVAPORATOR OPE PRESS. = 486 PSIA			1. $\Delta P_{3\times24''\phi\ PIPE} = 35$ PSI 2. EVAPORATOR OPE. PRESS. = 458 PSIA		
DETAILS	CALCU'D HEAT EXCH.			CALCU'D HEAT EXCH.		
	ONE	TWO		ONE	TWO	
		NO. 1[X]	NO. 2[X]		NO. 1	NO. 2
TUBE SIDE:						
FLOW RATE, #	603,470	603,470	603,470	613,158	613,158	613,158
TEMP. INLET °F	484.90	484.90	378	484.94	[illegible]4.94	378
TEMP. OUT, °F	263.93	378	263.93	276.16	378	276.16
SHELL SIDE						
FLOW RATE, #	577,102	577,102	577,102	577,102	577,102	577,102
TEMP. INLET, °F	223	343.77	223	223	332.21	223
TEMP. OUT, °F	458	458	343.77	450	450	332.21
HEAT TRANSF, BTU	140,396,821	71,680,166	71,613,784	151,015,408	72,912,995	64,399,984
CORR. LMTD	28.40	24.64	32.23	36.46	34.10	43.45
SURF. AREA REQ'D. A.* FT^2	21,971	12,929	9,875	18,408	9,502	6,587
No. OF PASSES, SHELL	6	3	3	6	3	3
No. OF PASSES, TUBE	12	6	6	12	6	6
ESTD SHELL I.D.‡ " (U_o=300)	63	49	43	58	42	36
" (U_o=350)	58	45	40	54	39	33

TABLE VIII. 3

SUMMARY OF HEAT EXCHANGER CALCULATIONS

			NO. 1	NO. 2
NO. 1	691	h OUTSIDE	2276	
NO. 2	756	h OUTSIDE		3129
U_C		CALCULATED	530	591
U_D		USED	300	300
R_d		CALCULATED	0.000976	0.001170
R_d		REQUIRED	0.000500	0.000500
r_w		REQUIRED	0.000471	0.000471
ΔP_s		ALLOWED	SHELL 5.00psi	SHELL 5.00psi
ΔP_t		ALLOWED	TUBE 10.0psi	TUBE 10.0psi
ΔP_s		CALCULATED	0.36psi	0.24psi
ΔP_t		CALCULATED	1.05psi	1.80psi
ΔP_r		CALCULATED	3.29psi	6.00psi
ΔP_T		CALCULATED	4.34psi	7.80psi

TABLE VIII. 4

SUMMARY OF HEAT EXCHANGER CALCULATIONS

			NO. 1	NO. 2
NO. 1	899	h OUTSIDE	2447	
NO. 2	918	h OUTSIDE		2906
U_C		CALCULATED	657	698
U_D		USED	350	350
R_d		CALCULATED	0.000864	0.000953
R_d		REQUIRED	0.000500	0.000500
r_w		REQUIRED	0.000471	0.000471
ΔP_s		ALLOWED	SHELL 5.00psi	SHELL 5.00psi
ΔP_t		ALLOWED	TUBE 10.0psi	TUBE 10.0psi
ΔP_s		CALCULATED	0.38psi	0.42psi
ΔP_t		CALCULATED	1.50psi	2.00psi
ΔP_r		CALCULATED	6.00psi	8.00psi
ΔP_T		CALCULATED	7.50psi	10.00psi

APPENDIX IX

PEGGING STEAM CALCULATIONS

IX.1 Water treating deaeration system

The water outlet temperature from the deaerator has to be determined as following :

It is a good practice to operate a deaerator at 2 to 8 psig above atmospheric pressure. 3 psig above the atmospheric pressure is chosen in this case, which means the deaerator will be operated at 18 psig at which the saturation temperature is 222°F. Since there is evaporator feed water pump must be required and the required discharge pressure from the pump has to overcome friction loss of piping system, hydraulic head, and cooler/preheaters, all pressure drops have been estimated and allowed. At a dishcarge pressure of 564 psig based upon the designed evaporator operating pressure, the water temperature has 1°F increment from 222°F to 223°F due to the increased pressure from 18 psia to 564 psia. This water temperature of 223°F is the feed water supply temperature criteria to the SSG system. Refer to Fig. 5 and Fig. 8.

Both W_1 and $S^{(1)}$ are unknown, W_2 is the only value known.

By applying energy equation :

$$\overset{(1)}{S}\,h_s + W_1\,h_{w1} = W_2\,h_{w2} \qquad - (1)$$

$$W_1 = W_2 - \overset{(1)}{S} \qquad \_\_\_\_\_\_ (2)$$

Substitute (2) into (1)

$$\overset{(1)}{S}h_s + (W_2 - \overset{(1)}{S})\, h_{w1} = W_2\, h_{w2}$$

Or $$\overset{(1)}{S}h_s + W_2\, h_{w1} - \overset{(1)}{S}h_{w1} = W_2\, h_{w2}$$

Re-arrange the above equation

$$\overset{(1)}{S}\,(h_s - h_{w1}) = W_2\,(h_{w2} - h_{w1})$$

$$\overset{(1)}{S} = \frac{W_2\;(h_{w2} - h_{w1})}{(h_s - h_{w1})} \qquad (IX.1.1)$$

This equation is developed for the calculation of deaerator pegging requirement when W_2 is known. Actually the blowdown is unknown because of $\overset{(1)}{S}$, W_1, W_2 are unknown. But the following relationship is true

$$W_2 = \left[1 + X \% \right] \left[\begin{array}{l} \text{secondary steam production(3)} \\ + \text{ equivalent weight of steam delivery} \\ \quad \text{pipeline heat loss (2)} \qquad\qquad (1) \\ + \text{ amount of pegging steam required } S \end{array} \right]$$

(IX.1.2)

Since the blowdown is based on the total steam requirement in which the pegging steam is unknown, the equation then can be further developed as in the following:

Let: $t_{w1} = 33^oF$

$h_{w1} = 1$ Btu /lb

blowdown allowed $= 1 \% \left[(3)+(2)+ \overset{(1)}{S} \right]$

$t_{w2} = 223^oF$

Then:

$$W_2 = 1.01\left[(3)+(2)+\overset{(1)}{S}\right] \qquad (1)$$

$$h_{w2}W_2 = 1.01\left[(3)+(2)\right] + h_s\overset{(1)}{S} \qquad (2)$$

Substitute (1) into (2)

$$h_{w2}\ 1.01\left[(3)+(2)+\overset{(1)}{S}\right] = 1.01\left[(3)+(2)\right] + h_s\overset{(1)}{S}$$

$$(h_s - 1.01\,h_{w2})\,\overset{(1)}{S} = 1.01\,h_{w2}\left[(3)+(2)\right] - 1.01\left[(3)+(2)\right]$$

$$\overset{(1)}{S} = \frac{1.01\,h_{w2}\left[(3)+(2)\right] - 1.01\left[(3)+(2)\right]}{h_s - 1.01\,h_{w2}} \qquad (\text{IX.1.3})$$

And $W_1 = W_2 - \overset{(1)}{S}$ (IX.1.4)

Results of calculation can be referred to table 5.1 through table 5.4

IX. 2 Condensate deaeration system

A condensate mixing tank is provided for the purpose of having an even condensate temperature return to the original existing deaerator.

The condensate return from the original circuit is at 200°F and the condensate return from the SSGS has a different temperature return.

Refer to Fig. 5 and Fig. 9 . The following general equations have been developed for the simplified calculations for the pegging steam required at different condition of condensate return :

$$W_2 = W_m + \overset{(1)}{S} \qquad \text{———— (1)}$$

$$W_m = W_s + W_p \qquad \text{———— (2)}$$

$$W_2 = W_s + W_p + \overset{(1)}{S} \qquad \text{———— (3)}$$

Where W_p and $\overset{(1)}{S}$ are known

$$W_2\, h_{w2} = W_s\, h_{ws} + W_p\, h_{wp} + \overset{(1)}{S} h_s \qquad \text{———— (4)}$$

(3) x h_{wp} and obtain:

$$W_2\, h_{wp} = W_s\, h_{wp} + W_p\, h_{wp} + \overset{(1)}{S}\, h_{wp} \qquad \text{———— (5)}$$

(5) -(4)

$$W_2\,(h_{wp} - h_{w2}) = W_s\,(h_{wp} - h_{ws}) + \overset{(1)}{S}(h_{wp} - h_s)$$

Or
$$\overset{(1)}{S} = \frac{W_2\,(h_{wp} - h_{w2}) - W_s\,(h_{wp} - h_{ws})}{(h_{wp} - h_s)} \qquad (\text{IX.2.1})$$

And
$$W_p = W_2 - W_s - \overset{(1)}{S} \qquad (\text{IX.2.2})$$

Results of calculations can be referred to table 5.1 through table 5.4.

APPENDIX X

BLOWDOWN SYSTEM CALCULATIONS

X.1 Sizing of blow - off tank

All calculations has been made in reference to reference (31) (26)

The following equations have been developed and used:

$$S_F\% = \frac{SH - SL}{LH} \times 100\,\% \qquad (X.1.1)$$

And

$$W_B = \frac{V_e}{V_1} \qquad (X.1.2)$$

$$W_F = W_B \times S_F\,\% \qquad (X.1.3)$$

$$W_N = W_B - W_F \qquad (X.1.4)$$

$$V_H = W_N - V_2 \qquad (X.1.5)$$

$$V_W = 2 \times V_H \qquad (X.1.6)$$

$$V = 2 \times V_W \qquad (X.1.7)$$

$$A_a = \frac{\pi D^2}{4} \qquad (X.1.8)$$

$$D = \sqrt{\frac{4\,A_a}{\pi}} \qquad (X.1.9)$$

$$L = \frac{V}{A_a} \qquad (X.1.10)$$

And the calculated results are shown in table X.1.1

TABLE X. 1. 1

RESULTS OF BLOWDOWN TANK CALCULATIONS

$S_F\%$	28.11%	
V_B	255	
W_B	12,978	
W_F	3648	
W_N	9330	
V_H	157	
V_W	314	
V	628	
A_a	12.6	
D	4	8
L	50	22

X.2 Sizing of vent pipe and water dischage pipe:

Equations have been developed in the following:

Ref. (31) and (26).

$$F_T = F_B \times d \qquad (X.2.1)$$

$$F_A = F_T \times S_F\,\% \qquad (X.2.2)$$

$$F_D = F_T - F_A \qquad (X.2.3)$$

$$GPM = \frac{F_D}{S_G \times 60} \qquad (X.2.4)$$

Constraint:

$$F_A \overset{\text{must}}{\ll} F_T$$

Results of calculations:

Dia. of vent pipe = 6 inch.

Dia. of disch. pipe = 6 inch.

X. 3. Blowdown cooler calculation.

The amount of cold water required to cool the blowdown to 80°F can be calculated according to the following developed formula:

$$G_{hb} = \frac{t_{mb} - t_{cb}}{t_{hb} - t_{mb}} \qquad (X.3.1)$$

Results:

G_{hb}: 0.23 calculated

t_{mb}: 80°F assumed

t_{cb}: 50°F assumed

t_{hb}: 212°F assumed

X . 4 Blowdown heat transfer calculation

Refer to Fig. 10

References : (31, 20, 21)

The following formulaes are developed and used:

$$Q_{in} = B \; (h_{B1} - h_{B2})$$

$$Q_{out} = W_1 \; (h_{W1'} - 0)$$

$$Q_{in} = Q_{out}$$

$$B \; (h_{B1} - h_{B2}) = W_1 \; h_{W1'}$$

$$h_{W1'} = \frac{B \; (h_{B1} - h_{B2})}{W_1} \qquad (X.4.1)$$

And $$S_B = \frac{W_2 \; (h_{W2} - h_{W1}')}{h_S - h_{W1}'} \qquad (X.4.2)$$

Steam saved = $S - S_B$

$$= \frac{W_2 \; (h_{W2} - h_{W1})}{(h_S - h_{W1})} - \frac{W_2 \; (h_{W2} - h_{W1}')}{(h_S - h_{W1}')} \qquad (X.4.3)$$

Results:

$h_{W1}' = 4.69$ btu/lb

$t_{W1}' = 36.7^{\circ}F$

$S - S_B = 7,595$ lb/hr Table V.1, and V.3

$S - S_B = 5,228$ lb/hr Table V.2 and V.4

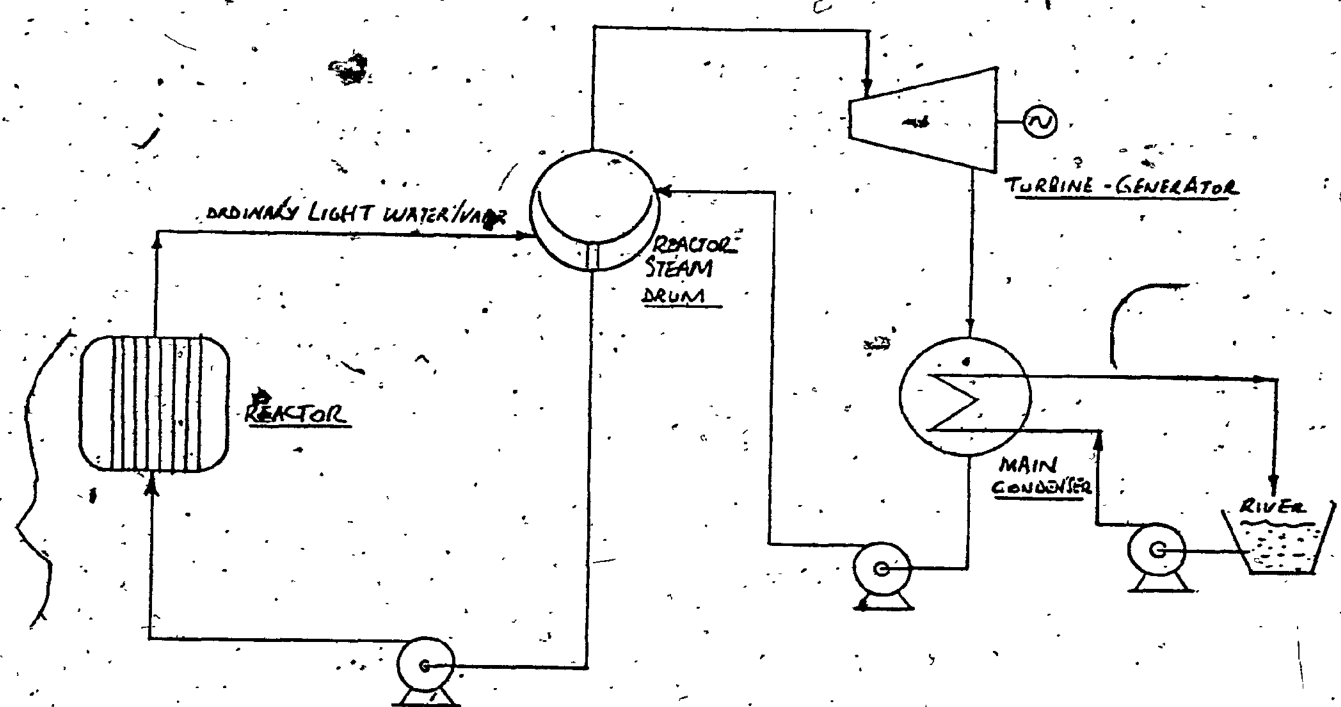


FIG. 1

CANDU BLW-NUCLEAR POWER GENERATION
GENERAL FLOW DIAGRAM

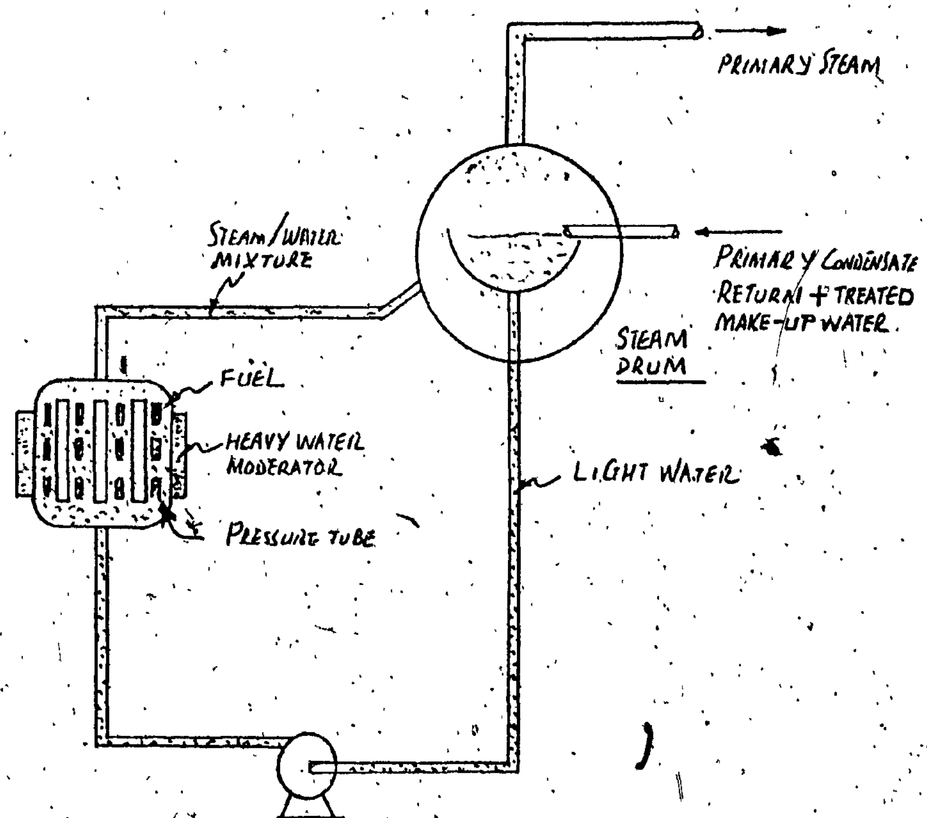


FIG. 2
SKEMATIC DIAGRAM OF CANDU BLW-NUCLEAR Flow DIAGRAM.

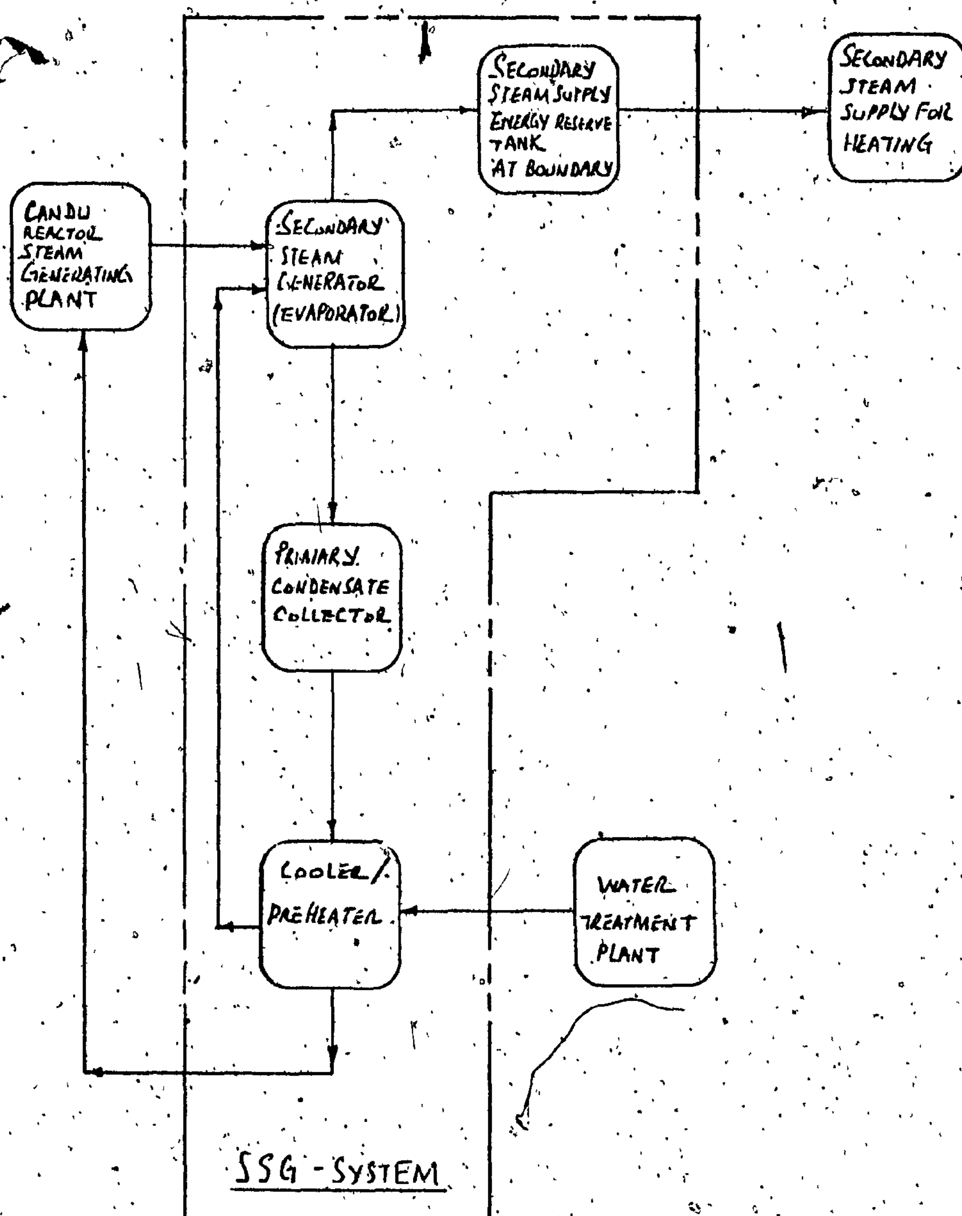


FIG. 3

SSG SYSTEM BLOCK DIAGRAM

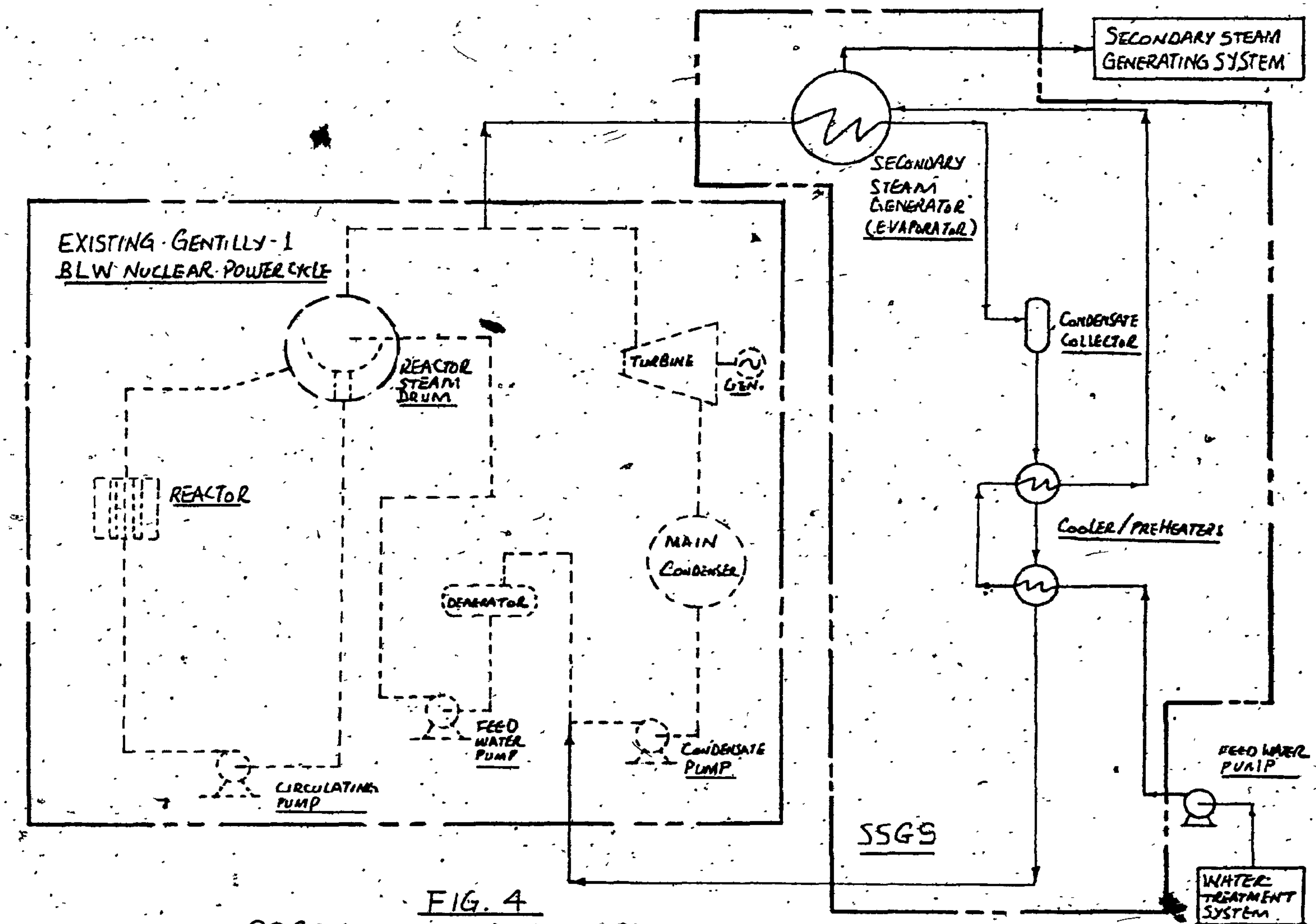


FIG. 4
SSGS AND EXISTING CANDU BLW-NUCLEAR POWER GENERATION FLOW DIAGRAM

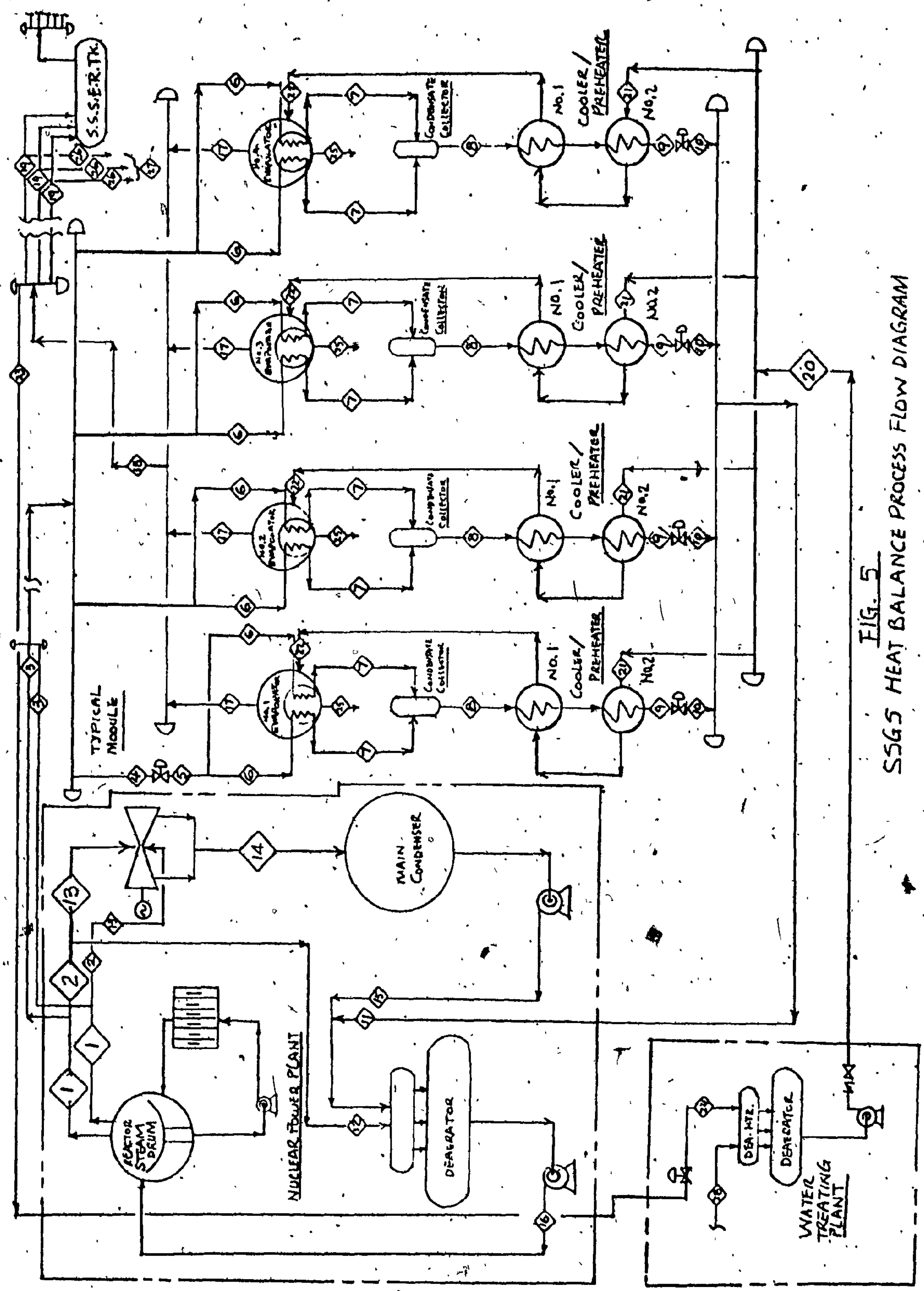


FIG. 5

SSGS HEAT BALANCE PROCESS FLOW DIAGRAM

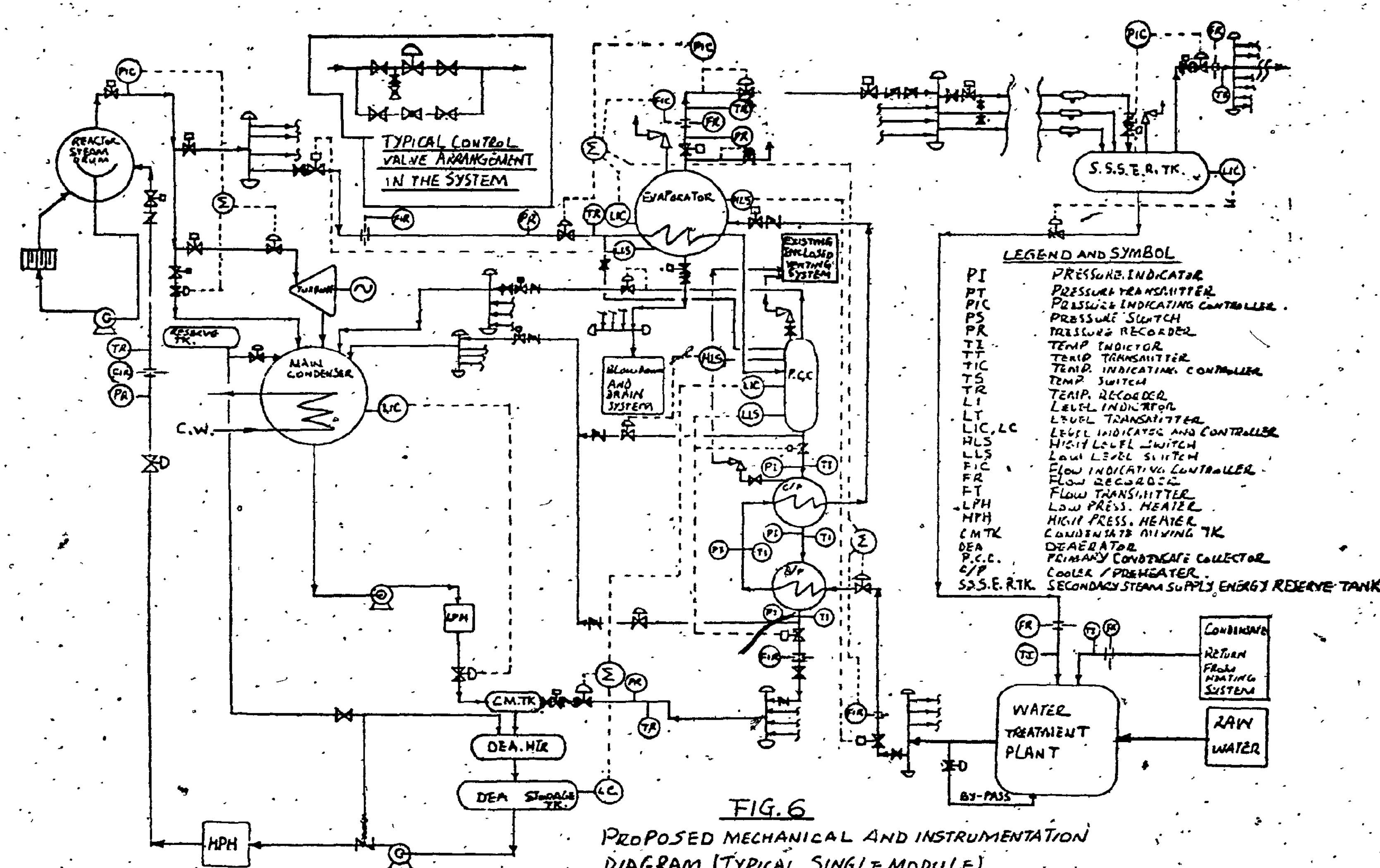
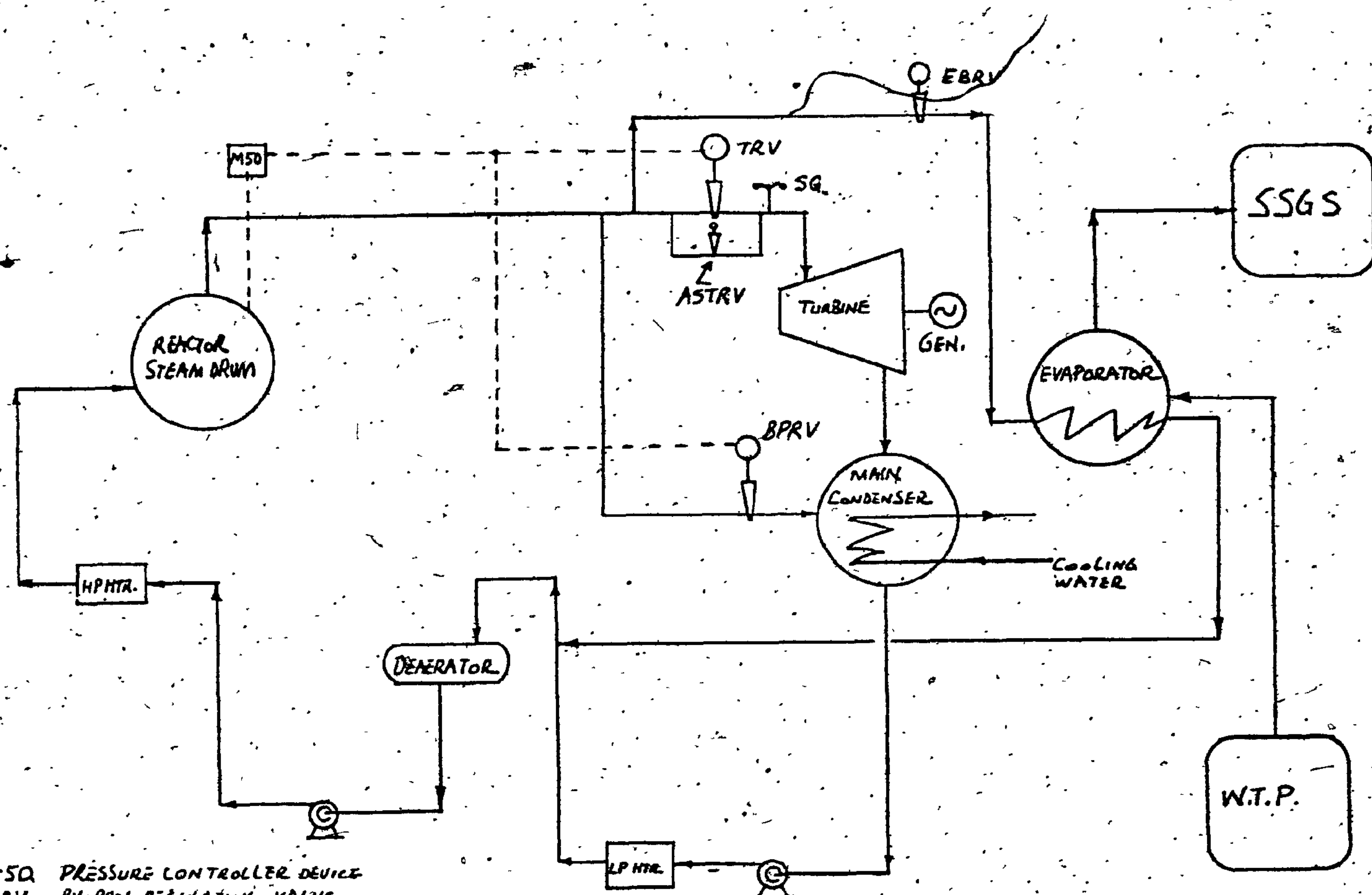


FIG. 6

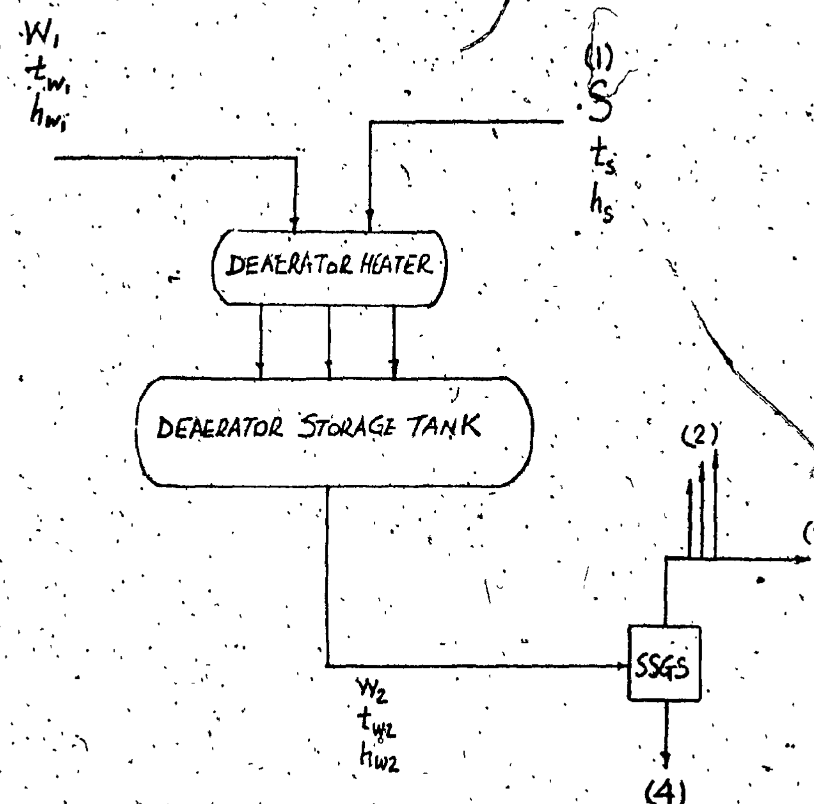
PROPOSED MECHANICAL AND INSTRUMENTATION DIAGRAM (TYPICAL SINGLE MODULE)



M-50 PRESSURE CONTROLLER DEVICE
BPRV BY-PASS REGULATING VALVE
TRV TURBINE REGULATING VALVE
SG SPEED GOVERNOR
EBRV EVAPORATOR REGULATING VALVE
WTP WATER TREATMENT PLANT
ASTRV ADDED SMALL TURBINE CONTROL VALVE

FIG. 7

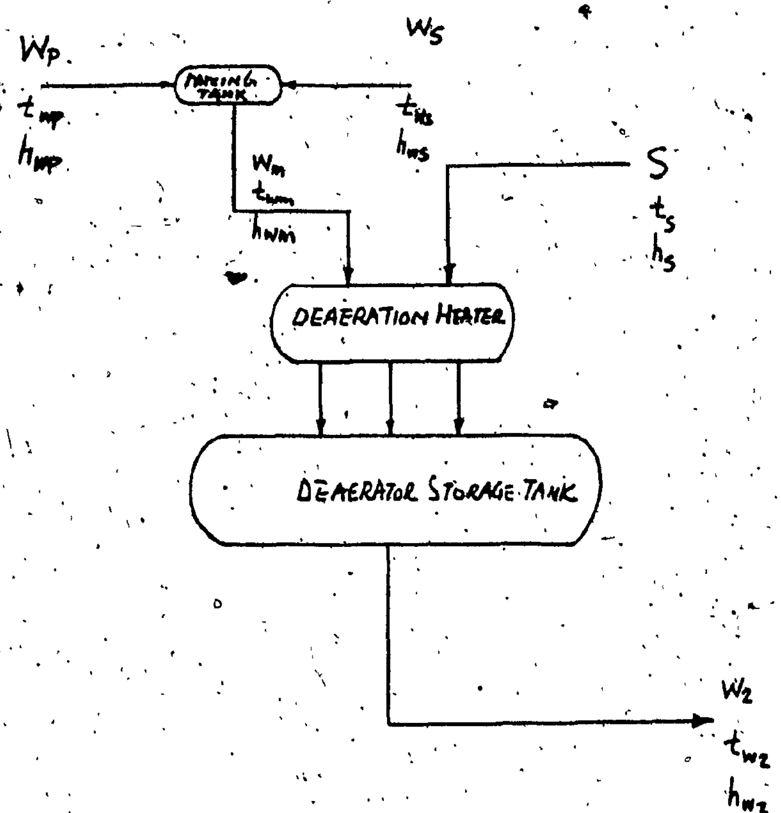
FLOW AND PRESSURE CONTROL SCHEMATIC DIAGRAM



(1)
S = PEGGING STEAM LB/HR.
W_1 = TREATED WATER INLET LB/HR
W_2 = TREATED FEED WATER OUTLET LB/HR.
t = TEMPERATURE °F
h = ENTHALPY BTU/LB.
SUBSCRIPTS : s, W_1 AND W_2 REPRESENT PEGGING STEAM INLET, WATER INLET AND WATER OUTLET RESPECTIVELY.
(2) = STEAM DELIVERY PIPELINE HEAT LOSS, BTU/#
(3) = SECONDARY REQUIRED STEAM, LB/HR
(4) = BLOWDOWN 1% IS ALLOWED, LB/HR.

FIG. 8

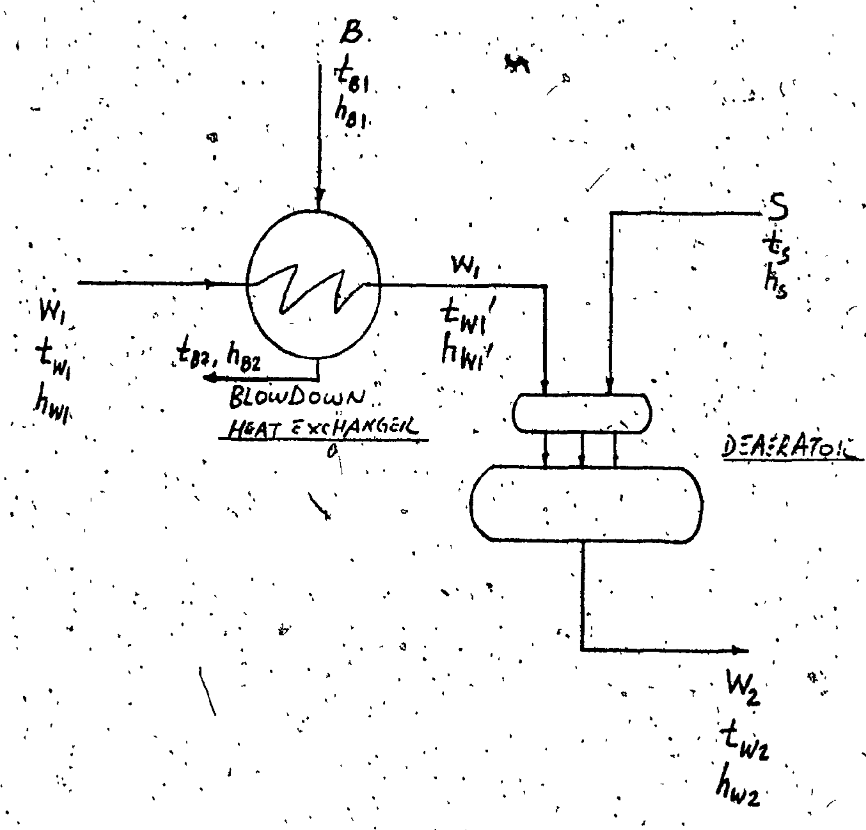
WATER TREATING DEAERATION SYSTEM DIAGRAM



W_m = MIXED CONDENSATE , LB/HR.
S = PEGGING STEAM LB/HR.
W_P = CONDENSATE RETURN FROM ORIGINAL CIRCUIT LB/HR.
W_S = PRIMARY CONDENSATE RETURNED FROM SSGS LB/HR.
t = TEMPERATURE °F
h = ENTHALPY BTU/LB.
SUBSCRIPTS : S, W_P, W_S, W_m, AND W_2 REPRESENT PEGGING STEAM INLET. CONDENSATE INLET AND WATER OUTLET RESPECTIVELY.

FIG. 9

CONDENSATE DEAERATION SYSTEM DIAGRAM



S = PEGGING STEAM LB/HR.
W1 = TREATED WATER INLET LB/HR.
W2 = TREATED FEED WATER OUTLET LB/HR.
B = BLOWDOWN LB/HR.
t = TEMPERATURE °F
h = ENTHALPY BTU/LB.
SUBSCRIPTS : S, W1, W2, AND B REPRESENT PEGGING STEAM INLET, WATER INLET, WATER OUTLET AND BLOWDOWN INDIVIDULLY

FIG. 10

BLOWDOWN HEAT EXCHANGER IN COMBINATION WITH WATER TREATING SYSTEM DEAERATION SYSTEM DIAGRAM

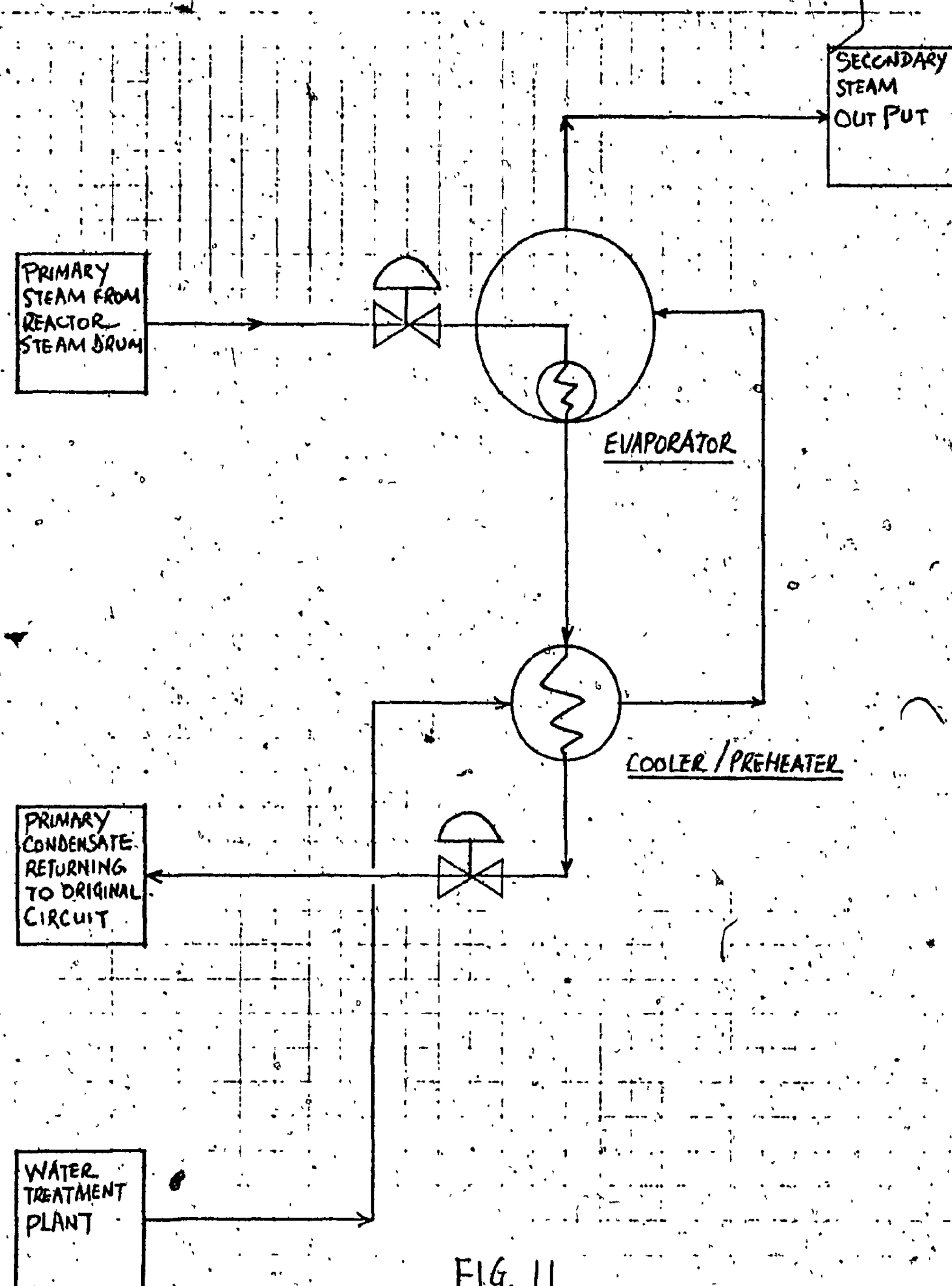


FIG. 11

EVAPORATOR AND COOLER / PREHEATER ONLY
WITHOUT CONDENSATE COLLECTOR

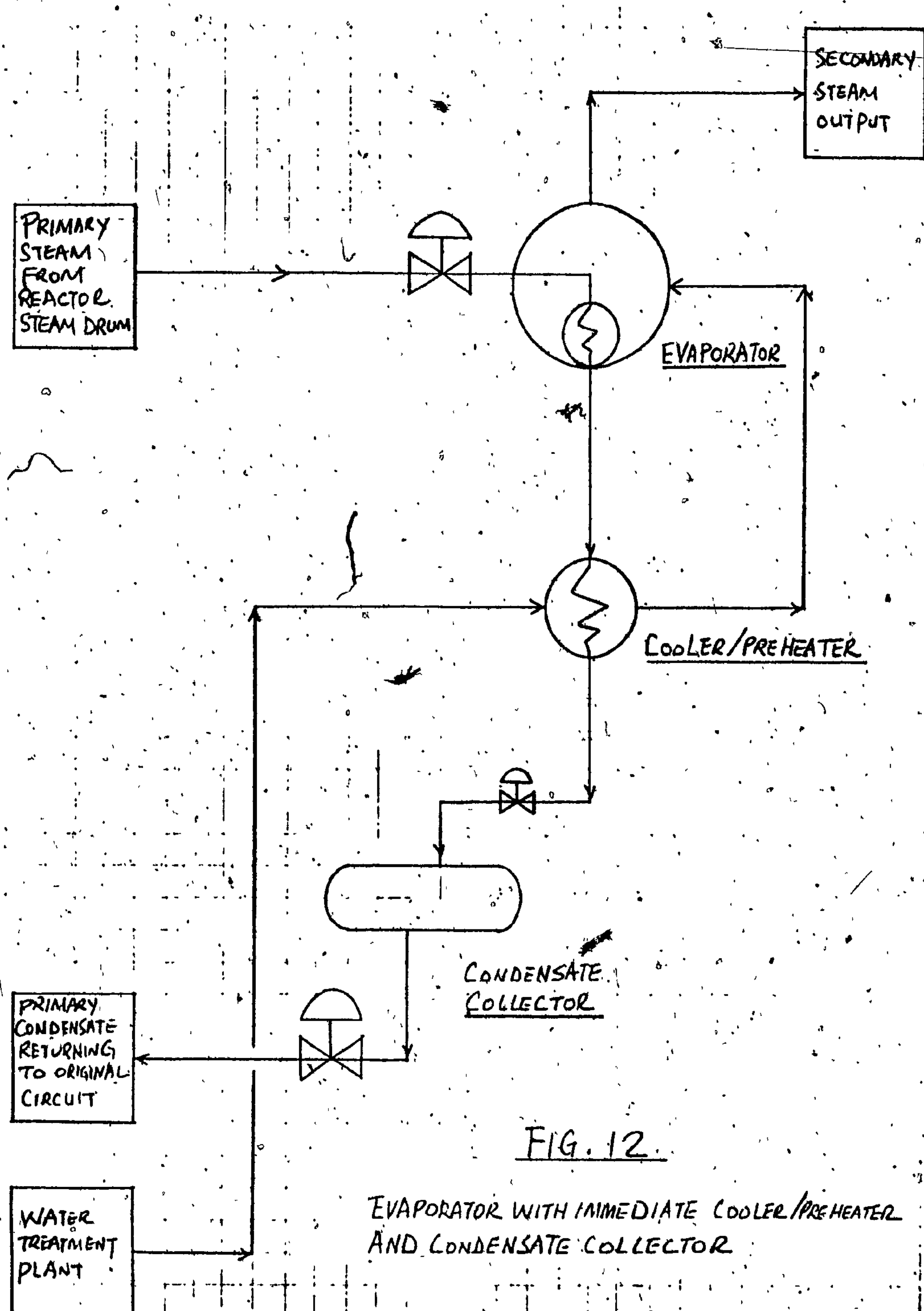


FIG. 12.

EVAPORATOR WITH IMMEDIATE COOLER/PREHEATER AND CONDENSATE COLLECTOR

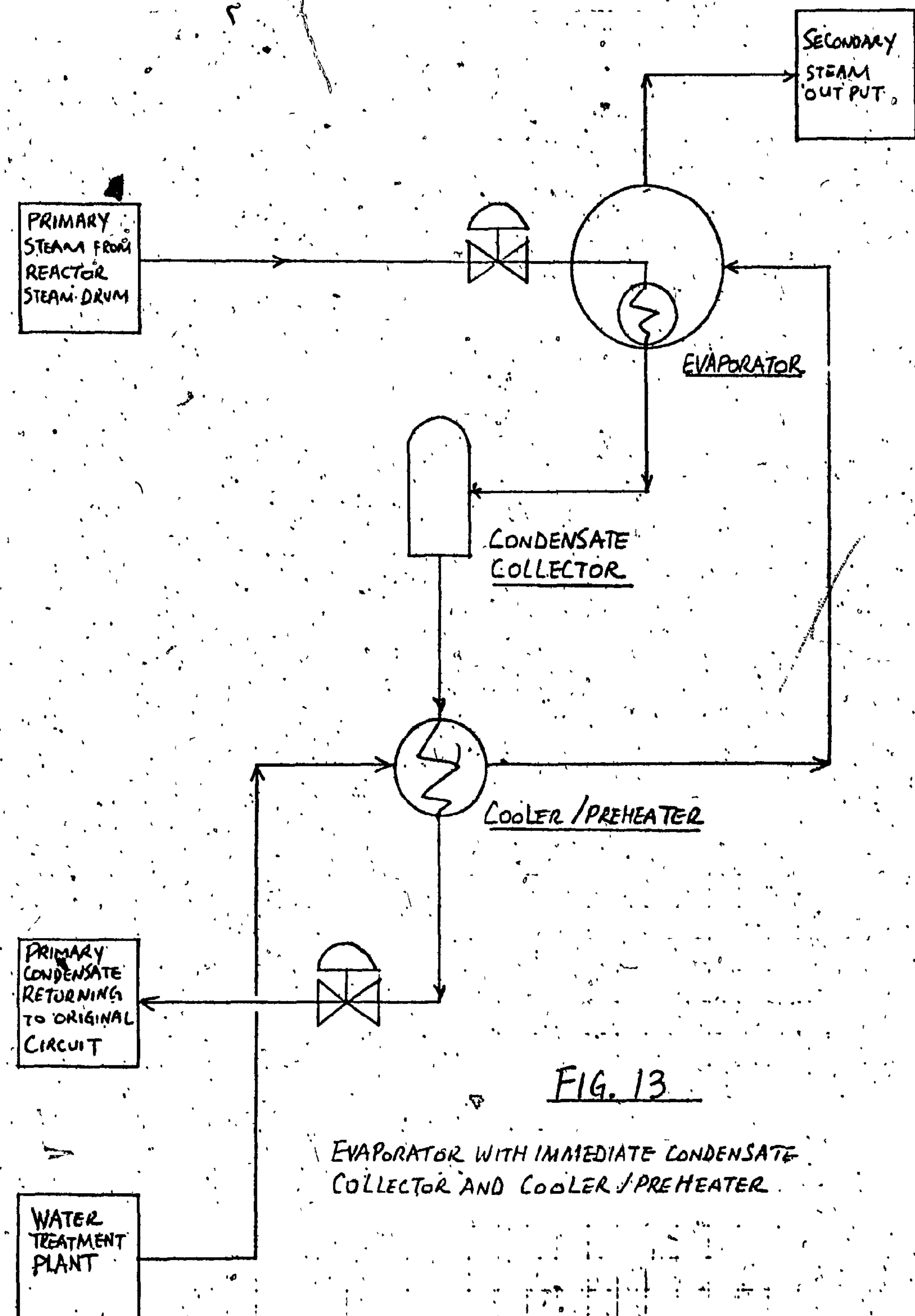


FIG. 13

EVAPORATOR WITH IMMEDIATE CONDENSATE COLLECTOR AND COOLER / PREHEATER

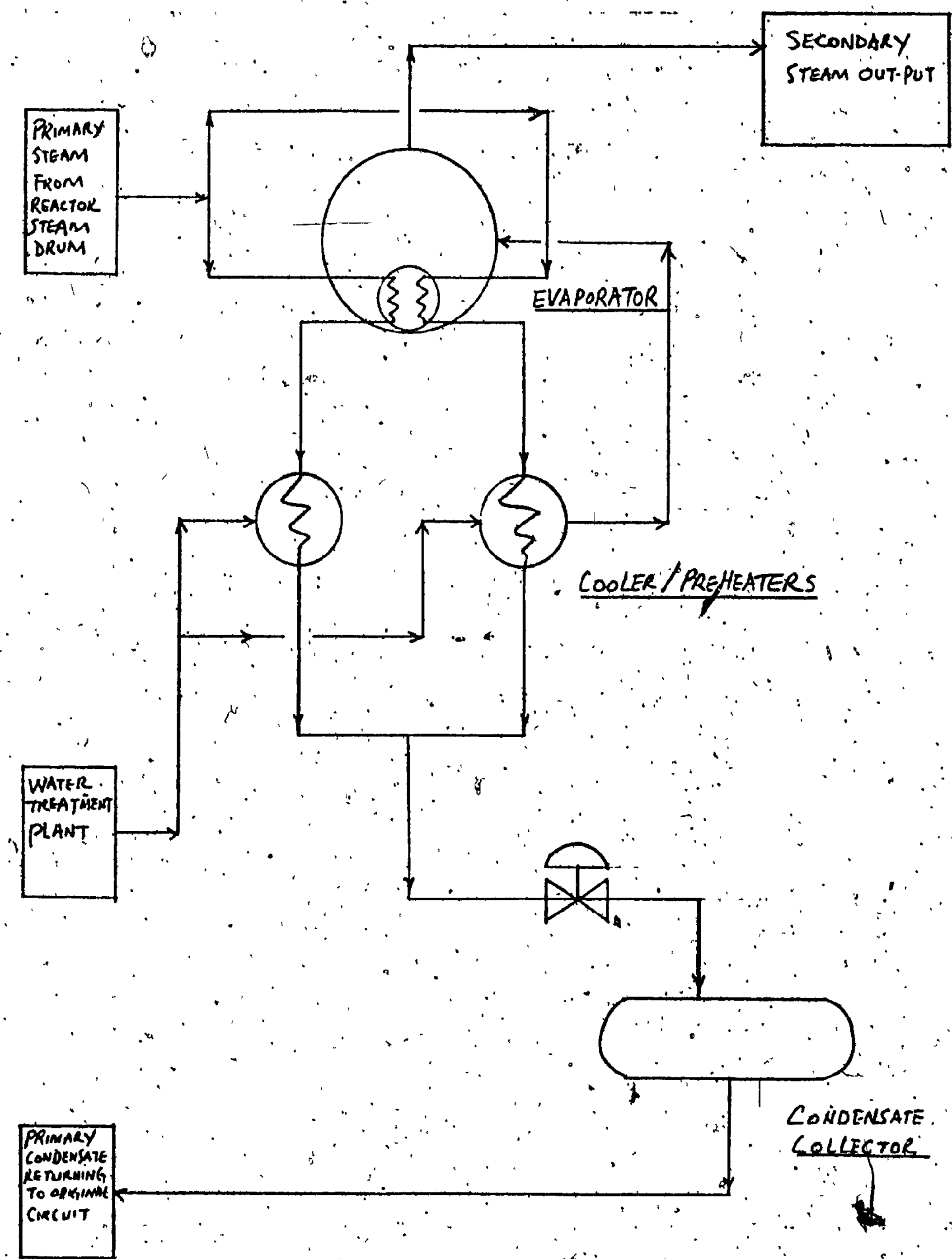


FIG. 14

EVAPORATOR WITH SPLIT COOLER / PREHEATER IN PARALLEL AND CONDENSATE COLLECTOR

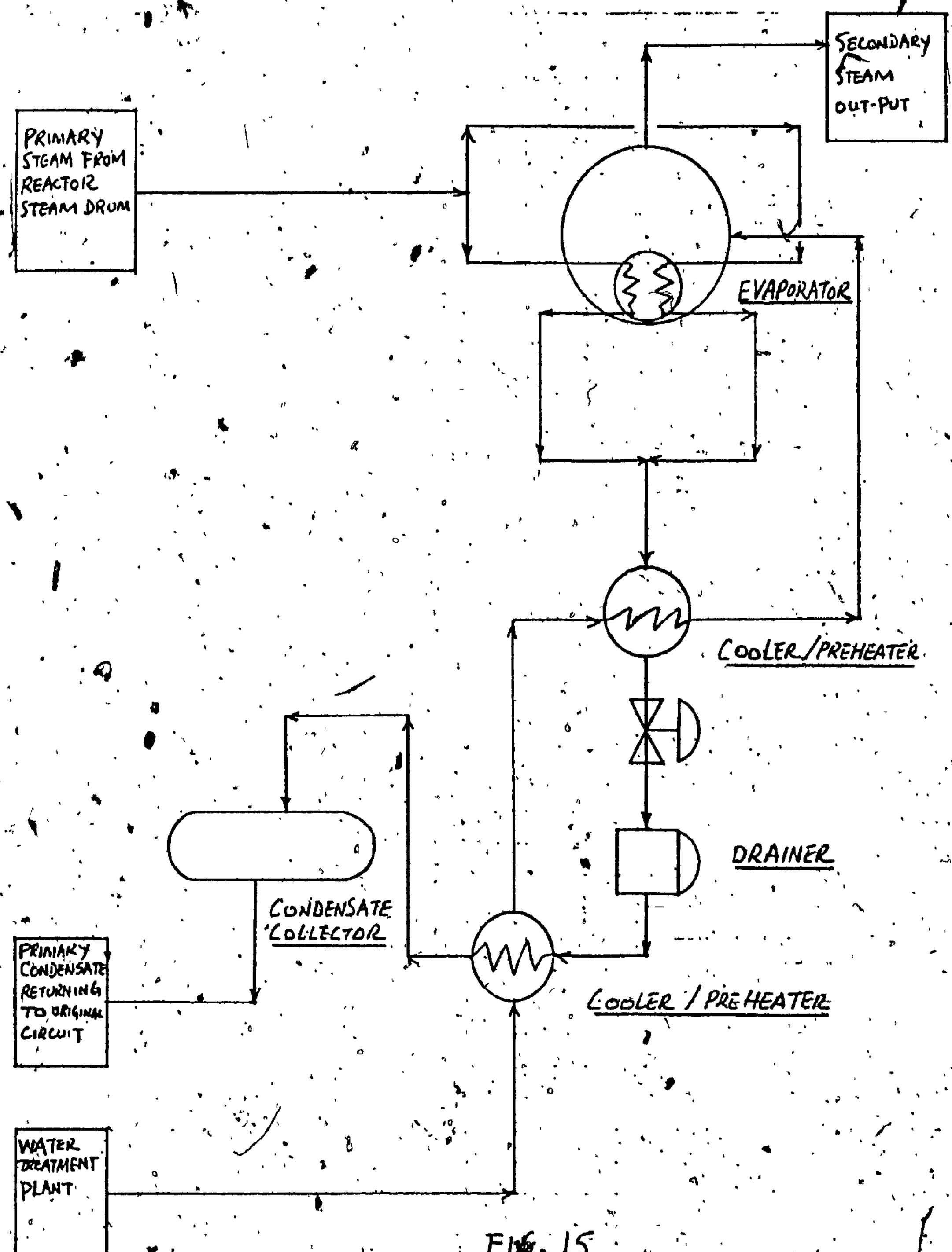


FIG. 15

EVAPORATOR WITH SPLIT COOLER/PREHEATER IN SERIES AND DRAINER IN BETWEEN, AND CONDENSATE COLLECTOR

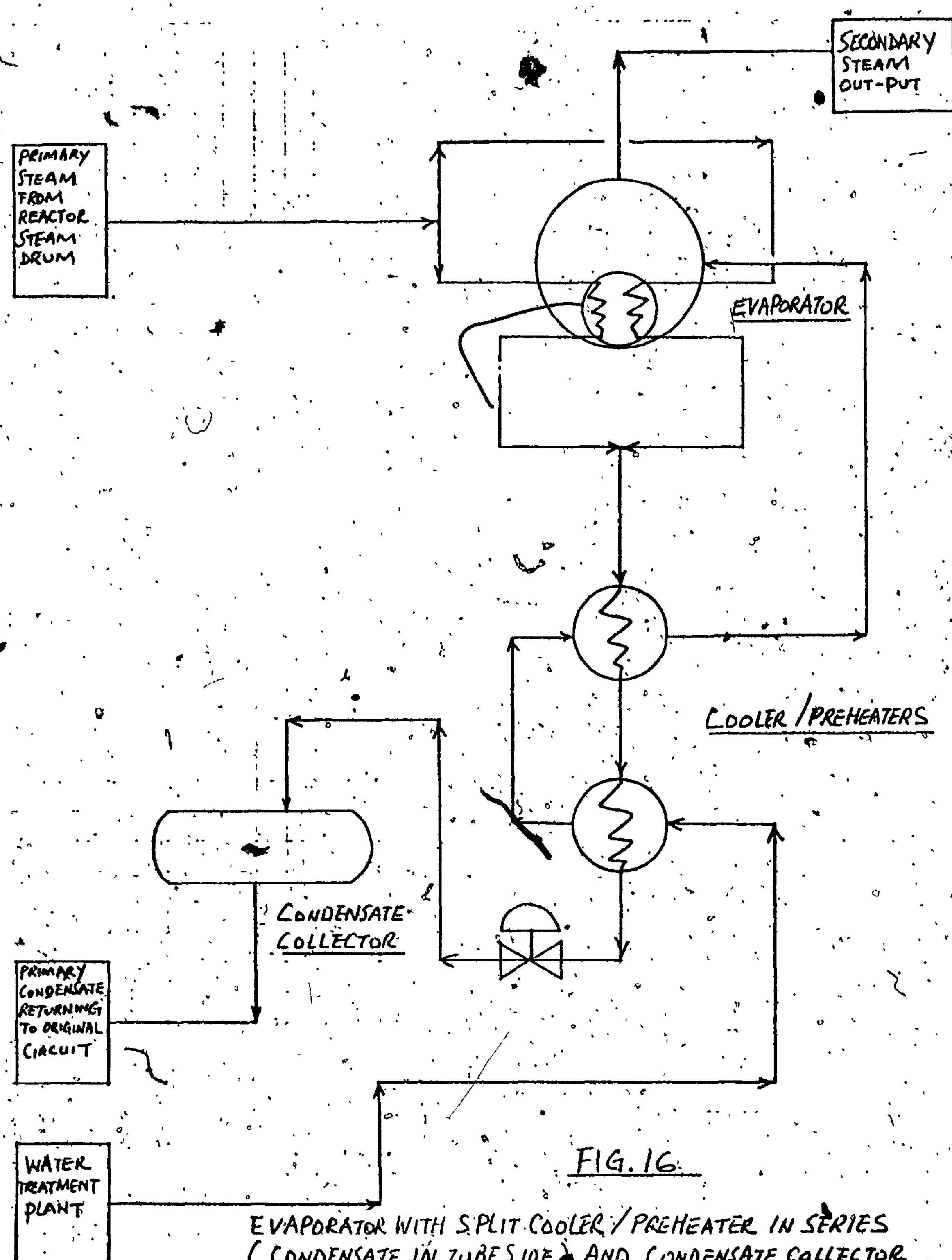


FIG. 16

EVAPORATOR WITH SPLIT COOLER / PREHEATER IN SERIES (CONDENSATE IN TUBE SIDE) AND CONDENSATE COLLECTOR

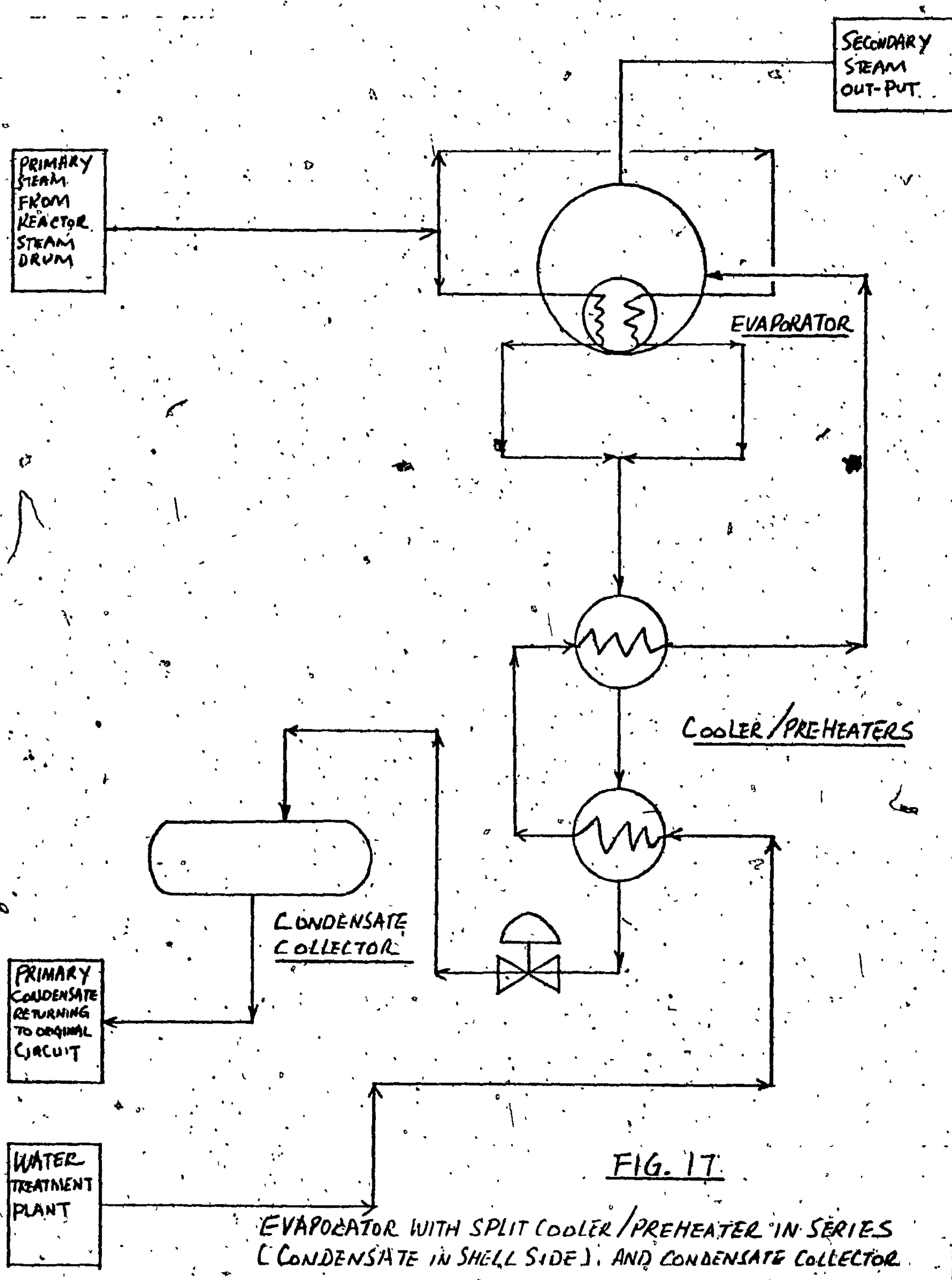


FIG. 17

EVAPORATOR WITH SPLIT COOLER/PREHEATER IN SERIES (CONDENSATE IN SHELL SIDE), AND CONDENSATE COLLECTOR

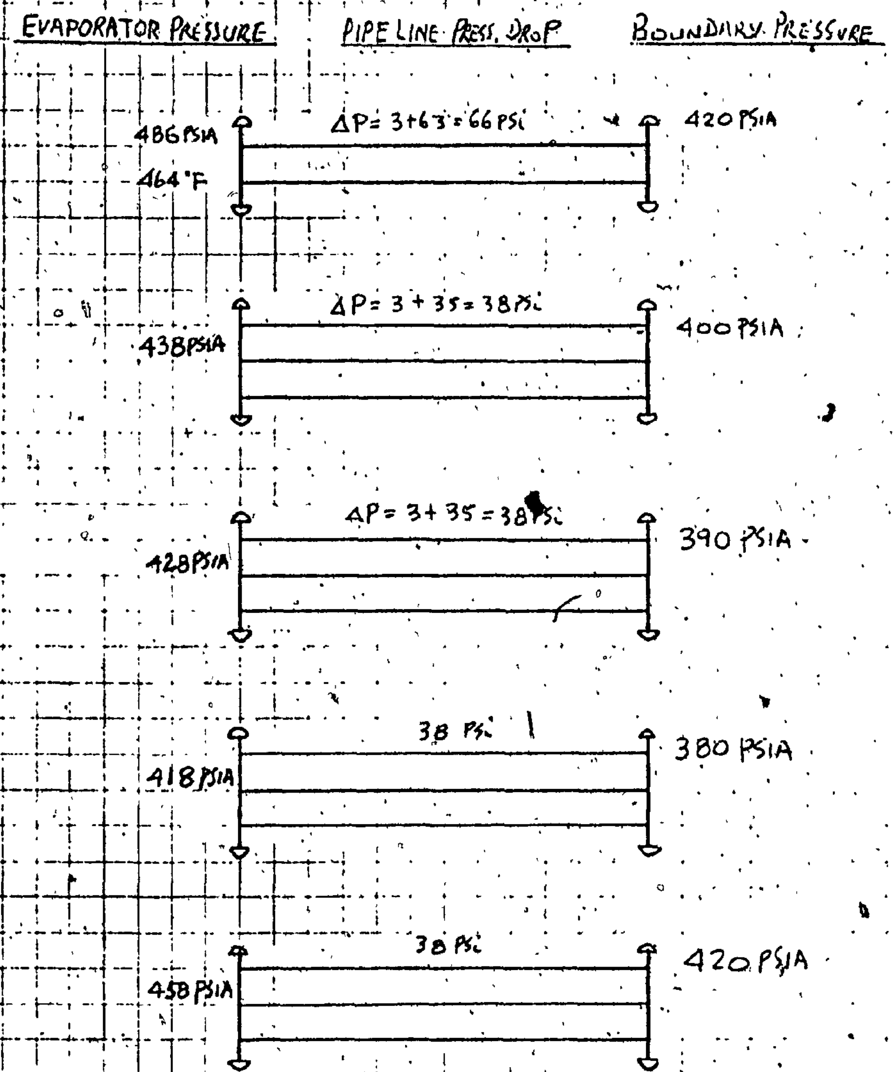


FIG. 18

STEAM DELIVERY LINES.