جامعة القاهرة_ كلية الهندسة
قسم هندسة القوى الميكانيكية
معمل التحكم الأوتوماتيكي

IAET Institute of Aviation Engineering and Technology
معهد هندسة وتكنولوجيا الطيران

ACC Virtual Labs
Automatic Control Circuits & Virtual Labs for Mechanical Power Systems

MEP 440 Embaba B. Sc. Design Project- Year 2011/2012

Investigation & Verification of 3 Automatic Control Virtual Labs: Water Tube Boiler, Steam Turbine Plant & Pumping Plant

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Abstract:

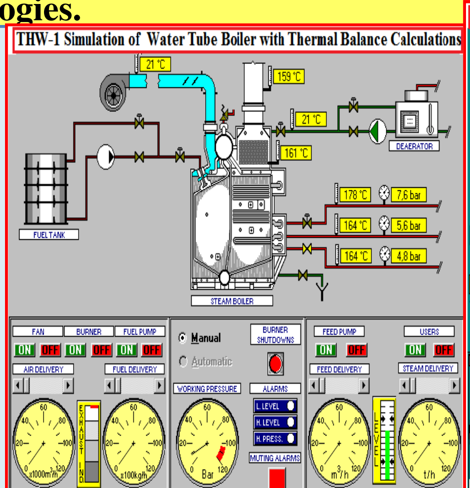
The goals of this project are to investigate- in details- and to examine the operation and performance of 3 Automatic Control Virtual Lab Application. These Labs are: Water Tube Boiler, Steam Turbine Power Plant, and Central Pumping Plant with Tank Filling program.

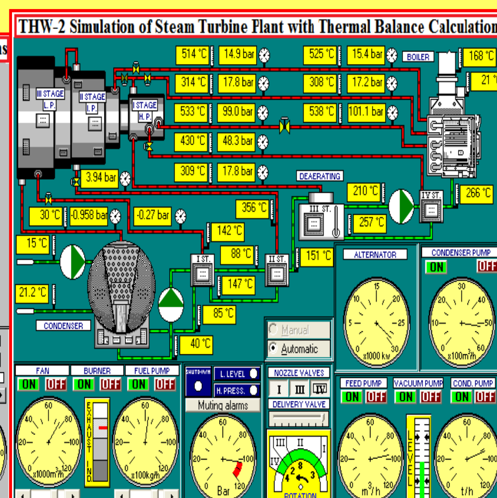
Specific Objectives of this design project:

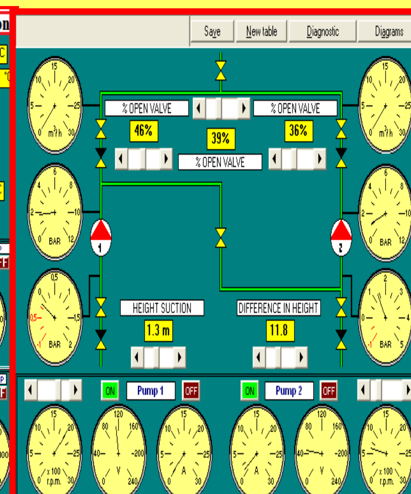
1) **Identifying the main concepts of industrial automatic control systems in three mechanical power applications by modern computer-based programs which simulate those practical control systems.**
2) **Investigation of the tree Applications Automatic Control Virtual Labs to understand their functions, how they work and what are their input and output signals …etc (there are more than 16 virtual labs in ACC).**
3) **Verification of accuracy and validity of the results obtained by those three virtual labs through performing engineering and scientific calibrations for those virtual labs. The calibration is done by comparing internal calculations done by those virtual labs with external engineering calculations using thermo-dynamic, conservation equations, and thermo-fluid relations to get the same output results.**
4) **Training students and engineers on Technical Report Writing and Presentation Skills for each Lab.**
5) **Enhancing the skills of Searching for information and adopting self learning capabilities related to Automatic systems and modern computer technologies.**

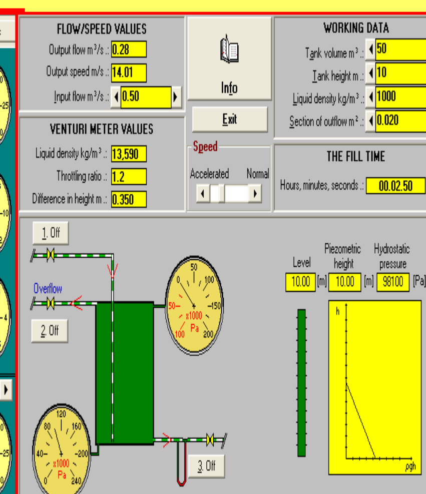
Contents:

- **Water tube boiler**
  - Virtual Lab.User manual
  - Controlling Technology
  - Virtual Lab.Calibration
- **Steam turbine Plant**
  - Virtual Lab.User manual
  - Controlling Technology
  - Virtual Lab.Calibration
- **Central Pumping plant**
  - Virtual Lab.User manual
  - Controlling Technology
  - Virtual Lab.Calibration

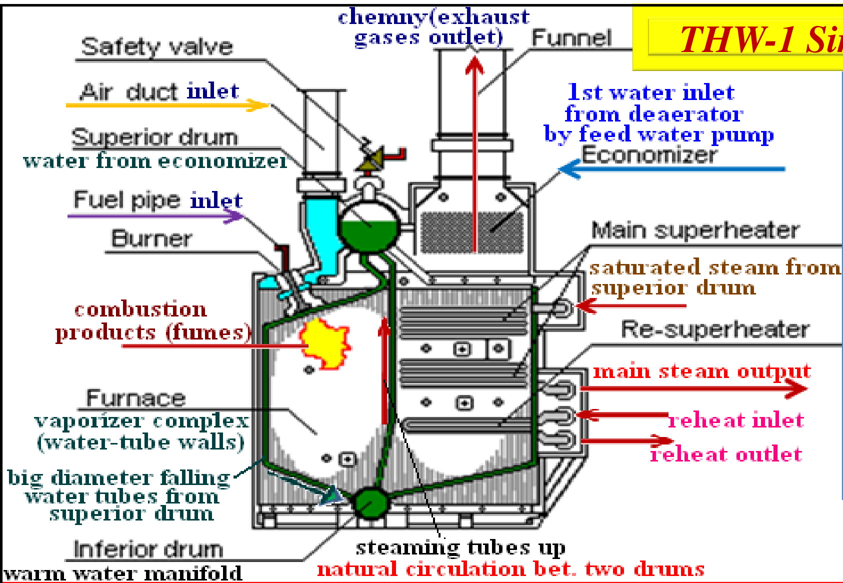


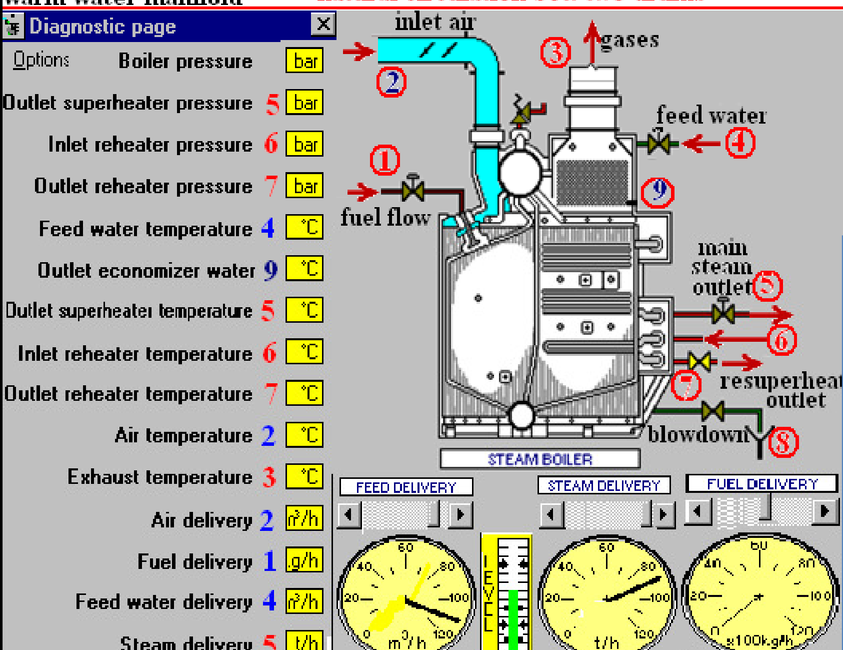


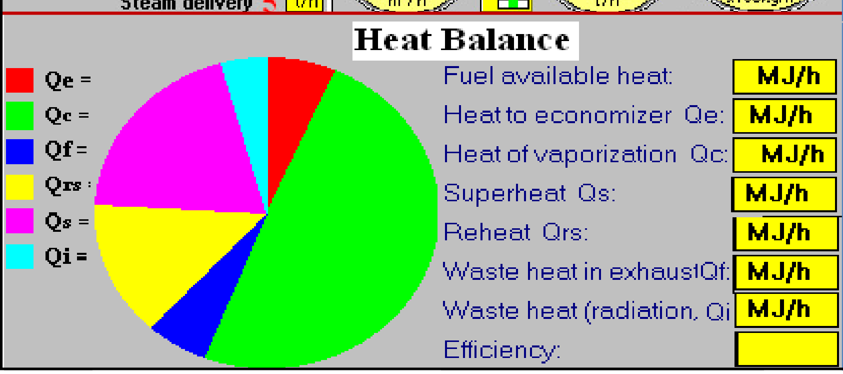


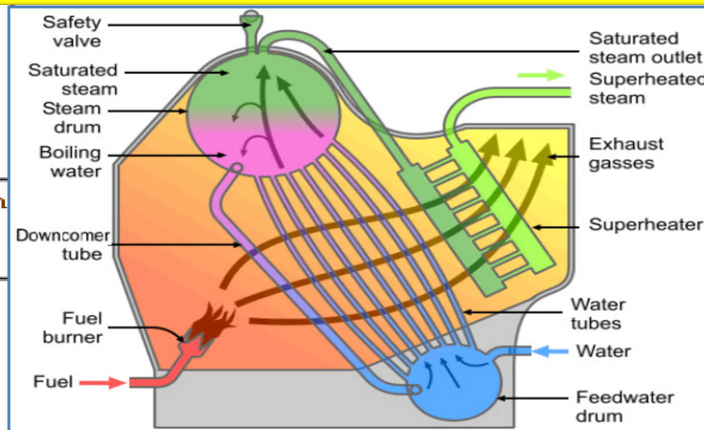


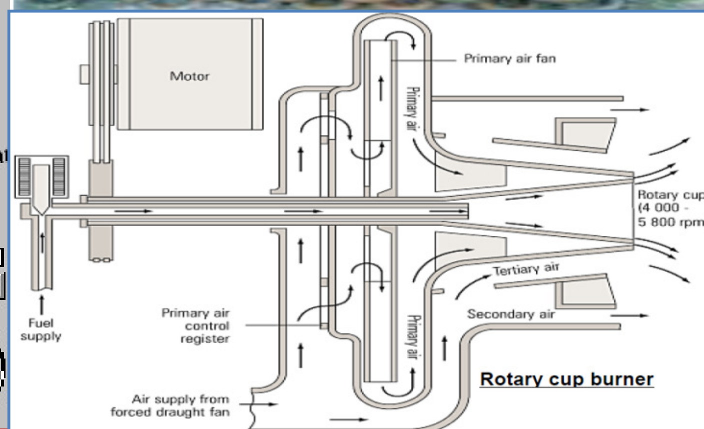
THW-1 Simulation of a water-tube boiler heating unit with thermal balance calculation

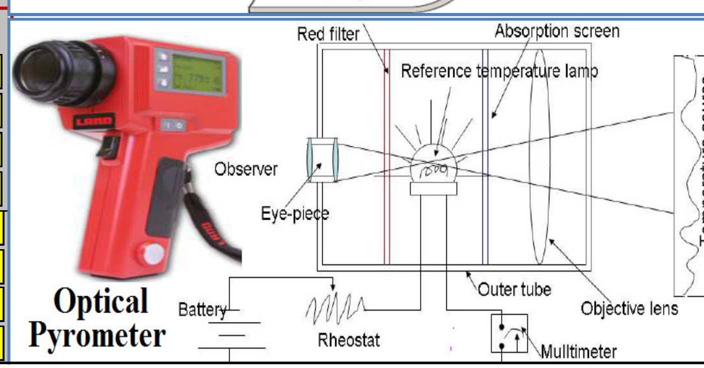


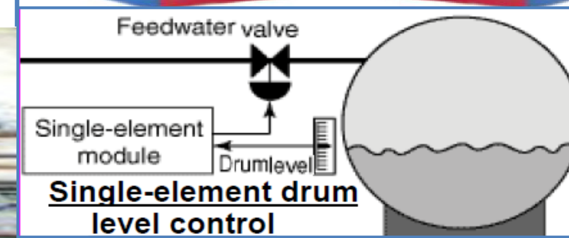


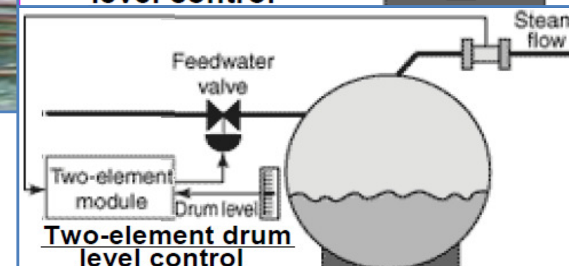


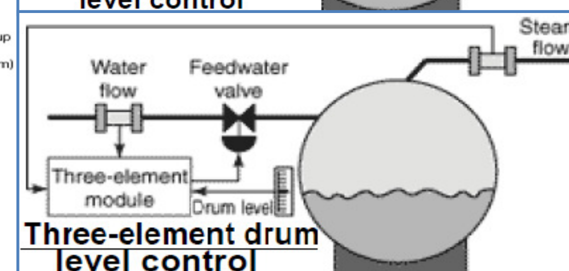


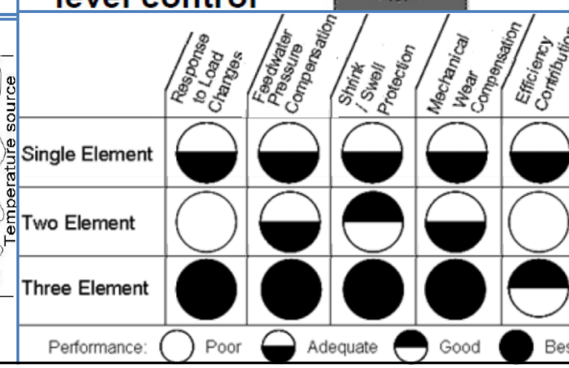


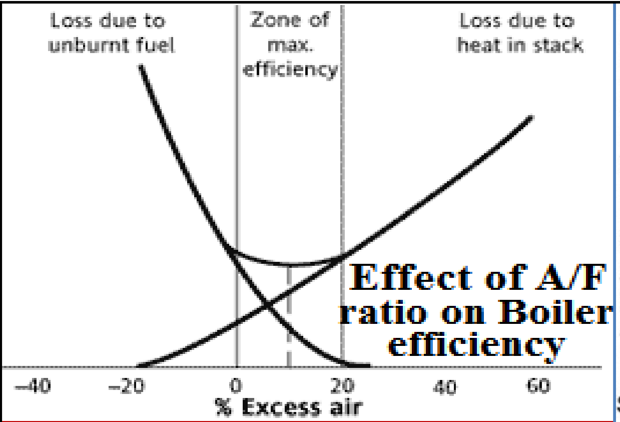








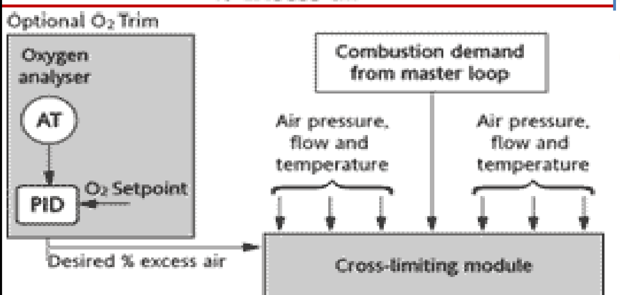




Boiler pressure	101.9 bar
Outlet superheater pressure	100.1 bar
Inlet reheater pressure	17.3 bar
Outlet reheater pressure	15.5 bar
Feed water temperature	206 °C
Outlet economizer water temp	260 °C
Outlet superheater temperature	537 °C
Inlet reheater temperature	306 °C
Outlet reheater temperature	525 °C
Air temperature	10 °C
Exhaust temperature	160 °C
Air delivery	85024.1175 m³/h
Fuel delivery	7185.3715 kg/h
Feed water delivery	94 m³/h
Steam delivery	90 t/h

Calibration of Virtual Lab. Program

Fuel available heat:....308,458 MJ/h
Heat to economizer....Qe:20,160 MJ/h
Heat of vaporization.Qc:147,330 MJ/h
Superheat............Qs:66,330 MJ/h
Reheat...............Qrs:43,650 MJ/h
Waste heat in funnel by exhaust gas Qf:18,007 MJ/h
Waste heat(radiation,unburnt) Qi:12,980 MJ/h
Efficiency.........................:0.900



Cross-limiting combustion control modulewith 6 trim (Enhanced Cross-limitting)

The Re-superheater is activated (the re-superheater outlet valve is opened)

Quantity of available heat in the fuel

$Q_d : Q_d = P_{cs} * P_g$ = 42.070*7185.3715=302289 MJ/hr $\quad Q_d = 302.289 \frac{MJ}{hr}$

Quantity of heat used in the economizer

$Q_e = P_v * (h_v - h_i) = 90 * 10^3 * 4270 * (260 - 206) = 20752 \frac{MJ}{hr} \quad Q_e = 20.752 \frac{MJ}{hr}$

Quantity of heat used in the generator tubes

$Q_g = P_v * (h_v - h_e) \quad h_e = 1110.58 \frac{KJ}{Kg}$

To get h_e , $Q_e = 20752 = 90 * 10^3 * (h_e - h_i)$, So

$Q_g = 90 * 10^3 * (2720 - 1110.58) = 144878 \frac{MJ}{hr} \quad Q_g = 144.878 \frac{MJ}{hr}$

Quantity of heat used in the superheater :

$Q_s = P_v * (h_s - h_v) = 90 * 10^3 * (3465.2 - 2720) \quad Q_s = 67.068 \frac{MJ}{hr}$

Quantity of heat used in the re-superheater :

$Q_{rs} = P_v * (h_{ru} - h_{re}) = 90 * 10^3 * (3540 - 3050) \quad Q_{rs} = 44.100 \frac{MJ}{hr}$

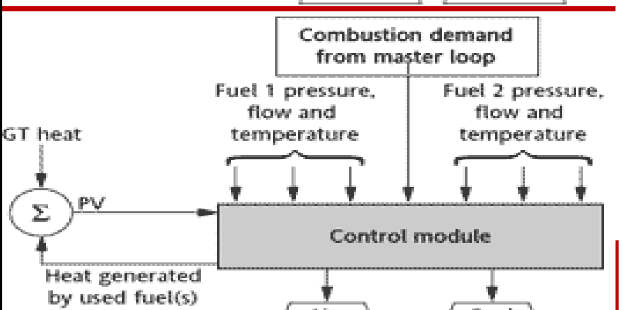
Quantity of heat lost in the funnel for fumes :

$Q_f = (P_a * \beta + P_g) * C_{mr} * (T_f - T_a)$

$Q_f = (85024.1175 * 1.225 + 7185.3715) * 1.025 * (160 - 10) \quad Q_f = 17.119 \frac{MJ}{hr}$

Quantity of heat lost for radiation and unburned:

$Q_i = Q_d - (Q_e + Q_g + Q_s + Q_{rs} + Q_f) \quad Q_i = 8.502 \frac{MJ}{hr}$



Total Heat Control System

Calibration

Comparison table between our calculations and the program calculation

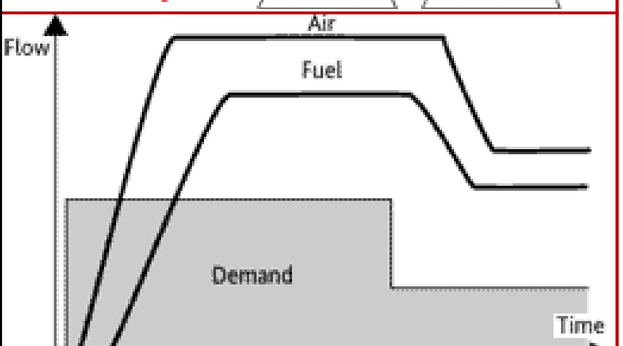
	Our Calculaions	Program Calculations	Error
Q_d	302.289 MJ/hr	308.458 MJ/hr	2%
Q_e	20.752 MJ/hr	20.160 MJ/hr	-3%
Q_g	144.848 MJ/hr	147.330 MJ/hr	1.7%
Q_s	67.068 MJ/hr	66.330 MJ/hr	-1.1%
Q_{rs}	44.100 MJ/hr	43.650 MJ/hr	-1.03%
Q_f	17.119 MJ/hr	18.007 MJ/hr	4.9%
Q_i	8.502 MJ/hr	12.980 MJ/hr	34.5%

$$\text{Efficiency} = \frac{Q_e + Q_g + Q_s + Q_{rs}}{Q_d}$$

Efficiency Calculated = 0.916 =91.6%

Efficiency by program = 0.9 = 90%

$$\text{Error} = \frac{0.9 - 0.916}{0.9} = -1.78\%$$

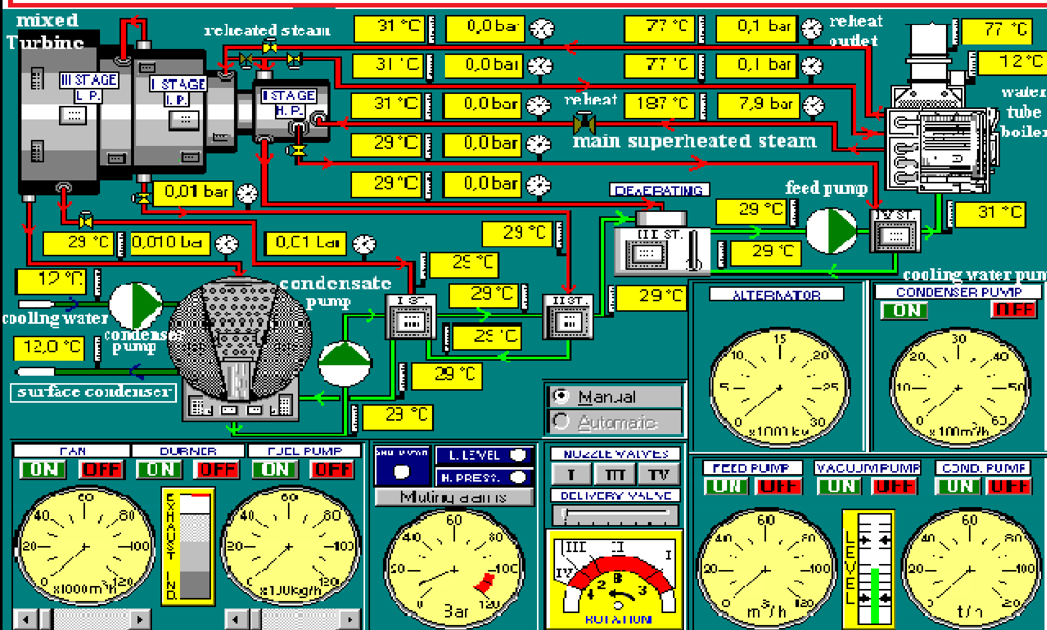


Cross-limiting Combustion contrl mechanism

Comments and Notes :

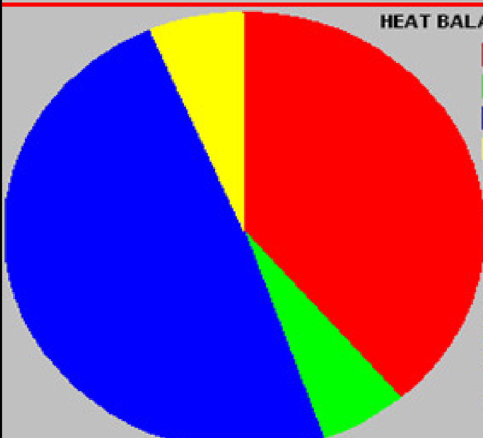
When we put error +- 4% , We found all values are correct except Q_i error = 34.5%
We have this large error because of C.V used in program is not correct

THW-2 Simulation of a Steam Turbine Plant with thermal balance calculation

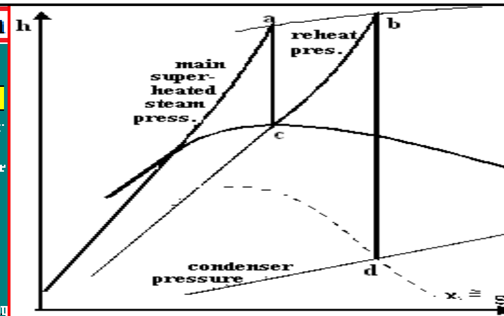


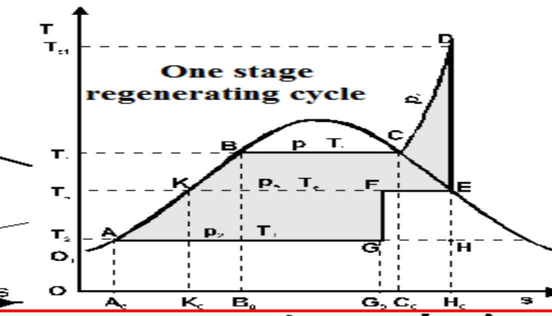
Boiler pressure	43.8 bar	Electric power	0 kW
Outlet Superheater pressure	43.0 bar	Outlet Superheater temperature	430 °C
Inlet Hight Pressure Turbine pressure	0.0 bar	Inlet Hight Pressure Turbine temperature	29 °C
Outlet Hight Pressure Turbine pressure	0.0 bar	Outlet Hight Pressure Turbine temperature	29 °C
Inlet Reheater pressure	0.1 bar	Inlet Reheater temperature	73 °C
Outlet Reheater pressure	0.1 bar	Outlet Reheater temperature	73 °C
Inlet Intermediate Pressure Turbine pressure	0.0 bar	Inlet Intermediate Pressure Turbine temperature	29 °C
Inlet Condenser pressure	-0.19 bar	Inlet Condenser temperature	27 °C
Inlet I Stage steam pressure	0.01 bar	Inlet I Stage steam temperature	27 °C
Inlet II Stage steam pressure	0.0 bar	Inlet II Stage steam temperature	27 °C
Inlet III Stage steam pressure	0.0 bar	Inlet III Stage steam temperature	27 °C
Inlet IV Stage steam pressure	0.0 bar	Inlet IV Stage steam temperature	27 °C
Steam delivery	0.00 t/h	Water vapor ratio	0.000
I Bleeding delivery	0.00 t/h	II Bleeding delivery	0.00 t/h
III Bleeding delivery	0.00 t/h	IV Bleeding delivery	0.00 t/h
Air temperature	10 °C	Exhaust temperature	151 °C
Air delivery	18,200 m³/h	Fuel delivery	1,454 Kg/h
Feed water delivery	0.00 m³/h	Condenser cooling water delivery	5149.4 m³/h
Inlet Condenser cooling water temperature	4 °C	Outlet Condenser cooling water temperature	4.0 °C
Inlet I Stage feed water temperature	27 °C	Inlet Condenser condensate temperature	27 °C
Inlet II Stage feed water temperature	27 °C	Inlet I Stage condensate temperature	27 °C
Inlet III Stage feed water temperature	27 °C	Inlet III Stage condensate temperature	27 °C
Inlet IV Stage feed water temperature	27 °C	Feed water temperature	29 °C

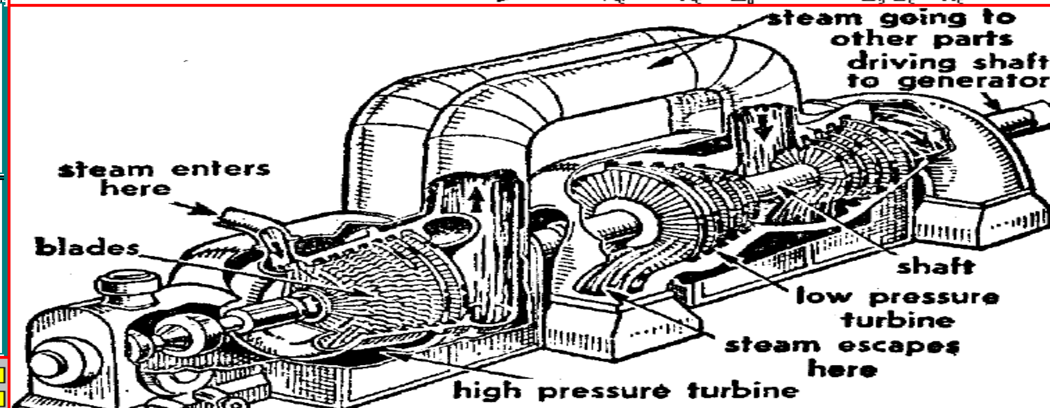
HEAT BALANCE



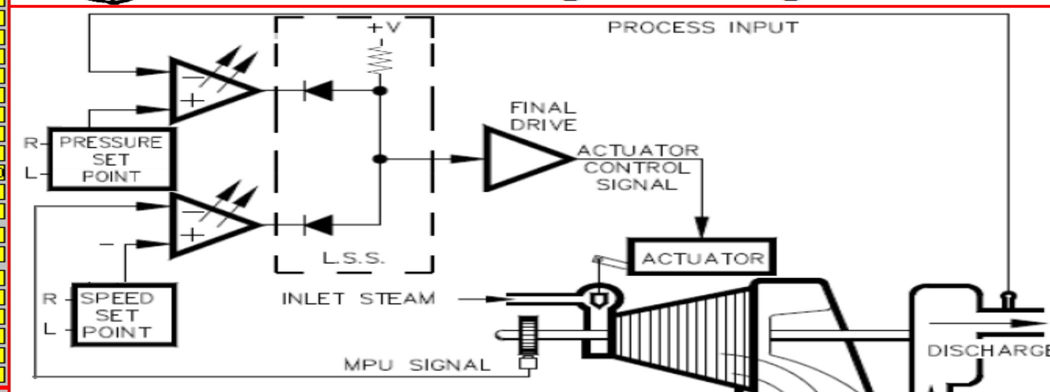
Fuel available heat	266,583 MJ/h
Shaft horsepower Pu	103,168 MJ/h
Haet lost in the funnel by exhaust gas Qf	15,563 MJ/h
Subtracted heat from the condenser Qa	131,010 MJ/h
Other loss (radiation, mechanical,..) Qi	16,841 MJ/h
Efficiency	0.387
Steam specific consumption	0.221 kg/kWh
Fuel specific consumption	3.16 kg/kWh



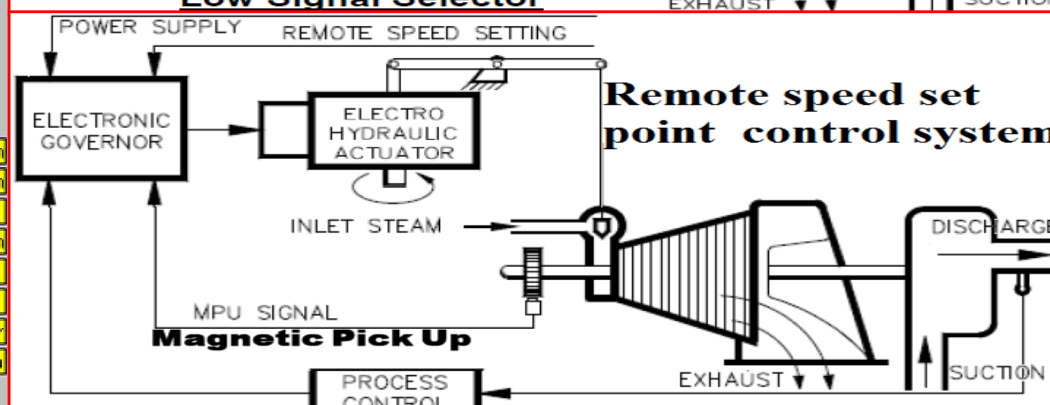




Non-extracting multi-stage Turbine

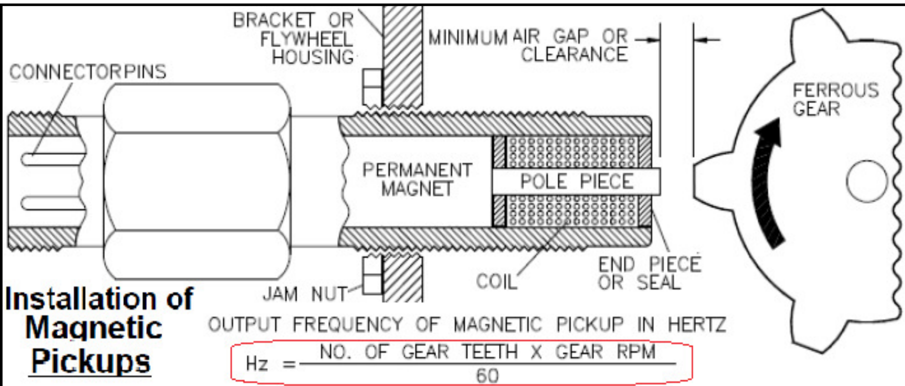


Low Signal Selector

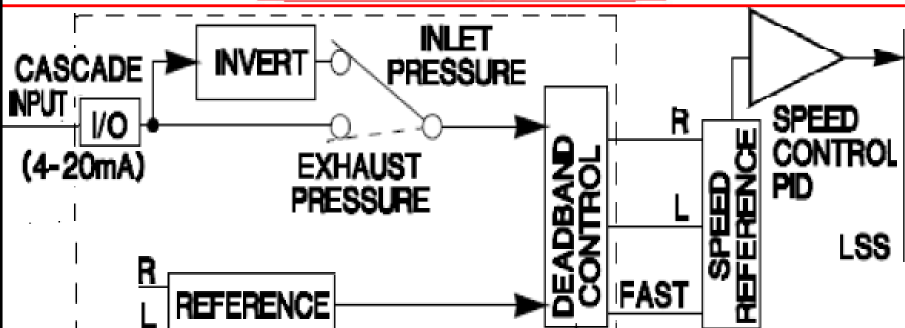


Remote speed set point control system

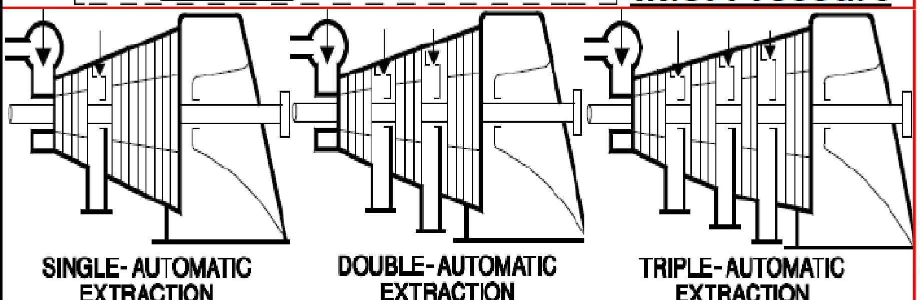
Magnetic Pick Up

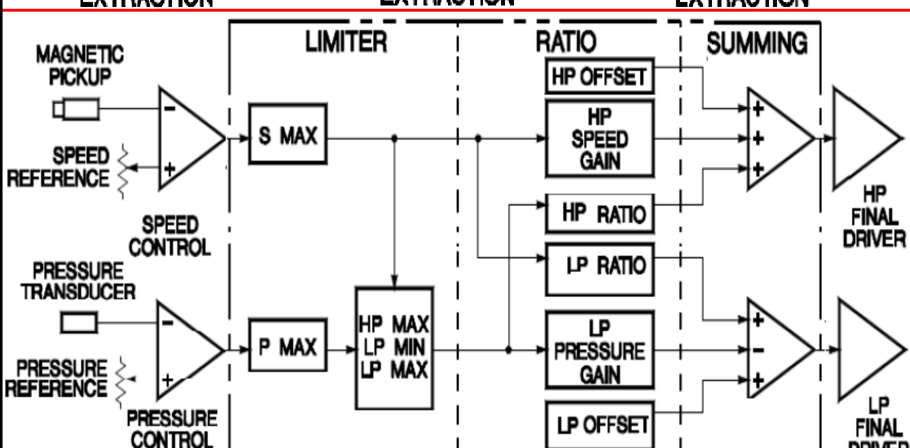


Installation of Magnetic Pickups

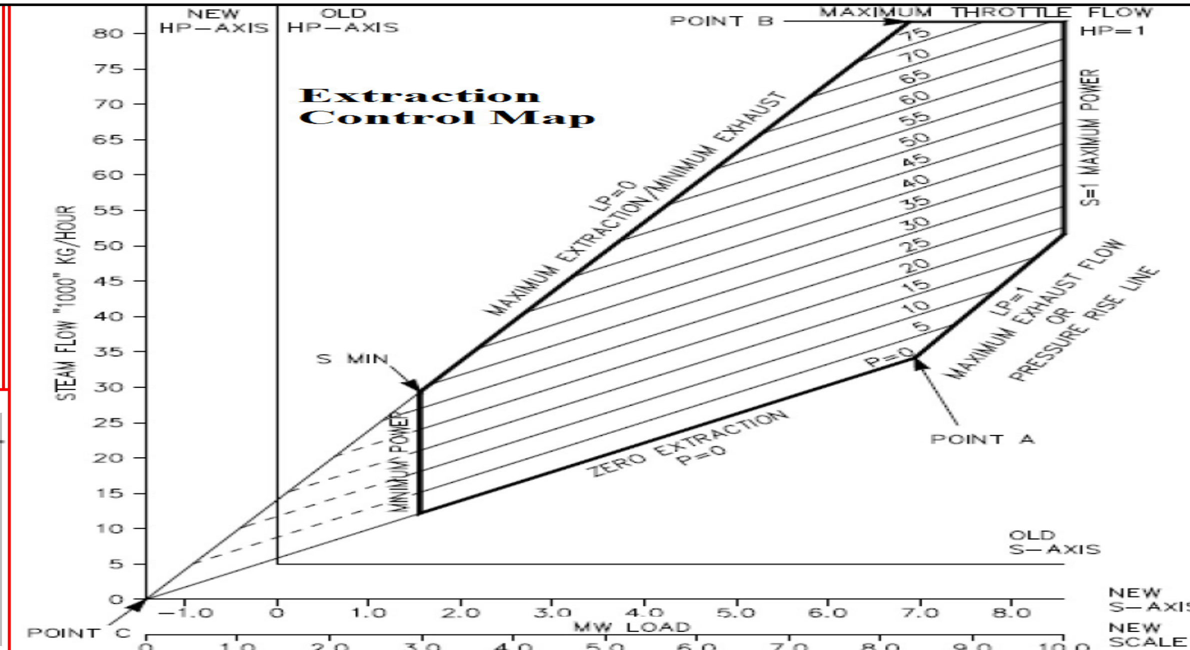


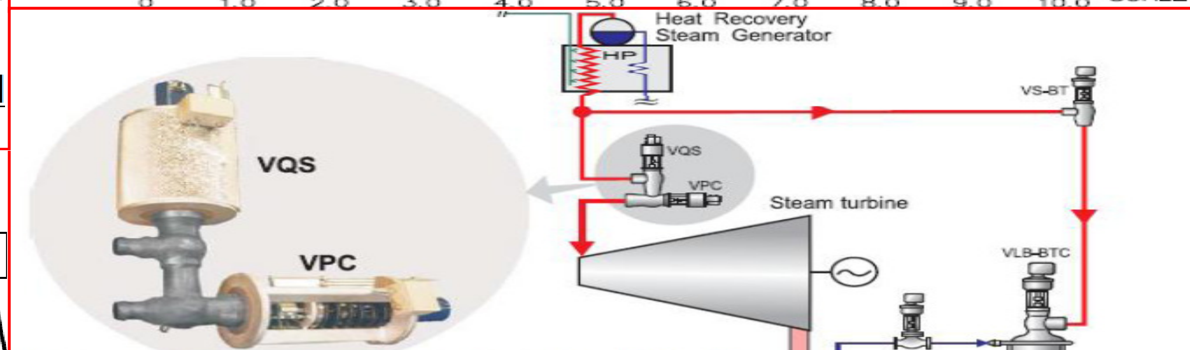
Cascade Control Inlet Pressure





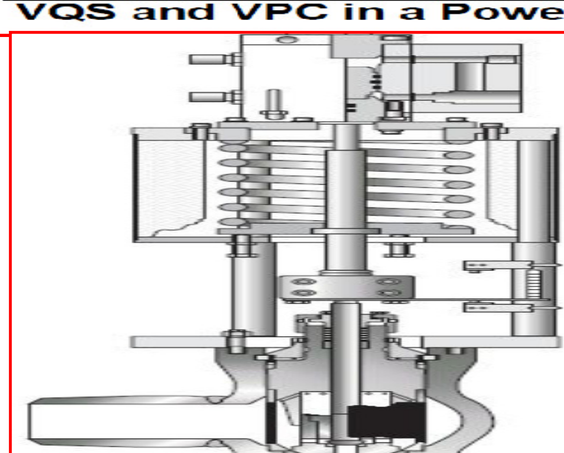
Control of Extraction different Ratios



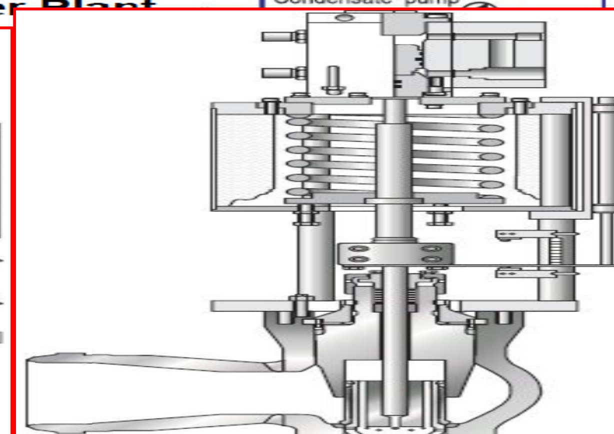


A typical installation of the CCI turbine quick stop valve VQS and turbine flow control valve VPC in a power station. (Example: Combined cycle single pressure)

Typical installation of Steam Turbine VQS and VPC in a Power Plant

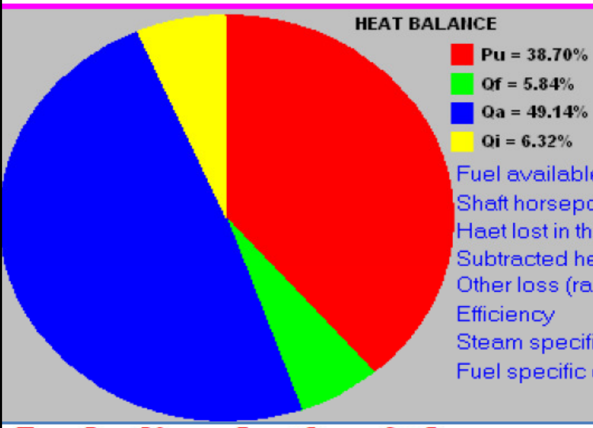


the VQS turbine quick stop valve



VPC turbine flow control valve with the optional quick closing function

Boiler pressure	100.1 bar	Electric power	28,658 kW
Outlet Superheater pressure	98.3 bar	Outlet Superheater temperature	533 °C
Inlet Hight Pressure Turbine pressure	96.2 bar	Inlet Hight Pressure Turbine temperature	528 °C
Outlet Hight Pressure Turbine pressure	16.3 bar	Outlet Hight Pressure Turbine temperature	312 °C
Inlet Reheater pressure	15.8 bar	Inlet Reheater temperature	306 °C
Outlet Reheater pressure	14.0 bar	Outlet Reheater temperature	525 °C
Inlet Intermediate Pressure Turbine pressure	13.6 bar	Inlet Intermediate Pressure Turbine temperature	514 °C
Inlet Condenser pressure	-0.97 bar	Inlet Condenser temperature	24 °C
Inlet I Stage steam pressure	-0.40 bar	Inlet I Stage steam temperature	129 °C
Inlet II Stage steam pressure	3.3 bar	Inlet II Stage steam temperature	348 °C
Inlet III Stage steam pressure	16.3 bar	Inlet III Stage steam temperature	307 °C
Inlet IV Stage steam pressure	45.9 bar	Inlet IV Stage steam temperature	426 °C
Steam delivery	90.53 t/h	Water vapor ratio	0.911
I Bleeding delivery	5.81 t/h	II Bleeding delivery	7.14 t/h
III Bleeding delivery	7.28 t/h	IV Bleeding delivery	10.16 t/h
Air temperature	8 °C	Exhaust temperature	155 °C
Air delivery	75,009 m³/h	Fuel delivery	6,339 Kg/h
Feed water delivery	94.53 m³/h	Condenser cooling water delivery	5071.6 m³/h
Inlet Condenser cooling water temperature	2 °C	Outlet Condenser cooling water temperature	8.2 °C
Inlet I Stage feed water temperature	34 °C	Inlet Condenser condensate temperature	82 °C
Inlet II Stage feed water temperature	85 °C	Inlet I Stage condensate temperature	141 °C
Inlet III Stage feed water temperature	145 °C	Inlet III Stage condensate temperature	254 °C
Inlet IV Stage feed water temperature	204 °C	Feed water temperature	263 °C



Fuel available heat	266,583 MJ/h
Shaft horsepower Pu	103,168 MJ/h
Haet lost in the funnel by exhaust gas Qf	15,563 MJ/h
Subtracted heat from the condenser Qa	131,010 MJ/h
Other loss (radiation, mechanical,..) Qi	16,841 MJ/h
Efficiency	0.387
Steam specific consumption	0.221 kg/kWh
Fuel specific consumption	3.16 kg/kWh

Including both of the re-superheater and all of four water heaters # I, II, III, and IV in the plant:

Boiler pressure 100.2 bar	Outlet Superheater pressure 98.4 bar
Inlet Hight Pressure Turbine pressure	96.3 bar
Outlet Hight Pressure Turbine pressure	16.4 bar
Inlet Reheater pressure	15.8 bar
Outlet Reheater pressure	14.0 bar
Inlet Intermediate Pressure Turbine pressure	13.6 bar
Inlet Condenser pressure	-0.97 bar
Inlet I Stage steam pressure	-0.40 bar
Inlet II Stage steam pressure	3.4 bar
Inlet III Stage steam pressure	16.4 bar
Inlet IV Stage steam pressure	46.0 bar
Outlet Superheater temperature	534 °C
Inlet Hight Pressure Turbine temperature	529 °C
Outlet Hight Pressure Turbine temperature	312 °C
Inlet Reheater temperature	306 °C
Outlet Reheater temperature	525 °C
Inlet Intermediate Pressure Turbine temperature	514 °C
Inlet Condenser temperature	24 °C
Inlet I Stage steam temperature	130 °C
Inlet II Stage steam temperature	348 °C
Inlet III Stage steam temperature	307 °C
Inlet IV Stage steam temperature	427 °C

Calibration of Virtual Lab. Program

Fuel available heat:..............................269,542 MJ/h
Shaft horsepower..............................Pu:104,314 MJ/h
Haet lost in the funnel by exhaust gas............Qf:15,736 MJ/h
Subtracted heat from the condenser...........Qa:132,519 MJ/h
Other loss (radiation, mechanical,..)... .Qi:16,975 MJ/h
Efficiency..:0.387
Steam specific consumption.........................:0.221 kg/kWh
Fuel specific consumption..........................:3.16 kg/kWh

- **Available heat (Q_d) :** $Q_d = m_f \cdot C.V$ =6.409*42=269.178 MJ/Kg
- **Power at Turbine axle (P_u) :** $Q_d = 269.178 \frac{MJ}{Kg}$

$$P_u = m_s \cdot (h_2 - h_7) + (m_s - m_{IV})(h_7 - h_8) + (m_s - m_{IV} - m_{III})(h_6 - h_9) + (m_s - m_{IV} - m_{III} - m_{II})(h_9 - h_{10}) + (m_s - m_{IV}$$

Knowing that :

State	Mass Rate (Kg/hr)
m_s	91.54
m_I	5.85
m_{II}	7.22
m_{III}	7.36
m_{IV}	10.27

So , Sub in the above equation by enthalpies and mass flow rates , get : $P_u = 112.43 \frac{MJ}{hr}$

- **Pump Power (P_{pump}):** $P_{pump} = (m_s - m_{IV} - m_{III})(h_{18} - h_{17}) + (m_s)(h_{13} - h_{12})$

$$P_{pump} = (91.54 - 10.27 - 7.36) \cdot (147 - 47) + 91.54 \cdot (1184 - 1128) = 9.229 \frac{MJ}{hr}$$

$P_{pump} = 9.229 \frac{MJ}{hr}$

So , • **Power Net (P_{net}) :** $P_{net} = P_u - P_{pump} = 112.43 - 9.223 = 103.207 \frac{MJ}{hr}$

So , $P_{net} = 103.702 \frac{MJ}{hr}$

- **Lost heat at chimney for fume (Q_f) :**

Q_f = [air delivery (V_a) * air density (ρ_a) + fuel delivery (m_f)] * [fume specific heat $(C_p)_{fume}$

$$Q_f = (75.842 \cdot 1.2 + 6.409) \cdot 1 \cdot (158 - 11) = 14257 \frac{KJ}{hr}$$

So , $Q_f = 14.257 \frac{MJ}{hr}$

- **Heat taken from the condenser (Q_a) :**

Q_a =[condenser cooling water delivery * (water temperature at condenser exit - water temperature at condenser inlet) * water specific heat] $Q_a = (\rho_w \cdot V_w) \cdot Cp_w \cdot (T_{w,o} - T_{w,in})$

$$Q_a = (10^3 \cdot 5127.9) \cdot 4.18 \cdot (11.2 - 5) = 132.894656 \frac{MJ}{hr}$$

Therefore , $Q_a = 132.894656 \frac{MJ}{hr}$

- **Other losses (radiation, incombustibles, mechanicals,etc.)(Q_i):**

$$Q_i = Q_d - Q_i - Q_f - P_{net}$$

$$Q_i = 269.78 - (132.895 + 14.257 + 103.207) = 18.783 \frac{MJ}{h}$$

- **Efficiency :** $\eta = \frac{P_{net}}{Q_d} = \frac{103.207}{269.78} = 0.383$
- **Steam Specific Consumption (C_s) :**

$$C_s = \frac{m_s}{P_{turbine}} = \frac{91.54 \cdot 10^3}{103.207 \cdot \frac{10^3}{3600}} = 3193 \frac{Kg}{KW_h}$$

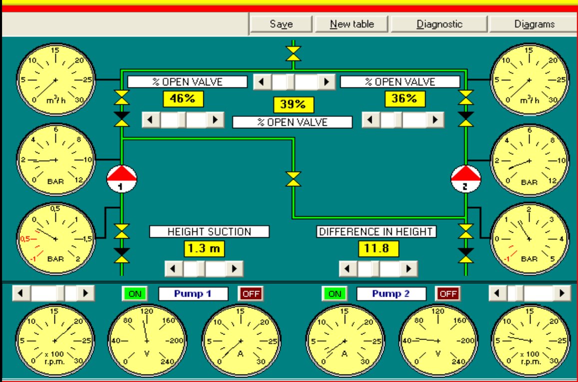
$C_s = 3193 \frac{Kg}{KW_h}$

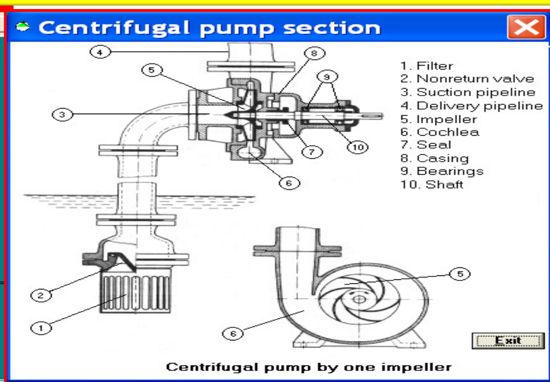
- **Fuel Specific Consumption (C_f) :**

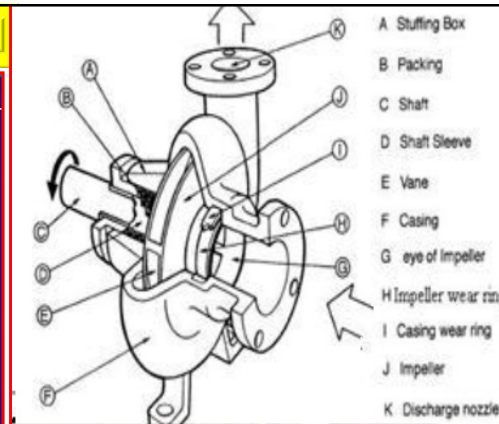
$$C_f = \frac{m_f}{P_{Turbine}} = \frac{6.409 \cdot 10^3}{103.207 \cdot \frac{10^3}{3600}} = 223.6 \frac{Kg}{KW_h}$$

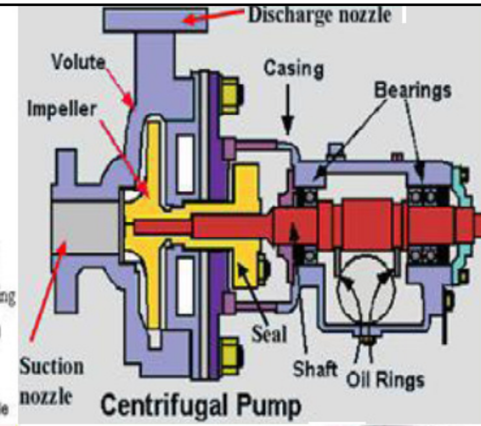
$C_f = 223.6 \frac{Kg}{KW_h}$

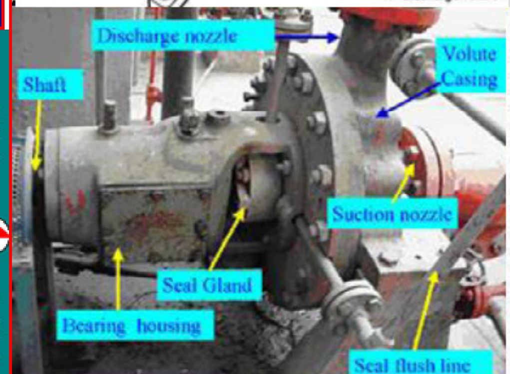
THW-10 Principles of Hydraulics &Centrifuge Pump Plant(2 programs)

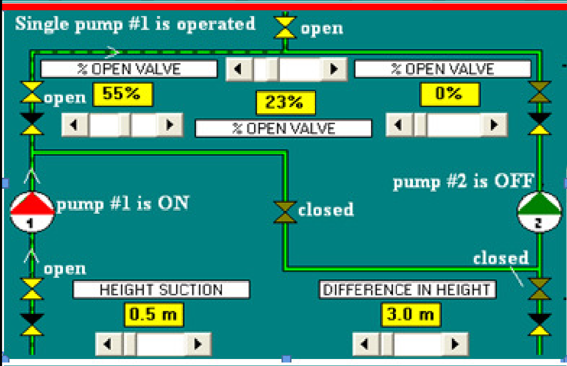


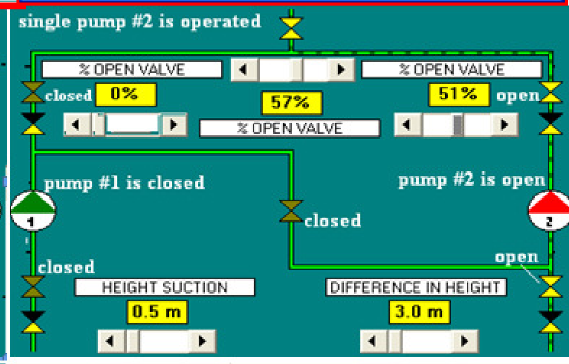


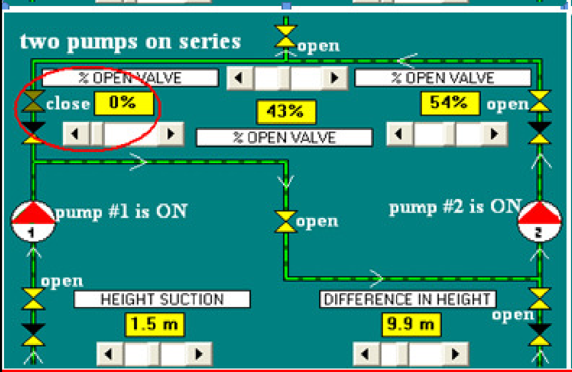


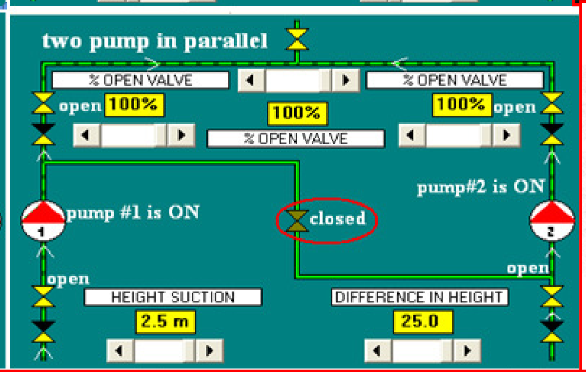


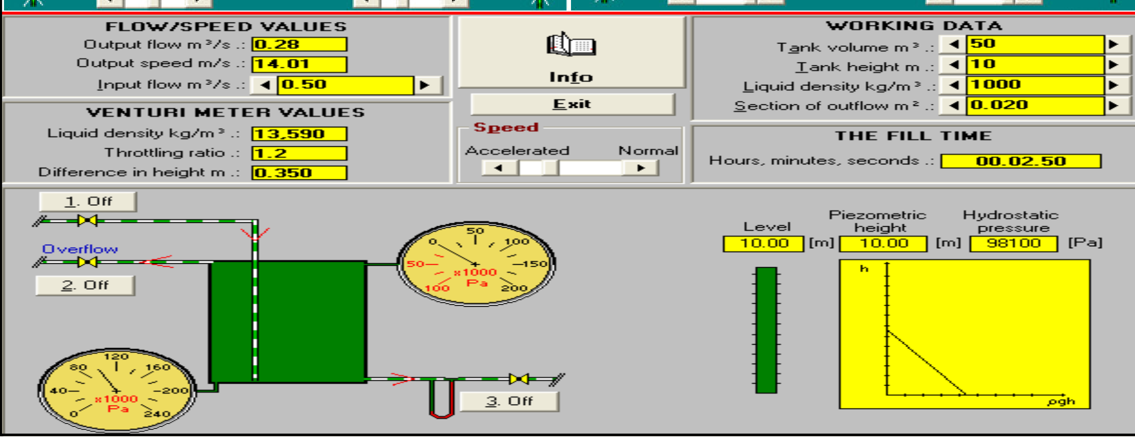


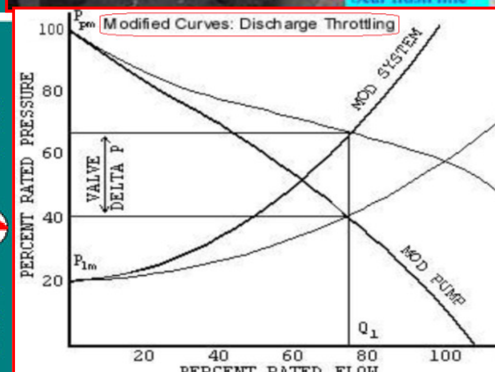


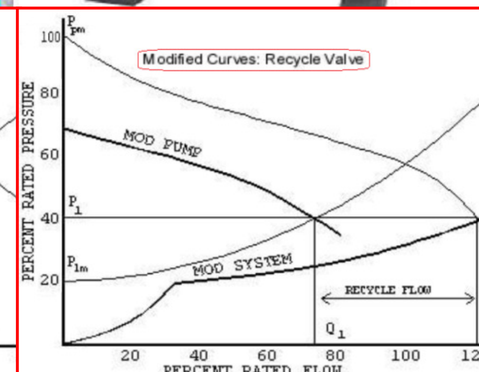


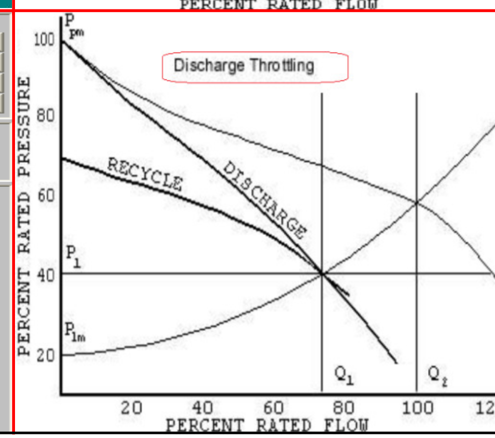


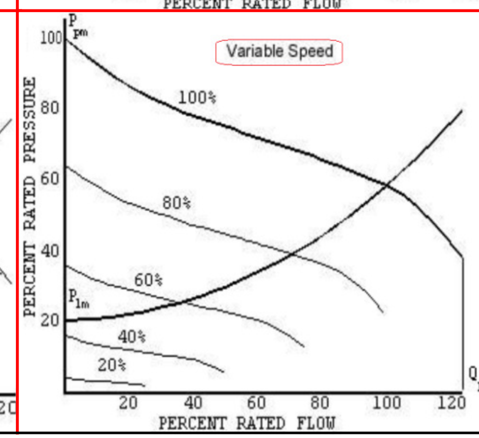


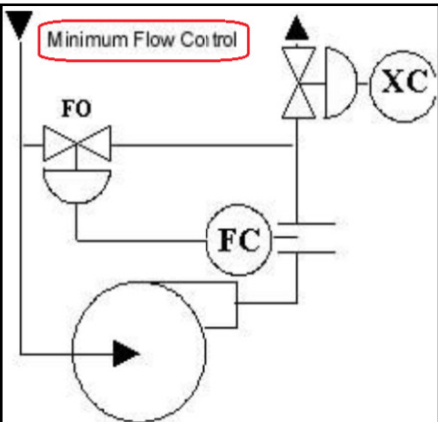


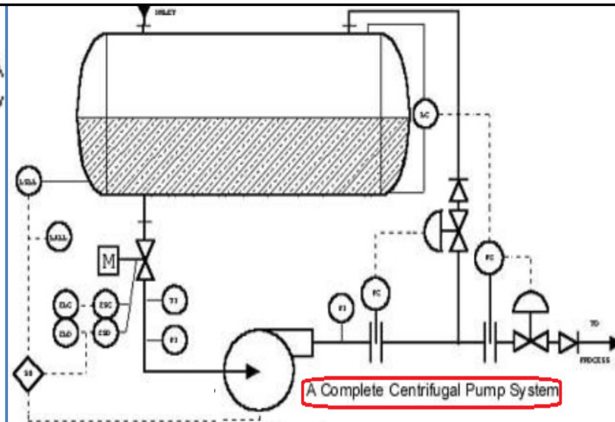


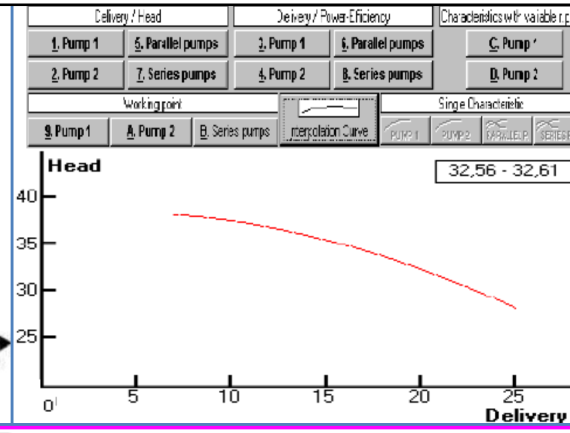


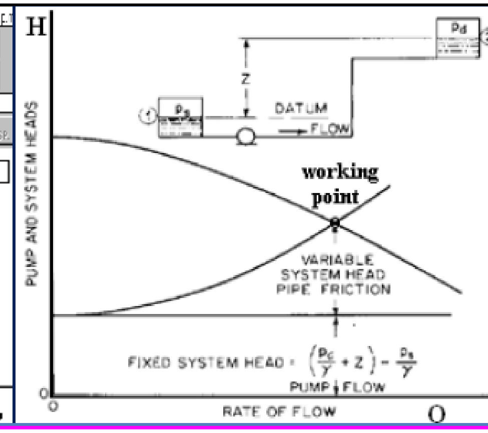












Diagnostic (C:\Program Files\PumpsHydraulic\PARALLEL.DTS

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Table 1 of 1

Pump 1	Voltage	Current	Delivery	Delivery Pressure	Suction Pressure	Revolution
	Volt	Ampere	m³/h	Bar	Bar	r.p.m.
1	140	11	7.1	3.4	-0.08	2842
2	140	11	8	3.37	-0.09	2842
3	140	12	8.9	3.34	-0.11	2842
4	140	12	9.8	3.3	-0.12	2842
5	140	13	10.8	3.25	-0.14	2842
6	140	13	11.2	3.23	-0.15	2842
7	140	13	12.1	3.18	-0.16	2842
8	140	13	13.1	3.13	-0.18	2842
9	140	14	14	3.07	-0.19	2842
10	140	14	14.9	3.01	-0.21	2842
11	140	14	15.4	2.98	-0.22	2842
12	140	15	15.8	2.95	-0.22	2842
13	140	15	16.3	2.91	-0.23	2842
14	140	15	16.7	2.88	-0.24	2842
15	140	15	17	2.86	-0.24	2842
16	140	15	17.4	2.83	-0.25	2842
17	140	16	18.1	2.77	-0.26	2842
18	140	16	18.6	2.73	-0.27	2842

Pump 2	Voltage	Current	Delivery	Delivery Pressure	Suction Pressure	Revolution
	Volt	Ampere	m³/h	Bar	Bar	r.p.m.
1	139	10	7.1	3.4	-0.08	2761
2	139	11	8	3.37	-0.09	2761
3	139	11	8.9	3.34	-0.11	2761
4	139	11	9.8	3.3	-0.12	2761
5	139	12	10.8	3.25	-0.14	2761
6	139	12	11.2	3.23	-0.15	2761
7	139	13	12.1	3.18	-0.16	2761
8	139	13	13.1	3.13	-0.18	2761
9	139	13	14	3.07	-0.19	2761
10	139	14	14.9	3.01	-0.21	2761
11	139	14	16.1	2.93	-0.23	2761
12	139	15	17	2.86	-0.24	2761
13	139	15	17.9	2.79	-0.26	2761
14	139	16	18.8	2.71	-0.27	2761
15	139	16	19.7	2.63	-0.29	2761
16	139	17	20.7	2.55	-0.31	2761
17	139	17	21.6	2.46	-0.32	2761
18	139	18	22.7	2.35	-0.34	2761

PUMP TOTAL HEAD

The total head of a pump is the difference between the energy level at the pump discharge (point 2) and that at the pump suction (point 1), as shown in Figures 7 and 8. Applying Bernoulli's equation (Eq. 1) at each point, the pump total head TH in feet (meters) becomes

$$TH = H_d - H_s = \left(\frac{V_d^2}{2g} + \frac{p_d}{\gamma_d} + Z_d\right) - \left(\frac{V_s^2}{2g} + \frac{p_s}{\gamma_s} + Z_s\right)$$

where $H = \frac{V^2}{2g} + \frac{p}{\gamma} + Z$

H = total head of system, (+) or (–) ft (m) gage or (+) ft (m) abs

P = total pressure of system, (+) or (–) lb/ft² (N/m²) gage or (+) lb/ft² (N/m²) abs

V = velocity, ft/s (m/s)

p = pressure, (+) or (–) lb/ft² (N/m²) gage or (+) lb/ft² (N/m²) abs

Z = elevation above (+) or below (–) datum plane, ft (m)

γ = specific weight (force) of liquid, lb/ft³ (N/m³)

g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

pump differential pressure

$$TH = \frac{P_\Delta}{\gamma}$$

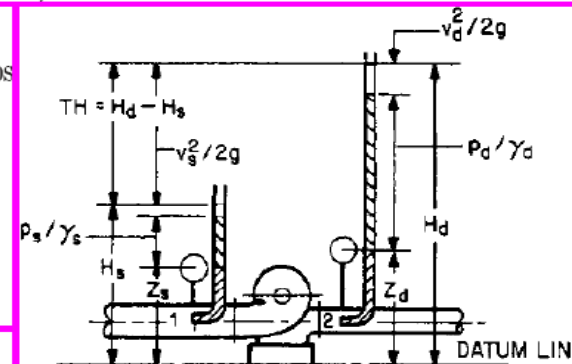

FIGURE 7 Centrifugal pump total head