NATIONAL EXAMINATIONS

December 2014

07-MEC-A6-1 FLUID MACHINERY

Three hours duration

Notes to Candidates

1. This is a **Closed Book** examination.

2. Exam consists of two Sections **Section A is Calculative (5 questions)** and **Section B is Descriptive (3 questions).**

3. **Do four (4) questions (including all parts of each question) from Section A (Calculative) and two (2) questions from Section B (Descriptive).**

4. **Six questions constitute a complete paper.** (Total 60 marks).

5. **All questions are of equal value.** (Each 10 marks).

6. If doubt exists as to the interpretation of any question, the candidate is urged to submit, with the answer paper, a clear statement of any assumptions made.

7. If any initial parts of a multi-part question cannot be solved the remaining parts may be worked by making appropriate assumptions for the first parts from the technical data given.

8. Candidates may use one of the approved **Casio** or **Sharp** calculators.

9. **Reference data** for particular questions are given in the Attachments on pages 8 to 11. **All pages from which data has been obtained or on which answers have been written are to be returned with the answer booklet to show any working. Candidate's names must be on these sheets.**

10. **Reference formulae and constants** are given on pages 12 to 16.

11. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams.

SECTION A CALCULATIVE QUESTIONS

QUESTION 1 PELTON WHEEL

An electrical generator is driven by a small single jet Pelton turbine designed to have the following technical parameters:

Specific speed	0.20
Effective head at the nozzle inlet	120 m
Nozzle velocity coefficient	0.985
Runner rotational speed	880 rev/min
Blade speed to jet speed ratio	0.47
Overall efficiency of the turbine	0.88 (based on shaft output)

(a) Calculate the shaft power output of the turbine.

(b) Calculate the volume flow rate.

(c) Calculate the jet flow area.

(d) Calculate the ratio of wheel diameter to jet diameter.

[10 marks]

QUESTION 2 COMPRESSOR AND TURBINE

Consider a simple single shaft gas turbine power plant having the following specifications:

Compressor pressure ratio	12.0	
Ambient pressure	100 kPa	
Ambient temperature	15°C	(288 K)
Turbine inlet temperature	1077°C	(1350 K)
Compressor efficiency	0.86	
Turbine efficiency	0.89	
Mechanical (shaft) efficiency	0.98	
Air/fuel ratio	50	

(a) Sketch a T-s diagram showing the complete thermodynamic cycle and identify by number all key points in the cycle.

(b) Determine and identify by number the temperatures at all key points in the cycle.

(c) Calculate the specific work output from the unit in kWs/kg of gas flow.

(d) Calculate the specific fuel consumption of the unit in kg/kWh.

[10 marks]

QUESTION 3 PUMP APPLICATION

Refer to the Examination Paper Attachments Page 8 **Pump Characteristics** and Page 9 **Cavitation Parameter.** Use the SI units on these charts.

A centrifugal pump is required to pump potable water from a water treatment plant to a reservoir for subsequent distribution. The required flow is 85 L/s and the head 30 m. The pump will be driven by an induction motor operating at 60 Hz and 400 V with a slip of 3% and electrical losses of 4%. To ensure satisfactory operation basic preliminary design information is required.

Assuming that the head loss in the pump inlet piping is 1.0 m and that the vapour pressure under the prevailing conditions is 2 kPa, determine the flowing:

(a) Pump specific speed

(b) Type of pump and sketch of impeller

(c) Diameter of impeller

(d) Critical cavitation parameter

(e) Desired net positive suction head

(f) Maximum elevation of pump relative to supply level

(g) Efficiency of pump

(h) Electric power consumption

[10 marks]

QUESTION 4 PUMP DESIGN

Refer to the Examination Paper Attachments Page 10 **Pump Velocity Diagram.**

A preliminary analysis for a particular centrifugal pump to deliver water under ambient conditions has yielded the following parameters:

Impeller inlet radius	100 mm
Impeller outlet radius	180 mm
Impeller inlet width	50 mm (in axial direction)
Impeller outlet width	30 mm (in axial direction)
Rotational speed	1720 rev/min
Volumetric flow rate	0.25 m^3/s
Delivery head	40 m

As a next step in the design of the pump the impeller blade angles need to be determined.

(a) Calculate the power required by the pump assuming ideal conditions (no friction).

(b) Calculate the mechanical torque required on the shaft.

(c) Calculate the blade inlet and outlet tangential velocities.

(d) Calculate the water inlet and outlet radial velocities neglecting blade thickness.

(e) Calculate the tangential water velocity at the outlet assuming pure radial flow at the inlet.

(f) Draw to scale the inlet and outlet velocity diagrams and determine the inlet and outlet blade angles.

Note: The velocity vector diagrams must be drawn sufficiently large to obtain a suitably accurate answer (10 mm = 5 m/s minimum). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.

[10 marks]

QUESTION 5 STEAM TURBINE BLADES

Refer to the Examination Paper Attachments Page 11 **Steam Turbine Velocity Diagram**.

One stage (set of fixed and moving blades) of a steam turbine receives steam at a mass flow rate of 30 kg/s. The fixed blade outlet angle θ is 20°. The moving blade is symmetrical such that the moving blade outlet angle γ is equal to the moving blade inlet angle Φ.

The initial absolute steam velocity V_{SI} leaving the fixed blades is 450 m/s. The blade velocity V_B is 250 m/s. Due to friction in the moving blades the outlet relative steam velocity V_{R2} from the moving blades is equal to 0.95 of the inlet relative steam velocity V_{R1} to these blades.

Draw to scale a velocity diagram for this turbine stage and determine the following:

(a) Final absolute steam velocity leaving the moving blades and its direction (angle δ)

(b) Power developed in this turbine stage due to change in momentum of the steam

(c) Blade efficiency related to the energy and flow of the initial steam jet

Note: The scale drawing should be to a large enough scale for accurate measurements (a scale of 10 m/s = 4 mm is suggested). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.

[10 marks]

SECTION B DESCRIPTIVE QUESTIONS

Note that each five mark part of each question requires a full page answer with complete explanations with sketches, if appropriate, to support the explanation.

QUESTION 6 BLADE DESIGN

PART I VANE CHARACTERISTICS

Describe with the aid of sketches the difference between forward curved and backward curved vanes in a centrifugal pump or fan. Explain how these configurations affect the velocity of the fluid leaving the impeller and hence the pressure rise in the diffuser. Sketch a typical head versus flow diagram (graph) for forward and backward curved vanes in a pump or fan.

(5 marks)

PART II NUMBER OF VANES

Centrifugal pump impellers are usually designed for an optimum number of vanes. Explain the effect on performance of a pump having both too many vanes or too few vanes. Explain the reasons for these effects.

(5 marks)

[10 marks]

QUESTION 7 TURBINE BLADE CHARACTERISTICS

PART I IMPULSE AND REACTION

Explain the difference between an impulse turbine and a reaction turbine. In particular refer to the changes in velocity in both the fixed and moving blades. Clarify how the forces developed are created and how they influence the transfer of energy from the fluid to the blades. If appropriate, show the difference between impulse and reaction in velocity diagrams for an axial flow gas or steam turbine.

(5 marks)

PART II OPTIMUM BLADE EFFICIENCY

With respect to a Pelton turbine show graphically in a sketch how and explain why the efficiency varies with turbine blade velocity (wheel rotational speed) when the water jet velocity remains constant. Consider the whole range of possibilities from a blade velocity of zero to a blade velocity equal to that of the jet. If appropriate, draw velocity diagrams to illustrate the explanation.

(5 marks)

[10 marks]

QUESTION 8 COMPRESSOR AND PUMP CHARACTERISTICS

PART I STALLING IN COMPRESSORS

Describe stalling in an axial flow compressor as used in a typical gas turbine. Clarify under what conditions stalling can occur. Explain how the phenomenon of stalling affects the design of the compressor especially with regard to the number of stages required.

(5 marks)

PART II CENTRIFUGAL PUMPS

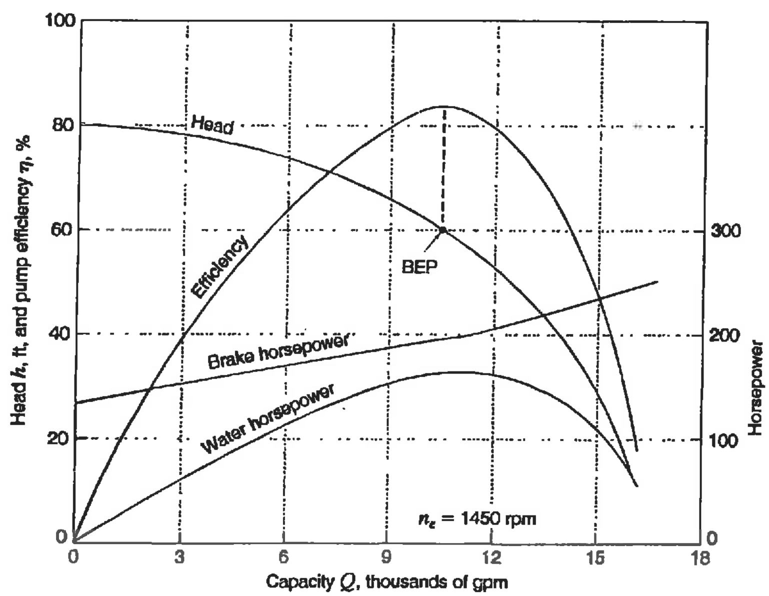
With reference to the figure below explain the following:

(a) Why the hydraulic efficiency (water horsepower) rises from zero to a peak and then declines towards zero

(b) Why the difference between the hydraulic power (water horsepower) and the mechanical power (brake horsepower) decreases to a low value and then increases to a value greater than the initial value.

(5 marks)

[10 marks]



Characteristic curves for a typical mixed-flow centrifugal pump.

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EXAMINATION PAPER ATTACHMENTS

QUESTION 3 PUMP CHARACTERISTICS

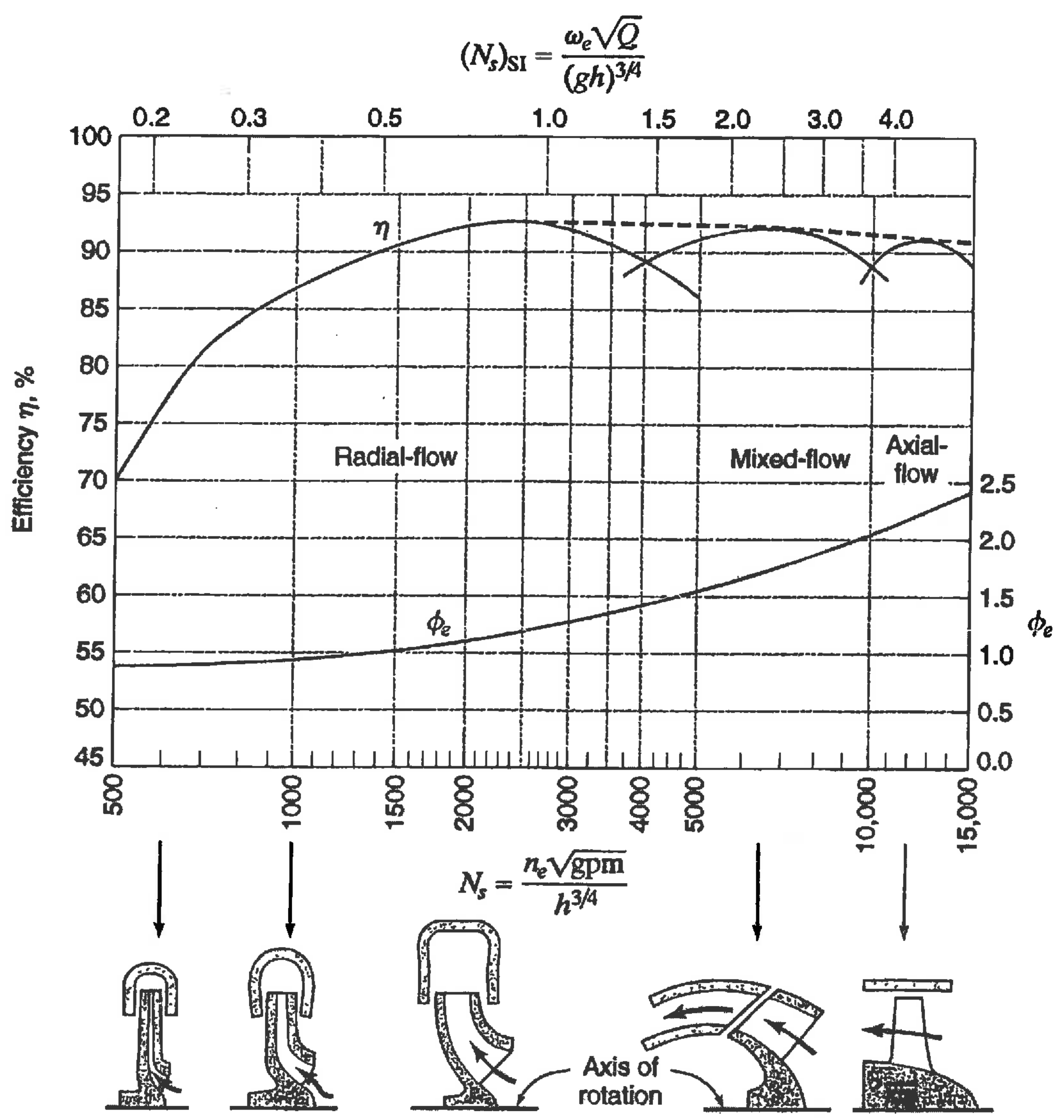


Figure 15.11
Optimum efficiency and typical values of ϕ_e for water pumps as a function of specific speed.

NAME ....

QUESTION 3 CAVITATION PARAMETER

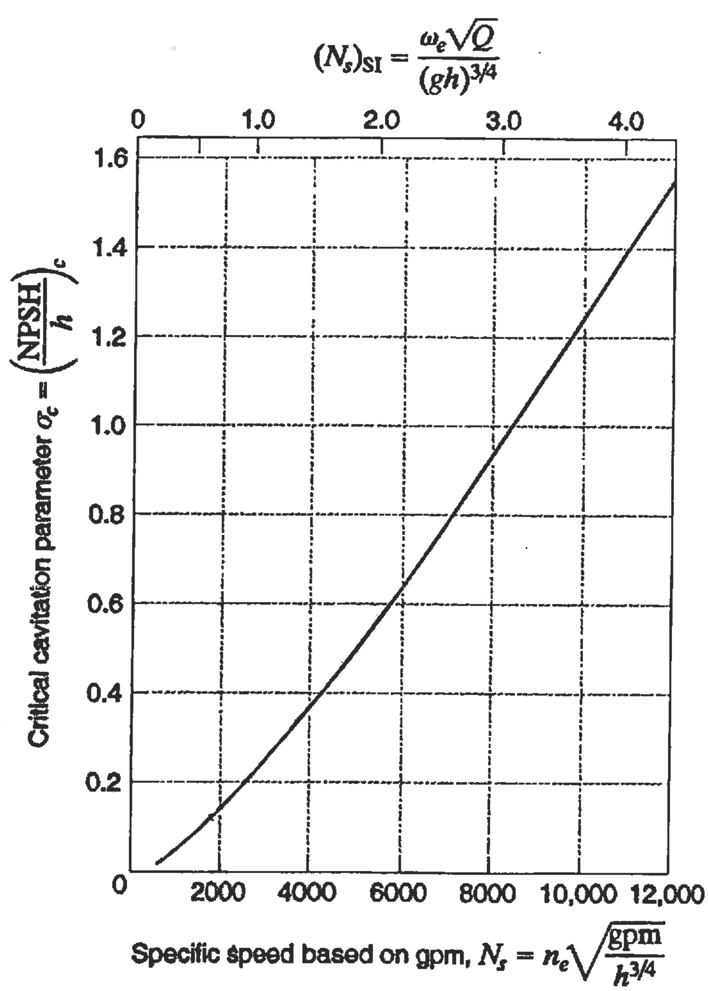
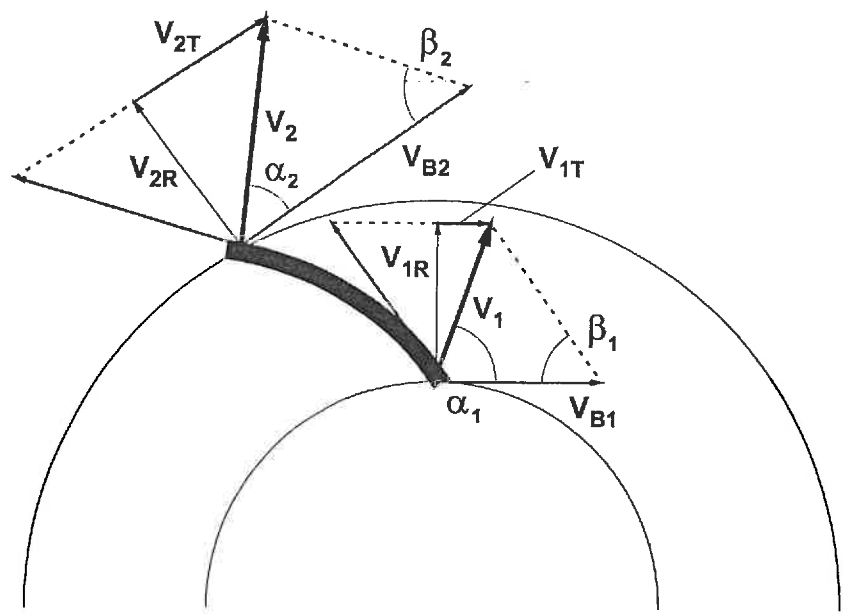


Figure 15.12
Approximate values of critical cavitation parameter σ_c as a function of specific speed.

QUESTION 4 PUMP VELOCITY DIAGRAM

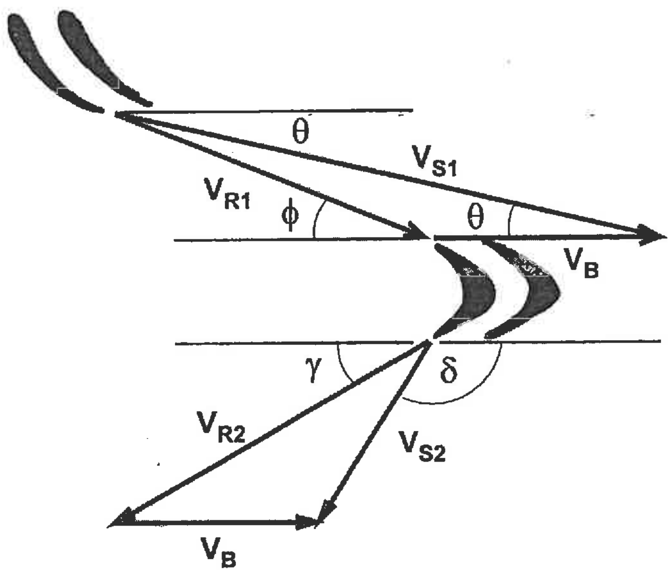
Nomenclature for velocity vectors and angles



V_1	Absolute water velocity at inlet
V_{B1}	Blade velocity at inlet
V_{1R}	Radial water velocity at inlet
V_{1T}	Tangential water velocity at inlet
V_2	Absolute water velocity at outlet
V_{B1}	Blade velocity at outlet
V_{2R}	Radial water velocity at outlet
V_{2T}	Tangential water velocity at outlet

QUESTION 5 STEAM TURBINE VELOCITY DIAGRAM

Nomenclature for velocity vectors and angles



V_{S1}	Absolute steam velocity entering moving blades
V_{R1}	Relative steam velocity entering moving blades
V_B	Moving blade velocity
V_{R2}	Relative steam velocity leaving moving blades
V_{S2}	Absolute steam velocity leaving moving blades

EXAMINATION REFERENCE MATERIAL

NOMENCLATURE FOR REFERENCE EQUATIONS (SI UNITS)

A	Flow area, Surface area	m^2
c_p	Specific heat at constant pressure	J/kg°C
c_v	Specific heat at constant volume	J/kg°C
b	Width	m
C	Velocity	m/s
D	Diameter	m
E	Energy	J
F	Force	N
g	Gravitational acceleration	m/s^2
h	Specific enthalpy	J/kg
h	System head	m
h_L	Head loss	m
H	Pump or turbine head	m
k	Ratio of specific heats	
L	Length	m
m	Mass	kg
M	Mass flow rate	kg/s
N	Rotational speed	rev/s
N_S	Specific Speed	
p	Pressure	Pa (N/m^2)
P	Power	W (J/s)
q	Heat transferred	J/kg
Q	Heat	J
Q	Flow rate	m^3/s
r	Radius	m
R	Specific gas constant	J/kg K
s	Entropy	J/kg K
T	Temperature	K
u	Specific internal energy	J/kg
U	Internal Energy	J
U	Velocity	m/s
v	Specific volume	m^3/kg
V	Velocity	m/s
w	Specific work	J/kg
W	Work	J
W	Velocity	m/s
x	Length	m
z	Elevation	m

α	Pump blade angle	°
α	Compressor blade angle	°
β	Pump blade angle	°
β	Compressor blade angle	°
γ	Turbine blade angle	°
φ	Turbine blade angle	°
δ	Turbine blade angle	°
η	Efficiency	
θ	Nozzle angle	°
μ	Dynamic viscosity	Ns/m^2
v	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
σ_C	Critical cavitation parameter	
τ	Thrust	N
τ	Torque	Nm
φ	Peripheral velocity factor	
ω	Rotational speed	rad/s
Ω	Heat transfer rate	J/s

GENERAL CONSTANTS

Use unless otherwise specified

Acceleration due to gravity:	$g = 9.81\ m/s^2$	
Atmospheric pressure:	$p_{atm} = 100$ kPa	
Water vapour pressure:	$p_{vapour} = 2.34$ kPa	(at 20°C)
Density of water:	$\rho_{water} = 1000\ kg/m^3$	
Density of air:	$\rho_{air} = 1.21\ kg/m^3$	(at 15°C)
Density of air:	$\rho_{air} = 1.19\ kg/m^3$	(at 20°C)
Specific heat of air:	$c_p = 1.005$ kJ/kg°C	
Specific heat of air:	$c_v = 0.718$ kJ/kg°C	
Specific heat of water:	$c_p = 4.19$ kJ/kg°C	

GENERAL REFERENCE EQUATIONS

Basic Thermodynamics

First Law:	$dE = \delta Q - \delta W$
Enthalpy:	$h = u + pv$
Continuity:	$\rho VA = \text{constant}$
Potential Energy:	$E_{PE} = mgz$
Kinetic Energy:	$E_{KE} = V^2/2$
Internal Energy:	$E_{IN} = U$
Flow Work:	$w = \Delta(pv)$
Energy Equation:	$zg + V^2/2 + u + pv + \Delta w + \Delta q = \text{constant}$

Ideal Gas Relationships

Gas Law:	$pv = RT$
Specific Heat at Constant Pressure:	$c_p = \Delta h/\Delta T$
Specific Heat at Constant Volume:	$c_v = \Delta u/\Delta T$
Specific Gas Constant	$R = c_p - c_v$
Ratio of Specific Heats	$k = c_p / c_v$
Isentropic Relations:	$p_1/p_2 = (v_2/v_1)^k = (T_1/T_2)^{k/(k-1)}$

FLUID MACHINERY REFERENCE EQUATIONS

Fluid Mechanics

Pressure	$p = \rho gh$
Continuity Equation:	$\rho_1 V_1 A_1 = \rho_2 V_2 A_2 = M$
Bernoulli's Equation:	$p_1/\rho g + z_1 + V_1^2/2g = p_2/\rho g + z_2 + V_2^2/2g$
Momentum Equation:	$F = p_1A_1 - p_2A_2 - \rho VA(V_2 - V_1)$ (one dimensional)

Steam Turbines

Nozzle Equation:	$h_1 - h_2 = (V_2^2 - V_1^2) / 2$
Work:	$w = [(V_1{}^2{}_{absolute} - V_2{}^2{}_{absolute}) + (V_2{}^2{}_{relative} - V_1{}^2{}_{relative})] / 2$
Work:	$w = (V_{S1}\cos\theta - V_{S2}\cos\delta)\, V_{blade}$
Power:	$P = wM$

Gas Turbines

State Equation:	$pv = RT$	
Isentropic Equation:	$(T_2/T_1) = (p_2/p_1)^{(k-1)/k}$	
Enthalpy Change:	$h_1 - h_2 = c_p(T_1 - T_2)$	(ideal gas)
Nozzle Equation:	$h_1 - h_2 = (V_2^2 - V_1^2) / 2$	
Work:	$w = (C_1 \sin\alpha_1 + C_2 \sin\alpha_2)\, U$	
Work:	$w = [(C_1^2 - C_2^2) + (W_2^2 - W_1^2)] / 2$	
Power:	$P = wM$	

Compressors

Work	$W = U(C_{Y2} - C_{Y1})$
Rotor Enthalpy Change	$h_1 + ½W_1^2 = h_2 + ½W_2^2$
Stator Enthalpy Change	$h_2 + ½C_2^2 = h_3 + ½C_3^2$
Isentropic Equation:	$(T_3/T_1) = (p_3/p_1)^{(k-1)/k}$

Jet Propulsion

Thrust:	$\tau = M(V_{jet} - V_{aircraft})$
Thrust Power:	$\tau V_{aircraft} = M(V_{jet} - V_{aircraft})V_{aircraft}$
Jet Power:	$P = M(V_{jet}^2 - V_{aircraft}^2) / 2$
Propulsion Efficiency:	$\eta_p = 2V_{aircraft}/(V_{jet} + V_{aircraft})$

Wind Turbine

Maximum Ideal Power:	$P_{max} = 8\,\rho\, AV_1^3 / 27$

Energy Equation

Pump and Turbine	$p_1/\rho g + z_1 + V_1^2/2g + w_{in}/g = p_2/\rho g + z_2 + V_2^2/2g + w_{out}/g$
With Friction:	$p_1/\rho g + z_1 + V_1^2/2g = p_2/\rho g + z_2 + V_2^2/2g + h_L$

Hydraulic Machines

Similarity Equations:	$Q_M/Q_P = (\omega_M/\omega_P)\,(D_M/D_P)^3$
	$H_M/H_P = (\omega_M/\omega_P)^2\,(D_M/D_P)^2$
	$P_M/P_P = (\rho_M/\rho_P)\,(\omega_M/\omega_P)^3\,(D_M/D_P)^5$
Pump Specific Speed:	$N_S = \omega\, Q^{1/2} / (gH)^{3/4}$
Turbine Specific Speed:	$N_S = \omega\, P^{1/2} / [\rho^{1/2}\,(gH)^{5/4}]$

Critical Cavitation Parameter: $\sigma = [\{(p_{atmosphere} - p_{vapour}) / \rho g \} - \Delta z] / H$
Moody Efficiency Relationship: $\eta_P = 1 - (1 - \eta_M) (D_M/D_P)^{1/4} (H_M/H_P)^{1/10}$
Approximate Moody Efficiency: $(1 - \eta_M)/(1 - \eta_P) \approx (D_P/D_M)^{1/5}$
Power: $P = \rho g Q H$

Pumps

Hydraulic Torque: $\tau = \rho Q (r_2 V_{2T} - r_1 V_{1T})$
Hydraulic Torque: $\tau = \rho Q (r_2 V_2 \cos\alpha_2 - r_1 V_1 \cos\alpha_1)$
Power: $P = 2\pi N \tau$
Net Positive Suction Head: $NPSH = [(p_{atmosphere} - p_{vapour}) / \rho g] - \Delta z - h_L$
Peripheral Velocity Factor: $\varphi = V_{B2} / (2gh)^{1/2}$
Critical Cavitation Parameter: $\sigma_C = NPSH / H$
Approximate Moody Efficiency: $(1 - \eta_P)/(1 - \eta_M) \approx (D_M/D_P)^{1/5}$

Steam Turbines

Force on Blades: $F = M (V_{S1}\cos\theta - V_{S2}\cos\delta)$
Power to Blades: $P = M (V_{S1}\cos\theta - V_{S2}\cos\delta) V_B$
Power to Blades: $P = M [(V_{S1}^2 - V_{S2}^2) + (V_{R2}^2 - V_{R1}^2)] / 2$