NATIONAL EXAMINATIONS

December 2018

16-MEC-B6 FLUID MACHINERY

Three hours duration

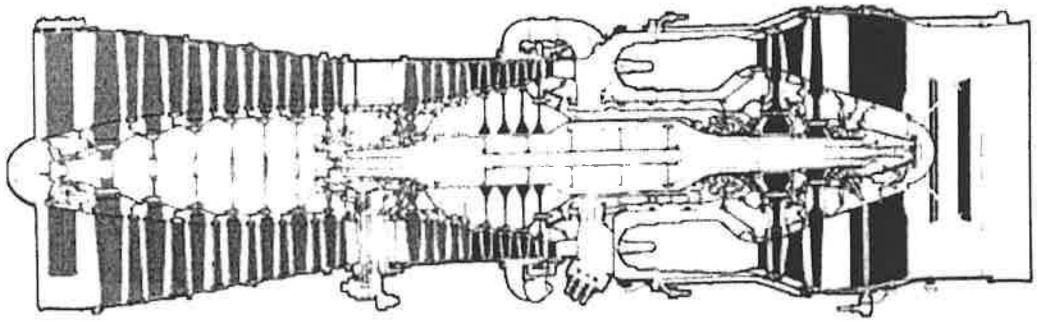
Notes to Candidates

1. This is a **Closed Book** examination.
2. Examination 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. Read the entire question before commencing the calculations and take note of hints or recommendations given.
8. 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.
9. Candidates may use one of the approved **Casio** or **Sharp** calculators.
10. **Reference data** for particular questions are given in the Attachments on pages 11 to 17. **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.**
11. **Reference formulae and constants** are given on pages 18 to 22.
12. **Drawing Instruments** (scale ruler, protractor and sharp pencil) are required for vector diagrams. While calculation of velocities by trigonometric ratios with reference to a sketch is acceptable it is longer and more time consuming.

SECTION A CALCULATIVE QUESTIONS

Show all steps in the calculations and state the units for all intermediate and final answers.

QUESTION 1 COMPRESSOR STAGE PERFORMANCE



Rolls-Royce/SNECMA Olympus 593 Mark 610 Afterburning Turbojet

The diagram above (for illustration only) shows a cross section of the engine of the Concorde supersonic aircraft (without the variable geometry intake and variable area nozzle). Relevant specifications and assumed operational conditions are as follows:

LP compressor stages	7	
LP compressor pressure ratio	3	(for whole compressor)
HP compressor stages	7	
HP compressor pressure ratio	5	(for whole compressor)
LP and HP isentropic efficiencies	0.90	
Mach number at compressor inlet	0.49	
Effective area at compressor inlet	0.80 m	
Air inlet pressure	80 kPa	
Air inlet temperature	80°C	

(a) Sketch the compression process on a T-s diagram and label all key points that will be required and referred to in the calculations. (1)

(b) Calculate the air temperature at the LP compressor exit and at the HP compressor exit. (4)

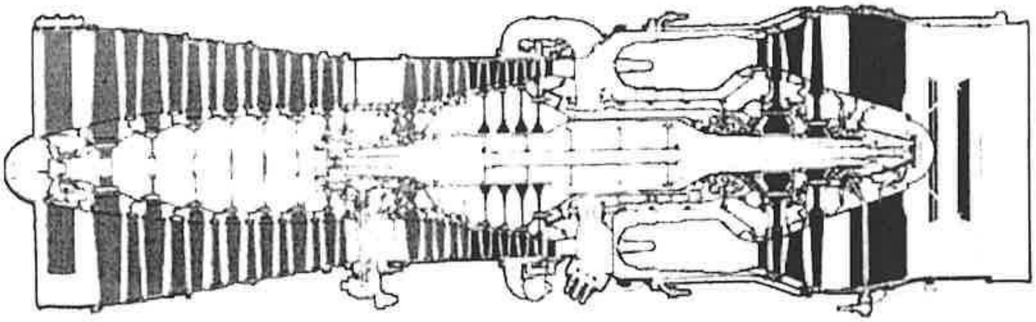
(c) Some air is drawn off after the fifth HP compressor stage for cooling of the turbine blades. Calculate the temperature of this air. (2)

(d) Calculate the air velocity at the compressor inlet. (1)

(e) Calculate the mass flow through the compressor (neglecting that extracted for cooling) and hence the power required to drive the compressor. (2)

[10 marks]

QUESTION 2 COMPRESSOR BLADE ANGLES



Rolls-Royce/SNECMA Olympus 593 Mark 610 Afterburning Turbojet

The diagram above (for illustrative purposes only) shows a cross section of the engine of the Concorde supersonic aircraft (without the variable geometry intake and variable area nozzle). Basic specifications and assumed operational conditions are as follows where HP and LP designate high pressure and low pressure respectively:

LP compressor stages	7	
LP compressor pressure ratio	3	(for whole compressor)
LP compressor speed	6500 rev/min	
HP compressor stages	7	
HP compressor pressure ratio	5	(for whole compressor)
HP compressor speed	8500 rev/min	
HP and LP isentropic efficiency	0.9	
LP first stage moving blade tip diameter	1.12 m	
LP first stage moving blade hub diameter	0.38 m	
Inlet air temperature to first stage	80°C	
Inlet air velocity to first stage	184 m/s	
Air mass flow rate	116 kg/s	

Consider the first stage moving blade conditions and assume no prewhirl at the inlet (air enters in a purely axial direction to give maximum flow).

Refer to the Examination Paper Attachments Page 11 **Compressor Velocity Diagram** and Page 12 **Compressor Inlet Blades** for guidance and reference.

This question is continued on the next page

Question 2 Continued

(a) Calculate the blade velocity at the tip and hub (root) of the first stage moving blade. (1)

(b) Calculate the temperature at the exit of the first stage. (2)

(c) Calculate the work done (J/kg) by the first stage and hence determine the values of the components (in the direction of blade motion) of the absolute air velocities (C_{Y1} and C_{Y2}) for both the blade tip and the blade root. (2)

(d) Draw to scale the velocity (vector) diagrams for the first stage moving blades at both the tip and hub and determine the inlet and outlet blade angles for these blades. *A scale of 1mm = 5 m/s is suggested.* (4)

(e) Refer to the photograph of the first stage inlet blades on Page 12 and assess whether the blade angles determined in (d) above are reasonably correct. Justify your answer. (1)

[10 marks]

QUESTION 3 HYDRO TURBINES

PART I KAPLAN TURBINE EFFICIENCY

Refer to the Examination Paper Attachments Page 13 **Kaplan Turbine** for illustrative purposes only.

Hydro turbines of the Kaplan type are installed at Mactaquac on the Saint John River. In order to determine the efficiency of the Mactaquac turbines the following hypothetical measurements are considered:

Turbine-generator speed	112.5 rev/min
Generator electrical output	110 MW
Water flow rate	354 m^3/s
Inlet pipe diameter (not in picture)	6.4 m
Outlet pipe diameter (bottom of picture)	7.0 m
Inlet water pressure	226 kPa gauge
Outlet water pressure	– 4.5 m H_2O

The elevation of the outlet pressure measuring point is 5.0 m below that of the inlet pressure measuring point.

Determine the following:

(a) Water flow velocities at inlet and outlet of the turbine. (1)

(b) Hydraulic power produced by the water (input to turbine-generator). (3)

(c) Electrical power output and efficiency of turbine-generator. (1)

(5 marks)

This question is continued on the next page ....

Question 3 Continued

PART II HYDRO TURBINE PLANT

Refer to the Examination Paper Attachments Page 14 **Hydro Power Plant**.

The figure shows a cross section of a small hydro power plant utilizing a Kaplan turbine in a low head application. The maximum power output of the turbine generator is 4 MW. Select an appropriate value for the turbine efficiency but assume that there is negligible hydraulic friction in the rest of the system:

(a) Determine the flow rate of water when operating at maximum power between normal low water level in the reservoir and full load water level in the tail race. (1)

(b) Determine the velocity of the water in the penstock when operating under the conditions in (a) above. Note that the penstock is circular in cross section and its diameter must be estimated from the information given on the drawing. (1)

(c) Should normal high water level in the reservoir be reached while that in the tailrace remains unchanged, state with reasons whether the velocity in the penstock will be greater or less than that given in (b) to achieve full power. (2)

(d) Determine the velocity of the water in the penstock when operating under the conditions in (c) above. (1)

(5 marks)

[10 marks]

QUESTION 4 FRANCIS HYDRO TURBINE

Refer to the Examination Paper Attachments Page 15 **Hydro Turbine Parameters** for nomenclature of velocities and angles.

As part of a feasibility study for a small new hydro plant some basic design information is required. The head available and power required indicate that a radial inward flow hydro turbine of the Francis type with a specific speed and efficiency as given below would be suitable. Basic parameters are as follows:

Head available $H = 100$ m
Power output $P = 4$ MW (shaft power)
Specific speed $N_s = 0.80$
Hydraulic efficiency $\eta = 0.90$

Assumed parameters for the preliminary design of a typical Francis turbine are as follows:

Peripheral velocity $u_1 = 0.7\ V_{JET}$ (where V_{JET} is jet velocity)
Outlet vane diameter $d_1 = \frac{3}{4}\,d_2$ (where d_2 is inlet vane diameter)
Guide vane angle $\alpha_1 = 20°$
Inlet vane angle $\beta_1 = 90°$ (radial flow with respect to runner)
Exit flow angle $\alpha_2 = 90°$ (radial flow with respect to draft tube)
Exit radial velocity $V_2 = 1.2$ Inlet radial velocity v_1

Assuming two dimensional flow conditions (no axial component) everywhere, calculate the following:

(a) Rotational speed of the turbine. (1)

(b) Maximum jet velocity (spouting velocity) due to available head. (1)

(c) Runner tip speed (vane velocity at inlet) and runner outlet speed (vane velocity at outlet). (1)

(d) Runner inlet and outlet diameters. (1)

(e) Volume flow rate. (1)

(f) Vane (blade) height at inlet. (2)

(g) Outlet vane angle and vane (blade) height at outlet. (3)

[10 marks]

QUESTION 5 CURTIS TYPE IMPULSE TURBINE

Refer to the Examination Paper Attachments Page 16 **Steam Turbine Velocity Diagram** (one stage only) for nomenclature of velocities and angles.

Steam exits the nozzles and enters the first stage moving blades of a velocity compounded two stage (Curtis) impulse turbine at 1411 m/s. The nozzle angle is 20° and the fixed blade exit angle is the same as its inlet angle, that is, the fixed blades are symmetrical. The moving blades of both stages are also symmetrical but with different angles. Assume zero fluid friction in nozzles and blades.

(a) Determine a blade velocity to give optimum work (minimum exit kinetic energy). (1)

(b) Draw to scale the velocity diagrams for the two stages. (3)

(c) Determine all the actual and relative steam velocities and blade angles and show them on the diagrams. (2)

(d) Calculate the work done by each stage, in kJ/kg of steam. (2)

(e) Calculate the total power output for a steam flow of 100 kg/s. (1)

(f) Calculate the blade efficiency. (1)

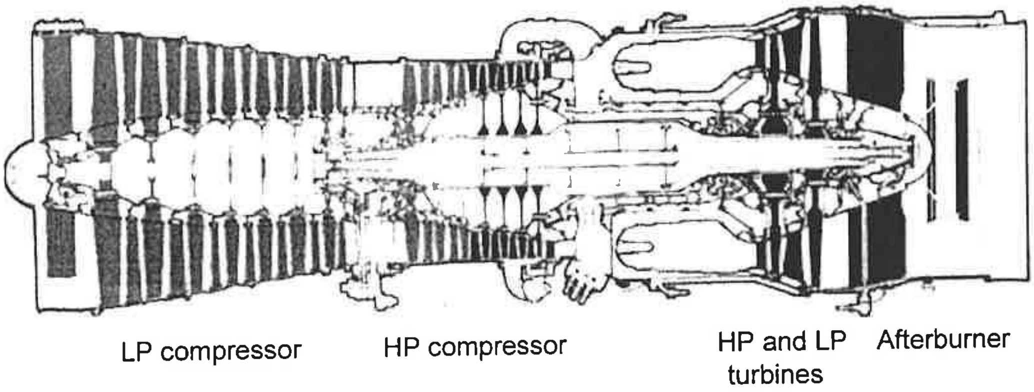
Note: The scale drawing should be to a large enough scale for accurate measurements (a scale of 1 mm = 10 m/s is suggested). While calculation of velocities by trigonometric ratios is acceptable it is longer and more time consuming.

[10 marks]

SECTION B DESCRIPTIVE QUESTIONS

Note that a ten mark question requires an answer of approximately two full pages of complete explanations with sketches, if appropriate, to support the explanation.

QUESTION 6 TURBINE AND COMPRESSOR STAGE LIMITATIONS



Rolls-Royce/SNECMA Olympus 593 Mark 610 Afterburning Turbojet

The diagram of the gas turbine turbojet engine above shows a high pressure (HP) turbine with just one stage driving a high pressure compressor with seven stages and similarly a low pressure (LP) turbine also with just one stage driving a low pressure compressor with seven stages. The two turbines drive the respective compressors through separate shafts at different speeds. *The diagram is for orientation only and no specific information need be obtained from it.*

(a) Explain why the compressors require so many stages compared with the turbines for an equivalent transfer of energy. Clarify the limiting factors in compressor design and operation. (4)

(b) Explain how with only one stage each the turbines can produce sufficient power to drive the compressors. (4)

(c) Explain why the low pressure and high pressure turbine-compressor shafts run at different speeds:

LP turbine-compressor shaft speed = 6500 rev/min
HP turbine-compressor shaft speed = 8500 rev/min

(2)

[10 marks]

QUESTION 7 FAN BLADE SHAPE

Refer to the Examination Paper Attachments Page 17 **Fan Characteristics**.

This page is to be returned with the answer booklet. Write your name on it.

Radial flow centrifugal fans may have blades that are radial or curved forwards or backwards. Explain what effect the shape of the blades has on the performance of the fan.

(a) Show graphically in sketches on Page 17 how the velocity diagram at the outlet is different for each type of fan and how the head versus flow curve is different for the three fans.

(b) Explain the advantages of forward or backward curved blades with respect to the other. Hence clarify with reasons which configuration is more common.

[10 marks]

QUESTION 8 HYDRO TURBINE OPERATION

PART I HYDRAULIC CAVITATION

Describe what determines the formation and collapse of vapour bubbles in a liquid. With reference to the mode of collapse, explain the phenomenon of cavitation and the mechanism of damage to the surface of hydraulic machine components. Clarify with reasons which parts of a turbine such as a Francis type could be damaged due to cavitation.

(5 marks)

PART II LOAD CHANGES

(a) Explain how water flow and hence power output is changed in a typical hydro turbine of the Francis type. Describe what happens in the penstock (inlet pipe) of such a turbine when the water flow is reduced to zero flow quickly (but not instantaneously). Explain the effects of varying the rates of flow reduction.

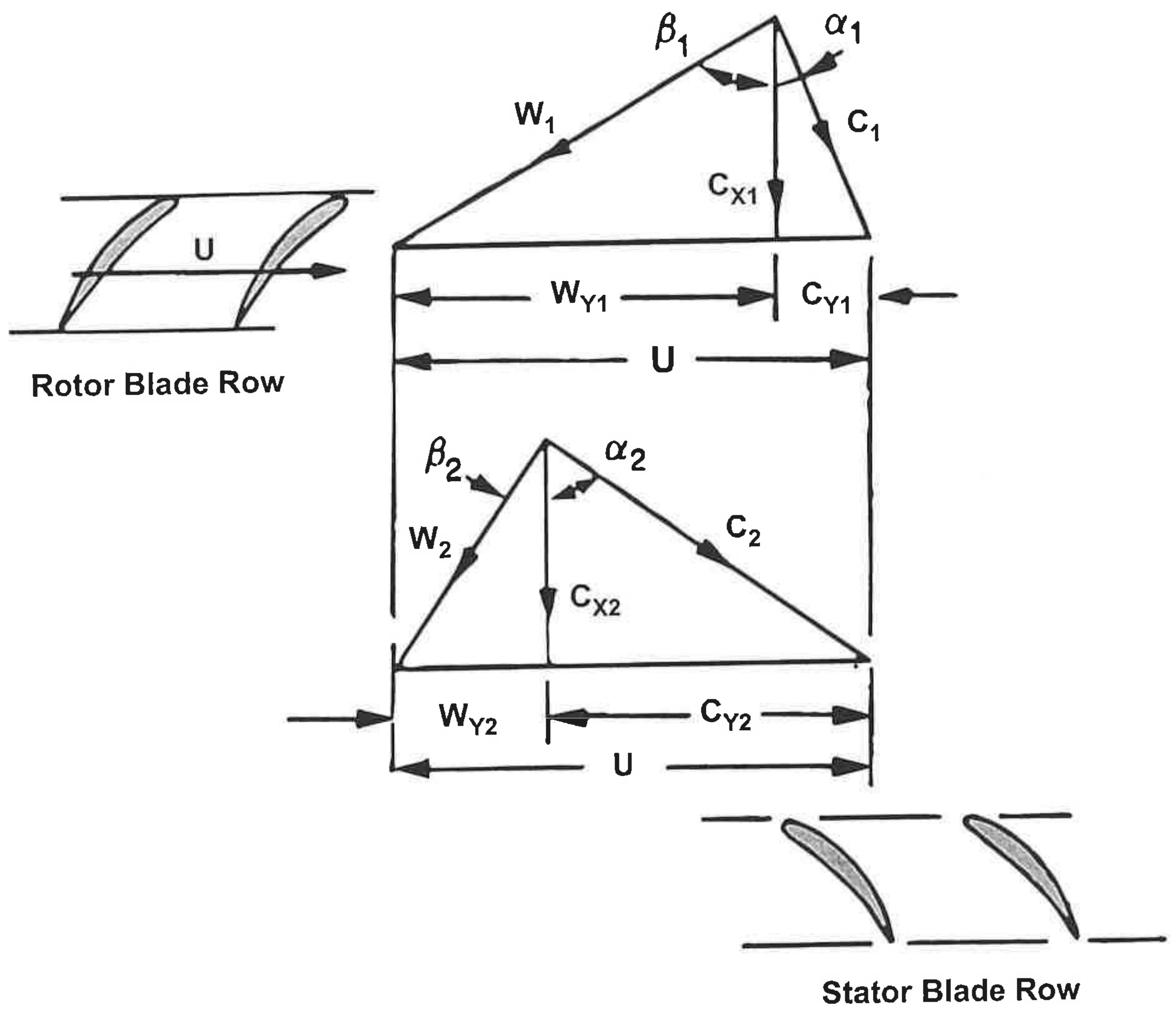
(b) Explain how the flow in the draught tube (outlet pipe) of a Francis type turbine is affected by low load (low flow) operation of a fixed speed (power generating) machine. Support the explanation with a sketch of the outlet velocity diagram showing the change from full load to low load conditions

(5 marks)

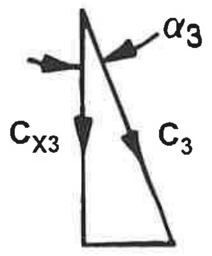
[10 marks]

EXAMINATION PAPER ATTACHMENTS

QUESTION 2 COMPRESSOR VELOCITY DIAGRAM

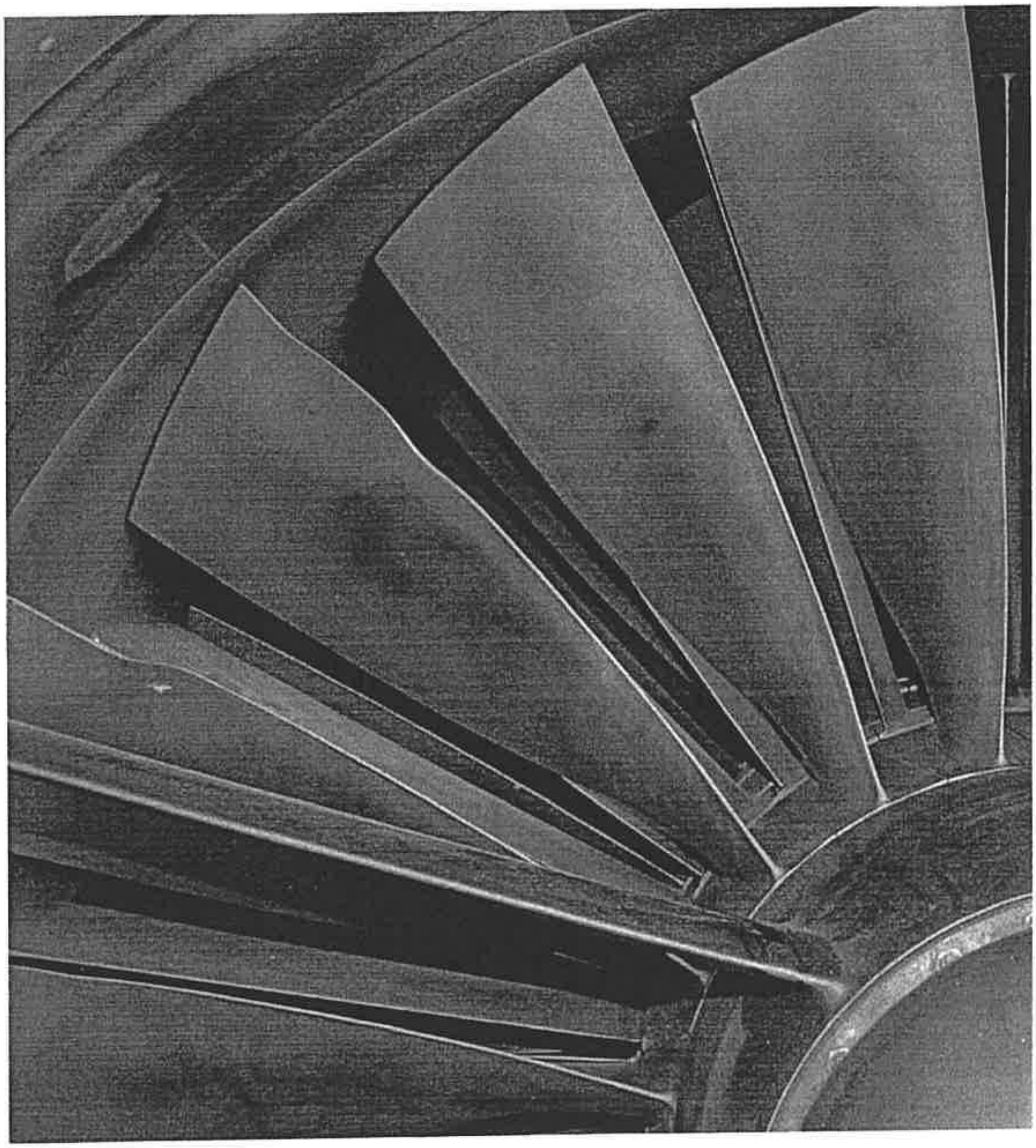


U	Blade velocity
C_1	Rotor blade absolute inlet velocity
W_1	Rotor blade relative inlet velocity
C_2	Rotor blade absolute outlet velocity
W_2	Rotor blade relative outlet velocity
C_3	Stator blade absolute outlet velocity



EXAMINATION PAPER ATTACHMENTS

QUESTION 2 COMPRESSOR INLET BLADES



Rolls-Royce Olympus 593-610 first stage low pressure compressor blades

EXAMINATION PAPER ATTACHMENTS

QUESTION 3 PART I KAPLAN TURBINE

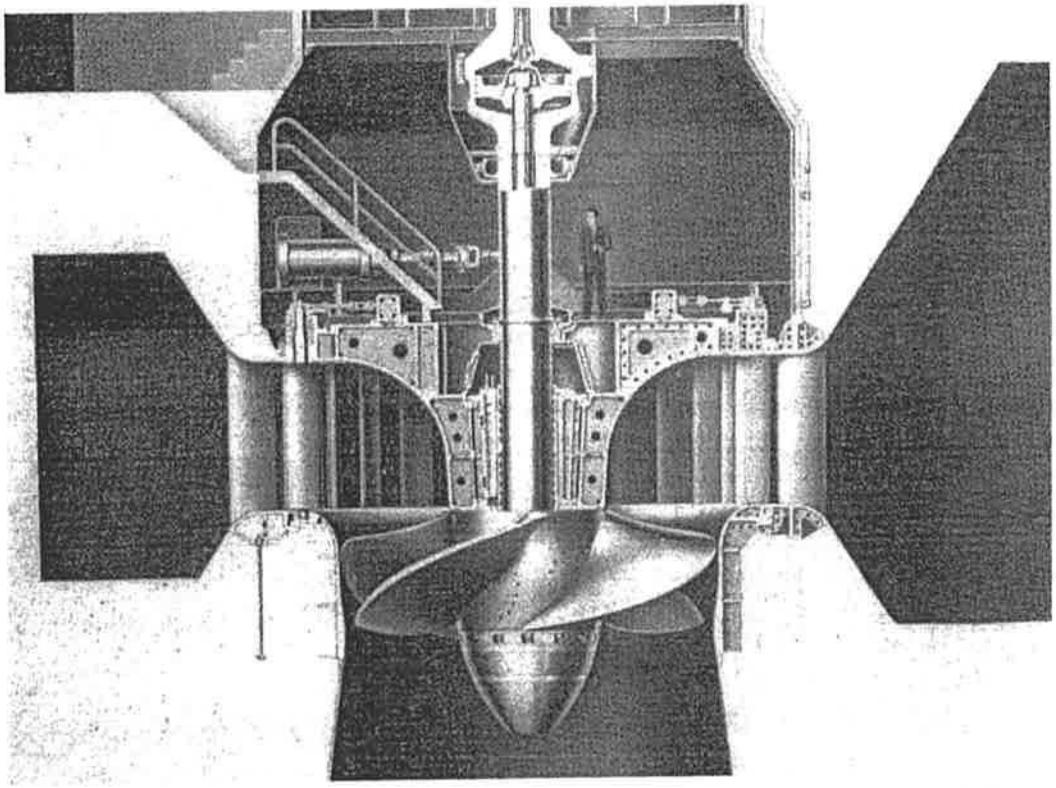
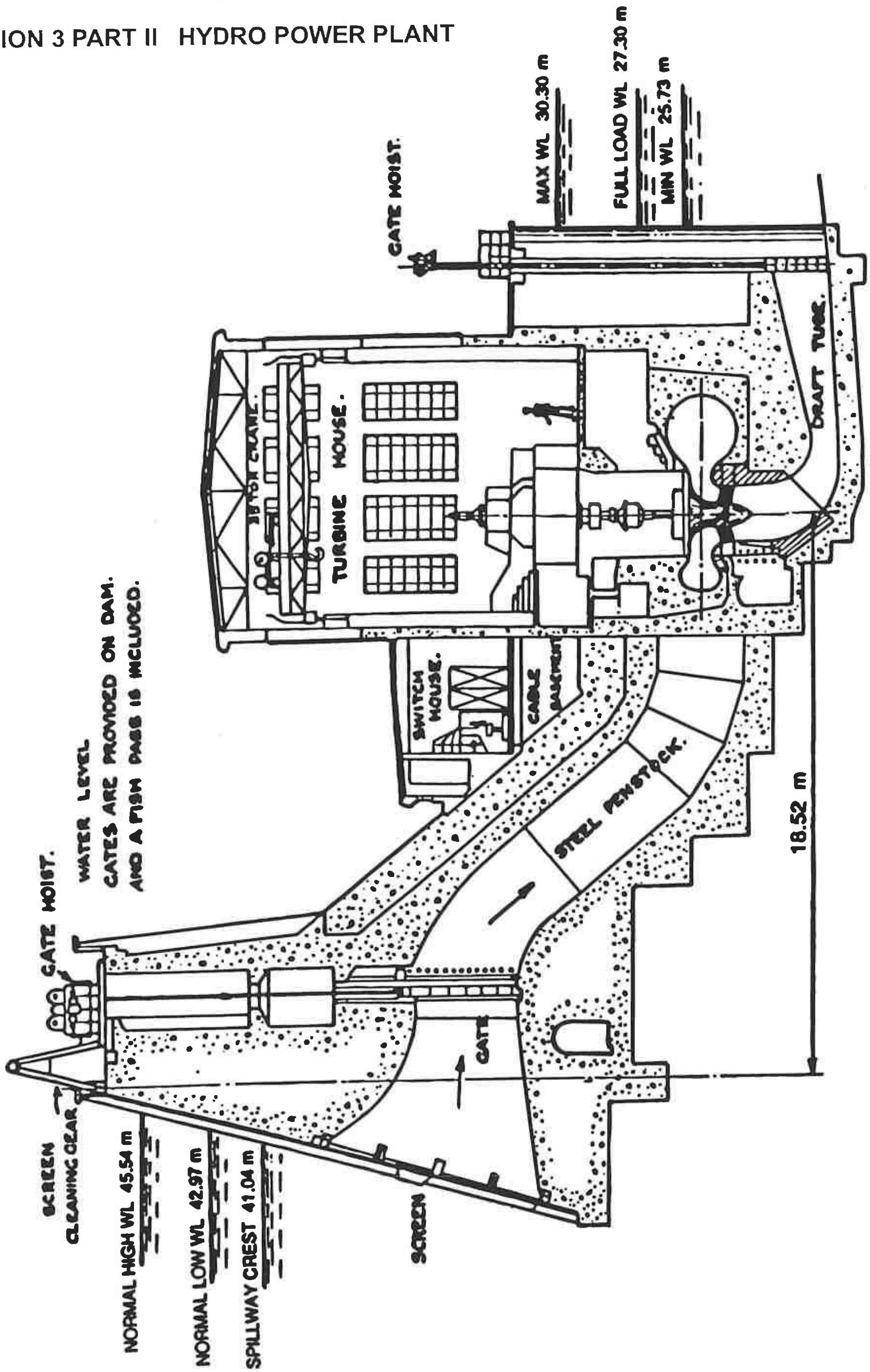


Figure 16.7 Kaplan turbine at Watts Bar Dam. 42,000 hp at 94.7 rpm under a head of 52 ft.

EXAMINATION PAPER ATTACHMENTS

QUESTION 3 PART II HYDRO POWER PLANT



4 MW Kaplan Turbine Setting.

EXAMINATION PAPER ATACHMENTS

QUESTION 4 HYDRO TURBINE PARAMETERS

Francis Turbine Runner

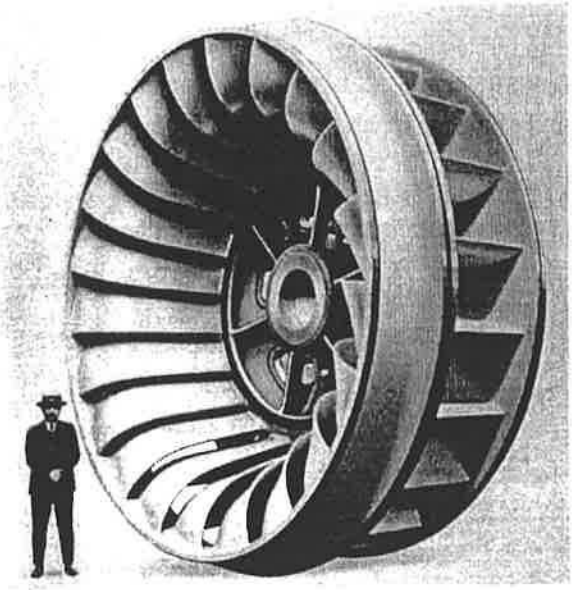
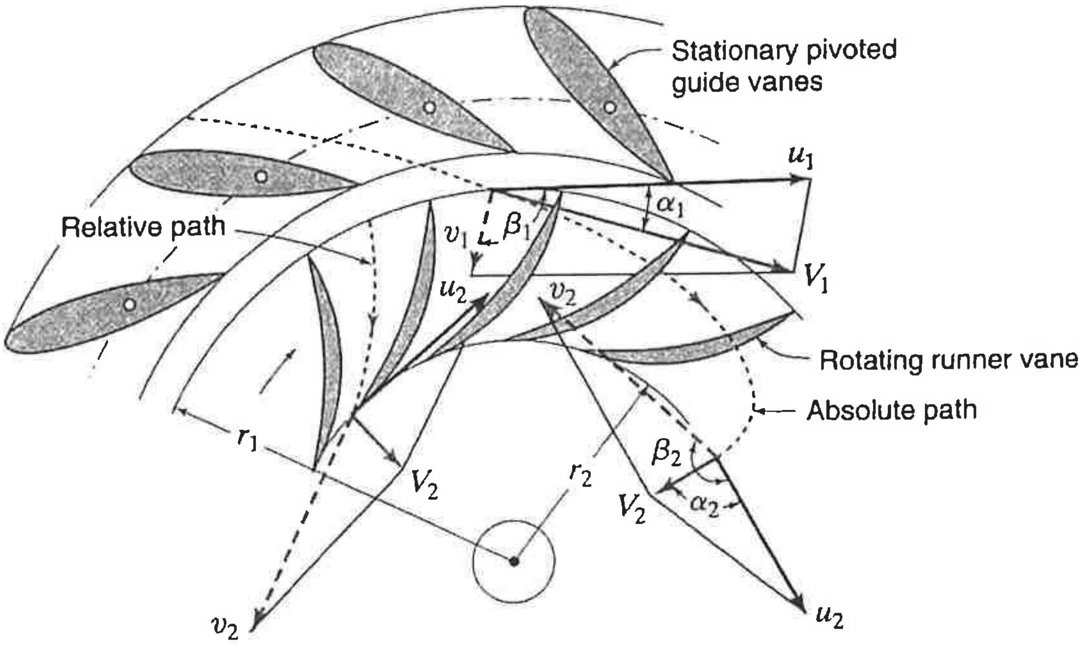


Figure 16.3 Francis runner at Niagara Falls. h = 214 ft, n = 107 rpm. η = 93.8 percent at 72,500 hp, diameter = 176 in, overall diameter at band = $183\frac{3}{8}$ in. *(Courtesy of Allis-Chalmers Mfg. Co.)*

Velocity Diagram

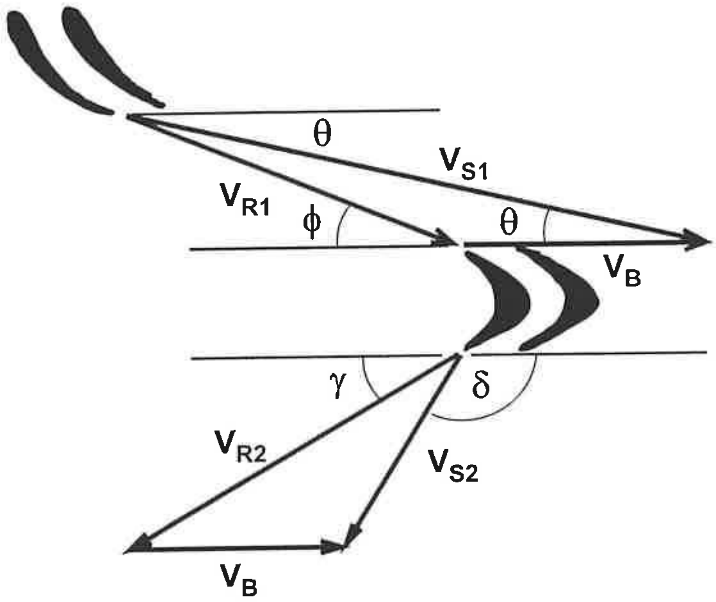


Velocity vectors for radial inward flow hydraulic turbine (Francis type)

EXAMINATION PAPER ATTACHMENTS

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

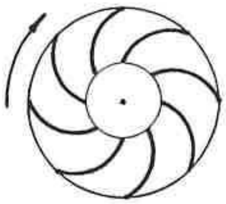
NAME ....

EXAMINATION PAPER ATTACHMENTS

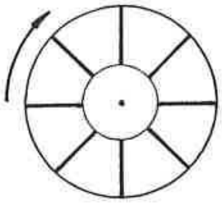
QUESTION 7 FAN CHARACTERISTICS

(a) Velocity Diagrams

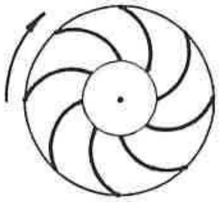
Draw velocity diagrams above each diagram and in line with the topmost blade for the three cases given.



Forward curved



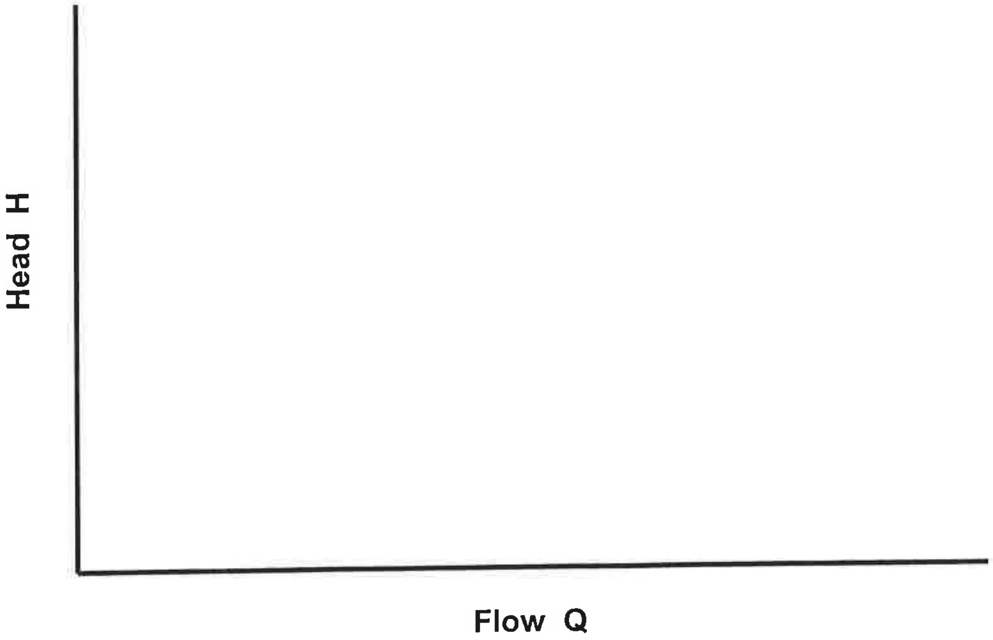
Radial



Backward curved

(a) Head versus Flow

On the axes given draw the head versus flow characteristics for the three cases above. Label the curves accordingly.



EXAMINATION REFERENCE MATERIAL

NOMENCLATURE FOR REFERENCE EQUATIONS (SI UNITS)

a	Sonic velocity	m/s
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
M_A	Mach number	
N	Rotational speed	rev/min
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
ν	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: $p_{vapour} = 1.71\ kPa$ (at 15°C)

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 $F = p_1 A_1 - p_2 A_2 - \rho VA(V_2 - V_1)$ (one dimensional)

Energy Equation

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

Compressible Flow

Mach Number $M_A = V/a$
Sonic Velocity $a = [kRT]^{1/2}$

Steam Turbines

Nozzle Equation: $h_1 - h_2 = (V_2^2 - V_1^2) / 2$
Work: $w = [(V_{1\,absolute}^2 - V_{2\,absolute}^2) + (V_{2\,relative}^2 - V_{1\,relative}^2)] / 2$
Work: $w = (V_{S1}\cos\theta - V_{S2}\cos\delta)\, V_{blade}$
Power: $P = wM$
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: $P = M\,[(V_{S1}^2 - V_{S2}^2) + (V_{R2}^2 - V_{R1}^2)] / 2$

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 $h_1 + \frac{1}{2}W_1^2 = h_2 + \frac{1}{2}W_2^2$
Stator Enthalpy Change $h_2 + \frac{1}{2}C_2^2 = h_3 + \frac{1}{2}C_3^2$
Isentropic Equation $(T_3/T_1) = (p_3/p_1)^{(k-1)/k}$

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: $\eta_P = 1 - (1 - \eta_M)\,(D_M/D_P)^{1/4}\,(H_M/H_P)^{1/10}$
Power: $P = \rho g Q H$

Pumps

Hydraulic Torque:	$\tau = \rho Q\,(r_2V_{2T} - r_1V_{1T})$
Hydraulic Torque:	$\tau = \rho Q\,(r_2V_2\cos\alpha_2 - r_1V_1\cos\alpha_1)$
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:	$(1 - \eta_P) / (1 - \eta_M) \approx (D_M/D_P)^{1/5}$

Hydro Turbines

Power	$P = \tau\omega$
Power	$P = \rho Q\omega\,(r_1V_1\cos\alpha_1 - r_2V_2\cos\alpha_2)$
Power	$P = \rho Q\,(U_1V_1\cos\alpha_1 - U_2V_2\cos\alpha_2)$
Peripheral Runner Velocity	$U_1 = \varphi\,(2gH)^{1/2}$
Absolute Water Velocity	$V_1 = C_1\,(2gH)^{1/2}$
Tangential Velocity	$U = \omega r$
Angular Velocity	$\omega = 2\pi N / 60$

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:	$\eta_p = 2\,V_{aircraft} / (V_{jet} + V_{aircraft})$

Wind Turbines

Maximum Ideal Power:	$P_{max} = 8\,\rho A V_1^3 / 27$