TURBO MACHINES

B.E, VSemester, Mechanical Engineering

[As per Choice Based Credit System (CBCS) scheme]

Course Code	17ME53	CIE Marks	40
Number of Lecture Hours/Week	04	SEE Marks	60
Total Number of Lecture Hours	50(10 Hours per Module)	Exam Hours	03
Credits – 04			

Course Objectives:

- **The course aims at giving an overview of different types of turbomachinery used for energy transformation, such as pumps, fans, compressors, as well as hydraulic and steam turbines.**
- **Explain the working principles of turbomachines and apply it to various types of machines**
- **It will focus on application of turbo machinery in power generation, power absorption and transportation sectors.**

Module - 1

Introduction: Definition of turbo machine, parts of turbo machines, Comparison with positive displacement machines, Classification, Dimensionless parameters and their significance, Effect of Reynolds number, Unit and specific quantities, model studies.

(Note: Since dimensional analysis is covered in Fluid Mechanics subject, questions on dimensional analysis may not be given. However, dimensional parameters and model studies may be given more weightage.)

Thermodynamics of fluid flow: Application of first and second law of thermodynamics to turbo machines, Efficiencies of turbo machines, Static and Stagnation states, Incompressible fluids and perfect gases, overall isentropic efficiency, stage efficiency (their comparison) and polytropic efficiency for both compression and expansion processes. Reheat factor for expansion process

Module - 2

Energy exchange in Turbo machines: Euler's turbine equation, Alternate form of Euler's turbine equation, Velocity triangles for different values of degree of reaction, Components of energy transfer, Degree of Reaction, utilization factor, Relation between degree of reaction and Utilization factor, Problems.

General Analysis of Turbo machines: Radial flow compressors and pumps – general analysis, Expression for degree of reaction, velocity triangles, Effect of blade discharge angle on energy transfer and degree of reaction, Effect of blade discharge angle on performance, Theoretical head – capacity relationship, General analysis of axial flow pumps and compressors, degree of reaction, velocity triangles, Problems.

Module - 3

Steam Turbines: Classification, Single stage impulse turbine, condition for maximum blade efficiency, stage efficiency, Need and methods of compounding, Multi-stage impulse turbine, expression for maximum utilization factor.

Reaction turbine – Parsons's turbine, condition for maximum utilization factor, reaction staging. Problems.

Module - 4

Hydraulic Turbines: Classification, various efficiencies.**Pelton turbine** – velocity triangles, design parameters, Maximum efficiency. **Francis turbine** - velocity triangles, design parameters, runner shapes for different blade speeds. Draft tubes- Types and functions. **Kaplan and Propeller turbines** - velocity triangles, design parameters. Problems.

Module - 5

Centrifugal Pumps: Classification and parts of centrifugal pump, different heads and efficiencies of centrifugal pump, Minimum speed for starting the flow, Maximum suction lift, Net positive suction head, Cavitation, Need for priming, Pumps in series and parallel. Problems.

Centrifugal Compressors: Stage velocity triangles, slip factor, power input factor, Stage work, Pressure developed, stage efficiency and surging and problems. Axial flow Compressors: Expression for pressure ratio developed in a stage, work done factor, efficiencies and stalling. Problems.

Course outcomes:

- **Able to give precise definition of turbomachinery**
- **Identify various types of turbo machinery**
- **Apply the Euler's equation for turbomachinery to analyse energy transfer in turbomachines**
- **Understand the principle of operation of pumps, fans, compressors and turbines.**
- **Perform the preliminary design of turbomachines (pumps, rotary compressors and turbines)**
- **Analyze the performance of turbo machinery.**

TEXT BOOKS:

1. An Introduction to Energy Conversion, Volume III, Turbo machinery, V. Kadambi and Manohar Prasad, New Age International Publishers, reprint 2008.
2. Turbo Machines ,B.U.Pai , 1st Editions, Wiley India Pvt, Ltd.
3. Turbines, Compressors & Fans, S. M. Yahya, Tata McGraw Hill Co. Ltd., 2nd edition, 2002

REFERENCE BOOKS

1. Principals of Turbo machines, D. G. Shepherd, The Macmillan Company (1964).
2. Fluid Mechanics & Thermodynamics of Turbo machines, S. L. Dixon, Elsevier (2005).
3. Text Book of Turbo machines, M. S. Govindegouda and A. M. Nagaraj, M. M. Publications, 4Th Ed, 2008.

|| Jai Sri Gurudev ||

Adichunchanagiri Shikshana Trust (R)

BGS INSTITUTE OF TECHNOLOGY

Department of Mechanical Engineering

CO-PO & CO-PSO mapping (17 Scheme)

Programme	Course Code	Subject	Credits	L-T-P	Assessment		Exam Duration
					SEE	CIA	
B.E	17ME53	Turbo Machine	04	3-2-0	60	40	3Hrs

Co's

17C303.1	Able to give precise definition of turbo machinery
17C303.2	Identify various types of turbo machinery.
17C303.3	Apply the Euler's equation for turbo machinery to analyze energy transfer in turbo machines.
17C303.4	Understand the principle of operation of pumps, fans, compressors and turbines
17C303.5	Perform the preliminary design of turbo machines (pumps, rotary compressors and turbines)
17C303.6	Analyze the performance of turbo machinery

Po & Pso's

PSO-1: Ability to acquire competencies in designing, analyzing and evaluating the mechanical components.

PSO-2: Ability to work professionally by applying manufacturing and management practices.

CO	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2
17C303.1	3	1	2	-	-	-	-	-	-	-	-	-	1	-
17C303.2	3	1	2	-	-	-	-	-	-	-	-	-	1	-
17C303.3	3	1	1	2	-	-	-	-	-	-	-	-	2	-
17C303.4	3	1	1	-	-	-	-	-	-	-	-	1	1	-
17C303.5	3	3	3	2	-	-	-	-	-	-	-	-	2	-
17C303.6	3	3	3	2	-	-	-	-	-	-	-	-	2	-
AVG	3	1.67	2	2	-	-	-	-	-	-	-	1	1.5	-

Course Owner

Verified

HOD

Head of the Department
Department of Mechanical Engineering
BGSIT B G Nagar-571448

	CIE (%)	SEE (%)	CES (%)	
COs	60	30	10	TOTAL
CO1	2.51	0.36	2.97	1.91
CO2	2.19	0.36	2.94	1.71
CO3	2.60	0.36	2.97	1.97
CO4	2.58	0.36	2.95	1.95
CO5	2.13	0.36	2.94	1.68
CO6	2.22	0.36	2.97	1.74

TURBOMACHINE		17ME43
	V sem	2019-2020
KEERTHI B L		

CO-PO/PSO Mapping Table

PO/PSO	Total	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2
CO1	1.91	3	1	2										1	
CO2	1.71	3	1	2										1	
CO3	1.97	3	1	1	2									2	
CO4	1.95	3	1	1									1	1	
CO5	1.68	3	3	3	2									2	
CO6	1.74	3	3	3	2									2	
Sum		18	10	12	6	0	0	0	0	0	0	0	1	9	0
Number		6	6	6	3	0	0	0	0	0	0	0	1	6	0
Average		3	1.666667	2	2	0	0	0	0	0	0	0	1	1.5	0
Weighted Sum		32.88713	17.79955	21.42796	10.76993	0	0	0	0	0	0	0	1.949007	16.347342	0
PO Attainment		1.83	0.99	1.19	1.20	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.65	0.91	0.00

Verified

Head of the Department
Department of Mechanical Engineering
BGSIT B G Nagar-571448

BGS Institute of Technology

Mandya

COURSE BOOK

Period of the Semester : **From** 29 Jul 2019 **To** 30 Nov 2019

Dept-Sem-Sec: ME-5-A

Subject with Code: TURBO MACHINES 17ME53

Time Slot

MON:	09:55 - 10:50	**TUE :**	13:45 - 14:35
WED:	09:55 - 10:50	**THU :**	
FRI :	09:00 - 09:55	**SAT :**	09:55 - 10:50

Name of the Teacher : B.L. Keerthi

BGSIT	Lesson Plan & Execution	

Name of the Faculty	***B.L. Keerthi***
Dept-Sem-Sec:	***ME-5-A***
Date of Commencement	***29 Jul 2019***
Last working day of Semester	***30 Nov 2019***
Source Material List	
1	Principals of Turbo machines, D. G. Shepherd, The Macmillan Company (1964).
2	Fluid Mechanics & Thermodynamics of Turbo machines, S. L. Dixon, Elsevier (2005).
3	Text Book of Turbo machines, M. S. Govindegouda and A. M. Nagaraj, M. M. Publications, 4Th Ed, 2008.
Course Outcome List	
1	Able to give precise definition of turbomachinery
2	Identify various types of turbo machinery
3	Apply the Euler 's equation for turbomachinery to analyse energy transfer in turbomachines
4	Understand the principle of operation of pumps, fans, compressors and turbines.
5	Perform the preliminary design of turbomachines (pumps, rotary compressors and turbines)
6	Analyze the performance of turbo machinery
Subject Name	TURBO MACHINES

Period	*Planned*			*Execution*		
	Date	***Topic***	***Source material to be referred***	***Date***	***Topic***	***Source material to be referred***
Module 1						
1	2 Aug 2019	Definition of turbo machine, parts of turbo machines		2 Aug 2019	Definition of turbo machine, parts of turbo machines	REF 1
2	3 Aug 2019	Comparison with positive displacement machines, Classification		3 Aug 2019	Comparison with positive displacement machines, Classification	REF 1 REF 2 REF 3
3	5 Aug 2019	Dimensionless parameters and their significance, Effect of Reynolds number		5 Aug 2019	Dimensionless parameters and their significance, Effect of Reynolds number	REF 1 REF 2 REF 3
4	6 Aug 2019	Unit and specific quantities, model studies		6 Aug 2019	Unit and specific quantities, model studies	REF 1 REF 2 REF 3
5	7 Aug 2019	(Note- Since dimensional analysis is covered in Fluid Mechanics subject, questions on dimensional analysis may not be given		7 Aug 2019	(Note- Since dimensional analysis is covered in Fluid Mechanics subject, questions on dimensional analysis may not be given	REF 1 REF 2 REF 3
6	9 Aug 2019	However		13 Aug 2019	model studies	REF 1 REF 2 REF 3
7	10 Aug 2019	dimensional parameters and model studies may be given more weightage)		13 Aug 2019	dimensional parameters and model studies may be given more weightage), questions on dimensional analysis may not be given	REF 1 REF 2 REF 3
8	12 Aug 2019	Application of first and second law of thermodynamics to turbo machines		16 Aug 2019	Application of first and second law of thermodynamics to turbo machines	REF 1 REF 2 REF 3
9	13 Aug 2019	Efficiencies of turbo machines		17 Aug 2019	Efficiencies of turbo machines	REF 1 REF 2 REF 3
10	14 Aug 2019	Static and Stagnation states		19 Aug 2019	Static and Stagnation states, Incompressible fluids and perfect gases	REF 1 REF 2 REF 3
11	16 Aug 2019	Incompressible fluids and perfect gases		20 Aug 2019	Incompressible fluids and perfect gases, stage efficiency (their comparison) and polytropic efficiency for both compression and expansion processes	REF 1 REF 2 REF 3

Period	Planned			Execution		
	Date	*Topic*	*Source material to be referred*	*Date*	*Topic*	*Source material to be referred*
12	17 Aug 2019	overall isentropic efficiency		21 Aug 2019	overall isentropic efficiency, stage efficiency (their comparison) and polytropic efficiency for both compression and expansion processes	REF 1 REF 2 REF 3
13	19 Aug 2019	stage efficiency (their comparison) and polytropic efficiency for both compression and expansion processes		19 Aug 2019	stage efficiency (their comparison) and polytropic efficiency for both compression and expansion processes, Reheat factor for expansion process	REF 1 REF 2 REF 3
14	20 Aug 2019	Reheat factor for expansion process		23 Aug 2019	Reheat factor for expansion process	REF 2 REF 1 REF 3
Module 2						
15	21 Aug 2019	Euler 's turbine equation, Alternate form of Euler 's turbine equation		26 Aug 2019	Euler 's turbine equation, Alternate form of Euler 's turbine equation	REF 1 REF 2 REF 3
16	23 Aug 2019	Velocity triangles for different values of degree of reaction, Components of energy transfer		23 Aug 2019	Velocity triangles for different values of degree of reaction, Components of energy transfer	REF 1 REF 2 REF 3
17	24 Aug 2019	Degree of Reaction		26 Aug 2019	Degree of Reaction	REF 1 REF 2 REF 3
18	26 Aug 2019	utilization factor		27 Aug 2019	utilization factor, Relation between degree of reaction and Utilization factor	REF 1 REF 2 REF 3
19	27 Aug 2019	Relation between degree of reaction and Utilization factor		27 Aug 2019	Relation between degree of reaction and Utilization factor, Problems.	REF 1 REF 2 REF 3
20	28 Aug 2019	Problems.		28 Aug 2019	Problems., Problems.	REF 1 REF 2 REF 3
21	30 Aug 2019	Radial flow compressors and pumps – general analysis, Expression for degree of reaction		30 Aug 2019	Radial flow compressors and pumps – general analysis, Expression for degree of reaction	REF 1 REF 2 REF 3

Period	*Planned*			*Execution*		
	Date	***Topic***	***Source material to be referred***	***Date***	***Topic***	***Source material to be referred***
22	31 Aug 2019	velocity triangles, Effect of blade discharge angle on energy transfer and degree of reaction		30 Aug 2019	velocity triangles, Expression for degree of reaction	REF 1 REF 2 REF 3
23	2 Sep 2019	Effect of blade discharge angle on performance, Theoretical head – capacity relationship		31 Aug 2019	Effect of blade discharge angle on performance, Theoretical head – capacity relationship	REF 1 REF 2 REF 3
24	3 Sep 2019	General analysis of axial flow pumps and compressors, degree of reaction		4 Sep 2019	General analysis of axial flow pumps and compressors, degree of reaction	REF 1 REF 2 REF 3
25	4 Sep 2019	velocity triangles		6 Sep 2019	velocity triangles, Problems.	REF 1 REF 2 REF 3
26	6 Sep 2019	Problems.		9 Sep 2019	Problems.	REF 1 REF 2 REF 3
Module 3						
27	7 Sep 2019	Classification, Single stage impulse turbine		9 Sep 2019	Classification, Single stage impulse turbine	REF 1 REF 2 REF 3
28	9 Sep 2019	condition for maximum blade efficiency		13 Sep 2019	condition for maximum blade efficiency, stage efficiency	REF 1 REF 2 REF 3
29	10 Sep 2019	stage efficiency		16 Sep 2019	Need and methods of compounding	REF 1 REF 2 REF 3
30	11 Sep 2019	Need and methods of compounding		17 Sep 2019	Need and methods of compounding, Multi-stage impulse turbine	REF 1 REF 2 REF 3
31	13 Sep 2019	Multi-stage impulse turbine		18 Sep 2019	Multi-stage impulse turbine, expression for maximum utilization factor.	REF 1 REF 2 REF 3
32	14 Sep 2019	expression for maximum utilization factor.		23 Sep 2019	expression for maximum utilization factor., expression for maximum utilization factor.	REF 1 REF 2 REF 3
33	16 Sep 2019	Parsons 's turbine		27 Sep 2019	Parsons 's turbine	REF 1 REF 2 REF 3
34	17 Sep 2019	Parsons 's turbine		30 Sep 2019	condition for maximum utilization factor	REF 1 REF 2 REF 3

Period	Planned			Execution		
	Date	***Topic***	***Source material to be referred***	***Date***	***Topic***	***Source material to be referred***
35	18 Sep 2019	condition for maximum utilization factor		1 Oct 2019	Problems.	REF 1
36	20 Sep 2019	condition for maximum utilization factor		4 Oct 2019	Problems.	REF 1
37	21 Sep 2019	reaction staging		4 Oct 2019	Problems.	REF 1 REF 2 REF 3
38	23 Sep 2019	Problems.		5 Oct 2019	Problems.	REF 1 REF 2 REF 3
Module 4						
39	1 Oct 2019	Classification, various efficiencies		9 Oct 2019	Classification, various efficiencies	REF 1 REF 2 REF 3
40	2 Oct 2019	Pelton turbine – velocity triangles, design parameters		10 Oct 2019	Pelton turbine – velocity triangles, design parameters	REF 1 REF 2 REF 3
41	4 Oct 2019	Maximum efficiency		14 Oct 2019	Maximum efficiency	REF 1 REF 2 REF 3
42	5 Oct 2019	Francis turbine - velocity triangles		15 Oct 2019	Francis turbine - velocity triangles	REF 1 REF 2 REF 3
43	7 Oct 2019	design parameters		16 Oct 2019	design parameters, Problems	REF 1 REF 2 REF 3
44	8 Oct 2019	runner shapes for different blade speeds		18 Oct 2019	runner shapes for different blade speeds, Draft tubes- Types and functions	REF 1 REF 2 REF 3
45	9 Oct 2019	Draft tubes- Types and functions		19 Oct 2019	Problems	REF 1 REF 2 REF 3
46	11 Oct 2019	Kaplan and Propeller turbines - velocity triangles		21 Oct 2019	Kaplan and Propeller turbines - velocity triangles	REF 1 REF 2 REF 3
47	12 Oct 2019	design parameters		22 Oct 2019	design parameters	REF 1 REF 2 REF 3
48	14 Oct 2019	Problems		23 Oct 2019	Problems	REF 1 REF 2 REF 3

Period	*Planned*			*Execution*		
	Date	*Topic*	*Source material to be referred*	*Date*	*Topic*	*Source material to be referred*
Module 5						
49	25 Oct 2019	Classification and parts of centrifugal pump, different heads and efficiencies of centrifugal pump		30 Oct 2019	Classification and parts of centrifugal pump, different heads and efficiencies of centrifugal pump	REF 1 REF 2 REF 3
50	26 Oct 2019	Minimum speed for starting the flow, Maximum suction lift		2 Nov 2019	Minimum speed for starting the flow, Maximum suction lift	REF 1 REF 2 REF 3
51	28 Oct 2019	Net positive suction head, Cavitation		4 Nov 2019	Net positive suction head, Cavitation	REF 1 REF 2 REF 3
52	29 Oct 2019	Need for priming, Pumps in series and parallel		5 Nov 2019	Need for priming, Pumps in series and parallel	REF 1 REF 2 REF 3
53	30 Oct 2019	Problems.		6 Nov 2019	Problems.	REF 1 REF 2 REF 3
54	1 Nov 2019	Stage velocity triangles, slip factor		8 Nov 2019	Stage velocity triangles, slip factor	REF 1 REF 2 REF 3
55	2 Nov 2019	power input factor, Stage work		11 Nov 2019	power input factor, Stage work	REF 1 REF 2 REF 3
56	4 Nov 2019	Pressure developed, stage efficiency and surging and problems		12 Nov 2019	Pressure developed, stage efficiency and surging and problems	REF 1 REF 2 REF 3
57	5 Nov 2019	Axial flow Compressors- Expression for pressure ratio developed in a stage, work done factor		13 Nov 2019	Axial flow Compressors- Expression for pressure ratio developed in a stage, work done factor	REF 1 REF 2 REF 3
58	6 Nov 2019	efficiencies and stalling, Problems		18 Nov 2019	efficiencies and stalling, Problems	REF 1 REF 2 REF 3

Module No.	# of Classes Planned(till date)	Planned Effort(till date)	# of Classes Executed(till date)	Actual Effort (till date)	% Coverage
1	14	12hrs 50min	14	12hrs 50min	100.0
2	12	11hrs 0min	12	11hrs 0min	100.0
3	12	11hrs 0min	12	11hrs 0min	100.0
4	10	9hrs 10min	10	9hrs 10min	100.0
5	10	9hrs 10min	10	9hrs 10min	100.0

Faculty in charge

HOD's Signature

Signature of Principal (&remark if any)

Module 1 : THERMODYNAMICS OF FLUID FLOW

This chapter deals with the some basic definitions of thermodynamics applies to the turbomacines and the discussions on the thermodynamics of the fluid flow through turbomachines.

3.1 Sonic Velocity and Mach Number:

Question No 3.1: Define Mach number and hence explain subsonic flow, sonic flow and supersonic flow. Or, write a note on Mach number. (VTU, Dec-09/Jan-10) Or,

Give classification of fluid flow based on Mach number and explain in brief. (VTU, Dec-12)

Answer: Sonic velocity (velocity of the sound) is referred to the speed of propagation of pressure wave in the medium. The velocity of the sound in a fluid at a local temperature *T* for an isentropic flow is given by

$$c = \sqrt{\gamma RT}.$$

Where γ, R and T are the ratio of specific heats, characteristic gas constant and the local temperature of the fluid respectively. At sea level the velocity of sound in air is given as 340 m/s.

Mach number is defined as the ratio of local velocity of fluid (V) to the sonic velocity (c) in that fluid. Thus

$$M = \frac{V}{c} = \frac{V}{\sqrt{\gamma RT}}$$

The fluid flow can be generally classified into subsonic flow, sonic flow and supersonic flow based on the value of Mach number.

Subsonic flow: If the Mach number is less than 1, then that type of flow is called subsonic flow, in which the velocity of the fluid is less than the velocity of the sound in that medium.

Sonic flow: If the Mach number is equal to 1, then that type of flow is called sonic flow, in which the velocity of the fluid is same as the velocity of the sound in that medium.

Supersonic flow: If the Mach number is greater than 1, then that type of flow is called supersonic flow, in which the velocity of the fluid is greater than the velocity of the sound in that medium.

3.2 Isentropic Flow for a Varying Flow Area:

Question No 3.2: For the isentropic flow through varying flow area, show that $\frac{dA}{A} = \frac{dp}{p}\left(\frac{1-M^2}{\gamma M^2}\right)$ ***and discuss the physical significance. Or, derive an expression for area ratio for isentropic flow through a passage of varying cross sectional area and discuss the significance of the expression. (VTU, Jun/Jul-13)***

Answer: The Continuity equation is given by,

$$\frac{dA}{A} + \frac{dV}{V} + \frac{d\rho}{\rho} = 0$$

Or,

$$\frac{dA}{A} = -\left(\frac{dV}{V} + \frac{d\rho}{\rho}\right)$$

But isentropic equation is,

$$\frac{dp}{p} = \gamma\frac{d\rho}{\rho} \Rightarrow \frac{dp}{\gamma p} = \frac{d\rho}{\rho}$$

But Euler's equation is,

$$\frac{dp}{\rho} + VdV = 0 \Rightarrow \frac{dp}{\rho V^2} + \frac{dV}{V}$$

$$\frac{dV}{V} = -\frac{dp}{\rho V^2}$$

From Mach number,

$$V^2 = M^2\gamma RT = M^2\gamma\left(\frac{p}{\rho}\right)$$

$$\rho V^2 = M^2\gamma p$$

Then,

$$\frac{dV}{V} = -\frac{dp}{M^2\gamma p} \qquad (3.1)$$

Therefore,

$$\frac{dA}{A} = -\left(-\frac{dp}{M^2\gamma p} + \frac{dp}{\gamma p}\right)$$

$$\frac{dA}{A} = \frac{dp}{M^2\gamma p} - \frac{dp}{\gamma p} = \frac{dp}{p}\left(\frac{1}{M^2\gamma} - \frac{1}{\gamma}\right)$$

$$\frac{dA}{A} = \frac{dp}{p}\left(\frac{1 - M^2}{\gamma M^2}\right) \qquad (3.2)$$

The significance of the equations (3.1) and (3.2) is discussed below:

The equation (3.1) shows that for nozzle pressure decreases as velocity increases and for diffuser velocity decreases as pressure increases.

For subsonic flow (M<1) the quantity $\left(\frac{1-M^2}{\gamma M^2}\right)$ is positive. In the nozzle pressure decreases, so the quantity $\frac{dp}{p}$ is negative; therefore from equation (3.2) the quantity $\frac{dA}{A}$ is also negative and hence area must decrease for subsonic nozzle in the direction of fluid flow. The shape of the subsonic nozzle (convergent nozzle) is as shown in figure 3.1.

In the diffuser pressure increases, so the quantity $\frac{dp}{p}$ is positive; therefore from equation (3.2) the quantity $\frac{dA}{A}$ is also positive and hence area must increase for subsonic diffuser in the direction of fluid flow. The shape of the subsonic diffuser (divergent diffuser) is as shown in figure 3.1.

For supersonic flow (M>1) the quantity $\left(\frac{1-M^2}{\gamma M^2}\right)$ is negative. In the nozzle pressure decreases, so the quantity $\frac{dp}{p}$ is negative; therefore from equation (3.2) the quantity $\frac{dA}{A}$ is positive and hence area must increase for supersonic nozzle in the direction of fluid flow. The shape of the supersonic nozzle (divergent nozzle) is as shown in figure 3.2.

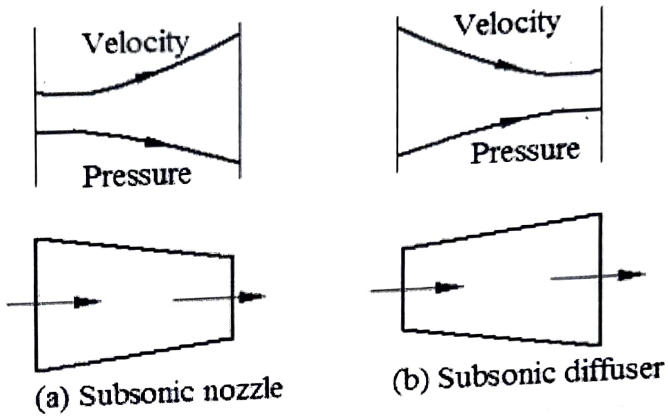

Fig. 3.1 Subsonic nozzle and diffuser

In the diffuser pressure increases, so the quantity $\frac{dp}{p}$ is positive; therefore from equation (3.2) the quantity $\frac{dA}{A}$ is negative and hence area must decrease for supersonic diffuser in the direction of fluid flow. The shape of the supersonic diffuser (convergent diffuser) is as shown in figure 3.2.

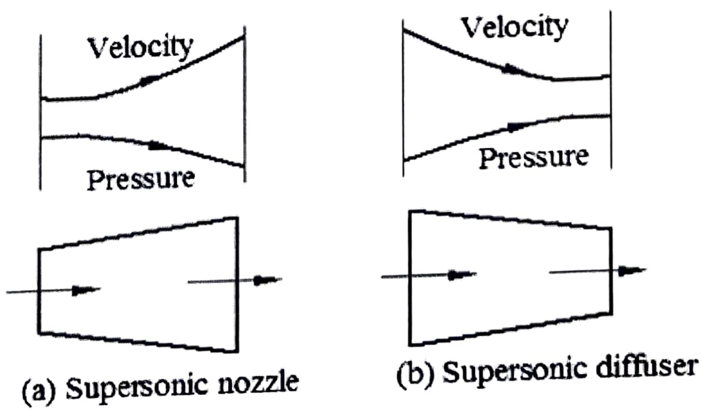

Fig. 3.2 Supersonic nozzle and diffuser

For sonic flow (M=1) the quantity $\left(\frac{1-M^2}{\gamma M^2}\right)$ is zero, from equation (3.2) the quantity $\frac{dA}{A}$ is also zero, i.e., area must be constant. This is the situation occurs at the throat portion of the convergent-divergent nozzle.

Note: The subsonic diffuser, subsonic nozzle and the supersonic nozzle are all of practical importance as for as the turbomachines are concerned, while the supersonic diffuser is of interest for wind tunnel and ram jet.

3.3 Static and Stagnation States:

Question No 3.3: Define static state and stagnation state for a fluid.

(VTU, Dec-11, Dec-12, Dec-14/Jan-15)

Answer: There are two kinds of state for the flowing fluid, namely static state and stagnation state.

(i) Static state: It is the state refers to those properties like pressure, temperature, density etc. which are measured when the measuring instruments are at rest relative to the flow of fluid.

(ii) Stagnation state: It is the final state of a fictitious, isentropic and work free process during which the final kinetic and potential energies of the fluid reduces to zero in a steady flow.

Question No 3.4: Write expressions for (i) stagnation enthalpy, (ii) stagnation temperature, (iii) stagnation pressure and (iv) stagnation density.

Answer: For a fictitious, isentropic and work free process the initial state is always the static state and final state is stagnation state. A steady flow energy equation (SFEE) for this fictitious process can be written as:

$$h_o + \frac{1}{2}V_o^2 + gZ_o + w = h + \frac{1}{2}V^2 + gZ + q$$

For isentropic and work free process, q=0 and w=0 and at the final state (stagnation state) of this process, ke=0 and pe=0. Thus steady flow energy equation is:

$$h_o = h + \frac{1}{2}V^2 + gZ$$

(i) Stagnation Enthalpy: It is defined as the enthalpy of a fluid when it is adiabatically decelerated to zero velocity. The stagnation enthalpy can be written as:

$$\boldsymbol{h_o = h + \frac{1}{2}V^2 + gZ}$$

Or,

$$\boldsymbol{h_o = h + \frac{1}{2}V^2}$$

(ii) Stagnation Temperature: It is defined as the temperature of a fluid when it is adiabatically decelerated to zero velocity. The stagnation temperature defined through stagnation enthalpy as:

$$c_pT_o = c_pT + \frac{1}{2}V^2$$

$$\boldsymbol{T_o = T + \frac{V^2}{2c_p}}$$

Or,

$$\frac{T_o}{T} = 1 + \frac{V^2}{2c_pT} = 1 + \frac{V^2(\gamma - 1)}{2\gamma RT} = 1 + \left(\frac{\gamma - 1}{2}\right)\frac{V^2}{c^2}$$

$$\boldsymbol{\frac{T_o}{T} = 1 + \left(\frac{\gamma - 1}{2}\right)M^2}$$

(iii) Stagnation Pressure: It is defined as the pressure of a fluid when it is adiabatically decelerated to zero velocity. The relation between the stagnation and static pressures can be written as:

$$\frac{p_o}{p} = \left(\frac{T_o}{T}\right)^{\frac{\gamma}{\gamma-1}}$$

$$\frac{p_o}{p} = \left[1 + \left(\frac{\gamma - 1}{2}\right) M^2\right]^{\frac{\gamma}{\gamma-1}}$$

For incompressible flows, $h = \frac{p}{\rho}$

$$\frac{p_o}{\rho} = \frac{p}{\rho} + \frac{V^2}{2}$$

$$p_o = p + \frac{\rho V^2}{2}$$

(iv) Stagnation Density: The stagnation density can be defined by using stagnation pressure and temperature. For an isentropic process,

$$\frac{\rho_o}{\rho} = \left(\frac{T_o}{T}\right)^{\frac{1}{\gamma-1}}$$

$$\frac{\rho_o}{\rho} = \left[1 + \left(\frac{\gamma - 1}{2}\right) M^2\right]^{\frac{1}{\gamma-1}}$$

3.4 Compression Process in Compressor:

3.4.1 Efficiency of Compression Process:

Question No 3.5: Define the following, with the help of a h-s diagram, for the power absorbing turbomachines: (i) Total-to-total efficiency, (ii) Total-to-static efficiency, (iii) Static-to-total efficiency, (iv) Static-to static efficiency. (VTU, Dec-06/Jan-07)

Answer: The h-s diagram for the compression process is shown in figure 3.3. The fluid has initially the static pressure and temperature determines by state 1, the state 01 is the corresponding stagnation state. After passing through the turbomachine, the final static properties of the fluid are determined by state 2 and state 02 is corresponding stagnation state. If the process is reversible, the final fluid static state would be 2' while stagnation state would be 02'. Line 1-2 in static coordinates and line 01-02 in stagnation coordinates represent the real process.

The actual work input for compression process is,

$$w = h_{02} - h_{01}$$

The ideal work input can be calculated by any one of the following four equations:

(i) Totol-to-total work input is the ideal work input for the stagnation ends,

$$w_{t-t} = h_{02'} - h_{01}$$

(ii) Total-to-static work input is the ideal work input for the stagnation inlet to the static exit,

$$w_{t-s} = h_{2'} - h_{01}$$

(iii) Static-to-total work input is the ideal work input for the static inlet to the stagnation exit,

$$w_{s-t} = h_{02'} - h_1$$

(iv) Static-to-static work input is the ideal work input for the static inlet to the static exit,

$$w_{s-s} = h_{2'} - h_1$$

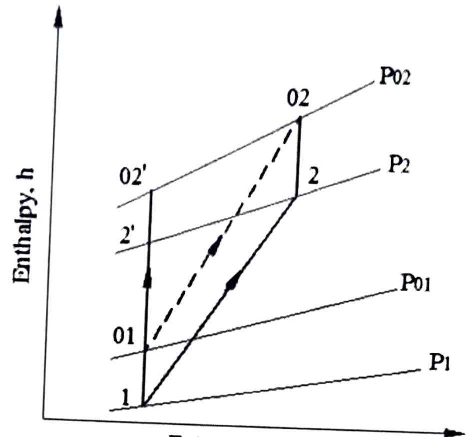

Fig. 3.3 h-s diagram for compression process

The efficiency of the compression process can be expressed by any one of the following equations:

(i) Total-to-total efficiency is defined as the ratio of total-to-total work input to the actual work input.

$$\eta_{t-t} = \frac{w_{t-t}}{w} = \frac{h_{02'} - h_{01}}{h_{02} - h_{01}}$$

(ii) Total-to-static efficiency is defined as the ratio of total-to-static work input to the actual work input.

$$\eta_{t-s} = \frac{w_{t-s}}{w} = \frac{h_{2'} - h_{01}}{h_{02} - h_{01}}$$

(iii) Static-to-total efficiency is defined as the ratio of static-to-total work input to the actual work input.

$$\eta_{s-t} = \frac{w_{s-t}}{w} = \frac{h_{02'} - h_1}{h_{02} - h_{01}}$$

(iv) Static-to-static efficiency is defined as the ratio of static-to-static work input to the actual work input.

$$\eta_{s-s} = \frac{w_{s-s}}{w} = \frac{h_{2'} - h_1}{h_{02} - h_{01}}$$

3.4.2 Effect of Pre-heat:

Question No 3.6: ***With the help of T-s diagram, show that the preheat factor in a multistage compressor is less than unity.*** *(VTU, May/Jun-10, Jun/Jul-11)*

Answer: The preheat factor for a compressor may be defined as the ratio of direct or Rankine isentropic work to the cumulative isentropic work.

The thermodynamic effect of multistage compression can be studied by considering three stage compressor working between inlet pressure p_1 and the delivery pressure p_2 as shown in the figure 3.4. The intermediate pressures are being p_A and p_B. The stage pressure ratio, p_r and the stage efficiency, η_{st} are assumed to be same for all stages. The process 1-2' and 1-2 are the isentropic and actual compression process respectively.

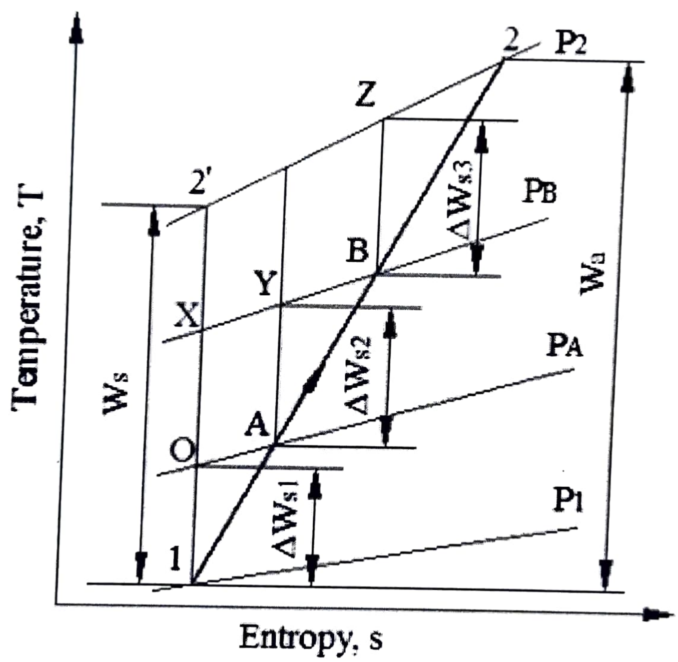

Fig. 3.4 Effect of preheat on compression process

As the constant pressure lines are diverging towards the right hand side of the temperature-entropy diagram, the isentropic work per stage increases as the temperature difference increases for the same pressure ratio and the stage efficiency. For example, in the second stage between pressures p_A and p_B, the isentropic temperature difference represented by the line A-Y is greater than that represented by the line X-O. It is therefore the isentropic work for the stage is greater by virtue of the inefficiency of the previous stage. Similarly for the next stage also.

Therefore, $$w_s < (\Delta w_{s1} + \Delta w_{s2} + \Delta w_{s3})$$

Or, $$w_s < \Sigma \Delta w_s$$

$$\frac{w_s}{\Sigma \Delta w_s} < 1$$

Therefore, the Preheat factor $\frac{w_s}{\Sigma \Delta w_s}$ is always less than unity for multistage compressor. This is due to the preheating of the fluid at the end of each compression stage and this appears as the losses in the subsequent stages.

Question No 3.7: For a multistage compressor, show that the overall efficiency is less than the stage efficiency using T-s diagram. *(VTU, Jun/Jul-08)*

Answer: Consider three stage compressor working between inlet pressure p_1 and the delivery pressure p_2 as shown in the figure 3.4. The intermediate pressures are being p_A and p_B. The stage pressure ratio, p_r and the stage efficiency, η_{st} are assumed to be same for all stages. The process 1-2' and 1-2 are the isentropic and actual compression process respectively.

If the overall efficiency of the multistage compressor is η_o, then the total actual work is given by,

$$w_a = \frac{w_s}{\eta_o}$$

Or,

$$w_s = \eta_o w_a$$

The total actual work can also be written as the sum of the actual work done in each stage,

$$w_a = w_{a1} + w_{a2} + w_{a3} = \frac{\Delta w_{s1}}{\eta_{st}} + \frac{\Delta w_{s2}}{\eta_{st}} + \frac{\Delta w_{s3}}{\eta_{st}}$$

$$w_a = \frac{1}{\eta_{st}}(\Delta w_{s1} + \Delta w_{s2} + \Delta w_{s3})$$

$$w_a = \frac{1}{\eta_{st}} \Sigma \Delta w_s$$

Or,

$$\Sigma \Delta w_s = \eta_{st} w_a$$

As the constant pressure lines are diverging towards the right hand side of the temperature-entropy diagram, the isentropic work per stage increases as the temperature difference increases for the same pressure ratio and the stage efficiency.

Therefore,

$$w_s < (\Delta w_{s1} + \Delta w_{s2} + \Delta w_{s3})$$

Or,

$$w_s < \Sigma \Delta w_s$$

$$\eta_o w_a < \eta_{st} w_a$$

$$\eta_o < \eta_{st}$$

For multistage compressor, the overall isentropic efficiency is less than the stage efficiency.

3.4.3 Infinitesimal Stage Efficiency or Polytropic Efficiency:

Question No 3.8: Obtain an expression for polytropic efficiency for a compressor in terms of pressure ratio and temperature ratio. Further express stage efficiency in terms of polytropic efficiency and pressure ratio. Also draw the relevant T-s diagram. *(VTU, Jun/Jul-13)* ***Or, Define the***

term infinitesimal stage efficiency of a compressor. Show that the polytropic efficiency during the compression process is given by $\eta_p = \frac{\frac{\gamma-1}{\gamma}\ln\left(\frac{p_2}{p_1}\right)}{\ln\left(\frac{T_2}{T_1}\right)}$ ***.(VTU, Dec-14/Jan-15)***

Answer: A finite compressor stage is made up of number of infinitesimal stages; the efficiency of these small stages is called polytropic efficiency or infinitesimal stage efficiency.

Consider a single stage compressor having its stage efficiency η_{st}, operates between the pressures p_1 and p_2, and an infinitesimal stage of efficiency η_p, working between the pressures p and p+dp as shown in figure 3.5.

The infinitesimal stage efficiency is given by,

$$\eta_p = \frac{Isentropic\ Temperature\ Rise}{Actual\ Temperature\ Rise} = \frac{dT'}{dT}$$

The actual temperature rise for infinitesimal stage is given by,

$$dT = \frac{dT'}{\eta_p} = \frac{T'-T}{\eta_p} = \frac{T\left(\frac{T'}{T}-1\right)}{\eta_p}$$

$$\frac{dT}{T} = \frac{\left[\left(\frac{p+dp}{p}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta_p}$$

$$\frac{dT}{T} = \frac{1}{\eta_p}\left[\left(1+\frac{dp}{p}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

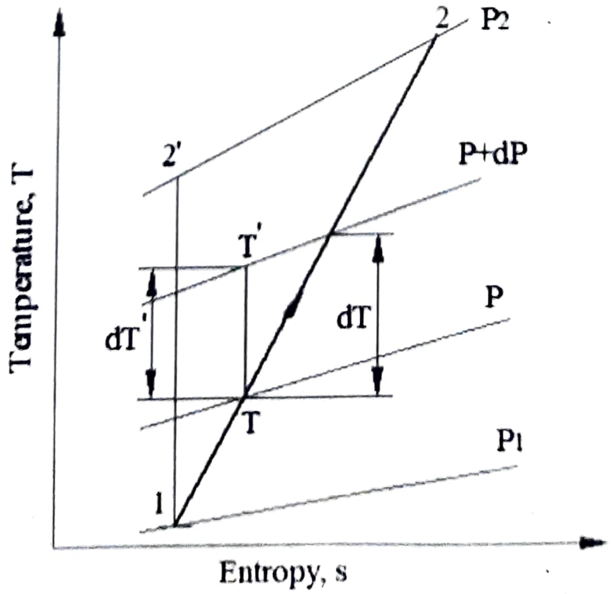

Fig. 3.5 Infinitesimal stage of a compressor

By series of expansion, $(1+x)^n = 1 + nx + \frac{n(n-1)}{2}x^2 + \cdots$ and neglecting second order differentials,

$$\frac{dT}{T} = \frac{1}{\eta_p}\left[1 + \frac{\gamma - 1}{\gamma}\frac{dp}{p} - 1\right]$$

$$\frac{dT}{T} = \frac{1}{\eta_p}\frac{\gamma - 1}{\gamma}\frac{dp}{p} \tag{3.3}$$

By integration with limits 1 to 2,

$$ln\left(\frac{T_2}{T_1}\right) = \frac{1}{\eta_p}\frac{\gamma - 1}{\gamma} ln\left(\frac{p_2}{p_1}\right)$$

$$\boldsymbol{\eta_p = \frac{\frac{\gamma - 1}{\gamma} ln\left(\frac{p_2}{p_1}\right)}{ln\left(\frac{T_2}{T_1}\right)}}$$

Question No 3.9: With the help of T-s diagram, show that polytropic efficiency during the compression process is given by $\eta_p = \left(\frac{\gamma-1}{\gamma}\right)\left(\frac{n}{n-1}\right)$(VTU, Jun/Jul-13)

Answer: From equation (3.3),

$$\frac{dT}{T} = \frac{1}{\eta_p}\frac{\gamma - 1}{\gamma}\frac{dp}{p}$$

By integration,

$$ln(T) = \frac{1}{\eta_p}\frac{\gamma - 1}{\gamma} ln(p) + Const$$

$$\frac{p^{\frac{\gamma-1}{\eta_p \gamma}}}{T} = Const$$

For actual compression process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_p \gamma}}$$

Assume actual compression process having polytropic index '*n*',

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

Therefore,

$$\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_p \gamma}} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

Equating the indices,

$$\frac{\gamma - 1}{\eta_p \gamma} = \frac{n-1}{n}$$

Or,

$$\eta_p = \left(\frac{\gamma - 1}{\gamma}\right)\left(\frac{n}{n-1}\right)$$

Question No 3.10: Derive an expression for stage efficiency of a compressor in terms of stage pressure ratio, polytropic efficiency and ratio of specific heats. Indicate the process on T-s diagram. (VTU, Dec-12) Or,

With the help of T-s diagram, show that stage efficiency of compressor is given by

$$\eta_{st} = \frac{P_r^{\frac{\gamma-1}{\gamma}} - 1}{P_r^{\frac{\gamma-1}{\eta_p\gamma}} - 1}$$

Answer: From the T-s diagram shown in figure 3.5, the compressor stage efficiency is given by,

$$\eta_{st} = \frac{Isentropic\ Temperature\ Rise}{Actual\ Temperature\ Rise}$$

$$\eta_{st} = \frac{T_{2'} - T_1}{T_2 - T_1} = \frac{T_1\left(\frac{T_{2'}}{T_1} - 1\right)}{T_1\left(\frac{T_2}{T_1} - 1\right)} = \frac{\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\eta_p\gamma}} - 1\right]}$$

Let, $p_r = \frac{p_2}{p_1}$

$$\eta_{st} = \frac{P_r^{\frac{\gamma-1}{\gamma}} - 1}{P_r^{\frac{\gamma-1}{\eta_p\gamma}} - 1}$$

3.4.4 Multistage Compressors:

Question No 3.11: Derive an expression for an overall isentropic efficiency for multistage compression in terms of pressure ratio, polytropic efficiency, number of stages and ratio of specific heats for a compressor. Or,

Show that for a multistage compression the overall isentropic efficiency is given by

$$\eta_o = \frac{P_r^{K\frac{\gamma-1}{\gamma}} - 1}{P_r^{K\frac{\gamma-1}{\eta_p\gamma}} - 1}$$

Where K= number of stages, P_r= pressure ratio per stage, η_p= polytropic efficiency, γ= ratio of specific heats.

Answer: The figure 3.6 shows the T-s diagram for compression process in multistage compressor operating between the pressures p_1 and p_{K+1}. If there are K stages with the overall pressure ratio $\frac{p_{K+1}}{p_1}$ and having equal stage efficiency and stage pressure ratio.

The overall efficiency of the multistage compressor is,

$$\eta_o = \frac{Total\ Isentropic\ Temperature\ Rise}{Total\ Actual\ Temperature\ Rise}$$

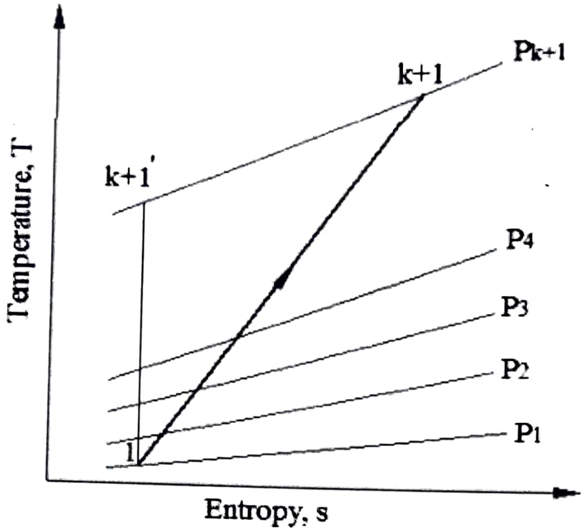

Fig. 3.6 Compression process in multistage compressor

$$\eta_o = \frac{T_{(K+1)'} - T_1}{T_{K+1} - T_1} = \frac{T_1\left(\frac{T_{(K+1)'}}{T_1} - 1\right)}{T_1\left(\frac{T_{K+1}}{T_1} - 1\right)}$$

$$\eta_o = \frac{P_{ro}^{\frac{\gamma-1}{\gamma}} - 1}{P_{ro}^{\frac{\gamma-1}{\eta_p \gamma}} - 1}$$

The overall pressure ratio can be written as, $p_{ro} = p_r^K$

Then overall efficiency of multistage compressor is,

$$\eta_o = \frac{P_r^{K\frac{\gamma-1}{\gamma}} - 1}{P_r^{K\frac{\gamma-1}{\eta_p \gamma}} - 1}$$

Question No 3.12: Derive an expression for an overall isentropic efficiency for finite number of stages of compression in terms of pressure ratio, stage efficiency, number of stages and ratio of specific heats for a compressor. **(VTU, May/Jun-10)** ***Or, Show that for a finite number of stages for compression the overall isentropic efficiency is given by***

$$\eta_o = \frac{P_r^{K\frac{\gamma-1}{\gamma}} - 1}{\left[1 + \frac{P_r^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{st}}\right]^K - 1}$$

Where K= number of stages, P_r= pressure ratio per stage, η_{st}= stage efficiency, γ= ratio of specific heats. (VTU, Jan/Feb-06)

Answer: If T_1 is the initial temperature at which the fluid enters the multistage compressor, K is the number of stages having equal pressure ratio p_r in each stage, then the actual temperature rise in each stage can be given as follows:

For first stage:

$$\Delta T_1 = (T_2 - T_1) = \frac{(T_{2'} - T_1)}{\eta_{st}} = \frac{T_1\left(\frac{T_{2'}}{T_1} - 1\right)}{\eta_{st}} = T_1 \frac{P_r^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{st}}$$

Let $A = \frac{P_r^{\frac{\gamma-1}{\gamma}} - 1}{\eta_{st}}$

$$\boldsymbol{\Delta T_1 = AT_1}$$

For second stage:

$$\Delta T_2 = (T_3 - T_2) = AT_2 = A(T_1 + AT_1)$$
$$\boldsymbol{\Delta T_2 = AT_1(1 + A)}$$

For third stage:

$$\Delta T_3 = (T_4 - T_3) = AT_3 = A[T_2 + AT_1(1 + A)]$$
$$\Delta T_3 = A[(T_1 + AT_1) + AT_1(1 + A)] = A[T_1(1 + A) + AT_1(1 + A)]$$
$$\Delta T_3 = A[(1 + A)(T_1 + AT_1)] = AT_1[(1 + A)(1 + A)]$$
$$\boldsymbol{\Delta T_3 = AT_1(1 + A)^2}$$

Similarly for fourth stage:

$$\boldsymbol{\Delta T_4 = AT_1(1 + A)^3}$$

And for K^th^ stage:

$$\boldsymbol{\Delta T_K = AT_1(1 + A)^{K-1}}$$

Total temperature rise across the multistage compressor is:

$$\Delta T_o = \Delta T_1 + \Delta T_2 + \Delta T_3 + \Delta T_4 + \cdots + \Delta T_K$$
$$\Delta T_o = AT_1 + AT_1(1 + A) + AT_1(1 + A)^2 + AT_1(1 + A)^3 + \cdots + AT_1(1 + A)^{K-1}$$
$$\Delta T_o = AT_1[1 + (1 + A) + (1 + A)^2 + (1 + A)^3 + \cdots + (1 + A)^{K-1}]$$
$$\Delta T_o = AT_1 S$$

Where

$$\boldsymbol{S = 1 + (1 + A) + (1 + A)^2 + (1 + A)^3 + \cdots + (1 + A)^{K-1}}$$
$$S = 1 + (1 + A)[1 + (1 + A) + (1 + A)^2 \ldots + (1 + A)^{K-2}]$$

Or,

$$S = 1 + (1 + A)[1 + (1 + A) + (1 + A)^2 \ldots + (1 + A)^{K-2} + (1 + A)^{K-1} - (1 + A)^{K-1}]$$
$$S = 1 + (1 + A)[S - (1 + A)^{K-1}]$$
$$S = 1 + S(1 + A) - (1 + A)^K = 1 + S + SA - (1 + A)^K$$
$$\boldsymbol{SA = (1 + A)^K - 1}$$

BGS Institute of Technology
Department: Mechanical Engineering
Assignment – I

USN:

Semester:V

Subject Name and Code: Turbo Machine (17ME53)

Max. Marks: 10

INSTRUCTIONS:

1. Answer all the question in yellow book

Q. No	QUESTIONS	CO	L	M
1.	With a neat sketch explain the parts of a turbo machine and also give its classifications.	2	2	1
2.	Differentiate between Turbo machine and Positive displacement machine.	1	2	1
3.	Derive the expression for pi-terms for the performance of a turbo machine.	1	2	1
4.	Derive the expression for specific speed of turbine and pump.	1	2	1
5.	Discuss the effect of Reynolds's number and also write the Moody's equation.	1	2	1
6.	Define first and second fan law.	1	2	1
7.	Discuss the first and second law of thermodynamics	1	2	1
8.	The following data were from the main t=characteristic of a Kaplan turbine of a runner diameter 1m. Pu=30.695, Qu=108.6, Nu= 63.6. Estimate i. the runner diameter II. the discharge III. the speed of a runner working under head of a 30m and developing 2000kW.Also IV. Determine specific speed of the runner.	6	3	1
9.	A Francis turbine model is built to scale 1:5 the data for model is P=4kW, N=3500rpm, H=2m and for prototype H=6m, assume overall efficiency of the model is 70%. Calculate (a) speed of the prototype (b) power of the prototype. Use Moody's equation.	6	3	1
10.	A turbine model 1:10 develops 2kW under head of 6m at 500rpm. Find the power developed by the prototype under head of 40m. Also find the speed of prototype and its specific speed. Assume the turbine efficiencies to remain same	6	4	1

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BGS Institute of Technology
Department: Mechanical Engineering
Assignment – II

USN:

Semester:V

Subject Name and Code: Turbo Machine (17ME53)

INSTRUCTIONS:

Max. Marks: 10

1. Answer all the question in yellow book

Q. No	QUESTIONS	CO	L	M
1.	Why compounding is necessary. Explain the different types of compounding with neat sketches.	4	2	1
2.	Distinguish between an impulse and reaction steam turbines.	2	2	1
3.	Explain the following (i)Nozzle efficiency (ii) diagram efficiency (iii) stage efficiency.	5	2	1
4.	Derive the expression for maximum blade efficiency of an impulse turbine.	4	3	1
5.	Derive the expression for maximum blade efficiency of an Parson's reaction turbine.	4	2	1
6.	Define hydraulic turbine. Give the complete classification of hydraulic turbines.	5	2	1
7.	With the reference to hydraulic turbine, define (i)hydraulic efficiency (ii) mechanical efficiency (iii) volumetric efficiency (iv) overall efficiency.	2	2	1
8.	Derive the expression for maximum hydraulic efficiency of a Pelton wheel. Also draw the inlet and exit velocity triangles.	5	3	1
9.	Dry saturated steam at 10atm pressure is supplied to a single rotor impulse wheel, the condenser pressure being 0.05 atm with the nozzle efficiency of 0.94 and the nozzle angle at the rotor inlet is 18° to the wheel plane. The rotor blades which move the speed of 450m/s are equiangular. If the co-efficient of velocity for rotor blades 0.92, find (i) The specific power output (ii) axial thrust (iii) the rotor efficiency (iv) the stage efficiency (v) the direction of steam exit.	5	3	1
10.	A double jet Pelton wheel is required to generate 7500kW when the available head at the base of nozzles is 400m. The jet is deflected through 165° and relative velocity of the jet is reduced by 15% in passage over the buckets. Determine (i) the diameter of each jet (ii) total flow (iii) force exerted by the jets in the tangential direction. Assume generator efficiency is 95%, η_o= 80%, speed ratio =0.47.	5	4	1

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Mandya

DEPARTMENT OF MECHANICAL ENGINEERING

III - OTHER ASSESSMENT (ASSIGNMENT)

Semester: 5-CBCS 2018

Subject: TURBO MACHINES (18ME53)

Faculty: Mr B L Keerthi

Max Marks: 10

Answer All Questions

Q.No		Marks	CO	BT/CL
1	Explain slip and slip coefficient in CF pump.	1	CO4	L2
2	What are the application of multistage CF pump? With a neat sketch explain the series and parallel.	1	CO4	L2
3	Explain the phenomenon of cavitation in CF pump. What are the causes ?? What are the steps to taken to avoid cavitation.	1	CO4	L3
4	Define CF pump. With usual notation derive the theoretical head capacity relationship.	1	CO4	L3
5	Define the following. (i) Manometric efficiency (ii) Mechanical efficiency (iii) Volumetric efficiency (iv) Overall efficiency.	1	CO5	L3
6	Explain with reference of CF pump (i) Net positive suction head (ii) Manometric head (iii)Need for priming.	1	CO4	L2
7	Derive the expression for pressure rise of an impeller of a CF pump	2	CO5	L3
8	Derive the expression for minimum starting speed of CF pump.	1	CO5	L3
9	Explain with neat sketch of CF pump with parts	1	CO4	L2

Head of the Department

Department of Mechanical Engineering

BGSIT B G Nagar-571448

USN :

BGS Institute of Technology, Mandya

DEPARTMENT OF MECHANICAL ENGINEERING

I - INTERNAL ASSESSMENT

Semester: 5-CBCS 2017
Subject: TURBO MACHINES (17ME53)
Faculty: Mr B L Keerthi

Date: 20 Sep 2019
Time: 09:30 AM - 10:30 AM
Max Marks: 30

Answer any 2 question(s)

Q.No			Marks	CO	BT/CL
1	a	Discuss the effect of Reynolds number and also write the Moody's equation	6	CO1	L2
	b	Define turbomachine. List the difference between positive displacement machine and turbomachine.	9	CO1,CO2	L2
		OR			
2	a	Define specific speed of pump. Derive the expression for specific speed of turbine	6	CO1,CO2	L2
	b	Test on a turbine runner 1.25m in diameter at 30m head gave the following results, the power developed is736kW, speed is 180rpm and discharge 2.7cubic meter per second. Find the diameter and discharge of a runner to operate at 45m head and gave 1472kW at the same efficiency. What is the specific speed of both turbines?	9	CO1	L3
3	a	Draw the velocity traingles at inlet and outlet of an axial flow turbine when (i) R=0 (ii) R=0.5	6	CO3	L3
	b	Derive the alternate forms of Euler's turbine equation and explain the significance of each energy component.	9	CO3	L3
		OR			
4	a	Define degree of reaction and show that the relatrion between utilization factor and degree of reaction for an axial flow turbine.	6	CO3	L3
	b	Derive an expression for Euler's energy for turbomachine.	9	CO3	L3

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BGSIT BG Nagara	Doc. Title: Internal Test Scheme Page 1 of 4 Date: 01.04.2018	Doc. No.: 06#Form#03 Rev. No. 00

CBCS Scheme (VTU)

DEPARTMENT: MECHANICAL ENGINEERING

Date 20.09.2019

Scheme & Solution - TEST - 1

Subject Code 17ME53

Semester VI   Subject Title: TURBOMACHINE

Question Number	Solution	Marks Allocated
1 (a)	$Re = \frac{\rho V D}{\mu} = \frac{VD}{\nu}$	01
	$Re < 2000$, Laminar Flow $Re > 4000$, Turbulent flow	01
	Explanation	03
	Moody's Equation $\eta_p = \left[1 - (1-\eta_m)\left(\frac{D_m}{D_p}\right)^{0.2}\right]$	01
		06
(b)	Definition	02
	Positive Displacement | Turbomachine * static | Dynamic * Volume changes | Pressure & momentum changes. * Unsteady flow | Steady flow * Low speed | High speed * complex in design | Simple in design. * Less volumetric efficiency | Nearly 100% * Low fluid handling capacity | High fluid handling capacity Any 7 difference	1×7 = 07
		09

Question Number	Solution	Marks Allocated
Q [a]	Specific Speed of pump Definition	01
	Power co-efficient $P/\rho N^3 D^5$	
	Head co-efficient $gH/N^2D^2 \Rightarrow D \propto \frac{H^{1/2}}{N}$	02
	$P \propto N^3 D^5$	
	$P = C\,H^{5/2}$, P = 1 kW, H = 1 m, N = Ns	02
	$N_{ST} = \frac{N P^{1/2}}{H^{5/2}}$	01
		06
[b]	**Data** model / prototype	
	Model: D = 1.25 m, H = 30 m, P = 736 kW, N = 180 rpm, Q = 2.7 m³/sec	
	Prototype: D = ?, H = 45 m, P = 1472 kW, Q = ?	
	$N_{ST} = ?$	
	$(N_{ST})_m = \frac{N_m P_m^{1/2}}{H_m^{5/4}} = \frac{180 \times \sqrt{736}}{(30)^{5/4}} = 69.55$	01
	$(N_{ST})_m = (N_{ST})_p \Rightarrow 69.55 = \frac{N_p \times (1472)^{1/2}}{(45)^{5/4}}$	02
	$N_p = 211.8$ rpm	
	$\eta_m = \frac{P_m}{(\rho g Q H)_m} = \frac{736 \times 10^3}{1000 \times 9.81 \times 2.7 \times 30} = 0.9262$	02
	$\eta_m = \eta_p \Rightarrow 0.9262 = \frac{1472 \times 10^3}{1000 \times 9.81 \times Q \times 45}$	02
	$Q_p = 3.6\ m^3/sec.$	
	$\left(\frac{Q}{ND^3}\right)_m = \left(\frac{Q}{ND^3}\right)_p$ (OR) $\left(\frac{gH}{N^2D^2}\right)_m = \left(\frac{gH}{N^2D^2}\right)_p$	
	$D_p^3 = \left(\frac{Q_p}{Q_m}\right)\left(\frac{N_m}{N_p}\right) D_m^3$	02
	$D_p = 1.3\ m$	
		09

Question Number	Solution	Marks Allocated

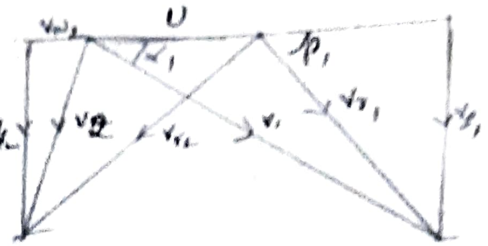
3 (a) For axial flow turbine $U_1 = U_2 = U$

(i) $R = 0$ [Impulse Turbine] — 03

$V_{r_2}^2 - V_{r_1}^2$

$V_{r_1} = V_{r_2}$

$\beta_1 = \beta_2$



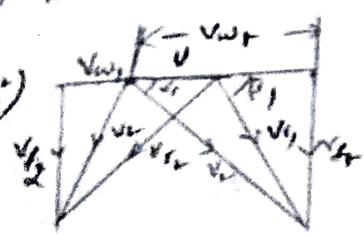
[02 + 01]

(ii) $R = 0.5$ [Parson Reaction Turbine] — 03

$$0.5 = \frac{(V_{r_2}^2 - V_{r_1}^2)}{(V_1^2 - V_2^2) + (V_{r_2}^2 - V_{r_1}^2)}$$

$V_1 = V_{r_2}, \quad V_2 = V_{r_1}$

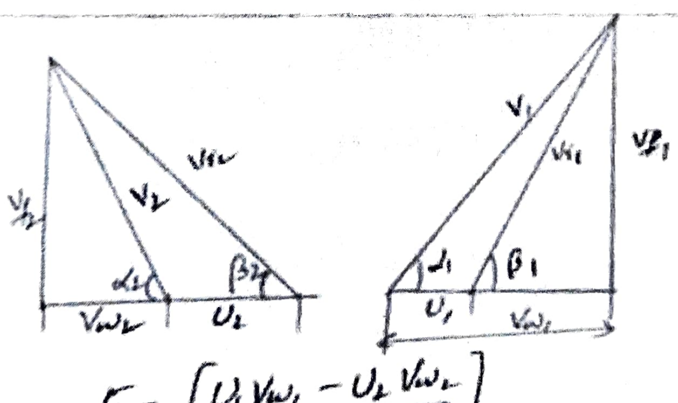
$\alpha_1 = \beta_2, \quad \beta_1 = \alpha_2$



[02 + 01]

06

(b)



02

01

$$E = \left[\frac{U_1 V_{w_1} - U_2 V_{w_2}}{g_c}\right]$$

From inlet vel Δ's, $U_1 V_{w_1} = \frac{1}{2}\left[V_1^2 + U_1^2 - V_{r_1}^2\right]$ — 02

$U_2 V_{w_2} = \frac{1}{2}\left[V_2^2 + U_2^2 - V_{r_2}^2\right]$

01

$$E = \frac{1}{2g_c}\left[V_1^2 + U_1^2 - V_{r_1}^2 - V_2^2 - U_2^2 + V_{r_2}^2\right]$$

$$E = \frac{(V_1^2 - V_2^2) + (U_1^2 - U_2^2) + (V_{r_2}^2 - V_{r_1}^2)}{2g_c}$$

03

Significance of each term 1×3

09

Question Number	Solution	Marks Allocated
4 (a)	Degree of Reaction : $R = \frac{\text{Static head}}{\text{Total head}}$	01
	At Axial flow turbine $U_1 = U_2 = U$	
	$\therefore R = \frac{(V_{r_2}^2 - V_{r_1}^2)}{(V_1^2 - V_2^2) + (V_{r_2}^2 - V_{r_1}^2)}$	01
	$(V_{r_2}^2 - V_{r_1}^2) = (R/1-R)(V_1^2 - V_2^2)$	01
	ε = Utilization factor $= \frac{(V_{r_2}^2 - V_{r_1}^2) + (V_1^2 - V_2^2)}{V_1^2 + (V_{r_2}^2 - V_{r_1}^2)}$	01
	$\varepsilon = \frac{\left(\frac{R}{1-R}\right)(V_1^2 - V_2^2) + (V_1^2 - V_2^2)}{V_1^2 + (R/1-R)(V_1^2 - V_2^2)}$	01
	$\varepsilon = \frac{V_1^2 - V_2^2}{V_1^2 - RV_2^2}$	01
		06
4 (b)	Euler's Turbine Equ.	
	Sketch	02
	$F = \frac{m_1}{g_c t} V_{w_1} - V_{w_2}$	01
	$F = \frac{m}{g_c}(V_{w_1} - V_{w_2})$	01
	$T = F \times r = \frac{m}{g_c}(V_{w_1} r_1 - V_{w_2} r_2)$	01
	$E = \omega \frac{m}{g_c}(V_{w_1} r_1 - V_{w_2} r_2)$	01
	$\omega \times r = U$ = Tangential velocity	
	$E = \frac{m}{g_c}(V_{w_1} U_1 - V_{w_2} U_2)$	01
	$E = \frac{m}{g_c}[U_1 V_{w_1} - U_2 V_{w_2}]$	01
	$E = \frac{E_0}{m}$ = Energy transfer in J/kg.	
	$E_0 = \left[\frac{U_1 V_{w_1} - U_2 V_{w_2}}{g_c}\right] = \Delta h_0$	01
		09

/

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DEPARTMENT OF MECHANICAL ENGINEERING

II - INTERNAL ASSESSMENT

Semester: 5-CBCS 2017	Date: 25 Oct 2019
Subject: TURBO MACHINES (17ME53)	Time: 09:30 AM - 10:30 AM
Faculty: Mr B L Keerthi	Max Marks: 30

<u>*Answer any 2 question(s)*</u>

Q.No			Marks	CO	BT/CL
1	a	Why compounding is necessary. With a neat sketch explain the pressure compounding.	6	CO4	L2
	b	Derive an expression for maximum blade efficiency a Parson's reaction turbine.	9	CO6	L3
		OR			
2	a	Define hydraulic turbines. Give the complete classification with examples. Define hydraulic and overall efficiency.	6	CO2	L2
	b	The penstock supplies water from a reservoir to the Pelton wheel with a gross head of 500m. One third head is lost due to frication in penstock. The rate of flow of water through the nozzle at the end of penstock is 2 cubic meter/sec. The angle of deflection of the jet is 165° . Determine the power given by the water to the runner and also hydraulic efficiency. take Cv=1, Speed ratio=0.45.	9	CO6	L3
3	a	What is draft tube. Discuss the functions of draft tubes and also give its classification	6	CO4	L2
	b	In a Francis turbine, the discharge is radial the blade speed at the inlet is 25m/s. At the inlet the tangential component of velocity is 18m/s. The radial velocity of flow is constant and equal to 2.5m/s. Water flows at a rate of 0.8 cubic meter/sec. The utilization factor is 0.82. Find(i) Euler's head (ii) power developed (iii) inlet blade angle (iv) degree of reaction. Draw velocity triangles.	9	CO6	L3
		OR			
4	a	Distinguish between impulse and reaction turbine.	6	CO2	L2
	b	Steam issuing from a nozzle to a De-Laval turbine with a velocity of 1000m/s. The nozzle is 20°, the mean blade speed is 400m/s. The blades are symmetrical, the mass flow rate is 1000 kg/hr, friction factor=0.8, nozzle efficiency =0.95. calculate (i) The blade angles (ii) axial thrust (iii) work done per kg of steam (iv) power developed (v) blade efficiency (vi) stage efficiency.	9	CO5	L3

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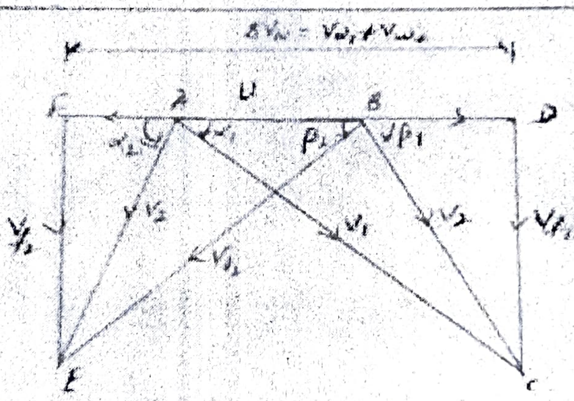
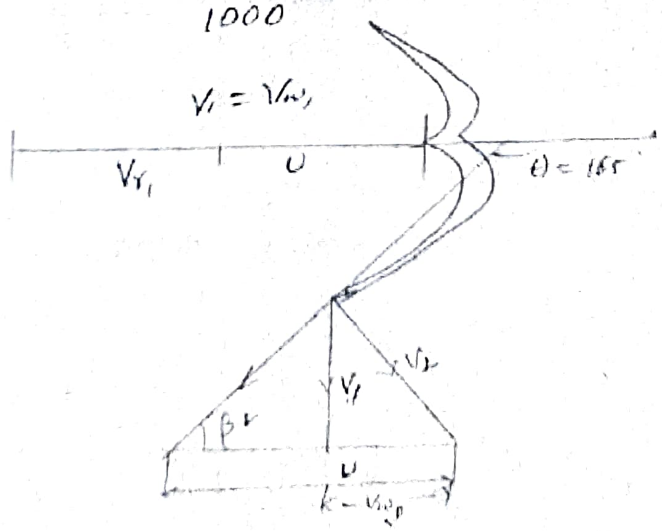
CBCS Scheme (VTU)

DEPARTMENT: MECHANICAL ENGINEERING

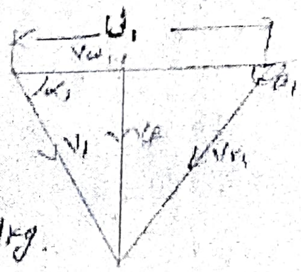
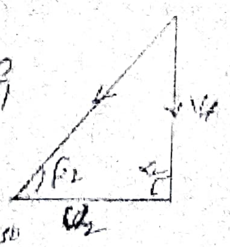
Scheme & Solution – TEST – II — **Date**: 25.10.2019

Semester: V — **Subject Title**: TURBOMACHINE — **Subject Code**: 17ME53

Question Number	Solution	Marks Allocated
1. a	Necessity of compounding	2M
	PV Diagram	2M
	Explanation	2M
		6M
b		
	For Parson's reaction turbine, R = 0.5	
	$\alpha_1 = \beta_2 \Rightarrow V_1 = V_{r_2}$	02
	$\alpha_2 = \beta_1 \Rightarrow V_2 = V_{r_1}$	
	$W = \frac{U \Delta V_w}{g_c} = \frac{U}{g_c}[V_1 \cos\alpha_1 + V_2 \cos\alpha_2]$	01
	$W = \frac{2UV_1 \cos\alpha_1 - U^2}{g_c} = \frac{V_1^2}{g_c}(2\phi \cos\alpha_1 - \phi^2)$	01
	$\Delta h = \frac{V_1^2}{2g_c} + \frac{V_{r_2}^2 - V_{r_1}^2}{2g_c}$	01
	$\Delta h = \frac{V_1^2}{2g_c}[1 + 2\phi \cos\alpha_1 - \phi^2]$	01
	$\eta_b = \frac{W}{\Delta h} = \frac{2\phi(\cos\alpha_1 - \phi^2)}{(1 + 2\phi \cos\alpha_1 - \phi^2)}$	01
	$\frac{\partial \eta_b}{\partial \phi} = 0 \Rightarrow \phi = U/V_1 = \cos\alpha_1$	01
	$\eta_{b_{max}} = \frac{2\cos^2\alpha_1}{1 + \cos^2\alpha_1}$	01
		4M

Question Number	Solution	Marks Allocated
2 (a)	Definition	01
	Classification with examples	03
	Hydraulic Efficiency $= \frac{RP}{WP} = \frac{P}{\rho g Q H}$	01
	Overall efficiency $= \frac{SP}{WP} = \frac{SP}{RP} \times \frac{RP}{WP} = \eta_{mech} \times \eta_H$	01
		06
(b)	$H = H_g - h_f = H_g - \frac{1}{3} H_g = 333.3\,m$, $\beta_2 = 15^{\circ}$	01
	$U = \phi \sqrt{2gH} = 36.38\,m/s$ $V_1 = C_v \sqrt{2gH} = 80.86\,m/s$ $V_{r_1} = V_1 - U = 44.473\,m/s$ $V_{r_2} = V_{r_1} = 44.43\,m/s$ $V_{w2} = V_{r_2} \cos\beta_2 - U = 6.57\,m/s$	03
	$P = \frac{\rho Q U (V_{w_1} + V_{w_2})}{1000} = 6362.63\,kW$	02
		01
	$\eta_H = \frac{2U(V_{w_1} + V_{w_2})}{V_1^2} = 97.37$	02
		09

#03-0

Question Number	Solution	Marks Allocated
3 (a)	Draft tube	01
	Functions	02
	Classification	
	a) Simple elbow tube	
	b) Moody spreading tube	03
	c) Straight divergent tube	
	d) Elbow with circular c/s	
		06
(b)	Radial discharge $V_{w_2} = 0$, $\alpha_2 = 90°$, $V_2 = V_f$	
	$U_1 = 25 m/s$, $V_{w_1} = 18 m/sec$, $V_f = 2.5 m/s = V_2$	01
	$Q = 0.8 m^3/s$, $\varepsilon = \eta_H = 0.82$	01
	Vel d/e 	
	Euler's head	
	$E = W = \frac{U_1 V_{w_1} + U_2 V_{w_2}}{g_c}$	
	$= \frac{25 \times 18}{1} = 450 J/kg$	
	$E = g He = 450$ 	
	$He = \frac{450}{g} = \frac{450}{9.81}$	02
	$He = 45.87m$	
	$P = \frac{\rho Q W}{1000 g_c} = \frac{1000 \times 0.8 \times 450}{1000}$	02
	$P = 360 kW$	
	$\tan \beta_1 = \frac{V_f}{(U_1 - V_{w_1})}$	
	$V_1 = \sqrt{V_f^2 + V_{w_1}^2} = 18.17 m/sec$	01
	$U_1 > V_{w_1}$ $\beta_1 = 19.650$ (-ve)	
	$\tan(180 - \beta_1) = \frac{V_f}{(U_1 - V_{w_1})}$ ⇒ $\beta_1 = 160.35$ (-ve)	
	$R = \frac{E - \frac{(V_1^2 - V_2^2)}{2g}}{E} = 0.64$	02
		09

Question Number	Solution	Marks Allocated
4 (a)	Any 6 difference 1×6	6M
(b)	$V_1 = 1000\ m/s$, $U = 400\ m/s$, $\beta_1 = \beta_2$	
	$m_s = 1000\ kg/hr = 0.278\ kg/s$, $K = \frac{V_{r2}}{V_{r1}} = 0.8$	
	$\eta_n = 0.95$	
	From Graphical method, Velocity d/c	02
	$V_{r1} = 640\ m/s$, $V_{r2} = 51\ m/sec$	
	$\Delta V_w = V_{w1} + V_{w2} = 971.2\ m/sec$	
	$\Delta V_f = V_{f1} - V_{f2} = 68.2\ m/sec$	
	$\beta_1 = \beta_2 = 32.4°$	02
	$F_a = \frac{m_s(V_{f1} - V_{f2})}{g_c} = 19\ N$	01
	$W = \frac{U \Delta V_w}{1000\ \eta_n} = 388.5\ kJ/kg$	01
	$P = m_s W = 108\ kW$	01
	$\eta_b = \frac{2 U \Delta V_w}{V_1^2} = 77.7\%$	01
	$\eta_{stage} = \eta_b \times \eta_{noz} = 73.8\%$	01
		09

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DEPARTMENT OF MECHANICAL ENGINEERING
III - INTERNAL ASSESSMENT

Semester: 5-CBCS 2017
Subject: TURBO MACHINES (17ME53)
Faculty: Mr B L Keerthi

Date: 22 Nov 2019
Time: 09:30 AM - 10:45 AM
Max Marks: 30

PART A

Answer any1 question(s)

Q.No			Marks	CO	BT/CL
1	a	Derive an expression for minimum starting speed of a centrifugal pump.	6	CO4	L3
	b	What are the applications of multistgae centrifugal pump? With a neat sketch explain centrifugal pumps in series and parallel.	9	CO4	L2
2	a	What is the cavitation? What are its effects?	6	CO4	L2
	b	Explain with a neat sketch , multistage centrifugal pump arrangement. Explain NPSH	9	CO4	L2

PART B

Answer any1 question(s)

Q.No			Marks	CO	BT/CL
3	a	With a neat sketch show the sectional arrangment of Kaplan turbine.	6	CO4	L2
	b	A propeller turbine has a outer diameter of 4.5m and inner diameter 2m. It develops 20605 kW under an head of 20m at 173rpm, the hydraulic efficiency is 0.94 , overall effciency is 0.88. Find the runner blade angles and discharge through the runner. Draw the velocity traingle.	9	CO6	L3
4	a	Exaplain the phenomenon of slip and slip factor. Define manometric head of a centrifugal pump.	6	CO4	L2
	b	A CF pump having outer diameter equal to 2 times the inner diameter and running at 1000rpm, working under a head of 30m. The velocity flow through the impeller is constant and equal to 2.5m/s. The vanes are set back at an angle 40° at the outlet. If the outer diameter is 50cm and width at outlet is 5cm, calculate (i) Vane angle at inlet (ii) Workdone by the impeller on water/ second (iii) manometric efficiency.	9	CO6	L3

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CBCS Scheme (VTU)

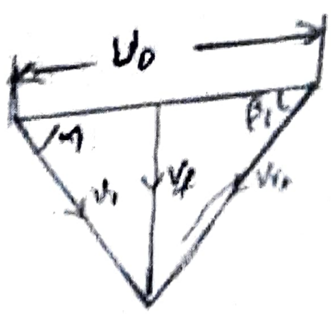
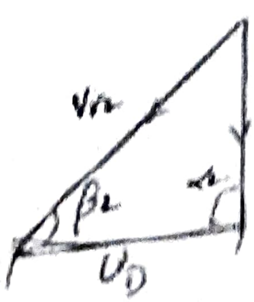
DEPARTMENT: MECHANICAL ENGINEERING

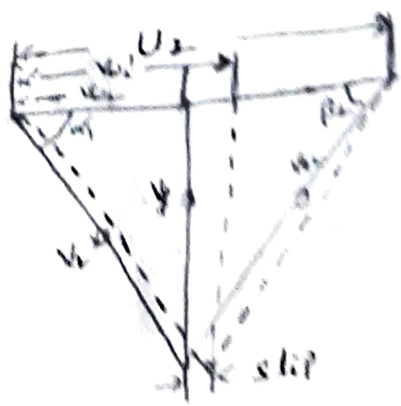
Scheme & Solution – TEST – III

Semester: V | Date: 22.11.2018

Subject Title: TURBOMACHINE | Subject Code: 17ME53

Question Number	Solution	Marks Allocated
1 (a)	The flow will commence only if	
	$\frac{U_2^2 - U_1^2}{2g} \geq H_m$	01
	$\frac{U_2^2 - U_1^2}{2g} = H_m = \eta_{mano} \times H_e$	01
	$\left(\frac{\pi N}{60}\right)^2 \times \left(\frac{D_2^2 - D_1^2}{2g}\right) = \eta_{mano} \times H_e$	02
	$N_{min} = \left(\frac{60}{\pi}\right)\left[\frac{2g \times \eta_{mano} \times H_e}{D_2^2 - D_1^2}\right]^{1/2}$ rpm	02
		06
(b)	Multi stage pumps	01
	Pumps in series – Sketch	02
	– Explanation	02
	pumps in parallel – Sketch	02
	– Explanation	02
		09
2 (a)	Cavitation	03
	Effects	03
		06
(b)	centrifugal pump	

Question Number	Solution	Marks Allocated
	Sketch – C.F pump	03
	– Explanation	03
	NPSH	02
	$NPSH = \frac{P_s}{\rho g} + \frac{V_s^2}{2g} - \frac{P_{vap}}{\rho g}$	01
		09
3 (a)	Kaplan turbine – Sketch	03
	– Explanation	03
		06
(b)	 I/L	02
	 $V_f = V_2$ O/L	

Question Number	Solution	Marks Allocated
	$U_D = \frac{\pi D N}{60} = 40.76$ m/sec $= U_1$	01
	$\eta_{overall} = \frac{SP}{\rho g Q H}$	
	$Q = \frac{20605 \times 1000}{0.88 \times 9.81 \times 1000 \times 20} = 119.34$ m³/sec	01
	$Q = \frac{\pi}{4}(D^2 - d^2) V_f$	
	$V_f = \left[\frac{4 \times 119.34}{(4.5^2 - 2^2)\pi}\right] = 9.35$ m/sec	01
	$\eta_H = \frac{(U_1 V_{w_1}/g_c)}{(gH/g_c)} = \frac{U_1 V_{w_1}}{gH}$	
	$V_{w_1} = \frac{0.94 \times 9.81 \times 20}{40.76} = 4.52$ m/sec	01
	$\tan\beta_1 = \frac{V_f}{U_1 - V_{w_1}} \Rightarrow \beta_1 = 14.66°$	01
	$\beta_2 = \tan^{-1}(V_f/U) = 12.91°$	01
	$\alpha_1 = \tan^{-1}\left(\frac{V_f}{V_{w_1}}\right) = 64.19°$	01
		07
4 (a)	Slip	01
	Sketch 	02
	Explanation	
	Manometric head	02

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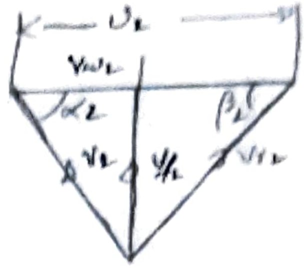
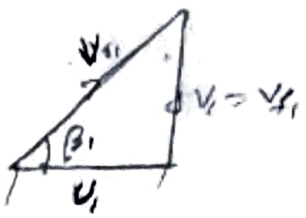
Question Number	Solution	Marks Allocated
	Slip = $H_e' - H_e$	
	Slip Co-efficient = $\mu = \frac{H_e}{H_e'}$	02
		06
4 (b)		01
		
	$U_2 = 26.18$ m/sec	01
	$U_1 = 13.09$ m/sec	
	$\tan\beta_1 = \frac{V_f}{U_1} \Rightarrow \beta_1 = 10.8$	01
	$Q = \pi D_2 B_2 V_f = 0.196$ m³/sec	01
	$m_g = \rho Q = 196$ kg/sec	
	$V_{w_2} = U_2 - \frac{V_f}{\tan\beta_2} = 23.2$ m/sec	01
	$W = \frac{U_2 V_{w_2}}{g_c} = 607$ J/kg $= gH_e$	01
	$P = 119$ kW	01
	$H_e = \frac{607}{9.81} = 61.88$ m	01
	$\eta_{mano} = \frac{H_m}{H_e} = \frac{30}{61.81} = 48.48\%$	01
		09

17ME53

USN

Fifth Semester B.E. Degree Examination, Dec.2019/Jan.2020
Turbomachines

Time: 3 hrs. Max. Marks: 100

Note: Answer any FIVE full questions, choosing ONE full question from each module.

Module-1

1 a. Define Turbomachine. With neat sketch, explain the parts of Turbomachine. (04 Marks)

b. Define specific speed of pump. Derive an expression for the same in terms of discharge speed and head. (06 Marks)

c. A Fransis turbine model is built to scale 1:5 the data for the model is P = 4kW, N = 3500rpm H = 2m and prototype H = 6m. Assume that the overall efficiency of the model as 70%. Calculate: i) Speed of the prototype ii) Power of the prototype. Use Moody's equation. (10 Marks)

OR

2 a. Define Polytropic Efficiency of turbine. Show that the Polytropic Efficiency during Expansion process is given by $\eta_p = \dfrac{\ln\dfrac{T_2}{T_1}}{\dfrac{\gamma-1}{\gamma}\ln\dfrac{P_2}{P_1}}$ (10 Marks)

b. In a three stage turbine the pressure ratio of each stage is 2 and stage efficiency is 0.75. Calculate overall efficiency and reheat factor. (10 Marks)

Module-2

3 a. Derive alternate form of Euler's turbine equation and explain the significance of each energy component. (10 Marks)

b. At a 50% reaction stage axial flow turbine the mean blade diameter is 0.6mts. The maximum utilization factor is 0.85 and steam flow rate is 12kg/sec. Calculate the inlet and outlet absolute velocities and power developed if the speed is 2500rpm. (10 Marks)

OR

4 a. In a turbomachine prove that the maximum utilization factor is given by

$\epsilon_{max} = \dfrac{2\phi\cos\alpha_1}{1+2\phi R\cos\alpha_1}$ where ϕ = speed ration, R = degree of reaction, α_1 = nozzle angle. (10 Marks)

b. Draw the velocity triangles at inlet and outlet of an axial flow compressor from the following data. Degree of reaction 0.5 inlet blade angle 45°. Axial velocity of flow which is constant throughout 120m/sec, speed of rotation 6500rpm, radius of rotation 20cm, blade speed of inlet is equal to blade speed at outlet. Calculate angles at inlet and outlet. Also calculate power needed to handle 1.5kg/s of air. (10 Marks)

Module-3

5 a. Why compounding of steam turbine necessary? Describe the velocity compounding of steam turbine with neat sketch. (08 Marks)

b. Show that for a two row Curtis steam turbine stage in the absence of friction for axial discharge at exit under maximum utilization condition $U/V_1 = \dfrac{\cos\alpha_1}{4}$ where U = blade speed V_1 = absolute velocity at inlet α_1 = nozzle angle at inlet. (12 Marks)

OR

6 a. Define degree of reaction for reaction turbine and derive an expression for the some for 50% reaction turbine. (10 Marks)

b. In a Parson's turbine, the axial velocity of flow of steam is 0.5 times the mean blade speed. The outlet angle of the blade is 20° diameter of the blade ring is 1.3m and rotational speed 3000rpm. Determine inlet blade angles, power developed for steam flow of 65kg/sec and isentropic enthalpy drop, if the stage efficiency is 80%. (10 Marks)

Module-4

7 a. Show that the specific speed of Pelton wheel is given by $n_s = 206.63 \frac{\sqrt{n}}{m}$ where n = number of jets used for the flow, m = wheel diameter to jet diameter ratio. Assume the jet velocity coefficient as 0.97 speed ratio as 0.45 and efficiency of the turbine as 0.89. (08 Marks)

b. A double overhung Pelton wheel unit is to produce 30000 kW of a generator under an effective head 300m at the base of the nozzle. Find the size of the jet. Mean diameter of the runner speed and specific speed of each Pelton turbine. Assume generator η = 93% Pelton wheel η = 0.85 speed ratio = 0.46 jet velocity co-coefficient = 0.97 and jet ratio = 12. (12 Marks)

OR

8 a. Draw a neat sketch of Francis turbine. Explain the function of draft tube. Also draw the typical velocity triangles of Fransis turbine. (08 Marks)

b. A Kaplan turbine working under head of 20m develops 11772kW of shaft power. The outer diameter of the runner is 3.5m and hub diameter is 1.75m. The guide blade angle of the extreme edge of the runner is 35°. The hydraulic and overall efficiencies of the turbine are 88% and 84% respectively. If the velocity of whirl is zero at outlet, determine: i) Runner vane angle at the inlet and outlet at the extreme edge of the runner ii) Speed of turbine. (12 Marks)

Module-5

9 a. Show that the pressure rise in the impeller of a centrifugal pump when the frictional and other losses in the impeller are neglected is given by $\Delta p = \frac{\rho}{2}\left[Vf_1^2 + U_2^2 - Vf_2^2 \operatorname{cosec}^2\beta_2\right]$.

Vf_1 and Vf_2 are velocity of flow at inlet and outlet of the impeller U_2 = tangential speed of impeller at exit, β_2 = exit blade angle. (10 Marks)

b. A centrifugal pump is running at 1000 rpm. The outlet vane angle of the impeller is 45° and the velocity of flow of the outlet is 2.5m/sec. The discharge through the pump is 0.2m^3/sec. When the pump is working against a head of 20m. If the monometric efficiency is 80% draw the outlet velocity diagram and calculate : i) The diameter of the impeller at the outlet ii) width of impeller at the outlet. (10 Marks)

OR

10 a. With reference to centrifugal air compressor, explain the following:
i) PreWhirl ii) Surging iii) Slip factor iv) Choking. (10 Marks)

b. A centrifugal compressor runs at a speed of 15000rpm and delivers air at 30kg/sec, exit radius is 0.35m, relative velocity and vane angles at exit are 100m/s and 75° respectively. Assuming axial inlet and inlet stagnation temperature and pressure as 300K and 1 bar respectively, calculate: i) Torque ii) The power required to drive compressor iii) The ideal head developed iv) The workdone v) The exit total pressure $(c_p)_{air}$ = 1.005kJ/kgK. (10 Marks)

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