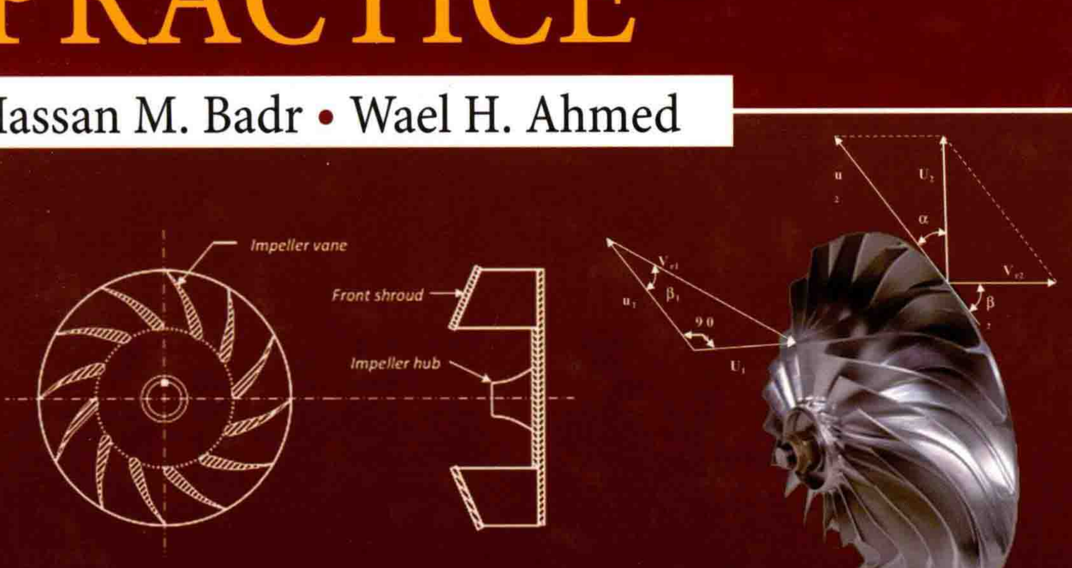
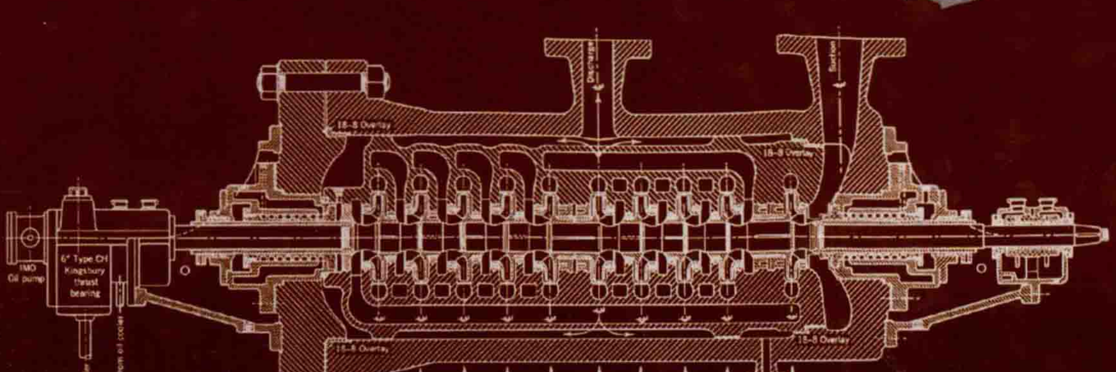
PUMPING MACHINERY THEORY AND PRACTICE

Hassan M. Badr • Wael H. Ahmed





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This edition first published 2015

Registered Office
John Wiley & Sons Ltd, The Atrium, Southern Gate, Chichester, West Sussex, PO19 8SQ, United Kingdom

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Library of Congress Cataloging-in-Publication Data

Badr, Hassan M.
Pumping machinery theory and practice / Hassan M. Badr, Wael H. Ahmed.
pages cm
Includes bibliographical references and index.
ISBN 978-1-118-93208-7 (hardback)
1. Pumping machinery. I. Ahmed, Wael H. II. Title.
TJ900.B193 2015
621.6–dc23

2014031896

A catalogue record for this book is available from the British Library.

Set in 10/12pt Times by SPi Publisher Services, Pondicherry, India
Printed in Singapore by C.O.S. Printers Pte Ltd

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To my parents, my dear wife and my children

Preface

Energy consumption in pumping systems accounts for approximately 20% of the world's electrical energy demand. Moreover, the operational cost of pumping machinery far outweighs their capital cost. Accordingly, engineers strive for optimum equipment performance for achieving economic operation. A thorough understanding of the components and principles of operation of these machines will provide an opportunity to dramatically reduce energy, operational and maintenance costs. Reducing energy consumption will also complement the current thrust towards protecting our environment.

This book is intended to be a basic reference on theoretical foundation and applications of various types of pumping machinery. In view of the great importance of pumps and compressors in almost every engineering system, this book presents the fundamental concepts underlying the flow processes taking place in these machines and the transformation of mechanical energy into fluid power. Special emphasis is given to basic theoretical formulation and design considerations of pumps and compressors in addition to improving problem-solving skills. This is achieved through the presentation of solved examples of applied nature using analytical means and/or basic engineering practices.

The book consists of ten chapters covering two main themes: the first smoothly introduces the essential terminology, basic principles, design considerations, and operational-type problems in pumping machinery. This part is supported by a good number of worked examples plus problems at the end of each chapter for the benefit of senior undergraduate students and junior engineers. This is considered a key feature of this book because other books in this area rarely provide enough worked problems and exercises. The second theme focuses on advanced topic such as two-phase flow pumping systems targeting practicing field engineers and introductory research scientists.

The authors wish to acknowledge their students' encouragement to write this book. The idea was initiated by the first author after searching for a good textbook for an undergraduate course in pumping machinery that he has been teaching for over 20 years. The absence of a suitable textbook demanded the preparation of a set of course notes to help the students to a better understanding of the subject. The support received from King Fahd University of Petroleum & Minerals under Grant # IN111025 for the preparation of this textbook is greatly appreciated.

Nomenclature

A	area
b	vane width
BP	brake power
C	speed of sound
C_c	contraction coefficient
C_H	head coefficient
C_P	power coefficient
C_Q	flow coefficient
D	impeller diameter
f	friction coefficient
g	gravitational acceleration
h	enthalpy
h_{ss}	static suction head
h_{sd}	static delivery head
h_s	suction head
h_d	delivery head
H	pump total head
$\underline{H}$	angular momentum
I	rotational enthalpy
k	specific heat ratio
K	loss coefficient for pipe fittings
L	length of connecting rod
$\dot{m}$	mass flow rate
M	Mach number
$\underline{M}$	moment
n_s	specific speed in SI
N_s	specific speed in the American system
$NPSH$	net positive suction head

N	speed of rotation in rpm
p	pressure
p_o	stagnation pressure
p_v	vapor pressure
P	power
Q	volume flow rate
r	crank radius
R	gas constant
S	suction specific speed
T	torque or temperature
T_u	unbalanced thrust
u	tangential velocity
v	flow velocity
V	whirl velocity component
V_r	relative velocity
x	coordinate
y	coordinate
Y	radial velocity component
z	elevation (measured from selected datum)

Greek Symbols

α	flow angle
β	vane angle
γ	specific weight of fluid
η	efficiency
λ	degree of reaction
μ	fluid viscosity
ν	fluid kinematic viscosity
θ	crank angle
ρ	fluid density
σ	Thoma's cavitation factor
ω	angular velocity

Subscripts

atm.	atmospheric
crit.	critical
d	discharge
e	Euler
f	friction
hyd.	hydraulic
L	leakage

mech.	mechanical
o	overall
r	relative
p	pressure
s	suction
sd	static delivery
sn	suction nozzle
ss	static suction
st	total static
u	unbalanced
v	vane
vol.	volumetric/volute
V	velocity

Contents

1

Essentials of Fluid Mechanics

The basic fundamentals of fluid mechanics are essential for understanding the fluid dynamics of pumping machinery. This chapter aims to provide a quick revision of the definitions and basic laws of fluid dynamics that are important for a thorough understanding of the material presented in this book. Of particular interest are the kinematics of fluid flow; the three conservation principles of mass, momentum, and energy; relevant dimensionless parameters; laminar and turbulent flows; and friction losses in piping systems. Some applications of relevance to pumping machinery are also considered.

1.1 Kinematics of Fluid Flow

To fully describe the fluid motion in a flow field it is necessary to know the flow velocity and acceleration of fluid particles at every point in the field. This may be a simple task in laminar flows but may be difficult in turbulent flows. If we use the Eulerian method and utilize Cartesian coordinates, the velocity vector at any point in a flow field can be expressed as

$$\underline{V} = u\underline{i} + v\underline{j} + w\underline{k} \tag{1.1}$$

where $\underline{V}$ is the velocity vector; u, v, and w are the velocity components in the x, y, and z directions; and $\underline{i}$, $\underline{j}$, and $\underline{k}$ are unit vectors in the respective directions. In general, each of the velocity components can be a function of position and time, and accordingly we can write

$$u = u(x, y, z, t), \quad v = v(x, y, z, t), \quad w = w(x, y, z, t) \tag{1.2}$$

The components of acceleration in the three directions can be expressed as

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$$a_x = \frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} \tag{1.3a}$$

$$a_y = \frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z} \tag{1.3b}$$

$$a_z = \frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z} \tag{1.3c}$$

The acceleration vector becomes

$$\underline{a} = a_x\underline{i} + a_y\underline{j} + a_z\underline{k} \tag{1.4}$$

This vector can be split into two components, the local component, $\underline{a}_{local}$, and the convective component, $\underline{a}_{conv.}$, that can be expressed as

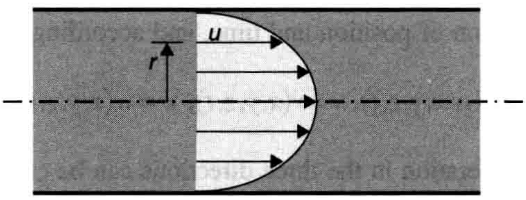
$$\underline{a}_{local} = \frac{\partial u}{\partial t}\underline{i} + \frac{\partial v}{\partial t}\underline{j} + \frac{\partial w}{\partial t}\underline{k} \tag{1.5a}$$

$$\underline{a}_{conv.} = \left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right)\underline{i} + \left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right)\underline{j} + \left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right)\underline{k} \tag{1.5b}$$

1.1.1 Types of Flows

The flow field can be described as *steady* or *unsteady*, *uniform* or *non-uniform*, *compressible* or *incompressible*, *rotational* or *irrotational*, *one-*, *two-*, or *three-dimensional*, and can also be described as *laminar* or *turbulent.* The flow is said to be steady if the velocity vector at any point in the flow field does not change with time.

Accordingly, the local component of acceleration ($\underline{a}_{local}$) vanishes if the flow is steady. The flow can also be described as uniform if the velocity vector does not change in the streamwise direction. For example, the pipe flow shown in Figure 1.1 is uniform since the velocity vector does not change downstream, but the flow in the bend shown in Figure 1.2 is non-uniform.



Laminar flow in a pipe

Figure 1.1 Laminar flow in a pipe as an example of uniform flow

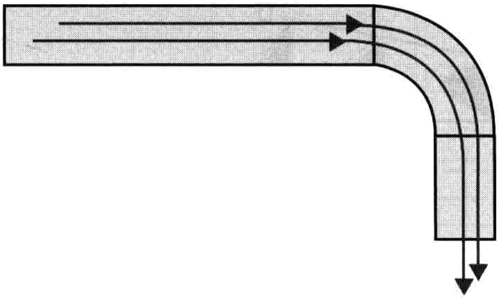


Figure 1.2 Flow in a 90° bend as an example of non-uniform flow

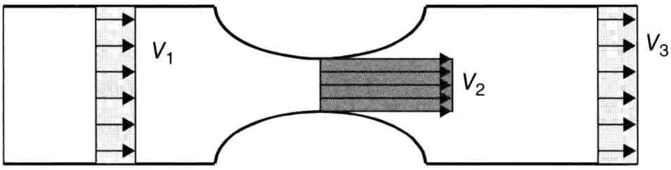


Figure 1.3 One-dimensional flow in a pipe with constriction

The flow is described as incompressible if the density change within the flow field does not exceed 5%. Accordingly, most of the flows in engineering applications are incompressible as, for example, flow of different liquids in pipelines and flow of air over a building. However, compressible flows occur in various applications such as flow in the nozzles of gas and steam turbines and in high speed flow in centrifugal and axial compressors. In general, the flow becomes compressible if the flow velocity is comparable to the local speed of sound. For example, the flow of air in any flow field can be assumed incompressible up to a Mach number of 0.3.

The flow is called one-dimensional (1-D) if the flow parameters are the same throughout any cross-section. These parameters (such as the velocity) may change from one section to another. As an approximation, we may call pipe or nozzle flows 1-D if we are interested in describing the average velocity and its variation along the flow passage. Figure 1.3 shows an example of 1-D flow in a pipe with constriction. On the other hand, the flow is called 2-D if it is not 1-D and is identical in parallel planes. For example, the viscous flow between the two diverging plates shown in Figure 1.4 is two-dimensional. In this case, two coordinates are needed to describe the velocity field.

If the flow is not 1-D or 2-D, it is then three-dimensional. For example, flow of exhaust gases out of a smoke stack is three-dimensional. Also, air flow over a car or over an airplane is three-dimensional.

1.1.2 Fluid Rotation and Vorticity

The rate of rotation of a fluid element represents the time rate of the angular displacement with respect to a given axis. The relationship between the velocity components and the rate of rotation can be expressed as

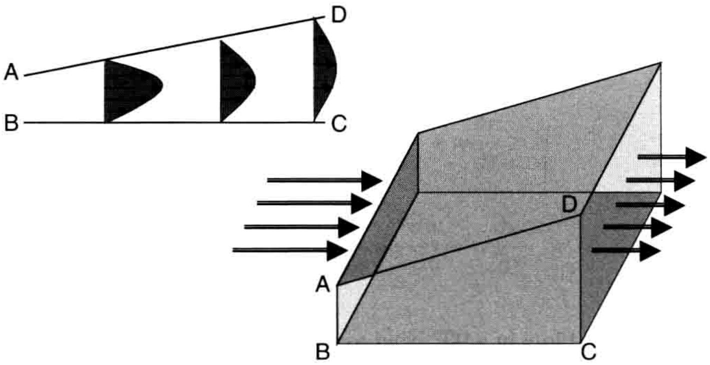


Figure 1.4 Two-dimensional flow between two diverging plates

$$\omega_x = \frac{1}{2}\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z}\right),\ \omega_y = \frac{1}{2}\left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x}\right),\ \omega_z = \frac{1}{2}\left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}\right) \tag{1.6}$$

where ω_x, ω_y, ω_z represent the rate of rotation around the x, y, and z axes.

The vorticity ζ is defined as twice the rate of rotation. Accordingly, the vorticity vector $\underline{\zeta}$ can be expressed as

$$\underline{\zeta} = \zeta_x \underline{i} + \zeta_y \underline{j} + \zeta_z \underline{k} = 2\omega_x \underline{i} + 2\omega_y \underline{j} + 2\omega_z \underline{k} \tag{1.7}$$

The flow is called irrotational when the rate of rotation around the three axes is zero. In this case, we must have $\zeta_x = \zeta_y = \zeta_z = 0$ for irrotational flow. The components of the vorticity vector in cylindrical coordinates can be written as

$$\zeta_r = \frac{1}{r}\frac{\partial v_z}{\partial \theta} - \frac{\partial v_\theta}{\partial z} \tag{1.8a}$$

$$\zeta_\theta = \frac{\partial v_r}{\partial z} - \frac{1}{r}\frac{\partial}{\partial r}(r v_z) \tag{1.8b}$$

$$\zeta_z = \frac{1}{r}\frac{\partial}{\partial r}(r v_\theta) - \frac{1}{r}\frac{\partial v_r}{\partial \theta} \tag{1.8c}$$

1.2 Conservation Principles

1.2.1 Conservation of Mass

Considering the general case of a compressible flow through the control volume (c.v.) shown in Figure 1.5 and assuming that $\boldsymbol{n}$ is a unit vector normal to the elementary surface area dA and $\boldsymbol{v}$ is the flow velocity through this area, then the conservation of mass equation takes the form

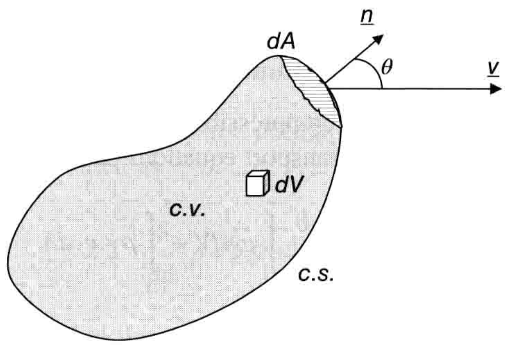


Figure 1.5 A schematic of an arbitrary control volume showing the flow velocity through a small elementary surface area

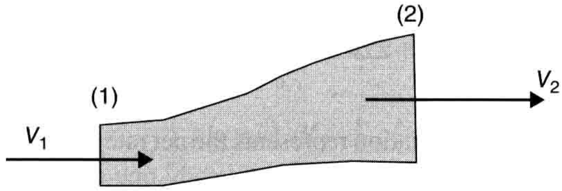


Figure 1.6 One-dimensional flow in a diverging flow passage

$$\frac{\partial}{\partial t}\int_{c.v.} \rho dV + \int_{c.s.} \rho \underline{v}.\underline{n}\, dA = 0 \tag{1.9}$$

where ρ is the fluid density, $\underline{v}$ is the fluid velocity, dV is an elementary volume, and t is the time.

When the control volume tends to a point, the equation tends to the differential form,

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \tag{1.10}$$

where u, v, and w are the velocity components in the x, y, and z directions. If the flow is incompressible, the above equation can be reduced to

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{1.11}$$

In the special case of 1-D steady flow in a control volume with one inlet and one exit (Figure 1.6), the conservation of mass equation takes the simple form,

$$\dot{m} = \rho_1 A_1 V_1 = \rho_2 A_2 V_2 = Const. \tag{1.12a}$$

where $\dot{m}$ is the mass flow rate, V is the flow velocity, and A is the cross-sectional area.