NUCLEAR TRAINING CENTRE

COURSE PI 30.1

5605N313

NUCLEAR TRAINING COURSE

COURSE PI 30.1

MECHANICAL EQUIPMENT

INDEX

Mechanical Equipment - Course PI 30.1

INTERIM OBJECTIVES AND COURSE NOTE SUPPLEMENT

430.10-0 Identification and Coding of Mechanical Equipment

The trainee will:

1. *State how equipment is identified:*

   (a) in field (by three (3) things)

   (b) on flow sheets (by two (2) things).

2. State colour codes for
   - Compressed Air
   - D_2O
   - H_2O
   - Steam
   - Oil

3. Draw symbols for
   - Centrifugal Pump
   - Recip. Pump
   - Heat Exchanger
   - Gate Valve
   - Safety Valve and Relief Valve
   - Compressors
   - Non-Return Valve
   - Globe Valve

430.10-1 Centrifugal Pumps

The trainee will:

1. Define:

   (a) suction lift
   (b) suction head
   (c) total (system head)

2. Label a simple schematic of a centrifugal pump, showing correct direction of impeller rotation, suction, discharge and fluids collector, (ie, *volute*, or *diffuser bowl*).

3. State six (6) basic features by which centrifugal pumps are classified and state options for each, ie, mount - vertical, horizontal, etc.

4. Be able to identify from a schematic or drawing:

   (a) type of mount
   (b) type of casing (ie, volute, diffuser bowl)
   (c) type of impeller
   (d) # stages
   (e) type of flow through the impeller
   (f) casing split

5. Compare efficiency, flow rate and, pressure at discharge for three (3) centrifugal pump impeller types.

6. Discuss causes, effects and symptoms of four (4) operational problems in centrifugal pumps; cavitation, air-locking, vapour-locking and loss of prime (lack of prime).

330.10-1 Centrifugal Pumps

The trainee will:

1. List those special design(s) which compensate for axial forces in centrifugal pumps.

2. List those special designs which compensate for radial forces in centrifugal pumps.

3. Explain briefly the operation of:

   (a) balance holes and wear rings
   (b) opposed impellers
   (c) balancing drum
   (d) balancing disc

   to compensate for axial forces in centrifugal pumps

4. Explain why a double suction impeller helps eliminate/alleviate cavitation.

430.10-2 Positive Displacement Pumps

The Trainee will:

1. Be able to name all eight (8) types of positive displacement pumps discussed in the text and know which ones are reciprocating or rotary.

2. State what is meant by:

   (a) double acting
   (b) single acting
   (c) duplex
   (d) triplex

3. Discuss causes, effects and symptoms of two (2) operational problems of positive displacement pumps.

   (a) cavitation
   (b) operation against blocked discharge (and necessity for pressure relief valve at discharge).

4. Compare centrifugal and positive displacement pumps with respect to:

   (a) principle of operation
   (b) how change in total head changes capacity
   (c) operation against a closed discharge and need for pressure relief valve
   (d) pressures and flow rates generated (capacities)
   (e) handled liquids
   (f) priming

5. Name one (1) type of pump which would offer "leak free" operation.

430.10-3 Compressors - Dynamic and Positive Displacement

The trainee will:

1. Name two (2) basic classes of compressors and compare their capacity, pressure and efficiency.

2. State that dynamic compressors are divided into two (2) types and name them.

3. Define "surge" and state in which type of compressors it occurs, ie, dynamic and/or positive displacement.

4. State characteristics (advantages; disadvantages) of reciprocating piston, diaphragm and screw compressors.

430.10-4 Air Systems

The trainee will:

1. For each of four (4) air systems in the NGD, state special requirements and kind(s) of compressor(s) used.

2. Draw and label a schematic of a typical air system.

3. Explain what compressor "unloading" is. State explicitly:

   (a) Is motor running during unloading?
   (b) Is air (gas) being compressed?
   (c) Is motor load reduced?
   (d) When is it used?

4. State why compressor "unloading" is used.

5. State the methods of compressor unloading used in NGD.

6. Explain what is meant by "automatic dual control" of compressors in the NGD. Name the two (2) modes of operation that are necessary.

230.11-1 Compressors

The trainee will:

1. Define:

   (a) capacity
   (b) volumetric efficiency
   (c) compression efficiency
   (d) compressor (shaft) efficiency

2. (a) Draw "Adiabatic" and "Isothermal" compression processes on a P-V diagram, both starting at some arbitrary point (V_0,P_0) and both terminating on the same isobaric line but at different points.

   (b) State that the compression work done in each case is represented by the area under the process curve.

3. For a two-stage reciprocating compressor, indicate on a P-V diagram that work savings due to:

   (a) water jacketing of LP cylinder
   (b) intercooler
   (c) water jacketing of HP cylinder

4. State some advantages of cooling, other than for work savings.

5. List three (3) or more "circumstances" or "sets of conditions" which would lead to explosions in a compressed air system.

430.10-5 Fans

The trainee will:

1. Name the two (2) basic types of fans and state the basic differences between them.

2. State which type of fan is used for air conditioning and ventilation systems with extensive ducting grids in large stations.

430.10-6 Vacuum Pumps

The trainee will:

1. State the two (2) basic classifications of vacuum pumps.

2. State the principle of operation of vapour vacuum pumps.

330.11-1 Heat Exchanges

The trainee will:

1. Be able to describe a typical shell and tube heat exchanger according to the following:

   (a) number of passes that tube fluid makes
   (b) type of tube bundle - straight; U shaped: coil

2. Briefly describe three (3) different "patterns of shell/tube flow" and two different "types of flow" possible in heat exchangers. State which combination is most efficient.

3. State which type of tube bundle is susceptible to contraction/expansion problems and state how they can be solved.

4. State two (2) functions of baffles in a heat exchanger.

430.12-1 Piping, Tubing and Joints

The trainee will:

1. State how to specify the size of a pipe and tube. State what N.S. and schedule # are and state how O.D., I.D. and wall thickness are given/found.

2. Compare pipes and tubes with respect to their distinguishing features:

   (a) variety of size available
   (b) tolerances
   (c) surface qualities
   (d) workability/bendability
   (e) cost

3. Name the various types of pipe joints and give the advantages and/or limitations specified in course notes:

   (a) permanent - welded
   (b) dismountable - flanged (welding neck); grayloc; victaulic

4. Name the dismountable pipe joint which is most commonly used in NGD for high pressures, and high temperatures. State why the victaulic joint is restricted to low temperatures.

5. Name the various types of dismountable tube joints and specify which one is most commonly preferred for use in NGD.

6. Name one (1) "zero leakage" (dismountable) pipe joint and one (1) "zero leakage" (dismountable) tube joint.

430.13-1 Valves

The trainee will:

1. State the (4) basic functions that valves perform in systems:

   (a) isolation
   (b) regulation
   (c) backflow prevention
   (d) pressure relief

   Give an example of a kind of valve used for each.

2. Name different types of stem seals. State the main advantage of a "bellows seal" in a valve.

3. State what functions a lantern ring serves in the stuffing box packing of a valve.

4. State what is meant by "zero leakage valve" and name two (2) types of "zero leakage" valves.

5. Compare gate valves and globe valves in respect to:

   (a) basic function (application)
   (b) pressure drop across valve when fully open
   (c) directionality (direction of fluid flow for various fluids)
   (d) flow pattern (changes in flow direction through the valve)

6. Be able to explain the reason for "double port" design in globe valves and "parallel slide disc" design in gate valves and state an application for each.

7. Compare "swing check" and "lift (piston) check" valves with respect to:

   (a) pressure drop across open valve
   (b) leakage in "backflow prevent mode"
   (c) use restrictions (horizontal, vertical)

8. Compare safety and relief valves with respect to:

   (a) type of fluid handled
   (b) valve action
   (c) amount of discharge
   (d) relative magnitude of opening and closing pressure

9. State, for the four (4) special valves listed below, which are used for isolation, control, or both isolation and control:

   (a) butterfly
   (b) ball
   (c) diaphragm
   (d) plug

10. State advantages of butterfly and ball valves over globe type valves.

430.14-1 Lubrication

The trainee will:

1. State the three (3) types (methods) of lubrication and indicate which one(s) have metal-to-metal contact; have an oil wedge.

2. Define each of the four (4) major properties of liquid lubricants:

   (a) viscosity
   (b) flashpoint
   (c) temperature stability
   (d) oiliness

3. Explain how viscosity varies with temperature, (ie, state the effect of temperature on viscosity).

430.14-2 Lubrication and Bearings - Unit I Bearings

The trainee will:

1. State the four (4) roles of bearings.

2. Be able to draw a classification tree of the major types and sub-types of bearings, and state whether each particular bearing is used for axial or radial support.

430.14-2 Lubrication and Bearings - Unit II Bearings

The trainee will:

1. State at least three (3) characteristics, for each of the three (3) methods of lubrication, ie, hydrodynamic, hydrostatic, boundary.

2. Be able to explain the concept of oil wedge lubrication as applied in radial and axial plain bearings.

3. State the types (methods) of lubrication in each of the two main types of bearings - plain and rolling element.

430.14-2 Lubrication and Bearings - Unit III Bearing Design

The trainee will:

1. With the aid of sketches, describe the construction of a tilting pad type bearing and explain briefly how lubrication is achieved at running speed. Name this type of lubrication.

2. State that tilting pad type bearings can be signed to compensate for radial loads or axial loads.

3. In Kingsbury (Mitchell) tilting pad axial thrust bearings, state whether the:

   (a) thrust collar
   (b) set (or sets) of tilting pads

   are

   (i) stationary (ie, fixed to housing or casing), or
   (ii) rotate with the shaft.

4. State that in Kingsbury (Mitchell) tilting pad axial thrust bearings, a rotating thrust collar may be in contact with a single set of tilting pads or it may be "sandwiched" between two sets of tilting pads.

230.12-1 Lubrication

The trainee will:

1. State the direction of viscosity increase from S.A.E. 20 to S.A.E. 50 (crankcase oils).

2. Define viscosity index (V.I.) of oils and state whether an oil with V.I.^{100} is more or less temperature stable than an oil with V.I.^{50}.

3. State three (3) advantages of a continuous circulating lubricating oil system over a "once through" lubrication system.

430.15-1 Sealing Devices

The trainee will:

1. State the qualities desired in gasketing materials.

2. State the types of gasket materials which might be used for high temperature and high pressure; which structure and material would form a gasket for high temperature AND high pressure use.

3. Explain the effects of having I.D. of gasketing:

   (a) too small
   (b) too large

4. Explain the purpose of a lantern ring in a pump stuffing box.

5. Explain the operation of a mechanical seal with the aid of a simple sketch.

330.14-1 Axial Mechanical Seals

The trainee will:

1. State advantages and disadvantages of mechanical seals as compared to packings with respect to the following:

   (a) long or short downtime to replace
   (b) cost
   (c) ease of installation
   (d) degree of failure (partial or total)
   (e) care required during handling and installation
   (f) lifetime
   (g) leakage control
   (h) friction (comparative magnitude)
   (i) shaft wear

330.13-1 Shaft Couplings

The trainee will:

1. Sketch or describe the types of shaft misalignments:

   (a) angular offset
   (b) parallel offset

2. Briefly explain each of the items in the simple motor/pump alignment procedure given below:

   (a) align motor to fixed pump (not vice-versa)
   (b) check shaft and coupling "run-out"
   (c) eliminate end-float
   (d) rough initial alignment
   (e) measurement of "parallel offset" and "angular offset" by taking "face" and "periphery" readings, 90° apart
   (f) correction in vertical plane and horizontal plane
   (g) check

3. State what functions the following perform in a motor/pump alignment procedure:

   (a) dial indicator
   (b) spreader mechanism

4. Name the two (2) basic types of couplings and know what degree of misalignment each can absorb.

5. List characteristics of rigid and also of flexible couplings with reference to:

   (a) torques transferred (large, medium, small)
   (b) necessity of lubrication of this drive component
   (c) speed limitation (high, medium, low)
   (d) driver/driven shaft (axis) orientation
   (e) positive drive or slip
   (f) R.P.M. change possible driver/driven or not

230.13-2 Belt Drives

The trainee will:

1. List characteristics of V-belt drives with reference to:

   (a) torques transferred (large, medium, small)
   (b) necessity of lubrication of this drive component
   (c) speed limitation (high, medium, low)
   (d) driver/driven shaft (axis) orientation
   (e) positive drive or slip
   (f) R.P.M. change possible driver/driven or not

2. List the items that should be checked during a routine inspection of a multiple V-belt drive of a compressor.

230.13-3 Chain Drives

The trainee will:

1. List characteristics of this type of drive with respect to:

   (a) torques transferred (large, medium, small)
   (b) necessity of lubrication of this drive component
   (c) speed limitation (high, medium, low)
   (d) driver/driven shaft (axis) orientation
   (e) positive drive or slip
   (f) R.P.M. change possible driver/driven or not

230.13-4 Gears and Gearing

The trainee will:

1. List the characteristics of this type of drive with reference to:

   (a) torques transferred (large, medium, small)
   (b) necessity of lubrication of this drive component
   (c) speed limitation (high, medium, low)
   (d) driver/driven or shaft (axis) orientation
   (e) positive drive or slip
   (f) R.P.M. change possible driver/driven or not

230.16-1 Gas Turbines

The trainee will:

1. State the type of dynamic compressor used in the gas turbines of the standby generators in NGD.

2. Explain what is meant by "surging" and "stalling".

3. Given a schematic of a gas turbine generator system used in NGD label the diagram fully to include:

   (a) DC starter motor
   (b) multi-stage axial compressor
   (c) combustion chambers including fuel nozzle, igniter
   (d) two stage compressor turbine
   (e) one stage power turbine
   (f) reduction gear
   (g) standby generator
   (h) compressor outlet pressure ≈ 500 kPa
   (i) combustion chamber exhaust temp ≈ 600°C
   (j) free power turbine - operating speed 7200 RPM
       - exhaust temp ≈ 450°C
   (k) standby generator operating speed 1800 R.P.M.

4. State the effect of the following on a gas turbine's output:

   (a) inlet ambient air temperature
   (b) inlet air pressure
   (c) build-up of combustion products on rotor/stator blades and other interior surfaces.

Mechanical Equipment - Course 430.1

IDENTIFICATION AND CODING OF MECHANICAL EQUIPMENT

Design, operation and maintenance of plant equipment requires the establishment and use of a system of positive identification of all systems, sub-systems and components in our plants.

A numbering system supplemented in the field by colour coding and tagging has been adopted. On flow sheets, a system of equipment symbols is used.

The numbering system is called **USI - Uniform Subject Index**. Although in principle it is identical in all our plants, it may vary in detail from station to station. In addition to mechanical equipment, USI specifies most of the equipment and activities in the plant. The subject index is sub-divided into Divisions. For example, at Bruce NGS, the Divisions are:

Division 0	General Project
Division 1	Site and Improvements
Division 2	Buildings, Structures and Shielding
Division 3	Reactor, Boiler and Auxiliaries
Division 4	Turbine, Generator and Auxiliaries
Division 5	Electric Power Systems
Division 6	Instrumentation and Control
Division 7	Common Processes and Services
Division 8	Construction Indirects

Each system, sub-system and component is assigned a five-digit number. An example from Division 4 explains the structure of USI:

Division	40000	Turbine, Generator and Auxiliaries
Major System	42000	Condensing System
System	42100	Main Condensing System
Sub-System	42120	Condenser Extraction System
Components	42121	Ejectors
	42122	Vacuum Pumps
	42123	Valves
	42128	Pipe Supports
	42129	Piping

So the first digit is indicative of a division, second of a major system in the division, third of a system within the major system, fourth of a sub-system in the system and finally the fifth digit classifies components in the sub-system.

In the field, the USI number accompanied by a brief written description is found either printed on the equipment or on a tag attached to the equipment.

Usually there is more than one component of the same kind within a sub-system, for example valves. To distinguish between identical components a letter code (P for pump, V for valve, etc) and a serial number is used to identify the particular component. This code, plus the serial number, is definitely found attached onto the component. Note that the letter code plus the serial number replaces the last digit which indicated the type of component in general.

For example,

4212V2

will be valve number two in the Condenser Extraction System.

On the flowsheets (system diagrams) pictorial symbols as well as USI numbers and letter symbols are used to achieve correspondence between the field and the documentation.

The letter symbols as well as pictorial symbols of various types of mechanical equipment are given in the Addendum with a complete Division 4 numbering system as an example. Also attached are two examples of flowsheets. All examples originate from Bruce NGS.

For quick field orientation, equipment and particularly piping is colour and letter coded so that it is immediately obvious what type of fluid is inside. Also an arrow is attached showing the direction of flow. The colours and code letters commonly used are:

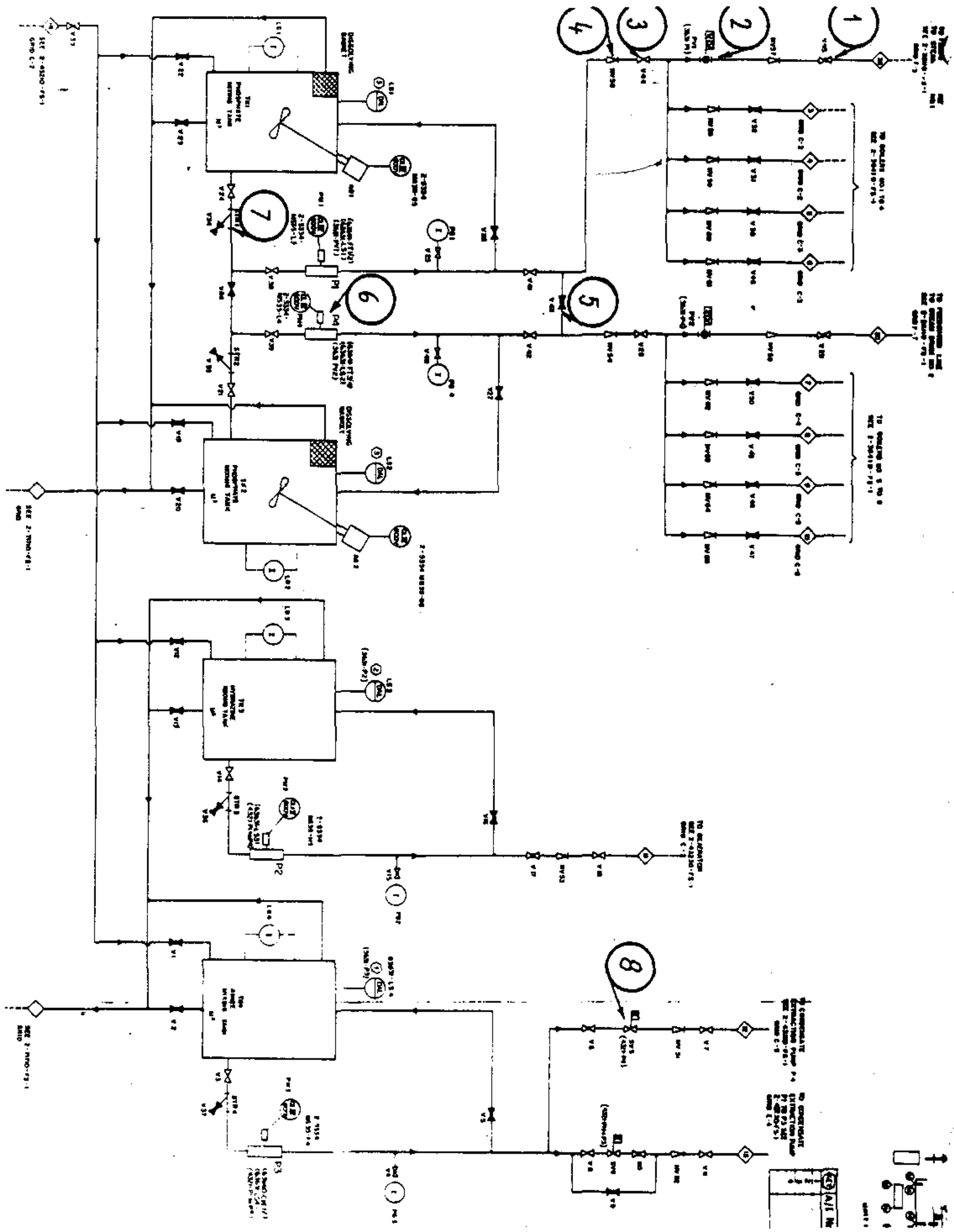
Air	A	Blue
Heavy Water	D	Pink
Light (Common) Water	W	Green
Steam	S	Silver (Aluminum), White at BHWP
Oil	O	Yellow
Helium	H	Brown
Other Gases	G	Brown
Bldg. Heating		White
Drains		Black
Fire Protection		Red
Vacuum		Purple
Chemicals		Orange

To summarize, each system, sub-system and component is given a USI number. Each similar component is given a serial number. In the field, systems are colour coded according to the fluid carried in them and each component is labeled with its USI number and serial number. All documentation (manuals, flowsheets, etc) refers to components by ~~their~~ USI and serial numbers.

ASSIGNMENTS

1. How is a piece of equipment identified in the field?
2. How is a piece of equipment identified in a flowsheet?
3. The USI number is 71310. What can you say about the equipment labelled by that number?
4. There are three identical pumps in the system numbered as 43230 - Boiler Feed System. Write a complete identification of all three of them.
5. A valve in a system is leaking. How would you identify it? In the field and in the flowsheet?
6. You identified a valve in a flowsheet. Its number is 4212V47. Describe how you would proceed in identifying it in the field.
7. Identify circled components in the attached sheet.

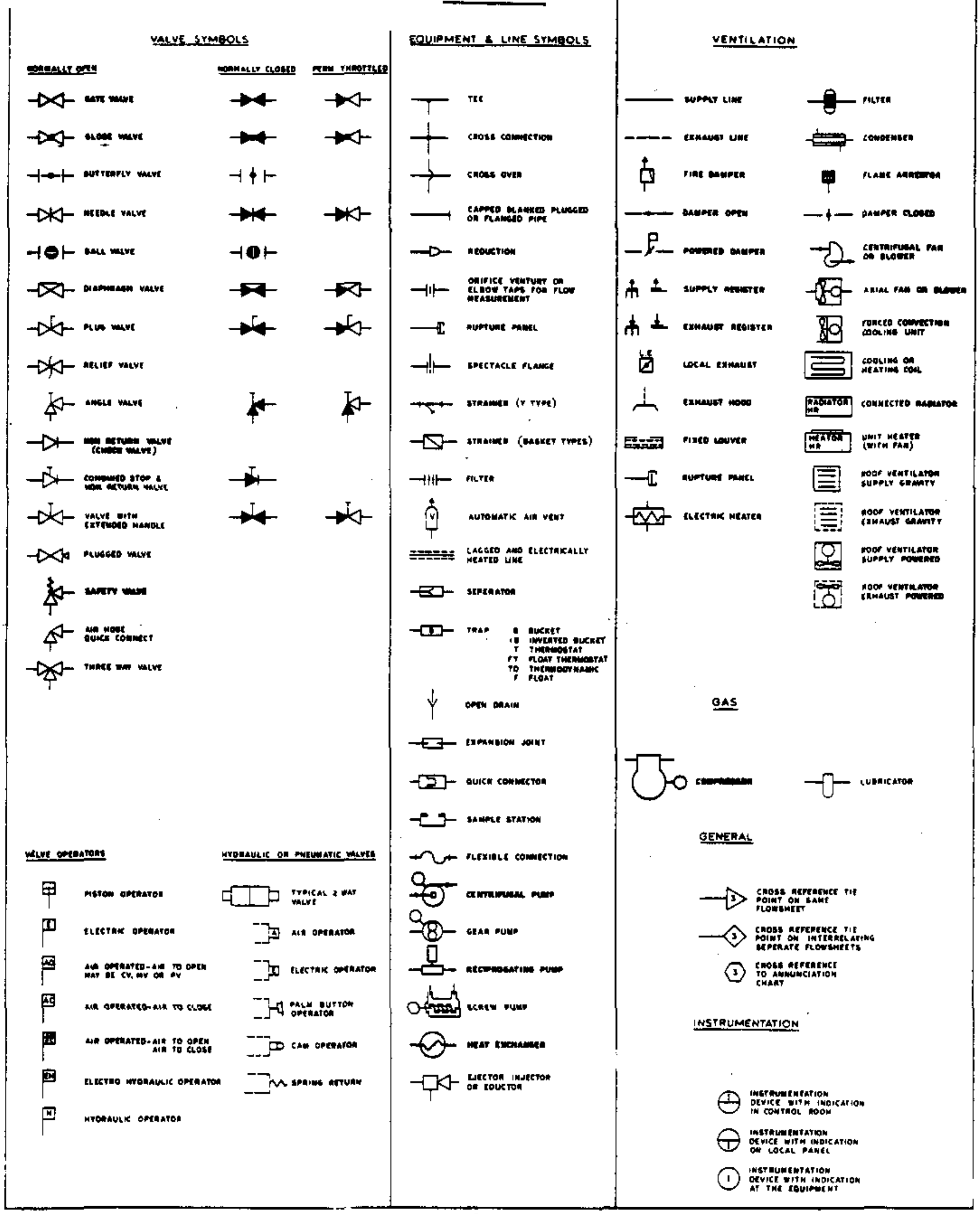
K. Mika


1
2
3
4
5
6
7
8
P1
P2
P3

ADDENDUM

Mechanical Equipment Device Code

ACU	Air Conditioning Unit	HTR	Heater
ARV	Air Release Valve	HTE	Heater Electrical
BO	Boiler	HX	Heat Exchanger
BRG	Bearing	LU	Lubricator
CCR	Cooling Coil - Refrig.	MV	Power Operated Valve
CCW	Cooling Coil - Water	NV	Non Return Valve (Electric, Air or Hydraulic)
CD	Condenser		
CP	Compressor	NZ	Nozzle
CR	Crane	P	Pump
CTU	Coolant Tube	PCV	Pressure Control Valve
CV	Control Valve	PO	Pneumatic Operator
DC	Drain Cooler	PRV	Pressure Regulating Valve
DP	Damper	RD	Rupture Disc
DR	Chemical Dryer	RV	Relief or Safety Valve
DY	Dryer	SC	Screen
EJ	Expansion Joint	SD	Steam Drum
F	Fan	SP	Separator
FC	Fluid Coupling	SF	Swaged Fitting
FCV	Flow Control Valve	STR	Strainer
FM	Fuelling Machine	SV	Solenoid Valve
G	Generator	SAE	Air Ejector (Steam Jet)
SG	Standby Generator	TK	Tank
GR	Gear Reducer	TP	Trap
HA	Hanger - Anchor	TRV	Temperature Regulating Valve
HCS	Heating Coil (Steam)		
HCW	Heating Coil (Water)	TU	Turbine
HR	Hanger Rigid (Pipes, etc)	V	Valve (Manually Operated)
HV	Hanger Variable		


VALVE SYMBOLS
NORMALLY OPEN
NORMALLY CLOSED
PERM THROTTLED
GATE VALVE
GLOBE VALVE
BUTTERFLY VALVE
NEEDLE VALVE
BALL VALVE
DIAPHRAGM VALVE
PLUG VALVE
RELIEF VALVE
ANGLE VALVE
NON RETURN VALVE (CHECK VALVE)
COMBINED STOP & NON RETURN VALVE
VALVE WITH EXTENDED HANDLE
PLUGGED VALVE
SAFETY VALVE
AIR HOSE QUICK CONNECT
THREE WAY VALVE
VALVE OPERATORS
PISTON OPERATOR
ELECTRIC OPERATOR
AIR OPERATED-AIR TO OPEN MAY BE CV, HV OR PV
AIR OPERATED-AIR TO CLOSE
AIR OPERATED-AIR TO OPEN AIR TO CLOSE
ELECTRO HYDRAULIC OPERATOR
HYDRAULIC OPERATOR
HYDRAULIC OR PNEUMATIC VALVES
TYPICAL 2 WAY VALVE
AIR OPERATOR
ELECTRIC OPERATOR
PALM BUTTON OPERATOR
CAM OPERATOR
SPRING RETURN
EQUIPMENT & LINE SYMBOLS
TEE
CROSS CONNECTION
CROSS OVER
CAPPED BLANKED PLUGGED OR FLANGED PIPE
REDUCTION
ORIFICE VENTURY OR ELBOW TAPS FOR FLOW MEASUREMENT
RUPTURE PANEL
SPECTACLE FLANGE
STRAINER (Y TYPE)
STRAINER (BASKET TYPES)
FILTER
AUTOMATIC AIR VENT
V
LAGGED AND ELECTRICALLY HEATED LINE
SEPERATOR
TRAP
B BUCKET
IB INVERTED BUCKET
T THERMOSTAT
FT FLOAT THERMOSTAT
TD THERMODYNAMIC
F FLOAT
OPEN DRAIN
EXPANSION JOINT
QUICK CONNECTOR
SAMPLE STATION
FLEXIBLE CONNECTION
CENTRIFUGAL PUMP
GEAR PUMP
RECIPROCATING PUMP
SCREW PUMP
HEAT EXCHANGER
EJECTOR INJECTOR OR EDUCTOR
VENTILATION
SUPPLY LINE
EXHAUST LINE
FIRE DAMPER
DAMPER OPEN
POWERED DAMPER
SUPPLY REGISTER
EXHAUST REGISTER
L.E.
LOCAL EXHAUST
EXHAUST HOOD
FIXED LOUVER
RUPTURE PANEL
ELECTRIC HEATER
FILTER
CONDENSER
FLAME ARRESTOR
DAMPER CLOSED
CENTRIFUGAL FAN OR BLOWER
AXIAL FAN OR BLOWER
FORCED CONVECTION COOLING UNIT
COOLING OR HEATING COIL
RADIATOR HR
CONNECTED RADIATOR
HEATER HR
UNIT HEATER (WITH FAN)
ROOF VENTILATOR SUPPLY GRAVITY
ROOF VENTILATOR EXHAUST GRAVITY
ROOF VENTILATOR SUPPLY POWERED
ROOF VENTILATOR EXHAUST POWERED
GAS
LUBRICATOR
GENERAL
3
CROSS REFERENCE TIE POINT ON SAME FLOWSHEET
CROSS REFERENCE TIE POINT ON INTERRELATING SEPERATE FLOWSHEETS
CROSS REFERENCE TO ANNUNCIATION CHART
INSTRUMENTATION
INSTRUMENTATION DEVICE WITH INDICATION IN CONTROL ROOM
INSTRUMENTATION DEVICE WITH INDICATION ON LOCAL PANEL
I
INSTRUMENTATION DEVICE WITH INDICATION AT THE EQUIPMENT

SUBJECT INDEX — BRUCE G.S.

DIVISION 4 TURBINE GENERATOR & AUXILIARIES

40000 TURBINE GENERATOR & AUXILIARIES
- 40010 Performance Testing

41000 TURBINE GENERATOR
- 41010 Tender Evaluation
- 41020 Lubricating System
- 41090 Maintenance Equipment
  - 41091 Handling equipment
  - 41092 Tools

41100 Turbine & Auxiliary Equipment
- 41110 Turbine
  - 41111 Bearings
  - 41112 H.P. casings and rotors
  - 41113 L.P. casings and rotors
- 41120 Emergency Stop & Governor Valves
- 41130 Reheat Stop & Intercept Valves
- 41140 Turning Gear
- 41150 Gland Seal System
- 41160 Turbovisory Equipment
- 41170 Governing System
- 41180 Steam Reject System
  - 41183 Valves
  - 41188 Pipe Supports
  - 41189 Piping
- 41190 L.P. Exhaust Cooling System

41200 Generator & Auxiliary Equipment
- 41210 Alternator
  - 41211 Bearings
  - 41212 Hydrogen seals
- 41220 Excitation
  - 41221 Exciters
  - 41222 Field cabling
  - 41223 Field breakers
- 41230 Hydrogen Cooling System
  - 41231 Heat exchangers
  - 41233 Valves
  - 41237 Purging equipment
  - 41238 Pipe supports
  - 41239 Piping
- 41240 Stator Cooling System
  - 41241 Heat exchangers
  - 41242 Pumps
  - 41243 Valves
  - 41244 Stator water tank
  - 41248 Pipe supports
  - 41249 Piping
- 41250 Seal Oil System
  - 41251 Heat exchangers
  - 41252 Pumps
  - 41253 Valves
  - 41258 Pipe supports
  - 41259 Piping

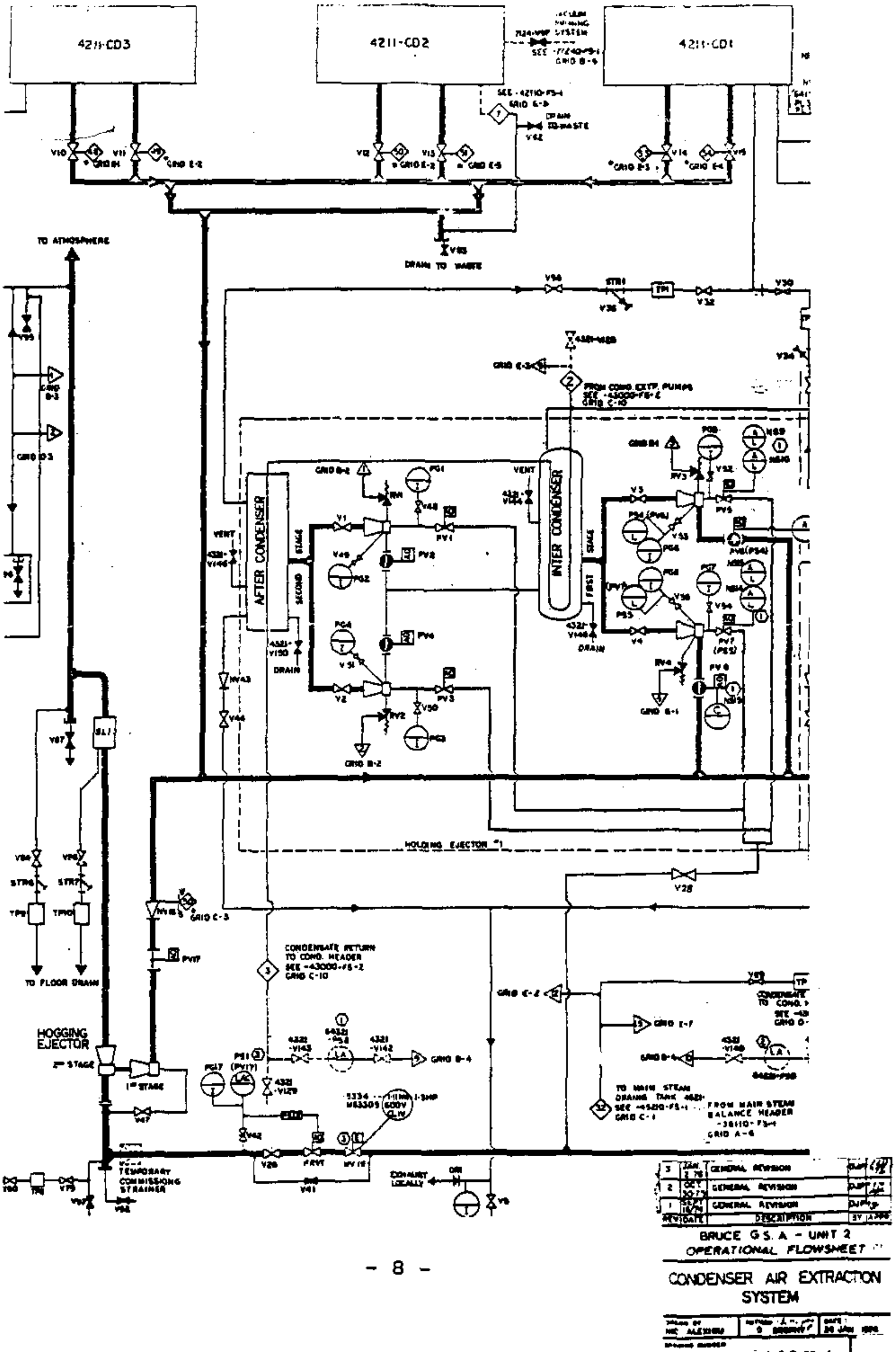
41400 Moisture Separation System
- 41410 Separator
- 41420 Separator Drains System
  - 41423 Valves
  - 41428 Pipe supports
  - 41429 Piping

41500 Steam Reheat System
- 41510 Live Reheat Steam System
- 41513 Valves
- 41518 Pipe supports
- 41519 Piping
- 41520 Reheater
  - 41521 Tubing
- 41530 Reheater Drains System
  - 41532 Drains pumps
  - 41533 Valves
  - 41534 Drains tank
  - 41538 Pipe supports
  - 41539 Piping
- 41540 Reheater Vent System
  - 41543 Valves
  - 41548 Pipe supports
  - 41549 Piping
- 41550 Reheater Blanket System
- 41560 Reheat Safety Valve System
  - 41563 Valves
  - 41568 Pipe supports
  - 41569 Piping
- 41570 Hot Reheat
  - 41573 Valves
  - 41578 Pipe supports
  - 41579 Piping
- 41580 Cold Reheat
  - 41583 Valves
  - 41588 Pipe supports
  - 41589 Piping

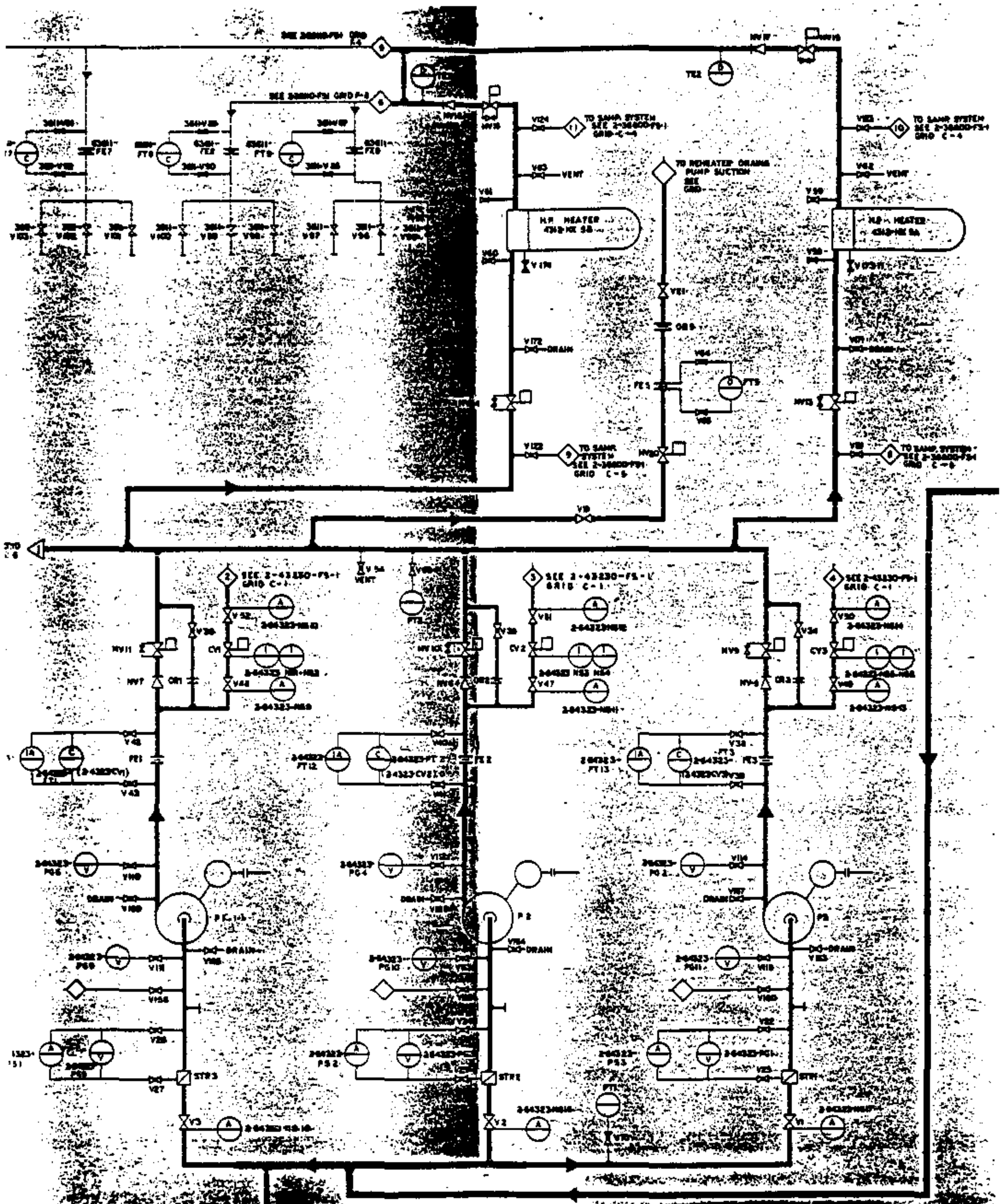
42000 CONDENSING SYSTEMS
- 42100 Main Condensing System
  - 42110 Main Condenser
    - 42111 Condenser
    - 42119 Tubing
  - 42120 Condenser Air Extraction System
    - 42121 Ejectors
    - 42122 Vacuum pumps
    - 42123 Valves
    - 42128 Pipe supports
    - 42129 Piping

43000 FEEDWATER SYSTEMS
- 43100 Feedwater Heating Systems
  - 43110 Extraction Steam System
    - 43113 Valves
    - 43118 Pipe supports
    - 43119 Piping
  - 43120 Feedwater Heaters
    - 43121 Closed heaters
    - 43122 Deaerator
    - 43123 Valves
  - 43130 Heater Drains System
    - 43132 Drain pumps
    - 43133 Valves
    - 43138 Pipe supports
    - 43139 Piping
  - 43140 Heater Vents System
    - 43143 Valves
    - 43148 Pipe supports
    - 43149 Piping
  - 43150 Heater Relief Valve System
    - 43153 Valves
    - 43158 Pipe supports
    - 43159 Piping
- 43200 Condensate & Feedwater System
  - 43210 Condensate System
    - 43212 Extraction pumps
    - 43213 Valves
    - 43218 Pipe supports
    - 43219 Piping
  - 43220 Condensate Make-up & Rejection System
    - 43222 Pumps
    - 43223 Valves
    - 43224 Storage tank
    - 43228 Pipe supports
    - 43229 Piping
  - 43230 Boiler Feed System
    - 43232 Pumps
    - 43233 Valves
    - 43234 Gland seal tank
    - 43235 Strainers
    - 43238 Pipe supports
    - 43239 Piping

45000 AUXILIARY SYSTEMS
- 45100 Sampling System
  - 45110 Sampling Circuit
    - 45111 Coolers
    - 45112 Pumps
    - 45113 Valves
    - 45118 Pipe supports
    - 45119 Piping
- 45200 Drain Systems
  - 45210 Steam Drain System
    - 45211 Traps
    - 45212 Pumps
    - 45213 Valves
    - 45214 Tanks
    - 45218 Pipe supports
    - 45219 Piping
  - 45220 Drain & Waste System
    - 45223 Valves
    - 45228 Pipe supports
    - 45229 Piping
  - 45230 Air Vents
    - 45233 Valves
    - 45238 Pipe supports
    - 45239 Piping
- 45300 Gland Injection System
  - 45310 Pump Gland Injection System
    - 45311 Coolers
    - 45312 Pumps
    - 45313 Valves
    - 45314 Tanks
    - 45318 Pipe supports
    - 45319 Piping



4211-CD3
4211-CD2
4211-CD1
TO ATMOSPHERE
DRAIN TO WASTE
AFTER CONDENSER
INTER CONDENSER
HOLDING EJECTOR #1
HOGGING EJECTOR
2nd STAGE
1st STAGE
TO FLOOR DRAIN
CONDENSATE RETURN TO COND. HEADER SEE -43000-FS-2 GRID C-10
TEMPORARY COMMISSIONING STRAINER
EXHAUST LOCALLY
GENERAL REVISION
DESCRIPTION
BRUCE G.S. A - UNIT 2
OPERATIONAL FLOWSHEET
CONDENSER AIR EXTRACTION SYSTEM



BRUCE G.S.

OPERATIONAL FLOW SHEET

H.P. FEEDWATER SYSTEM

NK21-OP-2-43230-FS-43 R-0

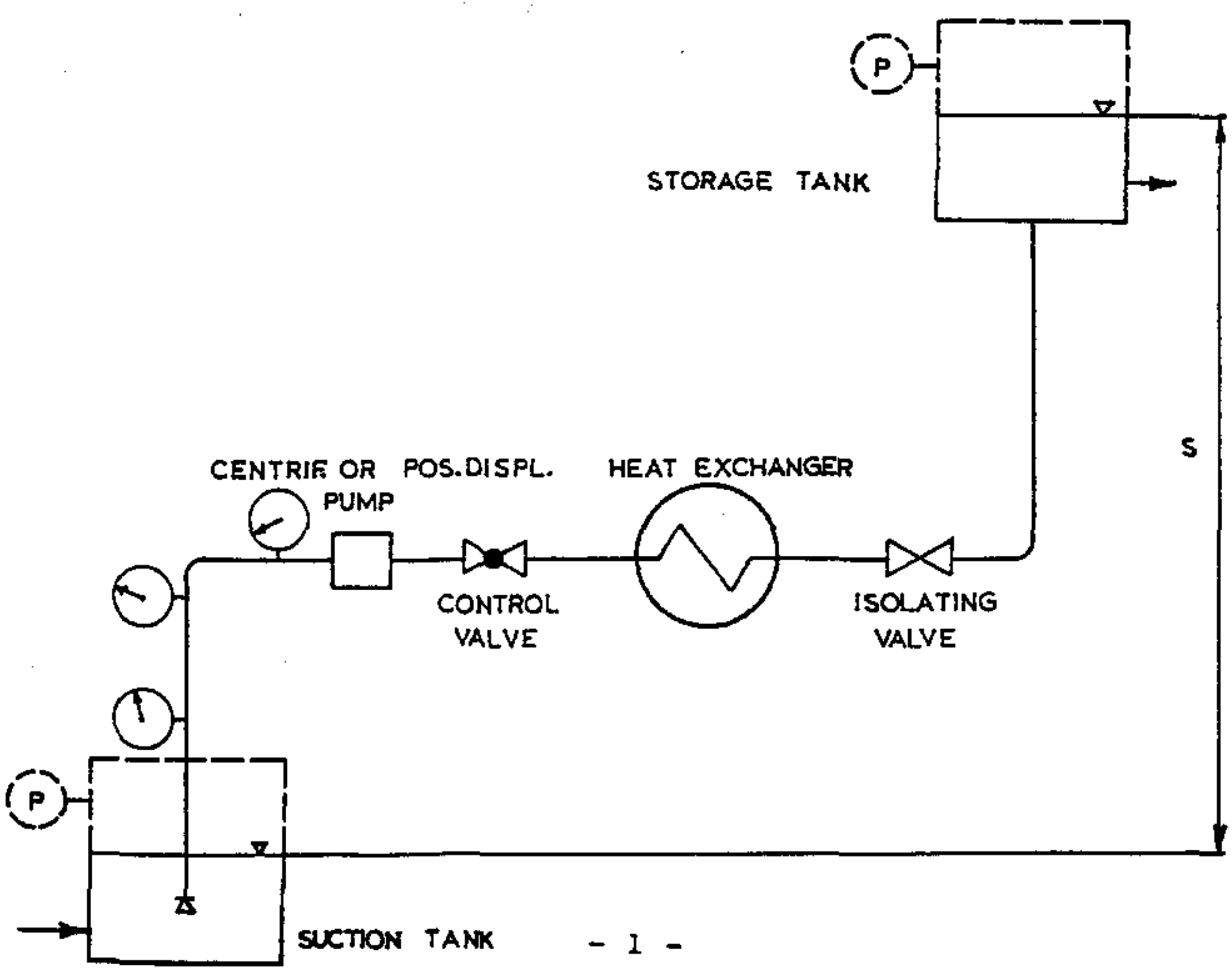
Mechanical Equipment - Course 430.1

CENTRIFUGAL PUMPS

In both nuclear and heavy water plants the process depends on the transport and pressurization of fluids which are either incompressible liquids or compressible vapours and gases.

Liquids are handled by devices called pumps. Depending on the basic principle of operation, the pumps can be divided into either kinetic or positive displacement. In each of these basic groups there is a number of different designs. Although we use a number of designs of positive displacement pumps, in the case of kinetic principle, our choice is practically limited to one design only which is the centrifugal pump.

At this point a question arises: What is the difference(s) between a centrifugal and a positive displacement pump? The best way to answer it, is to show and discuss the performance of the pump in a typical system in Figure 1.



The pump works with so-called suction lift, because its centerline is above the suction tank. If it was below, we would talk about the suction head. The pump delivers the liquid through a control valve, a heat exchanger and an isolating valve into a storage tank where it is ready to be used further in the process.

To perform this function, the pump has to overcome several resistances:

1. It has to work against gravitation to lift the liquid through the vertical distance S.

2. It has to overcome friction resistance to the flow of all system components, including piping.

3. Finally if suction and storage tanks are not both open to the atmosphere or under the same pressure in general, the pump has to work against the pressure difference.

The sum of all these work components is called a total head and for a pump to overcome this head it must develop high enough pressure. Obviously, if any of the resistances change, eg, the storage tank is put higher, control valve is closed more, the run of piping changed or a pressure or level in the suction tank drops, the total head will change accordingly.

Having understood how a system is applying a load on a pump, the differences between centrifugal and positive displacement pumps can be listed:

1. Change in Total Head - Change in Capacity.
2. Regulation of Flow.
3. Operation Against the Discharge Valve Closed.
4. Priming.
5. Capacities.
6. Pressures - Total Heads.
7. Handled Liquids.
8. Maintenance.

1. Change in Total Head - Change in Capacity

The basic difference between a centrifugal and a positive displacement pump is how they react to a change of the total head.

(a) A centrifugal pump is sensitive to changes of the total head and the change of the total head will result in a change of the delivered flow called 'capacity'. In the majority of cases the increase of the total head will cause the capacity to drop. If the total head drops, the capacity will increase.

(b) This is not true for positive displacement pumps.. They are the pumps which deliver a constant quantity of liquid per stroke or revolution regardless of the total head changes. Although it is not exactly true because with the increased discharge pressure the leakage along the shaft, along pistons, plungers and other moving components will go up, within practical limits this leakage is small compared with the discharge flow.

2. Regulation of Flow

The basic difference just explained will have implications as far as the system of regulation is concerned.

(a) Centrifugal pumps are sensitive to changes of the total head which can be easily accomplished by closing (= throttling) or opening (= dethrottling) of a regulating valve. Therefore the regulation of the flow in a system with a centrifugal pump is accomplished by having a regulating valve in the system, usually close to the discharge of the pump.

(b) This type of regulation would not work on positive displacement pumps because they are not sensitive to system head changes. The methods used on these pumps are either the change of speed of a driving motor or a change of the length of the stroke.

3. Operation Against the Discharge Valve Closed

Usually for isolation purposes the discharge from a pump is furnished with an isolating valve. During a shutdown, the discharge valve was closed. The pump was then started and the discharge valve was left closed. What happened?

(a) The impeller of a centrifugal pump started rotating but because the discharge is blocked, the liquid is churned within the pump. The discharge pressure increased slightly but no immediate damage occurred. If the mistake is not corrected, all energy from the motor is spent in churning the liquid and the temperature in the pump starts increasing. If nothing is done this temperature might increase so high that the problems like seizing, burning of seals, bearing damage will happen. The time when serious problems develop depends very much on the design and size of the pump and can vary from 30 seconds to 30 minutes.

As a matter of fact, some centrifugal pumps are deliberately started with discharge valve closed because starting power and torque requirements are cut down. As soon as the set reaches the operational speed the valve is opened.

(b) Positive displacement pumps deliver constant quantity per stroke or revolution. Even when the discharge valve is closed, the pump is trying to deliver its standard capacity and because it handles incompressible medium the discharge pressure increases rapidly to a point when the weakest point in the system would yield. It may be a pressure gauge, gasket, seal, lid, rotating parts, pipe, depending on a set-up. The conclusion is, an immediate damage to the pump or to the system follows. To prevent the damage, any system with a positive displacement pump should be equipped with a relief valve. This valve should be put between the pump and the first obstacle which usually is the valve itself.

4. Priming

Priming means filling the pump itself and the suction piping with handled liquid. If a suction tank is above the centerline of a pump, gravity will do the priming. But if the suction tank is below the pump as in Figure 1, then prior to or during a startup, the pump must prime itself if it is selfpriming or there must be some other means provided to prime it.

(a) Centrifugal pumps are not selfpriming and must be primed before a startup. Manufacturers sometimes offer "selfpriming centrifugal pumps" but on close inspection we always find some auxiliary equipment making the pump selfpriming. The most common design is a priming chamber which always holds a certain amount of water to prime the pump. A vast majority of pumps appearing in our plants work with suction head and priming is done automatically by gravity.

(b) Positive displacement pumps if in good shape are generally selfpriming. The manufacturer usually specifies a suction lift limit up to which a pump will prime itself.

5. Capacities

(a) Centrifugal pumps are available in a wide range of capacities, the largest ones being condenser cooling water pumps with the capacity of 13 m^3/s (~170,000 IGPM). No positive displacement pumps is available for such capacities.

(b) Positive displacement pumps are available for relatively small capacities, approximately up to 0.15 m^3/s (~2,000 IGPM).

6. Pressures - Total Heads

(a) A single stage centrifugal pump can develop only a limited pressure. For higher pressures, stages are compounded. But even then there is a practical limit to the size of a pump. In our plants we find 20 stage heat transport pressurizing pumps developing 10 MPa(g) (~1470 psig).

(b) Positive displacement pumps in general, plunger pumps in particular, are available for pressures up to 100 MPa (15,000 psi) and even higher.

7. Handled Liquids

(a) Centrifugal pumps can be designed to literally handle anything which has a slight tendency to flow. Slurries, paper stock, mixtures of abrasives, molasses are examples.

(b) Positive displacement pumps due to tight tolerances used in their design are vulnerable to abrasives and thick plugging liquids.

8. Maintenance

Because of a limited metal-to-metal contact and the overall simplicity of design, centrifugal pumps require less maintenance than positive displacement pumps.

To summarize, centrifugal pumps lend themselves better to most of the requirements in our plants and that is why the vast majority of pumps in our plants are of this type. But there are some systems where it is not possible to use them. In hydraulic systems where positive delivery of a relatvely small quantity of liquid is required at high pressure, in chemical injection systems where accurate metering is required, and as a standby in pressurizing systems, we find positive displacement pumps.

CENTRIFUGAL PUMPS

Centrifugal pumps come in a large variety of designs. There are several basic features according to which they can be classified:

1. Mount.
2. Staging.
3. Casing.
4. Impeller
5. Flow.
6. Energy Conversion.

1. Mount

Centrifugal pumps can be:

(a) horizontally mounted, Figure 2.
(b) vertically mounted, Figure 3.

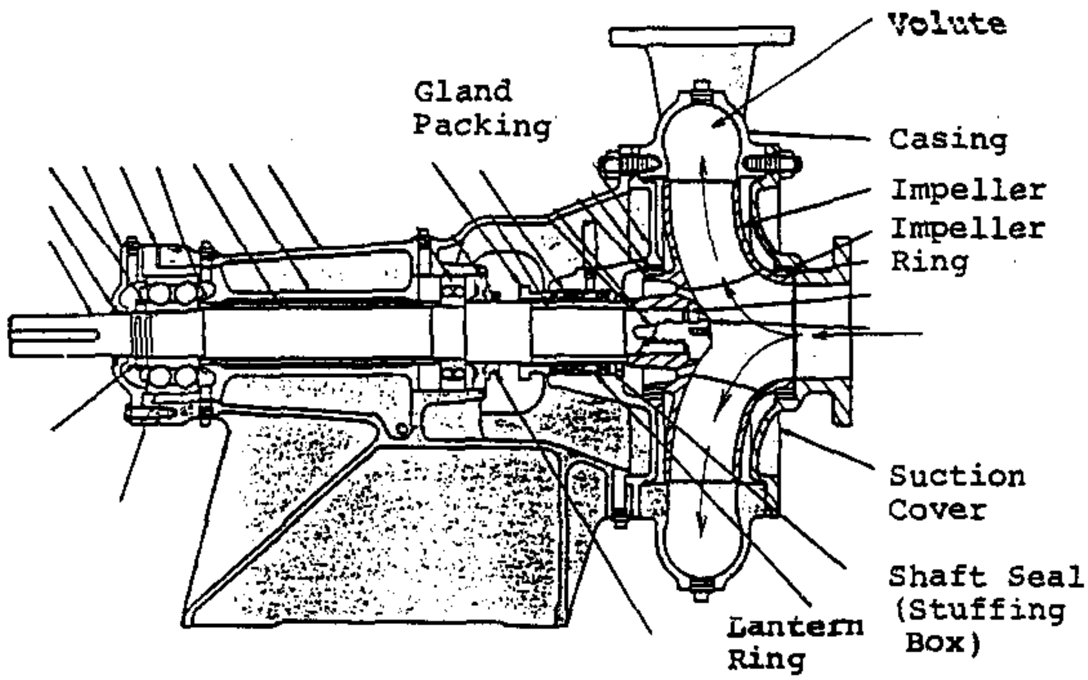


Figure 2

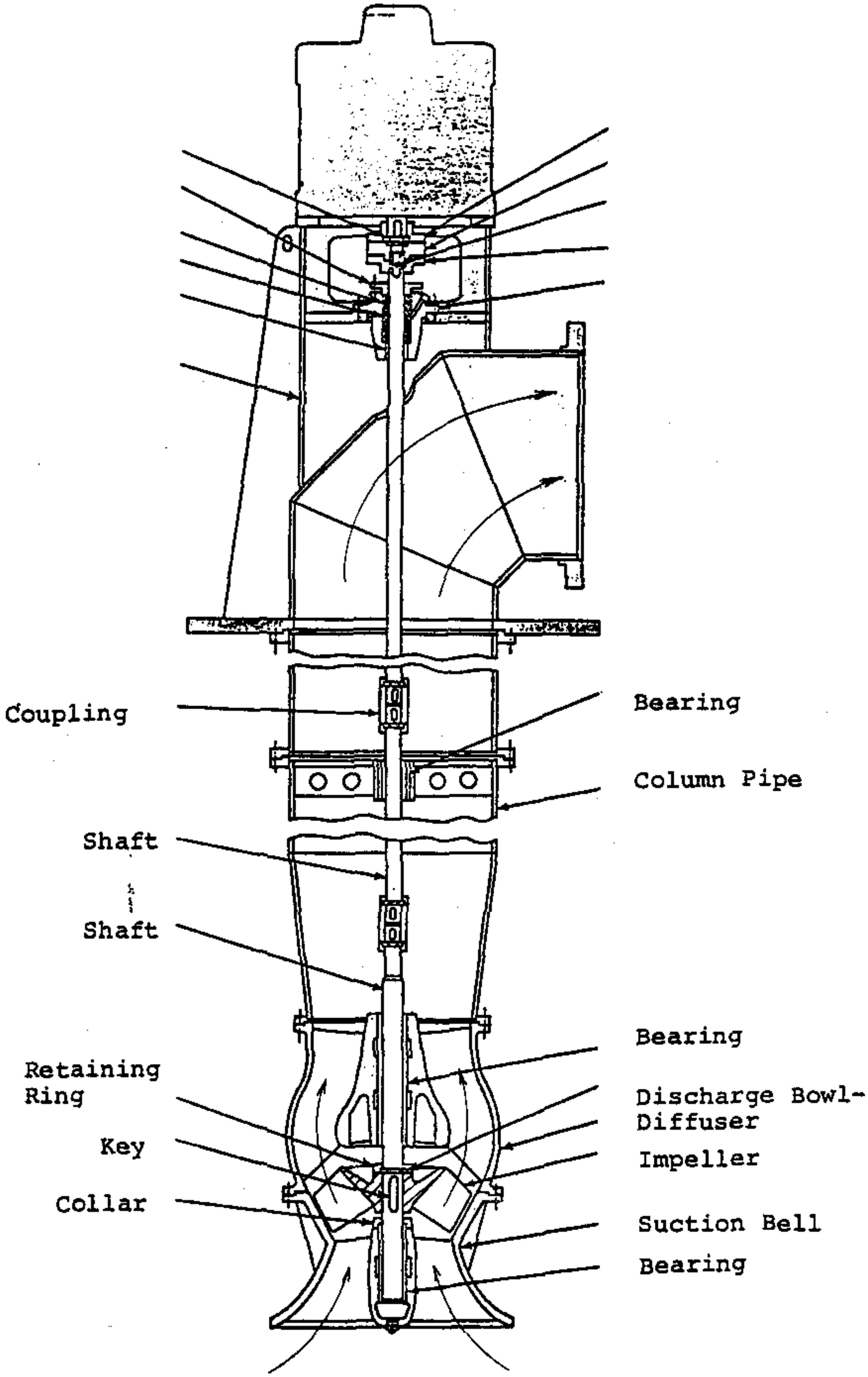

Coupling
Bearing
Column Pipe
Shaft
Shaft
Bearing
Retaining Ring
Discharge Bowl-Diffuser
Key
Impeller
Collar
Suction Bell
Bearing

Figure 3

2. Staging

Centrifugal pumps are:

(a) single stage, Figures 2, 3.
(b) multi stage (two or more stages), Figure 4.

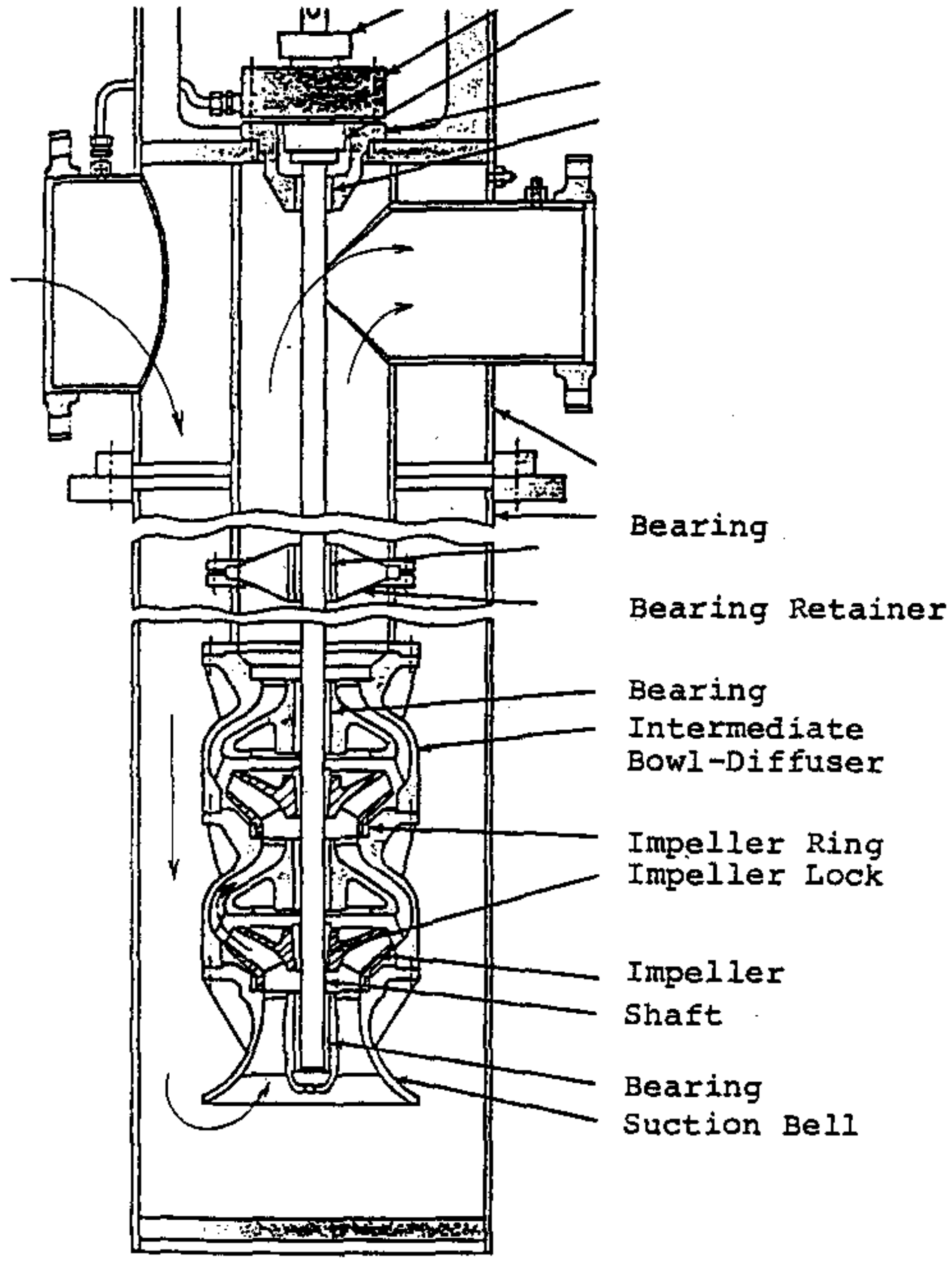


Figure 4

3. Casing

Types of casings found on centrifugal pumps are:

(a) Radially split casing, Figure 2. The plane of split is perpendicular to the shaft.

(b) Axially split casing, Figure 5. The plane of split runs through the axis of the shaft.

(c) Barrel type of casing, which is actually two casings, the internal axially or radially split and the external barrel. A simplified schematic of this type of casing is in Figure 6. It is used on high pressure, multi stage pumps.

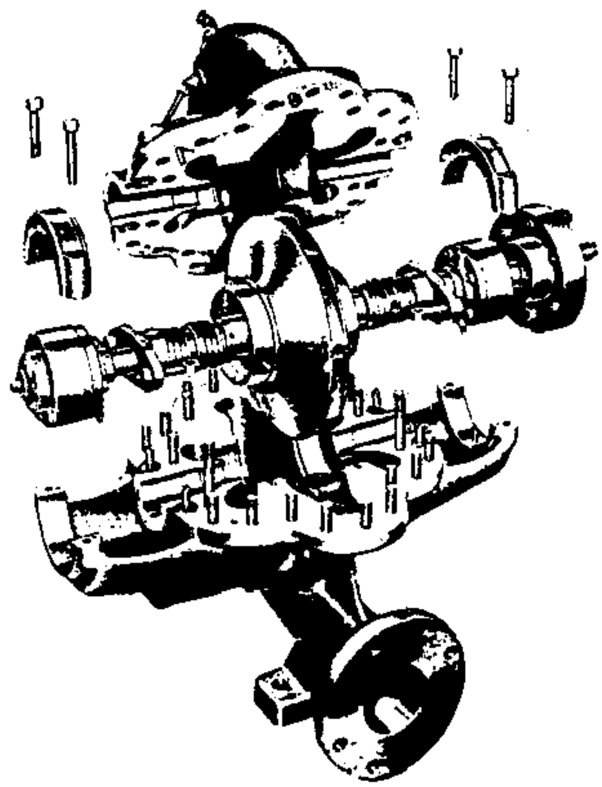


Figure 5

4. Impeller

According to the design, impellers can be:

(a) Fully Shrouded Impeller - Blades are sandwiched between two discs. They can develop higher pressures than other types of impellers. An example is in Figure 7 (a).

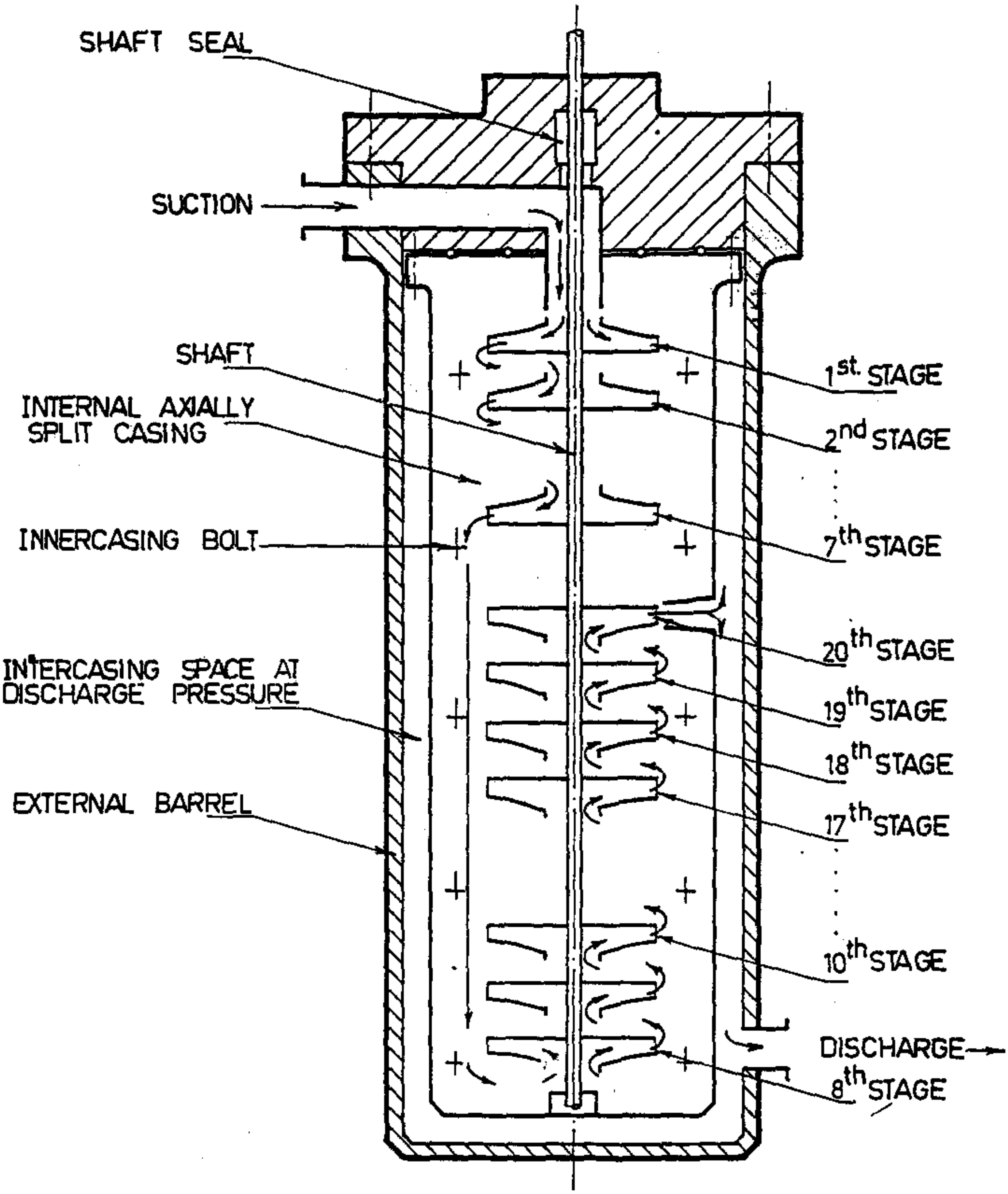

SHAFT SEAL
SUCTION
SHAFT
INTERNAL AXIALLY SPLIT CASING
INNERCASING BOLT
INTERCASING SPACE AT DISCHARGE PRESSURE
EXTERNAL BARREL
1st. STAGE
2nd STAGE
7th STAGE
20th STAGE
19th STAGE
18th STAGE
17th STAGE
10th STAGE
DISCHARGE
8th STAGE

Figure 6

(b) Semi-Shrouded Impellers - One disc is missing; blades are attached to one disc. They are cheaper, not able to develop as high pressures as fully shrouded impellers and are less efficient due to leakage along blades back to suction. An example is in Figure 7 (b).

(c) Open Impellers, Figure 7 (c), have blades attached to the hub with very little or no shroud at all. A special type of this kind is a propeller used on vertical pumps. It is shown in Figure 7 (d).

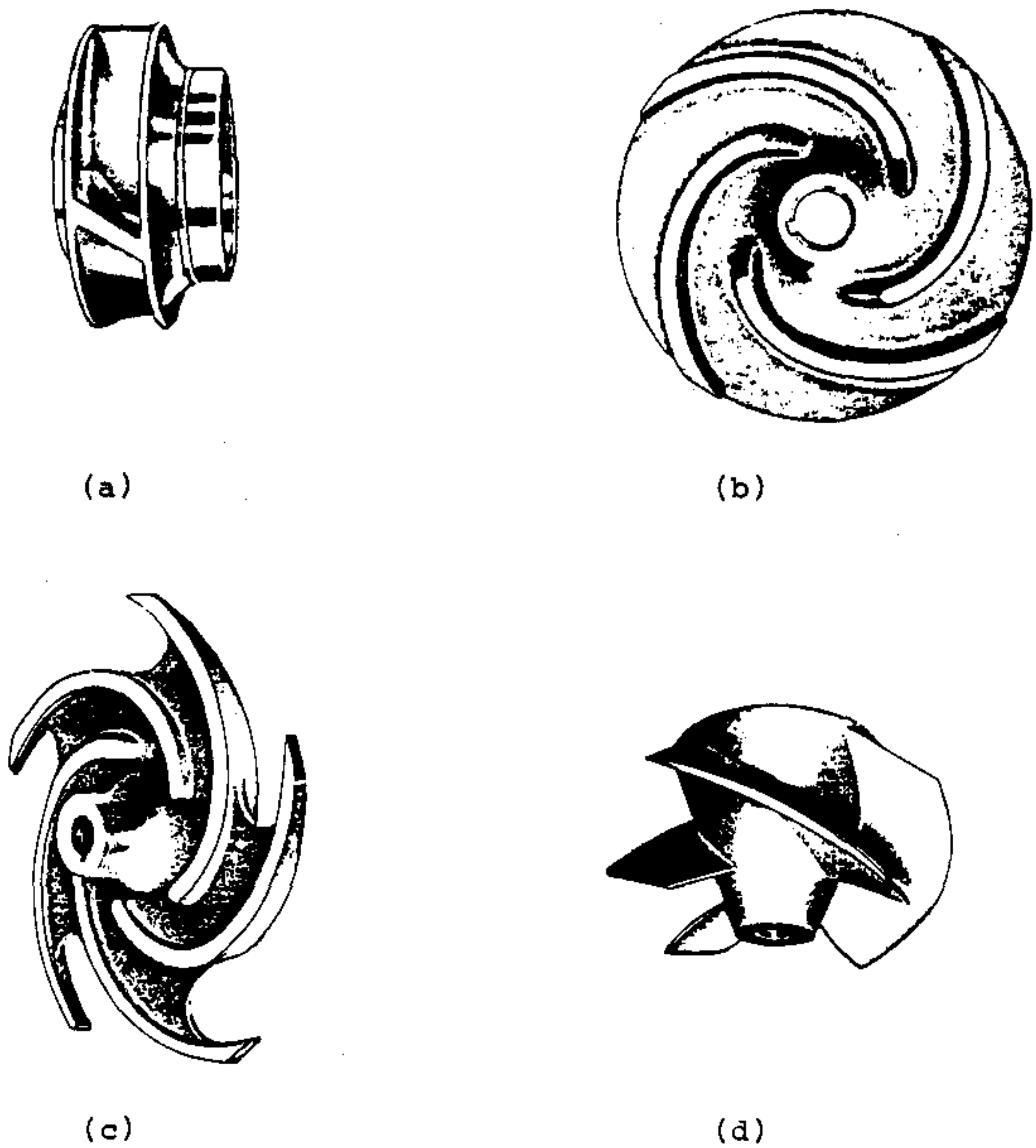


Figure 7

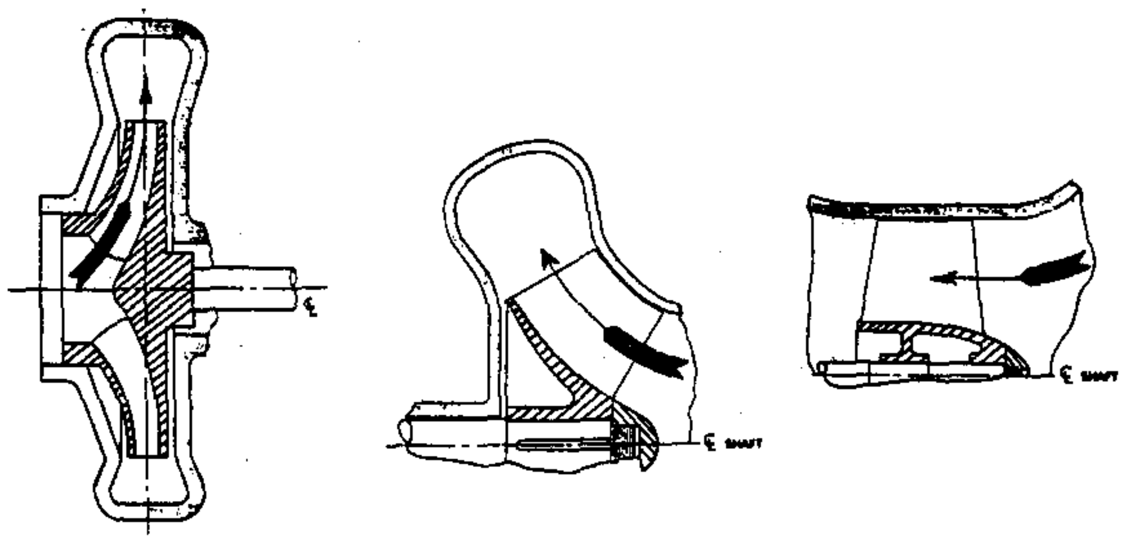
5. Flow

Depending on the flow through the impeller, centrifugal pumps are classified as:

(a) Radial Flow Pumps - flow enters the impeller axially and leaves radially as in Figure 8 (a).

(b) Mixed Flow Pumps - flow enters axially and leaves at some angle between radial and axial as in Figure 8 (b).

(c) Axial Flow Pumps - flow enters axially and leaves axially as in Figure 8 (c).



(a) (b) (c)

Figure 8

6. Energy Conversion

Centrifugal pumps are not pressurizing the liquid directly like positive displacement pumps. The mechanism creating a pressure head is as follows: liquid entering the impeller is flung into the volute with increased velocity (Figure 9). In other words the impeller imparts kinetic energy to the liquid. The liquid enters the volute, the cross section of which is increasing towards the discharge. This results in a decrease of velocity of the liquid and decrease of its kinetic energy. Because energy cannot be destroyed but only converted into another type of energy, decrease in

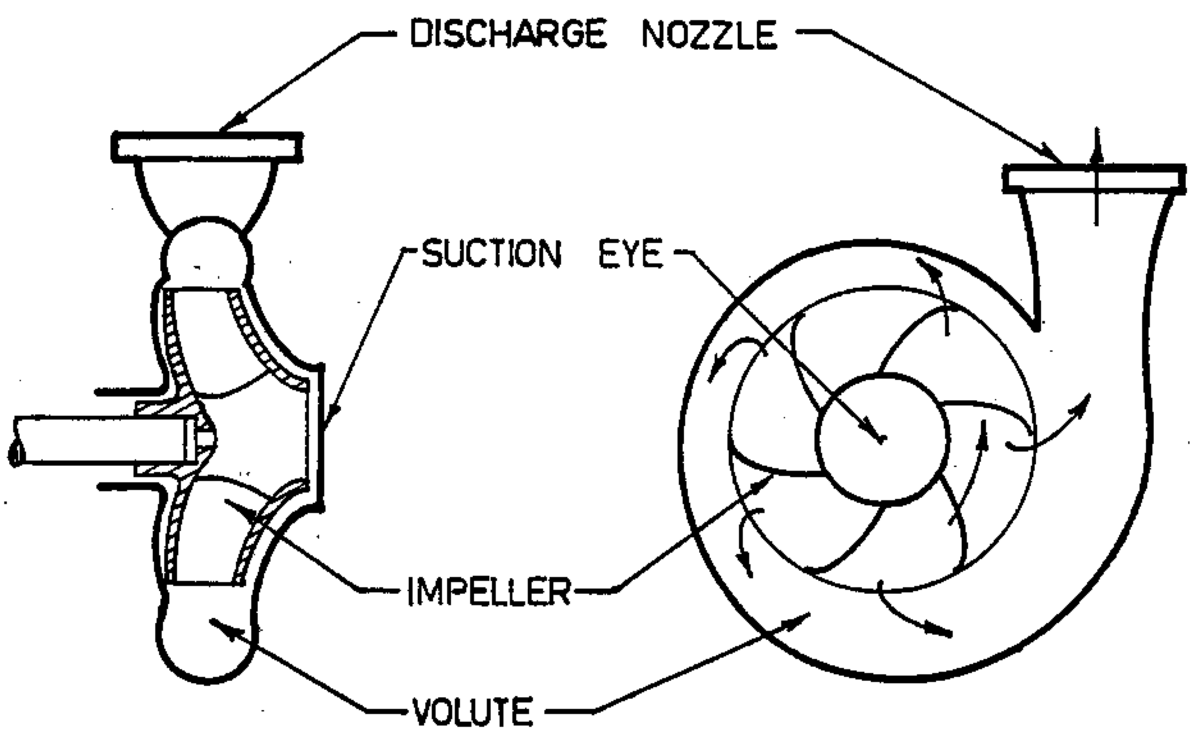


Figure 9

kinetic energy must result in the increase of some other type of energy. Total energy of flowing liquid in a pipe consists of several components. There is gravitational energy due to elevation of liquid above the ground, kinetic energy due to motion, pressure energy and heat energy. The elevation difference between the suction and discharge is very small and consequently the change in potential energy is insignificant. Also a change in temperature of in- and out-coming liquid is small, maximum 1°C, suggesting that the increase in heat energy due to internal friction is relatively small. So by the process of elimination the conclusion is that the change in kinetic energy results mainly in the change of pressure energy because it is the last remaining energy liquid can possess. If kinetic energy decreases, pressure energy must increase. Efficiency of a pump indicates how much of the supplied energy is converted into pressure energy and how much was wasted in friction which appears as heat in the liquid.

To summarize, the liquid is accelerated at first, then slowed down, the result is that the kinetic energy input at first is converted mainly into the pressure energy. This conversion by decelerating can be accomplished:

(a) in a volute, which is the scroll in a casing surrounding the impeller (Figure 9);

(b) in a diffuser, a stationary piece, adjacent to the impeller exit, which has multiple passages of increasing cross sectional area for converting velocity to pressure. Figure 10 shows the arrangement of a diffuser for a horizontal pump. Often this type is called a diffuser vane ring.

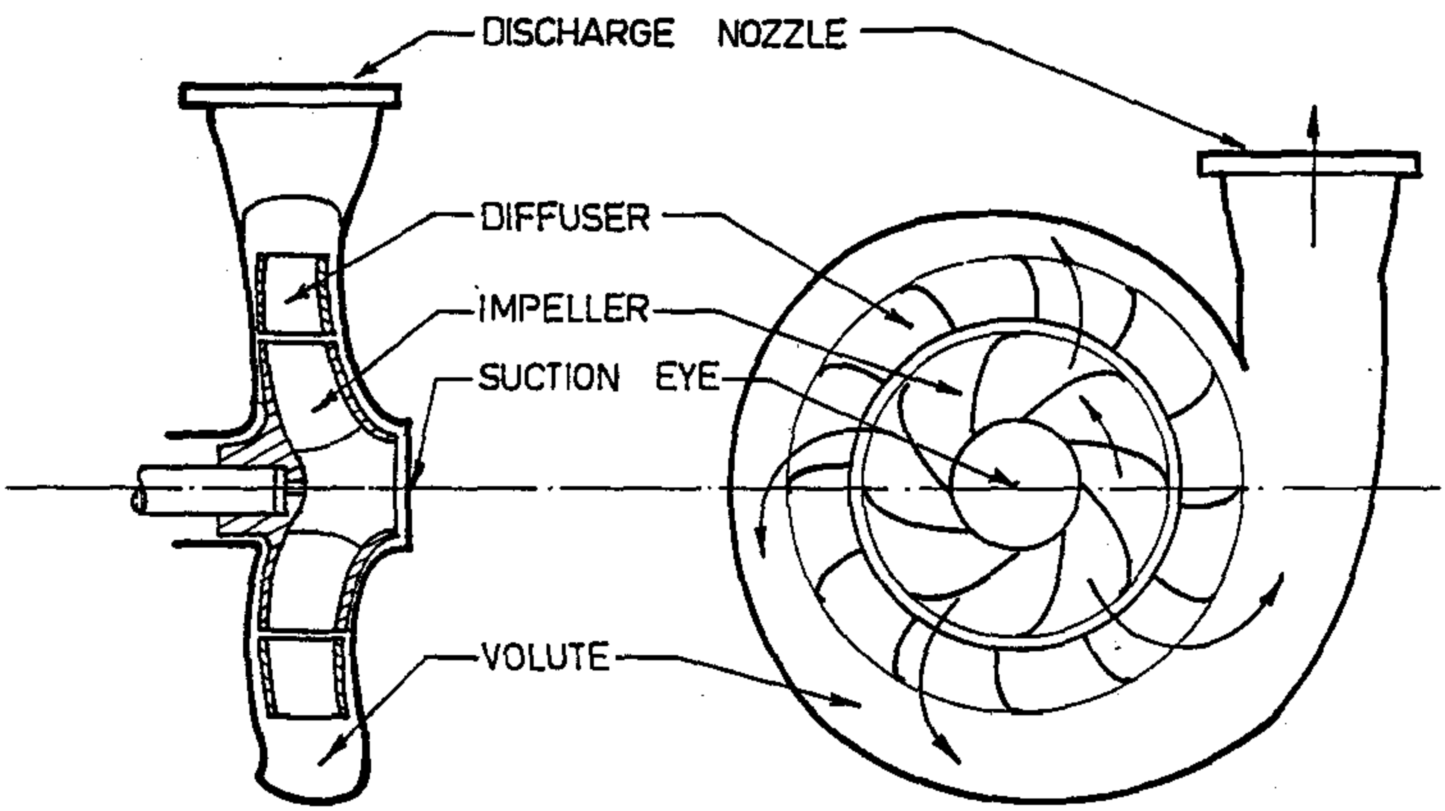


Figure 10

Another example of a diffuser on a vertical pumps is in Figure 3.

Diffusers are not common on horizontal units but are frequently found on vertical pumps.

OPERATION OF CENTRIFUGAL PUMPS

Before a centrifugal pump is started it is advisable to check a few things. All bolts on flanges and base are tight, covers and guards in place, shaft spinning freely, pump is primed. If after the startup the pump is not delivering at all or delivers less than expected, there are several operational problems to be suspected:

1. The pump is rotating in the wrong direction as a result of a mistake in the electrical hook-up.

2. The impeller is mounted the wrong way around. This is nearly impossible with single suction impellers, but a double suction impeller which is symmetrical can be reversed by mistake.

The described situations are shown in Figure 11.

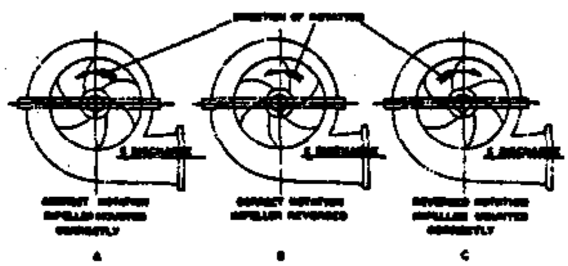


Figure 11

Experience shows that a vast majority of operational problems of centrifugal pumps originate in suction. They can be caused by:

1. Airlocking

   If a pump is working with a suction lift or takes suction from a vessel which is at a pressure way below the atmospheric pressure, suction piping and inlet to the pump might be under vacuum. In that case, any leakage due to poor piping joint or sealing arrangement will not be to the outside, but the ingress of air into the system will occur. Bubbles of air will also enter the system if there is not a sufficient submergence of the suction pipe or if bubbles of air are trapped by liquid falling into the suction tank are not separated before the liquid enters the suction pipe.

A centrifugal pump can handle a certain amount of air or, generally speaking, gas entrained in the liquid. But if the limit is exceeded, flow is interrupted and the pump is airlocked. If a pump works in other than air atmosphere, we talk about gaslocking.

2. Cavitation

In a system shown in Figure 1, the pressure at the surface of the liquid in an open tank will be the atmospheric pressure. As the liquid enters the suction piping it loses its pressure due to the friction losses and due to the lift. So the pressure progressively decreases and is the minimum as the liquid enters the impeller where the process of pressure buildup starts. The boiling point of any liquid depends on pressure and generally speaking it decreases with the decreasing pressure. If, during the passage through the suction piping, the pressure of liquid drops so much that it is equal or lower than boiling (= saturation) pressure corresponding to the temperature of the liquid, boiling will start. Small bubbles of steam, or generally vapour, will enter the impeller where the pressure is building up and again becomes higher than the saturation pressure of the liquid at the particular temperature. Vapour in the bubbles exposed to this higher pressure will condense leaving voids behind because the specific volume of liquid to vapour can be as high as 1 : 30,000. Surrounding liquid will immediately rush to fill these voids. This results in high speeds and resulting impacts. If a bubble and later a void happened to be at the metal surface of the impeller or casing, the inrushing liquid will hammer the surface. Millions of these impacts result in pitting and damage to pump parts. The whole process, starting with pressure drop and bubble formation up to the implosion and impact is called 'cavitation'.

Although the cavitation is most common in centrifugal pumps serving water systems it is stressed here that cavitation can occur in any component handling any liquid if the basic conditions exist, ie, drop in pressure under the saturation level and subsequent recovery of pressure above this level. Examples are cavitating piston pumps, valves, elbows, boat propellers and others.

It should be mentioned that components can be operating years under mild cavitation without a detrimental effect to their basic function. If it is not possible to get rid of cavitation by the change of design, material choice can make the lifetime of the component acceptable. As an example, alloy steels have better resistance to cavitation than mild carbon steels.

3. Vapourlocking

If the pressure in the suction piping drops grossly below the saturation pressure, vigorous boiling will take place and large bubbles of vapour will be generated. Similarly as in gaslocking, the vapour will fill the pump and interrupt the flow. If it happens unnoticed vapour and liquid will be churned within the pump, temperature will increase and the damage to the pump will follow.

Methods used to alleviate problems of cavitation and vapourlocking are explained at higher levels of this course where deeper background knowledge of operation characteristics of centrifugal pumps is available to a trainee.

ASSIGNMENT

1. Explain the terms used in pump operation:

    (a) suction lift.
    (b) suction head.
    (c) total head.

2. A pump is to be selected to provide cooling lake water for a number of station systems above water level. A large flow (25 000 GPM) at low pressure is required.

    (a) What type of pump would you choose? Why?
    (b) Is priming a problem? If so, suggest a solution.

3. How is the flow regulated in a system with:

    (a) centrifugal pump?
    (b) positive displacement pump?

4. A centrifugal pump delivers a large flow of cooling lake water (200 000 GPM) at low pressure. Suggest a pump design with respect to:

    (a) type of impeller.
    (b) direction of flow through impeller.
    (c) number of stages.

5. A throttle valve on the discharge side of a centrifugal pump is defective and closes. If the pump is still operational:

    (a) are there any immediate problems? Long term problems?

6. A pump under suction lift conditions is vibrating badly. A check with the control room indicates that motor current has dropped as well as discharge pressure which is unsteady.

    (a) Suggest a problem.
    (b) What are the possible long term effects?
    (c) Can you suggest a possible solution, if a pressurized tank is on suction.

7. Draw a simple sideview schematic of a centrifugal pump with appropriate labels which has:

   (a) fully shrouded impeller.
   (b) one stage.
   (c) radial flow through impeller.
   (d) radial split casing.
   (e) volute.

8. Explain the difference between airlocking and vapour locking.

9. Classify the pump shown in Figure 1 according to the six basic features.

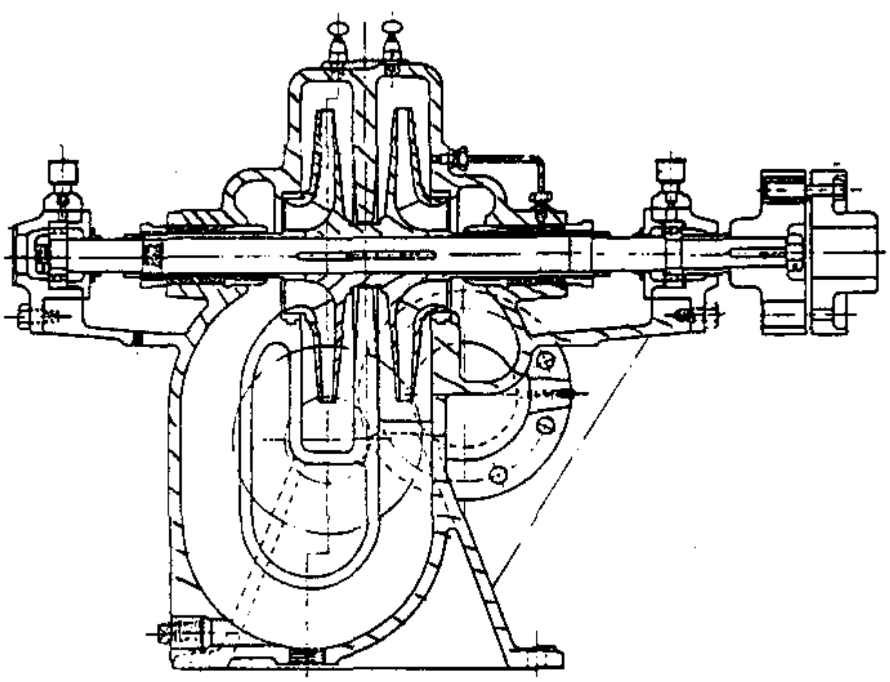


Figure 1

K. Mika

Mechanical Equipment - Course 330.1

CENTRIFUGAL PUMPS

INTRODUCTION

An outline of the principle of operation of the centrifugal pump, and of the nomenclature and classifications applied to describe the structure of the pump, was presented in 430.10-1.

In this chapter the classifications applied to centrifugal pumps are discussed in more detail. The major pump installations in the Pickering and Bruce Generating Stations and at the Bruce Heavy Water Plant are examined in terms of classification of the structure and the basic reasoning behind the selection of pump type for that particular application.

This chapter deals only with the pump prime mover and pump casing. Other pump components, namely bearing, seals, couplings, etc, are dealt with elsewhere in Mechanical Equipment course notes.

CLASSIFICATION OF STRUCTURE

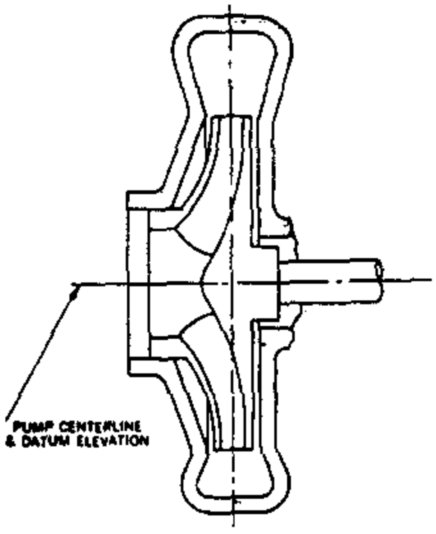
The classifications listed in Level 4, namely,

1. mount
2. staging
3. impeller type
4. direction of flow
5. casing split
6. energy conversion

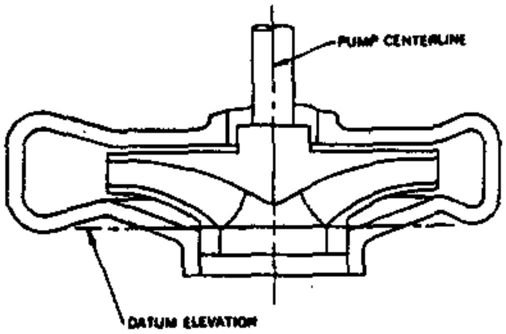
are reviewed below with a short discussion on the relative merits of each design type.

1. MOUNT: Figure 1: a) Horizontal
   b) Vertical

Space is obviously a major criterion in the choice between a vertical or a horizontal mount. A conventional horizontally mounted pump/motor set will occupy approximately three times the floor space which a similarly rated vertical pump will occupy. The headroom requirement for these pumps is, however, reversed.



HORIZONTAL UNIT



VERTICAL SINGLE SUCTION PUMP

FIGURE 1

A further important consideration in a very large pump installation is that of the method of pump/motor support.

The size and rigidity of base plate required to support a large pump/motor set horizontally leads to a more expensive installation. With a vertical mount, however, the full weight can be taken on tie bars slung from an overhead member with only relatively small lateral supports to maintain stability and alignment.

A further advantage of a vertical mount is that the application may allow the pump impeller to be immersed in the liquid to be pumped, thus eliminating the requirement for intake piping and pre-start priming.

On small pump/motor installations, however, the horizontal mount is often preferred due to the ease of maintenance which it can provide.

2. STAGING: a) Single Stage
b) Multistage

A single stage pump is one in which the head is developed by a single impeller. Often the total head to be developed requires the use of two or more impellers operating in series, each taking its suction from the discharge of the preceding impeller. For this purpose two or more single stage pumps may be connected in series or all the impellers may be incorporated in a single casing. The latter unit is called a Multistage Pump. See Figure 2.

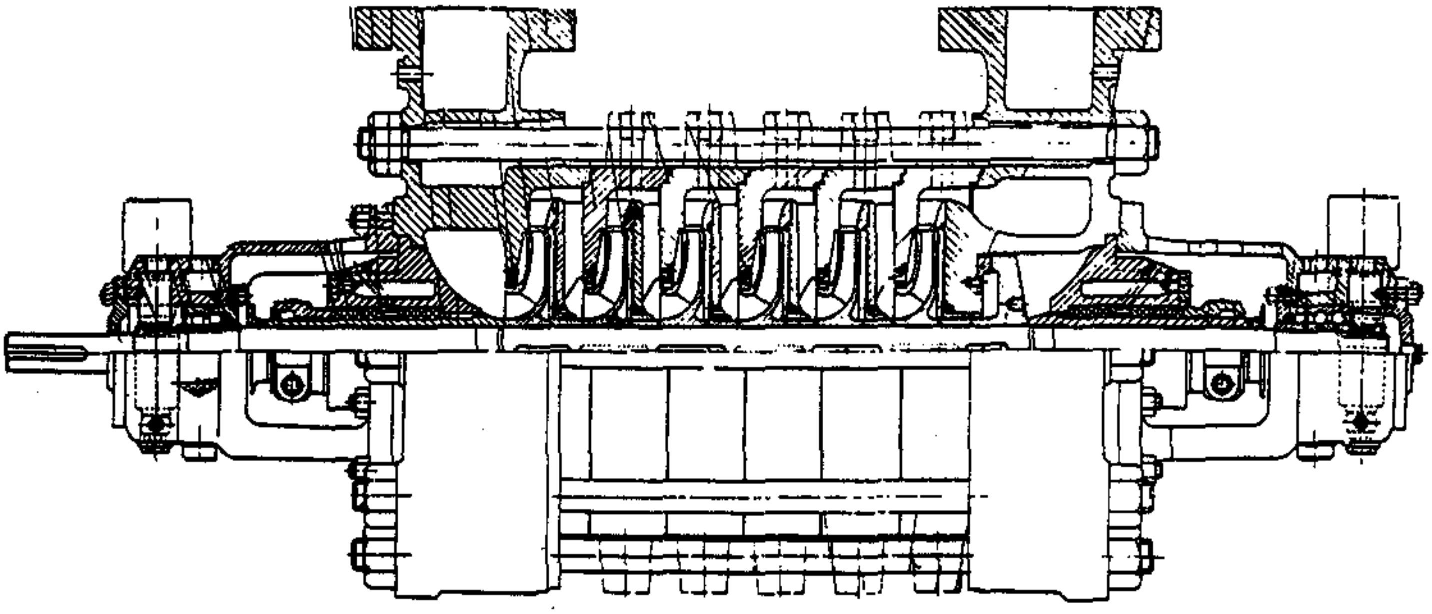


FIGURE 2: Multiple Stage Pump

When more than one impeller is used the impellers are often mounted on the shaft back to back or opposed (Figure 3). Opposed impellers have the effect of reducing the axial thrust on the shaft, thus decreasing the size of thrust bearing required.

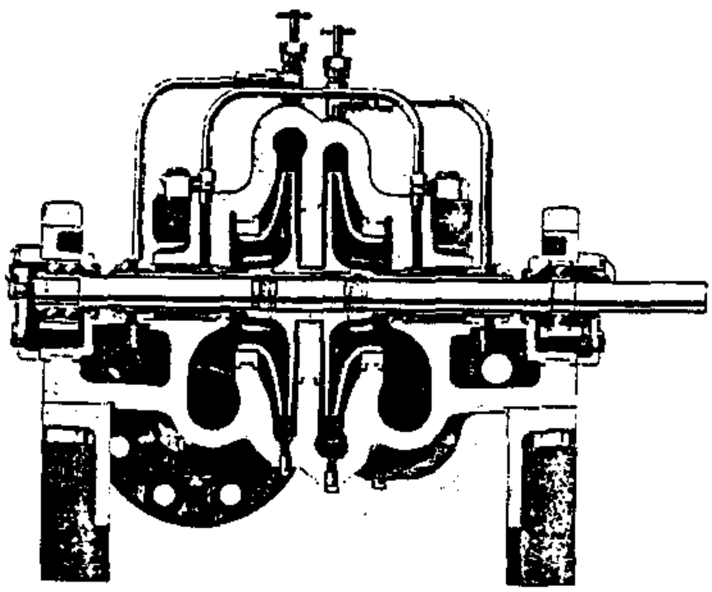


FIGURE 3: Two-stage horizontally split pump with opposed impellers

3. IMPELLER TYPE

The impeller of a centrifugal pump can be classified under the following categories:

a) single suction or double suction
b) shrouding design.

a) Single/Double Suction

In a single suction design the liquid enters the suction eye on one side of the impeller only. A double suction impeller is in effect, two single suction impellers arranged back to back in a single casting, the liquid entering the impeller from both sides. See Figure 4.

The two suction casing passageways are supplied from a common intake pipe.

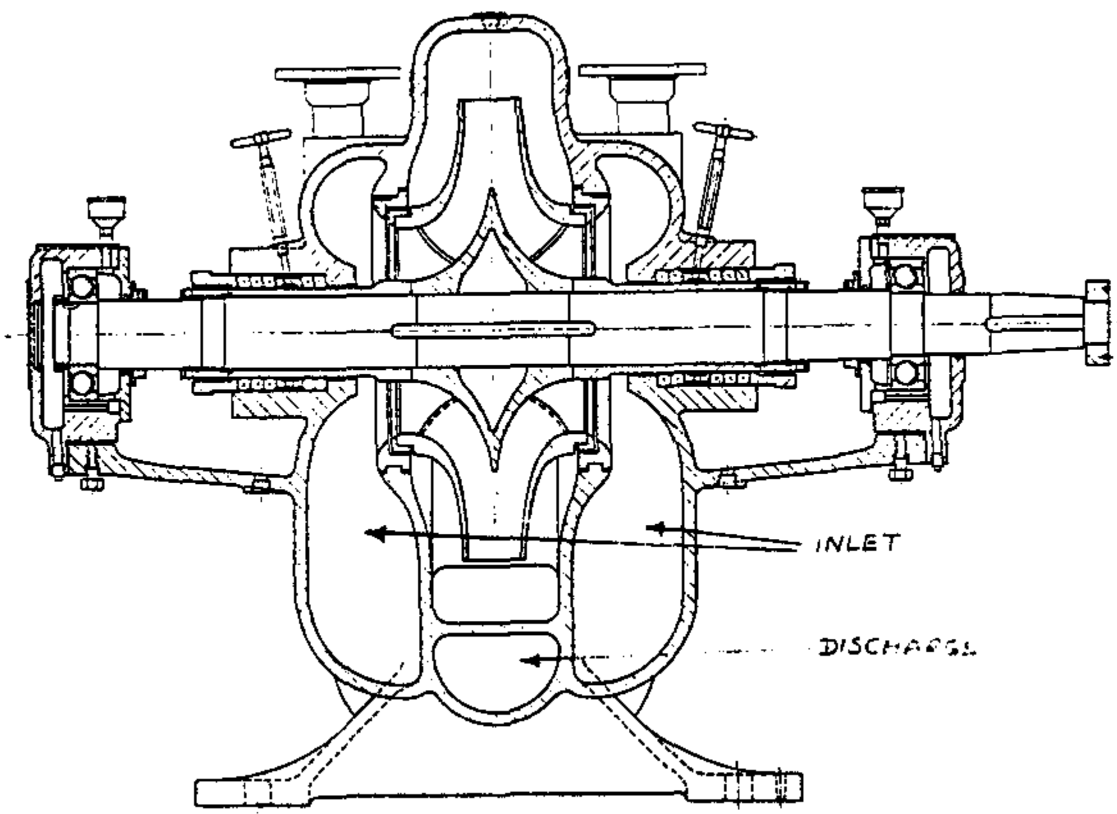


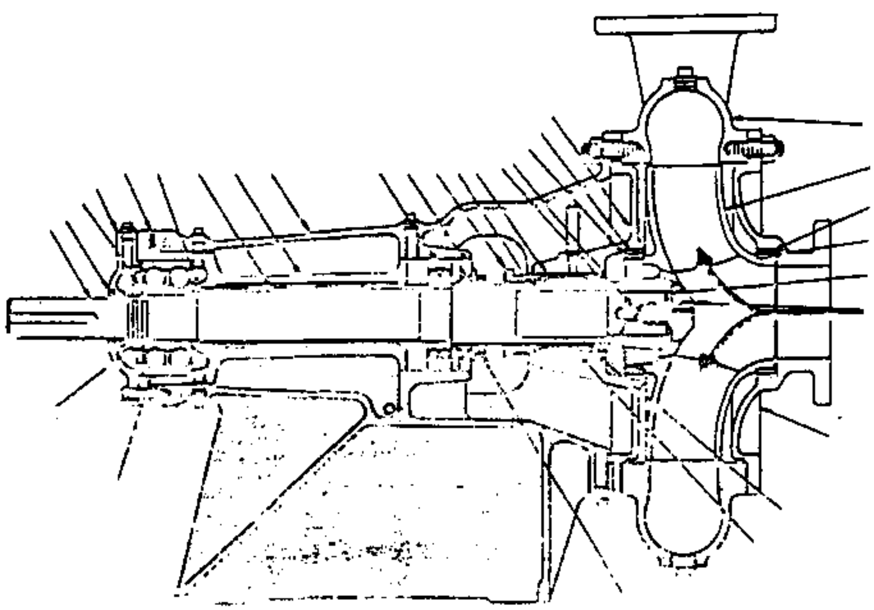
FIGURE 4: Double Suction Pump

A <u>double suction</u> impeller may be favoured for two reasons:

1. It is in axial hydraulic balance, reducing the size of thrust bearing required.

2. The greater impeller suction area (compared to a single suction design) permits the pump to operate with a lower pressure at the suction, for a given capacity, without cavitating.

An advantage of a <u>single suction</u> pump is that an overhung impeller design may be used - where the impeller is mounted on the end of the shaft. (Figure 5). Flow is allowed directly from the suction pipework to the eye of the impeller uninterrupted by the shaft. This reduces the turbulence at the entrance to the impeller and so reduces pressure losses.



<u>FIGURE 5:</u> Single Suction Pump With 'Overhung' Impeller

b) Shrouding Design

Impellers may be:
(i) Open
(ii) Semi Shrouded
(iii) Fully Shrouded

(i) Open Impeller - Figure 6 A, E, F, G

Consists of vanes attached to central hub without any form of integral backplate or shroud.

The liquid is channelled through the impeller by static ducting or shrouds, past which the impeller rotates with a clearance small enough to minimize back slip of liquid. The slippage increases as wear increases. To restore the pump to its original efficiency both the impeller and the sideplates must be replaced, involving a considerable expense.

The main disadvantages of the impeller is its structural weaknesses. If the vanes are long they must be strengthened by ribs or a partial shroud.

(ii) Semi Shrouded Impeller - Figure 6 B

Incorporates one shroud or integral sidewall. This shroud increases the strength of the impeller. Some slippage across the open face will still occur but the efficiency of the impeller is higher than that of the open impeller.

(iii) Fully Shrouded Impeller - Figure 6 C & D

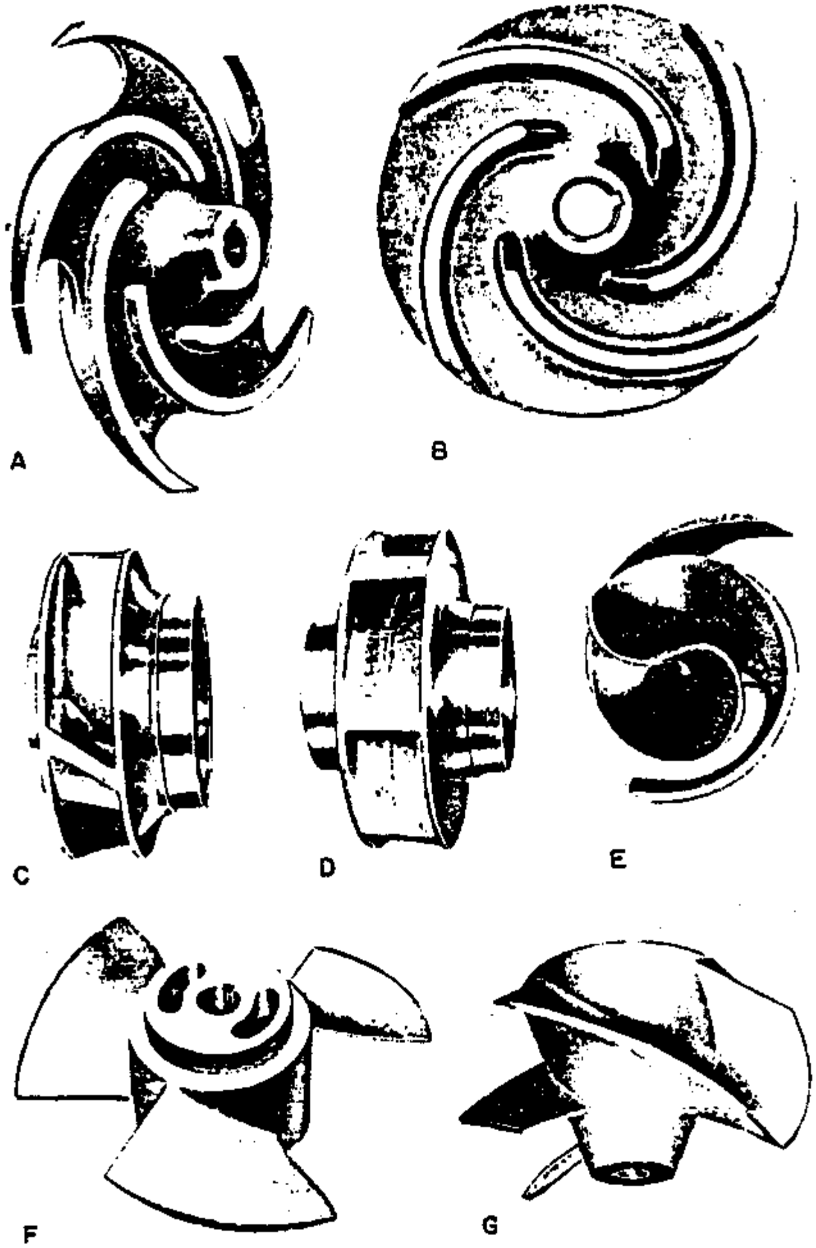
Used for handling clear liquids. Incorporates integral shrouds which totally enclose the impeller waterways from the suction eye to the periphery.

This design provides the highest efficiency in that no slippage can occur between the vanes and the shrouds. This design is, of course, expensive.

4. DIRECTION OF FLOW THROUGH IMPELLER

The direction of liquid flow through the impeller leads to the classifications:

i) Radial Flow Impeller
ii) Mixed Flow Impeller
iii) Axial Flow Impeller (Propeller)



A - Open Impeller with partial shroud

B - Semi shrouded impeller

C - Fully shrouded impeller

D - Double Suction fully shrouds impeller

E - Open impeller for viscous liquids

F - Axial flow impeller

G - Open mixed flow impeller

FIGURE 6

(i) Radial Flow Impeller

The liquid enters the impeller axially at the eye and flows radially to the periphery. The fluid is given Kinetic energy by the action of the centrifugal force causing the fluid to accelerate radially through the impeller. The kinetic energy gained in the impeller is converted to pressure energy as the liquid passes through the volute, (Figure 7a) which has a gradually increasing cross sectional area, reaching a maximum at the pump discharge.

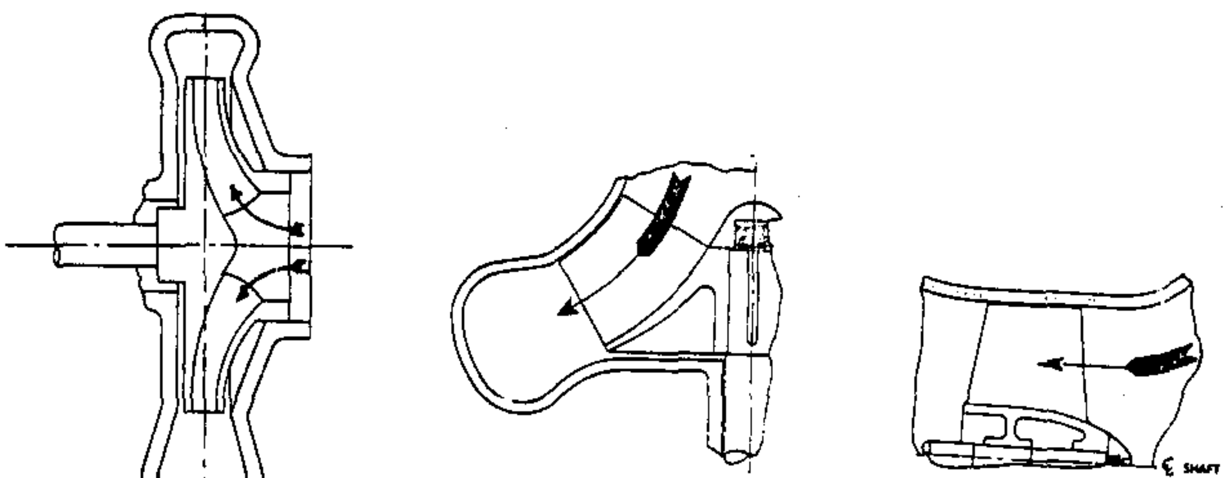
(ii) Mixed Flow Impeller

The head is developed partly by centrifugal force and partly by the lift of the vanes on the liquid. The flow enters axially and leaves with both a radial and axial component. (Figure 7b).

(iii) Axial Flow Impeller

The head is developed by the propelling or lifting action of the vanes on the liquid. The flow passes axially through the impeller with little change in direction. (Figure 7c).

The Radial Flow impeller is generally used when higher discharge pressures are required together with relatively low flow rates. An axial flow impeller would be used in high flow rate applications when just sufficient head is required to overcome friction losses. The mixed flow impeller is a compromise where some pressure rise, together with a reasonably high flow rate, is required.



a. Radial Flow Impeller

b. Mixed Flow Impeller

c. Axial Flow Impeller

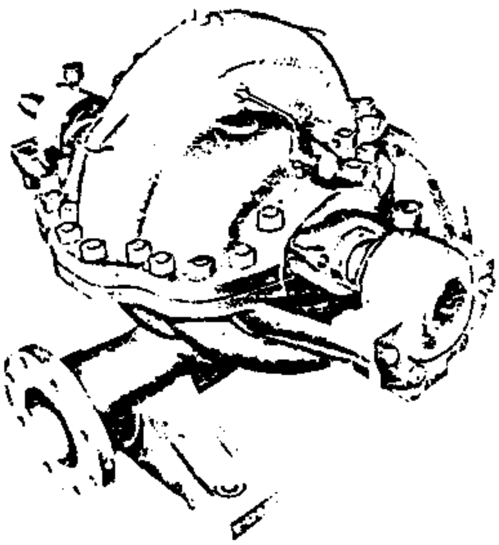
FIGURE 7

5. CASING SPLIT: a) Axial Split
b) Radial Split
c) Double Casing - Barrel

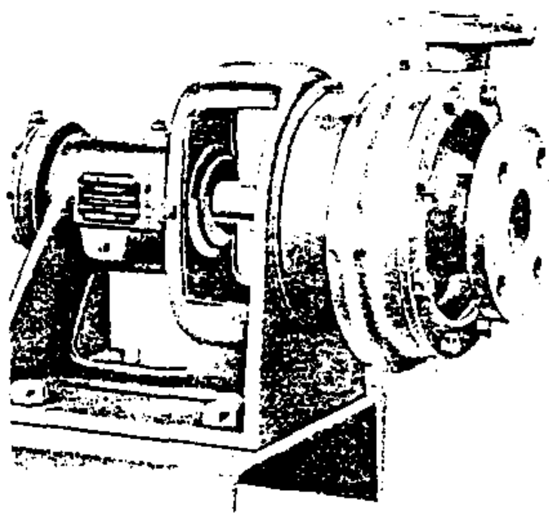
a) Axial Split - Figure 8a.

The joint between the two casing halves runs parallel to the shaft. The axially split casing has the advantage that a complete half casing may be removed to allow inspection of the pump internals without disturbing the bearings, seals or pipework.

The main disadvantage, however, is that the high pressure within the pump tends to force the casing halves apart, reducing the squeeze on the joint and leading to possible jointing problems.



a. Axial Split Casing



b. Radial Split Casing

FIGURE 8

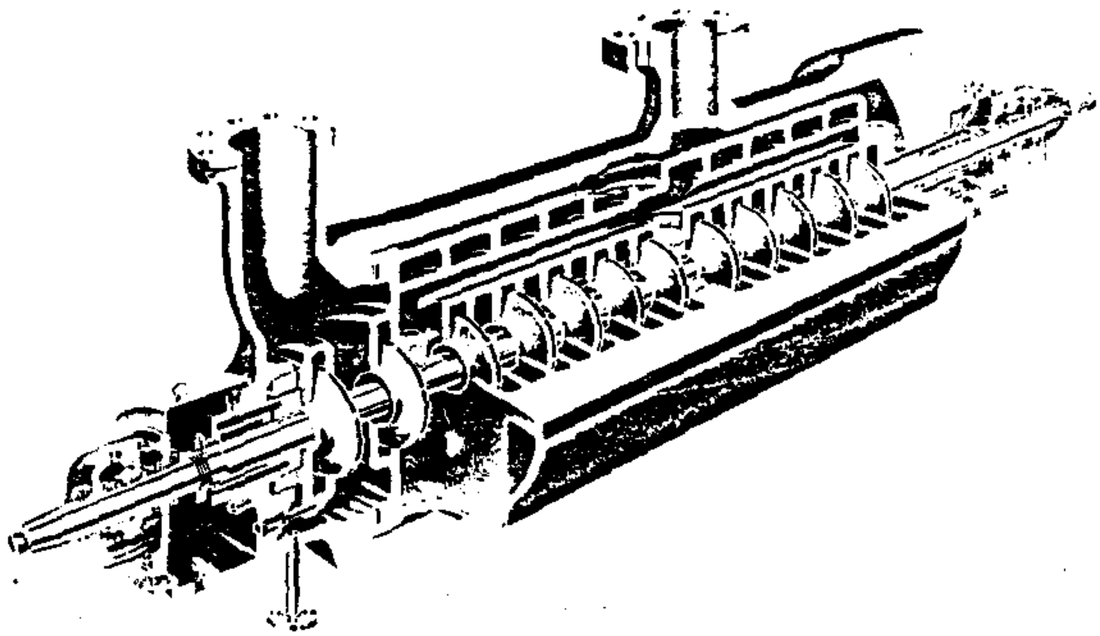
b) Radial Split - Figure 8b

The casing is split in a plane perpendicular to the axis of the pump.

A radially split casing has the advantage that the ducting (volute) which carries the high pressure liquid discharge from the impeller periphery to the discharge pipework is made from one casting.

This feature leads to an inherently stronger design than the axially split casing. The radially split casing generally requires that progressive removal be carried out of pump components, eg, pipework, bearing assembly, seals, etc, to gain access to the pump impeller. This disadvantage is particularly significant in a multistage pump.

c) Double Casing or Barrel - Figure 9



Double-casing multistage pump with axially split inner casing

(Courtesy Allis-Chalmers.)

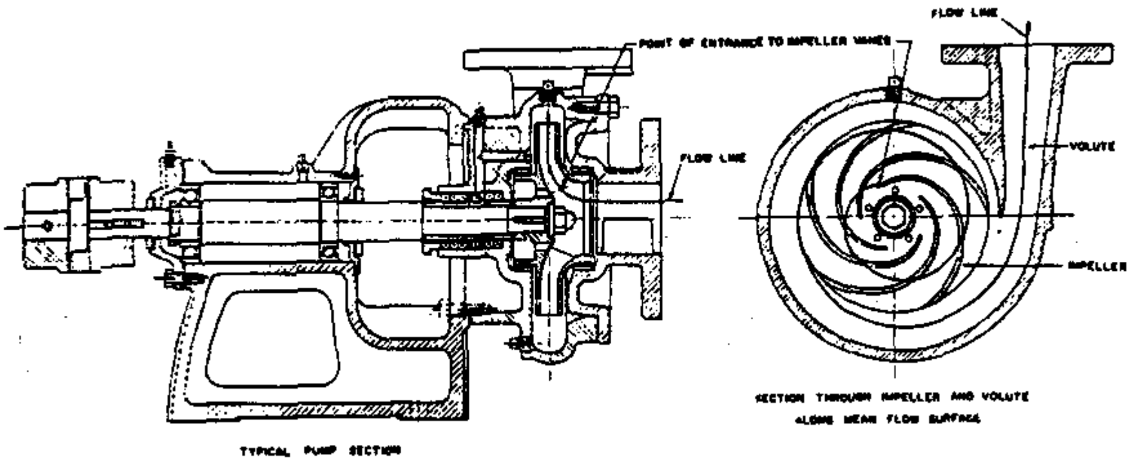
FIGURE 9

The Double Casing or "Barrel" casing has evolved out of the requirement for the convenience and expediency of an axial split casing together with the strength of the radial split casing. The basic principle consists of enclosing the working parts of a multistage centrifugal pump in an axial split casing and then locating a second radial casing or barrel around the inner casing. The space between the two casings is maintained at the discharge pressure of the pump. This arrangement ensures that the inner casing is under compression and the axial flanges will remain tight. This system does not, however, completely insure against interstage leakage.

6. <u>ENERGY CONVERSION</u>: a) Volute
b) Double Volute
c) Diffuser Vanes

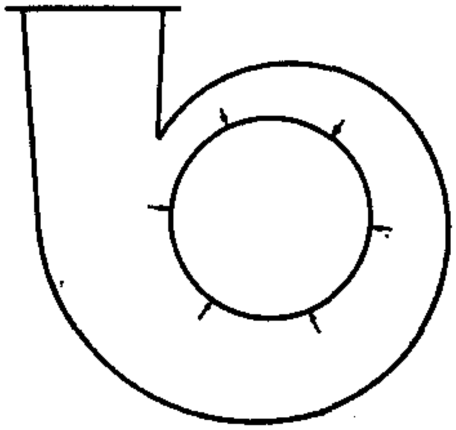
a) <u>Volute</u> - Figure 10

The volute is a chamber surrounding the impeller of which the cross sectional area increases steadily towards the discharge. This increase in cross-sectional area causes the conversion of the Kinetic Energy of the liquid to Pressure energy as it is directed from the impeller periphery to the pump discharge. The volute is used generally in pumps with radial flow impellers.



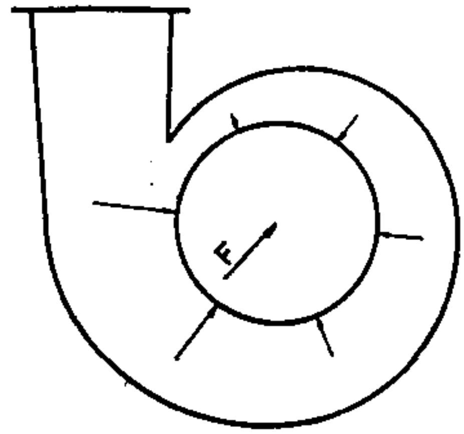
<u>FIGURE 10</u>

When a single volute pump casing design is used uniform pressures act around the periphery of the impeller only when the pump is operating at design capacities. At other capacities the pressures around the impeller, and therefore the radial forces acting on the impeller, are not uniform, and there is a resultant radial thrust. This thrust is usually the greatest at shut off (zero flow). (Figure 11). This thrust is not of great significance in small low pressure pump applications but pumps utilizing a large impeller with a high pressure rise across the pump require the use of a Double Volute.



Zero radial reaction in single-volute casing

Uniform pressures exist at design capacity.



Radial reaction in a single-volute casing

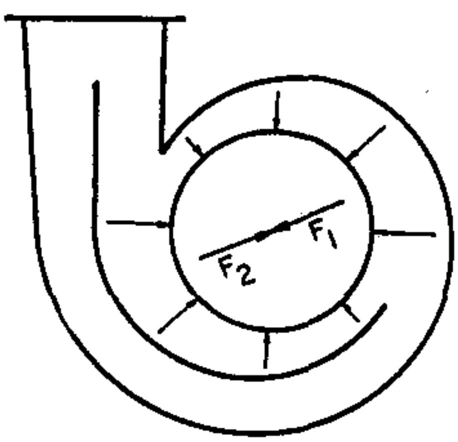
Uniform pressures do not exist at reduced capacities.

FIGURE 11

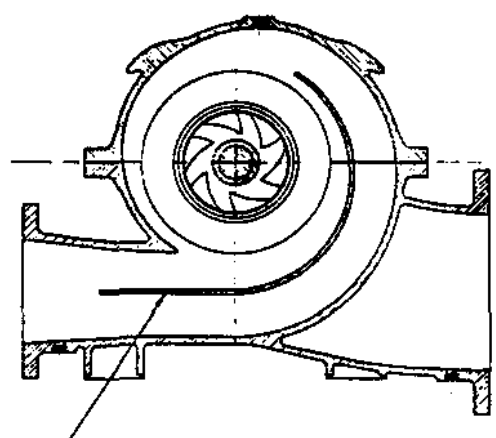
b) Double Volute

The Double Volute casing incorporates a wall which divides the original single volute into two 180° volutes. This design contributes towards a balance in the radial thrust exerted on the impeller. The reduction in radial thrust allows the use of a smaller diameter pump shaft with a smaller bearing surface area. (Figure 12).

An additional benefit is that the central rib in the volute helps to strengthen the pump casing.



Radial reactions in double-volute pump



Transverse view of double-volute casing pump

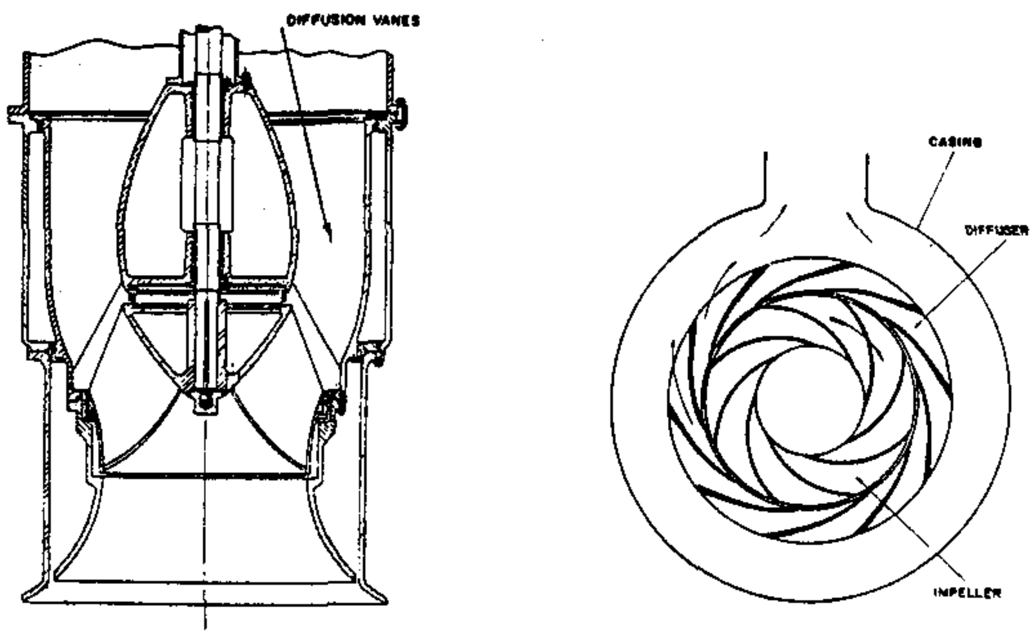
FIGURE 12

c) Diffuser Vanes

Diffuser vanes may be fitted to the discharge from a centrifugal pump impeller to reduce the turbulence generated in the flow as the liquid emerges from the impeller.

The diffuser is now seldom used in radial flow pumps since impeller/volute design has advanced to the degree that addition of diffuser vanes will not improve efficiency significantly. They are, however, used extensively in mixed and axial flow applications where a volute is impractical. (Figure 13).

A disadvantage of the diffuser vane is that the diffuser will generally improve the pump efficiency only at design flow conditions, when the angle of the vanes corresponds to the angle at which the liquid leaves the impeller. At other capacities the diffuser vanes can cause shock and increase turbulence.



a. Vertical mixed flow pump with diffuser

b. Radial flow pump with diffuser in volute

FIGURE 13

AXIAL THRUST IN CENTRIFUGAL PUMPS

Axial hydraulic thrust is the resultant of the forces acting on the impeller in the axial direction. (Figure 14a).

Reliable large capacity thrust bearings are now readily available so axial thrust in single stage pumps remains a problem only in larger units.

Methods by which axial load on a thrust bearing may be reduced are:

a) Double Suction Impeller - Figure 14b

Theoretically a double suction impeller is in hydraulic axial balance, with the pressures on one side equal to the pressures on the other. In practice the balance may not be achieved due to unequal or non uniform flows to the two sides caused by external conditions, such as an elbow being too close to the pump suction or due to internal casing conditions, such as assymetry of the suction passages or volutes.

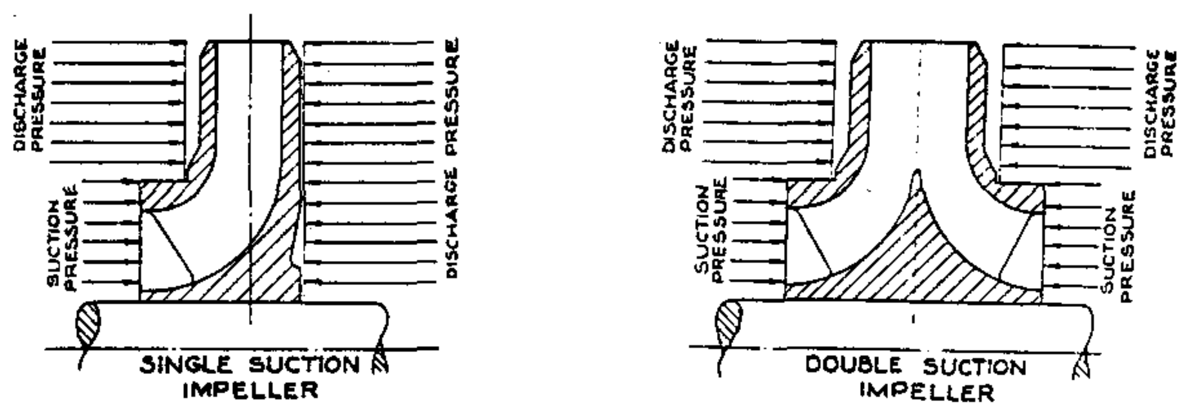


FIGURE 14

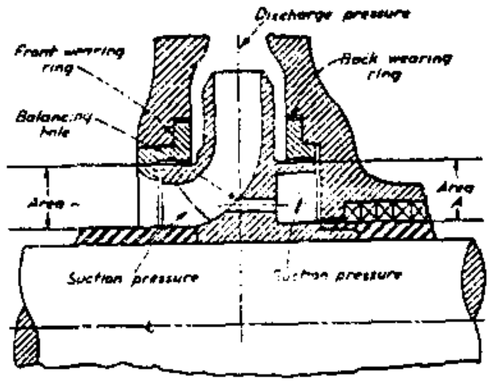
Combined, these factors create axial unbalance. To compensate for this, all centrifugal pumps, including those with double suction impellers, incorporate thrust bearings.

b) Balancing Holes

The high pressure behind the impeller is reduced by allowing liquid from the back of the impeller to be bled through to the front of the impeller by drilled holes.

c) Black Waring Rings - Figure 15

A single suction impeller can be provided with both front and back wearing rings. To equalize thrust areas, the diameter of both rings is made the same. Pressure approximately equal to the suction pressure is maintained in a chamber located under the back wearing ring by the use of balancing holes through the impeller.

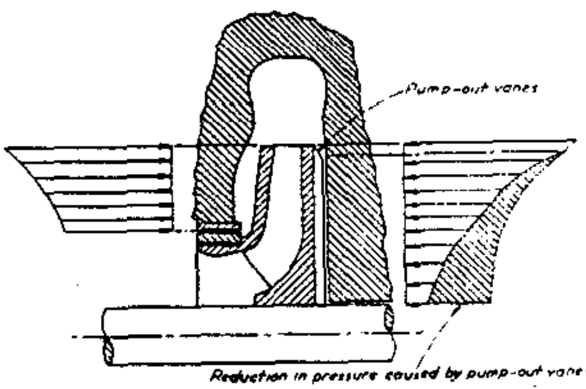


Balancing axial thrust of single-suction impeller with wearing ring on the back and balancing holes

FIGURE 15

d) Pump Out Vanes - Figure 16

Pump out vanes on the back shroud of single suction impellers have the effect of reducing the pressure acting on the back of the impeller by opposing the flow of the handled liquid from the volute towards the shaft behind the impeller.



Reducing axial thrust of single-suction impeller with pump-out vanes

Open impeller with partial shroud

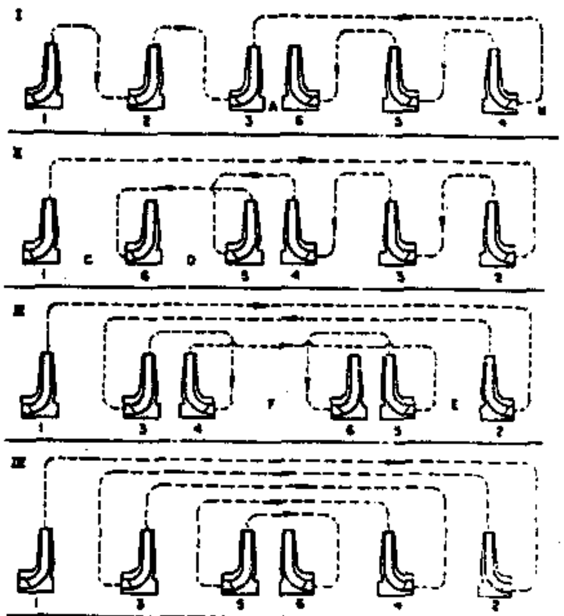
Front view (left) and back view (right) show pump-out vanes.

FIGURE 16

e) Opposed Impellers - Figure 17

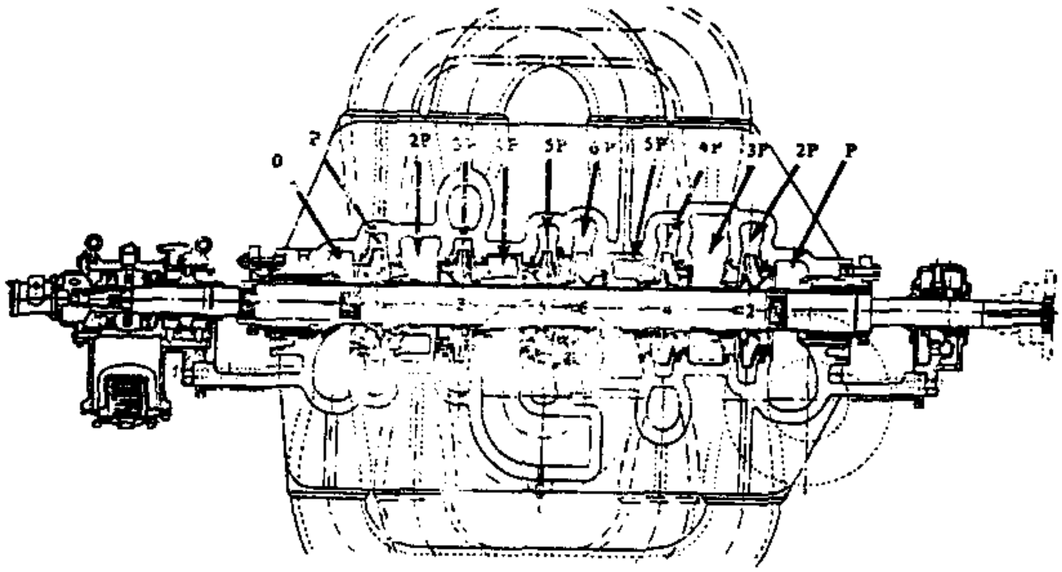
In multistage pumps axial thrust may be minimized by arranging the impellers on the shaft such that a number of impellers face the opposite direction to the remainder.

The sequence in which the individual impellers are to be arranged in the pump is decided by the manufacturer as a result of analysis of the number of running joints acceptable, the pressure differential across running joints and the pressure to which the end seal (stuffing box or mechanical seal) will be subject.



I. Joint A subject to three-stage pressure differential; one stuffing box under high pressure at B. II. Arrangement with two high-pressure joints, including four-stage pressure differential at C and two-stage differential at D. III. Joints E and F under two-stage pressure differential. IV. All running joints subject to only one-stage pressure differential.

Stage arrangements for six-stage axially balanced pump



Section of six-stage opposed-impeller pump

Suction pressure equal zero; pressure generated by each impeller is indicated by P.

FIGURE 17

f) Hydraulic Balancing Devices

i) Balancing Drum - Figure 18

A balancing chamber at the back of the last stage impeller is either keyed or screwed to the shaft and therefore rotates with the shaft. A small radial clearance separates the rotating drum from the static casing.

The pressure in the balance chamber situated behind the drum is maintained at pump suction pressure by interconnecting pipework. Thus a differential pressure exists across the drum creating an axial end force on the shaft. By careful design of the balancing drum can be made to balance the axial rotor thrust which would exist without the drum.

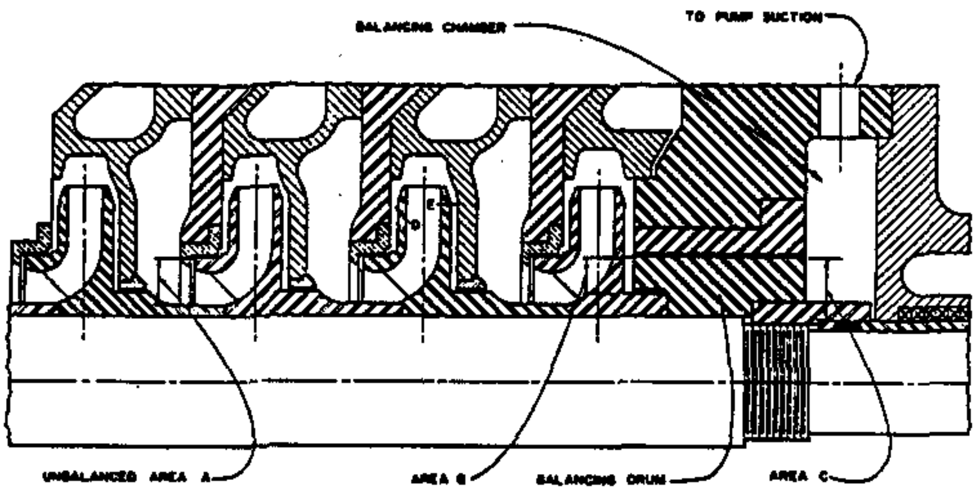


FIGURE 18: Balancing Drum

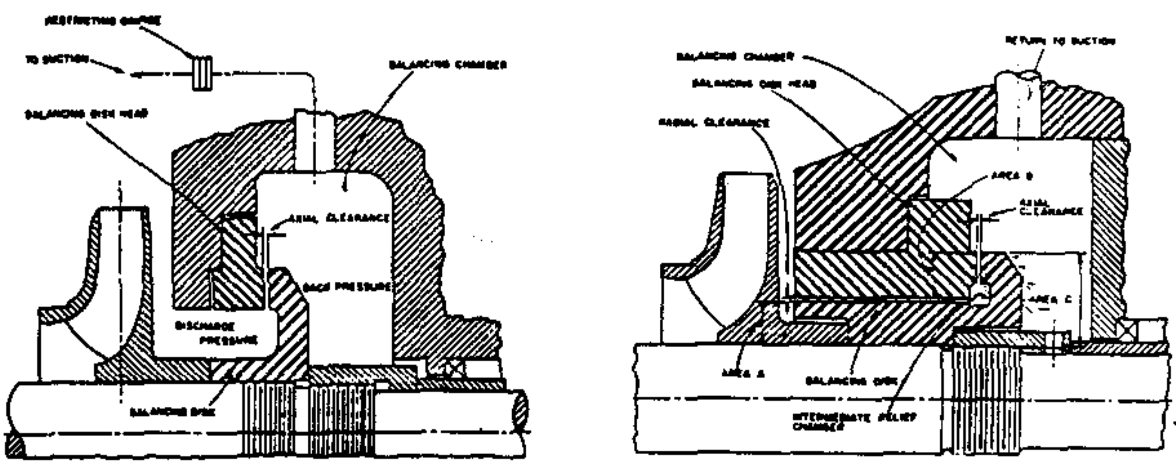
ii) Balancing Disc - Figure 19

The balancing disc is fixed to, and rotates with, the pump shaft, and is separated from the stationary balancing disc head by a small axial clearance. The liquid from behind the pump impeller can leak through this clearance to the balance chamber, from where it can flow back to the pump suction.

The pressure in the balance chamber will therefore act on the whole of the rear face of the balance disc and full pump discharge pressure will act on the smaller exposed area at the front of the disc. Small axial movement of the rotor shaft will adjust the clearance between the balancing disc head and the balance disc thus altering the flow through the clearance and causing the pressure in the balance chamber to vary. Altering the pressure in the balance chamber will change the axial force on the back of the balance piston.

Referring to Figure 19a, if the axial force on the impeller increases to the right, then the shaft will move in that direction increasing the axial clearance between the balancing disc and balancing disc head. The increase in clearance increases the flow into the balance chamber and hence increases the pressure in the balance chamber. This in turn increases the axial force on the back of the balancing disc and opposes the original increased impeller axial thrust.

It can be seen, therefore, that the balancing disc provides automatic compensation for any change in axial thrust caused by varying system characteristics at differential operating conditions. The thrust bearing must prevent excessive movement of the rotating element. This automatic compensation is the major feature that differentiates the balancing disc from the balancing drum.



a. Simple balancing disc

b. Combination balancing disc and drum

FIGURE 19

APPLICATIONS OF CENTRIFUGAL PUMPS

The classifications discussed in this chapter are applied to the following major pumps used in Candu and Heavy Water Plants.

Candu:
1. Primary Heat Transport Circulating Pump
2. Primary Heat Transport Pressurizing Pump
3. Boiler Feed Pump
4. Condenser Circulating Water Pump

BHWP:
5. In Line, Close Coupled Process Pump
6. Canned Rotor Process Pump
7. Cooling Water Pump

1. PRIMARY HEAT TRANSPORT CIRCULATING PUMP

The Byron Jackson PHT Circulating Pumps at both Pickering Generating Station and Bruce Generating Station are similar in construction although the BGS'A' pump is considerably larger than that at Pickering GS'A'. The Pickering PHT Circulating pump is shown in Figure 20, the Bruce PHT Circulating pump in Figure 21.

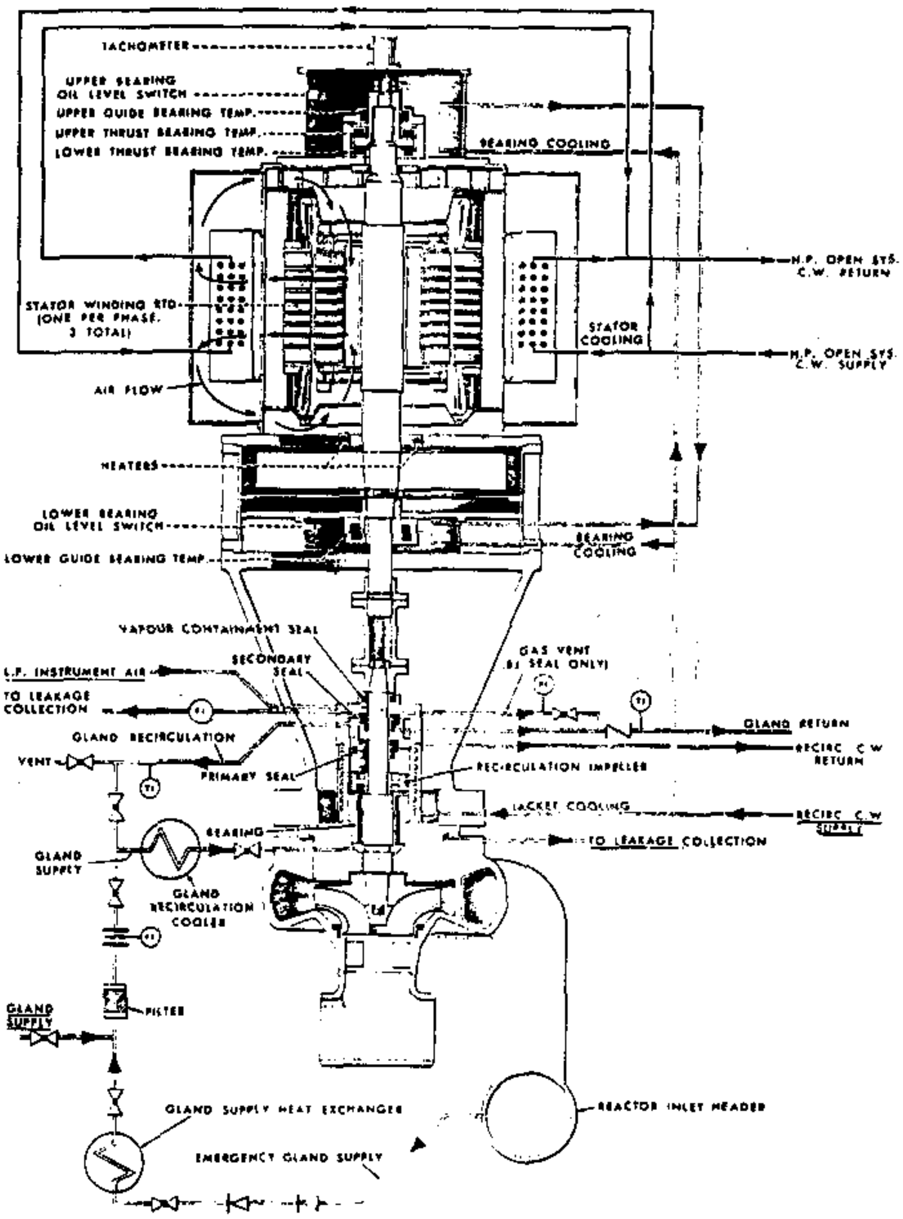
The purpose of the pump is to maintain a high flow rate of D_2O through the reactor to transport heat generated in the reactor to the Steam Generators.

a) Mount - Vertical

Reasons:
- a) Cheaper - fewer components - no base plate
- b) Occupies minimum floor space
- c) Suspended, therefore free to move slightly, easing pipework expansion problems
- d) Seal replacement less complex.

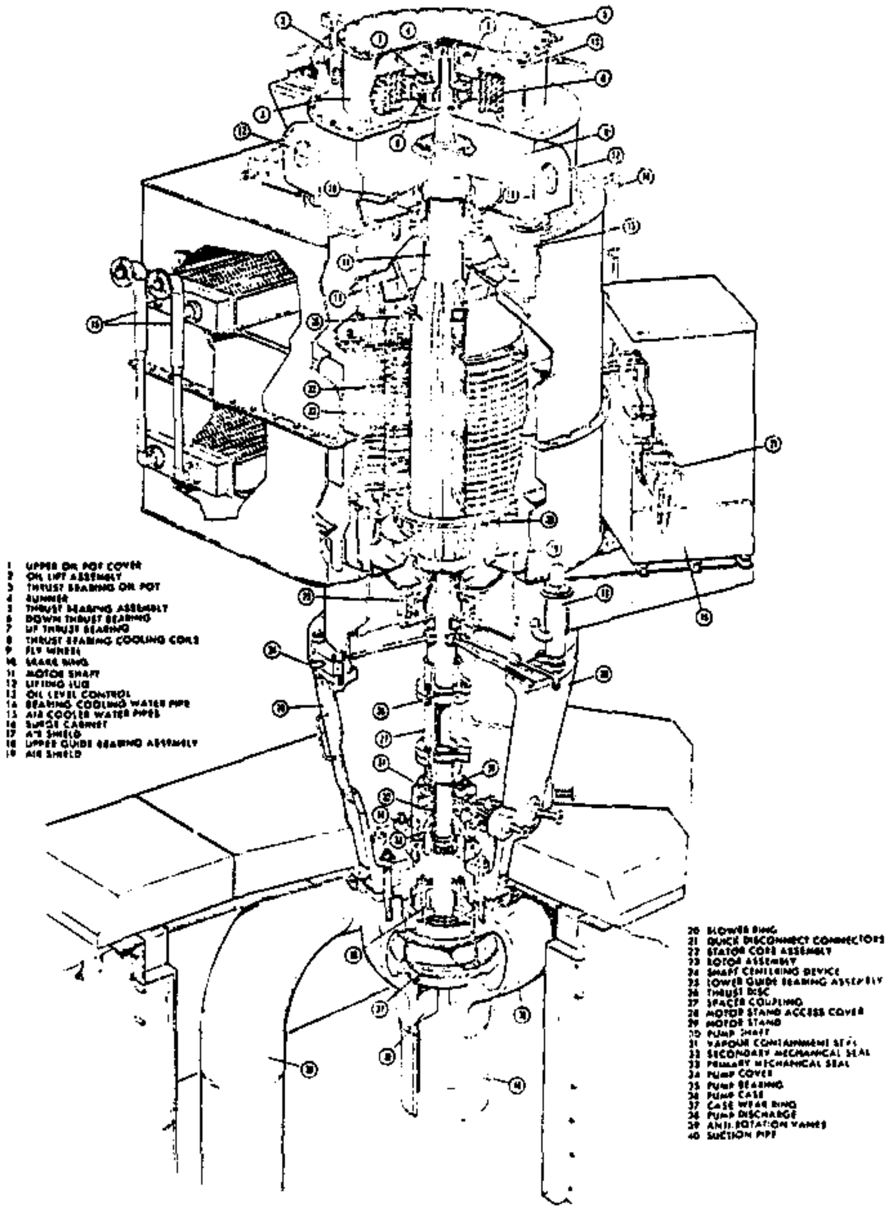
b) Staging and Impeller
- Single Stage
- Single Suction
- Radial Flow
- Fully Shrouded
- Stainless Steel.

The above combination enables the total design PHT flow rate of 12.1 m^3/s (160,000 Igpm) per reactor against a head of 144 m (480 ft) to be met at Pickering by 16 PHT pumps utilizing a 584 mm (23 in) diameter impeller, each with a rating of 0.76 m^3/s (10,000 Igpm).



Pickering GS'A' Heat Transport Pump - Instrumentation and Auxiliary Circuits Schematic

FIGURE 20



Bruce GS'A' Heat Transport Pump

FIGURE 21

The higher flow requirement at Bruce of 13.2 m^3/s (174,000 Igpm) against a head of 210 m (700 ft) is provided by a 4 PHT circulating pumps per reactor, each with a 800 mm (31.4 in) diameter impeller and a rating of 3.3 m^3/s (43,600 Igpm).

c) Casing Split - Radial

The design of the casing assembly is such that all internal pump components, except the shaft and the impeller, can be removed from the pump without disturbing the motor, providing ease and speed of maintenance. The rotating element, pump cover and all internal components can be removed vertically as a unit from the pump case after first removing the motor and motor mount.

d) Energy Conversion - Double Volute

The size of impeller dictates the use of a double volute to reduce the radial thrust on the impeller. This in turn reduces the required shaft diameter and the area of the bearing surface.

e) Axial Thrust Compensation - Impeller rear wearing ring and balance holes

The impeller has a rear wear ring of equal diameter to the front wear ring. Pressure balancing holes are drilled through the impeller inside the rear wear ring. A thrust bearing is fitted at the top of the motor shaft above the upper guide bearing.

f) Special Features - Flywheel

The flywheel increases the Moment of Inertia of the pump/motor rotating assembly to give the pump a rundown time sufficient to maintain reactor cooling until power is re-established after an electrical power failure.

h) Auxiliary Systems Required for Pump Operation

Before operation of the PHT circulating pump can be considered, the following auxiliary supplies must be established at the pump.

i) Gland Supply and Return - High pressure water from the PHT Pressurizing System for cooling and lubricating glands, seats and bearings.

ii) Pump seal leakage Collection & Venting - Leakage from mechanical seals is piped to the D_2O Collection System. Continuous venting is required to ensure that seals are completely immersed in water.

iii) Pump Jacket and motor bearing cooling - supplied by the Service Water Recirculating Cooling Water System.

iv) Motor Stator Cooling - 2 motor stator coolers for pump are cooled by the Service Water High Pressure Open System.

v) Vapour Containment Seal Air Supply - supplied from Reactor Building Instruments Air System.

2. PRIMARY HEAT TRANSPORT PRESSURIZING PUMP

The Primary Heat Transport pressurizing pumps are required to maintain PHT system pressure. They also supply the flow required for the PHT purification system, for PHT circulating pump gland sealing water, for the bleed condenser sprays and for pressurizing flow to the fuelling machines.

The two 100% Byron Jackson 363 BHP Type 15 HHH Hydropress centrifugal pumps are described below.

a) Mount - Vertical

b) Staging & Impeller - Multiple Stage (20)
- Single Suction
- Radial Flow
- Fully Shrouded

The very high head requirement of 890 m (2970 ft) with a low flow rate of 0.018 m^3/s (233 Igpm) at 38°C (100°F) has led to a multiple stage centrifugal pump design using 20 impellers of 0.165 m (6 in.) diameter. Fully shrouded, radial flow impellers are used for maximum efficiency.

c) Casing Split - Barrel, Figure 22

The inner casing is axially split to allow ease of inspection of the pump assembly. To minimize the difficulty involved in producing a leak tight flange to seal against the last stage discharge pressure of approximately 10 MPa(g) an external barrel is located over the axially split casing and the space between the two casings is pressurized by the pump discharge.

d) Energy Conversion - Volute

Each of the 20 stages has its own volute, the discharge of which is led to the eye of the next impeller.

e) Axial Thrust Compensation - Opposed Impellers - Figure 22

The impellers are arranged with the first 7 stage impellers being mounted on the shaft in the opposite direction to the last 13 to equalize axial thrust.

The Pressurizing Pumps at Bruce GS'A' are similar in principle but are horizontally mounted and have ten stages of fully shrouded, radial flow impellers. The casings are axially split with no barrel. This arrangement has called for an inherently stronger and heavier casing.

3. BOILER FEED PUMP

The Main Boiler Feed Pump at Pickering GS and Bruce GS are identical in all but impeller size. Both plants utilize three Byron Jackson pumps per unit, each of which are 50% capacity. These pumps take a suction from the Deaerator feedwater storage tank and discharge via a header, through two banks of feedwater heaters and feedwater regulating valves to the Steam Generator.

a) Mount - Horizontal

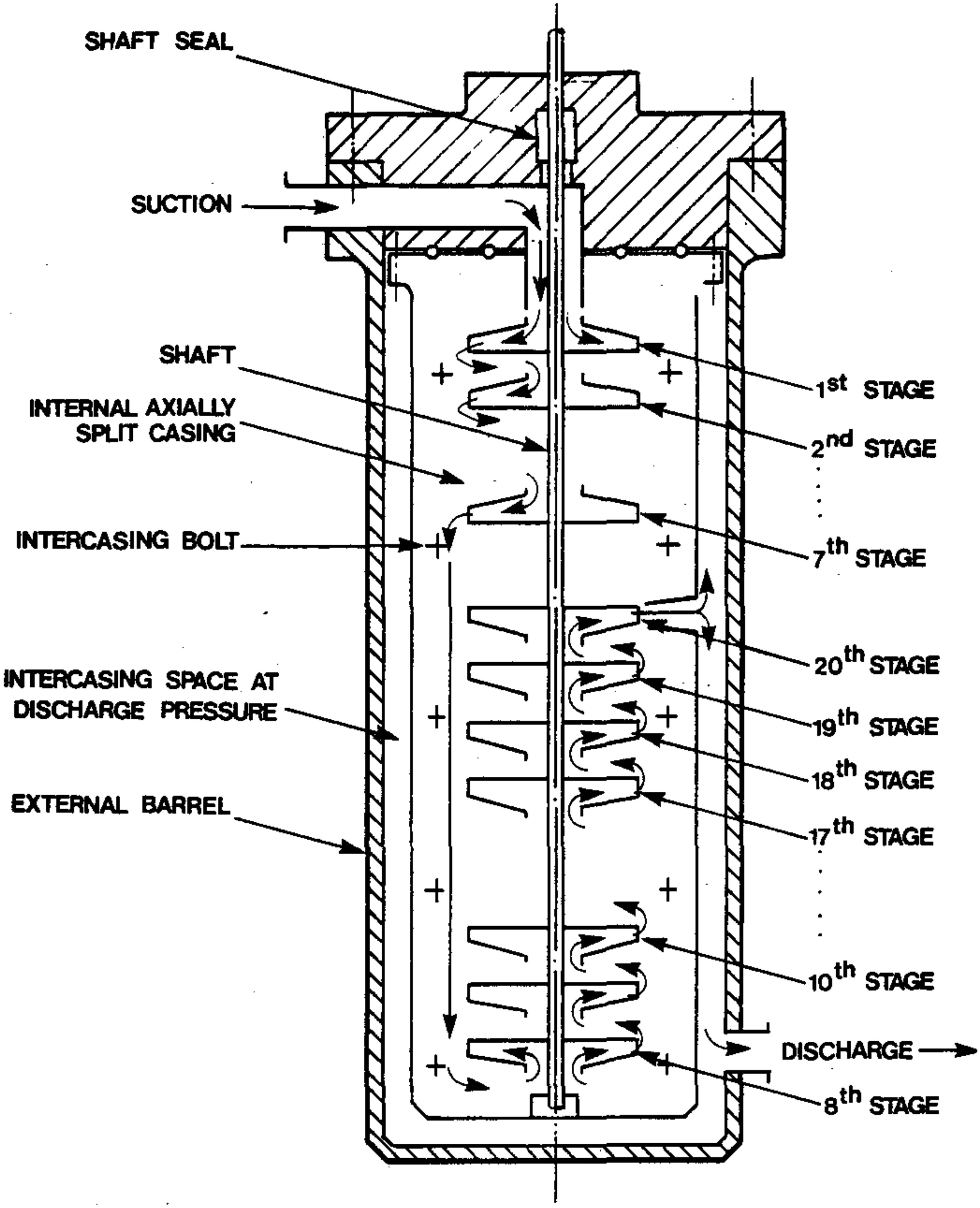
Being relatively small pumps the horizontal mount allows ease of maintenance without incurring high cost through the use of a large bed plate.

b) Staging & Impeller - Two Stage
- Single Suction
- Radial Flow
- Fully Shrouded

The two stage design is necessary to allow the pump to operate with a total head of 538 m (1765 ft) and a capacity of 0.651 m^3/s (8600 Igpm) at Bruce GS.

c) Casing Split - Axial

The maximum working pressure of 6.9 MPa(g) (1000 psig) can be contained by an axial split casing. The external pipework is attached to the lower half of the casing allowing the top cover to be removed without disturbing the pipework.



Operation of Pickering GS'A' HT Pressurizing Pump

FIGURE 22

d) Energy Conversion - Double Volute

The high pressure rise across the pump leads to a large radial thrust acting on the impeller. To minimize the radial thrust a double volute is used, allowing a smaller diameter pump shaft and a smaller bearing surface area to be used.

e) Axial Thrust Compensation - Opposed Impellers

The two impellers are opposed, contributing towards a balance in axial forces. The resulting reduced axial thrust is compensated by a thrust bearing.

f) Special Fittings - Continuous By-pass Flow

During plant power reduction the pump may be required to operate against a shut Boiler Feed Regulating Valve. Operation against a shut discharge valve leads to churning of the impeller with consequent overheating and eventual damage to the pump. To overcome this problem a recirculating line is fitted which allows 10% of the pump design flow to be directed from the discharge upstream of the Feed Regulating Valve back to the deaerator storage tank, allowing a continuous cooling flow to be maintained through the pump at all times.

4. CONDENSER CIRCULATING WATER PUMP

The Bruce GS condenser circulating water pump, (Figure 23) manufactured by Ingersoll Rand, provides a lake water coolant flow of 12.88 m^3/s (170,000 Igpm) to the Main Condensers for condensation of exhaust steam from the L.P. turbines. The pumps operate against a 6.4 m (21 ft) head.

a) Mount - Vertical

The pumps are vertically mounted submerged impeller (wet pit type) pump. Since the pump is below lake level there is no requirement for priming or for suction pipework. Also water lubricated cutless rubber bearings can be used since the lubricant will always be present at the bearing surface, although the primary bearing coolant and lubricant supply is from L.P. service water.

The pump casing is, in fact, part of the condenser inlet pipework. The pump shaft extends vertically up to the gearbox and motor above lake level.

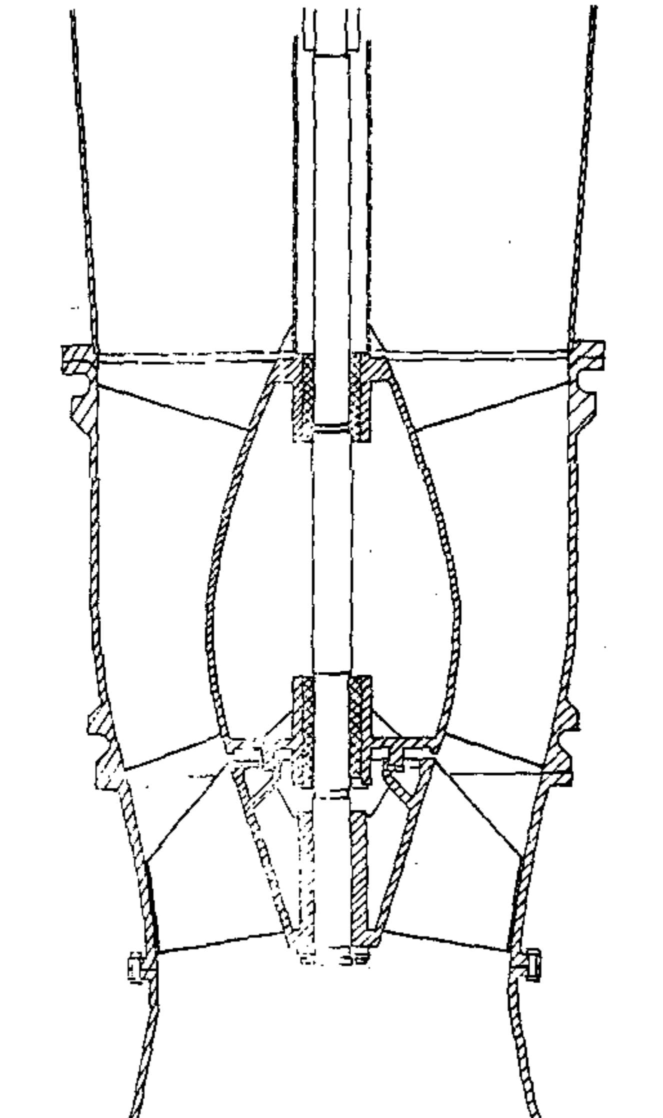


FIGURE 23: Pickering Condenser Circulating Water Pump

b) Staging & Impeller - Single Stage
- Single Suction
- Axial Flow

In installations requiring very high flow rates an axial flow impeller is used. The disadvantage of the axial flow impeller is that the discharge head which the impeller can supply is limited. To overcome this problem in the CANDU CCW installation the Main Condenser CCW System is completely primed by use of the Vacuum priming pumps. This allows coolant flow to be maintained by a syphon effect, with the CCW pumps supplying only sufficient head to overcome friction losses of the water through the system.

The 0.546 m (21.5 in) diameter impeller is capable of supplying the flow rate required at the head necessary to overcome friction losses using a single stage.

c) Casing Split - Radial

Since the casing is part of the Condenser inlet pipework an axial split casing would not be feasible.

d) Energy Conversion - Diffuser

A diffuser vane ring is the only practical method of energy conversion when an axial flow impeller is used. The operating conditions are constant, the only variation being due to build up of deposits on the pipework or partial blockage of condenser tubes or travelling screens, thus the impeller and diffuser vanes can be designed for maximum energy conversion with minimum turbulence at the design operating point with little anticipated deviation from that point.

e) Axial Thrust Compensation - Helical Gearing

Since the action of the axial flow impeller is that of lifting the water through the pump there is a downthrust on the impeller which must be compensated. This compensation is carried out by the use of single helical cut gearing in the gearbox between motor and pump. The gearing arrangement chosen provides an upward force on the pump shaft.

f) Special Features - Gearbox

Since the axial flow impeller is designed to run at low speed (200 rpm) a reduction gearbox must be used if it is chosen to run the pump using an induction motor. The gearbox, therefore, reduces the speed from 18000 to 200 rpm.

5. BHWP - Vertical, In Line, Close Coupled Centrifugal Pumps

United vertical, in line, close coupled centrifugal pumps (Figure 24) are used in the enriching units at the Bruce Heavy Water Plant in the following applications:

i) 1st, 2nd and 3rd Stages. Dehumidifier Pumps.
ii) 1st Stage Humidifier Pumps.
iii) 2nd and 3rd Stages Hot Tower Bottom Pumps.
iv) Effluent Strippers Reflux Pumps.
v) Steam Tracing Condensate Return Pumps.
vi) Tempered Water Pumps.

a) Mount - Vertical in line

Vertical in line pumps are common in industrial process service since they are designed with the same size suction and discharge pipework on the same vertical and horizontal centrelines 180° apart. Although similar pumps in small sizes may be supported by the pipe itself with no special supporting foundation the pumps used at BHWP are large enough to require considerable support.

b) Staging & Impeller - Single Stages
- Fully Shrouded
- Radial Flow

Since these pumps in effect act as Booster pumps they are required to be capable of increasing system pressure sufficient for the next stage of the process. To allow this pressure rise a radial flow impeller is used, sufficient pressure rise being available with one such impeller. To minimize the pump motor size a fully shrouded impeller is used providing maximum efficiency.

c) Energy Conversion - Volute

The Volute allows conversion of kinetic energy picked up in a radial flow impeller to pressure energy.

d) Casing Split

In line pumps invariably have radially split casings allowing dismantling of the pump without disturbing the pipework.

e) Axial Thrust Compensation

Axial thrust is reduced by the use of a rear wearing and balance holes in the impeller. Residual thrust is absorbed by the motor thrust bearing.

f) Special Features

Close coupled pumps do not have bearings within the pump assembly. The motor is connected directly to the pump via a solid coupling, the pump impeller is then completely supported by the motor bearings.

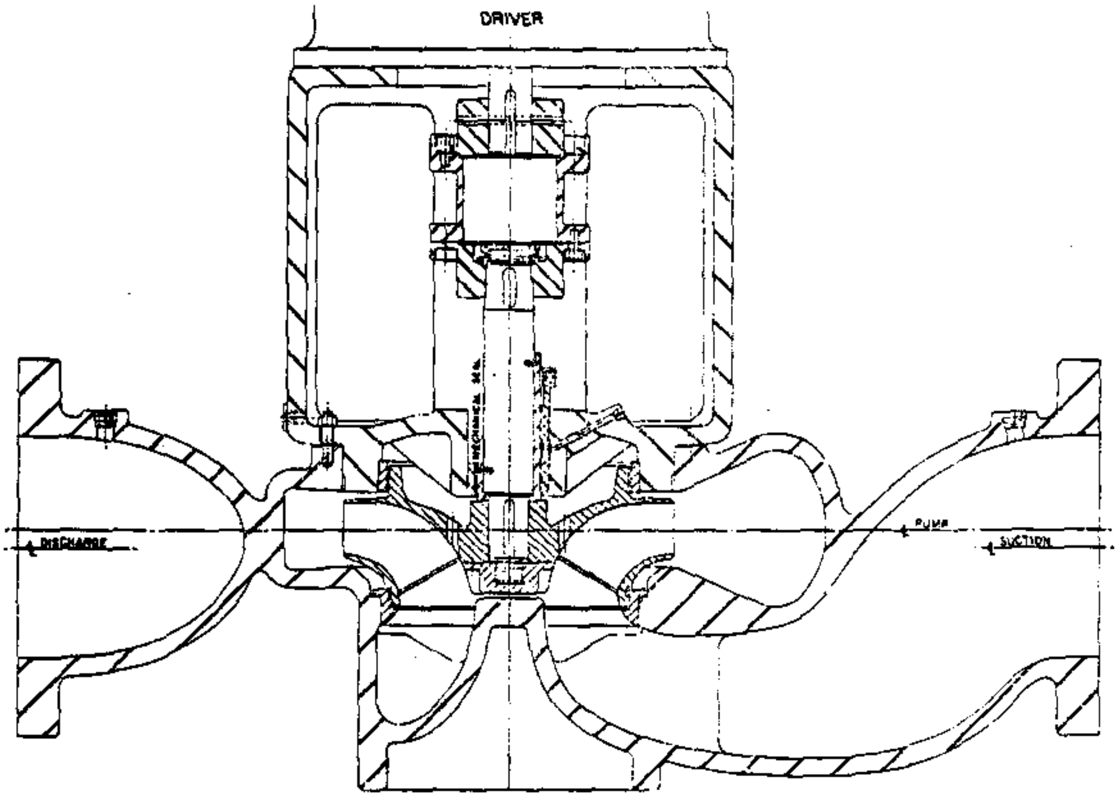


FIGURE 24

6. BHWP - Horizontal Canned Motor Centrifugal Pumps

Canned Motor Centrifugal Pumps are used in the Bruce Heavy Water Plant Finishing Units in the following applications:

i) Feed Storage Tank Fuel Pumps.
ii) D_2O Storage Transfer Pumps.
iii) 1st and 2nd Stage Bottoms Pumps.
iv) 1st Stage Reflux Pumps.

Application i) and ii) above use a pump with structures as shown in Figure 25. Appliction iii) and iv) differ from Figure 25 in that motor cooling is supplied by external ducting of liquid from the pump discharge. The design of the axial thrust compensation features. also differs in that applications iii) and iv) use a balance drum whereas applications i) and ii) use an automatic thrust control valve.

a) Mount - Horizontal

b) Staging & Impeller - Simple Stage
- Fully Shrouded
- Radial Flow

The impeller enables the pump to discharge a capacity of 1.43 l/s ( ) at 3600 rpm. The pump is powered by a 0.8 KW motor.

c) Energy Conversion - Volute

d) Casing Split - Radial, Canned rotor

In canned rotor pumps both motor and pump are enclosed in a common casing. The motor stator and rotor assemblies are sealed in two jackets or 'cans'.

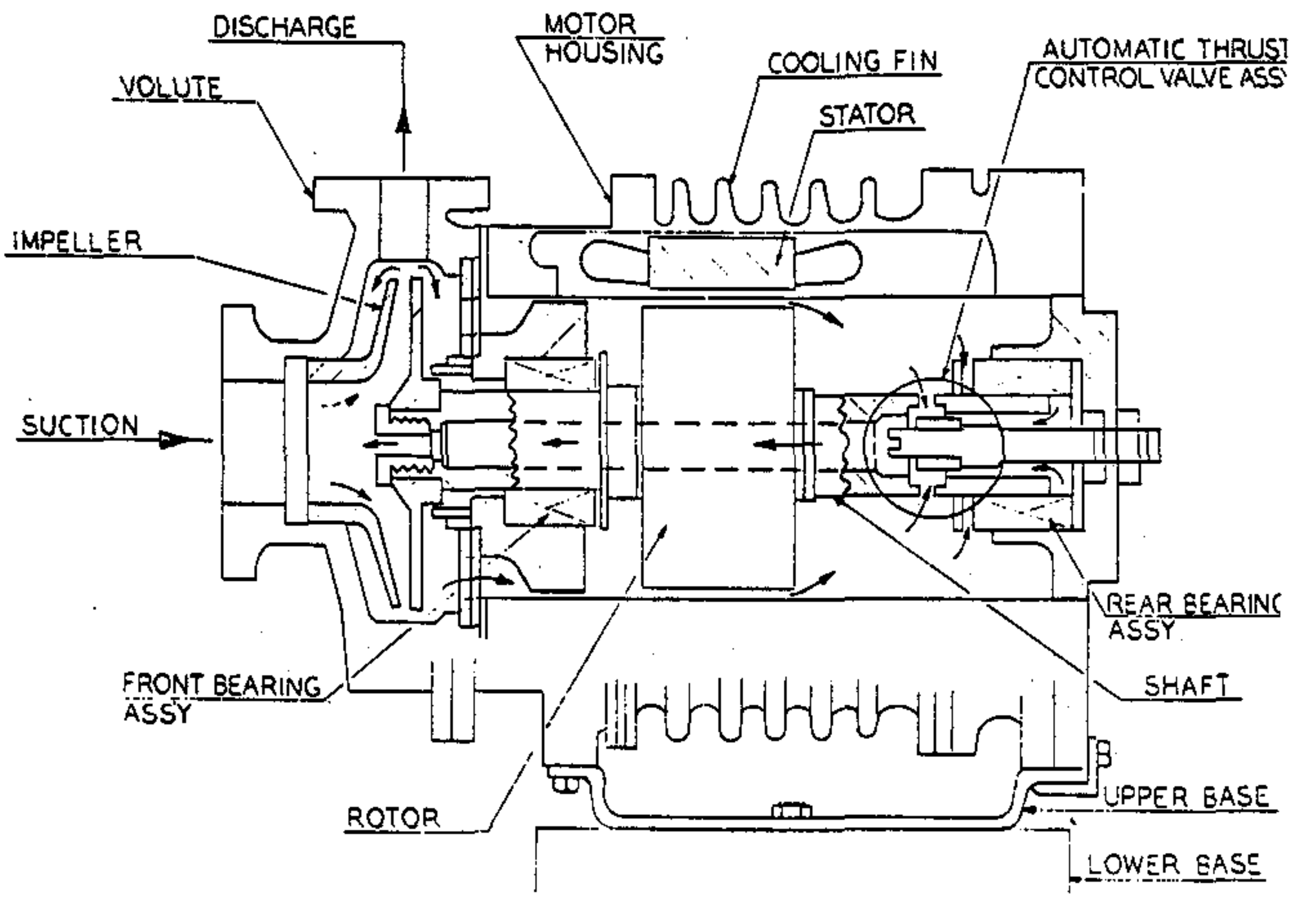
The canned rotor and shaft assembly is immersed in the fluid being pumped. Cooling for stator, rotor and bearings as well as liquid for bearing lubrication is provided by internal circulation of the pumped fluid, assisted by air cooling of the finned motor housing.

e) Axial Thrust Compensation - Automatic Thrust Control Valve

Liquid used for bearing and motor cooling returns to the impeller eye by means of a duct down the centre of the pump/motor shaft. Axial movement of the shaft due to a change in operating conditions causes a change in the flow-rate of cooling liquid return down the duct in the shaft by altering the opening of the automatic thrust motor valve. This change in liquid flow rate leads to a change in differential pressure between the front and back of the pump/motor shaft which tends to create a force opposing the original movements of the shaft and thus restore axial balance.

f) Special Features - Zero Leakage

Since the process liquid is used for motor cooling no conventional gland assemblies are required for canned centrifugal pumps. This eliminates the problems of prevention of leakage past a seal or contamination of process fluid due to leakage of seal coolants. This zero leakage feature makes canned rotor pumps dead for use in high grade D_2O systems where relatively small volumes are being transferred.



BHWP-Canned Rotor Centrifugal Pump With Internal Recirculation

FIGURE 25

7. BHWP - Common Services Cooling Water Pump

Two of the five Cooling Water Pumps, installed in the Cooling Water Pumphouse, supply the cooling water required for BHWP B, two will supply BHWP D and the fifth is available as a standby to replace any of the others.

The STORK VOA 80-75 pumps (Figure 26) are each capable of supplying a capacity of 5500 l/s (72,700 igpm) at a discharge pressure of 360 KPa (52 psig). Each pump is driven by a 2.5 MW (3300 Hp) motor.

a) Mount - Vertical Submerged

The submerged impeller removes the requirement for suction pipework and prestart priming. It also ensures that the water lubricates resin sleeve pump bearings are constantly immersed in water.

b) Staging & Impeller - Single Stage
- Fully Shrouded
- Mixed Flow

In view of the large pump size and high capacity a fully shrouded impeller is used to maximize efficiency and hence minimize motor size. The use of the mixed flow impeller is due to the requirement for a compromise believes the high capacity obtainable with a radial flow impeller and the pressure rise obtainable from a radial flow impeller. The impeller chosen is capable of meeting the pressure requirement in a single stage.

c) Energy Conversion - Diffuse

The diffuser allows the conversion of kinetic energy gained by the water in the mixed flow impeller to pressure energy.

d) Casing Split - Radial

Since the pump casing forms the cooling water ducting a radial split must be used.

e) Axial Thrust Compensation

The axial thrust excited downwards on the shift during operation is reduced by the use of balance holes in the hub of the impeller and a rear wear ring. Residual thrust is absorbed by a motor thrust bearing.

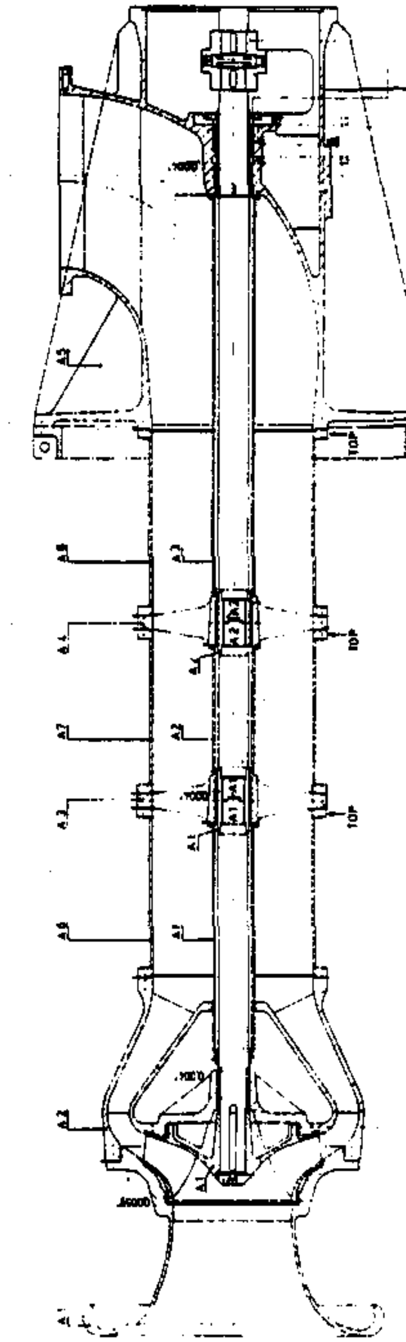


FIGURE 26: BHWP Cooling Water Pump

ASSIGNMENT

1. Draw a simple chart showing the major classifications of centrifugal pumps and the subdivisions of each of those classifications.

2. Give four applications of centrifugal pumps in a Nuclear Power Plant or Heavy Water Plant. Describe briefly the type of pump used in each case.

3. List and describe briefly five methods of compensating for axial thrust in a centrifugal pump. How can a large radial thrust on an impeller be compensated?

4. (a) Study the sectional drawings of centrifugal pumps shown in Figures 27 - 33. Classify each one in terms of:

   Number of stages
   Impeller type and direction of flow
   Energy conversion
   Casing split

   (b) Determine the means by which axial thrust in each of those pumps is compensated.

   (c) State the nature of the application for which you consider each pump might be used.

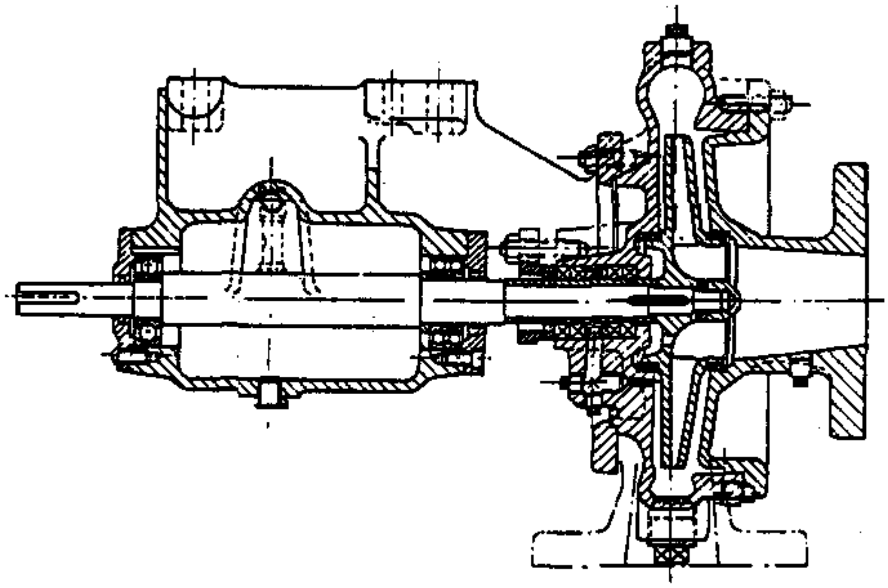


FIGURE 27

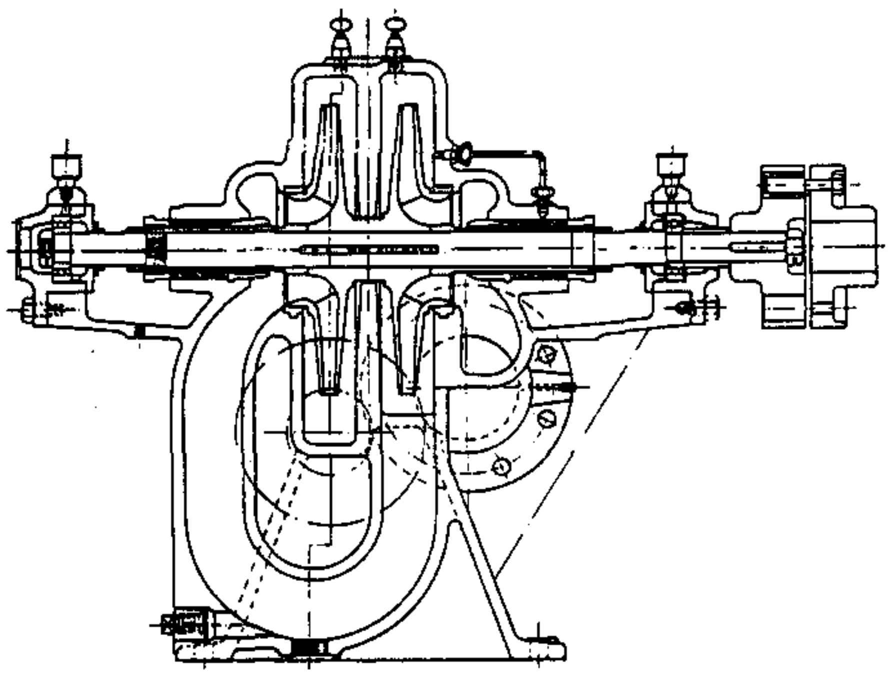


FIGURE 28

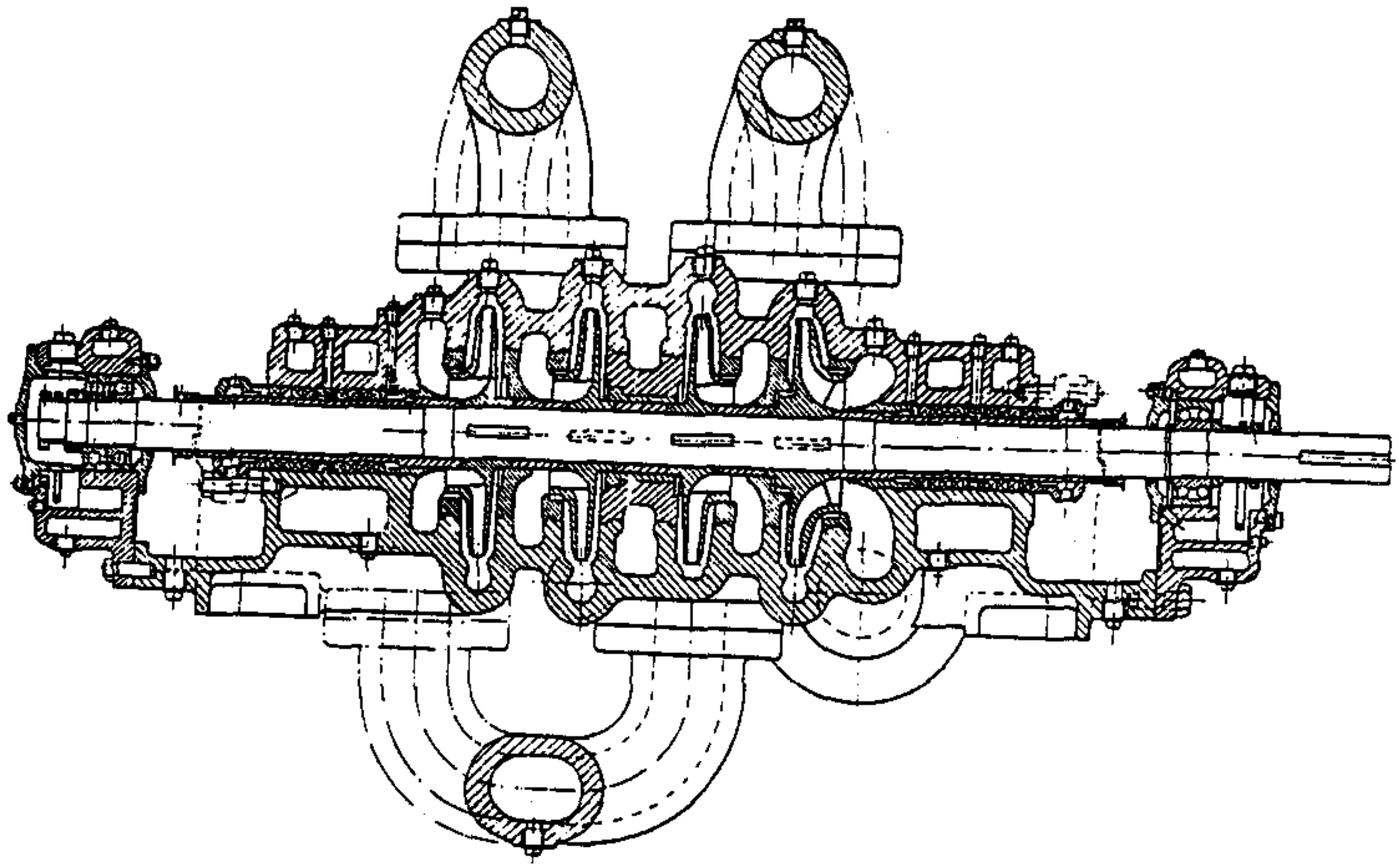


FIGURE 29

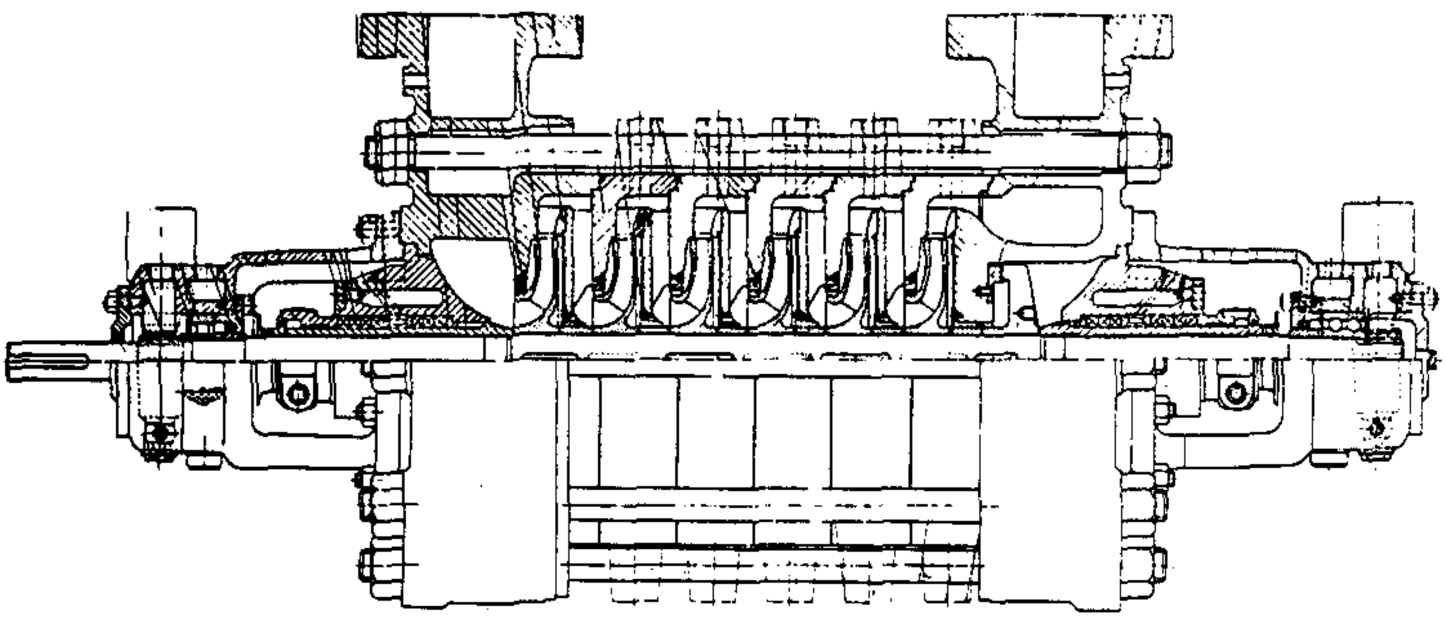


FIGURE 30

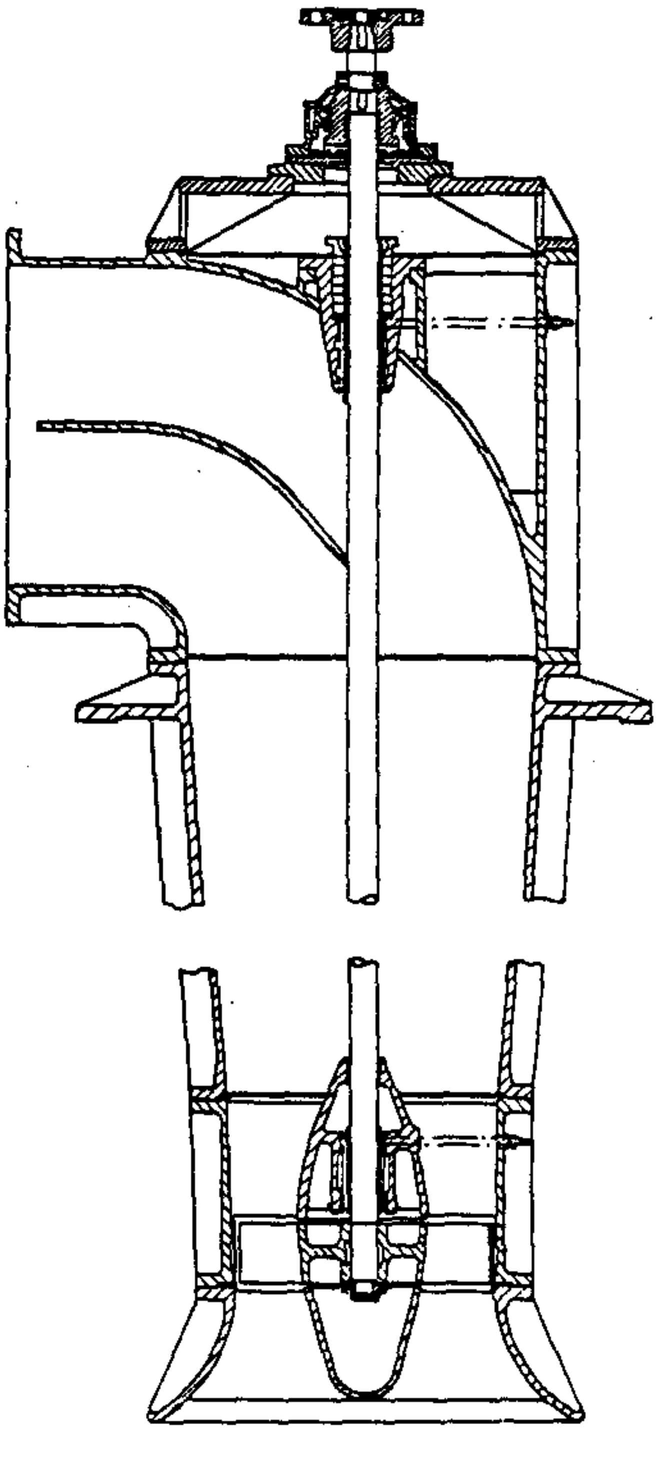


FIGURE 31

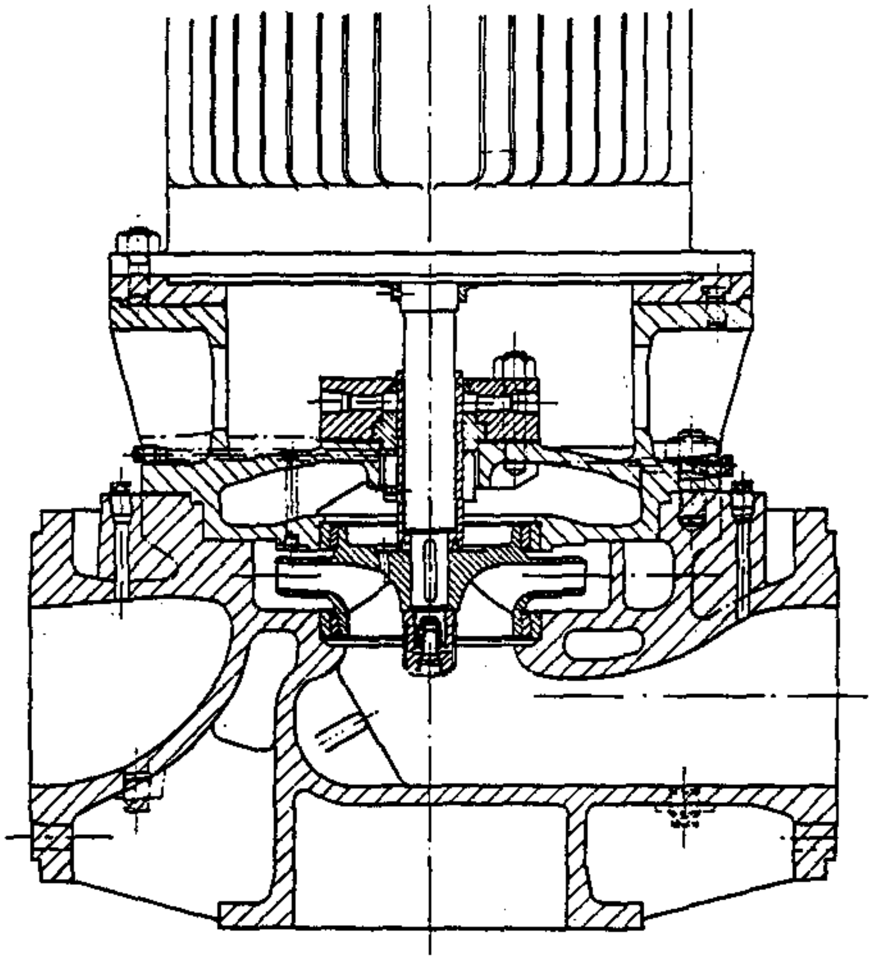


FIGURE 32

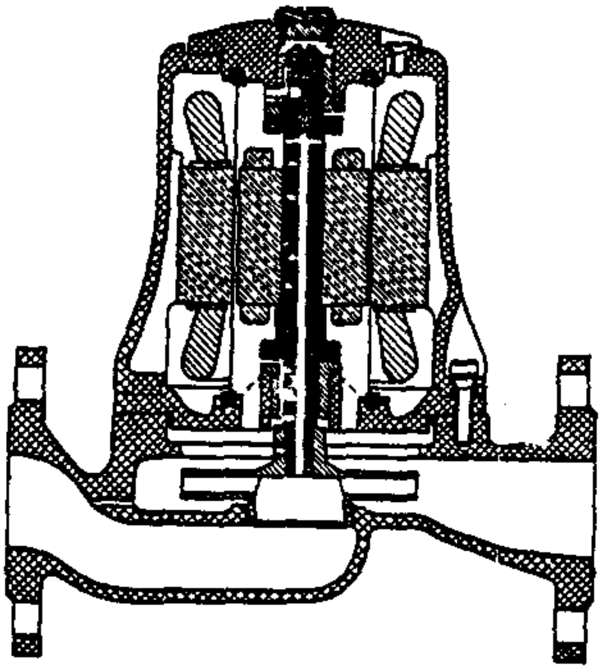


FIGURE 33

Mechanical Equipment - Course 430.1

POSITIVE DISPLACEMENT PUMPS

In the previous lesson it was explained that although centrifugal pumps have many operational and maintenance advantages over positive displacement pumps, there are services where positive displacement pumps cannot be replaced by centrifugal pumps. These services involve requirements for very high pressure, constant predetermined delivery which is not affected by system changes, and pumping of viscous liquids.

In our plants positive displacement pumps find limited application and are used in chemical injection systems (water treatment), hydraulic systems, especially fuel handling systems, pressurization systems (stand-by HT pressurizing pumps) and jacking oil systems.

Positive displacement pumps are either **rotary** or **reciprocating.**

Reciprocating Pumps

Reciprocating pumps are generally self-priming but this ability is limited and the manufacturer should be consulted if self-priming is the required feature. The discharge is pulsating and if the system demands a uniform pressure and flow an accumulator is employed on the discharge line which is a vessel partially filled with air cushioning pulses produced by the pump. They have to be fitted with check valves in suction and discharge. Reciprocating pumps are available as constant flow pumps or variable flow pumps. Variable delivery is usually accomplished by having a variable stroke feature on the pump. The second possibility of having a variable speed drive is seldom used with this type.

Three types of reciprocating pumps are available: **piston, plunger** and **diaphragm.**

Piston Pumps are usually double-acting which means that each side of the piston compresses the liquid. They can be horizontal or vertical. Depending on the number of cylinders in the pump, the pumps are referred to as simplex, with one cylinder, duplex with two cylinders, triplex with three cylinders and so on. Sealing of pistons is accomplished by piston rings carried by the piston.

Figure 1 shows a horizontal double-acting simplex piston pump.

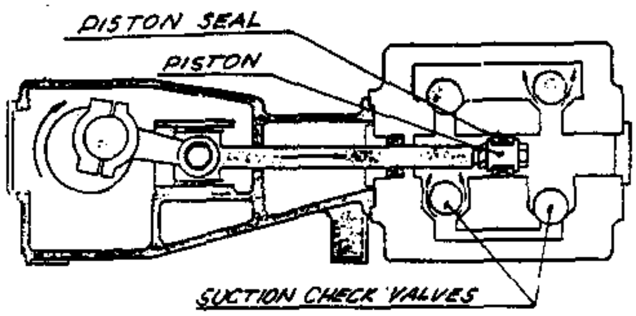


Figure 1

Plunger pumps are single-acting, are used for higher pressures than piston pumps and the plunger runs through a stationary seal, usually a stuffing box. They can be vertical or horizontal and simplex or multiplex. Designs with variable stroke are common. Plunger pumps are used for highest pressures in the industrial applications.

Figure 2 shows a horizontal single-acting simplex plunger pump.

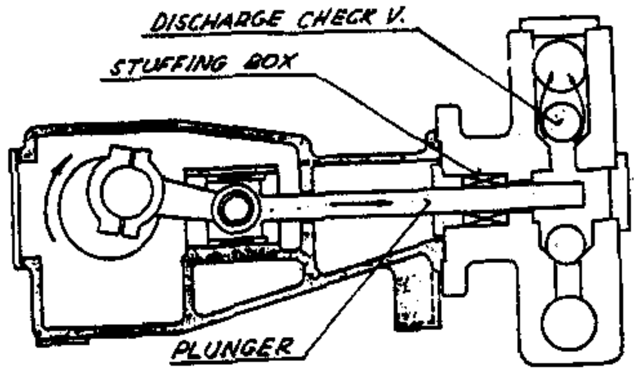


Figure 2

Diaphragm pumps can be either hydraulically or mechanically operated. In mechanically operated pumps the pumping action, resulting from the deflection of the diaphragm, is accomplished by direct pushing of a cam or a push rod on the diaphragm. In more common fluid operated pumps the deflection of a diaphragm is achieved by pressurized fluid. This fluid in turn is pressurized by a small plunger pump. The diaphragm pumps can be a simplex or multiplex design and can have a variable stroke arrangement.

Diaphragm pumps can be used when zero leakage is required or when contact by the pumped fluid with the plunger and cylinder might be detrimental to the pump.

Figure 3 shows a horizontal single-acting simplex flat diaphragm pump.

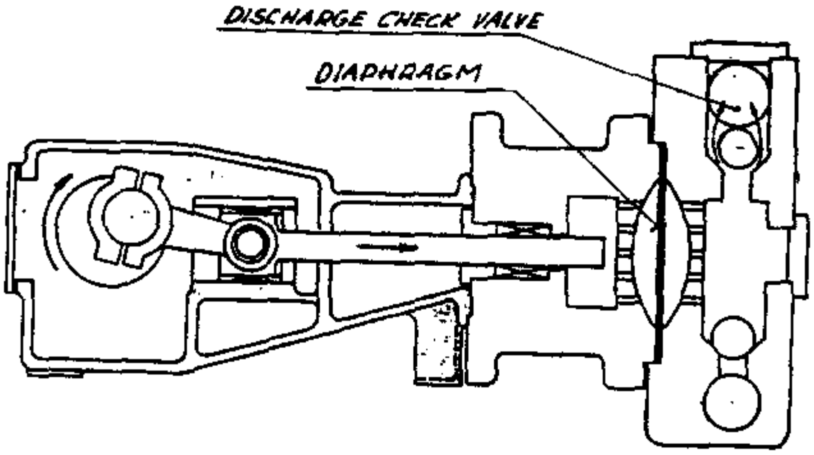


Figure 3

A special type of a piston pump is an axial piston pump. The principle of operation of these pumps is that liquid is drawn in an forced out by reciprocating pistons which rotate with the cylinders. They are multiple piston units and are available with constant or variable displacement. Figure 4 shows a constant displacement pump. The same unit can be used as a hydraulic motor.

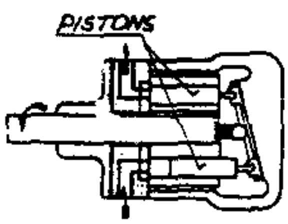


Figure 4

Rotary Pumps

Rotary positive displacement pumps are characterized by close running tolerances to minimize so called slip which is a leakage from the discharge back into the suction. They are generally self priming, do not require check valves in suction and discharge and produce negligible pulsations. Rotary pumps may be classified as: Vane, screw, gear and lobe type. Only the most common types are described here leaving a complete account for higher levels of this course.

Vane pumps usually have one or more vanes in the shape of blades sliding in radial slots in a rotor which is eccentrically located in the body of the pump. Pumps with blades in stator are also available. Figure 5 shows a blade-in-rotor sliding vane pump with constant delivery. A variable delivery feature of these pumps is accomplished by a variable speed drive or by changing the eccentricity of the rotor with respect to the stator-casing.

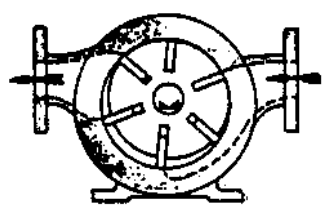
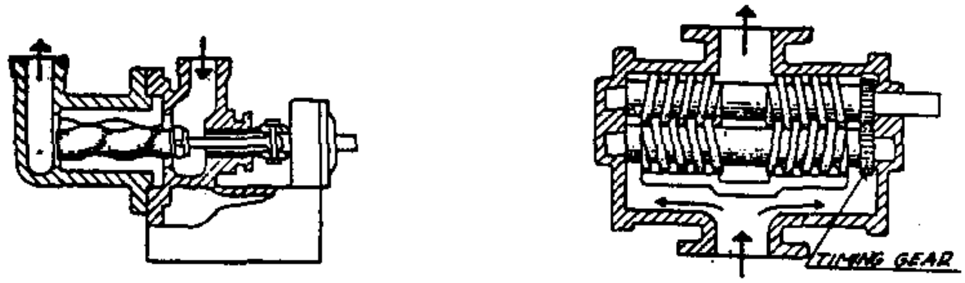


Figure 5

Screw pumps are available as single or multiple screw units shown in Figure 6(a) and (b).



(a) (b)

Figure 6

The fluid is carried between rotor screw threads and is displaced axially as they mesh either with internal threads on the stator (single screw) or with each other (multi-screw). The discharge from these pumps is uniform, they are self-priming, do not require check valves and can pump a substantial amount of solids, gases or vapours mixed in liquid. A timing gear is used to transfer the rotation from a motor driven shaft to the other shaft, enabling contact between the two screws to be eliminated, thus reducing wear.

Gear pumps are self-priming, do not need check valves and can handle pumpage which is fairly clean. They are easily damaged when run dry. Although internal gear pumps are in existence, external gear pumps are more common. An example of an external gear pump is in Figure 7. Fluid is carried between gear teeth and the casing and displaced when the teeth mesh.

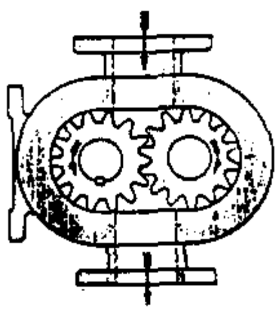


Figure 7

Changes in capacity can be realized with variable speed drives only. The motion from shaft to shaft is transferred by the gears themselves by direct tooth contact.

Lobe pumps are similar to gear pumps but the rotation from a driven shaft is transferred to the other shaft by the timing gear. The discharge is more pulsating because the rotor has only one to four lobes. Figure 8(a) shows a single lobe pump. 8(b) shows a three lobe pump. Liquid is trapped and carried between rotor lobe surfaces and the casing and discharged by meshing lobes.

(a) (b)

Figure 8

Flexible member pumps. The pumping action and sealing is derived from the elasticity of a flexible member which may be a vane, tube or some other less common element. They have good corrosion resistance, can pump in either direction and are light. They are temperature limited, discharge pressures and capacity are low. Figure 9(a) shows a flexible vane pump, 9(b) a flexible tube pump.

(a) (b)

Figure 9

Application of Positive Displacement Pumps

Positive displacement pumps have their place in pumping field and are applied in our plants for metering, pressurizing and hydraulic systems. The important fact is that due to their design, they cannot work with blocked or shut off discharge without immediately damaging the weakest member. That is why the discharge must be fitted with a relief valve which will open and prevent overpressurization and subsequent damage. Sometimes the pump is supplied with an internal relief valve but more often it is a responsibility of the user to fit the system with a relief valve which should be as close to the pump as possible and definitely before any other obstacle in the piping like a valve, pipe fitting or a heat exchanger.

Similarly, as in systems with centrifugal pumps majority of operating problems originate in suction. If a pump is working with excess suction lift the absolute pressure in the suction to the pump may drop below the vapour pressure corresponding to the temperature of pumpage and the pumpage starts boiling. As soon as it is in the pump, pressure increases, vapour condenses and the phenomenon of **cavitation,** explained in the previous lesson, damages the pump. The rules which should be followed to avoid cavitation are the same as in the case of centrifugal pump systems: put the pump as low as possible with respect to the suction tank, make suction piping as short and simple as possible, ie, as few elbows and fittings as possible, avoid valves and ensure that the diameter of the suction piping is adequate.

If the suction pressure is grossly below the vapour pressure a pump will suck in a variable quantity of liquid and boiled off vapour and its discharge flow rate will drop. Positive displacement pumps can discharge gases and vapours so that vapourlocking as understood in centrifugal pumps operations does not happen. Similarly airlocking is not a problem but it is important to realize the certain positive displacement pumps cannot run dry without being soon damaged.

ASSIGNMENT

1. Describe the difference between a piston and a plunger pump in design and application.

2. (a) Describe a problem associated with reciprocating pumps.

   (b) What solutions are possible?

3. How can flow rate be changed with a reciprocating piston pump?

4. Can the discharge from a gear pump be regulated by throttling (partially closing) a valve on the discharge? Explain.

5. Why is it necessary to have a relief valve in a system with a positive displacement pump?

6. Where into the system would you mount a relief valve?

7. Can a positive displacement pump cavitate? Explain.

8. Can airlocking be an operational problem in systems with positive displacement pumps? Explain.

K. Mika

Mechanical Equipment - Course 430.1

COMPRESSORS - DYNAMIC AND POSITIVE DISPLACEMENT

GENERAL

Until this point in time, the discussion has primarily focused on pumps, devices which move liquids. Understandably, there are also devices used to move air and gases. These devices are either compressors, blowers, fans or vacuum pumps. As indicated in Table 1, The American Society of Mechanical Engineers has devised a classification system to help us differentiate these devices. This system is based on discharge pressure.

Table 1

ASME CLASSIFICATION

DEVICE		DISCHARGE	PRESSURE
1.	Compressor	greater than 69 kPa(g)	(10 psig)
2.	Blower	13.8 kPa(g) to 69 kPa(g)	(2 to 10 psig)
3.	Fan	0 - 13.8 kPa(g)	(0 to 2 psig)
4.	Vacuum Pump	suction below atmospheric pressure	

COMPRESSORS

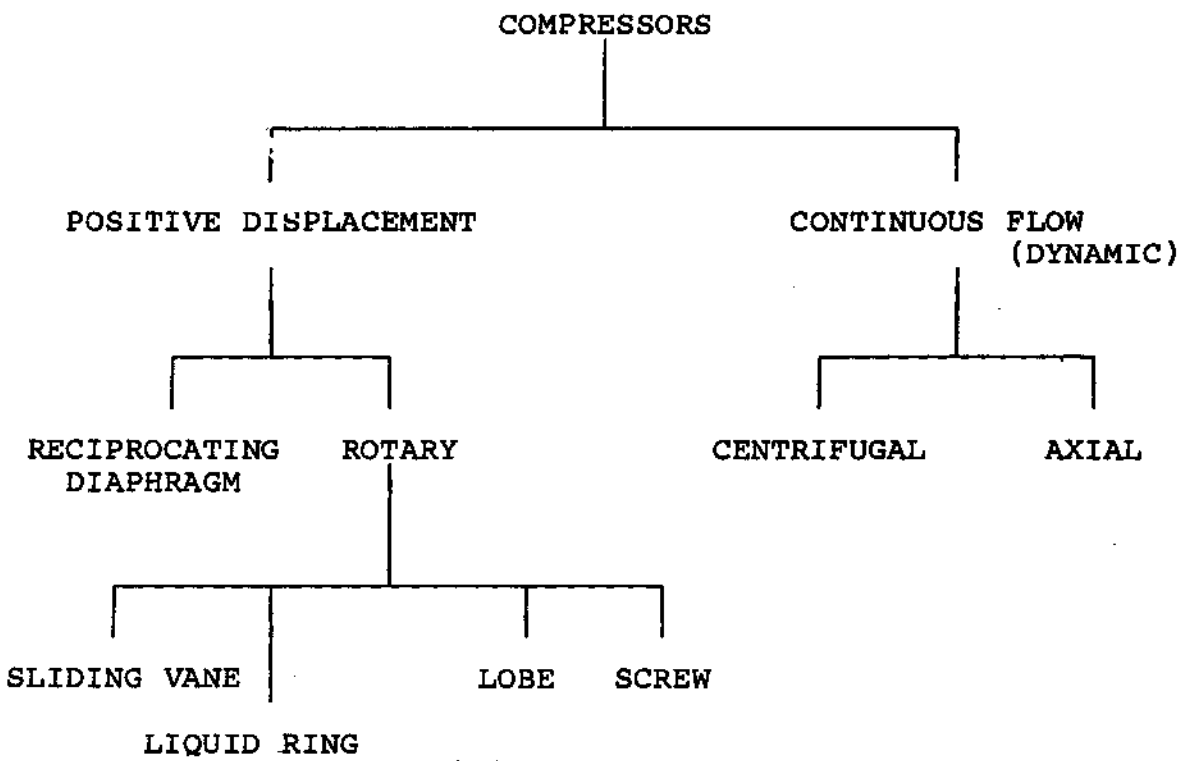
TYPES

A large variety of compressors of different sizes, shapes and designs are available from a number of manufacturers. However, all compressors can be very simply classified as being either a:

1. **Dynamic.**
2. **Positive Displacement.**

Within each major type, there are further subdivisions as shown in Table 2.

Table 2



As you probably noticed, the classification scheme in Table 2 indicates a similarity between pumps and compressors. In fact, there is little difference between the principle of operation of pumps and compressors. A comparison of the two basic types of compressors with respect to flow, pressures and efficiencies could have been a comparison of the two main types of pumps.

Table 3

TYPE		FLOW RATE	PRESSURE	EFFICIENCY
Axial	Dynamic	Highest	Lowest	Intermediate
Centrifugal		Intermediate	Intermediate	Lowest
Positive Displacement		Lowest	Highest	Highest

DYNAMIC COMPRESSORS

Principle of Operation

The principle of operation of the two dynamic compressors, **centrifugal** and **axial**, is basically the same. Two steps are involved:

1. A rapidly rotating element accelerates the gas as it passes through the machine. Centrifugal compressors use **impellers** which axial compressors use **blades** to accelerate the gas.

2. Compression of the gas occurs as velocity of the gas is converted into pressure by stationary elements. Centrifugal compressors employ a **diffuser** section and axial compressors use stationary **blades**.

The main difference which exists between the two is direction of air flow through the compressor. As indicated in Figure 1, centrifugal units have **radial** flow which **axial** units, Figure 2 have air flow parallel to shaft.

Applications

In NGS, centrifugal compressors are used in the **larger air conditioning units.** Although selection factors are many, in comparison with reciprocating units:

1. maintenance and failure rates are lower.

2. smaller, less expensive foundations are required.

3. they offer continuous, smooth, larger flows.

Axial units, on the other hand, are used in **gas turbines** of the standby generators at Pickering and Bruce because of the higher flow requirements. In comparison with centrifugal units, they are:

1. smaller,

2. lighter weight,

3. and require smaller foundations.

As the pressure rise per stage in lower with the axial units than centrifugal units, axial compressors have more stages for a given total pressure rise.

Unfortunately, both units suffer from a common operational problem - **surge.** By definition, surge arises when capacity is reduced to a point when insufficient pressure is

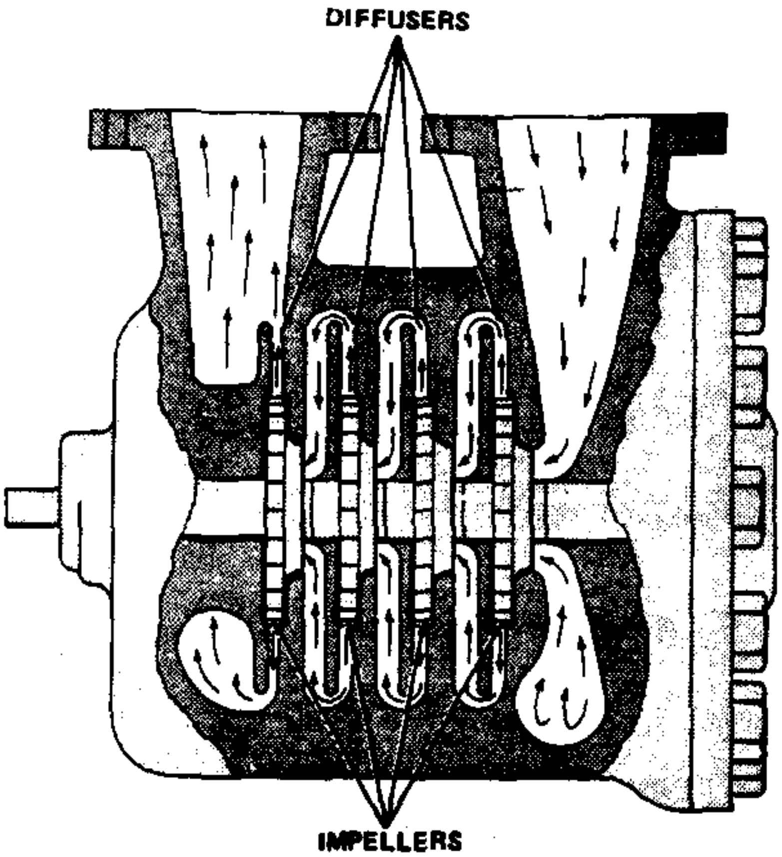


Figure 1

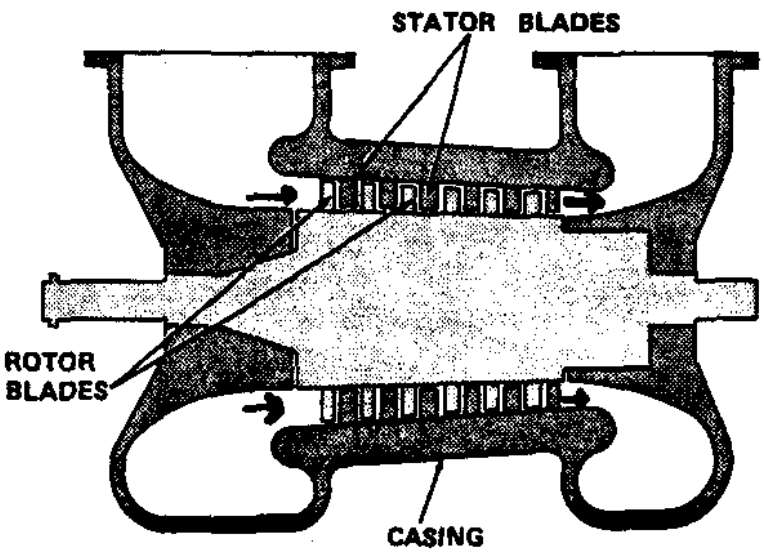


Figure 2

generated to maintain flow. Under this circumstance, a momentary reversal of flow within the compressor occurs. However, as soon as flow reverses, system discharge pressure drops and compressor assumes normal flow. These momentary pulsations are apt to be magnified in the discharge and result in excessive temperature rise, vibration, noise and excessive stress in the compressor.

POSITIVE DISPLACEMENT

As mentioned previously, the second class of compressor is the positive displacement type. The five basic types that belong in this class are:

1. (a) Reciprocating.
   (b) Diaphragm.

2. Rotary (i) sliding vane
   (ii) multiple lobe
   (iii) liquid ring

Principle of Operation

All positive displacement compressors have the same principle of operation. In each, a prime mover causes a reduction in volume of air which in turn initiates an increase in pressure.

RECIPROCATING COMPRESSOR

Of all compressor types, the reciprocating compressors are the most widely used, available in a wide range of sizes and shapes.

Characteristics

Advantages

1. Efficiency - as described previously in Table 1, they are the most efficient machines for most applications. With appropriate capacity controls, piston units can also maintain their efficiency at partial loads.

2. Type of Gas Handled - they can be built to handle any commercial gas (provided corrosion problems are solved).

3. Lubrication - cylinders can be either lubricated or non-lubricated.

4. Pressures - high discharge pressures are obtainable.

Disadvantages

1.	Vibrations	- because these units suffer from large inertia forces which tend to shake the units, large, costly foundations are required.
2.	Pulsating Flow	- if pulsating flow is a problem, units must be supplied with pulsation dampeners (accumulators).

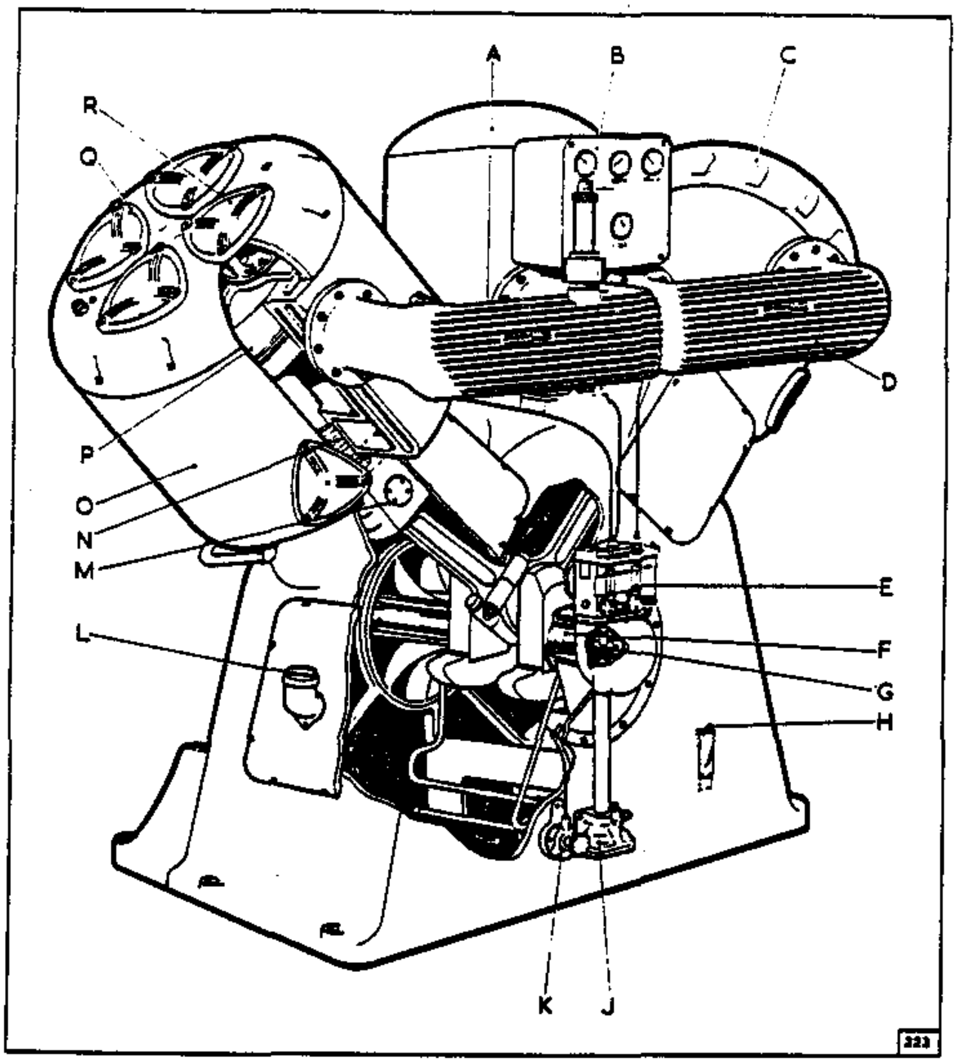
Description

Although these units appear complex on initial inspection, they can be simply described as positive displacement machines in which the compressing and displacing element is a piston having a reciprocating motion within a cylinder.

Figure 3 shows a schematic of a typical reciprocating compressor. For this example, a Broomwade model has been chosen to highlight the main components. Although models vary from station to station, components vary little.

This particular type is a two-stage, V-inclinded, double acting, intercooled, reciprocating compressor. An explanation of each term follows:

1.	Two-stages	- or two separate cylinders are used when it is necessary to have two steps of compression. Multi-staging a compressor reduces power consumption but also becomes necessary if conditions go beyond the capabilities of a single stage compressor to handle high discharge temperatures and stresses.
2.	Double-acting	- describes a unit which is capable of compression on both sides of a piston.
3.	V-inclined	- each cylinder is 45° from the vertical.
4.	Inter-cooling	- describes a process of cooling air leaving the first stage before entering the second stage to: (a) reduce the temperature of air. (b) reduce volume of air to be compressed in the succeeding stage. (c) save power. usually, the intercooler is a tube-in-shell water cooled heat exchanger.
5.	Reciprocating Piston	- is self-explanatory. Depending upon the system in which the compressor is used, the cylinder can be either lubricated or non-lubricated type employing **carbon** or **teflon** split rings.



Ref.	Part No.	Description	No.Off
A	C10190/213	Intercooler Sub-Assembly	1
B	C10190/474	Instrument Panel Assembly	1
C	C10190/471	H.P. Cylinder	1
D	C10190/132	Air Pipe - Intercooler to H.P. Cylinder	1
*E	A1413/727	Kirkham Two Feed Mechanical Lubricator	1
F	C10190/190	Crankshaft (N.C.I.)	1
G	C10190/178	Driven Gear (Pump Drive)	1
H	C10190/110	Dipstick	1
J	C10190/212	Oil Pump Sub-Assembly	1
K	A1413/481	Oil Filter	1
L	C10190/182	Oil Filler Cap Top	1
M	C10190/35	Crosshead	2
N	C10190/224	L.P. Gland Assembly	1
O	C10190/52	L.P. Cylinder	1
P	C10190/209	L.P. Piston Sub-Assembly	1
Q	C10190/199	L.P. Suction Valve Assembly	4
R	C10190/200	L.P. Delivery Valve Assembly	4

Figure 3

Diaphragm Compressors

Diaphragm compressors can be considered a special type of reciprocating compressor. In both cases, an oscillating prime mover physically displaces air. The diaphragm compressor would, of course, employ a diaphragm as a prime mover.

Although not as common as reciprocating piston compressors, diaphragm units offer a number of favourable characteristics.

Characteristics

1. In diaphragm units, the gas to be delivered is only in contact with the metal diaphragm. The gas is therefore compressed in a state of purity and without any lubricant.

2. There is no leakage of gas into the atmosphere.

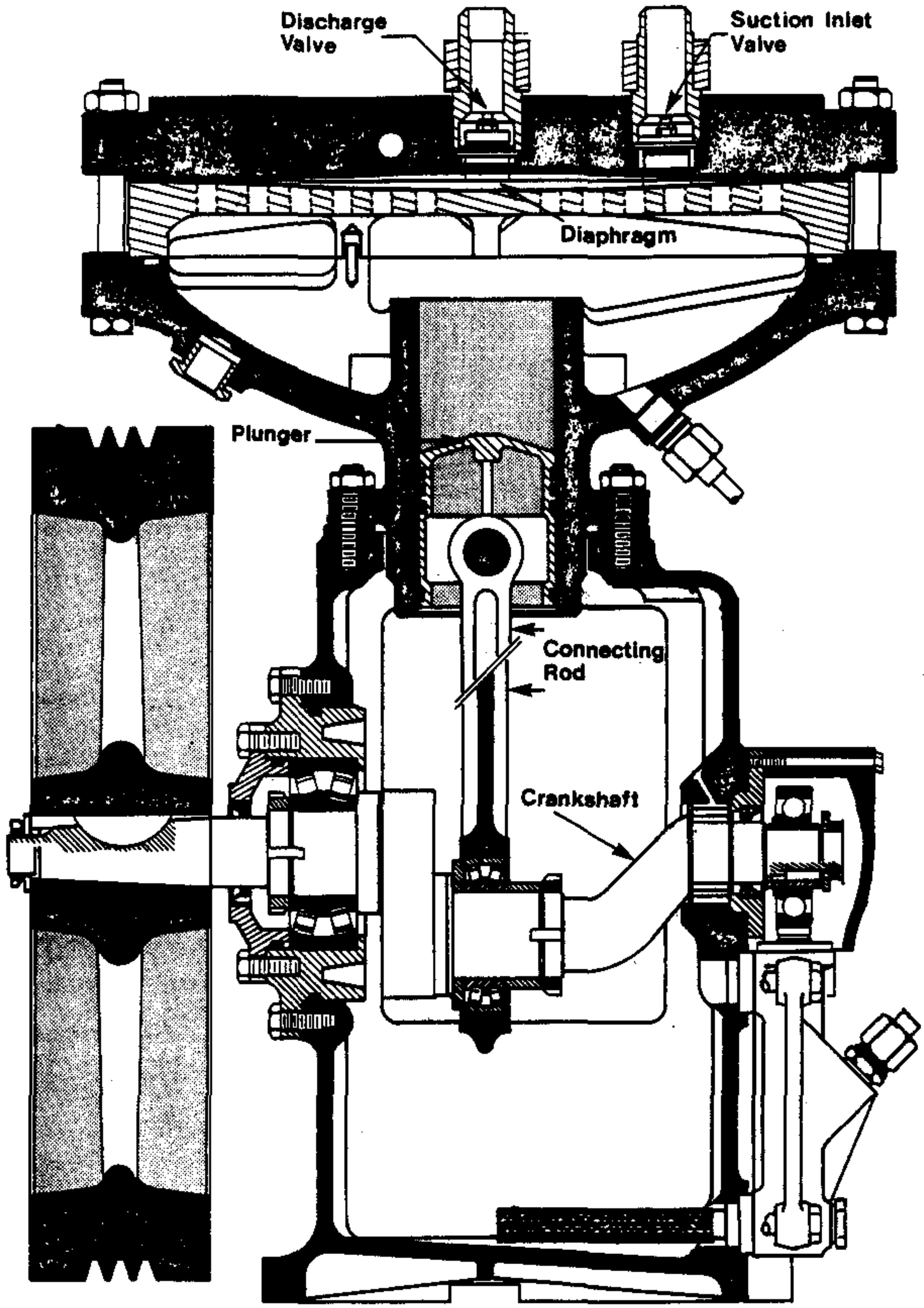
3. Because the diaphragm separates the gas completely from the oil, the compressor operates without any packed gland; this enables the compressor to operate with the least amount of power.

Description

These compressors shown in Figure 4 consist of two cylindrical plates hollowed out. Between the two plates, a flexible metallic diaphragm is gripped and held in position by bolts.

The lower plate is perforated to allow transfer of oil from the cylinder of the pump; the other plate carries suction and discharge valves.

A piston acts on the oil, with which the cylinder of the pump is filled and thus impacts motion to the diaphragm. This forces the diaphragm to contact the plate carrying suction and discharge valves to discharge compressed gas.



DIAPHRAGM COMPRESSOR

LIQUID RING COMPRESSOR

The liquid ring compressor uses a rotor with multiple forward turned blades turning about a central cone containing inlet and discharge ports driving a ring of liquid around the inside of an elliptical casing. As outline in Figure 5, a certain amount of liquid is trapped between adjacent blades and as the rotor turns, the liquid face moves in and out of this space due to casing shape. This creates a liquid piston.

Characterisitcs

Advantages

1. Non-Pulsating Air Flow

2. Oil-Free Air - lubrication is required only in the bearings external to the casing. The liquid acts as a coding, lubricating and sealing medium.

Disadvanges

1. Excess Moisture - because the water is a sealant, the air produced becomes water saturated. This therefore demands a separator or drier be used.

SCREW COMPRESSORS

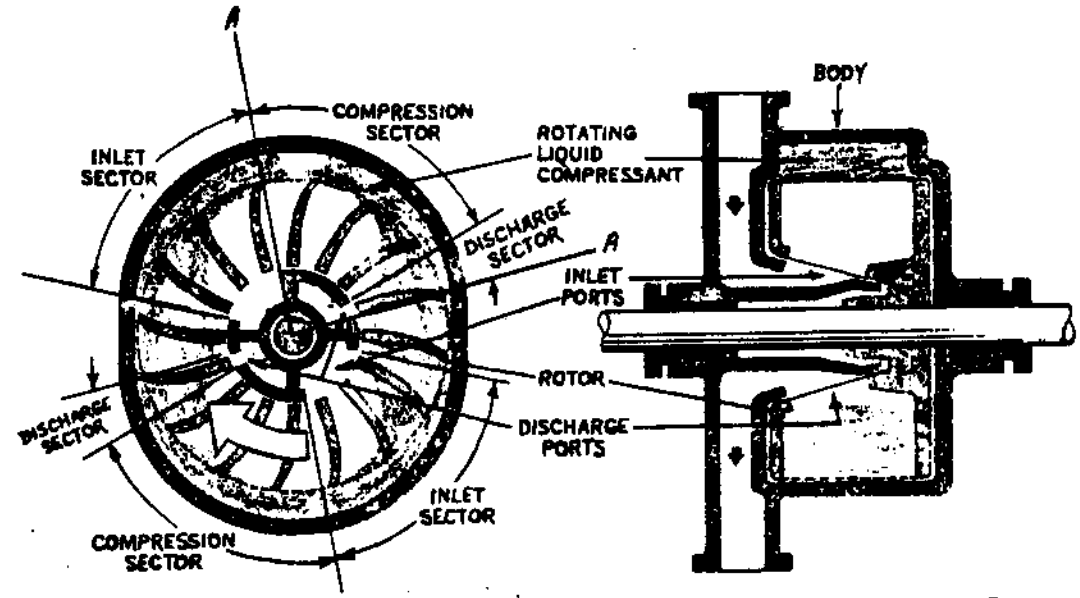
Helical or screw compressors are rotary, positive displacement machines in which two intermeshing rotors compress and discharge air. A description of the principle of operation is detailed in 430.10-6 and no further comment will be made.

A necessary component of a functioning screw unit is the external timing gears. Timing gears are used to transmit torque from driver to driven rotor and offer a number of advantages:

(a) Oil Free Air - since contact is external, the compression chamber is oil free. It should be noted however, that there are some screw compressors which do not have internal lubrication.

(b) Little Wear - results because of the small clearance space maintained between the two rotors.

(c) Little Maintenance - a result of little wear.



SCHEMATIC SECTION AT INLET AND DISCHARGE SECTORS
Through AA

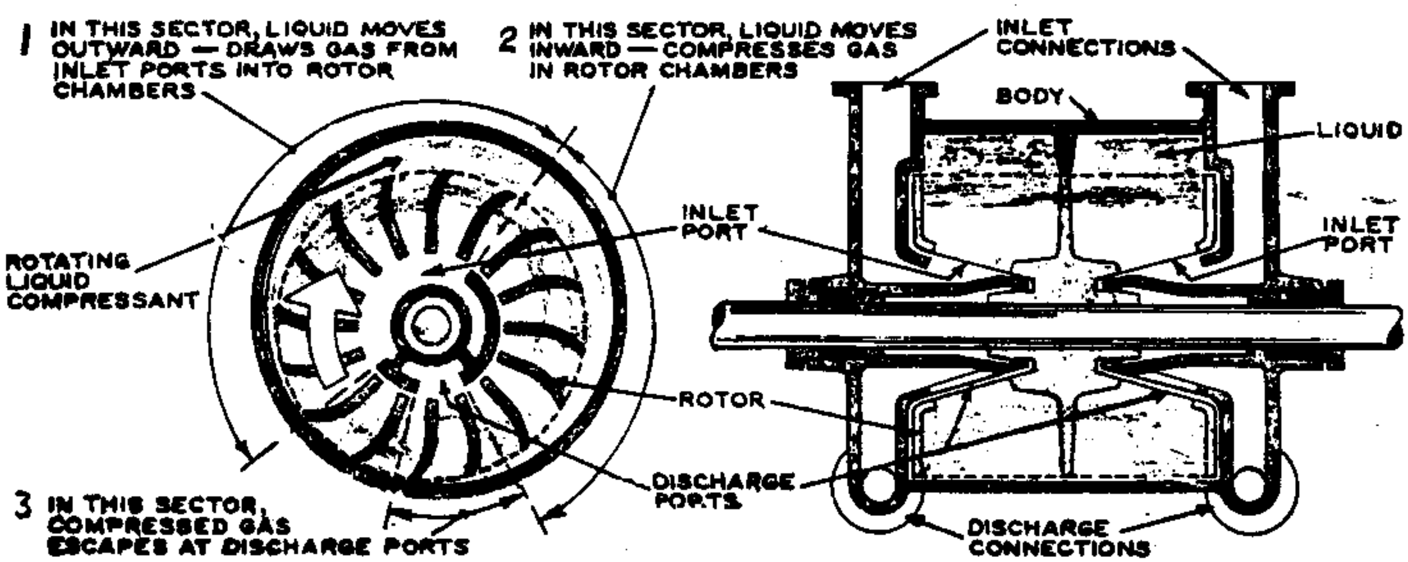


Figure 5

However, the disadvantage of their design are:

1. Low Discharge Pressures - because of leakage.

2. Lower Efficiency - as a result of recirculation of Air.

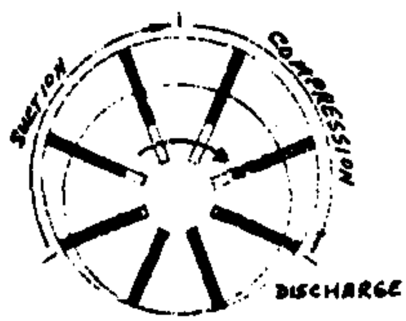
To maintain the clearances, the screw compressor must operate under certain limitations as follows:

1.	Discharge Temperature	- are limited to prevent excessive distortion of casing and rotors with consequent change in clearance.
2.	Temperature Rise Across the Unit	- to prevent excessive relative distortion of rotors.
3.	Pressure Differential	- limited for the same reason as temperature rise.

SLIDING VANE COMPRESSOR

Because sliding vane compressors are not used on any main air system in NGS, there will be only a brief description.

This rotary machine has as its basic element a cylindrical casing with rotor and head assembly. The general arrangement appears in Figure 6.



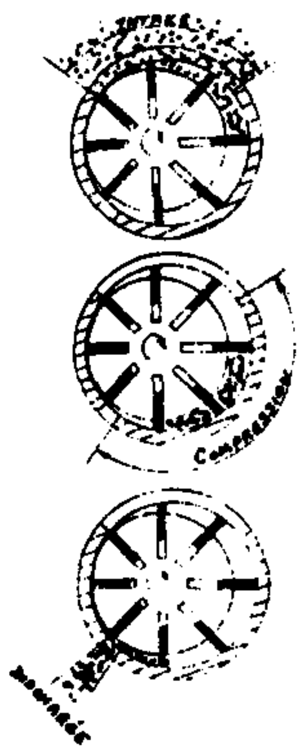


Figure 6

Compression takes place when the pocket volume decreases as the rotor turns.

With this type of compressor, it is important to maintain a lubricating film between vane and casing. The film not only reduces the wear but also acts as a sealant. Air entering the compressor should be clean to prevent any film disruption.

ASSIGNMENT

1. Name the four types of air moving devices. Describe how they are classified.

2. Compare the two main types of compressors with respect to flow rate, pressure and efficiency.

3. What advantages do centrifugal compressors have in comparison with reciprocating units?

4. What is one application of an axial compressor? Why has it been selected for this use?

5. Describe the problem of surge.

6. Discuss characteristics of reciprocating compressors.

7. Name the desirable characteristics of a diaphragm compressor.

8. Draw a typical reciprocating piston unit and explain the function of each main component.

9. Name two advantages of using a liquid ring compressor.

10. In the field, what screw compressor limitations should be monitored?

K.E. Keown

Mechanical Equipment - Course 430.1

AIR SYSTEM

Compressed air is a major source of industrial power with many advantages. It is relatively safe, economical, easily transmitted and adaptable. Its adaptability is evidenced by the many applications of compressed air in NGS.

There are four main air systems. For your convenience, a chart has been prepared describing each air system. As shown in the Table 1, each system demands its' own discharge pressures and special requirements.

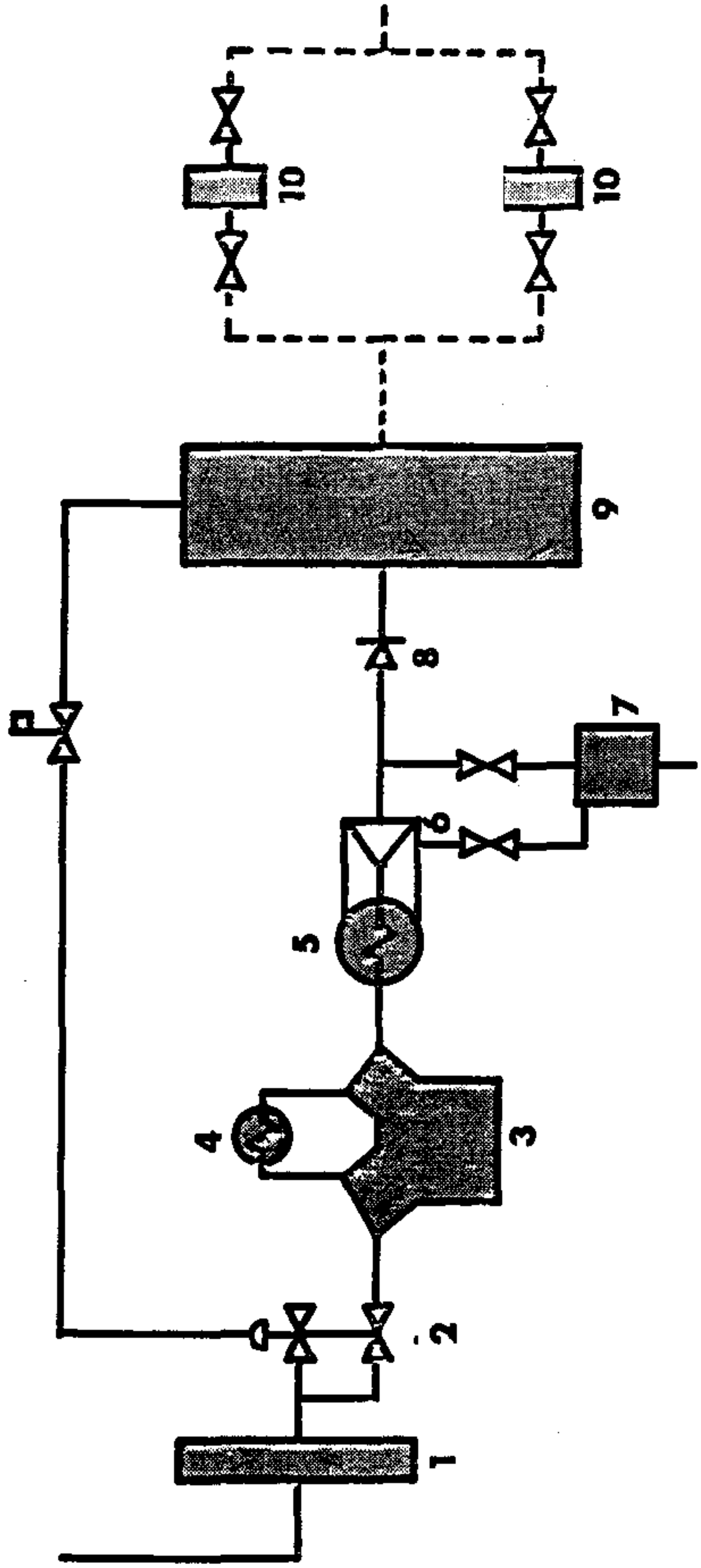
TABLE 1

SYSTEM	DISCHARGE PRESSURE		SPECIAL REQUIREMENTS	COMPRESSOR TYPE
1. Breathing Air	0.6 mPa(g)	(80 psig)	oil free	screw, liquid ring.
2. Service Air	0.9 mPa(g)	(130 psig)	-	Reciprocating piston.
3. Instrument Air				
H.P.	1.0 mPa(g)	(140 psig)	oil free	PNGS - Self lubricated Reciprocating piston.
L.P.	0.6 mPa(g)	(80 psig)	dry	
				BNGS - Lubricated Reciprocating piston with oil filter after compressor.
4. Switchyard Air	6.8 mPa(g)	(1000 psig)	oil free very dry	Self-Lubricated Reciprocating piston.

AIR SYSTEM - SCHEMATIC

Producing compressed air involves a number of devices. A typical air system with the major components appears in Figure 1. A description of each follows:

(1)	Filter/Silencer	- The main function of the filter is to ensure dirt particles do not enter the compressor. This prevents excessive sludge buildup which would lead to equipment failure.
(2)	Unloader Valve	- Controls compressor capacity.
(3)	Compressor	- Manufactures compressed air.
(4)	Intercooler	- A heat exchanger that reduces the temperature of air after the first stage that reduces power required.
(5)	Aftercooler	- Another heat exchanger, water cooled, which reduces temperature of compressed air to: (a) condense excessive moisture. (b) decrease volume of air to increase storage of air in air receiver.
(6)	Separator	- A device which separates moisture from the compressed air.
(7)	Trap	- Allows moisture to drain from separator without compressed air leakage.
(8)	Non-Return Valve	- Prevents the escape of compressed air from air receiver through a stopped compressor.
(9)	Receiver	- A storage vessel for compressed air.
(10)	Air Driers	- Are used to produce very dry air. Most air driers are molecular sieve types.


10
10
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4
3
2
1

COMPRESSOR CONTROL

Most of NGS air systems have varying demand characteristics. Occasionally, loads are heavy for only short time periods. Quite often, air system demand can be spread over longer time periods by use of receivers. In these circumstances, compressors may not be needed continuously.

Since capacity requirements do vary, some method of compressor control is essential. The method of control used by NGS is **automatic dual control** - a combination of unloading and ON/OFF process.

Unloading describes a process in which the compressor is operating but not compressing. There are three methods used in NGS to unload the compressors.

(1) Free Air Unloading
- with free air unloading, inlet valves are held open so that air discharge from the cylinder passes back into intake passages.
- usually found on 7" stroke compressors.

(2) Clearance Unloading
- used on compressors greater than 7" stroke.
- with clearance pocket unloading, the air is compressed into a clearance pocket when the piston travels in one direction and air returns to cylinder on the return stroke. Currently not used in NGS.

(3) Throttling Suction Air
- in this process, the air delivery is effected by throttling the inlet air whereby air delivery is completely stopped. Air present within compressor is then released to atmosphere.

The method of control used depends upon the type of compressor used. At Bruce NGS, for instance, throttling is used with screw type compressors, and free air unloading is employed in the instrument air - reciprocating piston compressors.

STARTING PROCEDURES

All compressors are unloaded before starting. Starting unloaded:

(1) reduces starting torque requirements. Since there is no compression at starting, power demands of the motor are reduced.

(2) enables the lubrication system to operate effectively before compression.

(3) enables the cooling water systems to operate effectively before compression.

ASSIGNMENT

1. Name the four air systems in NGS and describe the requirements of each.

2. Draw a typical air system showing the main components.

3. What method of compressor control is used in NGS?

4. Name and describe three methods of unloading.

5. How are compressors started? Why?

K.E. Keown

Mechanical Equipment - Course 430.1

FANS

Fans are devices designed to move air or, more generally, gases. No definite line exists between fans and their higher pressure counterparts - blowers and compressors. ASME (American Society of Mechanical Engineers) suggest in one of their standards that gas handling devices developing pressures below 13.8 kPa(g) (2 psig) be called fans.

The main application of fans in our plants is in ventilation and air conditioning systems. In these cases we have large flows at very low pressures, much lower than the mentioned limit of 13.8 kPa(g).

The principle of operation of fans is identical to that for centrifugal pumps or dynamic compressors. The air is accelerated by an impeller and subsequently slowed down by providing a larger cross-section in the casing or following ducting.

There are basically two types of fans:

1. Radial flow or centrifugal fans.
2. Axial flow fans.

CENTRIFUGAL FANS (Figure 1) consist of an impeller mounted on a shaft with the whole assembly rotating in a scroll-shaped housing. The housing and impeller, unlike centrifugal pumps and dynamic compressors, are welded from cut metal plates. Air enters the impeller axially, changes direction by 90° and leaves the impeller radially, entering the scroll-shaped casing where the pressure build-up is accomplished. Generally speaking, radial flow fans are used where higher pressures are involved, such as ventilation and air conditioning systems with extensive ducting grids. Examples are air conditioning systems at Pickering and Bruce NGS.

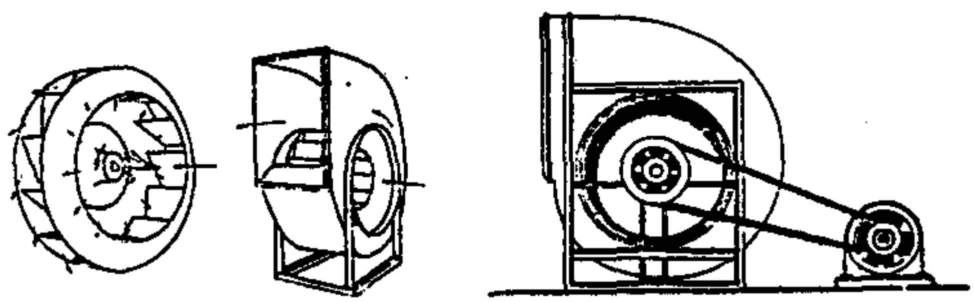


Figure 1

AXIAL FANS have two or more blades, usually air-foil blades mounted on a shaft. Air enters the impeller axially, is accelerated and leaves axially. Pressure build-up is accomplished in the discharge duct by installing in a diverging section or a diffuser.

The basic type is the PROPELLER FAN (Figure 2) whose blades spin in open atmosphere or are mounted in a plane sheet metal ring.

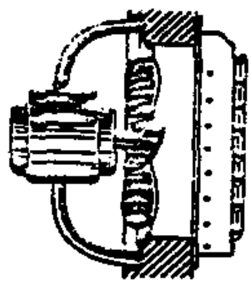


Figure 2

There is no ducting involved and this type of fan is used to move air just across the wall, in or out, or to induce air movement within the room.

The TUBEAXIAL FAN in Figure 3 has a surrounding cylinder and is used in a ducted system.

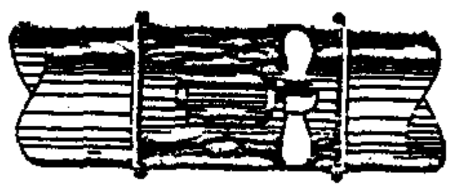


Figure 3

The VANEAXIAL FAN in Figure 4 has a set of air guide vanes mounted in a cylinder before or behind the airfoil-type impeller. They develop higher pressure than other axial fans and have begun to replace centrifugal fans in a number or services. Large fans of this type usually have variable-pitch blades which make them more versatile and economical for working under a wide range of operating conditions.

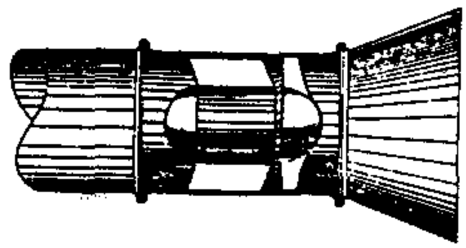


Figure 4

ASSIGNMENT

1. What are the two basic types of fans and what is the difference between them?

2. What is the difference between a centrifugal pump and a centrifugal fan?

3. What type of fan is usually found in systems with extensive grid of ducting?

K. Mika

Mechanical Equipment - Course 430.1

VACUUM PUMPS

A vacuum is generally considered to be a space, (a vessel, or system) containing air or any other gas at a pressure less than that of the atmosphere. There are several applications of vacuum in our nuclear plants, the major ones being the vacuum building, the condenser, shell side and condenser tube side (where vacuum establishes and perpetuates the syphon) and the finishing unit at the Bruce Heavy Water Plant.

Generally speaking there are two basic methods of producing a vacuum. In the first method gas or vapour molecules are physically removed from the space and exhausted to the atmosphere. This is accomplished by using vacuum pumps. In the second method the physical or chemical state of gas or vapour is changed. Typical methods are heat removal or a chemical reaction. It is within the scope of this course to explain only the first method, ie, to classify vacuum pumps.

Vacuum pumps can be classified into two groups: MECHANICAL and VAPOUR.

Mechanical Vacuum Pumps

The trainee at this point is familiar with operating principles of compressors, blowers and fans. It should not be of any surprise to find out that operating principles of mechanical vacuum pumps are identical to those of other gas handling devices; all of them take a suction from a lower pressure space, pressurize the gas and discharge it to a higher pressure space. The difference is that compressors, blowers and fans take suction from surrounding atmosphere and discharge to a close space or system at higher pressure, while vacuum pumps take a suction from a closed space at lower than atmospheric pressure and discharge to the surrounding atmosphere. Obviously, except for some minor design features, the devices can be and are almost identical.

In the design of mechanical vacuum pumps two principles are utilized: reciprocating or rotary or possibly a combination thereof. Both principles are characteristic of positive displacement devices. There are no dynamic vacuum pumps in existence.

The reciprocating principle is used in piston vacuum pumps. The construction is well known - the piston is moved back and forth by a crankshaft, usually driven by an electric motor. There may be one or more cylinders and pistons driven by a single crankshaft which can be mounted vertically or horizontally. The pistons are generally double-acting and, therefore, both sides of the cylinder must have suction and discharge valves. The application of piston vacuum pumps can be found in the Upgrading Unit (UPP) at Pickering NGS.

Another positive displacement reciprocating pump is a diaphragm pump but it is very seldom used in larger industrial applications.

There are several types of rotary pumps used for vacuum duties, most of which are again similar in general principle and design to rotary pumps used for normal pumping and compressing duties. In all cases there is a rotor which rotates within a hollow cylinder, which forms the body of the pump, the ends of which are closed by suitable plates. Rotary pumps can be dry running, oil drip lubricated or flood lubricated with oil or water. The inner surfaces of lubricated pumps are covered by a liquid film which forms a gas seal as well as cooling the pump.

The type most frequently found in our plants is a positive displacement, rotary, helical lobe, screw-type vacuum pump (Figure 1). They are used to create and maintain the

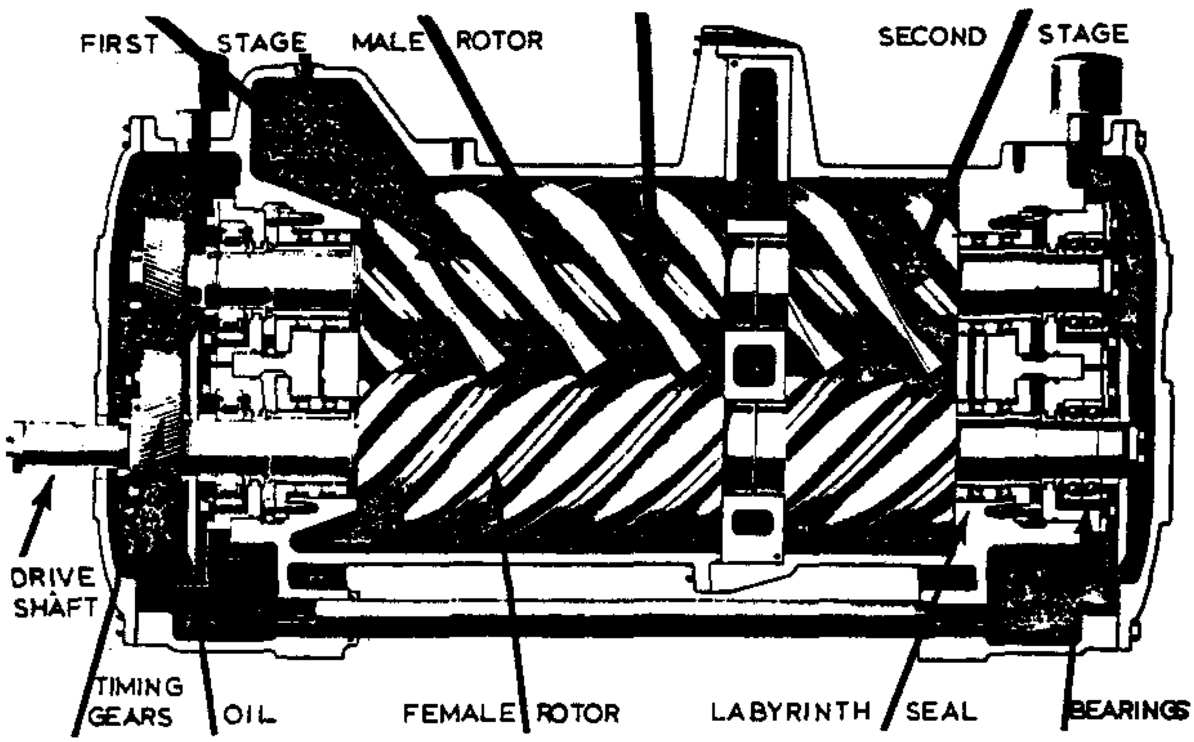


Figure 1

vacuum in the vacuum buildings at Bruce and Pickering NGS. The vacuum maintained there is 6.9 kPa(a) (2 in Hg abs). The same type of pump is also used on the main condenser (shell side) at Pickering NGS.

The displacement is obtained by meshing of two helical rotors on parallel shafts encased in a vertically split housing. Parts for suction and discharge are at opposite ends of the housing, with the inlet part at the driven end. The circular profile rotors are known as male and female rotors (Figure 2). Both are of the same diameter. Power input is

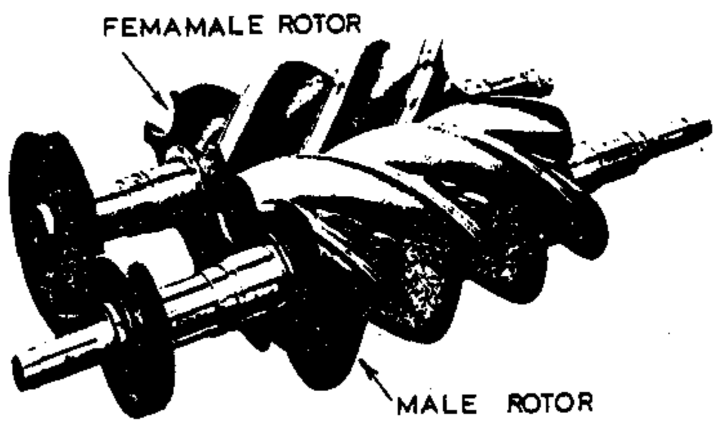


Figure 2

to the female rotor shaft. The torque is transmitted to the male rotor through the timing gears. In our case (pumps in the picture), it is a two-stage pump. These pumps can handle large quantities of air.

Another type of vacuum pump using the rotary principle is a sliding vane pump, (Figure 3). A rotor has diametrical slots which accommodate two or more free sliding vanes. They are held apart by springs that press them onto the surface of the stator. Inside the stator the rotor is mounted eccentrically with one point of it always in contact with the wall of the stator. The oil film between them forms an airtight seal.

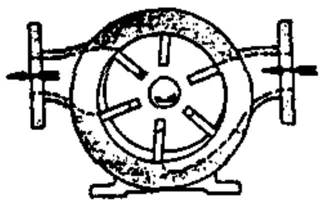


Figure 3

Lobe type vacuum pumps have lobed rotors which are gear driven to rotate at the same speed but in opposite directions and without touching each other. The rotors are enclosed within the casing or stator. As the clearances are from 0.005 to 0.01 inches and there usually is no oil seal, there is a back flow from the discharge to the suction. As there is no rubbing contact, higher speeds of rotation can be used. These vacuum pumps can handle considerable quantities of air which requires substantial cooling in the exhaust region.

The impellers come in a variety of shapes, two, three, four lobes. The two-lobe type, often called Roots, was shown in the lesson on positive displacement compressors. Figure 4 shows a four-lobe type.

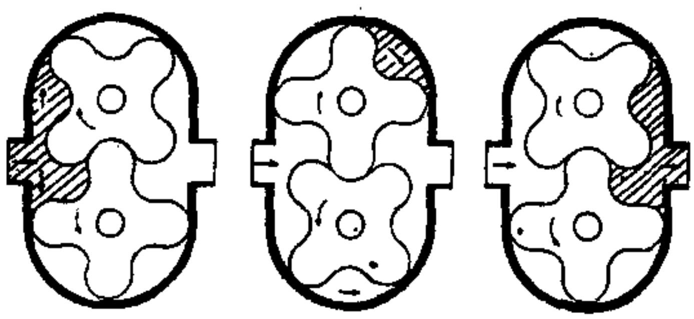


Figure 4

Rotary Air Pump in Figure 5 is a vacuum pump which uses rotary and centrifugal principles and to some extent a reciprocating action.

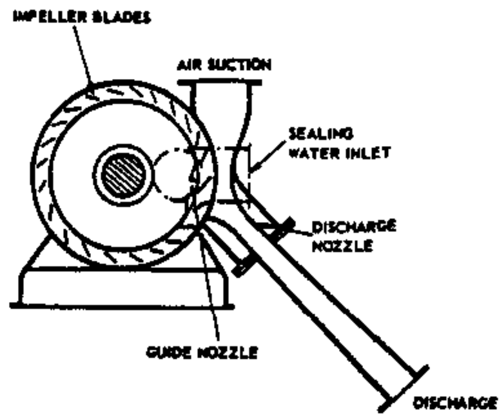


Figure 5

The impeller of the rotary air pump consists of a blanking plate mounted on a pump shaft. The closely spaced blades are attached to the rim of the blanking plate. Sealing water is fed into the pump and is broken into slugs by the impeller blades. These slugs of water pass to the discharge nozzle, draw the air in the effectively seal the pockets of air as they pass through the diffuser. From the diffuser, the air, non-condensable gases, and sealing water are discharged either to the drain or into a tank where water and gases are separated and water reused. The rotary air pump is used at NPD NGS on the main condenser shell side.

Finally there is a vacuum pump which use reciprocating, rotary and centrifugal action. It is called liquid-ring or liquid-piston vacuum pump. Again exactly the same principle is used in compressors. Figure 6 explains the pumping action.

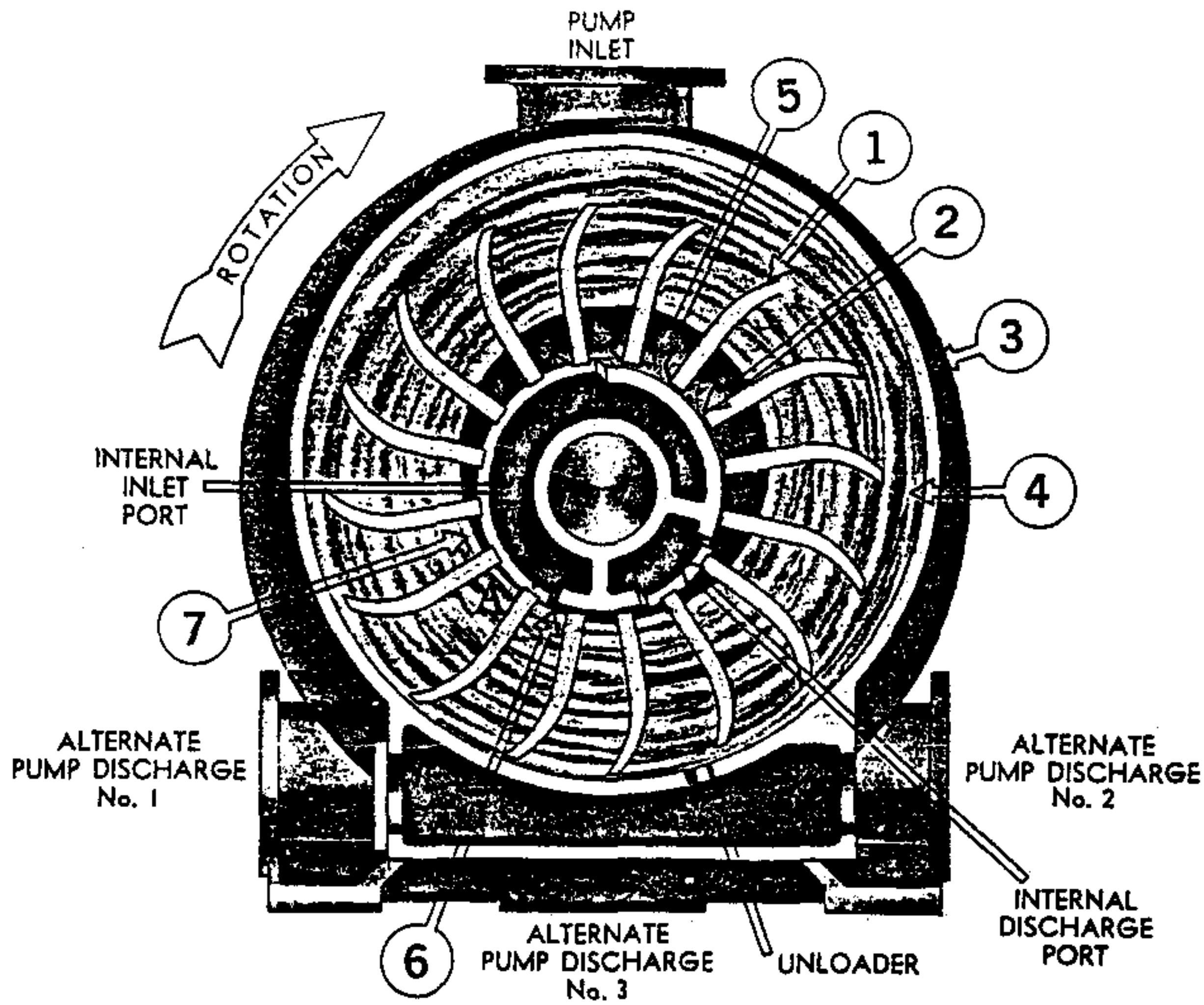


Figure 6

A rotor (1) revolves without metallic contact in a circular casing (3) containing liquid, usually water (4). The rotor is a casting consisting of a series of blades projecting from a hollow cylindrical hub. The blades are shrouded at the sides and form a series of chambers. Note that the curvature of the blades is in the direction of rotation unlike the rotor on centrifugal pumps. Starting at point "A", the chambers of the rotor are full of water. This water rotates with the rotor, but follows the contour of the casing (3), due to centrifugal force. The water (4) which entirely fills the rotor chamber at "A", recedes as the rotor advances until the rotor chamber attains its maximum size at (5). The converging casing forces the water back into the rotor chamber until it is again full at (6). This occurs once in each revolution. As the water is caused to recede from the motor chamber at (7), it is replaced by air drawn in through the inlet port in conical casting (2), connected with the pump inlet. As the rotor turns 360 degrees and the water is forced by the casing back into the rotor chamber, the air that has filled the chamber is forced through the discharge ports in the conical casting (2) into the pump discharge.

Liquid-ring pumps are used as vacuum priming pumps in CCW Systems (Condenser Cooling Water) for establishment and continuation of the syphon. They are also used in vacuum buildings in conjunction with screw pumps. Their function in this application is to keep vacuum in the upper vacuum chamber at the same level as in the main volume. In the event of an accident when the pressure rises in the main chamber of the vacuum building, the water from the storage tank will be pushed into the upper chamber and from there by gravity into the spray headers. If the vacuum was not maintained in the upper chamber, pressure there would resist water coming in and the dousing action would not be started. That, of course, is not tolerable.

Vapour Vacuum Pumps

Unlike mechanical vacuum pumps, vapour vacuum pumps operate on an entirely different principle which is not found in other gas pressurizing devices. A jet of vapour issuing from a nozzle is used as a means of pumping. A high velocity vapour jet is directed away from the pump inlet toward the pump discharge. A low pressure area is created, gas is drawn in and imparted directional velocity by the vapour jet and thus removed. There are two main types of vapour pumps - diffusion and ejector pumps.

Diffusion pumps use mercury or oil as a pumping fluid. The fluid is vaporized in the boiler by a heater, issued from the nozzle, recovered and returned to the boiler. These pumps are not suitable for large flows and are not used in our plants.

Ejector pumps use high pressure steam or compressed air as a pumping medium. Due to the availability of steam in our plants, steam ejectors are used at Douglas Point and Bruce NGS to establish and maintain vacuum in the condenser shell. At BHWP steam ejectors are used in the Finishing Unit for the distillation process. They are also used at the Upgrading Units at Pickering NGS. Steam ejectors are high capacity vacuum pumps. They are often designed as two or even more stage units. A typical steam ejector is shown in Figure 7.

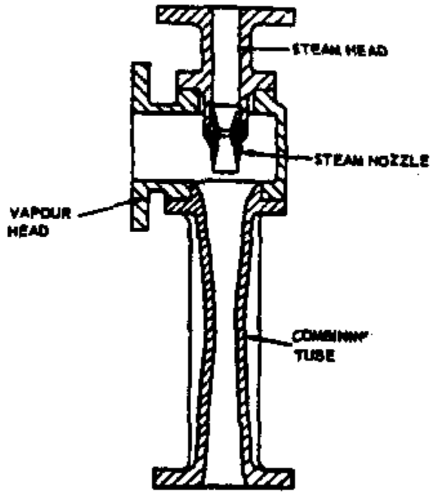


Figure 7

ASSIGNMENT

1. Give the basic classification of vacuum pumps.

2. What type of vacuum pumps are used:

   (a) in Vacuum Building main chamber?
   (b) in CCW Systems?
   (c) Finishing Unit at BHWP?

3. Explain the operation of:

   (a) liquid-piston vacuum pump.
   (b) steam ejector.

4. Which vacuum pumps are suitable for large capacities?

K. Mika

Mechanical Equipment - Course 230.1

COMPRESSORS

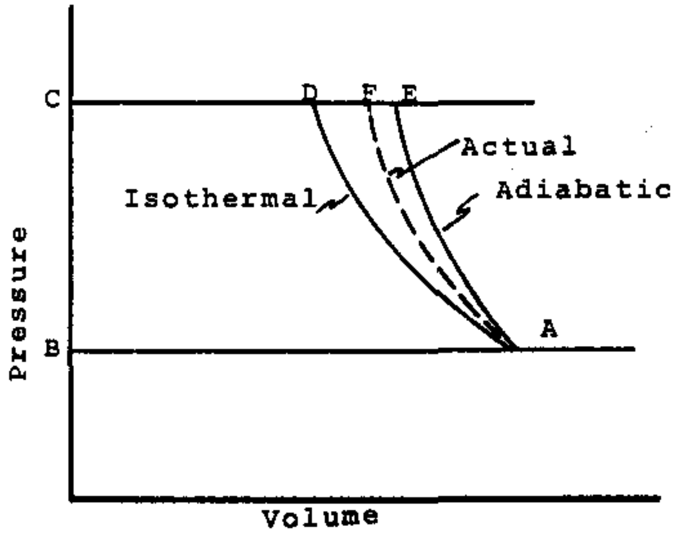
Various types of compressors were described in levels four and three mechanical equipment, the most important ones being reciprocating and rotary (liquid-ring) types.

Rather than discuss the physical characteristics of compressors in this lesson, the emphasis will be on the operational characteristics that operating people should be familiar with.

The Compression Process

In theory air may be compressed adiabatically (without the addition or removal of heat) or isothermally (at constant temperature). What this means is that the compression temperature of the air which has been adiabatically compressed will be much higher than the compression temperature of an isothermal compression. Since in most systems the compressed air is discharged into a receiver where it cools down to room temperature, that part of the compression energy which is subsequently lost by heat transfer through the receiver wall is wasted and represents increased power requirements at the compressor. The isothermal process which requires less work is therefore preferred and is approximated by cooling the air as it is compressed. Work is less because the air temperature is kept constant therefore the pressure is less as it is being compressed, therefore the piston doesn't have to do as much work. Keep in mind that this is an idealistic process.

Figure 1 illustrates a pressure volume diagram of the two compression processes. The dotted line AF is the actual compression curve, being closer to the adiabatic than to the isothermal. In a reciprocating machine it is not practical to have an isothermal process as it would have to occur over a long period of time. What is done however is that the cylinders are water jacketed, thus reducing the temperature somewhat. The area under each curve in Figure 1, isothermal (ADCBA), adiabatic (AECBA) and the actual (AFCBA) represents the work needed to compress the air. One can see that the adiabatic compression process requires more power. For example, an air system at 100 psig would result in 36 percent more power being required if it was an adiabatic compression process as opposed to an isothermal.



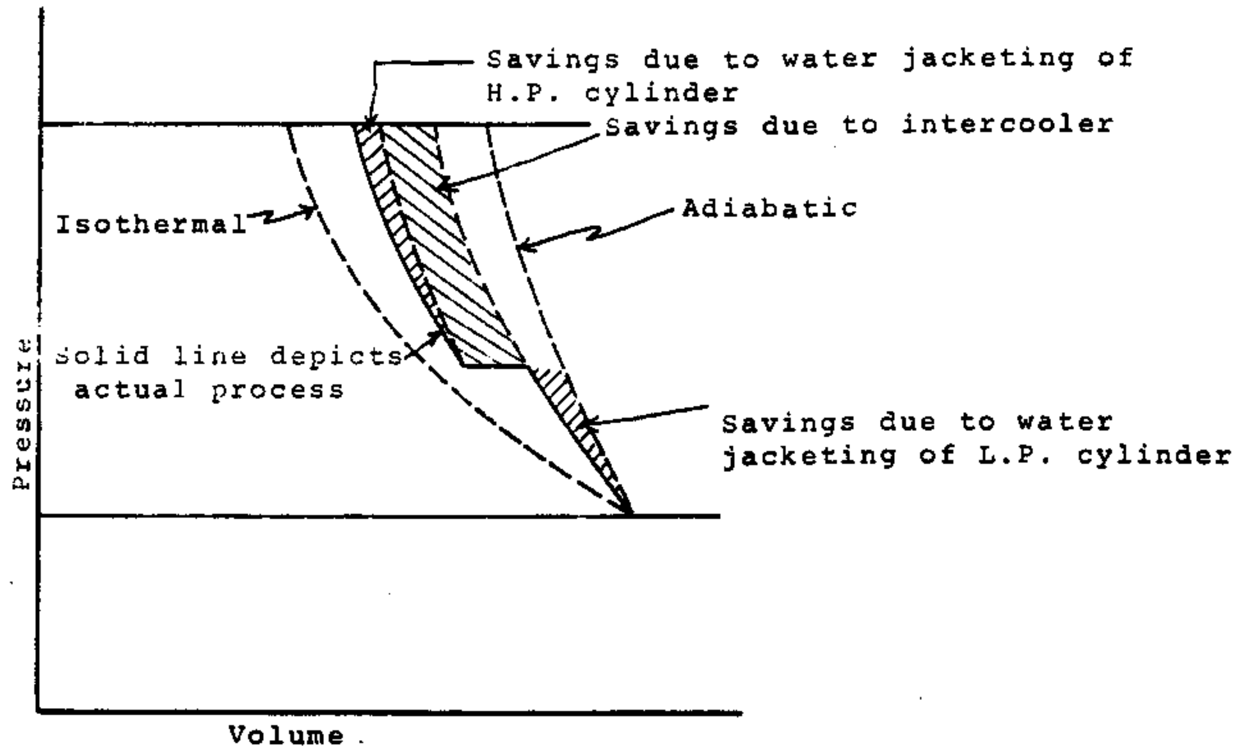
Compression Process

Figure 1

Multistage Compression

One practical method of minimizing the power losses arising from the heat of compression is to compress the air only part way to its final pressure, extract some of the heat then compress to the final pressure. This is done by compression in two or more stages. Cooling of the air between stages is then accomplished by passing it through an intercooler. The power caused by multistaging of positive displacement compressors depends upon several factors such as the ratio between the suction and discharge pressure, the cooling mediums used and the effectiveness of the intercooler.

Figure 2 illustrates the power savings effected by two-staging a water cooled reciprocating compressor and also indicates the saving due to water jacketting of the cylinders.



Multistage Compression Process

Figure 2

Compressor Capacity

Capacity of a compressor is the amount of air or gas measured at suction condition that is compressed and actually delivered through the discharge valves in one minute. The piston displacement is the volume swept by the piston in one minute.

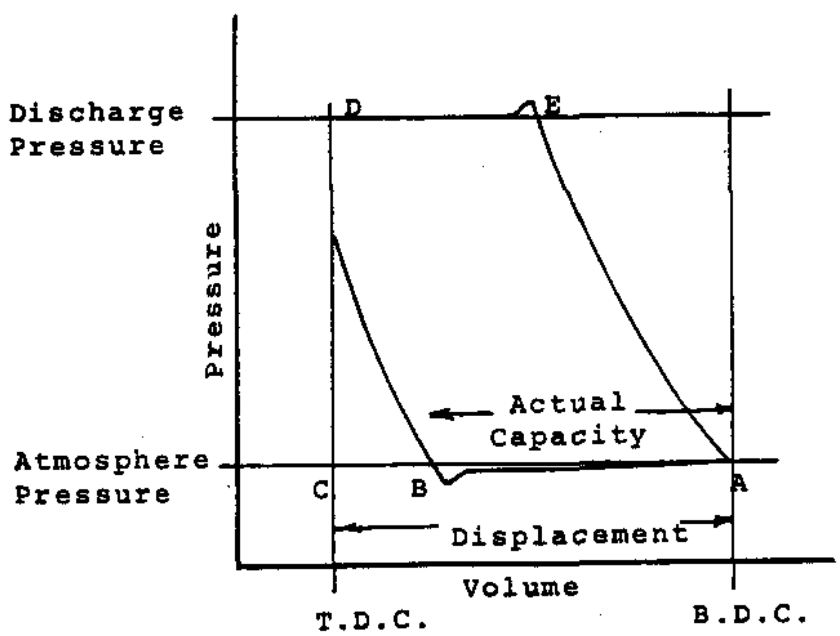
The actual delivered capacity divided by the piston displacement is referred to as the volumetric efficiency of the compressor and is expressed as a percentage. If air were an incompressible fluid, a compressor's capacity would be approximately equal to the piston displacement. However, since air expands and contracts with change in pressure and temperature, the actual capacity is always less than the piston displacement. This is also due to the necessary clearance between the piston and the cylinder head at the end of the stroke and to valve openings in the cylinder. At the end of the compression stroke, this clearance space is filled with air that has been compressed but not delivered through the discharge valve. On the return stroke this air will re-expand in the cylinder until the pressure is below the intake pressure which allows the intake valves to open.

Therefore, additional air is not admitted to the cylinder until the piston has travelled part way through its return or intake stroke. It can be seen in Figure 3 that suction valves open at point B and discharge at point E. It also becomes apparent that for a given cylinder volume, the greater the clearance, the later the suction valves will open and the lower the volumetric efficiency will be.

The volumetric efficiency involves every factor that causes a compressor to deliver less air that its piston displacement, and includes reduction in capacity due to preheating of air, slippage of air past piston rings and reversal of air flow through valves before closing.

Compression efficiency is the ratio of the horsepower required for a reversible adiabatic compression (theoretical process) to the horsepower actually expended in the cylinders, known as indicated horsepower. Indicated horsepower is the brake horsepower at the compressor shaft minus mechanical horsepower losses in the compressor.

Compressor efficiency is the ratio of the theoretical horsepower for a reversible adiabatic compression to the shaft horsepower input. It is equal to the product of the compression efficiency times the mechanical efficiency.



Effect of Cylinder Clearance on Volumetric Efficiency

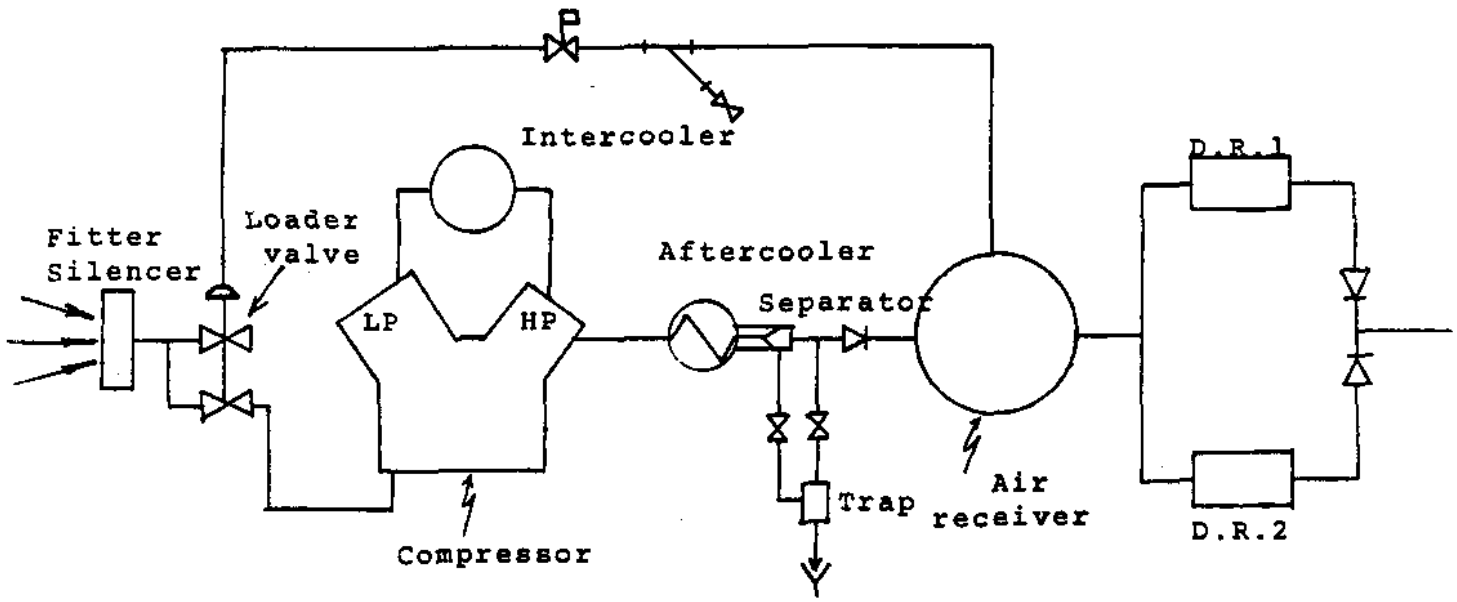
Figure 3

The Compressed Air System

In a simple air compressor system, air is drawn through a filter and the suction valves into the compressor where it is compressed. It is then forced through the discharge valves into an air receiver which acts as a storage tank or accumulator. This receiver assists in making regulation easier and aids the unloader in maintaining steady pressure. On most systems, compressor accessories such as intercoolers, aftercoolers, separators, traps, and dryers will be found. The functions of these accessories were discussed in Level 3 Mechanical Equipment so little more will be said here. However, it is worthwhile to look at typical systems one might find in the plants.

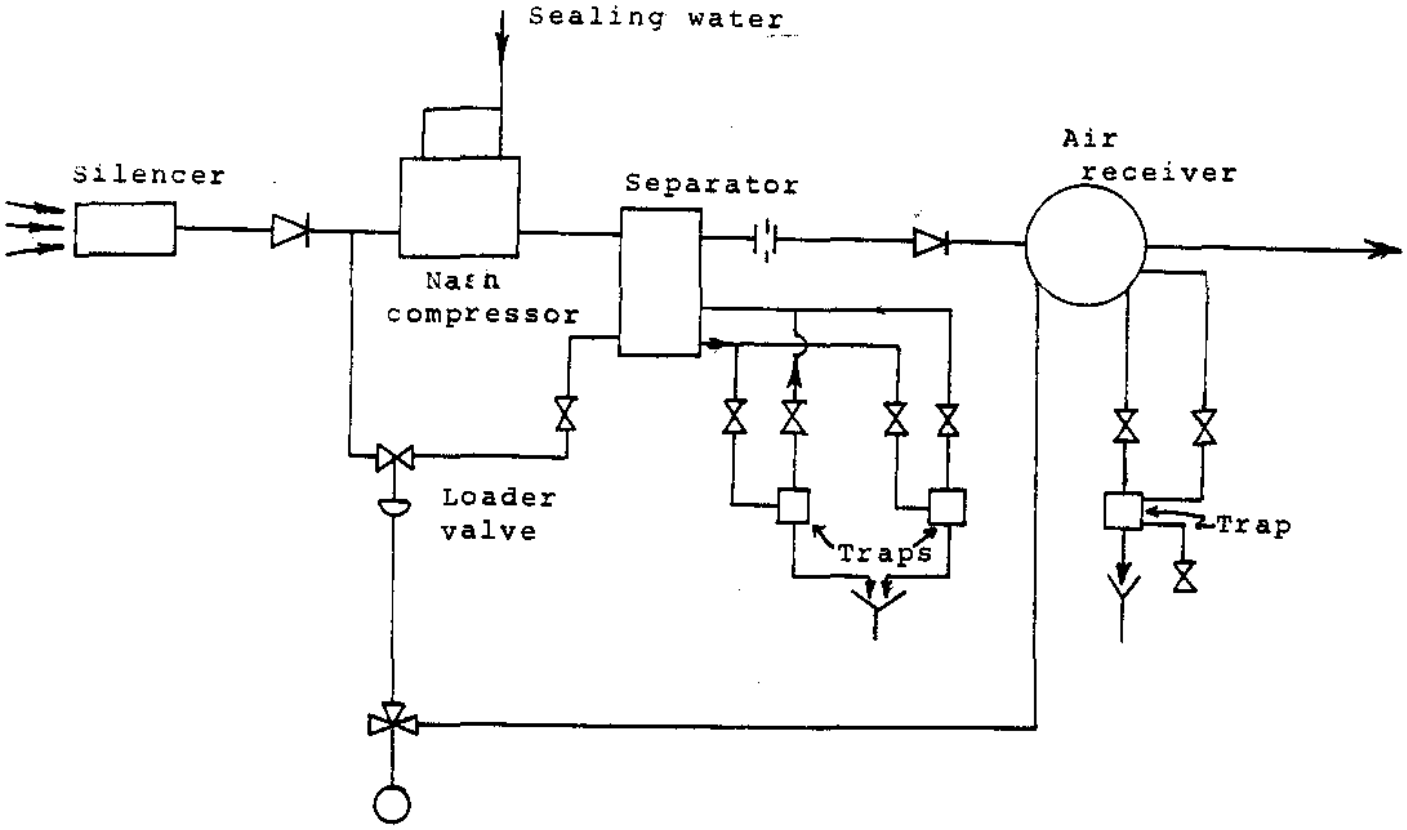
Figure 4 illustrates the high pressure instrument air system at Pickering G.S. This system consists of 6 Broomwade oil-free compressors of the two-stage double-acting type Vee apposed cylinders. The accessories as shown in Figure 4 consist of combination intake filter and silencer, intercooler, aftercooler, separators and dryers.

Figure 5 illustrates the breathing air system at Pickering G.S. This system provides moist oil and particulate-free air to breathing air stations in the reactor buildings. Two compressor units serve the entire station (enough to support 30 men in plastic suits). The two compressors are of rotary water seal type (Nash). The accessories with this system are an intake silencer, separator and receiver.



High Pressure Instrument Air Compressor

Figure 4



Breathing Air Compressor

Figure 5

As discussed in Level 3 there are various methods of controlling compressors and that at one plant the common method was the single step, unloading method. In the case of the H.P. Instrument air compressors (Figure 4) a pressure switch connected to the air supply header of the particular system controls the compressor. When the compressor is on auto position (compressor will run on demand of the appropriate pressure switch) and the pressure switch sensing a lag in pressure is closed, a solenoid valve on the low pressure service water supply opens to provide cooling water to the jackets and intercoolers and the compressor motor breaker closes starting the compressor. After a 10-second delay, the solenoid valve on the air supply to the pneumatic loader closes which results in the loading valve on the intake line to open, loading the compressor. When the demand decreases the cycle is reversed, the loader valve closed to unload the compressor. The motor continues to run for 10 minutes before the breaker opens, at which time the solenoid valve at the cooling water supply closes.

The breathing air compressor starts on demand of the appropriate pressure switch. The bypass solenoid valve opens to admit bypass water to maintain minimum flow to the compressor, and following a one-second delay, the sealing water solenoid valve opens to admit sealing water and the solenoid valve between the air receiver and the compressor loader opens to load the compressor. Decrease in demand results in the compressor being unloaded, bypass opens, and sealing water valve closes, after 10 minutes compressor stops.

Running Unloaded

Compressors, be they reciprocating or rotary should always be started unloaded. The most obvious reason for starting unloaded is to reduce the starting torque of the machine. On large reciprocating machines, the starting torques needed to overcome the static friction in the bearing etc, can be quite high, therefore starting loaded would only compound the problem.

Another reason for starting unloaded is to allow the working parts to become thoroughly covered with a lubricant, and allowing time for adequate cooling water flow to be established.

Common Compressor Troubles

Compressor valves are a very common source of trouble. Because of the nature of their design, that they are free to move vertically within a certain clearance, they tend to wear and the clearance increases, thus greater travel which results in even greater wear and noise, eventually leading to leakage.

What are the symptoms of compressor valve problems? The quickest way to spot valve trouble in a two-stage compressor is to look at the intercooler pressure. This pressure varies with the size of the cylinder and the intake pressure; it is usually 26 to 30 psi for a machine with atmospheric intake and 100 psi discharge. High intercooler pressure means trouble in the high pressure cylinders resulting from leaky intake or discharge valves, leakage caused by rings in poor condition, or a badly worn cylinder bore. Most compressor valves have slight leakage, but if you hear a continuous blow, it indicates excessive leakage.

If intercooler pressure is below normal, the trouble is generally in the low pressure cylinder.

The chart in Table 1 identifies some of the troubles that occur with compressors and also points out the possible causes.

Explosions have been known to occur in compressed air systems. In some cases explosions have occurred because of inadequate lubrication. If the cylinder is oil lubricated, it is important to choose an oil with the right viscosity and flash point. If the oil is very sluggish, it collects and mixes with dirt and forms hard deposit, mostly carbon which clogs piston rings and valves.

These carbon deposits, in quantity, plus moisture and heat are gas producers. The principle gas being formed is CO_2 which ignites at 1204°F. Thus an explosive mixture needs only ignition to produce an explosion. A leaky discharge valve would allow some of the hot compressed air to re-enter the cylinder to be re-compressed and temperature increased. This in time could supply the ignition temperature. Therefore it is most important that:

(a) the correct oil is used (check manufacturer's specifications)

(b) that the valves are kept in good condition.

Explosions can occur when piping to the safety valve and unloader become frozen due to low temperatures and an accumulation of moisture in the lines.

Explosions can also occur due to dirty intercoolers, defective unloading and failure of cooling water supply.

Corrosion, cracks or erosion in an air receiver can cause ruptures. Metal wastage can be kept to a minimum by ensuring that moisture is never allowed to accumulate and by periodic inspection. Excessive water in the lines could also result in slugs of water travelling through the pipe which, on closure of a valve, could result in water hammer with possibility of a rupture occurring.

To ensure reliable operation of a compressor it should be inspected at definite intervals. Suction and discharge valves should be removed and examined periodically. Intercoolers should be cleaned and lines checked for leaks. Crankshaft, crossheads and main bearings should be inspected regularly. In other words preventive maintenance should be practiced.

SYMPTOMS	POSSIBLE CAUSE
Noise or knocking	Loose or burned out bearings, loose valve or unloader, loose flywheel, motor rotor shunting back and forth from unlevel mounting or belt misalignment.
Squeal	Motor or compressor bearings tight, belts slipping, lack of oil, leaking gasket or joint.
Intercooler safety valve blows while running unloaded	Broken or leaking hp discharge valve, or suction unloader, defective or stuck lp unloader, blown gasket.
Intercooler valve blows while running loaded	Broken or leaking hp discharge or suction valve, defective hp unloader held in unloaded position, blown gasket.
Sudden capacity drop	Bad leak in air operated equipment or air lines, discharge piping clogged, suction filters blocked, broken or badly leaking valves, blown gaskets, leak in intercooler.
Gradual capacity drop	Accumulation of small leaks in air lines, poorly seating valves, restricted suction filters, worn rings or cylinders.
Receiver safety valve blows	Defective pop valve, defective pressure switch or pilot valve, leak in control line, inoperative suction unloaders.
Unit blows fuses	Fuses too small, low voltage, pressure switch differential setting too close, unit starting against full load, electrical trouble, motor or compressor tight.
Unit will not start	Blown fuse or tripped overload relay, motor or electrical trouble, defective pressure switch, motor or compressor binding.
Roughness and vibration	Base too light, improper shimming under unit, foundation bolts loose, unbalance from one cylinder not pumping.
Excessive oil consumption	Oil level too high, oil viscosity too light, too high oil pressure (if forcefeed lubricated), worn rings and cylinders

Troubleshooting Chart

Table 1

ASSIGNMENT

1. Explain the difference between adiabatic and isothermal compression.

2. What compression process actually occurs in a compressor?

3. What is the advantage of having a two-stage compressor over a single stage compressor?

4. Define:

   (a) Volumetric efficiency
   (b) Compressor efficiency
   (c) Compressor capacity.

5. For what reasons are compressors started unloaded?

6. What are the symptoms of leaky compressor valves?

7. Give two reasons why explosions might occur in a compressed air system.

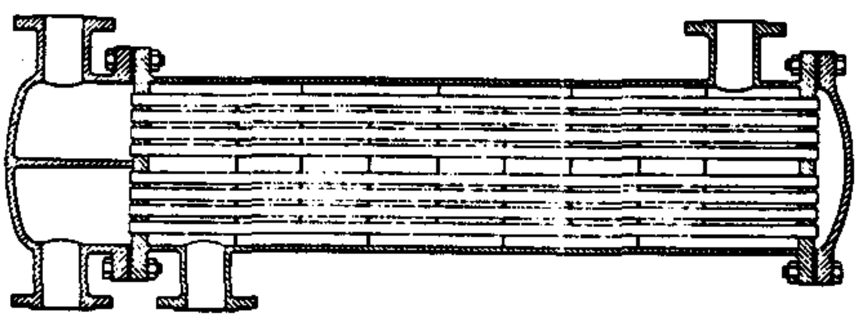
Mechanical Equipment - Course 330.1

HEAT EXCHANGERS

The basic principles of operation of heat exchangers were discussed in Lesson 430.11-1. In this lesson the construction of various types of heat exchangers will be dealt with. Also a more detailed look at the various types of heat exchangers found in nuclear power plants will be undertaken.

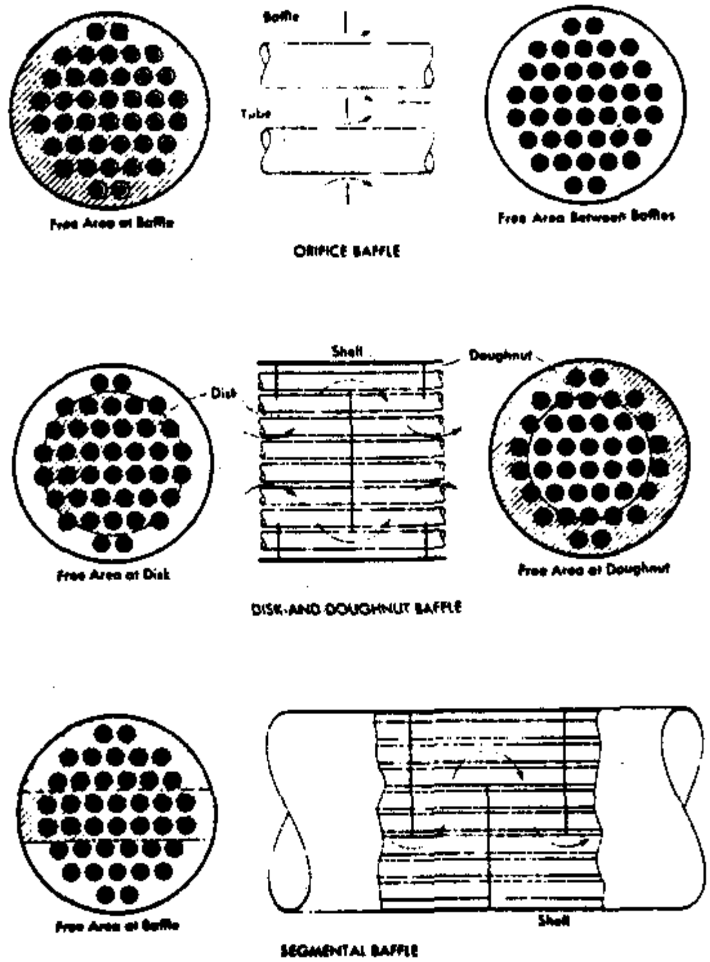
Construction

In Lesson 430.11-1 the basic type of heat exchanger, the tube and shell type, was discussed. The three basic flow patterns utilized with this type are parallel-flow, counter-flow and cross-flow, with the most efficient being the counter-flow method. In order to increase the effective heat transfer surface area per unit volume, most commercial heat exchangers provide more than a single pass through the tubes. The fluid flowing outside the tubes in the shell is routed back and forth by means of baffles. Figure 1 illustrates a cross section of a heat exchanger with two tube passes and one cross-baffled shell pass. The baffles are of the segmental type. This and other typical types of baffles are shown in Figure 2.



Shell and Tube Heat Exchanger

Figure 1



Three types of baffles used in shell and tube heat exchangers

Figure 2

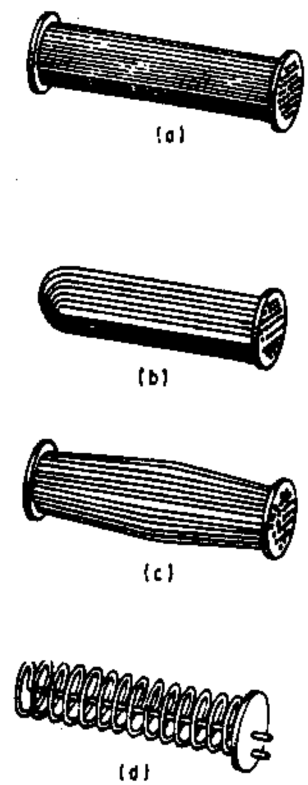
The heat exchanger illustrated in Figure 1 has fixed tube plates at each end and the tubes are welded or rolled into place. Other types of tube bundles used in shell and tube heat exchangers are illustrated in Figure 3.

Figure 3(a) shows the straight tube bundle. The advantage of this type is that it is easy to clean mechanically. To accommodate tube expansion, this type of bundle usually has a free end, commonly referred to as a floating head. Figure 7 illustrates a heat exchanger designed with a floating head.

Figure 3(b) shows a U-tube bundle. This type of bundle solves tube expansion problems. Also there are only one-half as many tube joints. This tube bundle is harder to clean.

Figure 3(c) illustrates a bowed tube bundle. It is bolted solidly to the shell at each end. As the bundle heats, it bows causing scale to crack off.

Figure 3(d) is the coil tube bundle which is used for very high pressures. It does away with gasketed joints in the high pressure circuit.



Four Typical Tube Bundles used in Heat Exchangers

Figure 3

The end plates into which the tubes are sealed are referred to as "Tube Sheets". These are the portions of a heat exchanger where leakage between the two fluids is most likely to occur. Figure 4 illustrates three types of tube joints.

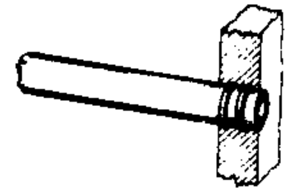
Applications

Shell and tube heat exchangers are among today's most popular design. Typical applications are: -

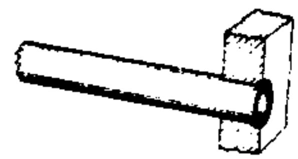
1) Feedwater heating
2) Surface condensers
3) Compressor inter-and-aftercoolers
4) Refrigeration condensers
5) Refrigeration evaporators
6) Lub-oil coolers

Typical types of heat exchangers for these applications will now be briefly described.

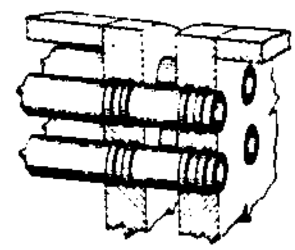
Rolled Tube Joint is the most common form of fastening tubes in tube sheets. Cold rolling flows the tube metal into annular grooves cut in the tube-sheet holes.



Welded Tube Joint usually remains tighter than a rolled joint where considerable expansion must be handled. Sometimes rolled joints are also welded.



Double Tube Sheets give positive protection against accidental leakage of shell fluid into tube fluid or the reverse. If tube joints leak in either tube sheet, the leakage can be collected in the cavity between the two sheets.



Tube Joints

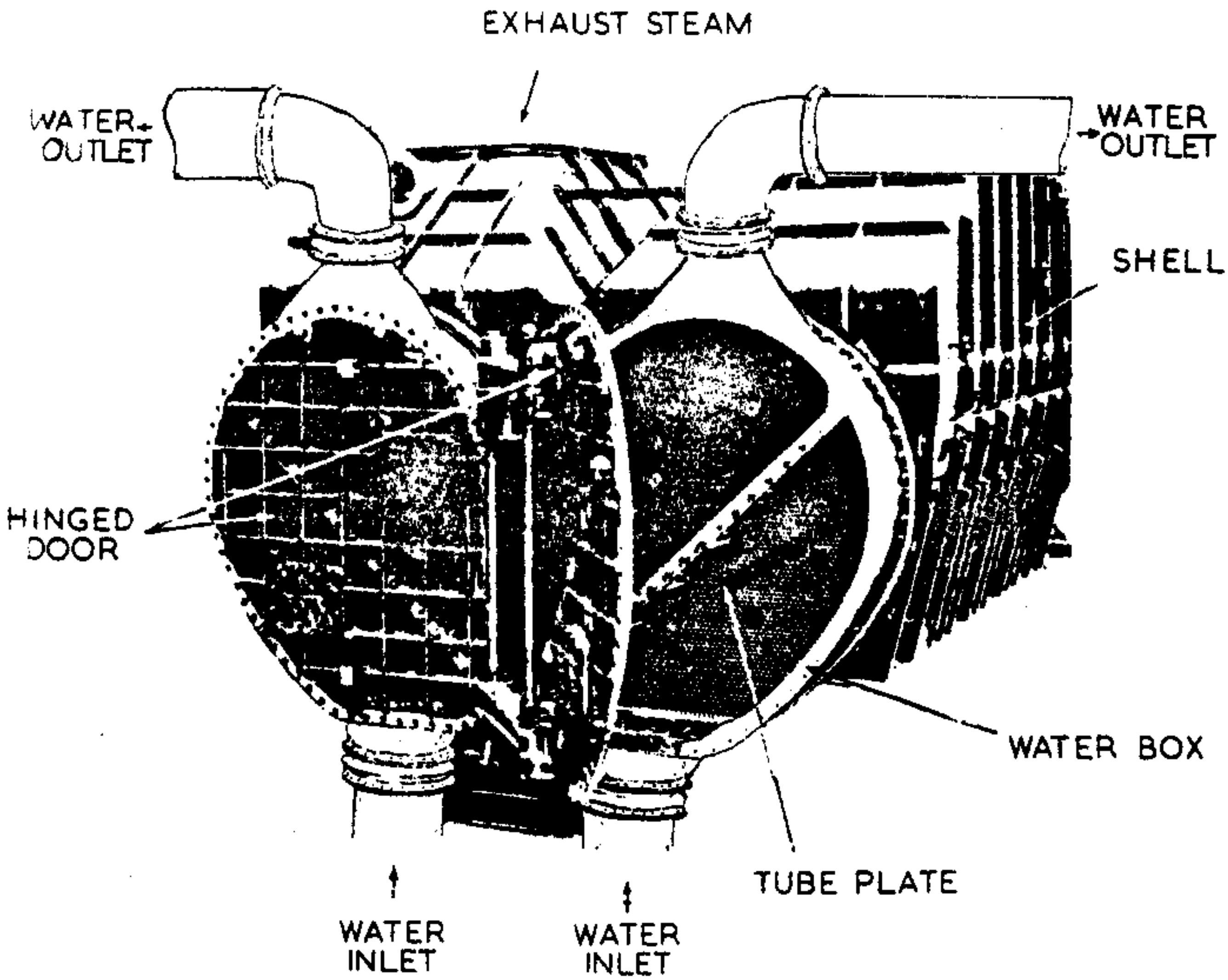
Figure 4

Surface Condenser

Steam surface condensers perform two functions; first, they recover condensate for boiler feed. As this water must be very pure, this results in a big saving in the cost of purifying the feedwater. Second, they reduce the back pressure on the turbines so maximum heat energy can be extracted from the steam.

Surface condensers are basically a vacuum tight shell and tube heat exchangers with cooling water flowing through the tubes and prime-mover exhaust steam surrounding the tubes. The steam condensed, collects in a hot well and is introduced in the feedwater system. To maintain vacuum various types of vacuum pumps can be used to remove air which leaks in and gases such as oxygen which is given off by the condensing steam. A more detailed description of the principle of operation of the surface condenser is given in the turbine, generator and auxiliaries course, therefore, little more will be said on this aspect of surface condensers.

Figure 5 illustrates a typical surface condenser. The particular condenser is of the twin shell construction, the shells being connected by means of a balance line. This type of condenser would be used where turbines have two steam exhausts. The shells are more or less independent of each other, having separate water inlets and outlets. This is advantageous in that repairs could be made to tubes in one of the shells without having to shut down completely. The condenser illustrated in Figure 5 is of two pass construction.



A Typical Condenser

Figure 5

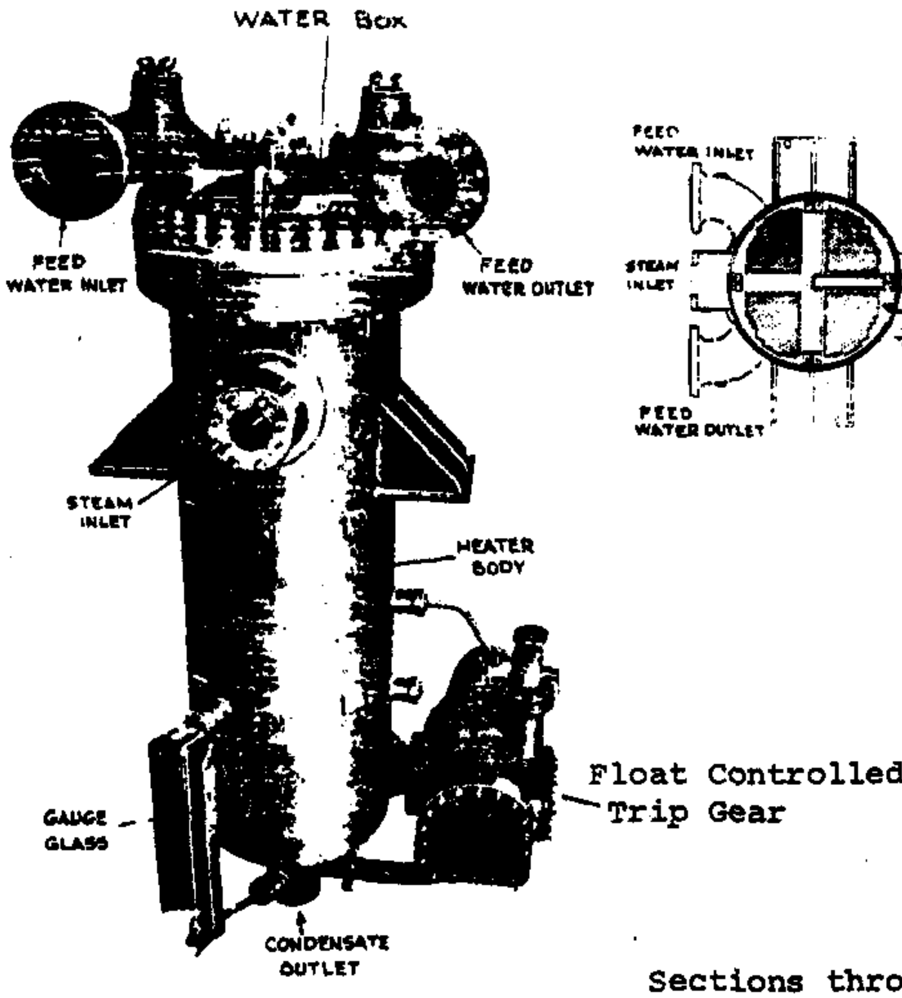
The name "surface condenser" is very appropriate because it contains a great number of cooling tubes. For example, the condenser tubes for the Pickering unit provides a cooling surface of 280,000 sq ft. It contains 27,000 one-inch diameter, 40 foot tubes. It requires 313,000 gallons per minute of lake water. The operating pressure for the water boxes is approximately 30 psig.

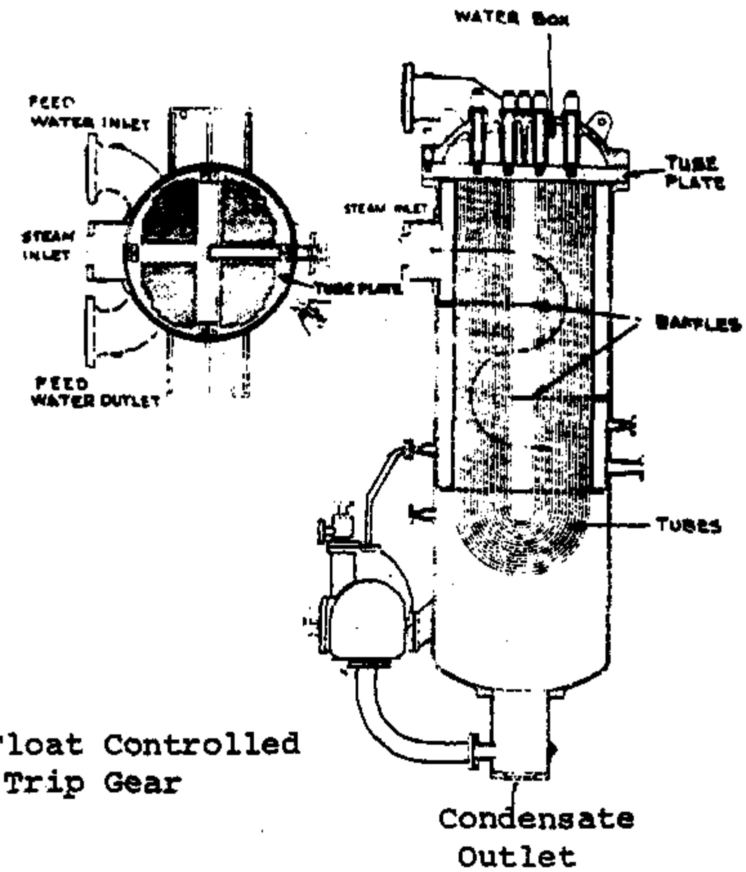
Tube sheets and tubes in condensers are normally made of compositions of non-ferrous metals such as admiralty brass. The criteria used for the metals chosen are usually resistant to corrosion and erosion, high thermal conductivity and cost. The shell is usually of a welded steel construction. (Cast iron is sometimes used for the shell material). The water box is usually made of cast iron. Hinge doors are illustrated in Figure 5. This facilitates cleaning or retubing the condenser.

Feedwater Heaters

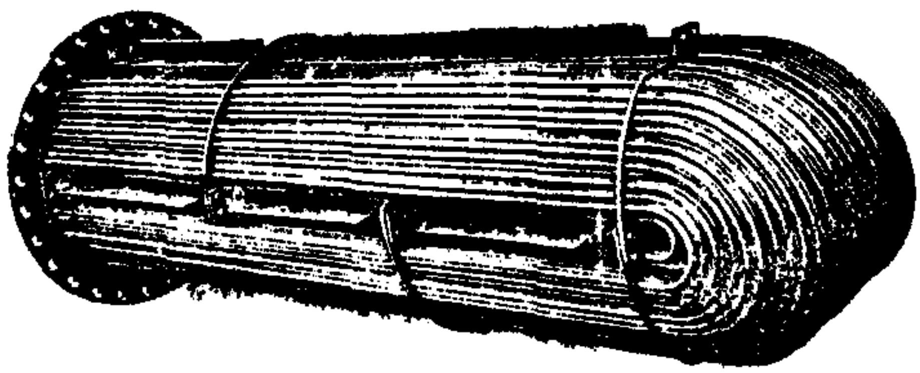
Feedwater heaters normally use steam bled from turbines as the energy supply. Their prime purpose is to raise cycle thermal efficiency by the regenerative heating principle. Modern plants will have low pressure and high pressure heaters.

Figures 6 and 7 illustrate typical vertical and a horizontal feedwater heaters respectively. Both heat exchangers illustrated are high pressure heaters, however low pressure heaters would be very similar in construction. Figures 6 and 7 illustrate two methods used for accommodating differential expansion. One is to use a U-tube and the other is to use a floating head. Generally in high pressure heaters the tubes are U-shaped. Horizontal-closed feedwater heaters as shown in Figure 7 require little headroom but need a clear floor space in front of the heater so that the bundle can be withdrawn from the heater. Vertical heaters may be used when headroom is ample and floor space is at a premium.





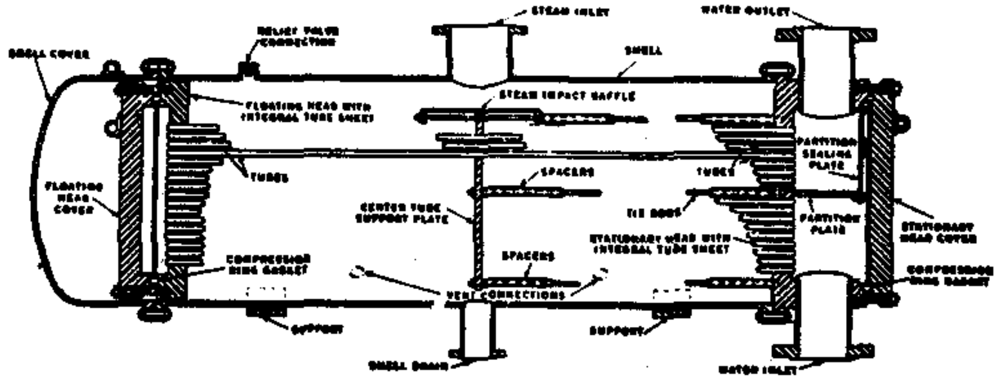
Sections through Heater to show Water Box and Steam Flow



Tube Nest

H.P. Feedwater Heater

Figure 6



Horizonal H.P. Feedwater Heater

Figure 7

It is interesting to note that both Douglas Point and Pickering generating stations utilize horizontal low pressure and high pressure U-tube heaters.

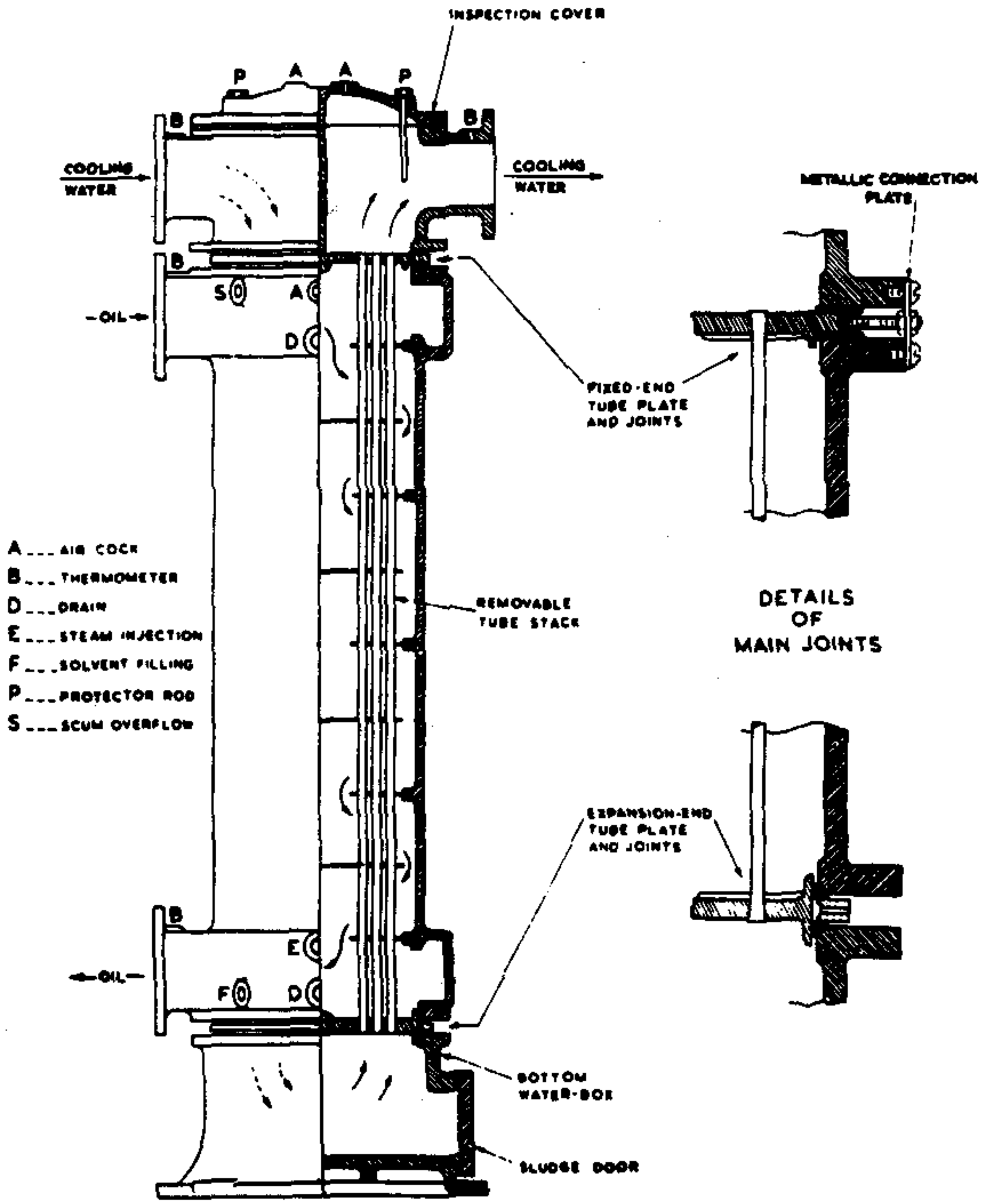
The material for these heaters is usually carbon steel because of the high operating pressures.

As steam condenses in either a vertical or horizontal heater, the condensate falls to the shell bottom and forms an effective seal against steam blowing through. This seal also prevents noncondensible gases from leaving the heater. The accumulation of these gases, however, can reduce the heaters effectiveness, therefore at each end of the steam space in the shell, vents are located. In addition to removing gases, the vents help to distribute the steam more uniformly throughout the shell.

Turbine Oil Coolers

The purpose of oil coolers is to cool the oil which is being supplied to the steam turbine bearings to a temperature of approximately 110-120°F. A typical oil cooler is illustrated in Figure 8. Cooling water flows through a two pass tube bundle while the oil passes over the outside of the tubes being directed by a series of internal baffle plates. Most coolers are placed in an upright position as illustrated in Figure 8. It is general practice to maintain the oil pressure in the cooler at a higher pressure than the circulating water to ensure that, in the event of tube failure, there will be no ingress of water into the oil system.

The coolers usually consist of a cast iron shell inside of which is a nest of brass tubes.



Typical Oil Cooler

Figure 8

General

As indicated in the earlier part of this discussion, there are many applications of tube and shell heat exchangers in Power Plants. Most of the examples taken were from the conventional side of power plants. There are many examples that could have been taken from the primary heat transport system and moderator system of Nuclear Power Plants. Some that come to mind are the main steam generators, which are U-tube and shell design, the shut down cooling heat exchangers, bleed condensers and moderator cooling heat exchangers.

All of these would be basically similar in construction to the ones already discussed. The main differences would be in the added precaution taken to ensure minimum heavy water leakage. This would involve welded tube joints, double gasketed joints, seal welds, etc.

ASSIGNMENT

1. Explain the purpose of increasing the number of tube passes in a heat exchanger.

2. What useful purpose does a double tube sheet perform?

3. Describe the methods by which axial expansion can be accommodated in heat exchanger tube nests.

4. For what purposes are vents installed in feedwater heaters?

5. Give two applications of tube and shell heat exchangers in a thermal power plant.

L.J. Laplante

Mechanical Equipment - Course 430.1

PIPING, TUBING AND JOINTS

Operation of all our plants depends on the transport of fluids (liquids, vapours and gases). This vital function is accomplished by using **fluid conduits.** This name is used to distinguish them from electrical conduits which are used for a different purpose and a substitution by each other is not acceptable.

There are two types of fluid conduits: **PIPES** and **TUBES** . Although they are used basically for the same purpose there is a number of distinguishing features which will influence the decision which one to use for a particular application. These features are listed and discussed in the following text.

SIZES

Piping is available in a limited number of sizes which are standardized for each type of material. An example of a standard for a carbon and alloy steel pipe is in the Table 1. Generally speaking any size outside this table would have to be specially ordered by the customer and it will be called tubing. Inspecting the table the first column specifies the **Nominal Pipe Size** (N.S.) in inches. The second column gives the Outside Diameter (O.D.) in inches. Up to and including N.S. 12" the N.S. is different from O.D. For N.S. 14" and up N.S. corresponds with O.D.

Wall thickness of a pipe is specified by the **Schedule Number** (Sch. No.). Columns three to 12 give wall thicknesses for different nominal sizes and various Schedule Numbers. Logically the Inside Diameter (I.D.) of a pipe is obtained by subtracting two wall thicknesses from the O.D.

Example: Find I.D. of the pipe N.S. = 1" Sch. No. 40, 80 and 160, from the Table:

			Nominal Wall Thicknesses		
	N.S.	O.D.	Sch. No. 40	80	160
	1"	1.315"	0.113"	0.179"	0.250"
I.D. =			1.089"	0.957"	0.815"

The O.D. for a particular nominal size remains identical for various Schedule Numbers, but the I.D. is changing accordingly.

Another important conclusion is that the nominal size up to 12" does not correspond with either O.D. or I.D.

CARBON AND ALLOY STEEL PIPE

		Nominal Wall Thickness									
Nominal Pipe Size	Outside Diameter	Sched. No. 10	Sched. No. 20	Sched. No. 30	Sched. No. 40	Sched. No. 60	Sched. No. 80	Sched. No. 100	Sched. No. 120	Sched. No. 140	Sched. No. 160
1/8	0.405	.....	.....	.....	0.068	.....	0.095	.....	.....	.....	.....
1/4	0.540	.....	.....	.....	0.088	.....	0.119	.....	.....	.....	.....
3/8	0.675	.....	.....	.....	0.091	.....	0.126	.....	.....	.....	.....
1/2	0.840	.....	.....	.....	0.109	.....	0.147	.....	.....	.....	0.187
3/4	1.050	.....	.....	.....	0.113	.....	0.154	.....	.....	.....	0.218
1	1.315	.....	.....	.....	0.133	.....	0.179	.....	.....	.....	0.250
1-1/4	1.660	.....	.....	.....	0.140	.....	0.191	.....	.....	.....	0.250
1-1/2	1.900	.....	.....	.....	0.145	.....	0.200	.....	.....	.....	0.281
2	2.375	.....	.....	.....	0.154	.....	0.218	.....	.....	.....	0.343
2-1/2	2.875	.....	.....	.....	0.203	.....	0.276	.....	.....	.....	0.375
3	3.500	.....	.....	.....	0.216	.....	0.300	.....	.....	.....	0.438
3-1/2	4.000	.....	.....	.....	0.226	.....	0.318	.....	.....	.....	.....
4	4.500	.....	.....	.....	0.237	.....	0.337	.....	0.438	.....	0.531
5	5.563	.....	.....	.....	0.258	.....	0.375	.....	0.500	.....	0.625
6	6.625	.....	.....	.....	0.280	.....	0.432	.....	0.562	.....	0.718
8	8.625	.....	0.250	0.277	0.322	0.406	0.500	0.593	0.718	0.812	0.906
10	10.750	.....	0.250	0.307	0.365	0.500	0.593	0.718	0.843	1.000	1.125
12	12.750	.....	0.250	0.330	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14	14.000	0.250	0.312	0.375	0.438	0.593	0.750	0.937	1.093	1.250	1.406
16	16.000	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.438	1.593
18	18.000	0.250	0.312	0.438	0.562	0.750	0.937	1.156	1.375	1.562	1.781
20	20.000	0.250	0.375	0.500	0.593	0.812	1.031	1.281	1.500	1.750	1.968
24	24.000	0.250	0.375	0.562	0.687	0.968	1.218	1.531	1.812	2.062	2.343
30	30.000	0.312	0.500	0.625	.....	.....	.....	.....	.....	.....	.....

TABLE 1

Pipes made of various materials having identical Nominal Size and identical Schedule Number will have identical O.D. but different I.D., because wall thicknesses for identical Schedule Numbers will be different.

Tubing in general is any fluid conduit with sizes outside the standard for pipe. Certain sizes of tubing are so much in demand that they are commercially available and do not have to be specified and ordered. Table 2 gives basic dimensions for commercially available tubing. It should be noted that the tubing is commercially available to 2" O.D. only. But up to this size, selection of wall thicknesses is much wider than in piping.

Tubing used for fluid transport is usually called **hydraulic tubing** to differentiate it from so called **mechanical tubing** which is used in structural design and does not conform with sizes in Table 2. Bicycle frame is an example of mechanical tubing application.

There is one more type of fluid conduit whose sizes do not conform either with pipe specifications or with tube specifications. It is called **streamline copper water tubing** and is available in four types, K, L, M, and DWV. Type K is of highest quality and is used for hot and cold water, gases and steam. Other types are used for less severe services than K, such as drains, vents and low-pressure service. Popularity of this conduit is based on ease of field erection.

SPECIFICATIONS

Piping is specified by the nominal size in inches and by the schedule number, for instance 2" Sch. No. 40 + material specifications. Originally the nominal size indicated the inside diameter of the pipe. However, the range of wall thicknesses now available provides a range of inside diameters.

Tubing is specified by the outside diameter in inches and by the wall thickness usually expressed as a decimal inch or rarely as a wire-gauge number.

For example: 1/4" 0.049 + material specifications.

TOLERANCES

Piping wall thickness tolerance is up to ±12% of the nominal dimension.

Table —Basic Dimensions for Commercially Available Tubing

Outside Diameter d_o(in.)	Wall Thickness t(in.)	Inside Diameter d_i(in.)	Outside Diameter d_o(in.)	Wall Thickness t(in.)	Inside Diameter d_i(in.)	Outside Diameter d_o(in.)	Wall Thickness t(in.)	Inside Diameter d_i(in.)	Outside Diameter d_o(in.)	Wall Thickness t(in.)	Inside Diameter d_i(in.)
⅛	0.028	0.069	½	0.035	0.430	⅞	0.049	0.777	1¼	0.120	1.010
	.032	.061		.042	.416		.058	.759			
	.035	.055		.049	.402		.065	.745	1½	0.065	1.370
				.058	.384		.072	.731		.072	1.356
3/16	0.032	0.1235		.065	.370		.083	.709		.083	1.334
	.035	.1175		.072	.356		.095	.685		.095	1.310
				.083	.334		.109	.657		.109	1.282
¼	0.035	0.180								.120	1.260
	.042	.166	⅝	0.035	0.555	1	0.049	0.902			
	.049	.152		.042	.541		.058	.884	1¾	0.065	1.620
	.058	.134		.049	.527		.065	.870		.072	1.606
	.065	.120		.058	.509		.072	.856		.083	1.584
				.065	.495		.083	.834		.095	1.560
5/16	0.035	0.2425		.072	.481		.095	.810		.109	1.532
	.042	.2285		.083	.459		.109	.782		.120	1.510
	.049	.2145		.095	.435		.120	.760		.134	1.482
	.058	.1965									
	.065	.1825	¾	0.049	0.652	1¼	0.049	1.152	2	0.065	1.870
				.058	.634		.058	1.134		.072	1.856
⅜	0.035	0.305		.065	.620		.065	1.120		.083	1.834
	.042	.291		.072	.606		.072	1.106		.095	1.810
	.049	.277		.083	.584		.083	1.084		.109	1.782
	.058	.259		.095	.560		.095	1.060		.120	1.760
	.065	.245		.109	.532		.109	1.032		.134	1.732

TABLE 2

Tubing wall thickness tolerance is up to ±10% of the nominal dimension, but the outside diameter is manufactured to close tolerances, for example 1/4" tubing will have O.D. within +0.004, -0.000 inch.

PLIABILITY - BENDING

Piping - Although it is possible to bend a pipe, this time-consuming and skill-requiring procedure is seldom used. Instead, fittings-elbows are used.

Tubing - is made of relatively malleable materials and can be bent easily, up to 1/2" manually. This reduces number of fittings, hence less leakage, better appearance and lower erection cost.

MATERIALS

Piping - Most common material of piping in our plants is steel. Majority of applications are satisfied by carbon steel. Stainless steel is found mainly in the Moderator System and certain Hydrogen Sulfide Systems.

Cast iron is used for low pressure water distribution and sewage disposal.

Wrought iron pipe is rare.

Copper alloy pipe is not much used.

Plastic pipes are slowly gaining popularity, particularly where corrosion is a problem and the system is at low pressure and temperature. Materials used are PVC, Polyethylene, ABS, Polybutylene and fiberglass reinforced epoxy or polyester.

Tubing - Carbon Steel tubing is used for steam tracing and generally on the turbine.

Stainless Steel tubing is used on high-pressure steam and water (D_2O in particular) instrumentation, collection and sampling and on the fuelling machine systems.

Copper tubing is used in low-pressure oil and hydraulic systems as well as instrument air systems.

Plastics - Various plastic material tubing is used in drain lines, low-pressure oil lines, hydrogen leak collection, wire joining and some more. The selection includes reinforced tygon, tygon, Polypenco, shrinkage PVC, flexible vinyl.

The only material used commonly in other industries but seldom in our plants is Aluminum.

COST

Although a cost of piping is lower than of tubing this disadvantage of tubing can be offset by the ease of erection and a lower possibility of leakage. Each application has to be analyzed and the best solution applied.

JOINTS

Joints may be of two general types - **permanent** and **dismountable** due to the explained differences between piping and tubing it is logical to expect differences between joints used on piping and on tubing. The most common types are explained in the following text.

Piping Joints

A permanent joint used on piping is a **welded joint**. Two ends of piping are prepared and then welded. It is a good joint, very seldom needing any further attention.

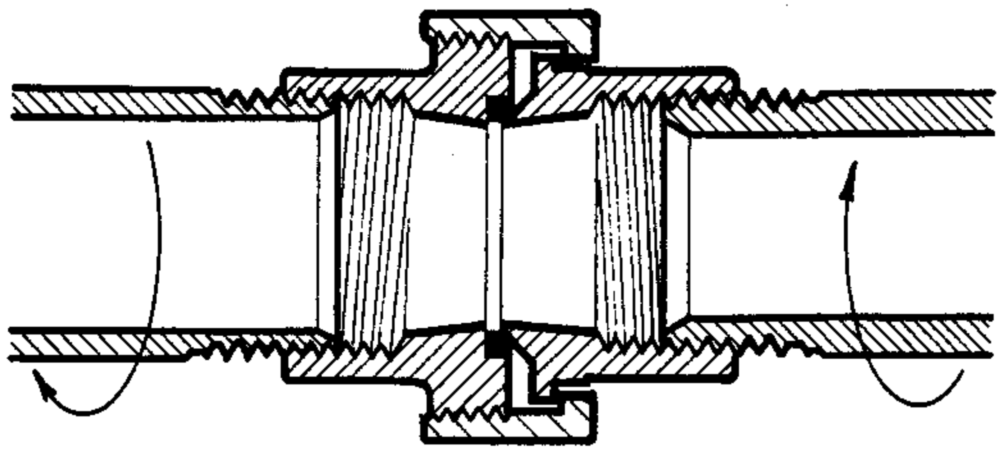
Dismountable joints used on piping are:

Threaded joint, two ends of piping are threaded using a pipe thread and they are held together and sealed by a coupling or a union both having the internal thread. They differ in that unions are made up of three different pieces, the coupling is one piece and pipe lines connected with unions do not have to be removed in order to change the fittings. A coupling and a union are shown in Figure 1.

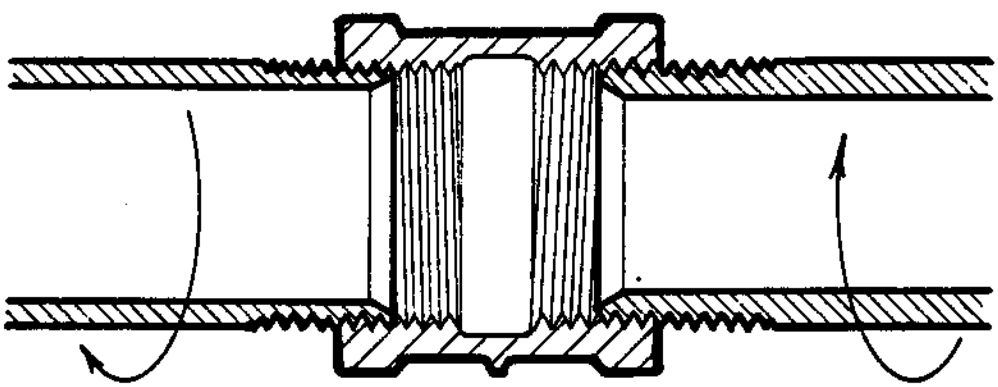
Flanged Joint, consists of two flanges, each attached to the end of to-be-connected pipe, bolts holding these two flanges together and squeezing a gasket, which seals the joint. Various types of flanges used in our plants are shown in Figure 2. The position of a pipe and the welds are shown in dotted lines. The strongest one is a welding neck flange which is used in high temperature, high pressure systems.

Victaulic is a trade name for a joint shown in Figure 3. It can be called a selfenergizing seal, because the process fluid has an access into the gasket (ring) and pushes it against the pipe. The higher the internal pressure the higher the sealing pressure.

This type of joint is mainly used in cold water systems, like service water system, cooling water system, chilled water system and so on where temperature is below approximately 95°C. Victaulic joints using better material gasket can be used up to 175°C.



a) Union



b) Coupling

Figure 1

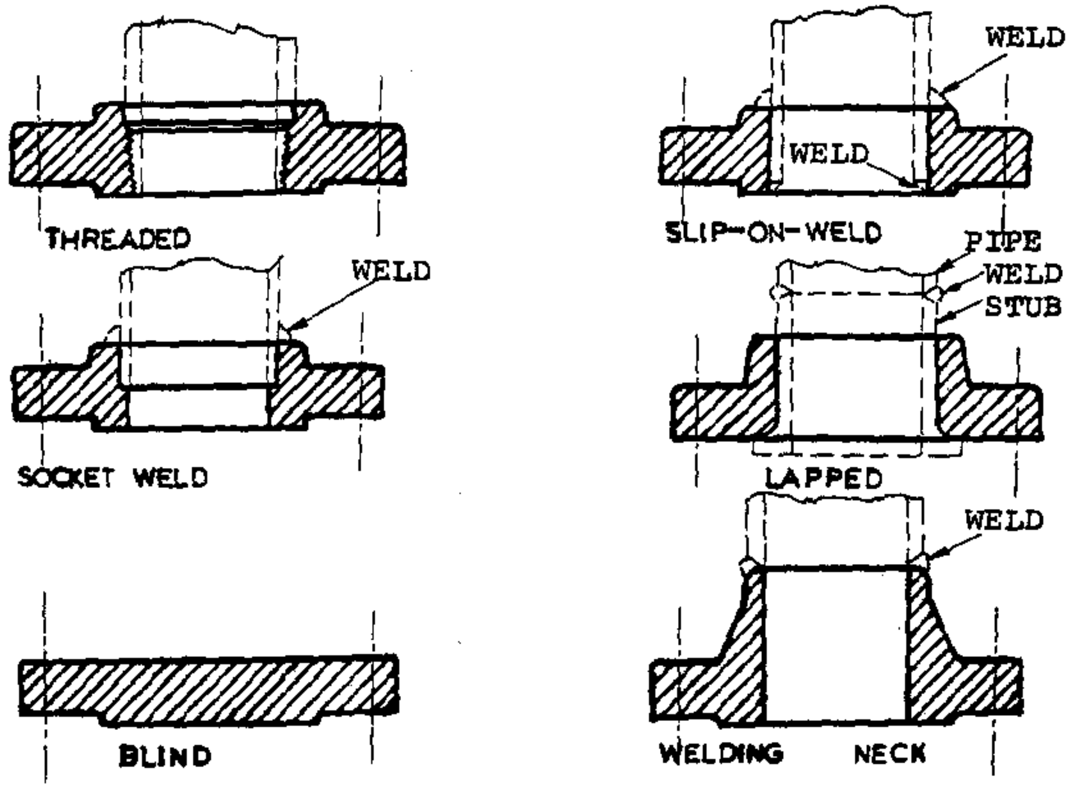

WELD
THREADED
WELD
SLIP-ON-WELD
WELD
PIPE
WELD
STUB
SOCKET WELD
LAPPED
WELD
BLIND
WELDING NECK

Figure 2

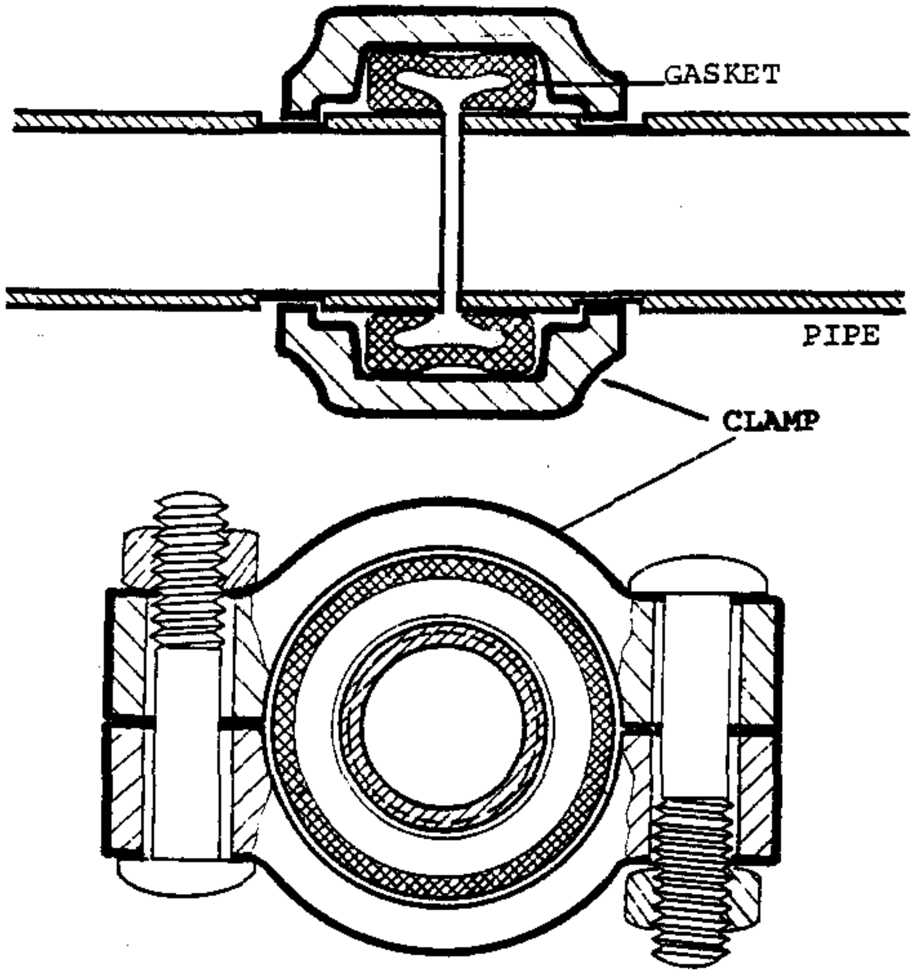
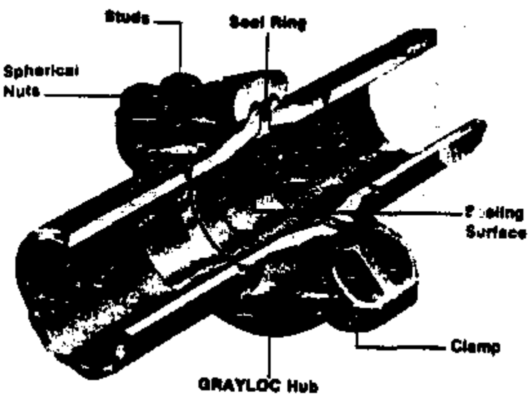


Figure 3

Grayloc is a trade name for a connection used for critical applications demanding "zero" leakage. Grayloc connectors are in service at pressures from hard vacuum to 420 MPa and at temperatures from -220°C to 925°C. In our plants they are used in Heat Transport System, particularly for the attachment of feeder pipes to the end fittings. Figure 4 shows the cut-away and the cross-section through this joint. Two "arms" of T-shaped seal ring are formed by the flexible, tapered sealing lips which during the make-up are forced against the slightly sharper taper of the sealing surfaces in the hub. The amount of their deflection is controlled by coming of the seal ring rib flash with the hub faces. Pressurized fluid inside makes the seal even tighter. This concept makes the grayloc ring a self-energized seal.

Tubing Joints - Permanent joints used on tubing are welded, brazed or adhesively bonded. The last method is not yet used in our plants, other types of permanent joints are not common either. The bulk of tube joints are dismountable.



A standard GRAYLOC assembly consists of two hubs, a metal seal ring, a two or three piece clamp, and bolting.

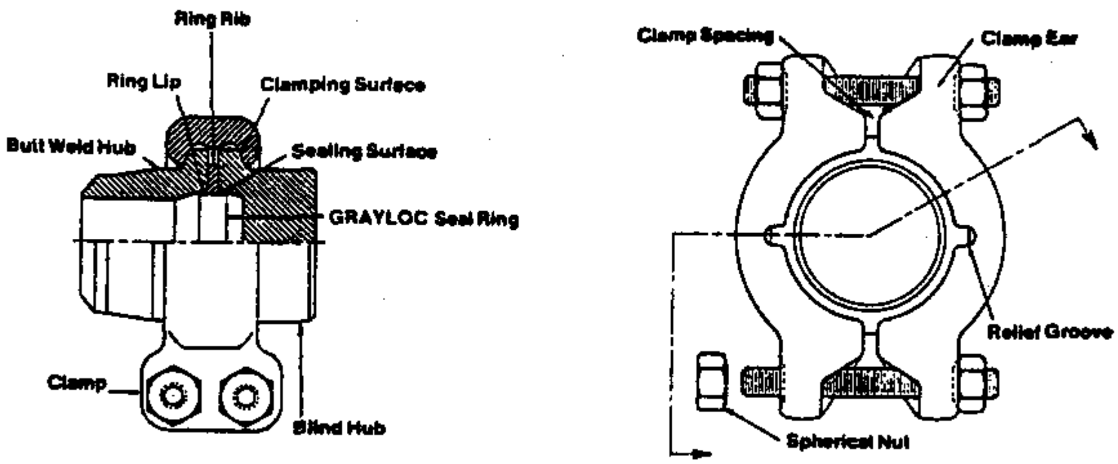
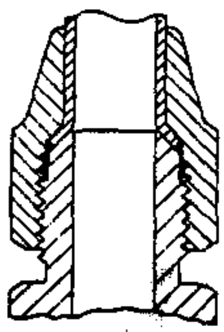


Figure 4

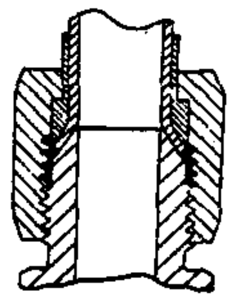
Flare Joint - The end of tubing is flared using a special flaring tool. The flare angle is either 37° or 45°. The flared end of the tubing is then pressed by the nut against the mating surface of the body of the fitting. Several different flared joints are available: a basic two-piece type (Figure 5 (a)) and an improved three-piece type (Figure 5 (b)).

They can be single or double flare as shown in Figure 6.



Two-Piece Type

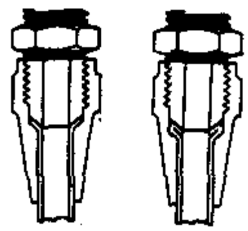
(a)



Three-Piece Type

(b)

Figure 5

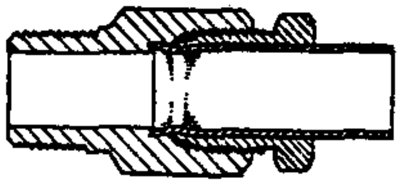


Single Flare Double Flare

Figure 6

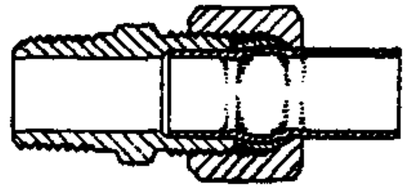
Flare fittings are not used for installation work in our plants but they can be found on purchased equipment.

<u>Compression Fittings</u> are also not common in our plants. Two types are available and shown in Figure 7.



Threaded Sleeve

(a)



Double Compression

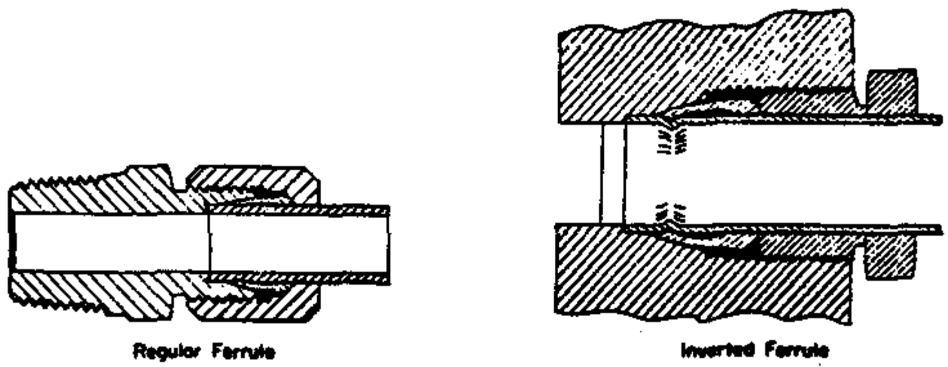
(b)

Figure 7

As the nut is tightened its end (Figure 7 (a)) or end of the sleeve (Figure 7 (b)) squeeze against tubing wall to form a slight compression deflection which produces sealing action. This fitting is limited to thin-walled tubing of soft material and vibrations tend to loosen it.

The almost exclusive tube fitting in our plants, used in the installation work, is a ferrule fitting. The sealing is achieved by forcing the edge of a small ring-ferrule into the tubing wall. This interference seal is leakproof up to high pressures and the fitting resists vibrations.

The simpler version is a single-ferrule fitting which is either of a regular ferrule (Figure 8 (a)) or of an inverted ferrule type (Figure 8 (b)).



(a) (b)

Figure 8

The bulk of tubing joints in our plants is accomplished by using double-ferrule fittings. The fitting is shown in Figure 9.

As the name implies, two ferrules are used resulting in two points of seal which makes the fitting more reliable and suitable for applications demanding "zero" leakage. It is a precision component and must be handled accordingly. Exact mounting procedures are to be followed to get the full benefit of this design. Very seldom the fitting is referred to as a double-ferrule fitting, rather a trade name is used to designate it. Although there are a number of these fittings on the market our nuclear generating stations use the Swagelok fitting and the heavy water plants use the Gyroloc fittings.

All joints discussed so far are **rigid joints** which do not allow any expansion due to temperature changes. But very often this expansion must be accommodated. Tubing systems almost always include bends and this feature along with the malleable nature of tubing materials guarantees absorption of expansion. In piping systems we talk about much longer runs of much larger size conduit and the fabrication of expansion absorbing bends would often be awkward if not impossible. In this case **expansion joints** are used. Figure 10 gives most common expansion joints used today.

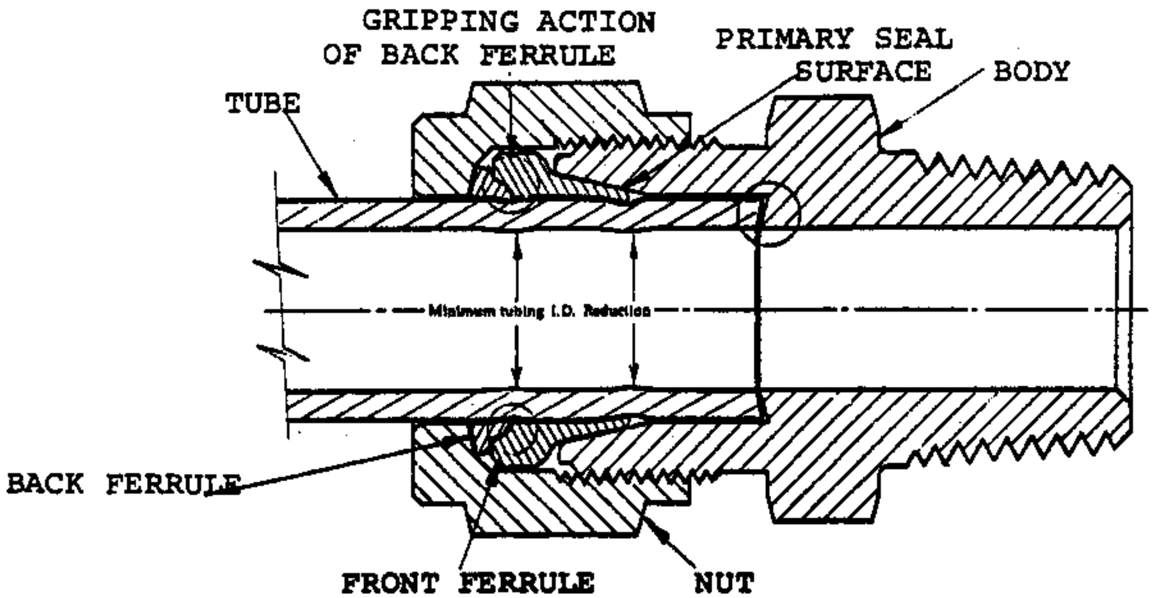
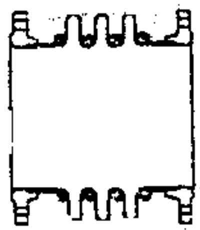
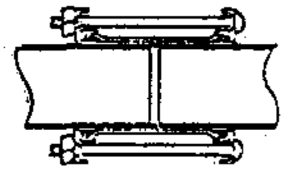


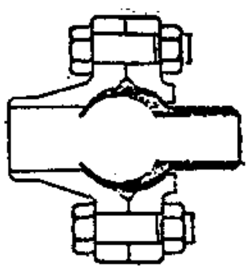
Figure 9



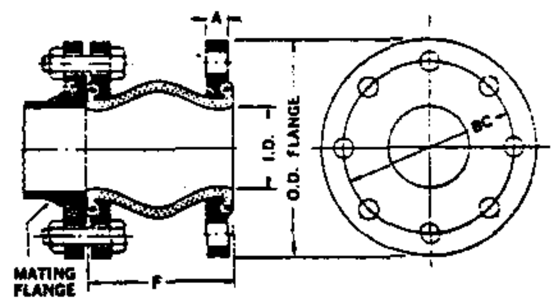
Bellows Expansion Joint can be braced for higher operating pressures.



Slip Type Expansion Joint is self sealing, provides some angular movement.



Ball Joints in series can be used instead of pipe loops for moderate pressure.



Garflex expansion joints feature a molded, elastomeric bellows having a ball-shaped design.

Figure 10

ASSIGNMENTS

1. What are the two types of fluid conduit used in our stations and what is the principal difference between them?

2. How is a pipe specified?
   How is a tube specified?

3. Explain the term "Nominal Size" and Schedule Number".

4. N.S. 4" Schedule 40 carbon steel pipe and N.S. 4" Schedule 40 brass pipe. What will be identical on these pipes?

5. Name the pipe joints and give the preferred ones for high temperatures, high pressure application and for extreme applications where "zero" leakage is demanded.

6. Which of the tube joints is most commonly used in our plants and why?

K. Mika

Mechanical Equipment - Course 430.1

VALVES

INTRODUCTION

A smoothly running nuclear station requires that systems within the station function properly. To reach this point of stability, valves are essential pieces of equipment. Not surprisingly, therefore, valves far out-number any other types of plant equipment with the exception of piping components.

VALVE FUNCTION

For a given system, different modes of operation may be required at different times. It may, for example, be necessary to isolate pieces of equipment, throttle pressure to reduce flow or to relieve excess pressure. One type of valve, unfortunately cannot handle all of these duties. Thus, there is one type of valve for a given function as listed in Table 1.

TABLE 1

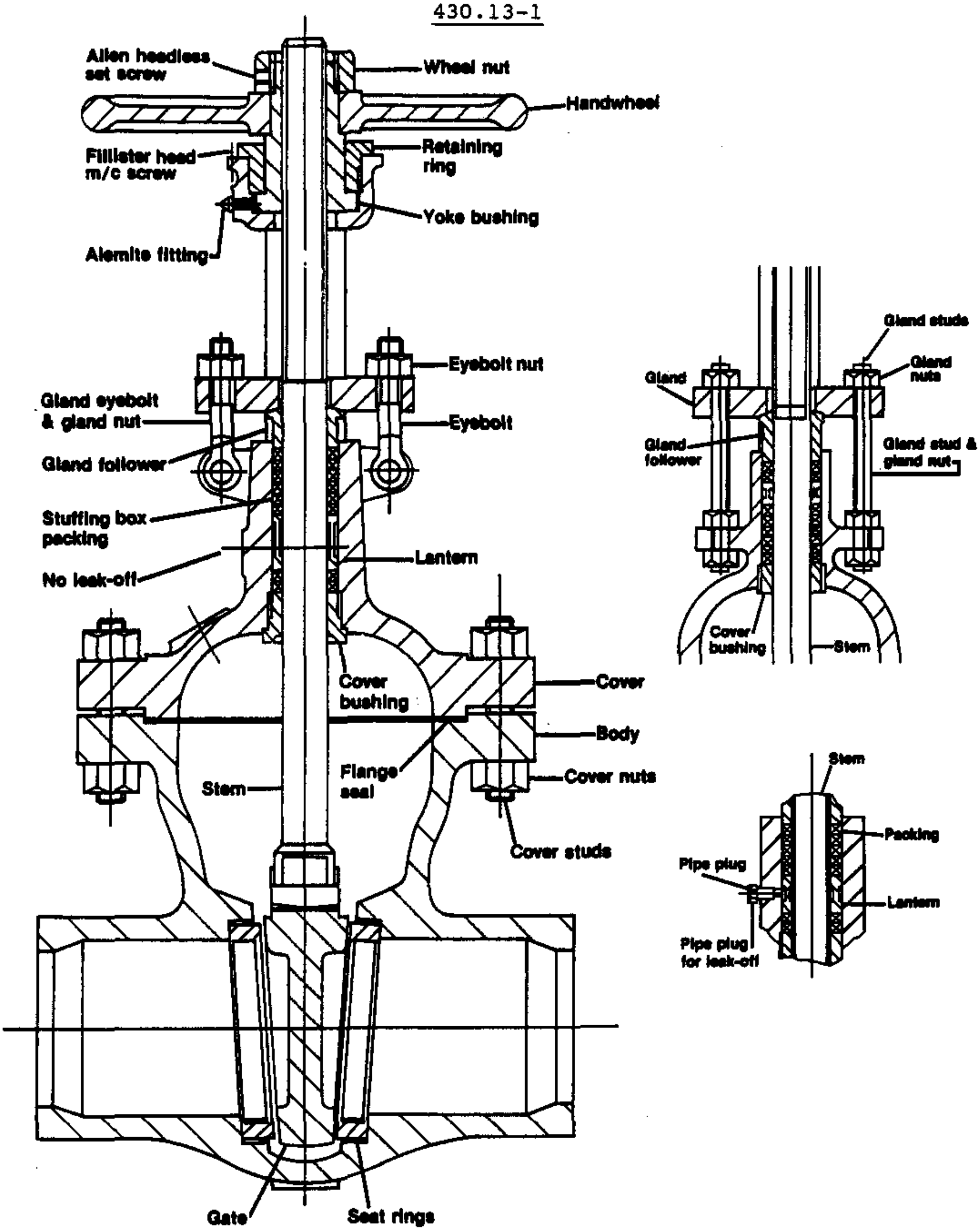
FUNCTION	EXAMPLE OF VALVE USED
1. Isolation	Gate
2. Control or Regulation	Globe
3. Backflow Prevention	Check
4. Pressure Relief	Safety, Relief, Safety, Relief

GATE VALVES

DESCRIPTION

The gate valve is probably the most commonly used type of valve in nuclear power stations. It is intended solely for ON-OFF service (isolation) and not for control. In any position except fully opened or closed, the gate and seat have a tendency to rapidly erode which would prevent tight shut-off.

A typical design of a gate valve appears in Figure 1 showing valve components.



Gate Valve

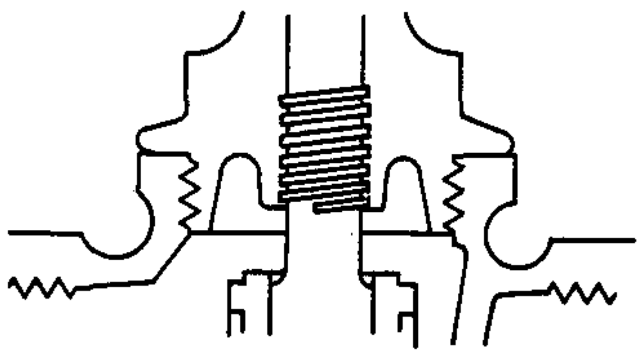
Figure 1

Valve Components

1. Valve Body - part of the valve which contains and regulates fluid flow.

2. Bonnet - is the valve component that gives a closure for the valve body. To gain access to the seat and disc, the bonnet must normally be removed. There are a number of different bonnet types, as indicated below.

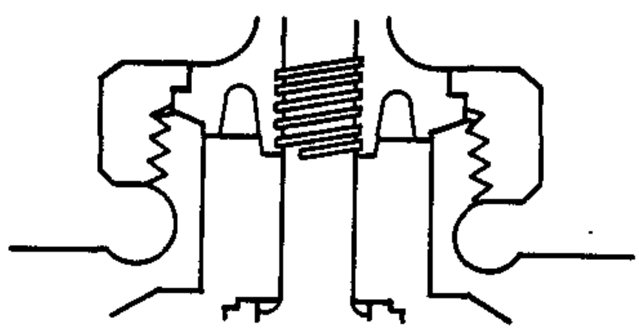
   (a) SCREW - is the simplest type. In this case the bonnet is screwed directly to the body. It is commonly found in smaller valves. See Figure 2.



Screw Bonnet

Figure 2

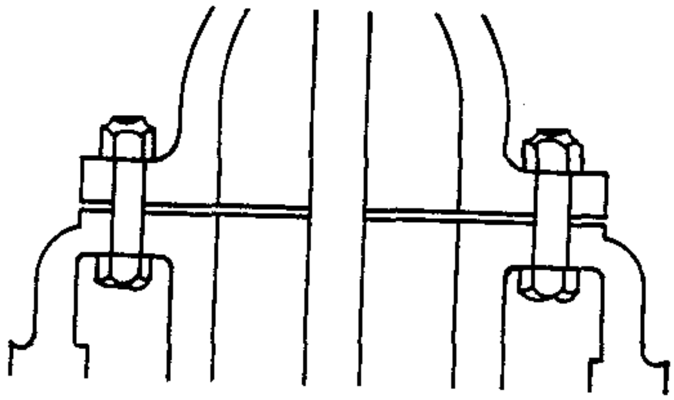
   (b) UNION - connects bonnet to body in a manner similar to a standard pipe union employing a gasket for sealing purposes. See Figure 3.



Union Bonnet

Figure 3

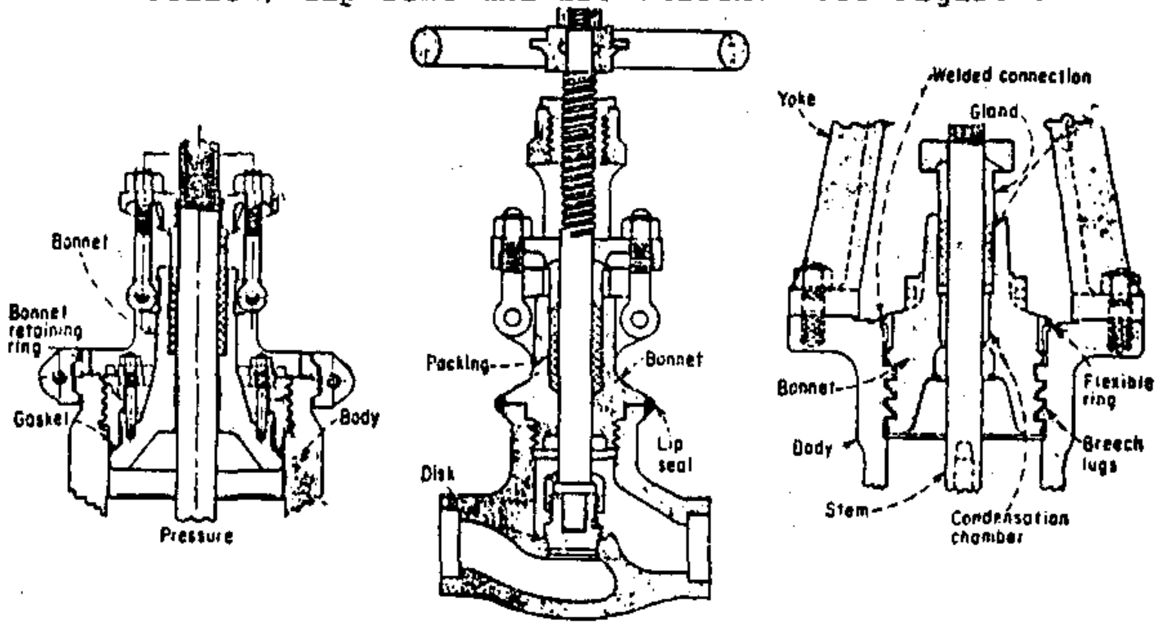
(c) BOLTED (Flanged) - is used for larger valves whenever corrosive solutions and high temperatures, pressures may be encountered. The bonnet flange is tightened to a body flange using a suitable gasket. See Figure 4.



Flanged Bonnet

Figure 4

(d) OTHERS - used for high pressures are the pressure-sealed, lip seal and breechlock. See Figure 5.



Section of an outside-screw and yoke valve showing a pressure-sealed bonnet. Pressure presses bonnet against gasket and retaining ring.

Lip-seal-bonnet globe valve with outside screw and yoke. Bonnet-body threads carry entire mechanical load, weld acting only to seal against leakage.

Section of valve showing the breech-lock joint used for high pressures. The lugs transmit the thrust on the bonnet to the body.

Figure 5

VALVE TRIM (refers to all the wetted parts of a valve excluding the body and bonnet).

3. DISC - is a flat or cylindrical fluid control element which is placed across the fluid pathway to block flow.

4. VALVE SEAT - in order to prevent downstream leakage when the disc is blocking flow there must be a tight fitting closure between the disc and valve seat. There are three types of seals possible.

   (a) metal-to-metal seal - provides the greatest strength but suffers from seizure and galling due to temperature effects and abrasion.

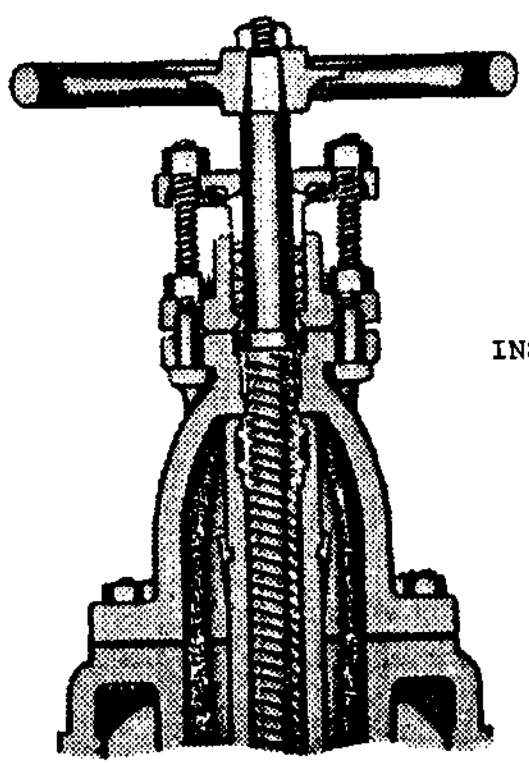
   (b) resilient seal - involves pressing a metal surface against a plastic or rubber one. This type of seal is usually used whenever a tight seal is required for fluids containing solid particles.

   (c) metal-to-metal seal with a resilient insert on one of the surfaces. This type of seal combines features of the other two types.

5. STEM OR SPINDLE - is the part of the valve used to slide the disc across the fluid pathway. The stem usually extends from the disc to the outside of the valve.

   Different stem arrangements are possible such as:

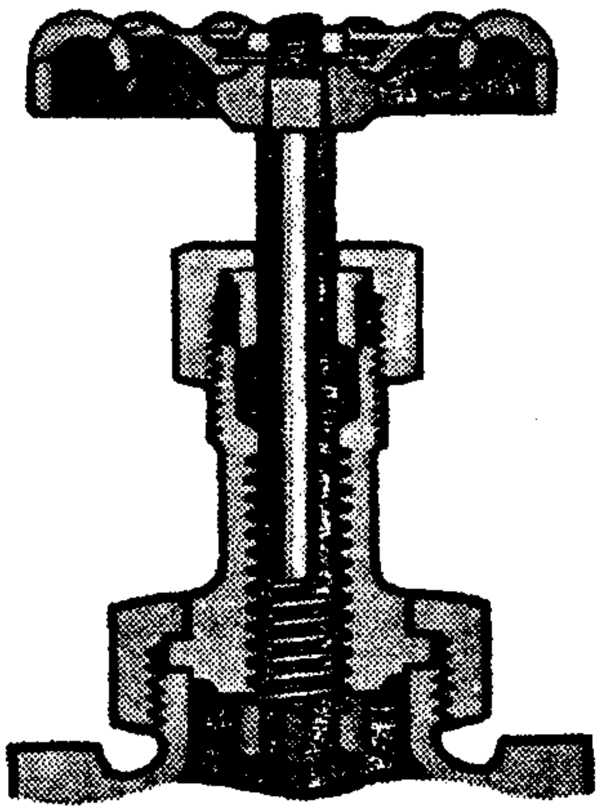
   (a) inside screw non-rising stem - has a disc which rises as stem screws into it. See Figure 6. Since stem screws are held within the body, this is an ideal arrangement especially where headroom is limited.



INSIDE SCREW, NON-RISING STEM

Figure 6

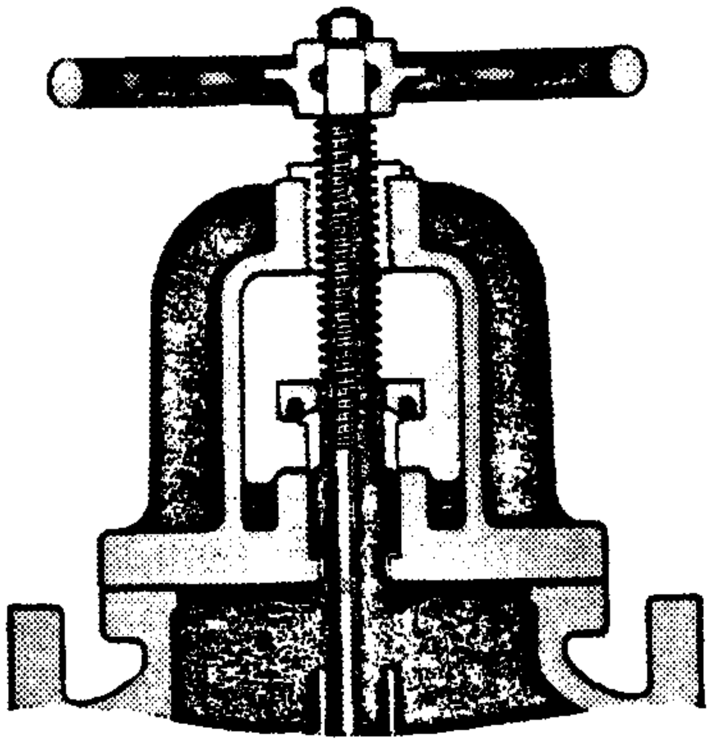
(b) inside screw rising stem - has a disc which lifts as screw turns. See Figure 7. Steam threads are held within the body. From an operator's viewpoint, therefore, disc position is readily indicated.



Inside Screw, Rising Stem

Figure 7

(c) outside screw rising stem - is a threaded stem which moves endwise only. Since the threads to not contact fluid, this type of stem is ideal for corrosive and/or high temperature applications. As in (b) the position of the disc is readily indicated. See Figure 8.



Outside Screw - Rising Stem

Figure 8

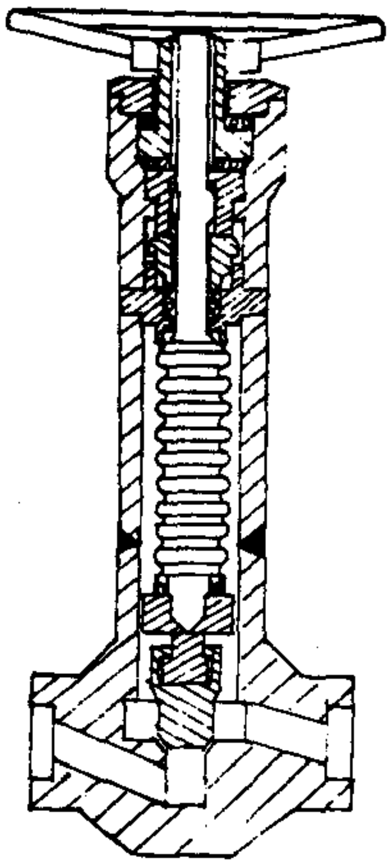
6. STEM SEALING - The most common method of sealing valve stems is to use a stuffing box packed with a flexible type of material.

   To retain the pressure fluids inside the valve, the packing is compressed within the stuffing box and against the stem by a packing nut or gland.

   When there must be absolutely no leakage to the outside, a conventional stem and stuffing box is unsatisfactory. A number of valves that use a packless method of sealing are available, for example:

   (a) diaphragm valve - which will be described later.

   (b) bellows valve which uses a metallic bellows between the body and bonnet. See Figure 9.



Bellows Seal

Figure 9

The last two "packless" sealing methods are particularly useful in preventing any heavy water leakage. Valves employing these methods are sometimes referred to as **zero leakage valves** .

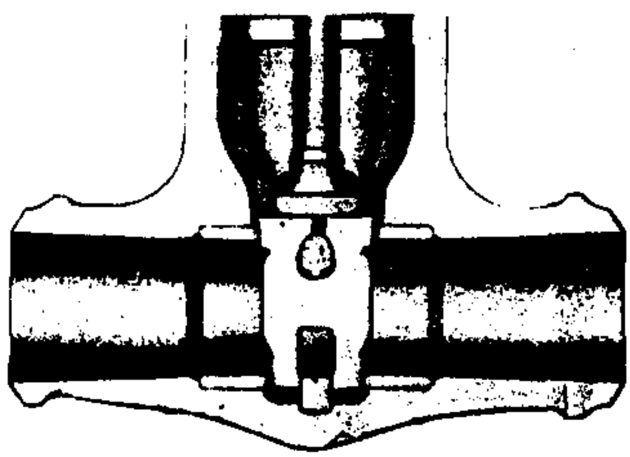
TYPES OF GATE VALVES

When fully opened, gate valves allow straight through flow in a passage that is equivalent to the inside diameter of the associated pipework. Thus, they impose a minimum pressure drop in the fluid flow system.

Gate valves are classified by the type of disc used. The various types of gate valves are listed below:

1. **Solid wedge disc with inclined seat** - This type of shape provides a good metal to metal seal but due to the solid disc it suffers from expansion effects because of high temperature operation. The expansion effects result in either valve seizure or poor disc to seat alignment which may lead to leakage. These valves are, therefore, normally found in cold water systems.

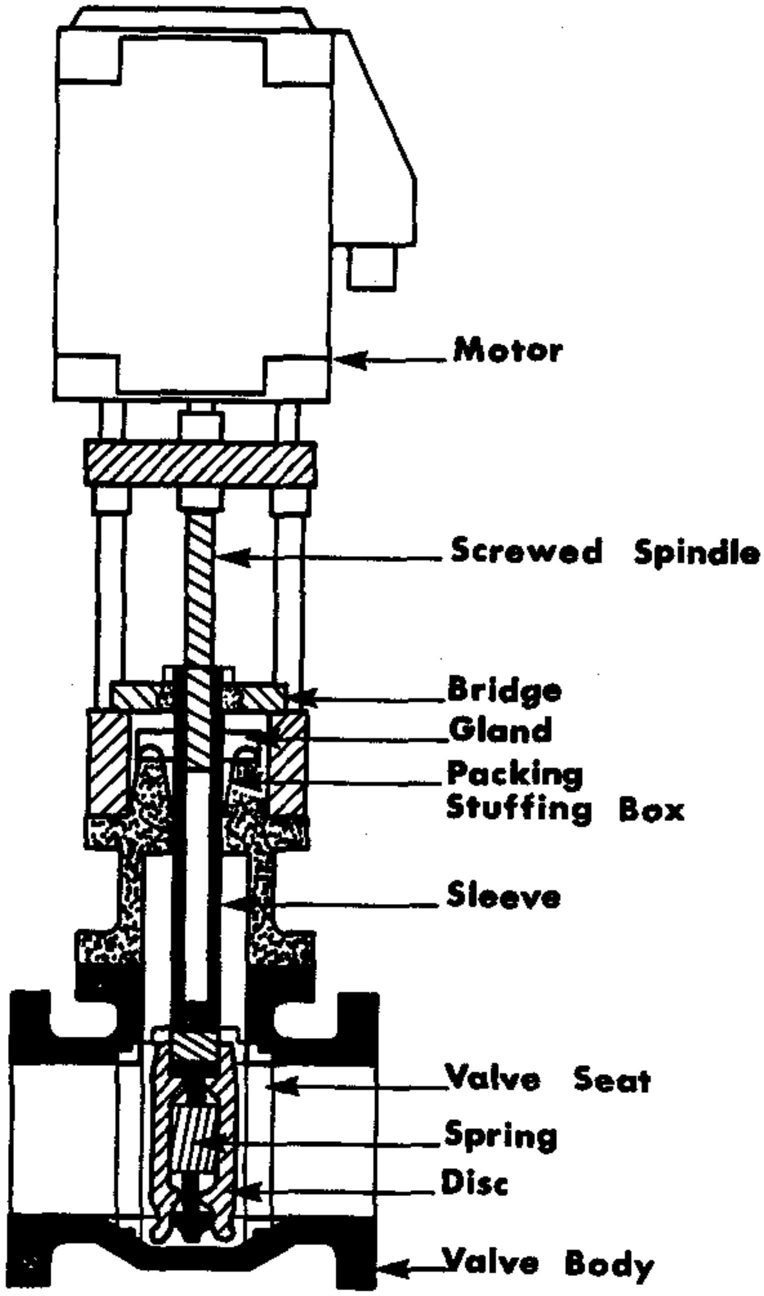
2. **Flexible wedge disc** - Figure 10 is a disc partially cut in halves. This disc type overcomes the temperature expansion problem.



Flexible Wedge

Figure 10

(3) **Parrallel slide disc valve** - Figure 11 - has two discs that are forced apart against parallel seats at the point of closure by a spring. Tight seals result as fluid pressure forces the valve disc against opposing seat.

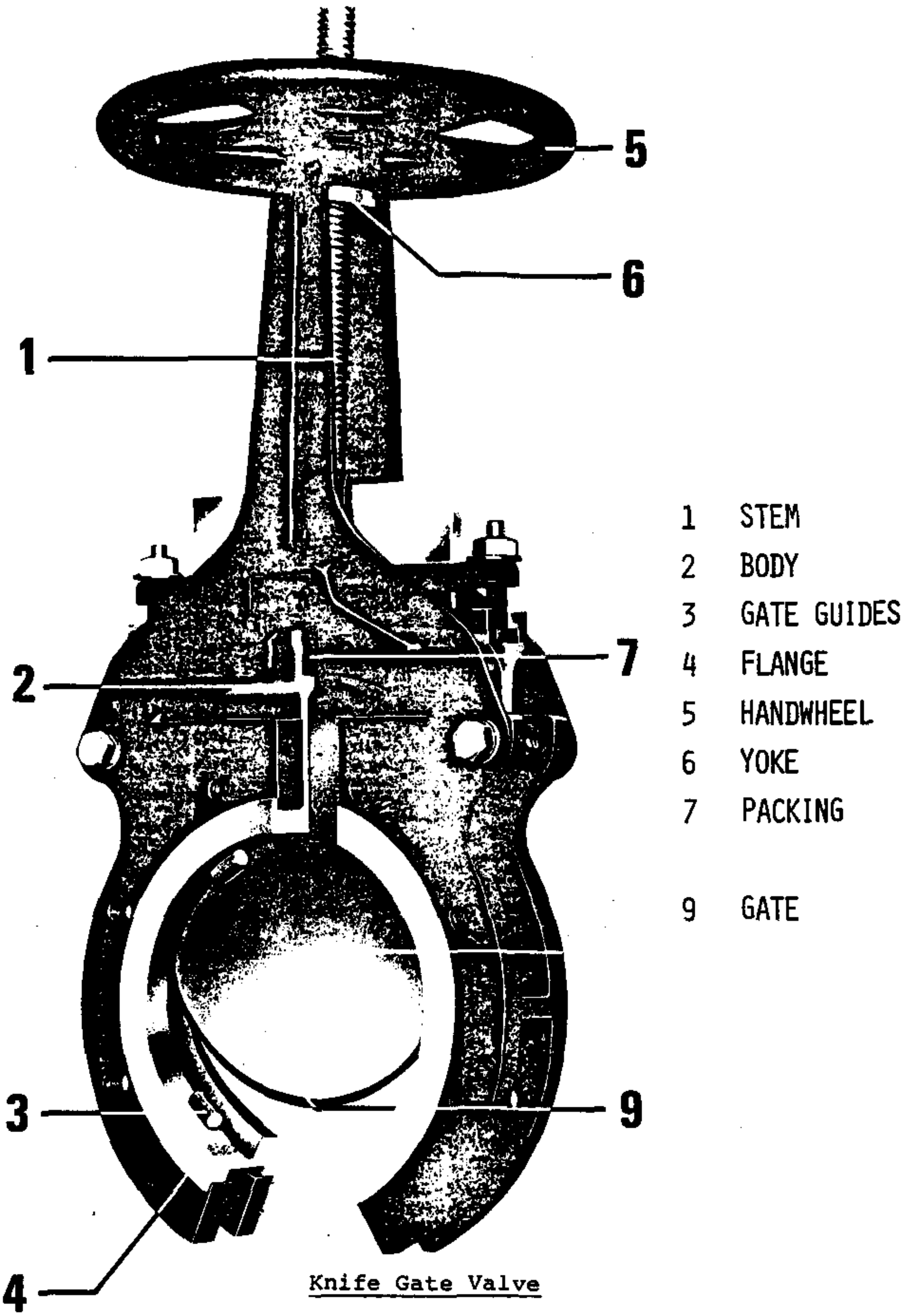


Parallel Slide Valve

Figure 11

(4) **Knife Gate Valves** - Figure 12 - have gates consisting of one or two discs that slide between parallel seats. There is no spreading mechanism: fluid pressure provides effective closure by forcing the downstream surface of the disc against the body seat.

These valves are used in low pressure systems of gases or liquids.

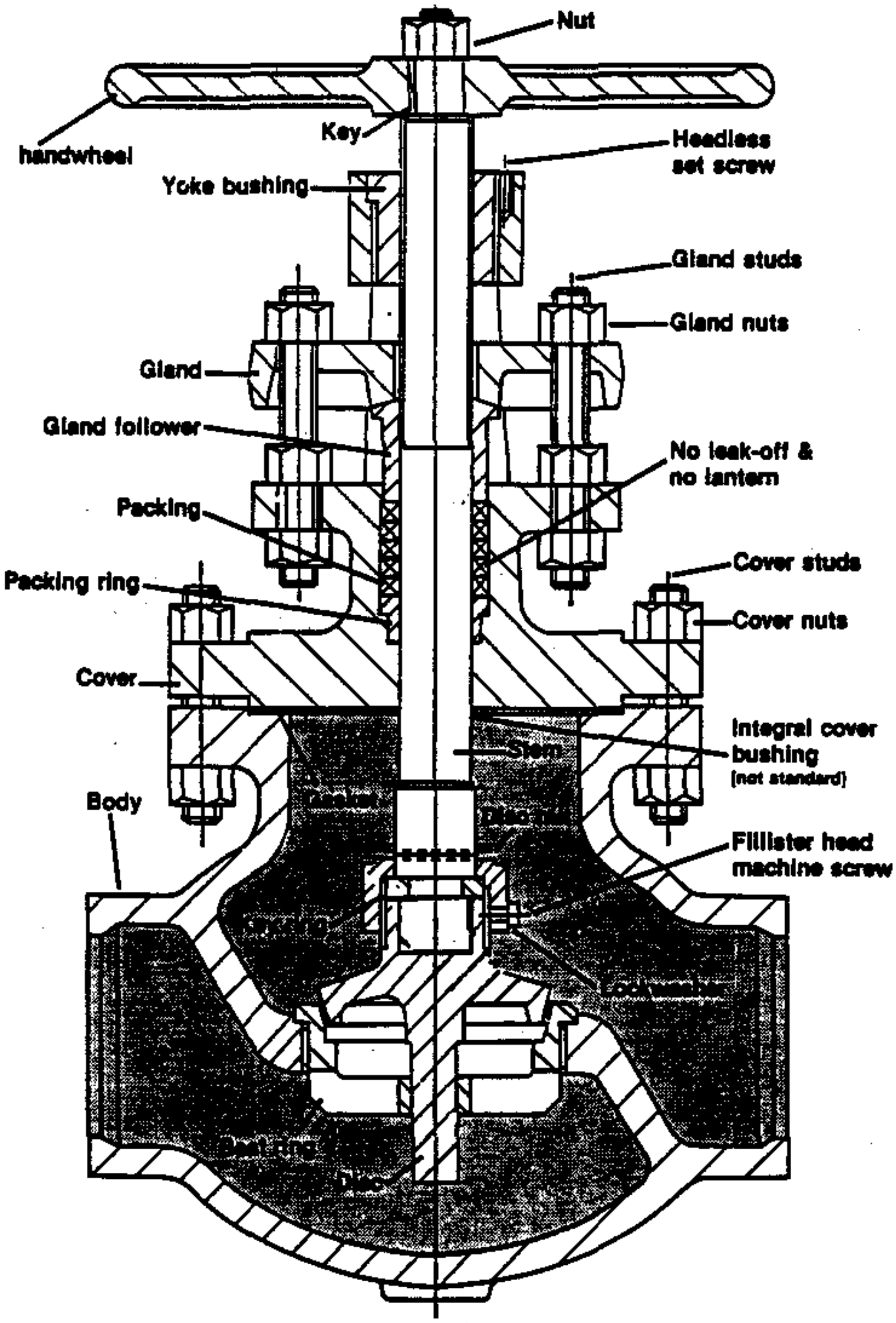


Knife Gate Valve

Figure 12

GLOBE VALVES

As mentioned previously, these valves are designed mainly for flow regulation. A typical globe valve, appearing in Figure 13, has basically the same components as a gate valve but obviously with different arrangements, as mentioned below.



Globe Valve

Figure 13

(1) Flow control for example, is accomplished by a plug or disc that seats on an orifice arranged at at 90° angle to the axis of flow passage. Since flow must make two right angled turns, pressure drop through the valve is much higher than in gate valves.

(2) For liquid service, flow is directed from underneath the plug. An arrow on the body gives correct flow direction. However, with steam, flow is in the reverse of direction of liquid. By directing the steam from above instead of from below, the stem will remain heated and will not contract as much if flow was directed from below. On valve closure, therefore, a tight seal will be maintained.

(3) Lastly, the valve body is globular in shape.

TYPES

(a) **Conventional, single ported valve** is a control valve which regulates flow using one plug.

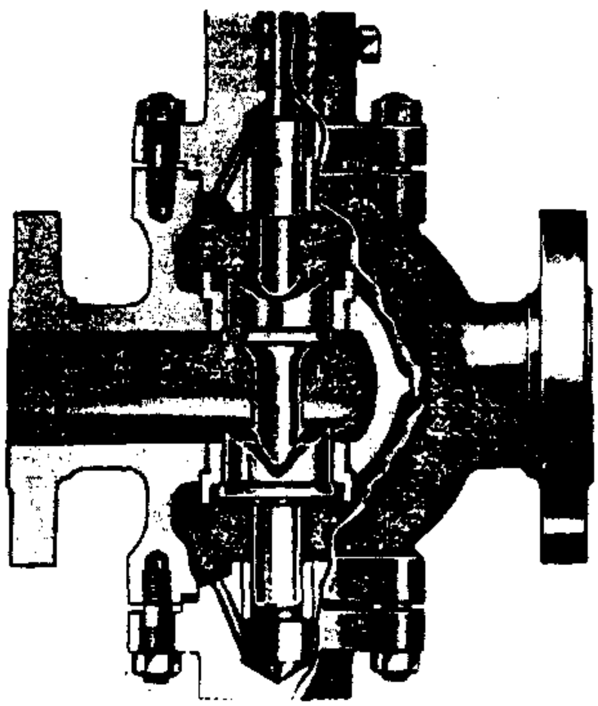
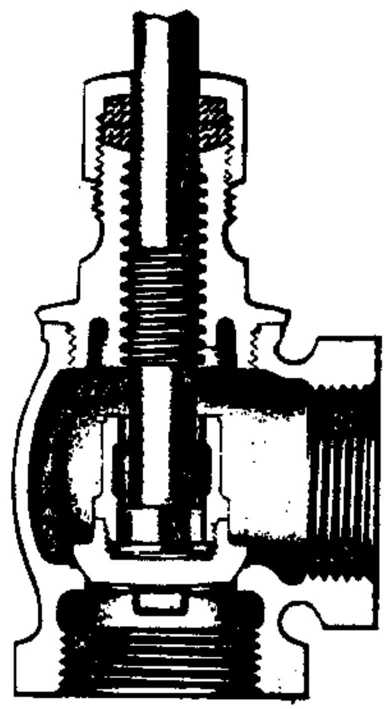


Figure 14

(b) **Double ported globe valve** - Figure 14, controls flow by the opening and closing of two plugs. The double ported or beat valve is used whenever fine control is required, for example the feedwater regulating valves. Fine control requires that axial forces acting on the stem be balanced. (The flow pushes one plug downward, the other upward.)

Unfortunately, this valve is not tight sealing but is preferred over the single ported valve if leakage is not a problem.



Angle Valve

Figure 15

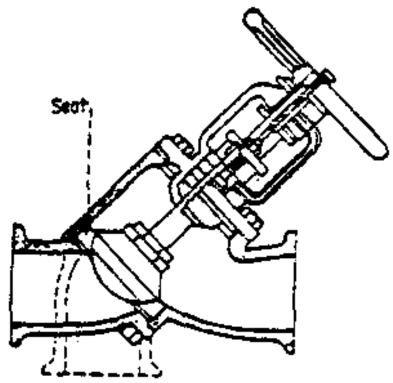
(c) **Angle valve** - Figure 15, is a variation of the basic globe valve design. The angle valve's body has two ends that are at right angles to each other with the axis of the stem in line with one of the ends. Governor steam valves are good examples of angled globe valves.

Two benefits of using this type of valve are:

(1) these valves present less of a pressure drop than globe valves, and

(2) result in a reduction of the number of fittings in a pipe system.

(d) A Y **Valve** - Figure 16 is similar to a glove valve except the orifice is usually at a 45° angle to the flow path. This design gives a low pressure drop across the valve yet with good throttling characteristics.

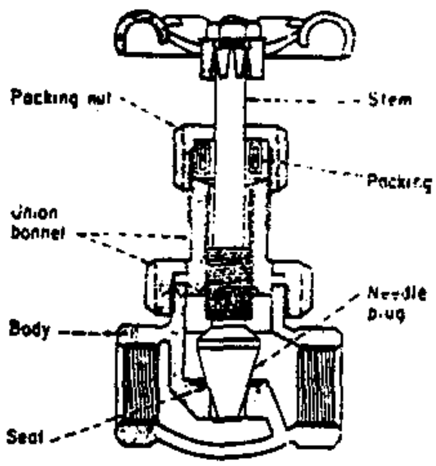


Y valve showing conversion to angle type

Y-VALVE

Figure 16

(e) **Needle valves** - Figure 17, allow close regulation of flow. Generally, a small sized valve, it has a tapered needle-like plug that fits accurately into the seat. Close regulation of flow is accomplished because of fine threading.



Needle valve showing needle-like plug closed

Needle Valve

Figure 17

CHECK VALVES

These valves are designed to prevent the reversal of flow in piping systems. Automatic in operation, opening results because of the pressure of the flowing fluid; closure happens as a result of either back pressure or weight of the check mechanism.

TYPES

(a) **Swing** - has a disk that is hinged at the top. Suited for both horizontal or vertical pipework, there is little pressure drop across the valve. Refer to Figure 18.

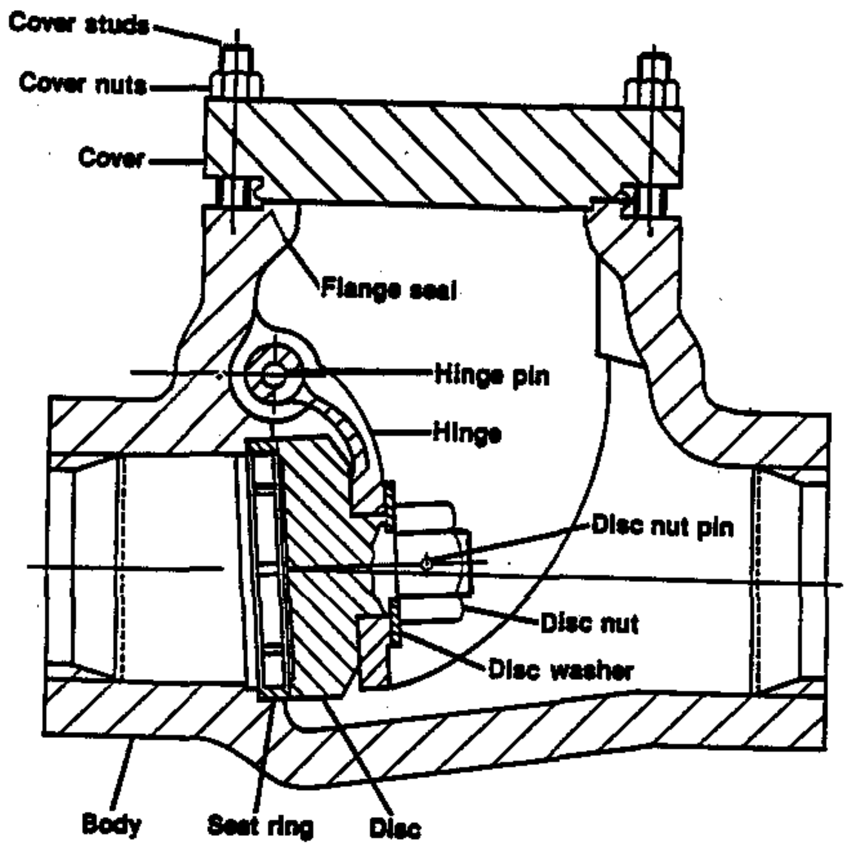
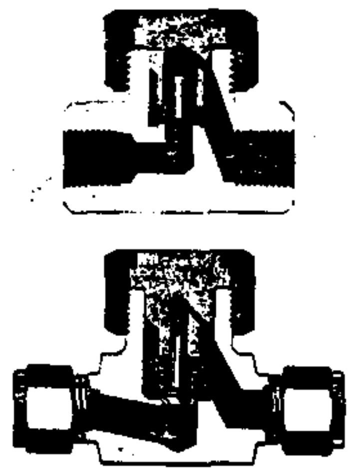


Figure 18

Occasionally swing check valves are equipped with an outside lever and weights to keep the valve from opening until desired pressure is reached. For example, instrumented swing check valves employing a piston actuator attached to an outside lever are found in extraction steam lines to L.P. feedheaters. The instrumental valve ensures quick closure preventing steam returning to the turbine when the turbine must be isolated.

(b) With **lift check valves**, a disk or ball is raised within guides by the pressure of the upward fluid flow. When flow reverses, the check device is forced back onto the seat by backflow and gravity.

(c) The **piston type** is essentially a disk valve with a dash-pot consisting of a piston and cylinder that provides a cushioning effect during operation. See Figure 19. More commonly found in horizontal pipework, these valves are suitable for services which have frequent changes in flow direction. However, higher pressure drops occur than with the swing type.



Lift Check Valve (Piston)

Figure 19

Check valves serve important roles in the functioning of pumps. If check valves appear on the dischargeable side, reverse rotation of the impeller is prevented. On the suction side, check valves, maintain pumps' prime and are normally referred to as **foot valves**.

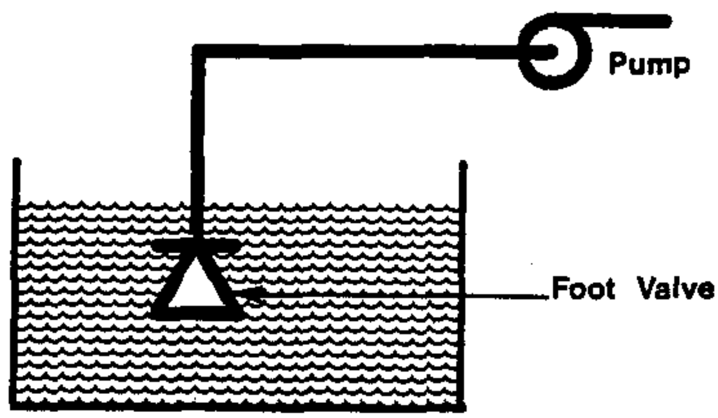
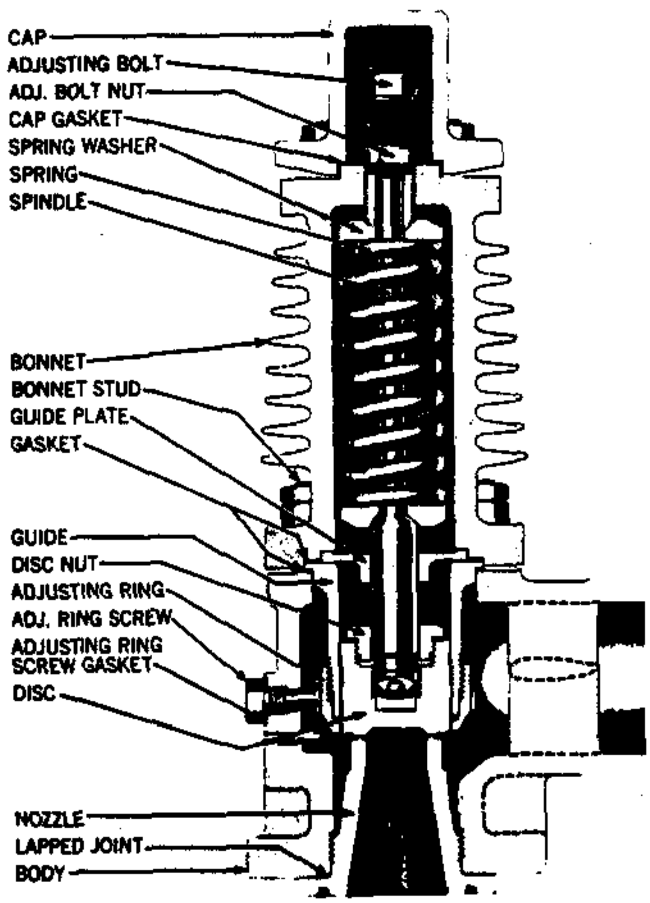


Figure 20

SAFETY AND RELIEF VALVES

Safety and relief valves are used to safeguard pressure systems against operating at dangerously high pressures. Both types automatically discharge fluid to relieve pressure, thus preventing a pre-determined safe pressure from being exceeded.

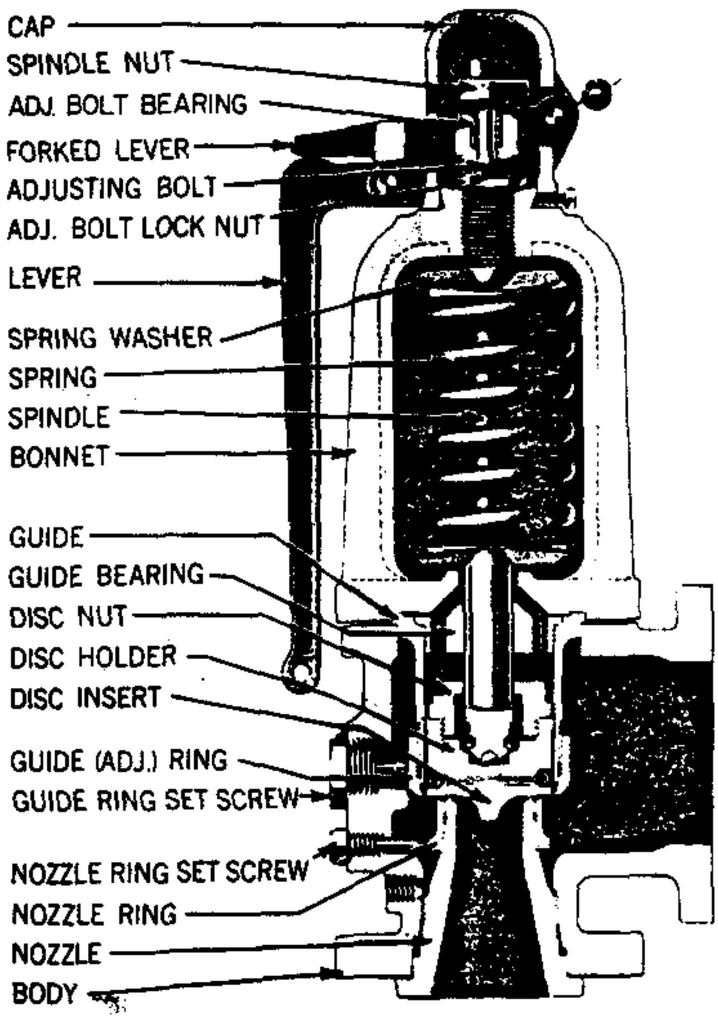
Safety valves are used with gases (or vapours), therefore have full-opening pop action to give immediate relief. Relief valves (Figure 21) are used primarily with noncompressible fluids where a relatively small discharge of liquid provides relief. Safety and relief valves are often designed such that they can be used interchangeably and are appropriately named safety relief valves.



Relief Valve

Figure 21

Both safety and relief valves usually operate by the lifting of a spring loaded disk which permits fluid to pass through. When sufficient pressure acting upward on the disc overcomes the force of the spring the valve opens. In safety valves, the disk over-hangs the seat to offer additional thrust area after the initial opening to produce a faster rise of the disk to the full open position. With relief valves, the area exposed to the over-pressure is constant whether the valve is open or closed, the result being a gradual lifting of the disc to the full open position. A typical safety valve is illustrated in Figure 22.



Safety Valve

Figure 22

Safety relief valves must combine characteristics of both safety and relief valves. For gas service, expansion effects of the gas provides the additional force underneath the disc to achieve popping action for immediate full lift. Liquid service also demands full lift in order to have the nozzle orifice control flow rate. To generate the additional forces underneath the disc, because expansive effects are absent, the liquid's direction of flow is changed 180°. Flow is therefore diverted downward upon contacting the inside of the disc holder skirt. This action adds reactive forces to lift the disc. Figure 23 demonstrates the action of a safety relief valve.

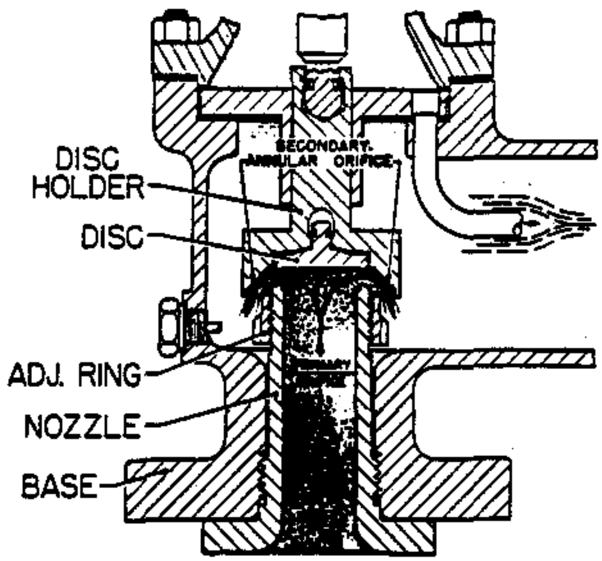


Figure 23

SPECIAL VALVES

BUTTERFLY VALVES

Built on the pipe damper principle, these valves are especially suited for large flows of gases and liquids at relatively low pressures.

Butterfly valves offer a number of advantages, they are:

(1) present low pressure drops to fluid flow.

(2) do not permit sediment build-up.

(3) are easy to install.

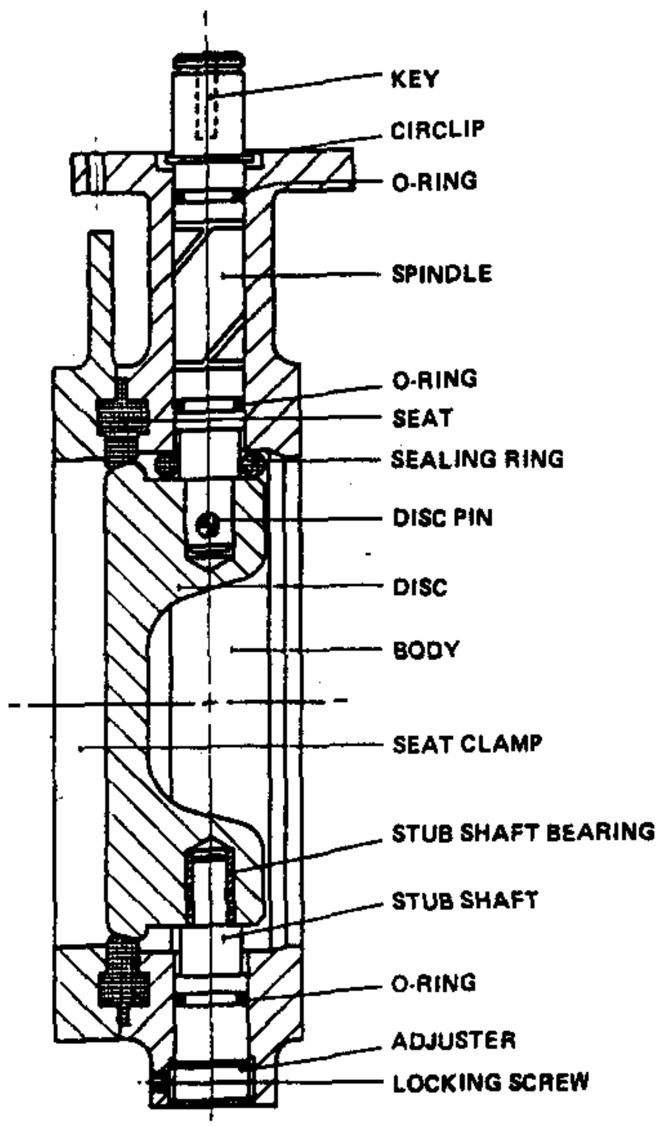
(4) are relatively low priced.

(5) are fast acting, since on quarter turn changes the valve from fully opened to fully closed.

(6) are light for their size compared with gate valves.

(7) can be used for either isolation or control.

The flow control element of this valve is a disc that swings on either a horizontal or vertical axis. In the former case, when the disc lies horizontal, the valve is full open and when the disc approaches the vertical position, the valve is shut. See Figure 24.



Wafer Design Butterfly Valve

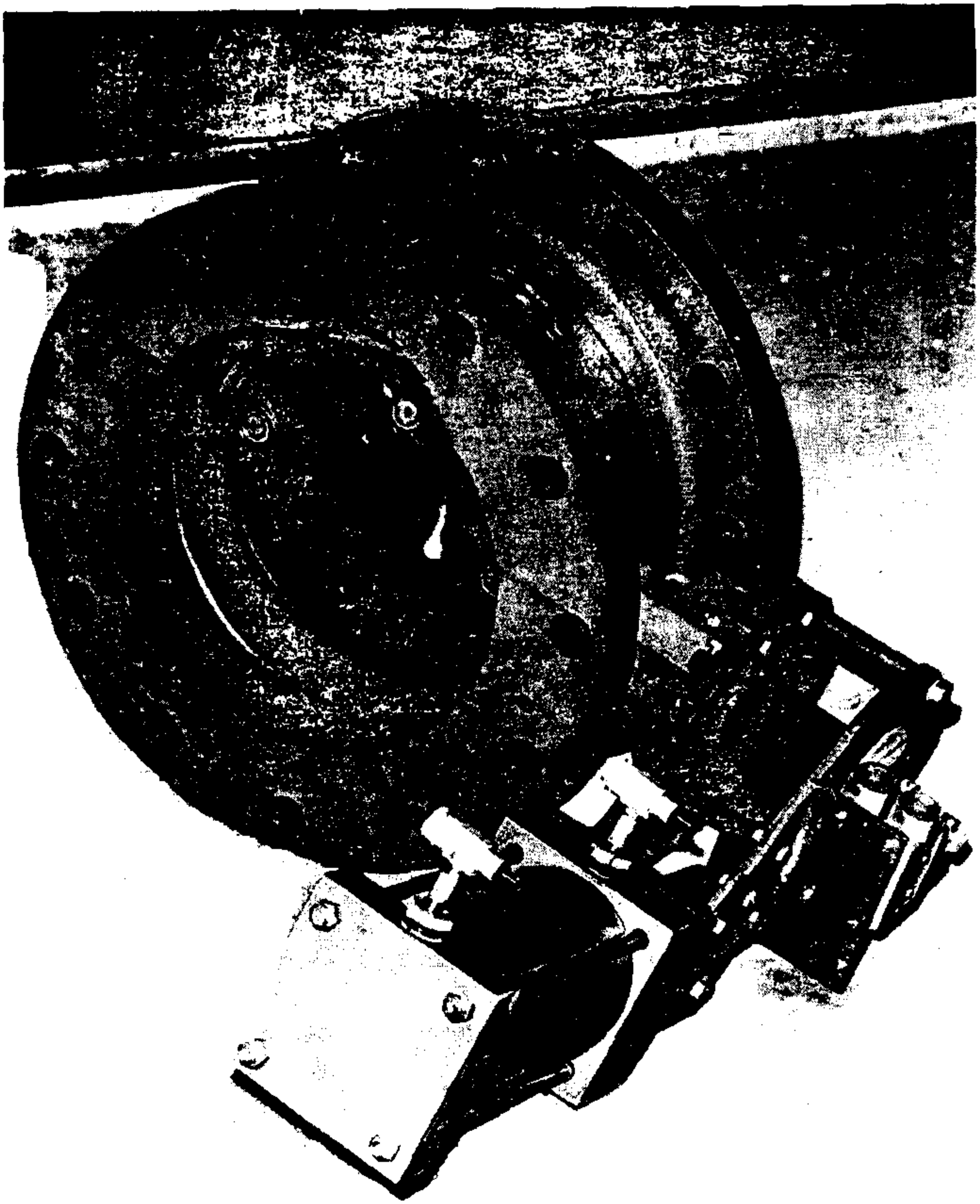
Figure 24

In Figure 24 and Figure 25, the two types of butterfly valves are shown:

(1) wafer design.

(2) flange design.

The wafer design is held in place between two pipe flanges by bolts that join the two flanges and pass through holes in the valve's outer casing. The flange design valve has flanged faces that are joined directly to the pipe flanges.



Flange Butterfly Valve (Piston Actuator)

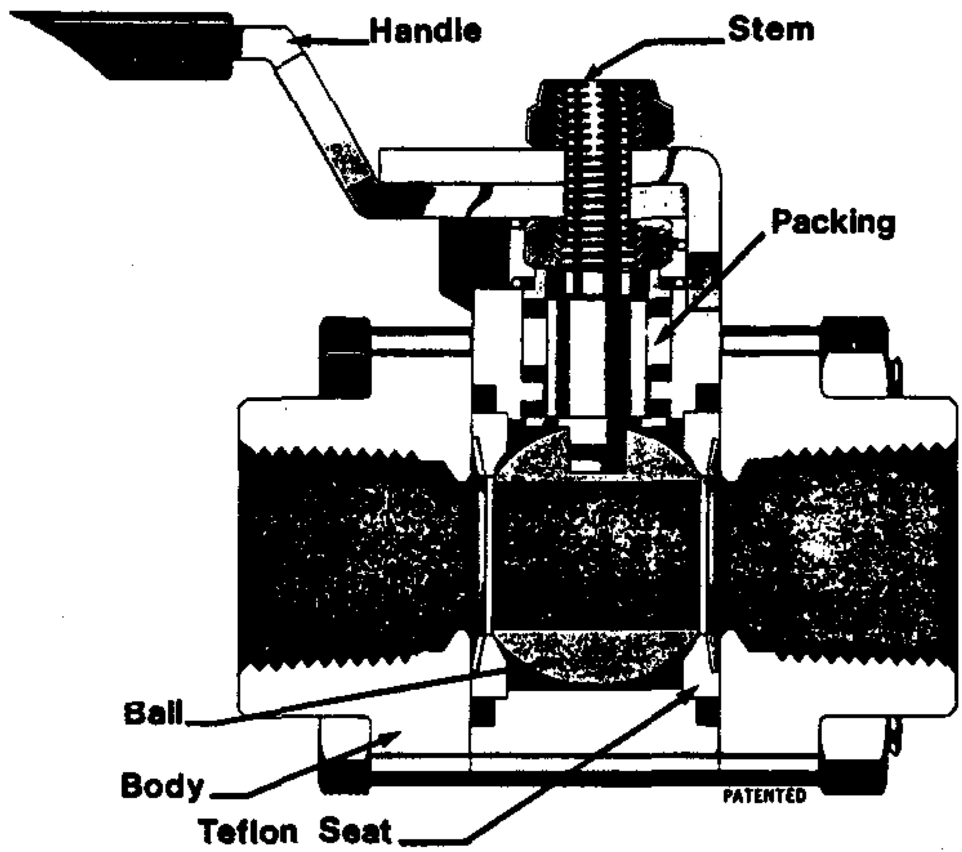
Figure 25

Ordinarily, butterfly valves will not close tightly. Leaks are prevented by using resilient seats or O-ring seats.

In the station, butterfly valves may be found either in an isolation or control function.

BALL VALVES

The ball valve is basically a ball with a hole through one axis that connects the inlet and outlet ports in the body. The ball rotates between resilient seats. In the open position, the flow is straight-through however, turning the ball 90°, completely blocks the passage. Refer to Figure 26.



Ball Valve

Figure 26

In addition to quick, quarter-turn, on-off operation, ball valves are compact, easy to maintain, require no lubrication and give tight sealing with low torque. Goodsealing results because fluid pressure forces with ball against the valve seat. Ball valves can be found for either isolation or control applications.

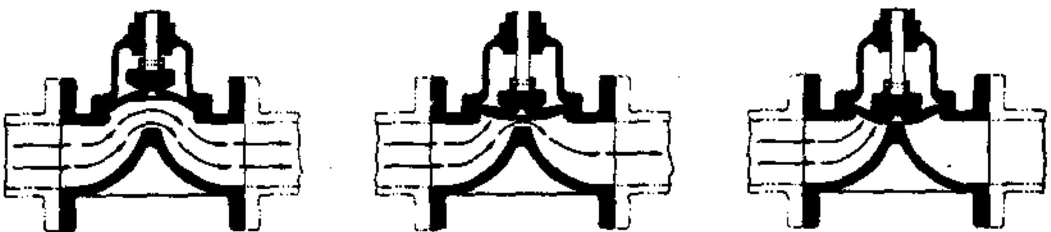
DIAPHRAGM VALVES

As mentioned previously, the diaphragm valve eliminates stem packing by using a flexible diaphragm to isolate the operating mechanisms from the fluid being handled. It consists basically of a body, bonnet, and flexible diaphragm.

DIAPHRAGM CONTROL VALVES

INTRODUCTION: The Conoflow Series HB diaphragm valve maintains a streamlined flow of many hard-to-handle fluids in a leakproof closure. These fluids include corrosive and erosive liquids, slurries, semi-solids, viscous substances, gases, etc. Numerous applications exist in less difficult services where simple packless construction and easy maintenance are desirable. This valve was formerly referred to as a "Saunders Patent-type" after P. K. Saunders, developer of the basic design.

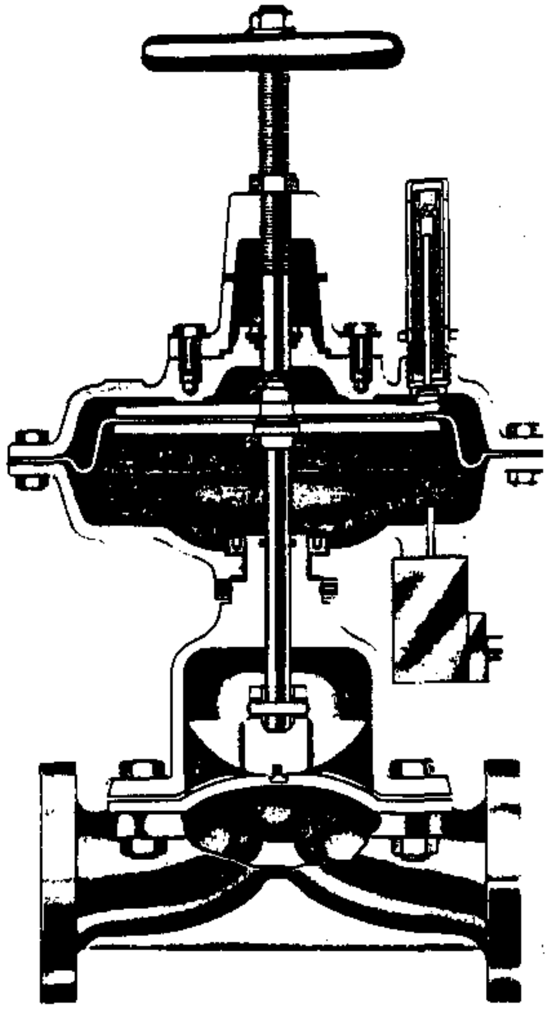
OPERATION: Principle of the Conoflow Series HB diaphragm valve is extremely simple. A resilient, flexible diaphragm is connected to a compressor by a stud molded into the diaphragm. The compressor is moved up and down by the valve stem. Thus, when the compressor is raised, the diaphragm is lifted out of the fluid path to allow streamlined flow in either direction, and is pressed against the body weir when the compressor is lowered. The diaphragm can also be placed in any intermediate position for throttling control.



Open Position **Throttling Position** **Closed Position**

Diaphragm Valves

Figure 27(a)



Diaphragm Valves

Figure 27(b)

When the valve is opened, the diaphragm is lifted out of the flow passage to allow smooth streamlined flow in either direction. In the closed position, the diaphragm is tightly seated against a weir or contoured area at the bottom of the valve. It may also be positioned at intermediate points in the fluid passage for throttling the flow.

The diaphragm valve is excellent for handling various substances, slurries or corrosive fluids.

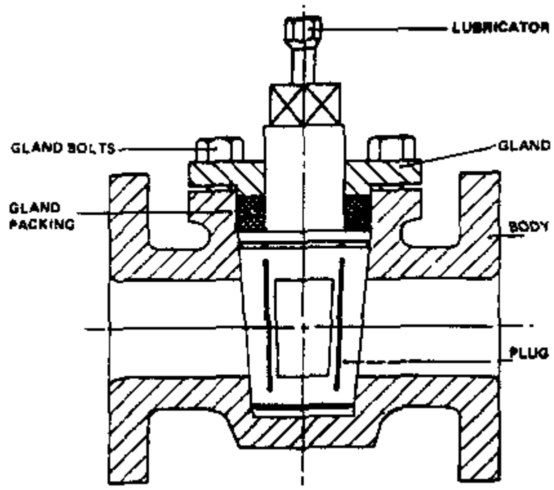
PLUG VALVES

The plug valve is one of the oldest members of the valve family. Like the gate valve it is used for on-off service.

The basic components of the plug valve are the body, plug and cover. The plug which may be either tapered or cylindrical, has an orifice which the fluid passes through. See Figure 28. In operation, the plug may be turned through 90° to allow fluid passage or to present a blank fuse to prevent flow.

These valves because of their design offer advantages of quick action, minimum installation space, simple operation, and low pressure drops.

The two basic types to be considered are the (1) non-lubricated type which incorporate passages or grooves in which lubricant/sealant can be applied under pressure. This serves to lift the plug for ease of operation.



PLUG VALVE

Figure 28

ASSIGNMENT

1. Name the four functions that valves must provide and an example of the valve type used for each function.

2. What are the consequences of using a gate valve for throttling?

3. Draw and label a typical gate and globe valve. Indicate on the sketch the direction of flow.

4. What type of gate valve would you recommend for:

   (a) cold water service?

   (b) high pressure, high temperature steam?

   Why?

5. What type of globe valve would you recommend for:

   (a) fine control for large flows of water - regulation?

   (b) governor steam valve?

   (c) fine control for low flowrate systems?

6. (a) Name the two types of nonreturn valves used and compare them with respect to leakage and pressure drop across them.

   (b) What type of valve would you use for a small pressure drop and vertical pipework use?

   (c) The main boiler feed pumps must not undergo reverse rotation. Would you, therefore, put the valve insuction or discharge pipework?

7. What type of pressure relief valve would be used to depressurize the steam generators? plunger chemical injection pump overpressure relief?

8. (a) Diaphragm valves can be used for what functions?

   (b) What is the main advantage of a diaphragm valve?

   (c) Where would you recommend use of diaphragm valves?

9. A large (approximately 2 metre diameter) butterfly valve is used on the discharge condenser circulating water line.

   (a) What are the other advantages of using butterfly valves?

   (b) What function do you think these valves perform?

Mechanical Equipment - Course 430.1

LUBRICATION

There are many reasons for lubrication of metal surfaces such as control of friction, reduced wear, reduced errosion, limiting temperature, cleaning, dampening shock, or forming a seal. Most of these requirements are interrelated, for example, a lubricant doing a relatively poor job of controlling friction must be able to assume the added burden of removing more heat. This additional heat may thin out the lubricant which results in a poor lubricant for the job at hand.

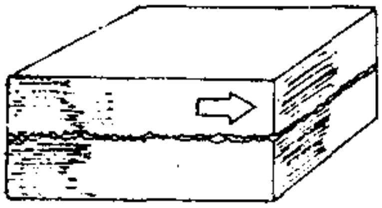
Of all the reasons for lubrication the two major ones are firstly to reduce friction and wear between surfaces and secondly to carry away heat generated by friction resulting from the metal surfaces or internally from the lube oil itself.

To understand the need for lubrication we should first look at friction and what causes it. Basically friction is the force that resists sliding motion. The term coefficient of Friction relates this friction force to the load, ie, the friction force divided by load. The thing which causes friction is the fact that surfaces are made up of peaks and depressions irregardless of how smooth they appear. When two such surfaces come together, these peaks and depressions interfere and cause resistance to slipping. Even rollers or balls suffer from this problem. When rollers or balls are placed between flat surfaces the materials deform and rolling elements slip under load giving rise to sliding friction.

We can divide the problem of friction into three main types Figure 1, sliding friction, rolling friction and fluid friction. The first is as we have mentioned, two metal surfaces being moved across each other. Examples here are a shaft turning in a bearing or pistons in a cylinder.

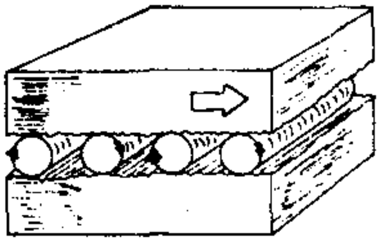
The second of these is rolling friction which, as we have said, is due to the deformation of the roller or ball material and the flat surfaces, and the tendency for rolling elements to slip under load.

The third type is fluid friction which is the resistance to movement within a fluid of one molecule relative to another molecule.



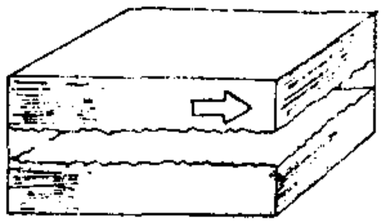
Sliding

This is most basic type of friction-one solid body pulled or pushed across the surface of another with adhesion, shearing and plowing coming directly into play. Examples are piston moving in a cylinder or shaft revolving in a bearing with no lubricant separating the surfaces.



Rolling

Any system of rolling elements reduces friction considerably. If balls or rollers and flat surfaces were smooth and inelastic, friction would be almost zero. But materials deform, rolling elements slip under load. However starting and running friction are about the same.



Fluid

When a film of liquid lubricant separates the surfaces, the only friction is from the motion within the fluid. The fluid splits into "layers". The top layer sticks to the top surface, the bottom layer to the lower surface. Each successive layer travels at lower speed, shearing the layers on either side.

Types of Friction

Figure 1

In order to reduce sliding friction we must attempt to keep the two surfaces apart. This can be done by rollers giving rise to rolling friction or by introducing a fluid which gives rise to fluid friction. If we use rollers of some sort we still get some sliding friction and the addition of a lubricant will reduce this. In these two cases we get back to fluid friction to some degree.

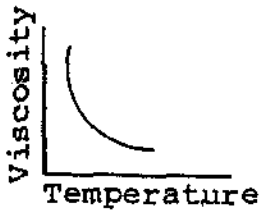
Taking a look at the properties of a lubricant which must exist for it to fulfil these functions, we find four major properties-viscosity, oiliness, flash point and temperature stability.

Oiliness is the ability of a lube-oil to cling to or be absorbed to the surface of a material. It is this characteristic which is used to aid in overcoming boundary friction. It is normally achieved today by means of additives to the oil which improve their natural adsorptive qualities.

Flash point of a lube-oil is the temperature at which vapour will be given off in enough quantities and ignite. This value should be high so that the oil does not break down during operation.

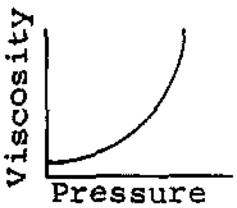
Temperature stability is the ability of a lube-oil to maintain its load carrying capacity over a range of temperature variation under which it is likely to be used.

Viscosity is by far the most important single factor when dealing with lube-oils. It can determine the friction loss, heat generation, load carrying capacity, film thickness, ability to flow and in many cases wear. The simplest definition of viscosity is the oil's internal resistance to motion. Oil with a high viscosity won't flow as easily as an oil with a low viscosity. Some of the characteristics of oil viscosity are shown in Figure 2. It is obvious that a combination of all these factors must be considered when selecting lubrication for any one problem. The use of lube-oil additives increases the scope of lube-oils for any given condition.



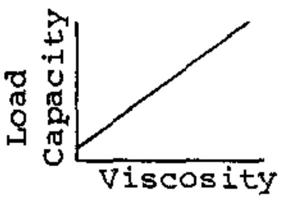
Temperature

Increasing temperature lowers oil viscosity. A high-viscosity oil can support a heavy load, especially at low temperatures. High-viscosity oils also have more internal friction.



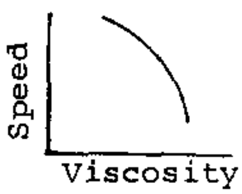
Pressure

Increasing pressure increases oil viscosity. However, this only becomes important when pressures are in the neighborhood of several thousand psi.



Load Capacity

Viscosity of an oil must be matched to the application. The oil must have enough viscosity to handle the load, yet increasing the viscosity causes an increase in fluid friction, which heats the oil and lowers the viscosity.



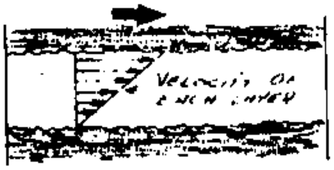
Shaft Speed

High speed means faster shearing of oil layers, and more fluid friction. As temperature goes up, viscosity goes down to decrease load capacity. However, a high speed helps to form a hydrodynamic wedge in bearings.

Oil Viscosity Characteristics

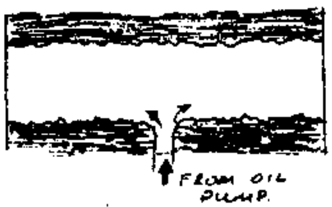
Figure 2

We have mentioned the three types of friction and we attempt to reduce them to fluid friction by means of lubricating materials. Now we should mention the types of lubrication. These are boundary or thin film, hydrodynamic and hydrostatic. (See Figure 3 on following page.)



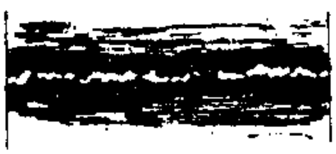
Hydrodynamic

Full film of oil, exaggerated above, is normally about 0.01 to 0.001 in. for 1-in. bearing-or lower.



Hydrostatic

Oil pressure from outside pump keeps surfaces apart. For slow, heavy loads or to cut down starting friction.



Thin-Film

Many bearings will operate with only 0.0001 in. or less oil film between surfaces; some metal-to-metal contact.

Types of Lubrication

Figure 3

Boundary lubrication is a very thin film of oil which adheres to the surface of the metals. There will be some metal to metal contact depending upon the oiliness characteristics of the lubricant.

Hydrodynamic lubrication is achieved by a relatively thick film of oil. Pressure, which is the key to separating surfaces, is built up within the bearing. This is accomplished either by temperature rise or by an oil wedge which will be explained later.

Hydrostatic lubrication is achieved by supplying a lubricant to a bearing area. In other words the pressure separating the two surfaces comes from an external source and physically lifts the two surfaces apart.

So far our discussion has dealt with fluid lubricants. However lubricants can be broadly classified as gas, liquid, semi-solid or solid and they may be grouped roughly into three general types - fluids, greases and solid-film lubricants.

The term oil covers a broad class of fluid lubricants and some of the general types are as follows:

1. Mineral oils - produced from crude or petrolium oil distillation. They are still the largest single type in general use.

2. Fixed Oils - produced from animals and plants. These are not generally used alone but are combined with mineral oils usually serving as oiliness agents.

3. Synthetic oils - these are man-made lubricants which have a wider range than petrolium oils. They can be carefully designed and additives can be more easily tailored for them.

Greases are essentially a mixture of a lubricating oil and a metallic soap which keeps the oil in suspension. The most common soaps in use are calcium, sodium, and lithium. Rather than using viscosity for the classification of greases the term consistency is used which is a measure of how easily the grease may be squeezed out from between the two parts being lubricated. Like oils greases may have special additives to enhance their lubricating qualities.

A solid lubricant is simply a solid material placed between two moving surfaces to prevent metal-to-metal contact. Therefore the application of solid lubricants is generally in the boundary area. They may be applied as dry powders, mixtures with grease and oil or mixtures with binders which form dry films when cured. Some of the solid lubricants found in use to-day are graphite powder, molybdenum disulfide, tungsten disulfide, teflon powder and other plastics.

ASSIGNMENT

1. What is friction?

2. What are the four major properties of lube oil?

3. Define viscosity.

4. What are the three main types of lubrication?

5. What is a grease?

G.S. Armstrong

Mechanical Equipment - Course 430.1

BEARINGS - UNIT 1

OBJECTIVES

From memory, the student will be able to:

1. Name the four roles of bearings.

2. Delineate the major types of bearings in a branch tree format as outlined in the text.

ROLES OF BEARINGS

By definition, bearings are parts of a machine in or on which another part revolves or slides. To rephrase the above, a bearing is simply a supporting part. This is the primary function of a bearing - to support. As we will see in later lessons, bearings are also typed according to the way they support the moving part.

So far, we have discussed the most obvious function of a bearing but this is not the only function. Without bearings, in particular, without well lubricated bearings in a machine, a significant amount of contact would take place. Understandably the machine would not be very efficient. A well lubricated bearing would reduce the amount of contact thereby reducing the amount of friction. As a result, the amount of wear would also be decreased.

Lastly, the design of a bearing generally incorporates a replaceable wear surface which is more economical to replace than a shaft. A shaft that is damaged would need to undergo remachining and heat treatments in order to regain its' original condition. This is a rather lengthy and costly exercise compared to replacing parts of a bearing.

EXERCISE

List the four roles of bearings below.

1.

2.

3.

4.

Solution is on next page.

SOLUTION

The four roles of bearings are to:

1. support moving parts,
2. reduce friction in the machine,
3. reduce wear in the machine,
4. provide a replaceable surface which is more economical to replace than a shaft.

TYPES OF BEARINGS

Bearings either belong to the plain or rolling element types. More will be said about each bearing type later. For now, plain bearings are bearings in which the primary motion is sliding. Rolling element bearings, on the other hand, are bearings which have relative motion between two loaded parts accommodated by rotation of balls or rollers.

The bearing tree, so far, looks like this:

BEARINGS

PLAIN

ROLLING ELEMENT

BALL

ROLLER

From the discussion on roles of bearings we mentioned that the primary role of a bearing is to support moving parts. Support means the bearing must be able to handle the different forces which act in the shaft.

If we analyse the types of forces which can be transmitted in a shaft, there are two main types:

1. radial forces - forces which act at 90° to the shaft.

2. axial forces - forces which act parallel to the shaft.

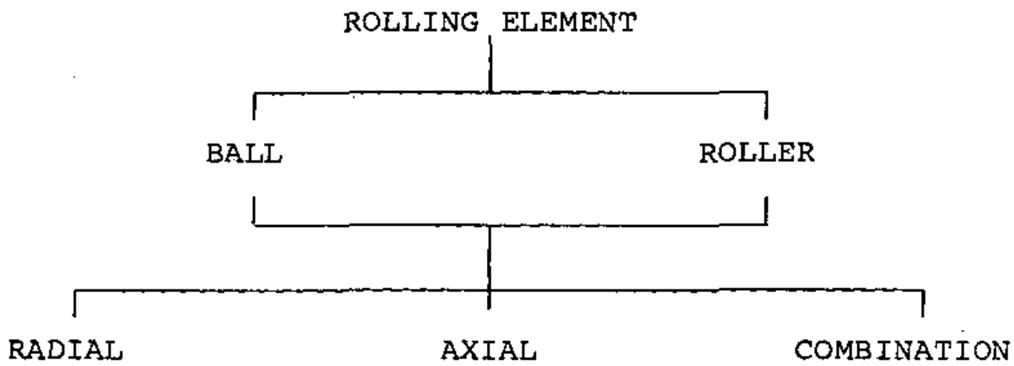
To cope with each force, the plain bearing type must use a different design. We can therefore breakdown the plain bearing group into the following:

PLAIN BEARINGS

JOURNAL BEARING (this type handles radial forces)

TILTING PAD THRUST BEARING (this type handles axial forces)

Rolling element bearings can also be typed according to the type of force handled. Unlike the plain bearing design, some rolling element types can handle a combination of radial and axial loads. Simply, the tree formation of rolling element bearings look like this:

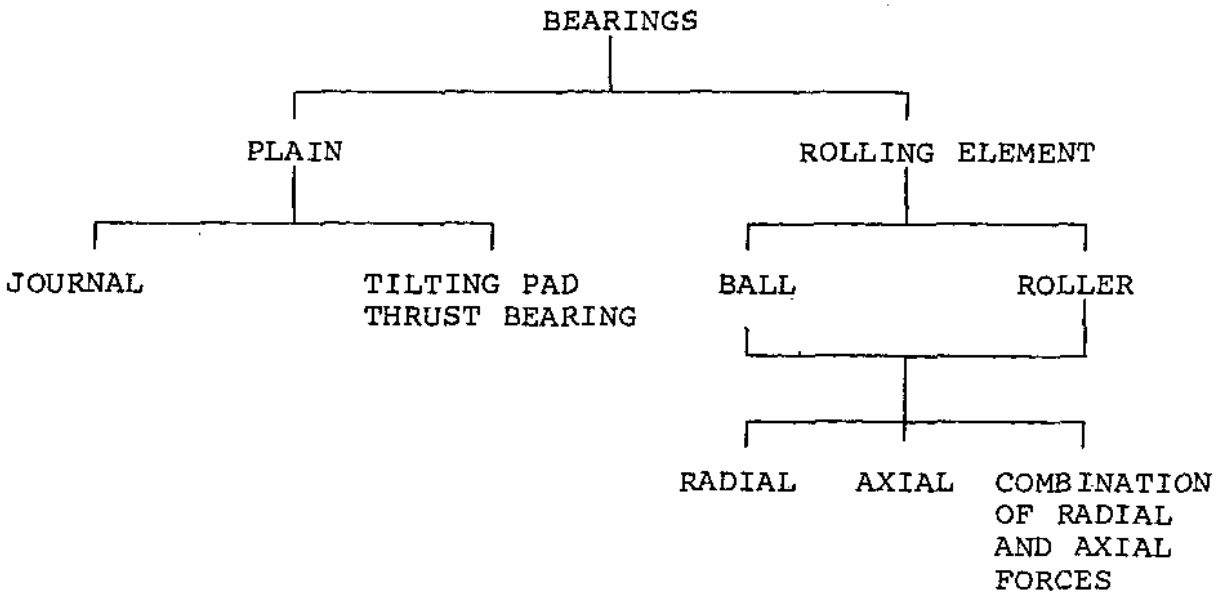


EXERCISE

From memory, draw a branch tree of the major types of bearings.

For the solution, turn to the next page.

SOLUTION


BEARINGS
PLAIN
ROLLING ELEMENT
JOURNAL
TILTING PAD THRUST BEARING
BALL
ROLLER
RADIAL
AXIAL
COMBINATION OF RADIAL AND AXIAL FORCES

Mechanical Equipment - Course 430.1

LUBRICATION METHODS - UNIT 2

OBJECTIVES

Given the three methods of lubrication, hydrodynamic, hydrostatic and boundary, the student will be able to list at least three characteristics for each as explain in the text.

LUBRICATION METHODS

As already mentioned, bearings help to reduce friction and wear in a machine. This we said earlier is more likely to occur in a well lubricated bearing.

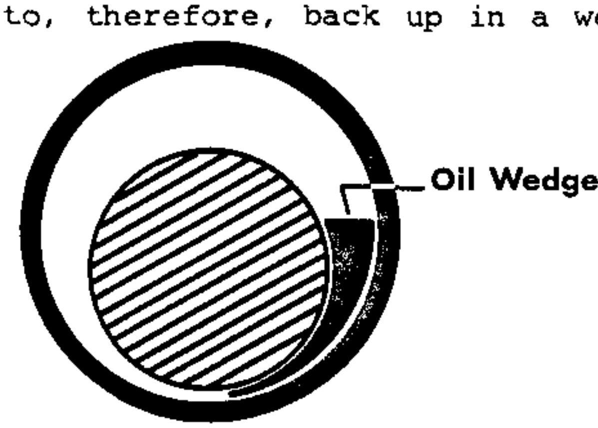
Plain bearings are mostly fluid film bearings. The fluid film generated separates the two surfaces of shaft and bearing and eliminates metal-to-metal contact. In doing so, friction and ultimately wear are reduced.

Two methods are employed to develop this fluid film. The methods are:

(1) hydrodynamic lubrication.

(2) hydrostatic lubrication.

Hydrodynamic lubrication uses a high pressure oil **wedge** to separate shaft and bearing surface materials. The "oil wedge" is developed internally by the bearing. When oil enters the bearing, it is picked up by the shaft surface. Oil on this surface is progressively fed towards the bottom of the shaft. Because of the geometry of the bearing, more oil enters an area towards the bottom of the shaft than leaves. Oil tends to, therefore, back up in a wedge-shaped area.



Since oil cannot be squeezed into a small volume, its pressure builds to separate the surfaces. To develop this oil wedge the following must be present:

(1) a supply of oil - low pressure.

(2) the shaft must be rotating.

(3) there must be a small clearance between shaft and bearing.

If a shaft in a journal bearing is not moving, how is the shaft separated from the bearing? Hydrostatic lubrication is used.

To separate the surfaces between a stationary shaft and a bearing, external high pressure oil is injected **underneath** the shaft. If the pressure is high enough, separation should take place. Again note that there should not be any metal-to-metal contact.

Rolling element bearings, on the other hand, permit motion between two loaded parts by rotation of balls and rollers. Metal-to-metal contact therefore occurs.

Although no special lubrication system is required, lube-oil or grease must certainly be present. Since most of the motion in these bearings is rolling of an element on a race with a small amount of slip, only a small amount of lubricant is required. Even though these bearings are called rolling element antifriction, a thin film of fluid separates the rollers and raceways except under very high load at low speeds (boundary lubrication). When low loads are combined with high speeds the tractive forces are not sufficient to maintain rolling, hence sliding occurs. This generates heat and additional lubrication must be available to keep temperatures down. Roller skidding can occur at constant speed but more common when rapid changes of speed are experienced. This skidding may occur in ball bearings but is more damaging in roller bearings.

EXERCISE

Write down as many points (characteristics) that you can think of for each of the following methods of lubrication in the space provided.

(1) Hydrodynamic.

(2) Hydrostatic.

(3) Boundary lubrication.

See next page of solution.

SOLUTION

Hydrodynamic Lubrication

- used in plain bearings.
- develops pressure by creating an oil wedge.
- no metal-to-metal contact is present.
- shaft must be rotating.
- clearances between shaft and bearing must be small.
- there must be a source of low pressure oil present.

Hydrostatic Lubrication

- no metal-to-metal contact.
- used on plain bearings.
- source of high pressure oil required underneath shaft

Boundary Lubrication

- thin layer of oil or grease present.
- used in rolling element bearings.
- there is some metal-to-metal contact.
- used to keep temperatures down when skidding occurs.

Mechanical Equipment - Course 430.1

BEARING DESIGN AND INSTALLATION - UNIT 3

OBJECTIVES

Given three bearing installations, the student will correctly title and label each installation.

In this lesson unit, we will look at typical construction and installations of a:

(1) journal bearing.

(2) tilting pad thrust bearing.

(3) ball bearing designed to cope with axial forces.

JOURNAL BEARING

As a member of the plain bearing family, a journal bearing supports a shaft by developing a high pressure oil wedge between the shaft and bearing surface.

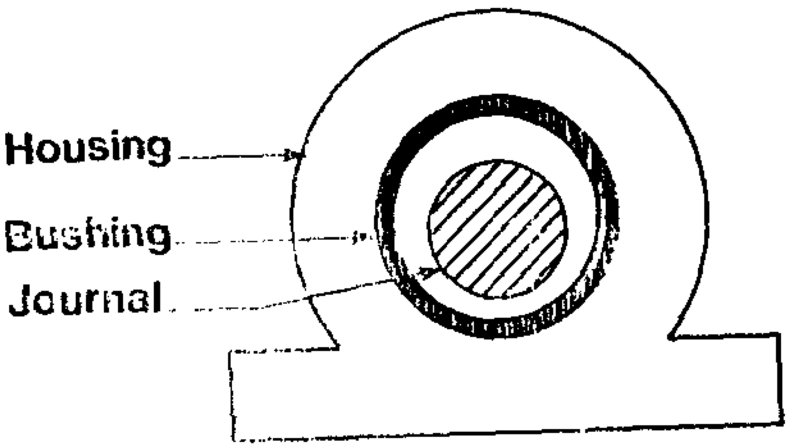
A common journal bearing has three main parts:

(1) a **housing** which holds the bearing.

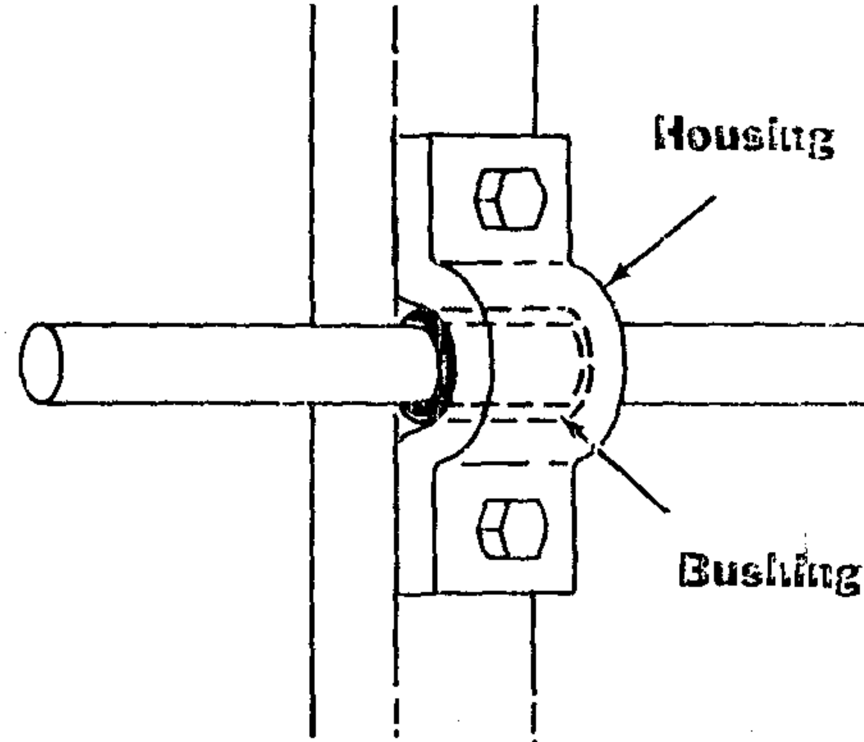
(2) a **bushing** (a soft inner surfaced wrapping) which surrounds.

(3) the **journal** or section of shaft surrounded by the bushing.

Figure 1 shows a journal bearing installation.



Journal Bearing - End View



Journal Bearing - Side View

Figure 1

TILTING PAD THRUST BEARING

Sometimes referred to by their brand name Kingsbury or Michell, this bearing specifically handles axial or thrust forces. Like the journal bearing, surfaces are separated by a high pressure oil wedge developed within the bearing.

Figure 2 and Figure 3 show the general construction of the bearing which consists of:

(1) a fixed collar attached to a shaft with,

(2) a concentric row of individual pivoted pads located on each side of the collar. The pivot allows each pad to tilt with respect to the collar surface to create a wedge shape gap. In turn, each pad's backplate attaches to,

(3) a bearing housing.

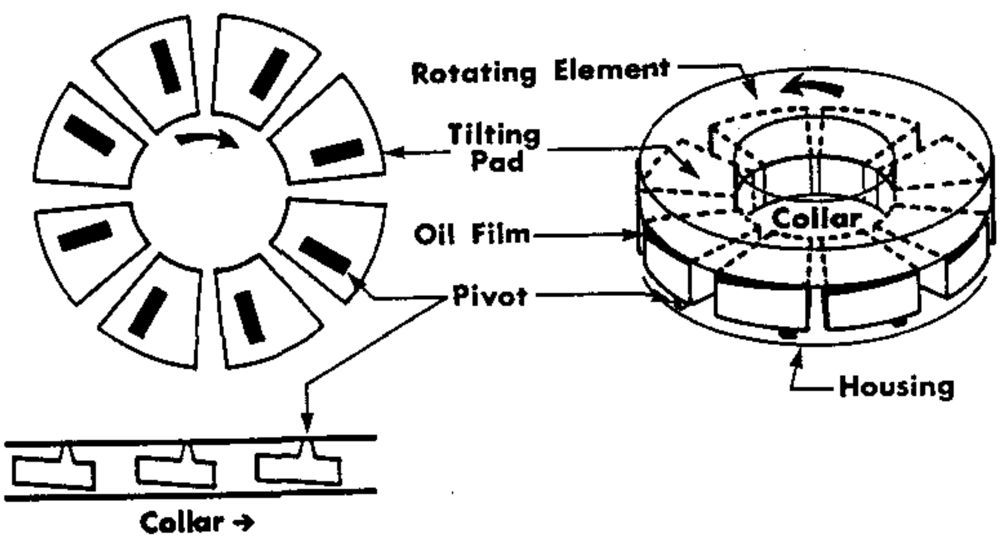


Figure 2 Figure 3

Tilting Pad Thrust Bearing

ROLLING ELEMENT

Ball Bearing (angular contact type)

Since the roller bearing does not significantly vary in design from a ball bearing, only the ball bearing will be discussed.

Each bearing (Figure 4) consists of:

(1) two hardened steel rings called **"races"**.

(2) hardened steel **balls** which roll between the races.

(3) optional **separators** or **cages** which space the rolling elements around the races.

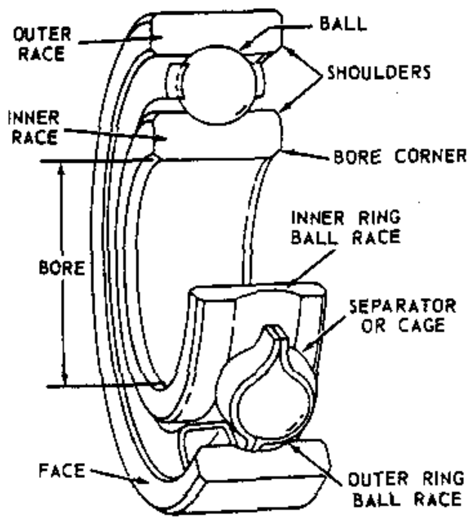


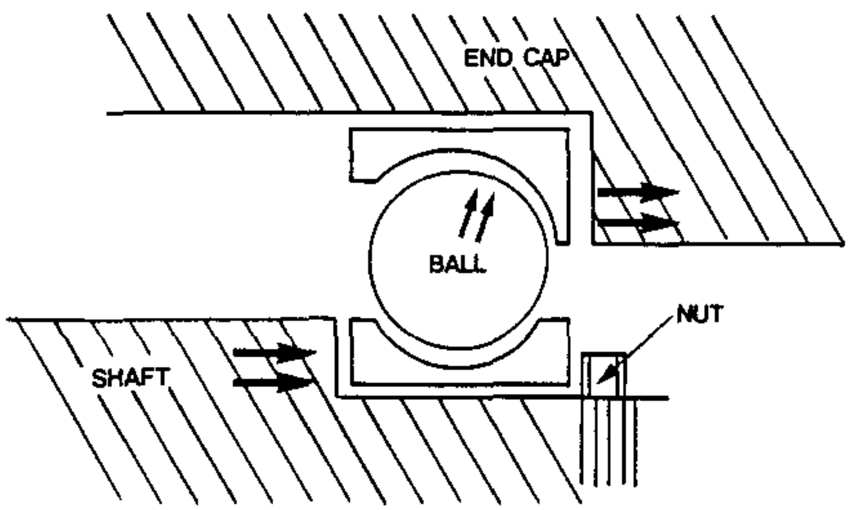
Figure 4

The angular contact ball bearing shown in Figure 5 is used to handle thrust (axial) loads. As the diagram indicates, forces are transmitted:

(a) from the shaft to the inner race of the bearing which butts against the shaft shoulder then.

(b) from the inner race to the balls and finally.

(c) from the balls to the outer race which contacts the bearing cover or end cap. To handle the large axial loads, the outer race of the bearing which contacts the bearing cover is deeper than the other lip which does not make any contact.



ANGULAR CONTACT BALL BEARING

Figure 5

EXERCISE

For the following diagrams of typical bearing installations, correctly name and label each one.

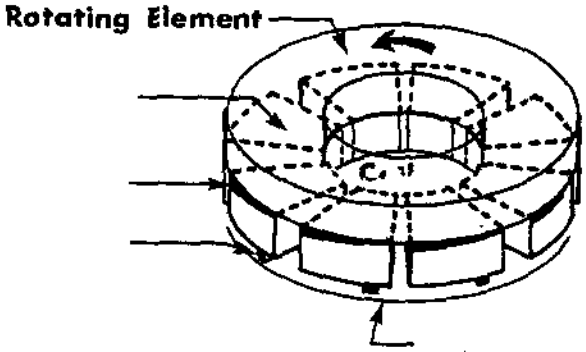


Figure 6

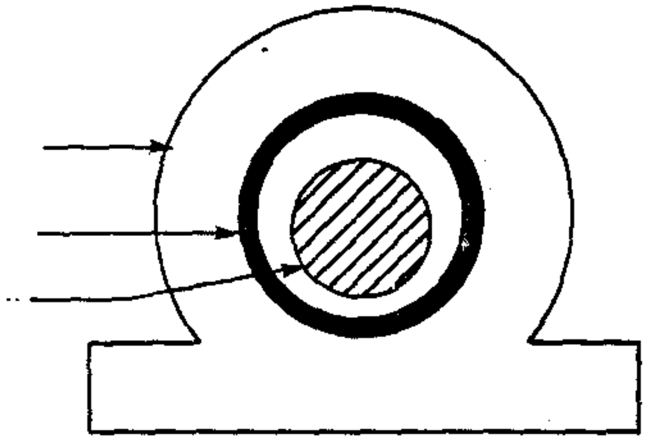


Figure 7

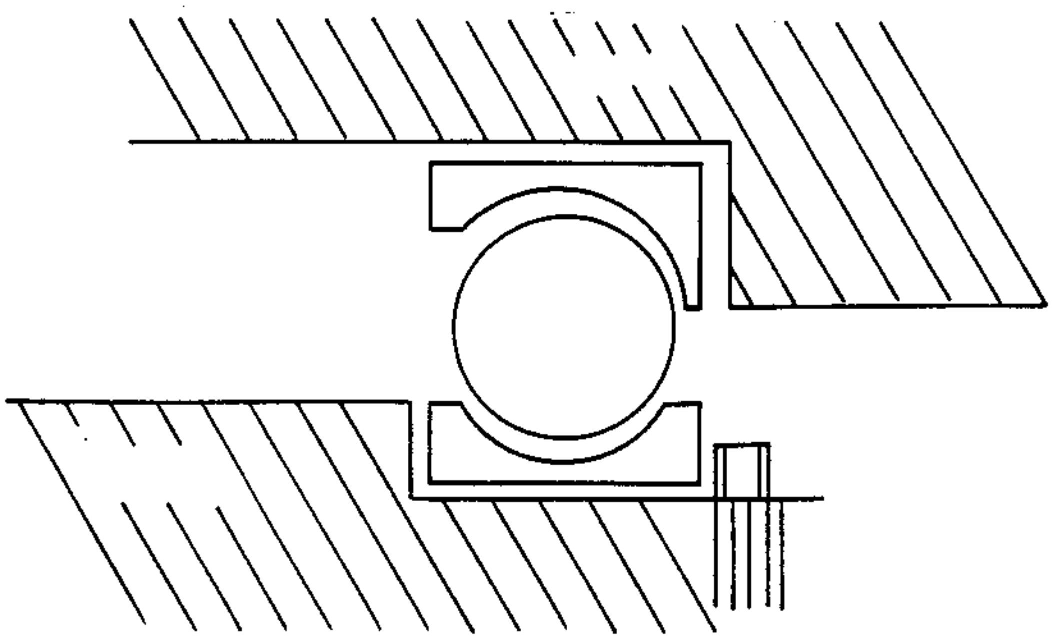


Figure 8

For the solution, check the diagrams which appear in the lesson material.

K. Keown

Mechanical Equipment - Course 230.1

LUBRICATION

Review of Lubrication Principles

Lesson 430.14-1 in the 430.1 Mechanical Equipment course very briefly outlined the types of friction which occurred, types of lubrication and lubrication properties. It would therefore be very helpful if the reader would read that particular lesson prior to taking this one.

In almost all rotating or reciprocating equipment there will be sliding, rolling and fluid friction forces at work. The main objectives of lubrication are therefore to reduce friction forces (reduce coefficient of friction), reduce wear of moving parts, cooling of bearing material, cleansing action and even to act as a shock absorber. In any bearing because of the friction forces, the temperature of the material will increase, therefore by having a flow of oil through the bearing the temperature rise will be reduced. Oil flowing through the bearing will help keep the bearing clean, flushing away any foreign material. A thick film of oil between metal surfaces will also tend to absorb any impact due to sudden load applications.

Types of Lubrication

There are essentially three modes of lubrication, they being:

(a) boundary lubrication

(b) hydrostatic lubrication

(c) hydrodynamic lubrication

Hydrodynamic and hydrostatic lubrication can also be classified as fluid lubrication in that there is a thick enough film of oil to ensure that the surface irregularities do not come into contact (no metal to metal contact). Hydrodynamic lubrication is where an oil wedge is formed within the bearing due to its design which increases the pressure of the oil, maintaining the metal surfaces apart. The oil wedge term is used because, what in fact happens is that the volume through which the oil flows is decreased, thus attempting to compress the oil increasing the pressure.

Hydrostatic lubrication is where oil is injected between two surfaces at high pressure forcing the two apart. Motion of the surfaces is not necessary. Resistance to motion in hydrostatic lubrication is due strictly to the oil viscosity as there is no metal to metal contact.

Boundary lubrication consists of separation of the bearing surfaces by a lubricant which is at best only a few molecules thick. Requirement of a good boundary lubricant is its ability to cling tenaciously to the surface of a material. The lubricant clings to the surface by either absorption or chemical reaction. In boundary lubrication the resistance to motion is influenced by both the lubricant viscosity and the surface requirements as the surface irregularities penetrate the film of oil, thus resulting in metal to metal contact. In actual fact viscosity will have little effect on the friction forces.

Normally a bearing is hydrodynamically lubricated but gradual transition to boundary can occur when as the speed changes or load is increased the oil wedges separating the surface become thinner and the surface peaks begin to penetrate.

Lubricating Properties

Fluid lubricant properties which are of common use:

1. Viscosity: oil's internal resistance to motion.
2. Oiliness: oil's ability to adhere to the metal surface.
3. Temperature Stability: temperature it can withstand without loss of lubricating properties.
4. Flash Point: temperature at which oil vaporizes.

The most important properties of a lubricating oil is its viscosity which largely determines its suitability for any particular application.

Absolute viscosity is determined either as kinematic viscosity in centistrokes or as dynamic viscosity in centipoise, obtained by multiplying the kinematic viscosity by the density of the oil at the temperature of measurement. Lubricating oils have viscosities ranging from 10 - 1000 centistoke at 100°F at which temperature water has a viscosity of about 1 centistoke.

For crankase oils for both gasoline and diesel engines, the Society of Automative Engineers (SAE) in America has adopted a system which grades the oils into seven categories with their viscosity specified at 210°F. The lightest three, SAE5W, SAE10W and SAE20W, are known as Winter or W grades; they have to meet a viscosity requirement at 210°F and 0°F. The other four grades, 20, 30, 40 and 50 oils, have viscosity at 210°F increasing in that order, but no requirement at 0°F.

Multigrade oils are now in use which fall within more than one SAE grade classification. They cover in one oil a Winter grade and a normal grade specification. Pure mineral oils do not normally fulfill this requirement, and thus additives have to be used. A similar classification is used for transmission and axle lubricants for which SAE grades 75, 85, 90, 140 and 250 are specified in terms of viscosities at 210°F and 0°F.

As was stated in the Level 4 course, viscosity is inversely proportional to temperature. The higher the temperature the thinner the oil (viscosity decreases). In a bearing therefore as the speed is increased, the temperature increases, therefore the oil becomes thinner.

Lubricant Additives

Adding something to lubricants is an old art. Steam cylinder oils compounded with animal fats and the marine steam engine lubricant that boasted of blown rape seed oil were among the first additive lubricants. The growth of additives since these early days has been rapid, meeting the needs of an expanding technological world. High bearing and gear loadings, smaller more powerful prime movers, greater speeds and widening range of operating temperatures have been behind the growth in additives.

There are many reasons for using additives in lubricants. Some are designed to protect the lubricant in service by limiting chemical change or deterioration. Others protect machines from effects of outside contamination (products of combustion, for example) which might form harmful deposits. Some additives improve a lubricant's physical properties or give completely new properties, and still others are designed to reduce surface wear.

A list of the common additives in use today, their functions and their applications, is given in Table 1. From this table, one can see what the requirements are for turbine lube oil, reciprocating and rotary equipment, etc.

Types of Lubricants

Lubricants can be classified into three groups, they being:

1. Fluid Lubrication

2. Greases.

3. Solid Lubricants.

Fluid lubrication, whose properties have been discussed, can be further divided into: mineral ois, fixed oils and synthetics. Mineral oils are extracted from crude petroleum and are the most commonly used. Fixed oils are of animal or vegetable origin. These particular oils have a high degree of oiliness and are used as additives with the mineral oils.

Synthetic oils, as the name implies, are built rather than derived. This particular group is becoming very popular, and has a wider range of application than mineral oils.

Lubricating greases are solid or semi-solid lubricants made by thickening lubricating oils with metallic soaps, silica gel, or other thickening agents. Greases are classified according to the type of thickener and their consistency. Consistency is measured in terms of "penetration", the distance a plunger penetrates into the grease under standard conditions. Groups are classified as 000 - 6 greases according to the penetration classification of the National Lubricating Grease Institue (NLGI) in America, 000 being the softest, 6 the hardest.

Greases in which soap is the thickener are known as soap-base greases and are sub-divided according to the types of soap into aluminum, calcium, lithium or sodium greases.

Aluminum greases are smooth, water resistant and adhesive, and are often used as chassis lubricants. Calcium greases are general purpose greases suitable for operative temperatures up to 50°C (120°F). They have drop points around 100°C (212°F) and are unaffected by water. Drop point of grease is defined as the temperature at which a drop of grease first falls through a small orifice at the bottom of a cup when heated. Sodium greases have higher drop points, about 160°C (320°F), and can be used when temperatures are too high for calcium greases. However they tend to emulsify in water. Lithium greases combine high drop point, up to 200°C (390°F) with good low temperature properties and resistance to water. They are used extensively for automotive and industrial applications. Greases thickened with inorganic thickeners such as clays, silica gel, are known as clay base or microgel greases. They have very high drop points of the order of 300°C (570°F) and can therefore be used to lubricate bearings at high temperatures. Gas turbine engines have created a demand for lubricants that will operate over a wider range of temperatures than can be obtained with conventional mineral oils.

Synthetic greases have a wide range of temperature stability but are also very expensive and their use is limited to specific applications.

Comparative Advantage of Grease and Oil in Bearing

The question is always asked whether an oil or a grease should be chosen for a particular application. Each has advantages which are listed as follows:

Advantages of Grease

1. Maintenance may be reduced, no level to maintain, regreasing is infrequent.

2. Proper grease quantity is easily confined in housing, simplifies design of bearing enclosure.

3. Freedom from leakage.

4. Improves efficiency of labyrinth enclosure, gives better bearing protection.

Advantage of Oil

1. Oil is easier to drain and refill, this is important if lubricating intervals are close together.

2. Use of oil makes it easier to control the correct amount of lubricant.

3. Same lubricant may be used on other types of bearing on the same machine.

4. If bearing must operate under high temperatures, conditions favor oil.

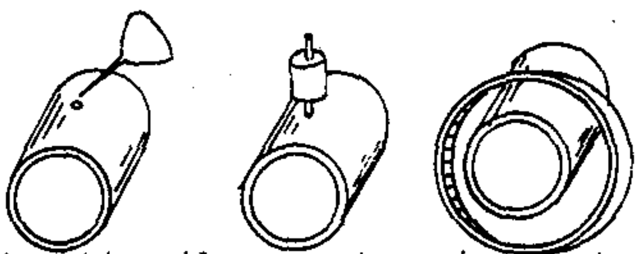
Solid Lubricants

Solid lubricant is simply a solid material between two moving surfaces to prevent metal-to-metal contact. Therefore the application of solid lubricants is generally for boundary lubrication. They may be applied as dry powerders, mixtures with grease and oil or mixtures with binders which form dry films when used. Some solid lubricants in use are graphite powder, molybdenum disulfide, pressed carbon, tungsten disulfide, telfon powder and other plastics.

Solid lubricants such as carbon bearings have the advantage that there is no contamination of the system fluid (D_2O). An example would be the main guide bearing (carbon) in the PHT circulating water pumps at Pickering.

Lubricating Systems

The simplest and oldest method of lubricating the single bearing is the hand operated oil can which is a feast of famine situation. This method is not suitable for the critical needs of todays machinery. Figure 1 illustrates various methods of supplying oil or grease to a single bearing. The automatic oiler (wickerfeed, gravity, etc) and the ring oiled bearing provides for a continuous supply of oil to the bearing. The ring picks up oil from a pool beneath the bearing and drags it through a bearing slot, where the moving shaft distributes it between shaft and bearing. Once through, the oil drains to the reservoir between the bearing and housing. For application of greases, three methods are direct application with grease gun, grease cup and finally the spring operated grease applicator.



OIL FEED from automatic oiler, centre, is constant while machine operates. Only attention needed is occasional refilling of reservoir

GREASE FED to bearing by spring pressure from reservoir, right. Spring compresses each time unit is filled from outside grease gun

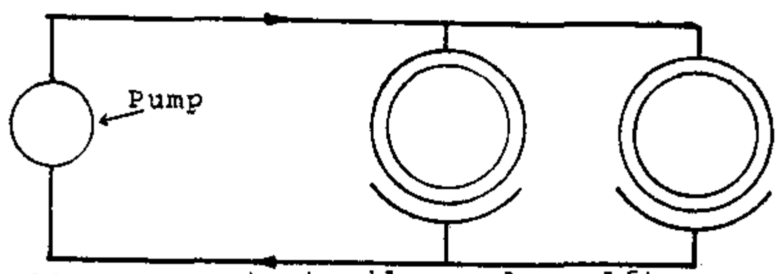
Single Bearing Lubrication

Figure 1

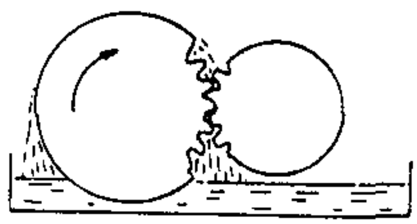
With the possible exception of the ring oiler, the methods just discussed are once through systems. Large power equipment generally uses continuous oil circulation. Advantages of a circulating system are:

1. Adequate oil supply for both lubrication and cooling.
2. Consumption cut by oil recirculation.
3. Dirt removed by flushing action.

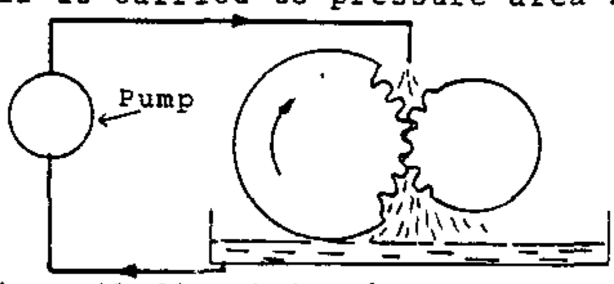
Figure 2 illustrates three basic circulating systems in use today.



EXTERNAL PUMP assures constant oil supply. After passing through bearings, the oil is collected and recirculated, reducing consumption



SPLASH LUBRICATION is simple form of circulating system (like a ring oiler). Oil is carried to pressure area by clinging on the teeth



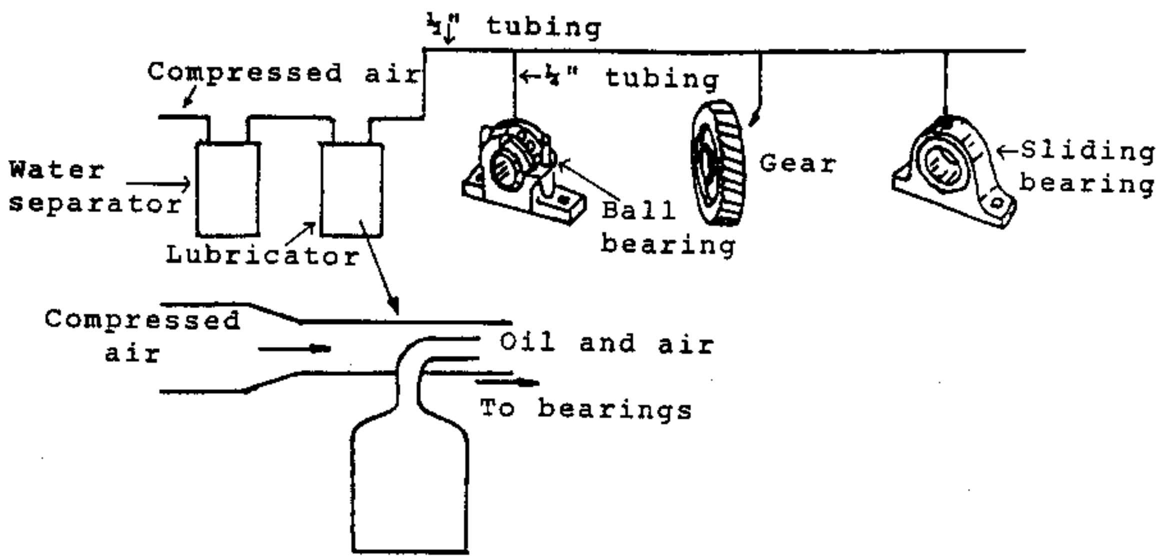
FORCE FEED applies oil direct to the pressure area. Oil conditioning unit can be installed in the feed line to extend life of the lubricant

Basic Circulating Systems

Figure 2

One other method of providing lubrication is to use the aerosol principle. This involves atomization of the oil, and mixing it with air. The mixture is distributed through tubing to bearing surfaces. Moving shaft in bearing actually removes oil out of the air stream, depositing it as a film between the shaft and bearing. Figure 3 illustrates a lubricating aerosol system. Such a system provides continuous lubrication to each bearing. The system is most economical, consumption being in some cases only one tenth as much as for other once through systems.

Airflow through bearings acts as a coolant. Like other systems previously discussed, there is no need for attention other than refilling lubricators as needed.



LUBRICATING AEROSOL carries fine oil particles in suspension until they hit moving surface. Shaft movement drags oil from airborne mixture into the bearing surfaces.

Lubricating Aerosol System

Figure 3

L. Laplante

Additive Type	Oxidation Inhibitor	Corrosion Inhibitor	Extreme Pressure, Oiliness or Film-Strength Improver	Detergent Dispersant	Rust Preventive	Metal Deactivator	Antiseptic (bactericide or disinfectant)	Pour-point Depressant	Emulsifier	Foam Inhibitor	Viscosity-index Improver
Typical compounds	Organics containing sulfur, phosphorus or nitrogen, such as organic amines, sulfides, hydroxy sulfides, phenols. Zinc organics are often used.	Organics containing active sulfur, phosphorus or nitrogen. Typical organic sulfides, phosphites, metal salts of thiophosphoric acid and sulfurized waxes.	Chlorine, phosphorus and sulfur compounds such as chlorinated waxes, organic phosphates (tricresyl phosphate), phosphites, lead soaps such as lead napthenate, fats.	Mettallo-organics (phosphates, phenolates, sulfonates and alcoholates). High-molecular-weight soaps containing tin, barium, magnesium, calcium.	Sulfonates, amines, fatty oils, fatty acids, oxidized-wax acids, phosphates, halogenated derivatives of some fatty acids.	Complex organic compounds containing nitrogen or sulfur; amines and sulfides, some soaps.	Some alcohols, aldehydes, phenals and mercuric compounds. Chlorine-containing compounds.	Wax alkylated napthalenes or phenols and their polymers. Methacrylate polymers are also used.	Certain soaps of fats and fatty acids, sulfonic acids or napthenic acids and surfactants	Silicone polymers and modified waxes.	Polymerized olefins or iso-olefins. Butylene polymers, methacrylic - acid - ester polymers, alkylated styrene polymers.
Why additive is used	Minimize formation of varnish and sludge on metal parts, prevent corrosion of alloy bearings	Protect alloy bearings and other metal surfaces from corrosion.	Reduce friction and wear, prevent galling, scoring and seizure.	Keep metal surfaces clean, prevent many types of deposits.	Minimize rusting of metal parts during equipment shutdown, storage and shipment.	Pacify, prevent or counteract catalytic effect of metals on oxidation process.	Control odor, metal staining and emulsion breaking in emulsion type oils.	Lower the pour point of the oil.	To produce a coolant-lubricant type fluid from soluble oils and water. Reduces water washing, rusting.	Prevent the formation of stable foam.	Reduce rate - of - change of viscosity with temperature.
How additive works	Reduces formation of acidic bodies by decreasing amount of oxygen taken up by the oil. Ends oil-oxidation reactions by forming inactive soluble compounds or by taking up oxygen. Additive may be oxidized instead of oil.	Inhibits oxidation, preventing formation of acidic bodies. Allows a protective film to form on metal surfaces. Catalytic oxidation of oil is decreased by this chemical film.	Chemical reaction forms a film on contacting metal surfaces. Film has lower shear strength than base metal, thereby reducing friction. Helps prevent welding, seizure of contacting surfaces when oil film is ruptured.	Chemical reaction prevents deposition of oxidation products and other substances.	Polar or chemical type surface-active materials are preferentially absorbed on metal. Film repels water attack.	Physical or chemical adsorption forms inactive protective film. Catalytically inactive complex is formed with soluble or insoluble metal ions.	Reduces or prevents micro-organism growth particularly harmful to emulsified oils.	Coats wax crystals in oil to prevent growth and interlacing at reduced operating temperatures.	Surface-active chemical agents reduce interfacial tensions so oil can be timely dispersed in water.	Reduces interfacial tension, allowing small bubbles to combine into larger air pockets. Latter will separate faster.	Improvers are less affected by temperature change than is oil. Example: They raise viscosity by a greater proportion at 200 F than they do at 100 F.
Typical lubricant applications and additive that may be used											
Turbines, electric motors, spindles, hydraulic and circulating systems	✓		✓		✓	✓		✓		✓	
Air compressors	✓		✓		✓			✓		✓	
Gears (hypoid, heavy-duty, worm)			✓		✓					✓	
I-c engines	✓	✓	✓	✓	✓	✓		✓		✓	✓
Soluble EP oils			✓		✓		✓		✓	✓	
Fire-resistant non-aqueous hydraulic fluids	✓		✓		✓					✓	✓
Ball- and roller-bearing greases	✓		✓		✓	✓					

ASSIGNMENT

1. Define the following terms:

   (a) Boundary lubrication

   (b) Hydrostatic lubrication

   (c) Hydrodynamic lubrication

2. List four lubricant properties that are considered in choosing the correct lubricant for a specific application.

3. List five lubricant additives stating typical applications.

4. Give four reasons why grease might be used instead of oil.

5. Give four reasons why oil might be used instead of grease.

6. What are the advantages of circulating oil system over a once through system?

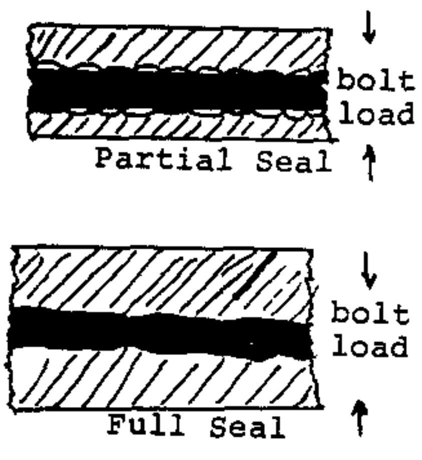
Mechanical Equipment - Course 430.1

SEALING DEVICES

There are three main methods by which a seal is made whether or not motion is involved. These are packings, gasketing and mechanical sealing. Each has its own specific use although some can serve double duties.

Taking first of all gaskets we must establish what a gasket is supposed to accomplish. In piping and machinery we have a problem of making pressure tight joints between two rigid elements. This can be done without the use of gaskets but requires the surfaces to be mated perfectly. In very large machines, ie, turbines or large pumps, this in fact is possible and not unusual to find these metal-to-metal or face-to-face joints. However these conditions rarely exist on smaller pumps or piping flanges so a gasket of some nature is used. Because they are designed to give, gaskets make up for imperfections of the average joint.

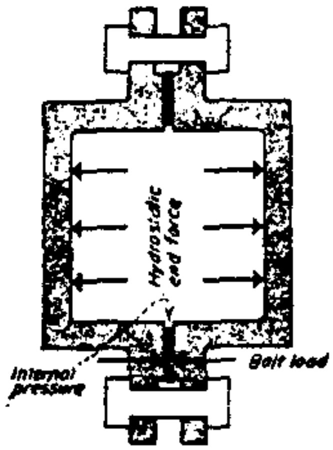
A gasket is a static seal by which a positive leak tight joint is made. In order to do this it must have two apparently conflicting properties. It must be soft enough to flow by compression into the hills or valleys of the joint face, Figure 1, and strong enough to resist internal pressures attempting to blow it out from between the joint faces.



Gasket Must Be Capable of Flowing, When Under Bolt Load, Into All Irregularities to Form a Full Seal

Figure 1

The compressability of a gasket will depend upon the service it is to be used in and hence the bolt load. It is of utmost importance that a plain gasket be thick enough to fill the irregularities and yet thin enough to give as little surface area as possible for the pressurized fluid to act upon. Another consideration to be taken into account is the I.D. of the gasket with respect to that of the joint. As seen in Figure 2, if the gasket is smaller than the flange then internal pressure acts on the flange faces and attempts to separate them thus reducing the squeeze on the gasket material. The closer the I.D. of the gasket is to that of the ftange then the lesser is the tendency to separate the joint.



When I.D. of Gasket is Greater Than Flange I.D.
Hydrostatic End Force Reduces Squeeze on Gasket

Figure 2

There are many materials available for gaskets. The selection is dependent upon the service conditions. The basic criterion are temperature, pressure and fluid being conveyed in the system. Some of the more common gasket materials are given below.

Asbestos - probably the most widely used material where heat is involved. It is pressed, woven, compounded and reinforced to give service qualities. It is supplied in bulk or in preformed or precut shapes.

Rubber - ideal gasket material because it is elastic and squeezes into joint with comparatively light bolt loading. Natural rubber is used mostly for hot or cold water, sometimes low pressure steam or gas. Synthetic rubber stands up to higher temperatures and some can be used in oil applications.

Silicone rubber or elastomers have excellent heat resistant qualities and good low temperature flexibility. Good with some oils but not solvents or steam-under pressure.

Plastics - ie, teflon or Kel-F good for high temperatures or corrosive fluids or replacing synthetic or natural rubber (Figure 3).

Woven asbestos, stainless steel core.

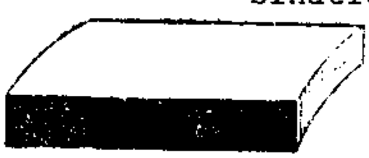
Stainless steel insert.

Woven asbestos compressed core.

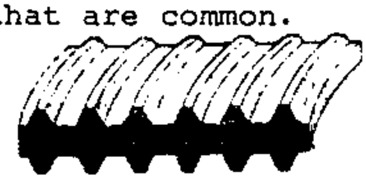
Typical Fillers in Teflon Jackets

Figure 3

Metals - pressure, temperature and corrosion resistance determines the materials and their construction. Lead, tin, copper, aluminum, brass, monel, nickel, silver, steel, platinum are all used for gaskets. Some examples are shown in Figure 4, to show the different shapes and combinations that are common.



Plain Solid

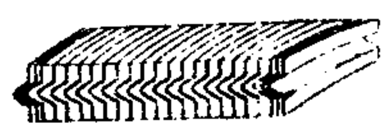


Serrated

Corrugated

Profile-Clad

Double Jacketed



Spiral-Wound

Metal Gaskets Showing Some Typical Configurations
Top Row - Plain Metal,
Bottom Row - Cladded or Filler Type Constr.

Figure 4

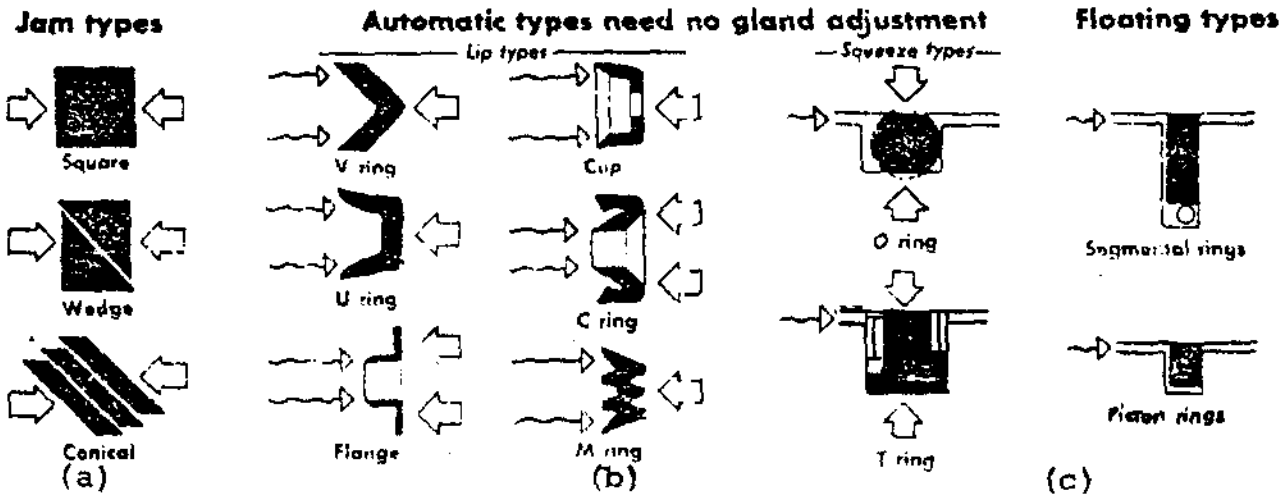
Let us now look at the problem of sealing moving components such as is found in pumps either rotating or reciprocating or helical found in valves. This problem is normally solved by using either packing or mechanical sealing devices.

Packing is a dynamic seal producing a relatively leak tight joint. By this we mean that packing throttles leakage not stop it altogether. This is because packing acts like a bearing and must be lubricated like one. Lubrication may come from a slight or controlled leakage from within themachines or in emergencies from a saturant in the packing. If these are not possible then packing must be lubricated in some other way as dry packing runs hot, hardens and either scores the shaft or allows excess leakage like any other bearing failure.

Packings fall into three broad classes. First there is the jam type (Figure 5 (a)) which includes any packing that is jammed into a stuffing box and adjusted from time to time by tightening the gland (Figure 6). These are normally braided, twisted, woven or laminated asbestos, cotten, rubber, leather, etc.

Secondly there is the automatic type (Figure 5(b)) which do not usually need gland adjustment. The fluid sealed supplies the pressure by forcing packing against a wearing face. They can be further divided into lip types and squeeze type. Common lip types are "U" or "V" ring, cup, flanged, etc. The squeeze type is the "O" ring.

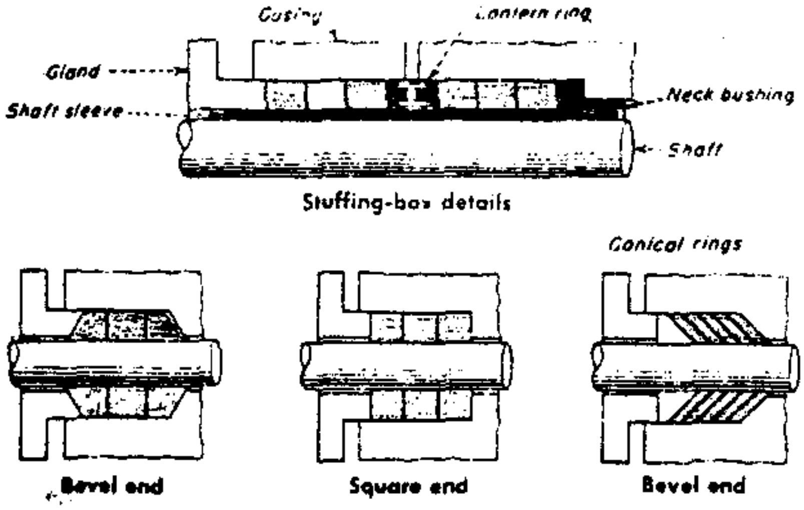
Third is the floating type (Figure 5(c)). These include segmental rings of carbon, metal, plastics, etc, held around the shaft by springs etc.



Types of Packings

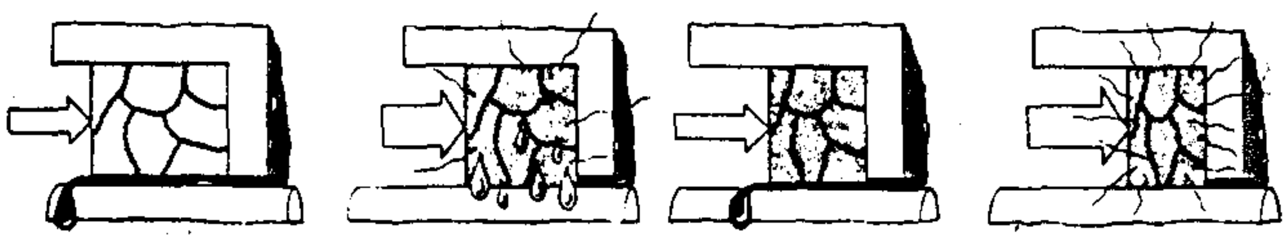
Figure 5

Lubrication of packing is achieved by allowing leakage along the moving member or by supplying a fluid to the gland so that it may travel along the shaft. This latter method is achieved by means of a lantern ring as shown in Figure 6. When there is no fluid leakage then packing must be capable of supplying the lubrication. This is done by saturating the packing with a lubricant. When this lubricant has been consumed through use, Figure 7, the packing becomes dry and brittle. At this point it must be replaced. It is quite common to find the first one or two rings of packing adjacent to the gland ring requiring renewal most frequently (Figure 8) as this is where the gland packing ring exerts its greatest force.



Types of Stuffing Boxes Showing Variations

Figure 6



(a) New Packing

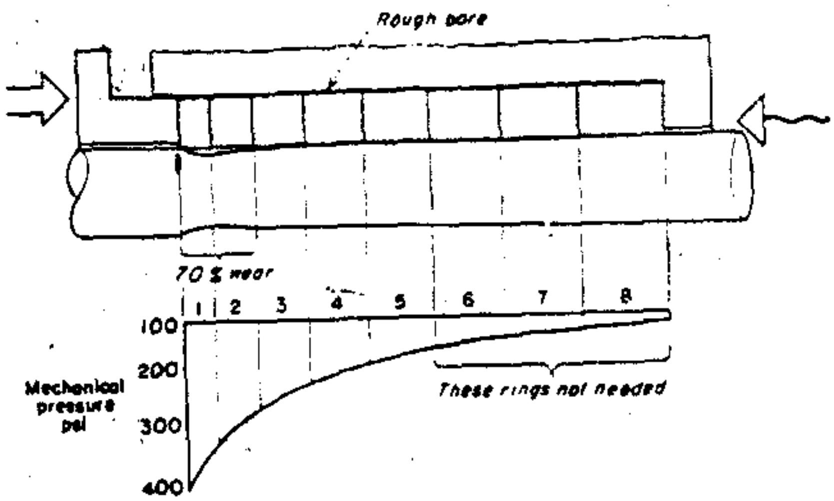
(b) No Fluid Leakage: Saturant Oozes Out.

(c) Fluid Lubricates.

(d) No Saturant Left.

Saturant In Packing Deteriorates Through Use When No Lubricant Is Supplied From Machine

Figure 7

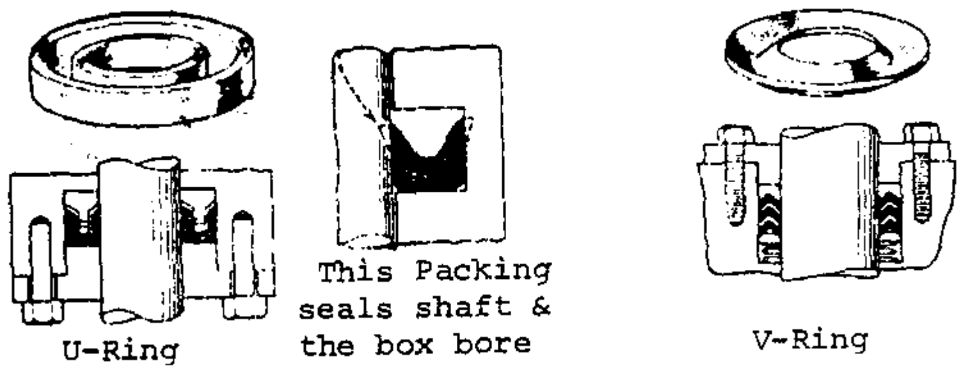


Load on Packing is Highest at Gland Ging and Hence These Packing Rings Need Most Frequent Replacing

Figure 8

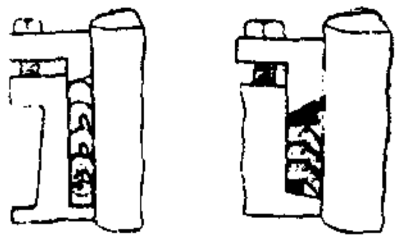
The use of automatic types helps to eliminate much of the human error in adjusting the jam type packing correctly. It is designed to make a seal by using the pumped fluid pressure. In some of the lip type packing small adjustments are required. Both variations are shown in Figure 9 and 10.

Lip Interference



Lip Type Packing Requiring No Gland Adjustment

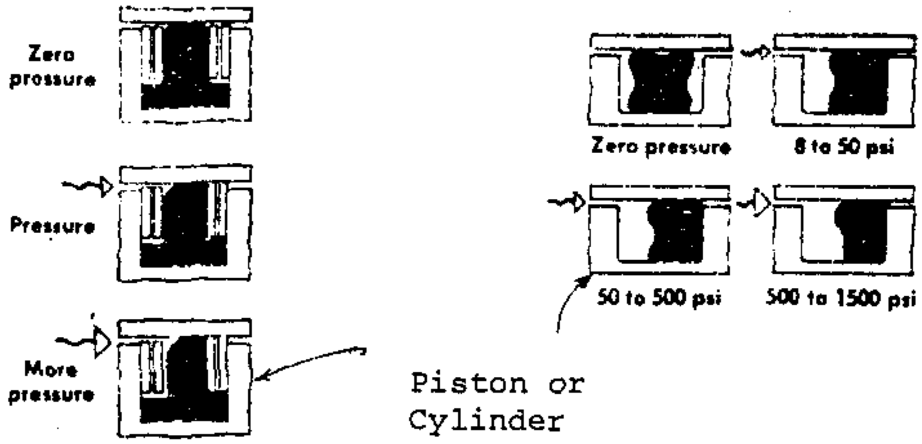
Figure 9



Lip Type Packing Requiring Gland Adjustment

Figure 10

The automatic squeeze type has an interference built into the ring causing it to be squeezed against the sliding and static surfaces without the use of the pumped fluid pressure. One of the drawbacks of the "O" ring is that under high pressures extrusion will occur due to internal fluid pressure. If this occurs the life of the ring will be shortened. The use of the back up rings is one method of reducing this effect (Figure 11).

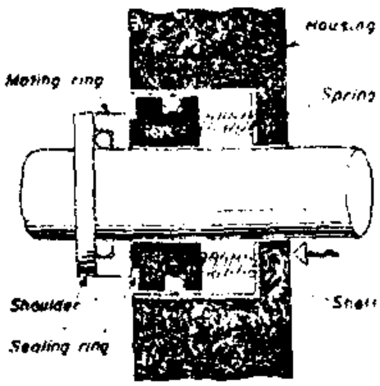


Squeeze Type "O" Ring Packing Showing the Effect of Pressure and the Use of Back-Up Rings

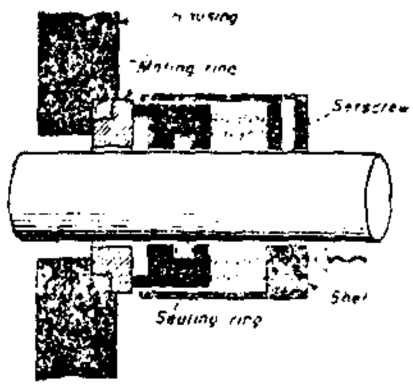
Figure 11

The third method of producing a seal is by means of a mechanical seal. It produces a positive leak tight seal. The various types of mechanical seals will be dealt with at level three and it is only intended to give a very basic description here.

The basic mechanical seal has two rings with wearing faces at right angles to the shaft. One ring is fastened to the shaft and revolves with it and the other is stationary and is held against the machine casing. The sealing ring is held against the stationary ring by means of a spring or springs and keeps a constant sealing pressure at the face. The sealing face can be very small and therefore minimum friction. Figure 12 shows the two basic types of mechanical seals.



Stationary Seal



Rotating Seal

Two Basic Types of Mechanical Seals

Figure 12

ASSIGNMENT

1. What is a gasket?

2. What is the most common material used for gaskets?

3. What are the types of packings?

4. What are the two basic types of mechanical seals?

G.S. Armstrong

Mechanical Equipment - Course 330

AXIAL MECHANICAL SEALS

There are two basic methods of obtaining shaft sealing in pump casings: conventional stuffing boxes, making use of soft pliable packing and face type mechanical seals, the latter being the newer method. Two special methods of sealing different from the above two use fixed labyrinths and floating seal rings. These two will not be discussed in this lesson. As the conventional stuffing box method was discussed in Lesson 430.15-1 no further mention of it will be made here.

Face type mechanical seals have certain advantages and disadvantages which are:-

Advantages

1. Reduce friction
2. Elimination of wear on shaft or shaft sleeve
3. Low controlled leakage over a long service life
4. Relative insensitivity to small shaft deflection or end play.
5. Freedom from periodic maintenance.

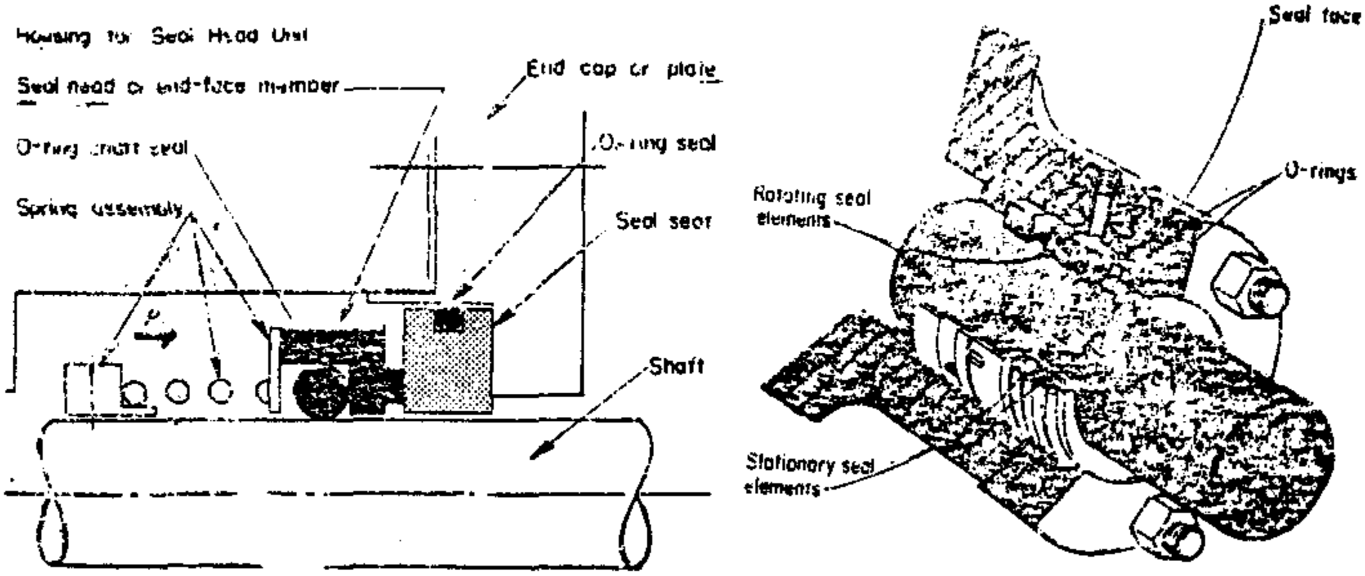
Disadvantages

1. Being a precision component it demands careful handling and installation.

2. Seal failure results in a lengthy shut-down, because to replace it one usually has to remove the driving motor and associated couplings.

Basic Design

A face type mechanical seal consists of two mating seal rings, one stationary one rotating, with extremely smooth parallel faces, a spring loading device and static seals. A typical seal is shown in section and cutaway in Figure 1.



Basic End-Face Mechanical Seal Design

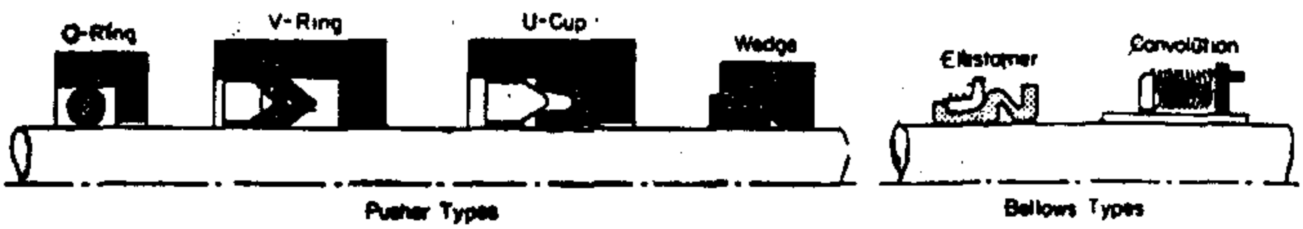
Figure 1

The rotating seal head which is fixed to the shaft is held against the stationary seal seat by means of the spring force and the hydraulic load acting upon it. Sealing takes place between the two surfaces of the seal head and seal seat. Since the rotating seal head is stationary with respect to the turning shaft, sealing at their junction point is accomplished easily through the use of an O-ring. A static seal is also obtained between the seal seat and the pump end cap by means of an O-ring.

Shaft sealing elements include O-rings, V-rings, U-cups, wedges and bellows (see Figure 2). The first four of these elements are referred to as "pusher-type" seals. As the seal face wears these pusher type elements are pushed forward along the shaft maintaining a seal. Typical pusher seal materials include elastomers, plastics, asbestos, and metal.

Bellows shaped sealing members differ from the pusher type in that it forms a static seal between itself and the shaft. All axial movement is taken up by the bellows flexure. Molded elastomers and corrugated metals are used for bellows.

The bellows type seal element unlike the pusher type member, is not subject to dust contamination. Any material collected in front of the pusher type member may produce a barrier. Also only the metal bellows type are applicable to extreme temperature conditions.



Shaft-sealing Configurations for Pusher and Bellow-type Seals

Figure 2

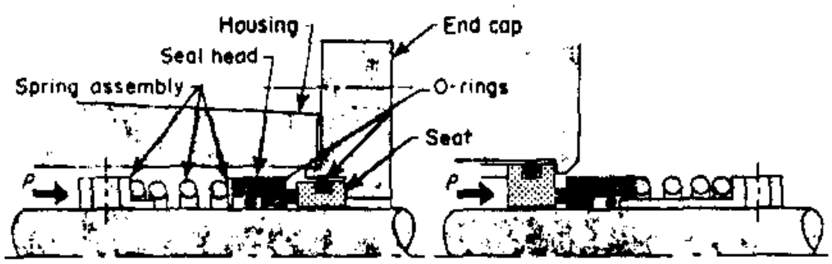
Spring assemblies are added to energize the end-face member axially, keeping the seal faces together during periods of shutdown or lack of hydraulic pressure in the unit.

Various types of springs are used in spring assemblies - single springs, multiple springs and wave springs. Multiple spring design is the most commonly used type. It has a shorter axial requirement than a single coil spring and resist unwinding to a higher degree than single coil spring when subjected to centrifugal force. Face loading can be more readily varied simply by adding or subtracting springs. Wave springs have the advantage of minimum space requirements but greater change in loading for a given deflection is required.

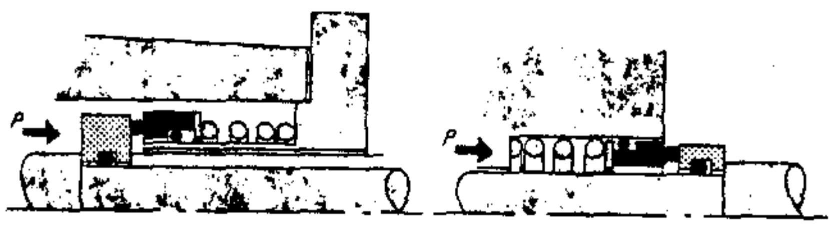
Positive drive is applied to seals to secure mechanical engagement to the shaft and to eliminate stress on the static seals. There are many types of positive drive, some of which are dent drives, key drives, set screw drives, pin, dowel and roll pin drive, snap ring drives, etc. The type of drive used depends upon loading requirements.

Stationary and Rotating Seals

The seal design used can have a rotating or a stationary seal head with respect to the shaft. The end face seal illustrated in Figure 1 has a rotating seal head and a stationary seal seat. Figure 3 illustrates both rotating and stationary seal heads, internally and externally mounted.



Rotating Seal Heads



Stationary Seal Heads

Figure 3

An internally mounted rotating seal design is the more commonly used type of seal arrangement in pumps, particularly where balancing is necessary. Balanced seals are discussed further on in this lesson.

Stationary heads are best applied when comparatively high speeds are encountered. The stationary seal head with its relatively simple rotating seat memrer requires less critical dynamic balancing than the rotating seal head with all of its components.

Externally mounted seals in some cases, simplify installation and removal, and also adjustment. For large equipment or cramped quarters cartridge seals have been developed which permit installation and removal of individual parts without having to remove bearings, couplings, housings and other components.

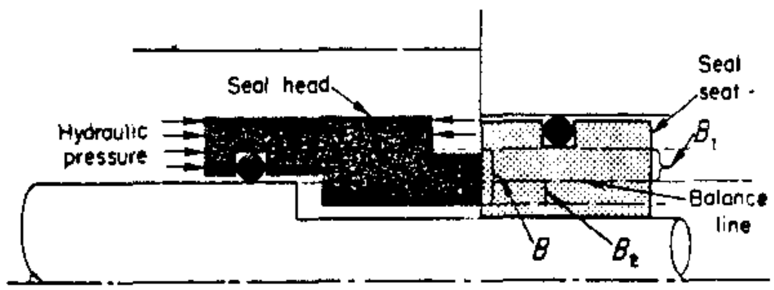
Axial Mechanical End Face Seal Operation

Flow through the seal face is limited by the resistance offered by the two faces. In most cases, leakage is so small that liquid passing through the interface path evaporates into the surrounding air, and the seal appears to be leakless. Some leakage is necessary as it acts as a lubricant to reduce the friction between the two faces. It also removes heat generated at the seal faces. The seal materials are usually poor conductors of heat.

The flow resistance developed by the seal faces depends upon the unit pressure between them. This unit pressure is produced by the hydraulic and spring force pressing the faces together. To help reduce friction between the stationary and rotating face the seal faces must be flat, parallel and very smooth. Also the face materials selected must have low coefficients of friction.

Counterbalancing a small part of the closing force is the pressure gradient acting across the seal interface. This pressure gradient is a result of the liquid pressure acting on one side of the interface and atmospheric pressure on the other.

If very high pressures must be sealed, a different design of seal is necessary which will reduce the crushing or closing force between the seal faces. This seal, called a balanced seal, is shown in Figure 4.



Partial balance achieved when head sealing face is lowered by means of a step cut in the sleeve. Hydraulic pressure acts against a portion of the total face area B. The factor of "per cent of balance" is created by distributing the area of B_1 and B_2 above and below the balance line respectively.

Balanced Mechanical Seal for High Pressure Service

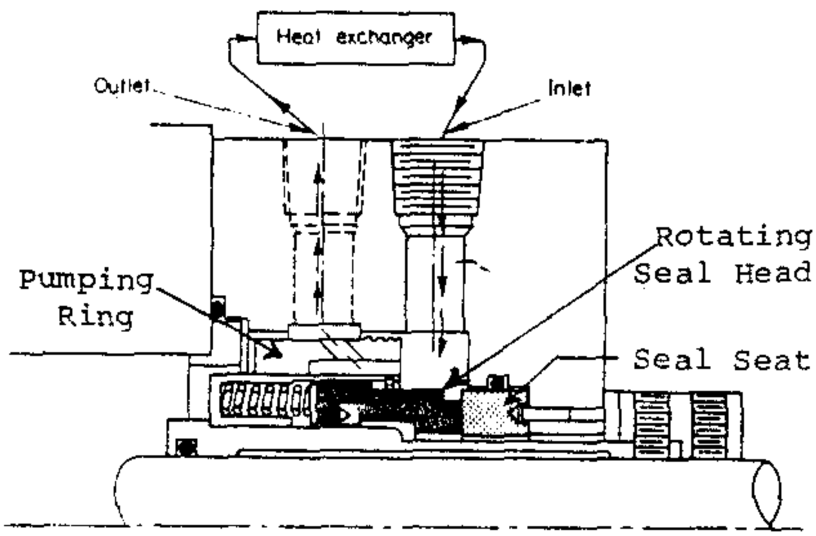
Figure 4

By machining a step on the rotating seal face and a shoulder on the shaft or the shaft sleeve part of the hydraulic force acting on the opposite end of the seal face is balanced. The shoulder on the shaft and the step on the rotating seal face considerably increase the cost of the seal.

Cooling and Lubricant

It is necessary to reduce the liquid temperature at the seal face to some value well below the liquid's atmospheric boiling point otherwise flashing or vaporization of the liquid will take place in the interface. The effect will be a loss of the liquid lubricant permitting the seal faces to run dry. Localized heating of the faces cause minute particles of material to break away forming leak paths. In addition as the liquid vaporizes or flashes, a large volume increase suddenly occurs. The effect is that of small explosions that causes the seal faces to separate, relieve pressure build up and then close rapidly because of the unbalanced spring and hydraulic forces. The rapid closing heavily loads the brittle faces and often breaks away more material.

A water jacket surrounding the whole seal assembly is one way to achieve cooling. Another method cools the liquid from the seal in a separate cooler and then injects the cooled liquid at the seal faces. In this method a small pumping ring on the drive collar of the seal provides the energy needed to overcome friction in the cooler. This method is illustrated in Figure 5.

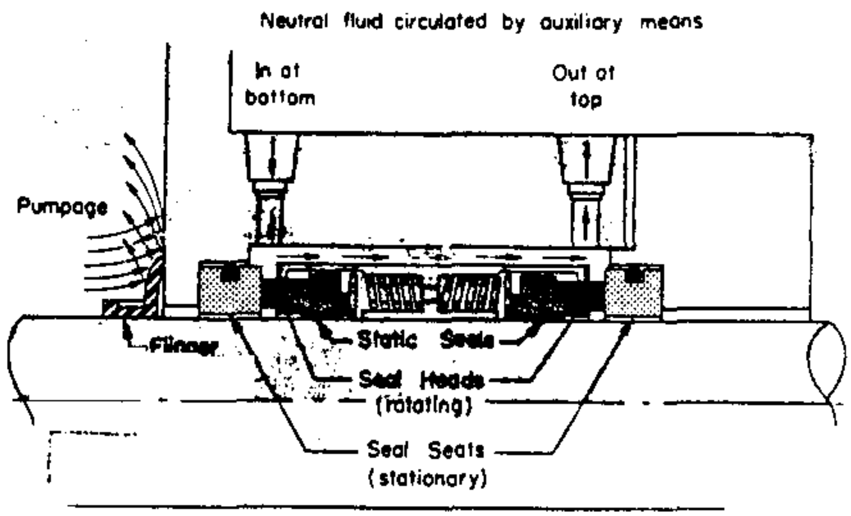


Heat Exchanger and Pumping Ring

Figure 5

Some provision is also necessary to prevent solid particles from damaging the seal faces. Several effective ways of accomplishing this are in use. In one method the seal faces are flushed with clear liquid. Another method simply injects filtered liquid from the pump discharge into the stuffing box.

In cases where contamination of the pumpage is not permitted a double seal design of the type shown in Figure 6 can be used. The neutral fluid is circulated between the two seals at a higher pressure than the pressure against the inboard seal faces thus preventing any outward flow of the pumpage.



Double-Seal Technique for Isolating Seal Fluid From Pumpage

Figure 6

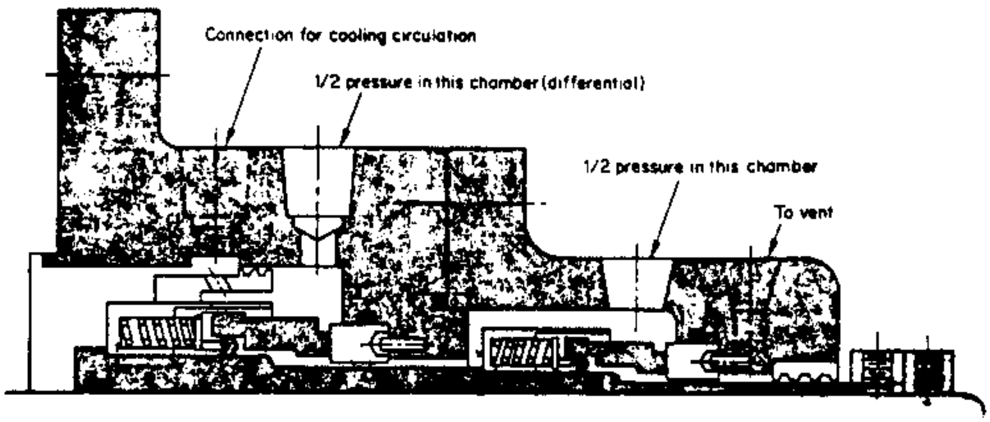
Face Material Combinations

The seal face material combinations depends primarily on the type of working medium. In high-pressure, high temperature water sealing systems found in nuclear power plants, stellites, tungsten carbide, and carbon graphite are commonly used materials. Stellites and carbides are usually used for the rotating seal face whereas the graphite is used for the stationary seal face.

Applications

One of the most critical areas of shaft sealing takes place in the Primary Heat Transport Circulating Pump found in Nuclear Power Plants. These particular pumps, using the

Pickering pump as an example, have a rated capacity of 10,000 Igpm. The heavy water circulated is at approximately 500°F and 1000 psig. These large pumps have a shaft seal of a type designed to collect seal leakage. The seal assembly contains two functionally identical face seals in tandem. A typical tandem seal arrangement is illustrated in Figure 7. (This is not the Pickering pump seal assembly). By breaking down pressures in the respective chambers, each seal faces only the resulting pressure differentials. Tandem seals are used where extra safety is needed, particularly when operating at high pressures.



Tandem Seal Arrangement

Figure 7

ASSIGNMENT

1. List the advantages and disadvantages of the axial mechanical seal.
2. What is the function of the spring in the seal assembly?
3. Why is leakage necessary through an axial mechanical seal?
4. What method is used to partially balance a mechanical seal?
5. Describe a tandem seal arrangement.

L.J. Laplante

Mechanical Equipment - Course 230.1

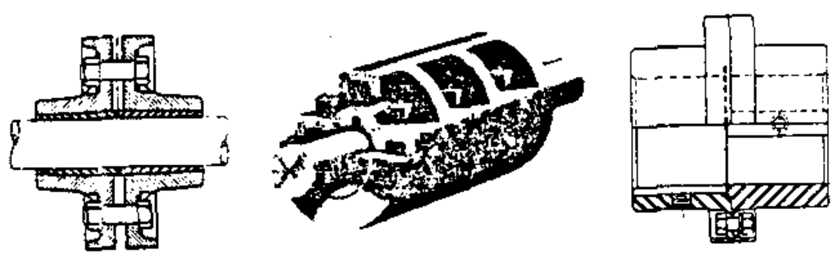
SHAFT COUPLINGS

Couplings

Couplings are used to join two shafts together and to provide some means of transmitting power from a driving source to a driven member. There are two main classifications of couplings, rigid or flexible.

Rigid Couplings

Three types of couplings are shown in Figure 1. Figure 1(a) shows a compression type. These couplings are used for light to medium loads. They consist of taper bored flanges and slotted, tapered sleeves bored to the shaft diameter. As the flanges are drawn together they squeeze the sleeves onto the shafts. The ribbed coupling in Figure 1(b) is used for heavy duty work and to connect two shafts of the same diameter. A single long key is fitted to both shafts and the two halves, which are bored and keyed to suit, are bolted together. The key rather than friction, provides a positive drive. Figure 1(c) illustrates a flanged-faced coupling which can be used to connect shafts of the same or different diameters. Each half of the coupling is bored and keyed to fit the shaft. Set screws secure the hubs to the shafts.

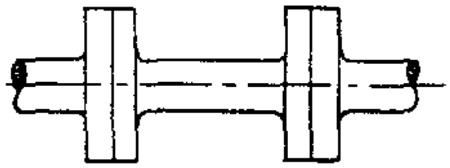


(a) (b) (c)

Couplings

Figure 1

The couplings on the turbo-generator sets at Pickering G.S. are short shafts with flanges forged on each end which mate with forged flanges on the rotor shafts as shown in the diagram below. The flanges are bolted together.



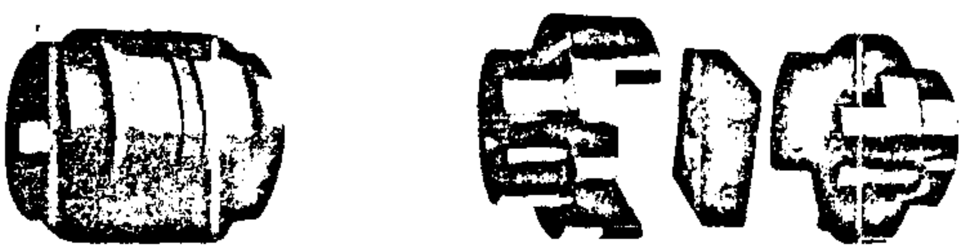
The main disadvantage of rigid couplings is that the alignment must be absolutely correct. If not then stresses can be created which fluctuate due to rotation and lead to fatigue problems.

Flexible Couplings

It is not possible to list all of the various flexible couplings, however they all have one characteristic in common. They will accommodate some misalignment such as may occur through temperature changes or settling of foundations. They are not a means of covering up poor initial alignment - they must still be done very carefully.

Flexible couplings use one of the three basic methods to achieve flexibility. One method is to use tightly fitted rigid parts with a sliding separator. A second method is to use loosely fitted rigid parts and the third method is the use of resilent or flexible parts.

Figure 2 shows two types of couplings employing a sliding separator. The one in Figure 2(a) consists of two flanges with slots milled in these faces. The separator is a disc with a rectangular ridge across each face. The ridge on one face being 90° from the ridge on the other. When assembled the ridges fit into the slots in the flanges and permit some sliding to take place. The exploded view in Figure 2(b) shows a square sliding centre member which engages the flanges. The block may be of self lubricating material or may be oiled.



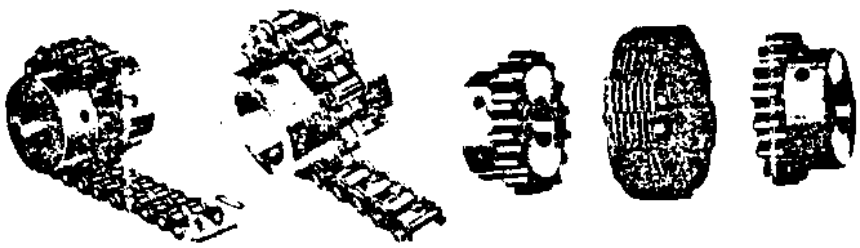
(a) (b)

Figure 2

The slider type of coupling will accept angular misalignment of up to 3°, parallel misalignment of 10% of nominal shaft diameter and end float of 1/32 to 1/4 of an inch. These terms will be explained later under the discussion of alignment. It should be noted that these tolerance do not apply to the initial alignment but are conditions brought on during operation. A flexible coupling does not eliminate the need for careful alignment.

Considering couplings which employ rigid parts loosely connected, Figure 3 illustrated three varieties of one type and Figure 4 shows another type.

The couplings in Figure 3 are called chain couplings. A hardened steel sprocket is fitted to the end of each shaft and then a length of chain is wrapped around engaging both sprockets and is connected at the ends. In Figure 3(a) the cain is a double roller one whereas in Figure 3(b) the chain is a wide single roller chain. Silent chain and sprockets with wide teeth are used in Figure 3(c).



(a) (b) (c)

Figure 3

The tolerances on the roller chain couplings are 1/2° to 1 1/2° angular misalignment, 0.01 to 2% chain pitch parallel misalignment, and .020 to 0.070 inch end play. For the silent chain the tolerances are 1° to 2° angular misalignment, parallel misalignment of 2% of chain pitch and end float of 1/8 to 1 inch. With proper installation, alignment, and maintenance they should have a long service life.

Figure 4 shows a gear coupling which consists of two identical hubs with external gear teeth and a covering sleeve with machined internal gear teeth. The teeth of the hubs mesh with the teeth of the sleeve. The working action is like a spline rather than gears since the parts do not rotate relative to each other. The sleeve may be one piece but for ease of installation it is often in two halves which are bolted together (as in illustration). Gear couplings will permit up to 2° angular misalignment or up to 7° with specially cut teeth. Parallel offset limit can be between .023 to .314 inches depending on the size of the coupling. The end float tolerance will need to be found in the manufacturer's handbook or specification.

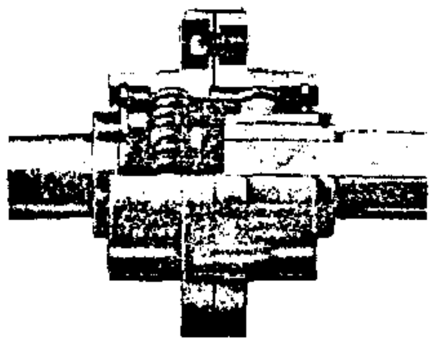
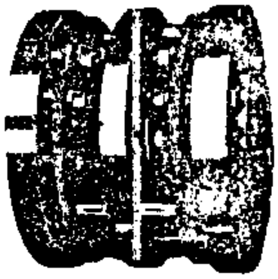


Figure 4

The final class of shaft couplings are those which use a flexible or resilient element in them. The resilient member can be metallic or non-metallic, the choice being dependent of the working environment and the loads involved.

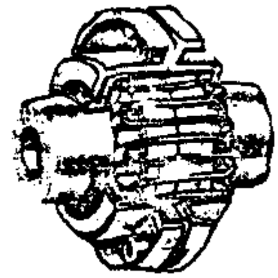
Figure 5 shows a selection of couplings which use metallic elements. Thin metallic discs are used in Figure 5(a), in 5(b) laminated pins. Laminated metal spokes provide the flexibility in 5(c) and a corrugated metal strip fitted into slots in the two halves of the coupling is the flexing element in 5(d). The coupling in 5(e) uses springs and that in 5(f) uses metal bellows. These all come in a variety of sizes, the last two have a particular use in instrumentation.



(a)

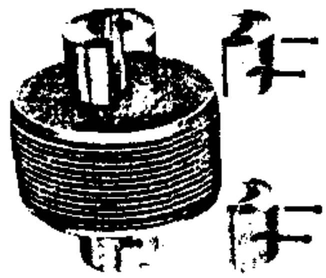
(b)

(c)



(d)

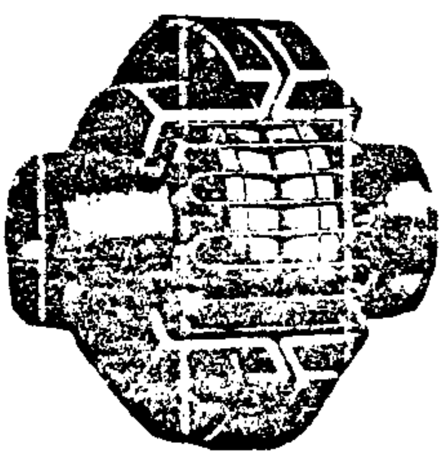
(e)



(f)

Figure 5

Figure 6 shows how one of the above couplings accommodates misalignment.



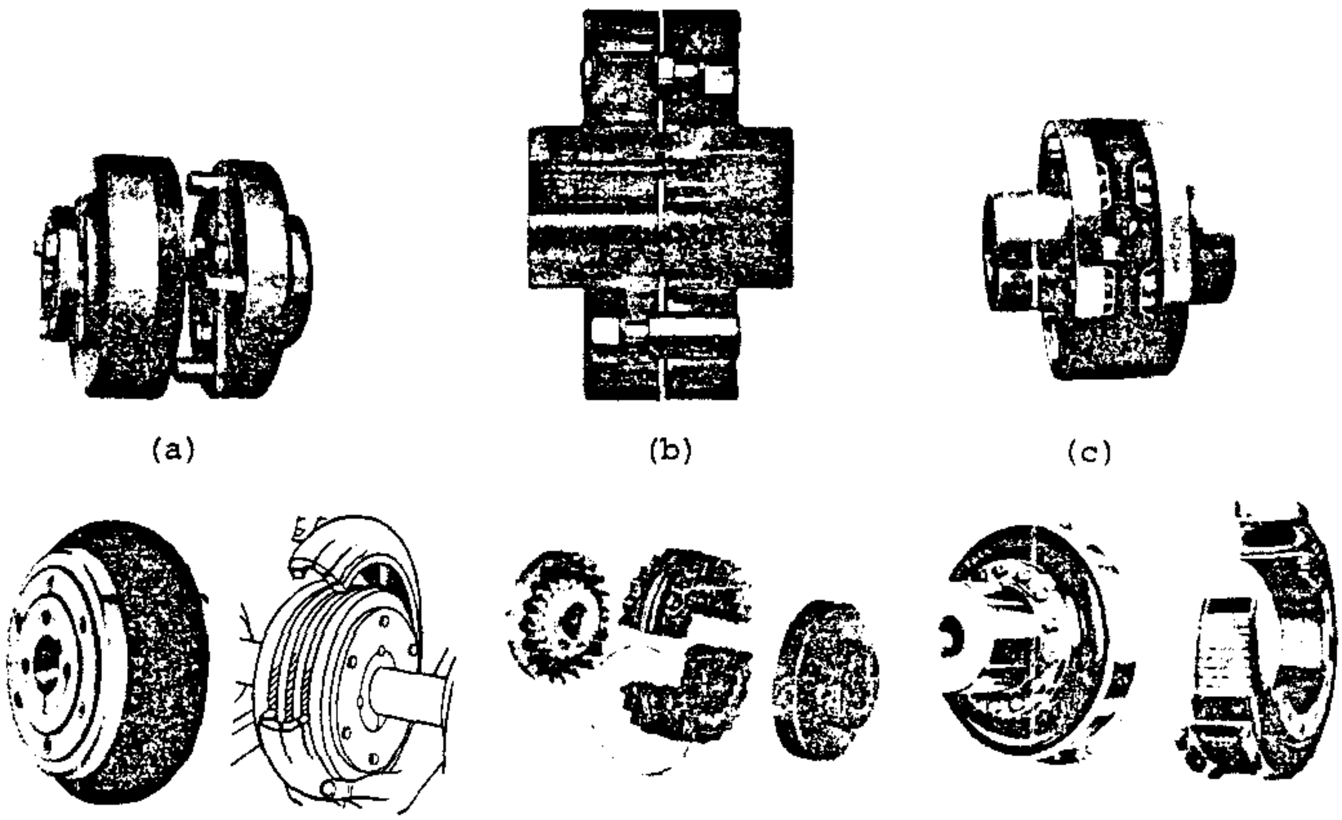
Angular Misalignment

End Float

Parallel Float

Figure 6

Examples of couplings using non-metallic flexible elements are shown in Figure 7. The coupling shown in Figure 7(a) consists of a non-metallic disc placed between two hubs. Pins in each flange engage alternate holes in the elastic disc. Figure 7(b) shows poins in each flange engaging holes bushed with flexible material in the opposite flange. Flexible spools are bolted between the two flanges in the coupling in Figure 7(c) and these absorb the energy of shock loading. In Figure 7(d) the coupling uses a split flexible element which looks like a top, bolted to the two flanges. The flexible element in the coupling in Figure 7(e) resembles a split internal-external gear. It fits into recesses in the flanges. Figure 7(f) illustrates a coupling with a hollow reinforced rubber torus bonded to an inner and an outer rim.



(a) (b) (c) (d) (e) (f)

Figure 7

Before leaving this general description of couplings it is necessary to repeat that flexible couplings are not intended to compensate for poor alignment techniques. Misalignment shortens their life.

Alignment

Three terms have been used in connection with the tolerance of a coupling for misalignment. They are angular offset, parallel offset and end float. Each of these terms are illustrated in Figure 8.

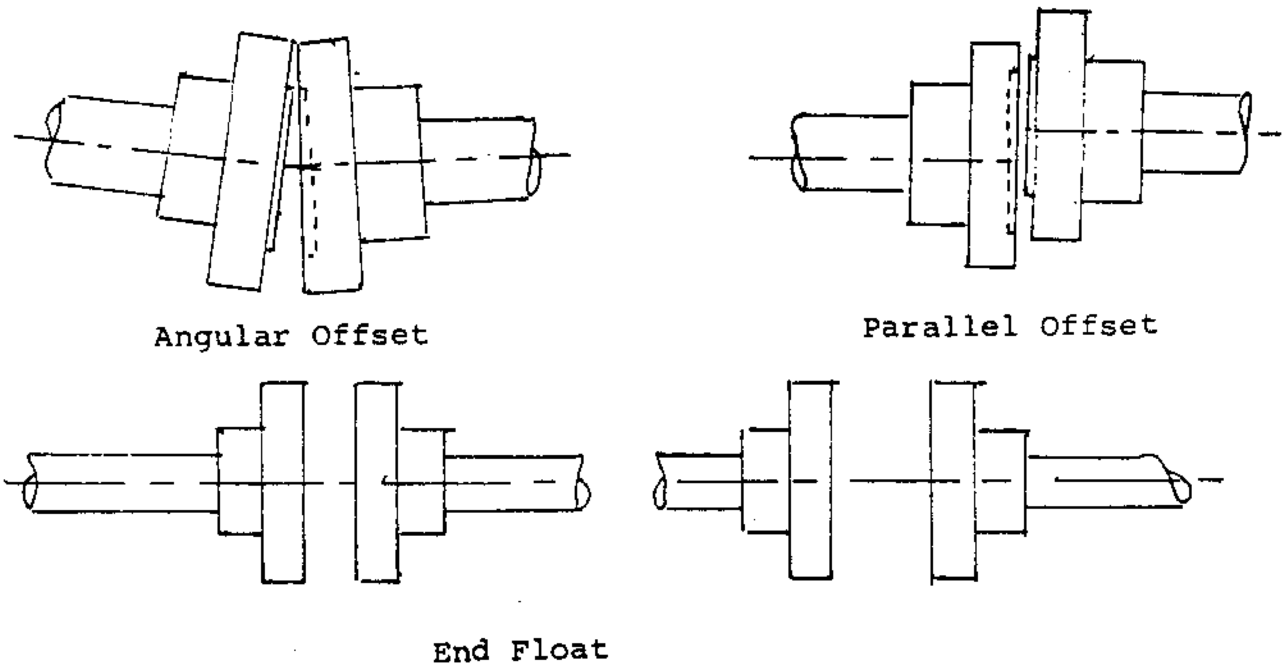
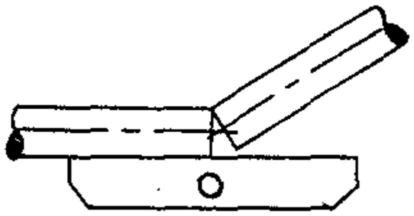


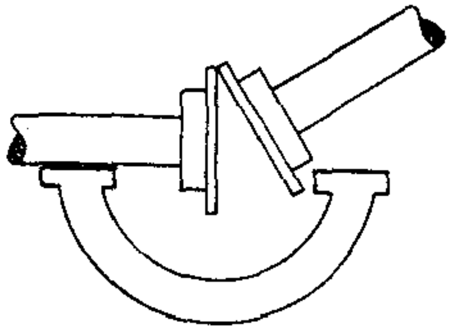
Figure 8

These terms also apply to shafts of course, as does the whole of the following discussion on alignment.

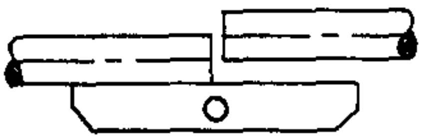
As an example consider the alignment of the shaft of a pump-motor set. First check that neither of the shafts are running-out. It will be impossible to align if one or the other of the shafts are bent or not running true. Make sure that all belt drives or chain drives are slackened off before making this check.

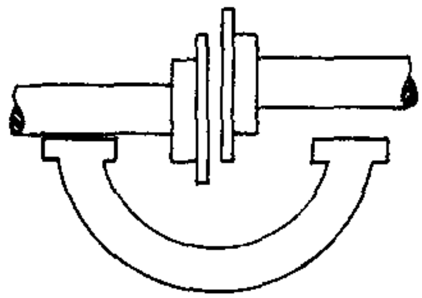
The preliminary alignment both angular and parallel can be done with a straight edge or a special gauge in the case where couplings are installed (Figure 9).





Angular Offset





Parallel Offset

Figure 9

Usually the adjustments are made to the motor and generally it will require shimming to the correct position.

Dial indicators are used to align shafts and couplings to obtain the precise alignment needed in high speed drives.

To check for angular misalignment the dial indicator should be set up in one of the following ways which are illustrated in Figure 10.

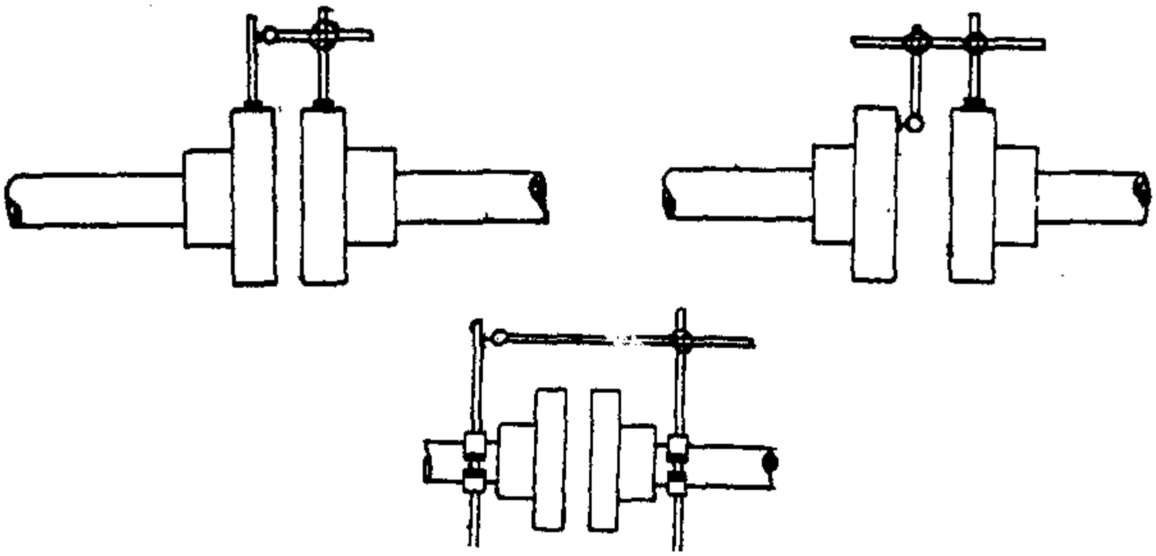


Figure 10

When this set-up is used BOTH SHAFTS MUST BE ROTATED TOGETHER and readings taken every 90°. If all of the readings are the same then the shafts have no angular offset. Under no circumstances should the coupling faces be assumed to be at right angles to the shaft. This is what would be assumed in the second case in Figure 10 if they were not rotated together.

The set-up for parallel offset is shown in Figure 11.

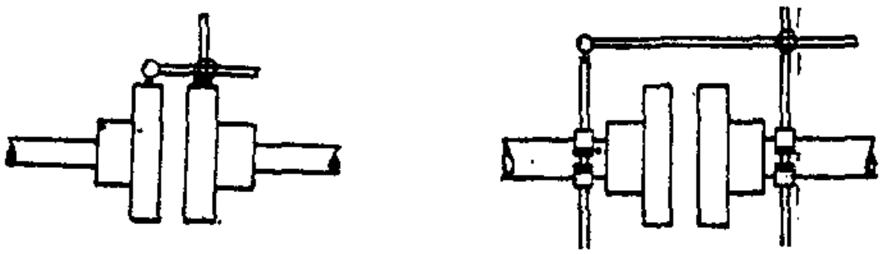


Figure 11

Again the shafts must be rotated together and readings taken every 90°. Corrections will be made by appropriate shimming in the vertical direction or by moving the motor assembly horizontally. The amount of offset will be equal to half the indicated reading for either the vertical or horizontal directions. For example, if the difference between the readings at the top and bottom is 0.016 inches; then the amount of offset is 0.008 inches.

The final check should be made with the components firmly bolted to the base.

Where the pump or motor operates at a high temperature the alignment will need to account for the differences in dimension caused by the change in temperature. Of course it might be possible to align the unit in the hot condition.

If there is insufficient clearance in the bolt holes, they may require to be enlarged.

Alignment requires patience and perserverance. There are no shortcuts.

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ASSIGNMENT

1. What are the differences between the three main types of shaft couplings?

2. What are the three basic types of misalignment?

3. What type of misalignment will the slides type of coupling accept?

4. Why should both shafts be rotated together when aligning them?

Mechanical Equipment - Course 230.1

BELT DRIVES

A common method of transmitting power is a combination of belts and pulleys. There are many types of belts and pulleys used for the transmission of power but, in general, the same principles apply to all of them.

The effectiveness of a belt drive is dependent on the friction developed between the belt and the pulley. Factors which affect the friction are belt tension, length of contact between belt and pulley (arc of contact), cleanliness of belt and condition of belts and pulleys.

The arc of contact should be as large as possible to obtain maximum power of transmission. However, Figure 1 shows that the smaller pulley has an arc of contact less than 180°. Properly designed belt drives should have an arc of contact greater than 120° and therefore the pulleys should be selected accordingly.

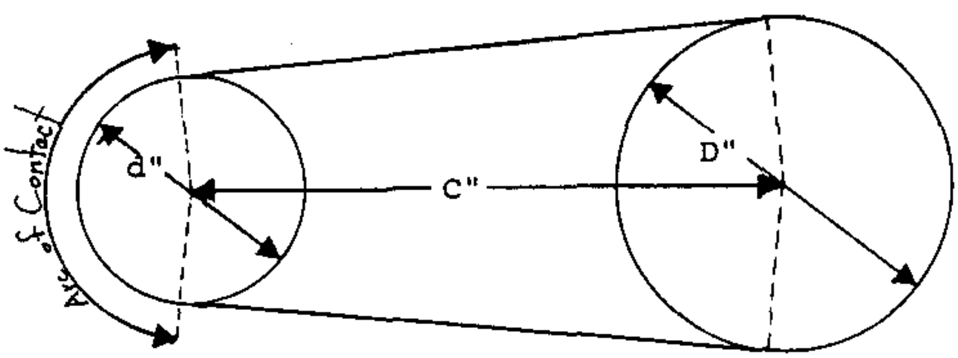

Figure 1

Some of the many types of belt and pulley combinations that will be encountered are:

- flat belts on flat, flanged pulleys or crowned pulleys,
- V belts on flat pulleys or V groove pulleys,
- toothed belts on gear type pulleys,
- round belts on half round grooved pulleys.

Flat Belts

At one time when all the machines in a machine shop were run from a single engine, the power was transmitted by flat belts, pulleys and shafts. The flat belt was probably the most popular form of belt drive. Today, however, flat belts are not used very much, particularly in Ontario Hydro's nuclear stations. Therefore only a few paragraphs will be devoted to discussing.

Flat belts are of various constructions. One type is leather belting made of several plies of leather bonded together. Figure 2 illustrates several other constructions. In Figure 2(a) the construction is plies of rubberized fabric; Figure 2(b) uses rubberized cord and Figure 2(c) is a combination of both (a) and (b). The belt illustrated in Figure 3 is used in instrumentation and is made of polyester film.

(a) (b) (c)

Figure 2

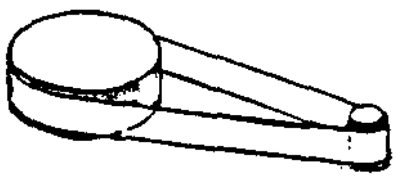

Figure 3

Flat belt pulleys are generally made of cast iron, but can also be made of wood, steel, or other suitable materials. The hub may be split or solid and there may be flanges on the sides of the face of the pulley. Figure 4 illustrates two typical pulleys; Figure 4(a) is split, steel pulley and 4(b) is a cast iron one. The pulleys' faces may be either flat or crowned.

(a) (b)

Figure 4

Crowned pulleys take advantage of the fact that a flat belt tends to move towards a larger diameter. Therefore making the centre of the face of larger diameter than the edges keeps the belt centred on the pulley. A cross section of one type of crowned pulley is shown in Figure 5.

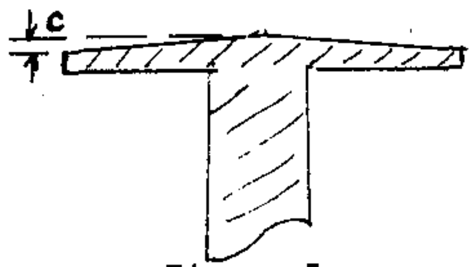

Figure 5

V-belts

Of all belt drives V-belts are probably the preferred drive for transmitted power. In contrast to the flat belt which depends only on the friction between the belt and the pulley to transmit power, the V-belt uses the wedging action of the belt in the grooves of the sheave to increase the frictional force.

The wedging action is a result of the belt being bent around the sheave. In Figure 6(a) the cross-section of a straight portion of the belt shows that the sides are straight, however when the same belt is bent as if around a pulley the sides would be seen to be bulging out as in Figure 6(b).

(a) (b)

Figure 6

Figure 7 illustrates the typical method of constructing a V-belt. The load-carrying cords may be of a variety of materials such as rayon, nylon, glass-fibre or even steel, and there may even be several layers of cords. The cover can be one of several rubberized fabrics depending on the type of service for which the belt is intended. The main body of the belt is made of some kind of rubber, again chosen to suit the working environment. Some materials are good for oil resistance, others stand up better to heat or ozone attack.

V-belt cross-section dimensions conform to a standard and use a letter designation (A, B, C, D, or E) as illustrated in Figure 8. However the method of specifying length varies between manufacturers and therefore their catalogues will need to be consulted.

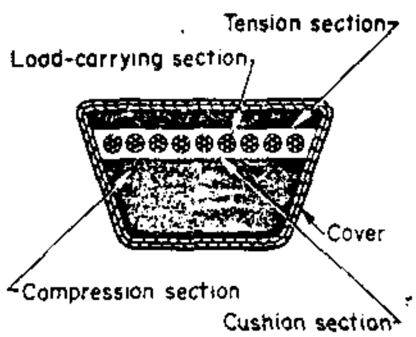

Figure 7

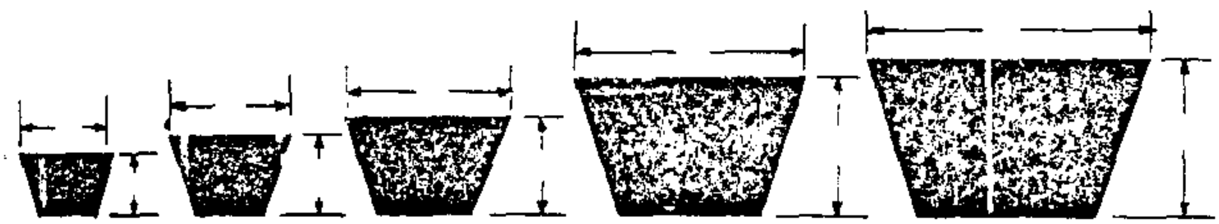

Figure 8

Different types of V-belt construction are shown in Figure 9. Two of these are worth highlighting.

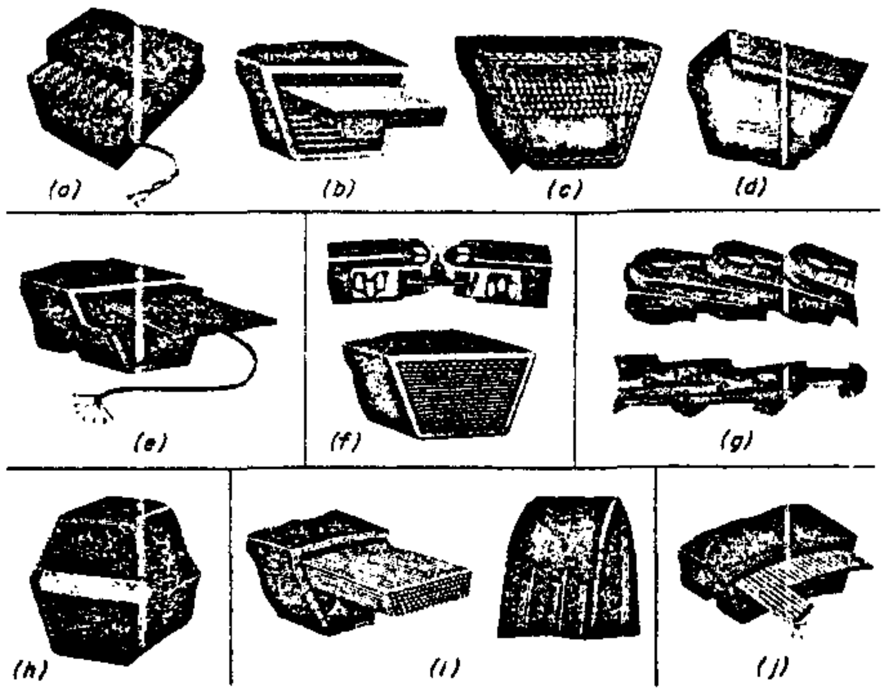

Figure 9

The belt in 9(d) is toothed or serrated on the underside. This is to help it to run cooler in high temperature applications. In "variomatic" or variable speed drive applications a belt like that in 9(j) would be used.

The pulleys or sheaves for V-belt drives are generally made from cast iron but any suitable material would serve. The grooves should have flat sides and should be deep enough so that the belt does not ride on the bottom. Figure 10 is a cross-section of a typical grooved sheave.

For proper running of a V-belt drive several things should be observed. All of the belts should be in good physical condition, not frayed, cracked or giving evidence of local wear. The belts should be riding in the grooves correctly as illustrated in Figure 11(a). No belts in the set should be appreciably longer than the others since this means that some belts are being overloaded. The belts should be tensioned properly.

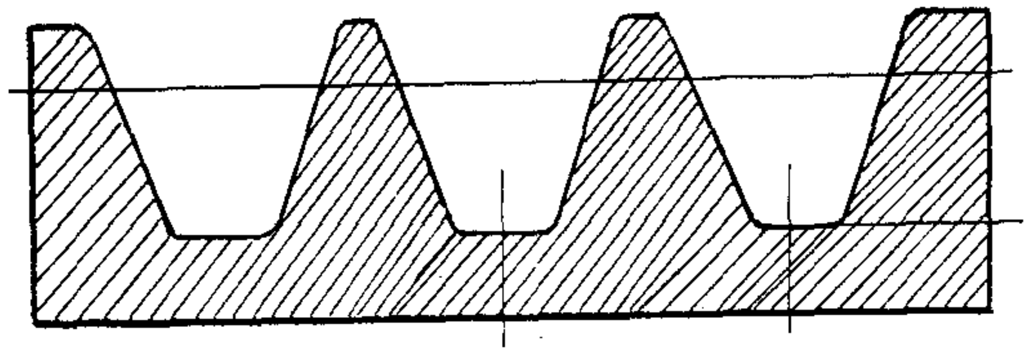

Figure 10

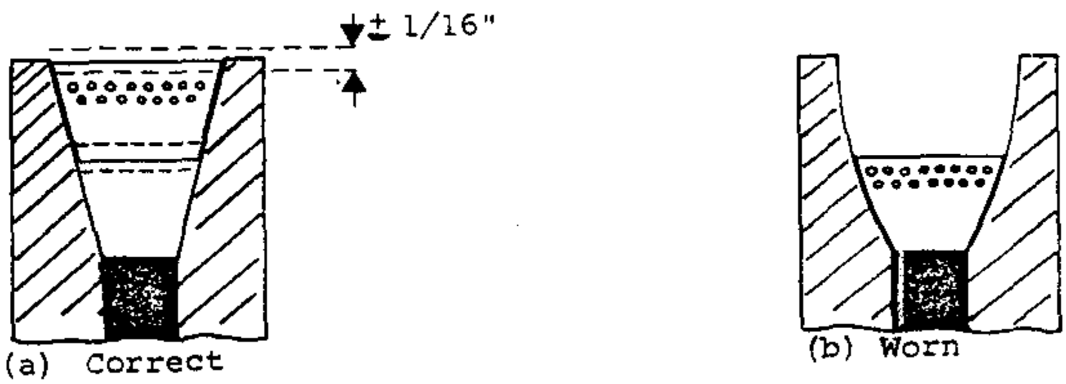

Figure 11

Problems with V-belt Drives

Some of the problems with V-belt drives can be detected early in their development by simple inspection. Preventive action can then be taken at the first appropriate time instead of waiting for a complete breakdown of the drive.

The following conditions can be observed while the drive is running. Under no circumstances should any component of the drive be touched.

If the belts are slipping, either the tension needs to be increased or the load is too great.

Check the tight side of the drive to see whether any belts appear slack. If so, then the groove may be worn, the belt may be stretched or the set was mismatched at the outset.

Look for wobble on the pulleys as this could mean a bent shaft which would require straightening.

Serious misalignment could probably be spotted by sighting along the belts, but alignment is better checked with the drive stopped. Other conditions can be seen best while drive is stopped and isolated.

Examine the belts for wear, fraying, cracking, broken cords or oil damage. Some of these things can be due to normal wear, however, they can be symptoms of other trouble particularly if they occur prematurely.

Examine the sheaves to see that they are not worn excessively nor cracked and chipped.

Immediately after stopping check the bearing temperatures. High temperatures can be due to lack of lubrication, or too tight belts.

Determine whether or not the belts are correctly tensioned. If a belt has flipped over it is generally an indication that one or more of the cords on one side are broken and therefore the belt should be replaced.

Wipe off any oil or grease on the belts using a cloth moistened with a suitable solvent. Oil or grease attack the belt material and soften it.

Handling Advice

Be careful. When turning the drive by hand remember that the inertia of the motor and driven equipment may keep the belts moving after the turning force is removed. If the drive is being rotated by pulling on the belts then hands and fingers can be pulled into the sheave accidentally. Apply the turning force by means of a lever on a sheave where possible.

Diagnosis of V-belt Failures

Condition: Belt material soft, spongy and tacky. Some peeling.

Cause: Oil or grease.

Prevention: Splash guards where possible; otherwise oil resistant belts.

****

Condition: Cover fabric ruptured at one location.

Cause: Belt pried on or some object falling into sheave groove.

Prevention: Install by moving motor.

****

Condition: Cracked or checked belts particularly on underside.

Cause: Backbending or high operating temperature.

Prevention: Check for cause of high temperature (eg, slippage) and rectify if possible to keep ambient temperature below 150°F.

****

Condition: Snapped belt.

Cause: Belt too loose causing it to snap tight on start-up.

Prevention: Maintain correct tension.

****

Condition: Worn or abraded belt sides.

Cause: Abrasive material on sheave, or misalignment.

Prevention: Keep Drive clean. Align properly.

Variable Speed Pulleys and Belts

When it is necessary to change the speed ratio between the driver and driven pulleys while the drive is in motion then a variable speed belt drive can be used.

These drives utilize a belt similar to that shown in Figure 12. A cross-section of the belt would show that it is arched. It also has ribs running across the belt which give it rigidity in that direction but still permitting flexibility in the lengthwise direction.

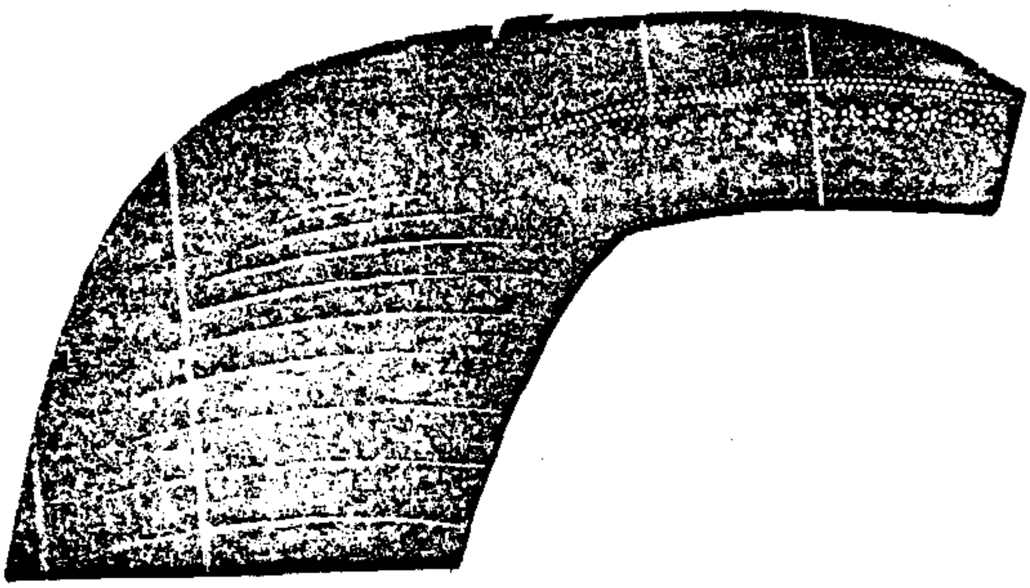

Figure 12

There are two methods of operating a variable speed drive. One is to use a pulley on which one flange can move against a spring while the other is fixed to the shaft as illustrated in Figure 13. As th centre distance between the pulleys is varied by moving the motor back and forth the speed ratio of the drive is varied. The other method is to use two pulleys on one of which the flanges open at the same rate as they close on the other. The movement of the flanges would be controlled by both the speed and the load.

For good efficient operation of these types of drives the pulleys need to be maintained in good condition with the working faces smooth. The belts will require inspection for defects as described for conventional V-belts. If the drive is working correctly the tension on the belt should be maintained automatically.

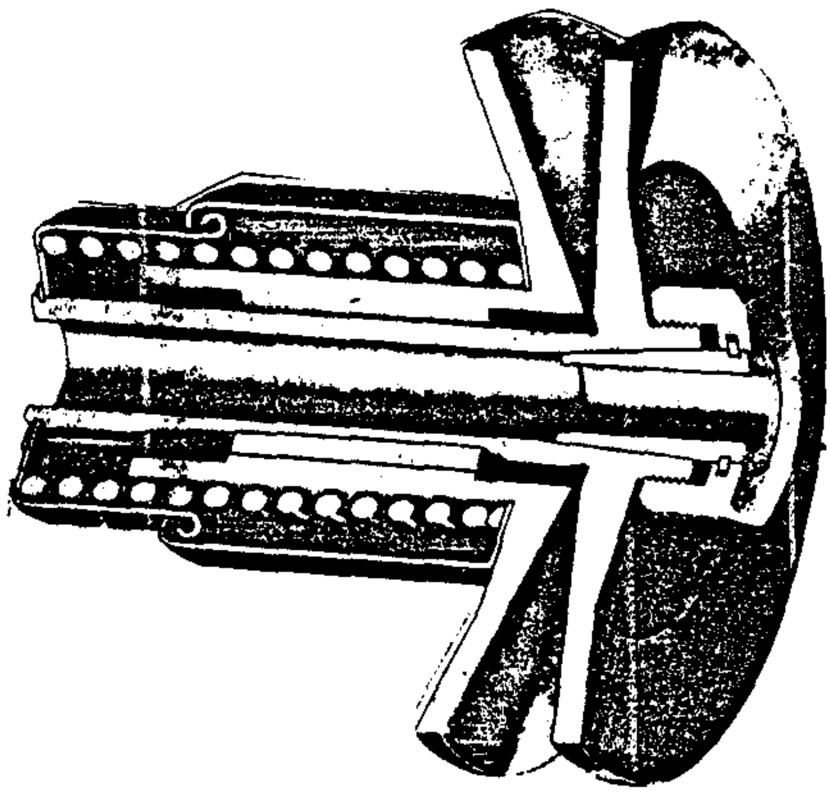

Figure 13

Toothed Belt Drives

There exist applications where the drive must be positive with no slippage, such as timing of valve operations. In these instances it is possible to use gear or chain drives, however, there exists a range of toothed belts which run in pulleys, resembling gears. For relatively low load situations these drives provide an inexpensive and quiet solution with adequate precision.

The construction of the belts is similar to that of V-belts except that the cords should resist stretching to maintain the rotational relationship betweent he two pulleys and the teeth should be able to withstand wear. For example, one timing belt for an overhead camshaft drive has glass fibre cords and nylon teeth. For lighter service other cords would be suitable as would rubber or neoprene cover and teeth.

The belts are available in several widths and lengths and with a range of pitches of the teeth. The pulleys must have teeth of the same pitch as the belt and one or both pulleys should be flanged to locate the belt and prevent it travelling off the edges.

Figure 14 illustrates a typical toothed belt drive.

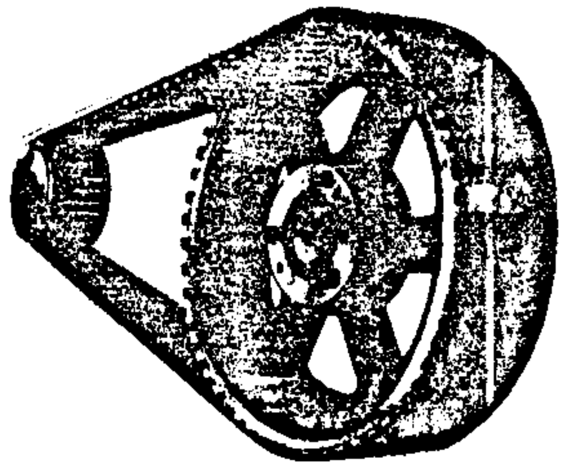

Figure 14

Idlers

Without a means for moving one of the components to create tension in the belt it may be necessary to use another pulley called an idler to act as a tensioning device. It is called an idler because it drives nothing.

The idler pulley may be placed on the outside of the belt or the inside and may be on the tight or slack sides.

An outside idler has the advantage of increasing the arc of contact but there is a limit to the amount of take-up possible since it would not be good to have the belts touching. It can be seen in Figure 15(a) and (b) that this possibility exists. Another disadvantage of the outside idler is the reverse bend which it puts on the belt. This reduces the life of the belt since it is generally not designed to take the tension created at the inside surface by a reverse bend.

The arc of contact is decreased by an inside idler but the amount of take-up virtually unlimited as shown in Figure 15(c) and (d).

The best location for an idler is on the slack side of the drive, however it must be on the tight side it should be close to the driven pully.

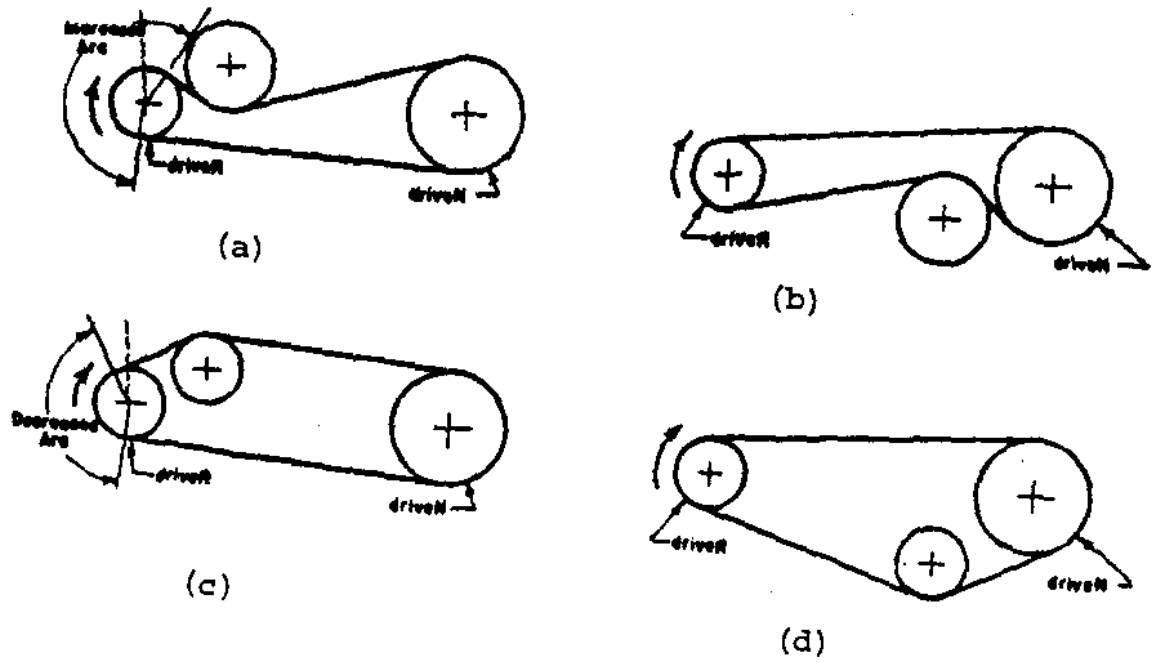

Figure 15

H. Timmins
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ASSIGNMENT

1. Why are crowned pulleys used for flat belts?

2. List four items that should be checked on a belt drive for proper operations.

3. Describe one type of variable speed drive pulley.

4. What is the purpose of an idler on a belt system?

Mechanical Equipment - Course 230.1

CHAIN DRIVES

Chain drives are used where the working environment is not suitable for belts, such as high temperatures, corrosive atmosphere, very high loads or where a positive drive with no slip can be tolerated. Gear drives could also be used in these situations however, a gear drive would be more expensive than an equivalent chain drive, generally. For example, consider the extra expense involved in using gears to drive a bicycle.

Chain and Sprockets

Probably the chain with which most people are familiar is the roller chain which is used on bicycles. The basic unit of a roller chain is the roller link as shown assembled in Figure 1. It is made up of pieces illustrated in Figure 2. Two rollers (a) are mounted on bushings (b) which are riveted into link plates (c). A chain is formed when the individual roller links are fastened together by the link pins (d) and the link plates (e). A section of single strand chain is shown in Figure 3(a) and a multistrand chain in Figure 3(b).

Roller Link

Figure 1

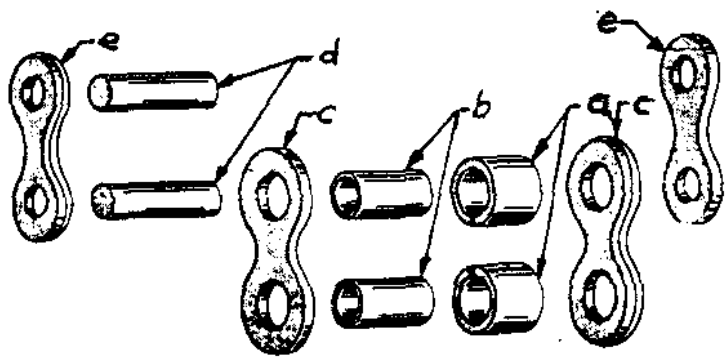
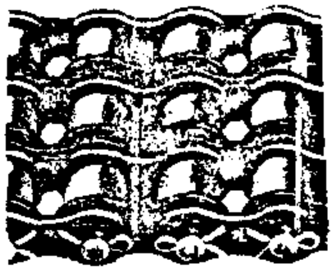


Figure 2

(a)



(b)

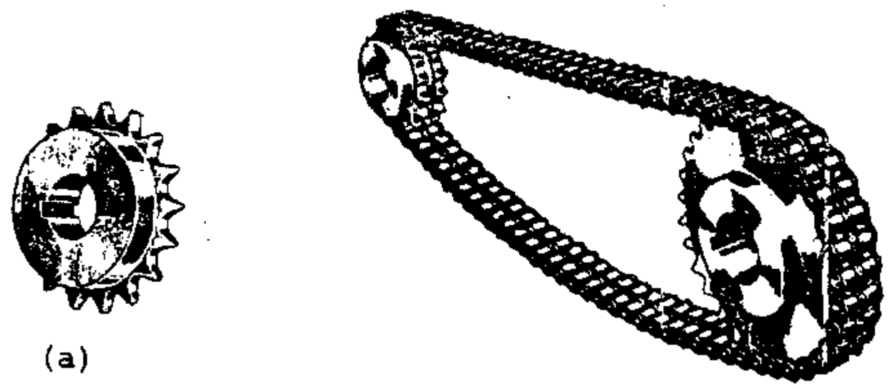
Figure 3

The pitch of the chain is the centre to centre distance between the link plate holes, therefore there is a close tolerance maintained on this dimension. The link pins, the bushings and the rollers are case hardened and ground for wear strength and precision. The pins may be riveted into the link plates or else they may be held in place by cotter pins as shown in Figure 3 (a) and (b).

Another style of chain used frequently as a drive chain on construction machinery is the offset side bar chain, Figure 4. The ends of the pins may be riveted instead of having cotter pins as in the illustration. Some offset chains are made without the rollers.

Figure 4

Sprockets for roller chains and offset side bar chains can be grouped according to their style of construction. There are steel plate sprockets without hubs, with hubs on one side or with hubs on both sides. They may be obtained with removable taper lock bushings which provide a quick and easy way to install and replace them or they can be keyed directly to the shaft. There are proper sprockets for single or multistrand chain drives. For proper meshing, the pitch of the sprocket teeth must match that of the chain, and with multistrand drives the distance between adjacent sets of teeth must also be the same as the chain. Figure 5(a) shows a sprocket with a hub on one side and Figure 5(b) is an illustration of a double strand drive.



(a) (b)

Figure 5

It is recommended that the chain wrap on the smaller sprocket be at least 120°. The slack should be on the lower side to prevent rubbing between the top and bottom. If it is necessary to keep the drive tight, and there is no provision for adjusting the centre to centre distance then an idler will be used. Figure 6 shows an installation using idlers.

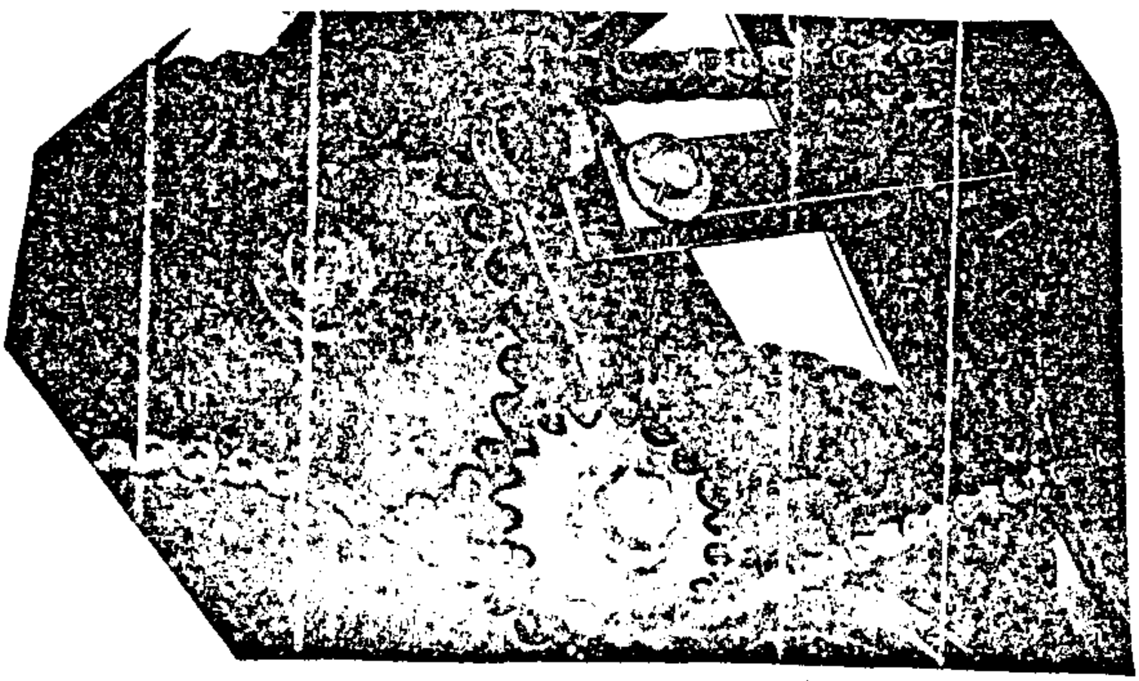
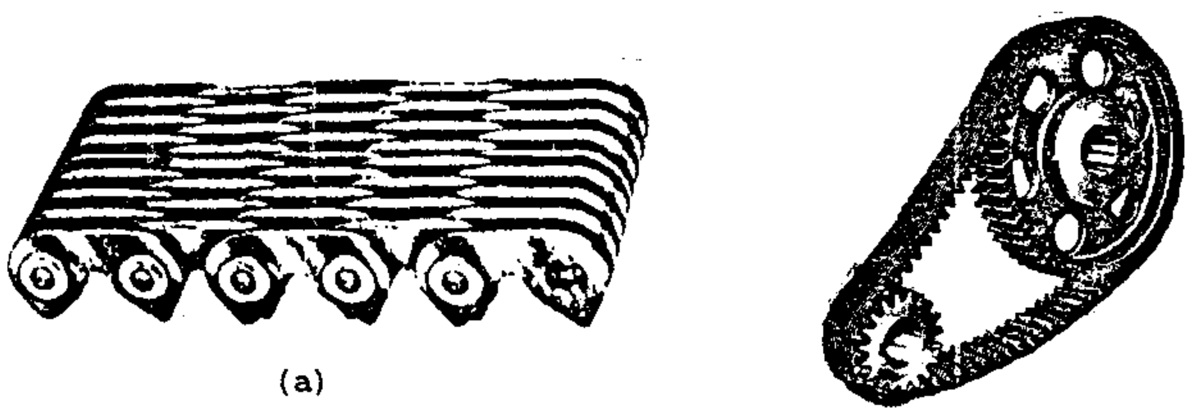


Figure 6

The silent or inverted tooth chain, shown in Figure 7(a) and (b) is a high speed chain used mainly for prime mover take-off drives, such as power shovels, machine tools and pumps.

These chains are made of a series of flat plates each of which have two projections or teeth, one at each end. The outer face of the tooth is ground to a given angle to work against the faces of the sprocket.

The chain passes over the face of a gear-like sprocket like a belt. The sprocket teeth do not protrude through the chain. In operation there is no sliding action between the chain and the sprocket resulting in a smooth, quiet action. The chain is held on the sprocket either by a row of centre guide plates, fitting a groove in the sprocket, Figure 7(b), or by side guide plates on each side of the chain straddling the sprocket.



(a) (b)

Figure 7

Maintenance

While the drive is running, watch the chain for excessive vibration or rubbing against adjacent parts. If the chain is running close to the tips of the teeth of the large sprocket, the chain should be replaced.

With the drive shut down and suitably tagged according to the safety regulations, the chain and sprockets should be examined for uneven or excessive wear on the sides of the sprocket teeth or the inside of the roller link plates.

Check the tension of the chain by trying to lift it away from the large sprocket.

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ASSIGNMENT

1. Why might you use a chain drive instead of a belt drive?

2. What are some of the items that should be checked to ensure correct chain drive operations?

Mechanical Equipment - Course 230.1

GEARS AND GEARING

Whenever rotary motion must be transferred from one shaft to another, gears should be considered. A gear is a toothed wheel used to transmit positive and uniform rotary motion from one shaft to another.

Gear Types

Gears can be classified according to the position which the teeth occupy with respect to the axis of rotation of the gear body. The more common types of gears in use are Spur gears, Bevel gears, Helical gears, Herringbone gears, Spiral gears, Worm gears and Planetary gears. Specially shaped gears such as eleptical gears and square gears are designed for a special purpose and are not used nearly as much as the first group mentioned.

Spur gears as shown in Figure 1 are gears having straight teeth that are cut parallel with the axis of rotation of the gear body. Spur gears are used to transmit motion from one shaft to another parallel shaft.

Helical gears shown in Figure 2 are gears having their teeth cut on a cylinder and at an angle with the axis of rotation of the gear body. These are sometimes called spiral gears by mistake.

Herringbone gears shown in Figure 3 are gears having helical teeth which diverge from the centre of the gear towards the sides of the gear body.

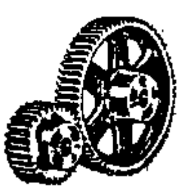


Figure 1

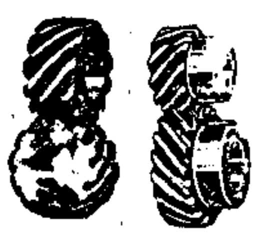


Figure 2

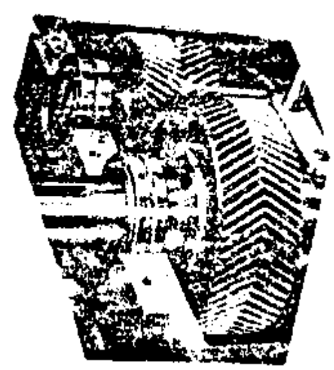


Figure 3

Spiral gears shown in Figure 4 are gears with teeth cut on a conical surface so that they curve continually away from or towards the apex of the cone upon which they are cut. These gears closely resemble bevel gears and are frequently called spiral bevel gears.

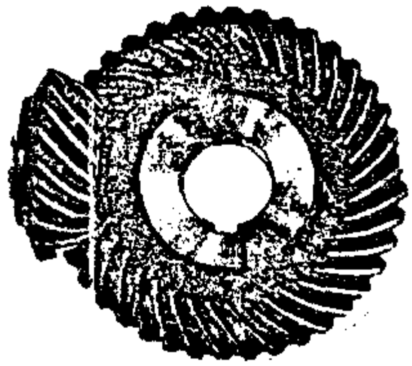


Figure 4

Bevel gears shown in Figure 5 are gears with their teeth cut on a conical surface so that the teeth radiate from the apex of the cone in a straight line. Miter gears are mating bevel gears having the same number of teeth and whose pitch cone angle is 45°. Miter gears are shown in Figure 5(a).

Figure 5

Figure 5(a)

A Worm gear, shown in Figure 6, is a gear having the the teeth cut angular to the axis of rotation and radially with the worm face. A worm gear is driven by a Worm, seen in Figure 6(a), which resembles a large screw thread.

Figure 6

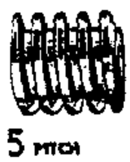


Figure 6(a)

Gear tooth profiles are commonly involute and this system has been developed from a basic rack. The rack as shown in Figure 7 and Figure 7(a) have straight sided teeth inclined at a standard pressue angle, usually 14 1/2° or 20°. This is an interchangeable system because all gears made to operate with this basic rack will operate with each other.

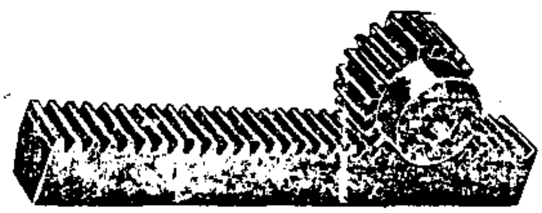


Figure 7

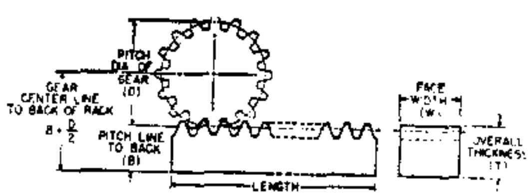


Figure 7(a)

To pressure angle is, as the name implies, the angle at which the tooth pressure is applied and distributed. In Figure 8, "Z" indicates the pressure angle.

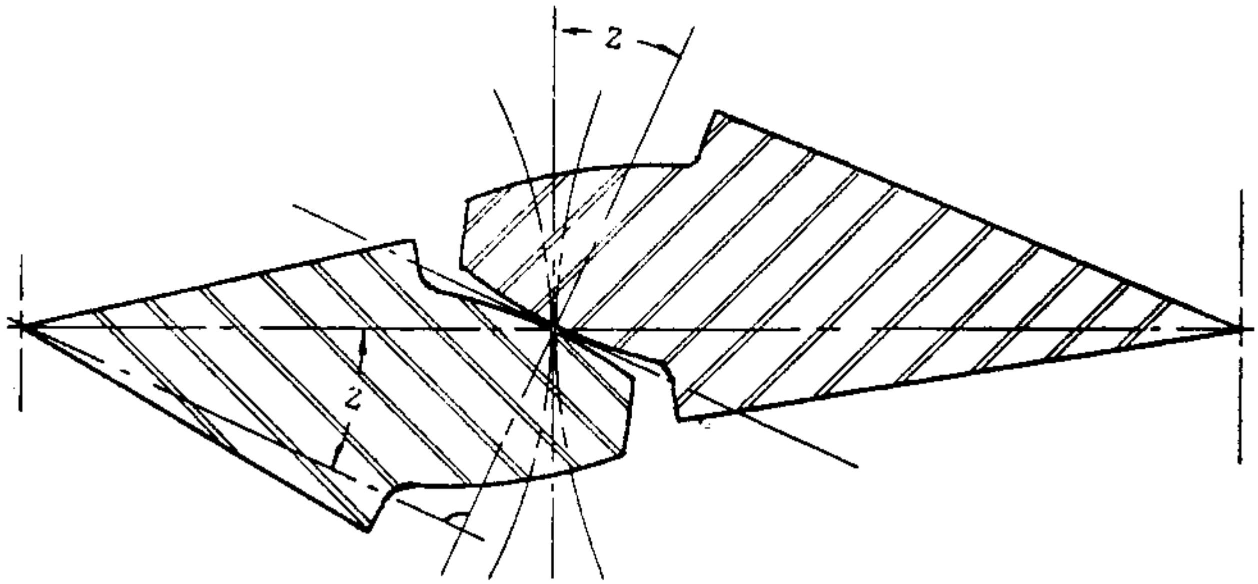


Figure 8

The 14 1/2° pressure angle gear has been used many years and remains useful for replacement or duplicate gears. The 20° pressure angle and more recently the 25° pressure angle have become standard for industry and the gear manufacturers. The gear designer's claim that the higher pressure angle (25°) permit the gears and the pinions to have fewer teeth without undercutting of the teeth.

Mating gears must have the same pressure angle or else they will not mesh properly.

Spur Gears

The gears shown in Figure 9 and the basic rack shown in Figure 10 give some of the terms used in calculating gear and tooth sizes.

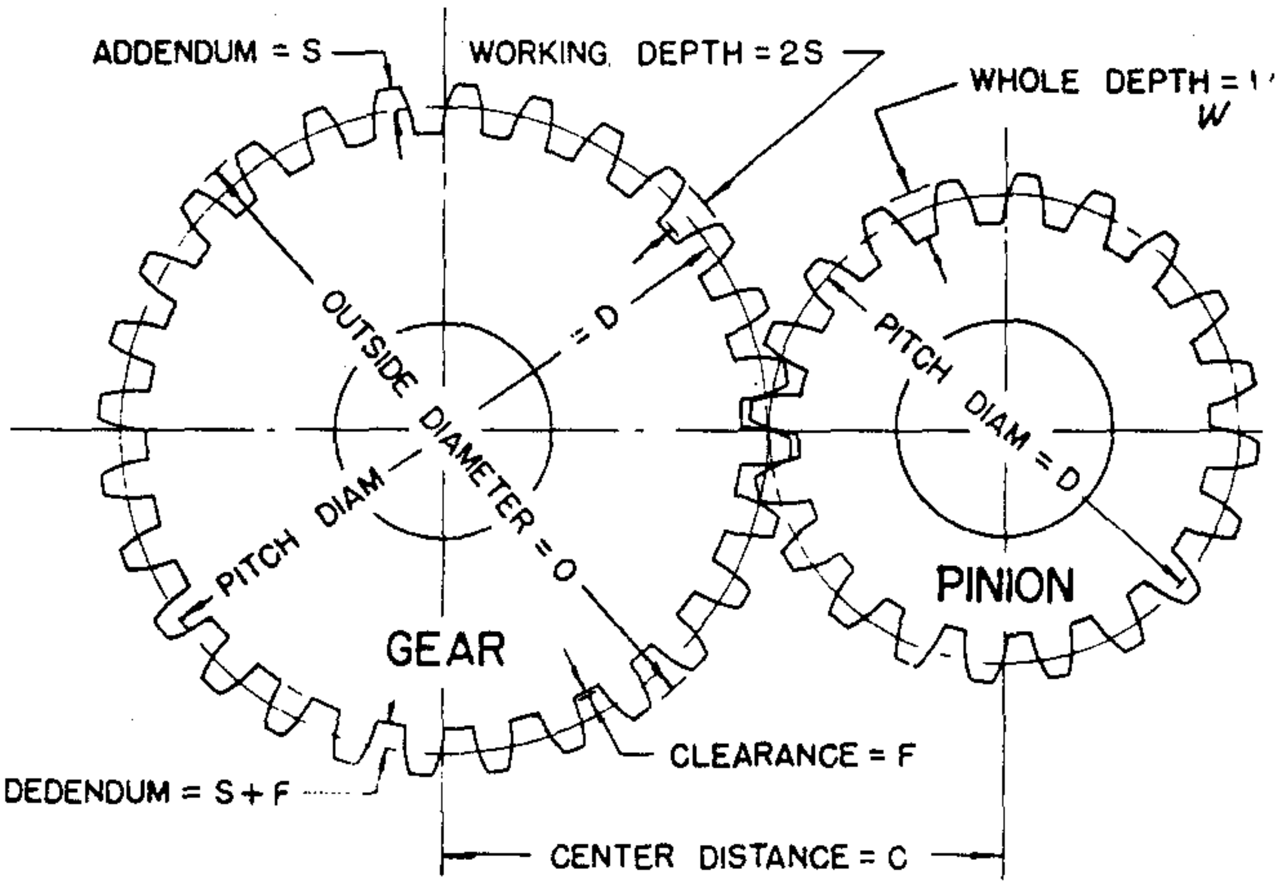
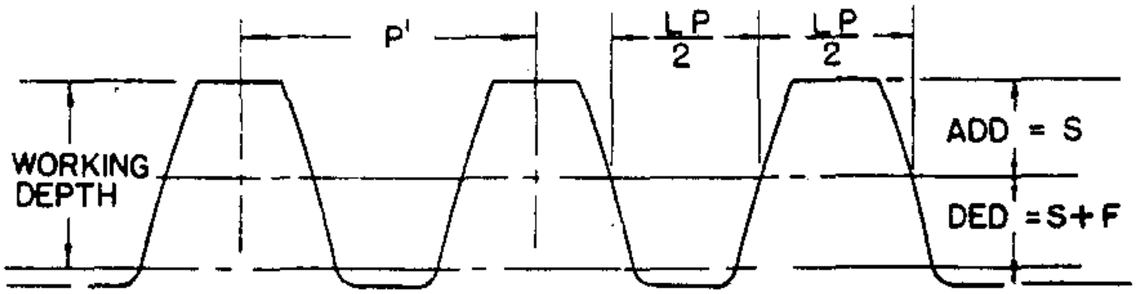

ADDENDUM = S
WORKING DEPTH = 2S
WHOLE DEPTH = W
OUTSIDE DIAMETER = O
= D
PITCH DIAM
PITCH DIAM = D
GEAR
PINION
CLEARANCE = F
DEDENDUM = S + F
CENTER DISTANCE = C

Figure 9



The Diametral Pitch (Pitch or P.) indicates the size of the tooth.

Addendum = $\frac{1}{P}$ Dedendum = $\frac{1.157}{P}$ Linear Pitch = $\frac{\Delta}{P}$

Working Depth = $\frac{2}{P}$ Whole Depth = $\frac{2.157}{P}$

Linear or Circular Pitch = P'

Rack Tooth

Figure 10

Pitch Circle: The pitch circle is an imaginary circle located about halfway down the gear tooth where both gears contact each other. Usually when a gear is spoken of as so many inches in diameter this is assumed to be the diameter of the pitch circle. The Pitch Diameter is the diameter of the pitch circle. See Figure 9.

Diametral Pitch (D.P.) is often abbreviated to "pitch" and the term pitch with respect to gears means "frequency". The pitch of a gear tooth is the number of teeth on the gear for each inch of pitch diameter. It is defined by the following equation:

$$P = \frac{N}{D}$$

Where P = Pitch
N = Number of Teeth
D = Pitch Diameter

Two gears must be of the same Diametral Pitch before they can mesh properly with each other, for example, two 8 D.P. gears will mesh properly but an 8 D.P. will not mesh with a 10 D.P.

Circular Pitch which is never abbreviated, is the sum of the tooth space plus the tooth width measured on the pitch circle, see Figure 9 and note that the circular pitch is a linear dimension and so must be measured in inches.

Addendum is the distance the tooth extends outside the pitch circle.

Dedendum is the distance from the pitch circle to the bottom of the gear bottom land and is equal to the addendum plus an allowance for clearance, or more simply, twice the addendum plust clearance.

Centre Distance is the sum of half the Pitch Diameter of one gear plus half the Pitch Diameter of the other gear, see Figure 9. The centre distance is very important because if the gears are correctly designed and manufactured they will mesh and operate freely and quietly, with the correct amount of backlash, at exactly that distance.

$$\text{eg, Standard C.D.} = \frac{\text{Pinion P.D.} + \text{gear P.D.}}{2}$$

It is most important that two mating gears must be of the same Diametral Pitch and the same Pressure Angle or they will not mesh properly.

Bevel and Miter Gears

Straight bevel gears can be used on shafts at any angle, though 90° is the most usual. Bevel and miter gears will provide smooth, quiet operation and long life when they are properly mounted and lubricated, however there are several important conditions to be met when assembling bevel gears:

Most standard bevel and ALL miter gears must be mounted on shafts that are at right angles (90°) to each other and as all bevel and miter gears develop radial and axial thrust loads these loads must be accommodated by use of suitable bearings. Figure 11 gives the various thrust directions for various combinations of gear sets.

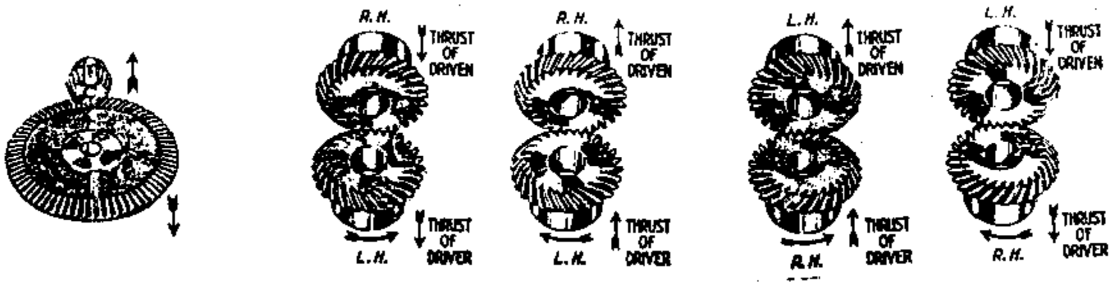


Figure 11

Helical Gears

Helical gears are used to transmit power or motion between non-intersecting shafts. The teeth of a helical gear may be cut with either a right or left hand helix. Two gears with teeth cut on the opposite hand operate on parallel shafts in much the same manner as spur gears. Figure 12 shows an opposite hand gear drive. This type of gear is recommened for high-speed and high torque applications. They will provide a stronger, smoother and quieter running drive than spur gears of the same pitch and size.

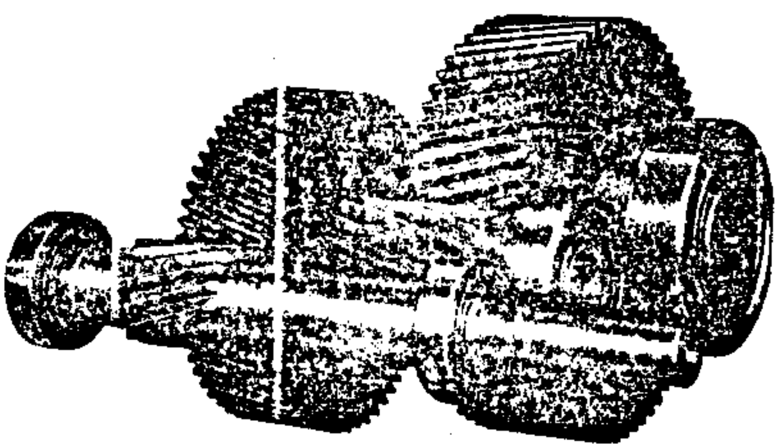


Figure 12

Figure 12(a)

Helical gears of the same hand as seen in Figure 12(a) operate on shafts that are 90° to each other. The load capacity of this type of drive is limited due to the reduction of tooth bearing area. Right angle shaft drives should be used for the transmisstion of motion only.

The rotation of helical gears generate an end thrust load and suitable bearings must be provided.

Herringbone Gears

Herringbone gears or double helical gears are used in a parallelled shaft transmission especially when a smooth continuous action is essential. The gear shown in Figure 13 is used in a speed increasing unit. This type of gear is used in marine reduction gears, in speed increasing units and in various tranmissions in connection with steam turbine and electric motor drives. Because of the double helix angle and the gear drive is self centering and large thrust bearings are not needed to take end thrust loads.

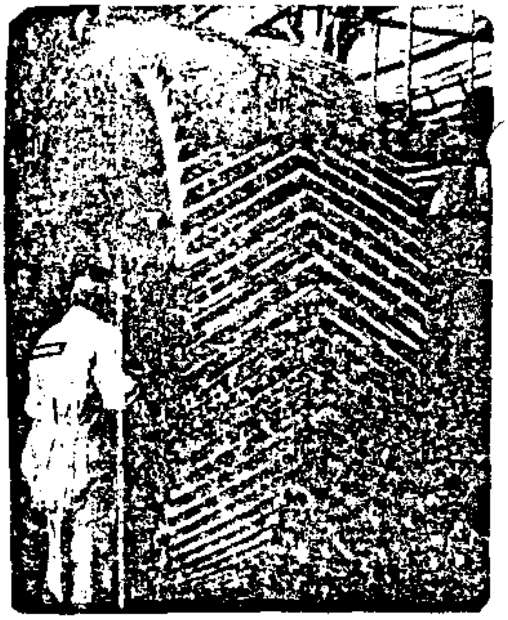


Figure 13

The failure of herringbone gears is rarely due to tooth breakage but is due to excess tooth wear or to sub-surface failure such as pitting or spalling.

Worm Gear Drive

A worm gear is essentially an endless screw wrapped around a cylinder. It has already been stated that a worm gear is driven by a worm. Figure 14 is an illustration of both a worm gear and a worm. The worm is manufactured with single, double, triple or quadruple starts on the worm screw. The worm gear and worm screw is used to convert rotary motion at high speed to rotary motion at low speed. The ratio or speed reduction equals the number of teeth cut on the worm wheel divided by the number of threads or starts on the worm screw. The ratios may range from 5:1 to 100:1.

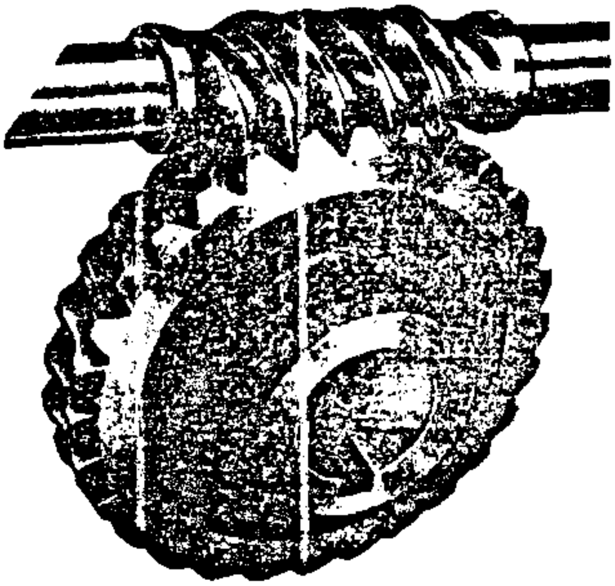


Figure 14

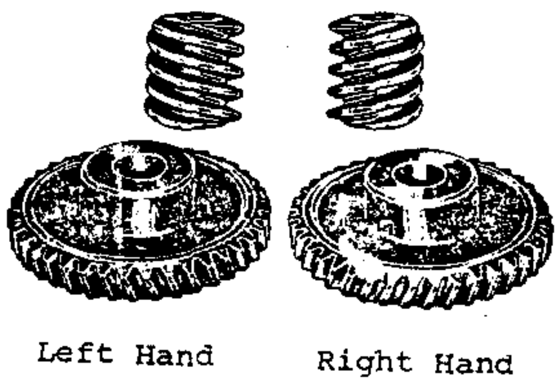


Figure 15

The teeth of the worm gear and the threads of the worm may be cut left hand or right hand and are not interchangeable with each other. To tell which is which, stand them upright as in Figure 15. The threads on the left hand cut lean to the left and the threads on the right hand cut lean to the right.

Backlash

Backlash is lost motion between mating gear teeth. It is a function of the actual centre distance of the two gears and the actual tooth thickness of each other. Backlash is caused by the difference of the thickness of the gear tooth and the space into which it meshes. The purpose of backlash or clearance is to prevent the gears from jamming together and making contact with both sides of the gear tooth at the same time. Lack of backlash will cause noise, overloading, overheating, rapid wear and perhaps the seizing and breaking of the gear teeth. Too much clearance or backlash will also give noisy gears and overloading of the gear tooth because of lack of contact area which will result in rapid wear of the gear teeth.

The proper clearance required for the backlash is built into the gear tooth profile when the gear is designed and cut. The only control a maintainer has over the backlash is the centre to centre distance in the case of parallel shafts and the Mounting Distance in the case of bevel gear sets. Shafts that should be parallel must be parallel otherwise clearance at one edge of the gear tooth will be different from the clearance at the other edge of the gear tooth.

These are cases where backlash must be eliminated. In applications of "zero backlash" the gears and mountings have been designed for this consideration and anti-backlash devices have been provided.

The following is a table of approximate backlash figures for various pitch gears. The correct backlash allowance for a set of gears can be found in the gear manufacturer's handbook.

Diametral Pitch	Backlash	Diametral Pitch	Backlash
3	.013"	8-9	.005"
4	.010"	10-13	.004"
5	.008"	14-32	.003"
6	.007"	33-64	.0025
7	.006"		

An increase or decrease in the center distance will cause an increase or decrease in backlash. The approximate relationship is that for each .001" change in center distance the backlash will change approximately .0007".

The minimum backlash should be at least enough to accommodate a lubricating film of oil on the teeth.

Tooth Wear

During the initial run-in period of a set of gears the action of rolling and sliding will smooth the surface of the gear teeth. With proper operating conditions and lubrication the teeth will show little signs of wear and normal wear will occur. Initial pitting of the gear teeth may continue until the local high spots are worn down at which time the gears will perform satisfactorily. Progressive pitting may occur after the initial run-in, this is indicated by visual inspection where it will be seen that pieces of metal have been torn from the gear tooth leaving pits in the surface.

Abrasion is seen as fine scratches extending from the root to the tip of the teeth. Abrasion is caused by gritty material in the air or in the lubricating oil. Abrasion can be controlled by frequent cleaning of the gear teeth and frequent changing of the lube oil.

Galling removes metal from the tooth surface and is due to the failure of the oil film to carry the load. Sometimes the metal is dragged over the tooth edge, creating a feather--edged appearance. Advance galling is indicated by a ridge developing at the pitch line of the driven teeth and a groove at the pitch line of the driving teeth.

Spalling is the condition when large chips or flakes break away from the gear tooth. This condition starts at the root of the tooth. Often small flakes of metal will be visible in the coil.

H. Timmins
L. Laplante

ASSIGNMENT

1. List five types of gears.

2. What is meant by the pressure angle on a gear?

3. Define the following terms:

   (a) Pitch
   (b) Pitch Circle
   (c) Diametral Pitch

4. What two conditions must be met in order that two mating gears mesh?

5. Define what is meant by backlash in a gear.

6. Is backlash necessary in gear operations?

7. Describe three types of wear that will take place on a gear tooth.

Mechanical Equipment - Course 230.1

GAS TURBINES

Two of our nuclear plants, Pickering G.S. and Bruce G.S. rely on open cycle gas turbines for standby power.

The Pickering G.S. unit was previously discussed in Level 3 Mechanical Equipment, are modified jet engines, with a free power turbine added to drive the generator. There are six units each rated at 5 MV, capable of peaking to 7 MW.

The purpose of this lesson is to discuss in more detail than was given in Level 3 the theory of operation of gas turbines, and some of the operating characteristics of these particular units.

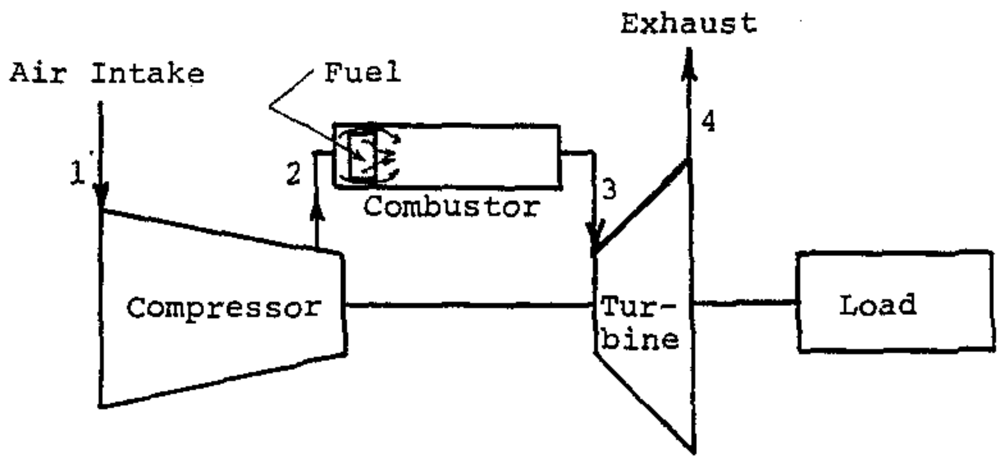
Gas Turbine Cycle

The theoretical cycle for the gas turbine is the Brayton cycle illustrated in Figure 1. It consists of isentropic compression, constant pressure heat additions, isentropic expansion, and constant pressure heat rejection.

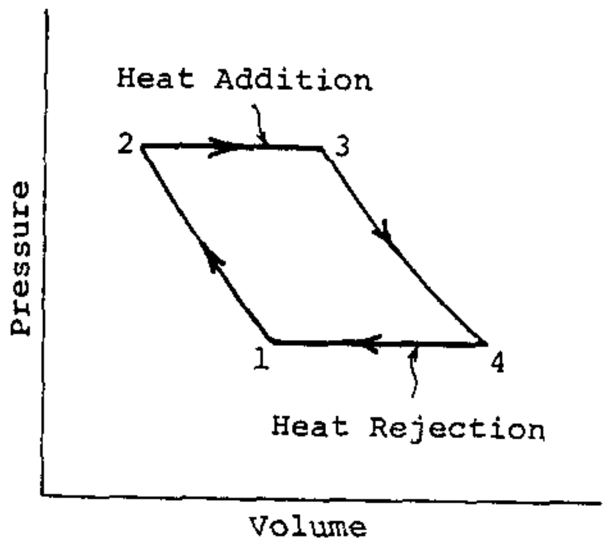
At this point it is well to distinguish between the open and closed cycle. The flow diagram of Figure 1 shows gases discharged to atmosphere from the turbine, and air entering the compressor from atmosphere. This is the open cycle. In the Brayton cycle diagram the point of exhaust (4) is connected to the point of intake (1) by a constant pressure process indicating that the gases leaving the turbine are cooled and then returned to the system. This is the closed cycle and requires an expensive heat exchanger between the turbine exhaust and the compressor intake and large amounts of air or water as the cooling medium.

Basic equipment for a simple gas turbine system includes a compressor, combustor, and turbine. A rotating compressor draws in atmospheric air, pressurizes it, and forces it into the combustor in a steady flow.

Fuel injected into the air burns, raising the temperature of the mixture of air and combustion products. This high energy mixture flows through the turbine, dropping its temperature and pressure as it does work on the moving blades. It leaves the turbine at high temperature and atmospheric pressure. The turbine drives the compressor and an external load such as a generator. The turbines at Pickering illustrated in Figure 2 consist of a ten stage axial flow



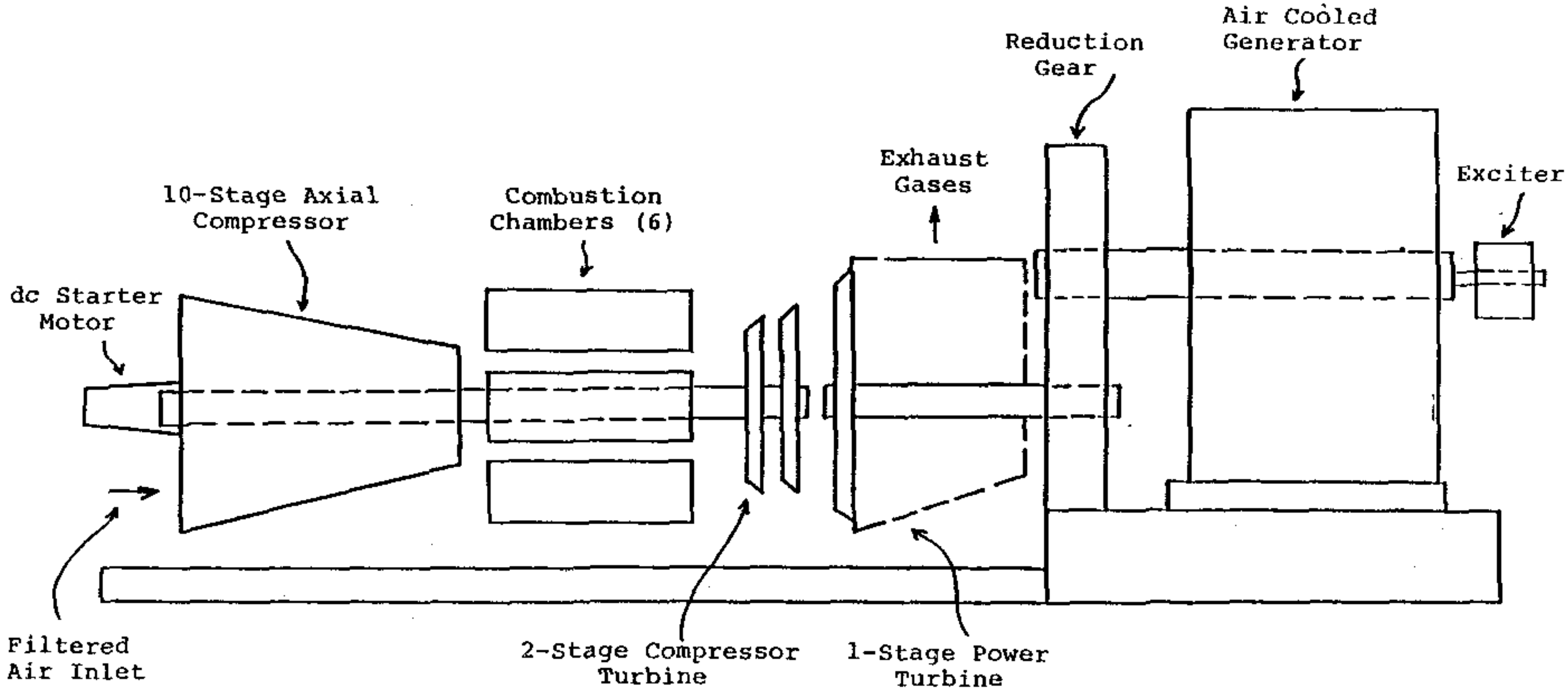
(a) Open Cycle Gas Turbine



(b) Brayton Cycle

Gas Turbine Cycle

Figure 1



Gas Turbine Generator Basic Components

Figure 1

compressor, six combustors, two stage compressor turbines and a single stage power turbine. The gas turbine consists of two shafts, one between the compressor and the compressor turbine and one between the power turbine and gear reducer driving the generator.

Compressor

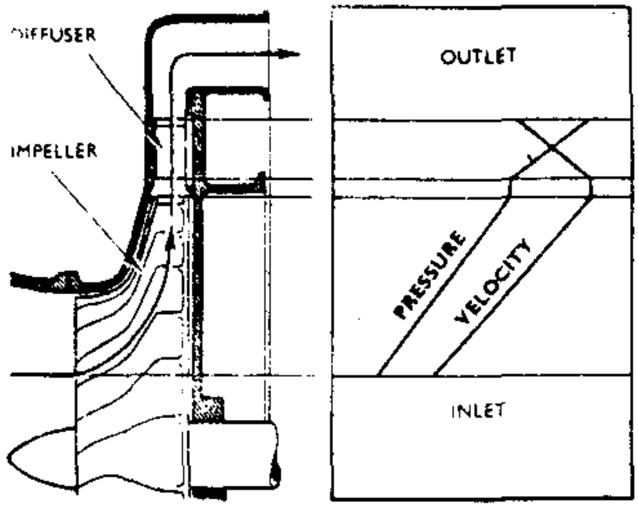
Efficient compression of a large volume flow of air is the key to a successful gas turbine cycle. This is achieved in two types of compression, the centrifugal (Figure 3) and the axial flow (Figure 4). The centrifugal compressor is a single or two-stage unit employing an impeller to accelerate the air and a diffuser to produce the required pressure rise. The axial flow compressor is a multi-stage unit employing alternating rows of rotating and stationing blades, to accelerate and diffuse the air until the required pressure rise is obtained.

With regard to the advantages and disadvantages of the two types, the centrifugal is usually more robust than the axial and is also easier to develop and manufacture. The axial, however, consumes far more air than a centrifugal of the same frontal area and can also be designed for high pressure ratios much easier. With the higher pressure ratios there is a higher engine efficiency due to an improved specific fuel consumption and thrust. The Pickering gas turbines use 10 stage axial flow compressors.

Each stage of a multi-stage compressor possesses certain air flow characteristics that are dissimilar from the adjacent stage. Therefore to design a workable and efficient compressor the characteristics of each stage must be matched. This is relatively simple to do for one set of conditions (design mass flow, pressure ratio and rotational speed) but it becomes more difficult if reasonable matching is to be retained when the compressor is operating over a wide range of conditions.

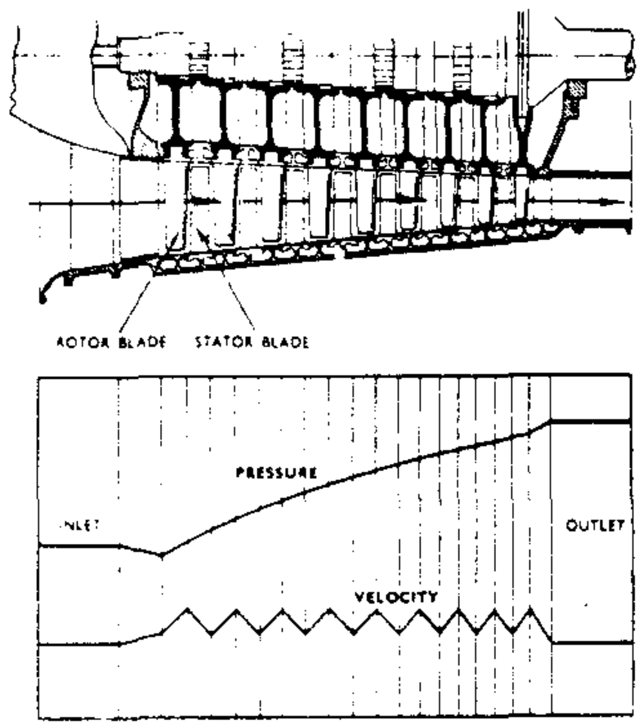
Outside the design conditions the flow around the blades tend to degenerate into violent turbulence and the smooth pattern of flow is destroyed. This condition is commonly referred to as stall or surge. A stall may affect only one stage or a group of stages, whilst a compressor surge generally refers to a complete flow breakdown through the compressor.

At low engine speeds a slight degree of blade stalling invariably occurs in the front stages of the compressor. This condition is not harmful or noticeable on engine operation. A more severe compressor stall is indicated by a rise in turbine gas temperature, vibration or coughing of the compressor. A surge is indicated by a bang of varying severity from the engine and a rise in turbine gas temperature. The rate of airflow and pressure ratio at which a surge occurs is termed the surge point. A line which gives all the surge points called the surge line (Figure 5) defines the minimum stable airflow which can be obtained at any rotative speed.



Pressure and Velocity Changes Through a Centrifugal Compressor

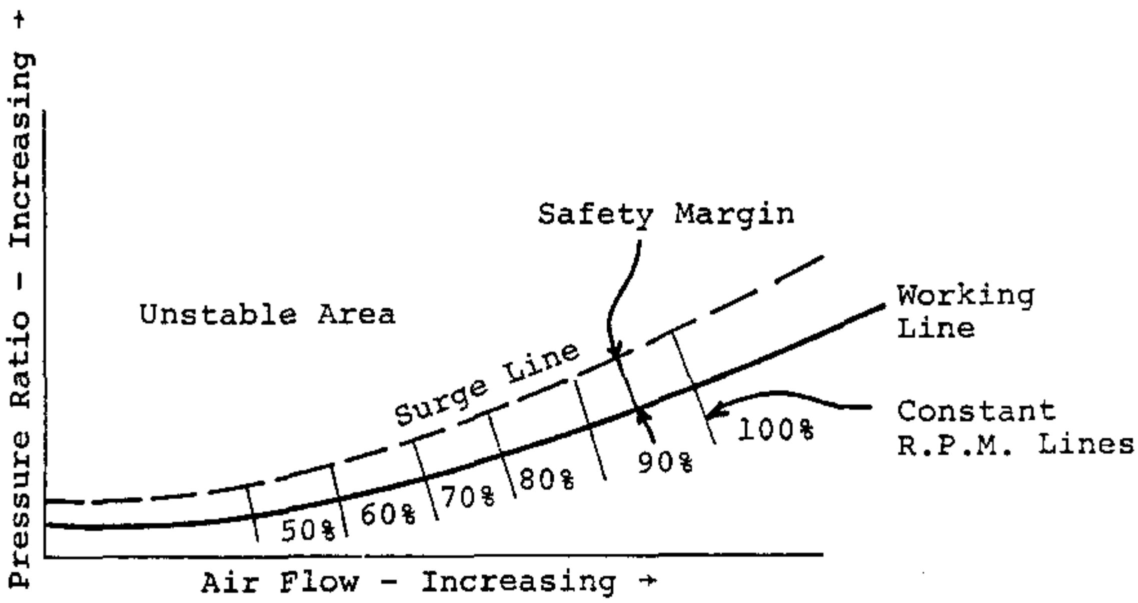
Figure 3



Pressure and Velocity Changes Through an Axial Compressor

Figure 4

A compressor is designed to have a good safety margin between the airflow and the compression ratio at which it will normally be operated and the airflow and the compression ratio at which a surge will occur.



Limits of Stable Air Flow

Figure 5

Combustion Chambers

The combustion has the difficult task of burning large quantities of fuel with extensive volumes of air, supplied by the compressor, and releasing the heat in such a manner that that the air is expanded and accelerated to give a smooth stream of uniformly heated gas at all conditions required by the turbine. This task must be accomplished with the minimum loss in pressure and with the maximum heat release for the limited space availab'e.

In a typical gas turbine engine such as that used at Pickering the combustion chamber consists of a two piece steel outer casing enclosing a flame tube fabricated from nickel-chrome sheet. An annular cooling air space is provided between the flame tube and the outer casing. Holes in the flame tube head admit the required amount of air for primary combustion and swirl vane incorporated in a diffuser impart a whirl velocity to the air entering around the burner.

Additional air is admitted through holes along the flame tube to cool the inner walls and dilute the products of combustion before entering the turbine. Figure 6 and Figure 7 illustrate combustion chambers similar to that found in the Pickering unit. Figure 7 illustrates the airflow pattern in a combustion chamber.

The liquid fuel system at Pickering consists of six duplex burners which deliver fuel to the combustion chamber in a finely atomized form. Each burner has two inlet passages, one a primary passage for low fuel flows (starting and idling) and the other a main passage for large flows at normal operating conditions.

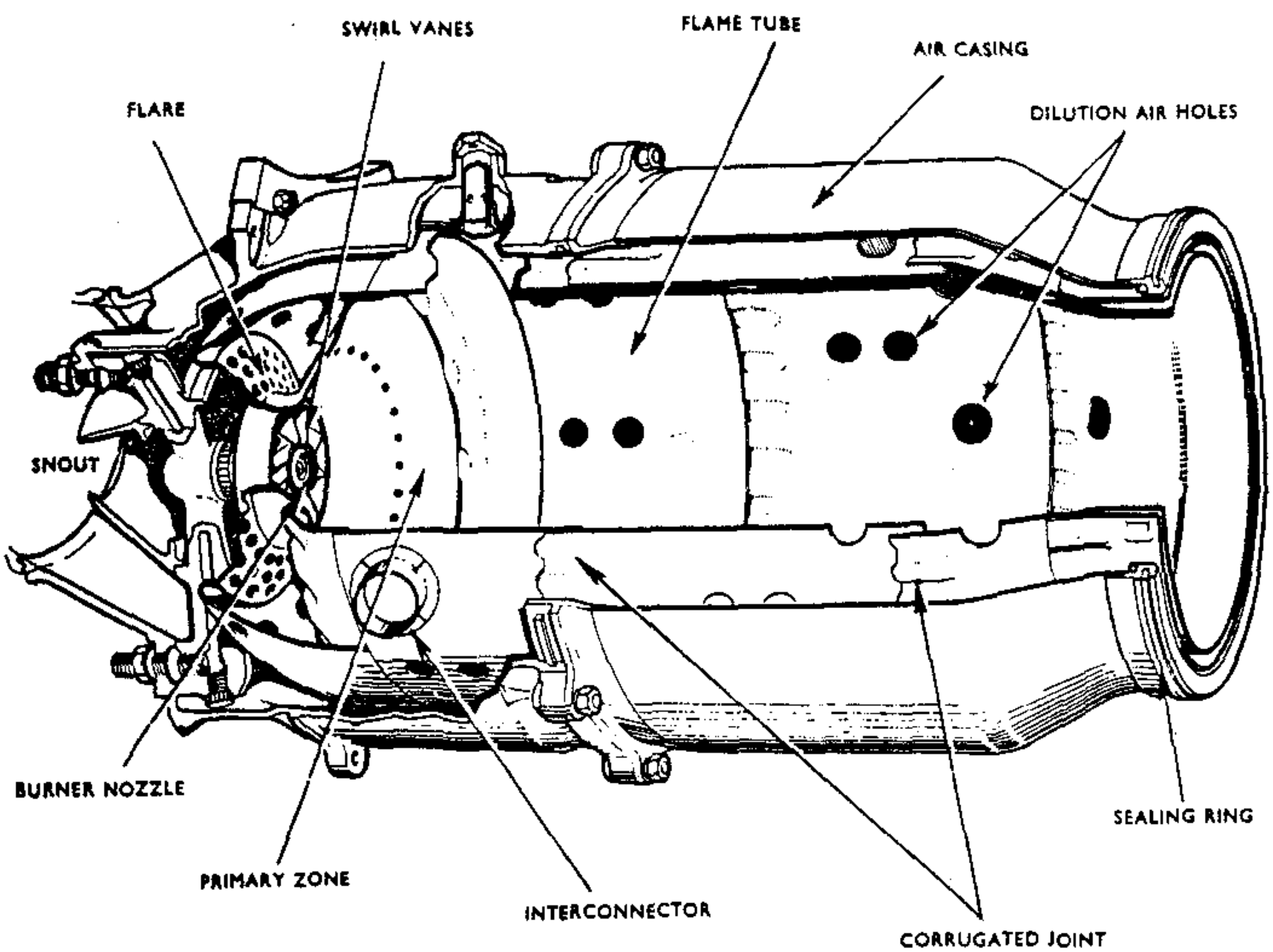
The temperature of the combustion gases released by the combustion zone could reach temperatures of 1800 to 2000% which is far too hot for entering the turbine. The air not used for combustion which is about 60 to 75% of the total airflow, is therefore introduced progressively into the flame tube. This dilution reduces the temperature of the gaseous mixture to approximately 1200°F and also keeps the temperature of the walls down.

The Gas Turbine

The gas turbine like steam turbines use familiar impulse and reaction principals. Because they work with lower pressure drops they have fewer stages, and less change in blade height from inlet to exhaust.

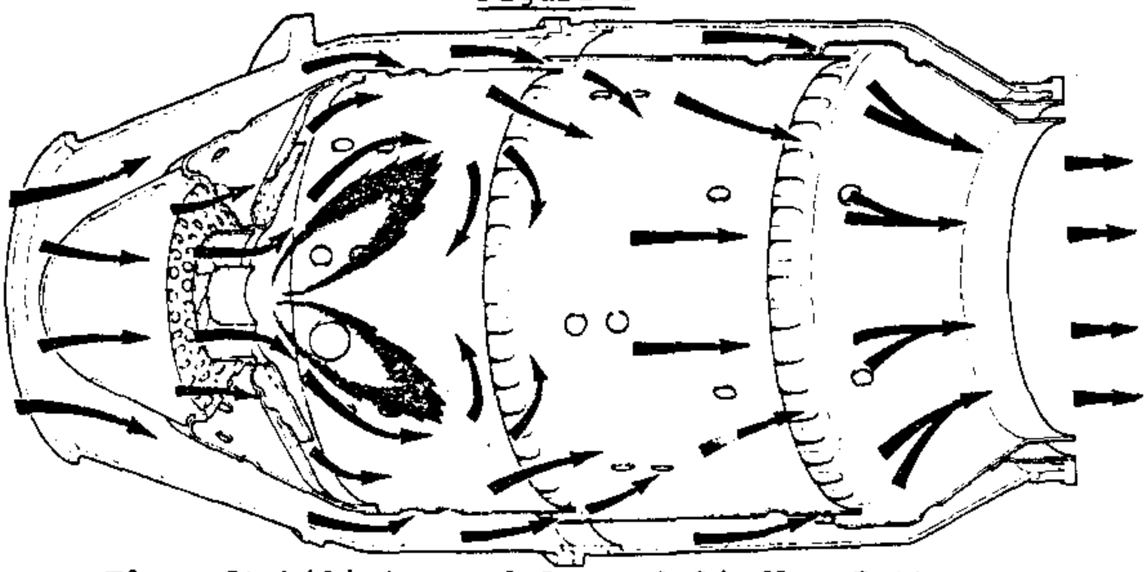
For best performance the gas turbine must work with high inlet gas temperatures. This poses service material problems that must be overcome by design. One approach is to cool the part subjected to highest temperatures and highest stresses. The most critical are the first stage moving blades and the disks carrying them. In one design air is bled from the compressor outlet and led over disk surfaces and blade roots to give film cooling, protecting the metal from the main gas flow temperature which would severely damage the rotor.

The turbine depends for its operation on the transfer of energy between the combustion gases and the turbine. The transfer occurs at approximately 90 percent efficiency, losses being due to thermodynamic and mechanical losses. The torque or turning power applied to the turbine is governed by the rate of gas flow and the energy change of the gas between the inlet and outlet of the turbine blade, thus if the energy is absorbed efficiently the induced whirl upon leaving the nozzle will be removed from the gas stream so that the flow at exit from the turbine will be straightened out. Excessive residual whirl reduces the efficiency of the exhaust system and also tends to produce vibrations.



A Typical Combustion Chamber

Figure 6



Flame Stabilizing and General Airflow Pattern

Figure 7

The degree of twist along the nozzle and blade is to make the gas flow from the combustion chambers do equal work at all positions along the length of the blade and to insure that the flow enters the exhaust system with a uniform axial velocity. The degree of reaction varies from root to tip, being lowest at the root and highest at the tip, with the mean section having a chosen value of about 50 percent.

Gas Turbine Performance Data

One major characteristic of any gas turbine is the variation of output with ambient inlet air temperature. With all other conditions and limits remaining constant a considerable increase in output results from a decrease in ambient air temperature. If the design point rating is at a relatively high temperature (27°C), a considerable increase in output is permissable over most of the year when the temperature is below the design temperatures. This characteristic is a major advantage for many power applications where winter load requirements are higher than summer loads. Figure 8 illustrates the variations in output with ambient air temperature.

The heat rate shown in Figure 8 is the criteria used to express unit efficiency. It is the energy input to the plant for each hp hour of output or kilowatt-hour of output.

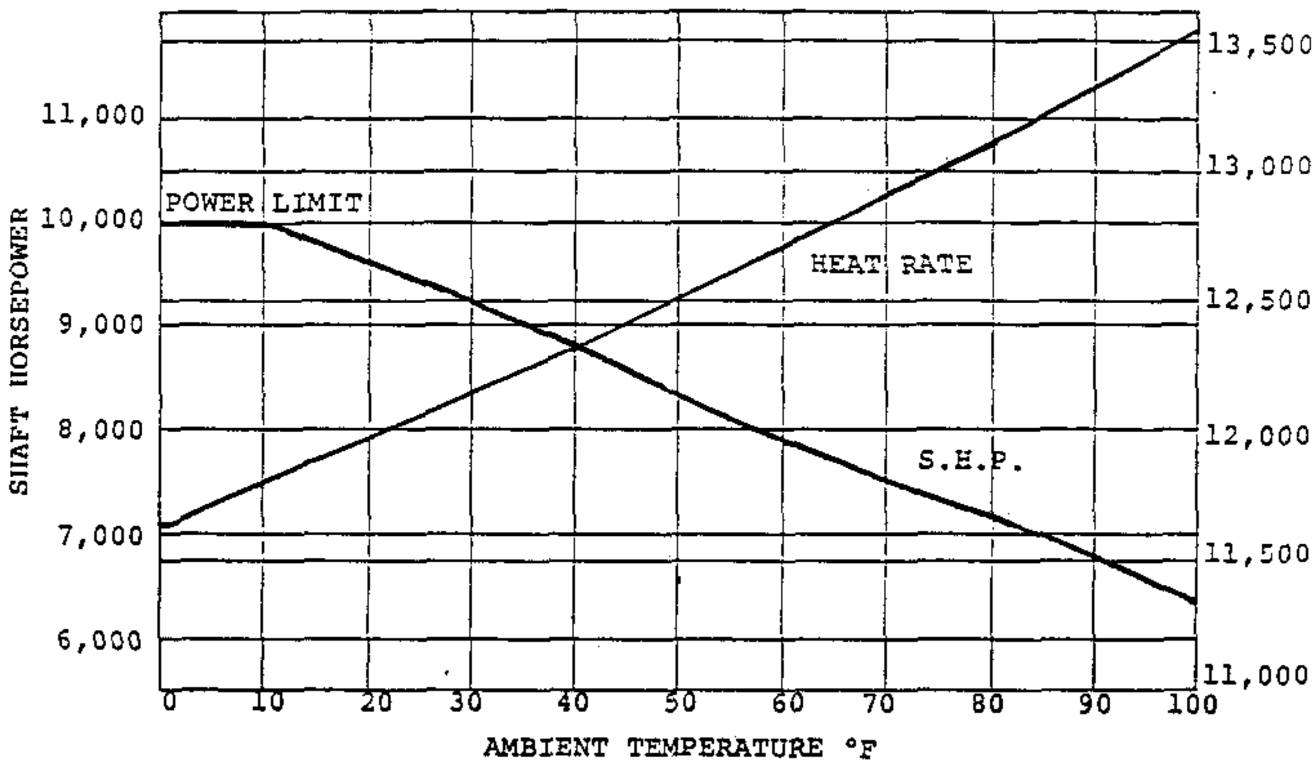
The output of a gas turbine also varies with ambient or inlet air pressure. As air pressure changes with altitude the output of a gas turbine will therefore change as well. As the inlet air pressure decreases or altitude increases, the output of the gas turbine will decrease as shown in Figure 9.

Such items as pressure losses in the intake and exhaust duct with affect the engine performance. For every one percent loss in total pressure in the intake ducting, the air flow and fuel flow will decrease by one percent, while the horsepower output of the power turbine will drop 2.6 percent.

For every one percent pressure loss in the total pressure in the exhaust ducting and stack there will be a decrease in power output of 1.6 percent.

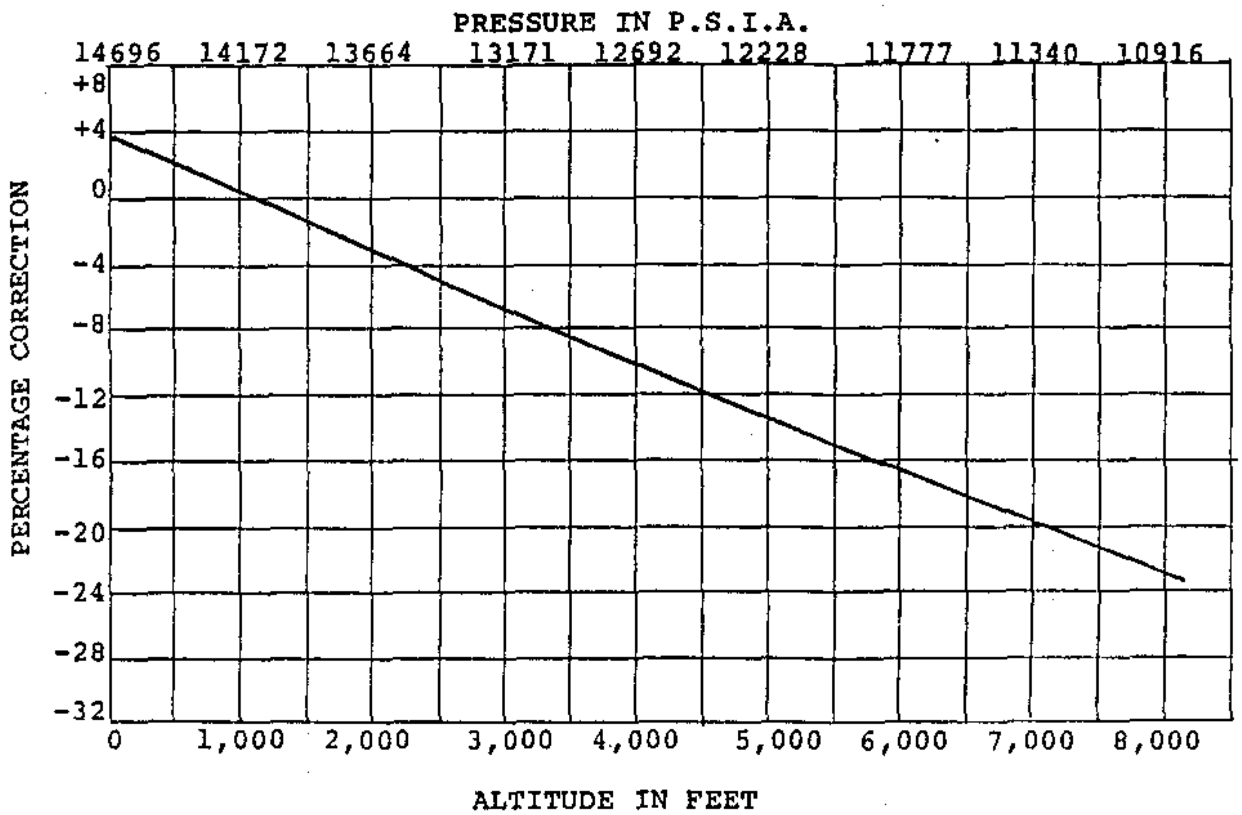
Operating Procedures

An outline of the Pickering gas turbine operation when call upon to start is given in Appendix A. The reader wil not be held responsible for the information contained in the appendix. It is only for those who are interested.



Variation of Power Output with Ambient Temperature

Figure 8



Variation of Power Output with Ambient Pressure

Figure 9

APPENDIX A

Control System

Each gas turbine generator has a separate control cubicle and is completely independent of other units in regard to auxiliaries and control equipment except for the common bulk fuel storage and the remote control centre in the main control room.

The units can be started and controlled from either the local or remote control centres.

Droop and isochronous governing facilities are provided with the selection of mode determined either by the operator or the emergency transfer system.

Auto and manual synchronizing equipment is provided for each unit.

The following is a general outline of the gas turbine operation when a unit is called to start.

1. Pre-start conditions such as: equipment temperatures, power supplies, control air pressures and governor position have to be in their correct state.

2. The power turbine lubrication system starts and establishes proper lubrication pressures. The starter motor starts to rotate the gas producer and speed builds up to an ignition speed of 800 RPM.

3. Following ignition the gas producer continues to accelerate to a self sustaining speed of 3000 RPM during which time the starter motor will have deenergized. Acceleration rate is controlled by a bias signal from the governing system and a limiter on the governing valve controlled by the gas producer discharge pressure since the electrical governor electrical signals are not yet available.

4. Rotation of the power turbine and the generator starts and voltage builds up to 4.16 KV with speed regulated for a generator frequency of 60 cycles.

Depending on the starting signal origin, the unit can be automatically or manually synchronized to the sytem following breaker selection by the operator or the unit will close in on a "dead bus" and start picking up loads sequentially as controlled by the emergency transfer system.

Following a successful start the units will be ready to accept load approximately 60 seconds from initiation of the start signal.

During unit operation of the following automatic limiting devices are effective to control unit output and override increase load signals.

1. High exhaust temperatures.
2. Gas producer speed (limited at 7500 RPM with overspeed trip at 7700 RPM).

Protective System

The follwing items are monitored and will automatically trip the units and lock out the start circuit:

1. Power turbine or gas producer overspeed.
2. Fire.
3. Generator fault.
4. Governor pump failure.
5. 125 V dc power failure.
6. Inverter ac failure (battery supplied).
7. Manual trip.

Lock out of the start circuit requires manual resetting.

The following items may or may not automically trip the unit depending on mode of operation (ie, peaking or emergency class 3 service) and the finalized design.

1. Low lube oil pressure.
2. High exhaust temperature.
3. High vibration.
4. High bearing drain oil temperature.
5. Fuel and lube oil temperatures low.
6. ac failure.

During start-up of the units, various sequences are timed and if they exceed a time limit, the unit will automatically shutdown but will not lock out. This allows a second start attempt. If the second start fails the start circuit is then locked out.

These sequences include the following:

1. Oil pressure in power turbine lube system fails to be established within 15 seconds.
2. Gas producer fails to reach self sustaining speed within 58 seconds.
3. ac power to auxiliaries must be established within 6 minutes.
4. Ignition has to occur within 20 seconds following switch on of fuel and ignitors.

L. Laplante

ASSIGNMENT

1. What is the advantage of an open cycle gas turbine over a closed cycle gas turbine?

2. Briefly describe the principle of operations of an open cycle gas turbine.

3. Explain the difference between compressor stall and compressor surge.

4. What are the symptoms of a stall or surge?

5. Explain why only a small percentage of the total air flow through a combustor actually undergoes combustion.

6. Why are there so few stages in a gas turbine compared to a steam turbine as found in our plants?

7. What effect does inlet air temperature have on gas turbine power output?

8. What effect does inlet air pressure have on gas turbine power output?