File No.T-18/1/2022-GATES(NW)

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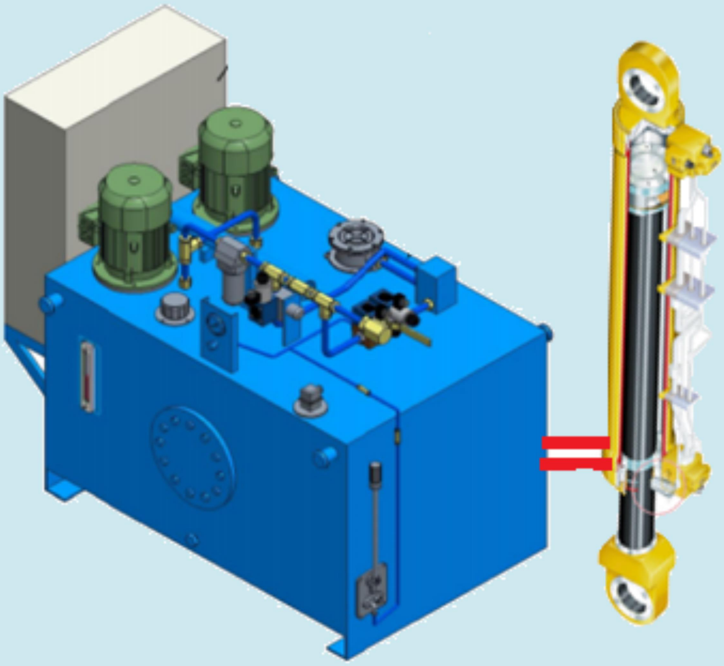

Design Manual on Hydraulic Hoist For operation of Hydraulic Gates

New Delhi

November 2022

अभिकल्प (उ.प.) संगठन

Design (NW) Organisation

फाटक अभिकल्प (उ.प.) निदेशालय

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FOREWORD

Hydraulic Gates are an important component of water storage / diversion structure for the regulation of water level, flood management, discharge regulation from reservoirs and other storage structures. Central Water Commission (CWC) has been publishing design standards for planning, execution, operation and maintenance of many components of the water resources projects. The availability of new technology demands updation of the standards to keep them relevant. Also, it is necessary that experiences learnt from the past should be incorporated into these standards to avoid those pitfalls bythe future generation.

CWC has prepared many design manuals related to hydro-mechanical equipment. Such manuals proved to be very useful and are being referred to byplanners and designers as well as field engineers. The necessity of a design manual for the hydraulic hoist system was being felt for long. This manual provides a general framework for the design engineering of hydraulic hoists for operating hydraulic gates.

The manual has been prepared by Gates Design (N&W) Directorate of CWC. I appreciate the effort of the officers who were involved in the preparation of this document. I am sure that the manual will be well received and useful for the design fraternity in this field.

(Dr. Rakesh Kumar Gupta)

New Delhi
November, 2022

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सत्यमेव जयते

MEMBER (Design & Research)
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Government of India.

PREFACE

The control structures comprising hydraulic gates and their operating equipment play an important role in the operation and safety of a dam or barrage. Hydraulic hoist which comprises of cylinder, piston and power pack is becoming the first choice of planners and designers because of their cost effectiveness. Unlike the more traditional cranes and lifts, hydraulic hoists rely on an oil-based piston mechanism instead of a large motorized operating system. This piston mechanism allows the hydraulic hoist to lift much larger loads.

Like all hydraulic-based systems, these hoists are susceptible to leaks or broken seals within the oil cavities, which require extensive repair and maintenance. Therefore, it is essential that proper procedure be followed during planning, designing, execution, operation and maintenance of these equipments.

The Design Manual on Hydraulic Hoist has been prepared based on the design practice being followed in Central Water Commission. The manual covers the general design features, criteria for selection of different components, hydraulic circuit and material.

The manual has been authored by Shri Satish Kamboj, Director under the guidance of Sh S K Sibal, Chief Engineer. I would like to thank all officers of Central Water Commission who were involved in preparing this manual. I sincerely hope that this manual will be a ready reference for concerned project designers, field engineers and other technical professionals.

(J. Chandrashekhar Iyer)
Member (D&R)
Central Water Commission

New Delhi
November, 2022

Officers and staff associated in preparation of the Design Manual

This Design Manual has been authored by Sh Satish Kamboj, Director, Gates Design (N&W) Directorate under the guidance of Sh S K Sibal, Chief Engineer, Design (N&W) Organisation. Sh Ghanshyam Patel, Assisstant Director, Gates Design (N&W) Dte has provided technical support for preparation of the manual. Drawings have been drawn by Sh Anuj Manohar Jadhav, Sr Draft's Man and Sh Biswajeet Biswas, Head Draft's man under supervision of Sh Sandeep Kumar, Assistant Director II, Sh Anurag Pal, Assistant Director II, Gates Design (N&W) Dte Gates Design (N&W) Dte provided logistic support.

The design manual has been reviewed by Sh Rahul Kumar Singh, Director, Gates Design (NW&S) Directorate and Sh Amit Ranjan, Director, Gates Design (E&NE) Directorate under overall guidance of Sh Gulshan Raj, Chief Engineer, Dam Safety Organisation.

Efforts of all the staff and officers who contributed directly or indirectly in preparation of this design manual are appreciated and gratefully acknowledged.

Disclaimer

Analysis of structures involved in this manual is dealt by adopting standard and popular structural analysis methods. However, depending upon specific applications, some advance or specialised analysis may be necessary, these are not dealt herein. Although every effort is made to ensure the correctness of information submitted for publication, the design manual may inadvertently contain technical inaccuracies or typographical errors. CWC assumes no responsibility for errors or failures of equipment resulting from adoption of design and analysis techniques presented in this manual and consequences thereof.

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List of symbols

a	outer radius, outer radius of piston,
A_r	Area ratio factor
b	inner radius, inner radius of piston, width of groove
Di	Inside Dia. of the Shell,
Do	Outer dia of the shell,
f	Allowable stress,
F_{pull}	Maximum pull load,
d’	‘O’ ring cross section diameter
h	depth of groove including radial clearance of matting parts
j	Joint Factor,
k	depth of groove
L_{ctc}	distance between rod eye centre and pivot centre.
L_x	distance from rod eye centre to bending moment point,
m	reciprocal of Poisson’s ratio,
M	Maximum bending moment in the plate,
n	Chamfer in radial direction,
P or P_d	Design pressure,
P_{LP}	pressure losses in all the elements along with the pipeline
P_{LT}	Pressure loss in return line
P_N	Total designed pressure required for lifting
q	load per unit area,
r_a	radius of rod eye bearing at moving surface,
R_a	reaction due to self-weight of hoist cylinder at rod eye,
R_{pivot}	reaction due to self-weight of hoist cylinder at pivot.
r_{pivot}	radius of pivot bearing at moving surface,
r_0	radius at start of distributed load,
t	thickness of plate/ piston / shell,
μ_1	Coefficient of friction of rod eye bearing,
μ_2	Coefficient of friction of pivot bearing,
v	poisson’s ratio,
w	self-weight of rod per unit length, pressure,
σ_t	tangential stress at surface of plate,
σ_r	radial stress at surface of plate,

1 Scope

This manual contains the design procedures to be followed for designing hydraulic hoist of gates for various applications in river valley projects. It is intended to provide practical guidance to engineers connected with the field of design, fabrication and erection of such equipment.

Although the primary intention of this document is to describe various design practices, it also covers some broad features of manufacturing, erection and testing of this equipment.

2 Limitations

This manual is based on the general design practices being followed in Central Water Commission (CWC). Although the application of this manual is recommended for Indian conditions only, it is attempted to conform to some prominent international guidelines of other countries also. Especially standards of USA, Germany, Japan and Sweden have been referred.

Analysis of structures involved in this manual is dealt by adopting standard and popular structural analysis methods. However, depending upon specific applications, some advance or specialised analysis may be necessary, these are not dealt herein.

3 Basic References

This manual is based on the broad guidelines contained in the Indian Standards. Some features from standards of other countries are also adopted as they are often followed in India. Details contained in catalogues of reputed manufacturers are also referred in this manual. List of various references is given at '**Annexure-1**'.

4 Introduction

Wide range of hoists is deployed for the operation of hydraulic gates. Most of the gates are required to be operated by a fixed hoist or by a moving device except gates which are operated by action of reservoir level. The fixed types of hoist are screw hoist, rope drum hoist, roller chain and hydraulic cylinder. Movable hoisting devices are used mainly in the operation of the stoplogs and diversion or bulkhead gates. Among the common moving type of hoists are manual or power driven hoists, overhead cranes, gantry cranes and wheel or tractor mounted cranes.

Many new gate designs utilize hydraulic cylinder hoist systems because they may be cost effective. However, these systems have some disadvantages and are not suited for all applications. Careful planning and close coordination of dam design engineer with the hydro-mechanical design engineer is required to optimize the hoist system.

Hydraulic Hoist consists of a cylinder in which a piston and piston stem is connected to the gate which lifts and lowers the gate by means of hydraulic pressure. Main components of hydraulic hoist are cylinder with upper and lower heads, piston, stem passing through a packing in the lower cylinder head, pumps, oil reservoir, control elements and piping. **Fig. 1** shows general arrangement of hydraulic hoist with radial gate installation.

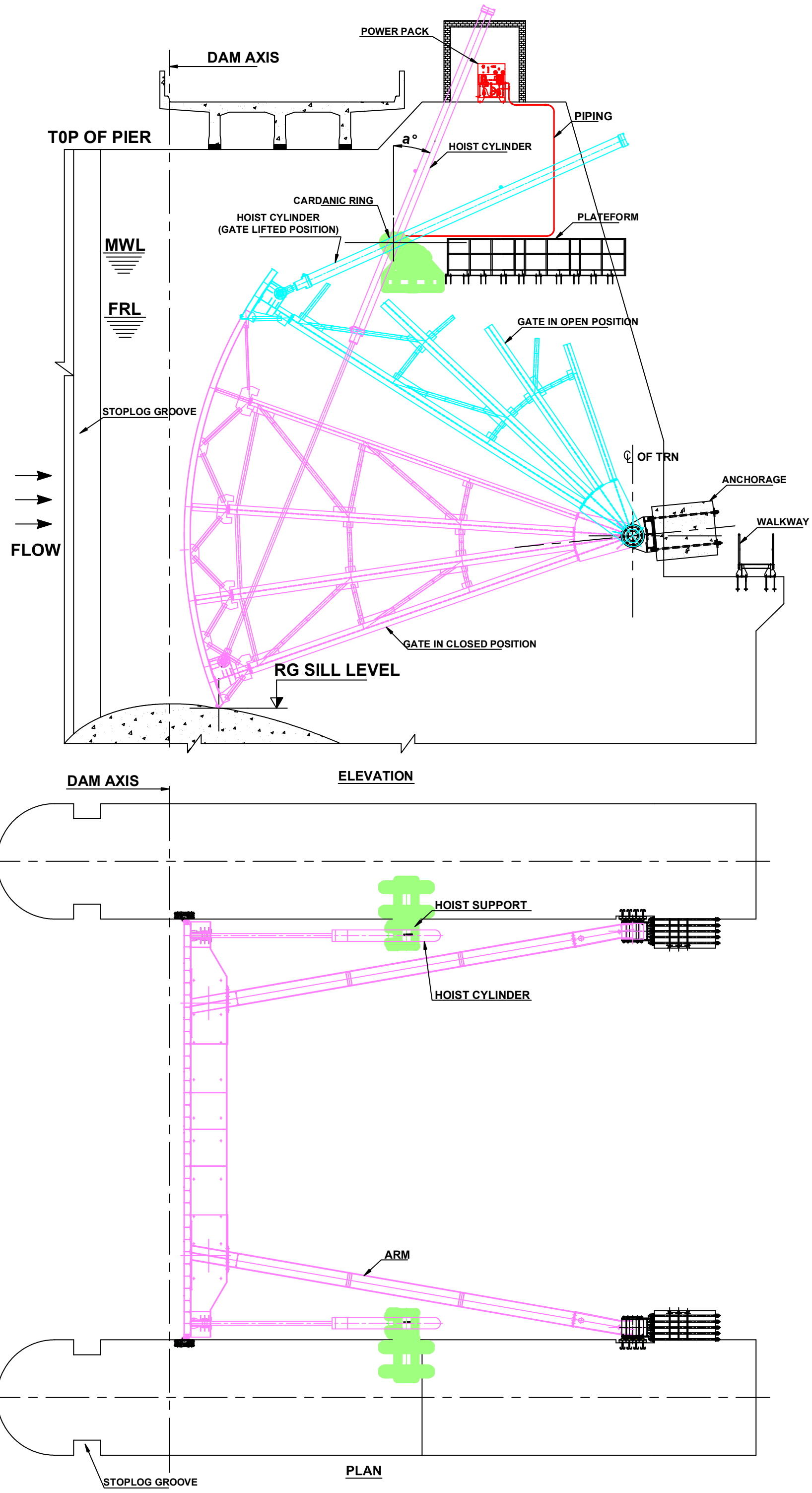

Figure 1: General Installation Radial Gate with Hydraulic Hoist

These hoists are very versatile and can be used in any type of application. They can be used for the operation of sluice gates where the hoist chamber is located within the body of the dam below the maximum water level in the reservoir. Telescoping cylinders are used for deep submerged gates. More than one cylinder is also used depending upon the hoisting force requirement and economics. Where dual lifting cylinders are utilized, synchronization of stem travel may be required for smooth non-binding gate motion.

Hydraulic hoist may consist of double acting cylinder as well which applies positive downward force also. The downward force is required to close the gate where gate is not self-closing i.e. where gate cannot be closed by its weight.

5 Factors governing selection of Hydraulic Hoist

Selection of type of hoisting arrangement may be the discretion of planners and designers. However, following factors govern the choice of hydraulic hoist:-

i) These hoists may be made for high capacity.
ii) Larger range of hoisting / lowering speeds.
iii) Faster speed is possible with dampening of vibrations of gate and cushioning at the end of stroke especially for gravity (emergency) closure of intake gates.
iv) Higher efficiency of the system as there are minimum numbers of moving parts as compared to other hoists.
v) There is minimum wear & tear due to auto lubrication in this type of hoists. Hence, smooth and noiseless operation.
vi) Faster response to control mechanism. Emergency closure of gates (quick closing) can be achieved very easily. This characteristic makes them highly suitable for Power Intake Gates.
vii) The use of hydraulic hoist leads to ease in gate operation programming as gate opening / closing speed can be more easily manipulated and hence effective remote control operation is possible.
viii) Requirement of positive thrust for closing the gate.
ix) Skilled manpower required for maintenance, replacement of cylinder parts may be tedious job.

6 Hoist Capacity

While calculating the hoist capacity, all the possible downward and upward forces may be considered. The capacity of the hoist should be based on the algebraic sum of the following:

a. All weights consisting of:
   i) gate leaf along with its components including ballast, if any; and
   ii) moving parts of the hoist like intermediate stem, gate stem, piston etc.
b. Water load on gate components including buoyancy, wherever necessary.
c. Frictional forces comprising of
   i) Wheel friction (rolling friction of wheel on track and the friction at the bearing of the wheel pin);
   ii) sliding friction;

iii) guide friction;
iv) trunnion friction, if hoist is used for radial gates;
v) seal friction including bearing pad friction in case of slide gates;
vi) friction of moving parts of hoist including mounting friction, if any;

d. Any hydrodynamic load like hydraulic downpull/ uplift.
e. Silt and ice load, wherever encountered.
f. Silt friction with gate components.
g. Seating load as given below:

Type of Gate	Minimum Seating Load
Low head fixed wheel or radial gates for spillway crest	2.5 kN/m length of gates
Medium head gates	5.0 kN/m length of gate
High head sluice gates/ High head radial gates	10.0 kN/m length of gate

h. Any other consideration specific to a particular site.

During the movement of gates, the combinations of above forces act depending on the direction of motion of gates. The worst combination of the above forces during either lowering or raising cycle should be considered. The hoist capacity arrived at should be increased by at least 20 percent as reserve.

6.1 Hoist Rating

General: Hoist rating and overload capacities, which are dependent on the hoist design, should satisfy the following conditions stated with respect to required forces:-

Hoist Rated Capacity: Hoist rated capacity is the pull and push the hoist will exert in rated conditions of speed, power, pressure, etc.

Hoist Overload Capacity: Hoist overload capacity is the exceptional pull/push the hoist can exert within stress and other limits specified for overload conditions.

6.2 Excessive or Irregular Hoisting Forces

General: Hoist connections, mountings, and other parts of the equipment subject to such forces should withstand without damage the maximum output pressure of the system. In case of twin cylinder hoist maximum hoisting force may be exerted in symmetrical or unsymmetrical condition.

Maximum Hoisting Forces with Symmetrical Reactions: In this overload condition of the gate, the portion of the maximum pull of the hoist which is in excess of the required normal pull force should be resisted by two equal "blocking forces" acting at the guiding arrangement on either side of the gate.

Maximum Hoisting Forces with Unsymmetrical Reactions: The gate should be evaluated for single cylinder holding or lowering.

7 Mounting of Hydraulic Hoists

7.1 Connection of Hoist to the Gate

Hydraulic hoist stem is generally connected to the gate while the cylinder shell is connected to civil structure. The connection is provided such that there is no undue stresses generated in the hoist parts. Pinned connection with spherical bearing is preferred for connecting the hoist stem with gate, as this type of connection has inherent capability of releasing the moment in two directions and hence undue stresses in hoist parts is avoided.

7.2 Connection of Hoist to the Civil Structure

Type of connection of hydraulic hoist to the civil structure depends upon many factors such as operation requirement, optimization of stroke length and effort, space availability etc. Hydraulic hoist of vertical lift gate may be fixed type or pinned type connection to civil structure whereas connection of radial gate hydraulic hoist requires angular movement.

7.2.1 Vertical lift Gate Hoist

The hoist for vertical lift gate can be single acting or double acting depending upon the operational requirement of gate. The hoist cylinder of vertical lift gate can be mounted on bonnet cover for sluice gate where the hoist chamber is located within the dam body below the maximum water level in the reservoir. Typical layout of bonnet cover gate is shown in **Fig. 2**.

In other case where gate is self-closing, single acting hydraulic hoist is used which can be mounted at top of dam with the help of flange mounting or pinned connection. The hoist stem is connected with the gate through tension members/ links.

7.2.2 Radial Gate Hoist

Generally single acting hoists are required to operate the radial gate except in few cases of sluice outlets where gate weight may not be sufficient for self-closing. Rotation of gate makes the hoist cylinder to rotate about the pivot point of hoist cylinder.

The hydraulic hoist can be connected to radial gate at different location on downstream of skin plate e.g. horizontal girder, radial arm. Connection of hydraulic hoist to the bottom most girder of the gate may require lesser stroke length but has lesser lever arm also. Whereas connecting the hydraulic hoist to top most girder of the gate may require higher stroke length but has higher lever arm also, this reduces the effort requirement. Also the high civil structure is required by connecting the hoist at top girder location. In some situation where civil structure is already high due to other requirements, the connection of hoist at top girder location may be economical choice.

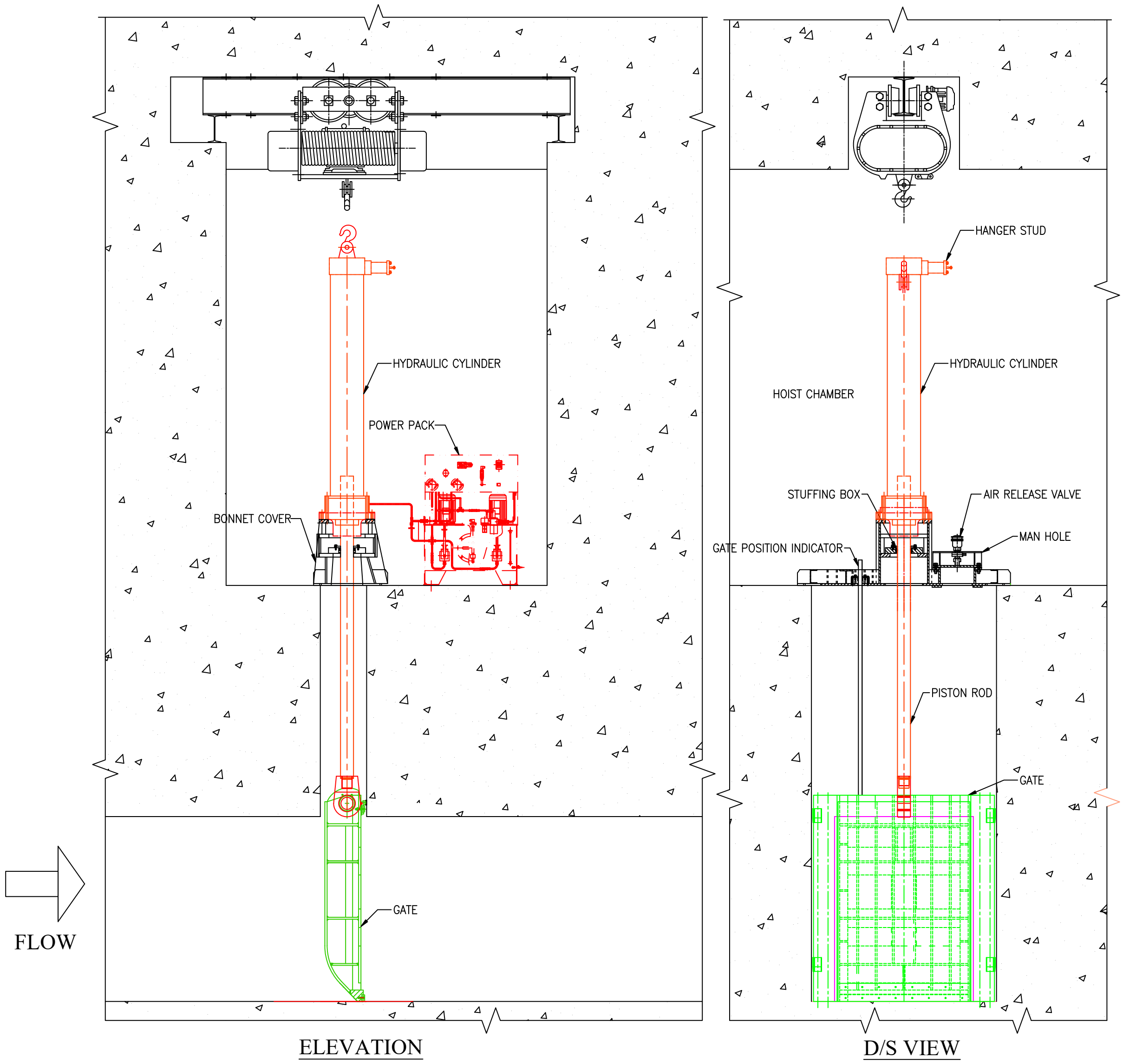

Figure 2 : General Installation of Sluice Gate

7.2.3 Pivot Point Location

The hydraulic hoist cylinder of radial gate is connected to the civil structure at a point known as pivot point. The planner should take into considerations the different factors while selecting the pivot point location as its Location is important for economic considerations apart from technical requirements. Vertical positioned cylinder, in gate fully closed position, develops less stresses as compared to non-vertical positioned cylinder.- Shifting the pivot point toward path of skin plate i.e. upstream reduces the effort required for lifting the gate, however, the required stroke length is also more. Keeping in view the limited length of cylinder tube honing facilities available in the country, the pivot point may require to be shifted towards downstream i.e. gate trunnion side. This reduces the stroke length but results in higher effort requirement.

Similarly, the level of pivot point is decided by considering the total lift of the gate, stroke length, lever arm, hoist connection point at gate, layout geometry and tail water level. The cylinder support bracket can also be provided on the hoist cylinder at a point between the cylinder heads such that over tilting is avoided during maintenance and also the stresses minimized after taking care of all geometric requirements. Alternatively the support can be provided at the piston end cover of the hoist cylinder.

7.2.4 Pivot angle:

The pivoting angle allowed by the mounting shall not be less than the maximum angle required for gate operation, maintenance or erection plus 75 mm clearance between support and cylinder in each direction. Planner should minimize the angle of hoist cylinder to the vertical as far as possible in gate closed position. This helps in reduction of stresses as well as deflection of hoist.

7.2.5 Cardanic Ring

For radial gate, hoist cylinder which is mounted at intermediate location between cylinder heads, cardanic ring is used, for allowing the movement of hydraulic cylinder in direction perpendicular to water flow in addition to rotation corresponding to gate operation. In other type of installation where the cylinder is mounted on piston side head, the bracket is connected through spherical bearing to pin which transfers the load to anchorages embedded in concrete structure. Typical arrangement of cardanic ring is shown in **Fig. 3**.

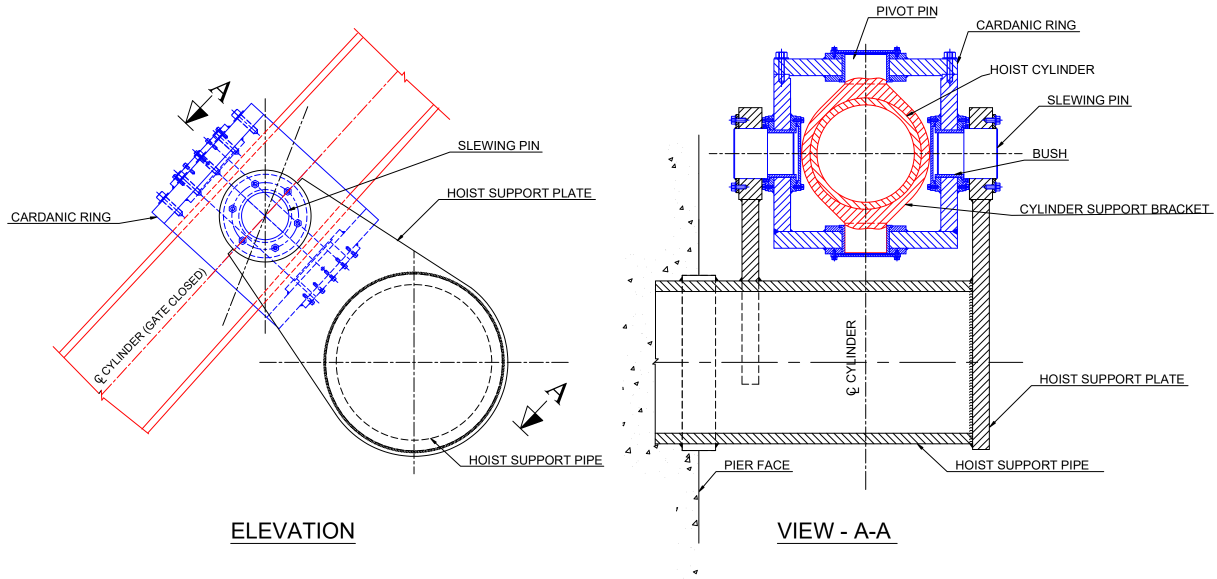

Figure 3 : Cardanic Ring

7.2.6 Bearings:

Bearings for cylinder mountings are self-lubricating bushings on corrosion-resistant pins. Rod end bearing is generally provided with self-lubricating spherical bush bearing.

8 Design Stroke.

Hydraulic cylinder assemblies are designed for stroke length such that when the gate and hoist assembly are erected exactly to the nominal design geometry, the piston rod has an available over-stroke in addition to the stroke required for complete rod extension to fully lower the gate. This over-stroke measured along the piston rod travel should not be less than 40mm. Similarly when the gate is completely raised, the hoist should have an available over-stroke of 25mm minimum. Actual extent of over-stroke should take into consideration all manufacturing and erection tolerances and other factors like bottom seal compression for gates.

9 Speed and End Cushion

Speed of travel of the hydraulic hoist depends upon the requirement of gate to be operated. Generally the operating speed of gate is 0.3m/minute to 0.7m/minute. However, higher speed can be provided for specific requirement such as for penstock gate. For radial gate arrangement, the speed of hoist cylinder is calculated from the geometry of gate. The hoist speed can be changed by flow control valve.

At the end of stroke during closing of gate, end cushion of 150mm travel is provided to avoid jerk. This cushioning should be designed by keeping in view the over stroke provision. Cushioning sleeve with cushioning adjustment screw is provided to reduce the speed at the end of closing cycle of gate. The cushioning sleeve is tapered for smooth reduction of speed. Ball type check valve is also provided in the lower head to bypass the end cushioning arrangement during lifting of gate.

Alternatively speed reduction at the end of stroke can be achieved by operating the limit switches by the moving gate. Type of end cushioning should be selected with due diligence because any malfunctioning in the speed reduction arrangement could have adverse effect on the gate.

10 Seal and Bearing Friction

The friction due to sealing should be calculated by sealing force generated by fluid pressure and coefficient of friction between seal and metal surface. In absence of accurate calculations, generally rod seal friction and piston seal frictions are assumed as 1.5% and 3% of hoisting force respectively. However, seal friction may be higher in case of radial gate hoists where bending of cylinder and rod create extra pressure on seals and bearings.

11 Material and Design Stresses

The recommended materials and design stresses for various components of hydraulic hoist are given in **Table 1**.

Table 1: Materials and design stresses for the components of Hydraulic Hoist

Sl. No.	Component Part	Recommended Material	Reference Specification	Allowable Design Stresses
(1)	(2)	(3)	(4)	(5)
i	Support frame	Structural steel	IS 808 IS 2062	0.5 YP
		Cast steel	IS 8500 IS 1030	0.55 YP
ii	Cylinder	a) Plate Steel b) Carbon Steel c) forging Steel	IS 2002 IS2041 IS 2004	0.30 YP
iii	Upper and lower cylinder head	Structural Steel Cast steel	IS 2002 IS 1030	0.25 YP
iv	Piston stem	a) Corrosion resistant steel b) Forged steel	IS 3444/ IS 1570(V) 18% chromium Min IS 2004	0.40 YP
v	Piston	a) Cast steel b) Forged steel c) Grey iron casting	IS1030 IS 2004 IS 210	0.25 YP
vi	Piston ring	a) Bronze b) Grey iron casting c) Synthetic PTFE	IS 318 IS 210	0.25 UTS
vii	Clevis pin	Corrosion resistant steel	IS 1570 (V) IS 6903	0.30 YP
viii	Gland, Clevis bushing	Cast manganese bronze Synthetic PTEF		0.30 YP
ix	Studs and bolts	Mild Steel	IS 1367	

12 Hydraulic Pressures

12.1 Required Cylinder pressures

The required fluid pressure in the cylinder on either side of the piston is based on the required operating forces. Weight of hoist moving parts, oil and seal/ bush friction in the cylinder should also be considered to calculate the required pressure in the cylinder.

12.2 Nominal System Design Pressures

Nominal system design pressures is the discharge pressures of pumps when functioning at the specified normal hoist pull capacity, considering pressure losses in the hydraulic system between the pump and cylinder. Nominal system design pressure is selected in the range of 100 to 200 kg/cm^2.

12.3 Maximum System Pressure

Maximum system pressure is the maximum pump output pressure or the pressure resulting from single cylinder holding force whichever is greater.

12.4 Minimum Effective Pressure

Minimum effective pressures in any part of the hydraulic system should not be lower than 0.703 kg/cm^2 (10 psi) absolute pressure during normal hoist operating conditions.

12.5 Hydrostatic Test Pressure

Hydrostatic test pressure is the higher of the following pressures as applicable for the hydraulic system, or parts of the hydraulic system:-

1. 1.5 times the nominal system design pressure
2. 1.1 times maximum system pressure

12.6 Pressure Loss

Pressure loss due to friction in the pipe line and hydraulic elements is considered to design the maximum design pressure. The pressure loss in the hydraulic elements can be taken from the graph of manufacturer catalogue. Following method may be used to calculate pressure loss in the hydraulic circuit:-

12.6.1 Pressure Loss during Lifting of Gate

This is the main pressure which govern the design pressure and it should be designed with due diligence. The pressure loss in the Hydraulic system while lifting of the gate can be divided into two parts as explained below:-

i) **Pressure loss in pressure lines:** It is the sum of the pressure losses in all the elements along with the pipeline. Let's assume it is $P_{LP.}$

ii) **Pressure loss in return line:** In the return line, the pressure losses in all the elements present in return line along with the pipeline losses are summed up (say P_{LT}) and multiplied by the Area ratio factor $\mathbf{A_r}$.

$$A_r = \frac{\text{Piston area}}{(\text{Piston Area} - \text{Rod area})}$$

So pressure loss in return line = P_{LT} X A_r.

Total designed pressure required for lifting $P_N = (P_d + P_{LP} + P_{LT} \text{ X } A_r)$

The value of **Area ratio factor** will always be more than 1 and is critical while designing the maximum Design Pressure of the system. If the Area ratio factor is not considered and the value of Maximum System Design Pressure is less than the P_N., the gate may not be lifted.

12.6.2 Pressure Loss during lowering of Gate

When the gate is lowered by the gravity, a minimum pressure of about 5 to 7 bars is maintained in the system. During lowering of the gate the pressure loss is encountered as per flow in pressure line and return line.

13 Design of Hydraulic Hoist Components

Main components of Hydraulic hoist consist of following:

a) **Cylinder assembly -** Main components of hoist cylinder are shown in **Fig-4**. The hoist cylinder consist of following components:
   i) Cylinder or shell
   ii) Cylinder Heads (Piston Side Head and Rod Side Head)
   iii) Piston Rod
   iv) Piston
   v) Piston and Piston Rod Bearings
   vi) Seals and Wipers
   vii) End cushioning arrangement
   viii) Cylinder support bracket/ flange/ cardanic ring
   ix) Bolted connections

b) **Power pack unit or control assembly -** consist of following main components:
   i) Oil tank
   ii) Pumps with motors and starting equipment
   iii) Hand Operated Pump
   iv) Manifold Block
   v) Shut off valve
   vi) Check / non return valve/ pilot operated check valve
   vii) Direction Control valves (manually, electrically or hydraulically operated)
   viii) Pressure relief valves/ solenoid controlled pressure relief valve
   ix) Flow control valve
   x) Accumulators
   xi) Logic valve
   xii) Rupture orifice
   xiii) Filters & Strainers
   xiv) Pressure gauges

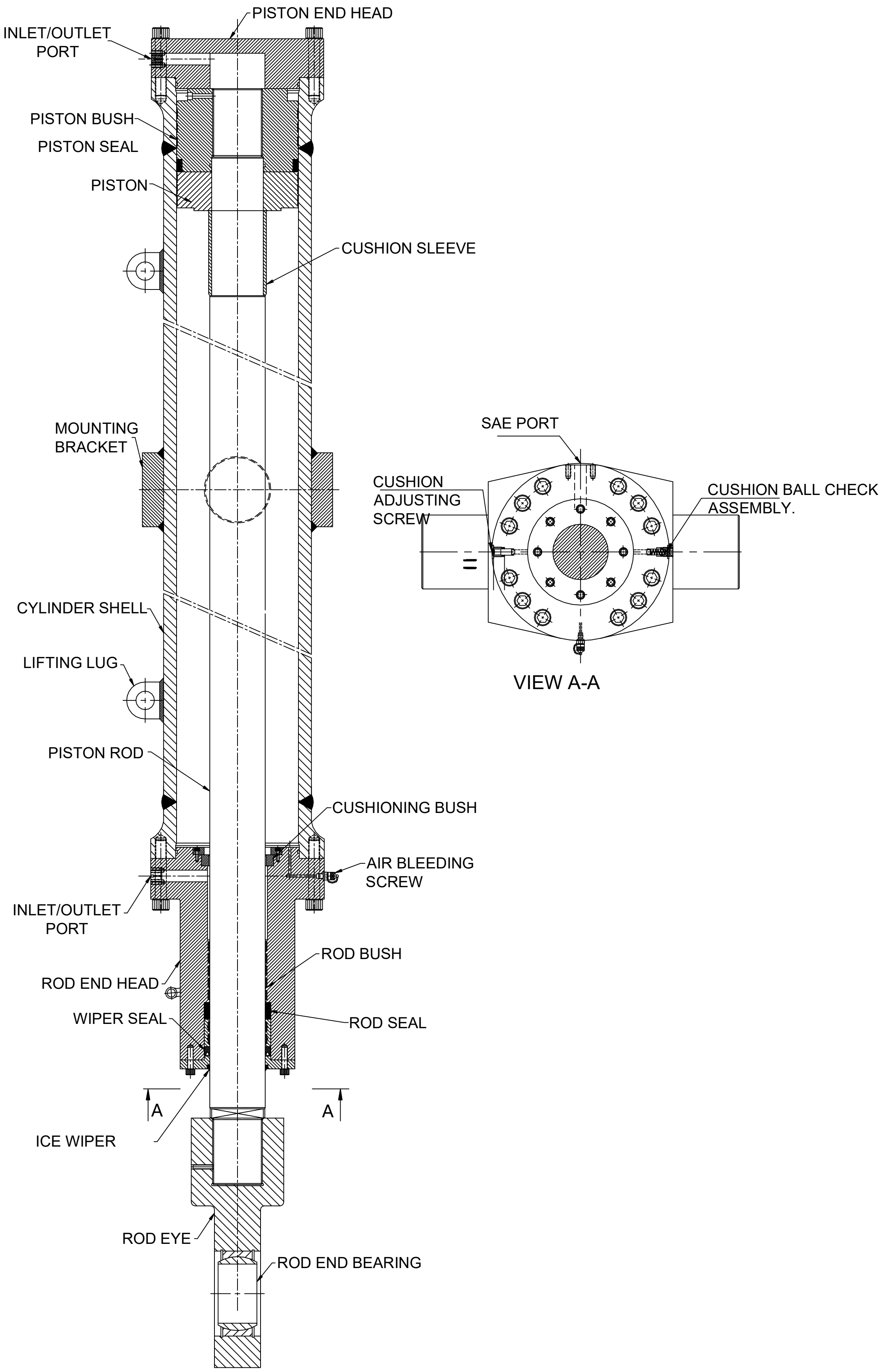

Figure 4 : Hydraulic Cylinder

xv) Test connections
xvi) Pressure switches
xvii) Hydraulic oil
xviii) Temperature Gauge
xix) Thermostat switch
xx) Oil level indicator
xxi) Oil level switch
xxii) Air breather
xxiii) Piping
xxiv) Push buttons, relays, limit switches and other electrical equipment for operating & controlling the system.

c) Hoist support structure
d) Support Bearings

13.1 Design Calculations of cylinder assembly

13.1.1 Cylinder

The operating pressure in the hydraulic cylinder is generally taken as the design pressure. The cylinder is designed as per IS:2825 and made out of cold drawn seamless steel tube preferably in single piece and the bore uniformly micro honed to a surface finish commensurate with the specified maximum leakage limit & long seal life. In any case it should not be coarser than 0.33 microns for smooth operation of the cylinder and long seal life. Flanges are welded at the ends of cylinder to bolt the cylinder heads. Alternatively, end pieces of higher thickness are butt welded to the cylinder. Butt welds are 100% radiographically tested for full strength. The cylinders are annealed for stress relieving before machining.

Strength calculation of the cylinder shell:

Minimum Wall thickness as per IS: 2825 CI.3.3.2.2

$$t = \frac{P\,D_i}{200\,f\,j - P} = \frac{P\,D_o}{200\,f\,j + P}$$

Where,

t = Shell thickness in mm

P = Design pressure in kg/ cm^2

D_i = Inside Dia. of the Shell

D_o = Outer Dia. of the Shell

f = Allowable stress in kg/mm^2

j = Joint Factor =1 for Seamless Tube

The shell thickness provided should be higher than the minimum allowable thickness computed for 3.3 factor of safety.

In addition to longitudinal, radial and hoop stresses developed due to pressure inside the cylinder, stresses due to gravitational forces and stresses due to friction at support/ mounting point should also be considered.

13.1.2 Cylinder Heads & Cap end Covers

These are designed as a thick flat plate held at the outer perimeter and made out of rolled steel accurately machined to ensure perfect alignment of piston rod bearings, piston rod, piston and tube etc. Extra strength due to shape of cylinder head / end cover is neglected.

Following procedure is adopted to determine cylinder Head/ cap stresses as per IS: 10210:-

a) Cylinder head (circular flat plate without hole as its centre) – **piston side**

$$Max\ \sigma_r = Max\ \sigma_t = \frac{3\,w\,a^2\,(m+1)}{8\,mt^2}$$

b) Cylinder head (circular flat plate with hole at its center) – **rod side**

at outer edge

$$Max\ \sigma_r = \frac{3\,w}{4\,t^2}\left[a^2 - 2b^2 + \frac{b^4(m-1) - 4\,b^4(m+1)\log\frac{a}{b} + a^2b^2(m+1)}{a^2(m-1) + b^2(m+1)}\right]$$

at inner edge

$$Max\ \sigma_r = \frac{3\,w\,(m^2-1)}{4\,mt^2}\left[\frac{a^4 - b^4 - 4a^2b^2\log\frac{a}{b}}{a^2(m-1) + b^2(m+1)}\right]$$

where

a	= outer radius in cm,
b	= inner radius in cm,
m	= reciprocal of Poisson's ratio,
σ_t	= tangential stress at surface of plate in N/mm^2,
σ_r	= radial stress at surface of plate,
t	= thickness of plate in cm,
w	= pressure in N/mm^2.

The greater of the two stresses should be used to determine the thickness of cylinder head.

13.1.3 Piston Rod

Piston rods are made of corrosion-resistant steel or forged steel cylindrically ground for accuracy. Outer surface of the piston rod, which contacts the piston rod guide bushing and seals, are ground and polished to a uniformly concentric finish having a surface roughness equal to, or better than 0.2 microns. The outer surface is hard chrome plated to at least 50 microns thickness to take care of damage & corrosion resistance. For application of forged steel rod in highly abrasive and corrosive environment such as silt flushing gate, the base layer of coating can be 150 microns thick and the top layer 200 micron thick.

The piston rod could have a High Velocity Oxy-fuel Flame (HVOF) sprayed Ni/Cr base layer and Plasma sprayed Cr_2 O_3 /Ti O_2 top layer. Hardness of the top layer may be provided in the range of 900 HV to 1000 HV and the surface shall be super finished to Ra=0.1 microns.

The coating should have resistance to corrosion and chemical attacks and special resistance to abrasive and erosive wear & tear. It should satisfy the corrosion tests according to the specified standards. (in particular to DIN 50021-SS 1000). Following tests may also be conducted on coatings:

- Bending tests
- Corrosion tests
- Scratch and wear test
- Seal behaviour test
- High Speed seal behaviour test
- Chemical exposure test
- Microscopic examination
- Impact test
- Roughness measurement
- Hardness measurement
- Adhesion test
- Hydraulic fluids

Connection of the piston rod to the piston: should be a rigid attachment and should permit disassembly for maintenance.

Permissible stress: Stresses in piston rod are limited to 0.4 of yield point at pressure setting of pump relief valve as per IS: 10210.

Double acting: In case of double acting hoist, the stem is also checked for buckling during closing cycle under compressive load.

Hoist stem for Radial Gate: Piston rod of radial gate should also be checked for bending due to gravitation force and moment due to bearing friction at support points. Following formula may be used to calculate bending moment in piston rod at a point when the gate is at just lifted position from sill:-

$$\mathrm{BM} = R_a L_x - \frac{w\,L_x^2}{2} + \sqrt{\left({F_{pull}}^2 + {R_a}^2\right)}\,\mu_1\, r_a\,\frac{(L_{ctc} - L_x)}{L_{ctc}} + \sqrt{\left({F_{pull}}^2 + {R_{pivot}}^2\right)}\,\frac{\mu_2\, r_{pivot}\, L_x}{L_{ctc}}$$

where

F_{pull} = Maximum pull load,

μ_1 = Coefficient of friction of rod eye bearing,

r_a = radius of rod eye bearing at moving surface,

R_a = reaction due to self-weight of hoist cylinder at rod eye $(\perp\ to\ axis)$,

L_x = distance from rod eye centre to bending moment point,

w = self-weight of rod per unit length $(\perp\ to\ axis)$,

L_{ctc} = distance between rod eye centre and pivot centre.

R_{pivot} = reaction due to self-weight of hoist cylinder at pivot $(\perp\ to\ axis)$,

μ_2 = Coefficient of friction of pivot bearing,

r_{pivot} = radius of pivot bearing at moving surface,

Loading and bending moment diagrams are shown in **Fig 5**.

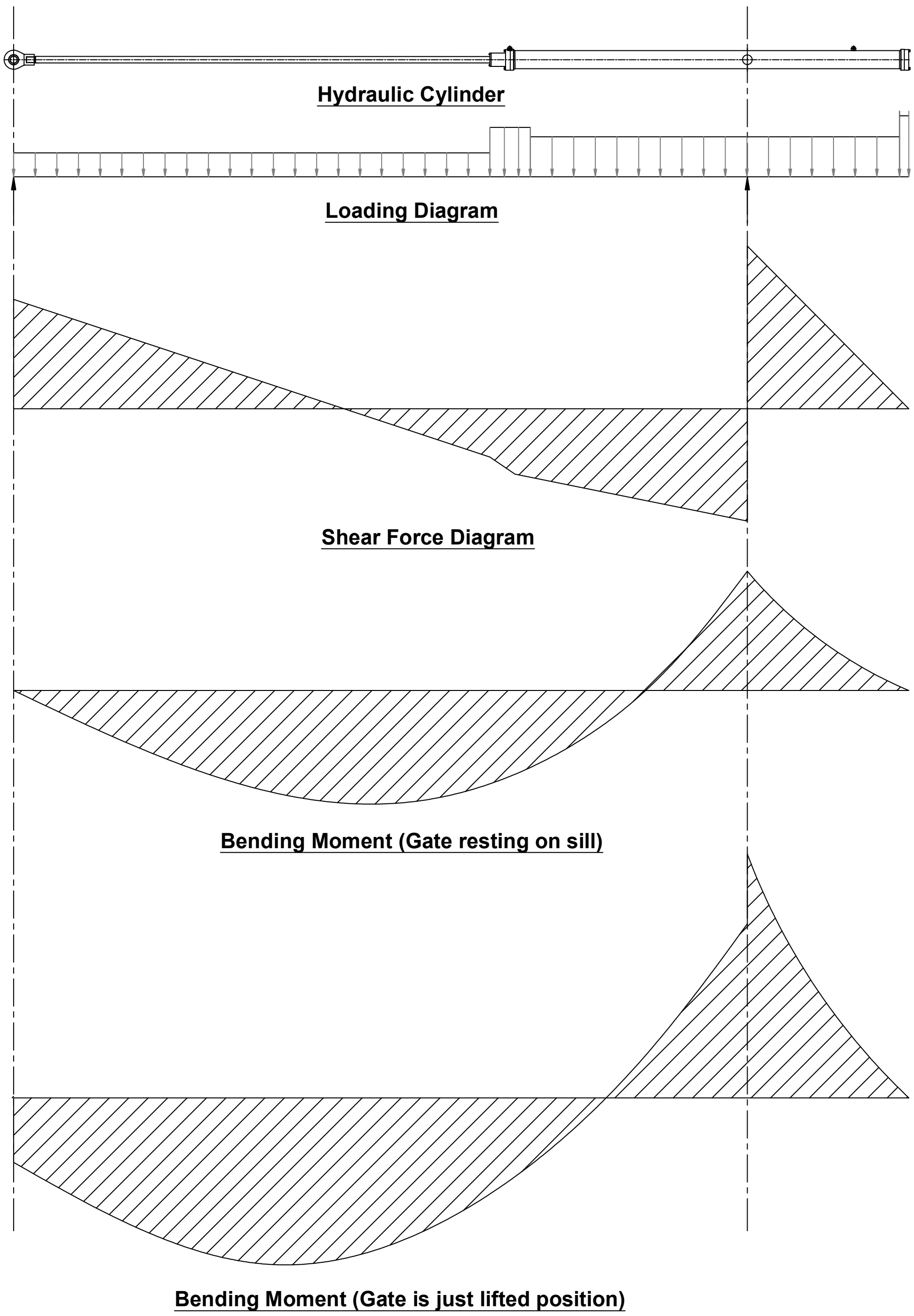

Figure 5 : Loading and bending moment diagrams of Radial Gate Hoist

Hoist cylinder of radial gate may be checked for maximum deflection due self-weight in extending position. Maximum deflection is so limited that there is no undue loading on the cylinder bushings and sealings. Counter weight may be provided if necessary, to limit the bending stresses and deflection.

13.1.4 Piston

The piston is designed for operating pressure and made of high grade forged or cast steel with large bearing area, pilot fitted to the piston rod and locked to it. The piston can be designed as circular disc fixed at inner edge and uniformly loaded. Following formula may be referred to calculate bending moment:-

$$M = \frac{qa^2}{C_1}\left[\frac{C_2}{2ab}(a^2 - r_0^2) - L\right]$$

where

M = Maximum bending moment in the plate,

q = load per unit area,

a = outer radius of piston,

r_0 = radius at start of distributed load,

$$C_1 = \frac{1}{2}\left[1 + \nu + (1 - \nu)\left(\frac{b}{a}\right)^2\right]$$

$$C_2 = \frac{b}{a}\left\{\frac{1+\nu}{2} ln\frac{a}{b} + \frac{(1-\nu)}{4}\left[1 - \left(\frac{b}{a}\right)^2\right]\right\}$$

$$L = \frac{1}{4}\left\{1 - \frac{1-\nu}{4}\left[1 - \left(\frac{r_0}{a}\right)^4\right] - \left(\frac{r_0}{a}\right)^2\left[1 + (1+\nu) ln\frac{a}{r_0}\right]\right\}$$

ν = poisson's ratio,

b = inner radius of piston,

13.1.5 Piston and Piston Rod Bearings

Bearing rings/bushes are provided in the piston as well as rod side head to transfer the lateral load due to jerk, misalignment, bending in case of radial gate hoist. Generally four to five bearing rings are provided in the piston as well as piston rod side head for hoists of vertical lift gate. However, these are higher in numbers for radial gate hoist as the bearings has additional load due to inclination of cylinder assembly.

Counterbalancing weights are also sometimes provided to minimise bearing pressures on the piston and piston guide rings, caused by the weight of the hoist assembly and contained hydraulic fluid in radial gate installations.

Replacing of piston rings and packing should be feasible by removal of piston side cylinder head for access and not of the hoist.

13.1.6 Seals and Wipers

Static Seals: Packings or static seals such as 'O' rings should be provided between all connected parts where leak-tight joints are required, such as between cylinder tube and heads or between piston and piston rod.

Piston seals and piston rod seals: Chevron type packing (V-packing) seals are generally used for piston and for piston rod, mechanically locked in place. Other types of seal such as 'O' ring with V packing which have better sealing capability can also be used. Seals should be capable to resist roll, turn, and extrusion. On hoist cylinders designed for fluid pressure acting from either side, a separate set of piston seals are provided on each side. Rod seals are V type fabric reinforced natural rubber rod seal packing with male & female headers having wear compensating lips to prevent rod end leakage.

Wiper Scraper and Ice Wiper: Wiper Scraper of Nitrite rubber lip type is used to remove foreign matter from piston rod to prevent damage to the V-packing. This prevents dust from entering & scoring the cylinder internal surface. Ice wiper is also used in applications where standard elastomer wipers wear too quickly as a result of contaminants adhering to the piston rod.

For radial gate cylinder assembly, the seals should be capable of sealing even when the forces and deflection due to bending are maximum i.e. when gate is fully closed. Cylinder and piston assembly should also be checked for these forces.

'O' rings: For static seals 'O' rings are used. The seals are selected from seal manufacturers' catalogue. The sizes of groove for 'O' ring should be selected carefully for proper sealing of matting surfaces. For general guidance compression rate of 'O' ring may be provided in the range of 8% to 30%. Compression rate is given by:

$$Compression\,rate = \frac{d' - h}{d'}\,X\,100$$

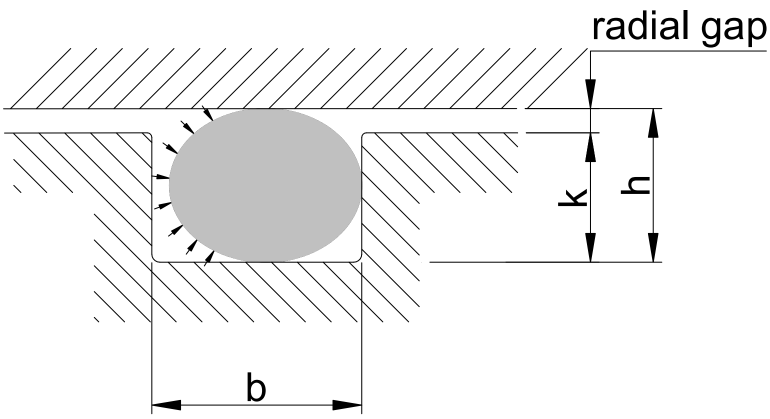

Figure 6 : 'O' ring groove for cylindrical sealing

Where d' = 'O' ring cross section diameter

h = depth of groove including radial clearance of matting parts = k + radial gap

k = depth of groove

Cylinder expansion due to oil pressure should also be considered for selection of radial gap / clearance. In case the radial clearance is higher, back ring may be used for the 'O' ring.

Groove width may be provided such that the 'O' ring doesn't occupy more than 90% of groove space. Occupancy percentage can be calculated by:

$$\text{Occupancy percentage} = \frac{\pi(\frac{d'}{2})^2}{bh} \text{ X } 100$$

b = width of groove

Surface roughness of better than R_a = 1.6 micron should be provided on the sealing surface for static sealing. However, better surface finish may be required for dynamic sealing. Chamfer should be provided on the surface where the 'O' ring enters. The chamfer amount, n (refer **Fig. 7**) should be higher than the projection of 'O' ring from the groove in uncompressed condition and the chamfer angle may be provided as 15^o to 20^o.

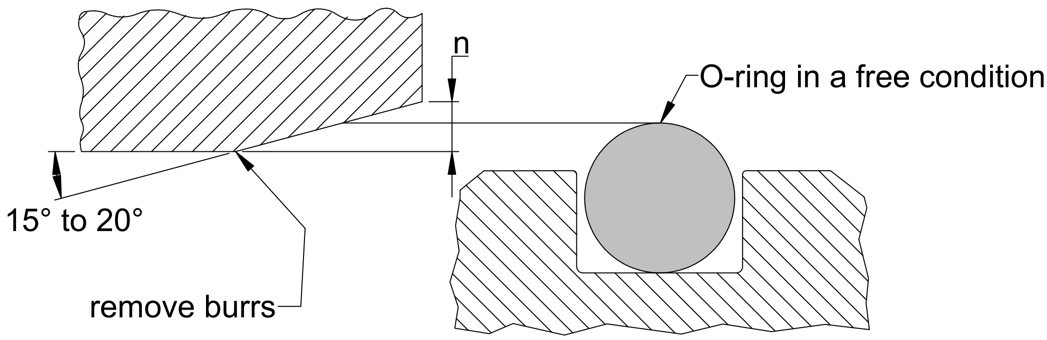

Figure 7 Chamfer in radial direction

As a general guidance, Table 1 may be followed for chamfer size in radial direction or as required by the 'O' ring manufacturer for effective sealing.

'O' ring cross section diameter		Minimum chamfer in radial direction (on surface where 'O' ring enters) "n"
over	upto	
-	2.4	0.9
2.4	3.5	1.1
3.5	5.7	1.3
5.7	8.4	1.5

Units: mm

13.2 Hanger Stud

A hanger stud is provided to lock the piston in fully retracted position. In case of the gate which is generally kept in fully lifted position, the piston is locked by hanger stud; hence the gate is locked at fully lifted position. The Hanger Stud can be provided in radial gate hoist cylinder to support the weight of piston and piston rod in fully retracted position (during transportation and for replacement of piston rod seal).

The hanger stud is designed to support the weight of the piston, piston rod and/or gate. For automatic operation of hoists, hydraulically operated hanger stud may be provided with spring loaded latches so that the piston is automatically locked up in stop stroke position every time. The lock of stud is automatically opened hydraulically when the push button is pressed for lowering operation. Typical arrangement of hanger stud is shown in **Fig 8**.

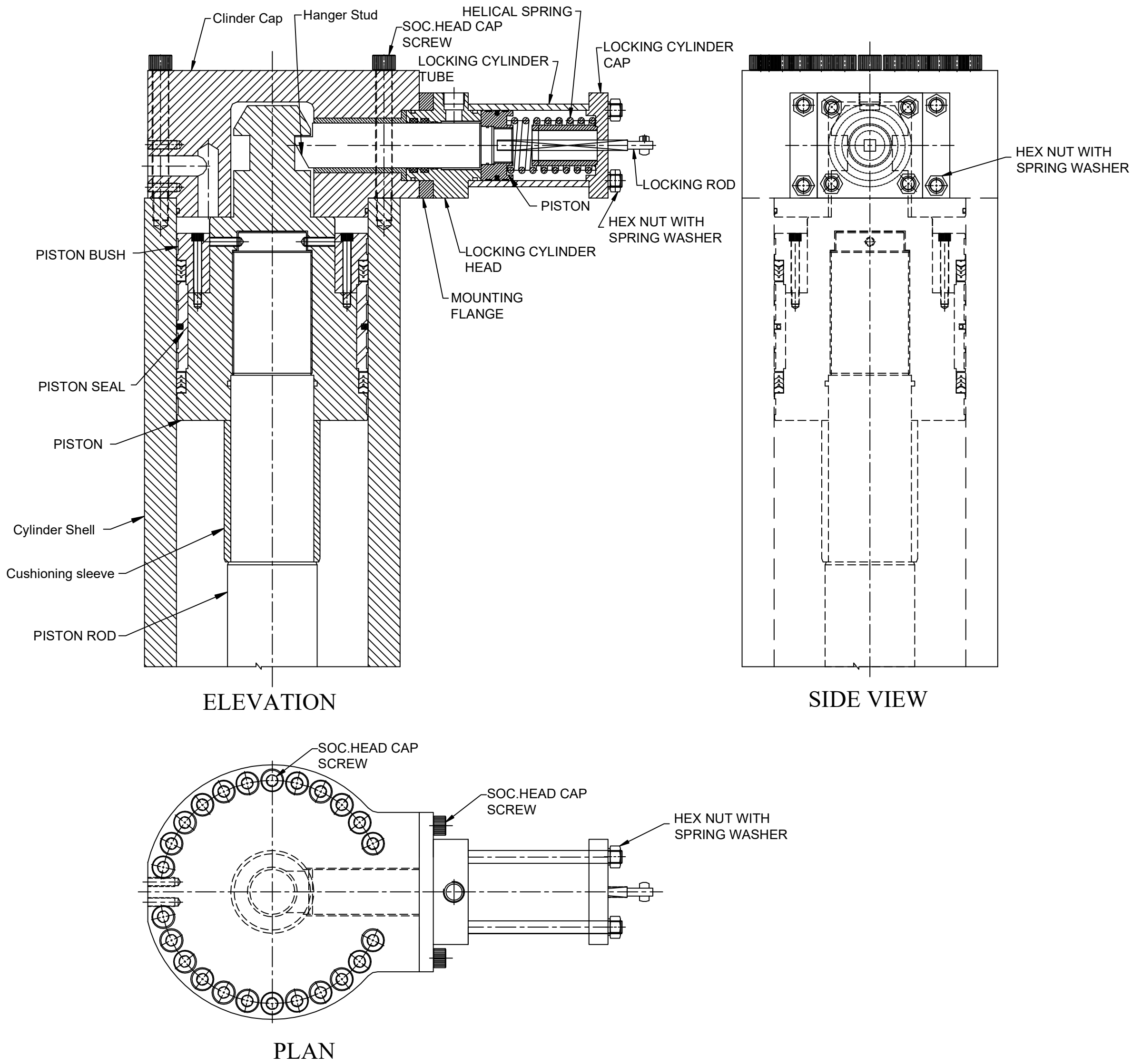

Figure 8 : Typical Hanger Stud

13.3 Hoist Support Frame

Hoist Support Frame is designed to withstand the maximum load occurring at the time of operation of the gates with 1.1 impact factor. Pivot support is designed to safely take the load of hoist when operated at designed capacity including weight of hoist and oil. It should also be checked for safe stresses in case of maximum pressure in the cylinder resulting from jamming of gate. Higher stresses may be allowed for checking of hoist support in occasional condition such as of jamming of gate. The hoist support should permit removal of the hoist cylinder without dismantling either the hoist support frame or trunnion.

Vertical lift gate hoist cylinder may be either flange mounted or pin mounted which is supported by structure of beams. The support beam for radial gate hoist is generally fabricated from seamless pipe or rolled pipe as the loading on the cylinder support changes with lifting of gate. Seamless pipe is preferred over rolled and welded pipe for better reliability. Longitudinal weld of rolled pipe should be tested radiographically for full strength otherwise weld efficiency may be taken. Typical arrangement of hoist support for radial gate installation is shown in **Fig 9**.

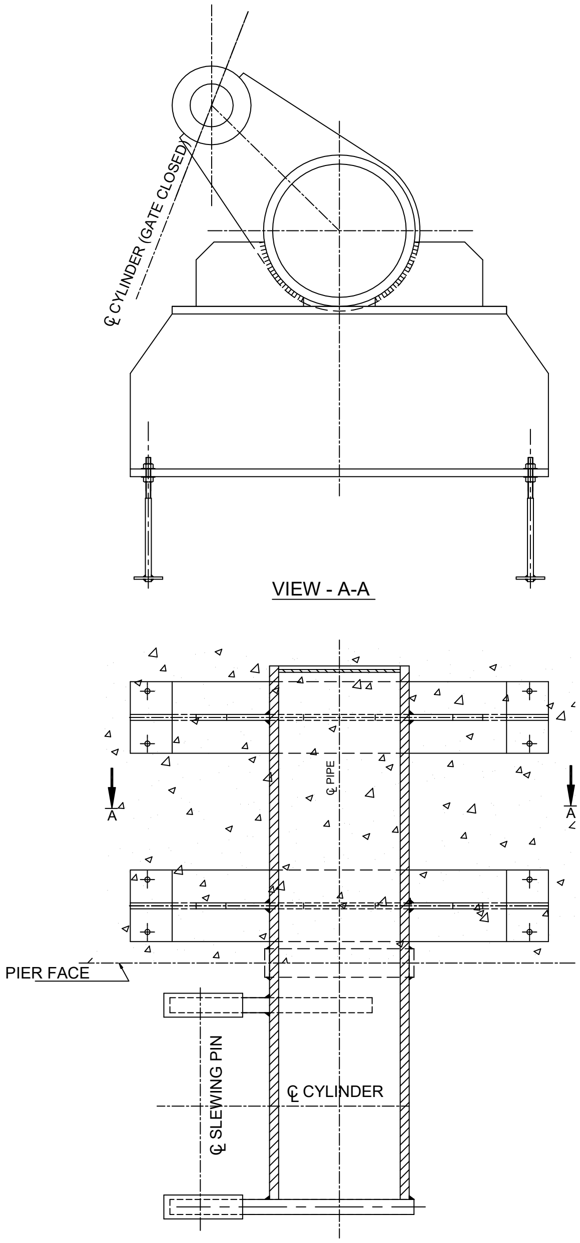

Figure 9 : Radial Gate Hydraulic Hoist Support

13.4 Access Ladders and Platforms

Wherever necessary, access ladder for providing access to the top of each hoist cylinder, along with support platform and guard railing should be provided so as to permit approach to the top of each hoist for inspection and maintenance.

14 Hoist control modules

Separate hoist control modules are required to be provided for each gate. However, they may be interconnected such that it is possible to operate two gates with one module in case necessary. The hoist control module consists of hydraulic power unit and an electrical control cabinet. Symbols of elements in the hydraulic circuit should be used as per IS:7513.

14.1 Hydraulic Circuit for Radial Gate

The hydraulic power unit include oil tank, two electric motor driven pumps, manual pump, oil reservoir, automatic controls, pressure relief valves, check valves, flow control valves, directional control valves, logic valve, pressure gauge, temperature gauge, oil level switch, oil filters, strainers, air breather, piping and accessories. The hydraulic control module is generally mounted on the oil tank and hoist cylinders. Typical hydraulic circuit for radial gate hydraulic hoist is shown in **Fig 10.** The system is capable of working in different operating conditions i.e. idle, gate opening, gate closing, emergency operation creep, jamming of gate and gravity closer.

14.1.1 Idle

With all the solenoids in de-energized condition, the electric motor(s) started. The pump output flows freely back into the reservoir through the pressure relief valve (PR1) and return line filter.

14.1.2 Gate Opening

To open the gate, press the “GATE OPEN” push button, which starts the pump operated by electric motor(s) in the system. After a few seconds time delay, the solenoid of ‘Solenoid operated Pressure Relief Valve’ (S1) and direction control (DC) valve (S2) get energized. This will helps the oil to flow from the pump to the piston rod side of both the hydraulic cylinders through Pressure line filter, DC valve and flow control valve. The piston rises and pushes out the flow from the piston side of cylinder through the return line filter.

The speed of lifting of gate can be controlled by the flow control valve. The hydraulic accumulator with safety pressure relief valve (PR5) is connected to pressure line to avoid pressure fluctuations and jerk. The pressure relief valve (PR1) limits the operating pressure and protects the system from overload. Once the gate reaches to fully open position the PLC will get the signal from angle position sensor or limit switch mounted on the gate assembly. At this point the solenoids of DC valve (S2) and Solenoid operated Pressure Relief Valve (S1) are de-energized, the piston stops moving and system reverts to idle.

14.1.3 Gate Closing

To close the gate, press the “GATE CLOSE” push button, which starts the pump operated by electric motor(s) in the system. After a few seconds time delay, the solenoid of Solenoid operated Pressure Relief Valve (S1) and DC valve solenoid (S3) get energized. Oil from pump is directed to piston side of the cylinder through check valve, pressure line filter and directional control valve. The pressure in the piston side is reduced by pressure relief valve (PR2). The pilot line also gets activated and this pilot pressure will operate the pilot operated check valve. The oil from rod end side of hydraulic cylinder will flow to the piston side of the cylinder through logic valve. The speed of the gate closing can be changed by adjusting the flow control valve. Once the gate is fully closed electrical signal will be cut off and the solenoids get de-energized, the piston stops moving and system reverts to idle.

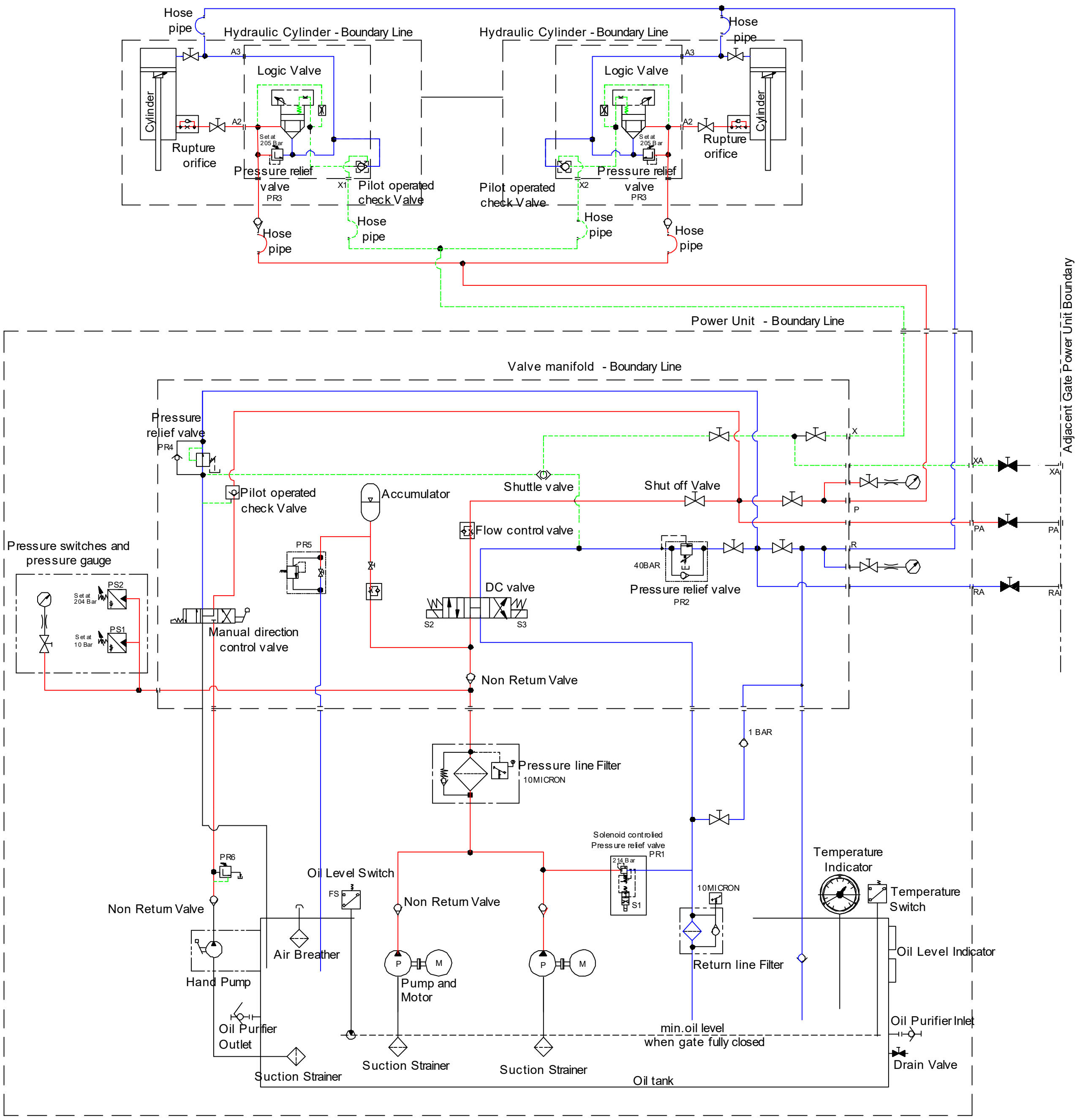

Figure 10 : Typical Hydraulic Circuit for Radial Gate Hoist

14.1.4 Emergency Operation

The gate can be operated in the event of power failure with the help of hand operated pump.

a) Opening of Gate - By operating the manual pump and manual direction control valve, the oil can be directed to rod end of cylinder to raise the piston. The piston side oil will flow to tank through return line filter.

b) Closing of Gate - The pressurised oil from manual pump will flow to pilot line through manual direction control valve which will operate the pilot operated check valve and hence the logic valve will direct the rod end oil to piston side of the cylinder. The oil from the tank can also flow directly to piston side through non return valve as required.

14.2 Hydraulic Circuit for Vertical Lift Gate

The hydraulic power unit include oil tank, two electric motor driven pumps, manual pump, oil reservoir, automatic controls, pressure relief valves, check valves, flow control valves, directional control valves, pressure gauge, temperature gauge, oil level switch, oil filters, strainers, air breather, piping and accessories. The hydraulic control module is generally mounted on the oil tank and hoist cylinders. Typical hydraulic circuit for hydraulic hoist of vertical lift gate is shown in **Fig 11.** The system is capable of working in different operating conditions i.e. idle, gate opening, gate closing, emergency operation creep, jamming of gate and gravity closer.

14.2.1 Idle

With all the solenoids in de-energized condition, the electric motor(s) started. The pump output flows freely back into the reservoir through the pressure relief valve (PR1) and return line filter.

14.2.2 Gate Opening

To open the gate, press the “GATE OPEN” push button, which starts the pump operated by electric motor(s) in the system. After a few seconds time delay, the solenoid of Solenoid operated Pressure Relief Valve (S1) and DC valves solenoid (S2, S4) get energized. This helps the oil to flow from the pump to the piston rod side of the hydraulic cylinders through Pressure line filter, DC valves and flow control valve. The piston rises and pushes out the flow from the piston side of cylinder through the return line pressure relief valve and return line filter.

The speed of lifting of gate can be changed by the flow control valve. The hydraulic accumulator with safety pressure relief valve is connected to pressure line to avoid pressure fluctuations and jerk. The pressure relief valve limits the operating pressure and protects the system from overload. Once the gate reaches to fully open position the PLC will get the signal from limit switch. At this point the solenoids of DC valve and Solenoid operated Pressure Relief Valve are de-energized, the piston stops moving and system reverts to idle.

14.2.3 Gate Closing

To close the gate, press the “GATE CLOSE” push button, which starts the pump operated by electric motor(s) in system. After a few seconds time delay, the solenoid of Solenoid operated Pressure Relief Valve (S1) and DC valve (S3, S5) get energized. Oil from pump is directed to head end of the cylinder through pressure line filter and directional control valve. The pressure in the piston side is reduced by pressure relief valve (PR3). The pilot line also gets activated by another DC valve that will operate the pilot operated check valve. The oil from rod end side of hydraulic cylinder will flow to piston side of the cylinder. The speed of the gate closing can be changed by adjusting the flow control valve. Once the gate is fully closed electrical signal will be cut off and the solenoids get de-energized, the piston stops moving and system reverts to idle.

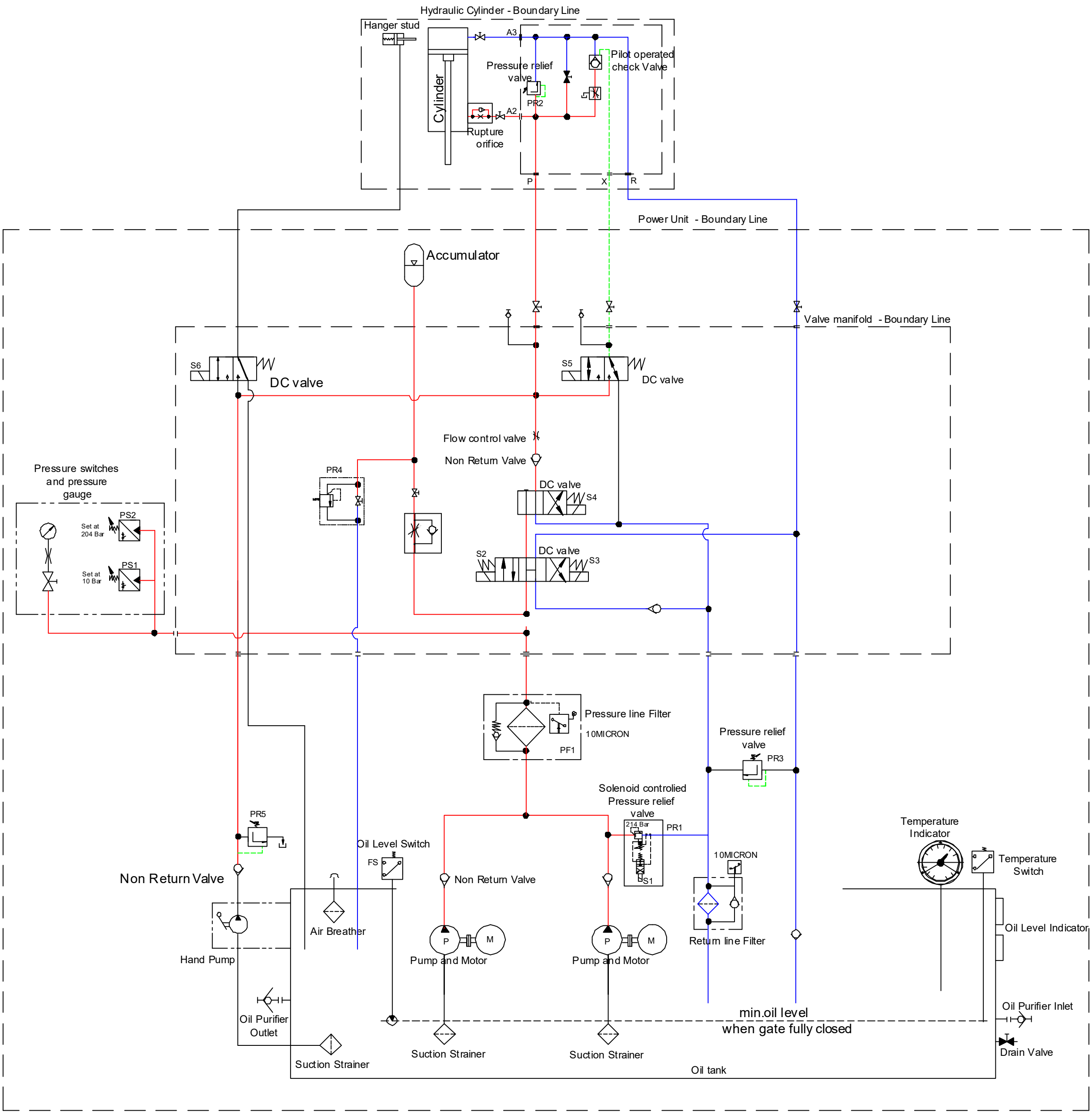

Figure 11 : Typical Hydraulic Circuit for Vertical Lift Gate Hoist

14.2.4 Emergency Operation

The gate can be operated in the event of power failure with the help of hand operated pump.

a) Opening of Gate - By operating the manual pump, the oil can be directed to rod end of cylinder to raise the piston. The piston side oil will flow to tank through return line filter.

b) Closing of Gate - The pressurised oil from manual pump will flow to pilot line through Pressure Relief valve which will operate the pilot operated check valve and hence the oil from rod end side will flow to the piston side of the cylinder.

14.3 Special Features

Some important aspects of the system other than normal operations are as described below:

14.3.1 Creep

When the gate is left standing in any intermediate position for long time, it is quite possible that under the weight of the moving parts, oil from the rod end side will leak past the piston to the piston side. The leakage rate may be very slow but if it occurs, the Gate will slowly creep down. Therefore it is also suggested to have the provision of auto creep restoration and alarm system when creep crosses the predetermined limit.

It may be achieved by deploying a limit switch/ position sensor/ tilt angle sensor/transducer/ tilt switch attached with gate assembly or hoist. The tilt angle sensor used for measuring the tilt of an object in multiple axes with reference to an absolute level plane. In case of the Hydraulic hoist of radial gate the cylinder movement is along the one axis only. Tilt switch needs to be calibrated as per tilt span of hoist for full stroke of cylinder and accordingly the logical program can be embedded in the PLC of control panel.

The position switch/ sensor send signal to control panel which starts the motor and pump assembly to restore the gate in its original position. The indication of creep is also displayed on local control panel.

14.3.2 Jamming of Gate

In case the gate jams and it is beyond the capacity of hydraulic hoist to lift it, pressure will rise to the setting of the pressure relief valve and the pump oil will return to tank without overloading the hoist system. However, to enable the operator to know that such a jamming has taken place, a pressure switch can be usefully employed. The pressure switch can sound an alarm or activate some other device in case the system pressure reaches the pre-set value. The pressure exerted by the hoist can be seen directly from the pressure gauge.

14.3.3 Holding up of Gate in any Intermediate Position

During normal operation, the gate will travel its complete length of stroke when appropriate push button is depressed. However, it is also possible to have a selection switch in the electrical panel, which will enable the gate to be moved only as long as the push button is kept depressed. In other words, the gate can be stopped in any intermediate position.

14.3.4 Gravity Closure

In case the gate is to be closed under gravity, the hydraulic system should be capable of closing the gate. In this case weight of moving parts will generate the oil pressure in rod end side which will be directed to piston side. Oil will also flow from the oil tank to the piston side of the cylinder.

14.4 Oil Tank

As per IS: 10210, oil tanks should have storage capacity considering the following:

a) Oil fully evacuated from one gate cylinder/cylinders at a time;
b) Displacement of oil due to piston stem of all the hoists;
c) 200 liters of spare oil or volume of one cylinder whichever is less; and
d) Free air spare equivalent in volume to 200 litres of oil or volume of one cylinder, whichever is less.

Additional capacity should also be provided for the volumetric displacement of hoist stem and for temperature produced volume changes. In case a central oil tank is provided for all the gates at an installation, the capacity of oil tank should be sufficient to meet the above requirements, with each requirement being met independently.

Oil tank should be properly painted to ensure cleanliness and to avoid rusting. The tanks should be provided with breather openings. Provision should also be made to drain water accumulations from the lower points in the oil tank and hoist cylinder, It should be of robust steel construction and suitable for floor mounting. It should be equipped with a sight fluid level gage, minimum oil level indicating device, dial type thermometer to indicate fluid temperature, valve drain connection, a magnetic plug type drain arranged to permit complete drainage, a filler pipe provided with a strainer, a desiccant filled breather cap with filter, and pipe connections to a fluid purification device. The filler cap should be a combination of air vent, dust screen and air filter. The oil tank should be provided with lifting and jacking lugs as required for its handling.

14.5 Motor driven pumps

Two motor driven oil pumps should be provided for the operating system to ensure the operation of gate or valve, in case one motor-pump unit fails. The oil pumps should be of the self-priming, positive and constant displacement type. As per the suitability, vane/gear/piston type oil pumps may be used.

The motor should conform to specified horsepower, speed. It should be totally enclosed, flame proof, air cooled, direct driven with normal starting torque and low starting current, continuous rating, three phase squirrel cage induction type. The motor winding should have insulation specially treated to withstand wet and humid conditions, and should be suitable for the required altitude.

14.6 Manual Pump

Lever operated piston type manual pump is used to operate the gate in case of power failure. The pump may be mounted on the front of the hydraulic fluid reservoir. Effort required to achieve nominal design system pressure may be limited to 14kg.

14.7 Valve Manifold Block

Hydraulic control valves are generally mounted on valve manifold block. One manifold block may be provided at the oil tank and another on each hydraulic cylinder. In case pipe mounted hydraulic control valves are used, special framed structure is provided on the oil tank to mount them. In both the cases, the components should be arranged in an orderly and functional manner, and they should be removable without removing the surrounding components and piping.

14.8 Hydraulic valves

14.8.1 General

The valves and fittings should have JIC or equivalent pressure ratings not less than the maximum system pressure. Electrically operated valves also have provisions for standby manual operation. The valves are selected from the manufacturer catalogue for the pressure and flow requirements. The valve operation should be smooth and without any jerk.

14.8.2 Check valves

Check valves should be spring loaded for closure with minimum shock. Check valve allow the flow in one direction only. Pilot operated check valves are used where one direction flow is free and other side flow is allowed only when pressure is applied. Check and pilot operated check valve are shown in Fig 12 and 13.

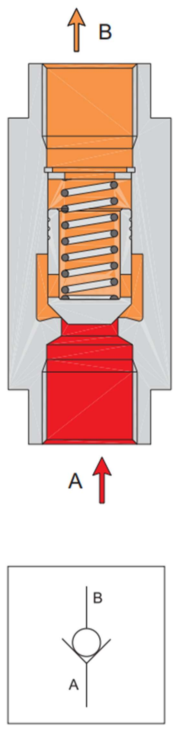

Figure 12 Check Valve

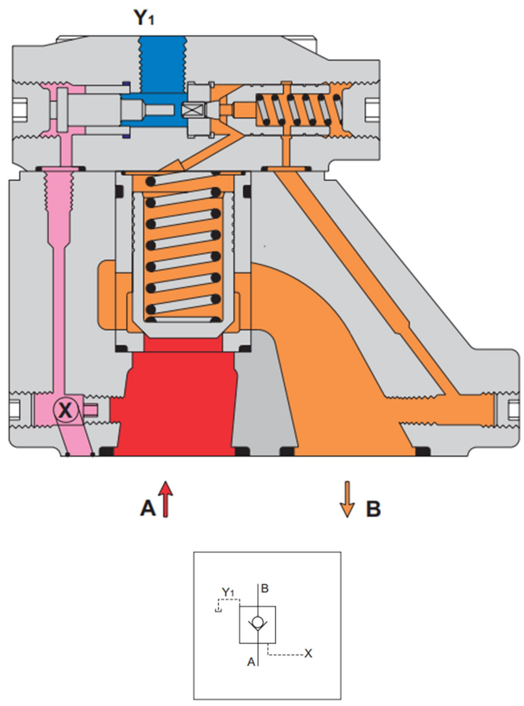

Figure 13 Pilot Operated Check Valve

14.8.3 Shut-off valves

Shut-off valves may be ball valves. These are provided to shut off the line as required. Shut off valve with hydraulic symbol is shown in **Fig. 14**.

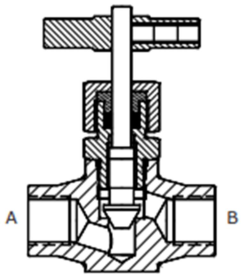

Figure 14 Shut off Valve

14.8.4 Pressure relief valves

Pressure relief valves are hydraulically operated type. The valves should be adjustable and should maintain the pressure within 5% of the pre-set value.

Pressure cut off valve should also be used to safeguard the hydraulic system in case pressure relief valve get stuck up. The pressure cut off valve may be set at 10% higher pressure than the pre-set value of pressure relief valve. Typical pressure relief valve with hydraulic symbol is shown in Fig 15.

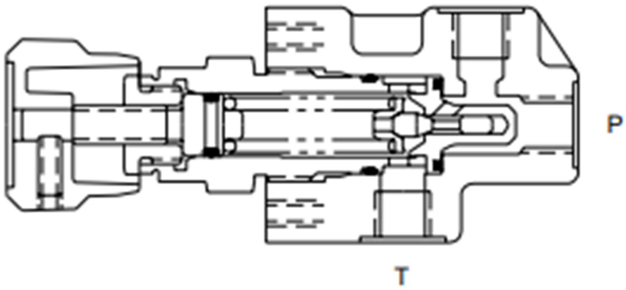

Figure 15 Pressure Relief Valve

14.8.5 Directional Control Valves

Direction control of different types is used to direct the flow in particular direction. During lifting the pressurised oil is directed to lower side and low pressure oil flow from upper side is directed to tank through return line filter.

Directional control valves should be rated for zero leakage. There should be minimum jerk in the system. The valve should be able to work at maximum system pressure and of required flow rate. These are generally magnetic control operated by electricity and should be provided with manual override. Normally three position valves are provided. Typical directional control valve and its functional symbols are shown in Fig 16.

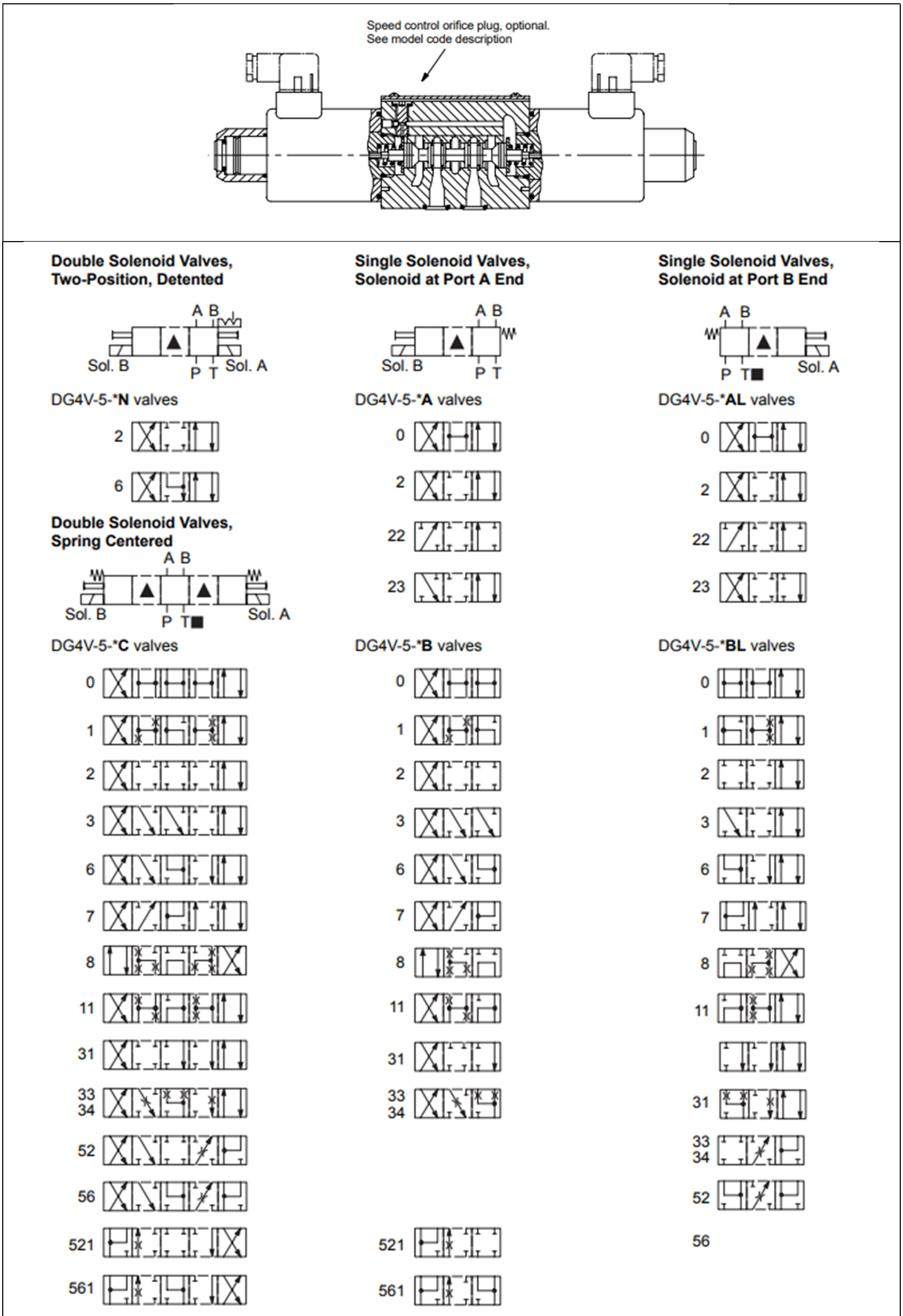

Figure 16 Directional Control Valve

14.8.6 Flow control valves

Flow control valves are used adjust the speed of gate by controlling the oil flow. These valves should be of adjustable pressure and temperature compensated type with an integral check valve for free return. The flow control valves may be shop tested to pass the required flow within 5%. Typical flow control valves with symbols are shown in **Fig 17 and 18**.

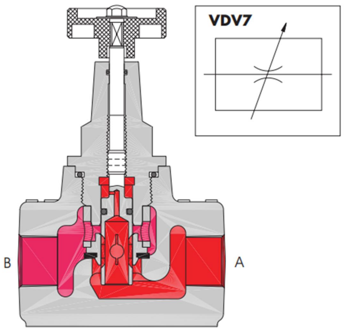

Figure 17 Flow Control Valve

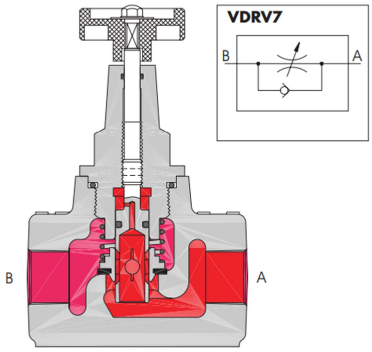

Figure 18 Unidirectional Flow Control Valve

14.8.7 Accumulators

Bladder type hydraulic accumulators may be used to absorb the jerk in the hydraulic system. The accumulators are pre-charged with nitrogen. The accumulator should be provided with an integral pressure relief valve, shut-off valves for isolation, removal, and drainage, and a valve charging connection.

14.8.8 Filters and Strainers

Generally filters are used in pressure line and in return line to protect the system from any foreign material which can otherwise affect the working of different valves and damage the parts of the hoist cylinder. Suction strainer is also used to protect the pump. Filters should be of the disposable, replaceable element type. Strainers should be of the cleanable, replaceable element type. Elements of all filters and strainers are preferable stainless steel or Monel and woven or wound wire.

Filter should be provided with a bypass valve which opens to pass the flow when the pressure drop across the filter element exceeds the allowable limit. The filter may be provided with a means to indicate the condition of the filter element by visual inspection.

Filter should also have a differential pressure or minimum pressure switch to energize a warning light when the pressure drop across the filter or strainer reaches a predetermined value.

Filter Media Rating Requirements: Filter elements may be screw on type and have following absolute ratings.

- Pump suction filter : 150 microns
- Low pressure line strainers : 160 microns.
- Pressure filters : 10 microns.
- Return line filters : 10 microns.

Pressure Drop Requirements: The maximum pressure drop across any clean filter or strainer at normal flow may be limited to following values in the normal operating temperature range:

- 0.0703 kg/cm^2 for low-pressure line strainers.
- 0.3515 kg/cm^2 for pressure filters.

By-pass Valves: The filters should be provided with bypass valves which set to open when the pressure drop across the filter elements exceeds 1.0545 kg/cm^2.

14.8.9 Pressure Gauges

Bourdon tube types with glycerine filled housing pressure gauges are generally used. The accuracy of pressure gauges should conform to Grade A of ANSI B40.1. The total measuring range of the gauge may be selected between 110 and 200% of the maximum pressure expected and may be provided with corrosion resistant shut off valves and snubbers.

14.8.10 Test Connections

The power unit should be provided with test connection at appropriate locations for attaching a pressure gage or transducer. The test connection is provided with a corrosion resistant steel minimum type connector equipped with check valves.

14.8.11 Air Vent

When the cylinders are initially installed, it is necessary to first vent the air out from the cylinder. This is done by opening air vent valve and raising the piston under pressure to push out the air on top of it. When the piston has reached the top of its stroke, vent valve can be opened to release any air trapped below the piston. After removing the air the vent valves are shut off. Similarly, when the cylinders are emptied of oil during maintenance, the air is to be removed while re-commissioning the unit.

14.8.12 Pressure Switches

Heavy duty piston type pressure switches may be provided as required.

14.9 Position Indicator

A rotary or linear position indicator/ sensor can be linked to the moving parts of the gate to indicate its position.

14.10 Hydraulic Oil

Hydraulic oil should be compatible to hydraulic elements used.

14.11 Locking of Bolted Connections

All screws, bolts, and nuts should be provided with a locking device. Selection of means may consider service conditions according to the following guidelines:

On parts which may be subject to vibration and on connections which require repeated adjustment or which are not readily accessible (such as bolts or nuts inside of a hoist cylinder), a positive and reusable locking device should be provided on bolts, such as a castellated nut with cotter pin or a soft copper locking plate or washer bent over the nut and a fixed point on the structure.

On other parts, either a positive reusable locking device or spring steel lock washers may be used on the bolts. Lock washers may be used in combination with a chemical thread locking compound when required. The selection of the compound should consider the requirements for disassembly and maintenance.

14.12 Flushing

14.12.1 Flushing Hydraulic Cylinders

Immediately before the operating cylinders for the gate are connected to the hydraulic system, the cylinders should be filled with hydraulic oil, and both piston side and rod end side of the cylinder should be flushed to remove the foreign matter by mechanically moving the piston back and forth for its full stroke.

14.12.2 Flushing piping

Hydraulic piping should be flushed by circulating hydraulic oil until returning oil meets ISO 4406, Code 19/16/13 requirements or equivalent.

14.12.3 Flushing Hydraulic Power Unit

Hydraulic power unit should be flushed preferably in the supplier’s shops. The hydraulic tank is filled with hydraulic oil and the oil filtration system actuated with a 10 micron filter. The oil should be circulated and filters changed until the unit runs for 24 hrs without becoming clogged.

14.12.4 Flushing Valve Manifolds

Prior to installation the valve manifold should be flushed by circulating hydraulic oil through all ports until the cleanliness of the return meets ISO 4406, Code 19/16/13 requirements.

14.13 Handling Provisions

All parts, components, and assemblies heavier than 14kg should be provided with suitable provisions for handling, such as eyebolts, lugs, hooks, tapped holes for eyebolts, or holes with rounded corners for passing slings.

Annexure -1

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36. Research Article No. J Therm Spray Tech (2022) 31:1443–1475 by Garima Mittal and Shiladitya Paul on “Suspension and Solution Precursor Plasma and HVOF Spray: A Review” – Springer

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