7. Mechanical and Electrical Equipment

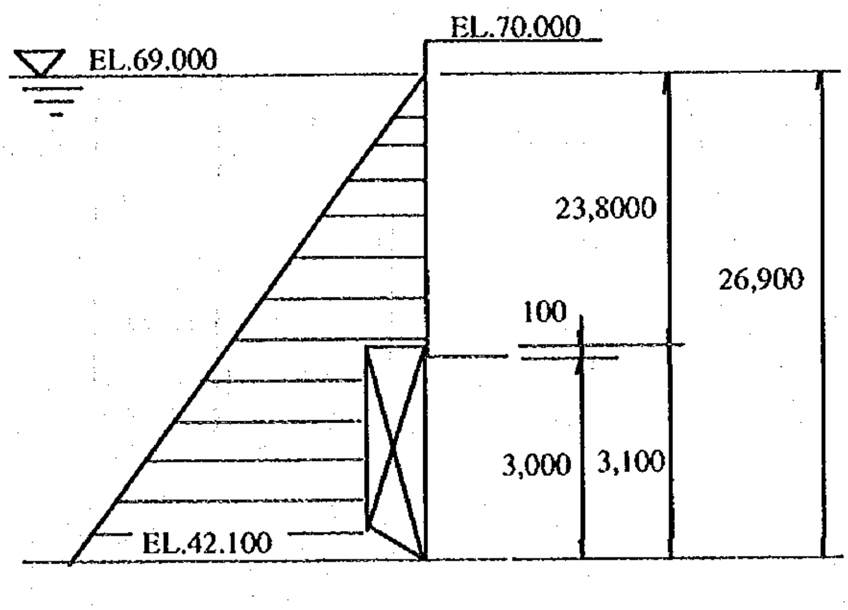
7.1 Severino Pumping Station

SEVERINO PUMPING STATION
INTAKE GATE AND GANTRY CRANE

1. Design Condition

Type	:	Steel made fixed-wheel gate
Quantity	:	Two (2) sets
Clear span	:	6.0 m
Clear height	:	3.0 m
Flood water level	:	EL.69.000 m
Sill elevation	:	EL.42.100 m
Design head	:	26.9 m
Sealing method	:	4 edges rubber seal at downstream face of gate
Maximum deflection of main horizontal beams	:	1 / 1000 of supporting span
Corrosion allowance	:	1.0 mm for skin plate and main structural members (usually asumed in air on the dogging device)
Type of hoist	:	Electrically driven wire rope wound type gantry crane
Operating speed	:	1 m/min. ± 10 %
Hoisting height	:	28.5 m
Operating method	:	Remote control from cabine

2. Hydraulic Load



$$P_T = \frac{1}{2} \times \left(H_2^2 - H_1^2\right) \times B \times G_W$$

where,

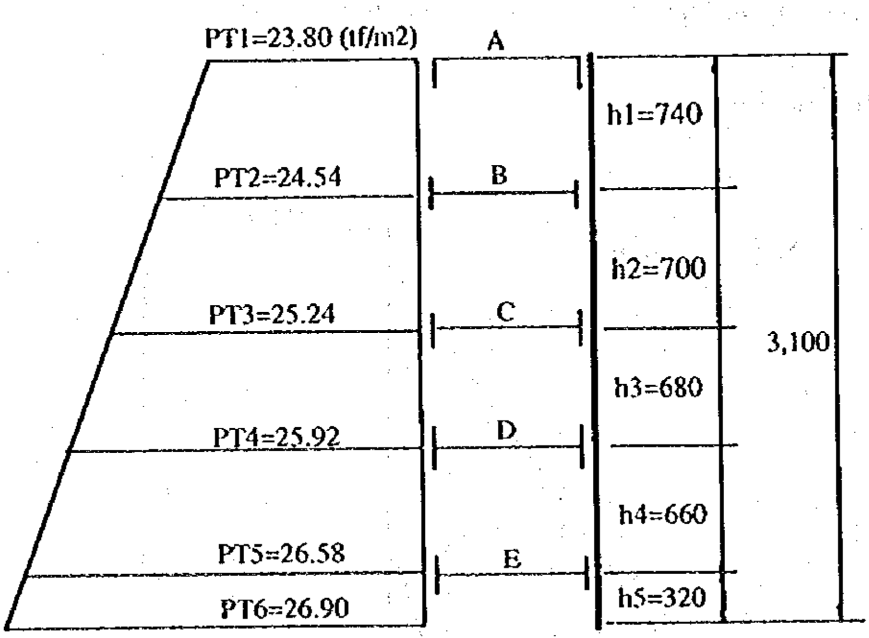
P_T	:	Hydraulic load (tf)	
H_1	:	Design head at gate top	23.800 m
H_2	:	Design head at gate bottom	26.900 m
B	:	Sealing span	3.100 m
G_W	:	Specific gravity of water	1.000 tf/m^3

Thus,

P_T = 0.5 x (26.9^2 - 23.8^2) x 3.1 x 1.0
= 243.614 tf

3. Main Horizontal Beams

(1) Arrangement of main horizontal beams
Six numbers of main horizontal beams are arranged as follows :



(2) Charging load on each beam

Charging load acting on each beam is calculated by the following equations :

$$Beam\ A = \frac{(2P_{T1} + P_{T2})}{6} \times h_1$$

$$Beam\ B = \frac{(P_{T1} + 2P_{T2})}{6} \times h_1 + \frac{(2P_{T2} + P_{T3})}{6} \times h_2$$

$$Beam\ C = \frac{(P_{T2} + 2P_{T3})}{6} \times h_2 + \frac{(2P_{T3} + P_{T4})}{6} \times h_3$$

$$Beam\ D = \frac{(P_{T3} + 2P_{T4})}{6} \times h_3 + \frac{(2P_{T4} + P_{T5})}{6} \times h_4$$

$$Beam\ E = \frac{(P_{T4} + 2P_{T5})}{6} \times h_4 + \frac{(P_{T5} + P_{T6})}{2} \times h_5$$

Thus, calculation result is as follows :

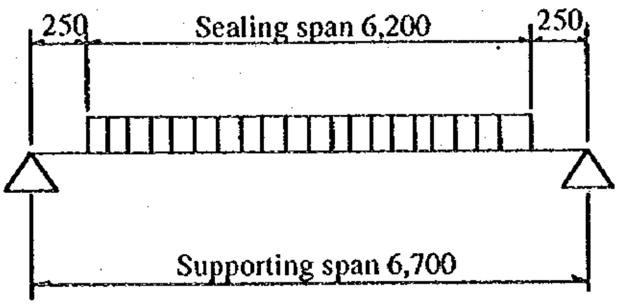
Beam A	Beam B	Beam C	Beam D	Beam E
8.897	17.659	17.411	17.362	17.256

(tf/m)

(3) Bending moment and shearing force

a. Bending moment

Maximum bending moment is calculated by the following equation :



$$M_{max.} = \frac{W \times (2 \times L - B)}{8}$$

where,

$M_{max.}$: Maximum bending moment (tf-m)
W : Design load acting on each beam (tf)
L : Supporting span 6.7 m
B : Sealing span 6.2 m

b. Maximum shearing force is calculated by the following equation :

$$S_{max.} = \frac{W}{2}$$

where,

$S_{max.}$: Maximum shearing force (tf)
W : Design load acting on each beam (tf)

c. Calculation result

The calculation result is as follows.

	Beam A	Beam B
W (tf)	55.163	109.487
Mmax. (tf-m)	49.647	98.538
Smax. (tf)	27.582	54.744

As maximum charging load is on Beam C, bending moment and shearing force are calculated only on Beam C

(4) Bending stress and shearing stress

a. Bending stress and shearing stress are calculated by the following equations:

$$\sigma_{max.} = \frac{M_{max.} \times 10^5}{Z} ,$$

$$\tau_{max.} = \frac{S_{max.} \times 10^3}{A_w}$$

where,

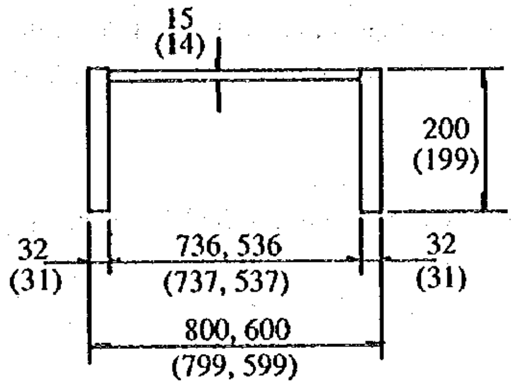
$\sigma_{max.}$: Maximum bending stress (kgf/cm^2)
$M_{max.}$: Maximum bending moment (tf-m)

Z	:	Modulus of section	(cm^3)
$\tau_{max.}$	:	Maximum shearing stress	(kgf/cm^2)
$S_{max.}$	:	Maximum shearing force	(tf)
A_w	:	Area of web at both ends	(cm^2)

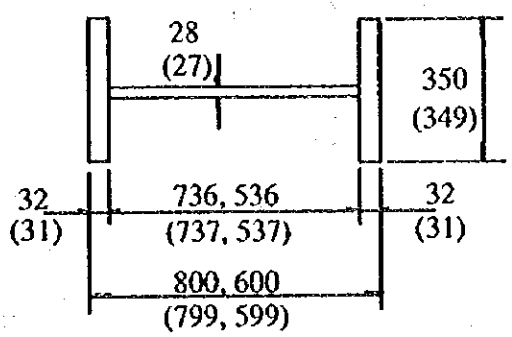
b. Sectional property of beams

Sectional dimension

Beam A



Beam B



		Beam A	Beam B
Moment of inertia	I (cm^4)	228,734	409,310
Modulus of section	Z (cm^3)	5,725	10,246
Area of web	A_w (cm^2)	75.18	144.99

(Aw : Reduced to 600 at both ends)

c. Calculation result

The calculation result is as follows.

		Beam A	Beam B
σ_{max}	(kgf/cm²)	867	962
σ_a	(kgf/cm²)	1,200	
$\tau_{max.}$	(kgf/cm²)	367	378
τ_a	(kgf/cm²)	700	

σ_a : Allowable bending stress = 0.5 σ_y = 1,200 kgf/cm²

τ_a : Allowable shearing stress = 0.3 $\sigma_y \neq$ 700 kgf/cm²

(5) Deflection

Maximum deflection of each beams is calculated by the following equation.

$$\delta_{max.} = \frac{W}{48 \times EI}\left(L^3 - \frac{L \times B^2}{2} + \frac{B^3}{8}\right)$$

where,

$\delta_{max.}$	: Maximum deflection of each beam		(cm)
W	: Design load on each beam		(kgf)
L	: Supporting span		670 cm
B	: Sealing span		620 cm
E	: Elastic modulus of steel	2.1 x 10^6	kgf/cm²
I	: Moment of inertia		(cm⁴)

Thus,

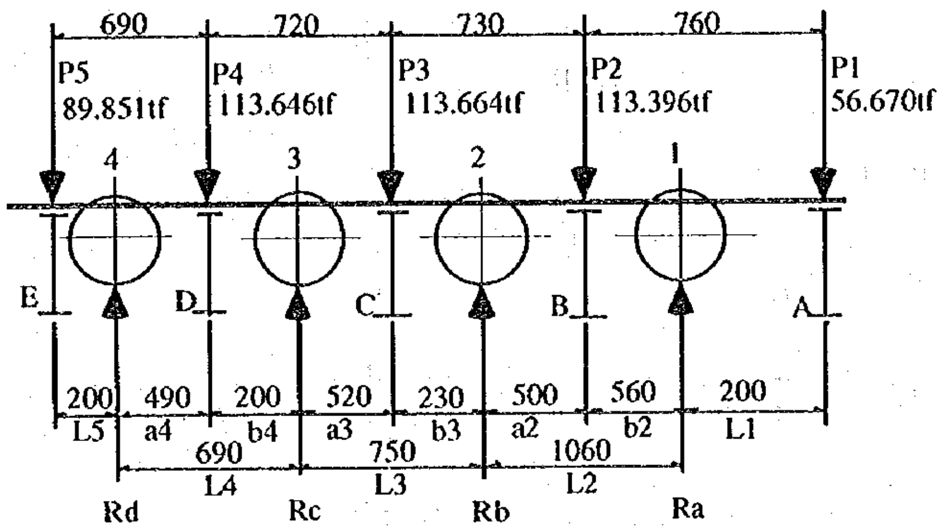
$$\delta_{max.} = 2.0018 \times \frac{W}{I}$$

		Beam A	Beam B
W	(kgf)	55,163	109,487
I	(cm⁴)	228,734	409,310
$\delta_{max.}$	(cm)	0.483	0.535
$\delta_{max.}$ / L		1/1,388	1/1,251
Allowable deflection :		1/1,000	

4. End Beam

(1) Arrangement of main wheels

Three main wheels are provided in each end beam of gate leaf and their arrangement is as follows :



(2) Bending moment

$$M_4 = -\frac{P_5}{2} \times L_5$$

$$L_4 \times M_4 + 2 \times (L_4 + L_3) \times M_3 + L_3 \times M_2 = -\frac{\frac{P_4}{2} \times a_4 \times (L_4^2 - a_4^2)}{L_4} - \frac{\frac{P_3}{2} \times b_3 \times (L_3^2 - b_3^2)}{L_3}$$

$$L_3 \times M_3 + 2 \times (L_3 + L_2) \times M_2 + L_2 \times M_1 = -\frac{\frac{P_3}{2} \times a_3 \times (L_3^2 - a_3^2)}{L_3} - \frac{\frac{P_2}{2} \times b_2 \times (L_2^2 - b_2^2)}{L_2}$$

$$M_1 = -\frac{P_1}{2} \times L_1$$

Thus,

M_1 = - 5.667 tf-m
M_2 = 0.310 tf-m
M_3 = - 4.319 tf-m
M_4 = - 8.985 tf-m

(3) Reaction force

$$R_a = \frac{P_2}{2} \times \frac{a_2}{L_2} + \frac{P_1}{2} + \frac{M_2 - M_1}{L_2}$$

$$R_b = \frac{P_3}{2} \times \frac{a_3}{L_3} + \frac{P_2}{2} \times \frac{b_2}{L_2} + \frac{M_3 - M_2}{L_3} + \frac{M_1 - M_2}{L_2}$$

$$R_c = \frac{P_4}{2} \times \frac{a_4}{L_4} + \frac{P_3}{2} \times \frac{b_3}{L_3} + \frac{M_4 - M_3}{L_4} + \frac{M_2 - M_3}{L_3}$$

$$R_d = \frac{P_5}{2} + \frac{P_4}{2} \times \frac{b_4}{L_4} + \frac{M_3 - M_4}{L_4}$$

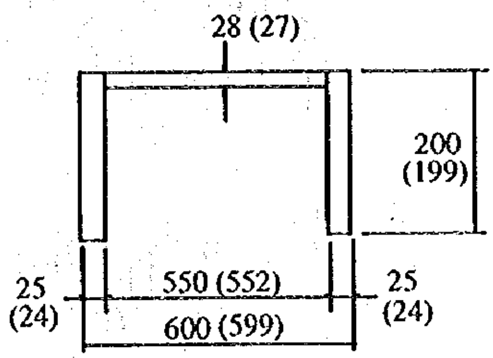
Thus, distributed load on each main wheel

R_a = 60.718 tf
R_b = 57.546 tf
R_c = 57.190 tf
R_d = 68.159 tf

(4) Shearing force

S1 = 28.335 tf
S2 = - 32.383 tf
S3 = 24.315 tf
S4 = - 33.231 tf
S5 = 23.600 tf
S6 = - 33.590 tf
S7 = 23.233 tf
S8 = - 44.926 tf

(5) Sectional property of end beam



$I = 120{,}339\ cm^4$

$Z = 4{,}018\ cm^3$

$A_W = 130.80\ cm^2$

(6) Bending and shearing stresses

Bending stress

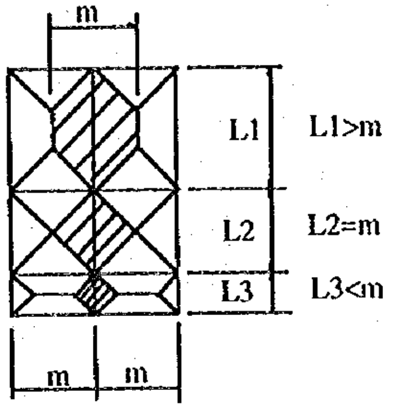
$$\sigma = \frac{M_{max.}}{Z} = \frac{8{,}985 \times 10^5}{4{,}018} = 224\ kgf/cm^2$$

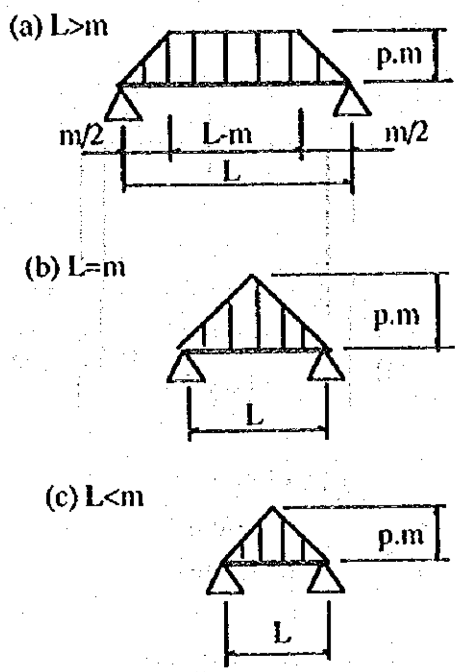
Shearing stress

$$\tau = \frac{S_{max.}}{A_W} = \frac{44.926 \times 10^3}{130.8} = 343\ kgf/cm^2$$

5. Vertical Girders

(1) Bending moment and shearing force
Bending moment and shearing force are calculated by the following formula.





(a) L > m

Bending moment

$$M = \frac{P \bullet m}{24}\left(3L^2 - m^2\right)$$

Shearing force

$$S = \frac{P \bullet m}{2}\left(L - \frac{m}{2}\right)$$

(b) L= m

Bending moment

$$M = \frac{P \bullet m \bullet L^2}{12}$$

Shearing force

$$S = \frac{P \bullet m \bullet L}{4}$$

(c) L< m

Bending moment

$$M = \frac{P \bullet L^3}{12}$$

Shearing force

$$S = \frac{P \bullet L^2}{4}$$

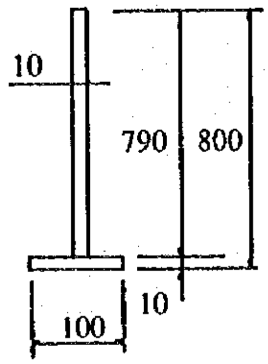
where,

M	:	Maximum bending moment	(kgf-cm)
P	:	Mean water pressure	(kgf/cm^2)
m	:	Pitch of vertical girder	45 cm
L	:	Distance between horizontal beams	(cm)
S	:	Maximum shearing force	(kgf)

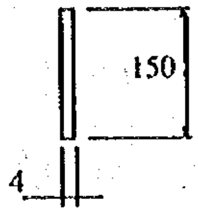
Portion	m (cm)	L (cm)	P (kgf/cm^2)	M (kgf-cm)	S (kgf)
1	50	76	2.418	74,696	3,083
2	50	73	2.493	70,034	2,991
3	50	72	2.565	69,747	3,014
4	50	69	2.636	64,696	2,899
5	50	20	2.680	4,467	268

(2) Sectional property

JIS G 3192 hot rolled steel section [150 x 75 x 6.5/10 and following sections are used at upper and bottom part of gate leaf.



Upper part
(Portion 1 to 4)



Bottom part
(Portion 5)

(A) Hot rolled section

Moment of inertia	I	= 861 cm^4
Modulus of section	Z	= 115 cm^3
Area of web	A_w	= 8.45 cm^2

(B) Plate girder steel section

< Upper part >

Moment of inertia	I	= 56,888 cm^4
Modulus of section	Z	= 1,422 cm^3
Area of web	A_w	= 79 cm^2

< Bottom part >

Moment of inertia	I	= 113 cm^4
Modulus of section	Z	= 15 cm^3
Area of web	A_w	= 6 cm^2

(3) Bending and shearing stresses

As the value of girder of [section is a minor than those of the plate girder sectional in all respects of its sectional properties, the calculation is limited to the [section.

$$\sigma = M/Z$$

where,

σ	:	Maximum bending stress	(kgf/cm^2)
M	:	Maximum bending moment	(kgf-cm)
Z	:	Minor modulus of section (Z=15 cm^3 for Portion 5)	115 cm^3

$$\tau = S/A_w$$

where,

τ	:	Maximum shearing stress	(kgf/cm^2)
S	:	Maximum shearing forcce	(kgf)
A_w	:	Minor modulus of section (A_w=6 cm^3 for Portion 5)	8.45 cm^2

Result of calculation

σ_1	=	650	kgf/cm^2	τ_1	=	365	kgf/cm^2
σ_2	=	609	kgf/cm^2	τ_2	=	354	kgf/cm^2
σ_3	=	606	kgf/cm^2	τ_3	=	357	kgf/cm^2
σ_4	=	563	kgf/cm^2	τ_4	=	343	kgf/cm^2
σ_5	=	298	kgf/cm^2	τ_5	=	45	kgf/cm^2

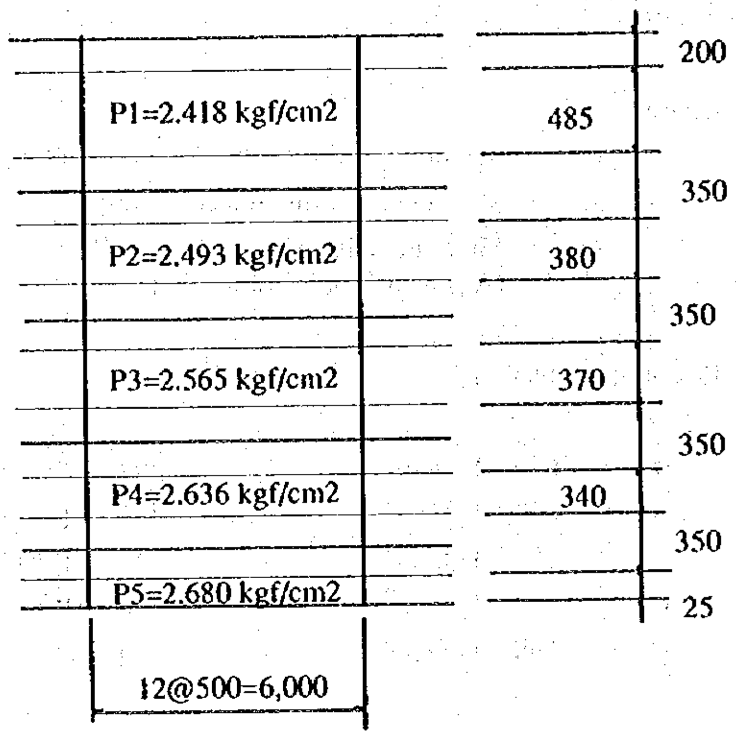
6. Skin Plate

Bending stress of skin plate is calculated in accordance with following Timoshenko's formula.

$$\sigma = \frac{K \times a^2 \times P}{100 \times (t - \varepsilon)^2}$$

where,

σ	:	Bending stress	(kgf/cm^2)
K	:	Coefficient by "b/a"	
a	:	Short span of plate	(cm)
b	:	Long span of plate	(cm)
P	:	Mean design pressure	(kgf/cm^2)
t	:	Thickness of plate	(cm)
ε	:	Corrosion allowance	0.1 cm



	No. 1	No. 2	No. 3	No. 4	No. 5
a (cm)	48.5	38.0	37.0	34.0	2.5
b (cm)	50.0	50.0	50.0	50.0	50.0
b/a	1.03	1.32	1.35	1.47	20.00
K	32.0	42.0	42.8	45.0	50.0
P (kgf/cm^2)	2.418	2.493	2.565	2.636	2.680
t (cm)	1.4	1.4	1.4	1.4	1.4
σ (kgf/cm^2)	1,077	894	889	811	5

7. Main Wheels

Main wheels are of point contact type, and their strength is calculated by the following Hertz's formula.

$$\rho = 0.418 \times \sqrt{\frac{P \times E}{B_o \times R}}$$

$$C = 1.52 \times \sqrt{\frac{P \times R}{B_o \times E}}$$

$$Z = 0.78 \times C$$

where,

ρ	:	Hertz's contact stress	(kgf/cm^2)
P	:	Working loaded one wheel	82,000 kgf
E	:	Modulus of elasticity of wheel	2.1×10^6 kgf/cm^2
R	:	Radius of roller	30 cm
B_o	:	Width of roller	20 cm
C	:	Contact width	(cm)
Z	:	Depth where maximum shearing stress occurs (cm)	

Thus,

$\rho = 7{,}081$ kgf/cm^2

$Z = 0.29$ cm

Allowable contact stress

$$\rho_a = \frac{100}{2 \times V} \times H_B$$

where,

V	:	Safty factor	1.3
H_B	:	Brinell hardness of roller JIS SCMn2B	190 kgf/cm^2

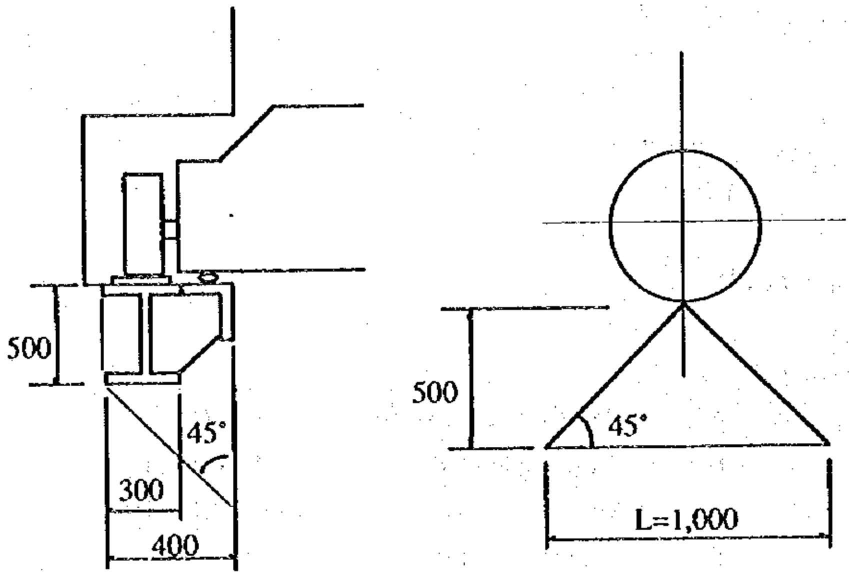
Thus,

$\rho_a = 7{,}308$ $kgf/cm^2 > \rho$

Thickness of track rail : T = 4 x Z

T = 12 mm

8. Strength of Concrete



(1) Bearing stress

$$\sigma_c = P/(L \times B_o)$$

where,

P	:	Working loaded one wheel	82,000 kgf
B_0	:	Width of roller	20 cm

Thus,

$\sigma_c = 6.8\ kgf/cm^2 < 60\ kgf/cm^2$

(2) Shearing stress

$$\tau_c = P/A_c$$

where,

P	:	Working loaded one wheel	82,000 kgf
Ac	:	Shearing area of concrete (50 + 40 x 1.414) x 100	10,656 cm^2

Thus,

$\tau_c = 7.7\ kgf/cm^2 < 8\ kgf/cm^2$

9. Operation Load

(1) Weight of gate leaf Wg = 16 tf

(2) Friction force due to main wheel

$$F_W = P \times (2 \times \mu_1 + \mu_2 \times d)/D$$

where,

P	:	Design load	243.614 tf
μ_1	:	Rolling frictional coefficient to main roller	0.1
μ_2	:	Rolling frictional coefficient to roller bearing	0.02
d	:	Mean diameter of roller shaft	20 cm
D	:	Diameter of main roller	60 cm

Thus,

$F_W = 2.44$ tf

(3) Friction due to rubber seal

$$F_r = \mu \times (q + P \times b) \times \Sigma l$$

where,

μ	:	Friction coefficient of rubber seal	
		at starting	1.5
		at sliding	0.7
q	:	Initial compression load on rubber seal	0.05 tf/m
P	:	Mean design pressure	25.4 tf/m^2
b	:	Contact width of rubber seal	0.04 m
Σ l	:	Total sliding length of rubber seal	6.2 m

Thus,

a) at Raising

$F_{rR} = 9.914$ tf

b) at Lowering

$F_{rL} = 6.626$ tf

(4) Buoyancy

$$F_b = \frac{W_g}{\gamma}$$

where,

Wg	:	Weight of gate leaf	16 tf
γ	:	Specific weight of steel	7.85 tf/m^3

Thus,

$F_b = 2.038$ tf

(5) Total operation load

	Raising load (tf)	Lowering load (tf)
Gate weight (wg)	16	16
Friction force due to main wheel (Fw)	2.44	- 2.44
Friction due to rubber seal (Fr)	9.914	- 6.626
Buoyancy (Fb)	-	- 2.038
Total	28.354	4.896

Thus,

Operating load at

Raising : 29 tf (including allowance)

Lowering : 4.9 tf

10. Wire Rope

(1) Tensile force

$$T_r = \frac{F}{N \times \eta}$$

where,

T_r	: Tensile force (tf)		
F	: Total hoisting load	29 tf	
N	: Numbers of wire rope	4	
η	: Total efficiency of sheave		0.95

Thus,

$$T_r = \frac{29}{4 \times 0.95} = 7.63 \quad tf$$

(2) Selection of wire rope

Type	: JIS G3525 6 x 37 Galvanized Grade A
Diameter	: 35.5 mmØ
Breaking strength	: 62.9 tf

Safty factor

$$S = \frac{Breaking\ strength}{Tensile\ force} = \frac{62.9}{7.63} = 8.24 \quad > \quad 8$$

11. Wire Drum and Sheave

Each diameter of drum and sheave is calculated as follows :

$$D \geq T \times D_w$$

where,

- D : Diameter (mm)
- T : Coefficient
- D_W : Diameter of wire rope 35.5 mm

Thus,

	Drum	Sheave
T	19	17
D_W (mm)	35.5	35.5
D (mm)	675 → 800	604 → 650

12. Output of Hoisting Motor

$$P = \frac{F \times V}{6.12 \times \eta}$$

where,

- P : Output of hoisting motor (kw)
- F : Hoisting load 29 tf
- V : Hoisting speed 1 m/min.

η : Efficiency of rake

$\eta_d \times \eta_s \times \eta_{g1} \times \eta_{g2} \times \eta_{g3} = 0.41$

η_d : Efficiency of drum	0.95
η_s : Efficiency of sheave	0.95
η_{g1} : Efficiency of worm reducer	0.5
η_{g2} : Efficiency of spur gear	0.95
η_{g3} : Efficiency of spur gear	0.95

Thus,

$$P = \frac{29 \times 1}{6.12 \times 0.41} = 11.56 \quad kw$$

= 15 kw, 6 pole motor is used.

13. Required Number of Reduction

$$iR = \frac{N_m}{V_o / (\pi \times D_d)}$$

where,

V_o : Operating speed 1 m/min.

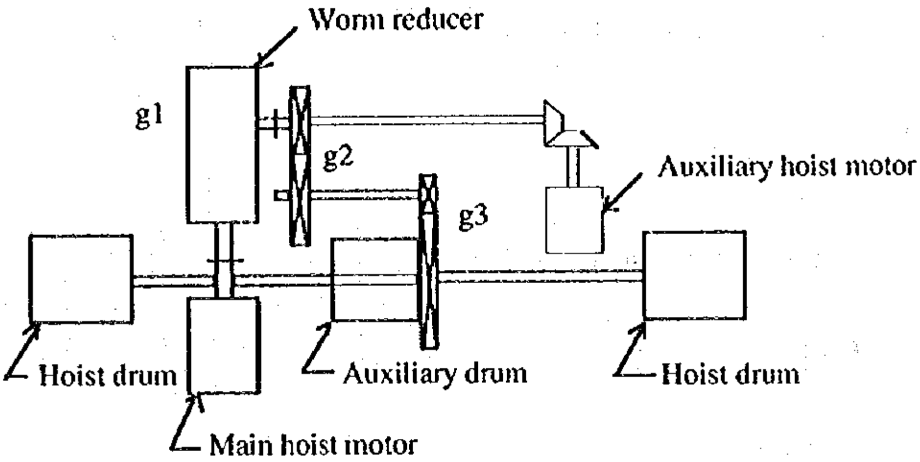
D_d : Diameter of drum 0.8 m

N_m : Revolution per minute of motor 1,160 r.p.m

Thus,

iR = 2,914

14. Arrangement of Hoisting Unit



15. Travelling Unit

(1) Friction force due to wheel

$$F_W = (W_r + W_c) \times (2 \times \mu_1 + \mu_2 \times d) / D$$

where,

Wr	:	Dead weight of gantry crane	35 tf
Wc	:	Hoisting load	17 tf
μ_1	:	Rolling friction coefficient	0.1
μ_2	:	Bearing friction coefficient	0.02
d	:	Diameter of wheel shaft	12 cm
D	:	Diameter of wheel	50 cm

Thus,

Fw = 0.46 tf

(2) Motor output

$$P_m = \frac{F_W \times V}{6.12 \times \eta_1 \times \eta_2 \times n}$$

where,

F_W	:	Operation force (including allowance)	0.5 tf
V	:	Travelling speed	10 m/min

η_1	:	Efficiency of chain sprocket	0.95
η_2	:	Efficiency of worm reducer	0.5
n	:	Number of motor	2

Thus,

$P_W = 0.86$

1.5 kw, 6 pole motor is used.

16.Traversing Unit

(1)Friction force due to wheel

$$F_W = (W_r + W_c) \times (2 \times \mu_1 + \mu_2 \times d) / D$$

where,

Wr	:	Dead weight of trolley	20 tf
Wc	:	Hoisting load	17 tf
μ_1	:	Rolling friction coefficient	0.1
μ_2	:	Bearing friction coefficient	0.02
d	:	Diameter of wheel shaft	5 cm
D	:	Diameter of wheel	25 cm

Thus,

Fw = 0.74 tf

(2) Motor output

$$P_m = \frac{F_W \times V}{6.12 \times \eta_1 \times \eta_2 \times n}$$

where,

F_W	:	Operation force (including allowance)	0.8 tf
V	:	Travelling speed	10 m/min
η_1	:	Efficiency of chain sprocket	0.95
η_2	:	Efficiency of worm reducer	0.5
n	:	Number of motor	2

Thus,

$P_W = 1.38$

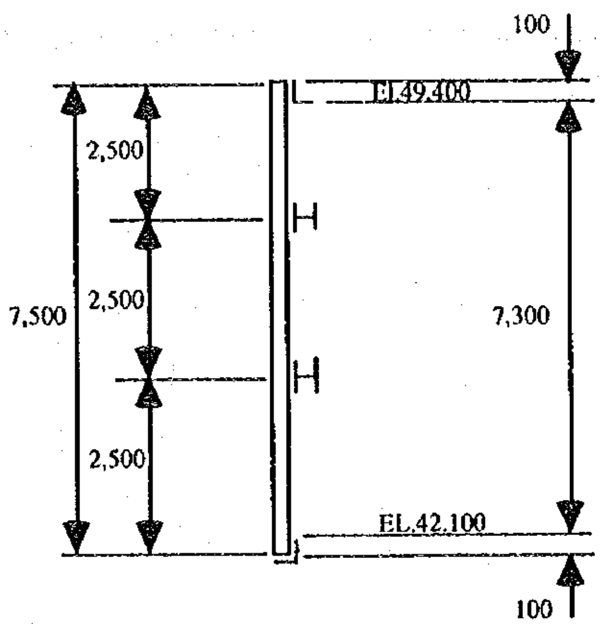
1.5 kw, 6 pole motor is used.

SEVERINO PUMPING STATION
INTAKE FIXED TRASHRACKS

1.Design Conditions

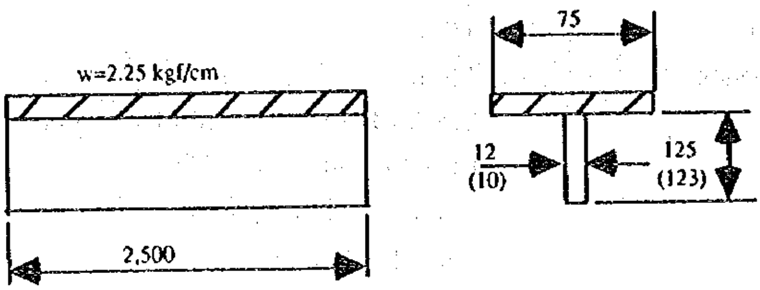
Type	: Vertical type fixed trashrack
Quantity	: 6 set
Clear span	: 6.0 m
Vertical height	: 7.5 m (Deck EL.49.500 m - Sill EL.42.000 m)
Gradient	: 1 : 0 (Θ = 90° ~ 0' ~ 0")
Bar pitch	: 75 mm (center to center)
Design head	: Water head of 3.0 m
Maximum deflection of supporting beams	: 1/600 of supporting span
Corrosion allowance	: 2.0 mm for bar elements and supporting beams

2.Arrangement of Trashrack



3.Bar Elements

(1) Bending moment and stress



a. Bending moment

$$M = \frac{W \times L^2}{8}$$

where,

M : Bending moment (kgf-cm)

W : Unit load on a bar

0.3 kgf/cm^2 x 7.5 cm = 2.25 kgf/cm

L : Maximum distance of center to center of supporting beam

250 cm

Thus,

$$M = \frac{2.25 \times 250^2}{8} = 17,578\ kgf - cm$$

b. Bending stress

$$\sigma_b = \frac{M}{Z}$$

where,

σ_b : Bending stress (kgf/cm^2)

M : Bending moment 17,578 kgf-cm

Z : Modulus of section

FB 125 x 12 25.2 cm^3

(123 x 10)

Thus,

$$\sigma_b = \frac{17,578}{25.2} = 698\ kgf/cm^2$$

(2) Critical stress considering horizontal buckling

$$C_r = 0.6 \times Y \times (1.23 - 0.0153 \times L/T)$$

where,

C_r : Critical stress (kgf/cm^2)

Y : Yield strength of material 2,400 kgf/cm^2

L : Laterally unsupporting length 30 cm

T : Thickness of bar 1.0 cm

Thus,

C_r = 0.6 x 2,400 x (1.23 - 0.0153 x 30 / 1.0)

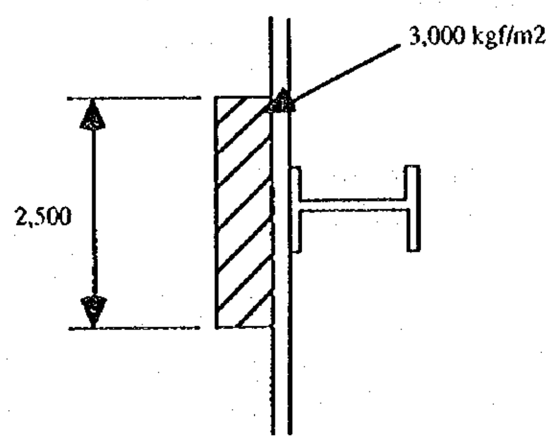
= 1,110 kgf/cm^2

σ_b = 698 kgf/cm^2 < C_r = 1,110 kgf/cm^2

4. Intermediate Supporting Beams

(1)Water pressure load

Water pressure load acted on each beam is as follows:



$W_w = L \times h \times B$

where,

W_w : Water pressure load (kgf)

L : Distance of center to center of supporting beams 2.500 m

h : Design head 3,000 kgf/m^2

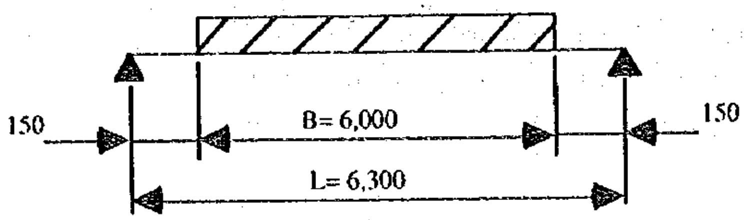
B : Clear span 6.0 m

Thus,

W_w = 2.500 x 3,000 x 6.0

= 45,000 kgf

(2) Bending moment and shearing force due to water load



a. Bending moment

$M_x = W_w \times (2L - B)/8$

where,

M_x : Bending moment due to water load (kgf-cm)

W_w : Water pressure load 45,000 kgf

L : Supporting span (B+30) 630 cm

B : Clear span 600 cm

Thus,

M_x = 45,000 x (2 x 630 - 600) / 8

= 3,712,500 kg-cm

b. Shearing force

$S_x = W_w /2$

where,

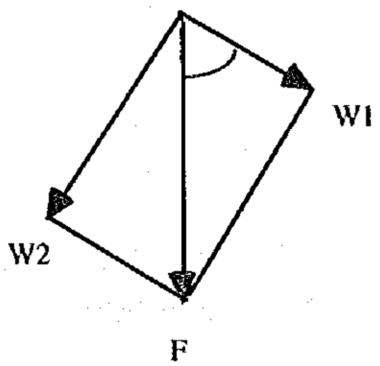
S_X : Shearing force due to water load (kgf)
W_W: Water pressure load 45,000 kgf

Thus,

$S_X = 45{,}000 / 2$
$= 22{,}500$ kgf

(3) Bending moment and shearing force due to own weight

The force due to own weight is distributed as follows :



$W1 = F \cos\theta$
$W2 = F \sin\theta$

where,

W1,W2 : Unit load of each direction (kgf/cm)
F : Unit load due to own weight 1.510 kgf/cm
θ : Angle between "F" and "W1" 90° ~ 0' ~ 0"

Thus,

W1 = 1.510 x cos 90° ~ 0' ~ 0"
= 0 kgf/cm
W2 = 1.510 x sin 90° ~ 0' ~ 0"
= 1.510 kgf/cm

a. Bending moment

$M_1 = W_1 \times L^2 / 8$
$M_2 = W_2 \times L^2 / 8$

where,

M1,M2 : Bending moment of each direction (kgf-cm)

W1,W2 : Unit load of each direction (kgf/cm)

W1 = 0 kgf/cm

W2 = 1.510 kgf/cm

L : Supporting span 630 cm

Thus,

$M1 = 0 \times 630^2/8$

$= 0$ kgf-cm

$M2 = 1.510 \times 630^2/8$

$= 74{,}915$ kgf-cm

b. Shearing force

$$S_1 = W_1 \times L/2$$

where,

S1 : Shearing force due to own weight (kgf)

W1,L : Same as the above

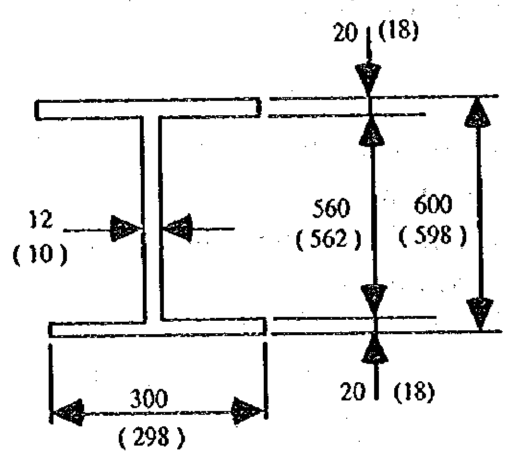
Thus,

$S1 = 0 \times 630 / 2$

$= 0$ kgf

(4) Bending and shearing stresses

Section properties



JIS G 3192 H-600 x 300 x 12/20 is used.

Moment of inertia

$Ix = 118{,}000\ cm^4$

$Iy = 9{,}020\ cm^4$

Modulus of section

$Zx = 4{,}020\ cm^3$

$Zy = 601\ cm^3$

Area of web $Aw = 56\ cm^2$

a. Bending stress

$$\sigma_b = (M_x + M_1)/Z_x + M_2/Z_y$$

where,

σ_b	: Bending stress (kgf/cm^3)	
Mx	: Bending moment due to water load	3,712,500 kgf-cm
M1	: Bending moment due to own weight	0 kgf-cm
Zx	: Modulus of section	4, 020 cm^3
M2	: Bending moment due to own weight	74,915 kgf-cm
Zy	: Modulus of section	601 cm^3

Thus,

$\sigma_b = (\ 3{,}712{,}500 + 0\ ) / 4{,}020 + 74{,}915 / 601$

$= 1{,}048\ kgf/cm^2 \quad < \quad \sigma_{ba} = 1{,}200\ kgf/cm^2$

σ_{ba} : Allowable bending stress

b. Shearing stress

$$\tau_c = (S_x + S_1)/A_w$$

where,

τ_c	: Shearing stress (kgf/cm^2)	
Sx	: Shearing force due to water pressure	22,500 kgf
S1	: Shearing force due to own weight	0 kgf
Aw	: Area of web	56 cm^2

Thus,

τ_c = (22,500 + 0) / 56
= 402 kgf/cm^2 < τ_{ca} = 700 kgf/cm^2

τ_{ca} : Allowable shearing stress

(5) Deflection

$$\delta = \frac{(W_w + W_1 \times B)}{48 \times E \times I_x}\left(L^3 - L \times B^2/2 + B^3/8\right)$$

where,

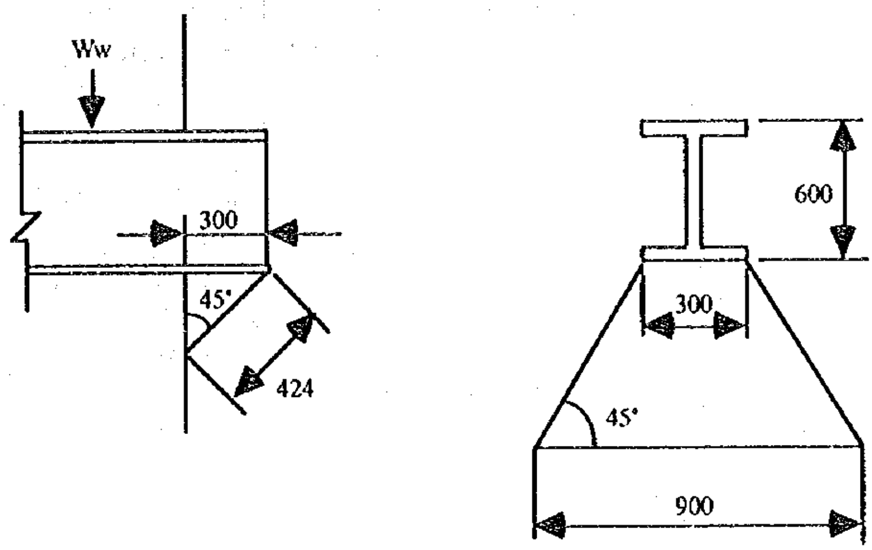
δ : Deflection of beam	(cm)
W_w : Water pressure load	45,000 kgf
W1 : Unit load due to own weight	0 kgf/cm
B : Clear span	600 cm
L : Supporting span	630 cm
E : Young's modulus	2.1 x 10^6 kgf/cm^2
Ix : Moment of inertia	118,000 cm^4

Thus,

$$\delta = \frac{(45,000 + 0 \times 600)}{48 \times 2.1 \times 10^6 \times 118,000}\left(630^3 - 630 \times 600^2/2 + 600^3/8\right)$$
= 0.619 cm

δ/L = 0.619 / 630 = 1 / 1,018 < 1 / 600

5. Strength of Concrete



(1) Bearing stress of concrete

$$\sigma_c = W_w / (2 \times A_1)$$

where,

σ_c : Bearing stress of concrete (kgf/cm^2)
W_w : Water pressure load 45,000 kgf
A1 : Bearing area of concrete 30 x 30 = 900 cm^2

Thus,

σ_c = 45,000 / (2 x 900)
= 25 kgf/cm^2 < 60 kgf/cm^2
(Allowable concrete bearing stress)

(2) Shearing stress of concrete

$$\tau_c = W_w / (2 \times A_2)$$

where,

τ_c : Shearing stress of concrete (kgf/cm^2)
W_w : Water pressure load 45,000 kgf
A2 : Shearing area of concrete 4,344 cm^2
60 x 30 + (30 + 90) / 2 x 42.4

Thus,

$$\tau_c = 45{,}000 / (2 \times 4{,}344)$$

$$= 5.18\ \text{kgf/cm}^2 \quad < \quad 8\ \text{kgf/cm}^2$$

(Allowable concrete shearing stress)

SEVERINO PUMPING STATION
PUMP TOTAL HEAD

1. Design Condition

(1) Design flow	9.6 m^3/s per lane (3.2 m^3/s per unit)
(2) Pipe diameter	
Suction side	1,500 - 1,200 - 1,100 mm
Discharge side	800 - 1,000 - 1,500 - 2,000 -2,400 mm
(3) Pipe length	
Suction side	9.3 m
Discharge side	179.3 m
(4) Water level	
Suction pit	EL. 58.50 m
Head tank	EL. 114.02 m

2. Calculation

(1) Pipe Friction Loss Head

A. Suction Pipe Friction Loss Head

A.1 Friction loss of channel inlet

$$h_{a1} = f_{a1} \times \frac{V_1^2}{2g}$$

where,

f_{a1}	:	Friction coefficient	0.25
V_1	:	Velocity	0.033 m/s

$$V_1 = Q_1/(H \times B) = 3.2/(16.4 \times 6) = 0.033$$

g	:	Gravitational acceleration	9.8 m/s^2

Thus,

h_{a1} = 0.00001 m

A.2 Friction loss of channel

$$h_{a2} = \frac{n^2 \times V_1^2}{rh^{4/3}} \times L_{a2}$$

where,

n : Coefficient of roughness 0.015

rh : $\frac{H \times B}{2H+B} = \frac{16.4 \times 6}{2 \times 16.4 + 6} =$ 2.54 m

L_{a2} : Channel length 4.0 m

Thus,

$h_{a2} = 0.00000$ m

A.3 Hydraulic loss of trash rack

$h_{a3} = 0.3$ m

A.4 Loss of culvert inlet

$$h_{a4} = f_{a4} \times \frac{V_2^2}{2g}$$

where,

f_{a4} : Friction factor 0.5

V_2 : Velocity 0.071m/s

$$V_2 = Q_1/(H \times B) = 3.2/(7.5 \times 6) = 0.071$$

Thus,

$h_{a4} = 0.00013$ m

A.5 Friction loss of culvert

$$h_{a5} = \frac{n^2 \times V_2^2}{rh^{4/3}} \times L_{a5}$$

where,

n : Coefficient of roughness 0.014

rh : $\frac{H \times B}{2 \times (H+B)} = \frac{7.5 \times 6}{2 \times (7.5+6)} =$ 1.67 m

L_{a5} : Channel length 0.5 m

Thus,

h_{a5} = 0.00000 m

A.6 Loss of culvert outlet

$$h_{a6} = f_{a6} \times \frac{V_2^2}{2g}$$

where,

f_{a6} : Friction coefficient 1.2

Thus,

h_{a6} = 0.00031 m

A.7 Loss of culvert inlet

$$h_{a7} = f_{a7} \times \frac{V_3^2}{2g}$$

where,

f_{a7} : Friction coefficient 0.5

V_3 : Velocity 0.178 m/s

$$V_3 = Q_1/(H \times B) = 3.2/(3 \times 6) = 0.178$$

Thus,

h_{a7} = 0.00081 m

A.8 Convergent loss of culvert

$$h_{a8} = f_{a8} \times \frac{V_4^2 - V_3^2}{2g}$$

where,

f_{a8} : Friction coefficient 0.3

V_4 : Velocity 0.213 m/s

$$V_4 = Q_1/(H \times B) = 3.2/(2.5 \times 6) = 0.213$$

Thus,

h_{a8} = 0.00021 m

A.9 Friction loss of culvert

$$h_{a9} = \frac{n^2 \times V_5^2}{rh^{4/3}} \times L_{a9}$$

where,

n : Coefficient of roughness 0.014

rh : $\dfrac{H \times B}{2 \times (H + B)} = \dfrac{\frac{3+2.5}{2} \times 6}{2 \times \left(\frac{3+2.5}{2} + 6\right)} = 0.943$ m

L_{a9} : Channel length 1.0 m

Thus,

h_{a9} = 0.00000 m

A.10 Loss of suction pipe inlet (Dia. 1,500)

$$h_{a10} = f_{a10} \times \frac{V_6^2}{2g}$$

where,

f_{a10} : Friction coefficient 0.5

V_6 : Velocity 1.811 m/s

$$V_6 = Q_1 \Big/ \left(\pi \times \frac{D_1^2}{4}\right) = 3.2 \Big/ \left(3.14 \times \frac{1.5^2}{4}\right) = 1.811$$

Thus,

h_{a10} = 0.08365 m

A.11 Convergent loss of suction pipe (Dia. 1,500 - 1,200)

$$h_{a11} = f_{a11} \times \frac{V_7^2 - V_6^2}{2g}$$

where,

f_{a11} : Friction coefficient 0.1
V_7 : Velocity 2.829 m/s

$$V_7 = Q_1 \Big/ \left(\pi \times \frac{D_2^{\,2}}{4} \right) = 3.2 \Big/ \left(3.14 \times \frac{1.2^2}{4} \right) = 2.829$$

Thus,

h_{a11} = 0.02411 m

A.12 Bend loss of suction pipe (Dia. 1,200)

$$h_{a12} = f_{a12} \times \frac{V_7^{\,2}}{2g}$$

where,

f_{a12} : Friction coefficient 0.248

$$f_{a12} = \left(0.131 + 1.847 \times (D_2/2R)^{3.5} \right) \times (\theta/90)^{0.5}$$

D_2 : 1.2 m
R : 1.32 m
θ : 90°

Thus,

h_{a12} = 0.24795 m

A.13 Convergent loss of suction pipe (Dia. 1,200 - 1,100)

$$h_{a13} = f_{a13} \times \frac{V_8^{\,2} - V_7^{\,2}}{2g}$$

where,

f_{a13} : Friction coefficient 0.15
V_8 : Velocity 3.367 m/s

$$V_8 = Q_1 \Big/ \left(\pi \times \frac{D_3^{\,2}}{4} \right) = 3.2 \Big/ \left(3.14 \times \frac{1.1^2}{4} \right) = 3.367$$

Thus,

h_{a13} = 0.02551 m

A.14 Friction loss of suction pipe (Dia. 1,200)

$$h_{a14} = f_{a14} \times \frac{L}{D_3} \times \frac{V_7^2}{2g}$$

where,

f_{a14} : Friction coefficient 0.029

$$f_{a13} = \{0.0144 + 9.5/(1000 \times \sqrt{V_7})\} \times 1.5$$

L_{a14} : Suction pipe length 9.3 m

Thus,

h_{a14} = 0.14183 m

A.15 Total suction loss head

$h_a = h_{a1} + h_{a2} + h_{a3} + h_{a4} + h_{a5} + h_{a6} + h_{a7} + h_{a8} + h_{a9} + h_{a10} + h_{a11} + h_{a12} + h_{a13} + h_{a14}$

= 0.00001 + 0.00000 + 0.3 + 0.00013 + 0.00000 + 0.00031 + 0.00081 + 0.00021 + 0.00000 + 0.08365 + 0.02411 +0.24795 + 0.02551 + 0.14183

= 0.82452 m

B. Discharge Pipe Friction Loss Head

B.1 Friction loss (Dia. 800)

$$h_{b1} = f_{b1} \times \frac{L_{b1}}{D_4} \times \frac{V_9^2}{2g}$$

where,

f_{b1} : Friction coefficient 0.027

$$f_{b1} = \{0.0144 + 9.5/(1000 \times \sqrt{V_9})\} \times 1.5$$

L_{b1} : Discharge pipe length 3.80 m

V_9 : Velocity 6.366 m/s

$$V_9 = Q_1 \Big/ \left(\pi \times \frac{D_4^2}{4}\right) = 3.2 \Big/ \left(3.14 \times \frac{0.8^2}{4}\right) = 6.366$$

Thus,

h_{b1} = 0.28484 m

B.2 Enlargement loss of discharge pipe (Dia. 800 - 1,000)

$$h_{b2} = f_{b2} \times \frac{(V_9 - V_{10})^2}{2g}$$

where,

f_{b2} : Friction coefficient 0.121

V_{10} : Velocity 4.074 m/s

$$V_{10} = Q_1 \Big/ \left(\pi \times \frac{D_5^2}{4} \right) = 3.2 \Big/ \left(3.14 \times \frac{1^2}{4} \right) = 4.074$$

Thus,

h_{b2} = 0.03251 m

B.3 Friction loss of non-return valve (Dia. 1,000)

$$h_{b3} = f_{b3} \times \frac{V_{10}^2}{2g}$$

where,

f_{b3} : Friction coefficient 0.9

Thus,

h_{b3} = 0.76213 m

B.4 Friction loss of butterfly valve (Dia. 1,000)

$$h_{b4} = f_{b4} \times \frac{V_{10}^2}{2g}$$

where,

f_{b4} : Friction coefficient 0.3

Thus,

h_{b4} = 0.25404 m

B.5 Bend loss of discharge pipe (Dia. 1,000)

$$h_{b5} = f_{b5} \times \frac{V_{10}^2}{2g}$$

where,

f_{b5} : Friction coefficient 0.132

$$f_{b5} = \left(0.131 + 1.847 \times (D_5/2R)^{3.5}\right) \times (\theta/90)^{0.5}$$

D_{b5} : 1.0 m

R : 2.0 m

θ : 90°

Thus,

h_{b5} = 0.11203 m

B.6 Friction loss (Dia. 1,000)

$$h_{b6} = f_{b6} \times \frac{L_{b6}}{D_5} \times \frac{V_{10}^2}{2g}$$

where,

f_{b6} : Friction coefficient 0.030

$$f_{b6} = \left\{0.0144 + 9.5/\left(1000 \times \sqrt{V_{10}}\right)\right\} \times 1.5$$

L_{b6} : Discharge pipe length 16 m

Thus,

h_{b6} = 0.40654 m

B.7 Enlargement loss of discharge pipe (Dia. 1,000 - 1,500)

$$h_{b7} = f_{b7} \times \frac{(V_{10} - V_{11})^2}{2g}$$

where,

f_{b7} : Friction coefficient 0.281

V_{11} : Velocity 1.811 m/s

$$V_{11} = Q_1 / \left(\pi \times \frac{D_6^2}{4} \right) = 3.2 / \left(3.14 \times \frac{1.5^2}{4} \right) = 1.811$$

Thus,

h_{b7} = 0.07351 m

B.8 Friction loss (Dia. 1,500)

$$h_{b8} = f_{b8} \times \frac{L_{b8}}{D_6} \times \frac{V_{12}^2}{2g}$$

where,

f_{b8}	:	Friction coefficient	0.03

$$f_{b8} = \left\{ 0.0144 + 9.5 / \left(1000 \times \sqrt{V_{12}} \right) \right\} \times 1.5$$

L_{b8}	:	Discharge pipe length	12 m
V_{12}	:	Velocity	3.622 m/s

$$V_{12} = Q_2 / \left(\pi \times \frac{D_6^2}{4} \right) = 6.4 / \left(3.14 \times \frac{1.5^2}{4} \right) = 3.622$$

Thus,

h_{b8} = 0.16061 m

B.9 Enlargement loss of discharge pipe (Dia. 1,500 - 2,000)

$$h_{b9} = f_{b9} \times \frac{\left(V_{12} - V_{13} \right)^2}{2g}$$

where,

f_{b9}	:	Friction coefficient	0.281
V_{13}	:	Velocity	2.037 m/s

$$V_{13} = Q_2 / \left(\pi \times \frac{D_7^2}{4} \right) = 6.4 / \left(3.14 \times \frac{2.0^2}{4} \right) = 2.037$$

Thus,

h_{b9} = 0.03602 m

B.10 Bend loss of discharge pipe (Dia. 2,000 : I.P.1)

$$h_{b10} = f_{b10} \times \frac{V_{14}^2}{2g}$$

where,

f_{b10} : Friction coefficient 0.145

$$f_{b10} = \left\{0.131 + 1.847 \times (D_7/2R)^{3.5}\right\} \times (\theta/90)^{0.5}$$

D_7 : 2.0 m

R : 4.0 m

θ : 90°

V_{14} : Velocity 3.056 m/s

$$V_{14} = Q_3 \Big/ \left(\pi \times \frac{D_7^2}{4}\right) = 9.6 \Big/ \left(3.14 \times \frac{2.0^2}{4}\right) = 3.056$$

Thus,

h_{b10} = 0.06929 m

B.11 Bend loss of discharge pipe (Dia. 2,000 : I.P.2)

$$h_{b11} = f_{b11} \times \frac{V_{14}^2}{2g}$$

where,

f_{b11} : Friction coefficient 0.096

$$f_{b11} = \left(0.131 + 1.847 \times (D_7/2R)^{3.5}\right) \times (\theta/90)^{0.5}$$

D_7 : 2.0 m

R : 4.0 m

θ : 38° 52' 27"

Thus,

h_{b11} = 0.04554 m

B.12 Bend loss of discharge pipe (Dia. 2,000 : I.P.3)

$$h_{b12} = f_{b12} \times \frac{V_{14}^2}{2g}$$

where,

f_{b12} : Friction coefficient 0.085

$$f_{b12} = \left(0.131 + 1.847 \times \left(D_7/2R\right)^{3.5}\right) \times \left(\theta/90\right)^{0.5}$$

D_7 : 2.0 m
R : 4.0 m
θ : 30° 28' 29"

Thus,

h_{b12} = 0.04032 m

B.13 Bend loss of discharge pipe (Dia. 2,000 : I.P.4)

$$h_{b13} = f_{b13} \times \frac{V_{14}{}^2}{2g}$$

where,

f_{b13} : Friction coefficient 0.049

$$f_{b13} = \left(0.131 + 1.847 \times \left(D_7/2R\right)^{3.5}\right) \times \left(\theta/90\right)^{0.5}$$

D_7 : 2.0 m
R : 4.0 m
θ : 10° 4' 56"

Thus,

h_{b13} = 0.02320 m

B.14 Enlargement loss of discharge pipe (Dia. 2,000 - 2,400)

$$h_{b14} = f_{b14} \times \frac{\left(V_{14} - V_{15}\right)^2}{2g}$$

where,

f_{b14} : Friction coefficient 0.191
V_{15} : Velocity 2.122 m/s

$$V_{15} = Q_3 \Big/ \left(\pi \times \frac{D_8^{\,2}}{4} \right) = 9.6 \Big/ \left(3.14 \times \frac{2.4^2}{4} \right) = 2.122$$

Thus,

h_{b14} = 0.00851 m

B.15 Velocity head loss of discharge pipe (Dia. 2,400)

$$h_{b15} = f_{b15} \times \frac{V_{15}^{\,2}}{2g}$$

where,

f_{b15} : Friction coefficient 1.0

Thus,

h_{b15} = 0.22975 m

B.16 Friction loss (Dia. 2,000)

$$h_{b16} = f_{b16} \times \frac{L_{b16}}{D_7} \times \frac{V_{14}^{\,2}}{2g}$$

where,

f_{b16} : Friction coefficient 0.03

$$f_{b16} = \left\{ 0.0144 + 9.5 \Big/ \left(1000 \times \sqrt{V_{14}} \right) \right\} \times 1.5$$

L_{b16} : Discharge pipe length 156 m

Thus,

h_{b16} = 1.11481 m

B.17 Total discharge pipe loss head

$h_b = h_{b1} + h_{b2} + h_{b3} + h_{b4} + h_{b5} + h_{b6} + h_{b7} + h_{b8} + h_{b9} + h_{b10}$
$+ h_{b11} + h_{b12} + h_{b13} + h_{b14} + h_{b15}$

= 0.28484 + 0.03251 + 0.76213 + 0.25404 + 0.11203 + 0.40654 +

0.07351 + 0.16061 + 0.03602 + 0.06929 + 0.04554 + 0.04032 +
0.02320 + 0.00851 + 0.22975 + 1.11481

= 3.65365 m

C. Total Loss Head

$h_L = h_a + h_b$
$= 0.82452 + 3.65365$
$= 4.47817$ m → 4.48 m

(2) Actual Head

A. Weighted averaged Water Level at suction side EL. 58.50 m

B. High Water level at Head Tank EL. 114.02 m

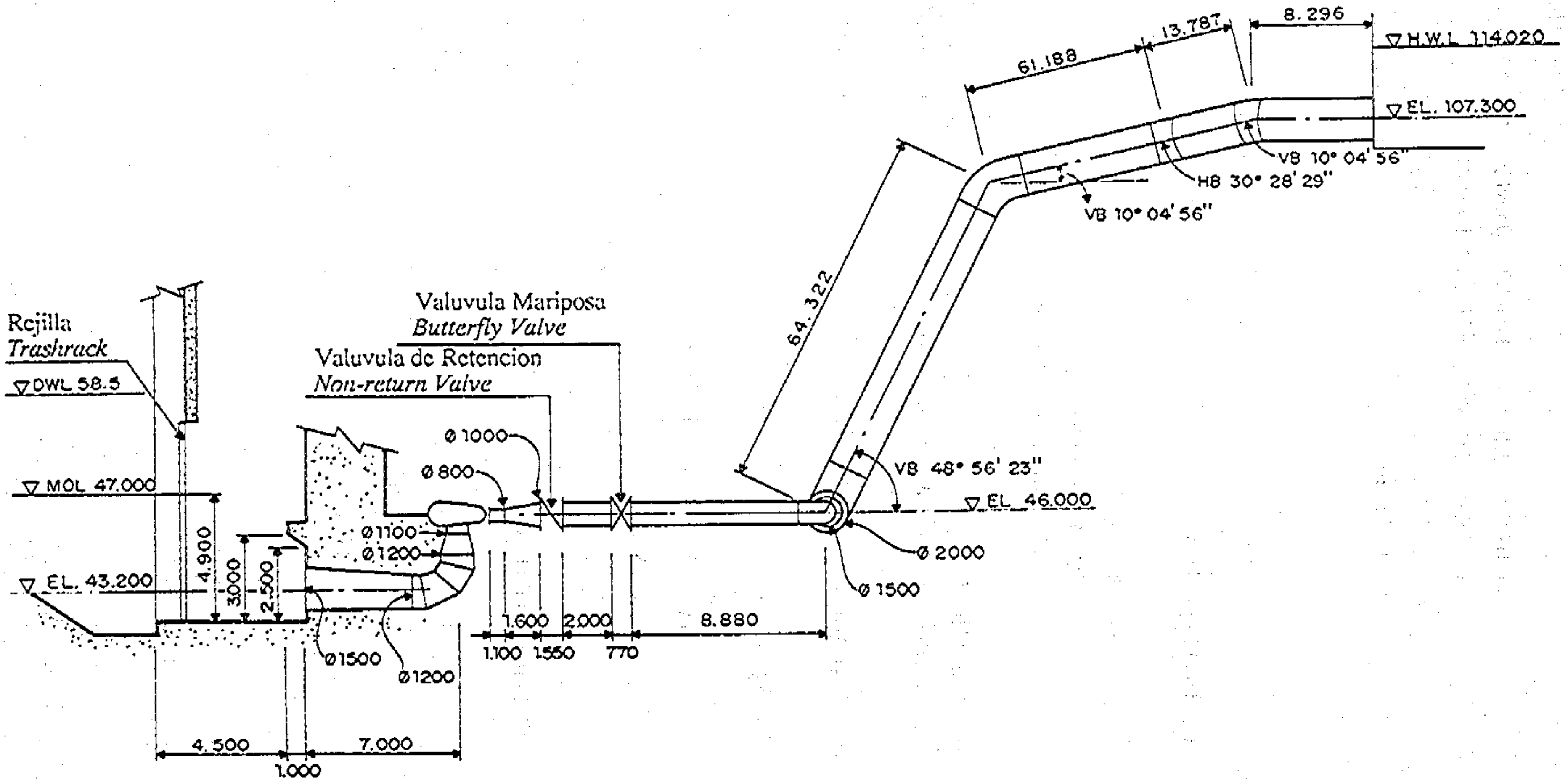
C. Actual head

h_A = EL. 114.02 - EL. 58.50
= 55.52 m

(3) Pump Total Head

$H_T = h_L + h_A$
$= 4.48 + 55.52$
$= 60.0$ m

PERFIL DE LA TUBERIA DE PRECION #1
PROFILE OF #1 PENSTOCK



SEVERINO PENSTOCK
WATER HAMMER

1. Design Condition

(1) Design discharge — 480 m^3/min (160 m^3/min per unit)

(2) Water level

Suction pit (L.W.L) — EL. 47.00 m

Head tank (H.W.L) — EL. 114.02 m

(3) Actual head — 67.02 m

(4) Number of pumps in operation — 3 units

(5) Discharge pipeline

Dia. (mm)	800	800-1000	1000	1000-1500	1500	1500-2000	2000
Length (m)	2.2	1.6	15.3	2.0	6.0	2.0	150.8
Material	JIS G 3106 SM 400 (ASTM A516 Grade 60/70)						
Shell thickness (mm)	9	9	9	9	9	10	9 - 10

2. Basic Data

(1) Pump

(1) Pump	Type	Vertical Shaft Single Suction Volute Pump
	Bore	800 mm
	Discharge	160 m^3/min
	Speed	600 rpm
	Efficiency	86 %
	Total head	69.8 m
(2) Motor	Type	Three-phase Wound - Rotor Type Induction Motor
	Output	2,400 kw
	Speed	600 rpm
	Frequency	60 Hz
	Voltage	4,160 kv

3. Calculation

(1) Pump shaft power

$$P = 9.8 \times \gamma \times Q \times H / \eta$$

where,

γ	:	Specific weight of water	1.0
Q	:	Pump discharge	2.67 m^3/s
H	:	Total head	69.8 m
η	:	Pump efficiency	0.86

Thus,

P = 2,123.7 kw

(2) Pump torque

$$M = 974 \times P / N$$

Where,

N : Speed 600 rpm

Thus,

M = 3,447 kg-m

(3) Flywheel effect (GD^2)

Motor	3,500 kg-m^2
Pump	740 kg-m^2
Total	4,240 kg-m^2

(4) Coefficient of pump inertia

$$K = \frac{375 \times M}{GD^2 \times N}$$

where,

M	:	Pump torque	3,447 kg-m
GD^2	:	Total flywheel effect	4,240 kg-m^2
N	:	Speed	600 rpm

Thus,

K = 0.51

(5) Average velocity in discharge pipeline

$$V = \frac{Q}{\frac{\pi}{4}D^2}$$

Pipe dia. (m)	0.8	1.0	1.5	2.0
Pipe length (m)	3.0	17.1	8.0	151.8
Flow (m^3/s)	3.2	3.2	6.4	9.6
Velocity (m/s)	6.4	4.1	3.6	3.1

Thus,

V = 3.3 m/s

(6) Propagation velocity of pressure wave

$$a = \frac{1{,}425}{\sqrt{1 + \frac{k}{E} \times \frac{D}{t}}}$$

where,

k/E : In case of Steel 0.01

D : Inside diameter of discharge pipe

t : Shell thickness of discharge pipe

Pipe dia. D (m)	0.8	1.0	1.5	2.0
Shell thickness t (mm)	9	9	9	9 - 10

Thus,

a = 841.7 m/s

(7) Coefficient of penstock

$$2\rho = \frac{a \times V}{g \times H}$$

where,

g : Gravitational acceleration 9.8 m/s^2

Thus,

$2\rho = 4.0$

(8) Reciprocating time of pressure wave

$$\mu = \frac{2 \times L}{a}$$

where,

L : Pipeline length 180.4 m

Thus,

$\mu = 0.43$ sec

(9) Ratio of penstock loss head

$$R = \frac{H - Ha}{H} \times 100$$

where,

H : Total head 69.8 m

Ha : Actual head 67.0 m

Thus,

R = 4.0 %

4. Water Head Diagram

(1) Value of pressure variation

The values are obtained from the J.Parmakian's chart.

Position	Min. press.	Max. press.
Pump	23 % 16.1 m	175 % 122.2 m
L/2	55 % 38.4 m	144 % 100.5 m
3L/4	74 % 51.7 m	126 % 87.9 m

SEVERINO PENSTOCK
SHELL THICKNESS

1. Exposed pipeline

(1) Shell thickness due to internal pressure

$$t \geq \frac{H \times D}{2\sigma \times \eta_w} + \varepsilon$$

where,

H : Design head (cm)

$H = H_1 + H_2$

H_1 : Static head (cm)

H_2 : Water hammer (cm)

D : Penstock inside diameter (cm)

σ : Allowable tensile stress 1300 kgf/cm^2

η_w : Welding efficiency 0.9

ε : Corrosion allowance 0.2 cm

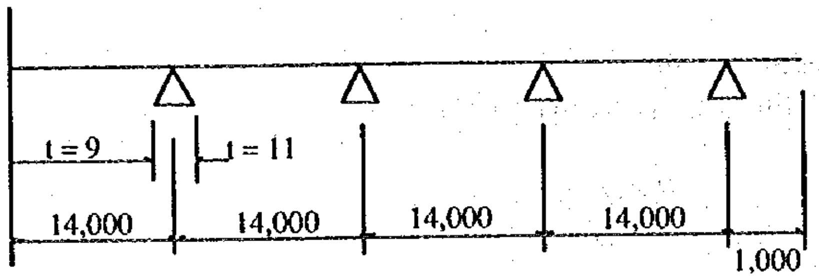
Materials	t : Thickness	σ (kgf/cm^2)
SS 400 JIS G 3101	≤ 40 mm > 40 mm	1300 1200
SM 400 JIS G 3106	≤ 40 mm > 40 mm	1300 1200
SM 490 JIS G 3106	≤ 40 mm > 40 mm	1750 1650

Location	D (cm)	H_1 (cm)	H_2 (cm)	t (cm)	t_m (cm)
1	100	67,500	36,000	0.8	0.8
2	150	67,500	34,500	0.8	0.8
3	200	67,500	32,500	1.0	0.9
4	200	17,500	16,000	0.9	0.9
5	200	7,000	0	0.9	0.9

t_m : Minimum thickness

$$t_m = \frac{D + 800}{400}$$

(2) Bending stress



- Continuous beam in 3-span, dia. 2.0 m, t = 9 mm

$W = W_t + W_w$ $\qquad$ $W_t = 0.44$ t/m

$= 3.58$ t/m $\qquad$ $W_w = \pi d^2/4 = 3.14$ t/m

$M_{max} = -0.1WL^2$

$= -0.1 \times 3.58 \times 14^2$

$= -70.17$ t-m

$\sigma = M / Z$

$= M / (\pi\, t\, r_m^2)$

$= -70.17 \times 10^5 / (\pi \times 0.9 \times 100.45^2)$

$= -246$ kgf/cm^2 $< \sigma_a$

- Max. allowable span, restricting due to max. allowable deflection of 1/350

$$\frac{\delta}{L} = \frac{5WL^3}{384EI}$$

where,

$W = 3.58$ t/m

$\delta/L = 1/350$

$E = 2.1 \times 10^6$ kgf/cm^2

$I = \pi/64 \times (D_0^4 - D_1^4) = 2,393,533$ cm^4

Thus,

$L = 31.3$ m > 14.0 m

(3) Stress at half-filled condition

$$\delta_{max} = 445.2 \times r_i^3/t = 445.2 \times 100.45^3/(0.9\text{-}0.15)^2 = 802 \text{ kgf/cm}^2$$

(4) Critical buckling pressure due to external pressure of 2.0 m
(Pressure difference at inside and outside pipeline)

$$P_k = \frac{2E}{1-\nu^2}\left(\frac{t}{D_0}\right)^3 \times \frac{1}{n}$$

where,

E : 2.1×10^6 (kgf/cm^2)

ν : Poisson's number 0.3

n : SF, 2 for exposed

Thus,

$P_k = 0.12$ kgf/cm^2 < 0.2 kgf/cm^2

2. Embedded pipeline

(1) Condition

- dia. 2.0 m, t = 10 mm concrete-filled
- External pressure

EL.70 - EL.46 = 24 m

(2) Pipeline having dia. 2.0 m, t = 10 mm without stiffener rings, material in σ=1,300 kgf/cm^2 class, is withstood against external pressure up to 60 m by calculation of E. Amstutz formula. The design is therefore accepted.

SEVERINO PENSTOCK
ONE-WAY SURGE TANK

1. Design Condition

(1) Design flow : 9.6 m3/sec
(one penstock under operation of three pump units)

(2) Penstock inside dia. : 1.0 - 1.5 - 2.0 m

(3) Penstock length : 180.4 m

(4) Water level
Suction pit : EL. 58.5
Head tank : EL.114.02
Surge tank : EL.101.0

2. Calculation

(1) Effective volume of surge tank

$$V_e = \frac{Q^2}{2g} \times \left(\frac{L_2}{A_2 \times H_2} - \frac{L_1}{A_1 \times H_1} \right) \times \alpha$$

where,

Q	:	Design flow	9.6 m3/sec
L_1	:	Penstock length between pump and surge tank	105.6 m
L_2	:	Penstock length between surge tank and head tank	74.8 m
$A_{1,2}$	:	Sectional area of penstock	3.14 m^2
H_1	:	Actual head between water level in suction pit and surge tank	42.5 m
H2	:	Actual head between water level in surge tank and head tank	13.02 m
α	:	Safety factor	3

Thus,

$V_e = 14.6\ m^3 \quad \rightarrow \quad 15\ m^3$

7.2 Transmission Line

SAG COMPUTATION OF ACSR ORIOLE
AND
O.H. EARTHWIRE 55 SQ.MM

A. POWER CONDUCTOR

A-1. Particulars of Power Conductor

Type of Conductor	: ACSR Oriole (ASTM B-232-78)
Stranding	: Aluminium : 30/2.69mm + St: 7/2.69mm
Sectional Area	: Aluminium : $A_a = 170.5mm^2$,
	Steel : $A_s = 39.8mm^2$,
	Total : $A = 210.3mm^2$
Diameter of Conductor	: D = 18.83mm
Unit Weight of Conductor	: w = 0.737kg/m
Ultimate Tensile Strength	: $T_u = 7{,}590kg$
Young's Modulus	: Aluminium : $E_a = 6{,}300kg/mm^2$
	Steel : $E_s = 21{,}000kg/mm^2$
Linear Expansion Coefficient	: Aluminium $\alpha a = 23 \times 10^{-6}/°C$
	Steel $\alpha s = 1.5 \times 10^{-6}/°C$
Equivalent Span Length	: $S_e = 350m$
Wind Pressure	: $w_p = 39kg/m^2$
Minimum Factor of Safety to Tu	: 2.5 for Max. Working Tension
	: 4.0 for Every Day stress
Conductor Temperature	: Maximum 60°C, Minimum 5°C,
	Every Day 25°C

A-2. Composite Young's Modulus (E) of ACSR

$m = A_a/A_s = 170.5/39.8 = 4.284$

$E = (m \times E_a + E_s)/(m + 1) = (4.284 \times 6{,}300 + 21{,}000)/(4.284 + 1) = 9.082\ (kg/mm^2)$

A-3. Composite Expansion Coefficient (α) of ACSR

$\alpha = (m \times E_a \times \alpha_a + \alpha_s \times E_s)/(m \times E_a + E_s)$

$= (4.284 \times 23 \times 10^{-6} \times 6{,}300 + 1.5 \times 10^{-6} \times 21{,}000)/(4.284 \times 6{,}300 + 21{,}000)$

$= 13.59 \times 10^{-6}/°C$

A-4. Wind Pressure and Weight of Equivalent Spans

(a) Equivalent Span (Se) = 350m

(b) Wind Press. $W_w(350) = S_e \times D \times 39 \times 10^{-3}kg = 350 \times 18.83 \times 10^{-3} \times 39 = 257kg$

$W_w(400) = 400 \times 18.83 \times 39 \times 10^{-3} = 294kg$

(c) Weight $W_t(350) = S \times {}_w\ kg = 350 \times 0.737 = 258kg$

$W_t(400) = 400 \times 0.737 = 295kg$

A-5. Loading Coefficient
(for maximum wind pressure at minimum temperature)

$S_e = 350m$: $q = (W_w^2 + W_t^2)^{1/2} / W_t = (257^2 + 258^2)^{1/2} / 258 = 1.41$

$S_e = 400m$: $q = (294^2 + 295^2)^{1/2} / 295 = 1.41$

A-6. Conductor Unit Weight per Section

$\delta = w/A = 0.737/210.3 = 3.50 \times 10^{-3}$ (kg/m.mm²)

A-7. Maximum Working Tension of Power Conductor (based on EDS)

EDS of the power conductor is less than 25% of the ultimate tensile strength of the power conductor (T_u), i.e., factor of safety of EDS is more than 4.0 to the T_u.

$f_1 = 0.25 \times T_u/A = 0.25 \times 7,590 / 210.3 = 9.022$ (kg/mm²)

$\alpha.E.t = 13.59 \times 10^{-6} \times 9.082 \times 10^3 \times (18-25) = -0.864$

$q_1 = 1.00, \quad q_2 = 1.41$

$K = f_1 - (q_1.\delta)^2 . Se^2 . E/24 . f_1^2$

$= 9.022 - (1.00 \times 3.50)^2 \times 10^{-6} \times 9.082 \times 10^3 \times Se^2 / 24 \times 9.022^2$

$= 9.022 - 0.0569 \times 10^{-3}.Se^2$

Se = 350m $K = 9.022 - 6.97 = 2.05$

Se = 400m $K = 9.022 - 9.104 = -0.082$

$M = (q_2.\delta)^2 . E . Se^2 / 24 = (1.41 \times 3.50)^2 \times 10^{-6} \times 9.082 \times 10^3 . Se^2 / 24$

$= 9.22 \times 10^{-3} . Se^2$

Se = 350m M = 1,129

Se = 400m M = 1,475

<u>Maximum working stress and tension of power conductors</u>

$f_2^2\{f_2 - (K-\alpha.E.t)\} = M$

Se = 350m $f_2^2\{f_2-(2.05+0.864)\} = 1{,}129$

$f_2^2(f_2-2.914) = 1{,}129$

$f_2 = 11.48$ (kg/mm^2) Max.work.Tension: $T = f_2.A = 2{,}414$kg

Se = 400m $f_2^2\{f_2-(-0.082+0.864)\} = 1{,}475$

$f_2 = 11.65$ (kg/mm^2) Max.Work.Tension: $T = f_2.A = 2{,}450$kg

Maximum working tension of the power conductor is set at 2,400kg. Factor of safety of the maximum working tension against its ultimate tensile strength is 7,590/2,400 = 3.10, i.e., more than 2.5 required.

A-8. Maximum Sag and Minimum Sag of Power Conductor

$f_1 = 2{,}400 / 210.3 = 11.41$ (kg/mm^2), $q_1 = 1.41$, $q_2 = 1.0$,

$\alpha.E.t_{max} = 13.59 \times 10^{-6} \times 9.082 \times 10^3 \times (60-18) = 5.184$

$\alpha.E.t_{min} = 13.59 \times 10^{-6} \times 9.082 \times 10^3 \times (5-18) = -1.604$

$K = f_1 - (q_1.\delta)^2.E.Se^2 / 24 \times f_1^2$

$= 11.41 - 1.41^2 \times 3.50^2 \times 10^{-6} \times 9.082 \times 10^3.Se^2 / 24 \times 11.41^2$

$= 11.41 - 0.071 \times 10^{-3}.Se^2$

Se = 350m $K = 11.41 - 8.70 = 2.71$

Se = 400m $K = 11.41 - 11.36 = 0.05$

$M = (q_2.\delta)^2.E.Se^2 / 24 = 3.50^2 \times 10^{-6} \times 9.082 \times 10^3.Se^2 / 24 = 4.636 \times 10^{-3}.Se^2$

Se = 350m $M = 467.91$

Se = 400m $M = 741.76$

<u>Conductor stress for maximum sag</u>

Se = 350m	$f_2^2\{f_2 - (2.71 - 5.184)\} = 567.91$	$f_2 = 7.533$ (kg/mm^2)
Se = 400m	$f_2^2\{f_2 - (0.05 - 5.184)\} = 741.76$	$f_2 = 7.623$ (kg/mm^2)

<u>Conductor stress for minimum sag</u>

Se = 350m	$f_2^2\{f_2 - (2.71 + 1.604)\} = 567.91$	$f_2 = 9.995$ (kg/mm^2)
Se = 400m	$f_2^2\{f_2 - (0.05 + 1.604)\} = 741.76$	$f_2 = 9.635$ (kg/mm^2)

A-9. Maximum and minimum sags of power conductor

Maximum sags

$D_{350m} = \delta.q_2.S_e^2 / 8 \times f_2 = 3.50 \times 10^{-3}.S_e^2 / 8 \times 7.533 = 0.0581 \times 10^{-3} \times S_e^2$

$D_{400m} = 3.50 \times 10^{-3}.S_e^2 / 8 \times 7.623 = 0.0574 \times 10^{-3} \times S_e^2$

Minimum sags

$D_{350m} = 3.50 \times 10^{-3}.S_e^2 / 8 \times 9.995 = 0.0438 \times 10^{-3} \times S_e^2$

$D_{400m} = 3.50 \times 10^{-3}.S_e^2 / 8 \times 9.635 = 0.0454 \times 10^{-3} \times S_e^2$

	Maximum Sag (m)		Minimum Sag (m)	
	Equivalent Span (m)		Equivalent Span (m)	
	350m	400m	350m	400m
50m	0.15	0.14	0.11	0.11
100m	0.58	0.57	0.44	0.45
150m	1.31	1.29	0.99	1.02
200m	2.32	2.30	1.75	1.82
250m	3.63	3.59	2.74	2.84
300m	5.23	5.17	3.94	4.09
350m	7.12	7.03	5.37	5.56
400m	9.30	9.18	7.01	7.26
450m	11.77	11.62	8.87	9.19
500m	14.59	14.35	10.95	11.35
550m	17.57	17.36	13.25	13.73
600m	20.92	20.66	15.77	16.34
650m	24.55	24.25	18.51	19.18
700m	28.47	28.13	21.46	22.25
800m	37.18	36.74	28.03	29.06
900m	47.06	46.49	35.48	36.77
1,000m	58.10	57.40	43.80	45.40
1,100m	70.30	69.45	53.00	54.93
1,200m	83.66	82.66	63.07	65.38
1,500m	130.73	129.15	98.55	102.15
2,000m	232.40	229.60	175.20	181.6
2,500m	363.13	158.75	273.75	283.74

B. OVERHEAD EARTHWIRE

B-1. Conditions for Sag Computation

Sag of O.H. earthwire is computed so that its sag in the equivalent span of 350m at 5°C, still air condition is 80% of sag of the power conductor in the same condition.

Sag of the power conductor in the equivalent span of 350m is 5.37m at 5°C and still air. The sag of O.H. earthwire at the condition should be around 5.37 x 0.8 = 4.30m.

B-2. Particulars of Overhead Earthwire (Galvanized Steel Stranded Wire)

Stranding	: St. 7/3.2mm
Sectional Area	: $A = 56.29\text{mm}^2$,
Diameter of Wire	: $D = 9.60\text{mm}$
Unit Weight of Wire	: $w = 0.446\text{kg/m}$
Ultimate Tensile Strength	: $T_u = 4{,}660\text{kg}$
Young's Modulus	: $E_s = 21{,}000\text{kg/mm}^2$
Linear Expansion Coefficient	: $\alpha s = 1.5 \times 10^{-6}/°C$
Wind Pressure	: $w_p = 39\text{kg/m}^2$
Minimum Safety of Factor	: 2.5 for against T_u
Earthwire Temperature	: Maximum: 40°C, Minimum: 5°C, Everyday: 25°C

B-3. Wind Pressure of Wire, Loading Coefficient and Unit Weight per Section

Wind pressure	$W_w = D \times 10^{-3} \times 39 = 0.374$ (kg/m)
Loading Coefficient	$q = (W_w^2 + W^2)^{1/2} / W = 1.305$
Unit Weight per Section	$\delta = w/A = 7.923 \times 10^{-3}$ (kg/m.mm²)

B-4. Stress of Wire at the Condition of 5°C and Still Air

$f = \delta . q_2 . S_e^2 / 8 \times d_2$

where, $d_2 = 4.30$m (80% of the power conductor's sag)

$f = 7.923 \times 10^{-3} \times 1.0 \times 350^2 / 8 \times 4.30 = 28.214$ (kg/mm²)

B-5. Maximum Working Tension of Earthwire

$\alpha.E.t = 1.5 \times 10^{-6} \times 21 \times 10^3 \times (18\text{-}5) = 0.410$

$K = f_1 - (q_1.\delta)^2.E.S_e^2 / 24 \times f_1^2,$

$M = (q_2.\delta)^2.E.S_e^2 / 24$

$K = 28.214 - (1 \times 7.923 \times 10^{-3})^2 \times 21 \times 10^3 \times 350^2 / 24 \times 28.214^2 = 19.76$

$M = 1.305^2 \times 7.923^2 \times 10^{-6} \times 21 \times 10^3 \times 350^2 / 24 = 11{,}459$

$f_2^2 \{f_2 - (K - \alpha.E.t)\} = M$

$f_2^2 (f_2 - 19.35) = 11{,}459$

$f_2 = 31.155$ (kg/mm²),

accordingly, maximum working tension: $T = f_2.A = 1{,}754$ (kg)

Factor of safety of the maximum working tension against its ultimate tensile strength is more than 2.5 (4,660/1,754 = 2.66).

7.3 Conguillo Inlet

CONGUILLO INLET
PIPE LOSS HEAD

1. Design Condition

(1)	Design flow	18 m^3/s (9 m^3/s per unit)
(2)	Pipeline inside dia	1,200 – 1,400 mm
(3)	Pipe length	5.7 m – 26 m

2. Calculation

(1) Hydraulic loss of trash rack

$$h_1 = 0.3\,m$$

(2) Loss of pipe inlet

$$h_2 = f_e \times \frac{V_1^2}{2g}$$

where,

f_c	:	Friction coefficient (Bellmouth)	0.15
V_1	:	Velocity	7.96 m/s

$$V_1 = \frac{9}{\pi \times 1.2^2/4} = 7.96$$

Thus,

h_2 = 0.48491 m

(3) Butterfly valve loss

$$h_3 = f_e \times \frac{V_1^2}{2g}$$

where,

f_c : Friction coefficient 0.2

Thus,

h_3 = 0.64618 m

(4) Friction loss of pipe

$$h_4 = f_e \times \frac{L_1}{D_1} \times \frac{V_1^2}{2g}$$

where,

f_e : Friction coefficient 0.027

$$f_e = \{0.0144 + 2.5/(1{,}000 \times \sqrt{V_1})\} \times 1.5$$

L_1 : Pipe length 5.7 m
D_1 : Pipe inside diameter 1.2 m
g : Gravitational acceleration 9.8 m/s^2

Thus,

h_4 = 0.41436 m

(5) Enlargement pipe loss

$$h_5 = \frac{(V_1 - V_2)^2}{2g} \times f_e$$

where,

f_e : Friction coefficient 0.33

$$V_2 = \frac{9}{\pi \times 1.4^2/4} = 5.85$$

Thus,

h_5 = 0.07513 m

(6) Bend pipe loss

$$h_6 = f_e \times \frac{V_2^2}{2g}$$

where,

f_e : Friction coefficient 0.083

$$f_e = \{0.131 + 1.847 \times (D_2/2R)^{3.5}\} \times (\theta/90)^{0.5}$$

D_2 = 1.4 m
R = 3.5 m
θ = 33° 1' 26"

Thus,

$h_6 = 0.14537$ m

(7) Bend pipe loss

$h_7 = 0.14537$

(8) Butterfly valve loss

$$h_8 = f_c \times \frac{V_2^2}{2g}$$

where,

f_c : Friction coefficient 0.2

Thus,

$h_8 = 0.34879$ m

(9) Butterfly valve loss

$h_9 = 0.34879$

(10) Friction loss of pipe

$$h_{10} = f_e \times \frac{L_2}{D_2} \times \frac{V_2^2}{2g}$$

where,

f_e : Friction coefficient 0.027

$$f_e = \left\{0.0144 + 9.5/\left(1{,}000 \times \sqrt{V_2}\right)\right\} \times 1.5$$

L_2 : Pipe length 33 m

D_2 : Pipe inside diameter 1.4 m

Thus,

$h_{10} = 1.10991$ m

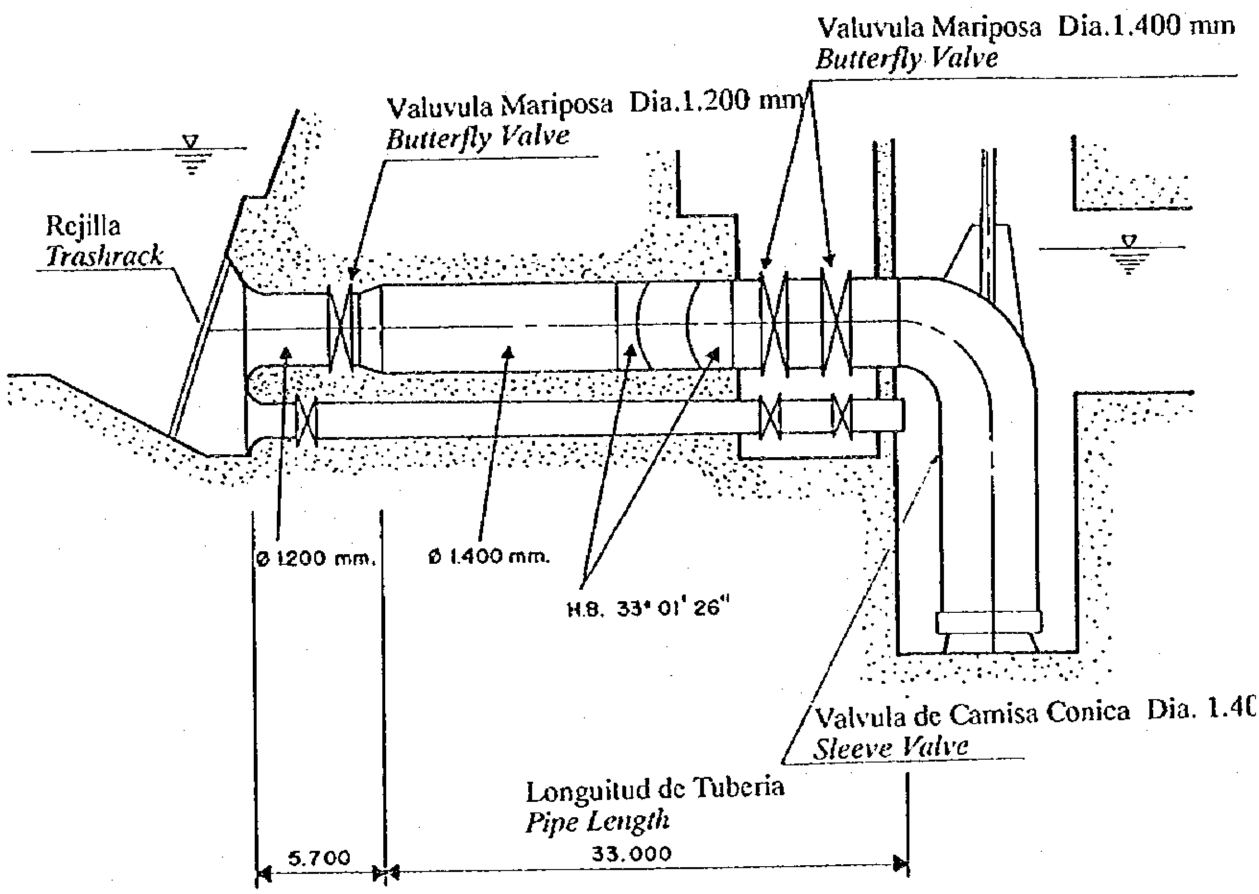
(11) Friction loss of pipe

$$h_e = h_1 + h_2 + h_3 + h_4 + h_5 + h_6 + h_7 + h_8 + h_9 + h_{10}$$

= 0.3 + 0.48491 + 0.64618 + 0.41436 + 0.07513 + 0.14537 + 0.14537 + 0.34879 + 0.34879 + 1.10991

= 3.9437 m → 4.0 m

ARREGLO DE LA BOCA DE ENTRADA A CONGUILLO
ARRANGEMENT OF CONGUILLO INLET



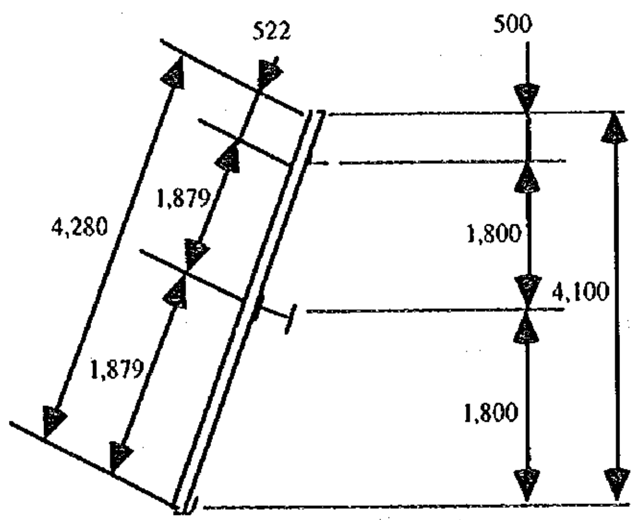
7.4 Poza Honda Inlet

POZA HONDA INLET
INTAKE FIXED TRASHRACK

1.Design Conditions

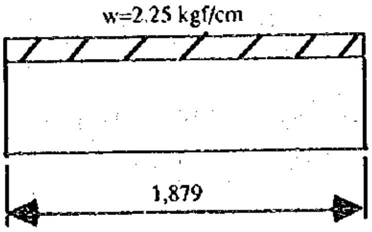
Type	:	Slant type fixed trashrack
Quantity	:	1 set
Clear span	:	4.0 m
Vertical height	:	4.1 m (Deck EL.95.500 m - Sill EL.91.400 m)
Gradient	:	1 : 0.3 (Θ = 73° ~ 18' ~ 3")
Slant length	:	4.280 m
Bar pitch	:	75 mm (center to center)
Design head	:	Water head difference of 3.0 m
Maximum deflection of supporting beams	:	1/600 of supporting span
Corrosion allowance	:	2.0 mm for bar elements and supporting beams (usually submerged in water condition)

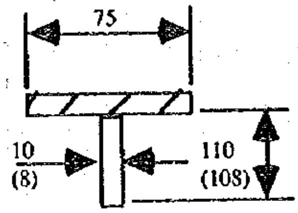
2.Arrangement of Trashrack



3.Bar Elements

(1) Bending moment and stress





a. Bending moment

$$M = \frac{W \times L^2}{8}$$

where,

M : Bending moment (kgf-cm)

W : Unit load on a bar

$0.3\ kgf/cm^2 \times 7.5\ cm = 2.25\ kgf/cm$

L : Maximum distance of center to center of supporting beam

187.9 cm

Thus,

$$M = \frac{2.25 \times 187.9^2}{8} = 9{,}930\ kgf - cm$$

b. Bending stress

$$\sigma_b = \frac{M}{Z}$$

where,

σ_b : Bending stress (kgf/cm^2)

M : Bending moment 9,930 kgf-cm

Z : Modulus of section

FB 110 x 10 15.5 cm^3
(108 x 8)

Thus,

$$\sigma_b = \frac{9,930}{15.5} = 641\ kgf/cm^2$$

(2) Critical stress considering horizontal buckling

$$C_r = 0.6 \times Y \times (1.23 - 0.0153 \times L/T)$$

where,

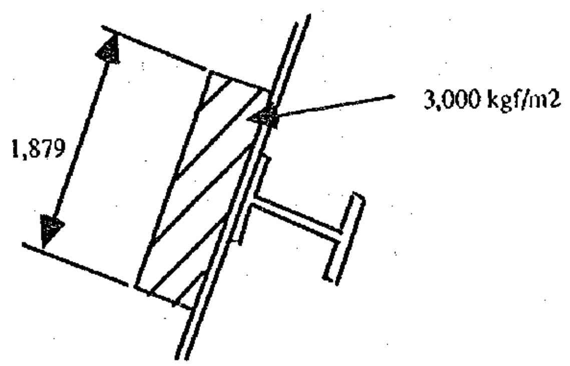
C_r : Critical stress (kgf/cm^2)
Y : Yield strength of material 2,400 kgf/cm^2
L : Laterally unsupporting length 35 cm
T : Thickness of bar o.8 cm

Thus,
C_r = 0.6 x 2,400 x (1.23 - 0.0153 x 35 / 0.8)
= 807 kgf/cm^2
σ_b = 641 kgf/cm^2 < C_r = 807 kgf/cm^2

4. Intermediate Supporting Beams

(1)Water pressure load

Water pressure load acted on each beam is as follows:



$$W_w = L \times h \times B$$

where,

W_w : Water pressure load (kgf)

L : Distance of center to center of supporting beams 1.879 m

h : Design head 3,000 kgf/m^2

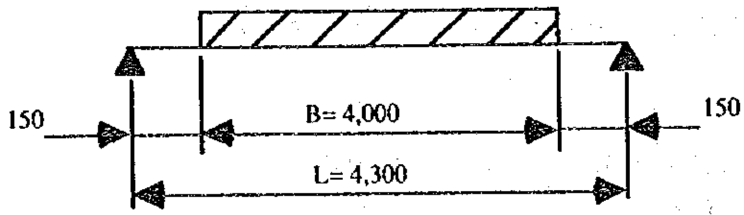
B : Clear span 4.0 m

Thus,

W_w = 1.879 x 3,000 x 4.0

= 22,548 kgf

(2) Bending moment and shearing force due to water load



a. Bending moment

$$M_x = W_w \times (2L - B)/8$$

where,

M_x : Bending moment due to water load (kgf-cm)

W_w : Water pressure load 22,548 kgf

L : Supporting span (B+30) 430 cm

B : Clear span 400 cm

Thus,

M_x = 22,548 x (2 x 430 - 400) / 8

= 1,296,510 kg-cm

b. Shearing force

$$S_x = W_w /2$$

where,

S_X : Shearing force due to water load (kgf)

W_W: Water pressure load 22,548 kgf

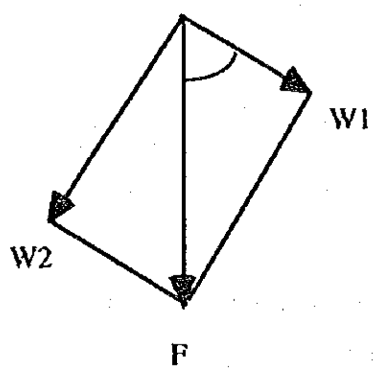
Thus,

$S_X = 22{,}548 / 2$

$= 11{,}274$ kgf

(3) Bending moment and shearing force due to own weight

The force due to own weight is distributed as follows :



W1 = F cos θ

W2 = F sin θ

where,

W1,W2 : Unit load of each direction (kgf/cm)

F : Unit load due to own weight 0.897 kgf/cm

θ : Angle between "F" and "W1" 73° ~ 18' ~ 3"

Thus,

W1 = 0.897 x cos 73° ~ 18' ~ 3"

= 0.258 kgf/cm

W2 = 0.897 x sin 73° ~ 18' ~ 3"

= 0.859 kgf/cm

a. Bending moment

$$M_1 = W_1 \times L^2 /8$$

$$M_2 = W_2 \times L^2 /8$$

where,

M1,M2 : Bending moment of each direction (kgf-cm)

W1,W2 : Unit load of each direction (kgf/cm)

W1 = 0.258 kgf/cm

W2 = 0.859 kgf/cm

L : Supporting span 430 cm

Thus,

$M1 = 0.258 \times 430^2 / 8$

$= 5{,}963$ kgf-cm

$M2 = 0.859 \times 430^2 / 8$

$= 19{,}853$ kgf-cm

b. Shearing force

$$S_1 = W_1 \times L / 2$$

where,

S1 : Shearing force due to own weight (kgf)

W1,L : Same as the above

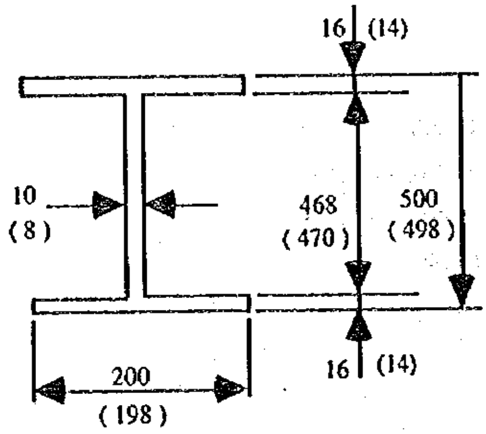
Thus,

$S1 = 0.258 \times 430 / 2$

$= 55$ kgf

(4) Bending and shearing stresses

Section properties



JIS G 3192 H-500 x 200 x 10/16 is used.

Moment of inertia

$Ix = 47{,}800\ cm^4$

$Iy = 2{,}140\ cm^4$

Modulus of section

$Zx = 1{,}910\ cm^3$

$Zy = 214\ cm^3$

Area of web $\quad Aw = 38\ cm^2$

a. Bending stress

$$\sigma_b = (M_x + M_1)/Z_x + M_2/Z_y$$

where,

σ_b : Bending stress (kgf/cm^3)	
Mx : Bending moment due to water load	1,296,510 kgf-cm
M1 : Bending moment due to own weight	5,963 kgf-cm
Zx : Modulus of section	1, 910 cm^3
M2 : Bending moment due to own weight	19,853 kgf-cm
Zy : Modulus of section	214 cm^3

Thus,

σ_b = (1,296,510 + 5,963) / 1,910 + 19,853 / 214

= 775 kgf/cm^2 $\quad < \quad \sigma_{ba} = 0.5\ \sigma_y = 1{,}200\ kgf/cm^2$

σ_{ba} : Allowable bending stress

b. Shearing stress

$$\tau_c = (S_x + S_1)/A_w$$

where,

τ_c : Shearing stress (kgf/cm^2)	
Sx : Shearing force due to water pressure	11,274 kgf
S1 : Shearing force due to own weight	55 kgf
Aw : Area of web	38 cm^2

Thus,

$\tau_c = (11,274 + 55) / 38$
$= 298$ kgf/cm^2 < $\tau_{ca} = 0.3\ \tau_y \neq 700$ kgf/cm^2

τ_{ca} : Allowable shearing stress

(5) Deflection

$$\delta = \frac{(W_w + W_1 \times B)}{48 \times E \times I_x}\left(L^3 - L \times B^2/2 + B^3/8\right)$$

where,

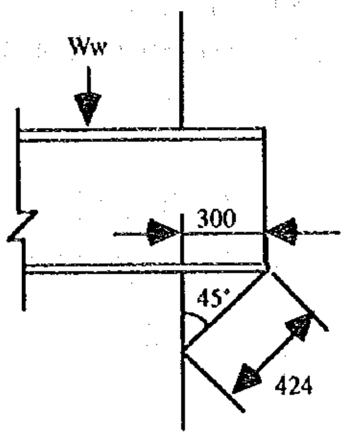
δ : Deflection of beam	(cm)
W_w : Water pressure load	22,548 kgf
W1 : Unit load due to own weight	0.258 kgf/cm
B : Clear span	400 cm
L : Supporting span	430 cm
E : Young's modulus	2.1 x 10^6 kgf/cm^2
Ix : Moment of inertia	47,800 cm^4

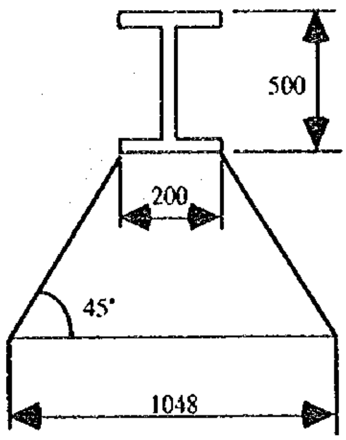
Thus,

$$\delta = \frac{(22,548 + 0.258 \times 400)}{48 \times 2.1 \times 10^6 \times 47,800}\left(430^3 - 430 \times 400^2/2 + 400^3/8\right)$$
$$= 0.250 \text{ cm}$$

$\delta/L = 0.250 / 430 = 1 / 1,720$ < $1 / 600$

5. Strength of Concrete





(1) Bearing stress of concrete

$$\sigma_c = W_w / (2 \times A_1)$$

where,

σ_c : Bearing stress of concrete (kgf/cm^2)

W_w : Water pressure load 22,548 kgf

A1 : Bearing area of concrete 30 x 20 = 600 cm^2

Thus,

σ_c = 22,548 / (2 x 600)

= 18.79 kgf/cm^2 < 60 kgf/cm^2

(Allowable concrete bearing stress)

(2) Shearing stress of concrete

$$\tau_c = W_w / (2 \times A_2)$$

where,

τ_c : Shearing stress of concrete (kgf/cm^2)

W_w : Water pressure load 22,548 kgf

A2 : Shearing area of concrete 2,646 cm^2

(20 + 104.8) / 2 x 42.4

Thus,

$\tau_c = 22{,}548 / (2 \times 2{,}646)$

$= 4.3\ kgf/cm^2 \quad < \quad 8\ kgf/cm^2$

(Allowable concrete shearing stress)

POZA HONDA INLET
PIPE LOSS HEAD

1. Design Condition

(1)	Design flow	4 m^3/s (2 m^3/s per unit)
(2)	Culvert size	2.5 x 4.0 m
(3)	Culvert length	39.3 m
(4)	Pipeline inside dia	900 mm
(5)	Pipe length	15.9 m

2. Calculation

(1) Hydraulic loss of trash rack

$$h_1 = 0.3\ m$$

(2) Loss of culvert inlet

$$h_2 = f_e \times \frac{V_1^2}{2g}$$

where,

f_c	:	Friction coefficient (Bellmouth)	0.2
V_1	:	Velocity	0.4 m/s

$$V_1 = \frac{4}{2.5 \times 4} = 0.4$$

Thus,

h_2 = 0.00163 m

(3) Friction loss of culvert

$$h_3 = \frac{n^2 \times V_1^2}{rh^{4/3}} \times L_1$$

where,

n	:	Roughness coefficient	0.015
V_1	:	Velocity	0.4 m/s
L_1	:	Culvert length	5.8 m
rh	:	Hydraulic radius	0.77 m

$$rh = \frac{2.5 \times 4}{2 \times (2.5 + 4)} = 0.77$$

Thus,

h_3 = 0.00030 m

(4) Transition culvert loss

$$h_4 = f_e \times \frac{V_1^2 - V_2^2}{2g}$$

where,

f_e	:	Friction coefficient	0.05
V_2	:	Velocity	

$$V_2 = \frac{4}{\{4 \times 0.5 + \pi \times 2^2 / 2\}} = 0.48$$

Thus,

h_4 = 0.00019 m

(5) Friction loss of culvert

$$h_5 = \frac{n^2 \times V_2^2}{rh^{4/3}} \times L_2$$

where,

n	:	Roughness coefficient	0.015
V_2	:	Velocity	0.48 m/s
L_2	:	Culvert length	33.5 m
rh	:	Hydraulic radius	1.07 m

$$rh = \frac{\frac{\pi \times 2^2}{2} + 4 \times 2.5}{\pi \times 2 + 2 \times 2.5 + 4} = 1.07$$

Thus,

h_5 = 0.00159 m

(6) Loss of pipe inlet

$$h_6 = f_e \times \frac{V_3^2}{2g}$$

where,

f_c	:	Friction coefficient (Bellmouth)	0.15
V_3	:	Velocity	3.14 m/s

$$V_3 = \frac{2}{\pi \times 0.9^2/4} = 3.14$$

Thus,

h_6 = 0.07564 m

(7) Bend pipe loss

$$h_7 = f_e \times \frac{V_3^2}{2g}$$

where,

f_e	:	Friction coefficient	0.078

$$f_e = \left\{0.131 + 1.847 \times (D/2R)^{3.5}\right\} \times (\theta/90)^{0.5}$$

D = 0.9 m
R = 2.25 m
θ = 28° 48' 38.84"

Thus,

h_7 = 0.03926 m

(8) Bend pipe loss

h_8 = 0.03926

(9) Butterfly valve loss

$$h_9 = f_e \times \frac{V_3^2}{2g}$$

where,

f_c	:	Friction coefficient	0.2

Thus,

h_9 = 0.10085 m

(10) Friction loss of pipe

$$h_{10} = f_e \times \frac{L}{D} \times \frac{V_3^2}{2g}$$

where,

f_e	:	Friction coefficient	0.030

$$f_e = \{0.0144 + 9.5/(1{,}000 \times \sqrt{V_3})\} \times 1.5$$

L	:	Pipe length	15.9 m
D	:	Pipe inside diameter	0.9 m

Thus,

h_{10} = 0.26726 m

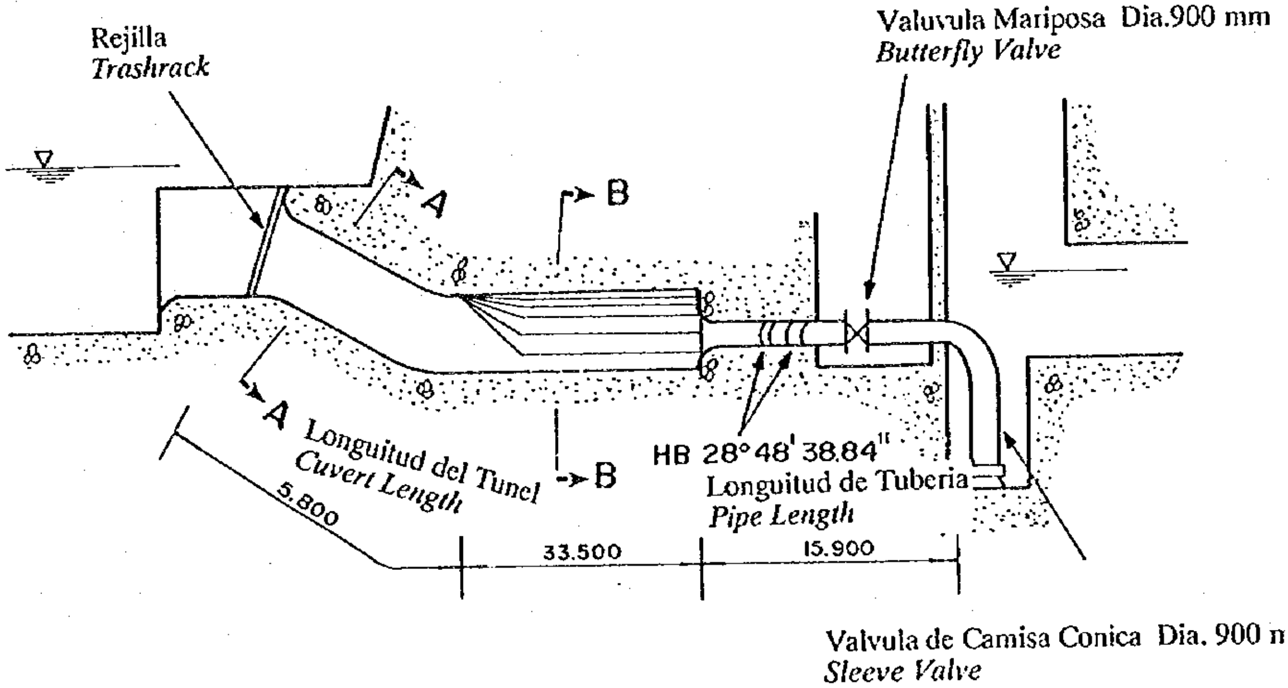
(11) Friction loss of pipe

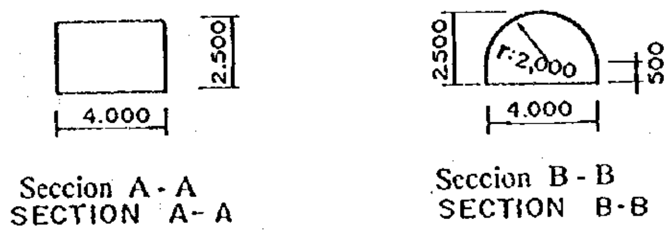
$$h_e = h_1 + h_2 + h_3 + h_4 + h_5 + h_6 + h_7 + h_8 + h_9 + h_{10}$$

= 0.3 + 0.00163 + 0.00030 + 0.00019 + 0.00159+ 0.07564 + 0.03926 + 0.03926 + 0.10085 + 0.26726

= 0.826 m → 0.85 m

ARREGRO DE LA BOCA DE ENTRADA A POZA HONDA
ARRANGEMENT OF POZA HONDA INLET





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