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Manual for the Design of a Simple Mechanical Water-Hydraulic Speed Governor
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by: Ueli Meier

HARNESSING WATER POWER ON A SMALL SCALE

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VOL. 5: MANUAL FOR THE DESIGN OF A SIMPLE
MECHANICAL WATER-HYDRAULIC SPEED GOVERNOR

AUTHOR: U. MEIER

(DRAFT)

SKAT

PREFACE

Energy is one of the prerequisites for rural development in developing countries while, at the same time, a worldwide scarcity of the traditional energy resources of the industrial era is imminent.

Many governments of developing countries have realised the huge potential of hydro power, a so far mostly untapped renewable energy resource not only in the megawatt range, but also on a small scale (10 - 100 kW), which seems the most appropriate for rural village electrification. It is mostly cost that has stood in the way of speedy and large scale development of small hydro power potentials. Imported and sophisticated equipment becomes costlier and costlier and is in most cases not economically feasible. This applies mostly to hydraulic equipment such as water turbines, accessories and governing devices. The case is different for alternators and switchgear which are produced in great number in industrialised countries and are therefore relatively cheap.

Experience in Nepal shows that it is possible to reduce costs of hydro electric generation projects vastly by minimising civil engineering and structural works and by producing hydraulic equipments in local workshops with simple designs and technology.

Nonavailability of a simple mechanical governor has long been a major obstacle in implementing small hydro projects with acceptable standards of safety. The Swiss Association for Technical Assistance, HELVETAS, in Zurich has sponsored a project to develop a simple governor that would be sufficiently accurate and reliable and could be manufactured by local workshops in Nepal. The work was undertaken by Messrs. H. Scheuer, A. Kilchmann and H. Jaeckle at the Institute of Hydraulics under Prof. Dr. B. Chaix of the Swiss Federal University of Technology (ETH) at Zurich, Switzerland, with the financial assistance of Service Overseas, Stuttgart.

Field information and criteria for design were supplied by Balaju Yantra Shala (P. Ltd. BYS, Nepal, where a prototype was built and tested in early 1979. The governor was designed for operating the gate of a Cross Flow Turbine (Banki), but may in fact be utilised on the flow regulator of any turbine. Because of a lack of other designs of simple governors and due to successful testing and introduction of this governor in Nepal, it was decided to prepare a instruction manual that may enable other organisations and individuals to adapt this governor to their own needs and improve it further as a contribution to the design of simple but reliable hydraulic equipment for small electricity generation units.

Manual for the Design of a Simple Mechanical Water-Hydraulic Governor

1. INTRODUCTION

For the control of hydraulic turbine used for electricity generation, there are
basically two possibilities:

a) load control, where the flow of water through the turbine is kept constant
   and where therefore the load has to be kept constant within tolerable
   limits. This is achieved today mainly with electronic controllers that
   switch any part of the load not consumed by the regular circuit into a
   ballast circuit thus keeping the total load on the turbine-alternator
   set constant;

b) flow control, of which the governor described here is an example and where
   the volume of water flowing through the turbine is adjusted depending on
   the load on the turbine-generator set.

Both systems keep the speed of the turbine within tolerable limits which, in turn,
results in a voltage and frequency of the generator that remains within a
specified range.

There exist virtually dozens of designs of governing devices for flow control.
Most are complex, costly, difficult to manufacture for a not specialised workshop
and really more accurate than would be required for a village electrification
scheme on a small scale.

For the sake of simplicity, the governor presented here was designed as
proportional governor, where each part load on the generator results in a respec-
tive speed of the turbine, remaining within ± 10% max. of rated speed, e.g. rated
speed + 10% at no load and rated speed - 10% at full load. This relatively large
speed variation ascertains good stability of the system. Initial experience shows
that it may be possible to reduce speed variation to ± 5% and still maintain
sufficient stability.

2. DESCRIPTION OF THE SYSTEM

A mechanical-hydraulic type of governing system was chosen. The working medium is water in a open circuit with the working pressure supplied by the head of the turbine installation. The device consists of the following components : (refer to Fig. 1)

1. Flyball assembly (centrifugal pendulum)
2. Throttle / pilot valve system
3. Lever linkage
4. Servo cylinder with closing spring
5. Turbine flow regulator (gate)
6. Water supply line with filter
7. Rotating masses (flywheel etc.)

H = Head
Q = Flow volume
P_0 = Pressure before throttle
P_1 = Pressure after throttle
N = Turbine speed
O = No load
F = Full load
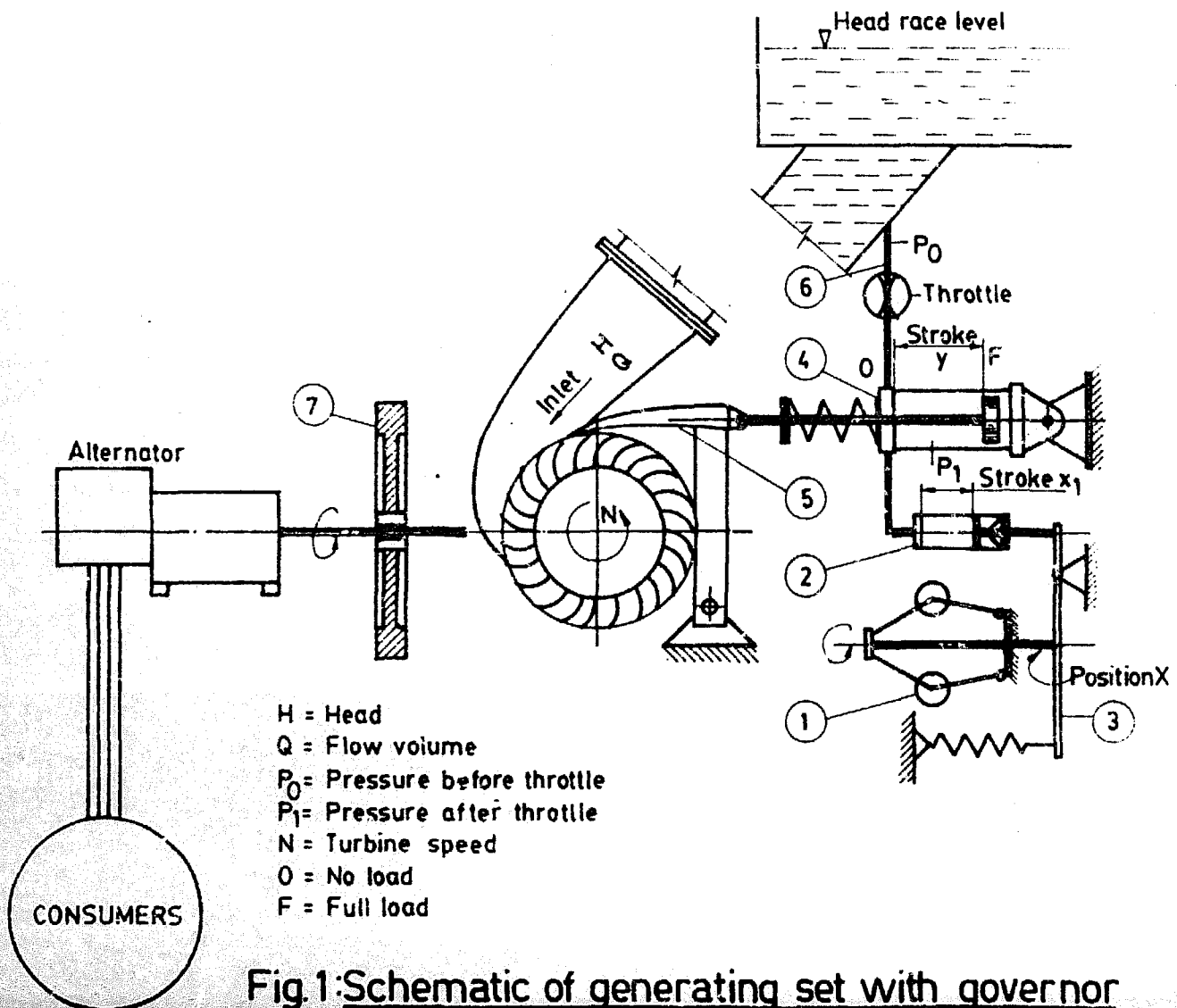
Fig.1:Schematic of generating set with governor   5

2.1 FLYBALL ASSEMBLY

The centrifugal pendulum is directly mounted on the turbine shaft and serves as a tachometer. The flyball mass will shift a pushrod against a spring depending of turbine speed. Thus for each turbine speed (within ± 10% of rated speed) there is a respective position of the flyball-pushrod. This position may be altered by varying the pretension of the flyball assembly spring. This is used for the adjustment of the nominal speed when the governor is tuned initially.
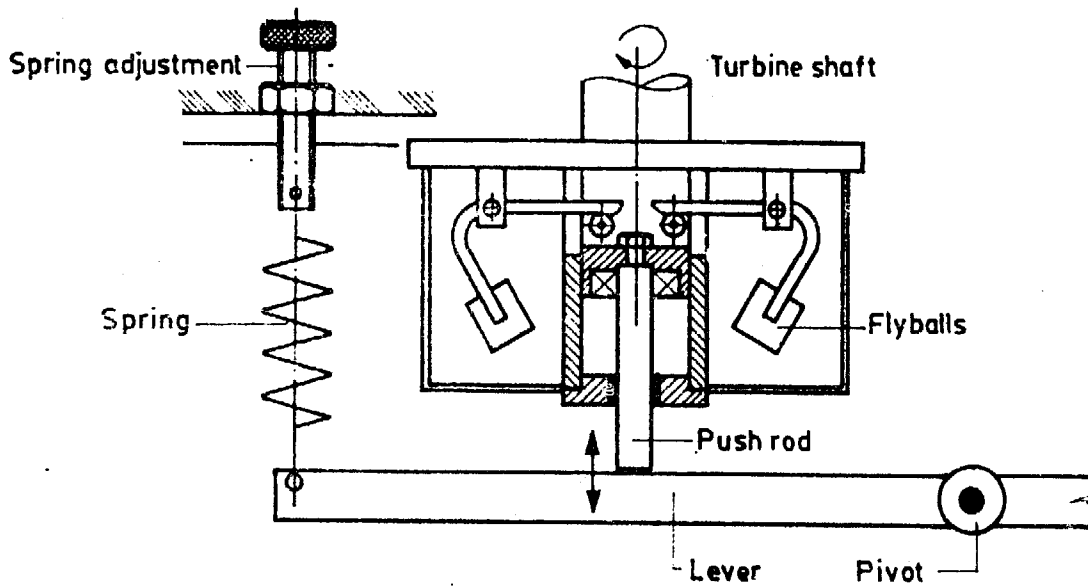
Fig. 2 Schematic: Flyball assembly

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2.2 THROTTLE / PILOT VALVE SYSTEM

The principal function of these components is to vary the water pressure in the servo cylinder as required by varying load conditions of the turbine. Water, under the static head of the installation is brought in a separate pipe to the governor; it passes first through the throttle and is discharged into the turbine pit through the pilot valve. Between throttle and pilot valve, the hydraulic cylinder is connected to the system as indicated in fig. 3.
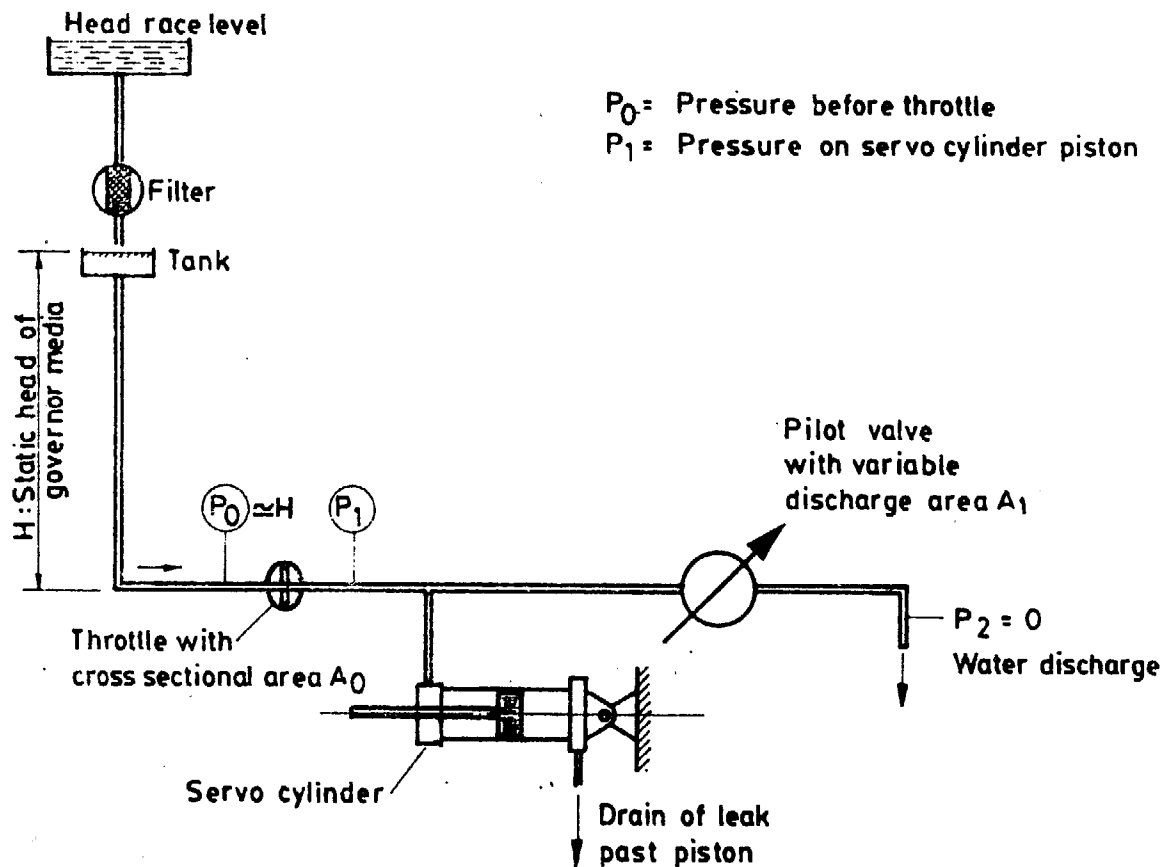
Fig. 3: Schematic of Throttle/pilot valve system

The pressure P_1 may be varied by varying the discharge area A2 in the pilot valve. The shape of the pilot valve piston is designed such that the characteristic of P_0/P_1 over the stroke of the pilot valve is linear, thus making it possible to get a speed and load defined pressure P_1, acting on the servo cylinder if the pilot valve piston is connected via a lever linkage to the flyball/pushrod.

2.3 LEVER LINKAGE

The lever as indicated in Fig. 4 serves to transmit the reaction of the
flyball assembly when turbine speed changes to the pilot valve piston. The
flyball spring is attached to the same lever, to counteract the flyball
effect. As mentioned in para. 2.1, the nominal position of the flyball
pushrod may be adjusted by a arrangement of varying the spring tension.
Another arrangement shown in fig. 4 serves to adjust the position of the
pilot valve piston in relation to the flyball pushrod position.
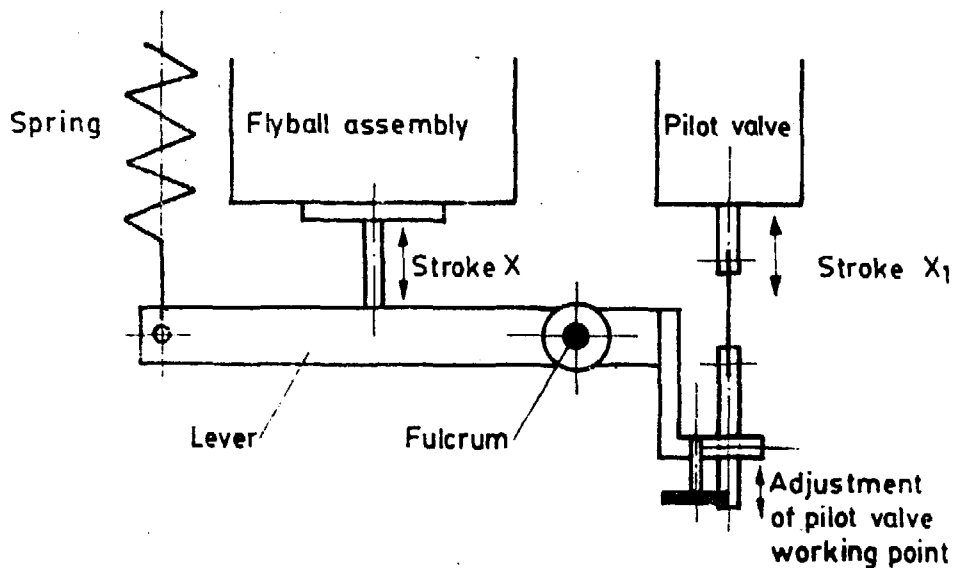
Fig. 4 : Lever linkage (schematic)

2.4 SERVO CYLINDER / CLOSING SPRING

This is the actual power component of the governor that moves the turbine
regulator as commanded by the flyball assembly. Hydraulic forces act in the
direction of opening the turbine gate only while forces required for closing
are supplied by a helical compression spring. In the static state, as long
as no load and therefore no speed changes occur, hydraulic forces in the
cylinder and spring forces are in balance, keeping the turbine gate in a
speed (load) defined position. It is necessary to be able to adjust the
gate position initially.

As shown in fig. 5, an arrangement is therefore made to vary the spring tension.
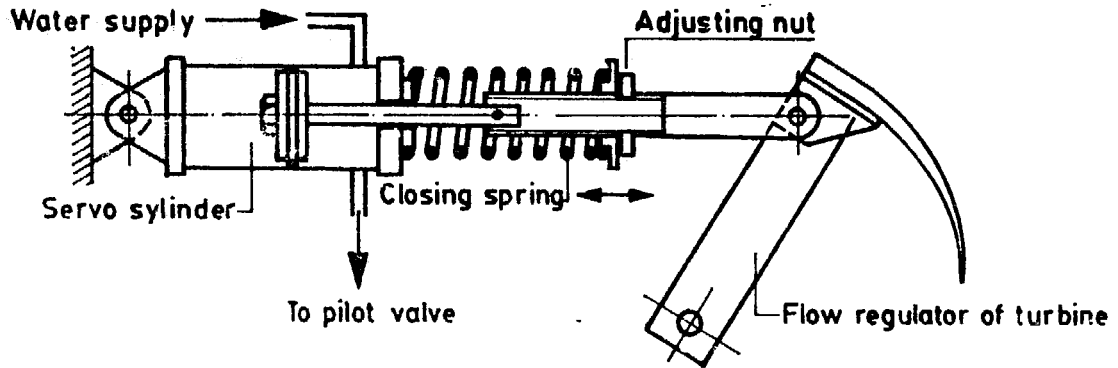
Fig. 5: Schematic of servo cylinder/closing spring arrangment

2.5 TURBINE FLOW REGULATOR

The gate of the turbine for which the governor was initially designed, has a physical shape as indicated in fig. 5. Any other design of the flow regulator is however possible, if the hydraulic forces acting on it remain within limits. Big forces required to move the gate have a direct bearing on the size of servo cylinder and closing spring. With a unduly large cylinder, governor action may be intolerably slow.

2.6 WATER SUPPLY LINE WITH FILTER

In most situations it will be necessary to filter the water that is used in the governor. A simple basket strainer type of filter was used so far. The basket consisting of very fine meshed stainless steel gauze has to be removed manually for cleaning. Any particle of foreign matter having a size of over 0.04 mm should be removed from the water to avoid abrasion or jamming of the pilot valve. Filter pressure required is between 2 and 3 meters. The filtered water is collected in a tank of approx. 100 lt volume from where a 50 mm HDPE pipe brings water to the governor. Several stop valves are incorporated in the installation to make cleaning of the filter, flushing of the line after

cleaning and "switching on" of the governor possible. The separate governor supply line also prevents shock waves that may occur in the penstock pipe from adversely affecting the governor performance.
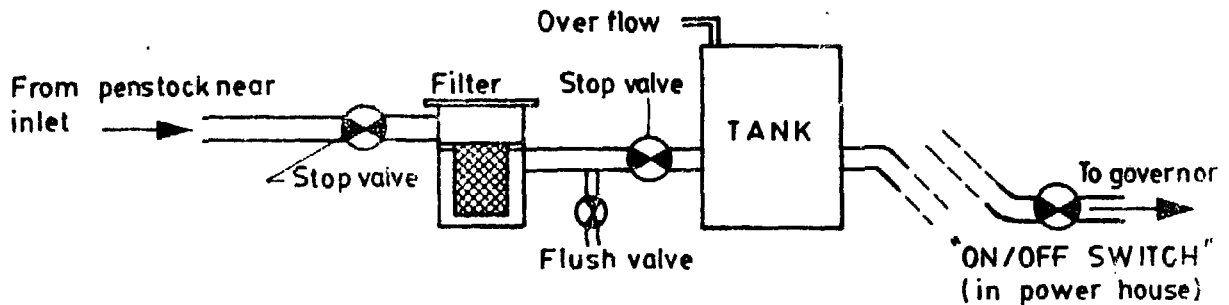

Fig.6: Schematic of governor supply system

2.7 ROTATING MASSES

In a system using a flow control governor, a flywheel is usually incorporated. The totality of rotating mass consists then of the following parts : turbine rotor, transmission components such as pulleys and belts or gears of a gear box, flywheel, coupling and rotor of the alternator. The value of these rotating masses may be expressed as the GD^2 in units of kg m^2 usually reduced to the shaft with highest speed, e.g. all masses revolving at lower speed have to be multiplied by the inverse speed ratio and may then be added to the masses revolving at highest speed (usually 1500 RPM).

If a turbine set to work at a certain load, is suddenly relieved to no-load or part-load, or is inversely loaded from part-load to full-load, the governor cannot in an instant move the flow regulator to the new position called for by the change of load, because the piston of the servo-cylinder moves with a definite velocity and requires a definite time to travel a given stroke. A transitory state accurs, during which the power generated by the turbine rotor is higher (or lower in the case of load increase) than the output of the alternator. The surplus (deficiency) energy is taken up (supplied) by the revolving masses of the set which are being accelerated (retarded). Momentarily the set acquires higher (lower) speed which is later brought to a defined value by the governor. The maximum deviation from the initial speed is depending on the percentage of load being switched, the totality of rotating masses inherent in

the set and the characteristics of the governor. For the same permissible
maximum speed deviation, a set of higher output requires a bigger rotating
mass and values of GD^2 for the rotating mass of 1 - 1,5 kg m^2/kW have been
found to be acceptable in tests with this simple governor.

The flywheel effect is lessened by such factors as high head and relatively
long lengths of penstock, because in such cases pressure changes may be con-
siderable. Calculations for the required flywheel effect (GD^2) become rather
complex and exceed the scope of this manual. The serious reader may refer to
relevant literature on hydraulic turbines such as "Water Power Engineering"
by H.K. Barrows, published by McGraw Hill, p. 549 ff.

3. DETERMINING CHARACTERISTIC CURVES OF THE SYSTEM AND ITS COMPONENTS

Characteristics of flyball assembly, throttle/pilot valve system and flow regu-
lator forces are interdependent and have to be matched to get a functioning governor.
In order to be able to select such standard elements as hydraulic cylinders, tension
spring (flyball) and compression spring (gate closure), these characteristic curves
have to be worked out.

3.1 FLYBALL ASSEMBLY PUSHROD FORCES

A force/stroke diagramme is drawn first of forces on the pushrod due to the
flyball effect (centrifugal forces). This may be done by measuring the
forces at various pushrod positions at constant speed. Ideally, 3 curves
are established at rated speed (N), N + 10% (max. speed) and N - 10% (min.
speed). The relation of pushrod forces and speed may be expressed by the
equation

$$F_p = m \cdot r \cdot w^2$$

where F_p is the force on the pushrod, m is the rotating mass, r is the radius
of rotation and w^2 is the angular velocity squared; by measuring the character-
istic curve of the flyball pushrod assembly at any speed, curves for other
speeds may be extrapolated since m and r remain identical at identical push -
rod positions. To get a speed-defined position of the flyball pushrod, a
spring has to counteract the flyball forces. To determine the c-value of
this spring, the point of full load (N - 10%) is connected with the point
of no load (N + 10%) in the characteristic diagramme of the flyball assembly

as shown in fig. 7. The spring characteristic so determined has still to be corrected for the lever ratio between flyball assembly and spring, as will be shown later on.

$$\text{Spring characteristic } C' = \frac{(F_{x_2} - F_{x_1})}{(X_2 - X_1)} = \frac{N}{mm}$$
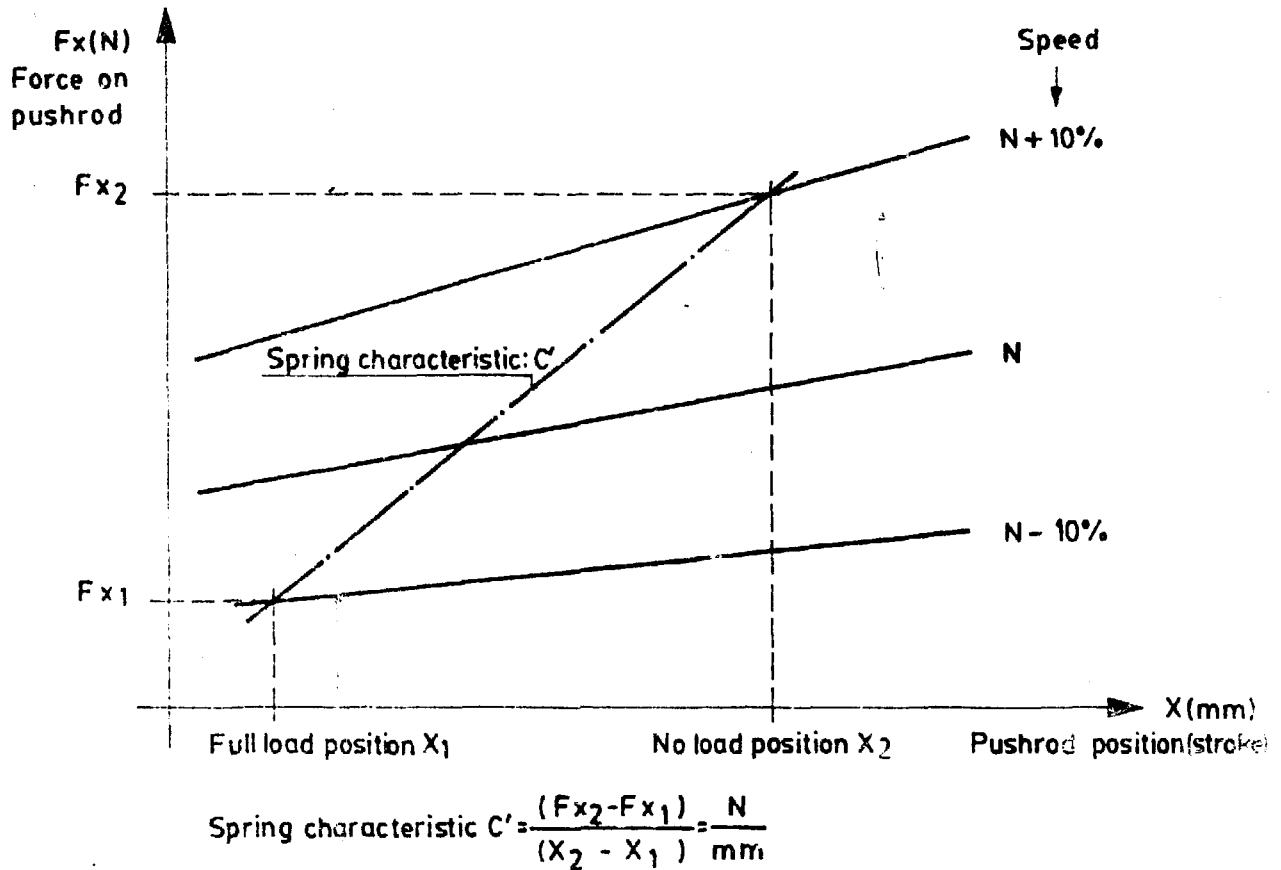
Fig.7: Characteristic diagramme of flyball pushod

The speed/stroke diagramme may now be drawn with the uncorrected c-value of the spring as shown in fig. 8, with pushrod positions taken from fig. 7.
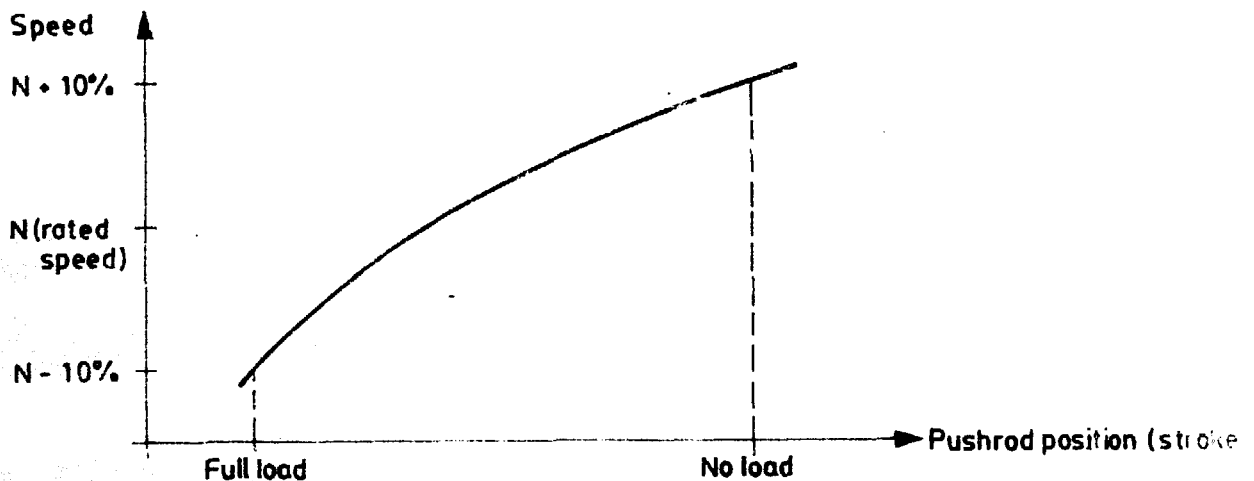

Fig.8: Fly-ball assembly: speed/stroke diagramme

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3.2 OPERATING FORCES ON FLOW REGULATOR

These are optimally measured on a turbine under operation at various gate
positions. The characteristic curve needs to be regular although not linear.
For an irregular characteristic where the same operating force is required at
different openings, this simple governing system would not work properly and
such a flow regulator would have to be redesigned. Fig. 9 gives an idea how
a gate characteristic may look like.
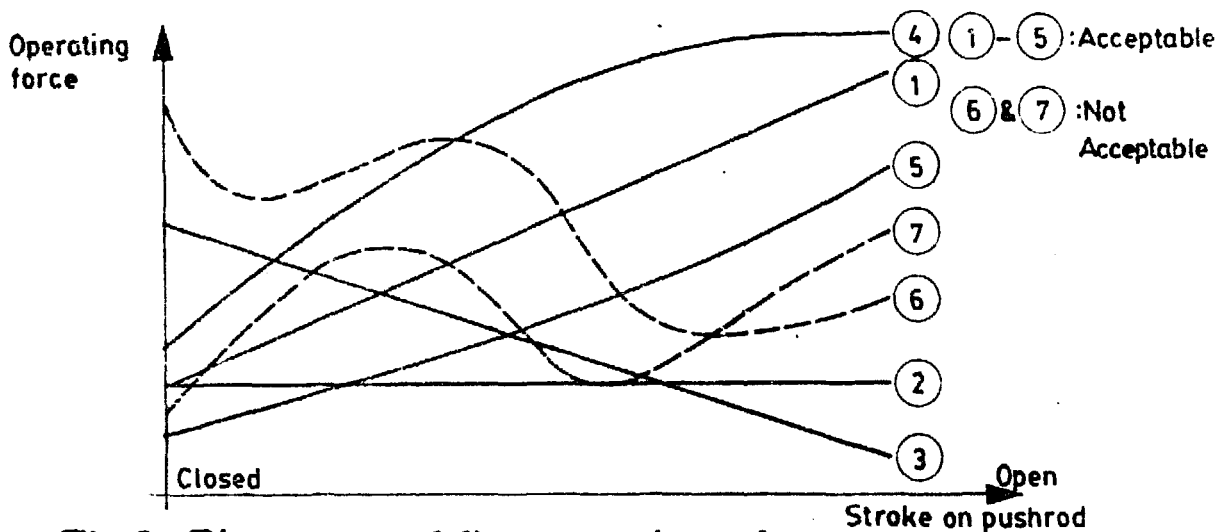
Fig.9: Diagramm of flow regulator forces

If a turbine gate is reliably selfclosing and rather linear in characteristic,
it is possible to make a design without closing spring. In such a case, the
servo cylinder would work against the forces of the gate when load changes
occur and forces would balance against each other under constant load condi-
tions. In all other cases, a closing spring is required and the spring
characteristic is imposed on the gate characteristic, resulting in the
required servo cylinder characteristic curve as illustrated in fig. 10.
However, dimensioning of servo cylinder diameter and spring will still have
to wait till the pilot valve is designed.
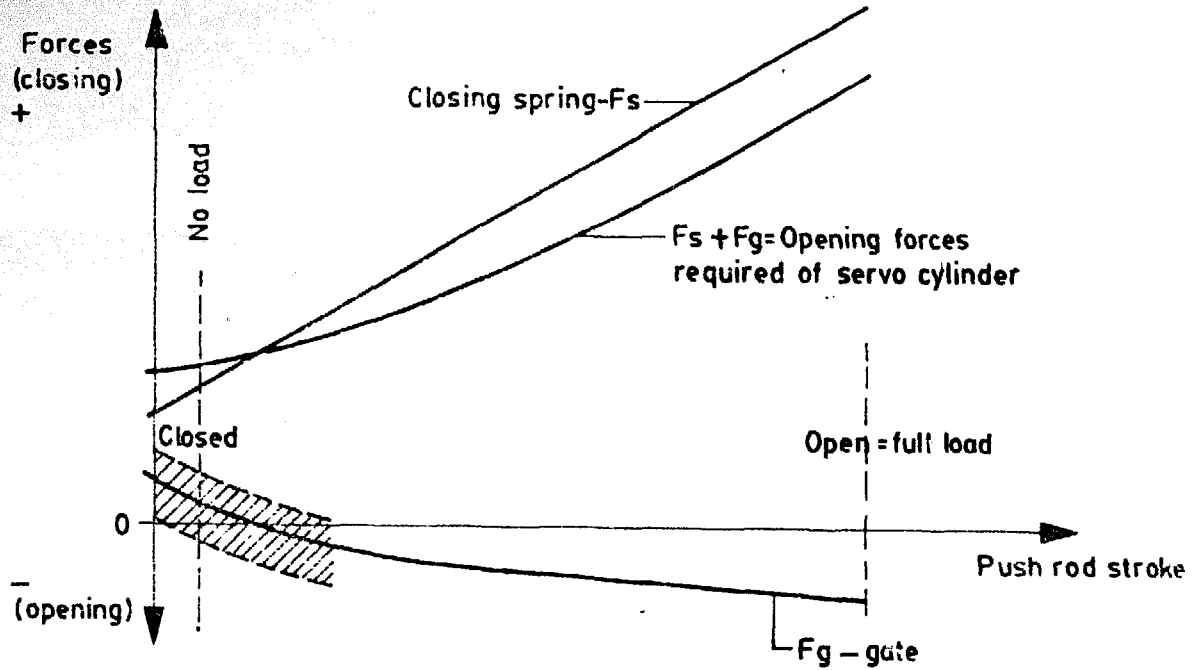
Fig.10: Flow regulator forces diagramme

3.3 DESIGN OF THROTTLE / PILOT VALVE SYSTEM

With pressure available in the system (Po) given and an assumed orifice diameter (Ao) of the throttle, the shape of the pilot valve piston - slit (A1) may be calculated to get the desired characteristic curve.
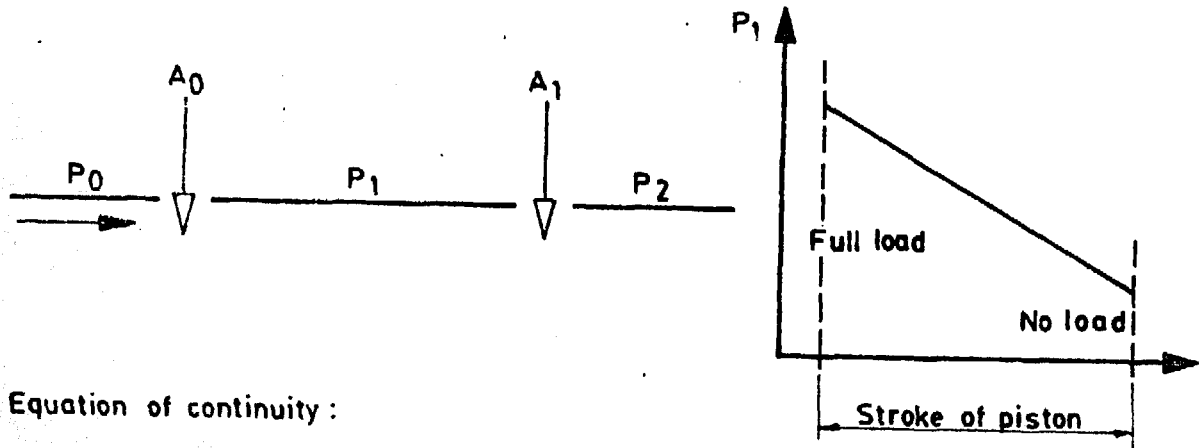
Equation of continuity :

$$A_0 \propto_0 \sqrt{2\,\frac{P_0 - P_1}{\gamma}} = A_1 \propto_1 \sqrt{2\,\frac{P_1 - P_2}{\gamma}}$$

may be simplified under the condition of $\propto_0 = \propto_1$ and limited to $P_2 = 0$ (drain), thus :

$$\frac{A_1}{A_0} = \sqrt{\frac{P_0}{P_1} - 1}$$

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This function is drawn in fig. 11 (see page 15) which may be used as an aid in slit calculations. The system has a certain leakage which shall be treated like an additional opening of the pilot valve. An increase of the still possible ratio P_1/P_0 may be achieved by increasing the orifice diameter of the throttle.

For the design of the pilot valve piston, we select now a stroke by which the piston should move between "no load" and "full load" of the set, of say 12 mm. Also, we have to decide on the minimum and maximum pressure ratio in the respective end positions of the piston. Values of 0,3 and 0,7 at no load and full load respectively have proven very satisfactory in our tests. Between these values the pressure ratio curve should be linear e.g. $\frac{(0,7 - 0,3)}{12} = 0,033$ of stroke as shown in fig. 12.
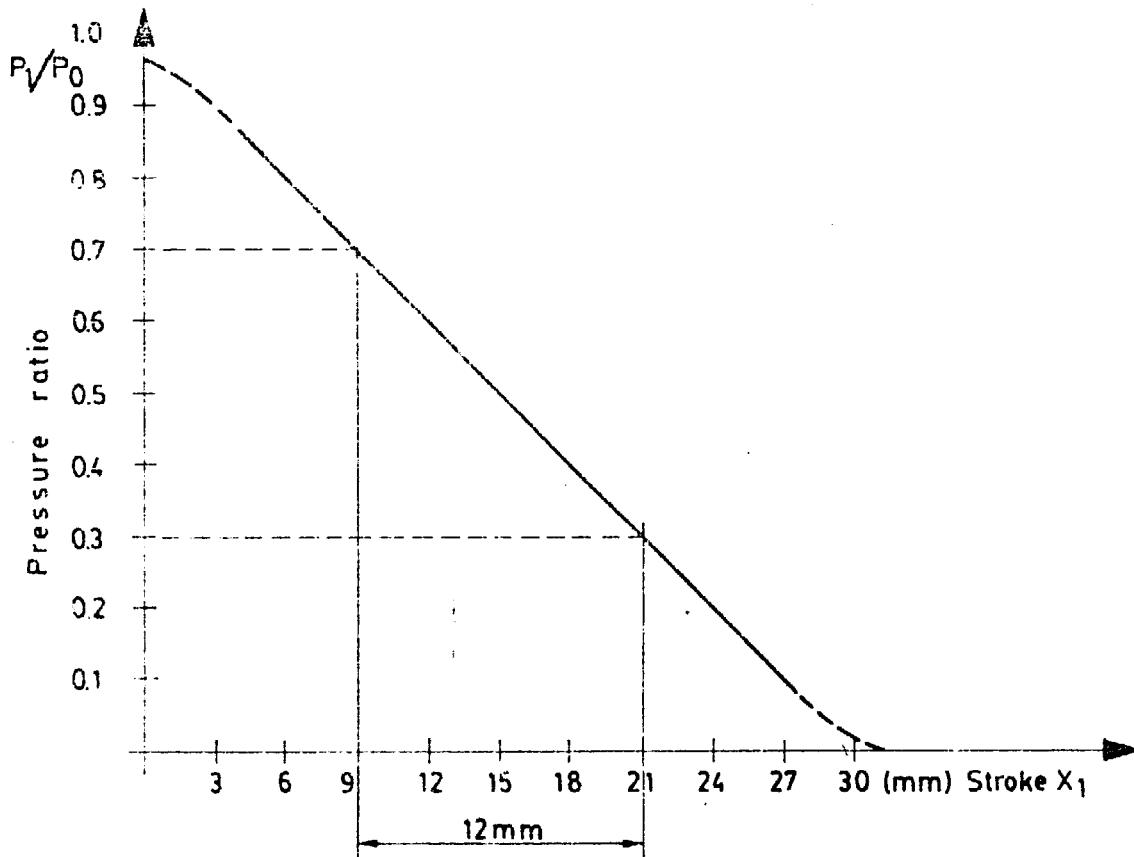
Fig.12: Characteristic curve for throttle / pilot valve system

Assuming a throttle orifice diameter of 18 mm, the width of slits may now be calculated in a tabular form, according to the formula $P_1/P_0 = \frac{1}{(A_1/A_0)^2 + 1}$ or by using the diagramme in fig. 11 (page 15).

The increment of discharge area (ΔA) has to be divided by the stroke increment (s = 3 mm) to get the discharge arc (2a). We execute the piston with two identical slits, arranged symmetrically on the piston. The value of 2a has therefore to be divided by 2 and of this arc, the chord (c) is calculated. The biggest possible discharge arc is limited by the piston housing and is 140° in our design as indicated in fig. 13. C then is the width of the slit at this point of the piston. Values of c are then plotted into the diagramme in fig. 1⁵ This diagramme is later used to find the required cutting depth for the respective cutter diameter in fabricating the piston slit. For practical purposes the first portion of the curve is simplified as shown with a dotted line.
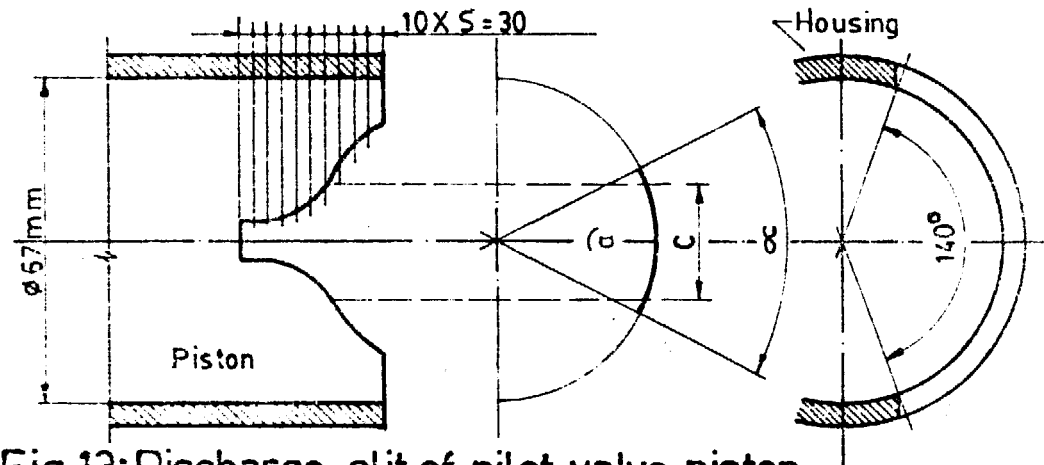
Fig.13: Discharge slit of pilot valve piston

Table I: Calculation table for slit width; $A_1 = 254.47 mm^2$ (φ18)

Stroke X_1 (mm)	P_1/P_0 design	$A_1'(mm^2)$ incl.leak 20mm²	$A_1 (mm^2)$ (net)	ΔA	2 â	∝(°)	C(mm)
0	0.95	58	38				
				27	9	7.7	4.5
3	0.9	85	65				
				42	14	12.0	7.0
6	0.8	127	107				
				40	13.3	11.4	6.65
9	0.7	167	147				
				41	13.7	11 7	6.83
12	0.6	208	188				
				46	15.3	13.0	7.6
15	0.5	254	234				
				58	19.3	16 5	9.6
18	0.4	312	292				
				77	25.7	22.0	12.8
21	0.3	389	369				
				117	39.0	33.4	19.2
24	0.2	506	486				
				257	85 7	73.3	40.0
27	0.1	763	743				
				346	115.3	98.6	50.8
30	0.05	1109	1089				

Hydraulic system

$$\frac{A_1}{A_0} = \sqrt{\frac{P_0}{P_1} - 1}$$

Fig. 11: Diagramme: Pressure ratio as a function of area ratio

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Fig. 14 : Diagramme of slit width

4. SELECTING SERVO CYLINDER, CLOSING SPRING, FLYBALL-SPRING AND LEVER RATIO OF LINKAGE

All these components may conveniently be chosen by using a set of calculation forms for each installation. In order to reduce the number of standard components such as cylinders and sp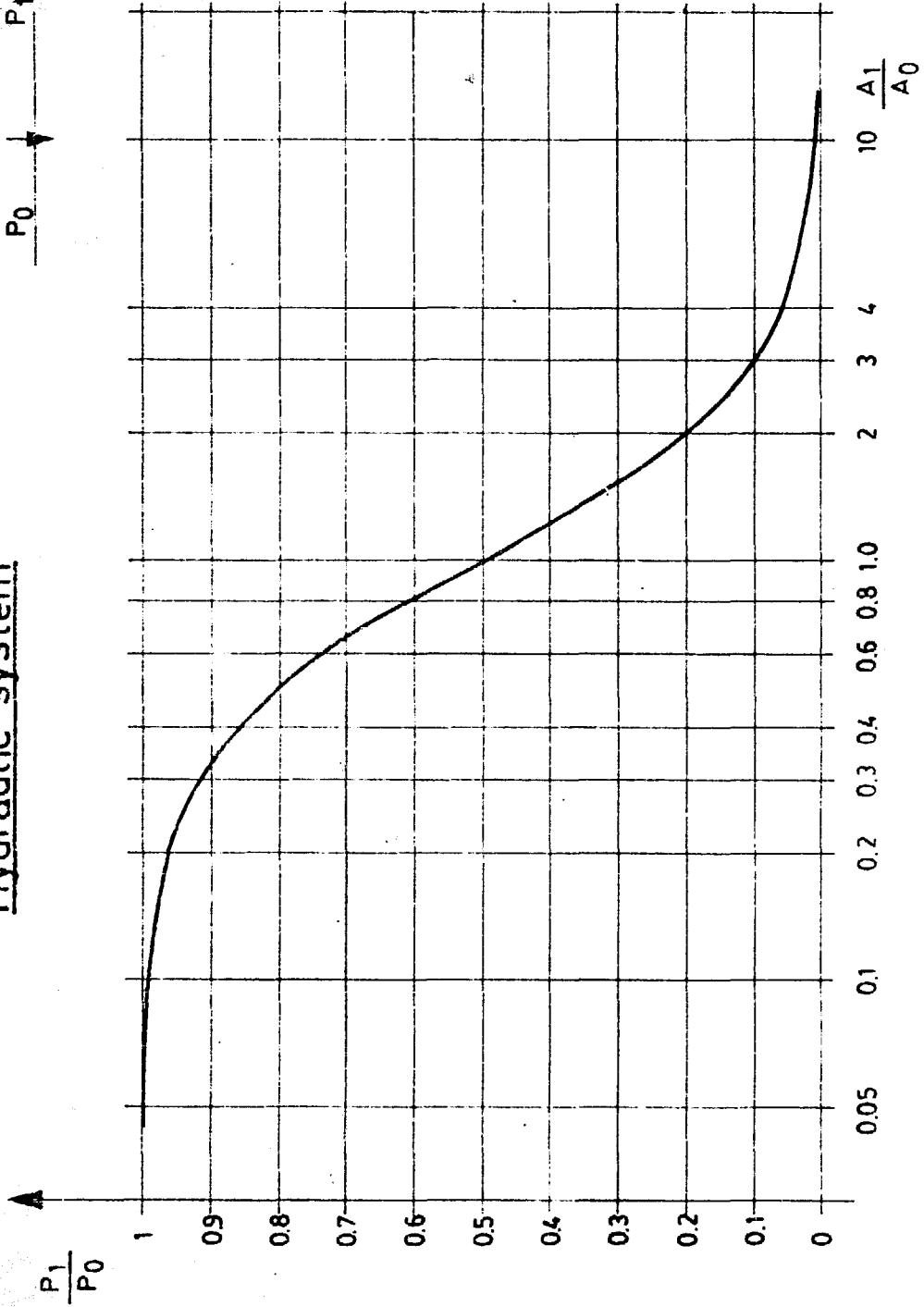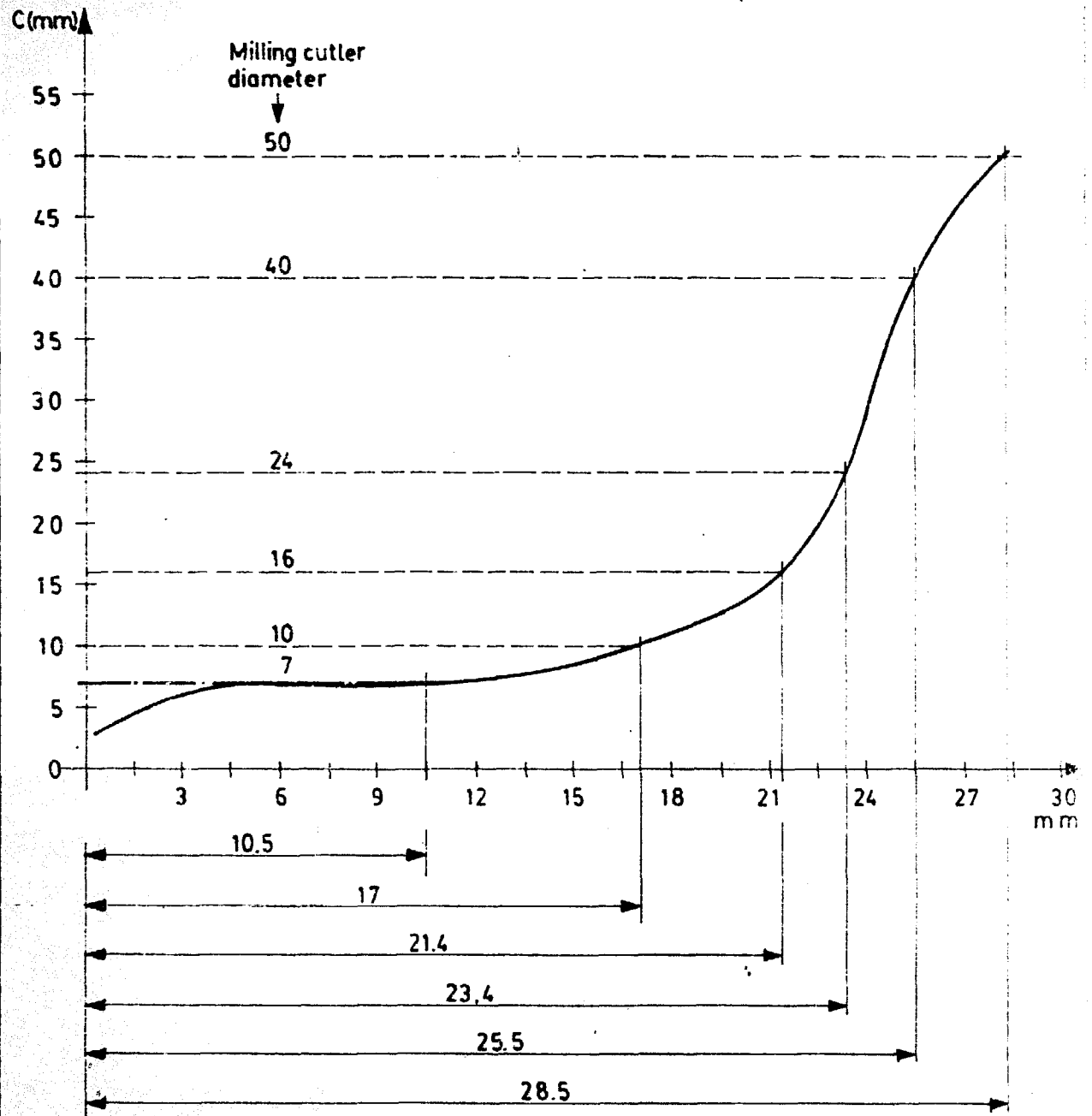rings for different design parameters, the design stroke of the pilot valve and the lever ration of the linkage between the flyball assembly and pilot valve may be varied. This makes it possible to design the governor components for heads from 10 to 40 meters, using only three different sizes of servo cylinder, three closing springs with different c-values and also three different flyball springs.

4.1. SELECTION OF CYLINDER DIAMETER

Table II may be used for a tentative selection of the cylinders diameter. This table is based on experience with the gate design as described in para. 2.5. Other designs may require different cylinders and the table should therefore serve as a guide only.

CYLINDER PISTON ⌀	PRESSURE HEAD(m)							
	10	20	30	40	50	60	70	80
⌀ 50				▬▬▬▬▬▬▬▬▬▬▬▬▬▬▬▬▬				
⌀ 60			▬▬▬▬▬▬▬▬▬					
⌀ 80		▬▬▬▬						
⌀ 100		▬▬▬						
⌀ 125	▬▬							
⌀ 160	▬▬							

TABLE II : Tentative selection of piston diameter

If the pressure in the servo cylinder is measured in an actual installation, we find that the full static head is equivalent to Po in the cylinder only at fully closed pilot valve position ($P1/Po = 1$).

In all other positions, represented by values of P_1/P_0, only a fraction of the static head is available as the actual cylinder pressure P_0.

This is shown in fig. 15, where the actual value of P_0 is plotted as a function of the pressure ratio P_1/P_0. The diagramme is again based on actual measurements and may have to be corrected for a specific pilot valve design and for other pressure heads.
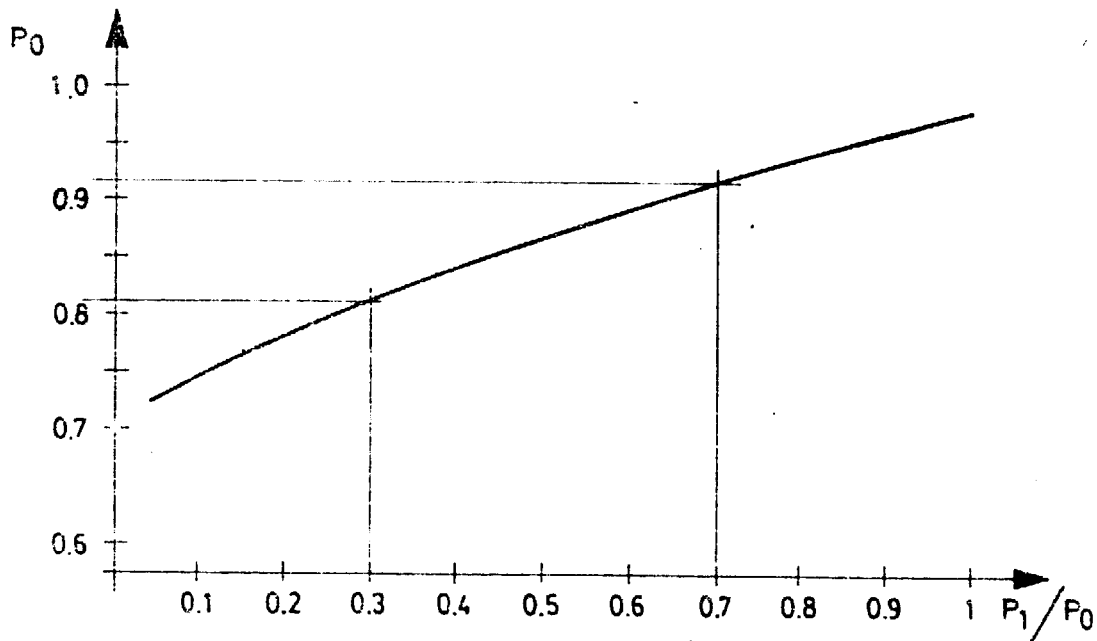
Fig.15:Available pressure P_0 along piston stroke
Testing condition $H_G = 11\,m$ throttle $\phi\,18mm$

Using form MEG 1 - 02 as attached, we may now calculate the cylinder forces available at the respective ratio of P_1/P_0 by multiplying the static pressure head by the effective factor of P_0 and finally multiplying these values with the effective cross sectional area of the selected servo cylinder.
(Please refer to design examples in annex I and II.)

4.2 SELECTION OF CLOSING SPRING

Still using form MEG 1 - 02 we calculate now a tentative c-value of the closing spring by substracting the cylinder force at $P_1/P_0 = 0,3$ from the respective value at $P_1/P_0 = 0,7$ and by dividing the result by the full stroke on the gate pushrod from no-load to full load; as shown in the examples, we then simply choose the nearest standard spring.

4.3 DETERMINING REQUIRED STROKE $x1$ OF THE PILOT VALVE

Continuing on form MEG 1 - 02 on the left side, we determine now the required closing spring forces at no-load (position $P_1/P_0 = 0,3$) by adding up cylinder forces in this position with turbine gate forces and friction inherent in the system (gate forces : + if in opening direction, - if in closing direction).

To find the required ratio of P_1/P_0 at full load, we continue on the right side of the same form by adding the force differential of the closing spring between stroke 0 and full stroke to the value Fn1 found before, plus gate forces and friction (attention : since we are now investigating the forces required to open, the signs for gate forces change, e.g. closing forces + and opening forces -). With the value found for Ff1, we then find the value of P_1/P_0 by interpolation from the table of servo cylinder forces. The stroke required to reach the value of P_1/P_0 at full-load so determined may now be found in the diagramme as shown in examples in annex I and II/3.

4.4 FLY-BALL SPRING/LEVER-LINKAGE DESIGN

For these calculations we use the centrifugal pendulum characteristic curves from form MEG 1 - 04 (annex I and II/4) where pushrod forces for full/load speed may be found at 3 mm stroke and no-load forces at 21 mm stroke. On form MEG 1 - 05 these forces should be entered in the column for a speed deviation of $\pm$ 10% and also for $\pm$ 7% since we will first try to achieve the latter.

The pilot valve stroke x1 is known from previous calculations and the lever length lx may be calculated to get this troke by the formula given. We then choose one of the three flyball spring (FBS 01 - 03) and calculate the spring

lever length lx' by the given formula for both cases $\pm$ 10% and $\pm$ 7% speed deviation. If the result for lx' is outside the possible adjustment range, we repeat the procedure with one of the other springs. Finally, we ascertain that the calculated maximum. spring deflection is permissible and repeat the calculating procedure again with another spring, if this is not the case. With this then, all the governor components are determined.

5. Physical Design Aspects of the Governor

Although the design of the governor is rather simple, it is evident that manufacturing tolerances will affect performance and trouble-free operation over a long time period. Wear will certainly play its role and it can safely be assumed that certain improvements will have to be made as experience with the operation of the existing design might show.

5.1 Flyball Assembly

The flyball assembly is made from mild steel except for bearing pins that should be hardened. The pushrod is provided with a deep grove ball-bearing and is prevented from rotating by the lever linkage that fits into the slot at the pushrod end. The actual flyweights are bolted to their respective arms by 1/4 " bolt and nut;these connections shall have to be prevented from loosening. The whole assembly is mounted directly on the turbine shaft.

5.2 Throttle Pilot Valve System

Function : water flows through the throttle into the cylinder and from there through the holes in the end cap and the pilot valve piston through the control slits out of the pilot valve housing.

All parts in contact with water should be made from corrosion free materials. Since the cylinder end plate is made from aluminium, we have chosen aluminium for the parts that connect directly to the cylinder. At the entrance to the cylinder a bush is screwed-in that takes up the actual throttle. Different throttle diameters may therefore be used simply by exchanging this bush. On top of the throttle bush a nozzle is screwed to which the flexible plastic inlet pipe is connected. At the cylinder outlet side an identical nozzle is

screwed directly into the endplate for the connecting flexible pipe to the pilot valve inlet.

The pilot valve assembly is made entirely from brass with a gap between the housing inner diameter and the piston of approx. 0,1 mm. The piston pushrod is pressed into the piston central bore and secured with a set screw. The whole assembly is fitted into the pilot valve assembly chamber welded up from steel sheet with O-rings as a seal at both ends. In order to provide a slight negative pressure for easy water escape, this chamber is provided with a drain pipe at the bottom that reaches just below the tail race level (drafttube effect). It is important that the pilot valve housing is mounted with the two slots exactly in a vertical position and there fixed with a pin.

5.3 LEVER LINKAGE

To connect the lever to the pilot valve pushrod without any play, a length of flat spring steel of 0,5 mm thickness is used. This provides for slack free transmission of the lever movement despite the radial movement of the lever as opposed to the linear movement of the pushrod.
The pivot point of the lever is fitted with a brass bush that fits clearance free onto its bearing pin. Care must however be taken to keep friction to a minimum. At the spring end of the lever, a slot is provided that permits to vary the spring lever length.

5.4 GOVERNOR FRAME AND BASE

The frame of the governor consists of a angle reinforced plate that accomodates the pilot valve chamber, and the adjustable spring take-up plate. Bores in the chamber, the height of the bearing pin support and its bore and the slots in the bottom of the frame are all machined after welding up the frame to provide for accuracy of alignment and the relative positions of bores and slots. The base consists of another plate that is fitted directly to the turbine frame. By providing longitudinal slots in the base and with lateral slots in the frame, it becomes possible to adjust the relative position of the whole frame assembly in relation to the flyball assembly for adjusting the designed lever ratio of the linkage bar. The relative height and alignment of this lever may be adjusted by three sets of bolts and nuts that connect the frame to the base.

5.5 THE HYDRAULIC CYLINDER ASSEMBLY

We have been using pneumatic cylinders made from stainless steel and with
aluminium endplates. The cylinder is rear trunion mounted to a cylinder
support to allow for the slight up and down movement imposed by the gate
pushrod. For the same reason, a flexible rubber bellows is fixed to the
gate pushrod and the bellows holder on the turbine. This bellows serves
to prevent water leakage from the turbine.

The connection between cylinder pushrod and gate pushrod finally is made
by threading the cylinder pushrod into the internal thread at the end of
the gate pushrod. Closing spring take-up bushes with suitable dimensions are
provided for to hold the spring in place. At the gate pushrod side, a
adjusting nut is provided on the threaded part of the gate pushrod, to make
the adjustment of the tension of the closing spring possible.
On the top of the cylinder end plate, a air bleeding screw should be provi-
ded, so that the air can be bled initially.

6. GOVERNOR TUNING

6.1 STATIC ADJUSTMENTS (in running condition)

6.1.1 Hold lever linkage with flyball-pushrod at postion of full load
(27 mm stroke).

6.1.2 Adjust pilot-valve to designed P_1/P_0 ratio; measurement by pressure
gauges.

6.1.3 Adjust closing spring at full loading of turbine to reach full load speed
(N - 7% ev. 10%),

or a l t e r n a t i v e l y

1.1 hold lever linkage at position of no load (flyball pushrod stroke 3 mm).

1.2 Adjust pilot-valve to pressure ratio $P_1/P_0 = 0,3$.

1.3 Adjust closing spring to get a no load speed of N + 7% (10%).

1.4 Check either no-load or full load speed (N - 7% ev. 10%)
   a) vary closing spring tension and/or pilot valve position slightly to
      optimise speed; re-check 1.1;
   b) vary lever length lx to increase or decrease pilot valve piston stroke
      to increase and decrease speed respectively.

6.2 DYNAMIC ADJUSTMENT

6.2.1 Adjust design lever length $1x'$, re-do 1.3

6.2.2 Adjust spring tension to reach no load speed (at no-load).

6.2.3 Check speed at full load, if incorrect, vary $1x'$.

6.2.4 If system is instable under varying load conditions, increase speed deviation by increasing $1x'$ and decreasing flyball spring tension.

6.2.5 If system is stable at all loading points but is not acting (insensitive) at small load changes, decrease speed deviation by decreasing $1x'$ and increasing pre-tension of spring.

DESIGN PARAMETERS FOR GOVERNOR | Transing : No 35

Gross head (H.total): ___ 30 ___ m, Filter head (H F) _____ 3,5 ___ m

Governor head (H$_G$) : ___ 26,5 ___ m

Turbine speed: ___ 530 ___ RPM

Stroke on regulator shaft : ___ 84 mm

1. Hydraulic cylinder :

Stroke required : Y (mm) : ___ 84 mm _____

ϕ of piston / piston rod (cm): ___ 80 / 20 ___

Effective piston area : A p (cm^2): ___ 47,1 cm^2 ___

Order specification : Bibus 80/150 ___

2. Turbine gate operation:

Pilot valve characteristic

$P_1/P_0 \longrightarrow$	0.25	0.3	0.35	0.4	0.5	0.55	0.6	0.65	0.7	0.75
Effecti. factor of P$_0$	0.20	0.24	0.28	0.32	0,42	0.47	0.52	0.57	0.62	0.68
Pressur available width P$_0$ =2,65 Kg⁴	0,53	0,44	0,74	0,85	1,11	1,25	1,38	1,51	1,64	1,80
Force available with Ap = 47,1 cm^2	25	30,1	34,9	40	52,3	58,9	65	71,1	77,2	84,8

Tentative closing spring selection : $\dfrac{Fo\,7 - Fo\,3}{Y}$ = C'= $\dfrac{77.2 - 30.1}{84}$ = 0,56 choose nearest standard spring

→ Spec CS-C3 : C = 0,457 Kgf/mm ΔF = C·Y = 0,457 · 84 = 38,4 Kgf

Required counter force of spring (to close) in no-load position (P$_1$/P$_0$ 0.3)

Force cyl. pushrod : ___ 30,1 ___

Force inherent in gate (±): ___ − 7 ___

+ Friction : ___ 10 ___

F nl (spring force) total : ___ 33,1 kgf

Required force of cyl. pushrod in full load position (to open)

F nl .... : ___ 33,1 ___

ΔF .... : ___ 38,4 ___

Force inherent in gate (±) : ___ + 3 ___

+ Friction : ___ 10 ___

F fl (force cyl. pushrod : ___ 84,5 kgt

⟹ P$_1$/P$_0$ (at full load) ~ : ___ 0,75 ___

3. Selection of throttle ϕ/pilot valve stroke :

ϕ throttle = _____ 18 _____ mm   Stroke X$_1$ = _____ 12,9 ___ mm

Designer ___ KG ___   Date ___ 19.79 ___

MECHANICAL GOVEROR MEG-1

Form : MEG 1-02

Throttle/pilot valve MEG1/Pv-01

$$P_1/P_0 = \frac{1}{(\frac{A_1'}{A_0})^2 + 1}$$

Throttle Ø ⟶		22	20	18	16	14
X_1 (mm)	A_1' (mm²)	P_1/P_0				
0	58	0,98	0,97	0,95	0,92	0,88
5	113	0,92	0,89	0,84	0,76	0,65
6	127	0,90	0,86	0,80	0,71	0,60
7	140	0,88	0,83	0,77	0,67	0,55
8	154	0,86	0,81	0,73	0,63	0,50
9	167	0,84	0,78	0,70	0,59	0,46
10	181	0,82	0,75	0,66	0,55	0,42
11	194	0,79	0,72	0,63	0,52	0,39
12	208	0,77	0,70	0,60	0,48	0,35
13	223	0,74	0,66	0,57	0,45	0,32
14	239	0,72	0,63	0,53	0,41	0,29
15	254	0,69	0,60	0,50	0,39	0,27
16	273	0,66	0,57	0,46	0,35	0,24
17	293	0,63	0,53	0,43	0,32	0,22
18	312	0,60	0,50	0,40	0,29	0,20
19	338	0,56	0,46	0,36	0,26	0,17
20	363	0,52	0,43	0,33	0,23	0,15
21	389	0,49	0,39	0,30	0,21	0,13
22	428	0,44	0,35	0,26	0,18	0,11
23	467	0,40	0,31	0,23	0,16	0,10
24	506	0,36	0,28	0,20	0,14	0,08
25	592	0,29	0,22	0,16	0,10	0,06
30	1109	0,11	0,07	0,05	0,03	0,02

⟹ SELECTION OF THROTTLE / PILOT VALVE COMBINATION

MECHANICAL GOVERNOR MEG 1

"Thansing"

PILOT VALVE: PV-01

0.9

0.8

0.75

0.7

0.6

0.5

0.4 ⌀22 ⌀20 ⌀18 ⌀16 ⌀14

0.3

0.2

0.1

12,9 mm

5 10 15 20 25 30 Stroke
X_1(mm)
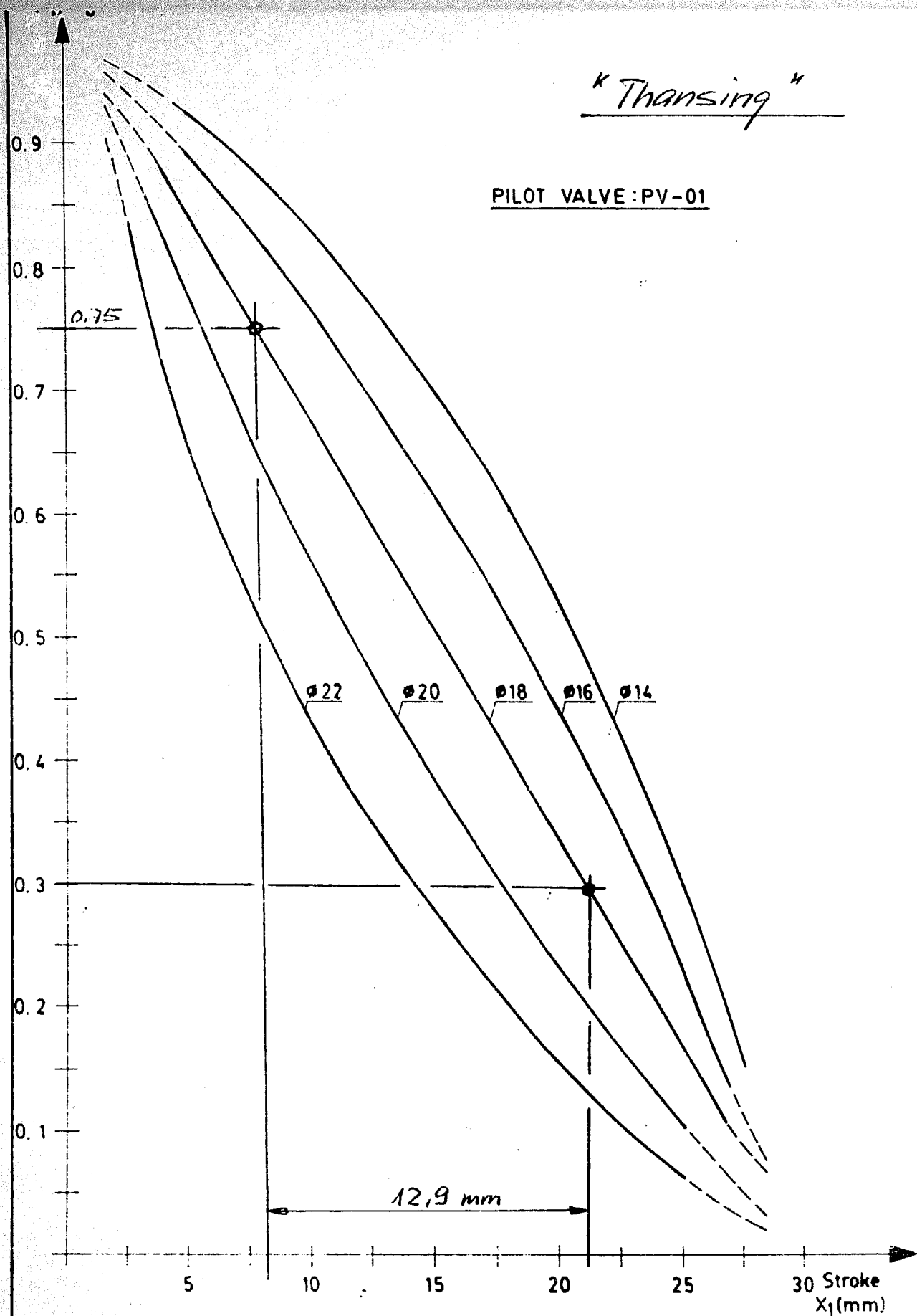
CHARACTERISTIC CURVES FOR DIFFERENT THROTTLE

MECHANICALGOVERNOR MEG 1

Form: MEG 1-01

Annexure II / 3

28

Centrifugal pendulum characteristic "Thansing"

Weight ≡ 0,7 kg   . (MODIFIED PROTOTYPE)

standard stroke 18 mm

F_x (kp)

Turbine speed (RPM)

Stroke X_1 (mm)
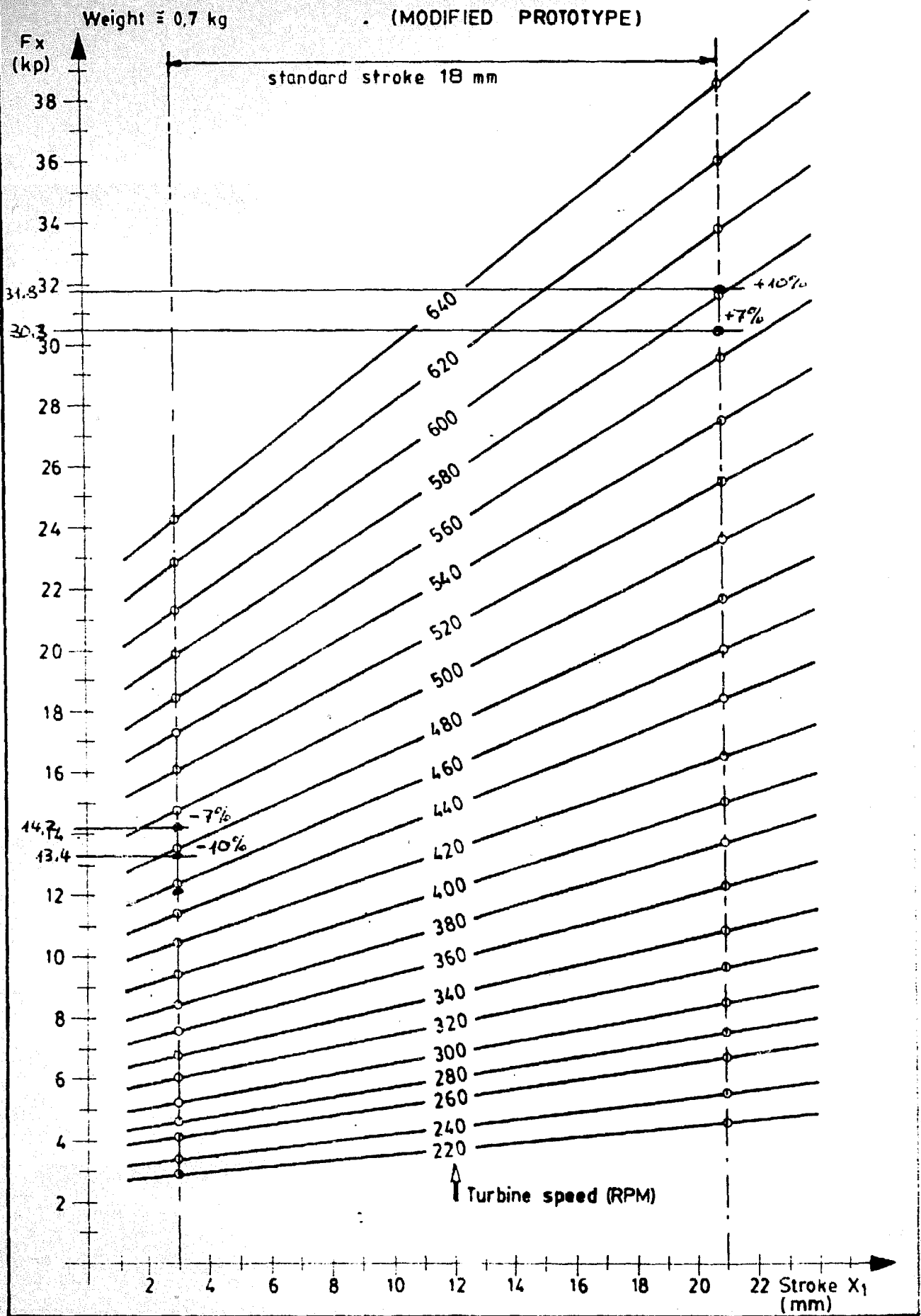
MECHANICAL GOVERNOR MEG 1

Form: MEG1-04

Flyball assembly - spring / lever design

Spring:
Force : F_s
Deflection: x'

Flyball assay:
Force : Fx
Stroke : x =18

Pilot valve:
Force ≅ 0
Stroke x_1

Pivot

$l_x = 95 - 125$ $l_{x_1} = 80$ mm

$l_{x'}$
$(245 - 320)$
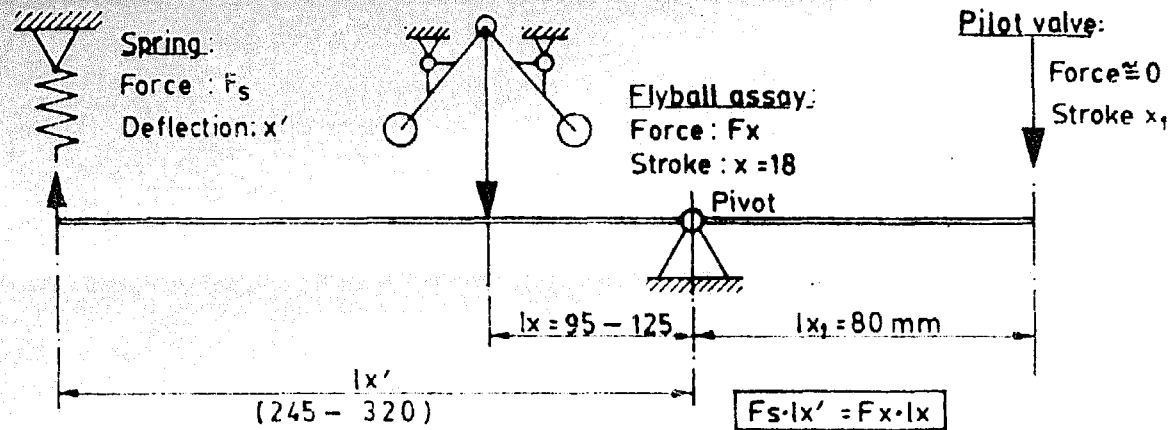
$$F_s \cdot l_{x'} = F_x \cdot l_x$$

<u>DATA</u> : Turbine speed: NT = _530_ RPM

$$x_1 = \_\_ 12.9 \_\_ \text{mm}$$

$$l_x = \frac{x \cdot l_{x_1}}{x_1} = \frac{18 \cdot 80}{x_1} = \frac{18 \cdot 80}{12.9} = \_\_111,6\_\_ \text{mm}$$

4.⇒ <u>Lever length</u> $l_{x'}$: (Fx from diagramm : MEG1 - 04)

$\triangle n \pm 10\%$	$\triangle n \pm 7\%$
⇒ Spring :_ _ _ _ _ C= _0,17_	Spring :_ _ _ _ _ C= _0,17_
n max :_583_ RPM → Fmax: _31,8_ kp	n max: _567_ RPM → Fmax: _30,3_ kp
n min :_477_ RPM → Fmin _13,4_ kp	nmin : _493_ RPM → Fmin: _14,2_ kp
$\triangle F_x = \_\_18,4\_\_$ kp	$\triangle F_x = \_\_16,1\_\_$ kp

$$l_{x'} = \sqrt{\frac{\triangle F_x \cdot l_x \cdot l_{x_1}}{C_0 \cdot x_1}}$$

$l_{x'} = \sqrt{\dfrac{18,4 \cdot 111,6 \cdot 80}{0,17 \cdot 12,9}} = \underline{273,7}\_$ mm	$l_{x'} = \sqrt{\dfrac{16,1 \cdot 111,6 \cdot 80}{0,17 \cdot 12,9}} = \underline{256}\_$ mm

---

⇒ <u>Spring deflection</u> $\triangle x'$:

Max. deflection at no load:

$\triangle x'_2 : \dfrac{Fmax \cdot l_x}{l_{x'} \cdot C} = \dfrac{31,8 \cdot 111,6}{273,7 \cdot 0,17} = \_76,3\_$ mm	$\triangle x'_2 : \dfrac{30,3 \cdot 111,6}{256 \cdot 0,17} = \_\_\_77,7\_$ mm

Pretension at full load:

$\triangle x'_1 : \dfrac{Fmin \cdot l_x}{l_{x'} \cdot C} = \dfrac{13,4 \cdot 111,6}{273,7 \cdot 0,17} = \_32,1\_$ mm	$\triangle x'_1 : \dfrac{14,4 \cdot 111,6}{256 \cdot 0,17} = \_\_\_36,4\_$ mm

---

Designer _____ ℛ _____   Date_ 1.9.79 _____
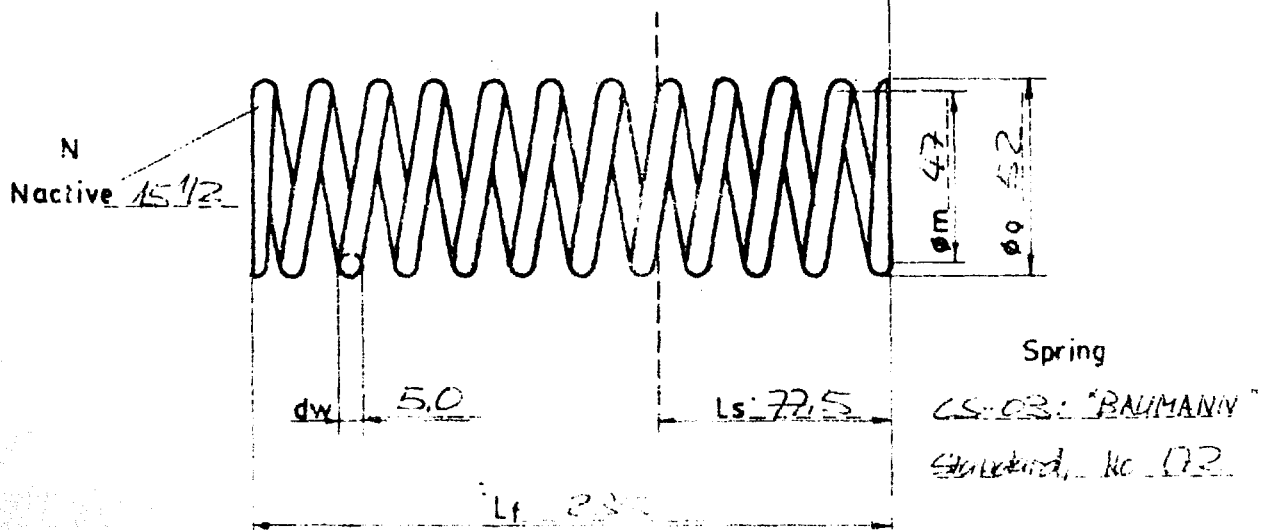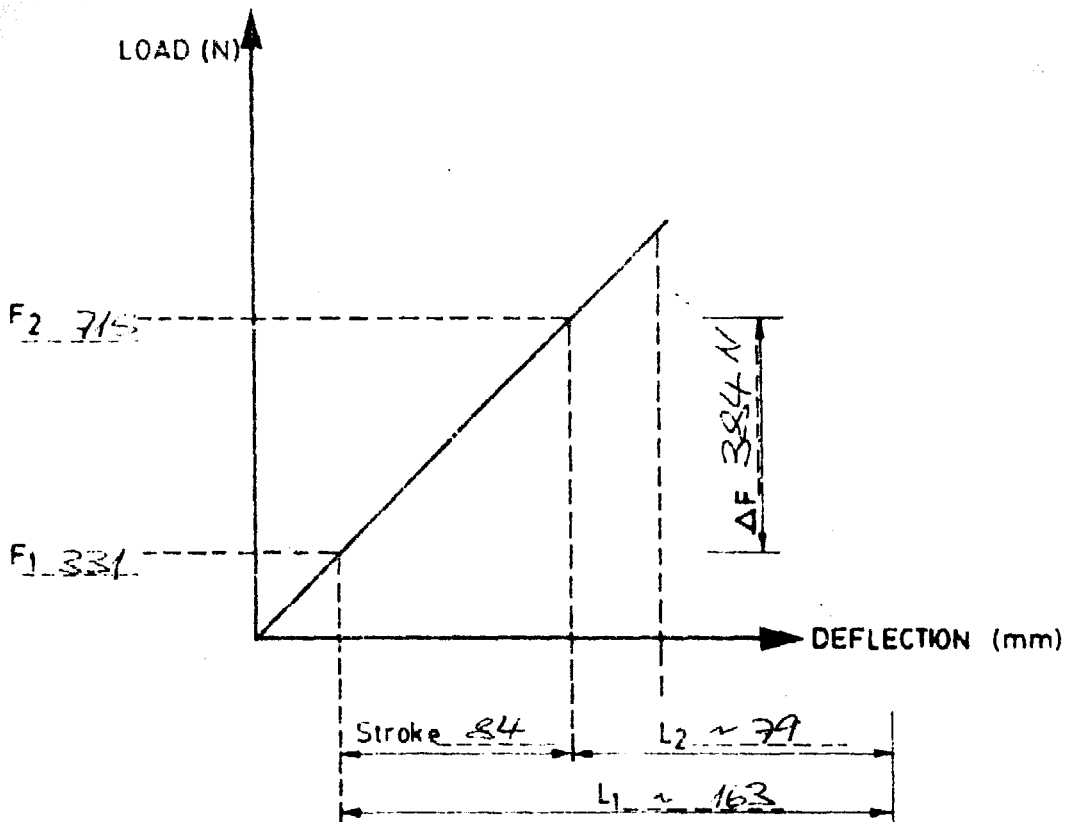
MECHANICAL GOVERNOR   MEG 1

Closing spring design

Name of installation_____ _Thürsing_ _____ ProjNo _ 3̶6̶5̶

Parameters acc. to. form MEG 1-01
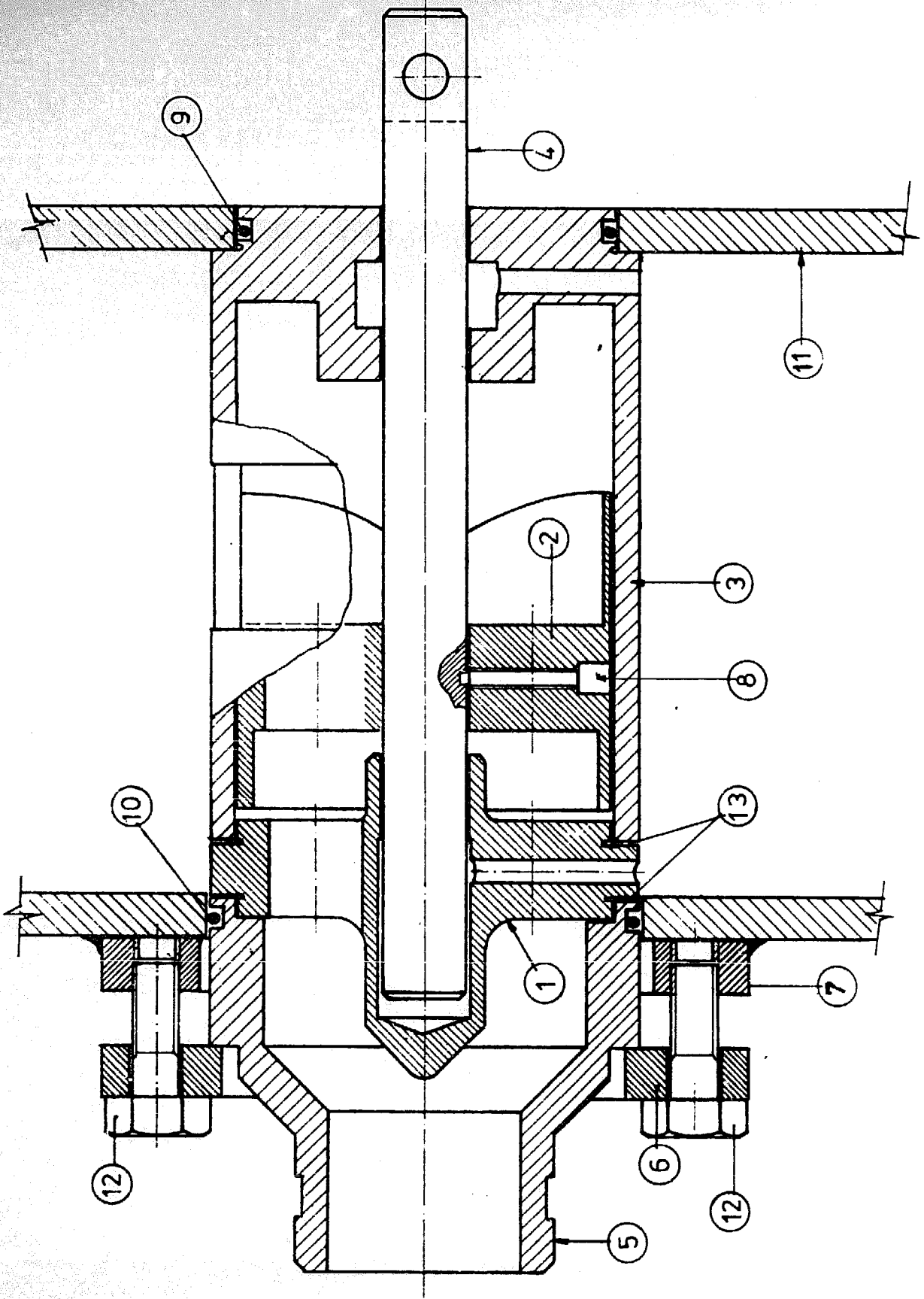
Spring constant : $= \dfrac{\Delta F}{Y}$ _4.57_ N/mm

LOAD (N)

F_2 _715_

F_1 _331_

ΔF 384 N

DEFLECTION (mm)

Stroke _84_

$L_2 \sim 79$

$L_1 \sim 163$

N
Nactive _15 1/2_

⌀m _42_

⌀o _52_

dw _5.0_

Ls _77.5_

Lf _235_

Spring
CS.03: "BAUMANN"
Standard, No _03_

Designer:_____ _UC_

Date: _1.9.79_ _____

MECHANICAL GOVERNOR   MEG 1 / CF 1

Annexure II / 6   3̶l̶

Pos.No	No. of item	Item	Drawing No.	Specification
1	1	End cap	PV1-01	Brass rod $3\frac{1}{2}$"
2	1	Piston 16	PV1-02	Brass rod 3" ⎫
2	1	Piston 18	PV1-03	Brass rod 3" ⎬ ALTERNATIVE
2	1	Piston 20	PV1-04	Brass rod 3" ⎭
3	1	Piston housing	PV1-05	Brass rod $3\frac{1}{2}$"
4	1	Push rod	PV1-06	Brass rod $\frac{3}{4}$"
5	1	Inlet nozzle	PV1-07	Brass rod $3\frac{1}{2}$"
6	1	Flange	PV1-08	M.S Plate 10mm
7	1	Threaded Flange	PV1-08	M.S. Plate 10mm
8	2	Set Screw	- - - - -	M4 X10, DIN 915
9	1	O-ring	- - - - -	64.5 X 3
10	1	O-ring	- - - -	74.5 X 3
11	1	Governor base	- - - -	
12	4	Hex.head bolt	- - - -	W $\frac{3}{8}$ X 1"
13	2	Rubber gasket	PV1-09	0 5 X ϕ 70 X 79

Part list

MECHANICAL GOVERNOR   MEG1/ PV1-00.0

Barant 036.3 26

PILOT VALVE ASSEMBLY   MEG 1/PV1— 00·0
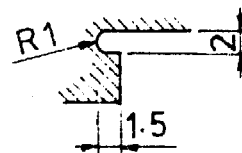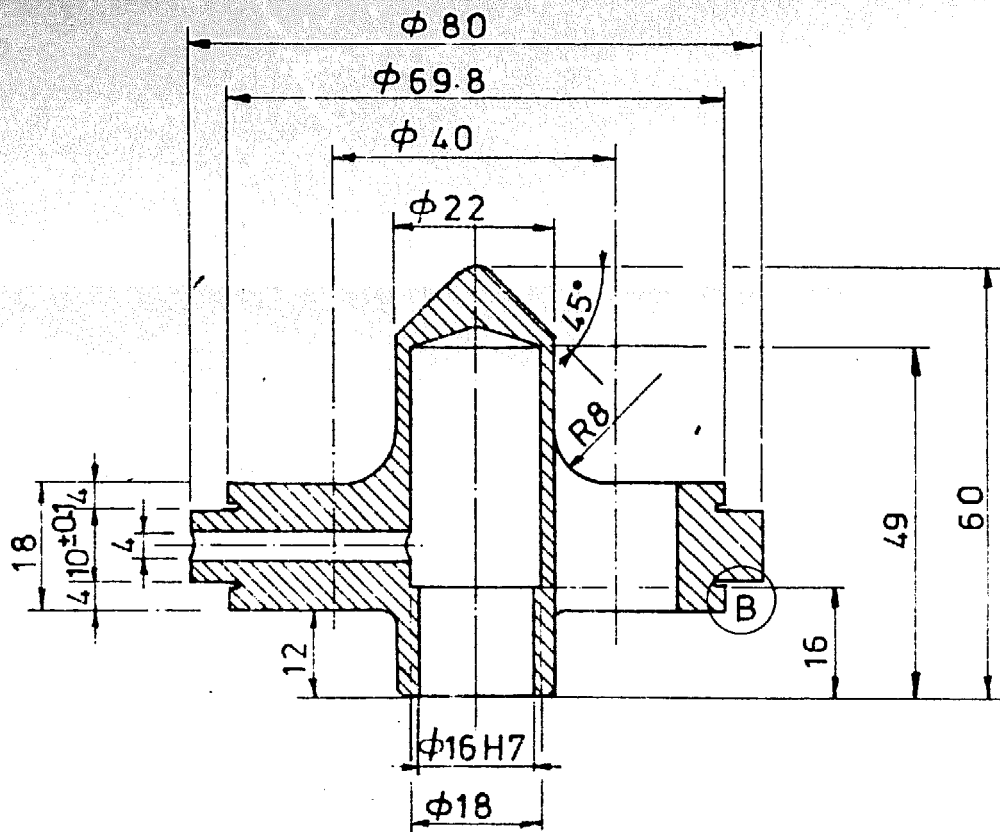
33

Basant 36 3.18

φ 80
φ 69.8
φ 40
φ 22
45°
R8
18
10±0.1
4
12
49
60
16
B
φ16 H7
φ18

R1
2
1.5

DETAIL B
SCALE 2:1

φ17
72°

NO 1
BRASS ROD φ 3½"

SCALE 1:1

ENDCAP

PV1-01

3

ilot Valve

72°

72°

35°

M4

φ6

11
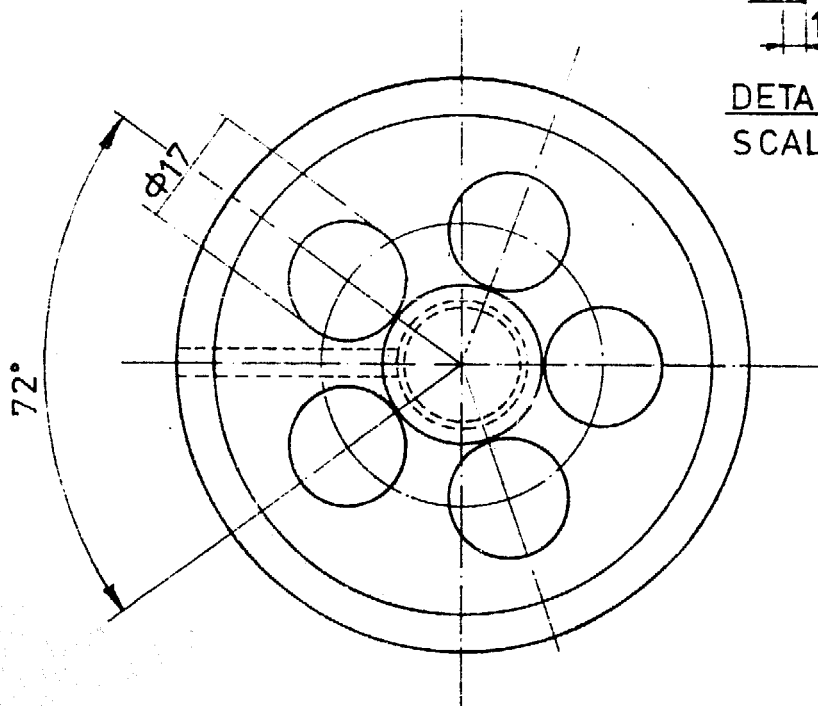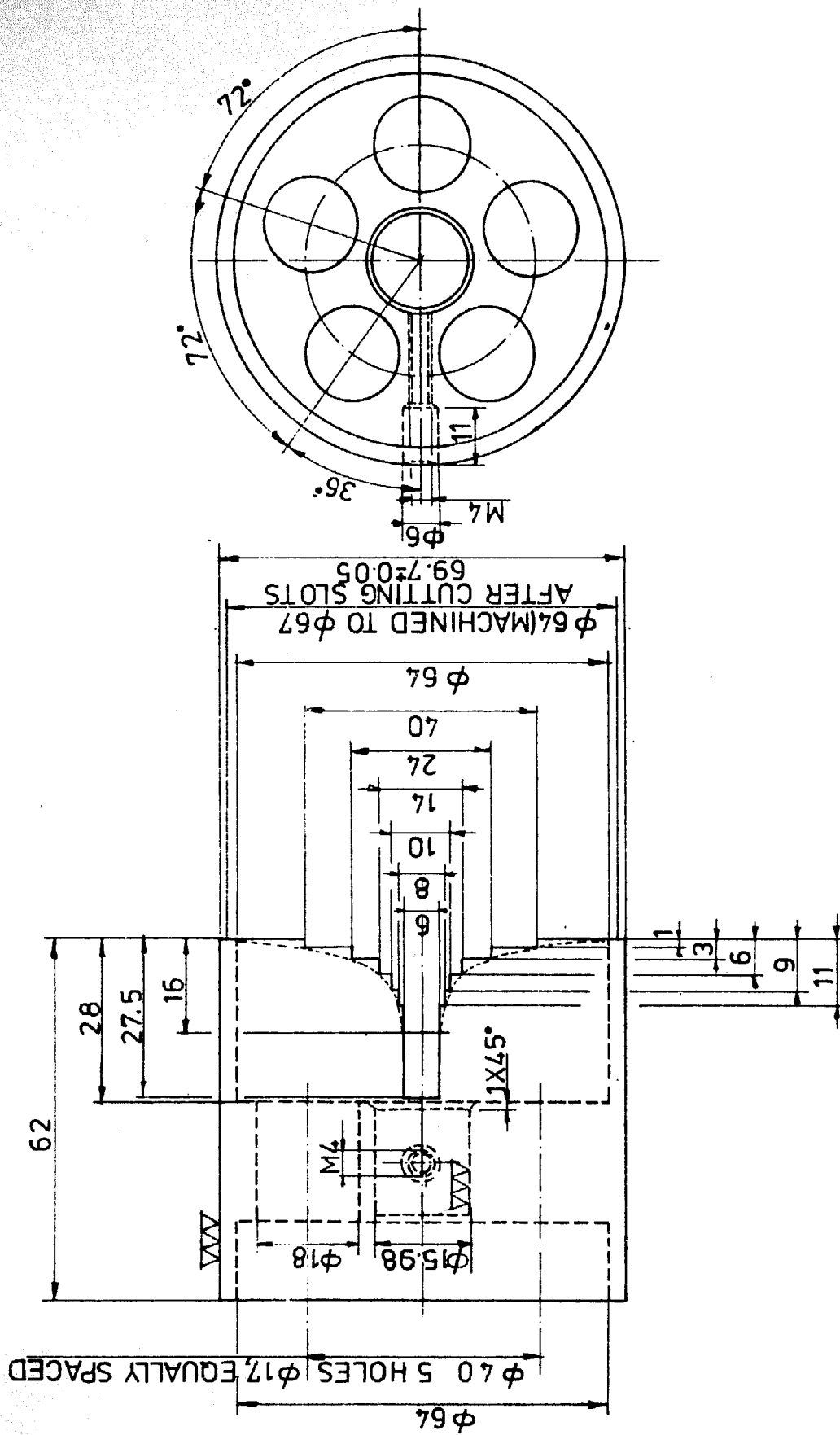
AFTER CUTTING SLOTS
69.7±0.05

φ64(MACHINED TO φ67

φ64

40

72

71

10

8

6

1
3
6
9
11

28
27.5
16

62

1X45°

M4

φ15.98

φ18

φ64

φ40 5 HOLES φ17 EQUALLY SPACED

SCALE 1:1

PISTON 16
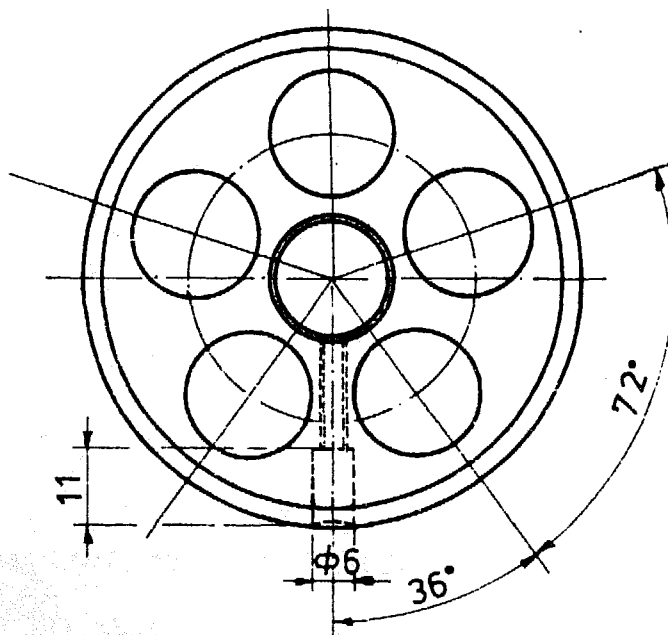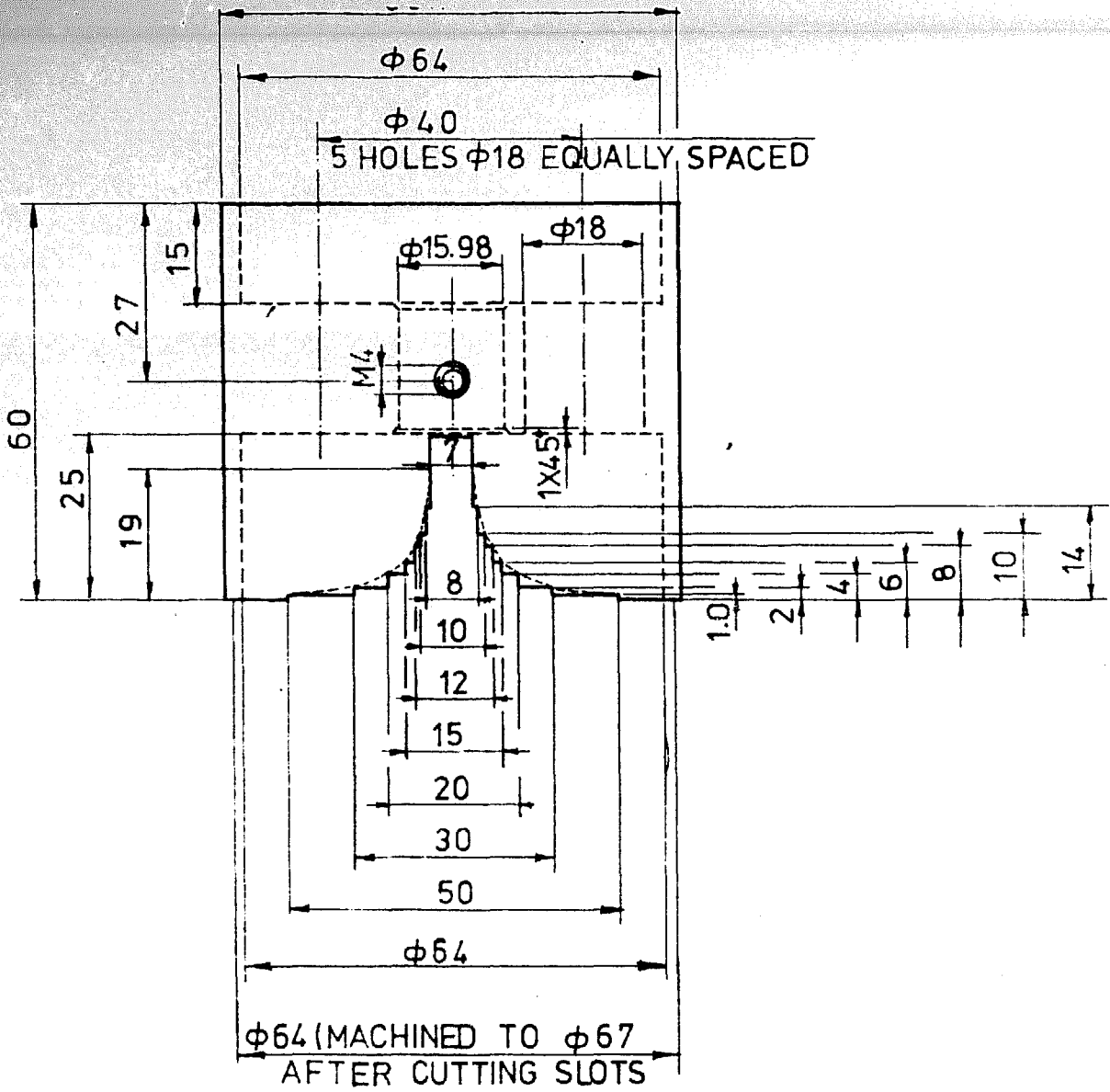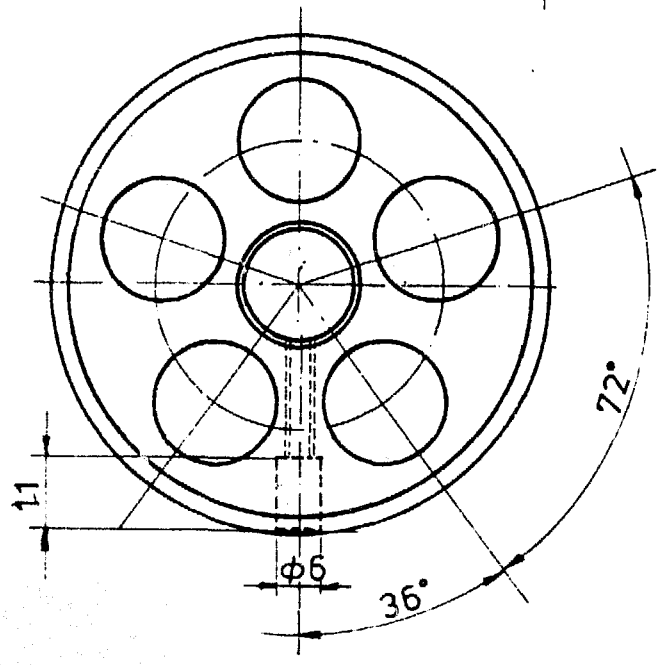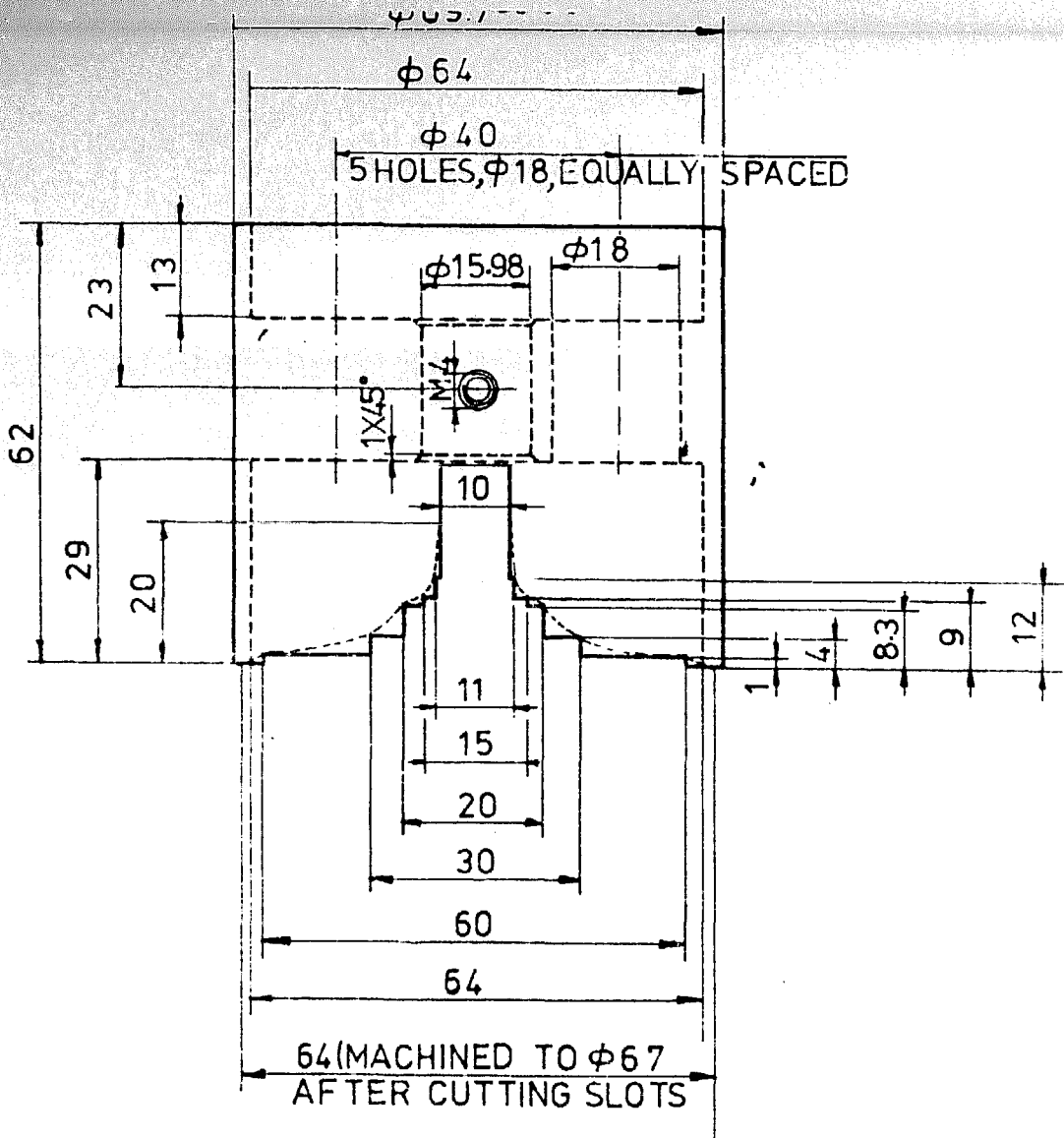
PV1—02

Pilot Valve

35

φ64

φ40
5 HOLES φ18 EQUALLY SPACED

φ15.98   φ18

M4

15
27
60
25
19
7
1×45°
8
10
12
15
20
30
50
φ64

φ64 (MACHINED TO φ67
AFTER CUTTING SLOTS

1.0   2   4   6   8   10   14

11
φ6
36°
72°

φ64

φ64

φ40
5 HOLES, φ18, EQUALLY SPACED

23   13

φ15.98   φ18

62

M4   1X45°

29   20

10

11

15

20

30

60

64

1   4   8·3   9   12

64 (MACHINED TO φ67
AFTER CUTTING SLOTS

72°

11

φ6

36°

DETAIL A
SCALE 4:1

R0.2
R0.5
2.4

φ69.8
φ65
φ26
φ16 H7
φ40
5
5

26

30 ±0.1
40
φ70 ±0.05

101
110
118

φ79.5

A

PISTON HOUSING

Pilot Valve

PV1-05

38

20.5

φ10

0.5X45°

φ16±0.02

60

183

12

POS—4
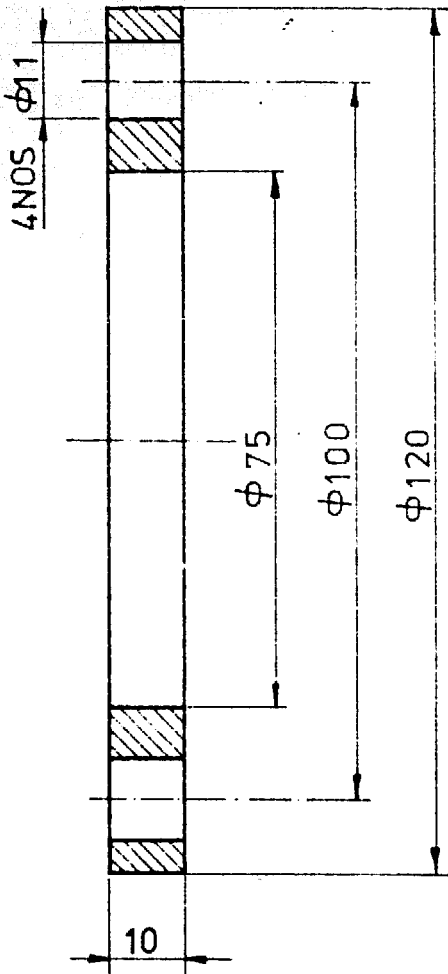
1 NO

BRASS ROD

40
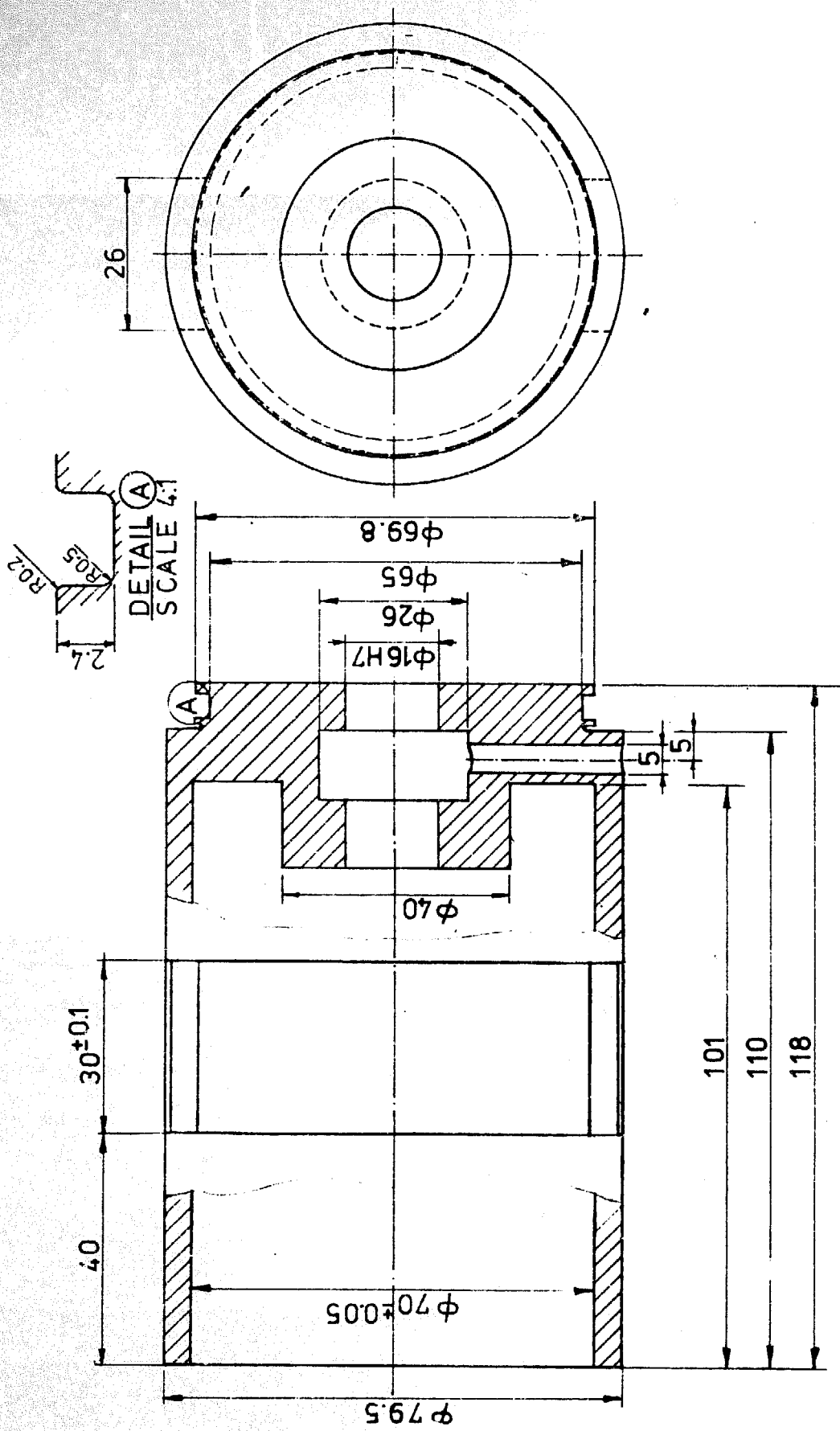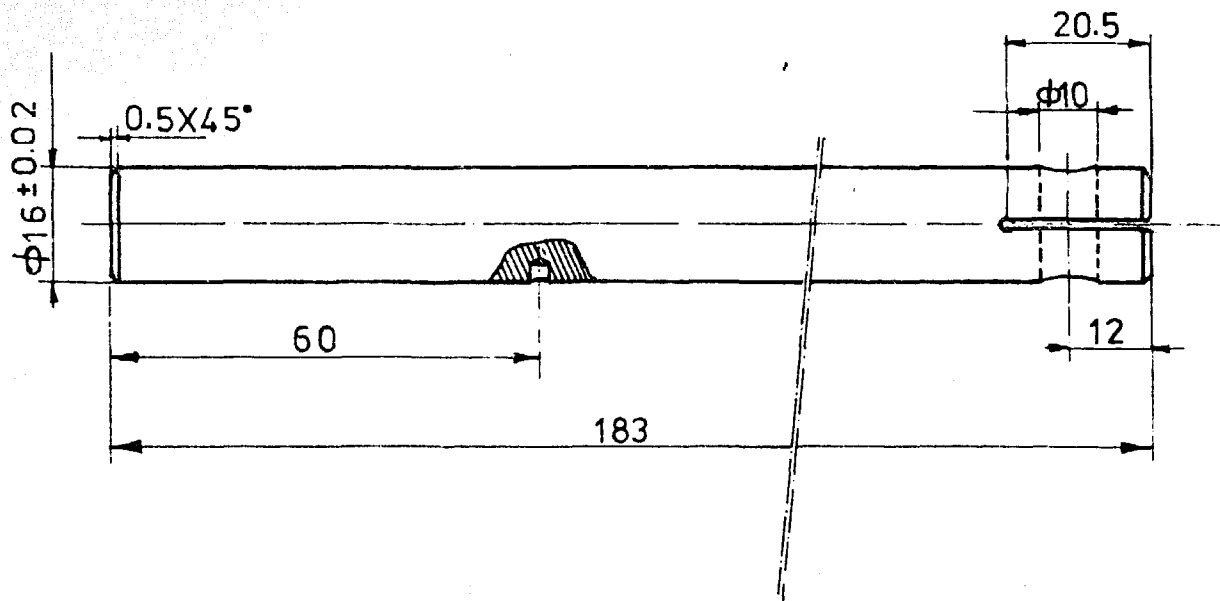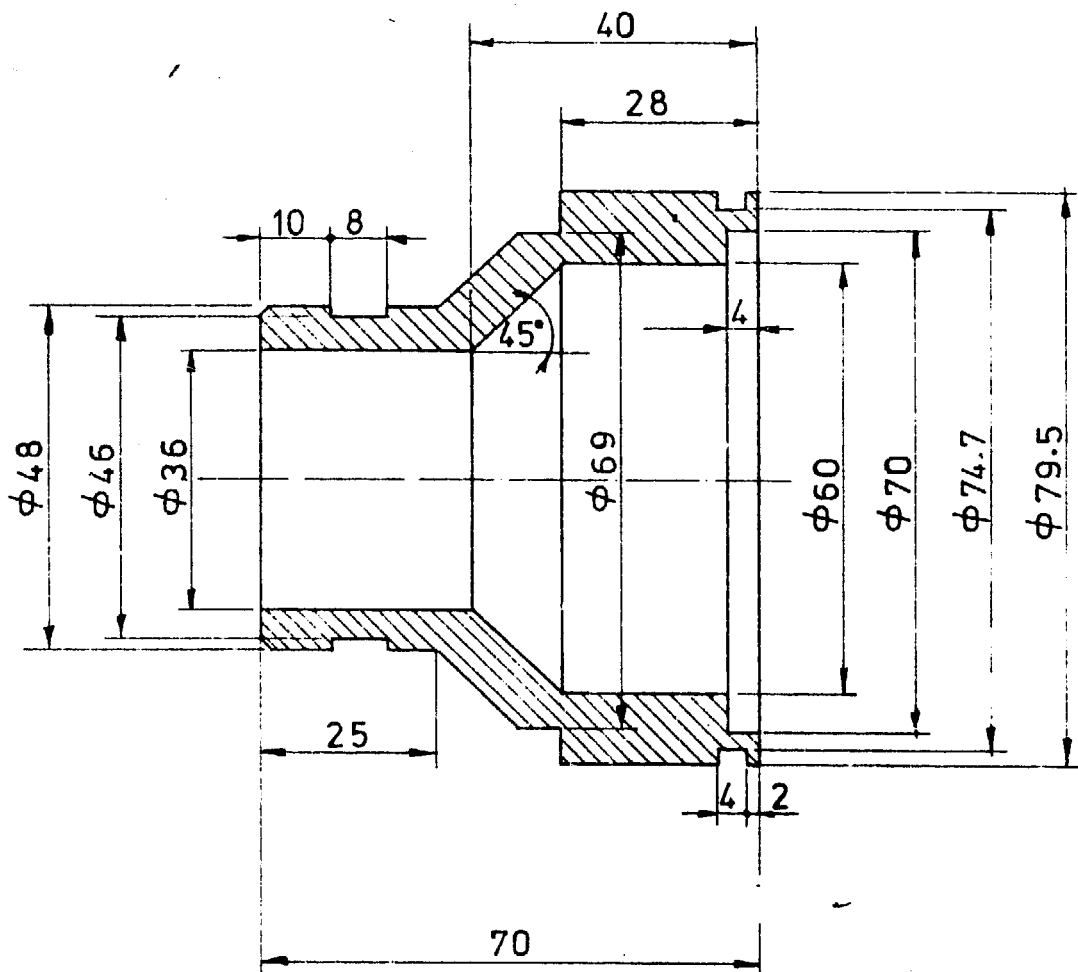
POS-5
1 NO
BRASS ROD $\phi 3\tfrac{1}{2}''$

SCALE 1:1

INLET NOZZLE

PV1- 07

Pilot Valve

POS-6
1 NO
M.S. PLATE
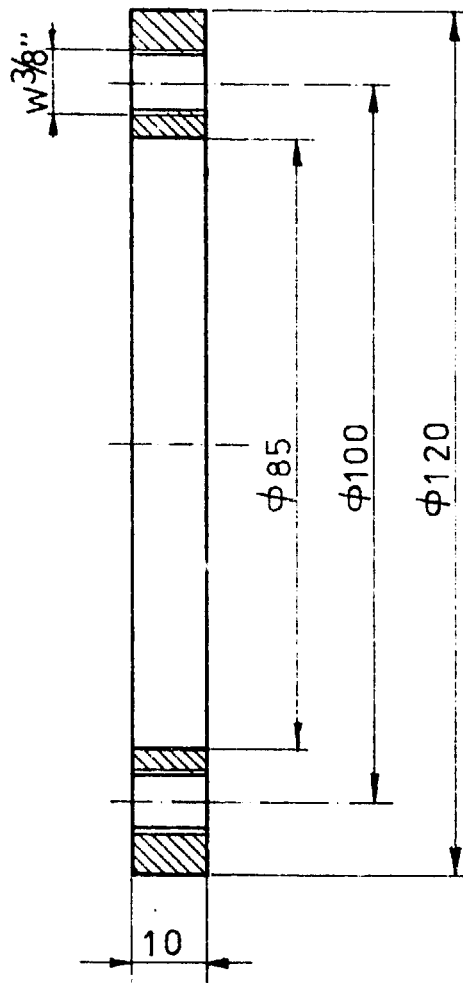
POS-7
1 NO
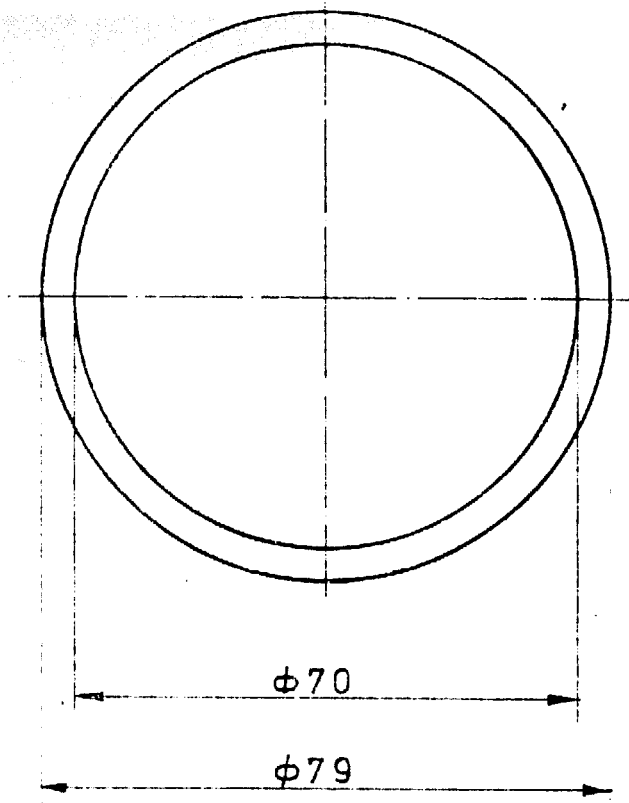M.S. PLATE

SCALE 1:1

FLANGES

PV1-08

Pilot Valve

41

$\phi 70$

$\phi 79$

POS-13

2 NOS

RUBBER 0.5 THICK

RUBBER GASKET

PV1-09

42

Basanta36 3 20