The Current Tendency

of

Hot Strip Mill

OCTOBER 1978

Presented by

MITSUBISHI HEAVY INDUSTRIES, LTD.

TOKYO JAPAN

The Current tendency of the hot strip mill in Japan

1. Introduction

Before the oil crisis happened in 1973, Japanese steel industry went on to make thier production facilities more productive installing high performance equipment in order to meet the demand from steel consumers. Since the oil crisis, however, the situation of the steel industries are changed, because of the recession of the world economy and also thier nature which the steel plant does have to require the huge amount of energy on thier steel production process.

Having the said back ground, the new designs and the improved operation techniques have been under development to meet the requirements from below-mentioned criteria and some of them are already tried in the practical operation.

The new tendency of the hot strip mill in Japan is mainly depend upon the save of energy, material and labor force.

2. The tendency of the layout

The recently ordered hot strip mill to Mitsubishi Heavy Industries, Ltd, have been planned out as 3/4-continuous hot strip mill by the below listed main reasons.

(1) Economization of investment cost for mechanical, civil, and electrical equipment.

(2) Equalization of the production capability between roughing train and finishing train.

(3) The better flexibility of the selection of pass number on roughing train. This will aim very much the rolling of the low alloy steel and high tensile steel.

(4) The better flexibility of the temperature control of the slab on the roughing train.

3. Current techniques

3-1 In order to save energy:-

Among the equipment and facilities for hot strip mill, the slab reheating furnace is the biggest energy consumer and also is acting as the principal equipment for the slab temperature control.

Therefore, the saving the energy on the slab reheating furnaces are most effective ways to achieve the energy saving in the hot strip mill. In accordance with these condition, the following techniques are already developed and applied.

(1) Direct rolling the hot slab from slabbing mill.

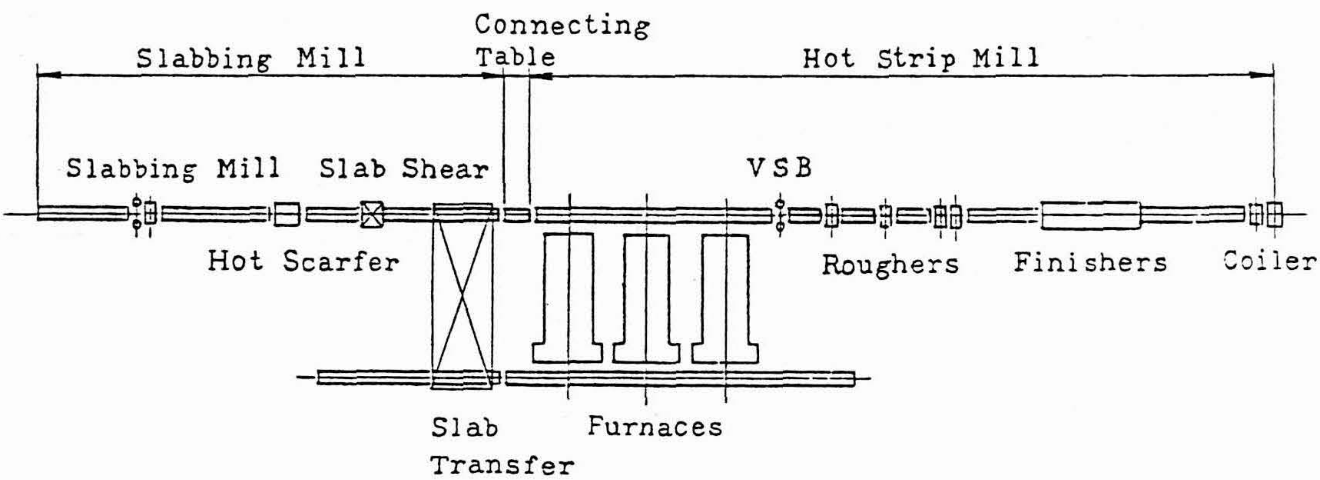
(2) Hot charging the slab to reheating furnaces.

(3) Low temperature extracting the slab from reheating furnace.

(4) Temperature controlled rolling.

3-1-1 Direct rolling the hot slab from slabbing mill.

The several mills, more than four mills are rolling the slab directly from the slabbing mill without reheating in the furnace, in order to achieve this operation, the slabbing mill and the hot strip mill are to be located in tandem as shown below.



The hot scarfing machine located on the rear side of the slabbing mill, will be operated automaticaly for the conditioning of the slab.

The automatic inspection device for the defect on the slab surface is under development at this time.

As a benefit of the direct rolling the hot slab without reheating in the furnace, the energy consumption of the reheating furnace will come down to approximately 80,000 ~ 100,000 Kcal/ton instead of the 360,000 ~ 400,000 Kcal/ton.

3-1-2 Hot charging the slab to reheating furnaces

Another major operation technique to save the energy consumption in the furnaces is Hot charging the slab having temperature arround 500°C ~ 700°C at entry end of the furnaces.

In this case, the slabbing mill or continuous slab casting machine can be installed in any place and the slab transfer between hot strip mill furnace charging end and them are required.

3-1-3 Low temperature extracting the slab from the reehating furnaces

It have been almost common sense to extract the slab from the reheating furnace and feed to the rolling mill at 1,250°C in the temperature.

Now, however, from energy saving and improvement of the quality of the steel, lower extraction temperature became common, which will be arround 1,150°C.

As the advantages of this operation, 10% of the fuel may be saved and the tensile strength of the steel may be increased maintaining the grain size within the range under 8.

For the latter purpose, of course, the temperature control throughout rolling are also essentially required.

3-1-4 Temperature controlled rolling

As stated on the bottom half, the improvement of the tensile strength of the steel can be done, controlling the temperature of the bar in roughing train or finishing train.

3-2 In order to save the material

The most effective mean to save the material is the improvement of yield and control of dimensional accuracy of the product. For this purpose, the following technique are developed and improved.

(1) Automatic gauge control

(2) Compensation of Back up Roll eccentricity

(3) Minimize of Head end mark on the bore of coils

(4) Automatic width control

(5) Minimize of necking of strip

As the future step, the followings are envisaged for the better yield of the hot strip mill

(a) Precision crown control

(b) Minimize of edge drop of the strip

(c) Change over the thickness in one strip

(d) Looper less control on finishing train

3-2-1 Automatic gauge control

The automatic gauge control on the hot strip mill have been done employing the electrical guage control system, MHI have, however, developed recently the hyd-grease type gauge control system which are installed on the top of the top back up roll chock as conpact assembly having better performances such as quick response, steady operation, less maintenance etc, and applied to the 186 inches Japanese plate mill in successfull operation.

Of course, MHI is also manufacturing another type hydraulic A.G.C system consisting of the fluid cylinder, servo valves and inductsyn for the constant roll gap control. The ideal arrangement of hydraulic will have also been studied.

3-2-2 Compensation of the Back up roll eccentricity

The eccentricity of journal and roll barrel of Back up roll come from roll grinding can cause cyclic variation of roll gap due to it's revolution. This will cause the thickness fluctuation of the strip up to today. This cyclic fluctuation of roll gap could not be compensated by Automatic gauge control.

Mitsubishi Heavy Industries, Ltd. have recently developed the compensation control system to eliminate the influence of the back-up roll eccentricity utilizing the load signal from load cells the position signal of the roll rotation and computor system.

The trial achievement of this control had taken the place in Japanese cold mill in last year, after several trial achievement, it has been proven that the system is quite stable and effective to eliminate the influence of Back up roll eccentricity from strip thickness.

3-2-3 Minimize of Head end mark on the bore of coils

When the heavy gauge strip is coiled up on the downcoiler mandrel, first few wraps have to suffer from the printing or deformation due to the step of the strip head end.

In order to release this problem, **Mitsubishi Heavy Industries, Ltd.** had applied 3-step expanding mandrel for downcoiler, on the past design. However, recently head end mark on the cooled strip became more severe due to increasing the needs to roll hard thicker material for line pipes.

As a solution for this, recently MHI have developed new design which is providing the pneumatic spring between unit roller bearing and it's frame having spring force variable control to establish adequate spring force according to the strip thickness in order to minimize the contact force of unit roller to coiling strip.

3-2-4 Automatic width control

According to the installation of the continuous slab casting machine, The width step variation of the slabs are grouped into several pattern to minimize the changing of the casting moulds. Therefore on the hot strip mill, the width of the slabs have to be adjusted to the specified strip width.

And also heavy thick slabs are introduced to bring the mill productivity higher, therefore on the edging mill, edging force have to be adjusted to maintain slab width constant, because of the fluctuation of thermal distribution.

From above demand, MHI have developed the heavy duty edging mill having below specification for heavy continuous cast slab rolling.

3-2-5 Minimizing of necking of strip

When strip tension in between last mill stand and mandrel are fluctuated by some reason such as different acceleration or deceleration performance of each drive, the width deviation will be happened, which are called as the necking of the strip.

From actual operation, it can be said that the necking of the strip will be happend frequently on the strip having the sectional area less than 2,000 mm^2.

Therefore, in order to avoid this necking, Down coiler mandrel drive will be equipped with change-over type gear drive to select best possible inertia for the tension control of the thin gauge material.

Reference data

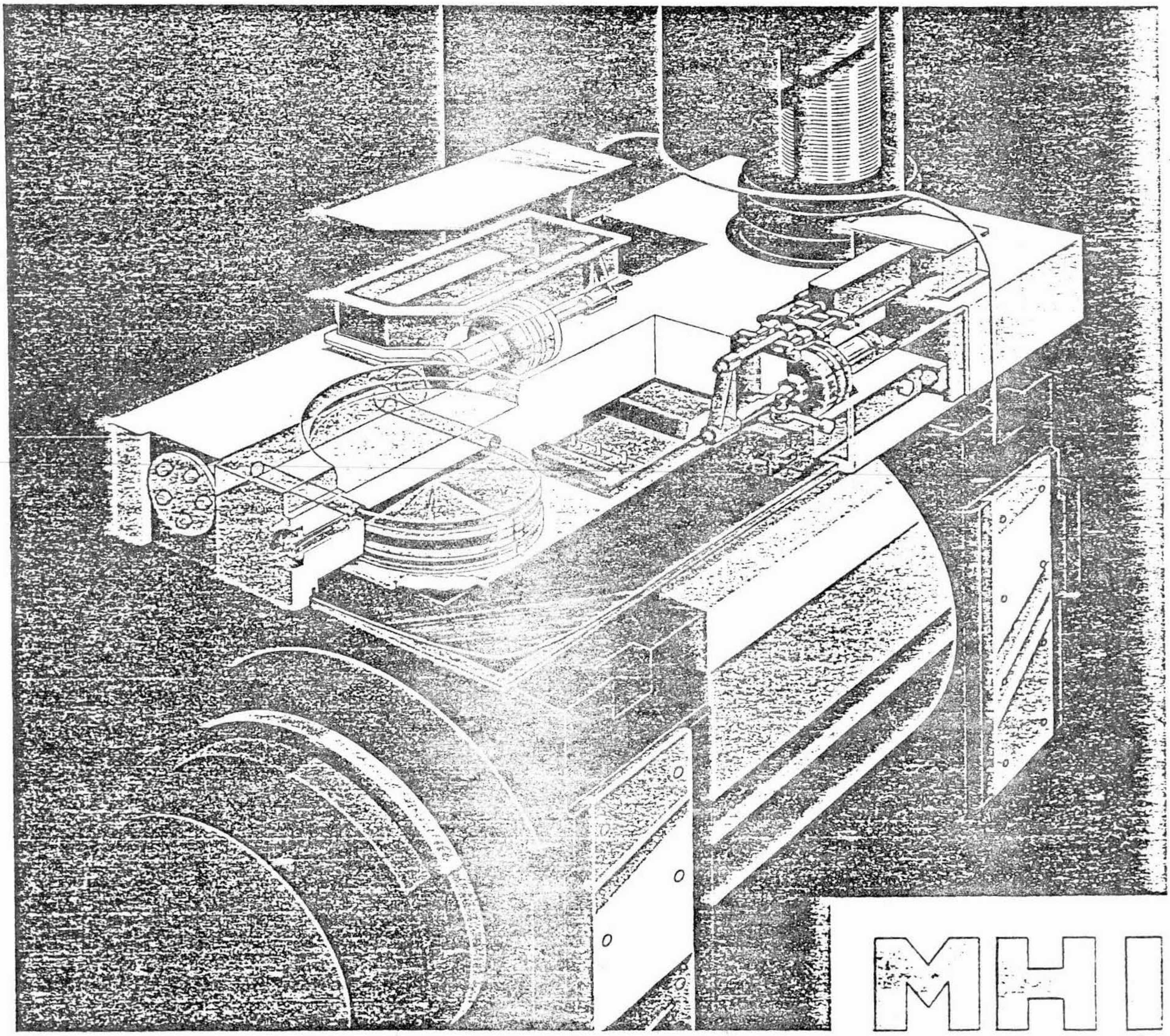
Heavy Edging Mill

Slabs to be rolled	Max. 350 mm thick
Max. draft	100 mm
Edging speed	70 m/min
Roll adjust mechanism	Hyd.-Mechanical
Roll adjusting speed	Max. 80 mm/sec
RAWC	Screw-nut sliding by Hydraulic
Normal roll adjust	by Mechanical
Adjusting force	Max. 500 tons

SERMES AGC SYSTEM

SERMES AGC system developed by Maschineufabrik SACK GmbH in West Germany, has been successfully put into practical use in rolling mills and at present 12 units are being operated satisfactorily and 2 units are under manufacture.

In order to meet new requirements for rolling mills, our Mitsubishi Heavy Industries, Ltd. (MHI) is now ready to start the manufacture of the SERMES AGC system under a license from SACK, in addition to the hydraulic AGC system which has been developed and been manufactured up to the present by our company.



1. Features

The SERMES AGC system mainly consists of main cylinders and intensifier cylinders. The main cylinder is controlled by the intensifier cylinder using grease as a medium.

As the main cylinder is of a grease operation type, it has a high capability of sealing so as to enable it to make a high pressure operation, and therefore when the system has been applied to respective kinds of rolling mill, it can make the best use of the following features.

(1) High responsiveness

Because this system has a good responsiveness of about 2 times as high as that of the electrically operated AGC system, it can be expected that improvement is made in the strip thickness accuracy.

(2) AGC rolling under a high load

As the grease operating pressure in the main cylinder is allowed to be raised up to max. 450 kg/cm^2, the AGC rolling can be performed under a high load.

(3) Variable mill modulus control

Rolling can be carried out at the optimum mill modulus selected according to the characteristics of each stand.

(4) Unjamming function

Jamming of the mill rolls can be easily released by the pull back of the main cylinders.

(5) Adaptability to existing rolling mills

As the main cylinder is installed at the upper part of the mill housing, it can be additionally installed easily without changing the pass-line, that is, by only some adjustment of the roll gap.

(6) Decrease of investment cost

As a flat cylinder is adopted for the main cylinder, this system can make the height of the mill housing lower as compared with other type hydraulic AGC systems and moreover, it does not require to provide dummy blocks for fixing securely the location of the pass line. Therefore this system can serve also for the saving of investment cost.

(7) Maintenace

As the main cylinder is installed at the upper part of the mill housing, its enviromental conditions are relatively good as compared with other type hydraulic AGC systems and also its removal can be made easily, which facilitates the maintenance work.

2. Construction

The main cylinders and the intensifier cylinders are assembled in the cross frame which is installed under the screw-down screws and is ranging from the work side to the drive side of the rolling mill. The cross frame is also provided, by the manifold type fitting method, with the piping and equipment for pilot cylinder operation.

Grease is filled between the main cylinder and the intensifier cylinder, and by using the grease as a medium, the position of the main cylinder is controlled by hydraulic control of the piston position of the intensifier cylinder.

As the pressure intensifying ratio of the intensifier cylinder is selected to about 2 times, the working oil pressure for the intensifier cylinder becomes 250 kg/cm^2 at the highest for the 450 kg/cm^2 maximum working grease pressure of the main cylinder, and therefore this SERMES AGC system has a universality in selecting the hydraulic source equipment and the electro-hydraulic servo-valve.

In designing the SERMES unit, importance must be attached particularly to the resistance against the vibrations originating from the rolling load, and in this respect, considerations are given as shown below.

(1) Especially, as for the wear of the sliding surface between the main cylinder and the plunger, the bush construction is adopted so that maintenance (replacement) can be carried out easily. Similar considerations are given also to the intensifier cylinder, etc.

(2) Intensifier cylinder and position detecting device

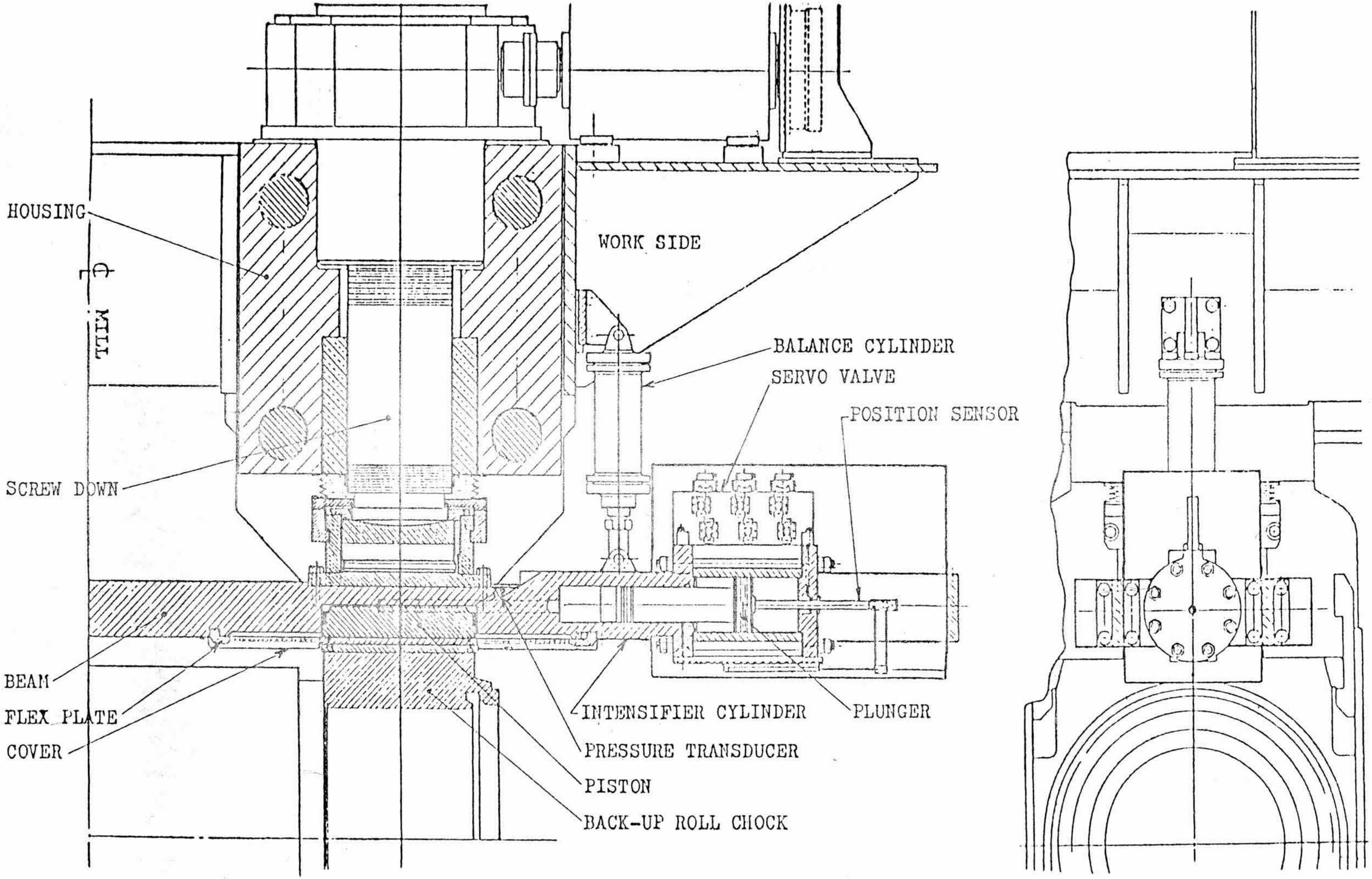
Especially, in the case of a rolling mill which is subjected to large impacts, a vertical type arrangement is adopted for the intensifier cylinder and the position detecting device so that unexpected abnormal loads will not act on the cylinder rod and the detecting rod.

Further, the detecting rod is provided with a guide damper to slacken the influence by the impact load.

(3) The vibration resistance becomes a problem also for the piping parts.

In this respect, however, a manifold block construction is adopted to decrease the number of joints as far as possible.

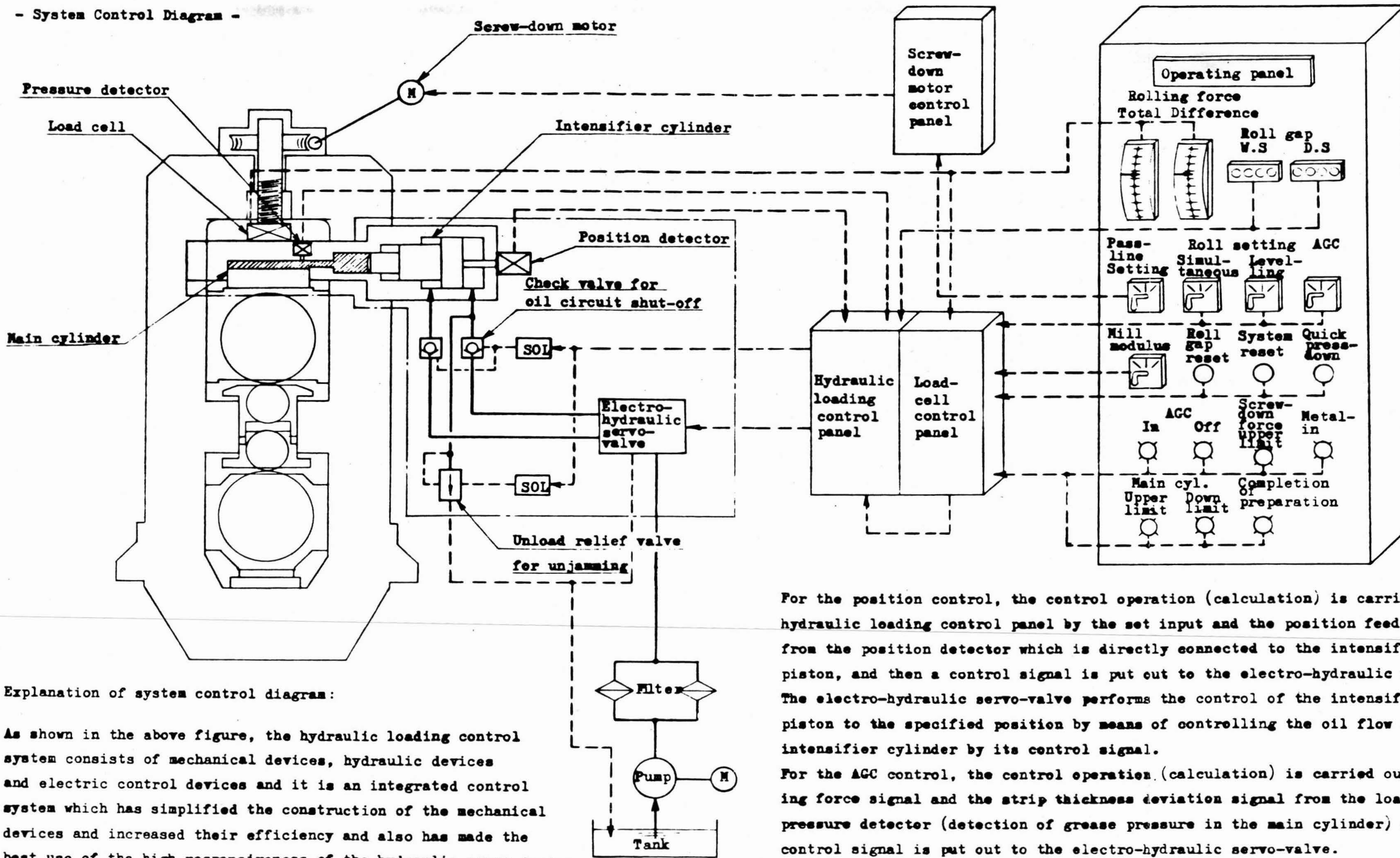
As for the detailed construction of the SERMES unit, please refer to the following page.



SERMES ELECTRO-HYDRAULIC ROLL POSITIONING SYSTEM

SERMES AGC SYSTEM

- System Control Diagram -



Explanation of system control diagram:

As shown in the above figure, the hydraulic loading control system consists of mechanical devices, hydraulic devices and electric control devices and it is an integrated control system which has simplified the construction of the mechanical devices and increased their efficiency and also has made the best use of the high responsiveness of the hydraulic servo device.

In the SERMES AGC system, grease is filled between the main cylinder and the intensifier cylinder and all the positioning of the main cylinder is carried out by setting the position of the intensifier cylinder through grease transmission.

For the position control, the control operation (calculation) is carried out in the hydraulic loading control panel by the set input and the position feed back signal from the position detector which is directly connected to the intensifier cylinder piston, and then a control signal is put out to the electro-hydraulic servo-valve. The electro-hydraulic servo-valve performs the control of the intensifier cylinder piston to the specified position by means of controlling the oil flow rate to the intensifier cylinder by its control signal.

For the AGC control, the control operation (calculation) is carried out by the rolling force signal and the strip thickness deviation signal from the load cell or the pressure detector (detection of grease pressure in the main cylinder) and then a control signal is put out to the electro-hydraulic servo-valve.

Beside the above, the SERMES AGC system has the following main control functions.

- Variable mill modulus AGC control
- Roll setting
- Unjamming operation
- Oil circuit shut-off control
- Monitor AGC control

Production records of SERMES AGC system

No.	Baujahr Year	Anlage Mill Type	Gerüstabmessungen Mill Size [mm]	Geschwindigkeit Mill Speed [m/min]	Nominale Walzlast Rolling Load [KN]	Kenngröße SERMES-Size	Hersteller Builder	Auftraggeber und Ort Customer and Location
1	1971	Kaltband-Reversiergerüst	465 Ø x 1420 Ø x 1650	600	18 000	565 x 12	CLE / SACK	Rasselstein AG / Werk Neuwied / BRD
2	1972	5gerüstige Kalt-Tandemstraße	585 Ø x 1350 Ø x 1260	1800	24 000	565 x 10	CLE	J. J. Carnaud / Basse-Indre / Frankreich
3	1972	6gerüstige Alu-Kaltbandstraße	585 Ø x 1410 Ø x 1525	2300	20 000	610 x 9,5	CLE / MESTA	Alcoa / Warrick Works / Indiana / USA
4	1973	2gerüstige DCR-Straße	600 Ø x 1420 Ø x 1320	2130	16 000	565 x 12	SACK	Hoesch Hüttenwerke AG / Dortmund / BRD
5	1973	Kaltband-Reversiergerüst	430 Ø x 1270 Ø x 1220	680	18 000	650 x 12,5	SACK	TERNI SpA / Terni / Italien
6	1973	Kaltband-Reversiergerüst	535 Ø x 1525 Ø x 1830	1000	18 000	565 x 12,5	SACK	Usine de Gustav Boel SA, La Louviere, Belgien
7	1974	5gerüstige Kalt-Tandemstraße (Gerüst 1)	460 Ø x 1250 Ø x 1070	1370	18 000	520 x 13	MESTA	Altos Hornos de Mexico, Monclovia, Mexico
8	1974	7gerüstige Warmbandfertigstraße (Gerüst 5+6+7)	675 Ø x 1300 Ø x 1700	720	20 000	650 x 20	SACK	August Thyssen-Hütte AG, Werk Bruckhausen / BRD
9	1974	4gerüstige Kalt-Tandemstraße (Gerüst 1)	580 Ø x 1420 Ø x 2030	1200	20 000	650 x 10	SACK	Stahlwerke Peine-Salzgitter AG, Werk Salzgitter / BRD
10	1975	Kaltband-Reversiergerüst	420 Ø x 1500 Ø x 2250	105	25 000	650 x 15	SACK	Avesta Jernverks AB, Avesta, Schweden
11	1975	Blechstraße	995 Ø x 2030 Ø x 2800	300	94 000	1016 x 25	MESTA	Bethlehem Steel Corp., Burus Harbor Plant, USA
12	1975	Kaltband-Reversiergerüst	150 Ø x 380 Ø x 250	16,7/50	2 000	177,8 x 8,75	CLE	Irsid-Paris, Frankreich
13	1976	5gerüstige Kalt-Tandemstraße	585 Ø x 1350 Ø x 1220	2080	12 000	565 x 12,5	SACK	CORNIGLIANO, Genua, Italien
14	1976	Blechstraße	950Øx2100Øx3700	360	77 000	1000x20	SACK	August Thyssen-Hütte AG, Werk Huckingen/BRD

Comparison table of various kinds of AGC systems (1/2)

	SERMES AGC system	Hydraulic AGC system	Electric AGC system
(1) Installation position	Usually the cylinder is installed in the window upper part of the mill housing.	Usually the cylinder is installed in the window lower part of the mill housing.	Usually this system is installed on the top of the mill housing.
(2) Main operating members	Grease operated main cylinder and hydraulically operated intensifier cylinder	Hydraulically operated cylinder	D.C. motor and screw-down screw
(3) Strength of electrically operated screw-down device	As the electrically operated screw-down device does not work at the time of AGC rolling, high load does not work on the screw-down worm gear set.		As the electrically operated screw-down device performs the AGC rolling, a high load design is required for this system.
(4) AGC rolling load	Because the max. working pressure of the main cylinder is 450 kg/cm^2, a high load rolling can be carried out.	Because the max. working pressure of the cylinder is 350 kg/cm^2, this system is inferior to the SERMES AGC system.	As the rolling load is limitted by the strength of the driving mechanism such as worm & wheel, screw & nut, etc., this system is inferior to the others in the high load rolling.
(5) AGC operation responsiveness	As the main cylinder is controlled by the intensifier cylinder, using grease as the medium, this system is inferior to the hydraulic AGC system in the point of responsiveness.	As the cylinder is controlled directly, this system is most excellent in the responsiveness.	The motor itself has a large inertia, and therefore there is a limitation in the responsiveness.
At position control	30 rad/sec	30 rad/sec	7.5 rad/sec
At BISRA control	10 rad/sec	30 rad/sec	4 rad/sec

Comparson table of various kinds of AGC systems (2/2)

<table>
<tr><th></th><th>SERMES AGC system</th><th>Hydraulic AGC system</th><th>Electric AGC system</th></tr>
<tr><td>(6) Setting accuracy of roll gap</td><td colspan="2">Setting accuracy can be raised by making fine adjustment with the cylinder after setting of roll gap by electrically operated screw-down device.</td><td>Setting accuracy is decided depending on the degree of accuracy of setting by the electrically operated screw-down device.</td></tr>
<tr><td>(7) Taper rolling</td><td colspan="2">In the case of plate rolling, taper rolling can be carried out within the stroke range of the cylinder.</td><td>Taper rolling is restricted by the shortage of capacity of the AGC screw-down motor.</td></tr>
<tr><td>(8) Unjamming function</td><td colspan="2">By the pull-back of the cylinder, unjamming of rolls can be carried out easily.</td><td>In the AGC screw-down motor, roll unjamming capacity can not be expected.</td></tr>
<tr><td rowspan="2">(9) Maintenance items</td><td>Maintenance for the wear parts in the SERMES unit.
o Bushes and packings for the each cylinder
o Position detectors
o Piping parts</td><td>Maintenance of the relative parts of the cylinder.
o Bushes and packings for the cylinders
o Position detectors
o Piping parts</td><td rowspan="2">Periodical inspection for the wear of mechanical parts.
o Screw-down worm gear sets
o Screw-down screws and nuts
o Bearings
o Clutches, etc.</td></tr>
<tr><td colspan="2">Maintenance of hydraulic working device
o Control of working oil
o Maintenance of the servo- valve and filter
o Maintenance of the hydraulic source equipment</td></tr>
<tr><td>(10) Large spare parts</td><td>o Bushes and packings for the SERMES unit
o Position detectors
o Servo-valve
o Filter element
o Others</td><td>o Cylinder assembly
o Position detectors
o Servo-valve
o Filter element
o Others</td><td>o Screw-down worm gaer sets
o Screw-down screws and nuts
o Bearings
o Others</td></tr>
</table>

ROLLING ECCENTRICITY CONTROL

OF

STRIP MILL

JULY, 1977

MITSUBISHI HEAVY INDUSTRIES, LTD.
Hiroshima Shipyard & Engine Works

1. Preface

Recently, demands for thickness accuracy of rolled products have become increasingly strict and severe from the standpoint of improvement of yield, and the hydraulic push up system developed as a countermeasure for the above and excellent in controllability, is exhibiting its noticeable effect.

In this respect, if the spring constant of mill (mill modulus) is raised by control, it is effective for preventing product thickness variations due to incoming strip sheet, however, it has a reverse effect on variations of product thickness due to roll eccentricity of mill itself. Therefore a control system for removing the influence of roll eccentrcitiy has now become one of important subjects of development in the field of steel industry.

For this reason, our company has developed a roll eccentricity compensation control system as explained hereunder on the basis of our measurements, researches and experience of actual test and operation concerning the roll eccentricity in the past.

Our roll eccentricity compensation system has two modes. In mode 1, the roll eccentricity is detected before start of rolling on the rolls being rotated in contact condition and this data is used for roll eccentricity control during rolling; off-line detection of roll eccentricty. In mode 2, the roll eccentricity control is made through detection of roll eccentricity during rolling; on-line detection of roll eccentricity. A combined use of the two modes will permit effective compensation control of the roll eccentricity.

Fig. 1 shows system control diagram.

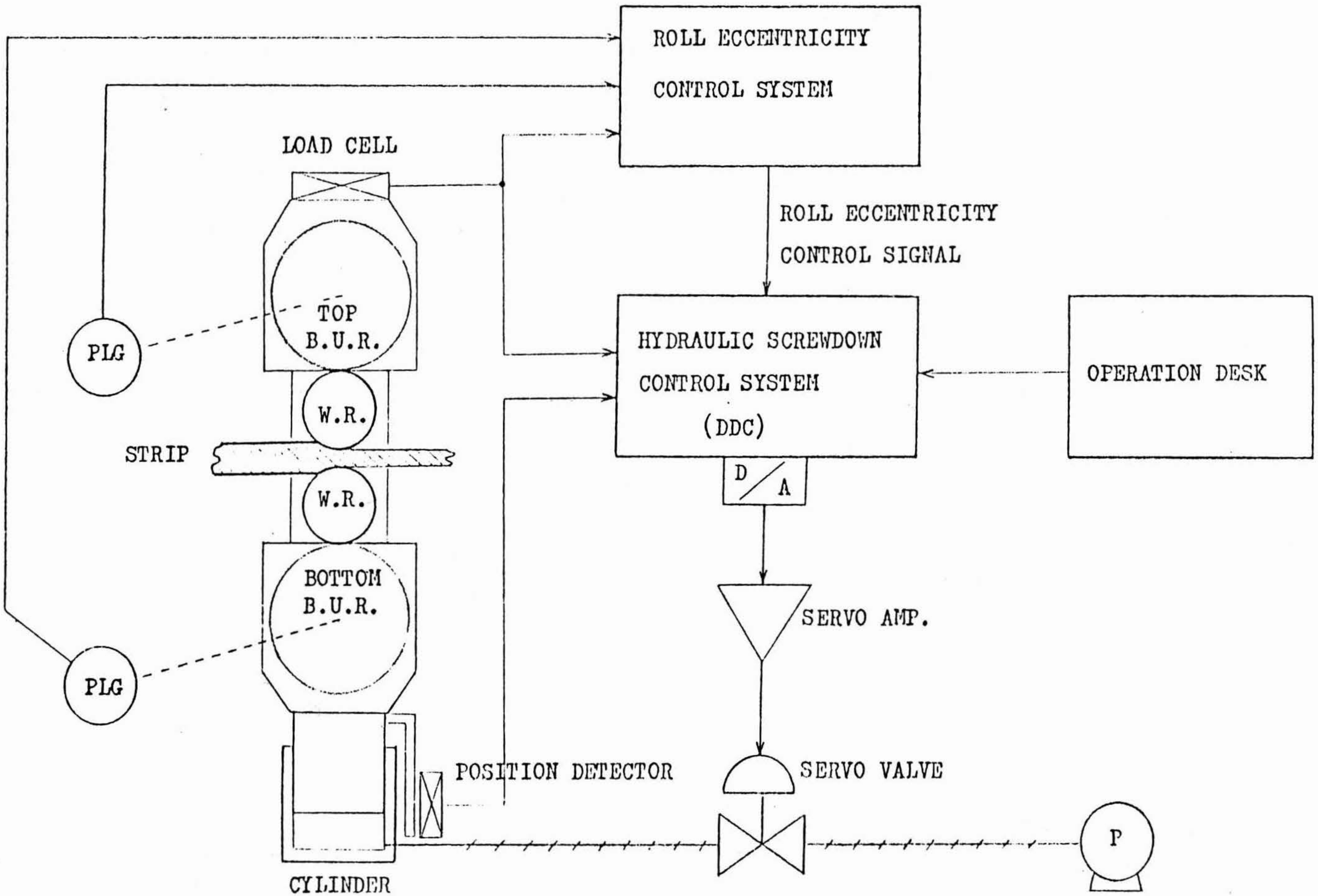


Fig. 1 SYSTEM CONTROL DIAGRAM

2. Roll eccentricity of mill

In a 4-high mill, roll gap causes variation due to eccentricity of back up roll at every revolution of back up roll, as its one cycle, which results in lack of uniformity in product thickness.

2.1 Influence of roll eccentricity

If the gap variation due to roll eccentricity is deemed as B, the amount of variation to be shifted to product thickness becomes KB (K is an influence coefficient) in no BISRA control conditions and the variation is decreased to a few tenths.

However, in case the BISRA control is being carried out, product thickness is influenced by roll eccentricity of nearly 100%, for the following reason.

2.2 Relation between BISRA control and roll eccentricity

In the BISRA control, the mill is deemed as one spring and roll gap is controlled by detection of rolling force, defining "increase of rolling force = expansion of roll gap" and "decrease of rolling force = contraction of roll gap".

On the other hand, in the roll eccentricity it can be said that "increase of rolling force = contraction of roll gap" and "decrease of rolling force = expansion of roll gap".

In other words, the BISRA control and the roll eccentricity are inversely related with each other.

Consequently the roll eccentricity compensation control is particularly important, when the BISRA control is carried out.

3. MHI roll eccentricity compensation control system (patent pending)

This system has been designed after full considerations and examinations of characteristics of roll eccentricity, and its principle, action and characteristics are shown below.

3.1 Control principle

1) Mode 1

Before start of rolling, the rolls are rotated in contact condition under a certain rolling force. The variation of the rolling force is detected along with the rotation angle of the top and bottom back up rolls and memorized by mini computer. The rotation angle is detected by the pulse generators connected to the top and bottom back up rolls.

The separation of roll eccentricity waves of the top and bottom back up rolls is achieved by simple calculation on a mini computer from the above data.

During rolling, the separated eccentricities of the top and bottom back up rolls are combined in contrast with the rotation angle and converted to roll eccentricity compensation signal.

2) Mode 2

From the rolling force variation signal just after start of rolling, a roll eccentricity signal for one cycle is separarated and calculated by computer.

The separation is carried out by utilizing the fact that the period of roll eccentricity coincides with the period of back up roll revolution. In the 2nd cycle, control is carried out by putting out the eccentricity signal sought out in the 1st cycle, in synchronization with the revolution of back up roll, and also if any rolling force variation due to roll eccentricity remains, it is calculated from the rolling force signal.

In the 3rd cycle, control is carried out with a signal which has been obtained by means of compensating the eccentricity signal sought for in the 1st cycle by the signal sought for in the 2nd cycle. In the same method control is carried out in and after the 4th cycle.

3.2 Characteristics of MHI roll eccentricity compensation system

As mentioned above this system has adopted the method of separating and extracting the roll eccentricity control signal by computer from the rolling force signal, and has the following features.

1. It is not necessary to provide any special equipment at the mill for the detection of roll eccentricity.
2. The detection of roll eccentricity, before start of rolling, is done automatically by mini computer.
3. Since this system has adopted a method that the roll eccentricity signal calculated by computer is compensated at every eccentricity compensation control cycle, a signal of high accuracy can be obtained.
4. The main point of roll eccentricity compensation control is the detection of eccentricity signal. Effective roll eccentricity compensation control can be expected by the combination of two roll eccentricity control modes.

4. Results of actual test

We made operation test of the roll eccentricity compensation system to affirm the effects of the roll eccentricity control in Kashima Works of Sumitomo Metal Industries and obtained good results as shown in Figs. 2 and 3.

Fig. 2 shows the results of the roll eccentricity control in mode 2 with the rolls being rotated in contact condition. The measurements indicate that the variation of rolling force due to the roll eccentricity was reduced from 88 to 20 tons or to about one-fourth through roll eccentricity control.

The rolling conditions during this test are as follows:

1) Rolling force : 1,000 tons
2) Rolling speed : 200 m/min(at No.1 st'd)
3) Mill modulus on BISRA AGC : 2,100 tons/mm

Fig. 3 shows the results of the roll eccentricity control in mode 2 during rolling. The measurements indicate that thickness deviation owing to the roll eccentricity at No.1 stand was improved from 31μ to 16μ at the delivery side of No.2 stand through roll eccentricity control.

The rolling conditions during this test are as follows:

1) Rolling force of No.1 stand : 1,000 tons
2) Rolling speed of No.1 stand : 398 m/min
3) Material thickness : 2.3 mm
   Product thickness : 0.5 mm

The effects of the roll eccentricity control in mode 1 will be measured in Kashima Works on and from July 22.

Note: Since patent is applied for the method and system of the roll eccentricity compensation control described above, please refrain from disclosing any information contained to a third party.

Fig.2 The effect of roll eccentricity control at the kiss roll (Mode 2: on line detection of roll eccentricity)

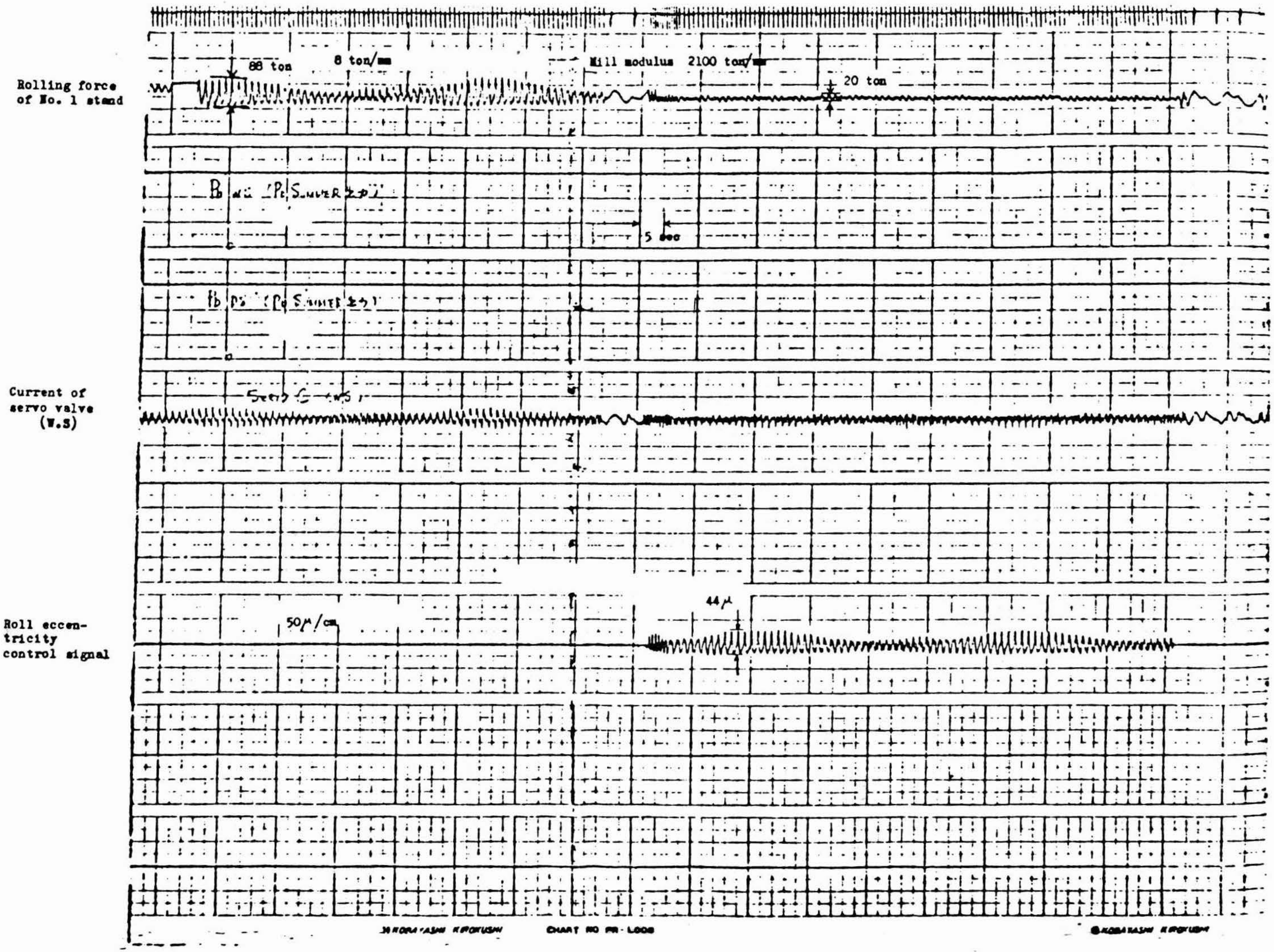
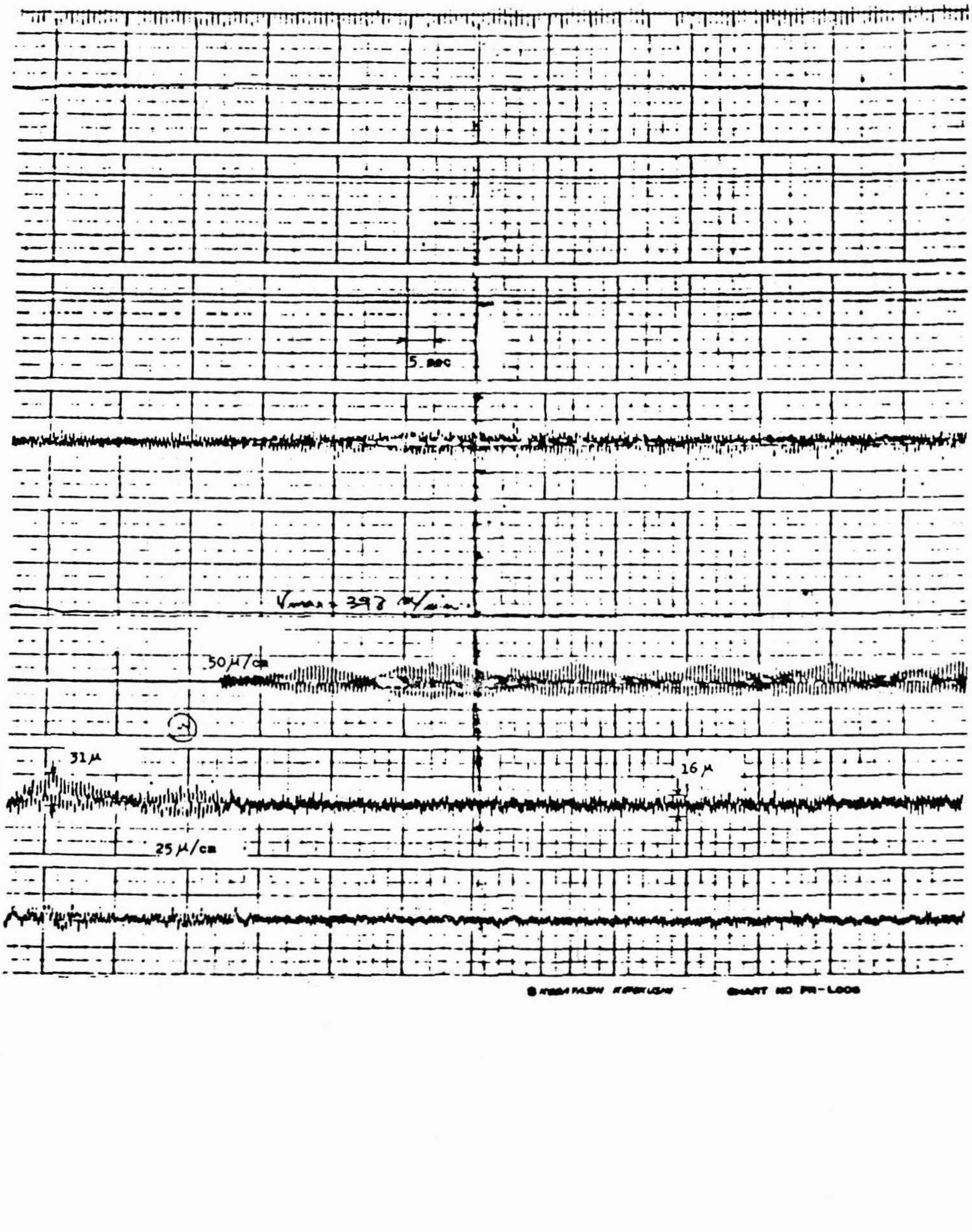


Fig. 3 The effect of roll eccentricity control at the rolling of strip
(Mode 2: on line detection of roll eccentricity)



STUDY ON THE ARRANGEMENT OF
HYDRAULIC SCREW-DOWN EQUIPMENT
IN A TANDEM HOT STRIP MILL AND
A SUITABLE LOOPER

146-71402

September, 1978

Study on the Arrangement of Hydraulic Screw-down Equipment in a Tandem Hot Strip Mill and a Suitable Looper

There is a tendency in recent years to adopt hydraulic screw-down equipment in rolling mills: in tandem hot strip mills as well as in tandem cold strip mills and plate mills.
In this paper the authors discuss the arrangement of hydraulic screw-down equipment in a tandem hot strip mill and the kind of looper suited to the mill in order to obtain referential data for the design of a tandem hot strip mill.
The results can be summarized as follows:

(1) A tandem hot strip mill having hydraulic screw-down equipment in the last two stands has the same gage variations as that having hydraulic screw-down equipment in all stands.

(2) In order to make tension variation smaller it is necessary to use a new type low inertia looper.

1. Introduction

Recently a hydraulic screw-down system is used in rolling mills, not only in the tandem cold strip mill but also in the plate mill. This trend is also seen in the hot strip mill. The reason is as follows.

A motor screw-down system is no longer sufficient in both mechanical strength and response characteristic because of the increase in AGC load and skid-mark temperature difference due to lower slab heating required by energy saving and the reduction in skid-mark period due to larger product thickness.

There are, on the other hand, the requirements for higher product thickness accuracy and larger yield.

For references in planning construction of new or improved hot strip mills in the future, studies have been made on how to arrange hydraulic screw-down stands and the suitable kinds of looper, using a dynamic simulator of the hot strip mill developed jointly with Mitsubishi Electric Corporation.

2. Simulation model and conditions

In evaluation with the simulation, problems are specifications of an object to be studied, validity of the mathematical model and specifications of the disturbances.

2.1 Specifications of a rolling mill

Tables 1~3 show main specifications of a rolling mill, specifications of screw-down equipment and specification of looper.

Table 1

Main specification of rolling mill

Items \ Stand No.	#1	#2	#3	#4	#5	#6	#7
Roll diameter (mm)	762	762	762	762	762	762	762
Motor output (kW)	9000	9000	9000	9000	9000	9000	7400
Motor speed (rpm)	115/300	115/300	115/300	115/300	152/395	190/495	212/550
Reduction ratio	3.14	2.02	1.35	-	-	-	-
Mill constant (t/mm)	600	600	600	600	600	600	600

Table 2

Specifications of screw-down equipment

Screw-down equipment	Item	Specification
Motor screw-down	Motor Reduction ratio Screw lead Back-lash	2setsx75/150kWx515/1030rpm 1/360 25 mm $\pm$ 15 μ
Hydraulic screw-down	Cylinder diameter x stroke Cylinder maximum push Servo-valve	920x10 mm 4000 t/stand (pressure 300 kgf/cm^2) MOOG 72-101

Table 3

Specifications of loopers

Item	Low inertia looper	Conventional looper
Motor	2setsx37.5kWx44rpm	1setx75kWx515rpm
Reduction gear	None	11.6

2.2 Mathematical model

In order for the dynamic simulator to be used generally, the mathematical model is in detail, as follows.

(1) Rolling theoretical formulae

For total roll force, torque and forward slip ratio, considering the tension, Sims' formulae are used. For mėan flow strees, Shida's formula is used. These are all solved as non-linear equation.

(2) Control systems

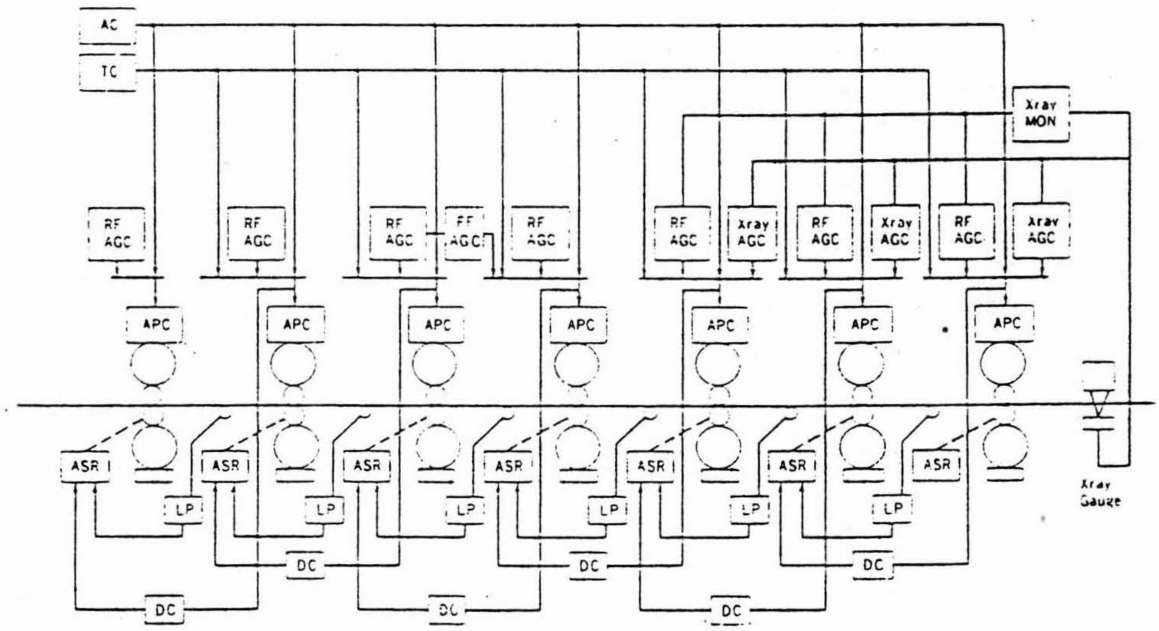
Control systems constitution is shown in Fig. 1. Each control loop includes non-linear friction, back-lash, saturation, dead zone and various minor loops, so that the simulation is in high accuracy even for small and large amplitudes. However, the present purpose is to examine the effects of arrangement of hydraulic screw-down equipments and the performance of loopers comparatively. In the constitution, only screw-down control and looper control systems are thus involved.

2.3 Simulation conditions

With the simulation, a system is evaluated only by strip gage variation and tension variation; strip shape is not considered.

(1) Screw-down equipments arrangement

To compare the performance between the hydraulic screw-down equipments arrangement and the looper, 7 cases of motor and hydraulic screw-down equipments combination in Table 4 are simulated. Mill rigidities



RF AGC	: Roll Force (BISRA)AGC	FF AGC	: Feed Forward AGC	DC	: Draft compensation
				LP	: Looper control
Xray AGC	: X ray AGC	AC	: Acceleration compensation	APC	: Automatic position control
Xray MON	: X ray Monitor	TC	: Tail end compensation	ASR	: Automatic speed regulator

Fig. 1 Dynamic simulation of a hot strip mill

Table 4

Arrangement of screw-down equipments

Case \ Stand No.	F1	F2	F3	F4	F5	F6	F7
1	M	M	M	M	M	M	M
2	H	H	H	H	H	H	H
3	M	M	M	M	M	M	H
4	M	M	M	M	M	H	H
5	M	M	M	M	H	H	H
6	M	M	M	H	H	H	H
7	H	H	H	M	M	M	M

Notes: 1. M is motor screw-down equipment, and H is hydraulic screw-down equipment.

2. Mill rigidities are all 3,000 t/mm, mill rigidity of the strip mill is 600 t/mm.

3. Conventional loopers for case 1; GD^2 $\frac{1}{4}$ low inertia loopers for the rest.

Table 5

Rolling conditions

Items	Condition
Bar thickness	30 mm
Strip thickness	3.0 mm
Strip width	1565 mm
Rolling speed	1200 m/min
Entry bar temperature at #1 stand	1071 °C
Exit strip temperature at #7 stand	937 °C
Between-stands tension	0.2 ~ 0.8 kg/mm^2

are all 3000 t/mm with motor and hydraulic screw-down stands controlled. In Table 6 are shown 11 cases of mill rigidities and screw-down stands arrangement combination.

(2) Rolling conditions

Conditions are shown in Table 5. Strip temperature is in steady state in each stand, except the portion of skid mark. The value is obtained with other program.

(3) Disturbances

For disturbances in a hot strip mill, there are that of temperature such as skid mark and thermal run-down, entry strip gage variation, oil film change and roll eccentricity. Of these, skid mark and roll eccentricity are severe in the strip gage control system. However, the roll eccentricity is eliminated with a roll eccentricity controlling device. In the present study, only the skid mark as in Fig. 2 considered.

(4) Response characteristics of screw-down equipments

For step 200 μ entry gage variation, control gains in motor screw-down equipment and hydraulic screw-down equipment are so adjusted to give response characteristics in Fig. 3. Response wave-form in motor screw-down equipment is distorted due to back-lash of screw and reduction gear, non-linear friction of screw and dead zone suppressing the self-excited vibration caused thereby.

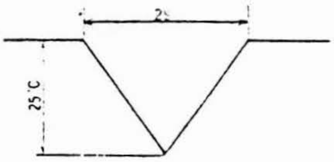


Fig. 2 Skid mark

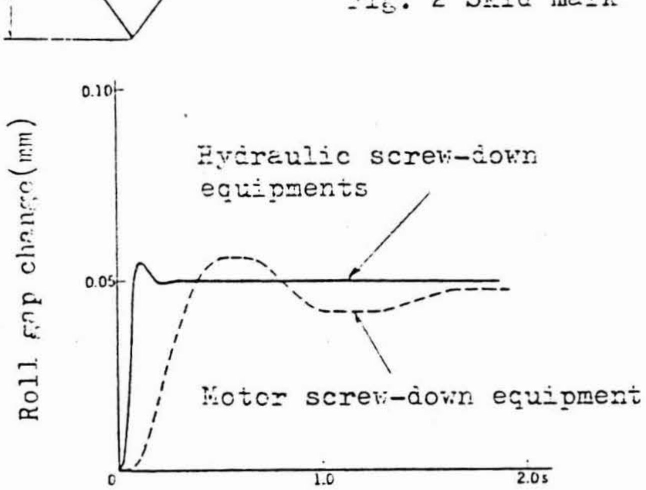


Fig. 3 Dynamic performance of screw-down equipments (Step 0.2mm entry gage variation)

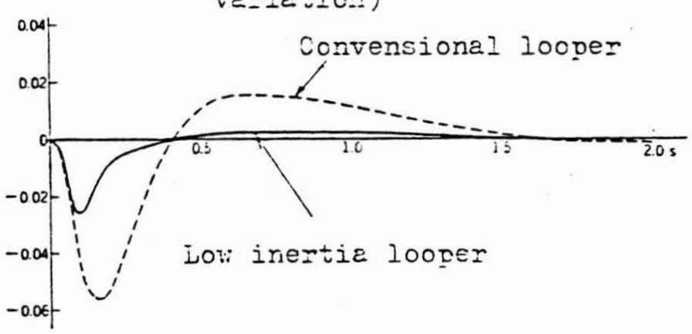


Fig. 4 Dynamic performance of loopers between #1 and #2 stand. (Step 0.3mm rpm#2 stand speed variation)

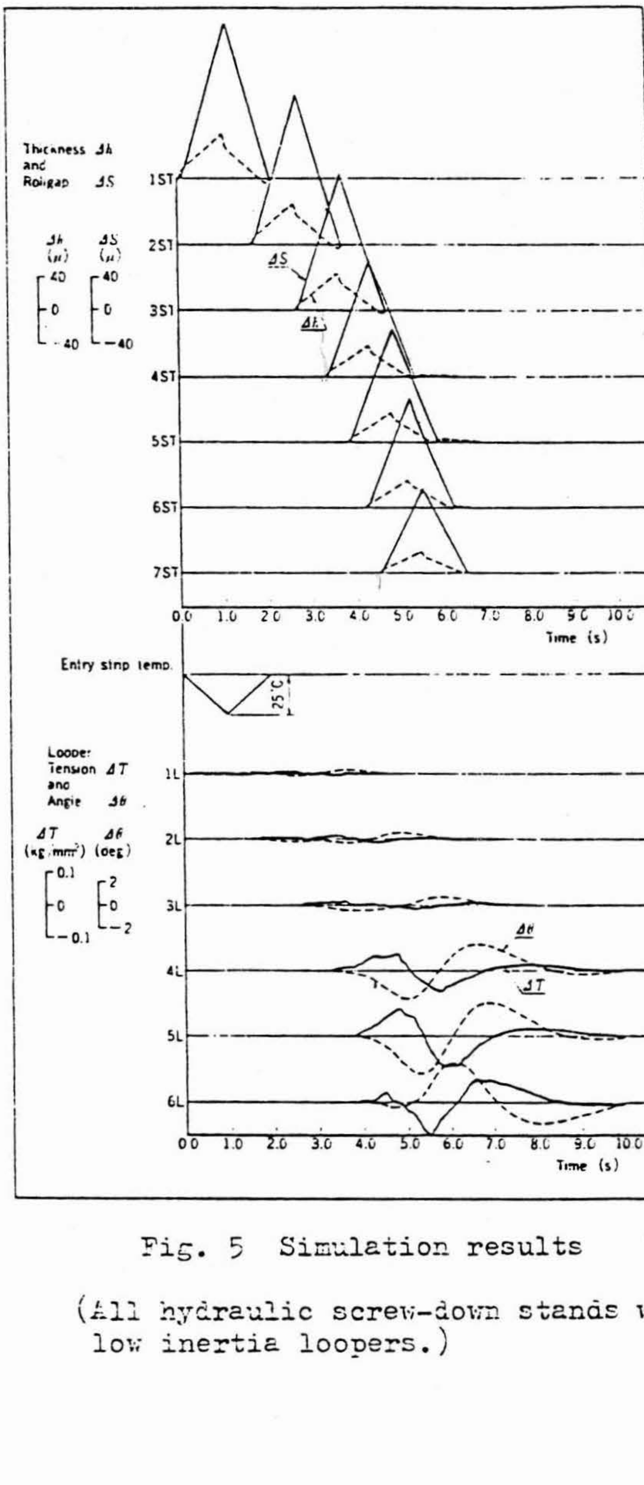


Fig. 5 Simulation results

(All hydraulic screw-down stands with low inertia loopers.)

(5) Response characteristic of looper
For 0.3 rpm roll speed step decrease in No. 2 stand, the tension between No. 1 and No. 2 stand shows response characteristics in conventional looper control system and low inertia looper control system as in Fig. 4.

3. Simulation results
Simulation was made with large computer CDC CYBER 173; the results were output with a plotter. Fig. 5 shows the results of simulation in the case of all hydraulic screw-down stands. Reading maximum deviations (P-P: peak-to peak) value) from the plotter output, results are presented in Figs. 6 to 11. The computing time (excluding the plotting time with a plotter) is about 60 times the actual time.

3.1 Screw-down equipments arrangement and exit gage deviations
To have an indication of proper screw-down equipments arrangement, exit strip gage deviations will be examined comparatively between 5 cases: i.e. no screw-down AGC, all motor screw-down stands, all hydraulic screw-down stands, former-half hydraulic screw-down stands, and latter-half hydraulic screw-down stands. As seen in Fig. 6, in the case of all same screw-down stands, the exit gage deviation in each stand decreases down the stream. In the case of latter-half 4 hydraulic screw-down stands, the curve first in motor screw-down stands shifts to that in hydraulic screw-down ones.

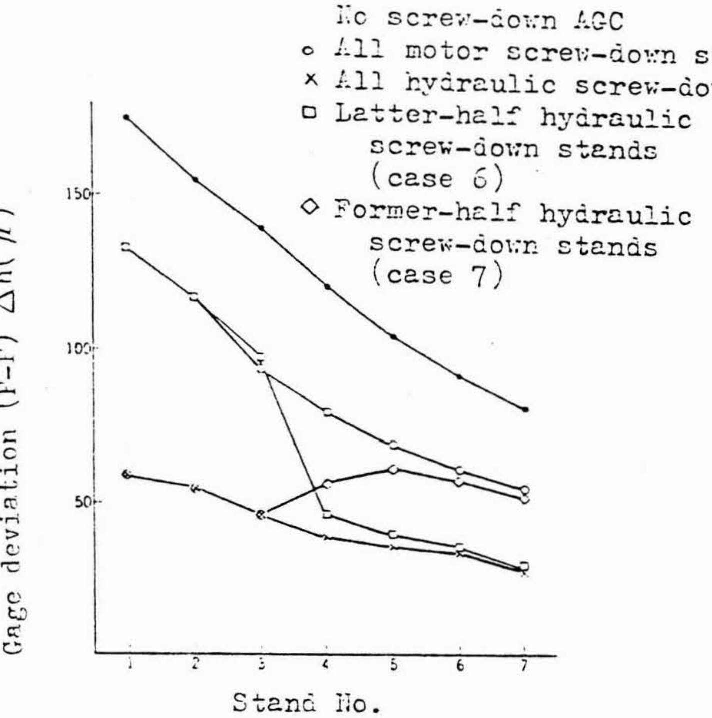


Fig. 6 Screw-down equipments arrangements and exit gage deviations

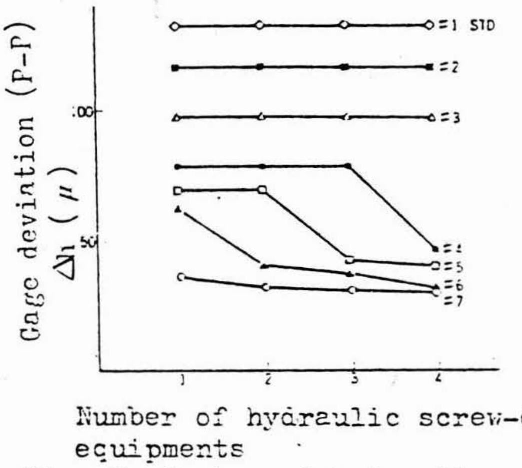


Fig. 8 Number of hydraulic screw-down equipments and gage deviations

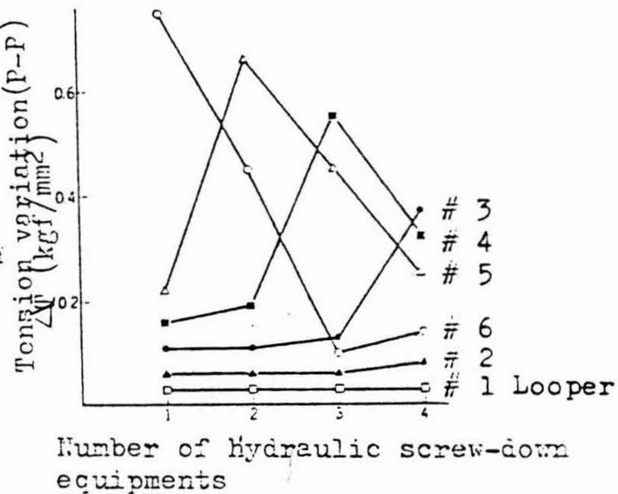


Fig. 9 Number of hydraulic screw-down equipments and tension variations.

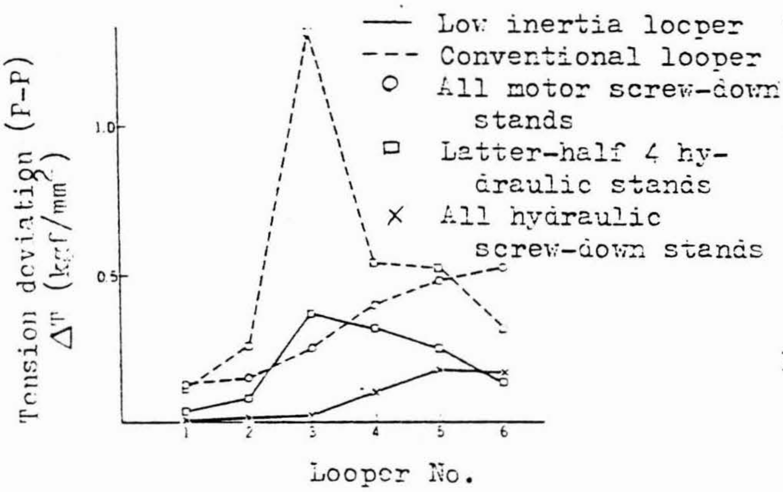


Fig. 7 Screw-down equipments arrangement and tension variations

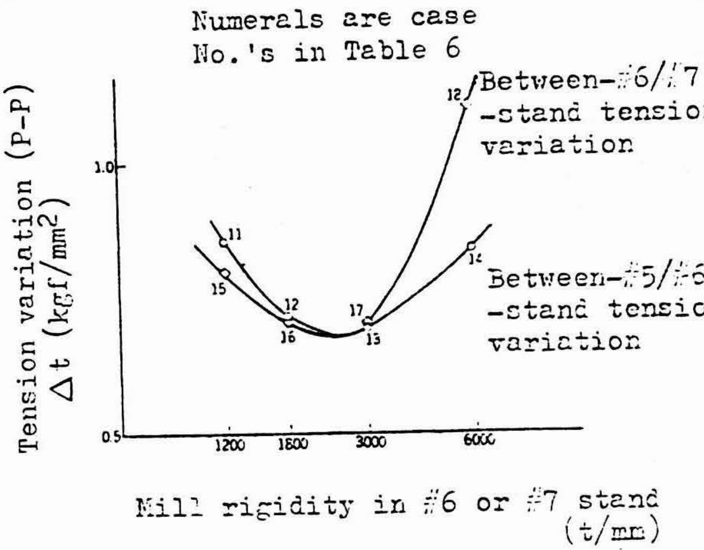


Fig. 11 #6 and #7 stand mill rigidity and tension variation

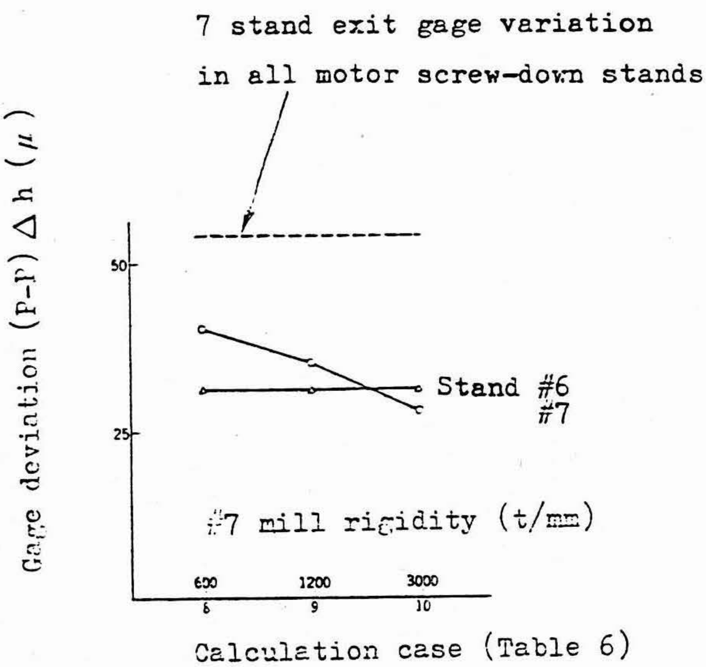


Fig. 10 Mill rigidities and gage deviations in stand #6 and #7

Then, in the case of former-half 4 hydraulic screw-down stands, curve is the opposite.

Results are concluded as follows.

(1) # 7 stand exit gage deviation in latter-half 4 hydraulic screw-down stands and in all hydraulic screw-down stands are similar.

(2) There is the following relationship between # 7 stand exit gage deviation and screw-down equipments arrangement.

All motor screw-down stands $\simeq$ 2/3 x no screw-down AGC
All hydraulic screw-down stands
Latter-half 4 hydraulic screw-down stands
$\simeq$ 1/3 x no screw-down AGC

(3) In certain could strip mills, hydraulic screw-down equipment is used only in # 1 stand. This positioning of hydraulic screw-down equipment is not useful for strip temperature disturbance such as in a hot strip mill.

3.2 Screw-down equipments arrangement, looper type and tension variation

To have an indication of how the tension variations are with hydraulic screw-down equipments arrangement and looper type (conventional, and low inertia), such will be examined comparatively between 4 cases: i.e. all motor screw-down stands with conventional loopers as in a conventional hot strip mill, all hydraulic screw-down stands with low inertia loopers as in a future strip mill, latter-half 4 hydraulic screw-down stands with low intertia loopers as in a future strip mill, and latter-half 4 hydraulic stands with conventional loopers. Results are indicated in Fig. 7, which are concluded as follows.

(1) By use of low intertia loopers, in both all hydraulic screw-down stands and latter-half 4 hydraulic screw-down stands, the tension variations are smaller than a maximum tension variation in all motor screw-down stands with conventional loopers.

(2) In the cases of all same screw-down stands, the tension variation between stands increases down the stream. In latter-half 4 hydraulic screw-down stands, there appears a maximum tension between the adjoining motor and hydraulic screw-down stands. In latter-half 4 hydraulic stands with conventional loopers, the maximum tension variation between the adjoining motor and hydraulic screw-down stands is several times larger than the maximum tension variation in all motor screw-down stands with conventional loopers.

3.3 Number of hydraulic screw-down stands and exit gage deviation in respective stands

The number of hydraulic screw-down equipments in # 7 stand varies toward the upper stream from 1 to 4, and the exit gage deviations in all stands are observed. The relationship revealed is shown in Fig. 8. Numerals at right ends of the bent lines are stand Nos. Each line is the variation in gage deviation at the respective stand. The bent line at bottom, for example, shows how the gage deviation in # 7 stand changes with the number of hydraulic screw-down stands. As seen, the exit gage deviation in # 7 stand extremely little changed beyond 2 hydraulic screw-down stands.

3.4 Number of hydraulic screw-down stands and tension variations

Similary to Chap. 3.3 above, the relationship between the number of hydraulic screw-down equipments up the stream and the tension variations in all stands was examined, which is shown in Fig. 9. Numerals at right ends of the bent lines are looper Nos. As described in Chap. 3.2, the tension is the largest between motor and hydraulic screw-down stands. In the case of 1 hydraulic screw-down stand, the tension variation in # 6 looper is thus the largest, with decrease of the number of hydraulic screw-down equipment, the junction between motor and hydraulic screw-down stands moves downstream, so that the maximum tension variation itself also rises.

3,5 # 7 stand mill rigidity and # 7 stand exit gage deviation

In the results so far given, mill rigidities in the respective stands are uniformly 3000 t/mm. Actually, however, taking into consideration the strip gage deviations and the between-stands tensions, a suitable distribution of mill rigidity in all respective stands is necessary. It was examined how the exit gage deviation in # 7 and # 6 stand would be with such mill rigidities, for cases No. 8 to 10 in Table 6. The mill rigidities, in # 4 to # 6 hydraulic screw-down stands are consecutively 1200, 2400 and 6000 t/mm in all the cases. In # 7 stand ,

Table 6. Arrangement of mill rigidities

Case \ Stand No.	#1		#2		#3		#4		#5		#6		#7	
8	M	3000	M	3000	M	3000	H	1200	H	2400	H	6000	M	600
9	"	"	"	"	"	"	"	"	"	"	"	"	H	1200
10	"	"	"	"	"	"	"	"	"	"	"	"	"	3000
11	"	"	"	"	"	"	M	3000	M	3000	"	1200	"	"
12	"	"	"	"	"	"	"	"	"	"	"	1800	"	"
13	"	"	"	"	"	"	"	"	"	"	"	3000	"	"
14	"	"	"	"	"	"	"	"	"	"	"	6000	"	"
15	"	"	"	"	"	"	"	"	"	"	M	3000	"	1200
16	"	"	"	"	"	"	"	"	"	"	"	"	"	1800
17	"	"	"	"	"	"	"	"	"	"	"	"	"	3000
18	"	"	"	"	"	"	"	"	"	"	"	"	"	6000

Notes: M ; Motor screw-down stand

H ; Hydraulic screw-down stand. Numerals on right side are mill rigidities. Loopers are all of low inertia type.

the rigidity changes are 600 t/mm in motor screw-down stand, 1200 and 3000 t/mm in hydraulic screw-down stand, respectively. The results are shown in Fig. 10. With increase of the mill rigidity in # 7 stand, the gage deviation in # 7 stand decreases. However, unless the rigidity is over about 2000 t/mm the gage deviation in # 7 stand is larger than that in # 6 stand.

3.6 Tension variation between motor and hydraulic screw-down stands and mill rigidity in the hydraulic screw-down stand

As described in Chap. 3.2, there appears the maximum tension variation in junction of motor and hydraulic screw-down stands. It was examined if this maximum tension variation could be reduced by changing the mill rigidity in the hydraulic screw-down stand. The tension variation between # 5 and # 6 stand was thus observed for cases No. 11 to 14 in Table 6: mill rigidities in # 5 motor screw-down stand are 3000 t/mm in all cases, and mill rigidities in # 6 hydraulic screw-down are 1200, 1800, 3000 and 6000 t/mm respectively. The tension variation between # 6 and # 7 stand was then examined for cases No. 15 to 18: mill rigidities in # 6 motor screw-down stands are 3000 t/mm and mill rigidity in # 7 hydraulic stands is changed similarly.

The results are shown in Fig. 11. As indicated, the maximum tension variation can be minimized by way of the mill rigidity in hydraulic screw-down stand somewhat smaller than that in the preceding motor screw-down stand.

4. Conclusion

From the above results described, studies made on proper arrangement of hydraulic and motor screw-down stands and suitable types of loopers are summarized as follows.

4.1 Screw-down stands arrangement

(1) There is no difference in # 7 stand exit gage deviation between downstream 2 hydraulic screw-down stands and all hydraulic screw-down stands. In this case, with skid-mark time width 2 sec, the # 7 stand exit gage deviation may be about 1/2 the deviation in all motor screw-down stands.

(2) Mill rigidity in the last stand has large influence on thickness deviation of the product plate. So, it must be fairly high. Or otherwise, the gage deviation occasionally becomes larger than that in the preceding stand.

(3) There occurs large tension variation in junction between motor and hydraulic screw-down stand. This large variation, however, can be minimized by way of the hydraulic screw-down stand mill rigidity somewhat smaller than the upstream motor screw-down stand mill rigidity.

(4) A front hydraulic screw-down stand is useful against an entry gage disturbance, but not so against a strip temperature disturbance. Therefore, the means is

not effective in a hot rolling mill, though effective in a cold rolling mill.

4.2 Suitable looper

In motor and hydraulic screw-down stands combination with conventional motor loopers, the tension variations are occasionally several times larger than those in motor screw-down stands. By using GD^2 about 1/4 low inertia motor loopers instead, the tension variations may be reduced below those in all motor screw-down stands.

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1) Rolling theories and the applications, compiled by The Iron and Steel Institute of Japan.

2) Shida, Empirical formulae of flow strees in carbon steel, Journal of Japan Society for Technology of Plasticity, Vol. 10, No. 103

3) Yanagi, Strip Temperature Prediction in Hot Mills, Mitsubishi Juko Giho, Vol. 11, No. 1 (1974)