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(54) **ROLLING MILL**

(57) In the rolling mill, an orifice (20) (contracted flow part) and a chamber (21) (expanded part) are provided in a hydraulic line (19) for a hydraulic cylinder (17) (hydraulic pressing means). The orifice (20) is disposed more to the hydraulic cylinder (17) side than the chamber (21). The internal diameter of the orifice (20) is set to be 2.5 mm (φ) or greater and 15 - 85% of the internal diameter of the hydraulic line (19). Thus, it is possible to control milling vibration.

Fig. 1(a)

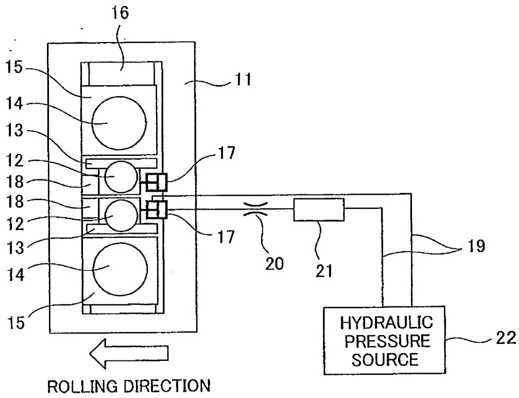

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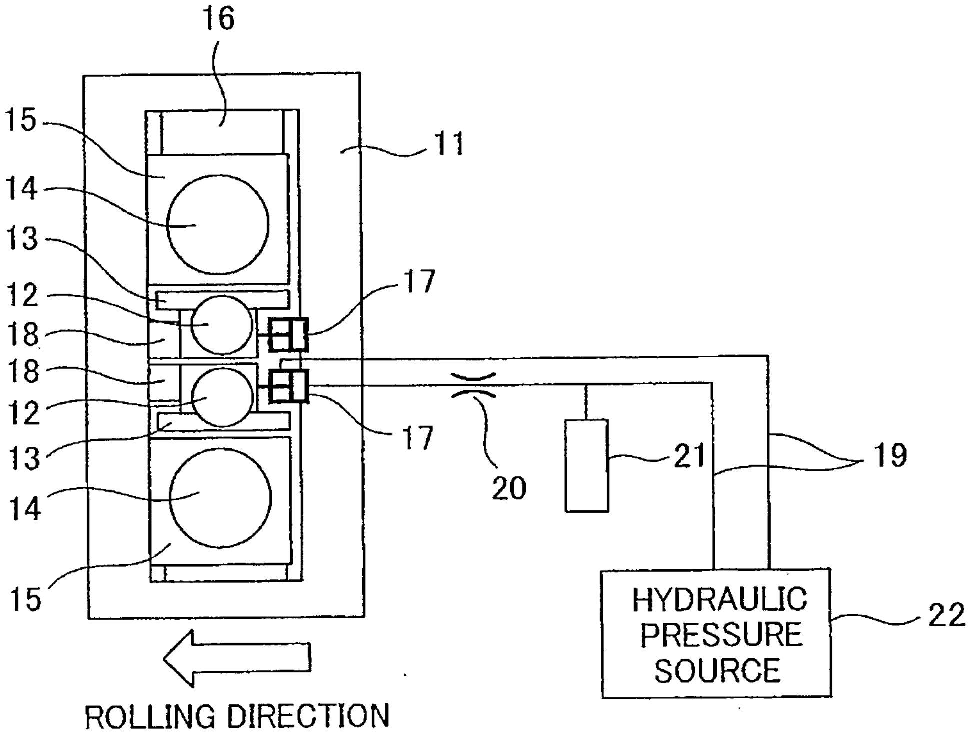
Fig. 1(b)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HYDRAULIC PRESSURE SOURCE
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ROLLING DIRECTION

Description

TECHNICAL FIELD

[0001] The present invention relates to a rolling mill, or more specifically, to a device to control vibration of a hot rolling mill which occurs in the course of rolling with the rolling mill.

BACKGROUND ART

[0002] Hot rolling may cause mill vibration in the course of rolling. The mill vibration means vibration of upper and lower work rolls (WRs) in a horizontal direction (a rolling direction) and in mutually reverse phases. Here, the mutually reverse phases represent a phenomenon that the lower WR moves to a downstream side when the upper WR moves to an upstream side, and on the other hand, the lower WR moves to the upstream side when the upper WR moves to the downstream side. The mill vibration leads to a fluctuation in strip thickness, loosening of various fastening bolts used in the rolling mill, vibration of pipes, and the like.

[0003] The vibration has heretofore been controlled by focusing on a static stiffness. Specifically, a conventional concept has been designed to eliminate a gap between a housing and a work roll chock in a rolling mill by applying pressure with a hydraulic cylinder, and thus to improve the static stiffness in a horizontal direction. Further, by way of extension, the static stiffness has been improved by reducing a diameter (an orifice diameter) of an orifice provided to a hydraulic supply-discharge pipe of a hydraulic cylinder (see Patent Document 1 below).

PRIOR ART DOCUMENT

PATENT DOCUMENT

[0004] Patent Document 1: Japanese Patent Application Publication No. 2001-113308

SUMMARY OF THE INVENTION

PROBLEMS TO BE SOLVED BY THE INVENTION

[0005] As described above, the conventional technique provides the reduced orifice diameter (about ϕ2.0 mm or below) as the means for further improving the static stiffness. Nonetheless, the reduction in the orifice diameter has a limitation because too small an orifice diameter may cause dust clogging, a failure to achieve a designed cylinder operation speed, and the like. Hence, there has been a problem that a sufficient vibration control effect was not actually available therefrom.

[0006] In view of the above, an object of the present invention is to provide a rolling mill which is capable of controlling mill vibration without having to reduce an orifice diameter excessively.

MEANS FOR SOLVING THE PROBLEMS

[0007] A rolling mill according to a first aspect of the invention to solve the above problems is characterized in that the rolling mill comprises:

a housing;
a pair of upper and lower work roll chocks supported by the housing;
a pair of upper and lower work rolls opposed to each other and pivotally supported by the pair of upper and lower work roll chocks, respectively;
roll gap controlling means for applying a predetermined pressure to the work rolls;
a pair of upper and lower first supporting means provided to the housing at positions on one side in a rolling direction for supporting the pair of upper and lower work roll chocks; and
a pair of upper and lower second supporting means provided to the housing at positions on another side in the rolling direction for supporting the pair of upper and lower work roll chocks, wherein
the first supporting means is used as hydraulic pressing means and is made capable of pressing the pair of upper and lower work roll chocks in a horizontal direction,
a flow contracting unit and an expanding unit are provided to a portion of a hydraulic supply-discharge pipe on a head side of the hydraulic pressing means, while disposing the flow contracting unit closer to the hydraulic pressing means than the expanding unit, and
an inside diameter of the flow contracting unit is set equal to or above ϕ2.5 mm and in a size of from 15% to 85%

relative to an inside diameter of the hydraulic supply-discharge pipe.

[0008] A rolling mill according to a second aspect of the invention to solve the above problems is the rolling mill according to the first aspect of the invention, characterized in that a volume of the expanding unit is set in a range from 7% to 180% relative to a volume of the hydraulic pressing means.

[0009] A rolling mill according to a third aspect of the invention to solve the above problems is the rolling mill according to the first or second aspect of the invention, characterized in that a distance between the flow contracting unit at the hydraulic supply-discharge pipe and the hydraulic pressing means is set equal to or below 7 m.

[0010] A rolling mill according to a fourth aspect of the invention to solve the above problems is the rolling mill according to any one of the first to third aspects of the invention, characterized in that a distance between the expanding unit and the flow contracting unit at the hydraulic supply-discharge pipe is set equal to or below 3.5 m.

EFFECT OF THE INVENTION

[0011] According to a rolling mill of the present invention, it is possible to control mill vibration without having to reduce an orifice diameter excessively.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012]

[Fig. 1] Fig. 1 includes schematic diagrams of a rolling mill according to a first embodiment of the present invention.
[Fig. 2] Fig. 2 includes graphs showing relations of an orifice diameter with a static stiffness, a damping ratio, and a dynamic stiffness, respectively, in a conventional rolling mill.
[Fig. 3] Fig. 3 includes graphs showing relations of an orifice diameter with a static stiffness, a damping ratio, and a dynamic stiffness, respectively, in the rolling mill according to the first embodiment of the present invention.
[Fig. 4] Fig. 4 is an analysis model diagram concerning the dynamic stiffness of the rolling mill according to the first embodiment of the present invention.
[Fig. 5] Fig. 5 is a graph showing an excitation force and a work roll displacement in the rolling mill according to the first embodiment of the present invention.
[Fig. 6] Fig. 6 is a graph showing a relation between an excitation frequency and the dynamic stiffness in the rolling mill according to the first embodiment of the present invention.
[Fig. 7] Fig. 7 illustrates graphs showing a relation between the orifice diameter and a dynamic stiffness ratio in the rolling mill according to the first embodiment of the present invention.
[Fig. 8] Fig. 8 illustrates graphs showing a relation between a chamber volume and the dynamic stiffness ratio in the rolling mill according to the first embodiment of the present invention.
[Fig. 9] Fig. 9 is a graph showing a relation between a cylinder-to-orifice distance and the dynamic stiffness ratio in the rolling mill according to the first embodiment of the present invention.
[Fig. 10] Fig. 10 is a graph showing a relation between an orifice-to-chamber distance and the dynamic stiffness ratio in the rolling mill according to the first embodiment of the present invention.

MODE FOR CARRYING OUT THE INVENTION

[0013] In regard to a rolling mill according to the present invention, the earnest investigations by the inventors have revealed a characteristic that a damping ratio varies with an orifice diameter by providing an appropriate chamber, and also the existence of an appropriate range for the orifice diameter from the viewpoint of controlling mill vibration by focusing on a dynamic stiffness derived from a static stiffness and the damping ratio. In addition, the existence of an appropriate range for a chamber volume has also been revealed. A rolling mill according to the present invention will be described below in the form of an embodiment and by using the drawings.

[First Embodiment]

[0014] First, a rolling mill according to a first embodiment of the present invention will be described by using Fig. 1. Fig. 1 includes schematic diagrams of the rolling mill according to the first embodiment of the present invention.

[0015] As shown in Fig. 1 (a), the rolling mill according to the first embodiment of the present invention includes a housing 11, work rolls 12, work roll chocks 13, backup rolls 14, backup roll chocks 15, roll gap controlling means 16, hydraulic cylinders 17 (hydraulic pressing means, first supporting means), housing liners 18 (second supporting means), a hydraulic supply-discharge pipe 19, an orifice 20 (a flow contracting unit), a chamber 21 (an expanding unit), and a hydraulic pressure source 22.

[0016] The pair of upper and lower work roll chocks 13 are supported by the housing 11.

[0017] The pair of upper and lower work rolls 12 are opposed to each other and are pivotally supported by the pair of upper and lower work roll chocks 13, respectively.

[0018] The pair of upper and lower backup rolls 14 are pivotally supported by the pair of upper and backup roll chocks 15 and are opposed to the pair of upper and lower work rolls 12, respectively.

[0019] The roll gap controlling means 16 applies a predetermined pressure to the work rolls 12 through the backup rolls 14.

[0020] The pair of upper and lower hydraulic cylinders 17 are provided to the housing 11 at positions on one side in a rolling direction so as to support the pair of upper and lower work roll chocks 13, and are made capable of pressing the pair of upper and lower work roll chocks 13 in a horizontal direction.

[0021] The pair of upper and lower housing liners 18 are provided to the housing 11 at positions on the other side in the rolling direction so as to support the pair of upper and lower work roll chocks 13.

[0022] The orifice 20 and the chamber 21 are provided to a portion of the hydraulic supply-discharge pipe 19 on a head side of the corresponding hydraulic cylinder 17 such that the orifice 20 is disposed closer to the hydraulic cylinder 17 than the chamber 21 is. Alternatively, as shown in Fig. 1(b), the rolling mill according to the first embodiment of the present invention may be configured to dispose the chamber 21 while branching off a pipe from the hydraulic supply-discharge pipe 19.

[0023] In the following, a description will be given of an orifice diameter (an inside diameter of the orifice 20).

[0024] The rolling mill according to the first embodiment of the present invention focuses on an improvement of a dynamic stiffness in the horizontal direction of the rolling mill in order to control mill vibration. The dynamic stiffness (K_d) is expressed by 2 $\times$ a static stiffness (K) $\times$ a damping ratio (ζ).

[0025] Fig. 2 includes graphs showing relations of an orifice diameter with a static stiffness, a damping ratio, and a dynamic stiffness, respectively, in a conventional rolling mill. Fig. 3 includes graphs showing relations of the orifice diameter with the static stiffness, the damping ratio, and the dynamic stiffness, respectively, in the rolling mill according to the first embodiment of the present invention. Figs. 2 (a) and 3 (a) are each a graph showing the relation between the static stiffness and the orifice diameter. Figs. 2(b) and 3(b) are each a graph showing the relation between the damping ratio and the orifice diameter. Figs. 2(c) and 3(c) are each a graph showing the relation between the dynamic stiffness and the orifice diameter.

[0026] As shown in Figs. 2(a) and 2(b), the static stiffness and the dynamic stiffness have heretofore been considered to become larger as the orifice diameter is made smaller based on the concept that the damping ratio remains constant irrespective of the orifice diameter. However, the earnest investigations by the inventors have revealed the characteristic that the damping ratio varies with the orifice diameter as shown in Fig. 3(b) by providing an appropriate chamber.

[0027] Specifically, as shown in Figs. 3(a) and 3(b), while the static stiffness becomes larger as the orifice diameter is made smaller, the damping ratio is reduced because oil in the hydraulic supply-discharge pipe 19 flows less smoothly in the orifice 20. On the other hand, while the static stiffness becomes smaller as the orifice diameter is made larger, the damping ratio is increased because the oil in the hydraulic supply-discharge pipe 19 flows more smoothly in the orifice 20.

[0028] Further, as shown in Fig. 3(c), it turned out that the dynamic stiffness was improved in particular by setting the orifice diameter within a predetermined range (to be described later).

[0029] In the past, nonetheless, in the case where only an orifice 20 was provided to the hydraulic supply-discharge pipe 19, the orifice diameter has been reduced (to about ϕ 2.0 mm or below) while focusing only on the static stiffness based on the concept as shown in Figs. 2 (a) and 2(b) that the damping ratio remains constant even if the orifice diameter is expanded.

[0030] However, it turned out that the installation of the chamber 21 made it possible to improve the damping ratio along with the expansion of the orifice diameter as mentioned above. Accordingly, this embodiment provides the chamber 21, and finds an appropriate range of the orifice diameter, which can further increase a vibration control effect, by focusing on the orifice diameter and the damping ratio, i.e., the dynamic stiffness.

[0031] In the meantime, a valve stand is located distant from the hydraulic cylinder 17 and the diameter of the hydraulic supply-discharge pipe 19 is reduced by using the orifice 20 as well. Accordingly, the mere provision of the orifice 20 can result in a situation where the oil inside the hydraulic supply-discharge pipe 19 flows less smoothly whereby the damping ratio is kept from increasing.

[0032] On the other hand, according to the rolling mill of the first embodiment of the present invention, the chamber is installed in the middle of the hydraulic supply-discharge pipe 19 and on an outlet side of orifice 20, so that the damping ratio can be improved by creating a pressure difference while feeding the oil through the orifice 20.

[0033] Meanwhile, from the viewpoint of the dynamic stiffness, it also turned out that the dynamic stiffness was improved in particular by setting a volume of the chamber 21 within predetermined range (to be described later).

[0034] Now, the range of the orifice diameter with which to improve the dynamic stiffness in particular will be determined.

[0035] Fig. 4 is a simulation model diagram in terms of the dynamic stiffness. Fig. 4 depicts models by using A as the orifice 20, B as the hydraulic supply-discharge pipe 19, K1 as a housing spring constant, K as the static stiffness of the model as a whole, c as a damping coefficient of a structure, D as the housing 11, E as the work rolls 12 and the work roll chocks 13, F as the hydraulic cylinder 17, and P as the hydraulic pump, respectively. Here, motion equations of the work rolls 12 and the work roll chocks 13, and a characteristic of the orifice 20 to determine its flow rate depending on the pressure difference are incorporated.

[0036] According to Fig. 4, the motion equations are expressed by the following formulae (1) and (2):
[Formula 1]

$$m\ddot{X} + c\dot{X} + kX = f_0 \sin \omega t + F_{oil} \qquad (1),$$

where f_0: excitation force, ω: excitation frequency, F_{oil}: force from hydraulic pressure cylinder, X: work roll displacement, m: mass of work roll and work roll chock, c: damping coefficient, k = K1: housing spring constant, and t: time; and
[Formula 2]

$$Q_{or} = co \cdot A_{or} \sqrt{\frac{\Delta P}{\rho}} \qquad (2),$$

where Q_{or}: orifice flow rate, A_{or}: orifice cross-sectional area, ΔP: pressure difference between back and front of orifice, ρ: hydraulic oil density, and co: flow rate coefficient.

[0037] In addition, the dynamic stiffness K_d is expressed by the following formula (3):
[Formula 3]

$$K_d = \frac{2f_0}{2X} \qquad (3).$$

[0038] From the formula (1), a relation between the excitation force f_0 and the work roll displacement X is derived as shown in a graph of Fig. 5.

[0039] Then, the work roll displacement X for each excitation frequency ω is calculated and a ratio of the excitation force f_0 to the work roll displacement X is calculated (the formula (3)). Note that values of this ratio vary with values of the excitation frequency ω as shown in Fig. 6.

[0040] Therefore, the smallest value out of the ratios of the excitation force f_0 to the work roll displacement X that vary with the values of the excitation frequency ω is evaluated as the dynamic stiffness K_d. Specifically, the dynamic stiffness K_d is defined as the smallest value out of the ratios of the excitation force f_0 to the work roll displacement X, which are obtained by giving values of the excitation force f_0 to the respective excitation frequency ω. The dynamic stiffness K_d is a value to determine movement at the time of vibration.

[0041] The values of the dynamic stiffness K_d are evaluated as described above by using various orifice diameters. Results are shown in Figs. 7(a) and 7(b).

[0042] Fig. 7 (a) is a graph showing a relation between the orifice diameter and the dynamic stiffness ratio, and Fig. 7(b) is a graph showing a relation between the dynamic stiffness ratio and a ratio of the orifice diameter to an inside diameter of the hydraulic supply-discharge pipe 19 (an orifice diameter-to-pipe-inside-diameter ratio). Here, Fig. 7(a) shows the case where the inside diameter (the pipe inside diameter) of the hydraulic supply-discharge pipe 19 is equal to ϕ18 mm. In the meantime, the dynamic stiffness ratio means a ratio to the dynamic stiffness when the orifice diameter is equal to 0, i.e., when the hydraulic cylinder is fully closed (the same applies to Figs. 8 and 9).

[0043] As shown in Figs. 7(a) and 7(b), the orifice diameter previously designed to be equal to or below about ϕ2.0 mm is then designed in a larger size to the contrary. In this way, it turned out that the dynamic stiffness ratio was improved further. Particularly, in Fig. 7(a), there are inflection points where the orifice diameter is equal to 2.5 mm and 15 mm, respectively, and the dynamic stiffness ratio rises sharply in the range from 2.5 mm to 15 mm inclusive. In Fig. 7 (b), there are inflection points where the orifice diameter-to-pipe-inside-diameter ratio is equal to 0.15 (15%) and 0.85 (85%), respectively, and the dynamic stiffness ratio rises sharply in the range from 15% to 85% inclusive.

[0044] In addition, the previously problematic dust clogging is less likely to occur by setting the orifice diameter equal to or above ϕ2.5 mm.

[0045] Accordingly, in the rolling mill of the first embodiment of the present invention, the orifice diameter is set to such a size equal to or above ϕ2.5 mm with the ratio to the pipe inside diameter within the range of 15% to 85%. In this case, the dynamic stiffness ratio is improved to 1.2 or above.

[0046] Meanwhile, a relation between a volume of the chamber 21 and the dynamic stiffness ratio has also been sought. Results turned out as shown in Figs. 8(a) and 8(b).

[0047] Fig. 8 (a) is a graph showing a relation between the volume of the chamber 21 (a chamber volume) and the dynamic stiffness ratio, and Fig. 8(b) is a graph showing a relation between the dynamic stiffness ratio and a ratio of the chamber volume to a volume of the hydraulic cylinder 17 (a chamber-volume-to-cylinder-volume ratio).

[0048] Here, the cylinder volume is defined as a volume to be determined by a cylinder diameter and a stroke. In a specific example, when the cylinder sizes is described as D 250 mm (a head diameter) / d 230 mm (a rod diameter) $\times$ a 90-mm stroke, the cylinder volume Vc is expressed by $Vc = (\pi/4) \times 25^2 \times 9$ (cm^3), which is equal to about 4.4 liters. Accordingly, the chamber-volume-to-cylinder-volume ratio is in a range from 0.07 to 1.8 in the case of the chamber volume in a range from 0.3 to 8.0 liters as described later.

[0049] As shown in Fig. 8(a), when the chamber volume is equal to or above 0.3 liter, the dynamic stiffness ratio is improved to 1.2 and above, or up to about 3.0 at the maximum. In other words, as shown in Fig. 8(b), when the chamber-volume-to-cylinder-volume ratio is equal to or above 0.07, the dynamic stiffness ratio is improved to 1.2 and above, or up to about 3.0 at the maximum.

[0050] Here, regarding values of the chamber volume greater than 8.0 liters in Fig. 8(a), i.e., values of the chamber-volume-to-cylinder-volume ratio greater than 1.8 in Fig. 8(b), the graphs reach saturation levels where the dynamic stiffness ratio is hardly improved anymore. Thus, the increase in chamber volume above the aforementioned value turned out not to contribute to a significant increase in effect.

[0051] Accordingly, in the rolling mill of the first embodiment of the present invention, the chamber-volume-to-cylinder-volume ratio is set equal to or above 0.07 and equal to or below 1.8 (i.e., equal to or above 7% and equal to or below 180%). Here, the dynamic stiffness ratio is equal to or above 1.2 in this case.

[0052] In the meantime, a relation between a distance from the hydraulic cylinder 17 to the orifice 20 (a cylinder-to-orifice distance) and the dynamic stiffness ratio has also been investigated. As a consequence, the dynamic stiffness ratio turns out to be equal to or above 1.2 when the cylinder-to-orifice distance is equal to or below 7.0 m as shown in a graph of Fig. 9.

[0053] Accordingly, in the rolling mill of the first embodiment of the present invention, the cylinder-to-orifice distance is set equal or below 7.0 m.

[0054] In addition, a relation between a distance from the orifice 20 to the chamber 21 (an orifice-to-chamber distance) and the dynamic stiffness ratio has also been investigated. As a consequence, the dynamic stiffness ratio turns out to be equal to or above 1.2 when the orifice-to-chamber distance is equal to or below 3.5 m as shown in a graph of Fig. 10.

[0055] Accordingly, in the rolling mill of the first embodiment of the present invention, the orifice-to-chamber distance is set equal or below 3.5 m.

[0056] The rolling mill according to the first embodiment of the present invention has been described above. In Figs. 1 and 2, the rolling mill according to the first embodiment of the present invention is provided with the orifice 20 and the chamber 21 only on the head side of the hydraulic supply-discharge pipe 19. In addition, however, another orifice and another chamber may be provided on a rod side thereof. Alternatively, on the rod side of the hydraulic supply-discharge tube 19, only the orifice may be provided while not providing the chamber. In any case, the effect of the orifice 20 and the chamber 21 remains the same.

[0057] By adopting the above-described configuration, the rolling mill according to the first embodiment of the present invention can control mill vibration without having to reduce the orifice diameter excessively.

INDUSTRIAL APPLICABILITY

[0058] The present invention is suitable for a rolling mill, or more specifically, to a device to control vibration of a hot rolling mill which occurs in the course of rolling with the rolling mill.

REFERENCE SINGS LIST

[0059]

11 housing
12 work roll
13 work roll chock
14 backup roll
15 backup roll chock
16 roll gap controlling means

17 hydraulic cylinder
18 housing liner
19 hydraulic supply-discharge pipe
20 orifice
21 chamber
22 hydraulic pressure source

Claims

1. A rolling mill **characterized in that** the rolling mill comprises:

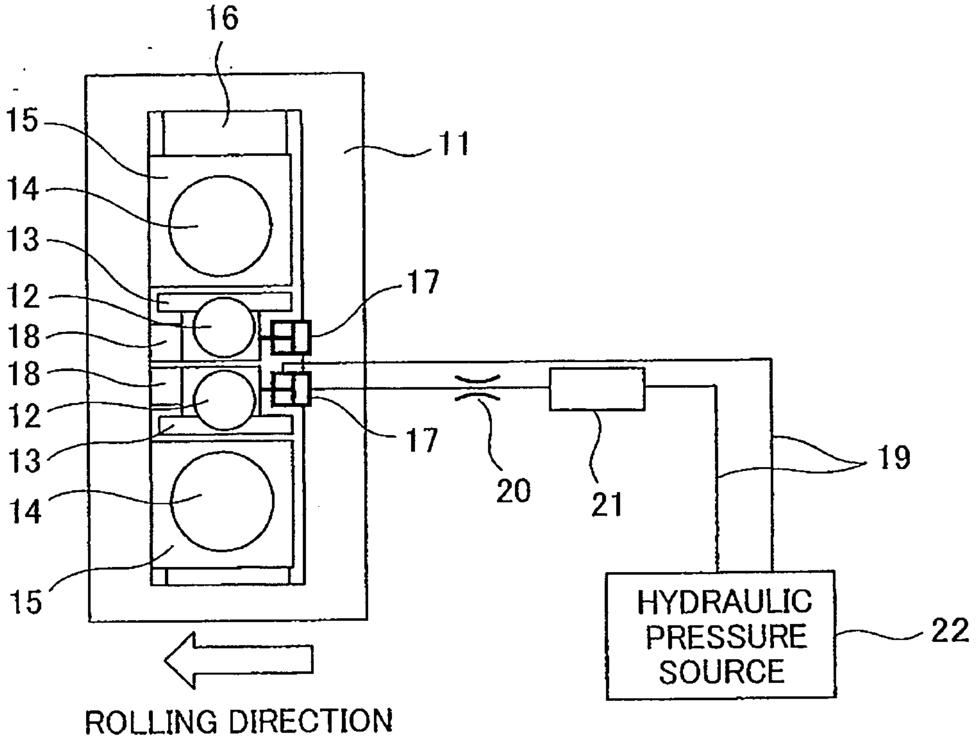
a housing;
a pair of upper and lower work roll chocks supported by the housing;
a pair of upper and lower work rolls opposed to each other and pivotally supported by the pair of upper and lower work roll chocks, respectively;
roll gap controlling means for applying a predetermined pressure to the work rolls;
a pair of upper and lower first supporting means provided to the housing at positions on one side in a rolling direction for supporting the pair of upper and lower work roll chocks; and
a pair of upper and lower second supporting means provided to the housing at positions on another side in the rolling direction for supporting the pair of upper and lower work roll chocks, wherein
the first supporting means is used as hydraulic pressing means and is made capable of pressing the pair of upper and lower work roll chocks in a horizontal direction,
a flow contracting unit and an expanding unit are provided to a portion of a hydraulic supply-discharge pipe on a head side of the hydraulic pressing means, while disposing the flow contracting unit closer to the hydraulic pressing means than the expanding unit, and
an inside diameter of the flow contracting unit is set equal to or above ϕ2.5 mm and in a size of from 15% to 85% relative to an inside diameter of the hydraulic supply-discharge pipe.

2. The rolling mill according to claim 1, **characterized in that** a volume of the expanding unit is set in a range from 7% to 180% relative to a volume of the hydraulic pressing means.

3. The rolling mill according to claim 1 or 2, **characterized in that** a distance between the flow contracting unit at the hydraulic supply-discharge pipe and the hydraulic pressing means is set equal to or below 7 m.

4. The rolling mill according to any one of claims 1 to 3, **characterized in that** a distance between the expanding unit and the flow contracting unit at the hydraulic supply-discharge pipe is set equal to or below 3.5 m.

Fig. 1(a)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HYDRAULIC PRESSURE SOURCE
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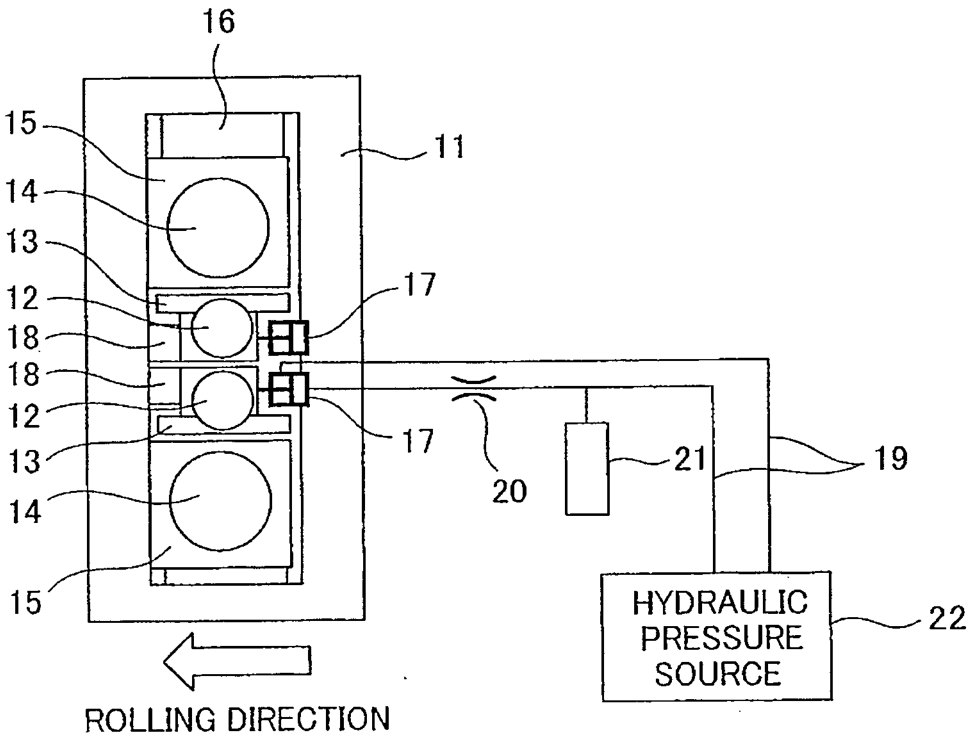
ROLLING DIRECTION

Fig. 1(b)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HYDRAULIC PRESSURE SOURCE
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ROLLING DIRECTION

Fig.2(a)

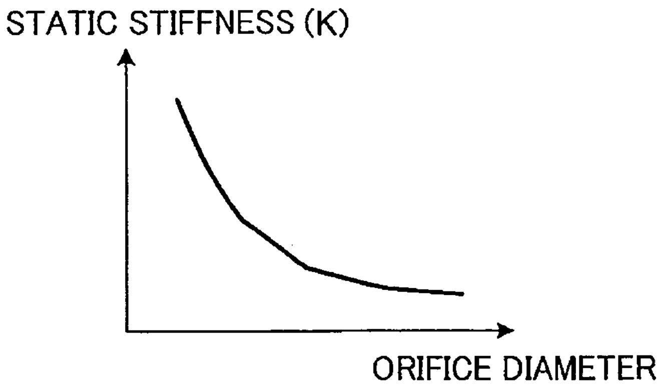

Fig.2(b)

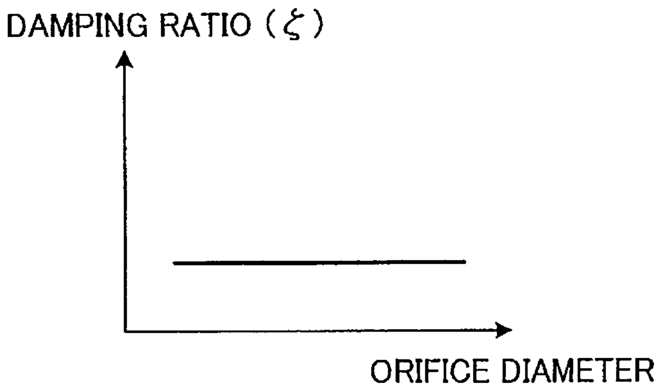

Fig.2(c)

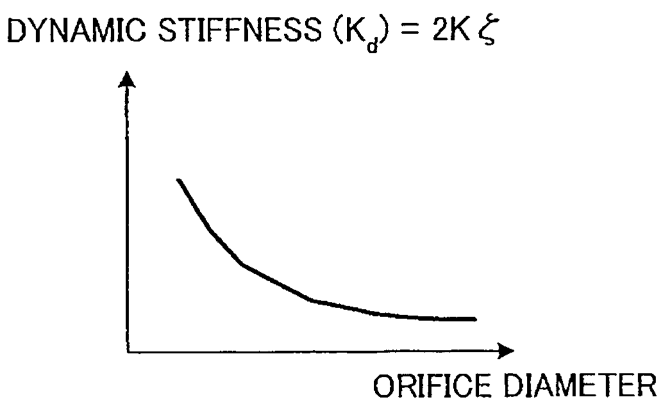

Fig.3(a)

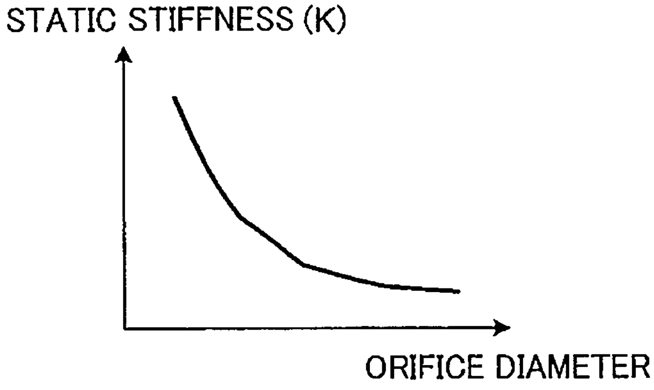

Fig.3(b)

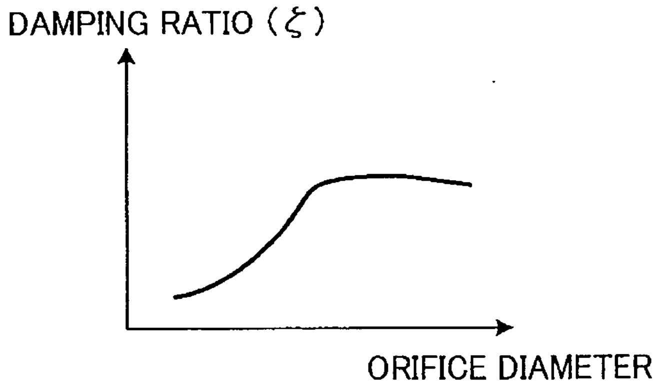

Fig.3(c)

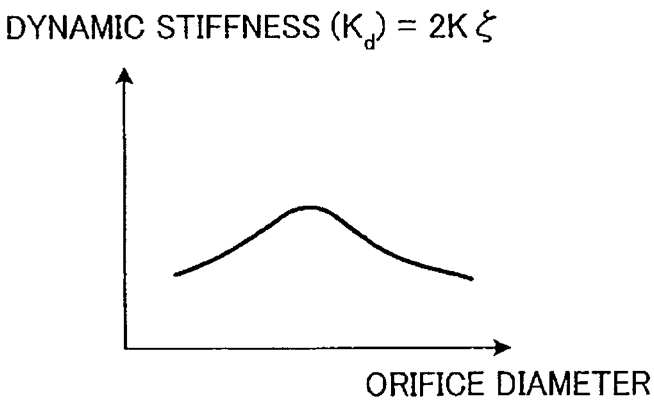

Fig. 4

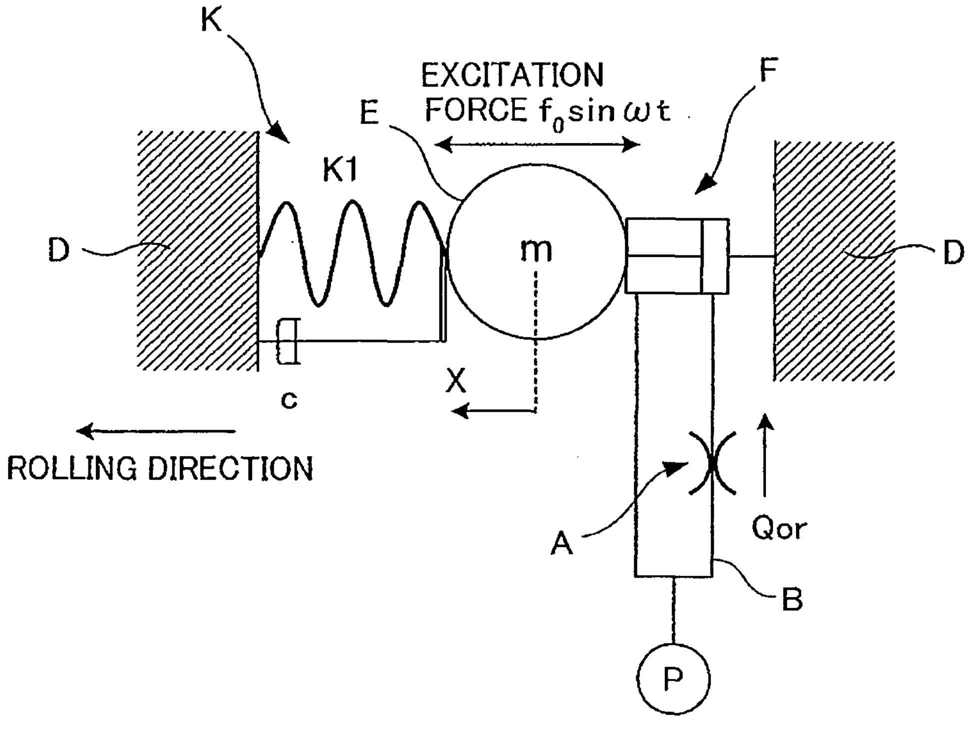
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Fig. 5

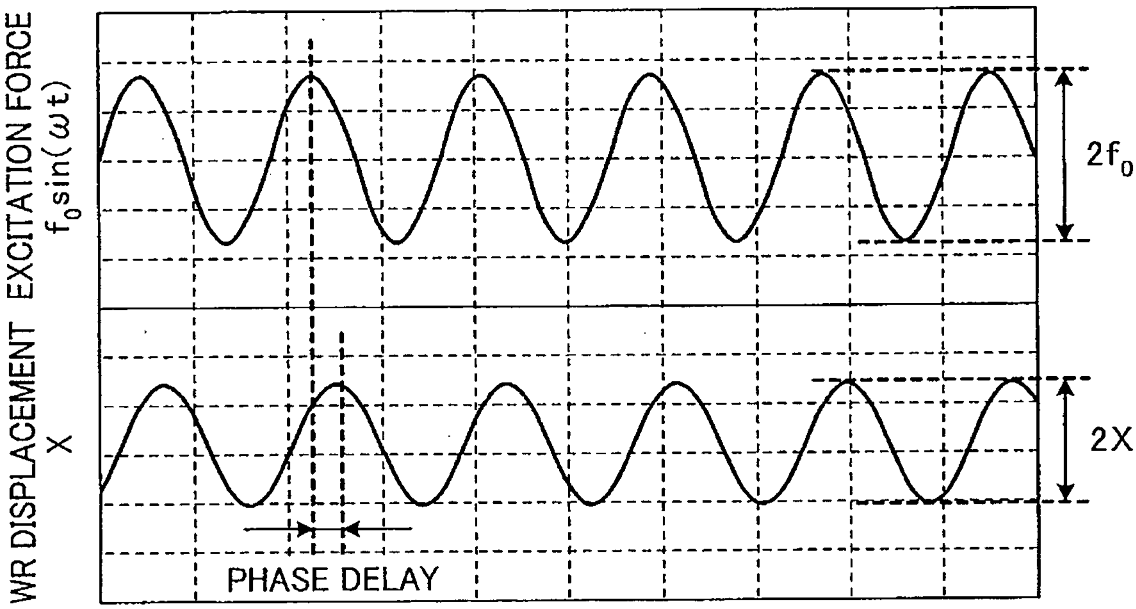

EXCITATION FORCE
$f_0 \sin(\omega t)$
WR DISPLACEMENT
X
$2f_0$
2X
PHASE DELAY

Fig.6

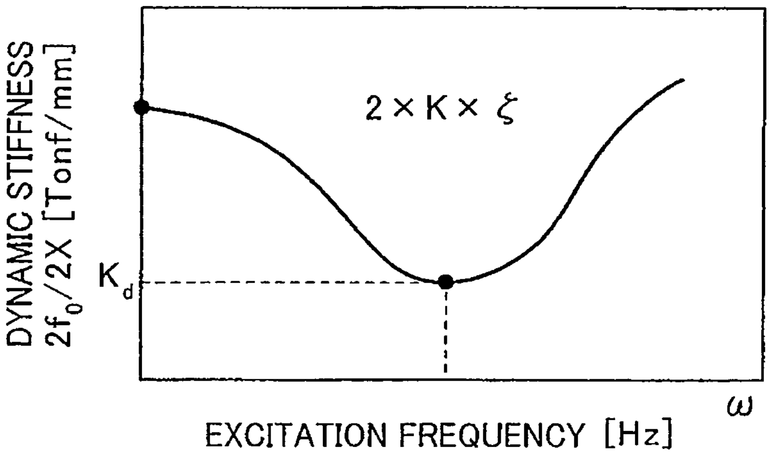

DYNAMIC STIFFNESS
2f0/2X [Tonf/mm]
2 × K × ζ
Kd
ω
EXCITATION FREQUENCY [Hz]

Fig. 7(a)

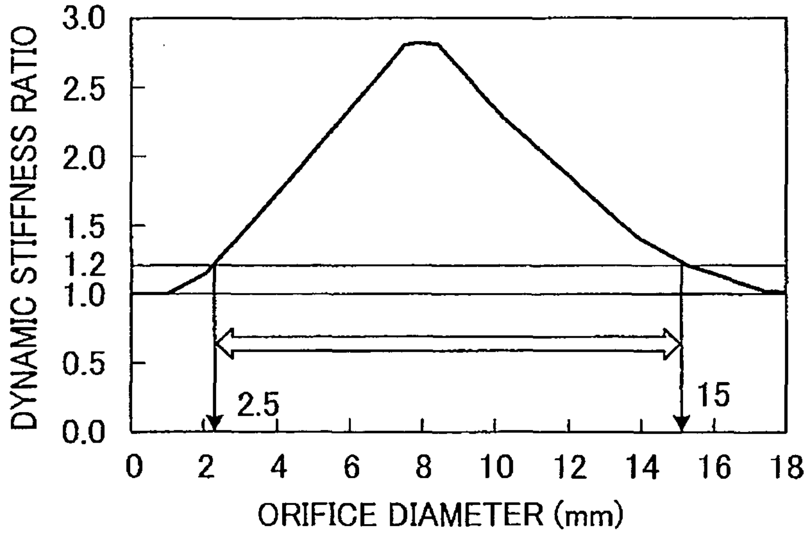

DYNAMIC STIFFNESS RATIO
3.0
2.5
2.0
1.5
1.2
1.0
0.5
0.0
2.5
15
0
2
4
6
8
10
12
14
16
18
ORIFICE DIAMETER (mm)

Fig. 7(b)

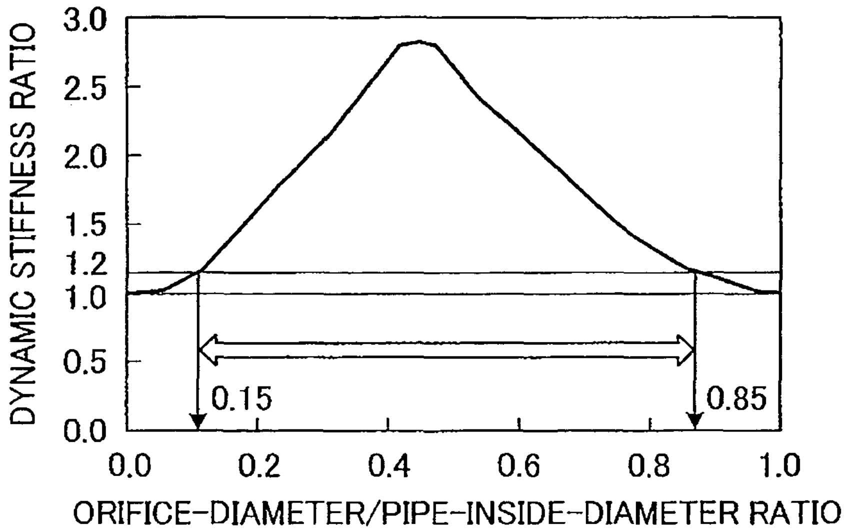

DYNAMIC STIFFNESS RATIO
3.0
2.5
2.0
1.5
1.2
1.0
0.5
0.0
0.15
0.85
0.0
0.2
0.4
0.6
0.8
1.0
ORIFICE-DIAMETER/PIPE-INSIDE-DIAMETER RATIO

Fig.8(a)

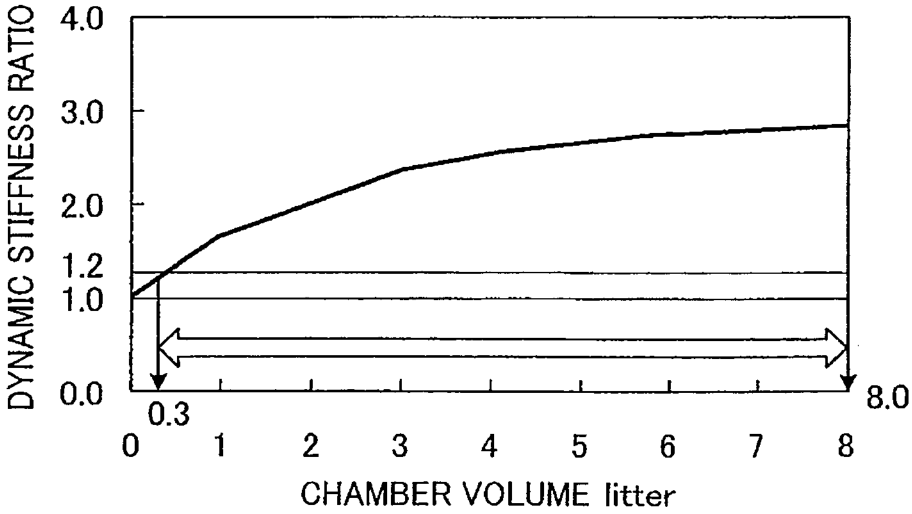

DYNAMIC STIFFNESS RATIO
4.0
3.0
2.0
1.2
1.0
0.0
0.3
8.0
0
1
2
3
4
5
6
7
8
CHAMBER VOLUME litter

Fig.8(b)

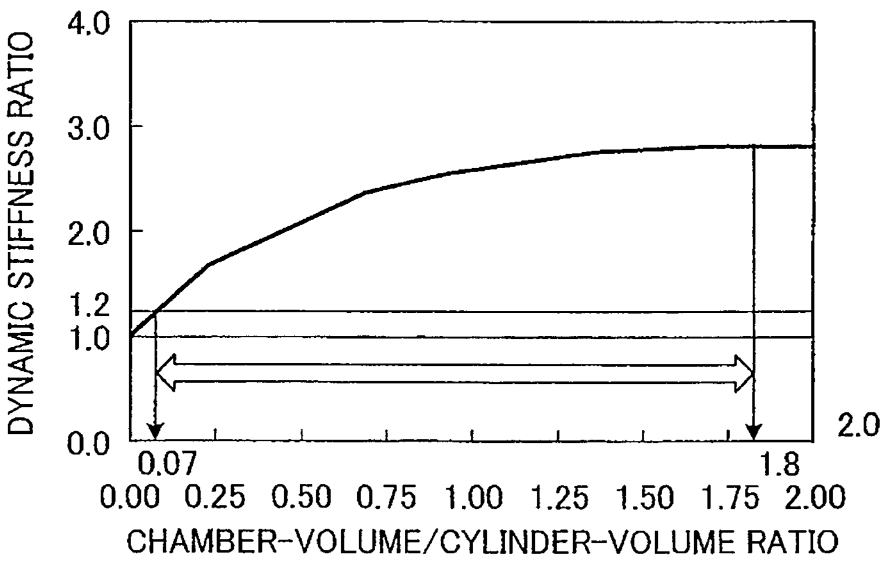

DYNAMIC STIFFNESS RATIO
4.0
3.0
2.0
1.2
1.0
0.0
0.07
1.8
2.0
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
CHAMBER-VOLUME/CYLINDER-VOLUME RATIO

Fig. 9

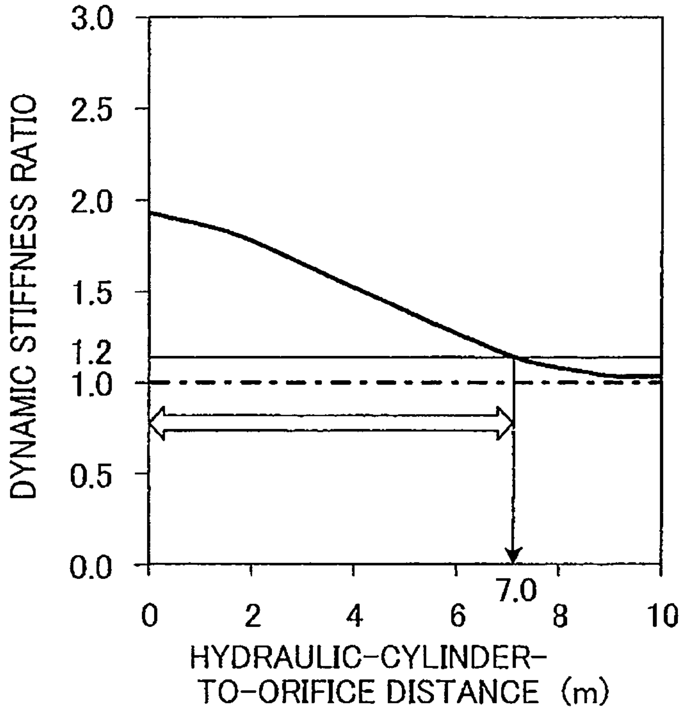
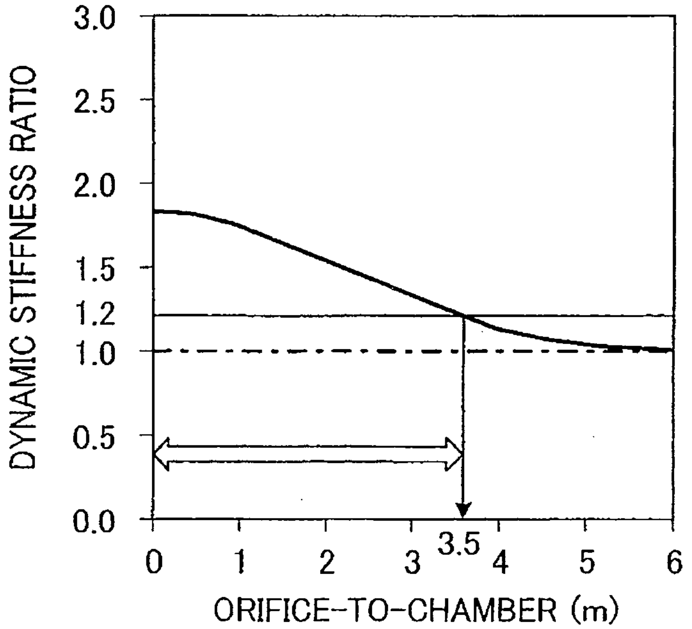

3.0
2.5
2.0
1.5
1.2
1.0
0.5
0.0
DYNAMIC STIFFNESS RATIO
7.0
0
2
4
6
8
10
HYDRAULIC-CYLINDER-TO-ORIFICE DISTANCE (m)

Fig. 10


3.0
2.5
2.0
1.5
1.2
1.0
0.5
0.0
DYNAMIC STIFFNESS RATIO
3.5
0
1
2
3
4
5
6
ORIFICE-TO-CHAMBER (m)

INTERNATIONAL SEARCH REPORT

International application No. PCT/JP2014/077052

A. CLASSIFICATION OF SUBJECT MATTER
B21B31/02(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
B21B31/02, B21B13/02

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Jitsuyo Shinan Koho	1922-1996	Jitsuyo Shinan Toroku Koho	1996-2014
Kokai Jitsuyo Shinan Koho	1971-2014	Toroku Jitsuyo Shinan Koho	1994-2014

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	JP 2001-113308 A (Mitsubishi Heavy Industries, Ltd.), 24 April 2001 (24.04.2001), paragraphs [0025] to [0030], [0046] to [0052]; fig. 11 to 12 & US 6510721 B1 & EP 1120172 A1 & WO 2001/012353 A1 & CN 1320064 A & KR 10-0429729 B1	1-4
A	JP 2-151310 A (Sumitomo Light Metal Industries, Ltd.), 11 June 1990 (11.06.1990), page 2, lower right column, line 7 to page 4, upper right column, line 17; fig. 1 to 4 (Family: none)	1-4

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

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"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
22 December 2014 (22.12.14)	13 January 2015 (13.01.15)
Name and mailing address of the ISA/ Japan Patent Office	Authorized officer
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 2009)

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Patent documents cited in the description

- JP 2001113308 A **[0004]**