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(54) A laundry washer with improved dynamic equilibration system

(57) A laundry washer for domestic use having a drum with a dynamic equilibration system comprising a plurality of annular hollow bodies fixedly connected to the drum and a plurality of movable bodies of cylindrical shaped disposed for free displacement within the hollow bodies. A fluid is contained in the hollow bodies and adapted to distribute itself along the circumference of the hollow bodies. The fluid has a viscosity between 40 and 130 mPa.s. The cylindrical movable bodies have a diameter between 10 and 18 mm, a length which is less than the internal width of said hollow bodies by an amount of between 1 and 2 mm, and an outer diameter less than the internal height of said hollow bodies by an amount not exceeding 0.75 mm.

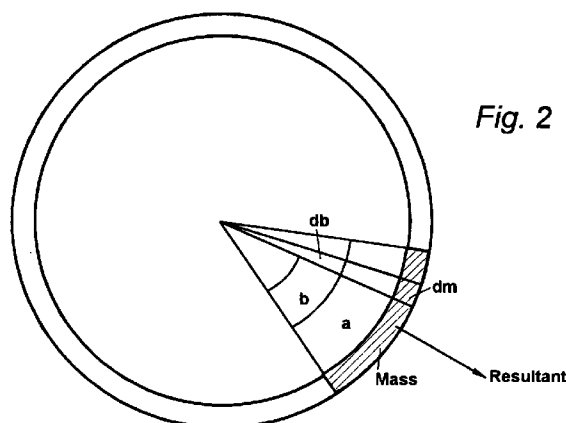


Fig. 2

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Description

The invention relates to a laundry washer, particularly of the domestic type, provided with an improved system for the dynamic equilibration of the drum.

Although the present invention is specifically directed to a laundry washer of the front-loading type, and the following description makes reference, for the sake of simplicity, to a laundry washer of this type, it is also advantageously applicable to other types of laundry washers, for instance to a laundry washer of the top-loading type, or to machines with a drum mounted for rotation about a vertical axis.

It is known that in laundry washers, particularly domestic ones, one of the major problems confronting the designer regards the balancing of the drum containing the laundry charge during the spin-drying phases for the rapid extraction of the rinsing water; this problem is particularly noted when the laundry charge is imbalanced, resulting in undesirable and often unacceptable oscillation of the drum, which is transmitted to the tub and from there to all parts of the machine.

This problem is generally known and shall therefore not be further discussed. Among the various solutions to this problem there has been proposed a solution of the dynamic type, which consists in having the centrifugal force of the imbalanced charge opposed by an imbalance of the same magnitude, but of opposite vector. Solutions of this kind are described for instance in French Patent no. 1,213,067 (Swiss priority 8.11.1957), and Italian Patent no. 1,108,367 (Japanese priority 12.1.1978). These documents describe dynamic equilibration systems applicable to the drums of laundry washers, particularly for domestic use, consisting of annular hollow bodies connected to the drum and containing equilibration masses freely displaceable within the hollow bodies so as to assume positions in which they oppose the imbalance of the drum caused by the imbalance of the charge. This solution, although basically feasible, has not found general acceptance for various reasons, among which in particular the following:

a) in the case of a laundry charge sufficiently balanced in and by itself, and if the hollow bodies do not also contain a liquid acting to entrain or to brake the equilibration masses, i.e. when the displacements thereof are not dampened, they tend to distribute along the circumference of the annular hollow bodies at substantially indifferent, although mutually opposed positions for thus maintaining the equilibration of the entire system; it is particularly because these positions are substantially indifferent, that irregularities of movement, tolerances in the mounting of the annular body with the masses therein and the deformation of the annular body, and vibrations caused by even the slightest imbalance produce an immediate equilibration response of the masses, causing them to drift to a new position of equilibrium.

In the course of this equilibration process, the masses may collide with one another with considerable violence, resulting in substantial noise generation.

Once the new state of equilibrium has been established, it is not maintained for any length of time, because any new imbalance of the charge will again start the just described imbalance/equilibration cycle.

This results in an extremely unsatisfactory situation, because on the one hand the vibrations are not definitely suppressed, and on the other hand the machine periodically emits considerable noise.

b) The construction of the drums has to be modified to improve their strength for permitting them to carry the annular hollow bodies, the latter acting on the drum's walls with considerable centrifugal forces caused by the weight of the equilibration masses.

c) The annular hollow bodies employed at present are made of rather light materials and therefore deformable to a certain degree under the centrifugal forces exerted by the equilibration masses; this deformation of the hollow bodies results in that the equilibration masses do not assume their optimum positions for outbalancing the imbalance of the charge, with the final result that the equilibration effect is imperfect or incomplete.

Also known from PCT Patent WO 93/23687 in the name of BALANCE TECHNOLOGY LIMITED PARTNERSHIP LA PLAIDERIE TRUST CO, LTD. is a laundry washing machine provided with a plurality of hollow bodies associated to a hermetically closed tub rotating about an axis, the hollow bodies containing a plurality of equilibration masses displaceable within the hollow bodies, the masses being of different dimensions in accordance with those of the various hollow bodies. This solution is not of any interest, it having been found that the employ of floating equilibration masses of different dimensions does not convey any appreciable advantage to the equilibration process, but above all, because in conventional domestic laundry washers the tub is stationary rather than rotatable, the imbalance to be corrected being that of the perforate drum and not that of the tub. In addition, the employ of hollow bodies of different and determined dimensions with the associated equilibration masses would involve considerable complications in the construction of the machine.

Known from Italian patent application no. PN94A000005 of the present applicant is an equilibration system comprising hollow bodies with equilibration masses contained therein, in which the hollow bodies are provided with projecting lobes effective to facilitate the aggregation of the masses at preferential positions for optimum equilibration. This solution, however, requires a considerable construction effort with particular specifications for any laundering assembly, to result in definite improvements without, however, offering a decisive solution.

Also known from European patent application no. 0,607,678, filed by the Whirlpool Corporation, is an equilibration system substantially consisting of a plurality of annular hollow bodies filled with respective liquid equilibration masses. In the course of extensive experimentation this solution has been found to be scarcely effective for reducing an existing imbalance due to the fluid nature of the equilibration masses which tend to distribute themselves along the entire circumference of the hollow bodies, resulting in a reduction of the equilibration effect.

Otherwise all of the solutions employing floating equilibration masses of solid nature suffer from the common shortcoming that the liquid in which the floating masses have to be immersed for avoiding excessive noise generation and for assisting the displacement of the masses by its drag effect, is normally an oil- and/or silicone-based liquid and thus subject to considerable variations of its viscosity as a function of temperature.

Since the starting temperature of a laundry washer may be as low as 5 °C, and may rise as high as 85 °C during operation, it is obvious that this high variability of the temperature favours a corresponding variability of the viscosity characteristics of these liquids, to thereby reduce their capability of adequately dragging the floating masses along as well as their capability of dampening the noise resulting from the repetitive collisions of the masses among themselves in the starting phase of the drum and above all in the deceleration and stopping phases.

In any case, the characteristics of the liquid are not the only factor affecting the general performance and noise generation, because it is also the number of the floating masses, the number of the annular hollow bodies containing them, the circumferential angle of the annular hollow bodies occupied by the respective floating masses, and the dimensional play between the floating masses and the respective hollow bodies in two directions orthogonal to the direction of displacement of the floating masses, which are elements of profound influence, and which have therefore to be carefully determined.

It is therefore an object of the present invention to improve the construction of a laundry washer drum by providing it with elements acting to retain the masses in determined positions as long as an existing imbalance remains below a predetermined value, and to permit the masses to freely position themselves in response to an imbalance above this limit value; it is another object of the invention to define one or more combinations of values to be assumed by the most important variables, such as the viscosity of the liquid, the configuration and number of the annular hollow bodies, their positioning relative to one another, the shape of the floating equilibration masses and their dimensions relative to the internal cavity of the hollow bodies wherein they are displaceably retained. The system is devised with the characteristics defined in the appended claims and effective to eliminate the described shortcomings without the necessity of major structural modifications and alterations in the basic construction of the machines.

The invention will be more clearly understood from the following description, given by way of non-limitative example.

In the following description, the term "ring" will be used as an equivalent to the term "annular hollow body", and the term "roller" as an equivalent to the term "cylinder", without the use of one or the other term compromising the lucidity of the presentation, given the context in which these terms are used, as will presently be obvious to those skilled in the art.

As explained above, the invention defines the optimum values of a number of fundamental variables determining the performance of the dynamic equilibration system.

For determining these values, an extensive test program has been planned and executed, and contemporaneously therewith, a mathematical model has been developed for in the first place accompanying the planning and execution of the tests, and in the second place, for the interpretation of the various results.

With a view to fully explain the tenor of the invention, the description contains all of the considerations, evaluations, elaborations and mathematic models, experiments, results and their interpretations substantially in the form of a logic nexus between cause, effect and final significance, so that one skilled in the art will not encounter any difficulty to understanding the operative environment in which the invention and its scope of application matured.

Behaviour of an oscillating system

When an oscillating system is excited by a periodic force, it oscillates with the frequency of the applied force. The amplitude of the oscillation and the phase angle (with respect to the applied force) depend on the frequency as shown by way of example in the diagrams of figs. 1A and 1B.

It is in particular noted that on exceeding the resonance frequency, the oscillating system is retarded to oscillate in phase opposition to the exciting force.

The principle on which the equilibration system using hollow rings is based is that of resonant systems: in response to an imbalancing force (internal of the drum of the laundry washer), any mass capable of free displacement within the single ring assumes the counterposition with respect thereto as soon as the value of the resonance frequency of the vibration is exceeded.

On the base of this principle, use is made of concentric rings containing floatingly displaceable objects.

The occupation angle of the equilibration masses

The number of objects to be contained in the rings is dependent upon their dimensions and weight, since the centrifugal force generated by the objects in combination is, for a given mass and speed of rotation, all the smaller as the circumferential angle occupied by the objects in the ring becomes greater. And the smaller the resultant centrifugal force thus developed, the smaller is the quantity of the concentrated mass capable of being equilibrated.

With reference to the diagrammatic presentation of fig. 2, the centrifugal force developed by an element of the mass "dm" in the direction of the resultant is given as:

$$dF_{\text{centrifuga}} = dm \cdot \cos(a/2 - b) \cdot w^2 \cdot r$$

wherein:

- w = angular speed
- r = radius of the ring.

From this follows that, for a given radius "r" and angular speed "w", the centrifugal force increases as the angle "a/2 - b" diminishes. The mass element can be expressed in terms of its "occupation angle":

$$dm = m \cdot db/a$$

By inserting this expression into the preceding one and integrating, the resultant is obtained as:

$$\text{resultant} = \int_0^a \left(\frac{m}{a}\right) \cdot \cos(a/2 - b) \cdot w^2 \cdot r \cdot db = 2 \cdot m \cdot w^2 \cdot r \cdot \frac{\sin(a/2)}{a}$$

Assuming m = 1, it is seen that with the increase of the angle occupied by the mass its "effectiveness", i.e. the centrifugal force developed, diminishes from 1 to 0 (for the occupation of the entire ring), as illustrated in the diagram of fig. 3.

The employ of only a single equilibration body is inconceivable, however, for two principal reasons:

- * it would be extremely difficult to cause the single body to travel around the ring;
- * when the mass causing the imbalance to be compensated is smaller than that of the equilibration body, the latter would drastically lose its effectiveness by being unable to distribute its mass along the ring.

The reduction of the angle occupied by the floating bodies could on the other hand be achieved by placing the bodies in a number of rings disposed in a concentric arrangement with their opposing surfaces in mutual contact as diagrammatically shown in fig. 4.

The viscous fluid

In addition to containing the equilibration bodies, the rings are filled with a fluid characterized by a specific viscosity value.

The provision of integrating the fluid in the annular equilibration system is principally due to the following considerations:

- * Without the fluid, and at low rotary speeds of the drum, the equilibration bodies would roll along in the lowermost part of the ring due to their weight, resulting in excessive noise.
- * On increasing the rotary speed of the drum, the equilibration bodies have to be accelerated by friction so as to enable them to assume their positions; it is therefore necessary that the fluid be sufficiently viscous.
- * In the deceleration phase at the end of the spin-drying step, or on inversion of the rotation of the drum in the course of the laundering operation, the drop of the equilibration bodies towards the lowermost part of the ring should be slowed down to thereby effectively dampen the metallic noise resulting from the bodies colliding with one another.

The viscosity as a function of temperature

The viscosity "η" of a fluid is the measure of its internal friction. It is dependent on the value of temperature in

accordance with an exponential function expressed as

$$\eta(T) = A \cdot e^{-B/T} \text{ (fig. 5).}$$

5 wherein

T = the temperature value

A and B = two parameters principally dependent on the type of the liquid.

10 A rise in temperature thus results in a lowering of the intrinsic viscosity of the fluid. The viscosity of a fluid is important because the viscous friction force " F_η " is dependent thereon according to the law:

$$F_\eta = \eta \cdot S \cdot \frac{dv}{dz}$$

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wherein

η = the coefficient of the dynamic viscosity of the fluid as defined above,

20 S = the contact surface area between a body and the liquid,

$\frac{dv}{dz}$ = the speed gradient of the fluid over the distance between the wall of the ring and the contact surface of the floating body.

In view of this law it is noted that for a given speed gradient of the floating body relative to the fluid, the friction force F_η can be adjusted by altering either the dimension of the contact surface " S " of the body or the viscosity coefficient " η " of the fluid.

Since the coefficient of viscosity is dependent on the temperature value, it follows that also the viscous friction force is dependent on the temperature.

Since the rings are attached to the drum of the laundry washer in direct contact with the washing liquid, they are subjected to temperature variations between about +5 °C and +80 °C. This calls for a fluid whose viscosity range at these working temperatures permits specific requirements to be satisfied, namely, reduced noise generation and the capability of dragging the floating bodies along throughout the ring.

In consideration of these arguments, one might be inclined to select a high viscosity liquid. This selection would be put into question, however, by the following aspects:

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* When the viscosity is extremely high, the floating bodies will right away be dragged along at the rotational speed of the laundering step, to effectively act as a single body. This may cause overheating of the motor due to the elevated output demand for the upward movement of the bodies.

* When the viscosity is extremely high, the positioning of the bodies in opposition to the imbalance of the charge proceeds extremely slowly, with the danger that the forces of the imbalance are further increased.

* It is known that in the case of high-viscosity liquids, the viscosity undergoes excessive variations in response to temperature variations. While, for example, the viscosity of water varies by the factor 3 in the temperature range between 5 °C and 85 °C, that of a typical lubricating oil may vary by the factor 50. In view of the requirement that the behaviour of the system should not vary to any higher degree within this temperature range, it becomes necessary to direct one's attention to less viscous fluids.

For these reasons, a number of liquids of different viscosities has been prepared in a series of experiments. This has been accomplished by mixing various liquids of known viscosity. The viscosity of the mixture can be estimated in the following manner:

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The viscosity of mixtures

For a single liquid, the effect of its viscosity is described with the aid of the following model:

55 * A body of the liquid is confined between two walls.

* One of the walls is fixed, while the other, disposed at a distance " d " from the fixed wall, moves at a speed " v ".

* The liquid disposed close to the walls assumes the speed of the respective wall: speed " v " adjacent the moving wall, and " 0 " adjacent the fixed wall (Fig. 6).

In this figure, the horizontal arrows indicate the speed vector of the liquid at varying distances from the fixed wall. For this model there applies the formula for the force required for moving the movable wall, the surface area of which is assumed as being equal to 1:

$$F = \eta * \frac{(v-0)}{d}$$

In analogy, in a first approximation a mixture (m1 + m2) of two liquids in the amounts of m1 and m2, respectively, may be considered as the two liquids superimposed rather than mixed with one another (Fig. 7).

At the interface between the two liquids it is assumed that liquid 1 is dragged along by liquid 2. The liquids and surfaces, or interface, respectively, are subjected to the action of entraining or braking drag forces (Fig. 8A).

Since all of the liquid layers basically move at uniform velocity, i.e. without any accelerations, the forces F1 and F2 have to be equal.

$$F1 = F2$$

For liquid 1 there applies:

$$F1 = \eta_1 * \frac{v1}{d1}$$

By mathematically elaborating these relations, and expanding them to the case of the two contemporaneously employed liquids, one obtains a formula for the "mean" viscosity:

$$\eta = \frac{\eta_1 * \eta_2 * (d1 + d2)}{\eta_1 * d2 + \eta_2 * d1}$$

For purposes of the model the thickness (d1, d2) of the layers may be replaced by the quantities (m1, m2) of the two liquids.

The resultant reasoning justifies the universal application of the formula representing the mixture with two superimposed layers. Instead of two layers it is thus for example possible to describe a four-layer model as represented in fig. 8B.

The formulation of the respective equations for the four layers analogous to the formulation with respect to two layers will finally result in the same formula valid for the two layers. This is also evident from fig. 8B, in which the overall velocity profile is similar to that of the two-layer model composed of liquids 1 and 2, with the flow speeds of any two liquids at their interface being of course equal.

The upper limit of this breakdown is an infinite number of layers, corresponding to a substantially homogenous mixture.

The viscosity of a mixture composed of

x% of a liquid of 400 cP and
100 - x% of a liquid of 100 cP

is represented by way of example in the diagram of fig. 9.

Calculation of the forces acting on the rollers in the equilibration ring:

For optimum results, the equilibration mass has been determined as consisting of solid bodies in the form of rollers or cylinders, because the mass thereof is maximized transversely of its direction of movement.

A roller located in an annular cavity is subjected to three forces, as diagrammatically shown in fig. 10:

- * gravity : fg
- * thrust of the fluid : fp
- * viscous drag of the fluid : fa

These forces are calculated as shown in the following.

The force of gravity:

This force acts on the displacement of the roller solely with the component:

$$f_{ga} = r^2 \pi x (\rho_r - \rho_f) g \sin(w)$$

wherein: r = radius of the roller
 x = length of the roller
 ρ_r = density of the roller
 ρ_f = density of the fluid
 g = gravity acceleration
 w = angle between the roller and the vertical

Thrust force exerted by the fluid:

When a ring is partially filled with a liquid and uniformly rotated in one direction, the liquid will be dragged along due to the friction between the walls of the ring and the liquid, resulting in a level offset at the two end faces of the liquid body as diagrammatically shown in fig. 11A.

As viewed from the rotating ring, the liquid appears to flow there in in response to the pressure difference "dp".

This flow "V" of the liquid is calculated as follows: The formula for the flow of a liquid in a circular tube is found in relevant literature under the designation "Hagen-Poiseuille formula". In the case of a ring of rectangular section this formula is not, however, directly applicable. By making use of the concept of equivalent radius, however, a rectangular section may be treated approximately in the same manner as a circular section.

$$\text{equivalent radius} = \frac{2 \cdot \text{sectional area}}{\text{circumference}}$$

That is:

$$R = \frac{a \cdot b}{a + b}$$

wherein: a = width of the ring's section
 b = height of ring's section

This equivalent radius is included in the Hagen-Poiseuille formula:

$$dp = \frac{8 \eta V}{R^4 \pi}$$

wherein "V" is the flow of the fluid expressed in m³/sec to be calculated from the rotary speed "v" of the ring and its sectional area:

$$V_{\text{provisional}} = \left(\frac{v \cdot D_{\text{ring}} \cdot \pi}{60} \right) \cdot a \cdot b \quad (\text{without correction for the flow through gaps}).$$

"dp" is thus the pressure exerted by the liquid on a partition inserted into the ring so as to impede the rotation of the fluid (Fig. 11B).

In the case of an equilibration ring containing one roller therein, the pressure "dp" would act on the roller with the force:

$$f_p = dp \cdot (x \cdot y)$$

wherein x = length of the roller
 y = diameter of the roller

When inserting the flow "V" into the formula, account has to be taken in this case of the fact that the roller does not completely obstruct the flow of the oil within the ring, as there is always a small fraction of the flow passing through gaps between the roller and the walls of the ring:

$$V = V_{\text{provisional}} - V_a - V_b$$

wherein

V_a = liquid flow through gap between side "a" and side "x" of the roller

V_b = liquid flow through gaps between the two sides "b" and side "y" of the roller.

This leaves a formula for the calculation of V_a and V_b to be found.

Friction force in gaps between the roller and the ring:

With reference to fig. 12, there applies the known friction law :

$$f = \eta \cdot S \cdot \frac{du}{dz}$$

In the present case, the friction force " f_a " for the lateral gap of the roller is expressed as:

$$f_a = \eta \cdot 2 \cdot S_{\text{pastiglia}} \cdot \frac{u}{sb}$$

The expression for the flow through the lateral gap is:

$$V_b = b \cdot sb \cdot \frac{u}{2}$$

This is because relative to the roller the liquid flows at the speed ($u/2$) for reasons of symmetry. For the upper gap the drag force exerted by the wall of the ring prevails, because the flow speed of the liquid relative to the roller is approximately " u ":

$$V_a = a \cdot sa \cdot u$$

" V_a " and " V_b " are used for correcting the flow "V" in the calculation of the thrust force, and the corrected value "V" is used for the calculation of " dp " and thus " fp ".

Calculation of the displacement angle:

With reference to fig. 10, in the state of equilibrium:

$$f_{ga} = f_p + f_a,$$

accordingly, when the ring is put in rotation, the single roller positions itself according to the angle:

$$w = \arcsin \frac{f_p + f_a}{r^2 \cdot \pi \cdot x \cdot (\rho_r - \rho_l) \cdot g}$$

The shape factor

From the theoretical considerations discussed in the preceding paragraph with respect to a single roller, it is recognized that the resultant force acting thereon can be expressed as follows:

$$F = \eta^* \text{ (a factor comprising physical and geometric parameters)}$$

Since this factor is not dependent on temperature, it is convenient to select it to have a sufficiently great magnitude so as to obtain a friction force corresponding to that of low-viscosity liquids. In view of the fact that for fluids of this type the dependence of the viscosity on temperature is of a smaller magnitude, the friction force will also depend on temperature to a lesser degree.

The shape factor can be partially acted upon when it is possible to accept production tolerances particularly with regard to the thickness of the rings and the dimensions of the floating bodies.

As regards the shape of an individual floating body, it is preferred to select bodies of circular geometry for their capability of rolling along within the rings. A specific selection is to be made between spherical and cylindrical bodies.

Referring once more to the relation between the viscous friction force "F" and the area of the contact surface "S", it is preferred to select bodies of cylindrical shape, the contact surface of which is at all points equidistant from the wall of the ring. In the present case, this implies rollers moving within a ring of rectangular cross-section (as already assumed in the preceding analyses).

The shape factor becomes all the more important with the diminishing width of the gap between the roller and the wall of the ring.

Synopsis and definition of the operating conditions:

The preceding considerations and initial tests have allowed to circumscribe the operating conditions of the system:

- * at the laundering speed, the rollers should be distributed in a uniform or random manner so as not to create problems in terms of noise generation by rolling or sliding about in the lower part of the ring, or in terms of non-uniform load on the motor caused by the effort of raising a "block" of the rollers.
- * At the spin-drying speed, the rollers should be rapidly positioned, without the creation of noise at the end of the spin-drying operation.

Diagrammatically shown in figs. 13A, 13B and 14 is the behaviour of the rollers (rolling, distribution, locking) at the laundering and spin-drying speeds, respectively, as dependent on the total friction acting on the rollers, and the respective positioning of the rollers. From the above considerations, it follows that the "friction" is determined by:

- the viscosity of the fluid,
- the generic shape factor (a function of the distance between the roller and the wall of the ring, and
- the dimensions of the rollers.

Experimentation

By varying the type of fluid, the shape factor and the dimensions of the rollers in an experimental ring, it was endeavoured to find the optimum combination for obtaining the desired functional behaviour. These experiments had to be carried out over the full range of possible operating temperatures.

With the aim of verifying the above formulated theoretical and mathematical assumptions, a series of experiments had to be devised and executed for examining the actual behaviour to arrive at the optimum solution.

The prototypes

For the experimentation explained above, two different prototypes of machines were used, each of which comprised a conventional laundering assembly mounted in a rigid test frame in which the assembly was suspended from above by two springs, and supported from below by two friction shock absorbers.

Each laundering assembly had an electric AC collector motor operating at a transmission ratio of 1 : 10 by using a transmission belt of the type commercially known as MEGADYNE EL1200 J6 between the motor and drum pulleys.

Fixedly attached to the front and rear of each drum were respective numbers of concentric rings for containing a fluid and a number of rollers devised to counterbalance any imbalance occurring in the spin-drying phase.

The rings are of two-part construction comprising a base and a cover secured thereto by screws. The cover is made of polymetamethylacrylate, and the base, of a polycarbonate with calcium carbonate as a filler material.

A first prototype was equipped with four concentric rings (cf. fig. 15A), and the other one, with three concentric rings (cf. fig. 15B) on each side, the rings of the respective groups being disposed closely adjacent one another.

The rollers contained in each ring were of a ferrous material.

The first experiments of the series were carried out with the following configurations:

PROTOTYPE 1

132 rollers of the same type were disposed in both groups of the rings. Each roller had a diameter of 13 mm, a length of 9 mm, and a weight of 10 gr. The gaps between each roller and the walls of the respective rings were 0.5 mm in the plane of the roller's diameter, and 0.75 mm in the plane of its length.

PROTOTYPE 2

57 identical rollers were disposed in the rings of the front and rear groups. Each roller had a diameter of 17 mm, a length of 10 mm, and a weight of 17.5 gr. The gaps between each roller and the walls of the ring in the planes of diameter and length were 0.5 mm.

The fluids

Since the range of temperatures to which the fluid is subjected during an operating cycle of a laundry washer extends, as has already been pointed out, from about +5 °C to +80 °C, it is necessary to investigate the properties of suitable fluids in terms of their viscosity within this temperature range. The respective values were determined in the first place by using a BROOKFIELD RVT viscosimeter.

The different temperatures of the fluids between +5 °C and +80 °C were obtained by placing the respective fluids in a refrigerator and a microwave oven, respectively.

The thus analyzed fluids are commercially known under the designations:

1. SPINESSO 10
2. SPINESSO 22
3. MASCHERPA 1579A
4. K (Produkt der Firma Henkel Chimica S.p.A.)
5. KLUEBER FLUID 9R 100
6. STRUCTOVIS FHD
7. H (Produkt der Firma Henkel Chimica S.p.A.)

The fluids "H" and "K" are detergents for industrial use, the remainder being oils.

For each of the fluids, a diagram has been established showing its viscosity over temperature as depicted in figs. 16 to 22. As these diagrams show, the range of viscosity variation varies from one fluid to another, with significant relationships between temperature and viscosity ranges. In particular, among the fluids analyzed, the one designated STRUCTOVIS FHD was found to have the highest viscosity of 1600 mPa.s at +5 °C and decreasing to 62.4 mPa.s at +80°C, while the fluid designated SPINESSO 10 had the lowest viscosity within this temperature range, decreasing from 74 mPa.s at +5 °C to 22 mPa.s at 80 °C (mPa.s=millipascal.sec.)

For permitting the obtained values to be compared to one another, they are summarized in table A, showing the lowest and highest operating temperatures and an assumed ambient temperature of 20 °C.

TABLE A

Fluid	5 °C	20 °C	80 °C
SPINESSO 10	74	54	20
MASCHERPA 1579A	77.6	54	22
SPINESSO 22	140	84	24
K (Henkel)	440	224	56
KLUEBER 9R 100	950	400	44
STRUCTOVIS FHD	1600	580	62,4
H (Henkel)	3360	2000	126
	mPa.s		

A first analysis of the obtained data will result in certain conclusions. Inasmuch as a fluid having a constant viscosity

at varying temperatures (a "Newton" fluid) does not come under consideration, the parameter governing the selection should be the smallest viscosity variation within the given temperature range. Under this aspect, the choice would fall upon the fluid SPINESSO 10, with its viscosity variation range from 74 to 20 = 54.

This characteristic alone is not, of course, sufficient for justifying the selection of this fluid, inasmuch as it remains to be determined whether the respective viscosity values are suitable for permitting the floating bodies, i.e. rollers, to assume their correct positions.

A second significant parameter for the selection of a suitable fluid is the relationship between the maximum and minimum viscosity values; the smaller this relationship, the greater is obviously the probability that the respective fluid is found suitable.

The relationship between the maximum and minimum viscosity values allows only a relative evaluation, however, because it does not reflect the absolute viscosity value; in any case, however, it is an indication of quality to be taken into consideration in the investigation of specific mixtures.

Among the fluids thus analyzed, the one presenting the best such relationship is that designated MASCHERPA 1579A with a value of 77,6 : 22, i.e. about 3.5.

It is known that one of the fluids having a modest maximum-minimum viscosity ratio within the temperature range under consideration is distilled water with a viscosity variation ranges between about 16 mPa.s and 7 mPa.s in the given temperature range.

In view of its low cost, and irrespective of the fact that its absolute viscosity values are too low for its direct use in an equilibration system under consideration, distilled water may still play an important role for the adjustment of the viscosity of some of the fluids named above.

While on the one hand water is not really suitable for being mixed with oils, it is highly useful for being mixed with detergents.

The tests

A series of tests was executed with the prototypes described above, with the individual variables being selectively varied.

A first group of tests was directed to observe, at an ambient temperature of about 25 °C, the distribution of the rollers at the laundering speed of 55 rpm of the drum, and at different viscosity values of the fluid contained in the rings to simulate different operating temperatures.

The fluid employed in these tests was a mixture of a detergent commercially available under the trademark DASH, and distilled water.

In both of the prototypes employed in the first group of tests, the rings were formed with crenellated internal wall surfaces, the tests resulting in the finding that the internal wall surfaces of the rings should preferably be smooth.

In the case of the prototype with three concentric rings on each wall, a good distribution of the floating bodies or rollers was observed under these conditions with viscosity values of the liquid between 70 mPa.s and 200 mPa.s. It is important to observe that the rollers, as long as they were grouped in the lower parts of the ring at low rotary speeds of the drum, interfered with one another so as to slide on the wall of the ring rather than rolling therealong, resulting in the generation of noise. The highest noise level was attained, however, when the rotation of the drum was stopped, and at a viscosity of the fluid between about 70 and 150 mPa.s, this noise resulting prevalently from the rollers colliding with one another.

In the case of the prototype employing four rings on each side, it was observed that the floating bodies, i.e. rollers were already dragged along in compact groups at rather low viscosity values of about 20 mPa.s, with evident difficulties on the part of the motor in maintaining a uniform rotary speed, and excessive oscillation of the drum-tub assembly.

The laundering speed of an electronically controlled laundry washer of a type actually in production attains the value of 55 rpm, followed by a reduction of this value to 40 rpm for a short period of about 23 seconds, as illustrated by the diagram of fig. 23. On observing the criteria stated above under these cyclic speed variation conditions, it was noted that the undesirable noise generation effect was even more pronounced.

For this reason, and also in view of the fact that the rollers in the configuration as employed are subjected to excessive centrifugal forces which may cause plastic deformation on the roller-supporting surfaces, further tests were executed with a modified shape factor in the four-ring system employing rollers of smaller dimensions.

The original rollers were thus replaced by similarly shaped rollers having a diameter of 13.5 mm, resulting in a gap between the rollers and the ring walls of 0.75 mm in the plane of the length of the rollers (=9 mm), and of 1.0 mm in the plane corresponding to the diameter of the rollers, as diagrammatically illustrated in fig. 24.

The groups of rollers extended over an angle of about 140° circumferentially of the respective rings, and represented a mass of about 830 gr, resulting in a total mass of 1 660 gr.

Under these conditions, and in view of the increased gravity forces, it was expected that the rollers would be carried around each of the rings as a compact group in the presence of a fluid of a higher viscosity than formerly established.

In a series of tests carried out with this configuration and employing fluid of different viscosities, it was found that

the best distribution of the rollers is obtained in the presence of a fluid having a viscosity between about 30 mPa.s and somewhat above 200 mPa.s.

At the same time it was noted that the noise generation was considerably reduced as compared to the three-ring system, although it was still considered excessive in the case of fluids having viscosities below 40 mPa.s.

After termination of this series of tests, further investigations were carried out for determining the behaviour of the system at high rotary speeds with a concentrated imbalance mass positioned within the drum.

This subsequent series of tests was directed to the operation of the four-ring prototype employing rollers with the diameter of 13,5 mm, and at rotary speeds of between 400 rpm and 700 rpm. These speeds were attained by departing from 55 rpm, followed by a short period of about 14 seconds at 85 rpm. The operation at 85 rpm also served for determining the "counterbalance mass", a coefficient resultant from a mathematical function derived from the rotary speed of the drum. In the case of an imbalance mass of 1.5 kg positioned centrally of the drum, this coefficient resulted in a maximum value of 4680. The tests thus executed demonstrated a satisfactory behaviour with rapid positioning of the rollers in opposition to the imbalance mass, as long as the viscosity of the fluid did not exceed about 120 to 130 mPa.s. At higher viscosity values the fluid exerts excessive drag on the rollers, causing them to maintain their spaced positions around their respective rings.

The results of these tests are set forth below in table B:

Table B

Viscosity (mPa.s)	Effect on the distribution of the rollers	Effect on noise generation
130	no correct positioning for spin-drying	low noise level
70 - 130	rollers dragged along as a group at 55 rpm	attenuated noise
40 - 70	rollers correctly distributed at 55 rpm	pronounced noise in the stopping phase
40	rollers not correctly distributed at 55 rpm (remain in lowermost position)	pronounced noise at 55 rpm and in the stopping phase

Of significance are the different configurations assumed by the rollers in response to the magnitude of the imbalance mass.

In the case of a mass of 1.5 kg centrally positioned within the drum, all of the rollers tend to position themselves in opposition thereto, while in the case of an imbalance mass of 1 kg, some of the rollers position themselves in phase therewith so as not to exceed the mass to be compensated.

As regards the counterbalance coefficient attained at the speed of 85 rpm, the maximum value lay just below 3000.

For the purpose of comparison, the counterbalance coefficient has been determined for a production-type laundry washer with the same tub-drum assembly, the same motor and the same belt drive transmission ratio. The thus ascertained values were always about 4680.

Table C, below, shows the results of the tests giving the most representative results with reference to the roller configuration diagrammatically shown in Fig. 24:

Table C

Test	rings	ring walls	viscosity	rollers	Ø	length	speed	results
1	4	crenellated	20-200	132	13 mm	9 mm	55 rpm	overall behaviour bad
2	3	crenellated	20-200	57	17	10	55	overall behaviour bad
3	4	smooth	20-80	132	13	9	55	rollers dragged in a group
4	3	smooth	70-200	57	17	10	55	excessive noise in stop phase
5	4	smooth	20-200	132	13	9	55-40	excessive noise in general
6	4	smooth	20-200	132	13,5	9	55-40	good distribution above 30 mPa.s modest noise above 40 mPa.s
7	4	smooth	20-200	132	13,5	9	400-700	correct positioning at spin-drying below 130 mPa.s

In all of the tests thus performed, the best results were obtained with an equilibration system with the following characteristics:

Rings and Rollers:

- 5 two discs of four rings each disposed at front and back of the drum;
 walls of rings smooth, measuring 10.5 x 14.5 mm, with radii $r_1 = 210$ mm, $r_2 = 194$ mm, $r_3 = 177$ mm, $r_4 = 166$ mm;
 132 rollers for each disc, disposed in the individual rings so as to occupy an angle of less than about 145° ;
 diameter of rollers 13,5 mm, length 9 mm, weight about 10 gr.

10 Viscosity of the fluid (fig. 25)

Satisfactory overall results and improvements over prior art have been obtained within given tolerances with respect to:

- 15 * the dimensions of the rollers,
 * the gaps between rollers and respective rings,
 * the angular positioning of the rollers and
 * the viscosity of the fluid

20 as defined in the attached claims.

Although the invention has been described with reference to preferred embodiments by way of example and in a known terminology, it is not restricted thereto, but rather encompasses various modifications conceivable by one skilled in the art.

- 25 These obvious modifications conceivable by one skilled in the art are thus considered as lying within the spirit and purview of the present invention as defined in the attached claims.

Claims

- 30 1. A laundry washer, particularly for domestic use, comprising an outer housing, a laundering tub, a perforate drum of cylindrical shape mounted in said tub for rotation about its axis during the laundering and spin-drying phases, said drum being provided with a dynamic equilibration system comprising a plurality of annular hollow bodies of closed rectangular cross-sectional configuration and fixedly connected to said drum with their axes coinciding with the axis of rotation of said drum, a plurality of movable bodies of cylindrical shape disposed for free displacement within said hollow bodies with their respective axes parallel to said axis of rotation of said drum, and a fluid having lubricating properties, particularly oil, contained in said hollow bodies and adapted to distribute itself all along the circumference of said hollow bodies and between said cylindrical bodies, characterized in that:
- 35 * said lubricant fluid has a viscosity between 40 and 130 mPa.s,
 * said cylindrical bodies, or rollers, have a diameter of between 10 and 18 mm,
 40 * the length of said rollers is less than the internal width of said hollow bodies by an amount of between 1 and 2 mm, and
 * the outer diameter of said rollers is less than the internal height of said hollow bodies by an amount not exceeding 0.75 mm.
- 45 2. A laundry washer according to claim 1, characterized in that said annular hollow bodies are of a diameter between 10 and 18 mm.
3. A laundry washer according to claim 1 or 2, characterized in that the length of said rollers is between 6 and 12 mm.
- 50 4. A laundry washer according to any of the preceding claims, characterized in that four of said annular hollow bodies are located at each end face of said drum.
5. A laundry washer according to claim 4, characterized in that in each pair of mutually adjacent hollow bodies the external cylindrical surface of the inner hollow body is completely in contact with the internal cylindrical surface of the outer hollow body.
- 55 6. A laundry washer according to claim 5, characterized in that said annular hollow bodies are disposed in symmetric arrangement on the respective end walls of said drum with respect to a plane orthogonal to the axis of rotation of said drum.

7. A laundry washer according to any of the preceding claims, characterized in that in each of said annular hollow bodies said cylindrical bodies or rollers contained therein are disposed over an angle of no more than 160° and centered upon the intersection of the axis of rotation of said drum with a plane substantially defined by the respective hollow body.

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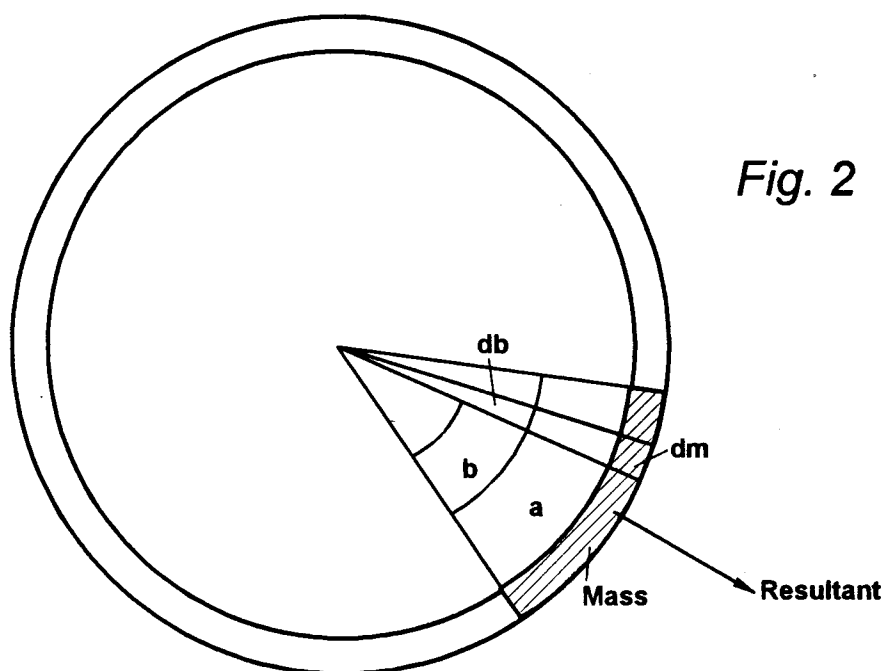
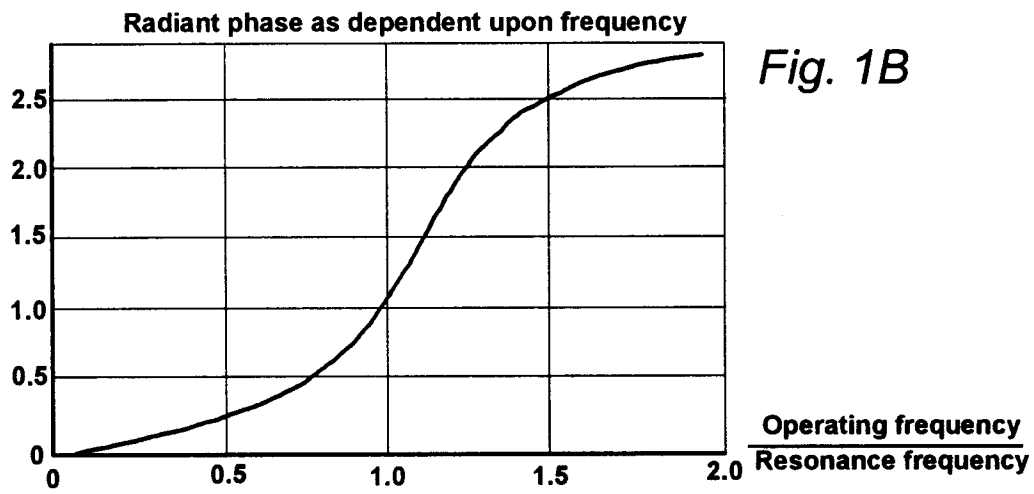
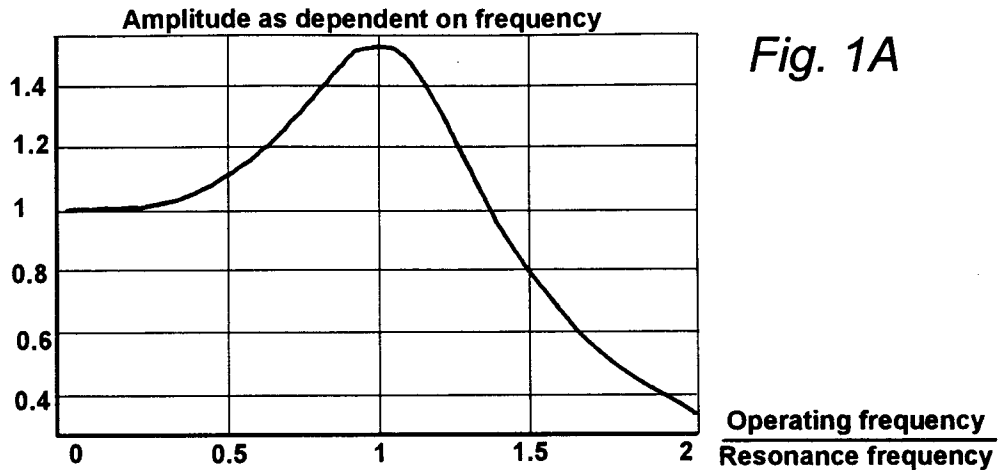
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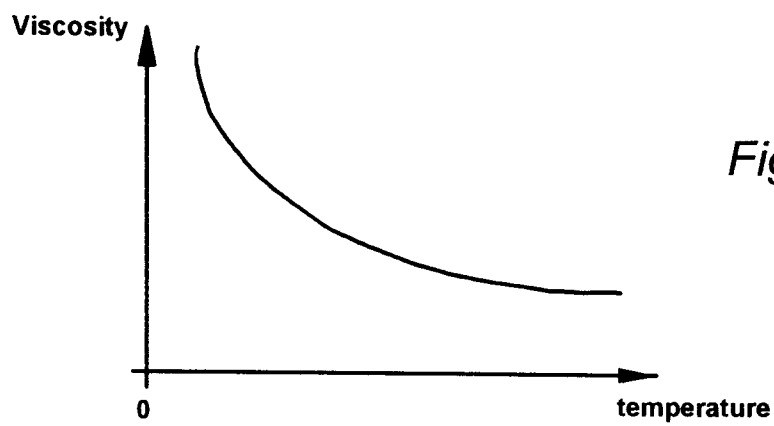
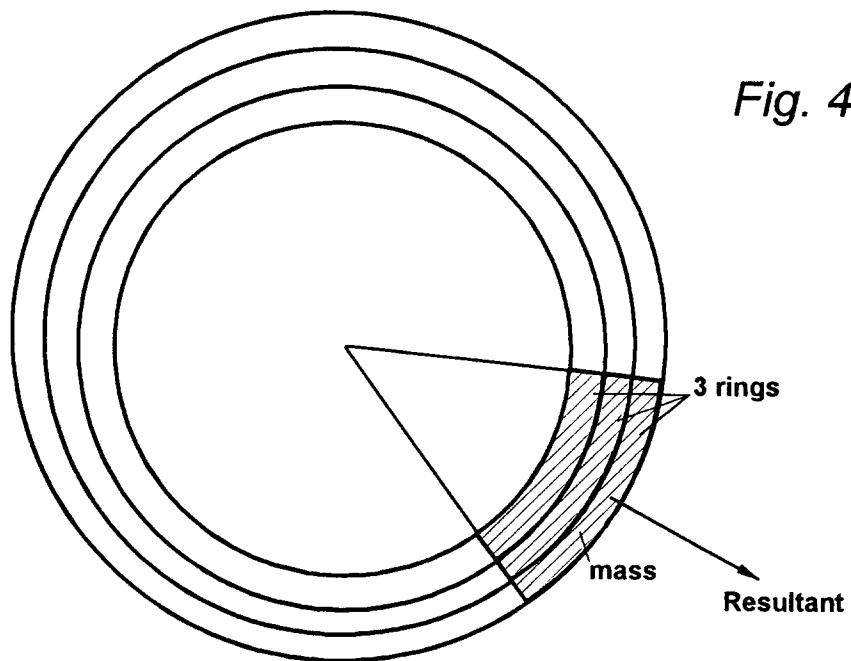
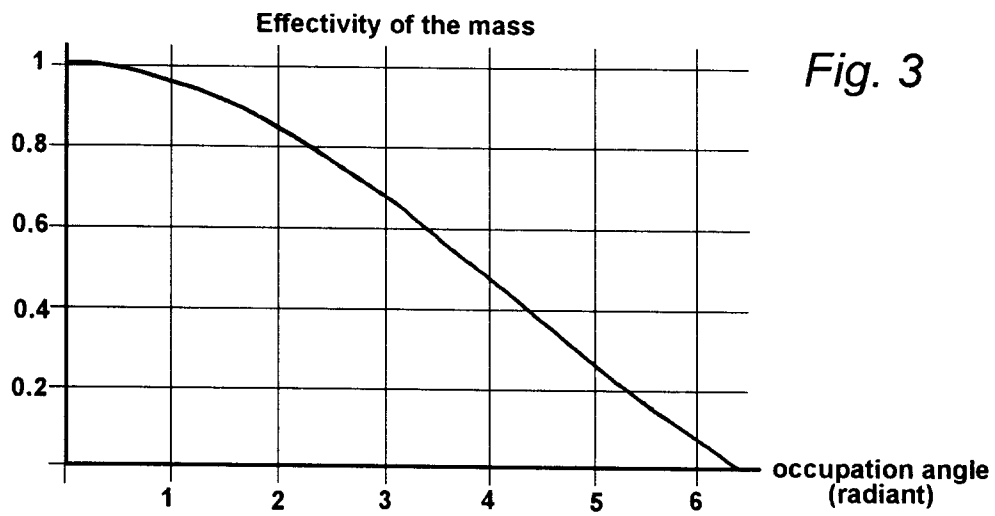
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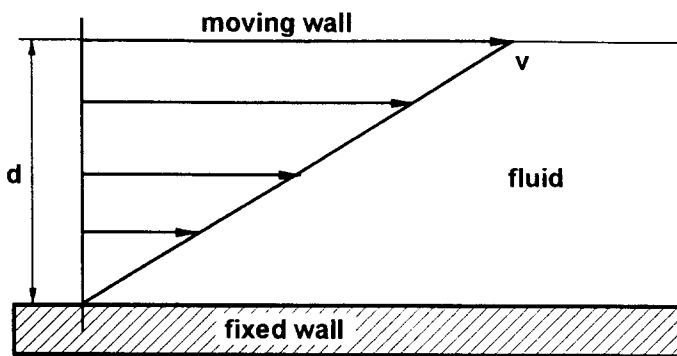


Fig. 6

Fig. 7

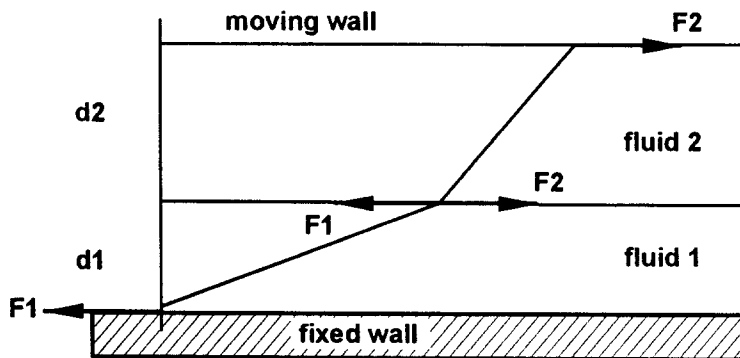
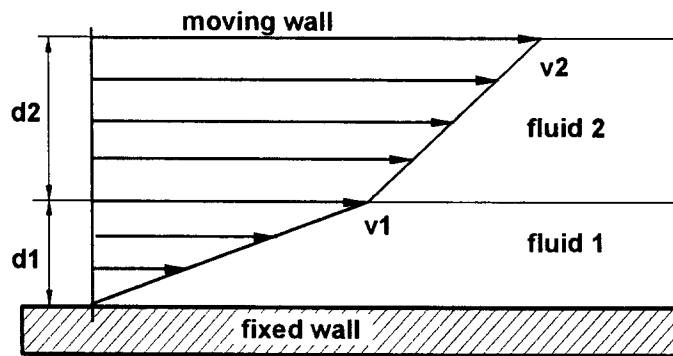
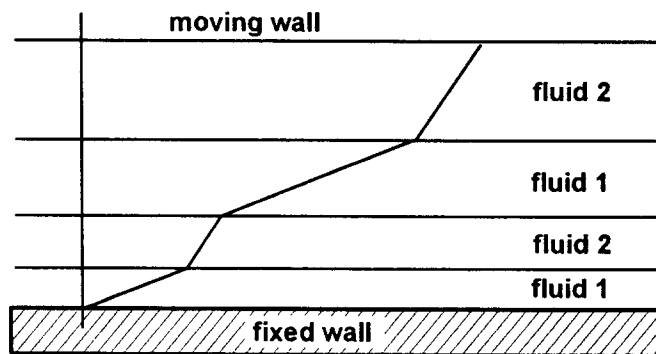


Fig. 8A

Fig. 8B



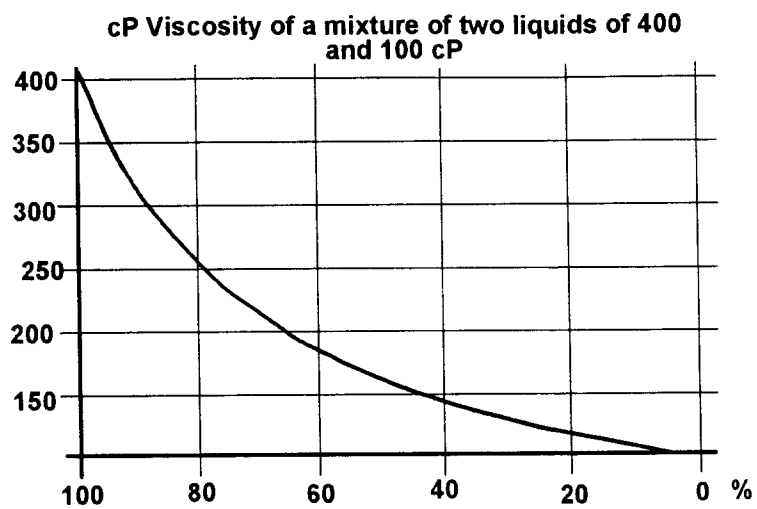


Fig. 9

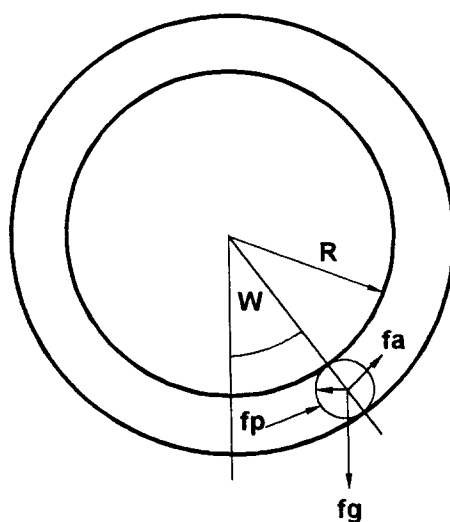


Fig. 10

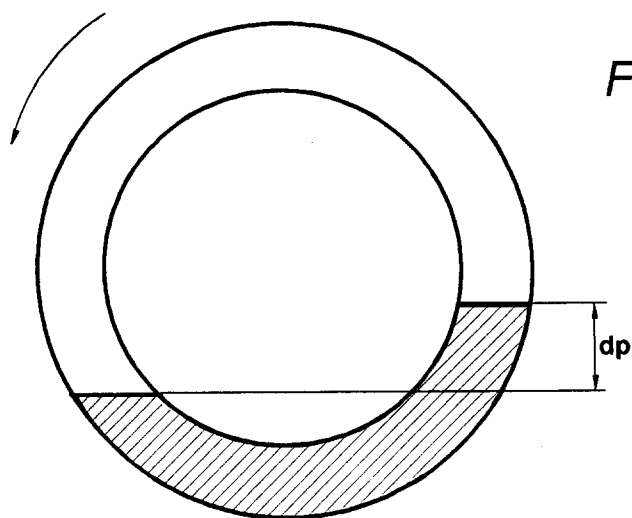


Fig. 11A

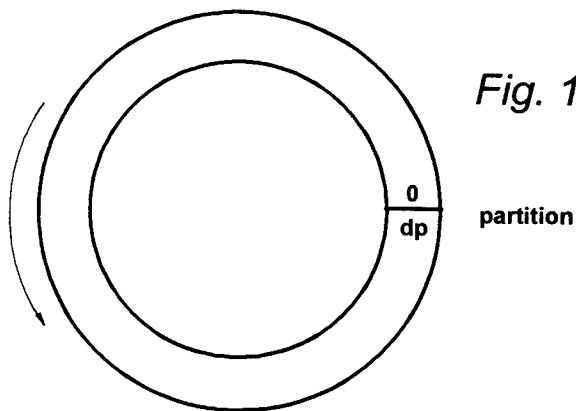


Fig. 11B

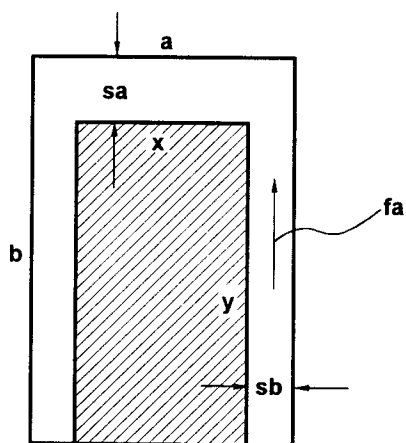


Fig. 12

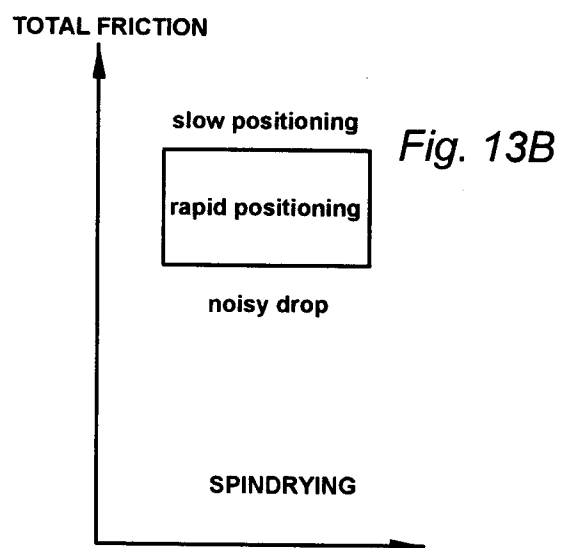
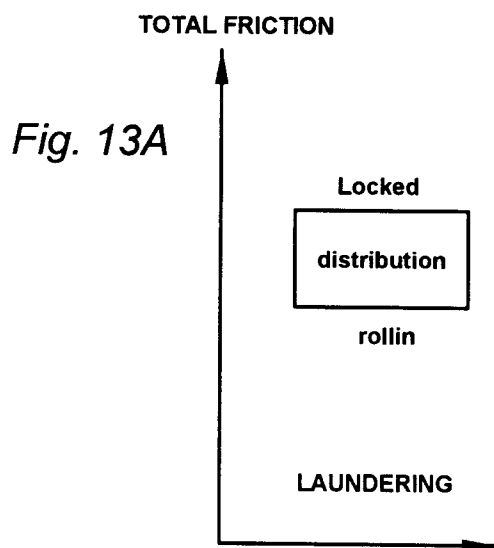


Fig. 14

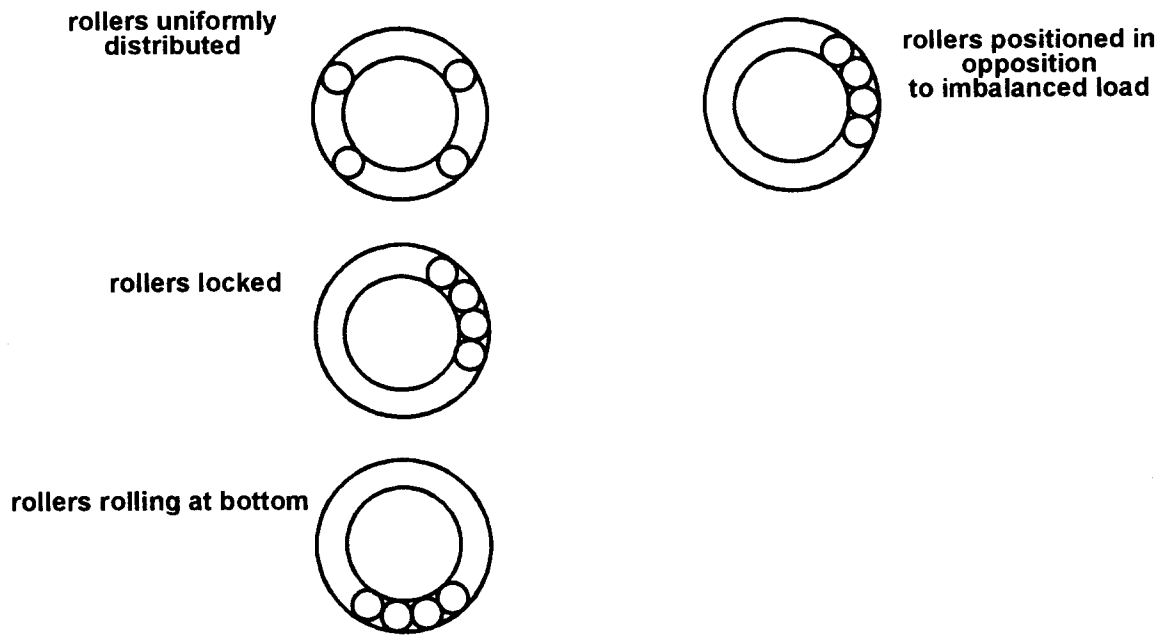


Fig. 15A

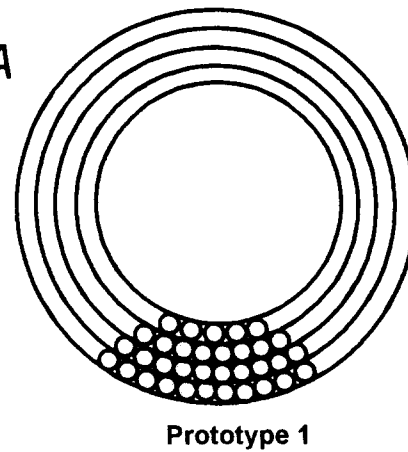


Fig. 15B

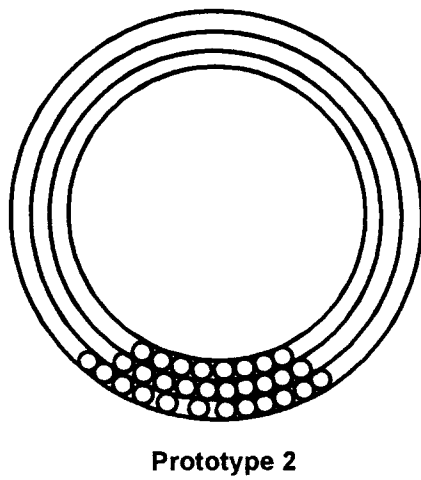


Fig. 16

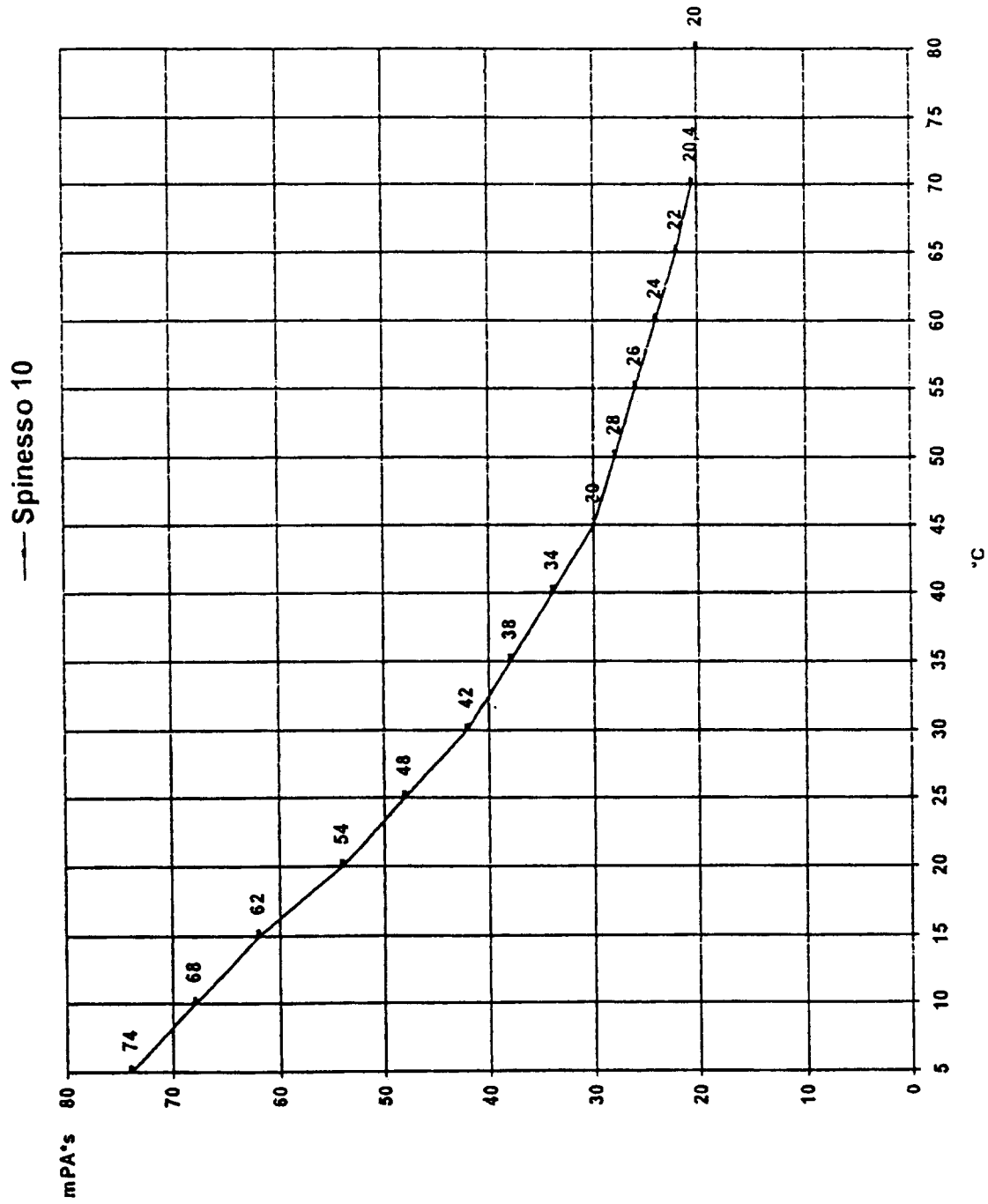


Fig. 17

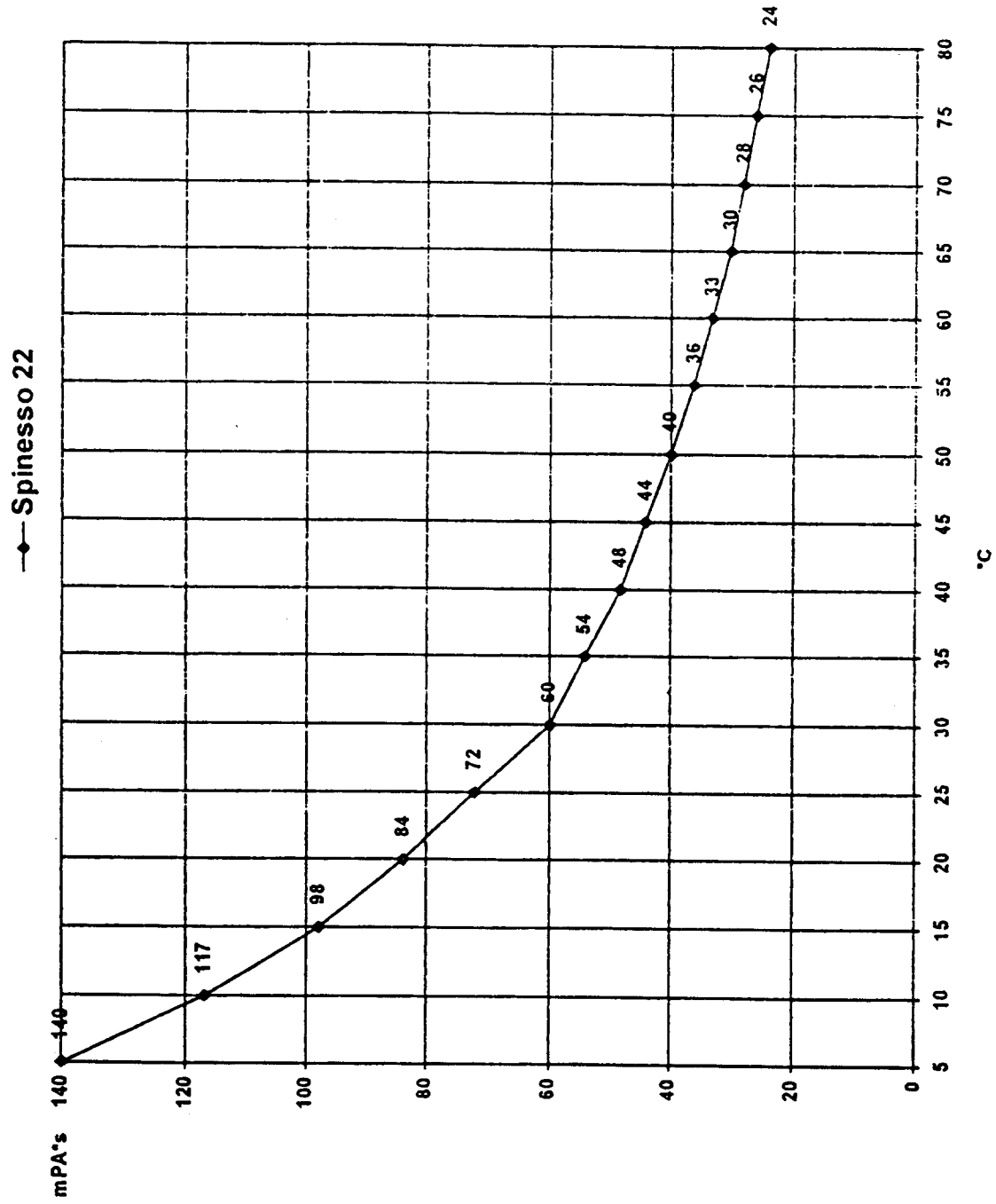


Fig. 18

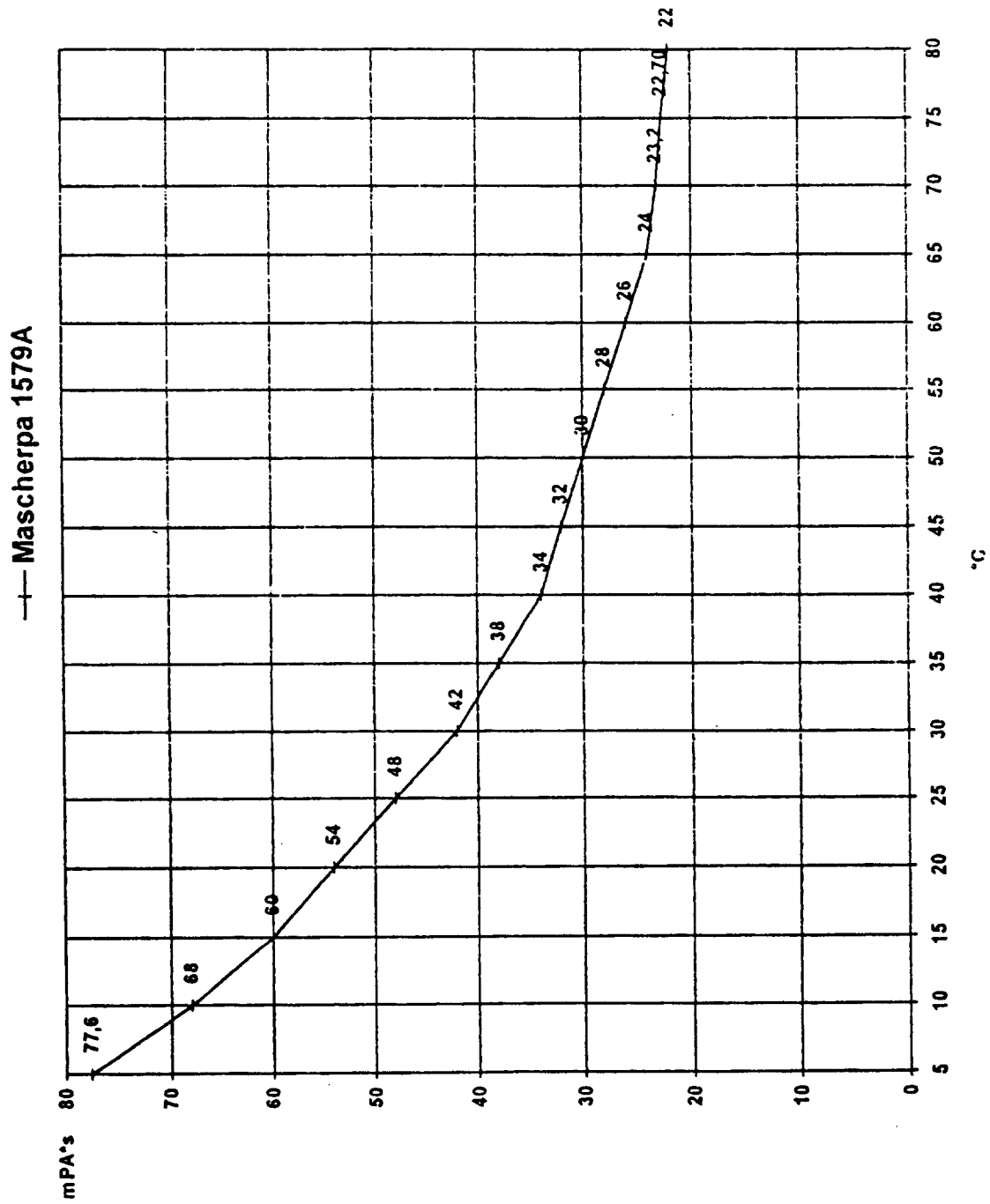


Fig. 19

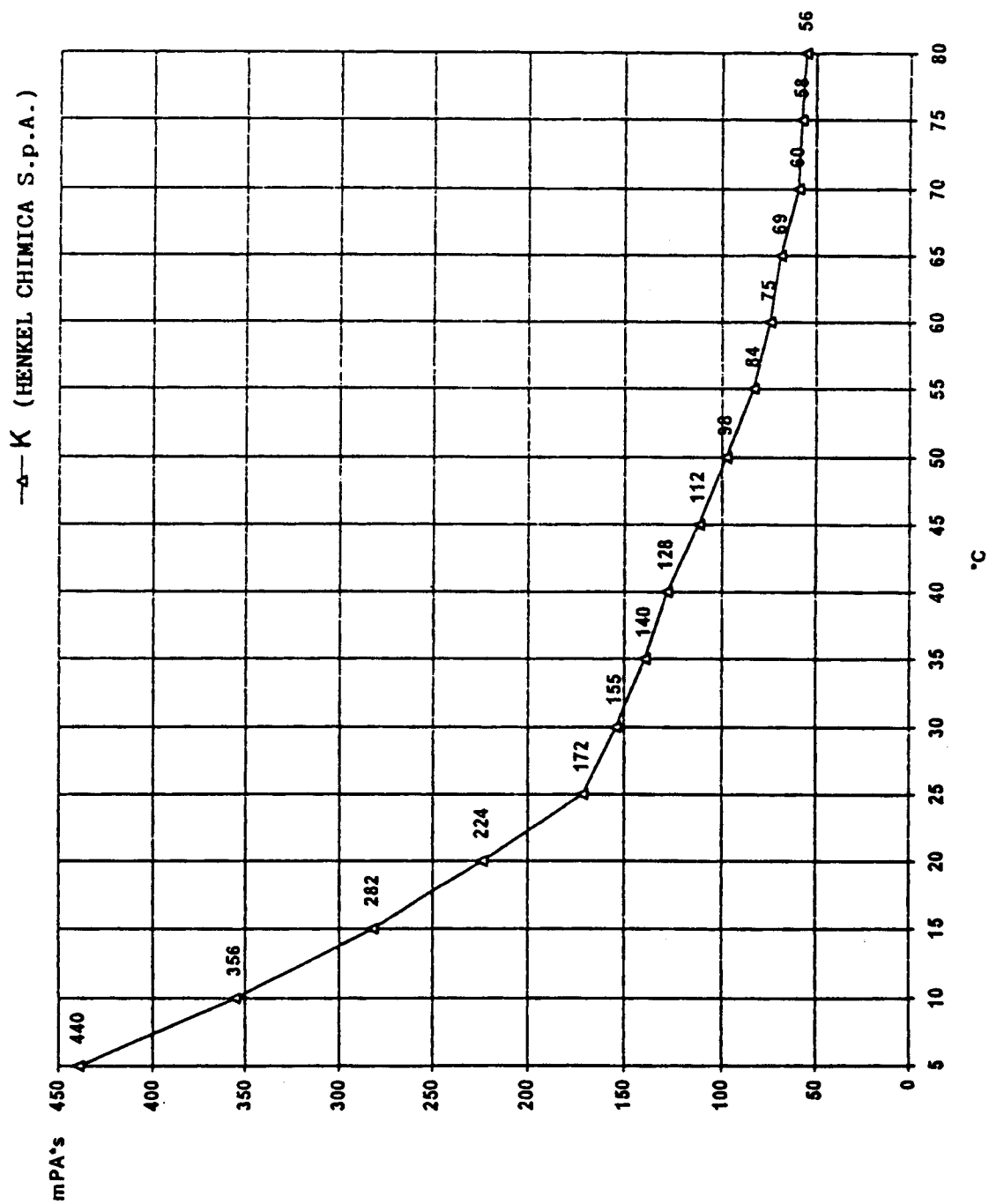


Fig. 20

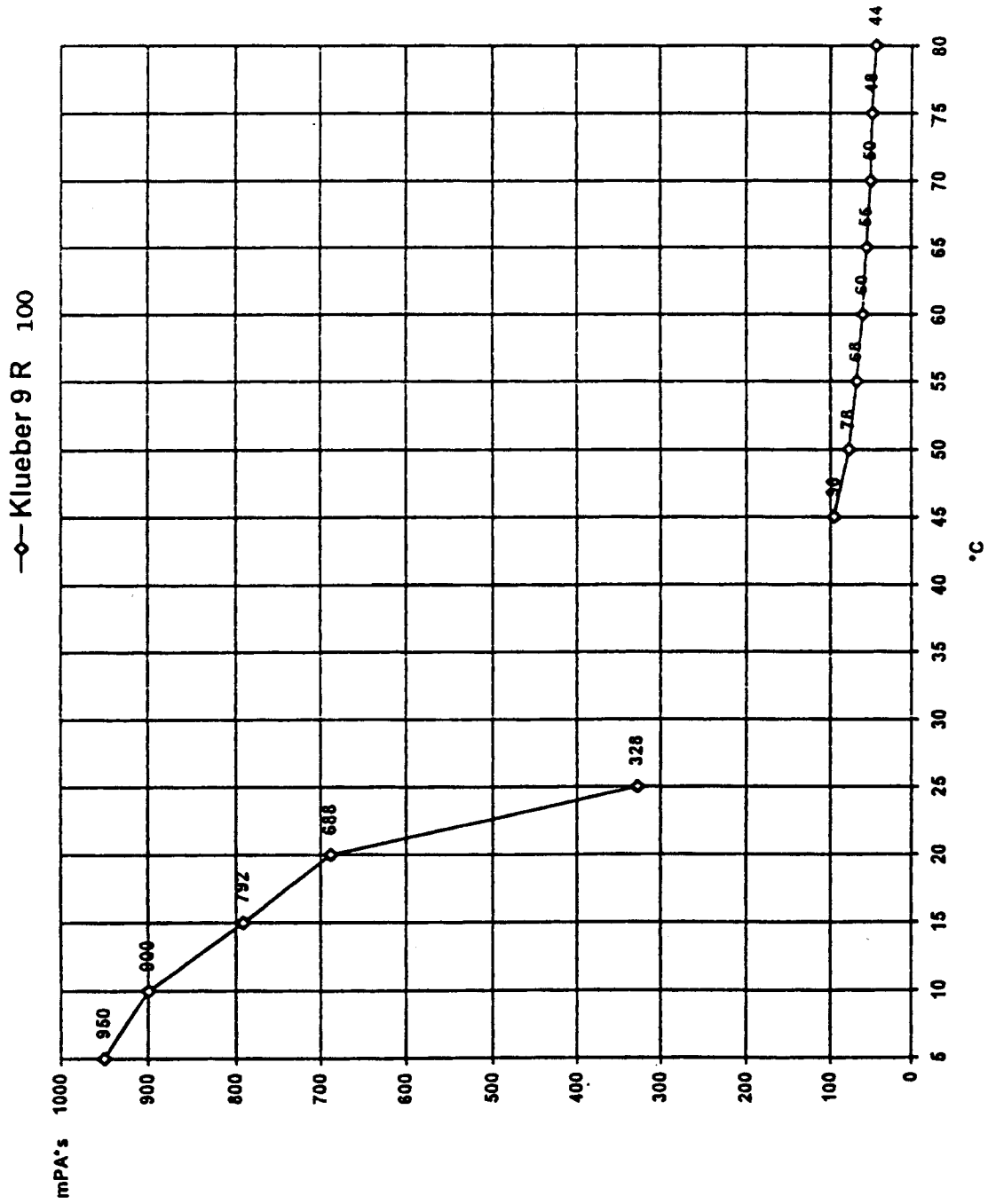


Fig. 21

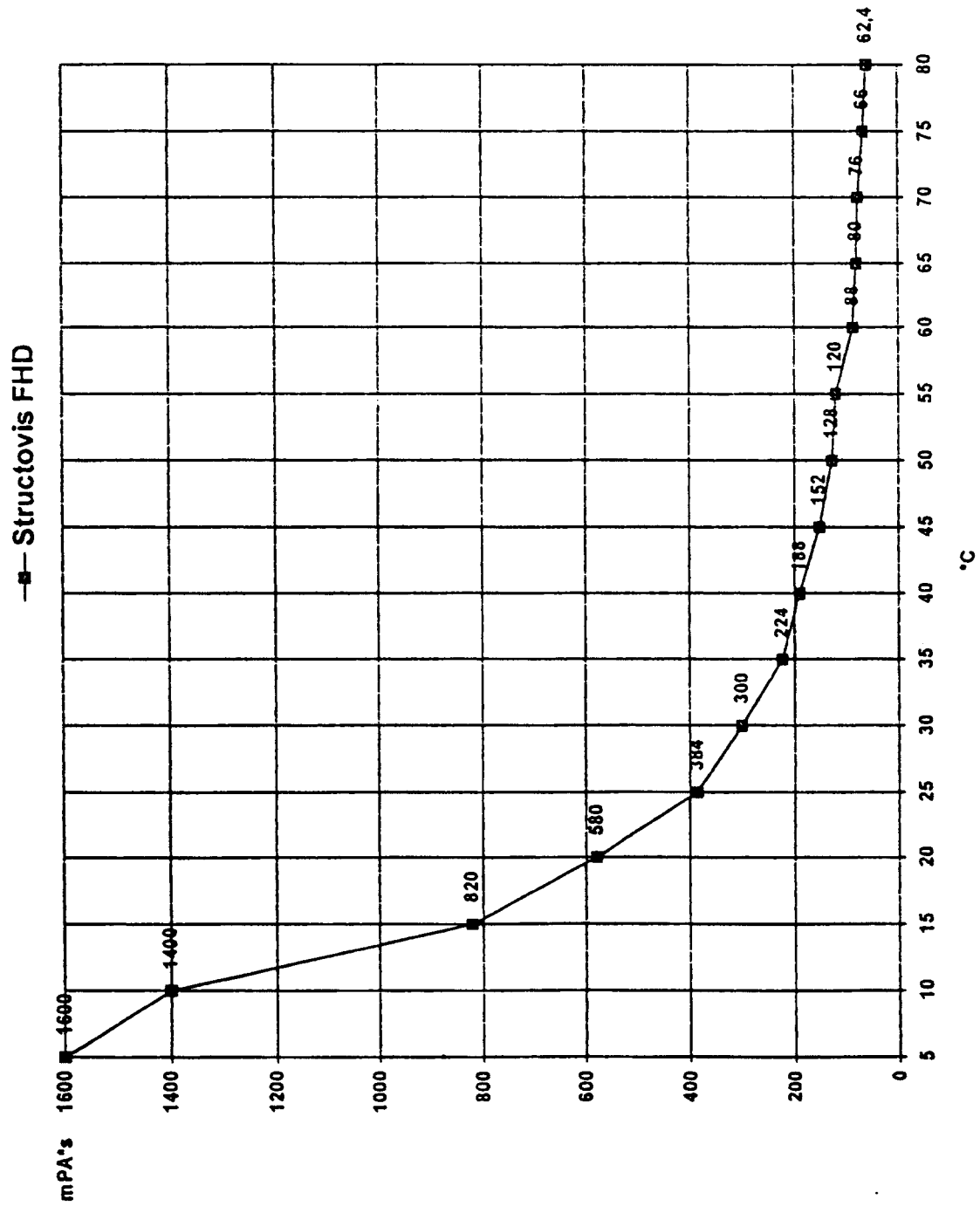


Fig. 22

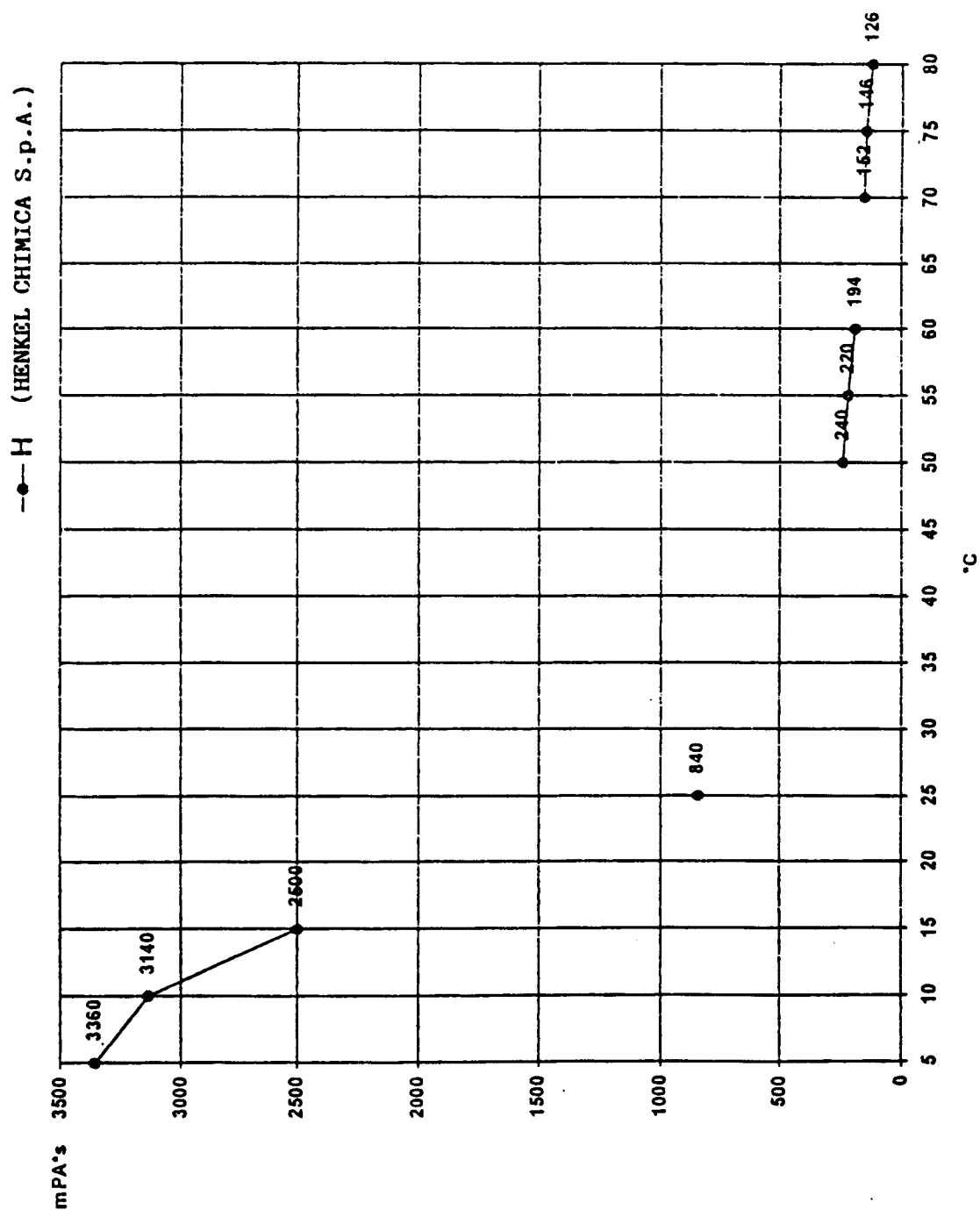


Fig. 23

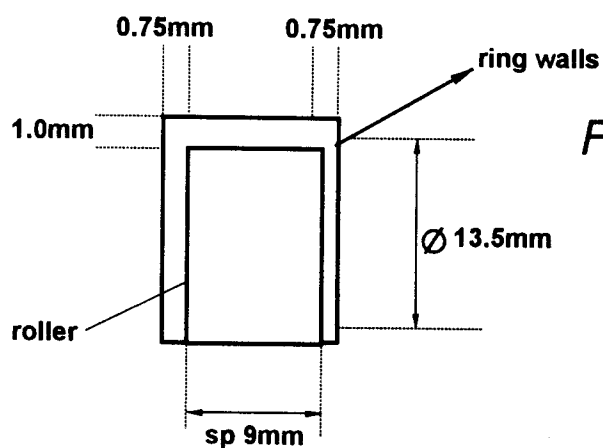
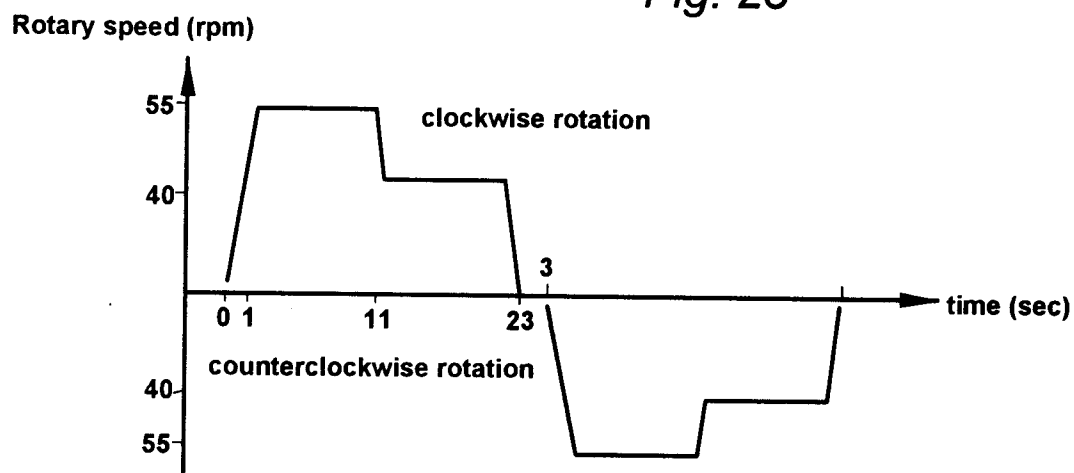


Fig. 24

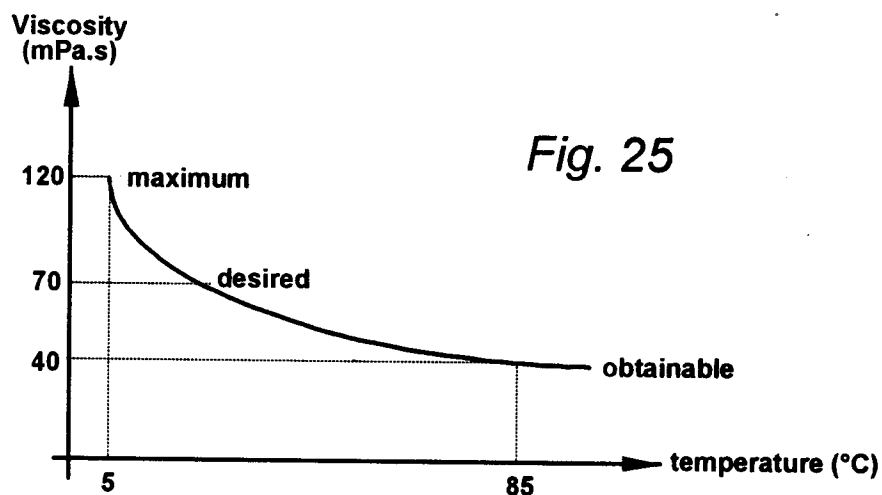


Fig. 25