KONI Self levelling hydro pneumatic suspension unit

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List of symbols

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KONI SELF LEVELLING HYDRO PNEUMATIC SUSPENSION UNIT.

I General technical remarks about vehicles.

II Choice of system with brief discussion.

III Technical and commercial points of view.

I-1 General requirements to suspension.

For the construction of the hydro pneumatic suspension unit a certain number of fundamental requirements which the car suspension has to meet, have been taken into account, such as e.g. the following :

a. vibration isolation
b. simplicity of construction
c. safety and adaptability to the vehicle
d. road holding characteristics.

I-2 Restricting ourselves to the characteristics and possibilities of the construction, the first striking feature is the simplicity of construction. With respect to its appearance the suspension unit may be compared with a shock absorber fitted upside down with a slightly larger diameter. We would emphasize, however, that it is not a shock absorber, but a suspension unit with or without a built-in shock absorber.

The characteristics may be classified as follows :

A suspension unit

a. being self levelling
b. having a constant natural frequency
c. having a built-in load sensitive damping
d. with a very low spring rate.

I-3 Vibration isolation.

This suspension unit only ensuring the vertical suspension, we shall only discuss this theme passing over the horizontal suspension which is at least as important especially with the low stiffnesses required and the present development of the tyres.
In order to obtain an optimal ride comfort for the passengers it will be necessary, in general, to keep the accelerations as small as possible by means of a proper vibration isolation.

The isolating factor is determined by the characteristics of the following :

1. tyre
2. spring and damper
3. seat

As KONI cannot practically influence the characteristics of tyres and seat we shall restrict ourselves to the spring and damper viz. to a single degree of freedom system.
The meaning of the letters used in the under mentioned formula is as follows :

M = mass of the suspended part per unit
C = spring rate
m = mass of the unsuspended part

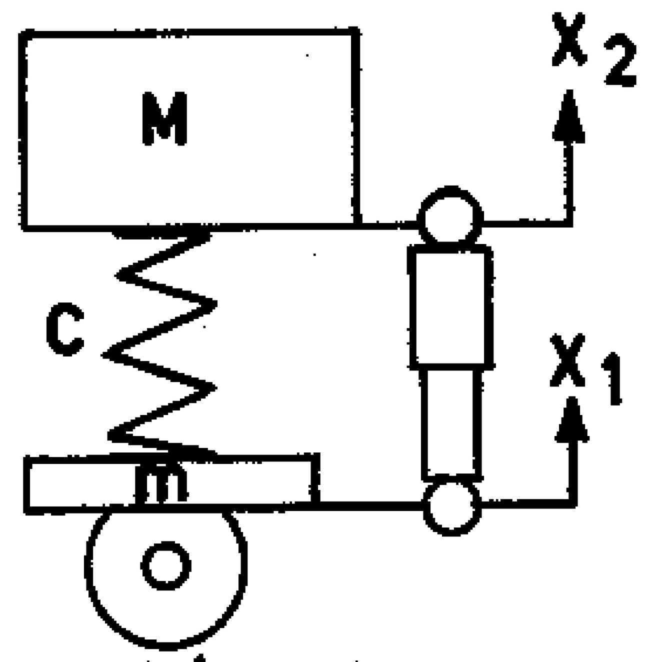


fig. I-1

$$\frac{x_1}{x_2} = \text{amplitude ratio}$$

$$\frac{\ddot{x}_1}{\ddot{x}_2} = \text{isolating factor}$$

In order to obtain an isolating factor as great as possible it is necessary to keep C as low as possible on temporarily by-passing the damping.

As it is difficult to compare spring rates of different suspension systems it is better to add the following value :

N_e = resonance frequency

showing the relation between M and C :

$$N_e = \frac{1}{2\pi}\sqrt{\frac{C}{M}}$$

It will be clear that when taking a low C, N_e is low as well. However, when increasing M without changing C, the value N_e will decrease.
This is the case with practically any motor car being more or less linear sprung. In other words a large load variation results in a relatively large variation N_e, i.e. a large difference in suspension comfort!

From the above follows when C being constant :

$$N_e = \frac{5}{\sqrt{f}}$$

so that with a given N_e the static deflection is also known.

The static deflection f will be high for a low N_e.

For stiffness and strength calculations of attachments and chassis the shock factor k is important. According to Garrett (in Automobile Dynamic Loads - Aut. Eng. '53) the most probable max. value for k is 3.

His calculation with a safety factor of 1,5 is based on :

$$P_{max.\ vert.} = 4{,}5\ R \text{ with } R = (M+m)\ g$$

Fixing -g- at 10 m/sec^2 the max. occurring vertical acceleration is thus 3 g.

As 4,5 R cannot be obtained in the mostly rather small and limited total spring travel, the suspension system is provided with bump rubbers of high stiffness and preferably with a progressive characteristic.
Depending on the purpose, for which the vehicle is used, a value k of 1,6 - 2 is usual for spring calculations.

With given values k, N_e and M the min. necessary spring travel on bump is consequently f_{dyn}.

From N_e and M follows C and

$$f_{st.} = \frac{M}{C} \cdot g \qquad f_{dyn.} = \frac{k\ Mg.}{C}$$

From the above theoretically follows :

$$f_{total} = \frac{(1+k)\ Mg}{C} \qquad (f_{st.} = \text{static deflection})$$

In other words with given values for f_{total}, k and M the obtainable N_e follows automatically.

As N_e decreases, when M increases, the really dynamic spring travels become proportionally larger under the same ride conditions.
The suspension now seems to become softer without changing anything to the spring construction.

As $f_{st.}$ now also increases, i.e. proportionally to the load rate, i.e.:

$$f_{st.laden} = \frac{G\ laden}{G\ unladen} \quad f_{st.unladen}$$

and, as already explained :

$$f_{dyn.laden} = \frac{G\ laden}{G\ unladen} \quad f_{dyn.unladen}$$

the spring starts bottoming there, where this did not happen under unladen conditions.

Consequently for calculations it is necessary to start from

$$k.G_{laden} \quad or \quad C = \frac{k.G\ max.}{f_{total}}$$

When indicating the load rate with K, this will give the following :

$$K = \frac{G\ max.}{G\ unladen} \quad or \quad C = \frac{k.K.G}{f_{total}}$$

From the above clearly follows that with a linear spring the spring rate increases with an increasing load factor. Specially cars with the centre of gravity far forward or far backward will have a high C or N_e at the opposite side and be experienced as uncomfortable. Especially with the small light cars there are many examples.

Further it is clear that

$$N_{e\,laden} = \frac{1}{\sqrt{K}} \cdot N_{e\,unladen}$$

The critical damping for the above mentioned suspension system is :

$$D_{crit.} = 2 \sqrt{C.M.} \quad kg/cm/sec.$$

and under laden conditions :

$$D_{crit.} = 2 \sqrt{C.K.M._o}$$

In other words with a linear suspension the damping must vary with $\sqrt{K}$ which is generally disregarded.

I-4 Progressive suspension.

If nevertheless we want to apply a low N_e lower than the linear value which would follow from the given values k and f_{total}, the spring characteristic must be progressive (fig. I-2). From this follows that a progressive characteristic automatically gives a lower C with an unchanged k.

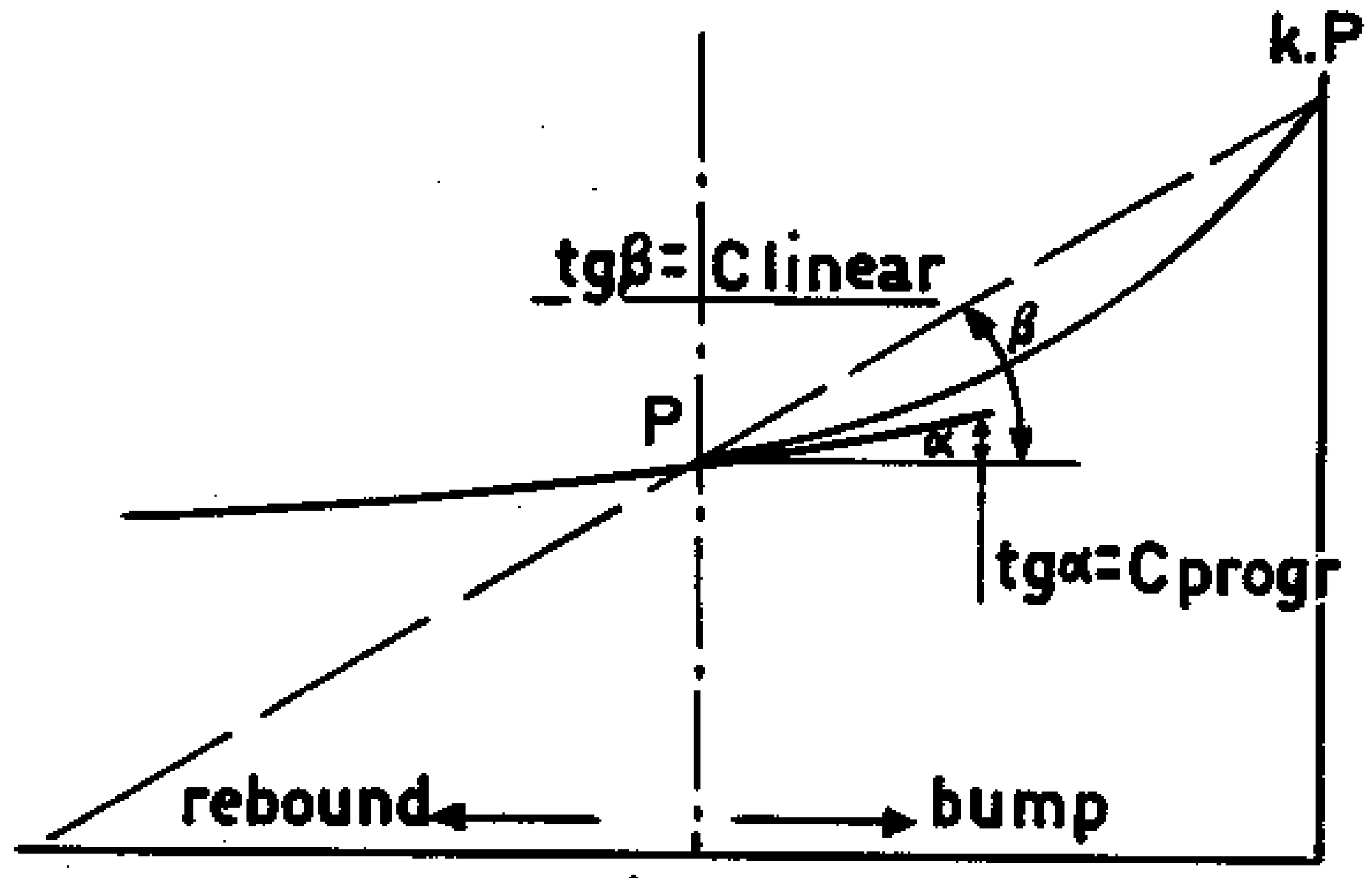


fig. I-2

It is possible to give the spring characteristic such a form that the spring constant proportionally increases with the load, so that N_e remains constant.

This results in very great features, such as invariable comfort irrespective of the load conditions.

In order to keep the natural frequency unchanged the following conditions must be met with :

Linear : $N_{e_{laden}} = \frac{1}{\sqrt{K}} \quad N_{e_o}$ and $f_{dyn._{laden}} = K.f_{unladen}$

N_e = constant, if the value of fraction $\frac{C}{M}$ remains constant, i.e. :

$C_{laden} = K.C_o$ and $f_{dyn._{laden}} = f_{dyn._{unladen}}$

Conclusion : With a given f_{total} the C_o can be much lower than that with a linear system, i.e. with progressive suspension N_e is lower than the one with a comparable linear system.

With load increase the deflection of a progressive system, however, becomes higher than the one with a linear system (fig. I-2).

Mere progression thus gives no improvement.

With constant level suspension (e.g. air suspension) the deflection due to extra load is still present in the first instance, however, after levelling the deflection disappears, so that then the total $f_{dyn.}$ is again available of the suspension under laden conditions.
In other words a system with a constant natural frequency and constant ride height requires the smallest f_{total} for certain values k and M or with given values k and M you obtain the lowest possible N_e with a suspension system with constant N_e and constant ride height (fig. I-3).

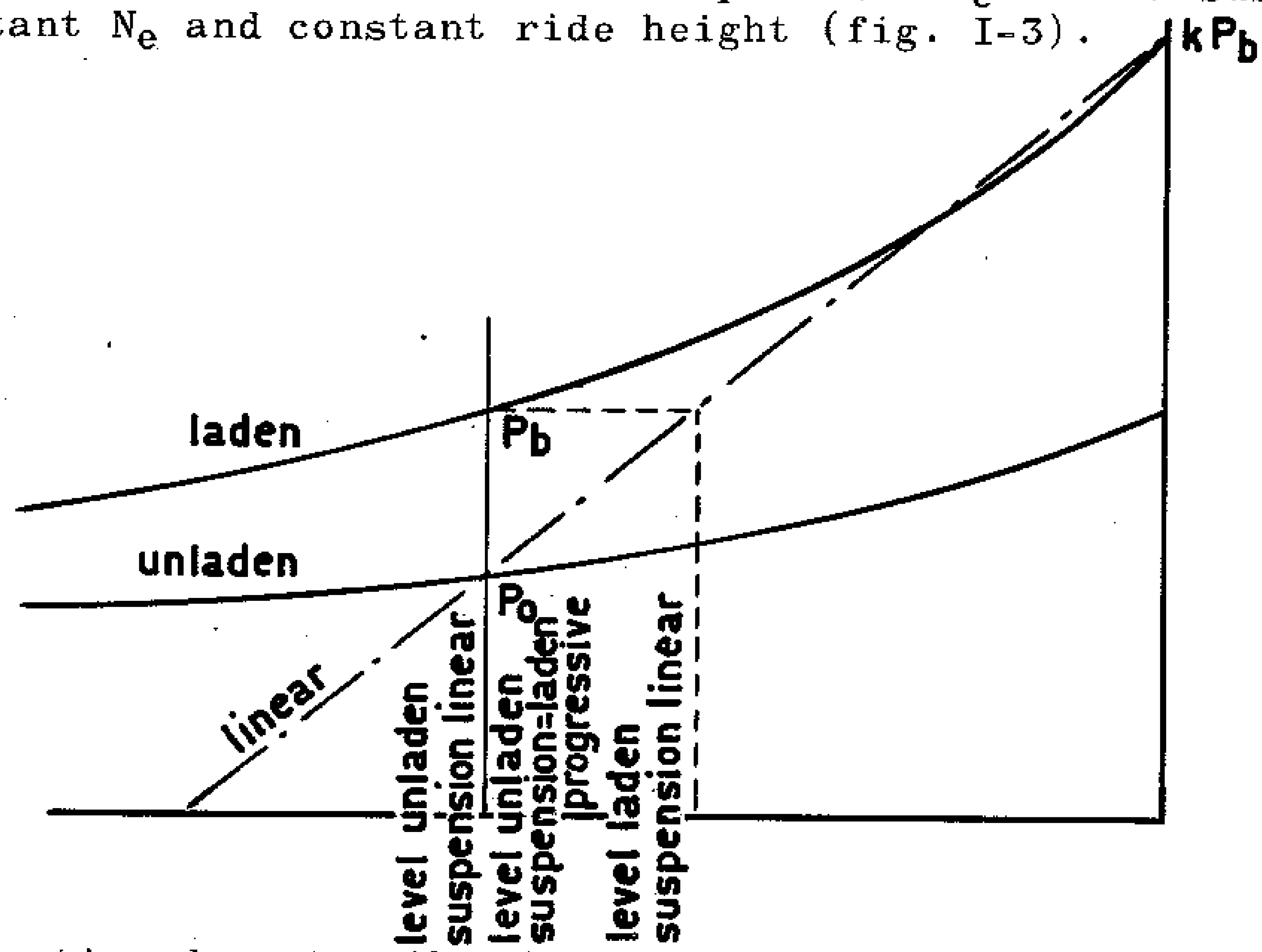


fig.I-3

In above mentioned system the damping must be completely load sensitive, because :

$$D_{laden} = 2\rho\sqrt{K.M_o}.K.C_o \quad \text{or} \quad D_{laden} = K.D_{unladen}$$

The resulting great feature is that with the unladen low C_o it is not necessary that the damping is as high as with the normal compromise damping.

II From the large range of possibilities a hydro pneumatic suspension system was chosen.

Fig. II-1 gives a schematic representation of the well known principle of a hydro pneumatic suspension unit.

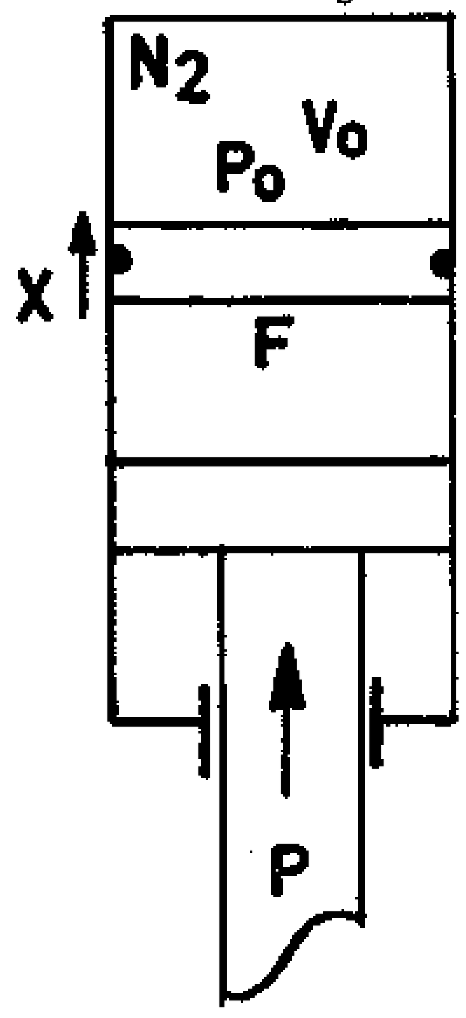


fig.II-1

The hydraulic part merely serves to transmit the gas pressure (generally nitrogen) to a piston or plunger, whereas the gas or gas compression and expansion serves as suspension medium. Actually the unit is an air suspension, however, with an invariable gas weight.

In principle such a system has no invariable natural frequency owing to the following :

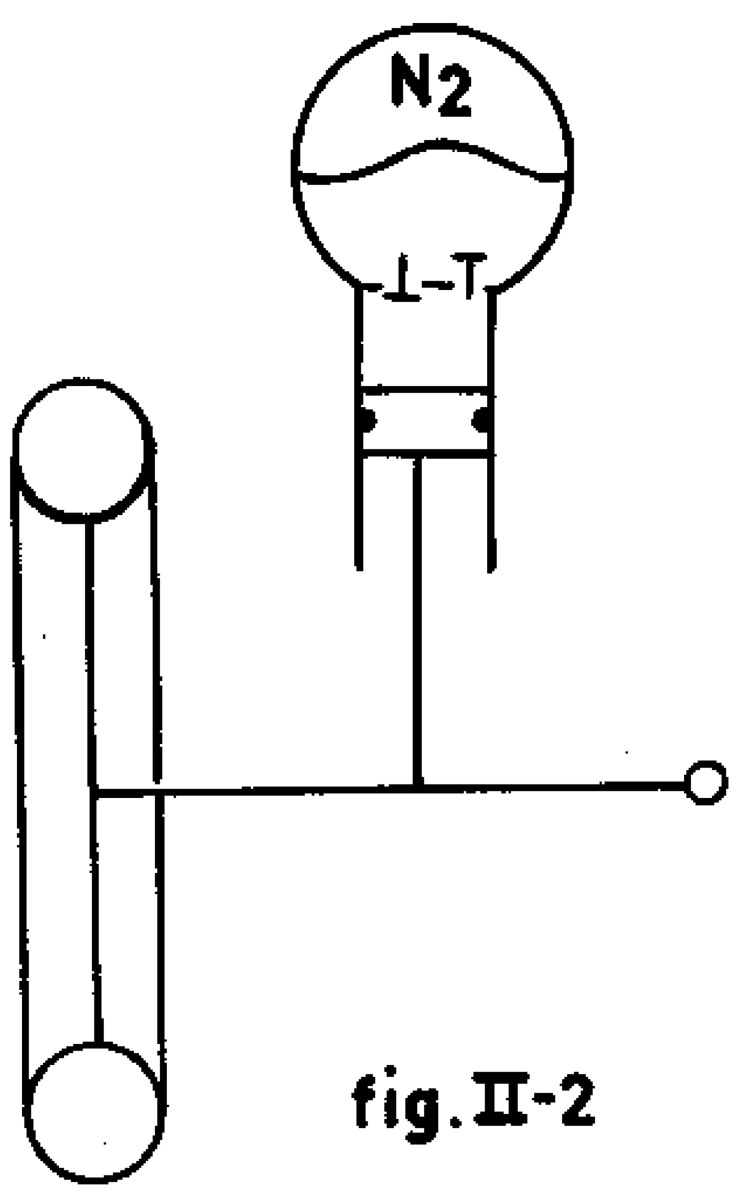


fig.II-2

According to Boyle: $p.V^n = \text{Constant}$

further:

$$C = \frac{dP}{dX} \text{ from which follows} \rightarrow C_o = \frac{n.p_o.F^2}{V_o}$$

$$\text{and } C_b = \frac{n.pb.F^2}{V_b}$$

$$N_e = \frac{1}{2\pi}\sqrt{\frac{C}{M}} \text{ thus:}$$

$$N_e = \frac{1}{2\pi}\sqrt{\frac{F^2.n.p_o}{V_o.M}} = \frac{1}{2\pi}\sqrt{\frac{F.M.g.n}{V_o.M}} = \frac{1}{2\pi}\sqrt{\frac{F.n.g.}{V_o}}$$

As V_o is not constant, neither is N_e

$$\frac{V}{V_o} = \left(\frac{p_o}{p}\right)^{\frac{1}{n}} = \left(\frac{G_o}{G}\right)^{\frac{1}{n}}, \text{ for } \frac{p_o}{p} \approx \frac{G_o}{G}$$

$$\text{thus } V = V_o\left(\frac{G_o}{G}\right)^{\frac{1}{n}}$$

$$N_{e\,laden} = \frac{1}{2\pi}\sqrt{\frac{F.n.g.}{V_o}\left(\frac{G}{G_o}\right)^{\frac{1}{n}}}$$

$$N_{e_b} = \frac{1}{2\pi}\left(\frac{G}{G_o}\right)^{\frac{1}{2n}}\sqrt{\frac{Fng}{V_o}} \text{ and if } \frac{G}{G_o} = K \text{ (load variation)}$$

$$\text{Consequently } \frac{N_{e_b}}{N_{e_o}} = K^{\frac{1}{2n}}$$

n will vary between 1,2 - 1,4 for polytropic or adiabatic compression, dependent on the bump velocity.
From the above follows that N_e increases with the load with the consequence that due to loading, thus completely contrary to that with linear suspension, the suspension becomes stiffer.
We have now examined the possibility to combine the good characteristics of above system with a natural frequency still being constant. Compare fig. II-3 with II-4.

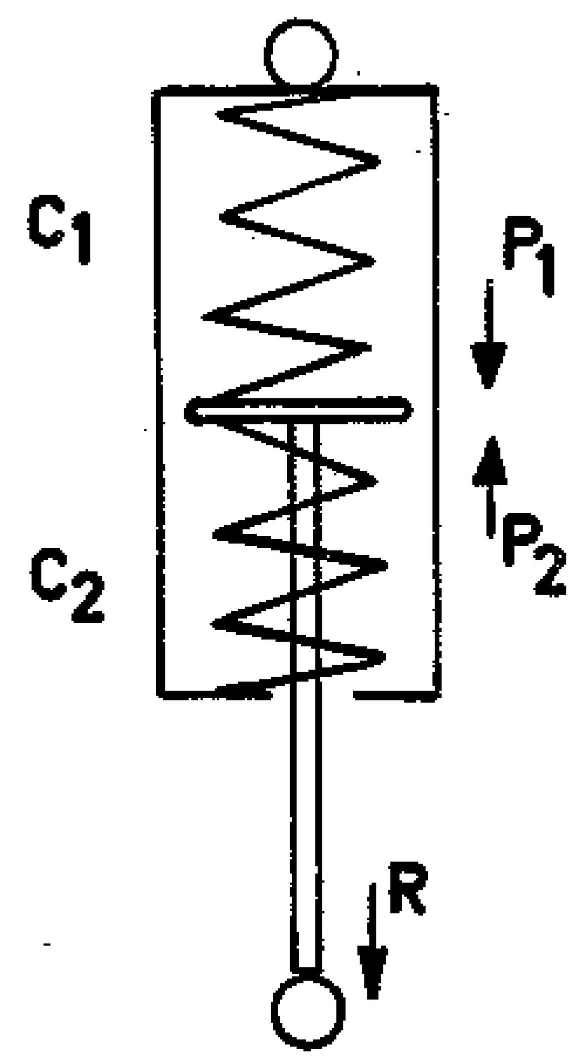


fig. II-3

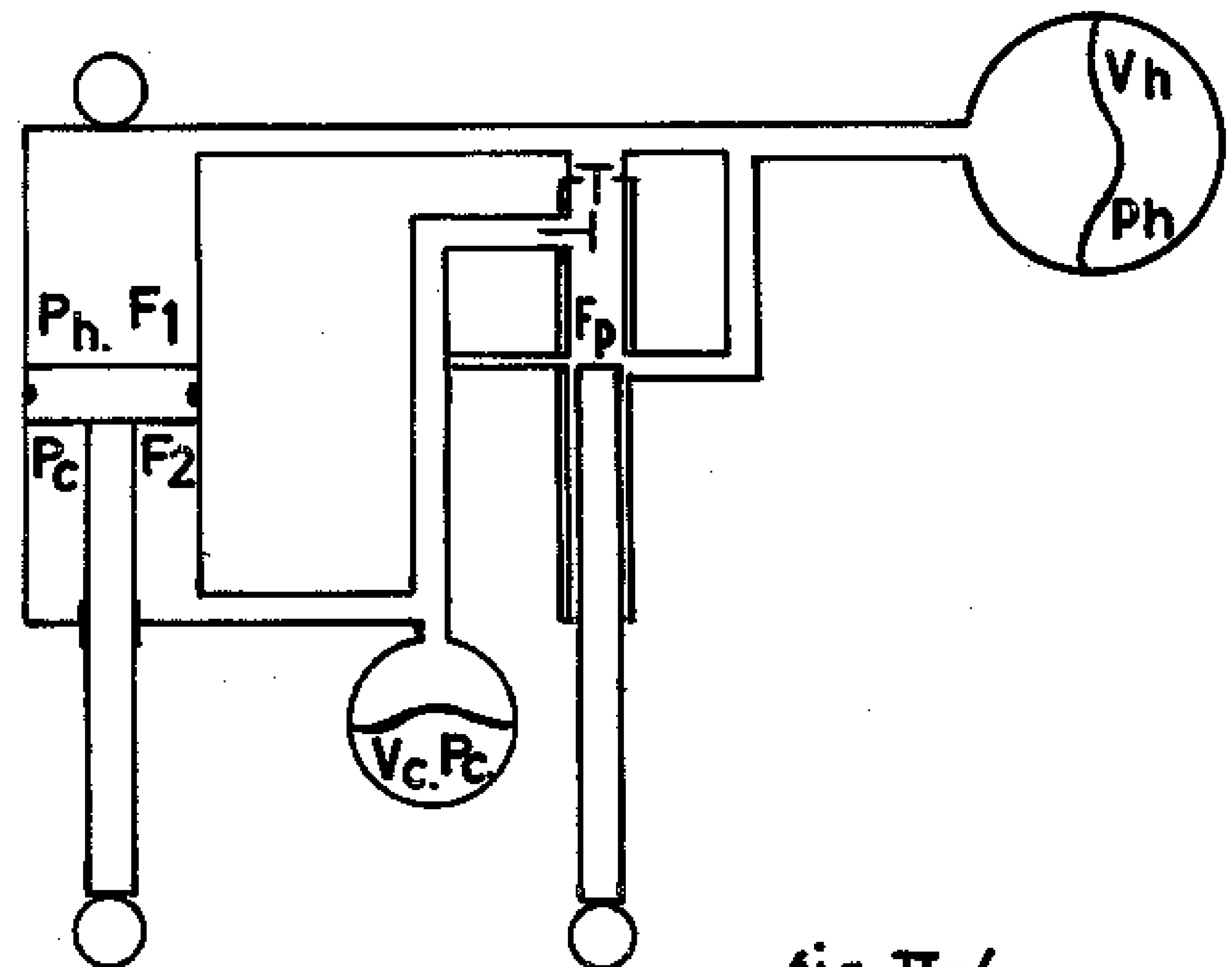


fig. II-4

Fig. II-3 shows that the resulting force is the difference between the two spring forces, whereas the resulting stiffness is equal to the sum of both spring rates.
This is the same with fig. II-4 schematically representing the KONI self levelling hydro pneumatic suspension unit.

The resulting force on the piston rod is here $\{p_h\ (F_1 + F_p) - p_c \cdot F_2\}$ kg whereas the resulting spring rate is the sum of C_h and C_c which each are dependent on their corresponding gas volumes V_h and V_c and F_1 and F_2.

With the suspension system under load the spring will first be compressed, so that the pump enters its operational range displacing at the same time an amount of oil F_p.X to V_h.

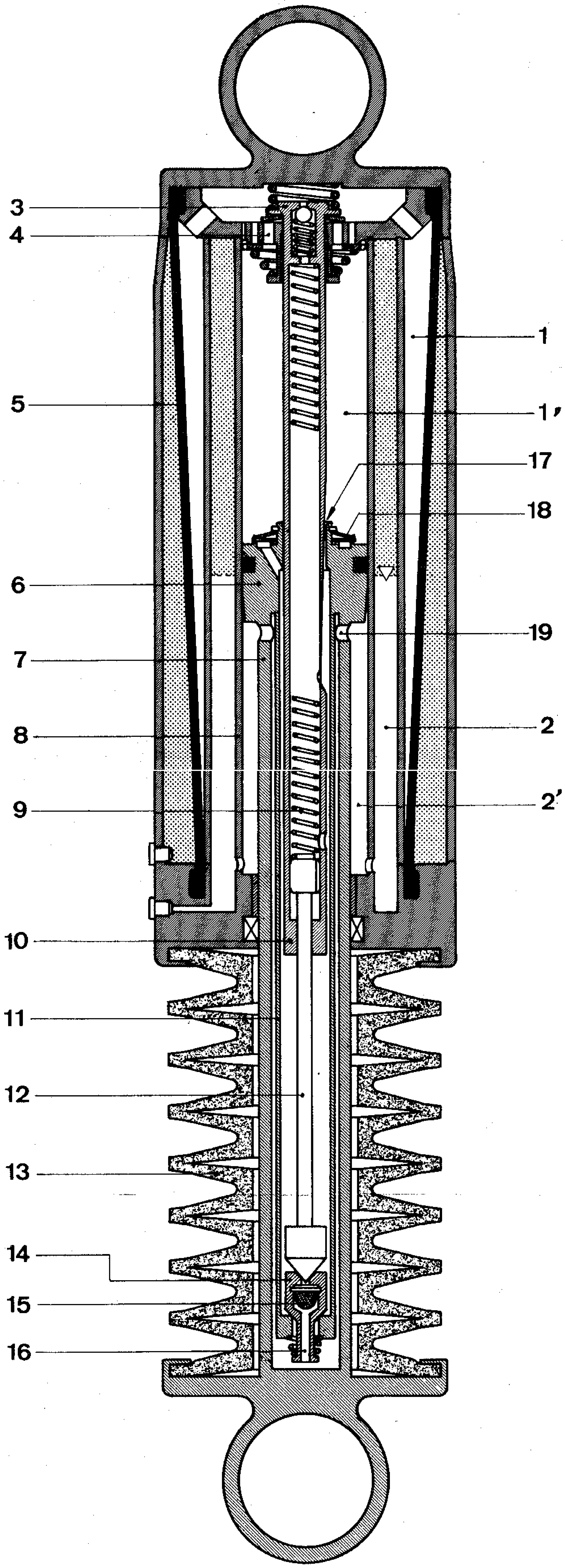
On rebound travel the pump sucks oil from V_c into the pump cylinder which is then again pumped to V_h during the following bump stroke.

This continues, until the pressure difference P_h - P_c becomes so large that the spring piston has reached the ride height, i.e. centre to centre distance between the eyes of the spring unit has obtained a predestined value giving the vehicle the normal ride height.

Whenever the pump makes a rebound stroke, the plunger opens a connecting channel from the pump cylinder to the main gas volume, after which the pump is by-passed and no oil is being pumped from V_c to V_h.

Under this load the vehicle always remains at the normal ride height. When removing the load, the spring will first extend. The pump then opens a return flow channel, through which oil continues to flow from V_h to V_c, until the correct centre to centre dimension is again reached, after which the pump closes the return flow channel.

The components shown beside each other in fig. II-4 are in reality all grouped concentrically around the cylinder giving a compact entirely self-supporting suspension unit (principle scheme fig. II-5).



1	MAIN SPRING ROOM
1'	HIGH PRESSURE CHAMBER
2	CORRECTION CHAMBER
2'	LOW PRESSURE CHAMBER
3	RELIEF VALVE
4	DAMPER ASSEMBLY
5	DIAPHRAGM
6	PISTON
7	PISTON ROD
8	CYLINDER
9	COIL SPRING
10	PUMP PLUNGER
11	PUMP CYLINDER
12	LEVELLING ROD
13	BUMP RUBBER
14	SUCTION VALVE
15	FILTER
16	RETURNFLOW ORIFICE
17	BYPASS SLOT
18	VALVE
19	SUCTION BORE

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Piston 6, which is screwed on the hollow piston rod 7 divides the working cylinder in two parts 1' and 2'. Between the working cylinder and a partition there is a space 2, in which there is oil and gas of the correction spring. Between the partition and the outer tube there is the main spring room 1 divided into an oil space and a gas volume by diaphragm 5.

The main spring room is connected with the damper assembly by channels, and with the space above the piston through the valves and orifices of that damper assembly, so that the pressure of the gas in the main spring room acts on top of piston 6 by means of the oil.

The correction spring in space 2 is connected with 2' by channels, so that the gas pressure of the correction spring acts on the bottom of piston 6.

The hollow pump plunger 10 is flexibly suspended in longitudinal sense in the damper by two springs and is self centring, so that the pump can align itself in the guide in piston 6.

The pump thus protrudes through its guide in piston 6 in the pump cylinder 11 which through suction valve 14 and space between 11 and 7 is connected with the correction spring.
In the suction valve 14 there is a small bore 16 covered at the pump chamber side by a small filter 15 which is built in the suction valve assembly.

Working principle.

On compressing the spring unit oil flows from chamber 1' through the damper assembly to the main spring room and increases the pressure of the gas. Moreover slot 17 disappears in the piston and an amount of oil $F_{pl}.X$ flows to the same spring room so that the total amount of oil which flows from 1' to 1 through the damper valves and orifices is $X\ (F_1 + F_{pl})$ on bump.
The pressure in chamber 2' decreases, as the oil follows the lower piston surface. On bump the levelling rod 12 is lifted off its abutment against the tension of spring 9.

On rebound the lower surface of the piston pushes back again the same amount of oil to space 2 less the quantity of oil sucked in by the pump through bore 19 and valve 14. This aspiration continues until bypass slot 17 opens. Valve 14 closes as the oil can flow in freely from high pressure chamber 1' through slot 17.When the spring is extended sufficiently, the levelling rod 12 is lifted off its seat in the suction valve by means of its abutment, and from then on oil can return to space 2' through a small bore 16, space between 7 and 11 and bore 19.

Now there is made a direct connection, be it choked by bore 16, between the main gas spring and the correction gas spring.
If these conditions occur several times or during a few seconds, this results in a pressure equalisation, so that finally the pressures on F_1 and F_2 are equal. The resulting force equals the pressure multiplied by the sectional area of the piston rod.

During the rebound and bump strokes oil must flow through the damper assembly which causes a pressure drop dependent on the velocity and direction of the oil flow.

During the first half of the rebound stroke the above pressure acts on the piston crown area less the sectional area of the plunger whereas the second half of the stroke is effected by the pressure on the sum of the piston area plus the plunger area, since from then on the pressure in pump chamber 11 is equalised with chamber 1' through the bypass slot 17.

Obviously a hysteresis is formed now in direct proportion to the pump action.

As with heavy loads the pressure in 1' is much higher than in 2' and under unladen conditions the pressure difference in static position is 0, this pump action which acts as a damping action is entirely load sensitive.

The pump thus automatically ensures a load sensitive damping which is not velocity sensitive. Therefore it is necessary to have an additional damping which gives the car the damping it needs in the unladen condition, when the pump action is still zero. This takes place in the well known manner by means of spring loaded valves and constant bores in damper assembly 4.

There is in piston 6 a seal ensuring -together with levelling rod 12 which closes the return flow constant- that once a pressure difference has been established by the pump action under influence of a certain load this same pressure difference cannot be equalised, unless this load is removed from the vehicle. In other words the vehicle will not sag e.g. due to leakage when stationary.

The point at which the pump is by-passed and the point at which the levelling rod opens the return flow bore, determine the nominal level of the spring unit.
This nominal level must correspond with the ride height of the vehicle.

For each application it will further be necessary to separately consider, whether the initial drop due to a large load variation will not become so excessive that the piston cannot move up and down any more in relation to the foot valve. In such a case the unit must be equipped with a bump stop, if not being already present.

It is clear that this remaining stroke is necessary to start pumping; once on normal ride height the bump will not or scarcely be in operation.

The afore mentioned equalizing pressure or basic load pressure may never correspond with the unladen conditions of the vehicle, but has to remain somewhat lower than same dependent on the temperature variations to be expected. By doing so we prevent that due to a temperature rise and a consequent relatively large expansion of the gas volume with the pressure remaining constant, there would be no possibility to blow off causing the vehicle to stand too high on the spring.

Furthermore a safety valve has been incorporated in the pump base which prevents that under overload conditions of the vehicle the pressure in the main gas volume would rise to the bursting point of the outer cylinder. The construction of this safety valve is such that on blowing off the pressure in the main gas volume will not drop at once, but merely stops the pump working.

For the construction of a suspension unit the following data are required:

1. What is the lowest permissible natural frequency in connection with the roll stiffness of the vehicle on the spot, where such a suspension system has been planned?

2. What are the load variations to be split into:

   2a Max. permissible weight between wheel and road surface.
   2b Min. weight on this same spot.
   2c Exact weight of unsprung parts.

3. What is the lever ratio of the spring or mechanical advantage?

4. What is the centre to centre distance of the spring attachments with the vehicle at static level and at curb weight?

5. Has a soft bump rubber been incorporated in the design already? If so, what are the characteristics, bump travel and max. permissible load?

6. Drawings of the surroundings of the suspension unit clearly showing the available space.

7. What is the max. permissible spring respectively wheel travel?

8. Is there already a spring (leaf spring or otherwise) incorporated in the design? If so, this spring has then to be reduced in load carrying capacity <u>and</u> stiffness. Is this possible and to what extent?

III From the above two chapters follows that the product in question has requirements essentially different to those of the shock absorber. It actually covers a field which is closely connected to that of the damper, but that up to now entirely belonged to the responsability of the car designer.
Above is explained why the conventional suspension and damping are always a compromise solution. Hence the following sales arguments :

1. In general can be said that starting from this conventional suspension there is a large improvement of comfort due to the following points :

   a. constant resonance frequency
   b. constant level suspension
   c. load sensitive damping
   d. consequently, a very soft suspension.

2. A very compact construction without piping, height adjustment valves, tanks, pressure accumulators (very expensive) and compressors respectively motor pumps.

3. Easily to be mounted on the spot, where normally the shock absorber would have been fitted.

4. Possibility to use the unit as main suspension which is advisable in all cases, whereby all remaining suspension elements are disregarded in so far they are not required for the axle location or brake and drive reactions.

5. Combination with existing spring elements, e.g. a leaf spring reduced in carrying capacity and stiffness.
   We would like to observe in this connection that our suspension units under unladen conditions must already carry a part of the car weight as large as possible in order to keep the characteristics of the suspension unit.

6. Road holding characteristics can be improved by the following :

   a. By maintaining a constant ride height the camber and caster angles of the suspension remain unchanged.
   b. Load sensitive damping.
   c. Roll stiffness increases proportionally with load.
   d. No bottoming of suspension under max. load conditions.

7. Head lamps always properly aimed.

KONI AUXILIARY SELF LEVELLING HYDRO PNEUMATIC SUSPENSION UNIT

When the load variation, used in an absolute sense, is not so large, there is also a further possibility viz. the auxiliary self levelling suspension unit.
We suppose in the above case that the greater part of the suspended mass is carried by a conventional spring (coil spring, leaf spring, torsion spring) of the unladen vehicle. A load up to a maximum of 20% of the unladen weight of the vehicle is carried by an auxiliary suspension unit which, starting from this 20%, will carry the total load variation, whereas the coil spring load remains the same.
An example will clarify the matter:

Unladen weight of vehicle	rear	1435 lbs
	unsprung	120 lbs
	sprung	1315 lbs

i.e. for each wheel 658 lbs and when i =1, then it is the total weight for each combination of spring and suspension unit under unladen conditions. Supposing that the total load variation (difference between unladen and laden vehicle) of the rear axle is 800 lbs this means 400 lbs per wheel. Thus the maximum carrying capacity of spring and suspension unit is 1058 lbs ($\frac{1315}{2}$ + 400). The suspension unit has a basic carrying capacity of 131,5 lbs being 20% of 658 and a maximum of 530 lbs. Consequently the coil spring should have a carrying capacity of 658 less 131, 5 is 526,5 lbs.

As the conventional springs with which this kind of suspension units must co-operate, have mostly linear characteristics, it would not be sufficient to use an auxiliary s.l. unit with an approx. constant natural frequency. This auxiliary s.l. unit should have a more than linear increase of spring rate with respect to increase of load. Already at an earlier stage we drew the attention to the advantage of a constant natural frequency. When using a main suspension unit it would not be sufficient to apply a single acting suspension unit, as this would show an increasing natural frequency in case of increase of load.

However, such characteristics are now necessary for an auxiliary s.l. unit. This could be realized with a one plunger suspension unit, i.e. one plunger surface with one elastic gas volume. The pressure of this last mentioned gas volume is enlarged or reduced by a pump system identical to that of the double acting system.

The reservoir is no longer connected directly with the working cylinder, but with the pump chamber behind the suction valve. Either is there a separate pump cylinder; the hollow piston rod itself acts as pump chamber.

The conception of the mere suspension unit is thus rather simple. Properly speaking it is a normal shock absorber with a reservoir in which a variable pressure prevails acting on the rod surface and supplying in this way an extending force necessary for the carrying function of the suspension unit.

Our example shows that though the load variation is only 60%, the variation of the load carrying capacity which the suspension unit has to supply -from 131 to 530- is more than 400%.
When considering the formula for the spring rate of a single surface suspension unit with an invariable gas weight mentioned on page 6 and doubling e.g. the load -thus the pressure p- having therefore to reduce the gas volume by half, then the spring rate is found not to be doubled, but it becomes 4 times as high.
Therefore we can say that the spring rate of a single surface suspension unit increases by the square of the load variation. In our example the spring rate would increase by a factor 4^2. When the dimensions are chosen in such a way that the basic spring rate is 5,5 lbs/1" e.g., then the spring rate is 90 lbs/1" at full load at level length.
When we choose an acceptable natural frequency e.g. 1,25 Hz. at full load (dependent on the available wheel travel), then the corresponding spring rate is 166 lbs/1", i.e. the spring rate of the coil spring amounts to 166-90 = 76 lbs/in.

Under unladen conditions the weight is 658 lbs and the spring rate 76 + 5,5 = 81,5lbs/in, so that the natural frequency is 1,12 Hz.

From the above follows that the natural frequency does not remain so nicely constant as that of e.g. the air suspension or the double acting suspension unit. Nevertheless as a compromise it is no doubt acceptable.
Whenever we incorporated both the suspension unit and the reservoir into one body, we found that after some time gas collected in the pump chamber, so that the pump action gradually diminished.

This is the main reason why we separated these two parts and always place the reservoir above the suction valve, so that the gas absorbed by the oil in the reservoir, finally returns again in the reservoir.
However, in the starting position -thus unladen- the gas pressure above the oil in the reservoir and in the high pressure chamber are equal.

The damper piston II is fitted to the piston rod 10 which acts as a spring plunger. The damper piston has holes closed by valves of variable stiffness and pre-tension which are provided with orifices, so that in position of rest the pressure above and below the piston is always equal. This construction enables to perform any damping characteristic desired, whereas the damping process takes place at an increased level of pressure.
In this way, at the same time, possible formation of vapour or air bubbles is avoided at the low pressure side of the damper piston during the movement.
This pressure level varies with the load, but in unladen position the pressure is always high enough to realize an exact damping.

The working of this auxiliary self levelling suspension unit is as follows.
When the vehicle is loaded the main springs will show a certain deflexion moving at the same time the suspension unit which in unladen position practically has no stiffness. Consequently the piston rod 10 will enter the working cylinder more deeply, so that also the pump plunger 14 will farther penetrate the hollow piston rod.

When the vehicle starts moving, the relative movement of the wheels with regard to the chassis will cause a corresponding relative movement of the piston rod with respect to the pump plunger and of the piston in the working cylinder 3,
During rebound movement the volume of the pump plunger which disappears from the piston rod, is compensated by oil from the reservoir 1 through the suction valve 7.
During the next bump movement the suction valve 7 is closed first, after which the pump plunger 14 displaces a quantity of oil through the delivery valve 12 to the working cylinder 3.

This process is repeated and continues as long as the pump plunger remains functioning in a position being under the predestined length of the suspension unit coinciding with the exact ride level of the vehicle .
As consequently the quantity of oil being in the space behind the membrane 4 in the high pressure chamber, increases, the pressure of the gas being at the opposite side of the membrane, will also increase. This increasing pressure raises the carrying capacity of the piston rod which will therefore relieve the main springs after some time.
As a consequence these springs will extend to their normal position again.

As soon as this position is attained, the pump will stop actioning, as a number of leak holes in the pump connect the pump chamber with the working cylinder and thus equalise the pressures, so that no further oil is sucked from the reservoir.
Just as with the double acting suspension unit a levelling rod 9 with cone prevents oil from flowing back to the reservoir. This will only take place, when the load is removed from the vehicle and the suspension unit will rise above ride level for some time.
This flowing back of oil is delayed by the return flow orifice 5 in the suction valve . To prevent this orifice from being obstructed it is equipped with a filter with fine mashes.

The pump system is thus identical to that of the double acting suspension unit. The difference is that the pressure which can be increased by pumping and decreased by opening the return flow orifice 5, only acts on the piston rod surface.

During the bump and rebound movement near the normal level position the oil displacement of the piston rod is received in the gas volume in chamber 2 only. The gas volume in chamber 1 does not act and consequently does not influence the suspension characteristic. This fact characterizes this suspension unit as a single acting suspension unit with a constant gas weight, for which is applicable that the spring rate increases by the square of the load variation and rod surface.

Also in this case there is a damping characteristic composed of a non velocity sensitive, but load sensitive part and a velocity sensitive part. The latter is caused by the damping components in the damper piston whereas the former comes from the pump action that is performed by every movement.

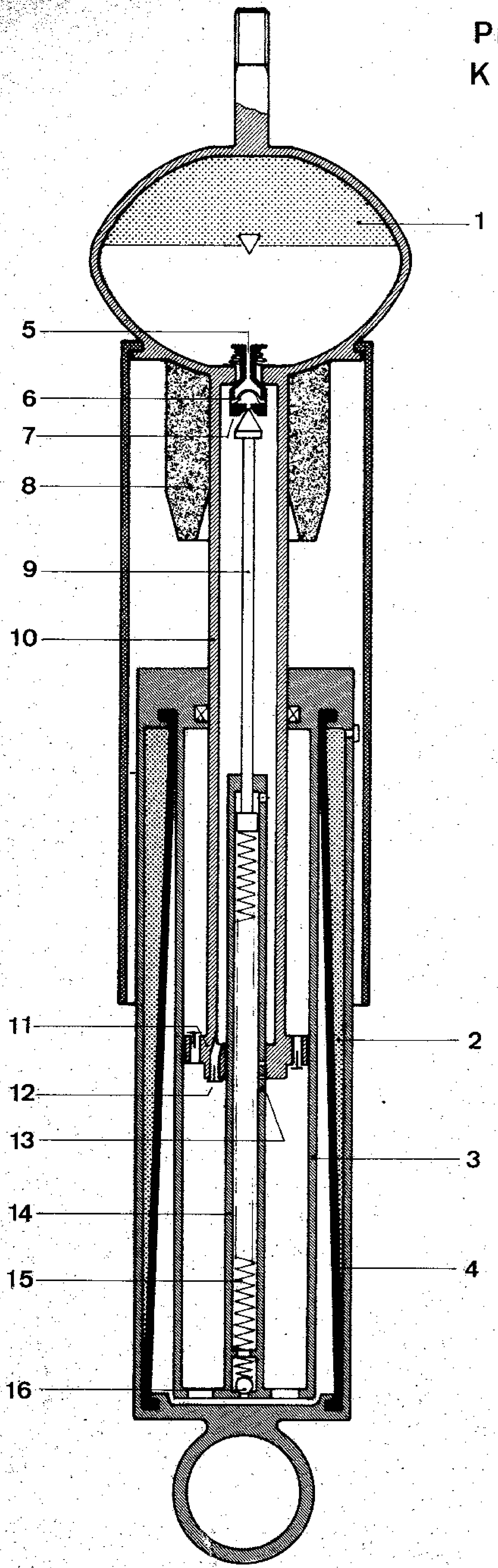
This suspension unit can be equipped with spring seats resulting in an assembly of damper and coil spring which can be considered as a complete main suspension unit with the same proporties as the double acting suspension unit. As a matter of fact the suspension unit can also be installed separated from the coil spring or leaf spring and -in case of sufficient space- located on the spot of the shock absorber.
In our arrangment the reservoir should be in line with the piston rod, because of which the fitting lengths are longer than those of the double acting suspension unit. In general, however, the single acting suspension unit will have a more slender shape (see drwg. 3018).

In those cases, where it is impossible to modify the space available in the vehicle to suit the lengths, the reservoir could be separated from the suspension unit using a flexible high pressure tube as connection (see drwg. 3019).
To facilitate matters the above flexible tube is provided with a self closing disconnect coupler fitted to the reservoir.
When mounting or dismounting a possible loss of oil or pressure from the reservoir or suspension unit is prevented in this way.

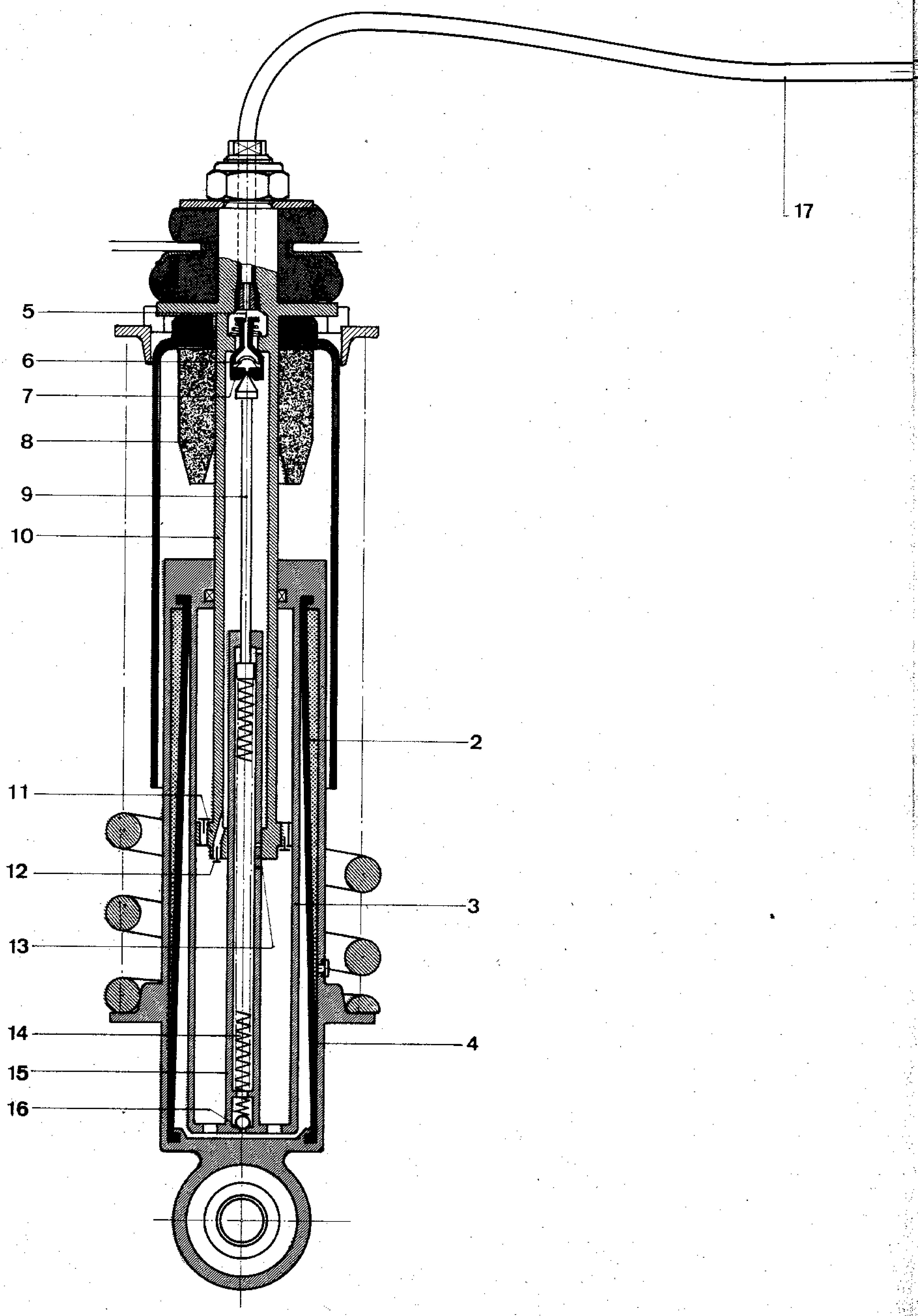
It is obvious that this solution is much more expensive, whereas the suspension unit is more vulnerable, because -if not treated carefully- the flexible tube may be damaged. Therefore we do prefer the normal execution with the reservoir in line with the piston rod.

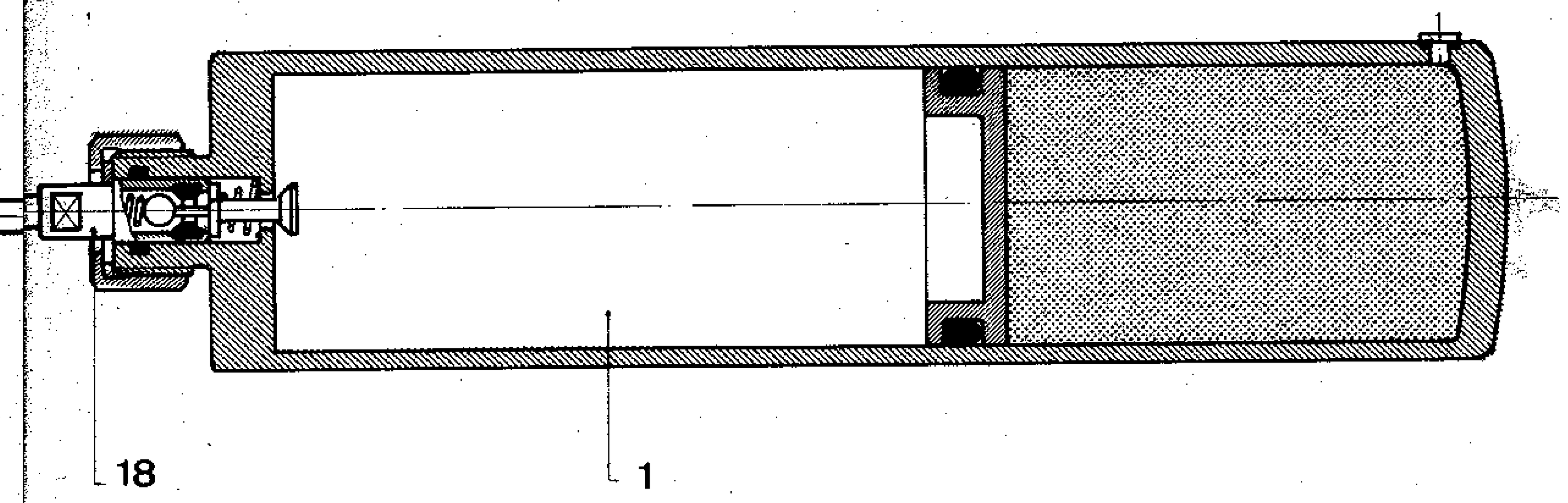
To submit a suitable offer for such a suspension unit the same details are required as those for the double acting suspension unit, as stated in the corresponding questionnaire (see the questions 1 up to 8 at page 10).

Principle Scheme of an auxiliary KONI Self Levelling Spring Unit
(Type 7110)



1 Low Pressure Chamber
2 High Pressure Chamber
3 Working Cylinder
4 Membrane
5 Return Flow Orifices
6 Filter
7 Suction Valve
8 Bumprubber
9 Levelling Rod
10 Piston Rod
11 Damper Piston
12 Delivery Valve
13 Bypass Orifices
14 Pump Plunger
15 Coil Spring
16 Relief Valve





Principle Scheme of an auxiliary KONI Self Levelling Spring Unit (Type 7110)

1 Low Pressure Reservoir
2 High Pressure Chamber
3 Working Cylinder
4 Membrane
5 Return Flow Orifices
6 Filter
7 Suction Valve
8 Bumprubber
9 Levelling Rod
10 Piston Rod
11 Damper Piston
12 Delivery Valve
13 Bypass Orifices
14 Pump Plunger
15 Coil Spring
16 Relief Valve
17 Nylon Tube
18 Disconnect Coupler

List of symbols used:

M = Mass (sprung)

m = mass (unsprung)

G = Weight (sprung)

C = Spring rate kg/cm (lbs/in)

N_e = Natural frequency (Hz)

N_{e_o} = unladen

N_{e_b} = laden (cycles/sec.)

f_{st} = static deflection

f_{dyn} = spring travel on bump

f_t = total deflection

k = shock factor

K = Load rate

D_{cr} = Critical damping factor kg/cm/sec. (lbs. sec/ft.)

P_o = pressure unladen kg/cm^2 (p.s.i)

P_b = pressure laden kg/cm^2 (p.s.i)

F = Piston surface cm^2 (sq.in)

F_p = pump surface cm^2 (sq.in)

V_h = Gas volume spring cm^3 (cub.in)

V_c = Gas volume correction spring cm^3 (cub.in)

V_o = Gas volume unladen

n = polytropic or adiabatic exponent.

i = lever ratio