

## Mechanical and Hydraulic Noise of Gear Pumps The Drastic Evolution Marked by “Continuous Contact” Pumps

### Part I

*Aspects tied to tothing, encapsulation and pulsation: how these problems are linked to pump design.*

*Analysis of the problems regarding external and internal gear pumps and the screw pump. Single contact, double contact and continuous contact. Wave form and power transported by the wave.*

*The drastic evolution marked by “continuous contact” pumps*

As everyone knows, today there is a lot of pressure on obtaining machines having low noise and vibration.

Hydraulic pumps, above all those for high pressure, entail noise and/or vibration levels that are often unacceptable for new applications.

Obviously the attempt to lower noise and pulsation has to come out of the analysis of their origin.

### Causes of noise in gear pumps

The sound emissions of gear pumps have two very distinct origins:

- mechanical noise
- hydraulic noise

#### Parameters influencing the emissions of mechanical noise

Mechanical noise is what one can expect from any pair of gears.

This does not at all mean that it is a simple aspect of the problem. On the contrary. The peculiarity of the pump gears make it more difficult than usual to tackle because the transmission needs are secondary to the hydraulic needs.

Taken for granted that the tothing geometry is optimised in view of the hydraulic needs, the most influencing aspects are technological in nature:

- 1) the precision class of the gears
- 2) the surface roughness, as entity and as “texture”
- 3) the determination of the advisable topological micro-corrections

- 1) The precision class of the gears makes a big impact by determining the maximum profile, helix, step and eccentricity errors. Obviously the improvement of the precision class entails use of better technologies and/or the refinement of existing technologies. I do not take it for granted that the problem has been solved; on the contrary, it is being studied and refined in order to get better quality at steady or dropping costs. All of us know that a gear wheel that is precision-ground (for example, in DIN 4-5 class) and subjected to honing has excellent characteristics, but it is not probable that the costs are compatible with the final product.
- 2) The entity and pattern of the surface roughness have a large impact on the high-frequency noise emissions. The roughness should be low, around Ra 0.3 (which is hard to get with grinding), and have a pattern non-perpendicular with the direction of the slipping speed. For example, the “texture” of the grinding roughness is unfavourable – above all that of the form grinder – whereas the “fish-bone” crossing generated by the shaving is clearly better. Honing as well offers favourable performance.
- 3) Topological micro-corrections are essential for correct operation of a gear. One significant parallel is depicted by a car’s tyres. None of us has ever seen a tyre made like a cylinder with sharp edges. In the same way the gears cannot be made satisfactorily making the

geometric solid that the kinematics would indicate. --

Just like the tyre has rounded edges and a swelling of the centre section, gears are not exactly involute helicoids. Both the involute and the side of the helicoids should diverge from the theoretical. The involute should be discarded at the beginning and end of the contact so as to allow for profile and step errors, as well as the step error caused by the tooth's flexures under load. In the same way "the helix", or the side of the tooth, must allow for the error of alignment of the axes, and of their inflexion and torsion.

### Parameters influencing the emissions of hydraulic noise

"Hydraulic" noise is generated by the pressure perturbations of the fluid going through the pump. This noise and/or vibration arises from three fundamental components:

- 1- the cavitation
- 2- the pressure peaks arising from the trapping of fluid between the top and bottom of the tooth
- 3- the "ripple", or the flow pulsation during meshing that causes a pressure pulsation

1)- The cavitation phenomena and at times the highly turbulent motion of the fluid can cause serious noise, in addition to structural damage, of the mechanical components. These phenomena are tied to the average speed of the fluid in the pipes and to the local changes of speed due to throttling and/or sharp changes of size of the pipes, in addition to the internal friction of the treated fluid (viscosity). These phenomena are normally fairly easy to control because they appear with very high viscosity levels and/or excessive fluid speeds, so they can be prevented by making the pumps operate at lower rotation speeds or by increasing the diameters of openings and pipes. On the contrary, sometimes they show up if the pump is running too slow and is unable to "draw" fluid. These problems become critical in particular cases, but are not a constant in gear pumps.

2)- The pressure peaks arising from trapping of the fluid between the top and bottom of the tooth are a serious problem that all involute tooth gear pumps have. Actually, there is a rotation phase when the fluid is no longer between the box and profile of the teeth, but only between the conjugate profiles. As you can see in the following figure, the geometry of the meshes makes it inevitable that during meshing the volume between the tooth top and bottom undergoes a change, a change that cannot be properly controlled through the intake and delivery openings radially made in the box because these volumes are found between the conjugate profiles and not between the profile and box.

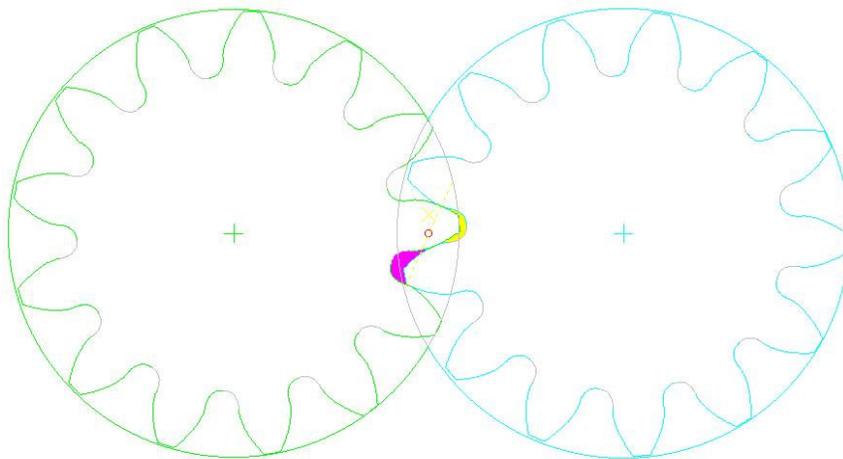
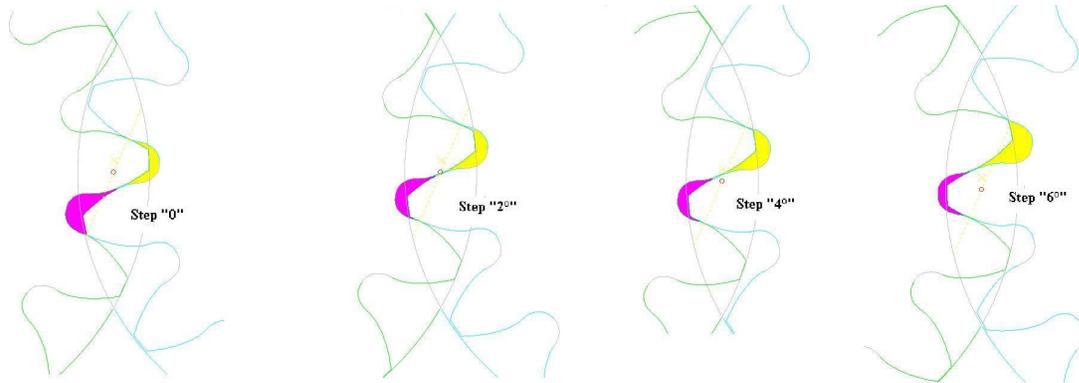


Fig. 1 - graphic simulation for a Z13 geometry made using ingw12 software

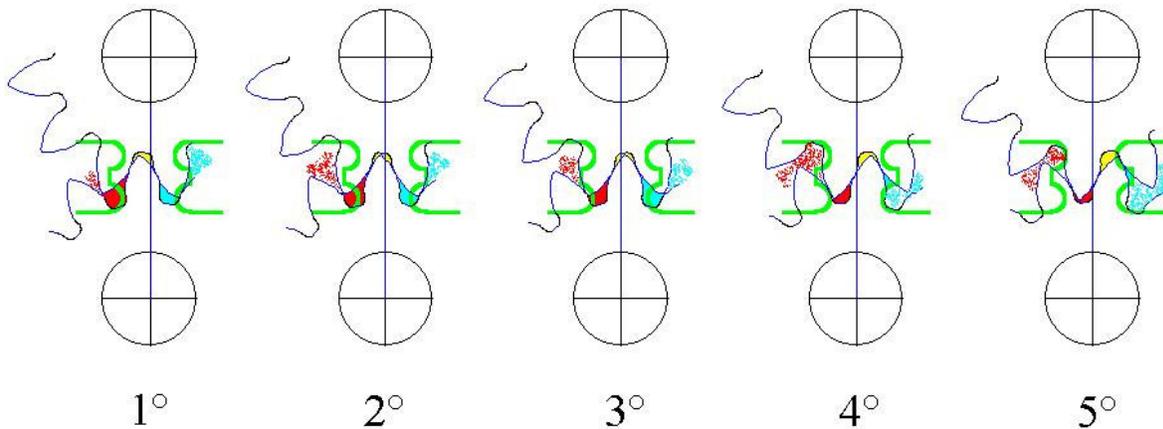


Volumes subject to changes not allowed by the substantial incompressibility of the fluid are seen in purple and yellow. The fluid is found in a volume defined by the conjugate profiles and by the lateral thrust washers. There is no radial communication towards the box.

Fig. 2 - graphic simulation for a Z13 geometry using ingw12 software in 2° steps

When this change is positive (intake phase), noise similar to cavitation is caused. Bubbles of steam upon the fluid pressure at the working temperature that are usually of few mbar are generated, so when this volume opens up towards the intake, the fluid fills these bubbles with violent implosions. When this change is negative (delivery phase), the poorly compressible fluid reaches instant, extremely high pressure values. The peaks in these transients are never as high as the theoretical expectation because the fluid passes through the available openings, although small, and the geometry of the pump (box, centre distance) changes elastically. In order cope with this problem entailing a large amount of noise, vibrations and excess mechanical stresses, drain pockets on the surfaces that axially delimit the pumps have been studied.

An example referring to a classic external gear pump follows.



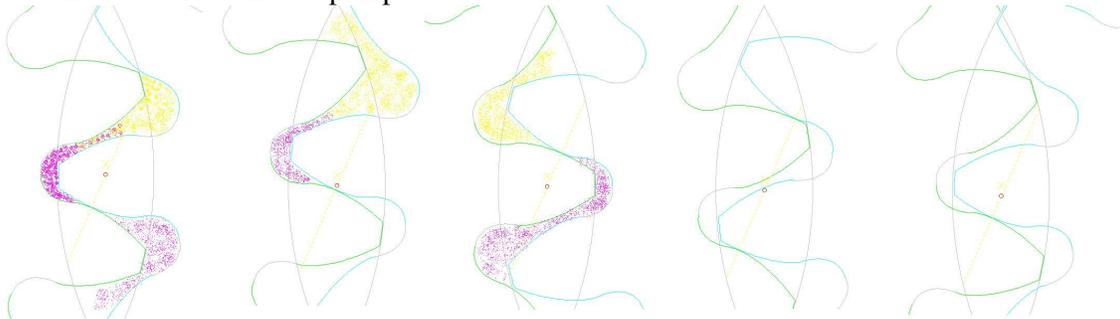
In the figure the intake area is coloured blue, the delivery red, and what remains encapsulated between the profiles is yellow. The full red, yellow and blue volumes are those subjected to changes that are partially allowed by the green drains or “pockets”, which are machined on the lateral faces (bushings) with axial thrust washer of the gears.

Fig. 3 – diagram of drain pockets machined on the axial pump delimitation surfaces

These pockets make the clearances (intake or delivery, respectively) communicate with the “trapped” volumes subject to change. The technical solution is undoubtedly valid, but cannot be “perfect” due to chains of machining tolerances and dynamic effects.

The upper theoretical limit of the size of the openings is the limit that involves a bypass between delivery and intake, with a loss of volumetric efficiency. If the situation is dynamic, this limit varies

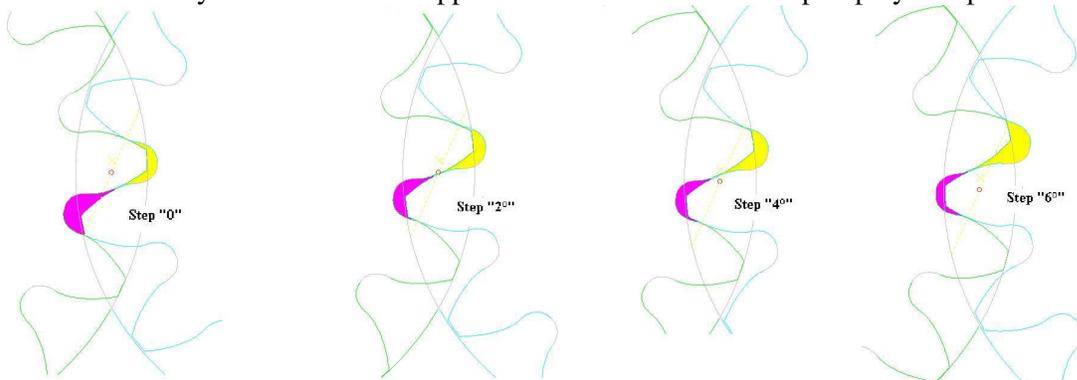
due to the fluid's inertia. Like what happens for explosion engines, the opening and closing "timing" can be optimised for just one given speed (and viscosity . . .). This timing in fact is not the "geometric" type that would allow opening to the bypass's limit, but presents slight advances in order to allow for the inertial effects. The optimum opening "advances" are logically variable with the pump's geometry, the rotation speed, the specific weight of the fluid and its viscosity. Another significant parameter is the meshing clearance in the traditional "single contact" pumps. All manufacturers have developed their own geometries and timings, but we can say that the technology is mature and therefore extremely optimised. The problem is noticed in the classic single contact external gear pumps, but is seen even more so in the double contact external gear pumps because of the better seal that the pulled flank carries out on the trapped volume. Indeed, the change of volume in the theoretically closed areas is possible not because of the very modest change of fluid volume, but by virtue of the narrow passages between gears and axial thrust washers, in addition to between gear and gear. The presence of a noticeable gap on the flanks not meshed clearly allows the overpressures to be considerably restricted, what is partially prevented in the so-called "double contact" pumps.



*It is clear how the presence of the clearance between the flanks not used for transmitting the torque makes the leakage of the fluid easier when there are changes of volume in the nominally closed areas, but in this way not completely trapped.*

Fig. 4 - encapsulation of the single contact external gear pumps - graphic simulation carried out using ingw12 software

Now I will show you what instead happens in the "double contact" pump by comparison



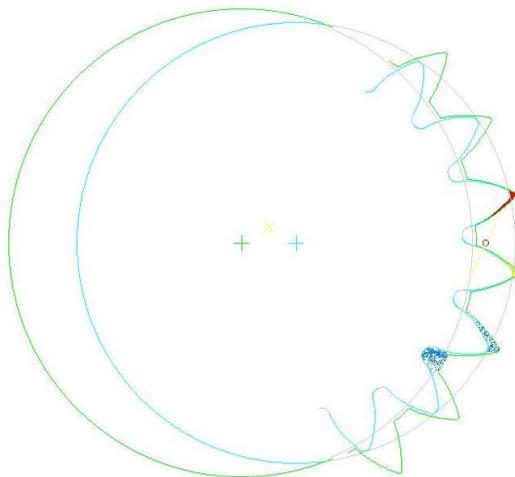
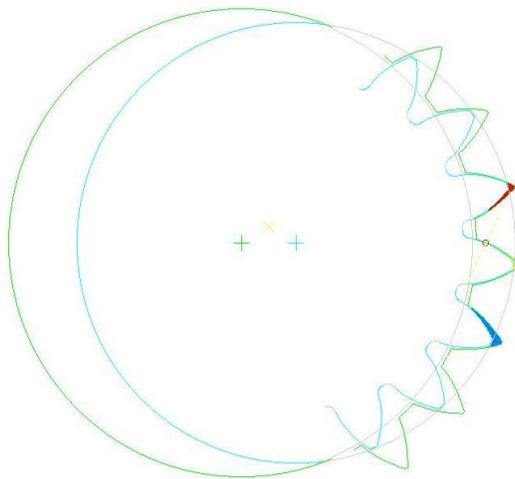
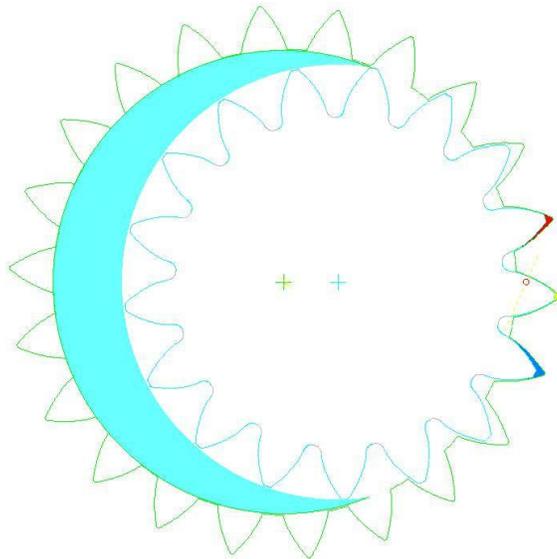
*The absence of clearance between the flanks not used for transmitting the torque makes the leakage of fluid difficult when there are changes of volume in the nominally closed areas, which appear completely trapped (except for the axial clearances and the modest clearance of the secondary flanks).*

Fig. 5 - encapsulation of the double contact external gear pumps - graphic simulation carried out using ingw12 software

- - Work carried out by Mario Antonio Morselli

Short description of the internal gear pumps.

The problem is less serious in internal gear pumps due to the smaller change of volume in play, but it is nevertheless present, even if control of the delivery and intake openings – by necessity axial – allows the role played by the previously illustrated "pockets" to be incorporated in them.



*It is evident in the figure how the small "trapped" area in yellow shows slight changes, whereas the blue (intake) and red (delivery) areas subject to greater changes can better be placed in communication with the respective openings, since the bypass danger is less. Maybe it is more proper to say that the yellow (trapped) area can be put in communication with the openings without bypass between intake and delivery when significant changes start to show up.*

Fig. 6 - encapsulation of the internal gear pumps - graphic simulation carried out using ingw12 software

The high profile overlap attainable with internal gear pumps is favourable from both the mechanical and the hydraulic noise points of view.

On the other hand, internal gear pumps have other problems constructive in nature tied to the presence of the so-called "lunette" (blue in the figure) and to the high slipping speed of the internal

crown that makes high rotation speeds problematic, in addition to the technological difficulty of economically making a high quality internal gear.

These pumps offer significant performance from the noise viewpoint only when designed with sophisticated compensation systems and built very carefully, so only the top of the range models allow their potentials to be tapped.

They allow even very high pressures to be attained with optimum efficiencies provided that the rotation speeds are not high and that the fluid is filtered very well. Pulsation is on rather low levels.

#### Short description of the triple screw pumps

The encapsulation problem is on the other hand almost completely overcome in the screw pumps owing to the different fluid transport method, which is parallel instead of perpendicular to the axes. The fluid would be encapsulated from the radial transport point of view, but with the axial transport method the phenomenon is irrelevant because there is a kind of three-dimensional “worm” whose section actually varies during rotation but that, although closed radially, is open axially.

As it is a question of “screws”, during rotation the figure is “translated” axially and this is what generates the oil’s transport. The section coloured in yellow in figure 7) is what I am referring to. The sections in red are the satellite screws, and that in blue is the main screw.

Screw pumps are usually very silent pumps with low pulsation and for medium – not high – pressures, and are suitable for operating with a deltaP of a few dozen bar. It is unlikely they can run at more than 40 bar.

Several special executions, for example by Settima Meccanica, have more than one meshed step and can reach 60 bar, although heavily paying the price in terms of volumetric efficiency.

One classic application example for these relatively unfamiliar pumps are hydraulic lifting systems (elevators, home-lifts, etc.) and lubrication devices and/or heat exchangers.

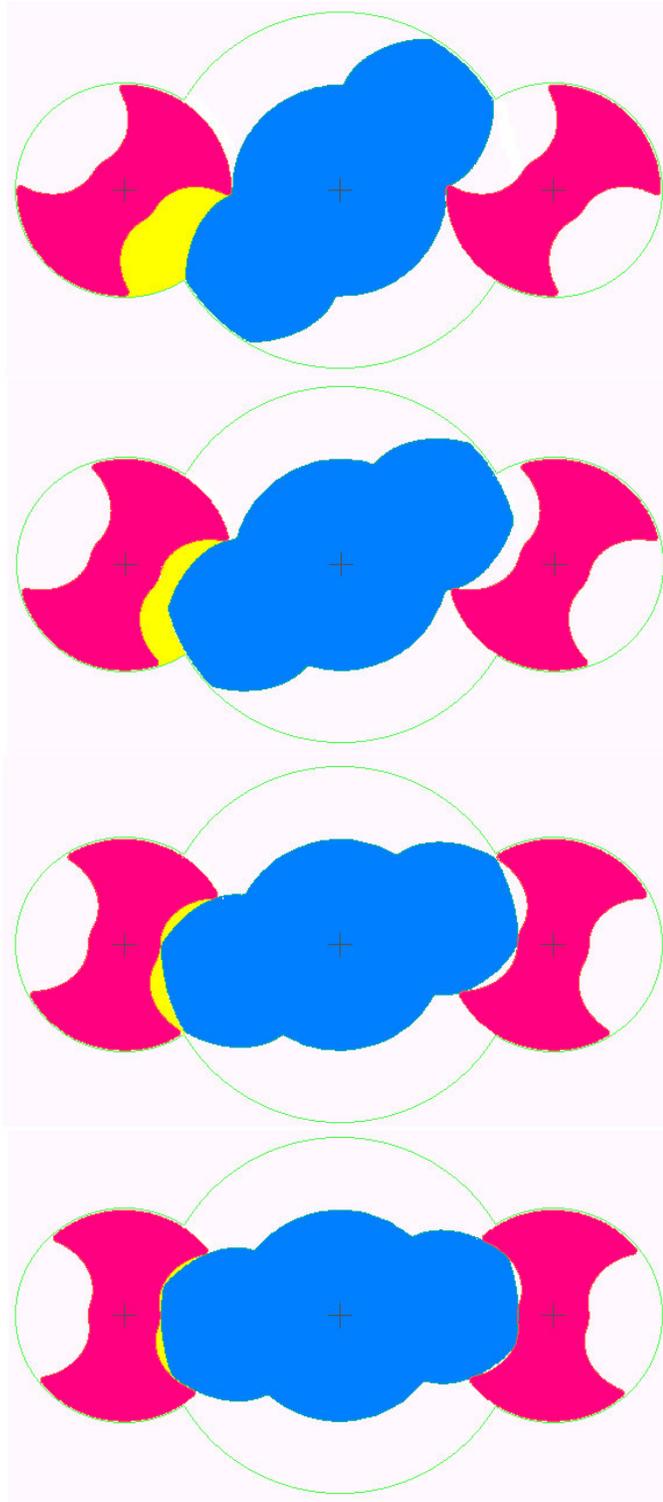
A proposal of subsequent rotation steps of the three screws is seen in the following figure.

Note that they could on the other hand be considered sections with different heights, as the screws are helical.

The area coloured in yellow does not represent an encapsulation of the fluid as the fluid’s transport is axial, so the area is “open” towards the passageways axially.

From this point of view it is irrelevant that the area be bounded partly by the box and partly by the profiles, or entirely by the profiles.

A small amount of disturbance is on the other hand caused by the change of area when it is “split”, like in the penultimate and last sequence. Settima Meccanica expressly patented a method for eliminating this problem based on a modification of the profiles at the axial ends of the screws.



*Fig. 7 – almost total absence of encapsulation of the screw pumps due to the axial mode of transport; significant example of Settima Meccanica.*

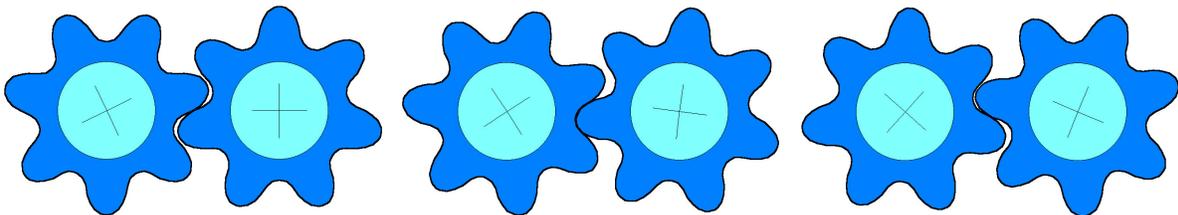


Fig. 8 Cutaway view of a triple screw pump, courtesy of Settima Meccanica

Here the three-dimensional course of the “worms” that are axially transported is clear.

Short description of the “continuous contact” pumps

The problem is solved in an even more sophisticated way in the new “continuous contact” pumps that the author designed and patented (\*). In this situation the transport flow is perpendicular to the axes like in the classic gear pumps, but the specially studied profile does not trap any volume while the profiles have a helical course. The aspect and the meshing methods make it a sort of hybrid among the gear and screw pumps.



*We can note from the sequence that at least from the practical point of view there is just one single point of contact between the profiles. There is no encapsulated volume between the profiles, and this completely eliminates the need for pockets on the axial thrust washers and any noise linked to this problem.*

Fig. 9 - Complete absence encapsulation of the “continuous contact” pumps

(\*) Patent pending in Europe, U.S.A., Canada, Japan

-- Work carried out by Mario Antonio Morselli

A particularly profound reader will note that the conjugate profiles do not seem to have enough profile overlap for transmitting the torque. I ask these readers to bear with me. They will find the completion of the description on the following pages. The helical course of the profile plays the role of completing the overlap, making a gentle transmission of movement and minimisation of the pulsation possible.



Fig. 10 - Aspect of the rotors-gears of a continuous contact pump  
These pumps are currently made and industrialized with the Continuum® trademark by Settima Flow Mechanism.

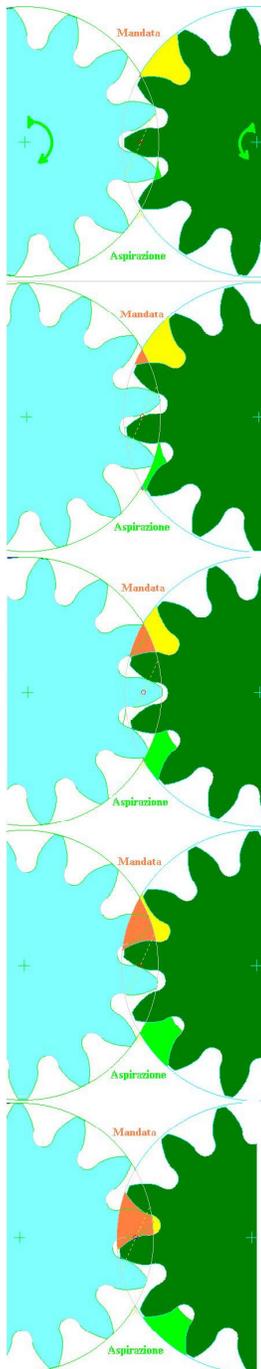
# Mechanical and Hydraulic Noise of Gear Pumps

## The drastic evolution marked by “continuous contact” pumps

### Part II

*Theoretical analysis of the flow pulsations, single contact, double contact and continuous contact pumps. Response of the pressure pulsations. Wave form and power transported by the wave. The drastic evolution marked by “continuous contact” pumps*

A pressure pulsation follows the flow pulsation during meshing (also called “ripple”), which is one of the major causes of problems tied to noise, vibrations and anomalous failures. What is the “ripple”, and where does it come from?



In order to fully understand the phenomenon, we have to concentrate on “how” a pump transfers the fluid from the intake throat to the outlet.

One of the clearest studies on this subject was that of the distinguished Prof. Claudio Bonacini, “On the Delivery of Gear Pumps”. Since I can’t quote the entire work, I will try to briefly illustrate some of the conclusions of this excellent essay.

Transfer of the fluid occurs when the volume of fluid between two teeth and the cylinder head “exits” the cylinder of the other gear (intake in light green) or is transferred in penetration into the cylinder of the other gear (delivery in orange). It is obvious that this volume in one case is generated (intake) and in the other is lacking (delivery).

Starting from above, we see the “complete” compartment of gear 2 (dark green) in yellow, which progressively enters the defining cylinder of gear 1 (blue). The portion that gradually enters is marked in orange. It is the volume that, as it runs short, is delivery. On the other side we see the progressive exit of the compartment that “fills up” in light green. It appears evident how the area vibrations are not linear with rotation. The effects of the drains machined on the axial thrust washers for controlling the changes of encapsulated volumes between the profiles are not evaluated in this diagram.

Fig. 11 - Method of transporting the fluid in the gear pump

Since the teeth are “discrete” entities, a volume transferred with a certain law lies with every tooth-compartment entity. Prof. Bonacini demonstrates that this law is a parabola for the involute geometry, and that other geometries (cycloidal for example) are worse because the law resting with them has more marked maximums and minimums.

It is also deduce that the lower the flow pulsation is, the higher the overlap because this allows the parabolas to be “overlapped”.

Below is a figure taken from the above-cited study of Prof. Bonacini that illustrates what has been stated.

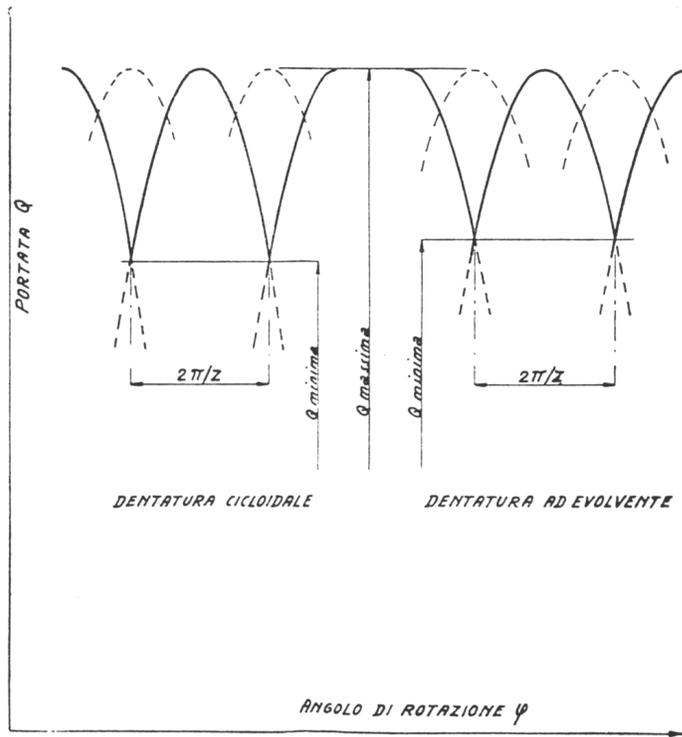


Fig. 12 – course of the flow depending on the angle of rotation  $\varphi$   
The course shown in the diagram comes from the following relation:

$$Q = \omega \cdot b \cdot (R_e^2 - R^2 - R \cdot \varphi^2 \cdot \cos(\theta)^2)$$

Where:

$Q$ = flow	$\omega$ = angular velocity = $d\varphi/d\tau$	$\varphi$ = angle of rotation
$b$ = band	$R$ = pitch radius	$R_e$ = external radius or of gear head
$\theta$ = angle of toothing pressure		$Z$ = number of gear teeth

The function  $Q(\varphi)$  represents, except the width of band  $b$ , the variation of the area that comprises the transported fluid in the section perpendicular to the axes  $Q(\varphi) = b \cdot dA(\varphi)/d\varphi$

A variety of solutions were put forward over the years so as to limit this problem.

The guiding principles of development emerging are those of building pumps with the maximum external radius possible (towards the limit of the pointed teeth), reduction of the angle of pressure (to the limit of the undercut), and an increase of the number of teeth to reduce  $\varphi_{max}$ , just like the adoption of double contact pumps (that is to say, with very small clearance on the flank not engaged in transmitting the torque) (Fig. 15).

This latter solution allows the  $\varphi_{max}$  to be cut in half, doubling the number of pulsations/rev.

Halving the  $\varphi_{max}$  reduces the amplitude of the pulsation by about 4 times (and this is the theoretical advantage).

In addition to the double contact pump, the other solution for getting the same theoretical result is the “duo” pump with two toothings that pump in parallel staggered by  $\frac{1}{2} * \phi_{max}$ .

I am convinced that the “duo” pump is virtually more effective in reducing the pulsation because the double contact situation is uncertain. In mechanics, “zero” clearance does not exist unless as an instantaneous and random transient, so there is either a slight clearance or a slight preloading. Slight clearance means a limited double contact effect, therefore not a complete doubling of the pulsation peaks. A slight preloading means important overloads on the pump’s components. In addition to this aspect, there is the major problem of wear. All of the wear phenomena entail a deterioration of the “double contact” situation.

From a purely geometric point of view, it is possible to assess the flow pulsation phenomenon and the most widely used parameter for assessing it is “irregularity” defined as:  
 Irregularity =  $(Q_{max} - Q_{min})/Q_{med}$ , where  $Q_{max}$  is the maximum value,  $Q_{min}$  the minimum value, and  $Q_{med}$  the average value of the flow ( $Q_{med}$  is the one that can be derived from the nominal displacement).

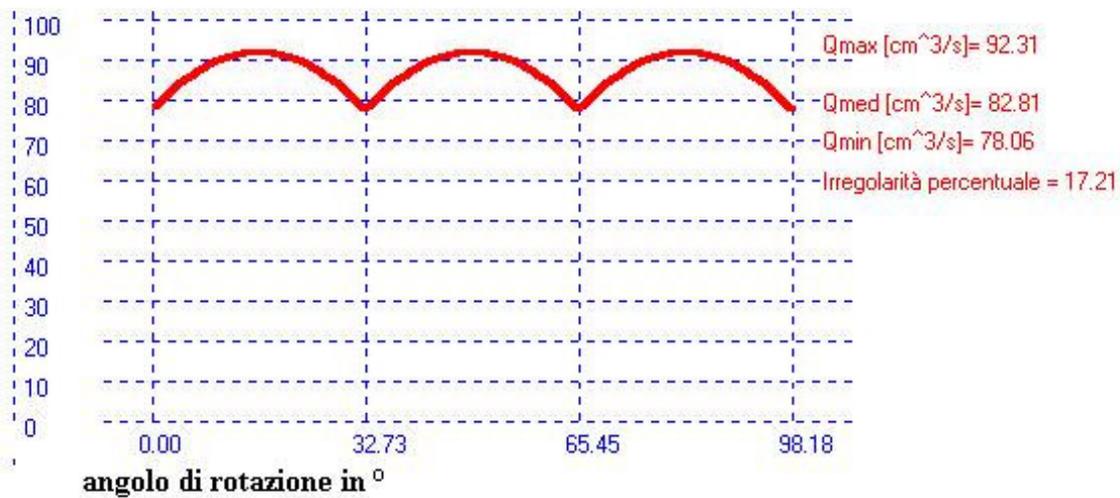


Fig. 13 Simulation of the behaviour of a Z11 single contact pump using ingW12 software – “bridge” between the drain pockets at the maximum value. “Trapped” fluid completely drained at intake.

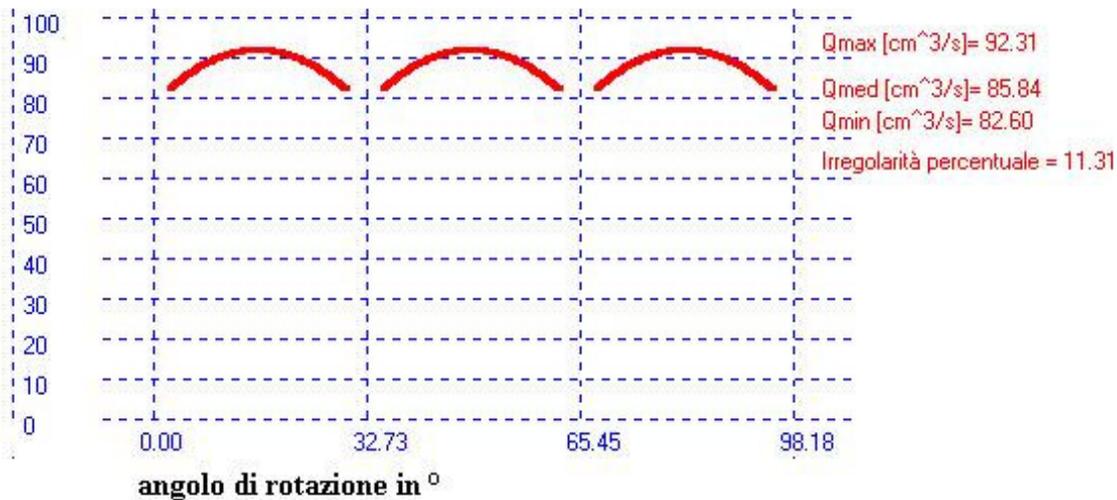


Fig. 14 Simulation of the behaviour of a Z11 single contact pump using ingW12 software – “bridge” between the drain pockets at an optimised value. Recovery of a part of the “trapped” fluid at the delivery.

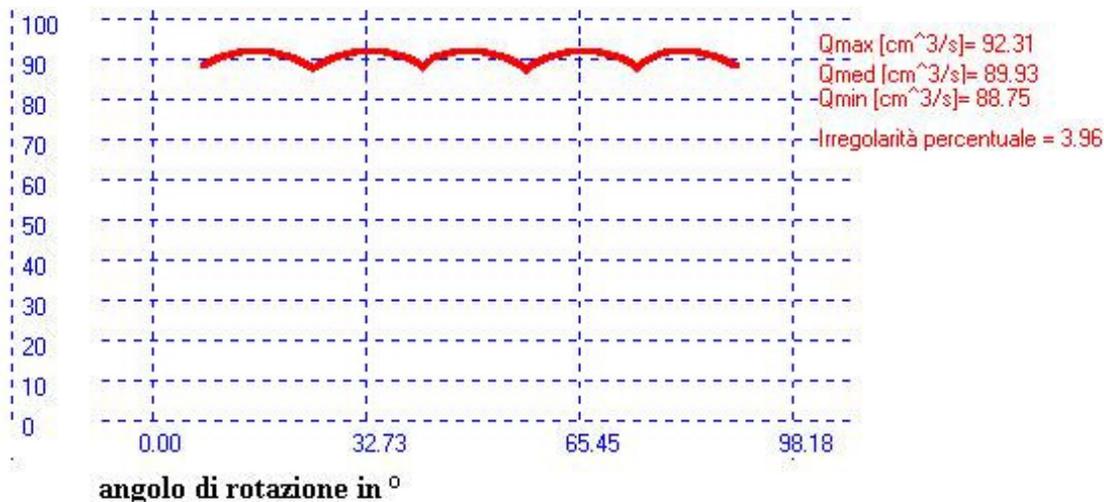


Fig. 15 Simulation of the behaviour of a Z11 double contact pump using ingW12 software – “bridge” between the drain pockets at a minimum value. Complete recovery of the “trapped” fluid at the delivery. The simulation cannot be faithfully reproduced in practice because the hypotheses are never achieved. The real situation of a double contact pump proves to be an average between Fig 13 and Fig 14.

In looking at these theoretical analyses, what further possibilities are there for reducing pulsation and noise?

Some have developed external helical gear pumps so as to get a softer meshing and to dampen the pulsation by overlapping staggered ripple waves.

These pumps have a few problems. First of all, if the helix angle is fairly high so as to provide significant results, the space between the tooth bottom and head becomes a three-dimensional “worm” that puts intake and delivery in bypass.

As far as the first aspect of the problem is concerned, two bypass development paths have been taken:

- keep a sufficiently low helix angle so as to avoid the bypass
- keep a tooth proportioning such as to have transversal overlap greater than 2 in order to prevent the three-dimensional “worm” from becoming a true communication;

Both solutions have given poor satisfaction. The first has led to pumps with a modest reduction of pulsations. The second has considerably complicated control of the drain pockets on the thrust washers, a problem that, as we have already seen, does not have a satisfactory solution even for the simpler pumps.

So the continuous contact pump lends itself to simultaneously solving the two problems: elimination of the encapsulation and a drastic reduction of pulsation. The  $dA(\varphi)/d\varphi$  function shown above at this point no longer presents discontinuity. Meshing is continuous, and so is the function.

As for the mathematical discussion of the helical pumps' flow, please see Appendix 1.

### Pressure response to the changes of flow of the gear pumps

The pressure response of the circuit system to the change of flow logically depends on its characteristics, but the pressure readings with high frequency sampling of the external gear pumps show a course with a great amount of conformity with the theoretical analysis.

Actually, the change of flow (with negligible compressibility fluids) entails an equivalent variation of speed, hence acceleration. The fluid has a mass, so in order to give it acceleration a force is needed, hence pressure. In our case it is a change of pressure as to the constant circuit pressure. The pressure pulsations cause noteworthy vibrations and noise in the user systems, sometimes very serious if the system's frequencies are close to the stressing frequencies. The pulsations therefore reduce the operator's comfort and can cause structural breakdowns of various elements, such as pipes, joints and filters.

The major problem is generally in any case the sound emission tied to propagation of the disturbance along the pipes. Below is a survey carried out on a Z12 double contact external gear pump.



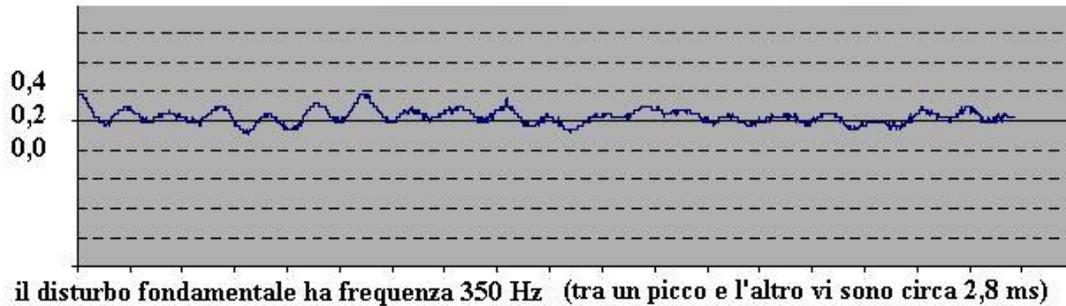
**il disturbo fondamentale ha frequenza 600 Hz (tra un picco e l'altro vi sono circa 1,7 ms)**

*As you can see in the figure, the pulsation in the test circuit was around 0.8-1 bar from peak to valley. The valleys have a peak with pronounced discontinuity of the first derivative and another very high frequency disturbance is found overlapping the carrier wave.*

Fig. 16 – Reading of the pressure pulsation of a good quality “double contact” external gear pump 20cc/rev at 1500 rpm, 150 bar in delivery. The fluid is a common mineral-base hydraulic oil ISO 46 at 50 °C – sampling at 40 kHz.

In other circuits, or with other pumps, the pulsation can also be considerably higher. Mediocre quality single contact pumps can reach +/- 4-5 bar. Average values for pumps of this size are +/- 1.5-2 bar.

Comparison with a continuous contact pump in the same working conditions is sensational.



*As you can see in the figure, the pulsation in the test circuit is about 0.2-0.3 bar from peak to valley. The valleys have peaks and valleys without discontinuity of the first derivative and the higher frequencies are practically absent. Disturbance is close to the sensitivity limit of the piezoelectric pickup used.*

Fig. 17 – Reading of the pressure pulsation of a continuous contact pump  
20 cc/rev at 1500 rpm, 150 bar in delivery. The fluid is a common mineral-base hydraulic oil ISO 46  
at 50°C – sampling at 40 kHz.

The magnitude of the measured ripple to a large extent depends on the user circuit.

As confirmation of what is stated above, there are precise regulations for measuring pulsations that prescribe the stiffness of the pipe downstream of the pump, its length, where to position the pressure transducer, and how to build the throttling. All of this makes the tests certainly comparable, but risks leading us along the wrong track as far as what then awaits us in the field is concerned. The user circuits rarely have steel pipes of a set length and needle throttle values without elastic elements. It is for this reason that I believe it useful to perform pulsation and noise tests on several circuits having characteristics similar to those of the end use.

#### Waves, frequency, transported amplitude and power, acoustic emissions.

Apart from the “optical” effect of great impact, what can be said about the compared measurements?

Without wanting to go into excessively technical dissertations, what is immediately glaring is that the continuous contact pump generates a disturbance that:

- 1) has much less amplitude
- 2) has a lower frequency
- 3) does not present sharp reversals
- 4) is free from high frequency components

Now the “average power” of a wave in its period is proportional to the square of the amplitude and to the square of its frequency.

It is evident that the average power of the ripple generated by the continuous contact pump is more than one size lower than that of the compared external gear pump.

Actually, the disturbance’s wave – at least that of a traditional gear pump – does not look much like a sine curve and contains very high frequency components. Moreover, the discontinuity of the first derivative in the shape of the flow disturbance, which is reflected on the shape of the frequency disturbance, depicts a sort of “collision” whose effect depends a lot on the response of the downstream circuit.

- - Work carried out by Mario Antonio Morselli

We are used to thinking that the noise generated by the better quality pumps is primarily a hydraulic noise. That is why we can expect an enormous improvement in the emission of the continuous contact pump over the traditional gear pump.

If we want to make a rough comparison by placing “1” as the value of the standard external gear pump, we can say that the pulsation’s power changes like this:

said  $P_{s\_R}$  the average power of the disturbance generated by a single contact external gear pump, we put  $P_{s\_R} = 1$

said  $P_{d\_R}$  the average power of the disturbance generated by a double contact external gear pump will be a disturbance with the following proportions on the foregoing:

amplitude nominally reduced 4 times, actually 3 time  
frequency doubled

$P_{d\_R} = P_{s\_R} * (1/3)^2 * 2^2 = (1/9)*2 = 0.44 \Rightarrow$  the power has dropped by a factor of 2.3

said  $P_{c\_R}$  the average power of the disturbance generated by a continuous contact gear pump will be a disturbance with the following proportions on the foregoing:

amplitude reduced 3.5 times (from reading)  
frequency decreased by  $600/350 = 1.7$  times

$P_{d\_R} = P_{s\_R} * (1/3.5)^2 * (1/1.71)^2 = 0.028 \Rightarrow$  the power has dropped by a factor of 35!

Based on the foregoing considerations, what could be said with regard to the acoustic emissions generated?

A rough line of reasoning (which I hope the sector’s experts will pardon) could be this: if the power of the source of sound is proportional to the power of the hydraulic disturbance, then one can say that the ratios between average disturbance power and those of the acoustic emissions are within the same ratios. Having established the following symbols:

$P_{s\_A}$  = acoustic power of the single contact pump

$P_{d\_A}$  = acoustic power of the double contact pump

$P_{c\_A}$  = acoustic power of the continuous contact pump

$P_{rif}$  = acoustic power of reference

$L_{p-s}$  = level of sound pressure in dB of the acoustic emissions of the single contact gear pump;

$L_{p-d}$  = level of sound pressure in dB of the acoustic emissions of the double contact gear pump;

$L_{p-c}$  = level of sound pressure in dB of the acoustic emissions of the continuous contact gear pump;

The sound pressure levels should be in these proportions between them.

$$\begin{aligned} L_{p-d} &= 10 * \text{Log}_{10}(P_{d\_A}/P_{rif}) = 10 * \text{Log}_{10}(P_{s\_A}/P_{rif} * P_{d\_A}/P_{s\_A}) \\ &= 10 * \text{Log}_{10}(P_{s\_A}/P_{rif}) - 10 * \text{Log}_{10}(P_{s\_A}/P_{d\_A}) \end{aligned}$$

if  $P_{s\_A}/P_{d\_A} = P_{s\_R}/P_{d\_R} = 2.3$  then

$$L_{p-d} = L_{p-s} - 10 * \text{Log}_{10}(2.3) = L_{p-s} - 3.5 \text{ dB}$$

So the double contact pump should generate 3.5 dB less than the single contact pump.

$$\begin{aligned} L_{p-c} &= 10 * \text{Log}_{10}(P_{c\_A}/P_{rif}) = 10 * \text{Log}_{10}(P_{d\_A}/P_{rif} * P_{c\_A}/P_{d\_A}) \\ &= 10 * \text{Log}_{10}(P_{d\_A}/P_{rif}) - 10 * \text{Log}_{10}(P_{d\_A}/P_{c\_A}) \end{aligned}$$

if  $P_{d\_A}/P_{c\_A} = P_{d\_R}/P_{c\_R} = 35$  then

$$L_{p-c} = L_{p-d} - 10 * \text{Log}_{10}(35) = L_{p-d} - 15.4 \text{ dB}$$

So the continuous contact pump should generate 15 dB less than the double contact pump!

Actually, this analysis does not bear in mind the source of mechanical noise, which is lower, but it exists and becomes increasingly influential the lower the hydraulic source is.

Below are a few measurements in dB of 20 cc/rev pumps (actually this is acoustic power, not acoustic pressure) in a rather common condition: 1500 rpm / 150 bar / hydraulic mineral oil ISO-VG 46 at 50°C

	Single	Double
	150 bar	150 bar
1500 rpm	68.16	66.51

The difference measured is about 1.5 dB, so the improvement is considerably less than the assumed value.

	Double	Continuous
	150 bar	150 bar
1500 rpm	66.51	59.11

The difference measured is about 7.5 dB, so here as well the improvement is considerably less than the assumed value, but it is still highly significant.

It is evident that the result in absolute terms is extremely remarkable, just as it is evident that by lowering the sound level by so much the noise of mechanical origin becomes perceptible.

I am showing as an example a table with assumed mechanical noise levels to be added to the hydraulic noise in the proportions that come out of the measured ripple:

dB hydr	dB mech	dB tot		
67.5	59.0	68.1	'single contact'	dB tot values measured at 150 bar and 1500 rpm with oil ISO-VG 46 at 50°C background noise < 40 dB
63.9	63.0	66.5	'double contact'	
48.4	58.8	59.1	'continuous contact'	

The indicated noise levels meet the following conditions:

- the total noise has the same acoustic power measured
- the hydraulic noise has powers in the ratios derived above

The mechanical noise is what meets the above-mentioned conditions, and seems to be definitely plausible. A mechanical noise of the double contact pump considerably higher than the other configurations is logical because the meshing conditions are unfavourable from the mechanical point of view. On the other hand, it is interesting to observe that on the acoustic emission levels of the continuous contact pump it is no longer true that a large part of the acoustic power comes from hydraulic problems, but instead the contrary is true. Once again I wish to point out that the acoustic observations made above are rough, and are just indicative. I mentioned them because even if they are not strict, they provide broad indications that allow us to get our bearings amidst the measurements and assess the proportions between the magnitudes involved.

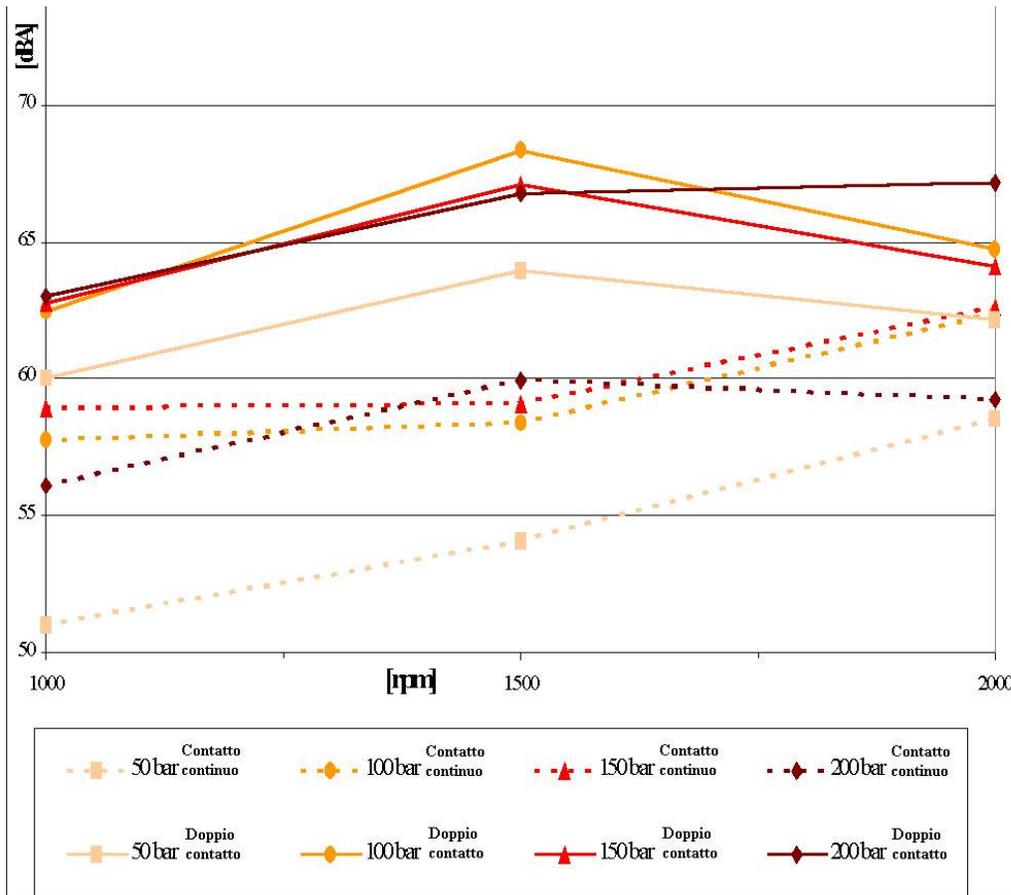


Fig. 18

Compared measurements of sound pressure in scale "A" for two 20cc/rev pumps with oil ISO-VG 46 at 50°C  
Background noise < 40 dBA

## Appendix 1

Short discussion of the mathematical description of flow in helical pumps  
Aside from the type of conjugate profile

The pump is made up of infinite slices of infinitesimal thickness  $dh$  high.

If we call the area of the infinitesimal "slice" containing the fluid  $A$ , the flow depends on the variation of  $A$  over time, but  $A = A(h, \varphi(t))$  and the angle of rotation  $\varphi = \varphi(t)$ ; so the integral along the height of area "b" of the infinitesimal flows  $(\delta A(h, \varphi(t)) / \delta h) * dh$  that lie with the individual "slices" of infinitesimal thickness involve this expression as they are put out of phase at an angle:

$$Q(t) = \delta \left( \int_{\varphi}^b \delta A(h, \varphi(t)) / \delta h * dh \right) / \delta t$$

whose variations with  $\varphi$  are lower than in the straight gear pump because they are indirect.

Naturally the expression for straight gear pumps is simplified because the function

$A = A(\varphi(t))$  becomes independent from  $h$ , and thus "extractable" from the integral, and the partial derivatives become simple derivatives.

So this expression  $Q(t) = \delta \left( \int_{\varphi}^b \delta A(\varphi(t)) / \delta h * dh \right) / \delta t$  is simplified bearing in mind that:

-- Work carried out by Mario Antonio Morselli

$$\int_0^b dh = b = \text{width of area}$$

and  $\int_0^b (\delta A(\varphi(t))/\delta h) * dh = A(\varphi(t))$  as A is independent from h

$$Q(t) == d( A(\varphi) * \int_0^b dh ) / dt = d( A(\varphi) * b ) / dt = b * dA(\varphi)/d\varphi * d\varphi/dt$$

Where  $dA(\varphi)/d\varphi$  is the function found by Bonacini, which overlapped with the various teeth in simultaneous meshing provides the diagrams from Fig. 13 to Fig. 15, and  $d\omega/dt$  is the angular velocity.

The problem is now posed in these terms:

“what are the conjugate continuous contact profiles that involve a  $dA(\varphi)/d\varphi$  function such that

the integral  $\int_0^b \delta A(h,\varphi)/\delta h * dh$  is constant with “t”, as  $\varphi = \varphi(t)$ ?”

In order to be able to best analyse the problem, we have to go by the concept according to which the section of a helicoid found following movement along the axis (variation in h) of the helix is equivalent to a rotation around the axis (variation of  $\varphi$ ).

The relationship binding the two variables is that of the helix, which with a movement along the axis equivalent to the pitch of the helix makes a  $2*\pi$  rotation; said pz being the helix pitch, we will have “b” expressed in terms of  $\varphi$ , i.e. angle of rotation becomes:

$$\varphi(b) = (b/pz) * 2 * \pi \quad \text{or: } b = pz * \varphi(b) / (2 * \pi)$$

in the same way for h and dh  $h = pz * \varphi / (2 * \pi)$  and thus  $dh = pz * d\varphi / (2 * \pi)$

Therefore, the variable “ $\omega$ ” can be substituted for h and the operations on “h”.

And so the integral  $\int_0^b \delta A(h,\varphi)/\delta h * dh$  becomes:  $\int_{\varphi_0}^{\varphi_0 + \varphi_b} \delta A(\varphi)/\delta \varphi * d\varphi = A(\varphi_0 + \varphi_b) - A(\varphi_0)$

The integral above is constant with  $\varphi_0$  (i.e. it is independent from the position) if A is a periodic function with  $\varphi$  of period  $\varphi_b$ : a very simple example is the sine function,  $\text{sine}(\varphi) = \text{sine}(\varphi + 2*\pi)$ .

The definition of conjugate profiles that meet the conditions set out above is a lot less simple.