Reducing Flow Pulsation with the Floating Cup Pump: Theoretical Analysis.

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SYNOPSIS

With the recent introduction of the floating cup (FC) displacement principle, axial piston units can be realised using low cost mass production techniques instead of the more machine shop type of production techniques used to date. As a consequence, a far larger number of displacement volumes can be realised at the same or lower unit costs. An obvious advantage of a larger number of pistons is the reduction of flow pulsation from the unit.
This paper focuses on this flow pulsation. In the ideal theoretical case that two out of phase halves of an FC pump directly deliver to the pump outlet, the odd harmonics in the 12 piston based flows from each pump half, will cancel out totally. In reality however, both sides are connected by a duct of finite length. The geometry of that duct and the location of the pump outlet, can significantly influence both the 'external' flow pulsation – at the pump outlet – and the 'internal' flow pulsation – in the duct. The paper shows that if the outlet duct is arranged symmetrical with respect to the two pump halves, perfect cancellation of the odd harmonics is indeed reached but only at the cost of high resonant pressure peaks within the duct. These pressure peaks might cause pump noise. They can be attenuated by either arranging the outlet in a non-symmetrical position or by adding Helmholtz resonators to the duct. Both options were studied; both show potential.

1 INTRODUCTION

At its start, the development of the Floating Cup (FC) axial piston displacement principle [1] was driven by the need for a positive displacement principle that could be used as the basis for an Innas Hydraulic Transformer (IHT). This displacement principle should at least match the high efficiency and power density of conventional axial piston displacement units. For application in the IHT however, it should also allow for a far larger number of pistons than conventional bent-axis or swash-plate units can offer, but at the same or lower production costs. In the FC displacement principle these seemingly conflicting demands were matched by adopting a design that it is fit for low-cost mass production techniques. By nature, conventional axial piston units have to be made using more expensive machine-shop kind of production techniques.

It was soon clear that the characteristics of the FC rotary group would be just as beneficial for pumps and motors. Consequently, the main development effort for the FC displacement principle was shifted to a 24 piston, 28 cm^3 fixed displacement pump [2]. This pump type has been the development work horse for the FC displacement principle.

This paper focuses in detail on one of the important advantages of the FC principle when applied to pumps: the reduction of the output flow pulsation due to the larger number of pistons. In section 1, the FC displacement principle is briefly described, referring to the 24 piston, 28 cm^3 fixed displacement pump prototype.
Section 3 compares the theoretical output flow pulsation of this FC design to that of a conventional 28 cm^3 bent axis pump with 7 pistons. In the FC pump, the internal ducting and the location of the outlet can influence the internal and external flow pulsation. Consequently, the influence of these parameters has been studied. Section 4 describes the calculation tools that have been used for this study. Section 5 presents the results for a symmetrical arrangement of the outlet connection. Section 6 shows how a non-symmetrical arrangement of the outlet duct or the addition of Helmholtz resonators, may improve the internal pressure pulsation in the FC pump. Finally, section 7 presents the conclusions.

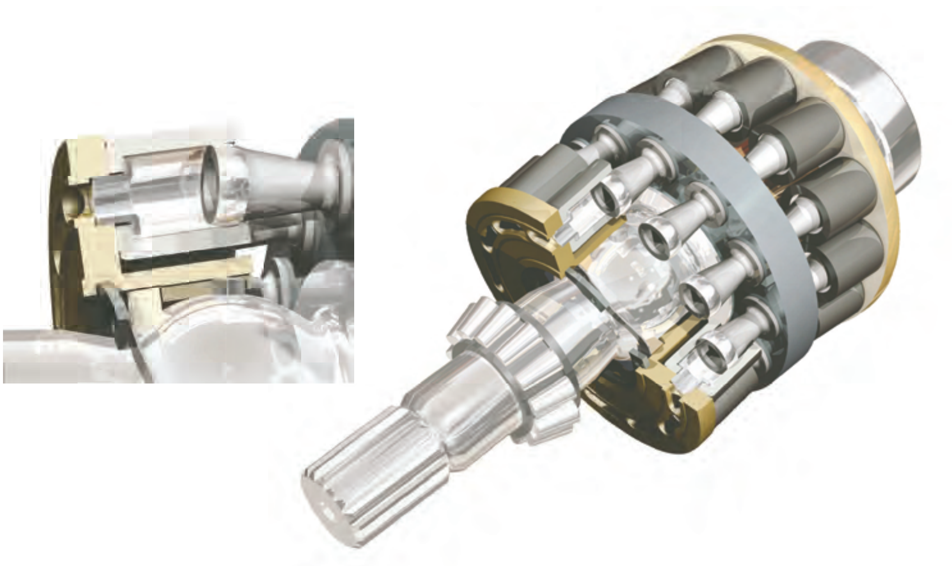

Figure 1: The rotary group of the 28 cm^3 Floating Cup pump prototype

One remark should be made here: this paper focuses on the pump's outlet side only. The inlet side of the floating cup pump was also analysed but as it proved to be less critical for the FC pump design, it is not treated in this paper.

2 THE FLOATING CUP DISPLACEMENT PRINCIPLE

Figure 1 shows the rotary group of the 28 cm^3 Floating Cup pump prototype. In this Floating Cup design, a rotor is fixed rigidly on to the axle and 12 two-sided pistons are fixed rigidly in the rotor. Both ends of each piston are spherical, each end has its own cup-like cylinder. This amounts to 24 'cups' in total, grouped in two sets of 12 at each side of the rotor. All cups on one side of the rotor rest on a common 'barrel plate'. Each barrel plate runs on the inclined face of a port plate (not shown in figure 1) and this ensures that the pistons move relative to the cups and pump oil.

A hole in the bottom of each cup connects to a corresponding hole in the barrel plate. The hole in the cup and the seal land around it are carefully tuned to give a resulting hydrostatic force that is large enough to press the cup down on the barrel plate. The cups are thus free to move over the barrel plate and this 'floating' motion provides the degree of freedom which allows the cups to follow the ellipsoidal track which is the projection of the piston circle on the barrel plane.
In each cup, a clamping screw ensures that the cups do not break out at high rotational speeds and low internal pressures.
The barrel plates are mounted on spherical joints on the axle. In each joint, a simple synchronising pin forces the barrel plate to rotate nearly synchronously with the axle.

In order to create effectively 24 pumping elements from the 12 two-sided pistons, the commutation of one pump half has been off-set to that of the other half. This has been achieved by giving the port plate planes an angular distance of 15 degrees along the main axis direction (figure 2).

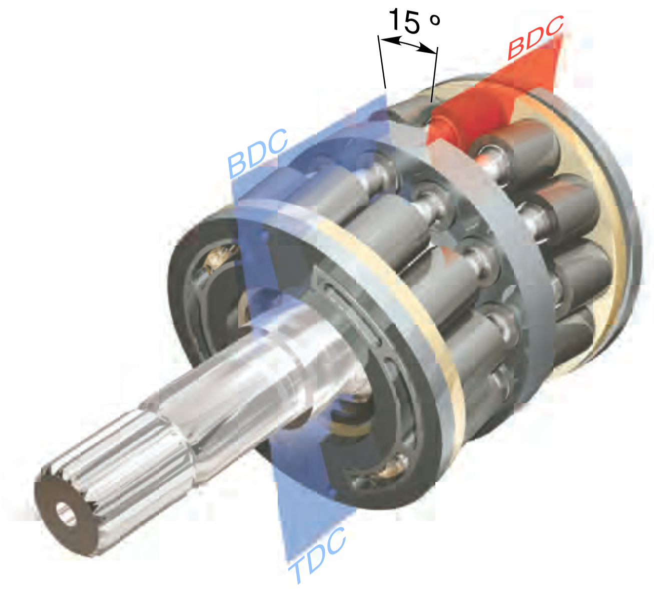

Figure 2: Off-setting the pump halves by 15 degrees

The flow from the 12 pistons at one rotor side and the flow from the 12 pistons at the other rotor side are joined in an internal manifold, which leads to the pump outlet. The ducts inside the pump are shown in figure 3.

If the ducts in both sides of the pump are identical and if the outlet of the pump is connected to

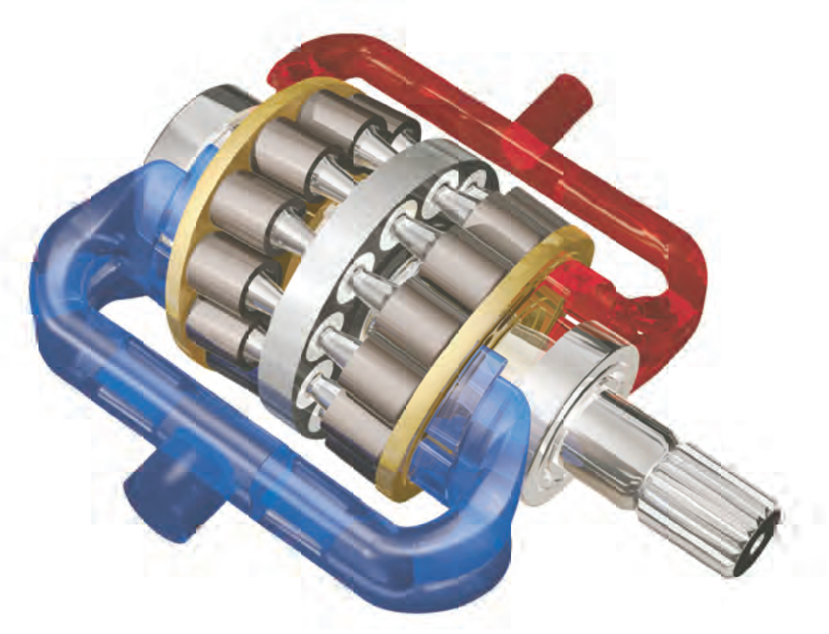

Figure 3: The ducts inside the 28 cm^3 Floating Cup pump prototype

the middle of the connecting duct, all odd harmonics in the 12 piston based flow from one side of the pump will be cancelled by the odd harmonics in the flow from the other side. In this case, the flow pulsation at the outlet of the pump will be analogous to the flow from a one-sided 24 piston pump.

3 THEORETICAL OUTPUT FLOW PULSATION

In order to assess the benefits of increasing the number of pistons, the outlet flow ripple generated by a 28 cm^3 FC pump with 24 pistons has been compared to the flow ripple generated by a conventional bent axis unit with 7 pistons (Bosch-Rexroth A2FO28). The simulations have been performed using the AMESIM software package and are based on displacement volumes which empty to or fill through the flow areas formed by the interface between the ports in the barrel and the kidneys in the port plate. Each pump's inlet and outlet are modeled as constant pressure sinks, which are assumed to extend from the outside world into the pump, into the respective kidneys.
For the A2FO28 model, measured values were used for stroke and bore of the pistons, for the dead volumes, for the angles of precompression and pre-expansion and for the geometries of the corresponding grooves. For the FCP28, these parameters were taken from the prototype design.

Rotational speed sweeps were calculated, the speed changing from 1000 to 6000 rpm in a total time of 0.4 seconds. In figure 4 the instantaneous flows coming out of the two different pumps for a rotational speed of 3000 rpm and a constant pump outlet pressure of 35 MPa are compared.

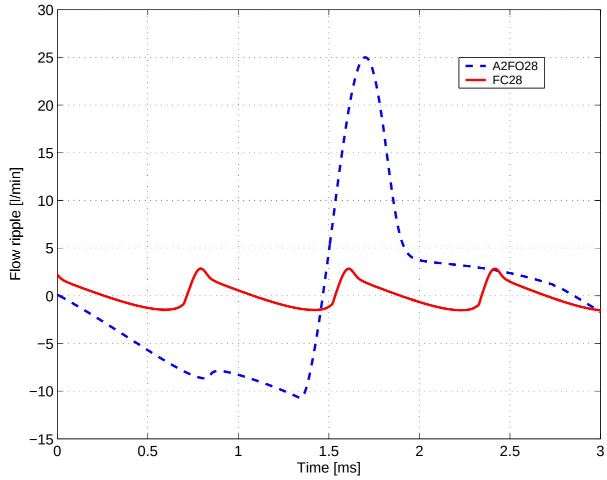

Figure 4: Comparison of the flow ripple at 35 MPa and 3000 rpm

The peak to peak amplitude of the dynamic part of the flow is significantly lower for the FC pump.

The calculated dynamic output flows from the two pumps were also compared in the frequency domain. To this end, an FFT of the time domain flow dynamic signal has been calculated. Since the information given by the frequency spectrum of the flow ripple is only relevant at the frequencies of the harmonics, the FFT of the time domain data has been calculated on a cycle by cycle basis. That is, the time length T of the sample has been reduced as the rotational speed increased, keeping the following relationship to the frequency f_1 of the first harmonic:

$$T = \frac{1}{f_1} \tag{1}$$

In figure 5 the frequency spectra at 3000 and 5000 rpm for both the conventional pump (left) and the new FC pump (right) are compared.

The difference is striking. Not only the amplitudes of the predominant harmonics for the FC pump are dramatically lower than those of the conventional pump, but also their frequencies are higher, which implies that for a given frequency range the number of harmonics significantly contributing to the flow ripple, is lower for the FC pump. The benefit increases as the rotational speed increases. At 5000 rpm there are 5 harmonics of the conventional 7-piston pump that contribute significantly to the ripple below 5000 Hz, while for the FC 24-piston pump there are only 2.

In order to get a measure of the total flow ripple at each rotational speed the root mean square (RMS) value of the flow ripple has been calculated for frequencies up to 10 kHz using:

$$Q_{rms}(n) = \sqrt{\frac{1}{2}\sum_{h=1}^{n_h} |Q_h(n)|^2} \tag{2}$$

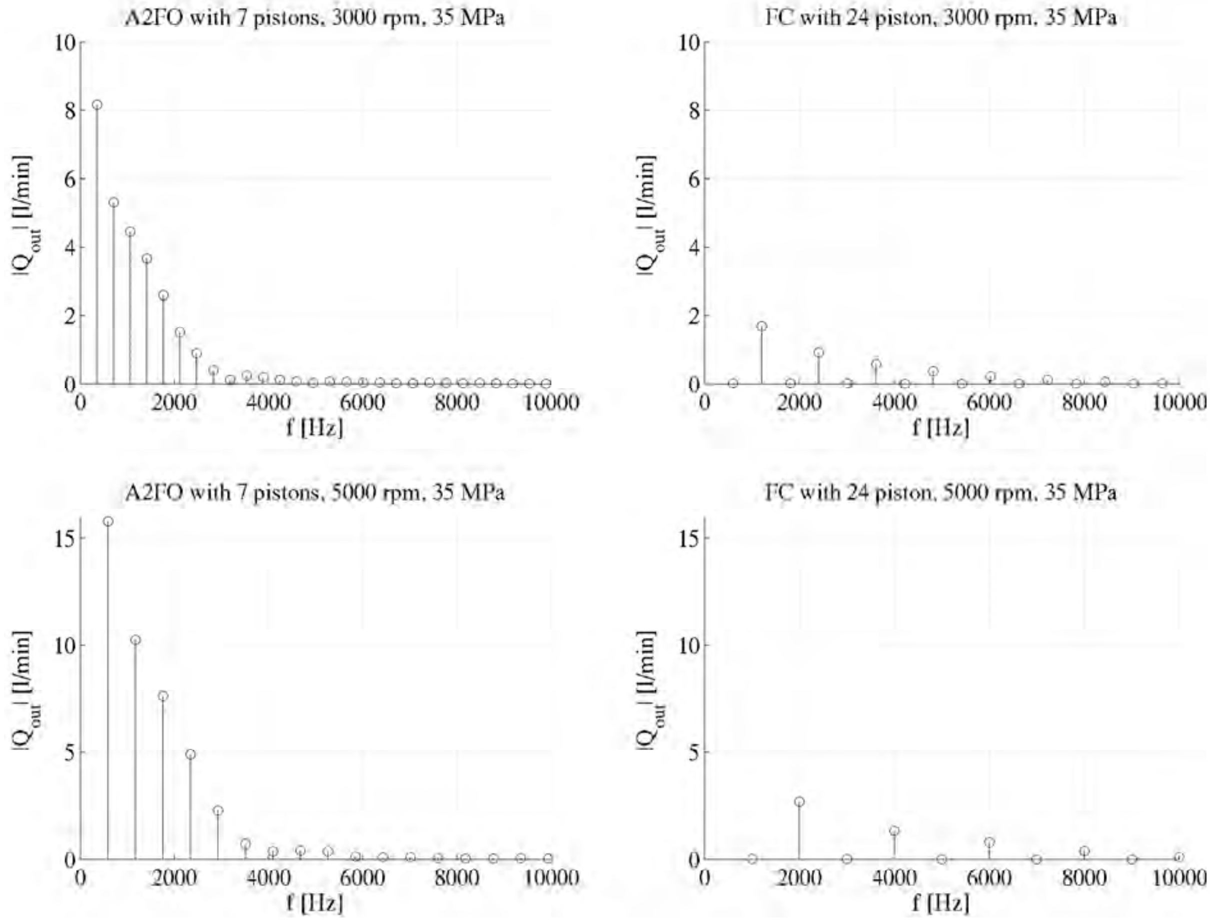

Figure 5: Flow ripple spectra at 35 MPa

Where n is the rotational speed in rpm and n_h is the number of harmonics below 10 kHz. Figure 6 shows the result. The benefit of this large an increase in the number of pistons is clear. The RMS value of the flow ripple is significantly lower for the 24-piston FC pump, at all rotational speeds.

4 CALCULATING THE DYNAMIC FLOW IN THE DUCTS

As mentioned in section 1 , with a symmetrical arrangement of the pump's connecting ducts and its outlet, the odd harmonics in the 12 piston based flow from each pump half cancel at the point where the the outlet attaches to the duct. Only the even harmonics will enter the outlet and thus, the pump appears as a 24 piston pump to the attached hydraulic system. At other positions in the connecting duct, however, the odd harmonics of the 12 will still be present.

When the first FC pump prototype was designed, the pressure pulsation in the connecting duct caused some concerns, as this pulsation excites the walls of the ducts and might cause noise. As no empirical knowledge of comparable constructions was available, it was decided to perform a theoretical study into the flow ripple in the internal ducts. The goals of this study were to determine if there would be a risk for severe internal pulsation and, if so, to identify ways to

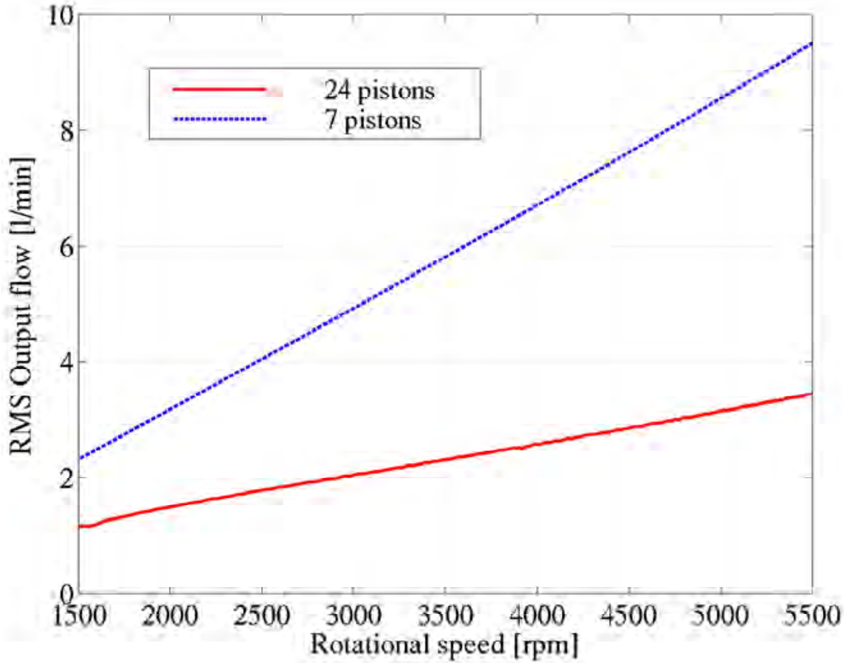

Figure 6: Comparison of change of RMS flow amplitude with rotational speed

minimise this internal pulsation without sacrificing the FC's capacity for a low external flow ripple.

In this context, it was necessary to choose a way modeling that can take fluid dynamic effects into account but calculates fast enough to enable design optimisation studies. An approach was adopted in which a hydrostatic system is broken up in a series of components that transfer energy at their interfaces. The state at each interface can be fully described by two time dependent parameters: a flux parameter and a potential parameter (flow and pressure in a hydrostatic system).
In the special case that the system under study may be linearised around a static working point, the dynamic content of these interface parameters can be transformed into the Laplace or the Fourier domain. The hydrostatic system components can then be fully characterised by the transfer functions between all interface parameters. As these transfer functions are simple algebraic relationships, the task of coupling components and solving the resulting system equations is replaced by a series of simple multiplications of transfer functions.

This way of modelling linear systems originated from the field of electronics (see for instance [3]). There, each interfacing side of a component is called a port. In analogy to electronic components, each port has two poles. The potential parameter is the difference in potential between these poles, the flux enters and leaves the port through the poles. Systems can be assembled using only three types of component models:
- active two-poles, that supply energy to the system,
- passive two-poles, that can dissipate energy and
- passive four-poles: that transfer energy from one component to the next.

Figure 7 shows the pole model of a hydrostatic system, containing all three types of components.

The dynamic content of flows and pressures are presented as the time dependant quantities $q(t)$ and $p(t)$.

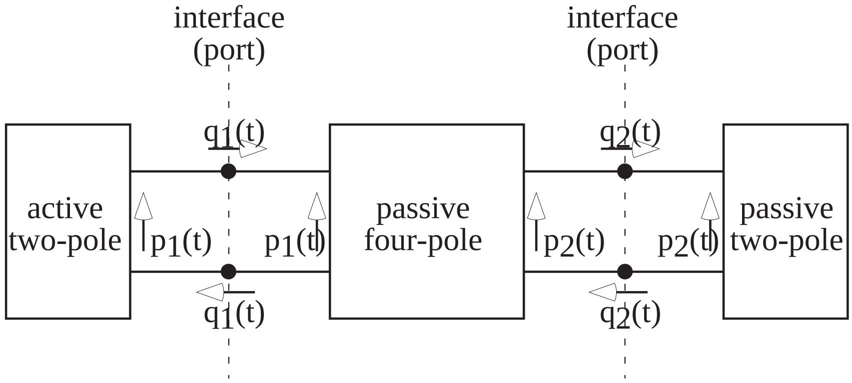

Figure 7: A hydrostatic system with two two-poles and a four-pole

The so-called 'four-pole equations', describing a general four-pole between interface i and $i+1$, can be written as a matrix multiplication between a vector with the pole parameters of port i and a vector with the pole parameters port $i+1$. If the Fourier formulation is used, the port parameters and the four elements of the matrix are functions of the angular frequency ω:

$$\begin{bmatrix} P_i(\omega) \\ Q_i(\omega) \end{bmatrix} = \begin{bmatrix} A_i(\omega) & B_i(\omega) \\ C_i(\omega) & D_i(\omega) \end{bmatrix} \begin{bmatrix} P_{i+1}(\omega) \\ Q_{i+1}(\omega) \end{bmatrix} \tag{3}$$

The passive two-poles can be written as a relationship between the flux and the potential parameter at the end of a system branch. In the Fourier formulation:

$$P_e(\omega) = Z_e(\omega) Q_e(\omega) \tag{4}$$

Z_e, generally complex and a function of ω, is called the 'end-impedance' of the system. The source two-pole defines either the flow or the pressure at its port, as a function of ω.

If port $i+1$ of a general four-pole is coupled to the end-impedance e, we can find, by a simple matrix multiplication for each ω:

$$P_i(\omega) = A_i(\omega) Z_e(\omega) Q_e(\omega) + B_i(\omega) Q_e(\omega) \tag{5}$$
$$Q_i(\omega) = C_i(\omega) Z_e(\omega) Q_e(\omega) + D_i(\omega) Q_e(\omega) \tag{6}$$

Eliminating $Q_e(\omega)$ from these equations leads to:

$$P_i(\omega) = \frac{A(\omega) Z_e(\omega) + B(\omega)}{C(\omega) Z_e(\omega) + D(\omega)} Q_i(\omega) = Z_e^T(\omega) Q_i(\omega) \tag{7}$$

This shows that an end-impedance coupled to a four-pole can be replaced by 'transformed' end-impedance Z_e^T. By repeating the process, a series of four-poles terminated by an end-impedance can be replaced by one transformed end-impedance Z_e^T. If this transformed end-impedance is coupled to a source two-pole for which either $P_s(\omega)$ or $Q_s(\omega)$ is known, the other source parameter can be directly calculated. Thus, both $P_s(\omega)$ and $Q_s(\omega)$ are known and with them $P_i(\omega)$ and $Q_i(\omega)$ at all other interface ports can be sequentially calculated.

At Innas, the 'HYFREQ' toolbox was created, that implements this way of modeling in MATLAB. The toolbox was created for use with the periodic input signals that originate from both the IFPE and the IHT and therefore it uses a Fourier formulation, in this case the one described by Stulemeier [4]. Four-pole and two-pole equations for hydrostatic components were taken from the same source but also from work done at the Delft University of Technology [5], at the University of Bath [6, 7, 8] and at the Okayama University [9]. In some cases they were derived in-house. This way of modeling in the frequency domain, is especially suited for a correct and fast-calculating description of inertial and dissipative effects in hydrostatic lines. The line model is based on a two-dimensional viscous compressible flow model, for which a good description can be found in [10, 11].

The HYFREQ toolbox calculates an exact Fourier transform of a periodic input signal and uses that as the known source in the frequency domain. It calculates all frequency domain responses at all interface points in the system in the way described above and uses an inverse Fourier transform to calculated the time domain responses at all interface points.

Initially, the HYFREQ toolbox allowed for only one source of disturbance in the system. Branched passive systems could be included in the model but no parallel loops or active branches. Due to its double-sided nature, if the flow ripples inside the FC pump are to be modeled, two active branches are required. The HYFREQ toolbox had to be expanded to achieve this:

A four-pole representation of a FC pump connected to a hydraulic system is given in figure 8. The figure shows three subsystems which represent the left side of the pump (System 1), the right side of the pump (System 2) and the exterior system (System 3).

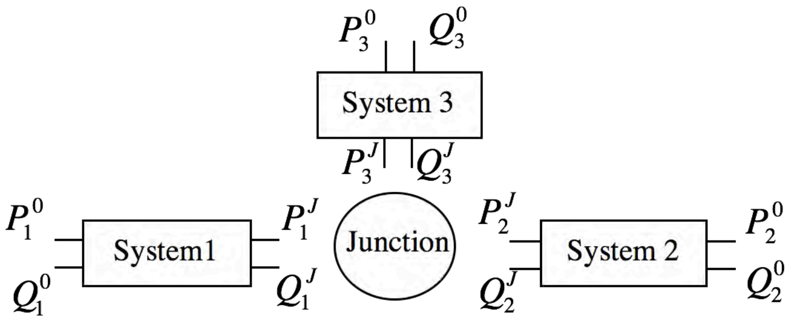

Figure 8: Four-pole representation of the FC pump and connected system

Each of these systems is a combination of four-poles and can be treated as such. Therefore, the relationship between pressure and flow at the limits of each of the subsystems can be written as:

$$\begin{bmatrix} P_i^0(\omega) \\ Q_i^0(\omega) \end{bmatrix} = \begin{bmatrix} A_i^0(\omega) & B_i^0(\omega) \\ C_i^0(\omega) & D_i^0(\omega) \end{bmatrix} \begin{bmatrix} P_i^J(\omega) \\ Q_i^J(\omega) \end{bmatrix} \qquad i = 1, 2, 3 \tag{8}$$

The input flows for systems 1 and 2, Q_1^0 and Q_2^0, are known and the pressure and flow at the limits of the three systems have to be determined. At the junction between the three subsystems the following relationships hold:

$$\begin{aligned} P_1^J(\omega) = P_2^J(\omega) = P_3^J(\omega) \\ Q_1^J(\omega) + Q_2^J(\omega) + Q_3^J(\omega) = 0 \end{aligned} \tag{9}$$

In general, it can be assumed that the end impedance of the exterior system is known:

$$Z_3^0(\omega) = \frac{P_3^0(\omega)}{Q_3^0(\omega)} \tag{10}$$

If the relationship given in equation 10 is substituted in equation 8, the impedance of the exterior system at the junction can be found.

$$Z_3^0(\omega) = \frac{P_3^J(\omega)}{Q_3^J(\omega)} = \frac{D_3(\omega)Z_3^0(\omega) - B_3(\omega)}{A_3(\omega) - C_3(\omega)Z_3^0(\omega)} \tag{11}$$

Finally, if equations 11 and 9 are introduced in equation 8 for systems 1 and 2 the following relationships can be found between the input flows and the flows at the junction.

$$\begin{bmatrix} Q_1^0(\omega) \\ Q_2^0(\omega) \\ 0 \end{bmatrix} = \begin{bmatrix} D_1(\omega) & 0 & C_1(\omega)Z_3^J(\omega) \\ 0 & D_2(\omega) & C_2(\omega)Z_3^J(\omega) \\ 1 & 1 & -1 \end{bmatrix} \begin{bmatrix} Q_1^J(\omega) \\ Q_2^J(\omega) \\ Q_3^J(\omega) \end{bmatrix} \tag{12}$$

Using equation 12, the flows at the junction can be determined and with those and equation 8, the rest of the interface parameters.

5 INTERNAL AND EXTERNAL FLOW PULSATION IN THE STANDARD FC DESIGN

This section shows how the design of the pump outlet affects the internal and external flow ripples. First however, the models used and the assumptions made will be described and the resulting limitations will be made explicit.

5.1 Description of the model and its limitations

The HYFREQ model of the high pressure side of the FC pump in the frequency domain, essentially consists of a combination of one-dimensional, distributed parameter models for the hydraulic lines and lumped parameter models for components like volumes. Where a fully established dynamic flow is modeled at a certain distance from the flow source (for example the external pump flow) this approach is perfectly valid and corresponds to standard practice. When modeling the interior of the FC Pump in this way, however, the method is pushed to its limits; possibly even over them, for two important reasons:

- The flow sources for the frequency domain modeling were derived from the AMESIM calculations that were presented in section 3. They assume a constant pressure sink in the whole internal duct, including the transitions and the kidneys. The dynamic parts of the so-calculated output flows, were subsequently taken as point flow sources for the HYFREQ models, located at the interface between the displacement chambers and the kidneys.
  This assumption of a point flow source is questionable: the dynamic parts of the pump flows calculated with AMESIM are mostly determined by phenomena in the precompression phase of a displacement chamber entering a kidney. This represents approximately the first 10% of the kidney length. Nevertheless, it may be argued that the flow source should be treated as a distributed flow source over the total kidney length and for this kind of flow source, the method provides no means.

- It is not entirely clear what should be taken as an appropriate model for the ducts in each pump half: should they be modelled as hydrostatic lines only or should the kidneys and the transitions to the circular ducts be modeled as volumes and the rest of the ducts as hydrostatic lines?

These are severe limitations when a reliable quantification of the pulsation is the goal. As in this case the goal was to prevent for possible problems with pulsation in the internal ducting, qualitative insight in the underlying mechanisms and possible solutions was sought after. The models should be appropriate for that.

Both ways of modelling the internal ducts have been examined. Figure 9 defines the options studied.

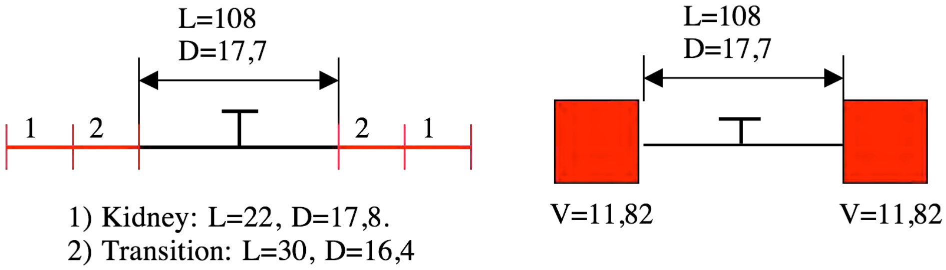

Figure 9: Four-pole models of the FC pump. Dimensions in mm and volumes in cm^3

For the system connected to the outlet of the pump, a reflection-free termination has been chosen in order to compare the different solutions without the influence of the 'outside world'. A pipe of 15 mm length and 19 mm diameter has been considered with an end-impedance equal to the characteristic impedance for that pipe: $Z_e = \rho c/A$. In this equation, ρ is the mass density of the oil, c the speed of sound in that oil at the average outlet pressure and A is the area of the outlet pipe.

The source flows for the excitation of the two pump halves have been taken from the same AMESIM simulations of a 1000 to 6000 rpm rotational speed sweeps that have been mentioned in section 3. The time domain data calculated with AMESIM were Fourier transformed in order to calculate the frequency spectra of the input flows. The frequency content of the flow data is known a priori, since only frequencies which are multiples of the rotational speed times the number of pistons per side will appear. For this reason, the FFT of the time domain data has been calculated taking 1 cycle at a time, which implies that the underlying time samples vary in length. Figure 10 shows the source flow magnitude spectra ($|Q_{source}|$). Each spectrum has a different frequency step, that corresponds to the frequency of the first harmonic for that rotational speed. 40 harmonics have been calculated for each rotational speed, which means that the band-width of the waterfall spectrum in figure 10 increases with rotational speed.

5.2 Results of the simulations

With the input flows shown above, the internal (in the kidney) and the external (at the pump outlet) pressure ripples have been calculated for the case where the outlet is connected symmetrically

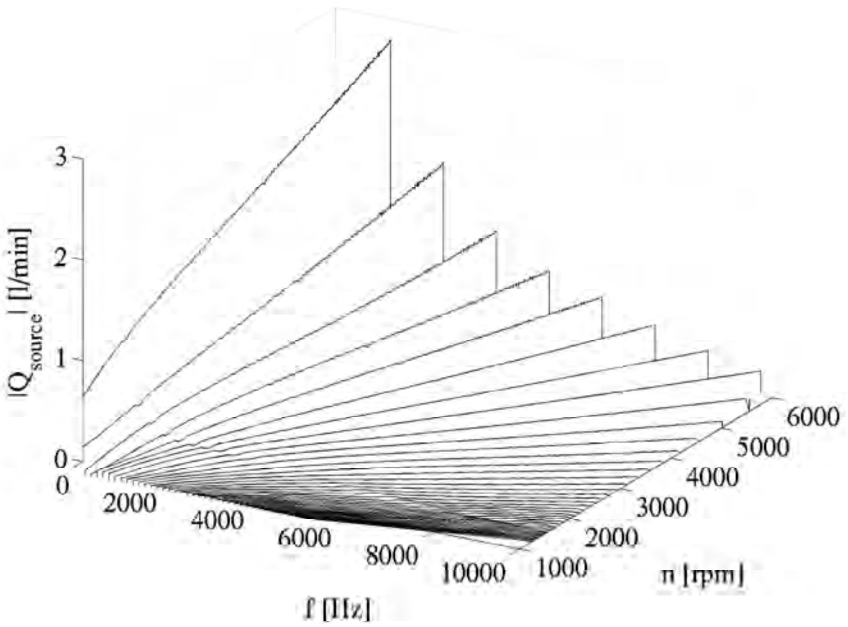

Figure 10: Waterfall of the source flow magnitudes out of one pump half

to the internal duct. Both possible internal duct models were calculated but the results obtained with the two models for this symmetric case are qualitatively equal. For that reason, only the model with only hydrostatic lines, for which the highest pulsation amplitudes are calculated, is treated here. Figure 11 shows the calculation results. The upper two plots, show waterfalls of the magnitude of the complex dynamic pressure signals ($|p_{dyn}|$) calculated for each order of 12 pistons.

The figure shows a clear a clear internal resonant frequency at 3490 Hz, which corresponds to a mode in the oil column in the hydrostatic line between the two kidneys. The amplitude of the peak increases as the rotational speed increases because the input flow amplitude of the corresponding harmonic increases. It is clarifying to look at the variation of the amplitude of the pressure magnitude along the line, for separate odd orders of 12 pistons, at the resonant frequency of 3490 Hz. These are essentially mode shapes, representing standing waves in the line. A 'negative magnitude' indicates that the pressure wave in this section is 180^o out of phase with the pressure wave in the part where 'positive magnitudes' are shown.

The mode shapes in the plot reveal that the resonance occurs every time the wavelength of the pressure pulsation for a given odd harmonic equals twice the length of the line. This resonance affects only the odd harmonics because the input flows for these harmonics on either side of the pump are 180 degrees out of phase, which perfectly matches the mode shape.

The results show that the external pressure pulsation is dramatically lower than the internal pressure pulsation. The explanation can be found by looking at the plot of the mode shapes. The outlet is located exactly at the position where the pressure pulsation due to the resonance is zero (a nodal point), which means that the perturbation is not transmitted to the outside. It can also be seen in the plot of the external pulsation that only the even harmonics contribute to the pressure pulsation at the outlet. The maximum pulsation values correspond to the second

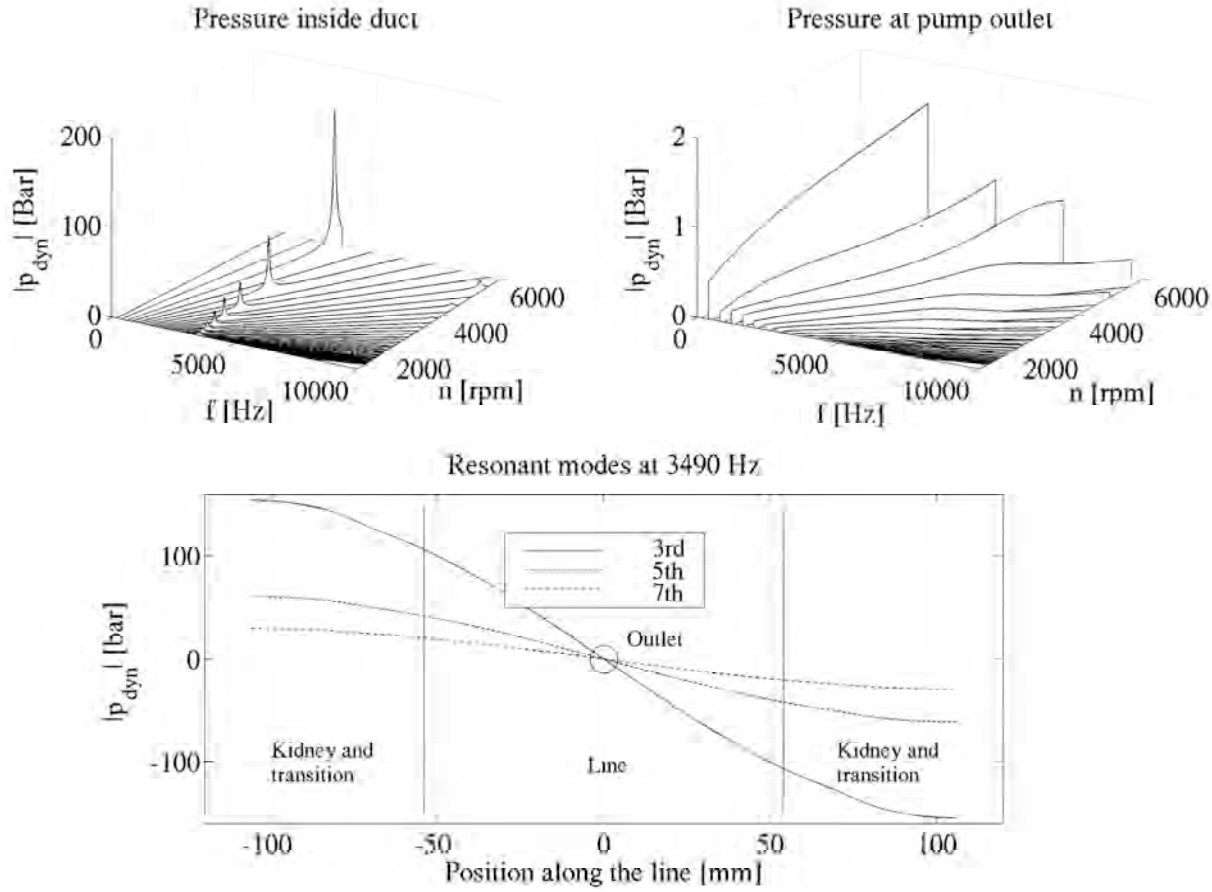

Figure 11: Results, symmetric connection (model with only hydraulic lines)

harmonic. The frequency of the peak increases with rotational speed, which indicates that there is no resonant phenomenon associated to those peaks. The increase in pulsation amplitude with rotational speed is due to the increase of the input flow amplitude.

An important remark on the validity of the calculation result, should be made here:
The calculated internal pulsation levels are extremely high, even to the point that the assumption of small changes, necessary to allow for this way of modeling in the frequency domain, cannot hold true. Moreover, the assumption of constant pressure sink, used to calculate the source flows in AMESIM, is not valid anymore. Notwithstanding these limitations, the potential problem of a resonant mode in the symmetrical design should be taken seriously, it would also appear if a more sophisticated model was adopted.

6 DESIGN ALTERNATIVES

The high internal pressure peaks at the resonant frequency could be a significant source of noise generated within the pump structure. As already explained, the pulsation peaks are caused by a resonant phenomenon inside the pump. Design alternatives have been sought, in order to reduce or eliminate this resonance. The two alternatives studied are:

- A non-symmetrical position of the pump outlet with respect to the internal duct.
- Helmholtz resonators tuned at the resonant frequency of the internal duct with symmetrical outlet position.

In the first case it is expected that the resonance will be eliminated at the cost of higher pulsation amplitudes at the outlet, as the perfect cancellation of the odd harmonics in the 12 piston based flows from each half is sacrificed. In the second case, the amplitude of the resonance peak should be greatly reduced and the outlet flow is expected to be the same as in the symmetric case.

6.1 Non-symmetrical position of the pump outlet

In a preliminary study, performed to determine the optimal location of the outlet, it was seen that the lowest pulsation at the outlet are obtained when the outlet is located close to either of the kidneys. In the modelled situation, the outlet has been positioned at one pump half, immediately next to the transition duct between the kidney and the main duct.

The calculation results for the the model with only hydrostatic lines in the internal duct, are shown in figure 12. The waterfall of the magnitudes of the internal pressure (in the kidney) shows a

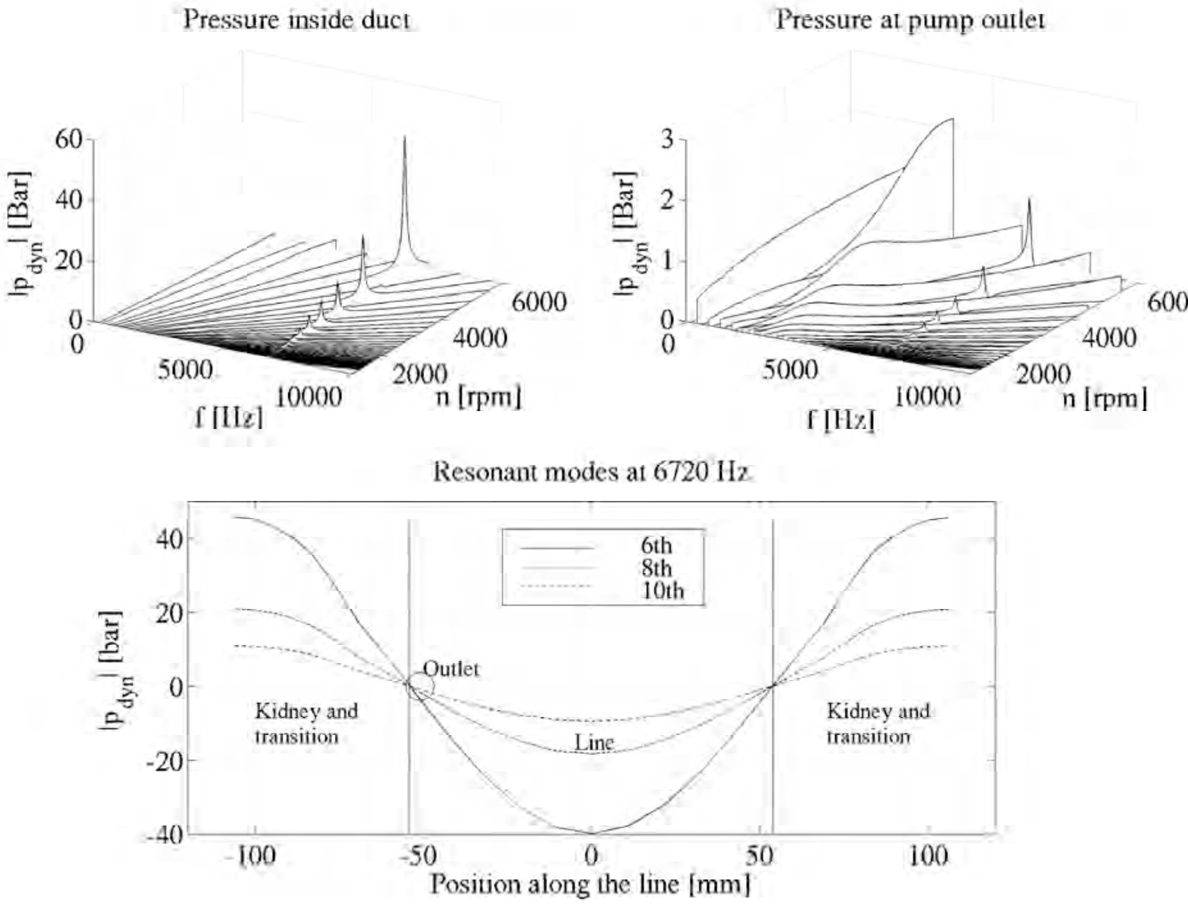

Figure 12: Results, non-symmetric connection (model with only hydraulic lines)

new peak at 6720 Hz. Its frequency remains constant with the rotational speed, which implies that the peak is due to a resonant phenomenon inside the pump. The high peaks occur every time the frequency of an even harmonic comes close to the frequency of 6720. In the rotational speed range analysed, the 6th harmonic is the highest that reaches the resonance frequency. The differences in the peak amplitude of the different harmonics are due to the differences in input flow amplitudes for the harmonics.

The mode shapes for the 6th, 8th and 10th harmonic of 12 pistons show that the resonance occurs when the wavelength of the pressure pulsation for an even harmonic is equal to the length of the line. This resonance affects only the even harmonics because the input flows at the ends of the line are in-phase for these harmonics. The mode shapes show that also in this case, the outlet is located close to a nodal point of the internal resonant mode shape. This implies that the high pulsation levels at the internal resonant frequency are not significantly transmitted to the outside.

Further analysis of this resonant phenomenon has shown that – with this duct model – the outlet is located almost exactly at the location where the even harmonics of the both pump sides match two different modes of the oil column in the duct. For this choice of outlet location and model chosen, the outlet is accidentally situated close to the coinciding nodal points of the two modes. In order to check the sensitivity to this choice, the position of the outlet was moved by around 10 mm in the model. This reduces the predicted peak amplitude of the pressure in the duct by a factor of ten, without significantly increasing the external flow pulsation.

The pressure pulsation at the outlet of the FC pump shows some resonant behavior for the odd harmonics at approximately 3500 Hz. This is probably due to the the internal mode at 3490 Hz. It is still present and sensitive to the odd harmonics but, because in this case the outlet is not located in a nodal point of this mode, the pressure pulsation will also pass to the outlet. This seems to attenuate the mode significantly, without shifting the frequency a lot. The external pressure pulsation is somewhat higher than in the symmetric situation, but it is still rather low. For high rotational speeds the highest pulsation amplitudes are due to the 3th harmonic passing through this resonant mode at about 3490 Hz. In the speed range where the 2nd harmonic predominates, the highest outlet pressure pulsation amplitudes are similar to those obtained with the symmetric configuration.

The system was also simulated using the model with the kidneys and part of the transitions modeled as volumes. Figure 13 shows the results. The internal pressure pulsation calculated for this case is much lower than for the model with only pipes. The high peaks at 6720 Hz can no longer be seen. The internal pressure pulsation matches the external pressure pulsation almost exactly. In both a mode at about 3500 Hz, sensitive to the odd harmonics, can again be seen. This is probably a mode, in which the total mass of the oil in the line bounces between the oil springs defined by the volumes at both ends. The outlet is not located at a nodal point, so it takes part in the mode, much in the same way as was described for the model with only lines and a non-symmetrical connection.

A peak of low amplitude and constant frequency can be seen at about 8740 Hz. This could be the same resonant mode shape found in the previous case. The plot of the modeshape of the 8th order indeed shows that this mode shape is similar to that calculated for the model containing only lines, apart from the fact that it is truncated at both ends due to the constant pressure in the volumes. The higher resonance frequency predicted is due to the shorter effective length of the line. The peak at 8740 Hz does not contribute significantly to the external pressure pulsation. From the above it can be concluded that the results obtained for the nonsymmetric configuration are very dependent on the type of model used. The main conclusions of the simulations of the

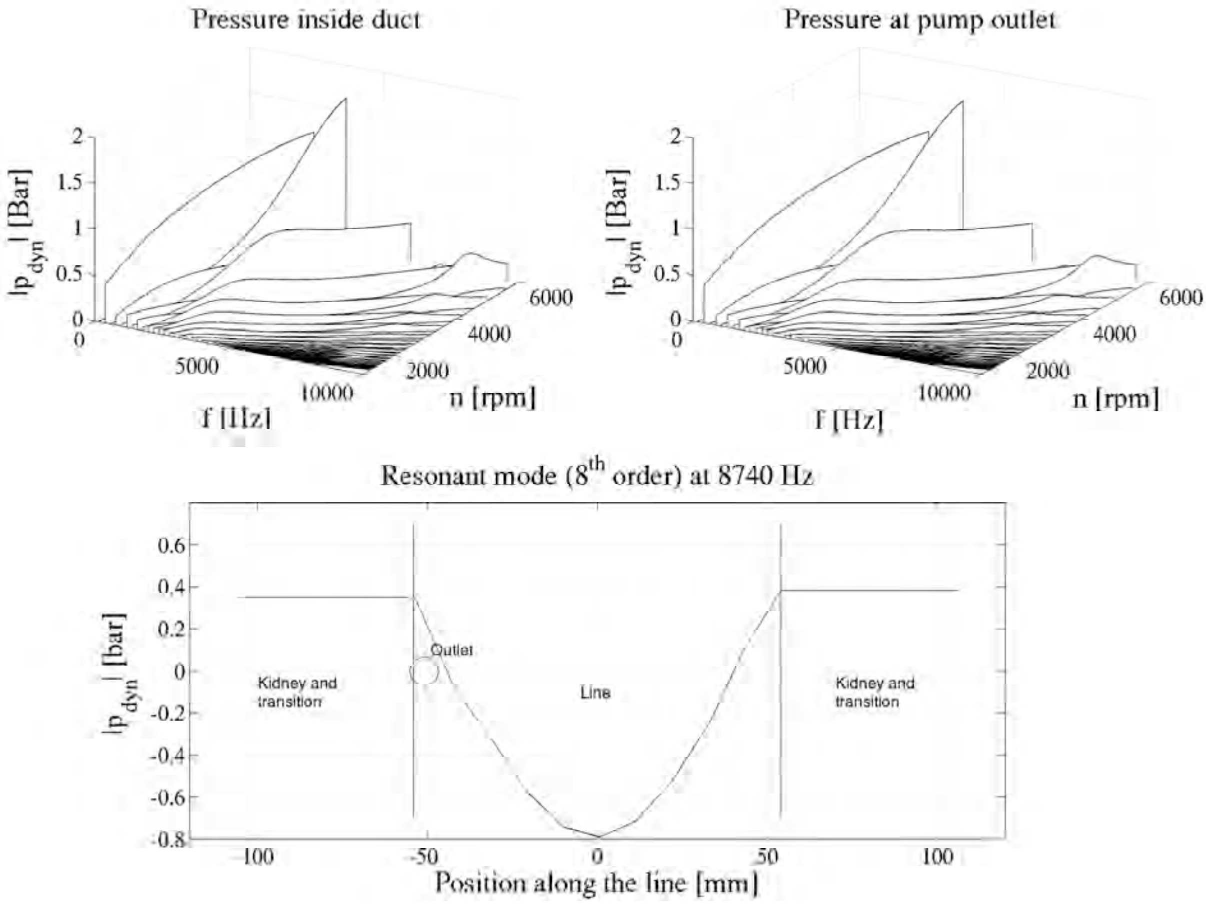

Figure 13: Results, non-symmetric connection (model with volumes and hydraulic lines)

non-symmetric configuration are:

- The pressure pulsation at the outlet of the pump can be expected to be low. The results obtained with both models are similar. Compared to the symmetric configuration, the results for the outlet pressure pulsation in the non-symmetric case are better than expected. The non-symmetric configuration spoils the cancellation of the odd harmonics and, therefore, it was expected that it would to produce higher pulsation levels at the outlet. It does, but not to a great extent. There is still a resonant mode in the connecting duct, which is sensitive to the odd harmonics of the source flow. That the outlet is not at a nodal point of this internal mode, seems to attenuate the pressure pulsation caused by this mode, both internal and external.
- The resonance seen with the internal duct modeled only with hydrostatic lines, is unique for that particular arrangement of the different pipe segments. If the position of the outlet is slightly modified, the amplitude of the resonance is greatly reduced.
- From the simulations it is difficult to establish whether there will be a resonance inside the pump. If the kidney and transition behave as a volume a resonance is very unlikely. If their behaviour is better approximated by a pipe there is a chance that a resonance occurs. In that case, the amplitude of the peaks will depend strongly on the exact position of the outlet. This implies a strong but hard to predict sensitivity to the design parameters.

6.2 Helmholtz resonators

The second alternative proposed is to use Helmholtz resonators to cancel out the internal resonance. The resonators should be located on both sides and as close as possible to the kidney, because that is where the maximum pressure levels occur. A HYFREQ modeling block for the Helmholtz resonators was constructed using [9] and used in the model of the FC pump. The results of the simulations obtained with the two different ways of modelling the internal duct are similar, so here only the data for the model containing hydrostatic lines are presented. The

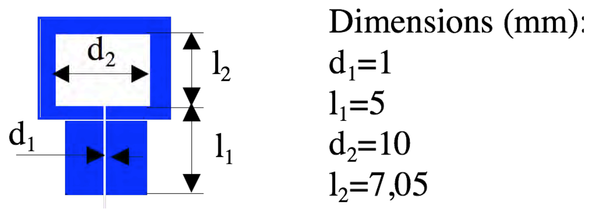

Figure 14: Dimensions of the Helmholtz resonators

dimensions of the Helmholtz resonators were optimised to give an appropriate damping value at a resonant frequency of 3490 Hz. They are given in figure 14.

The waterfall spectra of the internal and external pressure pulsation, calculated for a system with two Helmholtz resonators of these dimensions, located as close to the kidneys as possible, can be seen in figure 15 The pressure pulsation amplitudes in the kidney are greatly reduced

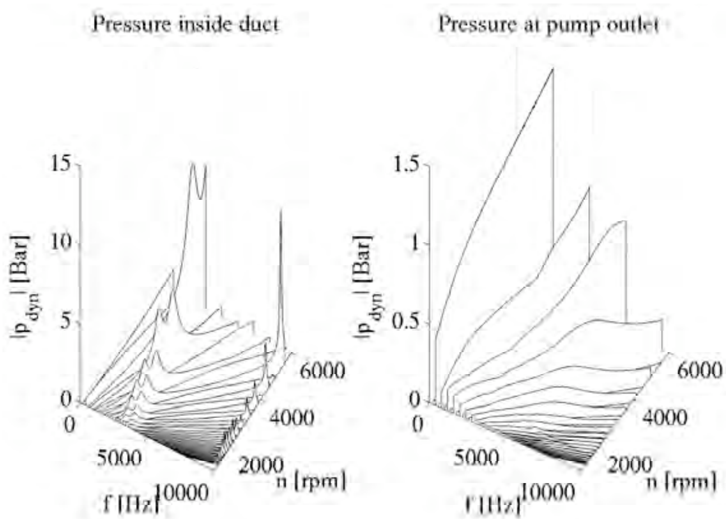

Figure 15: Results, symmetric connection and Helmholtz resonators (model with only hydraulic lines)

with respect to the situation without resonators. As should be expected from the use of a tuned resonator, there are now 2 peaks in the neighbourhood of 3500 Hz, but these are heavily damped.

The peaks at about 10 kHz where also present without resonators, but now their amplitude is comparable to the amplitude of the lower frequency peaks.

7 CONCLUSION

In this paper, a flow pulsation analysis for the new FC pump with 24 pistons has been presented. The comparison to the flow pulsation produced by a conventional bent axis 7-piston pump has shown that the amplitude of the flow ripple is greatly reduced with the FC pump and that the frequency content of the pulsation is pushed to higher frequency regions.

Due to the two-sided configuration of the FC pump, the internal pulsation in the connecting duct could be a problem. Simulation results for an FC pump with a symmetric outlet connection reveal an internal resonance that can lead to very high dynamic pressure levels inside the pump. As there are more ways to model the internal pump duct, two different ways of modeling were used. For this symmetrical configuration, they produce similar results. Thus, it is possible that for the symmetrical arrangement of the pump outlet, the high internal pulsation level will cause significant noise outside the pump. Two alternatives have been proposed, that can reduce this internal pressure pulsation.

The simulation results for a non-symmetrical outlet connection – the first alternative – show that in any case, the pulsation at the outlet is somewhat higher than with the symmetric outlet arrangement. The predicted increase is so small, that it is unlikely that this will produce a significant increase in measured noise.
The calculated internal pressure pulsation, however, differs with the duct model used. It seems that there is a slight chance that a non-symmetrical outlet position might still lead to high internal pressure pulsation.

The simulation results for an FC pump with Helmholtz resonators attached to the duct – the second alternative – show that the internal resonance is dramatically reduced and that the pulsation at the outlet is not deteriorated.

In the near future, measurements of the FC Pump will be performed. The prototype has been designed such that all variants – symmetric outlet connection, non-symmetric outlet connection and symmetric outlet connection with Helmholtz resonators –10 can be tested. If the tests of the symmetric connected outlet configuration indicate noise problems due to resonant phenomena in the internal duct, the other variants will be tested. In this case, the non-symmetric solution is preferred because it is simpler and cheaper to apply. Only if the resulting pulsation at the outlet is unacceptable should the Helmholtz resonators be introduced.

It should be noted that the models used for the calculations are not suited to give exact quantitative predictions of the internal pressure pulsation. They have been used to gain qualitative insight in the phenomena that might play a role in this new pump type, in order to be able to prepare appropriate countermeasures already in the design stage. To this end, the adopted modeling approach is considered to be appropriate.

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