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INVESTIGATIONS ON THE ACOUSTIC OPTIMISATION OF A VARIABLE DISPLACEMENT PUMP USING VIRTUAL PROTOTYPING

Summary. In modern vehicles the steering systems are still widely equipped with power-assisted steering pumps. In most cases vane pumps are used which limit the fluid volume flow in dependence of required pressure and running speed by a special design of the internal control valve. This control valve internally redirects the volume flow inside the pump leading still to unnecessary fluid circulation. Variable displacement pumps now offer an additional opportunity to eliminate the internal volume flow in dependence of the required load with reduced losses and hence increased efficiency. This is realized by a variable adjustment of the displacement cells, but simultaneously the variable force and load distributions inside the pump make the acoustic optimization even more difficult. In this paper the kinematics of the vane pump are modelled with a combined analytical and numerical approach. The data out of this model are used as input data for the hydraulic model of the variable displacement vane pump with a commercial tool. Both models are validated with data from test rig investigations. With this validated virtual prototype different design options are developed and finally successfully investigated on a test rig and in a passenger vehicle.

OPTYMALIZACJA AKUSTYCZNA POMPY O ZMIENNEJ OBJĘTOŚCI ZA POMOCĄ WIRTUALNEGO PROTOTYPOWANIA

Streszczenie. W nowoczesnych pojazdach układy kierownicze obecnie wyposażane są w pompy wspomagające. W wielu przypadkach pompy te ograniczają objętościowe natężenie przepływu w zależności od ciśnienia i prędkości obrotowej za pomocą specjalnego zaworu regulacyjnego. Zawór ten ustala przepływ wewnątrz pompy, powodując niepożądane cyrkulacje rzeczy. Pompy o zmiennej objętości zapewniają obecnie dodatkowe korzyści dostosowując przepływ wewnątrz w zależności od obciążenia przy zapewnieniu mniejszych strat i większej sprawności. Jest to realizowane przez regulacje zmian objętości komór, jednakże jednoczesne zmiany sił i rozkładu obciążenia wewnątrz pompy sprawiają, że proces jej optymalizacji akustycznej jest bardzo trudny. W artykule przedstawiono kinematyczny model pompy wraz z jego analitycznym i numerycznym opisem. Dane wyjściowe tego modelu wykorzystano jako dane wejściowe do modelu hydraulicznego pompy o zmiennej objętości, utworzonego za pomocą standardowych narzędzi. Oba modele zweryfikowano za pomocą danych ze stanowiska badawczego. W wyniku tej weryfikacji opracowano różne warianty konstrukcji pomp, które następnie przebadano na stanowisku badawczym i w samochodzie osobowym.

1. INTRODUCTION

Since several decades different kind of hydraulic pumps are in use for supplying automotive vehicles with highly pressurized fluids. In purely hydraulically assisted steering systems mainly vane pumps are used. These pumps have meanwhile been highly optimized with regards to several criteria. Nowadays electro-hydraulic or purely electric steering systems are developed and increasingly implemented in vehicles. These systems offer additional opportunities for an increased fuel efficiency and simultaneously reduced emissions. On basis of the existing economic demands and the continuously raising customer expectations the criteria costs and acoustics have an increased impact on the development and refinement of the systems and their acceptance at the market.

The advantage of conventional vane pumps as energy transformer in steering systems is based on the fact that this design concept offers a valuable compromise between required package space, pre-defined speed ranges and requested volume flow at an acceptable level of pressure pulsations. Nevertheless in the vehicle development process the steering systems always need to be tuned acoustically in order to minimize the impact on the passengers from the pressure pulsations of the power-assisted steering pump.

Conventional, non-electrical systems are usually belt-driven by the combustion engine. This leads to the effect that the systems are in operation even when no steering support is required. For steering systems this is valid for straight ahead drive or at idle. A complete decoupling of the energy transformer from the drive, as it is used for air conditioning compressors, is not an option for power-assisted steering pumps. The ramp-up time would be too long for the requirements of steering systems in case of shortly required power-demand during parking at standstill or suddenly required steering manoeuvres.

Therefore in conventional vane pumps the volume flow at the pressure outlet is controlled in dependence of the pump speed by an internal control valve. This offers the opportunity of easy-to-design volume flow characteristics versus speed which have an impact on the subjectively perceived steering performance of the vehicle. The control valve redirects the volume flow back internally inside the pump. Hence there is still unnecessarily an internal volume flow produced which is not required for the steering assistance.

The design concept of variable displacement pumps addresses this issue by adjusting the internal displacement in dependence of required pressure and volume flow so that the losses with increased engine speed are further reduced. This variability of the displacement leads simultaneously to variable internal force balances and variable vibration excitation. The task of acoustic system optimization is becoming even more challenging as the excitation mechanisms are more complex.

In this paper the kinematics of the variable displacement mechanism are modelled by a virtual model of the pump internals on basis of real world data. The output of this virtual model is then used as input for a virtual model of the dynamics of the hydraulic pump behaviour. By a combined use of both models alternative design variants are developed and evaluated in order to reduce the vibration excitation of the variable displacement pump. Finally one of the design variants is realized in hardware and evaluated with regard to the noise and vibration performance on a test rig as well as in a vehicle. This design variant is then compared to another design variant on a mechanical basis, which is more cost intensive.

2. OPERATING PRINCIPLE OF VARIABLE DISPLACEMENT PUMPS

In conventional vane pumps the volume flow per revolution is pre-defined by the fixed geometry of the displacement volume. This leads to a linear increase of volume flow at the pressure outlet with an increase of the pump speed as shown in the left part of Fig. 1. The implementation of a control valve limits the maximum volume flow at an upper limit. The characteristics of the control valve can be designed in such a way that with increasing speed the volume flow remains for a defined speed range at a constant level and then is further reduced to a lower level for constant system pressure. The volume flow for an external consumer is therefore reduced in dependency of the pump speed. Inside the pump the volume flow is still increasing, which leads to a large loss area as shown in Fig. 1.

In variable displacement pumps the geometry of the displacement volume is designed in such a way that it can be varied. The flow volume inside the pump is therefore adjustable by positioning the outer boundary of the displacement volume. This is realised by a hydraulic control loop in dependence of external load and pump speed. Theoretically the losses can be reduced to zero, see right part of Fig. 1. By adjustment of the system to a lower internal volume flow the required drive torque is reduced as well. This leads by the reduced energy consumption of the pump to lower fuel consumption and also to lower emissions of the combustion engine. In the literature a fuel saving of up to 0.15 l per 100 kilometres is quoted. In lubricating oil pumps the design concept of a variable displacement pump is already used since a longer time.

The designs of vane pumps from different manufacturers are quite similar. In the left part of Fig. 2 the design of a typical conventional vane pump is shown. It is a two-stroke system with ten vanes and strong contouring of the inner side of the housing. This pump is designed in such a way that in each cell the process from soak to output of the fluid is conducted twice per revolution. The advantage of a two-stroke system is that the rotor which carries the vanes and the rotating shaft are symmetrically loaded by the two strokes. This leads to a reduction of internal stresses on the components. An adjustment of a two-stroke system has yet not been realized by simultaneously maintaining the advantages. The contouring of the inner shape of the displacement volume defines on one side the volume flow per revolution and on the other side it ensures that the vanes are guided back in radial direction to the rotor at higher speeds. In addition the left part of Fig. 2 shows the components of the control valve, which limits the external volume flow at higher speeds.

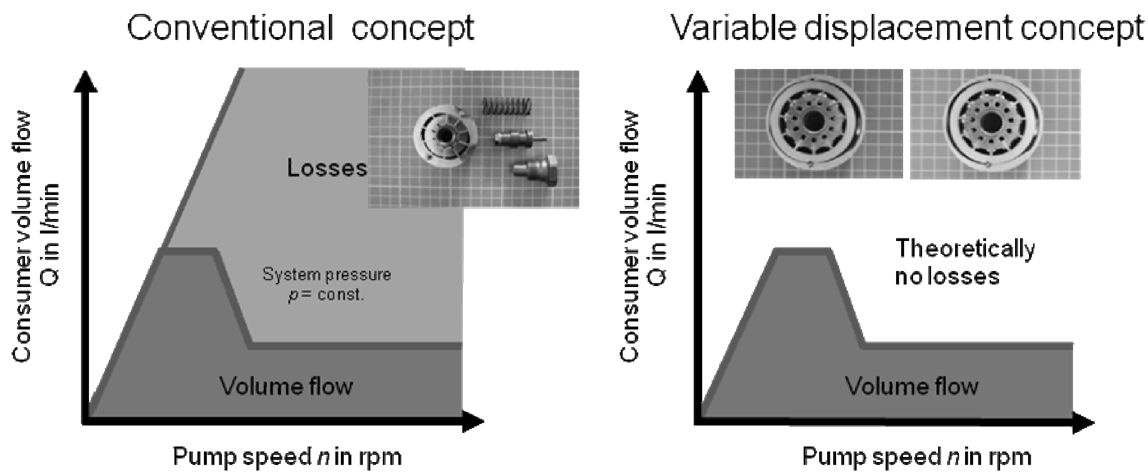

Fig. 1. Comparison of conventional and variable displacement pumps
Rys. 1. Porównanie pompy konwencjonalnej z pompom o zmiennej objętości

Actually different design concepts of variable displacement pumps are available for steering systems at the market. The right part of Fig. 2 shows an existing design concept of a variable displacement vane with single-stroke and eleven vanes. The displacement volume is limited by an outer ring, which is supported at its outer bottom end with a pin. This pin is inserted with free play to an insert of the pump housing. The positioning of the outer ring is dependent on the balance of the hydraulic forces at its outer side. Another design concept, which is already available at the market, is using a free-flow support of the outer ring.

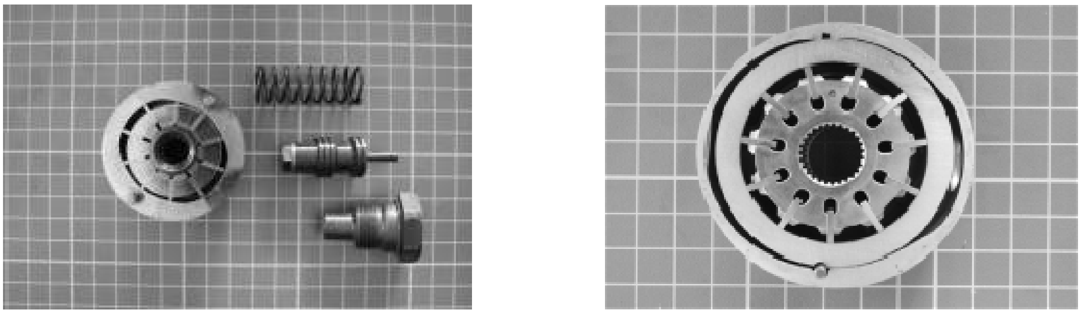

Fig. 2. Examples of a conventional (left) and a variable displacement (right) vane pump
Rys. 2. Przykłady pompy konwencjonalnej (lewa) i pompy o zmiennej objętości (prawa)

Independent of the chosen design concept the variable positioning of the outer ring inside the vane pump leads to variable hydraulic load conditions of the different components. Simultaneously the pressure build-up inside the cells is changing in each cell for each position. This is caused by the fact that the inlet and outlet orifices in the housing are still fixed in the same positions. The variability in the pressure gradient in dependence of the position of the outer ring leads to a variable dynamic excitation of the system components and respectively to a variable acoustic behaviour. The acoustic optimisation of steering systems with variable excitation spectra is therefore even more demanding.

The design principle of vane pumps lead always to a dependency of the pressure fluctuations from the number of cells between the pump vanes. Depending on the positioning of the inlet orifice and the outlet orifice relative to the individual cell, the actual position of the rotor as well as the actual contour of the inner side of the outer ring, which is limiting each cell at its outer side, the characteristic pressure profile is build-up. Special attention needs to be put on the gradient of the pressure during the build-up phase, the peak pressure inside the cell as well as the pressure fluctuations in the cell, see Fig. 3.

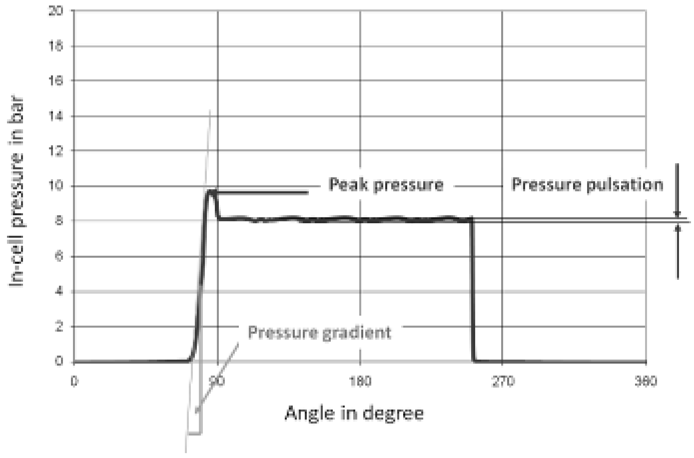

Fig. 3. Characteristic pressure profile a cell per revolution
Rys. 3. Przebieg zmian ciśnienia wewnątrz pompy w funkcji kąta obrotu

Hydraulic steering systems in vehicles are usually belt-driven. By use of a pre-tensioner the belt is pre-tensioned in such a way that it ensures a proper transmission of the required torque to the pulley of the pump. This leads to a force loading of the pump shaft and hence to a static bending of the shaft relative to the housing. In conventional vane pumps this effect is compensated by fine tuning of the housing design in order to ensure that the centre line of the rotor is in coincidence with the centre line of the housing after pre-tensioning the pump in installed conditions.

During the operation of the pump this static deflection is additionally superimposed by the elastic behaviour of the pulley, the shaft and the housing under load. In principle the rotating components can be modelled as an elastic shaft, which is statically pre-tensioned, and supported in an elastic structure. Due to the skewness of the shaft and the pulley caused by the pre-tensioning and the elastic behaviour of all components gyroscopic effects are occurring at higher speeds. These effects like the speed dependency of the eigenfrequencies of the rotating system are well-known in rotor dynamics.

For variable displacement pumps the relative positioning of the centre line of the outer ring relative to the centre line of the shaft and the rotor is now additionally changing in dependency of the actual position of the outer ring. The above described compensation of the elastic deformation of the rotor shaft by an offset of the support point is now no longer valid for all possible positions of the outer ring. Therefore beside the intended variability in the so-called displacement curve the variability of the outer ring position changes the dynamic excitation of the resulting forces on the shaft. This leads to varying excitations of the mechanical system and simultaneously to varying hydraulic loads in the system. Hence this physically variable vibration and acoustic behaviour is leading to phenomena, which are perceived by the customers as different noises inside the vehicle.

3. MODELLING OF KINEMATICS

The displacement volume per revolution of a vane pump can be calculated for single-stroke systems from the difference between the maximum and the minimum volume per cell during one revolution. With known position of the centre line of the shaft and known position of the outer ring contour the displacement curve of the pump can be calculated by modelling the kinematics of the pump. Special attention needs to be put on the inner contour of the outer ring and the modelling of the contact area between the outer ring and its counterpart at the insert in the pump housing. After the calculation of the volume for each cell during one revolution, see Fig. 4, the so-called displacement curve of the pump can be calculated.

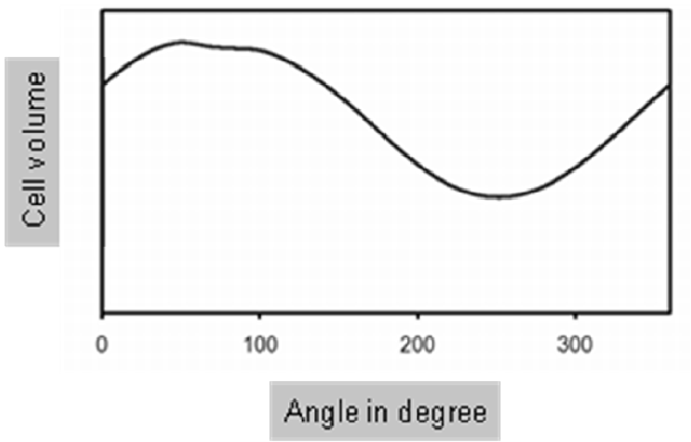

Fig. 4. Calculated displacement curve from the kinematic model
Rys. 4. Obliczona krzywa przebiegu zmian objętości na podstawie modelu kinematycznego

The kinematic model contains the geometry of the components and their constraints. As the data for the inner contour of the outer ring are available from measurement of the real world components a special treatment for de-noising the measurement data has to be applied. The same applies to the measurement data for the contact area between the outer ring and the insert in the pump housing. The area of each cell is physically limited by the two adjacent vanes, the outer radius of the rotor and the adjacent part of the outer ring contour. This area for each cell in each position is calculated by use of an analytical integration scheme for bi-quadratic approximation of the boundary as it is often used in finite-element-modelling.

With such a kinematic model the displacement curve can be calculated for any chosen position of the outer ring and any position of the rotor centre line. The investigations have shown a high sensitivity of the displacement curve with regard to the positioning of the outer ring as changes of the outer ring position in the range of micro-meters already lead to visible changes in the displacement curve. The displacement curve which has been calculated with this kinematic model is now been used as input data for the prediction of the pressure profile for each cell with the hydraulic model.

4. MODELLING OF HYDRAULICS

For the build-up of the virtual hydraulic model the geometry of the inlet and the outlet orifice by which the fluid flow into and out of the cells in dependence of the position of the individual vane position is required. In addition relevant leakage flow into and out of the cells is taken into account. In combination with the displacement curve from the kinematic model and with setting of speed and outlet pressure the pressure profile for each cell for one revolution can be calculated. The objective is to model the hydraulic behaviour under consideration of fluid dynamic effects inside the pump.

The model is build-up using the software DSHplus, which is offered by the company Fluidon, Aachen, Germany. At first a model for a single cell of the vane pump is developed. By measurements of the in-cell pressure of a real world pump on a test-rig this model is validated for different operating and build conditions. Afterwards the model is extended to an eleven-cell model.

The comparison of the calculated pressure profile in one cell with the measured in-cell pressure profile shows a good agreement, see Fig. 5. This is valid for the peak pressure in the cell as well as for the gradient of the pressure build-up. It has to be emphasised that the measurement data show the pressure profile of all cells at a fixed position of the pressure transducer in the housing for one revolution. In Fig. 5 the calculation result is shown for the same operating and build condition but in reference to an observer who is rotating with the cell. By superposition of the calculated data from one cell the profile for eleven cells is obtained.

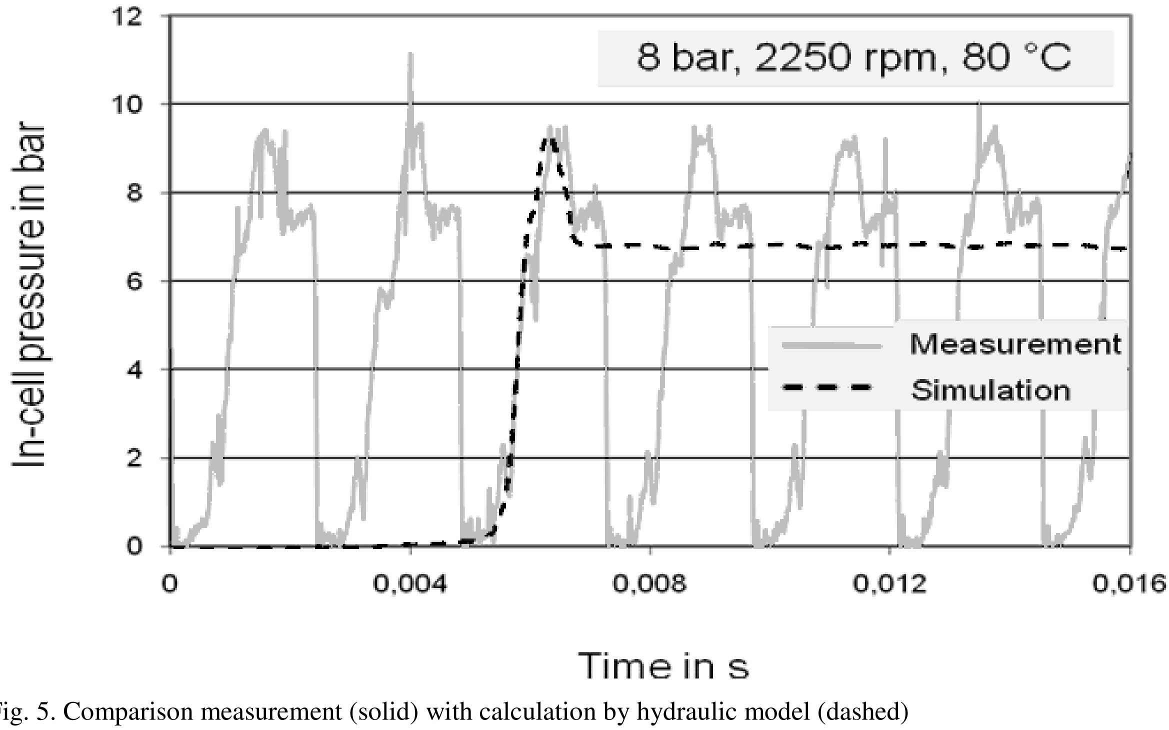

Fig. 5. Comparison measurement (solid) with calculation by hydraulic model (dashed)

Rys. 5. Porównanie wyników pomiarów (linia ciągła) z obliczeniami na podstawie modelu hydraulicznego (linia przerywana)

5. DEVELOPMENTS OF DESIGN VARIANTS

The validated virtual hydraulic model offers now in conjunction with the validated kinematic model the opportunity to develop optional design variants which reduced the observed pressure pulsation excitation inside the cells. In order to manage the large number of possible parameters, which can be modified at first a reference state has been selected, see left part of Fig. 6. This reference state shows with the validated hydraulic model an overshoot of the peak pressure inside the cells although the pressure pulsations at the pump outlet are relatively low. This is also observed in the measurements on the test rig.

For the reduction of the overshoot in the peak pressure different options are available. Firstly, by the introduction of hydraulic damping the pressure build-up in the relevant speed range can be influenced. This is described in the following as design variant A in more detail. Alternatively, the pressure build-up inside the cells can be modified by changing the displacement curve in dependence of the geometric positioning of the outer ring. This is evaluated below as design variant B.

In design variant A the overshoot of the peak pressure is reduced by the introduction of an additional flow channel between the cells and the pressure side of the pump at a specific location and with a specific design shape, see right part of Fig. 6. As a penalty the pressure pulsations at the pump outlet are increased, but this is evaluated as being still acceptable.

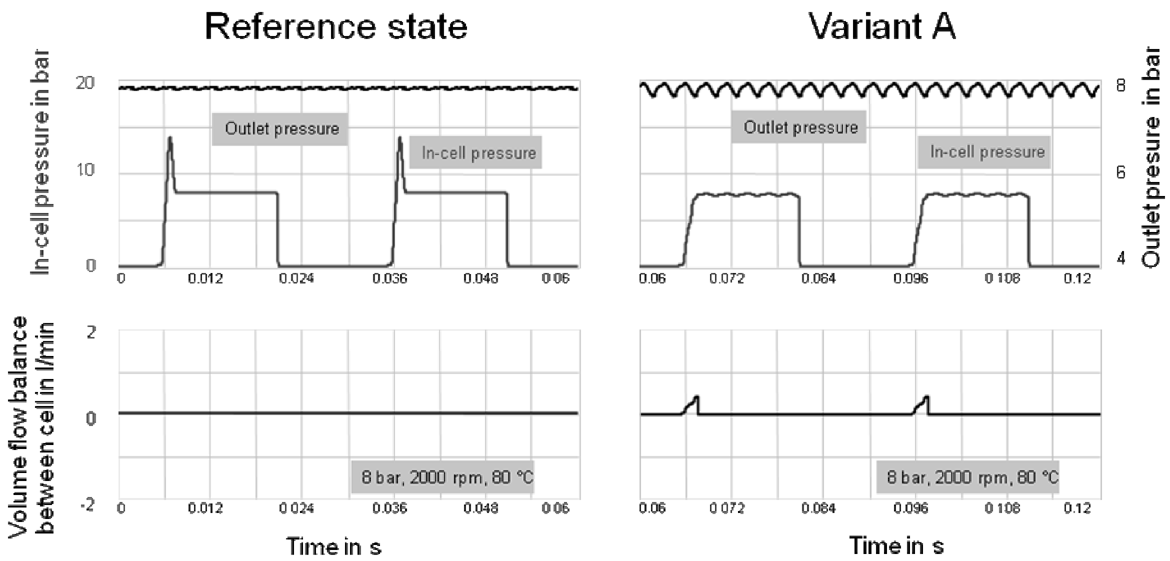

Fig. 6. Comparison simulation: reference state vs. state with additional variable hydraulic damping (design variant A)

Rys. 6. Porównanie wyników symulacji: pompy konwencjonalnej i pompy z dodatkowym zmiennym tłumieniem hydraulicznym (wariant A)

In design variant B the displacement curve can be varied by a horizontal movement of the outer ring in dependence of the pump speed. This would require a different internal design of the pump kinematics. The calculation results with the hydraulic model here show as well a reduction of the peak pressure and simultaneously a slight increase in the pressure pulsations at the pump outlet, see right part of Fig. 7.

6. VALIDATIONS

As the realisation of a horizontal movement of the outer ring requires larger modification to the pump internals the validation by hardware has been realised for design variant A.

In order to prevent the effect of in-vehicle installation the validation has been conducted in the first step on a test-rig. The vane pump is driven by an electric motor with pre-defined belt tension. The structure-borne excitation is measured by an accelerometer on the pump housing. According to the build principle of the vane pump the 11th, the 22nd and the 33rd order are the main excitation orders of the pump. In real world the pump speed range between 1.500 and 3.200 rpm has been identified as the most critical range for an unloaded system. Hence these orders in this speed range are evaluated in more detail.

The evaluation of the orders show a clear reduction of the acceleration level on the pump housing for design variant A with the additional variable hydraulic damping in the relevant speed range, see Fig. 8.

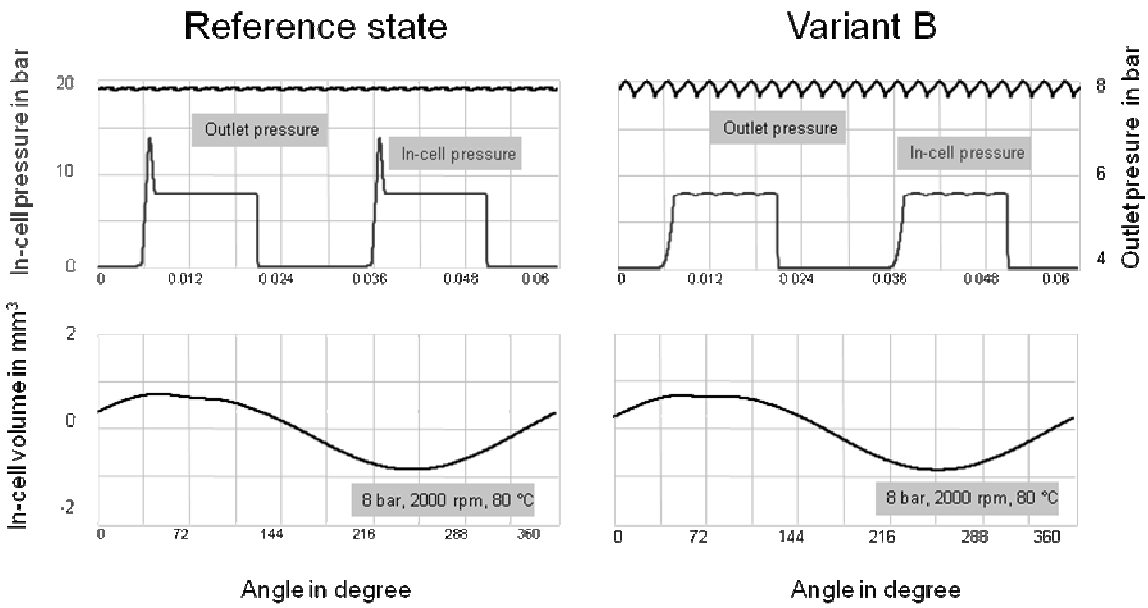

Fig. 7. Comparison simulation: reference state vs. state with horizontal shift of outer ring (design variant B)
Rys. 7. Porównanie wyników symulacji: pompy konwencjonalnej i pompy z poziomym przemieszczeniem pierścienia zewnętrznego (wariant B)

In the next step the pump with design variant A is evaluated installed in a vehicle. The reference state is used again as a basis. This state is compared to an optimised state in two build conditions. This is on one side the realisation of design variant A implemented in an optimized pump. In addition a second build condition is evaluated where design variant A is substituted by a mechanical measure. This mechanical measure is a dynamic absorber which is fitted to the pulley and tuned to the relevant speed range. It has to be pointed out that the dynamic absorber is at a significant higher cost level than the measure from design variant A. For the same system performance the developed design variant A therefore would be the better choice.

In Fig. 9 it is visible that the additional mechanical measure reduces the interior noise for all three orders in the relevant speed range in comparison to the reference state. The installation of the developed hydraulic measure, design variant A, shows the same potential for improvement of the interior noise for all three orders in the relevant speed range like the mechanical measure. Above 2.250 rpm a slight increase in the 22nd order is visible for design variant A. This could be resolved by further tuning the exact design parameters for the additional openings between the cells and the pressure side.

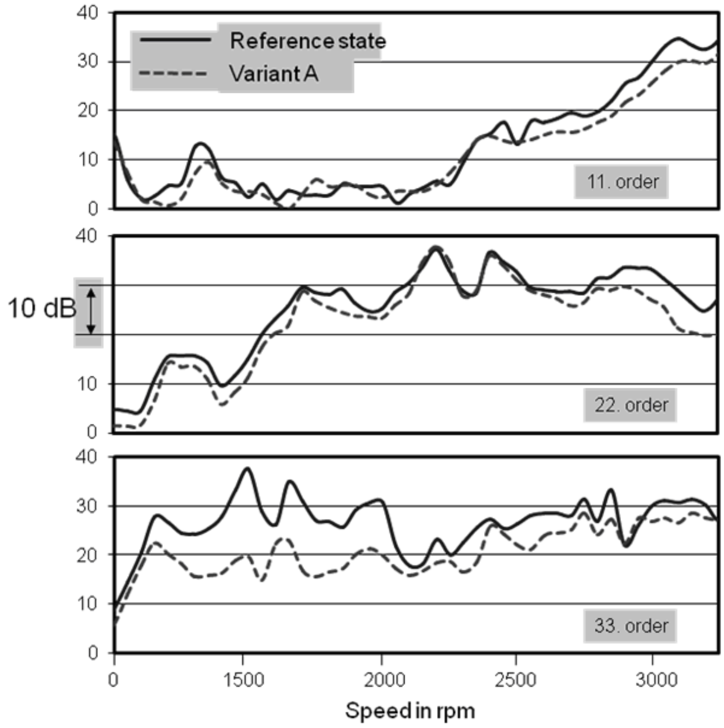

Fig. 8. Comparison rig test pump housing acceleration level: reference state (solid) vs. state with additional variable hydraulic damping (dashed)

Rys. 8. Porównanie poziomu przyspieszeń drgań obudowy pompy na stanowisku badawczym: pompy konwencjonalnej i pompy z dodatkowym zmiennym tłumieniem hydraulicznym

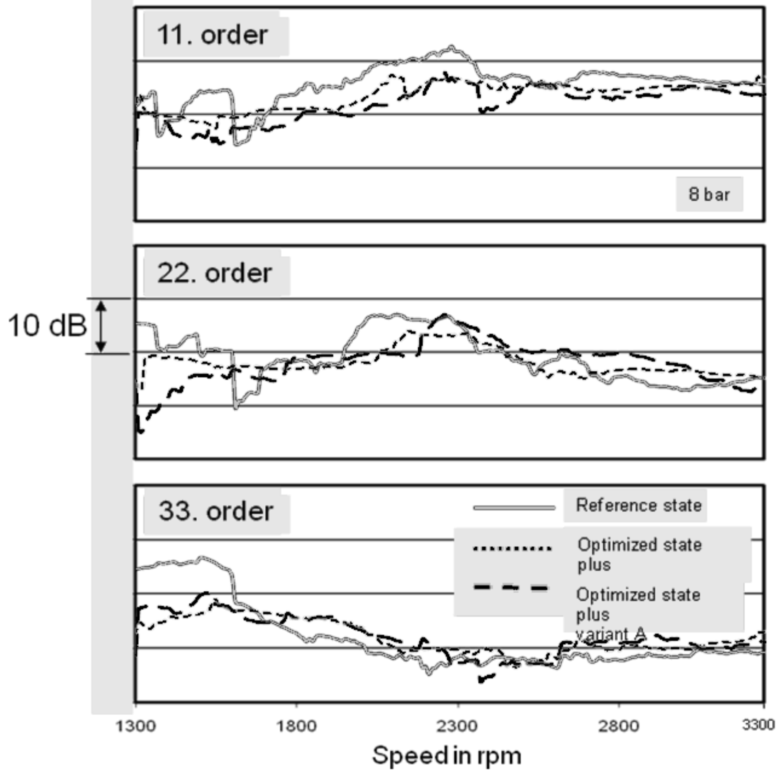

Fig. 9. Comparison in-vehicle interior noise: reference state (solid) vs. optimised state with additional mechanical measure (dotted) vs. optimised state with additional variable hydraulic damping (dashed)

Rys. 9. Porównanie poziomu hałasu wewnątrz pojazdu: pompa konwencjonalna (linia ciągła), zoptymalizowana pompa z tłumieniem mechanicznym (linia kropkowana), zoptymalizowana z dodatkowym zmiennym tłumieniem hydraulicznym (linia przerywana)

7. CONCLUSIONS

The complex interactions in variable displacement pumps can be modelled by relatively simple virtual models of the kinematic and the hydraulic behaviour with acceptable efforts. The comparison of measured data for the in-cell pressure on a test rig show a good correlation to the predicted values for different operation speeds and load conditions. On basis of this validated hydraulic model different design variants are developed in order to reduce the vibration excitation of the system. The realisation of a first prototype shows on a test-rig as well as in-vehicle promising results which emphasises the potential of the developed design variants with significant lower costs than the implementation of an alternative mechanical measure like a dynamic absorber.

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