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Evaluation of Pressure Resonance Phenomena in DCT Actuation Circuits

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Abstract: The paper investigates hydraulic wave propagation phenomena through hydraulic circuits of power transmission systems by means of numerical approaches. The actuation circuit of a Dual-Clutch Transmission (DCT) power transmission system supplied by a Gerotor pump is analyzed. A steady state approach is adopted to detect resonance phenomena due to Gerotor design parameters and circuit lengths, while one-dimensional numerical models are implemented to predict the pressure oscillations through the hydraulic ducts for the whole pump operating domain. CFD-1D pipelines are adopted to address the pressure oscillation behavior through the hydraulic pipeline, while spectral maps and order tracking techniques are used to evaluate their fluctuation intensity in function of the pump speed rate. The numerical models are validated with experimental tests performed on an ad hoc test rig for power transmission systems and a good match is found between the numerical and the experimental results. Pump design parameters as well as hydraulic accumulators and resonators are numerically investigated to quantitatively evaluate their improvement on the circuits' hydro-dynamic behavior. Furthermore, simplified numerical models are implemented to investigate the frequency response behavior of the hydraulic circuits by means of linear analysis. This approach resulted to be particularly effective for the prediction of the resonance frequencies location, and it can be adopted as an optimization tool since significant simulation time can be saved. Finally, the performance of the circuits operating with an eco-friendly fluid is evaluated numerically and the results are compared with the ones obtained with a traditional petroleum-based oil.

Keywords: Gerotor pump; actuation circuit; hydraulic resonance; pressure oscillation; CFD-1D; order tracking

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Citation: Terzi, S.; Montorsi, L.; Milani, M. Evaluation of Pressure Resonance Phenomena in DCT Actuation Circuits. *Energies* **2021**, *14*, 1354. https://doi.org/10.3390/en14051354

Academic Editors: Robin Roche and Teuvo Suntio

Received: 17 December 2020
Accepted: 24 February 2021
Published: 2 March 2021

1. Introduction

Numerical models are important tools in the design of hydraulic circuits for the prediction of their hydrodynamic behavior to evaluate their performance and detect unexpected anomalies, such as pressure oscillations and resonances that may generate instabilities and undesired noises. Moreover, simulation enables to investigate alternative and innovative solutions for improving the sustainability of the systems, such as by evaluating the performance of the circuit operating with eco-friendly innovative fluids instead of petroleum based hydraulic fluids. The hydraulic circuit of the power transmission systems of high-performance cars are generally characterized by strict geometrical constraints, due to the necessity of lightweight and compact systems design, and high speed volumetric pump as supply system, such as Gerotor gear pumps system since they guarantee high performance for the wide operating range of the engine speed.

In this paper, the hydraulic circuit of a Dual-Clutch Transmission (DCT) power transmission system adopted in high performance cars is analyzed by means of lumped and distributed parameters models for the investigation of resonance phenomena due to system design. In fact, these circuits are characterized by several components, such as pumps, valves, pistons, and ducts, which could resonate in specific actual operating conditions.

For instance, in [1], an aeronautical hydraulic pipeline system, which risks resonance due to the excitation of the pump vibration, is analyzed by means of an anti-resonance design approach based on a sensitivity index for the identification of the most significant variables for the circuit design. Gerotor gear pumps are employed in automotive applications, especially when high speed rotation rate is required, and numerical models are implemented to predict their fluid dynamic behavior and therefore to design the hydraulic circuits. Lumped and distributed models demonstrated to be accurate for the investigation and the prediction of the hydraulic behavior of Gerotor pumps, such as in [2] where a 1D approach is proposed for a Gerotor pump and the comparison with the experimental results proved good agreement by addressing the delivery flow rate prediction as well as the volumetric efficiency and the temperature influence. More detailed results can be obtained simulating the real component geometry with Computation Fluid Dynamics (CFD) numerical tools, such as in [3], where the fluid dynamics behavior of a double inlet Gerotor pump is analyzed under actual operating conditions by adopting an overset mesh approach. Nevertheless, the computational effort required for the simulation of CFD numerical models is not suitable for the simulation of complex hydraulic circuits and for the investigation of the whole operating conditions domain. Lumped and distributed numerical models are employed to evaluate the dynamic performance of power transmission systems, such as in [4], where a dynamic model of a DCT's hydraulic circuit is implemented for the prediction of the circuit characteristic, i.e., clutch filling and lubrication, and good agreement with the experimental results is found. Figure 1 reports the hydraulic scheme of the DCT power transmission system analyzed in the paper. It presents a Gerotor gear pump actuated by the engine, a Pressure Regulation Valve (PRV) for the regulation of the system pressure, the p_s, which is the system pressure transducer, and three Proportional Valves (PVs) for the actuation of the two clutches and of the differential piston. The two clutches enable power transmission from the engine to the odd and to the even gearsets, while the locking and unlocking of the differential is controlled by actuating the differential piston, based on the vehicle torque requirements. A hydraulic plate (Clutch Control Module) is adopted to manufacture the hydraulic circuit; the pipeline is drilled to design the hydraulic ducts, while the valves are located in their specific housing.

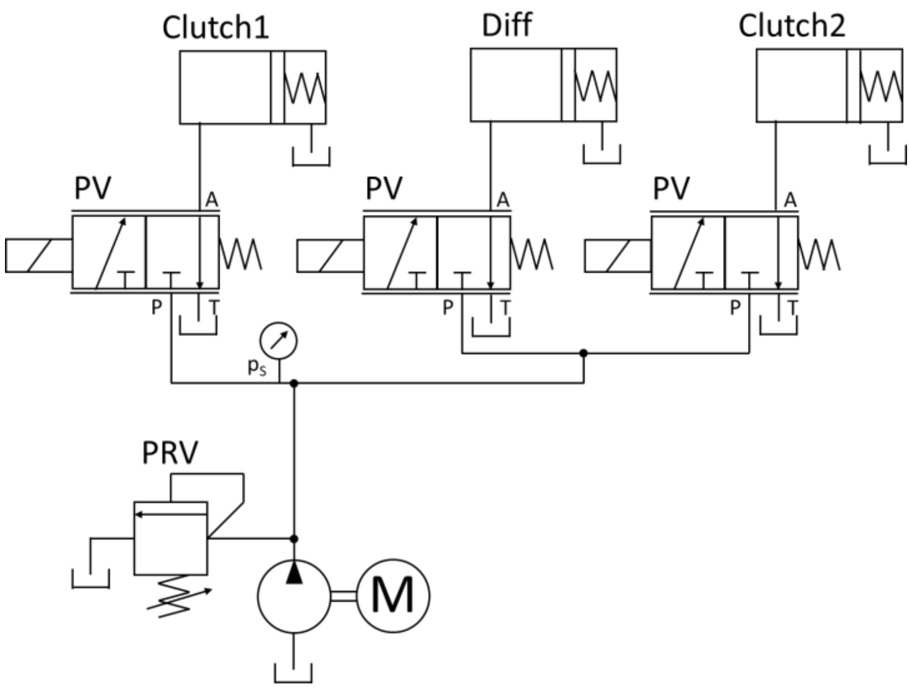


Figure 1. Hydraulic scheme of the Dual-Clutch Transmission (DCT) power transmission system.

In literature, wave propagation through hydraulic pipeline is deeply theoretically and experimentally investigated to analyze the fluid dynamic behavior. For instance, in [5], a four-equation model, which assumes 1D liquid flow, is adopted to simulate the fluid-structure interaction of water hammer in a pipeline fastened with a particular support system and it reported a good agreement between numerical and experimental results in terms of pressure oscillations. Hydraulic resonances may be induced by specific operating conditions and they should be generally avoided and constrained to achieve good performance to prevent components failure. For instance, in [6], the failure of grey cast iron pipe due to resonance phenomenon is numerically investigated by predicting the steady-oscillatory flow and evaluating the maximum stresses in the pipe. Wave attenuation is therefore an important topic and engineering solutions have been developed to reduce pressure oscillations where design constraints oblige to operate in resonant operating conditions. Helmholtz resonators are employed in hydraulic circuits to reduce flow pulsation and numerical models are used to design the resonator frequency, such as in [7], where a novel lumped parameter model is presented for the prediction of the resonance frequency of a three degrees of freedom Helmholtz resonator in hydraulic system. Numerical tools have been developed to analyze oscillating signals and to highlight their effectiveness by varying the operating parameters. Spectral maps are employed to evaluate instability behaviors in the frequency domain, such as in [8], where pressure signals recorded at different throttling valve position have been analyzed through spectral maps to describe the process of onset surge of a centrifugal blower. Other methods are also used for vibration analysis, such as the order tracking, which enables to identify speed-related vibrations by using as frequency base running speeds multiples. In [9], an order tracking method is investigated and the influence of the factors and assumptions are examined to evaluate their effectiveness on the results accuracy. Furthermore, simplified models can be implemented to evaluate and detect resonance behavior through linear response of hydraulic circuit by reducing simulation computational efforts. For instance, in [10], a modal testing approach is proposed for the experimental identification of the frequency response functions of hydraulic pipelines and results show a good match between experimental and calculated response functions. Numerical models are therefore accurate and reliable tools for pressure wave propagation analysis, but literature lacks numerical methodologies that dynamically predict resonance phenomena and investigate their causes in DCT power transmission systems. Furthermore, numerical tools can be adopted to optimize DCT hydraulic circuits' design parameters and predict their dynamic behavior by addressing the strict geometrical constraints that characterize this technology.

Nowadays, environmental compatibility as well as biodegradability are important topics for hydraulic applications. Eco-friendly operating fluids are developed to enhance environmental sustainability for industrial process, such as in [11], where vegetable oils performance is evaluated for metalworking applications and compared to petroleum based cutting fluid. In the power transmission field, several architectures present hydraulic systems operating with standard mineral oils for the actuation and the lubrication of clutches and gearset, and these applications require oil substitution due to properties degradation. The worldwide diffusion of these systems results into an important environmental impact that can be reduced by adopting eco-friendly compatible fluids.

In this paper, the wave propagation through the actuation circuit of a DCT power transmission system generated by a Gerotor pump supply is numerically investigated. A preliminary steady state model is realized to determine the resonance frequency of the circuit, while lumped and distributed parameter models are developed to numerically predict the dynamic of the pressure oscillations through the real geometrical domain of the pipeline. Resonance phenomena are detected and the influence of design parameters as well as components for the resonance attenuation are evaluated, i.e., pump teeth, hydraulic accumulators, and Helmholtz resonators. The pressure fluctuations are analyzed through spectral maps and order tracking techniques to evaluate resonance frequencies and amplitude as well as the pump orders which generate oscillations. The models are validated by comparing the pressure profiles obtained numerically with the ones measured experimentally on an ad hoc test rig for power transmission system. Furthermore, a linear response analysis of simplified models is investigated, highlighting frequency resonances locations by reducing the simulation computational effort. Finally, the performance of the circuit operating with petroleum-based oil and eco-friendly fluid are compared to evaluate the applicability of biodegradable fluid in the power transmission field to reduce the environmental impact.

2. Materials and Methods

In this chapter the numerical models used in the analysis are reported. Three approaches are implemented: a steady state approach, for the prediction of the resonance frequencies based on the pump features and the pipe lengths, a 0D/1D dynamic modeling approach for the simulation of the actual hydraulic circuit, to determine resonances amplitudes, frequencies, and orders, and finally a 0D/1D linear modeling approach, for the computation of the pipeline resonance response. Sensitivity analysis are performed with the 0D/1D models to optimize the dynamic behavior of the pressure wave propagations by varying the design parameters of the circuit.

2.1. Steady State Model

Through steady state calculation, it is possible to estimate pipe resonance frequencies due to pipes' geometrical features, fluid properties, and pump parameters. Table 1 reports the parameters of the hydraulic circuit, while Figure 2 shows the hydraulic circuit layout, where it is possible to recognize the Gerotor gear pump, the pipeline with the two lines to the PVs, the PRV, the p_s, and the Low Pressure (LP) circuit.

Table 1. Hydraulic circuit parameters.

Component	Parameters
Gerotor Pump	z_1 = 12 teeth z_2 = 11 teeth
Pipe 1	$(L/d)_1$ = 5.0; V_1 = 22.90 cc
Pipe 2	$(L/d)_2$ = 14.0; V_2 = 10.99 cc
Pipe 3	$(L/d)_3$ = 1.5; V_3 = 1.18 cc
Pipe 4	$(L/d)_4$ = 6.0; V_4 = 4.71 cc
Pipe 5	$(L/d)_5$ = 1.5; V_1 = 1.18 cc
Pipe 6	$(L/d)_6$ = 1.5; V_1 = 1.18 cc
Pipe 7	$(L/d)_7$ = 6.5; V_1 = 5.11 cc
PV	Closed
PRV	p^*

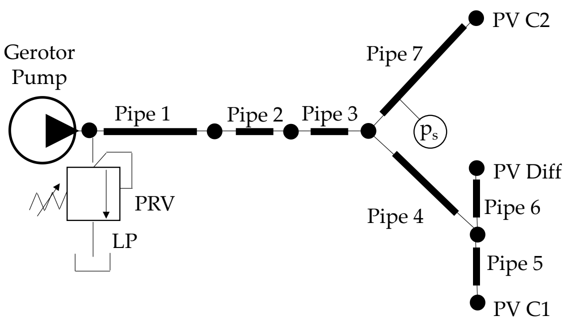


Figure 2. Layout of the hydraulic circuit.

The celerity in this specific hydraulic fluid is fundamental to address the wave propagation through the ducts and it is computed through Equation (1), where β is the Bulk Modulus and ρ is the fluid density:

$$c = (\beta/\rho)^{1/2}, \tag{1}$$

The wavelength λ is equal to four times the pipeline length (L) as reported in Equation (2) since the analyzed layout presents closed pipe architecture, while Equation (3) can be used to predict the resonance where the pump order (n) is an odd number representing the resonance node and therefore the harmonic:

$$\lambda = 4L, \tag{2}$$

$$f = n\,c/\lambda;\ n = [1, 3, 5, \ldots], \tag{3}$$

Finally, it is possible to predict the pump speed rate that excites the system at the frequency f due to the active wheel teeth through Equation (4), as reported in [12]:

$$n_{Pump} = f/z_2, \tag{4}$$

To investigate all the actual operating conditions of the pump for different pump orders, the algorithm shown in Figure 3 is used to detect the pump speed rates that generates resonance for the analyzed hydraulic circuit:

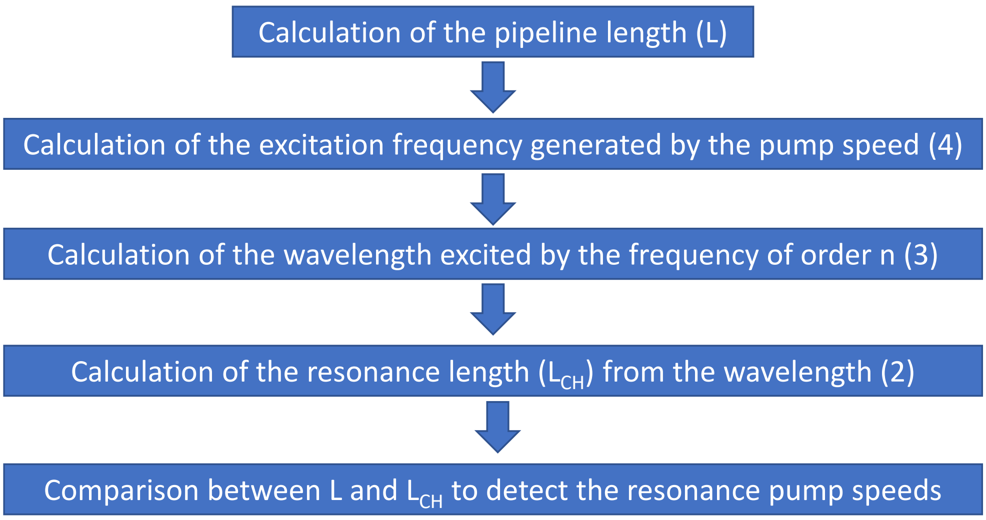


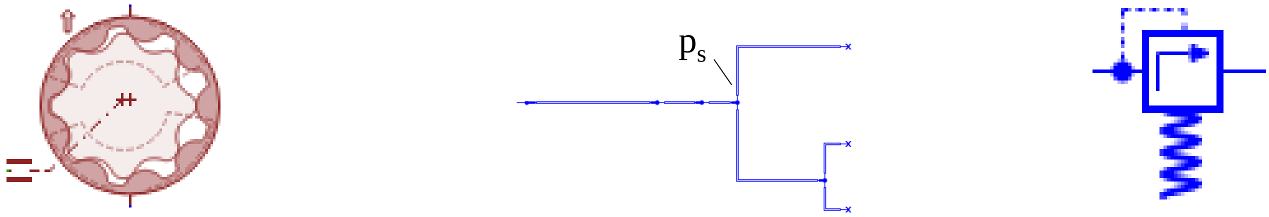
Figure 3. Steady state algorithm.

2.2. 0D/1D Model for Dynamics Analysis

Numerical models are developed to investigate the dynamic behavior of the circuit operating under actual operating conditions. The models of the hydraulic circuit are implemented in the software simulation environment Simcenter Amesim [13]. Specific models are chosen to simulate the hydraulic behavior of each component by addressing the main geometrical and physical features. The Gerotor gear pump is implemented through the component [HCDGP0], reported in Figure 4a. A Gerotor with 12 teeth for the external gear and 11 teeth for the internal gear is set accordingly with the pump parameters defined in Table 1. The internal and external gears' geometry are modeled through the Gerotor configuration tool where it is possible to define the geometrical parameters of the teeth as well as the inlet and outlet ports' angles. The leakages between the teeth are taken into account by using a Poiseuille law and a Couette law. The pipeline is modeled by implementing each pipe with the submodel [HLLWx] which is a CFD-1D hydraulic straight pipe model. A CFD-1D solver is dedicated in the software to integrate the partial differential equations in time and space to accurately predict the propagation of physical quantities, i.e., the flow and pressure oscillations through the pipe. Two-step Lax-Wendroff method is implemented in the software as CFD-1D solver to compute the continuity (5) and the momentum (6) equations in the numerical domain, where p is the absolute pressure, u is the velocity, S is the cross-sectional area, C_g is the gravity source, and C_f is the friction source. Five nodes are used to discretize each pipe. The pipeline is closed at the extremities with zero hydraulic flow sources (Q000) to address the boundary condition defined by the closed valve position and to enable wave reflection. Figure 4b reports the pipeline of the circuit, where the position of the pressure transducer, ps, is highlighted. The relief valve (RV010), represented in Figure 4c, is used to regulate the system pressure of the circuit and its opening pressure is set at the value p*. Figure 5 reports the base circuit, assembled with the single components, in order to analyze the dynamic response of the hydraulic circuit where resonance phenomena are expected.

$$\frac{\partial(\rho S)}{\partial t} + \frac{\partial(\rho u S)}{\partial x} = 0, \tag{5}$$

$$\frac{\partial(\rho u S)}{\partial t} + \frac{\partial(\rho u^2 S + pS)}{\partial x} + C_g + C_f = P\frac{\partial S}{\partial x} \tag{6}$$



(a) **(b)** **(c)**

Figure 4. Components submodels: (**a**) Gerotor gear pump (HCDGP0); (**b**) Pipeline (HLLWx); (**c**) Relief valve (RV010).

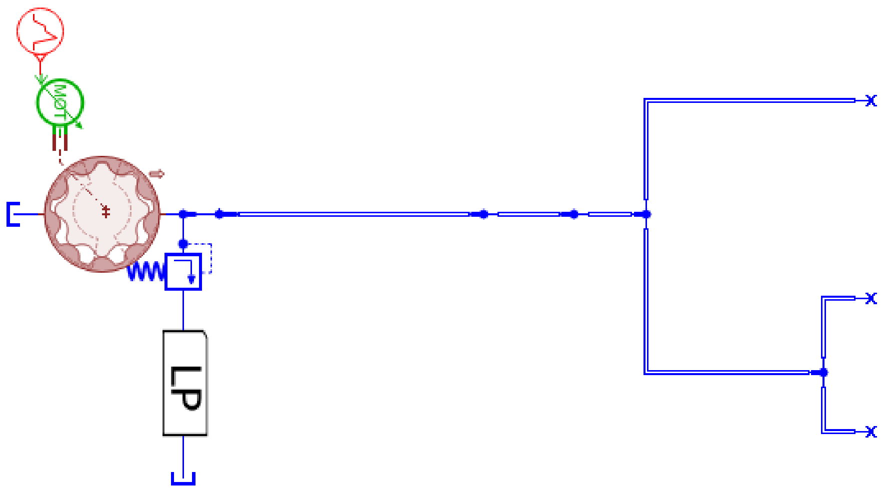


Figure 5. 0D/1D dynamic model of the base circuit.

Resonance phenomena can be reduced by modifying the design parameters of the circuit components or adding specific components, such as hydraulic accumulators and resonators. In this paper, three solutions for resonance reduction are numerically investigated by modifying the base circuit. The first investigated solution consists in the modification of the pump design parameter in order to the pump excitation frequency, e.g., by increasing the teeth number of the Gerotor pump. In this paper the performance of a pump with 19 teeth of the internal gear, i.e., z_1 = 20 and z_2 = 19, is investigated. The second solution is to increase the circuit volume by adding a hydraulic accumulator. The performance of a circuit with an accumulator of 0.1 L and an inlet port of 5 mm positioned at the end of the pump pipeline is analyzed. The third investigated solution is the use of a Helmholtz resonator to attenuate a desired resonance frequency. This component can be easily integrated in the hydraulic plate due to the compact design. The Helmholtz resonator is sized through the Equation (7), as reported in [7], where d_N is the diameter of the cylindrical neck, A_N is the cross sectional area of the neck, L_N is the neck length and V is the resonator cavity, as represented in Figure 6:

$$f = c/(2\pi) \times (A/(L_N V_C))^{1/2}; A_N = \pi\, d_N{}^2/4, \tag{7}$$

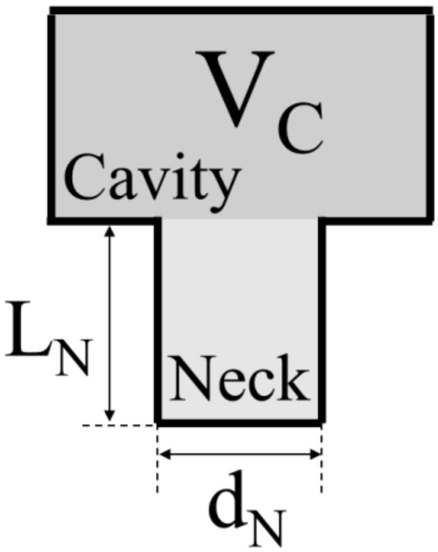


Figure 6. Helmholtz resonator layout.

Due to geometrical constraints of the hydraulic plate of the transmission, some parameters are design constraints, i.e., V_C and L_N. The volume V_C must be maximized to increase the magnitude of the resonator response, but it has to be small enough to keep be embedded within the hydraulic plate. Thus, the maximum volume available in the hydraulic plate is chosen in this analysis, which is 27 cm^3. The channel length L is also an important design parameter since it has to be small to obtain high frequency response, but it has to be sufficiently thick to avoid mechanical failures. For this reason, the channel length is sized with a fraction of the plate wall thickness, i.e., 1.5 mm. Thus, through Equation (7) it is possible to compute the diameter of the Helmholtz resonator neck, e.g., 1.1 mm for a frequency of 1100 Hz. The Helmholtz resonator is implemented in the numerical model through a pipe (HLLWx) with three nodes, which represents the neck, and a fixed volume for the cavity. The wall compliance of the resonator is modeled by considering the Young's modulus of the hydraulic plate as well as the wall thickness of the neck. The resonator is located at the middle length of Pipe 4. Table 2 reports the Helmholtz resonator parameters adopted in the sensitivity analysis. Figure 7 shows the three models implemented for the investigation of the three modified solutions. The results obtained with the models are analyzed in the result chapter, where the pressure oscillations measured at the system pressure transducer are evaluated in function of the pump speed rate. The actual pump operating range is considered and the simulations are performed between 0 rpm and 10,000 rpm with a slope of 500 rpm/s. Furthermore, the fluctuations of the pressure signals are investigated through spectral maps analysis as well as order tracking techniques, where the oscillating profiles are obtained by subtracting the mean pressure value to the system pressure signal predicted at the transducer.

Table 2. Helmholtz resonator parameters.

Parameter	Value
Vc	27 cm^3
L_N	1.5 mm
D_N	1.1 mm
f	1100 Hz

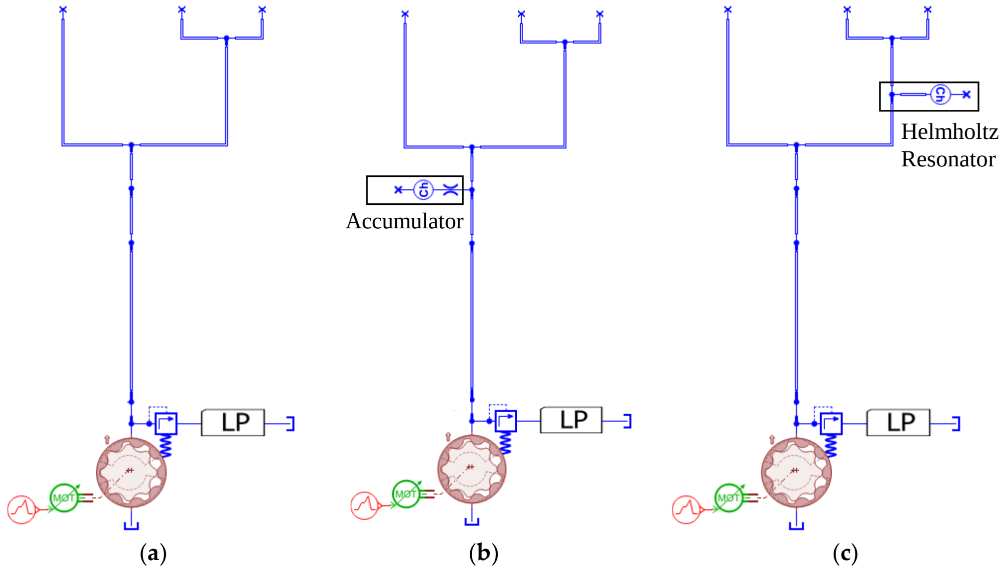
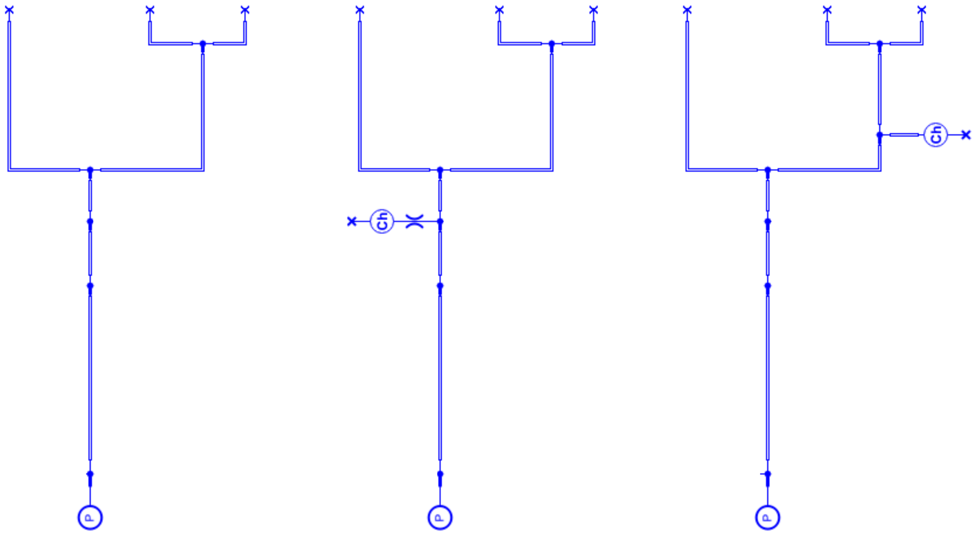


Figure 7. Modified models: (**a**) with 19 teeth Gerotor pump; (**b**) with the accumulator; (**c**) with the Helmholtz resonator.

2.3. 0D/1D Model for Linear Analysis

Simplified models, which analyze the frequency response of the circuit through linear analysis, have been implemented to reduce the simulation time and to permit circuit optimization since the computational effort for the simulation of dynamic models is relatively high in terms of computational time. Linear analysis enables to predict the pipeline resonant frequencies, by computing the systems' eigenvalues and eigenvectors derived from the Jacobian matrix, and to evaluate their frequency response in specific points of the hydraulic circuit in terms of magnitude and phase shift on Bode diagrams. Therefore, the linear analysis has been adopted to predict the frequency response of the system at the system pressure transducer position and to highlight resonance peaks due to system's eigenvalues and eigenvectors. The pipelines adopted in the dynamic analysis are investigated through linear analysis to compare the analogies and the differences of the two methodologies. The first difference is that the linear analysis does not allow to investigate the pump architecture, but only the pipeline domains, and therefore, the influence of the pump teeth number cannot be investigated. For this reason, the pipeline architectures analyzed through linear analysis are the base circuit, the base circuit with the accumulator, and the base circuit with the Helmholtz resonator. Figure 8 reports the three models of the analyzed circuits. Constant pressure is defined at the inlet boundary of the circuits since they are analyzed at the regulation pressure of the PRV, while the PRV is neglected since only the frequency response of the pipeline is considered in the analysis. To evaluate the frequency response at the system pressure transducer, the state observer is set at the transducer location, while the inlet pressure is set as control. Furthermore, the distributive inertia model (HL004x) is set for the pipeline, instead of the submodel (HLLWx), since it embeds frequency dependent friction and it results as particularly suitable for this analysis. Each pipeline is discretized in five points, while two points are defined in the Helmholtz resonator neck, as defined in the dynamic analysis. The magnitude of the Bode diagrams is used to analyze the frequency response of the circuits.



(a) (b) (c)

Figure 8. Hydraulic circuits for the linear analysis: (**a**) base circuit; (**b**) base circuit with the accumulator; (**c**) base circuit with the Helmholtz resonator.

2.4. Fluid Properties

The performance of the hydraulic circuit operating with petroleum based as well as eco-friendly oils is numerically investigated. As a petroleum-based oil, the Automatic Transmission Fluid (ATF) used in the power transmission is considered. The fluid properties are set in the model (FP04) in terms of density, bulk modulus, and viscosity at 90 °C and their values are reported in Table 3. Isothermal flow is considered in the analysis. Furthermore, the hydraulic behavior of the circuit is investigated by implementing in the model the fluid properties of an eco-friendly oil, the BIOHYDRAN TMP, which is a synthetic biodegradable hydraulic fluid. This oil is characterized by high viscosity at high temperatures, which is a fundamental property for this application since high temperatures are reached at regime and the efficiency of the system is therefore guaranteed. The model (FP04) is adopted also in this case to implement the fluid properties within the numerical model and the density and the viscosity of the ECO oil are collected in Table 3.

Table 3. Fluid Properties.

Fluid ()	β (MPa)	ρ (kg/m^3)	ν (cSt)
ATF	1350	780	7.9
ECO	1350	873	11.4

2.5. Simulated Cases

Table 4 reports the simulated cases. Cases #1 to #4 are the cases of the dynamic analysis of the circuits operating with the ATF, where Base_Dyn_11 is the base circuit case, while Base_Dyn_19, Accum_Dyn_11, and Accum_Dyn_11 are the cases of the modified circuits, i.e., with the pump with 19 teeth, with the accumulator, and with the Helmholtz resonator, respectively. Cases #5 to #7 are the models investigated through linear analysis, where Base_Lin is the base circuit, while Accum_Lin and HR_Lin are the cases with the accumulator and the Helmholtz resonator, respectively. Finally, Base_Dyn_11_ECO is the case where the eco-friendly oil is adopted as operating fluid of the base circuit and dynamic analysis is performed.

Table 4. Simulated cases.

Case	Simulation Name	Analysis	z_2	Accumulator	Helmholtz Resonator	Fluid
#1	Base_Dyn_11	Dynamic	11	No	No	ATF
#2	Base_Dyn_19	Dynamic	19	No	No	ATF
#3	Accum_Dyn_11	Dynamic	11	Yes	No	ATF
#4	HR_Dyn_11	Dynamic	11	No	Yes	ATF
#5	Base_Lin	Linear	11	No	No	ATF
#6	Accum_Lin	Linear	11	Yes	No	ATF
#7	HR_Lin	Linear	11	No	Yes	ATF
#8	Base_Dyn_11_ECO	Dynamic	11	No	No	ECO

3. Results

The results obtained numerically are reported in this chapter. The results computed with the steady state model are firstly analyzed to find resonance frequencies in function of the pump speed rate, while the effect of the resonance is furtherly investigated through dynamic simulations. Furthermore, the results obtained through linear analysis are reported as frequency response to highlight the resonance frequencies of the circuit. Finally, the results obtained adopting the eco-friendly fluid as operating fluid are compared to the ones obtained with the ATF in order to evaluate the performance of the circuit operating with the biodegradable fluid.

3.1. Steady State Model

The pump speed rates that may generate resonance are investigated by implementing the algorithm reported in Figure 3. In fact, by comparing the pipeline lengths with the resonance lengths predicted at different pump speeds and orders, it is possible to detect the pump speeds that generate resonances. The pipeline lengths analyzed are the lengths from the pump to the circuit extremities, i.e., $L_{PV\,C1}$ and $L_{PV\,C2}$, where $L_{PV\,C1}$ = 320 mm and $L_{PV\,C2}$ = 310 mm. Table 5 reports the calculation sheets implemented to localize the resonance lengths comparable with the circuit lengths and it is easy to recognize that the pipe lengths $L_{PV\,C1}$ and $L_{PV\,C2}$ are excited by the pump speeds of 5500 and 5750 rpm, respectively. This result shows that the pump speed generates resonance phenomena about these speeds and further investigation must be conducted to evaluate the resonance intensity which may affect the hydraulic pipeline. Numerical models have been therefore implemented to deeper investigate and predict the resonance behavior that may affect the actual hydraulic domain.

Table 5. Steady state algorithm results.

	Steady State Model												
	Pump Speed = n_{Pump} (rpm)												
	1000	2000	3000	4000	5000	5250	**5500**	**5750**	6000	7000	8000	9000	10,000
	Frequency = f (Hz)												
	183	367	550	733	917	963	**1008**	**1054**	1100	1283	1467	1650	1833
Order	**Wavelength = λ(mm)**												
1	7144	3572	2381	1786	1429	1361	**1299**	**1242**	1191	1021	893	794	714
	Resonance Length = L_{CH} (mm)												
1	1786	893	595	446	357	340	**325**	**311**	298	255	223	198	179

3.2. Dynamic Analysis of the 0D/1D Models

The numerical results obtained through dynamic analysis are reported in this sub-chapter. The pressure ripple measured at the sensor is investigated for the different models. The signals are analyzed in function of the pump speed to identify the resonances locations. Furthermore, the oscillating profiles are analyzed through spectral maps and order tracking techniques to evaluate the resonance phenomena and the pumps orders that generates it.

3.2.1. Base Circuit

Figure 9a reports the pressure measured at the system pressure transducer for the base circuit, i.e., Base_Dyn_11, where the results are shown in function of the pump speed. From the figure, it is possible to see that a main resonance phenomenon is predicted by the model at 5220 rpm. This behavior confirms the results of the steady state model, which approximated the resonance between 5500 rpm and 5750 rpm by considering only the circuits' lengths, while the numerical approach extended the analysis to the real pipeline geometry by considering the interaction between the two hydraulic branches as well as the real circuit dimensions. Furthermore, this approach enables to evaluate the intensity of the resonance and it highlights that the pressure oscillations at the resonance speed exceed the ones at lower speeds several times, e.g., over 5 times the oscillation amplitude of 2500 rpm. The pressure oscillations are analyzed through spectral analysis and order tracking technique to investigate the frequencies and the orders that affect the dynamic response of the circuit. Figure 9b reports the spectral maps that show the intensity of the pressure oscillations in function of the pump speed and the frequency. From the map, it is possible to see that along the pump excitation strip the maximum oscillation intensity is reached at 5220 rpm, which corresponds to about 1000 Hz of excitation frequency. This result confirms the prediction of the steady state model and shows the importance of addressing the real geometry to predict the actual hydraulic behavior. Figure 9c shows the order intensities in function of the pump speed to shows the most influent on the pressure oscillation excitation. Only the order 11 and 12 and their multiples are shown below since they are the teeth numbers of the Gerotor gear pump and the most influent. In fact, the order tracking technique uses multiples of the running speeds (orders) as frequency base and the hydraulic circuit is excited by multiple of the gears' active teeth for each pump revolution. It is interesting to note that the 11th order, which leads to the internal gear teeth, is the most influent and almost generates the whole oscillation amplitude. In fact, for each revolution of the Gerotor pump, the internal gear generates 11 flow ripple pulsations and therefore this flow excitation is the most influent for pressure ripple excitation. Furthermore, accordingly with the steady state analysis, the 11th order generates the resonance phenomenon, while the influence of its multiple is negligible.

Figure 10 highlights the accuracy of the numerical model for the prediction of the pressure oscillations through the pipeline by comparing the simulated signal with the experimental one. An ad hoc test rig, designed to test instrumented power transmission systems on actual operating conditions, is adopted to perform the test. A fly-wheel layout is adopted, where the transmission is actuated with an electric motor in the range [500, 10,000] rpm. The pressure is acquired with the system pressure transducer at the acquisition frequency of 20 kHz. As it is possible to see from the graph, a speed ramp is tested, and the resonance phenomena is experimentally measured at about 5200 rpm on the growing and the decreasing sides of the ramp. By comparing the experimental and the numerical results it is possible to see that the numerical model accurately predict the resonance behavior in terms of signal shape and amplitude.

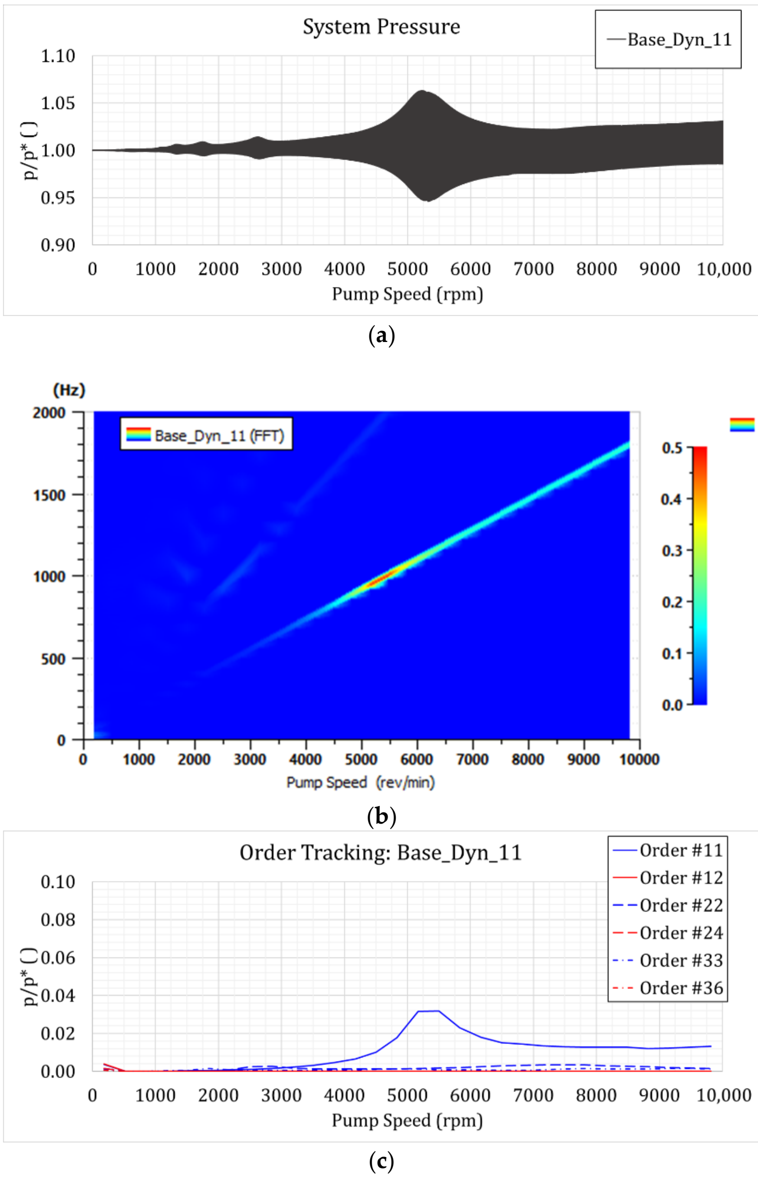


Figure 9. Results of Base_Dyn_11: (**a**) system pressure; (**b**) spectral map; (**c**) order tracking.

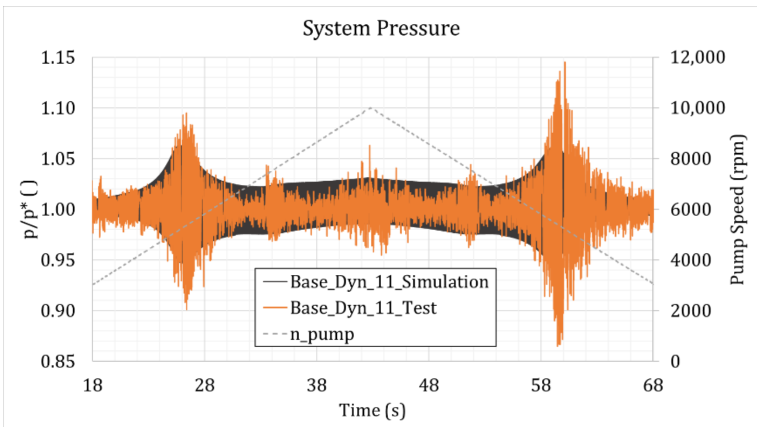
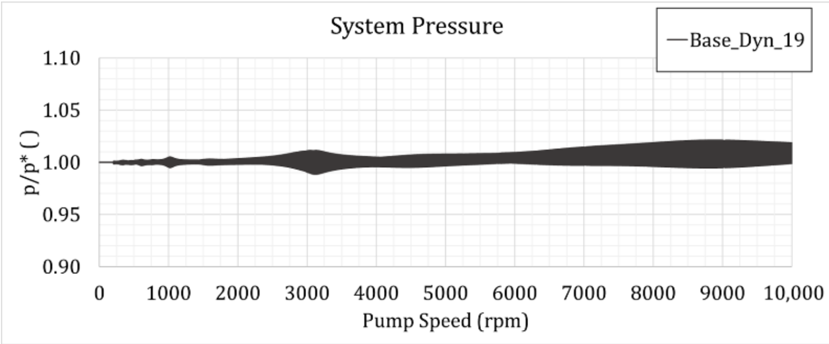


Figure 10. Validation of the Base_Dyn_11 numerical model: Simulation vs. Test.

Therefore, with these numerical tools it is possible to evaluate and predict resonance behavior as well as pressure waves propagation through the pipelines, highlighting resonance regimes and specific orders that generate them. These resonance behaviors should be reduced since they might affect the performance of the hydraulic circuit operating at specific regimes, compromising the dynamic performance, and generating instabilities. In the following paragraph, some possible solutions for resonance attenuation are numerically investigated by simulating the modified hydraulic circuits.

3.2.2. Modified Circuits

The results obtained with the modified circuits are collected in this paragraph, i.e., Base_Dyn_19, Accum_Dyn_11, and HR_Dyn_11, and they are compared with the results of the base circuit, i.e., Base_Dyn_11, in terms of pressure oscillations, spectral maps, and orders. Figure 11 shows the pressure oscillation measured at the system pressure transducer for the three cases. It is possible to see that the resonance predicted with the base circuit is attenuated in all the modified circuits, but the dynamic responses of the modified circuits present different shapes based on the adopted solution. Figure 11a reports the pressure predicted if a Gerotor pump with 19 teeth for the internal gear is adopted. This solution reduces the flow ripple, due to the increment of the pumping vanes, and modifies the excitation frequency of the pump, resulting into a more damped pressure response with a small resonance at 3130 rpm. Therefore, this approach enables to perform preliminary simulations based on real pump geometrical features and the prediction of resonance locations and magnitudes due to pump design. Preliminary analysis are particularly effective in terms of cost savings for industrial products with large scale market since modification during the product development prototyping may result into high costs for component design, chassis modifications and testing. Figure 11b reports the dynamic response when the accumulator is adopted. This solution shows that the resonance behavior is attenuated, and the peak is predicted at lower pump speeds, i.e., 2310 rpm. This solution results to be particularly efficient, but it requires high volumes that for power transmission systems of high-performance cars is not always available due to their compact design. This numerical tool is then useful for evaluating the design of the optimum accumulator volume based on geometrical constraints. Finally, Figure 11c reports the pressure oscillations predicted for the circuit with the Helmholtz resonator. The graph shows that the peak at 5220 rpm is suppressed, but two new peaks are predicted at lower and higher speeds. In fact, the resonator modifies the response of the circuit and in this specific case it determines two new resonances. The first resonance, at 3100 rpm, has a lower amplitude than the one predicted with Base_Dyn_11, and the introduction of this Helmholtz resonator can be considered an improvement for the attenuation of the pressure peak, while the second one occurs at high speed rate, i.e., 9320 rpm, and it may generate instabilities in critical conditions if the pump must address these high velocity rates. Therefore, this modeling strategy can predict the dynamic response of a system where a Helmholtz resonator is integrated in a specific position of the circuit and the influence of the resonator design parameters can be investigated. Helmholtz resonators are more compact solutions than accumulators, resulting more suitable for high-performance transmission systems where the geometrical constraints are generally strict, but they are more sensible to geometrical tolerances and require particular attention of the manufacturing process, especially for the resonator neck.



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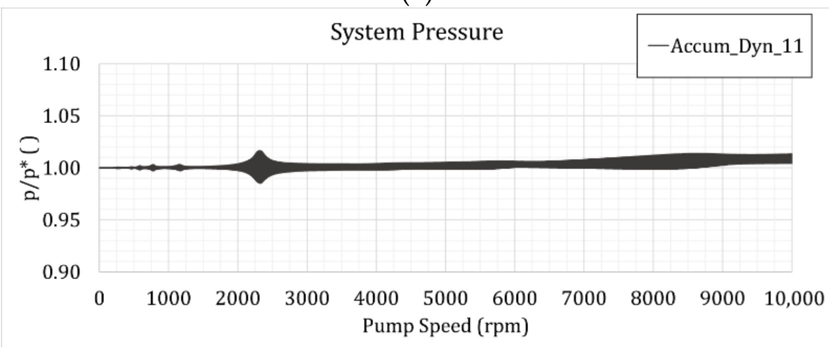

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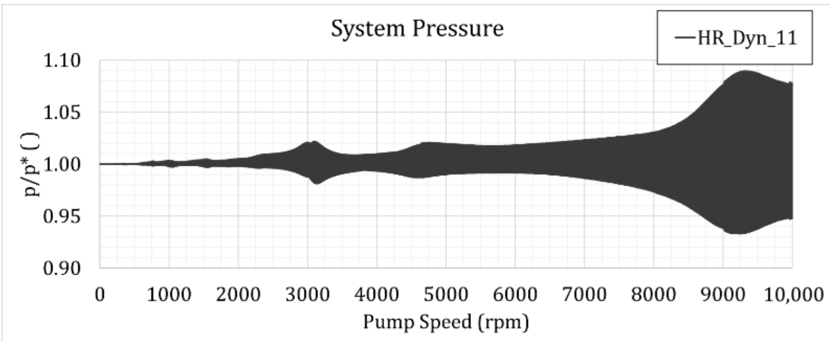


(**c**)

Figure 11. System pressure of: (**a**) Base_Dyn_19; (**b**) Accum_Dyn_11; (**c**) HR_Dyn_11.

Figure 12 reports the spectral maps of the fluctuations of the system pressures for the three analyzed circuits. As we saw in the base circuit analysis, these maps highlight the intensity of the resonation in function of the pump speed and frequency. Figure 12a shows the spectral map obtained by analyzing the oscillating pressure signal of the base circuit operating with the 19 teeth pump. By comparing this map with the one of the base circuit, see Figure 9b, it is possible to notice that the resonance spot predicted at 1000 Hz for the 11-teeth pump is predicted in this case at lower speed and it results attenuated in amplitude. From the spectral map it is possible to notice that the slope of the pump excitation frequency is higher for the 19-teeth pump than the 11 teeth, accordingly with Equation (4), and the maximum amplitude is predicted when the pump strip crosses the excitation frequency of the pipeline, i.e., at the resonance speed of 3130 rpm and 1000 Hz, accordingly with the pipeline characteristic lengths. Furthermore, by increasing the teeth number of the pump, the pump flow ripple decreases and consequently the dynamic response of the pipeline decreases, too. Figure 12b shows that by introducing an accumulator in the circuit, the magnitude of the oscillations is damped along of the pump strip, where the maximum amplitude of the resonance is found around 2310 rpm and 462 Hz. Figure 12c reports the spectral map of the base case with the Helmholtz resonator, where it is possible to recognize the two resonances predicted previously with the dynamic response at 3100 rpm and 9320 rpm and about 580 Hz and 1730 Hz, respectively.

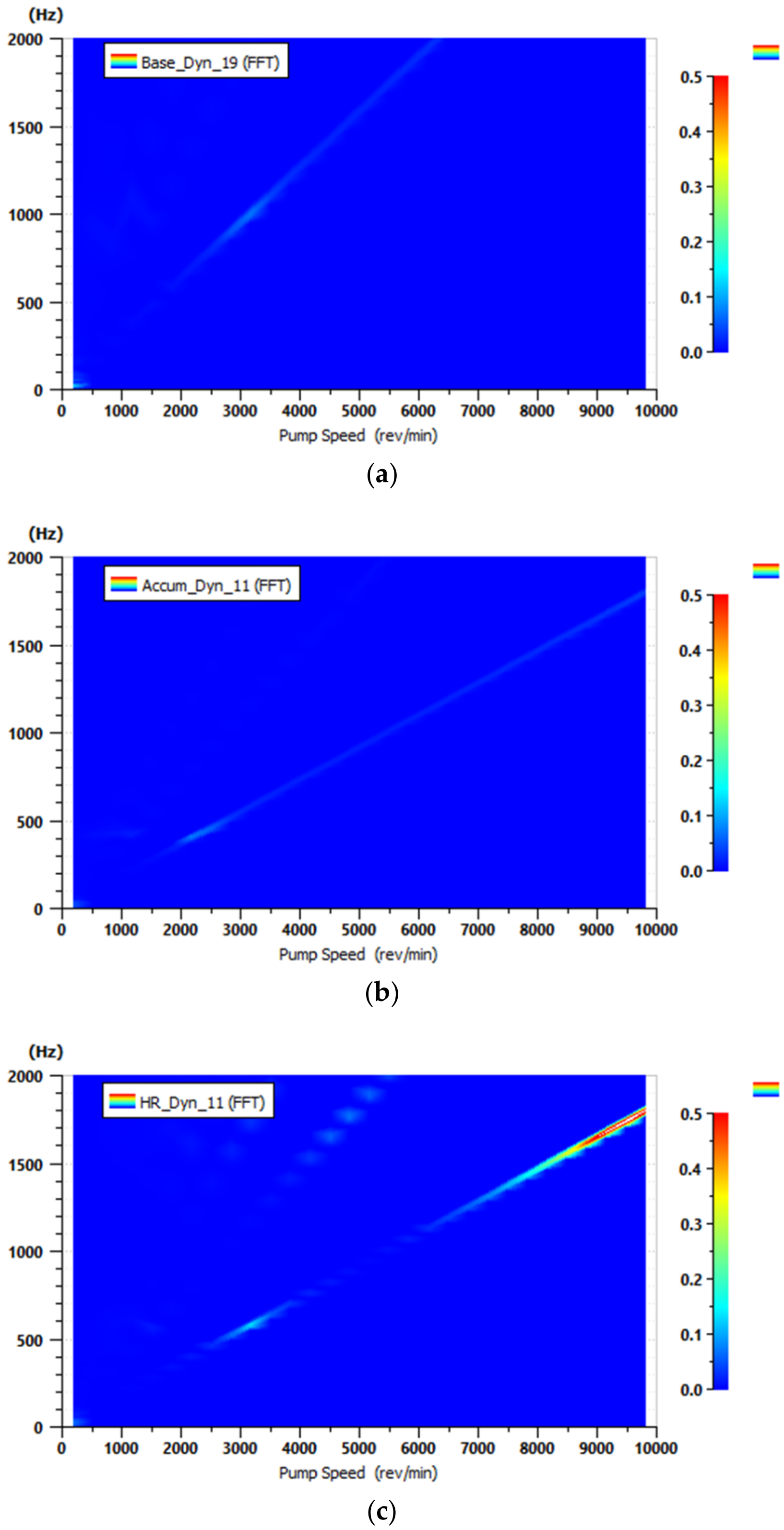


Figure 12. Spectral maps of: (**a**) Base_Dyn_19; (**b**) Accum_Dyn_11; (**c**) HR_Dyn_11.

Finally, the fluctuations are analyzed through order tracking technique to identify the most influent orders on the fluctuation prediction. As mentioned in the previous paragraph, the order 11 resulted to be the most influent for the base circuit where the peak was highlighted at about 5220, see Figure 9c. Figure 13a analyzes the order 19 and 20 and their multiples for Base_Dyn_19 since the circuit presents a pump with 19 teeth for the internal gear and 20 for the external one. From the figure, it is possible to see that the orders amplitudes are smaller than the ones of the base circuit, but the order of the internal gear is also in this case the most significant in magnitude since it is the teeth number of the active gear. As expected, the maximum value is predicted for the resonance peak at 3130 rpm where the pump excites the circuit with a ripple with a frequency comparable with the circuit characteristic lengths. This graph shows that it is fundamental to identify specific orders, based on the pump design parameters, to analyze with order tracking technique the pressure fluctuations. No agreement with pressure and spectral map results will be found if wrong orders are selected and therefore no significant results will be obtained. Figure 13b shows the graph for the case with the accumulator, i.e., Accum_Dyn_11, and accordingly to the dynamic analysis as well as the spectral map, the small resonance peak is predicted about 2310 rpm for the order 11. This behavior confirms that the introduction of an accumulator damps the dynamic behavior of Base_Dyn_11 by reducing the intensity and the speed rate of the resonance. Figure 13c reports the graph for the base circuit with the Helmholtz resonator, i.e., HR_Dyn_11. From the figure, it is possible to see that the two resonant behaviors are predicted at about the pump speed of 3100 rpm and 9320 rpm, where also in this case the pump order 11 is the responsible of the pressure fluctuations. Therefore, through order tracking technique, it is possible to identify and quantify the orders that generate resonant behaviors, which, also in this case, is the teeth number of the internal gear of the Gerotor. Table 6 collects the resonance pump speeds predicted numerically and the excitation frequency calculated through Equation (4).

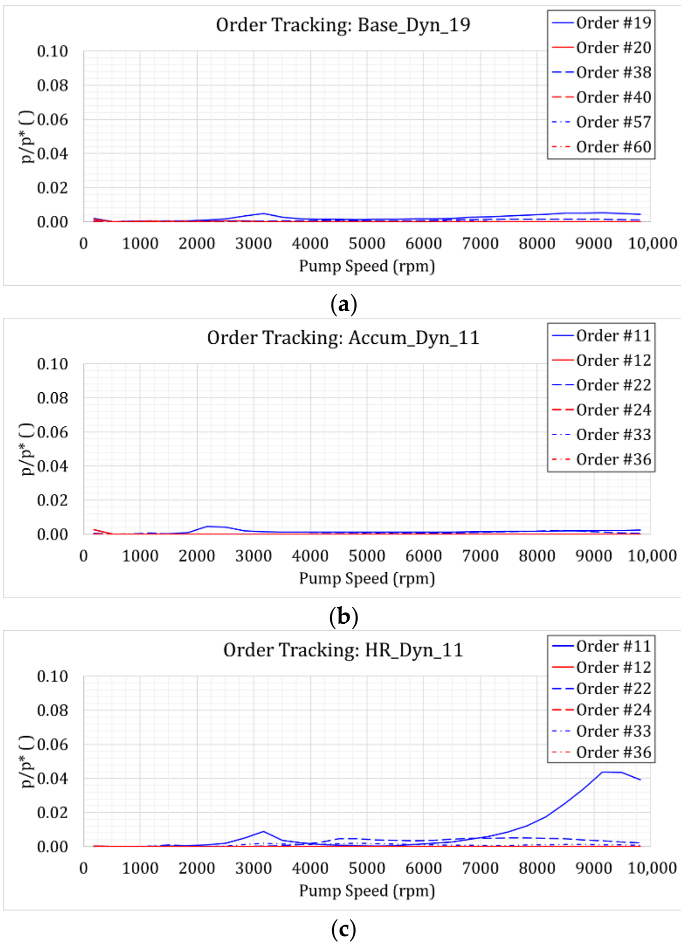


Figure 13. Order tracking graphs of: (**a**) Base_Dyn_19; (**b**) Accum_Dyn_11; (**c**) HR_Dyn_11.

Table 6. Resonance pump speeds.

Simulation	Pump Speed (rpm)	f (Hz)
Base_Dyn_11	5220	957
Base_Dyn_19	3130	991
Accum_Dyn_11	2310	423
HR_Dyn_11	3100	568
	9320	1708

3.3. Linear Analysis of the 0D/1D Models

In this subchapter, the frequency responses of the circuits are evaluated through linear analysis. Figure 14 compares the responses of the three circuits. The resonance response is analyzed in the interval [1, 2000] Hz since the maximum pump speed is 10,000 rpm and it leads to a maximum excitation frequency of 2000 Hz for a pump with 11 teeth of the internal gear. The frequency response of the base circuit, i.e., Base_Lin_11, predicts a resonance peak at 987 Hz, according to the steady state and dynamic analysis. The analysis of the circuit with the accumulator shows that by adding this component the peak moves to a lower frequency, i.e., 410 Hz, and it decreases in magnitude. The accumulator does not significantly affect the shape of the frequency response of the system, but it modifies the position of the peak and it damps the magnitude of the pressure oscillations. Finally, the diagram of the base circuit with the Helmholtz resonator shows that this component particularly affects the response, by suppressing the resonance frequency at 987 Hz, by generating a different response. In fact, the resonator modifies the system eigenvalues and in this case two new resonance frequencies at lower and higher frequencies are generated around the damped frequency, i.e., 545 Hz and 1676 Hz.

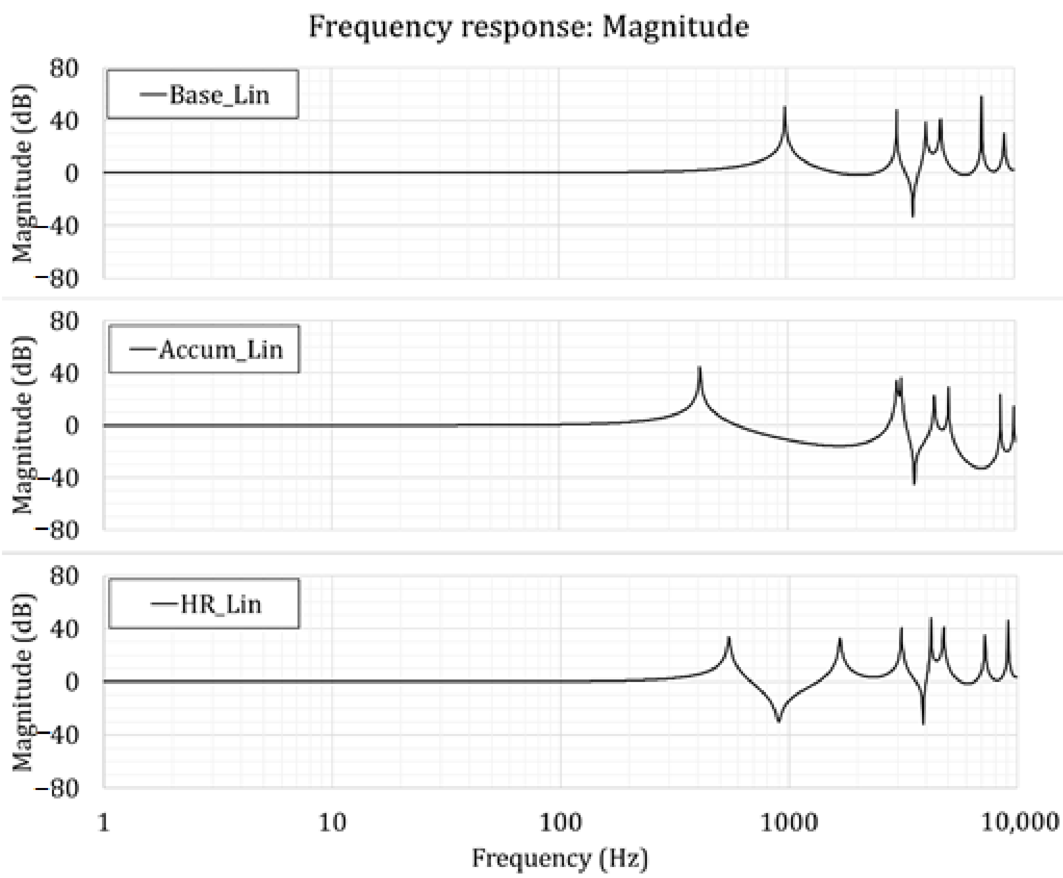


Figure 14. Frequency response of Base_Lin, Accum_Lin and HR_Lin.

Table 7 collects the resonance frequencies determined with the linear analysis and the corresponding pump speeds calculated through Equation (4). Figure 15 compares the results obtained with the dynamic and the linear analysis in terms of resonance frequency and pump speed. From the figure, it is possible to see that the values are close, and that the linear analysis can be considered a good approximation for the prediction of the resonant frequencies. The two approaches are therefore complementary since the dynamic one evaluates the pipeline frequency resonance as function of the pump regime, which excites the ducts, while the linear one predicts the frequency resonance of the pipeline due to the geometrical domain, and therefore it is possible to calculate the pump speed rate which excites the predicted resonance frequency. Furthermore, it is possible to affirm that the linear analysis can be considered a good design tool for the evaluation of the parameters influence due to the demonstrated accuracy and the computational effort required to perform sensitivity analysis can be significantly reduced. Nevertheless, dynamic analysis is required to evaluate non-linearities as well as the actual magnitude of the pressure oscillations.

Table 7. Resonance frequency.

Simulation	f (Hz)	Pump Speed (rpm)
Base_Lin	987	5385
Accum_Lin	412	2246
HR_Lin	545	2974
	1676	9144

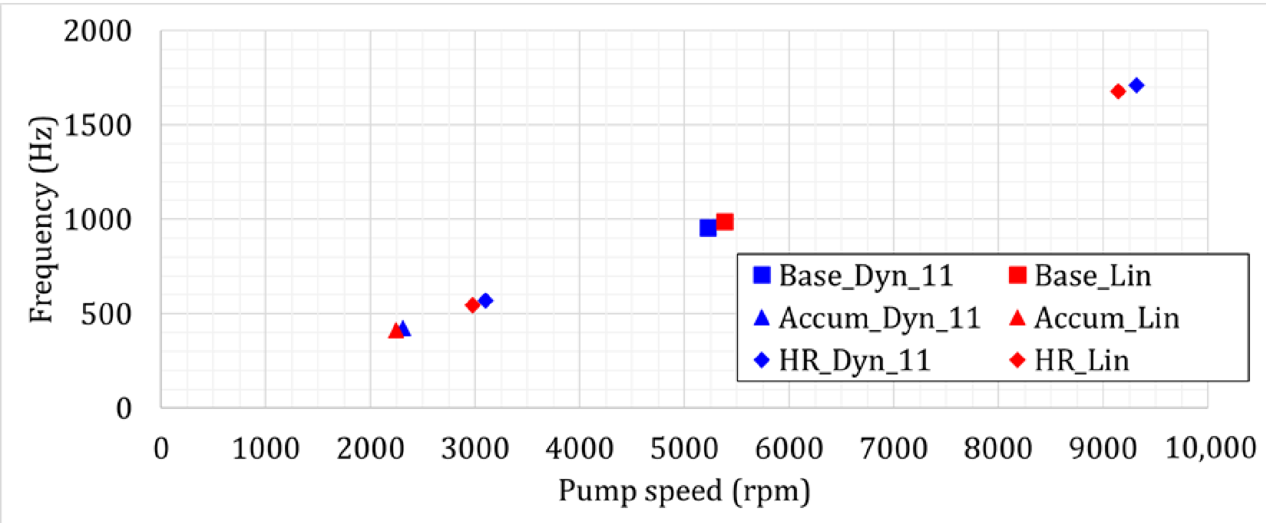
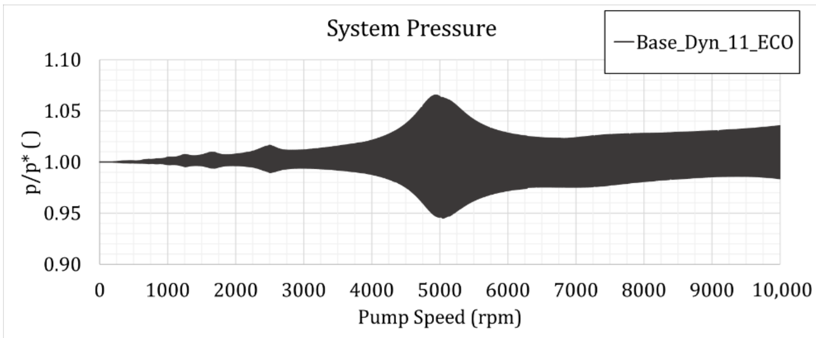


Figure 15. Resonance prediction: dynamic vs. linear analysis.

3.4. Fluid Sensitivity

In this paragraph, the results obtained with the model operating with the eco-friendly fluid are reported. The dynamic analysis is conducted to evaluate the performance of the circuit and the results are also in this case reported in terms of system pressure prediction and fluctuation analysis through spectral map and order tracking. Figure 16a shows the system pressure where it is possible to see that, also in this case, a resonance behavior is predicted at about 5000 rpm. This behavior agrees with the prediction that can be computed through steady state analysis. In fact, the density of this eco-friendly oil is higher than the one of the ATF at 90 °C and it results into a lower celerity value and therefore a lower pump speed excitation rate. The amplitude of the resonance resulted to be close to the one predicted with the ATF and this result confirms that a similar fluid dynamic behavior can be obtained by adopting the ECO fluid in the hydraulic circuit of the power transmission without reducing the performance of the system. Furthermore, by analyzing the spectral map, Figure 16b, and the order tracking graph, Figure 16c, it is possible to see that the same phenomena predicted with the ATF are calculated with the ECO fluid, such as the resonance behavior at 5000 rpm, which leads to 916 Hz and the order 11 is the most influent for the pressure oscillation excitation. Thus, this analysis confirms that the eco-friendly fluid could be used in this specific application addressing the fluid dynamic behavior of the ATF. Nevertheless, for this specific application, endurance of the oil performance must be guaranteed during the product life span, avoiding fluid degradation, and therefore, particular attention will be devoted on this topic for further developments and analysis.



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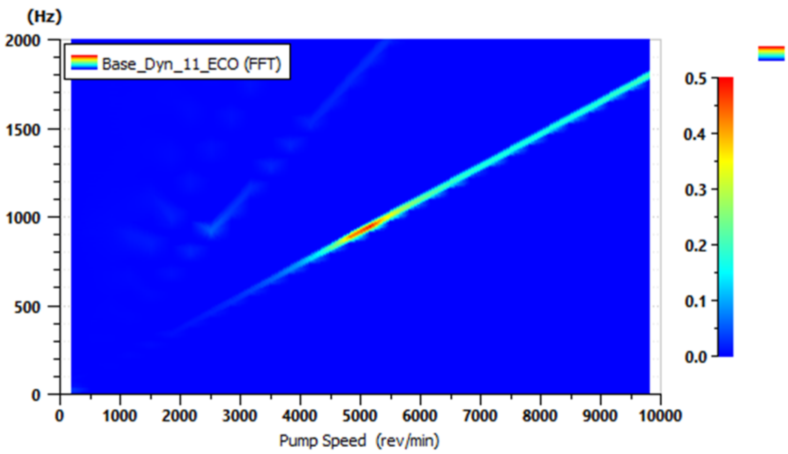

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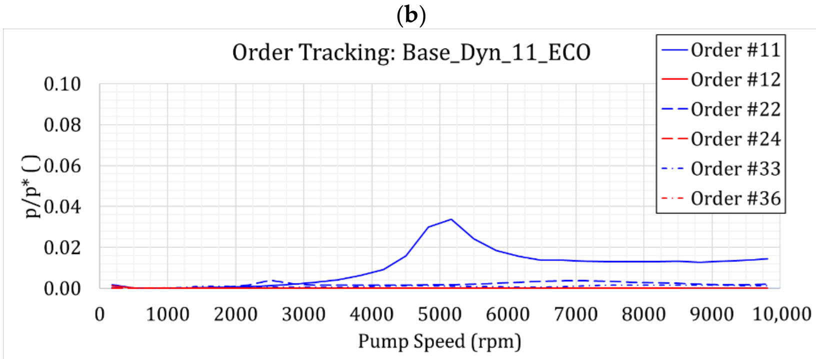

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Figure 16. Base_Dyn_11_ECO's results: (**a**) system pressure; (**b**) spectral map; (**c**) order tracking.

4. Conclusions

In conclusion, this paper analyzed wave propagation through the actuation circuit of a power transmission system excited by a Gerotor gear pump by means of numerical approaches. The actuation circuit of a DCT power transmission system has been numerically investigated since numerical models are important tools for the circuit design especially for the engineering applications where strict geometrical constraints and high performance are required. A steady state analysis was conducted to identify resonance behaviors due to circuit lengths and pump frequency excitation based on closed ducts wave propagation theory. By comparing the pump excitation frequency with the circuit lengths, two resonance frequencies were found at 1008 Hz and 1054 Hz, which leads to 5500 rpm and 5750 rpm of the Gerotor pump speed.

Therefore, a dynamic model of the actuation circuit was implemented to investigate the hydraulic behavior of the real geometrical domain of the hydraulic circuit, where CFD-1D pipes were used to address the pressure propagation through the pipelines and the pressure was evaluated at the system pressure transducer location. The numerical model confirmed the resonance behavior at 5220 rpm accordingly to the results of the steady state model. Spectral maps as well as order tracking techniques were adopted to analyze the pressure fluctuations, where the former analysis highlighted the resonance frequency of 957 Hz along the pump excitation strip, while the latter showed that the order 11, which is the internal gear teeth number, was the most influent. Furthermore, three possible design solutions for resonance attenuation were numerically investigated, i.e., a Gerotor pump with a larger number of internal gears, an accumulator and a Helmholtz resonator. A larger number of teeth reduced the flow ripple and only a small resonance behavior was predicted when the excitation frequency resulted comparable with the circuit lengths, 991 Hz and 3130 rpm. By adding an accumulator to the circuit, a significant attenuation of the resonance was found, and the peak moved to lower speed rate. The accumulator requires high volumes that are usually not suitable for high-performance cars' transmission systems, therefore a more compact solution, a Helmholtz resonator, was designed to suppress the resonance frequency. The results showed the resonance suppression, but the generation of two new resonances at lower and at higher speed rates, where the former had a lower magnitude and can be considered an improvement, while the latter had a significant intensity and may generate instability at high speed rates. A numerical approach, which adopted linear analysis to evaluate the resonance frequency due to the circuit geometrical features, was also implemented to decrease significantly the computational effort required for the simulation of each case. The results showed that the resonance frequencies predicted with the linear analysis agreed with the ones of the dynamic analysis and this result confirmed that linear analysis can be adopted to find optimized solutions, while dynamic analysis is necessary to evaluate the magnitude of the phenomena. Finally, the fluid dynamic behavior of the circuit operating with an eco-friendly fluid was evaluated and by comparing the results with the ones obtained with the ATF, similar performance was obtained. Therefore, eco-friendly fluids are suitable for this application and may improve the eco compatibility of the transmission systems.

Author Contributions: Conceptualization, M.M. and S.T.; methodology, M.M.; software, S.T.; validation, M.M. and S.T.; formal analysis, M.M. and L.M.; investigation, S.T.; resources, M.M. and L.M.; data curation, S.T.; writing—original draft preparation, M.M. and S.T; writing—review and editing, M.M. and S.T.; visualization, S.T.; supervision, M.M.; project administration, M.M.; funding acquisition, M.M. and L.M. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by Emilia Romagna Region—project PG/2018/632152, Efflics—Eco Friendly Fluidics for Fluid Power.

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: The data presented in this study are available on request from the corresponding author. The data are not publicly available due to privacy restrictions.

Conflicts of Interest: The authors declare no conflict of interest.

Abbreviations

β	Bulk Modulus
λ	Wavelength
ρ	Density
c	Celerity
A_N	Helmholtz resonator neck cross sectional area
ATF	Automatic transmission fluid
CFD	Computational Fluid Dynamics
C_f	Friction Source
C_g	Gravity source
d_N	Helmholtz resonator neck's diameter
$(L/d)_x$	Length/diameter of the pipe x; x = [1, 7]
DCT	Dual-Clutch Transmission
ECO	Eco-friendly oil
f	Frequency
L	Pipeline length
L_N	Helmholtz resonator neck length
L_{CH}	Resonance length
LP	Low Pressure
$L_{PV\,C1}$	Length from the pump to the PV of clutch 1
$L_{PV\,C2}$	Length from the pump to the PV of clutch 2
n	Pump order
n_{Pump}	Pump speed rate
p	Absolute pressure
p^*	PRV cracking pressure
PRV	Pressure Regulation Valve
p_s	System pressure
PV	Proportional Valve
PVs	Proportional Valves
S	Cross-sectional area
u	Velocity
V_C	Helmholtz resonator volume
V_x	Volume of the pipe x; x = [1, 7]
z_1	Teeth of the Gerotor external gear
z_2	Teeth of the Gerotor internal gear
1D	One-dimensional

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