

Active Fluid Borne Noise Reduction for Aviation Hydraulic Pumps

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Abstract

The aviation environment holds challenging application constraints for efficient hydraulic system noise reduction devices. Besides strong limits on component weight and size, high safety and reliability standards demand simple solutions. Hence, basic silencers like inline expansion chambers and Helmholtz-Resonators are state-of-the-art aboard commercial aircrafts. Unfortunately, they do not meet today's noise attenuation aims regarding passenger comfort and equipment durability. Significant attenuation performance is expected from active concepts that generate anti-phase noise. However, such concepts remain a long term approach unless related costs, e.g. due to additional power allocation and real-time control equipment can be avoided. In this paper an active fluid borne noise attenuation concept is discussed that accounts for the mentioned constraints. An aircraft hydraulic pump is considered as main noise source. The active attenuator is an in-house rotary valve design. The basic feature is a known direct shaft coupling principle of pump and rotary valve, so no speed/ frequency control of the valve and no separate power supply are required. The common-shaft principle is further simplified here and proposed as integral feature of future "smart pumps".

KEYWORDS: Aircraft Hydraulic System, Hydraulic System Noise, Fluid Borne Noise, Pump Flow Ripple, Pump Impedance, Active Attenuation, Direct Drain Principle, Rotary Valve, Common Shaft Principle

1. Introduction

The potential & feasibility to reduce flow/pressure ripple – fluid borne noise (FBN) – of hydraulic pumps by a rotary valve in common shaft application was presented by Goenechea, see e.g. /1/. Taking the common shaft principle as starting point, further modifications are discussed in the concept-section, chapter 2, in order to satisfy the aviation constraints. Related design parameters and a simulation model are presented in chapter 3 and 4. The model is built using the time domain FBN analysis capabilities of the software DSHplus v3.9. In chapter 5, a laboratory demonstrator for experimental validation is shown. It uses the pump test circuit of a dedicated FBN test rig at TUHH. Relevant issues are listed below. Due to paper space only 1 to 3 are further discussed:

1. The attenuation performance in terms of pump flow ripple (source flow ripple) reduction in the pump discharge line for anechoic (reflection-free) conditions.
2. The attenuator impact on the pumps suction line, because todays passive reference silencers at pump discharge port tend to induce suction flow ripple.
3. The attenuator impact on the pump power balance in terms of a decreased stationary pressure/ flow or increased power consumption of the electric motor.
4. The attenuator impact on the apparent pump impedance (source impedance), which is a critical characteristic for interaction with (reflected) system ripples.

2. Concept

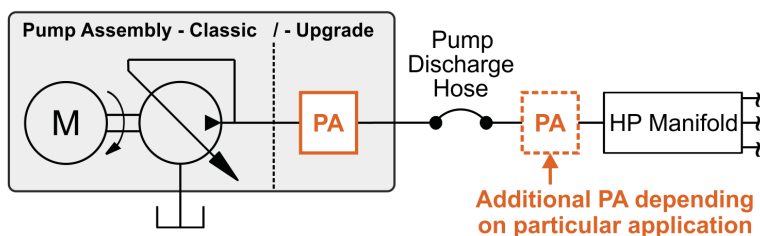
Active attenuators tend to be an attractive alternative to passive silencers regarding piston pump FBN reduction, especially for the fundamental and first harmonic pumping frequency. These carry most of the ripple power and passive solutions usually require large capacitance or inductance for lower frequencies, increasing silencer size/ weight.

In /1/ & /2/ some active attenuator concepts were compared, e.g. a Direct Principle draining dynamic flow to a lower pressure level, and an Active Resistance Principle, storing/releasing dynamic flow in/out a capacitance. Their use in aviation hydraulic systems was considered by Kohlberg et al. in /3/. One of the considered attenuator concepts was built and tested using a piezo-driven membrane actuator and is discussed in more detail in /4/. A high attenuation performance was achieved but came with many efforts. A realization by a rotary valve with pump shaft coupling enables to spare many efforts like a separate power supply and a speed/ frequency control.

A rotary valve (RV) can be designed compact and robust. However, a realistic aviation application would also demand to spare/ minimize any other onboard hardware and software for RV amplitude and phase control. Hence, a knowledge-based (pre-) design

of the pump-attenuator-assembly is required that only allows for simple adaptation functionalities, e.g. during maintenance on ground. Starting from typical state-of-the-art silencer solutions on aircrafts – in **Figure 1** shown with the proposed alternative concept – two technical realizations of the concept are given in **Figure 2**.

State-of-the-Art: **Passive Attenuators (PA) - Inline Silencer or Helmholtz Types**



Proposed Alternative: **Active Attenuator (AA) - Common Shaft Rotary Valve (RV)**

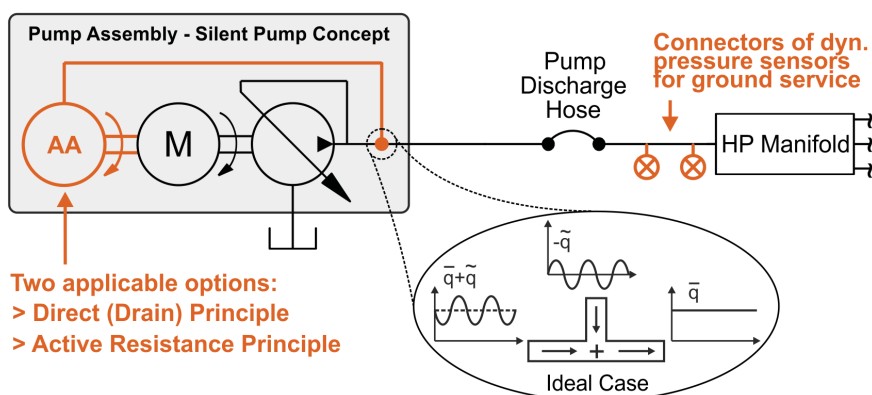


Figure 1: Fluid borne noise (FBN) reduction for aviation hydraulic pumps

The two technical realizations – Figure 2 – of the proposed active attenuator concept represent promising baseline candidates for an aviation specific application. That means a safety critical application where also failure cases of the RV are taken into account, like a stationary pressure drop due to RV sealing wear-out or burst. In that case, one or multiple switching valves (see SW or SW1/2) would have to disconnect the RV from the pump discharge line. The remaining components of the assembly would have to provide a minimum FBN attenuation performance until the next aircraft maintenance. E.g. for the shown Direct (Drain) Principle, the connection line length (L_1) has to be designed such, that the connection line itself acts like a passive side branch resonator (SW is closed) for the desired pumping frequencies during RV failure.

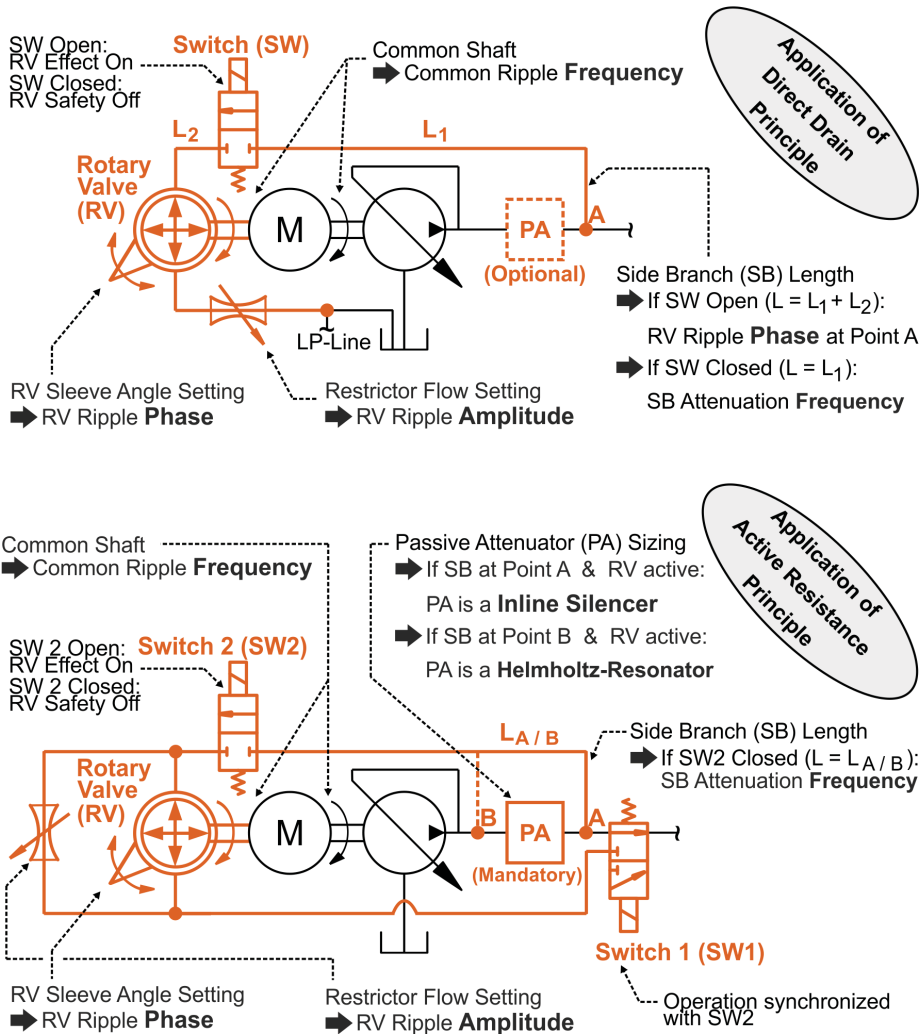


Figure 2: Considered technical realizations of the Silent Pump Concept and primary effect of dedicated tuning parameters on FBN (bold arrows)

3. Design

It is assumed that the RV ripple amplitude and phase are adjustable parameters acc. to Figure 2 during aircraft ground service. So the RV connection line length ($L_1/2$ or L_A/B) to Point “A” – and also the pump line length to Point “A” – become key preliminary design parameters of the assembly. This is not only for above mentioned RV failure cases. During normal operation, these lengths have great influence on the ripple phase of the respective ripple sources at Point “A”. This fact will remain for any active attenuator assembly acc. to Figure 2, where the RV is assumed as an add-on feature

for existing pumps and not yet as an integrated feature inside the housing of future “smart pumps” (negligible lengths). The relevant relations can be taken from a general solution of the Helmholtz equations /5/:

$$Q(x) = \left[F e^{-j\frac{\omega x}{c}} - G e^{j\frac{\omega x}{c}} \right] / Z_0 \tag{1}$$

$$Q(x)_{prog} = \left[(|F| e^{j\omega t}) e^{-j\omega x/c} \right] / Z_0 = \left[|F| e^{j(\varphi - \omega x/c)} \right] / Z_0 \tag{2}$$

In eq. (1), $Q(x)$ denotes a flow wave along a line, $F e^{-j\frac{\omega x}{c}}$ a pressure wave in positive x-direction (progressive) and $G e^{j\frac{\omega x}{c}}$ in negative x-direction (regressive), all travel with speed of sound c in the fluid. $e^{-j\frac{\omega x}{c}}$ & $e^{j\frac{\omega x}{c}}$ represent phase shifts increasing with distance and hence wave motion along the line. F & G are complex numbers for the line end boundary conditions, e.g. for pump & rotary valve. Z_0 is the characteristic pipe impedance. In eq. (2), the progressive part of eq. (1) is shown in detail. It is an important design equation for the proposed concepts: If $Q(x)_{prog}$ of pump & RV have the same amplitudes but are in anti-phase at intersection point A, than no progressive wave remains in a common discharge line and also no regressive (reflected) wave.

4. Simulation Model

The simulation models were built using the software DSHplus v3.9, see e.g. **Figure 3**.

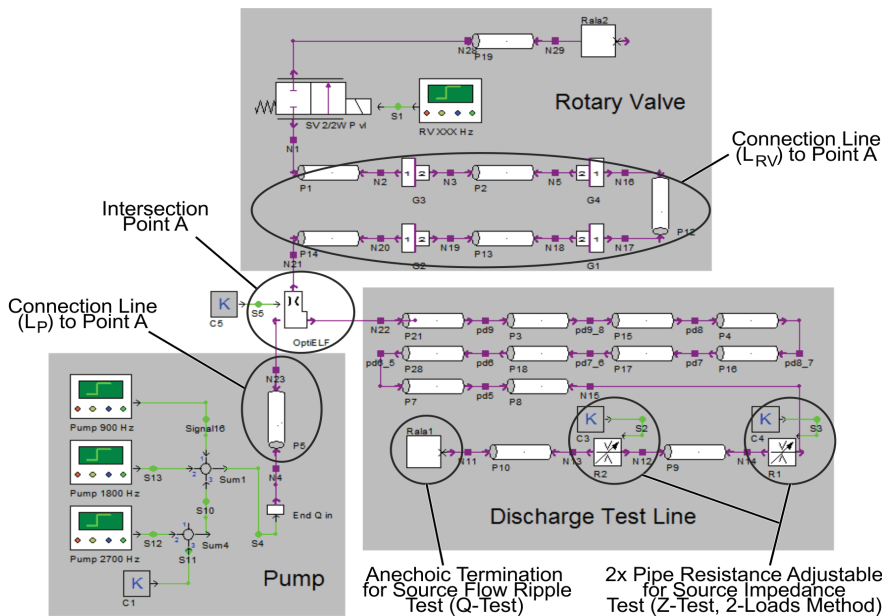


Figure 3: Model for Direct Drain Principle in DSHplus acc. to test rig setup (Figure 6/7)

Simulation and test results are limited here to the Direct Drain Principle, Figure 2, due to limited paper space. Besides, it was considered to be the most simple & compact technical realization, and hence to be most attractive one for an aviation application.

The Distributed Parameter Hydraulic Toolbox was applied which allows for ripple/wave investigation in time domain. The element discretization (discharge line pipe elements), Fig.3, is chosen acc. to a test rig to account for its sensor spacing, pipe diameters etc. The noise sources can be represented in multiple ways. The pump can be given by Function Generators to build sinusoidal signals. They form the flow ripple input of an open-ended pipe model ($Q(0) = Max, P(x) = 0$). Such flow ripple sources were also built from Pump Characteristic Models (look up tables, e.g. test data). **Figure 4** gives simulation results for the connection line length of pump (L_P) & rotary valve (L_{RV}). The three markers in Figure 4 (O, L1, L2) are related to the experimental setup below.

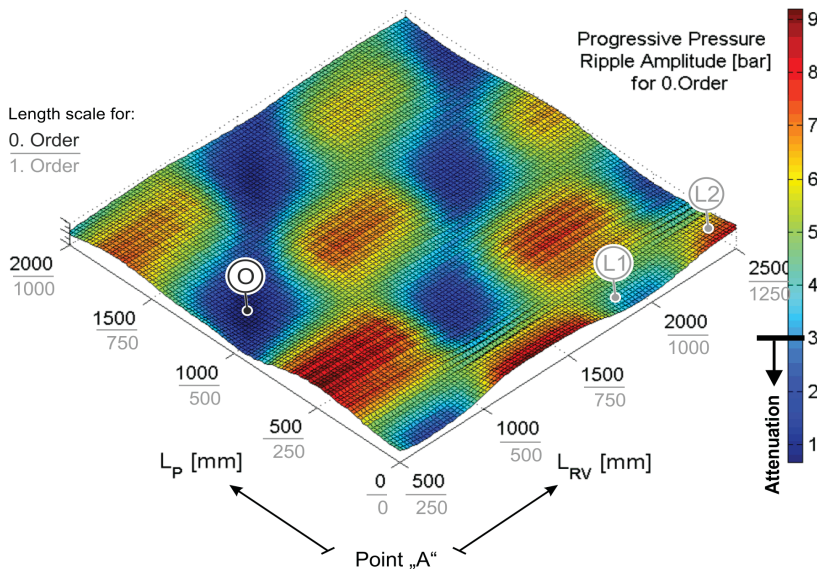


Figure 4: Simulation results for 0/(1). order ripple frequency in the discharge test line

Starting from a reference amplitude of approx. 3 bar (pump acting alone), attenuation is achieved in the blue marked areas as function of the connection line lengths. Of course, suitable lengths depend on the desired tuning frequency (see qualitatively the impact of the 1. order instead of the 0. order, Fig 2). Note the studied aviation pump is a constant speed pump, which helps to find suitable connection pipe lengths via simple optimization algorithms (e.g. MATLAB pattern search). Fluid parameters have a minor impact on the lengths, because the mean pressure is constant & temperature variation is small. Here, the fluid Skydrol 500B4 is applied with $c(200bar, 38^{\circ}C) \approx 1100m/s$.

5. Experimental Setup

The utilized noise test rig at TUHH follows dedicated ISO- standards, ISO 15086-1/2/3 & ISO 10767-1/2/3. Procedures in /7/, /8/ & /9/ are also included. The key equipment features (1.) test pipes with multiple dynamic pressure transducers for progressive/regressive wave detection & (2.) test ripple sources, here rotary valves (RV), **Figure 5**.

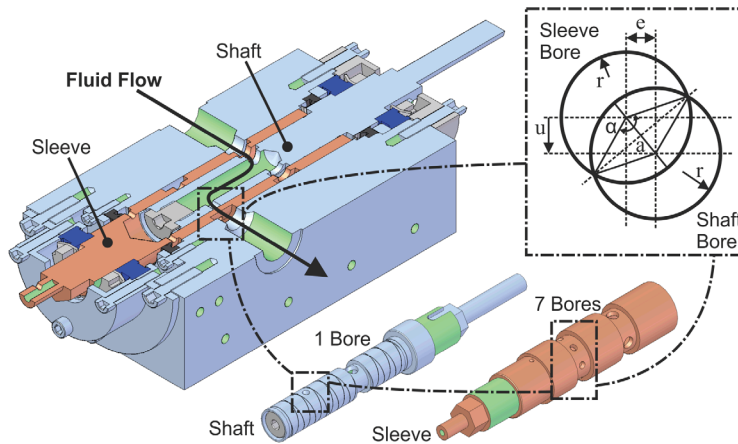


Figure 5: In-house rotary valve design at TUHH, more details available in /6/

The RV shaft speed controls the RV ripple fundamental frequency. The sleeve angle adjusts the ripple phase. The ripple shape is adjusted by the sleeve axial position (e). The amplitude is set by a needle valve that sets the lower pressure level for the RV. In the presented study, the RV is part of the component under test (EMP+RV), **Figure 6** and **Figure 7**. The needle valve is located in Load Block 2. As mentioned before, simulation and test results are limited here to the Direct Drain Principle, Figure 2, due to limited paper space. Besides, this principle is the most simple & compact technical realization, and hence assumed to be more attractive for an aviation application.

6. Study Cases

6.1. Flow Ripple in Discharge Line

In Figure 4, marker O shows an optimal line combination. The test rig geometry did not allow this. If the pump line is minimal, $L_p \approx 100\text{mm}$, then $L_{RV} = L_1 \approx 900\text{mm}$ is the shortest RV length, marker L1. It provides fair attenuation for 0. & 1. order, Fig. 4, while $L_{RV} = L_2 \approx 1200\text{mm}$ increases ripple. These expectations match test results for L1 & L2, **Figure 8**. Here, flow ripple are shown and the influence of RV angles is given. Noticeable attenuation is achieved even with none-optimal connection line lengths. The pump alone would have amplitudes of 1.5 L/min for 0. Order and 1.0 L/min for 1. Order.

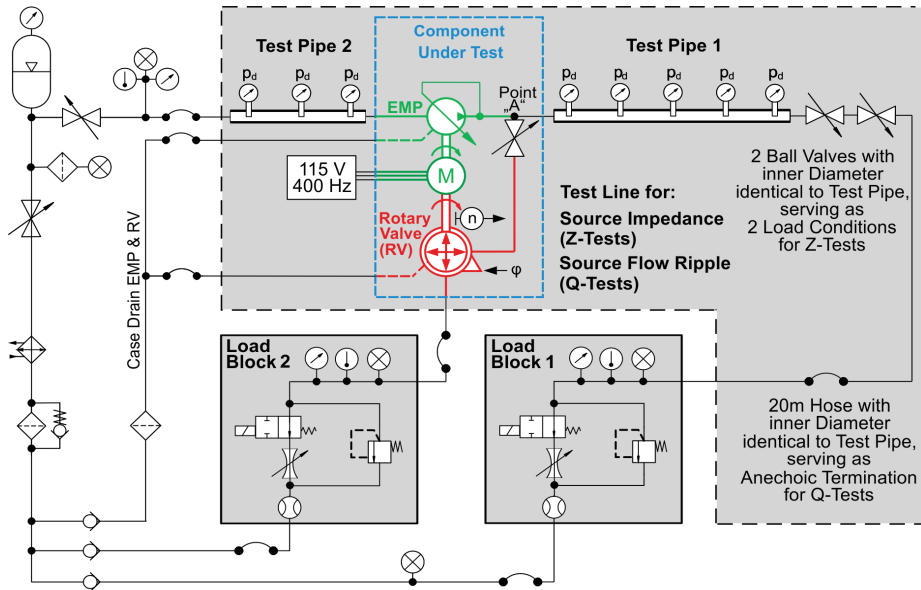


Figure 6: Hydraulic Diagram of the experimental setup for Direct Drain Principle

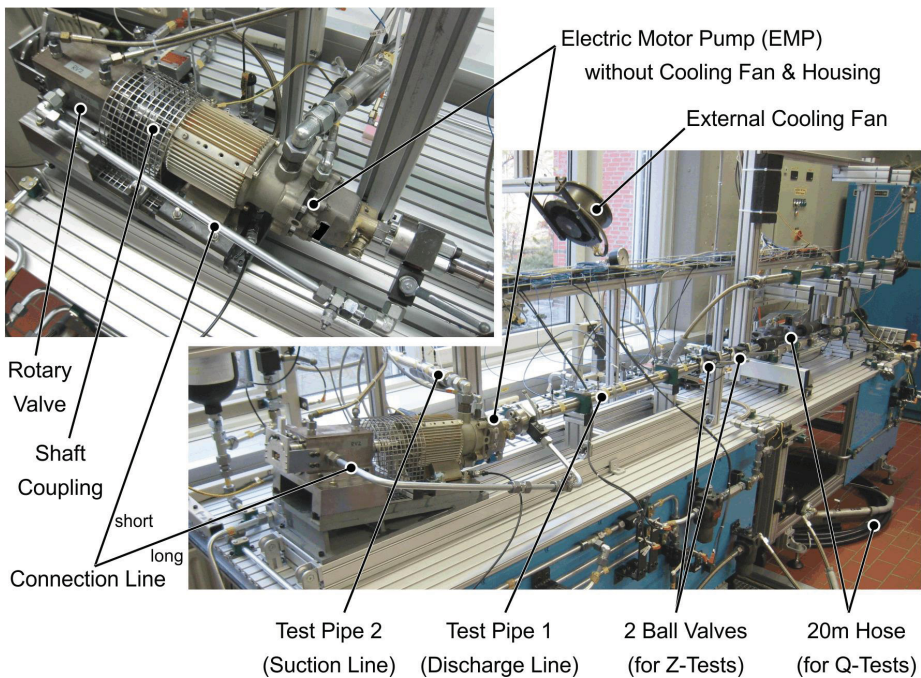


Figure 7: Picture of the experimental setup; pump test circuit of noise test rig at TUHH

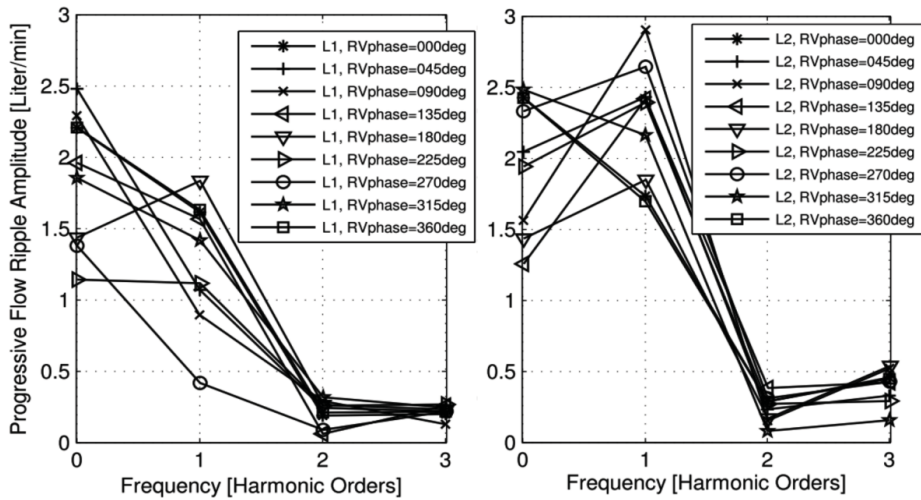


Figure 8: Experimental source flow ripple results for different RV connection line lengths (L1/ L2) and RV phase angles – Results for pump discharge line

Under the assumption of zero phase difference between pump and RV, the influence of phase changes due to the line lengths is shown in **Figure 9** for 1. Order acc. to eq. (2). It can be seen that L1 would be an almost optimal RV line length, if $L_p = 0$. Because this is not the case, the optimal lengths for RV are O1/2/3 to cancel the pump ripple.

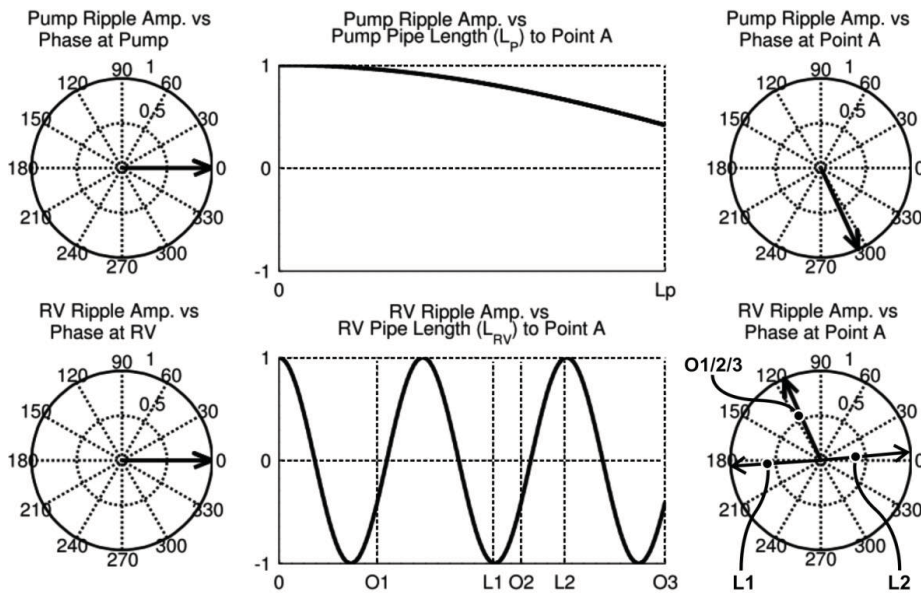


Figure 9: Analytical discussion of the source flow ripple results for a given/fixed pump pipe length (LP) and a variable rotary valve connection pipe length (LRV)

6.2. Flow Ripple in Suction Line

From typical passive silencers in the pump discharge line it is known that their blocked/ reflected ripples travel partially back to the pump and into the pumps suction line. Hence, passive silencers increase the pumps vibration load and cavitation risk at its suction port. Both reduce life time & reliability. So, an investigation of suction line ripple is introduced as indicator for pump (+ RV) suffering from its own fluid borne noise.

Corresponding to Figure 8, suction line test results are given in **Figure 10**. Similar to passive silencers, the main impact is at lowest frequency order(s), for both, connection line lengths & RV phase. The RV phase has the greater impact but with opposing trend to the results in Figure 8. So “L1, RVphase=270deg” tends to be the best in the discharge line, but the worst in the suction line. In case of L2 it is quite similar. This has multiple reasons related to the pump impedance and the lower pressure level of 3 bars in the suction line (different fluid properties). This should always be taken into account.

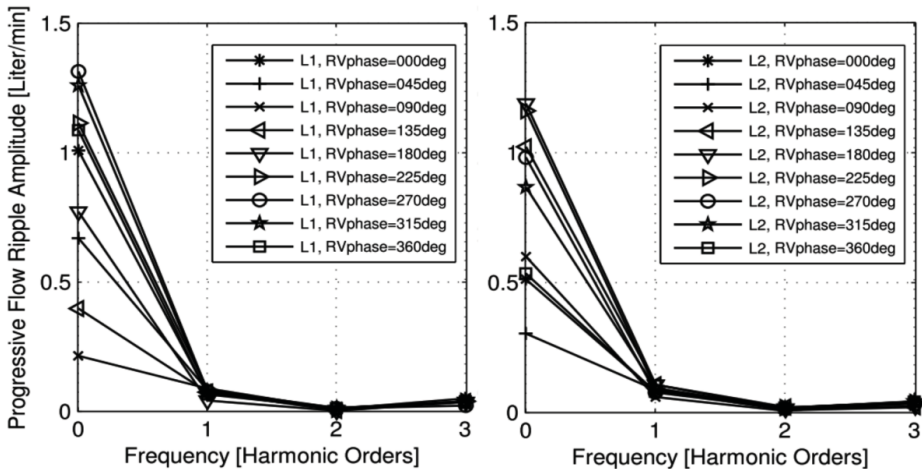


Figure 10: Experimental source flow ripple results for different RV connection line lengths (L1/ L2) and RV phase angles – Results for pump suction line

6.3. Overall Power Balance

For the Direct Drain Principle the RV ripple power is taken from the fluid. A separate power supply is spared, but means a small decrease of hydraulic power regarding pump mean pressure/flow characteristic. The mechanical power to drive the RV shaft & the change of electrical input power are negligible. In cruise condition – mean pressure ~200 bar & consumer flow ~5 L/min – the RV drain flow is ~3 L/min to achieve required amplitudes. This is ~10% of the pump nominal flow, ~30 L/min, and is tolerable up to a consumer flow of 25 L/min. Beyond, the mean pressure will drop below required limits.

7. Discussion

The connection line lengths of pump and RV to an intersection point A were highlighted as key (pre-) design parameters for the proposed active noise attenuation. The line lengths have strong impact on the noise phase at point A & hence on the attenuation performance. Although other parameters impact the phase, e.g. shaft speed & fluid properties, their impact is minor: A constant pressure hydraulic system – with means of temperature control – keeps the fluid properties in small limits. Besides, a constant speed electric motor pump was studied. If a variable speed pump/motor is intended, e.g. an engine driven pump, length-tuning to a major operating point, is recommended. Safety valves (SW) can reduce the RV impact outside this operating point, so passive side branch attenuation would remain of the connection line, Fig. 2, L1. Phase issues due to connection line lengths will vanish if active attenuation is integrated in future smart pumps. The concept here can serve as add-on for existing pumps, **Figure 11**.

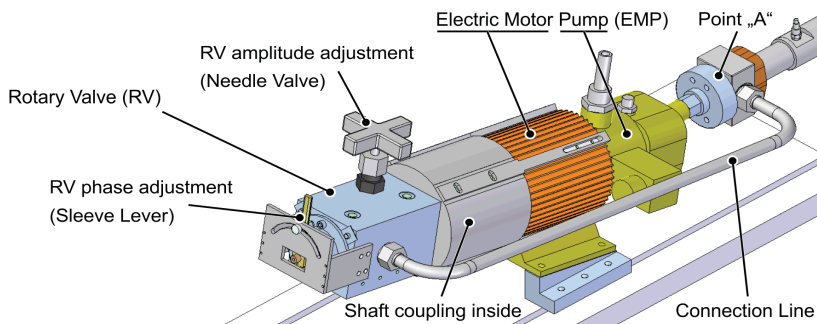


Figure 11: CAD assembly of a more compact laboratory prototype, moving the proposed attenuation concept towards an add-on feature for existing pumps

8. Conclusion

Active attenuation is known for high fluid borne noise reduction performance. It is considered for the fundamental & first harmonic frequency of pumps. These carry most of the noise power, and passive silencers require large capacitance/ inductance in the lower frequency range, increasing silencer size/ weight. This is an impact on aircraft production/ operation costs. Constraints related to flight performance & safety demand compact, robust & low-cost equipment. Hence, the paper focused challenges to get active attenuation drastically simple, but competitive in performance to passive silencers. Realistic concepts were discussed by simulation & test. The feasibility was shown. Control equipment is minimized by knowledge-based pre-design. It is referred to e.g. /10/ if RT-control (with FIR-filters, LMS-analysis, etc.) is required & e.g. to /11/ if time domain FBN modelling of pumps is desired including the pump impedance.

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