15th IFK
Proc. of the 15th International Fluid Power Conference <Symposium/Conference>
<DOI>

Next Generation Fixed Displacement Pumps for Electrified Mobile Applications

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Abstract.

Electrification of mobile machinery imposes new demands on hydraulic components, notably higher operating speeds, greater power density, and stricter noise requirements. This paper presents developments in the next generation of Parker fixed displacement pumps – gear, bent axis, and vane – designed specifically for electrified applications. We introduce a pragmatic low complexity gear pump approach retaining straight teeth while leveraging advanced manufacturing methods for higher shaft speeds at acceptable costs. For bent axis pumps, we demonstrate an integrated ripple chamber that reduces flow ripple yielding multi-decibel reductions in system noise while valve plate and suction channel refinements enable significant higher self-priming speeds, up to 17% depending on the displacement. The next generation vane pump incorporates tribological coatings for higher pressures and exceptional self-priming speed capabilities. Virtual and physical validation qualify the proposed concepts. Finally, the paper outlines a novel approach to integrate these pumps into ePump architectures while keeping modularity.

Keywords. Hydraulic pumps, electrification, fluid power electrification, fixed displacement pumps, noise, efficiency, integration, semi-integration, ePumps.

1. Introduction

Electrification is reshaping mobile fluid-power systems by removing the masking effect of Internal Combustion Engines (ICE) on one hand and by driving demand for higher power density due to batteries reducing available space and target weight for the hydraulics. As a result, hydraulic pumps – the primary flow sources in these systems – must deliver higher shaft speeds while meeting increasingly stringent Noise, Vibration, Harshness (NVH) requirements, higher efficiency while becoming smaller and lighter. Traditional pump designs, optimized for ICE-driven platforms, are therefore being re-evaluated for the new constraints imposed by electrified architectures. Although variable piston pumps dominate the hydraulic pump markets, fixed displacement units remain foundational in many mobile applications. However, the shift to electric prime movers alters the trade space for pump selection and optimization. Recent literature reports a broad spectrum of hydraulic pump concepts under investigation for designing electrified flow sources (ePumps), including vane, internal and external gear, bent axis piston, axial piston and radial piston designs to address the abovementioned requirements. Concurrently, to achieve the required power density of electrified mobile applications, system level integration considerations become more important as well, e.g. Integrated ePumps (IEPs).

This paper focuses on next generation fixed displacement pumps tailored for electrified mobile machinery, more specifically for open circuit work functions. Section 2 reviews the state of the art, summarizing the electrification-driven requirements and surveying relevant pump technologies. Different ePump integration levels are explained. Section 3 presents design concepts and validation results for the proposed next generation gear, bent axis, and vane pumps. Section 4 outlines a novel approach of integration for ePumps by combining modularity of non-integrated and the advantages of highly integrated ePumps. Section 5 summarizes the present contribution to provide an outlook on further research needs.

2. State of the Art

2.1. Mobile application perspective: Electrification leaves less space for hydraulics

Fischer et al. [1] provide an extensive overview on the state of the art, challenges and chances of electrification for hydraulic pumps and their operating limits. They summarize key literature dealing with different layouts of electrified mobile applications (centralized vs. de-centralized hydraulics as dealt in Casoli et al. [2]) and their maturity in terms of whether it is a phase/generation 1 electrification (pure conversion of existing ICE application as explained by Myers [3]) or a fully de-centralized layout in phase/generation 3 systems with one dedicated Electrohydraulic Pump (EHP) or ePump per function (as outlined by Nagel & Zhang [4]). Overall, it appears that the more electrified and de-centralized the hydraulics in the application are, the lesser space is available for the hydraulics due to increased electrical content.

2.2. System perspective: EHPs with higher power density

Both variable and fixed pump technologies have their use-cases in today's applications. This applies also to tomorrow's electrified mobile applications as well. While the Electric Motor (EM) has the flexibility of variable speeds, a Hydraulic Machine (HM) – this more generic term will be used in the following to consider multi-quadrant operation and not only pumping operation – using fixed displacement would require many and high accelerations and decelerations for the pump to implement an electric Load-Sensing (eLS) functionality. Using variable displacement technology adds mechanical complexity, though it provides an additional degree of freedom, as the flow can not only be adjusted by varying the EM's speed, but also through adjusting the displacement within the pump, e.g. through Electronic Displacement Control (EDC). Those fewer accelerations and deceleration of the pump's, its drivetrain and the EM's inertia enable efficiency optimized operating strategies as outlined by Myers [3]. Since the focus of this paper is fixed displacement technology, variable displacement pumps will not be further detailed. Ultimately, the choice for the technology depends on many factors to be weighted, the work function and the desired duty cycle to be performed.

In both cases however and as described by Fischer et al. [1] electrification demands more integration to compensate for decreasing available space in the mobile applications as well as tougher requirements to the target weight of hydraulics, as batteries occupy the bulk of the vehicle's size and increase vehicle weight accordingly. One first step for moderate integration is the direct coupling of EM to the HM as proposed by Cailteux [5] and Pelz et al. [6]. Having this direct connection instead of a coupling allowing compensation of misalignments and/or damping oscillations in circumferential direction requires however specific lubrication solutions for the splines to avoid fretting corrosion and ensure appropriate durability over lifetime. [1]

In the context of Parker's SMART Electrification portfolio (refer to [7]), pumps directly connected to Parker electric motors are referred to as Configured ePumps. Despite having the advantage of direct coupling with appropriate spline lubrication, Configured ePumps are highly modular, as they feature a broad spectrum of different pump technologies, displacements and power ranges as well as motor types, featuring not only Parker's GVM but also its NX motor both as low (LV) and high voltage (HV) versions. They can further be combined with Parker's Global Vehicle Inverter (GVI, refer to Parker Hannifin [8]), depending on the voltage level. Another direction is a more consequent type of integration, which can be seen on Figure 2.1, called Integrated ePump (IEP).

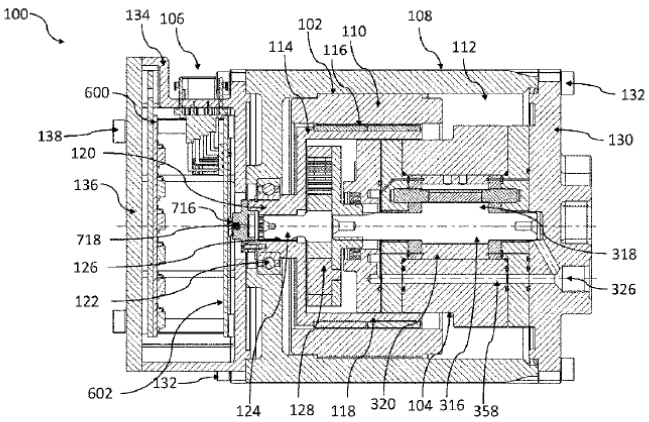


Figure 2.1: Integrated ePump (IEP) with Planetary Gearset (PGS) acc. to Zhang et al. [9]

There are multiple approaches on how to integrate HM with the EM, as a recent publication by Heeger, West & Ericson [10] outlines. The authors describe systematically whether HM and EM are axially or radially stacked and then whether the HM is direct-driven or gearbox-coupled. Combinations are also presented, e.g. radially integrated gearbox and HM. In related work, Lindh et al. [11] demonstrated a gearbox-coupled configuration connecting a bent axis pump with a permanent-magnet motor and examined the feasibility and performance of hydraulic-oil based cooling strategies for the motor. Ponomarev et. al [12] integrated a permanent magnet synchronous machine with a bent axis piston pump. The thesis also explores use of hydraulic fluid to cool the electric motor. Pietryzk et al. [13] investigated the potential of implementing high speeds at the EM with a Planetary Gearset (PGS) between EM and HM for a high-power density Electrohydraulic Actuator (EHA). Gadiyar et. al [14] designed a single casing integrated axial flux electric motor with a radial piston pump unit, allowing a reduction in the overall package volume and removing shafts, bearings, and bearing seals. Along the same lines, Sarode [15] have designed and experimentally validated an integrated fixed displacement axial piston machine with a permanent magnet electric motor. These single-casing motor–pump integrations enable highly compact packaging well suited to electrified mobile machinery. In the concept of Figure 2.1, the HM and the gearbox no. 128 are radially integrated into the EM's rotor no. 114. With that, a concept like the gearbox type of integrated ePumps from Heeger, West & Ericson [10] arises and has in addition a motor controller no. 106, which is also integrated axially with joint cooling system. This approach of combining hydraulic pump, electric motor, gearbox, and inverter represents one of the most highly integrated ePump architectures considered in this work, delivering a compact package and an in-situ thermal management solution.

2.3. Component perspective: Higher speed and low noise pumps

While the previously mentioned integration reduces size of the ePump thus weight, it appears that if each of those sub-components is as well optimized itself towards higher power density, a further optimized application size and weight is enabled. Hence, it becomes crucial to increase Self-Priming Speed (SPS) of HM to reach higher speeds for higher flows, compensating for a smaller and lighter pump with lower displacement. The displacement is a key factor defining the torque needed to drive a HM in pump mode at a certain pressure, as depicted in the following basic formula (refer to Ivantysyn & Ivantysynova [16]):

$$T_P = \frac{\Delta p \cdot V_{D,P}}{2 \cdot \pi \cdot \eta_{hm}} \quad (2.1)$$

At a given pressure differential Δp, reducing the displacement $V_{D,P}$ of a pump compared to a reference with lower SPS, it appears, that the needed torque T_P is significantly lower, assuming the hydromechanical efficiency η_{hm} staying constant. This drives secondary downsizing potential for the EM connected to the HM, since – as outlined by Schweigert et al. [17] – the required torque has the main influence on the size and hence the costs of an EM, which is – amongst others – one of the drivers of high-speed concepts for EHPs.

A wide variety of hydraulic pump architectures are employed in ePump designs: Sarode et al. [18] systematically compared several ePump configurations that combine commercially available hydraulic pumps to meet the demands of a representative mobile-hydraulic application. The study evaluated drive-cycle efficiency across four utility drive cycles and benchmarked mass, volume, cost, and power density for each architecture. By providing this generalized, utilization-based comparison, the work offers a practical guideline for selecting appropriate ePump architectures in electrified mobile hydraulic systems. These conclusions inform trade-off decisions when matching pump architecture to application requirements e.g., favoring bent axis designs where power density and efficiency are paramount, gear solutions where cost and simplicity dominate and vane where noise needs to be reduced.

Indeed, while external gear pump technology is cost-efficient and robust, it is usually rather loud at higher speeds and pressures, when having straight teeth. Using helical teeth for a more continuous meshing of the gears brings advantages in terms of NVH behavior. Transitioning to alternative profile geometries (e.g. trochoidal) instead of conventional involute profile allows to reduce dead-volume hence reduces flow ripple as outlined for example by Morselli [19] or Zhao & Vacca [20]. The latter ones discuss a Continuous-

Contact Helical Gear Pump (CCHGP) design combining both helical teeth and alternative profile geometries yielding improved NVH behavior. A more complex overall pump design compared to straight teeth gear pumps is usually resulting, as then either complexity increases in the design by adding balancing features or in the manufacturing by adding a double-helical gear layout (e.g. Morselli [21]) similarly to herringbone gears. Despite external gear technology, internal gear technology can also be used in medium pressure areas, typically being less noisy than external straight gear technology.

Vane pump technology operating in medium pressure often features a symmetric layout in circumferential direction, hence it is radially balanced, reducing nominal loads on the shaft compared to other technologies with a similar displacement. In addition to that, vane technology is rather quiet and offers great NVH behavior, due to a different operating principle bringing however disadvantages in terms of robustness and overall efficiency. [1]

Ultimately, bent axis pump technology typically offers best-in-class efficiency, however at the price of higher noise emissions inherent to its operating principle. Further details on advantages and disadvantages of each pump concept and type can be looked upon at Vacca & Franzoni [22], Sarode et al. [18] or Fischer et al. [1].

3. Next Generation Fixed Displacement Pumps for EHPs

3.1. Next Generation Gear Pumps for Electrified Applications

3.1.1. Concept and/or design

As mentioned in section 2.3, novel gear pump designs using helical teeth to enable lower noise may become more complex as they need special features to ensure equalization of the axial forces generated by the helical angle. The proposed design here features minor changes on today's gear pumps having straight gears, i.e. major process and quality improvements of the critical tribological interfaces, that cannot be detailed further due to confidentiality reasons. Those modifications enable achieving best-in-class efficiencies for straight gear pumps as well as improved durability at higher speeds, hence higher SPS.

3.1.2. Virtual and/or physical validation and limitations of this study

An exemplary SPS assessment performed for the 14cc sample can be seen in Figure 3.1 (left). The SPS gain potential for different displacements is outlined in Figure 3.1 (right).

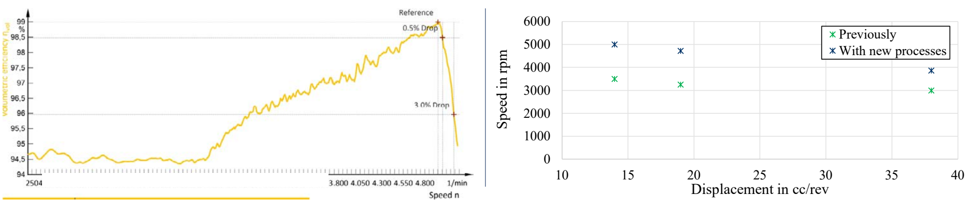


Figure 3.1: Exemplary SPS measurement at 200 bar and 0.8 bara inlet pressure (left) and SPS comparison before and after using the new manufacturing processes (right)

However, it must be acknowledged that those SPS gains are currently purely based on the formal definition of SPS. Additionally, the product maturity is at an A-Sample stage, hence a full endurance validation is to be performed. This will give more insights into the practically reachable SPS, whether this value can be driven continuously, intermittently or is just to be considered as a peak value.

3.1.3. Exemplary use-case related to mobile electrification

Since this new process is in A-Sample stage, there are currently no real use-cases demonstrating its advantages in mobile applications being in the field. However, any cost-sensitive electrified application where the noise performance with straight gears is acceptable, but an increased SPS is required while benefitting from the robustness and simplicity of a gear pump could be applicable.

3.2. Next Generation Bent Axis Pumps for Electrified Applications

3.2.1. Concept and/or design

The next generation bent axis pumps features an integrated Ripple Chamber (RC), which is novel for this type of pump. It has been applied to both the F1 truck pump, making it the F1e and the heavy-duty F12 pump, making it the F12e (refer to Parker Hannifin [23]). While the RC is integrated in the end cap of the F1e, the RC on the F12e is integrated into the barrel housing. The RC lowers significantly the flow ripple, hence also the dB(A) noise emissions in the lab but yields even more noise reduction potential in real applications. Additional design improvements on valve plate and suction channels increase the SPS by up to ≈ 17%, depending on the developed displacement of the F12e.

3.2.2. Virtual and/or physical validation and limitations of this study

The optimizations have been firstly virtually conceptualized and validated by means of Computational Fluid Dynamics (CFD) and then physically validated through extensive, physical testing, covering all typical qualification demands. It must be highlighted, that as opposed to the gear pump improvements described in section 3.1, for those new bent axis pump series F1e and F12e, full qualification took place. This includes a very demanding Steady-State Tests (SST), where a factor of the maximum pressure is applied – increasing the pressure beyond maximum pressure – and the pump is operated at the same time at the increased SPS for the whole testing duration covering several weeks. The SPS values of the F12 vs. the F12e having passed this demanding duty-cycle are on Figure 3.2 (right).

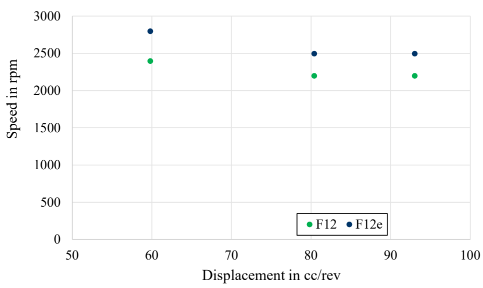


Figure 3.2: The new F1e and F12e series (left) and exemplary validated SPS values of the F12e vs. F12 pump versions at 1 bara inlet pressure (right)

There are no known limitations to this study at this stage, as the product has passed in C-/D-Sample maturity both all internal lab validations as well as the field testing successfully.

3.2.3. Exemplary use-case related to mobile electrification

The F1e has been tested on a truck application (tipper) to be compared to the standard F1. To avoid disruption of running operation where the measurement took place, it was not possible to compare on the same layout F1 vs. F1e during tipping operation. This can be seen in Figure 3.3 (left) where using the F1e, it appears that parasitic noises dominate further away from the pump compared to the same analysis with the F1. Hence, this made the F1e measurement look worse, than it actually was, though still achieving an improvement compared to the F1. The F12e however could be measured in exactly the same refuse truck side loader application to enable a side-by-side comparison. Representative measurement results can be seen on Figure 3.3 (right), showing excellent noise emission reductions.

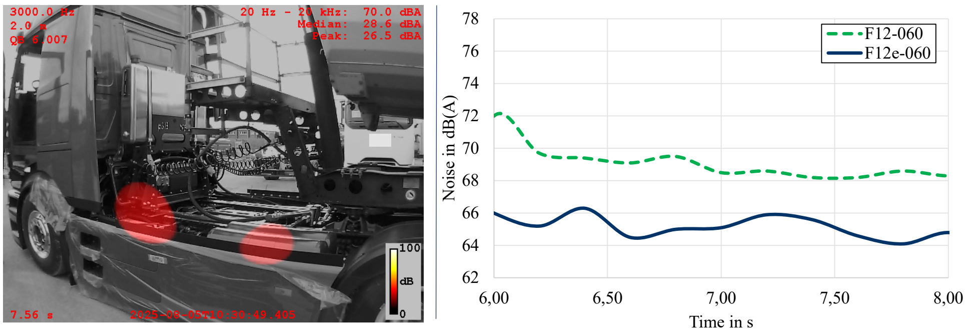


Figure 3.3: Exemplary F1e (left) measurements with acoustic camera during tipping operation and F12 vs. F12e noise measurement (right) results with acoustic camera performed on a refuse truck (ejection beam in operation at driver side)

In both cases, the external truck operators mentioned that they perceived significant noise improvements when switching from the F1 to the F1e and from the F12 to the F12e, which shows a real gain from psychoacoustic perspective in addition to the measurements results.

3.3. Next Generation Vane Pumps for Electrified Applications

3.3.1. Concept and/or design

Vane pumps are typically medium pressure pumps, but they still can reach higher pressures than today. Parker's new T8mini (refer to Parker Hannifin [24]) features 280 bar continuous pressure, while allowing 300 bar intermittent. This is made possible – amongst other changes – through novel coatings on the critical tribological interfaces between pressure and rear plates that interface with the rotor side faces. Additionally, a novel design of the port plates and housing in conjunction with additional radial holes in the cam profile enable excellent filling behaviour, yielding exceptional SPS performance. Together with a simpler mechanical design – for instance 10 instead of 12 vanes – the hydromechanical efficiency of this pump as well achieves highest levels. This enables a new breakthrough of overall efficiency for vane technology compared to the already rather high efficiency of the T7 series. The T8mini reaches on all his cam profiles enabling $V_{D,P} = 6 \dots 12\ cc/rev$ an SPS of 4,800 rpm at 0.8 bara. [24]

3.3.2. Virtual and/or physical validation and limitations of this study

The T8mini has been conceptualized by means of virtual validation (e.g. 1D-Simulation) and detailed using modern CFD simulations, e.g. to improve filling behaviour as described in Jeganathan et al. [25] and high volumetric efficiencies. The standard validation program in lab environment has been performed, e.g. from performance measurements (efficiency), noise emissions in reverberant rooms as well as high-pressure and endurance tests. Additional tests have been performed to meet special requirements demanded in new applications, e.g. BEV trucks both in on- and off-highway applications. Here, tests with high-viscosity oils (e.g. ISO 680 oils) were performed to reach kinematic viscosities of $\nu \approx 5{,}000\ cSt$ to simulate the effect of cold temperatures < 0° C. It must be mentioned that such tests do not replace specific in-vehicle tests at such low temperatures, nevertheless, they provide good-enough insights on cold-start capabilities of the pump. The test layout for this specific lab test can be seen on Figure 3.4 (left). Tests have been performed both in standard configuration (Figure 3.4, right) as well as submerged in the oil tank (not shown below).

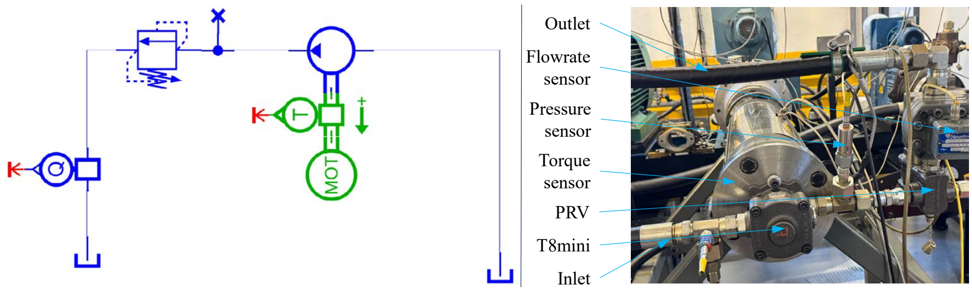


Figure 3.4: Hydraulic scheme & standard testing of T8mini at high viscosity

The submerged variant considered the intended layout in the final application by mimicking the application's oil tank volume and showed particularly good heat-up behaviour of the oil as well as an even better subjective noise assessment due to the surrounding oil, damping noise emissions. Ultimately, submerged pumps typically yield improved suction performance. As those investigations have been performed for a specific displacement of the T8mini and at a B-Sample maturity, further investigations are to be done for C-Samples at multiple displacements and also yielding higher production volumes.

3.3.3. Exemplary use-case related to mobile electrification

Since the T8mini has been optimized towards high-speeds and higher pressures, the detailed characterization at combinations of low speeds (≤ 1,000 rpm) at low pressures (≤ 50 bar) as well as high-speeds (> 4,000 rpm) at low pressures (≤ 50 bar) was not in the initial scope of the development. Nevertheless, special tests have been performed to check compliance with requirements demanded for an auxiliary low pressure 24 V ePump for On- and Off-Highway BEV Truck applications. Here, the T8mini's high-speed and cold-start capabilities as well as its resistance against high pressures spikes were key to achieve the high-power density, low noise, low envelope size and weight requirements as well as reliability for such a heavy-duty application. While the B-Sample featured a bell housing between the Brushless Direct Current (BLDC) motor with integrated motor controller rated for 1.85 kW (refer to Figure 3.5), the C-Samples have been designed featuring a direct connection. The potential of axial space saving resulting by that can be seen in Figure 3.5. Further reduction of envelope size however requires more extensive integration, that will be outlined in the next section.

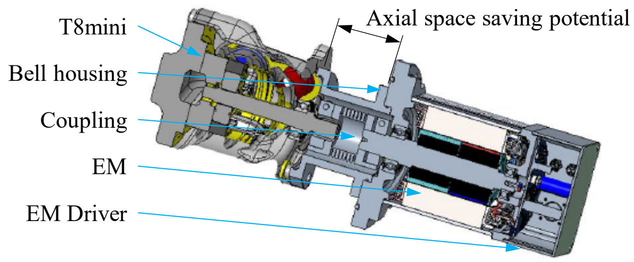


Figure 3.5: T8mini HM as 24 V auxiliary EHP for HV BEV Trucks, here in B-Sample maturity with moderate integration (still using a bell housing with coupling)

4. Discussion: A Novel Approach to EHP Integration

4.1. Limitations of increasing solely the pump's power density

The previous section demonstrated that with nowadays development tools like Computer Aided Engineering (CAE) – more specifically CFD – further advancements can be done on existing pump technologies, to increase their power density or to develop new pump generations. Increasing the SPS while improving the NVH behaviour at the same time enables reducing the pump size. In other words, assuming a certain flow Q_P is required from the application's perspective, if the pump's SPS n_P can be increased, its displacement $V_{D,P}$ can be subsequently reduced, as it can be seen in the following basic formula:

$$Q_P = n_P \cdot V_{D,P} \cdot \eta_{vol} \qquad (4.1)$$

If an intermittent flow of around 160 l/min is required, with the standard F12-060 for example, the requirement could not be reached without boosting the inlet, and it would be needed to take the next bigger size, i.e. the F12-080. With the F12e-060, this flow could be met through the increased SPS (refer to path a) in Figure 4.1) assuming the same volumetric efficiency η_{vol} for F12 and F12e at those different speeds: A significant reduction in displacement, hence pump size would be enabled. Another option would be, to maintain the F12-080 and operate it at lower speed to achieve similar flow Q_{des} at reduced noise (refer to path b) in Figure 4.1). However, having this example in mind, the HM downsizing remains marginal, compared to the size of the EM that typically represents the bulk of the EHP's size, weight and costs. Additionally, the speed limiting factor often being the HM's SPS, it remains difficult to operate the EM frequently in its efficiency optimum areas that typically are at higher speeds.

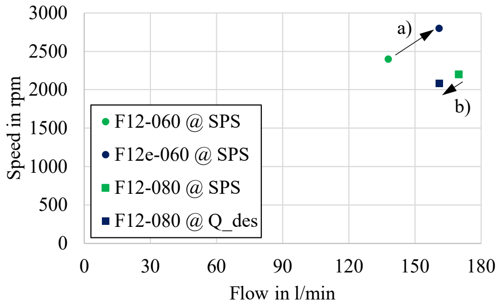


Figure 4.1: Exemplary calculation study to assess benefits and limitations of SPS increases

4.2. Semi-integration: High power density EHPs that can remain modular

One approach to tackle the beforementioned challenge to match HM and EM speeds, is to add a gearbox between EM and HM. The added complexity, losses and costs of the gearbox can only be justified, if the EM's downsizing is such, that from system perspective the overall efficiency remains constant or even improves through operating the EM in more efficient areas of its maps. Additionally, the system costs should ideally be lower than before. While Heeger, West & Ericson [10] showcase most possibilities to combine HM, EM and gearbox in Figure 4.2 (concepts (a) to (e), there may be another option (f) (Figure 4.2, below) that may appear counterintuitive at first sight.

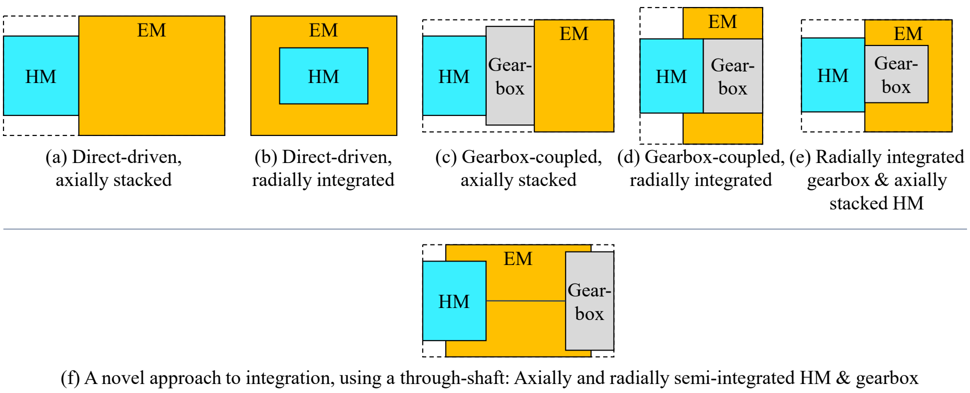


Figure 4.2: Concepts for combining Hydraulic Machines (HM) and Electric Motors (EM) into electro-hydraulic energy converters according to Heeger, West & Ericson [10] (left, with concepts (a) to (e)) and novel approach (f) for semi-integration (patent pending)

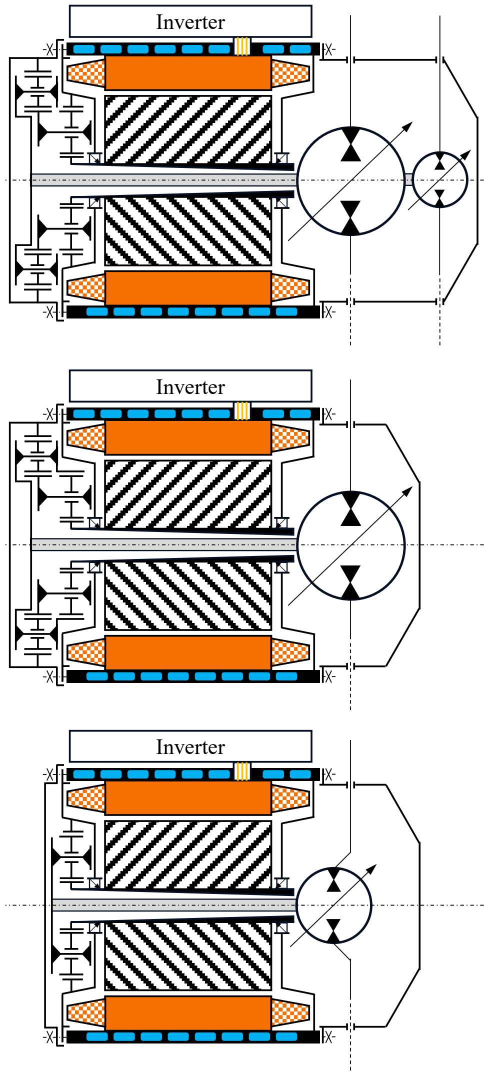
This option is to deliberately separate the gearbox and HM from the EM, so that both are arranged on opposed sides of the EM. This requires a through-shaft going through the EM which subsequently needs to have a hollow-shaft rotor and resolver design to have the gearbox connected to the HM. What appears to be more complex and costly at first sight, brings a lot of advantages as such a layout still enables a certain degree of integration, while offering more flexibility for modular design of the overall system and its components: Hence it can be referred to as "semi-integrated" EHP concept. The advantages of such an unconventional approach to integration are the following:

- Easier scalability of the outer diameter of the EM, while keeping the possibility to scale the EM in axial length. This gives additional flexibility for the design of a modular system comprising certain EM, HM and gearbox variants.
- The EM's rotor diameter can be much smaller, as it is not necessary to integrate gearbox and HM inside of the rotor, hence the only limiting factor of the EM's inner diameter is the very moderate diameter of the through-shaft. This is essential for high-speed EM concepts that require small diameters.
- The radial dimensions of HM and gearbox are less limited in radial direction than in other integrated concepts, offering additional possibilities to improve the scalability of those components to increase modularity of the EHP system.
- An oil cooling of the EM's rotor can easily be implemented via the through-shaft and/or the hollow rotor shaft, if a common oil sump between gearbox and HM is possible.
- If a common oil sump between gearbox and HM is chosen, the through-shaft layout extends the size of the oil sump, enabling using the oil sump as integrated tank.
- The through-shaft further enables integration of features to compensate radial and axial misalignments between gearbox and HM.
- In case a Planetary Gearset (PGS) is used as a gearbox – which has the benefit of highest power density and radially equalized forces – the overall EHP is fully co-axially arranged (concentric). This simplifies design of housing parts and provides an axially extending layout, that is beneficial if the radial extension of the available space is limited. Also, such a co-axial design supports a common part strategy.
- Gearbox and HM can be assembled nearly at the same time in parallel on each side of the EM, which reduces assembly time.
- Gearbox and HM can be dis-assembled and/or serviced independently from each other, which also eases maintenance, if required.

Such a concept is shown schematically in Figure 4.3 (i) using a 2-stage PGS on the left side with high PGS ratio, that is partly fitted under the windings of the EM. The HM is to be thought of a cartridge type of pump, that can be either of fixed or variable displacement type and can have 1- to 4-quadrant-capability, or any in-between step, depending on the requirements. The suction side of one or both pumps can be fed through the oil sump of the system or be led to an external tank. The outlets can be separated or together, depending on the application.

Another variant with only one minimal change can be seen in Figure 4.3 (ii). Here the only difference is that the hollow gear of the PGS stage I is connected to the carrier of the PGS stage II. This enables a lower ratio with only a minor modification of the PGS, supporting a common parts strategy.

Using only stage I of this same PGS – consistently driving common parts strategy – enables a further reduced ratio, that may be sufficient for intermediate displacements, see Figure 4.3 (iii).



(i) Semi-Integrated ePump concept (patent pending) with large main pump, auxiliary pump and 2-stage PGS connecting the pumps to the EM via through-shaft

(ii) Semi-Integrated ePump concept (patent pending) with large main pump, 2-stage PGS with modified ratio through connecting stage I hollow gear with stage II carrier

(iii) Semi-Integrated ePump concept (patent pending) with intermediate sized main pump, 1-stage PGS

Figure 4.3: Semi-Integrated ePump concept and exemplary variants (patent pending[1])

Further variants can be thought of, e.g. increasing torque capacity of the PGS through increasing either the number of planets, their width, or both together. Changes in the teeth numbers with or without addendum modifications enable slightly modified ratios. The schematically indicated position of the inverter is deliberate, as being on top allows proximity to the EM, shortening connection distances on one hand. On the other hand, having it not axially integrated ensures that gearbox and HM can be radially scaled. Ultimately, the use of a gearbox does not only limit itself to be used to allow higher speeds hence higher efficiencies at the EM (EM “high-speeding”) but may also be used to lower or

[1] A patent application covering aspects of this work has been filed; details will be available upon publication of the application.

at least keep the pump speed constant for purpose of à posteriori noise reduction in mobile applications particularly sensitive to noise emissions (HM "low-speeding").

4.3. Potential limitations to tackle when dealing with semi-integration

While the abovementioned approach for semi-integration is rather new and shows considerable advantages on many aspects, it brings also new challenges with it that need to be further investigated. Certainly, achieving the right balance between integration and modularity represents one of the biggest challenges. Adding a gearbox will increase inertia, which may be a limiting factor for highly dynamic applications. Depending on the lubrication concept, using a mono-fluid for the gearbox and the hydraulics might require compromises on either one of those sub-components' performance. Choosing separate, dedicated fluids on the other hand would complexify the sealing and eventually the assembly concept. All those aspects need to be investigated both theoretically and experimentally before implementation of semi-integration.

5. Summary

The present contribution shows advancements in the development of the next generation fixed hydraulic pumps for electrified mobile applications, more specifically for electrified open circuit work functions. In a first step, the current state of the art summarizes chances and challenges of electrification for hydraulic pumps. A main takeaway is, that due to increased need of space for batteries on HEV and BEV, the hydraulics need to be more power dense than ever to compensate a bigger portion of the vehicle's weight being accounted by the battery. Additionally, noise emissions of hydraulic pumps need to be reduced, being not masked by ICE noise anymore. In a second step, a moderate approach outlines, how to achieve those development targets from pump perspective presenting three examples of Parker's eReady pumps from lowest to highest degree of modifications: A new combination of manufacturing processes enabling higher SPS on gear pumps; an integrated ripple chamber in the new bent axis pumps F1e and F12e to reduce flow ripple, hence noise and further optimizations to increase SPS to up to 17% and ultimately a completely new generation of vane pumps designed for electrification. System level considerations have then been performed to reflect on the shown pump's power density improvements and to enable even higher density of EHPs. A novel approach was presented – that tries to reconcile high integration and high modularity – described as "semi-integration". Further research is needed in terms of endurance validation of the outlined SPS improvements for gear pumps as well as on the design and implementation of the outlined modular EHP system leveraging semi-integration.

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