Responding to sealing challenges in Electric Vehicles (EV)

Dr. Bas van der Vorst, Mr. Jeroen Lenten, Dr. Mickael Sansalone

Recent developments in electric driveline technologies require that seal testing capabilities, seal virtual predictive platforms and, most importantly, sealing solutions be upgraded to cope with the challenging high-speed levels and specific lubrication conditions present in Electric Vehicles (EV). Since the EV global market is also very dynamic, related sealing product specifications and test requirements also constantly varying.

This paper discusses a strategy to develop sealing solutions able to properly perform in the severe and diverse conditions related to EV applications. It requires test equipment that not only fulfils the test requirements but also rapidly adapts to changes ensuring that products can be properly developed and validated. Here parameters such as seal reverse pumping ability, friction and temperature are discussed, which are key in responding to the challenges related to Electric Vehicle applications.

1 Introduction (Background)

Over the last two decades the pressure by societies to reduce CO2 emissions has been increased globally. This has resulted in global legal regulations with respect to CO2 emission and penalty fines when emissions are too high. Figure 1 shows the reduction over time of CO2 produced by automotive vehicles for several regions and countries. In addition, it shows the future emissions targets based on the regulations in place aiming to further reduce CO2 emission.

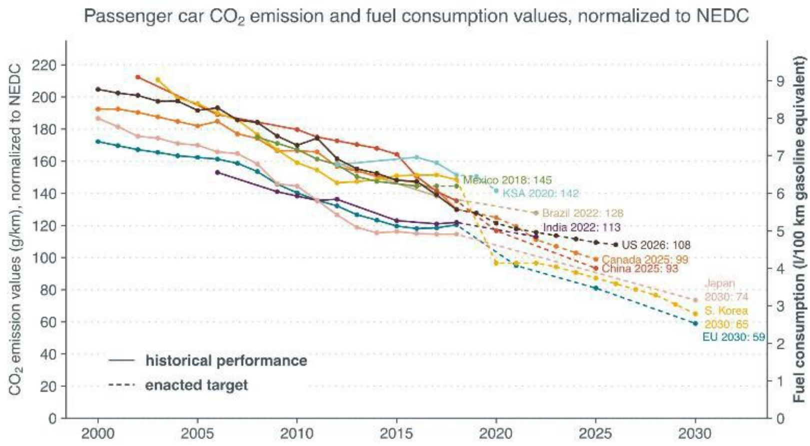

Figure 1: Reduction of CO2 emission over the last 20 years and the predicted reduction based on targets in current regulations (Global Comparison of Passenger Car and Light-commercial Vehicle Fuel Economy/GHG Emissions Standards., 2021).

Due to these strong regulations on CO_2 emission, the automotive industry is innovating at a very fast pace and will have to keep this pace in the near future. Here the driveline is one of the main parts where innovation takes place, i.e. here the basic idea is to use an electric motor to reduce the use of the internal combustion engine or even to completely replace it. This results in a highly dynamic automotive market where the drive-line technology inside is changing continuously. As a result, a wide variety of technologies coexist on the automotive market today. This ranges from hybrid systems that perform simple start-stop functionality for saving fuel when a vehicle is not moving all the way to full electric vehicles.

Figure 2 shows an overview of the most common electric drivelines currently on the market. Together with the internal combustion engine, the drive lines are generally divided in three classes as follows:

1. Combustion Vehicles
   IC : Internal Combustion

2. Hybrid Vehicles
   MHEV : Mild Hybrid Electric Vehicle
   FHEV : Full Hybrid Electric Vehicles
   PHEV : Plug-in Hybrid Electric Vehicle
   EREV : Extended Range Electric Vehicle

3. Full electric Vehicles
   BEV : Battery Electric Vehicle
   FCEV : Fuel Cell Electric Vehicle
   PFCEV : Plug-in Fuel Cell Electric Vehicle

It is forecasted that internal combustion vehicles (IC) will have a global presence well into the 2030's but, due to fast progress to full electric vehicles (BEV and FCEV) in some European markets and in China, electrified technologies are expected dominate globally by 2028.

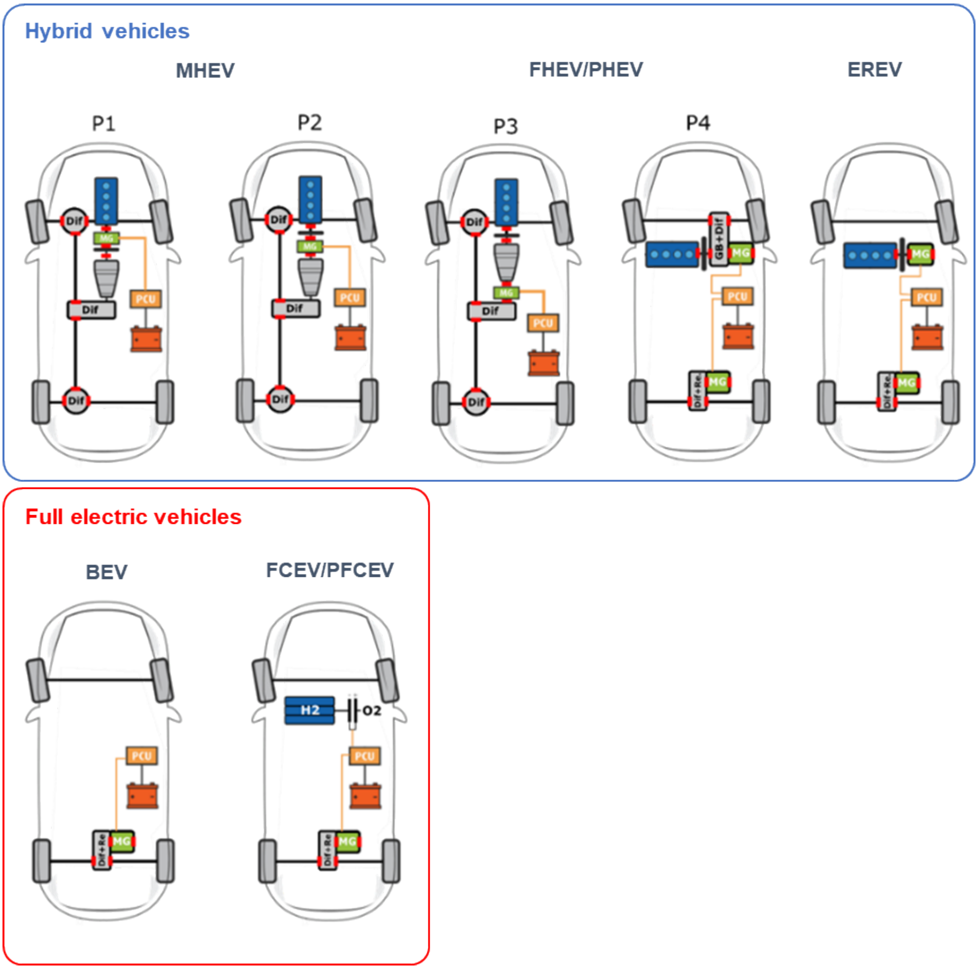

Figure 2: Schematic overview of most common drivelines for hybrid and full electric vehicles. Here one recognizes the combustion engine (bleu), the electric motor/generator (MG, green), the mechanical gearbox (grey), the power control unit (PCU, yellow), the electric battery (orange) and the differential (light grey, dif.) which is for some case integrated with a reduction gearbox (Diff+Re.). For the fuel cell electric vehicles (FCEV/PFCEV), H2 is the hydrogen tank and O2 is air intake supplying the oxygen (source https://ac.umicore.com/en/applications/hybrid-alternative-fuels/).

2 Impact on seal technology requirements

With regards to the impact on the sealing technology required to facilitate the electrification of the automotive drivelines, one would prefer to use a different driveline classification than shown in Figure 2 which consist of two classes.

The first class consists of the drivelines P1 up to P4 for which the internal combustion driveline and the electric motor are mechanically connected. In this case, the rotational speed of the electric motor is limited by the combustion powertrain. For this class, the sealing technology that already existed to design seals for combustion engines could, in general, be used to develop sealing solutions for the electrified drivelines.

The second class of drivelines does solely contain an electric driveline or, in case a combustion engine is used (P4 and EREV), it is only partially connected mechanically to the electric driveline. As a result, for this class of driveline designs the rotational speed of the electric motor is no longer limited by the maximum rotational speed of the combustion engine. This allowed designers to increase the power density of the e-powertrain by using smaller electric motors operating at higher speed in combination with a reduction gearbox to connect to the wheel shaft (see Figure 3). The most common design is the so-called 'dry motor' for which the main sealing challenge is to avoid gearbox lubricant to enter the dry environment of the electric motor, i.e. the cavity between stator and rotor.

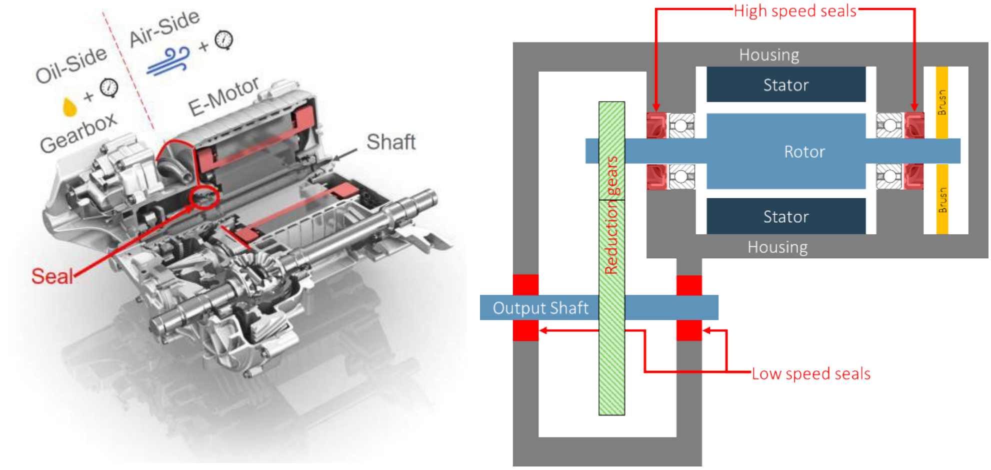

Figure 3: E-powertrain design typical for latest EV models (see figure 1), consisting of a high-speed electric motor combined with reduction gearbox (source https://www.zf.com/mobile/en/stories_30784.html).

This seal is first of all challenging due to the fact that these high-power density E-drivelines are designed to run at very high speed and temperature, i.e. sliding speeds in the range of 40 m/s and continuous operating temperature of 110°C as specified by OEM's. Consequently, this leads to a need for a temperature resistant material and for a low friction seal in order to avoid very high contact temperatures, accelerated wear and oil carbonization. In order to reduce seal friction, most seals designs aim to reduce the seal force, sometimes in combination with a low friction material such as PTFE that is highly temperature resistant as well.

Furthermore, since the temperature is expected to change significantly after a start or stop this can result in pressure differences. Since, in general, high-speed seals do have very low loads these pressures can have a significant effect on the contact force. However, whether this pressure does build up, strongly depends on the exact design of the application. This is reflected in the large spread of customer requirements with regards to pressure, i.e. ranging from 0 to 500 mbar (see Figure 4). In addition, also other requirements strongly depend on the particular E-driveline design. Here some applications require bidirectionality of the seal, for instance.

Finally, also the lubricant type and the lubrication method used for the gearbox are strongly varying between applications. In general, oil is used as a lubricant but, to have additional cooling capability, currently also solutions based on water(glycol)-based lubricants are under development. With respect to the lubrication method some applications use an oil mist while other use an oil bath at different oil levels. The dynamic misalignment of shaft is expected to be very low to avoid high vibration levels in the application. Therefore, this is expected to be less of a challenge for the seal.

The above clearly shows that the technical requirements relevant to the sealing technology needed for E-powertrains are varying significantly. Figure 4 gives an overview of the variations observed over the application field. This diversity of requirements is expected to remain for the forthcoming decade.

On top of the requirement variations discussed above, there might be a fundamental upcoming e-driveline design change promoted by major players in the EV field which are using so-called 'wet EV motor'. In this design, the lubricant is allowed to enter the space between rotor and stator, thereby cancelling the need for a (high-speed) seal between gearbox and the electric motor. In this case, there might still be a need for a high-speed seal at the brush side (see Figure 3) when the brush, that has to conduct electric current from the shaft to the frame, still has to operate in a dry environment.

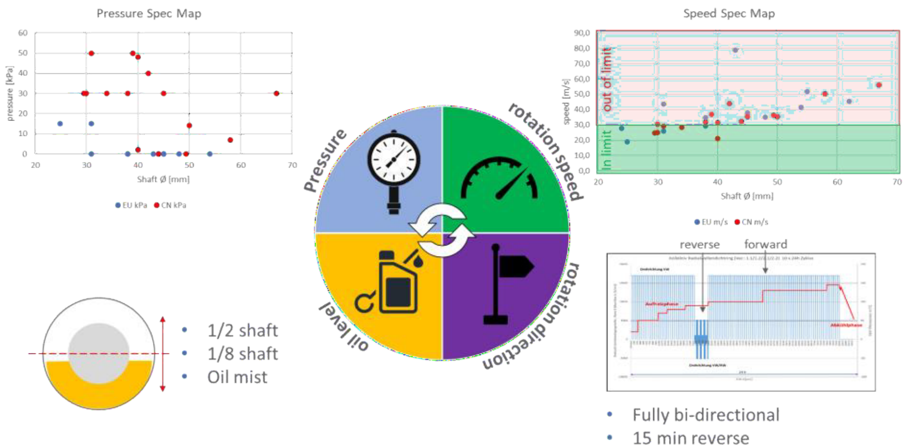

Figure 4: Variation of requirements in the E-powertrain market.

3 Responding to the EV market

Since the requirements for high-speed EV seals are varying within this market and are also not expected to converge on the short term, most probably no single sealing solution will be sufficient to cover all applications in the market. In other words, it can be expected that the sealing solutions vary from one EV design to the other. Responding to this variation, requires speed in design and design optimization. This can be achieved by a smart combination of model predictions, simulation, and testing.

As a starting point for testing, the application conditions have to be set. Here the test strategy is based on the most severe application conditions that are more or less common for all application. Firstly, based on customer requirements (Figure 4), tests are performed up to 50 m/s sliding speed at 110°C using an oil bath at ½ shaft level, since these conditions are expected to be most demanding for the seal with regards to retaining the oil. Compared to an oil bath, oil mist lubrication is expected to be less demanding with regards to oil retention, although it might be more severe with regards to starvation of the seal contact which probably results in high contact temperatures. Furthermore, the aim is to have a bidirectional seal which is more demanding for the design compared to a unidirectional seal.

Next step, depending on the specific application, is adding pressure (or vacuum) to the seal as it is expected to have a strong influence on the seal behavior. The latter is especially true considering the low load (low stiffness) seal designs required for these high-speed applications. Shaft misalignment is kept at a minimum for the test since it is expected to be very low as these have great impact on the vibration level of bearings and other mechanical components in the application.

These tests, combined with numerical prediction tools available, enable to analyze the sensitivity of the seal design with respect to lubrication, friction, heat conduction environment as well as application conditions such as pressure.

Figure 5: High-speed test rigs for seal performance testing. Testing capabilities: High speed testing up to 50.000 rpm, High temperature up to 180°C, Controlled pressure/vacuum up to 100 mbar +/- 0.2 mbar, oil bath or oil mist, pumping and leakage (visual, mass).

As a start, standard elastomer low force (plain) seals, designed for mid-range sliding speeds up to 20 m/s, were tested at 110°C in an ½ shaft oil bath to check when leakage starts and how it behaves at very high speeds, i.e. up to 50 m/s.

Figure 6 shows a typical result where the seal starts to leak the first few droplets at 30 m/s sliding speed and the leakage rate increases with speed up to 40 m/s after which the leakage rate decreases again. This might be an indication for the fact that less oil reaches the seal lip contact, for this particular seal design, due to oil spinning away from the high-speed shaft.

For these conditions, this shows that up to mid-range speeds the seal performs even a bit better than expected. Additional design changes are required to stop the leak rate for increasing speed. Here one can think of increasing the seal load, but this comes at the cost of increasing friction which is unwanted and furthermore results in higher contact temperature leading to oil carbonization and wear. Another route to decrease the leakage rate is to increase the reverse pumping ability of the seal. This can be done by adding pumping aids (helices) or using a wave type of seal (see figure) that induces a hydrodynamic pressure over the seal during rotation pushing the oil back into the application.

At very high speeds (>50 m/s), apart from the pumping required to stop the seal from leaking, here the seal might start to starve (run dry) when less oil reaches the contact which then leads to high contact temperatures. To counteract possible wear and carbonization due to starvation one might think of using low friction materials such as for instance PTFE. However, seals made from this type of materials many times show leakage problems at low speeds or during stand still when the reverse pumping is low.

Therefore, one optimization route for EV high-speed applications is to improve the standard seal design by including and optimizing the reverse pump rate of the seal which will be discussed in next section.

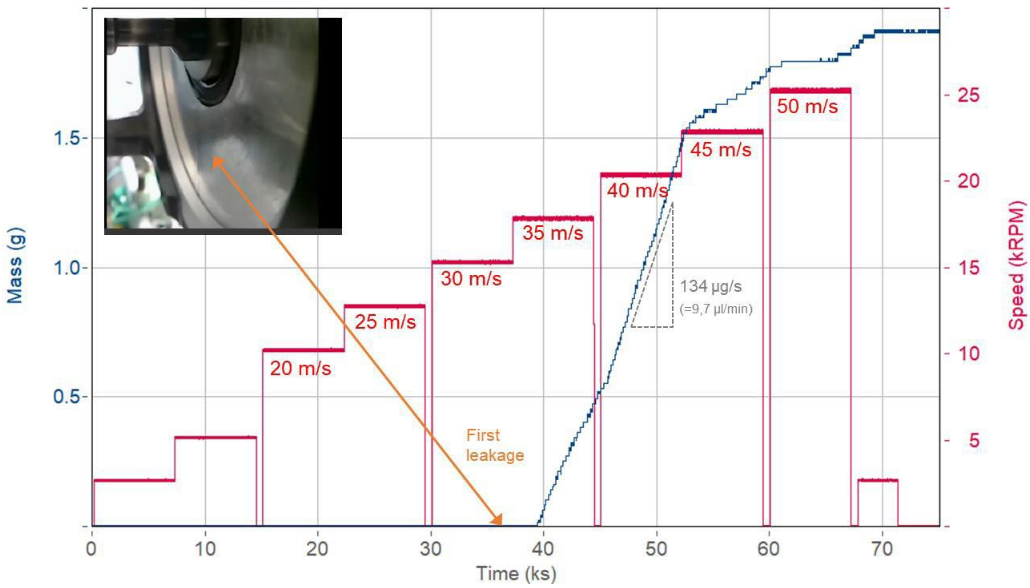

Figure 6: Performance test on a standard plain seal designed for sliding speeds up to 20 m/s. First oil leakage, starts at 30 m/s as observed with the camera. The total amount leaked during this speed is still too low before reaching the leakage sensor. Leakage rate further increases to 6,5 and to 9,7 µl/min for 35 and 40 m/s respectively. Finally, at very high speeds of 45 and 50 m/s the leakage rate drops to respectively 1,7 and 1,3 µl/min (note: 1 µg/s = 0,0723 µl/min for ρ = 0,83 g/ml).

4 Optimization with regards to reverse pumping

4.1 Seal design concepts with reverse pumping

In order to optimize the sealing capabilities of the standard seal shown in previous section for high-speed EV applications, the reverse pumping ability of the seal needs to be improved. For this, there are basically three design options available: Plain seals, wave seals and seals with hydrodynamic features (Figure 7). The basic difference between these design classes is the contact area between the seal lip and the shaft, i.e. the seal footprint, which results in different pumping mechanisms and typical pump rates ranges as shown in Figure 8 as a function of speed.

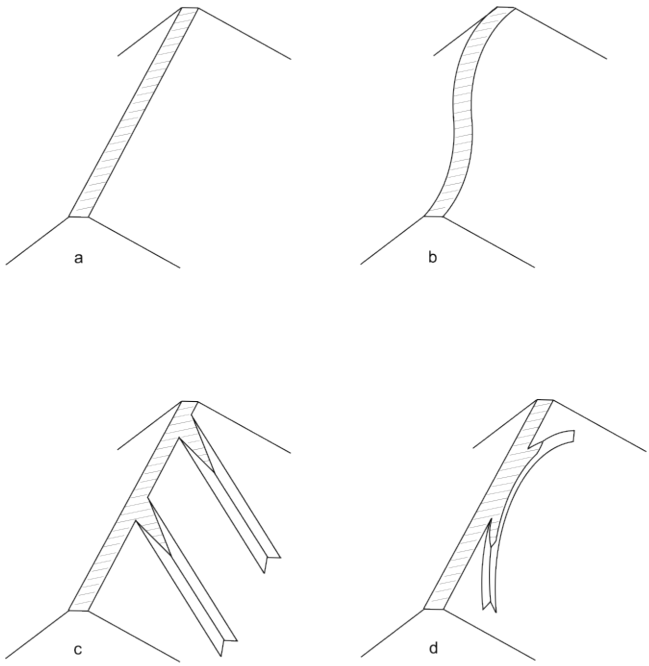

Figure 7: Different seal classes and their contact foot-prints. Here pane a shows the plain seal, i.e. the basic seal without any hydrodynamic pumping features; pane b the wave seal; pane c the helix rib seal and pane d the sinusoidal rib seal.

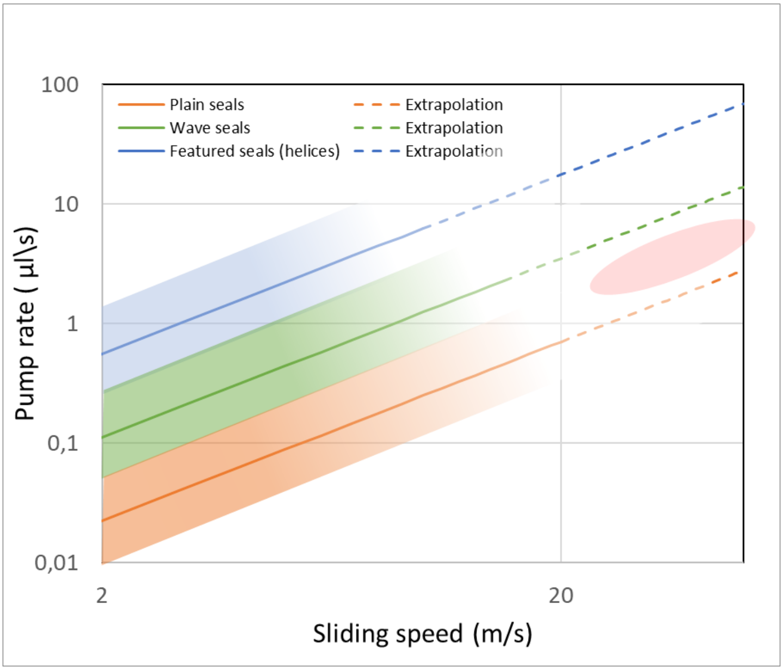

Figure 8: The lines indicate the typical pump rate behavior as a function of speed for the 3 classes of seals, the dashed lines are the model extrapolations above speeds where test data and experience is available with regards to seal pumping. The colored areas indicate the variation due to specific design and operating parameters. The red zone shows the pump rate required at high speeds to counteract the leakage rates observed during the test on the plain seal (Figure 6).

For the plain seal (Figure 7a), the reverse pumping is caused by asymmetric deformation of micro-asperities in the contact area due to shear stress caused by the friction between the seal lip and shaft during motion. Here the asymmetric pressure distribution generated by the seal lip design leads to the asymmetric deformation pattern in the seal contact. Many aspects of the rotary sealing mechanism and the reverse pumping capability of plain seals has been analyzed by several authors (Horve, 1996) (Kammüller, 1986) (Salant & Flaherty, 1995) (Salant, 1997) (Salant, 1999) (Salant & Rocke, 2004). However, when dynamic operating conditions are extreme, the reverse pumping of a plain lip rotary shaft seal is not always sufficient to guarantee secure sealing during service life.

Next, wave seal designs have a higher reverse pumping rates than plain seals. The contact footprint of these seals describes a sinusoidal path on the shaft (Figure 7b) in contrary to a straight contact path for plain seals. In this case, the total pump rate is determined as the net flow between inward and outward pumping (Horve, 1996) respectively occurring in the region of in-pumping and the region of out-pumping as will be discussed in more detail later.

A third design option to provide sufficient reverse pumping during service life is to mold additional features onto the plain lip surface that work as vanes. During shaft rotation these vanes obstruct the shear flow of the oil between the shaft and seal and as a result a hydrodynamic pressure builds up between the contact area of the feature and the plain seal lip. This creates locally a hydrodynamic pressure gradient over the plain seal lip that results in a reverse oil flow in axial direction. The strength of this flow is determined by the pressure depending on speed and vane geometry and parameters such as oil viscosity, oil film thickness and the contact width between shaft and seal.

For this third class of seals, many different seal feature designs can be identified such as helix ribs (Figure 7c), sinusoidal ribs (Figure 7d) and triangular pads. In principle, each individual vane only creates a reverse oil flow for a single rotation direction of the shaft. For example, the helix seal shown in Figure 7c is a unidirectional seal since all the vanes are oriented in the same direction. A bi-directional seal can be created by having a combination of vanes in both directions. This can be achieved, for example, by helices oriented in both directions or with the sinusoidal seal as shown in Figure 7d. However, in practice, the total pumping rate of this type bi-directional seals will be less due to the fact that part of the vanes are pumping in the other direction.

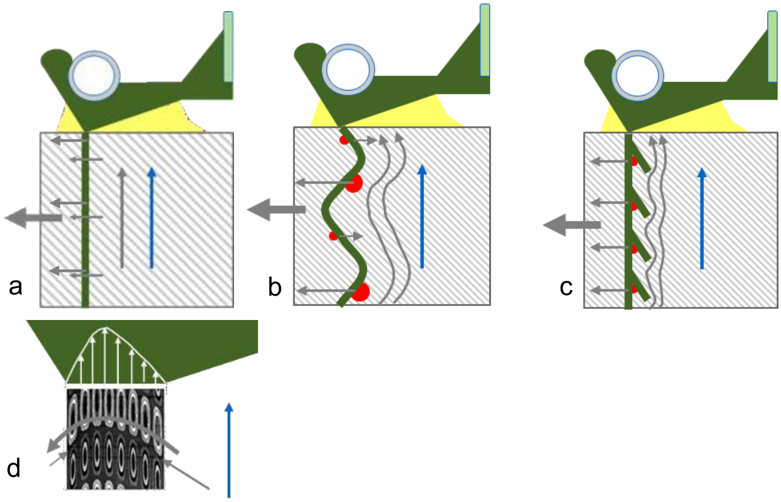

Figure 9: Contact footprints of different seal type classes. Here pane a shows the plain seal, i.e. the basic seal without any hydrodynamic features; pane b the wave seal; pane c the uni-directional helix rib seal. Pane d shows the tangential deformation pattern of the micro-asperities in the seal contact during rotation.
Here the blue arrow shows the shaft ration direction, the big grey arrow is the net reverse flow direction, the other grey arrows the local oil flow directions and the red dots the zones of high hydrodynamic pressure.

In the next section the wave seal has been selected to be further developed for high-speed EV applications. First of all, because from Figure 8, it is expected that wave seals have reverse pump rates in the order required to prevent the leakage observed in Figure 6 for plain seals. The required pump rate zone in Figure 8 is estimated as the pump rate of plain seals extrapolated to high speed (dashed orange line) to which the leakage rate for plain seals is added as observed in Figure 6 at high speeds. Secondly, wave seals are bi-directional for which, due to the in- and out-pumping, the contact is expected to be well lubricated resulting in higher film thickness and lower friction. Finally, the wave geometry, and consequently the pump rate, is relatively insensitive to wear and contamination.

Here it is noted that also helix seals with even higher pump rates are expected to prevent leakage. However, too high pump rates might lead to unwanted starvation of the seal contact leading to high contact temperatures, friction and wear.

4.2 Wave seal pumping and optimization

Similar performance test as performed on the standard (plain) seal shown in 6 has been performed on a standard low force wave seal. As expected, for this seal first leakage (few drops) starts at a higher speed compared to the standard plain seal as show in Figure 10, respectively first droplet is observed at 40 m/s compared to 30 m/s (Figure 6)

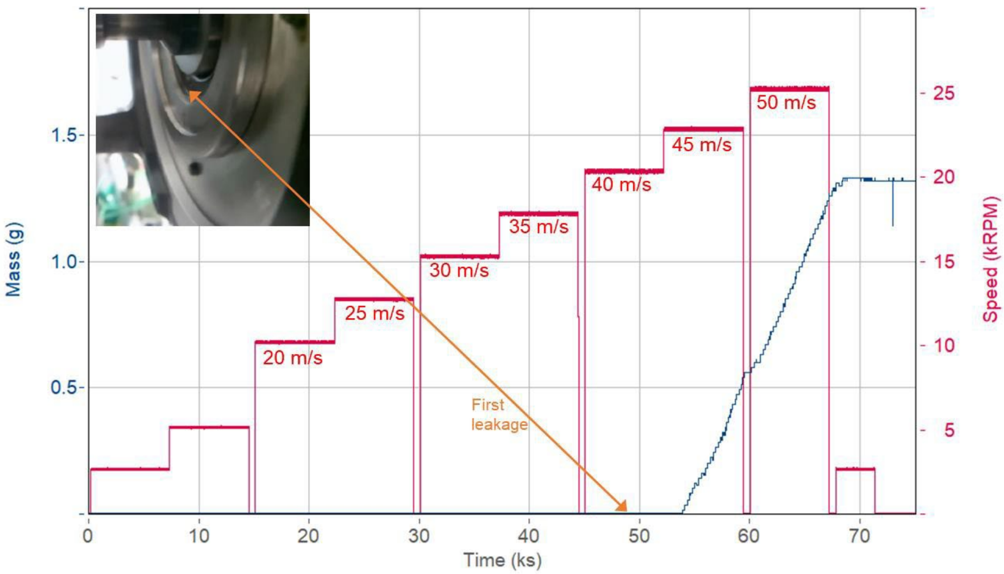

Figure 10: Performance test on a standard wave designed for sliding speeds up to 20 m/s. First oil leakage, starts at 40 m/s as observed with the camera. The total amount leaked during this speed stage is still too low before reaching the leakage sensor. The leakage rate further increases for 45 and 50 m/s.

In order to guide further improvements, the reverse pumping of the standard wave seal has been measured. In Figure 11, the measured pump rate is compared to the leakage rate it needs to compensate for, as observed in Figure 8. This shows that the reverse pump rate for mid-range speeds (35 to 40 m/s) is on the same order as the expected leakage rate whereas for higher speeds the pump rate is much larger than the leakage rate. This suggests a higher risk of leakage only at these mid-range speeds.

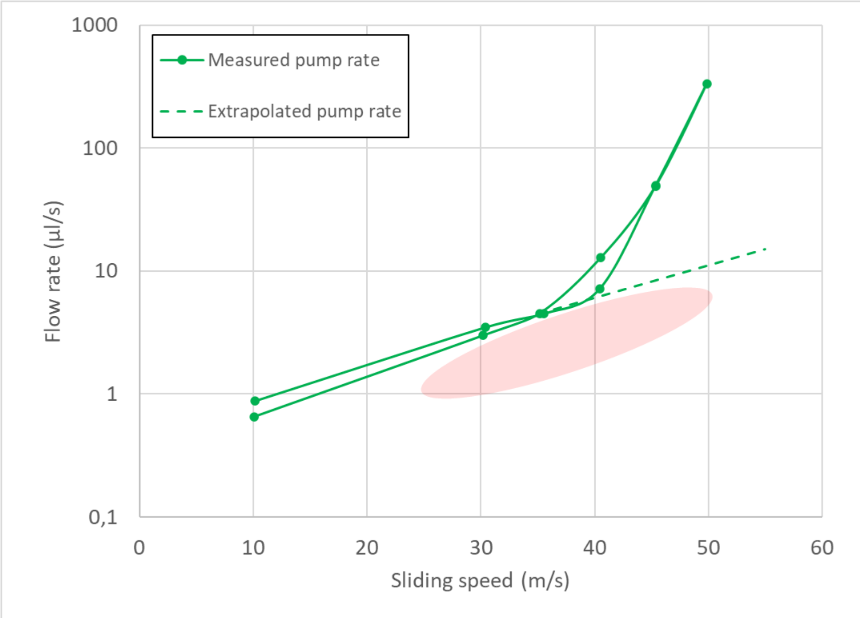

Figure 11: Comparison of wave seal measured pump rate with the required pump rate as shown in Figure 8. The dashed green line shows the high-speed pump rate expected based on the low-speed behavior of this seal and the modelling analysis in Figure 12. The red zone shows the pump rate required at high speeds to counteract the leakage as also indicated in Figure 8.

However, in Figure 12 the reverse pump rate measured for this wave is also compared to the existing pumping model for wave seals together with results for other wave seals designed for low speeds.

From this analysis is becomes clear that at high speeds the measured pump rate increases much faster with speed than might be expected from the model as well as what can be expected from the other wave seals. This is due to the nature of the test setup used to measure reverse pumping for which typically the air is completely filled with oil to assure the availability of oil around the seal circumference (Figure 13). The oil side is also completely filled with oil to avoid a pressure head over the seal during the test.

Consequently, during shaft rotation, the oil flow due to pumping is accompanied by a hydrodynamic pressure on the air side of the seal. At high flow rates, i.e. high speeds, this hydrodynamic force becomes significant with respect to the contact force and is able to lift-off the seal. In other words, the hydrodynamic force increase the gap or film thickness increases due to this effect and leads to higher pump rates than expected since in a realistic application the air side of the seal is not completely filled with oil. This type of systematic error in measuring reverse pumping is very hard, if not impossible, to solve by modifying the test setup.

Therefore, at high speeds the only way to get information on the reverse pump rate is to extrapolate the pumping model based on low-speed tests to high-speeds.

After these tests, the wave seal has been further optimized for high speeds and first performance tests has been finalized. During the performance testing, this seal did not show leakage at high speeds (Figure 14). Depending on the specific request, next the seal is up for tests in pressurized environment and for endurance test.

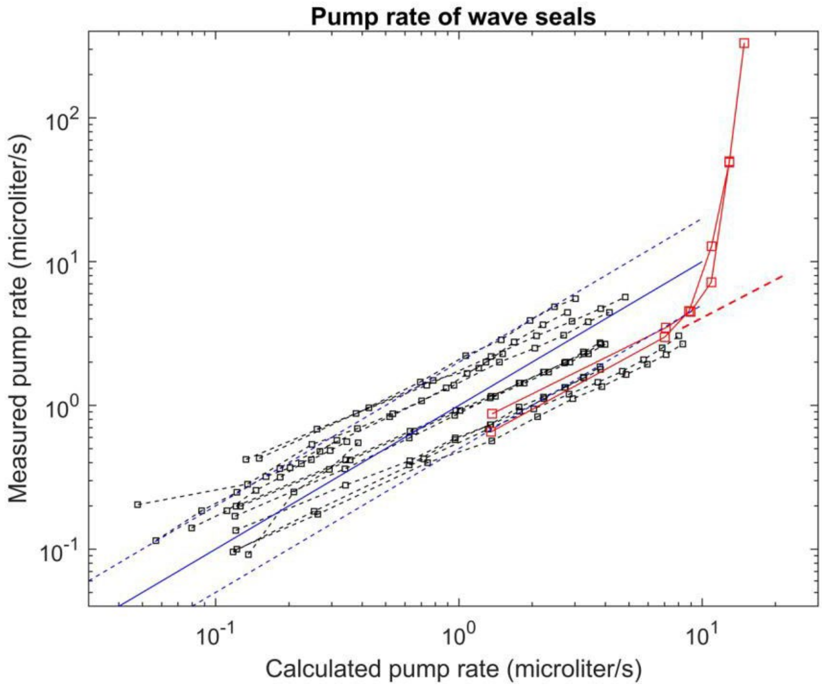

Figure 12: Comparison of wave seal pumping predictions to measurements. The solid blue line indicates a perfect match between experiment and model. The black lines are results for low-speed wave seals and the red line are the results for the high-speed wave seal. The dashed red line shows the high-speed pump rate expected based on the low-speed behavior of this seal together with the other test results and model shown.

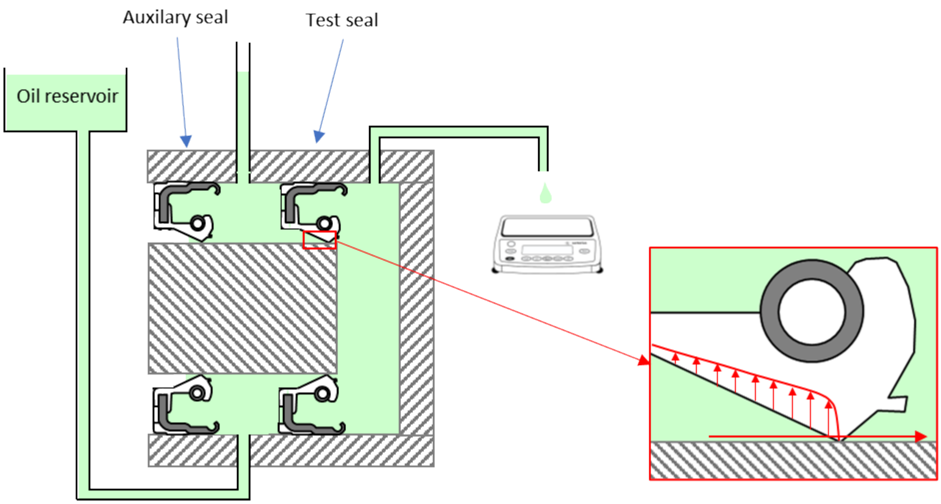

Figure 13:.Typical test setup for measuring reverse pumping of seals ([1], [2]), [3]). The chamber at the air side is completely filled with oil to assure continuous availability of oil over the full seal circumference during the test. In addition, also the chamber at the oil side of the seal is complete filled to assure no static pressure head exists over the seal lip. On the right the hydrodynamic pressure profile is shown caused by the oil flow generated by the seal during shaft rotation.

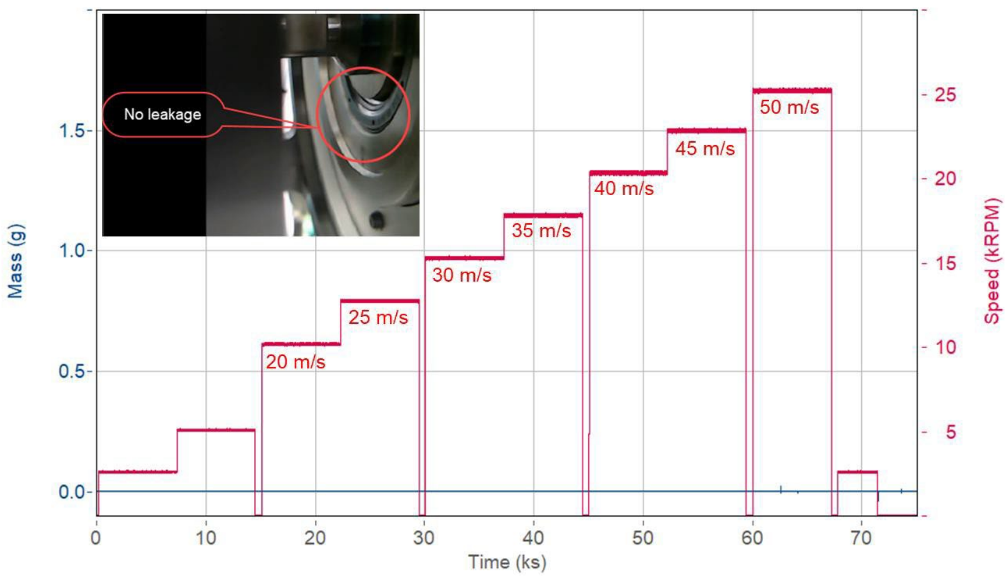

Figure 14: Wave seal design that has been optimized with regards to reverse pump rate at high speed. No leakage was observed by the sensor nor visually by the camera.

5 Discussion & Conclusion

In order to respond to the high-speed seal challenges in automotive electric drivelines, this paper shows how sealing solutions can be provided which satisfy the most demanding conditions related to speed, operating temperature and lubrication type, based on common requirements within the EV market.

First of all, it is shown that sealing ability can be improved by increasing the reverse pump rate of the seal without increasing seal contact force. This allows to keep friction and contact temperatures low with regards to carbon emission as well as to seal durability

Furthermore, the results in this paper indicate behavior that can be related to the very high speeds. For example the very high pump rates measured at high speeds on low stiffness seals are very likely due to seal lift-off caused by the reverse oil flow. Since this effect is very hard to avoid in an experiment, pump rates at high speeds have to be extrapolated from existing models or by numerical means, i.e. CFD.

High-speed tests carried out tend to indicate that the maximum leakage risk is not necessarily occurring at the maximum speed. At very high shafts speeds the oil seems to no longer reach the seal lip due to high centrifugal forces. When designing seals for high-speed applications, these types of oil inertia effects need to be taken into account.

6 References

[1] C. Burkhart, K. Peter, S. Thielen and B. Sauer, "Online determination of reverse pumping values of radial shaft seals and their tribologically equivalent system," in *22th International Colloquium onTribology 2020. Ostfildern: Techn. Akad. Esslingen*, Ostfildern: Techn. Akad. Esslingen, 2020.

[2] T. Kunstfeld, "Einfluss der Wellenoberfläche auf das Dichtverhalten von Radial-Wellendichtungen," 2003.

[3] Y. Kawahara and H. Hirabayashi, "A Study of Sealing Phenomena on Oil Seals," *A S L E Transactions 22,* p. 46–55, 1979.

[4] R. F. Salant, "Theory of lubrication of elastomeric rotary shaft seals," *Proc. Inst. Mech. Eng., J. Eng. Tribol.,vol. 213,* pp. 189-201, 1999.

[5] R. F. Salant, "Modelling rotary lip seals," *Wear vol. 207,* pp. 92-99, 1997.

[6] R. F. Salant and A. H. Rocke, "Hydrodynamic analysis of the flow in a rotary lip seal using flow factors," *ASME J.Tribol. vol. 126,* pp. 156-161, 2004.

[7] R. F. Salant and A. L. Flaherty, "Elastohydrodynamic analysis of reverse pumping in rotary lip seals with microasperities," *ASME J. Tribol., vol. 117,* pp. 53-59, 1995.

[8] L. A. Horve, Shaft Seals for Dynamic Applications, vol. 10th print, Marcel Dekker Inc. (ISBN-0-8247-9716-7), 1996.

[9] M. Kammüller, "Zur Abdichtwirkung von Radial-Wellendichtringen,," *Berichte aus dem Institut für Maschienelement und Gestaltungslehre,* Vols. Bericht nr. 20 (ISBN-3-921920-19-1), 1986.

[10] "Global Comparison of Passenger Car and Light-commercial Vehicle Fuel Economy/GHG Emissions Standards.," July 2021. [Online]. Available: https://theicct.org/pv-fuel-economy/.

7 Authors

SKF Seals Business Unit, Global Research & Innovation
Meidoornkade 14, 3992 AE Houten, The Netherlands:

Bas Van der Vorst, Dr. ir., bas.van.der.vorst@skf.com
Jeroen Lenten, Ing., jeroen.lenten@skf.com
Mickael Sansalone, Dr., mickael.sansalone@skf.com