LOCTITE® BEYOND THE BOND

LIQUID GASKETING DESIGN GUIDE

Henkel

Henkel Adhesive Technologies

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1. INTRODUCTION

Lightweight designs and the constant increase of capacity lead to highly stressed components and potential deformations at critical areas such as joints, sealing flanges, bolting elements, and attached housing areas. Also, the need for better environmental resistance with zero-leak requirements for flanged joints increases the difficulty for the designer.

To ensure that zero-leak requirements are met, it is essential that the joint be designed in accordance with gasket-specific guidelines. It is the intention of this guide to provide general design considerations that are gasket-type independent as well as specific guidelines for liquid gaskets, such as formed-in-place gaskets with anaerobic cure and room temperature vulcanizing RTV Elastomers.

The design guidelines are based on results obtained by dynamic fatigue tests, the analysis of existing gasketed joints, experience, and the results of analytical and numerical calculations within Henkel2 to 28 and independent institutes1. The general design considerations described in Section 3 summarize the gained knowledge of three projects at the University of Stuttgart sponsored by the Forschungsvereinigung Antriebstechnik, in which Henkel is an active partner.

2. DEFINITIONS

A gasket is a material positioned between two flanges which are held together by fasteners. Gaskets prevent leaking of fluids or gases by completely filling the space between the surfaces of the flanges. It is necessary for the seal to remain intact and leak-free for a prolonged time. The gasket must be resistant to the medium being sealed and able to withstand the application temperature, pressure, and micromovements caused by vibration as well as thermal expansion/contraction.

Flange Seals are classified as static or dynamic, depending on whether the sealed parts move relative to each other. A rotating shaft in a housing is an example of a typical dynamic system. While flanges are classified as static systems, they encounter small micromovements because of vibration, temperature changes, pressure changes, shock, impact or transmitting loads. The static gaskets or seals are categorized as shown in Figure 1.

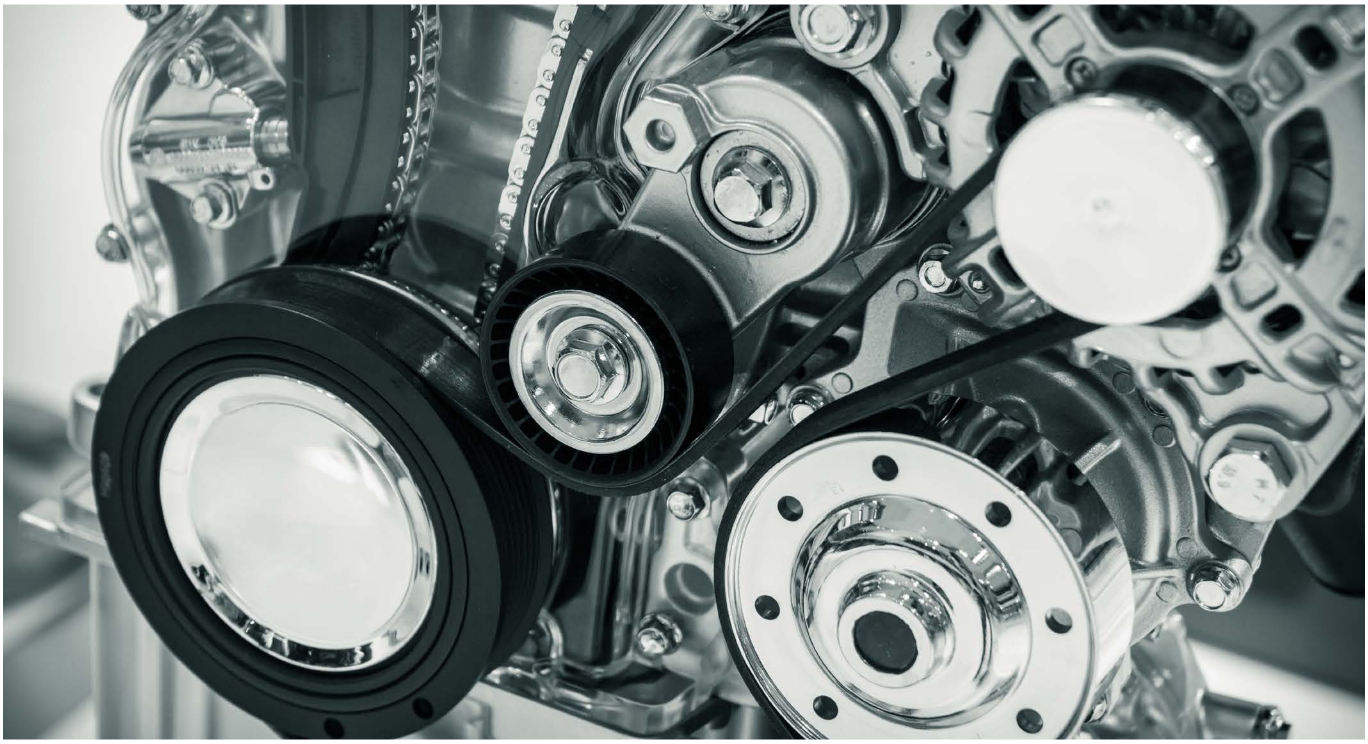


FIGURE 1: GASKET TYPES

FIP (Formed-In-Place) Gaskets are formed by the application of a bead or by screen print of liquid elastomer or anaerobic sealant, which is then assembled in the uncured state. On assembly, the sealant is smeared between the flanges and forced into surface imperfections to provide total contact between the two faces, and cures to form a durable seal.

CIP (Cured-In-Place) Gaskets are formed by the application of a bead of elastomer to one flange that is cured before the flanges are assembled. The gasket is then compressed by the mating flange to form the seal.

IIP (Injected-In-Place) Gaskets are liquid gaskets that are injected, after the assembly of the joint, into a groove between the two flange faces and then cured.

MIP (Molded-In-Place) Gaskets are molded directly onto one of the mating parts, usually into a groove.

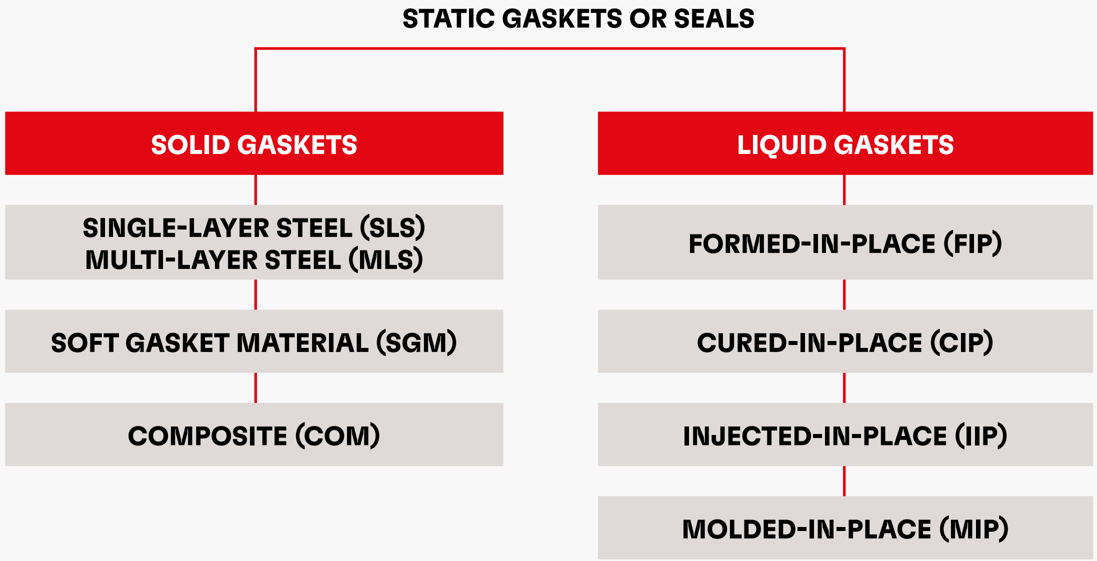


FIGURE 1: *GASKET TYPES*

3. GENERAL DESIGN CONSIDERATIONS

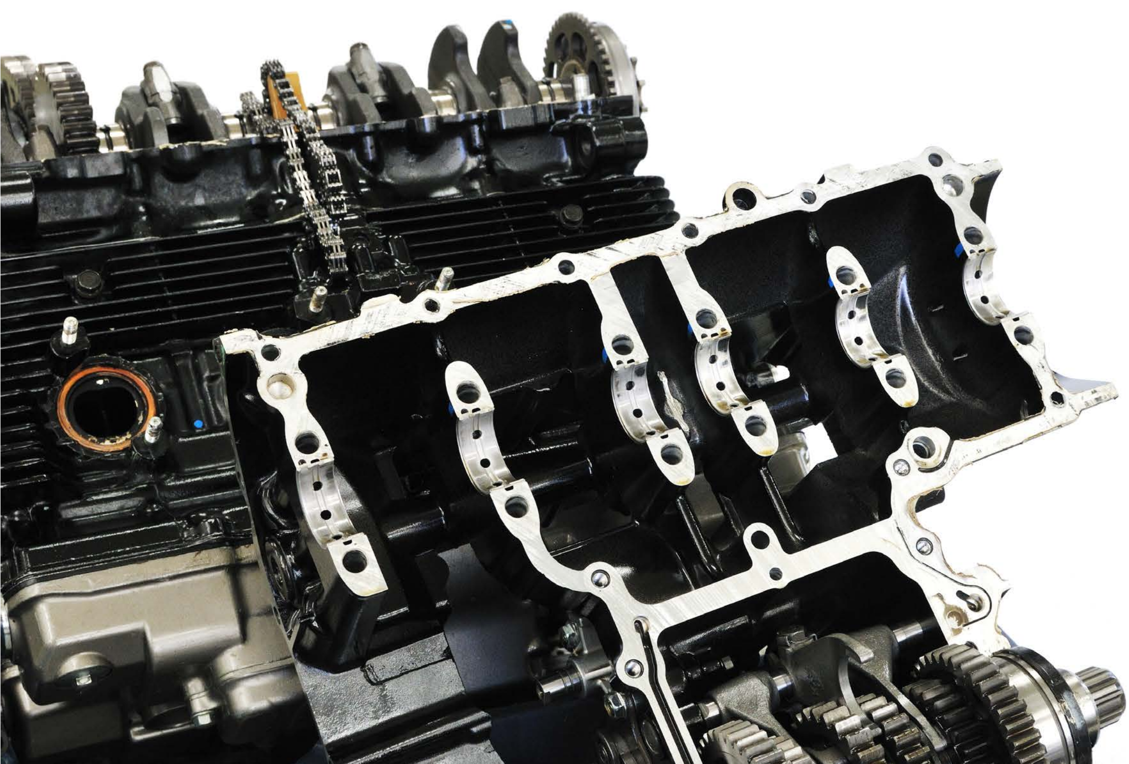
3.1 GENERAL GUIDELINES[1]

For the design of highly stressed gasketed joints, the following basic rules should be followed:

- **Flange pressure distribution**
  It is necessary to achieve uniform flange pressure distribution within the permissible limits – which include the critical minimum sealing stress of the gasket and the admissible compressive strength of the flange material and the gasket.
- **Operating load of the gasketed joint**
  The gasketed joint has to be as rigid as possible in order to minimize deformations and relative movements.

While following the above-mentioned requirements, it is essential to follow the rules explained below for the design of joints independent of the sealing material:

- **Rigidity of sealing flanges**
  The rigidity of the sealing flange is indicated by the pressure distribution in the seal joint. Select the amount of rigidity correctly, so that the critical sealing stress of the gasket is reached in all flange areas.
- **Bolt preload**
  In order to minimize the loss of initial bolt load due to the relaxation of the gasket, it is necessary to ensure sufficient compliance of the flanges and the bolts (preload reserves).
- **Consider different thermal expansions**
  Due to different thermal expansions of aluminum housings with steel bolts, a cold environment can cause greater contraction of the aluminum flange and unload both the flange and the gasket. The minimum flange pressure required for a leak free joint might then be compromised. High temperatures have the opposite effect, increasing the bolt and gasket load. In this case, the yield strength of the bolt and the compressive strength of the flange and the gasket are the limiting factors. Bolts and housings should have the same thermal expansion coefficient, if possible.
- **Stress and strain of the gasketed joint caused by external forces**
  In cases where the entire housing acts as a structure, the gasket joint should be as far as possible from the location where the forces feed into the housing.



- **Compressive stress distribution in the seal flanges**
  Compressive stress distribution in the seal flanges For optimum distribution of the bolt clamping load along the flange to the mid-point between the bolts, the bearing area of the bolt head should be as far away from the sealing area as possible. If the sealing area is in the middle of the effective bolt length (see Figure 7), the adjusting compressive stress distribution in the housing is optimized. The theoretical straight connection lines between bolts (see Figures 4 and 5) should not deviate significantly from the centerline of the gasket to allow a uniform compressive stress distribution within the whole flange width.
- **Adjusting the flange width to the compressive stress distribution**
  The bearing surface of the joint should be enlarged in the area of the bolt and reduced at midpoint between the bolts in order to obtain a more uniform compressive stress distribution in the seal joint.

3.2 BASIC DESIGN OF HOUSINGS[1]

To develop a suitable sealing concept for the housing, the following basic rules for the design are recommended:

- **Create small, spherical housing openings**
- **Use same materials for the seal flanges**
- **Try to achieve uniform temperature distribution**

3.3 FLANGE RIGIDITY[1]

The operational safety of a gasketed joint can be strongly influenced by varying the flange rigidity. The compressive stress distribution in the seal gap and in the flange between bolt head and seal gap indicates the rigidity of the joint. Figure 2 shows three possible variations of flange design and their influence on the component rigidity.

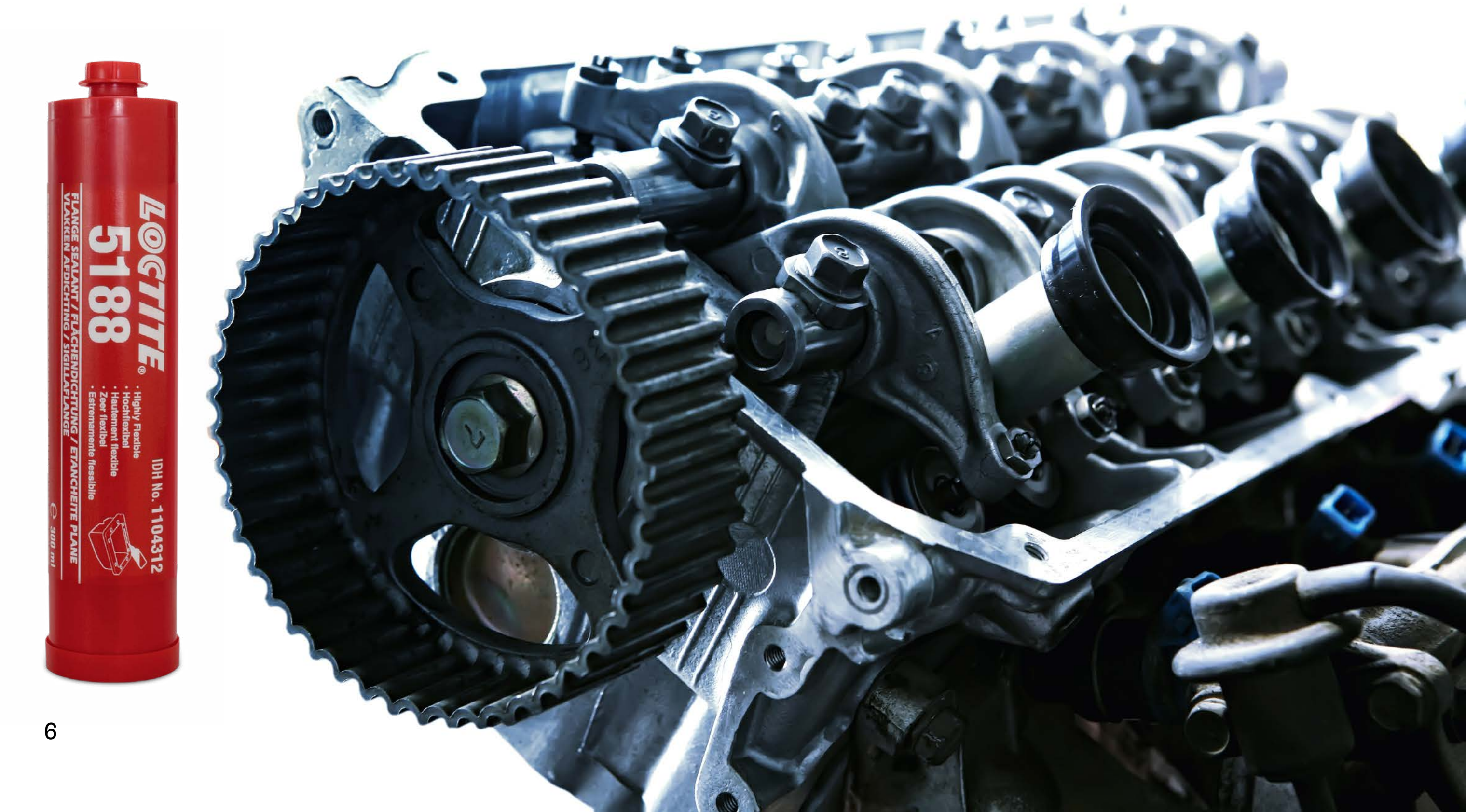
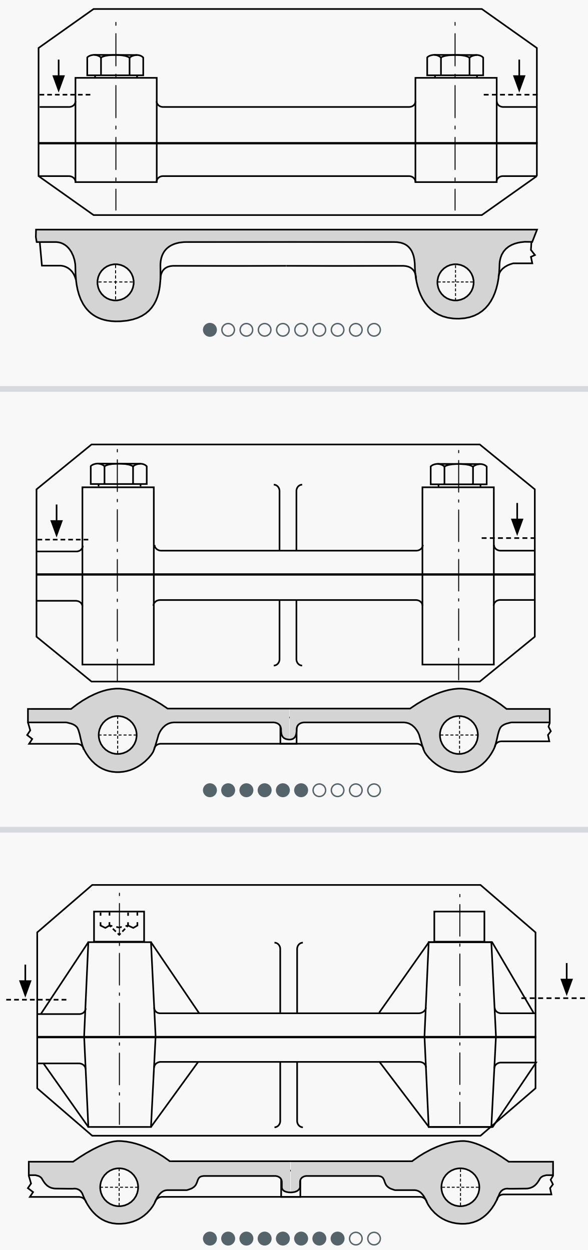


FIGURE 2:

Evaluation of the flange design in regard to the compressive stress distribution (qualitative), stiffening ribs, bosses at bolt location – Best Solution Bottom One.[1]

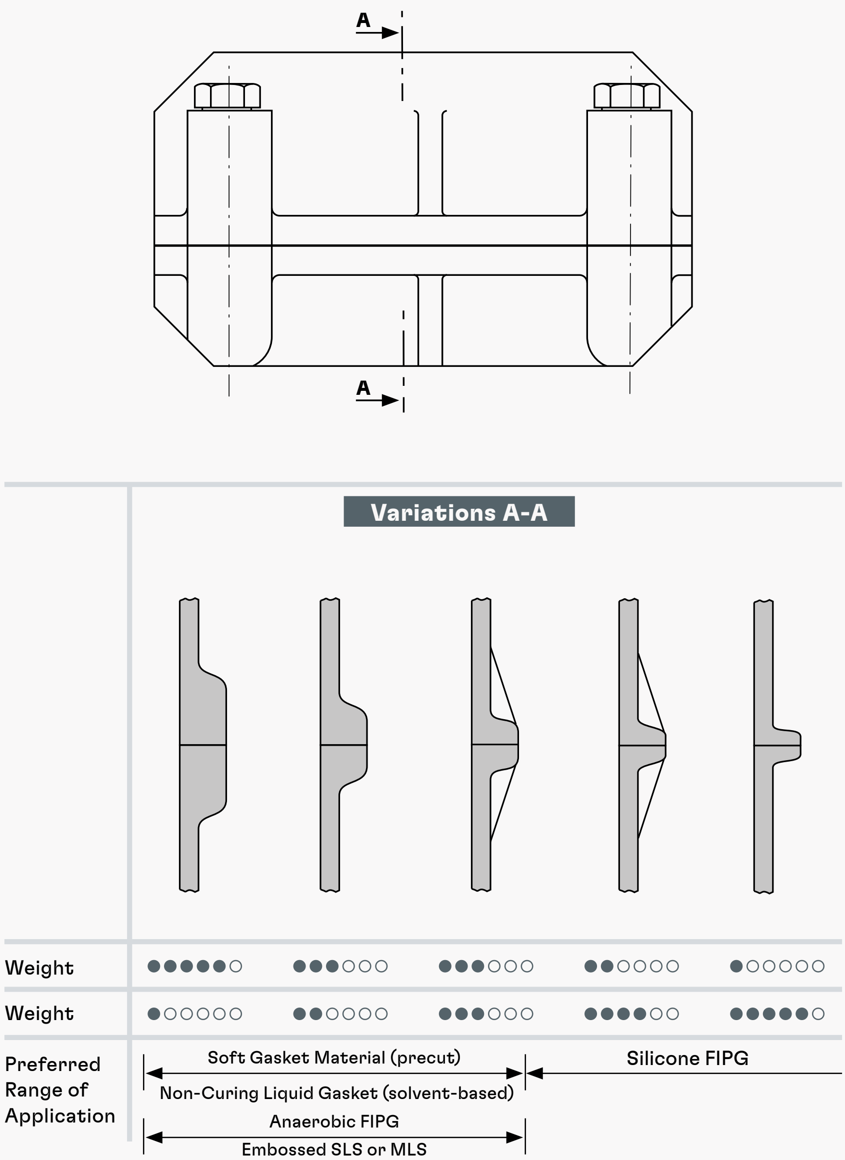


As flange rigidity decreases, the more complicated it becomes to obtain the required minimum surface pressure at mid-point between two bolts. Figure 3 is based on external research work[1] and illustrates the influence of flange rigidity on the use of static gasket types. Soft gasket material and non-curing liquid gaskets are only suitable for flanges with adequate bending rigidity. If the minimum bending rigidity value is reached at the flanges, a change to anaerobic formed-in-place gaskets or embossed single-layer or multi-layer steel gaskets is required. The range of application for anaerobic formed-in-place gaskets is even wider, covering the whole range of very rigid flanges to flanges with medium rigidity.

Internal research work[2,13] has shown that in cases where the required minimum surface pressure for anaerobics is not achieved or in cases of flexible flanges, like metal sheet parts, RTV Elastomers formed-in-place gaskets are suitable. The gasket has to become even more flexible with the potential to flex in bowing and shearing direction due to additional integral design features such as designed gaps, chamfers or retention grooves (see Section 6).

FIGURE 3:

Influence of flange rigidity on the use of static gasket types demonstrated on cast parts.



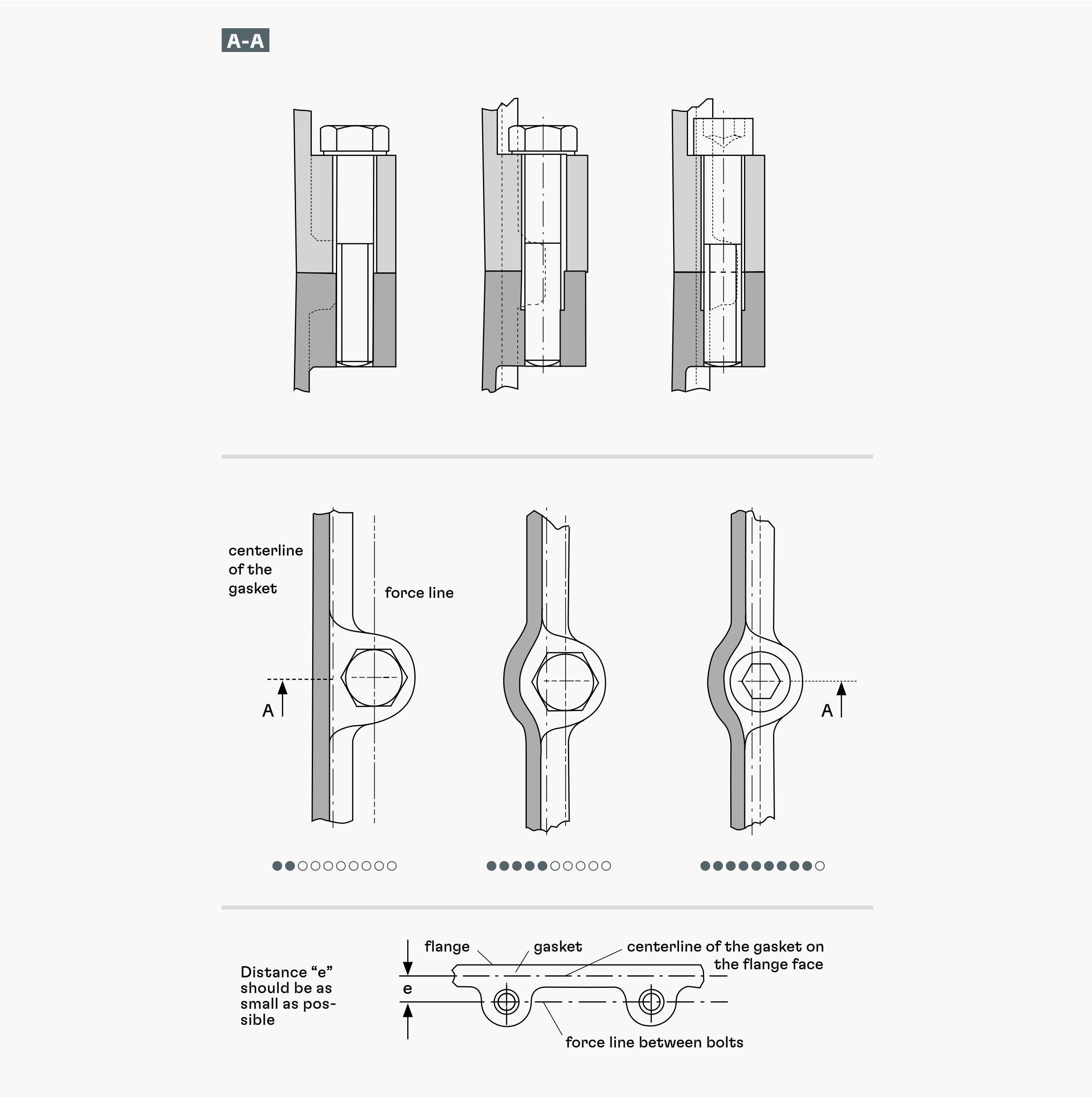
3.4 BOLT POSITIONING AND SPACING1

The best clamping pattern is invariably a combination of the maximum practical number of bolts, even spacing, and optimum positioning.

- Straight lines drawn from bolt to bolt, called bolt force lines, should be as close to the centerline of the gasket as possible to achieve uniform flange pressure distribution and avoid separation of the flanges due to potential prying (see Figure 4).

FIGURE 4:

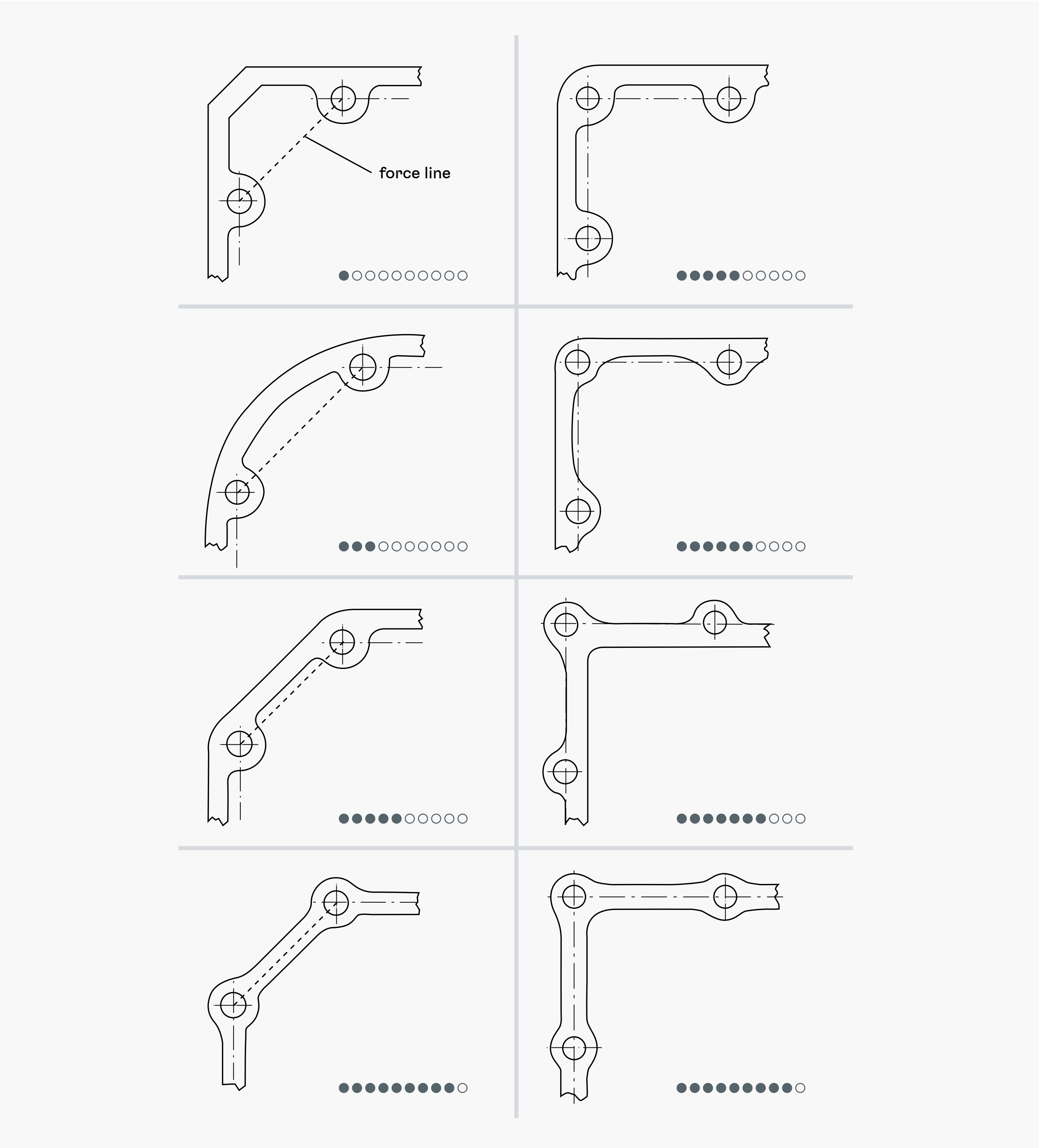
Evaluation of the distance between bolt force lines and centerline of the gasket in regard to the compressive stress distribution in the seal gap[1]



In addition, the position of the bolts is very important for the design of flange corner locations. Figure 5 shows different design variations and their valuation.

FIGURE 5:

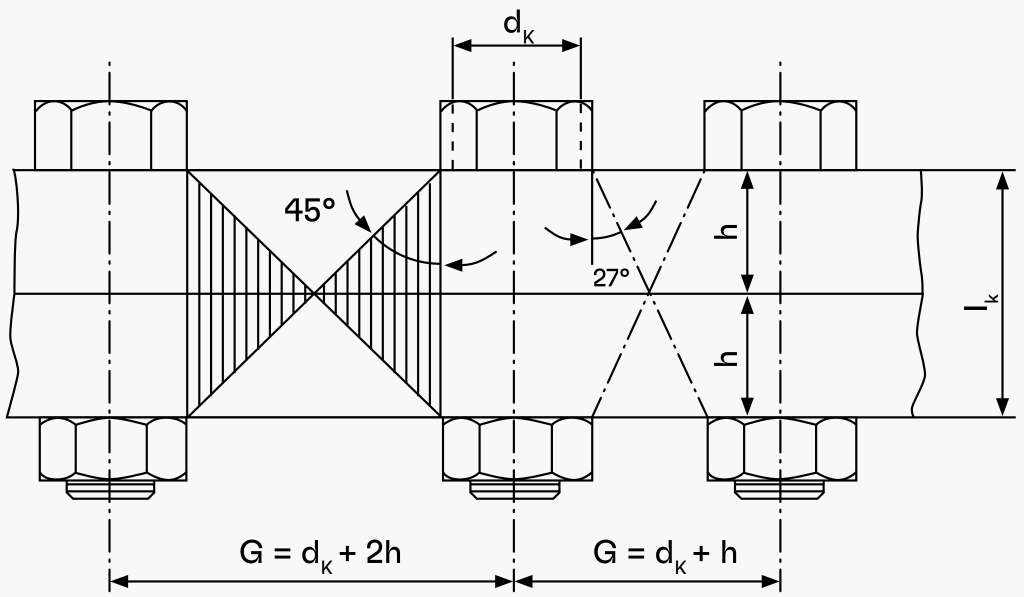
Evaluation of the position of the force lines to the centerline of the gasket in relation to the compressive stress distribution.[1]

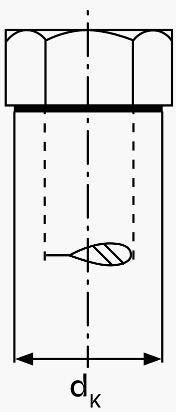


Theoretically, the bolt spacing can be calculated using the idealized model proposed by Rötscher. Rötscher's model says that the compressive stress in the flange between bolt head and seal gap is distributed as a cone with a half angular aperture of 45°, as shown in figure 6. For optimum bolt spacing, the pressure cones should at least touch each other or preferably overlap as demonstrated with the half angular aperture of 27°. Rötscher's model recommends overlapping cones for bolted joints without gaskets. For gasketed joints, research projects1 have shown that touching cones with a half angular aperture of 45° or greater are useful for the calculation of bolt spacing. For highly stressed gasketed joints, bolt spacing between the two limits, 27° and 45°, is recommended.

FIGURE 6: PRESSURE CONE ACCORDING TO RÖTSCHER.

The equation shows that the flange rigidity as well as the effective bolt length are important parameters for bolt spacing. Figure 7 shows the resulting compressive stress distribution in the joint with variations of these parameters.





$d_k + h \leq G \leq d_k + 2h$

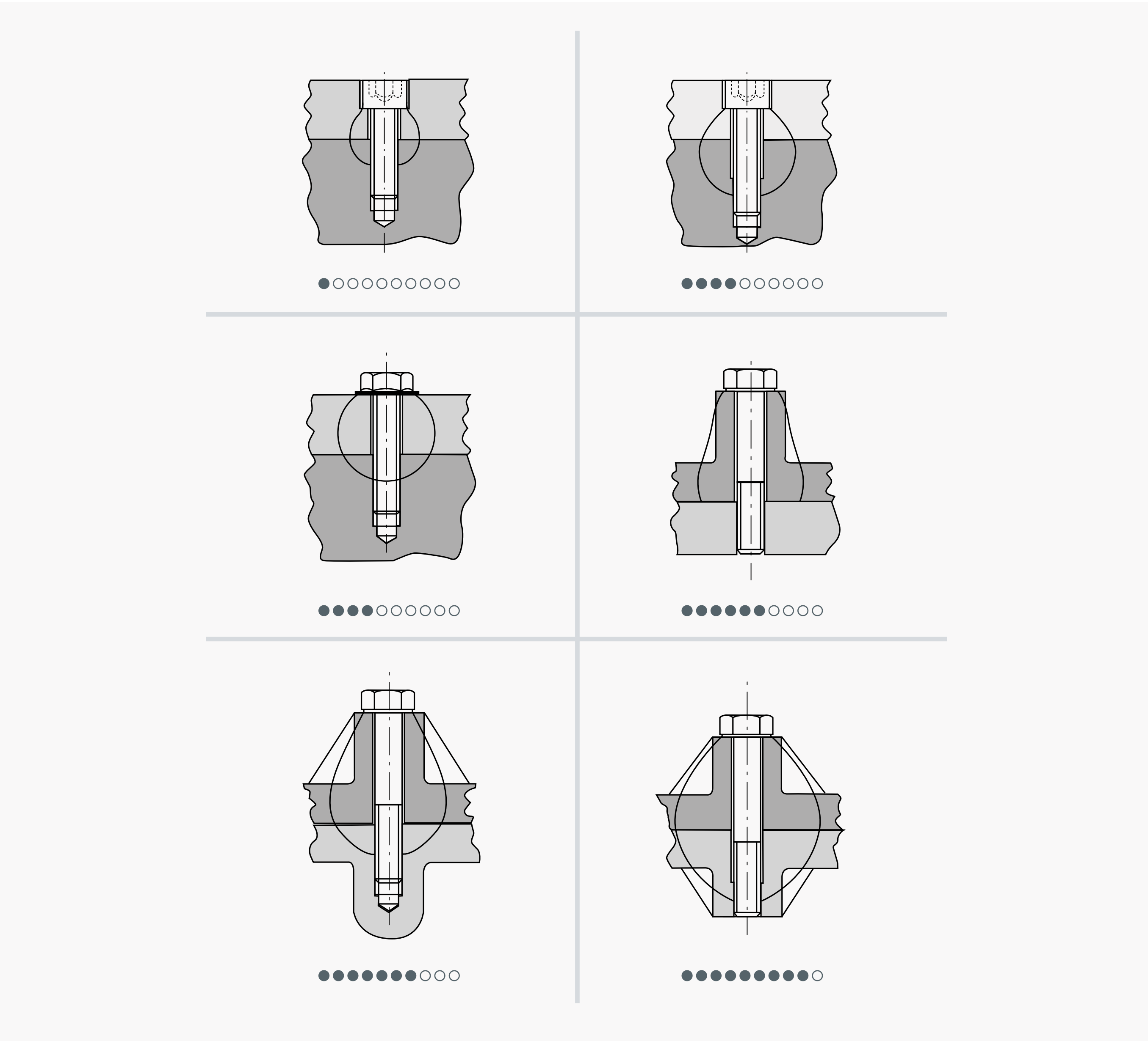
d_k ... diameter of the bearing area of the bolt head

3.5 BOLT GRADE AND LENGTH

- Select a bolt where the required initial bolt load is 80% of the proof load.
- Select a bolt that an initial bolt load of roughly 3 to 3.5 times the normal operating tensile loads can be applied (internal pressure, temperature effects, and external loads).
- Rule of thumb – when the length of the bolt is five times greater than the diameter, it can be elongated sufficiently to work as a spring between two flanges and dampen vibration.
- Optimum thread engagement length for steel is 1.2 times the diameter of the bolt; for cast iron it is 1.5 times; for aluminum, it is 1.6 times plus the tolerance for the run out of the thread or, for dynamic loading, plus 20%.

FIGURE 7: INFLUENCE OF THE FLANGE DESIGN ON THE COMPRESSIVE STRESS DISTRIBUTION OF THE FLANGES.[1]

The equation shows that the flange rigidity as well as the effective bolt length are important parameters for bolt spacing. Figure 7 shows the resulting compressive stress distribution in the joint with variations of these parameters.



3.6 FLANGE DESIGN VERIFICATION

- **One of the basic rules for the design of highly stressed gasketed joints is to achieve a uniform flange pressure distribution within the gasket-specific permissible limits.**
- **It is necessary to know in the early stages of the design process if the required flange pressure is achieved in the seal gap.**

The flange pressure distribution can be evaluated in a very early state with a Finite Element Analysis (FEA) or later on using prototypes with a pressure sensitive film, manufactured by Fuji Photo Film Company. A completely new design will rely on both systems since the FEA allows the design optimization in a cost-effective way and the imprint of the pressure sensitive film is needed for the confirmation of the numerical calculation.

To evaluate the stress distribution, the film is pre-cut to the shape of the mating flanges and holes are pre-punched for the fasteners. The film is then placed between the flanges and the bolts tightened to the specified torque. When pressure is applied, microcapsules break and a color-forming material is released. The microcapsules are adjusted to break at different pressure levels, with the resulting color density dependent on the amount of pressure. Thick red color indicates the applied pressure is high while fainter shades indicate the applied pressure is low. Using a commercially available Fuji densitometer, impression color density can be directly converted to stress readings.

A disadvantage of the system is that only the maximum applied force is recorded, whereas the unloading of the gasket under operating conditions such as temperature, pressure or dynamic loads cannot be measured. These effects have to be evaluated by FEA. The film gives an indication of the weak points in gasketed joints, such as areas with low or no flange pressure, and focuses the FEA on these points for optimization. The film also shows machining marks and problems with flange mating tolerances, such as flatness and overlapping.

Besides the FEA, real-time flange pressure mapping is possible with the Tekscan thin-film pressure profile measurement system at the prototype stage. With the Tekscan system, a high resolution, matrix-based tactile sensor is placed between the flanges. The software supplied with the sensor is capable of dynamic data collection and display where pressure cycling or the recording of an event might be important.

3.7 SURFACE FINISH

Surface finish or surface textures are terms used to describe the general quality of a workpiece surface. Surface finish consists of roughness, waviness, lay and flaws. Bearing and locating surfaces usually require close dimensional and surface finish control for proper functioning.

The surface finish is most important for conventional gaskets, since the initial compressive load required to deform the gasket into the flange surface irregularities increases with rougher surface finish.

For FIP sealants, the surface roughness has no influence on the initial compressive load because the product is in the fluid state during the assembly process; however, it does have an effect on the formation of the sealant layer thickness.

- At higher surface roughness Ra ≥3.0 µm (ten point height Rz ≥17 µm), metal-to-metal contact is achieved independent of the compressive stress.

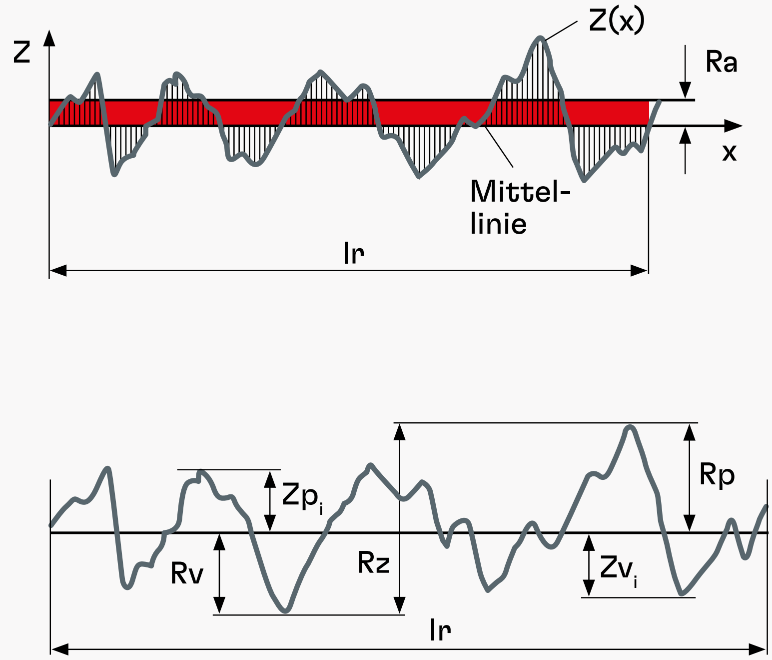
- At lower surface roughness Ra ≤0.3 µm (ten point height Rz ≤3 µm), FIP sealants tend to generate an adhesive layer which decreases with increasing flange pressure. Metal-to-metal contact is achieved only in the areas near the bolts. (The actual metal-to-metal contact between the most carefully finished interfacing parts does not exceed 25-35%.)

- Smoother surface finish aids cleaning and ensures removal of surface contamination before sealant application

- Blowout resistance decreases with increasing gaps. Therefore, surface finish is important for blowout pressure tests during assembly, when the FIP sealant is still in the uncured state.

The surface characteristics are determined primarily by means of electrical stylus instruments according to DIN EN ISO 4287:1998 (ANSI B46.1-1971). The two most common measurements of surface finish are Ra and Rz, as shown in figure 8.

FIGURE 8: RA AND RZ ACCORDING TO DIN EN ISO 4287:1998 (ANSI B46.1-1971).



$$Ra = \frac{1}{lr} \int_{0}^{lr} |Z(x)|\,dx$$

Ra is the mean value of the absolute values for the profile deviations within one individual measurement section lr. Rz is the maximum peak-to-valley height within one individual measurement section lr.

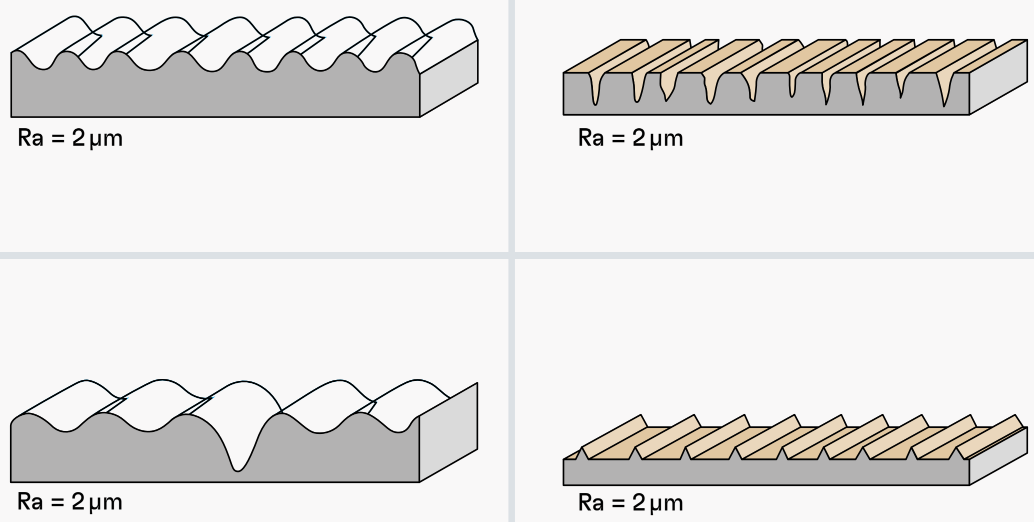
The formerly used ten point height uses Rz, which is the mean value of the absolute values of the heights of the five highest peaks and of the five highest valleys for five individual measurement sections within the evaluation length (Rz ≥ ten point height Rz).

The Ra value alone is not sufficient to determine roughness, since different surface textures can have the same Ra value, as visualized in figure 9. At least Ra and Rz should be measured, but using Ra, Rz, Rmax and Wt provide a more accurate picture.

Maximum peak-to-valley height, Rmax, is the greatest individual peak-to-valley height of the entire measurement section ln (= 5 x lr).

Wt is the maximum profile height of the filtered waviness profile within the entire measurement section ln.

FIGURE 9: DIFFERENT PROFILES WITH THE SAME RA VALUES.



GENERAL GUIDELINES FOR FLANGE SURFACE ROUGHNESS

125 micro inch (5 microns)	▪ Significant design improvement for liquid flange sealants. There is virtually no upper limit as roughness improves surface area and bond strength
60 micro inch (1.5 microns)	▪ Typical flange finish
40 micro inch (1 micron)	▪ Adequate surface roughness
20 micro inch (1/2 micron)	▪ Should be thoroughly tested
10 micro inch (1/4 micron)	▪ Too smooth

4. FIPG APPLICATION AREAS, ADVANTAGES OVER SOLID GASKETS

Unlike conventional gaskets, FIP technology does not require extreme compressive loading to form a seal, due to the adhesion of the cured FIPG to all the members in the joint. The main benefits of FIP over solid gaskets are:

- **No gasket relaxation** – FIP sealants allow metal-to-metal contact in most applications, which ensures correct bolt tension throughout the life of the assembly and eliminates the need for retorquing.
- **Non-shimming** – The metal-to-metal contact eliminates the need for gasket thickness, so tolerances can be more accurately maintained.
- **Relaxed surface finish** – FIP sealants allow relaxation of surface finish and flatness tolerances. Scratches and scored surfaces can be sealed without reworking the damaged surfaces.
- **Chemical compatibility** – FIP sealants demonstrate good to excellent solvent resistance.
- **Reduced inventory costs** – FIP sealants can seal different flange geometries, unlike solid gaskets that require stocking many different gaskets to fit the different geometries.
- **Automatic application** – FIP sealants can be applied by fully automated robotic dispensing or screen or stencil printing systems.
- **Easier handling of vertical components** – FIP sealants can be applied to both horizontal and vertical flange faces. Unlike solid gaskets, they do not require additional adhesive to maintain their position on vertical flange faces.
- **Reduced hydrocarbon emissions** – The reduced seal gap decreases hydrocarbon emissions compared to solid gaskets.

To achieve the required sealing performance on a wide range of flanges, 2 types of FIP sealants are frequently used:

- Anaerobic sealants
- Room temperature vulcanizing RTV Elastomers

ANAEROBIC SEALANTS

Anaerobic sealants cure in the absence of air and the presence of active metal surfaces. These products are best suited to seal rigid flanges, which are designed to:

- Achieve optimum stiffness between two mating parts
- Minimize movement between two parts
- Transmit forces from one part to another

Typical examples of rigid flanges are found in vehicles including gearbox housings, bedplate to crankcase, water pump to engine block, and cam cover to cylinder head.

Anaerobic FIP sealants are used for rigid bolted joints because they:

- Offer metal-to-metal contact
- Ensure correct bolt tension
- Accurately maintain final dimension tolerances
- Add structural strength and reduce micromovement
- Can be easily disassembled by applying a cleavage load to the joint
- Offer high-pressure resistance if sufficient clamp load is provided
- Remain in liquid form; unlike other FIP sealants, anaerobic sealants cure only between flange faces. Excess material can be wiped away from exterior surfaces or flushed away from interior surfaces (liquid sealants are miscible in many fluids, e.g., oil).
- Offer extensive on-part life when exposed to air – making multiple application methods possible, and reducing the problems associated with the use of volatile and/or moisture-cured sealants.

ROOM TEMPERATURE VULCANIZING RTV ELASTOMERS

RTV Elastomers cure to a rubbery solid by reacting with moisture in the environment. These products are best suited to seal flexible flanges, such as gearbox covers, timing chain covers, stamped sheet steel parts, thin-walled metal castings, and oil pans. Unlike rigid flanges, they do not usually support the function of the component, therefore micromovement between the flanges can be tolerated and an optimum clamp load distribution is not necessary.

Flexible flanges are normally used to:

- Cover an opening in a housing
- Seal a liquid inside a component or protect it from external contamination
- Cover moving parts to increase safety
- Encapsulate components to reduce noise

Apart from flexible flanges, there are additional types of flange designs that require flexible gaskets, such as:

- Parts where the required compressive stress distribution for anaerobics cannot be achieved
- Assemblies with different flange materials and large differences in their thermal expansion coefficients, which can result in bowing of the flanges
- Flanges where more than two parts are mounted together, forming T-joints

RTV Elastomers can be used for flexible joints as well as for stiff joints, and offer a number of benefits, including:

- High-gap filling
- Capability to seal joints with micromovements
- Ensuring metal-to-metal contact
- Achievement of correct bolt tension with no setting
- Sealing of T-joints
- Creating seals on non-machined flanges
- Potential for robotic application also for AN gaskets
- Creating seals between metal and plastic components or even between plastic and plastic materials

5. DESIGN RECOMMENDATIONS FOR ANAEROBIC FIPGS

To achieve optimum seal performance on rigid bolted joints, the general design considerations of Section 3 should be observed. In addition, several design features specific to anaerobic FIP sealants should be followed, including:

- Machined flanges with surface characteristics:
    - Ra 0.8 to 3.2 μm
    - Rz 3 to 21 μm (10 point height)
    - Rmax 4 to 30 μm
    - Flatness 0.1 mm @ 400 mm
- A minimum overlapping flange width of 5 mm (to ensure reliable curing)
- A minimum overlapping flange width around bolts of 3 mm (to ensure reliable curing)
- Chamfer dowel and bolt holes to eliminate raised metal and shimming
- The maximum gap at surface imperfections or machining marks must be within the maximum cure through volume range (0.1 to 0.25 mm sealant dependent)
- Typical minimum flange contact pressure for passenger car applications of 2.5 MPa
- Use alignment dowels for assembly of large parts to prevent smearing of the sealant and ensure accurate positioning of the two mating surfaces
- Conduct instant seal tests at least 20 minutes after assembly with test pressures ≤0.03 MPa/0.3 bar/4.3 psi for minimum possible duration

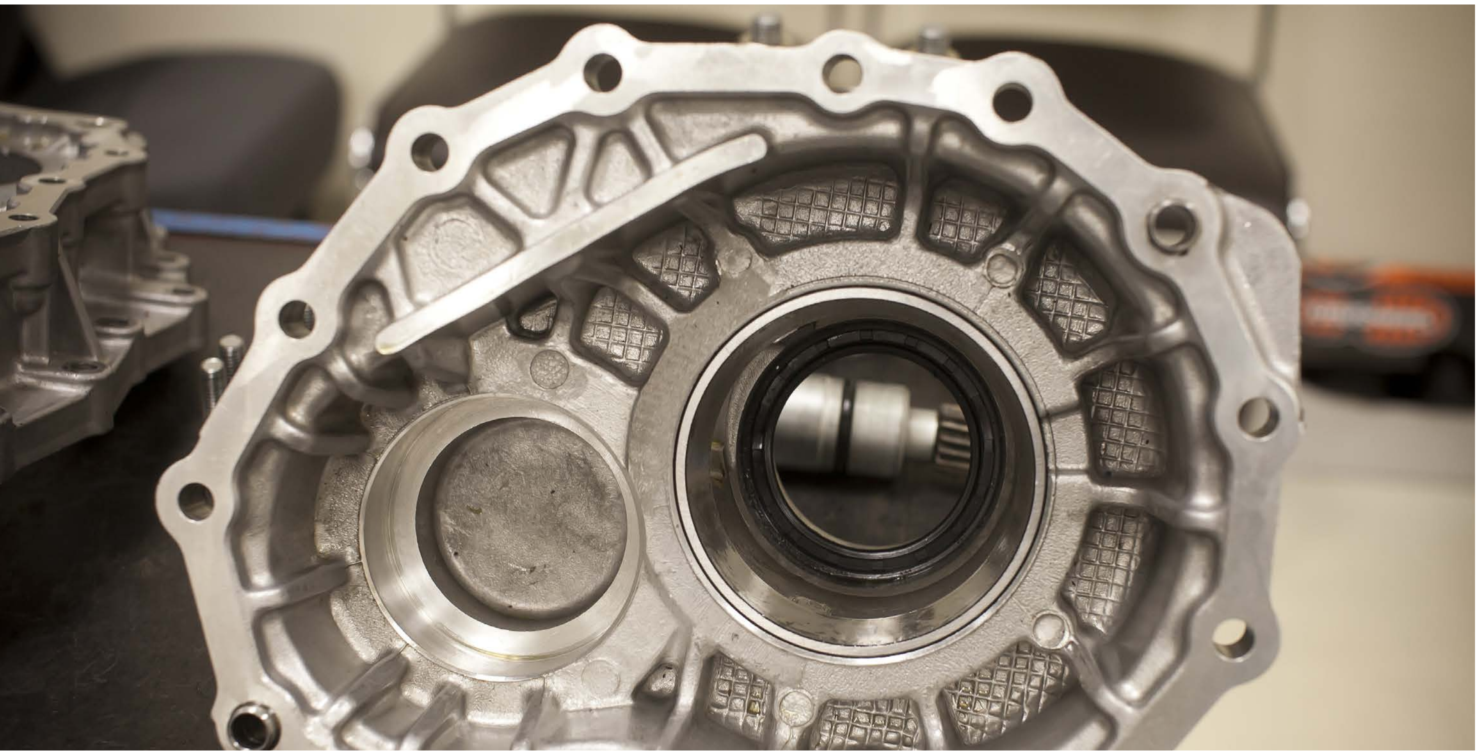
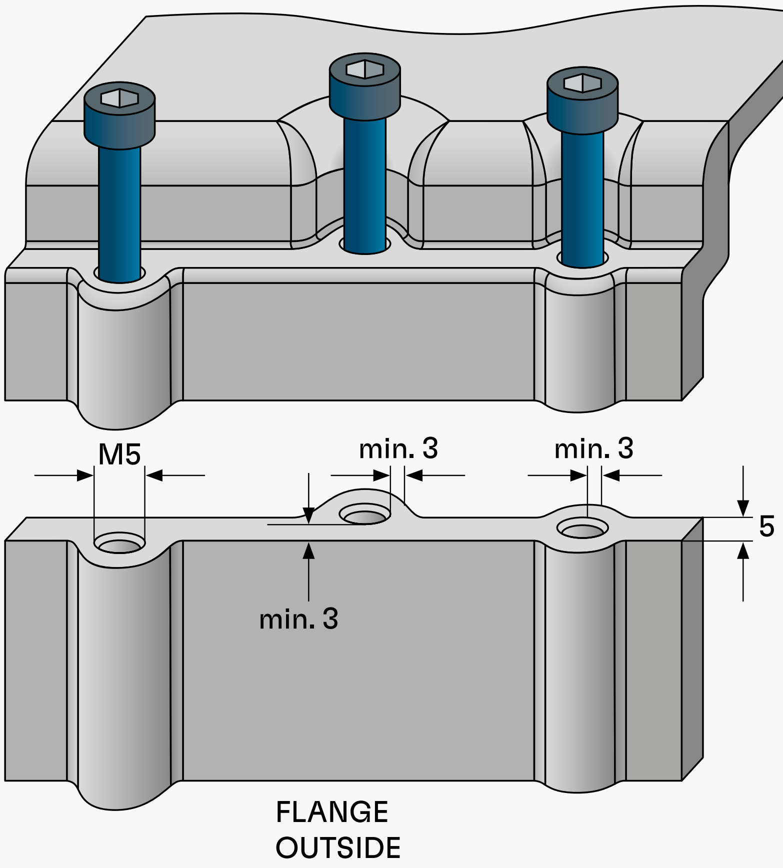


FIGURE 10: FLANGE DESIGN FOR ANAEROBIC FIP SEALANTS.



In addition to the correct design of the flange, reliable sealing with FIP gaskets depends on adhesion to the flange surface. The adhesion is strongly influenced by the cure performance. The anaerobic cure starts when atmospheric oxygen is excluded. Then free radicals are formed and under the effect of metal ions (Cu, Fe), these free radicals initiate the polymerization process.

Reliable curing and maximum adhesion are achieved by:

- Cleaning the flange faces
- Using activators or heat for substrates which are less active (stainless steel, high-alloy steel, aluminum with low copper content, anodic coatings, or chromate films)

6. DESIGN RECOMMENDATIONS FOR RTV ELASTOMERS

RTV Elastomers are able to seal joints with poor flange design and also critical areas such as T-joints, where anaerobic sealants or hard gaskets will have problems. Nevertheless, the general design considerations of Section 3 should be followed to achieve a reliable, durable seal.

WHY CHAMFER?

Performance tests and experience showed that an inside chamfer often is the best design to seal a joint when using RTV Elastomers. The main advantages of using a chamfer include:

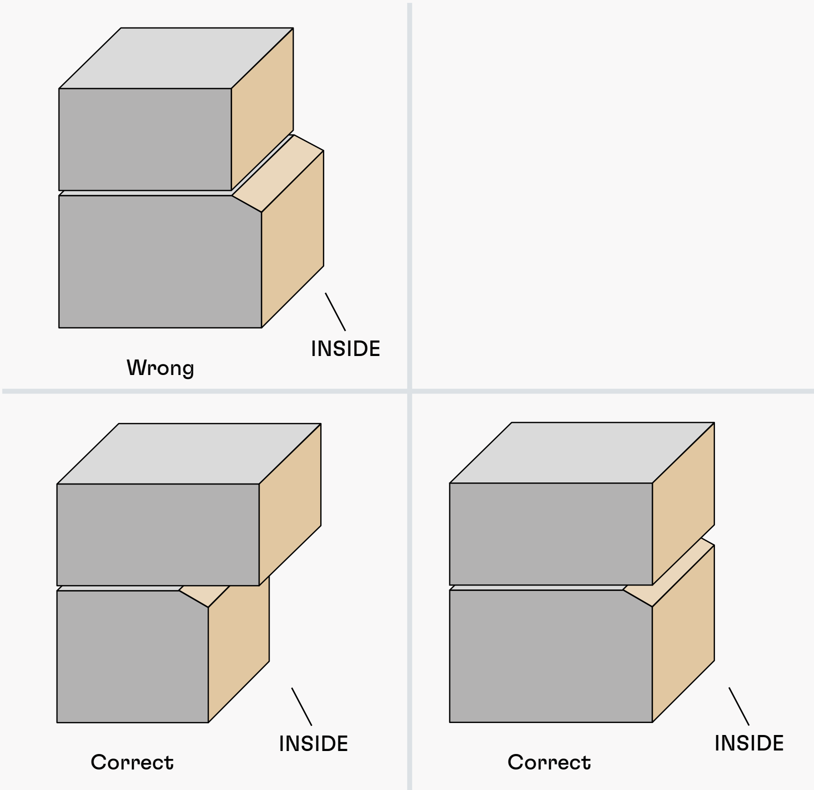
- Defined flow of product
    - No release of product pieces due to squeeze-out
- Defined, filled gap
    - Good durability because of product layer (in the chamfer)
- Fast curing
    - Product is fully cured when supplied to final customer
- Easy manufacturing
    - Cast surface leads to cost reduction
- Low product consumption
    - Cost reduction
- Oil exchange
- Guide for manual dispensing
    - Quick for manual dispensing

6.1 BASIC FLANGE DESIGN

Flanges with surface characteristics as diverse as stamped sheet steel and cast surfaces can be sealed.
The following recommendations are important for designing a joint sealed by a RTV Elastomer:

- Recommended surface characteristics:
  - Ra 0.5 to 8 µm
  - Rz 5 to 90 µm (10 point height)
  - Rmax <100 µm
- Flatness of both parts has to be defined in a way to avoid gaps of more than 0.3 mm between flange surfaces
- Maximum gap of 0.3 mm to allow an instant seal test after 20 minutes with 0.05 MPa/0.5 bar/7.2 psi or an early engine start (related to flow characteristics)
- Minimum overlapping flange width of 5 mm not including the chamfer (for instant sealing capabilities)
- Minimum overlapping flange width of 3 mm around bolts, not including the chamfer (for product dispensing)
- Inside cast chamfer width 2 mm minimum with angle 30° or alternative stamped inside radius 4.5 mm (for long-term sealing capability)
- Chamfer or radius only on one flange must be fully covered by mating part in all areas (see Figure 11)
- Use alignment dowels for assembly of large parts to prevent smearing of the sealant and ensure accurate positioning of the two mating surfaces
- When filling the chamfer area:
  - The minimum defined chamfer area has to be completely filled
  - Completely fill the minimum overlapping flange as shown in Figure 11

FIGURE 11: FLANGE ALIGNMENT.



When the required minimum surface pressure for anaerobics is not achieved, the gasket has to become more flexible. In those cases, RTV Elastomers are able to cope with movements in bowing and shearing direction due to their flexibility and integrated design features (e.g., chamfer).

It is obvious that the missing surface pressure and the movements also require an RTV Elastomer sealant with excellent adhesion to the substrate.

High adhesion and a reliable seal can only be achieved by:

- Properly cleaning the flange faces
- Selecting the correct product
- Assembling the parts within the skin-over time as listed on the Technical Data Sheet
- Ensuring the correct product quantity – typical bead size is 2.5 ± 0.5 mm (passenger car applications)
- Ensuring the correct bead location – always apply on the flat flange area, located 1 mm from the start of chamfer as shown in Figure 12
- Dispensing on either side of the joint if possible – not necessary to apply on the chamfered side

FIGURE 12: CHAMFER DESIGN.

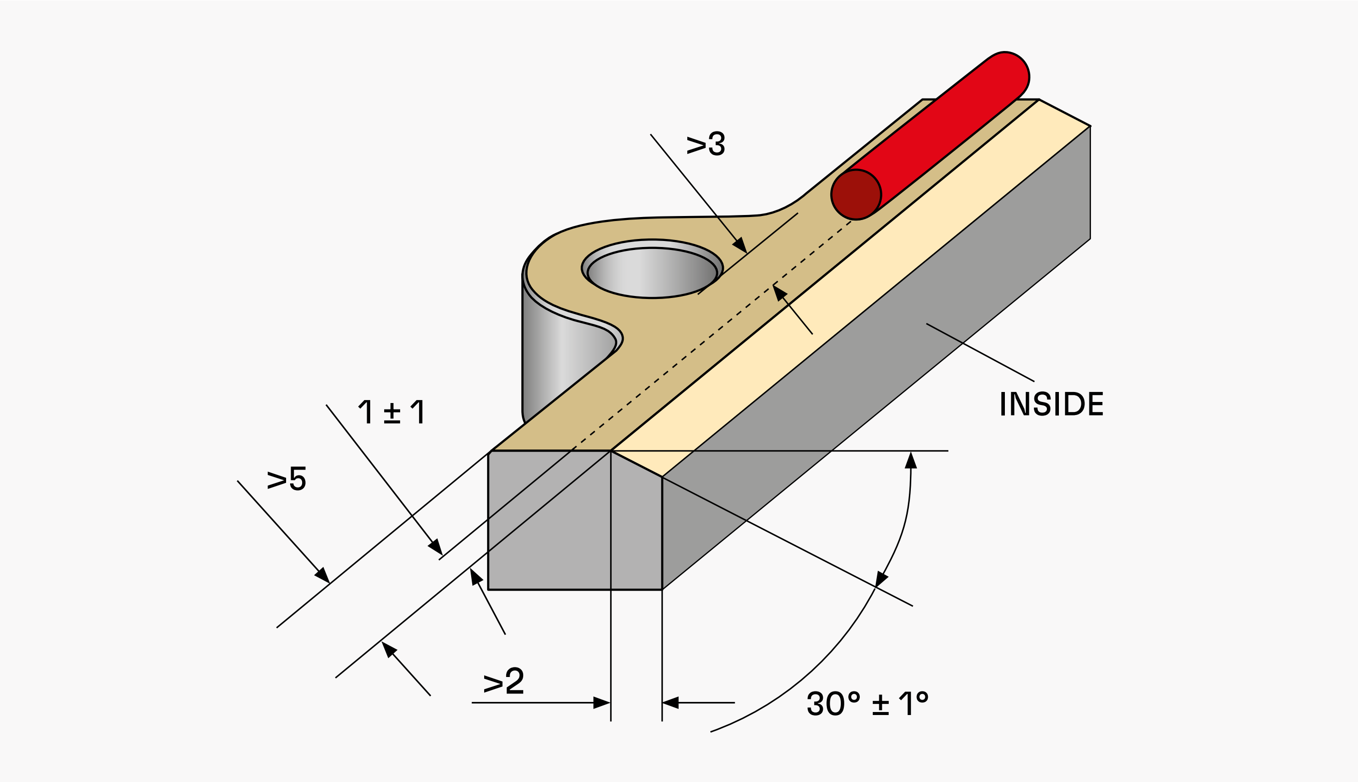
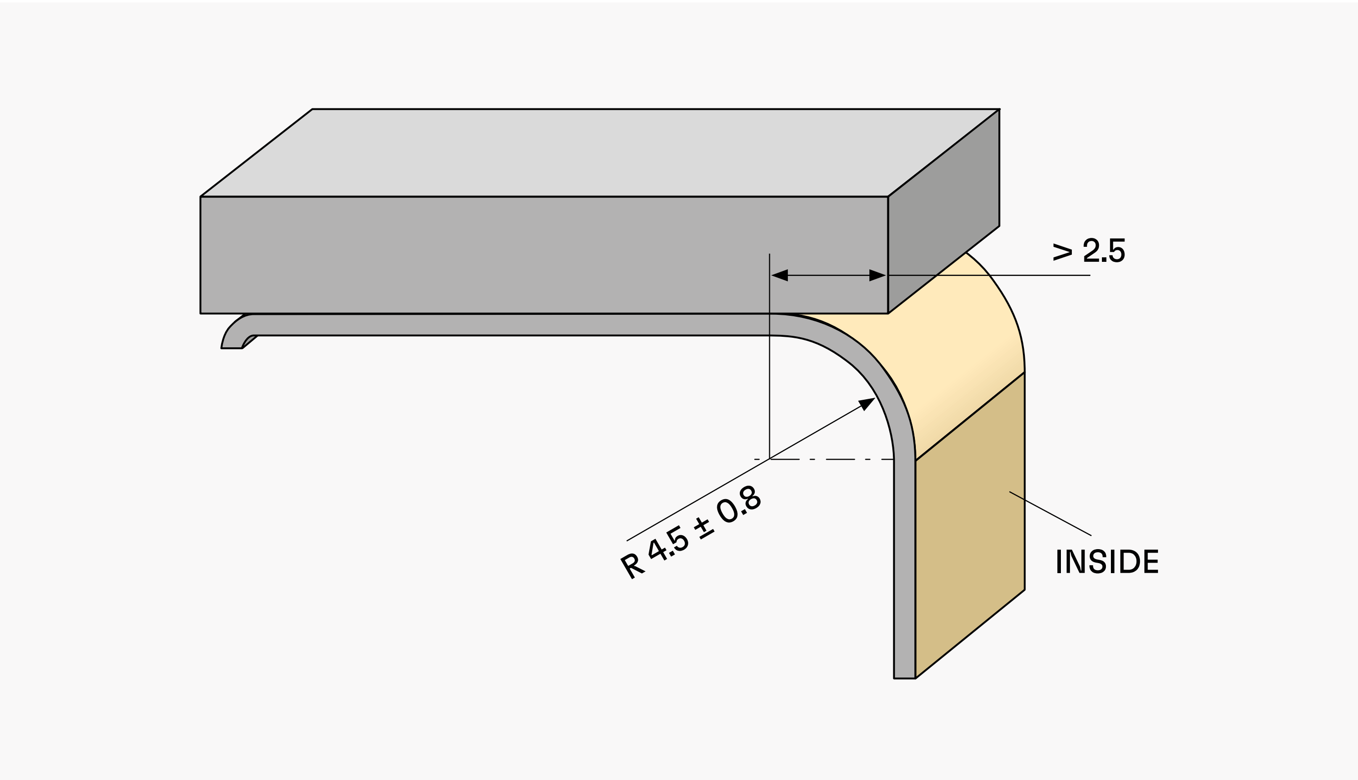


FIGURE 13: RADIUS DESIGN FOR STAMPED PARTS.

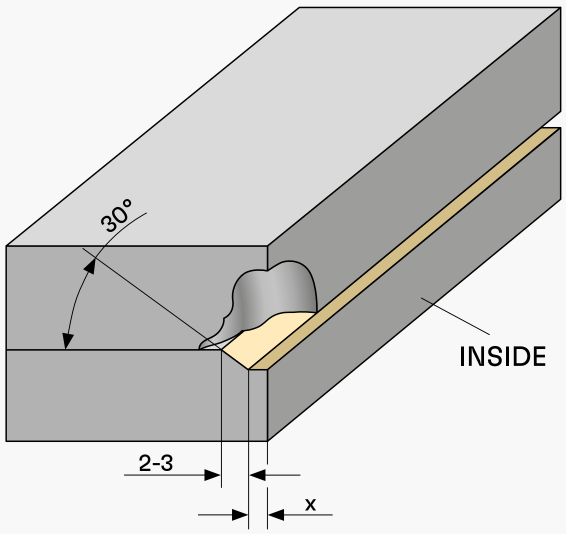


6.2 ALTERNATIVE DESIGN

CHAMFER-STEP

RTV Elastomer sealants usually need many hours to achieve a full cure. In areas where the uncured RTV Elastomer sealant can come in contact with the flowing media (e.g., oil pressure bores during an early engine start), a standard chamfer design is not sufficient. The liquid RTV Elastomer sealant might be washed away. To avoid direct contact with the flowing media, add an additional step to protect the RTV Elastomer sealant.

FIGURE 14: CHAMFER-STEP DESIGN / USE: E.G., FOR HIGH PRESSURE HOLES.



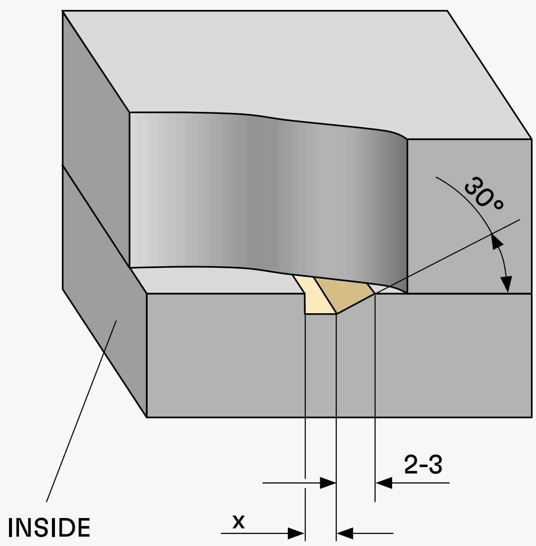
For this design, only the chamfer area has to be completely filled with RTV Elastomers. Dimension x must be able to pick up all the sealant that will be extruded through the chamfer. Worst case dimensions must be taken into account. Common dimension for x is 2 mm.

CHAMFER-GROOVE

On a real flange, a simple chamfer design often cannot be used in areas close to additional supporting bolts or bearings. The only possible design feature is a groove. To achieve fast curing, good product flow and dispensing freedom, a 30° chamfered groove on one side is the best solution.

The groove should be designed to avoid completely filling it with sealant, even under the worst case conditions (maximum product volume with minimum groove size). This can be achieved by designing a wider chamfer, having a wider flat bottom of the groove, or using both design elements.

FIGURE 15: CHAMFER-GROOVE DESIGN / USE: E.G., FOR BEDPLATE.



Note:
The groove chamfer never has to be completely filled with RTV Elastomer sealant, even under worst case tolerance conditions. Only the chamfer area has to be completely filled with sealant. Common dimension for x is 2 mm.

GROOVE

The groove design, in general, is not recommended for RTV Elastomer sealants because of the long curing time of the entrapped product. In addition, excess sealant can squeeze out at both the inside (fluid side) and outside of the flange. The excess sealant on the fluid side can break off and contaminate the engine fluid being sealed. On the outside edge the excess squeeze-out can cause cosmetic concerns. Even under worst conditions, the groove must be completely filled.

Common groove shape is a semicircle:

- width: 3.0 + 0.5 mm typical
- depth: 1.5 + 0.5 mm typical
- distance groove/bolt holes: 2 to 3 mm typical

T-JOINTS

The most critical areas to seal are where three sealing surfaces meet each other, known as the T-joint. The typical gaskets that will exist at a T-joint are FIPG, SLS (single-layer steel), MLS (multi-layer steel), or molded rubber types (press-in-place gasket, backbone carrier gasket, void-volume carrier gasket, edge molded carrier gasket, etc.) Special attention should be paid to joints when sealing with FIPG hard gaskets. To achieve a reliable seal, focus on the correct design and the tolerance of these joints, as well as on the assembly process and the product dispensing (see also Section 7.2 Dispensing).

The general rule is to avoid gaps of more than 0.3 mm under the worst conditions.

THERE ARE SEVERAL POSSIBLE SCENARIOS:

A) When sealing anaerobic to RTV Elastomers, the flange with the anaerobic always has to be assembled first to avoid curing and adhesion problems. Excess squeeze-out on the T-joint surface must be wiped off prior to applying the RTV Elastomers on the second sealing surface.

B) When sealing RTV Elastomers to RTV Elastomers, a continuous chamfer at the inside has to be formed after assembly (see Figure 16).

C) When sealing RTV Elastomer to molded hard rubber, cork, or metal gaskets, there are two possibilities:

When the first assembled joint is RTV Elastomer sealant and the second assembled joint is molded rubber, apply enough RTV Elastomer sealant to provide squeeze-out at the T-joint. Excess squeeze-out must be either wiped off or the mating parts must be assembled within the recommended RTV Elastomer sealant open time.

For the second assembled joint, the molded rubber gasket should incorporate a flat pad at the T-joint location. An RTV Elastomer sealant “dollop” must be dispensed at the T-joint location on the first flange position at the T-joint centerline (split line).

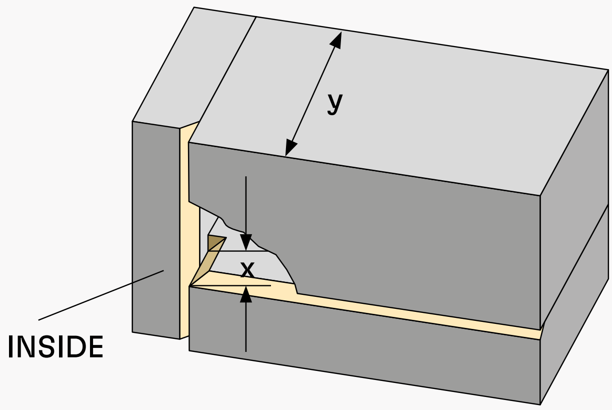
When the first assembled joint is the molded rubber gasket and the second assembled joint is the RTV Elastomer sealant, enough RTV Elastomer sealant must be applied to the second assembled joint to provide squeeze-out at the T-joint. The recommended position for the molded rubber gasket is normally flush to the surface of the T-joint. The molded rubber gasket may not protrude more than 0.5 mm at the end of the flange and the gasket recession must not be greater than 0.5 mm.

D) When sealing a SLS or MLS to RTV Elastomer sealant gasket, the first assembled joint is the SLS or MLS and the second assembled joint is the RTV Elastomer sealant. Apply sufficient RTV Elastomer sealant to the second assembled joint to provide squeeze-out at the T-joint. The SLS or MLS gasket must be greater than 0.5 mm thick and be flush (no protrusion allowed) to 1 mm recessed maximum. The RTV Elastomer sealant bead size for the T-joint will depend on the flange design of the second assembled joint with additional RTV Elastomer sealant at the T-joint. It is not recommended to have a T-joint design where the first assembled joint is the RTV Elastomer sealant and the second assembled joint is the SLS or MLS gasket.

When sealing to the SLS or MLS gasket, the RTV Elastomer sealant should have good contact to the sealing area of the hard gasket. Adhesion to the SLS or MLS gasket is essential. The hard gasket has to have sufficient surface pressure even at the end of the joint to avoid leakage through the hard gasket and movement of the hard gasket relative to the flange surfaces (different thermal expansion). One way to better cope with such relative movements is to integrate a semi-chamfer similar to Figure 16 (with no inside chamfer). See also Figures 17 and 18.

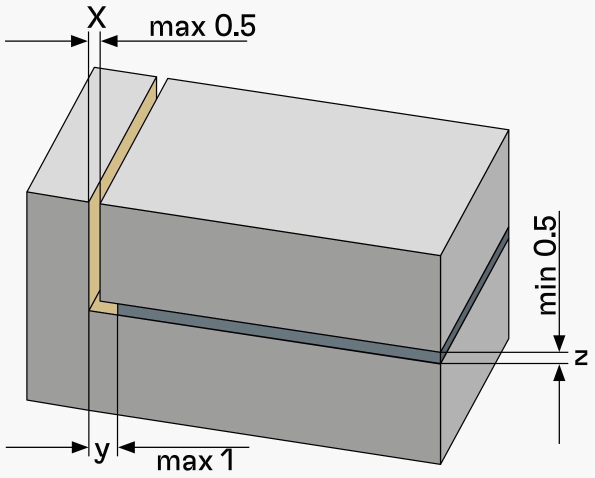
E) When sealing a SLS or MLS gasket to molded rubber gaskets, the first assembled joint is the SLS or MLS gasket and the second assembled joint is the molded rubber gasket. The SLS or MLS must be greater than 0.5 mm thick and be flush (no protrusion allowed) to 1 mm recessed maximum. For the second assembled joint, a flat pad at the T-joint should be incorporated into the molded rubber gasket at the T-joint location to allow a maximum width to assist in sealing the joint. An RTV Elastomer sealant "dollop" of typically 8 mm in diameter at the base must be dispensed at the T-joint location on the first flange positioned at the T-joint centerline. Again it is not recommended to have a T-joint design where the first assembled joint is the molded rubber gasket and the second assembled joint is the SLS or MLS gasket.

FIGURE 16: FLANGE GEOMETRY FOR RTV ELASTOMER SEALANTS AT T-JOINT.



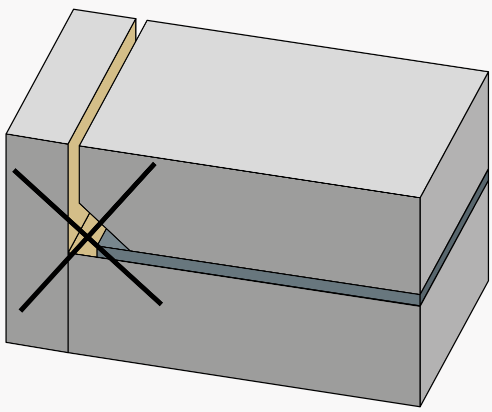
The dimension x always has to be smaller than the overall flange width y.
Common dimension:
x = 3 mm (y-x > 3 !)
The minimum recommended flange width y is 10 mm.

FIGURE 17: RECOMMENDED FLANGE GEOMETRY FOR RTV ELASTOMER SEALANTS AT T-JOINT WITH HARD GASKET (IN BLUE).



Machining and positioning tolerances may cause gap x to be up to 0.5 mm and above. Dimension y may be 0.8 mm or more. Gasket thickness z may exceed even 0.8 mm. In those cases it is necessary to understand the assembly procedure in detail and know the timing for cold or hot tests with the maximum occurring pressure and duration. Further, it will be necessary to apply more sealant at the T-joint (see also under Section 7.2, figure 20).
Verification tests under worst-case conditions are a must.

FIGURE 18: ADDITIONAL CHAMFER AT T-JOINT IS NOT RECOMMENDED. THERE IS A RISK OF LOSING PRESSURE ON HARD GASKET AND POOR INSTANT SEALING CAPABILITY.



7. FIPG SOLUTIONS FOR USE IN LIMITED DESIGN SPACE APPLICATIONS

Increasing lightweight demands, growing integration of digital systems and powertrain electrification are leading to increase in the number of components and systems, denoted in the previous guideline part as flexible or high-compliant. The housings of these components are thin-walled and made from aluminium die casting, high-performance plastics or deep-drawn metal sheets, and are by nature less stiff than gearbox or combustion engine housings. Examples include inverter housings, electronic control unit cases, sensor housings, door closing actuators, car lighting carriers or electronic module covers.

Unlike rigid flanges, the mating parts often do not support structural requirements or are not bearing sufficient loads such as provided by reaction forces. This may frequently lead to poor bolt positioning and large distances, resulting in insufficient contact pressure conditions. However, operational conditions in conjunction with the low structural stiffness may result in significant micromovement.

This type of applications is frequently addressed by RTV silicones, as recommended in Section 4. Depending on the requirements and the substrate materials to be sealed, the application of other liquid sealants with the ability to accommodate higher micromovement such as silane-modified polymers or polyurethane-based grades can be also purposeful. The use of PU-based adhesives may be purposeful when structural requirements

FIGURE 19: DESIGN CONSTRAINTS IN COMPLIANT APPLICATION EXAMPLE

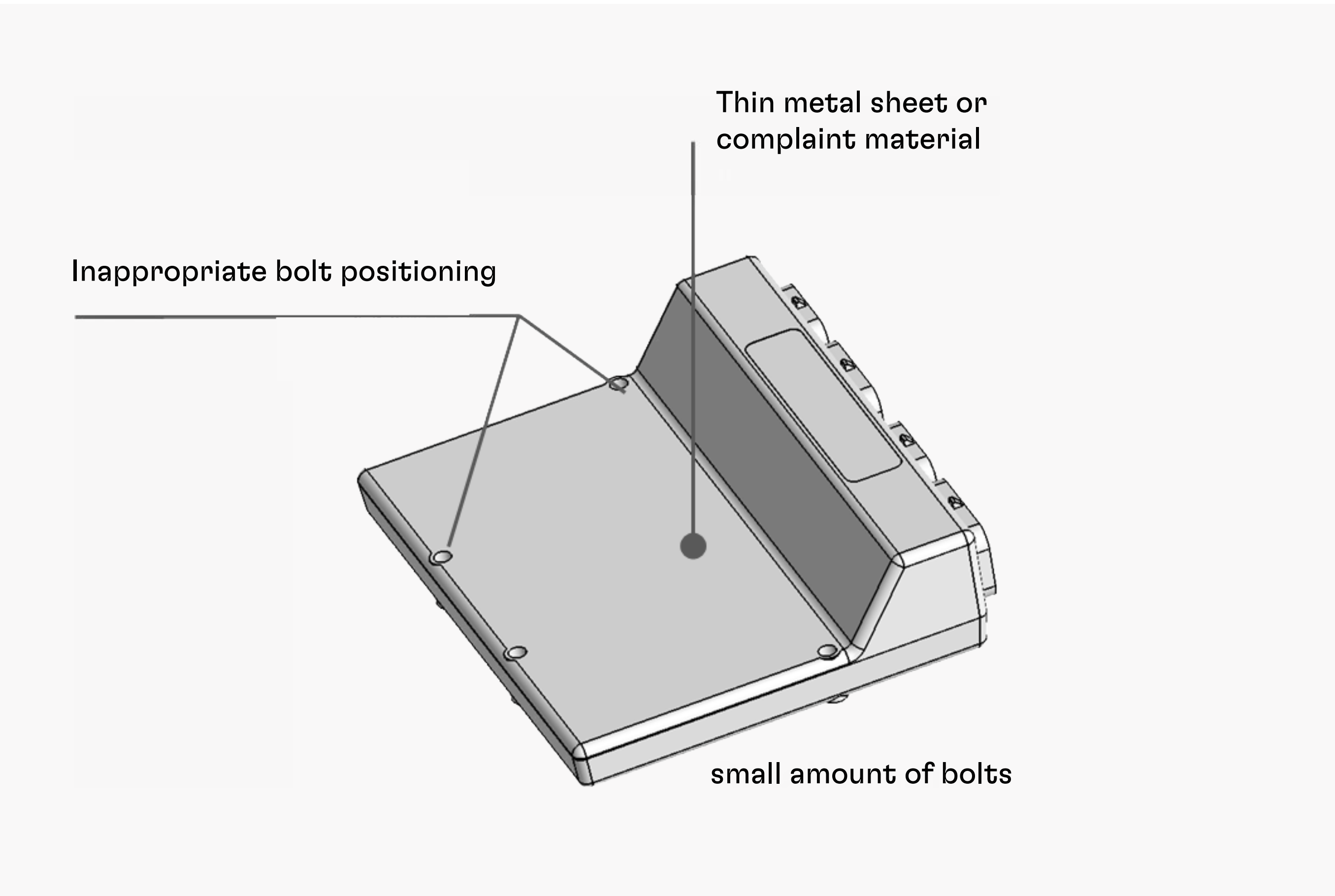


FIGURE 20: TYPICAL APPLICATIONS

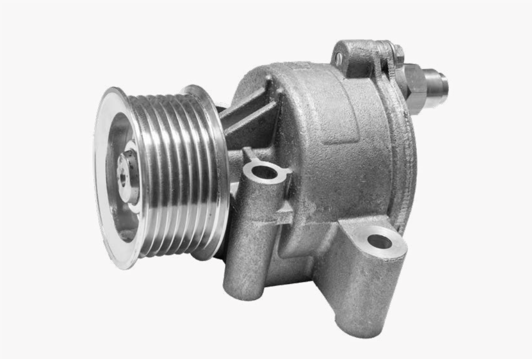
Electronic Control Unit

Actuators

Inverter/Converter

Water Pump

Oil Pump



Vacuum Pump

Steering Pump

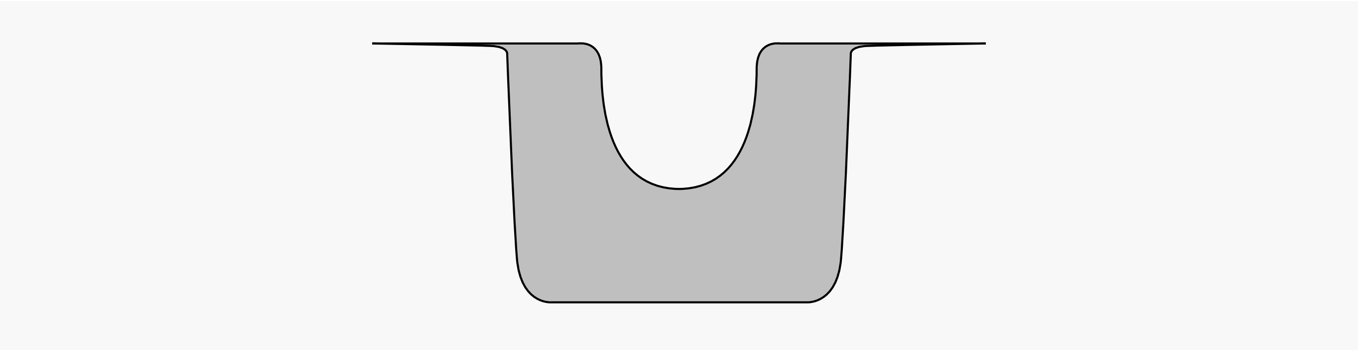
As parts are essentially thin, limited available space may hinder the application of the chamfer- or chamfer-step design proposed in Section 4, which require larger flange widths to form sufficient bondline thickness. To overcome basic limitations of a pure groove design, an interlocking design option has been developed.

7.2 DESIGN CONCEPT

The design comprises of an appropriately shaped profile projection of one mating surface, named a tongue, inserted into a corresponding groove formed into the opposite flange surface, into which the liquid sealant has been dispensed. After assembly, the liquid sealant cures and forms a gasket adhering to both mating surfaces.

The design does not intend to secure the flanges against in-plane micromovement but to ensure good adhesion of the sealant to both parts involved. Also, it transforms critical out-of-plane micromovement, usually producing tensile stress and interface separation, to more beneficial shear stresses.

FIGURE 21: GASKET ELEMENT (TONGUE) INSERTED INTO GROOVE FILLED WITH LIQUID SEALANT.



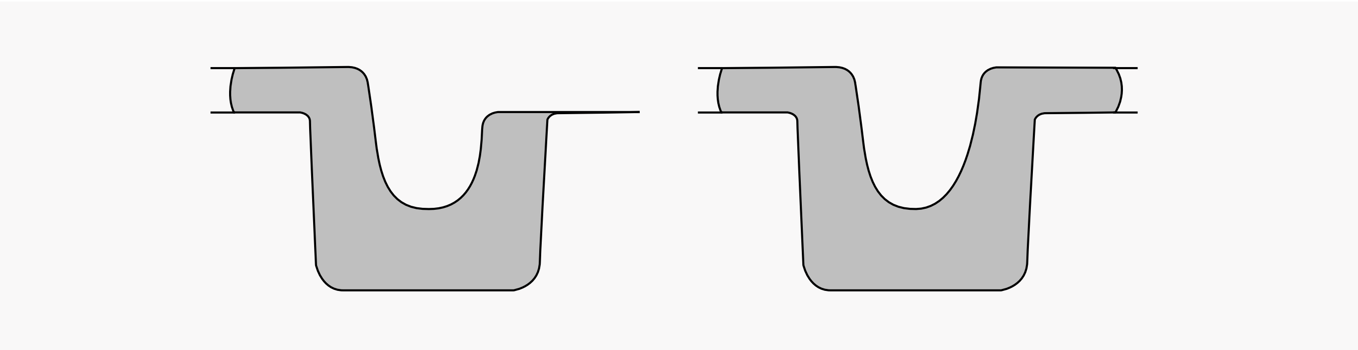
To avoid condensation residues at the interfaces and to support sealant cure by moisture access, a small one- or two-sided gap between the mating surfaces of about 0.5 mm filled with sealant is recommended (Figure 22).

The gap is filled along small length by the liquid sealant escaping from the groove, which it has been dispensed into, during the assembly process.

A smaller gap or a zero-gap designed on the side of the differential pressure or medium may be required to withstand instant seal test.

FIGURE 22: GASKET OPTIONS WITH MINIMUM THICKNESS SEAL GAP FORMED BY PRODUCT EXCEEDING THE GROOVE DURING ASSEMBLY. GAP IS MAINTAINED BY APPROPRIATE SPACERS SUCH AS SURFACE STEPS OR RAISED BOSSES ON WHICH COVER RESTS TO PROVIDE CLEARANCE.

Gasket options with minimum thickness seal gap formed by product exceeding the groove during assembly. Gap is maintained by appropriate spacers such as surface steps or raised bosses on which cover rests to provide clearance.



LOCTITE

7.3 RELEVANT SEALANT TECHNOLOGIES

Typically, adhesives and sealant materials are recommended that can accommodate relative movement of several 1/10 mm between the part surfaces. They comprise of RTV silicones, silane-modified polymers, or high-performance polyurethane-based materials.

RTV silicones are typically 1-component products that don't require mixing. Curing starts immediately at room temperature after the product is exposed to the atmosphere. They show high temperature performance and very good resistance to automotive oils and can withstand on-line, low-pressure tests carried out before product fully cures.

Silane-modified polymers show excellent adhesion ability to variety of substrate materials. To ensure proper in-depth curing, 2-component systems are required.

PU-based sealants provide good stiffness and structural performance. They are recommended for engineering plastics, especially for grades with low surface energy. The use of 2-component systems ensures sufficient cure speed and depth.

Cure may be affected by the level of moisture and temperature. Higher temperatures accelerate the cure speed and support adhesion build-up.

TABLE 1: FLEXIBLE LIQUID SEAL GRADES

Product class	Key advantages	Typical product	Product type/ components	Operating temperature range
RTV silicones	High temperature performance, excellent resistance to automotive oils. Requires no mixing	SI 5970	1-component, requires no mixing	-40°C to 150°C
Silane-modified polymers	Excellent adhesion ability to variety of substrate materials. Solvent-, isocyanate- and silicon-free	MS 647	2-component (A: MS 647, B: MS 9371B) Mixing ratio 10:1	-40°C to 100°C
Polyurethane-based sealants	Stiffness, structural performance Excellent adhesion ability to various kinds of plastics, specifically low surface energy types such as PP Talc, PP LGF, PC/ABS, PC/PET	UK 2073	2-component (A: UK 2073 (polyol), B: UK 2173 (isocyanate)) Mixing ratio 2:1	-40°C to 80°C

7.4 DESIGN RECOMMENDATIONS

The design recommendations following are based upon numerical analysis and experimental testing. As the design parameters are complex, the recommendations are of general nature and valid as sound proposal at initial design phase.

The material behaviour of elastomeric materials practically enclosed in a cavity is manifold and defies straightforward treatment. Any numerical analysis for sealants cured in very small grooves suffers from strong idealization and from lack of practical observation of damage onset and growth.

A better understanding of the basic material behaviour and its limits as well as of the effects of the basic design parameters is essential for successfully designing this type of gasket solutions.

7.4.1 TYPICAL LOADING CONDITIONS

Given the complex geometry and the variety of applications to address, it is difficult to establish generally valid operational conditions. The micromovement that may typically occur is of several 1/10 mm, but the loads and hence stresses which are likely to be encountered in practice depend on the stiffness of the parts and the position of the mechanical fasteners involved.

Insufficiently constrained, compliant covers may lead to some peel at the seal position. Appropriate design with ribs and local stiffeners and the use of smart materials such as fiber-reinforced plastics usually counteract to a large degree to asymmetric deformation at the seal area.

Neglecting deformation gradients, the stresses and strains occurring in a typical gasket may be derived by imposing micromovement between the mating parts directly at the seal positions (Figures 23, 24).

Typically, the 1st principal strain is suggested to represent material straining and probability of failure for rubber-like materials (compare section 7.5). In addition, hyperelastic soft materials are susceptible to micro cracking or growth of flaws under tensile hydrostatic loading. The hydrostatic pressure is the second parameter used to assess material loading and to identify potential failure areas or conditions.

When a typical seal is subjected to tangential micromovement of 0.5 mm in circumferential direction, the strains that occur are, by disregarding singular values at small corners, substantially low (Figure 23). The sealants recommended for use have been proved in lap shear testing to withstand shear strains larger than 200% even in thin bond layers (Table 4, Section 7.9). No noteworthy hydrostatic tension arises.

FIGURE 22: LOADING CONDITIONS WHEN 0.5 MM RELATIVE MICROMOVEMENT IS IMPOSED BETWEEN THE MATING SURFACES IN TYPICAL SEAL GEOMETRY IN TANGENTIAL DIRECTION (SI 5970 AT ROOM TEMPERATURE)

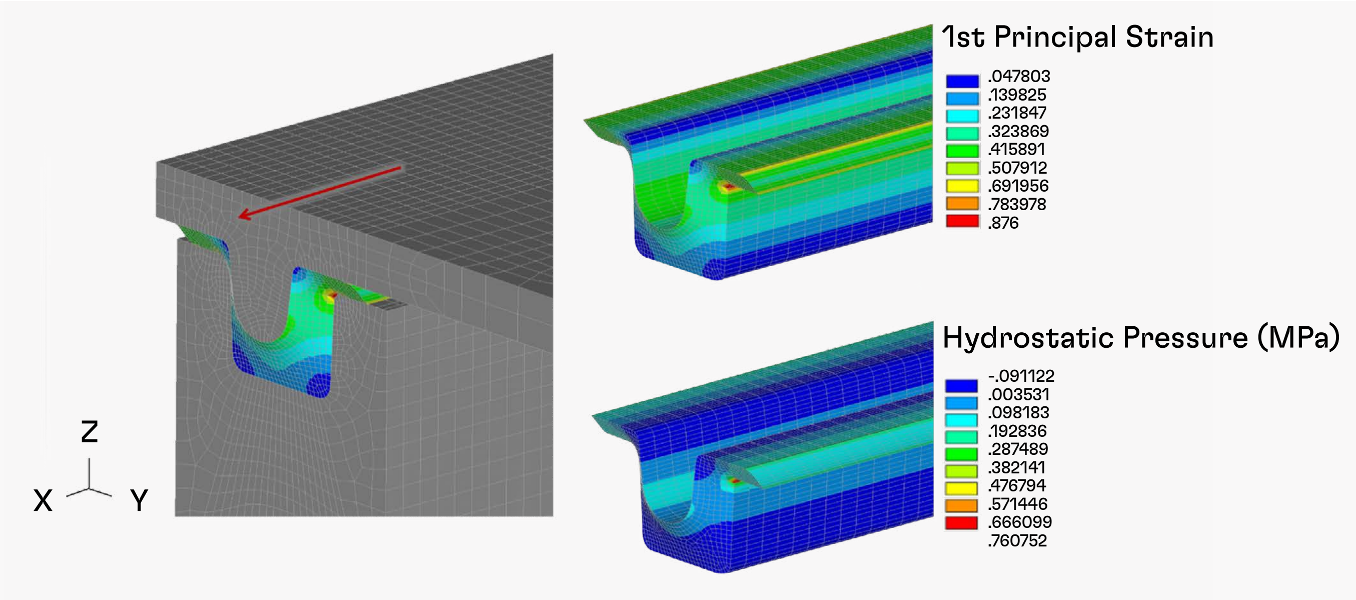
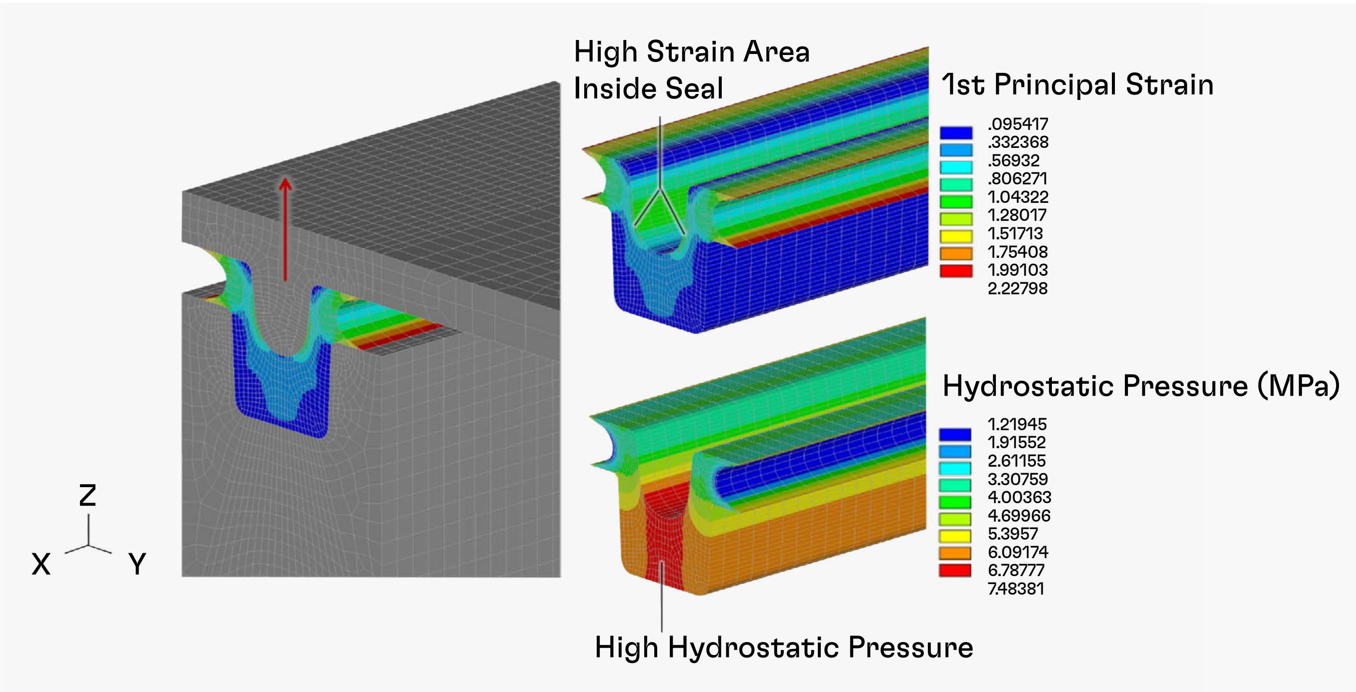


FIGURE 23: LOADING CONDITIONS WHEN 0.5 MM RELATIVE MICROMOVEMENT IS IMPOSED BETWEEN THE MATING SURFACES IN TYPICAL SEAL GEOMETRY IN DIRECTION PERPENDICULAR TO THE INTERFACE (SI 5970 AT ROOM TEMPERATURE)

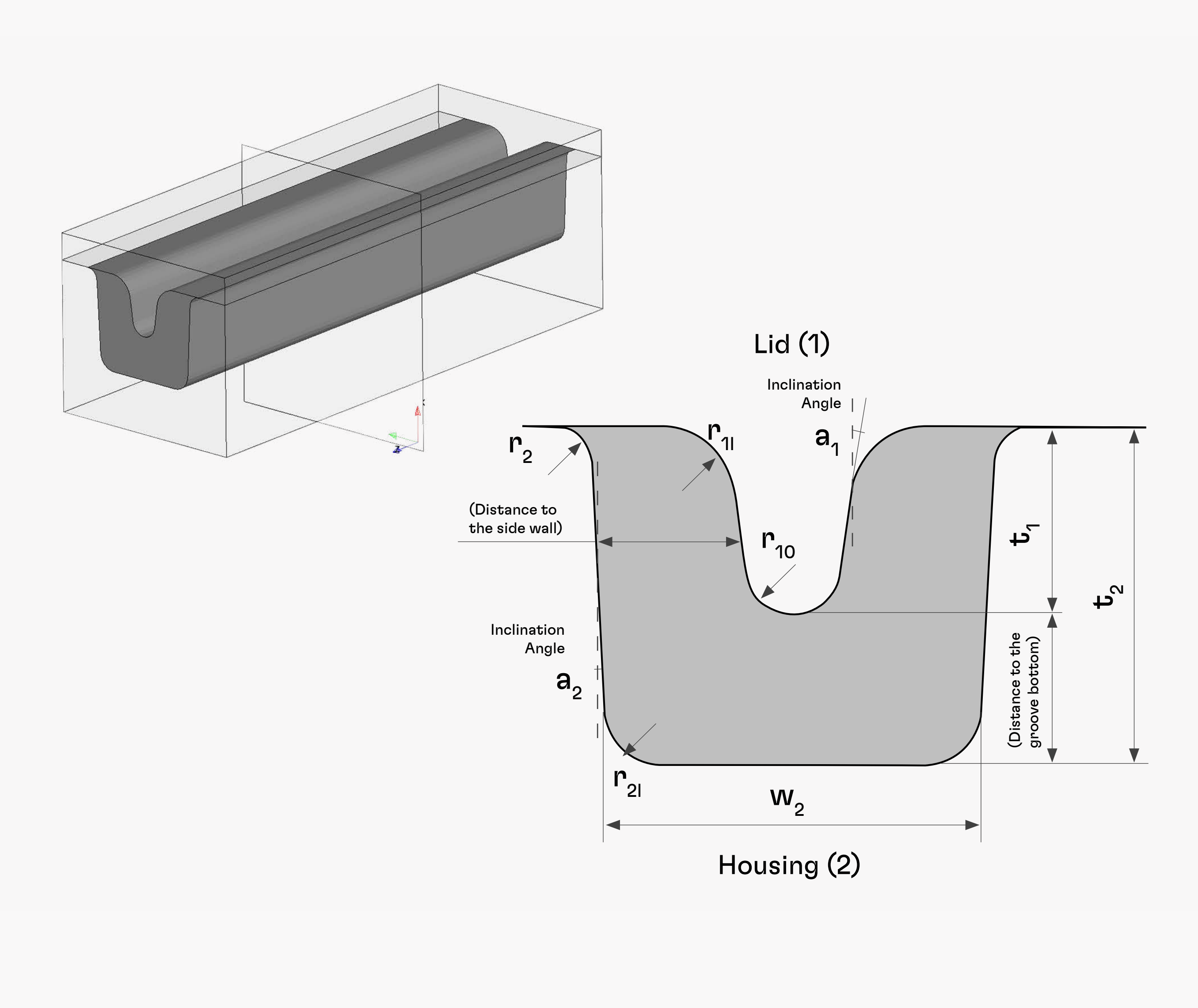


corners of the small gap, considerable material strain can be observed at both sides of the tongue inside the main seal area. In addition, significant hydrostatic pressure results underneath the tongue throughout the main seal cross section (Figure 23).

Micromovement perpendicular to the seal interface, as may have been expected in standard bonded joint design, appears thus critical and was therefore used in order to derive basic design rules.

7.4.2 BASIC DESIGN PARAMETERS

FIGURE 24: BASIC DESIGN PARAMETERS



The main design parameters affecting the loading conditions and thus the seal performance are the length of the tongue t1 corresponding to the immersion depth, the radius r10 at its tip, the width of the groove denoted w2, and its depth denoted t2. The base radii r1l and r2l are considered of minor importance but shall be sufficient to facilitate liquid product displacement and proper wetting during assembly. The rounding radius of the groove r2u is mostly given, determined by manufacturing constraints.

Parameter variation by means of finite element analysis supported the understanding of the underlying mechanisms and parameter interaction. Details of the numerical approach may be found in Section 7.5.

To ensure computational convergence and numerical stability when modelling a standard design with a seal which is practically fully confined (as illustrated in Figure 24), assumptions were made that included an initial separation of the sealant at the interface to the lid in the vicinity of the corners or throughout the entire top area (compare Section 7.5.3). Remaining strain or stress peaks at small corners were considered as attributed to stress singularities and not causal for the failure of the whole seal. The data evaluated for comparison and design integrity assessment are

a) the maximum 1st principal strain and
b) the maximum hydrostatic pressure (tension) within the main seal area. In addition
c) the hydrostatic pressure at the bottom of the groove was also determined.

In direct comparison to (b), the later parameter describes a possible reduction of the hydrostatic pressure along the mid-section of the seal.

The results are derived using a considerably fine mesh with an element length of 0.05 mm without further investigation of mesh density effects. The hydrostatic pressure, developed underneath the tongue is suggested to be less affected by the mesh density than the maximum principal strain probably does.

The data evaluated shall be seen as indication for the failure conditions and used for comparison purposes. Failure limits in FE analysis of bonded joints are often applied using a certain characteristic length over which stresses or strains have been averaged or a corresponding volume. This usually helps to account for stress or stress gradients. For the purposes of the present analysis, strains and stresses were averaged along the element area, the elements for the seal material having a typical length of 0.05 mm. This element size is far smaller than the characteristic lengths or areas typically used to assess the failure probability of adhesive joints, and it can therefore be assumed that the derived results lie well on the safe side. In addition, strain limits from experimental work are nominal values and do not account for stress or strain peaks arising at the overlap ends.

The strain and hydrostatic stress results cannot be therefore directly used to predict material failure without further data treatment. Any prediction based on that would probably lead to significant underestimation of the operational ability.

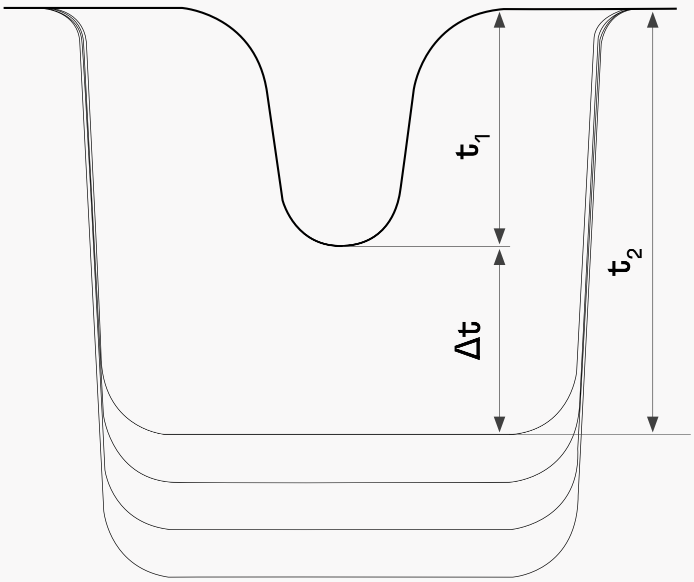
The results and the conclusions drawn from the parameter variation are presented below. The are derived by FE analysis using material data evaluated for a typical RTV silicon grade (SI 5970).

GROOVE DEPTH

Increasing the depth of the groove was found to lead to significant decrease of hydrostatic tension that occurs underneath the tongue when the seal is subjected to micromovement perpendicular to the mating interface. Decrease in strain is also considerable but less pronounced (Figure 25). Increasing the distance between the tongue and the bottom of the grove leads in addition to a drop in strain occurring at the outer corners of the sealant top interface which is however excluded from closer consideration.

FIGURE 25: EFFECT ON STRAIN AND HYDROSTATIC PRESSURE BY INCREASING THE DEPTH OF THE GROOVE AND SO THE DISTANCE BETWEEN THE TONGUE AND THE BOTTOM OF THE GROOVE

t_1	mm	1.0			
t_2	mm	1.8	2.0	2.2	2.4
Δt	mm	0.8	1.0	1.2	1.4
r_{10}	mm	0.25			
W_2	mm	2.0			
r_{1l}	mm	0.4			
r_{2l}	mm	0.3			
r_{2u}	mm	0.15			
a_1	°	8			
a_2	°	3			



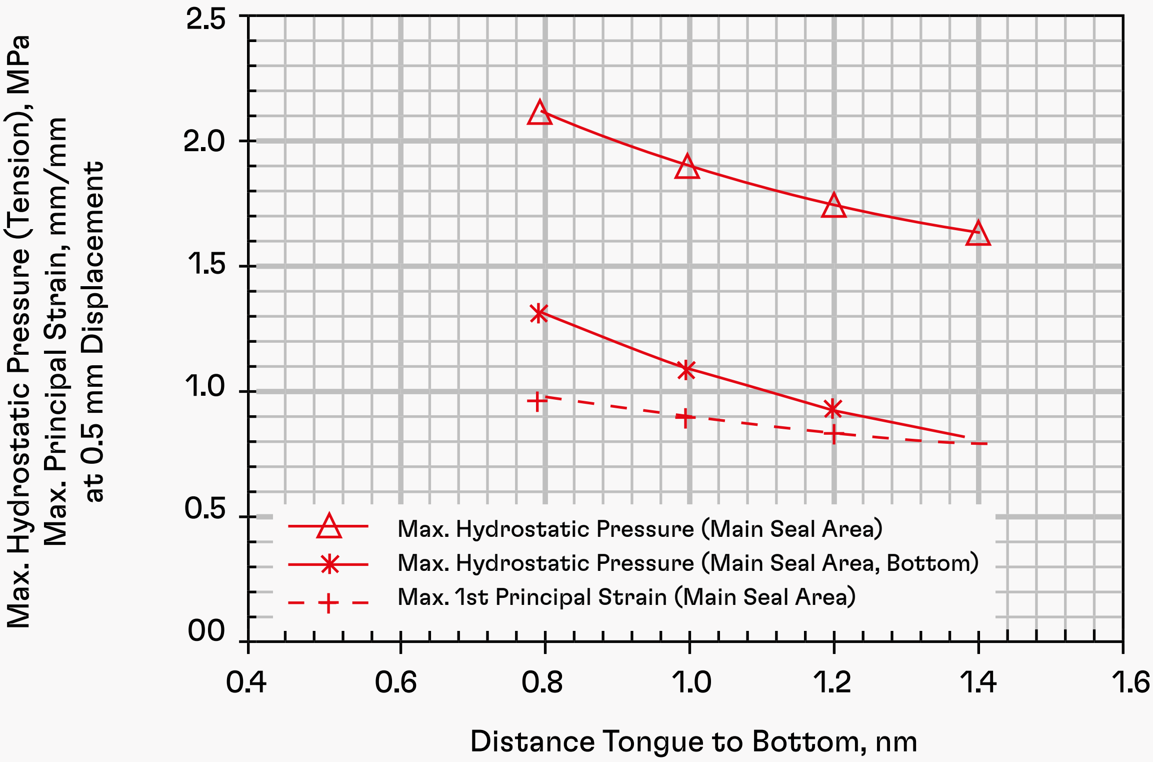
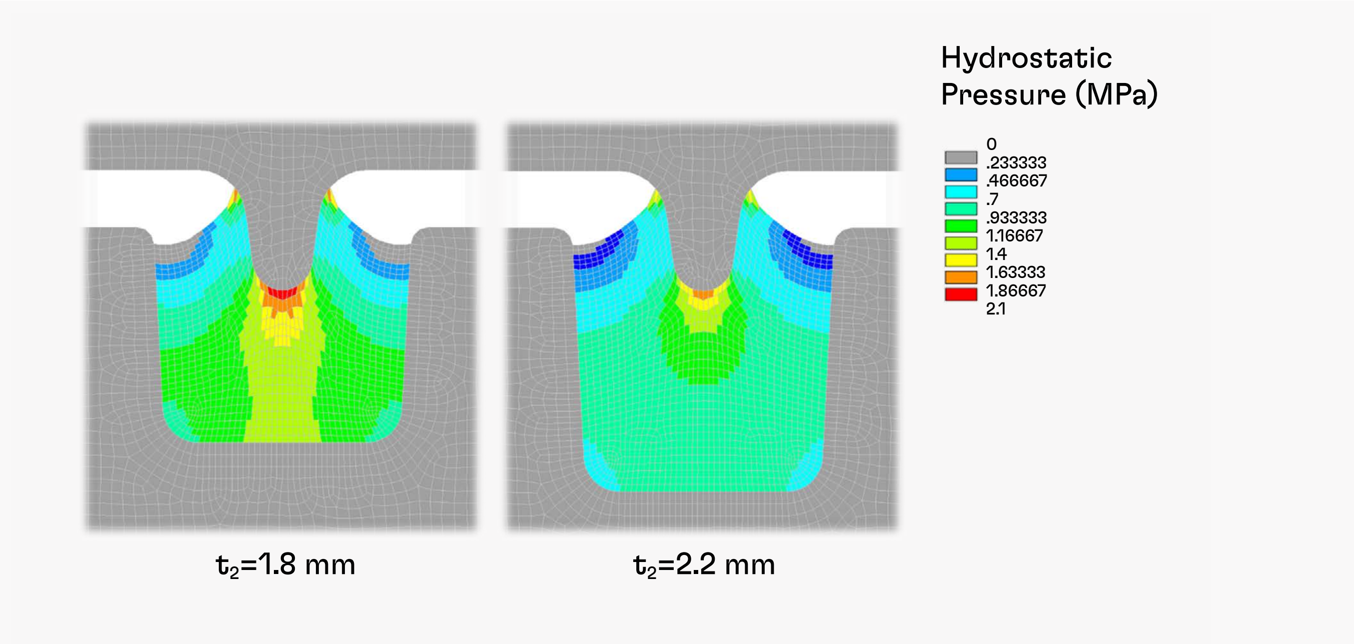


FIGURE 26: VARIATION OF THE DEPTH OF THE GROOVE: COMPARISON OF HYDROSTATIC PRESSURE OCCURRING WHEN 0.5 MM DISPLACEMENT IS IMPOSED TO THE UPPER PART. INCREASING THE DEPTH OF THE GROOVE LEADS TO CONSIDERABLE HYDROSTATIC TENSION REDUCTION.

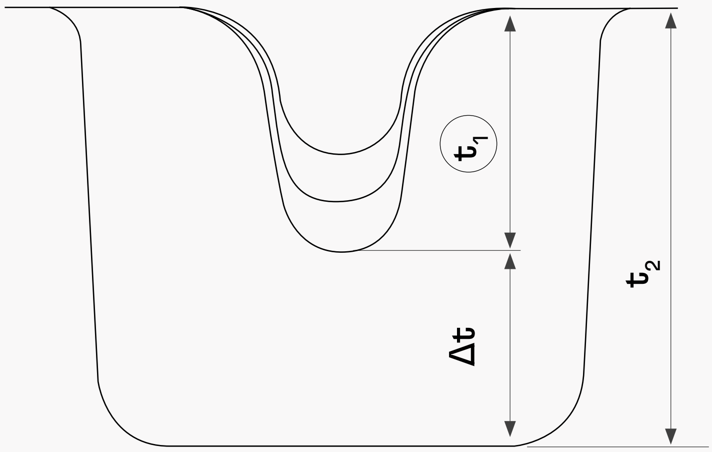


LENGTH OF THE TONGUE

While increasing the distance (gap) between the tongue and the bottom of the groove by decreasing the length of the tongue leads to comparable significant reduction of the hydrostatic tension as by increasing the depth of the groove, the impact on critical strain appears less pronounced. The increase of the gap relaxes straining the material arising from constrained lateral deformation but obviously increases the notch effects, as the tip of the tongue approaches the upper boundary of the seal.

FIGURE 27: EFFECT ON STRAIN AND HYDROSTATIC PRESSURE BY DECREASING THE LENGTH OF THE TONGUE AND SO INCREASING THE DISTANCE BETWEEN THE TONGUE AND THE BOTTOM OF THE GROOVE

t_1	mm	1.0	0.8	0.6
t_2	mm	1.8		
Δt	mm	0.8	1.0	1.2
r_{10}	mm	0.25		
W_2	mm	2.0		
r_{1l}	mm	0.4		
r_{2l}	mm	0.3		
r_{2u}	mm	0.15		
a_1	°	8		
a_2	°	3		



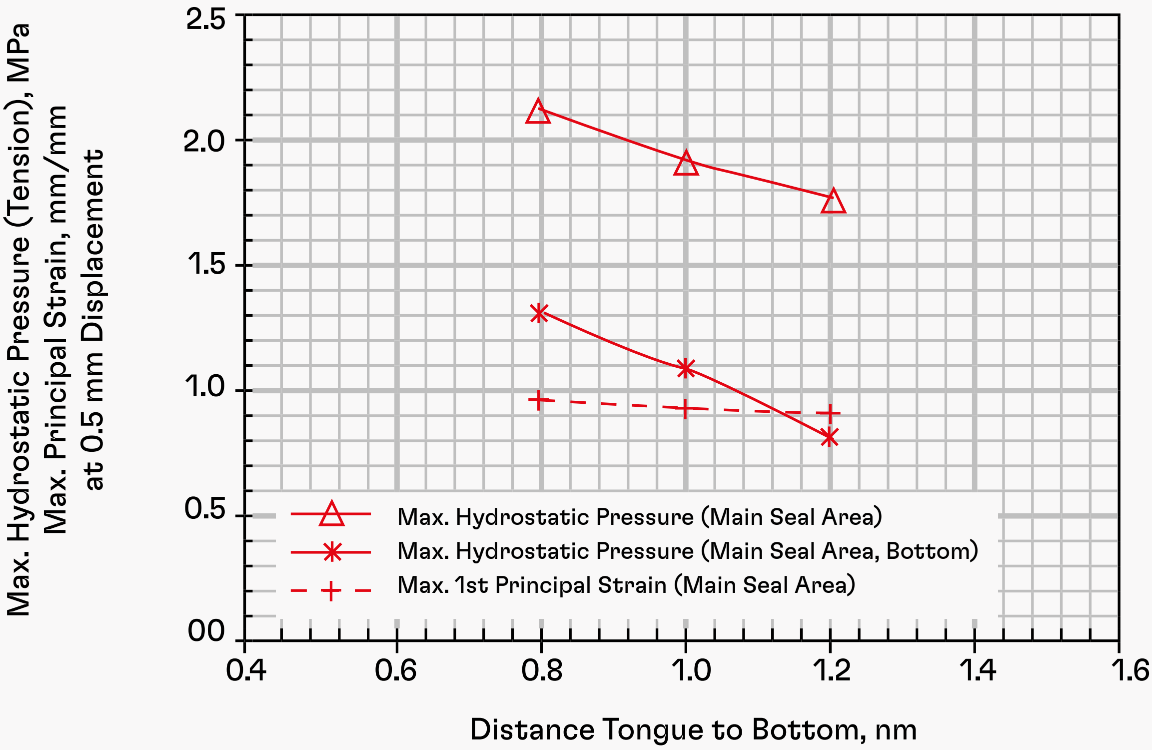


FIGURE 28: VARIATION OF THE LENGTH OF THE TONGUE: COMPARISON OF MAXIMUM PRINCIPAL STRAIN OCCURRING WHEN 0.5 MM DISPLACEMENT IS IMPOSED TO THE UPPER PART. INCREASING THE DISTANCE TO THE BOTTOM BY DECREASING THE LENGTH OF THE TONGUE HAS POSITIVE BUT LESS PRONOUNCED IMPACT ON THE STRAIN CONDITIONS THAN BY INCREASING THE DEPTH OF THE GROOVE.

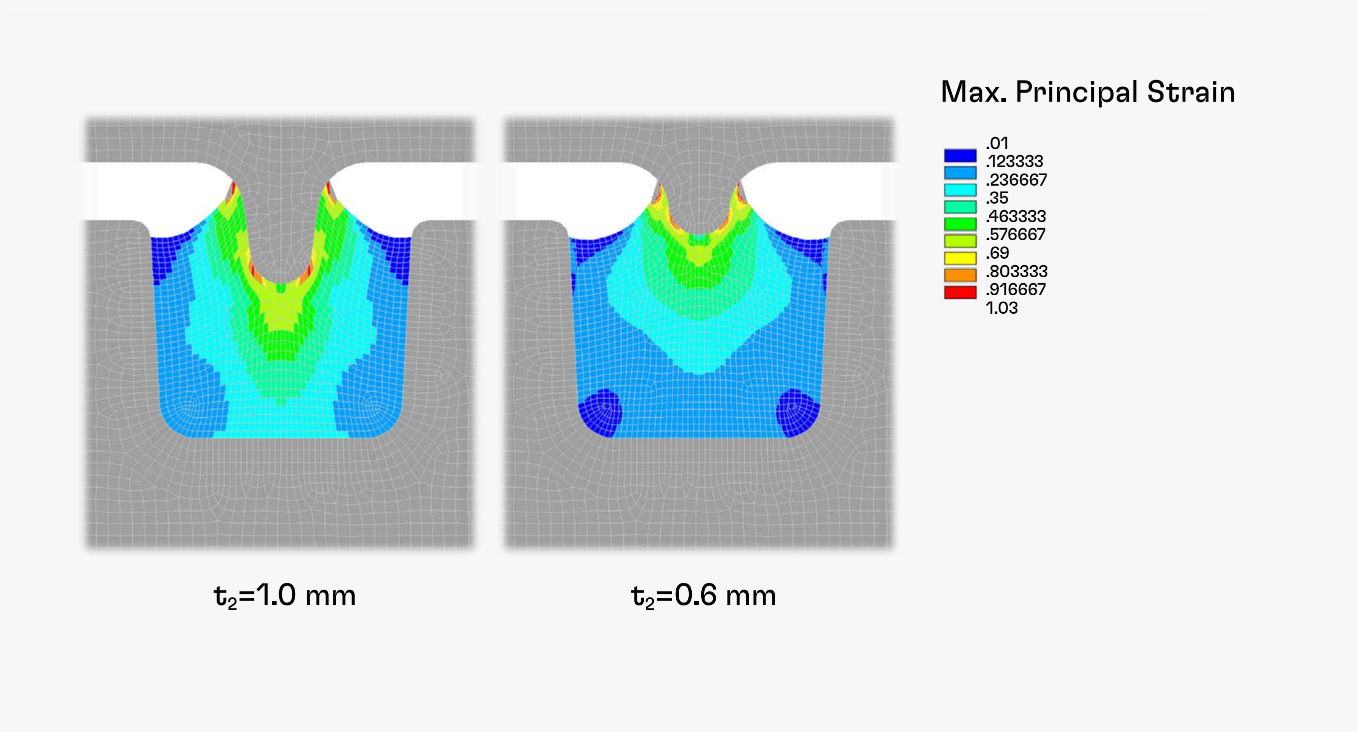
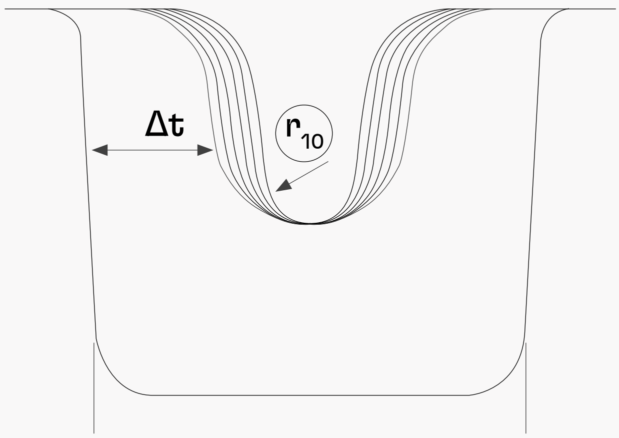


FIGURE 29: EFFECT ON STRAIN AND HYDROSTATIC PRESSURE BY DECREASING THE RADIUS OF THE TONGUE AND THUS, INCREASING THE DISTANCE BETWEEN THE TONGUE AND THE SIDE WALLS OF THE GROOVE

The radius at the tip of the tongue practically determines its width, provided the inclination angle is small. When assuming the groove width to remain constant, it also derives the gap between side wall and tongue. The distance to the side wall affects obviously to a major degree the hydrostatic tension generated at the bottom of the groove (Figure 33, 34) but appears to not essentially alters the maximum principal strain or the hydrostatic tension near the tip radius. One may assume that the stress and strain reduction by increased distance to the side wall is being contradicted by notch effects when the tip of the tongue becomes sharper due to smaller radius. The design of a wider radius helps also to relax the strain occurring at the outer corners of the sealant which is excluded here from closer consideration.

t_1	mm	1.0					
t_2	mm	1.8					
r_{10}	mm	0.2	0.25	0.3	0.35	0.4	0.45
W_{h2}	mm	2.0					
Δt	mm	0.8	0.75	0.7	0.65	0.6	0.55
r_{1l}	mm	0.4					
r_{2l}	mm	0.3					
r_{2u}	mm	0.15					
a_1	°	8					
a_2	°	3					



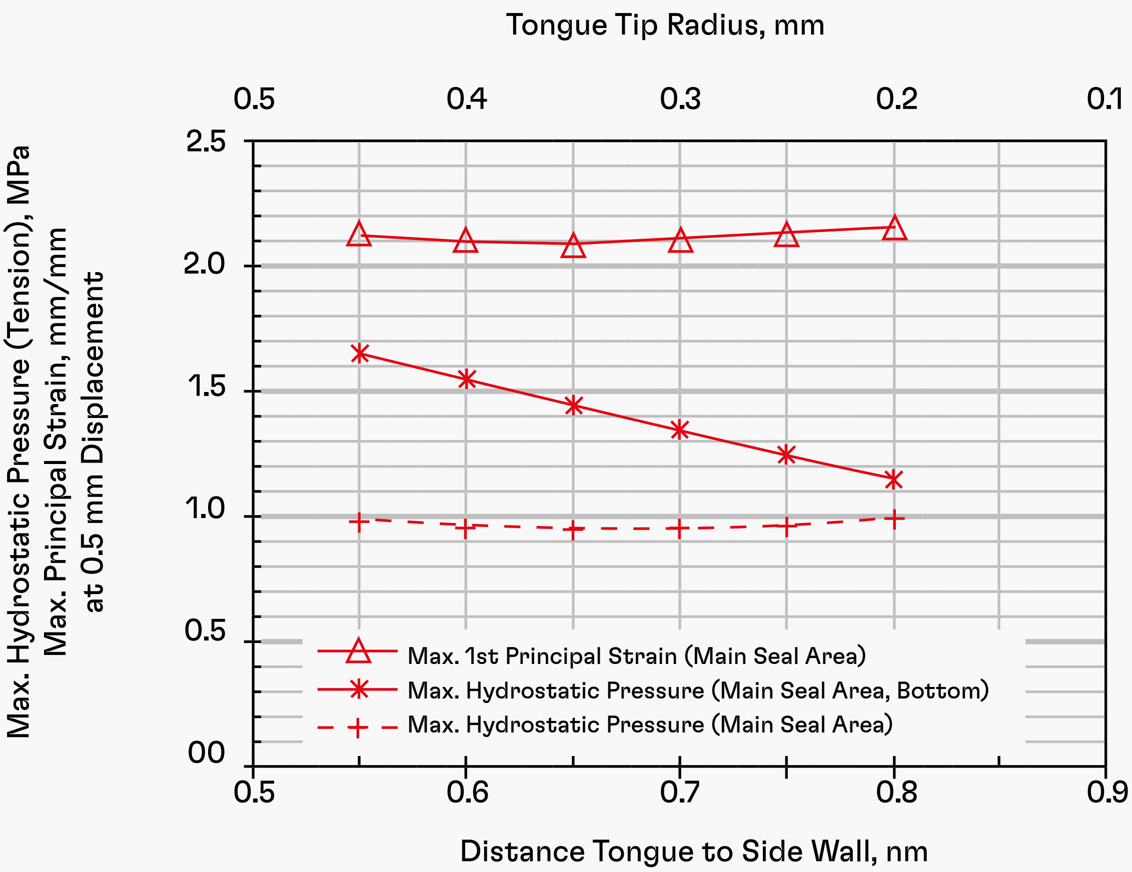
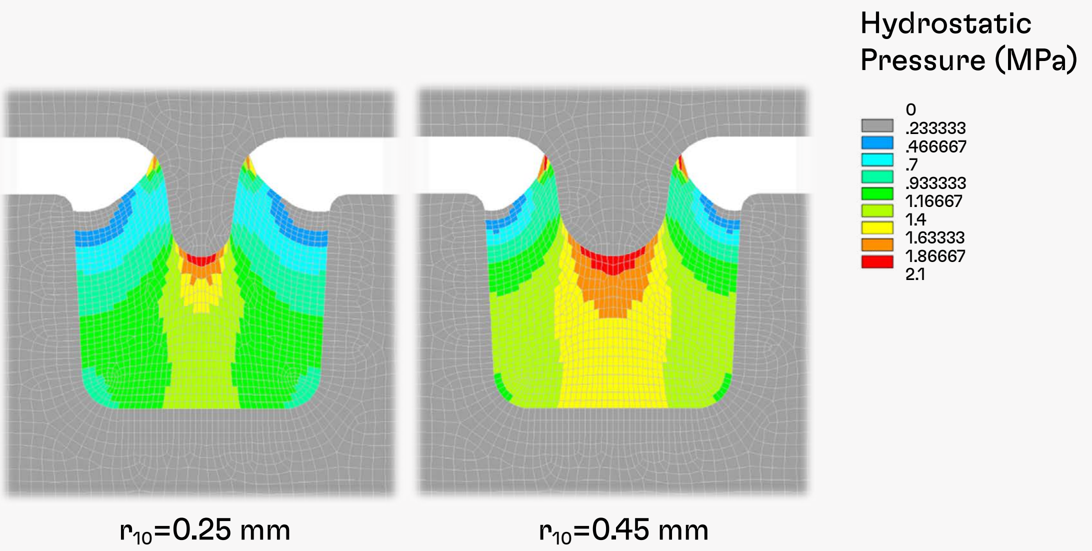


FIGURE 30: VARIATION OF TONGUE TIP RADIUS: COMPARISON OF HYDROSTATIC PRESSURE OCCURRING WHEN 0.5 MM DISPLACEMENT IS IMPOSED TO THE UPPER PART. SMALLER TONGUE TIP RADIUS AND WIDTH RESULT INTO REMARKABLE HYDROSTATIC TENSION REDUCTION UNDERNEATH THE TONGUE.



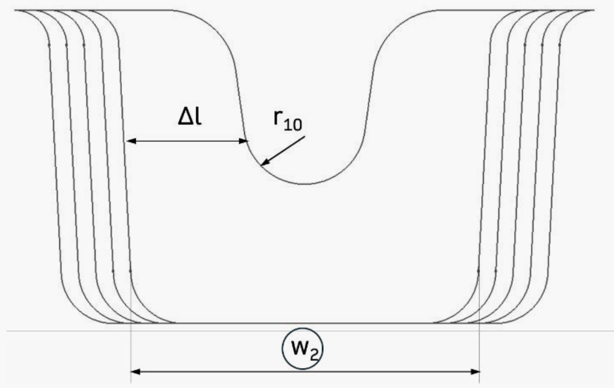
GROOVE WIDTH

Widening the groove while the profile of the tongue remains unaffected was found to result in continuous decrease of the 1st principal strain and the hydrostatic tension underneath the tongue (Figure 31).

However, the strain and stress reduction by increasing the side gap appears significantly less pronounced as it does when the distance between the tongue and the bottom increases (s. Figure 25).

FIGURE 31: EFFECT ON STRAIN AND HYDROSTATIC PRESSURE BY INCREASING THE WIDTH OF THE GROOVE AND THUS, INCREASING THE DISTANCE BETWEEN THE TONGUE AND THE SIDE WALLS.

t_1	mm	1.0				
t_2	mm	1.8				
r_{10}	mm	0.35				
W_{h2}	mm	2.0	2.2	2.4	2.6	2.8
Δt	mm	0.65	0.75	0.95	1.05	1.15
r_{1l}	mm	0.4				
r_{2l}	mm	0.3				
r_{2u}	mm	0.15				
a_1	°	8				
a_2	°	3				



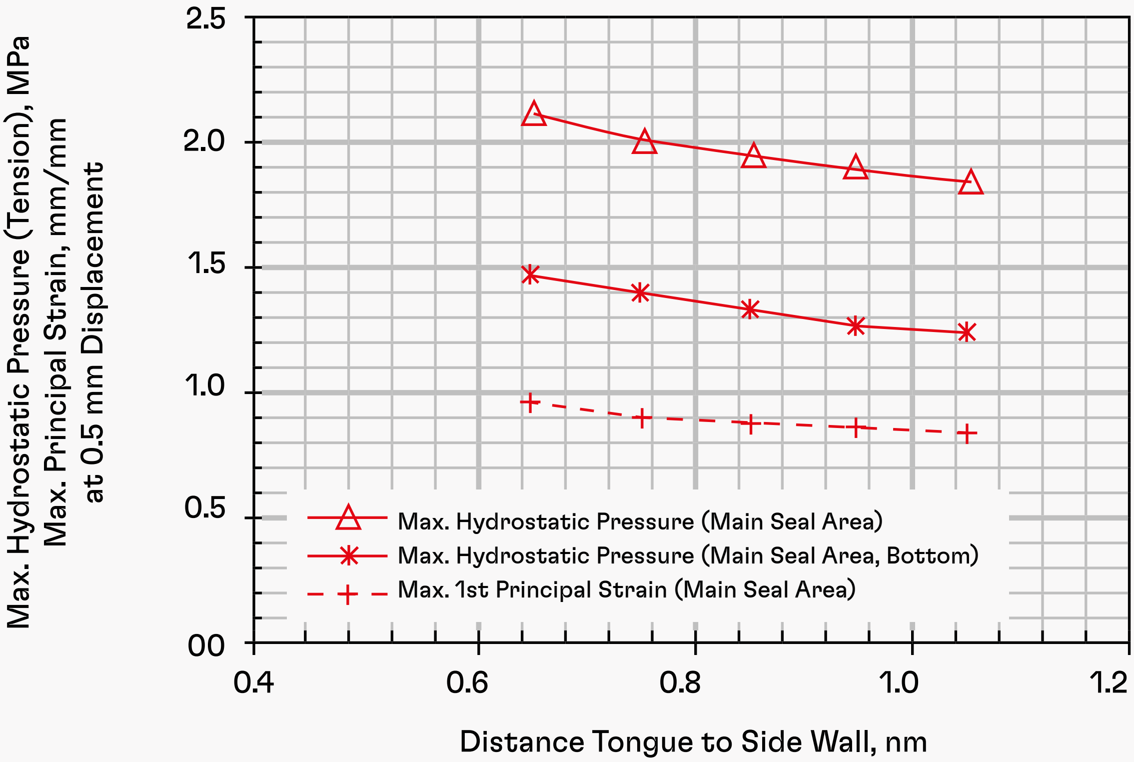
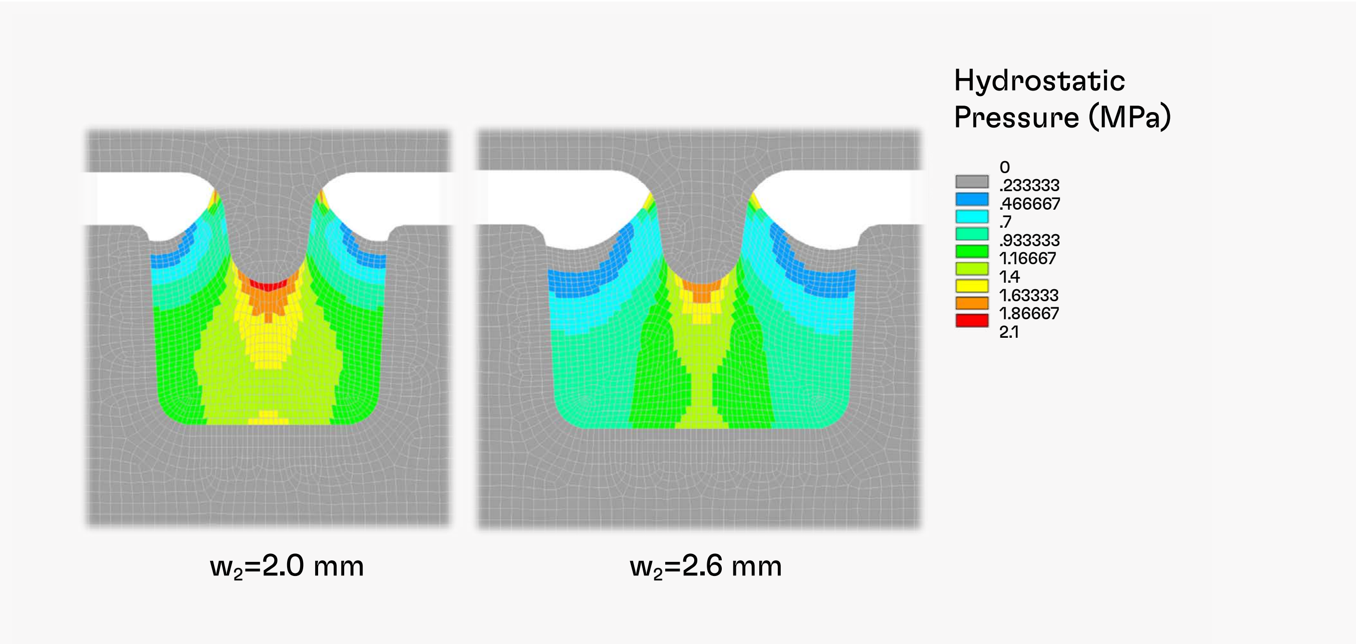


FIGURE 32: VARIATION OF GROOVE WIDTH: COMPARISON OF HYDROSTATIC PRESSURE OCCURRING WHEN 0.5 MM DISPLACEMENT IS IMPOSED TO THE UPPER PART. LARGER GROOVE WIDTH RESULTS INTO REMARKABLE HYDROSTATIC TENSION REDUCTION UNDERNEATH THE TONGUE.



Comparison of the effects achieved by decreasing the tip radius of the tongue and by increasing the width of the groove

When micromovement in normal direction occurs, the side gap between the tongue and the side walls was found to affect the strain resulting in the seal material and the hydrostatic tension underneath the tongue tip. Small distances to the side walls hinder the sealant material to contract when the tongue pulls it out.

The gap between the tongue and the side walls can be increased in two ways: either by reducing the tip radius and thus the tongue width, or by increasing the groove width.

The first option, i.e. increasing the side gap by decreasing the width or tip radius of the tongue, apparently lead to very strong reduction of the hydrostatic pressure at the bottom of the groove: However, 1st principal strain and maximum hydrostatic pressure remain practically unaffected. In addition, any benefit may be limited as the tongue becomes very narrow and susceptible to deformation or difficult to manufacture.

The second option, i.e. increasing the groove width, obviously provides more effective reduction of strain and hydrostatic pressure (Figure 33) although it is certainly more expensive in terms of design space. Widening the groove leads in contrast to narrowing the tongue to sustainable decrease of all loading parameters.

FIGURE 33: COMPARISON OF LOADING CONDITIONS ARISING FROM INCREASING THE GAP TO THE SIDE WALLS BY INCREASING THE WIDTH OF THE GROOVE (RED CHART POINTS) OR DECREASING THE WIDTH OF THE TONGUE (BLACK CHART POINTS)

THE IMPACT OF THE BASIC DESIGN PARAMETERS TO STRAIN AND STRESS CONDITIONS IN A TYPICAL SEAL IS SUMMARIZED IN TABLE 2.

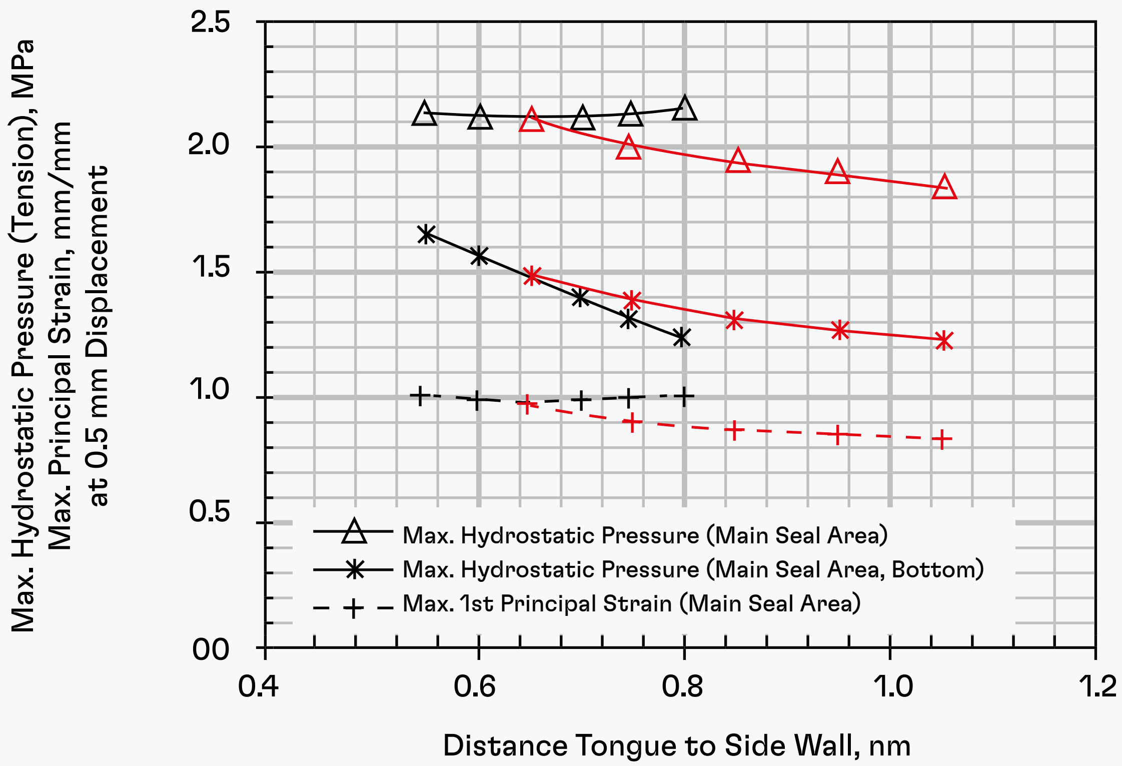


TABLE 2: OVERVIEW OF THE EFFECTS OF BASIC DESIGN PARAMETERS ON THE LOADING CONDITIONS OF THE SEAL MATERIAL WHEN THE GASKET IS SUBJECTED TO MICROMOVEMENT IN NORMAL DIRECTION. POSITIVE IMPACT IS HIGHLIGHTED WITH PLUS SIGNS (+), WHILE INDIFFERENT ONE WITH O.

Design Measure	Decrease Tip Radius/ Tongue Width	Increase Groove Width	Increase Groove Depth	Decrease Tongue Length
Impact on	Increase Gap to Side Walls		Increase Distance to Bottom	
Max. Principal Strain	0	+	+	0
Max. Hydrostatic Tension	++	+	++	++
Max. Hydrostatic Tension at Bottom	0	+	++	++

LOCTITE

7.4.3 GENERAL RECOMMENDATIONS

The side gap provided by the difference between the width of the groove and the width of the tongue, the later suggested simplified to twice its tip radius, shall be sufficient to accommodate shear strain generated by micromovement.

If micromovement occurs in the circumferential direction, we may approximately request

$$\frac{\Delta u_t}{\left(\frac{w_2}{2} - r_{10}\right)} \leq f \cdot \gamma_{nom,limit}$$

and in similar manner

$$\frac{\Delta u_t}{(t_2 - t_1)} \leq f \cdot \gamma_{nom,limit}$$

where Δut denotes the actual tangential displacement, f is a safety factor, and Ynom,limit the nominal shear strain at failure observed in shear testing. Failure limits for representative products are available from lap shear testing 1 mm thick bonded joints under various temperatures and operational conditions and listed in Table 4 in the Appendix.

A safety factor of 1,5 appears sufficient, given the fact that the shear strain limit from experimental work is recorded as nominal one and does not account for strain peaks occurring at overlap ends or adhesive layer corners.

Micromovement in normal (perpendicular) direction results in more complex material loading conditions, to which both shear (to the side wall) and tensile deformation (to the bottom of the groove) contribute. As the sealant material adheres to the side walls and to the bottom of the groove when the tongue is then pulled apart, constrained lateral deformation leads to hydrostatic tension arising underneath the tongue tip. Strains and stresses can be estimated in sufficient accuracy by means of appropriate FE analysis.

The width of the groove significantly affects the stress conditions in the seal material in loading in normal direction. A minimum width of 2.5 mm is recommended.

The groove width shall be large enough to ensure sufficient gap between the tongue and the side walls of the groove when talking into account part tolerances. A minimum gap of 0.6 mm shall be maintained.

The length of the tongue as inserted into the groove shall clearly exceed the expected micromovement in out-of-plane direction. This is necessary to ensure effective barrier against blow out events.

The distance between the tip of the tongue and the bottom of the groove is a crucial parameter. It directly affects the hydrostatic tension that occurs within the seal material underneath the tongue when the latter is pulled apart, and thus, the probability of damage and microcracking. The distance derives the required groove depth, once the tongue length has been determined. A minimum distance of 1 mm is recommended. Whenever possible with respect to the required length of the tongue, the gap to the bottom shall be chosen even larger than this value.

The tip radius of the tongue recommended is 0.3 mm. A larger radius may decrease the gap to the side wall of the groove in essential manner and thus, may result in higher hydrostatic tension under tensile loading. A much smaller radius however may lead to notch effects and contradict so stress and strain reduction. As the radius is associated with the width of the tongue, provision shall me made to avoid plastic deformation or rupture when its thickness significantly decreases.

Inclination angles are driven by the manufacturing process and constraints for the parts involved. They shall not lead to reduction of the gap available to accommodate shear stresses and strains.

A large base radius at the end of the tongue may also facilitate wetting and smooth product displacement during assembly. A radius of 0.4 mm appears sufficient.

Use larger wall inclination angle for the tongue than for the groove to easy liquid sealant displacement during assembling and ensure good wetting of the tongue profile without air cavities in the case of higher viscosity sealant. An angle of 8°-10° is recommended.

Avoid sharp edges of the groove at the interface of the mating surface. A radius of 0.2 mm minimum is recommended. A small 30° chamfer is preferred over a standard 45° one.

To avoid condensation residues at the interfaces and support sealant curing by moisture access, a small one- or two-sided gap between the mating surfaces of about 0.5 mm is recommended (Figure 22). A smaller gap or zero-gap to the side of the differential pressure or medium is required to withstand instant seal test. To ensure full wetting the tongue throughout its entire length, the condition must be fulfilled

$$l_0 = \frac{1}{t_0} \cdot \left[r_{10}^2 \cdot \left(\frac{\pi}{4} - 1 \right) + (t_2 - t_0) \cdot r_{10} \right] - \frac{1}{2} \cdot (w_2 - 2 \cdot r_{10}) \geq 0$$

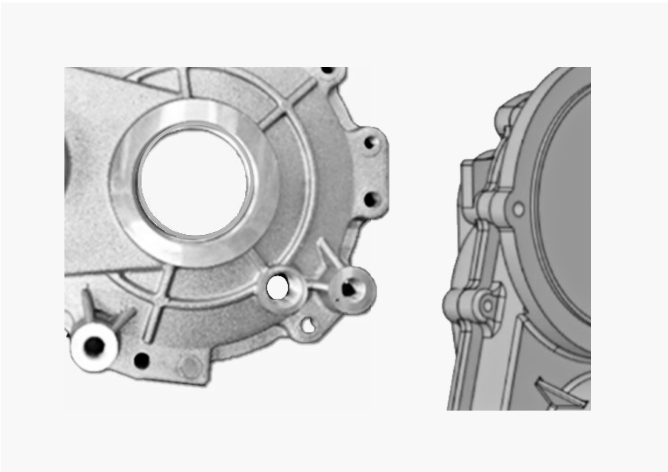
l_0 denotes the length of a two-sided gap of thickness h_0 along which product excess is forming a bond. The equation is an approximation within ±20% accuracy by neglecting inclination angles and smaller radii. $2 \cdot r_{10}$ corresponds approximately to the width of the tongue, provided that the inclination angle is small.

7.4.4 FLANGE DESIGN

Low material thickness and large bolt spaces can lead to non-symmetric deformation involving peel stresses. This may be more pronounced when internal pressure is applied. To avoid peel stresses and gross deformation gradients, appropriate design of the parts to be sealed is required.
Thicker raised edges or rim running all around the outer boundary of the lid and in general perimeter reinforcement prevent lid warping at edges due to internal pressure or when bolts are overtightened.

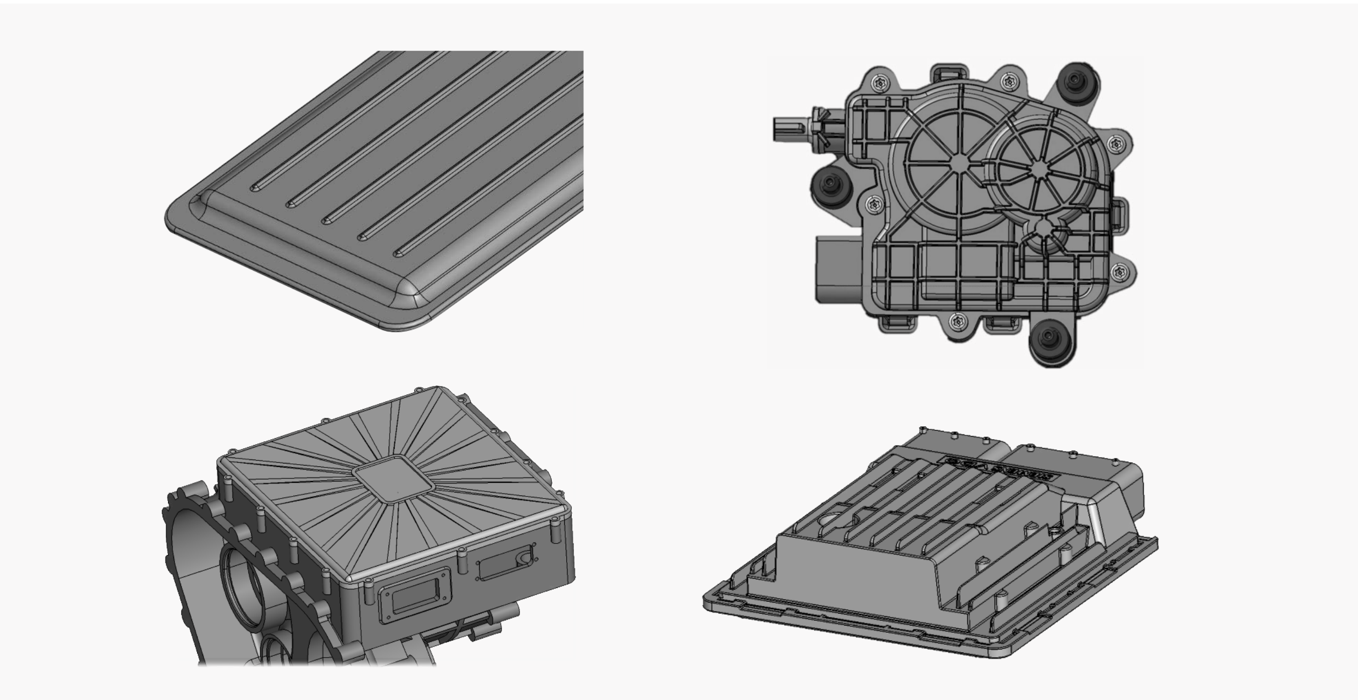
As the bolts may be often eccentrically positioned to the seal, increase material thickness around screw holes, if necessary, with additional ribs or reinforcing stiffeners (Figure 34).

FIGURE 34: STIFFNESS REINFORCEMENT AROUND SCREW HOLES



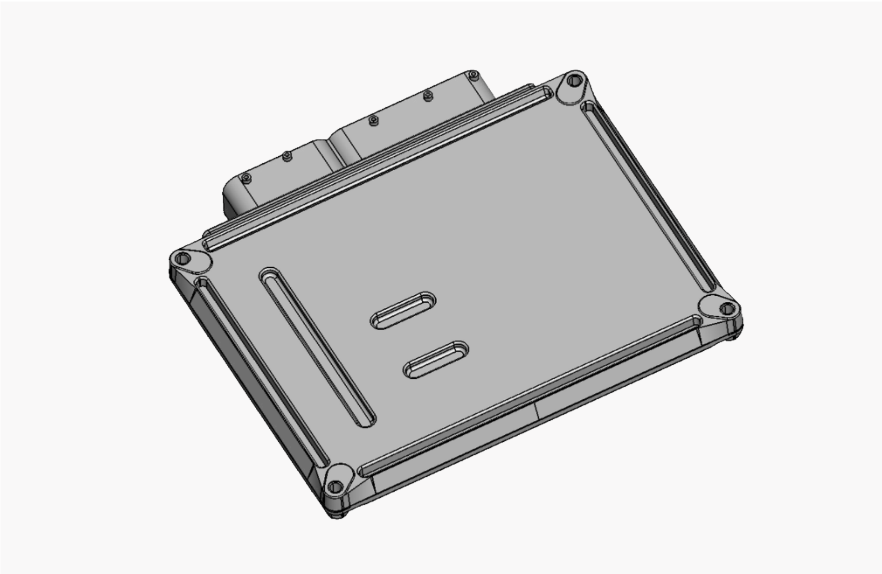
Use ribs running across the lid surface, connecting the perimeter edges, forming a grid or cross pattern to increase bending stiffness and limit flexing (Figure 35).

FIGURE 35: EXAMPLES OF RIB DESIGN FORMING GRID OR CROSS PATTERNS



In metal parts, add sickens (Figure 36) as subtle stiffening features to reduce flex without adding bulk.

FIGURE 36: TYPICAL SICKENS STIFFENING A METAL COVER



7.4.5 SURFACE CONDITIONS

Both assembly parts must be free from oil, grease, moisture, dirt and release agents. From the Henkel portfolio, for example, TEROSON PU 8550 or TEROSON VR 20 can be used as cleaners. When using solvents, an adequate flash off time (typically > 5 minutes) is essential to ensure dry surfaces and freedom of residues.

In case of plastics a pre-treatment like flaming, plasma or corona can be necessary to ensure good adhesion to the substrate. For MS-polymers, the use of a primer like TEROSON PU 8527 GM is also an option.

7.5 ENGINEERING APPROACH

A comprehensive assessment of the loading conditions for an intended seal design is feasible based upon numerical analysis by means of the finite element method. A finite element analysis (FEA) offers highly detailed geometric capabilities and allows for consideration of complex material behaviour, provided that corresponding material models are available and purposeful material data can be generated with appropriate accuracy.

7.5.1 MATERIAL RESPONSE AND ENGINEERING DATA

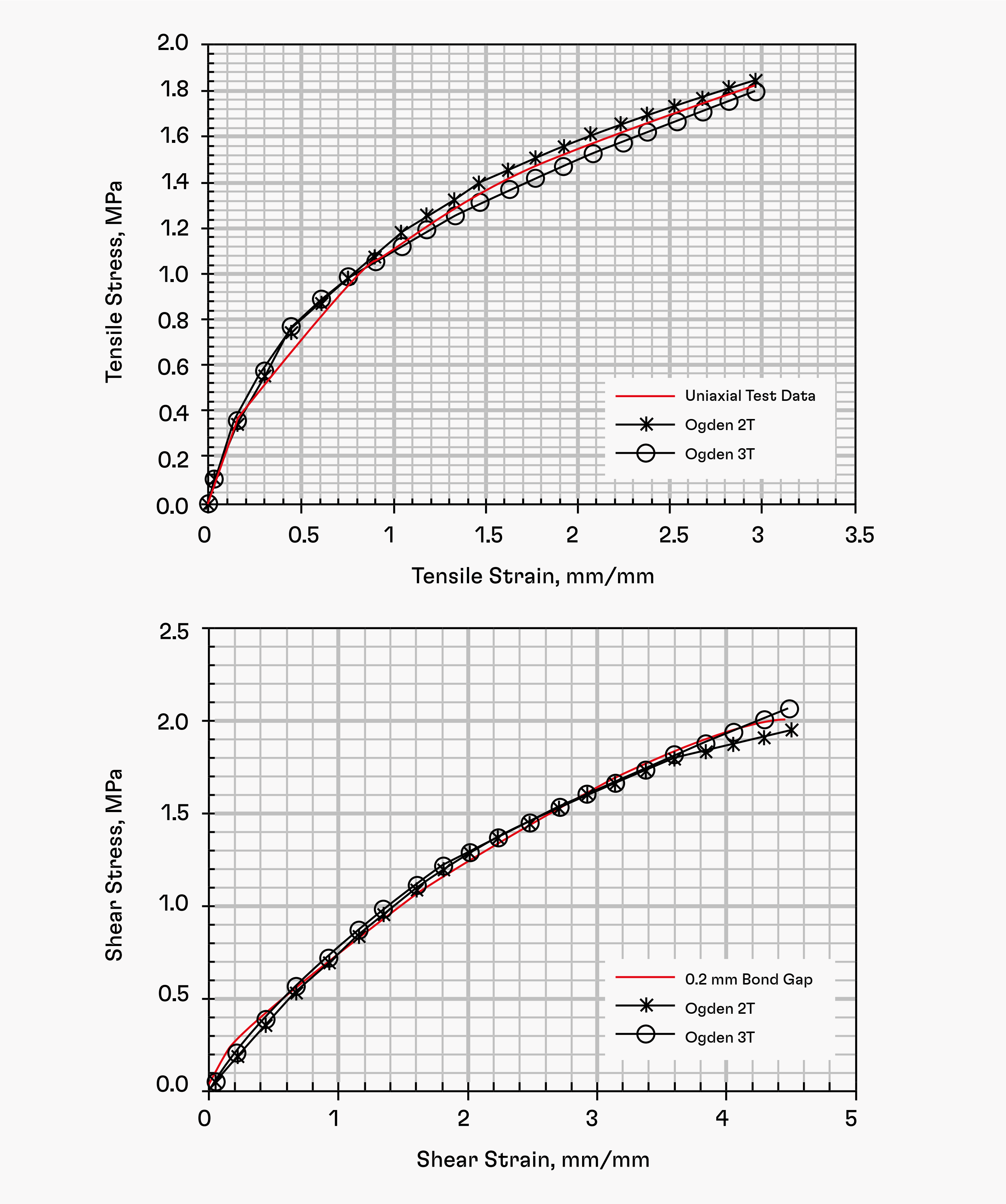
Flexible FIP sealants able to withstand large deformation are typically rubber-like materials. Quintessential features of their stress-strain response are its nonlinearity, and its recoverability. They can undergo large elastic deformations in the order of >150%, and, although not considered as incompressible, they show smaller volume change under applied stresses than polymer materials in general. Their behaviour may be primarily attributed to the deformation mode related to straightening of molecular chains.

The mechanical behavior of rubber-like materials is rather entropy-driven than energy-driven elasticity and it is usually referred to as hyper-elasticity. The mathematical description of the stress-strain behaviour in hyperelasticity makes use of the strain energy density function, W, which is some arbitrary function of the extension (stretch) ratios in the three principal directions.

There are several distinct approaches of the strain energy density function available, most of them leading to polynomial expressions including higher-order terms. Common strain energy functions are among others Ogden, Mooney-Rivlin, Yeoh, or a general form of higher-order polynomials.

Hyperelastic material constants for rubber-like materials are obtained through curve fitting of stress-strain data of preferably several fundamental strain states /1, 2/. For adhesives and sealants, typically lap shear tests are conducted beyond uniaxial tensile bulk material testing. The shear tests provide also evidence of the adhesion ability of the tested grades to near-application substrate materials. Data from lap shear testing (denoted “simple shear”), although not required for standard rubber-like materials, can be used in addition to the uniaxial tensile test data for curve fitting with the help of non-linear optimization.

FIGURE 37: EXAMPLE OF HYPERELASTIC MATERIAL CURVE FITTING (SI 5970, 23°C). UNIAXIAL TENSILE AND LAP SHEAR TEST RECORDS COMPARED TO PREDICTION USING 2- AND 3-TERMS OGDEN-TYPE HYPERELASTIC STRAIN DENSITY MODELS.



All grades recommended as flexible sealants exhibit glass transition temperatures below -40°C, and thus, they are suggested to behave rubber-like throughout the operational temperature range. Testing after exposure to elevated temperature for 500 h has also proven that the sealants still behave hyperelastic after heat ageing.

7.5.2 FAILURE CONDITIONS

No unified criterion exists for predicting the rupture of rubber-like materials under arbitrary loading conditions. In general, deformation- or strain-based criteria are preferred. Strains or stretches seem appropriate measures of the failure of soft polymers as they express limitations of chain segments extensibility of cross-linked macromolecules.

A widely used approach assumes failure to occur when the maximum principal strain exceeds a critical value. It has been found to provide good approximation of test data for rubber-like materials under varying loading conditions /3/.

The strain limit values applying can be estimated from uniaxial tensile or simple shear testing in terms of 1st principal strain. However, bulk test specimens may fail prematurely if the specimen contains flaws or irregularities included during manufacturing or die cutting. Shear tests provide more reliable failure limits although they are still affected, especially in the case of thicker bondlines, by peak strains and tensile stresses, which develop at the overlap ends due to the large deformation. Strains at failure obtained in lap shear testing using 0.2 mm thin bond gaps are assumed to better apply, although they may still underestimate the real failure limits.

In general, the maximum principal strain criterion is suggested to describe in good approximation distortional failure. A comprehensive failure analysis requires taking both, distortional and dilatational failure into account. The latter manifests in the onset and growth of damage when the material is subjected to triaxial (hydrostatic) tension. It is still controversial if such damage takes place when the conditions for growth of pre-existing micro-voids or infinitesimal flaws are met, or if stresses lead to formation and subsequently growth of microcracks and cavities (cavitation).

This type of damage obviously occurs only if the materials are sufficiently soft at large tensile stretches. It applies therefore primarily to RTV silicones and silane-modified polymers but appears not relevant for stiffer PU-based sealants.

After damage starts, it typically grows with stresses remaining practically constant for a large amount of deformation without rupture. A bond layer of 0.5 mm thickness typically withstands displacements of several mm in butt joint tensile testing prior to rupture. Material models to describe nucleation and growth of damage under hydrostatic tension do not have been established yet despite some recent development and are not available for numerical simulation.

An early formulated simple and therefore popular criterion indicates that damage onset occurs when the mean stress (hydrostatic tension) σm exceeds 5/6 E_{init}, where E_{init} denotes the initial modulus at infinitesimal strain /4/

$$\sigma_m = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} > \frac{5}{6} \cdot E_{init}$$

and all 3 principal stresses σi (i=1,2,3) are positive (tensile).

The criterion however was formulated for and applies strictly to a certain idealized type (Neo-Hook) of incompressible material response.

It is suggested that the ratio of the critical hydrostatic pressure to the small strain modulus varies depending on the compressibility of the rubber-like material /4/. The ratio then may drop to far below the initial modulus when compressibility increases, e.g. <40%. It has been however also proven that the ratio depends in addition on the hardening behaviour of the material at large stretches /5/. For large hardening exponents of a standard one-term Ogden material model, the critical pressure limit may achieve values several times greater than the small strain modulus. This is of particular interest for the RTV silicone SI 5970 tested after ageing in 5W30 synthetic oil. It typically exhibits then considerably low modulus at infinitesimal strains but remarkably stiffens at increasing stretches.

7.5.3 MODELLING ISSUES

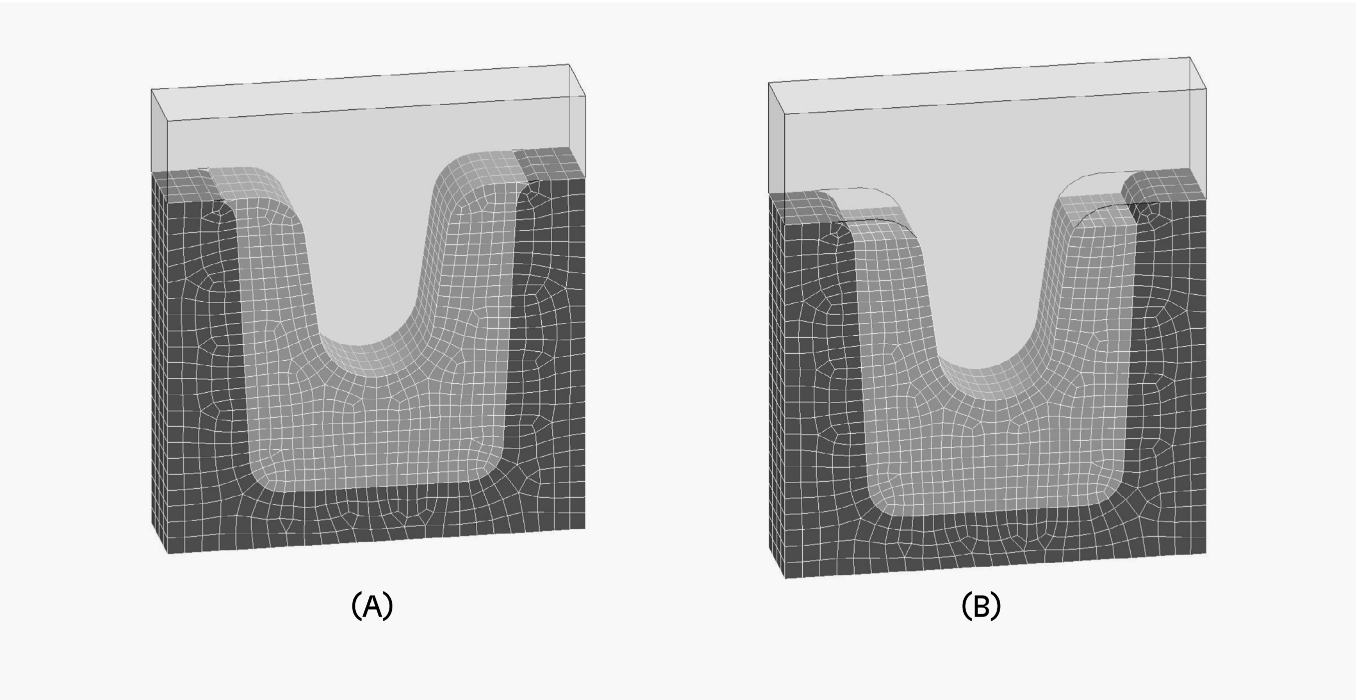
The small size of the proposed seal solution is a particular challenge. Accurate representation leads to high mesh density and thus, to high computational effort. A convenient way out is applying submodeling techniques which enable the deformation of the small gasket region to be describe in greater detail using boundary displacements from a global analysis.

High discretization is also required to prevent gross distortion of the elements when they undergo large deformation. One of the frequently encountered errors when using hyperelastic materials is the excessive element distortion. Due to the highly nonlinear nature of the simulation, certain local regions of the model may undergo excessive deformation. This, in turn, may lead to a non-physical shape of certain elements in the mesh, and therefore to numerical instability and sudden termination of the analysis.

Mesh replacement, or rezoning, is a technique that may be used to control element distortion in cases where large deformation occurs by replacing the mesh with an updated one fulfilling certain preconditions and continuing the analysis. Elements with nodes shared by all three parts involved, i.e. where seal material and both mating surfaces coincide in gasket design without an adhesive layer formed outside the groove, cannot be handled numerically when differential micromovement is applied and result so in analysis abortion. Elements at such corners shall be removed prior to run the analysis or the corresponding areas shall be excluded from meshing (Figure 38 a).

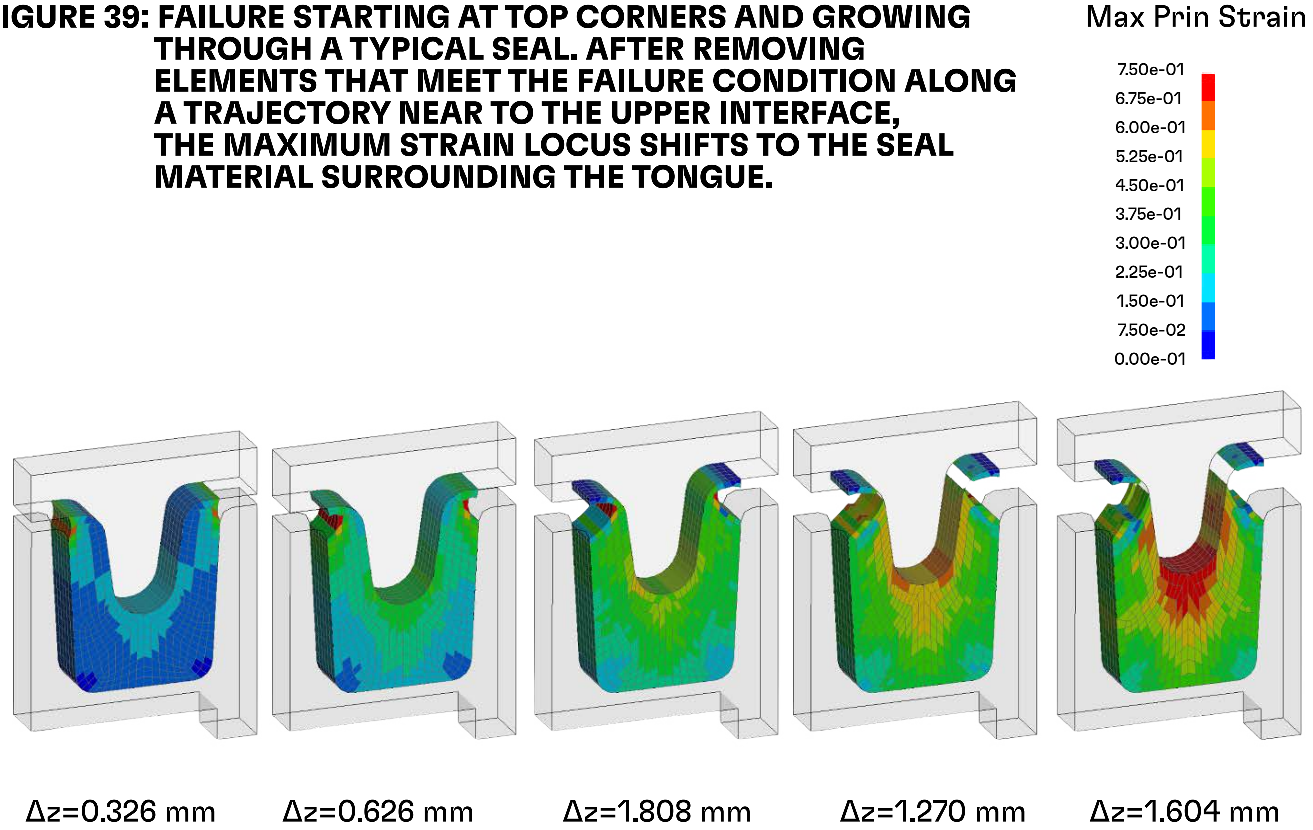
Further simplifications may be required to ensure numerical convergence. A purposeful one to prevent distortion of elements susceptible to gross deformation is to assume the groove to either being not completely filled during assembly or the seal at the top to be initially damaged and thus already separated from the mating surface (Figure 38 b).

FIGURE 38: ASSUMPTIONS TO FACILITATE FE ANALYSIS FOR SEAL MATERIAL CONFINED INTO GROOVE CAVITY AND TO PREVENT GROSS ELEMENT DISTORTION OR SINGULAR STRAINS AT OUTER CORNERS: (A) ELEMENTS AT OUTER CORNERS THAT MAY HAVE NODES SHARED BY ALL 3 PARTS ARE REMOVED TO OVERCOME UNPHYSICAL DOF CONSTRAINTS; (B) SEAL AT THE TOP IS ASSUMED TO BE INITIALLY DAMAGED AND THUS ALREADY SEPARATED FROM THE MATING SURFACE.



Releasing elements from the top of the cavity results in FE analysis in conservative sense. It may also reflect realistic conditions in practical applications. Computations using LS-Dyna FE solver with explicit time integration, which allows for physically removing elements due to failure at the time they meet the strain limit criterion, support the underlying assumption. After critical elements with excessive deformation failed starting from the top corners and continuing along a trajectory near to the upper interface, the maximum strain locus shifts to the seal material surrounding the tongue, typically at an angle of 30°-60° to its axis before spreading further out (Figure 39).

FIGURE 39: FAILURE STARTING AT TOP CORNERS AND GROWING THROUGH A TYPICAL SEAL. AFTER REMOVING ELEMENTS THAT MEET THE FAILURE CONDITION ALONG A TRAJECTORY NEAR TO THE UPPER INTERFACE, THE MAXIMUM STRAIN LOCUS SHIFTS TO THE SEAL MATERIAL SURROUNDING THE TONGUE.



The issue remaining is the application of the failure conditions and limits. The value of strains and stresses at critical areas depend on the mesh density and element length. Refining the mesh to prevent element distortion can lead to unrealistically high strain concentrations and failure predictions that are unrealistic. Frequently, failure limits in FE analysis of bonded joints are applied using a certain characteristic length over which stresses or strains have been averaged or a corresponding volume. This may help to account for high stress or stress gradients and relax unrealistic values. In applications with dimensional characteristics other than uniform bond gap, nonlocal methods may be used which average the local strains from a finite neighbouring volume to obtain a nonlocal strain.

For the purposes of the present guideline and the analysis conducted to investigate parameter variation, limits from experimental testing appropriately converted into maximum principal strain were applied to elements with a typical length of 0.05. Strains and stresses were averaged along each element area. This element size is far smaller than the characteristic lengths or areas typically used to assess the failure probability of adhesive joints. It is suggested that the small element size provides conservative failure assessment, and that the derived results lie therefore by far on the safe side. In addition, strain limits from experimental work are nominal values and do not account for stress or strain peaks arising at the overlap ends. A realistic failure prediction may need appropriate failure data adjustment to the actual element size in use.

Although simplifications are necessary to ensure analysis convergence and numerical stability, and despite the idealization of very small parts involved in the analysis, the FE method allows for consideration of complex material behaviour and helps to understand and identify failure risk areas and conditions.

7.6 BLOW-OUT TESTING

An appropriate test program was conducted to prove the sealing ability of the sealants under blow out test conditions by using appropriate test flanges that undergone 500 h heat ageing prior to testing. The flanges contained a gasket which was formed using a typical tongue and groove geometry and either provided a gap of 1 mm (Figure 40) to the inside and outside by means of suitable spacers or was gap free as the mating surfaces come into direct contact. After completion of the heat ageing period, the flanges were successively subjected to increasing internal pressure in steps of 0.5 bar, which was held constant for 30 seconds each time before being further increased to a maximum of 6 bar. All test flanges passed the blow-out tests without leakage.

FIGURE 40: TEST SET UP USED FOR BLOW-OUT TESTING AFTER 500 H HEAT AGEING

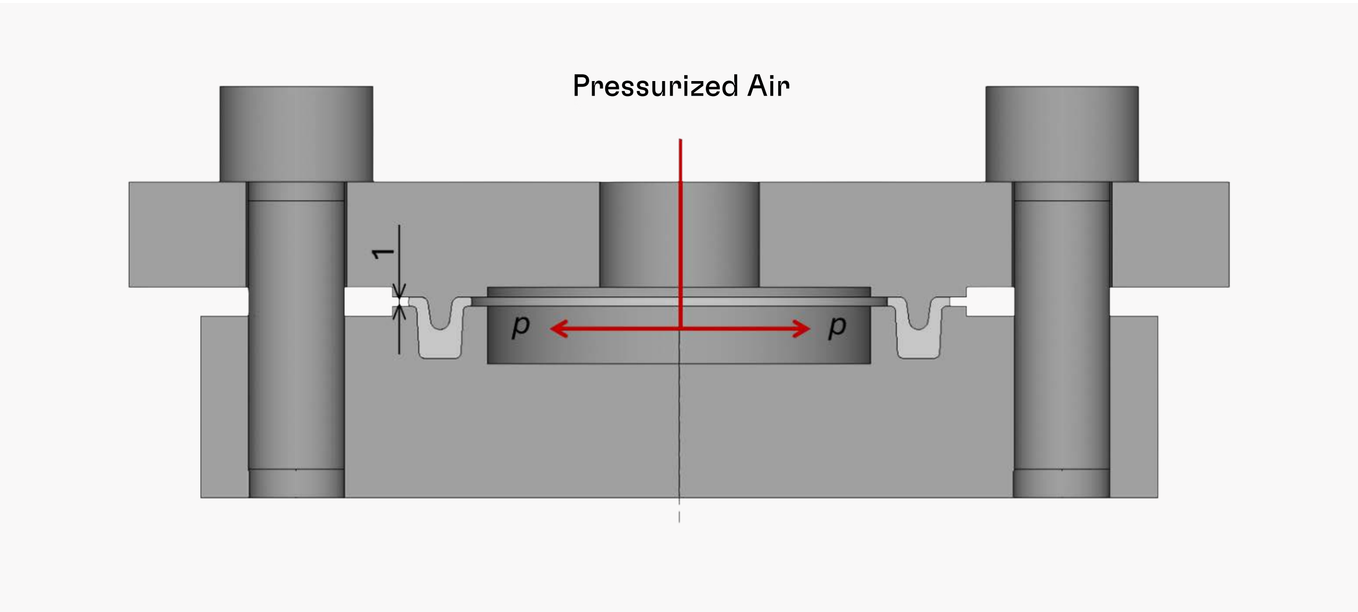


TABLE 3: AGEING CONDITIONS AND BLOW OUT TEST RESULTS

Product	Gap	Ageing Conditions			Blow Out Test Result
		Temperature	Medium	Duration	
UK 2073	1 mm	80°C	none	500 h	No leakage
	-				No leakage
MS 647	1 mm	100°C	none	500 h	No leakage
	-				No leakage
SI 5970	1 mm	150°C		500 h	No leakage
	-				No leakage

7.7 APPLICATION SUGGESTIONS

2-component sealants need to be mixed in a suitable 2-part dosing and mixing equipment prior to use. Apply using a meter mix pump for continuous in-line mixing and accurate ratio control. Use only static mix head. To ensure proper mixing, static mixers with at least 24 mixing elements are required.

Room temperature vulcanizing silicones such as SI 5970 do not need mixing. For full automatic applications a volumetric dispensing system shall be used.

UK 2073 is temperature sensitive with a short potlife. Extrude the required amount as a bead into the groove, then position rapidly the opposite part with the help of appropriate elements to maintain proper alignment and assembly as immediately as possible.

Curing occurs at room temperature and starts immediately either after the product is exposed to the atmosphere (1-component RTV silicone) or after component mixing (2-component sealants). It may be also affected by the level of moisture and temperature. Therefore, parts to be assembled should be mated within a few minutes after the product is dispensed. Follow the open time. In the case of delayed assembly, the product dispensed into the groove may start to gel or form a skin at top, which may result in inadequate adhesion build-up to the mating surface.

The rate of cure is temperature-dependent and thus, temperature affects open time and time to handling strength. High temperatures accelerate the cure speed and support adhesion build-up, while lower ones slow-it down. When accelerated curing by temperature (typically at 60°C) is considered, ensure that there is no differential thermal expansion between the parts involved or they are effectively constrained to avoid damage or inappropriate positioning of the sealing geometry due to resulting displacement.

Meter-mix systems require purge procedures and regular cleaning. Purge and rinse mix head regularly to avoid fouling.

8. PRODUCT DISPENSING AND ASSEMBLY

In volume production, the following points have to be considered:

8.1. CLEANING

All manufactured parts must be cleaned after machining. To achieve consistent quality of the cleaning process, the procedures provided by the detergent supplier should be followed. In general, RTV Elastomer sealants are less sensitive to contamination than anaerobics.

Anaerobics:
For anaerobics, both surfaces must be clean and dry before dispensing and assembly. Contamination on the flange could inhibit the cure of the product or reduce the adhesion to the substrate.

RTV Elastomer Sealants:
RTV Elastomer sealants also need clean, dry flanges to achieve a high quality and durable seal. Wet or oily surfaces could reduce the adhesion significantly and the dispensed bead could slip away from the correct position, which would impact the sealing performance.

General:
It is recommended to run basic adhesion tests with the production washing solution. Test conditions should be worst-case scenarios.

8.2. DISPENSING

Robot dispensing
The most flexible and reliable way to apply any kind of sealant is robotically or using an XY-Table. For this reason, Henkel recommends this technology especially for high-volume production.

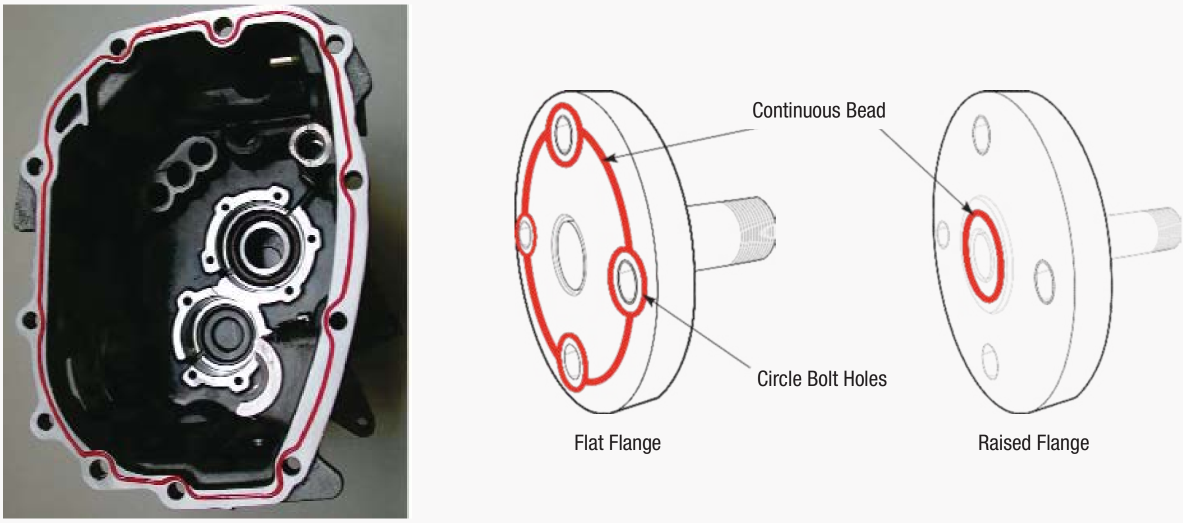
Henkel has developed its own dispensing systems that are able to apply high-viscosity anaerobics and RTV Elastomer sealants with high speed and excellent quality.

Quality systems like flow monitors or visual inspection systems are recommended to achieve consistent high quality.

EXAMPLE FOR ANAEROBIC

- Bead diameter 1.5 ± 0.5 mm
- Position the bead on the centerline of the mating surface with an accuracy of ± 1 mm
- Apply a continuous bead of material inside or around dowel and bolt holes
- Inspect the bead for uniform thickness, air pockets, voids, and continuity
- Distance between nozzle tip and the flange should be (1.3 ± 0.2 mm) bead diameter
- Dispensing speed of 80 to 130 mm/sec

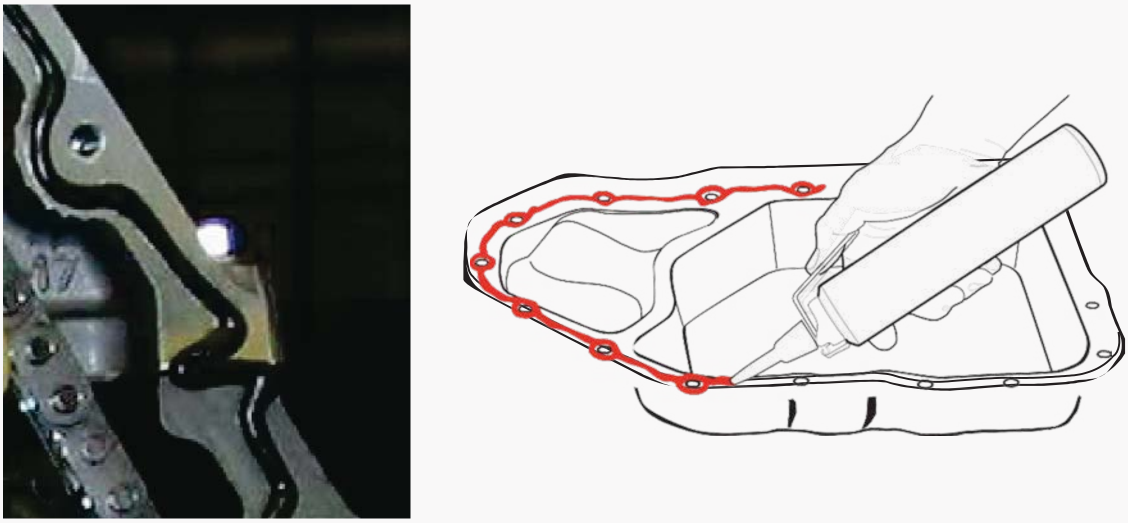
FIGURE 41: ANAEROBIC PRODUCT BEAD ON GEARBOX FLANGE.



EXAMPLE FOR RTV ELASTOMER SEALANT

- Bead diameter 2.5 ± 0.5 mm
- Positioned on flat flange area, center of bead 1 ± 1 mm from chamfer
- Apply a continuous bead of material inside or around dowel and bolt holes
- Inspect the bead for uniform thickness, air pocket voids, and continuity
- Distance between nozzle tip and flange should be (1.3 mm ± 0.2 mm) bead diameter (See also under Section 6.1 / Figure 12)
- Dispensing speed 80 to 130 mm/sec
- Note: T-joints may require a higher amount of RTV Elastomer sealant product. An increased bead diameter or a special dispensing path are proper ways to fill the void in that area. Dispensing studies are required to evaluate the correct product quantity. When a "dollop" is required to seal a T-joint, it is typically 8 mm diameter at the base.

FIGURE 42: POSSIBLE RTV ELASTOMER SEALANT BEAD ON T-JOINT.



SCREEN PRINTING

This process can be used to apply anaerobics. Screen printing is especially suitable for medium scale production, and where flexibility is not required.

A flat surface (e.g., no dowels) is necessary for the screen printing process.

Screens do wear and, therefore, have to be replaced from time to time.

Screen printing cannot be used for the application of RTV Elastomer sealants.

8.3. ASSEMBLY

It is essential to understand the procedure on the assembly line where FIPGs will be used. The assembly conditions, steps and cycle times have a major effect on the selection of the sealant, and later on the quality of the whole process.

It is important to avoid contamination of the flanges prior to assembly of the parts. When parts have to be moved, the dispensed product has to stay in place. The product should never be touched before assembly.

Quality inspection systems can help monitor dispensing and detect bead misplacement or bead interruption.

Once the applied product connects with both flange surfaces, any movement relative to the joint surface must be avoided.

ANAEROBICS:

The basic curing chemistry of anaerobic sealant allows an unlimited open time. Nevertheless, fast curing sealants can start to precure even before assembly. This so-called shimming effect will influence the sealing performance and cause a gap between the joint surfaces.

It is highly recommended to fully torque down all bolts immediately after joining the flange faces to avoid shimming and, later, leakage. Sub-assembled parts may require slave fasteners.

RTV ELASTOMERS:

Once the product is dispensed, the flanges have to be assembled within the skin over time of the product (approximately 5 to 15 minutes for most products, refer to TDS). Full torque down is not immediately necessary. Depending on the size and stiffness of the flanges, it is allowable to run down several bolts first, with full torque run down to follow within 20 to 30 minutes.

IMPORTANT:

Line shutdown or breaks must be taken into account. Proper planning of the assembly line can avoid the need for scrap or reworking of parts.

9. SERVICE AND REPAIR

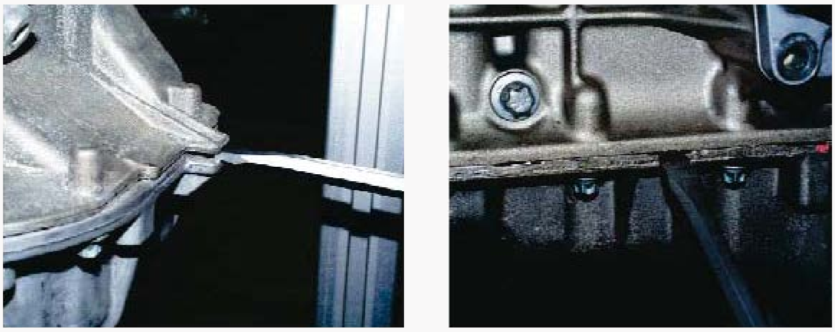
9.1. DISASSEMBLY

With the correct design, product and process, the joint will maintain the sealing capability throughout the life of the vehicle. Disassembly will, therefore, only be necessary for mechanical repair.

Design for Disassembly – Anaerobic / RTV Elastomer Sealants:
A highly effective and inexpensive method for disassembly is to implement special design features during component development.

The following pictures show two ways to achieve this:
Depending on part size and accessibility, two or more of those bosses or recesses are needed on each joint.

FIGURE 43: POSSIBLE DESIGN FEATURES FOR EASY DISASSEMBLY OF CAST PARTS. IN SOME CASES, ACCESSIBILITY OR SPACE REQUIREMENTS MIGHT LEAD TO THE USE OF JACK SCREWS. THE HOLES USED FOR BOLTING THE PARTS CAN BE USED AS JACK SCREW HOLES.

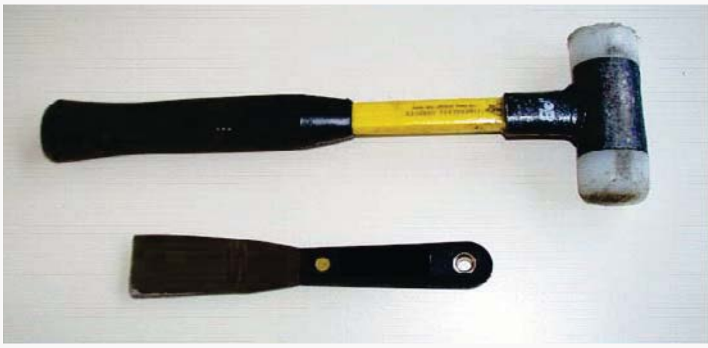


DISASSEMBLY TOOL – RTV ELASTOMER SEALANTS

Another common disassembly procedure, especially for stamped parts, is the spatula + hammer method (see Figure 22). This is valid for OEMs as well as service garages. The spatulas have to be partially modified depending on the access and handling conditions. The front and side edge should be sharpened for easy insertion and easy cutting.

The main advantages of this method are:
- • Good availability – every tool box is equipped with those tools
- • Little to no surface damage
- • Mechanics are used to this technique – used also for hard gasket disassembly
- • Low cost
- • For both cast aluminum and stamped steel oil pans
- • Compatible to the most common engines

FIGURE 44: EXAMPLE OF SPATULA WITH A PLASTIC HAMMER.



9.2. CLEANING

For FIP gaskets, it is essential to have an appropriate cleaning process in place to achieve a high quality seal.

After the disassembly of the parts, both flange surfaces must be cleaned and inspected.

ANAEROBIC

Old gasket residues must be completely removed to avoid shimming, which could cause a gap.
All dirt or fluids must be removed from the sealing surfaces to guarantee good product curing and adhesion to the substrate.
Contamination of the flanges prior to the assembly must be avoided. Therefore, it might also be necessary to clean parts or areas in the neighborhood of the sealed joint.

RTV ELASTOMER SEALANTS

The old gasket must be removed. Depending on the application, small amounts of residue are acceptable because, generally, the fresh RTV Elastomer sealant has good adhesion to old RTV Elastomer sealant. If shimming is an issue, the flange must be completely clean to avoid any gap or mispositioning.
Dirt or fluids must be removed from the sealing surfaces. The product must be applied to a dry flange.
Contamination of the flanges prior to the assembly must be avoided. Therefore, it might also be necessary to clean parts or areas in the neighborhood of the sealed joint.

Cleaners and Tools

- Loctite SF 7200 Parts Cleaner - Gasket Remover
- LOCTITE SF 790 Gasket Remover
- Spatula, plastic scraper
- Scour Pads

Do not use petroleum cleaner or mineral spirits that leave a residue and prevent adhesion or curing.

9.3. APPLICATION AND ASSEMBLY

For service, the only practical application method for a FIPG is manual bead dispensing.

It is important to describe where the product has to be dispensed and in what quantity. This should be shown in the service manual.

Apply sealant to only one of the sealing surfaces.

Inspect bead position, quantity and continuity, and repair imperfections immediately after dispensing.

ANAEROBICS:

Anaerobic products should be applied in a straight line in the middle of the flange (like robot application in volume production). The quantity must follow recommendations shown under Section 7.2 Dispensing.

RTV ELASTOMER SEALANTS:

When dispensing RTV Elastomer sealants, it is usually easier to find the proper bead location when the product is applied on the part with the chamfer. The location and quantity of the bead has to follow recommendations shown under Section 7.2 Dispensing.

10. SCOPE AND LIMITATIONS

This guideline is based on Henkel application experience of more than 25 years, fortified by extensive testing done in the GEC in Munich since 1991. With the knowledge we have accumulated, Henkel is able to demonstrate how a reliable seal can be achieved.

The content should be used to assist during development or to discuss occurring failures in the field. It can also be used to optimize existing flanges.

The guideline cannot replace detailed discussions between customer and the local Henkel sealing expert. Experience shows that every flange and application is different and, therefore, in-depth knowledge of product, design and process is necessary to find the best solution for each case. Exceptions to the given rules might be required and should be verified.

11. ABBREVIATIONS

SLS Single-Layer Steel – A gasket constructed from a single layer of steel. Typically, the steel will include an embossed sealing bead and may incorporate an additional surface coating or treatment.

MLS Multi-Layer Steel – A gasket constructed from two or more layers of steel. Typically, one or more layers will include embossed sealing beads and the gasket may incorporate an additional surface coating or treatment.

SGM Soft Gasket Material (including fiber, beater addition, paper, flexible graphite and cut rubber sheet) – A die-cut soft material that compresses to conform to the joint and create a seal. The gasket may additionally be treated with printed beads, surface coatings, sealing grommets, pressed beads, or saturants.

Com Composite – A gasket formed by combining one or more layers of a soft gasket material with one or more metallic layers. Layers may be mechanically or chemically bonded.

FIPG Formed-In-Place Gasket – See Section 2 for definition.

CIPG Cured-In-Place Gasket – See Section 2 for definition.

IIP Injected-In-Place – See Section 2 for definition.

MIP Molded-In-Place – See Section 2 for definition.

RTV Room Temperature Vulcanizing, a curing mechanism, e.g., for RTV Elastomers – see also Section 4 RTV Elastomer sealants.

Cu Copper

GEC Global Engineering Center

T-joint Area where two joints meet – See also Section 6.2.

12. BIBLIOGRAPHY

1 Klöpfer M., Jäckle, M., Lechner G, Universität Stuttgart, Institut fÜr Maschinenelemente: Deckeldichtungen. Abschlußbericht 152/III, FVA-Forschungsheft-Nr. 463, Frankfurt: Forschungsvereinigung Antriebstechnik e.V., 1995

2 Kreuzer R.: MAE.0001 – Comparison of Anaerobic (518) and Silicone (5699) as Gasketing Materials. MRD 94-04 June 17, 1994.

3 Prediger B., Kleiner F.: MAE.0021 – Blowout Behavior of Silicones. MRD 95-01 January 31, 1995.

4 Ritter K-H., Schmatz T.: Design Instruction RTV Silicone – Sealant. September 1996.

5 Ritter, K-H., Clauss, R.: The New Challenge for Sealant Supplier – A Partnership with the Automotive Industry. SAE 960212, Society of Automotive Engineers, Inc., Warrendale, PA, 1996.

6 Kreuzer R.: MNA.0436 – Jaguar Engine AJ30/31. MRD 96-01 April 9, 1996.

7 Kreuzer R., McClelland B., Garnich F.: MAE.0420 – Design Guidelines for Gasketing Applications. MRD 96-03 July 17, 1996.

8 Kleiner F.: MNA.0006 – Anaerobic Solution for a Bedplate Application. MRD 96-04 August 23, 1996.

9 Kreuzer R.: MTP.0497 – Gasketing Fatigue Test with 5900. MRD 96-07 December 13, 1996.

10 Ritter, K-H.: Design Guidelines and Concepts for Zero-Gap Bedplate Sealing. SAE 1999-01-0593, Society of Automotive Engineers, Inc., Warrendale, PA.

11 Kreuzer R.: MAE.0019 – Comparative Testing of Gasketing Materials. Reports MRD 97-03 May 16, 1997, MRD 98-04 July 6, 1998, MRD 99-04 May 18, 1999.

12 Kleiner F.: MAE.0623 – Surface Roughness Investigation. MRD 99-09 July 31, 1999.

13 Kreuzer R., Romanos G.: Zuverlässigkeit von Flächendichtungen auf Basis von Flussigdichtmitteln unter dynamischer Beanspruchung. VDI Berichte Nr. 1579, 2000.

14 Kreuzer R.: MAE.0747 – Comparative Fatigue Testing. MRD 00-04 May 18, 2000.

15 Kreuzer R.: MDP.1020 – Ford RTV Key Life Test. MRD 00-05 June 5, 2000.

16 Kreuzer R.: MAE.0749 – FEA Model of a Sealed Joint. MRD 00-21 December 11, 2000.

17 Kreuzer R.: MAE.5017 – Comparative Fatigue Testing. MRD 01-01 January 16, 2001.

18 Kreuzer R.: MAE.5041 – Fatigue Tests at Gasket Test Rig 2. MRD 01-06 August 21, 2001.

19 Kreuzer R.: MDP.5045 – Ford-KLT 5970 Investigation. MRD 01-05 September 24, 2001.

20 Schmatz T.: Guideline to Seal the T-Joint. September 2001.

21 Schmatz T.: MDP.0722 – T-Joint Investigation for Ford. MRD 01-07 October 18, 2001.

22 Kreuzer R.: MAE.5040 – Adhesive Layer Thickness. MRD 02-03 February 4, 2002.

23 Kreuzer R.: MAE.0748 – Gearbox Fatigue Testing. MRD 02-05 July 1, 2002.

[24] Kreuzer R.: MAE.5067 – Component Fatigue Testing – Gearbox. MRD 03-02 January 20, 2003.

[25] Kirchberger P.: Ford Puma T-Joint. MRD 03-04 February 27, 2003.

[26] Schmatz T.: MDP.0905 – Next Generation RTV. MRD 03-06 March 21, 2003.

[27] Becher J.: MAE.5068 – Adhesive Joint Behavior. MRD 03-07 March 20, 2003

[28] LOCTITE Worldwide Design Handbook. 2nd Edition 1998

[30] Treloar, L.R.G.: The Physics of Rubber Elasticity. 2nd ed. Oxford: Oxford University Press 1958

[31] Miller, K.: Testing Elastomers for Hyperelastic Material Models in Finite Element Analysis. Axel Products, Inc., 2000

[32] Rosendahl, P.L.; Drass, M.; Schneider, J.: Distortional-dilatational strain failure mode concept for hyperelastic materials. ce/papers - Proceedings in Civil Engineering 4 (2021) 6, 301-322

[33] Lopez-Pamies, O.: Onset of Cavitation in Compressible, Isotropic, Hyperelastic Solids. J Elast 94 (2009), 115–145.

[34] Horgan, C.; Polignone, D: Cavitation in Nonlinearly Elastic Solids: A Review. Appl Mech Rev 48 (1995) 8, 471-485

12.1. SUPPLEMENTARY INFORMATION (APPENDIX)

TABLE 4: NOMINAL SHEAR STRAIN AT FAILURE AT DIFFERENT TEST TEMPERATURES BEFORE AND AFTER HEAT AGEING (AVERAGE VALUES FROM LAP SHEAR TESTING 1.0 MM THICK BOND GAPS)

Nominal Shear Strain at Failure, mm/mm 1.0 mm Bond Gap / Virgin			
Test	**SI 5970**	**MS 647**	**UK 2073**
Temperature	**not aged (virgin)**		
-40	4.35	6.47	2.40
23	2.89	4.34	2.31
80	-	-	2.55
100	-	3.38	-
150	1.74	-	-

Nominal Shear Strain at Failure, mm/mm 1.0 mm Bond Gap / Aged			
	SI 5970	**MS 647**	**UK 2073**
Test	**aged 500 h at**		
Temperature	**150°C/oil**	**100°C/air**	**80°C/air *)**
-40	5.75	4.59	1.82
23	5.96	4.51	1.76
80	-	-	1.94
100	-	3.63	-
150	4.96	-	-

*) Shear strain limits for UK 2073 after ageing at 80°C are estimated values based upon comparison to corresponding uniaxial test data

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