Standardized Connectors for Automotive Brake Tubing Overview: Modern Requirements, Perfection Targets and Approaches

Stanislav I. Pliassounov

Connectors are essential components of contemporary hydraulic brake systems since desired layout of brake lines (tubing) would not be feasible without joining them together. Generic connector's arrangement includes a receiver (port), a tube (which is intended to be connected with that port) and a clamping mechanism to force them together. The seat (designated part of port's surface) is intended to receive the tube's flare with the purpose of developing a fluid-tight seal through mating of their sealing surfaces.

It is important to stress that contemporary brake tubing connectors do not rely on any sort of intermittent sealing media like sealing rings, putty, glue, washers etc. The seal must be achieved solely by clamping force, which makes metal to metal mating tight enough to contain high pressure inside the tubing. (Automotive brake tubing is designed to withstand operational pressure around 14 MPa or 2000 PSI and most of OEMs guarantee burst resistance to 80 MPa). Therefore, a good and robust connector seal may be expected only if an adequate clamping force has been developed onto the mutual contact area of an uninterrupted shape of a required size.

When being put together a brake tubing connector must perform two primary functions. First is to align the flare with the seat. Second is to force the flare and the seat against each other. The alignment function has the purpose to develop the mutual contact of uninterrupted (annular) shape. The clamping function is designed to create and maintain an adequate clamping force at that mutual contact zone.

Automotive tubing connectors are standardized components. International standard ISO 4038 [1] (introduced more than 30 years ago) outlines generic design of such connectors among other components of automotive hydraulic brake systems mostly focusing on their interchangeability. More detailed specification of standardized connector's design (the sizes, materials, geometry etc.) can be found in industry-wide standards, for example those maintained by SAE (Society of Automotive Engineers). In turn, when dealing with manufacturing of connector's components OEM developed corporate level detailed standards (specifications) deriving from those international or industry-wide ones.

Currently in automotive industry, only two distinct arrangements of standardized mass-produced hydraulic tubing connectors are utilized. Both the arrangements are based on a cone-to-cone style of mating between the flare and the seat. They utilize threaded joints to develop the clamping force and then to keep such tightened (secured) condition during the entire lifespan of the connector. Typically the alignment is accomplished either before or during the earlier phase of torqueing (securing) process. Virtually all contemporary mass-produced passenger vehicles incorporate such brake tubing connectors.

One arrangement incorporates a male type (convex) seat situated inside the connector's port and extending into it. Accordingly, a female (concave) flare with its inner surface dedicated for abutment against the seat with the purpose of forming the fluid seal, is required. This arrangement is called the JASO/SAE connector design. Its nominal sealing surfaces' shape is a frustum (portion of a cone with cut off vertex). Its double inverted flare (a.k.a. funnel or trumpet) has an inner (concave) frustum which is intended for sealing onto the "external" (extending into the port) seat. The design is defined by SAE 1533 [2] and JASO F402 [3] standards (which are similar to each other); see Fig1.

Fig. 1

The other standardized arrangement incorporates the reversed combination—a female type (concave) seat interacting with a male type (convex) flare. This one is called the ISO connector design. It incorporates the "bubble" flare with its external surface dedicated for abutment against the seat with the purpose of forming the fluid seal. Nominal shape of its sealing surface is also frustoconical. The frustoconical concave seat is an integrated part of the connector port - contrary to the JASO/SAE there is nothing extending into the port. The metric version of that ISO design is defined by the SAE standard J 1290 [4]; see Fig2.

The use of the ISO connector design (a concave seat interacting with a convex flare) has many advantages comparing to the JASO/SAE one (a concave flare interacting with a convex seat) in terms of flow consistency, manufacturing feasibility, cost and quality control.

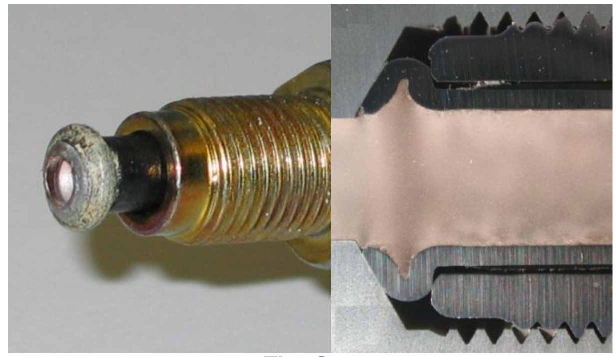


Fig. 2

The manufacturing process for concave ISO seat has less steps and the tooling is simpler than convex JASO/SAE ones (because the concave seat shape is simpler as no parts extend into the port). It is also easier to control the quality of a concave seat and the measurement fixtures are simpler as well. Moreover, it is easier to detect a defect on the external surface of a convex flare compared to the internal surface of a concave flare. Taken as a whole it is less expensive to produce the components for the ISO design and there is less probability for a defect to escape.

The only advantage which the SAE/JASO flare (concave) has over the ISO (convex) one is the opportunity to incorporate coining. Coining delivers precise and smooth sealing surface. Sadly, in case of the ISO flare it is impossible to use coining as there is no buttress available in the course of the flare forming process to squeeze its sealing surface against. That is why the ISO flare's sealing surface may not be as smooth and precise as the SAE/JASO's one. However, it is important to emphasize that the coining operation must be tuned into the process properly. Otherwise the position of such better sealing surface may get additional variation against the rest of the flare which in turn may worsen its mating with the seat and eventually create more problems effectively cancelling that potential advantage. Variation of the sealing surface position leads to the variation of actual datum of the mating. That, in turn, requires additional torque to engage more self-adjustment making the assembly process less predictable. And on top of that, it is possible to get a specific defect – the edge where the sealing surface's frustum meets the flare's outer part. Such defect significantly increases the probability of unwanted single point initial contact (which will be explained shortly) and is easily able to escape quality control.

Despite of its overall disadvantages the SAE/JASO arrangement is still widely utilized because there is a popular belief that this arrangement provides less probability to get a leak than the ISO one. Seemingly easier repair in case of the SAE/JASO arrangement in fact hides an acquired defect (flow reduction) and may lead to untimely part replacement in the field. Since torque engages connector's self-adjustment, it is a common practice to apply some extra torque in order to repair the connector if there is a leak. Usage of excessive extra torque is virtually undetectable in case of the SAE/JASO. The flare (female/concave) in this case envelops the seat (convex/male) and eventually translates all the deformations sustained due to excessive torque onto the seat. Accordingly the initial geometry of such male (convex) seat may get changed significantly. Returned part analysis revealed [5] that such over-torqueing may lead to a substantial alteration of the diameter of the passage, which in turn may cause a considerable flow rate decline. Contrary, in case of the ISO flare, excessive torque leads to a crash of the flare and its replacement. Obviously, it is more difficult to replace the tube than just to apply extra torque, but it is definitely worth the hassle as the defect gets surely detected and replaced along with the broken flare (no impact to the connector's performance).

Current standardized brake tubing connectors provide cost effective and relatively simple solutions for brake tubing connecting needs. This basic design could be traced as far as a century back when the tubing was usually made of relatively soft material (copper or similar). At that time there was the third function implied into the connector's design. In addition to the need to align and force the components together a connector was also supposed to accomplish the forming of desired geometry onto one or both mating components. Such forming consumed significant amount of torque required to secure the threaded joint. Moreover the torque variation was huge as the amount of necessary deformation to form the mating component was also unpredictable due to dimensional variation, and variation of the material properties. Gradually, ongoing perfection of the design has finally progressed up to contemporary state when incorporation of brazed double walled steel tubing brought to the design more similarity with an ordinary threaded joint (which is supposed to build up a certain level of stress into its components in order to perform the job).

Accordingly the requirements for brake tubing connectors have been progressively changing. In addition to the fundamental requirement to assure no brake fluid leaks from sealed joints for the entire life of the vehicle, there are many extra ones. And the most important ones among those extras are the requirements emerged from the assembly process excellence mindset and lean manufacturing philosophy. In modern assembly process each connector must be assembled and sealed with the following conditions: at a designated assembly station; by using specified torque; on the first attempt;

within time allocated for that operation; despite the presence of unavoidable disturbances and variations of the assembly process; and on top of that - there is no immediate verification available at the assembly station to check the result (has the connector got sealed or not). Taking into account that excellence and lean approaches imply only a few failures per million allowed, it becomes clear that the design of a modern connector must be capable to deliver very high probability to get it assembled and then turned into the sealed status on the very first attempt.

A contemporary passenger vehicle is usually equipped with a features loaded brake system (nowadays the ABS function became a kind of a minimum performance standard while vehicle stability control or similar functionalities are currently usually more desired options). Such sophisticated brake systems require a complex brake tubing layout with more than 20 connectors per vehicle. However, even one unsealed connector is enough to produce a no-brake unit which is subject for towing-off at the end of the assembly line. When a vehicle has been rejected in the course of the assembly process by the evac-and-fill equipment, it cannot be driven off in normal operational way since the brake system is not functional. Consequently a designated repair bay with brake fill and diagnostic equipment is necessary for those off-line repairs. A repair in that designated bay includes examining and repairing. Even in best case scenario, when only re-torqueing of the affected connector was sufficient, the duration would not be shorter than 20 minutes (getting the vehicle in the repair bay, setting it on the hoist, performing diagnostic, fixing the failure, getting the vehicle off the hoist). In case of using the force-fill technique for uncovering the leak location, that minimum repair time would be extended to around 30 minutes. Correspondingly, an assembly line may afford maximum 16…20 of such rejects a shift to avoid piling them up for later repair. Analysis and calculations [6] reveal that at least 99.9% probability to assemble and seal each connector from the very first attempt at the designated station is necessary to manage that acceptable level of those 16…20 no-brakes rejects a shift. In other words, this is the performance level of 1 failure per 1000 units or better per each connector (assuming a typical production shift of 500 …600 vehicles with 20…30 connectors each).

This probability includes both the likelihood of operators' unintended mistake and the connectors' designed capacity to deliver desired sealed status when designated torque and other necessary operations have been correctly applied in the course of the assembly process. Utilizing the process approach, it is possible to imagine all relevant connector's assembly and securing (torqueing) operations as a black box. Then the inputs are: designated torque (and other relevant tooling), all relevant assembly steps, connector's components etc.; subsequently the output is the number of sealed connectors. And (in process terminology) that desired 99.9% probability is the first pass yield of such assembly process. Thus, 99.9% first pass yield at each connector (or better) is the modern requirement based on process excellence mindset and lean manufacturing philosophy.

Should equal split between those mentioned above probabilities (operator related and design related) be assumed as simplified approach, then the connector's design must be robust enough to deliver the first attempt seal probability at each connector around 99.95% (500 failures per million). That level of performance matches to only one failed unit out of each 2000 in average when proper input to the process has been delivered (excluding operator dependency and assuming absolute conformity of the parts). Correspondingly, in the framework of that simplified approach, same performance of only 1 failure per each 2000 connectors or better should be expected from the operators.

It is obviously too optimistic of an expectation for the performance of operators dealing with current standardized connectors. 1 failure out of 2000 connectors means that each operator is allowed to slope the performance and forfeit an unintended mistake only once after 4 shifts in the raw (assuming same as above typical production shift). Unfortunately a little more realistic operator's performance of one failure a shift is already not sufficient. In this case the probability of success would be only 99.8% which is worse than the desired 99.9% and therefore unacceptable (same as above 500 units shift is assumed). And if the operators will deliver almost perfect performance of 1 failure per 1000 connectors on average the connector's design must in turn be capable to deliver 100% in order to attain that 99.9% target. Most excellent operator performance cannot compensate poor connector design. Correspondingly, a good connector design must be forgiving to unintended operator's mistakes making it possible for the operators to produce no failure during several production shifts in the raw.

Current standardized mass-produced connectors are obviously not able to support these modern requirements as their assembly process is heavily operator dependent. In order to resolve that modern lean and excellence challenge, connectors' design must be improved in such a way which would virtually eliminate the operator dependency. Moreover, the design must also be robust enough to other unavoidable variations and disturbances of the connectors' assembly process. Therefore the more robustness is available from the connector design – the better. That is why the prominent Six-Sigma level of less than 4 allowed failures per million units should not be regarded as excessive target when a modern brake tubing connector will be designed. Tangible acceptable robustness level (between those 3 and 500…1000 failures per million connectors) should be decided by each OEM based on tolerable level of relevant rework and waste.

It is important to emphasize that while existing connectors are not up to the challenge to meet those relatively new requirements that have emerged from modern assembly process performance demands; they are still good enough for the field. Current practice, field data and returned part analysis suggest acceptable performance with respect to the fundamental requirement to assure no brake fluid leak from the sealed joint for the entire life of the vehicle. At least those general purpose connectors (not involved in special applications like for example racing cars) do demonstrate the sealing performance acceptable to general public expectations. Ironically the modern assembly related requirements are stricter than the ones in the field. And there is no choice for each connector but to reach the end of the assembly process in order to begin demonstrating its worth in the field. That is why such assembly related requirements must be taken into account while designing new and improved connector arrangements especially in case of so-called quick (snapping) connectors.

Pursuing connector design excellence by targeting elimination of the operator dependency essentially means perfecting them by elimination of the failure modes associated with possible operator mistakes.

Therefore a connectors design must incorporate the solutions capable to provide resilience to the operator dependency. For example, a cross-thread is a typical operator mistake during the hand-start operation (the initial step of connector assembly process). It is a known fact that a fine thread is prone to cross-threading while a coarse one is not. Therefore, a connector design should incorporate a coarse thread of such size which would make cross-threading by the operator practically impossible. However, too much coarseness makes it more difficult to align and therefore seal the tube against the seat because of inherent shortcomings of the cone-to-cone mating (which will be explained shortly).

Hence, either the type of mating should be changed or some sort of a compromise solution is to be incorporated. And so far such compromising approach to keep current mating has been the mainstream in improving standardized connectors' design. There are many known solutions to facilitate the hand start when a relatively fine thread is utilized: dog points, chamfered thread start, wobble nuts, coned thread starts etc. However even the most prominent solution from this group - the MAThread - did help to reduce the operator dependency but did not eliminate it. (The MAThread features rounded thread at the start with gradual transformation of its external diameter into a conventional thread – see the nut on Fig.2 cross-section).

Despite incorporation of numerous innovative solutions current connectors' design is still not good enough. The assembly first pass yield of current mass produced standardized connectors is definitely below the outlined 99.9% target. Still some re-torqueing is usually required in the course of the assembly process in order to seal a number of them. Accordingly each assembly plant is compelled to allocate substantial resources for the associated detection and rework process as they strive for zero leaks in the field. Part suppliers are forced to supply the components unnecessarily close to the specification targets, which seriously restricts their operational window. Overall cost to maintain desired quality is too high and all associated expenses and waste are avoidable. This is an evident saving opportunity justifying the need for further improvement of the connectors' design. Regrettably numerous investigations and root cause analyses reveal a fundamental shortcoming with respect to the seal developments inherent to the design of cone-to-cone based standardized connectors.

As it was stressed above, robust connector seal may be expected only if adequate clamping force has been built onto the contact ring of sufficient size between the sealing surfaces. And because of that fundamental shortcoming such ring (annular shaped mating zone) may be developed only if certain conditions are met. The ring may be expected only if the axes of both the flare and the port coincide. Otherwise, it is common that the result of cone frustum side surfaces crossing (i.e., having a geometry entity which belongs to both frustums) is just a single point. Since a ring-like shaped initial contact may not be always anticipated a connector is expected to provide certain degree of self-adjustment (forcing, if necessary, the components to align against each other properly after initial contact occurrence). Correspondingly, some extra torque is usually necessary to secure connectors when initial contact takes place at a single point. That typically is sufficient to correct the mutual positions of the components toward development of a ring-like contact area between the cone frustums. Therefore by the design intent the torque to seal variation has been implied into the design.

And this variation is tough to forecast because actual degree of misalignment is somewhat unpredictable. Actual amount of such misalignment is the function of actual deviations and disturbances occurring for a particular connector but it reveals itself only when the connector's securing begins. True variation of the torques sealed a batch of connectors without disturbances and deviations may be obtained only in ideal (lab like) environment. In real assembly process it is usually masked by the variation coming from self-adjustment. While it is possible to perform statistical inferences for that batch based on lab test it is then nearly impossible to predict how much torque would be necessary to seal same connectors in real-life assembly process as the level of unavoidable disturbances and deviations is predictable merely arbitrarily.

That is why the variation of actual torque to seal observed in the assembly process may become too high. And it may easily overlap available operational window. Here is an example of typical assembly plant scenario. In the framework of implementation of another design change the torque setting at a particular assembly station had been specified at 15NM based on the connectors' parts validation as per relevant PPAP. After a few weeks of relatively smooth operations, something has been changed (either at the body shop or at the tube bundles supplier) but still being within allowed dimensional variation limits - at least relevant reports indicate parts' conformity to the specifications. This caused a slight shift – as much as 15mm - between the tubes' ends to be connected. To make it worse, that extreme shift of the tubes' centers against each other is observed for only 25% of the units – the rest do not demonstrate such problem (for many of them the shift is less than noticeable at 3 mm). Those 25% expose difficulty to perform the hand-start of connectors' thread. That is why about half of them do not receive proper hand-start and correspondingly have not been sealed by prescribed 15NM torque. The repair data indicate that the torque of 23NM (manually applied by brake repairman) is sufficient to fix the affected connectors. The tooling setting allows changing the torque target from current 15NM to 23NM. However the available tooling setting of 23NM has its own scatter, so if changed, the tooling's actual torque may periodically reach up to 25NM. At the same time that PPAP also instructs that torque of 24NM and higher may cause crash of the flare. Therefore that setting of 23NM may cause another problem while fixing the existing one. Eventually that assembly plant has no choice but to deal with those additional repairs while the ends' shift is getting fixed.

Unfortunately, no assembly process can distinguish which connector needs "usual" torque and which one needs "the bumped" one even if precisely controlled DC nut-runners are available. The process is simply not designed to deal with subpopulations. Therefore the less torque to seal variation is available - the better. And similarly to the operator dependency the connector design must be as resilient as possible to such unavoidable noise (deviations and disturbances) from the assembly process.

Regrettably, reasonable extra torque is not always helpful. There are certain known limitations of the self-adjustment capacity of currently produced connectors based on cone-to-cone type of mating [6]. Under certain conditions friction may lock the flare in a misaligned state against the port's seat. Simply put, if the initial contact occurs on a single point, then the flare gets locally squeezed there between the threaded nut and the port. If the effective friction coefficient at that squeezed area becomes greater than certain threshold, then the flare gets locked (pinned against the seat). Such locking inhibits self-adjustment as realignment becomes restricted. A crescent shaped damage of the seat clearly signifies such locked misalignment phenomenon [6]; see Fig 3. Correspondingly, usual application of extra torque now is not able to correct poor initial

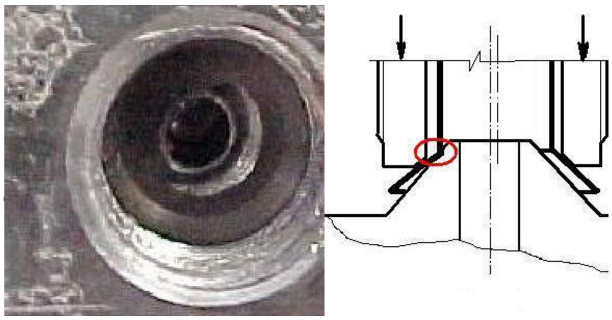


Fig. 3

contact into an uninterrupted ring-like line. At the same time reasonable torque increase may also turn out to be insufficient to provide the deformation, which becomes required to close the gap between the sealing surfaces. In this case further torque increase leads to squashing of joint's components, which in turn may permanently preclude development of the seal. Then actual torque to seal in this situation becomes infinite, which make any statistical inferences useless. It is important to stress that such locking in the misaligned condition is quite possible when connector's components meet their specification limits.

Since it is either impossible to predict (in certain and relatively frequently occurring conditions) actually necessary torque or its predicted variation may become greater than the operational window, then it is obviously impossible to expect the 99.9% first pass yield. Since torque variation is implied into the cone-to-cone mating, there is no other choice but to mull over elimination of such failure mode via abolishing of that mating type. Thus alternative mating types resilient to the misalignment (characterized by either less or no torque variation) ought to be considered.

Geometry fundamentals teach that there are only two options available when a circumference is sought as common result of two simple bodies' intersection. They are: sphere-to-sphere interaction and sphere-to-cone one. And both the options have been already incorporated into known arrangements proposed for brake tubing connectors. The US Patent 1894700 [7] stipulates the usage of a sphere-to-sphere mating and the US Patent 8152204 [8] – sphere-to-cone one.

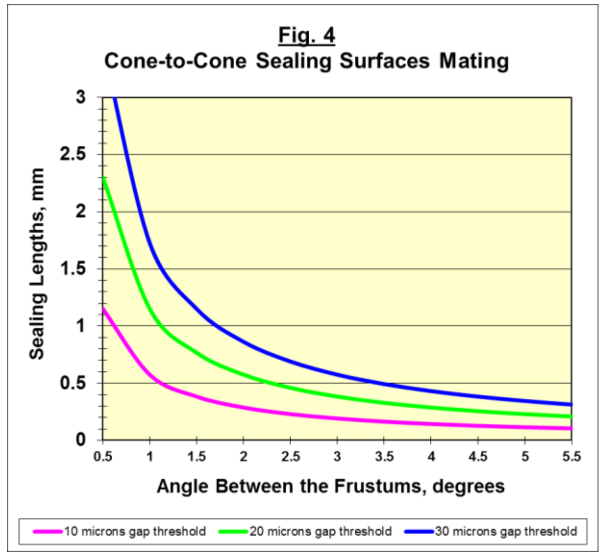
Delivering the initial contact between the flare and the seat which is always shaped as a ring is not the only advantage of a sphere-to-cone mating. Another important advantage is much wider operational window with respect to the sealing length [9]. The sealing length is nothing else but the width of the mutual contact ring (the mating zone between the seat and the flare).

In case of standardized cone-to-cone mating the sealing length (when the cones are properly aligned) is the function of the difference between the cones' angles. In case of a sphere-to-sphere mating the sealing lengths is the function of the relative difference between the spheres' radii. It is important to stress that the sealing length variation absorbs the variations of two mating components in both the cases above. In contrast, in case of sphere-to-cone mating, the sealing length is the function of the sphere radius only. It is always easier to control the variation coming from only one source than the one coming from two or more sources. And on top of that, the sealing length transfer function in case of a sphere-to-cone- mating is much favorable comparing to both the cone-to-cone mating and the sphere-to-sphere one.

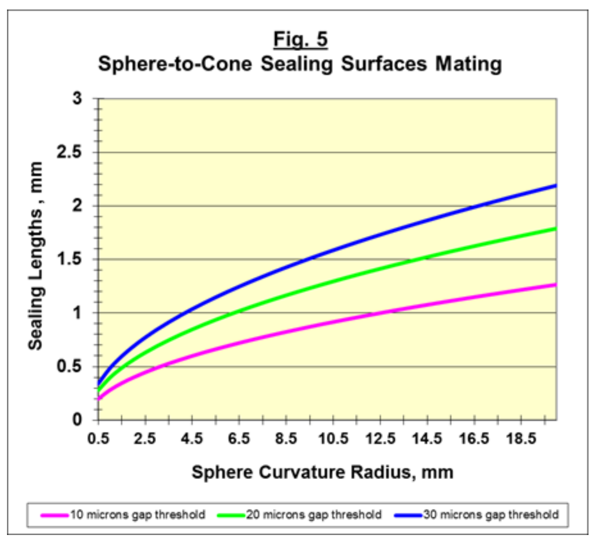
The sealing length is the function of connector's parameters at the mating zone. It is industry-wide practice to use simplified quasi-static rigid model and evaluate the sealing length geometrically via a "closable" gap. Instead of taking into account the contact stresses and corresponding local deformations (Hertzian contact mechanics approach loaded with complicated formulas) it is presumed that a small enough gap will be "closed" by the securing torque generated deformations. Accordingly in framework of that simplified approach a certain threshold value of the gap has been established. Then all the gaps less than that threshold are pronounced as "no gap" zone upon completion of the connector securing process. And the ring's width which is associated with the calculated gap less than that threshold (within intended contact zone) - is the sealing length in the context of that simplified approach. The threshold gap is usually assumed at 25 microns. Sometimes a stricter threshold of 10 microns may be utilized in order to take into account more severe cases with extra propensity for local defects and surface deviations and imperfections. Thus in the context of that approach the sealing length can be calculated via mating zone geometrical parameters. Correspondingly the transfer function in this context is the relationship between the sealing length and the relevant flare and seat dimensions.

Analysis of the transfer functions and operational windows comparison [9] revealed fundamental difference between cone-to-cone and sphere-to-cone types of mating. See the graphs below (Fig 4 and Fig 5 [9]).

The cone-to-cone transfer function clearly shows that relatively long sealing lengths are available only when fairly strict tolerances have been imposed. Moreover, the longer sealing length we want the less angles' differences we will have to maintain.



In contrast, sphere-to-cone mating delivers longer sealing lengths with virtually any radius suitable for connector's dimensional stack-up. (The sphere must be bigger than the tube's outer diameter, therefore such suitable radii range should be around 4mm to 15mm, depending on the tube size).



For example a sealing lengths span of 0.5...1.0mm in case of cone-to-cone mating (assuming 25 microns gap threshold) needs rigorous range of 1.5...2.5 degrees for the angles' difference. Same sealing lengths span in case of sphere-to-cone mating is supported by the range of radii from 1mm to 5mm. And since the suitable radii range just begins around 4mm, actual sealing lengths in a sphere-to-cone based connector may be expected nearly always better than that 1mm. Additionally, that premium 1+ mm sealing length range requires no upper specification limit for the sphere radius. Should such longer than 1 mm sealing length become desired in a cone-to-cone based connector then painstaking (and probably cost prohibiting) specification of 1.5 degrees or less for the angles' difference would be required. Hence the advantage of sphere-to-cone operational window is evident.

Let us use the standardized connectors as a benchmark (same 25 microns gap threshold is assumed). The ISO and JASO/SAE nominal sealing length is expected to be in the neighbourhood of 500 microns (when both the frustums are perfectly aligned and their dimensions are right on their specification targets). As to the tolerances, the standards stipulate the range of the angles' difference between 1 and 5 degrees at one side. This range corresponds to the sealing lengths span from approximately 300 microns to 1.4 millimeters (as shown on Fig.4).

Note, that sealing length of 1.4mm (which is sequential to the JASO/SAE specification limit delivering best result) is comparable to the lower end performance in a sphere-to-cone based connector. Mentioned above minimum suitable radius of 4 mm delivers 900 microns of the sealing length and 5mm one - approximately 1mm (as per Fig 5). It means that a sphere-to-cone based connector is much easier to seal than a cone-to-cone based one: nearly worst performance level with respect to the sealing length of the first is comparable to the latter's best.

The sphere-to-sphere sealing length transfer function is not much different comparing to the cone-to-cone one. Fig 6 clearly shows no advantages against the latter.

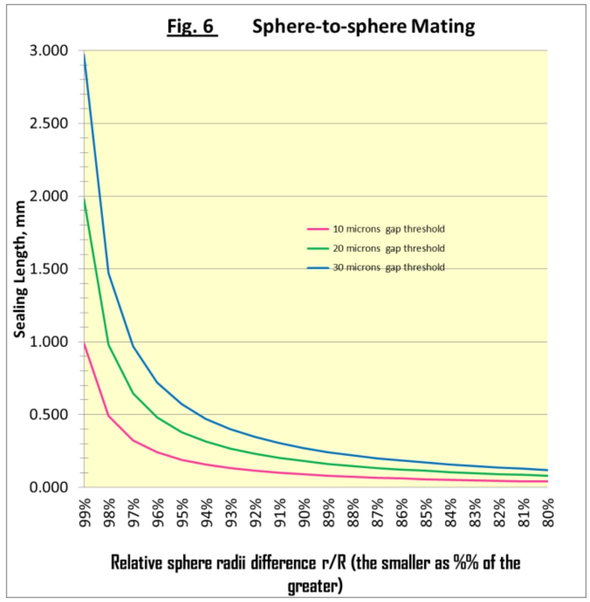


Fig. 6 Sphere-to-sphere Mating

Let us to perform another comparison to demonstrate lack of advantages of a sphere-to-sphere mating over a cone-to-cone one with respect to the sealing length. (Same 25 microns threshold gap is assumed). The sphere-to-sphere mating delivers 0.5 mm sealing length (same as the standardized cone-to-cone mating nominal) when the relative difference between the spheres is 95%. Assuming for simplicity's sake that the greater sphere radius is at 10mm, the small one should be 9.5mm. Such 0.5mm of radii difference is not easy to measure and therefore assure. Should longer than 1 mm sealing length become desired the sphere differences would be 97% and stricter. In the same as above 10mm example the nominal difference between the diameters would be 0.3mm – most likely its monitoring would be cost prohibiting.

Projecting the standardized connectors' sealing lengths range (300 microns to 1.4 millimeters) into the case of sphere-to-sphere mating, the sphere difference limits should be expected at approximately 92% to 98% (as per Fig.6).

Using the same 10mm greater sphere radius example, the smaller one should then be between 9.2mm and 9.8mm if the greater one would not have dimensional variation at all. However both spheres' variations affect the sealing length (not just one as in the case of sphere-to-cone mating). That is why in practical terms the tolerances corresponding to these 92% to 98% limits must be balanced out between both spheres. It may be accommodated, for example, as follows: 10.0 mm +0.3/-0.1 for the greater sphere and 9.6mm +0.1/-0.1 for the smaller one. (Then the difference between 10.3 mm and 9.5 mm yields 92% of relative difference and the combination of 9.9 and 9.7 – 98%). It is clear that controlling such tolerances most likely would be more challenging than the ones in case of the JASO/SAE connectors. Just compare angular specification allowance of +/- 2.5 degrees with those 0.2mm and 0.4 mm of the radial allowances. And if we want to accommodate more relaxed sphere radii allowances we will face the sealing length shorter than one in standardized cone-to-cone based connectors.

Though a sphere-to-sphere mating has no advantages over a cone-to-cone one with respect to the sealing length, it does have the advantage with respect to the alignment. There is no need to spend the torque reaching desired ring-shaped contact - it is always present. It is important to stress that sphere-to-sphere mating requires no realignment once the initial contact occurred. Consequently the variation of the torque to seal is expected to be significantly less for the latter.

A sphere-to-cone mating has exactly the same resilience to the misalignment as a sphere-to-sphere one – no self-adjustment needed once initial contact has occurred. Yet on top of that, its sealing length is longer, more stable and enables wider operational window. That is why the usage of a sphere-to-cone mating is superior. Thus, future connectors' development aiming for perfection must be focused on incorporation of such sphere-to-cone mating.

Sphere-to-cone mating incorporation is also promising to boost development of modern quick connectors. Many known quick connectors failed to deliver acceptable sealing robustness simply because they incorporated conventional cone-to-cone mating. This is the property of conventional cone-to-cone mating which prohibits simultaneous execution of the alignment and the clamping. Faster clamping is desired but certain time must be allocated to realign the flare and the seat to correct poor initial contact when necessary. Likewise relatively slow alignment process (moving the parts after initial contact occurrence) forbids momentary clamping. That makes a quick connector arrangement based on cone-to-cone mating quite complicated as the "latch" must get actuated only when the alignment completed. Since sphere-to-cone mating is resilient to misalignment, no moving is needed after initial contact. Accordingly the clamping function may be executed immediately upon initial contact occurrence – by simpler mechanisms and faster. On top of that, longer sealing length of the sphere-to-cone mating may require less clamping force providing additional opportunity to simplify the arrangement.

A quick connector incorporating a sphere-to-cone mating must also provide resilience to the operator dependency. Accordingly, only simple ("switch-on like") actuation of the connector's clamping mechanism by operator is wanted. Then the desired 99.9% first pass yield at each connector will become a reality.

A robust cost effective quick connector incorporating sphere-to-cone mating is a realistic solution to the modern lean and process excellence challenges. And it should become the mainstream of forthcoming perfection of the automotive brake tubing connectors.

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CONTACT

Stanislav I. Pliassounov, Ph.D., P.Eng. has 6 patented inventions regarding brakes and more than 20 years of experience on this field. When employed by Ford Motor Company in 1998-2007 he was focused on sealing robustness improvement with respect to hydraulic brake connectors. He is also proficient in Design for Six-Sigma – certified Master Black Belt. Dr. Pliassounov can be reached via e-mail: spliassounov@cogeco.ca