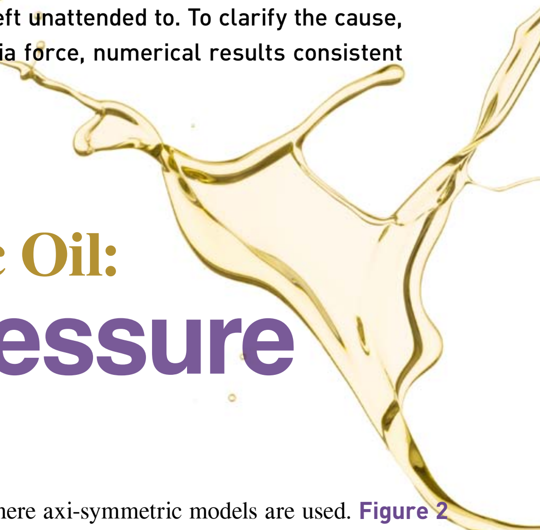
Hydraulic machines such as hydraulic motors are comprised of many precision parts. One of them, called a plug, is a part looked like a short-headed bolt. A potential problem with plugs is the oil leakage from the bearing surface. Its primary cause is increasing operating oil pressure. Clarifying the mechanism of the occurrence of leakage is very important from the perspective of reducing the size and weight of hydraulic machines. As a first step, this article will make clear the relationship between torque applied to the head and the generated axial force in the fastening process. Using the results, contact pressure distributions on the bearing surface are evaluated via finite element analysis. Next, the relationship between the axial force of the plug and the amount of hydraulic pressure are shown. Furthermore, explained are the specific distribution patterns of contact pressure under plug head, which significantly affect the sealing performance. Incidentally, though the leakage from the plugs in actual hydraulic machines does not occur frequently, it is not something to be left unattended to. To clarify the cause, focusing on the increasing rate of hydraulic pressure and regarding the effect as inertia force, numerical results consistent with the troubles in actual machines are obtained, which will be introduced in this article.



"Fastener Troubles, Causes & Solutions" Series

Leakage of High Pressure Hydraulic Oil: Is the Inertia of High Pressure the Driving Factor?

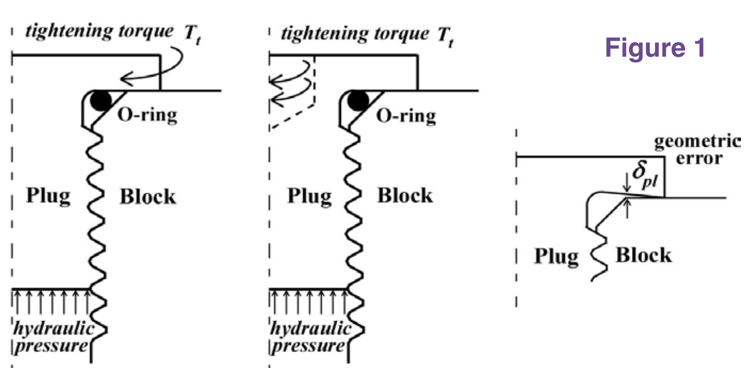
by Toshimichi Fukuoka

Shape and Fastening Characteristics of Plugs Used for Hydraulic Machines

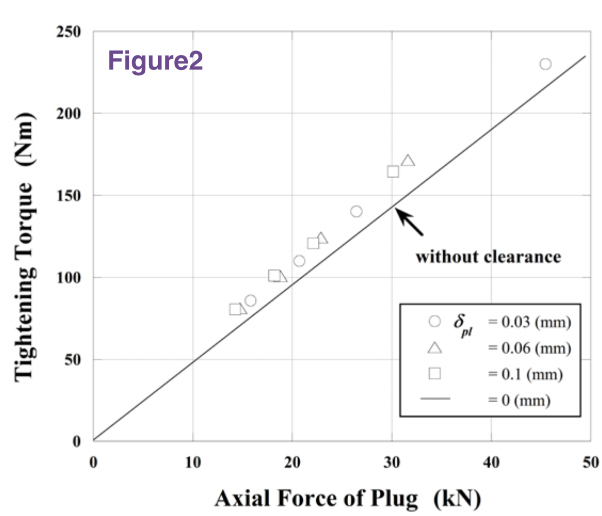
Threaded fasteners called plugs are extensively used in hydraulic machines. **Figures 1 (a)** and **(b)** show a typical shape of the plug. They are respectively called hexagon plugs and hexagon socket plugs. Generally, the former ones are fastened with a torque wrench, and the latter ones are with a hexagon wrench. The shape of both plugs resembles a bolt but the head height is quite short. Consequently, the stiffness in the axial direction is low, and as stated later, the bearing surface shows characteristic contact pressure distributions. The accuracy of the dimensions of plugs used in hydraulic machines are generally not very high. Therefore, there may be geometric errors δ_{pl} under the plug head, as shown in **Fig 1. (C)**. Such geometric errors, together with the effect of low head height, could produce a characteristic relationship between torque and axial force during fastening, which is different from that of ordinary bolts.

To make this point clear, finite element analyses were conducted to obtain the relationship between torque and axial force. Details of the analytical methods are omitted here. The analytical target is parallel pipe threads (G1/2) , where axi-symmetric models are used. **Figure 2** shows the analytical results. The parameter is the geometric error δ_{pl} around the bearing surface, shown in **Fig. 1 (C)**. Here, δ_{pl} =0 represents the case without clearance on the bearing surface at the initial state. The relationship between axial force and torque becomes naturally linear when the clearance does not exist. In the case with the existence of clearance, on the other hand, both are almost linearly related regardless of the extent. However, the amount of axial force generated for the same torque becomes slightly smaller, comparing to the case without clearance. The reason is as follows. If there is clearance as shown in **Fig. 1(c)**, the equivalent diameter of friction on the bearing surface becomes larger than the case without clearance. As a result, the proportion of torque for generating axial force decreases because of the increase of the proportion consumed by friction. Regarding the influence of δ_{pl}, the head has low stiffness, and therefore it seems no significant influence appears because the bearing surface is in full contact even under small axial force. For the cases with/without clearance, values of nut factor K numerically

Figure 1



(a) Hexagon Head Plug (b) Hexagon Socket Head Plug (c) Geometric error in Hexagon Head Plug



obtained are shown below in a similar manner to bolt fastening. Here, nut factor represents the relationship between applied torque and generated axial force. The product of axial force and thread nominal diameter multiplied by nut factor becomes the fastening torque.

Case without clearance:

K=1.029μ+0.0185 (hexagon head plug)

K=1.027μ+0.0186 (hexagon socket head plug) (1)

Case with clearance:

K=1.157μ+0.0171 (hexagon head plug)

K=1.018μ+0.0171 (hexagon socket head plug) (2)

In the above expressions, it is presumed that the coefficients of friction μ of the threaded surface and the bearing surface of a plug head are equal. Considering that actual plugs possibly have some geometric errors, Expression (2) is more practical. Then, substituting 0.15 for μ in Eq.(2), K becomes 0.190 and 0.170 respectively, which are slightly smaller than the nut factor of 0.2 for a bolt-nut connection.

Distribution Characteristics of Bearing Surface Pressure & Sealing Performance

Figure 3 shows the numerical results of bearing surface pressure distribution obtained by using the same analytical model used for analyzing the relationship between torque and axial force. The target of analysis is a plug with thread nominal diameter of G1/2 clamped by axial force of 25.7kN. Unlike ordinary bolts, only the inner side of the bearing surface is in contact. Therefore, the contact pressure sharply increases inward. This phenomenon results from a large bending deformation of the head by the action of axial force because of the low stiffness of the plug head. In this case, it is predicted that the sealing performance is achieved owing to high contact pressure appearing around the inner periphery of the bearing surface. The Figure shows the results for a hexagonal plug and hexagonal socket plug. No distinct difference is observed between the two. The patterns of contact pressure distributions in the initial fastening state, shown in **Figure 3**, won't change unless the effect of geometric errors appears due to a substantial reduction of the axial force. From the above, the amount of the axial force of plug under the action of hydraulic pressure can be regarded as a criterion for evaluating the sealing performance.

Figure 4 shows the relationship between the magnitude of hydraulic pressure acting on the plug and axial force. The analytical objects here are two plugs with nominal diameter of G1/2 and G1/4. The axial force in the initial fastening state is the value obtained corresponding to the recommended torque. Although exhibiting a complicated behavior for varying geometric errors, the axial force of plug wholly drops almost linearly with increasing hydraulic pressure. Referring to the results shown in the Figure, however, it is considered that the leakage does not occur over the widely used range of 25MPa-35MPa of hydraulic pressure. Meanwhile, in some actual hydraulic machines, hydraulic pressure acts dynamically and it could reach the maximum value in a very short period of time. In order to examine the influence of inertia in those cases, in the following section one-dimensional spring models are employed to evaluate the dynamic effect of hydraulic pressure upon the axial force of the plug.

Evaluation of the Influence of Inertia Using One-dimensional Spring Model

The equation of motion excluding the effect of viscosity is as follows.

[M]{ü}+[K]{u}={R} (3)

Figure3

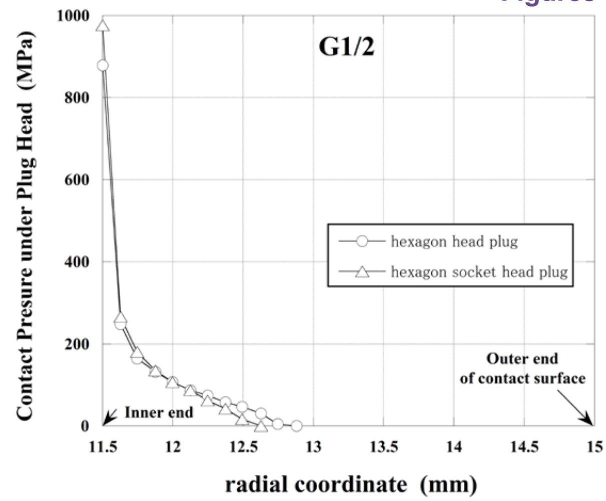


Figure4

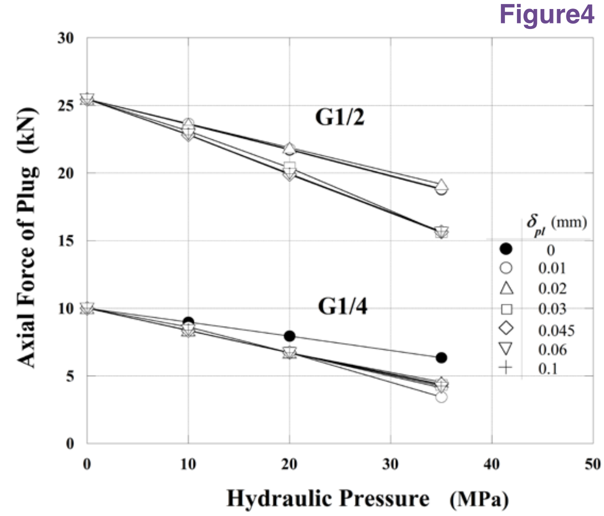
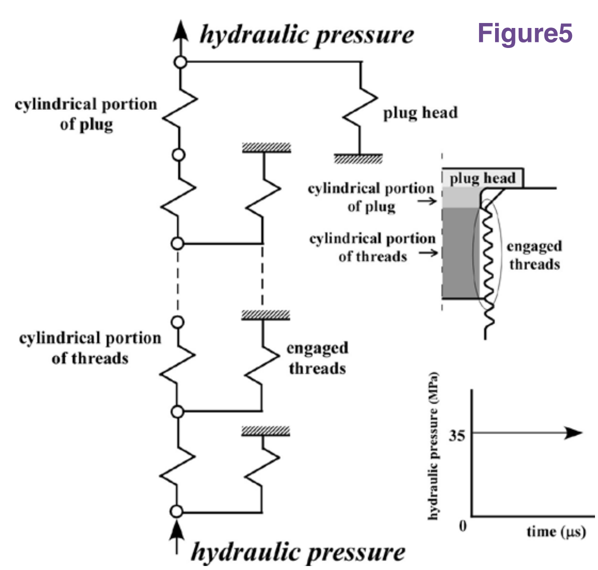
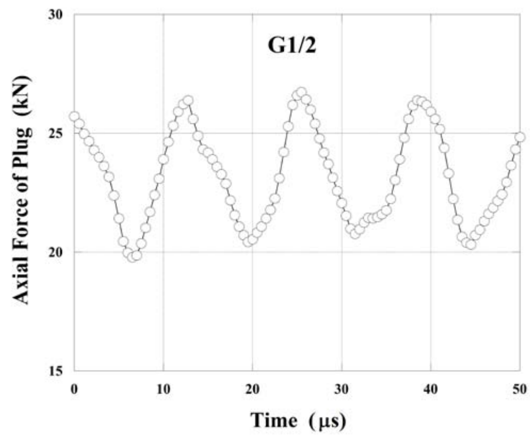


Figure5



[M] and [K] are mass matrix and stiffness matrix respectively, [R] is load vector, {ü}and {u} are acceleration vector and displacement vector. Incidentally, inertia does not act in the fastening process, so the term {ü} disappears. In **Figure 5**, the female-thread portion, which is machined on the main body of the hydraulic machine, is regarded as a rigid body, and the plug is divided into a threaded portion and a cylinder portion, and then it is represented using one-dimensional spring models. Nominal diameter of the plug is G1/2, and its total length is 21mm and the plug is tightened with axial force of 25.7kN. The pressure of the hydraulic oil is 35MPa. As shown in the Figure, hydraulic pressure is applied stepwise assuming the most severe conditions. **Figure 6** shows the variations of the plug force with time. In order to consider the stiffness of the female-thread portion, the stiffness of engaged threads is simply reduced by half in the actual analysis. As is obvious from the Figure, the axial force of the plug substantially oscillates against the initial value due to the dynamic effect of hydraulic pressure. It is considered that the sum of the axial force shown in **Figure 4** and the current result corresponds to the variations of axial force in actual machines. The aforementioned analytical results agree with the phenomenon in the actual machines in that the static action of hydraulic pressure causes the reduction of plug force and then the addition of dynamic effect may cause the leakage, though the probability is low.

Figure6



Conclusion

Sealing problems of threaded parts can be regarded as the problem whether the axial fastening force is adequately applied against the active inner pressure to maintain the contact pressure. Furthermore, the influence of inertia must be taken into account if the inner pressure rises sharply. In order to precisely evaluate the sealing performance of the plugs used in hydraulic machines discussed in this article, numerical models considering the effect of engaged threads would be necessary. On the other hand, if the purpose is to evaluate how much hydraulic pressure is allowed while maintaining the sealing performance, the one-dimensional spring models proposed here are considered to be a practical approach. The next article will discuss a pipe flange used in the connecting portion of pipelines, which is a typical example that the axial force of a bolt determines the sealing performance.

Reference

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