

Design and Evaluation of Quadruple System Protection Valve Piston Assembly using Finite Element Method

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ABSTRACT

This paper presents a systematic procedure for design and evaluation of snap fit for Quadruple System Protection Valve (QSPV) piston assembly. The QSPV piston is assembled with housing by means of snap joint. Snap joints are a very simple, economical and rapid way of joining two different components. All types of snap joints have in common the principle that a protruding part of one component, e.g., a hook, stud or bead is deflected briefly during the joining operation and catches in a depression (undercut) in the mating component. After the joining operation, the snap-fit features should return to a stress-free condition. The joint may be separable or inseparable depending on the shape of the undercut; the force required to separate the components varies greatly according to the design. It is particularly important to bear the following factors in mind when designing snap joints: Mechanical load during the assembly operation and force required for assembly.

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1. INTRODUCTION

One of the most important rules of design for assembly (DFA) is to replace loose fasteners, such as bolts, rivets, etc, by integrated fasteners [1]. As a consequence, many companies have set up programs to eliminate separate fasteners from their products. The use of integrated fasteners is facilitated by the ease of incorporation of special features into plastic molded parts. For example, in the automotive industries where snap fits were frequently limited to tasks such as attaching trimmings and electrical components, engineers are now looking for other applications [2]. However, snap fixtures have often been plagued by their reputation for failure. Furthermore, under the pressure of ecologists, engineers must take into account the complete product life cycle and in particular the recycling aspect. This means that designers should examine fastenings that can be separated. These problems can be alleviated to a large extent by proper design.

Most of the major plastic resin suppliers provide snap fit design booklets [3,4]. These usually include design formulae for calculating cantilever hooks and some general information such as the minimal radius at the base of the beam, one can

also find in the literature a technical analysis of snap fits. For example, Lee et al. [5] and Reiff [6] discuss the discrepancies between predicted behavior and actual performances of snap fits. They use finite-element analysis to evaluate the deflection of short cantilever beams and deformations on the wall base. They introduce the concept of a deflection magnification factor that can be used to correct the classical Euler/Timoshenko beam theory [7]. Although Reiff [6] gives some guidelines for snap fit design when maximum snap retention is required, little information on design, such as selection of the best beam shape (constant cross-section beam, tapered beam, "L" shape, "U" shape, etc.), ease of robot assembly, possibility of dismantling, is available. It is clear that the fastening community had traditionally concentrated on threaded and other fasteners, and could not provide all the answers for snap fits. In this paper a systematic method for design of annular snap fit with regard to several criteria such as available space and ease of dismantling is presented. It is hoped that this will help to design the best snap fit for the application.

2. CLASSIFICATION SNAP JOINTS

Fig.1 represents a simple classification according to the main stress in the joint: Fig.1-a annular snap joints are rotationally symmetrical and involve multi-axial stresses, Fig.1-b for

snap joints shear stresses carry the load; Fig.1-c in cantilever snap joint the load is mainly flexural. The paper will focus on the snap fits based on an annular snap joint.

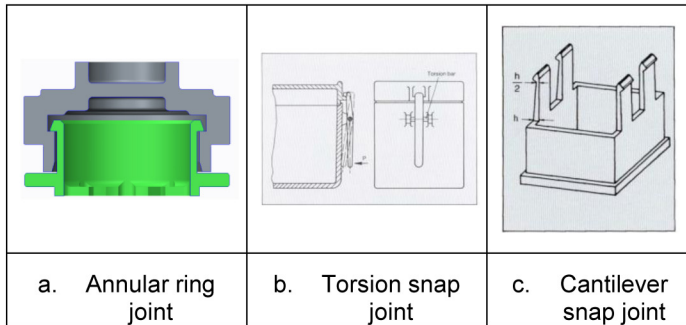


Figure-1. Types of Snap joints

3. DESIGN PROBLEM

In commercial vehicle air braking system, Quadruple System Protection Valve (QSPV) as shown in Fig.2 is used to distribute the air from air processing unit to front axle, rear axle pneumatic brake circuits, parking brake circuits and to the auxiliary applications like suspension system, door operating system and clutch operation. The valve should meet the regulatory compliance as per IS 11852 norms. If the piston assembly joint fails it will not serve its purpose. The causes for the failure are:

- Excessive mechanical load during the assembly operation.
- High interference between housing and piston will lead to permanent plastic deformation.

This paper provides solution for the above problem by introducing a method called annular snap joint. Also it provides a systematic approach to evaluate the design using finite element method.

3.1. QSPV Piston Assembly

The Fig.2 shows the Piston assembly used in Quadruple System Protection Valve (QSPV) is made of plastic material which consists of piston and retainer. The Piston is made up of PBT 30% GF and the housing is made up of Nylon 6 30% GF material. Both materials have different material properties in terms of flexural modulus, tensile strength and density. The piston is assembled with housing by means of snap fit mechanism to achieve the flexibility in integral locking feature.

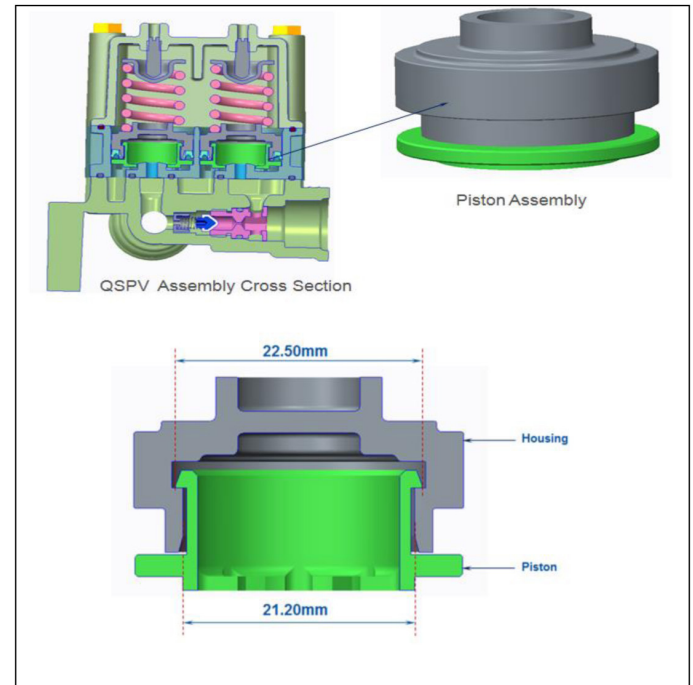


Figure-2. QSPV Piston Assembly

4. SNAP JOINT MECHANISM

The snap-fit is a mechanical joint system where part-to-part attachment is accomplished with locating and locking features (constraint features) that are homogenous with one or the other of the components being joined. Joining requires the (flexible) locking features to move aside for engagement with the mating part, followed by return of the locking feature toward its original position to accomplish the interference required to latch the components together. Locator features, the second type of constraint feature, are inflexible, providing strength and stability in the attachment. Enhancements complete the snap-fit system, adding robustness and user-friendliness to the attachment.

4.1. Concept of Annular Snap Joints

The annular snap joint is a convenient form of joint between two rotationally symmetric parts. In annular snap, a largely stress-free, positive joint is normally obtained. The joint can be either detachable as shown in Fig.3 difficult to disassemble or inseparable as shown in Fig.4 depending on the dimension of the bead and the return angle. Inseparable designs should be avoided in view of the complex tooling required (split cavity mold).

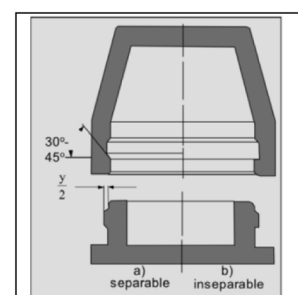


Figure-3. Annular Snap joints



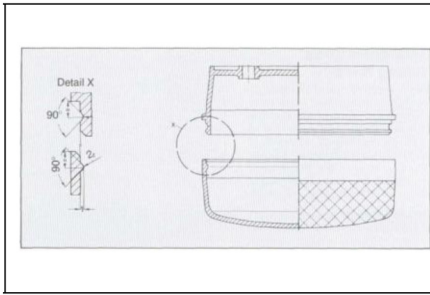


Figure-4. Annular snap joint on a lamp housing

The allowable deformation should not be exceeded either during the ejection of the part from the mold or during the joining operation.

The permissible undercut as shown in Fig. 5 is limited by the maximum permissible strain.

Permissible undercut = Permissible strain (ϵ) x diameter of the joint (d)

This is based on the assumption that one of the mating part remains rigid. If this is not the case, then the actual load on the material is correspondingly smaller. (with components of equal flexibility, the strain is halved i.e. the undercut can be twice as large).

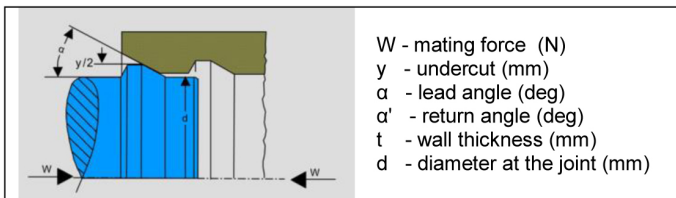


Figure-5. Annular snap joint

5. DEFLECTION FORCE, MATING FORCE

The determination of the mating force W is somewhat more complicated for annular snap joints. This is because the snap-fitting bead on the shaft expands a relatively large portion of the tube as shown in Fig. 6. Accordingly, the stress is also distributed over a large area of the material surrounding the bead.

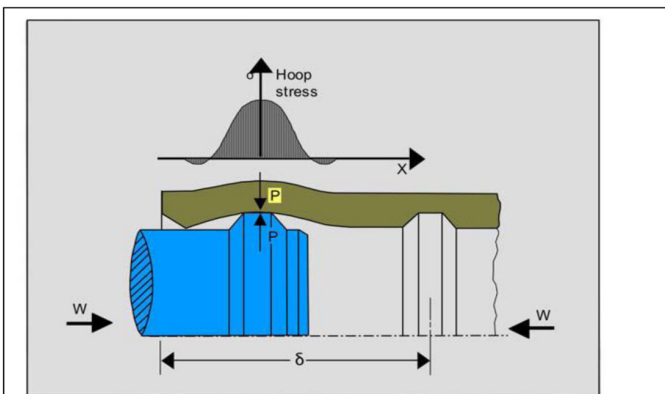


Figure-6. Stress distribution during joining

Experimentally proven answers to this problem are based on the "theory of a beam of infinite length resting on a resilient foundation." Two extreme cases are depicted in Fig. 7.

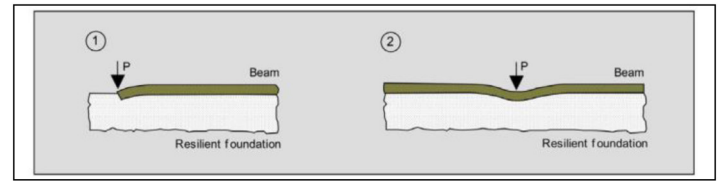


Figure-7. Beam resting on a resilient foundation

1. The force P is applied at the end of the beam. (This corresponds to a snap joint with the groove at the end of the tube.)
2. The force P is applied at long distance from the end of the beam (This is equivalent to an annular snap joint with the groove remote from the end of the tube)

A somewhat simplified version of the theory may be expressed as follows for joints near the end of the tube:

$$P = y \cdot d \cdot E_s \cdot X \quad (1)$$

Where

P - Transverse force (N)

y - Undercut (mm)

d - Diameter at the joint (mm)

E_s - Secant modulus (N/mm²)

X - Geometric factor

The geometric factor X takes into account the geometric rigidity.

As far as the mating force is concerned, friction conditions and joint angles must also be taken into consideration.

$$W = P \frac{\mu + \tan \alpha}{1 - \mu \tan \alpha} \quad (2)$$

Where μ = coefficient of friction

α = lead angle

The geometric factor, assuming that the shaft is rigid and the outer tube (hub) is elastic is as follows:

$$X_N = 0.62 \frac{\sqrt{(d_0/d - 1)/(d_0/d + 1)}}{[(d_0/d)^2 + 1] / [(d_0/d)^2 - 1] + \gamma} \quad (3)$$