UGP: A Universal Grip Port for the Manipulation of Everyday Objects by Humans and Machines

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Abstract. A standard mechanical port on everyday objects would allow a machine to take hold of a door, a drawer or an appliance without reproducing the human hand. Dexterous hands and learned manipulation provide part of the solution, but the main benefits are lost if every new object still requires the machine to perceive where it can be grasped, plan a multi-contact grasp and be trained on its variations. I propose to move the difficulty out of the robot and into the interface. The object carries a passive stud, 36 mm (1.42 in) across and 20 mm (0.787 in) tall, solid, with no moving parts, which a person uses as a knob. The machine carries a socket that accepts a lateral error of ±5 mm (±0.197 in), locks onto the stud with six bearing balls wedged by a spring-loaded sleeve, and seats on three ribs in V-grooves intended to constrain the relative pose in all six degrees of freedom. Aligned insertion needs a push of less than 30 N (6.74 lbf); indexing additionally uses controlled wrist torque and the collar actuator, with 30 ± 1 N (6.74 ± 0.225 lbf) net rib load; holding requires no energy; release of an externally unloaded port is estimated to require less than 110 N (24.7 lbf) over 5.8 mm (0.228 in), subject to the preload and friction assumptions below. The design is defined by a parametric CAD model and preliminary calculations; it has not yet been built or tested. Retention without friction is established from rest by an energy barrier, but not against shock or vibration, and the complete load path to the object remains unverified, so the calculated contact limits are not certified working loads. The assembly order, the fits and every fastener are specified, and the geometry of each is checked; drawings, supplier spring data and tests are not. If the stud profile and the free space around it are fixed by an open standard, compatible machines could operate prepared objects within their qualified load, perception and motion capabilities, and robots can be adapted to human environments without waiting for hands to be built or trained.

1. Introduction

Robots are leaving the cells and conveyors they were built around and are being sent into factories and warehouses, building sites and farms, shops, hospitals and homes. Whatever their form, humanoid or not, they meet a world in which interfaces were shaped for one end-effector, the human hand. The hand has 27 bones, more than twenty degrees of freedom and, in its glabrous skin alone, about 17,000 tactile units [1]. Robotics has not reproduced it, an instance of Moravec's observation that the sensorimotor skills which are effortless for people are among the hardest to give to machines [2]. The usual response is to make the machine more like a person: dexterous hands, tactile skins, and manipulation policies trained on large sets of demonstrations. Whatever its progress, this approach keeps one structural cost: every object is a new problem. A handle must be recognised as a handle, and a grasp must be found, executed through several simultaneous contacts, and held against slip. Uncertainty is added at each step (Figure 1).

The alternative is an interface based on geometry instead of dexterity, intended to let compatible machines take hold of prepared objects without hands or object-specific grasp training, within their sensing, actuation and load capabilities. Industry has done this before. Freight containers carry identical corner fittings, specified by an international standard, onto which the twistlocks of cranes, ships and trucks engage [3, 4]. Machine tools change cutters through a standard hollow taper [5]. Industrial robots accept tools through a standard mounting plate [6]. Spacecraft of different origins can dock because the docking interface is published [7]. In each case a passive feature on the many things is matched by an active mechanism on the few machines, and the value comes from the standard.

In this paper I propose such a port for everyday objects, the Universal Grip Port (UGP). I define one size, UGP-M: geometry, mechanism, docking sequence and calculated load ratings. A parametric CAD model and a calculation file accompany the paper. Figures 2 to 4 are sections of that model, and the reported numerical results are generated from that file. Checks verify specified model cases; they do not establish manufacturing feasibility or physical performance. The port is designed in millimetres, and every quantity is followed by its imperial equivalent in parentheses, rounded to three significant figures; the drawing of Figure 2 carries inches in brackets, as a dual-dimensioned drawing does, and the limit dimensions of Table 3 are metric only, because a drawing is toleranced in one system.

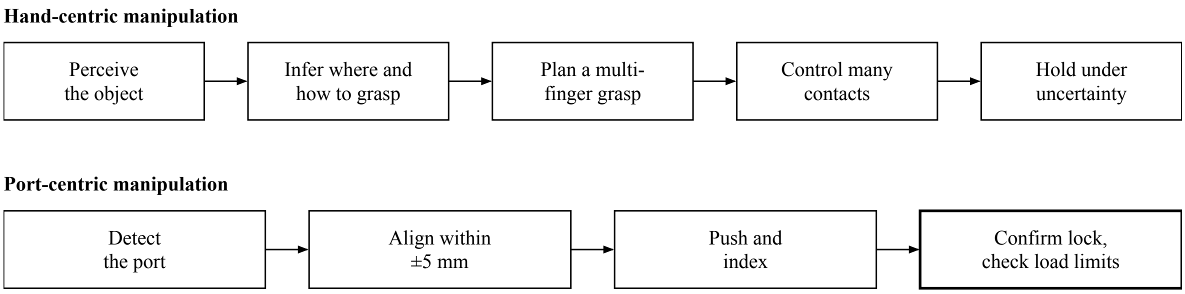

Figure 1. Hand-centric manipulation solves a chain of problems for each object. The proposed sequence aligns, pushes and indexes, then confirms the lock. Geometry constrains the seated pose; the robot cameras identify the object orientation. The load limits remain preliminary model results.

2. Requirements

I require the following of a port that is to be fitted to a very large number of objects and used by people and machines alike.

1. *Passive on the object.* No moving part, no power, nothing to adjust. The object side must cost what a knob costs.
2. *Usable by a hand*, unaided and without instruction.
3. *Tolerant capture.* The machine must be allowed an error of several millimetres and dock by axial approach followed, when needed, by the bounded torque-assisted indexing procedure.
4. *Positive lock, without power.* Once made, the connection must hold with the machine switched off, and the load must not be able to open it.
5. *Defined pose.* When locked, the pose of the object relative to the machine must be fixed in all six degrees of freedom, so that the machine can compute instead of feel.
6. *Rated.* The connection must carry a declared load on which a planner can rely.
7. *No cavity on the object*, since cavities collect grease and dust.
8. *Small, and fixed with an existing fastener*, so that existing objects can be converted.

3. Stud on the Object, Socket on the Machine

Electrical connectors put the receptacle on the device. I do the opposite, for four reasons. A protruding stud is already a knob, which answers requirement 2 by shape. The stud avoids a large receiving cavity; its small tag recess must be filled and sealed to meet requirement 7. Its main body can be turned; the 18 grooves, rounded roofs and mounting spurs require additional forming or machining operations. Casting and moulding are possible routes for separately qualified material variants, not substitutes for the hardened-steel contact specification used in the calculations. And every moving part is kept in the wrist of the machine, where it is protected and maintained with the machine. An object needs one stud per handle; a machine needs one socket.

The price of this choice is that the stud stands 20 mm (0.787 in) proud of the surface. Where that is unacceptable the same profile can be set in a recessed cup. The standard must therefore fix two things: the profile of the stud, and the free space around its axis that the socket occupies: a cylinder 44 mm (1.73 in) in diameter over the first 31 mm (1.22 in) from the mounting surface, and 63 mm (2.48 in) in diameter beyond, each enlarged by the lateral error allowed during the approach.

4. The Stud

The UGP-M stud (Figure 2) has a mainly rotational body 36 mm (1.42 in) in diameter and 20 mm (0.787 in) tall. From the mounting face outward it consists of a flange 3 mm (0.118 in) thick, a neck of 11.3 mm (0.445 in), and a head of 16 mm (0.630 in) that ends in a 45° lead-in cone. The underside of the head is a toroidal seat of radius 3.75 mm (0.148 in), that is 0.75 times the diameter of the 5 mm (0.197 in) locking balls (Section 7.2). The face of the flange carries 18 radial V-grooves, 90° wide and 1.2 mm (0.0472 in) deep, spaced at 20°, and between them the land is not flat but a roof with planar portions at 15° and rounded crests and shoulders. Its sharp-edge envelope is 0.49 mm (0.0193 in) high at the outer end, an upper bound rather than the finished crest height, over the band of radius 14 to 17.5 mm (0.551 to 0.689 in) that a rib can touch. The rounded roofs guide a rib toward a groove; deterministic departure from a crest requires the torque-assisted procedure of Section 6. A recess 6.4 mm (0.252 in) across and 1.5 mm (0.0591 in) deep in the top of the head takes an on-metal proximity tag 6 mm (0.236 in) in diameter and up to 1.4 mm (0.0551 in) thick (Section 8). Tags of this diameter are sold in several thicknesses; one supplier lists 0.86 mm (0.0339 in) in its data table and 0.85 mm (0.0335 in) in its description [16]; the ordered part requires a confirmed supplier drawing, and the recess is not sized on any one product. The stud is held from behind by one screw in a blind M4 × 0.7 thread. The CAD draws that hole at its 4 mm (0.157 in) major diameter over 9 mm (0.354 in), with neither the thread form nor the drill-point allowance; the tapping drill and the usable thread length belong on the production drawing (Section 9.2). Usable engagement and bottom clearance must be specified on the manufacturing drawing. A #8-32 version needs a separate thread and pilot specification; the threads are not interchangeable. Two truncated conical spurs under the flange, 2.4 mm (0.0945 in) at the base, 0.4 mm (0.0157 in) at the tip and 1.2 mm (0.0472 in) high, stop it from turning; they bite into wood and need two 2.4 mm (0.0945 in) holes in hard materials. The tips are flattened because a perfect point is neither machinable nor meshable. A person pinches the head between two fingers and pulls.

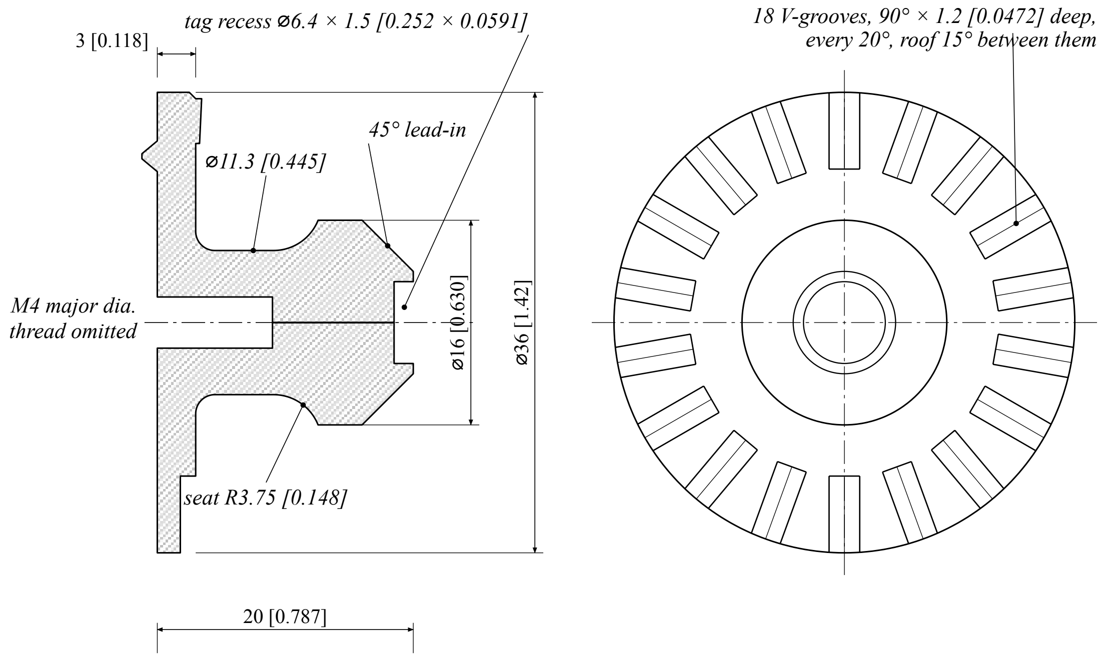

Figure 2. UGP-M stud, section and front view. The upper half of the section passes between two grooves and through a spur, the lower half along a groove. Dimensions in millimetres, inches in brackets.

5. The Socket

The socket (Figure 3) is 36 mm (1.42 in) in diameter at the nose and 42 mm (1.65 in) over the release collar, and measures 40.3 mm (1.59 in) from the nominal nose face to its mounting face. The ribs project 1.6 mm (0.0630 in) ahead of that face and the centring spigot projects 3 mm (0.118 in) behind the mounting face; the full axial envelope is 44.9 mm (1.77 in). Its mounting pattern targets ISO 9409-1–50–4–M6 [6] with four M6 screws on a 50 mm (1.97 in) pitch circle, a centring spigot and a hole for the locating pin; this flange is 63 mm (2.48 in) in diameter. Pilot fit, pin orientation and depth, screw engagement, and the cable passage must be checked against the selected robot drawing; this is not a universal robot mount. When the socket is seated, the mounting face stands 44 mm (1.73 in) from the surface of the object. The socket has five machined parts, six bearing balls, two helical springs, one stop screw, 3 M3 nose screws and two M4 dog-point screws, to which an antenna post can be added (Section 8). Its assembly and disassembly order is fixed (Section 10).

The *body* combines the mounting flange with a thin-walled *cage*, whose bore admits the head of the stud and whose six radial holes hold the balls. Around the cage slides the *locking sleeve*. From front to back its bore has a pocket into which the balls can retreat, a 45° transition, a cylindrical land 1.5 mm (0.0591 in) long, and a ramp inclined at 4° to the axis. A spring pushes the sleeve toward the object. The sleeve has 0.2 mm (0.00787 in) of nominal radial float, limited by its outer clearance to the housing, so that it can centre on the balls. The *nose* is a tube whose front face carries three relieved ribs with toroidal bearing caps. It is centred on a short boss of the body and ends in three rear tabs, 49.0 mm (1.93 in) across, each pierced for an M3 screw driven axially from the front into the flange face. Those screws sit at 90°, 180° and 270°, interleaved at 45° with the four M6 mounting screws and 90° from the ISO locating pin; their heads stay clear of the nose tube, of the M6 heads and of the collar's whole travel. Radial screws were tried first and abandoned: on this diameter their heads necessarily overlap the mounting flange. A *release collar*, a plain ring, slides onto the nose from the front and is then joined to the sleeve by two dog-point screws, driven in radially from the outside through the collar, the nose slots and into holes through the sleeve wall. They and the stop screw are the only special parts of the design, and Section 9.2 defines all three: no catalogue set screw reaches far enough, since the span from the collar thread to the sleeve is 6.3 mm (0.248 in) against the 2.25 mm (0.0886 in) of a DIN 915 M4. The dog points engage 2.8 mm (0.110 in) of sleeve and their threads take 2.7 mm (0.106 in) of collar wall; 0.25 mm (0.00984 in) of radial clearance in the sleeve holes preserves the float of the sleeve. The collar is the only part an actuator, or a person, has to move. Inside the bore slides a spring-loaded *plunger*. It is a tube, 12 mm (0.472 in) in bore, whose front rim is hollowed to the shape of the lead-in cone of the stud, and it is retained by a *stop screw* whose plain dog runs in a slot cut along it. Its rear outer edge carries the same 0.8 mm (0.0315 in) lead-in, because that is the edge which enters first when the socket is assembled: square, it would meet a ball at 2.3° and wedge instead of lifting; chamfered it meets one at 21°. Without the optional antenna post, a coaxial circular rod thinner than 12 mm (0.472 in) can enter the tube without directly moving it; with the post installed, the rod may hit the post. A coaxial circular body spanning the tube bore holds the balls at a radius of 8.5 mm (0.335 in) or more, where their centres are still inside the wall of the cage, whose bore radius is 8.2 mm (0.323 in); the balls then cannot fall into the bore. A thin tool pressed against the rim off the axis defeats this protection (Section 12).

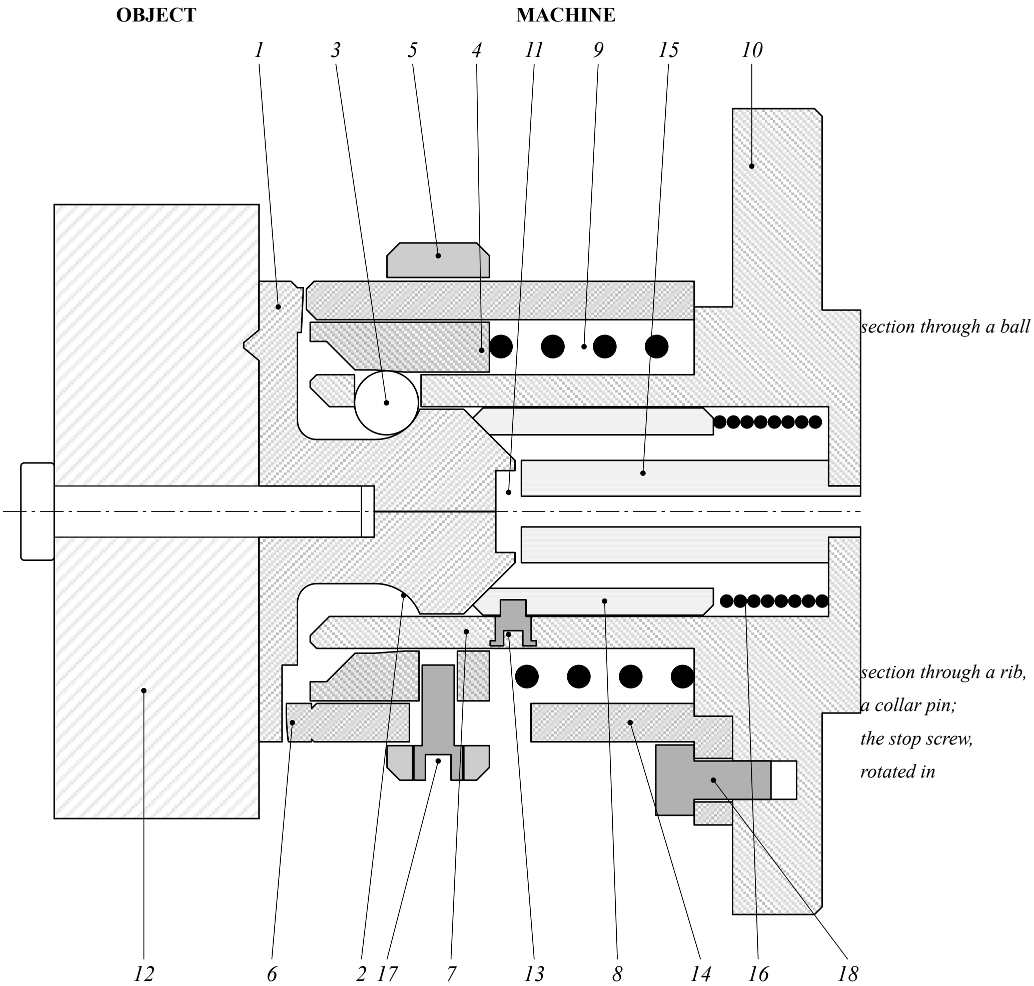

Figure 3. UGP-M mated and locked, CAD section with schematic springs and fixing screw. 1 stud; 2 toroidal seat; 3 ball (six, 5 mm (0.197 in)); 4 locking sleeve; 5 release collar; 6 rib of the nose seated in a V-groove; 7 body and cage; 8 arming plunger; 9 sleeve spring; 10 flange ISO 9409-1–50–4–M6; 11 tag recess; 12 panel of the object and fixing screw; 13 stop screw of the plunger, which lies at 30° and is rotated into the plane of the section; 14 nose; 15 antenna post; 16 plunger spring; 17 special dog-point screw, M4 thread, through collar, nose slot and sleeve; 18 M3 nose screw into the flange face.

6. Docking, Holding and Release

Armed. When the socket is empty the plunger stands under the six balls and holds them outward (Figure 4a). The balls block the 45° transition of the sleeve, so the sleeve is held back, 5.5 mm (0.217 in) from its locked position, against its spring. In this state the socket consumes nothing.

Capture. The machine pushes the socket over the stud. The lead-in cone of the head meets the chamfer of the cage and slides into the bore. The geometry accepts a lateral error of 5.7 mm (0.224 in), which I quote as ±5 mm (±0.197 in), on condition that the arm or the object can yield sideways by that amount; inside the bore the short cylinder of the head leaves 6.6° of angular freedom until the ribs seat. The head pushes the plunger back and takes its place under the balls (Figure 4b). If the head arrives off-centre it moves the plunger early, by at most 1.7 mm (0.0669 in), which leaves the balls supported with a margin of 2.8 mm (0.110 in). The push required is the force of the plunger spring plus the friction of the balls on the head, 24 to 30 N (5.40 to 6.74 lbf) for friction coefficients from 0.08 to 0.20, not counting the force needed to move the arm or the object sideways. An object that cannot react this force, such as a light object standing free on a table, cannot be docked from the side.

Lock. When the head has passed the balls they move inward onto the toroidal seat, the sleeve is freed, and its spring drives the ramp over the balls (Figure 4c). The wedge closes the lock, draws the stud into the socket, and presses the three ribs of the nose into three of the 18 V-grooves of the flange. The seating force depends on friction: between 50 N (11.2 lbf), with a coefficient of 0.20 on the ramp, the cage and the seat, and 530 N (119 lbf) without friction. Above a coefficient of 0.42 the spring alone no longer seats the stud against the plunger, and the arm has to push it home; the wedge then holds it there. The contact model idealises the three ribs and grooves as six unilateral point contacts, inspired by Maxwell couplings [8, 9]. The CAD now uses toroidal rib caps against the planar V flanks. The relieved rib bodies do not carry; both the ball paths and the six toroidal contacts have explicit Hertz limits. Actual repeatability and tolerance-dependent load sharing remain to be measured. The coupling seats in any of 18 orientations 20° apart. After seating, the spring closes the lock. During indexing the release actuator holds the collar retracted until seating is confirmed.

The quoted ±2.3° roll window is a geometric entry window, not a range of fully seated orientations. With a fixed roll error of 1°, the CAD still requires 0.275 mm (0.0108 in) of axial separation; at 2° it requires 0.549 mm (0.0216 in), and at 2.8° 0.713 mm (0.0281 in). Torsional compliance is needed to reach full seating, including inside this window.

Indexing in roll. The 18 grooves retain a 20° pitch. The rib tips are now toroidal caps, with a transverse radius of 0.8 mm (0.0315 in), a longitudinal radius of 18 mm (0.709 in) and the apex at radius 15.6 mm (0.614 in). The apex height is 2.1314 mm (0.0839 in); the cap joins its supporting body at 2.6 mm (0.102 in), and the body flanks are relieved by 0.3 mm (0.0118 in). Roof crests and the convex transitions from the V flanks to the 15° roofs have radius 0.8 mm (0.0315 in). These surfaces are present in the CAD and exports, not merely requested in comments.

A smooth periodic profile still has a maximum, and friction can hold the ribs near it. No passive departure from every orientation is claimed. The specified wrist therefore supplies a fixed positive roll torque, rather than searching back and forth: 0.32 ± 0.01 N·m (0.236 ± 0.00738 lbf·ft), with parasitic roll torque limited to 0.02 N·m (0.0148 lbf·ft). The net axial force on the ribs is controlled to 30 ± 1 N (6.74 ± 0.225 lbf), excluding the plunger spring force. It requires a calibrated spring-force estimate as well as a force sensor; the total actuator push is greater than this net value. At $\mu \leq 0.20$ the worst uphill roof torque is 0.276 N·m (0.204 lbf·ft), below the available 0.29 N·m (0.214 lbf·ft) after errors and parasitic torque.

The collar is held fully retracted during this operation. An axial separation measurement, accurate to ±0.02 mm (±0.000787 in) relative to the seated datum, switches off the torque bias at 1.10 mm (0.0433 in) separation. Below that threshold, axial force seats the V contacts with a back-drivable wrist; the conservative downhill torque on a planar roof is 0.028 N·m (0.0207 lbf·ft), exceeding the 0.02 N·m (0.0148 lbf·ft) parasitic bound. The downstream servo must limit speed to 1°/s with dissipative rate control. The quoted torque is the bias available at rest; rate-dependent braking may reduce it while moving and must vanish at rest, apart from the 0.02 N·m (0.0148 lbf·ft) parasitic allowance. A constant torque sent to an uncontrolled joint is not this protocol. Only after axial seating, with measured separation no greater than 0.01 mm (0.000394 in) and roll speed no greater than 0.05°/s, may the collar close. A separate sleeve-position signal must confirm the lock before carrying load. Abort, unload and keep the collar retracted if a force/speed limit is exceeded or seating is not reached within one 20° pitch or 30 s. There is no 100 N (22.5 lbf) or 300 N (67.4 lbf) indexing mode. The supplied supervisor implements these state transitions. After locking, contradictory collar/lock signals or separation outside the configured service interval revoke load authorization and latch a fault without commanding opening. The integrator must supply a validated separation interval that includes sensor uncertainty, permitted play and elastic displacement under service loads; no universal default is supplied. Actual torque monitoring, safe unloading and collar holding during withdrawal or emergency stop belong to the robot integration. Servo integration and physical tests remain required.

The pressure screen includes the curvatures of both convex surfaces, radii at their tolerance limits and principal directions differing by up to 15°. It assigns the entire conservative normal-force sum, 54.8 N (12.3 lbf), to one rib, rather than assuming three equal loads. Its peak Hertz estimate is 4131 MPa (599 ksi) against the selected 4200 MPa (609 ksi) indentation criterion. The maximum computed semi-major axis is 0.280 mm (0.0110 in). This is a quasi-static contact screen, not a test result or a finite-element solution through changes of curvature. The separate BRep path check locates the actual contacts and verifies that they stay off the physical edges. Once seated, the caps bear on both sides of the symmetric V, while the roofs clear the nose face by at least 0.21 mm (0.00827 in).

Release. An actuator draws the collar and the sleeve back 5.8 mm (0.228 in) against the spring. The pull needed when the port carries no load is at most 75 N (16.9 lbf) if the wedge was seated with friction in the range above, and 110 N (24.7 lbf) in the design cases of Section 7.4; the work done on the spring is 0.26 J (0.192 ft·lbf) (Figure 4d). As the machine withdraws, the head pushes the balls out into the pocket, the plunger follows the head and takes its place under the balls, and the socket is armed again. The collar must be held back for the whole withdrawal. Released early, the sleeve presses the balls onto the head and then onto the plunger cone with 49 N (11.0 lbf), far more than the 12 N (2.70 lbf) its spring can give, so the plunger cannot advance; if the stud then leaves entirely, a ball can drop into the bore. Holding the collar is therefore an operating rule, not a convenience (Section 10). Once the plunger is home the ball rests on its cylinder, whose normal is radial, so the sleeve's axial force goes into the wall of the cage hole and not into the plunger: the stop screw then carries only the plunger spring plus what friction the radial ball load can develop, 22 N (4.95 lbf). The worse case is the handover itself, while the ball is still on the 45° cone and the sleeve's 49 N (11.0 lbf) do become an axial push; that is the load behind the 82 MPa (11.9 ksi) of bending quoted for its dog, and the 21 MPa (3.05 ksi) with which that dog bears on the end of the acetal slot. Loss of actuator power does not command release. Retention still depends on the mechanical conditions of Section 7.1, and the quoted manual release force applies only after unloading the port (Section 10).

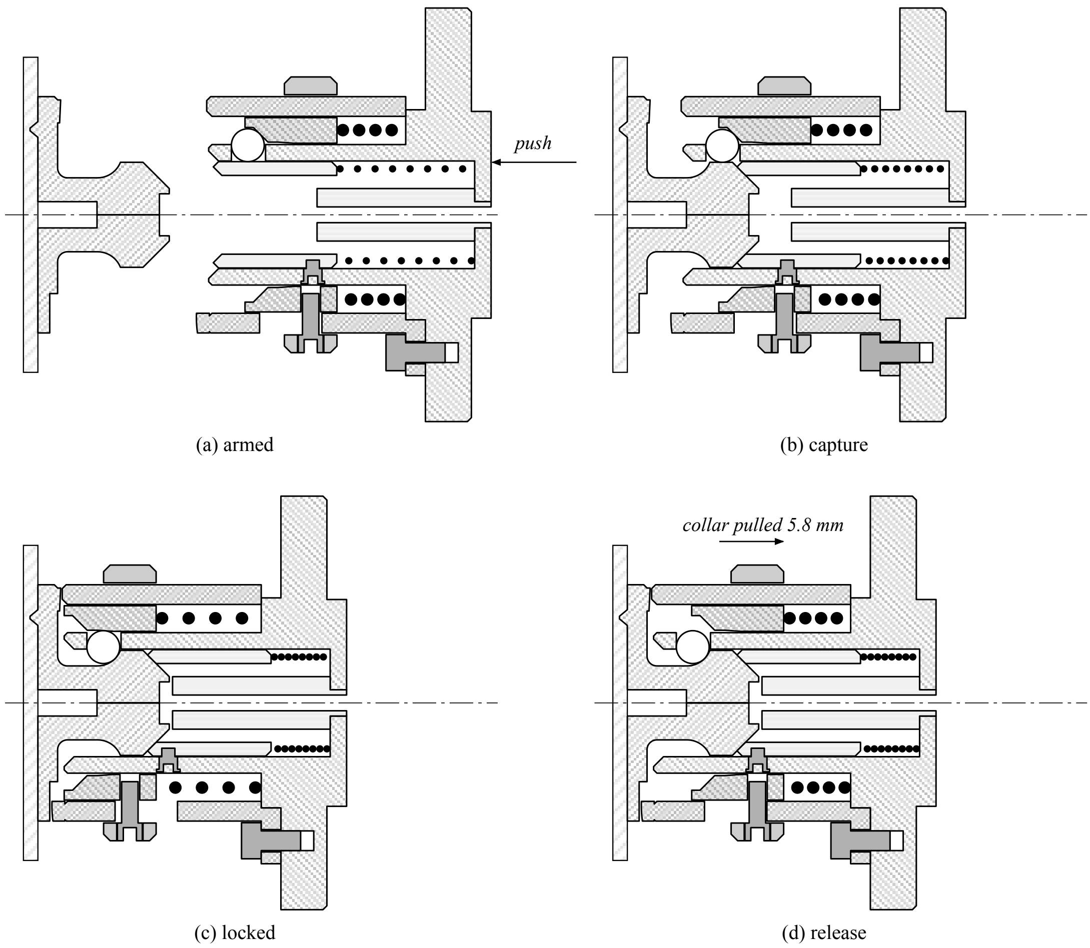

Figure 4. Docking sequence, drawn from the kinematic model. (a) Armed: the plunger holds the balls out and the balls hold the sleeve back. (b) Capture: the head has replaced the plunger under the balls. (c) Locked: the sleeve has wedged the balls onto the seat. (d) Release: the collar is drawn back and the balls are free.

7. Mechanics

7.1 Equilibrium of a ball and retention

Consider one ball while the stud is pulled away from the socket with a force F_a per ball (Figure 5). The seat pushes the ball outward and toward the object along a normal at 45°; the ramp, inclined at α to the axis, pushes it inward; the wall of the cage hole carries the axial remainder. Neglecting friction,

$$N_g = \sqrt{2}\, F_a, \quad N_r = F_a / \cos\alpha, \quad N_c = F_a\,(1 + \tan\alpha).$$

The load tries to drive the sleeve back with a force $N_r \sin\alpha$, and friction on the ramp resists with $\mu N_r \cos\alpha$. Making the ball spin instead of slide does not help the load, because the ball would then have to slide on the seat and on the cage, where the normal forces are larger. The wedge therefore holds by friction alone if

$$\tan\alpha < \mu.$$

With α = 4°, tan α = 0.070. The sensitivity calculations assume friction coefficients from 0.08 to 0.20, without measurements for this interface, so the condition is met within that assumed range but with too little margin to rest a safety claim on, and the coefficient must anyway be measured for the chosen finish, lubricant, wear and contamination. Retention is therefore not left to friction at all.

Instead the sleeve is stopped by an energy barrier. Its spring is sized, and the cylindrical land lengthened to 1.5 mm (0.0591 in), so that the work the spring absorbs exceeds the work the load can release. Suppose the worst case: zero friction everywhere, a pull held at F_0, and the plunger spring helping the load. As the sleeve retreats along the ramp the ball moves out, the stud withdraws, and the load does work; the peak surplus is 0 N·mm. The sphere then reaches the land after 1.0 mm (0.0394 in) of travel, having given the stud 0.07 mm (0.00276 in) of withdrawal and keeping 1.9 mm (0.0748 in) of engagement behind the head. On the cylinder the contact normal has no axial component, so the load does no further work while the spring keeps absorbing. By the end of the land the net energy is 64 N·mm *negative*. The sleeve cannot reach the release zone from rest, and no path to the release zone exists for a sleeve that starts at rest. That is the exact claim: the barrier is a statement about $K(s) = K(0) + W(s)$, and it holds only while $K(0)$ is zero. The 0.064 J (0.0472 ft·lbf) deficit at the end of the land is not an unlocking threshold: the work deficit can increase further along the following transition. For scale only, this energy corresponds to 1.61 m/s (5.3 ft/s) for the current CAD sleeve and collar mass of 49.37 g (1.74 oz), excluding the dog screws, spring and coupled inertias. This equivalent speed is not a validated impact limit. Shock, vibration and the inertia of the machine are therefore not covered, and are the first thing a test must measure. The same calculation stays negative at 1.5 F_0, which says nothing about the contacts there: they are rated at F_0. An independent rigid-body check of the same geometry gives −0.064 J (−0.0472 ft·lbf) at the end of the land, negative, confirming it. This is an energy argument on rigid bodies with the springs as calculated; it is not a dynamic simulation, and it does not cover a spring that has relaxed, broken or been fitted wrongly.

The shallow ramp limits the radial take-up of the wedge to −0.06 / +0.07 mm (0.00276 in) about the nominal position of the balls. A tolerance of ±0.05 mm (±0.00197 in) on the height of the seat above the grooves moves the balls by −0.05 / +0.05 mm (0.00197 in), which the wedge takes up. What remains, 0.01 mm (0.000394 in) inward and 0.02 mm (0.000787 in) outward, is all that is left for radial errors of the seat and of the ramp. Both exceed the 10 µm (0.39 thou) form band required in Section 7.2, which is what makes that band a requirement rather than an estimate, but not by much: these are the tightest tolerances of the design.

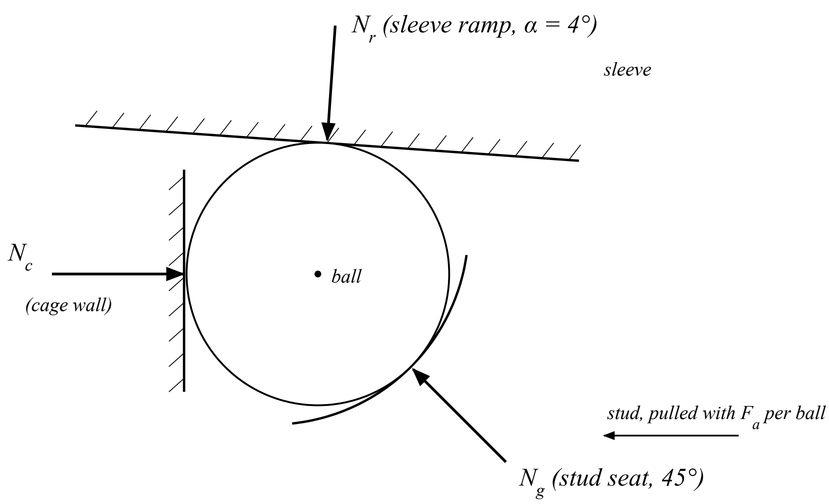

Figure 5. Forces on one locking ball when the stud is pulled.

7.2 Contact ratings

Each of the three contacts is an elliptical Hertz contact, which I solve exactly [10] for steel (E = 210 GPa (30.5 Mpsi), ν = 0.30). As a static criterion for hardened bearing steel I take the one used for ball bearings, a peak pressure of 4200 MPa (609 ksi), associated with a permanent set of about one ten-thousandth of the ball diameter in the bearing applications covered by that standard [11]. Its use here, including on the toroidal caps, is a design assumption requiring material and contact validation; it is not ISO qualification of the assembly. This permits a small permanent indentation (about 0.5 µm (0.020 thou) for a 5 mm (0.197 in) ball); it is neither a zero-marking criterion nor an assembly strength rating. It applies to a stud whose seat is hardened like a bearing race.

Table 1 gives the result for a seat at its nominal height. Against a plain 45° cone the ball would reach the limit at 100 N (22.5 lbf) per ball; the toroidal seat raises this by a factor of 2.6. The ramp is the governing contact in this idealisation. In particular, the cage row extrapolates an infinite-surface Hertz model to a large patch; it is not a validated capacity of the finite, thin cage wall.

Table 1. Ideal Hertz contact limits at 4200 MPa (609 ksi), hardened steel throughout; not complete component capacities.

Contact	Normal force at limit	Contact ellipse	Axial load per ball
Ball on toroidal seat (stud)	371 N (83.4 lbf)	0.64 × 0.26 mm (0.0252 × 0.0102 in)	262 N (58.9 lbf)
Ball on sleeve ramp	231 N (51.9 lbf)	0.35 × 0.30 mm (0.0138 × 0.0118 in)	**231 N (51.9 lbf)**
Ball on cage wall	2299 N (517 lbf)	2.92 × 0.36 mm (0.115 × 0.0142 in)	2149 N (483 lbf)

The radius of the seat is a compromise. A seat that fits the ball as closely as a bearing race, with a ratio of 0.53, would carry more at its nominal height, but the direction of the contact normal, and with it the load thrown onto the ramp, would then depend on the height of the seat to within a few hundredths of a millimetre: an error of +0.05 mm (0.00197 in) would turn the normal from 45° to 22°, and an error of −0.05 mm (−0.00197 in) would prevent the ball from seating at all. With a ratio of 0.75 the same errors turn the normal between 41.8° and 48.3°, and the limit per ball between 207 and 259 N (46.5 and 58.2 lbf), the ramp governing throughout. With equal load sharing, the ideal contact model gives approximately 6 × 207 N (46.5 lbf) ≈ 1.24 kN (279 lbf). The ±0.05 mm (±0.00197 in) is the combined error of the stud and of the socket on the height of the seat relative to the ball.

Six balls share a load only as far as the parts are accurate. A ball path, with its three contacts in series, closes by only 21 µm (0.83 thou) when it reaches its limit, so errors of form of a few micrometres decide how the load is shared. The stud is free to tilt and the sleeve is free to float, which removes any error that varies once around the axis; what remains is taken through the Hertz law, in which force grows as the 3/2 power of the approach. Of five patterns of error tried, the worst is the one that alternates from ball to ball: it leaves 0.87, 0.76 and 0.58 of the capacity when the seat, the six ball stops and the ramp together lie within a band of 5, 10 and 20 µm (0.20, 0.39 and 0.79 thou). I assume an axial-equivalent error band of 10 µm (0.39 thou) and apply the empirical model factor 0.76 to every ball limit from here on. Five error patterns do not bound every possible manufacturing error, and a uniform reduction of all contact limits does not explicitly simulate imperfect parts under combined loads. These estimates require further tolerance analysis. The balls also carry the plunger spring, which pushes the stud out with 20 N (4.50 lbf), so the six ball paths can carry 0.76 × 1.24 kN (279 lbf) − 20 N (4.50 lbf) = 0.92 kN (207 lbf) before any external load is applied. That is not yet the rating. The wedge seats the joint with a preload that the balls also carry, and in the least favourable case of Section 7.3 it costs a further 412 N (92.6 lbf), so the axial rating is F_0 = 0.51 kN (115 lbf). Reported values are rounded; the calculation carries full precision.

7.3 Moment, torque, shear and combined loads

Loads other than a straight pull are evaluated on a model of the locked port as a rigid stud held by twelve frictionless elastic contacts, the six balls on the seat and the six flanks of the ribs in the grooves, each able to push but not to pull. The stud has six degrees of freedom and the loads are applied at the centre of the rib plane. The sleeve floats: its lateral position is a further unknown with no stiffness of its own, so that the radial forces of the balls balance on it. The calculation compares a floating sleeve with a sleeve held centrally and keeps the lower result; these two cases do not prove a bound for all frictional states. In the sampled admissible cases the floating sleeve moves by at most 0.18 mm (0.00709 in), of the 0.2 mm (0.00787 in) available. Equilibrium is the minimum of a convex energy; the solver was checked against an exhaustive enumeration of the 4096 possible sets of closed contacts, with which it agrees. The model returns the axial rating F_0. For axial torque, both groups are checked: the ball-path capacity and the individual toroidal rib contacts. The latter now govern the torque limit; the contact pitch radius is 15.6 mm (0.614 in). The previous ball-only torque rating no longer applies.

The model needs the ratio of a rib-contact stiffness to that of a ball path. The Hertz approaches give 24 kN/mm for the three ball contacts in series at their rating. A toroidal rib contact has a tangent stiffness of 38 kN/mm at 100 N (22.5 lbf) normal load; it varies with load. The scan uses stiffness ratios 0.25, 1 and 3, including values below unity, and reaches beyond the computed ratio 2 at the rib contact limit. Nominal seat height and both tolerance limits, both seating-force bounds, and directions every 15° are evaluated. These linearised samples do not prove a bound for every nonlinear, imperfect contact state. Each toroidal rib contact is limited to 261 N (58.7 lbf) by the same indentation criterion; at that limit its semi-major contact axis is 0.522 mm (0.0206 in). Both ball and rib limits are enforced in the twelve-contact model.

For a bending moment, M_0 = 4.5 N·m (3.32 lbf·ft). In the worst direction the results run from 4.5 to 5.0 N·m (3.32 to 3.69 lbf·ft) over the stiffnesses and seating forces, showing the sampled stiffness sensitivity; in the most favourable direction the port carries up to 5.8 N·m (4.28 lbf·ft). For a torque, T_0 = 8.6 N·m (6.34 lbf·ft) in every case. For a transverse force through the centre of the rib plane, V_0 = 0.27 kN (60.7 lbf), with 0.27 to 0.28 kN (60.7 to 62.9 lbf) in the worst direction and up to 0.32 kN (71.9 lbf) in the best. The simple estimate that a transverse force V separates the ribs with a force 4V/3 gives 0.38 kN (85.4 lbf) and is not safe, because the unequal forces on the three ribs also tilt the stud.

In 500 random mixtures of the four loads, in random directions, the most loaded ball or rib contact reached at most 0.942 of its limit when the four ratios sum to one, and exceeded it when they sum to 1.1. This supports the following preliminary screening rule, without proving it for all mixtures, tolerance patterns or intermediate stiffness ratios. Here F is outward axial pull, V and M are transverse-force and bending-moment magnitudes, and T is the absolute axial torque:

$$F/F_0 + V/V_0 + M/M_0 + T/T_0 \le 1.$$

A weight applies a force and a moment together, so practical cases are computed directly. With the port axis horizontal and the centre of gravity on the object side of the axis at z = −150 mm (−5.91 in) relative to its mounting surface (z = 0), the port carries 3.0 kg (6.61 lb). Lifting from above, with the port axis vertical, it carries 51 kg (112 lb) if the centre of gravity is on the axis and 3.1 kg (6.83 lb) if it is 150 mm (5.91 in) to one side. For illustration only, equal sharing of a push F_0 gives a peak rib Hertz pressure of 3245 MPa (471 ksi); the rated checks use each individual contact force, not this sharing assumption.

7.4 *Tightening of the wedge*

A self-locked wedge can retain an unwanted preload. When the machine pushes the stud toward the socket with a force Q, the ribs are compressed, the balls are relieved, and the sleeve follows them in with the seating force S. When the push stops, the ribs relax against balls that can no longer retreat, and a preload stays locked in. At the nominal 45° seat angle, approximating both contact groups by linear axial springs, with ρ the stiffness ratio of Section 7.3 and $Q \geq S$,

$$P = S + max(0, Q - S) / (1 + \rho).$$

The declared push rating Q_0 = 0.40 kN (89.9 lbf) is below the worst spring seating force 530 N (119 lbf). In this linear model its post-push preload is therefore no greater than the preload already included in the ratings. The contact model checks the individual limits after the push at stiffness ratios 0.25, 1 and 3 and in the sampled directions. This preload model does not include nonlinear Hertz stiffness, sleeve bottoming, off-axis pushes or tolerance-dependent spring force. The locked-in preload also raises the pull needed on the collar, which is the spring force plus (P + 20 N (4.50 lbf))($\mu - \tan \alpha$), since the balls carry the plunger spring as well: 110 N (24.7 lbf) after a push of Q_0, or after a seating without friction, with $\mu = 0.20$ at breakaway, and 110 N (24.7 lbf) after an over-push equal to F_0. The actuator and the hand collar are to be sized for 110 N (24.7 lbf); undocking and docking again resets the wedge.

7.5 *What yields first*

In axial pull the ball-path contact screen is governed by the ramp; in torque the rounded rib contacts now govern. Reaching the indentation criterion is not a complete strength proof; deformation beyond it requires an elastic-plastic model. The following axial strength screens compare several idealised sections with the fixing screw tensile load; they do not establish the first failure of the assembly. An M4 screw has a stress area of 8.78 mm² and a minimum breaking load of 3.7 kN (832 lbf) in property class 4.8 and 7.0 kN (1570 lbf) in class 8.8 [13], that is 7.2 and 13.8 times F_0. At the class 4.8 load, using a provisional yield stress of 350 MPa (50.8 ksi) for the steel sections, the simple estimates are: 2.9 kN (652 lbf) per ball for the cage wall in front of each hole, 43 kN (9670 lbf) for the head in shear, 31 kN (6970 lbf) for the neck in tension over its net section of 87 mm², and a stress of 54 MPa (7.83 ksi) in the sleeve treated as a ring under six point loads. Missing checks include female-thread stripping and usable engagement, flange bending, the three nose screws under moment and torque, rib edge stresses, collar pins, and fixation to the object. The M4 tensile load alone cannot validate shear, bending or torque at the mounting surface. For example, transmitting 8.6 N·m (6.34 lbf·ft) through two spurs at radius 13 mm (0.512 in) requires roughly 332 N (74.6 lbf) tangential force at each spur; their tapered contacts and the panel have not been rated. A certified assembly working load, including fatigue, shock and safety factors, cannot yet be assigned.

Table 2. UGP-M preliminary model results. Contact limits are not certified assembly working loads; none has been measured.

Quantity	Value	Basis
Stud above object	⌀36 × 20 mm (1.42 × 0.787 in)	head ⌀16 mm (0.630 in), neck ⌀11.3 mm (0.445 in); spurs add 1.2 mm (0.0472 in) below surface
Stud mass	39 g (1.38 oz) / 33 g (1.16 oz)	steel 7.85 / zinc alloy 6.7 g/cm³; volume 5.0 cm³ (0.305 in³)
Socket full envelope	⌀63 × 44.9 mm (2.48 × 1.77 in)	collar ⌀42 mm (1.65 in); nose face to mounting face 40.3 mm (1.59 in)
Socket mass	360 g (12.7 oz)	steel; acetal plunger and post; without mounting screws
Lateral capture	±5 mm (±0.197 in)	geometric limit 5.7 mm (0.224 in); needs lateral compliance
Angular freedom in the bore	6.6°	0.4 mm (0.0157 in) clearance over a 3.4 mm (0.134 in) head cylinder
Roll capture	±2.3°	about any of 18 positions 20° apart
Aligned insertion push	24–30 N (5.40–6.74 lbf)	$\mu = 0.08$–0.20
Indexing rib load	30 ± 1 N (6.74 ± 0.225 lbf)	net of plunger force; torque and sensor requirements in Section 6
Seating force	50–530 N (11.2–119 lbf)	$\mu = 0.20$–0; sleeve spring 38 N (8.54 lbf) when locked
Release	5.8 mm (0.228 in); ≤ 110 N (24.7 lbf)	unloaded port, after at most a push of Q_0; spring work 0.26 J (0.192 ft·lbf), excluding friction; passive holding
Axial rating F_0	0.51 kN (115 lbf)	ISO 76 criterion; seat height ±0.05 mm (±0.00197 in); form within 10 µm (0.39 thou); net of the plunger spring
Push rating Q_0	0.40 kN (89.9 lbf)	preliminary linear preload model; checked at $\rho = 0.25, 1, 3$
Shear rating V_0	0.27 kN (60.7 lbf)	through the rib plane; lowest sampled case
Moment rating M_0	4.5 N·m (3.32 lbf·ft)	lowest sampled case
Torque rating T_0	8.6 N·m (6.34 lbf·ft)	about the port axis
Combined loads	sum of the four ratios ≤ 1	500 random mixtures
Mass carried, axis horizontal	3.0 kg (6.61 lb)	centre of gravity on the object axis at $z = -150$ mm (−5.91 in)
Mass lifted from above	51 kg (112 lb) / 3.1 kg (6.83 lb)	centre of gravity on the axis / 150 mm (5.91 in) off it

The spring calculations assume two helical compression springs with closed and ground ends. The sleeve spring has 1.8 mm (0.0709 in) wire, a mean diameter of 25.8 mm (1.02 in) and 5 coils, and gives 38 N (8.54 lbf) locked and 50 N (11.2 lbf) released; the plunger spring has 1.0 mm (0.0394 in) wire, a mean diameter of 14.0 mm (0.551 in) and 7.6 coils, and gives 12 N (2.70 lbf) extended and 20 N (4.50 lbf) compressed. The actual nominal slot stops permit 6.0 mm (0.236 in) of sleeve retraction and 0.5 mm (0.0197 in) of extra plunger compression beyond docking. At these stops, the calculated spring margins are 1.0 and 0.9 mm (0.0394 and 0.0354 in) above solid height, respectively, and peak stresses remain below 800 MPa (116 ksi); manufacturing tolerances must still be deducted. The 5.8 mm (0.228 in) release stroke is a command value, not the sleeve hard stop. The calculation assumes a static allowable stress of half the tensile strength and requires wire of at least 1800 MPa (261 ksi). That fraction is material- and process-dependent, not a universal allowance. Spring selection must confirm it, including fatigue, set, tolerances and the unusually small number of active sleeve coils. The CAD helices have uniform pitch and unground ends; they illustrate the spring envelopes, not manufacture-ready closed and ground springs.

7.6 What reaches the object

T_0 is a preliminary contact-model limit, not an installation rating. An assumed M4 class 8.8 screw preload of 60 % of proof load gives 3.1 kN (697 lbf), 3.0 MPa (0.435 ksi) mean substrate pressure, and 7.4 N·m (5.46 lbf·ft) of frictional torque at $\mu = 0.20$. A separate projected-area screen for the two spurs gives 0.44 N·m (0.325 lbf·ft) in softwood, 1.3 N·m (0.959 lbf·ft) in hardwood, 8.7 N·m (6.42 lbf·ft) in aluminium and 17.5 N·m (12.9 lbf·ft) in steel, using assumed bearing strengths of 10, 30, 200 and 400 MPa (1.45, 4.35, 29.0 and 58.0 ksi). At T_0, each spur would transmit 332 N (74.6 lbf), or 197 MPa (28.6 ksi) on that projected area. This is just below the assumed aluminium value and below the steel value; neither comparison qualifies the actual conical contacts or their holes. Wood creep may reduce screw preload. Even on metal, adding friction and spur capacities assumes compatible load sharing and maintained preload, neither established here. The arithmetic sums, 16.2 N·m (11.9 lbf·ft) for aluminium and 24.9 N·m (18.4 lbf·ft) for steel, are illustrations, not durable capacities. Species, moisture, grain, actual alloy, fits and cyclic loading require assessment. The fixation must be designed and tested for the object, potentially using an insert, a metal boss or a wider bolt pattern.

8. The Information Layer

A mechanical port tells the machine how to hold an object, not what it is holding. The recess in the head is intended for a passive proximity tag [15, 16], to be read by an antenna at the tip of a fixed acetal post that stands inside the plunger tube, 0.5 mm (0.0197 in) from the head when the port is docked. Tags of this size are read at a few millimetres at most [16]; reading one from inside a steel cage has not been demonstrated. The tag is meant to answer the questions that vision answers poorly: the mass of the object and the position of its centre of gravity, the load rating of the stud, and the kinematics of what lies behind it, for instance “hinge on the left, radius 397 mm”, “drawer, travel 450 mm” or “rotary control, 270°”. The robot cameras are responsible for identifying the object orientation; the coupling constrains the pose once seated. With that visual estimate and the constraint described by the tag, the robot can plan the opening trajectory. Camera integration and its accuracy are outside this mechanical design.

The flange is also a possible optical target: a disc of known diameter with 18 regular grooves.

9. Materials and Sizes

The estimates of Section 7 assume a stud whose seat has bearing-race hardness; the steel grade, hardness, case depth and surface finish still need manufacturing specifications. A stud in a zinc alloy or in a polymer will carry less, because its seat yields and, under sustained load, creeps; its rating cannot be calculated with confidence and must be established by test. I give none here. The rating of each stud is meant to be declared on its tag.

The socket consists of turned and milled parts, standard balls, pins and screws, and two springs. The sleeve and the cage must be hardened, since they carry the ball contacts, and the 10 µm (0.39 thou) band of Section 7.2 applies to them and to the seat of the stud together. I have not costed either half, and claim only that the stud has the geometry and the fastener of a cabinet knob.

Only the M size is defined here. A smaller size for light objects and a larger one for heavy doors and containers are reserved.

9.1 Fits

A fit designation only means something when both parts share a nominal size. Several interfaces here do not: the model deliberately leaves the collar 0.6 mm (0.0236 in) larger than the nose, and the nose bore 0.4 mm (0.0157 in) larger than the sleeve, because those gaps are the float the mechanism needs. Writing “H11/c11” across such a pair would be meaningless. Table 3 therefore gives two limit dimensions for every interface and the clearance they produce, whether or not the nominal sizes agree. These are design intents to be confirmed on the drawings, not measured results, and the clearances that the kinematics depend on are re-checked against these limits in the calculation file.

Table 3. Limit dimensions and the resulting diametral clearance, in millimetres only: a drawing is dimensioned in one system, and the port is metric. Divide by 25.4 for inches. Tapped holes are drawn at their major diameter; threads are not modelled.

Interface	Hole min / max	Shaft min / max	Clearance min / max	Designation
Nose flange bore on the body boss	32.000 / 32.025	31.975 / 31.991	0.009 / 0.050	H7/g6
Antenna post stem in the cable passage	4.000 / 4.012	3.998 / 4.006	0.006 int. / 0.014 cl.	H7/j6
Collar bore over the nose	36.600 / 36.760	35.840 / 36.000	0.600 / 0.920	36.6 H11 / 36 h11
Sleeve in the nose bore	30.000 / 30.130	29.470 / 29.600	0.400 / 0.660	30 H11 / 29.6 h11
Sleeve bore over the cage	21.810 / 21.940	21.270 / 21.400	0.410 / 0.670	21.81 H11 / 21.4 h11
Dog point in the sleeve hole	3.000 / 3.060	2.475 / 2.500	0.500 / 0.585	3 H11 / 2.5 h9
Nose screw through its tab	3.400 / 3.475	2.940 / 3.000	0.400 / 0.535	3.4 H11 / ISO 4762 shank
Ball in its cage hole	5.200 / 5.275	4.999 / 5.000	0.200 / 0.276	5.2 H11 / G10 ball
Stud head in the cage bore	16.400 / 16.510	15.890 / 16.000	0.400 / 0.620	16.4 H11 / 16 h11
Plunger in the cage bore	16.400 / 16.510	16.090 / 16.200	0.200 / 0.420	16.4 H11 / 16.2 h11

9.2 Threads, springs and processes

The stud takes one M4 × 0.7 screw in a blind thread, or a #8-32 in the imperial variant; the two are separate part numbers and are not interchangeable. The tapped depth is 9 mm (0.354 in) for at least 6 mm (0.236 in) of usable thread. The nose screws are M3 × 0.5 × 6 ISO 4762, the coarse pitch of an M3 being 0.5 mm (0.0197 in) and not the 0.7 mm (0.0276 in) of the M4 used elsewhere. Each passes through a 3 mm (0.118 in) tab and engages 3 mm (0.118 in) of a thread tapped 5 mm (0.197 in) deep, so the screw seats on the tab with 2 mm (0.0787 in) still to go before it could bottom, and 2 mm (0.0787 in) of metal is left under the tapped hole; they are secured with a medium-strength anaerobic adhesive; the two dog-point screws and the stop screw of the plunger are the three special parts, and are removed for internal disassembly with the nose left on the body. Complete disassembly also removes the three M3 nose screws. Each of the two collar dog screws is an M4 × 0.7 thread 2.7 mm (0.106 in) long followed by a plain dog 2.5 mm (0.0984 in) in diameter and 6.3 mm (0.248 in) long, with a 2 mm (0.0787 in) hexagon socket 2 mm (0.0787 in) deep in its outer end, in steel of property class 45H, secured with the same adhesive. Because the hole through the sleeve is open, the depth is fixed by a setting dimension rather than by a seat: the outer end is set flush with the collar, within 0.2 mm (0.00787 in), and even set that much too deep the tip still stops 1.1 mm (0.0433 in) short of the cage. A drawing of this part belongs with the others named in Section 9.4. The stop screw of the plunger is the third special part, and the smallest: a collar 0.4 mm (0.0157 in) thick and 3.6 mm (0.142 in) across, then an M3 × 0.5 thread 1.9 mm (0.0748 in) long, then a plain dog 2 mm (0.0787 in) in diameter and 1.3 mm (0.0512 in) long, 3.6 mm (0.142 in) in all, with a 1.5 mm (0.0591 in) hexagon socket in its outer face, in the same property class 45H and the same adhesive. Its depth is set by the collar bottoming in a spotface machined in the skin of the cage, so that there is nothing to measure in the blind: seated, the dog tip stands 0.1 mm (0.00394 in) clear of the bottom of the plunger slot and engages 1.2 mm (0.0472 in) of it, the thread keeps 3.8 turns in 1.9 mm (0.0748 in) of cage wall, and the outer face comes to rest 0.2 mm (0.00787 in) below the outside of the cage, which a depth gauge through the 5 mm (0.197 in) access hole confirms. The collar is sunk rather than left flush because a flat face carried on a radius stands further out at its edge than at its centre: seated, that edge lies at a radius of 10.65 mm (0.419 in), clear of the 10.91 mm (0.430 in) bore of the sleeve that slides over it. Both access holes are sized on that collar, which is the largest part of the screw and the part that has to travel through them, and not on its thread: 4.2 mm (0.165 in) through the sleeve and 5 mm (0.197 in) through the nose, which leaves 0.3 mm (0.0118 in) of radial play on the collar, more than the 0.10 mm (0.00394 in) the fixture is allowed on the hold position. The hexagon key reaches 7.3 mm (0.287 in) in from the outside of the nose. Both springs are cold-coiled from patented or oil-tempered wire of at least 1800 MPa (261 ksi) tensile strength, with closed and ground ends; the supplier drawing must give the force at each working length, the maximum solid height, the tolerance class and the fatigue duty, because the calculated margins here assume ends that a uniform modelled helix does not have. The bore of the body is bored, not drilled, to a flat floor: a twist drill reaching full diameter at that depth would break through into the cable passage. The space that houses the sleeve and its spring is not a groove to be machined: the nose is a separate part, so the cage of the body is an external cylinder, 21.4 mm (0.843 in) across over 30 mm (1.18 in), turned with the tool in open air. What the process sheet does have to settle is the flat-bottomed bore, the shoulder undercuts and the finish of the ball seats after hardening.

The indexing surfaces require a separate finishing operation after hardening: a toroidal cap of minor radius 0.8 mm (0.0315 in) and major radius 17.2 mm (0.677 in), centred at radial coordinate 15.6 mm (0.614 in) and z = 20.1314 mm (0.793 in), with its axis tangent to the pitch circle; the lower apex and join heights are specified in Section 6. All three cap locations reference the nose axis and mounting datum. Roof crest and shoulder radii are 0.8 mm (0.0315 in). Specify radius errors no greater than ±0.01 mm (±0.000394 in) and a total surface-profile zone of 0.01 mm (0.000394 in) relative to these datums, including the cap location. Copy the exact STEP surfaces into the toolpath; the convex surfaces are accessible from the front for form grinding or ball-end finishing. This is a process proposal, not supplier approval. Verify both curvatures, cap location, tangent continuity and the remaining edge margins after treatment. The nose bearing caps and stud indexing roofs require the same hardened contact-surface specification as the load-bearing seats.

9.3 Inspection

The 10 µm (0.39 thou) form band of Section 7.2 is a requirement on the finished, heat-treated parts, not a modelling convenience: the seat of the stud, the six ball stops of the cage and the ramp of the sleeve must lie within it, measured from the port axis as datum and the mounting face as the axial datum. The ±0.05 mm (±0.00197 in) on the height of the seat above the grooves governs the axial wedge take-up; the surface-profile, radius and fit requirements are separately tighter.

9.4 What is released

The design leaves the project as files, and those files are checked rather than assumed. Each of the 15 line items of the bill of materials carries its material, its treatment and its calculated mass; the socket adds up to 362 g (12.8 oz) without its mounting screws. Every part is exported as a STEP solid and re-imported to confirm that it comes back as one valid solid of the same volume, and the assembly file is confirmed to carry every part, the six balls and both springs. Every part is also exported as an STL and checked to be a closed manifold, so that a prototype can be printed at twice scale, where the balls become 10 mm (0.394 in), a catalogue size, and the thinnest wall becomes 1.6 mm (0.0630 in). The parametric model and the calculation file go with them, so that the implemented model checks can be rerun after changing a dimension; the publication also requires editorial review.

What is not released is as important. There are no dimensioned drawings with geometrical tolerances, no supplier drawing for either spring, no drawing for any of the three special screws, no design for the assembly fixture that holds the sleeve, and no tolerance stack on capture, seating and stroke. Those three are the work between this paper and a first part.

10. Assembly and Operation

The order below is not a suggestion: each step exists because the next one would otherwise be impossible. The sleeve only passes balls that are sunk, since its smallest bore is 21.8 mm (0.858 in); the plunger and its spring only pass balls that are raised, and of the two it is the plunger that is tight, with 0.2 mm (0.00787 in) of radial clearance against 0.8 mm (0.0315 in) for the 15 mm (0.591 in) spring. The balls therefore have to go in between those two parts, and nothing before or after them can be swapped.

The stop screw comes after both, for a reason of the same kind. The slot it runs in is closed at both ends, and its rear end *is* the forward stop that keeps the plunger spring from ejecting the plunger, so that material cannot be opened and the plunger cannot be threaded onto a stop already fitted: the model measures 3.71 mm³ of solid overlap if one tries. The screw is therefore driven last, radially, through a hole in the nose and a hole in the sleeve that face it only while the sleeve is held at its assembly position. Both sit at 30°, which is 30° from the nearest ball, nose slot, rib or tab: a slot in the plunger passing under a ball would let that ball drop 0.26 mm (0.0102 in) into it.

Holding the sleeve back through all of this is a fixture, not two loose pins. The nose slots are longitudinal, so a pin free in one simply travels with the sleeve; the pins have to be clamped, and the clamp has to react 50 N (11.2 lbf) and set the sleeve to 5.86 mm (0.231 in). That figure has to fall inside a window, 5.72 to 5.96 mm (0.225 to 0.235 in), which presents the pocket to the *whole* ball and not merely to its equator: comparing the equator with the ends of the pocket would have allowed 4.69 mm (0.185 in), where the 45° transition cuts into the ball and the plunger no longer passes under it. The nominal setting keeps 0.10 mm (0.00394 in) to the nearer end of that window, which is a requirement on the fixture and not a manufacturing tolerance. Holding the sleeve also keeps its spring out of the insertion force: pushed against it instead, the plunger would meet about 993 N (223 lbf) where the ramp first takes the load, on its chamfered leading corner. That figure is a local frictionless estimate at one point of the path, not a peak over the whole insertion, and the clamped order removes the term altogether. Disassembly is the reverse, and starts by unscrewing the two dog screws.

1. Bare body: drill the six ball holes, and spotface and tap the stop-screw hole, while the cage is uncovered.
2. Antenna post from the FRONT, dia 4 stem first, into the cable passage (optional layer). It cannot enter from the mounting face: its dia 8 body would have to cross the dia 4 passage.
3. Nose onto the centring boss; three M3 screws driven axially from the front.
4. Sleeve spring, then the sleeve, from the front through the nose bore, onto the bare cage. No ball is fitted yet: the sleeve could not pass a ball held out, nor could the plunger spring pass a ball left sunk.
5. Push the sleeve back to the assembly position with two pins through the nose slots into its dog holes, and CLAMP those pins: the slots are longitudinal, so a pin left free in one travels with the

sleeve and is not a stop. The fixture reacts about 50 N (11.2 lbf) and sets the position, which also brings the sleeve's access hole in line with the stop-screw hole.

6. Six balls through the front bore into their holes, pushed outward into the sleeve pocket and held there with assembly grease.
7. Plunger spring, then plunger, from the front: both pass under the raised balls, and the chamfered rear edge of the plunger lifts any ball that has sagged. Nothing yet retains the plunger; its own spring pushes it back out.
8. Holding the plunger pushed fully in, and turned so that its slot faces the access, drive the stop screw with a 1.5 mm (0.0591 in) hex key through the access hole in the nose and the sleeve hole facing it, until its collar seats in the spotface and can go no further; thread locker. Nothing keys the plunger in rotation until this moment, and the screw simply refuses to seat if the slot is not brought round to it. Its dog now runs in that slot.
9. Release the plunger, then the fixture: the sleeve advances, seats the balls on the plunger and the socket is armed.
10. Collar slid onto the nose from the front, up to the slots.
11. Two dog-point screws through collar and slots into the sleeve, set flush with the collar outside diameter within 0.2 mm (0.00787 in), checked with a depth gauge; thread locker.
12. Disassembly: the reverse. Dog-point screws and collar first; then the sleeve pushed back to the assembly position and clamped there, which frees the balls and lines the access up again; then the stop screw unscrewed, which releases the plunger; then the plunger and its spring, the balls, and last the sleeve.

The control system must enforce the following operating rules.

1. Indexing requires torque control and a calibrated estimate of the axial load on the ribs, excluding the plunger spring load. Keep that load at 30 ± 1 N (6.74 ± 0.225 lbf) and speed below 1 deg/s.
2. Hold the collar fully retracted during indexing. Above 1.10 mm (0.0433 in) axial mis-seat, apply +0.32 ± 0.01 N·m (0.236 ± 0.00738 lbf·ft) in one fixed direction; parasitic roll torque must stay below 0.02 N·m (0.0148 lbf·ft). Below that height, remove the bias and let the axial load seat the V flanks.
3. Require an axial position measurement accurate to ±0.02 mm (±0.000787 in). Release the collar only after full seating; verify the locked sleeve position before carrying load. Abort and unload if not seated within one 20-degree pitch or if any force, torque or speed limit is exceeded. Never increase indexing push to 100 or 300 N (22.5 or 67.4 lbf).
4. Release: pull the collar, then HOLD it back for the whole withdrawal of the stud.
5. Let the collar go only once the plunger has reached its forward stop (socket armed).
6. An emergency stop during withdrawal must leave the collar pulled back.

11. Relation to Prior Work

Nothing in the mechanism is new, and this is deliberate: a standard should be built from elements whose behaviour is known. The container corner fitting and the twistlock [3, 4] show that a passive fitting and an active lock can organise an industry. Spacecraft grapple fixtures and probe-and-drogue docking systems use a passive target and a capture cone, and the docking standard [7] shows that such an interface can be shared. Maxwell's coupling [8, 9] gives the repeatable seat. Balls driven by a wedge or a cam retain tools in robot tool changers and workpieces in zero-point clamping systems, and push-to-connect couplers lock on insertion. Standard interfaces for on-orbit servicing such as HOTDOCK [14] combine mechanical, electrical and data connections for modular spacecraft.

All of these connect a machine to a machine. To my knowledge none is designed so that the passive half is at the same time a handle for a person, fixed with the screw of a cabinet knob. That combination, and the proposal that it be standardised, is the contribution of this paper. I have not searched the patent literature.

12. Limitations and Open Work

Assembly and manufacture. Two assembly conflicts found by an earlier audit of this design have been removed rather than documented: the collar no longer carries integral pins that could not cross its slots, and the nose is no longer held by radial screws whose heads overlapped the mounting flange. The fastener envelopes now modelled include the M6 mounting screws and their key, and a sweep of the collar along its front assembly path is collision-free. What remains open is the step from a defined design to production: drawings with datums and geometrical tolerances, a supplier drawing for each spring and one for the special dog-point screw, a tolerance stack that carries the limits of Table 3 through capture, seating, stroke and spring length, and confirmation of materials and heat treatment. The control that holds the collar back until the socket has re-armed, including through an emergency stop, is a written rule here and not yet a mechanism. No single feature dominates the process choice; the flat-bottomed bore and the finishing of the hardened ball seats are the two that need a written operation.

No prototype has been built. The principal open issues are retention against shock and vibration, the complete structural load path into the object, tolerance-dependent operation, and everything that only a test can settle. The energy barrier of Section 7.1 covers a sleeve that starts at rest and nothing else; the machine's own inertia during an emergency stop is exactly the case it does not cover. The geometry has been checked in the CAD model for the absence of interference in the locked, armed and released positions, and the first aligned insertion check sampled 49 positions. A second analytic kinematic check evaluated all six balls and moving rigid parts at 33 positions, spring envelopes at eight positions, and prescribed lateral centring paths at 30 poses, without a detected collision in those sampled operating poses. These checks do not cover all off-axis approaches or tolerance combinations. The springs, forces and contact limits use the approximations described above; nothing has been confirmed by finite-element analysis or by test. The following also remain unknown: the real sharing of load between the six balls, calculated here for idealised patterns of error only; wear of the seat and of the ramp over many dockings; behaviour with a dirty or damaged stud; the effect of vibration on the wedge; the stiffness of the rib flanks relative to the ball paths, sampled here at ratios 0.25, 1 and 3; the ratings of studs that are not hardened; and whether a tag can be read from inside the cage. The port also has limits that follow from its principle. Aligned insertion needs up to 30 N (6.74 lbf); the indexing protocol additionally requires 30 ± 1 N (6.74 ± 0.225 lbf) net rib load, plus plunger spring load, and a bounded roll torque that the object must react, and a lateral compliance of 5 mm (0.197 in) in the arm or the object. A head of 16 mm (0.630 in) is small for a hand on a heavy door, where a larger size, or a conventional bar handle carrying a stud, is more appropriate; its ergonomics have not been tested. A machine locked to a fixed object is a hazard: the manual release, a breakaway load and the behaviour on loss of power belong in the standard. So is the armed socket itself, which closes on any object between 12 and 16 mm (0.472 and 0.630 in) across that is pushed into it, a finger included, with a radial force of about 8 N (1.80 lbf) per ball, and up to 91 N (20.5 lbf) per ball for an object close to 12 mm (0.472 in); it is opened by the collar. A thin tool pressed against the rim of the plunger, off the axis, can let a ball drop into the bore, after which the socket must be taken apart; so can releasing the collar before the stud is clear. Finally, ball-lock couplings are the subject of many patents, and no search of prior art or freedom-to-operate analysis has been performed.

13. Conclusion

I have proposed a port that lets machines manipulate everyday objects without hands. Interfaces made for the human hand leave the machine to solve perception, grasp planning and contact control anew for each object. To remove that burden I proposed a passive stud that is a knob for the person and a datum for the machine, and a socket that accepts ±5 mm (±0.197 in) of error, locks after controlled axial and torsional seating, holds without continuous power through a wedge whose retention rests on an energy barrier rather than on friction, and seats on a coupling that defines the pose by geometry. The stud is a single solid part; the difficulty is concentrated in the socket, of which each machine needs one. Its elements are borrowed from interfaces that have long carried freight, tools and spacecraft. What remains is to build it, test it, and agree on it.

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