INSTALLATION TOOLS FOR HYDRAULICALLY FITTED COUPLING HUBS—PRECAUTIONS AND DESIGN REQUIREMENTS

by

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ABSTRACT

The tools used to install hydraulically fitted hubs need to be as carefully designed and manufactured as the couplings themselves. The authors describe an installation of coupling hubs on upgraded machines where tools that were designed to be used on a coupling for a 6.0 in diameter shaft were adapted to mount a coupling on an 8.0 in shaft.

The larger forces involved caused a threaded rod to fracture and the coupling hub to be expelled from the shaft end with considerable force and velocity. An after-the-fact analysis of these "design alpha" tools is presented, along with a technical description of the forces acting upon a newly mounted hub and the velocity with which such a hub could be expelled from a shaft end. The 8.0 in hub was successfully installed on the shaft using modified “design alpha” tooling for expediency, but this modification was considered to be only a stopgap solution.

The authors designed new “design beta” tools to accomplish the installation of the coupling hubs safely. The analysis of the “design beta” tools is described along with the important criteria to be considered for coupling hub installation tools.

INITIAL INSTALLATION ATTEMPT

The coupling hub for the 8.0 in diameter motor shaft was to be installed while the motor was on a transport trailer, prior to being mounted on its base plate. The plant technicians had reviewed the limited instructions that had come with the “design alpha” tooling intended for the 6.0 in diameter shaft couplings and attempted to use this same equipment on the 8.0 in motor shaft coupling. The machinery train arrangement (Figure 1) is an expander/axial compressor/steam turbine/gear/motor layout. Figure 2 is a cross section view of the motor shaft with its coupling hub.

![Figure 1. Schematic of Equipment Train.](image)

During the first try to mount the motor hub, the “design alpha” tooling was used as-supplied with its “20 ton, two inch stroke” hydraulic cylinder. The configuration of the “design alpha” tools is shown schematically in Figure 3. The dilation pressure was raised to 25,000 psi and the advancing pressure was raised to 8000 psi on the 4.70 sq in cylinder reaching about 38,000 lb. The coupling was advanced to within ⅛ in of the desired position. However, it was not possible to advance the coupling any further with the existing equipment.

For the second attempt, the technicians used a larger “30 ton, 2.5 in stroke” hydraulic cylinder that had been stored with the tooling and had been used previously with the “design alpha” tooling on the 6.0 in shaft coupling hubs. Again, the dilation pressure was raised to 23,000 psi and the advancing pressure to 8000 psi. This was sufficient to advance the coupling hub to the desired position. However, with the larger piston area, 7.22 vs 4.70 sq in, the axial
force on the tools was now over 55,000 lb. As the position of the hub was being checked and prior to the release of the hydraulic pressure, the hollow 1.0 in-8 UNC (ASTM A193 Grade B7 with an approximate yield strength of 120,000 psi) threaded retaining rod fractured, propelling the coupling and the installation tools from the shaft end, hitting some railroad ties (used to support the dilation pump) and the steel trailer frame.

The "design alpha" tools are pictured (Figure 4), with one piece of the hollow fractured rod still threaded into the tooling adapter and the second piece with its intact nut. A photograph is shown in Figure 5 of the fractured rod posed on the inner 3/8 in hydraulic tubing, along with the locking nut that was still intact. Each end of the rod fracture is shown in a photograph (Figure 6).

POST INCIDENT INVESTIGATION

Although no one was significantly injured during the incident, the configuration of the hydraulic tubing leading to the dilation pump could have placed a technician directly in line with the shaft end.

Note in the photograph (Figure 7) that the dilation pump is connected with rigid steel tubing.

The following safety deficiencies of the "design alpha" tooling were highlighted:
The design margin for the tooling was only 1.1 times yield, even for the 6.0 in diameter shaft couplings for which it was designed.

The tooling was not capable of being used with each item of equipment in the machinery train.

The design of the tooling required that the technician operating the dilation pump stand in close proximity to the shaft end.

No instruction manual was provided, only limited instructions were detailed on the tooling assembly drawing.

The instructions provided did not specify any operating restrictions.

The tooling could readily be used improperly with larger than intended hydraulic cylinders. The tools should not have been able to be used with other general hydraulic equipment.

The use of these special purpose hydraulic tools should have been restricted to trained personnel.

MODIFIED “DESIGN ALPHA” TOOLS

The tools were modified to use a more robust hollow threaded rod to complete the installation during the limited time available. A 1.50 in-8 UN threaded rod was made from ASTM A193 Grade B7 material, with a 7/16 in diameter hole. This increased the strength of the rod by 325 percent over the one that fractured. Additional hydraulic tubing was used between the dilation pump and the center port on the machine shaft to allow the technician to stand further away from, and out-of-line with, the shaft end.

The same “30 ton, 2.5 in stroke” cylinder was used and the hub was successfully installed. This tool arrangement is pictured in Figure 8. However, this stopgap solution was not considered satisfactory for future use, so a new set of tools, specifically designed for the machine train, was ordered.

Where:

\[ P = \text{Dilation pressure, psi} \]
\[ k_l = \text{Ratio of total hub length to pressurized hub length} \]
\[ E = \text{Tensile modulus, 30,000,000 psi} \]
\[ I = \text{Diametrical interference, in/in} \]
\[ q = \text{Hub ID/OD ratio, in/in} \]

For this case:

\[ \text{hub OD} = 12.140 \text{ in, nominal} \]
\[ \text{hub ID} = 7.898 \text{ in, average} \]
\[ q = 0.6506 \text{ in/in} \]
\[ I = (8.0000 \text{ to } 7.9830) / 8.0000 = 0.002125 \text{ in/in} \]
\[ k_l = 9.719/9.418 = 1.032 \]
\[ P = 18,970 \text{ psi} \]

Because of the taper, there is an axial component of this dilation pressure acting on the hub. The annular area against which this pressure acts is defined by the seal diameter at the large end and the shaft diameter at the small end.

\[ F_a = P \pi/4 (d_{seal}^2 - d_{shaft}^2) \] (2)

Where:

\[ P = \text{Dilation pressure, 18,970 psi} \]
\[ d_{seal} = \text{Seal diameter, 7.9920 in} \]
\[ d_{shaft} = \text{Shaft diameter, 7.8302 in} \]
\[ F_a = \text{Axial force, 38,125 lb} \]

For a short period of time after the dilation pressure is released while the fluid is still trapped in the hub/shaft interface, the hub can readily move off of the tapered shaft with considerable force and, if unconstrained, with considerable speed. The trapped fluid apparently acts as a hydrodynamic film with very little friction.

The potential energy, PE, stored in the elastic expansion of the hub is equivalent to one half the product of the force required to expand the hub, Fexp, times the amount of radial expansion.

\[ F_{exp} = P \pi d_1 \] (3)

Where:

\[ P = \text{Dilation pressure, 18,970 psi} \]
\[ d = \text{Hub ID, 7.898 in, average} \]
\[ l = \text{Hub length, 9.719 in} \]
\[ F_{exp} = 4,575,100 \text{ lb} \]

\[ PE = \frac{1}{2} F_{exp} l d/2 \] (4)

Where:

\[ l = \text{Diametrical interference, 0.002125 in/in} \]
\[ PE = 19,196 \text{ in-lb} \]

The velocity with which the hub could be expelled from the shaft end would be the result of complete conversion of the potential energy, PE, into kinetic energy, KE.

\[ KE = \frac{1}{2} m v^2 \text{ or since } KE = PE \]
\[ v = \sqrt{2} (2 \text{ PE} / m) \] (5)

Where:

\[ m = \text{Hub mass, 0.5 lb sec}^2/\text{in, (193 lb weight)} \]

Thus:

\[ v = 277 \text{ in/sec or 23 ft/sec or 16 mph} \]
That velocity does not seem very high, but it is equivalent to
dropping the 193 lb hub from an 8 1/4 ft height.

\[ h = \frac{v^2}{2g} \]  

(6)

Where:
\[ \begin{align*}
  v &= \text{Velocity, 277 in/sec} \\
  g &= \text{Acceleration due to gravity, 386.4 in/sec}^2
\end{align*} \]

Thus:
\[ h = \text{Height, 99.3 in or 8 1/4 ft} \]

A list is shown in Table 1 of the equivalent drop height in feet
for commonly used interference fits. This list is for generic steel
coupling hubs with a nominal ID/OD ratio of 1.5 to one. Table 1
was constructed from:

\[ H = \frac{E \rho^2}{12} \left( \frac{1}{q^2} - 1 \right) \left( 1 + q^2 \right) \left( 1 - q^2 \right) + 1 \]  

(7)

Where:
\[ \begin{align*}
  H &= \text{Height, ft} \\
  \rho &= \text{Density of steel, 0.28 lb/in}^3
\end{align*} \]

Equation (7) is a combination of Equations (1), (2), (3), (4), (5),
and (6), plus an expression for the hub weight.

**Table 1. Expulsion Velocity Equivalent Drop Height.**

<table>
<thead>
<tr>
<th>Interference Fit inch/inch</th>
<th>Equivalent Drop Height ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0015</td>
<td>4.4</td>
</tr>
<tr>
<td>0.0020</td>
<td>7.8</td>
</tr>
<tr>
<td>0.0025</td>
<td>12.1</td>
</tr>
<tr>
<td>0.0030</td>
<td>17.5</td>
</tr>
</tbody>
</table>

The “design alpha” tools (Figure 3) consist of:
- Inner pressure tube (1) to supply dilation pressure to the shaft
center
- Externally threaded (2.5 in-12) adapter (2)
- Internally threaded coupler (3) with a 2.5 in-12 thread to connect
to adapter (2) and an internal one inch-8 UNC thread to connect to
- Hollow threaded (one inch-8 UNC) rod (4) connected to coupler (3)
- Hollow support cup (5) (a welded assembly)
- Commercially available hollow hydraulic cylinder (6) (shown
  pictorially only)
- Nut (7) and washer (8)

The adapter (2), coupler (3), and the hollow threaded rod (4),
become tensile members when the hydraulic cylinder is
pressurized to push the hub up the taper against the axial force
component of the dilation pressure, specifically the 38,125 lb force
noted previously in Equation (2). The base of the hydraulic
cylinder (6), and the hollow support cup (5), become compressive
members during installation.

The hollow threaded rod is the highest stressed member and is
the part that fractured. Note that the threaded rod (Figure 5)
fractured in the middle of its length, not near the junction at either
end as would be expected. A possible reason for the unexpected
fracture location could be a machining mismatch, resulting from
boring the rod one half the depth from each end, meeting in the
middle.

A finite element analysis (Figure 9) was done of the threaded rod
(4). The rod is modelled as threaded into the coupler (3), which
was fixed at its bottom, and loaded by the nut (8) at the top. The
model is an axisymmetric model with the centerline on the left. The
loading for this analysis was 8000 psi on the 4.70 sq in area of the
“20 ton, two inch stroke” hydraulic cylinder for which the tool was
designed, thus 37,600 lb force.

![Finite Element Model of “Design Alpha” Tools—Overall View.](image)

Note in the closeup view (Figure 10) of the portion of the rod
threaded into the coupler (3), the nonuniform loading of the first
few threads. These threads are stressed well above their yield
strength that was estimated at 110,000 psi maximum, thus allowing
deforination and elongation to occur.

A closeup view is shown in Figure 11 of the center portion of the
threaded rod. Although the “bulk” stress in the rod is in the 70,000
to 80,000 psi range, the peak stress at the root of the threads is
123,000 psi. (Note that the rod was actually loaded to about 57,800
lb force with the “30 ton, 2.5 in stroke” hydraulic cylinder giving
stresses of about 190,000 psi.)

Note in the closeup view (Figure 12) of the nut/rod junction, that
a condition exists that is similar to that at the coupler end (Figure
10). A failure at either end of the rod would not have been
unexpected.

A finite element analysis was done of the complete tool
assembly (Figure 9). This analysis did not indicate any other areas
of concern.

**“DESIGN BETA” TOOLS**

The “design beta” tools were implemented to install all seven
coupling hubs in the FCCU power recovery train with as much
commonality of parts as possible, while maintaining adequate
strength margins for safety. This was achieved using three
assembly configurations (Figure 13) with each having three core
pieces (1), (2), and (3) while sharing two seal adapters (4) and (5),
and two pushing adapters (6) and (7). Each configuration uses
the same dilation pressure tube (8). Note that the middle configuration
is specific for the coupling hub on the axial compressor at the
thrust bearing end. This hub has a restricted nut recess diameter
limiting the OD of cylinder (3). The two pusher adapters (6) and
(7) enable using the one cylinder (3) with all seven hubs. Since the
configuration with the highest axial load is the one for mounting the hub on the 8.0 in motor shaft, that configuration was analyzed by a finite element analysis overall model (Figure 14).

The axial force needed to balance the axial component of the dilation pressure remains the same as that applied to "design alpha," namely 38,125 lb. The hydraulic pressure needed to develop the required axial force is determined by the effective piston area.

\[ \text{Phyd} = \frac{Fa}{(\pi/4 \ (d_a^2 - d_i^2))} \]  

Where:
- \( d_o \) = Outer seal diameter, 6.500 in
- \( d_i \) = Inner seal diameter, 5.000 in
- \( Fa \) = Axial force, 38,125 lb
- \( \text{Phyd} \) = Hydraulic pressure, 2800 psi

The threaded (2.5 in-12) stud (1) holds the pusher piston (2) in place against the axial force. This stud also holds the seal adapter (4) in place allowing a small amount of axial clearance. This stud becomes a tensile member during installation.

The seal adapter (4) for two of the configurations, including that for the motor shaft, exposes one face to the full dilation pressure of 18,970 psi.

\[ F_{\text{seal}} = \frac{P}{(\pi/4 \ (d_{sa}^2))} \]  

Where:
- \( P \) = Dilation pressure, 18,970 psi
- \( d_{sa} \) = Seal adapter diameter, 1.000 in
- \( F_{\text{seal}} \) = Seal assembly axial force, 14,900 lb

This force loads the 2.5 in-12 threads in the same direction as the piston/cylinder hydraulic axial force.

The highest axial stress, 21,700 psi, is seen at the root of the external threads of the stud (1) and is adequately within the material yield strength of 110,000 psi. Note the closeup view of that portion (Figure 15). The cylinder (3) becomes an internally pressurized vessel during the installation and experiences a stress of 24,400 psi. Note the closeup view of that portion (Figure 16).

This local area has an adequate factor of safety of 4.5 compared to the 110,000 psi yield strength. The thin wall over the O-ring groove becomes a tensile member loaded by the 2800 psi hydraulic pressure against the area of the O-ring groove.

\[ S_{\text{groove}} = \frac{\text{Phyd}}{(\pi/4 \ (d_g^2 - d_c^2))} \]  

Where:
- \( \text{Phyd} \) = Hydraulic pressure, 2800 psi
- \( d_g \) = Groove diameter, 6.845 in
- \( d_c \) = Cylinder ID, 6.500 in
- \( d_{OD} \) = Cylinder OD, 7.250 in
- \( S_{\text{groove}} \) = Stress, average tensile, 2260 psi

This average tensile stress of 2260 psi is trivial compared with the yield strength. The peak longitudinal stress, noted in the closeup view (Figure 17) near the O-ring groove, appears on the OD and is 10,120 psi.

A photograph of the "design beta" tools is shown in Figure 18. Note: the two assemblies in the upper left of the photo are for smaller machines in the same plant and are not part of the subject of this study.
CONCLUSION

Improperly designed or used high pressure hydraulic tooling can potentially cause significant injury to personnel and damage to
dangerous manner. High pressure flexible hoses, (or adequate length solid tubing configurations), must be used to keep personnel away from, and out-of-line with, the shaft end. The two hydraulic pumps, 35,000 psi for dilation and 10,000 psi for pusher piston/cylinders, should be mounted on a frame for stability and each must have a pressure gauge. A popular arrangement of frame mounted pumps is pictured (Figure 19).

![Frame Mounted Pumps](image)

Detailed instructions must be prepared by the tooling designer and reviewed by the tooling user to ensure safe operation of the tools. These instructions must clearly specify the operation limits, inclusive of the maximum pressures that may be applied.

The using organization must train the technicians who will use the tools on the proper procedures and the potential hazards inherent with high pressure hydraulic equipment.

The pump mounting frame, the connecting hoses (or tubing), and the pusher tools should be kept in a reusable storage container to protect them from loss or damage between overhaul actions.

REFERENCES
