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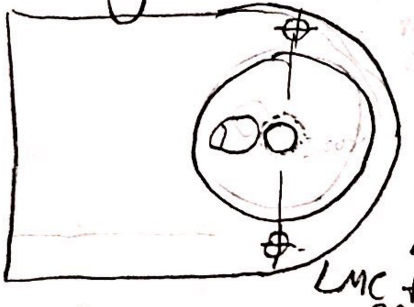
Homework 10: Tolerance stack-up

Mounting Block



for position of boss center, total tolerance is given by:

LMC of boss diameter

$\varnothing = 1.067$ $1.072 - 1.067 = $ .005 bonus

+.005 ⌖

.004

⌖.014

diametrical tolerance zone for position.

$\varnothing = .2505$

LMC for B-CBORE, as shown below

Side calculation for position of datum B - CBORE

LMC of CBORE diameter

+.003 ⊥

$\varnothing = .2505$ $.2505 - .2495 = $ .001 bonus

MMC

⌖ .004

hence $\frac{.014}{2} = \boxed{.007}$ radial tolerance zone and for the worst case scenario of this problem, the boss would be .007 lower (as depicted in my diagram) b/c gravity will then bring the displacer even lower and create risk of contact with the bottom of the air chamber.

For the mounting block's 4-40 clearance holes, positional tolerance given by:

MMC of hole

+.005 ⌖

$\varnothing = .111$ $.121 - .111 = $ .010 bonus

.003 ⊥ (from C)

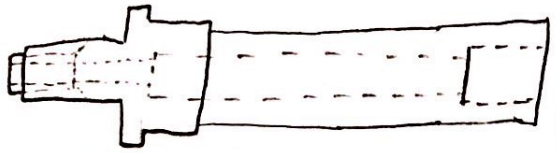
LMC of hole

.018

Thus the total radial tolerance in 4-40 clearance hole position is $\frac{.018}{2} = \boxed{.009}$ and in the worst case scenario for this problem, will move the holes (and subsequently air chamber) up by .009 for greatest risk of contact w/ the displacer

The GD&T callout for these holes actually references C(M) but I already accounted for the boss in this stackup so I didn't do it again. I also chose to ignore the angular tolerance in hole positions since the change to 180° makes these positions easier to get precise, and 1° is pretty negligible to begin with.

(2) Displacer Bushing Tube



Note: I assume perfect flushness between the DBT's largest front surface and the mounting block, since no contradictory GD&T data is given, and after physically machining both parts, I saw no opening between the two surfaces upon close inspection

Variance in fit between DBT's external thread and Mounting Block's internal thread:

DBT external thread - major $\varnothing .242 - .005 = \varnothing .237$ LMC for major diameter

Mounting Block int. thread - 1/4-28 UNF-2B

major $\varnothing .2500$ minimum MMC (by chart online) only minimum was listed for internal threads so I'll add .008 (to get major $\varnothing .258$ maximum 2B) since the .006 - .010 range seemed to be the standard difference between min and max for internal thread minor $\varnothing$ and I assumed similar for major

Note: I compared the LMC's (least mat conditions) b/c less material means more space for movement, approaching worst case scenario

Mismatch = $\varnothing .258 - \varnothing .237 = \frac{.021}{2} = .0105$ radial movement possible between int/ext threads, we will assume the bottoms interact (aka $\boxed{\text{DBT falls .0105 below}}$ perfect concentricity w/ mounting block hole

LMC for mounting block UNF-2B

LMC for DBT UNF-2A

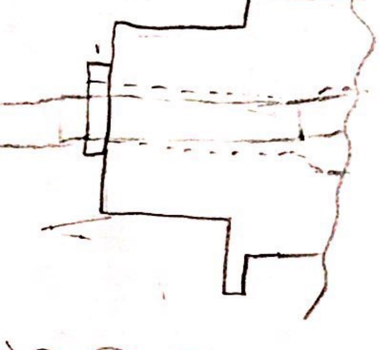
Now I will evaluate the fit between Displacer's rod and DBT's $\varnothing .1250$ reamed hole

First, note that the total positional tolerance of DBT's hole is $\varnothing .005$ (or radially, $\frac{.005}{2} = \boxed{.0025}$) since the GD&T callout doesn't say anything about Ⓜ for min/max diameter so we don't need to worry about bonus tolerance for anything. Additionally of note, I'll add that the $\varnothing .1250 {}^{+.0005}_{-.0000}$ callout for this hole implies a max $\varnothing .1255$ which I will consider for worst case scenario purposes

③ DBT and Displacer's rod (continued)

Now I'll consider the displacer itself, specifically, the long rod at its end. According to the provided Solidworks file, this rod has length ≈3" and importantly, diameter $\varnothing .1250$. By finding analogous McMaster data, this gives us bilateral tolerance $^{+.0000}_{-.0002}$ and straightness .0036" (or .0012"/inch roughly). Thus the LMC (of concern for worst-case scenario) is $\varnothing .1248$. Given that we are considering DBT hole $\varnothing .1255$, this leaves $\varnothing .1255 - \varnothing .1248 = .0007$ of diametrical empty space, translating into $\frac{.0007}{2} \approx$ .0004 of radial wiggle room. In the worst-case scenario, the Displacer Rod will fall .0004" so it and the DBT's hole meet at each's bottom.

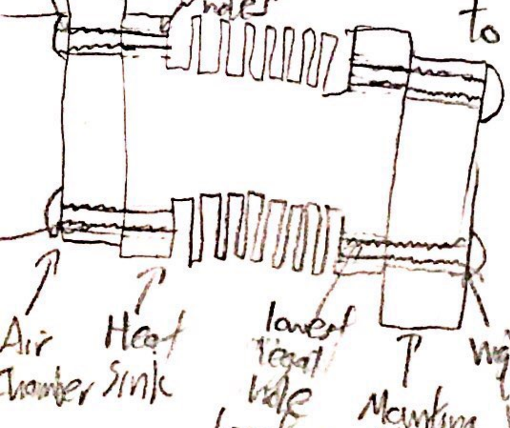
Note: Since this translational wiggle room is so small, I'll assume the angular wiggle room (in other words, the rod looking like this)



is even smaller, thus I'll neglect it, and assume the bottom edge of the rod is flush with the bottom edge of the $\varnothing .1250$ hole.

Mounting Blocks, Heat Sink, Air Chamber

Now I'll evaluate the interface (via 4-40 screws) between the mounting block and heat sink. First I'll note the worst case assembly will look like my ~~[illegible]~~ drawing to the left in terms of hole/4-40 screw locations (as I've labelled/drawn).



First, I'll evaluate the 2 4-40 screws connecting the mounting block to the heat sink. I've already (on page 1) accounted for the clearance holes being as high as possible on the mounting block, so now I'll account for the size of the screws relative to the clearance holes. For my bonus tolerance calculation on page 1, I assumed MMC of these holes (min $\varnothing .111$), so I'll stick with that since in a single scenario one feature maintains the same dimensions $\Longrightarrow$ *continued on next page*

(9) For the 4-40 UNC 2A fasteners, pg 283 of the book tells us the min major diameter is Ø.1061

This gives us a gap of .111 - .1061 = .0049

so half of this: $\frac{.0049}{2} \approx \boxed{.0025''}$ will be the amount the screw moves up from the center of the hole to get into position flush with the top of the mounting blocks clearance hole

Now I'll evaluate the tapped hole positions on either side of the heat sink (I'll actually consider just the right side, as drawn on my page 3, then multiply by 2 to account for the symmetrical left side).

- First, I'll assume the hole size is exactly the same as that of the fastener that threads into it — this is a standard assumption, because unlike screws fitting easily through clearance holes, screws and corresponding tapped holes must interface tightly (with little to no dimensional mismatch) for a strong fit
- Now the hole position is defined along a circle of Ø1.344, hence diametrical tolerance (for position) of assumed ±.005, translating into radial $\frac{.005}{2}$ = .0025. Now I'll assume a perpendicularity tolerance of .003 (since it's not specified otherwise, and $\boxed{\perp \mid .003}$ was called out for a similar hole on the mounting block. In terms of worst case scenario, we assume the feature looks something like this:

thus we can simply add .0025+.003=.0055 for the total effective positional tolerance generated by each pair of tapped 4-40 holes on the heat sink. Since there are two such pairs of holes we attain 2×.0055 = $\boxed{.011}$ as the total tolerance generated by the heat sink for our worst case scenario purposes.

Finally, I'll consider the fit (involving two 4-40 screws) between air chamber and heat sink. I just accounted for hole locations on the heat sink, so now I'll account for the 4-40 screws and the air chambers clearance holes they pass through. These clearance holes were machined to the same size Ø.116 as those in the mounting block, but now I'll assume LMC (max hole size Ø.121) since I've made no contradictory assumptions previously, and the biggest hole will allow the air chamber to travel as far up as possible, relative to the screws increasing risk of contact with the displacer.

(5.) Now as I said earlier, the 4-40 UNC 2A fasteners had minimum major diameter Ø.1061. This gives a gap of:

$.121 - .1061 = .0149$, so half of this:

$\frac{.0149}{2} \approx \boxed{.0075''}$ will be the amount the air chamber moves up (relative to concentricity between fasteners and holes)

Finally, we can say the hole positions (on the air chamber) themselves have tolerance $\boxed{.005}$ since similar holes on the mounting block engineering drawing had a $\boxed{⌖ | Ø.005}$ GD&T callout

Now onto the final step: the Displacer itself.

Note that I've already accounted for tolerances pertaining to its Rod, including the assumption of LMC min Ø.1248 and straightness .0036". Now I'll evaluate its fit into the Displacer cold cap. The cold cap has a .1250 hole, so assuming this was reamed with a .1250 reamer, I'll apply typical tolerance of $^{+.0002}_{-.0000}$ to get LMC of max Ø.1252 hole in displacer cold cap. This allows for a radial tolerance of $\frac{.1252 - .1248}{2} = .0002$, giving .0002" of wiggle room between

LMC for Cold Cap hole | LMC for Rod Ø

Rod and Cold Cap. Adding a positional tolerance of Ø.0001 (which is intentionally small since making the Cold Caps hole concentric to its OD should be easy by machining the whole part at once on the same machine zeroed at the center) gives a total $\boxed{Ø.0003}$ tolerance for the cold cap-rod fit.

Now for the fit between the Displacer and the Cold Cap. According to the Provided CAD files for these parts, the Cold Cap has Ø0.958 pressed into Displacers Ø0.938. This dimensional overlap lets us assume that the two parts are perfectly coincident along the entire body of the Cold Cap. So the only task left is the tolerance of the Displacer itself. I found a pdf online from ve-ca entitled Precision Pressings and Deep-Drawn Stampings which stated that the progressive draw process can leave "finished part tolerances as low as ±.02 mm" ($\approx$.0008")

(6.) So I'll assume the OD of the Displacer varies by diametrical tolerance ∅.0016 and thus can come up to $\boxed{.0008"}$ closer to contact with the bottom of the air chamber than its nominal size would suggest.

Finally, I'll put it all together!

The air chamber ID is dimensioned to ∅1.072, which after assuming negligable reaming tolerance, gives us an OD of ∅1.072 for the Displacer under ideal conditions (meaning when all nominal sizes end up exactly correct in the part, all holes and fixtures are perfectly concentric, etc.) All the tolerances I've gathered up (and circled) throughout this assignment contribute directly to a more likely contact between the bottom of the Displacer and the bottom of the air chamber, by either 1.) moving up the air chamber or 2) moving down the displacer. Thus to finish off, I can just subtract these all from ∅1.072 to get the worst case scenario displacer diameter. See below

∅1.0720
- .0070 for position of mounting block bars going down
- .0090 for position of mounting blocks 4-40 clearance holes going up
- .0105 for position of DBT going down, relative to mounting block
- .0036 for straightness tolerance of Displacer Rod bringing its far end down
- .0025 for position of DBTS hole (to fit Displacer Rod) going down
- .0004 for Displacer Rod to move down within DBTS hole for it
- .0025 for 4-40 screws to move up with mounting blocks clearance holes
- .0110 for heat sink tapped 4-40 holes to go up and down on either side
- .0075 for position of air chamber to move up relative to its 4-40 fasteners
- .0050 for position of air chamber holes to move up relative to the rest of the part
- .0003 for position of cold cap to move down relative to Displacer Rod
- .0008 for radial tolerance of the displacer itself

$\boxed{∅1.0119}$ as the maximum nominal diameter of the Displacer

Note that in the subtraction above, very often my "position" identifier actually accounted for other dimensions as well (such as diameters, perpendicularities etc.) which became absorbed into my "position" somewhere in my 6 pages of work

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