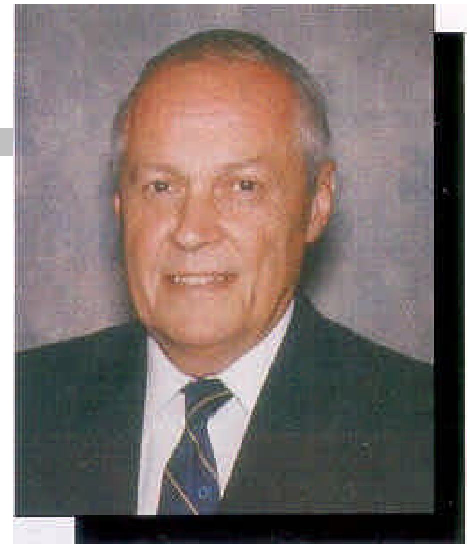
About the Author/BENGT BLENDULF

Bengt Blenduif is the president of Clemson EduPro, Inc. and is a professional educator specializing in fastener engineering education. Bengt's involvement in fastening technology goes back to 1966 when he joined one of the leading European fastener manufacturers. He has been in the mechanical/ metalworking field since 1961. Educated in mechanical engineering in Sweden, he moved to the United States in 1974 to start a subsidiary for a Swedish fastener manufacturer. Bengt also served for 8 years on the faculty in the College of Engineering and Science at Clemson University. Since 1997 he (Clemson EduPro, Inc.) teaches highly rated courses in Fastening Technology and Bolted/Screwed Joint Design in the U.S., Canada, and Mexico, primarily for engineers, but also for other fastener professionals. Bengt serves as chairman of the Bolting Technology Council and is the author of an extensive lecture book as well as over 80 technical papers and articles related to fasteners and international standardization. Bengt's business address is: Clemson EduPro, Inc., P0 Box 1862, Clemson, SC 29633-1862; phone/fax 864-654-1126; e-mail: edupro@carol.net, website: wwwedupro.us.



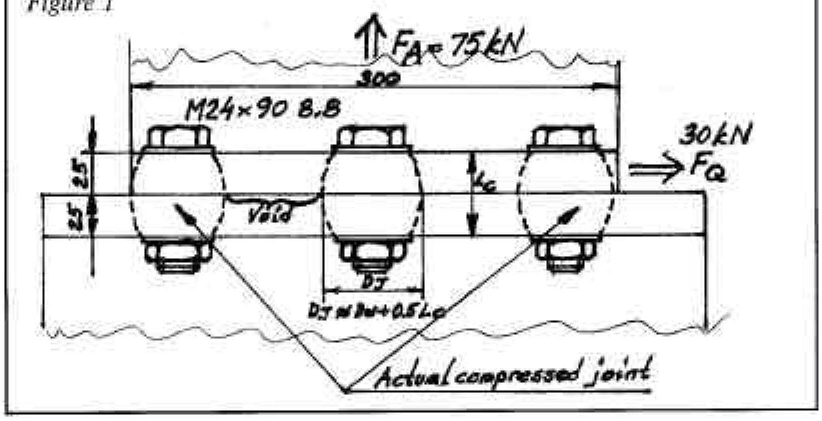
Big Bolts Better?

Choices for Performance and Economy

Few fastener engineers had a better understanding of bolted/screwed joint economy and performance than my mentor and teacher in fastening technology, Carl Dock of Sweden. Unfortunately, Carl passed away earlier this year and will be greatly missed on both sides of the Atlantic as a great person and fastener expert in ISO and many other areas. Carl developed the IPC concept (In-Place-Cost) that takes into account all aspects of a joint, such as drilling, tapping, purchasing, inventory keeping, handling, assembly, drive systems and other related issues. From this concept, many new fasteners have been developed, particularly in the area of thread rolling screws, screws and nuts with captive components (i.e. Sems) drilling screws, etc.

This section will look at ways of optimizing a joint system in order to find the best solution based on performance and economy. In Figure 1 we have details of a machine using 3 M24x90 8.8 bolts, Class 8 nuts and hard washers (HRC2O mm) to connect two parts with external loads going in both axial and transverse directions. The axial load F_A, 75 kN (approx. 17 000 lbf), and the transverse load F_Q, 30 kN (approx. 6 750 lbf), are carried by the 3 M24 bolts. This is the worst-case scenario for bolted joints; side loads are very tricky to deal with if we wish to have a frictional hold rather than a pure shear joint (which is a lot easier to design). This is particularly true when we also have an axial component trying to separate the clamped parts, thereby decreasing the pressure in the parting plane.

Figure 1

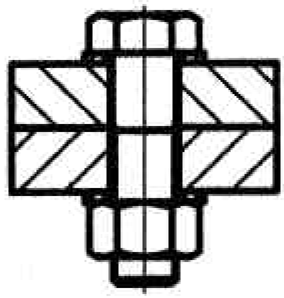


By using the 3 "big" bolts we are able to carry the two loads (F_A and F_Q), but if we look at the loading plane we find that the actual joint areas between the two joined plates are somewhat limited. A good approximation of the actual joint area diameter, having two plates of the same thickness, is to add the bearing surface diameter of the bolt head or nut face (D_W) and the thickness of one of the plates (or $D_W + 0.5L_C$, where L_C is total clamping length). In our case we have voids in the load distribution, and those voids will not help in preventing slipping between the two planes. I have used the design guideline VDI 2230 (German Engineering Society) to calculate and verify this joint and SR1 (a software based on VDI 2230) for the optimizations. VDI 2230 is, in my opinion, the most sophisticated calculation base for highly stressed bolt joints in existence today. In my engineering classes (Fastening Technology and Bolted/ Screwed Joint Design) I use the VDI approach since it is much superior to materials in our current college textbooks. The first edition was issued in 1976 and the latest edition in February 2003.

Figure 2 is a principal printout of one of the bolt locations, where the total loads have been divided by 3 (the number of bolts). Even with the VDI and SR1 we still have to break down a multiple axes joint into single axis bolt locations. I have written some explanatory notes on the table drawings in this chapter to familiarize you with some of the terms (abbreviations are from German). In this case we are using 90% of the bolts yield capacity, which is right at the proof load level (stress under proofing load). The Rp0.2 and proof load have been adjusted to take the torsional effect (thread friction during tightening) into account.

please turn to page 178

Figure 2



ISO 4014 - M24 x 90 - 8.8 SW36 WAF				
	OD	ID	thickn.	
i	da	di	l	material
Washer 1	45.0	25.0	4.0	AISI 1045 HEAT TR. Min HRC20
2	100.0	26.0	25.0	AISI 1018
3	100.0	26.0	25.0	AISI 1018
Washer 4	45.0	25.0	4.0	AISI 1045 HEAT TR. HRC20

bolted joint with nut (DSV)		
ANSI 1 - SW 36 WAF		
h M	mm	21.5

Company X
Clemson EduPro/Blendulf

Using 90% of reduced Rp 0.2

Tightening factor

LOAD (T=20°C)		
Axial load FA max	N	25000
FA min	N	0
Transverse FQ	N	10000
Clamping FKrequired	N	65000
FK min	N	66254
FM-Rp	N	187089
FM,max	N	168380
FMmax,req	N	166374
FMmin,req	N	103984
Relaxation fz	mm	0.0195
Fz	N	14941
FV min,req	N	89043
FV min	N	90296
FV max	N	153439
FSA max	N	957.1
FPA max	N	24043
F Rm	N	292578
F Rp	N	232653

ASSEMBLY (Nut driven)		
nue Rp		0.90
alpha A		1.60
MA max/min	Nm	775.3 / 484.5
alpha max/min	deg	26.37 / 15.48

FRICTION	min	max
μG thread	0.140	0.140
μK bearing	0.160	0.160
μTr parts	0.160	0.160
K factor	0.191	"nut factor"

FACTORS OF SAFETY (T=20°C)		
safety against loosening	FM,max/FMmax,req	1.01 ?!
safety yield point red.B	SF=Rp/Sig.redB	1.29
safety ag.fatigue fract.(concen.)	SD=Sig.AS/Sig.a	30.24
safety plate surface pressure	Sp=pG/pmax	1.55
safety against slipping due to FQ	SG=FKmin/FKQreq	1.06 ?!

$$\left.\begin{array}{l}\text{Load going to bolt}\\ \text{Load going to plates}\end{array}\right\}\ \textit{from}\ F_A$$

					Date	Name	
				Compl.	12/01/2003		BB65a
				Check			
				Stand.			
							1 (Page / Pg.)
Cond.	Modification	Date	Name	Bengt Blendulf Clemson, SC-USA			

C:\SR1\TRAIN\BB65a.SR1

12/01/2003 15:39 - SR1+ V12.11 #0000 - Bengt Blendulf Clemson, SC-USA - C:\SR1\TRAIN\BB65a.SR1

please turn to page 180

BIG BOLTS, *from page 178*

All safety factors are above 1, which is normally acceptable, two are dangerously close, namely 1.01 against risk of coming loose and 1.06 against possibility of slipping. Since the latter is crucial to this design (with a side load of 309 kN) we may consider a redesign to improve these numbers. To go higher than 90% of Rp0.2 (yield) is risky business with a torque wrench, however well calibrated (it will still cause a typical tension scatter of ±23% alpha A=1.6). The alpha A factor is the ratio between the highest assembly load and the lowest caused by the relative inaccuracy of the tightening equipment. Therefore, a designer would likely think in terms of bigger bolts. The next larger standard size is M27, so let's see what difference that would make for this joint.

M27×90 8.8 (3 bolts)

LOAD (T=20°C)		
FA max	N	25000
FA min	N	0
FQ	N	10000
FKreq	N	65000
FK min	N	81002
FM-Rp	N	246416
FM,max	N	197133
FMmax,req	N	171530
FMmin,req	N	107206
fz	mm	0.0196
Fz	N	18168
FV min,req	N	89041
FV min	N	108044
FV max	N	178969
FSA max	N	968.6
FPA max	N	24041
F Rm	N	381307
F Rp	N	303206

ASSEMBLY (Nut driven)		
nue Rp		0.80
alpha A		1.60
MA max/min	Nm	1017 / 635.6
alpha max/min	deg	25.39 / 15.87

FRICTION	min	max
μG	0.140	0.140
μK	0.160	0.160
μTr	0.160	0.160
K	0.191	

FACTORS OF SAFETY (T=20°C)		
safety against loosening	FM,max/FMmax,req	1.15 +
safety yield point red.B	SF=Rp/Sig.redB	1.44
safety ag fatigue fract.(concen.)	SD=Sig.AS/Sig.a	39.21
safety plate surface pressure	Sp=pG/pmax	1.63
safety against slipping due to FQ	SG=FKmin/FKQreq	1.30 +

Figure 3

Costs more than 2x!

In this case we are backing off to using 80% of yield since we can produce a residual clamping force FKmin of 81002 N, only needing about 65000 N. This alternative gives us good safety factors, but at the expense of cost. Using list prices from a 1998 catalog, the cost of the M24 example with 3 bolts M24x90, 3 M24 nuts and 6 hardened washers would be **$26.70**. With the M27 alternative the cost will more than double to **$68.49**! Not a very good prospect when everybody is pressured to play it economical (or even cheap). Add to that the cost of larger size holes to drill and much higher torque values. From an IPC standpoint, this is not a desirable direction to go.

Company X

Clemson EduPro/Bendulf

M20×80 8.8 (5 bolts)

LOAD (T=20°C)		
FA max	N	15000
FA min	N	0
FQ	N	6000
FKreq	N	40000
FK min	N	53843
FM-Rp	N	141302
FM,max	N	127172
FMmax,req	N	106024
FMmin,req	N	65640
fz	mm	0.0196
Fz	N	11167
FV min,req	N	54473
FV min	N	68316
FV max	N	116005
FSA max	N	527
FPA max	N	14473
F Rm	N	203179
F Rp	N	181564

ASSEMBLY (Nut driven)		
nue Rp		0.90
alpha A		1.60
MA max/min	Nm	491.5 / 307.2
alpha max/min	deg	31.98 / 19.99

FRICTION	min	max
μG	0.140	0.140
μK	0.160	0.160
μTr	0.160	0.160
K	0.193	

FACTORS OF SAFETY (T=20°C)		
safety against loosening	FM,max/FMmax,req	1.21 +
safety yield point red.B	SF=Rp/Sig.redB	1.19
safety ag.fatigue fract.(concen.)	SD=Sig.AS/Sig a	38.10
safety plate surface pressure	Sp=pG/pmax	1.30
safety against slipping due to FQ	SG=FKmin/FKQreq	1.44 +

Figure 4

Good safety factors
Cost = M24 alternative

Since this design leaves rather big distances between the bolts (when using 3) and uneven clamping on the total joined surface, we may want to try to use *more and smaller bolts*. Our next option could be to use 5 M20x80 8.8 bolts with nuts and hard washers. We can use 10 mm shorter bolts, since nuts and washers are a little thinner. As we can see in the safety factor table in Figure 4, we have as good or better result with 5 M20 bolts as we had on 3 M27 bolts. Also, the cost for this alternative with 5 M20x80 bolts, 5 M20 nuts and 10 hard washers is **$24.70** or about the same as for the original M24 alternative. So, smaller is often a better solution than going bigger! Of course, there will be 2 more holes to drill and 2 more nuts to tighten, but since they are also smaller it will not be that much of a difference. The IPC (In-Place-Cost) will not change very much, but the quality of the joint will be vastly improved.

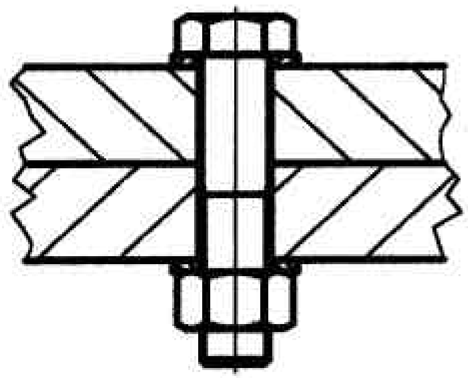
When we are on the roll of optimizing this joint, let's find a way of engaging the entire "stretch or row" of fasteners so that we have a better, continuous pattern for the frictional hold. We could look at the bridge builders and steel contractors for some principal approaches to this. Next time you drive past a bridge, take a look at the tight bolting pattern used to connect the beams for the span. (*But keep your eyes on the road so you don't run into the bridge!*)

This technique, with a close bolt pattern, develops a large friction surface, which will help the joint to stay close even when high transverse loads act on the bolted joints in the bridge span. If we "mimic" this approach for our mechanical joint, we can perhaps go one step further and use M16 fasteners. Let's see how this would work. (See Figure 5.)

Using the SR1 software I was able to carry out a variety of options for M16 fasteners relatively quickly. The best alternative proved to be using 6 M16 bolts in Class 10.9 (Figure 5). Since the machinery is to be used indoors, we did not need, for corrosion protection, any other surface treatment than the phos/oil that we usually get on these i 0.9 (Grade 8) products. Also, because of the strength of the material, only 75% of Rp0.2 (yield) is necessary for the preloading., As you can see in the safety factor table, we have almost as good values as we had for M20 and M27, and *a lot better* than the original M24 design. And the best part of all, the cost for 6 bolts, 6 nuts and 12 hard washers is now only **$14.70**, or about half of the original design. Even with more holes to be drilled and more (but smaller) fasteners to be tightened, this may be the lowest IPC of all the alternatives discussed here. With a mean torque of 243 Nm (179 lb ft) instead of 630 Nm (465 lb ft) it will be a lot easier to tighten these M16 fasteners using smaller torque wrenches and improving the ergonomics.

please turn to page 184

Figure 5



ISO 4014 - M16 x 80 - 10.9 SW24

l	de	di	l	material
1	30.0	17.0	3.0	AISI 1045 HEAT TR.
2	100.0	17.5	25.0	AISI 1018
3	100.0	17.5	25.0	AISI 1018
4	30.0	17.0	3.0	AISI 1045 HEAT TR.

bolted joint with nut (DSV)		
ANSI 1 - SW 24		
h M	mm	14.8

This would be my choice for this joint.

Company X
Clemson EduPro/Blendulf

LOAD (T=20°C)		
FA max	N	12500
FA min	N	0
FQ	N	5000
FKreq	N	32000
FK min	N	40342
FM-Rp	N	128799
FM,max	N	96699
FMmax,req	N	83252
FMmin,req	N	52032
fz	mm	0.0195
Fz	N	7956
FV min,req	N	44076
FV min	N	52419
FV max	N	88644
FSA max	N	423.6
FPA max	N	12076
F Rm	N	162936
F Rp	N	147268

180 *THE DISTRIBUTOR'S LINK*
BIG BOLTS, *from page 178*

MA max/min	Nm	299.2 / 187
alpha max/min	deg	42.62 / 26.64

FRICTION	min	max
µG	0.140	0.140
µK	0.160	0.160
µTr	0.160	0.160
K	0.193	

FACTORS OF SAFETY (T=20°C)		
safety against loosening	FM,max/FMmax,req	1.16
safety yield point red.B	SF=Rp/Sig.redB	1.42
safety ag.fatigue fract.(concen.)	SD=Sig.AS/Sig.a	32.28
safety plate surface pressure	Sp=pG/pmax	1.23
safety against slipping due to FQ	SG=FKmin/FKQreq	1.29

Good safety factors.
Low cost of fasteners.
Low IPS!

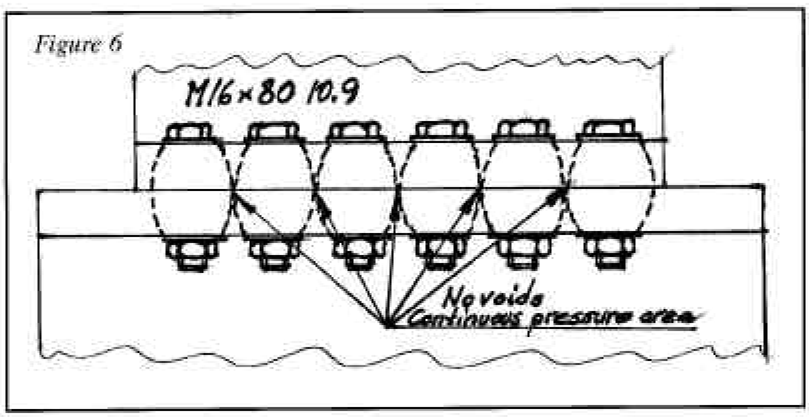
					Date	Name	
				Compt.	12/01/2003		BB65c
				Check			
				Stand.			
							1 (Page)
Cond.	Modification	Date	Name	Bengt Blendulf Clemson, SC-USA			Pg.

12/01/2003 15:00 - SR1+ V12.11 #0366 - Bengt Blendulf Clemson, SC-USA -

please turn to page 186

BIG BOLTS, *from page 184*

In figure 6 we can also see how the clamped "bodies" are developing a continuous pressure area, taking advantage of the better frictional hold we get with better spacing. This tight pattern is an absolute necessity if a joint is gasketed, because voids in the parting planes can cause leaks to develop. Gasket pressure should be kept uniform. There are a couple of other advantages to using smaller fasteners. The ratio between the diameter of the fastener and the clamping length is larger, which gives us more favorable elasticity signature of the joint (stiffer joint, springier fastener). Therefore, more of the external, axial load F_A will be absorbed by the compressive energy in the joint and less will go to the pre-loaded fasteners due to increased diameter to clamping length (d/L_C) ratio. Additionally, the endurance (fatigue) limits for smaller fasteners are higher per thread stress area unit than for larger diameter fasteners, which could be very important if external loads are cyclic/alternating.



The major lessons we have learned from optimizing the joint as we have done above is basically:

1. Bigger is not always better.
2. The geometry of the joint should be taken into account for best joint elasticity signature.
3. Transverse or combined loads require a lot more preload than pure axial loads.

The VDI 2230 Guideline

If we had calculated the joint alternatives discussed above by hand following the VDI 2230 it would have taken many hours because of the complexity of the mathematics and the many variables included. Having said that, the fact of the matter is that a highly stressed bolted/screwed joint is most often very difficult to design, particularly if "overdesign" is not an option due to economics and other restraints. For the serious design engineer involved in highly stressed bolted/screwed joint design, having access to the VDI 2230 Guideline is something I always recommend my students. It is now also issued as a German/English version (February 2003), which, of course, makes it a lot more accessible to the engineer in the U.S. It can be obtained from the publisher of all VDI guildelines:

Beuth Verlag GmbH
Burggrafenstrasse 6
D-10787 Berlin
Germany
Web: www.beuth.de

Software Version SR1

A very convenient way of making quick calculations of options can be found in SR1, a software program that contains not only the VDI 2230, but also other important data. It was developed by Ralph Shoberg of RS Technologies in Farmington Hills, MI in cooperation with Fritz Ruoss, a programming expert from Hexagon Software in Germany. With SRi we are able to analyze and verify our bolted joints more easily and accurately. This software can be obtained from:

RS Technologies
24350 Indoplex Circle
Farmington Hills, MI 48335
Phone 248-888-8260
Web: ww.rstechltd.com

Before we start using computer programs, however, we must fully understand what kind of "numbers" we are entering. No software in this design area will "think" for us. It will, however, speed up the verification of the designer's intentions and give us opportunities to quickly develop alternative solutions. The same thing applies to FEA (Finite Element Analysis) used by many engineers/designers today. Although very sophisticated, FEA does not design for us, but if properly used it give us very valuable guidance in our own decision making process.

In closing, big is not always better. Lean, mean and flexibility usually wins!