# Optimisation of Reinforcement

# Bar Couplers

by

Ian John Pugh

B.Eng. (Hons), A.M.I.Mech.E.

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## Synopsis

Sheffield based company Ancon Clark manufactures, amongst other products, a family of reinforcement bar connectors for the construction industry. These are mechanical devices used for end-on joining of two bars. Reinforcement bars are cast into concrete structures. Presently Ancon have approximately a 1% share of the world market for this product although 90% of their connector products are exported to the European market. High initial cost of the product means that market expansion is limited and an imminent all encompassing European Standard exceeds the current product performance capability.

The aim of this project was to redesign and develop this family of reinforcement bar connectors to produce a fully tested range to suit bar sizes from 8 to 50mm in diameter. The initial objectives were to reduce the product cost by 20% and increase product load carrying performance by 27%.

The Ancon connectors consist of a length of steel tube into which the ends of the two pieces of reinforcement bar (to be joined) are inserted. The bar ends are held in the tube by a series of aligned lock-shear bolts which are tightened and penetrate the bar whilst forcing it against two serrated 'grips'. Ancon also manufacture a number of connector variants such as connectors for two bars of differing diameter and connectors for situations where a structure will be completed/continued at a later date to the first section.

Two basic product development methods were considered, computer modelling and physical testing. Due to the complex nature of the product operating mechanisms computer modeling was seen as more costly and time consuming than testing of such an inexpensive product if the total number of tests was reasonable.

On this basis a physical testing approach was taken. One mid-size connector in the standard family range was developed through selective alteration of components,

through engineering analysis, and extensive physical testing to meet the performance and cost saving objectives.

The knowledge gained during development of this one size was extrapolated to the rest of the standard range. Seven out of ten of the connector range met the objective performance criteria with an average product cost saving of 23%. The other three (largest) couplers were shown to be beyond the product configuration in terms of both performance and economics.

The seven sizes redeveloped are the only sizes used in Europe so the bulk of the Ancon connector production is now less expensive and able to meet any impending European legislation.

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# CHAPTER 1 INTRODUCTION

## 1.1 Origins of Project

Ancon Clark Ltd is part of the Newmond Group, manufacturing steel fixings for the construction industry. The company was formed in 1992 after a merger between Ancon Stainless Steel Fixings Ltd and George Clark (Sheffield) Ltd. The company employs around 250 people across three sites; two in Sheffield, South Yorkshire, and one in Flint, North Wales and has an annual turnover of £17.5M (1995). The product range includes masonry support systems, windposts, parapet posts, lintels, wall ties, shear load connectors and channel/bolt fixings manufactured mainly in stainless steel but also in carbon steel. With this range of products Ancon Clark has around a 40% U.K. market share.

The company also manufacture connectors for concrete reinforcement bars, which are mechanical devices used for end-on joining of two bars. Reinforcement bars are cast into concrete structures to sustain tensile loading, and the connectors are generally used when other types of joint are either impractical or not permitted. Such products would tend to be classified as specialist, high value items.

Prior to the merger described above, in 1988, Ancon Clark was approached by a company called Metal Bond Technology, who had designed a reinforcement bar coupler and were looking to sell the manufacturing rights to the new product. A deal was agreed, whereby Ancon Clark assumed intellectual property ownership for the coupler.

Ancon Clark have since developed the coupler and several variants to suit all standard sizes of reinforcement bar from 8mm to 50mm diameter and by 1995 had captured approximately 10% of the U.K. market and 1% of the world market for all types of couplers, when product turnover approached £1M per annum. European Union countries account for 90% of the Ancon Clark coupler business, the remainder being in the US and the Far East.

At the outset of the project the high initial cost of the product meant that market expansion was limited and an imminent all-encompassing European standard was also likely to exceed the current product performance capacity. The project was proposed with the objective being to gain a clear understanding of the behaviour of the coupler under load and thereby enable an optimised device to be produced. This will lead to increased performance and reduced cost and opportunities to increase market share. The investigation was to be undertaken against a performance specification which anticipated the new all-European standard.

## 1.2 Historical Perspective

### 1.2.1 Concrete as a Building Material

Concrete is an artificial stone made from gravel or broken rock, sand, cement and water, the word itself being derived from the Latin 'concretus', meaning grown together or compounded. The materials are mixed together until a dense, uniform, and plastic mix is obtained. The mixture is then placed in a mould and allowed to set. The resultant material resembles natural stone in many of its properties. It is hard and brittle, strong in compression and weak in tension. Unlike natural stone, however, it can be produced in any shape without having to resort to the use of cutting tools and as the whole structure can be cast in one piece the need for (and weakness of) joints is eliminated.

Even though the use of concrete eliminates joints, its poor tensile properties are a limit to its sole structural use. Concrete members can be designed to be loaded in compression only, e.g. arches, or they can be cast with steel 'reinforcement' bars within them. The steel primarily serves to provide the tensile strength of the composite (concrete/steel) member.

The publication by Stanley (1) and the opening chapter of Cowan's text (2) give an excellent overview of the history of concrete and its use in construction. Examples of early concrete use has been dated as far back as 5600 BC. Excavations of Stone Age settlements on the river Danube, at Lepenski Vir, revealed 250 mm thick concrete hut floors composed of a mixture of red-lime, sand and gravel, to which water was added.

The earliest known illustrated use of concrete is depicted in a mural from Thebes in Egypt dating from about 1950 BC. It shows various stages in the manufacture and use of mortar and concrete. At this stage concrete was just used as an in-fill material for stone walls.

The art of making concrete eventually spread around the eastern Mediterranean and by 500 BC was being used in ancient Greece. It is thought that the Romans may have copied and developed the idea of making concrete from the ancient Greeks. Roman use and development of concrete eventually led to it being used as a structural material in its own right. Examples of early Roman concrete have been found dating back to 300 BC.

Some time during the second century BC the Romans made a major discovery that would revolutionise their use of the material. They started to quarry what they thought was sand from a source near Pozzuoli. It was actually a fine volcanic ash containing silica and alumina. When this was mixed with lime in the usual manner to produce concrete the result was a much stronger concrete than anything the Romans had previously been able to produce. The silica and alumina combined chemically with the lime to produce what became known as 'pozzolanic' cement. One of the first large scale uses of this material was in the amphitheatre at Pompeii constructed in 75 BC.

The Romans attempted to reinforce some of their concrete structures with bronze strips and rods. A good example of this exists in the roof of the Baths of Caracalla, Rome. Some improvement in the tensile strength of the concrete was achieved, however because the bronze has a higher rate of expansion and contraction than concrete composite members were prone to cracking.

This limitation led to the Romans designing their buildings to carry loads in compression which resulted in structures with massive thickness. As a result the Romans made developments in lightweight concrete. Initially this involved casting voids into arches and walls. Then lightweight aggregates were utilised such as crushed pumice, which is a porous volcanic rock. Lightweight concrete was used in arches of the Colosseum and also in the dome of the Pantheon. This 50 metre domed structure has survived intact to this day!.

The Romans brought their knowledge of concrete with them to Britain. They didn't transport their pozzolanic cement all the way to Britain however but made use of local materials. Typically lime concrete was used which was adequate for use as in-fill material in walls and floors. Hadrian's wall has a concrete core.

Over a period of 800 years the Romans developed concrete from a crude filling material to the position of being one of the main structural materials. Unfortunately most of the knowledge gained in the use of concrete disappeared almost completely with the decline of the Roman Empire.

The use of concrete in Britain in the middle ages was very limited and isolated. A number of Saxon concrete mixers have been found dating to around 700 AD. The Normans, however, were more liberal in their use of concrete. The Norman concrete work was not unlike that of the early Roman period used as in-fill material in walls. Their invasion of Britain led to a more widespread use of this concrete. An interesting example can be seen at Reading abbey where the stone facing has almost completely fallen away leaving what is in effect a concrete skeleton. Concrete was widely used in castles, including the White Tower in the Tower of London, Dover, Corfe and Rochester. In churches and cathedrals concrete was used principally for foundation work.

The Medieval and Renaissance periods saw very little use of concrete. Interest was revived in the middle of the 18th century. In 1756 Leeds engineer John Smeaton was commissioned to build the third Eddystone rocks lighthouse near Plymouth on the

English Channel. The two previous lighthouses had been timber structures. One burned down and the other was blown away during a fierce gale. Smeaton knew that a stone block structure was the only practical solution. However he also knew that no mortar existed to bind the blocks that could set in wet conditions. His investigations led to a mixture of Welsh limestone and Italian pozzolana which had excellent hardening properties when used under water. He had produced the first good-quality cement since the downfall of the Roman Empire.

Towards the end of the of the 18th century there was a considerable revival of interest in developing new types of cement, with many types of formulations which in essence were little better than Smeaton's attempts. It wasn't until 1824 that a significant advance was made. Joseph Aspdin, a Leeds bricklayer, took out a patent for the world's first Portland cement, so called, incidentally, because when it set Aspdin thought it resembled Portland stone in colour, and not, as people often think, because it was made in Portland. He produced it by heating fine powdered clay and limestone with water in a kiln. The cement had superior setting and strength qualities to anything else available at that time. When the roof of Isambard Kingdom Brunel's Thames tunnel collapsed he sealed the break by dumping tonnes of Portland cement into the river!.

One of the big drawbacks of Portland cement at the time was its cost - roughly ten times the relative cost of cement today. This was due to the high cost of its manufacture. It was after the invention of the rotary cement kiln in 1880 that the cost of cement reduced to reasonable levels.

### 1.2.2 Reinforcement in Concrete

The man generally credited with the invention of reinforced concrete is a little known Newcastle builder, William Wilkinson. This was recorded in his patent, first applied for in 1854, for 'Improvement in the construction of fireproof dwellings, warehouses, other buildings and parts of the same'. The patent specification states that a number of strips of hoop iron are to be laid on edge and embedded in mass concrete at distances of about 2ft. This system was designed for use in curved ceilings but more interestingly is the section dealing with flat ceilings. The patent suggests the use of second hand wire colliery ropes. The ropes were to be embedded in fresh concrete and the ends formed into loops or splayed, by opening out the strands, so that the ropes could not be pulled out when the concrete was loaded. The drawings which accompany the specification clearly show that Wilkinson understood the basic structural principals. A number of small buildings in the Newcastle area were erected in this manner. However, the main development of reinforced concrete construction in England took place after the registration of two French patents, by F. Hennebique in 1897 and by E. Coignet in 1904. By the end of the 19th century there were 43 different patent systems in use. Some employed complex arrangements of reinforcement, in others the arrangement of the reinforcement was similar to that in use today.

The first types of reinforcement to be commonly used were plain round steel bars. The Americans made extensive use of reinforced concrete in the early part of this century and at the same time developed the use of bars with protruding ribs on their outer edge. The ribs greatly improve the bar-to-concrete bond characteristics. Essentially the design of ribbed bars has not changed since its first use, however, as the rib pattern is not standard, multiple bar manufacturers the world over have differing rib geometry.

The applications of ribbed reinforcing bars is predominantly in large flat (or curved) concrete sections, columns and beams. Examples of large concrete sections are bridge decks, floors, parapets, reactor vessels and entire building external walls. Concrete columns and beams can be seen all around us.

## 1.3 Methods of joining reinforcement bars

### 1.3.1 General Background

Lancelot (3) describes the various methods of joining reinforcement. Manufacturing, fabrication, and transportation limitations make it impossible to provide full length continuous bars in most reinforced concrete structures. Therefore proper joining (or splicing) of reinforcement bars becomes essential to the integrity of reinforced concrete. He describes three main methods of splicing the bars :

- □ lapped bars - where the two bars to be joined are overlapped by a predetermined length with no fastening

- □ welded bars - the two bars to be joined are either welded end to end or a lap is welded along its length

- □ mechanically connected bars - a mechanical connector joins the two bars together by some means

Of the three, lap splicing is the most common. However, building codes frequently require such long laps that steel becomes congested at the splice location. Sometimes the lack of room at a joint makes a lap truly impossible. Location of construction joints, provision for future construction, or a particular method of construction can also make lap splices impractical.

Welded joints can be one of four types all of which are described in Gustafson's paper (21) . These are indirect and direct butt welds and indirect and direct lap welds. Direct butt welds involve welding the bars end to end by either bevel groove welds or fusion welding (where a mould surrounding the joint is filled with molten metal). An indirect butt weld involves welding the two ends to a common member such as a plate. A direct lap weld simply welds the two bars in contact along the lap length. In an indirect lap weld the bars are welded to a common plate. All welds require the use of heavy equipment on site.

When both lapped and welded joints are impossible or impractical a mechanical splice will be chosen. Lancelot (3), Harding (4) and an American standard document (5) all describe the various types of mechanical splices available. Most modern mechanical splices align and secure the joined reinforcement bars through an in line connection. All types of compression and tension couplers rely on mechanical interlock to achieve this using of some sort of 'sleeve' into which the two bar ends are inserted. The splices transfer tension or compression loads from one piece of reinforcement to another. Many splice systems which are designed for tension capability also satisfy compression splice requirements, but the converse is not true.

The most popular methods or devices are :

- metal-filled sleeves
- mortar or grout-filled sleeves
- swaging or forging
- threading
- friction and clamping

### 1.3.2 Compression only devices

Compression only devices generally consist of some sort of friction/clamping device and are used for connecting bars in columns that will only experience compressive loads (i.e. in non-seismic regions). The connector only has to have sufficient strength to ensure that the bars remain aligned.

### 1.3.3 Tension Devices

The first widely accepted commercial mechanical splice system was a metal-filled sleeve. The two bars were placed end to end into the sleeve which has internal ribs. The two sleeve ends are sealed and molten metal is then poured in via one of two tap holes. The metal flows between the bars and the sleeve, solidifying in the deformations of the ribbed bar and the internal ribs of the sleeve, forming a mechanical interlock. Displaced air escapes via the second tap hole and when metal rises in the second hole the sleeve is filled.

Bar ends must only be clean and dry. However, the interior of the splice cannot be inspected to ensure full mechanical interlock and there is a need for fire protection and protective clothing because of the heat given off during melting.

Grout or mortar filled sleeves work in much the same way as metal filled sleeves both by operation and installation. They are however much longer due to lower tensile properties of the filler material. This means they tend to be bulky. There is also a lack of heat resistance and fillers can take between 2 and 4 hours to set.

Splices based on hot or cold metal forming are also available and were developed following the filler type splices.They use interlocking mechanisms with hot and cold metal forming techniques which create interlock of the sleeve with bar deformations by applying external pressure to the sleeve. This forces the walls of the sleeve to collapse and conform to the bar deformations.

Hot forging was the earliest of the metal forming techniques. A furnace and a fuel source is required near to the immediate work area. The sleeves are heated in the furnace then placed over the two bar ends. The hot sleeve is forged into the deformations of both bars by a hydraulic ram. Contraction of the sleeve upon cooling improves bond and increases the splice strength.

Cold swaging and extrusion use a seamless sleeve placed over abutting ends of the two bars and a hydraulically powered extrusion press shapes the sleeve to the bars.

Threaded couplers are also available using parallel or tapered threads.

Parallel thread systems require the bar ends to be threaded which can be done by the bar manufacturer or on-site. The bars only have to be hand tightened with a strap/chain wrench. All threads must be protected against damage during shipping and site handling. Screwing in the larger bars is heavy work.

Tapered thread bars and couplers make alignment easier and assembly quicker i.e. fewer turns. Bars must be tightened to a predetermined torque. Originally both tapered and parallel threading of the bars reduced the nominal diameter and hence tensile strength. Now the bar ends are upset to enable a larger thread to be used.

Another type of threaded splice is available for reinforcing bars which have specially rolled thread-like deformations over their entire length. The coupler sleeves have internal threads to match the bar thread. The thread configuration is very coarse and two lock nuts are needed to eliminate the slack. The lock nuts must be tightened to a high torque.

### 1.3.4 Continuity Devices

All of the coupler types discussed in the previous sections are used for joining lengths of reinforcement that will be cast into concrete to produce whole members. Frequently steel must be continued across construction joints at a date later than the first cast. A typical application would be access hatches. This means that reinforcement bars must project out of the concrete to allow continuation/completion of structure at a later date which is hazardous to personnel. Specialised 'continuity' couplers have been developed to eliminate this hazard and at the same time reduce the amount of final formwork. Generally they consist of a male and a female half. The female component is cast against the formwork in the original cast with a plate against the outer surface to protect the mating parts. When continuation of the structure is required, the cover plate is removed the male half is connected to the female. A number of these continuity couplers exist. The mating part is always a threaded connection. The male and female parts are available in a number of the previously mentioned tension splice variants e.g. swaged and threaded couplers

## 1.4 Design Codes

There are a large number of design codes from numerous countries and governing bodies which specify various performance criteria for these reinforcement connectors. Specific code characteristics vary from country to country and from body to body but generally there are three main criteria to be met.

These are :

1. A minimum amount of permanent displacement or 'slip' after application of a specified load for each size of bar. This is intended to represent a maximum crack size for concrete.

2. A minimum failure load associated with each size of bar.

3. A minimum fatigue life or endurance under specified cyclic loading conditions.

Relevant codes include;

**United Kingdom** - BS8110 (6) and BS5400 (7), BBA(British Board of Agrément) Code (10)

**USA** - ACI (American Concrete Institute)318 (8) and 359 (15)

**Germany** - DIN 1045 (9)

**France** - NFA 03-162(11)

**Norway** - NS 3420 Norwegian Offshore Code (12)

**Sweden** - BBK 94 Volume 2 (13)

**Canada** - CSA N287.3 (14).

In addition, certain large scale construction projects have even created their own particular design code such as the Hong Kong Mass Transit and Rail Corporation project.

BS8110 requires that a reinforcement bar coupler must not have a permanent extension (slip) of more than 0.1mm after loading to 0.6Fy, where Fy is the yield stress of the bar and is currently 460N/mm². The coupler must also achieve a monotonic load of at least 1.1Fy before failing. All fatigue tests have a stress ratio of 0.2, which is the ratio between the smallest and largest stress cycles and are done at stress ranges of 140N/mm², 160N/mm² and 200N/mm² with the minimum number of cycles that must be sustained being specified as 3.5M, 1M and 0.3M respectively (7).

Other European codes are more stringent, demanding slip of less than 0.1mm at 0.7Fy where the yield stress of the bar is 500N/mm² (11). The static failure criterion is also more stringent at 1.25Fy (11). Endurance requirements are similar.

The ACI has a number of codes but none of them have a slip requirement. They do however stipulate a static failure criterion of at least 1.25Fy (15) for nuclear construction applications.

# CHAPTER 2 ANCON MBT COUPLERS

## 2.1 Coupler Design History

The Ancon MBT reinforcement coupler was designed originally by inventor Paul Hope, who filed a patent in 1983 (16) to protect his design. He had connections both in the construction and chemical industry and when ICI developed a new type of resin he immediately identified a market for its use as a filler for a resin reinforcement bar coupler. In the patent it states that the coupler consists of a sleeve, inside of which are two axial locating 'ribs'. Opposing the ribs are a number of bolts. When the two bars are fed into the sleeve the bolts are tightened forcing the bars against the locating ribs. Then the sleeve ends are sealed with putty and a resin is pumped into the sleeve via one of two tap holes. The stated resin is Seltite-Selfix®.

Tests performed upon this original design showed that the resin-bonded device alone could not meet the required slip performance. The coupler was redesigned so that the longitudinal locating ribs and the locating bolts contributed to the coupler stiffness as well as the resin. This was done by machining serrations, or teeth, onto the ribs and by machining a cone onto the end of the bolts. When the two bars are inserted the bolts are tightened so that the cone ends and the rib serrations penetrate the bar. The bolts are tightened to a pescribed torque value, achieved through the use of shear bands machined into the bolts just below the head. Both the bolt cones and the rib serrations are hardened. Then the resin is inserted as described previously.

It was discovered by testing at that time, that the bolts and serrated ribs, or saddles, could provide enough stiffness and strength to meet the performance without the need for resin. This revised design was patented in 1989 (17) and has remained in production since then with a size range of ten couplers to suit ten different sizes of bar. Figure 2.1 shows a standard coupler from the present range, the ET20. The cutaway in Figure 2.2 shows how the bolts penetrate the bar and also shows the location and operation of the saddles.

**Figure 2.1 - Present ET(European Type) 20 Coupler.**

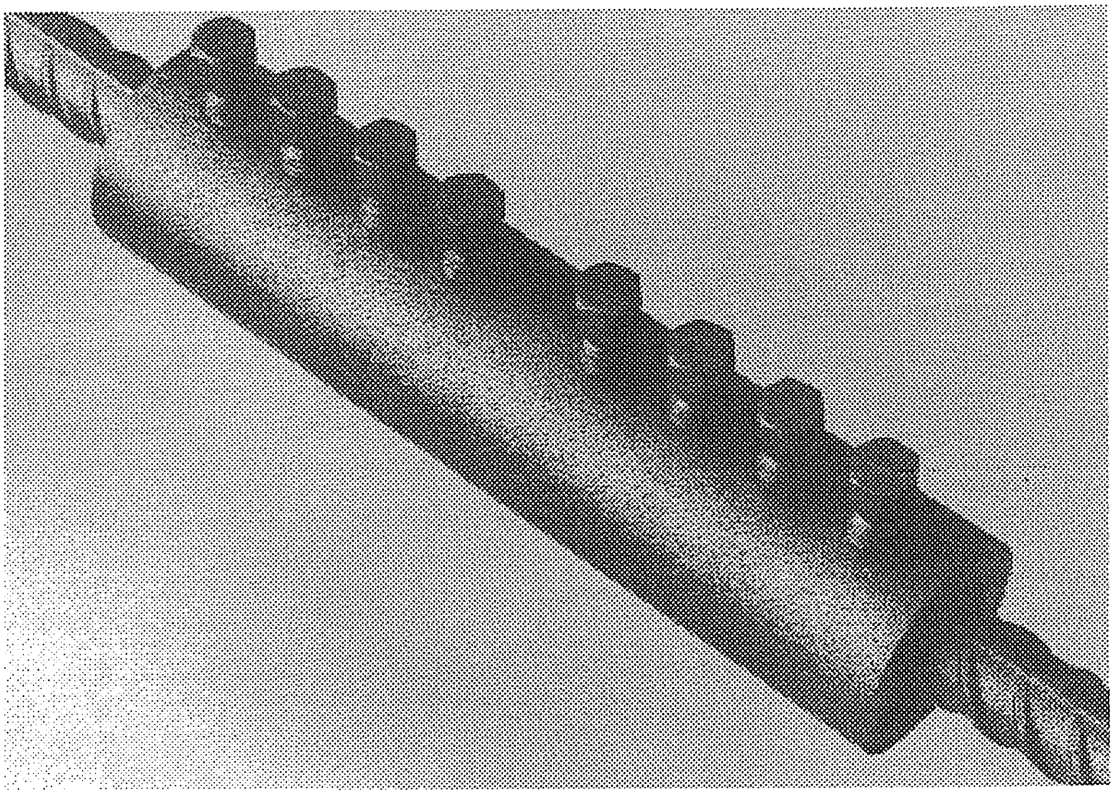
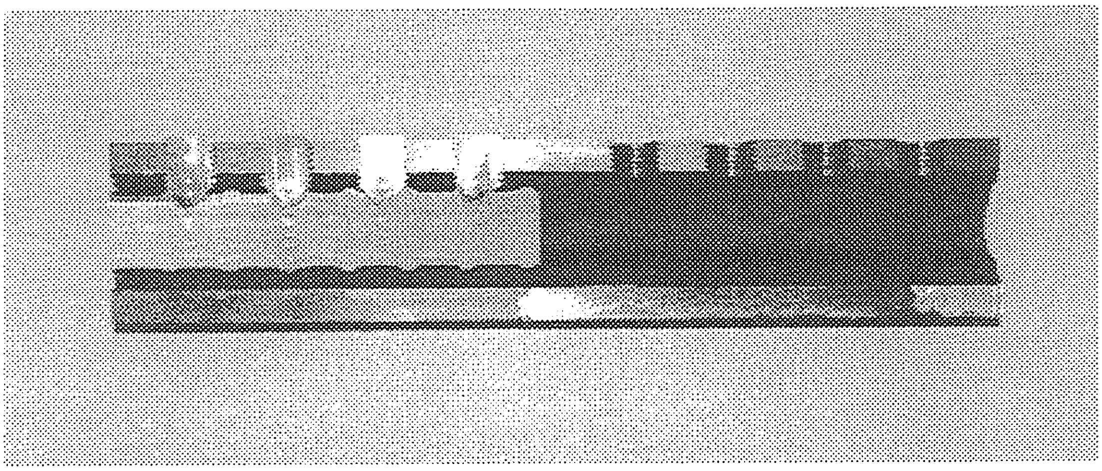


**Figure 2.2 Cutaway View Of ET 20 Coupler**



A continuity variant of the coupler was designed shortly after this and patented in 1993 (18). Other variants are the transition coupler, for connecting two different sized bars, and the compression-only variant, which is generally shorter than the standard version.

Figures 2.3 and 2.4 show continuity and transition type couplers.

**Figure 2.3 Ancon Clark Continuity Coupler**

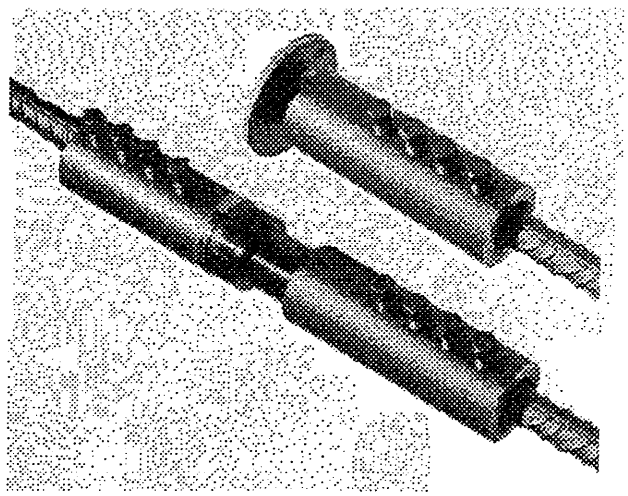
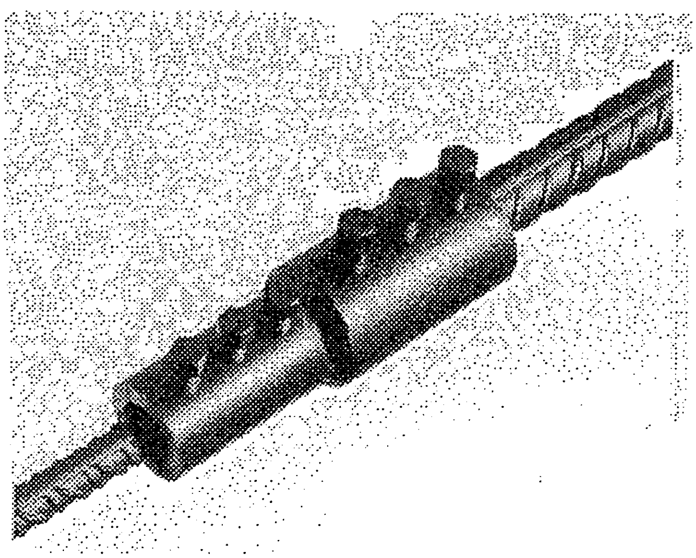


**Figure 2.4 Ancon Clark Transition Coupler**

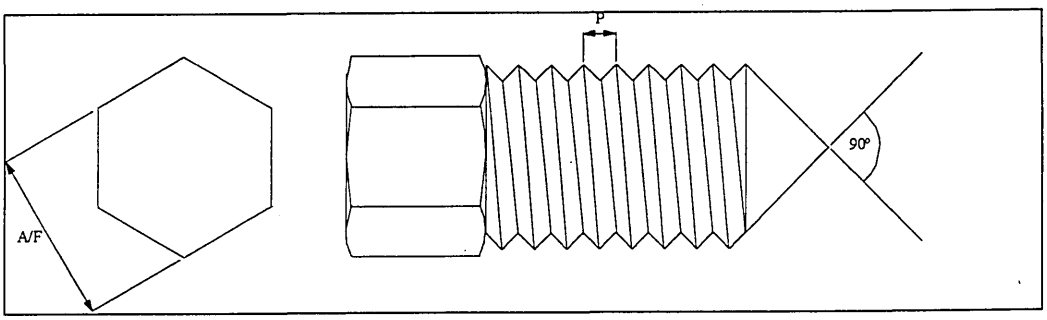


Ancon MBT couplers account for approximately 1% of world coupler sales. The couplers are much more expensive than their competitors, however their installation costs are much smaller. Table 2.1 shows the volume and relative value of sales by coupler size for the year 1995, which was used as the basis for selecting the ET20 as the first development size. All costs are relative to this coupler. The whole production range dimensions are given in Tables 2.2, 2.3 and 2.4.

**Table 2.1 ET Coupler Sales Volume and Relative Value**

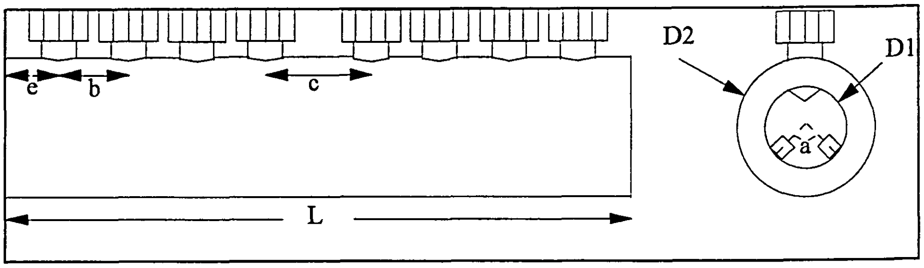
| Coupler Type | 1995 Sales Volume | 1995 Rel Sales Value |
|---|---|---|
| ET8 | 9054 | 4010 |
| ET10 | 3450 | 1568 |
| ET12 | 7325 | 4642 |
| ET16 | 15050 | 11655 |
| ET20 | 27000 | 27000 |
| ET25 | 12700 | 17010 |
| ET28 | 928 | 1685 |
| ET32 | 14050 | 28583 |
| ET40 | 6100 | 15834 |
| ET50 | 456 | 2202 |
| **totals** | 96113 | 114189 |

**Table 2.2 EuropeanType Coupler Bolt Dimensions**




| Coupler type | Number of bolts | Bolt type | Shear torque (Nm) | Pitch p (mm) | Drive A/F (") |
|---|---|---|---|---|---|
| ET8 | 4 | M10 | 55 | 1.50 | 1/2 |
| ET10 | 4 | M10 | 55 | 1.50 | 1/2 |
| ET12 | 6 | M10 | 55 | 1.50 | 1/2 |
| ET16 | 6 | M12 | 108 | 1.75 | 1/2 |
| ET20 | 8 | M12 | 108 | 1.75 | 1/2 |
| ET25 | 8 | M16B | 200 | 2.00 | 5/8 |
| ET28 | 10 | M16A | 360 | 2.00 | 5/8 |
| ET32 | 10 | M16A | 360 | 2.00 | 5/8 |
| ET40 | 10 | M20B | 450 | 2.50 | 3/4 |
| ET50 | 14 | M20A | 600 | 2.50 | 3/4 |

**Table 2.3 - European Type Coupler Sleeve Dimensions**




| Coupler type | End to first hole, e (mm) | Between holes, b (mm) | Between centre holes, c (mm) | Length L (mm) |
|---|---|---|---|---|
| ET8 | 15.00 | 20.00 | 30.00 | 100.00 |
| ET10 | 15.00 | 20.00 | 30.00 | 100.00 |
| ET12 | 15.00 | 20.00 | 30.00 | 140.00 |
| ET16 | 18.00 | 22.00 | 36.00 | 160.00 |
| ET20 | 18.00 | 22.00 | 36.00 | 204.00 |
| ET25 | 24.00 | 27.00 | 48.00 | 258.00 |
| ET28 | 24.00 | 27.00 | 48.00 | 312.00 |
| ET32 | 24.00 | 27.00 | 48.00 | 312.00 |
| ET40 | 25.00 | 32.00 | 50.00 | 356.00 |
| ET50 | 25.00 | 32.00 | 64.00 | 498.00 |

| Coupler type | Tube outer diameter, D2 (mm) | Tube inner diameter, D1 (mm) | Saddle position angle, a (Degrees) |
|---|---|---|---|
| ET8 | 33.40 | 20.70 | 84.00 |
| ET10 | 33.40 | 20.70 | 84.00 |
| ET12 | 33.40 | 20.70 | 84.00 |
| ET16 | 42.20 | 26.40 | 90.00 |
| ET20 | 48.30 | 31.30 | 90.00 |
| ET25 | 54.00 | 35.00 | 90.00 |
| ET28 | 66.70 | 41.70 | 90.00 |
| ET32 | 71.00 | 44.60 | 90.00 |
| ET40 | 81.00 | 56.00 | 90.00 |
| ET50 | 101.60 | 69.60 | 90.00 |

**Material specification**

Saddle material — Alloy steel 709M40, chemical composition to BS970 : Part 1 : 1991 - Table 4.

| | |
|---|---|
| Tube material | Desford T.I.6V and/or Hollomek 6V high strength seamless tube, with a minimum yield strength of 480 MPa and a minimum UTS of 600 MPa. |
| Bolt material | Alloy steel 606M36 (low sulphur content not exceeding 0.2%), chemical composition to BS970 : Part 3 : 1991 - Table 16 and Table 21 |

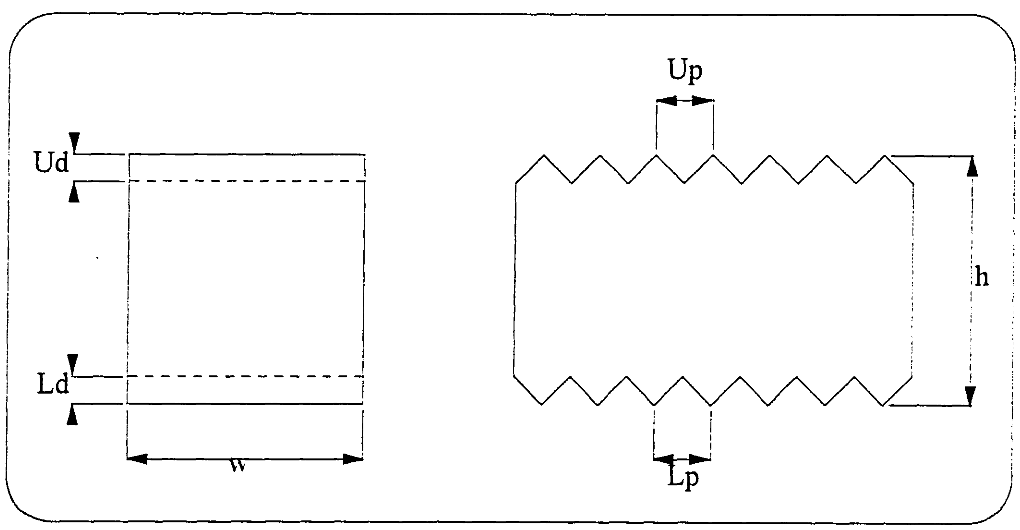
## 2.2 Coupler Manufacture

Everything except bolt manufacture and bolt/saddle heat treatment is performed in-house by Ancon Clark. The seamless tube is supplied in 3m lengths and cut to sleeve size using an auto saw.

The sleeves are then drilled and tapped in a CNC Fanuc Robodrill. An identification number is then stamped onto the sleeve end with a semi-automatic stamping machine.

The saddle material is supplied to cross-sectional size in 12ft lengths and cropped to length. The serrations are milled into the material with horizontal slab cutters one side at a time. A number of saddles are milled concurrently, clamped upon a flat bed depending upon saddle width.

**Table 2.4 Present European Type Saddle Dimensions**



| Coupler type | Saddle width, w (mm) | Saddle height, h (mm) | Saddle upper teeth pitch, Up (mm) | Saddle lower teeth pitch, Lp (mm) | Upper tooth depth, Ud (mm) | Lower tooth depth, Ld (mm) |
|---|---|---|---|---|---|---|
| ET8 | 6.00 | 6.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET10 | 5.00 | 5.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET12 | 4.00 | 4.00 | 3.00 | 1.50 | 1.50 | 0.75 |
| ET16 | 5.00 | 5.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET20 | 6.00 | 6.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET25 | 6.00 | 5.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET28 | 6.00 | 6.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET32 | 8.00 | 7.00 | 3.00 | 3.00 | 1.50 | 1.50 |
| ET40 | 10.00 | 9.00 | 4.00 | 4.00 | 2.00 | 2.00 |
| ET50 | 12.00 | 10.00 | 4.00 | 4.00 | 2.00 | 2.00 |

The saddles are hardened to 56 Rc by heating in an induction coil then cooling rapidly in brine.

Bolt blanks are machined directly from hexagonal bar of the same size as the bolt head using semi-automatic multi-spindle machines. The threads are then rolled onto the blanks one at a time manually.

Each bolt is through hardened to 32-39 Rc by controlled furnace heating and slow cooling. The bolt cones are then hardened to 56 Rc by heating each cone in a coil and cooling it rapidly.

Heat treated saddles are fixed into the sleeves by tack welds at each saddle end. This is done manually with jigs and MIG welding sets.Finally the finished bolts are assembled into the coupler body by hand.

### 2.3 Cost Analysis of Present Range

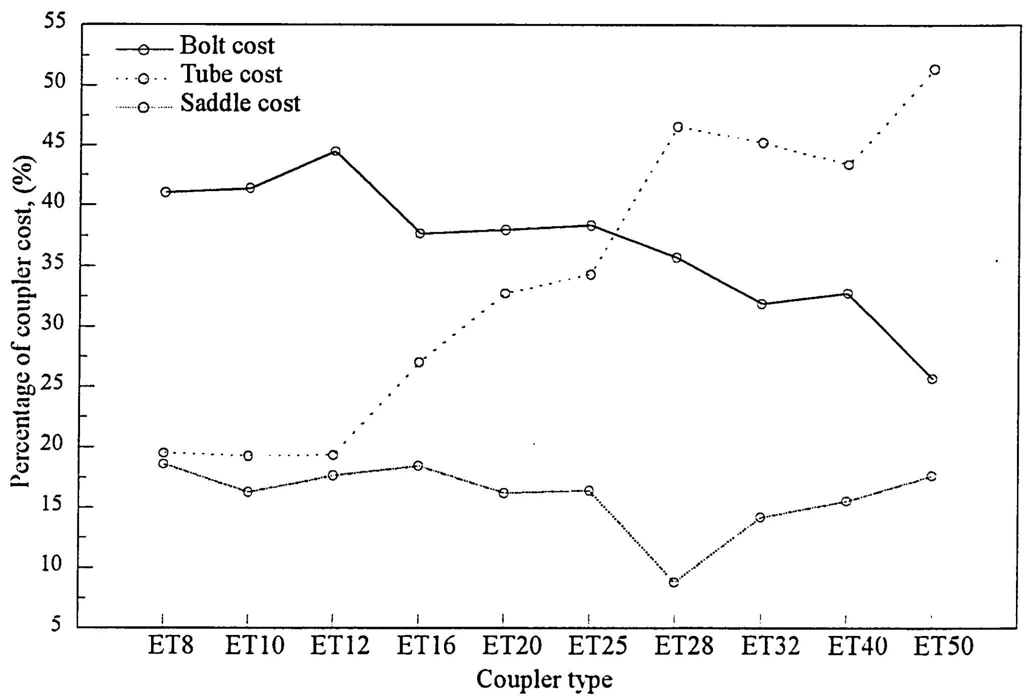
The sensitive commercial nature of this project does not permit the use of actual costs. As a result the cost analysis will be illustrated using arbitrary units, where one unit is the 1996 cost of an ET20 coupler. The relative costs are given in Table 2.5.

**Table 2.5 - Cost Ratios for ET Coupler Range**

| Coupler type | Cost (Units) |
|---|---|
| ET8 | 0.447 |
| ET10 | 0.452 |
| ET12 | 0.631 |
| ET16 | 0.767 |
| ET20 | 1.000 |
| ET25 | 1.334 |
| ET28 | 1.808 |
| ET32 | 2.025 |
| ET40 | 2.603 |
| ET50 | 4.811 |

The first stage in the project was to complete a detailed cost analysis of the whole product range. Fortunately the company had the information available and the whole range cost distribution is shown in graph 2.1.

**Graph 2.1 ET Coupler Cost Distribution**



The cost distribution varies for each coupler as the size increases. For the smaller couplers, the bolts contribute more than 40% of the total cost, but for larger sizes this falls to around 30%. Also, larger couplers have up to 50% of their cost in the tube , which falls to 20% for the smaller sizes. Saddle costs are relatively constant as a percentage throughout the range, apart from the ET28 which is dramatically low. This would be expected since the saddle for this size is very similar to the ones for the ET20 and 25 sizes.

To achieve meaningful cost savings it was initially thought sensible to reduce the tube cost in the larger couplers and reduce the bolt cost in the smaller couplers. Also any reduction in saddle dimensions and heat treatment costs was desirable. The remaining fractions of cost not included in the above analysis were not thought likely to be reduced independantly.

The ET20 coupler, which was the subject of all initial development, is a mid range coupler and did therefore require a reduction in number of bolts as much as a reduction in tube dimensions. A complete cost breakdown of the present ET20 is in Table 2.6, includng all material, manufacture and labour items. Similar tables are included in appendix E for whole range.

## 2.4 Project Aims and Objectives

The aim of this project was to develop the existing range of Ancon MBT couplers. The objective was to reduce product cost and to improve the product performance. Reducing the product cost would make the couplers more competitive and as a consequence capture more of the world market. Since the manufacturing system being used in the company was considered to be very efficient, the only way to reduce costs was by the redesign of some or all of the components to give reduced cost of material and bought out parts. In order to do this it was necessary to gain an understanding of the mechanism of operation of the coupler and the contribution of each component to the overall performance of the coupler.

**Table 2.6 Cost ratio analysis for present ET20 coupler**

| | | | | | |
|---|---|---|---|---|---|
| **TUBE COST** | | | **STAMP** | 500 | per HR |
| COST | 0.311 | | STAMP COST | 0.000 | |
| SCRAP | 0.016 | | SET-UP | 0.001 | |
| **TOTAL** | **0.327** | **Units** | **TOTAL** | **0.001** | **Units** |
| **BOLTS** | | | **DRILL & TAP** | 33 | per HR |
| 8 OFF M12 | 0.368 | | LABOUR COST | 0.042 | |
| TRANSPORT | 0.012 | | TOOL COST | 0.018 | |
| **TOTAL** | **0.380** | **Units** | **TOTAL** | **0.060** | **Units** |
| **SADDLE COST** | | | **WELD** | 40 | per HR |
| MATERIAL | 0.042 | | LABOUR COST | 0.035 | |
| MANFACTURING | 0.058 | | CONSUMABLES | 0.003 | |
| HEAT TREAMEN | 0.053 | | | | |
| TRANSPORT | 0.009 | | **TOTAL** | **0.038** | **Units** |
| **TOTAL** | **0.162** | **Units** | **ASSEMBLY & PACK** | 70 | per HR |
| **SAW** | 100 | per HR | ASSEMBLY LABOUR | 0.020 | **Units** |
| | | | PACK LABOUR | 0.002 | |
| CUT COST | | | | | |
| LOAD COST | 0.001 | | PACK COST | 0.009 | |
| BLADE COST | 0.001 | | | | |
| | | | **TOTAL** | **0.031** | **Units** |
| **TOTAL** | **0.002** | **Units** | | | |
| | | | **TOTAL COST** | **1.000** | **Units** |

# CHAPTER 3 NEW COUPLER REQUIREMENTS

## 3.1 Specification

The first stage in any product development project is to produce a suitable Design Specification. The following coupler specification contains details of all product constraints in a series of individual sections. The constraints must be adhered to during the development/design stage and as a consequence, are fully accurate yet not too detailed so as to become restrictive.

### Mechanical Loading and Dynamics

The loading is applied to the coupler via the reinforcement bar. In most applications this is tensile however a range of couplers are available for compressive loading applications. Cyclic loading is also possible where the concrete structure experiences dynamic loading e.g. a typical example of this is in bridge decks where the load being carried varies.

### Performance

According to DD ENV 1992-1-1 : 1992 (19) each coupler must have a tensile failure equal to or greater than 1.25 $F_Y$ where $F_Y$ is the yield stress of the couplers respective re-bar and will be taken as 500 N/mm$^2$ for the present work.

Also NFA 03-162 (11) states when a load of 0.7 Fy is applied to each coupler the permanent slip must not exceed 0.1mm.

BS5400 : Part 10 (7) states that the couplers must have an operational life identical to that of reinforcement which is 120 years. (See Appendix D for classification and required life cycles).

Presently couplers ET8 - ET32 meet the U.K. design codes only (with a special, more expensive ET8 - ET28 range for use with 500 Grade bar in Germany). The ET40 and ET50 do not consistently meet the permanent slip requirement but do achieve upper tensile load (U.T.L.) limits. The European commission has proposed one standard (19) which will combine all of the member states design codes into one. Since all of the European standard codes are more stringent than the U.K. code it is likely that the final design code will be more stringent. U.K. couplers will have to meet this new design code. This code has not yet been published but, to pre-empt the final code it is assumed that a combination of the most stringent requirements will be adopted i.e. 0.1 mm of slip at 0.7Fy and a minimum strength requirement of 1.25Fy.

**Environment**

The couplers are designed to be encased in concrete therefore, prior to casting, a period of time exposed to the atmosphere must not affect their ultimate performance.

Couplers must perform equally well throughout their environmental exposure time.

The couplers must be able to tolerate a period of two weeks exposed to construction site conditions without becoming visually unsettling due to surface corrosion. Continuity couplers must remain visually unsettling for up to one year when directly exposed to the atmosphere.

**Weight and Dimensions**

The coupler range should accommodate all standard sizes of re-bar available and rib orientation and be light enough to enable single operator installation. The outer dimensions of the couplers should be as small as possible so as not to restrict structural designers or installation.

All components incorporated into the design will be of metric dimensions.

**Installation**

The couplers should be installed via simple mechanical means within a sweep angle of access no greater than 30°. Any installation should comply with required concrete coverage and coupler spacing (2). A method for checking or ensuring correct installation should be included.

**Appearance**

The coupler should not have any sharp edges or any exposed mechanisms which may cause operator injury and the coupler must appear visually capable of the function and be supplied in a non-corroded state to promote customer confidence.

**Costs**

The retail cost of the coupler must be reduced by at least 20% either by an improvement of the product itself or by improvement of the processes involved.

**Manufacture**

The coupler range should be manufactured in accordance with ISO 9001 (20) where possible using internal capacity and facilities.

**Test procedure**

The slip and U.T.L. values had to be measured using one test sample. The permanent slip is measured externally to the test machine using displacement transducers and the U.T.L. is recorded both externally and by the test machine.

The test specimens consist of a ribbed control bar and two pieces of reinforcement bar (of equal combined length to the control bar) joined together by an assembled coupler. The control bar is tested first.

The specimen is placed into the test machine and zero load applied. The slip test is performed first followed by loading to failure. The displacement transducers are attached to the specimen via an extensometer. The extensometer is fixed to the bar by two clamps, one each side of the coupler over the gauge length, although these clamps are not fixed to each other. The transducers are arranged to detect any movement between these two clamps. The transducer output is fed into a graph plotter which also records the load output. The specimen is loaded to the specified slip load then unloaded to zero. The graph plotter records the permanent extension.

The extensometer is then removed and the specimen is loaded to failure. The graph plotter displays the load curve against a fixed time scale and the test machine records the peak load.

All of this procedure is then repeated for the coupler test specimen. The control bar slip is subtracted from the coupler specimen slip so that the true coupler slip is recorded.

## 3.2 Development Approach

As a simple product with a complex structure, each component must be optimised and to do this an understanding of the function of each component is required. This was completed as a modified failure modes analysis as follows:

### Tube

The tube holds the bolts which penetrate the reinforcement bar and thereby resists the tensile forces which are transmitted either during testing or in use. The tube inside diameter needs to be large enough to receive the size of reinforcement bar to its maximum dimension and also to enable the bolt cone to have cleared the tube inside diameter once penetration is complete. The tube wall thickness should resist radial loads as the bolt is screwed in and tensile loads transmitted by the bolts under test or in use. The threads in the tube wall may also be vulnerable to shear depending upon the tube wall thickness.

### Bolt

The bolt is obviously a key component and can vary in size or in quantity, the combination of which is crucial to cost. The bolts need to be large enough to enable good penetration into the reinforcement bar and also resist both shear and crushing failure under test or in use. Excessive increase in bolt size may reduce bolt quantities.

### Saddle

This component ties the reinforcement bar to the tube under the load of the bolts and resists tensile loads from tests or in use. The cross-section is necessary for the resistance of the loads. The numbers and size of serrations will affect cost. Large numbers of small serrations will have reduced individual direct load, but small root dimensions which may not resist tensile loads. It is difficult to establish the contribution of this component in comparison to the bolts where the penetration into the reinforcement bar will require a ploughing mechanism to induce failure in the bar.

Thus the approach to the project needed to accomodate all of these factors. Two basic approaches were possible; computer modelling or physical testing of a number of alternative configurations.

The book by Papalambros and Wilde (26) describes the principles behind computer modelling. Computer modelling typically uses finite element or numerical method software packages to create a computer model of the product or component. The model is adjusted until the product performance appears satisfactory and matches actual physical test data. Principally the model helps the engineer to better understand the product mechanisms making the task of development easier and reducing the amount of physical testing. In this example modelling screw threads and penetration are difficult to do and are seen as perhaps more costly and time consuming than testing of a relatively inexpensive product if the total number of tests is reasonable.

Factorial testing involves the changing of various product components, materials or forces testing the new design and comparing the results. If the number of elements that are altered are not chosen at random but as a result of engineering analysis then the number of tests can be reduced.

Statistical methods ascribed to Taguchi, described in his book Introduction to Quality Engineering (25), are used to discover how the product works by producing a number of designed experiments which take into account all of the variables present. The approach utilises a minimum number of physical tests. However, the approach is orientated towards one objective outcome only and would not be suitable for this case.

Hence it was decided to use a totally experimental approach for the ET20 varying physical sizes and component numbers using a parametric approach to identify key parameters which can then be applied to other sizes in the range.

# CHAPTER 4 BOLT TESTS

## 4.1 Introduction

It was felt useful at the outset of the project to undertake tests on the fastening bolts to establish some basic data about their load-torque relationships, maximum torque and compression loadings, particularly in relationship to their use within the tubes. The first job was to determine a relationship between the applied bolt torque, the depth of bolt cone penetration into a reinforcement bar and the bolt downward force. Bolt penetration into the reinforcement would obviously affect the load carrying capacity of the coupler. A number of load cells were available to measure the downward force. A rig which would combine a section of bar and several sizes of bolt was designed and manufactured.

## 4.2 Bolt Penetration Rig Specification

### Mechanical loading and dynamics

The penetration block needs to contain all of the bolt induced force within its assembly.

The weight of the block and any applied torque is transferred to the holding workbench vice.

The loading is gradual.

### Performance

All of the bolt downward force must be transmitted to the load cell. The maximum load cell capacity is 100 kN.

The maximum applied torque possible from an M16 shear bolt is 200 Nm.

### Environment

The bolt block will not be exposed to moisture or temperature extremes.

The testing will be performed in air.

**Weights and Dimensions**

The re-bar sample must have the same distance from its perimeter to the bolt hole as in the coupler. This gap is approximately 4 mm.

The block must accommodate the load cell which is 40 mm in diameter and 13 mm high.

Support for a 100 mm length of re-bar is adequate.

**Installation**

The block must be designed to fit into any workshop vice.

**Manufacture**

Only a single bolt penetration block is to be manufactured so production of the design must be possible using standard machine shop technology.

Standard size fasteners and materials to be used.

Manufacture must conform to ISO EN9001.

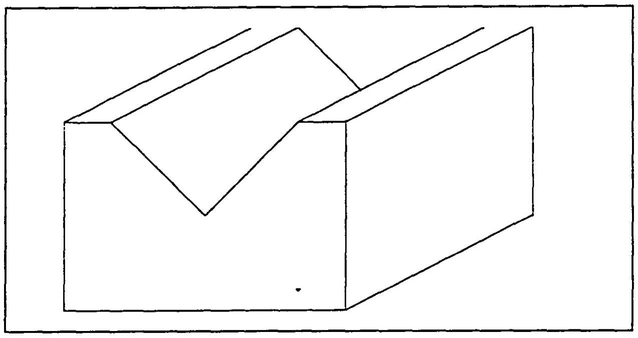
## 4.3 Conceptual Design of Bolt Penetration Rig

It is most important in this design that any constituent parts and any joints between the parts have a high stiffness to minimise any deflection due to the loading from the bolt. The nature of the load cell also requires contact surfaces that are not prone to distortion under load so these parts need to be as stiff as possible which may require a minimum component thickness and material hardening.

The basic constituents of the bolt block require some means of supporting the re-bar, some means of incorporating the load cell and a threaded hole for the bolt. All this needs to be held securely in place by components and joints with high stiffness.

The method of supporting the re-bar needs to be long enough for 100 mm of re-bar. A block with a V shaped groove cut into one side could be used for this purpose. A V block is shown in figure 4.1.

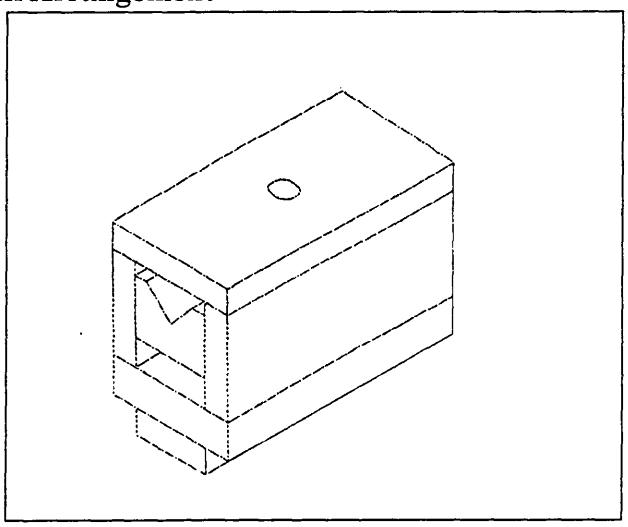
**Figure 4.1**
**Vee Block**



The bulk of the material below the bottom of the groove is to prevent any distortion of the bottom face due to the loading. Any distortion would produce an erroneous load cell output. The threaded hole for the bolt, needs to be approximately 4 mm away from the nominal edge of the reinforcement bar.

All this was incorporated into the frame arrangement shown in figure 4.2. The reinforcement bar will sit in the V-block which in turn, sits on top of the load cell which in turn sits on a base plate. By means of upright walls a plate, with the threaded hole tapped into it, is bolted to this base plate. So that the bolt block can be held in a vice a block was bolted to the base plate.

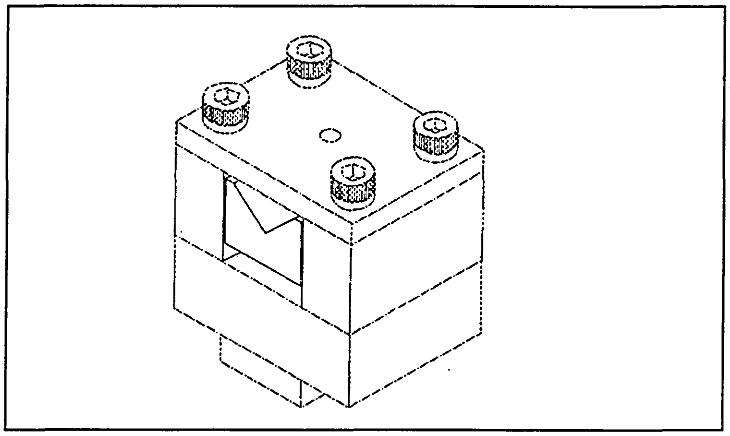
**Figure 4.2 Vee Block Arrangement**



All of the proposed bolted joints were designed to incorporate a high stiffness and be suitable for the required preloading and various strength requirements. The required plate thickness for the threaded hole plate was also calculated. These calculations and the calculations for the vice plate bolts are shown in Appendix B.

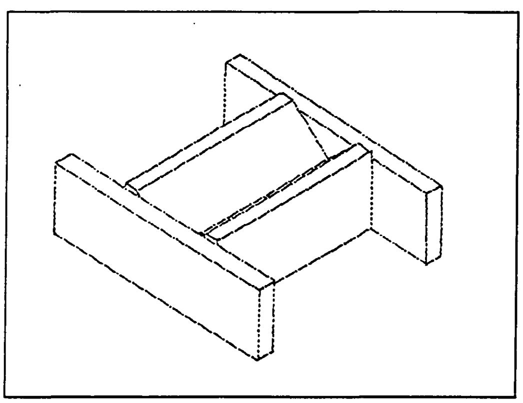
The designed bolt joints required an increase in dimensions of most of the bolt block frame components. The resulting design is shown in figure 4.3. The V-block has a clearance fit so that no friction from wall contact can falsify the load cell results. Due to stiffness and the load cell specifications the base plate upper surface, top plate lower surface, side plate joint surfaces and V-block underside require fine ground finishes. The load cell needed to be in contact with perfectly flat surfaces and the bolted joint contact surfaces need to have a high quality finish.

**Figure 4.3**



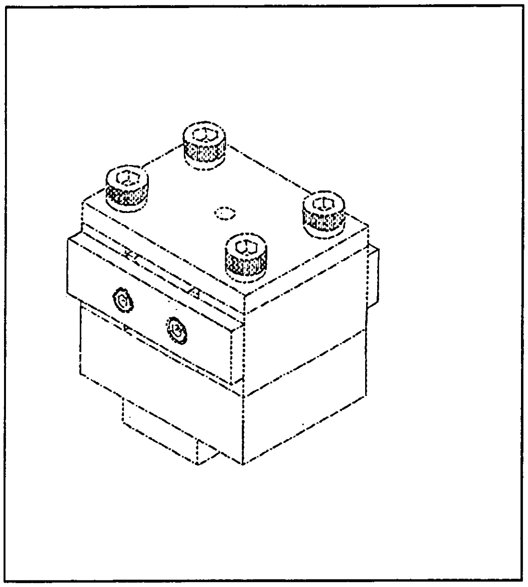
To prevent the re-bar sample from leaving the bolt block during tightening the V-block was be held in place by a retaining plate placed at each end of the block as shown in figure 4.4.

**Figure 4.4**



The finished bolt block assembly design drawing is shown in figure 4.5. Engineering drawings for this design are contained in Appendix A.

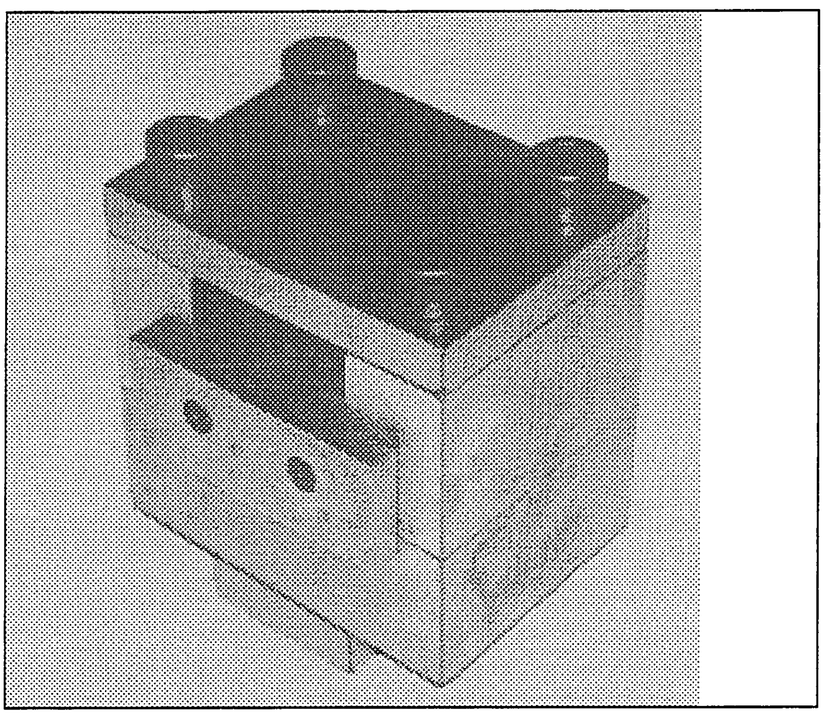
**Figure 4.5 Final Bolt Block Design**



### 4.4 Bolt Block Design Evaluation

The finished bolt block is shown fully assembled in figure 4.6. During assembly the bolted joints were tightened to the required torque using a calibrated torque wrench. No problems were encountered during this operation. The retaining plate bolts were only finger tightened as they are not strictly structural.

**Figure 4.6 Fully assembled bolt block without load cell fitted.**

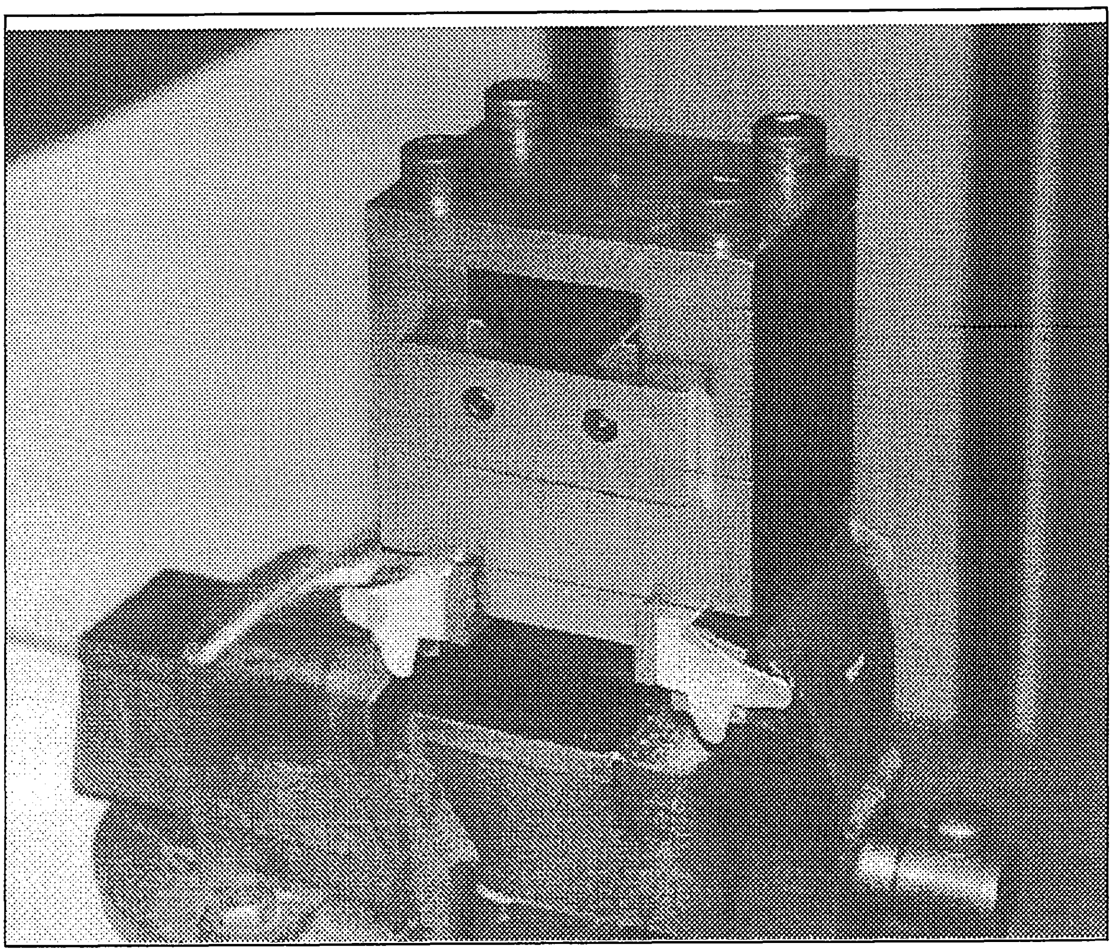


When the top plate is bolted down the block is held by the base plate, and not the vice plate, in a vice as the vice plate bolts were not designed for the torque required for the top plate bolts.

Load cell readings proved to have an acceptable repeatability for ten identical tests and the cell was easy to position into the block. The repeatable results prove that there was no distortion between the v block and the base plate during testing.

After a number of tests it became clear that the retaining plates were a rather redundant component of the design and removing them between tests became tedious. As a consequence the retaining plates were removed. No problems were encountered during further testing without them. The photograph in figure 4.7 shows the working bolt block mounted in a vice.

**Figure 4.7 Vice mounted bolt block with load cell.**



Following completion of the M12 tests, in order to test M14 and M16 bolts, two more top plates were produced with an M14 threaded hole and an M16 threaded hole respectively. It was calculated that the maximum torque, which is restricted by load cell range, that could be achieved with these bolts, could not exceed the strength of the block assembly.

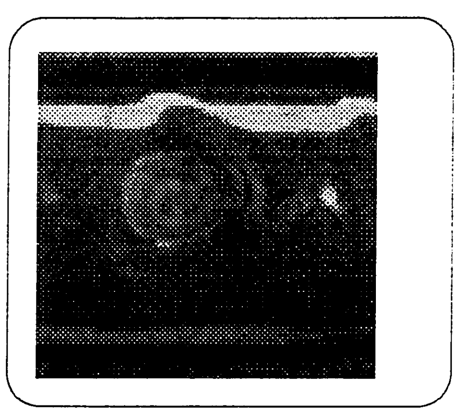
Again when the top plates are changed the block is held by the base plate within the vice jaws.

Repeatable performance and ruggedness make the bolt block an extremely good design. The top plates have been replaced repeatedly without effect. The versatility of this feature provides scope for further bolt and re-bar types. Varying re-bar sizes could be accommodated by varying the upright height. This could be done by producing a number of spacers which could act as extensions or as reducers when grouped.

### 4.4. Bolt Penetration Tests

Bolt penetration tests were performed first to determine a relationship between bolt torque, bolt penetration into the reinforcement bar and bolt downward force. This was done using the bolt penetration block. M14 bolts and 20mm ribbed reinforcement bar was used. A bolt was tightened to a certain torque and penetrated the bar. The bolt was then undone and the bar removed. A graph plotter recorded the maximum load experienced by the load cell and the dimensions of the indentation in the bar was measured. As the profile of the bolt cone is $90^0$ so the depth is half the hole diameter. The hole diameters were measured using a travelling optical microscope. Figure 4.8 shows a typical penetration test.

**Figure 4.8 Typical Bar Penetration**



Bolt penetration was measured for M12, M14 and M16 bolts, each using five different values of bolt torque, with four tests per value. A new bolt was used for each test. The expected bolt downward force for each torque value was calculated, using the following engineering formula (ref 22) :

$$\text{Bolt downward force } = T \times \frac{((6.2832 \times r)-(\mu \times P))}{(P+6.2832 \times \mu \times r)} \times \frac{1}{r} \qquad \text{Eq. 4.1}$$

where T = Torque, P = Thread pitch, r = Thread pitch radius, $\mu$= Coefficient of friction (= 0.3) The results are shown in tables 4.1, 4.2 and 4.3.

**Table 4.1 - Results for bolt penetration tests using M12 bolts on 20mm bar**

| Torque (Nm) | Hole 1 diameter (mm) | Hole 2 diameter (mm) | Hole 3 diameter (mm) | Hole 4 diameter (mm) | Average diameter (mm) | Measured force (kN) | Calculated force (kN) |
|---|---|---|---|---|---|---|---|
| 50 | 4.39 | 4.45 | 4.41 | 4.43 | 4.42 | 25 | 25.81 |
| 75 | 5.22 | 5.11 | 5.19 | 5.16 | 5.17 | 37 | 38.72 |
| 100 | 5.89 | 5.93 | 5.91 | 5.95 | 5.92 | 51 | 51.63 |
| 125 | 6.72 | 6.69 | 6.61 | 6.66 | 6.67 | 66 | 64.53 |
| 150 | 7.45 | 7.45 | 7.36 | 7.38 | 7.41 | 78 | 77.44 |

**Table 4.2 - Results for bolt penetration tests using M14 bolts on 20mm bar**

| Torque (Nm) | Hole 1 diameter (mm) | Hole 2 diameter (mm) | Hole 3 diameter (mm) | Hole 4 diameter (mm) | Average diameter (mm) | Measured force (kN) | Calculated force (kN) |
|---|---|---|---|---|---|---|---|
| 50 | 4.34 | 4.32 | 4.26 | 4.26 | 4.30 | 22 | 22.15 |
| 75 | 4.72 | 4.78 | 4.64 | 4.76 | 4.73 | 33 | 33.22 |
| 100 | 5.50 | 5.52 | 5.46 | 5.48 | 5.49 | 43 | 44.30 |
| 125 | 6.12 | 6.12 | 6.16 | 6.16 | 6.14 | 55 | 55.37 |
| 150 | 6.76 | 6.86 | 6.78 | 6.82 | 6.81 | 65 | 66.45 |

**Table 4.3 - Results for bolt penetration tests using M16 bolts on 20mm bar**

| Torque (Nm) | Hole 1 diameter (mm) | Hole 2 diameter (mm) | Hole 3 diameter (mm) | Hole 4 diameter (mm) | Average diameter (mm) | Measured force (kN) | Calculated force (kN) |
|---|---|---|---|---|---|---|---|
| 50 | 4.01 | 4.09 | 4.02 | 4.08 | 4.05 | 20 | 19.52 |
| 75 | 4.65 | 4.59 | 4.61 | 4.63 | 4.62 | 29 | 29.92 |
| 100 | 5.21 | 5.15 | 5.18 | 5.22 | 5.19 | 37 | 39.05 |
| 125 | 5.74 | 5.76 | 5.75 | 5.75 | 5.75 | 48 | 48.81 |
| 150 | 6.35 | 6.29 | 6.33 | 6.31 | 6.32 | 56 | 58.57 |

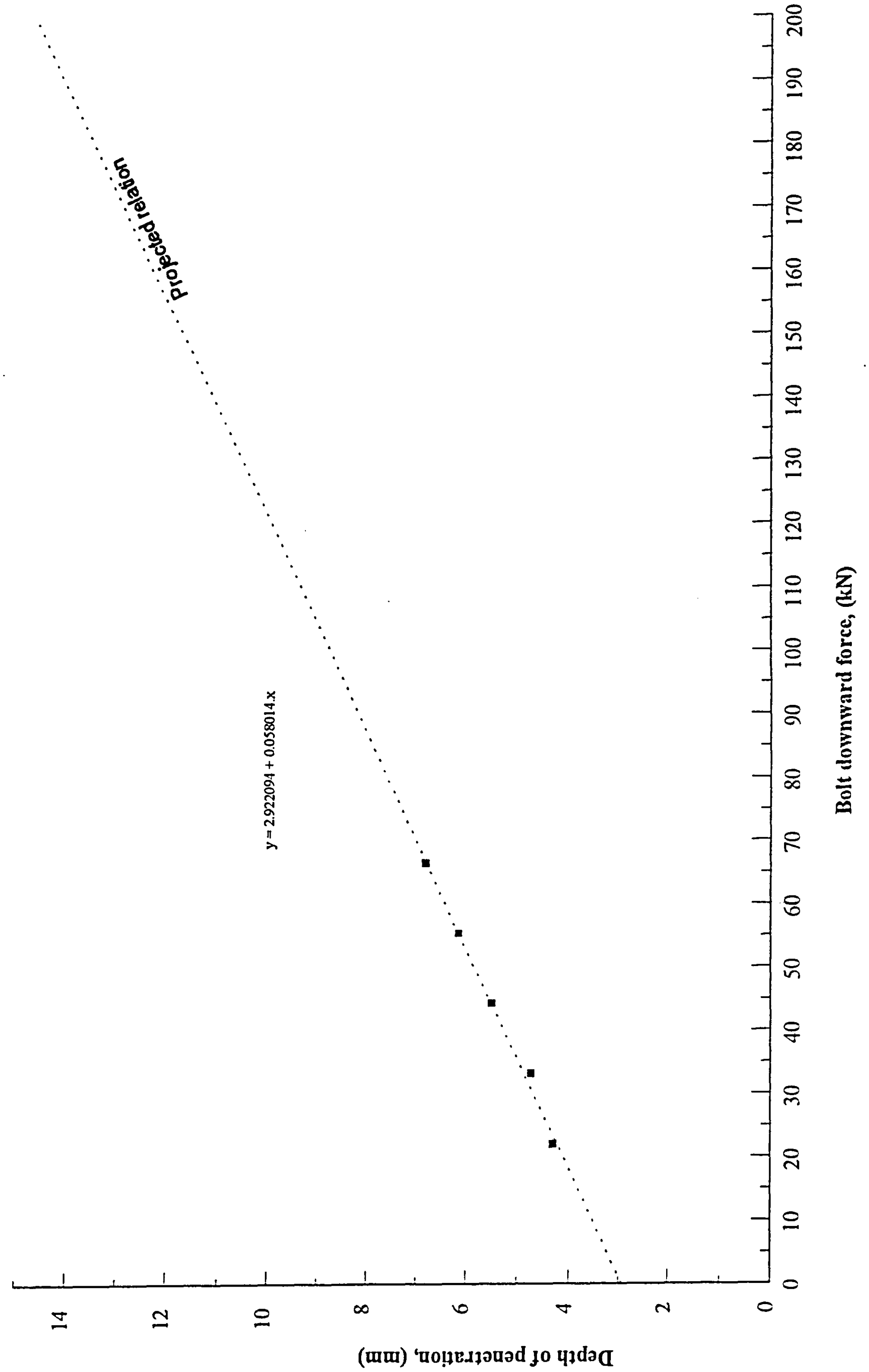
The measured and calculated results show little difference between them, so for convenience the calculated results were adopted. A graph showing depth of penetration against bolt downward force was plotted and a relationship determined between the two i.e.

Depth of penetration $= 2.922094 + (0.058014 \times \text{Bolt force(kN)})$

Since the bolt point was profiled at $90^0$ it was a simple calculation to determine depth of penetration from penetration diameter.

Graph 4.1 shows the curve extrapolated to include bolt downward force up to 200 kN.

## Graph 4.1 Relationship Between Bolt Penetration and Vertical Load



### 4.5 Maximum Torque for M12 bolts

As higher torque M12 bolts were likely to be investigated with the ET20 it was necessary to determine the maximum torque that could be applied to a non-shear M12 bolt used within an ET20 tube with dimensions of 48.3mm o/d and 33.4mm i/d i.e. the torque at which the tube threads strip or the torque at which the bolt shears across its minor diameter.

A standard ET20 coupler was fitted with eight non-shear M12 bolts. A piece of reinforcement bar, the same length as the coupler, was placed into the coupler and each bolt was tightened to 100Nm using a torque wrench. The torque for each bolt was then increased by increments of 5Nm until either the bolt or the tube thread failed. The results are given in Table 4.4.

**Table 4.4 Results of Maximum Torque Tests for M12 Bolt**

| Bolt no. | Failure torque (Nm) | Mode of failure |
|---|---|---|
| 1 | 190 | Bolt sheared |
| 2 | 195 | Bolt sheared |
| 3 | 195 | Bolt sheared |
| 4 | 195 | Bolt sheared |
| 5 | 185 | Bolt sheared |
| 6 | 195 | Bolt sheared |
| 7 | 195 | Bolt sheared |
| 8 | 195 | Bolt sheared |

With the lowest failure torque as 185Nm, it was considered reasonable that a maximum M12 bolt torque of 175Nm was permissible for further test purposes, which was considerably higher than the standard value of 108Nm for the current coupler.

### 4.6 Determination of bolt hole shear strength of tube

The developed ET20 tube thicknesses and the new prototype range tube thicknesses all needed to be determined in due course. To do this the shear strength of bolt hole tube threads needed to be found.

The tube material has a guaranteed manufacturers minimum tensile yield stress of 480N/mm$^2$ and a minimum ultimate tensile stress of 600N/mm$^2$. In theory the worst case bolt hole shear stress that could be experienced would be analogous with the minimum tube yield strength.

To perform the tests an Avery torsion testing machine was adapted to hold a coupler/bar sample and tighten a bolt through the tube and into the bar through the bolt hole. The machine is shown at figure 4.9 with a close-up of the jig at figure 4.10. The machine has a facility to record peak torque so that when the tube stripped in each case the maximum torque was recorded. A number of ET40 coupler samples were used and the tube stripped by a bolt in each case by progressively increasing torque until failure. Non-shear M20 bolts were used with the couplers.

**Figure 4.9 Avery Torsion Test Machine**

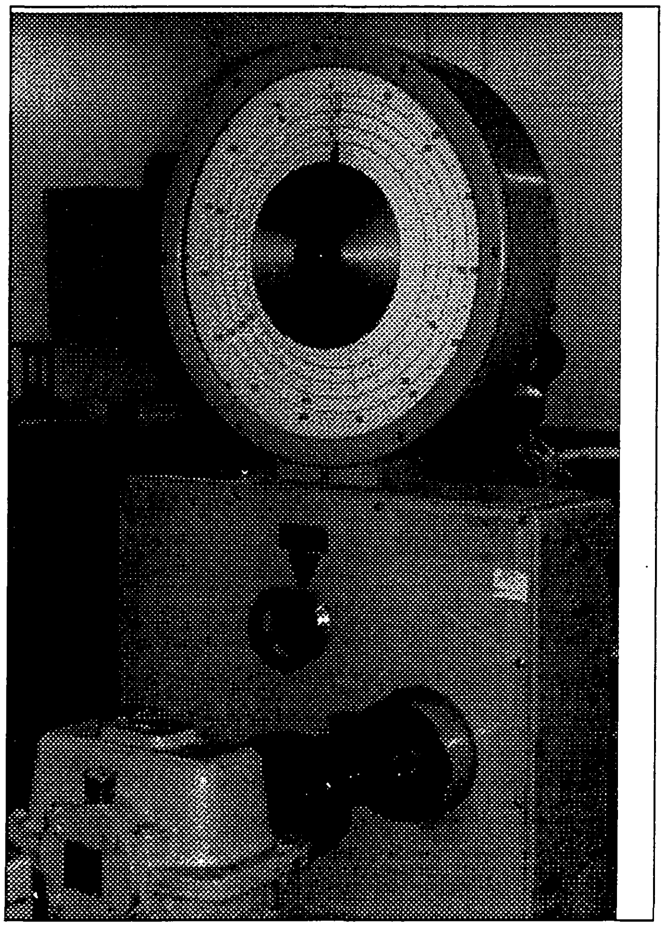
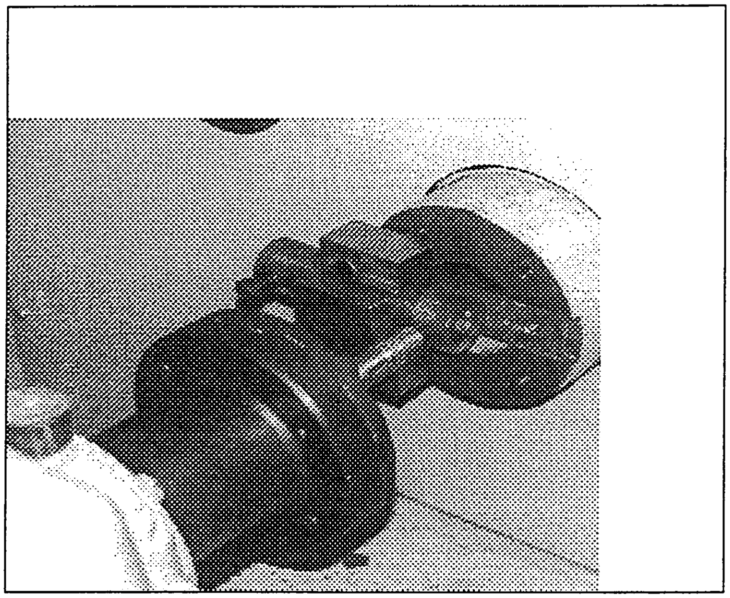


**Figure 4.10 Torsion testing machine set up for tube strip tests**



The test results are shown in Table 4.5. The tube threads stripped in each case and the respective nominal shear stress at maximum torque has been calculated as follows;

$$\text{Bolt hole shear stress} = \frac{2 \times \text{Bolt downward force}}{\text{Hole circumference} \times \text{Tube wall thickness}}$$

**Table 4.5 Tube strip test results for M20 bolts in an ET40 coupler**

| Specimen number | Bolt size | Maximum torque (Nm) | Associated downward force (kN) | Tube thickness (mm) | Nominal shear stress |
|---|---|---|---|---|---|
| 1a | M20 | 710 | 222.54 | 12.50 | 566.70 |
| 1b | M20 | 700 | 219.41 | 12.50 | 558.72 |
| 1c | M20 | 730 | 228.81 | 12.50 | 582.66 |
| 1d | M20 | 660 | 206.87 | 12.50 | 526.79 |
| 1e | M20 | 600 | 188.06 | 12.50 | 478.90 |

The lowest shear stress calculated is, as expected, analogous with the minimum tube yield strength. This value was used to calculate the prototype range tube thicknesses but first a factor of safety needed to be decided upon. The bolt hole stresses in the present range of couplers were calculated to see what factors of safety had been used to date. These values are given in Table 4.6.

**Table 4.6 Bolt hole stress factors of safety for present coupler range**

| Coupler type | Bolt type | Bolt torque (Nm) | Downward force (kN) | Tube thickness (mm) | Present bolt hole shear stress ($N/mm^2$) | Factor of safety |
|---|---|---|---|---|---|---|
| ET8 | M 10 | 55 | 34.06 | 6.35 | 341.47 | 1.40 |
| ET10 | M 10 | 55 | 34.06 | 6.35 | 341.45 | 1.40 |
| ET12 | M 10 | 55 | 34.06 | 6.35 | 341.45 | 1.40 |
| ET16 | M 12 | 108 | 55.75 | 7.90 | 374.36 | 1.28 |
| ET20 | M 12 | 108 | 55.75 | 8.50 | 347.94 | 1.38 |
| ET25 | M 16 B | 200 | 78.05 | 9.50 | 326.89 | 1.47 |
| ET28 | M 16 C | 360 | 87.86 | 12.50 | 279.65 | 1.71 |
| ET32 | M 16 A | 360 | 140.57 | 13.20 | 423.72 | 1.13 |
| ET40 | M 20 B | 450 | 141.05 | 12.50 | 359.17 | 1.33 |
| ET50 | M 20 A | 600 | 188.06 | 16.00 | 374.14 | 1.28 |

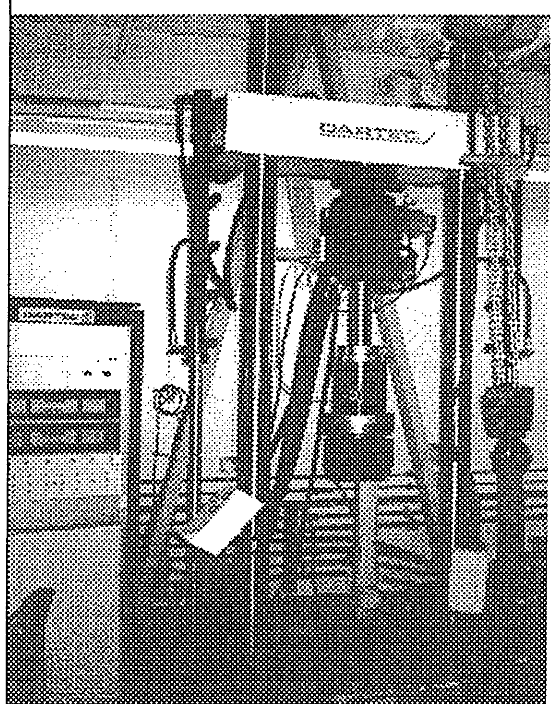
The lowest factors of safety is 1.13 for the ET32 coupler. This coupler was the third highest selling Ancon coupler for 1995 and there had been no reports of bolt stripping problems. It was decided then to adopt this factor of safety for all couplers in the range,and modify tube wall thicknesses accordingly.

# CHAPTER 5 COUPLER TEST EQUIPMENT AND INSTRUMENTATION

## 5.1 Test Equipment

Two alternative tensile testing machines were available for use during the project. Firstly a Dartec 250kN capacity servo hydraulic tensile testing machine, shown in Figure 5.1 below, which has adequate distance between the jaws and also fatigue test capability via external personal computer control.

**Figure 5.1 The Dartec Tensile Testing Machine**

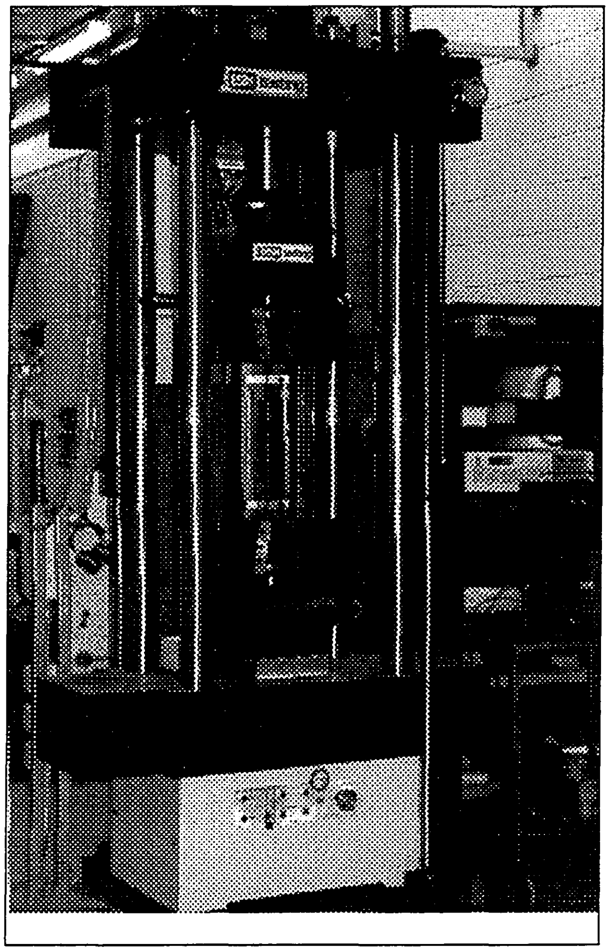


Secondly an ESH 600kN capacity servo hydraulic tensile testing machine was also available, see Figure 5.2 , however this machine does not have a fatigue loading capability.

The jaw limit for the Dartec machine was 28mm diameter while the ESH was able to accomodate up to 32 mm diameter. It was decided to use the Dartec for all static and fatigue tests up to 28 mm diameter and the ESH for static tests for 32 mm diameter. Fatigue tests for the latter were to be outsourced.

A five channel Siemens Kompensograph plotter mounted upon a trolley was used for recording data from either machine. Displacement measurements were performed using a pair of low voltage displacement transducers (LVDTs) with a range of 10mm accurate to ±0.001mm.

**Figure 5.2 ESH 600kN Tensile Testing Machine**



Unfortunately, there was no extensometer for use in this application, therefore one had to be designed and manufactured. Since it had been decided that all initial testing would be performed upon the ET20 size of coupler so the extensometer would be designed to suit this specimen.

## 5.2 Slip Measurement Extensometer

The objective was to design a slip measurement device for the permanent slip testing of reinforcement bar mechanical couplers. The constraints for the slip measurement testing rig follow as a series of individually headed items. They must be adhered to in the design process.

### Mechanical loading and dynamics

The testing rig should not transmit any of the tensile force involved in the mechanical testing. The design must incorporate some means of attaching the rig to the reinforcement bar accurately over the specified gauge length of 300mm.

Any attachment to the reinforcement bar will require a clamping force which will be supplied by the clamping device components.

The clamping force must be sufficient to support the weight of the rig.

Tensioning of the specimen may cause the arrangement to straighten and twist the rig which must be accomodated.

The rig assembly had to be rigid to avoid any erroneous contribution to the slip measurement.

Incorporation of clamping devices for the displacement transducers is required obviating the need for fine adjustment . Two or more transducers to be used to get an average measurement because of the bar straightening effect. Ideally then, these transducer clamps should be placed at opposite sides of the rig. Any relative movement of the rig must be smooth.

### Environment

The rig is used in testing where displacements are small and not subject to sudden, potentially damaging, failures.

The testing is performed in air.

**Weights and dimensions**

The rig should be as light as possible.

The specific gauge length to be measured is exactly 804 mm.

The electronic displacements are 130 mm in length with an outer diameter of 8 mm.

The outer diameter of the ET20 reinforcement bar mechanical coupler is 48.3 mm.

**Manufacture**

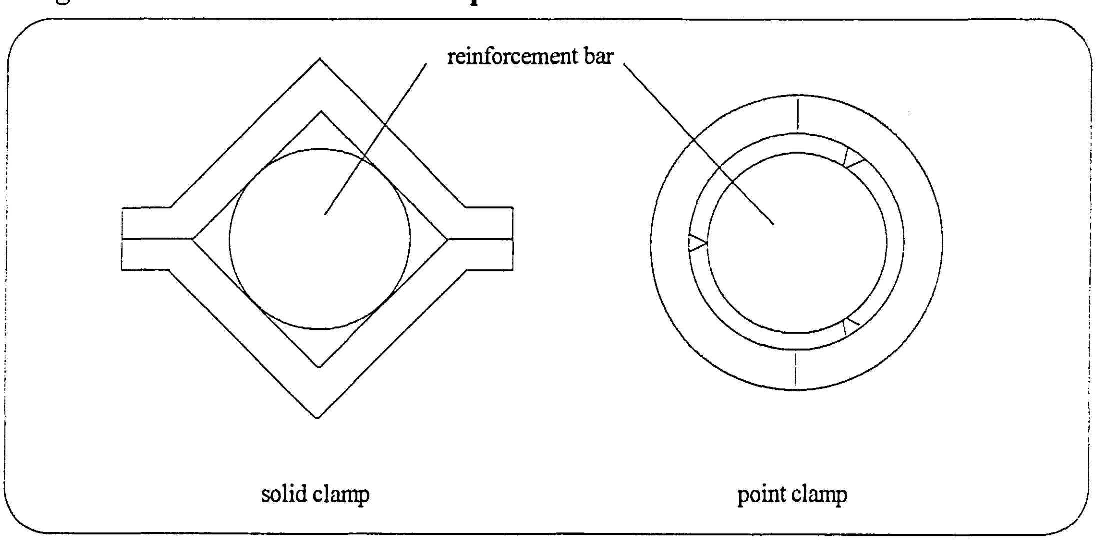
Only a single rig unit is to be manufactured so production of the design must be possible using standard machine shop technology. Any fasteners or materials used must be of a standard size.

## 5.3 Conceptual Design of Extensometer

When the specification was analysed it was clear that the design requirement could be divided into three constituents: the method of clamping the reinforcement bar, the method of connecting these clamps and the method of attaching the transducers.

Two design types were initially conceived , a solid face clamp and a point clamp. These are illustrated in figure 5.3.

**Figure 5.3 Solid and Point Clamp Alternatives**



Both types are made of two halves to enable easy assembly/dissassembly.

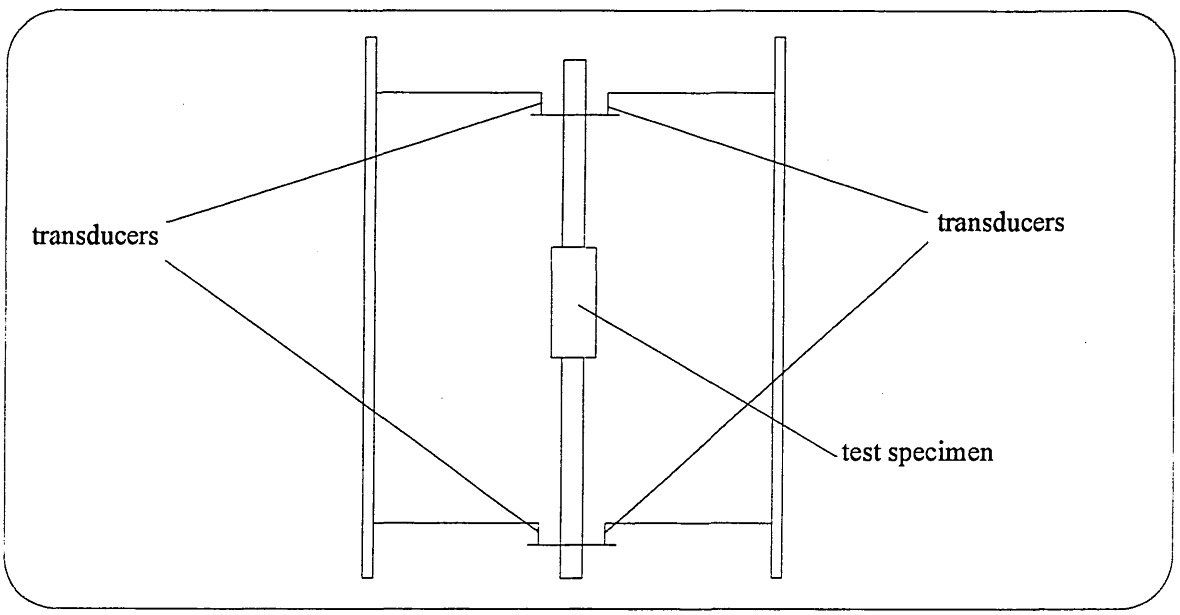
Both clamps could be aligned easily enough. The solid face clamp would involve more material and therefore be heavier. Also the amount of contact area with the reinforcement bar may affect the displacement measurement - a result of its high rigidity.

The point clamping was preferred as it would have minimal affect on the displacement measurement.

These clamps needed to be connected so that the displacement between the clamping points could be measured. Three configurations were devised for this.

The first one involved connecting the clamp points by external means as illustrated in figure 5.4.

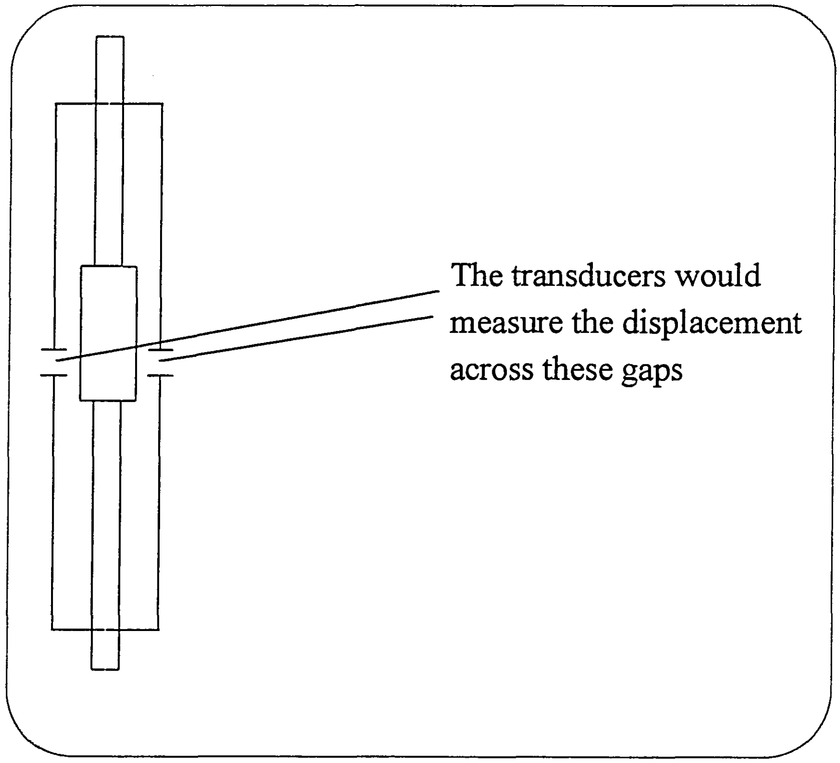
**Figure 5.4 External Connection of Clamps**




The uprights would either be fixed to the floor or to the test machine uprights. The displacement of each clamp would be measured and summed.

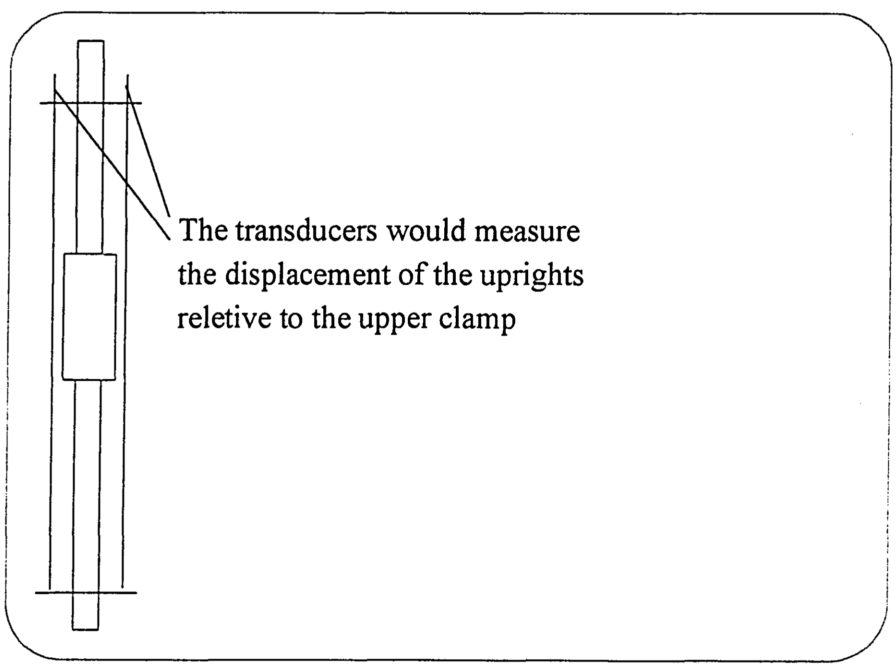
The other two configurations involved physical connection between the two clamps themselves. One involved two 'L' shaped connectors attached to each clamp, seen in figure 5.5.

**Figure 5.5 L-Connector Clamp**




The third configuration involved uprights that are fixed to one clamp and allowed to slide through the other clamp. This is shown in figure 5.6.

**Figure 5.6 End Connector Clamp**



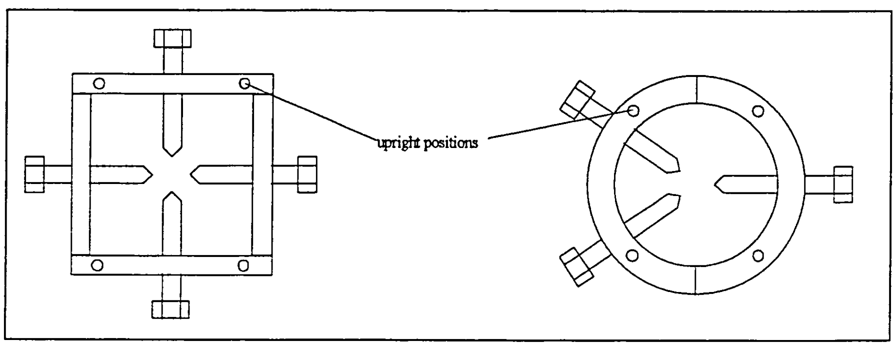
Rigidity and weight were the important factors for these connections. Undoubtedly the third configuration was the most rigid due to its simplistic arrangement, so that this configuration was combined with the point clamps.

With this arrangement the transducers simply needed to be attached to the steel uprights so that the displacement of the upper clamp could be measured. This just involved a block hole with a grub screw locking mechanism.

The selected configuration was then designed in detail.

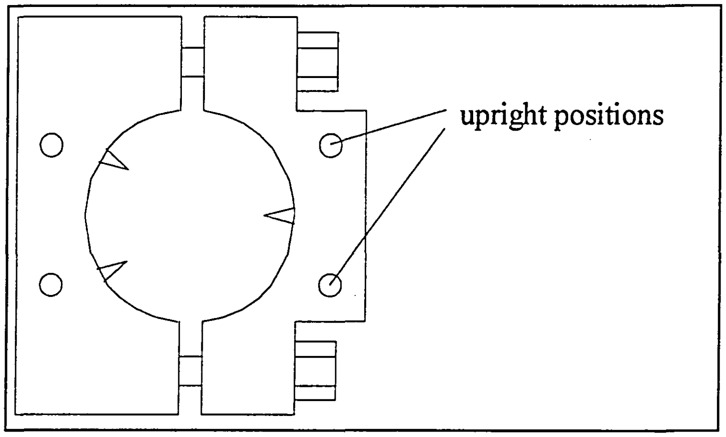
The size of the clamps was constrained by the coupler dimensions as the uprights must sit clear of the coupler. To save on weight long bolt points were considered in the clamp arrangements in figure 5.7.

**Figure 5.7 Alternative Clamping Configurations**



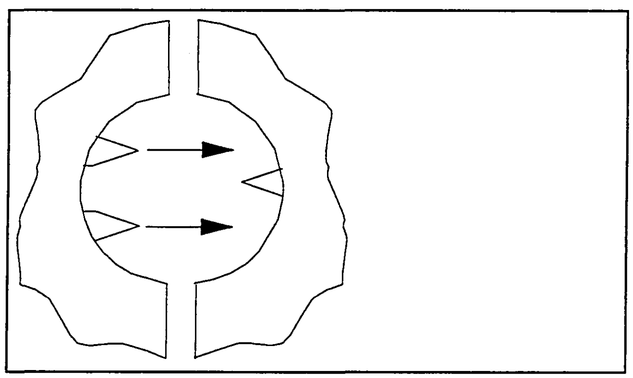

Hexagon socket head bolts provide the rapid assembly of the clamp halves/parts in each case. However, the distance, in each clamp arrangement, from the point to the threaded hole is large enough to allow distortion and misalignment of the points due to the bolt stiffness and thread fit. Consideration of this decided that to have rigid clamps with this degree of upright spacing would require as much solid material between the points and the upright positions as possible. Figure 5.8 shows the derived clamp design.

**Figure 5.8 Clamp arrangement**



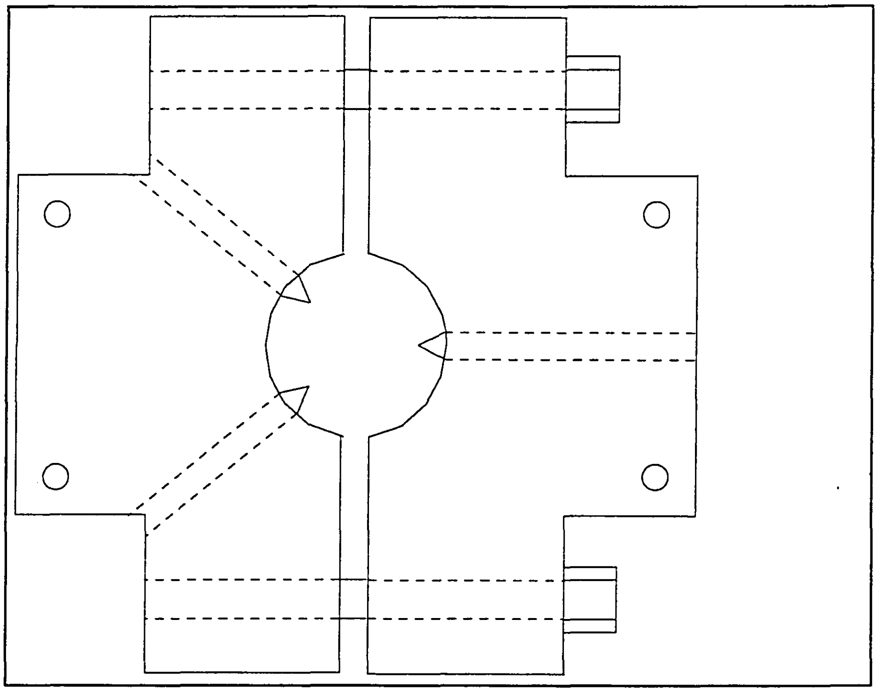

After much deliberation it was decided to further amend this design. As this design is clamped, via the bolts, the pins would be forced into the bar at an angle. This would tend to bend/break the pins and leave a void in the reinforcement bar behind the penetration path of the pin. It was felt to offer an improvement to align the pins with the clamping force direction as in figure 5.9,

**Figure 5.9**



but again this would tend to bend/break the pins. A much better method would be to use threaded pins which can be tightened against the reinforcement bar, independent of the clamping bolts. To ease manufacture these were made by turning points onto hex socket set screws and tip hardening them. The finished clamp design is shown in figure 5.10.

**Figure 5.10**

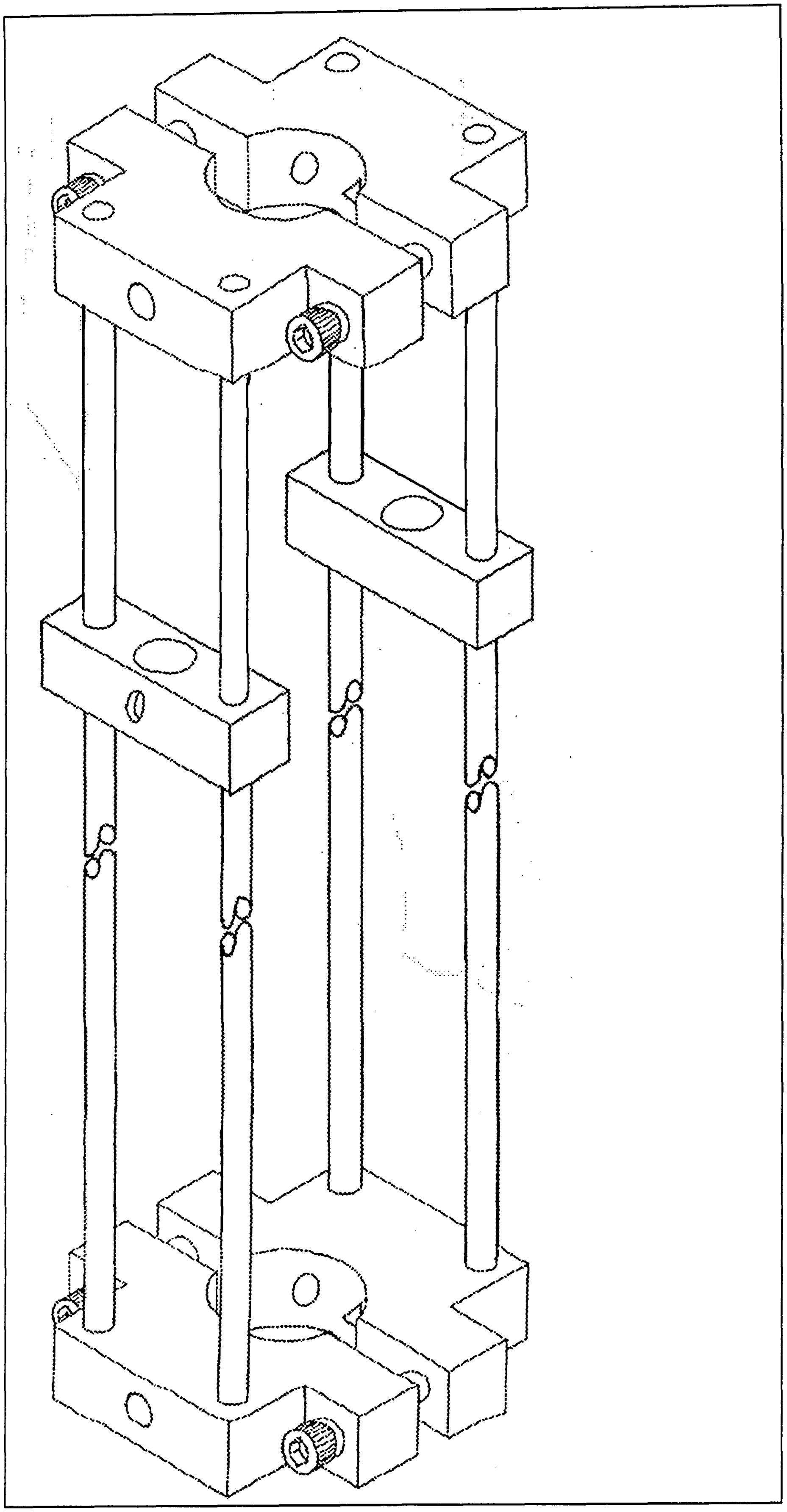


If required the set screws could be locked in place by a second 'locking' set screw.

The uprights were fixed into one clamp and allowed to slide through the other clamp. To attach the transducers a cross-head was fixed to the uprights below the upper moving clamp. This saved on the testing machine daylight required as opposed to attaching them above the upper clamp.

A final rig assembly drawing can be seen in figure 5.11. Engineering drawings for the production of the device are contained in Appendix A.

**Figure 5.11 Final Design for Extensiometer**



## 5.4 Extensometer Development

### 5.4.1 Design evaluation

Originally the uprights were to be glued into the lower clamp. It was decided to fix the uprights by incorporating grub screws into both the upper and lower clamps. Grub screws were incorporated in both clamps to ensure that the gauge length could be fixed until the clamps were attached.

Initial tests with the slip rig upon a standard coupling produced permanent extension results far higher than expected when compared with previous Ancon Clark results.

This initially resulted in a calibration check of the instrumentation involved and also a test involving the slip rig.

The calibration check validated all of the instrumentation concerned.

The test on the slip rig itself involved attaching each clamp to a small piece of re-bar. The free ends of the re-bar pieces were clamped into the testing machine jaws. The normal instrumentation was attached. The jaws were moved apart by the machine and the instrumentation produced a perfect trace.

At this stage the slip rig was considered to be working perfectly.

As testing machine daylight was no longer a concern the cross-heads were repositioned above the upper clamp to ease transducer attachment and detachment.

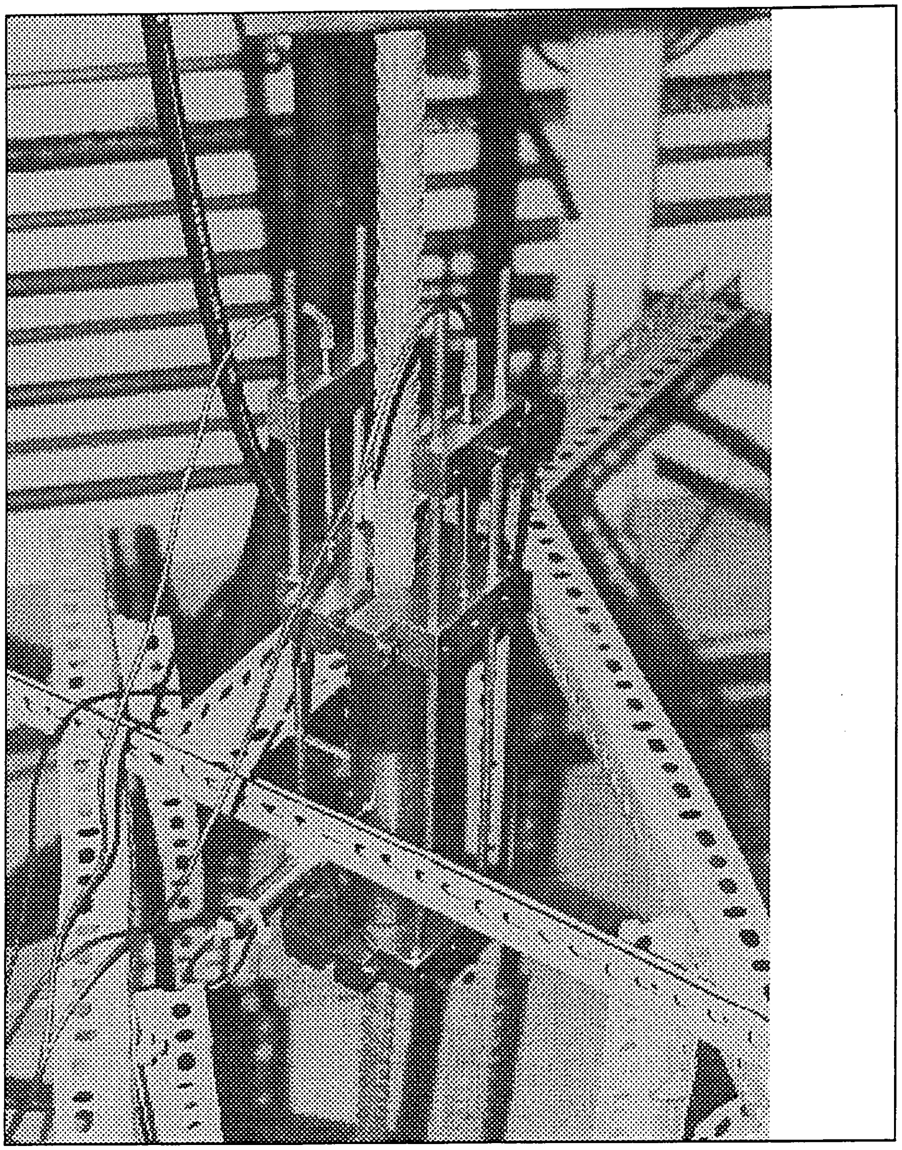
This revised coupler arrangement can be seen in figure 5.12.

Further tests with the slip rig still produced results higher than expected. This became a concern and it was considered that the specimens being measured by this rig (just visible in figure 5.12) were contributing to the erroneous results by straightening in the early stages of the tests. Further slip rig tests were undertaken to investigate the contribution to permanent slip by sample straightening. These involved the usual displacement transducers plus two more transducers, placed at

$90^0$ to each other, used to measure lateral displacement of the test specimen which is also visible in figure 5.12.

These test results demonstrated that straightening of the reinforcement bar has little contribution to permanent extension. An amount of lateral movement was detected, which, when translated into axial movement becomes negligable.

**Figure 5.12 Straightening test set-up**



A further test using the ESH test machine, which has hydraulic jaws and might be thought to provide more straightening during set-up, also confirmed this.

The next slip rig test used a single piece of re-bar connected between the testing machine jaws. A single piece of re-bar is known not to have any permanent extension at the loads in question. The slip rig was attached to the re-bar and tested. The results produced a permanent extension. This confirmed that the slip rig mechanism was affecting the extension results.

### 5.4.2 Refinement of Design

It was believed that the uprights were the problem with the operation of the extensiometer. The top clamp was removed and the bottom clamp released from the uprights. When examined manually any small (1 mm) deflection on the free end of an upright made it impossible to pull the upright from the bottom clamp.

Bending induced during slip rig attachment could easily provide this small displacement. It could also be induced by having varying distances between half collars, top and bottom.

Four 10 mm spacing rings were made to ensure that the half collars were identically spaced, top and bottom. The uprights were greased and this revision was tested - without success.

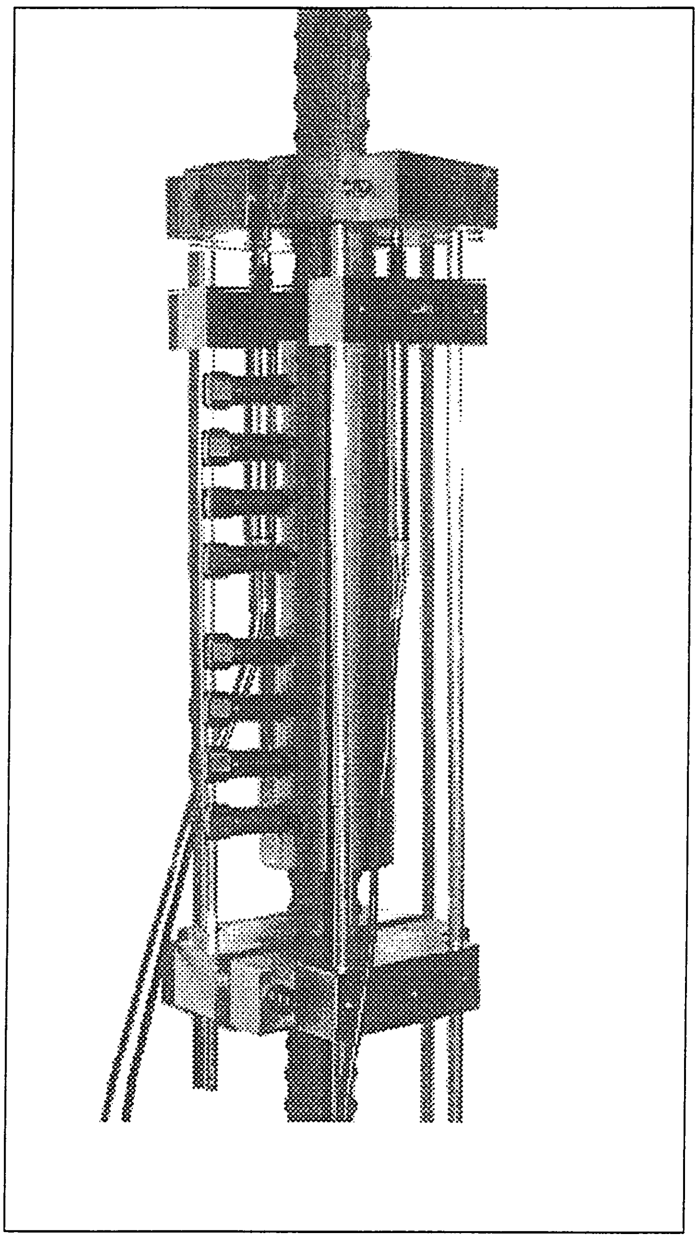
At this stage a decision was made to see how established test lab slip rigs were configured. A visit to Nottingham University testing labs gave one example. Their design consisted of two solid clamps and rigid uprights, fixed to the bottom clamp, which sit just below the upper clamp. Both solid clamps incorporated springs which ensured that a consistent clamping force was maintained during testing as the re-bar cross-section reduced.

Before any major changes were made to the slip rig it was decided to first try a rearrangement using the existing components.

First the spacing rings were removed. The uprights were fixed to the bottom clamp and made to sit below the upper clamp. The cross-heads were positioned a small distance below the upright upper ends. Small spring washers were placed beneath the caps of the main clamp bolts. The revised arrangement is shown in figure 5.13.

The arrangement was tested on three standard couplers. The results were excellent and the slip rig now worked. The design was adopted and showed an additional advantage. The uprights were measured to ensure the correct gauge length each time when assembled with 'shims'. The shims provided a 1 mm gap between the top clamp and the uprights allowed for any movement of the upper clamp due to any straightening. A final assembly drawing of this arrangement is given in Appendix A.

**Figure 5.13 The revised extensometer arrangement in operation.**
Notice the 1mm gap between the top clamp and uprights.



# CHAPTER 6 ET20 DEVELOPMENT

## 6.1 Original ET20 Coupler Tests

### 6.1.1 Pull Test on Standard ET20 Coupler

To provide a benchmark for the ET20 development work a sample of seven standard ET20 couplers from the present coupler range were tested in the Dartec machine to compare against the increased performance criteria. The couplers were of a standard configuration with eight M12 bolts torqued to 108 Nm and the results are given in Table 6.1.

**Table 6.1 Results of pull test on standard ET20**

| Specimen number | Permanent extension (mm) | Maximum load achieved (kN) | Target maximum (kN) | Nature of maximum load |
|---|---|---|---|---|
| 1a | 0.16 | 184.7 | 197 | End of m/c stroke |
| 1b | 0.16 | 188.6 | 197 | End of m/c stroke |
| 1c | 0.17 | 184.8 | 197 | End of m/c stroke |
| 1d | 0.18 | 189.5 | 197 | End of m/c stroke |
| 1e | 0.16 | 189.0 | 197 | End of m/c stroke |
| 1f | 0.14 | 188.7 | 197 | End of m/c stroke |
| 1g | 0.13 | 188.1 | 197 | End of m/c stroke |

Applying the t distribution (24) approach to these results gives confidence levels for determining the performance of a standard ET20 coupler (see Appendix B for calculations).

For a 95% confidence level the average extension of a standard ET20 coupler can be said to be 0.157 mm ± 0.017 mm.

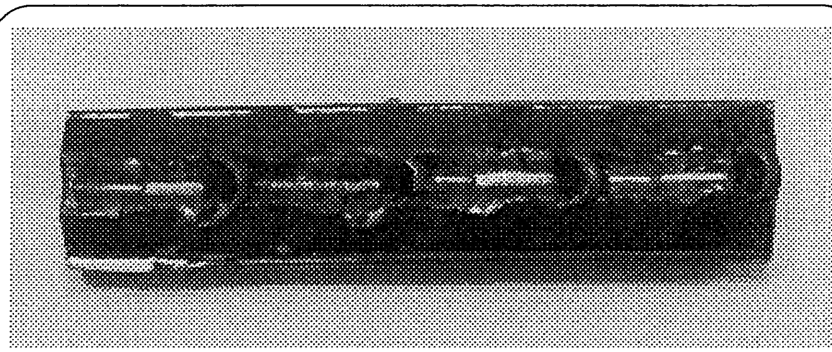
For a 99% confidence level the average extension of a standard ET20 coupler can be said to be 0.157 mm ± 0.026 mm.

The maximum load of 197 kN was not achieved due to the stroke limitation of the machine, which was overcome in future tests by resetting the stroke and performing a double operation.

### 6.1.2 Effect of Bolt Numbers on Upper Tensile Load (U.T.L.)

A series of tests were undertaken to identify the contribution of shear bolts to the coupler upper failure load performance. Four standard ET20 couplers were used for the monotonic pull tests, with differing numbers of M12 bolts i.e. the normal eight bolts, six bolts, four bolts and two bolts. The torque applied was 108Nm which is recommended for this range. The bolts were removed from each end of the coupler inwards. Each specimen was tested to failure; figure 6.1 illustrates a typical reinforcing bar condition following 'pull-out' failure.

**Figure 6.1 Typical Reinforcing Bar After 'Pull-Out' Failure**



The results of the tests are given in Table 6.2.

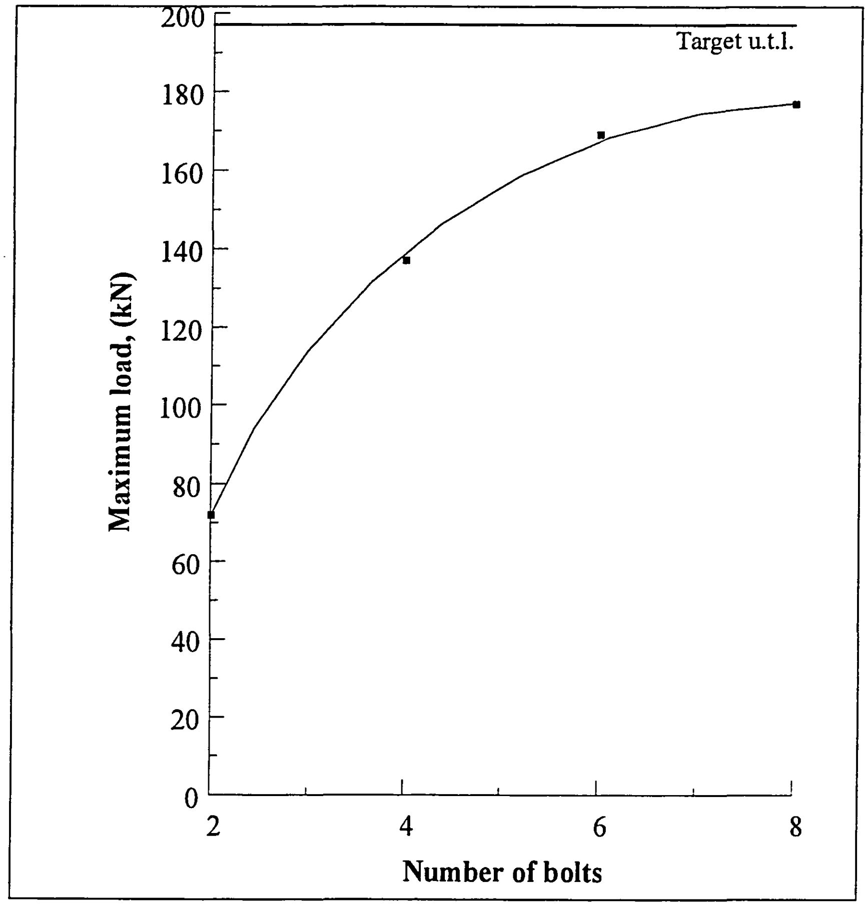
**Table 6.2 - Results of Pull Tests on ET20 with Varying Bolt Numbers**

| Specimen no. | No. of bolts | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 2 | 72.0 | Bar pull out |
| 1b | 4 | 137.0 | Bar pull out |
| 1c | 6 | 169.0 | Bar pull out |
| 1d | 8 | 177.0 | Ductile in bar |

Graph 6.1 illustrates the relationship between the upper failure load and the number of bolts.

The most significant feature of these results is that the fourth bolt contributes less than 10% of the maximum attainable load, suggesting an improvement in performance by changing the number of bolts and torque values, thereby changing penetration and clamping force. This was the objective of the subsequent series of tests, adjusting the size and number of bolts. From the cost analysis, this approach was also thought likely to generate the largest savings and provide consequential benefits in terms of tube material reduction..

**Graph 6.1 - Maximum Load vs Bolt Numbers for Standard ET20 Coupler**



## 6.2 Tests on ET20 with Varying Bolt Sizes and Quantities

### 6.2.1 Pull Test for ET20 with 8-M12 bolts and 120Nm torque

Each specimen consisted of a standard ET20 coupler with eight M12 bolts which were tightened to a torque value of 120 Nm in order to provide an increased clamping force and bolt penetration into the reinforcement bar The results are shown in Table 6.3.

**Table 6.3 Results for ET20 with 8-M12 bolts torqued to 120Nm**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) |
|---|---|---|
| 1a | 0.14 | 177.0 |
| 1b | 0.20 | 182.0 |
| 1c | 0.15 | 177.0 |

### 6.2.2 Pull test for ET20 with 6-M12 bolts and 130 Nm/150 Nm bolt torque

Two samples were tested, one with six M12 bolts torqued to 130Nm and a second with six M12 bolts torqued to 150Nm. The coupler length remained the same as standard and a sample size of three specimens was used.

Both samples were tested for increased slip and U.T.L. performance and the results are given in Table 6.4.

**Table 6.4 - Results of ET20 coupler tests with 6-M12 bolts torqued to 130 Nm**

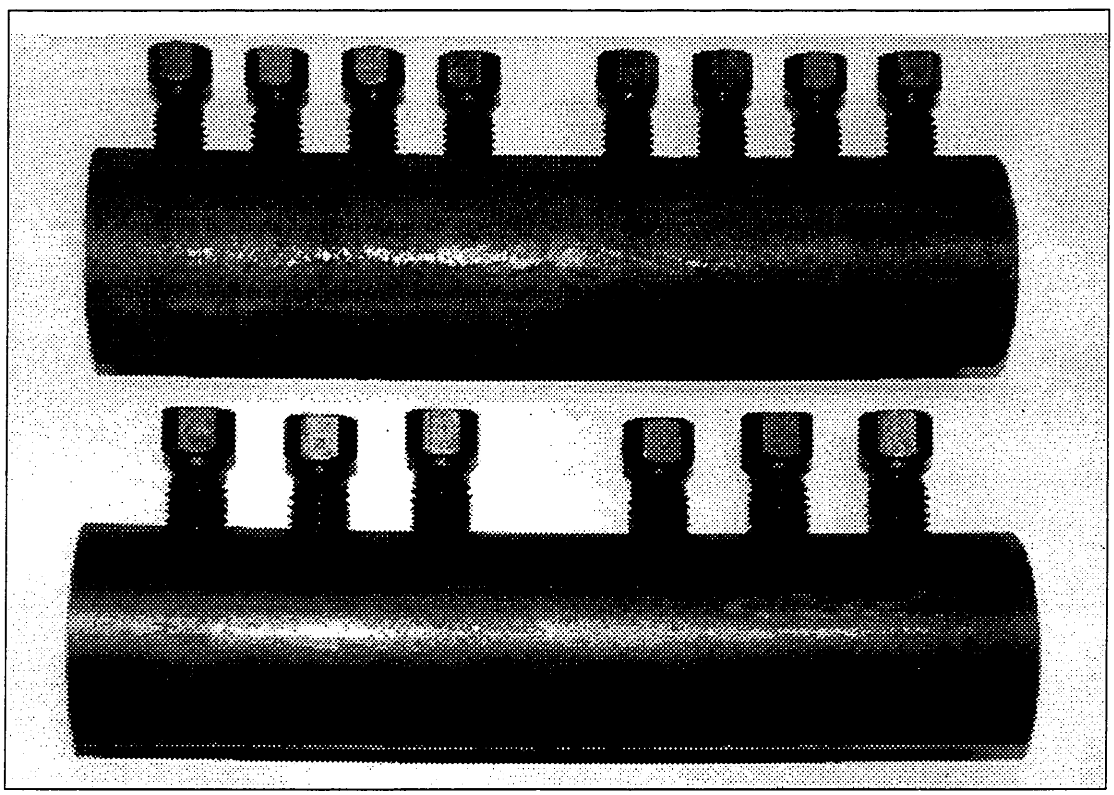
| Sample and specimen number | M12 bolt torque (Nm) | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) |
|---|---|---|---|
| 1a | 130 | - | 180.1 |
| 1b | 130 | 0.62 | 177.4 |
| 1c | 130 | - | 176.7 |
| 2a | 150 | 0.43 | 177.7 |
| 2b | 150 | 0.42 | 177.6 |
| 2c | 150 | 0.34 | 177.9 |

The two slip results for the first batch were not obtained due to failure of the instrumentation. The increased bolt torque did, in both cases, increase the upper failure load for the coupler. The overall stiffness of these arrangements does however result in poor slip performance; more than twice as much as a standard ET20 in the case of sample 2. It was felt that the stiffness may be improved by adding a further M12 bolt to each end of the coupler or by using six larger diameter bolts. Larger diameter bolts may have a stiffer seating because of the increased bolt hole size.

### 6.2.3 Pull Tests on ET20 with 6-M14 and 6-M16 bolts, each 150 Nm torque

Both samples used standard length ET20 couplers. The comparison between a standard ET20 and this new configuration can be seen in Figure 6.2.

**Figure 6.2 Comparison between standard ET20 and 6-M14 bolt specimen**



Both the M14 and M16 sample bolts were tightened to a torque value of 150 Nm. The results are given in Table 6.5.

**Table 6.5 ET20 coupler with 6-M14 and 6- M16 bolts, each 150 Nm torque**

| Sample and pecimen number | Bolt type | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) |
|---|---|---|---|
| 1a | M14 | 0.11 | 184.0 |
| 1b | M14 | 0.11 | 184.0 |
| 1c | M14 | 0.15 | 184.6 |
| 2a | M16 | 0.14 | 188.2 |
| 2b | M16 | 0.15 | 187.6 |
| 2c | M16 | 0.15 | 189.1 |

Due to the lower permanent slip results, the M14 configuration was clearly the better of the two, even with marginally lower UTL values. This arrangement also performed significantly better than the standard ET20 coupler, the results of which are shown in table 6.1, and was the best performing configuration to date. The one using M16 bolts was consistently poor with a perfomance similar to the coupler using 8 - M12 bolts torqued to 120 Nm. Based on these improvements further work was undertaken with the tube shortened to accomodate six bolts of M14 size.

### 6.2.4 Pull test on shorter ET20 with 6-M14 Bolts and 150 Nm torque

The coupler length was reduced to 160mm from the standard length of 204mm, as shown in Figure 6.3. The new length is simply the old length minus the two bolt pitches that have been removed. The saddles had standard tooth form and heat treatment. The results for a sample size of four are shown in Table 6.6.

**Figure 6.3 Comparison of standard and short ET20 with 6-M14 bolts**

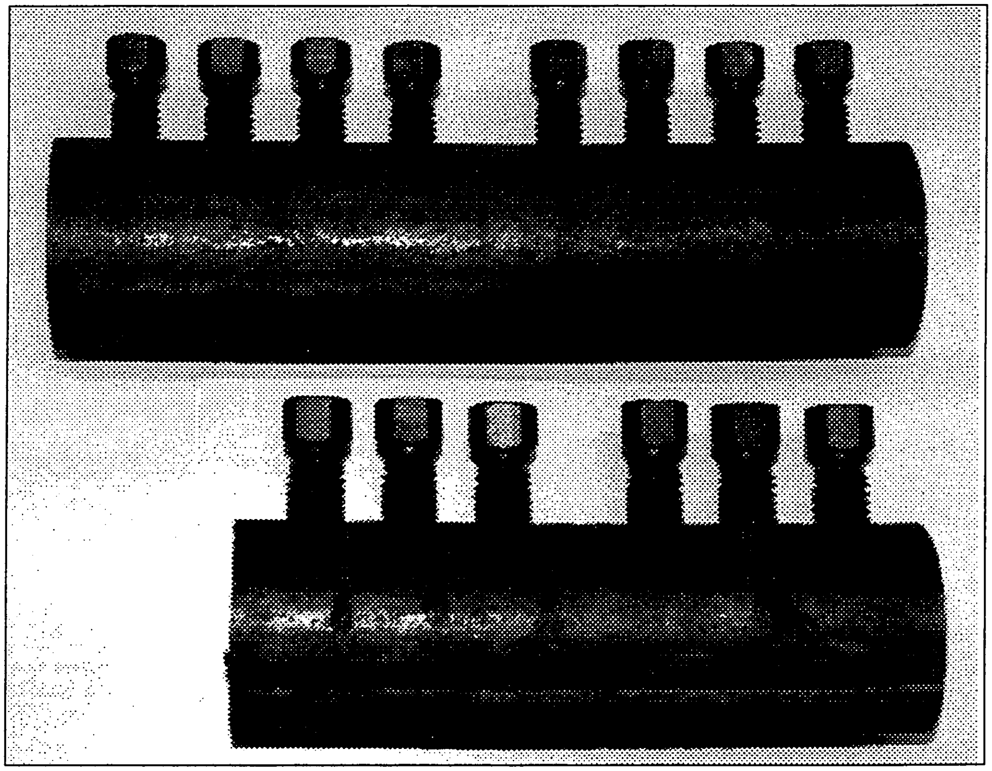
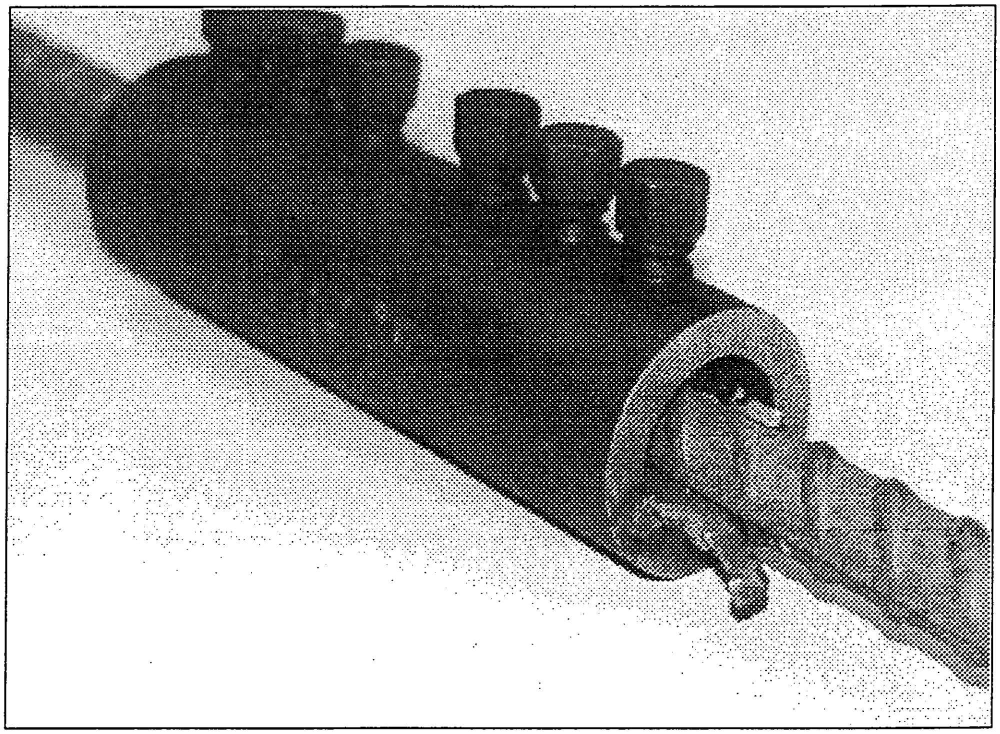


**Table 6.6 Results for short tube ET20 with 6-M14 bolts and 150Nm bolt torque**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.13 | 188.3 | Ductile in bar |
| 1b | 0.14 | 189.0 | Ductile in bar |
| 1c | 0.14 | 186.5 | Bar pull out |
| 1d | 0.13 | 187.9 | Ductile in bar |

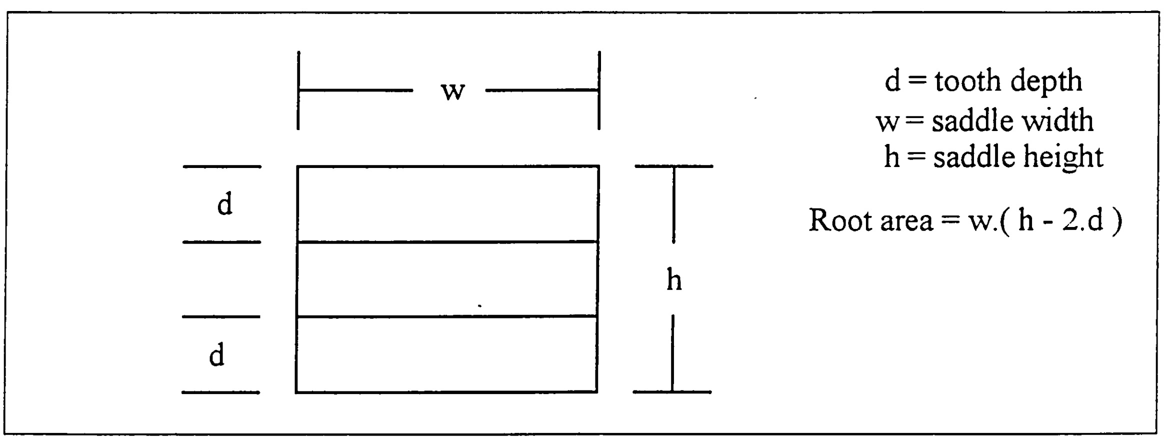
During testing it was observed that the saddles were breaking and were being pulled out of the coupler with the bar, shown in Figure 6.4. However, this configuration performed better than a standard ET20 in terms of slip. It was felt likely that the slip results were due to the shorter coupler having fewer teeth than a standard length coupler to key into the tube and bar.

**Figure 6.4 Saddle Pull Out**



It was felt that perhaps the induction heat treatment of the saddles was causing excessive brittleness in the saddle core. Possibly an alternative treatment could retain a softer core and maybe smaller teeth would reduce the effect by increasing the 'root' area of the saddle. The root area is illustrated in Figure 6.5 and the next series of tests were planned and executed to examine the effects of saddle geometry on coupler performance

**Figure 6.5 Saddle cross-section and root area**



### 6.2.5 Performance of tighter tolerance bolt hole threads for test 6.2.4

It was felt that the bolt fit may make a significant contribution to the coupler permanent slip. The present bolt and tube hole have medium fit threads (23). A sample of six M14 bolt short ET20 couplers was produced with a tighter tolerance bolt hole, i.e. M14 x 2.0 - 5H tap, to see if the slip result could be improved. These are in Table 6.7.

**Table 6.7 - Tighter tolerance hole coupler results**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.094 | 203.1 | Bar pull out |
| 1b | 0.056 | 198.5 | Bar pull out |
| 1c | 0.088 | 195.5 | Bar pull out |

These results indicated that tighter tolerance holes have a significant effect upon the slip performance. However, moving to a finer thread would create potential contamination site problems with dirty threads self-stripping and cross-threading, and it was expected that satisfactory performance may have been achieved using standard thread forms.

### 6.2.6 Summary

Figure 6.6 shows a tabular summary of the results for bolt variations suggesting that a coupler with 6 bolts of M14 type can be successful when torqued to 150 Nm , using close tolerance hole threads, although standard threads would have been preferred. The direct cost of using these bolts was 0.344 units compared to 0.368 for 8-M12 bolts. The cost of drilling and tapping was estimated to save 0.009 units. The reduction in bolts numbers enabled the tube length to be reduced from 204 mm to 160 mm reducing the cost of tube from 0.311 units to 0.244. These individual cost savings represented an overall total of 10% although more were anticipated from modifications to saddle geometry. At this stage seven configurations had been tested involving 22 samples in total.

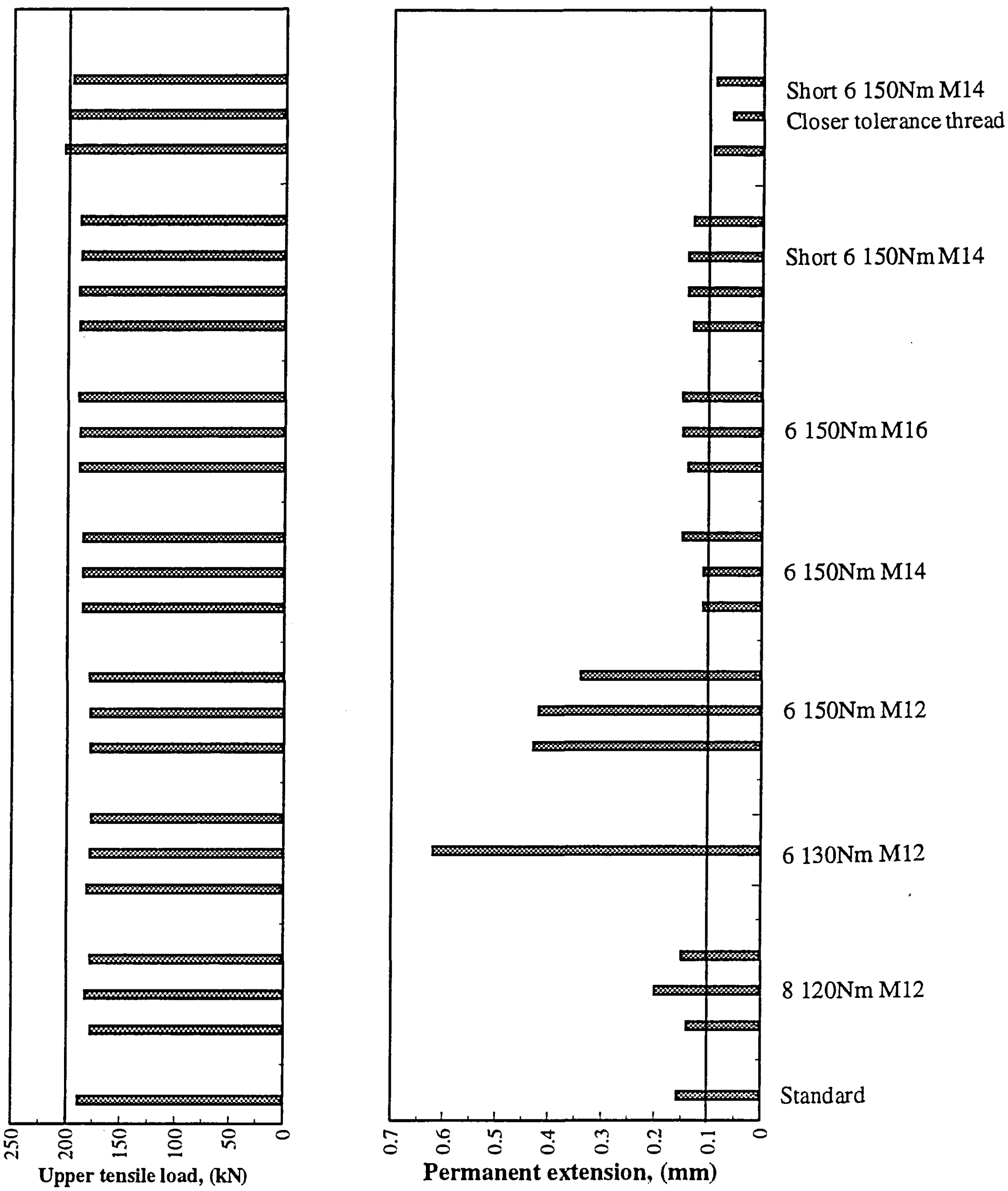



Figure 6.6 - Summary of ET20 bolt variation development

## 6.3 Serrated Saddle and Tube Modifications

### 6.3.1 Pull Test on ET20 with 6-M14 Bolts and Smooth Saddles

This configuration consisted of a standard length ET20 sleeve, six 150Nm M14 bolts and unserrated, untreated saddles. The object of the test was to determine the contribution to performance of serrated saddles by testing a smooth saddle configuration. A sample size of two specimens was tested, with the results tabulated below in Table 6.8.

**Table 6.8 Results for ET20 with smooth saddles, 6-M14 bolts and 150 Nm torque**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (mm) |
|---|---|---|
| 1a | 0.27 | 152.7 |
| 1b | 0.30 | 162.5 |

Comparing these results to those for couplers with serrated saddles, showed there was twice as much permanent slip and a 14% reduction in U.T.L.. It was a little surprising that the absence of teeth upon the saddles did not result in a much higher slip value. clearly some form of high friction resistance was necessary for effective operation. It was felt that maybe a less complicated means, other than teeth, could be found to provide the required stiffness or maybe smaller teeth would be beneficial. the first test was to compare nitriding with induction hardening for a standard coupler.

### 6.3.2 Pull Test on a short tube ET20 with 6-M14 bolts and standard teeth nitrided saddle

This configuration isolated the performance of nitride heat treatment alone without the small teeth. The results are given in Table 6.9.

**Table 6.9 - Results for Short Tube ET20 with Nitrided Standard Saddle**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.156 | 199.2 | Ductile in bar |
| 1b | 0.128 | 210.0 | Ductile in bar |
| 1c | 0.094 | 212.0 | Ductile in bar |

At least one saddle was broken (in half) in each test which indicated that the nitride treatment had no significantly improved performance. A natural step forward at this point was to try and determine the optimum saddle root area. Any reduction in saddle height would also enable a reduction in the sleeve outer diameter, but the first stage was to cut smaller teeth on a standard saddle.

### 6.3.3 Pull Test on Short Tube ET20 with 6-M14 bolts and small teeth nitrided saddles

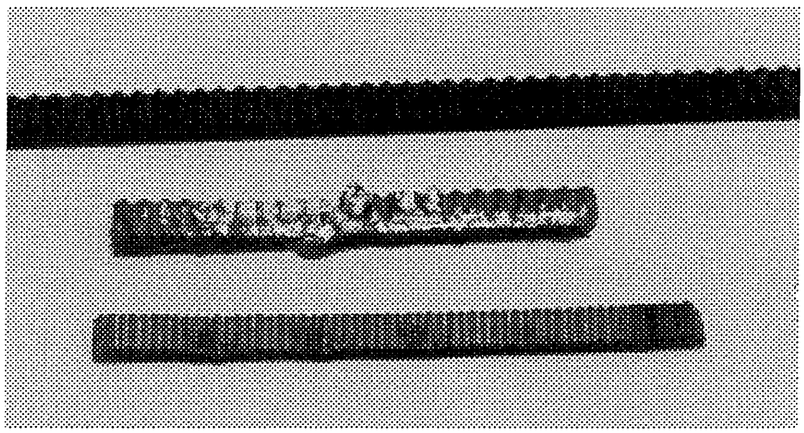
To clarify the benefits over the results in section 4.3.4 each specimen was made 160mm long using saddles with 1.5mm pitch teeth as opposed to the standard 3mm pitch teeth. The size of teeth was based upon the availability of tooling.
The results are given in Table 6.10.

**Table 6.10 ET20 with 1.5mm nitrided teeth**

| Specimen number | Permanent extension at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.081 | 200.5 | Bar pull out |
| 1b | 0.063 | 201.7 | Bar pull out |
| 1c | 0.063 | 206.2 | Bar pull out |

This configuration achieved the increased performance criteria required, in terms of slip and U.T.L. (required U.T.L. is 197kN). Smaller teeth and/or nitride hardness treatment obviously increased the coupler performance. The saddles remained, intact which was probably due to either the smaller tooth form, i.e. increased root area, lower load per tooth, or due to the nitride treatment, i.e. a less brittle saddle, or possibly a combination of the two. Figure 6.7 shows the difference in tooth damage between the standard and small teeth. The upper saddle is a standard unused one; the middle sample is a standard used saddle showing extensive deformation of the teeth; the lower picture shows a used saddle with small teeth with virtually no damage.

**Figure 6.7 Saddle Damage**



The next obvious step, with this configuration, was to reduce the bore of the tube which would reduce the outside diameter and hence the volume of material and the cost.

### 6.3.4 Pull Test on short tube, reduced bore ET20 with 1.5mm nitrided teeth saddles with 150 Nm torque

New tube internal dimensions were calculated at 27.5 mm using an iterative method incorporated into a visual basic computer program (see Appendix C). Wall thickness was set at 7.82 mm as previously calculated and the sample tube was machined from bar. The serrated saddles were 3.5mm in height and the test results are given in Table 6.11.

**Table 6.11 Reduced height serrated saddle results**

| Sample and specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.094 | 192.4 | Bar pull out |
| 1b | 0.100 | 192.2 | Bar pull out |
| 1c | 0.084 | 206.1 | Bar pull out |

The serrated saddle failure results were too inconsistent with only one out of three specimens achieving the desired failure load. These inconsistencies needed to be eliminated so it was decided to increase the bolt torque with a view to increasing the coupler failure performance.

### 6.3.5 Pull Test on Short, Reduced Bore ET20 with 6-M14 bolts torqued to 165 Nm, 3.5mm height saddle and 1.5mm pitch nitrided teeth.

The bolt torque was increased to 165Nm to see if the upper failure load could be increased. The results are given in Table 6.12

**Table 6.12 ET20 with 165 Nm bolt torque, 6-M14 bolts, 1.5mm teeth, 3.5mm saddle, short, small bore tube**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.081 | 211.9 | Ductile in bar |
| 1b | 0.100 | 213.3 | Ductile in bar |
| 1c | 0.088 | 208.8 | Bar pull out |

This configuration was the first to meet the peformance requirements and the serrated ET20 development could have stopped here, however, it was felt that there was scope to further reduce the saddle height, which would also enable a further reduction in tube inner diameter

### 6.3.6 Pull test for 6-165Nm M14 bolt smaller diameter, 160mm length ET20 couplers with 2.5mm height small teeth nitrided saddles

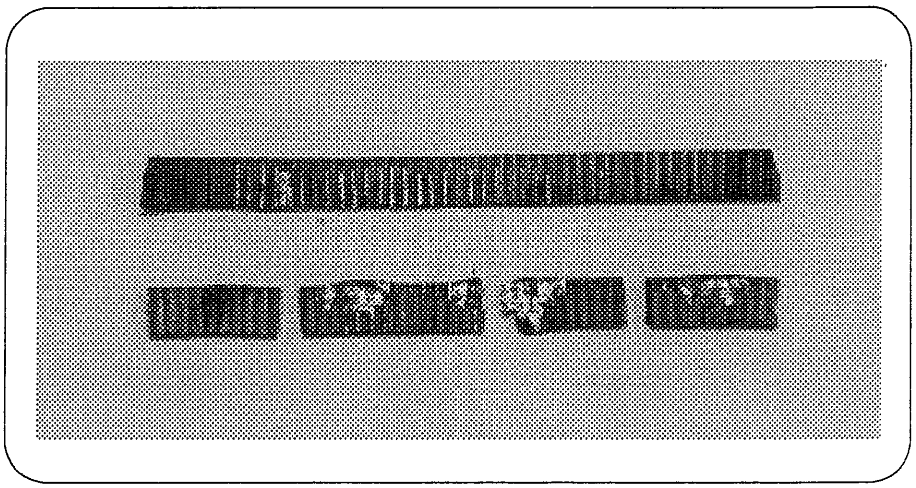
Each specimen had 6mm x 2.5mm saddles with 1.5mm pitch teeth. The results are given in Table 6.13.

**Table 6.13 Results for 2.5mm high, 1.5mm pitch nitrided teeth saddles**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.112 | 198.9 | Bar pull out |
| 1b | 0.081 | 194.8 | Bar pull out |
| 1c | 0.083 | 200.1 | Bar pull out |

For all three specimens the, saddles were broken and pulled out with the reinforcement. The performance of this arrangement was inconsistent between specimens. As reported in section 6.4, an abrasive saddle of section 6mm x 3mm had performed in a similar way, ie by fracturing and pulling out with the reinforcement. With this in mind it was decided to stay with the 6mm x 3.5mm serrated saddle and not test a 6mm x 3mm serrated saddle. Figure 6.8 contrasts the successful 3.5 mm high saddle with the failed 2.5mm high saddle.

**Figure 6.8 Small pitch saddles**



So the serrated configuration development had achieved the required performance criteria and cost savings could be determined in detail.

### 6.3.7 Pull Test for short tube ET20 with 6-M14 bolts, small teeth nitrided saddle assembled in alternative bolt order

All specimens tested to this point had been assembled in order from the centre two bolts outwards. The original specification states that the bolts can be assembled in any order. This means that any order of bolt tightening must produce the required performance.

This sample of tests was to determine the effect (if any) upon performance of an alternative order of bolt assembly. The bolts on each specimen were tightened in order from the outer bolts inwards. The results are given in Table 6.14

**Table 6.14 Results for alternative order of bolt assembly**

| Specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.175 | 195.1 | Bar pull out |
| 1b | 0.163 | 197.1 | Bar pull out |
| 1c | 0.156 | 191.7 | Bar pull out |

This order of tightening considerably reduces the coupler performance. It was decided to adopt a new specified tightening order of inner-bolts-outwards. A coupler that would perform, regardless of tightening order, would undoubtedly be more expensive.

### 6.3.8 Summary

A further 5 coupler configurations had been tested to produce these results, requiring 14 couplers to be tested, making 12 configurations and 34 tests in all to this point.

Figure 6.9 shows a summary of the results for the above tests which produced the first workable coupler as follows;

-Tube dimensions of 27.5 mm inside diameter, 7.82 mm wall thickness, 160 mm tube length.

-Six bolts, M14 size, torqued to 165 Nm.

-Saddles of 6 mm x 3.5 mm with 1.5 mm pitch teeth, nitride hardened.

This coupling produced a further reduction in tube material cost from the original 0.311 to 0.214 units. Saddle costs were reduced from 0.164 to 0.074 units, which included material, manufacturing and heat treatment. Table 6.15 shows the details of costs for this arrangement giving overall savings of 23.3%

**Figure 6.9 - Summary of ET20 serrated saddle development**

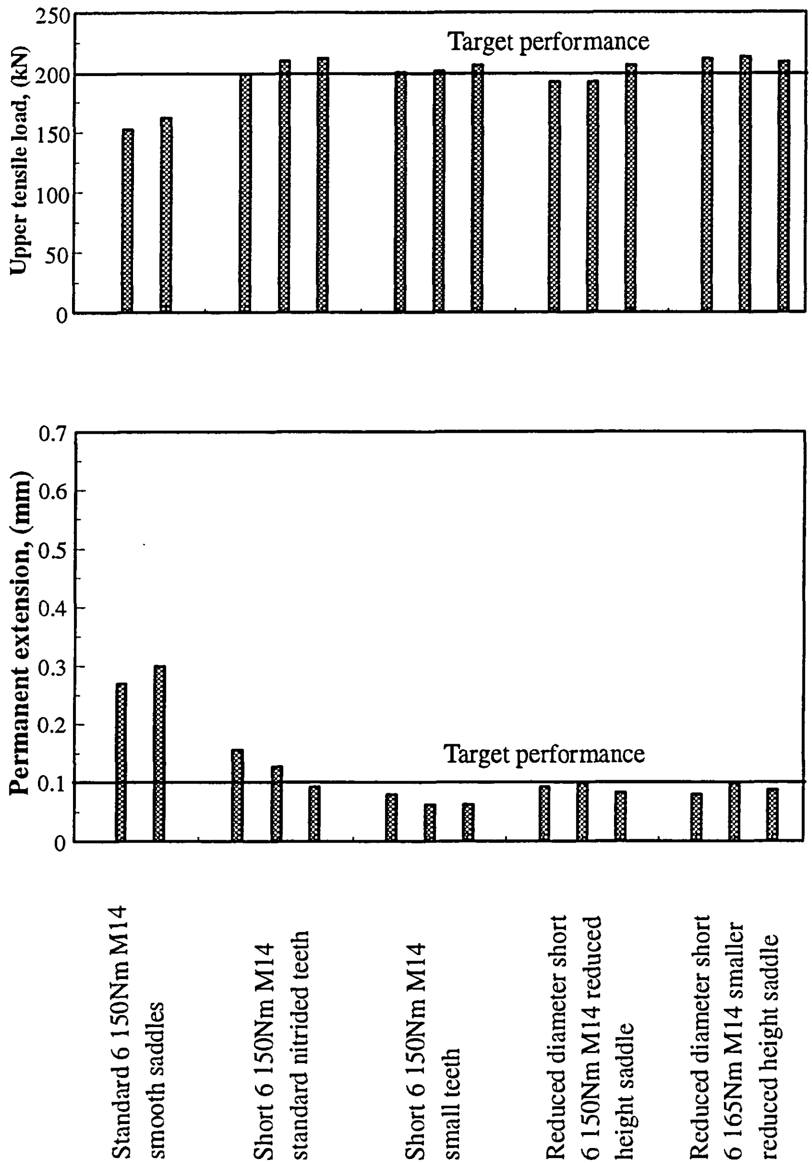


**Table 6.15 Cost Ratio Analysis for Developed ET 20 with 1.5mm Pitch Serrated Saddle**

| | | | | | |
|---|---|---|---|---|---|
| **TUBE COST** | | | **STAMP** | 500 | per HR |
| COST | 0.214 | | STAMP COST | 0.000 | |
| SCRAP | 0.016 | | SET-UP | 0.001 | |
| **TOTAL** | **0.230** | **Units** | **TOTAL** | **0.001** | **Units** |
| **BOLTS** | | | **DRILL & TAP** | 33 | per HR |
| 6 OFF M14 | 0.334 | | LABOUR COST | 0.031 | |
| TRANSPORT | 0.012 | | TOOL COST | 0.020 | |
| **TOTAL** | **0.345** | **Units** | **TOTAL** | **0.051** | **Units** |
| **SADDLE COST** | | | **WELD** | 40 | per HR |
| MATERIAL | 0.018 | | LABOUR COST | 0.035 | |
| MANFACTURING | 0.023 | | CONSUMABLES | 0.003 | |
| HEAT TREAMEN | 0.024 | | | | |
| TRANSPORT | 0.009 | | **TOTAL** | **0.038** | **Units** |
| **TOTAL** | **0.074** | **Units** | **ASSEMBLY & PACK** | 70 | per HR |
| **SAW** | 100 | per HR | ASSEMBLY LABOUR | 0.015 | **Units** |
| | | | PACK LABOUR | 0.002 | |
| CUT COST | | | | | |
| LOAD COST | 0.001 | | PACK COST | 0.009 | |
| BLADE COST | 0.001 | | | | |
| | | | **TOTAL** | **0.026** | **Units** |
| **TOTAL** | **0.002** | **Units** | | | |
| | | | **TOTAL COST** | **0.767** | **Units** |

## 6.4 Modifications to Abrasive Coated Saddles and Tubes

### 6.4.1 Pull Test on Standard ET20 with abrasive coated plain saddles

As a result of the encouraging results from the smooth saddle ET20 coupler tests it was decided to further test plain saddle couplers but this time with a friction coating applied to the saddle/bar and possibly the saddle/tube interfaces.

Metal sprayed coatings offered one solution. There are three types of metal spraying processes, flame spraying, arc spraying and plasma spraying, each of which finds regular use in industrial applications (25). Flame spraying involves spraying metal powders by feeding the powder into a flame which is fuelled by a combustible gas and compressed air. The air stream propels the powder, which consists of the abrasive particles and a matrix, onto the component. Only the matrix is melted in the flame so the abrasive particles are set in the cooling matrix upon the component. The process is very inaccurate and a suitable dispersion of the abrasive cannot be guaranteed. The arc spraying process is more accurate as the molten material is fed through a nozzle. The abrasive and matrix are fed as wire into an electric arc which melts them. The molten material is again propelled towards the workpiece in a stream of compressed air. Again only the matrix is fully molten but the higher velocity nozzle ensures a higher bond strength than achieved by flame spraying. Plasma spraying is as accurate as arc spraying but involves much higher stream velocity and flame temperature, hence, higher bond strengths. The material, in powder form, is fed into a hydrogen fuelled flame and propelled towards the workpiece in a high velocity nitrogen stream. Both the matrix and the abrasive particles are melted which leads to the high bond strength.

The first friction coating tried was a plasma sprayed tungsten carbide coating available from a specialist supplier. Two samples were tested, each one with a different saddle substrate material. One substrate was the standard saddle material, 709M40 (a key steel), and the other was a very soft low grade alloy steel. It was felt that the reinforcement bar ribs may penetrate into the softer steel more and effectively create a keying action between the bar and the saddles. The test results from these two samples were very encouraging and are given in Table 6.16.

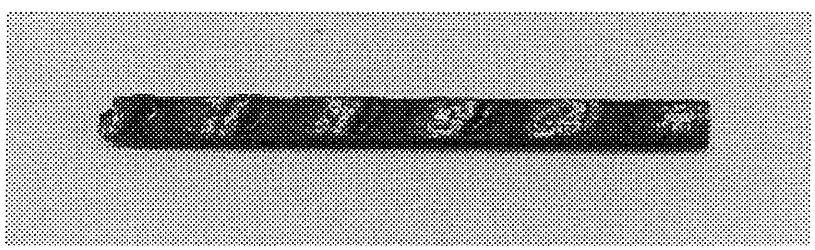
**Table 6.16 Results for plasma sprayed tungsten carbide coated mild and key steel saddle in standard ET20 couplers**

| Specimen and sample number | Saddle material | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | Key steel | 0.050 | 179.3 | Bar pull |
| 1b | Key steel | 0.069 | 190.6 | Bar pull |
| 1c | Key steel | 0.063 | 186.6 | Bar pull |
| 2a | Mild steel | 0.078 | 200.3 | Bar pull |
| 2b | Mild steel | 0.069 | 196.3 | Bar pull |
| 2c | Mild steel | 0.075 | 191.6 | Bar pull |

The key steel coated saddles produced a marginally better slip result than the mild steel saddles however the U.T.L. was below the required value of 197kN. The mild steel saddles performed better in terms of U.T.L. , and still satisfied the slip criteria. all subsequent tests were conducted using mild steel saddles.

Figure 6.10 shows a used saddle made of key steel and coated with tungsten carbide, clearly demonstrating the penetration of the reinforcement bar ribs.

**Figure 6.10 Coated Key Steel Saddle After Use**



### 6.4.2 Pull Test on Shorter tube ET20 with 6-M14 bolts and manually abrasive bauxite coated saddles

The next step was to follow the approach used with serrated saddles and test a shorter tube ET20 coupler using these type of saddles with 6-M14 bolts torqued to 150 Nm.

On the basis of the previous abrasive saddle results, Table 3.13, it was decided to try abrasive saddles that were produced manually by simply fixing loose abrasive grit to the saddle interfaces. Three different size grades of bauxite, which is a hard wearing material, were obtained. The bauxite was fixed to each saddle with epoxy resin. One sample of each size of grit was produced and the results are given in Table 6.17.

**Table 6.17 Manually applied bauxite coated saddle results**

| Specimen number | Grit size (mm) | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | 0.004 | 0.238 | 167.8 | Bar pull out |
| 1b | 0.004 | 0.206 | 143.0 | Bar pull out |
| 1c | 0.004 | 0.250 | 166.5 | Bar pull out |
| 2a | 0.006 | 0.750 | 147.6 | Bar pull out |
| 2b | 0.006 | 0.638 | 145.4 | Bar pull out |
| 3a | 0.008 | 0.906 | 145.8 | Bar pull out |
| 3b | 0.008 | 0.825 | 142.3 | Bar pull out |

Even though the smallest grit was the best, overall, these results were extremely poor indicating that the bond strength between the coating and the substrate metal is a key element in the performance. So the sprayed coatings were investigated further.

### 6.4.3 Pull Test on short tube ET20 with 6-M14 bolts and various sprayed saddle coatings

Three different coating samples were tested. The first was a flame sprayed, alumina based, ceramic coating, the second was an arc sprayed tungsten carbide coating and the third was a plasma sprayed tungsten carbide coating. The results are given in Table 6.18

**Table 6.18 Results for Various saddle coatings**

| Specimen/sample number | Coating description | Slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | Flame sprayed Al2O3 | 0.081 | 182.5 | Bar pull out |
| 1b | Flame sprayed Al2O3 | 0.094 | 199.2 | Bar pull out |
| 1c | Flame sprayed Al2O3 | 0.106 | 196.2 | Bar pull out |
| | | | | Bar pull out |
| 2a | Arc sprayed WC | 0.100 | 189.4 | Bar pull out |
| 2b | Arc sprayed WC | 0.088 | 195.4 | Bar pull out |
| 2c | Arc sprayed WC | 0.100 | 186.5 | Bar pull out |
| | | | | Bar pull out |
| 3a | Plasma sprayed WC | 0.081 | 202.5 | Bar pull out |
| 3b | Plasma sprayed WC | 0.063 | 201.8 | Bar pull out |
| 3c | Plasma sprayed WC | 0.100 | 197.1 | Bar pull out |

The plasma sprayed carbide coated saddles performed better than the other two coatings in terms of both slip and U.T.L.. The next step was to investigate reduced root area saddles which would enable the use of reduced diameter sleeves.

### 6.4.4 Pull Test on short tube, reduced bore ET20 with reduced height abrasive saddles

The new tube internal dimensions were calculated as previously using an iterative method incorporated into a visual basic computer program (see Appendix C). The mild steel abrasive saddles had a plasma sprayed tungsten carbide coating and were 3mm in height reduced from 6mm. The test results are given in Table 6.19.

**Table 6.19 Reduced height abrasive saddle results**

| Sample and specimen number | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| 1a | 0.100 | 160.3 | Bar pull out |
| 1b | 0.100 | 168.9 | Bar pull out |
| 1c | 0.084 | 173.0 | Bar pull out |

The abrasive saddle produced poor ultimate load failure results, the saddle having failed with a tensile failure so it was decided to try a further sample with increased height saddles and increased bolt torque.

### 6.4.5 Pull test for 6-165Nm M14 bolt smaller diameter, 160mm ET20 couplers with various coated 3.5mm height mild and key steel saddles

Further to the previous abrasive tests, specimens were produced with saddle sections 3.5mm x 6mm. There were two samples, one with mild and one with key steel saddles, of three specimens where each specimen had a different abrasive coating. As with the serrated development the bolt torque was increased to 165Nm and the results are given in Table 6.20.

**Table 6.20 Results for various abrasive coated saddles**

| Specimen number | Saddle material | Coating description | Coating thickness (") | Slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|---|---|
| 1a | Key steel | Arc sprayed FeCrB | 0.004 | 0.082 | 195.8 | Bar pull out |
| 1b | Key steel | Arc sprayed FECRAL | 0.004 | 0.078 | 174.8 | Bar pull out |
| 1c | Key steel | Plasma sprayed WC/Co | 0.004 | 0.073 | 208.4 | Bar pull out |
| 1d | Mild steel | Arc sprayed FeCrB | 0.004 | 0.056 | 178.5 | Bar pull out |
| 1e | Mild steel | Arc sprayed FECRAL | 0.004 | 0.073 | 176.7 | Bar pull out |
| 1f | Mild steel | Plasma sprayed WC/Co | 0.004 | 0.100 | 172.8 | Bar pull out |

Only configuration 1c met the increased performance requirements which was the key steel plasma sprayed tungsten carbide coating in a cobalt matrix. The mild steel saddle version of this arrangement had a poor failure performance which conflicted with earlier results where mild steel coated saddles had performed better in standard ET20 couplers. Obviously the reduction in saddle height required a higher strength saddle.

A couple of plasma sprayed tungsten carbide confirmation tests were also carried out to strengthen the single result. These results are shown in Table 6.21.

**Table 6.21 Confirmation of plasma sprayed tungsten carbide results**

| Specimen number | Saddle material | Coating description | Coating thickness (") | Slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|---|---|
| 1g | Key steel | Plasma sprayed WC/Co | 0.004 | 0.06 | 200.4 | Bar pull out |
| 1h | Key steel | Plasma sprayed WC/Co | 0.004 | 0.08 | 201.9 | Bar pull out |

Although marginally lower UTL values were obtained, the performance was confirmed.

It was interesting to speculate as to why the tungsten carbide was the only one to perform. Saddle material aside, plasma sprayed coatings produce the highest coating-to-substrate bond strength which was the obvious explanation. However, the plasma spraying process has a high initial capital cost in comparison to arc spraying so an arc sprayed coating with a similar bond strength to that produced by plasma spraying was considered to be well worth investigating.

Arc sprayed titanium carbide has a similar bond strength to plasma sprayed tungsten carbide so a sample of these was tested. The same bolt and tube configuration as the tests in Table 6.21 was used and the results are given in Table 6.22.

**Table 6.22 Results of titanium carbide coated saddle tests**

| Specimen number | Coating thickness (") | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | 0.004 | 0.07 | 191.9 | Bar pull out |
| 1b | 0.004 | 0.085 | 190.7 | Bar pull out |
| 1c | 0.004 | 0.068 | 193.3 | Bar pull out |

The failure performance of this configuration was unacceptable. It seemed that a combination of both process and coating was responsible for the performance and not the bond strength. Alternative plasma sprayed coatings had not been investigated to this point and may have been worthwhile because the tungsten carbide powders that had been used so far were one of the most expensive on the market. However it was possible to obtain a lower grade, and lower cost, tungsten carbide powder. This consisted of reconstituted cutting tools which are ground into suitable powder form. A sample of two specimens was tested with this lower grade coating. The results are tabulated in Table 6.23.

**Table 6.23 Results of lower grade tungsten carbide coated saddle tests**

| Specimen number | Coating thickness (") | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | 0.004 | 0.0975 | 201.8 | Bar pull out |
| 1b | 0.004 | 0.0725 | 202.5 | Bar pull out |

This configuration met the new performance requirements and as it stands, also offered cost savings. Suppliers also suggested that reducing the coating thickness might be beneficial, so a sample of three specimens was tested with lower grade tungsten carbide of 0.001" thickness. Unfortunately they did not perform satisfactorily as indicated by the results in table 6.24.

**Table 6.24 Results for thin low grade tungsten carbide coated saddles**

| Specimen number | Coating thickness (") | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 1a | 0.001 | 0.045 | 191.3 | Bar pull out |
| 1b | 0.001 | 0.120 | 178.5 | Bar pull out |
| 1c | 0.001 | 0.135 | 178.2 | Bar pull out |

### 6.4.6 Summary

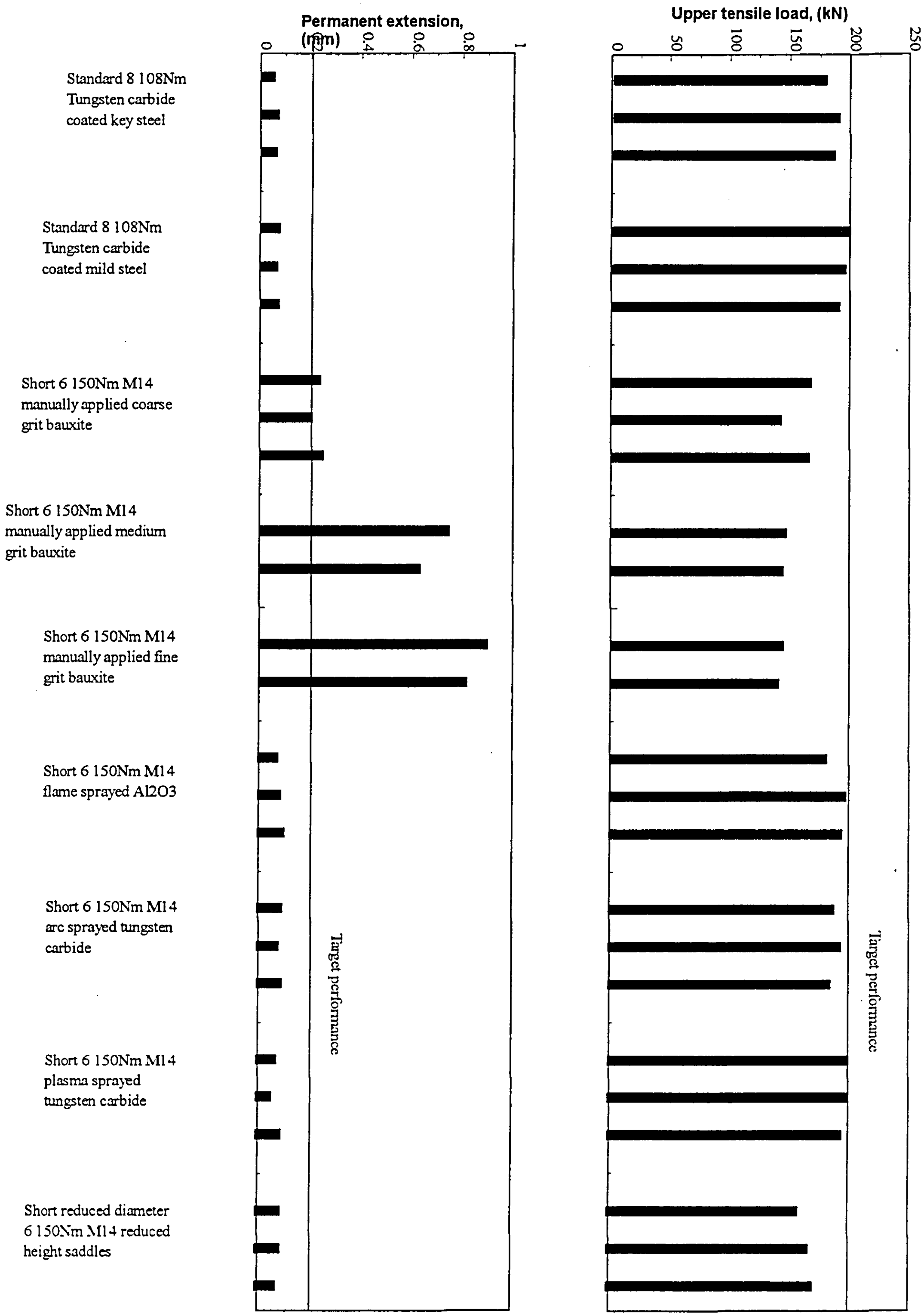
Figures 6.11 and 12 summarise the results for couplers with abrasive coated saddles. Up to this point 19 configurations had been tested, representing some 41 samples. Two suitable couplers have been identified, each of the same basic geometry , one using saddles coated with high grade tungsten carbide, the other a lower grade of the same material. Coating thickness in both cases was 0.004" and the coatings were applied using the plasma spraying technique; the following geometry was used;

-Tube dimensions of 27.5 mm inside diameter, 7.82 mm wall thickness, 160 mm tube length.

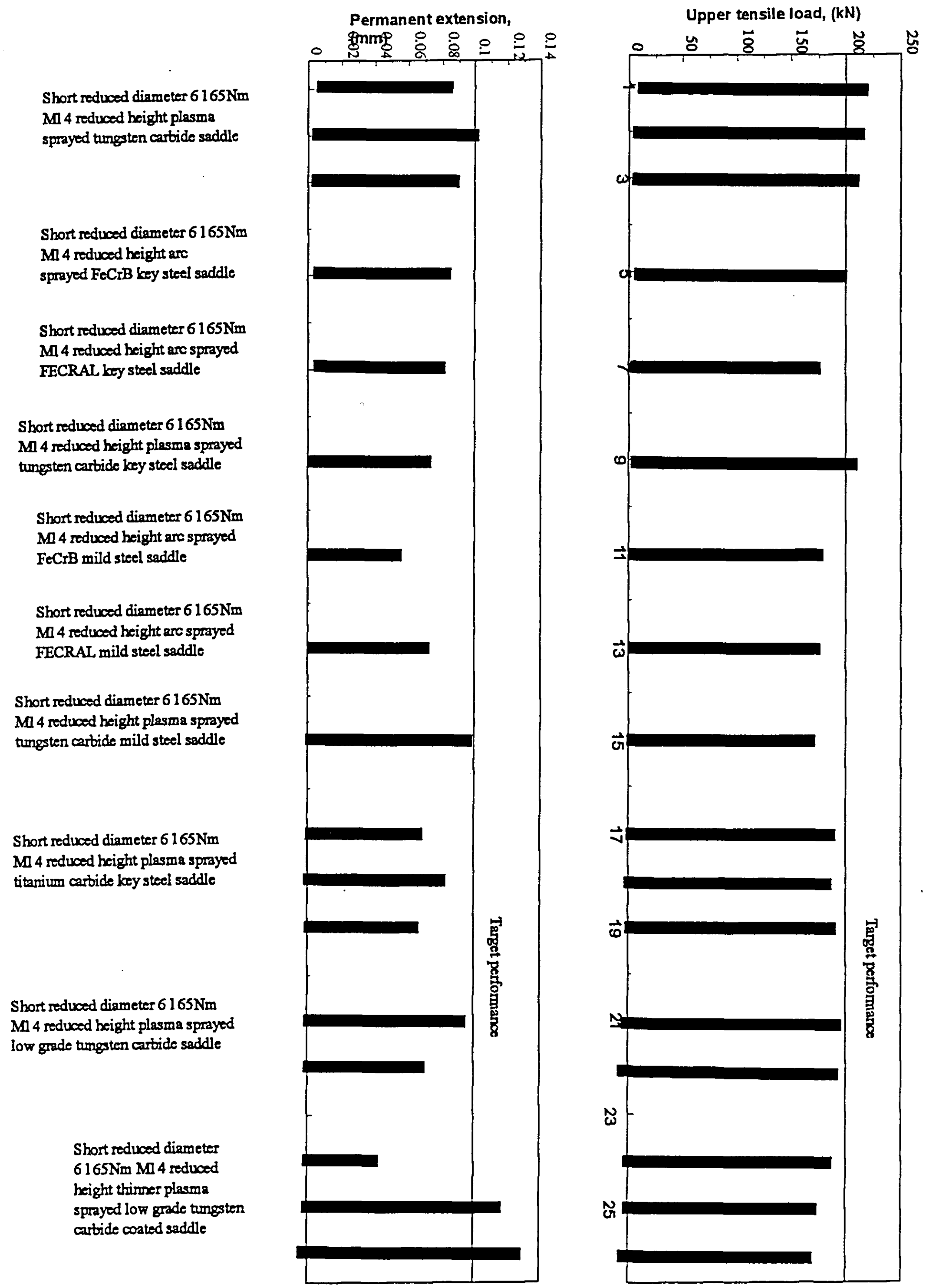
-Six bolts, M14 size, torqued to 165 Nm.

-Saddles of 6 mm x 3.5 mm.

## Figure 6.11 Summary of Results



# Figure 6.12 Summary of Results



The cost of coating the saddles was calculated on the basis of a per kg figure for the coating material obtained from the supplier. For the high grade coating this was 10.67 units/kg and for the low grade material a figure of 3.9 units/kg was used. Using these figures reduced the saddle cost from an original value of 0.162 units to 0.1 and 0.048 respectively, and compares well with the new serrated saddle cost of 0.074 units. It was also felt possible that the coated saddles could be even cheaper by the use of an in-house spraying machine. Thus at this stage in the project, the two coated saddles offered savings of 25.9% and 20.7% respectively. The cost details are shown in tables 6.25 and 6.26.

**Table 6.25 Cost Ratio Analysis of Low Quality Coated Saddles**

| | | | | | |
|---|---|---|---|---|---|
| **TUBE COST** | | | **STAMP** | 500 | per HR |
| COST | 0.214 | | STAMP COST | 0.000 | |
| SCRAP | 0.016 | | SET-UP | 0.001 | |
| **TOTAL** | **0.230** | **Units** | **TOTAL** | **0.001** | **Units** |
| **BOLTS** | | | **DRILL & TAP** | 44 | per HR |
| 6 OFF M14 | 0.334 | | LABOUR COST | 0.031 | |
| TRANSPORT | 0.012 | | TOOL COST | 0.020 | |
| **TOTAL** | **0.380** | **Units** | **TOTAL** | **0.051** | **Units** |
| **SADDLE COST** | | | **WELD** | 40 | per HR |
| MATERIAL | 0.018 | | LABOUR COST | 0.035 | |
| | | | CONSUMABLES | 0.003 | |
| COATING | 0.030 | | | | |
| | | | **TOTAL** | **0.038** | **Units** |
| **TOTAL** | **0.048** | **Units** | **ASSEMBLY & PACK** | 93 | per HR |
| SAW | 100 | per HR | ASSEMBLY LABOUR | 0.015 | **Units** |
| | | | PACK LABOUR | 0.002 | |
| CUT COST | | | | | |
| LOAD COST | 0.001 | | PACK COST | 0.009 | |
| BLADE COST | 0.001 | | | | |
| | | | **TOTAL** | **0.026** | **Units** |
| **TOTAL** | **0.002** | **Units** | | | |
| | | | **TOTAL COST** | **0.741** | **Units** |

**Table 6.26 Cost Ratio Analysis for High Quality Coated Saddles**

| | | | | | |
|---|---|---|---|---|---|
| **TUBE COST** | | | **STAMP** | 500 | per HR |
| COST | 0.214 | | STAMP COST | 0.000 | |
| SCRAP | 0.016 | | SET-UP | 0.001 | |
| **TOTAL** | **0.230** | **Units** | **TOTAL** | **0.001** | **Units** |
| **BOLTS** | | | **DRILL & TAP** | 44 | per HR |
| 6 OFF M14 | 0.334 | | LABOUR COST | 0.031 | |
| TRANSPORT | 0.012 | | TOOL COST | 0.020 | |
| **TOTAL** | **0.380** | **Units** | **TOTAL** | **0.051** | **Units** |
| **SADDLE COST** | | | **WELD** | 40 | per HR |
| MATERIAL | 0.018 | | LABOUR COST | 0.035 | |
| | | | CONSUMABLES | 0.003 | |
| COATING | 0.082 | | | | |
| | | | **TOTAL** | **0.038** | **Units** |
| **TOTAL** | **0.100** | **Units** | **ASSEMBLY & PACK** | 93 | per HR |
| **SAW** | 100 | per HR | ASSEMBLY LABOUR | 0.015 | **Units** |
| | | | PACK LABOUR | 0.002 | |
| CUT COST | | | | | |
| LOAD COST | 0.001 | | PACK COST | 0.009 | |
| BLADE COST | 0.001 | | | | |
| | | | **TOTAL** | **0.026** | **Units** |
| **TOTAL** | **0.002** | **Units** | | | |
| | | | **TOTAL COST** | **0.793** | **Units** |

Although it was decided to proceed to full range tests there was some concern as to the viability and customer acceptance of coated couplers. To provide some basis for decisions for the company, a SWOT analysis was undertaken and are summarised in tables 27 and 28.

**Table 6.27 Relative Advantages for Serrated saddles**

| Advantages | Disadvantages |
|---|---|
| Established process | Higher finished stock levels |
| Customer accepted | Reaction time |
| Machine breakdown cover | Low patentability |
| Easier to obtain product approval | Floor space |

Generally the advantages were related to the saddles being an established component and part of the product but the use of two machines also provided a breakdown cover. The disadvantages were more varied. The tooth cutting and heat treatment processes demand a long lead time (2 weeks) and because of this poor reaction time a certain amount of finished stock needs to be maintained. Also because this mechanism is already part of the product patent the possibility of strengthening the patent is slim.

**Table 6.28 Relative Advantages for Coated Saddles**

| Advantages | Disadvantages |
|---|---|
| Reaction time | Machine breakdown |
| Patentability | Quality control |
| Reduced finished stock | Customer acceptance |
| Floor space | Harder to obtain approval |
| Greater capacity | |

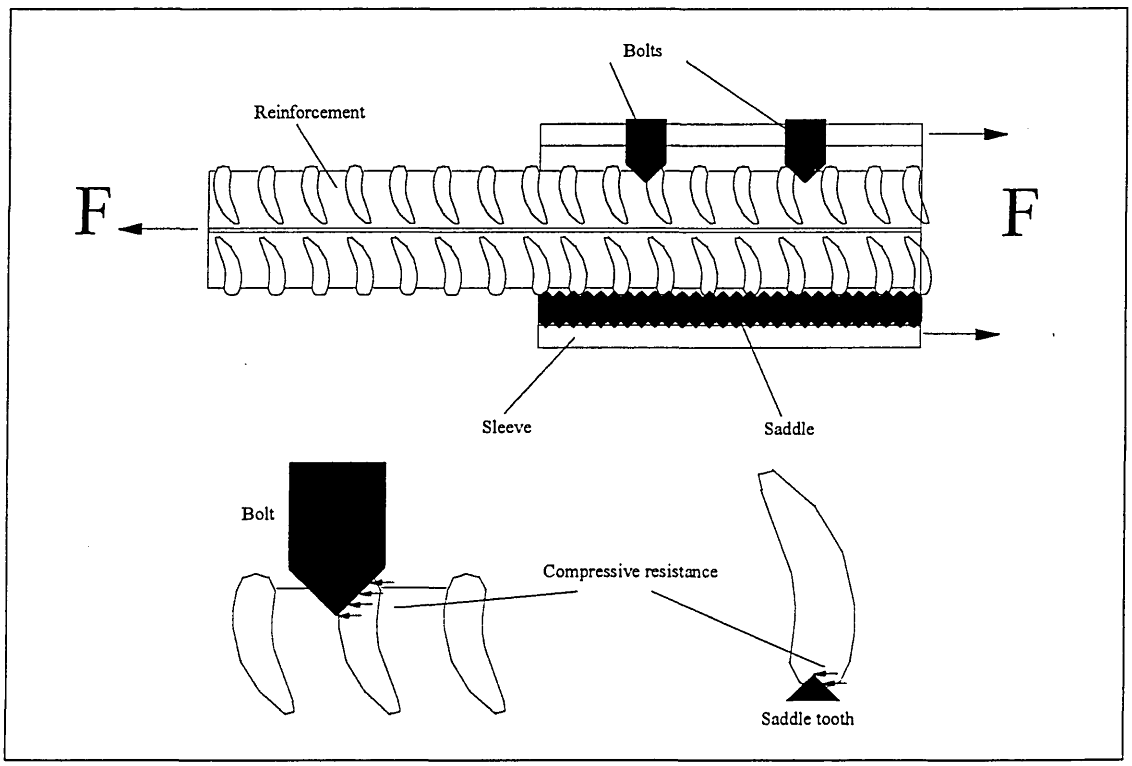
# CHAPTER 7 EXTENSION OF RESULTS TO ALL SIZES

## 7.1 Principles of Parameterisation

The key elements involved in the ET20 developed couplers were the reduction in numbers of bolts, increase in bolt size, increase in bolt torque and reduction in saddle root area. In terms of serrations the pitch was reduced, with smaller teeth in one case, and minute abrasions in the other.

When the coupler is loaded the resistance to the bar being pulled out of the sleeve is a combination of the bolt penetration, the bolt clamping force and the saddle penetration. All bar penetrating elements provide an area which resists compressive loading in the bar itself. In Figure 7.1 a half coupler is shown sectioned with the bar under tensile load, F. The areas of compressive loading in the bar are shown for a bolt and a saddle tooth.

**Figure 7.1 Areas of compressive loading in bar**




The smooth saddle tests that were performed illustrate that the bolts contribute most to the coupler performance. As the bolts will always penetrate the bar regardless of the rib pattern it was felt reasonable to use the bolt relation to develop the rest of the range. The amount of saddle penetration will vary with rib pattern and is difficult to quantify anyway.

Earlier work in this thesis showed that for a given torque a bolt cone will penetrate a bar to a certain depth relative to the nominal diameter. The projected area of penetration (i.e. area seen along the bar axis) can be calculated easily as the cone profile is known. Multiplying by the number of bolts gives the total coupler projected area. Stress is force per unit area, so if we divide the tensile load by the total coupler projected penetration area we can determine the compressive stress acting upon the bolts along the bar axis. We will call this the 'ploughing stress' as, if sufficiently high, the bolts will 'plough' through the bar leaving a 'V' shaped channel in the bar. Alternatively the bolt will probably shear or the cone will collapse.

The developed ET20 coupler satisfied the specified performance requirements and the ploughing stress is calculated for the developed ET20, assuming a linear relationship between load and ploughing stress, was extrapolated to the rest of the coupler range. Using this value the required projected penetration area for each size of coupler was determined and from this the numbers of bolts and applied torque selected to suit the limitations of the bolt and achieve minimum cost criteria where possible.

Taking the developed ET20 :

Yield load = 157.1kN

Bolt torque = 165Nm

No. of bolts = 6

Bolt downward force = 73.105kN (ref 22)

Depth of bolt penetration mm = 2.922094mm + (0.058014mm/kN.xkN)
= 7.16mm

Area of projected penetration = 25.66mm$^2$

Coupler projected penetration area = 153.9mm$^2$

Hence nominal ploughing stress = 1021N/mm$^2$

This is higher than the reinforcement bar compressive yield stress, but the real failure stress is likely to be lower since some load is taken by the saddle. However this was felt to be a suitable parameter on which to base the design.

Every coupler in the range was assumed to require a similar nominal ploughing stress value. The yield load for each size was known therefore the projected penetration area for each coupler was calculated. Table 7.1 gives the respective areas required for each coupler in the range.

**Table 7.1 - Projected penetration areas required by coupler range based on developed ET20**

| Coupler type | Yield load (kN) | Ploughing stress required (N/mm²) | Required projected penetration area (mm²) |
|---|---|---|---|
| ET 8 | 25.1 | 1021 | 24.62 |
| ET 10 | 39.3 | 1021 | 38.46 |
| ET 12 | 56.5 | 1021 | 55.39 |
| ET 16 | 100.5 | 1021 | 98.46 |
| ET 25 | 245.4 | 1021 | 240.39 |
| ET 28 | 307.9 | 1021 | 301.54 |
| ET 32 | 402.1 | 1021 | 393.85 |
| ET 40 | 628.3 | 1021 | 615.40 |
| ET 50 | 981.7 | 1021 | 961.56 |

Each size of coupler was examined to see where numbers of bolts could be reduced and bolt diameters increased. When an alternative configuration was selected upon, the required bolt torque was determined based upon the required ploughing area. Table 7.2 illustrates the first prototype range and compares the number of bolts with the previous range. In all but couplers ET10, ET40 and ET50 the number of bolts was reduced. The existing ET40 and ET50 range did not work to existing criteria at the outset of the project so the standard number of bolts was retained in the first instant.

**Table 7.2 - Bolts for proposed prototype range**

| Coupler type | Present no. of bolts | Present bolt size | Proposed no. of bolts | Proposed bolt size |
|---|---|---|---|---|
| ET8 | 4 | M10 | 2 | M10 |
| ET10 | 4 | M10 | 4 | M10 |
| ET12 | 6 | M10 | 4 | M12 |
| ET16 | 6 | M12 | 4 | M14 |
| ET25 | 8 | M16 | 8 | M14 |
| ET28 | 10 | M16 | 8 | M16 |
| ET32 | 10 | M16 | 8 | M16 |
| ET40 | 10 | M20 | 10 | M20 |
| ET50 | 14 | M20 | 14 | M20 |

| Coupler type | Required projected area ($mm^2$) | Required bolt torque (Nm) |
|---|---|---|
| ET8 | 24.62 | 55.66 |
| ET10 | 38.46 | 43.12 |
| ET12 | 55.39 | 41.24 |
| ET16 | 98.46 | 158.37 |
| ET25 | 240.39 | 194.07 |
| ET28 | 301.54 | 276.60 |
| ET32 | 393.85 | 360.49 |
| ET40 | 615.40 | 566.58 |
| ET50 | 961.56 | 632.99 |

At this stage the subject of design for manufacture, particularly commonality of components, was considered and any two like sized bolts that had similar bolt torque were replaced by one common bolt. A 56Nm M10 bolt was selected for both the ET8 and ET10 couplers and a 165Nm M14 bolt was used for both the ET16 and ET20 couplers.

The developed ET20 demonstrated that smaller teeth in the saddle improved slip performance. As a consequence all of the serrated prototype range were given reduced pitch teeth. Couplers ET8 to ET32 were given 1.5mm pitch teeth and couplers ET40 and ET50 were given 3mm pitch teeth, based upon availability of tooling.

The saddle root area for each coupler was calculated by a similar method. The developed ET20 saddle has a particular root area. It was assumed, for comparative analysis only, that this area was subjected to the coupler yield load and hence a stress could be calculated for the developed ET20 saddle. This value of stress could then be extrapolated to the rest of the range to determine the new saddle thicknesses, see Table 7.3.

Taking the developed ET20 :

$$\text{Yield load} = 157.1\text{kN}$$

$$\begin{aligned}\text{Root area} &= w.(h - 2.d) \\ &= 12\text{mm}^2\end{aligned}$$

$$\text{Root tensile stress} = 13.1\text{kN/mm}^2$$

**Table7.3 - Proposed saddle dimensions for first prototype range of couplers**

| Coupler type | Yield load (kN) | Required root stress (N/mm²) | Required root area (mm²) | Saddle width (mm) | Associated root area height (mm) | Tooth depth (mm) | Saddle height (mm) |
|---|---|---|---|---|---|---|---|
| ET8 | 25.10 | 13091.67 | 1.92 | 4.00 | 0.48 | 0.75 | 2.0 |
| ET10 | 39.30 | 13091.67 | 3.00 | 4.00 | 0.75 | 0.75 | 2.3 |
| ET12 | 56.50 | 13091.67 | 4.32 | 4.00 | 1.08 | 0.75 | 2.6 |
| ET16 | 100.50 | 13091.67 | 7.68 | 4.00 | 1.92 | 0.75 | 3.4 |
| ET25 | 245.50 | 13091.67 | 18.75 | 6.00 | 3.13 | 0.75 | 4.6 |
| ET28 | 308.00 | 13091.67 | 23.53 | 6.00 | 3.92 | 0.75 | 5.4 |
| ET32 | 402.00 | 13091.67 | 30.71 | 8.00 | 3.84 | 0.75 | 5.4 |
| ET40 | 628.00 | 13091.67 | 47.97 | 10.00 | 4.80 | 1.50 | 7.8 |
| ET50 | 981.50 | 13091.67 | 74.97 | 12.00 | 6.25 | 1.50 | 9.2 |

Again saddles of similar size were replaced by one common size. ET8 to ET16 saddles would all be 4mm x 3mm in cross-section. The above table suggests that for the ET16, a saddle with depth greater than 3mm would be required. First an ET16 with the 4mm x 3mm saddle was to be tried. If that failed then the developed ET20 saddle, of size 6mm x 3.5mm, would be tried.

All of these saddle dimensions were also adopted for the abrasive prototype range, following the principles used on the ET20.

All of the new prototypes would require a revised sleeve internal diameter, which was calculated using the bespoke program in Appendix C.

The new tube wall thicknesses were calculated using the bolt hole stress for the present range ET32 coupler, as reported in chapter 4 for tube strip results and analysis. Table 7.4 gives the proposed tube wall thicknesses.

**Table 7.4 - Tube thicknesses of proposed prototype couplers**

| Coupler type | Proposed bolt torque (Nm) | olt size | Associated downward force (kN) | Required tube thickness (mm) |
|---|---|---|---|---|
| ET8 | 56 | M 10 | 34.60 | 5.18 |
| ET10 | 56 | M 10 | 34.60 | 5.18 |
| ET12 | 75 | M 12 | 34.60 | 4.32 |
| ET16 | 165 | M 14 | 73.09 | 7.82 |
| ET20 | 165 | M 14 | 73.09 | 7.82 |
| ET25 | 195 | M 14 | 86.38 | 9.24 |
| ET28 | 277 | M 16 | 108.34 | 10.14 |
| ET32 | 363 | M 16 | 141.98 | 13.29 |
| ET40 | 567 | M 20 | 177.42 | 13.29 |
| ET50 | 633 | M 20 | 198.07 | 14.83 |

With the required tube thickness, minimum tube inner diameter and tube manufacturers tolerances known it was possible to determine the new tube outer diameter. The minimum tolerance tube inner diameter must coincide with the minimum required inner diameter. This worst case dimension was used to calculate the actual size of tube that would be supplied.

i.e. Standard tube O/D = Min. I/D + (2 x max. tolerance thickness) + O/D tolerance Eq.7.1

Standard tube I/D = Standard O/D - (2 x standard thickness) Eq.7.2

Table 7.5 gives the proposed tube dimensions and the associated tolerances, which were available from manufacturers to those dimensions provided minimal annual quantities are purchased.

**Table 7.5 - Dimensions of proposed prototype tube range**

| Coupler type | Tube thickness (mm) | O/D tolerance (mm) | Thickness tolerance (%) | Minimum I/D required (mm) | Standard O/D required to meet this (mm) | Standard size tube I/D (mm) |
|---|---|---|---|---|---|---|
| ET8 | 5.18 | 0.08 | 7.5 | 14.84 | 26.06 | 15.69 |
| ET10 | 5.18 | 0.15 | 7.5 | 16.80 | 28.09 | 17.73 |
| ET12 | 5.18 | 0.15 | 7.5 | 17.78 | 29.08 | 18.71 |
| ET16 | 7.82 | 0.40 | 7.5 | 22.72 | 39.94 | 24.30 |
| ET20 | 7.82 | 0.50 | 7.5 | 27.49 | 44.80 | 29.16 |
| ET25 | 9.24 | 0.50 | 7.5 | 34.70 | 55.07 | 36.58 |
| ET28 | 10.14 | 0.50 | 7.5 | 39.10 | 61.41 | 41.12 |
| ET32 | 13.29 | 0.60 | 7.5 | 43.76 | 72.94 | 46.35 |
| ET40 | 13.29 | 0.60 | 7.5 | 55.67 | 84.84 | 58.26 |
| ET50 | 14.83 | 0.75 | 7.5 | 68.46 | 101.10 | 71.43 |

Where the number of bolts had been reduced the overall coupler length was reduced accordingly by the number of bolt pitches. The present and proposed prototype coupler lengths are given in Table 7.6.

**Table 7.6 Present and proposed prototype coupler lengths**

| Coupler type | Present coupler length (mm) | Prototype coupler length (mm) |
|---|---|---|
| ET8 | 100 | 60 |
| ET10 | 100 | 100 |
| ET12 | 140 | 100 |
| ET16 | 160 | 116 |
| ET20 | 204 | 160 |
| ET25 | 258 | 228 |
| ET28 | 312 | 258 |
| ET32 | 312 | 258 |
| ET40 | 356 | 356 |
| ET50 | 498 | 498 |

At this stage the project had identified three similar priced, successful, alternative ET20 couplers. It was decided to extrapolate the ET20 development principles to the rest of the coupler range for each type of saddle configuration to see how the overall range cost savings compared against each other.

## 7.2 Projected Cost Savings for Proposed Prototype Range.

The bolt and tube configurations were to be the same for both the high and low cost abrasive and serrated saddle range. For each coupler size a spreadsheet was used to examine the cost of various configurations that met the required ploughing area. The removal of bolts and change of bolt size was done intuitively where applicable.

The same saddle dimensions that were derived for the serrated coupler range were also adopted for the two types of coatings for the abrasive saddle range. The cost savings for all of these ranges are given in Table 7.7, 7.8 and 7.9..

**Table 7.7 Projected Savings For Serrated Saddle Configuration**

| Serrated coupler type | Percentage cost saving (%) | 1995 coupler sales | Potential cost saving per year (units) |
|---|---|---|---|
| ET8 | 45.18 | 9054 | 1811.54 |
| ET10 | 10.64 | 3450 | 166.78 |
| ET12 | 36.88 | 7325 | 1711.78 |
| ET16 | 24.48 | 15050 | 2852.94 |
| ET20 | 23.32 | 27000 | 6296.40 |
| ET25 | 16.05 | 12700 | 2730.12 |
| ET28 | 26.96 | 928 | 454.39 |
| ET32 | 19.47 | 14050 | 5565.14 |
| ET40 | -0.58 | 6100 | -91.84 |
| ET50 | 5.30 | 456 | 116.69 |
| | | total | **21613.92** |

**Table 7.8 Projected Savings for High Grade Tungsten Carbide Coated Saddle Configuration**

| Abrasive coupler type | Percentage cost saving (%) | 1995 coupler sales | Potential cost saving per year (units) |
|---|---|---|---|
| ET8 | 44.84 | 9054 | 1798.16 |
| ET10 | 7.10 | 3450 | 111.32 |
| ET12 | 31.99 | 7325 | 1485.03 |
| ET16 | 24.26 | 15050 | 2826.78 |
| ET20 | 21.82 | 27000 | 5891.84 |
| ET25 | 12.21 | 12700 | 2077.98 |
| ET28 | 22.66 | 928 | 381.94 |
| ET32 | 16.72 | 14050 | 381.94 |
| ET40 | -6.18 | 6100 | -985.93 |
| ET50 | 1.43 | 456 | 31.56 |
| | | total | **18397.64** |

**Table 7.9 Projected Savings for Low Grade Tungsten Carbide Coated Saddle Configuration**

| Abrasive coupler type | Percentage cost saving (%) | 1995 coupler sales | Potential cost saving per year (units) |
|---|---|---|---|
| ET8 | 47.71 | 9054 | 1913.02 |
| ET10 | 11.90 | 3450 | 186.54 |
| ET12 | 35.44 | 7325 | 1644.74 |
| ET16 | 27.53 | 15050 | 3207.42 |
| ET20 | 27.06 | 27000 | 7304.69 |
| ET25 | 17.78 | 12700 | 3024.98 |
| ET28 | 27.31 | 928 | 460.24 |
| ET32 | 22.25 | 14050 | 6359.65 |
| ET40 | 1.24 | 6100 | 197.77 |
| ET50 | 8.17 | 456 | 180.10 |
| | | **Total** | **24479.17** |

# CHAPTER 8 PERFORMANCE OF WHOLE PROTOTYPE RANGE

## 8.1 Test Results

Since the only difference between the two alternative coated saddles was the coating material, it was felt appropriate to test the low cost coated couplers and the serrated saddle couplers initially. Clearly the former gives the best cost savings whilst the latter is the more traditional. If neither proved satisfactory, it was expected that the high cost coating would come into play.

Both serrated and coated abrasive couplers ET8 to ET32 were tested at Sheffield Hallam University. ET40 and ET50 couplers were tested at the University of Nottingham. Table 8.1 shows the first prototype results using three samples. The figures in bold type are those that do not meet the required performance. The slip must be less than 0.1mm with target failure loads as given.

**Table 8.1 - First prototype results**

| Serrated coupler | Slip at 0.7Fy (mm) | | | U.T.L. (kN) | | | Target U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|---|---|---|---|
| 8 | 0.078 | 0.090 | - | 29.6 | 29.6 | 29.4 | 31.0 | Ductile in bar |
| 10 | 0.000 | 0.020 | 0.038 | 47.3 | 48.1 | 47.9 | 49.0 | Ductile in bar |
| 12 | 0.033 | 0.055 | 0.050 | 70.6 | 70.6 | 70.1 | 71.0 | Ductile in bar |
| 16 | 0.063 | 0.068 | 0.053 | 120.5 | - | 121.1 | 126.0 | Ductile in bar |
| 20 | 0.100 | 0.085 | 0.078 | 210.9 | 207.1 | 205.9 | 196.0 | Ductile in bar |
| 25 | 0.065 | 0.075 | 0.108 | 290.0 | 291.5 | 293.0 | 307.0 | Ductile in bar |
| 28 | **0.176** | **0.195** | - | **349.1** | **347.1** | - | 385.0 | Pull out |
| 32 | **0.195** | **0.203** | **0.185** | **414.7** | **424.5** | **448.4** | 503.0 | Pull out |
| 40 | **0.381** | **0.443** | - | **757.0** | **757.0** | - | 785.0 | Pull out |
| 50 | **0.700** | **0.742** | - | **973.0** | **1011.0** | - | 1227.0 | Pull out |
| **Abrasive coupler** | **Slip at 0.7Fy (mm)** | | | **U.T.L. (kN)** | | | **Target U.T.L. (kN)** | |
| 8 | **0.175** | 0.000 | 0.055 | **26.6** | **25.5** | **26.0** | 31.0 | Pull out |
| 10 | 0.053 | 0.038 | 0.040 | 47.9 | 47.7 | 47.7 | 49.0 | Ductile in bar |
| 12 | 0.043 | 0.043 | 0.045 | 70.4 | 70.4 | 70.6 | 71.0 | Ductile in bar |
| 16 | **0.108** | **0.119** | **0.101** | 120.4 | 113.1 | 120.5 | 126.0 | Ductile in bar |
| 20 | 0.098 | 0.073 | - | 201.8 | 202.5 | - | 196.0 | Pull out |
| 25 | **0.115** | **0.155** | - | **292.9** | **275.9** | - | 307.0 | Pull out |
| 28 | **0.225** | - | - | **375.4** | **387.0** | **388.6** | 385.0 | Pull out |
| 32 | **0.305** | - | - | **375.0** | - | - | 503.0 | Pull out |
| 40 | **0.261** | **0.273** | **0.201** | **740.0** | **685.0** | **701.0** | 785.0 | Pull out |
| 50 | **0.439** | **0.390** | **0.427** | **1161.0** | 1275.0 | 1256.0 | 1227.0 | Pull out |

Generally for both types of saddle those couplers larger than ET20 did not perform to specification, whilst those smaller than ET20 were satisfactory. The ET25 was successful using serrated saddles but did not perform with coated saddles.The abrasive range, as a whole, did not perform as well as the serrated range, particularly with the smaller couplers. However for the ET40 and ET50 the coated saddles performed somewhat better than the serrated ones, albeit both were well below acceptable levels. On this basis the abrasive prototype coupler configuration was abandoned and all subsequent work was done with the serrated range prototypes.

Non-performers in the serrated range were couplers ET28, ET32, ET40 and ET50, which needed to be further developed.

The test specimens for those tests that were unsuccessful were disassembled and it was immediately obvious that the bolt penetration into the reinforcement was far below that expected. The penetrations were measured and a revised relationship was produced.

i.e. Depth of penetration mm = 2.324616mm + (0.021006mm/kN x Bolt downward force kN)

Both the original and the revised relationship are shown in Graph 8.1. The descrepancy between the two was thought to be due to the saddle penetration into the bar and tube. What was clear though, was the fact that the relationship between nominal ploughing stress and coupler load is non-linear.

It was still felt that the key to low cost coupler performance was bolt numbers and torque. With sufficient specimens available it was possible to test a number of ET28 couplers with progressively increased bolt torque. A working configuration was found using 8-M16 bolts torqued to 363Nm with 1.5mm pitch teeth. The results are shown in Table 8.2, where the target UTL of 385 kN was easily achieved. The small amount of slip over 0.1mm was thought to be acceptable and would be accounted for by a full test using a control bar.

## Graph 8.1 Comparing Projected and Actual Bolt Penetrations

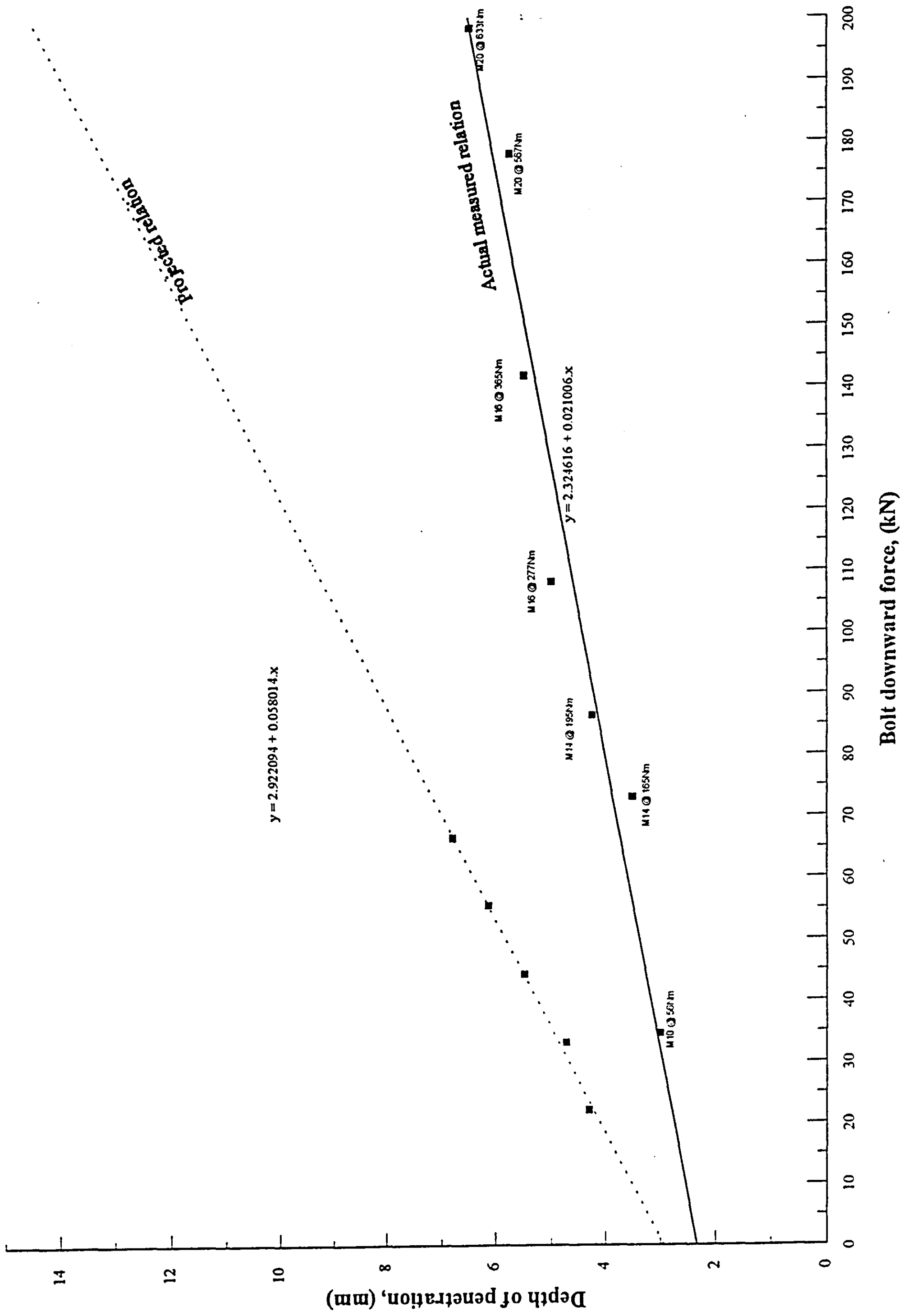


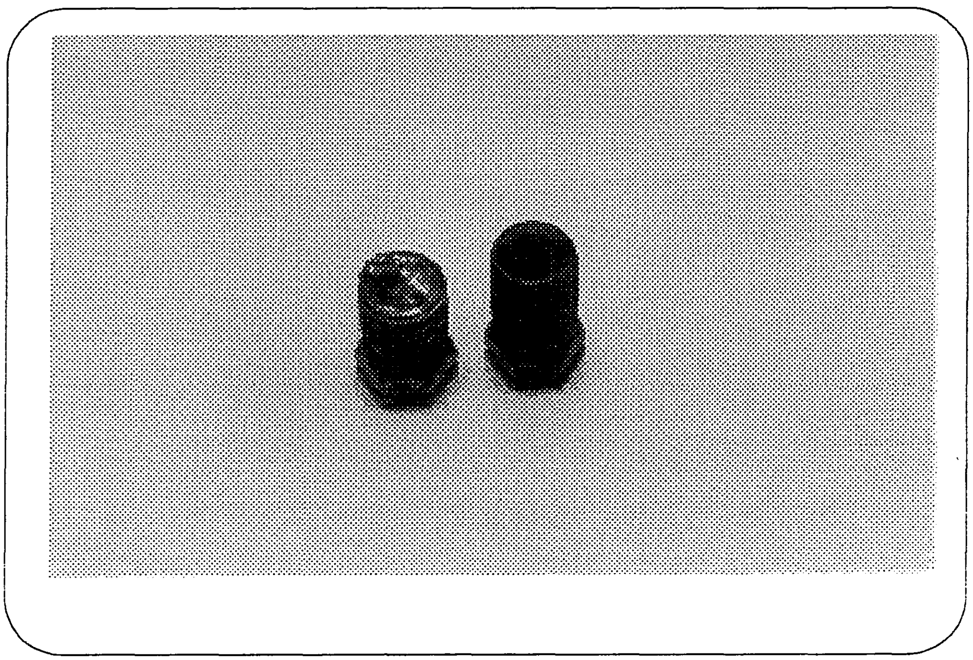
**Table 8.2 - Results for ET28 coupler with increased torque 8-M16 bolts**

| Specimen | Bolt torque (Nm) | Slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 28 | 320 | 0.175 | 415.5 | Bar pull out |
| 28 | 363 | 0.105 | 411.9 | Bar pull out |
| 28 | 363 | 0.100 | 414.2 | Bar pull out |
| 28 | 363 | 0.103 | 400.9 | Bar pull out |

## 8.2 Further development of couplers ET32, ET40 and ET50

The first prototype ET32 produced surprisingly poor upper load failure results. Examination of the disassembled specimens showed a number of bolts that had undergone severe cone collapse and failure (see fig. 8.1).

**Figure 8.1 Damaged Bolt Cone**



The bolts appeared to have failed because of internal material defects i.e. longitudinal inclusions. Any such inclusions were checked by ultrasonic testing. A number of M16 bolts from the same batch as the test bolts were examined with an ultrasonic probe. This gave no indication that there were material defects. One explanation for the failure was simply that the bolts were subjected to excessive compressive loading. To investigate this a readily available standard ET32, with

10-MI6 bolts torqued to 363Nm, was tested to the new performance criteria. The result is given in Table 8.3.

**Table 8.3 Standard ET32 with increased bolt torque**

| Coupler number | Slip at 0.7Fy (mm) | U.T.L. (kN) | Target U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| ET32 | 0.163 | 482.8 | 503 | Bar pull out |

A significant increase in upper failure load, compared to the first prototype results was observed, however, the disassembled bolts had still undergone severe cone failure. The rest of the coupler range from ET8 to ET28 were analysed to see if a relationship between numbers of bolts and failure load could be found. Table 8.4 illustrates calculations determining bolt shear stress for samples which had successfully sustained loads.

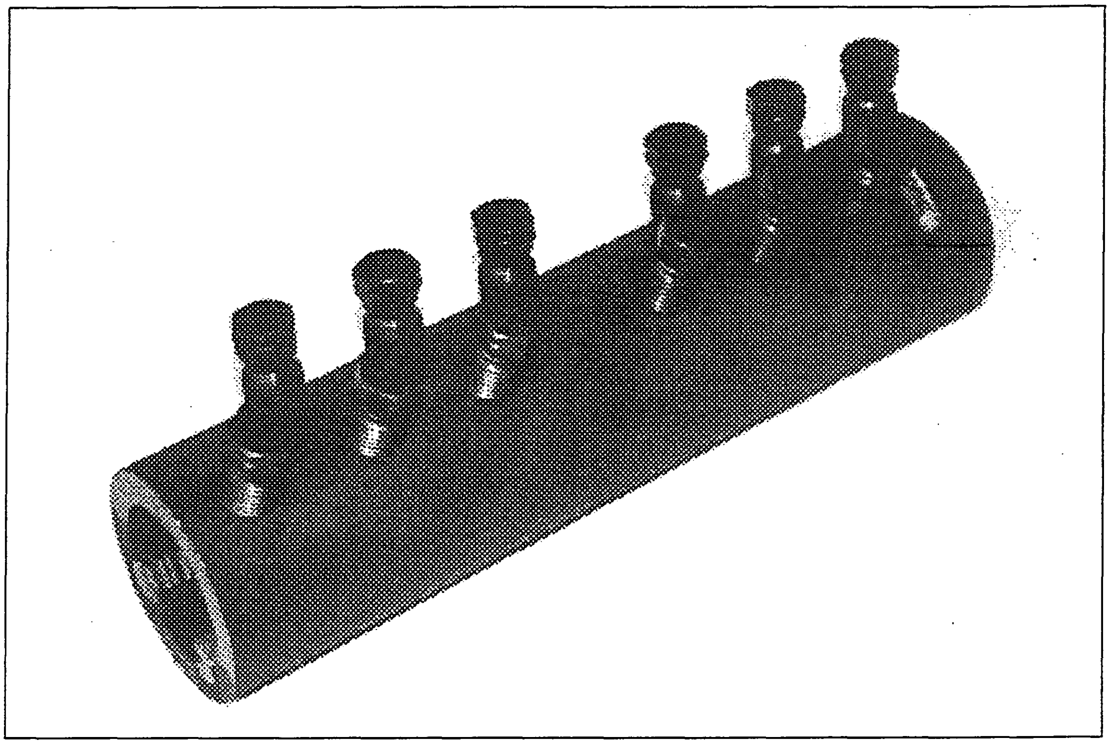
**Table 8.4 Bolt Shear Stress Calculations**

| Coupler type | 1.25Fy (kN) | Bolt type | Minor diameter (mm) | Minor shear area ($mm^2$) | Number of bolts | Total coupler bolts shear area ($mm^2$) | Shear stress in bolts ($N/mm^2$) |
|---|---|---|---|---|---|---|---|
| ET8 | 31 | 10 | 8.376 | 55.10 | 2 | 110.20 | 284.70 |
| ET10 | 49 | 10 | 8.376 | 55.10 | 4 | 220.41 | 222.88 |
| ET12 | 71 | 10 | 8.376 | 55.10 | 6 | 330.61 | 213.62 |
| ET16 | 126 | 14 | 11.84 | 110.01 | 4 | 440.03 | 285.49 |
| ET20 | 196 | 14 | 11.84 | 110.01 | 6 | 660.05 | 297.33 |
| ET25 | 307 | 14 | 11.84 | 110.01 | 8 | 880.07 | 348.69 |
| ET28 | 385 | 16 | 13.84 | 150.33 | 8 | 1202.65 | 320.13 |
| ET32 | 503 | 16 | 13.84 | 150.33 | 10 | 1503.31 | 334.26 |

Clearly the shear stress in ET 32 is not the highest in the range. Also there is no obvious pattern to be seen except that the ET28 shear stress was lower than the ET25 shear stress. If the ET32 shear stress needed to be progressively lower still, then a further two bolts per coupler would be needed. An in-line 12 M16 bolt, ET32 configuration would significantly increase the coupler length (and cost). A 12 M14 bolt, ET32 would not be as long but calculations showed that the shear area associated with 12 M14 bolts would not provide a lower shear stress.

One possible solution was to stagger the coupler bolts. The product specification stipulated a bolt sweep angle of no greater than $30^{0,}$ and the 12 M16 bolt, staggered ET32, that was produced is shown in Figure 8.2. All other components and bolt torques were the same as for the in-line configuration.

**Figure 8.2 ET32 with 12-M16 staggered bolts**



The results for the ET32 with staggered bolts were also poor considering that two extra two bolts had been added. The prototype ET40 and ET50 results were also unacceptable. On this basis the company decided that for couplers ET32, ET40 and ET50 the required slip performance would be set at 0.1mm for 0.6Fy, where Fy would remain at 500N/mm$^2$. If this performance was met with ease then 0.7Fy would be aimed for but first the slightly increased performance of 0.6Fy would be achieved if possible.

A sample of three staggered ET32 couplers were tested and the results which are given in Table 8.5 and are seen to be satisfactory for slip with a UTL of 1.0Fy being achieved..

**Table 8.5 Results for staggered 12 bolt ET32**

| Specimen number | Permanent slip at 0.6Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|
| ET32 | 0.0970 | 487.9 | Ductile in bar |
| ET32 | 0.0997 | 495.6 | Ductile in bar |
| ET32 | 0.0970 | 488.8 | Ductile in bar |

The coupler achieved the required initial slip performance and a UTL of 1.2Fy (483 kN) but it was clear that there was little likeliehood of the coupler succesfully satisfying the 0.7Fy slip criteria. Visual examination after the tests showed little damage to the bolts.

The shear stress in the bolts for this staggered ET32 at failure was calculated and the value used to produce staggered ET40 and ET50 equivalents. The staggered ET40 had 12 bolts and the staggered ET50 had 18 bolts. The coupler characteristics and the test results are given in Table 8.6.

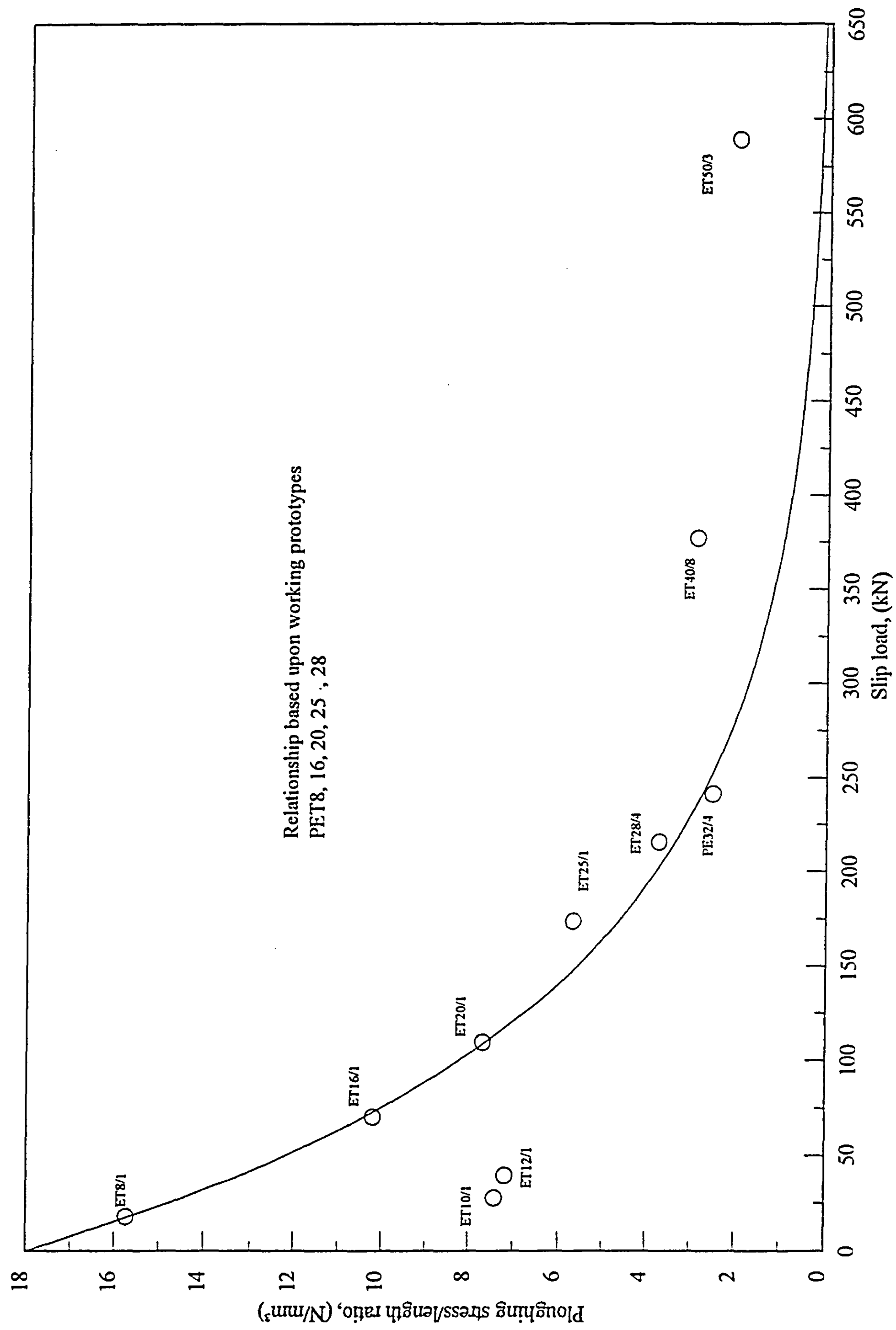
**Table 8.6 - Staggered ET40 and ET50 test results**

| Specimen number | Number bolts | Bolt torque (Nm) | Slip at 0.6Fy (mm) | Target UTL (kN) | Actual UTL (kN) | Mode of failure |
|---|---|---|---|---|---|---|
| ET40 | 12 | 567 | 0.1540 | 629 | 766 | Ductile in bar |
| ET40 | 12 | 567 | 0.1417 | 629 | 769 | Ductile in bar |
| ET40 | 12 | 567 | 0.1185 | 629 | 770 | Ductile in bar |
| ET50 | 18 | 633 | 0.3000 | 961 | 1256 | Ductile in bar |
| ET50 | 18 | 633 | 0.3000 | 961 | 1262 | Ductile in bar |
| ET50 | 18 | 633 | 0.3000 | 961 | - | Ductile in bar |

The target UTL of 1.2Fy was easily achieved in both cases, however the real problem with the larger couplers was achieving the required permanent slip, the results of which were extremely poor. It was felt appropriate to see if there was a relationship defining permanent slip performance.

A suitable exponential relationship was found between the ratio of ploughing stress and coupler length as a function of permanent slip load. This is shown in Graph 8.2. To bring the ET40 and ET50 couplers in line with this relationship was beyond economic and physical viability i.e. the ET40 would be 60% more expensive than the present range, have 20 M-16 torqued to 450Nm and the ET50 would be 140% more expensive with 80-M16 bolts torqued to 450 Nm and over 2m long!.

**Graph 8.2 Relationship Between Ploughing Stress/Length and Slip Load**



It was decided to be impractical to further develop the ET40 and ET50 couplers and that the company would market them stating their lower performance. This left a range of couplers from ET8 to ET32 satisfying the new criteria, where the ET32 was the only staggered configuration. It was felt however, that the oddness of one coupler, in a range of eight, was not a marketable option and an in-line 12 bolt ET32 coupler would be too expensive. The staggered ET32 was abandoned and the present ET32 was included with the ET40 and ET50 as a range of lower performance couplers.

More importantly the succesful ET8 to ET28 couplers are generally the only coupler sizes used in Europe, specifically Germany, and can clearly be marketed as a coherent range.

## 8.3 Accredited testing of ET8 to ET28

The ET8 to ET28 range needed to be accredited to the new specification at an approved external testing facility. This was necessary to obtain a new British Board of Agrèment approval certificate This was expected to be fairly straightforward, but unfortunately the ET25 was unsuccesful, particularly in regard to slip performance.

Graph 8.2 shows that the ET25 sits further above the line than any other working prototype. The tests were expected to be successful but the ET25 failed to perform, consistently producing slip results of 0.15mm. All possible test equipment, component and assembly factors were investigated to try to isolate any mistakes. None were found to explain the poor slip performance, although the graphical evidence suggested that the ET25 configuration would not work.

A working prototype ET25 would have to satisfy the relationship shown in Graph 8.2. Two alternatives were derived. One had eight M14 bolts torqued to 232 Nm and the other had six M16 bolts torqued to 262 Nm. For the latter, the tube wall thickness was marginally increased to accomodate the higher bolt loads.

Both configurations were tested and the results are given in Table 8.7.

**Table 8.7 Results of alternative ET25 tests**

| Sample/specimen number | Bolt type | Permanent slip at 0.7Fy (mm) | U.T.L. (kN) | Mode of failure |
|---|---|---|---|---|
| 25 | M14 | 0.1275 | n/a | n/a |
| 25 | M14 | 0.1538 | n/a | n/a |
| 25 | M14 | 0.1213 | n/a | n/a |
| 25 | M16 | 0.0844 | 315.3 | Ductile in bar |
| 25 | M16 | 0.0912 | 316.1 | Ductile in bar |
| 25 | M16 | 0.0655 | 317.6 | Ductile in bar |

The latter configuration met the performance requirements and was confirmed in testing at Nottingham University.

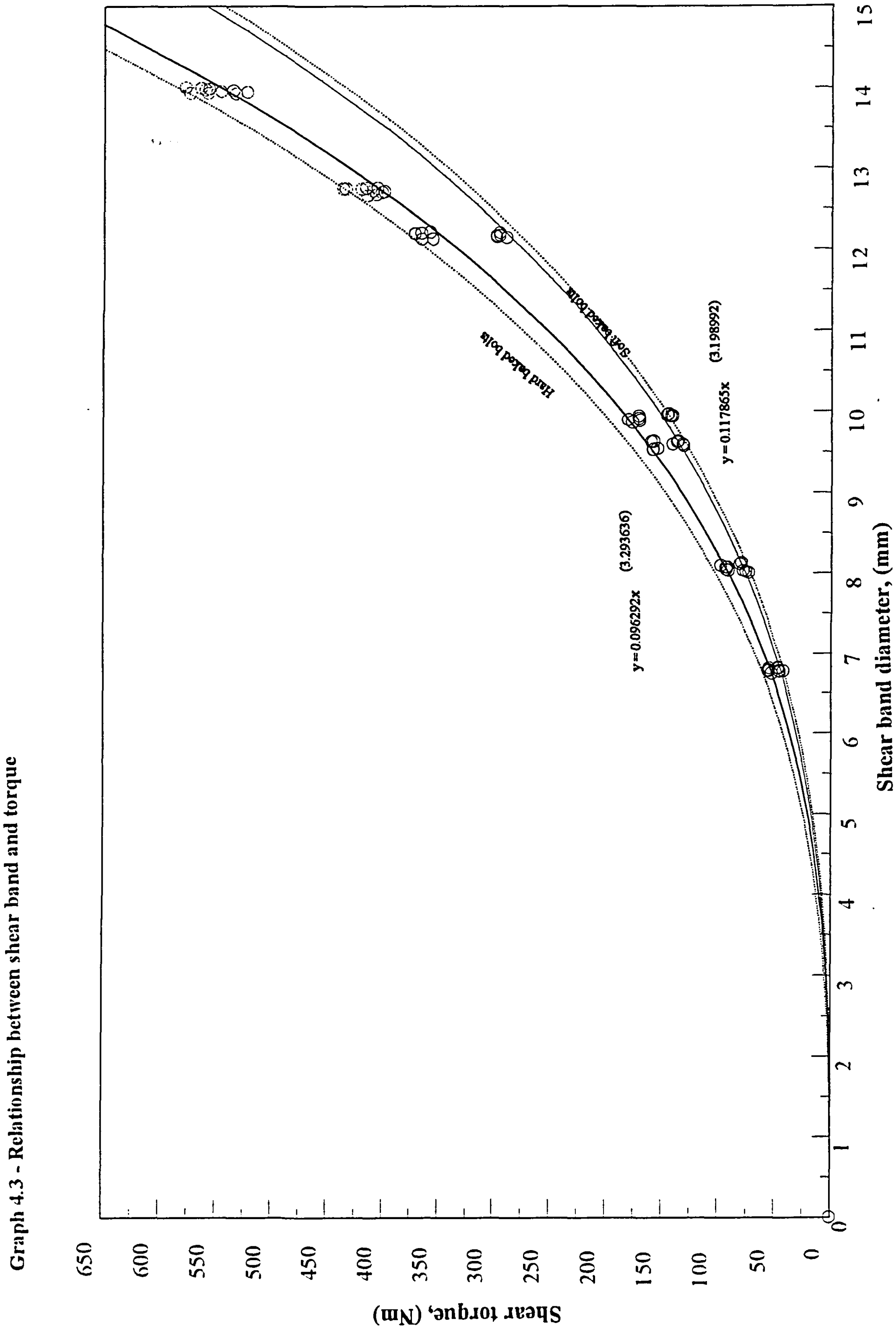
That completed a range of couplers from ET8 to ET28 that met the increased static performance requirements.

It had been decided at an early stage in the project that the static load requirementswould be dominant and would be the basis of the new couplers. Previous history of this product had shown that any coupler that met the static load requirements would satify the fatigue criterion. However, for accreditation, the British Board of Agrement required satisfactory fatigue testing of sizes ET8 and ET28 to the criteria in BS5400:part 4 (7). This was completed at Sheffield Hallam University and the couplers were found to be satisfactory.

### 8.4 New shear bolts

In all of the development tests, the bolts were tightened using a calibrated torque wrench. Actual production couplers use shear band bolts, where the head shears at a prescribed torque value. This alleviates the need for a torque wrench on-ssite and guarantees the fitting with the correct bolt torque. Since bolt torque, tube diameter and bolt diameter had changed in most cases for couplers ET8 to ET28, new bolt dimensions were required. Firstly the new shear band diameters were determined. To do this, the relationship between shear torque and shear band diameter was established.

## Graph 8.3 Relationship Between Shear Band and Bolt Torque



A large selection of bolt diameters were measured using a point micrometer. These bolts were then sheared using the Avery torsion machine and the maximum bolt torque recorded. The bolt hardness was also measured with a Vickers hardness tester. Some of the bolts were hard baked and some were soft baked. A relationship was found for both hard bolts and soft bolts, each of which is illustrated in Graph 8.3.

For hard baked bolts

$$sheartorque\_Nm = 0.096292\left[\frac{Nm}{mm^{(3.293636)}}\right]x^{(3.293636)}mm^{3.293636}$$

where x = shear band diameter

For soft baked bolts

$$sheartorque\_Nm = 0.117865\left[\frac{Nm}{mm^{3.198992}}\right]x^{(3.198992)}mm^{3.198992}$$

where x = shear band diameter

The bolt hardness of stock bolts is consistently at the hard baked end of the scale so the new shear band diameters were determined using the hard baked relation.

The new bolt lengths were determined by the bolt depth of penetration, tube dimensions and bar dimensions i.e.

Bolt thread + cone length

= (Depth penetration + Max. tube O/D) - (Min. tube thickness + saddle/bar daimeter)

### 8.5 Cost Analysis of Whole Range

Certain similar components such as bolts, saddles and tubes were grouped where possible. For example the ET8 and ET10 couplers were given a common tube size. The selected tube dimensions were designed for the ET10 so the ET8 had larger saddles incorporated. The economic reasons for doing this are illustrated in Table 8.8

**Table 8.8 - Cost analysis of ET8 and ET10 tube**

Cost implications of bespoke tube for both ET8 and ET10

| Coupler | O/D (m) | Price per tonne (units) | Minimum tonnage purchase (Tonnes) | Holding stock (units) | Tonnage sold based on 1995 figures (Tonnes) | Cost saving based on 1995 sales (units) |
|---|---|---|---|---|---|---|
| ET8 | 0.026 | 317 | 2 | 633 | 0.72 | 876 |
| ET10 | 0.028 | 260 | 2 | 519 | 1.01 | 178 |
| | | | Total | 1152 | Total | 1054 |

Cost implications of single tube size for both ET8 and ET10

| Coupler | O/D (m) | Price per tonne (units) | Minimum tonnage purchase (Tonnes) | Holding stock (units) | Tonnage sold based on 1995 figures (Tonnes) | Cost saving based on 1995 sales (units) |
|---|---|---|---|---|---|---|
| ET8/10 | 0.028 | 260 | 2 | 519 | 1.73 | 1043 |
| | | | Total | 519 | Total | 1043 |

Note : Bespoke ET8 price is £0.727 and common ET8
(ET10 tube with larger saddles) is £0.732

For the ET28 the increase in tube wall thickness reduced the savings from 26.96% to 19.06%. Actual savings are shown in Table 8.9.

**Table 8.9 Cost savings for whole range**

| Serrated coupler type | Percentage cost saving (%) | 1995 coupler sales | Potential cost saving per year (units) |
|---|---|---|---|
| ET8 | 45.18 | 9054 | 1811.54 |
| ET10 | 10.64 | 3450 | 166.78 |
| ET12 | 36.88 | 7325 | 1711.78 |
| ET16 | 24.48 | 15050 | 2852.94 |
| ET20 | 23.32 | 27000 | 6296.40 |
| ET25 | 16.05 | 12700 | 2730.12 |
| ET28 | 19.06 | 928 | 321.25 |
| | | Total | 15890.80 |
| | | Orig. Cost | 65570 |
| | | % saving | 23.5 |

Thus for the range from ET8 to ET28 which satisfied the new specification, savings of 23.5% were obtained. Since the remainder of the range had increased costs to enable them to operate against a reduced specification then these costs are not quoted.

# CHAPTER 9 DISCUSSION AND CONCLUSIONS

The objectives of this project at the outset, were to increase the product range permanent extension performance by 27% and increase the upper tensile load capacity by 9% whilst reducing product cost by up to 20%. At the conclusion of the project, Ancon Clark has a new lower cost, better performing range of seven connectors in sizes 8, 10, 12, 16, 20, 25, and 28mm respectively. The performance objectives have been achieved for these seven sizes and the average cost reduction across the range specified above is 23%

Even though the product is very simple in construction the actual operation of each of the components and their function during loading was found to be extremely complex. The method of development adopted in this project was undoubtedly appropriate as the final number of tests conducted was about 120 and was certainly less expensive than a computer model would have been both in terms of cost and time. A logical approach was taken by firstly isolating individual component contributions to performance with the 20mm connector. These were then altered with a view to establishing trends in cost reduction and increasing performance. Several key performance/cost relationships were established during these tests. Increasing the clamp bolt size generally improved load carrying capability to the point where the number of bolts could be reduced which would then reduce cost. To make this viable the torque applied to the bolt had to be increased towards the bolt failure condition to increase the penetration of the bolt into the bar and thus the resistance to pull through of the bars. The added advantage of reducing the number of bolts was a complementary reduction in tube length with associated cost savings. Another key factor was the use of serrated saddles with small serrations which reduced the depth of the saddle and hence the inside diameter of the tube. Due to the discrete values of bolt sizes and cutters for the saddle serrations, continuous variation of these parameters was not possible. The key principles by which performance was improved were found to be increased bolt size, bolt torque, decreased numbers of bolts and adoption of small serrations together with the order in which the bolts were tightened.

When these principles were applied to the rest of the connector range, however, only the sizes below the first developed connector satisfied the required performance criteria. Investigation of the failed test specimens showed that the principal bolt relationship, which had been used to redesign all couplers, was incorrect. Derivation of the original relationship ignored the saddle penetration into the bar and into the sleeve which, when the corrected and original relationships were compared, was significant. This meant that for the 25 and 28mm connectors insufficient bolt penetration was being achieved. The relationship was revised using bolt penetration measurements from fully assembled connectors. The data was used to redesign the 25 and 28mm connectors which subsequently satisfied the performance requirements. The larger three couplers, ie 32, 40, and 50mm, were shown to be beyond these key principles both economically and dimensionally. It must be stated that at the outset of the project the existing ET40 and ET50 couplers did not perform to the existing performance criteria.

The key performance requirement is 0.1mm permanent extension (or slip) after loading to 0.7Fy, where Fy is the yield strength of the bar. The 0.1mm permanent extension is determined by the maximum specified allowable concrete structure crack width under load, and therefore applies to all sizes of reinforcement bar and their respective connectors. The relationship derived from the working developed prototypes showed the magnitude of clamping force, bolt penetration and saddle penetration required for the larger connectors to meet the performance criteria. This configuration cannot accommodate these requirements dimensionally (ie too little space). for example the 50mm connector would have to be over 2m long with 60 bolts. It was felt that maybe the use of large bolts, with coarse threads, in the 32, 40 and 50mm connectors contributed to the permanent extension. Finer threads may have reduced this contribution but for the same torque value a finer thread bolt would produce a lower clamping force and therefore would require a higher torque which would create further design problems. I believe a different connector configuration altogether is required for the larger bar connectors. Literature which accompanies a competitors patented parallel threaded connector contains data which indicates that parallel threads meet the project objective permanent extension performance for all sizes in the 8 to 50mm bar range. This at least proves that the performance can be achieved.

Apart from the increase in connector performance and reduction in cost comparing the developed couplers and the existing range there is a general reduction in numbers of bolts and a reduction in overall length/weight of up to 40% (for the ET8). These factors will contribute to easier/quicker installation of the developed range. Problems had occurred with the existing range in cases where maximum tolerance bars would not fit into the connector sleeve. All of the new range will accommodate maximum tolerance bars.

What was surprising were the significant cost savings associated with the adoption of smaller serrations in the saddle. The savings made in manufacturing costs and hardness treatment had not been conceived at the outset of the project and totalled, on average, 4%. The developed range would ease manufacture because the reduced depth serrations could be machined twice as fast and generally fewer holes per connector would need to be drilled. The manufacture of the product is probably the next route to be investigated for cost savings. Certainly the adoption of smaller serrations may make alternative/less expensive manufacturing methods viable.

The investigation into an abrasive saddle looked very interesting in the early stages but proved incapable in the final analysis as only four out of ten of the original prototype sizes satisfied the performance criteria. It was agreed, however, that the patentability and potential cost savings justified the extra time involved.

To measure the success of the project all of the benefits to Ancon Clark must be listed. The company has a developed range of seven connectors which are easier to install and will conform to any new European legislation at a reduced cost of 15%. These sizes account for 90% of Ancon coupler sales annually. Cost savings made will pay off the development cost of the project in $1^1/_2$ years, at current sales levels. Most importantly Ancon has documented test data and design notes to deal with most eventualities, something that had not been available before. The opportunity to design bespoke connectors within this seven size connector range now exists. It has been recently suggested, however, that this cost saving will do little to increase the Ancon connector market share. Coupler prices were reduced by 30% last year to try and increase sales, with little effect. This suggests that the Ancon connectors are outpriced by more than 30% by the competitor connectors and cannot compete in the bulk supply market. Therefore, the new range cost savings may only be realised through current sales levels, with little impact on sales volume, but of course increased profitability on current sales volumes.

It must be accepted then that the products are used solely for refurbishment, repairs or applications where mistakes have been made (eg the bars cannot be rotated to allow fitment of threaded connectors) and the competitor coupler cannot be used. In these instances the customer has little choice but to use the Ancon connector, with the absence of a competitive alternative. There is thus an opportunity to refocus the market that the Ancon connector is aimed at and potentially increase sales volume as well as list price and ultimately profit levels, combining this with the achievements realised by this project and the future of this niche market for the Ancon connector looks secure for some years to come.

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| PART NO | DESCRIPTION | MATERIAL | QUANTITY | DWG NO. |
|---|---|---|---|---|
| 1 | MALE COLLAR – FIXED | MILD STEEL | 1 | 1 |
| 2 | M6 HEX SOCKET SET SCREW, POINTED | CARBON STEEL | 6 | 3 |
| 3 | M6 X 1.0 BOLT HEX SOCKET CAP SCREW<br>LENGTH 40mm<br>THREAD LENGTH 18mm | CARBON STEEL | 4 | – |
| 4 | FEMALE COLLAR – FIXED | MILD STEEL | 1 | 2 |
| 5 | M5 X 0.8 HEX SOCKET SET SCREW<br>LENGTH 16mm<br>THREAD LENGTH 16mm | CARBON STEEL | 2 | – |
| 6 | CROSS HEAD – FIXED | MILD STEEL | 2 | 5 |
| 7 | UPRIGHT | SILVER STEEL | 4 | 4 |
| 8 | MALE COLLAR – SLIDING | MILD STEEL | 1 | 1 |
| 9 | FEMALE COLLAR – SLIDING | MILD STEEL | 1 | 2 |

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CLIENT: MIT DEVELOPMENT

PROJECT: SLIP MEASUREMENT TESTING RIG

TITLE: PARTS LIST

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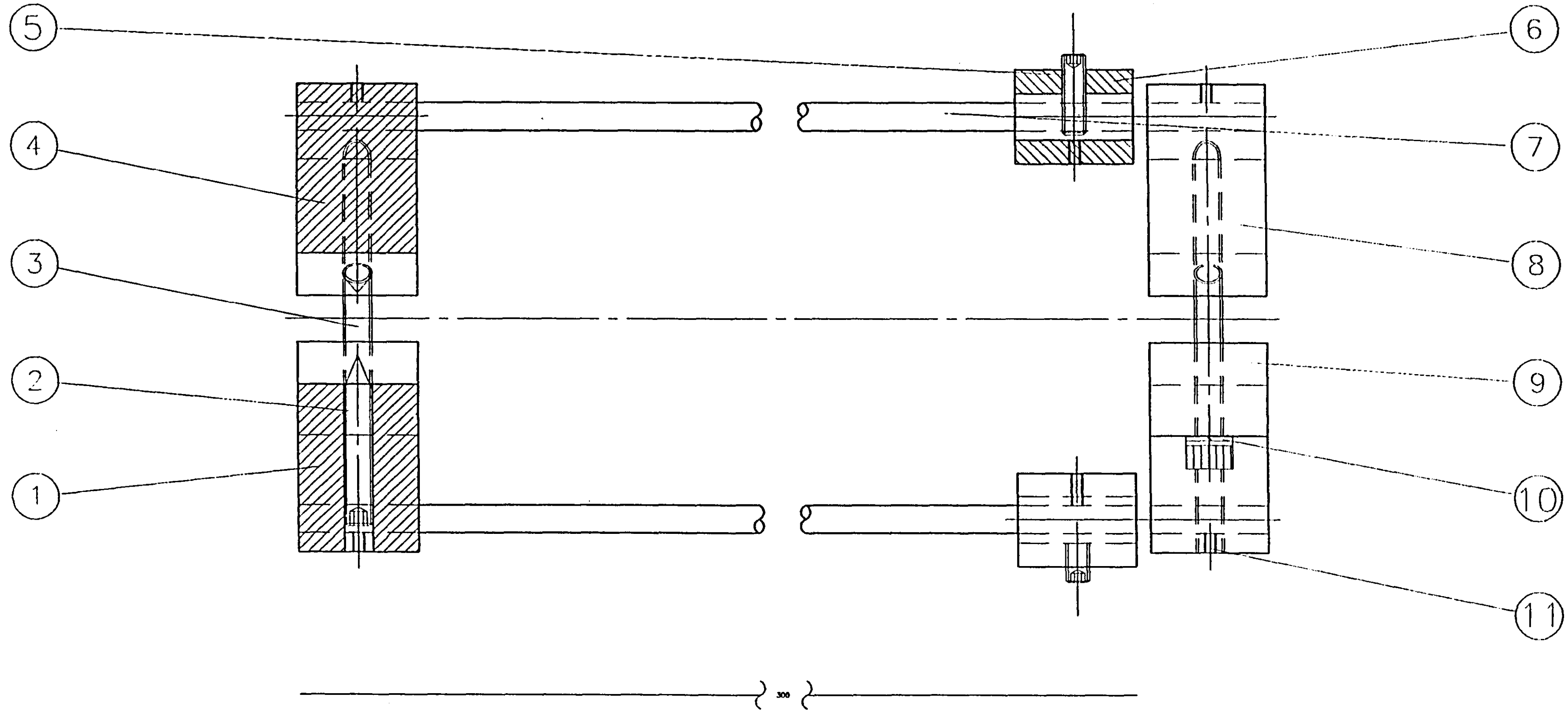


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COUPLER '97

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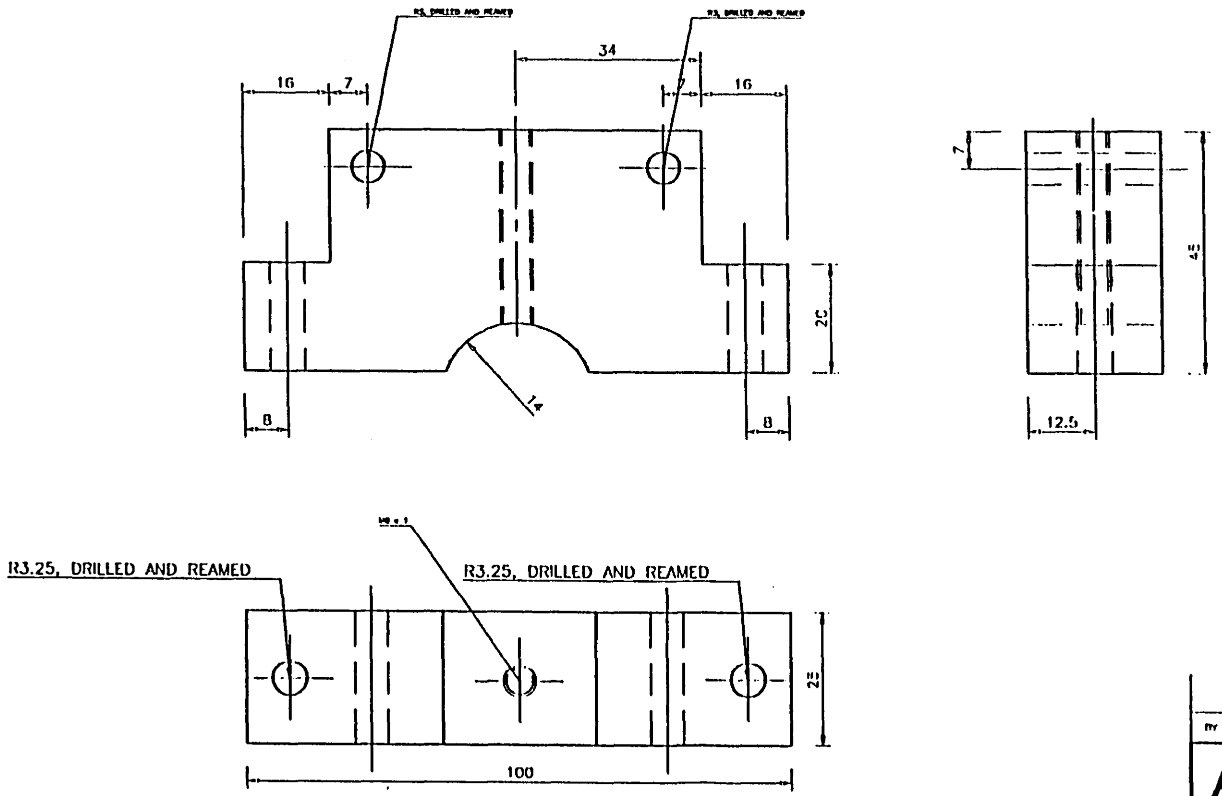


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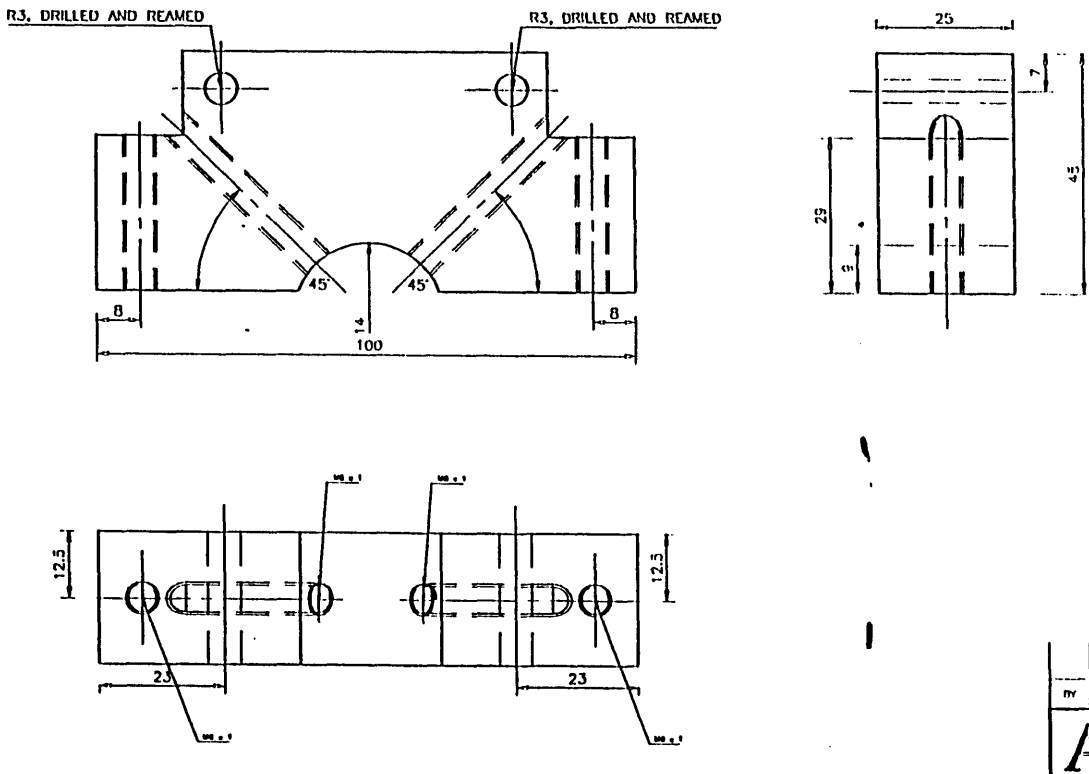


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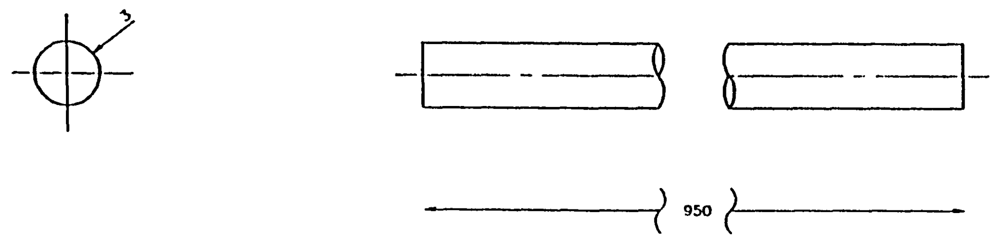


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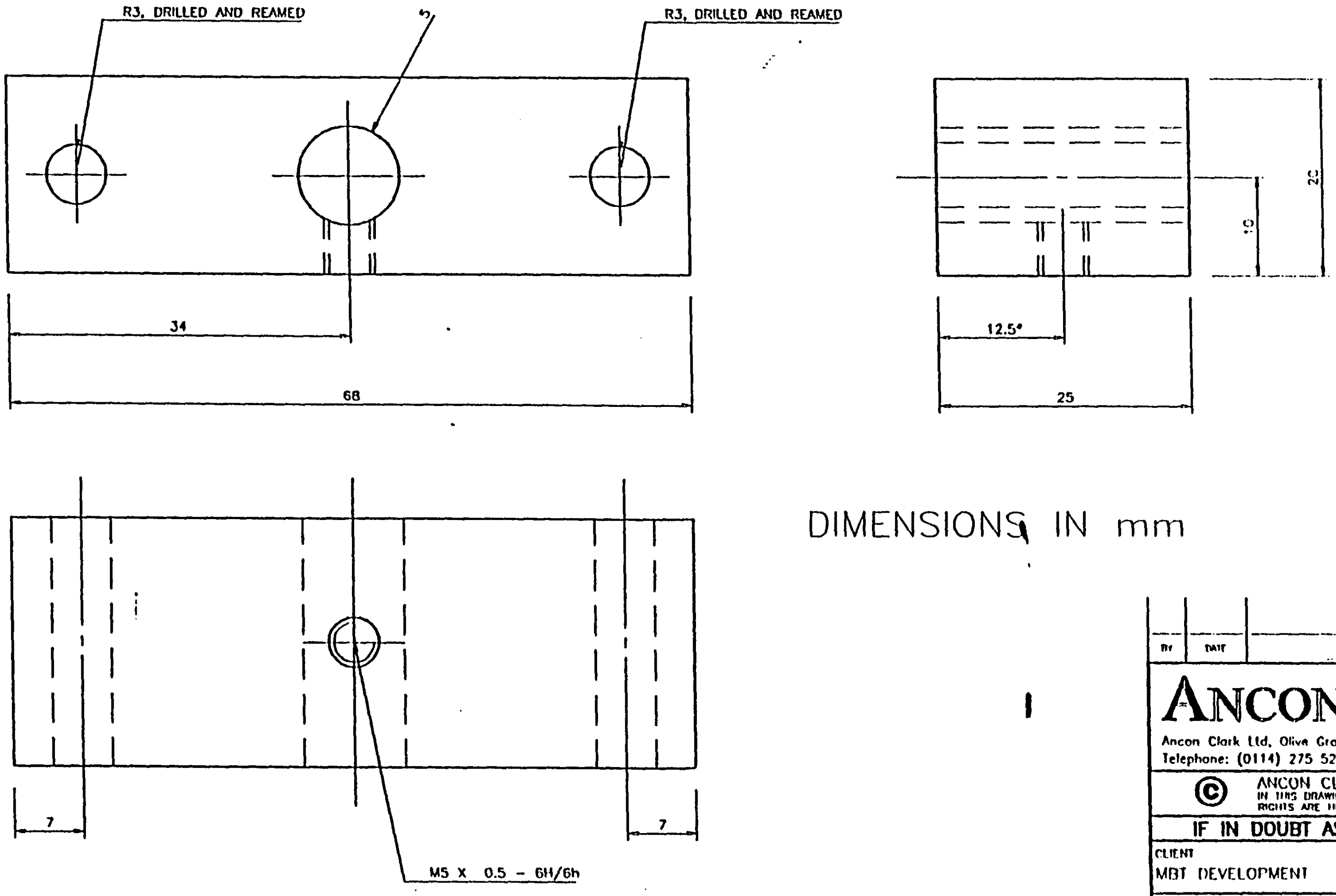


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Ancon Clark Ltd, Olive Grove Road, Sheffield, S2 3GB.
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MBT DEVELOPMENT

PROJECT
SLIP MEASUREMENT TESTING RIG

TITLE
CROSS HEAD WORKING DRAWING

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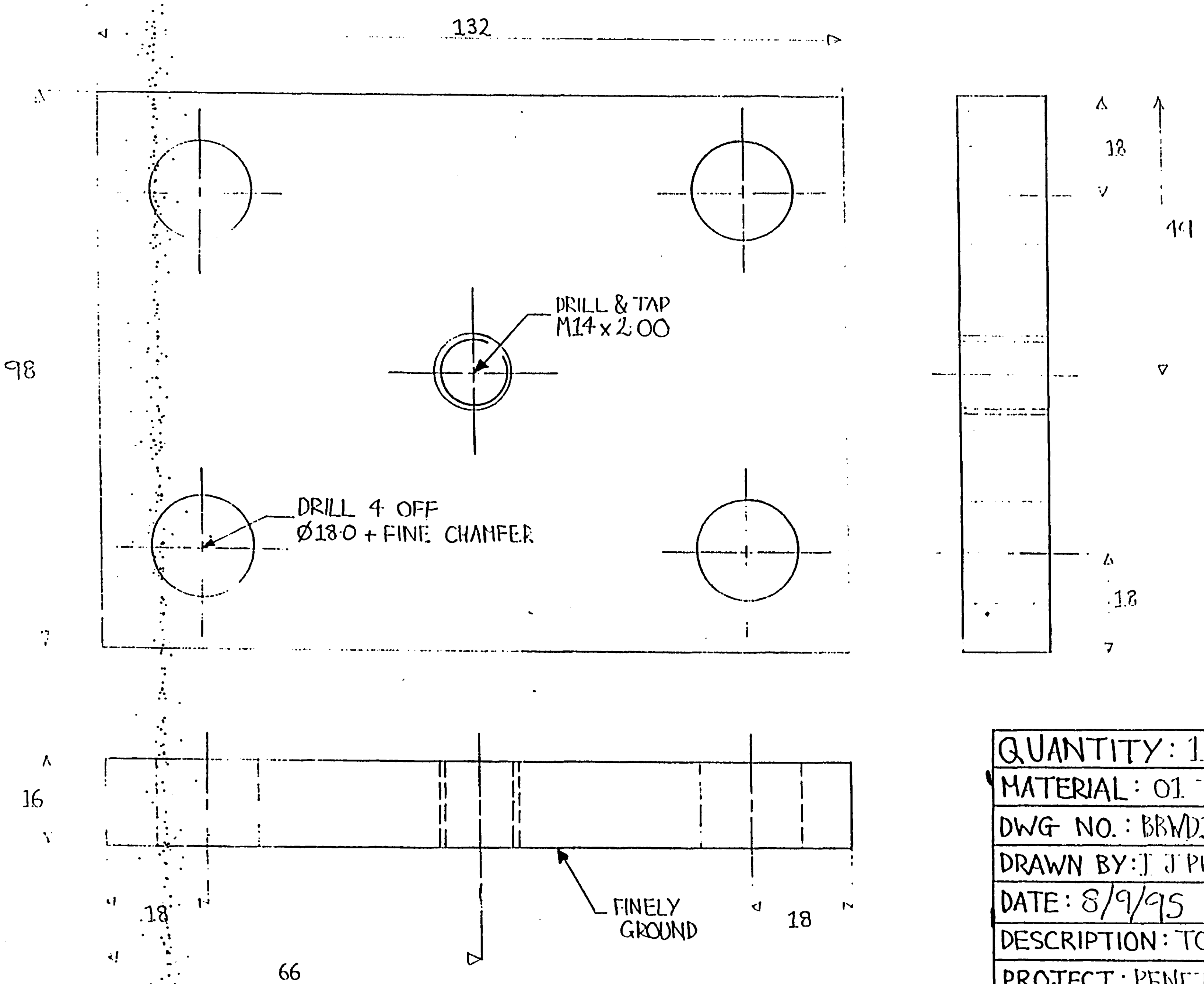


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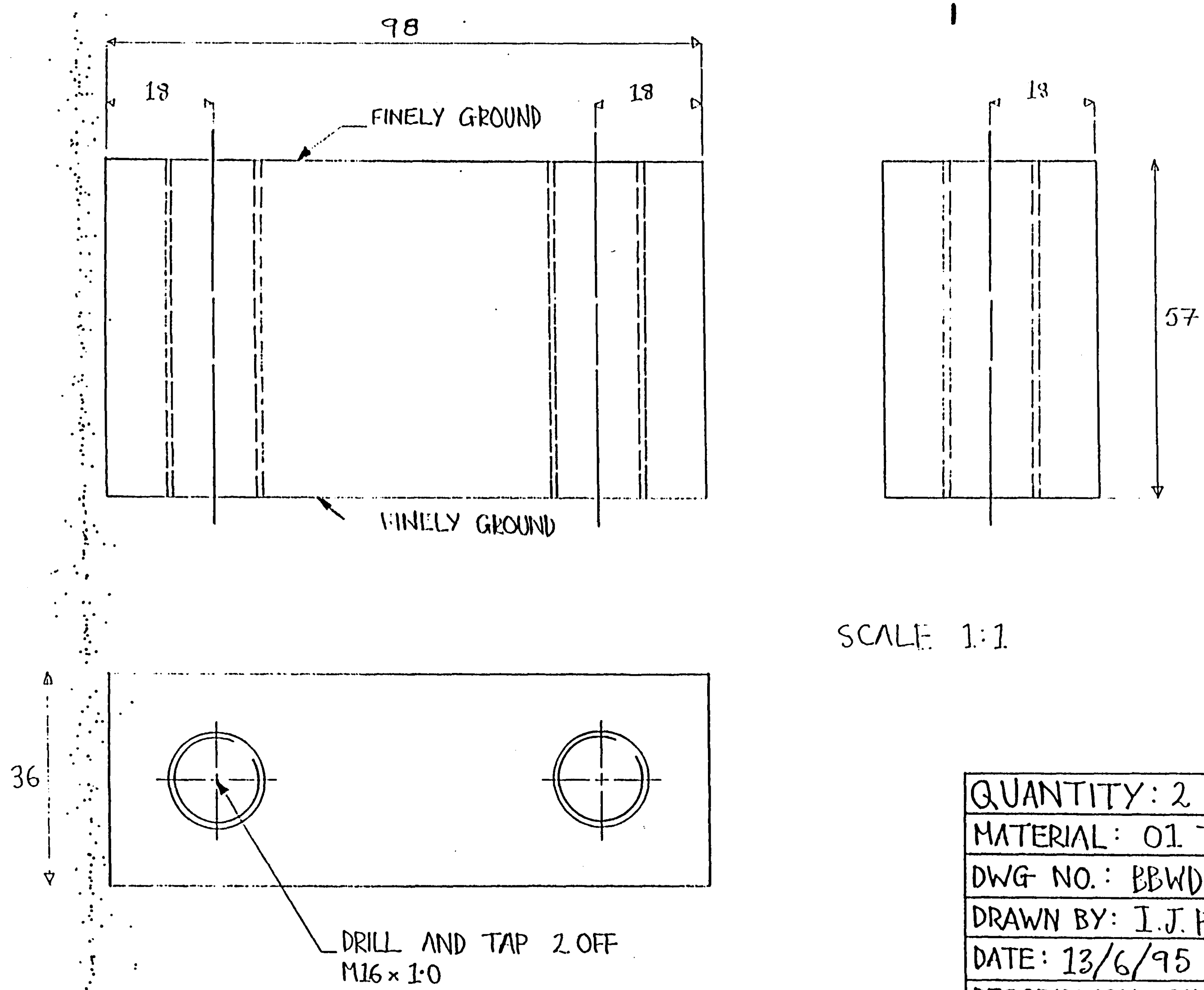


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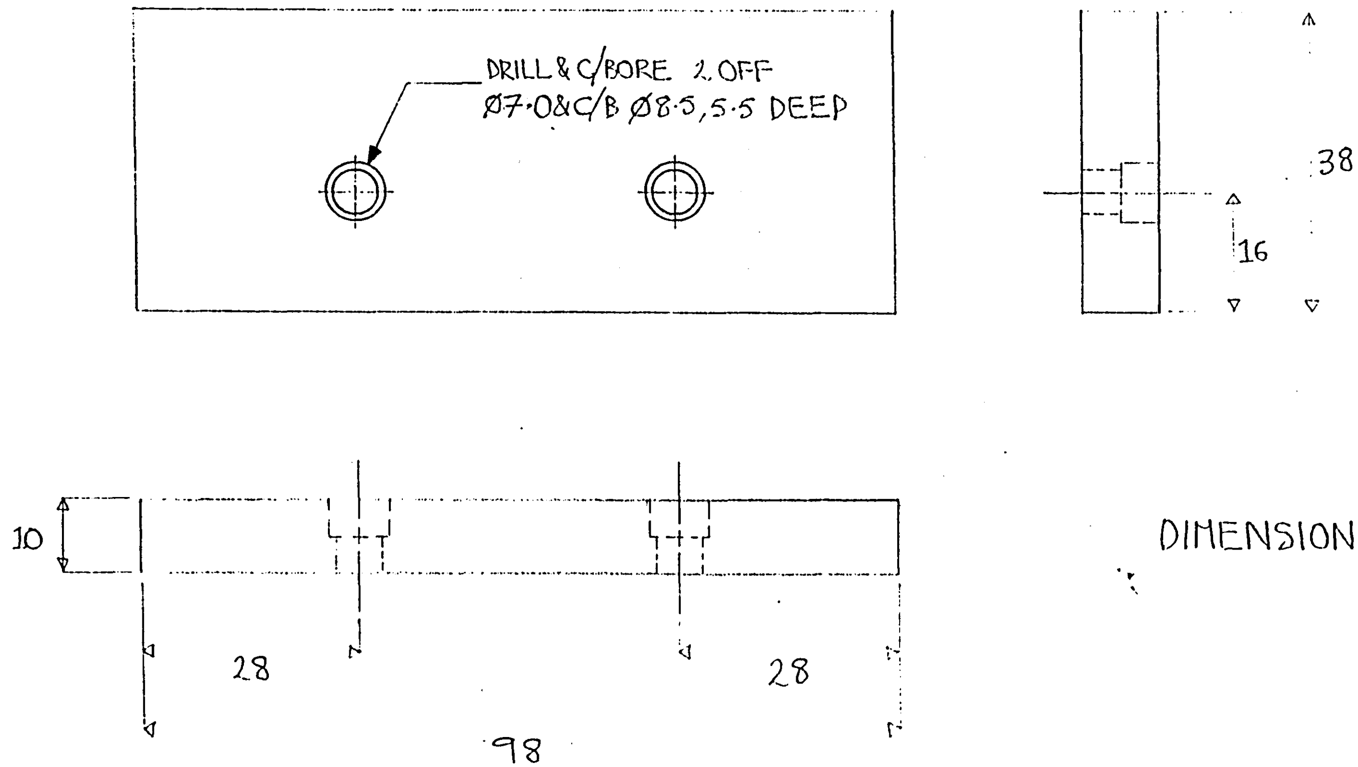


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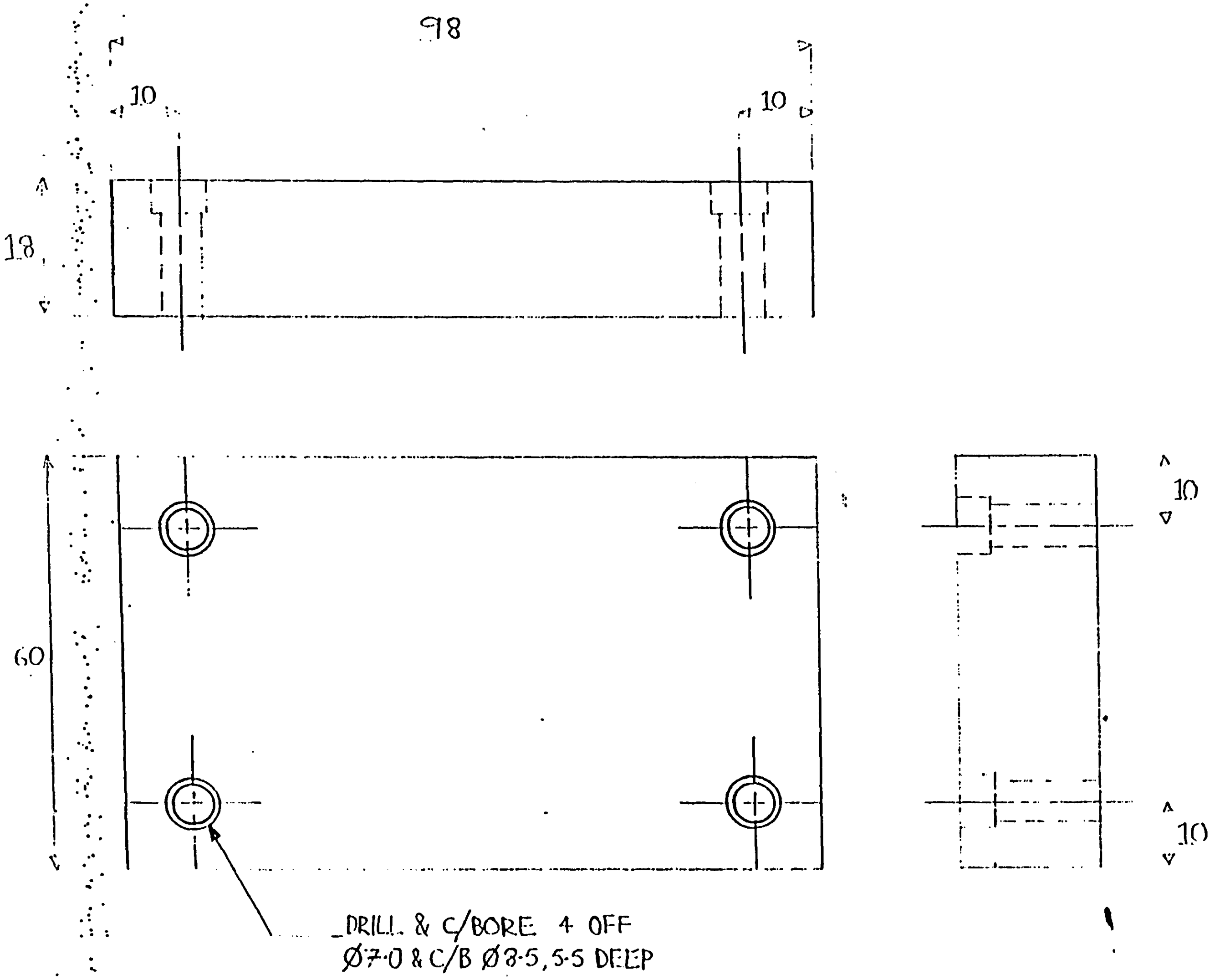
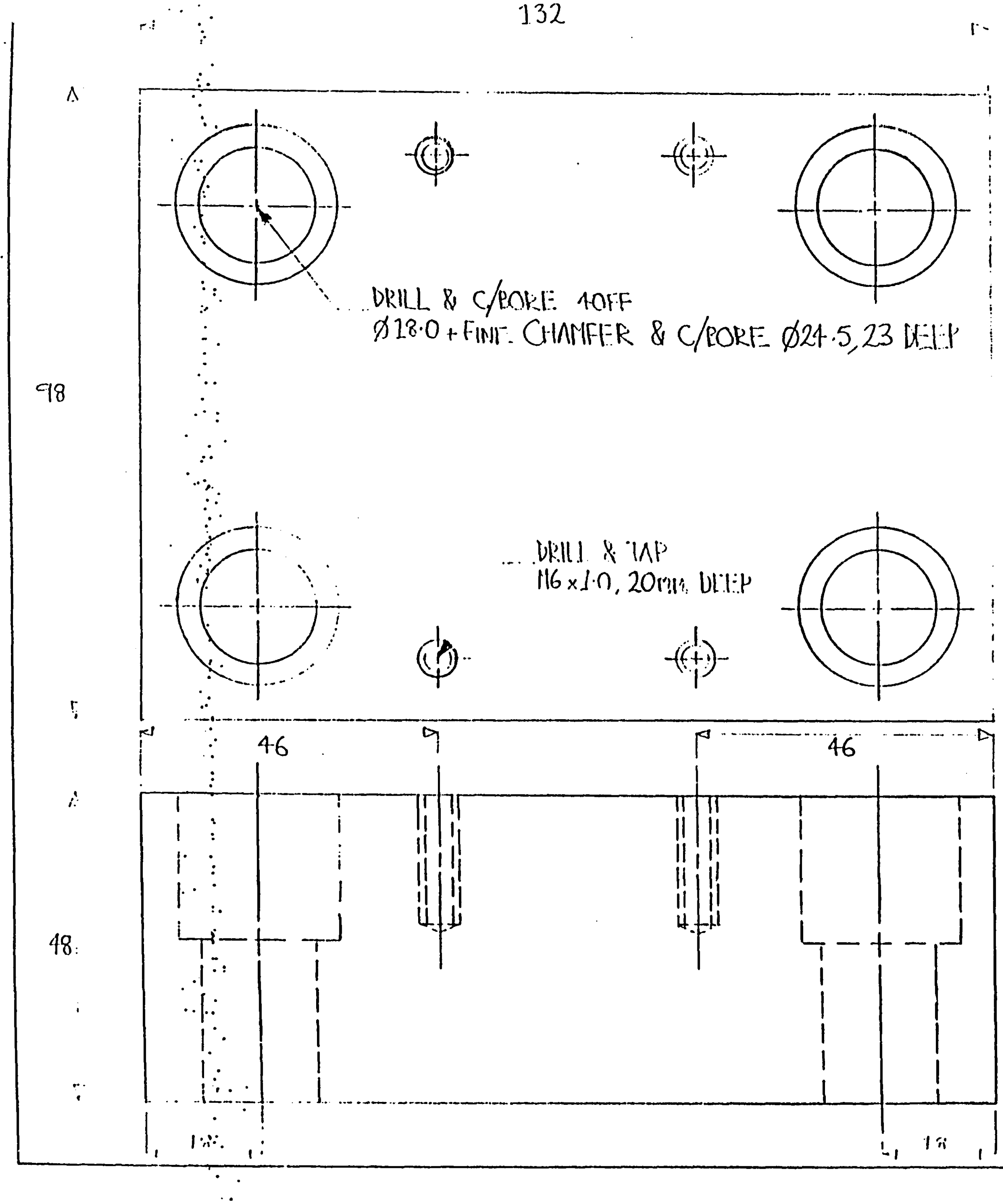
|---|---|
| QUANTITY: | 2 OFF |
| DESC: | RETAINING PLATE |
| MATERIAL: | 01 TOOL STEEL, HARDENED 50 |
| DWG NO.: | BBWD 3A |

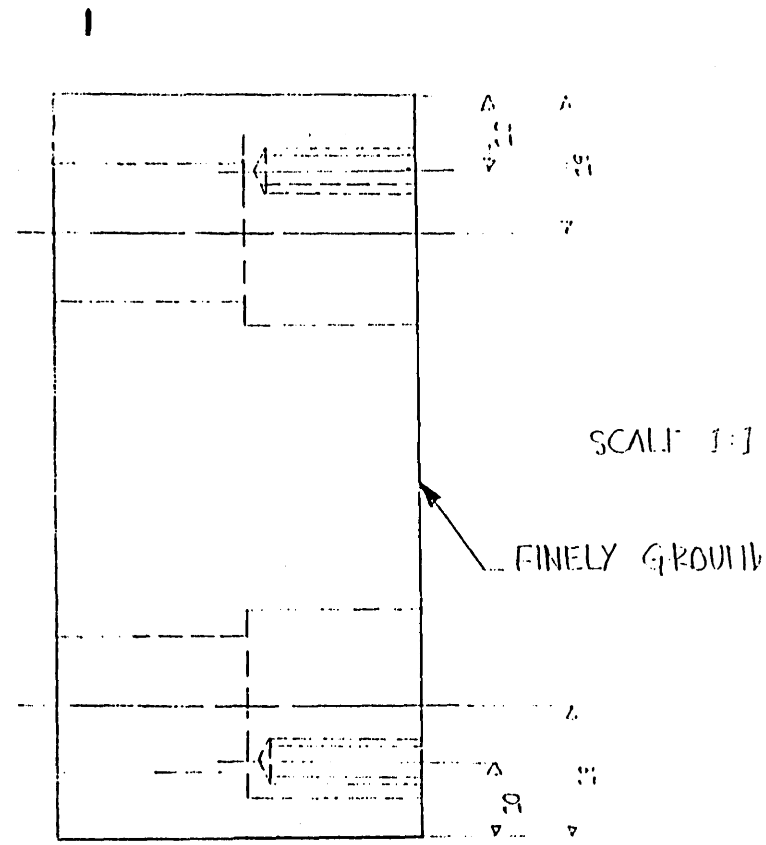



SCALE 1:1

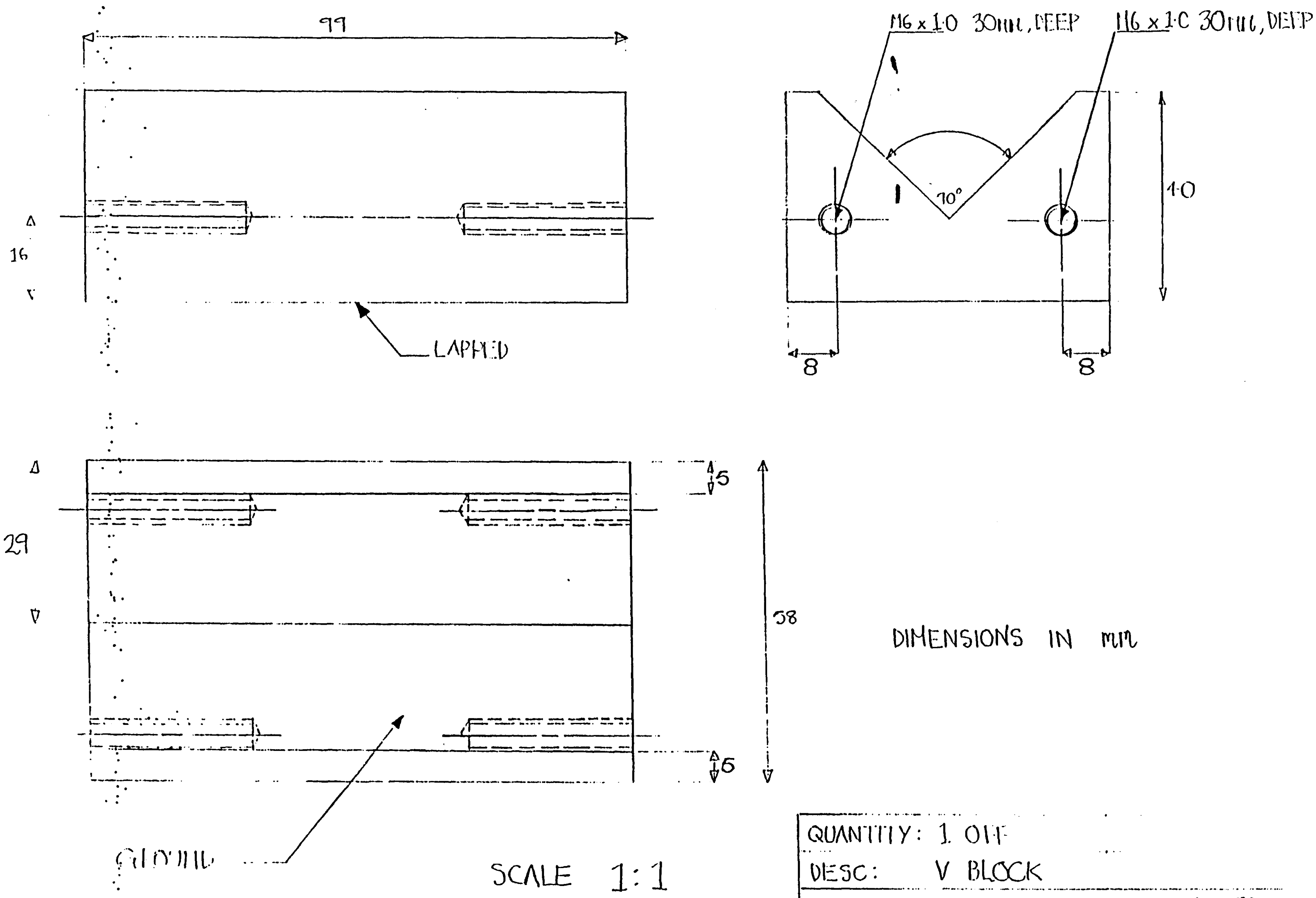
DIMENSIONS IN mm

| | |
|---|---|
| QUANTITY: | 1 OFF |
| DESC: | VICE PLATE |
| MATERIAL: | 01 TOOL STEEL, HAR[illegible] |
| DWG. NO.: | BBW014A |

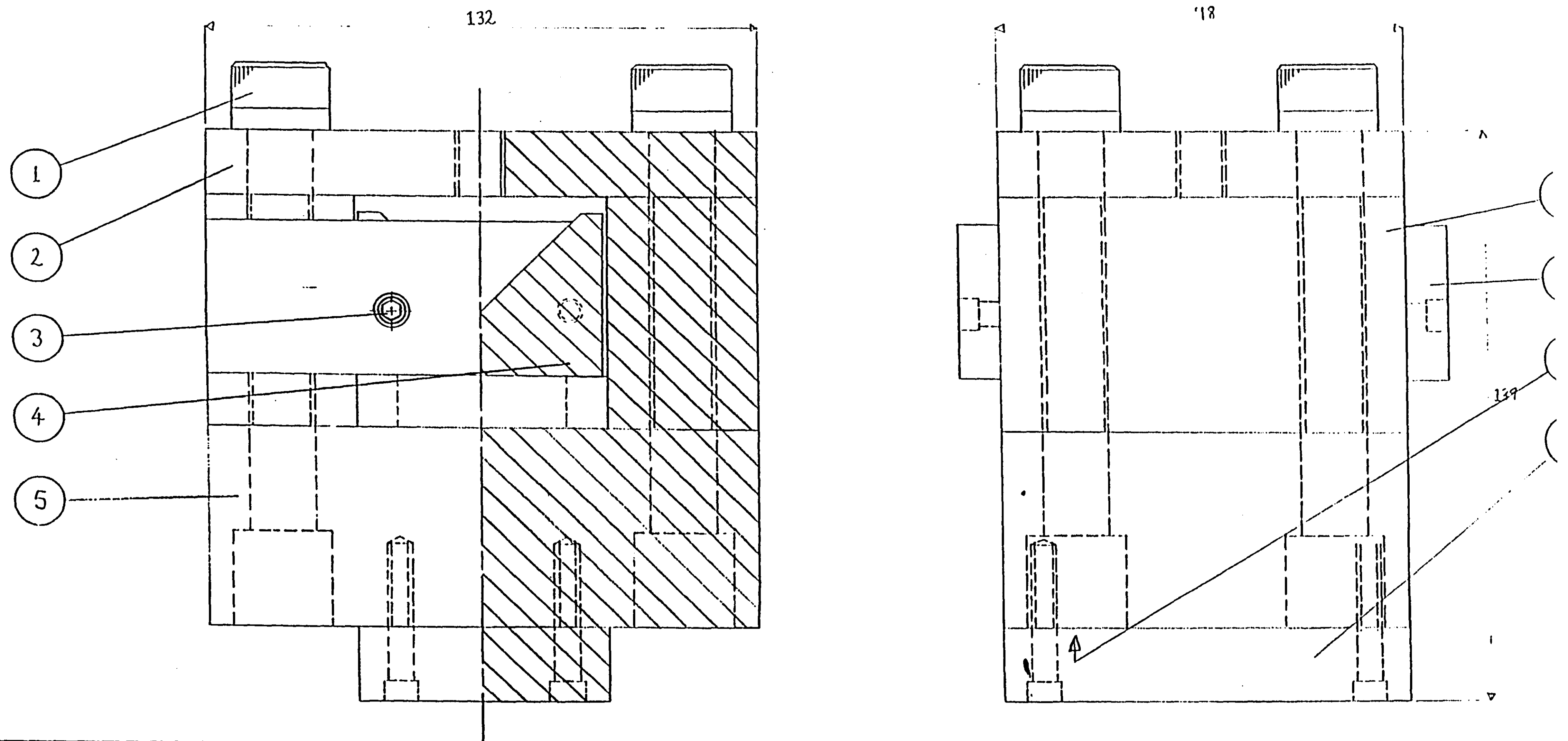



| | |
|---|---|
| QUANTITY: 1 OFF | |
| MATERIAL: O1 TOOL STEEL, HARDENED | |
| DWG NO.: BBWD5A | |
| DRAWN BY: I.J PUGH | |
| DATE: 13/6/95 | DIMENSIONS IN |
| DESCRIPTION: BASE PLATE | |
| PROJECT: PENETRATION BLOCK | |
| TITLE: BASE PLATE WORKING DRAWIN | |




| | |
|---|---|
| QUANTITY: | 1 OFF |
| DESC: | V BLOCK |
| MATERIAL: | TOOL STEEL, HARDENED TO 50RC |
| DWG NO.: | BBWD6A |




| | | | |
|---|---|---|---|
| 9 | VICE PLATE | 1 | BBWD4A. |
| 8 | M16 x 2·0, 45MM LONG, GRADE 12·9, HEX SOCKET CAP SCREW | 4 | OFF SHELF |
| 7 | RETAINING PLATE | 2 | BBWD3A |
| 6 | SIDE PLATE | 2 | BBWD2A |
| 5 | BASE PLATE | 1 | BBWD5A |
| 4 | V-BLOCK | 1 | BBWD6A |
| 3 | M6 x 1·0, 30MM LONG, GRADE 12·9, HEX SOCKET CAP SCREW | 8 | OFF SHELF |
| 2 | TOP PLATE | 1 | BBWD1A |
| 1 | M16 x 2·0, 40MM LONG, GRADE 12·9, HEX SOCKET CAP SCREW | 4 | OFF SHELF |
| PART NO. | DESCRIPTION | QUANTITY | WKG DWG. NO |

PARTS LIST

| | |
|---|---|
| QUANTITY: 1 OFF | |
| MATERIAL: TOOL STEEL | |
| DWG NO.: BBAD1A | |
| DRAWN BY: I. J. PUGH | |
| DATE: 15/6/95 | DIMENSIONS IN M |
| DESCRIPTION: BOLT PENETRATION BL | |
| PROJECT: PENETRATION BLOCK | |
| TITLE: PENETRATION BLOCK ASS'Y | |

## B.1 Bolt block calculations

**B1.1 Main assembly bolt thread engagement :-** Use M16 x 2.0 bolts, grade 12.9

To prevent stripping of the hole and bolt threads respectively, the engaged length of thread for a solid bolt must not exceed the greater of

$$0.87D_c\left[\frac{Sybolt}{Synut}\right] \qquad \text{or} \qquad 0.87D_c \qquad (22)$$

where $D_c$ = root diameter of thread

$= 13.835$ (23)

$S_y$ = yield stress

$= 1170\ N/mm^2$ for bolt

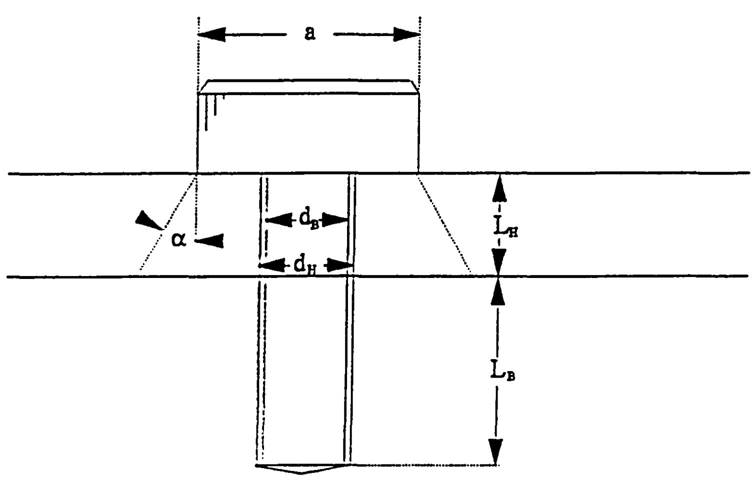
$= 700\ N/mm^2$ for nut

so $$0.87.13.835^2\left[\frac{1170}{700}\right] \qquad \text{or} \qquad 0.87.13.835$$

$= 23.470$ mm $\qquad = 12.036$ mm

therefore the engaged length of thread must exceed 23.47 mm.

**B.1.2 Calculation of preload and required bolt torque upon assembly bolts**



**B.1.3 Calculate ratio of stiffness,** $\beta = \frac{K_B}{K_H}$

where $K_B$= bolt stiffness

$K_H$= stiffness of clamped component

Method a - using equivalent cylinder for $K_H$.

$$K_B = \frac{E_B.A_B}{L_B}$$

where $E_B$= Bolt material Young's modulus

$A_B$= Area under bolt head

$L_B$= Thread engagement length

$$K_B = \frac{210x10^3 Nmm^{-2}.\pi.\left(\frac{14.701^2}{4}\right)mm^2}{24mm}$$

$$= 1.49 \times 10^6 \; Nmm^{-1}$$

$$K_H = \frac{A_H.E_H}{L_H}$$

where $A_H$= Effective area of compressed component

$$= \frac{\pi}{4}.\left[(a+L_H)^2 - d_H^2\right]$$

$$= 1002mm^2$$

therefore $$K_H = \frac{1002mm^2.210x10^3 Nmm^{-2}}{24mm}$$

$$= 8.77Nmm^{-1}$$

Ratio of stiffnesses, $$\beta = \frac{K_B}{K_H} = \frac{1.49x10^6}{8.77x10^6}$$

$$= 169.897 \times 10^{-3}$$

Maximum load induced by bolt

$$P_a = 100kN$$

Factor of safety = 1.5

Therefore $\quad P_s = 150\text{kN}$

Preload, $\quad P_o = \frac{P_s}{1+\beta} = \frac{150}{1.169897}$

$= 128.22\text{kN}$

Final bolt load, $\quad P = P_o + \frac{\beta.P_a}{1+\beta}$

$= 128.22 + \left(\frac{100.169.897\text{x}10^{-3}}{1+169.897\text{x}10^{-3}}\right)$

$= 142.74\text{kN}$

Bolt $\sigma = \frac{P}{A_b}$

$= \frac{142.74\text{kN}}{169.76\text{mm}^2}$

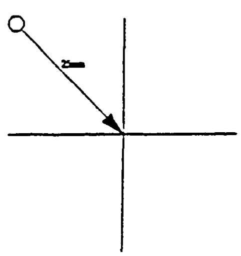
as bolt $Fy = 1170\text{Nmm}^{-2}$ $\quad$ FINE

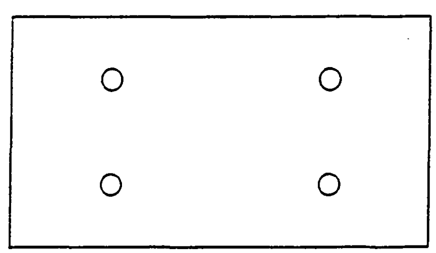
Torque for preload, $\quad T = Q.\frac{(P+6.2832.\mu.r)}{(6.2832.r-\mu.P)}.r$

$Q = 128.22\text{kN}, P = 2\text{x}10^{-3}, \mu = 0.3, r = 7.3505\text{x}10^{-3}\text{m}$

$T = 328\text{Nm}$

**B.1.4 Vice plate bolts**





Assume four bolts. Bolt torque centred on centre of vice plate. Tendency to shear fixing bolts.

At 25mm 190Nm becomes $\quad \frac{190\text{Nm}}{25\text{mm}}\left[\frac{1\text{mm}}{1\text{x}10^{-3}\text{m}}\right]$

$= 7.6\text{kN}$

Taken up by four bolts so shear stress, $\tau = \frac{7.6 \div 4}{28.278} \frac{kN}{mm^2}$

$$= 67.19 Nmm^{-2}$$

So select 4nos M6 x 1.0 bolts.

### B.2 Determining confidence levels for the performance of a standard ET20 coupler

Mean of slip results, $\bar{x} = \frac{0.13+0.16+0.16+0.17+0.18+0.16+0.14}{7}$

$$= 0.157$$

$$s = \sqrt{\frac{\Sigma x^2}{n} - \bar{x}^2}$$

$$= 0.017$$

$$\hat{s} = s.\sqrt{\frac{n}{n-1}}$$

$$= 0.017.\sqrt{\frac{7}{6}}$$

$$= 0.018$$

Degrees of freedom, d.f. = n-1 = 7-1 = 6 $\quad t_{0.05} = 2.447$

$$\bar{x} \pm t_{0.05}.x.\frac{s}{\sqrt{n-1}}$$

$$= 0.157 \pm \left(2.447x\frac{0.017}{\sqrt{6}}\right)$$

95% confidence level = 0.157 ± 0.017

99% confidence level = $0.157 \pm \left(3.707x\frac{0.017}{\sqrt{6}}\right)$

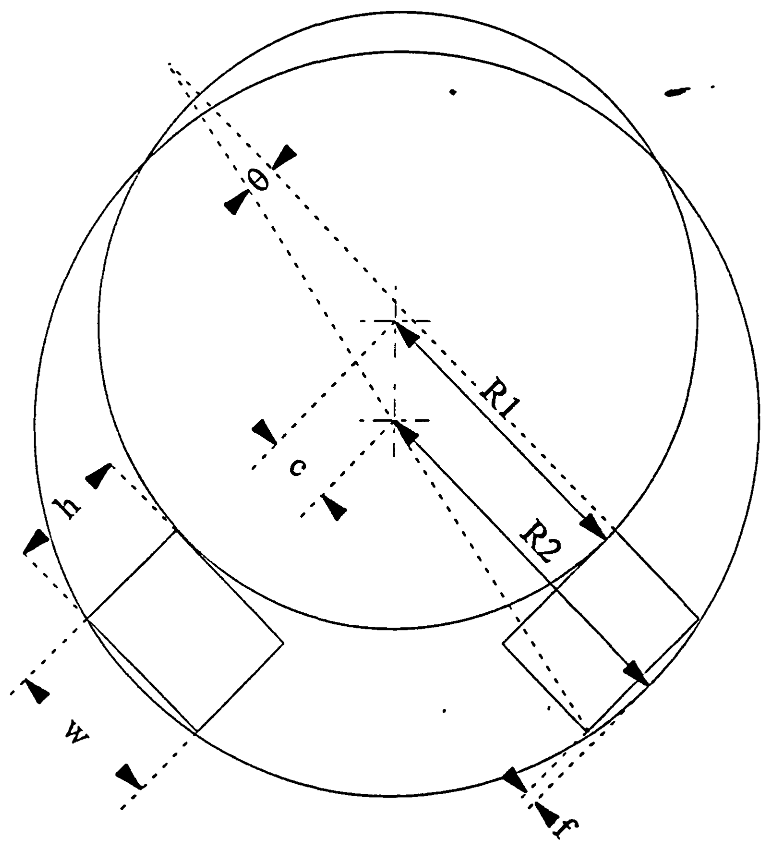
$$= 0.157 \pm 0.026$$

## C.1 Visual Basic program MBT_fit

This program was written to determine the minimum required inner tube diameter to accommodate a saddle and bar size. The geometry between the various components, i.e. tube, saddles and bar, is not related therefore an iterative method has to be used. Obviously a computer program is ideal for this application.

The program asks the user for the saddle height and width and also the bar diameter. Obviously the maximum bar diameter is entered to avoid the possibility of bar tube interference. The graphical situation shown in Figure C.1 illustrates the geometry the program code is working with.

Figure C.1 - Diagrammatic layout of saddle, bar and tube arrangement




The program code is listed in Figure C.2. Ultimately the code produces the correct inner radius R2 accurate to 0.0001mm.

This situation is true when $R2 = R1 + h + f - c$

The inner tube radius R2 is based upon the bar radius R1 plus a small increment. The saddle, width w and height h, is placed on the inner tube wall angular spaced at 90°. The bar rests upon these saddles. The angle θ is then calculated using the saddle width/tube inner diameter ratio. Using this angle the distance f is calculated. Then a value for c is calculated. This is only correct when both the top of the bar and the tube coincide, hence this is when the correct inner diameter is calculated. The value for R2 is increased incrementally in a FOR/NEXT loop until this situation is true. The loop is repeated for increased R2 accuracy from 0.1mm to 0.0001mm.

Figure C.2 - MBT_fit program code

```
Private Sub Command1_Click ()
'
   R1 = Val(Text1(0).Text) / 2
   h = Val(Text1(1).Text)
   w = Val(Text1(2).Text)
   If R1 <= 0 Or h <= 0 Or w <= 0 Then
      Exit Sub
   End If
   '
   DecPlace = .1
   Seed = R1
   '
   Do While DecPlace > .00001
      For R2 = Seed To (R1 + (10 * h)) Step DecPlace
         Fraction = (w / (2 * R2))
         'Atn(Fraction / Sqr(-Fraction * Fraction + 1)) is equivalent to
ArcSin(Fraction)
         Theta = Atn(Fraction / Sqr(-Fraction * Fraction + 1))
         f = R2 - (R2 * (Cos(Theta)))
         m = R2 - R1
         c = Sqr((m ^ 2) / 2)
         x = R1 + h + f - c
         Text2(0).Text = Format$(x - R2, "##0.0####")
         Text2(1).Text = Format$(R2 * 2, "##0.0####")
         Text2(0).Refresh
         Text2(1).Refresh
         If Abs(R2 - x) < DecPlace Then
            Exit For
         End If
      Next R2
      Seed = R2 - DecPlace
      DecPlace = DecPlace / 10
   Loop
End Sub
```

## D.1 Derivation of fatigue coupler life

The fatigue life of a coupler is derived from British Standard BS5400 : Part10 : 1980, particularly the section dealing with fatigue assessment of bridges carrying highway and railway loading.

The design life is that period in which a bridge is required to perform safely with an acceptable probability (2.3%) that it will not require repair. The design life is stated in the standard as 120 years. The number of load cycles that a component must endure in this time is dependant upon the magnitude of the cycles and the component classification.

Reinforcement bar couplers fall into classification D which is based upon welded joints between reinforcement bar ends.

The number of repetitions to failure, N, of any one stress range, $\sigma_r$, should be obtained from either of the following equations which have been plotted in Graph D.1 :

$$N \times \sigma_r^m = K_2 \qquad \text{Eq. D.1}$$

$$\mathrm{Log}_{10}N = \mathrm{Log}_{10}K_2 - (m \times \mathrm{Log}_{10}\sigma_r) \qquad \text{Eq. D.2}$$

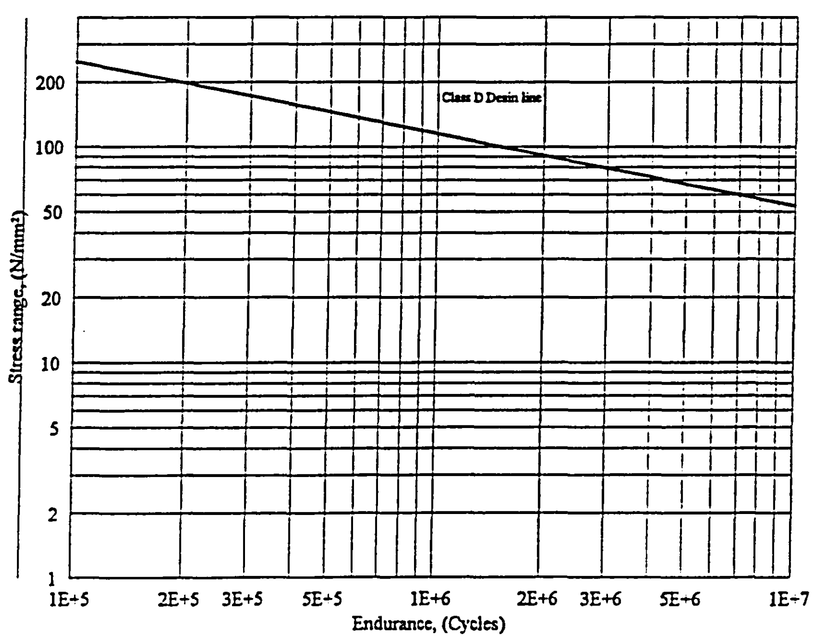
where

$N$ = the predicted number of cycles to failure of a stress range $\sigma_r$

$m$ = is the inverse slope for class D of the design line $\mathrm{Log}\sigma_r - \mathrm{Log}N$ curve

$K_2$ = is the constant term relating to a probability of failure of 2.3% within the design life

Graph D.1 - Design life S-N curve for reinforcement bar couplers



Couplers must achieve the minimum number of cycles for each stress range tested. Not all stress ranges are tested. The British Board of Agrément require test ranges of $200N/mm^2$, $160N/mm^2$ and $140N/mm^2$.

# Appendix E - Product range cost breakdown

## E.1 - ET8 cost breakdown

| | | | | | |
|---|---|---|---|---|---|
| **TUBE COST** | | | **STAMP** | 500 | per HR |
| COST | 0.083 | | STAMP COST | 0.000 | |
| SCRAP | 0.004 | | SET-UP | 0.001 | |
| **TOTAL** | **0.087** | **Units** | **TOTAL** | **0.001** | **Units** |
| **BOLTS** | | | **DRILL & TAP** | 33 | per HR |
| 8 OFF M12 | 0.180 | | LABOUR COST | 0.009 | |
| TRANSPORT | 0.004 | | TOOL COST | 0.009 | |
| **TOTAL** | **0.184** | **Units** | **TOTAL** | **0.018** | **Units** |
| **SADDLE COST** | | | **WELD** | 40 | PER HR |
| MATERIAL | 0.021 | | LABOUR COST | 0.046 | |
| MAUFACTURING | 0.029 | | CONSUMABLES | 0.005 | |
| HEAT TREAMENT | 0.029 | | | | |
| TRANSPORT | 0.004 | | **TOTAL** | **0.051** | **Units** |
| **TOTAL** | **0.083** | **Units** | **ASSEMBLY & PACK** | 70 | PER HR |
| **SAW** | 100 | per HR | ASSEMBLY LABOUR | 0.020 | **Units** |
| | | | PACK LABOUR | 0.002 | |
| CUT COST | | | | | |
| LOAD COST | 0.001 | | PACK COST | 0.001 | |
| BLADE COST | 0.001 | | | | |
| | | | **TOTAL** | **0.023** | **Units** |
| **TOTAL** | **0.002** | **Units** | | | |
| | | | **TOTAL COST** | **0.449** | **Units** |

## E.2 ET10 cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 0.083 | |
| SCRAP | 0.004 | |
| **TOTAL** | **0.087** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.180 | |
| TRANSPORT | 0.008 | |
| **TOTAL** | **0.188** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.017 | |
| MAUFACTURING | 0.026 | |
| HEAT TREAMENT | 0.027 | |
| TRANSPORT | 0.004 | |
| **TOTAL** | **0.074** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.031 | |
| TOOL COST | 0.009 | |
| **TOTAL** | **0.040** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.031 | |
| CONSUMABLES | 0.003 | |
| **TOTAL** | **0.034** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.020 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.008 | |
| **TOTAL** | **0.029** | **Units** |
| **TOTAL COST** | **0.454** | **Units** |

## E.3 ET12 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 0.117 | |
| SCRAP | 0.006 | |
| **TOTAL** | **0.122** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.270 | |
| TRANSPORT | 0.012 | |
| **TOTAL** | **0.282** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.017 | |
| MAUFACTURING | 0.046 | |
| HEAT TREAMENT | 0.040 | |
| TRANSPORT | 0.009 | |
| **TOTAL** | **0.112** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.035 | |
| TOOL COST | 0.013 | |
| **TOTAL** | **0.048** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.035 | |
| CONSUMABLES | 0.003 | |
| **TOTAL** | **0.038** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.020 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.008 | |
| **TOTAL** | **0.029** | **Units** |
| **TOTAL COST** | **0.634** | **Units** |

## E.4 ET16 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 0.199 | |
| SCRAP | 0.010 | |
| **TOTAL** | **0.209** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.276 | |
| TRANSPORT | 0.015 | |
| **TOTAL** | **0.291** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.024 | |
| MAUFACTURING | 0.052 | |
| HEAT TREAMENT | 0.058 | |
| TRANSPORT | 0.009 | |
| **TOTAL** | **0.143** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.042 | |
| TOOL COST | 0.018 | |
| **TOTAL** | **0.060** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.035 | |
| CONSUMABLES | 0.003 | |
| **TOTAL** | **0.038** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.020 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.009 | |
| **TOTAL** | **0.031** | **Units** |
| **TOTAL COST** | **0.774** | **Units** |

## E.5 ET25 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 0.437 | |
| SCRAP | 0.022 | |
| **TOTAL** | **0.459** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.502 | |
| TRANSPORT | 0.012 | |
| **TOTAL** | **0.513** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.051 | |
| MAUFACTURING | 0.098 | |
| HEAT TREAMENT | 0.062 | |
| TRANSPORT | 0.009 | |
| **TOTAL** | **0.220** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.046 | |
| TOOL COST | 0.027 | |
| **TOTAL** | **0.073** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.035 | |
| CONSUMABLES | 0.003 | |
| **TOTAL** | **0.038** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.020 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.012 | |
| **TOTAL** | **0.034** | **Units** |
| **TOTAL COST** | **1.340** | **Units** |

## E.6 ET28 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 0.805 | |
| SCRAP | 0.040 | |
| **TOTAL** | **0.846** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.637 | |
| TRANSPORT | 0.012 | |
| **TOTAL** | **0.649** | **Units** |
| **SADDLE COST** | — | |
| MATERIAL | 0.036 | |
| MAUFACTURING | 0.052 | |
| HEAT TREAMENT | 0.062 | |
| TRANSPORT | 0.009 | |
| **TOTAL** | **0.160** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.046 | |
| TOOL COST | 0.033 | |
| **TOTAL** | **0.079** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.035 | |
| CONSUMABLES | 0.003 | |
| **TOTAL** | **0.038** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.028 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.012 | |
| **TOTAL** | **0.042** | **Units** |
| **TOTAL COST** | **1.816** | **Units** |

## E.8 ET40 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 1.081 | |
| SCRAP | 0.054 | |
| **TOTAL** | **1.135** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 0.845 | |
| TRANSPORT | 0.012 | |
| **TOTAL** | **0.857** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.132 | |
| MAUFACTURING | 0.141 | |
| HEATTREAMENT | 0.125 | |
| TRANSPORT | 0.009 | |
| **TOTAL** | **0.406** | **Units** |
| **SAW** | 100 | per HR |
| CUTCOST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.055 | |
| TOOL COST | 0.045 | |
| **TOTAL** | **0.100** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.046 | |
| CONSUMABLES | 0.005 | |
| **TOTAL** | **0.051** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.028 | Units |
| PACK LABOUR | 0.002 | |
| PACIK COST | 0.033 | |
| **TOTAL** | **0.064** | **Units** |
| **TOTAL COST** | **2.615** | **Units** |

## E.9 ET50 Cost breakdown

| | | |
|---|---|---|
| **TUBE COST** | | |
| COST | 2.365 | |
| SCRAP | 0.118 | |
| **TOTAL** | **2.483** | **Units** |
| **BOLTS** | | |
| 8 OFF M12 | 1.230 | |
| TRANSPORT | 0.012 | |
| **TOTAL** | **1.241** | **Units** |
| **SADDLE COST** | | |
| MATERIAL | 0.291 | |
| MAUFACTURING | 0.295 | |
| HEAT TREAMENT | 0.249 | |
| TRANSPORT | 0.018 | |
| **TOTAL** | **0.853** | **Units** |
| **SAW** | 100 | per HR |
| CUT COST | | |
| LOAD COST | 0.001 | |
| BLADE COST | 0.001 | |
| **TOTAL** | **0.002** | **Units** |

| | | |
|---|---|---|
| **STAMP** | 500 | per HR |
| STAMP COST | 0.000 | |
| SET-UP | 0.001 | |
| **TOTAL** | **0.001** | **Units** |
| **DRILL & TAP** | 33 | per HR |
| LABOUR COST | 0.069 | |
| TOOL COST | 0.062 | |
| **TOTAL** | **0.131** | **Units** |
| **WELD** | 40 | PER HR |
| LABOUR COST | 0.069 | |
| CONSUMABLES | 0.007 | |
| **TOTAL** | **0.076** | **Units** |
| **ASSEMBLY & PACK** | 70 | PER HR |
| ASSEMBLY LABOUR | 0.039 | **Units** |
| PACK LABOUR | 0.002 | |
| PACK COST | 0.003 | |
| **TOTAL** | **0.044** | **Units** |
| **TOTAL COST** | **4.833** | **Units** |