Design characteristics of a pipe crawling robot

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DESIGN CHARACTERISTICS OF A PIPE CRAWLING ROBOT

by

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ABSTRACT

This thesis deals with the design characteristics of a pipe crawling vehicle which utilises a unique, innovative and patented drive system. The principle of the drive system is simple. That is, if a brush is inserted into a pipe and its bristles are swept back at an angle, then, it is easier to push the brush forwards through the pipe than it is to pull it backwards. Thus, if two brushes are interconnected by a reciprocating cylinder, then, by cycling the cylinder, it is possible for the vehicle to "crawl" through the pipe. The drive mechanism has two main advantages. The first is the ability of the bristles to deflect over or around obstacles, thus, the vehicles can be used in severely damaged pipes. Secondly, the drive mechanism is able to generate extremely high "grip" forces, thus, the vehicle has a high payload to weight ratio. This "simple" traction mechanism has subsequently been proven to be extremely capable in significantly hostile environments, for example, nuclear plants and sewers.

The development of the vehicle has resulted in brushes being considered as "engineering" components. This thesis considers the forces present when a brush moves forward through a pipe, further, it also considers the forces present if the brush is required to grip the walls of the pipe. A "simple" cantilever model has been developed which predicts the force required to push a brush forwards through the pipe. A second model has been developed which predicts the forward to reverse or "slip" to "grip" ratio of a brush, for given frictional conditions. This model is deemed satisfactory up to the onset of bristle buckling.

The experimental program determined three factors, they were, the force required to load a brush into a pipe, the force required to push a brush forward through a pipe and the reverse force a brush could support prior to failure.

It can be concluded that this vehicle, through its tractive capability and environmental compliance, is able to traverse irregularly shaped pipes. Ultimately, this allows tooling to be transported and used at previously unobtainable positions within such pipes.

To Trudy, James and my parents.

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"Thanks" to the Cairngorm mountains in Scotland where both my thoughts and body were often found, perhaps a little too often!. I shall never forget this period of my life, nor the exceptional people with whom I have been privileged to work.

DECLARATION

This thesis is the result of my own work. No part of the thesis has been submitted for any other degree in this or any other University.

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NOTATION

AMR	Autonomous Mobile Robot
CSA	Cross Sectional Area
DOF	Degrees of Freedom
E	Modulus of Elasticity
EI	Flexural Stiffness
SEM	Scanning Electron Microscope
F	Force
F_N	Normal Force
Gpa	Giga Pascals
HP	Horse Power
I	Second Moment of Area
ID	Internal Diameter
ISO	International Standards Organisation
L	Length
L_c	Chord Length
N	Newton
Pa	Pascal
PCD	Pitch Circle Diameter
PCT	Patent Co-operation Treaty
pd	Potential Difference
PTFE	Polytetrafluorethylene

PVC	Polyvinyl Chloride
R_a	Surface Roughness
R_L	Length of Radius
SBM	Shaft Boring Vehicle
TBM	Tunnel Boring Vehicle
TIV	Tethered Inspection Vehicle
V_L	Cantilever Deflection Length
W	Load
μ	Friction
μF	Friction Force
1.5D	Tightness of a bend, that is, the radius is 1.5 times the ID

CHAPTER 1

INTRODUCTION

1.1 AIMS AND OBJECTIVES

This thesis deals with the design characteristics of a pipe crawling vehicle which utilises a unique, innovative and patented drive system. The principle of the drive system is simple. That is, if a brush is inserted into a pipe and its bristles are swept back at an angle, then, it is easier to push the brush forwards through the pipe than it is to pull it backwards. Thus, if two brushes are interconnected by a reciprocating cylinder, then, by cycling the cylinder, it is possible for the vehicle to "crawl" through the pipe.

The development of the vehicle has resulted in brushes being considered as "engineering" components. This thesis considers the forces present when a brush moves forward through a pipe, further, it also considers the forces present when the brush is required to grip the walls of the pipe. Different pipe conditions may require different bristle/brush configurations to ensure adequate traction can be obtained. This thesis analyses some different pipe conditions and attempts to determine the most suitable bristle/brush designs.

1.2 BACKGROUND

The wheel was invented in approximately 3500 BC and since that time man has maintained an interest in locomotion. The twentieth century has seen significant advances in transportation and vehicle development. However, with many of these advances came the need for correspondingly smooth and uniform surfaces or roadways on which these vehicles could successfully operate. At present, a large percentage of the World's surface is considered unsuitable for any type of ground operating mechanised vehicle. The Earth and nearby planets, including the Moon and Mars, have considerable inhospitable land mass areas. The type of vehicle traction system required to cope with such environments may be different to that previously considered as a "conventional" traction system. Exploration by means of wheeled vehicles is in the main impractical or severely limited, thus, current research is being undertaken into other forms of locomotion.

A number of bodies are currently undertaking research into the various forms of locomotion. Of specific interest is legged vehicles using walking locomotion. Legged vehicles are able to obtain specific footholds via leg lift and place mechanisms, allowing the foot to be placed in an exact location and thus avoid obstacles. The wheel, when subjected to similar irregular terrain must be driven over or around obstacles or the vehicle risks being unable to continue its forward passage. Considerable research has already been undertaken into both wheeled and legged vehicles through "terratechnology", the study of the interface between the traction inducing component of a vehicle and its immediate environment. The majority of research in terratechnology has focused upon the reaction of the environment to traction, making extensive use of soil mechanics. However, the performance of the traction inducing drive mechanism itself is an area which has seen little research. Interest in man-made environments has moved research towards gaining an understanding of the mechanisms for climbing and traversing shafts and tunnels. Ill-constrained environments can be a natural cave or a man-made mine shaft. Vehicles or robots that climb are more reliant upon frictional forces than walking vehicles that traverse a horizontal plane. Gravity acting on a climbing vehicle has the effect of increasing the risk of slippage through shearing between the leg/wall interface. Hence, climbing vehicles require a reliable mechanism to prevent slippage, for example, additional legs. Thus, sufficient legs are required to satisfy the kinematic requirements and sufficient grip is required to overcome the gravitational force.

Early research indicated the complexities of utilising legged motion for traversing ill-constrained environments and many researchers are currently working on the problems relating to the control of such a vehicle. Wheeled vehicles do not require such complex control algorithms, however, other problems do exist, for example, loss of traction, insurmountable objects and risk of grounding the vehicle body. Literature reviews indicated that little research had been undertaken into vehicles that are able to traverse shafts and pipelines especially if the surface conditions of the environments are irregular. The type of traction inducing mechanism that would be required for such an environment was to form the nucleus of this current research.

1.3 TRACTION

For the purpose of this thesis *Traction* means the ability of the bristle driven vehicle to draw or pull a payload and/or umbilical. *Grip* refers to the vehicle's bristles being able to obtain a satisfactory purchase with the pipe wall whilst dragging a given payload and/or umbilical. The term *Drive Force* is the force a given fluid powered cylinder is able to generate to ensure the payload is able to be dragged.

Traction is important especially for off-road vehicles, specifically vehicles involved in agriculture, construction, mining, military operations and exploration. Prior to the publication of M G Becker's "*Theory of Land Locomotion*" in 1956 and his "*Off the Road Locomotion and Introduction to Terrain Vehicle Systems*" during the 1960's, development of this type of vehicle tended to be based on unscientific methods. Publication of these two works stimulated significant interest in this type of research, now known as "*Terramechanics*" Wong (1989).

1.3.1 TERRAMECHANICS

Terramechanics is the study of a mechanical vehicle and its interaction with the terrain it is traversing. There is a strong relationship between the wheel-terrain interaction and the topic commonly understood as "Soil Mechanics". Wong (1989) states that *Terramechanics* can be considered as having two main branches; the first, *terrain vehicle mechanics*, considers the tractive performance of a vehicle over unprepared terrain and additionally considers ride quality, handling, obstacle negotiation and water crossing. The second branch, *terrain implement mechanics* focuses in the main on the functionality of soil cultivating and earth moving equipment. An additional element has recently been introduced, although nameless at present, it considers the damage and impact such mechanised vehicles inflict onto the land, including such factors as soil compacting. Wong summarises the role of terramechanics succinctly in his book *"Terramechanics and Offroad Vehicles"* by means of a diagram reproduced in Figure 1.1

As the Earth's natural resources continue to become depleted, robots that are able to cope with such environments will become more necessary. Ultimately, it may prove necessary to explore and mine for valuable natural resources on planets other than our own. The Lunar Roving Vehicle for the Apollo space mission is a notable example of the inclusion of terramechanic design methodology into a successful finished vehicle, *(Bekker, 1964, 1967, 1969, 1981)*. A number of different traction possibilities were considered for the above project including, walking machines with various gaits and screw-driven vehicles, as well as a number of tracked and wheeled designs. Finally, a four-wheel drive vehicle employing a unique tyre design was chosen. The tyre was woven with steel wire and surrounded with titanium chevrons. The tyre produced favourable elasticity, traction, strength, lightness and durability. The vehicle also operated well in extremes of vacuum and temperature.

1.3.2 TRACKED VEHICLES

Track drive as known today, was originally conceived in the 18th century as a "portable railway" and tracked vehicles have been used widely since the turn of this century, Wong (1989). The fundamental advantage of tracked vehicles is their ability to induce greater traction over varied terrain, for example, their performance in sodden conditions is greatly enhanced over that of the wheeled vehicle. In a tracked vehicle the load is spread across a greater surface area, thus, the tracked vehicle is less likely to sink, further, it is able to generate increased traction, for example, a sandy beach. This is especially true across terrain that is prone to shearing. A tracked vehicle's performance can be defined by its motion resistance, tractive effort, drawbar pull and tractive efficiency, all of which are functions of slip. Slip, in turn is directly related to the normal and shear stress distributions at the track-terrain interface. Wong (1989) makes a special note in his book: *"It is generally recognised that the interaction between a tracked vehicle and terrain is very complex and is difficult to model accurately"*.

1.3.3 WHEELED VEHICLES

The problem of the interaction of the wheel with the terrain still remains an extremely complicated one, as it is influenced by a large number of variables. The soil can flow under the wheel in complex fashions, the flow patterns of the soil vary depending upon the tractive force and/or slip of the wheel, Wong (1989). Due to wheel-terrain interaction, normal and shear forces develop at the interface, however, their placement across the contact area varies depending upon the design and operational variables of the wheel and the terrain conditions. Variables can include temperature, wheel material, terrain type and makeup, tread pattern, contact area, diameter of wheel, terrain undulation and vehicle mass. Naturally occurring terrain, where man has yet to make inroads, cause the most problems to vehicle mobility. For example, snow, bog or mud are not generally homogeneous and are easily compressible. This results in further complications for the wheel-terrain interaction.

1.4 AUTOMATIC VEHICLES

The term *Autonomous Mobile Robot* (AMR) refers to a manipulator or a group of manipulators that are dedicated to performing a task independent of human interaction. *"Mobile"* refers to their ability to move, generally using wheels or tracks to give the tractive element. *Autonomy* in this situation is considered to mean the ability of the machine to make independent, intelligent decisions as the terrain demands. This can be summarised by stating that a human operator is required to initiate a move instruction and the robot is left to execute the move whilst taking into consideration its immediate environment.

In July 1997 a mobile robot played a substantial part in the United States Pathfinder mission to Mars, the small sampling robot known as ROVA was controlled from Earth. ROVA incorporated laser guidance, this allowed the vehicle to distinguish features of the terrain, for example, rocks and ravines. Ill-constrained environments are coming to the forefront of mobile robot exploration and this is one area of research still relatively untouched. In the future, this area of research may prove invaluable in producing vehicles that can be utilised in deserts, jungles, shafts, tunnels or on extra-terrestrial surfaces. Autonomous mobile robots could eventually be used in the rescue of people from extreme terrain conditions, for example, mountain, earthquake or cave rescue operations.

1.5 SHAFT AND TUNNELLING ROBOTS

Within the mining industry the boring of shafts or tunnels is carried out using large tunnel boring machines (TBM's) that are able to clamp to the shaft wall via a circumfrential arrangement of clamps, similar to the brake shoes on a car. The clamp arrangements exist at the front and back of the TBM body. The body is able to pass through these clamps, hence, it is able to move through the shaft slowly as the boring proceeds this occurs via a series of slide and clamp cycles, Goodell (1991). This traction method is slow and is dedicated to the diameter of the shaft or tunnel that is being bored.

A shaft climbing robot utilising conventional legs would require complex control algorithms. A complex sensory ability would also be necessary, for example, vision or other guidance system. An array of foot sensors analysing position and force may also be required. Progress though the shaft would be slow and dependence upon sensor feedback would be significant. The gravity force, so useful for wheeled or tracked robots moving in the horizontal plane, would hinder the movement of this type of vehicle. Technologies for shaft climbing or traversing robots, autonomous or not, have considerable technical problems to overcome.

The legs of such a robot would require considerable flexibility, thus, a number of joints would be required and this combination would have considerable mass. Therefore, the torque that would be required at the body-leg joint would be considerable and necessitate large electric motors and gearing, in turn a considerable power supply would be required.

1.6 COMMERCIAL PIPE PIGS

Commercial pipe pigging is a well-developed technology with many thousands of Kilometres of pipeline being inspected, cleaned or maintained annually. Propulsion or traction is derived from using the differential pressure of the pipeline product acting upon polyurethane sealing cups attached to the pig's body. This type of pipeline pig is rarely tethered to an umbilical but is usually "free-swimming" and is thus able to inspect many hundreds of Kilometres at a time. These pigs generally inspect pipelines of known internal diameters and profile, although slight internal diameter variances are normal. However, these pigs are unsuitable for exploration of ill-constrained environments because they are unable to accommodate changes in the environment and require the flow of the product for propulsion.

1.7 GAIT

Within the animal kingdom many animals utilise two, four, six, eight (or more) legs. The sequence and number of legs placed at a given time is known as the gait. A tortoise, for example, only moves a single leg at one time. This allows the remaining three legs to form a stable support. A horse utilises three separate gaits during the walking, trotting and galloping phases of its motion. In these more sophisticated gaits, say, a galloping horse, the animal may have only one leg in contact with the ground giving the tractive force. In this case it is necessary for the animal to have a sophisticated dynamic awareness and control of its motion. Some insects with six legs, three per body side, move the front and rear legs on one side together with the middle leg of the other side, this always leaves a stable tripod of legs for support. Climbing gaits seem to have been totally neglected to date and the initial consideration of the shaft climbing problem related to possible climbing gaits. Hence, for technical and design reasons the concept of a legged climbing machine was abandoned.

1.8 A NOVEL ALTERNATIVE

There are a number of fundamental problems associated with the traversing of terrain, whether highly constrained or ill-constrained. The problem being the terrain complexity and the tractive method required to successfully negotiate such terrain. This research is concerned with the investigation and development of an exploration robot that utilises a novel traction mechanism. The robot is able to ascend, descend and traverse shafts, tunnels, pipework, mine workings and sewers, essentially ill-constrained and hostile environments. Mine workings, sewers and bore holes are considered the most hostile of man-made shafts and tunnels. The fundamental traction principle is also applicable to the traversal of external surfaces, for example, pipes, cables or the legs of marine structures. Once again, many of these environments are hostile to humans. The traction mechanism can also be used to traverse between parallel plates, for example, for the inspection of the inner and outer skins of marine supertankers.

The innovative and novel section of this research lies within the unique traction mechanism. The traction mechanism is passive, it is designed to harness the severity of its environment in order to increase its tractive effort, thus eliminating the numerous control problems previously associated with other types of exploration vehicle. The principle of the drive mechanism is extremely simple, in its basic form it consists of a brush at each end of a reciprocating actuator. In one particular configuration, the bristles of the brush fan out radially from a cylindrical core. The brush diameter is greater than the shaft or pipe into which it is inserted. As the "robot" is inserted into the pipe the bristles become swept back. Even a lay person will be familiar with the fact that it is usually easier to insert a brush than it is to withdraw a brush against the sweep of the bristles. This could be termed the *"test-tube cleaning"* analysis. The recognition that the force required to advance a brush in a pipe is less than the force required to draw it back is the basic principle of this novel traction mechanism. However, although the mechanism can be appreciated at a subjective level, optimisation of the traction forces requires a more substantial understanding of the mechanism.

After insertion, the bristles form an interface between the pipe wall and brush core. The vehicle is reciprocating in nature. The traction process begins when the actuator pushes the leading brush further into the pipe, in doing this the rear set of bristles become subject to a reaction force and act as compliant struts, see Figure 1.2. The natural spring in the bristle material always ensures that the bristles are pushed out into the shaft wall. The reciprocating nature of the axial drive cylinder ensures that the mode of each brush is passively switched from forward "slip" to "rear" grip and back again. As a multitude of bristles are used at both the front and rear of the vehicle, it can be guaranteed that a large percentage will obtain a suitable foothold. When the actuator is reversed the trailing brush is drawn further into the pipe and the leading brush is subject to a reaction force. This now completes the traction cycle.

It has been found that the careful selection of bristle materials and hence, their mechanical properties, allows a vehicle to be designed that is capable of delivering a large "payload" to vehicle weight ratio. The loads can be transported both vertically and horizontally. Additionally, the bristles allow the vehicle to remain highly compliant within its environment. Thus, the machine is able to cope with shafts and pipework of varying diameters and profiles as well as surface conditions, for example, inclusions, omissions, offtakes and partial collapses.

1.9 CONCLUSION

The initial review of current research into robots for ill-constrained environments revealed that the largest problem appeared to be that of the control of such kinematic and dynamic vehicles. This research has concentrated on developing a new and fundamentally passive tractive drive mechanism for such a vehicle. The use of bristles for the traction mechanism is believed to be unique and prior to this research, bristles, which were formed into brushes were rarely considered to be "engineered" components.

This research centres on the analysis of the bristles when they are subject to the alternating loads experienced whilst the drive mechanism is employed. Optimisation of such a traction mechanism allows the payload capability of such a vehicle to be improved. Traction variations under different pipe conditions have also been reviewed, for example, changing internal pipe diameters, bristle length, bristle material, pipe wall roughness and any lubrication present.

Pipeline Integrity International Limited (PII), the main sponsor of the research, together with the University of Durham have subsequently funded the patenting of the brush traction principle on a world-wide basis. British Nuclear Fuels Limited (BNFL) have also contributed to the vehicles development and have provided interesting and challenging applications.

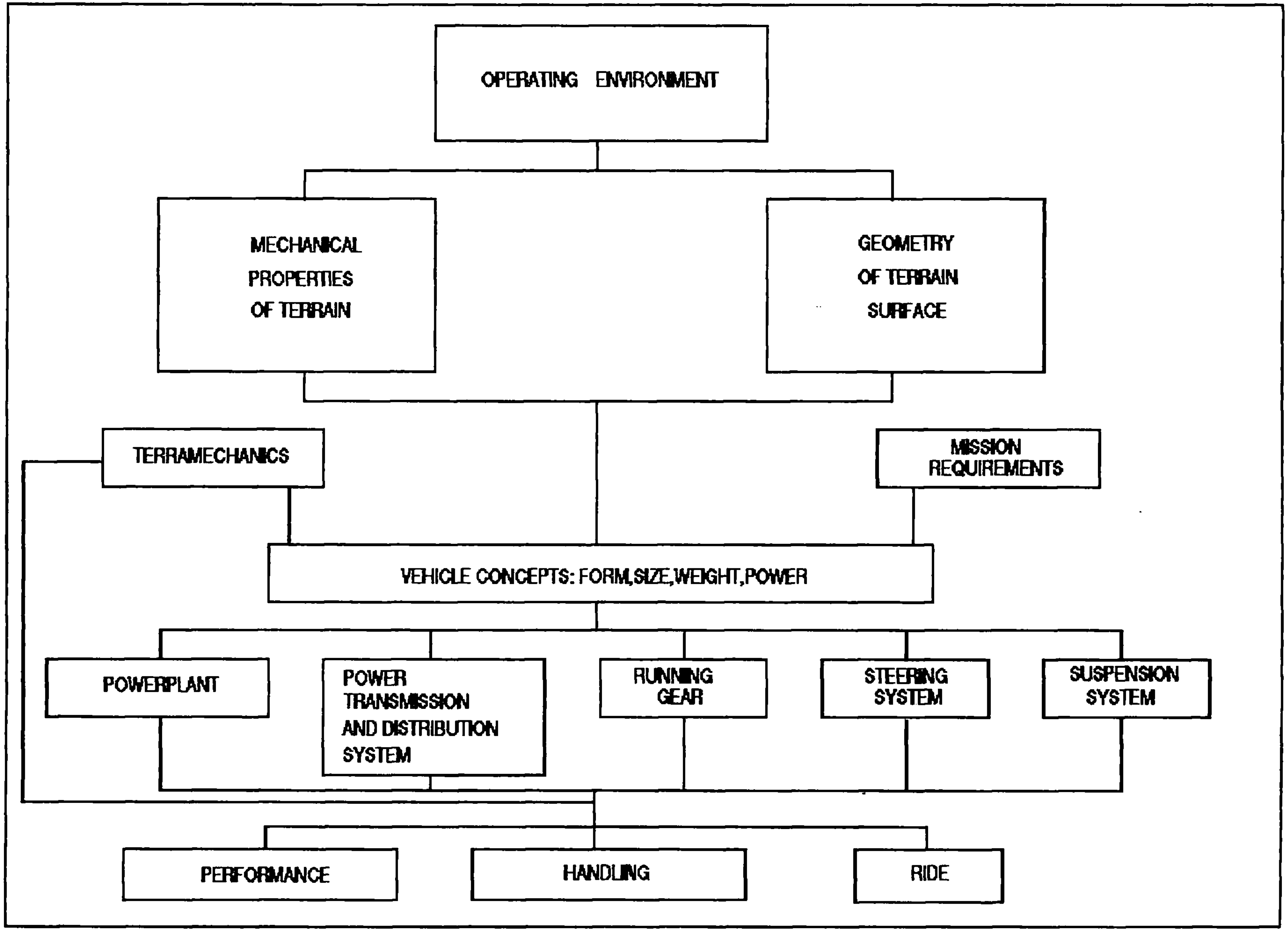


FIGURE 1.1 **ROLE OF TERRAMECHANICS** *TERRAMECHANICS AND OFF-ROAD VEHICLES J.Y. WONG*

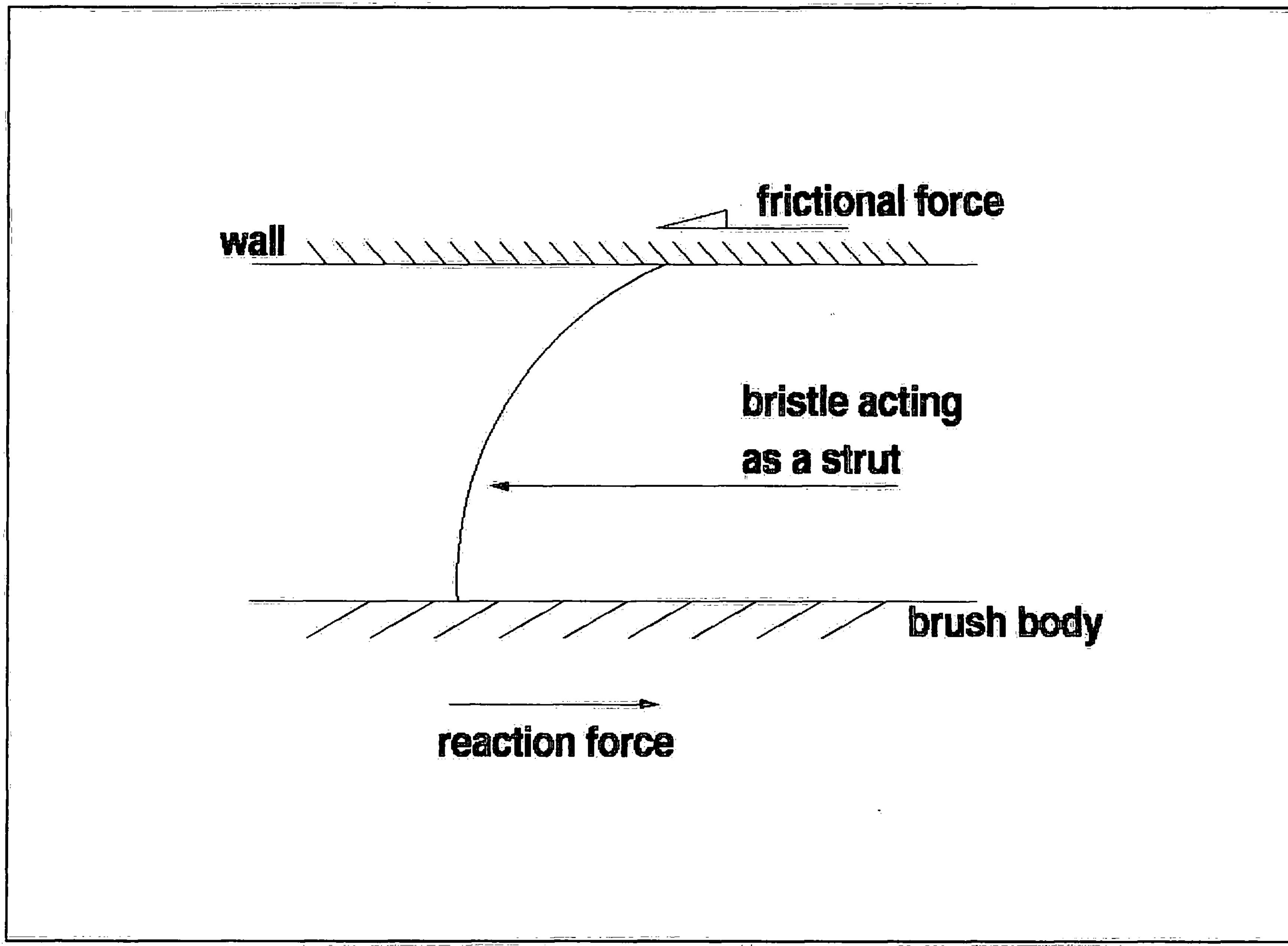


FIGURE 1.2 AN INDIVIDUAL BRISTLE ACTING AS A STRUT

CHAPTER 2

SHAFT AND TUNNEL ROBOTS: A CONSIDERATION OF CONCEPTS

2.1 INTRODUCTION

One area of application of robotics that is still wide open to "new" ideas is that of "exploration", both above and below ground. This chapter considers the problems associated with traversing non uniform terrain, specifically, shaft and tunnel negotiation. There are a large number of factors that influence the design and performance of such a vehicle. These factors are primarily external in nature, for example, shaft shape and wall materials. These factors can be dealt with through the careful design of the individual vehicle, notably its traction mechanism.

2.2 TRACTION MECHANISMS

As described in Chapter One an established device for boring and traversing shafts and tunnels is the Tunnel Boring Machine (TBM). A Shaft

Boring Machine (SBM) is a modified version of the TBM principle which allows the vertical boring of shafts. The SBM is able to both lower and raise itself via a system of grip rings and "walking cylinders" (Goodell 1991). Figure 2.1 illustrates the layout of an SBM. The upper and lower gripper rings clamp the shaft wall, in doing so they support the machine body. The lower gripper ring is an integrated part of the machine's body. A walking cycle would consist of the upper grip ring retaining its contact with the shaft wall as the lower ring is released. The walking cylinders extend, in doing so the machine's main body progresses up the shaft. The upper gripper ring is then released whilst the lower ring retains contact with the wall and the walking cylinders are closed. This completes the cycle.

The SBM's main purpose is for the boring of shafts, not shaft or tunnel exploration, however, it is able to ascend or descend for manoeuvrability purposes. As a shaft exploration vehicle the SBM would suffer some significant limitations. Firstly, it would be dedicated to the shaft diameter it had cut. Secondly, it would be slow and dependent upon the security of the shaft wall strata for adequate support.

Various devices for drilling and inspection of smaller boreholes and pipes use similar traction mechanisms. A recent development being the "Pipecat" or "PipeTrain" distributed by the Norwegian company AGR (GRØNER OFFSHORE I&M AS), Elbertse (1994). The drive mechanism is similar to that of the SBM. However, the AGR machine consists of a forward and rear clamping unit, with a section in the middle called a "travel unit". To obtain a drive cycle the rear clamping unit extends and clamps the pipe wall, the travel unit is then extended and the forward clamping unit extends and grips the wall. The rear clamping unit is then withdrawn and the travel unit closes. This completes the cycle. Figure 2.2 illustrates the principle.

2.3 ASCENDING/DESCENDING CONSIDERATIONS

A horizontal walking vehicle that loses traction and slips may cease to move forward, however, a climbing robot that loses traction will slip and fall to the bottom of the shaft. Based upon the above consideration, areas of potential traction difficulties have been identified. To ascend or descend a shaft requires a number of functional areas to be integrated.

Horizontal traversing vehicles are able to use gravity, in this case, a beneficial downward force which acts upon the friction conditions at the vehicle-terrain interface to increase horizontal traction forces. In contrast, a climbing vehicle is critically dependent upon the establishment of clamping and friction forces at the shaft wall interface. A machine designed to ascend or descend a shaft is required to overcome or redirect the effect of the gravity force. The redirection of the gravity force into a useful component may be difficult but not impossible.

2.4 THE MACHINE - WALL INTERFACE

A problem common to any climbing vehicle is obtaining sufficient traction at the foot and shaft wall interface. The force required to push a conventional flat foot against the shaft wall, via a leg, must be sufficient to support the weight of the vehicle and ensure that limiting friction is not exceeded. Conventional "leg" and "foot" arrangements have a number of fundamental drawbacks. Normally, adequate support forces can be generated but the shaft wall may not be able to provide an adequate reaction force. For example, the wall may compress or shear, resulting in instability and wall failure. Additional considerations include size, shape, cross-sectional area and surface profile (footprint) of the foot. Note the above assumes a static situation. If a vehicle of this type was descending a shaft or pipe then the leg/foot would have to generate additional forces to overcome the momentum of the vehicle due to it descending, that is, additional forces to ensure adequate deceleration of the vehicles body.

The gripper ring method of support, as used on the SBM, has a considerable advantage over a "legged" climbing vehicle. The SBM is able to apply the required vehicle holding force over a large surface area. Thus, localised wall conditions become less important and the potential for wall collapse, shear or compression is significantly lowered. However, the SBM also has a number of limitations, as previously mentioned. The main disadvantage being its lack of flexibility. A "legged" climbing vehicle is able to "decide" where to place its foot on the wall, the decision would be made "intelligently" through a series of on-board sensors incorporating feedback. However, a legged climbing vehicle has the disadvantage that it must place its "foot" within the limits of a "safety envelope". Should the vehicle place a foot outside the envelope then its gait may become unstable resulting in a fall. Thus, if a "legged" vehicle is to overcome some of the difficulties associated with shaft climbing then the interface between the foot and shaft is of fundamental importance for both traction and security.

2.5 ADDITIONAL "LEG" CONSIDERATIONS

"Conventional" legged vehicles are generally unsuitable for the exploration of pipelines or shafts due to their reliance on gravity. In shaft climbing, any leg exerting a force will require an equal and opposite force, from the strata or pipe, to maintain equilibrium and hold the robot safely in the shaft. The friction condition between the foot and wall interface is also important, foot design and shaft wall material will determine whether sufficient friction can be generated to secure the vehicle. In a climbing gait, it might be expected that most vehicles will move a single leg or group of legs at a given time. Thus, a legged climbing vehicle would require highly complex mechanics and control. In a vertical shaft or pipeline an incorrect phasing of the rise and fall gait could result in an unstable situation occurring, resulting in a fall.

An unstable situation may also occur where a leg contacts the shaft wall but the wall compresses or the foot is placed in a void. Assuming this leg was part of a tripod of forces, the two remaining legs will continue to extend in order to attempt to obtain a suitable foothold for the leg which is located in the void. Should this leg still fail to obtain a foothold, the two opposing legs will reach their maximum extension. The result is a vehicle that is no longer stable, neither centrally or vertically orientated. None of the three legs have now secured a safe hold due to the void or the wall's compressibility. The result is a tilted, unstable or falling vehicle.

As part of a preliminary investigation, the minimum number of legs required to ascend or descend a shaft was considered. From a kinematic point of view the minimum number of legs would be nine. A triple three legged gait would be used. Three legs, equally spaced at 120 degrees apart would project from the bottom and top of the vehicles body, with the third set located around the centre of the body, Figure 2.3 illustrates the layout. The machine, once inserted into the shaft, would centralise itself, via the nine legs. When the machine was required to take a "step" the upper group of three legs in the shaft would withdraw from the wall and the upper walking cylinder, in line with the vehicles body, would extend. The upper three legs would then relocate themselves in their new position. The central group of three legs would then withdraw and the upper walking cylinder would retract simultaneously with the lower walking cylinder extending. The central group of three legs would relocate themselves in their new position. Finally, the lower group of three legs would withdraw and the lower walking cylinder would retract. To complete the cycle the lower legs would extend to reposition themselves. At any given time the vehicle remains kinematically stable through its double tripod arrangement of legs.

It could be possible to reduce the number of "legs" to "two" by using the air bag principle, this configuration would not be kinematically stable. However, due to the length of the vehicles body it would only become unstable when the top air bag is released from the shaft wall. During this process the vehicle may begin to rotate but the rotation would be small because of its body length and a point would occur where the deflated bag would contact the shaft wall. This would not occur when the lower air bag is deflated. In this case the vehicle would linearise itself because of gravity, assuming the shaft was vertical.

2.6 PASSIVE OR DYNAMIC TRACTION

As previously mentioned, many of the designs that could arise from a simple consideration of kinematics and climbing gaits would rely upon complex control methods. Ultimately, this would result in complex design and manufacture problems, particularly with respect to the power to weight ratio of the vehicle. <u>At this stage consideration was given to a vehicle that was more passive and flexible in nature and through its robustness able to deal with ill-constrained environments.</u>

In many fields of engineering the world of nature can often yield potential solutions to complex mechanical problems. Two particular methods of traction appeared to be worthy of further consideration. The first was the millipede, which, at any given time may have a large percentage of its many legs in contact with the ground, yet forward movement is still able to occur. The control associated with each leg appears "limited", each following a continuous loop "waveform" pattern. The second natural mechanism was that of a worm negotiating a passage through soil grains in which it swells the front portion of its body and grips the passage wall and then contracts so that its tail segments move forward. These tail segments then swell and grip the wall, the front segment is released and the worm extends its body once again. The cycle is then repeated. The worm's principle of movement is not dissimilar to that of the SBM described earlier.

The above *"worm"* drive principle is not dissimilar to devices used in colonoscopy. The human colon is a long channel, the shape, diameter, texture and surface roughness of which change throughout its length. The colon can often be totally collapsed, that is, folded over on itself. Of significant importance is that whatever traction inducing mechanism is used it must not damage the surface of the colon. Toshio Fukuda, et al (1989) devised a vehicle that used rubber gas actuators, these were used for gripping the colon wall, forward movement of the vehicle and to provide sufficient flexibility to negotiate extremely tight bends. Joel Burdick, et al (1995) invented a robotic endoscope that uses inflatable balloons to grip the colon wall and expanding bellows to derive the forward movement of the vehicle. The reader will appreciate that the uses of sharp bristles would be extremely unsuitable for use within a human colon. The vehicles described above are generally known as *"Inchworms"*. A review of this technology may be found in Phee et al (1997).

2.7 MILLIPEDE OR BRISTLE DRIVE MECHANISM

As previously mentioned in Chapter One, a simple analogy of the above principle is that of inserting a test tube brush into a test tube with rough walls. It requires a lower force to push the brush forward through the tube, that is, the bristles travelling with their natural deflection, than it does to withdraw the brush against the natural deflection of the bristles. An attempt to withdraw the brush would result in some of the bristle tips securing a "foothold" into the test tube wall. The result is a brush that has a grip on the tube wall, however, as the withdrawal force increases some bristles may jump out of their "footholds" and skip along the tube wall. Other bristles may buckle or "flip through" by rotating out of plane.

The bristle drive mechanism has a number of advantages over a mechanism utilising feet for traction. A vehicle utilising feet requires a specific number of legs/feet, each foot being placed with relative precision and in a specific order. Due to the cost and additional weight of a vehicle using a leg and foot drive mechanism it is unlikely that any additional legs and feet can be incorporated, therefore, it is not cost effective to incorporate redundant legs. Thus, should a single leg/foot be misplaced then the vehicle will become unbalanced and a fall may occur. The bristle drive system incorporates gross "leg" redundancy for very low cost, both financial and weight.

As the bristle drive mechanism does not rely upon a “foot”, it is dependent on the surface roughness of the shaft or pipe wall. The compliance of the bristles allows them to position themselves such that some, or many, are able to find an adequate crevice from which they can obtain a suitable purchase. The mechanical properties of the bristle material, as well as the bristle’s physical dimensions can affect their tractive performance. The number of brush units and cylinders as well as the sequence in which they are triggered are also contributing factors to vehicle traction. A wave like motion, not dissimilar to that of the millipede could prove to be the optimum design.

The bristle system is both simple and passive in its operation. As described in Chapter One, in its simplest form, the system consists of two cores incorporating radially extending bristles interconnected by a hydraulic or pneumatic reciprocating cylinder. Each brush unit contains no mechanical, electromechanical or moving parts. The principle of operation is that the brush, consisting of radially extending bristles, is a larger diameter than the shaft or pipe in which it is inserted. As the brushes are inserted into the pipe the bristles are swept back, thus, orientating the bristle tips at a specific angle to the pipe wall. The natural compliance of the bristles is relied upon to push the bristle tips against the shaft or pipe wall, thus obtaining a suitable purchase. It is a combination of the friction conditions and the numerous bristle factors that combine to produce the traction mechanism. To move a brush forward requires a lower force than that to pull the brush back against the loading direction. Oscillation of the propulsion cylinder will cause the front and rear brush units to move forward in turn.

Backward movement is restricted by the grip of the brush on the pipe wall. Figure 2.4 illustrates the principle.

The bristle drive mechanism has the advantage that the bristles are simple, structural and passive in nature giving considerable compliance with the pipe wall. Should the vehicle attempt to bypass a section of the shaft or pipe that has partially collapsed, the bristles simply bend further backwards. The vehicle is then able to ride over the obstruction and the bristles positioned further along the vehicle continue to provide the required purchase. The vehicle is therefore able to pass the obstruction. Early tests indicated that the vehicles limiting factor, in terms of obstacle passing, was that of its brush core and/or cylinder diameter. More recently, the vehicle which does not incorporate guidance wheels appears able to cope with reductions in internal pipe diameter of up to 40%. Thus, shaft inclusions and omissions can be adequately accommodated.

A vehicle which uses the bristle drive principle is able to utilise gravity as a positive, contributing force during the vertical ascent of a shaft or pipeline. To harness gravity effectively requires the bristle tip to form a specific angle with the shaft wall, this angle varies slightly depending on ambient conditions. It can be seen that where a vehicle is required to operate in shafts of constantly changing conditions it would be useful to maintain the optimum bristle tip angle. One way of achieving this is to adjust the bristle length to suit the local conditions, thus optimum traction is maintained. This system is particularly useful where the vehicle would be required to operate in shafts where the pipeline diameter changes, lubrication conditions also require changes to the bristle angle.

2.8 SOIL MECHANICS

Soil mechanics is another consideration and there are a number of areas of soil mechanics that are relevant to a climbing robot;

- Plasticity of Soil
- Soil Compaction
- Permeability
- Response to Total Stress Change
- Shear Strength/Rate
- Bearing Capacity

The above are for consideration only, it is not proposed that any detail is entered into at this stage but the reader should be aware that these are necessary considerations for a vehicle climbing unlined bore holes.

A shaft climbing vehicle will encounter variable soil and rock conditions, thus, the six factors mentioned above could all prove to be potential problem areas. In climbing, to maintain the body in a vertical orientation requires the legs to "jack out" into often variable strata conditions, therefore, exact orientation will be difficult. Problems could also occur where a foot is placed across a boundary area, for example, between soil and rock, thus the body may move off centre and an unstable situation may occur.

2.9 SHAFT CONDITIONS

Many of the "legged" designs presented in the literature are limited by the legs themselves. Being an exploration vehicle, tunnels and shafts are not going to be straight or of constant diameter, thus, a legged robot may be limited to a specific range of diameters. As each leg would have to be individually placed, robot progress will be slow. It can be seen that the bristle traction system is able to passively expand and contract in line with the shaft or pipe diameter. By means of compliance, the bristle drive mechanism is able to react to the changes in pipe wall diameter instantly, without control or feedback intervention. If a number of bristle/brush units are combined, then caverns, voids and culverts can be traversed. Here, the rear brush unit continues to grip whilst the middle and front brush units "search" for a path through.

Legged robots will find non-conforming and irregularly shaped shaft environments especially difficult to traverse, due to their requirement to maintain a specific, balanced, regular and uniform gait. A legged robot would be particularly susceptible to trapping itself in such an irregular environment. In contrast, the bristle driven vehicle may accommodate these changes easily, with each individual bristle finding its own "foothold" or slipping to form a skid that easily deflects over or around obstacles.

2.10 SHAFT VOIDS AND PROTRUSIONS

If a void appears in the shaft or pipeline wall, the potential for a legged vehicle to misplace a foot is raised, hence, the necessity for control and feedback. Should a void appear in the wall it will then be difficult for the robot to reposition its foot due to the prerequisite requirement to maintain the static equilibrium of that specific type of gait. That is, the foot will only have a small envelope in which to be placed without the danger of disrupting the gait sequence and stability. Once again, the use of a multi-bristled drive mechanism allows some redundancy and is able to accommodate the instance where some of the bristles or even a complete brush finds itself in a void.

2.11 FRICTION CONDITIONS

Frictional considerations are important in traversing both vertical and horizontal shafts and for leg or bristle traction. A fuller analysis is given in Chapters Four and Five but at this point it will be adequate to bring out the point that the bristle mechanism uses the natural compliance of individual bristles to generate the normal force at the bristle/shaft wall interface, whereas in a legged robot the interface force must in some way be generated and controlled by the action of one or a number of the leg joints.

2.12 FOOT DESIGN CONSIDERATIONS

A conventional legged vehicle would benefit from a "spiked" foot, enabling it to obtain a more satisfactory purchase. A bristle uses a multitude of small gripping points. Whereas a spiked foot incorporates ridges and troughs, allowing a better foothold, the bristle tips "search" out the natural crevices in the roughness of a shaft or pipe wall. Under most circumstances, particularly where fine steel bristles are used, the forces generated at the bristle tip/shaft wall interface are large.

2.13 CONCLUSION

The problems a non-uniform unconstrained environment present are complex. Space is often restricted and the environment may contain unexpected and non-uniform obstacles. Any vehicle that is required to operate is such an environment must be either "intelligent" or have passive compliance with its environment. The bristle principle allows passive mechanics to accommodate such limitations and obstructions. The natural spring of the bristle ensures that traction can be achieved.

Other problems can also exist, for example, shaft environments can also be warm, wet, humid or corrosive. Sewers are typical of shafts, tunnels and pipelines that include damage, often as a result of cyclic loading and compressive forces from above, thus, collapsed sections are a regular obstacle to negotiate. A "simple" traction device which does not require complex mechanics and control is consequently very attractive.

The drive method developed was that of a bristle based system. With hindsight it has been found that the understanding and design of such a "simple" drive system has had many hidden complexities and this thesis reveals the evolution of this understanding and its application to the solution of a number of associated project design problems.

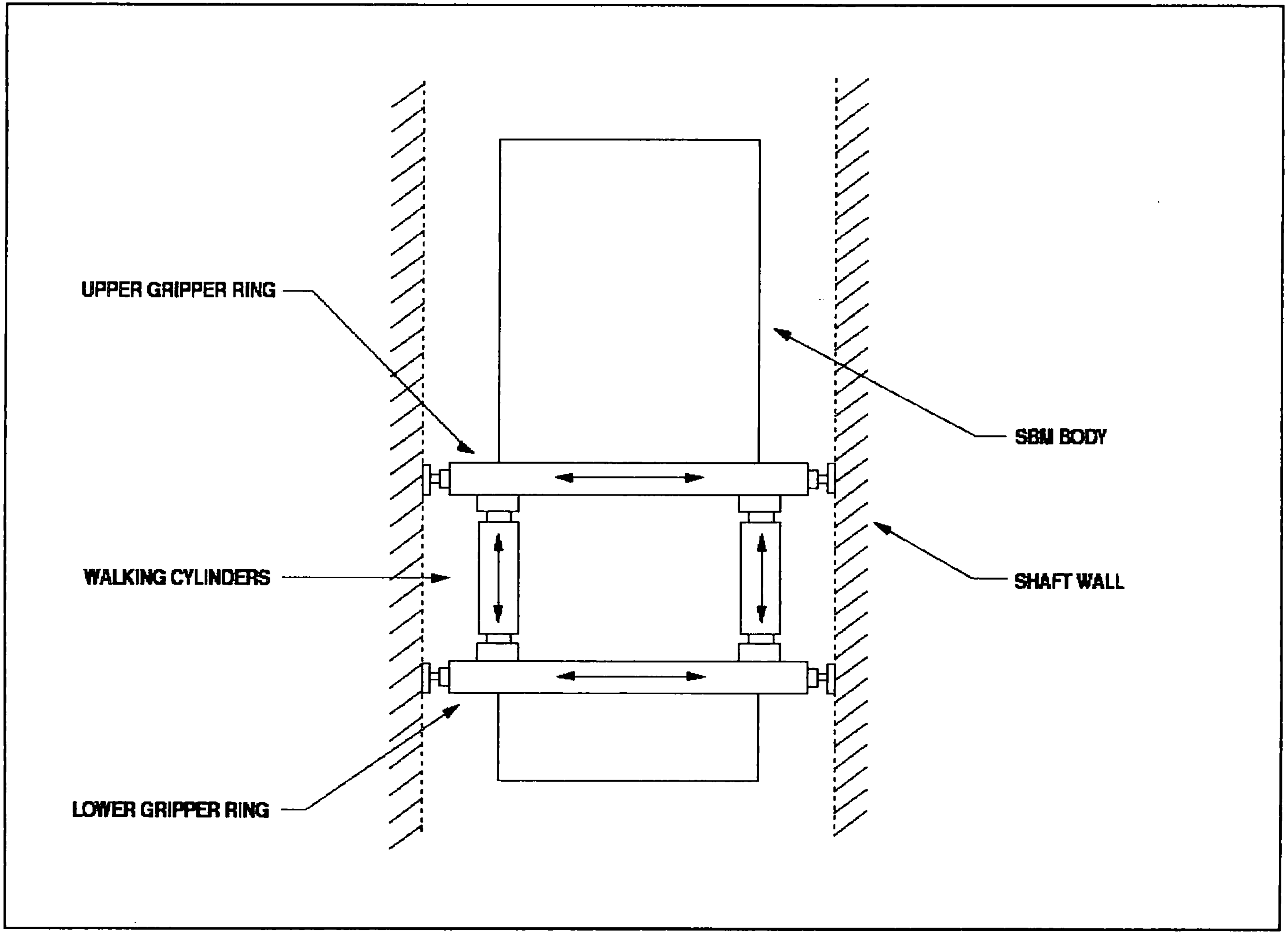


FIGURE 2.1 **SBM WALKING MECHANISM (GOODEL 1991)**

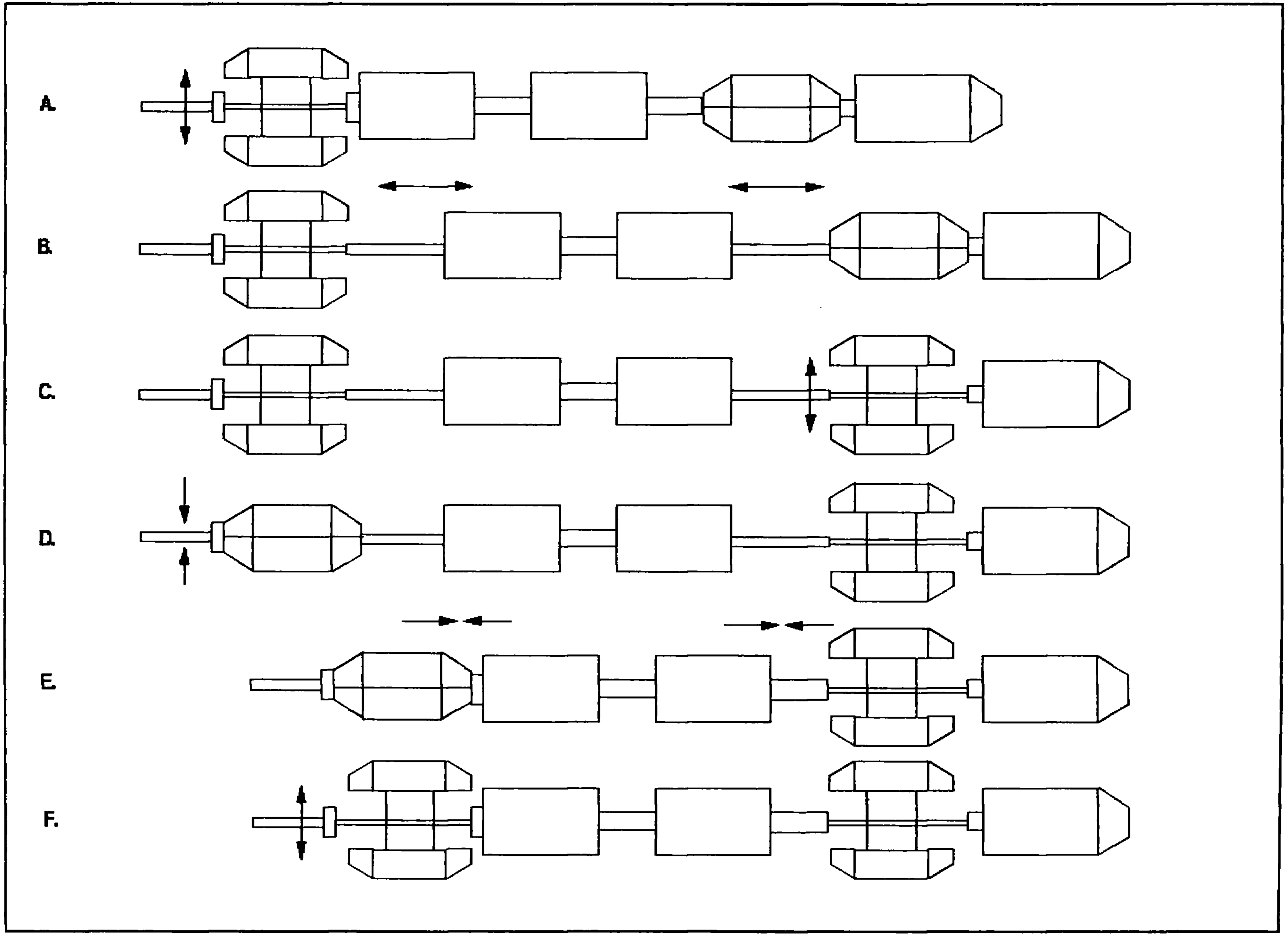


FIGURE 2.2 PIPECAT PRINCIPLE OF OPERATION (Scientific Surveys Ltd 1992)

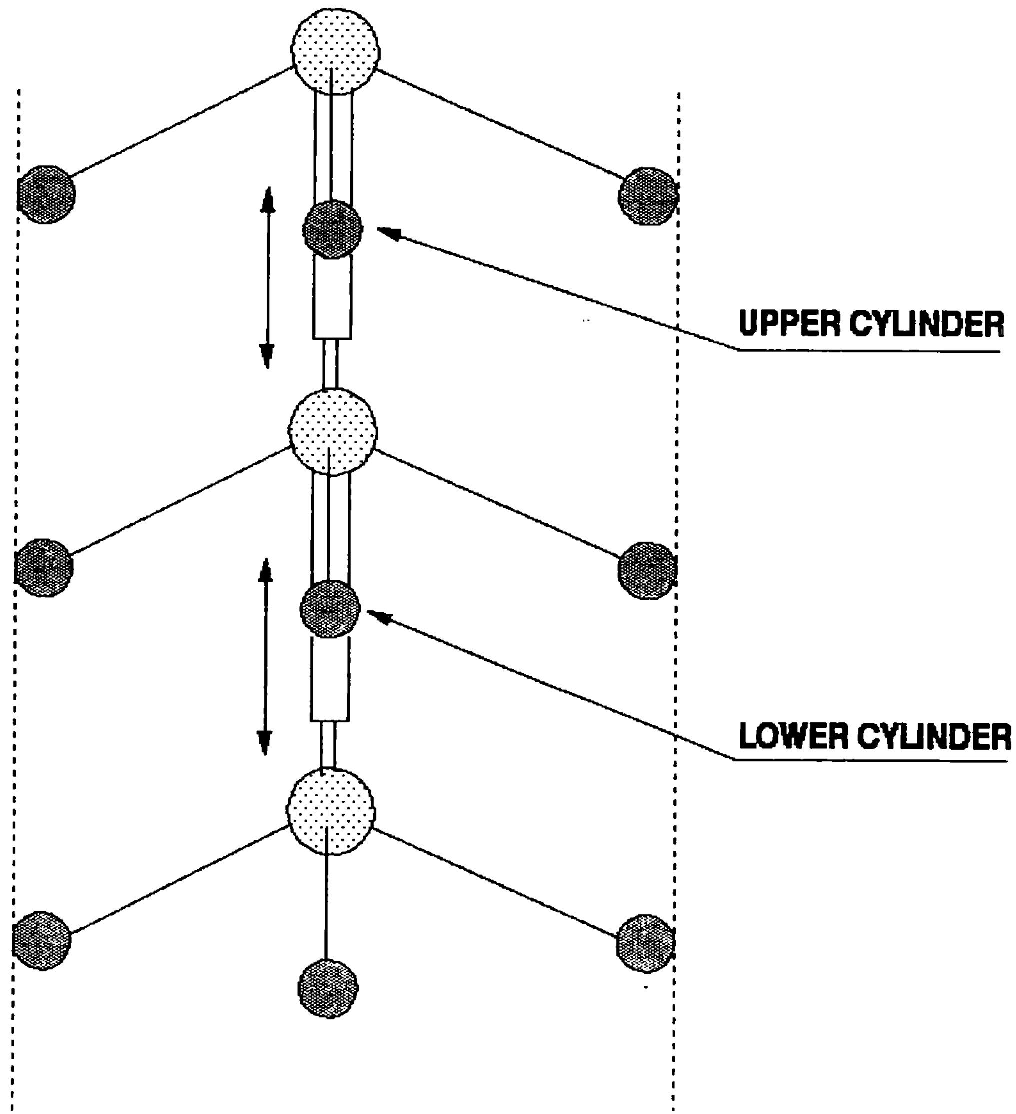


FIGURE 2.3 KINEMATICS: MINIMUM NUMBER OF LEGS REQUIRED TO CLIMB A SHAFT

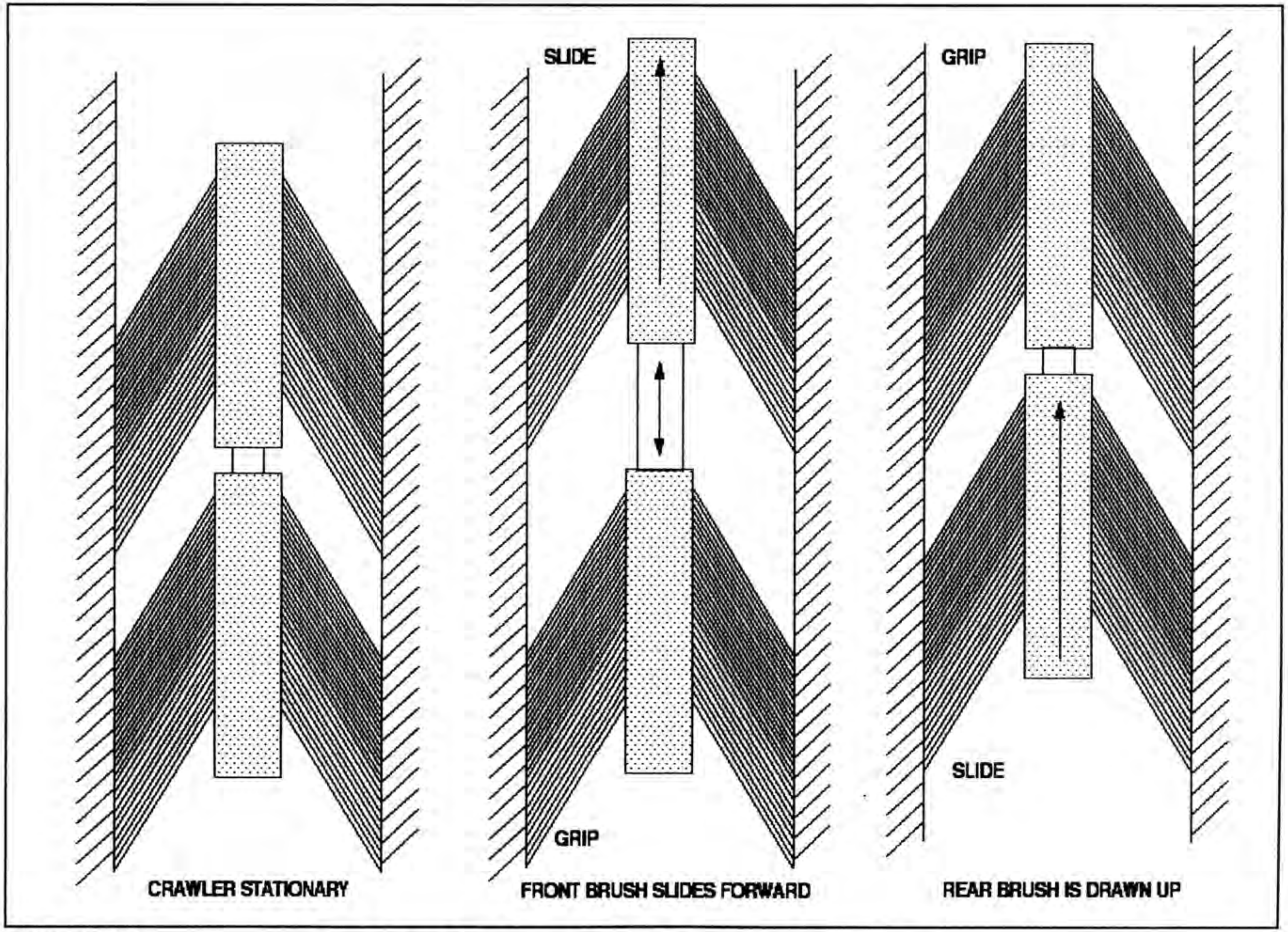


FIGURE 2.4 **THE DRIVE CYCLE OF A BRISTLE PROPELLED CRAWLER**

CHAPTER 3

PREVIOUS RELEVANT WORK

3.1 INTRODUCTION

This chapter considers previous relevant work, both research and commercial. Previous relevant patents are also considered, a number of which came to light during the UK and European (PCT) patent searches that were undertaken with respect to the University of Durham's patent application. Within this chapter consideration is given to papers relating to tracked and wheeled robots, legged walking robots and gait mechanisms. The University of Durham's pipeline vehicle emanated from thought and discussion relating to the above papers. The fundamental goal was the simplification of some of the above mechanisms and principles whilst maintaining reliability and compliance with the terrain. Commercial crawler and pigging technology is also considered

3.2 COMMERCIAL PIGGING

3.2.1 DEFINITION

The origin of the term pipeline pig is not clear. Two schools of thought exist, neither of which can be substantiated; the first suggests the term *"pig"* originates from the squealing sound it makes as it begins its traverse of the pipeline. Most pigs carry inspection components that are an interference fit with the pipeline internal diameter (I.D.), hence abrasion and therefore sound are by-products of the inspection run. The second opinion is that *pig* is an acronym for *Pipeline Inspection G......?*. Even the UK patent office cannot agree upon a suitable word for the *"G"*, perhaps it is *"Gadget"?*. The UK Patent Office defines a pig as *"a device which is passed a distance through a conduit such as a pipe, sewer or main to carry out one or more non-destructive operations on or in the original conduit such as inspecting, cleaning, coating, lining, batching, laying cables, welding, aligning and plugging"*. The Patent Office also includes; *"tethered pigs, crawlers and pipeline inspection vehicles"* in the above broad term.

3.2.2 CLEANING PIGS

The earliest form of the pig is thought to have been a bundle of straw enclosed in barbed wire, used for cleaning pipelines. Patent GB16399 of 1884 is the earliest known patent of a pig, it was classified as a *"Tube Cleaner for Water Mains"*, the applicant being S. Clayton, 32 Houghton Place, Bradford, Yorkshire, England. The 1920's saw the first pigs to employ the pressure of the fluid as the propellant, essentially spheres, bullets and plugs, the UK patent office noted that they were essentially formed from leather and rubber. (GB223073; *R. Arthur and J.T. Mannion, New Jersey, 1923, "Cleaning Tubes"*, GB293032; *J. Adolf, Cologne, Germany, 1928, "Cleaning Beer Pipes"*, GB336388; *I.T. Stephens, Ashton Under Lyne, England, 1929, "Cleaning Condenser"*).

GB287021; *J. Lewis, Monmouthshire, England, 1927, "Cleaning Pipes"* was the first pig patent to incorporate spring loaded arms for wear compensation and segmented drive cups. Sections of the drive cups are able to collapse where the pressure of the drive fluid exceeds a pre-determined level. This may be necessary where the pigs progress is restricted due to a blockage in the pipeline. Additionally, this can be beneficial in controlling the velocity of the pig through the pipeline.

The UK patent office notes that pig patent applications have continued to rise at a steady rate. Areas subject to patent attention include; polyurethane drive cups, the inclusion of limited fluid bypass mechanisms, allowing drive fluid to flow ahead of the pig and articulation between drive cups and body sections allowing the pig to negotiate tight bends.

Other areas of increased patent activity include disclosures referring to the need to accommodate variation in pipe size, improved sealing between drive cup and pipe wall and easier launching.

3.2.3 INSPECTING AND MEASURING PIGS

Early attempts at using electric current and magnetic flux induction techniques to identify defects in the pipe wall were appearing by the end of World War Two. Notable early examples include; GB591822; *D. J. Manning et al., 1945, "Detecting Pipe Leaks"*, GB641657; *D. I. Lawson, 1947, "Determining Physical Qualities and Dimensions of Materials"*. Patent applications started to accelerate during the sixties and early seventies. This is attributable to two main factors. Firstly, emergency pipelines laid during World War Two were placed into the ground without anti-corrosion coatings and/or cathodic protection. These lines remained in service after the war but were beginning to leak and for safety and environmental reasons inspection methods were sought. Secondly, as the oil and gas industries were growing rapidly the need to inspect the pipelines regularly was also increasing. Hence, these companies were spending money on research and development in order to achieve a greater market share. The patents of the 60's and 70's were for *"Intelligent Pigs"*, generally pigs with on-board inspection and storage facilities. An early example was GB1002465; *Camaco Inc., 1964, "Testing a Pipeline for Corrosion and for Defects in Protective Sheathing"*.

Prior to an inspection pig being used in a pipeline of unknown internal dimensions a *"profile pig"* is initially sent down the pipeline. This pig generally consists of two polyurethane drive cups with a metal gauging flange placed between the two cups. The flange is a clearance fit in a pipeline of a known internal diameter, hence, it should pass along the pipeline without incurring damage. If it is recovered with any of the flange fingers distorted or damaged, then, with the help of an on-board recorder and distance odometer, it is possible to identify the location of the bore change. Alternative inspection methods would then be implemented preventing expensive damage to the pig and/or pipeline. The profiling pigs are considered by some to be the most basic of the "intelligent pigs" (GB2040459). T. D. Williamson developed the *"Kaliper Pig"* during the early seventies. Williamson established a number of patents relating to profile pigs (GB1299321; *1971, "Measuring Deviations in Pipeline Bores; Recording Apparatus"*, GB1336246; *1971, "Pipe-Line Pig; Inspection Pig"*, GB1471674; *1974, "Pipe-Line Pig"* and GB2127548). EP0233683 includes on-board electronics and three odometer wheels, thus bend distances can be averaged and calculated with some accuracy.

Twice as many UK patents have been filed concerning magnetic flux leakage pigs than ultrasound, competition between the two methods still exists. Other detecting systems include radiation and infra red. GB1258435 and GB1312229 *(AMF)*, GB1471595 *(Vetco)* and GB1535252 and GB15102225 *(British Gas)* were the leading patents granted in the early seventies concerning magnetic flux leakage pigs. GB1510225 describes the magnetic system with specific relevance to the use of steel bristles for inducing the magnetic flux into the pipe wall. GB2034122 describes the use of laminated steel foils, for the same purpose, the foils directly replace the bristles. GB2149116 describes the use of both magnetic flux induction and ultrasonic inspection on the same pig.

3.2.4 RENOVATING PIGS

Patent GB282969 (*Stewarts & Lloyds Ltd, Glasgow, "Lining Pipes and Tubes", 1928*) is the earliest patent relating to a pig employed in pipeline coating. The above patent relates to a simple pig. Molten bitumen would be charged into the bend whilst a cooling fluid would be applied to the external wall of the bend. The "pig" or more precisely a heated ball would then be drawn through the pipe, thus, smoothing the bitumen layer and removing any excess coating. Bitumen, cement and plastics are all used to coat pipeline walls. Recently the use of flexible, semi-rigid and plastic liners in pipeline renovation has increased. This is classified as *"no-dig"* technology, that is, pigs that are able to enter pipelines, water mains or sewers and repair the fault whilst the lines are still in operation. Patents GB2225406, GB2223559, GB2193289, GB2094178 and GB2188695 are concerned with the use of liners in pipeline renovation. More recently intelligent pigs have been used to determine the position where a branch leaves a main line sewer. After it has been re-lined a cutter head is then used to remove the skin of the re-lining material covering the branch. Patents GB2192442, GB2113126, GB2113126 and GB2092493 detail this type of device.

Sewers and some other pipelines contain insufficient pressure and hence cannot generate sufficient force to drive a pig via the conventional drive cup method. In these cases alternative drive methods are sought, for example, winching or an on-board propulsion system can be employed.

A sub-group of pigs exist to repair leaks. These pigs contain inflatable annular rings, used to define the limit to which an anaerobic sealant will be pumped. Once the sealant has set, the annular rings are withdrawn from the pipe wall (GB2142703). This type of pig often has hollow sections that allow the product to continue to flow whilst the repair work is undertaken. Therefore, damaged pipelines can be repaired whilst the flow of product continues, thus, repair costs are lower. Inflatable sleeves and serrated shoes can also be used to clamp the pig to the pipe wall whilst repairs are undertaken (GB2215805, GB2235745, GB2243428 and GB2086525). In metallic pipelines pigs can also undertake welding repairs. On-board tooling can include alignment, cutting and de-burring equipment (GB2205143).

3.2.5 OTHER APPLICATIONS

The UK patent office notes that the majority of patents registered over the last 100 years have related to cleaning, renovating or inspecting, however, a significant number of other applications have also been applied for. The last 20 years has seen an increased number of patents relating to cable laying and most recently the laying of fibre optic cables. Companies, such as British Telecommunications PLC employ such technology. A recent patent details a device that attaches the cable to the pipe wall as it lays it (GB2241120, GB2163232).

The UK patent office also notes that pig patent activity is *"set to continue across all aspects of the industry with some areas such as cleaning and coating continuing at a similar rate as over the last 100 years. Others, such as intelligent pigging are experiencing fewer filings compared to the last 20 years and others such as the "re-lining" pigs displaying a higher rate of filing".* (The Patent Office, London, 1990.) The UK patent office continues; *"....it is noticeable that over recent years patents relating to pigging increasingly refer to usage in gas, oil, water and sewer lines with resulting cross-fertilisation of ideas across the industries. It is expected this will continue to an even greater degree, given particularly the many millions of pounds of business involved".*

Currently, UK and USA companies lead pig technology. Presently, the former Soviet Union is filing patents nationally and internationally and it is thought that they may have some interesting pig patents.

3.2.6 WINCHED PIGGING

If pipelines are decommissioned, that is, product free and generally open to the atmosphere, there are two methods of inspection. The first is to winch the pig through the pipeline. This is acceptable in straight pipes with few bends, however, in pipelines with a number of right angled bends of a tight radius (1 or 1.5D) considerable problems are encountered. As the winching cable passes the bend a "capstan effect" occurs between the two, as such, the winching load increases significantly as the bend population increases. The higher the winching loads the greater the capstan effect to the extent that the winching rope can and does cut grooves into the inner wall of the bend. The second method is the use of a crawler. A crawler is considered to be a self-propelled vehicle which generally incorporates an umbilical, however, they can also derive their power from an energy source stored on the vehicle.

3.3 DURHAM UNIVERSITY PATENTS

3.3.1 UK PATENTS SEARCH

The University of Durham's UK patent application for a "Surface Traversing Vehicle" (App. No. GB 9619482.4) was subject to a patent search. The UK patent search returned four patents which were considered by the examiner to be relevant to the University's application, that is, they contained similarities to the patent being applied for. A copy of this patent is located in Appendix B.

The first patent is (GB 1 418 492: *Inventor; Roy Butterfield, National Research Development Corporation, filed 11th December, 1972, "Improvements in and Relating to Apparatus for Moving Along or Through a Material"*). This patent considers a vehicle that is able to move along or through the ground. The invention comprises a front and rear unit, both have electrically conductive outer skins which are interconnected via a ram or jack, the two units are able to accept differing potential differences (PD). The front and rear units are insulated from one another. The vehicle can either bore its own hole or travel along a pre-formed hole. The principle of the above invention and the way it differs from the University of Durham's invention is as follows: The PD of the front unit is switched to cathodic (ground potential) whilst the rear unit is switched to +100V (anodic). The cathodic action of the front unit produces little resistance to it moving across the wall material. However, the anodic action of the rear unit has the effect of increasing the resistance to motion and it effectively stands still. The jack or ram is then expanded and the front unit of the vehicle moves forward. The PD is then reversed, the ram closes, subsequently, the front unit "grips" and the rear of the unit is drawn up. This completes a cycle.

This patent has reciprocating motion in common with the University's patent, however, traction is obtained via a completely different method. This vehicle utilises electro-osmotic principles, whereas the University's patent uses compressive forces generated through mechanical struts. Additionally, this mechanism could be argued to be active in its deployment, the University's is passive, that is, the bristles act to provide propulsion without having to be switched, moved or retracted.

The second patent is (EP 0 390 352 A: *inventor; Shaun E Egger et al, 1990, "Apparatus for Remotely Controlled Movement Through Tubular Conduit"*). This invention consists of a front and rear member *"which carries a plurality of extensible and retractable pneumatic cylinders for movement of frictional engagement elements into and out of engagement with the interior wall of the pipe. One or more axial drive cylinders rigidly connect the spaced* *members for movement thereof toward and away from one another".* It can be seen that this invention relies upon jacking out a gripper ring, obtaining a foothold and then extending or retracting the axial drive cylinder. This vehicle is again active and complex in nature, the opposite being compliant and passively gripping bristles. The principle is similar to the SBM and TBM's described earlier.

The third patent is (US 4 537 136: *Brian Douglas, Subscan Systems Ltd, 1985, "Pipeline Vehicle").* This vehicle has front and rear sets of wheels carried on sprung arms, both sets of wheels are lockable. The principle relies upon the rear set of wheels being locked, a linear axial ram is operated and the front wheels, being unlocked, slide forward. The principle is reversed and repeated, thus, reciprocating motion is achieved. Douglas' patent cites four other patents as reference, FR - App. No. 2 355 236 *(Rouland),* US Pat. No. 3,047,270 *(Moore),* US Pat. No. 2,518,330 *(Jasper),* GB - App. No. 1 124 732 *(Chicago Pneumatics).* The patent application continues; *"To provide a vehicle which can travel through a pipe or conduit by means of forward and rear wall-engaging means mounted on a main body and alternately operable to grip the wall, together with means operable to axially extend and retract the wall engaging means relative to each other in synchronism with the wall engaging means to cause the vehicle to advance stepwise in the pipe".* This patent refers to prior known arts as having two main disadvantages, they are; *"One is that it is not easy to adapt to varying diameters of pipe, each vehicle essentially being for a pre-determined pipe size. The other is that the vehicle is not capable of traversing any wall irregularity or other obstruction of more than very small dimensions".*

The above patent requires a secondary locking function, that is, the vehicle is unable to operate passively. The above three patents have the disadvantage that they are slow. To operate successfully they require a secondary function to obtain traction, for example, a locking, clamping or switching operation as well as the linear, axial operation.

Patent four is (DE 2405343 A: *Garda Schnell, 1974, "Propulsion for Sliding Vehicles")*. This patent initially caused concern. The patent details a device for sliding over unstable terrain, for example, water, mud, bog, meadow, sand and snow. The device incorporates angled faces on two independent surfaces. When one device is moved in a forward direction the friction resistance between the contact face of the other device and the surface over which it is passing increases resulting in the reaction force necessary for the first device to progress forwards. The procedure is then repeated resulting in a reciprocating action. Schnell indicates various types of contact face are possible including; blade, spoon, scale or brush type.

Upon closer examination of the patent it becomes evident that Schnell's patent relies upon each face generating its reaction force from the viscous drag of the fluid or material over which it is being driven. It can be seen this principle is significantly different to the brush mechanism where each individual bristle is subject to point contact forces, the forces are then transmitted through the strut axially. Schnell has a separate design that uses animal fur, this is the closest mechanism to bristles in the patent. It is clear that animal fur would not have sufficient strength to support a device as described by Schnell's patent. Schnell's device is planar in nature, thus, it relies upon gravitational force. The University's vehicle is designed for oval, round or irregularly shaped conduit and does not require gravity to assist in the traction mechanism. The University's vehicle provides its own "gravity" force in the form of equal and opposite reaction forces from opposing bristles.

3.3.2 PCT PATENT SEARCH (EUROPEAN)

As Pipeline Integrity International (PII) required the patent cover to be world-wide, the most effective way of achieving this was to file what is known as a PCT application (PCT/GB 96/02307). During the PCT patent search five patents were cited as having features similar the University's patent application.

The first patent is (FR,A,2 495 191; *1982, Edouard Remaut et al, Pipeline Service, France, "A Device for Monitoring the Soluble Anodes for Cathodic Protection of a Pipeline"*). It is unclear why this patent was listed as the vehicle is driven by conventional pipeline product acting as the propellant across the drive cups fitted to the vehicle; *"The intra-pipe controller assembly is propelled by the fluid in the pipeline which is to be inspected, such as crude petroleum, at a pre-determined speed of 1 to 1.5 meters/second"*. The vehicle does, however, employ metallic brushes, although these are only used to detect the differences in potential which occur between two points along the inside walls of the pipe.

The second patent is (EP,A, 0 523 880; *Kenneth Watson et al, British Gas Plc, 1993, "Pipeline Inspection Vehicle"*). "Conventional" pigs utilise steel bristles to induce the magnetic flux into the pipeline wall, however, some pigs use steel foils for the same purpose. This pig uses foils for magnetic flux induction. Drive is via the use of drive cups and product pressure. It can be assumed that a Boolean algebra search was employed by the patent examiners, hence, the word *"foils"* was returned because the University's patent mentions foils as an alternative to bristles for the drive element.

The third patent is (EP,A, 0 514 039; *David Smart, British Gas Plc, 1992, "Towing Swivel for Pipe Inspection Vehicle or Other Pipe Vehicle"*). Again it is assumed that this patent was returned during a Boolean algebra search for the key word *"foils"*. This patent relates to an invention for a towing swivel, the swivel is designed to ease the passage of the inspection vehicle around bends of tight radii. The swivel is able to rotate to a suitable position to prevent jamming. The vehicle is a conventional Tethered Inspection Vehicle (TIV), the vehicle that the University's invention is intended to replace. Again this patent appears to be of little relevance.

The fourth patent is (EP,A, 0 526 900; *Ronald E. Pelrine, Osaka Gas Company Limited, 1993, "Vehicle for Use in Pipes"*). This vehicle consists of a *"vehicle body, a drive assembly, a wheel comprising an outer wheel member having an annular peripheral wall, an inner wheel member including a portion of magnetically permeable material having an outside diameter....... "*. This patent refers to a vehicle that is attached to the inside or outside of a pipeline via magnetism, it is essentially a wheel driven train that manoeuvres its way through the pipeline.

The fifth patent is (DE,U, 93 11 145; *Siemens AG, 1995, "Device for Inspecting or Treating the Internal Surface of a Pipeline"*). This is a small vehicle that is designed to operate in pipelines of approximately 50mm (2") internal diameter. The vehicle is pushed into the pipe via a *"pressure-rigid hollow thrust hose"*, that is, a type of pushrod system. The vehicle incorporates a front manipulator for tool holding, for example, a grinding head. The vehicle does incorporate "drive vehicles" although these simply facilitate additional support for the thrust movement of the manipulator. This vehicle incorporates pneumatic air rams, although they are used to lock and support the manipulator during the tooling operation.

3.3.3 BNFL PATENT

A BNFL patent was brought to the attention of the University after the University had filed its own patent. The patent was registered by BNFL in 1985, the inventors being Edward Leonard Jones and Robin John Luxmoore (GB 216 7829 *"Improvements in Pipe Crawlers"*). This patent describes *"a double-acting piston and cylinder assembly having members on axially displaceable component parts of the assembly to support the assembly substantially centrally within the bore of a pipe or tube, the members being of an effective lateral dimension* *greater than the diameter of the bore and having a rigidity such that when the pipe crawler is inserted into the bore the members are inclined to the axis of the pipe and their outer edges are continuously biased into engagement with the wall of the pipe, and means for introducing pressure fluid to the assembly in such a manner as to effect step by step travel of the crawler along the bore, the members on the axially displaceable component parts of the assembly acting to alternately anchor and release the respective component parts, and an inflatable bladder connected to one of the component parts operable to anchor the part to effect reverse movement of the crawler...... Alternatively, instead of being annular plates or diaphragms the members 5 and 6 can be spiders engaging the wall of the pipe at angularly spaced apart regions".*

Again, initially, this patent caused the University some concern, however, upon detailed examination there are a number of fundamental differences between this and the University's patent. As part of the BNFL patent Claim One states that; "*.....and an inflatable bladder connected to one of the component parts operable to anchor the part to effect reverse movement of the crawler".* All BNFL's proceeding claims refer to Claim One. The University's patent does not require a bladder to effect reversal, therefore, the University's patent does not infringe the BNFL patent. Considering the BNFL patent from an engineering perspective; The stress conditions observed by a diaphragm are significantly different to those present in the University's bristle drive mechanism. The stiffness of a diaphragm subjected to the loading described by the BNFL patent is dependent upon the generation of secondary, compressive hoop stresses as the diaphragm deforms. Should the diaphragm be cut, for example, to leave a "spider" shape, then the hoop stresses cannot be formed, hence, the support or reverse load capability of the diaphragm will decrease significantly. The bristle principle does not rely upon the above mechanism, that is, hoop stresses cannot and do not need to be generated. Each bristle, whilst subject to a compressive axial force acts as an individual strut, the reaction force is generated by each bristle obtaining a "foothold" on the wall of the pipe. The support force is then transmitted axially along the length of the bristle into the pipe wall. Friction and surface roughness of the pipe wall are required for both traction principles, however, the University's mechanism is fundamentally different to that of BNFL. The BNFL design cannot easily accommodate pipeline inclusions or obstacles. A "diaphragm" or "spider" would cope less well than bristles when encountering an obstacle. The diaphragm or spider would have to ride over the obstacle, hence, deflecting in the radial-longitudinal plane. Deflection in this direction is undesirable because it leads to loss of traction. Additionally, a deflected diaphragm or spider may further lose traction due to the material either side of the collision point being lifted away from the pipe wall. The main problem with diaphragms or spiders is that they are made of continuous or semi-continuous flaps of material. Under the same conditions, as mentioned above, a brush may have only a few bristles deflected whilst traversing the same obstacle. Bristles tend to deflect or "flow" around an obstacle whilst being able to maintain satisfactory traction. Consequently, it is unlikely that the BNFL crawler will operate as well in deformed or damaged pipe work. Further, the BNFL vehicle requires the front diaphragm to be of a "weaker" material than the rear diaphragm in order for reversal to occur. Once again the University's crawler does not require this mechanism. If "diaphragms" or "spiders" were used in conjunction with universal joints to enable bend passing capability, then due to their limited aspect ratio it is likely that jack-knifing of the diaphragms may become a significant problem.

BNFL makes no mention of separating the drive and suspension functions of the crawler, therefore, the diaphragm must serve both purposes. The diaphragm material must be stiff enough to support the crawler whilst continuing to provide sufficient traction. The relative stiffness of the diaphragm may also result in a lack of compliance with the pipe wall environment. Finally, the diaphragm is designed to deflect a specific amount when inserted into a pipe, if the internal diameter (I.D.) of the pipe varies significantly then it will not be possible to maintain optimum traction. A pipe with a smaller I.D. will cause the diaphragm to deflect further backwards resulting in lowered tractive effort. A pipe with a larger I.D. may result in some or total traction loss if the diaphragm is too small for the I.D. of the pipe. If some contact is maintained a number of the diaphragms may be subjected to premature reversal.

A number of patents were referred to by the BNFL patent. The most relevant is patent GB2152622 A, *(Alan Bishop, Micro Consultants Limited, 1984, "Pneumatic Switching Devices and Linear Feed Motors Incorporating Such Devices")*. This patent considers a known type of duct motor, a device used to traverse ducting whilst towing a rope or cable. The rope is then used to haul cables through the duct.

In general, duct motors tend to have an axial pneumatic cylinder with an inflatable air bag or "balloon" at either end. The duct motor principle is similar to the worm drive principle mentioned in Chapter Two. The duct motor operation cycle is simple. The motor is inserted into the duct and the trailing balloon is inflated. The axial cylinder is extended and the leading balloon is inflated whilst the trailing balloon is deflated. The axial cylinder is closed, that is, the trailing balloon unit is drawn up towards the leading unit. This completes one cycle. Patent GB2152622 is similar to that of the duct motor except it cites... *"two pair of resiliently deformable, disc shaped diaphragms"* instead of the balloons, additionally, it states..." *the diaphragms are disk shaped but could be different shapes, for example, flexible fingers"*. The term "flexible fingers" can be considered to be as undefined and as ambiguous as BNFL's term, "spiders".

A number of other patents were cited by the BNFL Patent. The first is GB2153040 A (*Malcolm Wayman et al, British Gas Corporation, 1985, "Self Propelled Apparatus for Replacing (e.g. Gas) Mains"*). Wayman's patent from an engineering perspective is interesting. The device consists of a axial hydraulic cylinder, with opposing, tapered, sliding, grip rings at each end. When the cylinder is activated, the front tapered grip rings naturally release, as the rear cone begins to slide back the tapered grip ring is forced out into contact with the wall, hence, they lock. When the axial hydraulic cylinder is closed (evacuated), the reverse occurs, hence, passive, reciprocating forward motion. In line with the BNFL and University of Durham patents this is another device which is passive in operation, that is, it only requires a single traction stage to obtain forward motion. A disadvantage of this device is that it has limited obstacle avoidance capability and bend passing capability.

Patent two is GB1044201 (*James Douglass Hill, 1963, "Improvements In or Relating to Pneumatic Self-Propelled Apparatus")*. This patent is an early application for a duct motor, that stated, it is difficult to distinguish it from the two patents mentioned above.

Patents three is GB2126683 A (*Cyril Arthur Piper et al, Kinaut Instruments Limited, 1983. "Duct Motor"*) and GB 2059000 A (*John Ronald Slight, The Post Office, 1980, "Pneumatically Propelled Duct Motor"*). This patent considers improvements and variations to duct motors, the principle of which has already been described.

Patent four is GB2137719 A *(Patrick Noel Daly et al, P.N. Daly Limited (UK), 1984, "Pipe Replacement")*. This patent refers to a *"mole"* that is winched through an existing pipe, braking it up as it progresses, the mole tows a new pipe into place as it progresses. As with some previous patents it is unclear why this patent was referenced.

3.4 COMMERCIAL CRAWLERS

A number of commercial crawlers exist. The majority of commercial crawlers use either tracks or wheels, a lesser number incorporate circumferential clamping (grip rings). Some crawlers slide on skids having being inserted into the pipe via a rigid, stiff, "umbilical" push rod. Some of the inspection camera devices used for small bore pipework (<50mm ID) also use the push rod "drive" system. Additionally, they can incorporate camera centering devices which can be in the form of circumfrential brush rings or rubber ridges, in the former, the bristles have no drive function.

3.4.1 PEARPOINT LTD

A current leader in CCTV pipeline inspection is Pearpoint Limited. Pearpoint provide a considerable range of products from the small *Mini Flexiprobe* camera to fitting out complete vans with dedicated inspection equipment. The range includes the *Flexiprobe* camera (Patent No. 2172079) which is used in pipes of 25mm diameter or more. This has a range of up to 150 Metres and uses a pushrod. The Flexiprobe/Flexiscan can be added to the high torque P238 six wheel tractor with a potential range of 300 Metres. Changing the diameter of wheels allows the unit to operate between 6" and 24" pipelines. The Flexiprobe/Flexiscan can also be fitted with skids. An additional tractor is the P148 (Patents; 2210530B, 2215941B, 2215942B and 2211594B). Pearpoint have a range of explosion proof cameras and tractors in addition to the standard range. Including the P415 colour forward viewing camera, P403 Flexiscan camera, P420 six wheel explosion proof camera and the P400 explosion proof tractor.

Pearpoint's tractors utilise vehicle mass to increase tractive effort in what can often be damp and slippery pipes. Towing the umbilical induces drag, the better the traction the further the tractor can progress before reaching the limit of the vehicle's range. At this point the vehicle must reverse, sometimes being unable to complete its inspection run. Wheeled tractors find dislodged pipework very difficult to cope with, for example, a step where two pipes meet would be a significant problem. If the vehicle continued down the step then it may not be able to return the same way, that is, climb back up the step. To cope with such conditions requires the integration of a number of variables, for example, vehicle mass, wheel diameter, step size, friction conditions and traction. The "step" problem has been witnessed during field trials of the Pearpoint tractors. As previously mentioned, Pearpoint's vehicles rely on gravity acting on the mass of the vehicle to induce traction, as such, their tractor vehicles are unable to cope with vertical ascents or descents. Additionally, the tractors are unable to cope with 1.5D bends or below, even 3D bends can cause problems.

3.4.2 FASTFLOW PIPELINE SERVICES

Another company using crawler technology is Fastflow Pipeline Services (FPS) based in the North East of England, FPS is the UK licensee of the world-wide patented KA-TE system, manufactured by KA-TE SYSTEMS, Zurich. This is a tractor vehicle, instead of performing routine camera inspections, it specialises in carrying tools suitable for sewer repair and rehabilitation. Tools available include; hollow diamond tipped drills, grinding bits, epoxy grout spreaders, dedicated groove cutters (tree root removal tool), 4.5HP grinding motor for concrete and scale removal, inflatable seals and shuttering tools. The crawler module is similar to the Pearpoint vehicle, that is, it has considerable mass, relies on gravity for traction and is controlled via an umbilical. This type of tool relies on a skilled operator, it is able to accommodate pipe diameters ranging from 200-800mm.

3.4.3 THE CRAWLER INVASION

A number of other companies operate similar crawler vehicles. Some of these companies are briefly discussed below, they include:

TELESPEC; this company offers standard colour cameras as well as pan and rotate colour cameras. The midi crawler weights 14Kg and has a range in excess of 100 metres. The mainline crawler weights 34Kg and has a range in excess of 200 metres. Telespec also offer borehole colour cameras and precision root and intrusion cutters as well as a vehicle mounted on skids which incorporates a high pressure water cutter.

HYTEC (HYDRO-TECHNOLOGIE) this company carries a similar range of crawlers and vehicles to those mentioned above, they quote their products as; *"remote controlled vehicles, video systems and instrumented robots and tooling"*. Areas of activity include sub-sea, nuclear and pipelines, ducts and boreholes.

SCANPROBE TECHNIQUES offer a range of camera crawlers, again using a crawler body of considerable mass, traction is via wheels and control is via an umbilical.

DUCKBILL supply a track/wheel driven crawler with a pipe diameter range of 6" - 24", linear range approximately 200 metres. Within the product range is also a 3" diameter push rod camera, Duckbill claim this is able to accommodate 90 degree bends in 4" diameter pipe.

JENOPTIK JENA (formerly Carl Zeiss Jena) offer a range of crawlers and push rod camera devices. Their crawler range being of the "conventional", heavy, wheeled vehicle type. The vehicles use interchangeable wheels of variable diameters depending upon pipe diameter to be inspected, bolted to the vehicle body is the camera.

ROV TECHNOLOGIES INC. have *a "miniature wet or dry crawling climbing device"* - SCARAB 11. This vehicle weighs approximately 25Kg. It is different to those previously mentioned as it utilises independent front and rear drive tracks, ROV Technologies claim that it is able to negotiate obstacles up to 7" in height.

JME LIMITED produce a range of pipeline crawlers, varying in diameter from 6" - 72". These crawlers are "conventional" in that they obtain traction via wheels and a body of considerable mass, however, they are used to inspect the pipeline via X-Ray inspection devices, these are carried on board. JME claim the maximum gradient their crawler can negotiate is 27 degrees, for a *"clean, dry pipe"*. The crawlers carry their own power supply in the form of large lead acid batteries, hence, the weight of the 22" crawler depends on the battery Amp/hr rating, the 20 Amp/hr crawler weighs 150Kg, the 40 Amp/hr crawler weighs 240Kg.

None of the above crawlers are capable of vertical ascent or descent under their own power. However, there are at least two crawlers on the market that are able to perform this task, albeit they are slow in operation due to the additional drive steps required to move them forward, the two vehicles are discussed below.

3.4.4 INTERNAL PIPELINE VEHICLE (IPV)

The Internal Pipeline Vehicle (IPV *"Pipewalker"*) is manufactured by Remote Special Systems (UK) Limited, although it was originally designed by Subocean Projects and Engineering Ltd (SPEL). The vehicle has two pneumatic axial drive cylinders and a front and rear clamping mechanism, interconnected via universal joints. Operation is as follows; The rear clamping unit is expanded and both axial cylinders expand. The front clamping unit then expands whilst the rear clamping unit is retracted. The axial cylinders then retract. This completes the cycle. The cycle is then repeated. It can be seen that the Pipewalker requires two additional drive operations compared to the University's vehicle. These additional drive operations slow the vehicle. Localised pipe wall conditions also affect vehicle performance. The current linear range of this vehicle is quoted as 500 metres, the number and severity of bends would reduce the vehicles range accordingly. The vehicle is also classed as “fail safe” by its designers, that is, in the event of a power failure, the clamps would retract allowing manual retraction of the vehicle from the pipeline. This vehicle has the advantage that it is able to ascend and descend vertically. However, if a power failure occurs during ascent or descent, then, the vehicle may suffer a fall.

3.4.5 PIPECAT

Pipecat is manufactured by AGR (GRØNER OFFSHORE I&M AS Norway), it is a pneumatic, umbilical tethered crawler. Pipecat is capable of climbing vertically and has a quoted linear range of approximately 500 metres. The principle of operation is identical to that of the IPV described above, except that the clamping unit takes the form of a split circular pod, which, when expanded, clamps the pipe wall. The average velocity of Pipecat is quoted as 100m/hr, the University's crawler has a velocity of approximately 650m/hr. GRØNER claim Pipecat is able to negotiate 1.5D bends and pipeline diameters between 2”-24”. Pipecat inspects using colour CCD camera and ultrasound.

3.5 PREVIOUS RELEVANT RESEARCH

3.5.1 TRACKED AND WHEELED DRIVE

The ideas and thought processes which resulted in the bristle propelled vehicle stemmed from consideration of “conventional” areas of automation, notably legged and wheeled vehicles. Although the University's vehicle is clearly shaft or pipeline based, other variants are possible. For example, planar vehicles where payload, gravity and vehicle mass would combine to orientate the bristles to an appropriate angle suitable for vehicle traction to occur. The following section reviews research areas applicable to traction and negotiation of non-uniform terrain.

Some initial discussions considered the design problems relating to vehicles used to climb the external or internal surfaces of structures. Hagen Schemph et al (*1995, Neptune: "Above-Ground Storage Tank Inspection Robot System")* utilised magnetism for the "clamping" or "push force" required to overcome gravity and shear forces whilst attached to the vessel's walls. Neptune was designed to navigate around the base and walls of above-ground storage tanks. The crawler consists of a set of main drive tracks. At the front and rear of the main tracks are a set of hinged auxiliary tracks. The auxiliary tracks are permanently magnetised and they are angled and tapered at the front, this allows easy transition between floor and wall manoeuvres. The main drive tracks incorporate switchable magnets, this is to prevent high levels of debris attraction, for example, ferrous corrosion flakes and particulate. Switchable magnets also allow easy turning, one track remains magnetised and hence locked whilst the opposite track continues to drive. The vehicle then pivots around the stationary track.

C. R. Weisbin et al *(1994, "NASA Rover and Telerobotics Technology Program"), Planetary Rover Technology Program* noted that; *"The rover technology base emerging from this program has made possible the Mars/Pathfinder project Microrover, currently planned for launch in 1996".* However, of greater interest were the future goals considered necessary for the following four years up to 1998, they were;

1. Autonomously traverse 100m of rough terrain within sight of a lander.
2. Autonomously traverse 100m of rough terrain over the horizon and effect a return to the lander craft.
3. Autonomously traverse 1km of rough terrain with execution of select manipulation tasks.
4. Complete science/sample acquisition and return to lander with over the horizon navigation.

Weisbin also noted that; *"University and industrial researchers are contributing to the Rover Technology Program in such areas as legged versus wheeled mobility".*

At present, considerable research and thought is being directed towards mobile robot mission capabilities for the NASA Lunar Rover Mission. Weisbin considers a number of mobile robot prototypes including DANTE and AMBLER, both of which were developed at Carnegie Mellon University. DANTE and

AMBLER have both been subject to testing in extreme conditions including Mt. Erebus (Antarctica) and Mt. Spurr (Alaska).

Many universities and private companies continue to develop mobile robot technology, originally this was for "pure" research and development of the technology. More recently, companies and individuals are considering the commercial potential of such "vehicles". Simmy Grewal, in a recent article entitled *"Mobile Systems"* (1993) reviewed the capabilities of Numbat. Numbat is an eight wheeled crawler. Numbat is remotely controlled and designed to enter coalmines as a rescue vehicle. In field trials the vehicle has been used to enter underground mines in Queensland, Australia. At present only shallow entry drift mines are possible, Numbat can only enter if driven in under its own power. Numbat is a wheeled vehicle with a low centre of gravity, where vertical ascent or descent is not possible, negotiable grades are limited to *"25 degrees on loose material and 45 degrees on firm surfaces"*.

A number of companies in the United States are starting to utilise robots in security roles. Cheryl Pellerin considered this point when she reviewed Cybermotions' SR2 in her article *"Twenty-First Century Centries" (1993)*. SR2 is a wheel propelled security robot that can patrol a deserted factory for up to 12 hours before it requires recharging. SR2's functions include; video recording, sensing fire, intruders and air quality. If a fixed alarm is triggered in another part of the factory SR2 will take the most direct route and investigate. SR2 uses the K2A synchro-drive mobile base, this allows three wheels to be powered and steered in synchronisation. This vehicle is only suitable for flat and relatively uniform terrain, if the terrain was unstructured then the vehicle may become unstable. A number of commercial mobile robots are already in production including Denning Mobile Robotics Inc.'s range of vehicles including self-guided vacuum cleaners, floor scrubbers, security cameras and trolleys.

Shigeo Hirose and Akio Morishima (1990) consider some fundamental points in their article *"Design and Control of a Mobile Robot with an Articulated Body"*. They note that for robots to become more useful in society they must address the current problem of non-uniform terrain negotiation, both indoors and outdoors, outdoor examples cited include *"fields, mountains and sea bottoms"*. Hirose and Morishima consider the internal traversal of a nuclear reactor for their design study. Nuclear reactors are hostile and non-uniform environments in which to operate. The point is made that; *"limitation of the wheel and crawler track system is its low terrain adaptability. Although the crawler track is more easily adaptable than the wheels, the locomotion of both is restricted when there is any irregularity on the surface of the ground"*. Hirose and Morishima advance to consider legged locomotion. They note that legs have the ability to *"choose intelligently which spots on the ground are to be contacted. By discreetly choosing supporting leg points, high terrain adaptability can be exhibited, even when the surface is highly irregular"*. Few would disagree with this statement. In many ways this statement shows clearly the advantage of a multitude of passive bristles acting as "legs", during every stoke the brush unit is guaranteed that a percentage of the "legs" will obtain a satisfactory purchase with the wall.

Hirose and Morishima's design, the KR1, is a tracked vehicle, consisting of a number of modules, therefore, the leading and trailing modules are able to support the central module as the KR1 crosses a ditch. Each module can rise and fall with respect to one another, as such, the KR1 is able to ascend and descend stairs. KR1 relies upon gravity for traction, thus, it is unable to climb vertically as a suitable "push" force, required to prevent a fall, is unavailable.

Scott Y. Harman (1987) starts his paper (*"The Ground Surveillance Robot (GSR): An Autonomous Vehicle Designed to Transit Unknown Terrain")* by considering the same point as Hirose and Morishima, that is, the need for autonomous robots which are able to cope with extreme terrain. Harman quotes terrain examples as; combat, outer space and ocean exploration. The GSR is intended to transit from a known point to another known point whilst negotiating unmapped, extreme terrain. The paper is concerned with the control of the vehicle, however, this clearly illustrates the benefits a passive terrain robot may have over such a dynamic vehicle. Control is only necessary in order to determine obstacle avoidance, guidance can simply be obtained through a Global Positioning System (GPS) and auto-pilot. The GSR is an M114 armoured personnel carrier that has been converted for auto control. Although using tracks, the GSR is still subjected to terrain limitations, such as grounding.

Wheel and track drive mechanisms are well known, R. A. Bryson (1988) discusses obstacle avoidance in his paper *"Quantifying Battle tank Mobility"*. Bryson notes that the first tanks were designed to cross obstacles, this was a more desirable attribute than speed. Obstacle avoidance is an important issue but velocity is also becoming important. For a tank to climb steps, its design is important, it must have a high forward mounted idler, large approach and departure angles and an aggressive track. The centre of gravity must be low and forward on the tank, this helps eliminate the possibility of the tank flipping over. Bryson noted that void crossing capability was a function of centre of gravity position and length of track on the ground. Gradient climbing performance requires a powerful engine, low gearing, consistent traction and good brakes!. To traverse a steep sided slope in a tracked tank can be difficult, if it were ascending the slope directly the transverse bars and ridges under the tracks increase traction. However, traversing a slope with this type of track does little to prevent slippage. Bryson also notes the increased likelihood of grounding in a tank, notably, under muddy conditions, to the extent that a "V" shaped under belly is desirable as it reduces the suction between the tank and the mud. Water is also considered to be another type of "obstacle". Tanks are designed to wade, unprepared, to a depth of 1 metre, deeper water can be crossed via snorkel. Some lighter tanks and those fitted with wading screens can swim, for example, the Sherman DD tanks of World War Two.

3.5.2 LEGGED WALKING MACHINES

Walking machines can be considered to fit one of three groups, they are; static or stable gait walkers, hopping machines or bipeds. Initially, the exact format that this research should take was unknown, however, a paper entitled *"Configuration of Autonomous Walkers for Extreme Terrain"* by John E. Bares and William L. Whittaker (1993) became the starting point. The first sentence of the paper states *"Robots that can competently, efficiently and autonomously operate in extreme terrain do not exist"*. From initial discussions thought was apportioned to the complexity of the control needed for such a robot. It was at this stage that a decision was taken to attempt to determine a more simplistic method of obtaining traction for non-uniform terrain. Bares and Whittaker continued by discussing the complexities of three dimensional terrain, boulder covered slopes, steps, ditches and other natural obstacles. They also considered terrain material type, for example, rock, sand, mud or dust. Further, they stated that *"walking locomotion is uniquely advantageous to autonomous traversal of extreme terrain: because a walker adapts its feet to the terrain, it can avoid undesirable footholds, optimise stability and propel its body independent of terrain details"*. Again the advantages of a passive, continuous, non "thinking" method of terrain traversal are apparent. Bares and Whittaker considered three types of walker configuration, to their credit two, the Circulating Walker and the Weaving Walker incorporate the new and unique feature that the trailing legs, at the end of the gait cycle, are recovered from the rear of the body and placed ahead of the supporting legs. However, the fundamental problem of how to control such a vehicle still existed.

If one considers the “apparent ease” with which a human can traverse extreme terrain, some to the point of running, for example, a fell runner descending a scree slope with all the normal, gravity and shear forces, it is not unreasonable to consider a biped traversing extreme terrain. The control of such a robot would require balance, three dimensional terrain profiling ability, tactile, force and dead reckoning sensing. The second paper reviewed following the Bares and Whittaker paper was that of A.A. Grishin et al *(1994), “Dynamic Walking of a Vehicle With Two Telescopic Legs Controlled by Two Drives”*. Grishin concluded that the design of the control law contained two major flaws, the first was the constraint imposed on the foot standing on the support is not bilateral. When the vehicle walks, the leg can rise over the surface or slip along it, this causes confusion in the program and the vehicle tumbles over. The second problem relates to vehicle instability whilst in the single support phase.

The Adaptive Suspension Vehicle (ASV) is probably one of the most successful and well known attempts to design an extreme terrain walking vehicle. Dennis R. Pugh et al *(“Technical Description of the Adaptive Suspension Vehicle”, 1990)*. The ASV is one of the few robots to successfully demonstrate its terrain handling ability, for example, the crossing of a 6’ wide simulated ditch. The

ASV is not autonomous, as it requires a skilled on-board operator, although stable when static, due to its use of a six legged tripod gait. The ASV was first built in 1985 and the time and financial investment required to develop such a complex vehicle was extensive.

Further walking issues are addressed by; Makoto *Kaneko (1985, "A Hexapod Walking Machine with Decoupled Freedoms").* Peter Buhrle *(1996, "Modelling, Simulation and Realisation of an Autonomous Six Legged Walking Machine").* Ivan E. Sutherland et al *(1984, "Footprints in the Asphalt").* Joseph S. Byrd et al *(1990, "A Six-Legged Telerobot for Nuclear Applications Development").* E. F. Fichter et al *(1992, "Walking Machine Design Based on Certain Aspects of Insect Leg Design").* Eric Krotkov et *al (1992 "Performance of a Six-Legged Planetary Rover: Power, Positioning and Autonomous Walking").*

After considering Grishin and Bares papers it became apparent that for a "practical" extreme terrain vehicle a simpler traction mechanism should be sought. It was apparent that the *"control element"* was the largest of the obstacles to overcome, not the physical traction mechanism. Thus, consideration was given to "passive" traction mechanisms. For example, virtually all current research considers robots that utilise legs, that is, a gravity incorporating traction mechanism. It became apparent that a "robot" that was able to reliably ascend or descend a shaft or pipeline, irrespective of gravity, would be a suitable area for future research.

3.5.3 GAITS

McGhee (1968) was one of the first to investigate animal gaits as a basis for walking robots. E. F. Fichter et al (1988) noted that most current walking machines are designed and built for use on smooth, horizontal and regular surfaces, he states the main reason being so that the control and navigation systems could be further developed. This is an agreeable statement, however, control and navigation are fundamental parts of an autonomous extreme terrain vehicle. The question arises; *Should the control or the physical mechanics of the traction mechanism be developed first?*. Legged robots are generally restricted to three degrees of freedom (DOF) and most gaits employ between one and six legs. (D. Gan et al 1985, E. Fichter et al 1987, M. Raibert 1986, I. Sutherland and M. Ullner 1984, K. Waldron et al 1984). Fichter concludes that *"no current walking machine could work successfully in complex environments because neither leg mechanics nor control is sufficiently sophisticated"*.

Shin-Min Song and Yaw-Dong Chen (1991) consider the mobility of a walking machine, whilst negotiating extreme terrain. Generally, gaits for extreme terrain can be separated into two areas: *"gaits for planar rough terrain and gaits for three dimensional rough terrain"*. The latter can be reviewed in: S. Hirose and Y. Umetani (1980), S. M. Song and K. J. Waldron (1987), W. J. Lee and D. E.

Orin (1988), V. R. Kumar and K. J. Waldron (1988) and B. S. Choi and S. M. Song (1988). Gaits for planar rough terrain can be reviewed in: S. J. Tsai (1983), S. M. Song (1984) and S. M. Song and B. S. Choi (1989). The gaits for planar rough terrain include the *"Follow the Leader Gait"*, a special strategy of the *"Wave Gait"* for walking on rough terrain and the *"Free Gait"*. The "Follow the Leader Gait" requires a human operator to choose two stable footholds for the two fore legs, the following legs simply occupy the void left by the leading foot. Song and Chen's "Free Gait" relies upon a computer generated gait which selects the footholds and determines the leg movements based on certain set of programmed rules and a terrain map. A graph search approach to *"Free Gait"* generation is also considered by Prabhir K. Pal and K. Jayarajan (1991).

Some walking vehicles may need to rely upon more than one gait during negotiation of terrain. Chang-De Zhang and Shin-Min Song (1989) noted the need for three different gait types, *"Straight-Line Gaits"*, *"Turning Gaits"* and *"Stair Climbing Gaits"* in their work; *"Gaits and geometry of a Walking Chair for the Disabled"*. It is interesting to consider the number of gait variations a human employs whilst negotiating extreme terrain, arms as well as legs may be used. For example, ascending or descending a steep slope, often jumping is required, side stepping is employed, occasionally a crouching or sitting stance becomes necessary. Most robots or walking machines have limited degrees of freedom in their legs and joints, some are limited to 3 DOF. A human, however, employs significantly more when the whole body is considered. Zhang and Song's main consideration for their chair was the number of legs required, they concluded that the number should be four. Four legs are more compact and agile than six, six may cause problems when the chair negotiated stairs.

The vehicle developed at the University of Durham cannot be said to have a "gait", yet, it virtually guarantees traction whilst negotiating either ill-constrained planar surfaces or shafts/pipelines. Here, the planar vehicle is of particular interest, it is able to obtain a foothold on non-uniform terrain. If struts are unable to climb over an obstacle they are often able to deflect and pass to either side of it. Some evidence to support the principle of a "passive" gait can be derived from a study by K. G. Pearson and R. Franklin (1984) into the *"Characteristics of Leg Movements and Patterns of Co-ordinates in Locusts Walking on Rough Terrain"*. They concluded that; *"...the most obvious feature of leg movements in walking locusts is that individual legs have the capacity for finding support sites independently of the other legs and without input from the eyes and the antennae. Each leg can act as a single functional unit in finding a site for tarsal placement. Three tactics used by each leg to find a support site are (1) rhythmic searching movements, initiated when a leg fails to contact the substrate at the end of the swing phase, (2) elevation of the leg and the placement of the tarsus on an object when the leg strikes the object during swing and (3) rapid shifts of the tarsus from point to point on an object to locate a suitable support site once the object has been found"*.

Following on, Jessica K. Hodgins and Marc H. Raibert (1991, *"Adjusting Step Length for Rough Terrain Locomotion"*) consider the need to adjust the step length of a dynamic biped robot whilst traversing rough terrain. They note that a dynamic legged robot employing a dynamic gait should be able to traverse difficult terrain easier than a static gait system as they do not have the need for a continuous path of support. This type of biped attempts to act as a human would, that is, an ability to jump across voids, run up stairs or jump from rock to rock. The problems the biped face are similar to those faced by a human, for example, terrain sensing, planning a path, selecting a foothold and the step length adjustment. See also; M. H. Raibert et al (1986) *"Running on Four Legs as Though They Were One"*.

R. McN. Alexander (1984) in a paper entitled; *"The Gaits of Bipedal and Quadrupedal Animals"* considers the natural evolution of gaits integrated with the natural learning process of animals. He concludes that; *"The gaits of animals seem designed to minimise unwanted displacements, for example, the slow moving turtle, as well as minimising energy costs"*. He further concludes that robot engineers should consider the above points carefully, as the conclusion for animal gaits apply equally to robot gaits.

3.6 CONCLUSION

This literature review has considered two distinctive areas; pig technology, including, pipeline crawlers and previous research into mobile robots, some of which are autonomous. Although autonomous mobile robots have little to do with pipeline crawlers it has been necessary to review them in order to establish the origins of the thinking behind this research. Early research had shown that only limited research had been undertaken into the area of tunnel and/or shaft exploration. Although the University's vehicle is similar to that of a pipeline crawler, it extends further than the boundaries of conventional crawler theory. It extends into exploration, this has been clearly proven through some of the field trials in collapsed sewers.

The papers reviewed which have considered walking robots clearly indicate the complexities of the control and gait mechanisms involved in leg and foot placement whether static or dynamic. The benefit of a multitude of "uncontrolled", albeit, constrained "legs" becomes apparent. Control and gait problems, are such that, many researchers, for example, Marc Raibert have spent much of their working lives attempting to advance the understanding. However, few researchers have considered the problem from a wider perspective, that is, the simplification or "redesign" of the actual traction mechanism. Had the problem of the control of the legs and the gait drive sequence of a shaft climbing robot not been so complex, thought may not have transcended to the bristle drive principle. A simpler drive system requires less control, thus, the passive system appears to have a number of advantages over "traditional" leg designs. This is especially true where the robot is required to operate across non-uniform terrain or within an ill-constrained environment.

CHAPTER 4

PIPELINE CRAWLER: DESIGN CONSIDERATIONS

4.1 INTRODUCTION

Any pipeline inspection vehicle, whether conventional pipeline pig or self-propelled crawler has basic requirements. Some relate to the vehicle, some to the operating environment, for example. This chapter considers design factors relating to both the vehicle and environment. Consideration is also given to some of the design solutions developed for experimental vehicles.

4.1.1 TESTING THE PRINCIPLE: THE EARLY DAYS

Initially, it was unclear whether the principle of bristle traction would work. To enable the principle to be tried a crude test vehicle was assembled. This consisted of a pair of lavatory brushes, minus their handles, interconnected by a 16mm diameter double acting pneumatic cylinder and controlled by a solenoid valve. The "vehicle" was tested in an acrylic pipe, the bore of which was slightly roughened to improve grip. At this stage, without considering or understanding the mechanics of the bristle, some early force measurements were recorded using a spring balance. The force required to overcome the static friction of the "sliding" brush was approximately 20 N, whilst the "gripping" brush was able to support a vertical payload of, approximately, 70 N. Failure in traction was in the form of bristle buckling and eventually total bristle reversal. From these figures it was concluded that each lavatory brush could support approximately 90 N prior to failure. The experiment relied on the "gripping" brush generating sufficient force to overcome the static friction of the "sliding" brush (20 N) whilst also drawing up the payload of 70 N. A second test used four lavatory brushes at each end of a 25mm diameter double acting pneumatic cylinder. This pulled an approximate payload of 180 N vertically, again, failure occurred through bristle collapse and reversal.

4.2 VEHICLE JACK-KNIFING

A vehicle that is required to negotiate tight (1.5D) bends also requires joints. On early machines, which used joints, the vehicle would jack-knife whilst negotiating both straight and curved sections of pipe. It was realised that a vehicle that utilises a "push-pull" reciprocating traction method would induce jack-knifing. Jack-knifing occurs only during the "push" element of the traction mechanism. This is induced by the friction force exerted by the brush on the pipe wall ahead of the joint. If the brush-joint-cylinder-brush layout is perfectly aligned then the forces transmitted through such components will also be aligned. However, if there is a small misalignment of any component, then a moment will be generated and such a moment will cause the joint to flex, thus causing jack-knifing. Further, in jack-knifing, the joint rotates causing the forward stroke to decrease significantly.

Under certain circumstances the problem of jack-knifing was so extreme as to totally halt all forward motion, that is, the piston stroke was absorbed in translation and/or rotation of the joint. The problem of jack-knifing was a result of a number of factors including; the type and configuration of the joint, the aspect ratio of the brush unit and the reliance upon a single brush unit to provide guidance, suspension and traction. Jack-knifing can be eliminated through the incorporation of regularly spaced sets of guidance/suspension wheels along the vehicle's body. The guidance wheels also provide an additional benefit, they separate the traction and suspension elements of the drive. If the wheels are not included the brush units must perform both of these functions. An additional benefit of the wheels is that they help to guide the vehicle around tight bends, that is, the outer radius of the bend deflects the leading wheels located at the nose of the vehicle. Thus, the vehicle is "guided" around the bend.

4.2.1 VEHICLE ARTICULATION: EARLY IDEAS

The first vehicles manufactured were for sewer inspections. Sewers do not incorporate tight bends, bends, if any, are long with gentle radii. The bends employed in the pipelines inspected by Pipeline Integrity International (PII) incorporated extremely tight radii, down to 1.5 or 1D 90 degree bends. Such bend radii demand a vehicle which is able to accommodate such bends. Standard "universal" joints initially proved unsuitable, due to the problem of induced jack-knifing when subjected to compressive forces, as previously described.

In an attempt to obtain a "joint" that was flexible but did not induce jack-knifing, further designs were considered. A method of achieving articulation was to connect a number of individual brush units together using a tensile cable. A concave hollow was milled into the end of each brush unit. The brush units were separated with a ball which acted as a bearing, not dissimilar to a set of beads inserted on a string separated by tubes with concave ends. The main disadvantage of this type of "joint" was the fact that the cable tension required to retain the components in place resulted in the vehicle remaining relatively rigid whilst attempting to negotiate a bend. If the tension was relieved, the brush units, having a relatively small aspect ratio would jack-knife with respect to one another. A further consequence of relieving the tension in the cable was backlash at the bearing faces, thus, a reduction in axial drive stroke occurred. This effect became most evident on short stroke cylinders, notably 50mm or less, as illustrated in Figure 4.1. This design can be considered as a stiff beam requiring a given force (F) to deflect. For a vehicle of this type to negotiate a bend it must deflect, as such, the bristles on the leading and trailing brush units become hard pressed against the outer radius of the bend. Simultaneously, the main drive cylinder is forced into the inner radius of the bend. Additionally, the tighter the bend the greater the forces exerted, to the extent that bend passage becomes impossible due to the length of the vehicle, its inability to flex and the subsequent force build-up. Therefore, with the cable maintained tight to minimise backlash the brushes no longer act independently but as a uniform and integrated unit. Thus, the advantage of this type of joint design becomes lost, as does the flexibility and compliance of the individual brush units.

Another type of joint investigated was that of a long spring, designed for tension, running through the centre of a number of independent brush units. Separating the brush units and external to the spring were short sections of polyurethane tube. The complete assembly was held together with a tensile cable running through the centre of the spring. The spring became the "backbone" of the mechanism and allowed flexibility. The polyurethane stiffened the joint and helped to prevent jack-knifing whilst restricting the angular movement of the brush units through the natural compliance of the polyurethane. This type of "joint" showed considerable promise with some successful early tests. As with the previous joint the main limitation was the tensile cable. Ideally, the cable tension should be slackened during the negotiation of a bend and then re-tensioned on the straight sections.

The aspect ratio of the brush unit is the ratio of its length to its diameter. If the ratio falls to less than 2:1, that is, two units of diameter to one unit of length, then the brush unit risks becoming unstable and cocked in the pipe. A child's analogy would be that of inserting the lid of a tube of Smarties into the tube itself. Using a finger, it is difficult to push the lid down the tube without it cocking over, any slight force applied off the central axis of the lid will result in instability and hence the lid will rotate. It is able to move off centre and cock over because it is short with respect to its diameter. Due to its short body length there are limited wall reaction forces available to prevent rotation. Alternatively, if the long Smartie's tube were placed inside a pipe of a slightly larger diameter, flipping the tube would not be possible. Here, as the Smartie's tube rotated the reaction forces at the ends of the tube would build rapidly, thus, preventing rotation. The smaller diameter tube is now constrained by the larger diameter tube, as illustrated in Figure 4.2. If short section brush units are required, flipping can be prevented through the use of guidance wheel assemblies, as discussed earlier.

At this stage the requirement for the joint mechanism was beginning to be understood. It was realised that a joint must be as rigid as possible whilst negotiating the straight sections of pipe, thus, helping to prevent jack-knifing. Whilst negotiating a bend the joint should have an ability to deflect to accommodate the bend. It became apparent that an "intelligent" joint was required. Considerable thought was given to a central ball locking device, for example, a ratchet system. As a vehicle approaches a bend when using this type of joint, a "signal" is sent to the joint in the form of a torque induced by the leading guidance wheels on the vehicle. This torque acts on the joint causing it to deflect and relocate in the next ratchet tooth. The force/deflection "curve" of this type of joint is represented in Figure 4.3. It can be seen that the force builds as the joint is required to deflect, as the "tooth" climbs the incline and drops into the next available space on the ratchet wheel the force drops off. This is repeated until the required deflection is achieved. If a tooth stops on the gradient of the ratchet wheel due to the maximum curvature of the bend being reached, then, a constant force may continue to be applied to the joint. That is, the force may not fully decrease to zero. This is represented by the dotted lines on the graph in Figure 4.3.

Thought was also given to an electro-magnetic locking joint. The joint would consist of two disks, each containing segments of separately insulated switchable windings. The plates would be parallel to one another. They would be connected via rubber bobbins, these would provide insulation between the two plates. The rubber bobbins would also hold the joint together when no force was being applied. If the joint was used in a straight section of pipe the segments of the two plates would be charged the same, thus, a repelling force would be generated. This would produce a "locked" joint. If a bend was sensed the electro-magnetic forces could be powered down allowing the joint to deflect against a weak magnetic force field. Due to the fact that each of the sectors of the opposing disks could be charged differently, then, the joint could also be charged to obtain a pre-determined angle. For example, some magnets would repel whilst others would attract one another. The main disadvantage of this system was the size of the windings required to produce the force required to stiffen the joint. However, magnet material technology is changing rapidly, thus, this type of joint may be possible in the future.

4.2.2 RUBBER JOINTS

Progress in the design of joints came in the form of rubber anti-vibration mountings. This type of joint was successfully used on an early polymeric bristled camera inspection vehicle built for Pipeline Integrity International (PII). The photograph in Figure 4.4 illustrates a joint of this type. The joint consisted of two aluminium plates interconnected by three rubber anti-vibration mounts. The rubber mounts are arranged around the centre of the plate with a specific PCD. The Shore hardness, PCD and size of the rubber mountings can all be manipulated to obtain a specific joint stiffness. As the vehicle negotiates a bend the rubber bobbins are subjected to compressive or tensile forces depending upon their position with respect to the joint's deflection. As the vehicle re-enters the straight, the joint returns to a more neutral position, that is, the two aluminium plates return to being parallel to one another whilst the three rubber bobbins return to their unloaded state. As the vehicle negotiates straight pipe sections the rubber bobbins are subjected to alternating tensile and compressive forces.

One drawback of this type of joint is the hysteresis or backlash that can manifest itself in the form of loss of axial drive stroke. This is especially noticeable if the brush units exert high friction forces against the pipe wall. During testing of an "early vehicle" the backlash effect was recorded. A drive cylinder with an original stroke of 50mm delivered a working stroke of between 5mm to 10mm less than the expected value. This is dependent upon the number of brush units in front of or behind the drive cylinder. This test did not consider umbilical drag. The greater the distance the vehicle travels, the greater the weight of umbilical dragged. This results in further backlash. It is not unreasonable to consider that the backlash may eventually equal the stroke of the axial drive cylinder.

Figure 4.5 illustrates the general force/deflection characteristics of a joint incorporating rubber bobbins. A single rubber bobbin will behave in one of two ways depicted depending upon whether it is subjected to compressive or tensile forces. Generally, this type of joint utilises three rubber bobbins, as such, two may be compressed whilst the third is under tension, alternatively, two may be under tension whilst the third becomes compressed. If a rubber bobbin is subjected to a compressive force the force will build with the deflection. As the joint continues to deflect the rubber bobbin is further compressed, as the rubber does not behave in a linear fashion the force begins to rise more sharply. If a bobbin is subjected to a tensile force, the load initially rises sharply. However, the force begins to fall off quickly as the cross-sectional area of the bobbin decreases, due to it becoming stretched. The reader will appreciate that the actual curve will be somewhere between the two curves represented by Figure 4.5. The actual shape of the force/deflection curve will depend upon a number of variables. For example, the PCD of the rubber bobbins and the stiffness of the rubber used, both of which may effect the joints ability to deflect.

4.2.3 RUBBER JOINTS WITH STABILISERS

The crawler can be used very effectively inside badly damaged sewers due to its flexibility and compliance with its immediate environment. As previously stated, sewers do not normally incorporate tight bends, although they do have long sweeping bends. Normally, these shallow bends can be accommodated through the natural flexibility of the crawler's bristles. However, sewer vehicles can be long, for example, in excess of 2 metres for a 375mm (15") diameter vehicle. As such, some vehicle articulation is still required. This articulation is especially useful during the loading of the vehicle, notably, via manhole entry points which have restricted space. The "standard" rubber joint, as described previously, has proved to be extremely reliable for sewer vehicle use, however, the problem of jack-knifing still existed. Previously, to prevent jack-knifing the rubber bobbin type of joints were "stabilised" through the incorporation of guidance wheels. It is not possible to use wheels in the sewer environment, thus, another method to prevent jack-knifing was sought.

A joint suitable for sewer vehicles, as well as other applications, came in the form of the previously described rubber bobbin joint with the addition of three equally spaced steel pegs. The photograph in Figure 4.6 illustrates the joint design. The steel pegs effectively act to reduce the jack-knifing. The pegs share the 360 degree PCD with the rubber bobbins, that is, a rubber bobbin is located at 0 degrees, a steel peg is then located at 60 degrees etc. The pegs are fastened to one of the aluminium plates. The other ends of the pegs incorporate a convex radius, this radius locates into a concave radius milled into the opposing plate. The purpose of the pegs is to stabilise the vehicle during the transmission of compressive forces through the joints. As the compression force is transmitted through the joint, the three steel pegs located in the concave radii, effectively lock. Thus, joint rotation and vehicle jack-knifing is prevented, resulting in a full axial drive stroke being achieved. However, the joint is still able to deflect during bend negotiation. Here, a "signal" is sent to the joint by the leading brush unit as it is moved off its central axis. If this occurs the joints open. One or more rubber bobbins may be subjected to an increased tensile force whilst the remaining bobbin(s) are subjected to a tensile pre-load force, determined by the length of its pegs. During this procedure, one or more steel pegs disengage from their concave radii. The remaining steel peg(s) act as a suitable pivot point allowing the joint to deflect. This type of joint has been proved to be highly effective in sewer environments where the use of guidance wheels is not possible.

The force/deflection curve of this type of joint is illustrated in Figure 4.7. Again, this figure is a representation, the actual curve shape will depend upon a number of variables. This type of joint experiences a high initial load before deflection begins. This is due to the additional force required to move the joint off its locating pegs. After the initial movement is induced the load continues to rise as deflection increases, albeit at a slower rate. Approaching full deflection the load will rise sharply due to one or more of the rubber bobbins becoming fully compressed. This type of joint requires the individual rubber bobbins to stretch or compress further in order for the joint to obtain a suitable deflection. Thus, as the joint approaches full deflection the forces rise considerably.

4.2.4 ROSE BEARING JOINTS

During the development of a high towing capacity vehicle for Pipeline Integrity International (PII) it became evident that the backlash effect induced by the rubber bobbins was unsatisfactory. If the rubber bobbins were used in conjunction with steel bristles the backlash became equal to the stroke of the axial drive cylinder (50mm). A "new" type of joint was sought to overcome this problem. A joint was required that was able to allow angular deflection of the joint, thus allowing the vehicle to negotiate 1.5D, 90 degree bends. The joint was also required to withstand tensile and compressive forces up to 10,000 N. The vehicle that was designed for PII had a payload capability of, approximately, 1 metric Tonne.

The joint design used two Rose bearings inter-connected by a stainless steel shaft, not unlike the general arrangement of a dumbbell. A Rose bearing is a commercially available spherical bearing with a hole through its centre. The bearing is mounted in a housing which allows it to suitably rotate. Without a shaft fastened through the bearing, the bearing is able to rotate freely within its housing. If a shaft is fastened through the bearing the bearing rotation becomes limited by the shaft interfering with the housing, however, a significant deflection is still available. The photograph in Figure 4.8 illustrates a joint of this type. The joint consisted of two steel or aluminium plates parallel to one another. Each plate contains a counter bore of a diameter and depth suitable to accept the Rose bearing housing. The housing was held in the counter-bore by means of a small locking plate.

Three rubber anti-vibration type mountings were arranged around the shaft and served a number of purposes. The rubber bobbins helped to stiffen the joint. Without the rubber bobbins, this type of joint may have been more prone to "jack-knife", whether transmitting a compressive force or not. An additional benefit was that the vehicle was easier to handle, that is, stiffer, when external to the pipe.

The above type of joint has been used for approximately two years and has proved to be extremely reliable. A cyclic test on a joint of this design was recently undertaken at the University. The joint completed in excess of 500,000 full deflection cycles. As the joint deflects, two of the rubber bobbins are subjected to a compressive load, the third is subjected to a tensile load. As the joint is able to deflect through a considerable angle, the rubber bobbin, which experiences the tensile force must stretch a considerable distance. After, approximately, 100,000 cycles, this rubber bobbin failed. The Rose bearings and their PTFE liners showed no signs of "play" at the end of the test and were still fully serviceable. However, if this type of joint was used within a working pipeline environment, then, metallic particles, including rust, may be present. As such, the joint would be expected to wear more rapidly unless suitable protection was provided.

The force/deflection curve for the Rose bearing type of joint will be similar to that shown in Figure 4.5. The Rose bearings simply strengthen and stiffen the joint whilst still allowing it to suitably deflect. Thus, the joint is able to accommodate significantly higher force transmissions without jack-knifing or backlash problems.

4.2.5 INTELLIGENT JOINTS

From the above discussions it is evident that a joint that is stiff on the straight pipe sections and that can deflect during the negotiation of a bend is desirable. A number of "intelligent" joint designs have been considered, including, the magnetic joint described previously. An "intelligent" joint would allow the negotiation of bends without resorting to the use of "brute" force. If a joint was able to predict and pre-orientate itself to accommodate an approaching bend, then, the brush units would also manoeuvre and orientate themselves ready to negotiate the bend.

It is extremely important and beneficial to ensure that the brush units, or as near as possible, follow the central axis of the pipe as they negotiate a bend. This ensures the bristles remain able to induce positive tractive effort. If a brush fails to follow the central axis, some bristles negotiating the outer bend radius will become increasingly deflected to the point of collapse. At this point, the bristles produce zero tractive effort and effectively act as skids. Some bristles negotiating the inner radius will experience a similar effect. Additionally, some bristles on both the inner and outer radii will become remote from the wall, thus, tractive effort is no longer produced. This effect also occurs if the brush unit follows the central axis of the pipe, although to a lesser extent. It can be seen that variable length bristles on the brush cores may be a possible solution to this problem. However, this would be difficult to design, build and control. It can be seen that an "intelligent" joint may be beneficial. For example, if the joint could be "locked" on the straight sections of pipe then the problem of jack-knifing could be permanently eliminated.

One of the most exciting advances in joint design has been the development of a joint which incorporates rheological fluid. This fluid contains small metallic particles which are magnetic. If a current is passed through the fluid, the particles align to the shape of the field (Rheonetic Magnetic Fluids & Systems). Subsequently, the viscosity of the fluid increases as more current is applied. If the current is switched off the fluid returns to being a low viscosity "liquid" again. This technology is currently at an early stage. However, it can be seen that the stiffness of a joint could be changed. The joint would probably require some type of valving, although it may be possible to rely upon the shearing of the "liquid" between two moving components, for example, a ball and socket joint.

Figure 4.9 is a representation of how a rheological fluid type joint may perform. Starting at 0,0 the joint is "frozen", thus, high forces are induced without deflection. If the joint is required to deflect the liquid is "un-frozen". At a given point the joint can be "re-frozen", thus, setting at a pre-determined angle. The joint could then be “un-frozen” again and return to 0,0, that is, zero deflection, before being “re-frozen” once again. Again, this figure is a simple representation of what may occur, as such, the positive and negative gradients are not likely to be linear, as represented.

4.3 TRACTION AND SUSPENSION

In the early days of the vehicle’s development the bristles were unknowingly relied upon to provide both the traction and suspension functions. This was due to a fundamental lack of understanding of the role the bristles played with respect to the above individual functions. Generally, if the bristles were sufficiently stiff to support the weight of the vehicle, then, adequate traction was available.

As development of the bristle traction system progressed it was realised that optimum traction could be obtained by maintaining a specific angle between the bristle tip and the pipe wall. This angle varied depending upon the weight of the vehicle, hence, traction was directly influenced by suspension forces. As previously described the ability of the vehicle’s body to remain suspended as close to the central axis of the pipe becomes increasingly important as the vehicle negotiates a bend. Here, without adequate suspension, sectors of the bristles on the brush core increasingly move off the centre line. Some bristles effectively collapse whilst others on the opposite side of the brush core may lose contact with the pipe wall altogether. The result is a significant reduction in tractive capability coinciding with an increase in the force required to push or pull the brush unit around the bend, due to the additional friction conditions.

The problem of insufficiently stiff bristles emerged in the early stages of the research during a commercial sewer inspection. The sewer was badly blocked with silt deposits. The vehicle was designed with long bristles and a small body diameter to enable it to ride over the silt. The vehicle was designed to allow adequate tractive capability whilst retaining sufficient bristle compliance for obstacle negotiation. However, the weight of the vehicle exceeded the support capability of the bristles and the lower bristles buckled. During the inspection the vehicle started to dip at the nose. It was realised that relying upon the "tractive" bristles to perform both the drive and suspension function was not going to be satisfactory. A method was sought which would separate the two functions.

If a pipeline is of a known and uniform ID it is possible to incorporate suspension wheels on a carrier. These took the form of six equidistant radial wheels emerging from the vehicle body at specific intervals along the vehicle train. This allowed the bristle traction properties to be maximised, that is, dedicated to the pipeline in which it was to operate. The inclusion of the wheels had the secondary benefit of eliminating jack-knifing. In pipelines with 1.5D bends the suspension wheels also acted as guidance wheels whilst the vehicle negotiated the bend. Jack-knifing on the bends was reduced, resulting in the vehicle increasing its effective velocity through the bends by maximising the stroke of the cylinder available. Previously, without the guidance wheels the vehicle would attempt to drive into the outer radius of a bend relying solely on the stiffness of the bristles to adequately deflect the vehicle around the bend. During this manoeuvre significant slippage occurred resulting in loss of stroke and reduced payload capability.

If a vehicle was required to negotiate damaged sewers the incorporation of guidance wheels was not possible. The wheels would lock up against obstructions, for example, misplaced pipework, resulting in the vehicle becoming stuck. Sewer or exploration vehicles require the smallest body diameter, mated to the longest bristle length possible. This enhances its ability to bypass obstacles but can also cause the vehicle to collapse under its own weight. The designer has two choices. He/she can increase the diameter of the wire used for the bristles, thus, for a given length of bristle a higher weight can be supported before buckling is induced. This can cause problems due to the increased force required to propel the vehicle. The second possibility is the inclusion of "suspension" bristles. These are bristles that extend radially from the core and are a slight clearance fit inside the pipeline. As these bristles are perpendicular to the core and are not subject to preset bristle deflection they require a significantly higher force to buckle than the force required to buckle the "traction" bristles. As such, a small number of these bristles are able to support the weight of the vehicle and prevent jack-knifing. Additionally, the "suspension" bristles are still able to deflect easily when meeting obstacles.

The problem of traction and suspension, especially during bend negotiation is an extremely complex one. In the case of a "simple" three brush/two cylinder machine the vehicle will display the following characteristics during bend negotiation. As the leading brush enters the bend its bristles contact the outer radius of the pipe, this causes the brush to be deflected around the bend. The bristles contacting the outer radius become increasingly deflected due to the increased forces acting upon them. During this manoeuvre the leading brush moves off the centre line of the pipe. Simultaneously, the bristles in contact with the inner radius become straight. Thus, the bristles are no longer able to obtain optimum tractive effort from their contact with the pipe wall. Here, the swept back bristles simply act as skids whilst the bristles near the inner radius are unable to contribute to the tractive effort and are liable to flip. As the middle brush enters the bend the opposite bristle displacement occurs. That is, the bristles adjacent to the outer radius of the bend move away from the pipe wall and return to being radially straight. Simultaneously, the bristle nearest the inner radius become increasingly deflected. Again, the optimum tractive capability of the brush is lost. As the trailing brush enters the bend it mimics the displacement of the leading brush. In simple terms, the complete vehicle train is attempting to remain as straight as possible. Thus, the leading and trailing brush units are pressed towards the outer radius of the bend whilst the middle unit is pressed towards the inner radius [Figure 4.1]. This phenomenon was also noted on some of the early "experimental" vehicles built to test various types of joint configuration.

It can be seen that an ability to suspend the brush units as close to the centre line of the pipe is beneficial to the vehicles tractive capability. Generally, in mechanical terms, the bristles which provide the tractive element are those which exert an axial force and the bristles which provide the suspension element are those which exert the radial force. Therefore, it can be seen that traction and suspension are integrated functions and when designing a vehicle they must both be considered. In conclusion, it is evident that a number of benefits can be obtained by separating the two functions. This can be achieved by incorporating either wheels or "suspension" bristles for suspension and swept bristles for traction.

4.4 BEND NEGOTIATION: CYLINDERS AND JOINTS

As discussed above, the negotiation of tight bends can be difficult, especially whilst maintaining maximum traction. The pneumatic cylinders, which provide the axial drive, generally constitute the longest fixed sections of the vehicle train which must negotiate a bend. The negotiation of 1 and 1.5D bends necessitates short "body" lengths, interconnected by, regularly placed joints. As such, a vehicle designed to negotiate such a bend is limited to a specific pneumatic cylinder stroke length.

The maximum pneumatic cylinder stroke length that is able to negotiate a bend also depends upon the diameter of the piston inside the cylinder. For example, it may be possible for a 32mm diameter cylinder with a stroke length of 65mm to negotiate a 6" 1.5D 90 degree bend. Conversely, it may only be possible to get a 50mm diameter cylinder with a stroke length of 50mm around the same bend. The diameter of the piston, as well as the pressure it operates at, dictates the thrust and pull available from that particular configuration. This, in turn, dictates the payload capacity of the vehicle. As with all other sections of a vehicle train, the aspect ratio of the cylinder dictates its ability to negotiate a bend. That is, its length to width ratio.

Generally, the largest number of bends occur at the beginning of a pipeline. This is in the main due to the pipeline entering the ground and changing direction to avoid obstacles. If access to a pipeline can be obtained from both ends the largest bend population may be at both the entrance and exit. For example, a pipeline could be above ground, it could then submerge, traverse a main road and re-emerge again. It can therefore be seen that it would be advantageous to have long strokes on the straight sections of pipe, for example, 150 to 250mm. However, the maximum stroke that can be obtained in a 6" and 8" 1.5D 90 degree bend is, approximately, 50 and 60mm, respectively. A 50mm stroke pneumatic cylinder requires three times the number of valve cycles to cover the same distance as a 150mm cylinder. Therefore, the additional valve cycles require time, this can slow the vehicle down over long distances. Additionally, pneumatic cylinders require time to build up pressure prior to moving the load ahead of them. Therefore, a longer stroke cylinder would be more efficient.

A second disadvantage of using short stroke cylinders is their inability to accommodate hysterisis present in the vehicle system. As already described, hysteresis can emanate from the use of rubber bobbin joints, through the joints jack-knifing and if the vehicles are used without the use of guidance wheels. Hysterisis may also occur if the forces being applied to the brush units are significant, for example, during the movement of heavy payloads. Here, slight additional bristle deflections may occur which will manifest itself as hysterisis. Therefore, if hysterisis is present, it is beneficial to utilise longer pneumatic cylinder lengths. An additional benefit of long cylinder lengths is that their long aspect ratios are better able to resist jack-knifing compared to shorter cylinders.

Unfortunately, as previously described, increased pneumatic cylinder lengths are unable to pass tight bends without fouling. If extended cylinder strokes are used in pipelines with tight bends there is an increased risk of cylinder/shaft wall collision. However, prior to the collision, the cylinder may extend a short distance before jamming on the pipe wall. If this occurs the vehicle may still be able to "shuffle" a passage around the bend. However, this would not be practical from a commercial perspective as damage to the cylinder shaft and/or cylinder nose bearing could result. One possible solution may be the use of telescopic cylinders, this idea has not been given any further consideration.

As the vehicle uses double acting pneumatic cylinders one must note that their overall length, when extended, is often double the stroke length. This is due to the fact that piston length, sealing, rear wall thickness and the nose bearing assembly must be accounted for. Generally, from the commercially available cylinders, the ISO compact cylinders have the shortest closed body length for a given stroke length.

To conclude, it can be seen that short cylinders are able to negotiate the tight bend radii but are slower and less efficient on the long straight sections. However, they can be cycled very quickly and in real terms, they are only slightly slower than the long stroke cylinders. This statement assumes that both cylinders have the same piston diameter. If the cycling speed of the cylinder becomes important it is necessary to use the smallest diameter cylinder possible for the payload/umbilical weight. Larger diameter cylinders require longer to fill and exhaust. This significantly affects the overall velocity of the vehicle.

4.4.1 PIPE BORE CHANGES

Due to the dominance of the oil industry in terms of pipework design and specification, pipework is generally measured in the American/Imperial system. The tightness of the bends are specified according to their ID. For example, common pipe radii are 1, 1.5 or 3D. Others such as 0.75D and 5D are also manufactured. Some bends are continuously formed and others are of a mitred construction. A 6" 1.5D bend measures 9" from the flange face to the centre line of the radii, that is, 1.5x the pipe ID, hence, 1.5D. Pipe weights are specified as the schedule of the pipe. A pipe will have a nominal outside diameter, for example, 8.625" (219.08mm). However, the pipe wall thickness can vary, thus, reducing the internal diameter and increasing the weight per given length, Table 4.1 below shows the pipe weights and sizes for schedules 40, 80 and 120 8" nominal OD pipework.

	PIPE ID	OD (INCHES)	CIRCUMFERENCE (INCHES)	WEIGHT PER FOOT (POUNDS)
Schedule 40	7.981	8.625	27.09	28.6
Schedule 80	7.625	8.626	27.09	43.39
Schedule 120	7.189	8.625	27.09	60.70

Table 4.1 The Schedule Pipe System

The relationship between pipe schedule, internal diameter and weight of pipe is clearly evident. A smaller internal diameter makes 1 or 1.5D 90 degree bend negotiation more difficult, not due to the radius, but the reduced cross sectional area available to accommodate the vehicle hardware. This problem can be complicated further by the incorporation of fixed suspension wheels and fixed length bristles. It was realised that some degree of compliance would be required at the wheels in order for the wheel assembly to bypass internal weld beads and mitred bends. This was accomplished through the incorporation of relatively soft (Shore hardness 40) polyurethane tyres, which allow approximately 10mm of "suspension" across the diameter of the vehicle. The contact edge of the tyre also incorporates a radius. This reduces tyre drag as well as preventing the tyre from forming "feathered" edges. The latest generation of steel bristled vehicles have approximately 14mm of "suspension" across the diameter of the vehicle through the redesign of the tyre/wheel assemblies. In the future it may be desirable or necessary to design sprung wheel carrier assemblies.

Commonly, a commercial pipeline can and does change schedule, thus, during inspection, ID bore changes are common. As the vehicle incorporates "fixed" wheel assemblies, which includes tyres of limited compliance, it is not possible for the vehicle to be a perfect fit in both schedule 40 and 80 pipelines. Therefore, the guidance wheel assembly has been designed to be a slight interference fit in the smaller ID pipeline, for example, an interference of 0.5-1.5mm in schedule 80 pipe. In the larger ID pipe (schedule 40) the wheel assembly becomes a slight clearance fit.

As previously mentioned, future generations of the vehicle may incorporate sprung suspension wheels. This would allow the schedule of the pipe to be more precisely accommodated. Currently, PII, utilise sprung wheels on the front of their conventional pigs. This is to facilitate keeping the nose of the pig up. The nose dips due to a combination of friction generated at the bristle tips and the clamping effect of the bristles which are magnetically charged during inspection, further, gravity also pulls the nose of pigs down. The pigs used by PII are heavy in comparison to those used by the University. The wheels also perform a secondary function, that is, to deflect the vehicle as it negotiates bends. This reduces friction, prevents the pig from jamming and reduces wear and damage to the bristles and drive cups.

4.5 CYLINDER AND BRUSH CONFIGURATION

The vehicle developed by the University can use pneumatic, hydraulic or electric power. Brief consideration has also been given to petrol/diesel engine power. Some of the latest generation vehicles do not require an external source of energy. Currently, work is being undertaken into vehicles which are able to harness their energy from the ambient flow within the pipeline and subsequently crawl against the flow.

Pneumatic cylinders are used to deliver forward axial drive. Most pneumatic cylinder applications require pressures of between 7 and 10 bar (101.5psi to 145psi) to operate. If a vehicle is required to tow high payloads, pressures up to 20 bar (290psi) may be used. Due to the large payloads it is sometimes necessary to generate higher thrust and pull forces. This would normally be achieved by using a cylinder with an increased piston diameter. However, as the cylinders are required to operate in confined spaces this can sometimes only be achieved by using higher pressures on smaller piston diameters. Compressed air has the advantage that it is a "clean" source of energy, it can be easily generated and subsequently exhausted to atmosphere. In pneumatic terms *thrust* is the "compressive" force exerted by the piston rod as it extends, that is, the pressure acts over the total cross-sectional area (CSA) of the piston. In pneumatic terms, *pull* is the "tensile" force delivered at the piston rod as it retracts, that is, the pressure acting over the CSA of the piston minus the CSA of the piston rod.

Hydraulic cylinders have been considered on a number of occasions, mainly if the payload capacity has exceeded the ability of the pneumatic energy sources. For most applications the size of the hydraulic solenoids and cylinders prohibit their use in pipelines below 8" diameter. For example, the cylinder requires significantly more material to withstand the increased pressures, as such, the overall closed length of a 50mm hydraulic cylinder is too long for a 8" 1.5D, 90 degree bend. This problem increases further after a flexible joint has been added to each end of the cylinder. Even with the additional thrust available from the cylinder the weight of the umbilical may reduce the vehicles range. This may become more pronounced as it becomes necessary to drag the umbilical around increased numbers of bends. Hydraulic fluid must also be pumped back to a supply tank. Occasionally, the off-shore industry uses sea water as the "hydraulic fluid". This can have the advantage that it can be "exhausted" back to the sea. It may be possible to have an on-board hydraulic pump with a closed circuit design. To date, the University has not found it necessary to resort to hydraulic power. When PII required a high payload capacity vehicle the University tested some commercially available pneumatics at elevated pressures. PII were satisfied that the system was still "safe". In an 8" diameter pipe it was possible to generate payload forces in excess of 10,000N (1 metric tonne).

As development progressed the question of number of cylinders, number of brushes, cylinder phasing, cylinder positioning on the vehicle and vehicle velocity arose. As previously mentioned, the first test vehicles simply consisted of a brush at the front and rear interconnected by a pneumatic cylinder. This type of design is still very useful where speed is important. If traction and/or payload capacity is important, a vehicle which uses the three brush/two cylinder configuration is more desirable. If the vehicle is required to tow high payloads this brush configuration has a significant advantage over the two brush/one cylinder configuration. The vehicle design ensures that only one brush unit negotiates a bend at one time, thus, the vehicle has the benefit of always having two other brush units delivering maximum tractive effort. A three brush/two cylinder vehicle is especially useful in a sewer environment. Here, the vehicle is able to extend individual brush units over obstacles, this ensures that in a wet, slippery environment two brush units are always giving maximum tractive effort. This design is also crucial where bi-directionality is required. Increased vehicle length only has one disadvantage and that is the increased difficulty of loading. This is most evident during the loading of sewer vehicles where access via manholes can be extremely limited. However, if the leading brush unit of the vehicle can be loaded and a suitable reaction force can be applied to the trailing brush unit in the form of a force applied by an operator it is possible to drive the vehicle into the pipe under its own power. Extracting the vehicle from a manhole is also possible using a similar method, that is, it is driven out with the last brush to emerge being "helped" out by the use of a winch.

A vehicle with more than three brush units/two cylinders may be required in a pipeline with a large number of bends or if the payload is high. However, it must be remembered that it is the last cylinder on the vehicle train, which, when exerting a pull force, draws up the trailing brush unit, solenoid carrier and umbilical. Therefore, increasing the number of brushes and cylinders is unlikely to have any great effect on payload capacity, although traction during bend negotiation may be improved. If increased payload capacity is required, a larger cylinder and/or increased operating pressure may be the only answer. It may prove beneficial, for example, to use an air-motor or closed circuit on-board hydraulic cylinder for the trailing axial drive unit.

The three brush/two cylinder vehicle is driven through a logical switching sequence. As the leading pneumatic cylinder is filled it causes the leading brush unit to slide forward. Simultaneously, the middle and trailing brush units provide the grip on the pipe wall. The leading pneumatic cylinder is then exhausted whilst the trailing pneumatic cylinder is filled. Simultaneously, the leading and trailing brush units grip whilst the middle brush slides forwards. The trailing cylinder is then exhausted. As this happens the leading and middle brush units grip whilst the trailing brush unit is drawn up. This concludes one complete cycle, Figure 4.10 illustrates the principle. The reader will observe the cylinders are double acting, therefore, each cylinder must traverse 100mm to ensure the vehicle progresses by 50mm. Therefore, the vehicles pneumatic cylinders must travel 200 mm per cycle to obtain a vehicle movement of 50mm, if a 50mm stroke cylinders are used.

4.6 VEHICLE VELOCITY

During the early stages of development the solenoid valve was remote from the vehicle, that it, it remained outside the pipe being inspected. This was primarily for reliability reasons. This required exhausting all of the pneumatic line to the vehicle, as well as the cylinder. This resulted in a large additional volume of stored energy being released for every cylinder cycle. Thus, the time required to refill the cylinder and line depended upon the length of umbilical. This resulted in the vehicle being extremely slow. During some of the early commercial sewer trials this principle was adopted, again for security reasons. The "second generation" sewer vehicles were designed with on-board solenoid control valves. With this system only a short length of pneumatic line is required to exhaust per cycle, that is, the length between the solenoid valve and the cylinder. This basic change increased the vehicles velocity by a factor of 8.

It is important when designing pneumatic circuits to ensure that the complete circuit is designed to ensure maximum flow. If restrictions exist in the circuit this will effectively slow down the extension and retraction velocity of the cylinder. When designing the circuit it is necessary to ensure that the cylinder volume and port size matches the ID of the pneumatic line. The valve flow per hour must also be considered, too low and the overall velocity of the vehicle can be significantly reduced. The fittings or unions that are used to connect the pneumatic pipe work to the cylinders and valves must also be able to accommodate the required flow rates. For example, banjo type unions can cause severe restrictions to flow and are best avoided. The pneumatics industry uses a number of different methods to describe the flow capability of a component. The most common is the C_v factor. A C_v of one is equal to a flow rate of one US gallon of water per minute, with a pressure drop of one psi.

The polymeric bristled vehicle which utilises two 32mm diameter pistons x 50mm stroke cylinders has a recorded velocity of 0.124 m/s, 7.43 m/minute or 446m/hour. At the time the nearest competitor, Pipecat, had a quoted velocity of approximately 100m/hour. A steel bristled vehicle built for PII utilises two 80mm diameter pistons x 50mm stroke, this vehicle is slower, figures recorded were; 0.05 m/s, 2.92 m/minute or 175.4 m/hour. A special "sprinter" concept vehicle has been designed and built specifically aimed at the telecommunications/cable laying industries. This is a two brush/single cylinder vehicle. This has a velocity of 900m/hour.

4.7 VEHICLE RANGE

To calculate or even estimate the range a vehicle is capable of is extremely difficult. This depends upon a number of variable factors, notably bend radii and population and the pipe schedule. Ensuring a vehicle is adequately able to negotiate a bend is not difficult. However, the vehicle's umbilical is subjected to a capstan effect as it negotiates the bends. The greater the number of bends the greater the effect. It is not difficult to see that if a large number of tight bends occur at the beginning of a pipeline then the drag on the umbilical will rise quickly.

Subsequently, the vehicle may not be able to traverse the remainder of the pipe because the drag of the umbilical will exceed the mechanical capability of the vehicle.

The axial drive cylinders also limit the range of the vehicle. For a given pressure they are able to exert a specific thrust or pull. Additionally, the brush units require a specific force in order to move them, that is, to overcome the static friction generated at the bristle tip/pipe wall interface. Typical figures for an 8" PII vehicle traversing a dry, schedule 40 pipe are:

Force required for leading brush to overcome static friction	= 1687N
Force required for trailing brush to overcome static friction	= 1962N
Theoretical cylinder thrust (80mm Dia. piston) at 16 bar	= 8044N
As above less 20% for frictional losses	= 6435N
Theoretical cylinder pull (80mm Dia. piston) at 16 bar	= 7259N
As above less 20% hysteresis	= 5807N

Thus, briefly it can be calculated that:

5807N - 1962N = <u>3845N</u>

3845N is the approximate reserve force available for umbilical drag, if umbilical drag is dependent upon bend population and F= μ N.

Thus, equating the maximum available force from the cylinder to the total resistance to movement gives:

$$\mathbf{[(Cyl._D - Ptn._d).PRESS - F_F\ ] = [Bsh._{SF}+Bend_F+(Length\ Umb.)\ K]}\ ^{1}$$

(max. force available) = (total drag)

It can be seen that the major limiting factor to vehicle range is umbilical drag, that is, $Bsh._{SF}$, $Bend_F$, Length Umb. and K. Here, the drag increases from the capstan effect at the bends as well as the frictional drag in the straight pipe sections. Additionally, the drag increases further as the bend radii decreases, it also rises as the bend population increases. Umbilical drag varies due to other external factors. For example, the surface roughness of pipe wall and/or any "lubricant" present inside the line. However, the most significant factor in determining vehicle range is the number and severity of bends. The greater the number of bends in the pipe the greater the induced capstan effect as the umbilical is dragged around those bends. The reader will appreciate that the vehicle will eventually stop when the drag of the umbilical is equal to the force generated by the cylinder less the frictional factors already described. Sewer vehicles do not generally have the problem of bends. Most sections of sewer pipe are "straight" with the maximum distance between manholes being, approximately, 150 Metres. The average distance between manholes is approximately 100 Metres. Some of the early University sewer inspection vehicles had theoretical linear ranges in excess of

[1] Where: $Cyl._D$ is cylinder piston diameter, $Ptn._d$ is cylinder piston rod diameter, F_F is cylinder friction, $Bsh._{SF}$ is brush static friction, Length Umb. is umbilical length and K is the drag per unit length of umbilical.

7km. Unfortunately, most oil and gas transmission lines have considerable bend populations, therefore, the range of the vehicle in these pipelines is reduced considerably.

Early vehicles utilised a polyurethane sheathed umbilical which induced high drag because of the high coefficient of friction between the polyurethane and the pipe wall. Later nylon 11 and nylon 6 were used for the outer casing of the umbilical. This reduced the drag significantly. A simple "pull-through" experiment was carried out in order to determine the effect umbilical "jacket" materials had on the drag of the umbilical. Two materials were chosen for the "jacket", they were polyurethane and nylon, as previously described. Two 1.5D, 90 degree bends were bolted together to form a 180 degree "U" bend. As the two umbilicals passed the bend, their drag was recorded. To ensure the polyurethane and nylon pipes were kept taught various weights were attached to their ends. Two tests were undertaken, test A and test B. Test A consisted of the tension weight being pulled vertically, against gravity. Test B pulled the tension weight horizontally across a desk top. The results of the two tests are detailed in Table 4.2 below.

	TEST A	TEST A	TEST B	TEST B
WEIGHT	POLY.	NYLON	POLY.	NYLON
10 N	123N	27N	39N	10N
20 N	226N	49N	69N	12N
30 N	342N	74N	98N	20N
40 N	520N	118N	147N	34N

Table 4.2 Drag Due To Different Umbilical Sheath Materials

In addition to the standard load build up due to the umbilical being pulled around a bend, forces through multiple bends must be considered, that is, compound drag. The compound drag will build up rapidly and will also depend upon bend location. To date, a polymeric bristled vehicle has negotiated a 28.5m course consisting of 4x45degree bends and 4x90 degree bends. The straight pipework was in a vertical orientation. The vehicle was not overcome by the umbilical drag. A steel bristled vehicle similarly negotiated an 8" simulated river-crossing test track. This had an overall length of approximately 60 metres. It consisted of 8, 3D bends and additionally, the pipework alternated between schedule 40 and 80. During polymeric bristled vehicle trials the drag on the umbilical was recorded. To pull 30m of "raw" PVC cable required 147N. The PVC cables were then sealed inside a Nylon "jacket", the drag dropped to 17N. The correct choice of umbilical "jacket" material can be clearly seen to be important.

4.8 PIPE WALL CONDITION

In addition to the bristle theory, which is covered in Chapter Five, another factor governing vehicle performance is the roughness and lubrication condition of the pipe wall. It can be seen that the interface between the bristle tip and pipe wall is critical in terms of forward slippage and rear grip. Wall condition can be considered in two parts; surface roughness - the R_a value[2] and lubrication. The vehicle can be used to inspect pipelines carrying a diverse range of product, for example, oil, gas, ammonia, fuel oil, water, petrol and diesel. It can therefore be seen that the surface roughness and friction conditions of these pipelines will vary widely. Under lubricated conditions the force required to slide the brush unit forward can be reduced. However, under the same conditions, the ability of the brush to grip the pipewall is also reduced. Under lubricated conditions it is necessary to choose the bristle material, bristle diameter and the angle the bristle tip forms with the pipe wall carefully. This ensures that optimum tractive capability is maintained. Pipewall condition is important to vehicle traction, as is tyre choice to a Formula One racing car.

4.8.1 PIPE WALL ROUGHNESS

Roughness can be defined as "the irregularities in the surface texture which are inherent in the production process but excluding waviness and errors of form. Waviness may arise from such factors as machine or work deflections, vibrations,

[2] R_a is the arithmetical mean deviation. This the arithmetical average value of the variation of the profile above and below a reference line throughout the prescribed sampling length. The reference line may be the centre line, this being a line chosen so that the sums of the areas contained between it and those parts of the surface profile which lie on either side of it are equal.

heat treatment or warping strains" (Bolton 1989). "Roughness can be considered as the arithmetical mean deviation (R_a). This is the arithmetical average value of the variation of a profile above and below a reference line throughout the sample length." (Bolton 1989). The reference line is the centre line which keeps the peaks and troughs equal.

W Bolton defines some examples of surface roughness values as:

Surface Roughness	*R_a (μm)*
Very Rough	50
Rough	25
Semi-rough	12.5
Medium	6.3
Semi-fine	3.2
Fine	1.6
Course-ground	0.8
Medium-ground	0.4
Fine-ground	0.2
Super-fine	0.1

Table 4.3 Surface Roughness Designation

The table above means little, in terms of corroded steel, however, it does give a "bench-mark" to surface roughness. Bolton also lists surface roughness values for common surface finishes, they are;

Process	R_a (μm)
Sand Casting	12.5 to 25
Die casting	0.8 to 1.6
Investment casting	1.6 to 3.2
Hot rolling	12.5 to 25
Forging	3.2 to 12.5
Extrusion	0.8 to 3.2
Cold rolling	0.4 to 3.2
Turning	0.4 to 6
Planing and shaping	0.8 to 15
Milling	0.8 to 6.5
Grinding	0.1 to 1.6

Table 4.4 Typical Surface Roughness Values

The R_a value can increase the force required to move a brush unit forward through a pipe due to the increased drag. Additionally, a high R_a value can accelerate bristle tip wear, most notably when polymeric bristles are used. The R_a value of the pipe wall has a considerable effect on the performance of the vehicle. If the surface roughness is considered at a microscopic level, the greater the R_a value, the greater the surface "roughness". In terms of corroded pipework, this roughness is most noticeable in the form of surface pitting. This pitting increases the number of potential "footholds" in which the bristle is able obtain a suitable purchase.

4.8.2 LUBRICATION

Generally, lubrication has the effect of "contaminating" the surfaces in contact. In the case of the bristle tip and pipe wall interface, lubrication prevents adhesive contact. The lubricant shears easily when a shearing force is applied. This normally has the benefit of reducing the coefficient of friction. This is beneficial if a brush unit is travelling forwards, less so, if the bristles are required to grip.

Whilst the brush units are driven in a forward direction it is desirable to lower the friction force between the bristle tips and the pipe wall. A condition where this may occur would be in a pipe line carrying an oil based substance. The advantage of lowering this force becomes evident when the last axial drive cylinder in the line draws up the last brush and/or the "payload", including the umbilical. Therefore, the lower the total force required to drag the brush unit forward, the greater the additional force available for dragging the payload and umbilical. When the bristles are required to "grip", any lubricant remaining in the pipe could be considered as detrimental, perhaps inducing rearward slippage. If the surface of a pipe is smooth and a suitable purchase cannot be found, then some rear slippage may occur. Subsequently, the ratio between the forward "slip" force and reverse "grip" force is reduced.

Some "heavy" oils may cause other problems. For example, fuel oil is so viscous that it must be pre-heated simply to allow it to flow. If fuel oil is present, its viscosity is such that it is extremely difficult for the bristles to cut through to the pipe wall to obtain a satisfactory purchase. Under these circumstances forward drag is lowered but not significantly, due to an increase in viscous drag. Additionally, the bristles become heavily contaminated and the brush unit becomes very "rigid". If this occurs the bristles loose their compliance, thus, the brush acts as a "rigid" body. Consequently, in reverse, the valuable "grip" force is decreased.

4.9 BI-DIRECTIONAL VEHICLES

During the initial development of the vehicle considerable thought was given to tractive performance. As the development continued it became apparent that the vehicle design was becoming suited to uni-directional operation. The bristles were becoming shorter and stiffer making passive brush reversal more difficult. Thought was then given to designing a bi-directional crawler.

A "standard" brush that has a force applied axially along its core will transmit the force through its bristles and into the pipewall, hence it grips. However, as the axial force increases, the force transmitted through the bristles also increases. When the force reaches a critical level and the bristles are still engaged with the pipe wall, one of two conditions may occur. The first is that the brush becomes unstable due to the high loads being applied. Without warning the core begins to rotate, as if a couple had been applied to it. The rotation can be either clockwise or anti-clockwise. As the core continues to rotate the bristles move through an arc, that is, they "flip" or move out of plane. During this phenomenon the brush core is translated axially through the pipe. When the bristles have reached their maximum, out of plane rotation, the stored energy within the bristles is suddenly released. The result is that the core is propelled forwards (or rearwards) along the pipe as the bristles extend outwards. Figure 4.12 illustrates the bristle movement and core translation. At this point, the bristles have changed direction totally and the brush core, to accommodate the bristle movement, has translated axially. If the bristles are short and stiff this is less likely to happen due to the roughness of the wall not being sufficiently adequate to engage the bristles securely. Here, the bristles simply begin to "stutter" or slip across the surface of the pipe wall, resulting is brush slippage, this is the second condition.

If the bristles are able to flip out of plane, as described above, it may be possible to reverse the vehicle simply by using a tow rope. However, where bends occur the tow rope is subjected to a capstan effect, thus, after a short distance it is no longer possible to apply sufficient force through the tow rope to reverse the vehicle. Additionally, if cross-ply brush units are employed for high axial loads, passive reversal becomes impossible.

These reversal "mechanisms" can be reduced in dry pipes by the use of cross-ply (X-ply) brush units. X-ply brush units are designed to prevent brush rotation. This is effected by orientating 50% of the bristles at +5 degrees from the radial centre line whilst the remaining 50% are orientated at -5 degrees. The "couple" reversing mechanism described above is induced when the supporting bristles become unstable. That is, a majority "decide" that it is easier to rotate than to continue to support the load. This preferred orientation may occur through the bristle material being slightly inclined, perhaps introduced during the manufacturing stage. The cross-ply mechanism physically ensures that an equal but specific angle occurs. Hence, the bristles "fight" each other to rotate the core in their respective preferred directions. Core rotation does not occur and the only way of releasing the stress in the bristles is through slippage or plastic deformation.

Initial thoughts and experiments on intentional brush reversal considered the possibility of using two different types of brush on a vehicle. A vehicle was tested that used two different brushes, that is, the leading brush contained fewer bristles than the trailing brush. Reversal relied upon the fact that the leading brush, when traversing the pipe would come up against a T-junction or blockage. As the cylinder continued to attempt to drive forward an increase in force would occur at the nose of the vehicle. This force build-up would be transmitted back through the cylinder to the trailing brush. On the next stroke of the cylinder and assuming sufficient thrust or pull could be generated, the trailing brush would flip and the bristles would re-orientate to the opposite direction. As the trailing brush was able to deliver a greater reverse "grip" force than the leading brush, at the next stroke of the cylinder the leading brush would also flip and the vehicle would drive off on the return leg of its journey. This theory was proved in the laboratory by driving a small vehicle through a perspex pipe into a wall, the vehicle flipped successfully. Two main disadvantages exist with this vehicle; the first is that the vehicle can only reverse if a sufficiently high reaction force can be generated. However, it may be possible to add a brake to the vehicle which would perform the same function as the reaction force. The second disadvantage is that once the vehicle has reversed it will be on a one way trip back.

As previously mentioned, the "standard" vehicle design uses a three brush/two cylinder configuration. Some considerable time was spent attempting to use variable cylinder switching sequences. It was hoped that this may enable the vehicle to reverse simply by using the differential forces produced by different brushes. The theory was based around the fact that the forward sliding friction forces of two brushes may be enough to reverse the third, the "gripping" brush. A "standard" brush has an average slip to grip ratio of 1:4. Thus, it can be seen that, if two brushes were sliding and one was gripping the resultant ratio would be 2:4. It can be seen that this would not be enough to enable the "gripping" brush to flip. A group of Durham University undergraduates undertook a two week course in which they attempted to apply a different principle. They attempted to incorporate a rotary pneumatic cylinder into the design. It was hoped that the additional rotary motion would help to induce the bristle flipping mechanism as described previously. Unfortunately, it did not succeed. It was concluded that reversal did not occur due to the stroke of the drive cylinders being too short, thus, they were unable to translate the rotary motion into a useful component force. That is, the axial drive cylinder stroke was not long enough to complete the full flip through cycle of the brush units - the short stroke of the axial drive cylinders were effectively acting as a premature stop to the flipping process.

It was realised at this point that a more reliable method of reversal must be sought. If the forces transmitted through the bristles into the pipe wall could be reduced or removed altogether then it may be possible to suitably re-orientate them. It was realised that a method was required to collapse the bristles away from the wall. By sequencing the axial drive cylinders whilst re-extending the bristles outwards it was possible to re-orientate the bristles. Collapsible brush units were designed. The brush units feature a hollow tube which is sealed at both ends, effectively a pressure vessel. Around the perimeter of the tube are six individual plates running the length of the pressure vessel. These plates are made from a tube with an ID the same as the OD of the pressure vessel. The "tube" is then cut to form the six individual plates, these act as the bristle carrier plates. Each carrier plate is connected to four pistons, the pistons are sealed and extend radially out of the pressure vessel. Air pressure is introduced into the pressure vessel. Simultaneously, the pistons extend out, pushing the bristle carrier plates outwards. To collapse the bristles the air is exhausted from the pressure vessel/brush core and the natural spring effect of the bristles on the pipe wall push the bristle carrier plates flat against the brush core body. The procedure to reverse a three brush/two drive cylinder vehicle is as follows;

The vehicle is driven as far as required into the pipe, at this stage the brush cores are open and both axial drive cylinders are closed. The trailing brush unit is exhausted and the bristles retract away from the wall. The trailing brush unit is then re-filled forcing the bristles back out and into contact with the wall. A moment after this procedure the trailing axial drive cylinder is activated and extends. This causes the bristles to re-orientate in the opposite direction, the trailing brush unit is now in the reversed position. The next step is to exhaust the central brush unit. Once exhausted it is refilled and the bristles extend out towards the wall. The leading axial drive cylinder is activated and extends outwards. Simultaneously, the trailing axial drive cylinder is exhausted. This ensures that the re-orientation of the central brush unit is now complete. The final stage is to exhaust the leading brush unit and then re-fill it, this pushes the bristles back out and into contact with the pipewall. Simultaneously, the leading axial drive cylinder is exhausted, this causes the leading brush unit to re-orientate. This completes the reversal cycle, Figure 4.13 details the cycle. At any time the in-bound drive can be halted and the above procedure reversed, this allows the vehicle to reverse once again and continue on its original out-bound journey. The vehicle has the advantage that it can change direction as often as desired and the mechanism is both robust and reliable. Reversal can occur on vertical or horizontal sections with the vehicles nose up or down. Reversal can also occur on bends.

4.10 BRISTLE/BRUSH UNIT DESIGN

The design of the brush unit is essentially the heart of this research. Chapter Five considers the theory behind bristle design whilst Chapter Six consolidates the theory with experimental results. As previously mentioned, bristle/brush design requires the integration of a number of important factors to obtain optimum tractive ability under different pipe conditions.

4.10.1 A CONSIDERATION OF BRISTLE MATERIAL

The choice of bristle material is governed by a number of factors. They are; the traction forces required, the pipeline environmental conditions and the susceptibility of the pipe or pipe contents to damage.

4.10.1.1 POLYMERIC BRISTLES

Two types of polymeric bristles were used originally, they were nylon 66 and nylon 612. Polymeric bristles suffer a number of disadvantages over steel bristles. The main disadvantage is that they can lose a large percentage of their stiffness when exposed to water. For example, nylon 6 can lose up to 80% of its stiffness. Other polymer materials, for example, polyester and polypropylene are less susceptible to this problem, however, they have a significantly lower Modulus of Elasticity compared to either nylon 6 or 612. As the stiffness of the bristles is critical to the performance of the vehicle, then, it can be clearly seen that polymeric bristles are generally an inappropriate choice in wet environments, for example, sewers. A secondary disadvantage of polymeric bristles is that they suffer from creep. That is, after a vehicle has spent some time within a pipeline the bristles will have acquired a semi-permanent deflection which reduces the forces at the bristle tip/pipe wall interface. Further, polymeric bristles do not perform well when used in polymeric pipelines, as the smooth surface of the pipeline does not allow the bristles to obtain a suitable foothold. However, polymeric bristles do have one fundamental advantage over steel bristles, that is, they cause little abrasive damage to their immediate environment. Steel bristles tend to "cut" into soft wall materials, for example, perspex.

4.10.1.2 STEEL BRISTLES

A steel bristle of comparable length and diameter to a plastic bristle is approximately 66 times stiffer *(E for steel is approximately 200 GPa, E for nylon is approximately 3 GPa)*. Thus, if high tractive capability is required steel bristles are a more appropriate choice of material. To date, the steel bristles used in the development of experimental vehicles have been pre-hardened and tempered to BS2803, chemical composition 095A65. Depending upon the application they may also be tinned. Consideration is currently being given to bristles that are fully hardened.

The Modulus of Elasticity *E*, of steel is such that steel bristles are able to support substantially higher axial reverse forces than polymeric bristles. Such loads supported by steel brush units have exceeded 1.25 Tonnes. These brush units can be used individually or in multiples. The main disadvantage of steel bristles is that they may abrade the internal surface of a pipe and/or damage cabling within the pipe.

For the majority of applications it can be assumed that steel bristles will not be affected by stress relaxation or creep. However, at higher ambient operating temperatures, or if the vehicle is operating within a pipeline for an extended period then due consideration should be given to this problem.

It is assumed that the steel bristles deflect within their elastic limit and that plastic deformation of the material does not occur. If the steel bristles are relatively short and consequently stiff, it is not practical to reverse the brush by winching or tow rope. For example, a vehicle using three steel bristled brushes and two axial drive cylinders, may require over 3 Tonnes to reverse if winched backwards. This figure is based upon the fact that a single X-ply brush, under dry conditions was able to support 1.25 Tonnes when subjected to a reverse force experiment. If bends are present the additional capstan effect of the umbilical and winching cable must also be taken into consideration. These forces can quickly escalate, thus the vehicle can quickly become irreversible. However, it should be noted that sewer vehicles use steel bristles and are reversed by tow rope/winch. This is possible because of their relatively long and flexible bristles.

4.10.2 BRISTLE LENGTH

For a brush to induce maximum tractive capability a specific bristle length must be used. This length will depend upon a number of factors, including; bristle material, Modulus of Elasticity *(E)*, second moment of area *(I)*, internal diameter of pipe and diameter of brush core.

If a relatively short bristle is used and the bristle is relatively steep with respect to the pipe wall, then, care must be taken to ensure premature flipping of the bristle does not occur under reverse "grip" conditions. If a bristle is short with respect to its diameter it will be relatively stiff. If this occurs and the bristle is subjected to high reverse loading in a pipe with a smooth wall finish then slippage or bristle "stutter" (stick-slip) results.

If a bristles length is long compared to its cross-sectional area the bristle risks premature collapse due to it flipping through, even with a relatively small axial force. However, a long bristled vehicle is extremely useful especially if the vehicle is required to negotiate an irregular passage, for example, a sewer. Unfortunately, a secondary disadvantage of long bristles is their inability to support the weight of the vehicles body. The result is the vehicle may "nose-dive", a phenomenon that can be countered by the use of specialised suspension bristles which are shorter in length and are not swept back, consequently they are stiffer.

4.10.3 BRISTLE PRE-SWEEP

The orientation of the bristles is important. Generally, the bristles are inserted normal to the core axis, that is, they extend radially outwards. However, they may also be pre-swept, that is, they are swept back with respect to the axis of the core. The angle of the pre-sweep can be varied, but it is generally less than 15 degrees. Pre-sweeping the bristles reduces the force required to load the brush, it also reduces the force required to move a brush forwards through the pipe.

However, pre-sweeping the bristles also reduces the force required to reverse a brush. Further, pre-sweeping the bristles can have a detrimental affect upon reversibility. Generally, a pre-swept bristled brush is not considered to be reversible.

4.10.4 X-PLY BRISTLE CONFIGURATION

The X-ply bristle mechanism is designed to counter the affect of the couple induced into the brush core under reverse axial loading. The bristles extend from the brush core, however, alternate rows of bristles are at opposing offset angles of +/-5 degrees. This bristle layout is repeated along the length of the brush core, the only criterion being that there must be an equal number of bristles with the negative inclination as there are with the positive inclination. Using this mechanism it is possible to increase the force the brush can withstand prior to reversal.

4.10.5 WEAR: EFFECT ON TRACTION

Wear can be thought of as the transfer of material due to the adhesion occurring between the points of real contact between two surfaces. Two surfaces that rub together result in the unintentional removal of material from one or both surfaces. Bolton (1989) details four different wear conditions;

1. "Where the junctions between the surfaces are weaker than the sliding materials, shearing occurs at the interface and there is little transfer of

metal (material) from one surface to the other, i.e. little wear occurs, for example, a tin-base alloy sliding on steel".

2. "Where the junctions are stronger than one of the metals but weaker than the other, shearing takes place a small distance within the softer material. Wear of the softer material thus occurs and eventually a film of softer material builds up on the harder surface, for example, a lead-base alloy sliding on steel".

3. "Where the junctions are stronger than both metals, shearing will take place mainly in the softer of the two metals but some fragments of the harder metal may be ploughed out, for example, copper sliding on steel".

4. "Where both the sliding surfaces are the same, the process of deformation and sliding causes the junction material to work [harder]. As a consequence of this shearing occurs within both the metals and considerable wear can occur".

If polymeric bristles are used in conjunction with steel or vitrified clay pipes, bristle wear escalates, the softer material is subject to layer shear with each forward stroke. The same occurs if steel bristles are deployed in polymeric pipes, except in this case it is the pipe which wears.

If bristles have been employed for numerous pipe runs, wear of the bristle tip occurs. A uni-directional vehicle produces a bristle with a single chisel edge and a bi-directional vehicle produces a bristle with a double chiselled edge, see Figure 4.14(a). Due to increased wear rates, this effect is more pronounced on the polymer bristled vehicles than the steel bristled vehicles.

It should be realised that in terms of the surface roughness (R_a value) of the pipe walls, the "pits" are very small indeed (μm), compared to the diameter of the bristle passing over them. New bristles have sharp edges when used in a pipe for the first time, this edge is beneficial to the bristle, enabling it to obtain a secure purchase. Over time the edge becomes worn, in part due to high local stresses at the bristle tip, these stresses increase as the bristle is subjected to reverse axial loading. When this occurs the bristle tip material shears. Further, wear also occurs as the bristle moves forward through the pipe. From the time the edge begins to wear up until the chisel edge has completely formed, it is considered that the bristle's ability to provide adequate "grip", when subjected to reverse axial forces, is lowered. However, once the initial wear process is complete, it is considered that when the bristle is subjected to reverse axial forces, the bristle is subjected to a slight rotation about the tip. This enables the chisel edge to be "tilted" into and engage with the pipe wall and it is thought that this process allows the bristle tip to obtain a suitable purchase.

Using the above explanation it can be suggested that a large number of smaller diameter bristles are better able to obtain a purchase than a small number of larger diameter bristles. This is thought to be due to the fact that the bristle tip rotation may be more pronounced with smaller diameter bristles, as their flexural stiffness is lower, subsequently, they are better able to engage with the pipe wall. Larger diameter bristles are less likely to experience such a pronounced rotation due to their increased stiffness, as such, they are more likely to slip, see Figure 4.14(b).

4.10.6 WEAR: SEM IMAGES

To identify the magnitude of bristle wear Scanning Electron Microscope (SEM) images were taken during different stages of a bristle's "life". Images of both nylon 612 and steel were observed and recorded.

The nylon 612 is of the *Penter* type design. *"Penter"* is a nylon extruded to form a *"star"* cross section, with five points. Figure 4.15, 45x magnification (tilt 0 degrees), shows a bristle in the "as cut" state. The top right hand "star" shows evidence of plastic deformation due to the action of the blade, whilst the bottom "star" shows evidence of axial splitting. The point at which the blades met can be seen one third of the way across the face. Of note is the fact that, in general, the face is cut at 90 degrees and no external wear is evident.

Figure 4.16, 45x magnification (tilt 89.5 degrees), shows a side profile of a nylon 612 bristle. This bristle was used on a uni-directional vehicle. Significant wear can be seen on the lower face of the bristle tip. The small area of additional wear at the top of the bristle tip is caused by withdrawing the vehicle and occasionally driving it backwards. The detached fibres or "chains" of material have sheared from the bristle tip during normal vehicle operation. This bristle had completed approximately 1km of service in steel pipelines prior to being photographed.

Figure 4.17, 45x magnification (tilt 89.5 degrees), shows an end-on view of a penter nylon 612 bristle. This bristle had also completed 1km of uni-directional service in steel pipelines. The shearing effect of the polymer against the steel is evident at the bottom left hand corner of the image. Here, there has been an accumulation of fibre. The overall chamfer is clearly evident.

Figure 4.18, 300x magnification (tilt 93.5 degrees), shows a steel bristle, 0.193mm diameter (36SWG). The steel bristle is in the "as cut" state. Slight plastic deformation due to cutting is present at the bottom left and top right hand sides of the image.

Figure 4.19, 300x magnification (tilt 93.5 degree), shows a steel bristle that has completed approximately 500m of uni-directional service in steel pipelines. The slight curvature at the top of the figure and the flat area at the base are caused by plastic deformation due to the cutting process. The area to the right of the image is wear caused by the bristle traversing the pipe wall.

4.11 CONCLUSION

This research is essentially concerned with the brush/bristle design and the tractive performance derived from a vehicle incorporating bristle drive technology. This chapter has been written to indicate to the reader some of the design steps which were followed in order to arrive at the current technology.

What should have become apparent from this chapter is the degree of dependency that each of the design factors has on one another. The development of this vehicle has been based on both experiment and calculation. Difficulties have occurred due to a lack of understanding of bristle behaviour when they are used for traction. As such, a degree of "trial and error" was present in the early days. The vehicles can now be designed with considerable confidence, such that, the performance that it will deliver is suitable for the application for which they were designed.

A photographic history of the development of the vehicle is located in Appendix A. The photograph in Figure A.20 shows the initial vehicle, that is, a pneumatic cylinder and two lavatory brushes. The photograph in Figure A.21 shows the second vehicle manufactured, this vehicle had specifically manufactured brushes. The photograph in Figure A.22 shows the fore-runner to the sewer vehicles. This vehicle is 75mm in diameter and was designed to negotiate a badly deformed perspex pipe test track. The photograph in Figure A.23 shows a "sprinter" vehicle. This vehicle was specifically designed to pull draw cables through pipelines quickly. The vehicle does not use solenoid valves to switch the double acting pneumatic cylinder, instead the switching is performed by on-board reed switches. The photograph in Figure A.24 shows a vehicle used to ascend the external surface of a tubular structure, for example, a tensile cable. The photograph in Figure A.25 shows an electric powered vehicle, this vehicle was built to prove the concept of electric power, to date, no further development has been undertaken. The photograph in figure A.26 shows an "experimental vehicle" built to prove the principle of bi-directionality. Each of the brushes was capable of crudely expanding and contracting, however, the principle worked and this "vehicle" became the fore-runner of the more complex expanding brush pod designs. The photograph in Figure A.27 shows the parallel plane vehicle. This vehicle was designed to negotiate and inspect between the twin skins of an oil tanker [ship]. The photograph in Figure A.28 is the PII high capacity vehicle. As previously explained this vehicle has a towing capacity of approximately 1 metric Tonne. This is an 8" diameter vehicle. The photograph in Figure A.29 shows the "family" of sewer vehicles. In the foreground is the 225mm (9") diameter vehicle. In the centre is the 300mm (12") diameter vehicle and at the rear of the photograph is the 375mm (15") diameter vehicle. The photograph in Figure A.30 shows a vehicle designed to harness energy from fluid flow, this in turn allows the vehicle to crawl, using the bristle traction system, against the fluid flow. The photograph in Figure A.31 shows a 150mm (6") diameter bi-directional vehicle which utilises the expanding bristle pods. Finally, the photograph in Figure A.32 shows a partly dissembled expanding brush core. The radial pistons and bristle plates can be clearly seen.

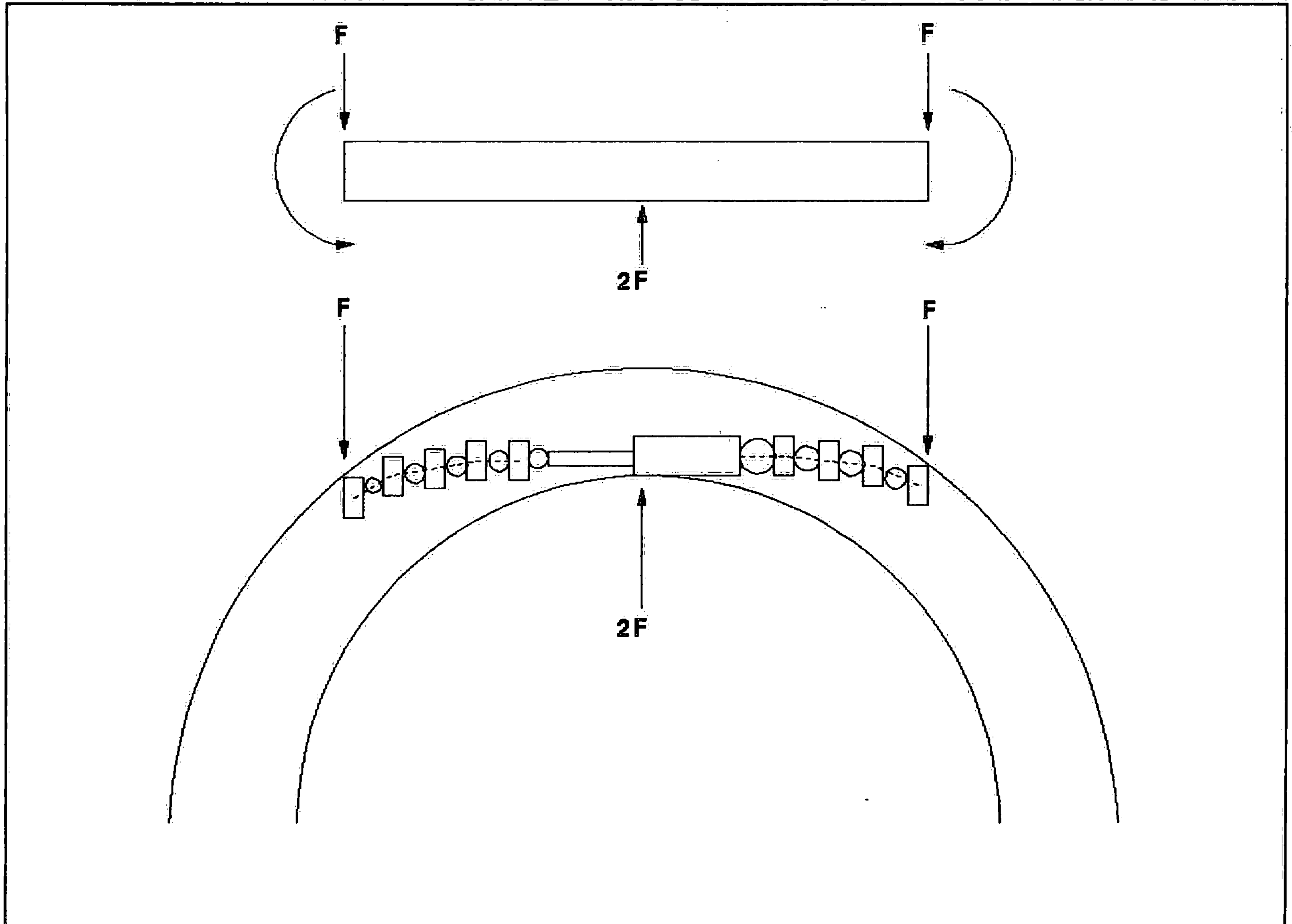


FIGURE 4.1 AN EXAMPLE OF A "TENSILE CABLE" JOINT

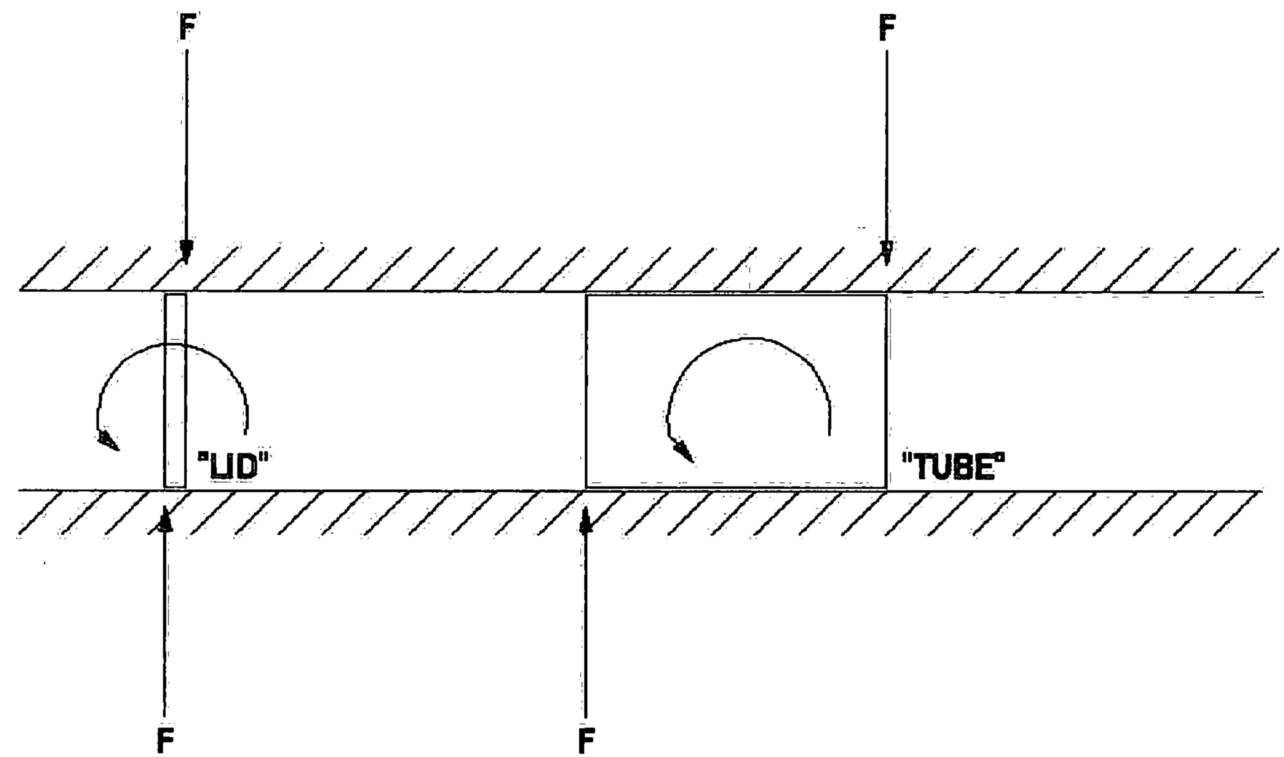


FIGURE 4.2 INSTABILITY: A FUNCTION OF BRUSH CORE DIAMETER TO LENGTH

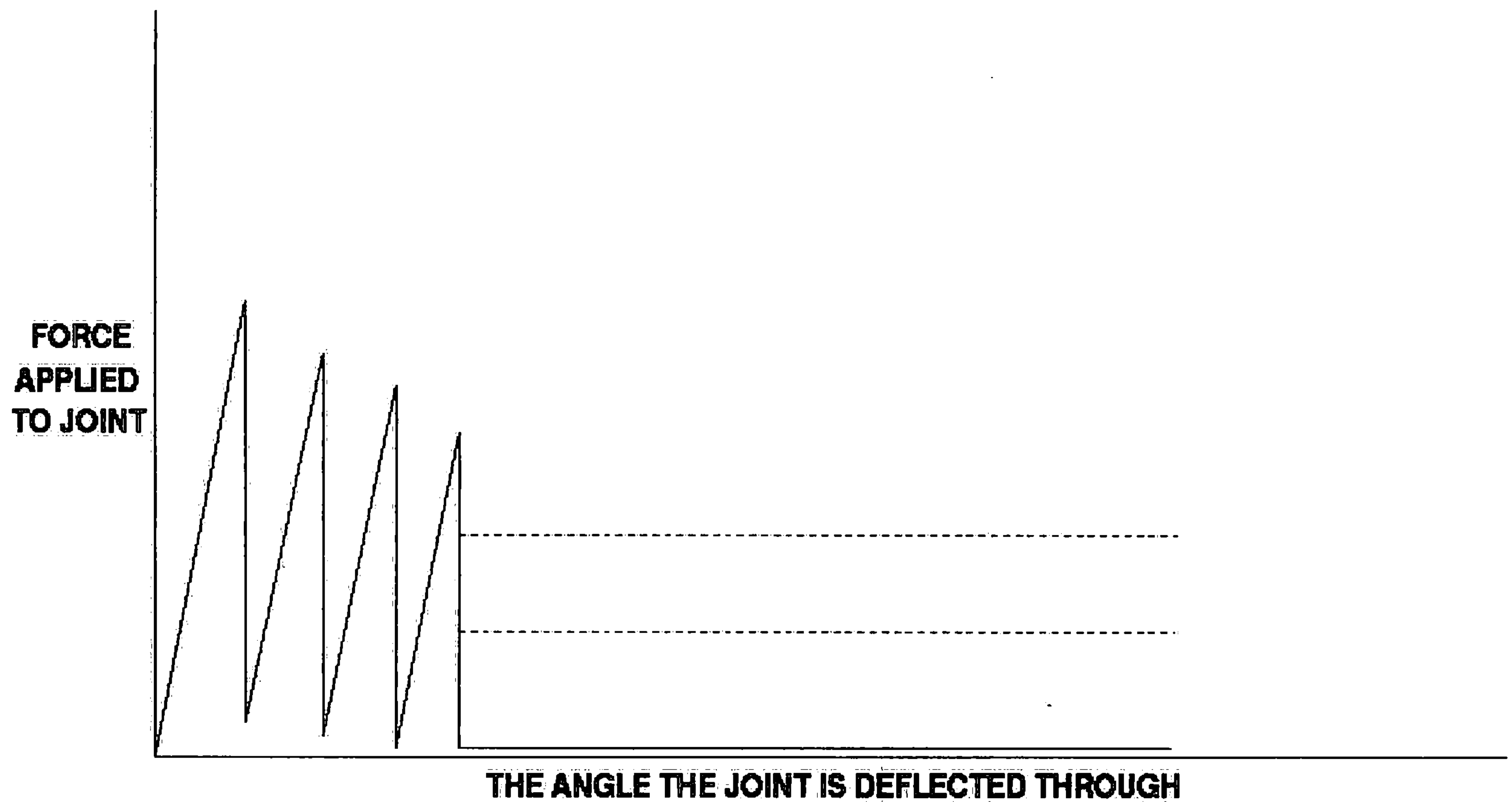


FIGURE 4.3 A FORCE/DEFLECTION CURVE FOR A RATCHET TYPE JOINT

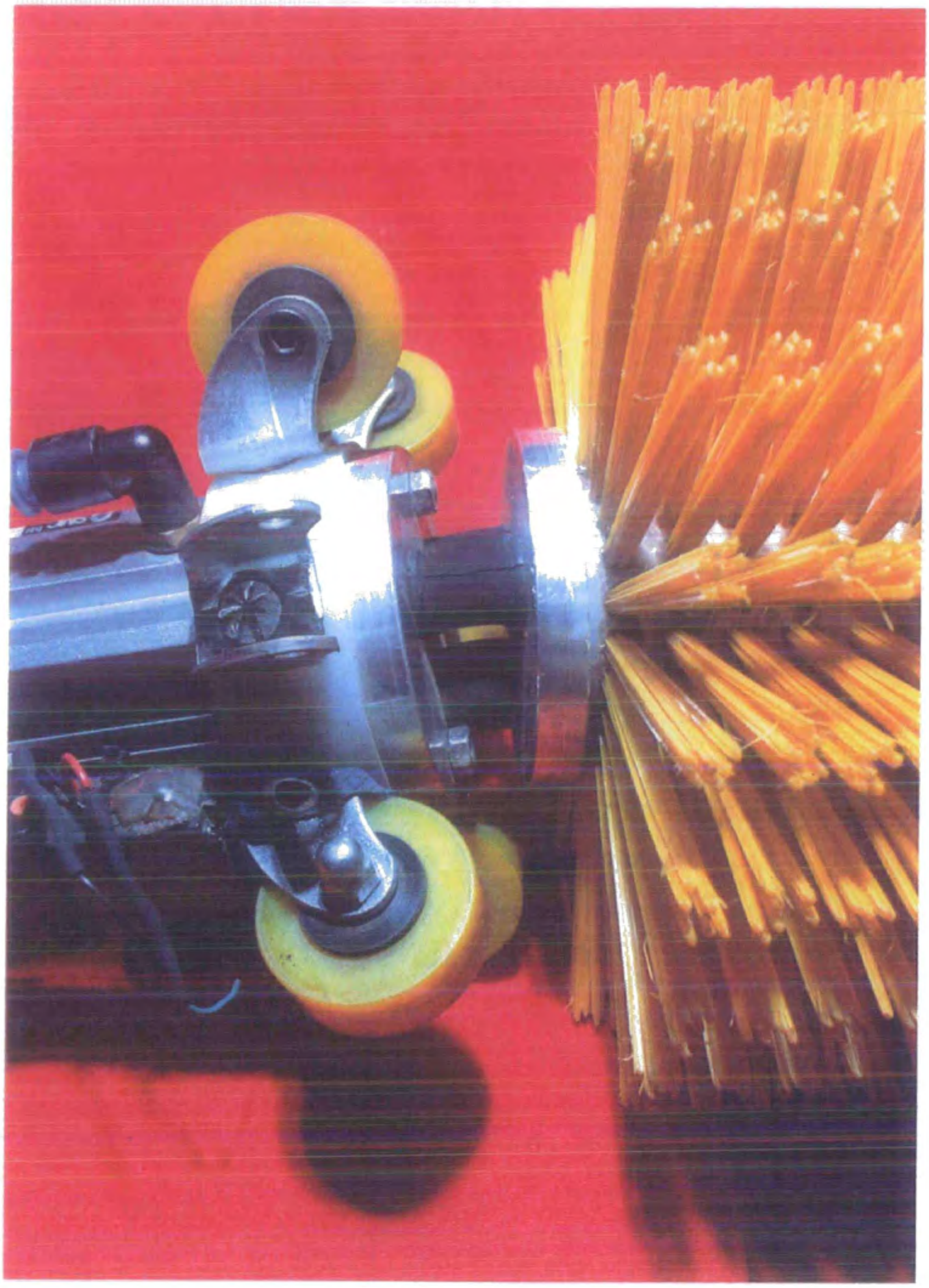


FIGURE 4.4 An example of a rubber bobbin joint fitted to a polymeric bristled vehicle.

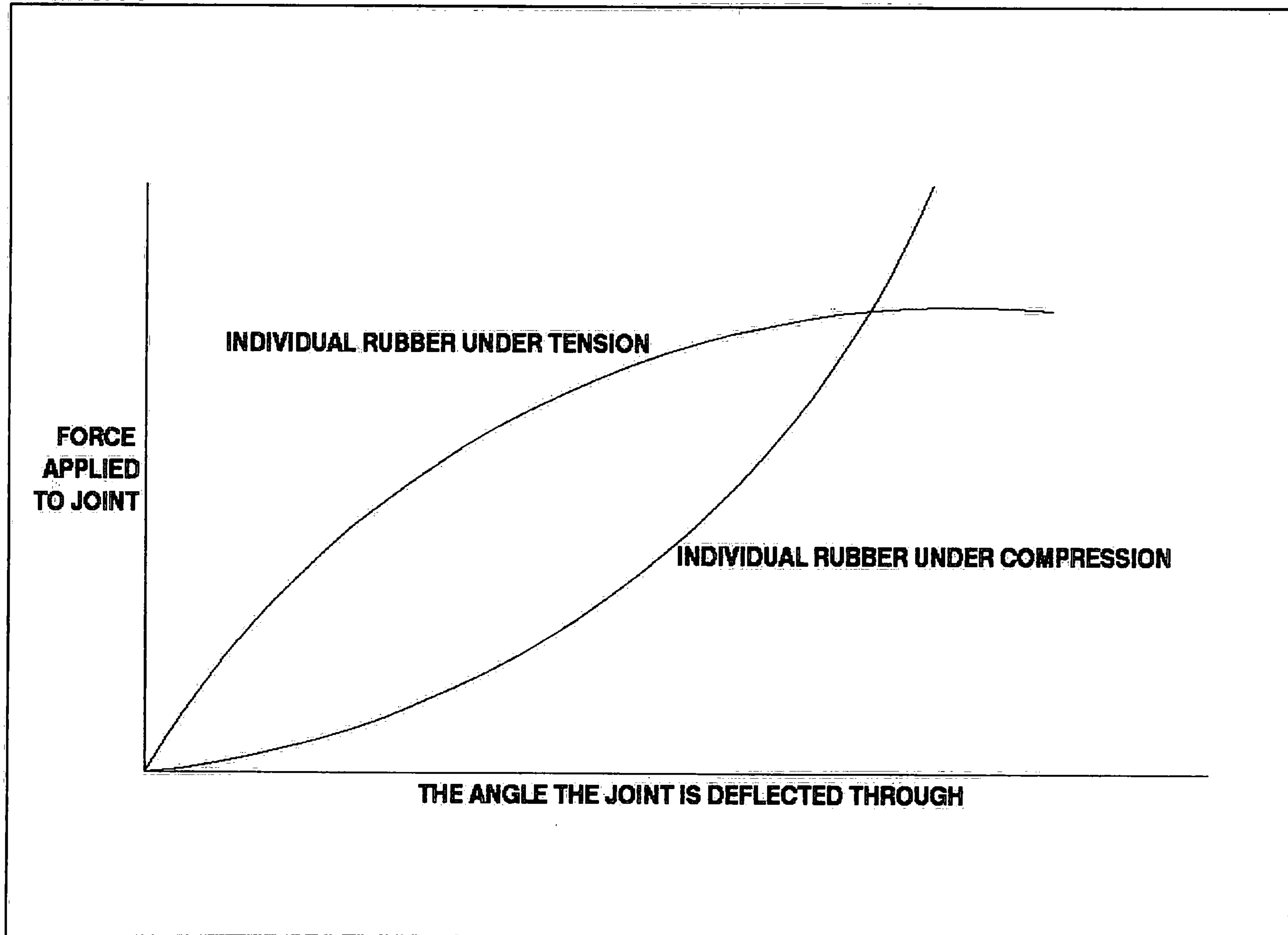


FIGURE 4.5 FORCE/DEFLECTION CURVE FOR A RUBBER OR ROSE BEARING JOINT

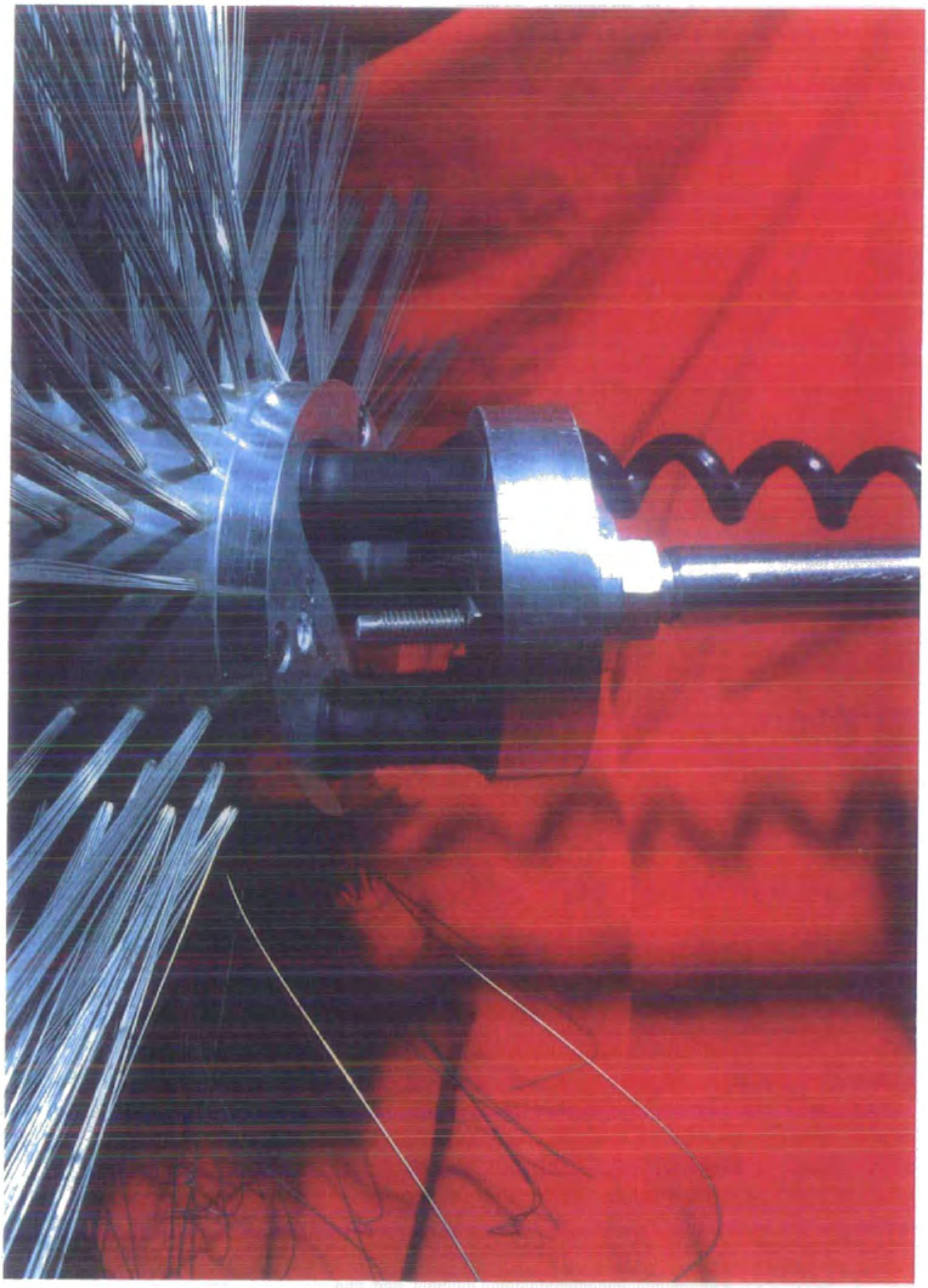


FIGURE 4.6 An example of a rubber bobbin joint incorporating stabiliser pegs.

The joint is fitted to a sewer vehicle.

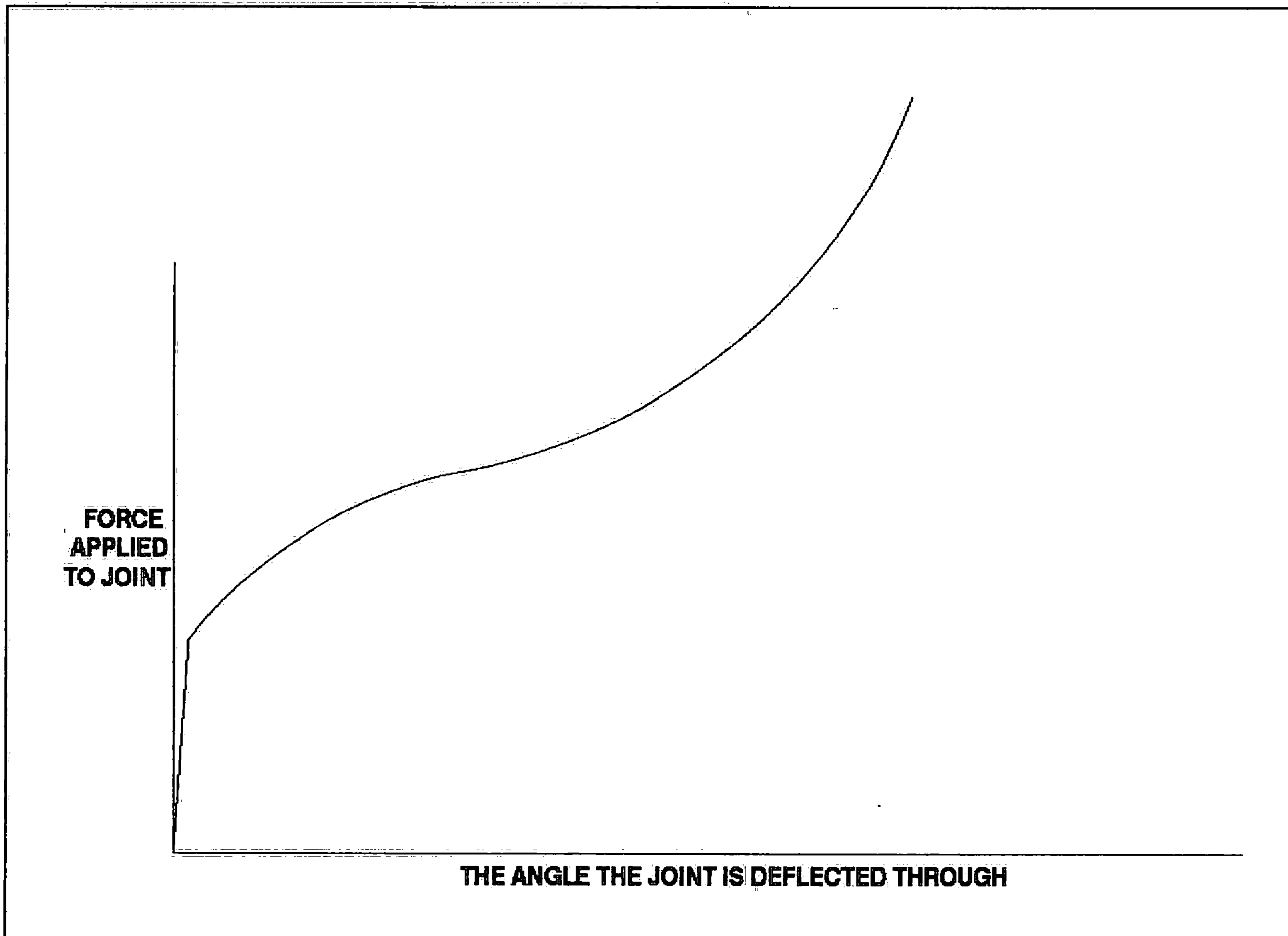


FIGURE 4.7 A FORCE/DEFLECTION CURVE FOR A RUBBER JOINT INCORPORATING STABILISERS

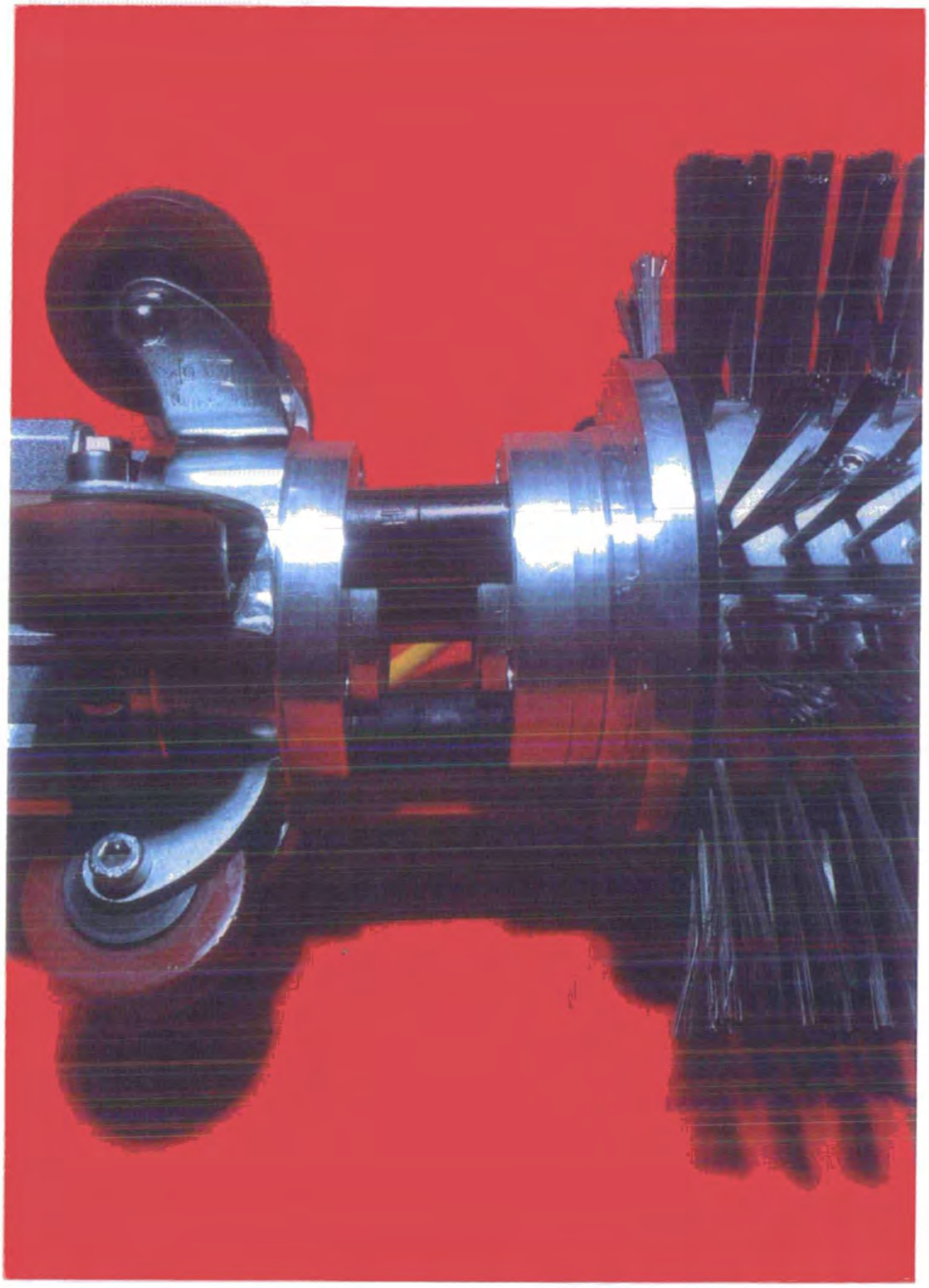


FIGURE 4.8 An example of a Rose bearing joint fitted to a 6" diameter bi-directional vehicle.

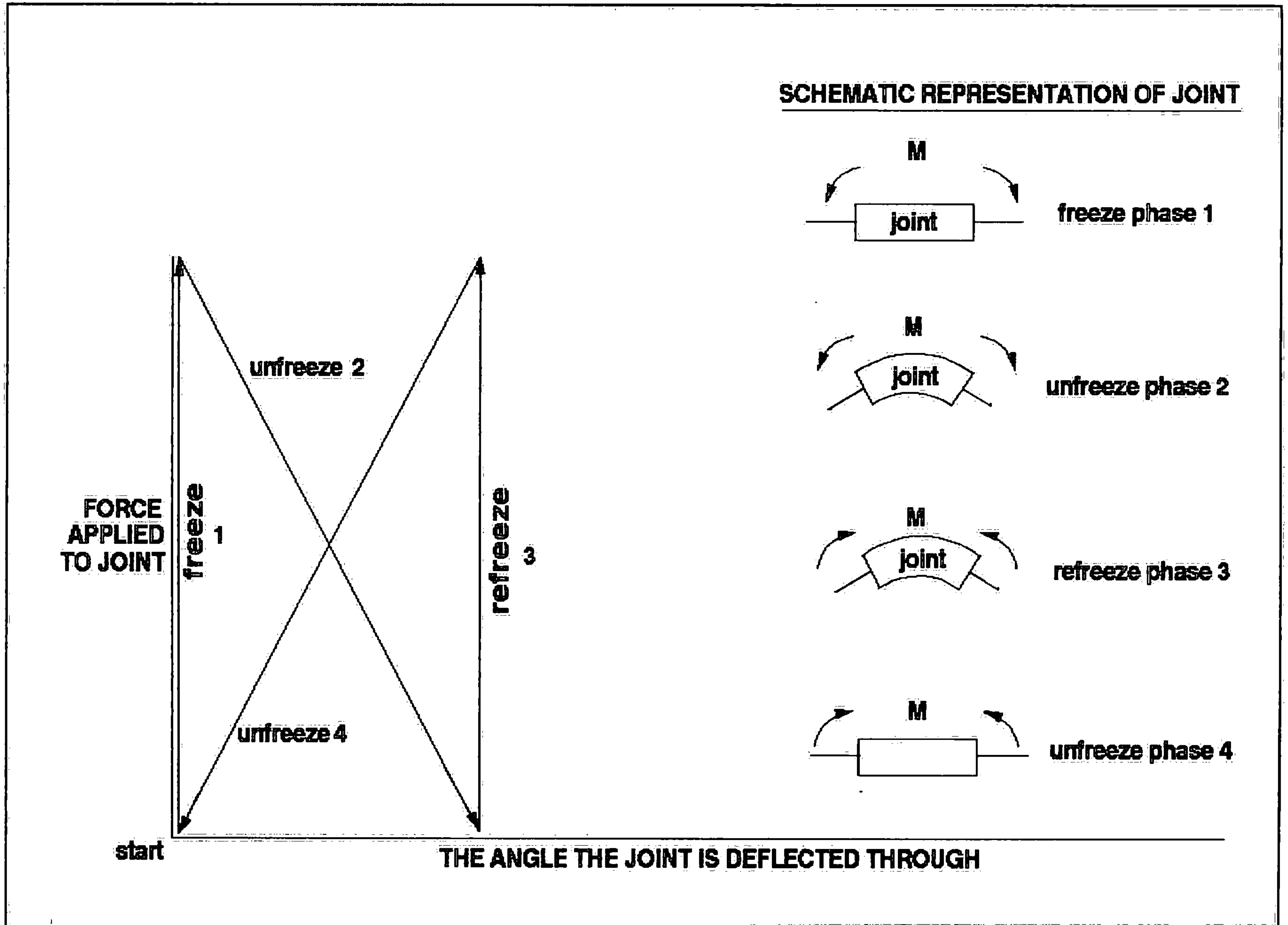


FIGURE 4.9 A FORCE/DEFLECTION CURVE FOR A RHEOLOGICAL FLUID JOINT

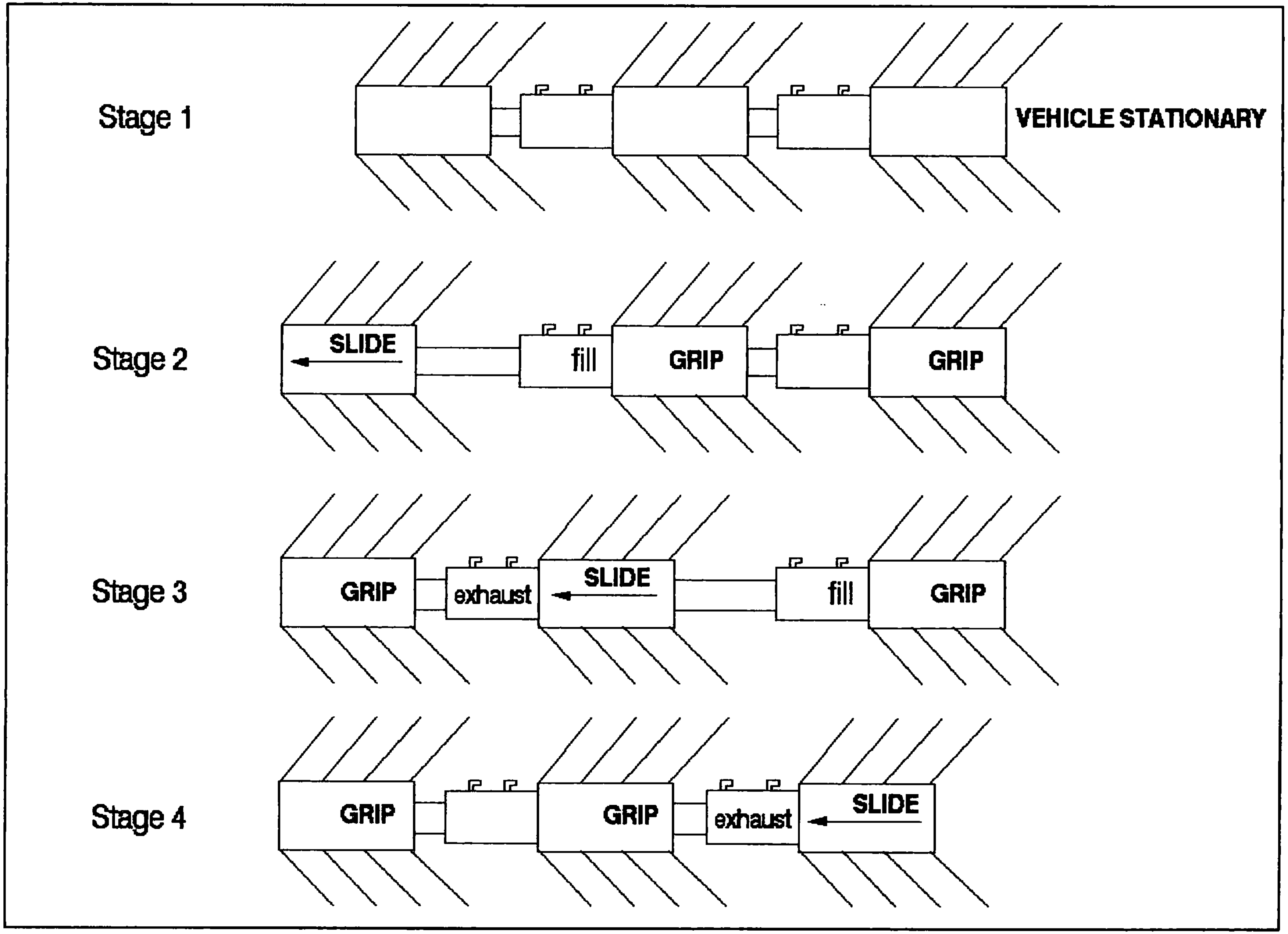
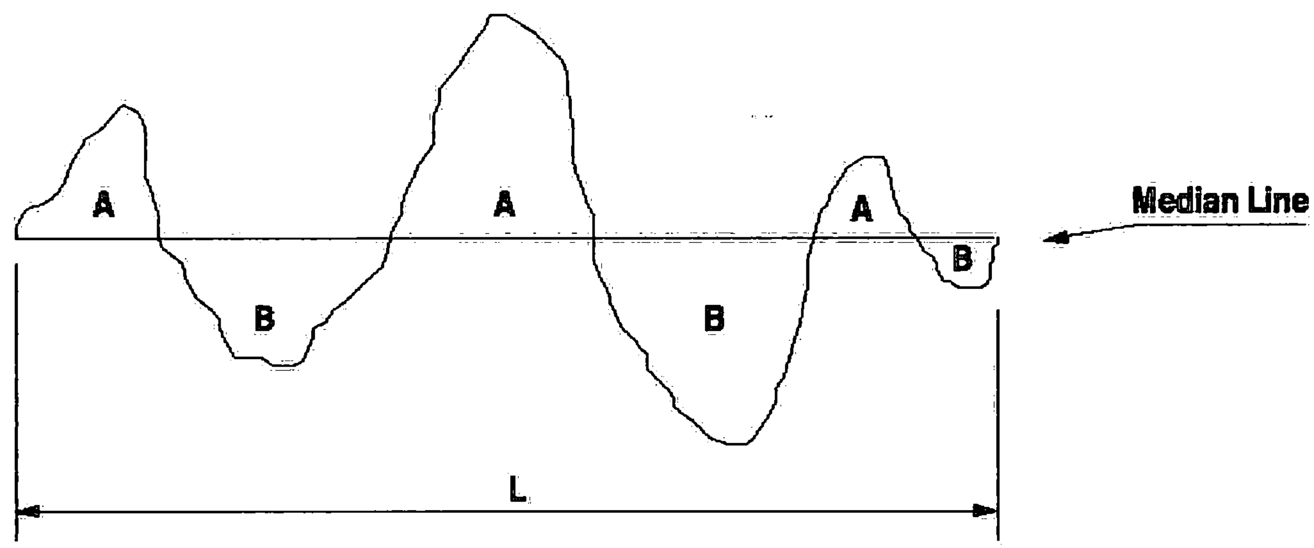


FIGURE 4.10 THE FORWARD DRIVE SEQUENCE FOR A 3 BRUSH/2 CYLINDER VEHICLE



Ra is the average distance (In Microns) between the Median Line and the peaks and troughs of the surface profile

FIGURE 4.11 **THE PEAKS AND TROUGHS OF SURFACE ROUGHNESS**

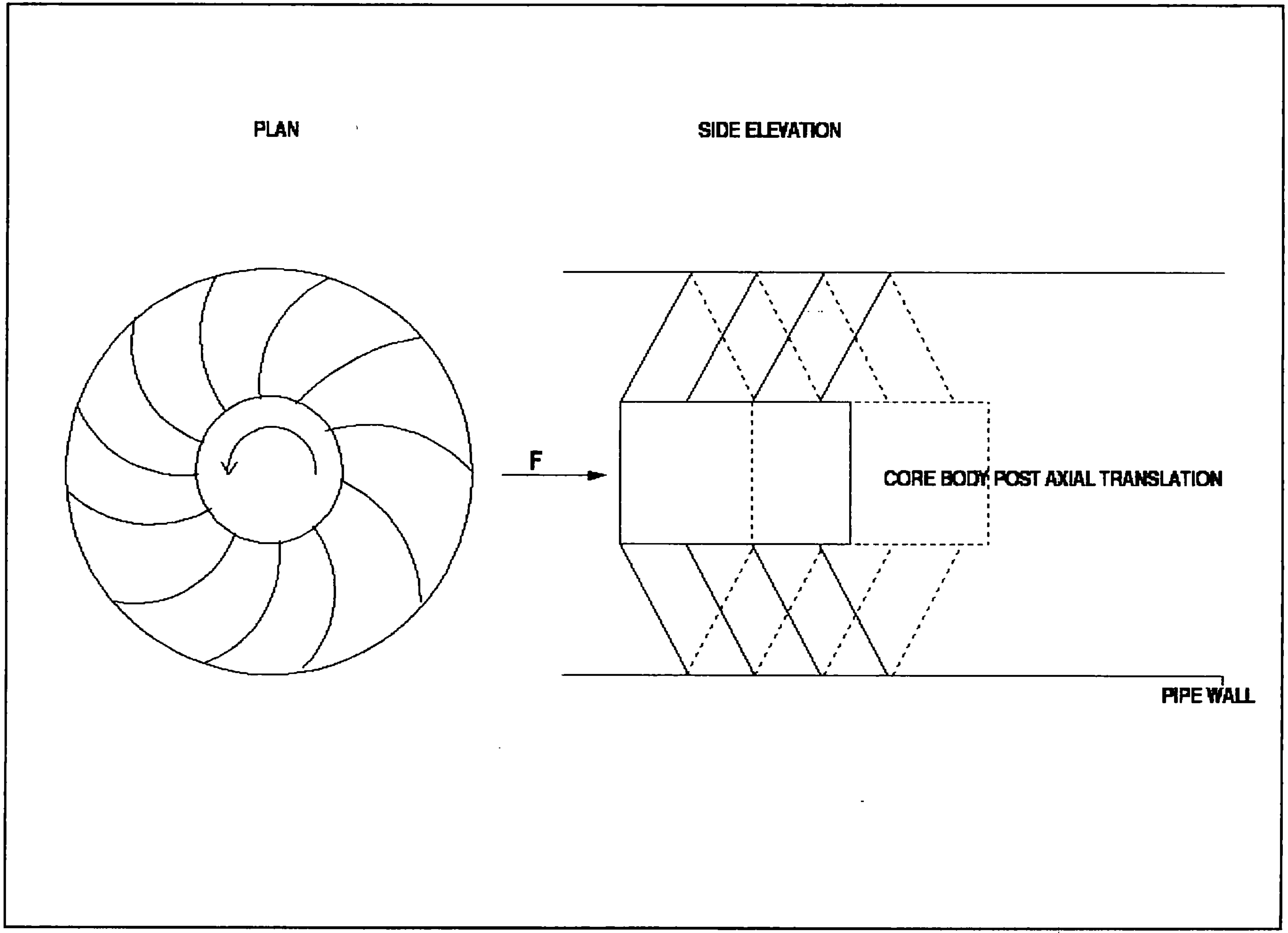
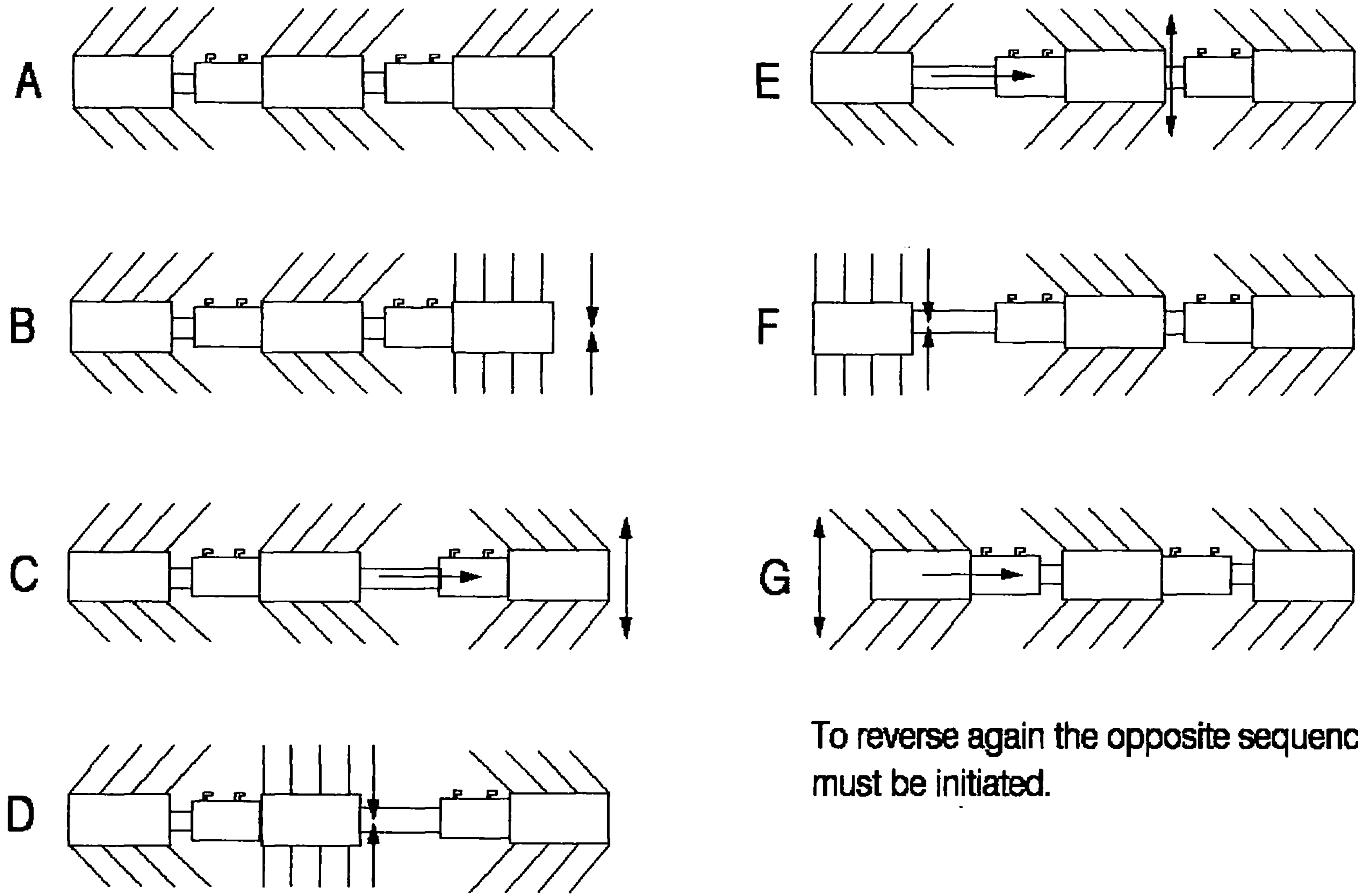


FIGURE 4.12 AN EXAMPLE OF A BRUSH ROTATING WHEN SUBJECTED TO A REVERSE FORCE EXPERIMENT



To reverse again the opposite sequence must be initiated.

FIGURE 4.13 AN EXAMPLE OF THE REVERSE SEQUENCE FOR A 3 BRUSH/2 CYLINDER VEHICLE

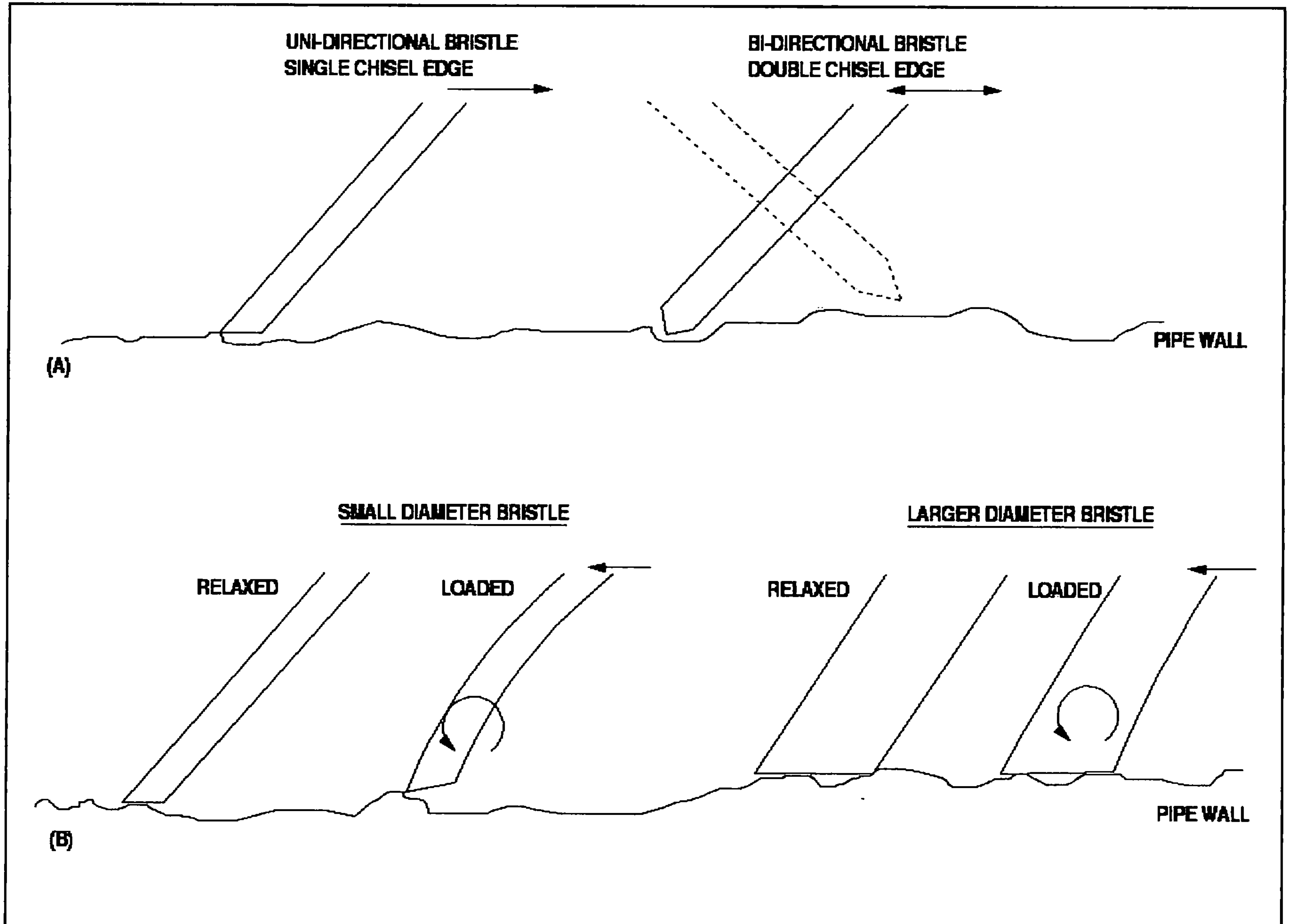


FIGURE 4.14 AN EXAMPLE OF BRISTLE TIP WEAR AND BRISTLE DIAMETER TO GRIP EFFECT

FIGURE 4.15 Nylon 612 bristle “as cut”, magnification 45X.

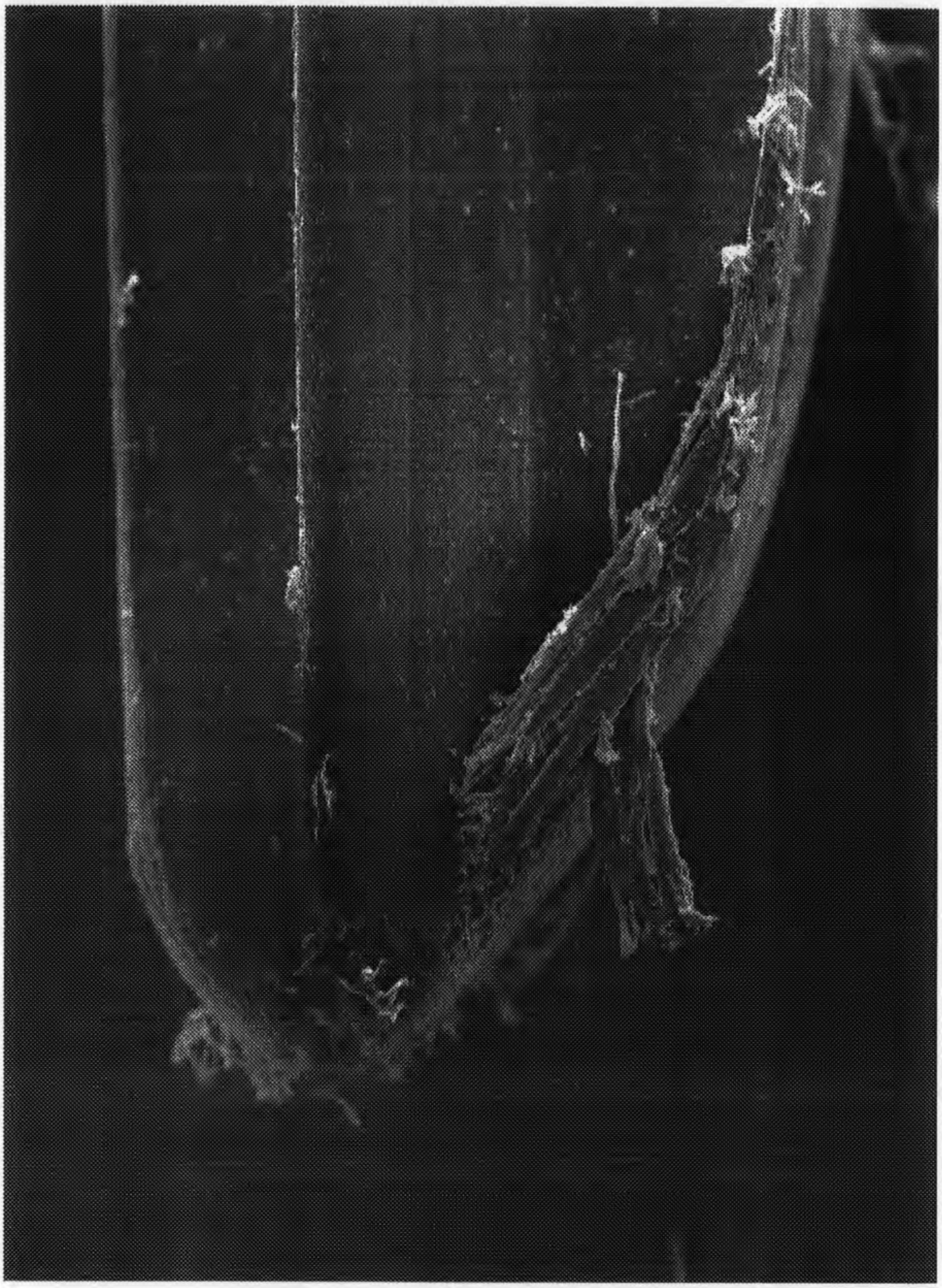


FIGURE 4.16 Nylon 612 bristle after completing 1Km of uni-directional travel, magnification 45X (side view).

FIGURE 4.17 Nylon 612 bristle after completing 1Km of uni-directional travel, magnification 45X ("foot" view).

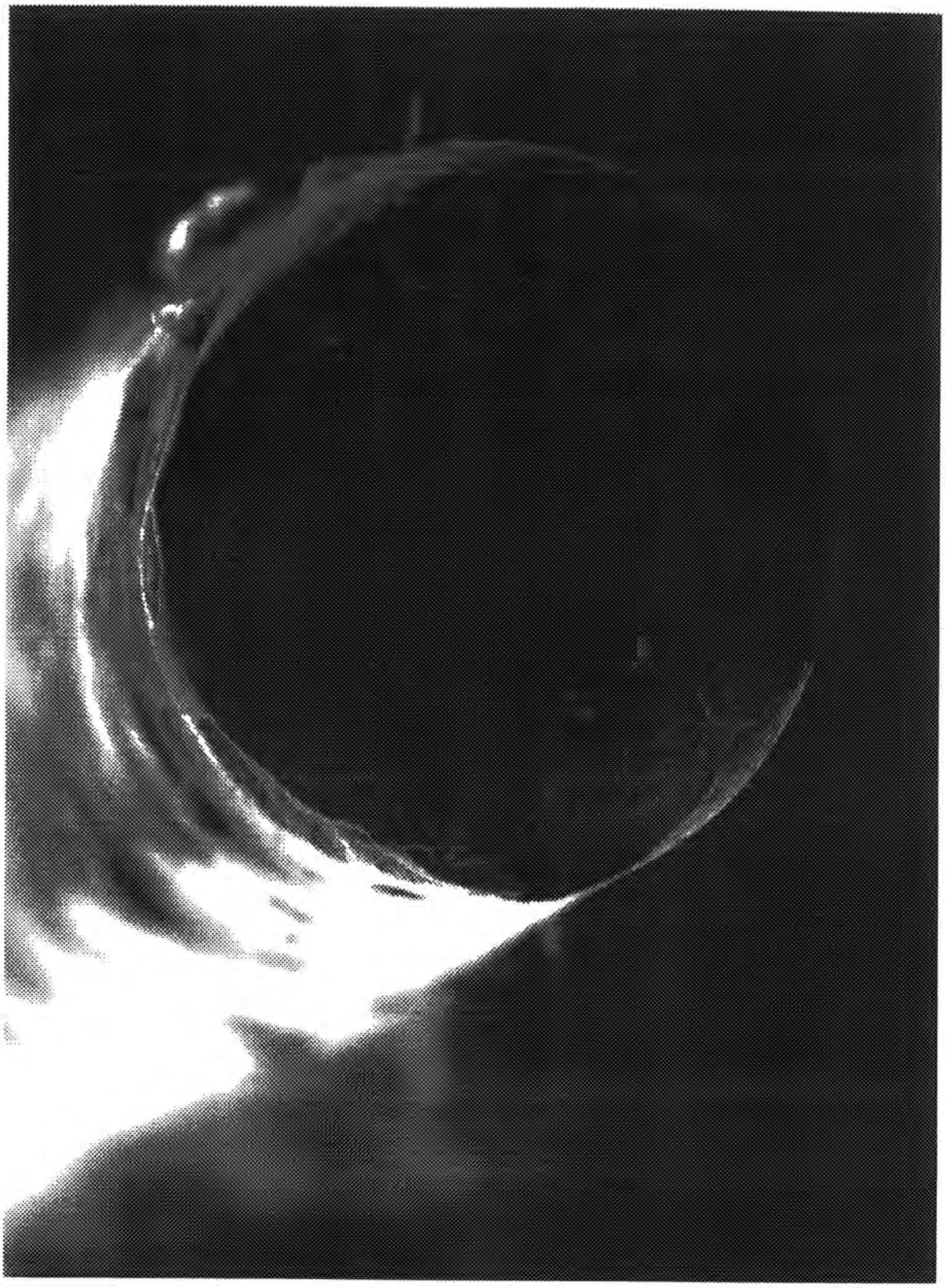


FIGURE 4.18 0.2mm diameter steel bristle, as cut, 300X magnification (end view).

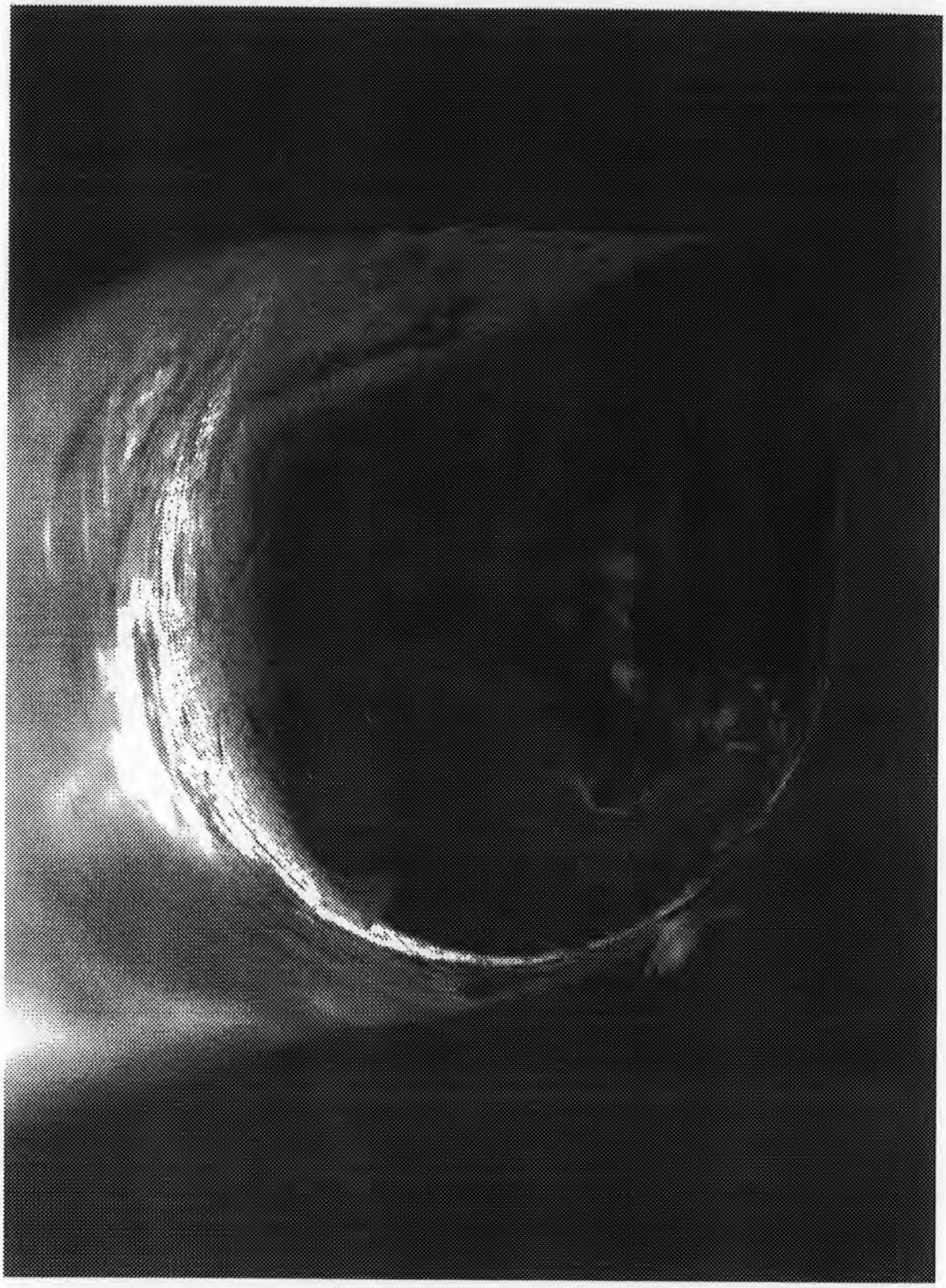


FIGURE 4.19 0.2mm diameter steel bristle after completing 500m of uni-directional travel, 300X magnification (end view).

CHAPTER 5

BRISTLE THEORY

5.1 INTRODUCTION

This chapter relates to the mechanics of individual bristles and complete brushes. To date, brushes have not been considered or used as "engineering" components. Subsequently, very little analysis is available on the static or dynamic behaviour of such items. As such, it is believed that the use of bristles for traction and drive is unique, hence, the current patent applications. The principle, as previously described, relies upon the fact that a brush with a head diameter larger than the pipe into which it is inserted will require less force to move forward than to move backwards. The highest ratio possible between forward sliding and reverse gripping is desirable. At the bristle tip/pipe wall interface, neglecting bending, the bristle normal force (F_N) and the friction force (μF_N) are present, as illustrated by Figure 5.1. To design a vehicle to travel through a specific internal pipeline diameter and wall condition requires a basic understanding of the bristle at an engineering level, specifically, the bristle tip interaction with the pipe wall. A number of factors are vital to the overall design including; bristle material, bristle free length, bristle diameter, the method of attaching the bristle to the brush core, the angle the bristle forms with the pipe wall, the R_a value of the pipe wall and any "lubrication" present. In engineering terms, once the bristle is inserted into the pipe it can be considered as a simple cantilever beam experiencing a concentrated end load. Most cantilevers experience loads exerted at a given point along their length, as such, they are subjected to a bending moment. A bristle is similar, except that the necessary deflection, which results from it being inserted into the pipe, acts upon the bristle stiffness to exert a force against the pipe wall.

If the bristle is subjected to an axial compressive force, for example, in the rearward or support mode, then it can be considered as a strut. Stephens (1970) describes a strut as *"a compression member which is long in comparison to its cross-sectional dimensions. Such a member will fail due to buckling before the direct compressive stress reaches the yield point"*. However, as the bristle is already curved, prediction of collapse using "textbook" buckling theory is very difficult, for any cases other than straight or relatively straight bristles.

This chapter starts by considering the force required for a bristle to slide forwards through a pipe. Additionally, it attempts to predict the onset of bristle failure through buckling when the bristle is subjected to axial loading whilst in "grip" mode. During the early stages of the vehicles development, forward to reverse ratios of between 1:3 and 1:4 were regularly observed. As the brush bristle mechanism became more understood ratios of 1:9 were achieved. It is currently considered that with "intelligent" traction control and a cross-ply (X-ply) bristle configuration, the "slip to grip" ratio may be further increased to 1:12.

5.2 FORWARD BRISTLE THEORY

In order for the traction mechanism to perform satisfactorily the internal diameter of the pipe into which the brush unit/vehicle is to be inserted must be less than the diameter of the brush head in its unloaded state. Upon insertion, the bristles are swept back and the bristle tips form a specific angle with the pipe wall. To achieve a basic understanding of the principle it is necessary to understand the forces being exerted at this interface. That is, for a given angle formed between bristle tip and the pipe wall and for known material/lubrication conditions, what force would be required for the bristle/brush to overcome the friction. As previously mentioned, as a first attempt at bristle analysis, each bristle can be considered as a simple uniform cantilever experiencing a concentrated end load, with flexural stiffness EI and length L. The cantilever is effectively "built-in" at one end (the brush core) and experiences a concentrated load W at the other end. It is possible to calculate the deflection of the cantilever (V_L) for a given load. Simple cantilever/beam theory is adequate to calculate the brush loading force and forward "slip" force, considering the cantilever below:

With reference to Figure 5.2 the bending moment at a distance z from the built-in end is:

$$M = -W(L-z)$$

It is generally considered that a sagging bending moment is positive, where a length of the beam is subjected to sagging bending moments the value of (dv/dz) along the length diminishes as z increases; hence, a sagging moment implies that the curvature is negative. Then;

$$EI\frac{d^2v}{dz^2} = -M$$

where M is the *sagging* bending moment, from the above equation

$$EI\frac{d^2v}{dz^2} = W(L - z)$$

then

$$EI\frac{dv}{dz} = W(Lz - \tfrac{1}{2}z^2) + A \qquad (5.2.1)$$

At this point it would be useful to calculate the angle the bristle tip forms with the pipe wall. This can be done by solving for A in equation 5.2.1 by use of the end conditions at the core, that is, $\frac{dv}{dz} = 0$, at Z=0, i.e. A=0. At Z=L the angle of the bristle at the pipe wall is given by:

$$\theta = \frac{W(L^2 - \frac{1}{2}L^2)}{EI}$$

thus

$$\theta = \frac{WL^2}{2EI}$$

following on from equation 5.2.1

$$EIv = W(\tfrac{1}{2}Lz^2 - \tfrac{1}{6}z^3) + B$$

Furthermore, at $z = 0$ there is also no deflection, so that B = 0. Then;

$$EIv = \frac{Wz^2}{6}.(3L - z)$$

At the free end, $z = L$,

$$V_L = \frac{WL^2}{3EI}$$

thus,

$$W = 3EI.\left(\frac{V_L}{L^3}\right) \qquad (5.2.2)$$

The flexural stiffness of the member EI must also be considered

Prior to finding W it is necessary to determine V_L. The reader will note that for the purpose of these calculations, the bristle, when inserted into the pipe will form an arc, in reality, the bristle shape is considerably more complex than this. Thus, referring to Figure 5.3, it can be seen that;

$$r_L \, \alpha = L \tag{5.2.3}$$

and

$$r^2 + [r_L - V_L]^2 = r_L^{\,2} \tag{5.2.4}$$

and

$$r^2 + V_L^{\,2} = L_c^{\,2} \tag{5.2.5}$$

Using 5.2.4

$$r^2 + [r_L - V_L]^2 = r_L^{\,2}$$

thus

$$r^2 + r_L^{\,2} - 2r_L V_L + V_L^{\,2} = r_L^{\,2}$$

thus

$$r^2 = 2r_L V_L - V_L^{\,2}$$

thus

$$r^2 + V_L^{\,2} = 2r_L V_L$$

and

$$\frac{r^2 + V_L{}^2}{2V_L} = r_L \qquad (5.2.6)$$

By Cosine rule;

$$L_c{}^2 = r_L{}^2 + r_L{}^2 - 2r_L{}^2 \cos\alpha$$

thus

$$L_c{}^2 = 2r_L{}^2(1 - \cos\alpha)$$

Also

$$L_c{}^2 = V_L{}^2 + r^2 = 2r_L{}^2(1 - \cos\alpha)$$

thus

$$V_L{}^2 = 2r_L{}^2(1 - \cos\alpha) - r^2 \qquad (5.2.7)$$

Substituting equation 5.2.6 into 5.2.7;

$$V_L{}^2 = 2\left(\frac{r^2 + V_L{}^2}{2V_L}\right)^2 (1 - \cos\alpha) - r^2 \qquad (5.2.8)$$

Substituting equation 5.2.3 into 5.2.8;

$$V_L{}^2 = 2\left(\frac{r^2 + V_L{}^2}{2V_L}\right)^2 \left(1 - \cos\frac{L}{r_L}\right) - r^2 \qquad (5.2.9)$$

thus from equation 5.2.6;

$$V_L^2 = 2\left(\frac{r^2 + V_L^2}{2V_L}\right)^2 \left(1 - \cos\frac{L}{\left(\frac{r^2 + V_L^2}{2V_L}\right)}\right) - r^2 \qquad (5.2.10)$$

Rearranging;

$$\frac{2}{V_L^2}\left(\frac{r^2 + V_L^2}{2V_L}\right)^2 \left(1 - \cos\frac{L}{\left(\frac{r^2 + V_L^2}{2V_L}\right)}\right) - \frac{r^2}{V_L^2} = 1 \qquad (5.2.11)$$

Using equation 5.2.11 it is possible to solve for V_L numerically and W can now be calculated using equation 5.2.2. W can be considered as axial to the pipe and it is necessary to calculate the normal force F_N. Thus, using the principle of virtual work, it is found that;

$$W.V_L = F_N.(L - r)$$

rearranging

$$\frac{W.V_L}{(L - r)} = F_N$$

and

$$F_{N}.\mu = \frac{\mu W V_L}{L - r} \qquad (5.2.12)$$

(where F_N μ is the friction drag on the bristle)

To obtain $F_N \mu$, the frictional drag, it is now a simple matter of calculating W and V_L.

5.2.1 SPREADSHEET DESIGN

As the forward force calculations were required regularly a spreadsheet was written. The spreadsheet was based upon the above "simple" beam deflection theory. Figure 5.4 shows an example of the spreadsheet. The variables include; brush core diameter, total brush head diameter, Young's Modulus (Modulus of Elasticity) of bristle material, internal diameter of pipe, diameter of bristle, number of bristle rows per core, number of bristle holes per circumference, number of bristles per hole and number of bristle cores per side of cylinder. The above variables are entered into the top left hand fields by the user. The spreadsheet calculates some fundamental forces and dimensions. One of the most important results is the "radial force" F_N exerted by the bristles against the pipe wall. This data can then be multiplied by μ (coefficient of friction) to determine the force required to move the brush unit through the pipe, thus, overcoming the friction conditions. Knowing this force allows the designer to specify the size of the pneumatic or other fluid power cylinder to be used. An additional section of the spreadsheet calculates the cylinder thrust and pull for a given piston diameter, piston rod diameter and pressure. A cylinder must be able to deliver sufficient thrust and pull to ensure that the friction between the bristles and pipe wall can be overcome. If a long umbilical and/or heavy payload is to be transported, then, additional cylinder pull must be made available. All vehicles, irrespective of number of drive cylinders or brush units require the "last" drive cylinder in the sequence to act in pull mode, as it draws up the final brush unit, umbilical and payload.

As mentioned above, Figure 5.4 shows an example of the spreadsheet. This example represents a brush actually built, tested and deployed. The technical data for the brush has been entered into the spreadsheet and the total "Normal Force" has been calculated. The forward force obtained for this brush during testing was 753.4N (76.8KgF). This figure was then entered into the "Test Force Recorded" field which calculated a μ value of 0.327 which is considered acceptable for steel bristles traversing a rusty, pitted, steel pipe. The spreadsheet also calculates the angle between the bristle and brush core body after the bristle/brush has been inserted into the pipe. Further, it calculates the angle between the bristle tip and the pipe wall.

5.3 REVERSE BRISTLE MODEL

It is important to know the force required to push a brush forwards through the pipe, thus, overcoming friction. This allows the thrust and pull of the axial drive cylinder to be calculated. If the thrust and pull of the cylinder is known it is possible to calculate the total force available for moving a "payload". The axial drive cylinder in pull mode must be capable of dragging the trailing brush, umbilical and payload. Figure 5.5 illustrates the principle. The designer must then satisfy him/herself that the leading and/or trailing brushes are able to support all the forces being "towed". This ensures the vehicle is able to continue moving forwards. To summarise, it can be said that; Available Cylinder Pull $\geq F_F + F_U + F_P$. Failure to ensure that a brush or brushes are able to support the load will result in the bristles buckling and/or flipping through. This will result in the vehicle effectively reversing in the pipe. Hence, it is important to be able to determine the capacity of a brush in support or "grip" mode.

A model has been developed which provides a method of predicting the forward to reverse, or "slip to grip", ratio, for given frictional conditions. This model is deemed satisfactory up to the onset of bristle buckling. Figure 5.6 illustrates the force diagram for the forward and reverse loading modes. Consider a bristle sliding forwards through a pipe. It can be assumed that the bristle forms an angle with the pipe wall, this is represented by line OH. If the bristle is moved to the left then its motion will be opposed by a friction force μF, where F is the force normal to the pipe wall caused by the bristles resilience. A brush inside a pipe will be in transverse equilibrium so that the sum of all the F values for all the bristles extending from the brush core will be zero and the sum of all the μF components will equal the total force required to move the brush through the pipe in a forward direction. The sum of the force vectors (F) perpendicular to the wall of the pipe and the frictional force μF is the resultant force OC. Note that in the limiting case, the angle between OA and OC is the friction angle θ such that $\mathrm{Tan}\,\theta = \mu$. The resultant force OC can be resolved into two components, one perpendicular to the bristle OD and one component along the bristle DC. If the bristle is relatively straight and the thrust along the bristle is not sufficient to cause buckling then the bending of the bristle can be assumed to be caused by the bending force OD acting perpendicular to the bristle.

If the brush is then put into gripping mode by pushing the core in the opposite direction, the frictional force direction will be reversed. Assuming that the value of μ is the same in both directions, then the resultant force acting on the bristle tip will be along the line OE where the angle between OE and the pipe wall is once again θ, the friction angle. Although the limiting line of action of the resultant force acting on the bristle in gripping mode is known at this stage, the magnitude of the resultant force has yet to be determined. In forward sliding mode it was assumed that the shape of the bristle, that is, the bending of the bristle was determined by the bending force OD. If it is further assumed that the shape of the bristle remains the same in gripping mode, then, in gripping mode, it is only the thrust along the bristle that increases. If the line of the thrust along the bristle is extended it will eventually intersect with the friction angle line OE giving a thrust magnitude DE. From Figure 5.6 it can be seen that in gripping mode the resultant force acting at the bristle tip is OE which can be resolved into the two components OD perpendicular to the bristle and DE along the bristle. Resolving the resultants into the direction along the pipe axis, in both forward travel and gripping modes, allows the computation of the traction ratio of the force required to push a brush forward and the limiting force available in the gripping mode, that is, GE/BC.

The force diagram in Figure 5.7 shows that as the angle between the bristle and pipe wall approaches the friction angle θ, the thrust along the bristle DE increases considerably. This can give a proportionately larger GE/BC ratio. The force diagram in Figure 5.8 shows the situation in which the angle between the bristle tip and pipe wall is equal to the friction angle θ. In this case the vector representing the thrust along the bristle (extension of DC) is parallel to the vector of the resultant acting along the frictional angle line OE. Thus, these lines theoretically meet at infinity indicating a theoretical thrust along the bristle of infinity and, therefore, a theoretical "slip" to "grip" ratio of infinity. This result is illustrated in the spreadsheet, Figure 5.9, which shows the traction ratio approaching infinity as the angle of the bristle approaches the friction angle.

However, the bristle and the pipe wall are not able to sustain thrust forces of infinity and the model needs to be modified to accommodate bristle or pipe wall failure. For example, a slender flexible bristle will collapse by buckling before reaching a thrust of infinity and before plastic failure. A short stubby bristle may collapse by plastic yielding or the failure of the wall of the pipe may also limit the magnitudes of the bristle thrust force. The buckling of a slender bristle is probably the most relevant mode of failure and in this case the magnitude of the thrust along the bristle is limited to the buckling load, DF in Figure 5.10. As the thrust along the bristle is limited to the buckling load it is not possible for the resultant force acting on the bristle tip to reach the frictional limit. Thus, in instances of the bristle failing by buckling and if the bristle angle is near or beyond the friction angle, the axial force will be less than the limiting friction value and therefore the bristle will grip without slip, irrespective of the force applied to the core. However, as the bristle tip rotates about O and the bristle angle moves beyond the friction angle, the axial force that the bristle can sustain decreases to zero. Beyond this point, the axial force applied by the bristle to the core is reversed and the bristle "flips through" pushing the core along the pipe until the bristle has achieved the mirror image of the force diagram in Figure 5.6, Figure 5.11 illustrates the principle. The failure of the bristle by a buckling mechanism is an instability phenomenon and as a consequence the flip through of the bristle at the limit of the gripping mode is also a rapid and unstable process, hence the descriptive term "flipping through".

The model has been used to compare two illustrative specific cases. Case one is the comparison of three different bristle angles (α) whilst μ remains constant. Case two compares a constant bristle angle with two different μ values. Case one is illustrated by Figures 5.12, 5.13 and 5.14. The model assumes a constant μ value of 0.4. It can be clearly seen in Figure 5.12, where the angle between the bristle and pipe wall is large, the bristle is likely to generate relatively high thrust loads along the bristle. However, in Figures 5.13 and 5.14, as the angle between the bristle and pipe wall decreases the thrust load along the bristle also decreases. Figure 5.15, illustrates a situation where the angle of the bristle approaches the friction angle and shows that the thrust along the bristle increases rapidly.

Case two is illustrated by Figures 5.16 and 5.17. Here, the bristle angle α remains constant and μ changes. It can be seen that a higher μ value gives a higher ratio. Both of the above cases are as expected and both substantiate the ability of the model to predict actual behaviour. It should be noted that the bristle tip is assumed to be straight for this model, however, in real terms, it will have a slight curvature. The effect of curvature at the tip will be discussed later in Chapter Six.

The model provides a description of how the mechanism of traction works. The main objective of the research was to obtain the highest possible ratio between forward "slip" and rear "grip". Therefore, to be able to determine, by calculation or other means, the highest force possible a bristle is able to support prior to "buckling" is both desirable and beneficial.

5.4 BRISTLE BUCKLING: EULER

The model above assumes that a bristle can be considered to be a *strut,* that is, a compression member which is long compared to its cross-sectional area. Buckling theory concludes that such a member will fail due to buckling before the direct compressive stress reaches the yield point. Both the Euler and Rankine theories consider the axial load required to buckle a strut, Euler considers long struts and Rankine shorter struts. Unfortunately, due to the following limitations it has not been possible to apply either of these theories directly. The main limitation is that both theories require the strut to be initially straight. Additionally, the load must be applied axially and the material must be homogenous. To further complicate matters, both of the above theories require the applied load to increase gradually, a brush however is generally subjected to rapid cyclic loading.

However, buckling theory does provide a useful insight into how a bristle behaves under different constraint conditions. Euler's Theory covers four different constraint cases; *strut with both ends pinned, strut with one end fixed and one end free, strut with both ends fixed and strut with one end fixed and one end pinned.* Figure 5.18 shows the bristles deflected whilst being subjected to the above conditions. Below are the equations for the four cases mentioned above (Stephens 1970):

Case A: Strut with both ends pinned:

$$\mathbf{P} = \frac{\pi^2 . E . I}{l^2} \tag{5.2.13}$$

Case B: Strut with one end fixed and one end free:

$$\mathbf{P} = \frac{\pi^2 . E . I}{4l^2} \tag{5.2.14}$$

Case C: Strut with both ends fixed:

$$\mathbf{P} = \frac{4\pi^2 . E . I}{l^2} \tag{5.2.15}$$

Case D: Strut with one end fixed and one end pinned;

$$\boldsymbol{P} = \boldsymbol{20.25} \frac{EI}{l^2} \approx \frac{2\pi^2 EI}{l^2} \tag{5.2.16}$$

A radially bristled brush when inserted into a pipe and subjected to a force, as if it were in reverse or "grip" mode, would normally reverse by core rotation. Here, the bristles move out of plane, that is, a couple is induced into the brush core. If Euler's Theory was used to attempt to calculate the force for such a brush unit then the most appropriate case for buckling would be *Case B "Strut with one end fixed and one end free"*. However, if a X-ply brush was subjected to reverse "grip" forces then it would be likely to buckle as in *Case D, "Strut with one end fixed and one end pinned"*. As such, it would be interesting to compare a radially bristled brush with a X-ply brush using the two Euler cases described above;

Case B:

$$\mathbf{P} = \frac{\pi^2.E.I}{4l^2} = \pi^2 . \frac{(200.10^9).(7.854.10^{-17})}{4.(48.10^{-3})^2} = \mathbf{P = 0.01682N}$$

P (per brush) = 0.01682 x (8x27x550) = 1998N.

Case D:

$$P = \frac{2\pi^2 EI}{l^2} = 2.\pi^2 . \frac{(200.10^9).(7.854.10^{-17})}{(48.10^{-3})^2} = \mathbf{P = 0.135N}$$

P (per brush) = 0.135 x (8x27x550) = 15,984N.

From the above, it can be seen that *Case D* can support 8x the force that *Case B* can support. However, the reader should appreciate that neither of the above examples obey the rules for Euler's theory of buckling. Further, it is considered that it is the bristle tip/pipe wall interface that becomes the limiting factor in determining whether a brush can support a specific "reverse" force, that is, a brush will slip down the pipe before the bristles experience the forces calculated above. However, it is known, through experimentation, that in dry conditions, X-plying the bristles does make an appreciable difference to the force a brush is able to support in the reverse or "grip" mode prior to failure.

5.5 LARGE ELASTIC BRISTLE DEFORMATION

Further theoretical and simulation studies have been undertaken by Wilson (1998). This analytical work seeks to take a more rigorous approach to the subject of the large elastic deformation of bristles. Although this material is not yet completed or published it does indicate a number of promising approaches to using simulation and numerical methods for obtaining bristle forces and shapes. Although this work is interesting it has not contributed directly to the present research but is a start on the development of more complex analytical models. The simple models described above have proved sufficient for the practical design work in developing a broad range of experimental and commercial vehicles.

The work by Wilson has focused upon:

1. Determining the force required to load a brush into a pipeline.
2. Determining the slope of the bristle under loaded conditions.
3. The out-of-plane deformation of the bristle during reversal.

5.6 CONCLUSION

This chapter begins by considering the force required to deflect a bristle and the force required to drag that bristle tip over the internal wall of a pipeline.

As a first approximation the bristle has been considered as a simple cantilever, however, in real terms, the mechanics are considerably more complex than this. This calculation has proven to be robust with a significant number of experimental and commercial vehicles being built using the results. As will be shown, the results of these calculations also satisfactorily match the experimental results.

To determine the force a brush is able to support prior to collapsing or deflecting out of plane is complicated. Again, a first approximation can be determined from the model presented within this chapter, however, this is also an extremely complex issue. If a brush is subjected to reverse or "grip" mode forces, it is extremely difficult to predict the point at which the stored energy in each bristle will be released or the direction in which the brush core will rotate. In many cases it is difficult to predict the mode of failure. However, the use of X-ply brush units, in dry conditions, significantly increases the force a brush can withstand prior to failure and this phenomenon can be predicted by means of the buckling theory.

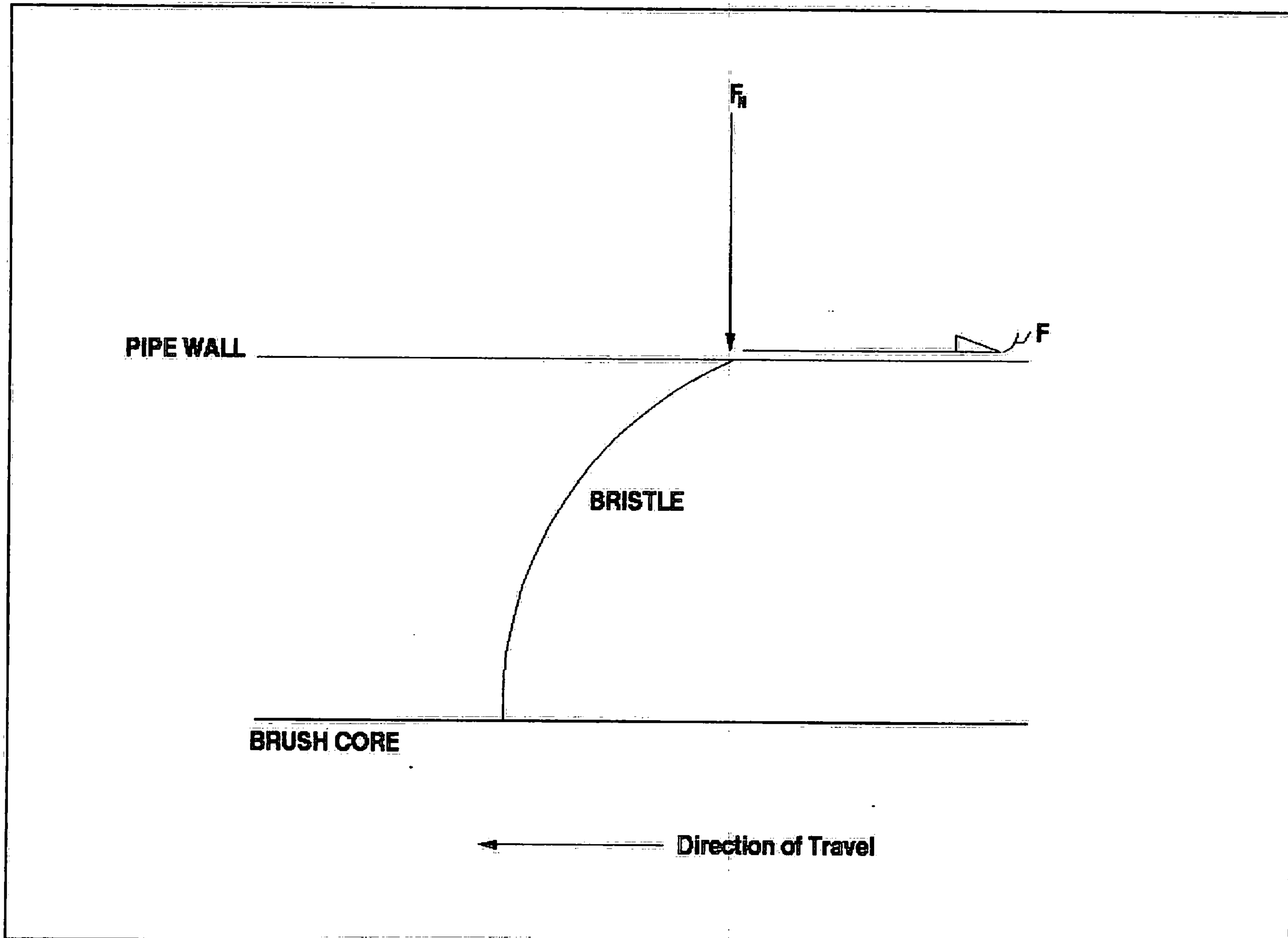


FIGURE 5.1 **THE FORCES PRESENT AT THE BRISTLE TIP/PIPE WALL INTERFACE**

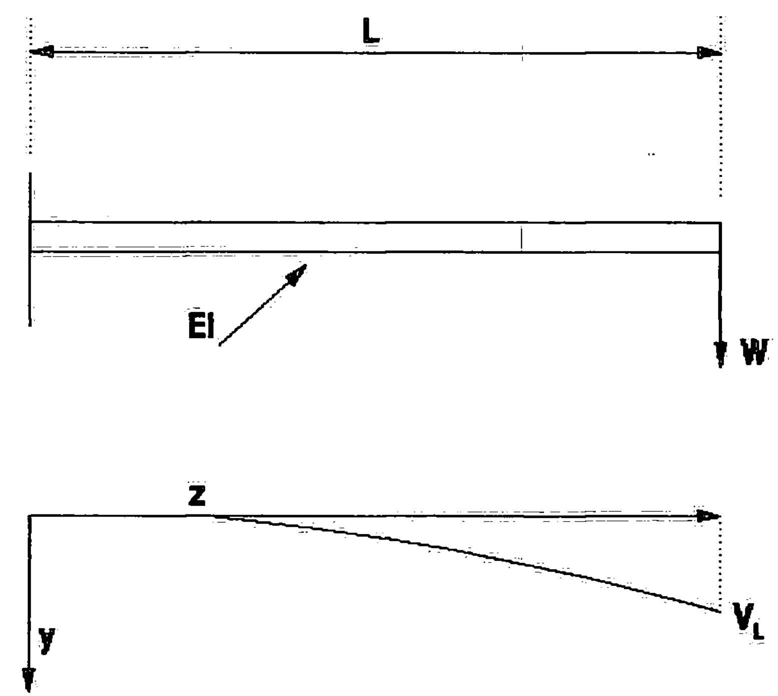


FIGURE 5.2 A SIMPLE CANTILEVER

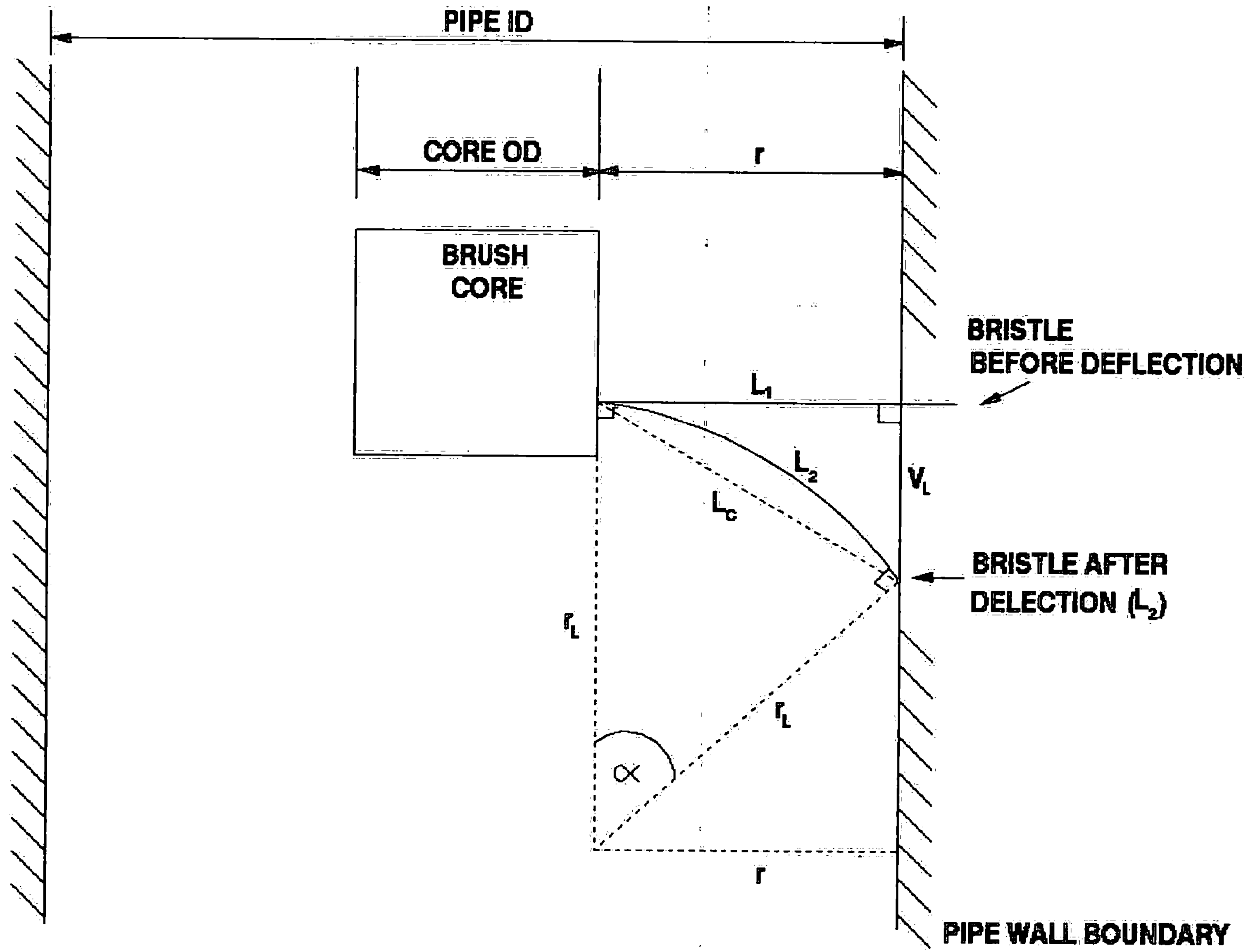


FIGURE 5.3 THE APPROXIMATE BRISTLE DEFLECTION AFTER INSERTION INTO A PIPE

ENTER DATA FOR ELEVEN FIELDS BELOW

CORE DIAMETER (M) =	0.082
HEAD DIAMETER (M) =	0.183
YOUNG'S MODULUS OF BRISTLE MATL.(GPa/NM	2E+11
INTERNAL DIAMETER OF PIPE (M) =	0.155
DIAMETER OF BRISTLE (M) =	0.0002
PRE-SWEEP FROM CORE NORMAL (DEGREES)	0
NO. OF ROWS PER CORE =	27
NO. OF HOLES/CIRCUMFERENCE =	7
NO. OF BRISTLES/HOLE =	550
TOTAL NO. BRISTLES/CORE =	103950
NO. OF CORES PER SIDE OF CYLINDER =	1
LENGTH OF BRISTLE (L) In (M) =	0.0505
BRUSH CAVITY LENGTH (r) IN (M) =	0.0365

NORMAL FORCE BRUSH APPLIES (Kg) 234.52

mu	F In Kg	mu	F In Kg	mu	F In Kg	mu	F In Kg
0.025	5.86301	0.275	64.4931	0.525	123.123	0.775	181.753
0.05	11.726	0.3	70.3561	0.55	128.986	0.8	187.616
0.075	17.589	0.325	76.2191	0.575	134.849	0.825	193.479
0.1	23.452	0.35	82.0821	0.6	140.712	0.85	199.342
0.125	29.315	0.375	87.9451	0.625	146.575	0.875	205.205
0.15	35.178	0.4	93.8081	0.65	152.438	0.9	211.068
0.175	41.0411	0.425	99.6711	0.675	158.301	0.925	216.931
0.2	46.9041	0.45	105.534	0.7	164.164	0.95	222.794
0.225	52.7671	0.475	111.397	0.725	170.027	0.975	228.657
0.25	58.6301	0.5	117.26	0.75	175.89	1	234.52

TEST FORCE RECORDED (Kg) =	76.8
mu THEREFORE =	0.32748

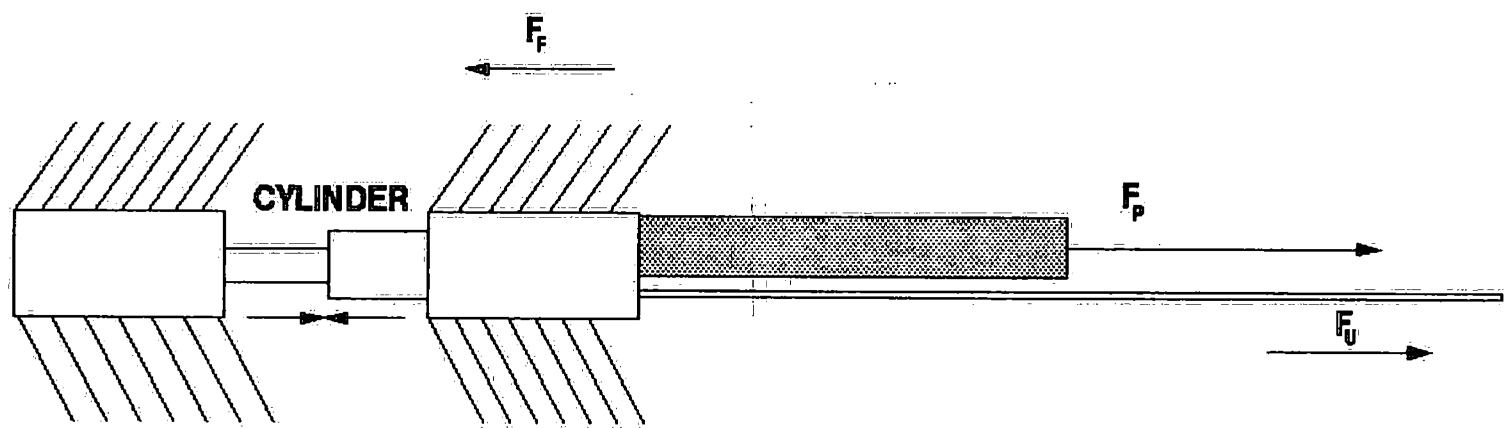
ANGLE BETWEEN BRISTLE NORMAL & SWEPT POSITION (DEGREES)=					38.564
ANGLE BETWEEN BRISTLE TIP AND PIPE WALL (DEGREES)=					51.436
CALCULATION OF LC (mm)=					0.04668
CALCULATION OF "VL" (M)					0.0291
VL CORRECT WHEN "RESULT" =1	0.0291				
RESULT	1.00858				
A	2361.8				
B	0.0014				
C	1.34879	77.3193	0.22018	0.77982	
D	1.57328				
CALCULATION OF "I"					7.9E-17
"I" = (PI*D^4)/64 =					
CALCULATION OF FORCE "W" PER BRISTLE (N)					0.01065
W=(3.E.I.(VL/L^3)) =					
COMPONENTS AS CALCULATED					
VL =	0.0291				
L =	0.0505				
E =	2E+11				
I =	7.9E-17				
W =	?				
FORCE "W" PER BRUSH (N)					1106.84
FORCE "W" PER BRUSH (Kg)					112.828
CALCULATION OF VL/(L-r)					2.07857

CALCUALATION TO OBTAIN EFFECTIVE SHORTENING OF VL DUE TO PRE-SWEEP	
NEW ANGLE BETWEEN BRISTLE AND CORE BODY (DEGREES)=	38.564
180-ANGLE BETWEEN BRISTLE TIP AND PIPE WALL (DEGREES=	90
SINE RULE USED TO CALCULATE NEW VL (M)=	0.62339
INTERMMEDIATE STEP..	1
NEW VL(M)=	0.0291
FORCE "W" PER BRISTLE (N)	0.01065
FORCE "W" PER BRUSH (N)	1106.84
CALCULATION OF VL/(L-r)	2.07857
NORMAL FORCE BRUSH APPLIES (Kg)	234.52

TO CALCULATE THRUST ENTER USER ENTRY FIELDS BELOW:		
PI	3.142	
PISTON DIAMETER (mm)	50	
GAUGE PRESSURE (BAR)	16	
THEORETICAL THRUST		THRUST LESS 20%
THRUST =	320.265	256.228

TO CALCULATE PULL ENTER USER ENTRY FIELDS BELOW:		
PI	3.142	
PISTON DIAMETER (mm)	50	
PISTON ROD DIAMETER (mm)	20	
GAUGE PRESSURE (BAR)	16	
THEORETICAL PULL		PULL LESS 20%
PULL =	269.04	215.232

FIGURE 5.4 AN EXAMPLE OF THE SPREADSHEET USED TO CALCULATE THE "FORWARD" FORCE



F_F = FORCE TO MOVE BRUSH

F_P = FORCE TO MOVE PAYLOAD

F_U = FORCE TO MOVE UMBILICAL

FIGURE 5.5 THE FORCES PRESENT AS THE TRAILING CYLINDER OPERATES IN "PULL" MODE

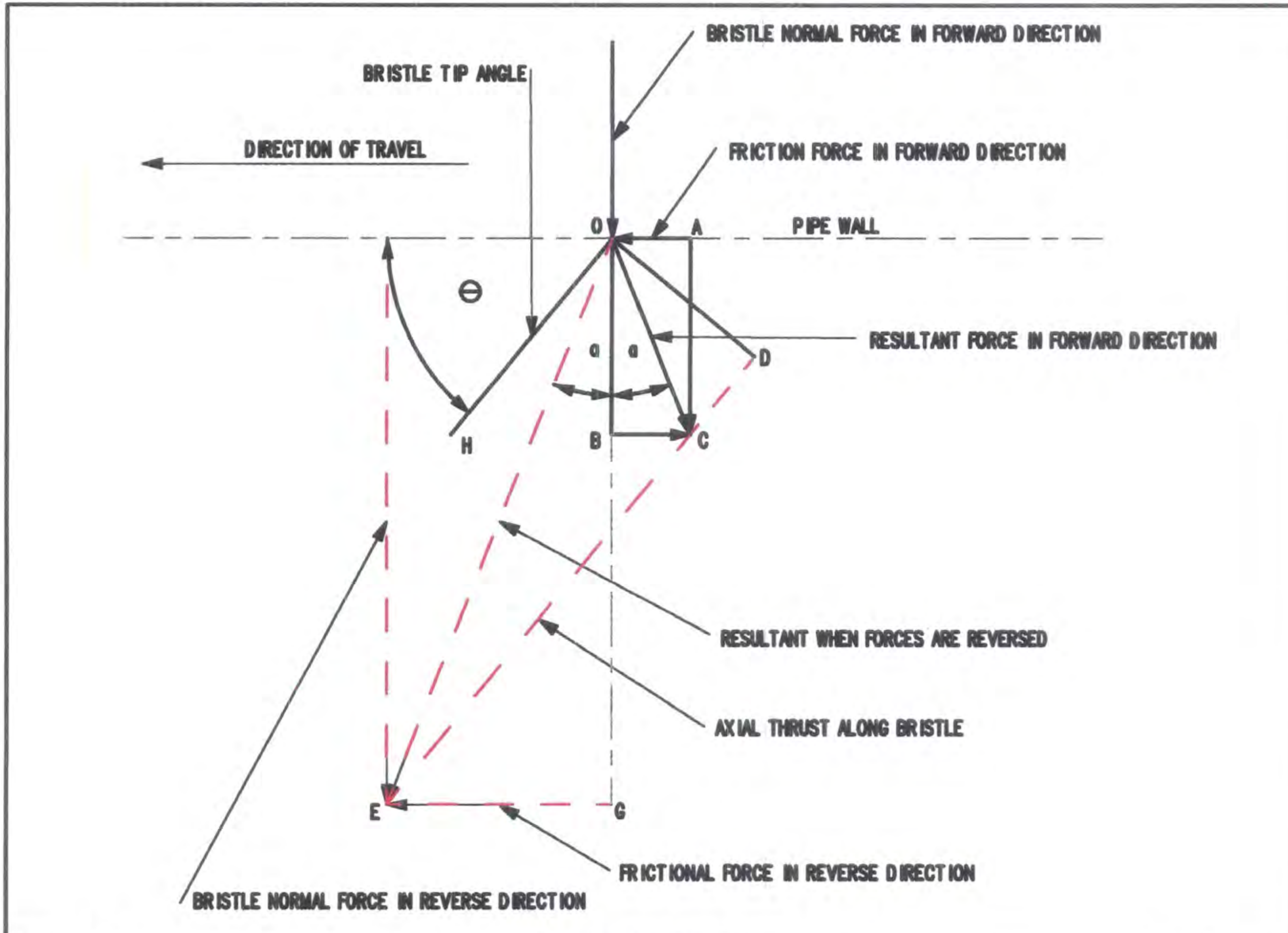


FIGURE 5.6 MODEL ILLUSTRATING FORCES PRESENT

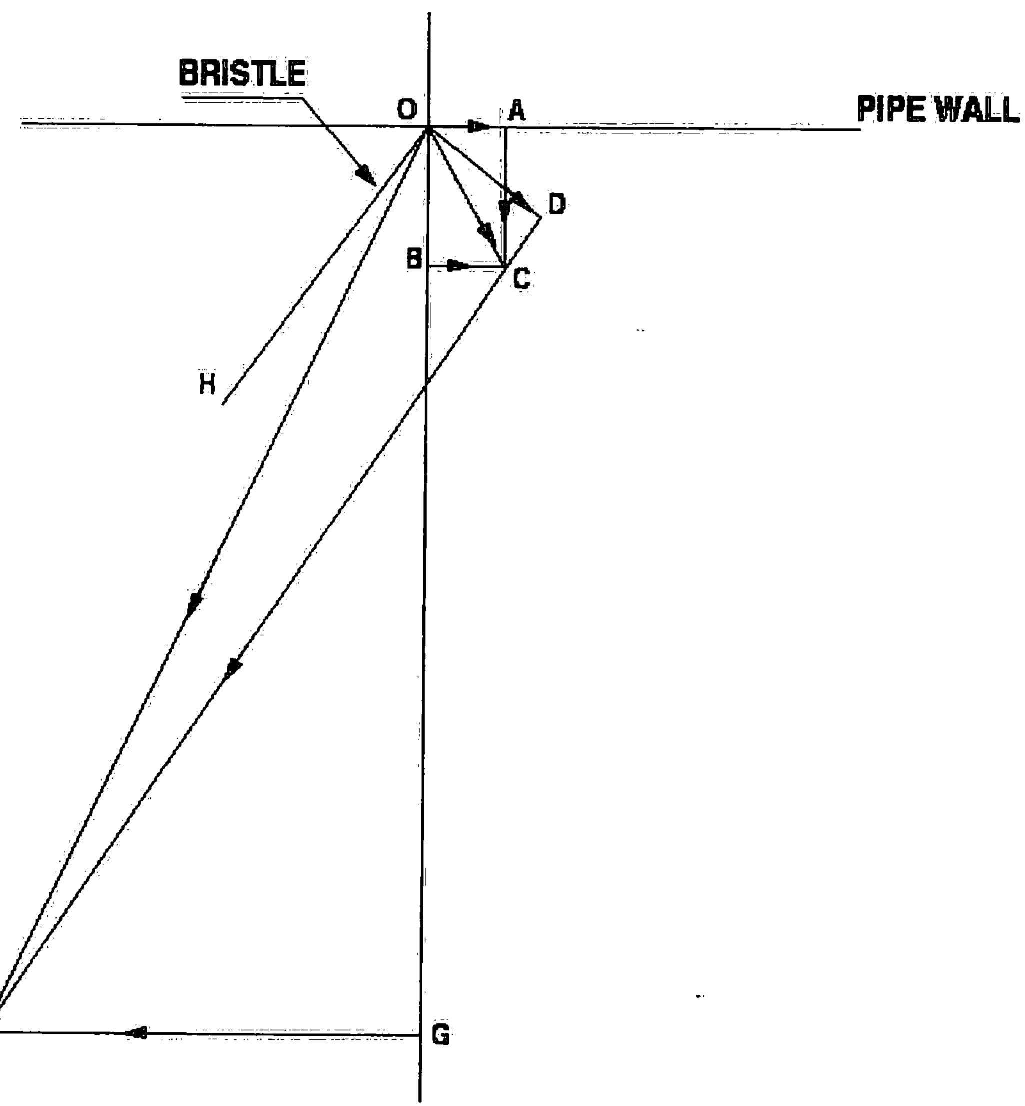


FIGURE 5.7 FORCES AT BRISTLE/PIPEWALL INTERFACE

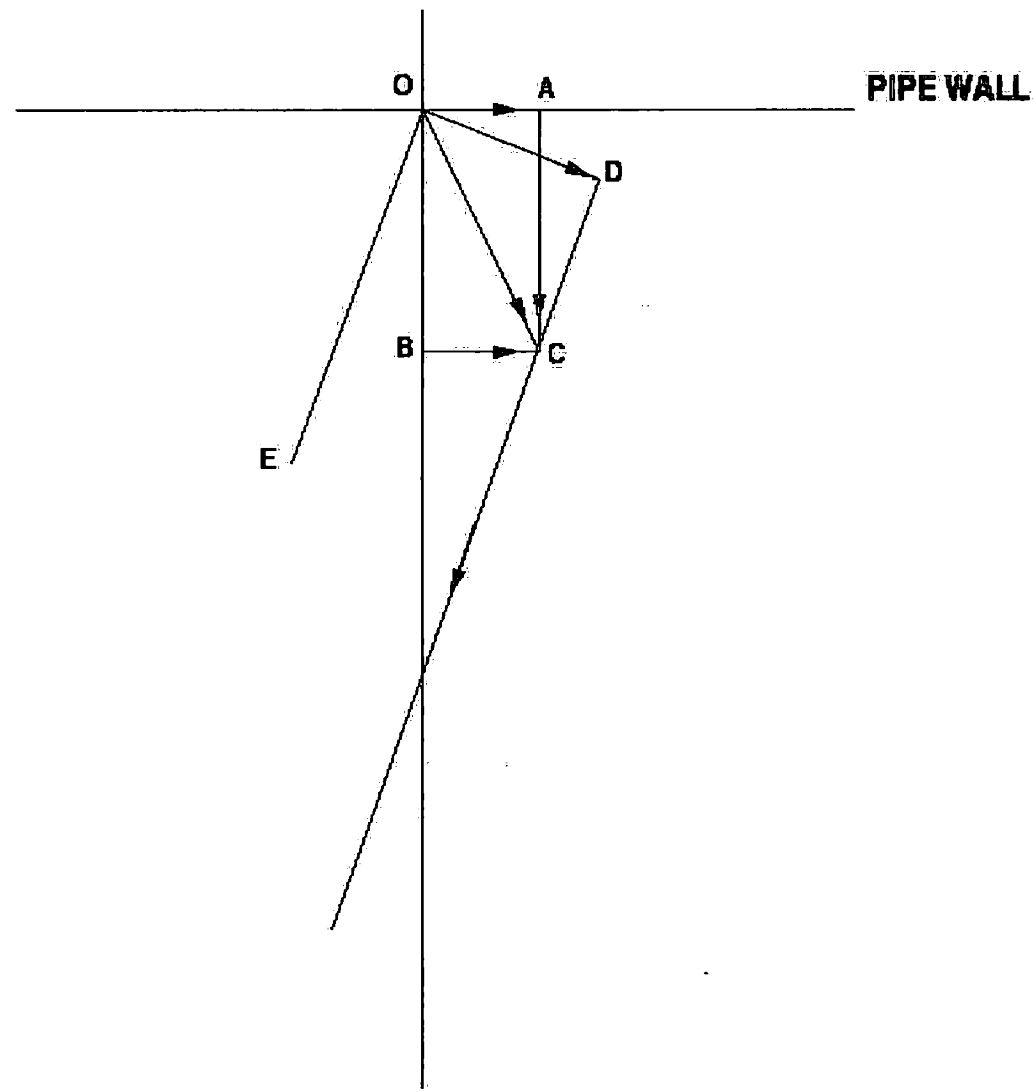


FIGURE 5.8 FORCES AT BRISTLE/PIPEWALL INTERFACE

									Friction Value												
		0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	0.55	0.6	0.65	0.7	0.75	0.8	0.85	0.9	0.95	1
	2	-5.63	-2.07	-1.6	-1.42	-1.33	-1.26	-1.22	-1.19	-1.17	-1.15	-1.14	-1.12	-1.11	-1.1	-1.09	-1.09	-1.08	-1.08	-1.07	-1.07
	4	6.02	-5.65	-2.75	-2.08	-1.78	-1.61	-1.5	-1.42	-1.37	-1.33	-1.29	-1.26	-1.24	-1.22	-1.21	-1.19	-1.18	-1.17	-1.16	-1.15
	6	2.82	40.18	-5.68	-3.22	-2.45	-2.08	-1.86	-1.71	-1.61	-1.53	-1.47	-1.42	-1.39	-1.35	-1.33	-1.3	-1.28	-1.26	-1.25	-1.23
	8	2.11	5.93	-30.7	-5.73	-3.57	-2.76	-2.34	-2.08	-1.91	-1.78	-1.69	-1.61	-1.55	-1.5	-1.46	-1.43	-1.4	-1.37	-1.35	-1.33
	10	1.79	3.62	12.4	-15.9	-5.79	-3.85	-3.03	-2.58	-2.29	-2.09	-1.94	-1.83	-1.75	-1.67	-1.61	-1.57	-1.52	-1.49	-1.46	-1.43
	12	1.62	2.78	5.8	32.86	-12.35	-5.86	-4.09	-3.27	-2.79	-2.48	-2.26	-2.1	-1.97	-1.87	-1.79	-1.72	-1.67	-1.62	-1.58	-1.54
	14	1.5	2.34	4.02	9.11	-743.1	-10.84	-5.95	-4.31	-3.49	-2.99	-2.66	-2.42	-2.25	-2.11	-2	-1.91	-1.83	-1.77	-1.71	-1.66
	16	1.42	2.07	3.19	5.61	14.61	-44.27	-10.07	-6.06	-4.51	-3.69	-3.18	-2.83	-2.58	-2.39	-2.24	-2.12	-2.02	-1.94	-1.86	-1.8
	18	1.36	1.89	2.72	4.2	7.67	25.06	-26.9	-9.66	-6.2	-4.71	-3.89	-3.36	-3	-2.73	-2.53	-2.37	-2.24	-2.13	-2.04	-1.96
	20	1.32	1.76	2.4	3.44	5.39	10.38	51.11	-21.2	-9.46	-6.35	-4.91	-4.08	-3.54	-3.17	-2.89	-2.67	-2.5	-2.36	-2.24	-2.14
	22	1.28	1.66	2.18	2.96	4.25	6.77	13.96	199.7	-18.58	-9.42	-6.54	-5.12	-4.29	-3.73	-3.34	-3.04	-2.81	-2.63	-2.48	-2.36
	24	1.25	1.58	2.02	2.63	3.56	5.13	8.35	18.69	-187.6	-17.26	-9.5	-6.75	-5.35	-4.5	-3.92	-3.51	-3.2	-2.96	-2.76	-2.61
	26	1.23	1.52	1.89	2.4	3.1	4.2	6.08	10.12	24.85	-80.5	-16.7	-9.69	-7.01	-5.6	-4.72	-4.12	-3.7	-3.37	-3.11	-2.9
	28	1.21	1.46	1.79	2.21	2.76	3.59	4.85	7.07	12.02	32.54	-59.1	-16.57	-10	-7.32	-5.87	-4.96	-4.34	-3.89	-3.54	-3.27
	30	1.19	1.42	1.7	2.06	2.53	3.16	4.08	5.51	8.07	13.93	41.22	-52	-16.89	-10.4	-7.69	-6.19	-5.24	-4.58	-4.1	-3.73
	32	1.17	1.38	1.63	1.94	2.33	2.85	3.55	4.56	6.15	9.01	15.69	49.25	-50.73	-17.63	-11	-8.14	-6.55	-5.54	-4.84	-4.33
	34	1.16	1.35	1.57	1.84	2.18	2.6	3.16	3.91	5.01	6.73	9.84	17.11	54.04	-53.9	-18.87	-11.75	-8.69	-6.98	-5.9	-4.14
	36	1.15	1.32	1.52	1.76	2.05	2.41	2.86	3.45	4.26	5.41	7.23	10.48	17.98	53.75	-62.95	-20.78	-12.77	-9.38	-7.5	-6.31
	38	1.14	1.29	1.48	1.69	1.94	2.25	2.62	3.1	3.72	4.56	5.76	7.62	10.9	18.22	48.95	-84.5	-23.74	-14.16	-10.26	-8.14
	40	1.13	1.27	1.44	1.63	1.85	2.11	2.43	2.82	3.31	3.95	4.81	6.02	7.86	11.07	17.84	41.92	-155	-28.56	-16.13	-11.4
	42	1.12	1.25	1.4	1.57	1.77	2	2.27	2.6	3	3.5	4.14	5	6.19	7.99	10.97	16.94	34.73	4455.96	-37.3	-19.08
Bristle	44	1.11	1.23	1.37	1.52	1.7	1.9	2.14	2.41	2.75	3.15	3.65	4.28	5.12	6.27	7.95	10.66	15.7	28.4	122.11	-57.3
Angle	46	1.1	1.21	1.34	1.48	1.64	1.82	2.02	2.26	2.54	2.87	3.27	3.76	4.37	5.17	6.25	7.79	10.16	14.28	23.21	57.29
	48	1.09	1.2	1.31	1.44	1.58	1.74	1.92	2.13	2.36	2.64	2.96	3.35	3.82	4.41	5.16	6.15	7.52	9.55	12.83	19.08
	50	1.09	1.18	1.29	1.4	1.53	1.67	1.83	2.01	2.21	2.45	2.71	3.03	3.4	3.85	4.4	5.08	5.97	7.1	8.86	11.43
	52	1.08	1.17	1.27	1.37	1.49	1.61	1.75	1.91	2.08	2.28	2.51	2.77	3.06	3.41	3.83	4.33	4.95	5.74	6.76	8.14
	54	1.08	1.16	1.25	1.34	1.44	1.56	1.68	1.82	1.97	2.14	2.33	2.55	2.79	3.07	3.4	3.78	4.23	4.78	5.46	6.31
	56	1.07	1.15	1.23	1.31	1.41	1.51	1.62	1.74	1.87	2.02	2.18	2.36	2.56	2.79	3.05	3.34	3.69	4.09	4.57	5.14
	58	1.07	1.13	1.21	1.29	1.37	1.46	1.56	1.67	1.78	1.91	2.05	2.2	2.37	2.56	2.76	3	3.27	3.57	3.92	4.33
	60	1.06	1.12	1.19	1.26	1.34	1.42	1.51	1.6	1.7	1.81	1.93	2.06	2.2	2.36	2.53	2.72	2.93	3.16	3.43	3.73

FIGURE 5.9 SPREADSHEET TO COMPUTE THE FORWARD TO REVERSE RATIOS FOR DIFFERENT VALUES OF BRISTLE ANGLE AND COEFFICIENT OF FRICTION

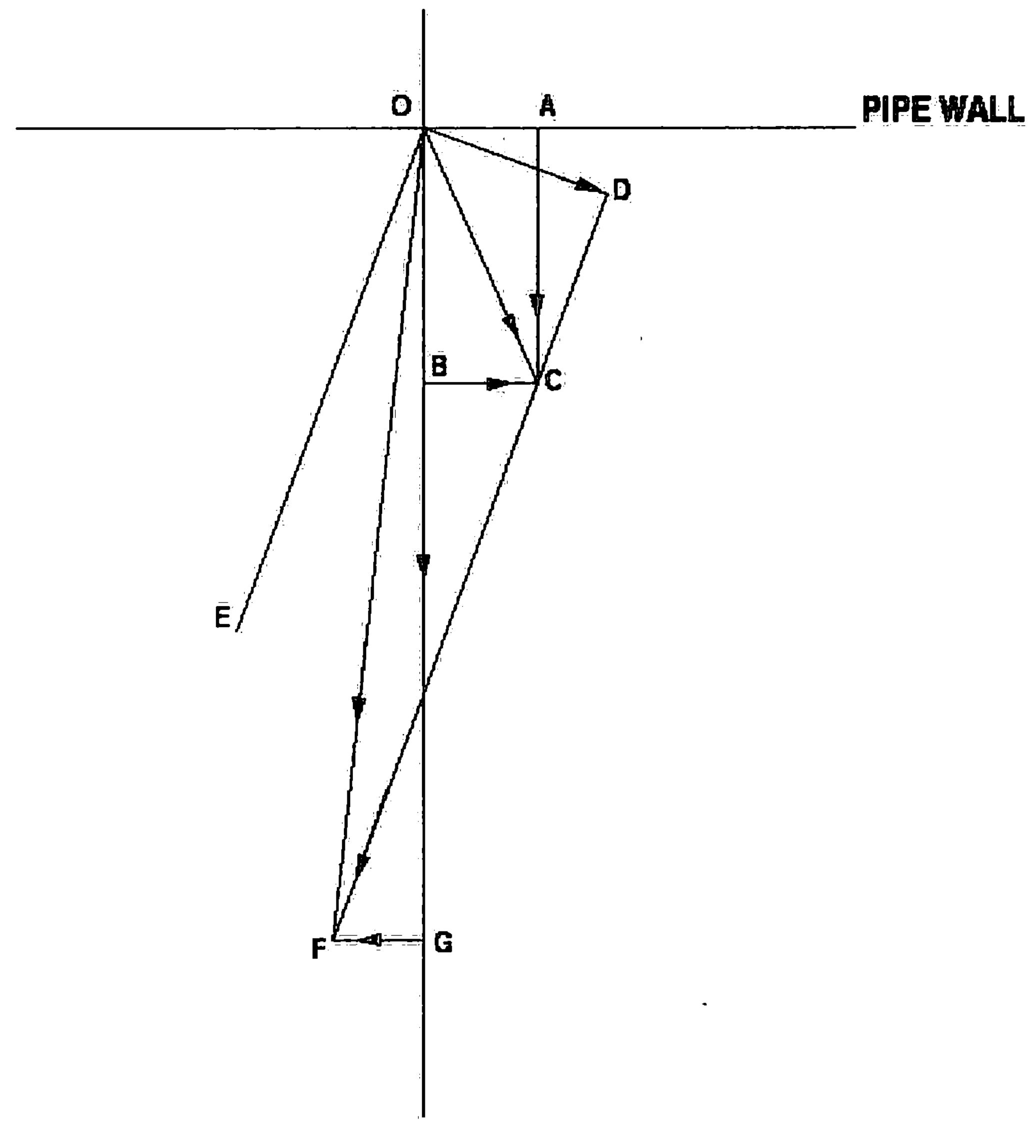


FIGURE 5.10 FORCES AT BRISTLE/PIPEWALL INTERFACE

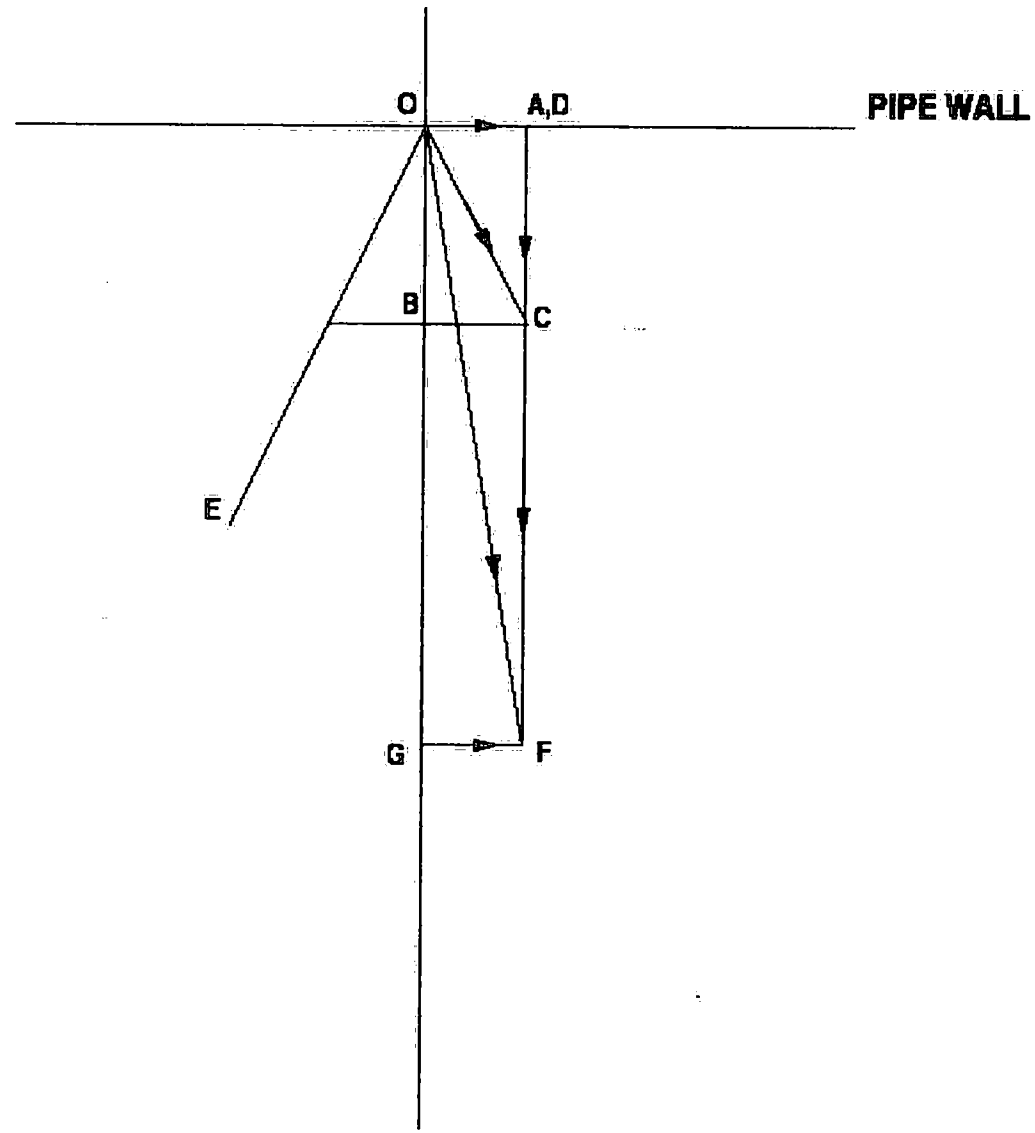


FIGURE 5.11 FORCES AT BRISTLE/PIPEWALL INTERFACE

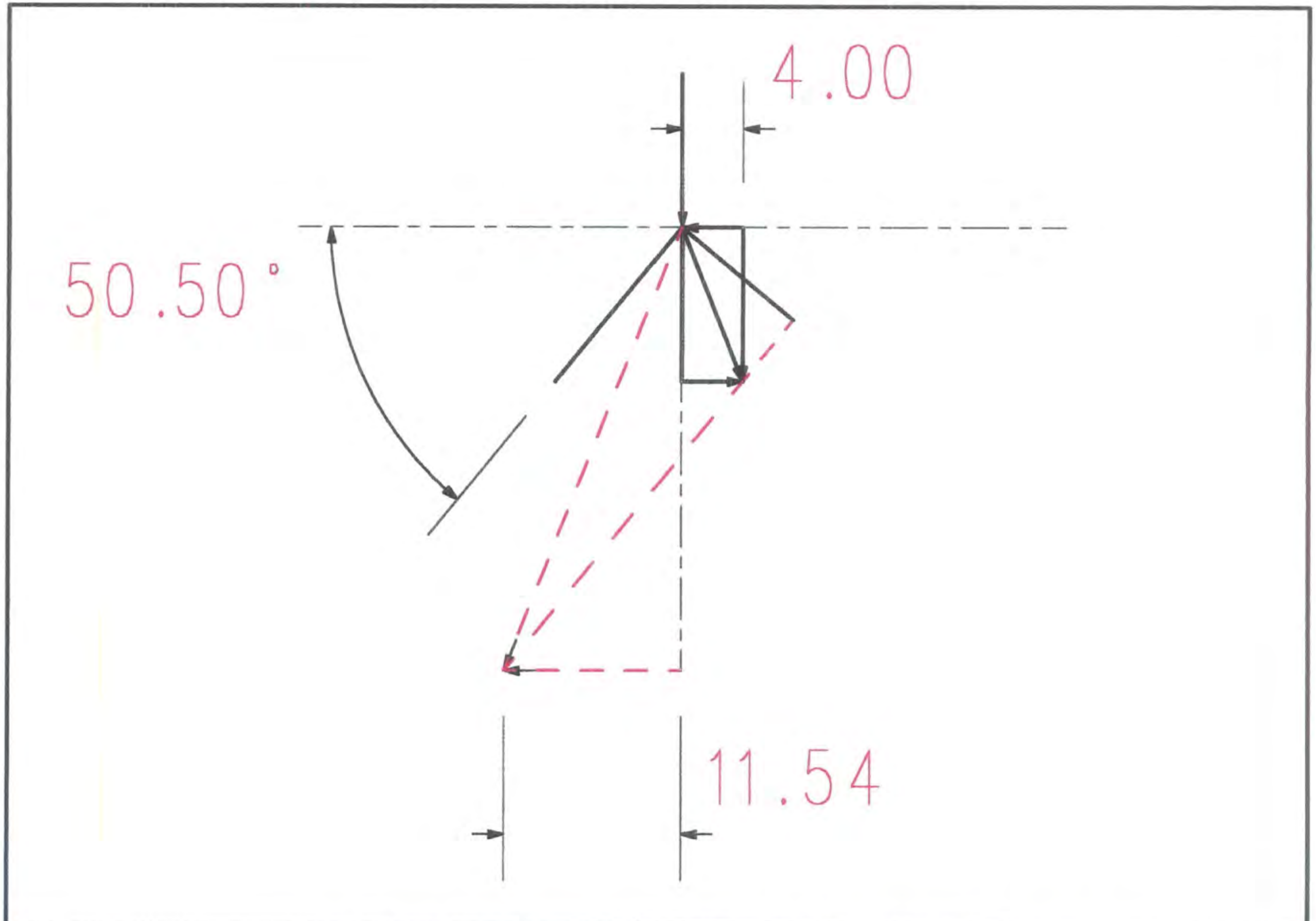


FIGURE 5.12 BRISTLE MODEL: BRISTLE ANGLE 50.5 DEGREES, FRICTION 0.4, FINAL RATIO 1:2.89

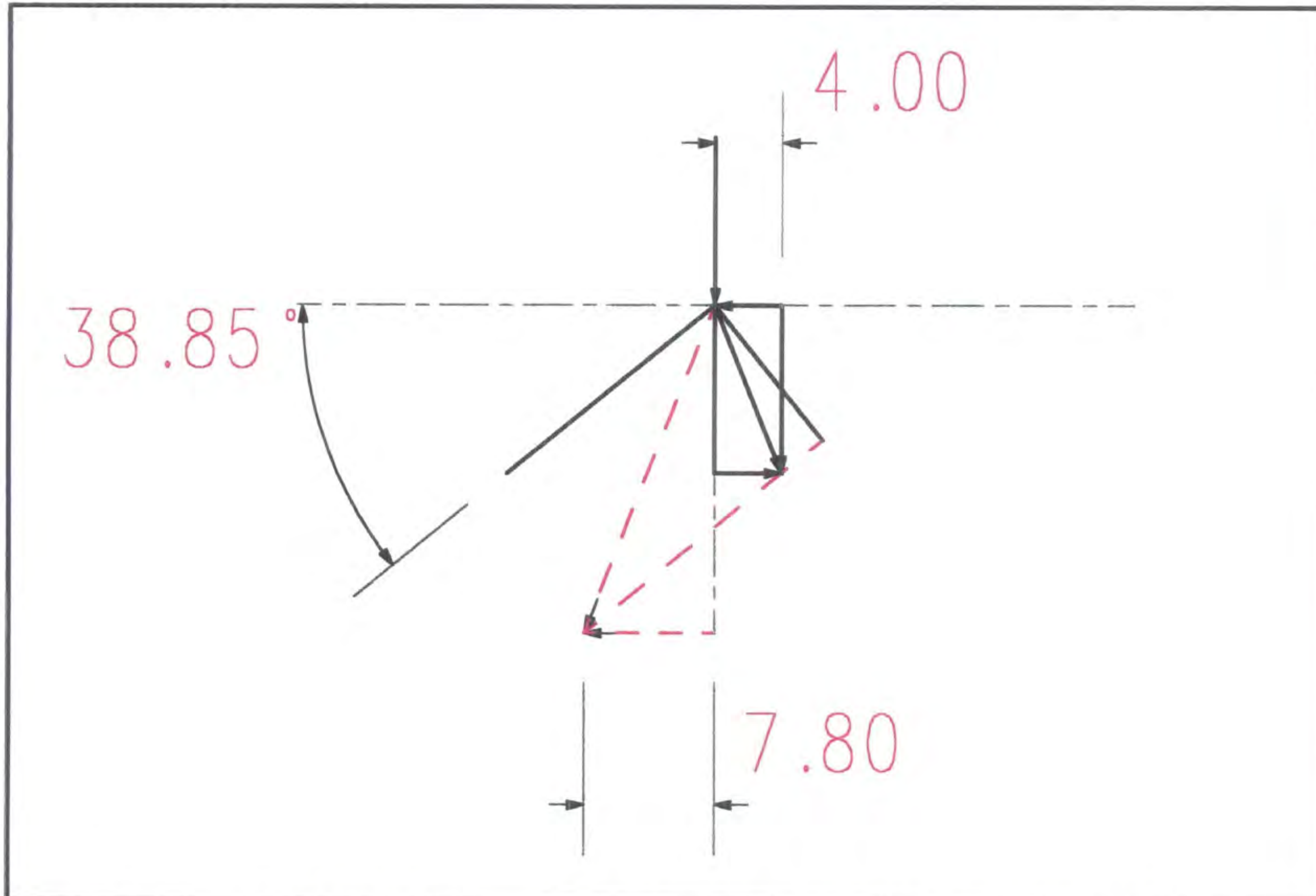


FIGURE 5.13 BRISTLE MODEL: BRISTLE ANGLE 38.85 DEGREES, FRICTION 0.4, FINAL RATIO 1:1.95

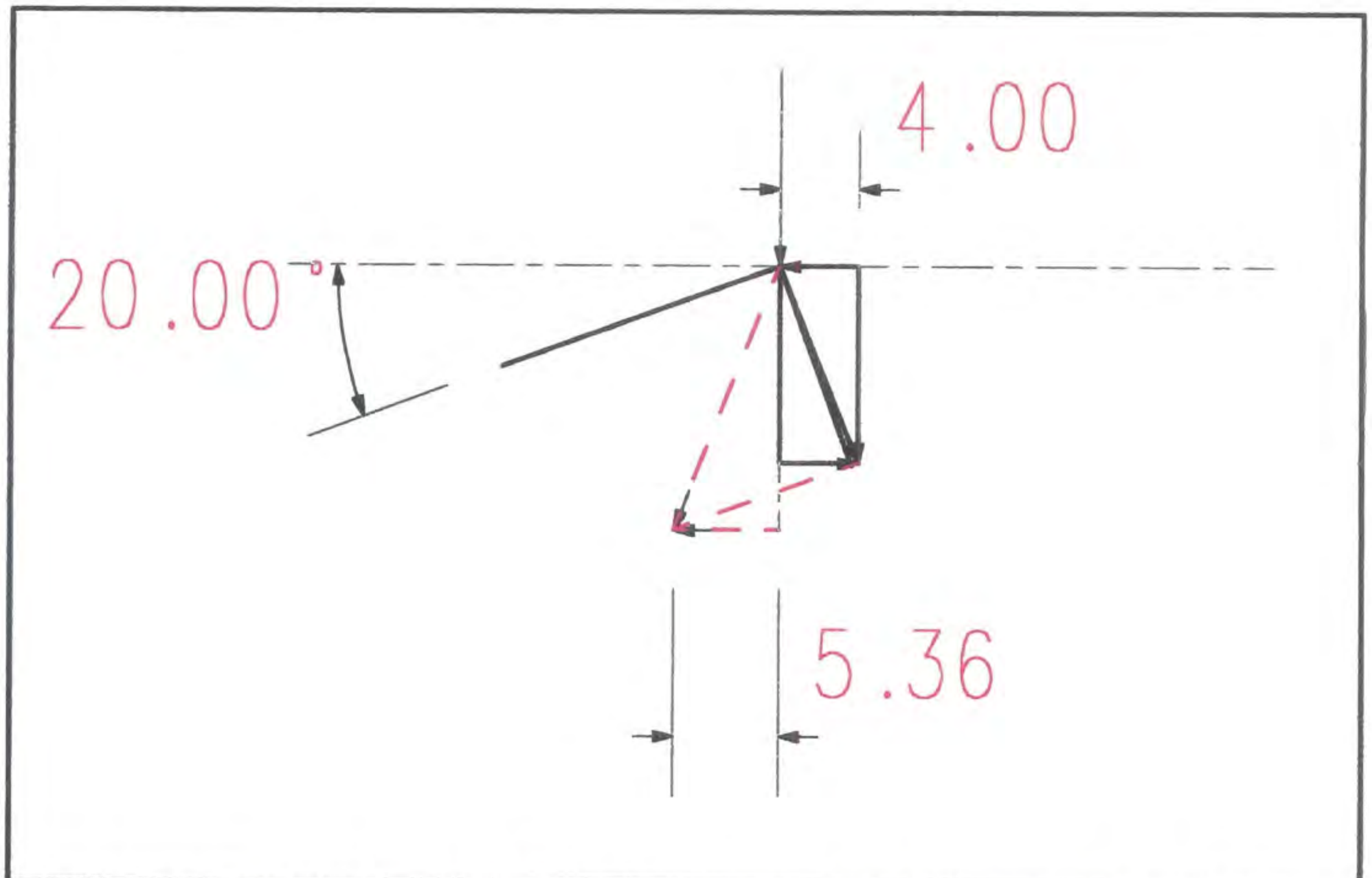


FIGURE 5.14 BRISTLE MODEL: BRISTLE ANGLE 20 DEGREES, FRICTION 0.4, FINAL RATIO 1:1.34

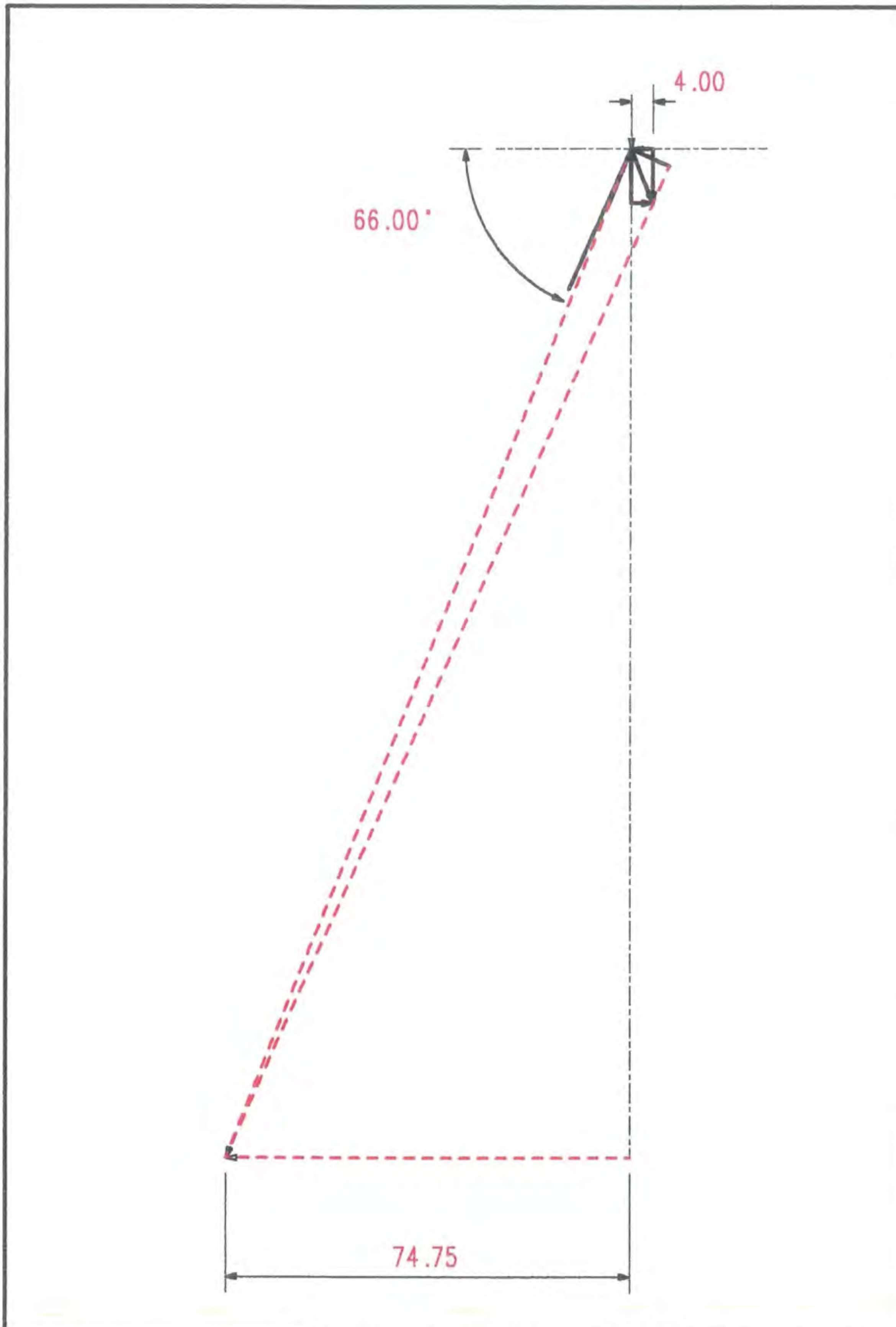


FIGURE 5.15 BRISTLE MODEL: BRISTLE ANGLE 66 DEGREES, FRICTION 0.4, FINAL RATIO 1:18.7

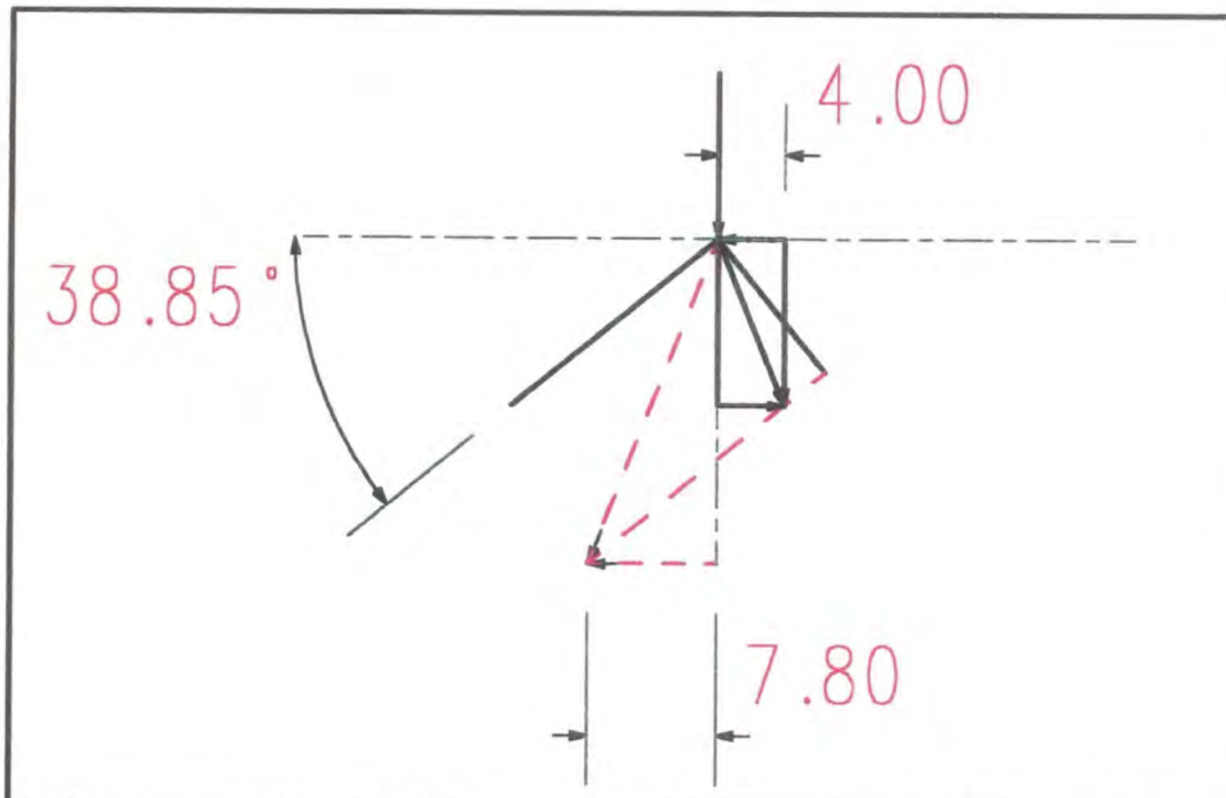


FIGURE 5.16 BRISTLE MODEL: BRISTLE ANGLE 38.85 DEGREES, FRICTION 0.4, FINAL RATIO, 1:1.95

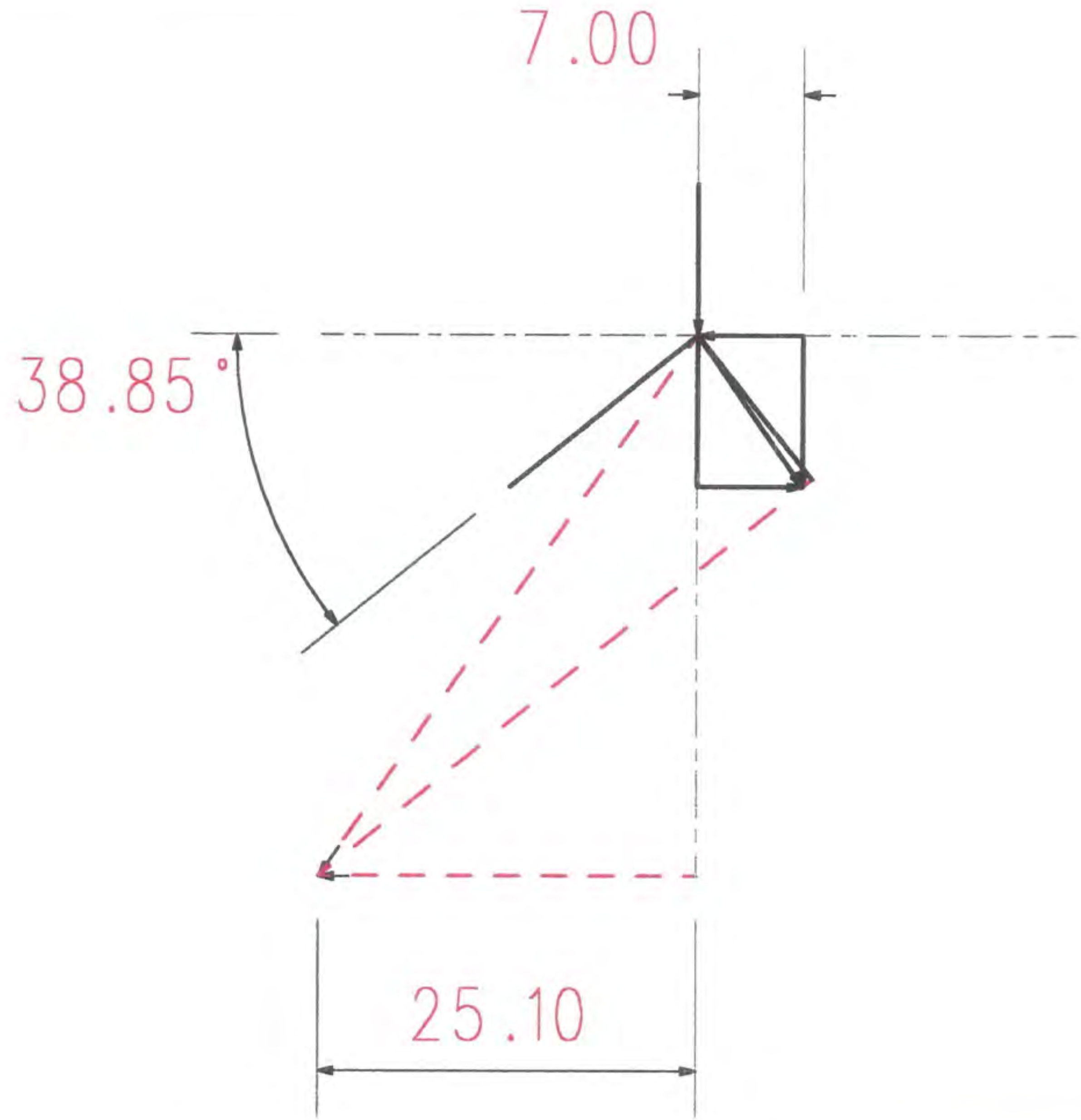
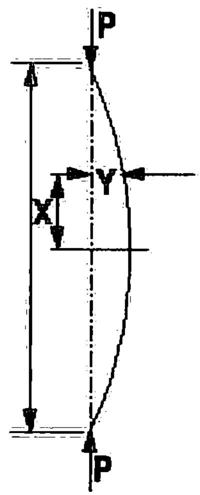
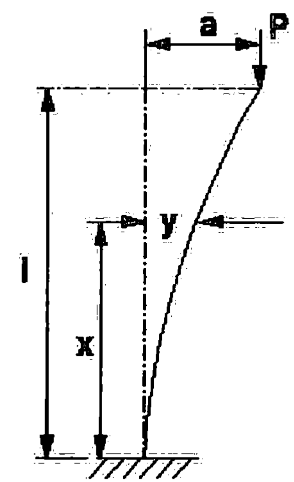


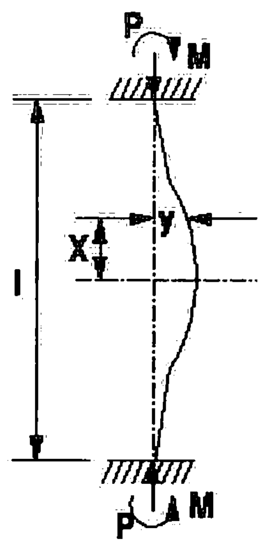
FIGURE 5.17 BRISTLE MODEL: BRISTLE ANGLE 38.85 DEGREES, FRICTION 0.7, FINAL RATIO 1:3.59



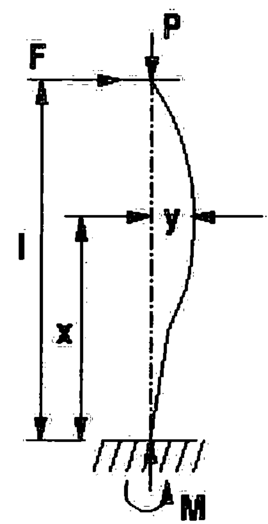
STRUT WITH BOTH ENDS PINNED



STRUT WITH ONE END FIXED & ONE END FREE



STRUT WITH BOTH ENDS FIXED



STRUT WITH ONE END FIXED & ONE END PINNED

FIGURE 5.18 EULER'S THEORY (Stephens 1970)

CHAPTER 6

EXPERIMENTAL INVESTIGATION

6.1 INTRODUCTION

The testing of the bristles could have taken one of two forms; either individual bristles could have been tested or complete brush ring assemblies. The decision was taken to test complete brush ring assemblies within short sections of pipe, each pipe section being approximately 300mm (12") long. The short pipe sections were cut from commercial pipelines which were not being used, thus, testing was as close to reality as possible. Both steel and polymeric bristled brushes were tested in a range of pipe sections with different internal diameters and surface roughness conditions. Additionally, the brushes were also tested under different lubrication conditions.

Each individual brush ring was tested in an attempt to determine the following:

1. The maximum force required to insert or load the brush into the pipe, this was subsequently termed the *"Loading Force"*.

2. The force required to push the brush unit forwards through the pipe, that is, the force required to overcome friction. This was termed *"Forward Force"*.
3. The force required to reverse the brush unit whilst in the pipe. Reversal occurred in a number of ways, each of which will be discussed later in this chapter. This was termed *"Reversal Force"*. Where the core of the brush was subjected to rotation during reversal, the "Direction of Rotation" (*DOR*) was recorded.

6.2 METHOD

The photograph in Figure 6.1 details the rig layout. A "Clockhouse" compression testing machine was used as the "platform" for the tests. As the brush units were to be "pushed" through the pipes, a compression testing machine provided an ideal platform on which to conduct the tests. All tests were carried out with the machine operating at its fastest speed, 0.174mm/s. However, it should be noted that this is a significantly lower speed than the brush unit would actually travel at whilst deployed in a pipeline. Additionally, a vehicle operating within a pipeline applies the axial thrust or pull to its brushes in a rapid cycle. Thus, in practice, a brush unit would experience rapid acceleration and deceleration.

The pipe sections were placed in a lathe and both ends faced off, this ensured that the axis of the pipe was maintained perpendicular to the loading plate of the testing machine. The "Clockhouse" rig consisted of a machined circular loading plate which was located on top of the lifting platform of the machine. The photograph in Figure 6.2 shows the incorporation of a linear transducer with its displacement shaft contacting the lifting platform. After calibration, this provided an accurate measurement of distance travelled (mm). To ensure that the brush unit remained as central as possible within the pipe an aluminium support with a machined shoulder was used, see photograph in Figure 6.3. The support was placed inside the brush core with the shoulder butted up against the core end face. The central axis of the shoulder was drilled and tapped M16. Thus, the support was attached to M16 studding. The opposite end of the studding was screwed into a steel shaft which ran inside an axial bearing housing on the compression testing machine. The bearing housing ensured that the shoulder pushed the brush as centrally as possible, whilst the bearings helped to reduce any drag present. The bearing housing was integrated into the cross-brace support of the compression testing machine. This assembly was further connected to a "proving" ring. Prior to beginning the experiments both the proving ring and the displacement transducer were calibrated. The ring used for testing the polymeric brush units was a 1000lb (453.6Kg) ring (No. 3262 - transducer: 0.00089kN/mv). The ring used for testing the steel brush unit was a 5000lb (2268Kg) ring (No. 2990 - transducer: 0.0041995kN/mv).

The operation of the test rig was straight forward, a typical set of experiments would take the following form:

After the brush had been located into position, the rig was switched on. The lifting platform was then raised, this subsequently raised the pipe section which was located on the circular plate. As the brush touched the top of the pipe section, the brush and proving ring assembly continued to rise a short distance further, this allowed the "backlash" to be taken up. The top of the proving ring then contacted the large cast iron reaction support. It was at this point that the brush began to be inserted into the pipe. During the test the loading force was recorded, as the test continued and the brush became fully inserted into the pipe, the force required to push the brush forwards was recorded. The support plate of the compression testing machine was then lowered and the pipe was inverted, whilst the brush remained inside. The above experiment was then repeated, however, this time, the brush was forced to reverse. The transducer and proving ring signals were logged using the logging function of the computer software Triax, Toll (1993). Subsequently, the force exerted was plotted against the vertical displacement of the brush unit.

6.2.1 POLYMERIC BRISTLES

The forces required to load a brush into a pipe, move it forwards through the pipe and subsequently reverse it in the pipe were the main aims of these experiments. The ratio between forward "slip" and rear "grip" was also important. Of additional interest was the effect the R_a value, lubrication and creep had on traction. Of particular interest was whether the bristles have an ability to "cut" through the lubrication film in order to obtain a suitable purchase. As previously mentioned, nylon loses its stiffness, significantly, when subjected to water contamination. Thus, an interest was also taken into the magnitude of this loss of stiffness.

At the time testing occurred, polymeric bristles were the only bristles used on the sewer vehicles. In the main, this was due to the relatively "low" forces they were able to exert, specifically, in the forward direction. Thus, they were relatively easy to load into the sewer via a restricted manhole. Further, they were easy to reverse, they did not rust and were extremely compliant with their immediate environment, a factor which allowed significant obstacle negotiation. As mentioned above, their main weakness was their loss of stiffness when they were exposed to water. As most sewers contain water, this was a problem. The nylon family is particularly susceptible to this phenomenon. For example, nylon 6 can lose up to 70-80% of its stiffness, nylon 66 can lose up to 50-60% of its stiffness and nylon 612 can lose up to 30-40 % of its stiffness, Delbanco Mayer (1995). Polyester and polypropylene do not suffer from the same problem, although as bristle material they still remain unsuitable as their initial Modulus of Elasticity is low (P.P. is half that of nylon 6). Loss of stiffness of bristle material in a sewer may also be accelerated due to the warm and humid conditions. The sewer machine relies upon the bristles for both suspension and traction, thus, this was considered an important area to investigate.

There were two types of polymeric bristles used in the experiments, nylon 66 and nylon 612. Six different nominal diameters of bristle were used: 0.70mm, 1.1mm, 1.3mm, 1.5mm, 1.625mm and 2.0mm. All except the 1.625mm were nylon 66, the 1.625mm was nylon 612. The nylon 66 bristles were round or slightly oval in profile, the nylon 612 was a "star" shape in section. The brush core diameter remained constant at 50mm, with a length of 80mm. Total number of bristles can be calculated by multiplying No. of bristles/hole x No. holes per circumference (24) x No. holes per core length (8). The bristle hole diameter was constant at 4.8mm across the range of brushes. None of the polymeric bristled brush units incorporated the X-ply mechanism. In summary, the polymeric bristled brush specifications are given by Table 6.1.

Bristle Diameter	0.7mm	1.1mm	1.3mm	1.5mm	1.625mm	1.5mm
Brush core diameter (mm)	50	50	50	50	50	50
Head diameter (mm)	180	180	180	180	180	180
Angle increase from normal	0	0	0	0	0	0
No. of bristle rows	8	8	8	8	8	8
No. of holes/circumference	24	24	24	24	24	24
No. of bristles/hole	36	14	8/9	7	7	4

Table 6.1 Polymeric Bristled Brush Specifications.

6.2.2 STEEL BRISTLES

Further tests were undertaken on the steel brush units, of particular interest was the effect a brush design would have on the forward "slip" to rear "grip" ratio. Additional tests were undertaken to determine what benefit, if any, the X-ply brush units had over the non-X-ply brush units when subjected to reverse axial forces. Again, the R_a value and lubrication conditions were of significant importance.

A number of different steel bristled brush units were chosen for the experiment. They included; Two pre-used brush units which incorporated a 15 degree pre-sweep (termed *OLD 15A* and *OLD 15B* during the tests) which were acquired from Pipeline Integrity International (PII). They were originally used to carry out some preliminary tests in order to determine a number of forward and reverse forces. After the above brush units had been tested it was noted that there was a significant reduction in the force required to push the brush forwards through the pipe as well as the force the brush could support, in reverse, prior to failure. Two additional 15 degree pre-swept brush units were also obtained, prior to testing they were unused (termed *NEW 15A* and *NEW 15B* during the tests). Two further brush units were obtained from PII, these brushes had their bristles normal to the core (termed *NEW 0A* and *NEW 0B* during the tests). Another two brush units were manufactured by the University, they were both X-ply units, one incorporated a 15 degree pre-sweep, the other had bristles normal to the core (termed *NEW 15X and NEW 0X*). A summary of the technical data for the eight steel bristled brush units used for testing is detailed in Table 6.2, below.

Brush Type	OLD/NEW 15A+15B	NEW 0A+0B	NEW 0X	NEW 15X
Brush core diameter	85mm	86mm	84mm	86mm
Head diameter	178mm	184mm	184mm	182mm
Bristle diameter	0.193mm	0.193mm	0.193mm	0.193mm
Angle increase from 90 degrees	15 degrees	0 degrees	0 degrees	15 degrees
No. of bristle rows	7	7	8*	8*
No. of holes/circumference	27	27	27	27
No. of bristles/hole	550	550	550	550

Table 6.2 Steel Bristled Brush Specifications.

*NOTE: An additional bristle row is required on the X-ply brush units to ensure an equal number of rows with the +/-5 degree offset. If the number of rows were unequal then the brush unit would become "unbalanced" and a couple could more easily be induced into the core when the brush was subjected to reverse axial forces.

6.3 TEST CONSIDERATIONS

6.3.1 R_a VALUE

Only one pipe I.D. was used for the polymeric bristled brush tests, although three stages of corrosion or surface roughness were obtained. For the purpose of the tests, these were classified as *Smooth, Medium and Rough*. The internal surfaces of the three pipe sections were measured using a surface roughness measurement device (Tencor Instruments), the resultant R_a values were 10.56, 12.36 and 22.68 μm, respectively.

The steel bristled brush tests used four pipe sections with different ID's The R_a values were 38.84 (139mm I.D.), 38.84 (144mm I.D.), 38.84 (149mm I.D.) and 4.47 μm (155mm I.D.), respectively.

6.3.2 PIPE INTERNAL DIAMETER

The nylon bristles were tested in different pipes sections with the same internal diameter under different lubrication conditions, the I.D. of the pipe sections were, nominally, 160mm. The unrestrained head diameters of the brush units tested were, nominally, 180mm, after insertion into the pipe, the bristle formed an angle with the pipe wall of, approximately, 65.2 degrees, this was calculated by the spreadsheet.

Four different pipe ID's were used for testing the steel bristled brush units, they were; 139mm, 144mm, 149mm and 155mm. Table 6.3 details the bristle to pipe wall angles calculated by the spreadsheet (degrees) in the four pipe ID's:

Pipe ID	Brush with 15 degree sweep	Brush with 0 degree sweep	X-ply inc. 15 degree sweep	X-ply inc. 0 degree sweep
139mm	45.7	46.2	45.1	46.8
144mm	48.3	48.8	47.8	49.3
149mm	50.7	51.1	50.2	51.6
155mm	53.2	53.6	52.8	54

Table 6.3 Steel Bristled Brush Specifications.

6.3.3 TEMPERATURE

Temperature throughout the testing was held as constant as possible, it was approximately 20 degrees Celsius.

6.3.4 VISCOSITY

The viscosity of the various lubricants was determined. The lubricants used during the tests included; diesel, commercial light oil (Castrol GTX) and fuel oil. Samples of the lubricants were collected and sent away for analysis. To summarise, the results were as follows: diesel 4cst, light oil 104cst and fuel oil 975 cst (centi-stokes at 40 degrees Celcius). The polymeric bristled brush units were tested under dry, water, diesel and GTX (light oil) conditions, the steel was tested under dry, diesel, GTX (light oil) and fuel oil conditions.

6.3.5 GENERAL EXPERIMENTAL OBSERVATIONS

The reader should appreciate when reviewing the following tables and figures (graphs) that during each pass a brush unit makes through a pipe a small amount of the surface layer of the pipe (and bristle) will be removed, that is, the pipe becomes progressively smoother. Thus, it would not be unreasonable to note a slight decline in the forces recorded for a specific brush after it had completed a number of passes through the same pipe section. To help reduce this effect separate pipe sections were used for both the polymer and steel brush units.

After a brush had passed through a lubricated pipe section in the forward direction, the effect of the bristles "ploughing" through the lubrication layer could be clearly seen. The internal wall of the pipe showed heavy "striping".

During the reverse testing of the steel bristled brush units it became necessary to bolt down the pipe sections to the lifting plate to prevent the pipe being pushed off centre. Further, by maintaining the brush unit central within the pipe it was possible to increase the axial force the brush was able to support prior to collapse.

Bolton (1989) quotes typical values of μ_s (coefficient of static friction) for various combinations of materials as;

Steel on steel	0.8
Mild steel on mild steel	0.5
Nylon on nylon	0.3
Nylon on steel	0.2

6.4 LOADING, FORWARD FORCE & REVERSAL

It is proposed that prior to discussing the results in detail, a brief description of the types of graph commonly recorded and the typical modes of brush failure, whilst in reverse "grip" mode, are discussed. A small number of graphs are included which are typical examples of the results recorded.

The graph (16LPFDS1.XLS) in Figure 6.4 is a typical example of a nylon 612 bristled brush being loaded and pushed forwards through a smooth, dry pipe. The initial positive gradient shown on the graph is the effect of the first bank of bristles contacting the rim of the pipe in which it is to be inserted. As the bristles begin to deflect rearwards the force begins to build, hence, the initial positive gradient. The point at which the gradient reaches the first peak and then produces a slight negative gradient is the point at which the first bank of bristles has entered the pipe. Thus, the force temporarily drops off. Subsequently, a second set of bristles begin to contact the rim, this registers as the second positive gradient.

Overall, the force continues to rise, as more bristles contact the edge of the pipe and subsequently enter into the pipe. A point is reached (approximately, after 70mm of displacement) where a number of the banks of bristles are in the pipe but the overall trend of the gradient turns negative. This is the balance point at which most of the bristle banks have entered into the pipe, subsequently, there is less interference from bristles that have yet to enter the pipe. The photograph in Figure 6.5 illustrates this point. The graph clearly shows the remaining three banks being pushed into the pipe. At this point the brush is pushed forwards through the pipe a short distance as indicated by the levelling off of the graph, the photograph in Figure 6.6 shows an example of this. The graph (N0XLD55.XLS) in Figure 6.7 shows the same load and push forward experiment for a steel brush unit. In this instance the same graph was also used to collect the data for the forward force experiment.

The graph (16FCVDS1.XLS) in Figure 6.8 shows the trace of a nylon 612 brush pre-inserted into a smooth, dry pipe and being subjected to a forward force. The force builds rapidly up to the point the brush begins to slide forward through the pipe. Once the initial friction has been overcome and the brush begins to move, the force declines slightly. Subsequently, the force climbs again and then begins to stabilise.

The graph (16REVDS1.XLS) in Figure 6.9 shows a nylon 612 brush unit being subjected to a reverse force experiment in a smooth, dry pipe. In this graph the force climbs rapidly and within 3mm of axial displacement the brush has reached the maximum load it is able to support prior to collapse, this position is illustrated by the photograph in Figure 6.10. At this point the load begins to decline and it was noted that when this occurred the bristles had moved out of plane and were translated from an axial orientation to align themselves with the transverse plane. This position is illustrated by the photograph in Figure 6.11. This occurs at approximately 38mm of travel. At this point, the force the bristles are able to resist axially is zero. The bristles still retain stored energy and as the testing machine continued to push the core rearwards a point was reached where the bristles were suddenly able to release their stored energy. At this point the brush unit rapidly sprang forward. The bristles were then pointing in the opposite orientation to which they started, that is, they had changed direction by 180 degrees.

The graph (16REVXM1.XLS) in Figure 6.12 shows a nylon 612 brush being subjected to a reverse force experiment in the medium roughness pipe with light oil (Castrol GTX) lubrication. The graph shows that, after reaching a peak reversal force, the forward force trace does not behave in a manner similar to the case described above. Instead, another type of mechanism occurred in which the brush was not retained centrally within the pipe. When this occurs, the bristles on one side of the core grip the wall and exert a higher radial force than bristles that are beginning to slip on the opposite side of the core. As the core experiences the axial reverse force, the brush is forced to move further off-centre due to the influence of the unequal bristle forces. Subsequently, the bristles experiencing the largest deflection begin to collapse as the bristles on the opposite side of the core begin to straighten. Thus, an unstable situation occurs and the brush core moves more easily off the centre axis. As a consequence, some bristles begin to deflect further, whilst the bristles on the opposite side of the core straighten and become detached from the pipe wall, the photograph in Figure 6.13 illustrates this phenomenon. If a lubricant is present, the bristles become less able to retain an adequate foothold and become more susceptible to slippage. Hence, in summary, if conditions are favourable there is a tendency for the brush core to move off the centre axis of the pipe and for some bristles to simply slide forward whilst others simply "flip" through. Subsequently, the tilted brush moves down the pipe against a frictional resistance.

The graph (N0XRD55.XLS) in Figure 6.14 shows a *NEW 0X* steel brush unit undergoing a reverse force experiment in a 155mm diameter dry pipe. Here, a slight reaction couple was induced at the core and considerable bristle slippage occurred. This slippage is thought to be due to the action of the core rotation caused by the reaction couple. As the core begins to rotate, even slightly, the "twisting" action induces the bristles to move out of plane. The deflection of the bristles has two effects. One effect is the reduction of the perpendicular force at the bristle tips which consequently reduces the frictional force. A second effect is the generation of a moment arm acting on the bristle about its location on the brush core. This moment subjects the bristle to a combination of axial thrust and torsion. The bristles are long and slender and thus have little resistance to torsion. This means that the force at the bristle tip is easily able to rotate the bristle about its axis at the core and thus the bristle provides little resistance to the axial movement of the core. To summarise, the graph shows that the force builds rapidly and begins to slowly decline. In this case although all of the bristles remained in the "reverse" direction, they slipped over the pipe wall, sometimes obtaining a "foothold" but losing it again when the force built up once again.

6.5 POLYMERIC BRISTLE RESULTS ANALYSIS

Discussed below are the main findings from the above experiments and included are the complete table of results for each brush unit tested, that is, the loading force, the forward force and the reversal force, see figures 6.15 to 6.20.

The tables are split into three sections. The first section deals with the loading force, the second section deals with the forward force and the third section deals with the reverse force. There are six separate tables for the polymeric bristles, that is, an individual table for the 0.7mm diameter bristles through to the 2.0mm diameter bristles. Each section is split into rows, each row contains the details of an experiment performed on the brush under different surface roughness and lubrication conditions. The columns represent a number of important variables. The first of these variables is the R_a value of the pipe sections, as recorded at the beginning of the tests. The *"P MAX"* value is the maximum force in any particular experiment. For example, in the loading force experiment it is the force required to load a brush into a pipe [Figure 6.4]. In the forward force experiments P MAX provides the maximum force required to push a brush forwards through a pipe, this was generally the force required to overcome limiting friction [Figure 6.8]. In the reverse force experiments P MAX provides the maximum axial force a brush can support prior to failure whilst in the reverse force mode [Figure 6.9]. *"P MIN"* is the minimum force recorded whilst the brush is pushed forwards through a pipe [Figure 6.8] and *"P AV."* is the average force required to push a brush forwards through a pipe [Figure 6.8]. The column detailing the temperature is self explanatory. The column designated *"VIS. cSt"*. is the viscosity of the lubricant used during the test, as previously described in 6.3.4. The *"COMMENTS"* column details any observations made during the experiments. This was primarily used for recording the mode of brush failure during reverse force experiments. Within the tables, values are provided for μ. These values are calculated by cross-referencing the actual forward force recorded (P max, P min and P Av.) during the experiment with data from the theoretical model spreadsheet for that particular brush design. The *"ANGLE"* column is the angle the bristle forms with the pipe wall, as calculated from the theoretical spreadsheet. The *"ROTATION"* column is the Direction of Rotation (DOR) that the brush core experienced. The column headed *"RATIO"* is the ratio of P Av. (forward force) and P MAX (reverse force).

6.5.1 DISCUSSION OF LOADING FORCE

If the forces required to load the polymeric brush units are reviewed [Figures 6.15 through 6.20] it is evident that, irrespective of bristle material type or bristle diameter, it requires a slightly higher force to load the brush unit under dry conditions than under "lubricated" conditions. However, there is no obvious decrease in force as the viscosity of the lubricant increases. That is, the force required to load a brush into a pipe, which has water lubrication present is similar to the force required to load a brush in a pipe that has either diesel or light oil lubrication present. To summarise, it can be said that any lubricant will decrease the force required to load a brush unit into a pipe but only by a small amount.

6.5.2 DISCUSSION OF FORWARD FORCE

As discussed above, three separate sets of forces were recorded during the forward force experiments, the "Maximum Force", the "Minimum Force" and the "Average Force". The average force is the average of all of the individual forces logged throughout the test. Unlike the loading force experiments, the presence of lubrication whilst a brush is being pushed forwards through a pipe does make a notable difference to the forward force. If water or diesel lubrication are present, a small decrease in the force required to push the brush forwards occurs. If light oil lubrication is present, there is a more noticeable decrease in force, an example being the forward forces recorded for the nylon 612 brush unit [Figure 6.19]. Here, under dry conditions, the smooth, medium and rough pipes gave forces of 79.5N, 131.5N and 140.3N, respectively. Under light oil lubrication the forces recorded were, 56.9N, 65.7N and 93.2N, respectively. The effect the surface roughness of the pipe has on the forward force is also apparent, this will be discussed in further detail in the following paragraphs.

Referring to Figure 6.15, the reader will notice that if the 0.7mm diameter bristles are exposed to water the force can decrease significantly, even when compared to the diesel or light oil. It is considered that this may be due to the small cross-sectional area of the bristles soaking up water and subsequently losing their stiffness, again, this will be dealt with in more detail later.

The μ values shown in Figure 6.15 (0.7mm bristles) for a brush being pushed through a dry pipe range from 0.16 to 0.25, for water they range from 0.05 to 0.1, for diesel they range from 0.09 to 0.2 and for light oil they range between 0.04 and 0.19. Thus, the dry pipe returns the smallest spread of μ values, the greatest spread occurs if the light oil lubrication is present. The μ values can also be seen to increase as the R_a values increase, which would be expected.

6.5.3 DISCUSSION OF REVERSE FORCE

Of significant interest was the fact that lubrication conditions, in general, have little effect upon reverse "grip" forces. If Figures 6.15 to 6.20 are reviewed it will be noted that there is a very slight decrease in the rear "grip" force for the lubricated conditions, compared with the dry conditions. In some instances a brush subjected to lubrication has been able to support a greater reverse force prior to collapsing than the same brush under dry conditions. For example, in Figure 6.17(nylon 66, 1.3mm diameter bristles) it can be seen that if the brush is placed in a smooth, dry pipe the peak reversal force is 652.4N prior to failure. The same brush in the same pipe when lubricated with light oil was able to support 800.5N prior to failure. There are a number of other examples of this phenomenon, including, water and diesel lubrication conditions. Further examples can be readily found, for example, Figure 6.20. Here, a nylon 66 bristled brush (2.0mm diameter bristles), if placed in a rough, dry pipe is able to support 2129.8N prior to failure, however, the same brush placed in the same pipe under diesel lubrication conditions was able to support 2150.4N prior to failure.

After considering the above it was concluded that lubrication had little effect. However, as evident from the tables, the results are well distributed, that is, the results are significantly scattered. The scatter is considered to be influenced by the unstable nature of the brush mechanism if subjected to a reverse axial force.

It will be noted that the 0.7mm bristled brush was able to reverse in the pipe by rotating its core and moving its bristles out of plane, see "comments", Figure 6.15. This case was also true for the 1.1mm diameter bristles (Figure 6.16). However, as the bristle diameter increased to 1.3mm, core offset and bristle slippage began to occur. Where the R_a value remains high ($22.68\,\mu m$) or the pipe wall remains dry, then the brush is still able to reverse by core rotation and the bristle "flipping" or moving out of plane. The 1.5mm bristled brush, 1.625mm bristled brush and 2.0mm bristled brush are all able to reverse by this method when the pipe remains un-lubricated or the R_a value (surface roughness) remains high, for example $22.68\,\mu m$. However, as soon as lubrication of any type is introduced slight core rotation occurs followed by the core becoming offset, subsequently followed by bristle slippage.

6.5.4 LUBRICATION EFFECT

As briefly discussed above, another factor which plays a significant role in determining the force required to overcome the static friction in the forward force direction is lubrication. The appended graph, (LUB-15F.XLS) in Figure 6.21, shows four forward force experiments under dry, water, diesel and light oil conditions. The brush used the 1.5mm diameter nylon 66 bristles and operated in the medium roughness pipe (R_a = 12.36 μm). In can be seen that the location of each lubrication condition appears on the graph as one would expect, that is, dry conditions produce the highest forward force, followed by water, diesel and light oil. The pipe ID remained constant for all four experiments.

One of the most important questions that arises from these experiments, irrespective of reverse failure mode is whether lubrication is detrimental, or not, to the ability of a brush to obtain a suitable purchase, whilst in reverse or "grip" mode. As previously mentioned, during the forward force experiments the force required to push the brush through the pipe decreases if lubrication is present. However, the axial force a brush is able to support whilst in reverse or "grip" mode remains similar to that observed in a dry pipe. As such, very high ratios between forward "slip" and rear "grip" are possible under lubricated conditions, in most cases significantly higher ratios than seen in dry pipes. This may allow a similar or higher payload to be transported through a lubricated pipeline.

During the tests conducted with diesel oil lubrication, the forward force results produced many "spikes" on the graph, it is considered that diesel may induce a type of "stick-slip" effect. That is, boundaries of high and low lubrication (Figure 6.21).

6.5.5 LUBRICATION: EFFECT ON RATIOS

Consider the force a brush is able to support in reverse or "grip" mode whilst subjected to water, diesel or light oil lubrication. Table 6.4, below, details the maximum force a brush was able to support prior to failure.

PIPE CONDITION	BRISTLE DIAMETER					
	0.7mm	1.1mm	1.3mm	1.5mm	1.625mm	2.0mm
S1 DRY	221.7N	430.7N	652.4N	735.8N	536.6N	1180.1N
M1 DRY	256N	451.3N	843.7N	795.6N	583.7N	1500N
R1 DRY	220.7N	437.5N	809.3N	887.8N	573.9N	2129.8N
S1 H2O	204.1N	395.3N	712.2N	676.9N	395.3N	467N*
M1 H2O	206N	273.7N	679.8N	862.3N	400.3N	970.2N*
R1 H2O	223.7N	334.5N	905.5N	734.8N	391.4N	1490.1N
S1 DIESEL	183.5N	294.3N	719.1N	660.2N	500.3N	1089.9N*
M1 DIESEL	206N	325.7N	684.7N	606.3N	488.5N	1089.9N*
R1 DIESEL	200.1N	380.6N	836.8N	755.4N	492.5N	2150.4N
S1 GTX	185.4N	439.5N	800.5N	781.9N	400.2N	759.3N*
M1 GTX	188.4N	349.2N	586.6N	651.4N	355.1N	825N*
R1 GTX	225.6N	421.8N	544.5N	812.3N	364.9N	1650N*

Table 6.4 Maximum Reverse Force Supported Prior To Failure.

NOTES: *Tests halted due to bristle slippage and core offset. S1, M1 and R1 designate surface roughness, that is, SMOOTH, MEDIUM AND ROUGH.

It can be seen that lubrication causes no significant reduction in the force a brush can withstand prior to failure, irrespective of the type of lubrication present. The forces in the forward direction are however lowered [Figure 6.21], thus, the forward "slip" to rear "grip" ratio increases. This is an extremely important finding, it gives increased confidence to companies such as PII. PII may use these vehicles in pipelines, which, although purged with Nitrogen and "clean" may still contain pockets of product. In most cases this product will have "lubricant" properties. One condition under which the brush units were not tested was that of wax residue. Wax can block pipelines completely if left to build up or the temperature of the oil within the pipeline decreases. As a comparison this wax can be as hard as household candle wax.

It is worth noting that the 1.625mm (nylon 612) bristled brush is unable to support as high a reverse force as the 1.3 and 1.5mm diameter bristled brushes (nylon 66). This may be due to a number of factors. The Modulus of Elasticity for the 1.625mm diameter bristles is approximately one third lower than for the nylon 66. Further, the cross-sectional area of the bristle is smaller than the 1.625mm diameter designation suggests, due to "star" shape sections of material having being "removed". The results for the 1.625mm diameter bristled brush are comparable to the 1.1mm brush.

Using the above reverse forces and the forward forces, consideration can now be given to the "slip" to "grip" ratios when the brush is exposed to dry, water, diesel and light oil conditions. These ratios are given in Table 6.5, below.

PIPE CONDITION	BRISTLE DIAMETER					
	0.7mm	1.1mm	1.3mm	1.5mm	1.625mm	2.0mm
S1 DRY	1:7.29	1:6.75	1:6.16	1:5.1	1:6.75	1:6.91
M1 DRY	1:7.45	1:4.74	1:5.48	1:4.56	1:4.44	1:5.9
R1 DRY	1:5.23	1:4.55	1:5.36	1:4.84	1:4.09	1:8.16
S1 H2O	1:14.9	1:6.2	1:6.85	1:5.48	1:4.43	1:2.29*
M1 H2O	1:14	1:3.72	1:5.68	1:5.71	1:4.58	1:4.64*
R1 H2O	1:15.2	1:3.88	1:7.5	1:4.8	1:3.17	1:6.58
S1 DIESEL	1:9.84	1:4.55	1:7.33	1:5.7	1:5.05	1:6.07*
M1 DIESEL	1:8.08	1:5.03	1:6.23	1:4.61	1:5.86	1:5.22*
R1 DIESEL	1:6.18	1:5.39	1:7.23	1:5.83	1:4.23	1:9.21
S1 GTX	1:10.5	1:9.33	1:11.3	1:7.66	1:7.04	1:5*
M1 GTX	1:19.2	1:7.91	1:6.23	1:6.85	1:5.4	1:4.81*
R1 GTX	1:7.67	1:5.89	1:4.48	1:5.79	1:3.92	1:8.41*

Table 6.5 Forward "Slip" to Rear "Grip" Ratio.

*Tests halted due to bristle slippage and core offset. S1, M1 and R1 designate surface roughness, that is, SMOOTH, MEDIUM AND ROUGH.

As illustrated by Table 6.5 the forward "slip" to reverse "grip" ratios, under lubricated conditions can increase.

6.5.6 R_a EFFECT

The R_a value effects both the forward "slip" and reverse "grip" capability of a brush unit. The force required to push a brush forward through a pipe is raised as the surface roughness increases. If a brush unit is used in the reverse or "grip" mode an increased surface roughness allows the bristle tips to obtain a more secure purchase. The graph, (RA-FCV07.xls) in Figure 6.22, illustrates the principle. In reverse, the axial force a brush unit can withstand prior to failure increases as the R_a value increases. The reverse experiment on the 2.0mm bristle diameter, nylon 66, brush unit shows that if the pipe conditions are suitable it is possible to increase the reverse axial force such a brush is able to support prior to failure. The graph (RA-REV20.XLS) in Figure 6.23 illustrates the principle. The graph in Figure 6.24 illustrates the relationship between surface roughness and the friction coefficient. The friction coefficient can be seen to rise as the roughness of the pipe wall increases. Further, the graph also shows that the friction coefficient decreases as the viscosity of the lubrication increases.

6.5.7 CREEP

Creep or stress relaxation of polymeric bristle material is of particular concern, especially if a brush remains loaded in a pipe for any length of time. An experiment was conducted under dry conditions to determine this effect. The experiment involved the loading and pushing forward of 0.7, 1.1, 1.5 and 2.0mm diameter bristled brushes into a pipe. After the brush unit had been loaded and had completed, approximately, 10mm of forward travel the testing machine was stopped and the rig was left for one hour. The rig was subsequently re-started and the results recorded, see Figures 6.25 to 6.28. Figure 6.29 shows the creep effect on a 2.00mm diameter bristled brush after 20 hours in a pipe. The difference between the initial forward travel of the brush and the forward travel after an hour was determined and a reduction in axial force was calculated as a percentage. These tests were conducted at a constant temperature of 16 degrees Celsius. Figure 6.30 shows the percentage reduction in force due to creep over a period of one hour for different bristle diameters. The nylon 612 (1.625mm diameter) bristles were further tested over a 16 hour period. It was noted that the total reduction in force due to creep occurred within the first 16 hours, the graph (16HR1625.XLS) in Figure 6.31 illustrates the experiment.

Note: the majority of sewer investigations would be completed within 1 hour or less. However, the reduction in axial force, due to creep, must also be considered with respect to further loss of bristle stiffness due to water contamination. This combination may result in significantly weakened brushes, thus, a reduction in vehicle performance may occur.

6.5.8 POLYMERIC BRISTLES MOVING IN POLYMERIC PIPE

Practical experience has shown that certain bristle materials work better with certain pipe materials. For example, steel bristles work well on steel pipes and polymeric bristles also work well on steel pipes. However, the problem of slip can occur if polymeric bristles are used in polymeric pipelines.

To indicate how catastrophic the results of choosing inappropriate bristle /pipe materials could be, a brief experiment was carried out using nylon 66 and nylon 612 bristled brushes traversing a UPVc pipe. The brushes were subjected to a forward force experiment and a reverse force experiment, these results were then reviewed as a forward "slip" to rear "grip" ratio. Figures 6.32 to 6.37 detail the results. The ratio is illustrated in table 6.6 below.

BRISTLE TYPE/DIAMETER	RATIO
NYLON 66 0.7mm	1:1.3
NYLON 66 1.1mm	1:0.875
NYLON 66 1.3mm	1:1.3
NYLON 66 1.5mm	1:0.75
NYLON 612 1.625mm	1:2.2
NYLON 66 2.0mm	1:1.15

Table 6.6 "Slip" to "Grip" Ratio for Polymeric Bristles Whilst Inside UPVc Pipe.

The results of the nylon 66 1.1mm and the nylon 66 1.5mm are of particular interest. Both results show that the force required to move the brush unit forwards through the pipe, that is, the force required to overcome friction, is higher than the force required to "reverse" the brush. Therefore, the ratio between forward "slip" and rear "grip" is less than 1:1. Thus, it can be assumed that if the vehicle was loaded into the pipe it would be more likely to slide rearwards, than travel forwards, even though the bristles would be pointing in the "forward" direction. It can be seen that if inappropriate materials are chosen for the bristles and pipe, then traction may be seriously affected.

6.5.9 WATER CONTAMINATION

Loss of stiffness due to water contamination appeared to be evident during the dry and then subsequent water lubrication testing of the 0.7mm nylon 66. The appended graph (07FCVH20.XLS), in Figure 6.38 shows the dry tests at the top, in blue, the water tests are below in red (R_a value 10.5), blue (R_a value 12.36) and green (R_a value 22.68). If Figures 6.16 through 6.20 are reviewed it can be seen that for all other bristle diameters a reduction in forward force occurs if the bristles are subjected to water lubrication/contamination. However, the reduction is not as great as it is for the 0.7mm diameter bristles. The test was re-performed after the brush and pipe were thoroughly dried, the second experiment gave the same results as the first. It is interesting to note that the reverse forces available, if water lubrication is present, remain high. Thus, forward "slip" to reverse "grip" ratios as high as 1:15 have been recorded. It is thought that the 0.7mm diameter bristles are affected to a larger extent due to their smaller cross-sectional area, thus, they lose their stiffness quicker due to an accelerated water absorption effect.

6.6 STEEL BRISTLE RESULTS ANALYSIS

In section 6.4, the overall shape of the graphs for polymeric bristles under loading, forward and reverse forces were discussed. These generic graphs are the same for the experimental results for steel bristled brushes. However, with the steel brushes, the loading and forward force experiments were integrated and the results appear on the same graph. The main findings from the above experiments are discussed below. The complete table of results for each brush unit tested are also included, that is, the loading force, the forward force and the reversal force, see Figures 6.39 to 6.46.

The tables are split into three sections. The first section deals with the loading force. Section two deals with the forward force and section three considers the reverse force. There are eight tables for the steel bristled brushes. A description of each brush tested has been provided in section 6.2.2. Each section is split into rows, each row contains the details of an experiment performed on the brush under different surface roughness and lubrication conditions. The columns represent a number of important variables and are the same as those used for the polymeric bristled tests except the "PI LOAD" is the peak force observed whilst loading the brush. Also "PR LOAD" is the maximum force a brush was able to support prior to failure when subjected to a reverse force experiment.

The steel bristled brushes were tested under dry, diesel, Castrol GTX (light oil) and fuel oil lubrication conditions. The lubricant viscosities were recorded as; 4, 104 and 975 centistokes, respectively. The R_a value of the pipes were recorded as $4.47\mu m$ for the 155mm ID pipe section and $38.84\mu m$ for the 149, 144 and 139mm ID pipe sections. The diameter of the steel bristles was consistent at 0.2mm throughout all of the experiments.

6.6.1 DISCUSSION OF LOADING AND FORWARD FORCE

An experiment was undertaken to determine what effect the change in internal diameter would have on the force required to load a steel bristled brush. As previously mentioned four pipe ID's were used, they were 155mm, 149mm, 144mm and 139mm. The graph in Figure 6.47 shows the maximum loading force for a radial steel bristled brush (NEW0A), this is a typical example of the shape of the graph obtained during the loading experiments, irrespective of brush design or lubrication present. It can be seen that the force required to load the steel brush depends upon the ID of the pipe. That is, a higher loading force is required to load the brush into the 139mm ID pipe than is required to load the brush into the 144mm ID pipe and so on. The graph clearly shows that the force required to load the steel bristled brush into the 139mm ID pipe was over double that required to load the same brush into the 155mm ID pipe.

The graph in Figure 6.48 shows the same steel bristled brush, described above, loaded into a 155mm ID pipe under dry, diesel, light oil and fuel oil lubrication conditions. It can be seen that if lubrication is present there is a very slight difference in the force required to load the brush into the pipe. However, it can be seen from the graph that under diesel lubrication conditions it requires, approximately, 0.2kN additional force to load the brush. However, under dry, light oil and fuel oil lubrication conditions the force to load the brush remains the same. If the other steel bristled brush units are loaded, they produce similar graphs, that is, lubrication makes only a very slight difference to the force required to load a brush into a pipe. This is considered to be due to the fact that as each circumferential row of bristles enters into the pipe, then, the force required to push those bristles forward through the pipe decreases, slightly, due to the lubrication.

A further brush loading experiment was undertaken to determine the difference in force required to load a radial, steel bristled brush (NEW0A) compared to the force required to load a steel bristled brush with 15 degree pre-swept bristles (NEW15A). Both brushes were compared being loaded into the same pipe. The graph in Figure 6.49 illustrates the results. The reader should compare the upper red trace with the lower red trace and so on. The lower red trace ("15/155") is the plot of the steel bristled brush with the 15 degree pre-swept bristles being inserted into the 155mm ID pipe. The upper red trace ("0/155") is the plot of the radially bristled brush being inserted into the 155mm ID pipe. If the reader then reviews the blue trace (149mm ID pipe), the pink trace

(144mm ID pipe) and the green trace (139mm ID pipe) the difference in force required to load a radial brush compared to a pre-swept brush can be observed. It is clearly evident that the force required to load a steel bristled radial brush is considerably higher than the force required to load a steel bristled brush with bristles pre-swept at 15 degrees.

An experiment was conducted to determine whether lubrication would, generally, reduce the forward force required to push a brush through a pipe. This experiment used the 139mm ID pipe which had an average R_a value of 38.84 microns. Further, it used a steel brush unit with bristles pre-swept at 15 degrees (NEW15A). The graph (FCV-LUB0.XLS) in Figure 6.50 illustrates the results. If the right hand side of the graph is reviewed, it can be seen that if lubrication is present in the pipe then there is a clear reduction in the force required to push the brush forward through the pipe. Further, as the viscosity of the lubricant increases, the forward force decreases. It is considered that this phenomenon may be due to a hydrodynamic lubrication effect occurring between the bristle tip and pipe wall as the brush is pushed forwards through the pipe. This is not unreasonable when the high pressures generated at the bristle tips are taken into account. Further, as the viscosity of the lubricant increases, the less chance there is of the lubricant being forced out of the crevices of the pipe wall when being subjected to the high pressures exerted by the bristle tips. The slight differences in force required to load the brush into the pipe when lubrication is present can also be clearly seen.

An experiment was conducted to determine whether during the forward force experiments a reduction in force occurred if the bristles were pre-swept as opposed to radial. Two brushes were chosen for comparison, they were the radial X-ply (NEW0X) and the 15 degree pre-swept X-ply (NEW15X). The brushes were compared under fuel oil lubrication. The graph in Figure 6.51 shows the results. The red trace showing the largest forward force is the radial brush being pushed forwards through a 139mm ID pipe, the lower red trace is the pre-swept brush being pushed forwards through the 139mm ID pipe. The blue traces represent the 144mm ID pipe, the green traces represent the 149mm ID pipe and the pink traces represent the 155mm ID pipe. As with the red trace, it is the radial bristled brushes which require the largest force to move forward through the pipe. The lower trace represents the pre-swept brush. It can be concluded that a higher force is required to push a radial bristled brush forward through a pipe than to push a pre-swept bristled brush through the same pipe. This is the case for all other types of lubrication condition as well as dry conditions.

6.6.2 DISCUSSION OF REVERSE FORCES

An experiment was undertaken to determine whether a steel brush unit could support the same reverse force under lubricated conditions as it could under dry conditions. For this experiment the pipe was tested under dry, diesel oil, light oil and fuel oil lubrication. The pipe ID was 139mm and its R_a value was 38.84 microns. The steel brush used had a 15 degree pre-sweep (OLD15A). This experiment was particularly important as it had already been determined that the polymeric bristles had an ability to "cut-through" the lubrication film and secure a foothold under similar conditions.

The graph (LUB-REV.XLS) in Figure 6.52 shows the results recorded for the four different pipe lubrication conditions. It can be seen that the dry condition produces a predictably high reverse force prior to failure. However, the diesel oil, light oil and fuel oil lubrication all affect the ability of the brush to obtain a suitable purchase. Of interest is the fact that under dry conditions the mode of brush failure was determined by the core becoming offset in the pipe followed by bristle slippage, as described in section 6.4. However, under lubricated conditions the mode of failure switches to significant bristle slippage and/or total bristle slippage, both cases experience a slight, initial, brush core rotation. Thus, it can be seen that lubrication present within a pipe will significantly affect the reverse axial force a steel bristled brush can support prior to failure.

Leading on from the above experiment, a further experiment was conducted to determine what benefit, if any, X-plying either a radial or pre-swept brush would have in terms of the brush being able to support higher axial reverse forces prior to failing. The peak reverse forces for a radial brush, a radial X-ply brush, a 15 degree pre-swept brush and a 15 degree pre-swept X-ply brush were recorded, the results are shown in Table 6.7.

PIPE CONDITION	BRUSH TYPE			
	RADIAL (NEW0A)	15 DEGREE PRE-SWEEP (NEW15A)	RADIAL X-PLY (NEW0X)	X-PLY 15 DEGREE PRE-SWEEP (NEW 15X)
149mm DRY	6517N	5644N	12475N	4549N
144mm DRY	7654N	4683N	10889N	4134N
139mm DRY	7841N	6792N	11576N	8769N
149mm DIESEL	4413N	3333N	5064N	3565N
144mm DIESEL	4110N	4000N	4209N	4082N
139mm DIESEL	3682N	2074N	4138N	2934N
149mm GTX	3347N	4378N	4188N	3671N
144mm GTX	2119N	2031N	3809N	3731N
139mm GTX	2536N	1570N	2638N	2025N
149mm F. OIL	3363N	2604N	4587N	3050N
144mm F. OIL	1607N	1424N	2094N	1725N
139mm F. OIL	1673N	875N	1639N	1380N

Table 6.7 Maximum Reverse Force Under Different Lubrication Conditions.

The graph in Figure 6.53 compares a radial brush with a radial X-ply brush. It can be seen that under lubricated conditions the X-ply makes no significant difference. However, if conditions are dry, then the X-ply mechanism has a significant effect on the reverse force a brush can support prior to failure. The graph in Figure 6.54 compares a 15 degree pre-swept brush with a 15 degree pre-swept X-ply brush. It can be seen that there appears to be no benefit in using the X-ply design if the brush is also pre-swept. This is true for both dry and lubricated conditions.

The graph (X-PLYREV.XLS) in Figure 6.55 compares the "dry" results for the radial steel bristled brush and the radial X-ply brush unit whilst subject to reverse force experiments. The differences are significant with peak reversing loads being significantly higher for the radial X-ply brush than for the radial brush. The 139mm, 144mm and 149mm ID pipes were used for this experiment, the R_a value remained constant at 38.84 microns. The radial X-ply unit can be seen, under dry conditions, to be able to improve the reverse force support capability over the radial brush unit by a significant amount. However, this benefit does not appear to exist if lubrication is present. Here, the reverse forces for the X-ply brush unit are lower than for the radial brush unit. The radial X-ply brush was able, under dry conditions, to support a maximum recorded load of 12,475N (1.271 Tonnes, approximately), a considerable achievement. Reverse failure occurred in the form of "violent" bristle slippage and brush core offset, that is, the brush core moved off the centre axis of the pipe.

As will be noted from the results above, the 15 degree pre-swept X-ply brush unit did not provide the desired results. Upon analysis, failure was consistently due to premature brush core rotation, most interestingly the DOR was consistently ACW. This may indicate a geometric machining fault, that is, the bristle holes may have been machined into the core at an incorrect offset/angle. Therefore, it must be recommended that further experiments should be performed on a re-manufactured version of the brush.

6.6.3 LUBRICATION AND PIPE DIAMETER

An experiment was conducted to determine whether the ID of the pipe affects the ability of the brush to support reverse forces whilst under lubricated conditions. During these experiments the R_a value was consistent at 38.38 microns. The brush used was of the radial type (NEW0X). The graph (DIESEL.XLS) in Figure 6.56 shows the results. It can be seen that under diesel oil lubrication, as the pipe ID decreases then so does the ability of the brush to grip the pipe wall. The graph (GTX.XLS) in Figure 6.57 shows a similar trend if the brush is subject to light oil lubrication. Again, it can be seen that as the pipe ID decreases the brush's ability to support high reverse forces also decreases. The graph (FUEL-OIL.XLS) in Figure 6.58 again shows the trend detailed by the two graphs described above. However, what is of particular interest is the fact that as the viscosity of the lubricant increases the ability of the brush to grip also decreases [see the 139mm and 144mm ID traces in Figures 6.56, 6.57 and 6.58].

Again, this is thought to be due to hydrodynamic lubrication conditions occurring at the bristle tip/pipe wall interface, as previously described in section 6.61.

Again, the reader should note the spiked trace of the diesel lubrication conditions in Figures 6.56.

6.6.4 USED BRUSHES v NEW BRUSHES

As previously explained two brushes with 15 degree pre-swept bristles (OLD15A and OLD15B) were obtained from PII and used in some preliminary tests. Subsequently, they were subject to considerable forward and reverse force "tests". A further experiment was undertaken to determine whether there was any significant difference between the "used" and "un-used" brushes, Specifically, in terms of the force required to push them forward through a pipe and the reverse force they could support prior to failure. Of interest was whether the "used" brushes would produce lower forces in the reverse direction than the "un-used" brush units. The results of the forward and reverse force experiments are detailed below in Table 6.8;

PIPE CONDITION	REVERSE FORCE			
	OLD15 DEGREE RADIAL: FORWARD	NEW15 DEGREE RADIAL: FORWARD	OLD15 DEGREE RADIAL: REVERSE	NEW15 DEGREE RADIAL: REVERSE
155mm DRY	599.4N	567N	2537.9N	2195.5N
149mm DRY	916.3N	907.4N	3950.5N	5643.7N
144mm DRY	968.3N	1074.2N	4261.5N	4683.3N
139mm DRY	1049.7N	981N	6414.8N	6791.5N
155mm DIESEL	525.8N	540.5N	1787.4N	2242.6N
149mm DIESEL	660.2N	721N	3768N	3333.4N
144mm DIESEL	812.3N	884.9N	3395.2N	3999.5N
139mm DIESEL	885.8N	914.3N	2686N	2073.8N
155mm GTX	511.1N	451.3N	2359.3N	1649.1N
149mm GTX	618N	702.4N	3576.7N	4379.2N
144mm GTX	737.7N	737.1N	2591.8N	2030.7N
139mm GTX	769.1N	814.2N	1526.4N	1569.6N
155mm F. OIL	422.8N	418.9N	1607.9N	2128.8N
149mm F. OIL	615.1N	690.6N	2600.6N	2603.6N
144mm F. OIL	683.8N	680.8N	1335.1N	1424.4N
139mm F.OIL	736.7N	669N	1112.5N	875.1N

Table 6.8 Comparison of "Used" and "Unused" Steel Brushes.

As can be determined from the above table there appears to be no consistent pattern in the results obtained for the "used" and "un-used" brushes. Therefore, it is not possible to conclude that any one brush, "used" or "un-used" is any "better" than any other.

6.6.5 R_a EFFECT

As previously mentioned, when the pipe sections used in the experiments were initially obtained, the R_a values were recorded. The 155mm diameter pipe recorded an R_a value of $4.47\,\mu m$. The 149, 144 and 139mm diameter pipes recorded R_a values of $38.84\,\mu m$. The R_a value can be seen to affect the forward force results as illustrated by the graph (RA-EFECT.XLS) in Figure 6.59. The right hand side of the graph shows the forward force results, whilst the positive gradient on the left hand side shows the loading force. The graph shows a significantly decreased forward force value for the $4.47\,\mu m$ pipe section.

An experiment was conducted to determine the affect the R_a value may have on the ability of a brush to grip the pipe wall during reverse force experiments. The brush used was of the radial X-ply design (NEW0X) and the experiment was performed in dry conditions. The brush was able to obtain a suitable purchase in the 139mm, 144mm and 149mm ID pipe sections ($38.84\,\mu m$). However, the brush was unable to obtain a suitable purchase in the 155mm ID pipe ($4.47\,\mu m$). The graph (RA-REV.XLS) in Figure 6.60 details the considerable difference. If the complete data tables are studied it can be seen that the R_a value significantly effects all forward and reverse results, irrespective of lubrication. As with the polymeric tests, the R_a value is one of the critical factors which determines how well a vehicle will perform. The graph in Figure 6.61 details the relationship between surface roughness/pipe ID and the friction coefficient, as the surface roughness increases and the pipe ID decreases, the friction value also increases. Further, it shows how the friction decreases as the lubrication viscosity increases.

6.6.6 BRISTLE ANGLE

The angle between the bristle tip and the pipe wall is important. If this angle is too great, there is a risk that the bristle may prematurely "flip-through". However, if this angle is too small then there is an increased risk that the bristle may simply slide rearwards over the pipe wall. It is very difficult to maintain a specific angle whilst a vehicle negotiates a pipe, this is due to the fact that the ID of the pipe can and does change along the length of the pipe. For example, it is possible for a single line, under inspection, to be made up of schedule 20, schedule 40, schedule 80 and possibly schedule 120 pipe in the same line. In reality, the variance is generally limited to between schedule 40 and schedule 80. Thus, maintaining an "optimum" bristle angle is not possible. In practice, the angle between the bristle and pipe wall should be between 50 and 60 degrees. However, this angle will vary depending upon a number of factors, for example, lubrication, whether the brush is of the X-ply type and pipe ID changes. It can be seen that, ideally, an ability to alter the effective length of the bristles whilst negotiating the pipe length would be desirable. This in turn would alter the angle formed between the bristle and the pipe wall. This angle could be constantly monitored and corrected depending upon the ambient conditions. Thus, optimum tractive capability could be maintained constantly.

6.7 CONCLUSION

This chapter illustrates the considerable complexities associated with using brushes as "engineering" components. Even after significant testing there are still a large number of un-answered questions. Specifically, the interaction of the bristle tip with the pipe wall when the bristle is subjected to reverse axial forces.

6.7.1 POLYMERIC BRISTLED BRUSHES

It can be concluded that the main disadvantage of polymeric bristles is that they lose their stiffness when in contact with water. Under these conditions, polymeric bristles can lose between 20 and 50 percent of their stiffness within one hour (at 20 degrees Celsius). Additionally, they also suffer from creep, as determined by experimentation. In many hotter countries, where the vehicle may be expected to operate, the rate of creep would be accelerated. Further, it was noted that if the bristles are left loaded inside a pipe for significantly longer periods, the loss of stiffness does not appear to increase above 50 percent.

Choice of bristle material and pipeline material is also important as seen from the UPVc pipe/nylon bristle tests, some materials are considered unsuitable and do not give favourable traction characteristics.

The R_a or surface roughness of the pipeline is important, as the roughness increases so does the force required to push the brush forward through the pipe, thus, overcoming friction. If the brush unit is subjected to reverse force loading, then an increased R_a value ensures the brush is better able to obtain a satisfactory purchase. The wear rates of polymeric bristles must also be considered, although no specific experiments were performed to determine the level of wear, the SEM images clearly indicate significant wear.

One advantage of polymeric bristles is that as the viscosity of lubrication increases, the force required to push the brush forward through the pipe decreases. Further, under similar lubrication conditions there is no drop in the reverse force the brush can withstand prior to failure. Subsequently, this phenomenon increases the forward "slip" to rear "grip" ratios substantially.

6.7.2 STEEL BRISTLED BRUSHES

The steel bristled brushes followed the basic findings of the polymer bristled brush experiments. However, two major differences were noted. The first was the effect the radial X-ply brush unit had on the reverse force experiments under dry conditions. The second was the inability of the steel bristles to obtain an adequate “foothold” under lubrication conditions when subjected to reverse force experiments.

Again, the R_a value of the pipe wall affects the forward force, as the surface roughness increases so does the force required to push the brush unit forward through the pipe.

Lubrication was found to lower the force required to push the brush forward through the pipe. Again, as the viscosity of the lubricant increased the force required to push the brush forward through the pipe decreased.

It was found that a significantly higher force was required to load and subsequently push a radial steel bristled brush forward through a pipe than to load and push forward a steel bristled brush with 15 degree pre-swept bristles.

The radial steel bristled X-ply brush showed a significant ability to increase the reverse force a brush could withstand when under dry conditions. As previously mentioned, it would be beneficial to re-manufacture and test the X-ply steel bristled brush with the 15 degree-pre-swept bristles.

6.7.3 SUMMARY

To conclude it can be said that the point at which a brush unit will fail when exposed to axial reverse forces is extremely difficult to determine. The process of failure is caused by instability, therefore predication of failure is complex. However, the radial X-ply brush appears to be able to defer this instability mechanism significantly when conditions are dry.

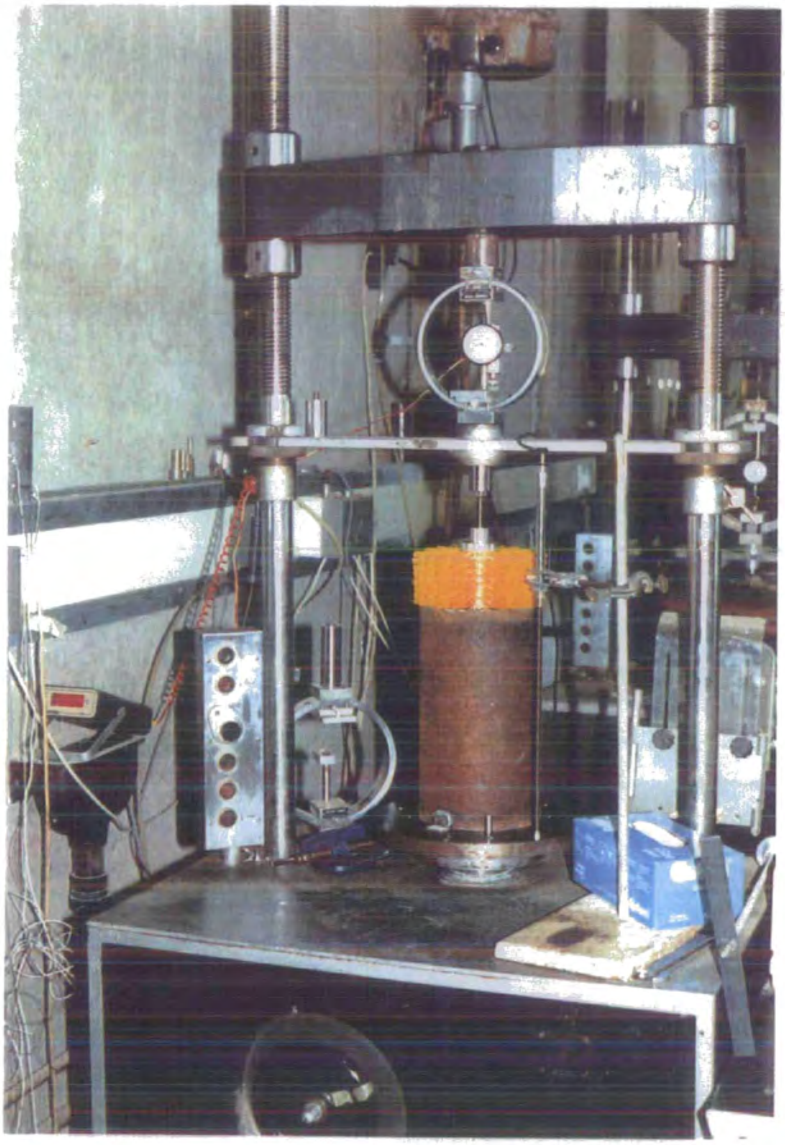


FIGURE 6.1 A "Clockhouse" compression testing machine, set up for testing brushes.

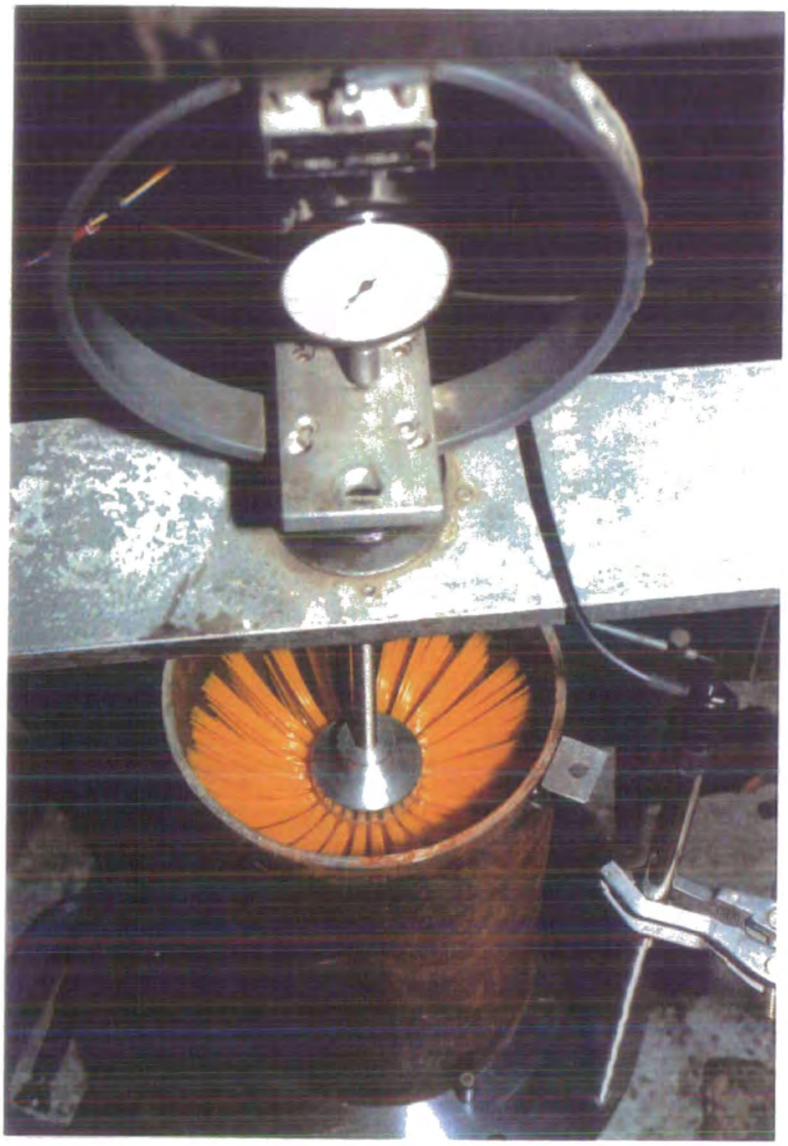


FIGURE 6.2 A photograph showing the location of the linear transducer which is located on the right hand side of the lifting plate.

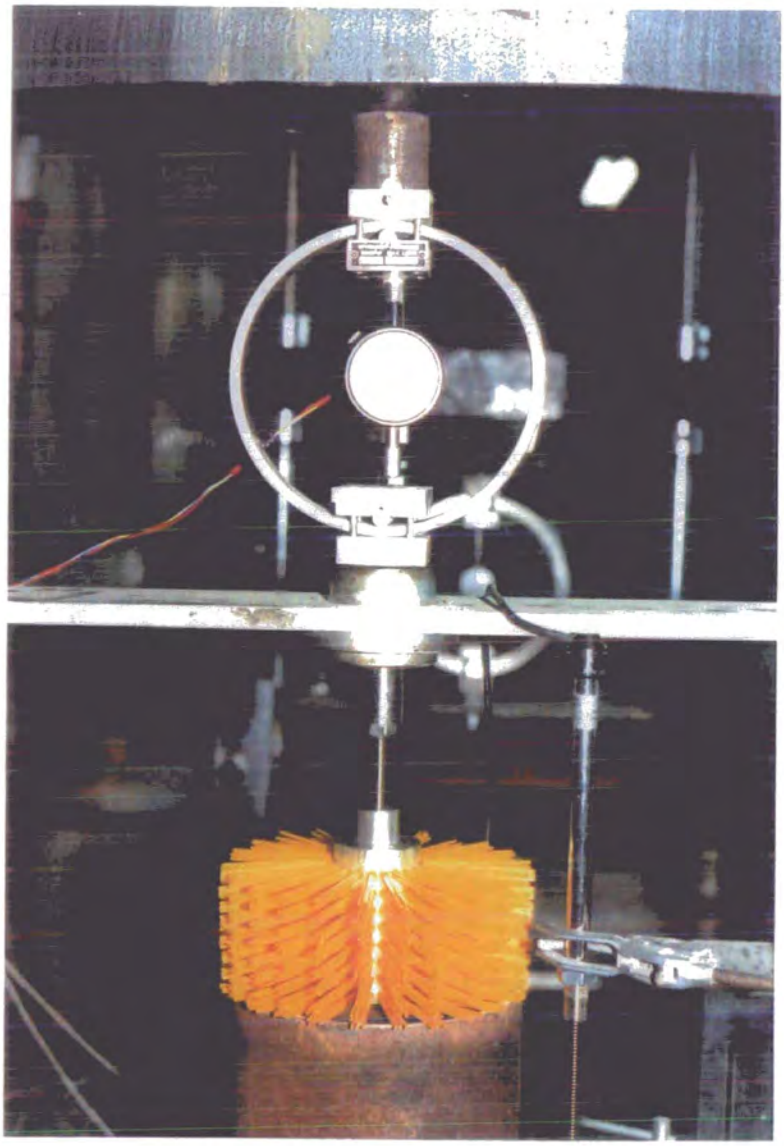


FIGURE 6.3 A photograph showing the aluminium support used to centralise the brush as it is pushed into the pipe.

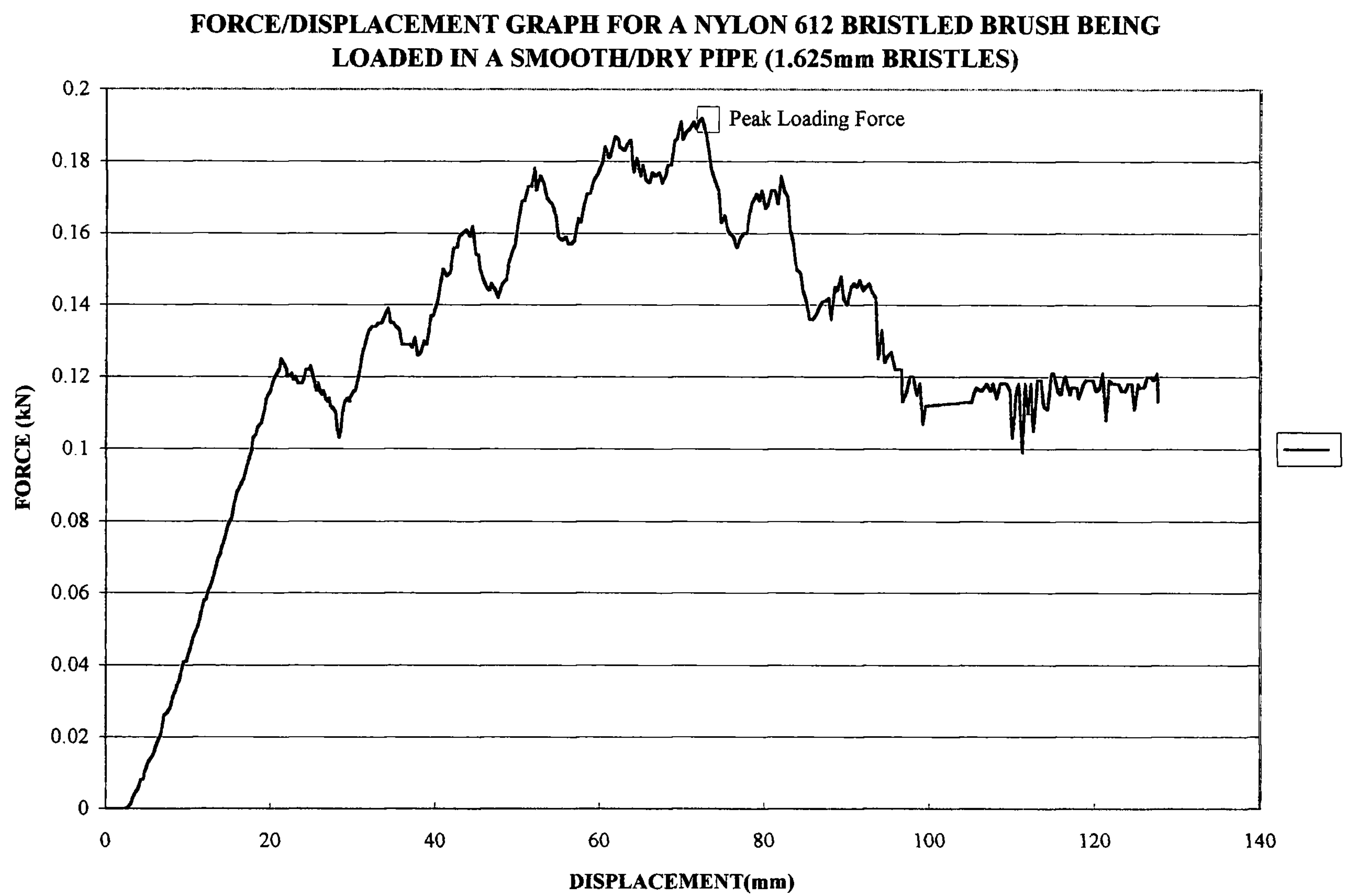


FIGURE 6.4 (16LPFDS1.XLS)

PEAK LOADING FORCE 0.192kN

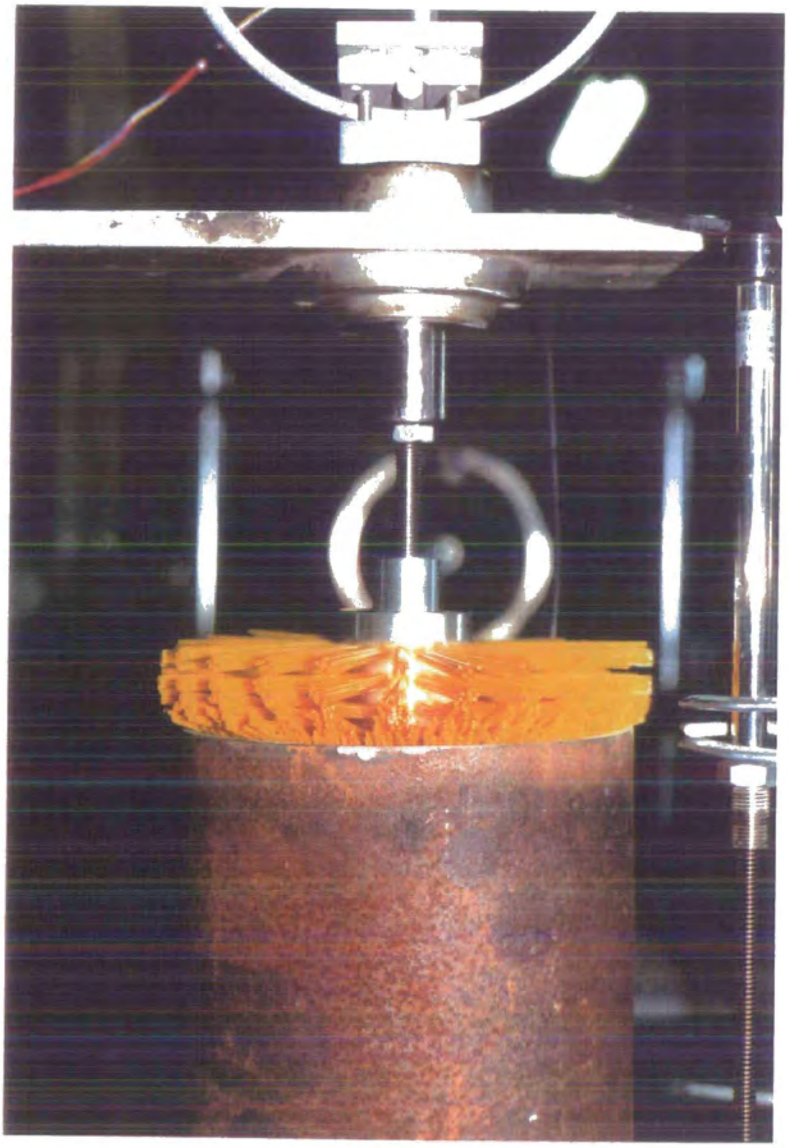


FIGURE 6.5 A photograph showing a polymeric bristled brush undergoing a load and push forward experiment.

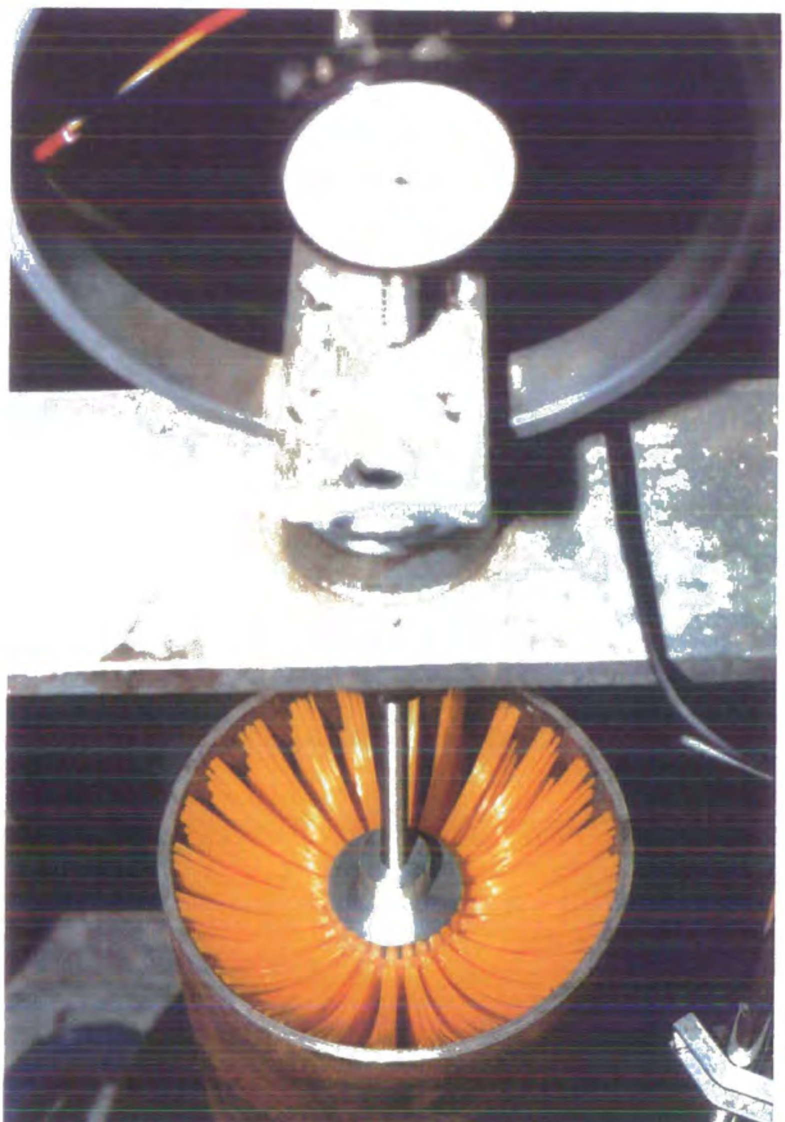
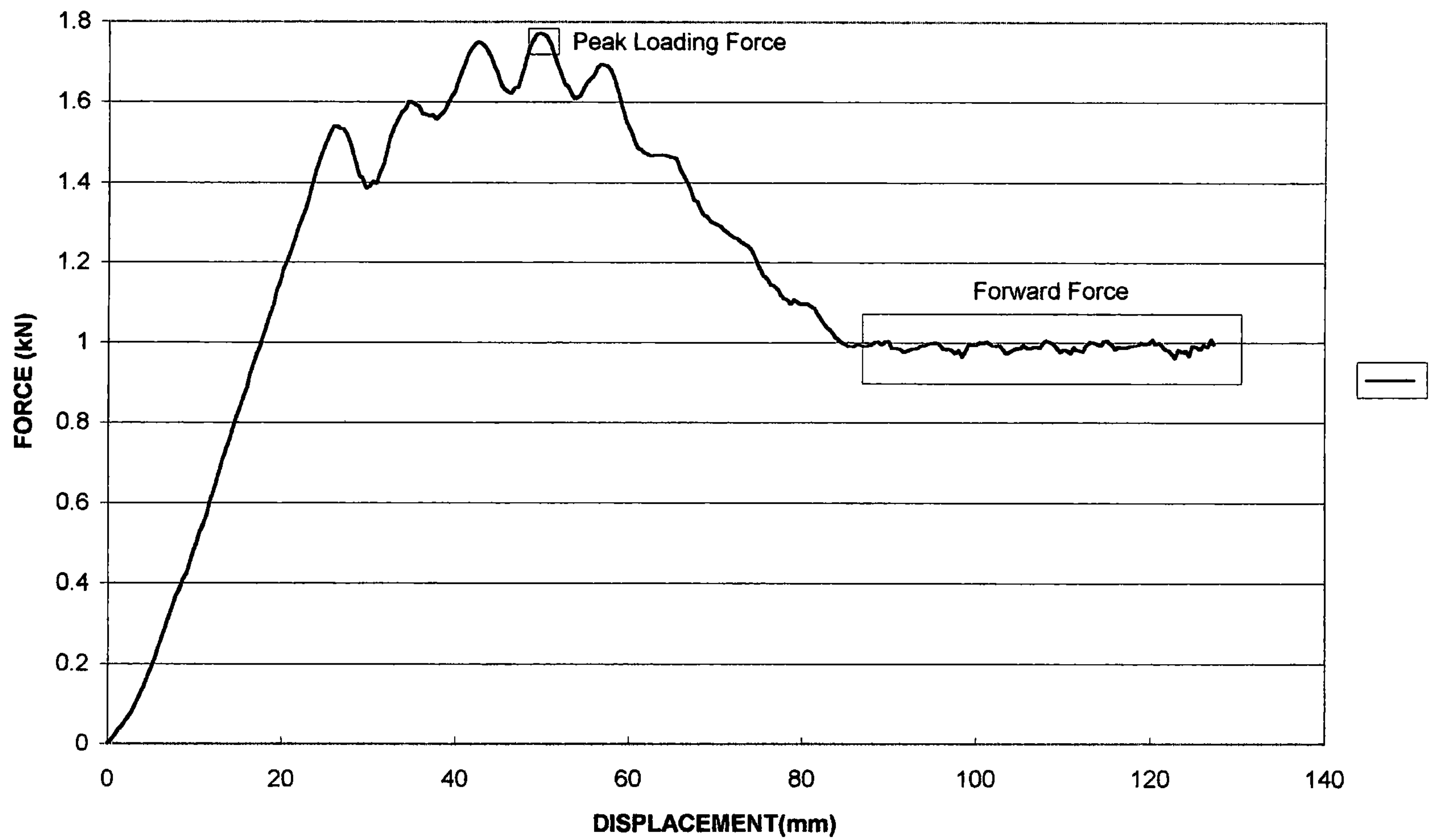


FIGURE 6.6 A photograph showing a polymeric bristled brush being pushed forward through a pipe.

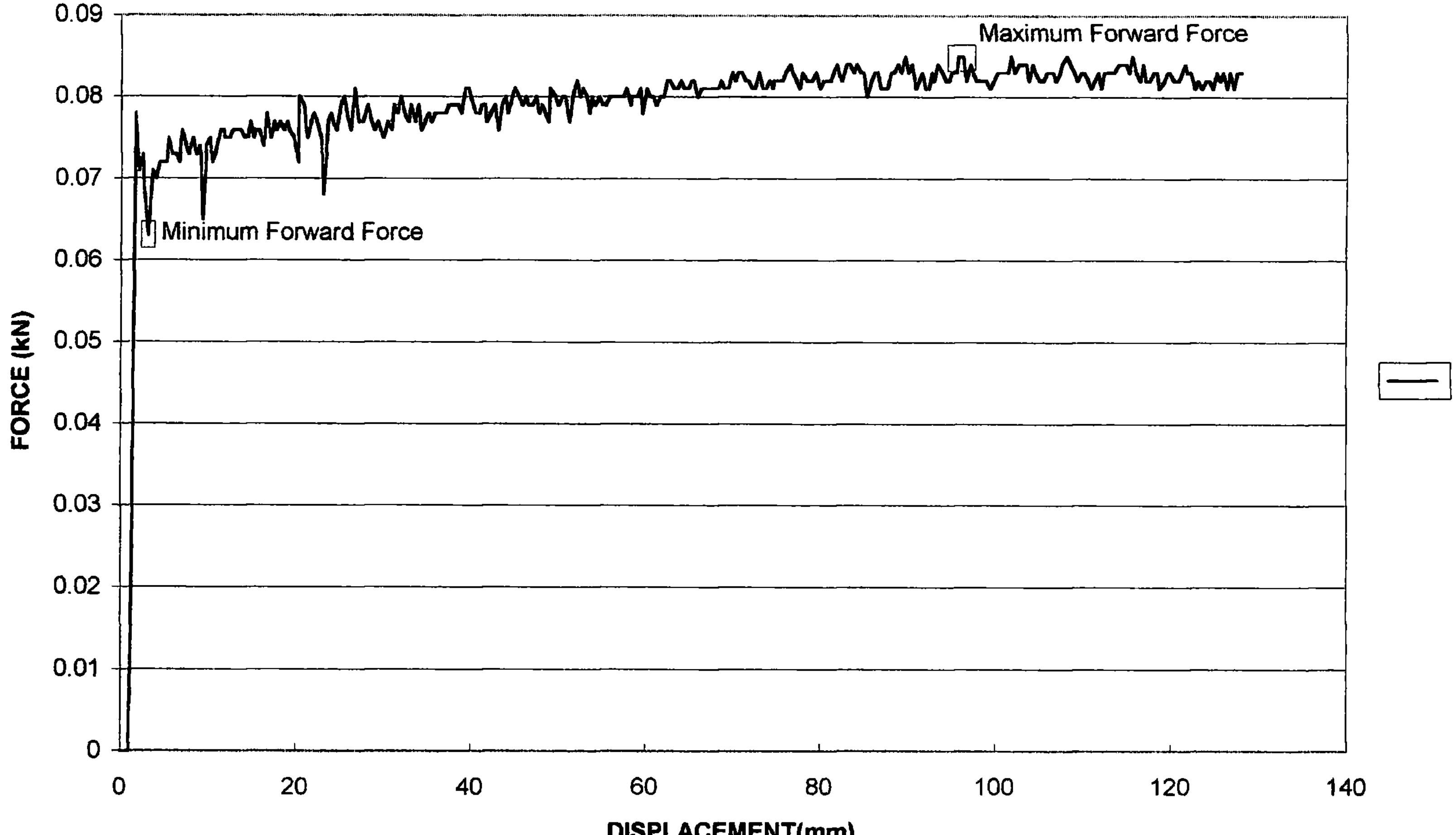
A FORCE/DISPLACEMENT GRAPH FOR A STEEL RADIAL X-PLY BRUSH BEING LOADED AND PUSHED FORWARD IN A 155mm DIAMETER, DRY PIPE



PEAK LOADING FORCE 1.770kN
AVERAGE FORCE TO PUSH BRUSH FORWARD 1kN

FIGURE 6.7 (N0XLD55.XLS)

A FORCE/DISPLACEMENT GRAPH FOR A NYLON 612 BRISTLED BRUSH BEING PUSHED FORWARD THROUGH A SMOOTH/DRY PIPE (1.625mm BRISTLES)



MAXIMUM FORCE 0.085kN
MINIMUM FORCE 0.063kN
AVERAGE FORCE 0.0799kN

"AVERAGE FORCE" IS THE AVERAGE OF ALL THE POINTS (NOT INCLUDING THE INITIAL POSITIVE GRADIENT)

FIGURE 6.8 (16FCVDS1.XLS)

A FORCE/DISPLACEMENT GRAPH FOR A NYLON 612 BRISTLED BRUSH IN A SMOOTH/DRY PIPE (1.625mm BRISTLES) BEING SUBJECT TO A REVERSE FORCE EXPERIMENT

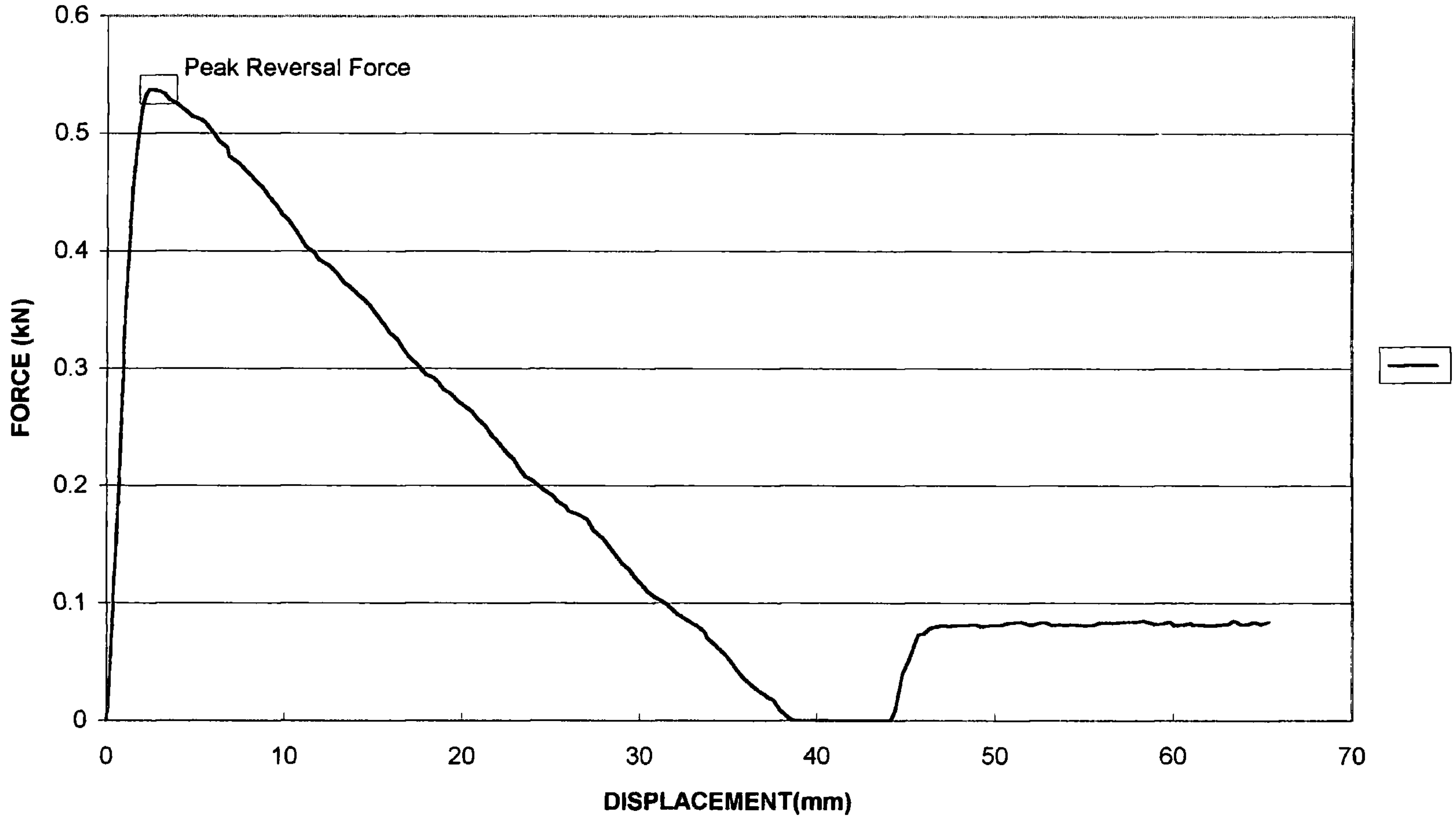


FIGURE 6.9 (16REVDS1.XLS) PEAK REVERSAL FORCE 0.537kN ROTATION C.W.

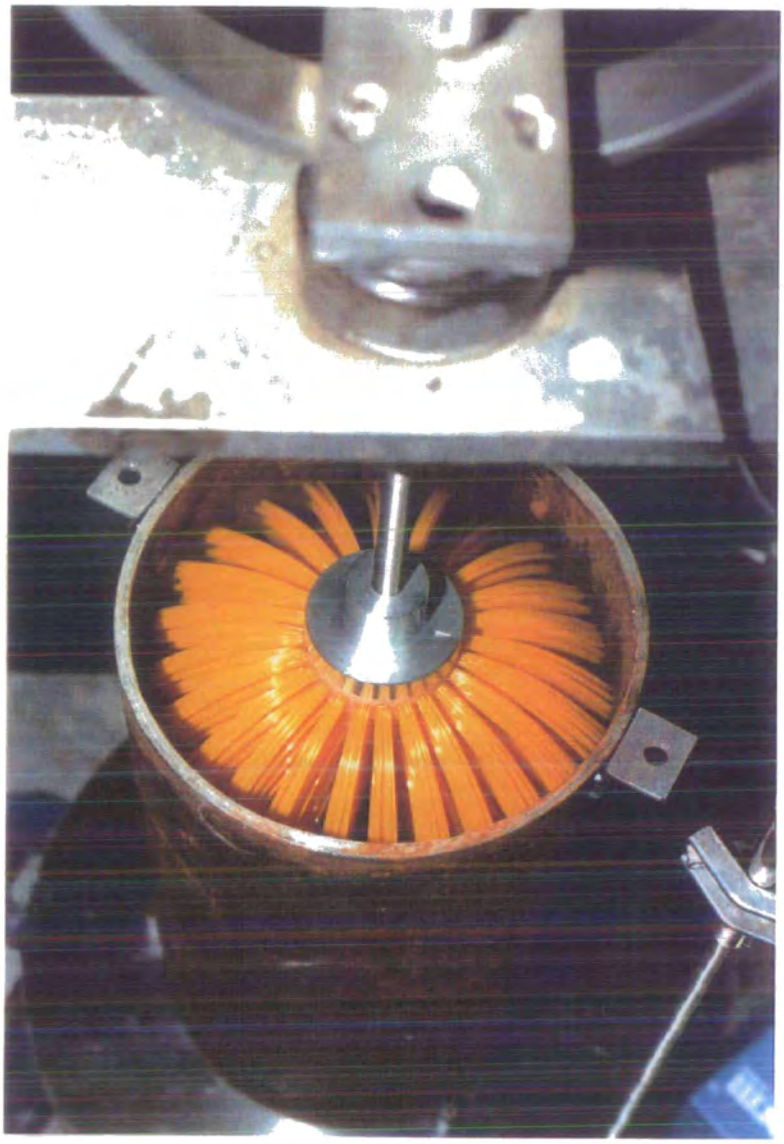


FIGURE 6.10 A photograph showing a polymeric bristled brush being subjected to a reverse force experiment.

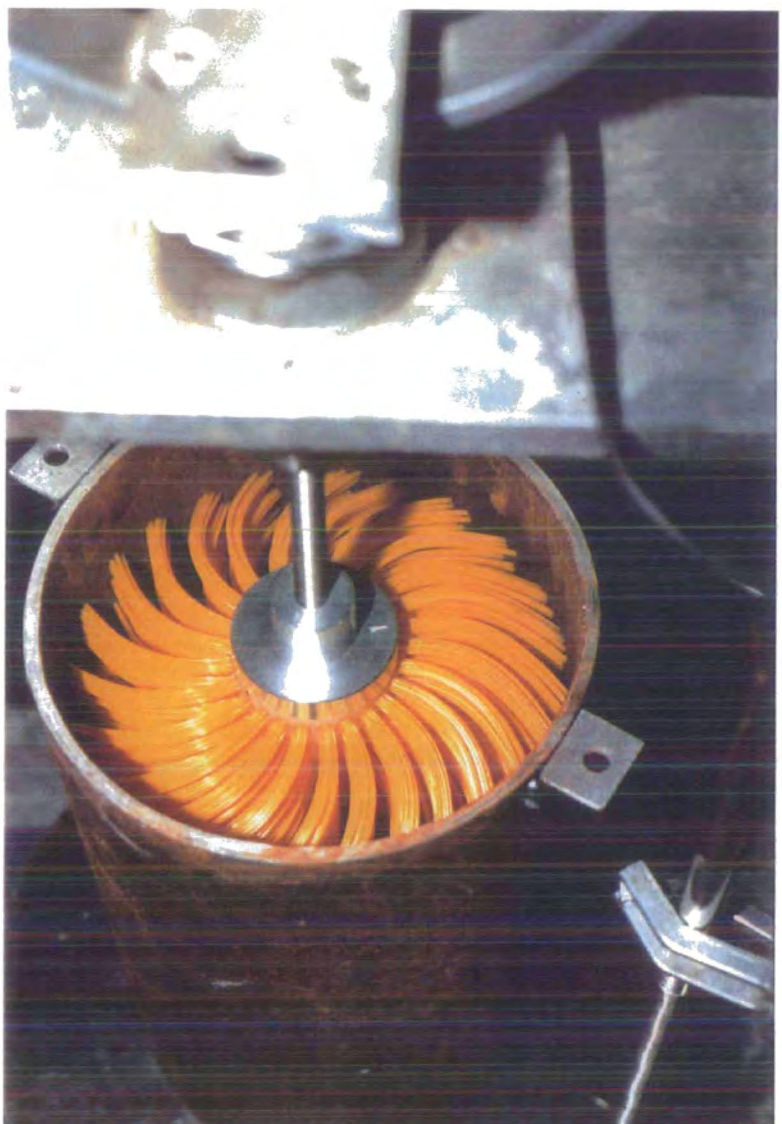


FIGURE 6.11 A photograph showing a polymeric bristled brush with its bristles displaced out of plane after being subjected to a reverse force experiment.

A FORCE/DISPLACEMENT GRAPH FOR A NYLON 612 BRISTLED BRUSH IN A MEDIUM ROUGHNESS/LIGHT OIL LUBRICATED PIPE (1.625mm BRISTLES) BEING SUBJECT TO A REVERSE FORCE EXPERIMENT

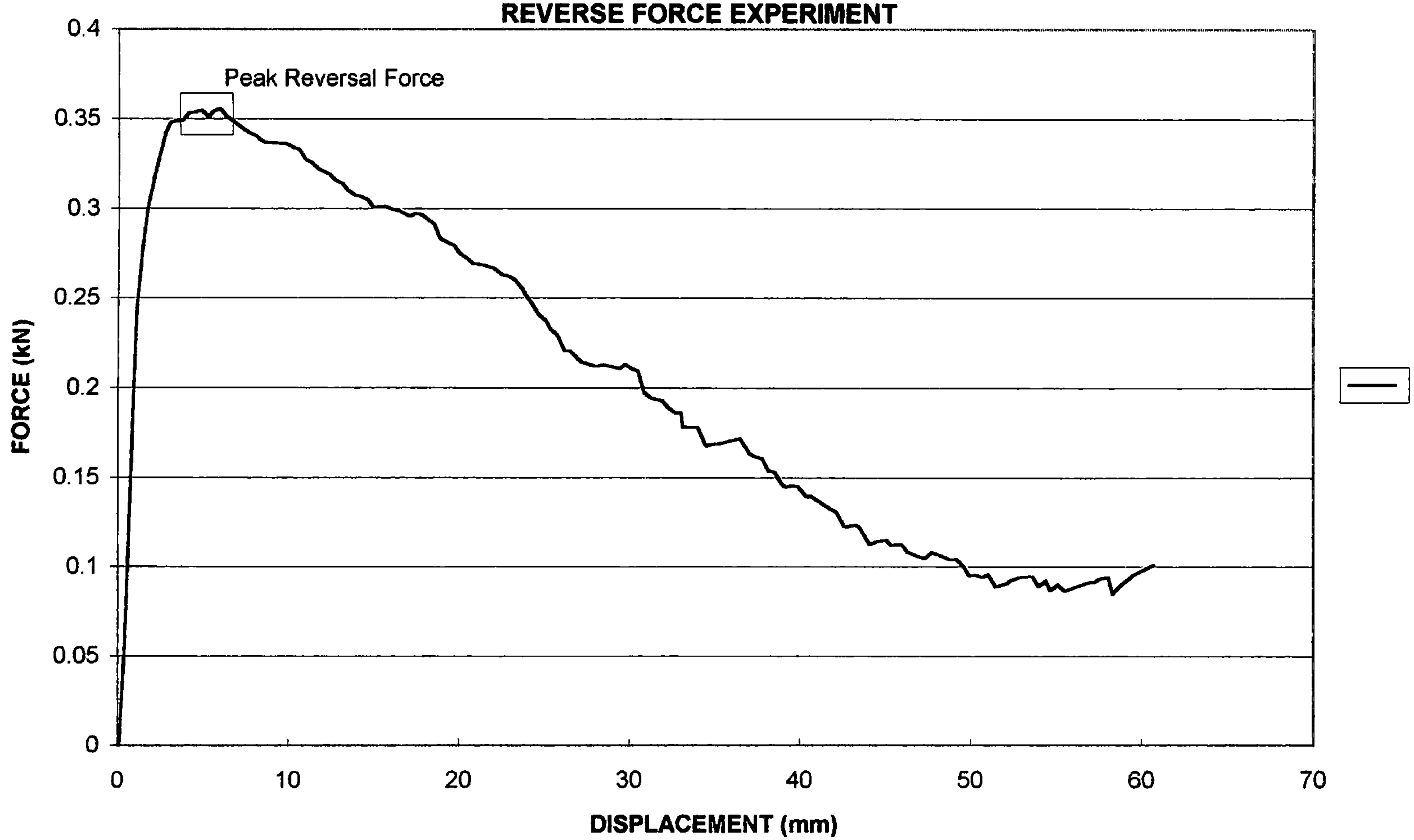


FIGURE 6.12 (16REVXM1.XLS) PEAK REVERSAL FORCE 0.355kN ROTATION A.C.W. LEADING BRISTLES DID NOT REVERSE

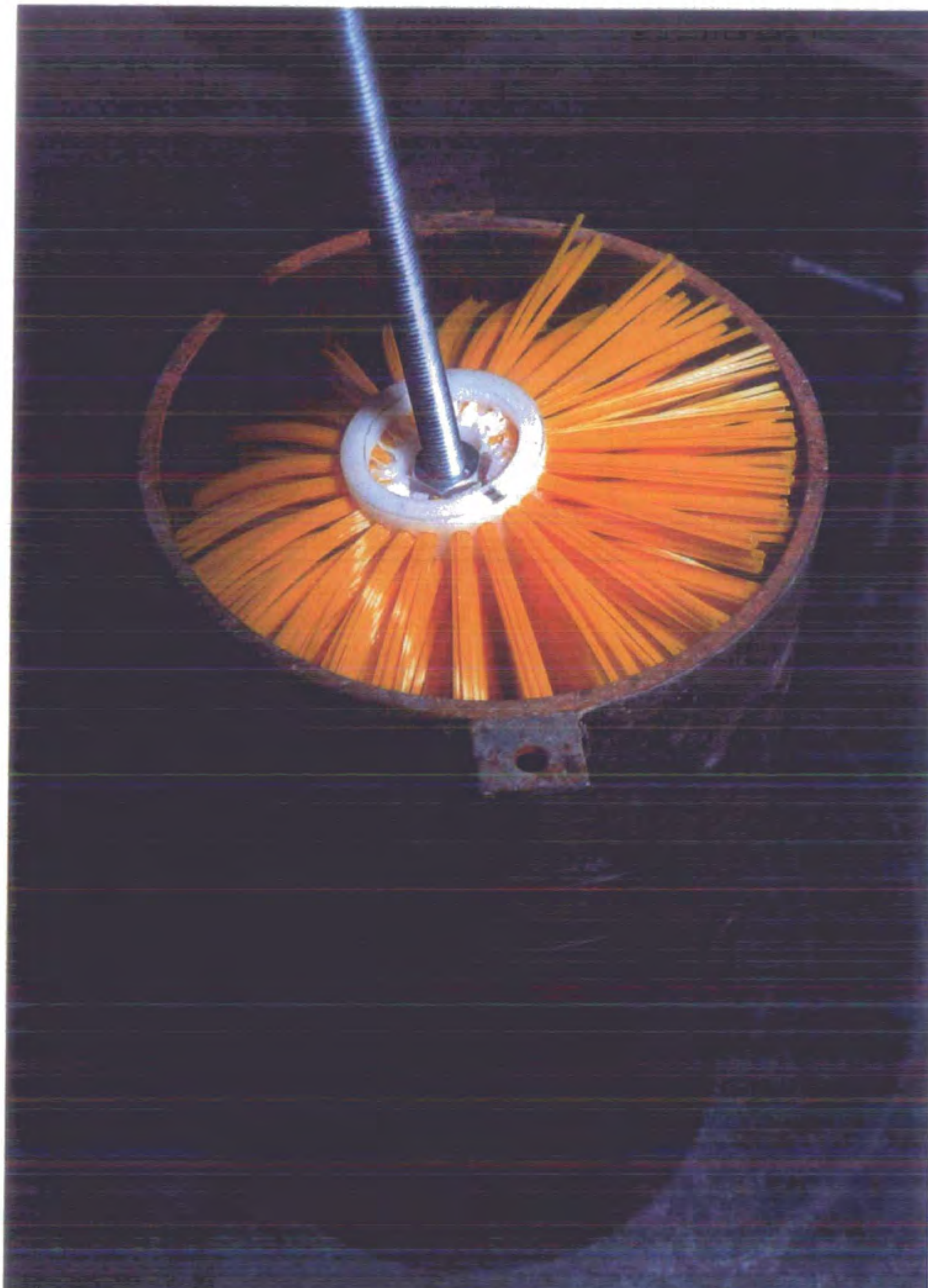


FIGURE 6.13 A photograph showing an example of a polymeric bristled brush which has moved off the centre axis of the pipe during a reverse force experiment.

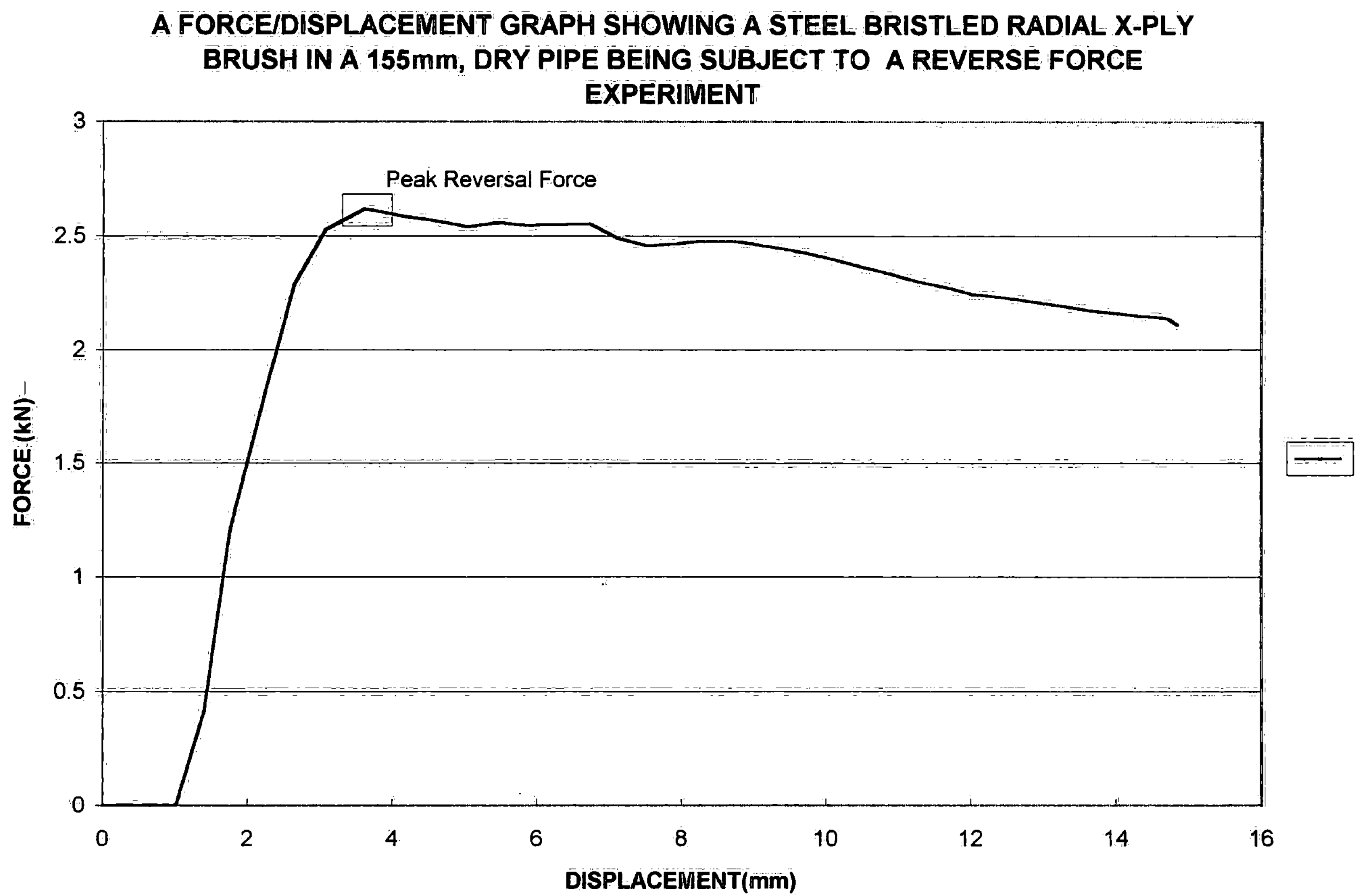


FIGURE 6.14 (N0XRD55.XLS)

PEAK REVERSAL FORCE 2.618kN

ROTATION A.C.W.
BRISTLE SLIPPAGE

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 66: 0.7mm BRISTLED BRUSH

LOADING FORCE											
PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	64			21'C		LINEAR & STEPS IN				65.2
M1 DRY	12.36	80			21'C		LOADING.				65.2
R1 DRY	22.68	68			18'C						65.2
S1 H2O	10.56	53			18'C						65.2
M1 H2O	12.36	48			21'C						65.2
R1 H2O	22.68	38			17'C						65.2
S1 DIESEL	10.56	59			19'C	4					65.2
M1 DIESEL	12.36	52			18'C	4					65.2
R1 DIESEL	22.68	66			16'C	4					65.2
S1 GTX	10.56	44			18'C	104					65.2
M1 GTX	12.36	62			17'C	104					65.2
R1 GTX	22.68	47			18'C	104					65.2

FORWARD FORCE											
PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	33	29	30	18'C			0.18	0.16	0.17	65.2
M1 DRY	12.36	39	31	34	19'C			0.21	0.17	0.19	65.2
R1 DRY	22.68	45	38	42	19'C			0.25	0.21	0.23	65.2
S1 H2O	10.56	16	10	14	18'C		0.7mm ABSORBED H2O	0.09	0.05	0.08	65.2
M1 H2O	12.36	17	11	15	17'C		AND BECAME LESS	0.09	0.06	0.08	65.2
R1 H2O	22.68	18	11	15	18'C		STIFF?	0.1	0.06	0.08	65.2
S1 DIESEL	10.56	22	17	19	18'C	4	LESS ABSORBANT OR	0.12	0.09	0.1	65.2
M1 DIESEL	12.36	29	22	26	18'C	4	VISCOUS DRAG?.	0.16	0.12	0.14	65.2
R1 DIESEL	22.68	36	29	32	18'C	4		0.2	0.16	0.18	65.2
S1 GTX	10.56	21	14	18	18'C	104		0.11	0.08	0.1	65.2
M1 GTX	12.36	12	8	10	17'C	104		0.06	0.04	0.05	65.2
R1 GTX	22.68	34	28	29	18'C	104		0.19	0.15	0.16	65.2

REVERSE FORCE													
PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	222			19'C		COUPLE REVERSAL				65.2	ACW	7.4
M1 DRY	12.36	256			20'C		COUPLE REVERSAL				65.2	CW	7.5294
R1 DRY	22.68	221			19'C		COUPLE REVERSAL				65.2	ACW	5.2619
S1 H2O	10.56	204			20'C		COUPLE REVERSAL				65.2	CW	14.571
M1 H2O	12.36	206			19'C		COUPLE REVERSAL				65.2	CW	13.733
R1 H2O	22.68	224			18'C		COUPLE REVERSAL				65.2	CW	14.933
S1 DIESEL	10.56	184			20'C	4	COUPLE REVERSAL				65.2	CW	9.6842
M1 DIESEL	12.36	206			17'C	4	COUPLE REVERSAL				65.2	CW	7.9231
R1 DIESEL	22.68	200			20'C	4	COUPLE REVERSAL				65.2	CW	6.25
S1 GTX	10.56	185			17'C	104	COUPLE REVERSAL				65.2	ACW	10.278
M1 GTX	12.36	188			18'C	104	COUPLE REVERSAL				65.2	ACW	18.8
R1 GTX	22.68	226			18'C	104	COUPLE REVERSAL				65.2	ACW	7.7931

FIGURE 6.15

VISCOSITY IN CENTISTOKES AT 40'C
RATIO=P AV./P MAX

TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 66, 1.1mm BRISTLED BRUSH

LOADING FORCE

PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	131			20'C		INCREASED STEP SIZE				65.2
M1 DRY	12.36	152			21'C						65.2
R1 DRY	22.68	159			18'C						65.2
S1 H20	10.56	130			18'C						65.2
M1 H20	12.36	121			20'C						65.2
R1 H2O	22.68	130			17'C						65.2
S1 DIESEL	10.56	119			19'C	4					65.2
M1 DIESEL	12.36	123			18'C	4					65.2
R1 DIESEL	22.68	140			16'C	4					65.2
S1 GTX	10.56	124			18'C	104					65.2
M1 GTX	12.36	128			18'C	104					65.2
R1 GTX	22.68	125			18'C	104					65.2

FORWARD FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	70	51	64	18'C			0.16	0.12	0.15	65.2
M1 DRY	12.36	107	87	95	19'C			0.25	0.2	0.22	65.2
R1 DRY	22.68	101	91	96	19'C			0.23	0.21	0.22	65.2
S1 H20	10.56	70	56	64	17'C		WATER ABSORBTION?	0.16	0.13	0.15	65.2
M1 H20	12.36	79	65	74	17'C			0.18	0.15	0.17	65.2
R1 H2O	22.68	93	80	86	18'C			0.21	0.18	0.2	65.2
S1 DIESEL	10.56	69	58	65	18'C	4		0.16	0.13	0.15	65.2
M1 DIESEL	12.36	70	61	65	18'C	4		0.16	0.14	0.15	65.2
R1 DIESEL	22.68	77	65	71	18'C	4		0.18	0.15	0.16	65.2
S1 GTX	10.56	51	43	47	18'C	104		0.12	0.1	0.11	65.2
M1 GTX	12.36	50	40	44	17'C	104		0.12	0.09	0.1	65.2
R1 GTX	22.68	84	65	72	17'C	104		0.18	0.15	0.16	65.2

REVERSE FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	431			19'C		COUPLE REVERSAL				65.2	CW	6.7344
M1 DRY	12.36	451			19'C		COUPLE REVERSAL				65.2	ACW	4.7474
R1 DRY	22.68	438			19'C		COUPLE REVERSAL				65.2	ACW	4.5625
S1 H20	10.56	395			19'C		COUPLE REVERSAL				65.2	CW	6.1719
M1 H20	12.36	274			19'C		COUPLE REVERSAL				65.2	CW	3.7027
R1 H2O	22.68	335			18'C		COUPLE REVERSAL				65.2	CW	3.8953
S1 DIESEL	10.56	294			21'C	4	COUPLE REVERSAL				65.2	CW	4.5231
M1 DIESEL	12.36	326			17'C	4	COUPLE REVERSAL				65.2	CW	5.0154
R1 DIESEL	22.68	381			19'C	4	COUPLE REVERSAL				65.2	CW	5.3662
S1 GTX	10.56	440			17'C	104	COUPLE REVERSAL				65.2	CW	9.3617
M1 GTX	12.36	349			18'C	104	COUPLE REVERSAL				65.2	CW	7.9318
R1 GTX	22.68	422			18'C	104	COUPLE REVERSAL				65.2	CW	5.8611

FIGURE: 6.16

VISCOSITY IN CENTISTOKES AT 40'C
RATIO=P AV./P MAX

TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 66, 1.3mm BRISTLED BRUSH

LOADING FORCE

PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	198			20°C						65.2
M1 DRY	12.36	243			21°C						65.2
R1 DRY	22.68	252			18°C						65.2
S1 H20	10.56	200			18°C						65.2
M1 H20	12.36	209			20°C						65.2
R1 H2O	22.68	223			16°C						65.2
S1 DIESEL	10.56	227			19°C	4					65.2
M1 DIESEL	12.36	201			18°C	4					65.2
R1 DIESEL	22.68	229			15°C	4					65.2
S1 GTX	10.56	215			18°C	104					65.2
M1 GTX	12.36	203			18°C	104					65.2
R1 GTX	22.68	210			18°C	104					65.2

FORWARD FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	114	81	106	18°C			0.22	0.16	0.21	65.2
M1 DRY	12.36	167	146	154	19°C			0.32	0.28	0.3	65.2
R1 DRY	22.68	157	143	151	20°C			0.3	0.28	0.29	65.2
S1 H20	10.56	114	90	104	17°C			0.22	0.18	0.2	65.2
M1 H20	12.36	128	114	120	17°C			0.25	0.22	0.23	65.2
R1 H2O	22.68	125	115	121	18°C			0.24	0.22	0.23	65.2
S1 DIESEL	10.56	108	83	98	18°C	4		0.21	0.16	0.19	65.2
M1 DIESEL	12.36	129	90	110	18°C	4		0.25	0.18	0.21	65.2
R1 DIESEL	22.68	125	103	116	17°C	4		0.24	0.2	0.22	65.2
S1 GTX	10.56	80	65	71	18°C	104		0.16	0.13	0.14	65.2
M1 GTX	12.36	107	86	94	17°C	104		0.21	0.17	0.18	65.2
R1 GTX	22.68	134	113	122	17°C	104		0.26	0.22	0.24	65.2

REVERSE FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	652			20°C		COUPLE REVERSAL				65.2	ACW	6.1509
M1 DRY	12.36	844			19°C		COUPLE REVERSAL				65.2	ACW	5.4805
R1 DRY	22.68	809			18°C		COUPLE REVERSAL				65.2	ACW	5.3576
S1 H20	10.56	712			20°C		CORE OFFSET, B. SLIP				65.2	NONE	6.8462
M1 H20	12.36	680			19°C		CORE OFFSET, B. SLIP				65.2	NONE	5.6667
R1 H2O	22.68	906			18°C		COUPLE REVERSAL				65.2	ACW	7.4876
S1 DIESEL	10.56	719			21°C	4	SOME BRISTLE SLIP				65.2	CW	7.3367
M1 DIESEL	12.36	685			17°C	4	SOME BRISTLE SLIP				65.2	ACW	6.2273
R1 DIESEL	22.68	837			19°C	4	COUPLE REVERSAL				65.2	ACW	7.2155
S1 GTX	10.56	801			17°C	104	SOME BRISTLE SLIP*				65.2	CW	11.282
M1 GTX	12.36	587			18°C	104	SOME BRISTLE SLIP*				65.2	CW	6.2447
R1 GTX	22.68	545			19°C	104	COUPLE REVERSAL				65.2	CW	4.4672

*LEADING BANK OF BRISTLES DID NOT REVERSE
RATIO= P AV/P MAX

VISCOSITY IN CENTISTOKES AT 40°C
TEST RIG VELOCITY=0.174mm/s

FIGURE 6.17

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 66, 1.5mm BRISTLED BRUSH

LOADING FORCE

PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	233			20'C						65.2
M1 DRY	12.36	313			21'C						65.2
R1 DRY	22.68	277			18'C						65.2
S1 H20	10.56	256			18'C						65.2
M1 H20	12.36	235			20'C						65.2
R1 H2O	22.68	283			16'C						65.2
S1 DIESEL	10.56	259			19'C	4					65.2
M1 DIESEL	12.36	251			17'C	4					65.2
R1 DIESEL	22.68	265			16'C	4					65.2
S1 GTX	10.56	250			18'C	104					65.2
M1 GTX	12.36	239			17'C	104					65.2
R1 GTX	22.68	221			18'C	104					65.2

FORWARD FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	154	113	144	18'C			0.21	0.15	0.19	65.2
M1 DRY	12.36	189	167	175	19'C			0.25	0.22	0.23	65.2
R1 DRY	22.68	194	177	184	20'C			0.26	0.23	0.24	65.2
S1 H20	10.56	138	107	124	17'C			0.18	0.14	0.16	65.2
M1 H20	12.36	158	143	151	18'C			0.21	0.19	0.2	65.2
R1 H2O	22.68	160	145	153	18'C			0.21	0.19	0.2	65.2
S1 DIESEL	10.56	128	93	116	18'C	4		0.17	0.12	0.15	65.2
M1 DIESEL	12.36	146	109	132	19'C	4		0.19	0.15	0.18	65.2
R1 DIESEL	22.68	143	111	130	18'C	4		0.19	0.15	0.17	65.2
S1 GTX	10.56	113	92	102	17'C	104		0.15	0.12	0.14	65.2
M1 GTX	12.36	100	90	95	17'C	104		0.13	0.12	0.13	65.2
R1 GTX	22.68	156	129	140	18'C	104		0.21	0.17	0.19	65.2

REVERSAL FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	736			18'C		COUPLE REVERSAL				65.2	ACW	5.1111
M1 DRY	12.36	796			20'C		COUPLE REVERSAL				65.2	CW	4.5486
R1 DRY	22.68	888			18'C		COUPLE REVERSAL				65.2	ACW	4.8261
S1 H20	10.56	677			20'C		COUPLE+SLIP+OFFSET				65.2	ACW	5.4597
M1 H20	12.36	862			19'C		COUPLE+SLIP+OFFSET				65.2	CW	5.7086
R1 H2O	22.68	735			18'C		COUPLE REVERSAL				65.2	ACW	4.8039
S1 DIESEL	10.56	660			21'C	4	COUPLE+SLIP+OFFSET				65.2	ACW	5.6897
M1 DIESEL	12.36	606			18'C	4	COUPLE+SLIP+OFFSET				65.2	CW	4.5909
R1 DIESEL	22.68	755			19'C	4	COUPLE REVERSAL				65.2	ACW	5.8077
S1 GTX	10.56	782			18'C	104	COUPLE+SLIP+OFFSET				65.2	ACW	7.6667
M1 GTX	12.36	651			18'C	104	COUPLE+SLIP+OFFSET				65.2	CW	6.8526
R1 GTX	22.68	812			19'C	104	COUPLE REVERSAL				66.2	ACW	5.8

FIGURE 6.18

*LEADING BANK OF BRISTLES DID NOT REVERSE
RATIO=PAV./P MAX

VISCOSITY IN CENTISTOKES AT 40'C
TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 612, 1.625mm BRISTLED BRUSH

LOADING FORCE											
PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	183			20'C						65.2
M1 DRY	12.36	206			18'C						65.2
R1 DRY	22.68	247			18'C						65.2
S1 H20	10.56	162			17'C	1					65.2
M1 H20	12.36	191			20'C						65.2
R1 H2O	22.68	188			16'C						65.2
S1 DIESEL	10.56	170			18'C	4					65.2
M1 DIESEL	12.36	215			16'C	4					65.2
R1 DIESEL	22.68	174			15'C	4					65.2
S1 GTX	10.56	195			18'C	104					65.2
M1 GTX	12.36	165			18'C	104					65.2
R1 GTX	22.68	182			18'C	104					65.2

FORWARD FORCE											
PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	85	63	80	18'C			0.13	0.1	0.12	65.2
M1 DRY	12.36	140	124	132	18'C			0.22	0.19	0.2	65.2
R1 DRY	22.68	147	135	140	20'C			0.23	0.21	0.22	65.2
S1 H20	10.56	97	80	89	17'C			0.15	0.12	0.14	65.2
M1 H20	12.36	96	81	87	18'C			0.15	0.13	0.14	65.2
R1 H2O	22.68	130	115	124	18'C			0.2	0.18	0.19	65.2
S1 DIESEL	10.56	105	92	99	18'C	4		0.16	0.14	0.15	65.2
M1 DIESEL	12.36	92	80	83	18'C	4		0.14	0.12	0.13	65.2
R1 DIESEL	22.68	124	111	117	18'C	4		0.19	0.17	0.18	65.2
S1 GTX	10.56	69	53	57	17'C	104		0.11	0.08	0.09	65.2
M1 GTX	12.36	87	60	66	17'C	104		0.14	0.09	0.1	65.2
R1 GTX	22.68	97	88	93	17'C	104		0.15	0.14	0.14	65.2

REVERSAL FORCE													
PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	537			20'C		COUPLE REVERSAL				65.2	CW	6.7125
M1 DRY	12.36	584			20'C		COUPLE REVERSAL				65.2	ACW	4.4242
R1 DRY	22.68	574			18'C		COUPLE REVERSAL				65.2	CW	4.1
S1 H20	10.56	395			20'C		COUPLE+SLIP+OFFSET				65.2	CW	4.4382
M1 H20	12.36	400			18'C		COUPLE+SLIP+OFFSET				65.2	CW	4.5977
R1 H2O	22.68	391			18'C		COUPLE REVERSAL				65.2	CW	3.1532
S1 DIESEL	10.56	500			21'C	4	COUPLE REVERSAL				65.2	CW	5.0505
M1 DIESEL	12.36	489			17'C	4	COUPLE REVERSAL				65.2	CW	5.8916
R1 DIESEL	22.68	493			18'C	4	COUPLE REVERSAL				65.2	CW	4.2137
S1 GTX	10.56	400			18'C	104	COUPLE+SLIP+OFFSET*				65.2	ACW	7.0175
M1 GTX	12.36	355			18'C	104	COUPLE+SLIP+OFFSET*				65.2	ACW	5.3788
R1 GTX	22.68	365			19'C	104	COUPLE REVERSAL				65.2	ACW	3.9247

*LEADING BANK OF BRISTLES DID NOT REVERSE
RATIO=P AV./P MAX

VISCOSITY IN CENTISTOKES AT 40'C
TEST RIG VELOCITY 0.174mm/s

FIGURE 6.19

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE NYLON 66, 2.0mm BRISTLED BRUSH

LOADING FORCE

PIPE	Ra VALUE	P MAX(N)	P MIN(N)	P Av.(N)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	440			21'C						65.2
M1 DRY	12.36	485			18'C						65.2
R1 DRY	22.68	446			18'C						65.2
S1 H20	10.56	371			16'C						65.2
M1 H20	12.36	398			20'C						65.2
R1 H2O	22.68	395			17'C						65.2
S1 DIESEL	10.56	331			18'C	4					65.2
M1 DIESEL	12.36	377			17'C	4					65.2
R1 DIESEL	22.68	422			16'C	4					65.2
S1 GTX	10.56	365			18'C	104					65.2
M1 GTX	12.36	413			17'C	104					65.2
R1 GTX	22.68	377			17'C	104					65.2

FORWARD FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
S1 DRY	10.56	187	114	171	17'C			0.14	0.08	0.13	65.2
M1 DRY	12.36	271	239	254	18'C			0.2	0.18	0.19	65.2
R1 DRY	22.68	274	237	261	20'C			0.2	0.18	0.19	65.2
S1 H20	10.56	228	153	204	17'C			0.17	0.11	0.15	65.2
M1 H20	12.36	218	155	209	18'C			0.16	0.11	0.16	65.2
R1 H2O	22.68	238	197	227	18'C			0.17	0.15	0.17	65.2
S1 DIESEL	10.56	196	166	180	18'C	4		0.14	0.12	0.13	65.2
M1 DIESEL	12.36	237	166	209	18'C	4		0.18	0.12	0.15	65.2
R1 DIESEL	22.68	251	215	234	18'C	4		0.19	0.16	0.17	65.2
S1 GTX	10.56	170	141	152	17'C	104		0.13	0.1	0.11	65.2
M1 GTX	12.36	208	152	172	17'C	104		0.15	0.11	0.13	65.2
R1 GTX	22.68	224	182	196	17'C	104		0.16	0.13	0.14	65.2

REVERSE FORCE

PIPE	Ra VALUE	P MAX (N)	P MIN (N)	P Av. (N)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
S1 DRY	10.56	1,180			20'C		COUPLE REVERSAL				65.2	ACW	6.8006
M1 DRY	12.36	1,500			20'C		COUPLE REVERSAL				65.2	ACW	5.8055
R1 DRY	22.68	2,130			18'C		COUPLE REVERSAL				65.2	ACW	8.1609
S1 H20	10.56	467			20'C		SLIP + OFFSET				65.2	HALTED	2.2892
M1 H20	12.36	970			18'C		SLIP + OFFSET				65.2	HALTED	4.8411
R1 H2O	22.68	1,490			18'C		COUPLE+SLIP+OFFSET				65.2	ACW	6.5639
S1 DIESEL	10.56	1,090			22'C	4	SLIP + OFFSET				65.2	HALTED	6.0656
M1 DIESEL	12.36	1,090			17'C	4	SLIP + OFFSET				65.2	HALTED	5.2153
R1 DIESEL	22.68	2,150			18'C	4	COUPLE REVERSAL				65.2	ACW	9.188
S1 GTX	10.56	759			18'C	104	SLIP + OFFSET				65.2	HALTED	4.8934
M1 GTX	12.36	825			18'C	104	SLIP + OFFSET				65.2	HALTED	4.7965
R1 GTX	22.68	1,650			19'C	104	SLIP + OFFSET				65.2	HALTED	8.4184

FIGURE 6.20

*LEADING BANK OF BRISTLES DID NOT REVERSE
RATIO=P AV./P MAX

VISCOSITY IN CENTISTOKES AT 40'C
TEST RIG VELOCITY 0.174mm/s

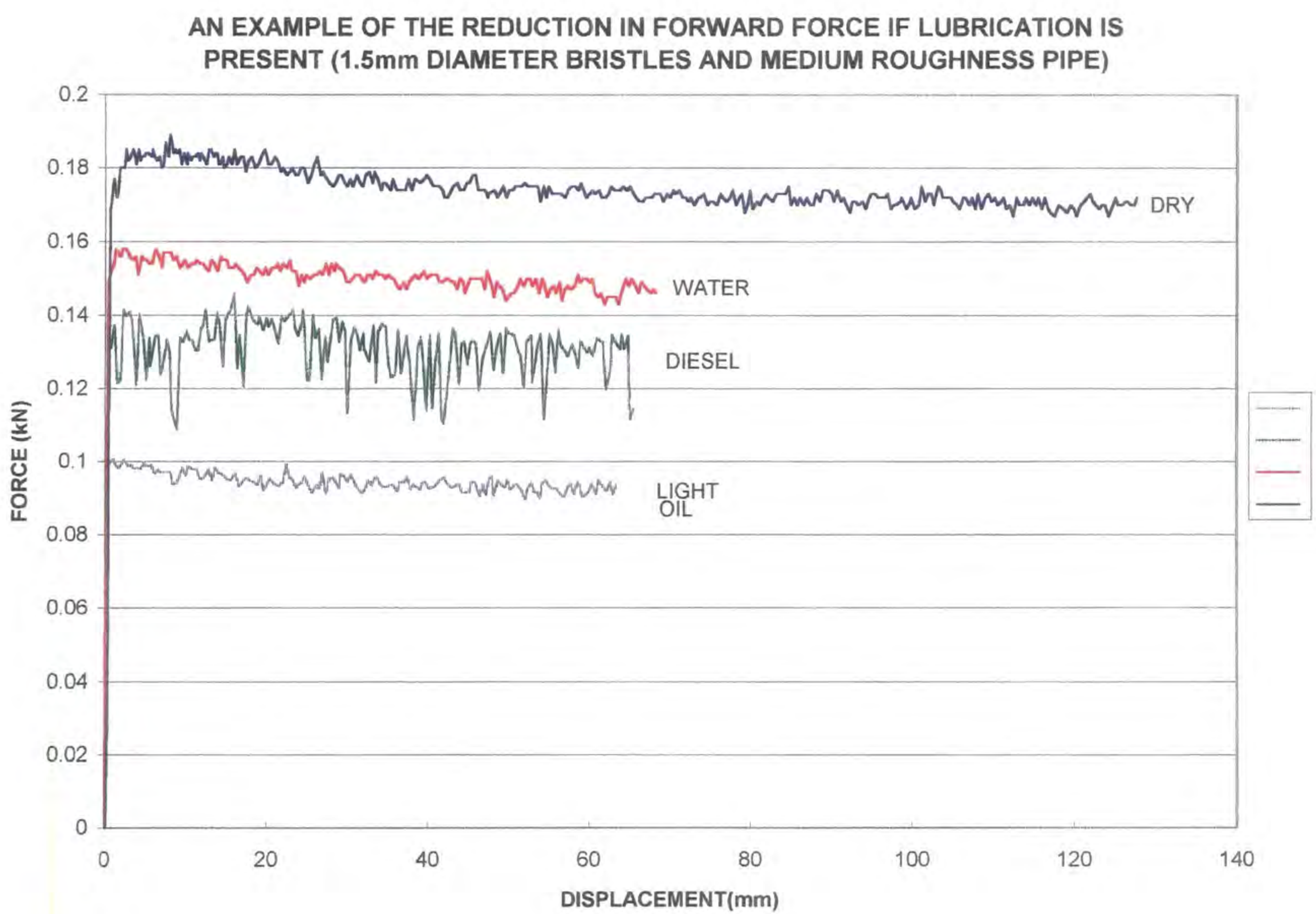


FIGURE 6.21 (LUB-15F.XLS)

AN EXAMPLE OF THE EFFECT SURFACE ROUGHNESS HAS ON THE FORCE REQUIRED TO PUSH A BRUSH FORWARD THROUGH A PIPE (0.7mm DIAMETER NYLON 66 BRISTLES-DRY PIPE)

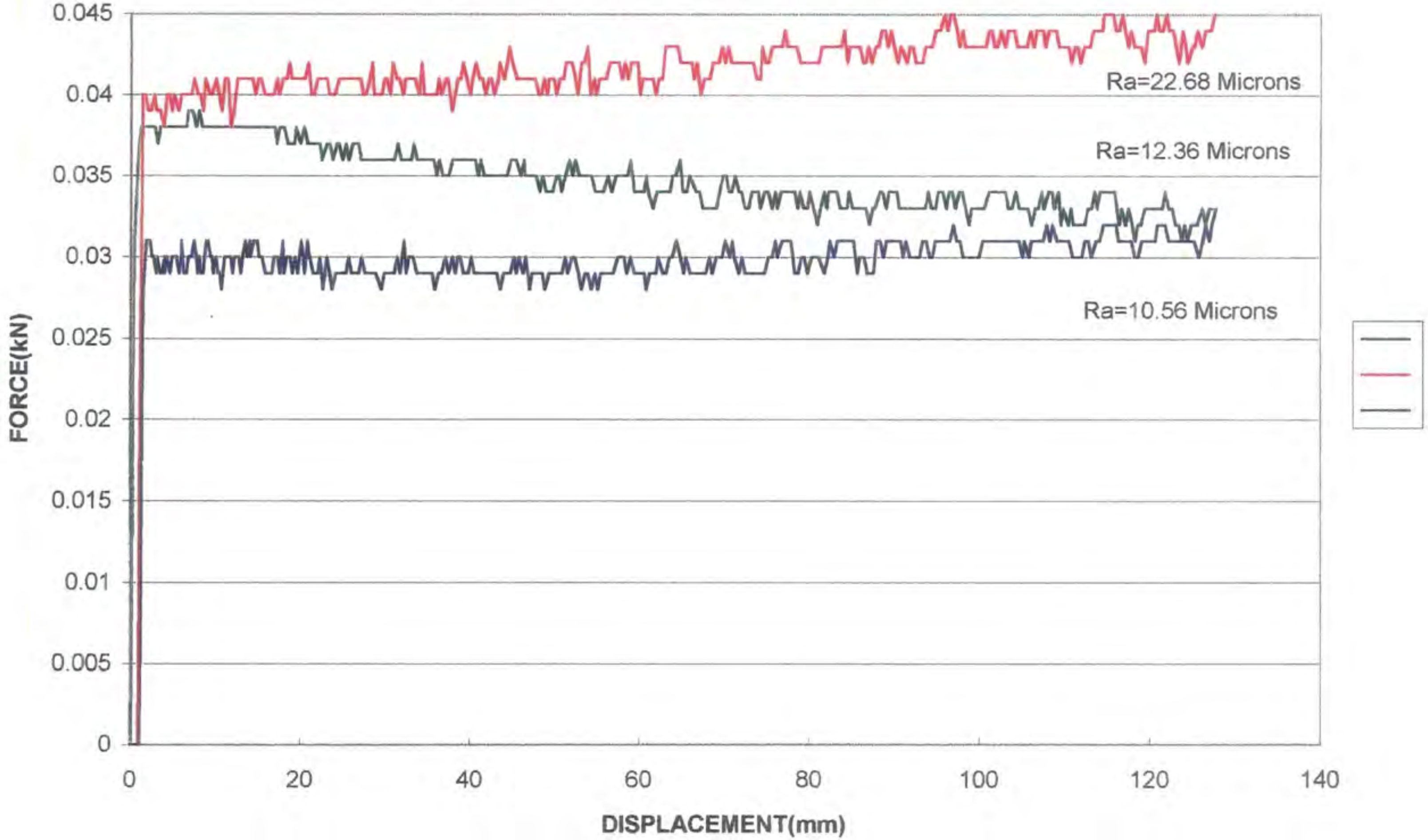


FIGURE 6.22
(RA-FCV07.XLS)

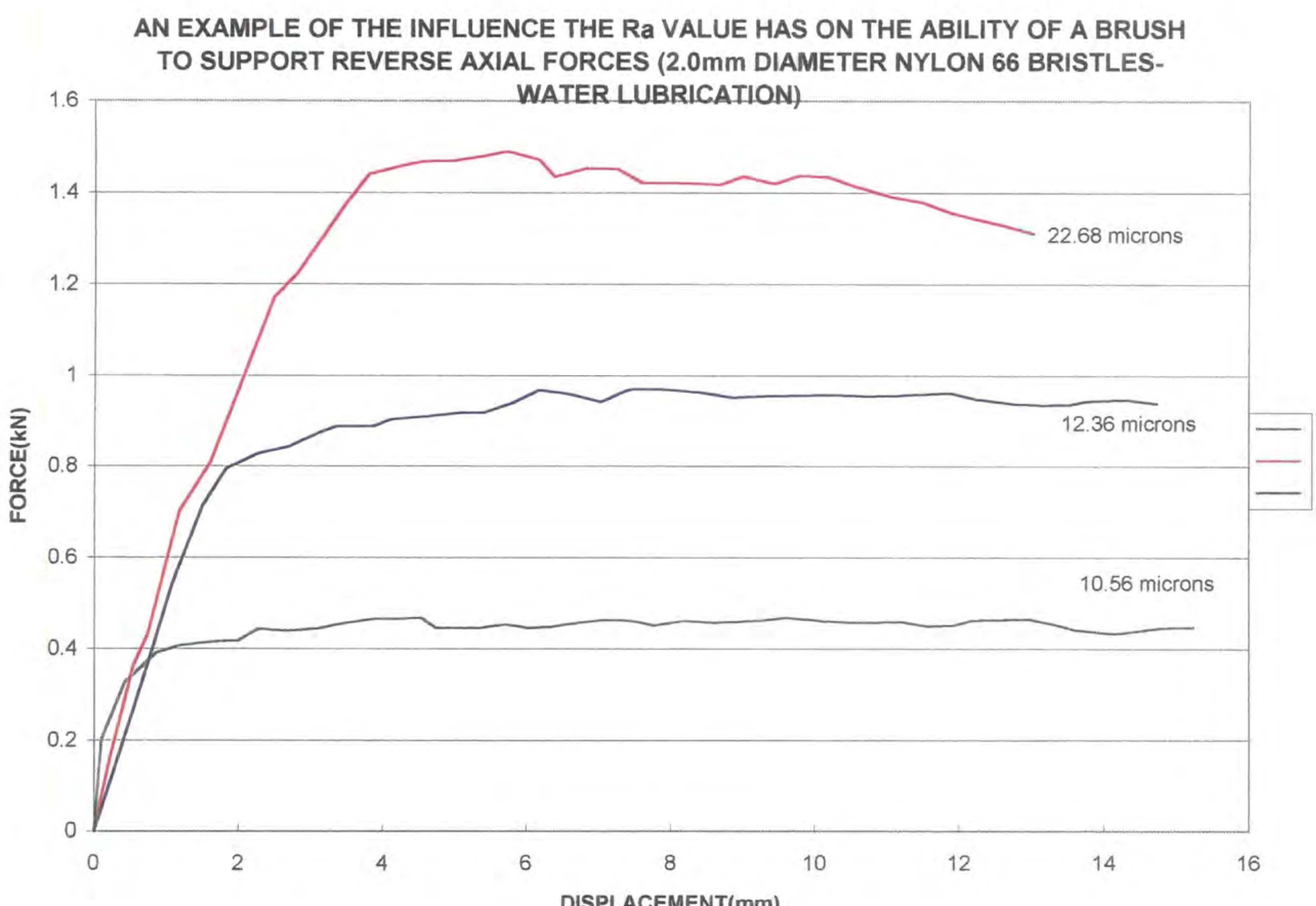


FIGURE 6.23
(RA-REV20.XLS)

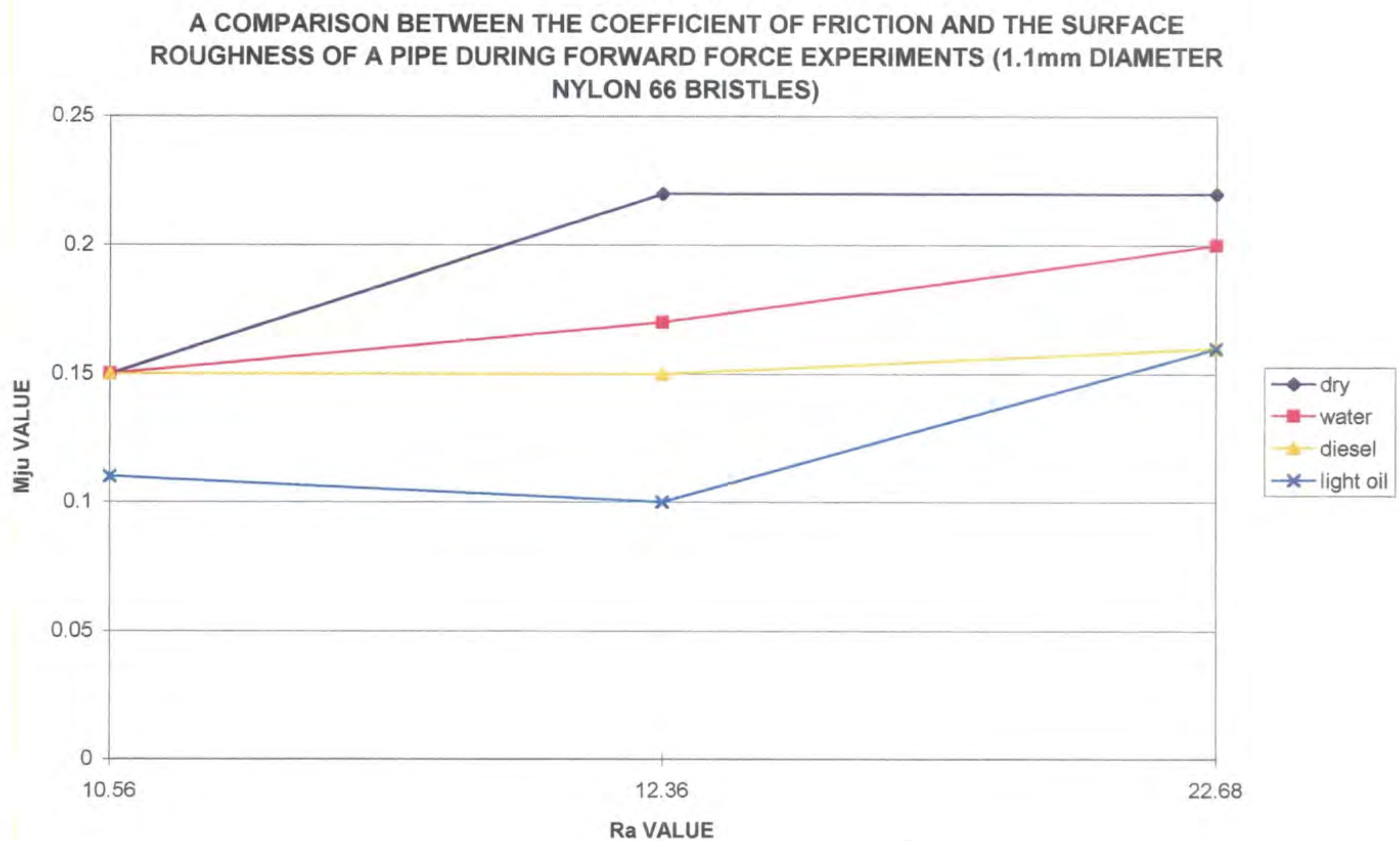

A COMPARISON BETWEEN THE COEFFICIENT OF FRICTION AND THE SURFACE ROUGHNESS OF A PIPE DURING FORWARD FORCE EXPERIMENTS (1.1mm DIAMETER NYLON 66 BRISTLES)
Mju VALUE
0.25
0.2
0.15
0.1
0.05
0
10.56
12.36
22.68
Ra VALUE
dry
water
diesel
light oil

FIGURE 6.24

A CREEP ANALYSIS OF NYLON 66 BRISTLES (0.7mm BRISTLE DIAMETER,160mm PIPE DIAMETER, DRY CONDITIONS)

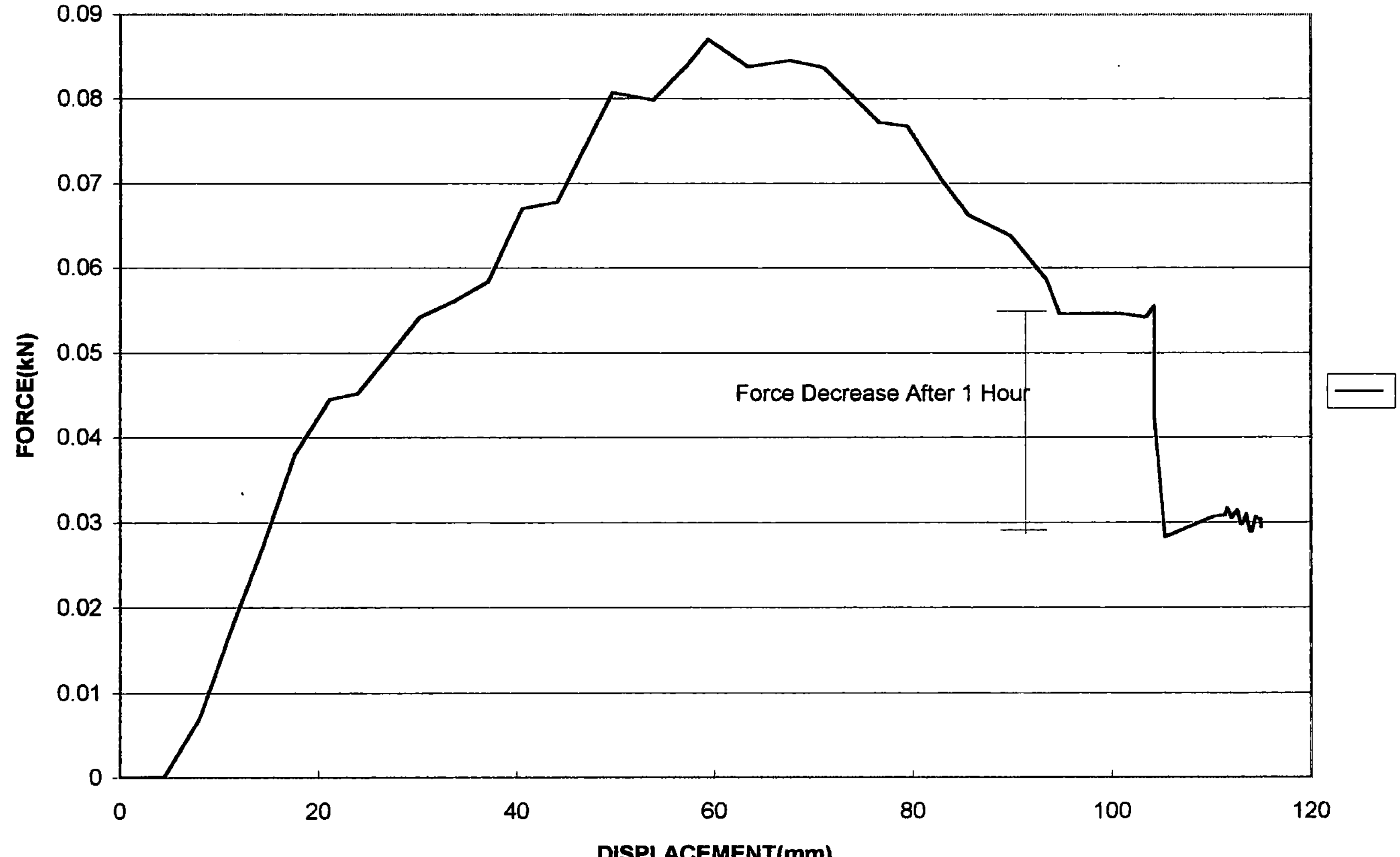


FIGURE 6.25
(CREEP07.XLS)

TEST TEMP. 16c

1 HOUR IN PIPE = 47.1%
REDUCTION IN AXIAL FORCE EXERTED

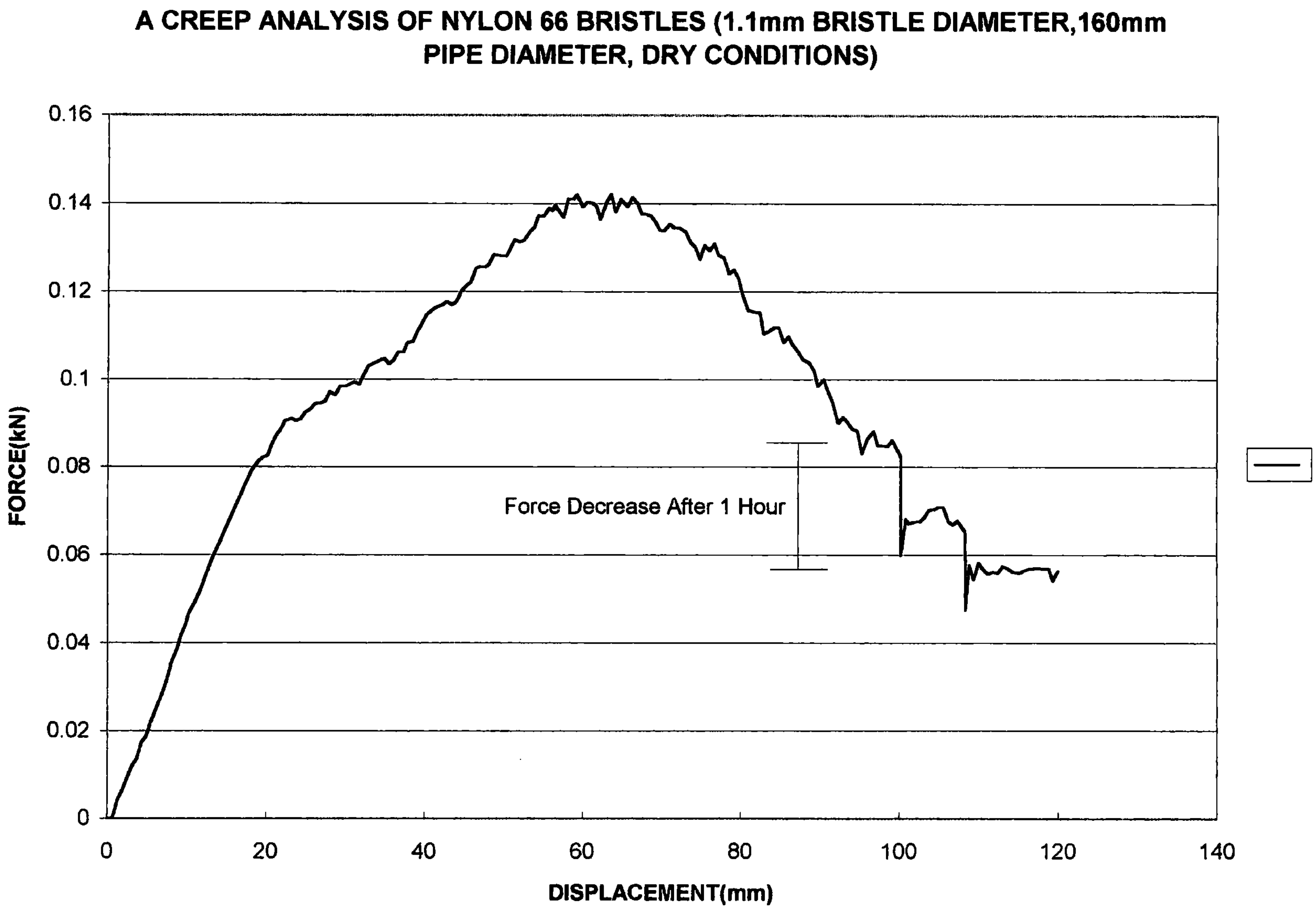


FIGURE 6.26
(CREEP11.XLS)

TEST TEMP. 16c

1 HOUR IN PIPE = 34.6%
REDUCTION IN AXIAL FORCE EXERTED

A CREEP ANALYSIS OF NYLON 66 BRISTLES (1.5mm BRISTLE DIAMETER,160mm PIPE DIAMETER, DRY CONDITIONS)

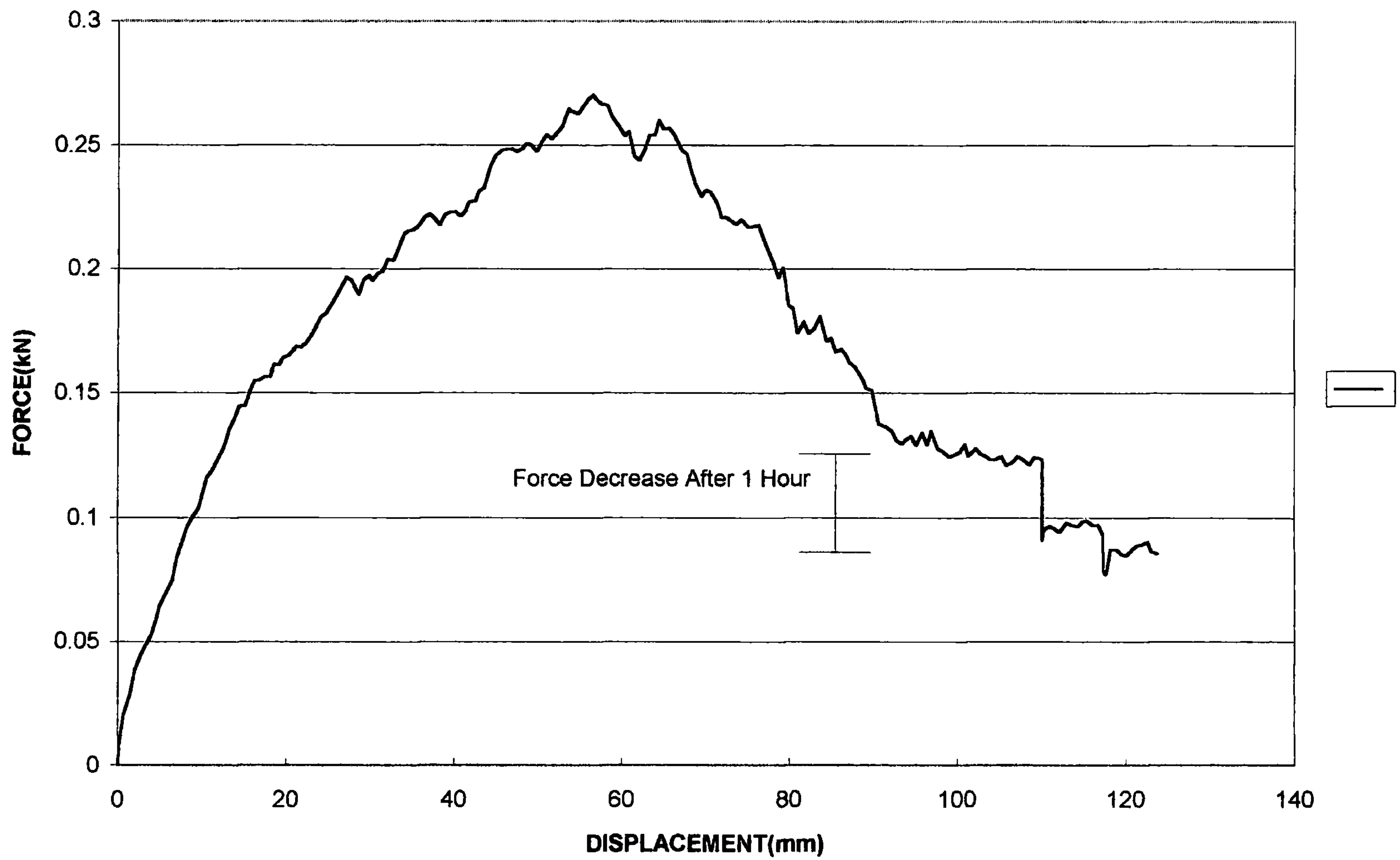


FIGURE 6.27
(CREEP15.XLS)

TEST TEMP. 16c

1 HOUR IN PIPE = 30.7%
REDUCTION IN AXIAL FORCE EXERTED

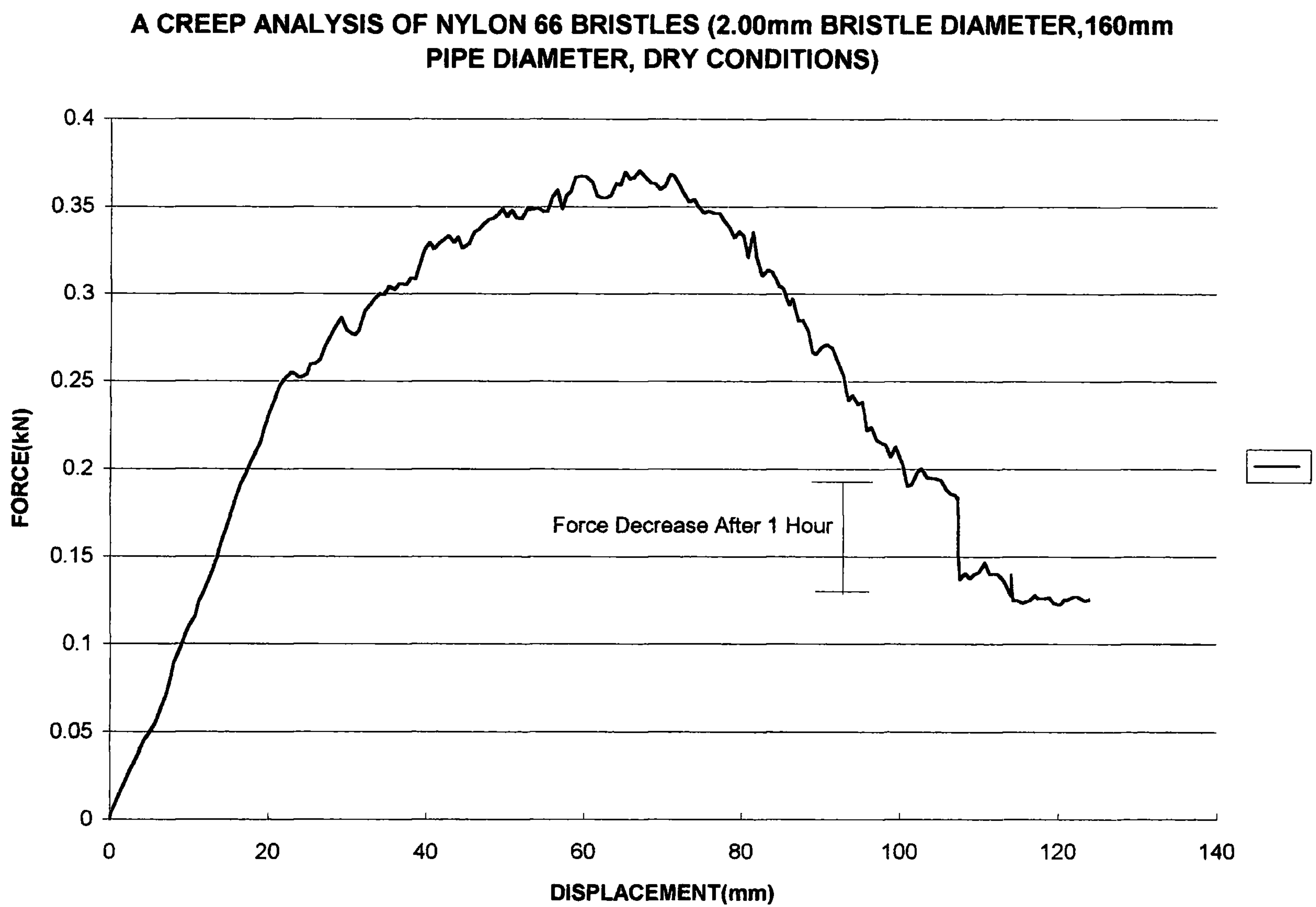


FIGURE 6.28
(CREEP20.XLS)

TEST TEMP. 16c

1 HOUR IN PIPE = 32.3% REDUCTION IN AXIAL FORCE EXERTED

20 HOUR CREEP ANALYSIS ON 2.00mm DIAMETER NYLON 66 BRISTLES,160mm INTERNAL DIAMETER PIPE)

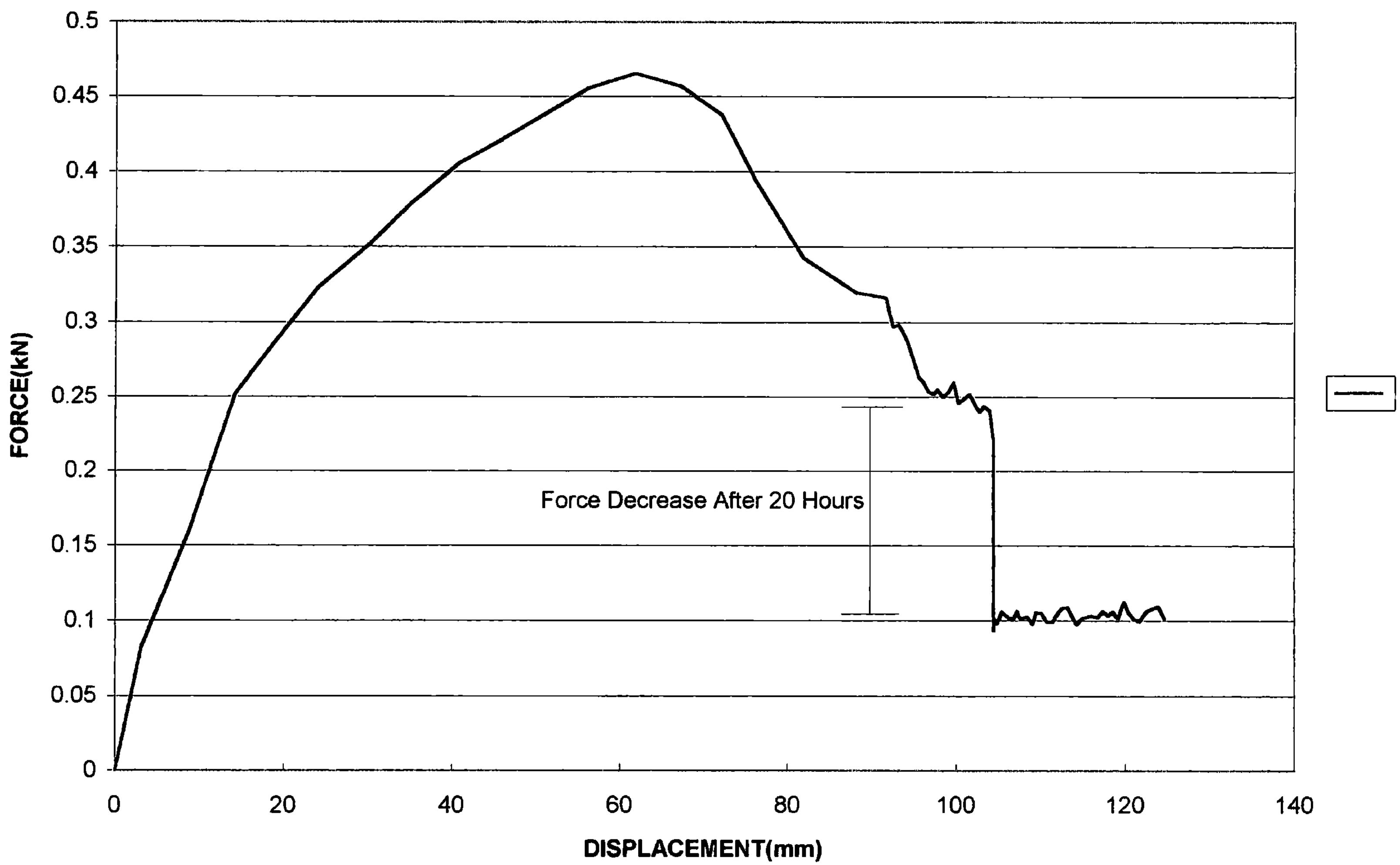


FIGURE 6.29
(16HR2000.XLS)

Av. TEMP. 16c

20 HOURS IN PIPE = 53.8%
REDUCTION IN AXIAL FORCE EXERTED

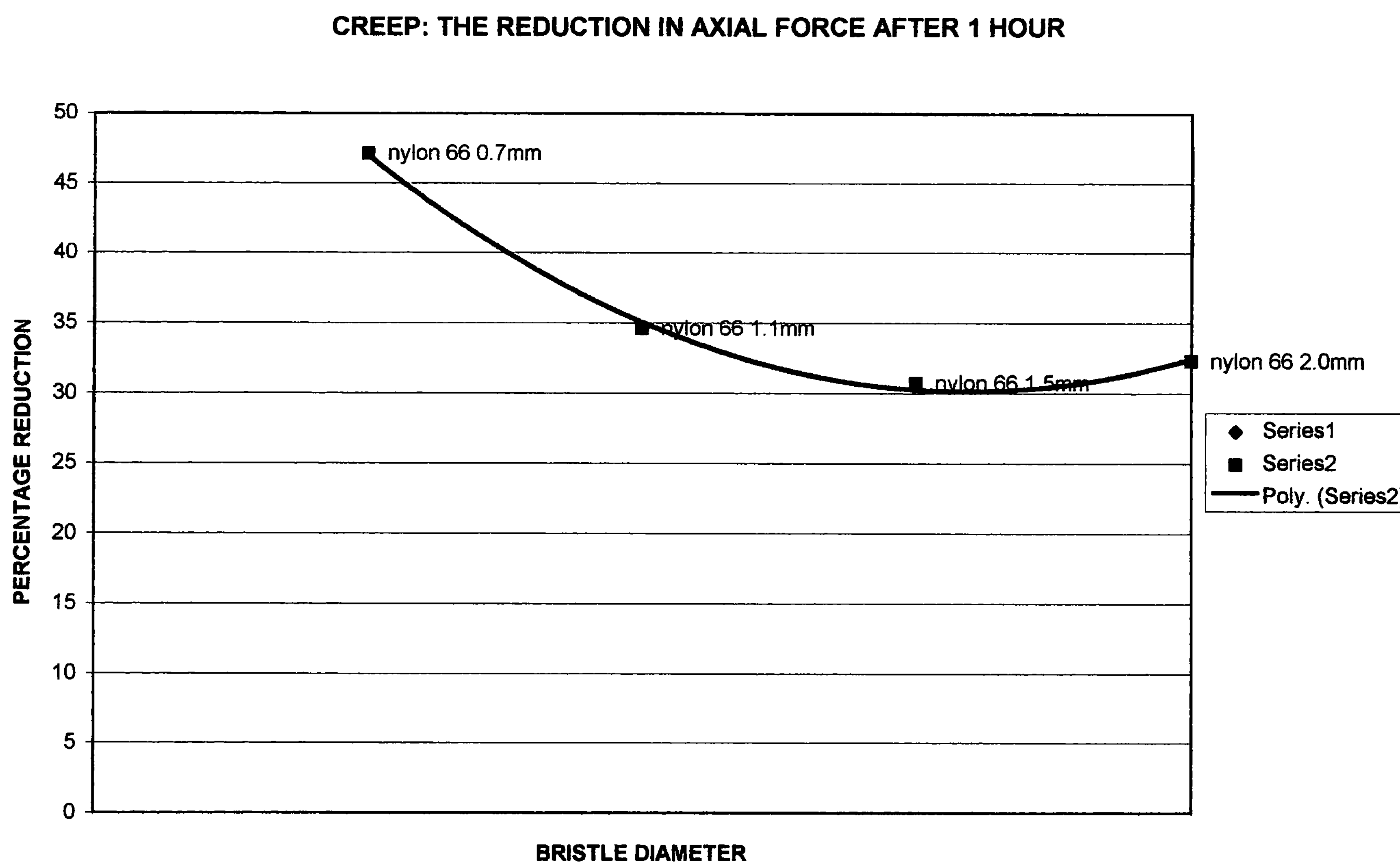

CREEP: THE REDUCTION IN AXIAL FORCE AFTER 1 HOUR
PERCENTAGE REDUCTION
50
45
40
35
30
25
20
15
10
5
0
nylon 66 0.7mm
nylon 66 1.1mm
nylon 66 1.5mm
nylon 66 2.0mm
Series1
Series2
Poly. (Series2)
BRISTLE DIAMETER

FIGURE 6.30

A 16 HOUR CREEP ANALYSIS ON1.625mm DIAMETER NYLON 612 BRISTLES (160mm INTERNAL DIAMETER PIPE, DRY CONDITIONS)

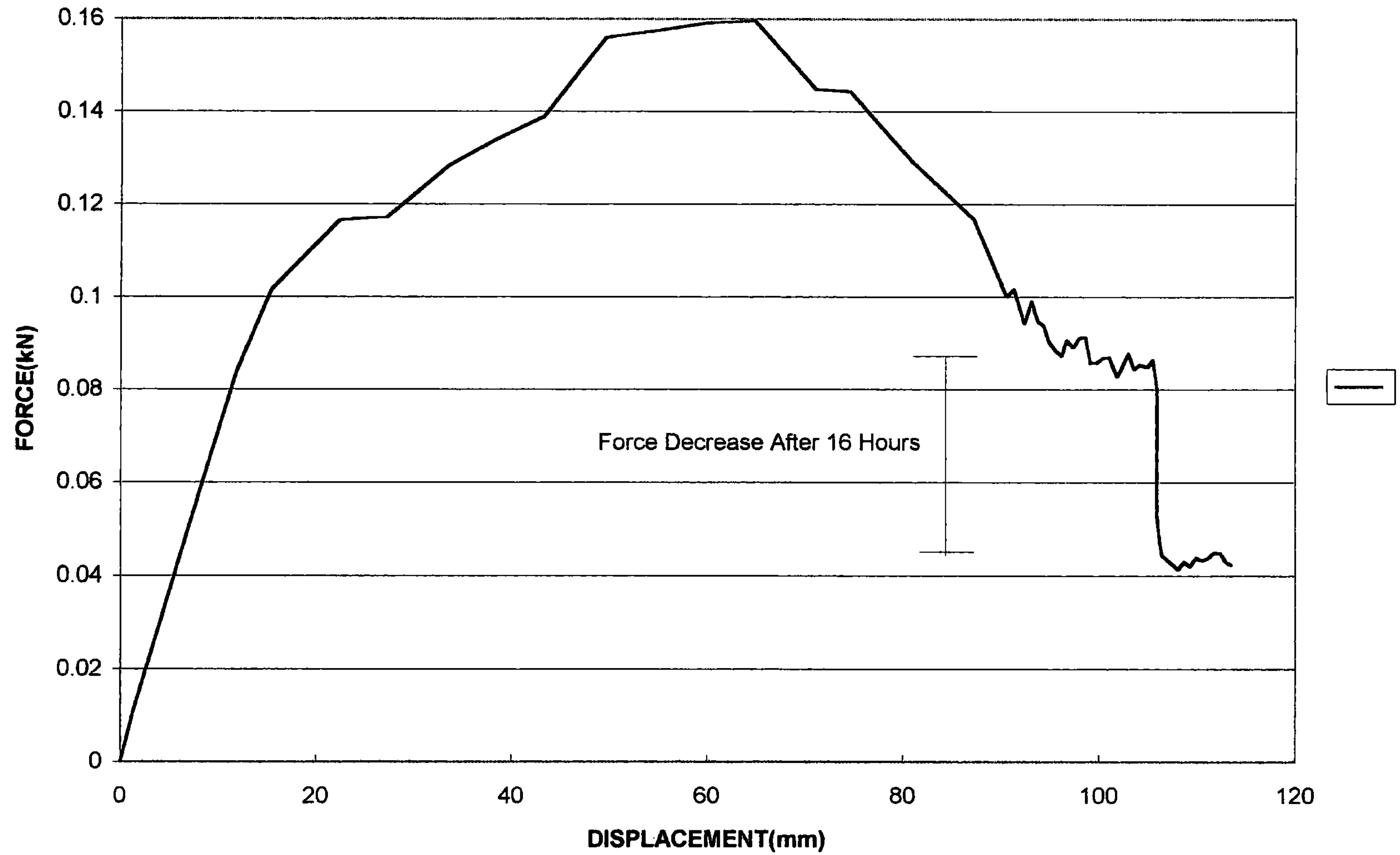


FIGURE 6.31
(16HR1625.XLS)

Av. TEMP. 16c

16 HOURS IN PIPE = 51.1%
REDUCTION IN AXIAL FORCE EXERTED

A COMPARISON OF NYLON 66 BRISTLES IN A PVCu PIPE (BRISTLE DIAMETER 0.7mm, DRY CONDITION)

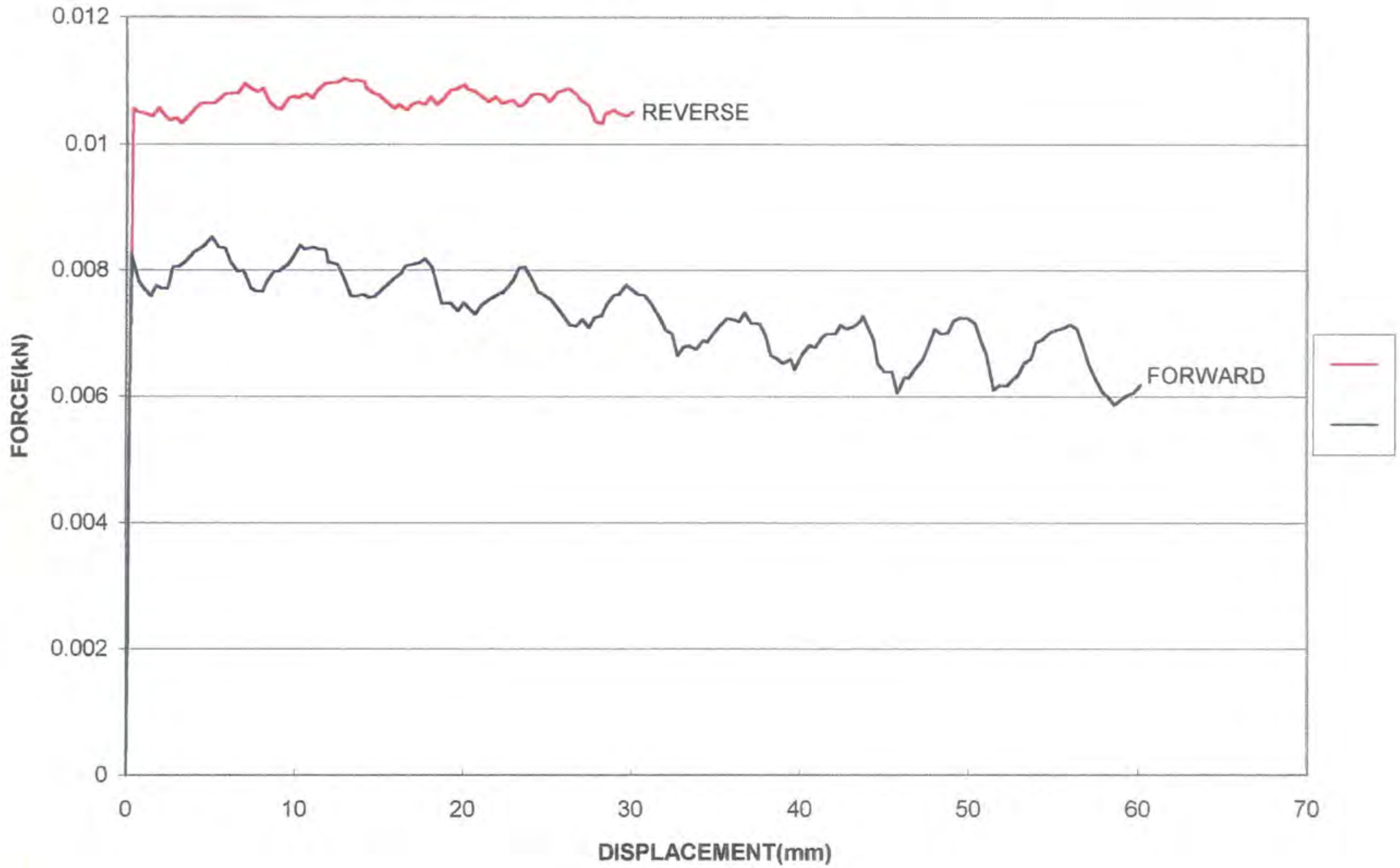


FIGURE 6.32
(07PVCFR.XLS)

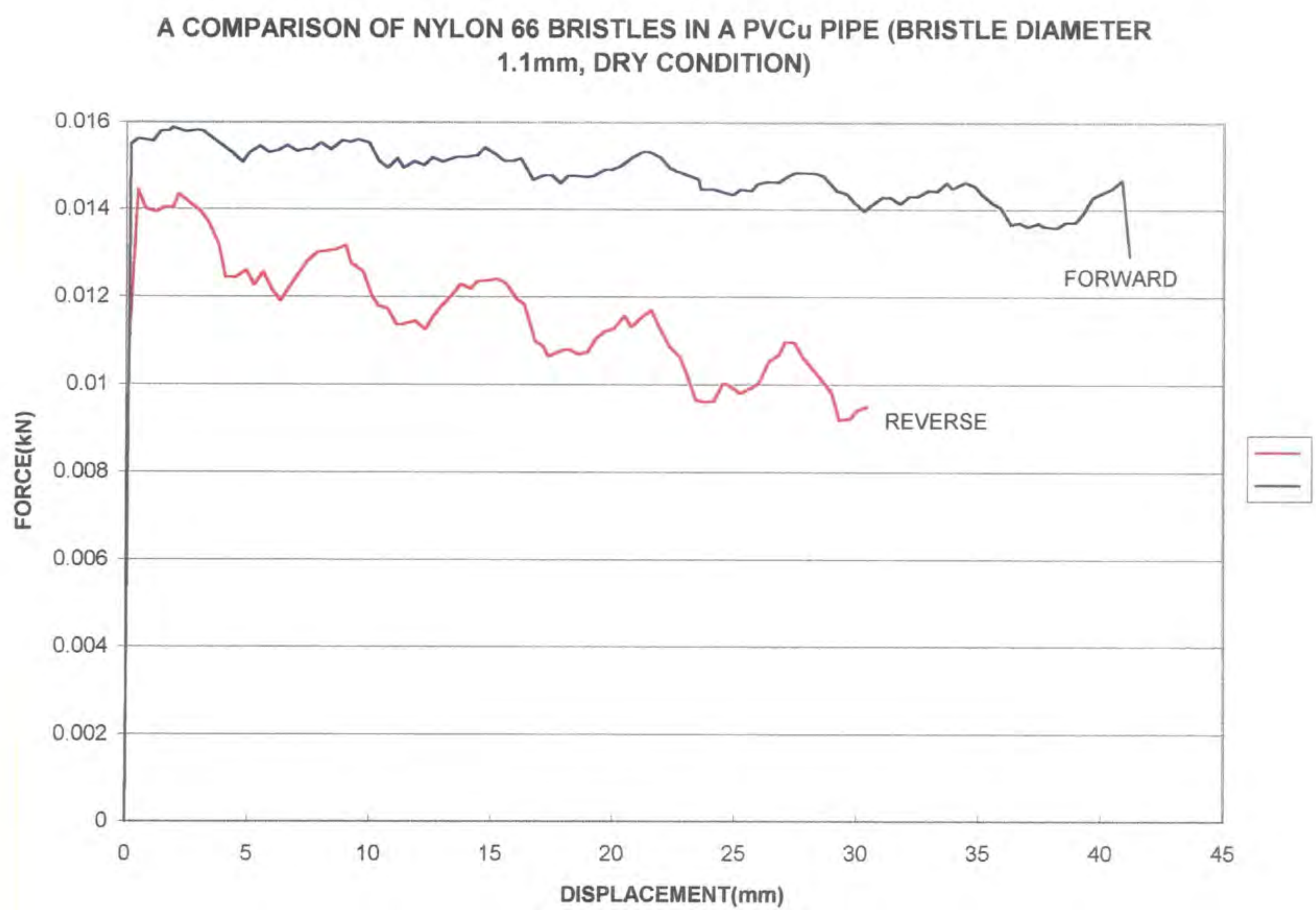


FIGURE 6.33
(11PVCFR.XLS)

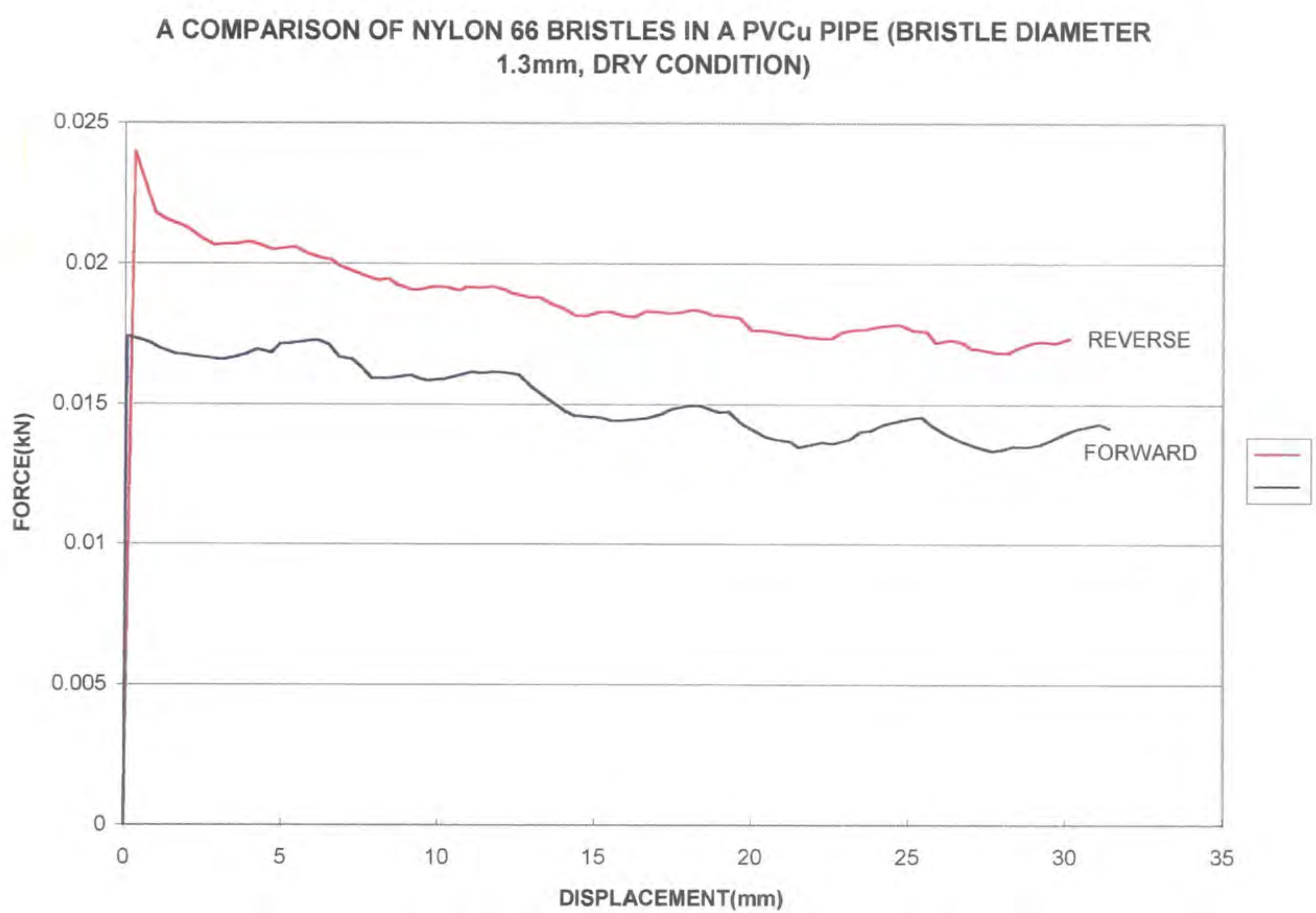


FIGURE 6.34
(13PVCFR.XLS)

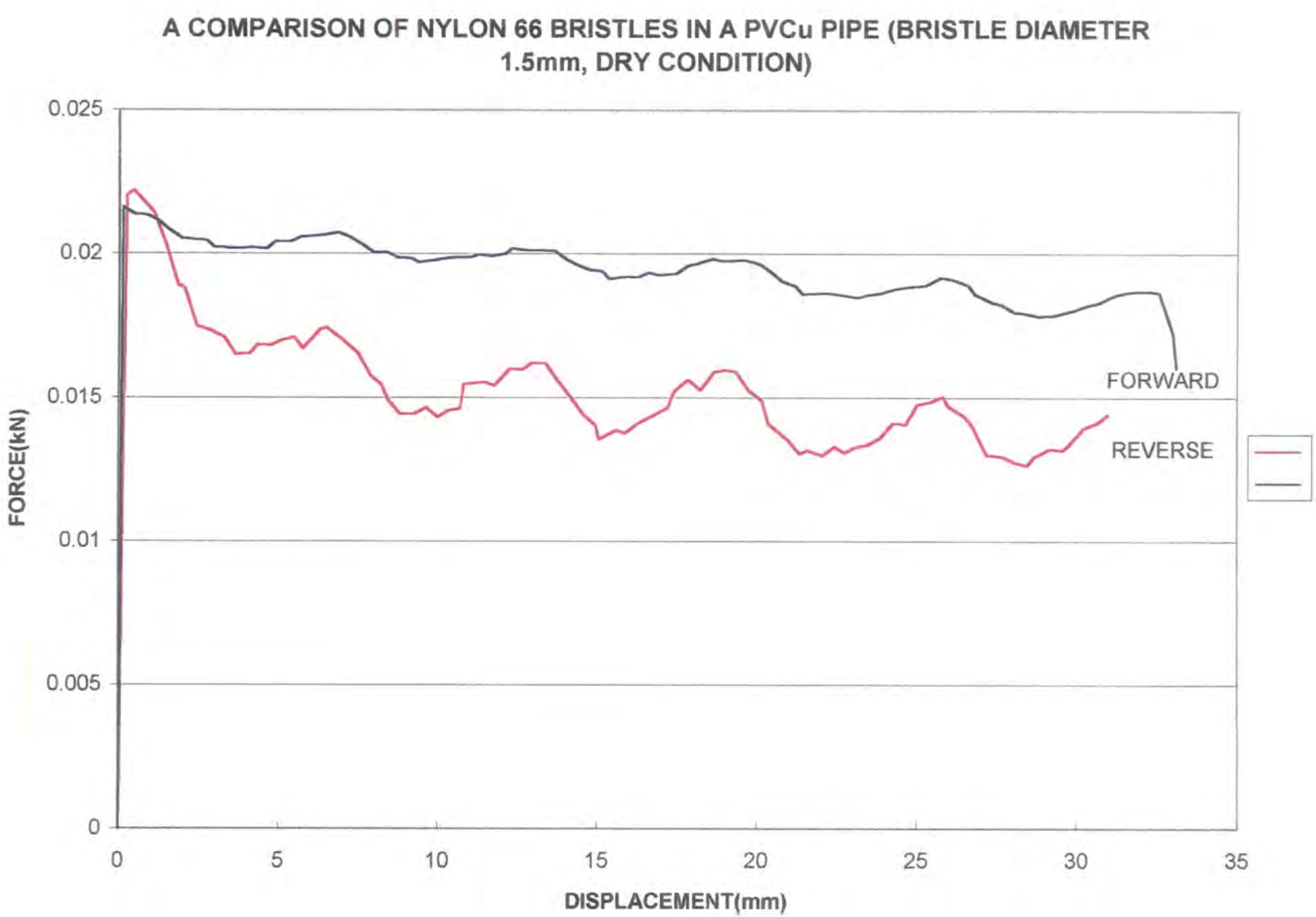


FIGURE 6.35
(15PVCFR.XLS)

A COMPARISON OF NYLON 612 BRISTLES IN PVCu PIPE (BRISTLE DIAMETER 1.625mm, DRY CONDITION)

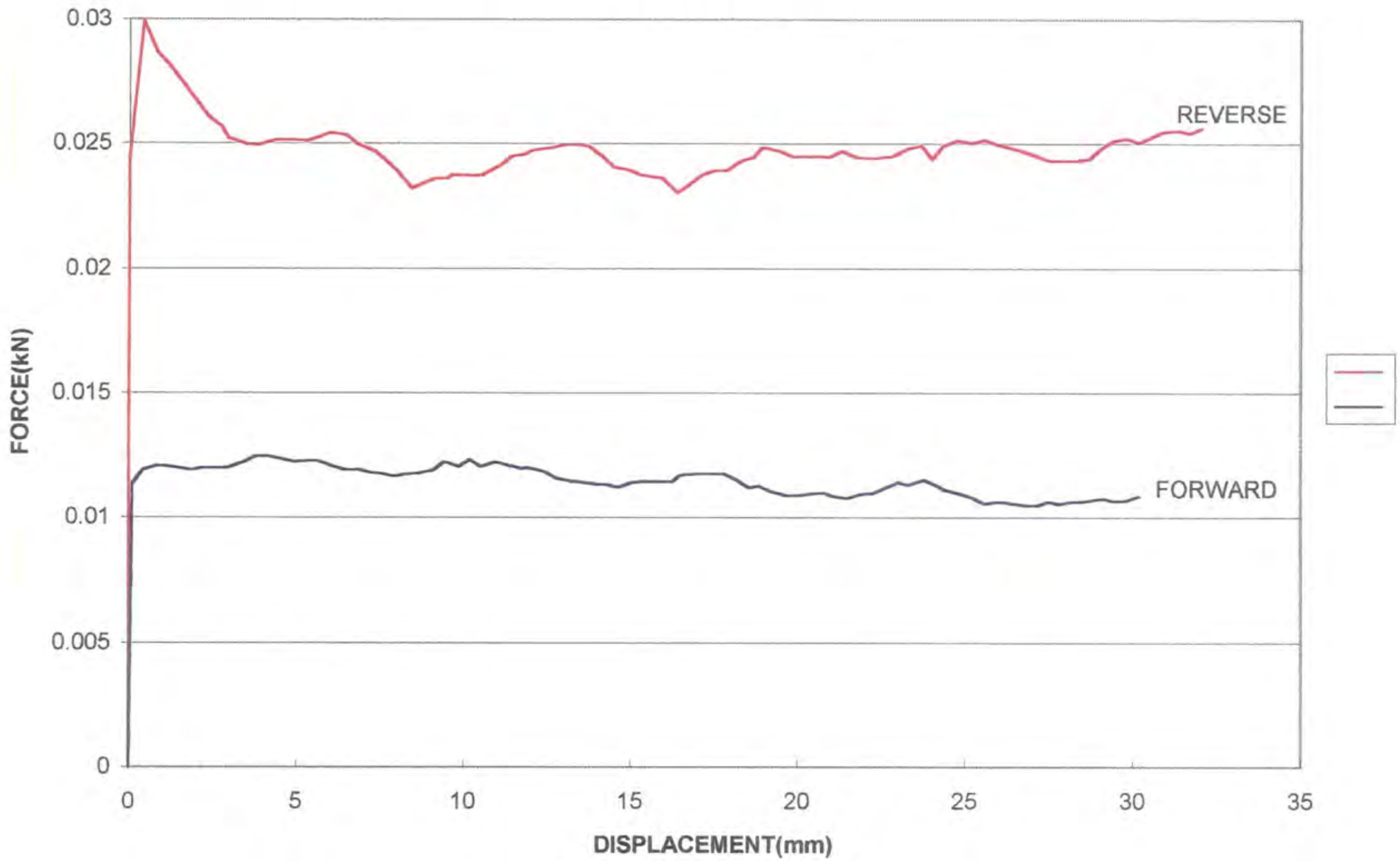


FIGURE 6.36
(16PVCFR.XLS)

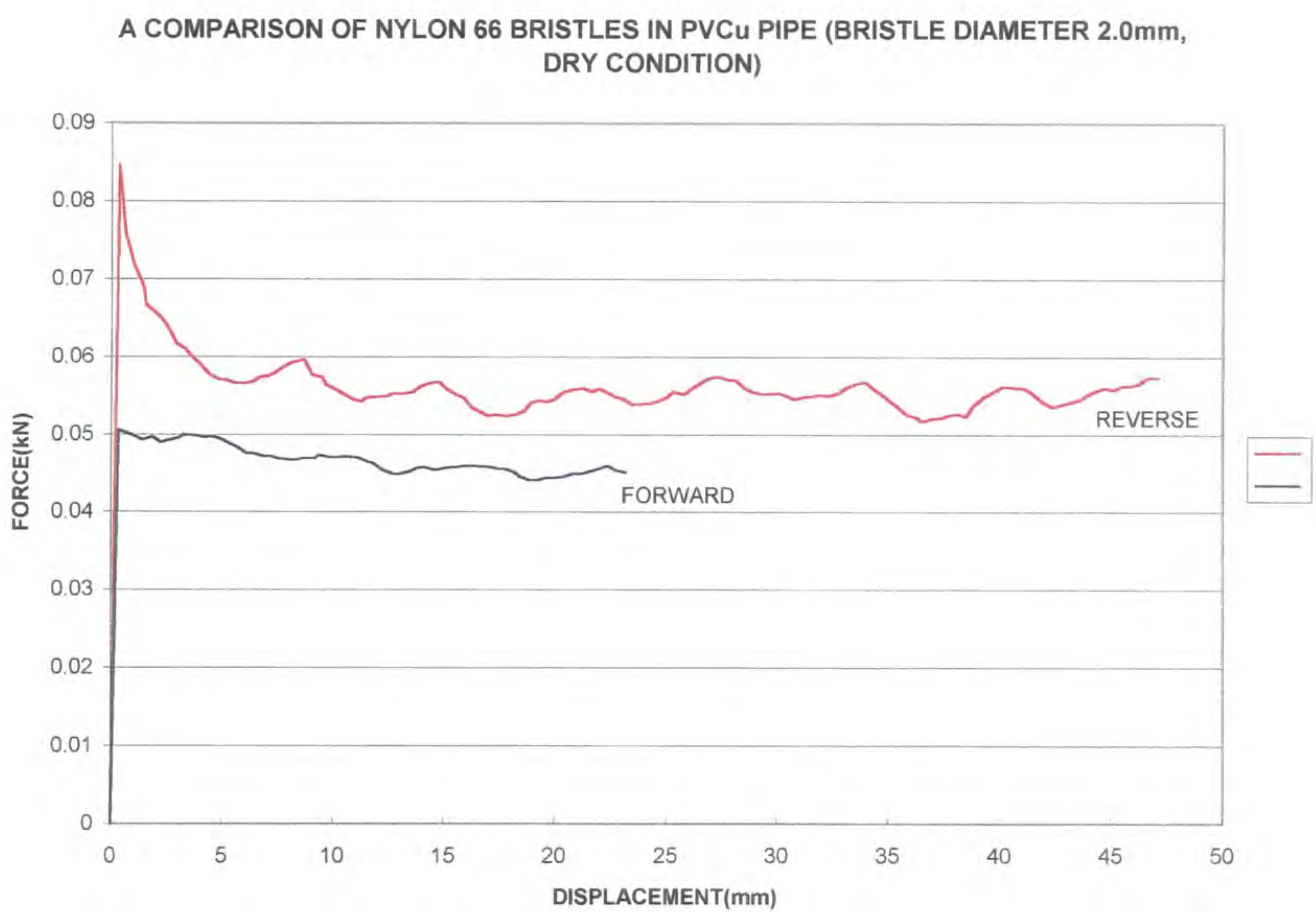


FIGURE 6.37
(20PVCFR.XLS)

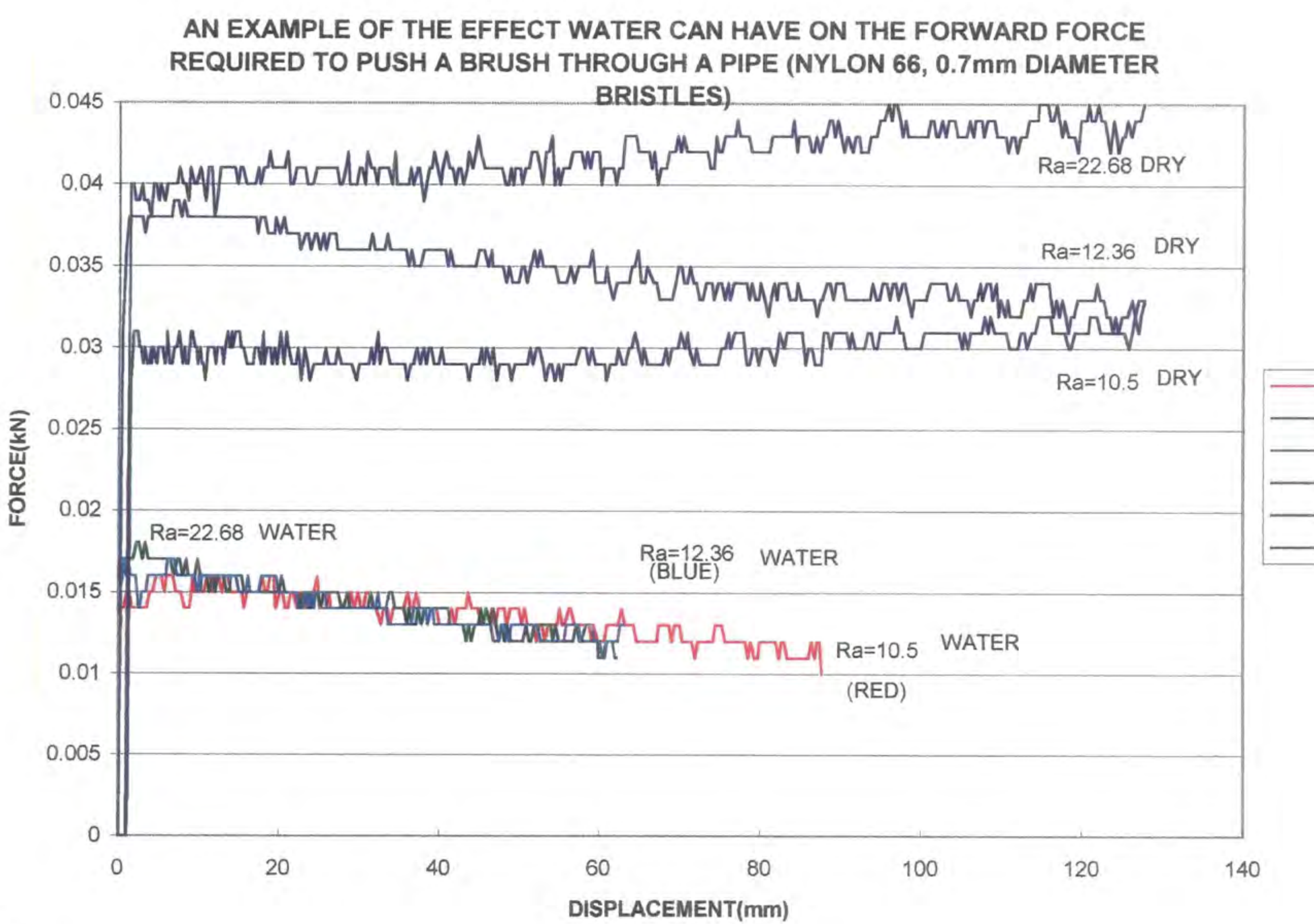


FIGURE 6.38
(07FCVH20.XLS)

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "OLD15A" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	0.943	0.612	0.585	0.599	19'C			0.33	0.31	0.32	53.2
149mm DRY	38.84	1.303	0.947	0.848	0.916	18'C			0.6	0.54	0.59	50.7
144mm DRY	38.84	1.568	0.987	0.948	0.968	18'C			0.72	0.69	0.7	48.3
139mm DRY	38.84	2.338	1.081	0.999	1.05	19'C			0.87	0.8	0.84	45.7
155mm DIESEL	4.47	0.89	0.548	0.501	0.526	18'C	4		0.29	0.27	0.28	53.2
149mm DIESEL	38.84	1.113	0.705	0.582	0.66	17'C	4		0.45	0.37	0.42	50.7
144mm DIESEL	38.84	1.513	0.855	0.736	0.812	16'C	4		0.62	0.53	0.59	48.3
139mm DIESEL	38.84	2.32	0.944	0.743	0.888	18'C	4		0.78	0.6	0.71	45.7
155mm GTX	4.47	0.869	0.536	0.489	0.511	17'C	104		0.29	0.26	0.27	53.2
149mm GTX	38.84	1.064	0.699	0.529	0.618	18'C	104		0.45	0.34	0.4	50.7
144mm GTX	38.84	1.525	0.797	0.644	0.738	18'C	104		0.58	0.47	0.53	48.3
139mm GTX	38.84	2.262	0.841	0.65	0.769	17'C	104		0.67	0.52	0.62	45.7
155mm FUEL OIL	4.47	0.801	0.444	0.398	0.423	17'C	975		0.24	0.21	0.73	53.2
149mm FUEL OIL	38.84	1.12	0.692	0.566	0.615	17'C	975		0.44	0.38	0.4	50.7
144mm FUEL OIL	38.84	1.451	0.709	0.657	0.684	17'C	975		0.51	0.48	0.5	48.3
139mm FUEL OIL	38.84	2.03	0.762	0.714	0.737	16'C	975		0.61	0.57	0.59	45.7

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.538				19'C		SLIPPAGE/OFFSET				53.2		4.237062
149mm DRY	38.84	3.951				17'C						50.7	CW	4.313319
144mm DRY	38.84	4.262				18'C						48.3	CW	4.402893
139mm DRY	38.84	6.415				19'C						45.7	CW	6.109524
155mm DIESEL	4.47	1.787				17'C	4	OFFSET & ROTATION				53.2	CW	3.397338
149mm DIESEL	38.84	3.768				17'C	4	OFFSET & ROTATION				50.7	CW	5.709091
144mm DIESEL	38.84	3.395				16'C	4	OFFSET & SLIPPAGE				48.3	CW	4.181034
139mm DIESEL	38.84	2.686				18'C	4	SLIPPAGE ONLY				45.7		3.031603
155mm GTX	4.47	2.359				17'C	104	SLIPPAGE & ROTATION				53.2	CW	4.616438
149mm GTX	38.84	3.577				18'C	104	SLIPPAGE & ROTATION				50.7	CW	5.788026
144mm GTX	38.84	2.592				18'C	104	SLIPPAGE ONLY				48.3		3.512195
139mm GTX	38.84	1.526				17'C	104	SLIPPAGE ONLY				45.7		1.984395
155mm FUEL OIL	4.47	11.608				16'C	975					53.2	ACW	3.801418
149mm FUEL OIL	38.84	2.601				17'C	975					50.7	CW	4.229268
144mm FUEL OIL	38.84	1.335				16'C	975	SLIPPAGE ONLY				48.3		1.951754
139mm FUEL OIL	38.84	1.113				17'C	975	SLIPPAGE ONLY				45.7		1.510176

FIGURE 6.39

VISCOSITY IN CENTISTOKES AT 40'C
RATIO=P AV./P MAX

TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "OLD15B" STEEL BRISTLED BRUSH.

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.L LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	0.94	0.601	0.574	0.586	19'C			0.32	0.31	0.31	53.2
149mm DRY	38.84	1.27	0.83	0.728	0.785	17'C			0.53	0.47	0.5	50.7
144mm DRY	38.84	1.722	0.973	0.921	0.946	18'C			0.71	0.67	0.69	48.3
139mm DRY	38.84	2.496	1.119	1.026	1.069	19'C			0.9	0.82	0.86	45.7
155mm DIESEL	4.47	0.898	0.5	0.463	0.481	17'C	4		0.27	0.25	0.26	53.2
149mm DIESEL	38.84	1.262	0.743	0.594	0.683	17'C	4		0.48	0.38	0.44	50.7
144mm DIESEL	38.84	1.705	0.855	0.759	0.819	16'C	4		0.62	0.55	0.59	48.3
139mm DIESEL	38.84	2.466	1.025	0.857	0.985	18'C	4		0.82	0.69	0.79	45.7
155mm GTX	4.47	0.829	0.438	0.378	0.412	18'C	104		0.23	0.2	0.22	53.2
149mm GTX	38.84	1.072	0.653	0.519	0.595	18'C	104		0.42	0.33	0.38	50.7
144mm GTX	38.84	1.529	0.72	0.589	0.672	16'C	104		0.52	0.43	0.49	48.3
139mm GTX	38.84	2.311	0.724	0.537	0.641	17'C	104		0.58	0.43	0.51	45.7
155mm FUEL OIL	4.47	0.819	0.409	0.354	0.383	16'C	975		0.22	0.19	0.2	53.2
149mm FUEL OIL	38.84	1.19	0.686	0.62	0.659	18'C	975		0.44	0.4	0.42	50.7
144mm FUEL OIL	38.84	1.428	0.688	0.65	0.671	16'C	975		0.5	0.47	0.49	48.3
139mm FUEL OIL	38.84	2.109	0.752	0.715	0.734	16'C	975		0.6	0.57	0.59	45.7

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.883				18'C		SLIPPAGE/OFFSET				53.2		4.9198
149mm DRY	38.84	5.96				18'C		SLIPPAGE/OFFSET				50.7		7.59236
144mm DRY	38.84	3.989				18'C						48.3	ACW	4.2167
139mm DRY	38.84	7.23				18'C						45.7	CW	6.76333
155mm DIESEL	4.47	2.272				17'C	4	OFFSET & ROTATION				53.2	CW	4.72349
149mm DIESEL	38.84	4.762				17'C	4	OFFSET & SLIPPAGE				50.7	CW	6.97218
144mm DIESEL	38.84	4.204				16'C	4	OFFSET & SLIPPAGE				48.3	CW	5.13309
139mm DIESEL	38.84	2.348				18'C	4	SLIPPAGE ONLY				45.7		2.38376
155mm GTX	4.47	1.322				18'C	104	SLIPPAGE & ROTATION				53.2	ACW	3.20874
149mm GTX	38.84	3.397				18'C	104	SLIPPAGE & ROTATION				50.7	ACW	5.70924
144mm GTX	38.84	2.272				18'C	104	SLIPPAGE ONLY				48.3		3.38095
139mm GTX	38.84	1.458				18'C	104	SLIPPAGE ONLY				45.7		2.27457
155mm FUEL OIL	4.47	2.231				17'C	975					53.2	CW	5.82507
149mm FUEL OIL	38.84	4.972				18'C	975					50.7	CW	7.54476
144mm FUEL OIL	38.84	1.256				16'C	975	SLIPPAGE ONLY				48.3		1.87183
139mm FUEL OIL	38.84	1.021				17'C	975	SLIPPAGE ONLY				45.7		1.39101

FIGURE 6.40 VISCOSITY IN CENTISTOKES AT 40'C TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "NEW15A" STEEL BRISTLED BRUSH

LOAD AND PUSH FORWARD FORCE

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	0.895	0.595	0.551	0.567	18'C			0.32	0.3	0.3	53.2
149mm DRY	38.84	1.232	0.948	0.849	0.907	19'C			0.61	0.55	0.58	50.7
144mm DRY	38.84	1.623	1.104	1.044	1.074	18'C			0.8	0.78	0.78	48.3
139mm DRY	38.84	2.084	1.013	0.948	0.981	18'C			0.81	0.76	0.79	45.7
155mm DIESEL	4.47	0.899	0.553	0.528	0.541	17'C	4		0.3	0.28	0.29	53.2
149mm DIESEL	38.84	1.182	0.781	0.647	0.721	17'C	4		0.5	0.42	0.46	50.7
144mm DIESEL	38.84	1.553	0.912	0.839	0.885	16'C	4		0.66	0.61	0.64	48.3
139mm DIESEL	38.84	2.214	0.961	0.784	0.914	18'C	4		0.77	0.63	0.73	45.7
155mm GTX	4.47	0.842	0.476	0.424	0.451	17'C	104		0.25	0.23	0.24	53.2
149mm GTX	38.84	1.251	0.774	0.608	0.702	18'C	104		0.5	0.39	0.45	50.7
144mm GTX	38.84	1.495	0.798	0.648	0.738	17'C	104		0.58	0.47	0.54	48.3
139mm GTX	38.84	2.249	0.912	0.662	0.814	17'C	104		0.73	0.53	0.65	45.7
155mm FUEL OIL	4.47	0.809	0.443	0.391	0.419	17'C	975		0.24	0.21	0.22	53.2
149mm FUEL OIL	38.84	1.179	0.72	0.666	0.691	18'C	975		0.48	0.43	0.44	50.7
144mm FUEL OIL	38.84	1.374	0.702	0.668	0.681	16'C	975		0.51	0.48	0.49	48.3
139mm FUEL OIL	38.84	1.945	0.687	0.648	0.669	17'C	975		0.55	0.52	0.54	45.7

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.198				19'C		SLIPPAGE/OFFSET				53.2		3.873018
149mm DRY	38.84	5.644				18'C						50.7	CW	6.222712
144mm DRY	38.84	4.683				18'C						48.3	ACW	4.360335
139mm DRY	38.84	6.792				18'C						45.7	ACW	6.923547
155mm DIESEL	4.47	2.243				17'C	4	OFFSET & SLIPPAGE				53.2		4.146028
149mm DIESEL	38.84	3.333				17'C	4	OFFSET & SLIPPAGE				50.7	ACW	4.622748
144mm DIESEL	38.84	4				16'C	4	OFFSET & SLIPPAGE				48.3	ACW	4.519774
139mm DIESEL	38.84	2.074				18'C	4	SLIPPAGE ONLY				45.7		2.269147
155mm GTX	4.47	1.649				18'C	104	SLIPPAGE & ROTATION				53.2	ACW	3.656319
149mm GTX	38.84	4.378				18'C	104	SLIPPAGE & ROTATION				50.7	ACW	6.236467
144mm GTX	38.84	2.031				17'C	104	SLIPPAGE ONLY				48.3		2.752033
139mm GTX	38.84	1.57				18'C	104					45.7		1.928747
155mm FUEL OIL	4.47	2.129				17'C	975					53.2	CW	5.081146
149mm FUEL OIL	38.84	2.604				17'C	975					50.7	ACW	3.768452
144mm FUEL OIL	38.84	1.424				16'C	975	SLIPPAGE ONLY				48.3		2.091043
139mm FUEL OIL	38.84	0.875				16'C	975	SLIPPAGE ONLY				45.7		1.307922

FIGURE 6.41 VISCOSITY IN CENTISTOKES AT 40'C TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "NEW15B" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	0.871	0.575	0.533	0.555	19'C			0.31	0.29	0.3	53.2
149mm DRY	38.84	1.183	0.894	0.79	0.848	19'C			0.57	0.51	0.54	50.7
144mm DRY	38.84	1.609	1.086	1.048	1.067	18'C			0.79	0.76	0.77	48.3
139mm DRY	38.84	2.168	1.127	1.048	1.091	18'C			0.9	0.84	0.87	45.7
155mm DIESEL	4.47	0.88	0.515	0.48	0.501	17'C	4		0.28	0.26	0.27	53.2
149mm DIESEL	38.84	1.212	0.759	0.624	0.698	16'C	4		0.49	0.4	0.45	50.7
144mm DIESEL	38.84	1.504	0.827	0.748	0.803	16'C	4		0.6	0.54	0.58	48.3
139mm DIESEL	38.84	2.236	1.034	0.944	0.999	18'C	4		0.83	0.78	0.8	45.7
155mm GTX	4.47	0.868	0.51	0.454	0.483	18'C	104		0.27	0.24	0.26	53.2
149mm GTX	38.84	1.149	0.703	0.526	0.618	18'C	104		0.45	0.34	0.4	50.7
144mm GTX	38.84	1.468	0.788	0.616	0.711	16'C	104		0.57	0.45	0.52	48.3
139mm GTX	38.84	2.341	0.943	0.655	0.81	17'C	104		0.76	0.53	0.65	45.7
155mm FUEL OIL	4.47	0.814	0.467	0.416	0.441	17'C	975		0.25	0.22	0.24	53.2
149mm FUEL OIL	38.84	1.168	0.672	0.617	0.645	17'C	975		0.43	0.4	0.41	50.7
144mm FUEL OIL	38.84	1.415	0.702	0.664	0.684	16'C	975		0.51	0.48	0.5	48.3
139mm FUEL OIL	38.84	2.032	0.763	0.717	0.738	15'C	975		0.61	0.57	0.59	45.7

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.652				19'C		SLIPPAGE/OFFSET				53.2		4.778378
149mm DRY	38.84	7.332				19'C		SLIPPAGE/OFFSET				50.7		8.646226
144mm DRY	38.84	4.174				18'C						48.3	CW	3.911903
139mm DRY	38.84	3.873				18'C						45.7	CW	3.549954
155mm DIESEL	4.47	1.687				17'C	4	OFFSET & SLIPPAGE				53.2		3.367265
149mm DIESEL	38.84	3.663				16'C	4	OFFSET & SLIPPAGE				50.7	CW	5.247851
144mm DIESEL	38.84	3.348				16'C	4	OFFSET & SLIPPAGE				48.3	CW	4.169365
139mm DIESEL	38.84	2.228				18'C	4	SLIPPAGE ONLY				45.7		2.23023
155mm GTX	4.47	2.274				18'C	104	SLIPPAGE & ROTATION				53.2	CW	4.708075
149mm GTX	38.84	2.819				17'C	104	SLIPPAGE & ROTATION				50.7	CW	4.561489
144mm GTX	38.84	1.845				16'C	104	SLIPPAGE ONLY				48.3		2.594937
139mm GTX	38.84	1.75				17'C	104	SLIPPAGE ONLY				45.7		2.160494
155mm FUEL OIL	4.47	2.175				17'C	975					53.2	CW	4.931973
149mm FUEL OIL	38.84	2.817				17'C	975					50.7	CW	4.367442
144mm FUEL OIL	38.84	1.369				16'C	975	SLIPPAGE ONLY				48.3		2.001462
139mm FUEL OIL	38.84	1.1				16'C	975	SLIPPAGE ONLY				45.7		1.490515

FIGURE 6.42

VISCOSITY IN CENTISTOKES AT 40'C

TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "NEW0A" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.L LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	1.649	0.897	0.811	0.845	19'C			0.45	0.41	0.43	53.6
149mm DRY	38.84	2.284	1.301	1.142	1.229	18'C			0.8	0.7	0.75	51.1
144mm DRY	38.84	3.219	2.226	1.495	1.671	18'C			1.57	1.05	1.18	48.8
139mm DRY	38.84	3.622	1.515	1.421	1.466	18'C			1.2	1.13	1.16	46.2
155mm DIESEL	4.47	1.836	0.782	0.726	0.761	17'C	4		0.4	0.37	0.39	53.6
149mm DIESEL	38.84	2.638	1.122	0.891	1.012	16'C	4		0.69	0.55	0.62	51.1
144mm DIESEL	38.84	3.23	1.222	1.025	1.166	16'C	4		0.86	0.72	0.82	48.8
139mm DIESEL	38.84	4.458	1.376	1.135	1.309	17'C	4		1.09	0.9	1.04	46.2
155mm GTX	4.47	1.873	0.879	0.617	0.646	18'C	104		0.34	0.31	0.33	53.6
149mm GTX	38.84	2.501	0.965	0.685	0.829	17'C	104		0.59	0.42	0.51	51.1
144mm GTX	38.84	3.116	0.988	0.769	0.874	16'C	104		0.7	0.54	0.62	48.8
139mm GTX	38.84	4.202	1.158	0.865	1.039	17'C	104		0.92	0.69	0.83	46.2
155mm FUEL OIL	4.47	1.663	0.66	0.581	0.615	17'C	975		0.33	0.29	0.31	53.6
149mm FUEL OIL	38.84	2.261	0.895	0.717	0.826	17'C	975		0.55	0.44	0.51	51.1
144mm FUEL OIL	38.84	2.627	0.913	0.851	0.877	16'C	975		0.64	0.6	0.62	48.8
139mm FUEL OIL	38.84	3.875	0.927	0.88	0.904	16'C	975		0.74	0.7	0.72	46.2

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.36				18'C		SLIPPAGE/OFFSET				53.6		2.7929
149mm DRY	38.84	6.517				18'C						51.1	CW	5.30269
144mm DRY	38.84	7.654				18'C						48.8	ACW	4.58049
139mm DRY	38.84	7.841				17'C						46.2	ACW	5.34857
155mm DIESEL	4.47	2.059				17'C	4	OFFSET & SLIPPAGE				53.6	ACW	2.70565
149mm DIESEL	38.84	4.413				16'C	4	OFFSET & SLIPPAGE				51.1	CW	4.36067
144mm DIESEL	38.84	4.11				16'C	4	OFFSET & SLIPPAGE				48.8	ACW	3.52487
139mm DIESEL	38.84	3.682				18'C	4	SLIPPAGE ONLY				46.2		2.81283
155mm GTX	4.47	2.096				18'C	104	SLIPPAGE & ROTATION				53.6	ACW	3.24458
149mm GTX	38.84	3.347				17'C	104	SLIPPAGE & ROTATION				51.1	ACW	4.03739
144mm GTX	38.84	2.119				16'C	104	SLIPPAGE ONLY				48.8		2.42449
139mm GTX	38.84	2.536				17'C	104	SLIPPAGE ONLY				46.2		2.44081
155mm FUEL OIL	4.47	2.262				17'C	975					53.6	ACW	3.67805
149mm FUEL OIL	38.84	3.363				17'C	975					51.1	ACW	4.07143
144mm FUEL OIL	38.84	1.607				16'C	975	SLIPPAGE ONLY				48.8		1.83238
139mm FUEL OIL	38.84	1.673				16'C	975	SLIPPAGE ONLY				46.2		1.85066

FIGURE 6.43 VISCOSITY IN CENTISTOKES AT 40'C TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "NEW0B" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	1.578	0.782	0.718	0.753	18'C			0.4	0.36	0.38	53.6
149mm DRY	38.84	2.145	1.162	0.989	1.076	18'C			0.71	0.61	0.66	51.1
144mm DRY	38.84	2.257	1.108	1.062	1.084	18'C			0.78	0.75	0.76	48.8
139mm DRY	38.84	3.76	1.473	1.278	1.414	17'C			1.17	1.02	1.12	46.2
155mm DIESEL	4.47	1.725	0.804	0.732	0.771	17'C	4		0.41	0.37	0.39	53.6
149mm DIESEL	38.84	2.142	0.998	0.828	0.928	16'C	4		0.61	0.51	0.57	51.1
144mm DIESEL	38.84	2.886	1.141	0.946	1.062	16'C	4		0.8	0.67	0.75	48.8
139mm DIESEL	38.84	3.778	1.312	0.998	1.192	17'C	4		1.04	0.79	0.95	46.2
155mm GTX	4.47	1.648	0.72	0.642	0.683	18'C	104		0.38	0.32	0.35	53.6
149mm GTX	38.84	2.306	0.942	0.729	0.829	17'C	104		0.58	0.45	0.51	51.1
144mm GTX	38.84	2.838	1.008	0.815	0.915	16'C	104		0.71	0.57	0.64	48.8
139mm GTX	38.84	3.921	1.115	0.857	1.003	17'C	104		0.89	0.68	0.8	46.2
155mm FUEL OIL	4.47	1.494	0.616	0.553	0.582	17'C	975		0.31	0.28	0.29	53.6
149mm FUEL OIL	38.84	2.146	0.891	0.759	0.841	17'C	975		0.55	0.47	0.52	51.1
144mm FUEL OIL	38.84	2.186	0.767	0.732	0.752	16'C	975		0.54	0.52	0.53	48.8
139mm FUEL OIL	38.84	3.584	0.944	0.902	0.92	16'C	975		0.75	0.72	0.73	46.2

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	3.626				18'C		SLIPPAGE/OFFSET				53.6	CW	4.815405
149mm DRY	38.84	5.557				18'C						51.1	ACW	5.164498
144mm DRY	38.84	5.511				19'C						48.8	ACW	5.083948
139mm DRY	38.84	6.976				17'C						46.2	ACW	4.933522
155mm DIESEL	4.47	2.311				17'C	4	OFFSET & SLIPPAGE				53.6	ACW	2.997406
149mm DIESEL	38.84	5.398				16'C	4	OFFSET & SLIPPAGE				51.1	CW	5.81681
144mm DIESEL	38.84	3.841				16'C	4	OFFSET & SLIPPAGE				48.8	ACW	3.616761
139mm DIESEL	38.84	3.327				18'C	4	SLIPPAGE ONLY				46.2		2.791107
155mm GTX	4.47	2.057				18'C	104	SLIPPAGE & ROTATION				53.6	ACW	3.011713
149mm GTX	38.84	5.478				18'C	104					51.1	CW	6.607861
144mm GTX	38.84	1.915				16'C	104	SLIPPAGE ONLY				48.8		2.092896
139mm GTX	38.84	2.133				17'C	104	SLIPPAGE ONLY				46.2		2.12662
155mm FUEL OIL	4.47	2.062				17'C	975					53.6	CW	3.542955
149mm FUEL OIL	38.84	3.024				17'C	975					51.1	ACW	3.595719
144mm FUEL OIL	38.84	1.553				16'C	975	SLIPPAGE ONLY				48.8		2.06516
139mm FUEL OIL	38.84	1.582				16'C	975	SLIPPAGE ONLY				46.2		1.719565

FIGURE 6.44

VISCOSITY IN CENTISTOKES AT 40'C

TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE EXPERIMENTS UNDERTAKEN ON THE "NEWOX" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCEPF

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	1.77	1.109	0.961	1	18'C			0.51	0.44	0.46	54
149mm DRY	38.84	2.197	1.515	1.435	1.473	18'C			0.84	0.79	0.82	51.6
144mm DRY	38.84	2.688	1.98	1.719	1.769	19'C			1.26	1.09	1.12	49.3
139mm DRY	38.84	3.99	1.897	1.489	1.544	18'C			1.36	1.07	1.11	46.8
155mm DIESEL	4.47	1.763	0.984	0.874	0.905	17'C	4		0.45	0.4	0.41	54
149mm DIESEL	38.84	2.253	1.168	0.958	1.066	16'C	4		0.65	0.53	0.59	51.6
144mm DIESEL	38.84	2.994	1.398	1.277	1.327	16'C	4		0.89	0.75	0.84	49.3
139mm DIESEL	38.84	4.087	1.518	1.113	1.36	18'C	4		1.09	0.8	0.97	46.8
155mm GTX	4.47	1.785	0.78	0.687	0.721	18'C	104		0.36	0.31	0.33	54
149mm GTX	38.84	2.341	1.165	0.823	0.961	18'C	104		0.65	0.46	0.53	51.6
144mm GTX	38.84	2.873	1.199	0.981	1.097	18'C	104		0.76	0.62	0.7	49.3
139mm GTX	38.84	3.74	1.258	1.017	1.153	17'C	104		0.9	0.73	0.83	46.8
155mm FUEL OIL	4.47	1.594	0.754	0.675	0.712	17'C	975		0.34	0.31	0.33	54
149mm FUEL OIL	38.84	2.178	1.04	0.894	0.972	17'C	975		0.58	0.5	0.54	51.6
144mm FUEL OIL	38.84	2.529	1.071	1.019	1.038	17'C	975		0.68	0.65	0.66	49.3
139mm FUEL OIL	38.84	3.809	1.1	1.066	1.081	16'C	975		0.79	0.73	0.77	46.8

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.618				17'C		SLIPPAGE				54	ACW	2.618
149mm DRY	38.84	12.475				17'C						51.6	CW	8.469111
144mm DRY	38.84	10.889				19'C						49.3	CW	6.155455
139mm DRY	38.84	11.576				18'C						46.8	ACW	7.497409
155mm DIESEL	4.47	2.157				17'C	4	OFFSET & SLIPPAGE				54		2.383425
149mm DIESEL	38.84	5.064				16'C	4	OFFSET & SLIPPAGE				51.6	CW	4.750469
144mm DIESEL	38.84	4.209				16'C	4	OFFSET & SLIPPAGE				49.3	ACW	3.171816
139mm DIESEL	38.84	4.138				18'C	4	SLIPPAGE ONLY				46.8		3.042647
155mm GTX	4.47	2.301				18'C	104	SLIPPAGE & ROTATION				54		3.191401
149mm GTX	38.84	4.188				18'C	104	SLIPPAGE & ROTATION				51.6	ACW	4.35796
144mm GTX	38.84	3.809				17'C	104	SLIPPAGE ONLY				49.3	CW	3.472197
139mm GTX	38.84	2.638				17'C	104	SLIPPAGE ONLY				46.8		2.287944
155mm FUEL OIL	4.47	1.902				17'C	975					54	CW	2.671348
149mm FUEL OIL	38.84	4.587				17'C	975					51.6	ACW	4.719136
144mm FUEL OIL	38.84	2.094				17'C	975	SLIPPAGE ONLY				49.3		2.017341
139mm FUEL OIL	38.84	1.639				16'C	975	SLIPPAGE ONLY				46.8		1.516189

FIGURE 6.45 VISCOSITY IN CENTISTOKES AT 40'C TEST RIG VELOCITY=0.174mm/s

A SUMMARY OF THE RESULTS FOR THE LOADING FORCE, FORWARD FORCE AND REVERSAL FORCE UNDERTAKEN ON THE "NEW15X" STEEL BRISTLED BRUSH

LOAD AND FORWARD FORCE

PIPE	Ra VALUE	P.I. LOAD(kN)	P MAX(kN)	P MIN(kN)	P Av.(kN)	TEMP	VIS. cSt	COMMENTS(NOTE)	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE
155mm DRY	4.47	1.074	0.754	0.718	0.736	18'C			0.39	0.37	0.38	52.8
149mm DRY	38.84	1.534	1.299	1.247	1.278	19'C			0.8	0.77	0.79	50.2
144mm DRY	38.84	1.897	1.364	1.319	1.339	18'C			0.94	0.91	0.92	47.8
139mm DRY	38.84	2.348	1.431	1.32	1.382	18'C			1.08	1	1.05	45.1
155mm DIESEL	4.47	1.032	0.644	0.594	0.62	17'C	4		0.34	0.31	0.32	52.8
149mm DIESEL	38.84	1.455	0.939	0.801	0.895	17'C	4		0.58	0.49	0.55	50.2
144mm DIESEL	38.84	1.784	1.102	0.916	1.042	16'C	4		0.76	0.63	0.72	47.8
139mm DIESEL	38.84	2.413	1.258	0.991	1.183	17'C	4		0.95	0.75	0.89	45.1
155mm GTX	4.47	0.992	0.551	0.49	0.515	18'C	104		0.29	0.26	0.27	52.8
149mm GTX	38.84	1.461	0.974	0.668	0.816	18'C	104		0.6	0.41	0.5	50.2
144mm GTX	38.84	1.743	0.95	0.77	0.87	16'C	104		0.65	0.53	0.6	47.8
139mm GTX	38.84	2.323	1.102	0.818	0.964	17'C	104		0.83	0.62	0.73	45.1
155mm FUEL OIL	4.47	0.928	0.503	0.441	0.474	17'C	975		0.26	0.23	0.25	52.8
149mm FUEL OIL	38.84	1.378	0.838	0.752	0.809	17'C	975		0.52	0.46	0.5	50.2
144mm FUEL OIL	38.84	1.698	0.894	0.851	0.874	16'C	975		0.62	0.59	0.6	47.8
139mm FUEL OIL	38.84	2.182	0.883	0.842	0.862	16'C	975		0.67	0.64	0.65	45.1

REVERSAL FORCE

PIPE	Ra VALUE	P.R. LOAD(kN)	P MAX (kN)	P MIN (kN)	P Av. (kN)	TEMP	VIS. cSt	COMMENTS	MU MAX.	MU MIN.	MU Av.	BRISTLE ANGLE	ROTATION	RATIO
155mm DRY	4.47	2.728				18'C		EARLY COUPLE				52.8	ACW	3.706522
149mm DRY	38.84	4.549				19'C		EARLY COUPLE				50.2	ACW	3.559468
144mm DRY	38.84	4.134				18'C		EARLY COUPLE				47.8	ACW	3.087379
139mm DRY	38.84	8.769				18'C						45.1	ACW	6.345152
155mm DIESEL	4.47	3.608				17'C	4	OFFSET & SLIPPAGE				52.8		5.819355
149mm DIESEL	38.84	3.565				16'C	4	OFFSET & SLIPPAGE				50.2	ACW	3.98324
144mm DIESEL	38.84	4.082				16'C	4	OFFSET & SLIPPAGE				47.8	ACW	3.917466
139mm DIESEL	38.84	2.934				18'C	4	SLIPPAGE ONLY				45.1		2.480135
155mm GTX	4.47	1.721				18'C	104	SLIPPAGE & ROTATION				52.8	ACW	3.341748
149mm GTX	38.84	3.671				18'C	104	SLIPPAGE & ROTATION				50.2	ACW	4.498775
144mm GTX	38.84	3.731				18'C	104					47.8	ACW	4.288506
139mm GTX	38.84	2.025				17'C	104	SLIPPAGE ONLY				45.1		2.100622
155mm FUEL OIL	4.47	2.744				17'C	975					52.8	ACW	5.78903
149mm FUEL OIL	38.84	3.05				18'C	975					50.2	ACW	3.770087
144mm FUEL OIL	38.84	1.725				17'C	975	SLIPPAGE ONLY				47.8		1.973684
139mm FUEL OIL	38.84	1.38				17'C	975	SLIPPAGE ONLY				45.1		1.600928

FIGURE 6.46 VISCOSITY IN CENTISTOKES AT 40'C TEST RIG VELOCITY=0.174mm/s

A GRAPH SHOWING THE FORCE REQUIRED TO LOAD A STEEL BRISTLED RADIAL BRUSH (NEW0A) INTO FOUR DIFFERENT ID PIPES UNDER DRY CONDITIONS

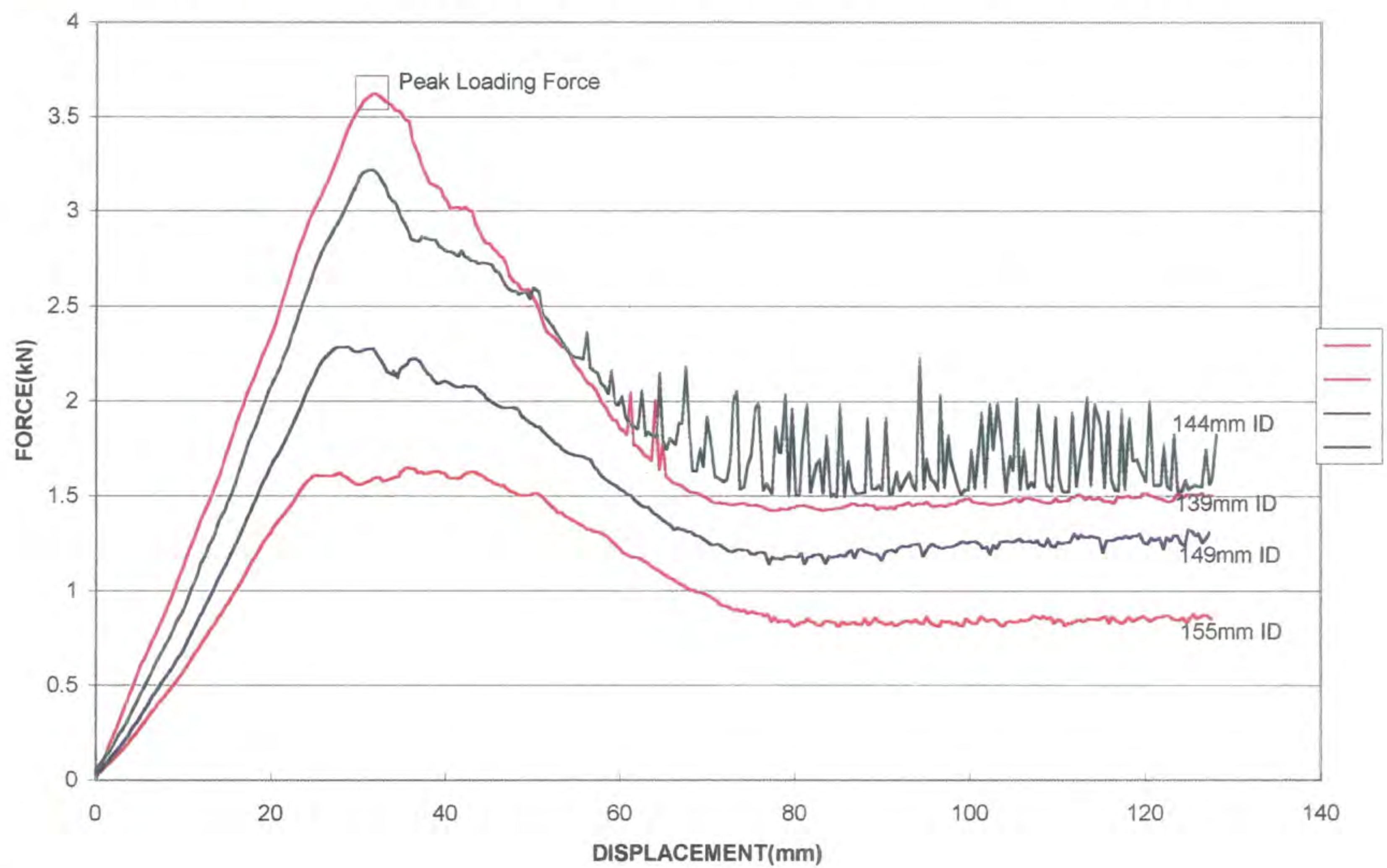
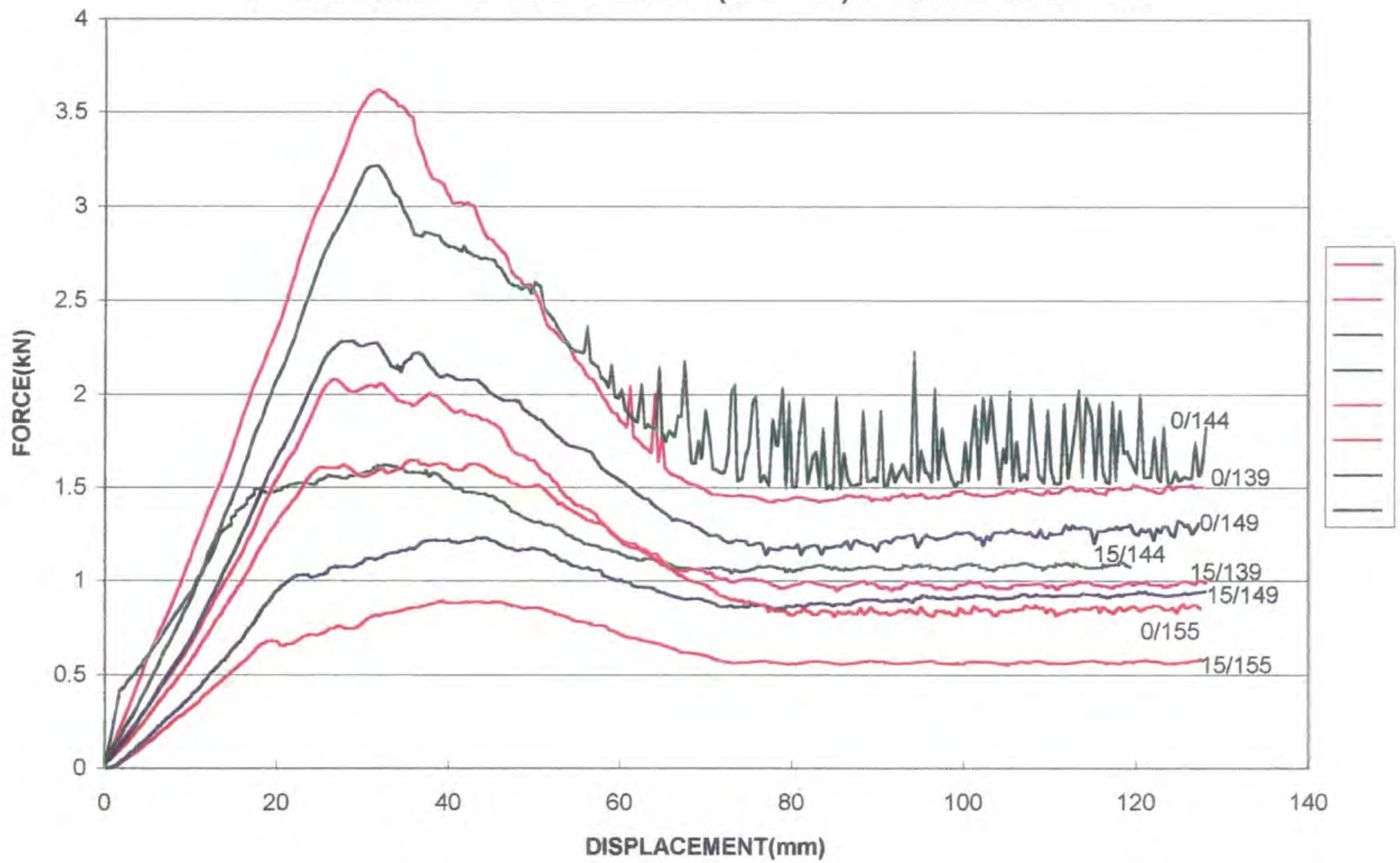


FIGURE 6-47

A GRAPH SHOWING THE FORCE REQUIRED TO LOAD A STEEL BRISTLED RADIAL BRUSH (NEW0A) INTO A 155mm ID PIPE UNDER DRY, DIESEL, LIGHT OIL AND FUEL OIL LUBRICATION

FIGURE 6.48

A GRAPH SHOWING THE FORCE REQUIRED TO LOAD A STEEL BRISTLED RADIAL BRUSH (NEW0A) INTO A DRY PIPE COMPARED TO THE FORCE REQUIRED TO LOAD A 15 DEGREE PRE-SWEPT BRUSH (NEW15A) INTO THE SAME PIPE



"0/139" = A RADIAL BRUSH BEING INSERTED INTO A 139mm ID PIPE
"15/139" = A 15 DEGREE PRE-SWEPT BRUSH BEING INSERTED INTO A 139mm ID PIPE

FIGURE 6.49

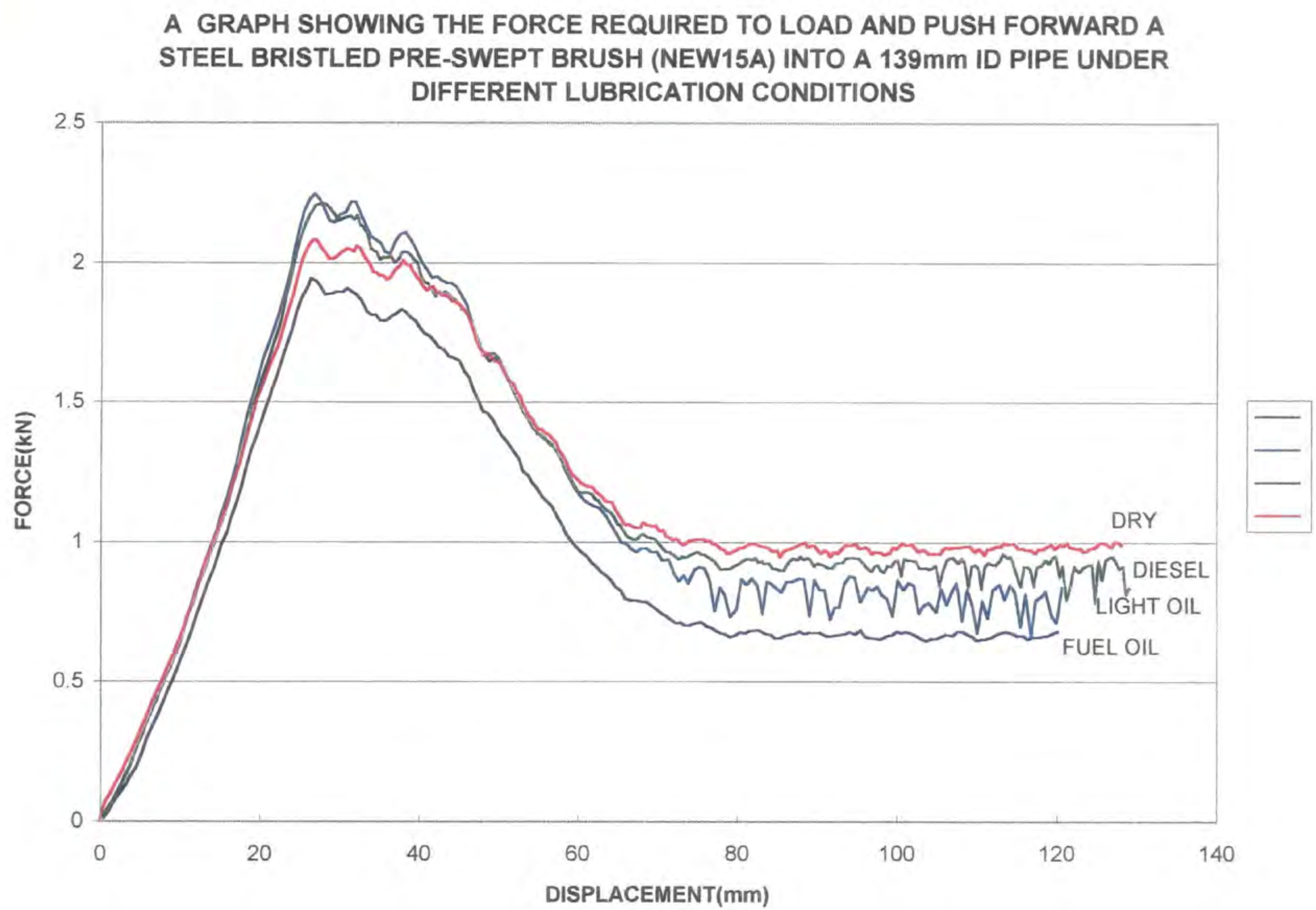
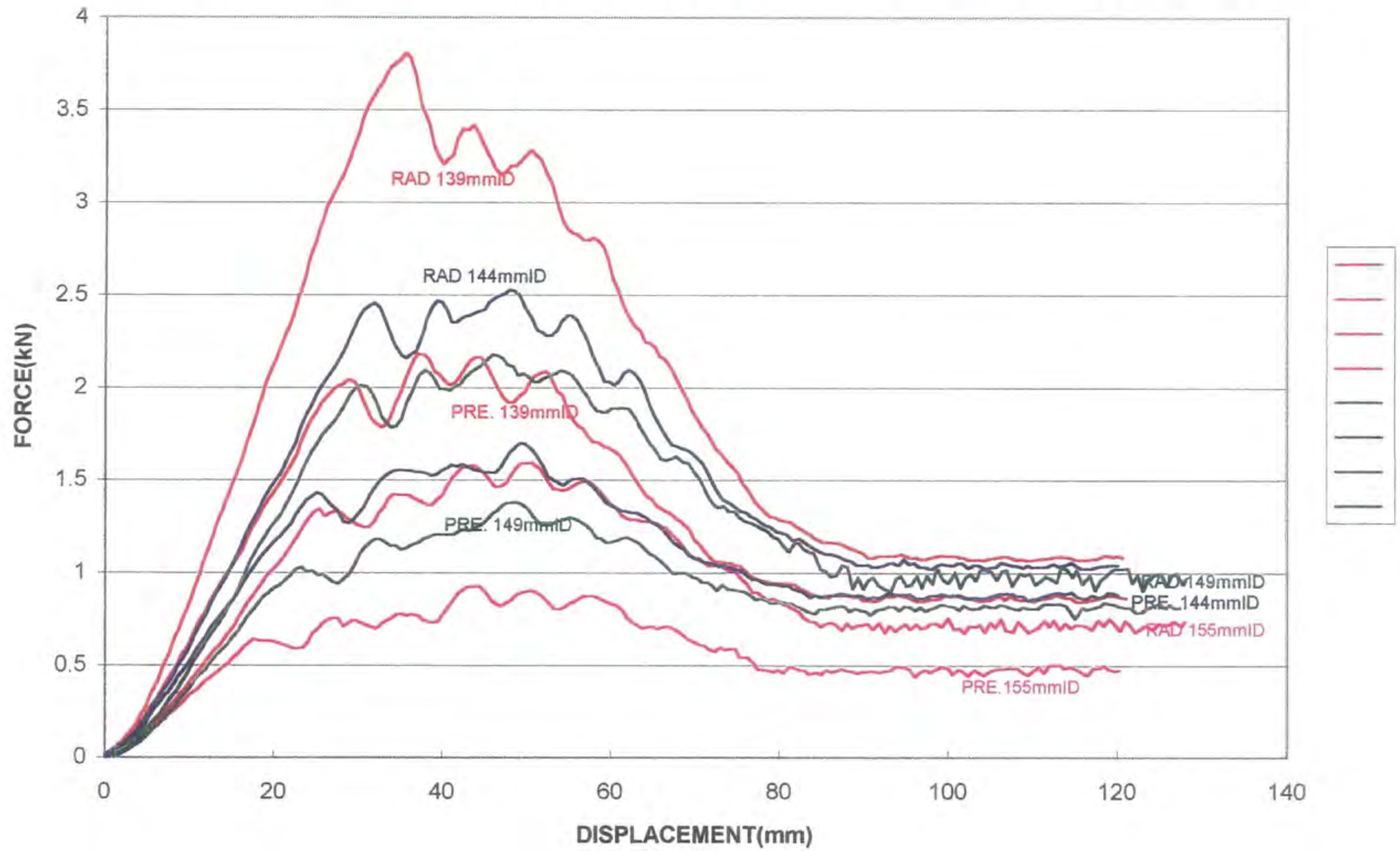


FIGURE 6.50
(FCV-LUB0.XLS)

Ra CONSTANT AT 38.84 MICRONS

A GRAPH SHOWING THE DIFFERENT FORCES REQUIRED TO LOAD & PUSH FORWARD A RADIAL BRISTLED BRUSH (NEW0A) COMPARED TO THE FORCE REQUIRED TO LOAD & PUSH FORWARD A PRE-SWEPT BRUSH (NEW15X)(FUEL OIL LUBRICATION PRESENT)



"RAD 139mmID"= RADIAL BRUSH INSERTED IN A 139mm ID PIPE
"PRE 139mmID" = PRE-SWEPT BRUSH INSERTED IN A 139mm ID PIPE

FIGURE 6.51

A GRAPH SHOWING THE AFFECT LUBRICATION HAS ON REVERSE FORCES,139mm ID PIPE, Ra=38.84, BRUSH PRE-SWEPT BRISTLES - 15 DEGREES (OLD15A)

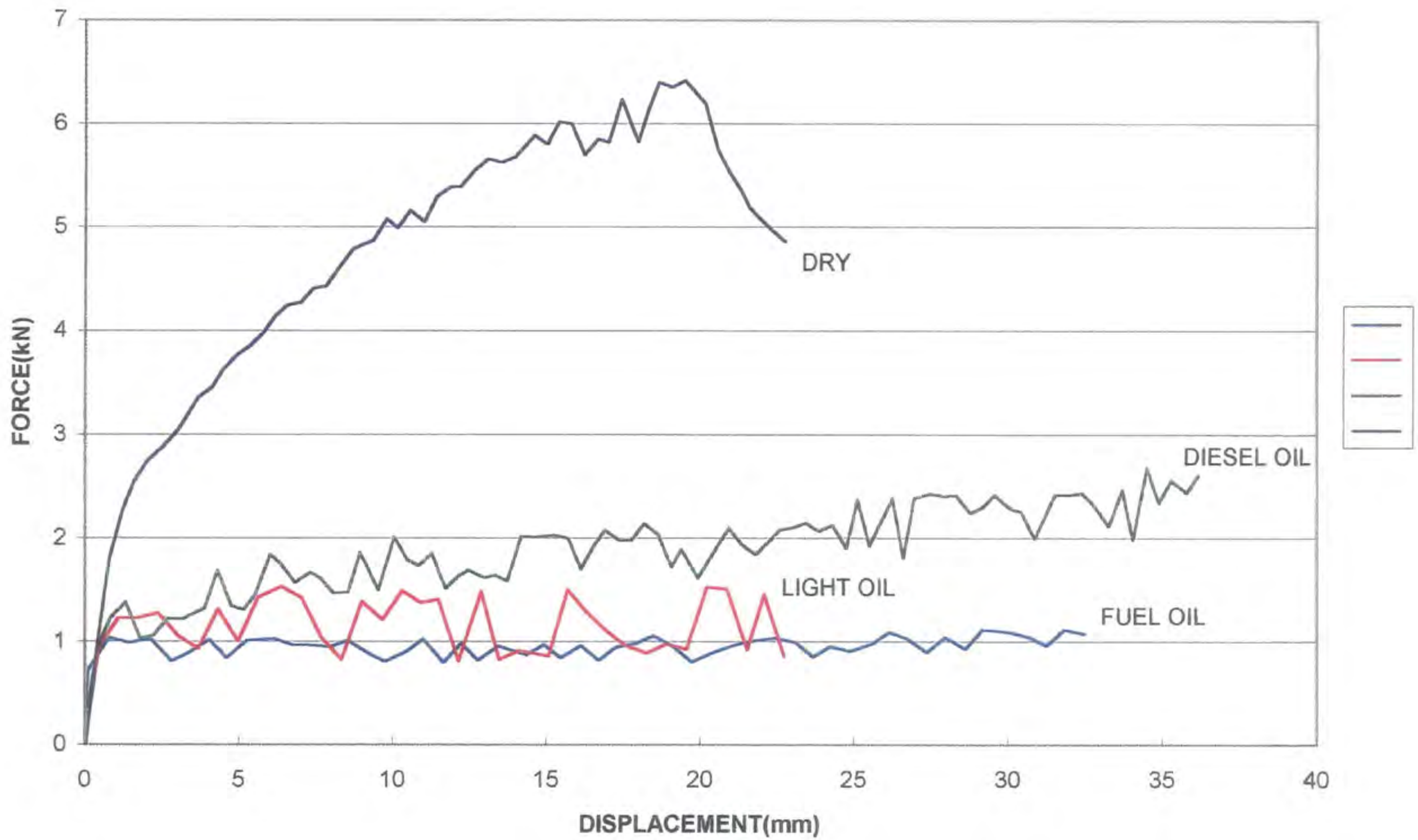
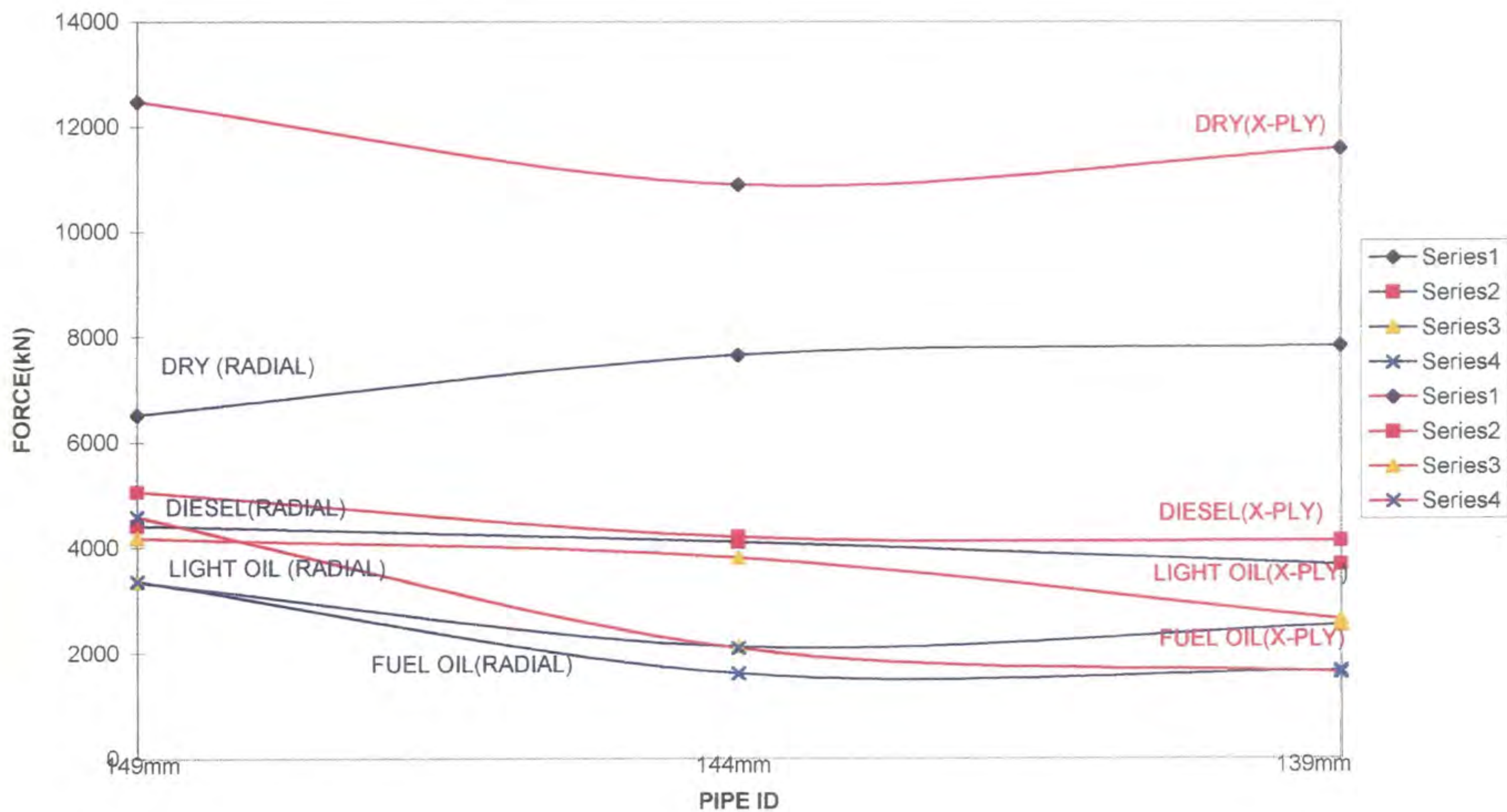


FIGURE 6.52
(LUB-REV.XLS)

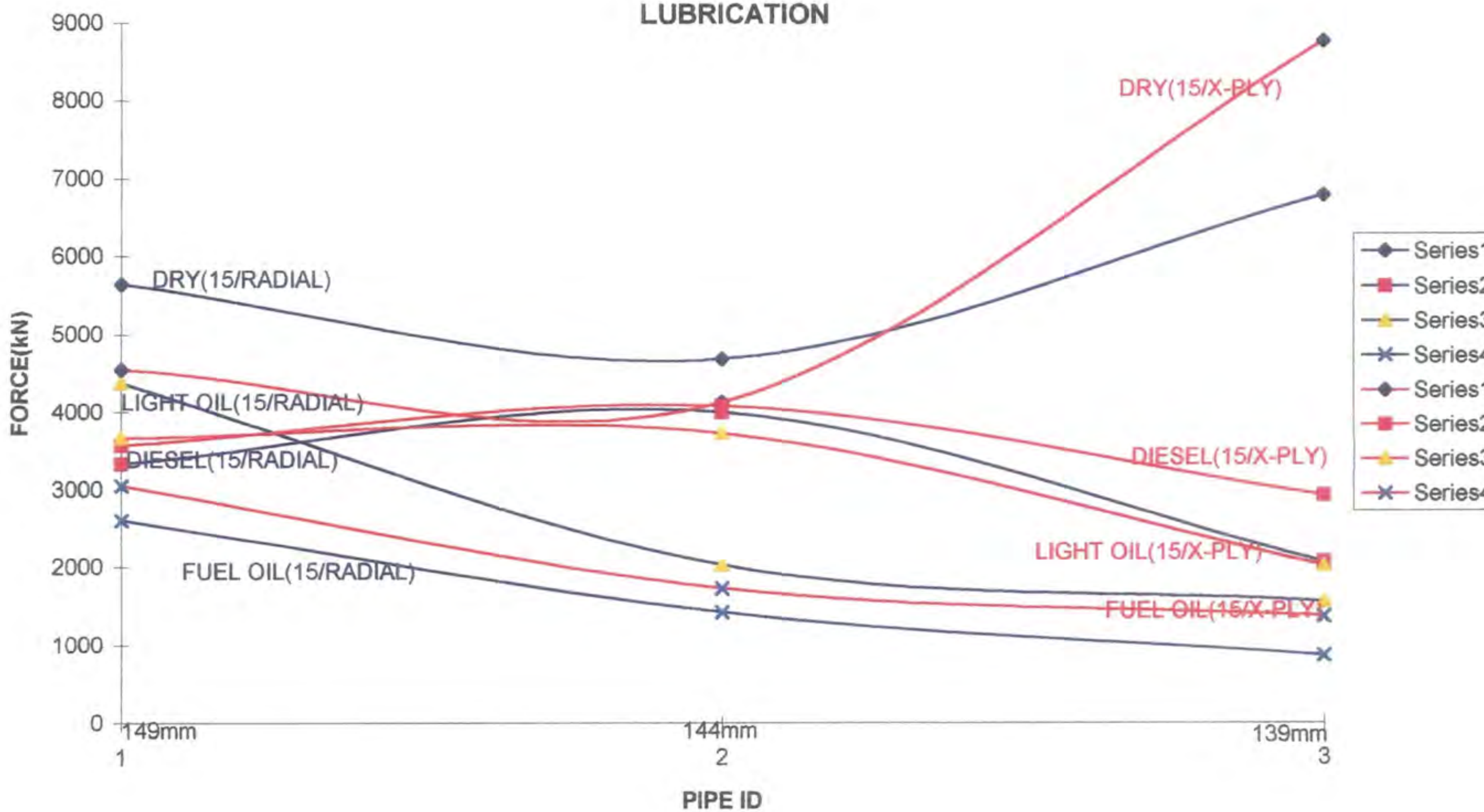
A GRAPH SHOWING THE DIFFERENCE BETWEEN A RADIAL BRUSH AND A RADIAL X-PLY BRUSH WHEN SUBJECTED TO REVERSE FORCES AND LUBRICATION



RED TRACES=X-PLY
BLUE TRACES=RADIAL

FIGURE 6.53

A GRAPH SHOWING THE DIFFERENCE BETWEEN A 15 DEGREE PRE-SWEPT BRUSH AND A 15 DEGREE PRE-SWEPT X-PLY BRUSH WHEN SUBJECTED TO REVERSE FORCES AND LUBRICATION



RED TRACES=PRE-SWEEP X-PLY
BLUE TRACES=PRE-SWEEP ONLY

FIGURE 6.54

AN ADDITIONAL GRAPH DETAILING THE BENEFIT OF RADIAL X-PLY BRUSHES OVER RADIAL NON X-PLY BRUSHES (DRY CONDITIONS)

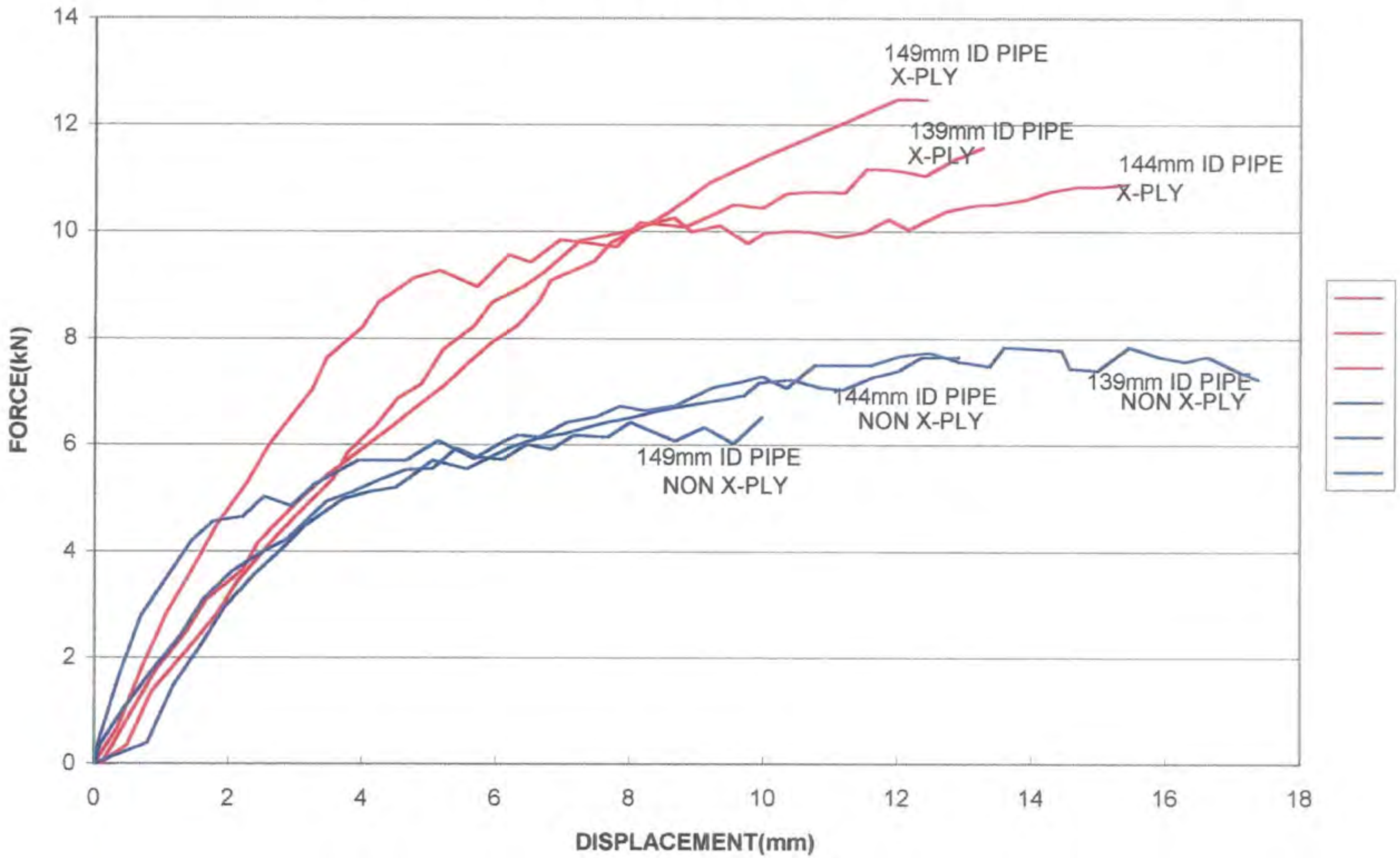


FIGURE 6.55
(X-PLYREV.XLS)

Ra CONSTANT THROUGHOUT AT 38.84

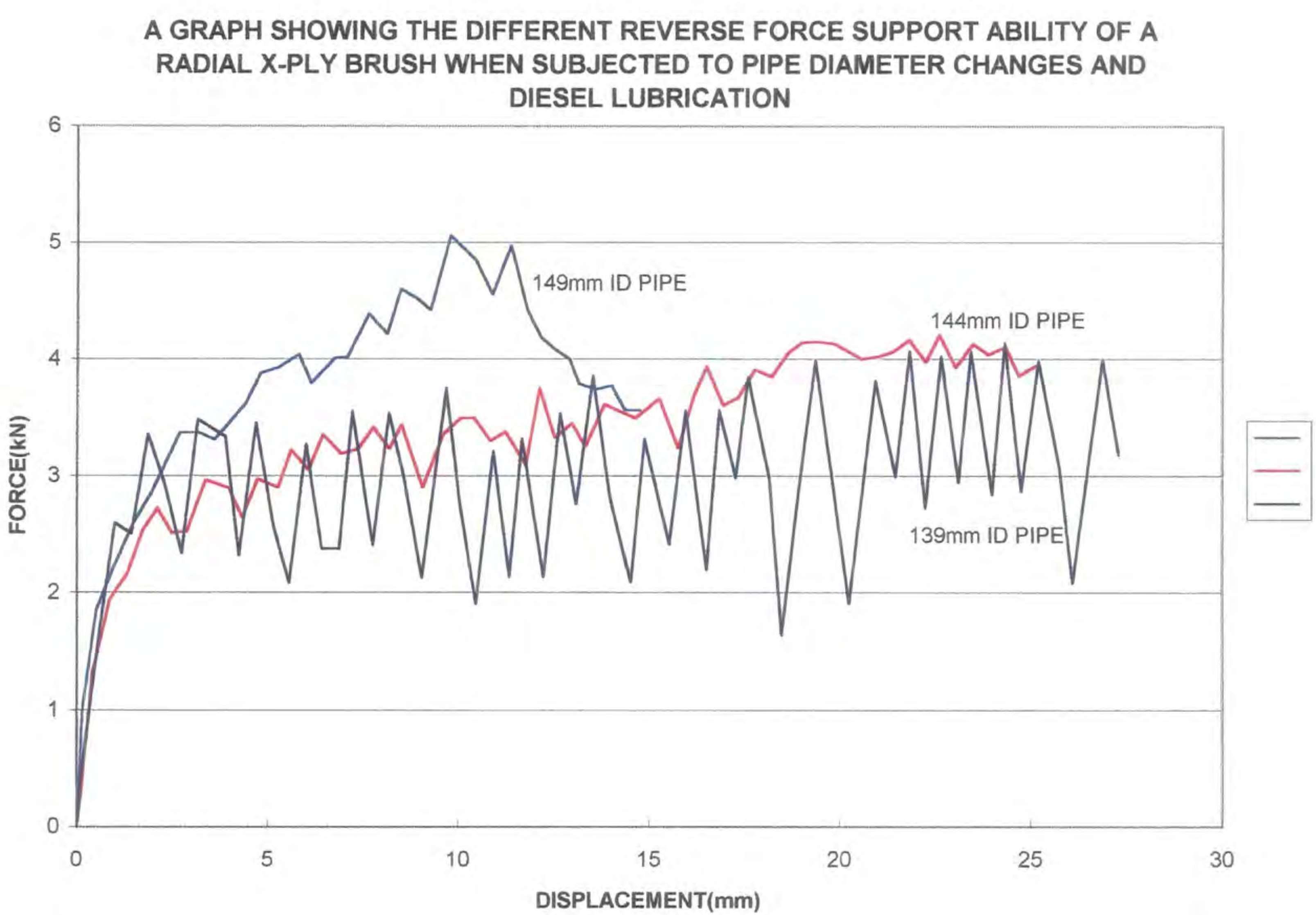


FIGURE 6.56
(DIESEL.XLS)

Ra=38.84 MICRONS

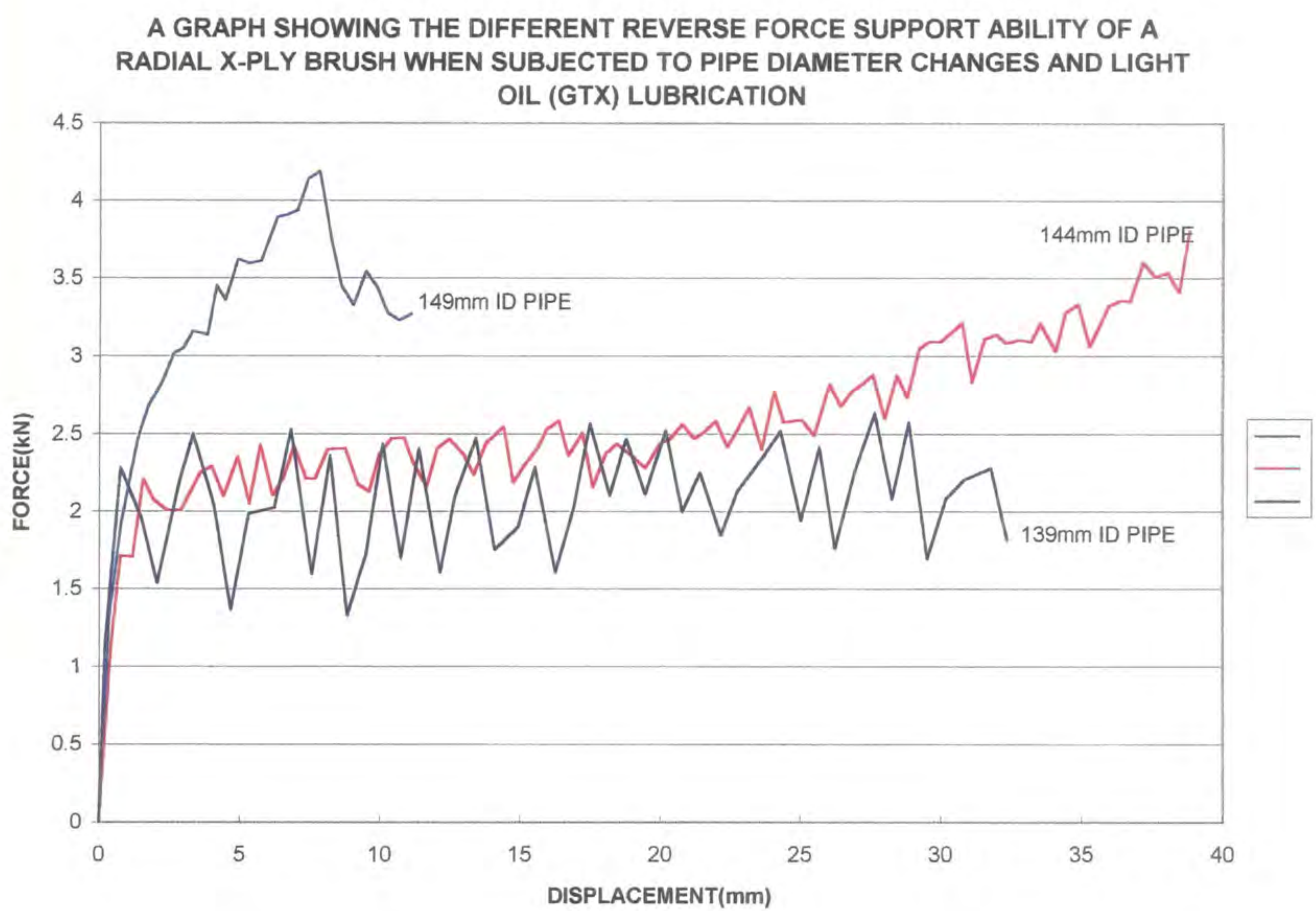


FIGURE 6.57
(GTX.XLS)

Ra=38.84 MICRONS

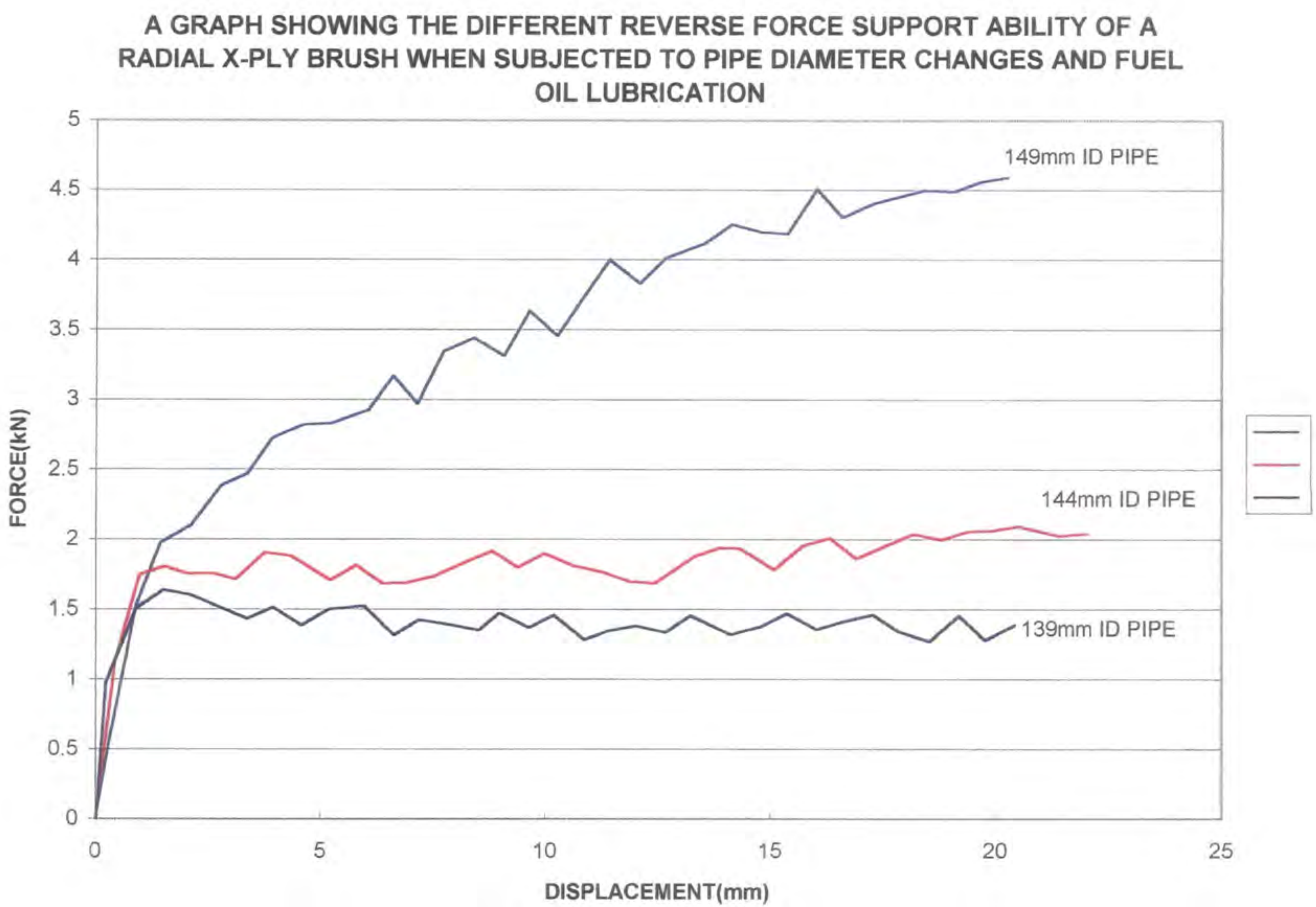


FIGURE 6.58
(FUEL-OIL.XLS)

Ra=38.84 MICRONS

A GRAPH SHOWING THE AFFECT THE Ra VALUE HAS ON THE FORWARD FORCE
(RADIAL X-PLY BRUSH, DRY CONDITIONS)

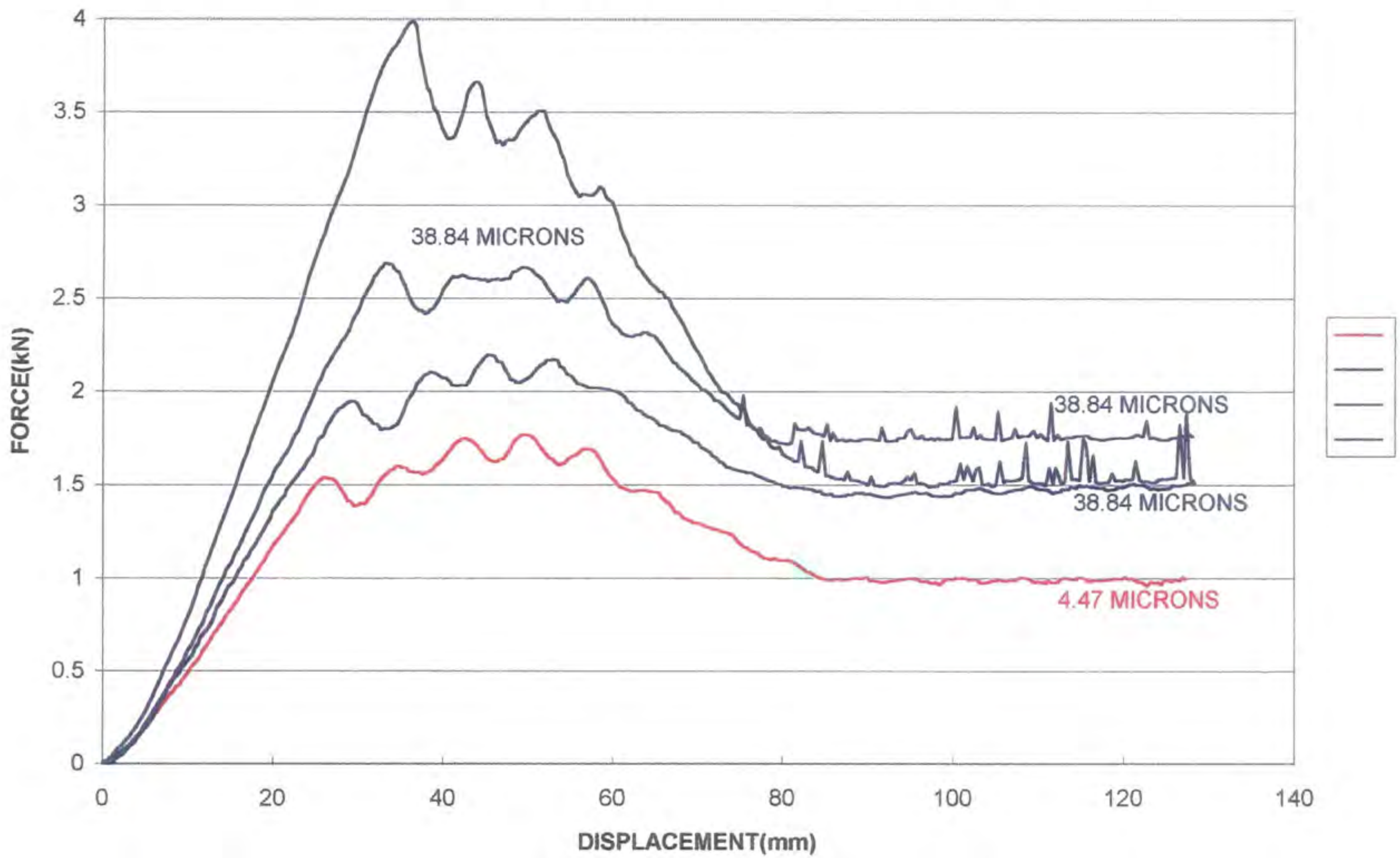


FIGURE 6.59
(RA-EFECT.XLS)

A GRAPH SHOWING THE AFFECT THE Ra VALUE HAS ON THE REVERSE FORCE SUPPORT ABILITY OF A RADIAL X-PLY BRUSH (NEW0X) UNDER DRY CONDITIONS

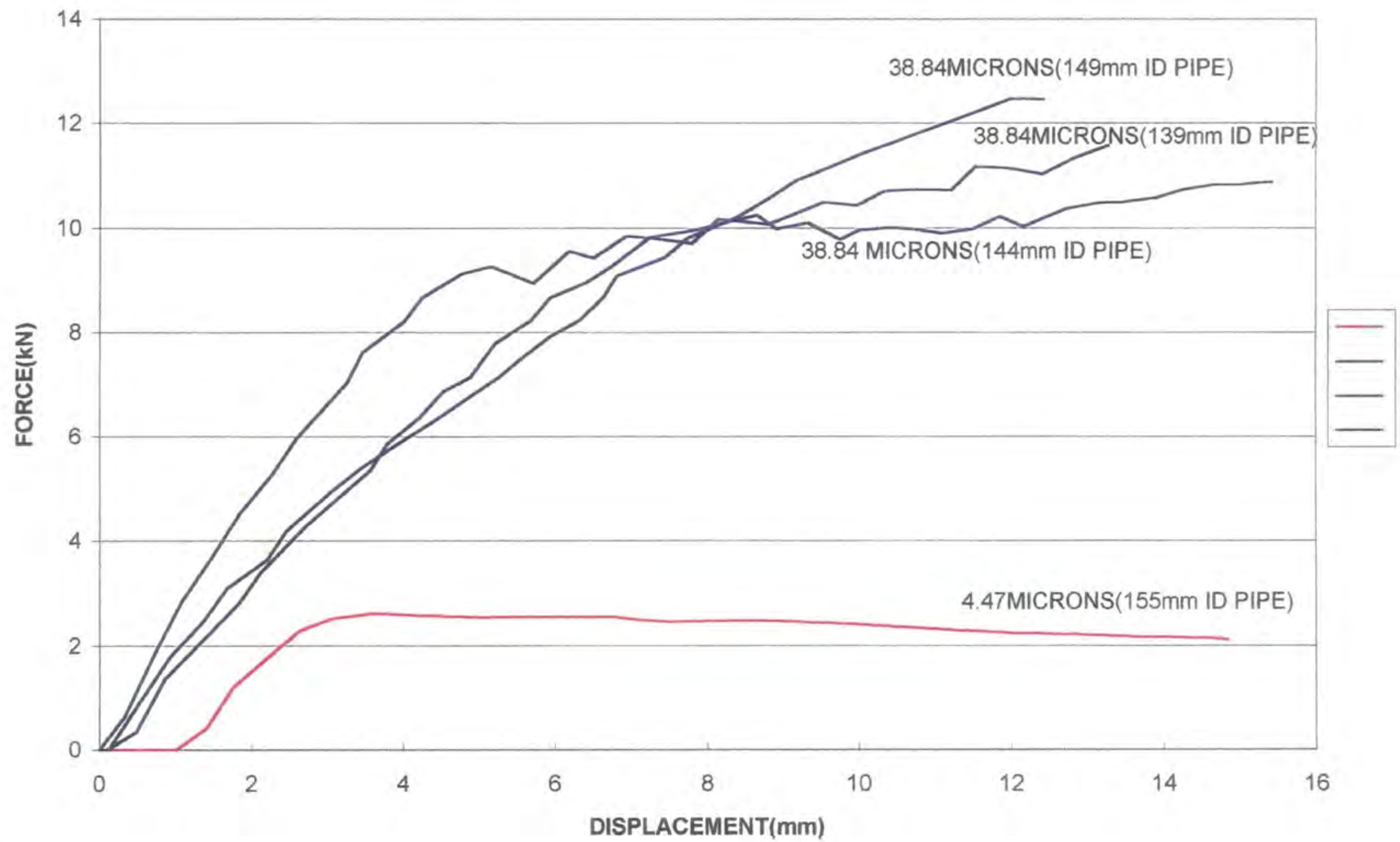


FIGURE 6-60
(RA-REV.XLS)

A GRAPH SHOWING THE FRICTION COEFFICIENT PLOTTED AGAINST THE SURFACE ROUGHNESS/PIPE ID

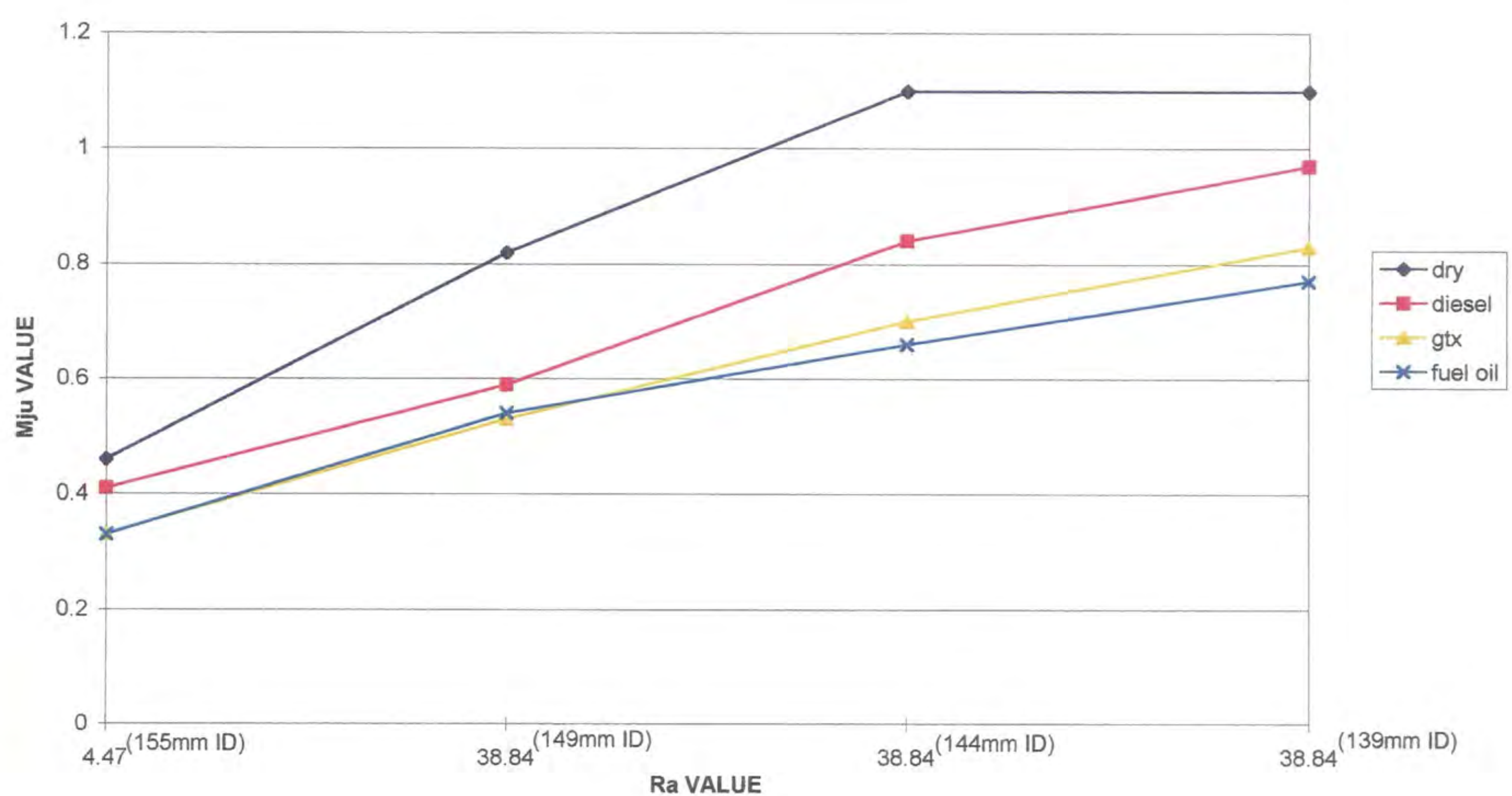


FIGURE 6.61

CHAPTER 7

CONCLUSION

The conclusions are discussed in three sections. Section one considers the loading force, the forward force and the reversal force of a brush in a pipe from an experimental perspective. Section two considers the models developed for analysis of the bristles and their correspondence to the experimental observations. Section three considers the overall design of this type of machine.

7.1 TRACTION FORCES

1. It was found that as both the polymeric and steel bristled brushes were loaded into a pipe the force rises as each circumfrential row of bristles entered into the pipe. A "balance" point was reached and the force slowly began to decline.
2. It was found that the force required to load a steel bristled brush unit increased as the pipe ID decreased. If lubricant was present there was a slight decrease in the force required to load the brushes.
3. It was found that if lubricant was present in a pipe, then, the force required to push a brush forward through the pipe decreased, this was true for both polymeric and steel bristled brushes.

4. It was found that there was a clear difference between the reverse force a polymeric bristled brush could sustain in dry conditions and the force it could sustain in lubricated conditions. Unfortunately, no discernible pattern could be determined from the relatively small amount of data obtained. Further research is required in this area.
5. It was further found that, under lubricated conditions, the polymeric bristled brushes yielded a higher forward "slip" to rear "grip" ratio. This was partly due to the lower force required to push the polymeric bristled brush forward through the pipe when lubrication was present.
6. It was found that if the steel bristled brushes were subject to the reverse force experiments they were unable to maintain the same grip under lubricated conditions as under dry conditions, consequently, rearward slippage resulted.
7. It was found that if a cross-ply (X-ply), radial, steel bristled brush, was used in dry conditions then it was possible to significantly increase the force the brush was able to support when subject to reverse forces.

It is considered that if a X-ply brush could be maintained centrally within the pipe and prevented from rotating, then, significantly higher reverse forces may be obtained. That is, if the bristles were forced to buckle in plane.

7.2 THEORETICAL MODELS

Two models were developed to analyse the forces present during forward and reverse force application. The first model considered an individual bristle which acted as a simple uniform cantilever experiencing a concentrated end load, with flexural stiffness EI and length l. By calculating the deflection of the cantilever, it was possible to determine the force the bristle exerted against the pipe wall. The force required to push the bristle/brush forwards could then be determined by $F=\mu.N$. This "simple" model was then incorporated into a spreadsheet which was used regularly to help in the development of vehicles. The second model is able to predict the forward "slip" to reverse "grip" ratio for known or pre-determined friction values. This model involves drawing the forces to scale.

1. It was found that the results predicted by the "simple" cantilever model correlated well to the forward force experiments performed on the polymeric bristled brushes. The results obtained from the practical experiments were processed by the spreadsheet and average μ values of 0.2, 0.16, 0.16 and 0.14 were calculated for the forward force experiments on the dry, water, diesel and light oil lubrication conditions, respectively. These friction values are considered acceptable.
2. It has been found that the second model, which predicted the forward "slip" to rear "grip" ratios is useful in a predictive sense, though the theory contains assumptions which restrict its applicability to conditions prior to the onset of the bristle buckling.
3. Using the model, it was found that as the angle between the bristle and pipe wall approached the friction angle, then the forward "slip" to reverse "grip" ratio increased significantly.

4. It was found that if the bristle angle and friction angles were equal, the model showed a theoretical ratio of infinity. It was realised that this situation could not occur as the bristle and/or pipe wall could not sustain such thrust forces
5. It was found that a slender flexible bristle will collapse by buckling before reaching a thrust of infinity and before plastic failure occurs. A short stubby bristle may collapse by plastic yielding or the wall of the pipe may fail to support such thrusts. Most bristles can be considered as "slender", that is, the magnitude of the thrust along the bristle is limited to the buckling load.
6. It was found that if a bristle fails by buckling and the bristle angle is near or beyond the friction angle, then, the axial force will be less than the limiting friction value, therefore, the bristle will grip without slip. If the bristle angle is greater than the friction angle, the axial force the bristle can sustain will decreases to zero, in this case bristle slippage would result.
7. It was found that the failure of a bristle by a buckling mechanism is an instability phenomenon, as a consequence, the failure of a brush when subjected to reverse forces is a rapid and unstable process. Further, the point at which failure occurs is very difficult to predict.

7.3 VEHICLE DESIGN CONSIDERATIONS

1. It was found that "simple" universal joints or couplings caused the vehicle to jack-knife on the straight sections of pipe.
2. It was found that a joint that remained stiff or "locked" in the straight and was able to deflect during bend negotiation was beneficial in reducing jack-knifing.

3. The first "successful" joint used two parallel aluminium plates, held apart, but, interconnected by, three anti-vibration rubber bobbins located around a specific PCD. It was found that this type of joint, although acceptable, had a number of inherent weaknesses. The main problem was the fact that when the vehicle was travelling in a straight pipe section, the joint would still induce a degree of vehicle or inter-body jack-knifing.
4. It was found that by incorporating three adjustable steel pins in the above joint it was possible to create a joint which was more stable in the straight sections but was still able to deflect to provide bend negotiation. The pins were screwed into one of the plates and their tips, which incorporated radii were allowed to rotate in radii machined into the opposing plate. Thus, in the straight, the forward force would be transmitted through the steel pins. However, as the vehicle negotiated a bend at least one of the pins "breaks-away" and the joint deflects, this leaves one or two pins to act as a pivot point. This has been an extremely successful joint and is currently used on the sewer vehicles.
5. If high payloads are to be carried it was found that a stronger joint, both in compression and tension was required. This was accomplished through the use of rose bearings, this joint still incorporated the rubber bobbins which helped to stiffen the joint.
6. It was found that by incorporating suspension wheels into the vehicle jack-knifing could be avoided.
7. It was found that if wheels could not be used due to obstacles inside the pipe then short, stubby bristles could be used as "suspension bristles".

8. It was concluded that the size (length and diameter), thrust and pull of a fluid cylinder was dependent upon the ID of the pipe in which it was to operate, the tightness of the bends it had to negotiate and the magnitude of the payload the vehicle had to transport. A tight bend would necessitate a short stroke cylinder and a small pipe ID would limit the diameter of the cylinder piston, thus, the thrust and pull the cylinder could generate would be lower. If a heavy payload is to be moved or densely populated brushes are used a cylinder is required that is able to generate high thrust and pull forces.
9. It was found that the aspect ratio of a brush, that is, its length to its width, makes a considerable difference to the stability of a vehicle when inside a pipe, if the brush is too short with respect to its diameter it will simply "cock" over in the pipe, that is, the vehicle will jack-knife.
10. It was found that the stiffness of the bristles is important, this is critical to both suspension and traction. If the bristles are insufficiently stiff, the vehicle, when inserted into the pipe will simply "nose-dive".
11. It was found that if a vehicle is required to operate at high speed then a two brush/one cylinder configuration is well suited. If high traction is required a three brush/two cylinder machine is better suited.

It will now be clear to the reader that this study is in no way complete in terms of a full understanding of the traction mechanism. However, a useful first analysis has been put forward which is able to establish some of the fundamental traction forces which occur whilst a bristled vehicle negotiates varying pipeline conditions. Specifically, the force required for a given brush to move forward through a pipe and the "grip" force the same brush is able to exert when in support mode. To conclude, specific vehicles can now be designed for specific pipeline conditions with a degree of confidence that they will perform satisfactorily.

7.4 FURTHER WORK

There are two main areas which are deemed to be in need of further work. The first concerns the forces generated by the bristles at the bristle tip/pipe wall interface and the reverse force a brush can support prior to failure. The second is the further development of the vehicle joints.

The structural properties of bristles during the gripping mode are not yet understood. The complex "buckling" mechanism and the flip-through process requires further study. It would be interesting to determine whether a brush located centrally within a pipe and prevented from rotating was able to generate the very high forward "slip" to rear "grip" ratios which were predicted by the theoretical model.

Lubrication clearly influences brush performance and further work could be undertaken into the interactions between the bristle tips and pipe wall. For example, does plastic yielding occur at the bristle tips, further, does micro fusion/welding occur between the bristle tip and pipe wall at the time the two are in contact.

The experimental work supported by the theoretical models indicate that the traction mechanism has some unstable characteristics. In the experimental machines instability has been overcome by the use of suspension wheels or bristles. Another mechanism for controlling machine stability could be by the careful design of joints. Although the vehicle joints work well in the experimental machines they are still crude. Further work could be undertaken into making these joints "intelligent". That is, the joint would remain locked when in a straight section of pipe and would "relax" or even pre-angle itself to accommodate an approaching bend. Perhaps further work could be undertaken into the use of rheological fluids to control and or manipulate the joints.

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APPENDIX A

A PHOTOGRAPHIC "HISTORY" OF THE VEHICLE'S DEVELOPMENT

FIGURE A20. A photograph showing the simple "lavatory brush" vehicle.

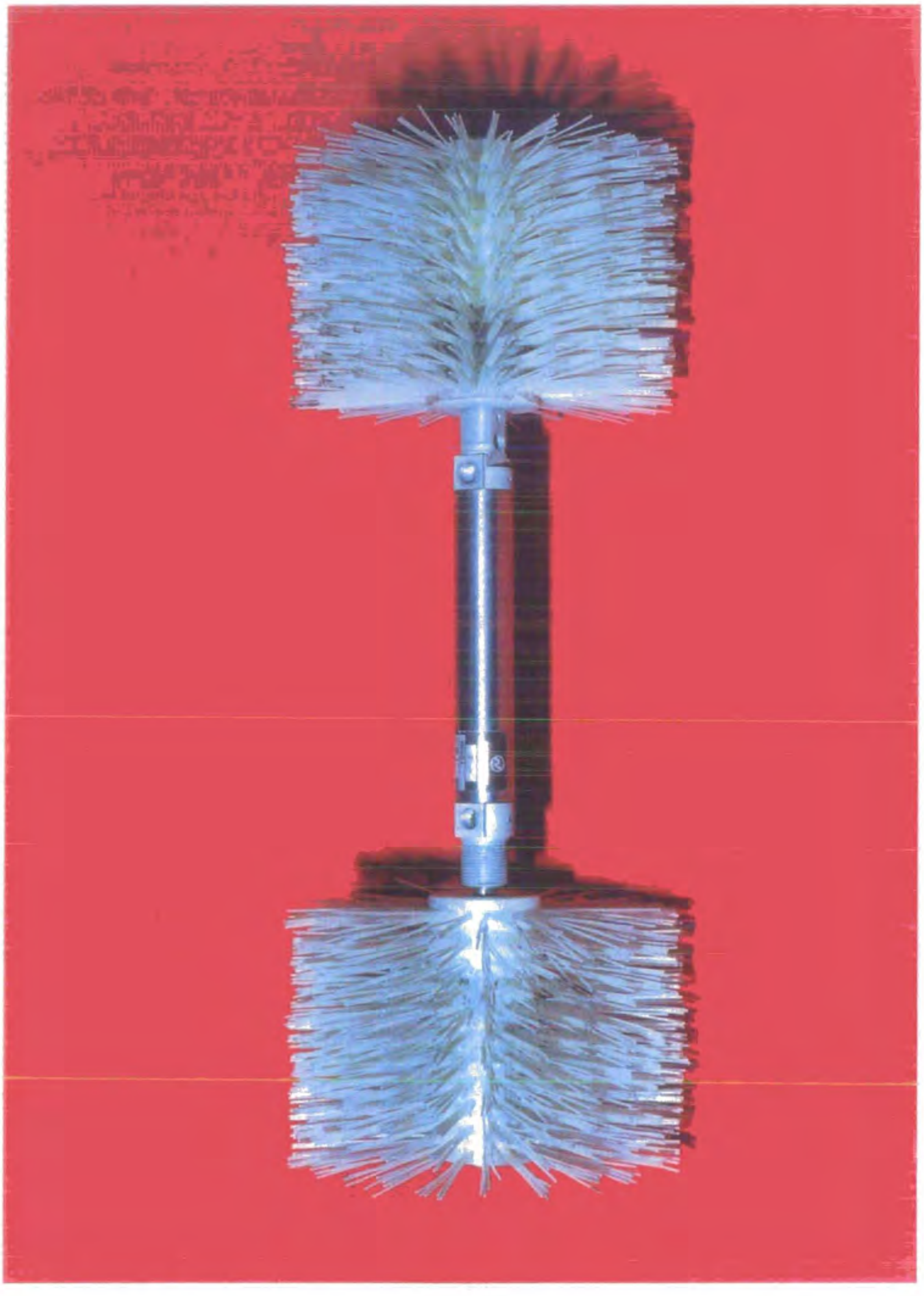

FIGURE A21. A photograph showing a simple vehicle which incorporates "designed" brushes.

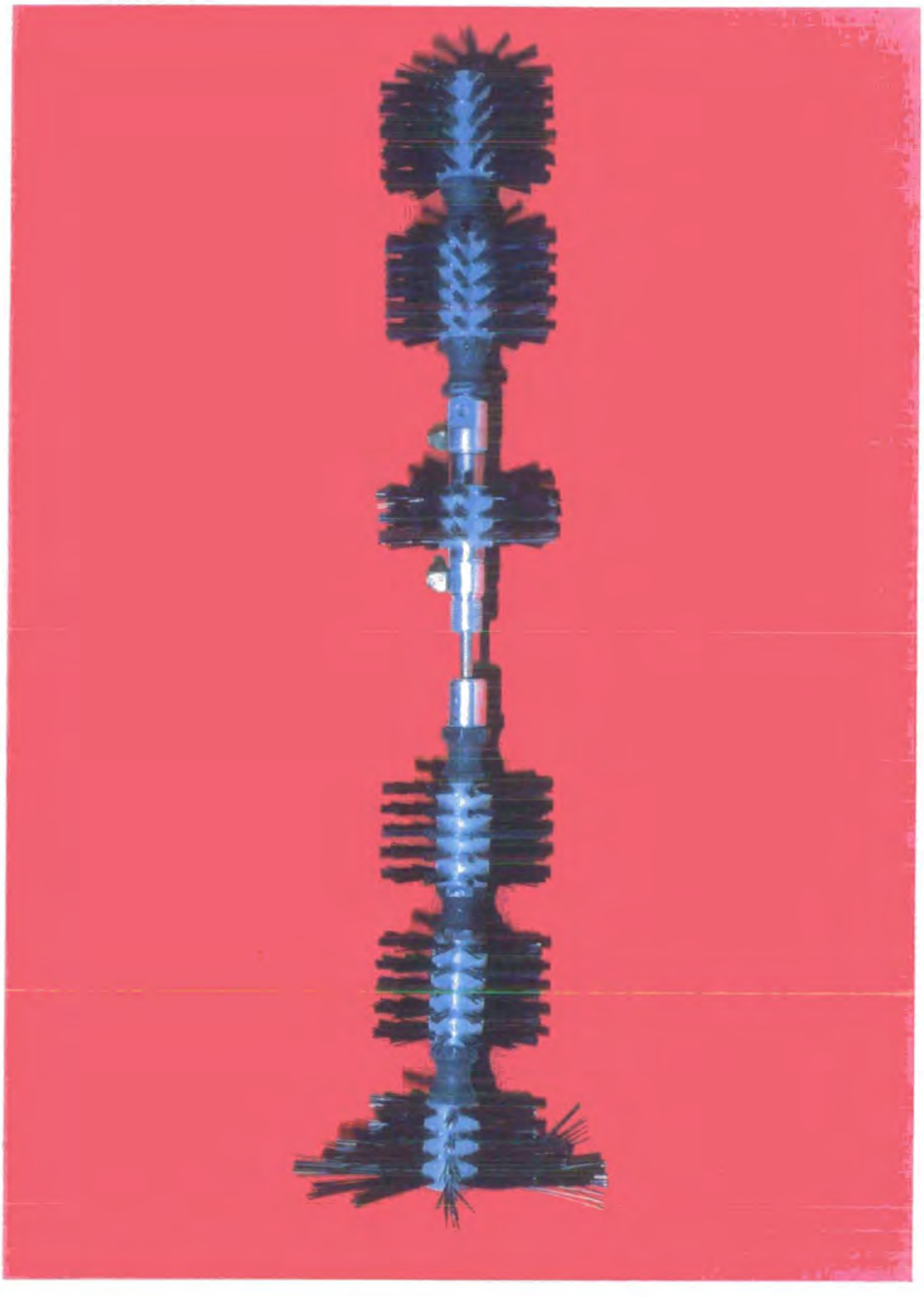

FIGURE A22. A photograph showing the first multi-jointed flexible vehicle.

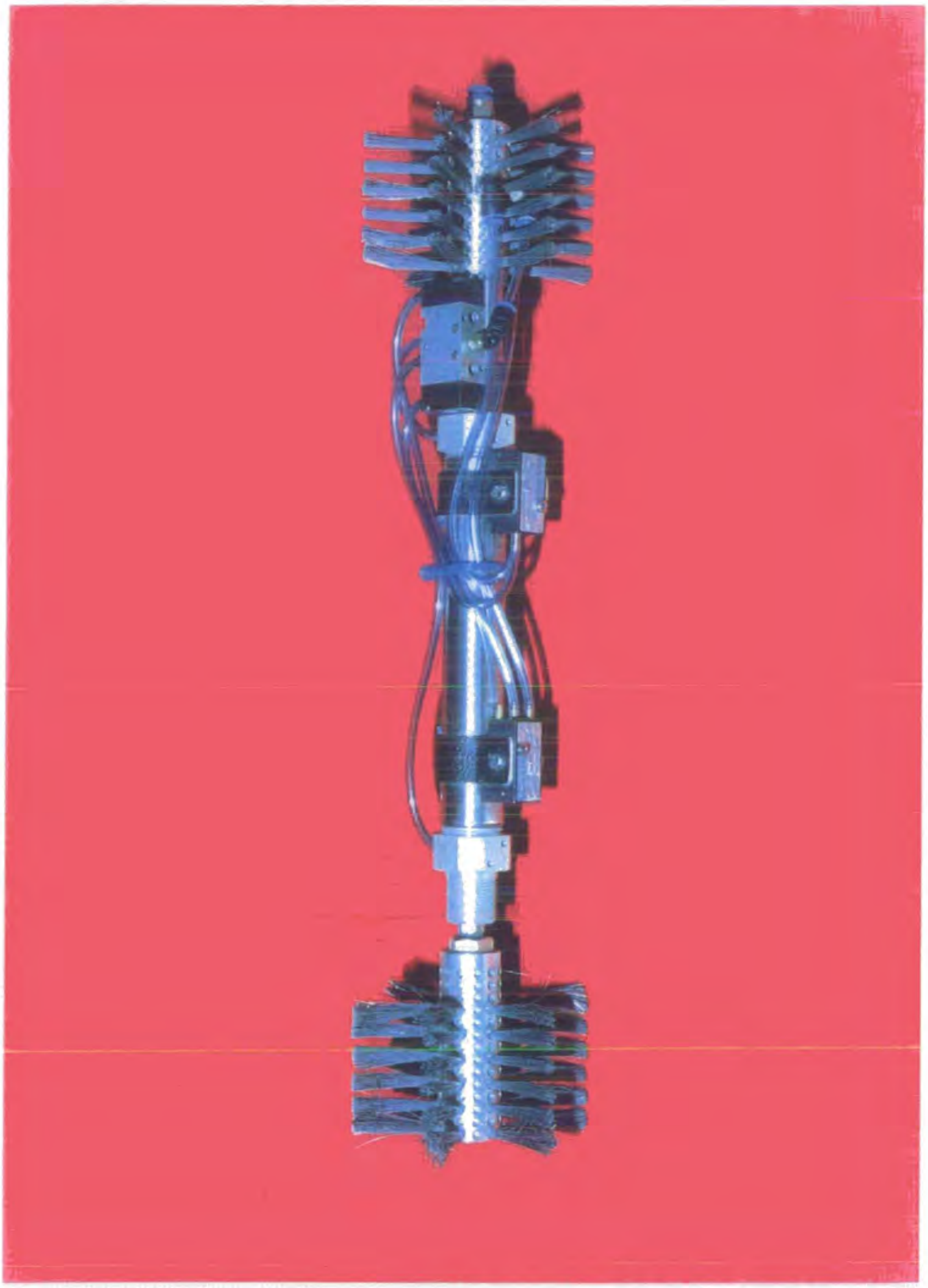

FIGURE A23. A photograph showing the "sprinter vehicle". This vehicle utilises pneumatics instead of electronics to switch the reciprocating cylinder.

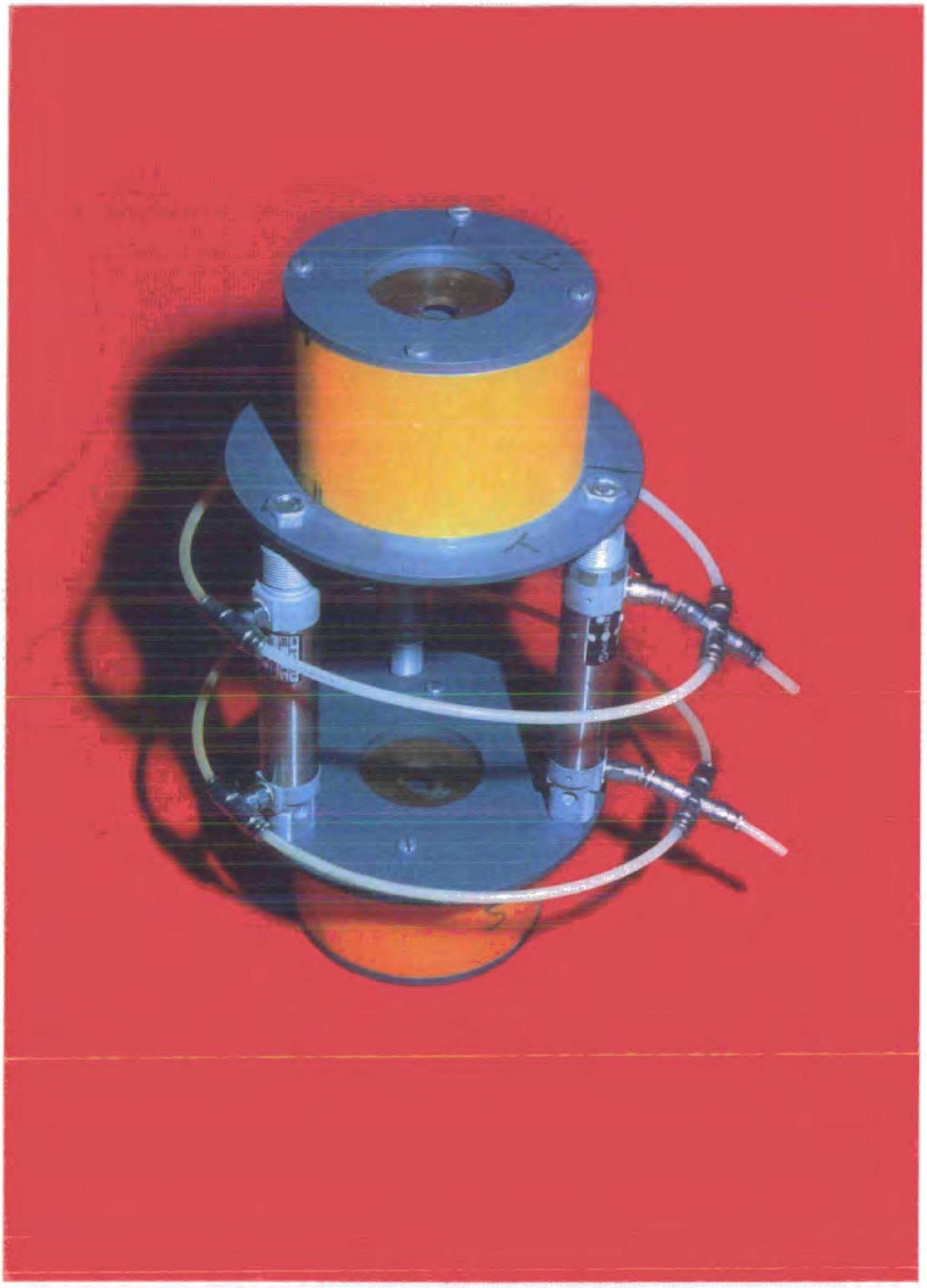

FIGURE A24. A photograph showing the external climbing vehicle. This vehicle is designed to climb up the external surface of tubes, pipes, cables or rods.

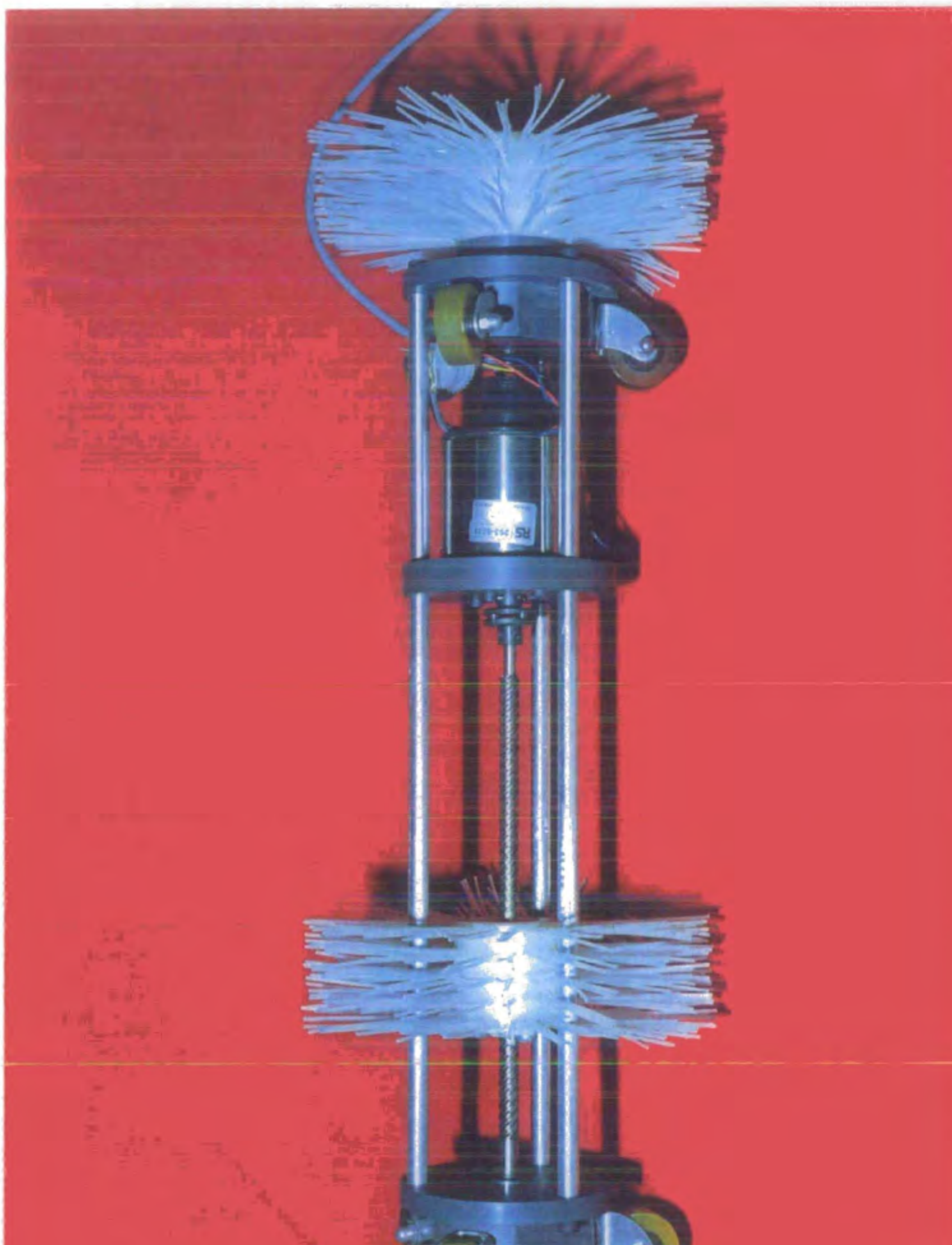

FIGURE A25. A photograph showing a totally electric vehicle.

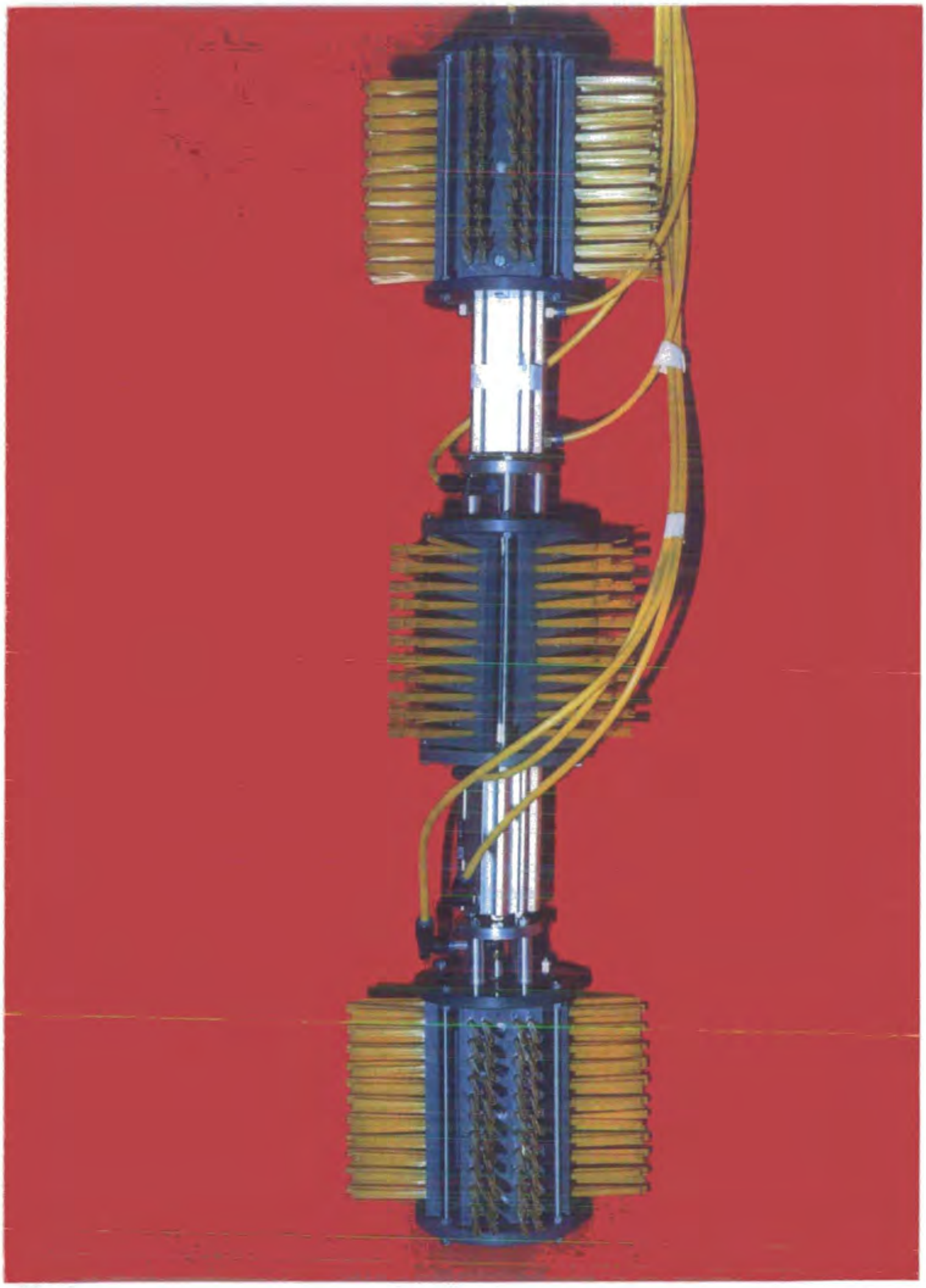

FIGURE A26. A photograph showing an experimental vehicle which was built to prove the principle of bi-directionality.

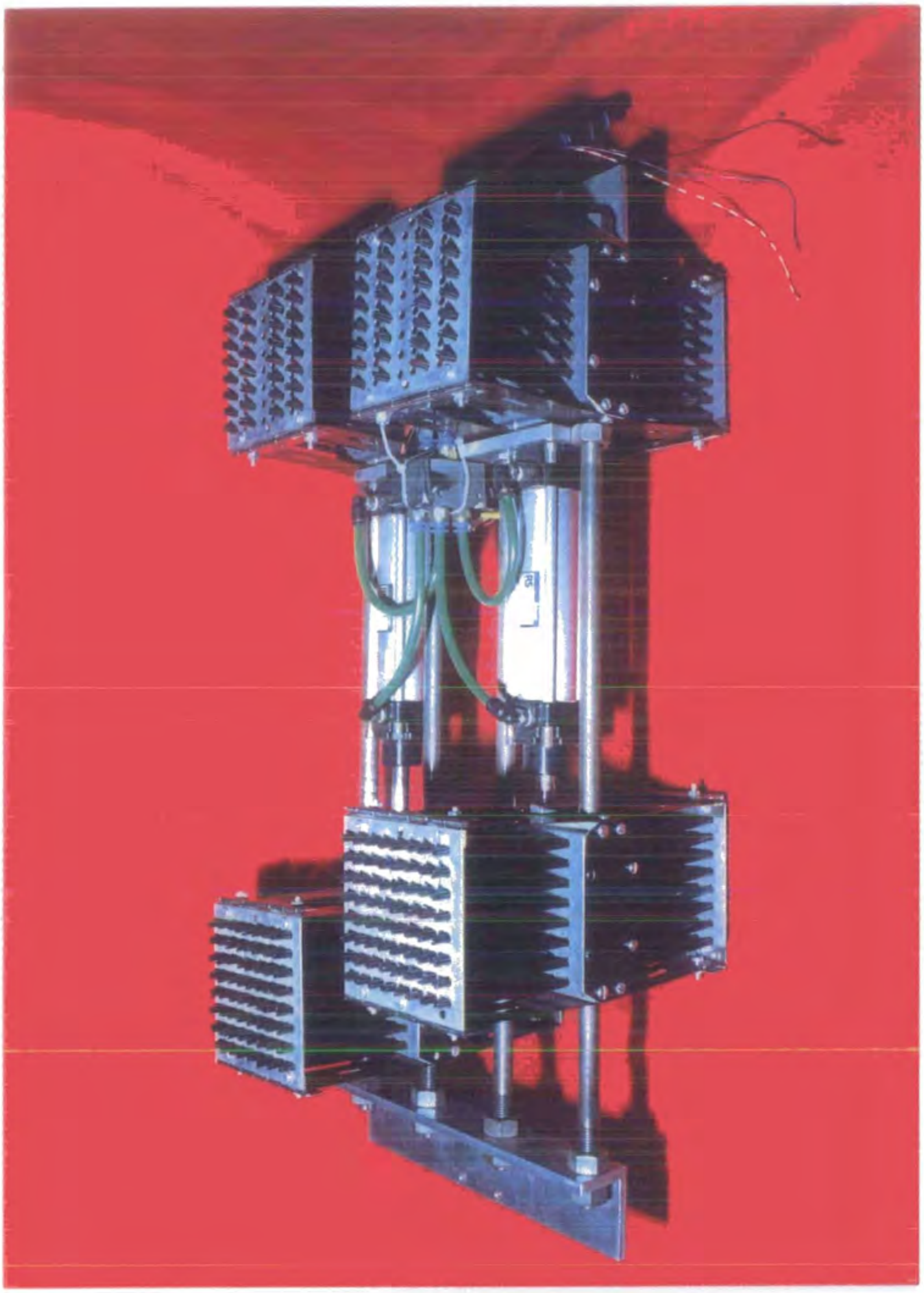

FIGURE A27. A photograph showing the parallel plane vehicle. This is an experimental vehicle designed to negotiate between the twin skins of a ship.

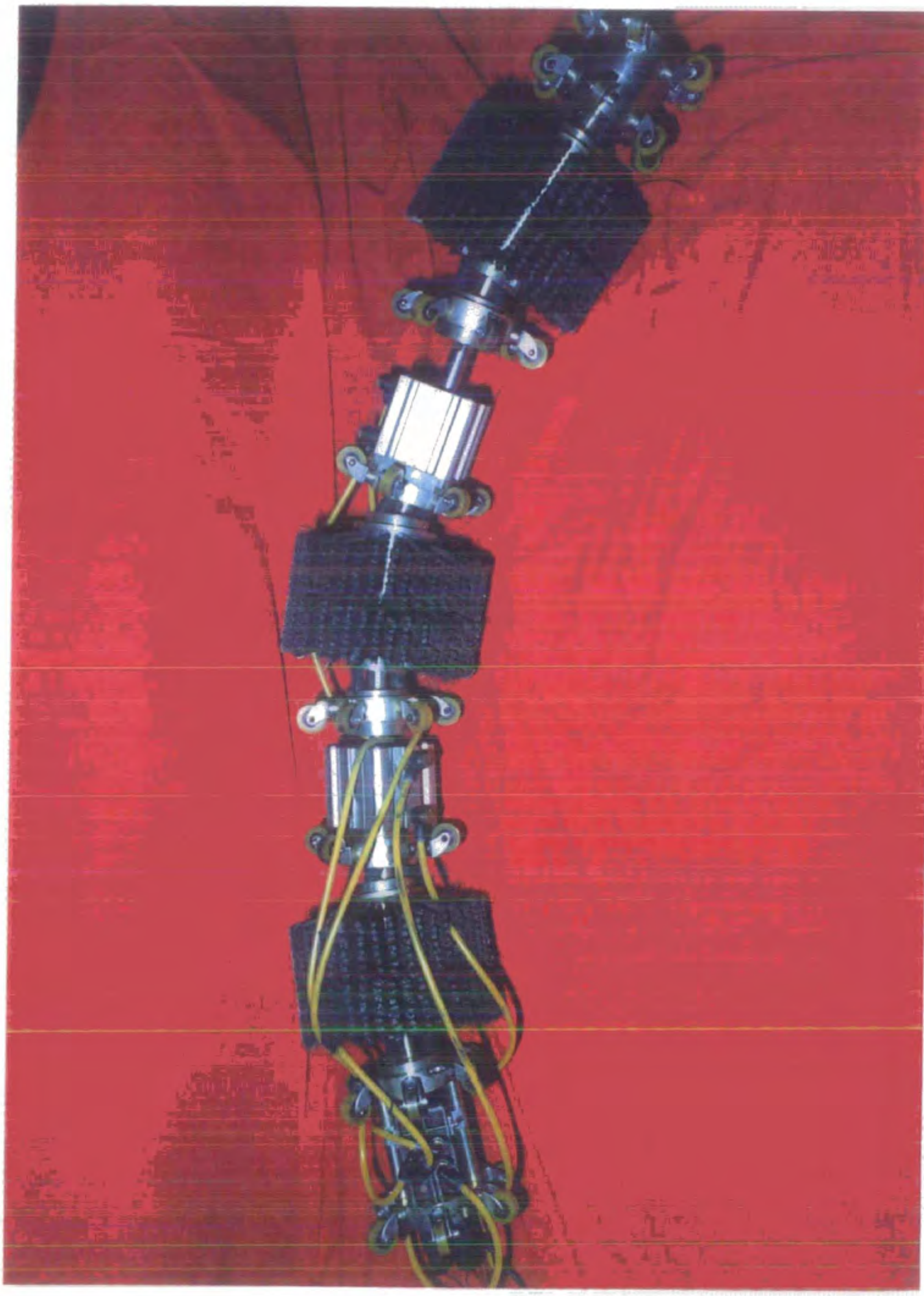

FIGURE A28. A photograph showing the 8" diameter, high capacity PII vehicle.

This vehicle has a tow capacity in excess of 1 Tonne.

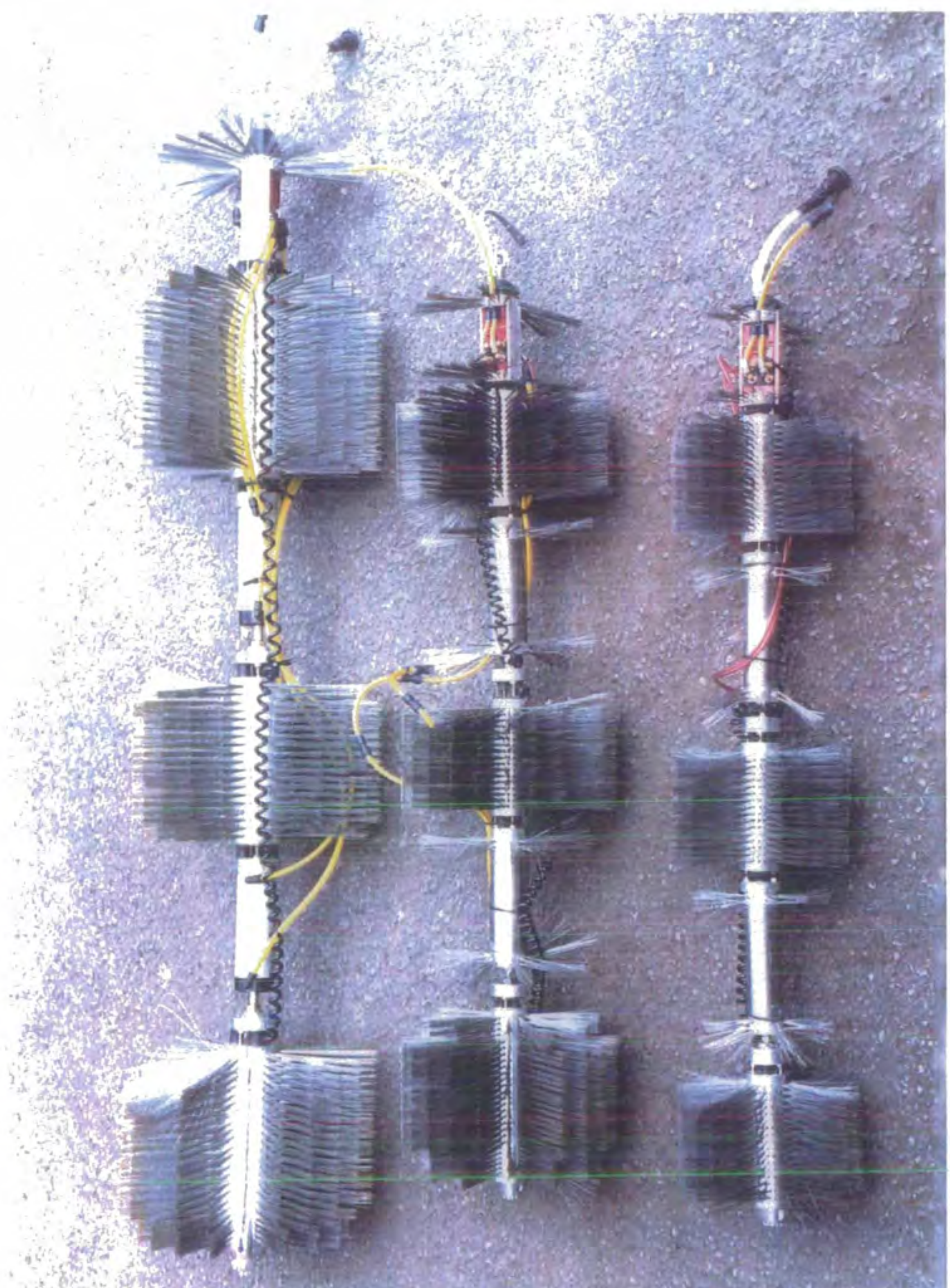

FIGURE A29. A photograph showing the "family" of sewer inspection vehicles. The 225mm diameter vehicle is in the foreground, the 300mm diameter vehicle is in the centre and the 375mm diameter vehicle is at the rear.

FIGURE A30. A photograph showing a vehicle that is designed to harness its energy source from fluid flowing through it. It is then able to crawl against the fluid flow.

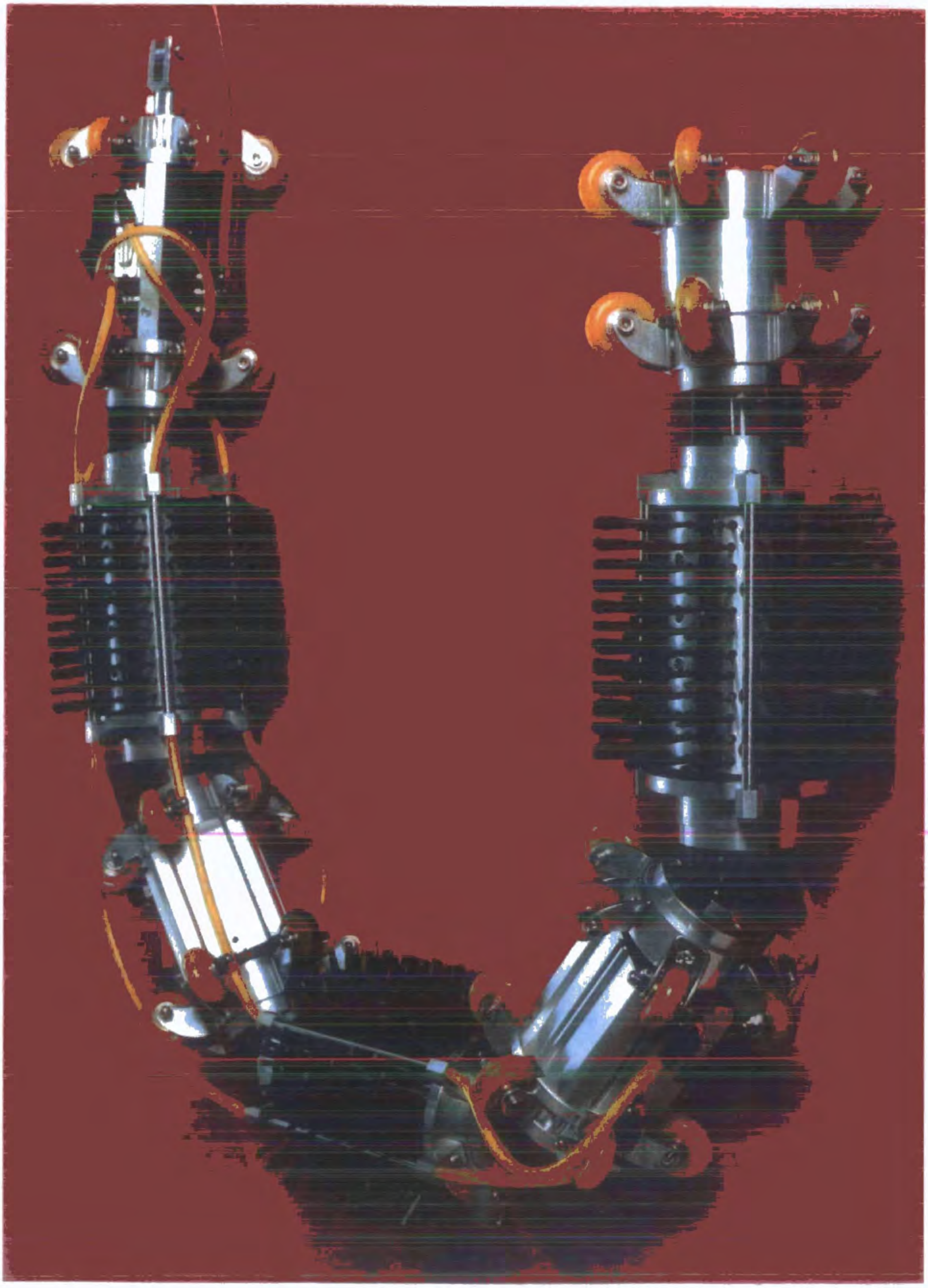

FIGURE A31 A photograph showing the 6" diameter bi-directional vehicle.

FIGURE A31. A photograph showing an expanding brush core.

APPENDIX B

THE UNIVERSITY OF DURHAM PATENT

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Surface traversing vehicle

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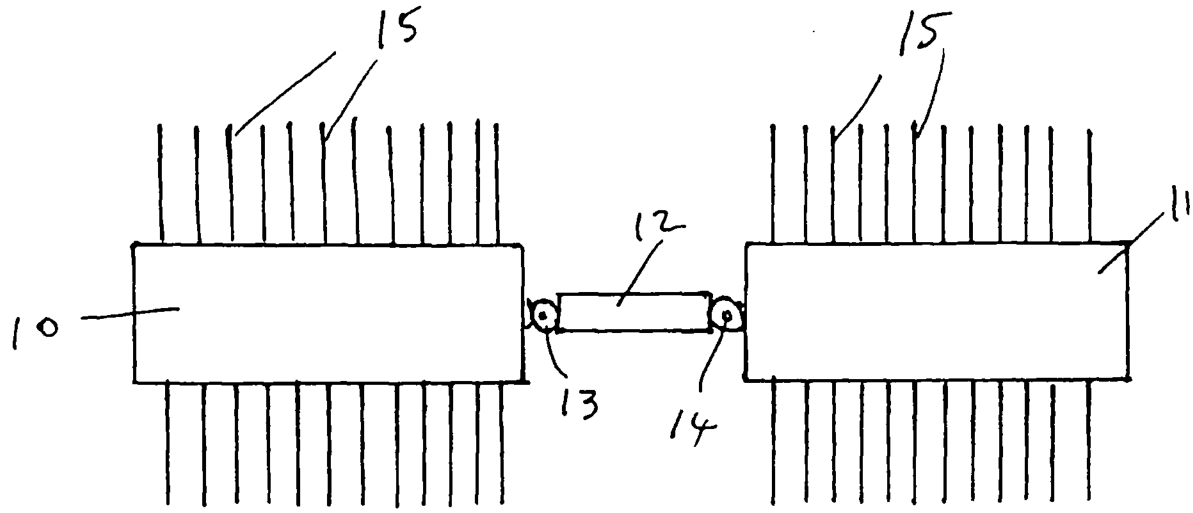

Fig. 1.

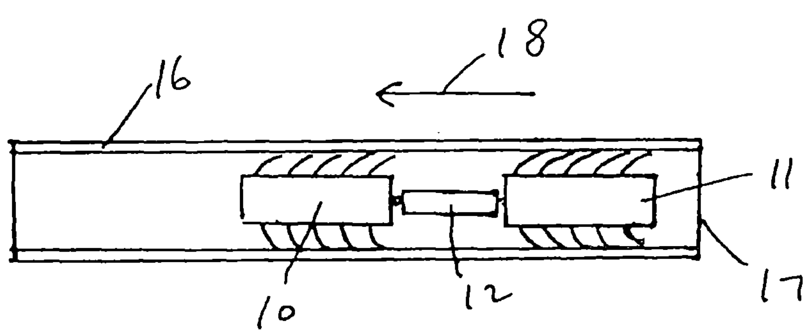

Fig. 2.

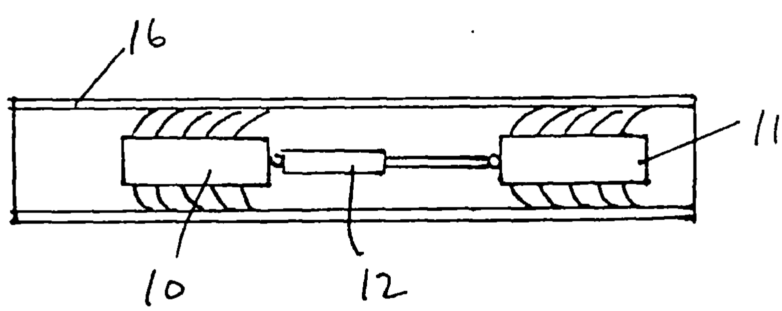

Fig. 3.

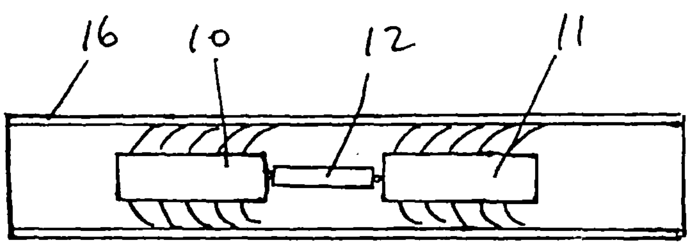

Fig. 4.

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Surface traversing vehicle

The invention is a vehicle for traversing a surface such as for carrying out an inspection, survey or maintenance operation upon that surface. For example, vehicles of this type may be used to traverse a flat surface, or a space between two such surfaces, or to traverse internally or externally the length of a generally tubular conduit such as a pipe, shaft, tunnel, drain, chimney or the like, in a horizontal, vertical or intermediate direction.

Vehicles and similar tools for use in carrying out operations such as inspection and maintenance of the interior of tubular conduits are known and used. Many such vehicles can be used only in conduits which are horizontal or do not depart greatly from the horizontal; many of these rely upon the conduit being more-or-less uniform in cross-section throughout its length.

If a vehicle is to traverse the length of a generally vertical or steeply-inclined surface, then the vehicle must be able to grip the face of the surface sufficiently well to support itself, and any equipment which it is required to carry or move, against the effects of gravity, friction and any other resistance. Thus vertically-traversing vehicles are more difficult to devise and such vehicles as are available for this purpose are usually suitable for use only in tubular conduits of uniform cross-section and having diameters of a specific value or lying within a narrowly-defined range.

Few, if any, prior vehicles are suitable for carrying out exploratory or other operations other than over the interior surfaces of such uniform tubular conduits. Such vehicles are not able satisfactorily to carry out such operations even in tubular conduits of varying cross-sectional size or shape, whether or not those conduits are generally horizontal. They are also usually unsuitable for use when the inner surface of a tubular conduit has irregularities such as hollows or localised projections.

Against this background, it is an object of the present invention to provide a surface traversing vehicle which is suitable for use upon a wide variety of smooth and/or irregular surfaces, including those of a wide range of generally tubular conduits of uniform or irregular cross-sectional and/or shape and also generally planar surfaces of uniform or irregular shape, any of which surfaces may be horizontal, vertical or intermediate these directions.

The surface traversing vehicle according to the present invention comprises at least two bodies interconnected by means to move adjacent said bodies towards and away from each other, each said body being supported upon a multiplicity of resilient bristles extending from it. It has unexpectedly been found that alternate moving of the bodies towards and away from each other causes the vehicle to move in successive steps along a generally linear path over the surface upon which the vehicle is supported. For example, when the vehicle is placed within a tubular conduit having an average inside diameter a little less than the maximum overall dimension of the bristles measured in that diametrical direction, the vehicle traverses the length of the conduit in this way.

Subject to the foregoing features, the particular detailed form of the surface traversing vehicle according to the present invention will depend upon the general nature of the surface to be traversed.

For example, the shape of the bristle-carrying bodies may be generally flat when the vehicle is to be used to traverse a generally planar surface or when it is to traverse a space between two adjacent generally planar, generally parallel surfaces. For use upon or within tubular conduits they may be elongate in the direction of the length of the vehicle or relatively short in that direction and are preferably rotationally symmetrical about that direction. Thus, for example, they may be generally cylindrical in shape. However, they may also be non-symmetrical or irregular in shape, in order to correspond to the cross-sectional shape of the conduit upon or within which they are intended to be used.

The bristles extending from these bodies are resilient and are directed generally towards the surface which is to be traversed. For example, if that surface is a single generally flat surface, then the bristles may extend generally parallel to each other in a single direction away from each of the bodies. If the surface is one of a pair of such surfaces, then the bristles will normally extend in two opposite directions. When the vehicle is intended for use within a generally tubular conduit, the bristles are directed outwardly from the bodies. However, the bodies may alternatively be of generally annular cross-section, with the bristles directed inwardly, for use to traverse the outside of a chimney, post, cable or the like.

The bristles upon which the bodies are supported may extend, when in an unstressed state, in a direction which is generally perpendicular to the surface of the body, for example radially outwardly or inwardly in the case of a generally cylindrical (including annular) body. Alternatively, the bristles may be slightly inclined to that direction. They may all be mounted in mutual parallel or they may be off-set from parallel, for example in pairs of mutually inclined bristles.

When the vehicle according to the present invention is in use, it is necessary for the bristles to be diverted to a greater or lesser extent from their unstressed orientation. The required resiliency of the bristles enables them to return, or to tend to return, to that unstressed orientation and then, if the vehicle is to be reversed, to be diverted beyond that orientation into a new inclination in an opposite direction. The bristles may be natural bristles or may be of any other material having the desired resiliency, for example a synthetic polymeric material or a metal. The material ideally displays a relatively high stiffness coupled with a high rate of elasticity. When the vehicle is used to carry a relatively light load, for example a camera to inspect the surface in question, then synthetic polymeric material bristles, for example of nylon, are suitable. When better traction is required, for example when the vehicle is to tow behind it a relatively heavy load, then metal bristles, for example of steel, are preferred. Mixtures of bristles of different materials and/or of different lengths may also be used.

If, having performed the desired function, the vehicle is required to move in the reverse direction, for example to enable it to be retrieved or because it has encountered an obstruction, it is necessary for the inclination of the bristles to be reversed. This result may be achieved in any of a number of different ways. Since movement of the vehicle over a surface requires one of the bodies to remain stationary while the other moves towards or away from it, the main requirement is to reverse the inclination of the bristles on a first body, preferably that one which is rearmost during the initial forward movement of the vehicle. The vehicle may therefore be constructed with, for example, shorter bristles on the first unit, to enable more ready reversal of the bristles.

As another approach to aiding of the vehicle, one or more of the bodies may be constructed with retractable bristles and/or a mechanism may be provided specifically for the purpose of reversing the inclination of the bristles. As a further alternative, the vehicle may be reversed simply by pulling the whole vehicle, or just the rearmost body, backwards by a distance sufficient to cause the bristles to move to the oppositely-inclined position. This may be achieved by pulling manually upon a line attached to the vehicle, or by, say, operating a pneumatic or hydraulic cylinder included in such a line.

When, as in most operating situations, the vehicle is required to change from forward motion to rearward motion within a limited space, for example within a tubular conduit, the movement of the bristles within that space may be aided by providing means to rotate one or more of the bodies about its axis.

In one form of the present invention, the vehicle is required to traverse a single flat surface. To that end, the vehicle may be retained in contact with that surface simply by gravity. However one preferred alternative, which may then allow the vehicle to traverse an inclined or even a vertical surface, is to provide means whereby to retain the vehicle against the surface magnetically, for example using a permanent magnet or electromagnetically.

The means for moving the bristle-carrying bodies towards and away from each other may take any desired form, being chosen to reflect various factors including the circumstances and/or conditions in which the vehicle is to be used. For example the means may be electrically-powered by a direct electrical line or by a battery, preferably a rechargeable battery. In one preferred form of the invention, the means is a pneumatic or hydraulic cylinder, by means of which the bodies may be moved apart when operating fluid is supplied to the cylinder and moved towards each other when the fluid flow is reversed.

The linking together of the bristle-carrying bodies may be rigid or relatively so, especially when the vehicle comprises only two such bodies. However, in general it is preferred that the bodies be flexibly interconnected, in particular to enable the vehicle to traverse non-linear, for example curved or angled, conduits.

While the bodies are supported upon the bristles which extend from them, some of the weight of the vehicle may be carried by one or more wheels, for example in pairs, located upon the bodies themselves and/or upon the links, for example pneumatic cylinders, disposed between adjacent bodies. Such wheels also may provide stability to the linear movement of the vehicle, which might otherwise jack-knife in some circumstances.

The vehicle may comprise only two bristle-carrying bodies or may comprise three or more such bodies. In the latter case, it is preferred that the mutual approaching and moving apart of adjacent bodies be phased so as to lead to a sequence of such movements along the length of the vehicle, thereby smoothing out the progression of the vehicle along the conduit. However in one alternative arrangement, the bodies may be coupled together in pairs, the two bodies in each pair being coupled at a fixed distance apart, to enable the effective length of each body to be increased.

The operation of a vehicle according to the invention comprising three or more of the bristle-carrying bodies, by bringing about the relative movement of adjacent bodies in a pre-determined sequence, may be effected automatically by means of a suitable controller, which may be located upon the vehicle or remote from it; in the latter case, an electrical link from the remote location to the vehicle may be by means of a direct electrical line - or a radio link may be provided for the purpose.

When the vehicle is designed to be operated pneumatically, an air line may be provided from a remote source of compressed air to the pneumatic cylinders. That line may be combined with an electrical line, in the form of an umbilical linking the remote control position to the vehicle. The umbilical may in turn be dragged behind the vehicle by means of a similar towing vehicle specifically provided for that purpose. As the vehicle proper moves further from the control position, supplementary such umbilical tugs may be added. Sensors in the line may monitor tension in the umbilical and in turn prompt an umbilical tug to respond by accelerating or decelerating briefly.

The vehicle according to the invention may be used for a wide range of purposes in a wide number of situations. It will most usually carry or convey a tool to apply some treatment to the interior surface of a tubular conduit, for example to clear debris or growth therefrom, or some form of monitoring device or instrument, for example to survey or explore the shape or condition of such a conduit. Thus it may be used in mine shafts, in chimneys, in tunnels and in pipes conveying utility services such as water, electrical and gas pipelines, telecommunication lines and sewers. In other forms, it may be used to traverse the space between parallel surfaces, for example between the hulls of a twin-hulled tanker or other sea-going vessel, or to survey or treat a single planar surface.

The vehicle is particularly suitable for use in hazardous environments, for example where there may be a risk of fire and/or explosion, because it does not require to have any electrical or electronic components.

The invention is further described and illustrated with reference to the accompanying drawings, which illustrate, by way of example only, one simple embodiment of the vehicle according to the present invention and wherein:

Fig. 1 is an elevation of the vehicle; and

Figs. 2 to 4 show, to a smaller scale, three successive positions of the vehicle in use within a pipe.

The illustrated vehicle comprises two short generally cylindrical bodies 10, 11, linked together by a pneumatic cylinder 12, to which the two bodies are pivotally coupled at 13 and 14 respectively.

Each of the bodies has a substantial number of resilient bristles 15 extending radially outwardly from around its curved surface.

Figs. 2 to 4 show how the vehicle is able to progress, from right to left as illustrated, along a pipe 16, only a short part of the length of which is illustrated. The vehicle is introduced to the pipe at its right-hand end 17 and, since the inside diameter of the pipe 16 is somewhat less than the maximum overall lateral diameter of the vehicle between the ends of the bristles 15, the bristles adopt a position in which they are curved and inclined towards the right, at an average angle of the order of between 15 and 45 degrees.

In order to advance the vehicle along the pipe in the direction of the arrow 18, air is introduced into the cylinder 12 and the bodies 10, 11 are thereby urged apart. The orientation of the bristles on the body 11 resists rearward movement of that body and the body 10 is therefore thrust forwards, the rearwardly-directed bristles thereon offering less resistance to that motion,so that the bodies adopt the positions shown in Fig. 3. Upon subsequent evacuation of the cylinder 12 (Fig. 4), the body 11 is drawn forwards towards the body 10 until the cylinder is fully retracted as shown. As will readily be understood, alternate extension and retraction of the cylinder thus causes the vehicle to advance, progressively and stepwise, through the pipe 16 in the direction of the arrow 18.

In experimental use, the illustrated vehicle has been shown to be able to advance vertically, horizontally and at intermediate inclinations along a tubular conduit and to take with it loads substantially greater than its own weight.

CLAIMS

1. A surface traversing vehicle comprising at least two bodies interconnected by means to move adjacent said bodies towards and away from each other, each said body being supported upon a multiplicity of resilient bristles extending from it.

2. A surface traversing vehicle as claimed in Claim 1, wherein each body is generally flat.

3. A surface traversing vehicle as claimed in Claim 2, wherein the bristles extend from a single flat face of each body.

4. A surface traversing vehicle as claimed in Claim 3, having means to retain the vehicle against a surface magnetically.

5. A surface traversing vehicle as claimed in Claim 2, wherein the bristles extend in opposite directions from opposite flat faces of each body.

6. A surface traversing vehicle as claimed in Claim 1, wherein each body is rotationally symmetrical about the length of the vehicle.

7. A surface traversing vehicle as claimed in Claim 6,

wherein each body is generally cylindrical and the bristles extend outwardly from the bodies.

8. A surface traversing vehicle as claimed in Claim 6, wherein each body is of generally annular cross-section and the bristles extend inwardly from the bodies.

9. A surface traversing vehicle as claimed in any of the preceding claims, wherein the bristles extend generally perpendicular to the surface of the body, or slightly inclined to that direction.

10. A surface traversing vehicle as claimed in any one of 1 to 8 claims, wherein the bristles are mounted off-set from mutual parallel.

11. A surface traversing vehicle as claimed in any of the preceding claims, wherein the bristles are natural bristles or of a synthetic polymeric material or a metal.

12. A surface traversing vehicle as claimed in Claim 11, wherein the bristles are of nylon or steel.

13. A surface traversing vehicle as claimed in any of the preceding claims, wherein the bristles are of different materials and/or of different lengths.

14. A surface traversing vehicle as claimed in any of the preceding claims, having retractable bristles and/or a mechanism for reversing the inclination of the bristles.

15. A surface traversing vehicle as claimed in any of Claims 1 to 13, having a line which includes a pneumatic or hydraulic cylinder to enable movement of the vehicle to be reversed.

16. A surface traversing vehicle as claimed in any of the preceding claims, having means to rotate one or more of the bodies about its axis.

17. A surface traversing vehicle as claimed in any of the preceding claims, wherein the means to move the bodies towards and away from each other is electrically powered.

18. A surface traversing vehicle as claimed in any of Claims 1 to 16, wherein the means to move the bodies towards and away from each other is a pneumatic or hydraulic cylinder.

19. A surface traversing vehicle as claimed in any of the preceding claims, wherein the bodies are flexibly interconnected.

20. A surface traversing vehicle as claimed in any of the preceding claims, having one or more wheels located upon the bodies and/or upon the links between adjacent bodies.

21. A surface traversing vehicle as claimed in any of the preceding claims, comprising more than two said bodies, coupled together in pairs at a fixed distance apart.

22. A surface traversing vehicle as claimed in any of Claims 1 to 20, comprising three or more said bodies, each interconnected for movement towards and away from the adjacent body or bodies.

23. A surface traversing vehicle as claimed in Claim 22, wherein relative movement of adjacent bodies is effected automatically by means of a controller.

24. A surface traversing vehicle as claimed in any of the preceding claims, having an umbilical linking the vehicle to a remote control position.

25. A surface traversing vehicle as claimed in Claim 24, further having a similar vehicle for towing the umbilical.

26. A surface traversing vehicle as claimed in Claim 25, having one or more sensors to monitor tension in the umbilical.

27. A surface traversing vehicle substantially as hereinbefore described with reference to, and as illustrated in, the accompanying drawings.