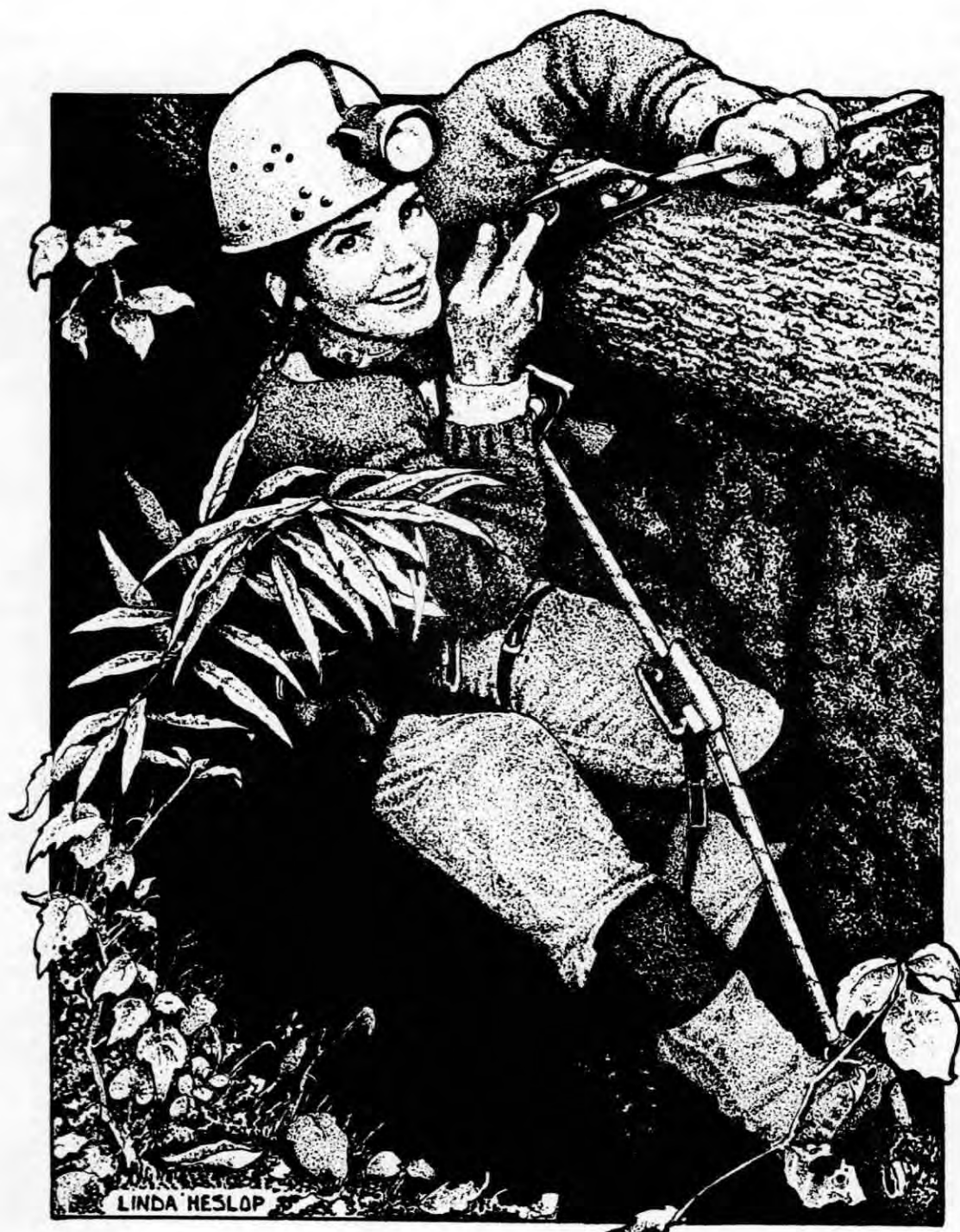


NYLON HIGHWAY

NO. 42



...ESPECIALLY FOR THE VERTICAL CAVER

NYLON HIGHWAY

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The Nylon Highway is published by the Vertical Section of the National Speleological Society on a semi-annual basis pending sufficient material. It is the intent of this publication to provide a vehicle for papers on vertical work. All submitted articles containing unsafe practices will be returned to the author.

Opinions expressed herein are credited to the author and do not necessarily agree with those of the Vertical Section, its members or its Executive Committee.

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Message from the Editor

Wm Shrewsbury, NSS #22677RL

In this issue we have an excellent assortment of articles. Gary Storrick's 'Prusik Rappel Safety questions if it is indeed a 'safety.' After reading his article, the answer is pretty obvious. The Coefficient of Friction of Three Common Ascenders - Matthew Barn's article may surprise you.... Art Fortini performed a simple yet 'common' testing method on webbing and cordage. Last, but definitely not least, is the our feature article, 'Rope System Analysis', by Stephen Attaway. You'll definitely modify your climbing and bolting methods after reading this one!

You will note that there are a couple of blank spots - like the inside of the back cover. Sure, I could move the Nylon Highway advertisement inside. It sure would be nice to have that great photo of yours on the cover somewhere.....

If you have an article or photo that you think we could use, send it to us. I know there are some cavers out there working on new gear (I'm involved in several myself!) What better way to introduce it than through the Nylon Highway? Send your articles and photos. I use MS-Word, but can read most formats. Reasonable handwriting is gladly accepted. E-mail submissions are even better.

Stay tuned for the next exciting issue featuring articles by Bruce Smith, Wm Shrewsbury and YOU! I for one can't wait to read about them....

Cave Softly and Carry a Long Rope,

Wm Shrewsbury, Editor

Front Cover

Hoopers Well, Alabama -
Drawing by Linda Heslop.

Right

Editor exiting a pulldown
through Carpenter-Swago, a wet multi-
drop system in West Virginia. Photo
by Liz Shrewsbury.



Prusik Rappel "Safety"

by Dr. Gary D. Storrick (Trafford, PA)

Introduction

A few months ago there was a discussion on the Internet (in rec. climbing) about the pros and cons on using a Prusik knot as a safety while rappelling. I made a few short comments that basically said I was against it and thought it might even be a dangerous practice. Naturally, I got a few comments back including many requests for more information. I replied with a long post, which Magnus Homann thought enough of to place in "The Climbing Archive" on the WWW (www.dtek.chalmers.se/Climbing/index.html)! The post was originally published without copyright and anyone is free to reproduce it.

Remember, this post was written for climbers and climbers often must rappel on double ropes. I never intended for this to be a "Nylon Highway" article, which explains the weird introductory paragraphs and also the unusual (for cavers) subject matter. The Editor thinks this is worth printing so I added a couple of figures at his request, inserted this introduction, fixed a few typos, but did little else. I hope, for once, I can get a "Nylon Highway" article out without the attendant hate mail from readers who don't like it. With that said, here is the post:

The Post

This is a more detailed post intended to explain some of my earlier posts discussing the use of a Prusik knot as a rappel safety. I'm putting this out for information only and not to try to get others to follow my advice. Each climber is responsible for their own decisions regarding the safety techniques they use. I have no ax to grind, I'm merely passing on my reason for coming to the decision not to use a Prusik Safety. THE REASON I'M POSTING SUCH A LONG DISCUSSION IS THAT I'VE RECEIVED NUMEROUS E-MAIL MESSAGES ASKING FOR MORE INFORMATION. Here it is:

My background is 25 years of caving, taken quite seriously. I collect rappelling and descending devices (many of the devices

shown in Tom Martin's *Rappelling* are from my collection). I was a reviewer for Allen Padgett and Bruce Smith's *On Rope* and have written several articles in the NSS Vertical Section's newsletter "The Nylon Highway". I don't know everything, but I'm also not a beginner. I don't do nearly enough climbing (I climb 5.n, where n is too small to print), but I have done enough to know the difference between rappelling above and below ground. I make the hopefully unnecessary apology for the gender pronouns used herein. I didn't invent the language, neither did the people I'm quoting.

My attitude is that safety is not given by any gadget but is the property of one's attitude and experience. In *Advanced Rockcraft*, p. 66, Royal Robbins wrote, "Safety in rock climbing lies almost entirely with this 'judgment' area. Little is left to chance. Equipment is a minor factor. With the best equipment in the world, the man with poor judgment is in mortal danger." True words, I feel, when applied to rappelling. I also believe that reliance on Prusik or other mechanical safety devices often causes a person to relax their vigilance. 'I can't prove this, I just believe it. Even if we assume that the presence of a Prusik Safety won't affect our judgment, will it work if needed? The answer appears to be "maybe" but most likely not.

Don Davison, Jr. (then Chairman, NSS Safety and Techniques Committee) discussed this in the August 1976 *NSS News*. On page 140 Don writes, "There were several difficulties which caused the use of the chest safety Prusik to be generally abandoned; the most significant being the requirement that the knot be released during a period of accelerating stress. The caver was asked to relinquish a 'firm' grip on the rope and relax in a panic situation. In releasing his grip, he was asked to perform a negative action, a type of behavior which only strenuous and repetitive drill can instill in the majority of individuals".

Don continues: "The tremendous urge of the rappeller to grip the rappel rope (already in his hand) during the period of stress was documented by the use of Dan Meier's three-rope rig ("The Tech Troglodyte", Vol. III, No. 2, Winter, 1965, pp. 31-33). [What if anything goes wrong. I don't recommend trying it, it can be as dangerous as many other aspects of climbing :^). Make your own choice---gds]. To set up a three-rope rig, a rappel point is rigged on a 60-100 foot high free fall cliff so that its end hangs down only about 20 feet. A second rope which reaches the ground is rigged from the same anchor. An overhead belay, well to the side of the main anchor is established with a third rope that can reach the ground.

In practice, the rappeller rigs his rappel device into the short rope and places his chest safety on the third rope to allow the rappeller to fall about half the total height off the drop before being caught at a measured height of about 20 feet above the ground [this is critical-gds]. Before the rappeller begins to descend the belayer 'locks off' the belay rope at the measured distance-thus the rappeller is already caught and the only variable is whether the rappeller will release the Prusik or be caught by the overhead belay. Most experienced cavers were not able to release the Prusik, especially when closed eyes were required after the free fall portion of the drop had been reached [closing the eyes makes the loss of control more of a surprise-gds]. The three rope rig was developed after the May 21, 1964 accident in Newberry-Banes Cave, Virginia ("The Tech Troglodyte", Vol. III, No. 1, Fall, 1964, pp. 18-21). In this accident, a caver "rode the Prusik down the rappel rope for over 100 feet before the knot was relieved of human interference when the caver's head struck a ledge."

Davison goes on to describe a 'Safety Rappel Cam' which he developed and I won't go into. It is difficult to build, only works on a single rope, and never became popular. In fact, other than Don and I, I don't know of anyone who actually built one.

In the January 1977 NSS News, "Hits And Near Misses", (p. 18), a caver

relates a story where two Prusik safeties failed to grab on a 200 foot free drop on Goldline. "All of a sudden I started falling real fast. I couldn't grab the rope, the Prusiks slid along with me and I dropped 110 to 120 feet until I hit a ledge with my feet-damaging my legs. The Prusiks grabbed and I swung about 60 feet across the pit where my head impacted the wall, cracking my helmet, and finally stabilized hanging upside down from the Prusiks 30 to 40 feet off the floor...Injuries included: Left femur broken just above the knee, head of left femur badly cracked, left ankle severely sprained, right heel fractured, deep gouge in right knee, bruised ribs, rope burned palms of both hands."

Don's comments in the evaluation included, "The failure of the chest Prusik to function was again caused by the human element when the victim 'instinctively grabbed the rope'; functioning was proper when relieved of human interference [but look what it took to relieve the human interference-gds]. Reliance upon the use of the safety Prusik should not be encouraged among human troglodytes."

It doesn't have to be a Prusik. In the June 1977 NSS News (p 128), another caver used a Gibbs ascender as a chest safety. "I don't know what caused it, but the rope slipped from my right, and braking hand, and swung to the other side of my body, where I couldn't get at it. I gripped 'the rope' instinctively with my left hand which was on the Gibbs, and rode it down for the ninety feet....Result: Broken right femur, lacerated right knee, assorted bruises, eight weeks in traction, four to six months to regain full use of the leg, possible operations to repair ligament and tendon damage."

The evaluation comments "The victim had used the chest safety Gibbs, during practice. ONCE AGAIN, the very real danger inherent in using safety rappel devices which are held "open" manually is well documented. In a time of stress, the 'negative action' of relaxing the grip around the rope, be the hand directly on the Gibbs, chest safety Prusik, etc., is an unlikely probability. No rappel safety device which requires a 'negative action' to activate

should be trusted. Hopefully, no more injuries will occur before cavers realize the dangers inherent in the various 'negative action' rappel [safety] devices."

In the July 1977 NSS News (p 148): "The inherent problem of human interference with the function of the 'negative action' chest Prusik safety has been well documented (NSS News, Aug. 1976, pp. 140-1; _ "Off Belay", December 1976, pp. 14-17); but this hazard is also characteristic of any rappel safety which is held 'open' on the rope above the descending device. Some cavers, however, feel that the use of a hand held Gibbs or Jumar will eliminate the danger—not realizing that the problem is not in the device but rather the human. During any period of unanticipated accelerating stress, a caver who is not heavily conditioned will tend to grip the rope and, in turn, the hand held Prusik, Gibbs, or Jumar."

The "Off Belay" article just referred to is Ray Smutek's 'The Questionable Prusik Safety'. It is too long to reproduce here. If you can get a copy, read it. It also provides relevant quotes from the "Tech Troglodyte" articles cited previously, which would be difficult to obtain at this time.

From the Troglodyte quotes: "The instinct to grab onto something when falling is very strong. During the use of this rig [the three-rope], five experienced cavers tried it out. Only two let go of their chest Prusik on the first drop. The other three panicked in various degrees, freezing on the rope. If this can be taken as a valid sample..." It can't. Cavers have repeated these tests (albeit undocumented in most cases), and the results indicate that the success rate is much lower than the 40% indicated by the first five.

Mr. Smutek's article goes on to discuss the possibility of the Prusik 'safety' melting or failing—both possible, but not my main point. Mr. Smutek concludes, "The protection provided by a Prusik 'safety' is highly suspect and quite possibly the illusion. Couple this with the puissance of the accidental jamming and the danger of getting hung up and it appears to me that a Prusik 'safety' does nothing except

needlessly complicate and already complex maneuver."

One alternative that seems better than the chest Prusik (but still not adequate to me) was presented by Larry Penberthy in "Off Belay" No.16, pp. 10-11. Since "the trouble with the chest/Prusik system is that a beginner [anyone-gds] may lose control, start to slide rapidly, panic and then grasp the Prusik even tighter, thus preventing it from working. It is contrary to instinct to let go of the rope to gain security." So, "At an MSR 'working' field trip, we devised a new method of security for rappel. The rope from above passes through a [rappel device] and then down to a security knot [Mr. Penberthy recommends either a Penberthy or a Penberthy-Pierson knot, not a Prusik-gds] attached to a webbing loop around one thigh [no days, attached to the harness-gds]. When descending normally, the braking (lower) hand grips the knot to prevent it from grabbing, and simultaneously applies enough friction to control the rate of descent.

As the climber descends, the rope slides upwards through the security knot, and then through the friction device. If the climber lets go with his braking hand completely, the knot grabs and stops him. If he grips the control knot tightly in panic, the extra braking friction force stops him, assuming the friction device has a high enough friction ratio."

But there is a problem and Mr. Penberthy recognized it: "CAUTION: The security knot must not be able to touch the [rappel device]. If it does, the security knot will not grab." My experience suggests that this disadvantage is enough to be a problem, so I do not use this technique.

"Off Belay" No. 30, p. 37 describes a suffocation death when a chest Prusik locked off. The climber was dead within about 30 minutes. In the evaluation, the Prusik safety is mentioned and "its use is a questionable practice".

"Off Belay", June 1977: An article titled 'The Prusik Safety Strikes Again', tells (quoting 'Mugelnoos') of a rappeller who lost control of her rappel, then fell ten feet until

her shirt tail tangled in her rappel device. At that point the Prusik safety engaged. "The girl was extremely lucky, since a Prusik Safety will NOT stop a climber once he or she begins to fall freely, unless something else slows the climber to a near standstill. In this case, it appears that the friction of the rope running across her back, and probably more important, her shirt tail jamming in the brake brought her fall to a stop, permitting the Prusik to grab."

In his book *Single Rope Technique: A Guide for Vertical Cavers* (Sydney Speleological Society Occasional Paper No. 7, 1977), Neil Montgomery writes "The value of a climbing knot [as a rappel safety] is questionable since in a panic it is likely that one will keep tight hold on the knot and prevent its gripping." I disagree with the follow-up statement that an ascender is better, for the reasons given earlier and because that is not what they were designed for. Neil describes the "Spelean Shunt" made from a Gibbs and a carabiner. A small minority of vertical cavers use these; most of us don't. Since they work on single ropes only, I won't describe them. I personally don't like them but as there are a few vertical cavers whose expertise I respect do, I mentioned them here.

Prusik rappel safeties are not mentioned, as far as I could find, in David Judson's (ed.) *Caving Practice and Equipment* (British Cave Research Association, 1991), but autolock descenders (e.g., Petzl Stop, Diablo, Dressler, SRT, Gemlock, Tracson, Lewis, etc.) are. Most of these devices are too heavy for most climbers to want to carry them. In the rappelling chapter (p. 57), Dave Elliot writes "One notable disadvantage of most existing autolock designs arises from the need to release the handle for the device to lock on the rope, whereas a thoughtless or panicking caver might instinctively grip the handle tighter and only worsen the situation." Mr. Elliot says essentially the same thing in his book *SRT* (Troll Safety Equipment Co., 1986, p. 8), once again not mentioning the Prusik Safety.

The use of a Petzl Shunt as a safety is mentioned in the early edition of Mike Meredith's book *Vertical Caving* (ca. 1979)

in a single sentence on page 23. The second (revised and enlarged) edition (1986) removed this reference. These books describe the continental European approach to vertical caving. Petzl still mentions the Shunt as a rappel safety in their catalog, but includes a warning that "releasing the shunt is essential for it to function properly."

Allen Padgett and Bruce Smith's discussion of the Prusik Safety in *On Rope* (National Speleological Society, 1987) explains, "It was learned through several bad accidents that if a problem occurs, instead of letting go, the natural reaction is to grab. Grabbing a Prusik allows it to slide down the rope, traveling faster every instant. If a person is actually able to come to his senses long enough to let go of the knot, the sling material may disintegrate, allowing the climber to descend even more rapidly than before. In actual usage, the Prusik Safety has proven to be troublesome and dangerous." They then mention the Spelean Shunt, Safety Rappel Cam, and Petzl Shunt as options, but correctly note that "No safe-belay device should interfere with rappelling technique. If it does interfere, it is counter productive in that it exists to help control a problem, but instead tends to create one."

One last quote, from John Long, on page 155 of the second edition of *How To Rock Climb*, "A sliding knot backup (commonly referred to as a Prusik backup) is rarely if ever used as a normal procedure. If you don't know how to rappel, get a belay. If you are doubtful that you can make a certain rappel, don't make that rappel. Only if you are doubtful and must rappel, and no belay is possible, should you consider the Prusik backup as an option..." All told, the Prusik backup is a highly contested technique. The only thing for certain is that it can be highly problematic.

The use of the Prusik Safety may be highly contested among climbers but not among cavers. It is almost universally rejected.

The Coefficient of Friction of Three Common Ascenders

by Matthew T. Barns (Bon Air, VA)

This experiment was performed by Vertical Section member Matthew (Matt) T. Barns, NSS #33543, when he was a fifteen year old freshman in high school. The paper was written and submitted for review to the Virginia Junior Academy of Science. Over 10,000 papers are submitted each year, and Matt's was one of approximately 700 accepted for presentation at the 1993 VJAS Convention; acceptance is an honor unto itself. Matt designed the experiment, performed the data gathering and statistical analysis and wrote the paper.

Several other Vertical Section and NSS members provided mentorship in physics, mathematics, statistics, and scientific report writing; all subjects to which Matt was new as a freshman earth science student. - Tray Murphy, NSS #29211

Abstract

Ascenders are mechanical cams that apply inward pressure on a rope in order to keep the ascender and the load securely on the rope. The ascender with the lowest coefficient of kinetic friction can be moved up the rope with the least amount of energy expended by the caver. With this in mind, the purpose of this project was to find the ascender with the lowest coefficient of kinetic friction. This was done by measuring the amount of force necessary to move each of the ascenders along a standard caving rope at a constant velocity. It was hypothesized that there would be a statistically significant difference between the three coefficients of friction. Three different types of ascenders were used to test the hypothesis. The ascenders used in this experiment were the Gibbs ascender, the Petzl ascender, and the Jumar ascender. The means of the coefficients of friction were found to be 0.787 for the Gibbs, 0.713 for the Petzl, and 0.473 for the Jumar. A t-test was performed which indicated there was a statistically significant difference between the Gibbs and the Jumar, the Petzl and the Jumar, and no statistically significant difference between the Gibbs and the Petzl. These results supported the hypothesis in two out of three cases.

Introduction

This project was intended to find the mechanical ascender with the lowest amount of friction by measuring the amount of force necessary to move each of the ascenders along a rope at a constant velocity. Ascenders are mechanical cams that open to allow a load to move up a rope when force is applied in an upward direction and prevent it from sliding back down by applying downward force on the ascender, thus closing the cam and preventing downward movement of the load. They are connected to a standing rope and the load is affixed to the lower attachment point. An upward force is applied to the ascender that causes it to slide up the rope. There are two major types of ascenders, the open shell and the closed shell ascender. The open shell ascender is designed to come off the rope quickly and easily with one hand. Open shell ascenders have a toothed cam that is spring-loaded to ensure positive closure on the rope. The cams of the two open shell ascenders used differ in the number and angle of the teeth on the face of the cam. The Jumar Ascender has 44 small straight protrusions about five-tenths to one millimeter long that grip the rope at a 90 degree angle (see Figure 3 in Appendix A). The Petzl has 24 teeth approximately the same length projecting downward at a ten degree angle (see Figure 4 in Appendix A). The closed shell ascender, Gibbs, has a shell in the shape of a taco shell. The cam pivots between the outer walls of the shell. Because of this design, it takes two hands to place the ascender onto the rope. The cam uses ridges and valleys to grip the rope rather than teeth. The cam on the Gibbs was also spring-loaded. The pivot point on the Gibbs ascender is directly between the face of the cam and the load attachment point (see Figures 5 and 6 in Appendix A), whereas the attachment points on the open shell ascenders are between the face of the cam and the pivot.

This experiment examined the forces acting on the ascenders and from this data, found the coefficient of kinetic friction for each of the three ascenders. The project was based on Newton's second law of

motion, which states "when an unbalanced force acts on an object, the object will be accelerated. The acceleration will vary directly with the unbalanced force applied and will be in the same direction as the applied force. It will vary inversely with the mass of the object." (Murphy, et al., 1986)

Force is a vector quantity that acts on an object whether it is in motion or at rest. The ascenders used in this experiment have multiple forces acting on them (see Figure 1, below). The weight (W) of the ascender is the force acting in the negative

y-axis direction. The force of friction (F_f) is equal to and acting in an opposite direction to the force (F_x), which is acting in the negative x-axis direction. The F_x force is the direction of motion. The normal force (N) is the force that opposes the force of weight on the ascender. The normal force is found by adding the weight to the vertical component (F_y). Friction is the "force opposing motion between two objects that are in contact" (Murphy, et al., 1986). In this case, the two objects were the cam and the shell of the ascender against the rope.

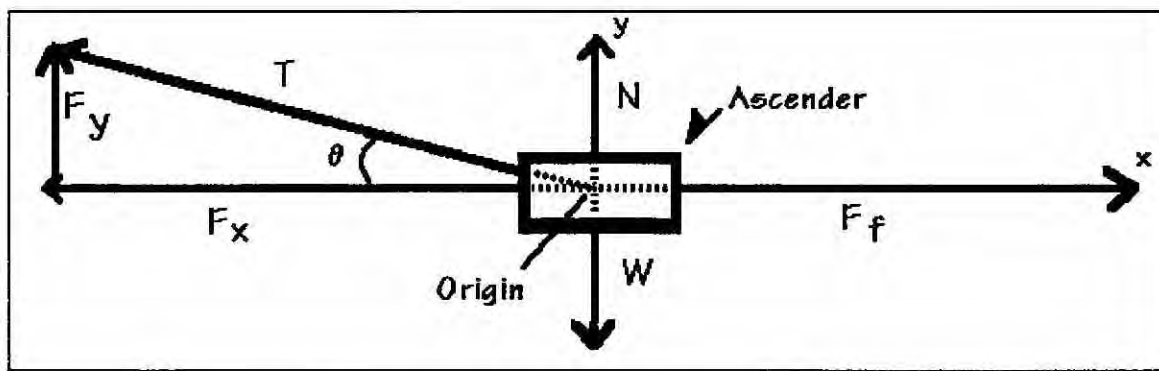


Figure 1. Free-body Diagram of Forces acting on the Ascender

Kinetic friction is the friction that exists when one object is in contact and in motion in relation to the other. The coefficient of kinetic friction is the ratio between the force of friction and the normal force. The coefficient of kinetic friction of the ascender was found when the net force on the ascender was zero. Newton's second law states that force equals mass times acceleration; therefore, if the ascender moves on the rope at a constant velocity, the net acceleration equals zero. Consequently, the horizontal force is balanced by the force of friction. Dividing the force of friction by the normal force gives the coefficient of kinetic friction. The coefficient of kinetic friction is a dimensionless constant. It is almost independent of the area of contact between the surfaces because on smaller surfaces, the forces are concentrated in a smaller area (Williams, et al., 1984).

Cavers use ascenders to climb out of caves into which they have rappelled. The ascender with the lowest coefficient of

kinetic friction can be moved up the rope with the least amount of energy expended by the caver. There are also vertical rope climbing competitions and by using an ascender with a low friction coefficient, the climbing times will presumably be faster. Therefore, finding the ascender with the lowest coefficient of kinetic friction will be of interest to many people in the caving community.

With these applications in mind, the purpose of this project was to test three different types of mechanical ascenders to determine the coefficients of kinetic friction between the faces of the cams and the standing rope. It was hypothesized that there would be a statistically significant difference between the coefficients of kinetic friction of the three ascenders tested, due to the difference in the designs of the shells and cams of the ascenders.

Methods and Materials

Three different types of ascenders were used to test the hypothesis. One

ascender that was tested was the Gibbs ascender, an enclosed shell camming device. The other two ascenders were open shell ascenders, a Petzl and a Jumar. The major difference in these ascenders was the design of the cam itself. The Gibbs uses a cam with ridges running perpendicular to the standing rope. The Petzl and Jumar use toothed cams with the teeth pointed downward to ease the release of the cam with upward force.

A length of standard 11 millimeter (7/16 inch) static nylon kernmantle caving rope was used to assemble the experiment. This type of rope is the standard rope that cavers use in vertical work. The length of rope was attached to anchors at either end, in a horizontal configuration. Since the coefficient of kinetic friction will not change with the orientation of the rope it was possible to use this horizontal experimental setup to simplify data collection. The rope was tensioned by a come-along in order to ensure a constant angle between the hauling line and the horizontal rope.

A tower was constructed on a wagon. One end of a spring scale (resolution: 1000g x 10g) was attached to the top edge of the tower. The other end of the spring scale was attached via a short piece of four millimeter accessory cord to the load attachment point of the Gibbs ascender. The hauling line was attached to the load attachment point of the Gibbs (see

Figure 8 in Appendix B). Measuring the tension in the hauling line in this manner was necessary so that the horizontal and vertical components of this force could be broken down later mathematically. The angle formed by the hauling line and the horizontal rope gave the angle θ used in calculations of these forces. The ascender was pulled along the rope by the wagon at a velocity of 25 centimeters per second. A constant velocity was achieved in the following manner: The rope was marked off into 25 centimeter increments. A stop watch was set to beep once a second. The wagon was pulled at a speed at which each mark and beep occurred simultaneously. This interval was selected after several trial runs, because it was found to be a convenient interval to achieve a constant velocity. An assistant kept the ascender in the proper orientation, that is, upright; and read the spring scale as the ascender was pulled. The ascender was pulled over a total distance of nine meters. Thirty trials were performed with the Gibbs ascender. The ascender was massed on a triple beam balance. From this, the weight was calculated.

This exact procedure was repeated for the Petzl ascender and again for the Jumar ascender. A diagram of the experimental setup can be found in Appendix B.

Results

The force that was measured with the thirty trials was force T . The tension T can be broken down into its horizontal and vertical components. The horizontal component of T is F_x . This was extracted by the following equation:

$$F_x = T \cos \theta \text{ (Equation 1.) (All equations from Serway, 1990)}$$

F_x , being equal and opposite to F_f , provides F_f .

$$F_x = F_f \text{ (Equation 2.)}$$

The vertical component of T provides F_y . This value was obtained by the following equation:

$$F_y = T \sin \theta \text{ (Equation 3.)}$$

When added to the weight of the ascender it provided the normal force (N).

$$N = F_y + W \text{ (Equation 4.)}$$

Then F_f was divided by the normal force (N) in order to provide μ_K .

$$\mu_K = \frac{F_f}{N} \text{ (Equation 5.)}$$

In all cases, forces were measured in kilograms and converted to Newton's with the following equation: $W = mg$.

The Gibbs weighed 0.203 kilograms on the triple beam balance. The tension (T) for the Gibbs ascender ranged from 0.190 to 0.245 Newton's. A coefficient of friction was calculated for each data point (T) using equations one through five above. The coefficients ranged from 0.740 to 0.906. The mean of these coefficients of friction was 0.787. The standard deviation was 0.041.

The Petzl weighed 0.177 kilograms. The tension (T) ranged from 0.150 to 0.170 Newton's. The calculated coefficients of friction ranged from 0.682 to 0.756. The mean of these thirty coefficients was 0.713. The standard deviation was 0.025.

The Jumar weighed 0.267 kilograms on the triple beam balance. The range of the tensions (T) was only 0.140 to 0.155

Newton's. The thirty coefficients of friction calculated for the Jumar ranged from 0.451 to 0.508, with a mean of 0.473. The standard deviation was 0.014.

A Student's t -test was run comparing each set of coefficients to the others. With a set at .05 and with 58 degrees of freedom, the critical value was 2.001. The t value for the Gibbs - Jumar test was 39.009, which indicated there was a statistically significant difference between the two sets of data. The t value for the Petzl - Jumar test was 8.277, which again showed a statistically significant difference between the two. The final test, between the Gibbs and Petzl, gave a result of -44.782, which indicated there was no statistically significant difference between the two ascenders' data.

Discussion & Conclusion

The purpose of this experiment was to find the mechanical ascender with the least amount of kinetic friction by measuring the amount of force necessary to move each of the ascenders along a rope at a constant velocity. The means of the coefficients of friction (μ_k) were 0.787 for the Gibbs, 0.713 for the Petzl, and 0.473 for the Jumar (see Figure 2).

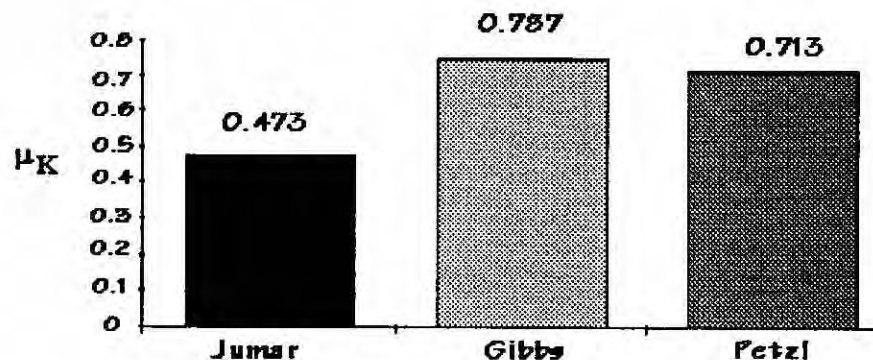


Figure 2. Chart of the three coefficients of kinetic friction

Therefore, the ascender with the least coefficient of kinetic friction was the Jumar. Some coefficients of kinetic friction have been tested and run the range from rubber on concrete with a μ_k of 0.8, to the synovial joints in humans with a μ_k of 0.003 (Serway, 1990). More specifically, these coefficients fell in between the coefficients of kinetic friction of smooth aluminum on smooth steel ($\mu_k = 0.47$), and the coefficient of kinetic friction of rubber on concrete ($\mu_k = 0.8$) (Serway, 1990). The shells of the ascenders were constructed of anodized aluminum (Gibbs and Petzl) or

painted aluminum (Jumar) to reduce rope drag. Also, the cam faces were anodized or polished for the same reason. The rope was made of woven nylon filaments so slick that one can not climb hand over hand as can be done with manila rope.

One of the reasons there were differences in the coefficients of kinetic friction may be explained by the ascenders' designs. The Jumar ascender had only four centimeters of its shell touching the rope at any given time. This limits the amount of friction created by the shell against the rope. Both the Petzl and the Gibbs had about the same coefficient of

kinetic friction. This could be explained by the amount of the shell each had touching the rope. The Gibbs had about ten centimeters of the shell in contact with the rope and the Petzl had about nine centimeters in contact with the rope. Because the shell was touching the rope, a greater F_x value was needed to make both F_x and F_f equal.

Theoretically, there should not be a difference in the coefficient of kinetic friction by switching to a vertical configuration, as opposed to the horizontal configuration tested. In order to improve the procedure, an experimental design that allows a vertical configuration could be used, so that the ascenders would be working in the direction that they were intended to work.

This would allow the ascenders to slide up the rope instead of along it, thus making the data more directly linked to the use of the ascenders. By the use of a winch, the ascenders could be pulled at a more nearly constant velocity than a person could pull it.

The hypothesis stated there would be a statistically significant difference between the coefficients of kinetic friction of all three ascenders. As the t-tests showed, there was in fact a statistically significant difference between the Gibbs and the Jumar, and between the Petzl and the Jumar. There was not a statistically significant difference between the Petzl and the Gibbs. Therefore; the hypothesis was supported in two of the three cases.

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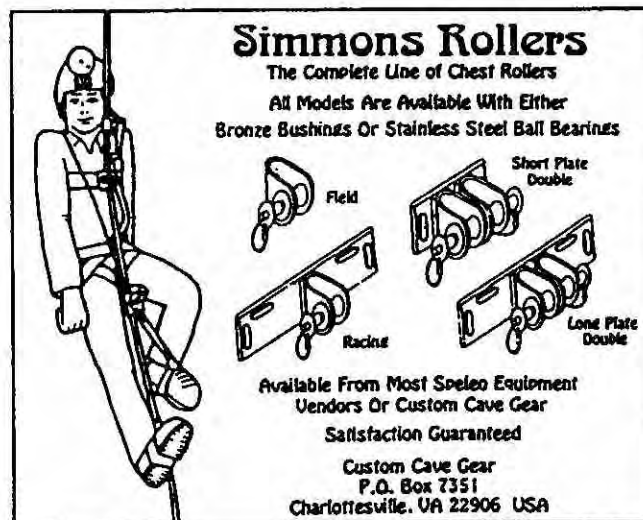
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Appendix A **Diagrams of Ascenders Used in This Experiment**

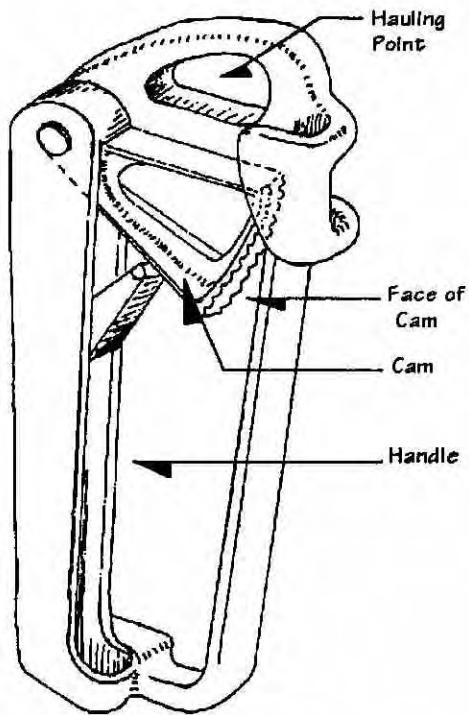


Figure 3. Jumar Ascender

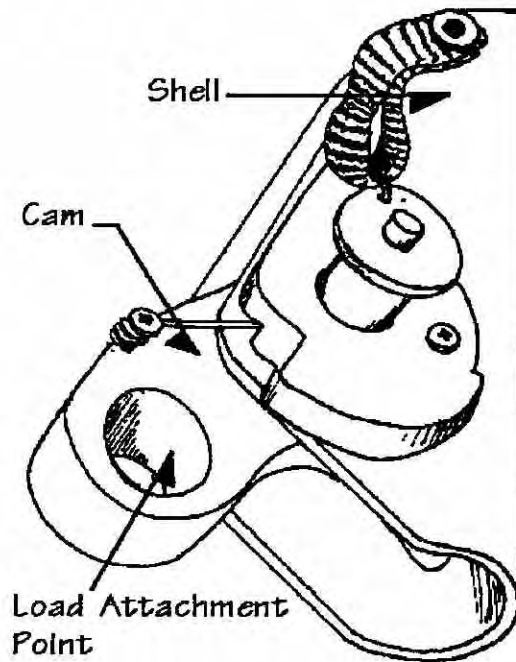


Figure 5. Spring-loaded Gibbs Ascender

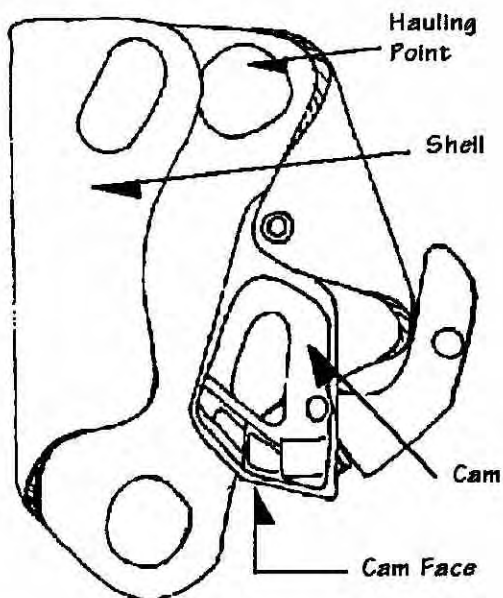


Figure 4. Petzl Ascender

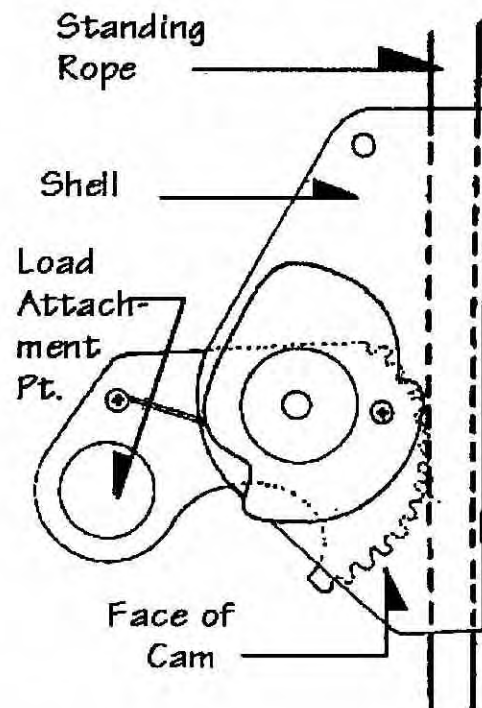


Figure 6. View of internal parts of Gibbs Ascender

Appendix B
Diagram of Experimental Setup

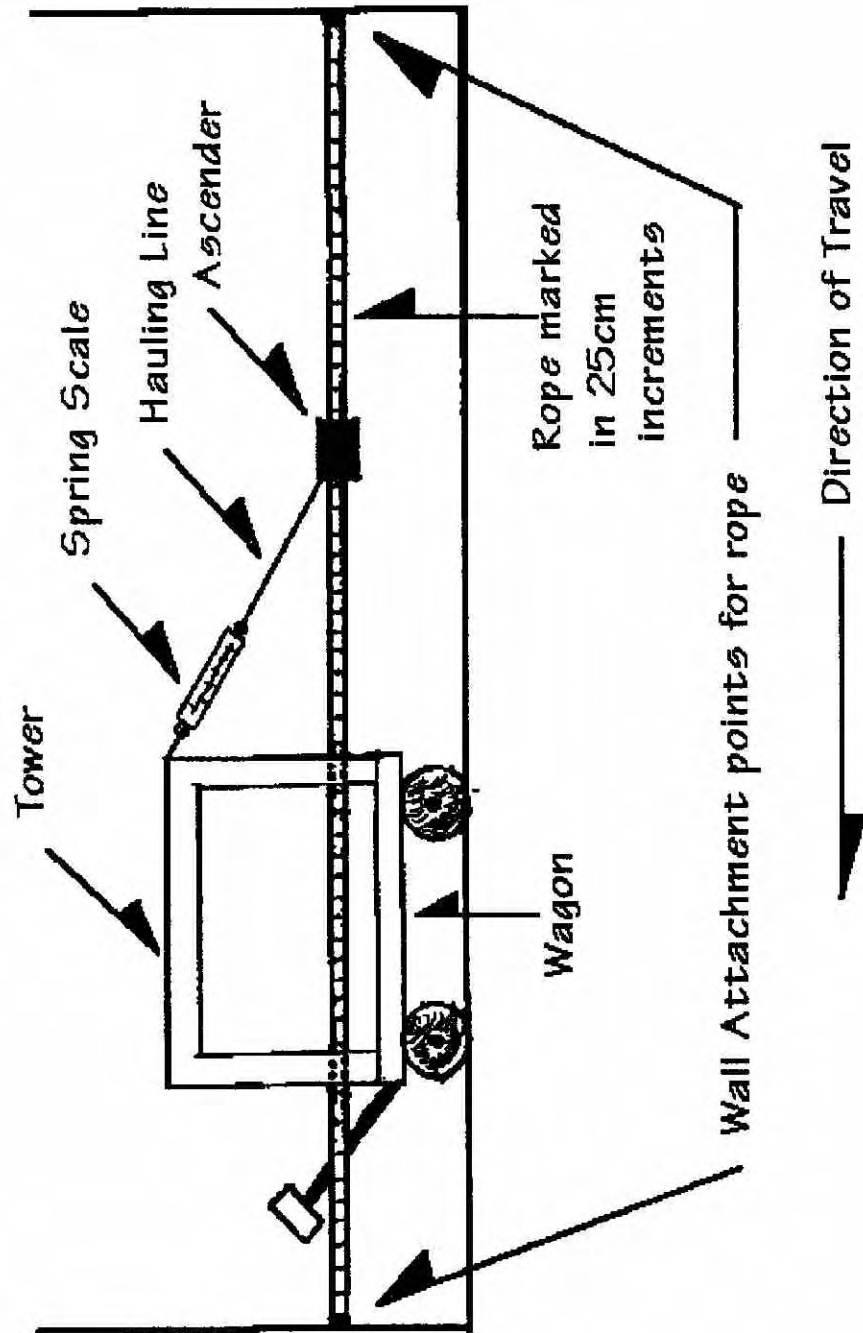
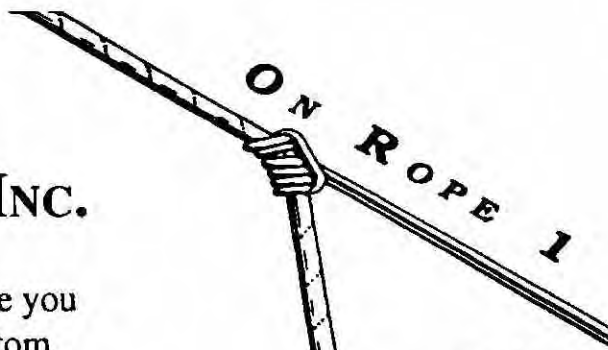


Diagram of experimental setup. (Not to scale)

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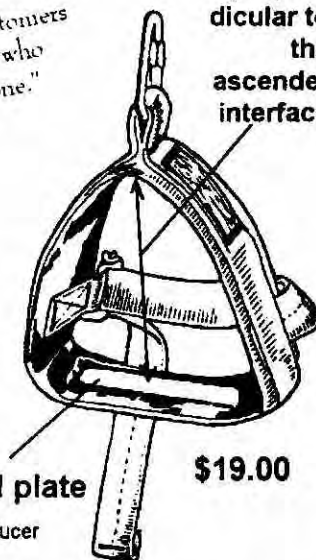
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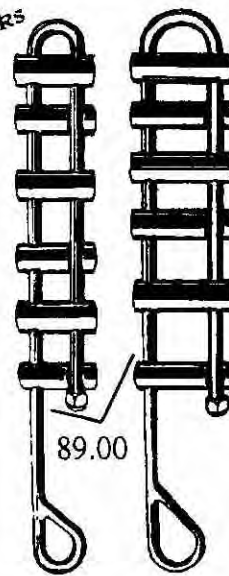
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ABRASION RESISTANCE OF WEBBING AND CORDAGE

by Art Fortini (Pasadena, CA)

By its very nature, cave exploring tends to be extremely abusive on equipment. Because of the inadequacy of standard mountaineering equipment in the cave environment, several equipment manufacturers have begun making gear specifically for caving (PMI ropes, Lost Creek packs, etc.) Unfortunately, not everything we take underground with us is cave-proof, and even the "cave-proof" things we do take eventually succumb to wear and tear.

While cavers may not agree on when a worn out Petzl suit should be retired and replaced, continued use is unlikely to have catastrophic results. Continuing to use worn out ropes and/or webbing, however, can have such results. To further complicate matters, the wear rates of ropes and webbing vary tremendously. Although several factors affect the useful life of ropes and webbing (age, exposure to UV light, dirt, etc.), few can shorten it as quickly as abrasion. In one report, a fixed rope in Lechuguilla Cave was reported to have been abraded to the point where only a few core fibers were left (Richards, 1994.)

The purpose of this study was to quantitatively rank cordage and webbing with regard to their abrasion resistance so that search and rescue personnel, cavers, and climbers can choose their equipment appropriately. It is likely that the UIAA, NFPA, and/or ASTM have formal test procedures for abrasion resistance, and if I were ambitious enough, I could have simply looked up the results. Generally speaking, however, I find it more satisfying to go out and break things myself.

Test Procedure

In a typical search and rescue situation, a rope is connected to a carabiner, which is then connected to a load. As the rope is raised/lowered over lips and ledges, the rope gets pinned between the rock and the carabiner and abrades. Furthermore, as more ledges are encountered, the same point on the rope gets abraded each time. In this case the

rope is bent around the carabiner, and the *outside* of the bend is being abraded.

In a typical caving situation the rope is tied off to an anchor and lowered down the pit. Abrasion at the lip is then due to cyclic stretching of the rope caused by ascending. In this case the rope is being abraded on the *inside* of the bend, rather than the outside.

Because the outside bend of a rope is under slighter greater tension than the inside of the bend, it will tend to abrade more quickly. As a result, a test procedure was developed which would abrade the outside bend of a tensioned piece of rope/webbing. The actual procedure was as follows:

1. A stokes basket was inverted and a 150 lb. block of cement was placed over the foot end, as close to the end as possible.
2. A 12" long, 1" diameter, smooth sided bar from a dumbbell was placed under the foot rail of the inverted litter.
3. The test sample was looped under the bar such that it was being compressed by the weight of the cement and litter.
4. The ends of the test sample were run upwards, and tied to a piece of 11 mm rope.
5. The 11 mm rope was run towards the head of the litter, through a carabiner near the front of the litter, and used as the drag line.
6. The carabiner near the front of the litter served to prevent rotation of the litter while still allowing the rope tension to be transmitted to the test sample (Figure 1).
7. The litter was dragged on a smooth cement floor (a firehouse bay) at a slow walking pace until the test sample failed catastrophically. The force required to drag the litter was ~30 lbs for all of the samples.

The total distance dragged was noted, and the data are summarized in Table 1. With the exception of the 11 mm static rope and 9/16" nylon webbing, all test samples were brand new and unused. The 11 mm rope and 9/16" webbing were ~4 and ~1 year old, respectively.

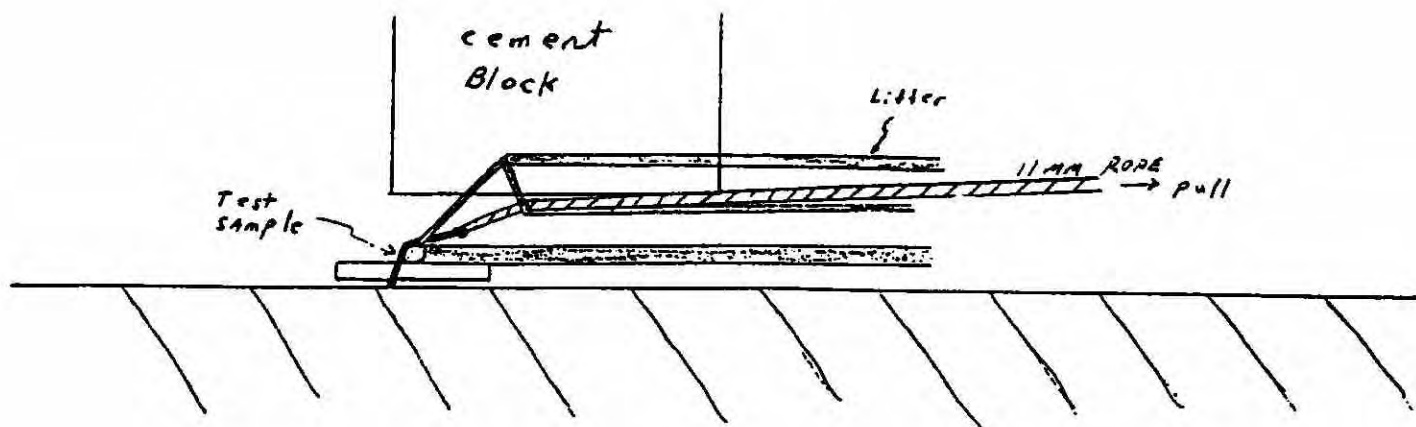


Figure 1

Table 1

Item	Average Distance (ft)	# of Samples tested	Standard Deviation (ft)	Nominal Tensile Strength (kN)	Linear Density ¹ (g/m)
9/16" Webbing	35	2	3	11.3 (2500 lb.)	26.1
1" Webbing	58	3	11	18.1 (4100 lb.)	38.2
9/16" Spectra-tape	161	2	4	14.5 (3300 lb.)	49.1
11 mm Rope Static	339	3	63	30.3 ² (6800 lb.)	81.9 ²
11 mm Rope (Sheath only)	115	1	NA	~3 ³ (700 lb.)	~8
3/8" Rope Static	228	2	11	25.8 ⁴ (5800 lb.)	74.5 ⁴
9 mm Rope Dynamic	230	2	14	18.5 (4100 lb.)	49.9
6 mm Rope Dynamic	113	4	33	8.0 (1800 lb.)	23.0
5.5 mm Spectra Cord ⁵ - 1	43	2	3	17.8 (4000 lb.)	19.5
5.5 mm Spectra Cord ⁵ - 2	60	2	8	17.8 (4000 lb.)	19.5
5.5 mm Spectra Cord ⁵ - 3	84	2	8	17.8 (4000 lb.)	19.5.
5.5 mm Kevlar ⁶	92	2	10	20.0 (4500 lb.)	19.5

¹ Based on REI data sheet, except as noted.

² Based on data from CMC catalog 111 for Wellington Puritan caving rope.

³ Estimated as 10% of the nominal rope strength.

⁴ Based on data from J.E. Weinel, Inc. Catalog #14 for PMI rescue rope.

⁵ Spectra Cords from REI taken from 3 different spools, possibly different manufacturers.

⁶ Although called Kevlar, it was actually a mixture of Kevlar and Spectra fibers (i.e.; what Black Diamond calls Gemini cord).

As can be seen, the abrasion resistance of 1" tubular webbing is quite poor. It was actually pretty scary to see how quickly it failed. The only materials which showed poorer performance were 9/16 tubular webbing (not surprising) and one of the Spectra cord samples. In spite of the fact that the 1" webbing had the largest contact area with the abrasive surface (and hence the lowest force per square millimeter of contact area) it performed very poorly. Due to its construction, all of the fibers in a piece of webbing come to the surface and are subject to abrasion. In a piece of rope, however, the outer fibers protect the inner fibers, thus providing the superior abrasion resistance.

In all of the samples tested, some melting and/or glazing of the material was noted at the point of failure. Heating of some of the fibers to the melting point thus appears to be a significant effect. This could explain why the various Spectra cords (which have a very low melting point) performed so poorly.

Interestingly, the Spectra tape samples fared quite well. In all of the tests done with the Spectra tape, however, the sample rolled into a "rope". This would tend to explain why it performed better than the 9/16" nylon webbing (which did not roll up).

With the Kevlar samples, melting/glazing was seen primarily on the nylon sheath and the Spectra fibers. The Kevlar fibers did not appear melted/glazed, but it was difficult to tell.

No significant difference in abrasion resistance was noted between static and dynamic ropes, although the published strength and density data for 9 mm and 3/8" ropes varies significantly.

Although a loose correlation exists between nominal strength and abrasion resistance (Figure 2), the 5.5 mm cordage, as well as the 1" webbing, are outliers on the low side of the norm.

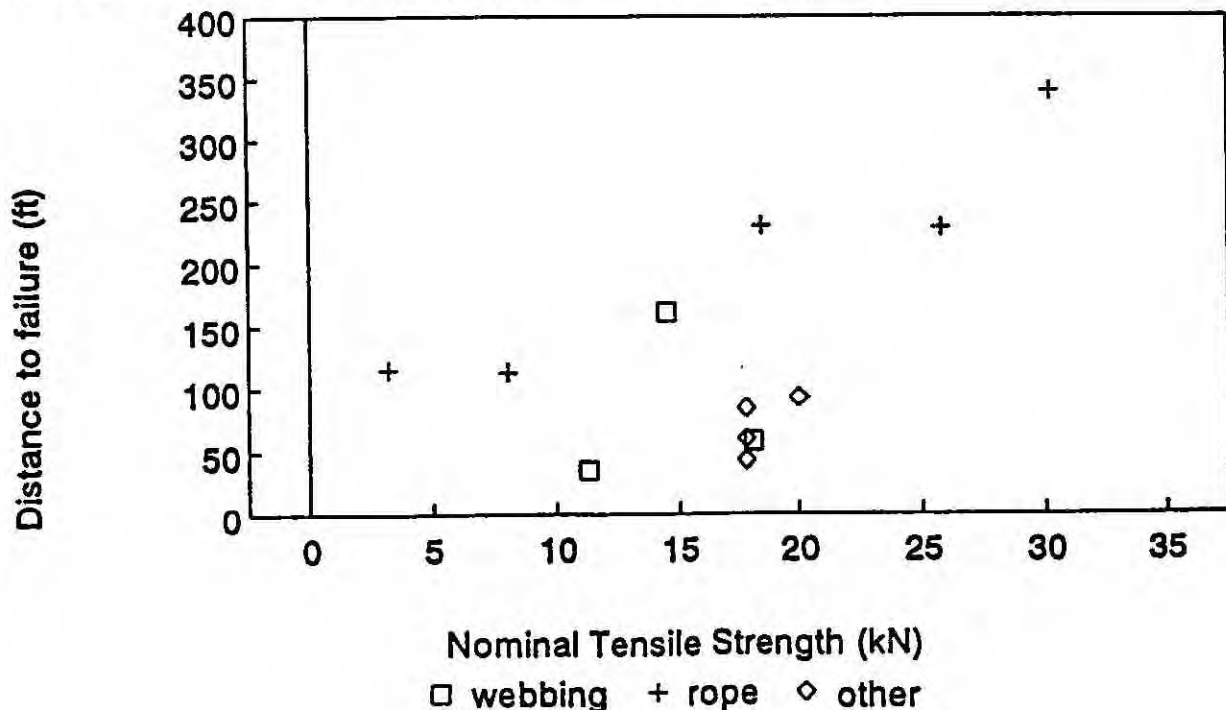


Figure 2

Conclusions

1" tubular webbing, as well as Kevlar and Spectra cord, lose their strength very rapidly in an abrasive environment. In the testing done, failure occurred when the tensile strength of the test item dropped below the force required to drag the litter; i.e., ~30 lb. In actual field use, the conditions will be much more severe than the test conditions used here. Specifically:

1. The dynamic tensile loads associated with a caver ascending a rope, or the static tensile loads associated with a loaded litter being raised or lowered over a lip, will be much greater than the ~30 lbs which caused failure in these tests.

2. The compressive force squeezing the rope/webbing against the floor during testing was constant at ~150 lb. A small caver with a light pack would simulate this, however, a heavy caver or a caver with a multi-day pack would apply much more than 150 lbs of compressive load between the rock and the rope/webbing. The same is true for a loaded litter.

By the way, this assumes that the compressive force between the rope and the rock is equal to the tension in the rope. This will be true if the rope is bent 60° from straight (i.e., a 120° included angle) as it goes over the lip. As the bend becomes sharper, the compressive load increases rapidly.

3. The cement floor used in the testing was quite smooth and not tremendously abrasive. In actual field use, the rock upon which the rope/webbing will be abrading is likely

to be much rougher and more abrasive.

Recommendations

Because actual field conditions are likely to be much more severe than those used during testing, it can only be concluded that failure will occur well before the distances listed in Table 1. I recommended the following:

1. Don't be fooled by tensile strength numbers. Although 1" webbing, Kevlar, and Spectra cord all possess high tensile strength (~19 kN, ~4000 lb.) and good strength to weight ratios (0.47, 0.91, and 1.03 kN-m/g, respectively) relative to nylon rope (0.36 kN-m/g), they have very poor abrasion resistance.

2. Avoid having long lengths of rope between the anchor and the lip. This is the segment of rope which will stretch and cause the rope to saw over the edge as people ascend. If possible, use an anchor relatively near the lip or double (or triple) the rope between the anchor and the lip.

3. Pad the lip and other abrasion points.

4. Inspect fixed ropes before you trust your life to them, even if the anchor point is a little out of the way or hard to see. I know this is common sense, but on a long expeditions in a multi-drop cave, fatigue (and laziness) can be overwhelming.

5. Use 9 mm or 3/8" rope instead of 1" webbing in applications where abrasion is likely.

Acknowledgments

Funding from the Sierra Madre Search and Rescue Team is gratefully acknowledged as is the assistance of Dave Johnson in performing the testing. My thanks to Ron Miller and Cathy Borer for reviewing the manuscript.

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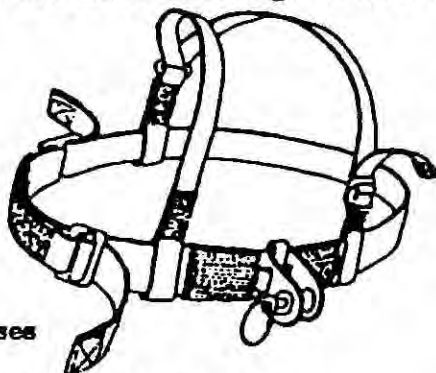
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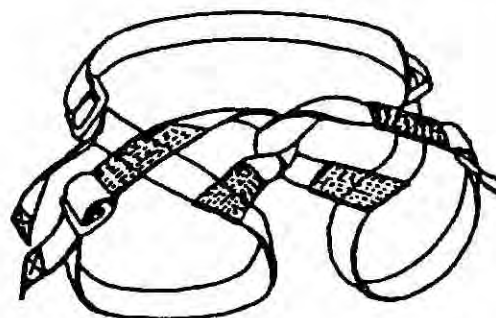
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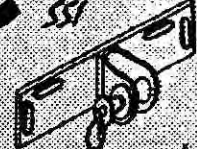
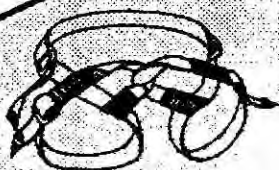
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Rope System Analysis

by Stephen W. Attaway (Sandia Park, NM)

Abstract

This paper presents an analysis of the loads in a typical climbing rope system subjected to a dynamic loading from a fall. Several examples are illustrated to show how to calculate the force on ropes and anchors subjected to dynamic loads that are experienced by a falling rock climber. The force in a rope that is generated when a falling weight is arrested depends on how fast the weight is stopped. We will use the energy method to solve for the maximum strain energy in the rope. The effects of friction, dynamic rope modulus, and rope condition will also be considered.

We developed some rules of thumb to help a lead climber place fall protection and understand the limitations. The amount of 'safe' lead out depended on the amount of rope that is between the belayer and the climber, the type and condition of the belay rope, and the type of anchor used.

Motivation

Rock Climbing is a technical sport. A good understanding of the mechanics of anchor placement, rope behavior, and impact dynamics is important to climber safety.

On June 23, 1996, three climbers fell to their death on the Warpy Moople route on the formation called Muralla Grande located in the Sandia Mountains east of Albuquerque, NM. Warpy Moople is a Grade III, class 5.9 climb with 8 pitches. At least one of the climbers had reached the top of the last pitch, which is rated 5.5-5.6. One plausible scenario is that the first climber reached the top and called "off belay" before placing his top rope belay anchor. The other two climbers may have begun removing their belay anchor and were getting ready to climb when the first climber fell. Only three pieces of protection were found on the 165 rope between the first climber and the next. One question that has been asked is: why did the protection fail. We know that at least 100 ft. of rope was between the lead climber and the belayer. We do not know the location of the protection. Two pieces of protection were of the cam type design, and the third was a chock with a wire. These three experienced climbers fell to their deaths believing that the cam type protection device could protect a fall of over 100ft without pulling out. Many other experienced rock climbers that I interviewed also believed that 50 to 100 ft. lead outs were 'safe'.

Here I will try to show that the amount of 'safe' lead out depends on the amount of rope that is between the belayer and the climber, the type and condition of the belay rope, and the type of anchor used. In the rest of this paper we will outline some of the methods of rock climbing and fall protection. Then, some equations that are useful for understanding rope behavior are derived. These equations will be applied to a typical leader fall to predict the magnitude of forces that an anchor must withstand. The effects of dynamic stiffness,

friction, rope condition, and belay device will be considered. Finally, we will consider how much "lead out" is safe for a given rope and anchor system. Several areas for research on dynamic rope behavior are suggested.

Methods of rock climbing fall protection

Some of the typical styles of rock climbing are shown in Figure 1. Top-roping has a belayer at the top of the climb with a near taut rope going down to the climber. If the climber falls, the rope is weighted quickly by the climber's weight.

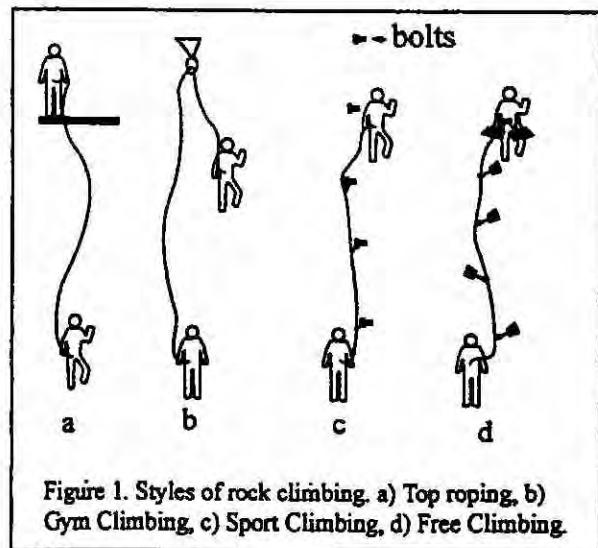


Figure 1. Styles of rock climbing. a) Top roping, b) Gym Climbing, c) Sport Climbing, d) Free Climbing.

Gym climbing usually has the belayer on the ground with the rope running up to a carabiner or pulley along the top of the climb. Here, if the climber falls, the rope is also quickly weighted by climber's weight, and the climber can be lowered to safety.

Sport climbing involves climbing a wall that has fixed

(permanent) points of protection along its path. Typically rock bolts are placed at short intervals. The climber will start from the ground with a rope being lead out by the belayer. The climber can clip the rope into each fixed protection point using a sling with carabiners. Since the climber may be quite a distance above their protection, a climber fall can lead to high impact forces in the rope and anchor systems.

Free climbing is similar to sport climbing except that there are no fixed points of protection. Instead, the climber must wedge 'chocks' or 'cam' devices into the rock at set intervals. Before the lead climber reaches the end of the rope, a belay station is rigged so that the second climber can be belayed from the top in a top-rope style. The second climber will typically remove the fall protection as they climb. Free climbing requires expert skill in placing the fall protection. If placed incorrectly, the dynamic force from a fall can rip all the fall protection out and lead to a 'grounding'.

Rope Deflection

Before looking at the complex rope system, let's look at the dynamics of a simpler mass-spring model. Mass-spring systems are well studied, and usually engineers and physics majors are tortured by the equations for this system in college exams. A fallen climber on rope behaves somewhat like a mass-spring system. The spring corresponds to the rope, and the mass corresponds to the weight of the fallen climber. An ideal mass-spring system is shown in Figure 2

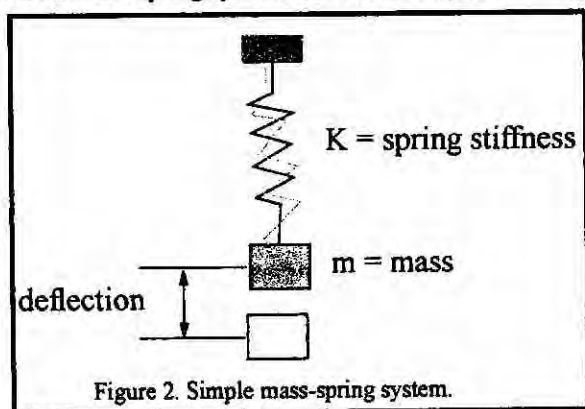


Figure 2. Simple mass-spring system.

If you were to follow a single strand of nylon in a rope, then you would find that it makes a spiral path along the length of the rope. This spiral path looks like the same path that a steel spring makes. This spring-like path is what gives a rope its 'bounce'. As the rope is loaded, the spring is stretched exactly as a steel spring is stretched.

Dynamic ropes are designed to stretch. For example, Gold Line climbing rope has nearly ten times more stretch than that of static PMI. Rock climbing ropes that are designed to take falls are also designed to stretch. Both static and dynamic ropes are made from the same type of nylon (Kevlar and Spectra have too little stretch to absorb energy). If you measured the stretch of a single strand of nylon, then you would find that it is much stiffer than the rope. The rope

construction (number of core strands, twist of core strands, sheath tightness, size, weave of the sheath yarn, etc.) determines the modulus of a rope. (H. C. Wu, 1992, has developed an accurate method to predict the static tensile strength of double-braided ropes based on the above properties.) Note that most webbing does not have a spiral, and, thus, has very little stretch. This makes webbing a poor energy absorber.

Static Deflection

Before we can compute the dynamic forces from a fall, we must first understand how a rope responds to a static load. The force in the rope (spring) is proportional to the amount of stretch in a rope. The relation is

$$F = K\delta \quad \text{Eq(1)}$$

where F is the force in the rope, K is called the stiffness of the rope, and δ is the displacement of the rope from its unstretched length.

The stiffness of the rope, K , depends on the length of the rope. Short ropes are very stiff, and long ropes are less stiff. The stiffness can be computed as:

$$K = \frac{M}{L} \quad \text{Eq(2)}$$

where M is called the rope modulus (change in force for a given stretch), and L is the rope length.¹ The rope modulus is computed from the stretch in a rope under a given load. The modulus is defined as the force per unit stretch:

$$M = \frac{F}{\epsilon}, \quad \text{Eq(3)}$$

where $\epsilon = \frac{\delta}{L}$ is the stretch or the change in length over the length. The stiffness of a rope may change with load or as the rope is used.

Table 1 shows the typical rope moduli for different types of ropes. Notice that the modulus of static rope is four to five times stiffer than dynamic rope.

As an example of using the equations of a spring, consider a weight of $W = 200$ lbs on PMI rope. At $L = 200$ ft., the static deflection should be given by:

$$\delta = \frac{W}{K} = \frac{WL}{M} = \frac{200 \times 200}{19555} = 2.0 \text{ ft.} \quad \text{Eq(4)}$$

If two 200 lb loads were on the rope at the same time, then the stretch would be 4 ft. If the rope were twice as long, say 400 ft., then the 200 lb load would stretch the rope 4 ft., an 800 ft. rope would stretch 8 ft. If Gold Line were used, on an 800 ft. rope, 80 ft. of stretch would be required to support a 200 lb load.

1. A note to engineers: This is not a conventional modulus. It has dimensions of force, not stress.

Table 1: Static modulus of different types of rope.

	Force	stretch (δ/L)	Modulus (lb/ ft/ft)
PMI [1]	176 lbs	0.009	19555
Blue Water II	176 lbs	0.011	16000
Gold Line	176 lbs	0.088	2000
dynamic climbing rope [2]			4000 - 8000

One way to determine the modulus of a rope is to measure the deflection for a given weight on 100 ft. of rope. The static modulus can then be computed based on:

$$M = \frac{\overline{WL}}{\overline{\delta}} \quad \text{Eq(5)}$$

where the bar above the quantities indicates measured results. Here we have used the term static modulus to indicate that the modulus was measured under a static (non-moving) loading condition. Later, we will introduce the concept of a dynamic modulus.

The modulus of a dynamic belay rope will change with use. A fall on a dynamic rope will straighten some of the fibers and cause the rope to become stiffer. The rope modulus can also change if it is used for climbing or rappelling.

Dynamic loads

Rope stretch is important because it governs the distance over which a falling load will stop. The shorter the stopping distance, the greater the deceleration. Since the dynamic force in the rope is equal to the mass times the deceleration, high decelerations mean high loads in the rope. (the fall does not hurt; its that sudden stop at the end).

One way of approximating the maximum dynamic force in a rope system is to use an energy balance equation. (see Spotts, 1978 [4]) The total energy of a fall must be balanced by the total strain energy in the rope. At the end of the fall, the rope will be stretched to its maximum length. At this point, the climber will have just come to zero velocity, and all the energy from the fall will be stored in the rope as strain energy. To compute the maximum load in the rope during a fall, we will need to know the maximum stretch in the rope.

All the energy added to a falling weight as it travels through the earth's gravity field must be converted into strain energy in the rope. The energy balance for the mass-spring system is:

$$PE = SE \quad \text{Eq(6)}$$

where PE is the potential energy, and SE is the strain energy in the rope.

Potential Energy

For a mass moving through the earth's gravity, the change in potential energy is given by:

$$PE = mgh \quad \text{Eq(7)}$$

where h is the height or distance of the fall, g is the gravity constant, and m is the mass. We have neglected air resistance here. One can argue that air drag is proportional to velocity to a good approximation and that for velocities encountered in 'safe' falls air drag is negligible. If you were going to bungee jump from the top of El Cap, then you may want to consider air drag.

If a weight falls a given height, h, the potential energy is converted to kinetic energy according to:

$$\frac{1}{2}mv^2 = mgh \quad \text{Eq(8)}$$

where m is the mass, and v is the velocity. To stop a fall, a rope must absorb all of the kinetic energy, which it does by converting it into strain energy.

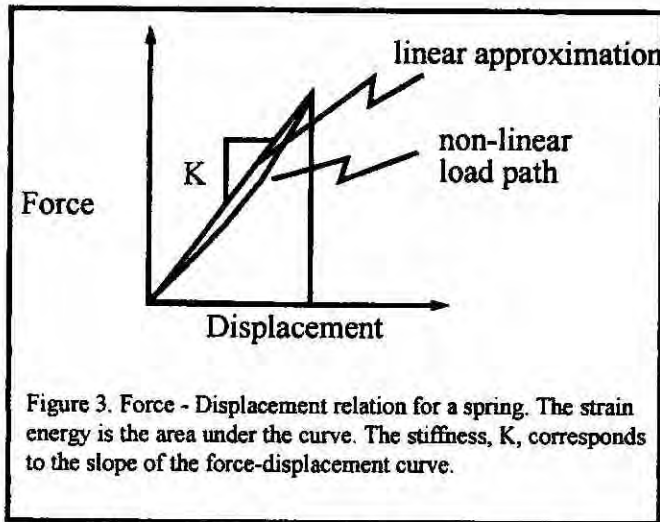
We measure the height, h, in the potential energy equation from the lead climbers location down to the point where the rope has no stretch. Because the mass will move downward as the spring stretches, we must also include the additional change in potential energy due to the stretch of the rope. The total change in potential energy due to the fall from height, h, and the deflection of the rope, δ , will be:

$$PE = mgh + mg\delta \quad \text{Eq(9)}$$

Strain Energy

Now, lets look at the energy used in stretching a spring. The strain energy (or work done) of the spring is given by the force in the spring integrated over the distance which it acts. Because the force changes as the rope is stretched, the strain energy is computed by integrating the force over the displacement of the spring.

$$SE = \int_0^{\delta} F(x)dx \quad \text{Eq(10)}$$



If the spring is linear (meaning it has the same stiffness for a given displacement of interest) then the strain energy is:

$$SE = \frac{1}{2}K\delta^2 \quad \text{Eq(11)}$$

where K is the stiffness in the spring (rope), and δ is the maximum displacement of the spring (stretch in the rope). Note that some ropes, like bungee cord, may not have a linear force displacement curve. The energy method can still be used; however, the math will become more difficult because of the integral in Eq(10). Often, a rope will be 'almost' linear. In this case, we approximate the stiffness of the rope as shown in Figure 3. For now, we will assume linear force displacement relations as a first approximation.

Now that we have defined the relations for strain energy, kinetic energy, and potential energy, we can use these relations to solve for the maximum force generated by a fall.

$$Wh + W\delta - \frac{1}{2}K\delta^2 = 0 \quad \text{Eq(12)}$$

where W is the weight of the climber. Recall that $W = mg$. We can now use the quadratic equation to solve for the displacement:

$$\delta = \frac{W \pm \sqrt{W^2 + 2KWh}}{K} \quad \text{Eq(13)}$$

This simplifies to,

$$K\delta = W + W\sqrt{1 + 2\frac{K}{W}h} \quad \text{Eq(14)}$$

We can express Eq(13) in terms of the static deflection of the rope under the weight, W, by using Eq(4) $\frac{W}{K} = \delta_{st}$.

Recalling from Eq(1) that $F = K\delta$, and assuming K is the

same under dynamic and static loading, allow us to express the ratio of the dynamic force in the rope to the climber's weight in terms of the initial fall height and the static rope deflection. From the solution to the quadratic equation and substitution we get:

$$\frac{F}{W} = 1 + \sqrt{1 + \frac{2h}{\delta_{st}}} \quad \text{Eq(15)}$$

The ratio F/W is called the **impact load factor**. Another way to look at the impact load factor is that F/W corresponds to the maximum number of g's that the climber will experience as he is decelerated by the rope. Typically, a 10g acceleration will cause a jet pilot to pass out due to all the blood being forced from the head to the legs. An acceleration of 16g's will cause damage to humans, i.e. the gravity load will be enough to break bones (your head weighs approximately 15 lbs, under 16 g's it will feel like it weighs 240 lbs).

UIAA limits:

UIAA defines a set of tests for measuring the performance of ropes. UIAA impact force test requires dynamic rope to be designed to limit the maximum dynamic load due to a falling weight of 80 kg (176 lbs) to 12 kN (2697 lb) when dropped 4.8 meters (15.7 ft) onto a 2.8 meter (9.2 ft) section of rope. The rope is passed over a 10 mm radius edge to simulate a carabiner. The test approximates a worst case fall (that's a fall factor 2, or falling twice the length of the rope). The impact force on the weight is limited on the first fall to 12 kN, and the rope must survive 4 falls. By limiting the impact force on the worst case fall, this test sets the design load that the rest of the climbing system must endure.

For purposes of comparison, here are the UIAA recommended minimum limits for strength in the safety system:[2]

Table 2: UIAA recommended Limits

Device	Minimum Limits
Anchors	25 kN (5620 lb)
Carabiners	20 kN (4496 lb)
Slings	22 kN (4945 lb)
Harnesses	15 kN (3372 lb)

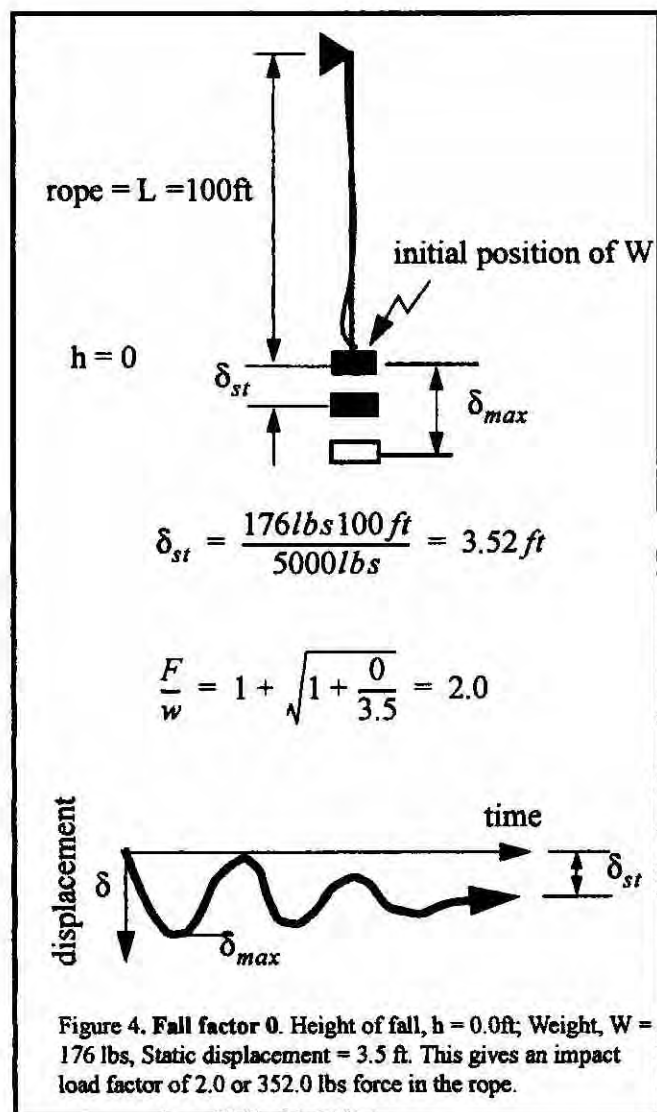
By designing ropes to generate no more than the UIAA limit of 12 kN, the forces in that the different components of lead climbing protection should have an upper bound.

Fall Analysis.

In this section, we will illustrate how to compute the impact load factor for some different types of falls. In all the examples we will assume the belayer is using a dynamic rope and that the rope does not slip in the belay device.

Example: Fall Factor 0.

Let's assume that a climber falls while being topped roped with a dynamic rope and that the belayer has all the slack out of the rope. Figure 4 shows the geometry for such a fall. Notice that for this fall, h will be zero, and the impact load factor will be 2.0. At first glance, this does not seem correct. In order to understand why the impact fall factor should be 2, think about a weight at the top of a ladder with the rope just barely taut. If someone kicks the ladder from under the weight, then the weight will stretch the rope and bounce. The force at the peak stretch in the bounce will be a factor of two higher than the static force (i.e. when you are not bouncing).



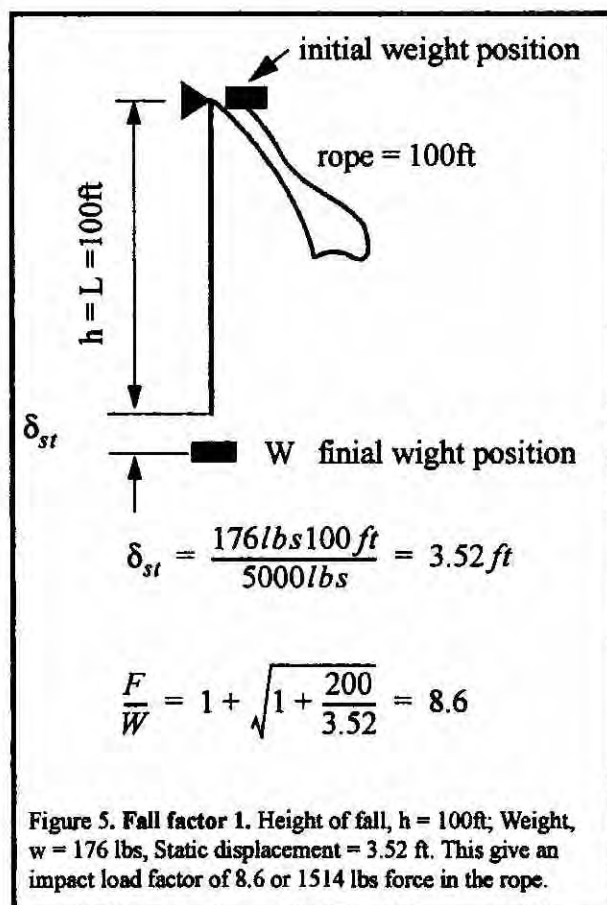
For an actual system, it is easy to measure δ_{st} : simply hang on the rope. If δ_{st} is very small compared to h , then the dynamic force in the rope will be very large. Thus, you should be able to get a good idea of how much impact load-factor will be generated by bouncing on the rope. If you

bounce your weight on the rope system, and you do not move very much, then it probably is not a good idea to fall.

Also shown in Figure 4 is a plot of what the displacement (force) would look like as a function of time. The displacement in the rope starts out at zero and climbs to its maximum. Once the peak displacement is reached, the climber will bounce about the static deflection until his motion is damped out by internal friction in the rope. Some ropes will have more damping than others. The peak force for a fall factor of zero will be the same regardless of the length of the rope. The duration of the force will increase as the rope length gets longer.

Example: Fall factor 1.

Now let's consider the case where a climber wants to test her rope. She goes to a nice high bridge and ties a 100 ft rope to the guardrail. After attaching the other end to her seat harness, she JUMPS off the bridge! Figure 5 shows the geometry for this fall factor of 1.



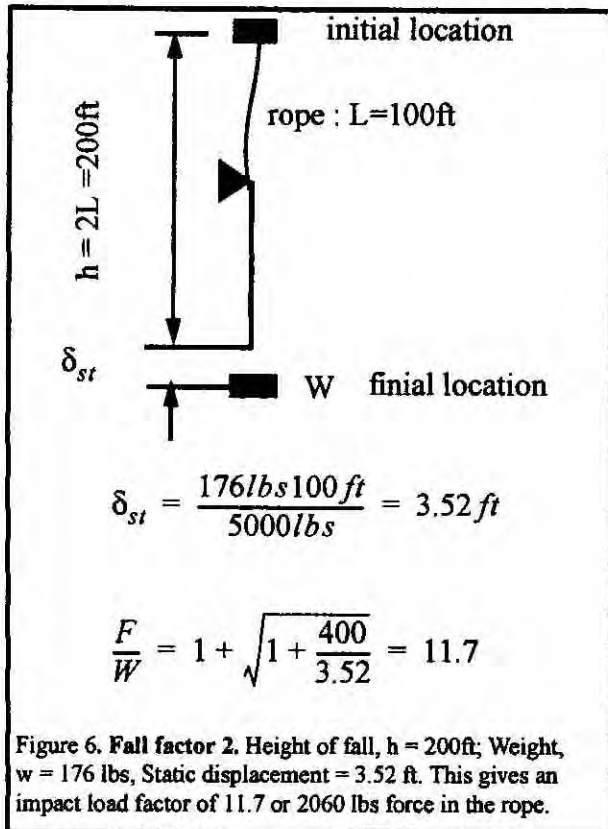
In this case the rope and the height of the fall is the same. For the case shown, the impact load factor will be just over $F/W = 8.6\text{ g's}$, or $F = 1514\text{ lb}$. If we assume static rope like PMI, then the modulus would be much higher and give $F/W = 15.1\text{ g's}$ or $F = 3035\text{ lbs}$ (don't try this at home! I know of two people that have taken 100+ ft. falls on PMI. In one case, the rope outer sheath was cut by their jumars, the sheath slid

down the rope 1.5 feet, and the core melted and fused for 2 feet. The climber was uninjured and climbed back up the rope. I assume they did not reach the full 15 g's because of the energy absorbed by sheath cutting and slipping).

Example: fall factor 2.

Ok, now lets consider what has to be the worst case fall factor. A climber is the full rope length above the belayer as shown in Figure 6 The fall would be twice the length of the rope, or a **fall factor of 2**. Even though the fall is from twice the height, the impact load factor only increases to 11.4. Most people consider the fall factor of 2 to be the worst case impact load. It is important to note that a fall factor of 2 can be generated on a very short fall.

By measuring the dynamic load on a worst case fall, manufacturers can give their rope customers an idea of how stiff their ropes are compared to others. For rope comparison, manufacturers list the impact force for their rope assuming a fall factor of 1.78 with a weight of 80 kG (176 lb). Table 3 lists some typical Manufacturer's rope specifications.



Example: The effect of rope length.

The example shown in Figure 7 shows two different scenarios. One that most sane climbers will avoid because of fear: a full rope length, fall factor 2 screamer. The other scenario is one that we all see on every climb that has a hanging belay: a 10 ft fall with only 5 feet of rope out. The

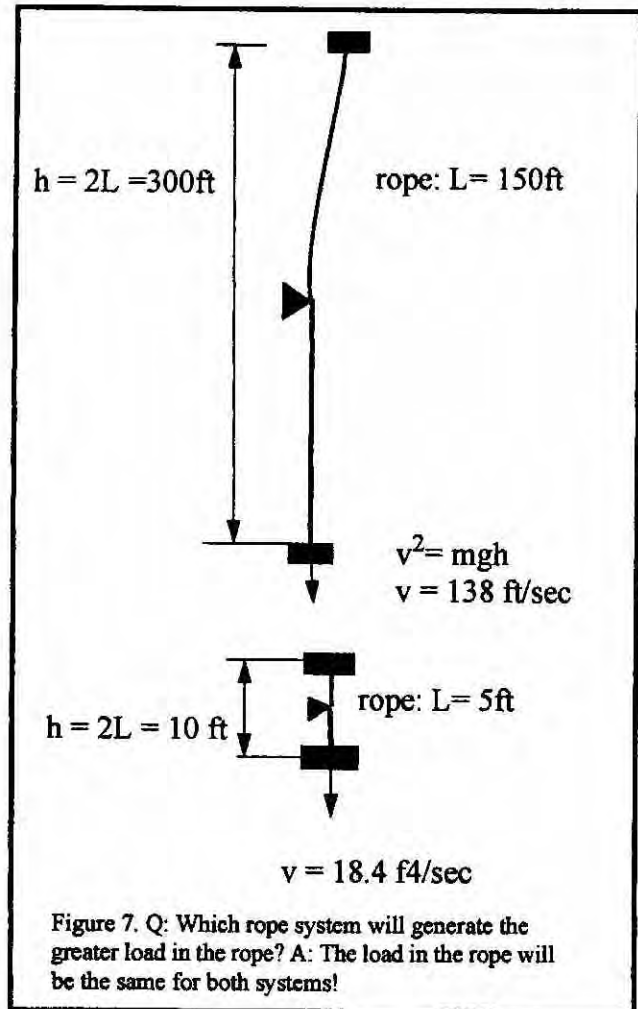


Table 3: Typical Rope Specifications^a for UIAA fall test.

Diameter mm	Impact Forces kN (lbs)	Impact load factor F/W (g's)
9.8 (a)	10.7 (2400)	13.6
10.5 (a)	11.4 (2562)	14.5
10.8 (a)	10.2 (2292)	13.0
11.0 (a)	10.2 (2292)	13.0
10.0 (b)	9.8 (2203)	12.5
10.2 (b)	9.9 (2225)	12.6
10.5 (b)	9.4 (2113)	12.0
11.0 (a)	9.7 ()	12.3

a. New England Rope, b. Mammut Rope. [3]

peak force for both of these falls will be the same. Even though the first fall is higher, there is more rope to absorb the fall energy.

The difference is that in the 300 ft fall, the duration of the force will be much longer than for the short fall. Even though the short fall's duration is much smaller, believe me, it will still yank very, very hard on your anchor and your body.

Leader Fall Analysis

Ok, so we conclude for the above analysis that the rope can catch a fall twice the length of the rope. Now, let's look in more detail at just what happens during a leader fall. Figure 8 shows a typical leader fall. The leader rope length (i.e. the "lead out") is L_2 , above his last protection. The climber will fall a distance $h = 2L_2$. The static deflection is based on the total length of rope $L = L_1 + L_2$. As an example consider $L_1 = 10$ ft. and $L_2 = 20$ ft. Assume a rope modulus of $M = 5000$ lbs/ft./ft. and a $W = 176$ lb climber. This will give a load factor of $F/W = 10.0$. The load in the rope would be $F = 1760$ lb.

To see this in more detail, let's go back to Eq(14) and insert the rope stiffness directly from Eq(2). This gives

$$\frac{F}{W} = 1 + \sqrt{1 + \frac{2hM}{WL}} \quad \text{Eq(16)}$$

where L is the total length of the rope, $L_1 + L_2$, and h is the distance that the climber falls, $2L_2$.

At the anchor point, the rope doubles back to form a 2:1 mechanical advantage. This generates twice the load at the anchor if we assume no friction. (The effects of friction will be considered later.)

Now, suppose that the climber is 100 ft. above his last anchor, $L_2 = 100$, then the total rope length will be $L = 110$ ft., to give a fall factor of 11.2. This would generate 1973 lbs of force in the rope and 3946 lbs of force on the anchor.

Typical cam anchors fail between 2000 - 3000 lb of static load, while chock type anchors on wires will also fail between 2000 - 3000 lbs. Hex style anchors on Spectra webbing typically are rated to around 5000 to 6000 lb. Bolted anchors in hard rock are typically good for 5000 to 6000 lb. So, you see, when we combine the 2:1 mechanical advantage with the impact load on the rope, we can easily generate enough force to pull out cam and wire type anchors. So, now for the question that every rock climber has asked at least once while on the rock.

How far can I lead out and still be safe?

Let's say you are 15 ft above a cam anchor with a rated strength of 2500 lbs and there is 50 ft of rope between that placement and the belayer. Just how safe are you? Figure 9 shows a plot that can be used to estimate the anchor forces generated for a given combination of lead outs.

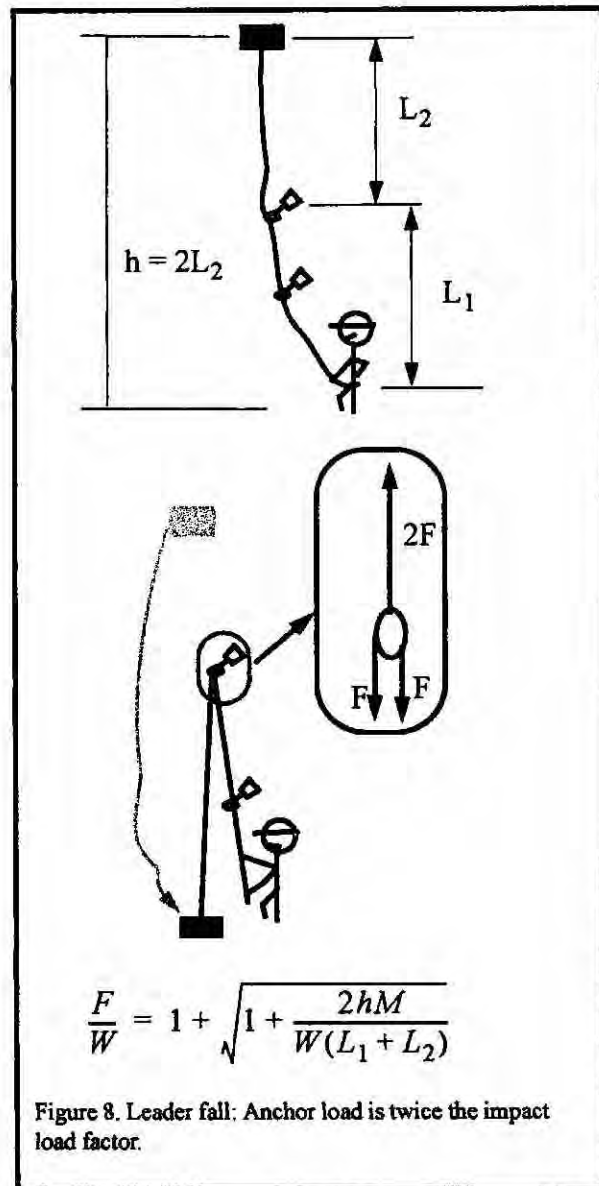


Figure 8. Leader fall: Anchor load is twice the impact load factor.

To see how to use this graph, let's consider a couple of examples. Let's look at a climber that is 20 ft above his last anchor, $L_2 = 20$, and a total of 40 ft of rope out, or $L_1 = 20$. Thus, the climber will fall $h = 40$ ft with $L = L_1 + L_2 = 40$ ft. (fall factor 1). This places him at point A on the plot which corresponds to an anchor impact force of 3400 lbs. Since a cam anchor cannot support this much impact force, we hope that a good strong bolt is used for this anchor.

First, let's consider a bolted route. Typically, bolt manufacturers claim that a well placed bolt can support 5600 lbs (25 kN). If we assume an old bolt in weak rock, such that the bolts can only support 3500 lb, then we can take a 20 ft lead beyond the last piece of pro at 20 feet. We can take an 80 ft lead from 80 feet. If we really have a bomb proof (nuclear, that is) anchor (5500 lbs), then we can take a 300+ fall and not fail the anchor. (Don't try this without lots of overhang: at 300 ft, the rock will be going past you at 95

m.p.h. You would not want to get out of your car if it was going this fast).

Now, let's look at cam anchors. Cams are typically strength rated for 2500 lb. We will assume that because of rock conditions, placement, etc. that a cam can support only 2000 lb of anchor force. Huber, 1995 has summarized some important findings on the strength of camming anchors and suggests that 2000 lbs may be overly optimistic.

Under these assumptions, a 176 lb climber would be limited to 7 ft of lead out above her pro at 40 ft, (point B), 15 ft. of lead out at 90 ft (point C), and 22 ft of lead out at 140 ft.

In general, anchors can be divided into two classes: high strength and low strength. The high strength class would include well placed bolts and well placed hexes on strong cord. The low strength class would include all active camming devices and nuts on wires. The difference between high strength anchors and low strength anchors needs to be emphasized. The failure to distinguish performance of the two allows some very bad assumptions on what is a safe lead out.

One problem is that many of the climbers are taking long leader falls on bolted routes, and then expecting their cams and wire chocks to hold similarly on lead climbs. A general rule of thumb for the low strength anchors is to not lead out more than 1/4 the length of the rope between the belayer and the highest piece of protection.

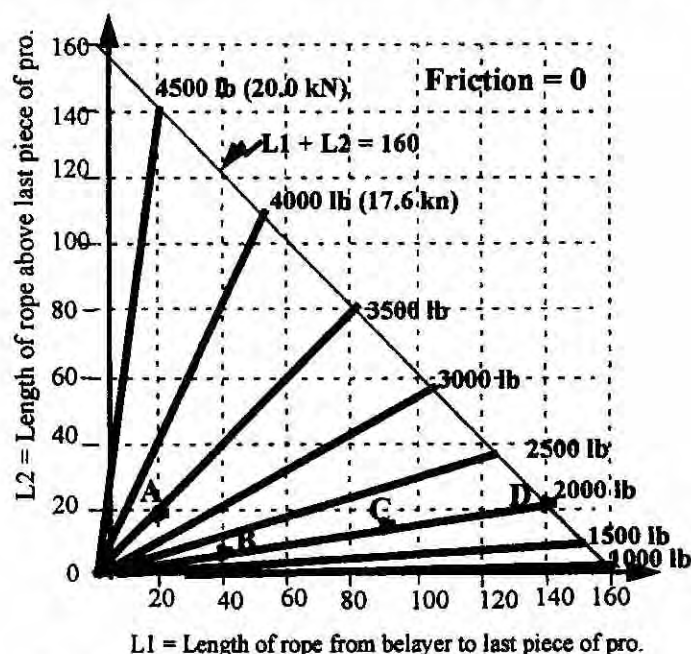


Figure 9 Plot of impact load on anchor for $W=176.0$ lb, $M=7000$ lb., for different combinations of lead out above last protection and distance from belayer to protection.

An interesting exercise is to work out the optimal number of anchors required for a 160 ft lead climb. First, we will assume that the first anchor is a great one, and that we can

get off the ground 6 ft. A fall from just one foot above this anchor would fail a 2000 lb anchor! Ok, so we put in a nice 4000 lb hex and climb to 12 ft and place a cam. At this point, we can climb about 2.5 ft above the 12 ft placement before the cam will overload at the 2000 lb limit. Now at 14.5 ft, we can climb another 3 ft to the next anchor, at 17.5 ft., 4 more ft to 21.5 ft, 5 ft to 26.5, 6 ft to 32.5, 7 ft to 39.5, 9 ft to 48.5, 12 ft to 60 ft., 14 ft to 75 ft, 17 ft to 91 ft, 19 ft to 110 then 25 ft to 135, 30 ft to the top. That's a minimum of 14 placements. If, say, at 75 ft we lead out 20 ft instead of 17, we could unzipper all the pieces and hit the ground.

In contrast, if 3500 lb anchors are set, then we could (may not want to) set only 5 pieces at 10, 20, 40, and 80. (Oops, don't forget about the stretch in the rope.)

The effects of friction.

Any climber will tell you that the above analysis is nonsense because we did not include the effects of friction. Ok, let's redo the analysis and consider friction. Testing has shown that the friction on a rope that bends 180 degrees over a carabiner will reduce the load that the belayer feels by 52 percent (Soles, 1995). This friction can reduce the overall anchor load because the force in the belay side will not be as high. (see Figure 10) This effect is offset somewhat by the reduction in stretch of the rope as the climber falls.

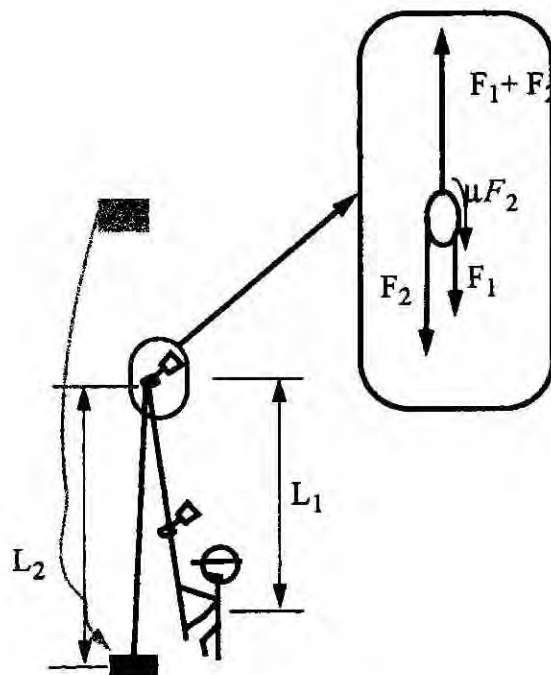


Figure 10 Friction effects on a leader fall can reduce the effectiveness of the pulley effect at the anchor.

The friction force can be expressed as a fraction, μ , of the impact force. The impact force will be balanced by the

friction force and the belay force:

$$F_2 = \mu F_2 + F_1 \quad \text{Eq(17)}$$

or

$$F_1 = (1 - \mu)F_2 \quad \text{Eq(18)}$$

The total displacement of the load will be the sum of the displacements in the belay and climber side of the rope.

$$\delta = \delta_1 + \delta_2 \quad \text{Eq(19)}$$

where

$$\delta_1 = \frac{F_1 L_1}{M} \text{ and } \delta_2 = \frac{F_2 L_2}{M} \quad \text{Eq(20)}$$

Substitution of δ_1 and δ_2 into Eq(18) gives:

$$\delta_1 = (1 - \mu) \frac{L_1}{L_2} \delta_2 \quad \text{Eq(21)}$$

The energy balance equation can still be used to compute the maximum displacement during a fall; however, we must include the work done by the friction force. The energy balance equation becomes:

$$PE = SE + W_f \quad \text{Eq(22)}$$

where the work done by friction is equal to the frictional force integrated over the distance that the rope moves through the carabiner.

$$W_f = \int_0^{\delta_1} \mu F_2 dx \quad \text{Eq(23)}$$

Using Eq(18) and Eq(20) we get,

$$W_f = \int_0^{\delta_1} \frac{\mu}{(1 - \mu) L_1} M x dx \quad \text{Eq(24)}$$

or

$$W_f = \frac{M}{2 L_1} \left(\frac{u}{(1 - u)} \right) \delta_1^2 \quad \text{Eq(25)}$$

Substituting Eq(21) into Eq(25) allows us to express the work done in terms of δ_2 :

$$W_f = \frac{M}{2 L_2} \mu (1 - \mu) \frac{L_1}{L_2} \delta_2^2 \quad \text{Eq(26)}$$

When $\mu = 0$ the work done by friction will be zero because the fractional force will be zero. When $\mu = 1$ the work done will also be zero because there will be no displacement through the carabiner.

The potential energy will be:

$$PE = mgh + mg(\delta_1 + \delta_2) \quad \text{Eq(27)}$$

or expressed in terms of δ_2

$$PE = mgh + mg \left((1 - \mu) \frac{L_1}{L_2} + 1 \right) \delta_2 \quad \text{Eq(28)}$$

The strain energy in the rope will be the some of the strain energy in the L_1 and L_2 sections:

$$SE = \frac{M}{2} \left(\frac{\delta_1^2}{L_1} + \frac{\delta_2^2}{L_2} \right) \quad \text{Eq(29)}$$

or

$$SE = \frac{M}{2 L_2} \left((1 - \mu)^2 \frac{L_2}{L_1} + 1 \right) \delta_2^2 \quad \text{Eq(30)}$$

Substitution of Eq(26), Eq(28), and Eq(30) into Eq(22) gives:

$$\begin{aligned} mgh + mg \left(1 - \mu \frac{L_1}{L_2} + 1 \right) \delta_2 \\ - \frac{M}{2 L_2} \left(1 - \mu \frac{L_2}{L_1} + 1 \right) \delta_2^2 = 0 \end{aligned} \quad \text{Eq(31)}$$

a quadratic equation in terms of the displacement of the length of rope above the last point of protection, L_2 . After solving for the displacement, we can get the total force on the anchor as:

$$F_{\text{anchor}} = (2 - \mu) \frac{M}{L_2} \delta_2 \quad \text{Eq(32)}$$

Figure 11 shows the solution to Eq(31) and Eq(32) for $W = 176$ lb, $M = 7000$ lb and $u = 0.5$. If $\mu = 0$, then we get the same solution as shown in Figure 9.

The effects of friction are greatest when the lead out is long compared with the amount of rope out. To see this compare Figure 9 with Figure 11. For this case, high frictional forces at the carabiner reduces the mechanical advantage at the anchor.

Friction does not significantly change the 'safe' lead out for a 2000 anchor. For cases where the belay line is long compared to the lead out, the effects of friction at the anchor are offset by a stiffer overall response. To understand this, consider the case where the friction is perfect, $\mu = 1$, and we have no movement of the rope through the carabiner. This case would lead to a fall factor of 2 impact load. The good news is that the belayer will not feel as much force when the climber falls. The bad news is that the anchor will still feel about the same pullout force. Here we have considered the effects of friction on only one anchor. I am not sure

how friction from a zig-zag rope system would effect the anchor loads.

Eq(33)

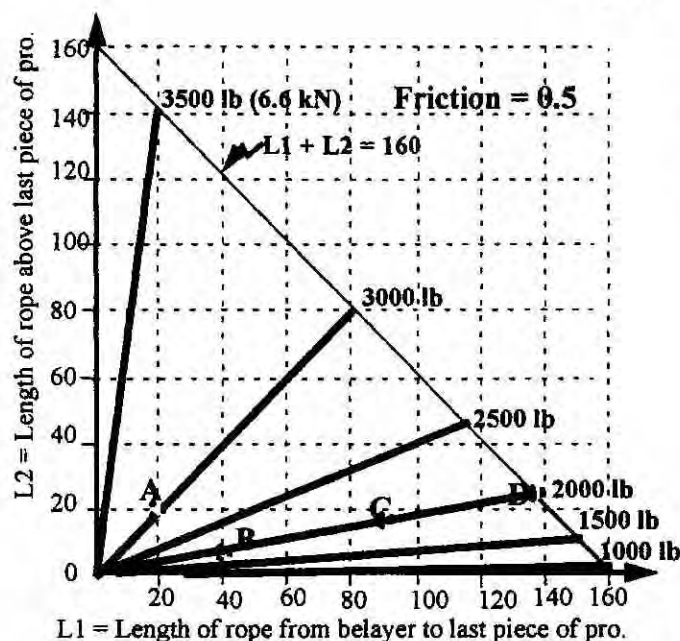


Figure 11. Plot of anchor load for $W=176.0$ lb, $M=7000$ lb, friction = 0.5, for different combinations of lead out above last pro and distance to pro.

Modulus as a function of impact force.

Now that we have completed the analysis of a fall with friction, we can use it to model the UIAA test. This will allow us to compute the dynamic modulus of the rope from the test data generated during the manufacturer's qualification test. So, you ask, what's a dynamic modulus? Well..

Toomey, 1988, showed that a dynamic loading of nylon rope used for ocean towing can behave differently under dynamic and static conditions. Toomey showed that the dynamic modulus (the local secant modulus or the apparent modulus) can exceed the quasi-static stiffness by a factor of 3 or 4 depending on the rope construction. Figure 13 shows typical dynamic force deflection curves for marine rope.

In order to convert the impact forces that manufacturers supply with their ropes to an equivalent dynamic modulus, Eq(20) and Eq(31) will be applied to the UIAA standard test in such a way that the effective dynamic modulus is solved for as an unknown. In the UIAA drop test, a weight, $W = 80$ kG, is dropped $h=4.6$ meters with $L_1 = 0.3$ m and $L_2 = 2.5$ m. Figure 12 shows a plot of the solution of M in terms of the impact force that a climber feels. A dynamic rope that has a rated impact load of 10 kN would correspond to a rope modulus of 30.0 kN or 6744 lb. Here, we assumed a friction factor of $\mu = 0.5$; however, because the length of the L_1 side of the rope is so short, friction makes less than a 5% difference in the impact force in the rope.

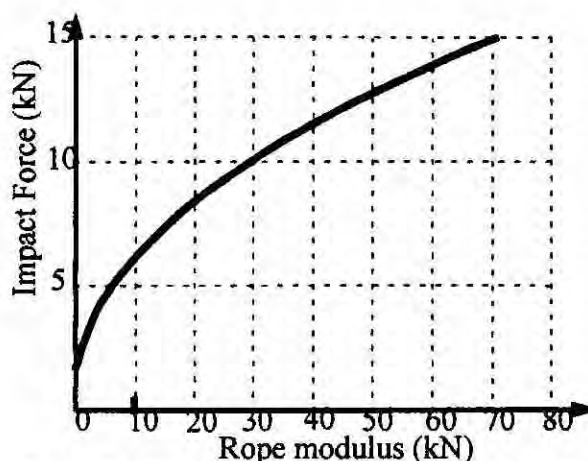


Figure 12. Impact force as a function of Rope modulus for UIAA test.

In the above calculations, we assumed that the rope modulus was constant, independent of stretch. In fact, the rope modulus is a function of rope stretch, which is evident from the lack of a straight line relationship between load and stretch in Figure 13. Assuming a linear relation for calculating maximum load should still be a good estimate of the strain energy in the rope, provided that the effective dynamic modulus is computed from the UIAA dynamic rope test data. I could not find any manufacturing data for dynamic climbing ropes that measures the rope modulus as a function of stretch.

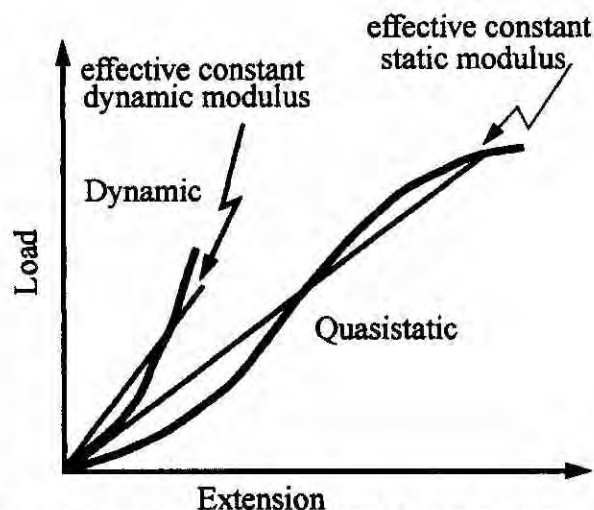


Figure 13 The difference between dynamic and quasi-static moduli of marine nylon rope. (Toomey, 1988)

Ok, so now we can compute the effective dynamic modulus. However, everyone knows that an old rope does not feel and work like a new rope. Could the rope modulus can also change with rope use? When drop tests are performed on a rope, the measured modulus increases with each drop. The

UIAA test requires that the rope survive 5 drops. The impact force is determined from the first drop. It is not unusual for the impact force to increase by 30 to 60% after four drops. If the rope generated 10 kN of force on its first drop, and increased to 15 kN by its fourth drop, this would mean that the modulus must increase by almost a factor of 2.4! It's no wonder that the rope fails after a set number of falls. Basically, the rope becomes stiffer and stiffer with each fall to the point that you might as well be using a static rope.

But wait, that's not all! In addition to increasing with each impact load, the modulus can also increase from rappelling, jumaring, and lowering. Anything that subjects the rope to forces that can straighten the rope fibers can also increase the rope modulus.

Ropes used in climbing gyms are often subjected to many small falls, each of which tends to increase the rope modulus. C. Soles, 1995, performed tests on ropes used in a climbing gym for 2 weeks. Two of the ropes he tested broke on the first fall. Not good news.

As a rope is strained, the mechanical conditioning through the structural realignment and deformation of the fibers contribute to an increased stiffness of the rope. The outer sheath on a rope acts like a "Chinese finger trap" as it is tightened under load. By providing a constraining force, the outer sheath generates internal friction that must be exceeded to elongate the twisted core fibers. This frictional work will not be stored as strain energy and will be converted to heat as the rope is stretched. As a rope is cycled under load, a hysteresis effect will occur as the rope loads under one path and unloads under another. Toomey, 1988, measured the hysteresis for 1/2 inch Samson Ocean Towing rope, Figure 14.

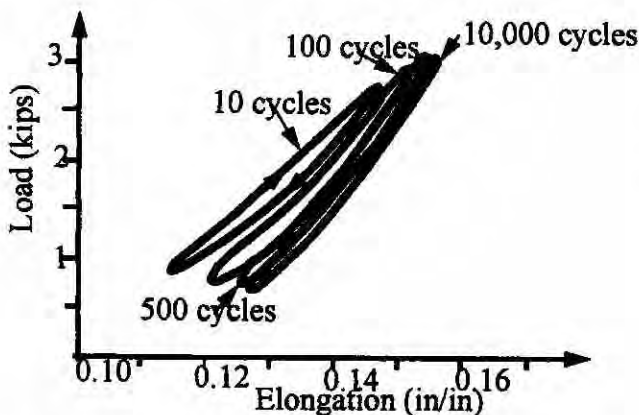


Figure 14. Hysteresis loops for nylon, dry, cycled at steady tension of 1.8 kips, frequency of 0.2 Hz and a strain amplitude of 0.017 in/in at the 10th, 100th, and 500th and 10,000th cycle. (from Toomey, 1988)

Two important observations can be made from Toomey's hysteresis measurement: 1). the dynamic modulus increased with load cycles; 2). the amount of hysteresis decreased with load cycles. **Does climbing rope experience the same sort**

of behavior? Can the rope be mechanically reconditioned to remove the effects of cyclic fatigue?

Figure 15 shows typical results for the change in dynamic modulus for a rope as a function of cyclic loading. The good news is that the dynamic modulus (for this type of rope construction, anyway) approaches a constant as it is cycled. **How does the dynamic modulus for different dynamic climbing ropes behave under cyclic loading?**

Based on the behavior of ocean towing ropes, we would conclude that an old rope might be a tired rope. One sport climber that I talked to said that after each fall he "worked" the rope to recondition it. I do not know if a rope can be "reconditioned". Until I find out, I will have two ropes. The one in mint condition, that I will use for big wall climbs. After a fall on any rope, it will be used as an "old" rope for work in the rock gym, top-roping or for rappelling.

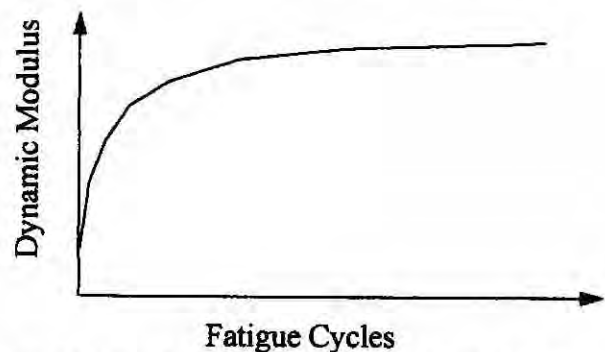


Figure 15. Typical change in dynamic modulus with fatigue cycles. [Pervorsek and Kwon, 1976]

Belay Devices: The good news!

In the above calculations for a leader fall, we have assumed that the belayer does not let the rope slip through his hands. Table 4 shows the typical breaking force of some belay devices.

Table 4: Breaking Forces for belay devices^a

Type	Breaking Force (kN)
Figure 8	1.5
Stitch plates	2.0
ATCs	2.0
Munter Hitch	3.0
Grigri	9.0

a. Clyde Soles, Rock and Ice Magazine, Vol. 117, No. 68

Clearly, the dynamic breaking provided by a slipping belay device will limit the impact load. Caution should be used when using any of these belay devices, because the rapid slippage of the rope can burn the belayers hands. Also, anyone who has gone just a little too fast on rappell can attest to the fact that these belay devices require strict attention to prevent mishap.

The force limiting nature of dynamic belay devices clearly has ramifications for rescue belays. Two methods of rescue belay are common. One uses a set of double prusik knots to belay the load. The other method is more time consuming and consists of pulling the rope up through a belay device. Since static ropes are used for hauling systems, the use of a dynamic belay device could greatly reduce the impact loads should the system be shock loaded.

Deflection of a rope on a traverse.

Traverses often are used on climbing routes and are frequently used during rescue operations for safety belays. The impact loads that result when a fall occurs on a traverse can be quite high. The rope loads depends on the geometry of the traverse. Consider a climber crossing a traverse with only a locking carabiner clipped into the traverse. If the fall occurs at either end of the traverse, then the climber would slide toward the center of the traverse. As the climber slides toward the center of the traverse, friction will dissipate some of the energy. The worst case would be to assume that the climber is located such that when he falls, he will not slide. For a traverse with equal height anchors, this 'horizontal equilibrium' (i.e. no sliding) will be at the center of the rope. If the anchors are at different heights, then the 'horizontal equilibrium' will not be at the center, but at the point where the angles θ_1 and θ_2 shown in Figure 16 are equal.

As an example, we will look at the traverse system as shown in Figure 16. The bolts on the left were set 10 ft. above the bolts on the right, giving the traverse about a slope.

Before we can compute the dynamic response of this

system, we must first compute the static deflection. (Remember that you can also measure the static deflection in the field.)

The equivalent stiffness of the traverse rope is computed by combining the stiffness of the rope to the left and the right of the weight. The actual formula for computing the equivalent stiffness is based on computing the stiffness along the length of the rope, then rotating the stiffness to the correct geometry. If you really want to see how this done see Martin[10].

The equivalent stiffness is

$$K_e = K_1(\sin \theta_1)^2 + K_2(\sin \theta_2)^2 \quad \text{Eq(34)}$$

where $K_1 = \frac{M}{L_1}$ and $K_2 = \frac{M}{L_2}$ are the stiffness of the lengths of rope. θ_1 and θ_2 are the angles between the ropes.

The deflection given by the above formula would be:

$$\delta_{st} = \frac{W}{K_1(\sin \theta_1)^2 + K_2(\sin \theta_2)^2} \quad \text{Eq(35)}$$

Eq(34) and Eq(35) are only valid for cases where the angles do not change much due to the deflection. If the rope is straight to start with, then the angles will be zero, giving an infinite deflection. What really happens is that the rope deflects, and the resulting deflections make the problem geometrically non-linear. This does not mean that the problem cannot be solved, it just means that the math becomes too hard to do by hand, and a computer is recommended. There are several commercial computer programs that can compute this sort of non-linear deflection. As it turns out, you do not want to rig a traverse so taut that it has a near zero angle. If you have a near zero angle, then the loads on the anchor will be very high.

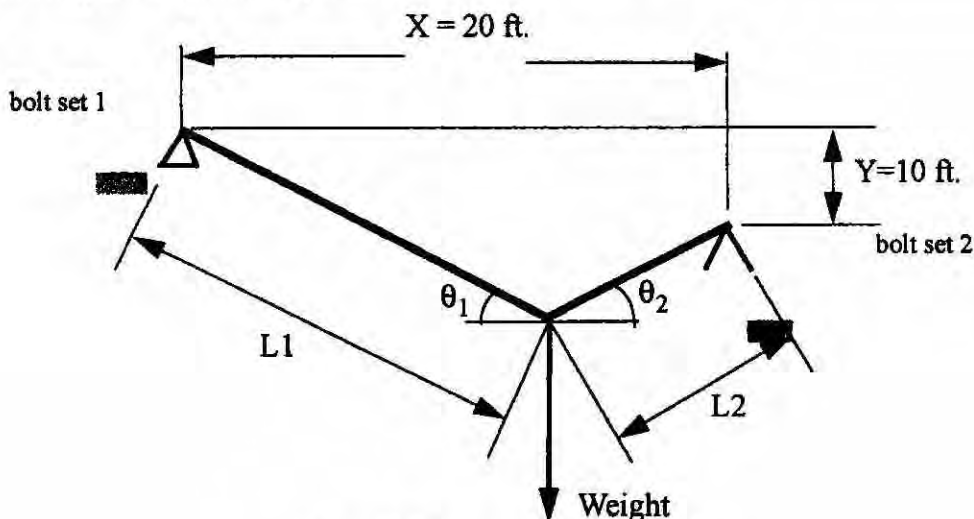


Figure 16 The traverse loaded by a weight.

Table 5: Loads on a traverse.

L (ft.)	h (ft.)	L1 (ft.)	L2 (ft.)	angle (deg)	F(lbs)
20	3	19.5	0.5	31.8	3635
21	3	19.0	1.98	35.95	1985
22	3	18.9	3.12	39.4	1645
23	3	18.9	4.07	42.34	1479
23	5	18.9	4.07	42.34	1862

Table 2 shows the dynamic impact forces computed for different lengths of PMI rope ($M=20000$ lbs/ft./ft.). The computations assume a 200 lb weight (160 lb with 40 lb pack). Notice how much difference 2.0 ft. of rope length make in the impact load factor!

Summary

We have presented equations for computing the dynamic impact load factor for typical rope systems used for rock climbing. The equations are based on the height of the fall, the deflection of the rope, and friction. Example calculations showed that it is easy to exceed the recommended maximum loads that are typical of cam type anchor devices. The calculations show that falls from any combination of lead out and belayed rope length usually will not exceed 4000 lbs of anchor force. However, falling from a leading out of more than 1/4 the belayed rope length could generate more than 2000 lbs of anchor force, the approximate force need to pull-out some types of climbing protection.

Recommendations:

- Build your anchors to withstand 25 kN (5500 lb) when possible.
- If you are going to use wire chocks and cams that have a typical strength of 2000 lbs, then don't lead out more than 1/4 the belayed rope length.
- Never exceed your climbing abilities on a big wall climb. Test your skills at a rock gym, on bolted routes, or under top-roped conditions.
- Use only 'new' ropes for lead climbing. To protect against shock loading of the anchors, use a rope with a low modulus or impact force rating.
- Use a dynamic belay device.

Future Work

In order to design safer climbing anchors it is necessary to understand the forces generated from climbing ropes. While the above methods present a first order prediction of the forces involved in a climbing rope, some measurements that would be useful in better predicting forces on anchors are:

- Static force-deflection curve.

- Dynamic force-deflection curve.
- Dynamic force-deflection curve for ropes subjected to different cyclic loadings such as repeated rappelling, climbing, or short falls.
- Measure the dynamic modulus and hystereses of ropes after they are subjected to repeated falls.
- Determine the strain-rate dependence of climbing rope. (Dynamic force-deflection at different strain rates.)
- Perform dynamic tests that simulate a leader fall and measure the force vs. time at the anchor and the belayer.

Dedication

This work is dedicated to Dr. Carlos Abad, Ms. Jane Teessen, and Dr. Glen Tietjen who died from a fall of 817 feet on June 23, 1996.

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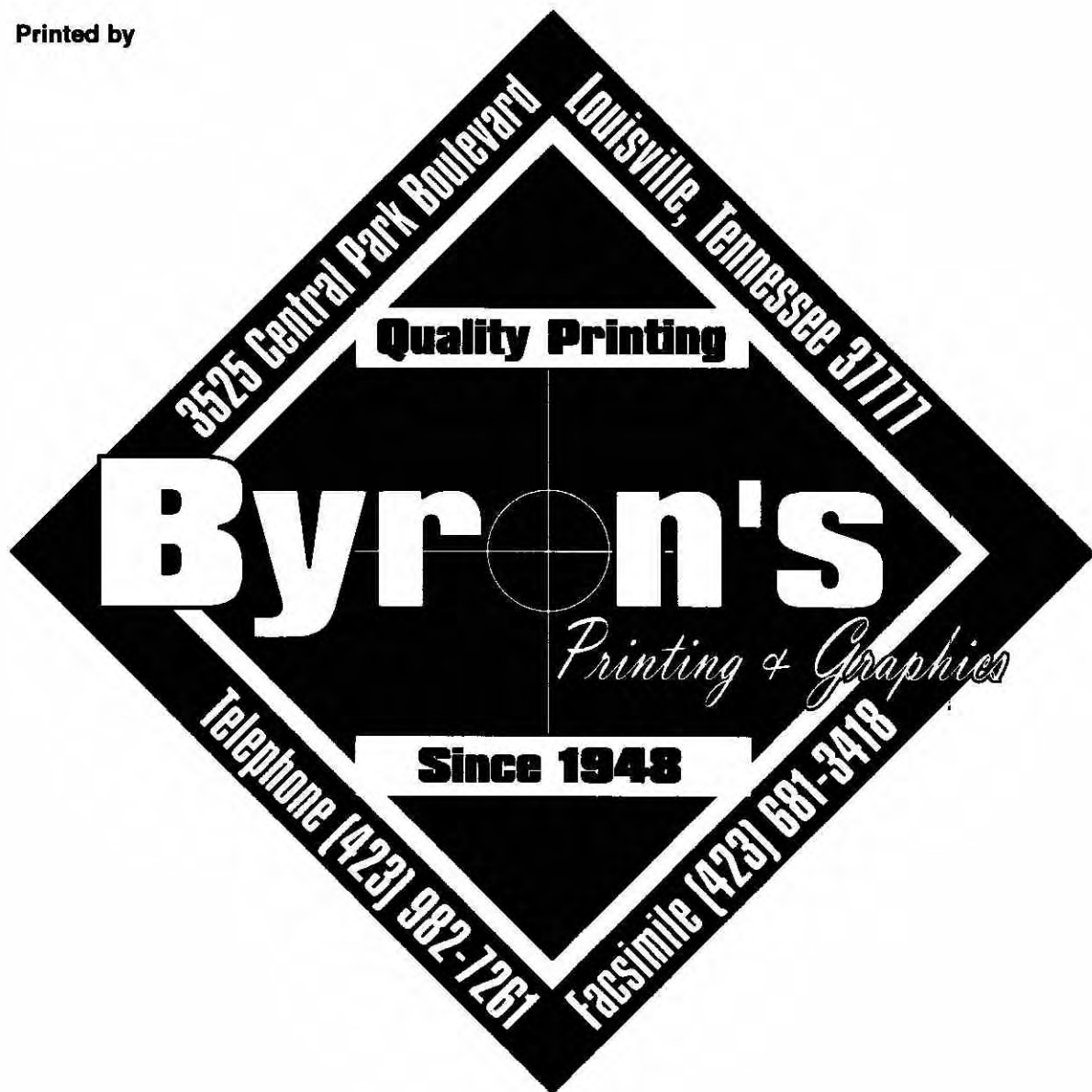


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