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Withdrawal and Combined Load Capacity of Threaded Fastener Wood Joints

by

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(ABSTRACT)

In this study, general models of the capacity of threaded fastener joints were developed from extensive experimental tests. One study objective was to develop a general model of threaded fastener withdrawal strength applicable to joints containing fasteners with widely varying thread geometries. A total of 419 tests of joints using six different fasteners and five species were tested. A multiplicative model containing wood specific gravity and the wood volume contained within the fastener threads provided very accurate predictions of withdrawal strength.

A second study objective was to assess the accuracy of existing design criteria for threaded fastener joints subject to combined axial withdrawal and lateral shear loading. A total of 321 joints using 3 different fasteners and two species were tested at five angles between 0^{o} and 90^{o} to fastener axis. Little to no interaction

was found between the lateral and withdrawal force components on joint capacity. Current design philosophy in the National Design Specification generally yielded conservative predictions of actual joint capacity. Improved design criteria suggested by experimental results were derived.

A pilot study was also conducted to assess the influence of wood desorption on the withdrawal resistance of tapping screws. Forty joints of two moisture conditions, green and dry at insertion and two species were tested in withdrawal. In general, maximum and proportional limit loads were not affected by desorption whereas stiffness was significantly reduced for joints which desorbed after insertion.

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1.0 Introduction

The strength and stability of any structure is dependent upon the adequacy of the connections between the individual structural components. One major advantage of wood as a structural material is the ease with which wood members can be joined by a wide variety of mechanical fasteners. The strength of a wood joint, however, is dependent on many variables such as wood strength and stiffness, type of loading, environmental influences, and fastener characteristics.

Threaded fasteners such as wood screws, lag screws, screw tie spikes, and spiral dowels are commonly used in structural connections to transmit either shear or withdrawal loads or a combination thereof. Smaller threaded fasteners, such as wood and drive screws, are subject to withdrawal and combined loading in furniture and mobile home applications as well as in metal-clad roof diaphragms used in agricultural construction. Lag screws, spiral dowels and other large fasteners are subject to similar loading in glulam and heavy timber construction. Screw tie spikes are also employed in withdrawal and combined loading situations in the fastening of tie plates and other equipment to wood ties. Threaded fasteners are of notable importance in withdrawal loading where they have an advantage over non-threaded fasteners such as bolts and nails (36). Threaded fasteners can also be removed and reinserted without significant loss of holding power (9) and, unlike nails, can be used in the fastening of brittle materials.

Design criteria for connections with threaded fasteners are based on a limited number of empirical tests using only shear or withdrawal loads. The results of these tests were cast into simplified expressions to predict joint strength. Reduction factors are then applied to these predictions to account for safety, environmental conditions, and duration of load.

Although several studies of threaded fastener joint strength have been conducted, few subsequent efforts have been made to interpret and correlate the individual results. The independence of previous test programs, the lack of consistent methodology, and extrapolation of test data have resulted in some anomilies. For example, lateral design values for large diameter wood screws are two to four times greater than those for lag screws of equal diameter. Wood screw lateral resistance is independent of whether the load is directed parallel or perpendicular to the grain whereas lag screw design values are significantly different in the two orientations. Also, a design value increase for metal side plates is available for wood screw joints but not lag screw joints in the perpendicular to grain orientation. These discrepancies are primarily related to independent test programs and failure to consider all variables which influence joint performance.

Thread geometry is among the more important variables ignored in previous studies. Past research has tended to employ fasteners of similar geometries in order to evaluate length and diameter effects.

The effect of wood desorption on joint strength has also not been studied. However, drying effects have been found to substantially influence the withdrawal resistance of threaded nails (33).

Previous threaded fastener research has also failed to consider combined loading effects. To derive a design equation for joints subject to combined loading it is necessary to determine if an interaction exists between the lateral and withdrawal force components. If an interaction does not exist, i.e. a lateral force does not influence withdrawal strength (or vice versa), then the resultant force can be split into its individual components. The individual components can then be compared with predictions generated from models for pure withdrawal or lateral loading. If an interaction does exist, a separate combined loading model must be derived based on the variation in joint strength with respect to the angle between the force and the fastener axis. Significant interactions of this type have been observed in the combined loading of nailed joints (10).

The wood design community is currently moving towards a reliability-based design (RBD) format similar to that adopted by the steel and concrete industries. The limited, fragmented, and at times contradictory, data base concerning fasteners exists as a major obstacle to the application of RBD to wood. This research proposes to establish models of threaded fastener performance which will contribute to the development of rational, uniform, design procedures for mechanically fastened wood joints.

2.0 Objectives

The overall objective of this research is to provide a rational, consistent methodology for establishing design capacity of threaded fastenings in wood subject to a) withdrawal loads or b) combined withdrawal and lateral loads. To accomplish this the following subobjectives are proposed.

a) Establish and verify a model of the action of a threaded fastener in withdrawal to predict connection strength. This model will consider fastener geometry such as thread characteristics in addition to wood related parameters.

b) To identify and characterize any interaction between lateral and withdrawal forces on the strength of joints subject to loads in both directions.

c) To propose an interaction relationship to account for these effects, if present, that would be suitable for design purposes.

d) To evaluate any effects of desorption on the withdrawal strength of selected threaded fasteners.

3.0 Model of Withdrawal Strength

A Mechanistic Approach to the Prediction of Threaded Fastener Withdrawal Strength

3.1 Introduction

Dowel type connectors such as nails and screws are often subject to forces acting parallel to the connector axis. These forces tend to withdraw the shank of the connector from the substrate. Resistance to withdrawal of smooth shank connectors is a function of friction and a complex interaction of the time dependent properties of wood and joint geometry. For threaded fasteners, an additional complexity involving thread geometry is introduced.

Several researchers have developed predictions of the withdrawal resistance of common threaded fasteners from wood. These studies were generally specific to a certain fastener type (i.e. thread geometry) such as wood screws, lag screws, spiral dowels, etc. As a result, a variety of empirical withdrawal strength models have appeared in the literature, each associated with an individual fastener design. A number of these formulas are illustrated in Table 1. Current design practice for axially loaded wood and lag screws stems from such studies conducted by Fairchild (13) and Newlin and Gahagan (19). Withdrawal strength for non-standard threaded fasteners, such as the numerous varieties employed in the railroad and construction industries, is based upon manufacturer's tests or fixed adjustments to predictions from an equation for a standard fastener (29).

Use of a separate empirical equation for allowable strength of lag and wood screws of comparable diameter results in 230-280% difference in capacity between the two types. Additionally, code writers are at a loss to provide guidance for capacity of screws when industrial thread standards are changed or when construction practice shifts (i.e. substitution of tapping screws for nails or wood screws). In the determination of current withdrawal strength models, little regard has been directed toward the mechanism of failure in fastener withdrawal or the influence of fastener geometry on this mechanism. The purpose of this study is to develop a model of threaded fastener withdrawal which considers both wood and fastener thread geometry variables. This model, coupled with some empirical data, can be used to set consistent withdrawal capacities for threaded fasteners of varying geometries.

Failure of an axially loaded threaded fastener joint can result from one of three mechanisms; 1) yielding of the fastener in tension, 2) failure of the wood component under the fastener head or washer-plate (head pull-through), or 3) withdrawal of the fastener shank as a result of wood failure near the thread-wood interface. Fastener yield load is easily obtained from a table of steel properties and fastener root diameters. Alternatively, one may rely on minimum quality specifications such as ASTM A307 (2). Usually, only a combination of small fastener diameter and dense wood species will produce this failure mode. Head pull-through is common with less dense species and thin materials. This can be prevented by appropriately sized washer-plates or other means of increasing bearing capacity.

In shank withdrawal, wood failure results when the stress on the wood positioned between the threads or at the thread-wood interface exceeds a critical value. This abrupt, generally catastrophic wood failure appears to be dominated by cleavage parallel to grain, shear perpendicular to grain, and to a lesser degree, compression perpendicular to grain. If the governing mechanism of wood failure is reasonably constant over a wide range of thread geometries, it may be possible to predict shank withdrawal strength using a quasi-mechanistic, rather than a purely statistical model. This will allow for derivation of a single withdrawal strength prediction equation applicable to a wide range of fastener thread geometries. Use of this equation can unify design specifications for various fasteners and eliminate existing inconstencies.

3.2 Withdrawal Mechanism

One simplification of the withdrawal mechanism is to consider that the maximum load is a linear function of the area over which the withdrawal force is transferred from the fastener. The design of bolted metal connections employs this approach where joint failure is expected to occur by shearing of the nut threads. The average shear stress in the thread of a nut is calculated by:

$$\sigma_s = \frac{2\ P_w}{\pi\ D_m\ H} \qquad [1]$$

where: σ_s = average shear stress in a single nut thread, psi,

P_w = axial bolt load, lbs.,

D_m = major diameter of the bolt, in.,

H = nut thickness, in.

A maximum axial force, P_w, can be calculated by rearranging and setting σ_s equal to a critical shear stress. If this approach is applied to an axially loaded fastener in wood, maximum withdrawal load, P_w in lbs., may be calculated by:

$$P_w = \sigma_v\ A \qquad [2]$$

where σl_v is a critical shear strength of wood perpendicular to grain and A is the area over which P_w is transferred. Area, A, can be calculated by rearranging [2] and substituting:

$$A = P_w/\sigma_s = (\pi/2)\ D_m\ L_e$$

where D_m becomes the outer thread diameter of the fastener and L_e is the effective length of engagement of the fastener thread. The effective length of engagement is equal to the depth of penetration minus some small adjustment to account for the fastener tip. For thread geometries common to fasteners manufactured for wood applications, the following definition of shear area may be more suitable:

$$A_s = \pi D_m L_e \qquad [3]$$

The term $\pi/2$ is used in metal connections since shear stress is distributed over one-half the inner nut surface. In wood joints, shear stress is more likely to be distributed over the complete wood-thread interface. Thus, the shear area is equal to the surface area of a cylinder as defined in Figure 1. For the purpose of this study, this failure consideration will be called the <u>shear area mechanism</u>.

Alternatively, fastener withdrawal may be considered as a bearing or compression failure of the wood component located between the threads. The average bearing stress in a single nut thread may be calculated as:

$$\sigma_b = \frac{P_w}{A_b} = \frac{P_w}{(\pi/4)\ (D_m^2 - D_r^2)\ (L_e/p)} \qquad [4]$$

where: σ_b = average bearing stress in a single nut thread, psi,
P_w = axial bolt load, lbs.,
A_b = area over which P_w is transferred, sq. in.,
D_m = fastener major diameter, in.,

D_r = fastener root diameter, in.,
L_e = effective length of thread engagement, in.,
p = thread pitch, in.

Thread pitch, p, is equal to the reciprocal of the number of threads per inch. The bearing area is defined as the horizontal projection of the upper surface of the thread over the entire effective length. See Figure 1. The critical bearing stress may be related to the compression strength perpendicular-to-grain. Fastener bearing area is defined by rearranging [4]:

$$A_b = P_w/\sigma_b = (\pi/4)\ TPI\ (D_m^2 - D_r^2)\ L_e \qquad [5]$$

where the number of threads per inch, TPI, is substituted for 1/p. In this study, this approach is identified as the <u>bearing area mechanism</u>.

A third alternative considers that failure of the wood component results from a combination of stresses and depends on the total volume of wood constrained within the fastener threads. There are several mechanical analogies that can be developed by considering the ring of wood between the threads that resists fastener movement. One possibility is idealization of a clamped circular plate with a hole subject to a uniformly distributed load. This plate has varying thickness corresponding to the pitch at the fixed end and either zero or a "flat" distance next to the thread shank or free end. The maximum stress or deflection of this plate is a function of bearing area, A_b, plate thickness, and the distance from the center of the fastener to the centroid of the torus formed by the wood ring. The resulting equations from plate theory are quite complex and may not apply to failure stress. However, pragmatically the wood thread volume is a product of these three components and hence is a useful simplified substitution:

$$VOL = \pi\ (f + p)\ [k\ R_m^2 + (k-1)\ R_r^2 + R_m\ R_r\ (1 - 2k)]\ TPI\ L_e \qquad [6]$$

where:

$$k = \frac{f + 2p}{3(f + p)}$$

and; VOL = volume of wood contained within the engaged fastener threads, cu. in.,
f = flat = distance along the fastener root between the base of adjacent threads, in.,
p = thread pitch, in.,
R_m = major radius, in.,
R_r = root radius, in.,
TPI = number of threads per inch,
L_e = effective length of thread engagement, in.

A maximum withdrawal load corresponds to the force required to create a maximum plate surface stress, cause a critical bending deflection, or some combination of effects. A clear mechanical analogy of volumetric stress similar to σ_b or σ_s could be developed but is not warranted.

For the purpose of this study it is assumed that the critical material property (σ_s, σ_b, or σ_{vol}) is an exponential function of the specific gravity of the wood material. This is consistent with previous work (see Table 1) and provides an easy transition of research results to design guidance.

3.3 Methods and Materials

As part of a companion study, a survey was made of 12 manufacturers of threaded fasteners for the construction and railroad maintenance industries. Based on this survey, six fasteners were selected to represent the range of available thread geometries. Fastener lengths were selected based on manufacturers' recommendations of most popular sizes. The fasteners were: 3/8" x 6" hex head lag screws, 5/8" x 6" hex head lag screws, a tapping screw similar to a No. 10 x 2" hex head type AB, and three proprietary fasteners manufactured for railroad applications. These are illustrated in Figure 2 with thread characteristics tabulated in Table 2.

Wood species were selected by specific gravity rather than shear or compression strength because specific gravity is easily determined and is strongly correlated with most wood strength properties. Commercially important species representing the full range of both hardwood and softwood groups were included. These were West coast spruce-pine-fir (S-P-F), yellow poplar, Southern yellow pine, red oak, and hickory.

Test specimens were constructed by laminating lumber segments into blocks of sufficient depth to accomodate a pre-selected length of fastener penetration. Each lamina in every block was planed before gluing to eliminate the possibility of gaps between lamina and to insure that the fastener would be withdrawn from a defect-free area in the orthotropic radial direction. Block widths and lengths were selected to discourage splitting of the specimen during fastener withdrawal. For the proprietary fasteners the blocks were 4 to 5.5 inches in width and 8 to 12 inches long. Specimens for the 5/8" lag screws were 3.5 to 5.5 inches wide and 8 to 10 inches in length. Specimens for the 3/8" lag screws measured from 2.5 to 3.5 inches in width and 8 to 10 inches in length. All specimens used to test the tapping screws measured 2" x 7". The wider and longer dimensions were used with wood species prone to splitting such as S-P-F and Southern pine. Smaller sizes were sufficient for poplar and oak which did not split as readily. Each block was used to test one fastener except for those receiving the 3/8" screw where up to three could be accomodated without interference. Twelve to 22 replications were tested for each experimental cell with a total sample size of 419 observations.

Lead hole size influences the withdrawal strength of threaded fasteners. The Wood Handbook (27), the National Design Specification (18), and Newlin and Gahagan (19) provided information aiding lead hole selection. Lead holes ranged from approximately 50% of fastener root diameter in S-P-F and poplar to just over 90% root diameter in hickory. Table 3 presents the lead hole sizes used with the various fasteners and wood species. All lead holes were drilled to a depth exceeding that of fastener penetration.

All fasteners except Fastener C and the tapping screws and were driven into test blocks using a pneumatic wrench. The spiral dowel-type thread on Fastener C was unreceptive to impact driving and these were hydraulically pressed into sample blocks. Tapping screws were driven using a 1/4" socket fitted to a variable speed electric drill. Wax lubrication was applied to the 3/8" lag screws, 5/8" lag screws, and railroad Fasteners A and B for insertion into the oak and hickory blocks. Withdrawal testing was conducted on a 20,000 lb. MTS hydraulic test machine at the ASTM D1761 (2) specified rate of stroke control of 0.10 in./min. Two LVDT's on either side of the fastener measured the separation of the fastener head from the sample block. Output from the two transducers was averaged to compensate for any misalignment or asymmetrical movement. Force was sensed with a load cell and a load-deformation graph recorded for each test. The test set-up is illustrated in Figure 3.

The depth of penetration of the threaded portion varied somewhat by fastener and species to insure that shank withdrawal was the governing failure mechanism. Actual penetrations used are shown in Table 3. These values were set based on test machine capacity and preliminary tests which showed the fastener steel yield stress of the 3/8" lag screws to be approximately 90 ksi. Depth of penetration for these fasteners was set to insure that the maximum fastener stress would not exceed 80 ksi. In some cases different depths of penetration were used for a single fastener-species combination to test whether the strength per inch of thread penetration was constant. For example, with the 3/8" screws and Southern pine there were 12 replications at 3.0" depth, 8 with 2.57", and 2 with 1.857".

No significant differences were found between strength on a per inch basis. Failure of the tapping screw in tension occurred at a steel tensile stress of approximately 177 ksi. This failure was not ductile as seen with lag screws but brittle resulting in the separation of the head and shank of the fastener. To insure shank withdrawal, a one-inch depth of penetration was used with all five wood species. This depth was the maximum possible for hickory samples.

After testing, specific gravity (oven-dry volume basis) and moisture content were determined on samples cut from each block. Maximum withdrawal load (P_{max}) and load and deflection at the proportional limit (P_{pl}, d_{pl}) were determined from the load-deformation curves.

3.4 Results and Discussion

The results of the withdrawal strength study are presented in Tables 4 and 5. To allow for comparison, P_{max} and P_{pl} were divided by the effective length of thread engagement used in each test. Joint stiffness was also calculated on a per inch basis by dividing P_{pl} per of thread engagement by deflection at the proportional limit. Like the per inch strength parameters, d_{pl} was reasonably independent of length of engagement and was remarkably consistent between fastener.

Fastener	Mean d_{pl} (in.)	Coeff. Var. (%)
3/8" Lag Screw	.0368	21.2
5/8" Lag Screw	.0334	17.3
Tapping Screw	.0123	18.4
Fastener A	.0355	17.0
Fastener B	.0359	16.5
Fastener C	.0276	42.3

The coefficients of variation (C.O.V.) for d_{pl} are similar for all except Fastener C joints. The large C.O.V. for Fastener C data is probably not related to differing L_e since very few depths of penetration were used with this fastener as shown in Table 3. In addition, the C.O.V for the 5/8" lag and tapping screw data, where one L_e was used, is not much different from that of the 3/8" lag screw or Fastener A and B data where variable L_e was employed. With d_{pl} relatively constant and P_{pl} increasing with L_e, overall joint stiffness increases with length of engagement.

The average moisture content of the 419 withdrawal samples was 7.6% with a coefficient of variation of 14.7%. Moisture content was not considered a variable in this study since sufficient variability in this parameter was not observed in the experimental data.

Shear and bearing areas and wood volume were calculated using [3], [5], and [6], respectively, for each fastener type. L_e was set to unity so that all results were in terms of capacity per unit length of thread engagement. Regression analyses were performed using multiplicative models as discussed earlier. Logarithmic transformation of all dependent and independent variables was made prior to regression analysis using the Statistical Analysis System (23). The regression models had the general form:

$$\text{Capacity} = K\,SG^{n}\,(\text{Property})^{m}$$

where:

Capacity = P_{max} or P_{pl} per inch of thread engagement,
Property = A_s, A_b, Vol,
K, n, m = fitted regression parameters.

The term $K\,SG^{n}$ is used to represent a critical wood property in terms of specific gravity.

The fit of the P_{max}/inch models to the data was quite good as indicated by a range of R^2's from .877 to .908. The quality of fit is further supported by the coefficient of variation (CV) which is defined as the standard error of the estimate divided by the mean of the dependent variable. CV ranged from 2.06% to 2.38% for the study regressions. The fit of the volume relationship [6] to the experimental data is illustrated in Figure 4.

Similar regression analyses were conducted with P_{pl}/inch as the dependent variable. The bearing area mechanism yielded the best fit with an R^2 of .793. Shear area and volume models did not fit as well as demonstrated by coefficients of determination of .597 and .629, respectively.

From residual analysis of the regression results it was noted that all three models under-predicted P_{max}/inch for Fastener C. The mechanistic P_{pl}/inch models, on the other hand, produced consistent over-predictions for this fastener. Load-deformation curves for "C" are characterized by a short linear section below the proportional limit followed by a long curvilinear section between proportional limit and maximum load. Compared to the abrupt, well-defined failure seen with the other fasteners, those of "C" could be considered ductile. The percent of proportional limit to maximum loads were found to average 38% for Fastener C while this percentage was 67 to 77% for the other fasteners. This behavior is typical for the withdrawal of threaded nails and indicates that a different resistance mechanism is operative.

In any threaded fastener subject to axial withdrawal, friction acts to resist turning of the screw and its subsequent axial displacement. For any one revolution of the screw, axial displacement is a function of the pitch and lead angle. If the friction is great enough (or the head is fixed), the withdrawal failure is due to wood thread failure (i.e. shear, bearing, or volumetric effects). If the lead angle is large and friction is not great enough to resist turning, thread damage will be minimal and joint separation is almost entirely dependent on friction. The large lead angle and shallow thread depth of Fastener C is very conducive to this type of behavior. In general, the mean prediction errors for P_{max1}/inch of Fastener C (15-20%) using the regression models were much less than those for P_{pl}/inch. This suggests that the proportional limit is more sensitive to friction than maximum load.

Data for Fastener C were removed from the data base and the regression analyses were repeated. The following resulted:

$$P_{max}/\text{in.} = 4073\ SG^{1.53}\ A_s^{.66} \qquad R^2 = .920 \qquad SEE = 1.160 \qquad [7]$$

$$P_{pl}/\text{in.} = 2526\ SG^{1.33}\ A_s^{.68} \qquad R^2 = .877 \qquad SEE = 1.1938 \qquad [8]$$

$$P_{max}/\text{in.} = 7167\ SG^{1.48}\ A_b^{.57} \qquad R^2 = .912 \qquad SEE = 1.168 \qquad [9]$$

$$P_{pl}/\text{in.} = 4509\ SG^{1.28}\ A_b^{.58} \qquad R^2 = .855 \qquad SEE = 1.212 \qquad [10]$$

$$P_{max}/\text{in.} = 12{,}805\ SG^{1.51}\ VOL^{.33} \qquad R^2 = .923 \qquad SEE = 1.157 \qquad [11]$$

$$P_{pl}/\text{in.} = 8193\ SG^{1.30}\ VOL^{.34} \qquad R^2 = .877 \qquad SEE = 1.1937 \qquad [12]$$

These regression results indicate that all three models are comparable for the prediction of withdrawal capacity. However, based on the Standard Error of the Estimate (SEE), the wood volume model is best. Another principal reason for this choice is the ability of the volume parameter to discriminate with respect to effective thread geometry. For example, compare the results in Table 4 for the 5/8" x 6" lag screw and Fastener B. Both of these fasteners have the same thread geometry except for thread depth. If the shear area model is used, no difference in withdrawal capacity can be detected between the two fasteners. Yet Fastener B clearly has greater withdrawal capacity. The volume parameter picks up the difference in geometry. The mean square errors of the predictions of P_{max} and P_{pl} for Fastener B using the shear area model are twice as great as those found using the volume model. It is this thread geometry discrimination that adds power to the use of volume for identifying withdrawal capacity.

For both P_{max} and P_{pl} models, specific gravity was the most significant parameter, accounting for 60% of the total amount of explained variation. The remaining 40% was explained by the thread geometry parameters. Equations [7], [9], and [11] are similar in form to the Wood Handbook predicting equation for lag screws. Note that A_b varies approximately as major diameter and that A_s and VOL are also related to D_m. If shank diameter is substituted for A_b, the resulting model has about the same exponents as the Wood Handbook model and similar predictive capability as the A_s model. Eckelman's (7) analysis of Fairchild's (13) wood screw data also shows that withdrawal strength is related to specific gravity to the 1.5 power.

This suggests that the volume model could be extended to fasteners outside the scope of this study.

To explore this hypothesis, a data set of simulated ultimate strengths was created using the Wood Handbook empirical relations. For this purpose specific gravity was varied in even increments over the range of 0.35 to 0.65; wood screw gauge from No. 8 (0.164 in.) to 24 gauge (0.372 in.); and lag screw size from 0.190 in. to 1.00 in. An "FPL" ultimate withdrawal strength was predicted using the two equations shown in Table 1 and the input parameters described above. The VOLUME model was used to predict this ultimate strength using standardized thread geometry and equations [6] and [11]. Predicted vs. "FPL" are shown plotted in Figure 5. The line of equality lies below most points but parallel to the trend. That is, the volume model tended to over-predict the strength as estimated from the Wood Handbook (27). This is not unexpected since the Wood Handbook equation under-predicted the study lag screw withdrawal strength by 13-26%. Also, it is likely that the FPL formulas are conservative estimates of withdrawal strength. This was also found by Eckelman who analyzed the wood screw data on which the Table 1 equation is based. The quality of the fit in Figure 5 indicates that the volume (or perhaps other models) may be successfully used to develop one equation for the withdrawal of all threaded fasteners. This hypothesis will have to be tested using actual rather than simulated data.

Wilkinson and Laatsch (29) found that tapping screws yield a withdrawal strength 10-16% greater than that of similar sized wood screws. This discrepancy can be attributed to differences in thread geometry. Due to the substantial taper found in the threaded portion of wood screws, the effective length of engagement of wood screws is less than that of comparably sized tapping screws. This reduction in L_e results in lower withdrawal strength. The A_s, A_b and VOL models can likely supply accurate withdrawal strength predictions for both fasteners if L_e data is obtained.

A modification to the bearing area models [9,10] was also tried by adding a third variable consisting of the product of the cosines of the lead and helix angles. Lead angle is the angle between the thread orientation and the fastener shank. The helix angle is that between the surface of an individual thread and a projection perpendicular to the fastener axis. This variable was created to increase the accuracy of predictions for fasteners where lead and helix angles become large enough to influence joint behavior. This practice is common in design of power screws (24). The addition of this variable was significant but resulted in a very slight overall model improvement that could not justify increased model complexity. However, this factor could be considered by fastener designers looking to optimize withdrawal strength of a proprietory fastener.

3.5 Conclusion

The study results show that threaded fastener withdrawal strength per inch of effective thread enagagement can be predicted quite accurately through models employing fastener shear area, fastener bearing area, or wood volume contained within the fastener threads. The selection of these variables was based on the premise that shank withdrawal results from localized shear and compression failure in the wood component within or adjacent to the threads. Strength per inch was evaluated in this study to allow comparison of fasteners of distinctly different thread geometries. Joint strength and stiffness were linearly related to length of thread engagement but deflection at the proportional limit remained relatively constant for each fastener.

Models based on three thread geometry parameters were successfully applied to experimental data from testing of fasteners with differing thread geometry. Predictions from a single wood volume model were favorably compared to strengths estimated from several empirical equations in the Wood Handbook for both wood and lag screws. However, the accuracy of the models decreased when applied to fasteners having a combination of large lead angle and shallow thread, such as spiral dowels. This combination results in a withdrawal strength governed by friction and less dependent on the resistance of the wood to shear and compression stresses. For such fasteners, a separate strength model based on surface contact area should be developed. This approach has been successfully applied to predict withdrawal strength of threaded nails (22).

For thread designs typical to wood and lag screws, the shank withdrawal models are accurate and tend to confirm the suspected shear and compression related mechanisms governing fastener withdrawal. The models are not applicable to tensile failure of the fastener material or head pull-through. Proper lead hole size is required to insure full thread action and guarantee maximum joint strength.

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Table 1: Empirical equations for prediction of withdrawal strength of threaded fasteners from wood side grain.

Fastener	Equation	Reference
Wood Screws	$14{,}220\ SG^{2}\ D\ L$	Fairchild (13), Wood Handbook (27)
	$10{,}762.3\ D^{.976}\ L^{.886}\ SG^{1.499}$	Eckelman (7)
	$13.16\ D^{1.155}\ (L - D)^{.702}\ V^{.840}$	Eckelman (7)
Lag Screws	$7500\ SG^{1.5}\ D^{.75}\ L$	Newlin & Gahagan (19), Wood Handbook (27)
Drift Pins	$6600\ SG^{2}\ D\ L$	Wood Handbook (27)
Threaded Inserts	$1.188\ D^{.25}\ L^{1.25}\ V$	Eckelman (11)
Spiral Dowels	$6760\ SG^{2}\ D^{.75}\ L$	Wood Handbook (27)

SG = specific gravity (oven-dry basis)
D = fastener shank diameter, in.
L = depth of penetration of threaded portion, in.
V = wood shear strength parallel to grain, psi.

Table 2: Characteristics of fasteners tested in withdrawal study.

Fastener Variable	3/8" Lag Screw	5/8" Lag Screw	Tapping Screw	Fastener A	Fastener B	Fastener C
# Leads	1	1	2	2	2	4
Threads per Inch	7	5	15	4.85	5	2
Pitch (in.)	.1429	.2000	.0667	.2062	.2000	.5000
Thread Angle (deg.)	30	30	30	25	40	120
Lead Angle (deg.)	6.8	5.9	13.8	12.8	11.7	48.4
Major Diameter (in.)	.381	.619	.176	.585	.620	.740
Root Diameter (in.)	.279	.469	.120	.385	.370	.550
Single Thread Depth (in.)	.051	.075	.028	.100	.125	.095
Flat at Root (in.)	.0615	.0861	.0287	.076	.085	.145
Bearing Area per Inch (sq. in.)	.3701	.6409	.1953	.7390	.9719	.3850
Shear Area per Inch (sq. in.)	1.1969	1.9446	.5529	1.8378	1.9478	2.3248

Pitch = 1 / (Threads per Inch)

Lead angle = (# Leads) (Tan^{-1} (Pitch / π D_m))

Bearing Area per Inch = (π/4) (Threads per Inch) ($D_m^2 - D_r^2$)

Shear Area per Inch = π D_m

Table 3: Nominal lead hole size (LH) and test depths of penetration (DP) by fastener and wood species. Note that the DP includes the tip portion of the fastener which may not be totally effective.

Fastener	S-P-F LH in.	S-P-F DP in.	Yellow Poplar LH in.	Yellow Poplar DP in.	Southern Pine LH in.	Southern Pine DP in.	Red Oak LH in.	Red Oak DP in.	Hickory LH in.	Hickory DP in.
3/8" Lag Screw	.172	(a)	.172	2.143	.234	(b)	.250	2.143	.250	1.714
5/8" Lag Screw	.313	3.5	.281	3.5	.391	3.5	.422	3.5	.453	3.5
Tapping Screw	.063	1.0	.063	1.0	.078	1.0	.094	1.0	.094	1.0
Fastener A	.328	4.625	.281	4.625	.328	4.625	.359	4.625	.359	2.941
Fastener B	.328	4.375	.328	4.375	.344	4.375	.359	4.375	.359	2.975
Fastener C	.453	4.0	.469	4.0	.484	4.0	.500	4.0	.516	3.0

a) 12 tests with 4.0", 10 tests with 3.0".
b) 12 tests with 3.0", 8 tests with 2.571", 2 tests with 1.857".

Table 4: Average results of single fastener withdrawal tests.

Species	Sample Size[a]	Specific Gravity	Moisture Content (%)	Joint Capacity/Inch Thread Penetration: Maximum Load (lbs/in)	Load at Proportional Limit (lbs/in)	Joint Stiffness (lbs/in / in)
			*** 3/8" x 6" Lag Screw ***			
S-P-F	22	.38 (11.8)	7.9 (13.3)	1046 (19.9)	834 (13.8)	23,400 (16.0)
Y. Poplar	12	.52 (2.0)	6.9 (1.8)	2076 (3.2)	1700 (5.0)	48,700 (6.9)
S. Pine	22	.47 (17.1)	8.4 (11.4)	1371 (28.9)	1017 (23.7)	30,400 (20.1)
Oak	12	.65 (1.4)	6.4 (5.9)	2273 (3.6)	1796 (7.1)	50,100 (4.9)
Hickory	12	.78 (6.0)	7.9 (2.9)	2831 (10.2)	2042 (14.9)	45,500 (13.1)
			*** 5/8" x 6" Lag Screw ***			
S-P-F	22	.43 (11.9)	7.7 (8.3)	1769 (23.6)	1300 (24.4)	36,000 (24.0)
Y. Poplar	13	.47 (5.3)	6.9 (4.5)	1988 (8.6)	1340 (9.5)	45,000 (10.0)
S. Pine	22	.49 (14.6)	7.5 (5.1)	1902 (8.8)	1412 (10.6)	45,400 (17.9)
Oak	12	.66 (3.9)	6.2 (4.2)	2991 (13.2)	2006 (13.0)	70,900 (32.2)
Hickory	12	.80 (2.9)	7.3 (0.9)	4578 (4.2)	2850 (5.0)	75,400 (9.3)
			*** Tapping Screw ***			
S-P-F	22	.44 (13.6)	7.8 (5.1)	690 (20.6)	504 (21.4)	45,600 (20.9)
Y. Poplar	12	.52 (1.1)	6.1 (2.0)	1193 (3.6)	859 (6.9)	68,500 (4.0)
S. Pine	22	.51 (13.1)	7.4 (8.6)	1030 (17.4)	726 (16.6)	63,400 (11.9)
Oak	12	.70 (2.8)	9.2 (16.1)	1457 (9.0)	954 (9.8)	68,400 (12.7)
Hickory	14	.70 (13.1)	7.2 (2.2)	1794 (20.3)	964 (17.3)	74,500 (11.4)

a) Sample size may vary by property.

Table 5: Results of single fastener withdrawal tests.

Species	Sample Size[a]	Specific Gravity	Moisture Content (%)	Joint Capacity/Inch Thread Penetration: Maximum Load (lbs/in)	Load at Proportional Limit (lbs/in)	Joint Stiffness (lbs/in / in)
				* * * Fastener A * * *		
S-P-F	12	.42 (9.0)	7.4 (4.5)	1496 (13.7)	1108 (14.2)	29,200 (15.3)
Y. Poplar	12	.46 (7.4)	5.6 (4.2)	2208 (15.7)	1467 (17.0)	47,800 (11.3)
S. Pine	12	.52 (9.0)	9.0 (10.4)	1880 (14.0)	1233 (14.3)	39,400 (13.7)
Oak	12	.70 (7.7)	7.6 (21.8)	3538 (13.5)	2283 (14.8)	64,500 (10.3)
Hickory	12	.78 (6.1)	7.5 (4.7)	4696 (14.3)	3356 (13.0)	81,000 (8.8)
				* * * Fastener B * * *		
S-P-F	12	.45 (6.1)	8.5 (4.4)	1892 (9.4)	1407 (10.5)	41,200 (14.0)
Y. Poplar	12	.47 (4.8)	6.5 (5.2)	2386 (10.0)	1783 (11.3)	50,600 (13.9)
S. Pine	12	.59 (9.7)	10.1 (7.0)	2549 (11.1)	1738 (18.0)	48,100 (11.6)
Oak	12	.67 (3.7)	7.3 (21.1)	3681 (13.4)	2688 (18.6)	67,400 (9.3)
Hickory	12	.71 (16.9)	7.3 (4.7)	4405 (26.8)	2878 (29.8)	85,000 (13.5)
				* * * Fastener C * * *		
S-P-F	12	.46 (14.3)	8.4 (10.5)	1552 (29.8)	592 (32.5)	24,000 (49.4)
Y. Poplar	12	.46 (9.2)	7.6 (3.6)	1842 (14.9)	563 (35.0)	24,700 (51.5)
S. Pine	12	.48 (10.4)	8.7 (4.6)	1735 (17.2)	591 (31.8)	31,100 (34.6)
Oak	12	.68 (2.3)	7.0 (2.5)	3223 (6.4)	1442 (16.2)	50,200 (19.7)
Hickory	8	.74 (8.9)	7.7 (3.7)	3855 (10.4)	1716 (20.9)	57,800 (27.3)

a) Sample size may vary by property.

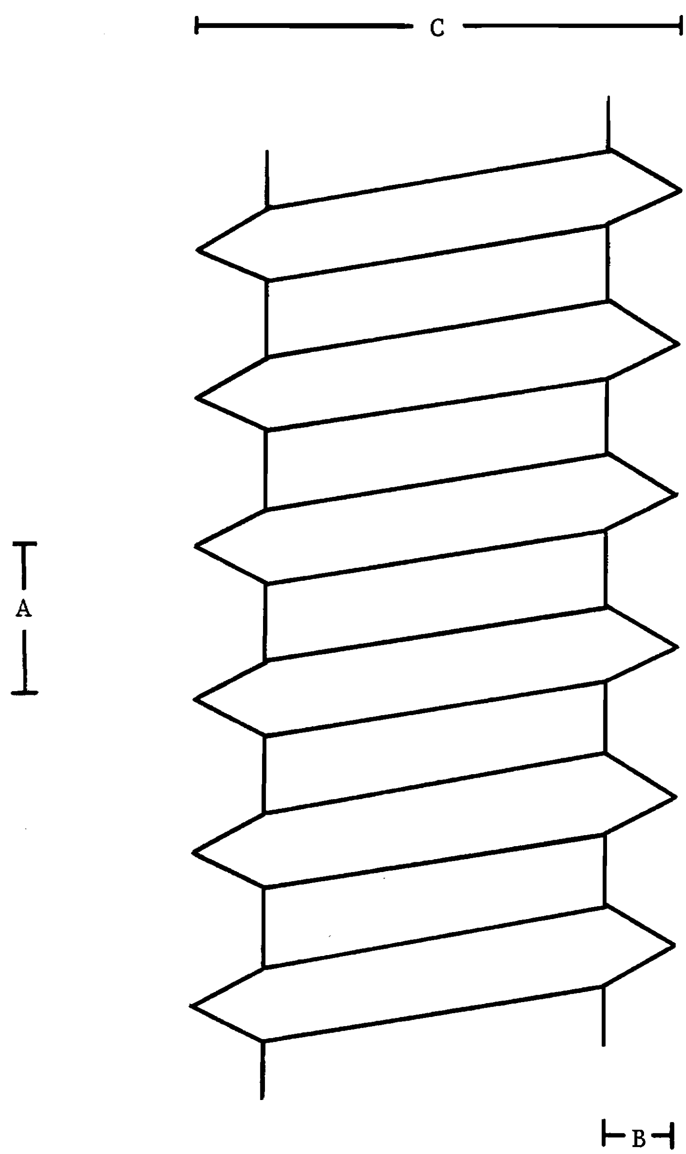

Figure 1: Description of fastener shear and bearing areas.

Shear area equals the surface area of the side of a right circular cylinder of diameter C and height A.

Bearing area equals the sum of the area of the horizontal projection B of the upper surface of the fastener threads.

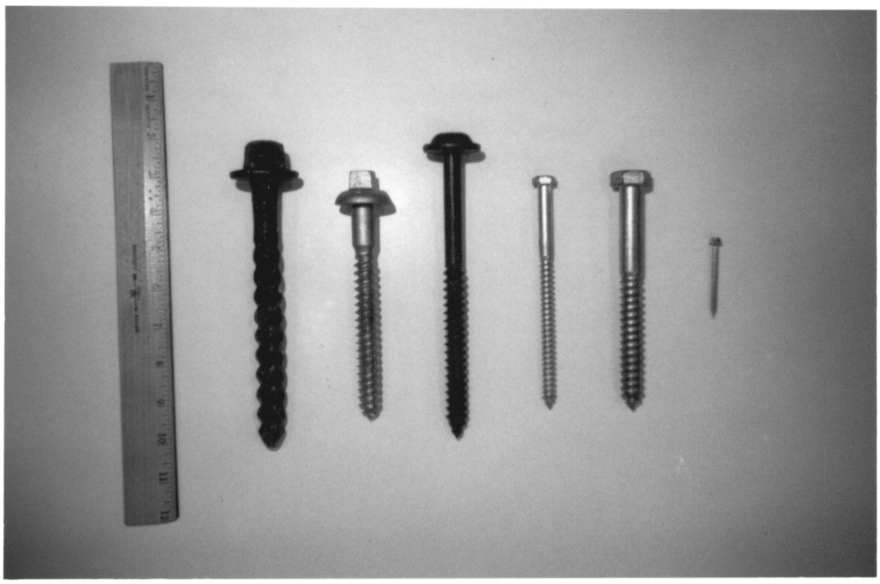

Figure 2: Fasteners tested in withdrawal strength study.

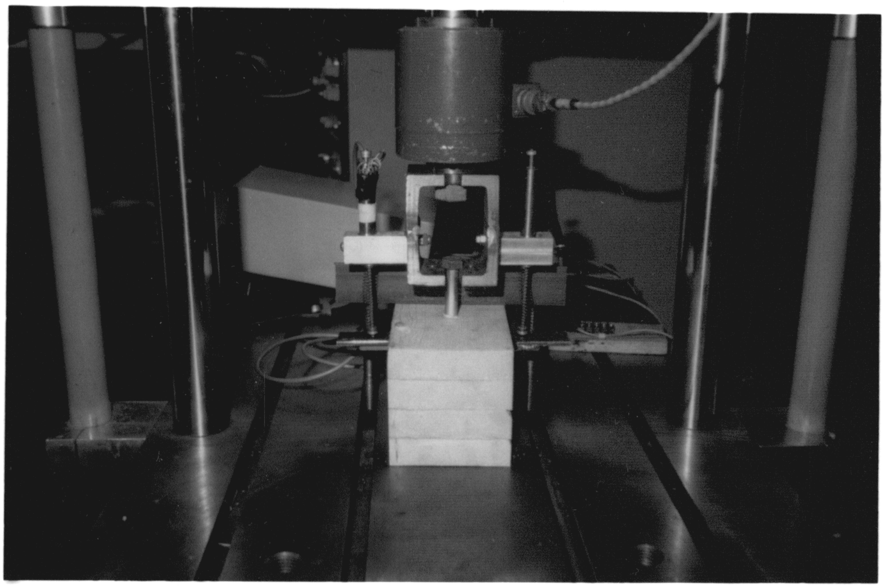

Figure 3: Test machine set-up used in withdrawal strength study.

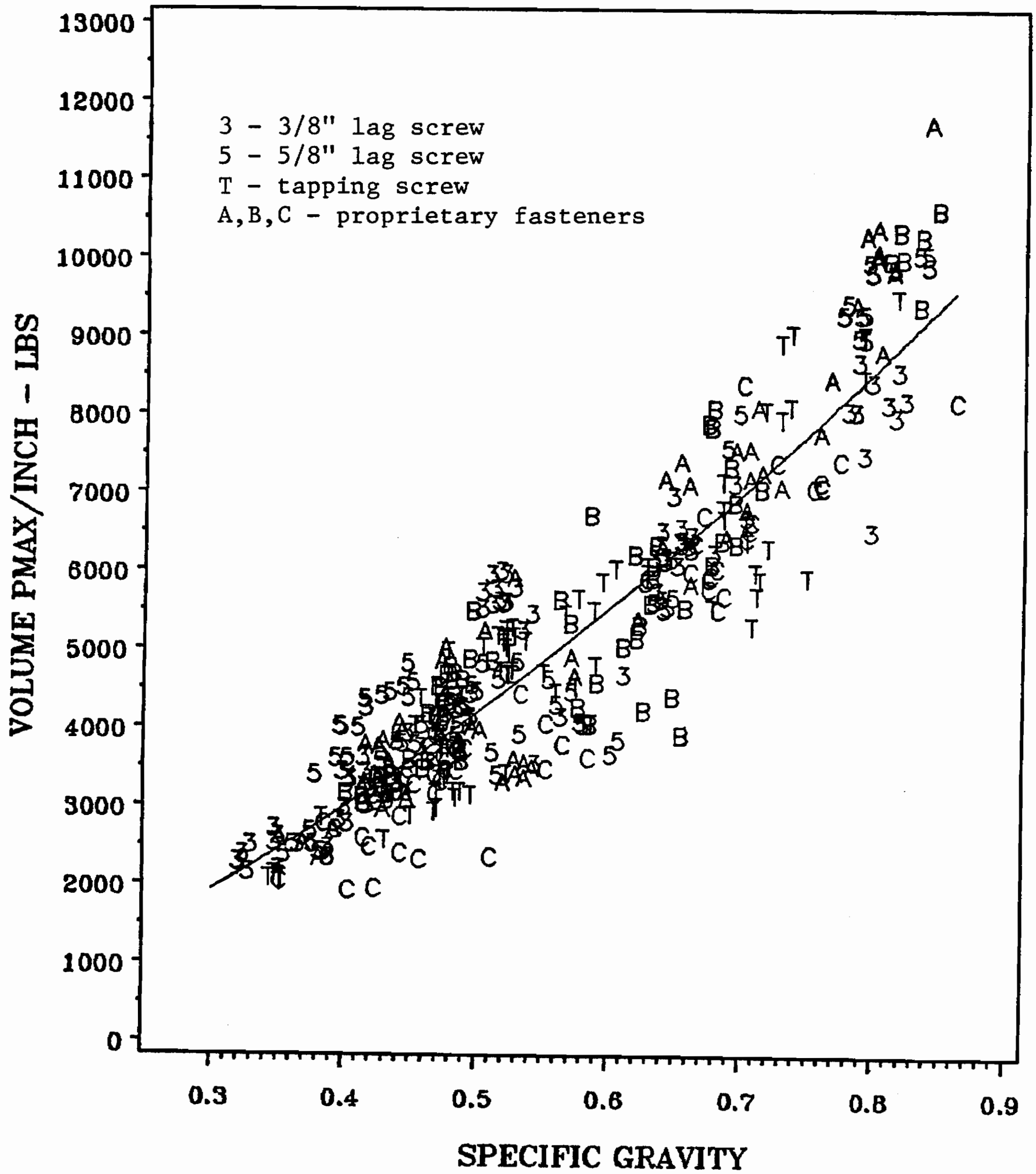

Figure 4: Comparison of predicted maximum withdrawal load using volume model to experimental data.

$$\text{Volume } P_{max}/\text{inch} = (P_{max}/\text{inch}) \,/\, \text{volume}^{.31}$$

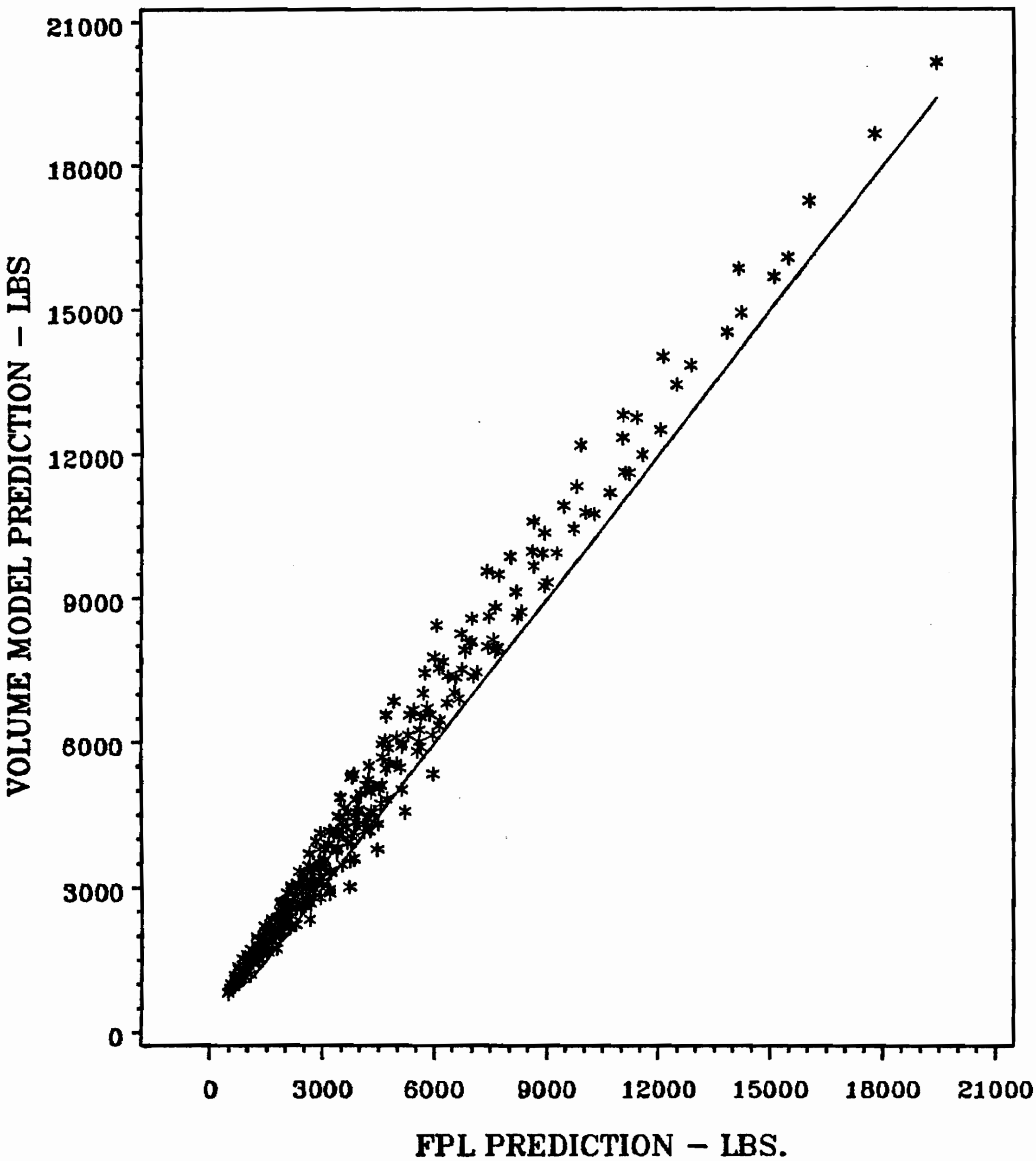

Figure 5: Comparison of predicted maximum withdrawal load using volume model and FPL (Table 1) equations for lag and wood screws.

4.0 Combined Loading Study

Combined Axial and Lateral Load Capacity of Threaded Fastener Wood Joints

4.1 Introduction

Lag screws and similar fasteners are frequently used to resist forces which simultaneously act parallel and perpendicular to the fastener axis. Common examples include attachment of metal down guy brackets in utility structures, transmission of rail forces to railroad ties (27), and the actions of wind forces of metal-clad roof diaphragms (13). These types of applied forces subject the screw to both axial withdrawal and lateral shear with proportions varying with the angle, Θ. The designer of these joints is faced with conflicting guidance. The National Design Specification (20) presumes no interaction between the effects of axial and lateral loading. Hence, the NDS safety checking equations are:

$$\begin{aligned} P_\Theta \cos \Theta &\leq P_{l,allow} \\ P_\Theta \sin \Theta &\leq P_{w,allow} \end{aligned} \qquad [1]$$

where:

P_Θ = design load in lbs. for a joint loaded at some angle (0^o - 90^o) to the wood surface,

$P_{l,allow}$, $P_{w,allow}$ = allowable fastener loads in lateral shear and withdrawal, respectively.

Implicit to the use of [1] is the assumption that the failure mechanisms are shank withdrawal in axial loading and proportional limit in lateral shear. Failure of the fastener itself is not considered.

The Timber Construction Manual (1) implies that there may be some interaction in the response of the joint to the orthogonal vector components. While not an explicit interaction relation, the TCM recommends use of the Hankinson formula to interpolate:

$$P_\Theta \le P_{\Theta allow} = \frac{(P_{w,allow})\ (P_{l,allow})}{(P_{w,allow})\ \cos^2 \Theta + (P_{l,allow})\ \sin^2 \Theta} \qquad [2]$$

where 0^o refers to pure lateral loading and 90^o refers to pure withdrawal. For threaded fasteners with a small lead angle, such as lag screws and wood screws, there are no published data to support the use of either [1] or [2]. The Hankinson formula was originally used to describe the magnitude of wood crushing strength at varying grain angles. For threaded fastener joints subject to combined loading, the formula is an arbitrary empirical description.

The draft Eurocode (7) for timber structures recommends for helically threaded nails that:

$$\left[\frac{P_\Theta \sin \Theta}{P_{w,allow}}\right]^2 + \left[\frac{P_\Theta \cos \Theta}{P_{l,allow}}\right]^2 \le 1.0 \qquad [3]$$

This second order interaction apparently stems from results of tests made by Ehlbeck (11) on threaded and ring shank nails. Debonis and Bodig (8) studied the interaction of withdrawal and lateral forces on smooth-shank nail joints. They discovered differences of 40-70% between experimental data and Hankinson formula interpolation for ultimate load. Based on their data, a first order interaction was identified which reduced differences to less than 15%:

$$\frac{P_\Theta \sin \Theta}{P_{w,allow}} + \frac{P_\Theta \cos \Theta}{P_{l,allow}} \leq Z \qquad [4]$$

where:

$P_{l,max}$ = maximum lateral load, lbs.,

$P_{w,max}$ = maximum withdrawal load, lbs.,

$Z = 1 + K \sin 2\Theta$

K = variable depending on depth of penetration and wood species.

Figure 1 demonstrates the difference between the recommendations of [1] and [2] for a 5/8" lag screw securing a metal plate to a Southern pine member. There are no practical differences between the results of equations [2] and [3]. However, the current NDS guidance, equation [1], deviates substantially from the other two at some angles.

The objective of this study was to experimentally evaluate the behavior of threaded fastener wood joints subject to combined withdrawal and lateral loading. Current design criteria will be applied to the experimental data to determine the accuracy of existing design methods. The results of this study may be used to provide design guidance for future code writers.

4.2 Methods and Materials

In overview, this study consisted of testing joints with metal side plates constructed with three different screws and two wood species. Tests were conducted at angles varying from pure lateral (0^{o}) to pure withdrawal (90^{o}). From the resulting data and test observations, an understanding of fastener action was gained and some guidance for the designer was developed.

To begin the study an apparatus was constructed to position forces at the following angles to the wood surface of a test joint: 0^{o} or pure lateral loading, 22.5 degrees, 45 degrees, 67.5 degrees, and 90^{o} or pure withdrawal. The apparatus was similar to that used by Debonis and Bodig (8) in nailed joint tests, but consisted of much sturdier construction in anticipation of greater load requirements.

The combined loading apparatus consisted of two sections. The lower section was used to position the sample joint at the desired test angle. This section was fitted with a large spring in its base to allow for uniform load distribution and to minimize any unwanted pre-load on the joint. The function of the upper section was to form the connection between the test machine load cell and the screw head. This section was designed to accomodate individual fittings thus allowing testing of numerous fasteners and plate thicknesses. The upper section had two brackets to position LVDT's for measurement of the axial deflection on either side of the screw head. The mean output from the two LVDT's was recorded with the corresponding load on a chart recorder. The lower section of the apparatus was fitted with a single LVDT for measurement of lateral deflections. The output from this LVDT was recorded with load on a second chart recorder. The combined loading apparatus and instrumentation is illustrated in Figure 2.

The three fasteners selected for the study were a 3/8" x 6" lag screw, a 5/8" x 6" lag screw, and a No. 9 x 2" type AB tapping screw (Figure 3). Lag screws are common in glulam and heavy timber construction while the tapping screws are used to fasten metal-clading to roofs and walls in agricultural structures. The two lag screws are identical to those tested in a companion study of withdrawal strength and were identified as very poplular sizes based on U.S. production. The tapping screw is typical of that used in the agricultural structures industry and elsewhere. It was selected for use because it was expected that its response would be dissimilar to that of the lag screws. The wood species chosen included S-P-F and Southern pine. Both are major structural species in North America and represent different density ranges as well as NDS (20) fastening species classes.

The side members of lag screw joints consisted consisted of a 1/2" steel cleat while tapping screw joints had a 1/4" cleat. Specimens were constructed by laminating lumber segments into blocks of sufficient depth to allow minimum depths of penetration of 5.5 and 1.75 inches for the lag and tapping screws, respectively. Only clear segments cut from an individual piece of lumber were used in each sample block. All S-P-F specimens were constructed from 2 x 6 material with block lengths of 8, 10, and 12 inches used for the tapping screw, 3/8" lag screw, and 5/8" lag screw joints, respectively. Southern pine blocks were similar except that 2 x 4 was laminated for the 3/8" lag screw joints.

In 5/8" lag and tapping screw joints, 15 replicates were tested for each fastener-species-angle combination at angles between 0 and 67.5 degrees. For 3/8" lag screw joints, 10 replicates were tested at those angles. Twenty-two replicates were tested in pure withdrawal (90^{o}) for each fastener-species as part of the companion study of withdrawal resistance. As a result there were some density differences between the 90^{o} samples and the others. However, regresssion analysis developed as part of the withdrawal study provided means to adjust to a common specific gravity. All screws were selected from a larger sample and each type were from the same manufacturer. The lag screws met the requirements of ANSI B18.2.1 (3) with the exception that the length of the threaded portion of the 3/8" screw was 4.0" instead of 3.5".

Lead holes for 5/8" lag screws were drilled completely through the sample blocks with a diameter of 5/16" in S-P-F and 25/64" in Southern pine. A second lead hole 19/32" in diameter was drilled into the original hole to depth of 1 7/8" to accomodate the unthreaded portion of the screw shank. For the 3/8" screws, lead holes for the threaded portion were 11/64" and 15/64" for S-P-F and Southern pine, respectively. The lead hole for the shank portion was 23/64" in diameter and was drilled to a depth of 1 7/16". For the tapping screws, a single lead hole 1/16" in diameter was drilled to a depth of approximately 1.75" for both the S-P-F and Southern pine specimens. Lag screws were driven into lead holes using a pneumatic wrench and air compressor until 1/2" of the screw shank extended from the block. Tapping screws were installed using a 1/4" socket fitted to a variable speed electric drill and were driven into the test specimens until 1/4" of the threaded portion extended from the block.

Testing was conducted on an MTS hydraulic test machine under a stroke control rate of 0.10 inches per minute. Prior to lag screw joint tests, a pre-load of approximately 100 lbs. was applied to the joint to take up slack in the combined loading apparatus. No pre-load was applied in the tapping screw joint tests. Two load-deformation curves were simultaneously generated for each joint during testing with both axial and lateral component deflections being recorded. However, tests of joints at angles of 0 and 22.5 degrees yielded meaningless withdrawal load-deformation curves due to the bending of the exposed portion of the screw shank towards the block. Withdrawal deflections were therefore recorded for only the 45^{o}, 67.5^{o}, and 90^{o} tests. Deflection in the lateral direction was recorded for all joint tests.

From the recorded joint test data, the following were identified and tabulated for further analysis: load and deflection at the proportional limit (for both withdrawal and lateral components, if favorable), maximum load, and yield load. A true maximum load was not recorded for the 0^{o} and 22.5^{o} lag screw tests since these joints experienced excessive deformations which could not be accomodated by the test apparatus. For the 5/8" lag screw joints, loading was ceased after approximately 0.5" of lateral deflection. The 3/8" lag screw joint tests were continued until lateral deflection had reached four-tenths of an inch. Maximum load was recorded for all 90^{o} and 67.5^{o} lag screw joint tests and two-thirds of the 5/8" lag screw joint 45 degree tests. A pronounced maximum load was not found for the 3/8" lag screw tests at 45^{o} or for many of the 5/8" lag screw Southern pine joints tested at this angle.

Yield load was recorded for the lag screw joints loaded at 0^{o}, 22.5^{o}, and 45 degrees. This definition of yield load has been used by a number of authors (19,22) to define a predictable point in the lateral deformation curve of dowel fastener joints. This definition is not the same as that used by Larsen and Reestrup (16) for laterally-loaded lag screw joints. A well-defined yield mechanism was not observed for lag screw joints at 67.5 or 90 degrees. Yield loads were not observed or recorded for the tapping screw joints tested at any of the five angles. In tapping screw joints subjected to 0 to 45^{o} loading, and in the 67.5^{o} pine joints, a combined bending and shear failure occurred in the fastener at the junction of the head and shank. The 67.5^{o} S-P-F joints failed primarily in fastener withdrawal. Therefore, maximum loads were recorded for tapping screw joints tested at all angles.

Moisture content and specific gravity (oven dry weight and volume) were calculated and recorded for each test sample. The ultimate tensile strength of the 3/8" lag and the tapping screws was determined from earlier withdrawal tests to be approximately 90 and 175 ksi, respectively. Tests were also conducted to determine the bending yield stress of the two lag screws. This information is important for subsequent interpretation of the combined loading data.

4.3 Results and Discussion

The results of the combined loading study are summarized in Tables 1-3. To adjust the withdrawal (90^{o}) data to a specific gravity similar to that of specimens tested at other angles, the following was used (5):

$$P_f = P_i \left[\frac{SG_f}{SG_i} \right]^k$$

where P_i and P_f are initial and adjusted withdrawal strengths, SG_i and SG_f are initial and target specific gravity, and $k = 1.3$ or 1.5 for adjustment of proportional limit (P_{pl}) or maximum load (P_{max}), respectively. These exponents were determined from the companion withdrawal strength study. For the 3/8" lag and tapping screws, the mean maximum loads in withdrawal were adjusted to 5642 lbs. and 1965 lbs., respectively. These values correspond to the ultimate tensile load for the two fasteners.

Connection proportional limit load was defined as the minimum P_{pl} from either the axial or lateral component curves. In general, the lateral component P_{pl} was critical at angles other than 90 degrees. Figures 4 and 5 show mean connection P_{pl} for tapping and 5/8" lag screw joints with Hankinson's formula superimposed. In Figure 4 Hankinson's formula is seen to fit the data trends reasonably well for both species. However, maximum errors between predicted and observed means were approximately 12% and 22% for S-P-F and Southern pine, respectively. In S-P-F, the maximum discrepancy occurred at the 45^{o} orientation. Proportional limit for these joints appeared to be wood related although mixed wood and fastener failures contributed to maximum load. In Southern pine joints, the maximum difference was found at 67.5 degrees. At this orientation proportional limit may be less related to wood response since fastener failure resulted in 11 of 15 tests.

Hankinson's formula did not fit connection P_{pl} in 5/8" lag screw joints as seen in Figure 5. Experimental means were underestimated by 9% for S-P-F and 17% for Southern pine at the 67.5^{o} orientation. Conversely, the formula overestimated proportional limit load by 8 to 39% at other intermediate angles. Fit of Hankinson's formula to 3/8" lag screw joint data was also quite poor with prediction errors ranging from 4 to 32%.

Equation [2] can also be applied to a connection ultimate load though in some instances the definition of "ultimate" becomes subjective. Failure of an axially loaded joint results at an abrupt, well-defined maximum load. However, in lateral loading of lag screw joints, a maximum tolerable distortion must be set. In this study, ultimate load was defined as the minimum of the true ultimate or load recorded at deflections of 0.4" or 0.5" for 3/8" and 5/8" lag screw joints, respectively. Hankinson's formula was applied to data using this definition with the result that equation [2] underestimated capacity at all intermediate angles of load by up to 25% in 3/8" lag screw joints and 20% in 5/8" lag screw joints. For tapping screws, where a true maximum load was observed at all angles, Hankinson's formula fit reasonably well but again underestimated capacity by 10-14% across the spectrum of intermediate angles.

The influence of any interaction between axial and lateral force components on joint capacity was explored graphically. A plot of this type for maximum load of tapping screw joints is shown in Figure 6. The ordinate is the ratio of $P_\Theta \sin \Theta$ to the average ultimate withdrawal load, P_w, for S-P-F or Southern pine. P_Θ corresponds to "ultimate" joint strength as defined earlier. The abscissa is a similar ratio of $P_\Theta \cos \Theta$ to the average lateral load capacity, P_l, for the corresponding species. Superimposed on Figure 6 are linear and non-linear curves illustrating interaction forms similar to those observed by DeBonis and Bodig (8) and Ehlbeck (11) for nailed joints. Neither form fit the tapping screw joint P_{max} data.

Figure 7 shows a similar plot for connection proportional limit in tapping screw joints in which P_{pl} data was substituted for P_Θ, P_w, and P_l. Here a function similar to equation [3] may describe an apparent interaction between axial and lateral components in P_{pl} of tapping screw joints.

Figure 8 shows data for both proportional limit and "ultimate" load for both 3/8" and 5/8" lag screw joints. A continuous function such as [2] does not adequately describe any interaction between axial and lateral force components for either P_{max} or P_{pl} in these joints.

In general, connection capacity (either P_{max} or P_{pl}) for the 90^o and 67.5^o orientations were very similar. This similarity is a primary reason for the lack of fit of Hankinson's formula and an apparent lack of an interaction in lag screw joints. At 45^o there was a substantial change in joint response. In effect, joint capacity in 90^o and 67.5^o loading was dominated by the withdrawal resistance of the fastener while in 0^o to 45^o loading, lateral response seemed dominant. Based on these observations, the following rules are suggested for lag screw joint response:

For load orientations of $67.5^o \leq \Theta \leq 90^o$ to the wood surface;

$$P_\Theta \sin \Theta \leq P_w \qquad [5]$$

For load orientations of $45^o \leq \Theta \leq 0^o$ to the wood surface;

$$P_\Theta \cos \Theta \leq P_l$$

where:

P_Θ = capacity for a joint loaded at an angle, Θ, to the wood surface,
P_w = joint capacity in withdrawal,
P_l = joint capacity in lateral shear.

Additional data is required to determine which criterion applies for load orientations between 45^o and 67.5 degrees. The latter rule will provide conservative design guidance at these orientations.

Figure 9 shows experimental maximum load vs. target value determined using [5]. For each species and fastener, mean values of P_w and P_l were used to predict P_Θ. Hence the scatter about the line of equality includes variation in specific gravity.

4.4 Conclusion

The study results showed that Hankinson's formula fit experimental proportional limit and maximum load data reasonably well for tapping screw joints, but when applied to lag screw joints, the formula yielded errors of up to 39%. Linear and non-linear interaction equations were also used to attempt to describe combined loading effects. A non-linear interaction equation was found to describe proportional limit in tapping screw joints. However, neither form of the interaction equation could accurately describe proportional limit or maximum load in lag screw joints, or maximum load in tapping screw joints. A modification of the National Design Specification (20) vector relationships was ultimately suggested for design purposes.

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Table 1: Summary results from combined load tests with 3/8" lag screws.

Angle[a] of Load (deg.)	Sample[b] Size	Specific Gravity	Moisture Content (%)	Load at Proportional Limit (lbs.)		Yield Load (lbs.)	Maximum Load (lbs.)		Stiffness (lbs./in)	
				Lateral	Axial		True	0.4" def.	Lateral	Axial
					*** Spruce-Pine-Fir ***					
0	10	0.40(9.8)	12.9(6.0)	616(15.9)	-	952(11.0)	-	1838(19.8)	16,800(21.9)	-
22.5	10	0.39(10.1)	12.3(9.8)	673(17.9)	-	954(12.2)	-	2126(21.2)	14,800(36.2)	-
45	10	0.38(10.1)	13.2(5.3)	903(15.7)	-	1277(8.6)	-	2664(12.5)	19,200(29.9)	58,100(36.9)
67.5	11	0.40(11.8)	13.3(4.7)	2893(22.3)	3577(12.5)	-	4481(13.9	-	21,100(16.0)	72,600(13.9)
90	22	0.38(11.8)	7.9(13.3)	-	3181(13.8)	-	3989(19.9)	-	-	89,400(16.0)
					*** Southern Pine ***					
0	10	0.61(15.0)	12.4(6.6)	880(17.3)	-	1138(12.1)	-	2807(17.0)	19,900(18.7)	-
22.5	10	0.59(8.6)	12.1(9.1)	929(14.5)	-	1219(8.1)	-	2920(9.7)	17,400(14.5)	-
45	10	0.58(12.4)	12.0(9.6)	1266(18.3)	-	1546(8.5)	-	3537(13.0)	23,200(20.7)	63,800(26.1)
67.5	10	0.61(11.1)	12.1(9.3)	3880(11.5)	3910(13.6)	-	6086(3.4)	-	28,300(21.6)	57,200(16.3)
90	22	0.47(17.1) [0.60 (d)]	8.4(11.4)	-	3877(23.7) [5326 (d)]	-	5227(28.9) [5642 (d)]	-	-	116,000(20.1)

Note: a) 0 degrees = Pure lateral loading; 90 degrees = Pure withdrawal.
b) Sample size may vary by property and angle.
c) Values in ( ) are C.O.V. in percent.
d) Value adjusted to SG as shown.

Table 2: Summary results from combined load tests with 5/8" lag screws.

Angle[a] of Load (deg.)	Sample[b] Size	Specific Gravity	Moisture Content (%)	Load at Proportional Limit (lbs.)		Yield Load (lbs.)	Maximum Load (lbs.)		Stiffness (lbs./in)	
				Lateral	Axial		True	0.5" def.	Lateral	Axial
					*** Spruce-Pine-Fir ***					
0	15	0.39(8.6)	10.9(9.7)	1528(21.4)	-	2504(13.1)	-	4749(20.5)	20,700(22.7)	-
22.5	15	0.38(12.5)	12.4(7.3)	1425(22.2)	-	2757(14.5)	-	5039(14.2)	37,000(24.3)	-
45	14	0.40(11.0)	12.0(10.8)	1744(22.2)	2978(25.2)	3473(13.8)	6738(17.7)	6755(17.9)	54,400(19.8)	13,400(27.6)
67.5	15	0.38(12.5)	11.9(11.4)	3999(32.6)	3742(28.2)	-	5947(25.0)	-	184,100(41.8)	12,500(27.3)
90	22	0.43(11.9) [0.39 (d)]	7.7(8.3)	-	4307(24.4) [3794 (d)]	-	5859(23.6) [5060 (d)]	-	-	119,100(24.0)
					*** Southern Pine ***					
0	15	0.61(9.9)	10.5(7.9)	2736(10.0)	-	3395(4.1)	-	6875(14.3)	29,600(16.4)	-
22.5	15	0.60(11.8)	11.3(5.7)	2768(15.2)	-	4010(15.9)	-	7530(9.1)	58,200(31.1)	-
45	15	0.59(11.6)	11.1(6.0)	2715(16.6)	3483(18.1)	4350(9.2)	8805(8.2)	9332(11.6)[e]	64,300(22.2)	25,000(15.9)
67.5	15	0.60(10.9)	11.4(5.8)	6198(14.0)	5699(16.1)	-	10310(11.6)	-	307,000(32.1)	20,000(17.1)
90	22	0.49(14.6) [0.60 (d)]	7.5(5.1)	-	4677(10.6) [6085 (d)]	-	6300(8.8) [8535 (d)]	-	-	150,200(17.9)

Note:
a) 0 degrees = Pure lateral loading; 90 degrees = Pure withdrawal.
b) Sample size may vary by property and angle.
c) Values in ( ) are C.O.V. in percent.
d) Values adjusted to SG as shown.
e) Less than half of samples exhibited a true maximum.

Table 3: Summary results from combined load tests with tapping screws.

Angle[a] of Load (deg.)	Sample[b] Size	Specific Gravity	Moisture Content (%)	Load at Proportional Limit (lbs.)		Maximum Load (lbs.)	Stiffness (lbs./in.)	
				Lateral	Axial		Lateral	Axial
				* * * Spruce-Pine-Fir * * *				
0	15	0.38(12.4)	12.0(6.0)	222(22.7)	-	384(12.9)	1973(14.9)	-
22.5	15	0.39(12.1)	12.0(5.3)	220(10.8)	-	479(9.3)	2627(21.0)	-
45	15	0.37(12.7)	11.9(4.3)	296(16.6)	416(24.9)[e]	590(13.9)	5217(33.4)	66,300(30.0)
67.5	15	0.39(10.9)	12.5(5.4)	516(19.9)	673(17.1)	810(19.5)	16,983(50.7)	22,900(40.9)
90	22	0.44(13.6) [0.38 (d)]	7.8(5.1)	-	807(21.4) [666 (d)]	1104(20.6) [886 (d)]	-	39,500(20.9)
				* * * Southern Pine * * *				
0	15	0.58(12.7)	11.5(5.7)	328(12.8)	-	440(8.7)	2805(37.1)	-
22.5	15	0.59(12.6)	11.7(4.6)	380(14.4)	-	578(10.6)	4399(27.4)	-
45	15	0.58(13.7)	11.4(4.9)	462(13.6)	536(16.6)[e]	804(11.1)	7859(32.4)	96,900(11.7)
67.5	15	0.62(11.2)	12.3(8.2)	780(11.4)	1108(7.0)	1299(7.3)	35,614(41.1)	33,700(46.6)
90	22	0.51(13.1) [0.59 (d)]	7.4(8.6)	-	1161(16.6) [1403 (d)]	1647(17.4) [1965 (d)]	-	55,000(11.9)

Note: a) 0 degrees = Pure lateral loading; 90 degrees = pure withdrawal.
b) Sample size may vary by property.
c) Values in ( ) are C.O.V. in percent.
d) Values adjusted to SG shown.
e) Less than half of samples exhibited an identifiable proportional limit in withdrawal.

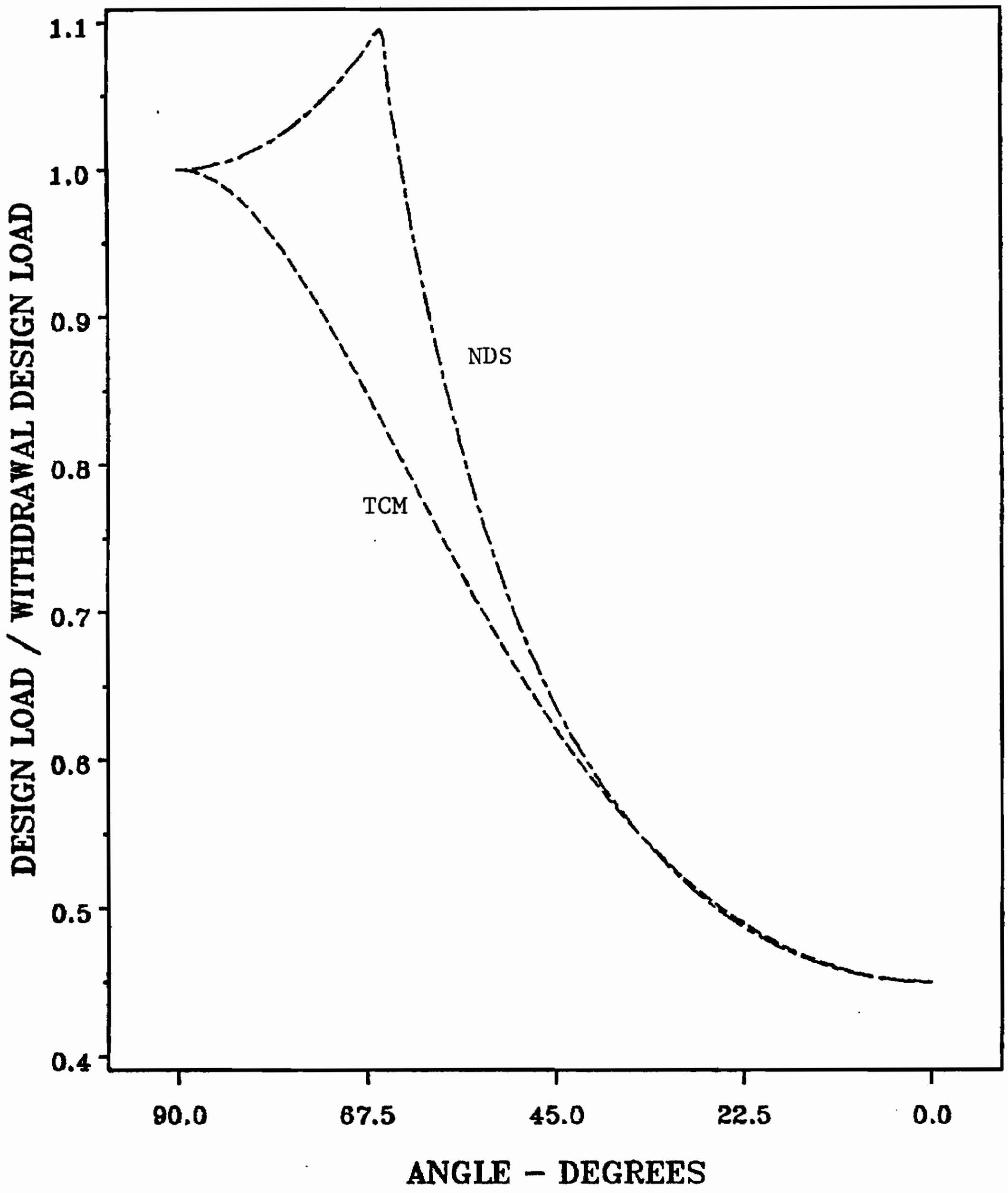

Figure 1: Comparison of NDS and TCM combined loading design criteria.

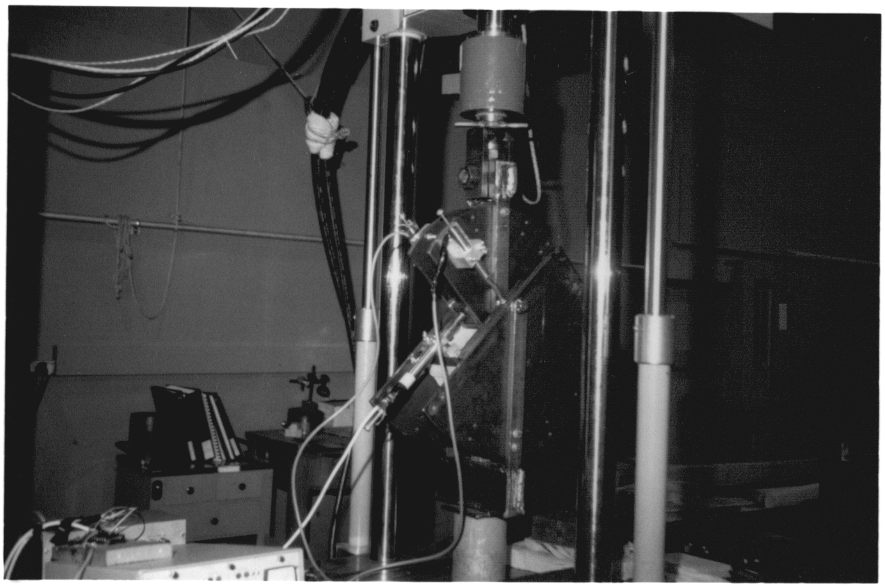

Figure 2: Combined loading apparatus and instrumentation.

Figure 3: Fasteners tested in combined loading study.

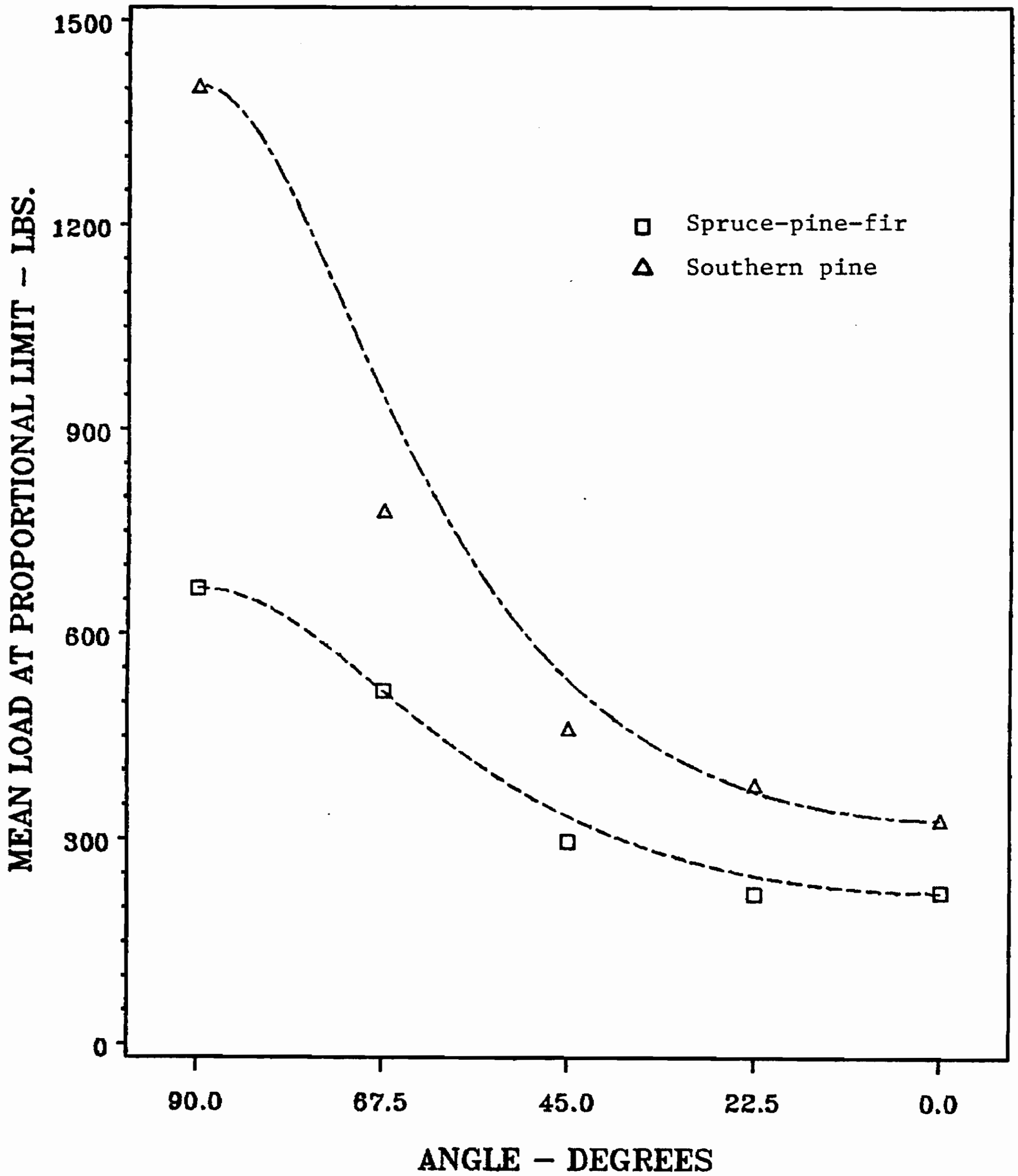

Figure 4: Fit of Hankinson's formula to experimental data - Tapping screw joints.

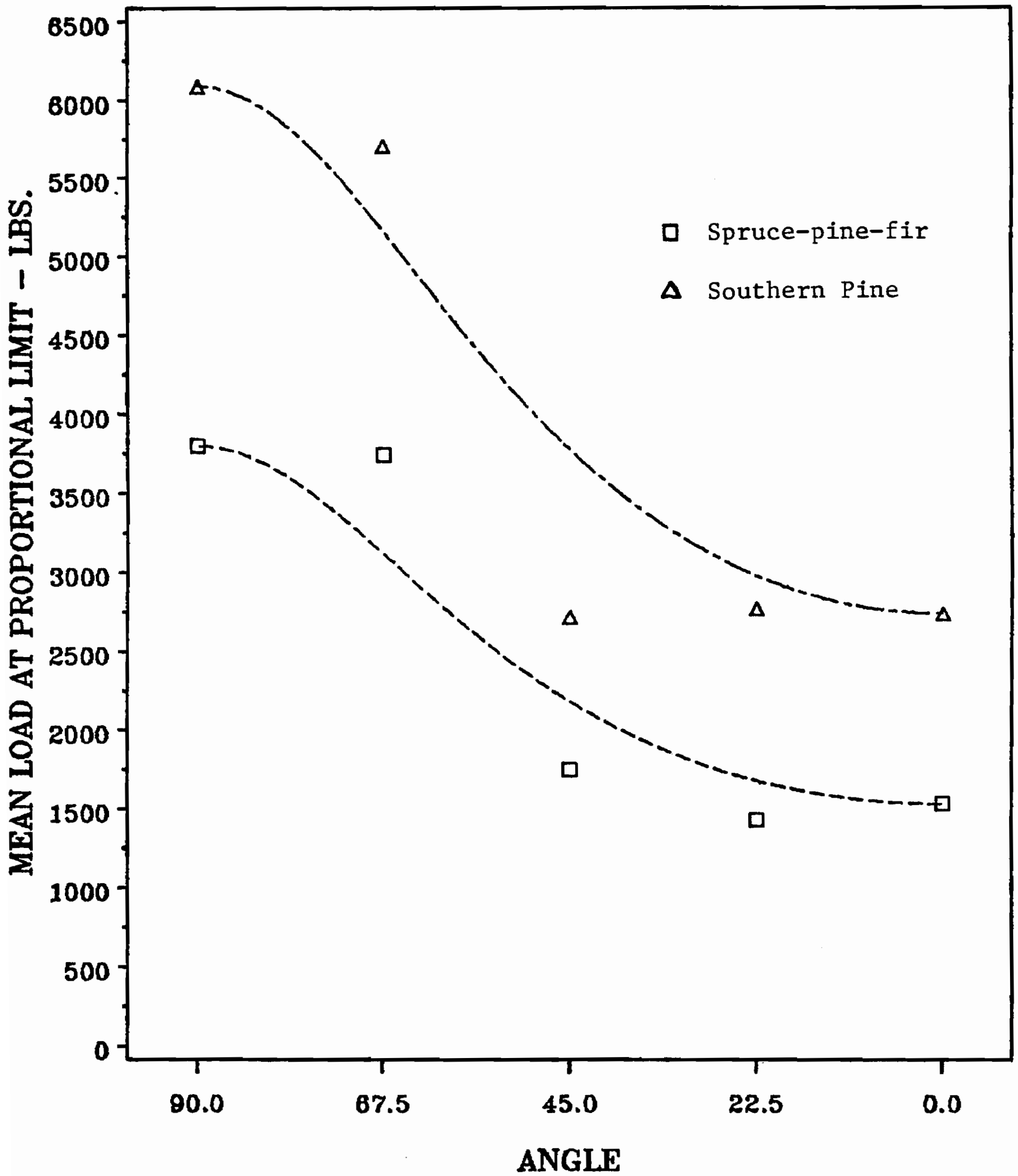

Figure 5: Fit of Hankinson's formula to experimental data - 5/8" lag screw joints.

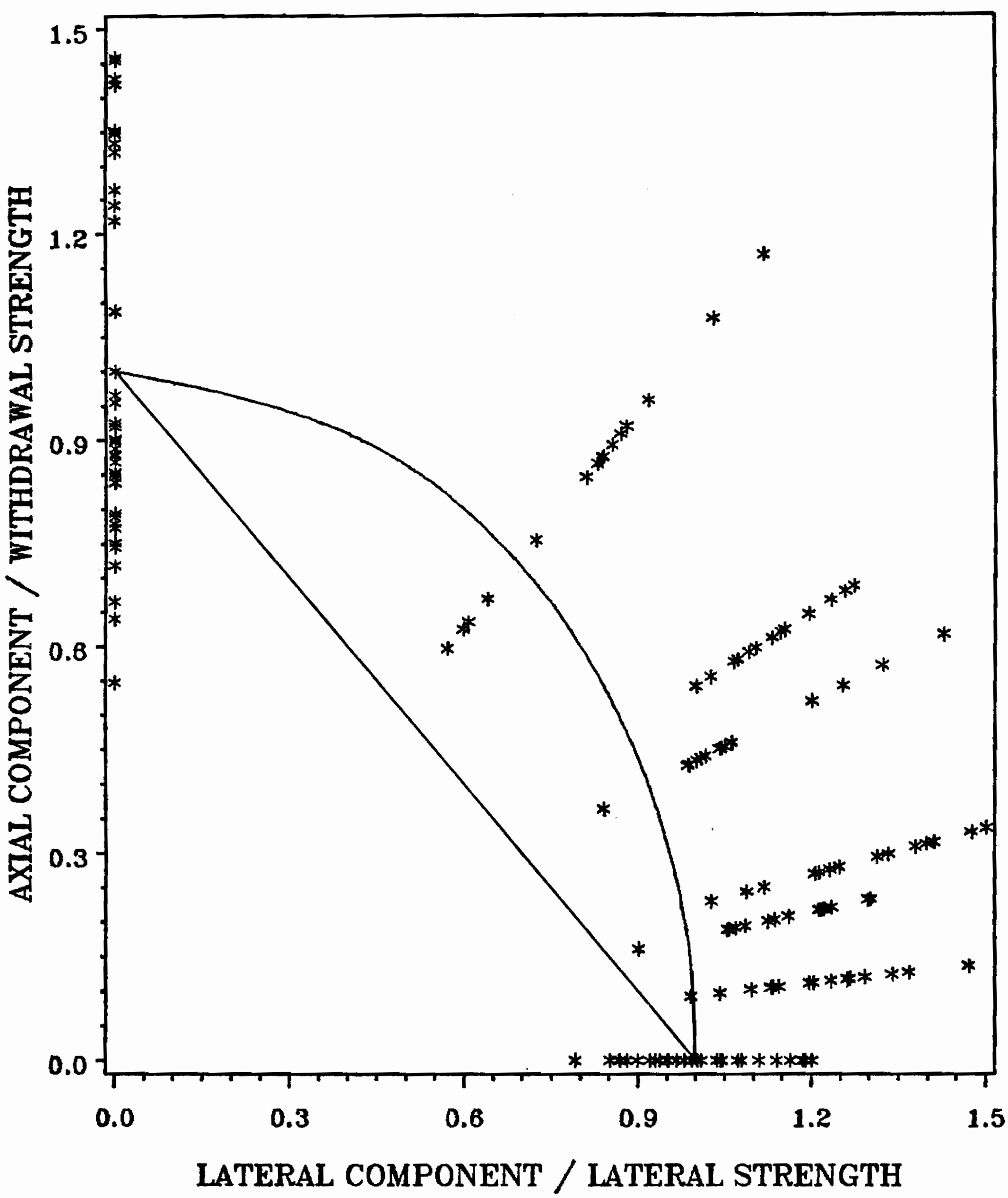

Figure 6: Interaction of axial and lateral components of maximum load - Tapping screw joints.

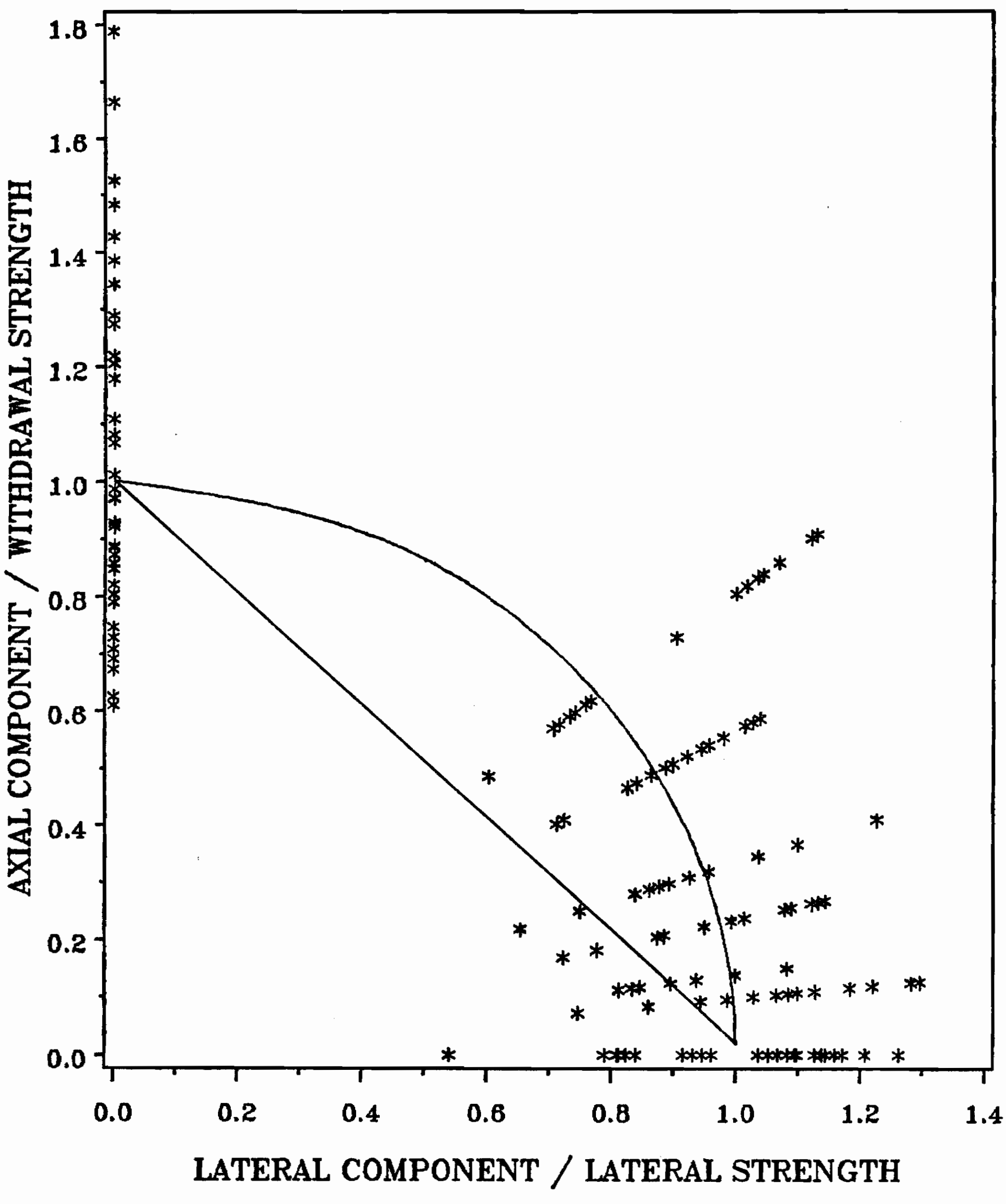

Figure 7: Interaction of axial and lateral components of load at the proportional limit - Tapping screw joints.

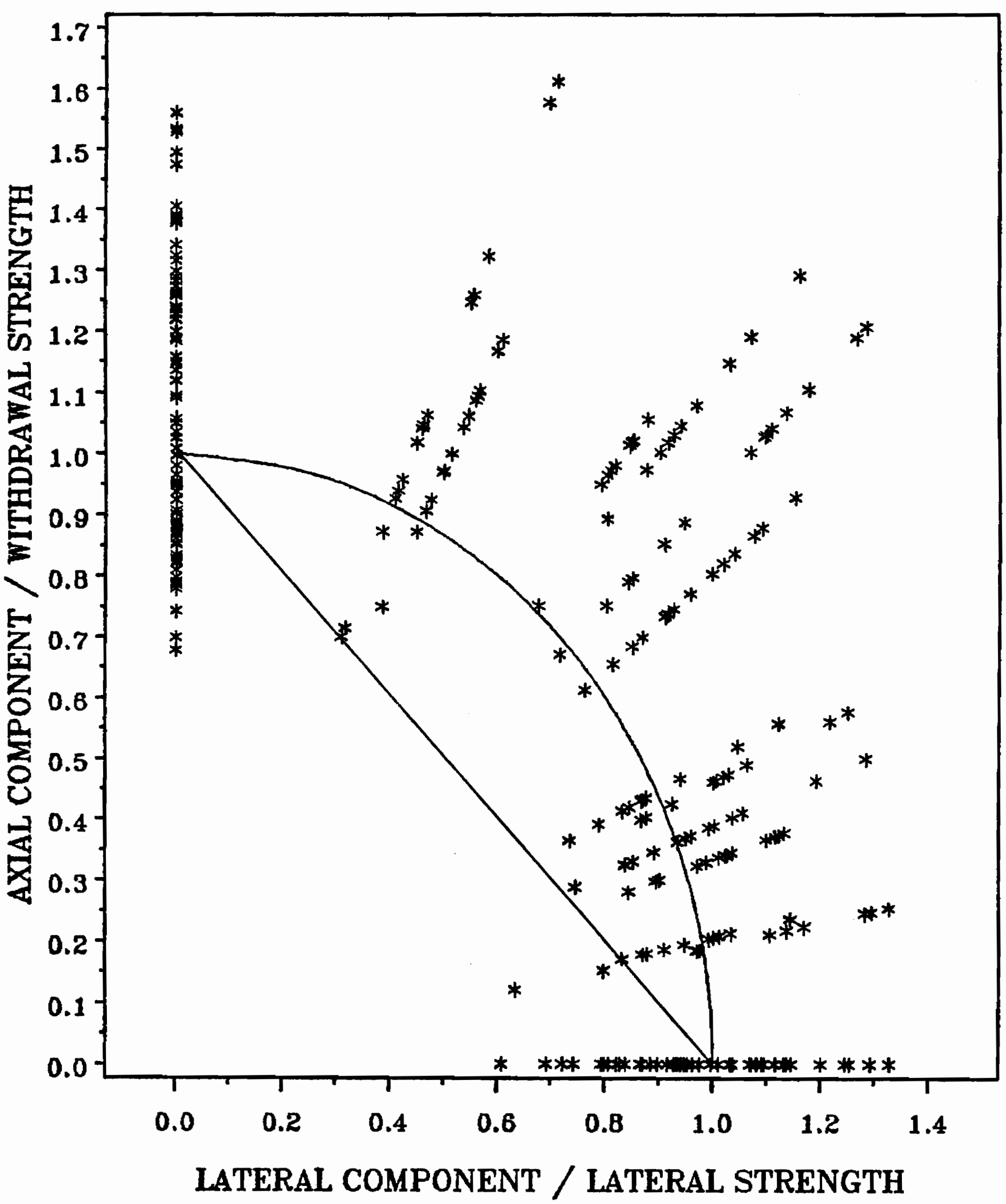

Figure 8: Interaction of axial and lateral force components of maximum and proportional limit load - lag screw joints.

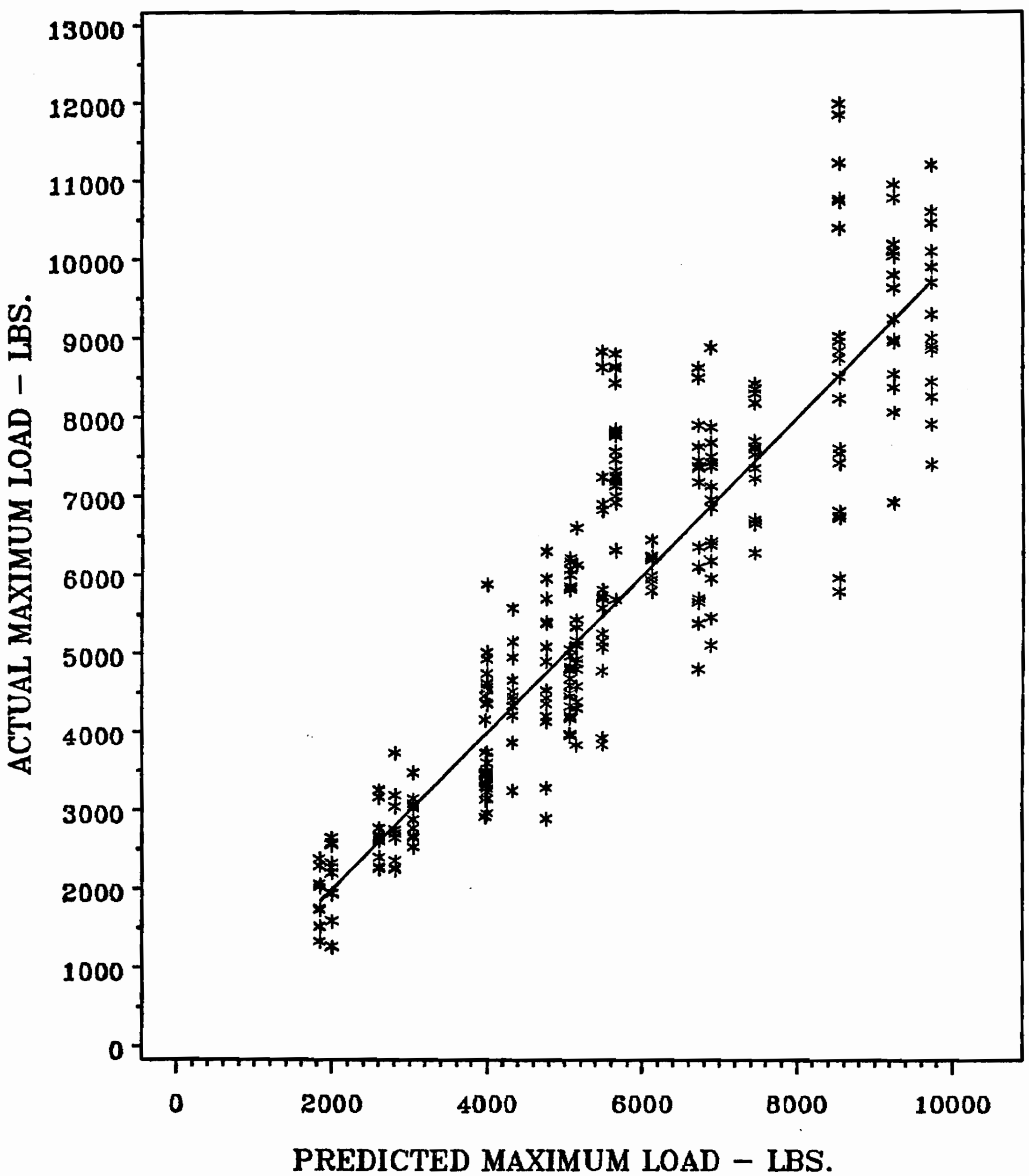

Figure 9: Comparison of actual and predicted maximum load - lag screw joints.

5.0 Desorption Effects Study

Effects of Moisture Desorption on Withdrawal Strength of Tapping Screws

5.1 Introduction

Plain shank nails driven into green wood lose as much as 80% of their initial withdrawal resistance as a result of wood drying to a point below the fiber saturation point (FSP) (12). The withdrawal resistance of threaded nails, however, remains constant or may slightly increase under similar conditions (12). Threaded nails driven into lumber dried below FSP will also have increased withdrawal resistance following further drying.

The Wood Handbook (13) specifies no reduction in withdrawal strength of threaded nails for desorption conditions. The same is implied for wood screws though there is no clear indication that withdrawal strength is exempt from reduction. No guidance is given for use of lag screws. The Timber Construction Manual (1) specifies a factor of 0.75 to be applied to withdrawal strength of a lag or wood screw inserted in green wood and then allowed to dry in place.

The purpose of this preliminary study is to determine if drying from an original moisture content above the fiber saturation point to a final moisture content below FSP has a significant effect on the withdrawal strength of tapping screws. These fasteners are common in the agricultural structures industry where they are used to attach metal clading to wood roof and wall systems. The results of this study will indicate whether the effects of wood desorption warrant use of an adjustment factor to calculate axial design loads for these fasteners.

5.2 Methods and Materials

Twenty spruce-pine-fir (S-P-F) and twenty Southern pine specimens were machined to approximately 3.75" x 7" x 1.5". These specimens were free from defects and were oriented to insure withdrawal testing in the radial direction. Specimens were manufactured in pairs with both specimens cut from a single piece of lumber. One sample from each pair was randomly assigned to one of two groups: 1) fastener inserted into wet wood and withdrawn following drying to below FSP, or 2) fastener inserted and withdrawn from dry wood.

Lead holes were drilled into all sample blocks to a depth of 1.5" with diameters of 1/16" or 5/64" for S-P-F and Southern pine, respectively. A variable speed electric drill was used to insert No. 9 x 2" type AB tapping screws to a depth of one inch into those blocks designated for dry testing. Withdrawal testing of dry samples followed immediately. The twenty blocks destined for the desorption series were weighed and then soaked for approximately 7 weeks at which time fasteners were inserted. Average moisture content of the wet blocks was approximately 66% and 61% for the S-P-F and pine samples, respectively. The desorption blocks were placed in a room where temperature and relative humidity were constant at approximately 72°F and 35%. The sample blocks remained in these conditions for a period of eight weeks. To slow the initial drying rate and thus reduce the possibility of degrade, the blocks were enclosed in plastic bags with holes in them during the first two weeks of drying. At the end of the eight week period the blocks were moved to new conditions of 88°F and 22% R.H. where they remained for three additional weeks. At the end of the eleven week period the desorption blocks had dried to approximately 10% moisture content. No noticeable defects were present in the samples at the conclusion of the desorption process.

The withdrawal testing was conducted on a 20,000 lb. MTS hydraulic test machine at the ASTM (2) specified rate of stroke control of 0.10 inch/ min. The withdrawal testing setup is illustrated in Figure 1. Load-deformation curves were generated during testing from which maximum withdrawal load (P_{max}) and load and deflection at the proportional limit (P_{pl}, d_{pl}) were obtained. The specific gravity (oven-dry weight and volume) and moisture content of each withdrawal sample was also determined.

5.3 Results and Discussion

The results of the desorption effects study are presented in Table 1. To evaluate the significance of any desorption effect on joint performance, a two-way analysis of variance (ANOVA) was performed between the dependent variables tabulated in Table 1 and the independent variables of initial moisture condition (wet or dry) and final moisture content. The ANOVA was performed for both combined and individual species data. The results of these analyses are presented in Table 2. No adjustment was made for the moisture content difference between the two desorption groups since moisture content effects were not significant in any of the ANOVA's. The dependent variables are plotted against specific gravity in Figures 2-5. The two moisture conditions have been identified in these plots by differing symbols.

Table 1 shows that the desorbed connections for both species had lower mean values of P_{max}, P_{pl} and stiffness and a higher mean d_{pl} than those found for the dry connections. However, ANOVA failed to show that the effect of wood desorption on P_{max} was significant at the .05 level for either the combined or individual species data. The effect of wood desorption on joint stiffness was significant in all analyses. Desorption effects were also significant on both P_{pl} and d_{pl} in Southern pine but not for either of these parameters in S-P-F. For the combined data, desorption effects were significant for d_{pl} but not P_{pl}.

From the individual species data it appears that desorption may have greater impact on screw withdrawal from Southern pine than from S-P-F. Southern pine joints experienced a 39% loss of stiffness and a 45% increase in d_{pl} following desorption. In S-P-F, stiffness decreased 29% while a 15% increase in d_{pl} was observed. Figures 4 and 5 illustrate this potential differential response of Southern pine to desorption effects where changes in stiffness and d_{pl} appear to be accentuated by variation in specific gravity. This differentiation is not obvious in P_{max} or P_{pl}.

It should be pointed out that a greater sample size is needed to make finer distinctions between capacity of desorbed and constant-condition joints. This is because of the high variation in the strength data. Nevertheless it is clear that joint stiffness was influenced negatively by desorption. This may have implications for the design of some metal clad building diaphragms.

5.4 Conclusion

The results of this study indicate that wood desorption effects may be more influential on the withdrawal stiffness than the withdrawal strength of tapping screw joints. Maximum withdrawal load was not significantly affected by wood desorption in either S-P-F or Southern pine. However, joints constructed from both species experienced significant reductions in stiffness following desorption. This reduction in stiffness resulted primarily from an increase in deflection at the proportional limit rather than a major decrease in P_{pl}. The increase in d_{pl} and the associated reduction in stiffness was also more pronounced in Southern pine than in S-P-F.

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Table 1: Results of desorption effects study.

Species	Moisture Condition [a]	Sample Size	Moisture Content (%)	Specific Gravity	Maximum Load (lbs.)	Load at Proportional Limit (lbs.)	Defl. at Proportional Limit (in.)	Joint Stiffness (lb/in)
S-P-F	Dry	10	8.2(1.6)	.42(17.5)	551(23.4)	412(23.5)	.0118(15.8)	35,100(22.9)
	Wet	10 [b]	10.5(1.2)	.41(15.6)	493(19.5)	348(24.9)	.0136(13.3)	25,300(15.8)
S. Pine	Dry	10	7.9(3.0)	.47(9.8)	851(17.7)	588(15.9)	.0109(16.9)	54,500(12.5)
	Wet	10	10.7(1.5)	.47(8.5)	724(17.2)	511(11.8)	.0158(16.2)	33,000(18.9)

a) Dry refers to specimens in which fasteners were inserted and withdrawn from wood at the moisture content presented in the table.

Wet refers to specimens in which fasteners were inserted into wet wood and withdrawn following drying of the wood to the moisture content presented in the table.

b) Proportional limit data was recorded for only 9 wet S-P-F specimens.

c) Values in ( ) are C.O.V. in percent.

Table 2: Results of two-factor analyses of variance.

Dependent Variable (species)	Independent Variables	F Statistic	Pr > F	Decision at .05 Level
P_{max} (both)	Condition	2.49	.1229	not sig.
	MC	0.64	.4305	not sig.
P_{pl} (both)	Condition	2.87	.0988	not sig.
	MC	0.30	.5886	not sig.
d_{pl} (both)	Condition	25.30	.0001	sig.
	MC	3.50	.0693	not sig.
Stiffness (both)	Condition	25.30	.0001	sig.
	MC	3.66	.0638	not sig.
P_{max} (SPF)	Condition	1.31	.2685	not sig.
	MC	1.36	.2600	not sig.
P_{pl} (SPF)	Condition	2.41	.1402	not sig.
	MC	2.02	.1743	not sig.
d_{pl} (SPF)	Condition	4.09	.0601	not sig.
	MC	0.14	.7177	not sig.
Stiffness (SPF)	Condition	11.59	.0036	sig.
	MC	2.08	.1686	not sig.
P_{max} (SYP)	Condition	4.02	.0612	not sig.
	MC	0.00	.9787	not sig.
P_{pl} (SYP)	Condition	4.48	.0493	sig.
	MC	0.00	.9538	not sig.
d_{pl} (SYP)	Condition	23.64	.0001	sig.
	MC	0.62	.4401	not sig.
Stiffness (SYP)	Condition	51.84	.0001	sig.
	MC	0.34	.5661	not sig.

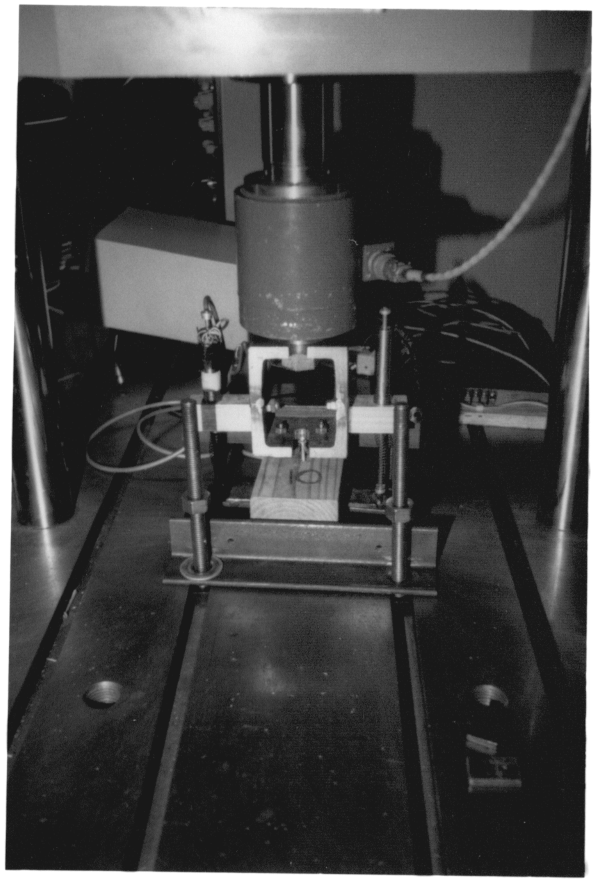

Figure 1: Test machine set-up used in desorption effects study.

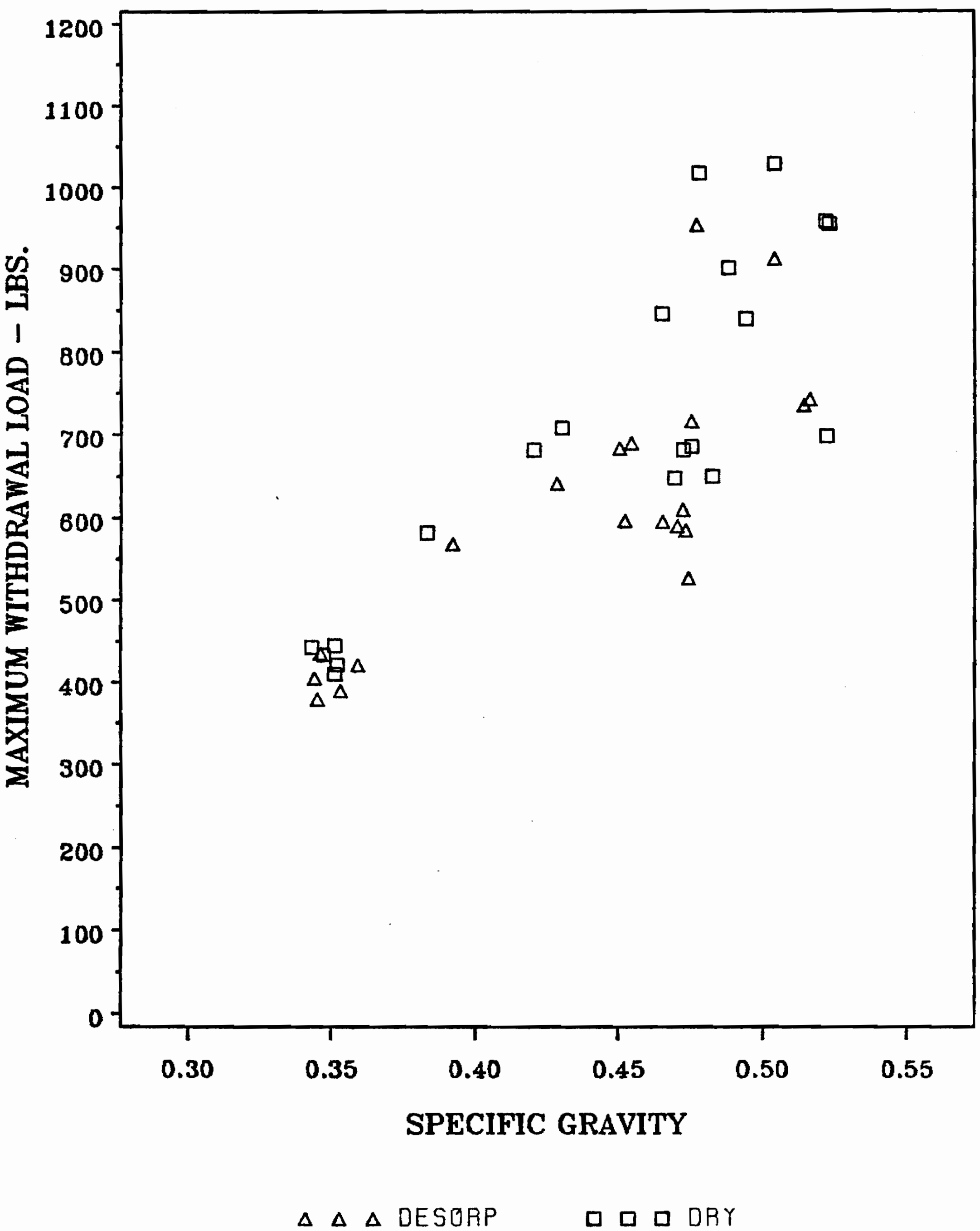

Figure 2: Results of desorption effects study - maximum withdrawal load.

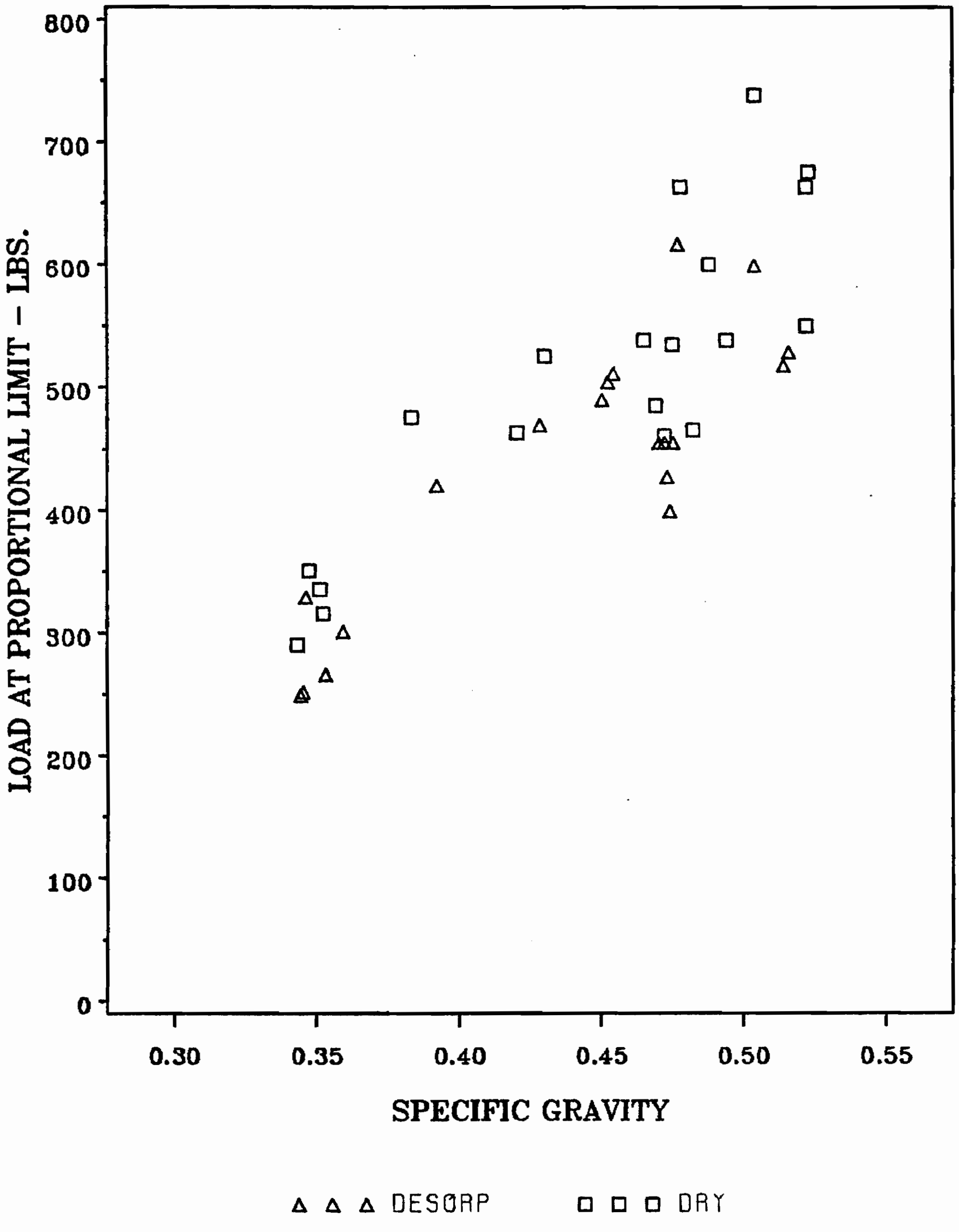

Figure 3: Results of desorption effects study - load at proportional limit.

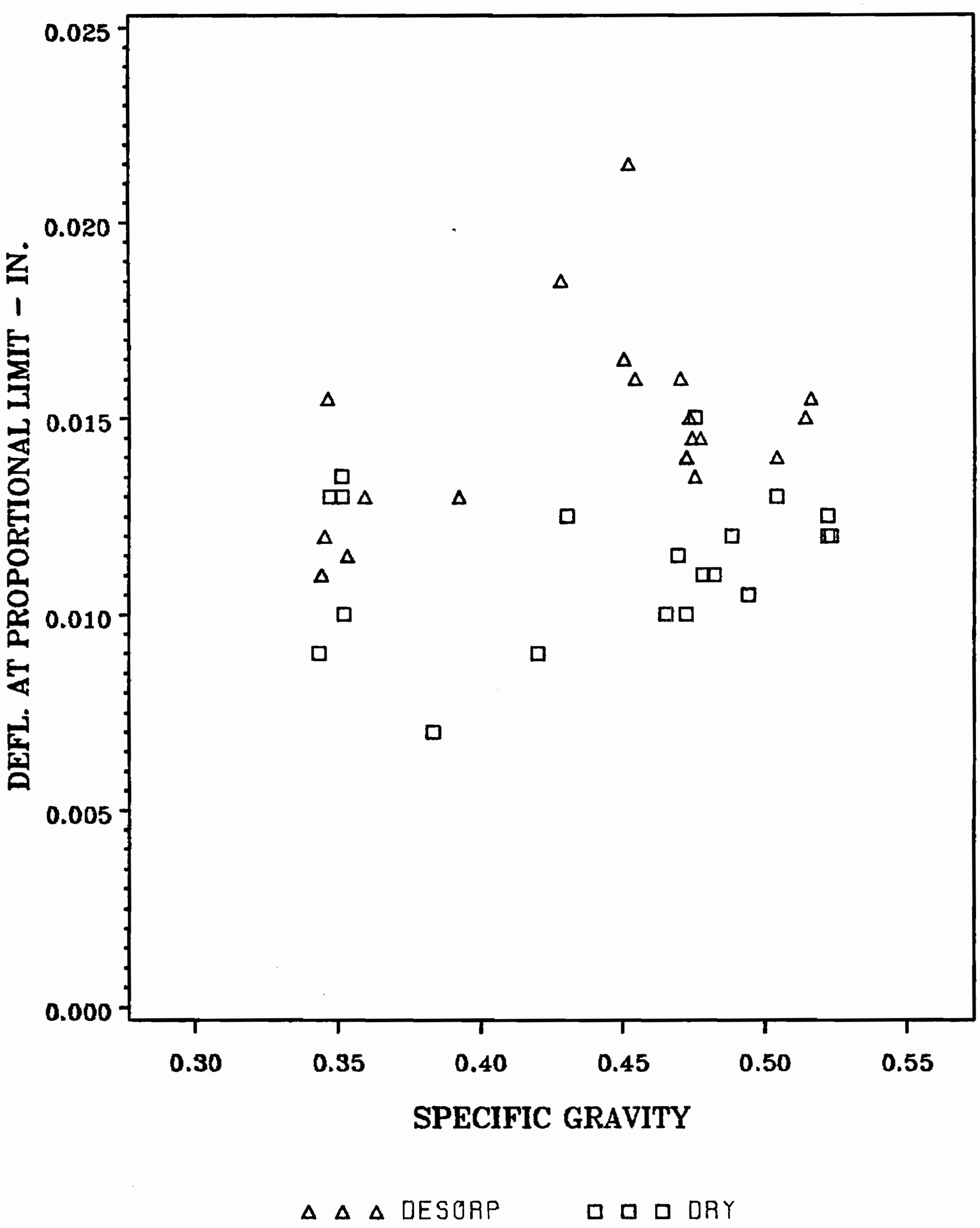

Figure 4: Results of desorption effects study - deflection at proportional limit.

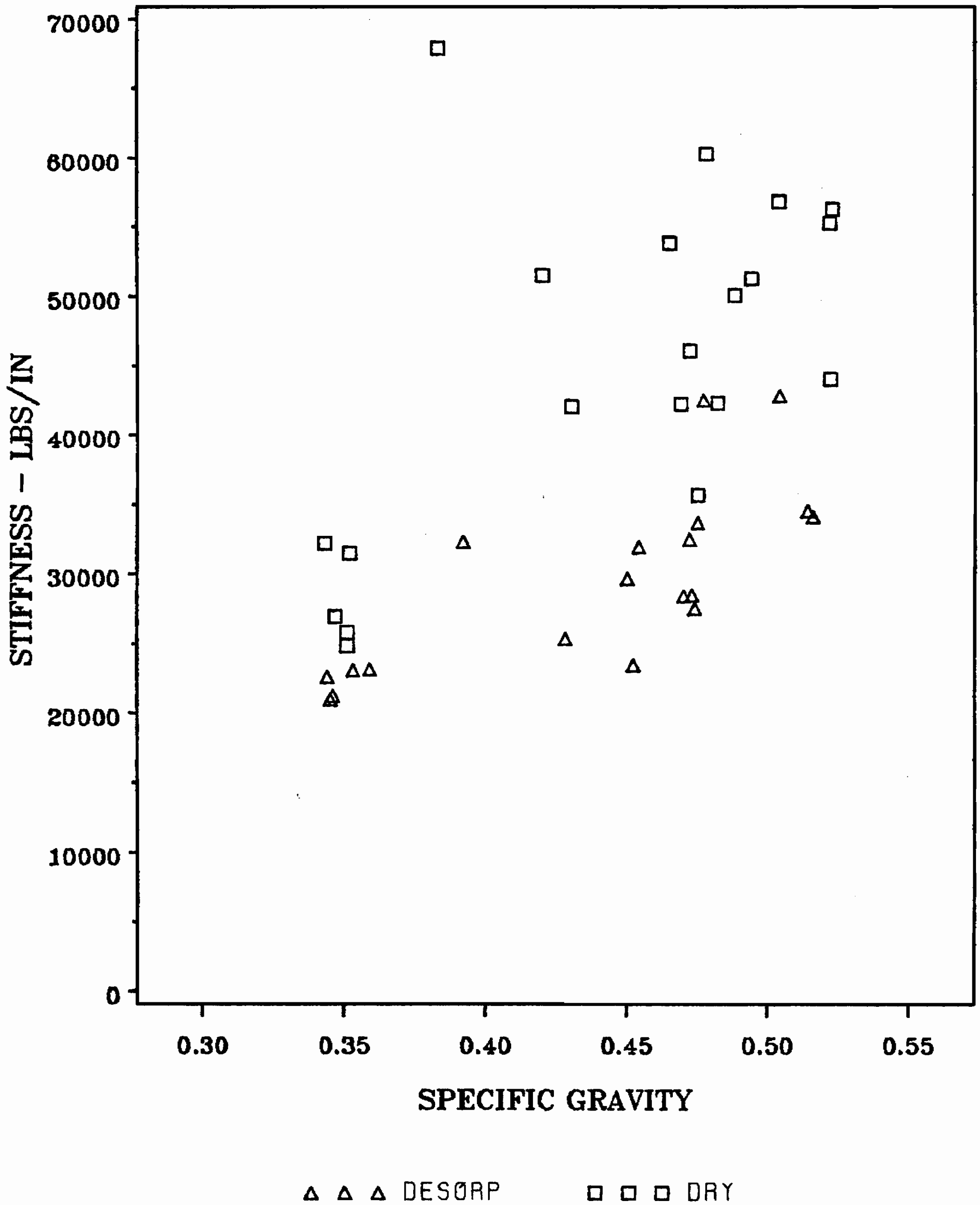

Figure 5: Results of desorption effects study - withdrawal stiffness.

6.0 Review of Literature

6.1 Threaded Fasteners - General

Wood screws are manufactured in lengths of 1/4 to 4 inches with diameters ranging from .060 to .372 inches. They have a continuous helical short-lead thread around a slightly tapered shank from which the thread projects (36). Wood screws are made of steel, brass, or other metals and alloys, or with specific finishes such as nickel, blued, chromium, or cadmium (38). Wood screws are generally used in light-duty applications and serve essentially the same purpose as nails. They are commonly employed in the manufacture of furniture, mobile homes, and manufactured housing. Drive screws, similar to wood screws but with a shallower thread, are commonly used to fasten dry wall and other sheathing material and to improve racking resistance in light industrial and commercial structures.

Lag screws are stouter than wood screws, have a square or hex head, a plain cylindrical shank section below the head, and a helically threaded cylindrical section extending to a pointed tip. Lag screws are generally made of hot-rolled steel, may be unfinished, electroplated, or galvanized, and are available in diameters 1/4" to 1 1/4" with lengths of 1 to 16" (28). The length of the threaded portion varies with the length of the screw and ranges from three-fourths inch for the 1 and 1 1/4 inch screws to half the length for all lengths greater than 10 inches (37).

Lag screws are commonly employed in fastening metal components to heavy timber members. Whereas bolts are generally applied in instances of lateral loading, lag screws can be used to transmit both shear and withdrawal loads. Lag screws have found considerable application in recent years due to the increase in construction involving glue laminated timber. Through bolts have been found to be inferior to lag screws in glulam applications due to their high cost and ineffectiveness related to the shrinkage of the wood (26). In addition, the use of through bolts has been found to considerably weaken the wood cross section (26). Lag screws are also commonly employed in situations where it is objectionable to have a nut on the surface or where difficulty in fastening a bolt or drilling a through hole is encountered.

Threaded dowels are headless, usually pointless, helically threaded wood, metal, or plastic rods primarily used to transmit shear forces between members. Threaded dowels are commonly used in the glulam industry in hidden connections, to provide between member continuity in bridge decking systems, and to fasten pile bents or stringers to caps in pile foundations. Plain and threaded wood dowels are also commonly used to join particleboard, medium-density fiberboard, and other wood composites in furniture construction. Wood dowels often replace drift pins or bolts in such applications due to their greater withdrawal resistance and superior performance in changing environmental conditions.

Screw tie spikes are cylindical and have a blunt point, a heavy cut thread, a taper near the top, and a washerlike flange surmounted by a square or oblong head (23). They are 6 1/2 to 8 inches in overall length and 13/16 to 15/16 inches in diameter. Screw tie spikes are used in the railroad industry to secure tie plates to wood ties. Screw tie spikes offer greater hold down strength and are less damaging to the wood ties than the typical, unthreaded track spike.

6.2 Environmental Influences

Cockrell (10) performed withdrawal testing of three sizes of screws from ten species of wood in which moisture content was taken into account. The results of this research showed that wood possessed, on an average, 50% greater withdrawal resistance at approximately 7% moisture content than at an MC above the fiber saturation point. The discrepancy between green and dry withdrawal strength was generally observed to increase with wood density.

McLain (28) states that a reduction factor must be applied to the allowable design loads of lag screw joints if the wood members are not dry at the time of joint construction, or if the moisture content of these members is expected to change in service. A table containing these reduction factors is presented in Appendix B.

Stern (37) studied the effects of wood desorption and dimensional change as related to the withdrawal resistance of plain and threaded nails. The following conclusions were recorded:

1) Plain shank nails lose as much as 80% of their initial withdrawal resistance during the drying of green lumber to a point below the fiber saturation point (FSP) while properly threaded nails retain or slightly increase withdrawal resistance under similar conditions.

2) Variations in moisture content occurring above FSP does not influence the withdrawal resistance of threaded nails providing the wood was above FSP when the nails were driven.

3) Threaded nails driven into lumber below FSP will display an increase in withdrawal resistance upon a further reduction in moisture content.

4) The greater the moisture content at the time threaded nails are driven, the greater the withdrawal resistance following drying below FSP, providing splitting does not occur at the nail locations.

Longworth and McMullin (27) performed an investigation of the effects of moisture content and dimensional change on the strength of bolted timber connections. They concluded that the strength of a bolted connection, as determined by its proportional limit, decreases with an increase in moisture content. The calculation of the proportional limit in these studies included an adjustment for the frictional resistance between the wood and bolt. This adjustment was warranted since at low moisture contents wood shrinkage caused a reduction or elimination of friction. Additional conclusions drawn from this study include:

1) Ultimate load of bolted connections is not influenced by moisture content.

2) Bolted connections seasoned after fabrication and retightened are as strong as connection fabricated in seasoned material.

3) There is considerably more slip at ultimate load in connections in green material, but slip at the proportional limit is not significantly affected by moisture content.

6.3 Withdrawal of Wood Screws

According to the Wood Handbook (38), the ultimate withdrawal strength of wood screws inserted into the side grain of seasoned wood may be expressed by:

$$P = 15{,}700\ G^{2}\ D\ L$$

where P is the ultimate withdrawal load in pounds, G is the specific gravity based on oven-dry weight and volume at 12% moisture content, D is shank diameter of the screw in inches, and L is length of penetration of the threaded portion of the screw in inches. This formula is applicable when lead holes are 70% that of the screw root diameter for softwoods and 90% for hardwoods. Withdrawal loads for screws inserted into the end grain should average 75% that of the side grain provided splitting during insertion is avoided.

The Fasteners Handbook (22) states that a .192 inch wood screw driven into "soft" wood perpendicular to the grain can resist 80 pounds of withdrawal force per inch of thread penetration. The withdrawal of a .192 inch round nail was stated as 25 pounds per inch of penetration. If large changes in moisture content occur after nailing, the nail figure must be divided by four. This publication recommends a lead hole size equal to 70% of the screw root diameter for soft woods, with 90% suggested for hard woods.

Cockrell (10) performed withdrawal testing of one-inch No. 6 (.138" shank diameter), No. 8 (.164"), and No. 10 (.190") flat-head wood screws from the following species of wood; sugar maple, beech, birch, red oak, black ash, red pine, red spruce, hemlock, white pine, and basswood. Screws were inserted into lead holes equal to 90% of the screw root diameter and were driven to a depth of three quarters of an inch. The influences of density, moisture content, and grain orientation were addressed in this study.

Cockrell found a strong, positively sloping linear relationship existing between specific gravity and withdrawal resistance. The one exception to this finding was that maple exhibited the greatest withdrawal strength of the ten species tested while birch and beech possessed slightly greater specific gravities. As previously stated, the woods tested displayed an average of 50% greater withdrawal resistance at 7% moisture content than at an MC above the fiber saturation point. The increase in withdrawal strength between green and 7% was greatest for maple (71%) and least for white pine (23%). The amount of increase due to moisture content reduction appeared to be related to wood density. Cockrell observed that diffuse porous hardwoods, especially those of high density, and conifers of irregular growth and uneven texture were stronger in screw holding ability from the tangential surface while ring porous hardwoods and regular growth, even texture, softwoods were stronger from their radial surface. Withdrawal resistance from the end grain of dry wood was found to average 65% that of the side grain for the species tested. Cockrell also states that a screw can be unscrewed and then reinserted into its original lead hole with no loss in withdrawal strength. Side hardness was the wood strength property Cockrell concluded to be most closely related to screw withdrawal resistance.

Fairchild (20) performed extensive withdrawal testing of over 10,000 screws from six species of wood including yellow poplar, cypress, sycamore, hard maple, white oak, and southern pine. Screws used in these tests ranged from No. 0 (.040" shank diameter) to No. 24 (.372") in lengths of 1/4 to 5 inches. The objective of these studies was to evaluate the influence exerted on withdrawal resistance by lead hole size, screw lubrication, cracks in the wood, and screw dimensions.

For soft woods, such as yellow poplar, Fairchild found that a lead hole equal to 50% of the screw root diameter gives the greatest withdrawal strength. The absence of a lead hole or the use of a lead hole diameter which approaches the screw diameter was found to greatly reduce withdrawal resistance in the less dense woods. For woods of medium density, such as sycamore, a lead hole size equal to 70% screw root diameter was recommended. The use of a large lead hole, or absence thereof, also caused a reduction in withdrawal strength though not as marked as found with the less dense woods. A lead hole equal to 90% was recommended for dense woods such as oak.

Variations in lead hole size were found to be less influential in denser woods.

From studies of screw dimensions, Fairchild concluded that withdrawal resistance increases with screw diameter up to a certain limit, beyond which an increase in diameter results in decreased withdrawal strength. This conclusion implies that a screw of given diameter may exhibit withdrawal strength equal to or even greater than one of same length but of larger diameter. Fairchild attributes this to the fact that the larger diameter screw has coarser threads and a larger point and thus may contain fewer effective threads. Fairchild also concluded that maximum withdrawal strength will be attained at a certain screw length, after which additional length will not significantly increase withdrawal resistance. Additional conclusions drawn from these studies include that lubrication of screws prior to insertion did not significantly affect withdrawal resistance, that the end grain withdrawal strength was 75% that of the side grain, and that even slight cracks in the wood in the vicinity of the screw resulted in a 10 to 25% reduction in withdrawal strength. While Fairchild did not address wood density in these studies, examination of the included data reveals a notable increase in withdrawal strength with increasing species density.

Eckelman (12-18) engaged in a further statistical analysis of the data generated by Fairchild. The results of this analysis indicated that withdrawal strenth was nonlinear with respect to screw diameter, screw length, and wood specific gravity, at the 95% confidence level. It also indicated that interaction effects were present between all three of these variables. Results of this analysis yielded the following predictor equation:

$$P_s = 10{,}762.3 \; D^{.976} \; L^{.886} \; G^{1.499}$$

with a coefficient of determination of .9407,

where P_s is the maximum withdrawal force from the side grain in pounds, D is the screw shank diameter in inches, L is depth of screw penetration in inches, and G is the specific gravity of the wood at its current moisture content. An examination of the accuracy of this equation determined it to over-predict the withdrawal strength of short screws. This result was expected since a portion of the tip of the screw does not contribute to withdrawal strength and thus the effective depth of penetration is proportionally less for shorter screws. An adjustment was subsequently made to the original regression model to account for this tip effect and the following equation was derived:

$$P_s = 17{,}699.5 \; D^{1.1678} \; (L - D)^{.7047} \; G^{1.5447}$$

with a coefficient of determination of .952.

An examination of the correlation of other material properties with withdrawal resistance suggested that shear strength may be a better indicator of withdrawal strength than specific gravity. Upon converting specific gravity to shear strength using Wood Handbook data, Eckelman performed a regression analysis which yielded:

$$P_s = 13.16\ D^{1.155}\ (L - D)^{.702}\ V^{.840}$$

with a coefficient of determination of .9272,

where V is the parallel to grain shear strength of the wood at its current moisture content in psi.

Eckelman also performed an additional analysis of the data generated by Cockrell from which the following equation was derived:

$$P_s = 3951.0\ D^{.525}\ G^{1.751}$$

with a coefficient of determination of .9412.

Two effects of interest appear in this analysis; first, there is an interaction between diameter and length such that diameter effects change as length changes. Second, the strength-diameter ratio decreases as the diameter of the screw is increased beyond a given size. It must be remembered that the Cockrell's data and its associated regression equation were generated from very limited testing while Fairchild evaluated approximately 100 different sizes of screws.

Eckelman also implemented the data generated by Fairchild and Cockrell in an analysis to determine whether shear strength or specific gravity is a better predictor of withdrawal resistance. In the mechanical testing, sugar maple was found to possess greater withdrawal strength than yellow birch or American beech. Calculations based on shear strength will predict this outcome while those based specific gravity produce the opposite result. Also, calculations based on specific gravity predict nearly identical withdrawal strength for cypress and yellow poplar while the experimental values differ by about 40%. Observations of this type led Eckelman to conclude that shear strength was a better withdrawal strength predictor than specific gravity. A final regression analysis of side grain data was conducted from which the following simplified predictor equation was derived:

$$P_s = 3.204 \quad D \quad (L - D)^{3/4} \quad V$$

with a coefficient of determination of .9335.

Analysis of the data of Fairchild and Cockrell was also used to develop the following predictor equation for end grain withdrawal strength:

$$P_e = 8.752 \quad D^{7/4} \quad (L - D)^{3/4} \quad V$$

with a coefficient of determination of .9023, where P_e is the maximum withdrawal force from the end grain in pounds. Also included with the above results were equations for predicting withdrawal strength of screws from plywood and particleboard.

Eckelman later performed a study to determine whether the equation presented above for side grain withdrawal resistance would acurately predict withdrawal strength of Type A sheet metal-type screws. A series of mechanical tests of sheet metal screws were completed and the experimental data were then compared to withdrawal figures predicted by the equation. This comparison determined that the equation under-predicted the withdrawal strength of the sheet metal screws by an average of 4%.

Eckelman observed in later studies that screws driven completely through the wood member exhibited an average of 16% greater withdrawal strength over those that were simply embedded to the full length of the screw. This result reinforces the tip effect addressed earlier.

In withdrawal testing of threaded metal inserts in wood, Eckelman observed that withdrawal strength is essentially independent of the insert diameter. The withdrawal resistance was found to be strongly related to both the depth of embedment of the insert and the shear strength of the wood.

Osborn (31) states that the withdrawal resistance of a plain shank fastener is a function of the surface contact area of the fastener in the nailing member. Surface area is also a factor in the withdrawal resistance of helically threaded nails though some shear perpendicular to the grain occurs in the lumber when the nail is withdrawn. A high quality thread not only increases the surface contact area but it also increases the area over which shear will occur. This results in greater withdrawal resistance. A poor quality thread increases surface contact area only slightly over that of a plain shank fastener, resulting in withdrawal resistance superior to a plain shank nail fastener but inferior to a high quality threaded nail.

Carroll (9) studied the relationship between withdrawal strength and driving torque exerted during the insertion of screws into plywood and particleboard. The results of this study indicated that an excessive torque applied during screw insertion will reduce withdrawal strength from both materials. This reduction in withdrawal resistance is the result of the stripping of the "threads" which are formed in the wood substrate during screw insertion. Stripping of these threads was found to occur when a screw was tightened to approximately one turn past the point where it was flush with the board. The stripping torque of screws set in softwood plywood was not greatly different from the stripping torque of the same screws set in 40 lb./cu. ft. particleboard.

Whittington and Walters (39) evaluated the withdrawal resistance of three types of wood screws from soft maple and particleboard. The types of screws studied were the common wood screw (13 threads per inch), the improved wood screw (16 t.p.i.), and the self-tapping screw (12 t.p.i.). All screws were 10 gage with two inch length. These studies found that the average withdrawal force from the soft maple lumber was 35% greater than that of the particleboard. The improved and self-tapping screws displayed significantly greater withdrawal resistance than the common wood screws in the maple lumber tests. No significant difference in withdrawal force was observed between the three screw types in the particleboard tests. Other than threads per inch, no additional description of thread geometry was presented in this publication.

Wilkinson and Laatsch (40) studied the lateral and withdrawal resistance of tapping screws in red oak, Douglas-fir, and red-wood. For comparison, withdrawal tests were also conducted using standard wood screws. Tapping screws differ from standard wood screws in that they are threaded over their entire length whereas only two-thirds of a wood screw shank is threaded. Tapping screws also tend to have more consistency in dimensions and finer, sharper threads than wood screws.

The results of this study determined that the optimum lead hole size for tapping screws was approximately 60% of the screw root diameter, with maximum withdrawal loads occurring at lead hole sizes from 50 to 70%. The resistance of tapping screws to withdrawal was also found to increase linearly with depth of penetration. At a depth of approximately 1 1/8 inches, the ultimate tensile strength of the screw was developed in red oak. Differences in withdrawal strength from the radial and tangential surfaces were not apparent, while end grain withdrawal strength was only 71 to 86% that of side grain. In the comparison to wood screws, tapping screws exhibited 6 to 15 percent greater withdrawal strength. This comparison is based on No. 10 screws inserted into lead holes equal to 60% of screw root diameter.

The National Design Specifications (29) include a table of allowable withdrawal design loads for wood screws in seasoned wood (Appendix F). Maximum allowable withdrawal load from this table is based on screw diameter, wood specific gravity, and the depth of penetration of the threaded portion of the screw. Values from this table apply when the following conditions are met; 1) the allowable tensile strength of the screw shall not be exceeded, and 2) the wood screw shall not be loaded in withdrawal from the end grain. These specifications also require lead holes to be bored to a diameter of 70 or 90% of the screw root diameter depending on wood species. Soap or other lubricant may be used to facilitate screw insertion. The NDS specifications do not address the issue of screw quality and are said to apply to any wood screw of sufficient strength to cause failure in the wood rather than the metal.

The American Society of Mechanical Engineers (4) published the American National Standard B18.6.1-1981, Wood Screws (Inch Series).

This standard covers the complete general and dimensional data for the various types of slotted and recessed head wood screws recognized as "American National Standard." This document states that wood screws shall have a coarse pitch spaced threads and a gimlet point. The threads may be either cut or rolled (cold formed) at manufacturer's option, unless desiginated by the purchaser. For cut threads, the length of the thread shall be equivalent to approximately two-thirds of the nominal length of the screw. In the case of rolled threads, the length of the thread shall be equivalent to at least four times the basic screw diameter or two-thirds of the nominal screw length, whichever is greater. Screws of nominal lengths which are too short to accommodate the minimum thread length shall have threads extending as close to the underside of the head as practicable. A table of thread dimensions and body diameters for wood screws is included in Appendix M.

6.4 Withdrawal of Lag Screws

The Wood Handbook (38) supplies the following equation to calculate the maximum load in direct withdrawal for lag screws inserted into the side grain of seasoned wood:

$$P_s = 8100 \; G^{3/2} \; D^{3/4} \; L$$

where P_s is the maximum withdrawal load from the side grain in pounds, G is the specific gravity of the wood based on ovendry weight and volume at 12% moisture content, D is the screw shank diameter in inches, and L is the length of penetration of the threaded portion of the screw in inches. The resistance of lag screws to withdrawal from the end grain is about 75% that of side grain resistance. Allowable withdrawal resistance has been calculated as one-fifth of the maximum load to account for test data variability and load duration effects. Lag screws, like wood screws require properly drilled lead holes to optimize withdrawal strength. The lead hole for the shank portion of the lag screw should be of equal diameter to the screw shank. The diameter of the lead hole for the threaded portion is a function of the wood density; for lightweight woods, such as the cedars and white pine, a lead hole of 40 to 70% of the shank diameter is recommended. For medium density woods, such as Douglas fir and southern pine, 65 to 75% is recommended. For dense woods such as the oaks, lead hole diameter should be 65 to 85% of the lag screw shank diameter (Appendix A). In determining withdrawal resistance, the allowable tensile strength of the lag screw root section should not be exceeded. Penetration of the threaded portion of the screw to a depth of seven times the shank diameter for dense species and 10 to 12 times the shank diameter in less dense species will develop approximately the ultimate lag screw tensile strength.

Newlin and Gahagan (30) reported the results of substantial lag screw withdrawal testing conducted at the U.S. Forest Products Laboratory. These tests involved seven diameters of screws ranging from 1/4" to 1" and five wood species including Douglas fir, Southern pine, redwood, white pine, and white oak. The influence of lead hole size, depth of screw penetration, screw diameter, and wood specific gravity were evaluated.

In lead hole studies using 5/8" screws, a lead hole diameter equal to 70% the screw shank diameter was associated with the greatest withdrawal loads for white oak. For Douglas fir and southern pine, the most effective lead hole size was 65% screw shank diameter. The most effective lead hole sizes for redwood and white pine were 60 and 50%, respectively. In addition to wood density, optimum lead hole size was also found to be related to lag screw diameter. The above figures should be slightly increased for screws larger than 5/8", and decreased for smaller screws.

Studies to evaluate the influence of the depth of penetration determined this parameter to be directly related to screw withdrawal strength. One observation of interest was that during the testing of 5/8" screws with a four inch depth of penetration in white oak, the ultimate strength of the screw was exceeded. Wood specific gravity and lag screw diameter were also found to be closely related to withdrawal resistance. The combined influence of these variables were expressed in the formula:

$$P_s = 7500 \quad D^{3/4} \quad G^{3/2}$$

where P_s is the maximum withdrawal load per inch of penetration from the wood side grain in lbs., D is lag screw shank diameter in inches, and G is the wood specific gravity based on oven-dry weight and volume when oven dry. The figure 7500 is a species constant related to the five implemented in these tests. This expression is very similar to that presented in the Wood Handbook. The change in the constant is related to the use of volume at 12% moisture content in the determination of specific gravity. The authors also concluded that no single wood strength property is as good a criterion for predicting lag screw withdrawal resistance as the specific gravity of the wood.

Hoadley (24) conducted an investigation to compare the performance of lag screws and threaded nails when used in a typical structural joint in southern pine glulam. Companion tests were also conducted in which individual fasteners were subject to direct shear and withdrawal loading. Lag screws evaluated in this research were four inches in length with a shank diameter of .303 inches. Low carbon steel, medium high carbon steel, and hardened steel nails of four inch length and .300 inch shank diameter were also tested.

Results of these tests showed that lag screws possessed twice the withdrawal resistance of the nails. In the tests of lateral resistance, the lag screws were equal to the low and medium carbon steel nails while the hardened nails were clearly superior. The hardened nails also out performed the other fasteners in the joint tests due to their greater bending resistance. Although the load was applied laterally in the joint tests, the bending of the fastener shanks introduced a component of withdrawal. For this reason the lags screws displayed a slight advantage over the low and medium carbon steel nails in the joint tests. Fastener bending with subsequent withdrawal appeared to be the predominant mode of failure for the lag screws and regular nails, while the greater yield strength of the hardened steel nails allowed for greater shear loads with eventual failure due to crushing under the fastener.

The National Design Specifications (29) present a table of allowable withdrawal design values for lag screws inserted into the side grain of seasoned wood that will remain dry in service (Appendix G). The allowable design loads from this table are based on lag screw diameter, wood specific gravity, and the depth of penentration of the threaded portion of the screw. In determining the withdrawal resistance, the maximum tensile strength of the fastener is not to be exceeded. Penetration of the threaded portion of the screw to a depth of 7 x screw diameter for Group 1 species (Appendix A), 8 x diameter for Group 2 species, 10 x diameter for Group 3 species, and 11 x diameter for Group 4 species will develop approximately the tensile strength of common lag screws in axial withdrawal. Lead hole specifications (Appendix A) are similar to that presented in the Wood Handbook. The withdrawal loading of lag screws inserted into the wood end grain is not recommended, though when unavoidable, design values should not exceed 75% of that for side grain withdrawal. The NDS design values and provisions are stated only to apply to lag screws of material conforming to ASTM Standard A307, Low-Carbon Steel Externally and Internally Threaded Standard Fasteners.

According to the National Design Specifications, design values for lag screws loaded at an angle to the grain other than 0 or 90 degrees shall be determined through the application of the Hankinson formula:

$$N = \frac{P\ Q}{P\ \sin^2 \Theta + Q\ \sin^2 \Theta}$$

where N is the allowable lateral load, in pounds, of one screw at an angle Θ to the direction of the grain, P is the allowable load, in pounds, for parallel to grain loading, Q is the allowable load, in pounds, for perpendicular to grain loading

The American Society of Mechanical Engineers (3) published the American National Standard B18.2.1-1981, Square and Hex Bolts and Screws Inch Series. This standard is intended to cover the complete general data for the various types of inch series square and hex bolts and screws recognized as American National Standard. This document states the minimum thread length shall be equal to one-half of the nominal screw length plus 0.50", or 6.00", whichever is shorter. Screws too short for the formula thread length shall be threaded as close to the head or shoulder as practicable. Threads on lag screws are to conform with Appendix J. Dimensions for square and hex lag screws are included in Appendixes K and L.

6.5 Withdrawal of Dowel-Type Fasteners

Eckelman (18) performed testing to evaluate the relationship between the withdrawal strength of unthreaded dowels from the side grain of wood and the shear strengths parallel to the grain of the wood members. Testing involved fifteen hardwood species commonly used in furniture applications. The shear strengths of these woods ranged from 990 to 2330 psi. All dowels used in the testing were machined from sugar maple. The following empirical expression was developed:

$$F = .834 \; D \; L^{.89} \; (.95 \, V_1 - V_2) \; A \; B$$

where F is the withdrawal strength of the dowel in pounds, D is the dowel diameter in inches, L is the length of imbedment of the dowel in inches, V_1 is the shear strength parallel to the grain of the wood in which the dowel is embedded, V_2 is the shear strength of the wood from which the dowel is constructed, A is an adhesive factor, and B is a dowel-hole clearence factor. This relationship implies that an interaction exists between the dowel pin and the members connected so that withdrawal is related to the average of the shear strengths of the components.

Test results were found to agree closely with values predicted from the above equation with one exception - observed values for sweetgum averaged 15 percent higher than predicted. In all other cases, the averages of the observed values differed by no more than 7 percent from predicted values. The author concluded that the withdrawal strength/shear strength relationship, as defined in the above expression, is valid over a wide range of shear strength values when dowels are constructed from a high shear strength species.

Eckelman (12) also performed studies aimed at developing design formulas to predict the ultimate strength of axially loaded, unthreaded-dowel joints. To begin the study, single pin dowel joints were tested in tension to determine ultimate strength and force-displacement relationships. Two types of specimens were tested: end-to-end grain and end-to-side grain. Specimens were constructed from several species and were bonded with a urea-formaldehyde resin.

The data generated in the testing was used to derive the following expression:

$$F = .45 \quad D \quad L^{.89} \quad (.95\, V_1 + V_2) \quad A \quad B$$

where F is the ultimate tensile strength of end-to-side grain joints in pounds, D is the dowel diameter in inches, L is the length of the dowel in inches, V_1 is the parallel to grain shear strength of the side grain member in psi, V_2 is the parallel to grain shear strength of the dowel in psi, A is a correction factor for adhesives other than urea-formaldehyde, and B is a correction factor for dowel-hole clearence.

For end-to-end grain joints the following expression should be used:

$$F = .45 \quad D \quad L^{.89} \quad (V_1 + V_2) \quad A \quad B$$

where F is the ultimate tensile strength of end-to-end grain joints in pounds. Of interest is the fact that the end-to-side grain specimens exhibited only 92% the strength of the end-to-end grain specimens. This is reflected in the above equations.

Analysis of the test data also lead to the conclusion that the strength of the dowel joint does not continue to increase indefinitely with dowel length. Maximum strength in each diameter class was obtained with dowels that had a length/diameter ratio of 8:1 or 9:1. As illustrated, the ultimate strength within the effective length was determined to be proportional to dowel length to the .89 power and directly proportional to dowel diameter. Joint displacement was inversely proportional to the square of the dowel diameter and was found to decrease as dowel length increased to about two inches. No decrease in displacement was noted beyond this length. Joint displacement was determined to remain linear with applied load until failure. Also, displacements in end-to-end grain joints were only 70.7% that of end-to-side grain joints.

6.6 Threaded Fasteners in Metal Connections

The Mechanical Design and Systems Handbook (33) states that fasteners possessing a fine thread are suited to joints subject to vibration. Fine threads offer greater strength at the thread root, may be used to provide fine adjustments, and are recommended for tapping hard nuts. The strength of fine threads is greater for sizes one inch and under, increasing as the size decreases. Coarse threads are desirable where the hole is tapped in weaker materials such as cast iron, aluminum, and magnesium alloys. Coarse threads are stronger in sizes one inch and over and are especially suited for tapping brittle materials.

For a bolt subjected to an axial load, the stress area can be calculated as:

$$A_s = \pi D_s^2 / 4 = F_e / \sigma_d$$

where: $D_s = (D_m + D_r) / 2$

A_s = Bolt stress area, in^2.,
D_s = Diameter corresponding to stress area, in.,
D_m = Major or nominal diameter, in.,
D_r = Root diameter, in.,
$\sigma_d = \sigma_y/N$ = Allowable nominal bolt stress, psi,
σ_y = Yield stress of bolt material, psi,
N = Factor of safety,
F_e = External bolt load, lbs.

From these relationships it can be noted that the strength contributed by the threads is small in comparison with that of the material properties. (Values for σ_y are included in Appendix

H). The allowable stress should be lowered for screws having a nominal diameter of 1/2 inch or smaller. The allowable nominal stress should also be reduced when a number of fasteners work together in a joint and an uneven load distribution among the fasteners is possible.

The allowble axial load per bolt may be reasonably expressed by:

$$F_e = C\ (A_s)^{1.42} = \sigma_w\ A_s$$

where:

F_e = External load per bolt, lbs.

C = Empirical constant

$\sigma_w = C\ (A_s)^{.42}$ = Working stress, psi

A_s = Bolt stress area, in^2.

This equation is applicable for bolts 3/4 inch in diameter and over made of steel containing 0.08 to 0.25 percent carbon. For bolts two inches and smaller, C = 5000 for carbon steel of ultimate strength 60,000 psi, increasing in proportion to the ultimate strength to 15,000 for alloy-steel bolts. For bolts two inches and larger, C is increased by 40 percent.

The influence of torsional stress on the bolt strength may be taken into account by decreasing the allowable nominal strength σ_d by 25 to 30 percent. The equation for stress area can then be modified:

$$2\ A_s = \pi\ D_s^2 / 4 = F_i / .75\ \sigma_d$$

where

F_i = Initial tightening force induced by the nut, lbs.

In metal-to-metal surfaces, the tightening force should be somewhat greater than F_e. If $F_e > F_i$, then F_e should be used in the above equation.

Black (7) presents the following expressions for the calculation of the thickness of a nut required for equality of the tensile strength of the bolt and thread shear strength. The strength of the bolt in tension can be calculated from:

$$F_t = (\pi\ D_r^2\ \sigma_t) / 4$$

where

F_t = Strength of the bolt in tension, lbs.,
D_r = Bolt minor diameter. in.,
σ_t = Yield stress of the bolt material in tension, psi.

The shear strength of the the bolt threads can be determined from:

$$F_s = (7\ \pi\ D_r\ H\ \sigma_s) / 8$$

where

F_s = Strength of the bolt threads in shear, lbs.,
D_r = Bolt minor diameter, in.,
σ_s = Yield stress of the bolt material in shear, psi,
H = Thickness of nut, in.

The thickness of the nut can be calculated by equating F_s and F_t. The required nut thickness is then:

$$H = .46\ D_m$$

where D_m is the major diameter of the bolt in inches.

This solution is based on the following assumptions:

a) Each turn of the threads supports an equal share of the load

b) Stress concentration is neglected

c) $D_r = .80\ D_m$ (National Coarse threads)

d) For the bolt steel, yield stress in shear σ_s is equal to one-half the yield stress in tension σ_t

Therefore, for National Coarse threads, the threads will be as strong in failure by shear as the bolt in tension if the thickness of the nut is .46 times the bolt major diameter. National Standard nuts are approximately 7/8 D_m in height and thus standard threads will not fail by shear.

However, the assumption that each thread supports an equal share of the load is incorrect because of the elongation of the bolt and the compression of the nut under load. Photoelastic tests indicate that the concentration of stress at the root of a National Coarse thread is large as illustrated by a static stress concentration factor equal to 5.62. This stress concentration is generally not as serious in bolts made of ductile material and subject to static loads. In dynamic loading, the concentration of stress has been found to reduce the endurance limit of National Coarse threaded bolts by fatigue stress-concentration factors equal to 2.84 for medium-carbon steel and 3.85 for SAE 2320 nickel steel heat-treated. These factors were determined for repeated tension loading of bolts threaded with no special care toward relieving stress concentration at the juncture of the thread and the shank of the bolt.

Shigley (34) presents the following expression to calculate the average shear stress in the threads of a square-threaded bolt in an axially loaded joint:

$$V_b = 2 F / \pi D_r H$$

where:

V_b = Average bolt-thread shear stress, psi,
F = Axial force on bolt, lbs.,
D_r = Bolt minor diameter, in.,
H = Thickness of nut, in.

The following equation is presented for the calculation of the average shear stress in the threads of the nut:

$$V_n = 2 F / \pi D_m H$$

where

V_n = Average nut-thread shear stress, psi,
D_m = Bolt major diameter, in.

The average bearing stress in the threads can be determined from:

$$\sigma_b = \frac{-4 p F}{\pi H (D_m^2 - D_r^2)}$$

where:

σ_b = Average bearing stress in the threads, psi,
p = Thread pitch, in.

All stresses determined from the above equations are average stresses since F is assumed to be uniformly distributed. Since this assumption is not always true, rather large safety factors, $n > 2$, should be used when these equations are applied in design purposes.

Faires (21) presents the following formula to calculate the design tensile stress of an axially loaded threaded fastener joint:

$$\sigma_t = \sigma_y / 6 A_s^{1/2}$$

where:

σ_t = Design tensile stress, psi,
σ_y = Yield strength of the fastener material in tension, psi,
A_s = Stress area, in^2.

The stress area is that area corresponding to a diameter which is equal to the average of the pitch and minor diameters of class 3 tolerances. The pitch diameter is characterized by a radius which intersects the screw thread at a height of one-half the pitch (Appendix I) (35).

The allowable tensile force in lbs., F_e, can be expressed as:

$$F_e = \sigma_t A_s$$

Upon substuting this relationship into the original equation, the following expression can be derived:

$$F_e = \sigma_y \ A_s^{3/2} \ / \ 6$$

This equation should give satisfactory results for axially loaded fasteners smaller than 1 3/4 inches in diameter where the bolts or screws are well tightened. For larger diameter fasteners, substitute the simple stress equation:

$$F_e = \sigma_t \ A_s$$

where: $\sigma_t = \sigma_y \ / \ 4$.

According to Spotts (35) a stress concentration is present in a bolt at the point where the load is transferred from the nut to the adjoining member. This stress concentration arises because the force in the bolt must shift outwardly to the region near the boundary as it is transferred from the bolt to the nut. Under ideal conditions, the tension in the bolt and the compression in the nut should be reduced uniformly starting from full load at the first contact between the bolt and nut. However, tension increases the pitch in the bolt and compression decreases the pitch in the nut so that correct compliance is not maintained in the loaded joint. The major portion of the load is transferred at the first pair of contacting threads and a large stress concentration is present in this region.

Although the stress concentration is somewhat relieved by the bending of the threads and expansion of the nut, most bolt failures occur at this point.

Three-dimensional photoelastic analyses have indicated a stress concentration factor of 3.85 at the root of the first engaged thread. A number of methods have been used to increase the flexibility of the nut and thus increase the area over which the transfer of force occurs. Tension nuts, containing a reduced cross-sectional area at the lip, have been successfully used to relieve stress concentrations. The reduced cross-sectional area of the lip deforms in tension with the bolt allowing the load to spread over a greater number of threads. Another method is to cut the thread of the nut on a very small taper, thus reducing the contact area of the first few threads. Since these threads will bend and carry less load, additional threads will come into service. Nuts manufactured from a material with an MOE smaller than that of the bolt have also been successful in spreading the load over a larger area. The material in the nut, however, must have a sufficient reserve of ductility to deform without breaking.

It has been estimated that bolt failures are distributed as follows: 15% under the head, 20% in the bolt at the end of the threads, and 65% in the bolt at the nut face.

6.7 Lateral Loading

Newlin and Gahagan (30) performed a lateral loading study involving four wood species and ten sizes of lag screws. Each test specimen was constructed from either a cleat and a block of the same wood species, or a metal cleat and a wooden block. The block is the section of the joint receiving the point of the screw while the cleat is the section immediately under the screw head. All loading was performed parallel to the wood grain.

The authors found that proportional limit loads varied by approximately the 3/4 power of specific gravity regardless of the screw shank diameter. Proportional limit loads were also found to vary by the square of the screw shank diameter. The resistance of the joint to lateral displacement in parallel to grain loading was also found to vary as the square root of the tensile yield stress of the fastener steel.

An increase in proportional limit load was also observed with an increase in the ratio of cleat thickness to shank diameter up to approximately 7 to 1, beyond which little additional increase occurred. For a ratio of cleat thickness to shank diameter of 3.5 to 1, a depth of penetration in the block of approximately 7 times the screw shank diameter for harder woods, and 11 or 12 times the shank diameter for softer species, is required to develop maximum joint strength. Greater cleat thickness-shank diameter ratios require greater depths of penetration while depths can be reduced for lower ratios. Proportional limit loads were found not to be affected until the depth of penetration is reduced to less than 5 times the shank diameter of the screw. Proportional limit loads were also found to be 25% greater using 1/2" metal cleats than for a joint of similar species and geometry constructed with a wooden cleat.

The authors concluded by presenting the following expression to calculate proportional limit loads for laterally loaded lag screw joints:

$$P = K \ D^2$$

where P is the proportional limit lateral load in pounds, D is the lag screw shank diameter in inches, and K is a species group constant. This equation is applicable for joints constructed with cleat thickness-shank diameter ratio of 3.5 or greater and a depth of penetration of 7 to 11 times the screw shank diameter depending on wood species. Furthermore, joints should be constructed of wood of 15% or less moisture content with lag screws possessing a tensile yield stress of at least 45,000 psi.

Johansen (25) developed expressions to predict the load-bearing capacity of single- and double-shear dowel joints. These expressions were based on the assumption that the dowel subjected to bending stresses and the wood subjected to embedding stresses yield by ideally plastic deformation. This assumption approximates the behavior of a steel dowel in bending though its fit to the behavior of wood in embedding is less exact (Au). Though the expressions developed by Johansen are regarded as simplistic, the assumptions and expressions were verifed in experimental tests. Johansen's yield theory was later successfully modified and applied to describing the behavior of nailed and bolted wood joints, and several European countries have adopted design codes for laterally loaded nailed joints based upon it.

Larsen and Reestrup (26) developed a theoretical model of the behavior of laterally loaded lag screw joints. In developing this model the authors established two theories describing joint failure; the flexible screw and the rigid screw theories. The flexible screw theory applies when the forces acting on the fastener are large enough that yielding of the screw occurs. Six failure patterns were developed in conjunction with the flexible screw theory: the screw yields in only the block in either the screw shank, threaded portion, or a the junction of the two sections; or yielding of the screw occurs in both the cleat and block. Yielding in the block can occur in either the shank, threaded portion, or junction while yielding in the cleat is assumed to always occur in the fastener shank. The rigid screw theory applies when the forces acting on the screw remain less than the fastener yield stress. Joint failure within the rigid screw theory occurs only by crushing of the wood in the cleat.

Mathematical expressions to predict ultimate load based on joint geometry, screw size and material properties, and wood embedding strength were derived for each of the seven failure patterns. An experimental test program was then used to successfully verify the accuracy of the expressions.

6.8 Combined Loading

The National Design Specifications (29) present the following criterion for combined lateral and withdrawal loading of lag screws:

$$P_\Theta \sin \Theta \leq W \qquad \text{and} \qquad P_\Theta \cos \Theta \leq L$$

where:

P_Θ is the allowable load, in pounds, for a lag screw inserted at an angle Θ to the direction of the grain, W is the allowable withdrawal load in pounds, and L is the allowable lateral load in pounds. This criterion assumes no interaction between the lateral and withdrawal force components.

The Timber Construction Manual (1) mandates use of the Hankinson formula in the design of lag screw joints subject to combined loading:

$$P_\Theta = \frac{W \; L}{W \; \sin^2 \Theta + L \; \cos^2 \Theta}$$

where the variables are the same as above.

DeBonis and Bodig (11) studied the effects of combined lateral and withdrawal loading on nailed joints. To accomplish this study, an apparatus (Appendix D) and test method were designed which enabled the authors to apply axial loads, lateral loads, and controlled ratios of axial to lateral loads to the nailed joints. Three species of wood were selected for testing: Engelman spruce, Douglas-fir, and red oak. Eight penney (.131 ") nails were inserted into 2" x 6" x 1.5" specimens to depth of 6, 10, and 14 times the nail diameter. Withdrawal or combined forces were then applied at one of eight angles to the fastener axis.

Several approaches were taken to describe the effects of combined withdrawal and lateral loading. Hankinson's formula proved to be a poor equation to describe the combined load effect, producing errors in excess of 500% in some cases. A second approach implemented a stress interaction equation:

$$\frac{P_{\Theta} \sin \Theta}{W} + \frac{P_{\Theta} \cos \Theta}{L} = 1.0$$

where the P_{Θ} is the maximum load attained at angle Θ, W is the ultimate withdrawal load, and L is the ultimate lateral load. (The restraint of 1.0 was removed to determine the value of the sum on the left produced by the substitution of actual experimemtal values for P, W, and L).

The interaction equation provided an accurate method for describing ultimate load with errors below 15%. The authors recommended this equation for design of nailed joints subjected to combined lateral and withdrawal loading.

Ehlbeck (19) conducted a similar combined loading study of threaded nail joints constructed with European whitewood. The following interaction equation provided the best results in describing joint behavior:

$$\left[\frac{P_\Theta \sin \Theta}{W}\right]^2 + \left[\frac{P_\Theta \cos \Theta}{L}\right]^2 = 1.0$$

where the variables are the same as above.

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Appendix A: Species grouping and recommended lead hole size for American woods (from National Design Specification).

Grouping for lag screw, drift bolt, nail, spike, wood screw and metal plate connector loads

Group	Species of wood	Specific gravity** (G)
Group I	Ash, Commercial White	0.62
	Beech	0.68
	Birch, Sweet & Yellow	0.66
	Hickory & Pecan	0.75
	Maple, Black & Sugar	0.66
	Oak, Red & White	0.67
Group II	Douglas Fir - Larch***	0.51
	Southern Pine	0.55
	Sweetgum & Tupelo	0.54
	Virginia Pine - Pond Pine	0.54
Group III	California Redwood	0.42
	Douglas Fir, South	0.48
	Eastern Hemlock	0.43
	Eastern Hemlock - Tamarack***	0.45
	Eastern Softwoods	0.42
	Eastern Spruce	0.43
	Hem - Fir***	0.42
	Lodgepole Pine	0.44
	Mountain Hemlock	0.47
	Mountain Hemlock - Hem Fir	0.44
	Northern Aspen	0.42
	Northern Pine	0.46
	Ponderosa Pine***	0.49
	Ponderosa Pine-Sugar Pine	0.42
	Red Pine****	0.42
	Sitka Spruce	0.43
	Southern Cypress	0.48
	Spruce-Pine-Fir	0.42
	Western Hemlock	0.48
	Yellow Poplar	0.46
Group IV	Aspen	0.40
	Balsam Fir	0.38
	Black Cottonwood	0.33
	California Redwood, Open grain	0.37
	Coast Sitka Spruce	0.39
	Coast Species	0.39
	Cottonwood, Eastern	0.41
	Eastern White Pine***	0.38
	Eastern Woods	0.38
	Engelmann Spruce - Alpine Fir	0.36
	Idaho White Pine	0.40
	Northern Species	0.35
	Northern White Cedar	0.31
	West Coast Woods (Mixed Species)	0.35
	Western Cedars***	0.35
	Western White Pine	0.40
	White Woods (Western Woods)	0.35

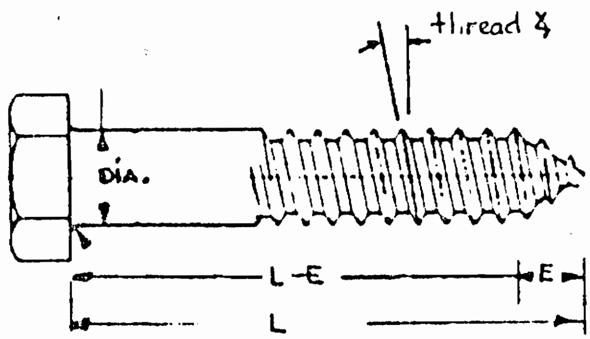
Appendix B: Adjustment factors for allowable design loads for lag screw joints subjected to moisture content variations (from McLain).

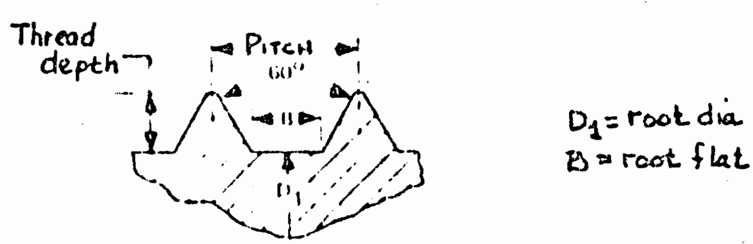
Condition of Wood[1]		
At Fabrication	In Service	Factor
Dry	Dry	1.0
Dry or Wet	Exposed to Weather	0.75
Dry or Wet	Wet	0.67
Partially Seasoned or Wet	Dry[2]	
One Fastener or two or more in a single row or several rows with separate splice plates		1.0
All other cases		0.4

[1]Dry wood has moisture content of 19% or less. Wet wood has moisture content of approximately 30%. Partially seasoned is in between.

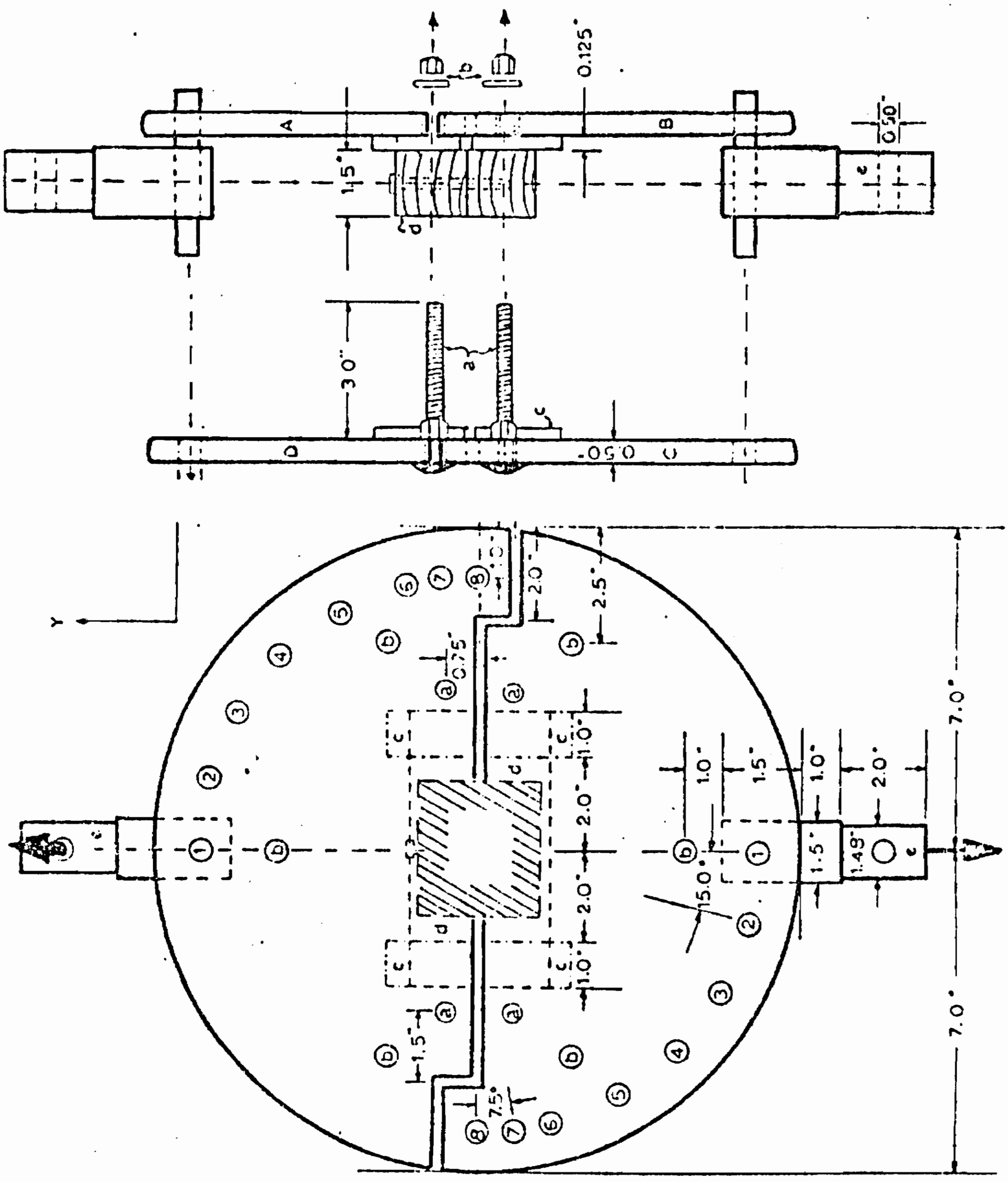
[2]If partially seasoned at fabrication, but dry before design load is applied, use interpolated reductions.

Appendix C: Fastener thread characteristics.

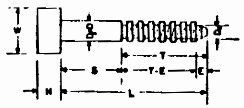




Appendix D: Apparatus designed for combined loading of nailed joints (from DeBonis and Bodig).



Appendix E: Typical dimensions of standard lag screws for wood (from ASME).



[All dimensions in inches]

D = Nominal diameter.
$Ds = D$ = Diameter of shank.
Dr = Diameter at root of thread.
W = Width of head across flats.
H = Height of head.
L = Nominal length.
S = Length of shank.
T = Length of thread.
E = Length of tapered tip.
N = Number of threads per inch.

Nominal length L in inches*	Item	Dimensions of lag screw with nominal diameter D of–												
		3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1-1/8	1-1/4
All lengths	$Ds=D$	0.190	0.250	0.3125	0.375	0.4375	0.500	0.5625	0.625	0.750	0.875	1.000	1.125	1.250
	Dr	0.120	0.173	0.227	0.265	0.328	0.371	0.435	0.471	0.579	0.683	0.780	0.887	1.012
	E	5/32	3/16	1/4	1/4	9/32	5/16	3/8	3/8	7/16	1/2	9/16	5/8	3/4
	H	9/64	11/64	13/64	1/4	19/64	21/64	3/8	27/64	1/2	19/32	21/32	3/4	27/32
	W	9/32	3/8	1/2	9/16	5/8	3/4	7/8	15/16	1-1/8	1-5/16	1-1/2	1-11/16	1-7/8
	N	11	10	9	7	7	6	6	5	4-1/2	4	3-1/2	3-1/4	3-1/4
1........	S	1/4	1/4	1/4	1/4	1/4	1/4	......	......	......	......	......	......	......
	T	3/4	3/4	3/4	3/4	3/4	3/4	......	......	......	......	......	......	......
	T-E	19/32	9/16	1/2	1/2	15/32	7/16	......	......	......	......	......	......	......
1½......	S	3/8	3/8	3/8	3/8	3/8	3/8	......	......	......	......	......	......	......
	T	1-1/8	1-1/8	1-1/8	1-1/8	1-1/8	1-1/8	......	......	......	......	......	......	......
	T-E	31/32	15/16	7/8	7/8	27/32	13/16	......	......	......	......	......	......	......
2.......	S	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	......	......	......	......	......
	T	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	......	......	......	......	......
	T-E	1-11/32	1-5/16	1-1/4	1-1/4	17/32	1-3/16	1-1/8	1-1/8	......	......	......	......	......
2½......	S	1	1	7/8	7/8	3/4	3/4	3/4	3/4	......	......	......	......	......
	T	1-1/2	1-1/2	1-5/8	1-5/8	1-3/4	1-3/4	1-3/4	1-3/4	......	......	......	......	......
	T-E	1-11/16	1-5/16	1-3/8	1-3/8	1-15/32	1-7/16	1-3/8	1-3/8	......	......	......	......	......
3.......	S	1	1	1	1	1	1	1	1	1	1	1	......	......
	T	2	2	2	2	2	2	2	2	2	2	2	......	......
	T-E	1-27/32	1-13/16	1-3/4	1-3/4	1-23/32	1-11/16	1-5/8	1-5/8	1-9/16	1-1/2	1-7/16	......	......
4.......	S	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2	1-1/2
	T	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2
	T-E	2-11/32	2-5/16	2-1/4	2-1/4	2-7/32	2-3/16	2-1/8	2-1/8	2-1/16	2	1-15/16	1-7/8	1-3/4
5.......	S	2	2	2	2	2	2	2	2	2	2	2	2	2
	T	3	3	3	3	3	3	3	3	3	3	3	3	3
	T-E	2-27/32	2-13/16	2-3/4	2-3/4	2-23/32	2-11/16	2-5/8	2-5/8	2-9/16	2-1/2	2-7/16	2-3/8	2-1/4
6.......	S	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2	2-1/2
	T	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2
	T-E	3-11/32	3-5/16	3-1/4	3-1/4	3-7/32	3-3/16	3-1/8	3-1/8	3-1/16	3	2-15/16	2-7/8	2-3/4
7.......	S	3	3	3	3	3	3	3	3	3	3	3	3	3
	T	4	4	4	4	4	4	4	4	4	4	4	4	4
	T-E	3-27/32	3-13/16	3-3/4	3-3/4	3-23/32	3-11/16	3-5/8	3-5/8	3-9/16	3-1/2	3-7/16	3-3/8	3-1/4
8.......	S	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2	3-1/2
	T	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2	4-1/2
	T-E	4-11/16	4-5/16	4-1/4	4-1/4	4-7/32	4-3/16	4-1/8	4-1/8	4-1/16	4	3-15/16	3-7/8	3-3/4
9.......	S	4	4	4	4	4	4	4	4	4	4	4	4	4
	T	5	5	5	5	5	5	5	5	5	5	5	5	5
	T-E	4-27/32	4-13/16	4-3/4	4-3/4	4-23/32	4-11/16	4-5/8	4-5/8	4-9/16	4-1/2	4-7/16	4-3/8	4-1/4
10.....	S	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4	4-3/4
	T	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4	5-1/4
	T-E	5-3/32	5-1/16	5	5	4-31/32	4-15/16	4-7/8	4-7/8	4-13/16	4-3/4	4-11/16	4-5/8	4-1/2
11.....	S	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2
	T	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2	5-1/2
	T-E	5-11/32	5-9/32	5-1/4	5-1/4	5-7/32	5-3/16	5-1/8	5-1/8	5-1/16	5	4-15/16	4-7/8	4-3/4
12.....	S	6	6	6	6	6	6	6	6	6	6	6	6	6
	T	6	6	6	6	6	6	6	6	6	6	6	6	6
	T-E	5-27/32	5-13/16	5-3/4	5-3/4	5-23/32	5-11/16	5-5/8	5-5/8	5-9/16	5-1/2	5-7/16	5-3/8	5-1/4

Appendix F: Withdrawal design loads for wood screws (from National Design Spec.).

Specific gravity G	Screw size											
	g =	6	7	8	9	10	12	14	16	18	20	24
	D =	0.138	0.151	0.164	0.177	0.190	0.216	0.242	0.268	0.294	0.320	0.372
0.75		220	241	262	283	304	345	387	428	470	511	594
0.68		181	198	215	232	250	284	318	352	386	420	489
0.67		176	193	209	226	242	275	309	342	375	408	474
0.66		171	187	203	219	235	267	299	332	364	396	460
0.62		151	165	179	193	207	236	264	293	321	349	406
0.55		119	130	141	152	163	186	208	230	253	275	320
0.54		114	125	136	147	157	179	200	222	243	265	308
0.51		102	112	121	131	140	160	179	198	217	236	275
0.49		94	103	112	121	130	147	165	183	200	218	254
0.48		90	99	107	116	124	141	158	175	192	209	243
0.47		87	95	103	111	119	136	152	168	184	201	233
0.46		83	91	99	106	114	130	145	161	177	192	224
0.45		79	87	94	102	109	124	139	154	169	184	214
0.44		76	83	90	97	104	119	133	147	162	176	205
0.43		72	79	86	93	100	113	127	141	154	168	195
0.42		69	76	82	89	95	108	121	134	147	160	186
0.41		66	72	78	85	91	103	116	128	140	153	178
0.40		63	69	75	80	86	98	110	122	134	145	169
0.39		60	65	71	76	82	93	105	116	127	138	161
0.38		57	62	67	73	78	89	99	110	121	131	153
0.37		54	59	64	69	74	84	94	104	114	124	145
0.36		51	56	60	65	70	80	89	99	108	118	137
0.35		48	53	57	62	66	75	84	93	102	111	129
0.33		43	47	51	55	59	67	75	83	91	99	115
0.31		38	41	45	48	52	59	66	73	80	87	102

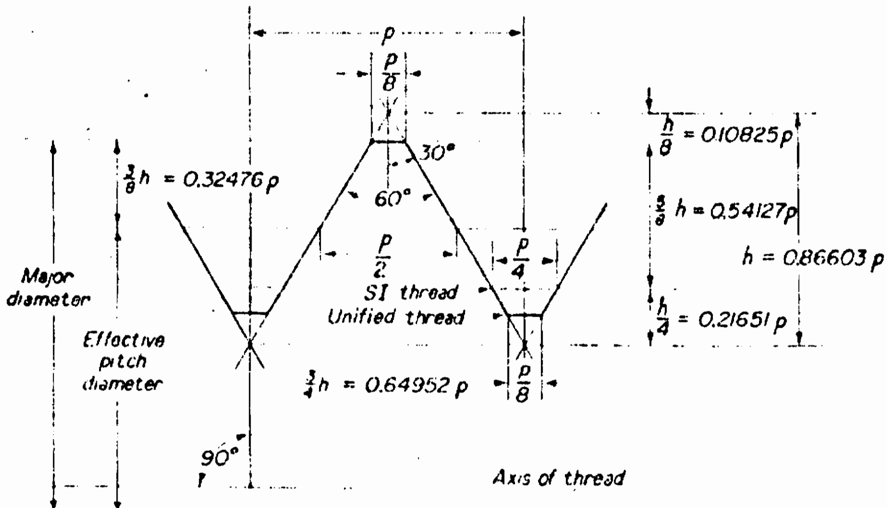
Appendix G: Withdrawal design loads for lag screws (from National Design Spec.).

Specific gravity *G*	Lag screw diameter *D*											
	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1-1/8	1-1/4
	0.250	0.3125	0.375	0.4375	0.500	0.5625	0.625	0.750	0.875	1.000	1.125	1.250
0.75	413	489	560	629	695	759	822	942	1058	1169	1277	1382
0.68	357	422	484	543	600	656	709	813	913	1009	1103	1193
0.67	349	413	473	531	587	641	694	796	893	987	1078	1167
0.66	341	403	463	519	574	627	678	778	873	965	1054	1141
0.62	311	367	421	473	523	571	618	708	795	879	960	1039
0.55	260	307	352	395	437	477	516	592	664	734	802	868
0.54	253	299	342	384	425	464	502	576	646	714	780	844
0.51	232	274	314	353	390	426	461	528	593	656	716	775
0.49	218	258	296	332	367	401	434	498	559	617	674	730
0.48	212	250	287	322	356	389	421	482	542	599	654	708
0.47	205	242	278	312	345	377	408	467	525	580	634	686
0.46	199	235	269	302	334	365	395	453	508	562	613	664
0.45	192	227	260	292	323	353	382	438	492	543	594	642
0.44	186	220	252	283	312	341	369	423	475	525	574	621
0.43	179	212	243	273	302	330	357	409	459	508	554	600
0.42	173	205	235	264	291	318	344	395	443	490	535	579
0.41	167	198	226	254	281	307	332	381	428	473	516	559
0.40	161	190	218	245	271	296	320	367	412	455	497	538
0.39	155	183	210	236	261	285	308	353	397	438	479	518
0.38	149	176	202	227	251	274	296	340	381	422	461	498
0.37	143	169	194	218	241	263	285	326	367	405	443	479
0.36	137	163	186	209	231	253	273	313	352	389	425	460
0.35	132	156	179	200	222	242	262	300	337	373	407	441
0.33	121	143	164	184	203	222	240	275	309	341	373	403
0.31	110	130	149	167	185	202	218	250	281	311	339	367

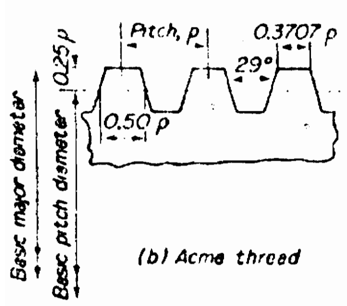
Appendix H: Properties of ASTM and SAE grades of steel bolts and cap screws.

SAE grade	Steel designation	Bolt size diam, in.	Yield strength σ_y, psi	Ultimate tensile strength (min) σ_u, psi	Hardness (Brinell)	Comments
0	Low-carbon	All sizes	....	....	....	General use
1	ASTM A307 low-carbon commercial	All sizes	....	55,000	207 max	Cold or hot heading
2	Low-carbon bright finish 0.28C, 0.04P, max 0.05S	Up to ½ incl. Over ½ to ¾ incl.	55,000 52,000	69,000 64,000	241 max	Cold-headed product. Automotive applications
3	Medium-carbon 0.28 to 0.55C, max 0.04P, max 0.05S. May be aged to 700°F to suit	Up to ½ incl. Over ½ to ¾ incl.	85,000 80,000	110,000 100,000	207-269	Cold worked by heading or roll threading. Used for high fatigue strength
4	Commercial	Over ¼ to 1½ incl.	28,000	55,000	207 max	
5	ASTM A325 medium-carbon 0.28 to 0.55C, max 0.04P, max 0.05S, quenched and tempered at 800°F	Up to ¾ incl. Over ¾ to 1 incl. Over 1 up 1½ incl.	85,000 78,000 74,000	120,000 115,000 105,000	241-302 235-302 223-285	Used where high preload is necessary
6	Special medium-carbon, oil-quenched and tempered at 800°F min	Up to ⅝ incl. Over ⅝ to ¾ incl.	110,000 105,000	140,000 133,000	285-331 269-331	Used where higher strength than grade 5 is required
7	Medium-carbon fine-grain alloy 0.28 to 0.55C, max 0.04P, max 0.05S, oil-quenched and tempered at 800°F min	Up to 1½ incl.	105,000	133,000	269-331	Roll-threaded after heat-treatment for improved fatigue strength
8	Same as grade 7 but higher strength	Up to 1½ incl.	120,000	150,000	302-352	

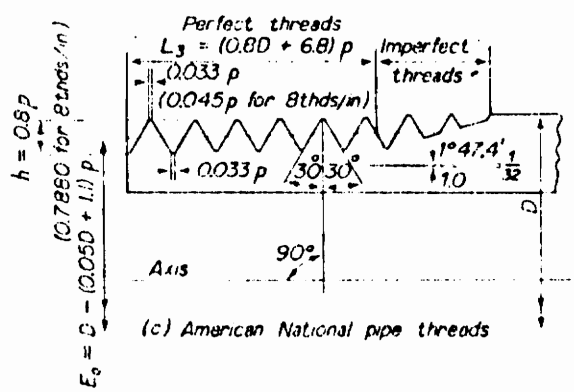
Appendix I: Basic forms of screw threads.



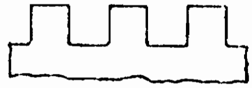
(a) Unified, American National and SI threads



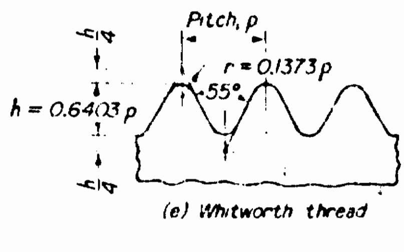
(b) Acme thread



(c) American National pipe threads



(d) Square thread



(e) Whitworth thread

Appendix J: Dimensions of lag screw threads.

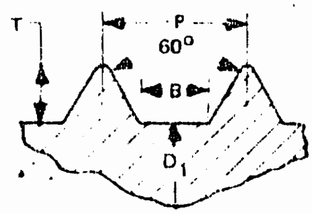


FIG. 5 DETAIL OF THREADS

TABLE 9A DIMENSIONS OF LAG SCREW THREADS

Nominal Size or Basic Product Dia		Threads per Inch	P Pitch	B Flat at Root	T Depth of Thread	D_1 Root Dia
No. 10	0.1900	11	0.091	0.039	0.035	0.120
1/4	0.2500	10	0.100	0.043	0.039	0.173
5/16	0.3125	9	0.111	0.048	0.043	0.227
3/8	0.3750	7	0.143	0.062	0.055	0.265
7/16	0.4375	7	0.143	0.062	0.055	0.328
1/2	0.5000	6	0.167	0.072	0.064	0.371
5/8	0.6250	5	0.200	0.086	0.077	0.471
3/4	0.7500	4 1/2	0.222	0.096	0.085	0.579
7/8	0.8750	4	0.250	0.108	0.096	0.683
1	1.0000	3 1/2	0.286	0.123	0.110	0.780
1 1/8	1.1250	3 1/4	0.308	0.133	0.119	0.887
1 1/4	1.2500	3 1/4	0.308	0.133	0.119	1.012

GENERAL NOTE:

Thread formulas are as follows:

$$\text{Pitch} = \frac{1}{\text{No. of threads per inch}}$$

Flat at root = pitch × 0.4305

Depth of single thread = pitch × 0.385

Appendix K: Dimensions of hex lag screws (from ASME).

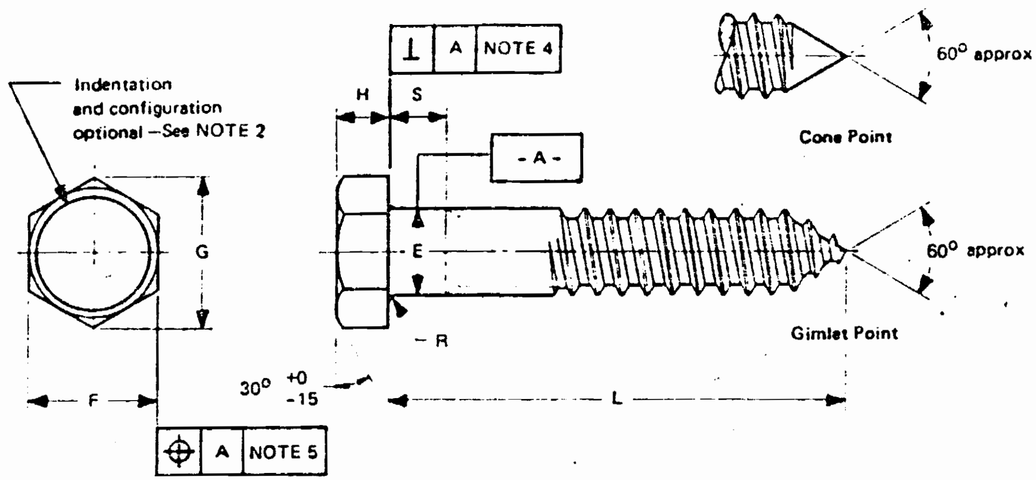


TABLE 9 DIMENSIONS OF HEX LAG SCREWS

Nominal Size or Basic Product Dia (12)		E Body or Shoulder Dia (6), (7)		F Width Across Flats (3)			G Width Across Corners		H Height			S Shoulder Length (7)	R Radius of Fillet	
		Max	Min	Basic	Max	Min	Max	Min	Basic	Max	Min	Min	Max	Min
No. 10	0.1900	0.199	0.178	9/32	0.281	0.271	0.323	0.309	1/8	0.140	0.110	0.094	0.03	0.01
1/4	0.2500	0.260	0.237	7/16	0.438	0.425	0.505	0.484	11/64	0.188	0.150	0.094	0.03	0.01
5/16	0.3125	0.324	0.298	1/2	0.500	0.484	0.577	0.552	7/32	0.235	0.195	0.125	0.03	0.01
3/8	0.3750	0.388	0.360	9/16	0.562	0.544	0.650	0.620	1/4	0.268	0.226	0.125	0.03	0.01
7/16	0.4375	0.452	0.421	5/8	0.625	0.603	0.722	0.687	19/64	0.316	0.272	0.156	0.03	0.01
1/2	0.5000	0.515	0.482	3/4	0.750	0.725	0.866	0.826	11/32	0.364	0.302	0.156	0.03	0.01
5/8	0.6250	0.642	0.605	1 5/16	0.938	0.906	1.083	1.033	27/64	0.444	0.378	0.312	0.06	0.02
3/4	0.7500	0.768	0.729	1 1/8	1.125	1.088	1.299	1.240	1/2	0.524	0.455	0.375	0.06	0.02
7/8	0.8750	0.895	0.852	1 5/16	1.312	1.269	1.516	1.447	37/64	0.604	0.531	0.375	0.06	0.02
1	1.0000	1.022	0.976	1 1/2	1.500	1.450	1.732	1.653	43/64	0.700	0.591	0.625	0.09	0.03
1 1/8	1.1250	1.149	1.098	1 11/16	1.688	1.631	1.949	1.859	3/4	0.780	0.658	0.625	0.09	0.03
1 1/4	1.2500	1.277	1.223	1 7/8	1.875	1.812	2.165	2.066	27/32	0.876	0.749	0.625	0.09	0.03

Appendix L: Dimensions of square lag screws (from ASME)

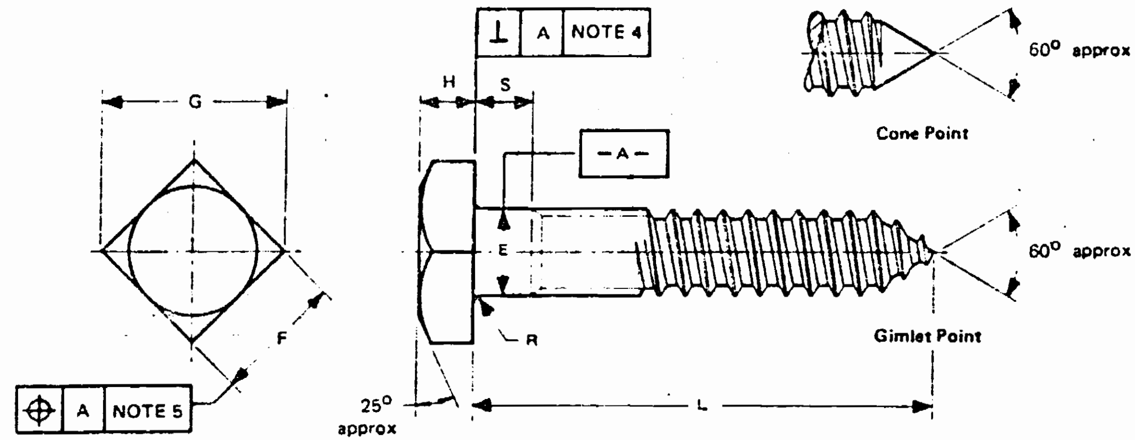
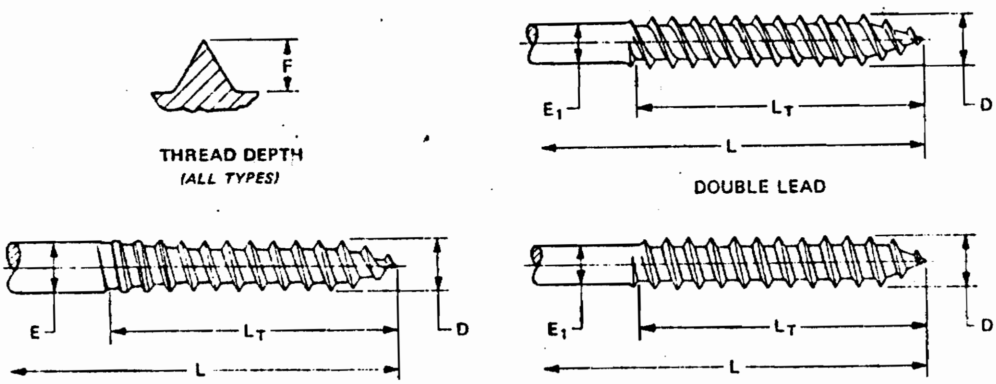


TABLE 8 DIMENSIONS OF SQUARE LAG SCREWS

Nominal Size or Basic Product Dia (12)		E		F			G		H			S	R	
		Body or Shoulder Dia (6), (7)		Width Across Flats (3)			Width Across Corners		Height			Shoulder Length (7)	Radius of Fillet	
		Max	Min	Basic	Max	Min	Max	Min	Basic	Max	Min	Min	Max	Min
No. 10	0.1900	0.199	0.178	9/32	0.281	0.271	0.398	0.372	1/8	0.140	0.110	0.094	0.03	0.01
1/4	0.2500	0.260	0.237	3/8	0.375	0.362	0.530	0.498	11/64	0.188	0.156	0.094	0.03	0.01
5/16	0.3125	0.324	0.298	1/2	0.500	0.484	0.707	0.665	13/64	0.220	0.186	0.125	0.03	0.01
3/8	0.3750	0.388	0.360	9/16	0.562	0.544	0.795	0.747	1/4	0.268	0.232	0.125	0.03	0.01
7/16	0.4375	0.452	0.421	5/8	0.625	0.603	0.884	0.828	19/64	0.316	0.278	0.156	0.03	0.01
1/2	0.5000	0.515	0.482	3/4	0.750	0.725	1.061	0.995	21/64	0.348	0.308	0.156	0.03	0.01
5/8	0.6250	0.642	0.605	1 5/16	0.938	0.906	1.326	1.244	27/64	0.444	0.400	0.312	0.06	0.02
3/4	0.7500	0.768	0.729	1 1/8	1.125	1.088	1.591	1.494	1/2	0.524	0.476	0.375	0.06	0.02
7/8	0.8750	0.895	0.852	1 5/16	1.312	1.269	1.856	1.742	19/32	0.620	0.568	0.375	0.06	0.02
1	1.0000	1.022	0.976	1 1/2	1.500	1.450	2.121	1.991	21/32	0.684	0.628	0.625	0.09	0.03
1 1/8	1.1250	1.149	1.098	1 11/16	1.688	1.631	2.386	2.239	3/4	0.780	0.720	0.625	0.09	0.03
1 1/4	1.2500	1.277	1.223	1 7/8	1.875	1.812	2.652	2.489	27/32	0.876	0.812	0.625	0.09	0.03

Appendix M: Dimensions of threads and body diameters for wood screws (from ASME).



SINGLE LEAD

CUT THREAD TYPE[3]

SINGLE LEAD

ROLLED THREAD TYPES[4]

Table 1 Dimensions of Threads and Body Diameters for Wood Screws

Nominal Size[1] or Basic Screw Diameter	Threads per Inch[2]	D Thread Major Diameter		E Body Diameter (Cut Thread)		E1 Body Diameter (Rolled Thread)		F Thread Depth
		Max	Min	Max	Min	Max	Min	Min
0 0.060	32	0.064	0.053	0.064	0.053	0.055	0.044	0.010
1 0.073	28	0.077	0.066	0.077	0.066	0.066	0.055	0.010
2 0.086	26	0.090	0.079	0.090	0.079	0.075	0.064	0.010
3 0.099	24	0.103	0.092	0.103	0.092	0.086	0.075	0.014
4 0.112	22	0.116	0.105	0.116	0.105	0.095	0.084	0.016
5 0.125	20	0.129	0.118	0.129	0.118	0.107	0.096	0.018
6 0.138	18	0.142	0.131	0.142	0.131	0.118	0.107	0.020
7 0.151	16	0.155	0.144	0.155	0.144	0.127	0.116	0.022
8 0.164	15	0.168	0.157	0.168	0.157	0.136	0.125	0.023
9 0.177	14	0.181	0.170	0.181	0.170	0.147	0.136	0.026
10 0.190	13	0.194	0.183	0.194	0.183	0.157	0.146	0.030
12 0.216	11	0.220	0.209	0.220	0.209	0.176	0.165	0.031
14 0.242	10	0.246	0.235	0.246	0.235	0.201	0.190	0.035
16 0.268	9	0.272	0.261	0.272	0.261	0.214	0.203	0.038
18 0.294	8	0.298	0.287	0.298	0.287	0.237	0.226	0.042
20 0.320	8	0.324	0.313	0.324	0.313	0.260	0.249	0.046
24 0.372	7	0.376	0.365	0.376	0.365	0.303	0.292	0.050

1 Where specifying nominal size in decimals, zeros preceding the decimal shall be omitted.

2 The maximum permissible variation in the number of threads per inch shall be plus or minus 10 per cent of that which is tabulated for the respective screw size.

3 Cut thread type screws are usually supplied with single lead threads for all screw lengths.

4 Rolled thread type screws may be supplied with either single lead or double lead threads at the option of the manufacturer, however, single lead threads are preferred where the nominal length of the screw is shorter than 4 times the basic screw diameter. Points shall be sharp, however, no extrusion of excess material beyond the apex of point resulting from thread rolling shall be permissible.

Vita

The author was born in Abington, Pa. on May 30, 1963. He grew up in Worcester, Pa. where he graduated from Methacton High School in June 1981.

College study was begun at the University of Maine where a Bachelor of Science in Wood Science and Technology was completed in December 1985. Graduate study was then pursued at Virginia Polytechnic Institute and State University. A Master of Science in Wood Science and Forest Products was completed in December 1988.