TRAFFIC POLE DESIGN

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ABSTRACT

An existing traffic pole was modeled and redesigned based on existing dimensions and functions. A MathCAD program was developed to assist solving stresses, deflection and stability. The redesigned pole meets all required design criterion of stress, deflection and stability requirements. With the exception of the modifiable welded joints, the new design has a factor of safety of 6. Deflection in the vertical direction are less than one foot at the arm tip. Production drawings and material specifications are provided.

INTRODUCTION

Throughout the world, the automobile has become the preferred method of transportation. As the the number of vehicles increases, sophisticated ordered methods of directing traffic at intersections are needed. Traffic lights direct the majority of this traffic, especially in high density transit situations such as cities and major highways. Many of these traffic lights are dependent upon the structure of a cantilevered pole. The cantilevered traffic pole has advantages of requiring only a relatively small ground surface area while allowing for many lanes of traffic. Traffic poles are a transparent part of the transportation system, yet they must work in the most extreme and harsh conditions without failure while also meeting the strict budget, appearance and safety conditions of the funding public. Designing a traffic pole involves simultaneously specifying a geometry, material and loading condition that doesn't exceed failure, force, deflection or stress limitations. This project is based on an existing traffic pole and will retain most of the dimensions while using a different material. No analysis will be performed below the shaft base junction. This design project calculates the response of the pole to static as well as severe ice and wind conditions. The new design needs to meet most design requirements and physical properties.

THEORY AND PROBLEM

Engineering design of mechanical structures consists of specifying the physical properties of a design that meet the criterion of stress, deflection and stability. Within the entire structure, stresses, deflection and stability must be such that the structure remains in a safe state.

The structure must be capable of adaptation to environmental conditions. In accordance with NCHRP Report 411, we analyized 3 environmental cases: Static, Wind and Ice loading.

Static loading considers the traffic pole with no external loading conditions. The existing traffic pole was observed in this condition. The design must at least support its own weight and the loading of the luminaires.

Wind loading considers the steady state response to wind flowing around the arm, shaft and luminaires of the design. The wind loading case is calculated at 90 miles per hour which is a typical 50 year wind maximum for the majority of the United States (Appendix C. Figure 1). This wind loading does not apply to hurricane or typhoon prone areas or to localized high wind areas.

Ice loading is caused by ice freezing to the poles. Because water has a high density, this loading can be significant. The ice loading considers the shaft covered with 1 inches of ice and the arm covered with 2 inches of ice plus 6 inches of ice hanging from the arm and luminaires. (Appendix A.)

The final loading, both ice and wind, considers the worst case of environmental loading. This loading is much more than $Wind + 1/2 \cdot Ice$ as NCHRP Report 411 suggests as a maximum. This load is not expected to occur in reality due to the ice breaking off in high wind; however, this case will be a good test of overloading the design for maximum safety.

METHOD OF CALCULATION

A Traffic pole in Stillwater was observered and its approximate dimensions measured. This design has a shaft of 25 feet and an arm that spans 2 lanes which is 25 feet. The original shaft has a diameter of 12 inches at the base and 10 inches at the top. The arm tapers from 10 inches at the base to 4 inches at the tip. The deflection of the arm was approximately 5 inches with no wind or ice loading. The deflection of the shaft was barely significant at less than 3 inches. The shaft material was steel of less than 1/8 inch thickness. The shaft was bolted in four places to a concrete pad set into the ground. The arm was attached to the shaft by bolts into a flange welded to the shaft. Although the type and history of the steel was unknown, it is assumed to be a non-hardened carbon steel such as the 10xx series. The Steel type was assumed to be AISA 1040. The surface finish appeared to be a coating of zinc.

Hapco Inc, a traffic pole manufacturing company, operates a web site describing their design specifications and dimensions. They design only aluminum poles due to corrosion and cost factors. The site also mentions the materials basic loading conditions they use in the design of their poles.

The large number of calculations combined with the testing of several similar loading cases led to using a computer for the calculations. The symbolic mathematics program MathCAD was used. This also allowed iterations in the design process to take place nearly instantaneously and thus more frequently. Changing single parameters was easy and led to a better understanding of the relationship between the parameter and reaction of the design. The analysis of the shaft, arm and the computation of external loading were computed in different documents and important variables passed between them via data files.

Assumptions and limitations were taken into consideration during the analysis of the traffic signals fabrication and overall performance. Since there are many different types of luminaires that have many different characteristics, like three to five light signals and multiple aesthetic accessories, the luminaires were assumed to weigh eighty pounds, have three light signals and have no aesthetic accessories. Also, it is assumed that the luminaires-to-arm attachments will not break in the extreme weather conditions. The wind gusts are assumed to be evenly distributed across the arm and shaft, and the wind forces only travel in the coordinate y and x directions (refer to AutoCAD drawings, Appendix A). The wind gust also were assumed to travel across a uniform diameter shaft and arm using the largest value of the taper for design safety. Even though it is highly unlikely, the thickness of the shaft and arm are assumed to be perfectly uniform without defects or forms of corrosion. The concrete base of the shaft is assumed to remain perpendicular to the ground and not shift from erosion of the ground or severe winds. The circuitry of the light is assumed not to be effeted by any weather conditions. The ice coats on the arm and shaft were assumed to have constant diameters taking the average of the taper and to be uniformly thick, one inch. The luminaires were considered boxes as far as dimensions for an ice coat for added weight. Many of the stated factors above could be effected by rare environmental conditions, but are not considered in a mechanical design analysis. One mechanical factor not considered is impact by an automobile. This was not given as a design criteria.

The cost calculation was limited because no information could be found on the cost of a basic luminary or material forming and finishing cost because companies required quotes and specifications for prices. The only available prices were last years average sale costs of raw aluminum and raw steel per pound. The overall analyses is limited to a two luminary bar system and does not take into affect the possibility of needing a new or different luminary due to street or traffic condition alterations.

The analyses for the arm and shaft for static and wind and ice loadings are given in *Traffic Arm: Static*, *Traffic Arm: Wind and Ice*,*Shaft: Static* and *Shaft: Wind and Ice:* (Appendix B). Each analysis is divided into 5 parts: initialization, forces, strength and deformation. The analyses of shafts also include a stability section.

The documents first define functions to easily compute the Moments of Inertia, diameters and principle stresses. The physical constants are entered, top and bottom diameter, length of the shaft, thickness and material properties. For ease of calculation with some complicated loading conditions, the poles are divided into 300 one inch long segments. The 2nd Moment of Inertia, Polar moment of inertia, area and weight are calculated and plotted versus the shaft height. This section also inputs the wind and ice loadings from *Wind and Ice* given in Appendix B. For the static cases, these loadings are then set to zero.

Force analysis computes the forces along the member resulting from the loading condition specified in the previous part of the analysis document. For the shaft, the force analysis first inputs the loading reactions resulting from the arm. The shear forces, axial forces and moments are computed along the length of the member for all three axises. Formulas for forces and moments are taken from loading condition 2,3 and 4 in Table A-9 of Shigley.

Strength analysis uses the values for moments and forces from the previous part and computes the bending and shear stresses along the length of the member. The maximum moment point is computed by converting the moments at the base to real and imaginary parts and then converting into an equivalent magnitude and direction. A maximum compressive force due to bending moment is computed using the magnitude of the previously found maximum moment. The stresses at the maximum moment point are entered into function to solve for the principle stresses and the result is stored into a matrix. Maximum shear stress is computed and printed from the difference in magnitude of the two largest resulting principle stresses.

The next part of the analysis is deformation. Due to the number of non-uniform loads and changing physical properties of the pole, the deflection is calculated by a finite difference method. Each deflection value is calculated as the addition of the previous deflection and an expression for the deflection of a one inch segment of the member. This method is similar to that given in Section 3-6 of Shigley. The resulting deflections and slopes are plotted. Finally, the total deflection in an axis direction is computed and plotted as the sum of all deflections in that direction.

Stability is considered only in the shaft. Stability of the shaft is considered with secant formula, *Eq 3-62* (Shigley), for eccentric loading. The radius of gyration and eccentricity are computed. The force at the top of the shaft required to buckle the shaft is computed for both the properties of the bottom and top of the shaft.

The *Drag and Ice* document given in Appendix B computes the drag and ice loadings on the shaft and arm. Drag and Ice loadings are output to a data file for later use. All of the calculations are in pounds and inches.

The drag data is calculated for the two sizes of tubes and the luminaries. A value for the velocity of the wind in miles per hour is input and converted to feet per second. Also entered are the properties of air and a function for drag (Young, Munson and Okiishi). An average value for the Effective Projected Area of the luminaries was found in NCHRP Report 411 *Table 10.* The coefficient of drag for the poles were found in Young, Munsion and Okiishi. Values for the drag are calculated and written to a data file for later use.

Ice loading was calculated for the two poles and the luminaries. The ice was one inch thick covering the average surface area of each pole and luminaire. In addition, hanging ice three inches by six inches hung from the entire length of the arm. For ease of later calculation, the resulting ice load was uniformly distributed along the appropriate member.

The MathCAD documents provided on the diskette allows for varying the material, thickness, diameter and length of the shaft and arm in either ice, drag or both environmental conditions for any other data needed.

RESULTS AND DISCUSSION

Calculations were performed as described above. The existing steel design and the new design were analyized. A full analysis of the new design and a summary of the existing design is given in Appendix A.

Analysis in the drag and ice was the same for for both the existing and the new design. Using a wind velocity of 90 miles per hour, the 10 inch tube had a drag of 0.864 pounds per linear inch. The 12 inch tube had a drag of 0.72 pounds per linear inch. The luminaries had a drag of 150 pounds each. Ice added a total of 262 pounds to the shaft which is 0.872 pounds per linear inch. The total added weight of ice to the luminaries and the arm was 438 pounds for a distributed load of 1.5 pounds per linear inch of the arm.

The new design uses Aluminum alloy 6063. Both the shaft and the arm are aluminum.

This material has a minimum yield strength of 60 ksi. Aluminum was selected for the resistance to corrosion, relative abundance, low density and ease of machining.

The static case of the arm and shaft is given in *Traffic Arm: Static* and *Shaft: Static* (Appendix A). This loading condition had only the weight of the components and luminaries. At the critical point of the arm, the maximus compressive stress is 3.5 ksi and the maximum shear 1.8 ksi. The arm deflected 2.29 inches due to arm bending plus 1.7 inches due to the shaft bending for a total of 4 inches deflection at the end of the arm. No x deflection occurred. At the arm mount, the arm creates a reaction of 304 pounds in the negative z direction and moment of -53000 inch pounds around the x axis. For the shaft, the maximum compressive stress at the shaft base was 2.5 ksi resulting in a maximum shear of 1.25 ksi. There was no shaft deflection in the x direction. In the y direction, the shaft deflected -1.7 inches. The secant formula for the critical buckling force yielded a value 2500 pounds to buckle the shaft assuming the diameter to be that of the top, 10 inches. 3600 pounds was required to buckle the shaft with the diameter that of the bottom, 12 inches.

The wind and ice loading case of the arm and shaft is given in *Shaft: Wind and Ice* and *Traffic Arm: Wind and Ice.* This case considered an ice and wind loading as given in *Wind and Ice* (Appendix A). At the critical point of the arm at the base, the maximum principle stress is 9.9 ksi and the maximum shear stress of 5 ksi. The total arm deflection in the z direction is 4.9 inches due to the arm and 3.2 inches due to the shaft rotation about the x axis for a total of 8.1 inches of z deflection at the arm tip. The arm deflected 1.25 inches in the x direction at the arm tip. The reaction force in the z direction at the arm mount is 742 pounds, 515 pounds in the negative x direction, a reaction moment of 86000 inch pounds around the z axis and 118000 inch pounds around the negative z direction. At the critical point of the shaft at the base, the maximum compressive stress was 6.8 ksi. When the compressive stress of the weight and the torsion of the arm moment is combined, the maximum principle stress is 10.3 ksi yielding a maximum shear stress of 6.2 ksi. The shaft deflected 3.2 inches in the y direction and 2.6 inches in the x direction. A total of 2.7 ksi at 159 inches from the centerline of the shaft was required to buckle the shaft assuming constant diameter of 10 inches.

Analysis was performed on the existing steel design for two cases, a static case and a wind plus ice case. Analysis summaries of the existing steel design are given in *Steel Traffic Pole Arm: Static*, *Steel Traffic Pole Arm: Wind and Ice*, *Steel Traffic Pole Shaft: Static* and *Steel Traffic Pole Shaft: Wind and Ice*. AISI 1040 steel has a yield strength of 80 ksi.

The static case of the existing steel design is given in *Steel Traffic Pole Arm: Static* and *Steel Traffic Pole Shaft: Static*. A bottom diameter of 12 inches and a top diameter of 10 inches was used in the calculations. The first luminaire was spaced 10 feet from the arm base and the second luminaire was spaced an additional 10 feet. The thickness of the steel was estimated at .125 inches. The stress maximum at the base of the arm is 7.6 ksi. The maximum stress at the critical point of shaft base is 6.0 ksi. The maximum deflection at the end of the arm is the deflection of the arm plus the deflection due to bending rotation of the shaft. From the analysis, the end deflection δ_{arm} equals $1.852 + 1.559 = 3.408\ inches$. There was no arm deflection in the x direction. The shaft deflected 1.559 inches in the negative y direction due to the reaction to the arm's loading. The shaft had no deflection in the x direction.

The wind and ice loading of the existing steel design is given in *Steel Traffic Pole Arm: Wind and Ice* and *Steel Traffic Pole Shaft: Wind and Ice.* Two 90 mile per hour winds were directed simultaneously towards both the negative x and negative y directions. The maximum stress at the critical point of the shaft was 10.7 ksi. The maximum stress at the critical point of the arm was 17 ksi. A maximum shaft deflection of 3.7 inches in the negative y direction and -2.7 inches in the negative x direction was calculated. The arm deflected .7 inches in the x direction. The z deflection for the arm was 3.245 inches due to the arm and 3.7 inches in the shaft for a total of 6.945 inches deflection in the z direction.

The design requires four welds. The locations of these welds are: shaft to the base flange, arm to the arm mounting flange, the arm mount to the shaft and the cap on the shaft. The shaft fits into the flange and then is welded at the junctions. The arm to the arm flange is welded similarly. These types of welds are given in NCHRP Report 412 *Appendix A-9* to withstand 13 ksi of stress. This is within the limits found in the above calculation. For added strength, gussets could be welded between the flange and the member such as *Example 14 Appendix A-11* in NCHRP Report 412. The arm is mounted by bolting the arm flange onto a built up box welded to the shaft. From NCHRP Report 412 *Appendix A-9)*, the member can withstand 13 ksi of stress. A cap is also welded onto the top of the shaft.

The new design uses bolted connections for connecting the shaft flange and the arm flange to the respective mounting location. SAE grade 5 bolts of one inch diameter will be used for the arm flange to arm mounting box connection. Similar bolts embedded into the concrete base are used to connect the shaft flange to the concrete base. This is shown in Appendix A. From table 8-4 in Shigley, these grade of bolts are rated at 85kpsi proof strength which is well within the stresses found in the flange junctions by at least a factor of 8.

Cost analysis only considers the difference in materials and machining. No extra luminaries or other hardware is considered.

The new design uses aluminum for both the shaft and arm. Aluminum was sold at an average price of 71 cents per pound. (Kaiser Aluminum Corporation.) Thus at 415 pounds total, the raw cost of the aluminum for the shaft and arm is 300 dollars. Machining and finishing of the poles is expected to cost four times the raw materials. Four welds are expected to cost 50 dollars apiece for labor, materials and setup overhead. Bolts and associated hardware will cost 50 dollars. The estimated cost of manufacturing the aluminum design is 1750 dollars per unit.

The existing design uses steel. Steel was sold at an average of 30.8 cents per pound. (Stelco Incorporated) At 530 pounds, the raw cost of the steel is 164 dollars. Machining and finishing of the poles is expected to cost three times the raw materials. Four welds are expected to cost 50 dollars apiece for labor, materials and setup overhead. Bolts and associated hardware will cost 50 dollars. The estimated cost of manufacturing the existing steel design is 905 dollars per unit.

CONCLUSION

The new design meets all design criteria. All stresses are within the limit for aluminum. Without the welded junctions, the largest stress created in the worst conditions is 10.3 ksi in the shaft base. Assuming that the welding junctions can be further improved to not be the limiting stress, the factor of safety is at least $60/10.3 \approx 6$. The maximum arm deflections of the new design are similar to the existing steel design; however, the new design's arm only has approximately half the taper as the existing steel design keeping the diameter at the tip for the new design is two inches larger. The new design does initially cost more but aluminum properly maintained should outlive the steel pole due to rust and corrosion problems with steel. Also, the scrap value of the aluminum pole is considerably higher than that of the steel (Hapco). The new design meets all design criteria. The new design performed as well as the old, has a better salvage value, fewer corrosion problems and will last longer.

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APPENDIX A

AutoCAD Drawings

APPENDIX B

MathCAD documents

APPENDIX C

Figures

FIGURE 1. Wind maximum (50 year)