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RECOMMENDED REVISIONS TO UNIT STRESSES IN MACHINERY PARTS

AMERICAN SOCIETY OF MECHANICAL ENGINEERS SECOND BIENNIAL MOVABLE BRIDGE SYMPOSIUM

presented by
Joseph J. Hosanna, Associate
Envirodyne Engineers, Inc.

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1. INTRODUCTION

I am sure, if you have ever designed a machinery part for a movable bridge, that you have read "it", and you read "it" more than once. And then you read "it" again, because "it" contains so much usefull information. The "it" I am talking about is the last paragraph of Article 2.5.11 - Unit Stresses in Machinery Parts of AASHTO's "Standard Specifications for Movable Highway Bridges."

Even though the paragraph is quite lengthy, I have often thought that its subject matter deserves much more space in the Specifications. This paper reviews the content of this paragraph and makes specific recommendations concerning changes and additions to it.

2. WHAT DOES IT SAY?

The last paragraph of Article 2.5.11 states that the allowable unit stresses included in the article provide an appropriate factor of safety against:

- o Static Failure
- o Fatigue with and without Stress Reversal
- o Stress-Raisers which produce stress concentrations of 140 percent of the computed stresses.

The article then goes on to describe, in general terms, the types of details covered by the 140 percent stress concentration factor. They include increases of stress:

- o At shoulders of the bearings for trunnions and counterweight sheave shafts.
- o Near the hub at gear arms.
- o At the faces of pinions for integral shafts and pinions.
- o At keyways.

Of all the stress raisers described only the keyways are further defined as to size and location. It states that keyways shall have a width not more than 1/4 the shaft diameter and a depth not more than 1/8 the shaft diameter. For other machinery elements it alludes to fillets of "reasonable radius". This area of the specifications on stress raisers is the one I feel should be expanded.

Lastly, the paragraph states that if a shaft has no keyways or other stress raisers the unit stresses in the shaft may be increased by 20 percent.

3. BACKGROUND

3.1 Factor of Safety

What is the actual factor of safety provided by the values listed in Article 2.5.11, based on the information provided in the last paragraph? Since "an appropriate factor of safety" is provided against failure by fatigue with a maximum stress concentration factor of 1.4 as stated in the last paragraph of Article 2.5.11 then the actual factor of safety is:

$$F_s = \frac{S_e}{K_t S_a}$$

Where:
F_s = Actual Factor of Safety
S_e = Endurance Limit Stress
K_t = Stress Concentration Factor
S_a = Allowable Unit Stress

F_s, K_t, and S_a are self explanatory. However, S_e needs further explanation. By definition the endurance limit stress is the maximum value of the completely reversed bending stress, which a plain specimen can sustain for 1,000,000 or more load cycles without failure. Or, in other words, if the stress in a machinery part never exceeds a certain value called the endurance limit the part will last indefinitely.

The endurance limit is dependent on the condition of the part's surface. For a part with machined surfaces at points of maximum stress, the endurance limit is approximately 35 to 40 percent of the ultimate strength of the material.

Upon inspection of the allowable unit stresses in the AASHTO Specifications it can be readily seen that the values for bending are equal to 20 percent of the ultimate strength of the material (S_u), or

$$S_a = \frac{S_u}{5} \qquad (2)$$

Rearranging equation (1) and substituting the values for S_e and K_t, we have

$$S_a = \frac{S_e}{K_t F_s} = \frac{0.4\ S_u}{1.4\ F_s} \qquad (3)$$

Equating equations (2) and (3), and solving for F_s, we find

$$\frac{0.4\ S_u}{1.4\ F_s} = \frac{S_u}{5}$$

Therefore, $F_s = 1.4$

Not much of a factor of safety, for a part undergoing a large number of cycles under the maximum loading.

An interesting point is that the stress concentration factor K_t times the factor of safety is equal to 2. Thus, if the stress concentration factor is greater than two the part will fail from fatigue at less than one million cycles of loading.

3.2 Stress Concentration Factors

Figure 3.1 below presents the stress concentration factor for a round bar in bending.

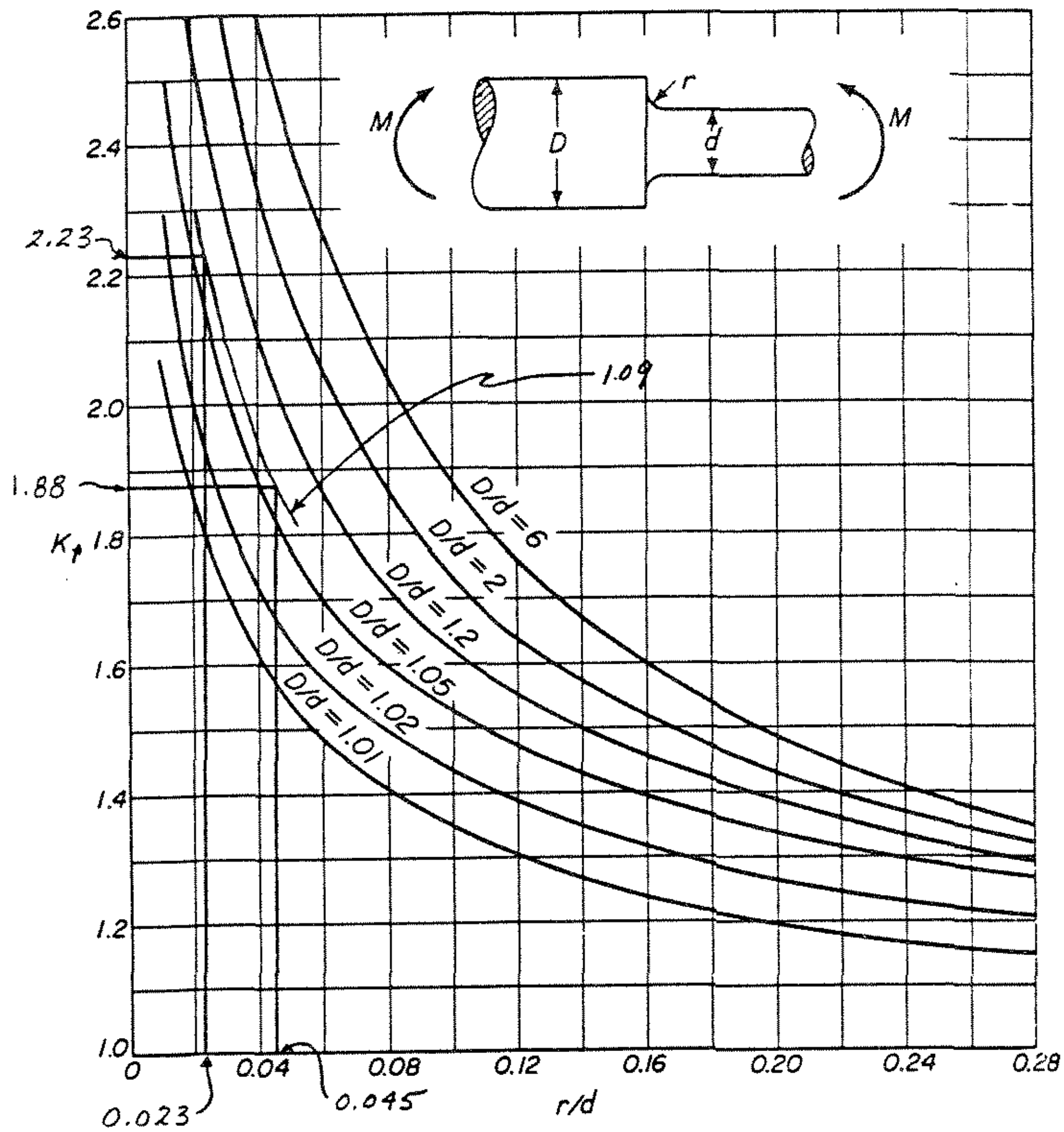

FIGURE 3.1 Factors of stress concentration *K* for various sizes of fillets for round bar in bending to be applied to the stress in the section of diameter *d*.

Let's look at an example and determine what the actual stress concentration factor is:

The minor diameter (d) of a counterweight sheave shaft is 22 inches, the major diameter (D) is 24 inches, and the radius of the fillet (r) is the maximum allowable based on the geometry, 1 inch. Therefore, D/d = 1.09 and r/d = 0.045. Using Figure 3.1 the stress concentration factor is 1.88.

If we are to maintain the implied factor of safety included in the specifications then the allowable stress for the sheave shaft should be reduced to 74 percent of the value listed in Article 2.5.11.

Now let's look at the sample problem but this time let the fillet be equal to 1/2 inch. D/d is still equal to 1.09 but r/d = 0.023. Again using Figure 3.1 the stress concentration factor is now 2.23. Since K_t is greater than 2, the factor of safety against fatigue failure is less than one. Which means the shaft will last less than 1,000,000 cycles.

Assuming that the counterweight sheave shaft is machined from a steel that has fatigue curves as shown in Figure 3.2 and the shaft is designed for a maximum allowable stress equal to 20 percent of the ultimate or 18 ksi, then for the case where the fillet is 1 inch the actual stress at the fillet, taking into account the stress concentration factor, is 33.8 ksi. However, for the shaft with a fillet radius of 1/2 inch the actual stress is 40.1 ksi. For a shaft with a machined finish from Figure 3.2 the shaft with the 1 inch radius should have a fatigue life of 1,000,000 cycles but the shaft with the 1/2 inch radius would have a fatigue life of only 200,000 cycles.

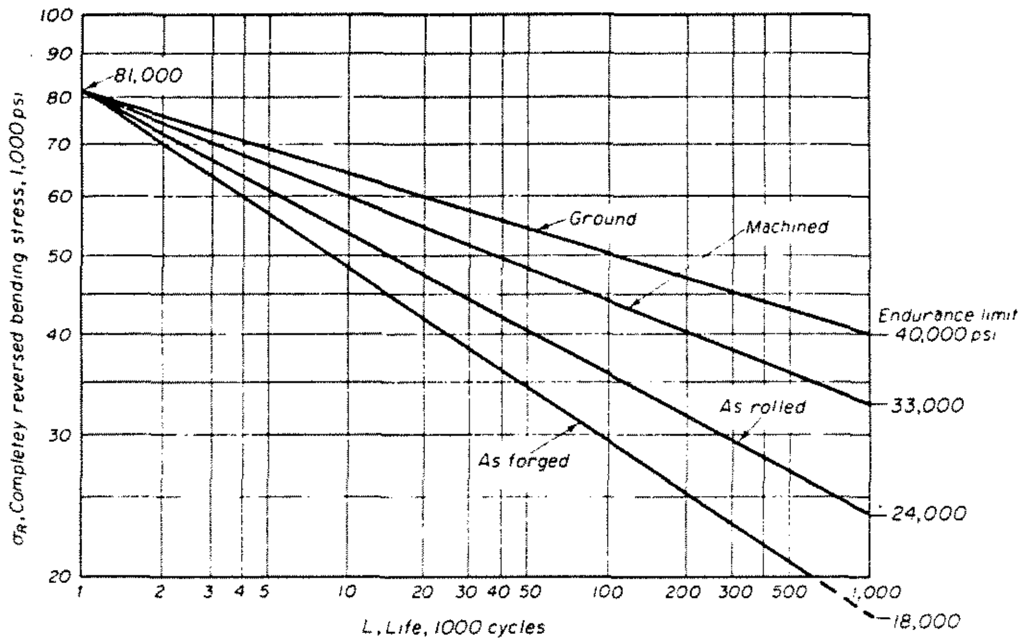

FIGURE 3.2 Typical fatigue curves: steel specimens; reversed bending; σ_{ult} 90,000 psi. Relationship between equivalent completely reversed stress σ_R and fatigue life L.

If the sheave shaft was for a vertical lift bridge which opens twenty times a day, every day per year and each time the bridge opens the sheave shaft rotates through two complete revolutions, then the usefull life of the shaft would be greater than 50 years if the 1 inch fillet were used but less than 14 years if the 1/4 inch fillet were used.

It should be noted that the stress concentration factors presented throughout this paper are based strictly on the form or shape of the part and not on the material or size. In reality the factor of stress concentration for fatigue, which is the ratio of the endurance limit of a plain specimen to the nominal stress at the endurance limit of a specimen containing a stress raiser, should be utilized to more accurately depict the real life behavior.

The stress concentration factor for fatigue not only depends on the shape of the part but also the type of material, and the actual size of the part. Methods are available for making quantitative estimates of the fatigue stress concentration factors but these methods are somewhat complicated. The full theoretical stress concentration factors, as presented in this paper, are easier to determine and are usually larger than the fatigue stress concentration factors. Therefore, when the factors presented herein are applied, the results will usually be on the safe side.

4. RECOMMENDED CHANGES TO ARTICLE 2.5.11

From the previous example, you can begin to see the importance that the details of a machinery part and the associated stress concentration factors have on the service life of the component. For this reason it is recommended that Article 2.5.11 be modified to stress the importance of the details by providing values for stress concentration factors, for the most common stress raisers.

The recommended change to this article is the addition of the following after the last paragraph of Article 2.5.11:

"The actual stress concentration factor for various details of machinery and other similar parts shall be computed by utilizing the figures contained in Appendix A. If the stress concentration factor determined from the Appendix is less than 1.4 the values of allowable unit stress tabulated in Article 2.5.11 may be utilized directly. If the determined stress concentration factor is greater than 1.4 then the allowable unit stress S_k shall be utilized for the design.

$$S_k = \frac{1.4\ S_a}{K_t}$$

Where: S_k = Reduced allowable unit stress
S_a = Allowable unit stress tabulated in Article 2.5.11
K_t = Stress concentration factor from Appendix A."

It is further recommended that the last sentence of Article 2.5.11 be deleted. This sentence reads, "In the

absence of keyways or other stress-raisers in a shaft, the unit stresses for torsion and flexure in a shaft may be increased 20 percent." This provision would be covered by the proposed addition of the following paragraph.

"If the stress concentration factor determined from Appendix A is less than 1.4 the formula for S_k may be utilized to increase the allowable unit stress. But in no case shall the stress concentration factor be taken as less than 1.2."

Until a change is made to the Specifications it is strongly recommended that all designers verify that the details they utilize for machinery components do not have stress raisers with stress concentration factors exceeding 1.4.

5. MODIFICATIONS TO THE AREA SPECIFICATIONS

The American Railway Engineering Association (AREA) Committee 15 recently approved a revision to the article in the AREA Specifications, which is analogous to Article 2.5.11 in the AASHTO Specifications. The revision lowered the allowable stress to 10,000 psi for trunnions with rotation more than 180 degrees. The explanation given for this change was that trunnions which rotate more than 180 degress experience full stress reversal, and that the fatigue life of a trunnion is reduced by keyways, shoulders and other section discontinuities.

The endurance limit for the two forged steels listed for trunnions in the specification is approximately 30,000 psi. Trunnions designed in accordance with the specifications would not experience stresses in excess of 30,000 psi, and therefore could not fail by fatigue, unless they were overloaded by some unexpected occurrence or a detail of the part created a locally high stress due to a stress raiser with a high stress concentration factor.

This paper presents a different approach to handling an apparent problem with the allowable stresses in trunnions. Rather than reducing the allowable stress just for trunnions, it is recommended that the modifications to the specifications contained herein be adopted.

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2. M.F. Spotts, "Design of Machine Elements", New Jersey, Prentice - Hall, Inc., 1985.

3. Raymond J. Roark and Warren C. Young, "Formulas for Stress and Strain", New York, McGraw - Hill Book Company, 1975.

APPENDIX A

FACTORS OF STRESS CONCENTRATION

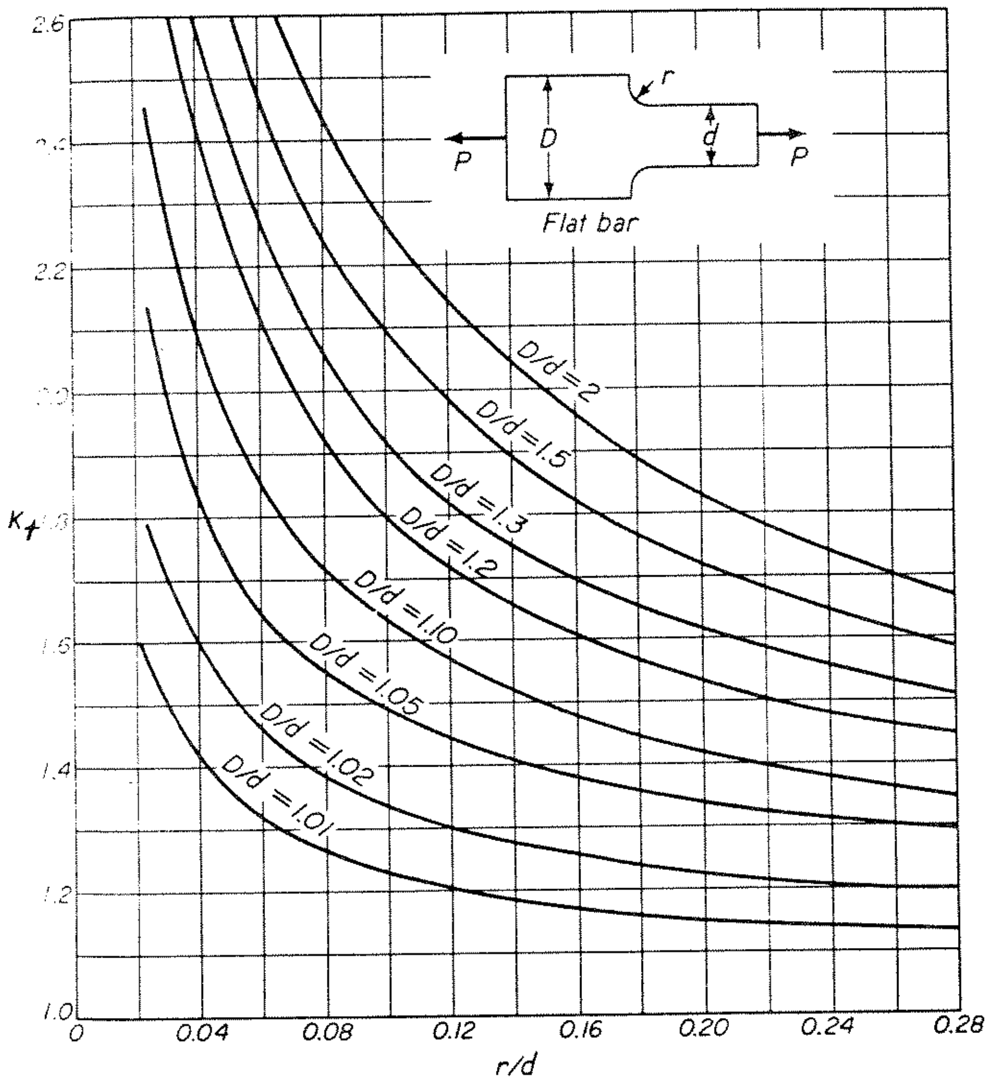

FIGURE A-1 Factors of stress concentration K for various sizes of fillets for flat bar in tension or compression to be applied to the stress in the section of width d.

APPENDIX A

FACTORS OF STRESS CONCENTRATION

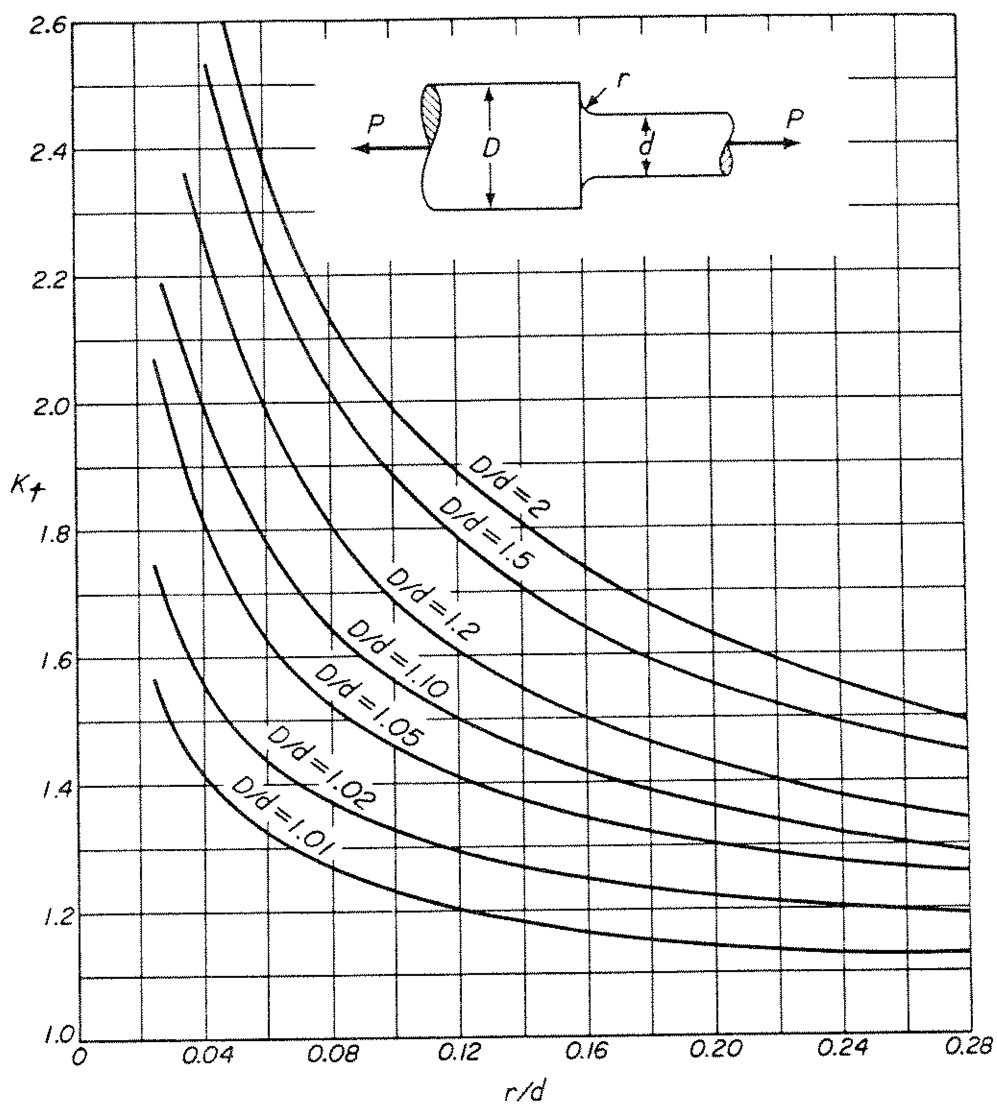

FIGURE A-2 Factors of stress concentration *K* for various sizes of fillets for round bar in tension or compression to be applied to the stress in the section diameter *d*.

APPENDIX A

FACTORS OF STRESS CONCENTRATION

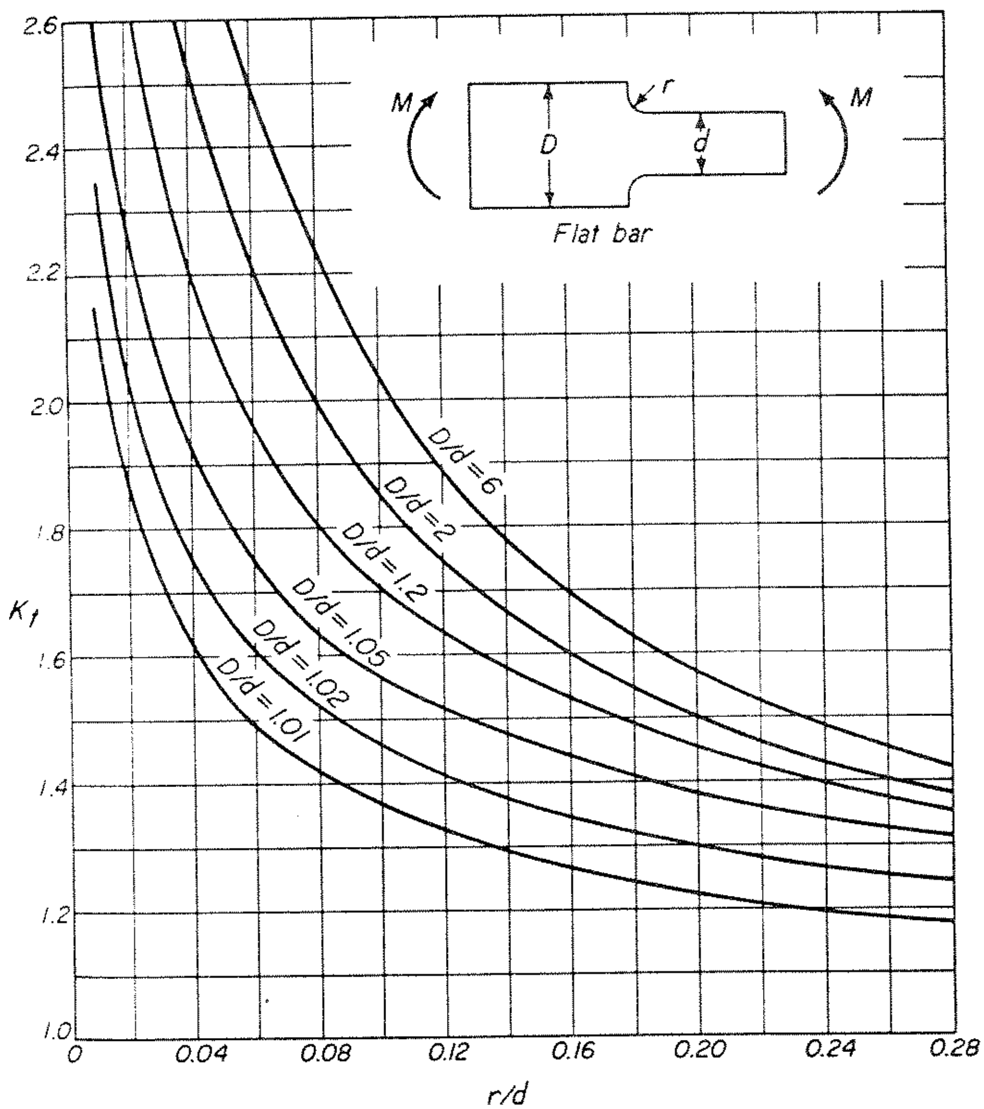

FIGURE A-3 Factors of stress concentration K for various sizes of fillets for flat bar in bending to be applied to the stress in the section of width d.

FACTORS OF STRESS CONCENTRATION

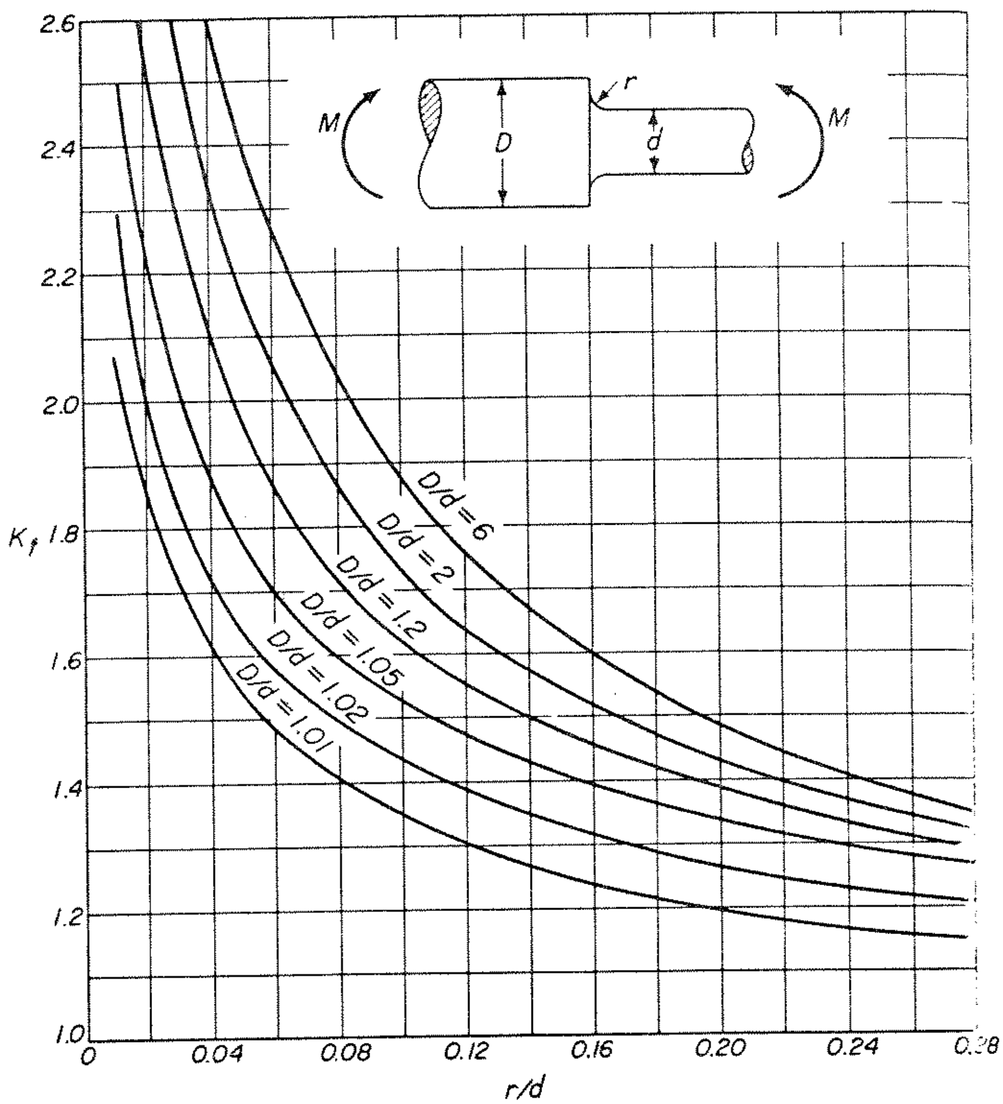

FIGURE A-4 Factors of stress concentration *K* for various sizes of fillets for round bar in bending to be applied to the stress in the section of diameter *d*.

APPENDIX A

FACTORS OF STRESS CONCENTRATION

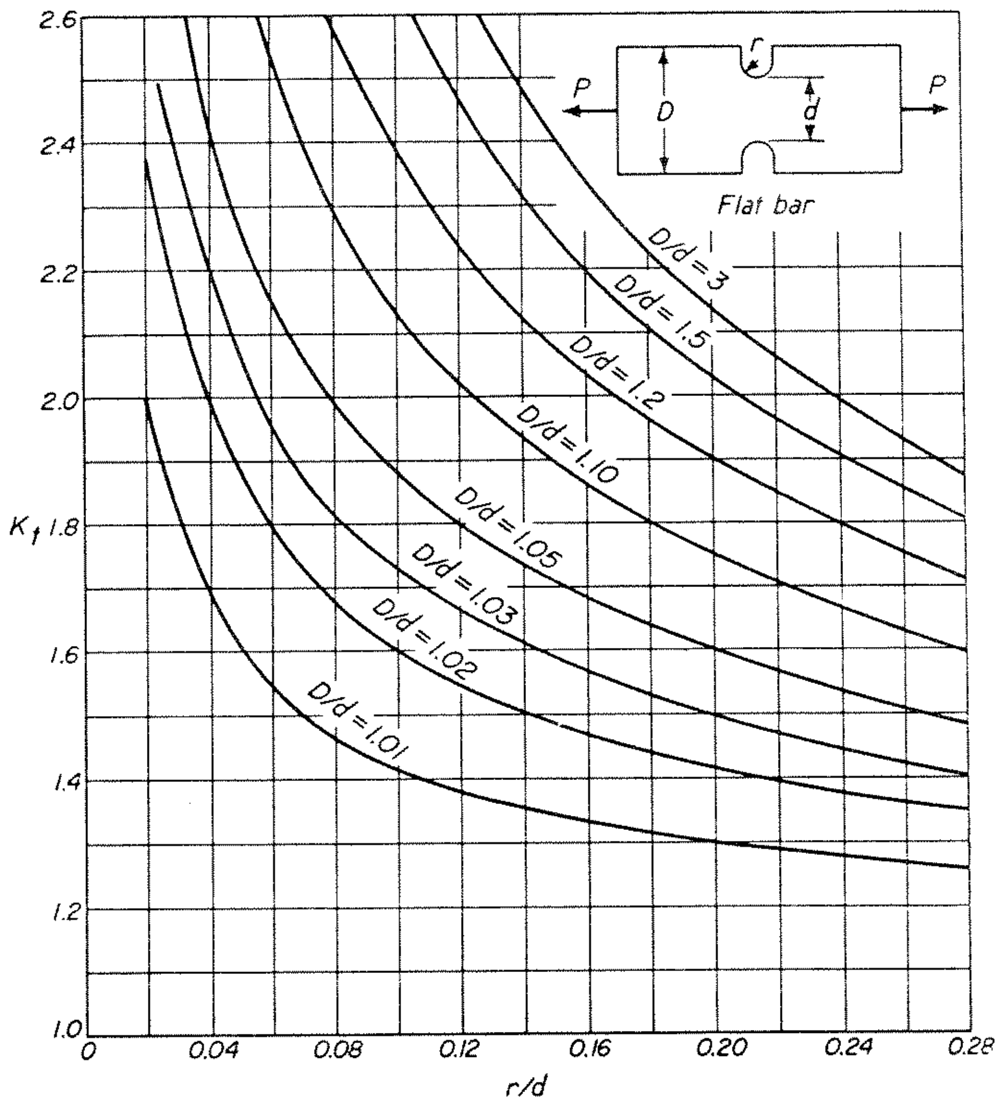

FIGURE A-5 Factors of stress concentration K for grooves of various depths in tension or compression to be applied to the stress in the section of the flat bar of width d.

APPENDIX A

FACTORS OF STRESS CONCENTRATION

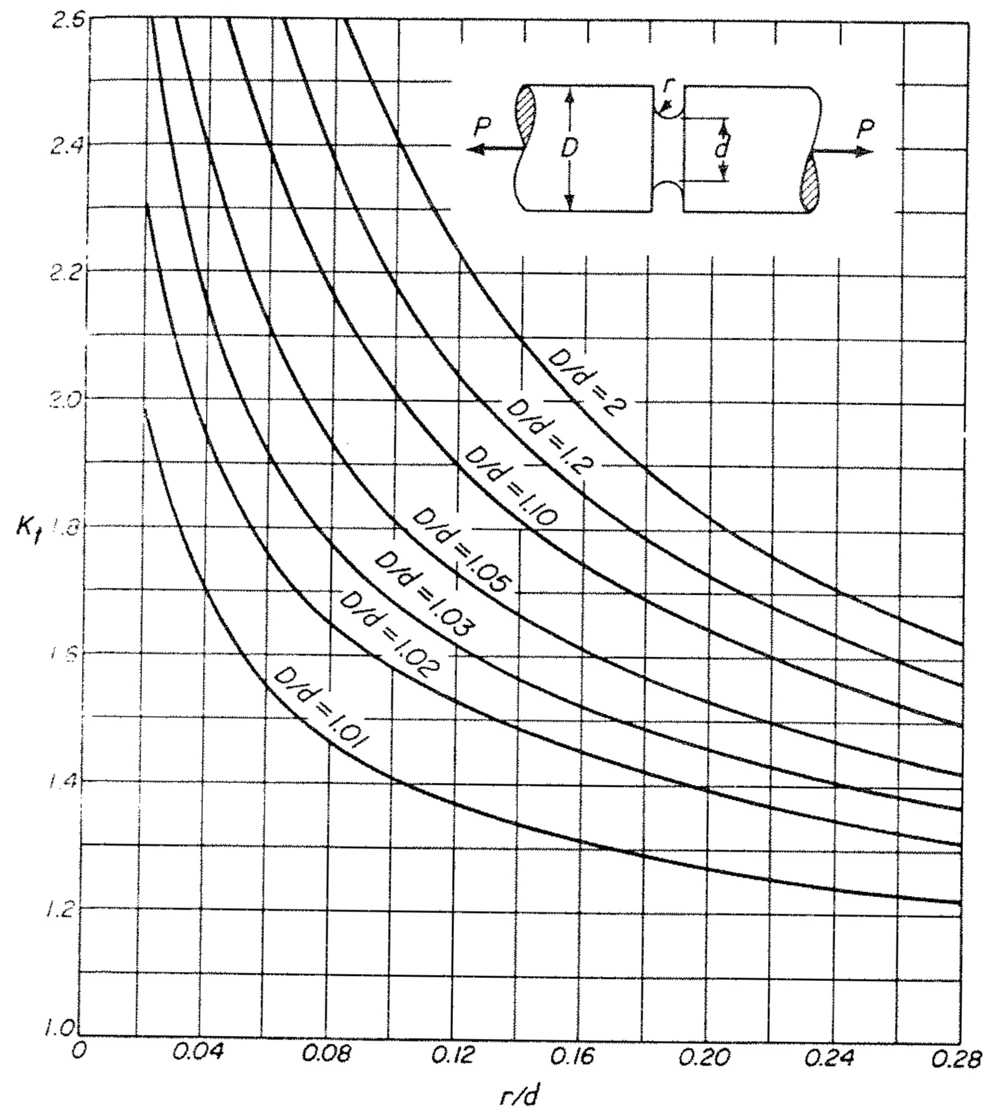

FIGURE A-6 Factors of stress concentration K for grooves of various depths for round bar in tension or compression to be applied to the stress in the section of diameter d.

FACTORS OF STRESS CONCENTRATION

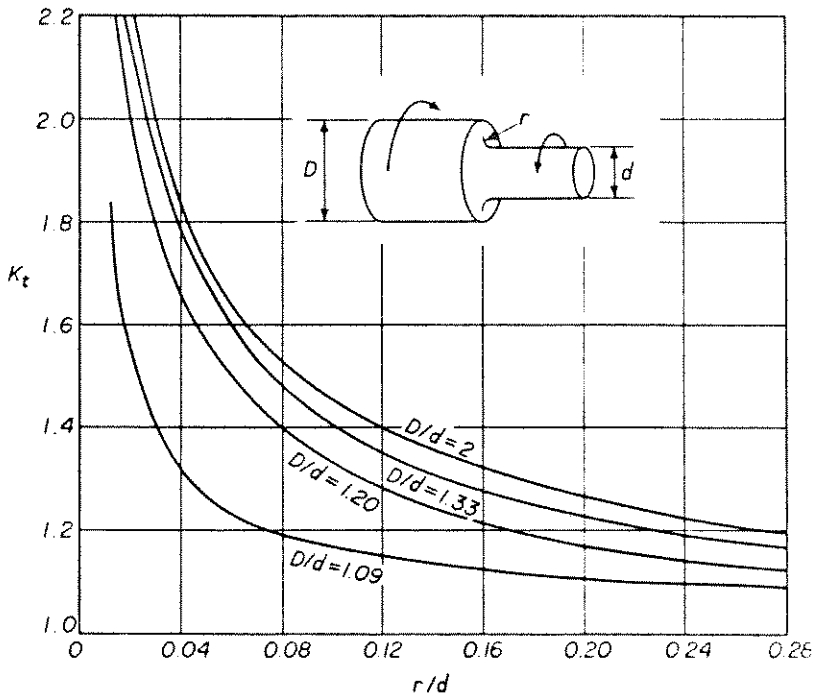

FIGURE A-7 Factors of torsional stress concentration K for circular shafts of two diameters to be applied to the shear stress in section of diameter d.

FACTORS OF STRESS CONCENTRATION

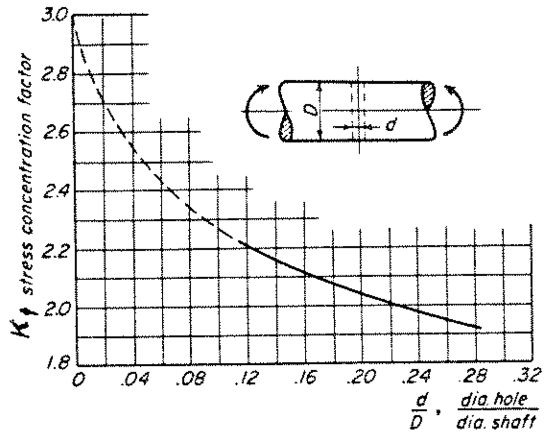

FIGURE A-8 Stress concentration factors for shaft with transverse hole loaded in bending. Based on section modulus of the net area.

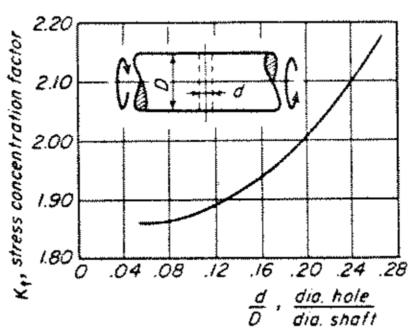

FIGURE A-9 Stress concentration factors for shaft with transverse hole. Torsional loading. Based on full cross sectional area. No reduction for hole.