PD 6438:1969

A review of present methods for

Design of bolted flanges for pressure vessels

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Contents

Foreword

This is the third memorandum in the series being prepared by Committee E/-/3 and reviews the methods of design for bolted flanges given in British Standards and other codes. It comments on the limits of application of the various rules and recommends where further study is required to evolve standard design methods to take into account all relevant parameters.

This memorandum has been prepared by Mr. P.J. Kemp and has been scrutinized and approved by the various committees responsible for particular British Standards for pressure vessels and bolted flanges.

Summary of pages

This document comprises a front cover, an inside front cover, pages i and ii, pages 1 to 7 and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

1 Introduction

The following review is limited to the design of bolted circular flanges for services outside the standard series. Excluded are pipe flanges such as those covered in sizes up to 24 in diameter in BS 1560 (1), BS 10 (2) and a British Standard for metric flanges now being prepared. The ASA series is used in Europe for the petroleum industry with inch-size bolting, but the ISA series of flanges is being used for many other purposes.

2 Existing Methods

2.1 The ASME method (7) for flange design is widely used in the British petro-chemical industry and has been adopted in:

BS 1515, "*Fusion welded pressure vessels for use in the chemical, petroleum and allied industries*", Part 1, "*Carbon and ferritic alloy steels*", and Part 2, "*Austenitic stainless steel*".

BS 3915, "*Carbon and low alloy steel pressure vessels for primary circuits of nuclear reactors*".

2.2 Significantly higher design stresses are permitted in these British Standards than allowed in ASME VIII (7). However, at the test pressure the amount of plastic strain that might occur in these British flange designs is no higher than could occur in ASME VIII flanges, as shown in Table 1.

2.3 BS 1500-1 (3) has retained the Lake and Boyd (28) method, which was introduced to provide lighter flanges than the ASME method. The comprehensive data on gasket factors and minimum design seating stresses for various gasket materials tabulated in the ASME procedure is unfortunately lacking in BS 1500.

2.4 It was known in 1957 that the ASME (Taylor Forge, ref. 13) method was liable to be unsatisfactory for large diameter flanges and, it was reported, could lead to designs that could not be made leak-tight.

2.5 Murray and Stuart (34), using theoretical and experimental evidence, showed that for large flanges the Taylor Forge method underestimates and the BS 1500 method over-estimates the stresses for large taper hub flanges. Consequently, for diameters over about 10 ft ASME flanges may be too thin and BS 1500 taper hub flanges may be uneconomically thick.

2.6 The discrepancies are due to the neglect of a particular integral in the original calculations. The Murray and Stuart method enables calculations to be made of the longitudinal stresses behind the hub and the rotation of the flange for individual cases. Printing errors in the equations in the original paper have to be corrected before solving the eight simultaneous linear equations.

2.7 DIN 2505 (40) includes a method for dealing with load deformation of the joint due to pressure. The Swedish Pressure Vessel Code (54) has a procedure for calculating full face flanges and non-circular plate flanges.

3 Particular cases

3.1 Flanges for cryogenic temperatures

3.1.1 When flanges tightened at ambient temperature are cooled the materials contract, usually causing relaxation of the bolt stress and hence of the gasket pressure. The joint may then leak at low temperature.

3.1.2 Bolted flanged joints should be avoided, if possible, for low temperature service by using all-welded or brazed joints. The use of joints fitted with bore seals such as those made by Messrs. Ruston Graylock Ltd. or High Duty Couplings Ltd. may be considered. In these cases the seal is at cone surfaces on a thin metal ring within the bore of a pair of flanges. The sealing ring material should have a coefficient of contraction not more than that of either of the flanges of the joint.

3.1.3 When flanged joints must be used at low temperature the bolting should be of material with a coefficient of contraction not less than that of the flanges. If possible, the bolts and flanges should be covered with thermal insulation to help minimise temperature gradients. The use of compensating washers of material with very low coefficient of contraction under the nuts will help ensure a tighter joint at low temperature.

3.1.4 If there is no satisfactory alternative to a pair of flanges of dissimilar metals the bolting may be provided with compensating sleeves or washers (37).

3.2 Flanges for high temperatures

3.2.1 When flanges tightened at ambient temperature are heated the flange material expands, usually causing the bolts, being at some what lower temperature, to tighten.

3.2.2 When exposed to high temperature the flanges and bolts will creep, causing relaxation of the bolt load and hence of the gasket pressure, and eventually the joint may leak.

3.2.3 When the joint is cooled down after exposure to high temperature the joint may leak, due to:

1) plastic strain of bolts during initial heating of flanges

2) creep of bolts under load

3) creep of flanges under load.

3.2.4 Information for the design of flanges in hot services is contained in references 12, 22, 30, 44, 45, 49 and 51.

3.3 Flanges for high pressure

3.3.1 The necessary information to design high pressure flanges with pressure-energized ring joint gaskets and made from any suitable material is provided in a paper by Eichenberg (61). These rules have been used for the design of the American Petroleum Institute Standard API — 10 000 lb and 15 000 lb flanges.

3.4 Flanges of materials other than steel

3.4.1 The Taylor Forge method assumes a constant modulus of elasticity as for carbon steel at ambient temperature. For a flange of different material a correction must be applied to allow for the effect of the different E at the temperature under consideration (86). Under a given bending moment the angle of rotation of a flange ring is inversely proportional to the value of E (34).

4 Deficiencies of ASME method

The ASME method does not meet all the requirements for flange design and has the following major deficiencies:

4.1 Satisfactory up to about 5 ft diameter, progressively more unsatisfactory above this and inadequate above 10 ft (34).

4.2 Flat face flanges with metal to metal contact beyond the bolt circle not covered (54) (80) (81) (82) (83).

4.3 Hoop stress due to internal pressure neglected (54).

4.4 Applies primarily to flanges with the same modulus of elasticity as carbon steel (34) (86).

4.5 Does not consider separately the deformation characteristics of the gasket under effects of pressure and temperature (56) (59) (79).

4.6 Designs with self-energizing seals not covered other than elastomer O rings (38).

4.7 Thermal effects neglected (12) (51) (54) (36) (62).

4.8 Designs with radial slotted holes not covered (13) (54).

4.9 Applies primarily to circular flanges (13) (57).

4.10 Stress concentrations at fillets and holes neglected (54).

4.11 Does not give rotation of flange (34).

5 Recommendations

A general study to evolve standard design methods taking into account all relevant parameters would appear to be justified, as none of the methods used in current codes is ideal for every case. For instance, the BS 1500 (3) and BS 10 (2) methods are not suitable for taper hub flanges and the use of the Taylor Forge method is subject to the limitations listed in Clause **4**. The aims of any further work should be:

a) To provide standard design charts over a wider range of parameters than is covered in current codes.

b) To provide a computer method suitable for universal use outside the range of the standard design charts.

The work should embrace flanges with full face gaskets and materials other than carbon steel.

Table 1 — Maximum stresses in carbon steel pressure vessels at ambient temperature expressed as a decimal of the ultimate tensile strength and yield strength

Hoop	UTS x	0.2 % Y x
Nominal design stress (S_{Fo})		
ASME VIII:1965, para. UA-500	0.250	0.625
ASA B31-3:1966, para. 302.3 i(c)	0.333	0.625
BS 1515:1965	0.425	0.666
BS 3915:1965	0.425	0.666
Nominal stress at test pressure		
ASME VIII:1965, factor 1.5	0.375	0.938
ASA B31-3:1966, factor 1.3	0.433	0.813
BS 1515:1965, factor 1.3	0.552	0.866
BS 3915:1965, factor 1.3	0.552	0.866
Flange bending		
Maximum longitudinal stress at design pressure ($1.5 \times S_{Fo}$)		
ASME VIII:1965	0.375	0.938
ASA B31-3:1966	0.500	0.938
BS 1515:1965	0.638	1.000
BS 3915:1965	0.638	1.000
At hydraulic test		
ASME VIII:1965, factor 1.5	0.563	1.408
ASA B31-3:1966, factor 1.3	0.650	1.220
BS 1515:1965, factor 1.3	0.830	1.300
BS 3915:1965, factor 1.3	0.830	1.300

NOTE At the hydraulic test pressure, in each case the maximum permissible longitudinal stress behind the flange is in the same part of the plastic region, i.e. 1.2 to 1.4 × 0.2 % yield stress, when the nominal design stress is two-thirds of the yield stress.

References

1. BS 1560:1958, "*Steel pipe flanges and flanged fittings (nominal sizes ½ in to 24 in) for the petroleum industry*".

2. BS 10:1962, "*Flanges and bolting for pipes, valves and fittings*".

3. BS 1500, "*Fusion welded pressure vessels for general purposes*", Part 1:1958, "*Carbon and low alloy steels*" and Part 3:1965, "*Aluminium*".

4. BS 1515, "*Fusion welded pressure vessels for use in the chemical, petroleum and allied industries*", Part 1:1965, "*Carbon and ferritic alloy steels*" and Part 2:1968, "*Austenitic stainless steel*".

5. BS 3915:1965, "*Carbon and low alloy steel pressure vessels for primary circuits of nuclear reactors*".

6. ASA, B31.3:1965, "Petroleum refinery piping".

7. ASME Code Sec. VIII:1968, "Rules for construction of unfired pressure vessels", New York.

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12. Waters, E.O., "Analysis of bolted joints at high temperature", ASME Trans., 1938.

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21. Timoshenko, S. and Goodier, V.N., "Theory of elasticity", McGraw Hill Book Co. Inc., New York, 1951, Art. 23.

22. Kerhof, W.P., "New stress calculations and temperature curves for integral flanges", Proc. Third World Petroleum Congress, 1951, Vol. 8, p 151.

23. Westrom, D.B. and Bergh, S.E., "Effect of internal pressure on stresses and strains in bolted flanged connections", Amer. Soc. Mech. Eng. Trans., 1951, Vol.73.

24. Jaep, W.F., "A design procedure for integral flanges with tapered hubs", Amer. Soc. Mech. Eng. Trans., 1951.

25. Waters, E.O. and Williams, F.S.G., "Stress conditions in flanged joints for low-pressure service", ASME Trans., 1952.

26. Freeman, A.R., "Gaskets for high-pressure vessels", Mech. Eng., 1952.

27. Davis, J.Y. and Heeley, E.J., "Strains in flanged pipes", British Welding Journal, July 1955.

28. Lake, G.F. and Boyd, G., "Design of bolted flanged joints of pressure vessels", Proc.I.Mech. E., 1957, Vol. 171, No.31.

29. Donald, M.B. and Salomon, J.M., "Behaviour of compressed asbestos-fibre gaskets in narrow-faced, bolted, flanged joints", Proc.I.Mech. E., 1957, Vol. 171, No.31.

30. Stafford, J.A. and Gemmill, M.G., "Stress relaxation behaviour of chromium-molybdenum and chromium molybdenum-vanadium bolting materials", Proc.I.Mech. E., 1957, Vol. 171, No.31.

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33. Dudley, W.M., "Deflection of heat exchanger flanged joints as affected by barreling and warping", ASME Trans., 1960, Paper 60 — WA70.

34. Murray, N.W. and Stuart, D.G., "Behaviour of large taper hub flanges", Proc.I. Mech. E., 1961 Symposium.

35. Kraus, H., "Flexure of a circular plate with a ring of holes", July, Appl. Mech., 1962.

36. Bernard, H.J., "Flanges theory and the revised BS 10:1962", Proc.I.Mech. E., 1963, Vol.178, Part 1, No.5.

37. Usher, J.W.C., "Development of a flanged joint between stainless steel and aluminium piping for liquid oxygen service", Proc.I.Mech. E., 1963, Vol.177, No.28.

38. Lee, D.E., "New development in flange seals", ASME Trans., October 1963, Paper 63-Pet-28.

39. Korelitz, T.H., "Cut vessel flange cost by computer", Hydrocarbon Processing and Petroleum Refiner, July 1964, Vol.42, No.7.

40. DIN 2505, "Berechnung von Flanschverbindungen Entwurf", Marz 1961. (This is a method of calculation. Standard weld neck flange dimensions are given in DIN 2627 etc and standard flange resistances are given in DIN 2501 etc).

41. Siebel, E. and Schwaigerer, S., (V.G.B. Merkblatt No.4 of 1951).

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44. Bailey, R.W. "Bolted flange connections in the presence of steady creep", Engineering Vol. 144, 1937, No.364.

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46. Almen, J.O., "Tightening is vital factor in bolt endurance", Machine Design, February 1944, p 158–162.

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49. Downey, St.C. and Draper, J.H.M., Paper on conference on thermal loading and creep in structures and components, Proc.I.Mech. E., London, 1964.

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56. Krageloh, E., "Die wesentlichen Prüfmethoden für It-Dichtungen", Gummi und Asbest, 11, (1957), S.628.

57. Kenny, B. et al., "Stiffness of broad-faced gasketted flanged joints", J. of Mech. Eng. Sci., March 1963, 5, (1), 1–14.

The mechanism by which broad-faced flanged joints retaining a circular plate exert restraint against the flexure of the plate due to pressure differentials is discussed and studied experimentally. The theory proposed by Yi-Yuan Yu for determining the stiffness of an ungasketted joint is reconsidered and modified to suit the observed behaviour of metal-to-metal joints and of joints here one or more gaskets are included between mating surfaces of the joint assembly. Hence, a more exact method for calculating stiffness factors for such joint assemblies is formulated. Experiments were conducted on a particular design of header to tube plate assembly and the results used to check the validity of the modified theory.

58. "How to design orifice flange assemblies". Heating, Piping and Air Conditioning, June 1967, **39**, 137–42.

Gives details of butt welding neck, raised face orifice flange assembly. A table gives major overall dimensions for various nominal pipe sizes and pressure ratings.

59. Mostoslavskaya, V.M., "Temperaturnye napryazheniya v kompozitnom soedinenii trub". Fnergomashinostroenie, November 1965, 10–12. (In Russian.)

Thermal stresses in composite pipe joints; mechanically joined or welded pipe joints with conical contact surfaces made from materials of different coefficients of expansion; assuming that joint is represented by cylindrical shell of revolution, relationships are derived enabling calculation of thermal stresses and deformation; distribution of stresses among individual layers of composite joint.

60. "Manual of bolted flanges ring type", Design and Research Associates, 863 Pleasant Valley Way, West Orange, New Jersey, 1962, 25. (European Agent, J.F. Kelly, 31 Priory Grove, Still-organ, Co. Dublin, Republic of Ireland.)

Contains about 30 000 flange designs conforming to Section VIII, Appendix II, of the ASME Boiler Code.

61. Eichenberg, R., "Design of high-pressure integral and welding neck flanges with pressure-energized ring joint gaskets", ASME Paper No. 63-Pet-3, J. of Engineering for Industry, May 1964, **86**, (2), 199-2-4.

This paper provides all necessary information to design high pressure flanges with pressure-energized ring joint gaskets, for any pressure and made from any suitable material. These rules have been used to design the American Petroleum Institute Standard API-10 000 lb and 15 000 lb flanges.

62. Mueller, K., "Die Festigkeitsberechnung von Bördelflanschen", Stahibau, February 1966, **35**, 57–62. (In German.)

Stress calculation of pipe flanges; lapped-end pipes made of high-alloyed steel, light metals, or plastics are bolted together by means of a pair of unalloyed steel rings; method derived from statical design of boiler bulkheads by M. Esslinger (1952) is developed for stress calculation of these joints; method is based on treating separately cylindrical section of pipe, curved section of flange and straight extension of flange; relationships are derived enabling calculating of all section forces, deformations and internal stresses in pipe, flange, and rings.

63. Webjorn, J., "Flange design in Sweden", ASME Paper No. 67-Pet-20. 9pp.

Presents a new type of flange which is being developed in Sweden. It is more compact and lighter in weight than the current standards. The basic principles behind the design are explained and their application to the various components of the flange assembly. There is a discussion of the experimental work that was performed, together with other background information. The dimensions and working pressures that have been determined for a proposed flange series designed on these principles are also included. Briefly, these proposals take advantage of the newer steelmaking processes and the abilities of modern seals, such as O-rings, to make available an alternate series of pipe flanges to supplement those currently in general use. The principal features of this design are stiff, full-face, reduced-diameter flanges and slender, resilient bolts.

64. Spijkers, A., "Flange design and calculations", Ingenieur, s'Grav., 3.11.61, **73**, (44), W167.

Gives a general introduction to flange design; different types of flanges are considered, with theoretical estimates of flange strength, number and strength of bolts required for particular duties and approximate estimates of the torques which a flange can experience; numerical assumptions in some of the above methods are criticized.

65. Schuplyak, I.A., "Kraschetu plotnosti flantsevykh sosdinenii s prokladkami iz polimernykh materialov", NI. Taganov, Vestnik Mashinostroeniya, January 1966, 32–4. (In Russian.)

Schuplyak, I.A., NI. Taganov, Vestnik Mashinostroeniya, January 1966, 32–4. (In Russian.)

Design for tightness of flange joints with plastic gaskets; tongue and groove flange pipe joint with Teflon and h.p. polyethylene gaskets are theoretically investigated, assuming that flange deformation is negligible compared to deformation of bolts and gaskets; formula is derived expressing pressure that must be applied to gasket in terms of pressure in pipe, gasket width, and coefficient of joint rigidity.

Witten, A.H., "Flanged joints must be expected and tested", Power, January 1964, **108**, 62–3.

Recommendations are made to compensate reduction in bolt stress when component parts of flanged joint are subjected to variety of tensile and compressive stresses of different intensities, especially when temperatures are high and magnitude of stresses changes, resulting in lowering of bolt stress.

67. Meincke, H., "Principles of design of neck-welding flanges", VDI-Z, May 1963, **105**, 549–556.

The author states at the outset that the dimensions of flanges for pipes and apparatus are determined in Germany according to DIN-Vornorm 2505, in England and America according to the ASME-Code or TEMA-Standards (Tubular Exchanger Manufacturers Association) and that this takes a great deal of time. He therefore describes a method of calculation he has developed which simplifies the process without any loss in accuracy. At the same time it gives the economically best form of flange. In conclusion, he gives proof of the accuracy of his method.

68. GES, Pavlov, P.A., "Nesushchaya sposobnost flantsevykh soedinenii detalei", Fnergomashinostroenie, July 1965, 22–5. (In Russian.)

Load capacity of flange joints for hydraulic turbine elements and conduits of hydroelectric power plants; formulas for determining ultimate load capacity of flange joints connecting pipes subjected to axial tensile stress, twisting moment, and inner pressure; theoretical results are compared with experimental data.

69. Alexander, J.M. and Lengyel, B., "In cold extrusion of flanges against high hydrostatic pressure", Inst. Metals-J., January 1965, **93**, 137–45.

Cold extruding large metal flanges against fluid pressure to delay onset of instability and fracture in flanges was found successful in experiments with HC copper and commercial aluminium, which were extruded against 10, 20 and 25 ton/in^2 fluid pressure to three different flange thicknesses. Approximate mathematical solution for extrusion pressure was developed by using techniques of limit analysis. This showed good agreement with experimental results. Predicted values of extrusion pressure for harder material were analyzed and found to be within practical limits.

70. Levy, S., "Bolt force to flatten warped flanges", ASME Paper No. 63-WA-274, Trans. of the ASME J. of Eng. for Ind., August 1964, **86**, (3), 269–72.

Initial lack of flatness of the flanges of pipe connectors can result in leakage if the bolt loads are not sufficient to achieve positive gasket compression at all points on the circumference. Equations are presented for computing the magnitude of the bolt load necessary to flatten the flange. Account is taken of the bending and twisting resistance of the flange itself, the membrane and hoop bending restraint afforded by the pipe and the fact that the bolt circle is displaced from the gasket circle. The analysis applied to flanges whose warping can be adequately described by considering it to vary as cos 20. Numerical examples are considered for several typical flanges.

71. Schleeh, W., "A simple method of calculating flange stresses", Beton-u., Stahlbetonb., 1964, **59**, (3), 49–56; (4), 91–4; (5), 111–9.

Navier's concept of elementary stress is used as the basis of calculation, and combined with normal stress, σy, fulfills all limiting and equilibrium conditions. The correction function, including additional stress, necessary to achieve complete accuracy can be calculated for all possible stress states. Weighting factors of additional stresses for the important boundary loads are given and the simplicity and speed of the method is demonstrated by a number of examples.

72. Robinson, J.N., et al., "Development of ring-joint flanges for use in the HRE-2". (Oak Ridge Nat. Lab., Tenn.), December 21, 1961, Contract W-7405-Eng-26. 54pp. (ORNL-3165.)

Ring-joint flanges were studied in thermal-cycle tests as part of the development work associated with Homogeneous Reactor Experiment No.2 (HRE-2). The purpose of this study was to provide criteria for design, installation, and operation of joints that would remain leak-tight under reactor operating temperatures and pressures.

73. "Pipe connection", Chemical Engineering, April 26, 1965, **72**, (9), 183–4.

Intended to serve the same function as a flanged connection, this unit is fastened with only four bolts, thus allowing much faster assembly and disassembly. It is available in ½ through 30 in sizes for temperatures from – 43 °F to + 1 500 °F, and for pressures to 50 000 pounds per square inch. The units may be butt-welded, socket-welded or screwed directly into the process piping system. The device also features a blowout-proof metal seal ring, which is reusable. The connection is said to be one fourth lighter and to require less space than flanges. Bolt-hole alignment is eliminated since the unit can be rotated into any position.Standard materials are carbon steel of 304 stainless, but the clamp can be furnished in a variety of other materials. Gray Tool Co., Houston.

74. Ponthir, L., "Calculating the elastic deformation strength of pipe flanges", Chal. et Ind., March 1961, **42**, (428), 83–96. (In French.)

Whatever the shape and dimensions of a flange brazed to a pipe the maximum stress will always be located in the pipe close to the joint, and more attention must be given to this stress than to that obtaining in the flange. The joint bolts are subjected to bending stresses which are significant as regards deformation of the flanges. To obviate these difficulties the flanges should be designed for a substantial thickness and as small as possible force leverages so as to reduce the angle of rotation and increase the flexibility of bolts.

75. Thomas, W.M., "Up-to-date codes and standards cut cost of piping", Oil and Gas Journal, May 22, 1967, **65**, 113–7.

A review of petroleum industry codes and standards for valves, flanges and gaskets.

76. Watson, I., "Flange bolt design", Engineering Materials and Design, October 1964, **7**, (10), 687–9.

Discusses the general design of bolts for flanges subjected to bending.

77. Gitzendanner, L.C., et al., "Flanged omega seal and diffusion bonded connector designs", Proc.

SAE and Marshall Space Flight Centre Conf. on the design of leak-tight fluid connectors, August 1965, 177–85. (NASA-TMX-5785.)

Two semi-permanent flanged fluid connector designs, applicable to large diameter ducting systems and intended specifically for insensitivity of sealing to reduction of bolt load, are described. The first, an omega seal connector designed for 4 700 pounds per square inch service at 1 440 °F, incorporates a hermetic seal by the fusion-welding of two segments of a thin toroidal shell about the periphery of the connector. In order to make and break the seal, special welding and weld cutting equipment is required. In that an alternate load path exists for the compressive loading across the connector and in that the toroidal omega seal has inherent flexibility, the system has the ability to withstand flange displacements and rotations. The design is similar to that used by the United States Navy on its primary loop nuclear submarine systems. The second design, utilizing a diffusion bond as the hermetic seal, allows the seal to be made in the field by the application of moderate heat and bolt stress. The diffusion bonded flanged connector was designed for 6 000 pounds per square inch service at temperatures ranging between – 450 °F and + 100 °F. Both designs are described, along with their inherent advantages and disadvantages. The results of the programme in which a prototype of each design was manufactured and tested are described.

78. "Steel pipe flanges and flanged fittings", ASA, 1961, B16.5. 150, 300, 400, 600, 900, 1 500 and 2 500 lb.

79. Donald, M.B. and Morris, C., "Effect of flange design on gasket performance in narrow faced bolted joints", Second International Conference on Fluid Sealing, Paper A4, British Hydromechanics Research Association, Cranfield, U.K., April 1964.

80. Schneider, R.W., "Flat face flanges with metal-to-metal contact beyond the bolt circle", Journal of Engineering for Power, ASME Trans., Series A, Vol.90, No.1, January 1968, pp 82–88.

81. Waters, E.O. and Schneider, R.W., "Axisymmetric, nonidentical, flat face flanges with metal-to- metal contact beyond the bolt circle", ASME Paper No. 68-WA/PVP-5.

82. Discussion/Author's reply on ref. 80 above, Journal of Engineering for Power, ASME Trans., Series A, Vol.90, No.3, July 1968, pp 296–298. (This presents an alternate design procedure which eliminates solution by successive approximations.)

83. Pressure Vessel Research Committee, 1968 Annual Report, p 7, "Subcommittee on bolted flanged connections". Also ASME Paper 68-PVP-22, p 3, "Stresses in bolted flanged connections". Also information concerning PVRC Design Division Problems, Nos.XIV and XV (ASME Topic No.3) from Mr. C.F. Larson, Pressure Vessel Research Committee, 345 East 47th Street, New York, 10017.

84. Bickell and Ruiz, "Pressure vessel design", Part 16.4 on bolted flanged connections, 1967, Macmillan.

85. ASME Code Sec. VIII, Division 2, 1968, has pages on design and bolting of flanges.

86. TEMA Code, 1968, Part R8.21, method for adjusting design thickness of flange to allow for variation of E with temperature.

87. Unpublished document, Kemp, P.J., "Preliminary review of flange design".

88. Unpublished document, Strawson, J.W., concerning flange design to BS 1515-1, and comments on a letter of May 21, 1968, from Kemp, P.J., regarding higher permissible design stresses.

89. Unpublished document, Strawson, J.W., corrects dimensionless parameter proposed in ref. 88.

90. Private communication from I.C.I. Mond Division (Mr. J.G.H. Hills) March 3, 1969, to BSI with data indicating that flanged joints in which there may be some plastic behaviour when the bolts are fully tightened can be satisfactory.

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