LEAKAGE ANALYSIS OF GASKETED FLANGED JOINTS UNDER COMBINED INTERNAL PRESSURE AND THERMAL LOADING

Muhammad Abid*, Kamran Ahmed Khan, Javed Ahmad Chattha, and David H Nash**

Faculty of Mechanical Engineering,
Ghulam Ishaq Khan Institute (GIKI), Topi, Pakistan
**Department of Mechanical Engineering,
University of Strathclyde, Glasgow, Scotland, UK

ABSTRACT

Leakage in Gasketed Flanged Joints (GFJs) have always been a great problem for the process industry. The sealing performance of a GFJ depends on its installation and applied loading conditions. This paper aims to finding the leak rate through ANSI class#150 flange joints using a compressed asbestos sheet (CAS) gasket under combined structural and thermal transient loading conditions using two different leak rate models and two different bolt-up levels. The first model is a Gasket Compressive Strain model in which strains are determined using finite element analysis. The other model is based on Porous Media Theory in which gasket is considered as porous media. Leakage rates are determined using both leak rate models and are compared against appropriate tightness classes and the effectiveness of each approach is presented.

INTRODUCTION

Conventional gasketed-flanged pipe joints are widely used in the oil and gas and process industries for connecting pipes to pipes and pipes to other pressure equipment. For bolted flange joints, the two main concerns are joint strength and sealing capability. Currently available design methods and codes address only the structural strength of the flange joint under internal pressure only and do not fully consider the effect of sealing capability of the joints under transient thermal loading. When gasketed bolted connections are used in mechanical structures such as pipe flange connections and covers of pressure vessels in chemical plants, and the cylinder head in combustion engines for example, they are usually under thermal conditions. In the available published literature, the thermal behaviour of pipe flange joints is discussed under steady state loading with or without internal pressure [1, 5-6, 11,14] and under transient loading condition without internal pressure [7,13]. In the present study, the sealing behaviour of different sizes of gasketed bolted flanges joint under combined internal pressure and thermal transient loading is determined using finite element analysis and two available analytical leak rate models.

FINITE ELEMENT MODELLING

In order to address the problem, ANSI 150# flange sizes of 0.5", 1", 2", 3", 4", 10", 14", 18" and 24" are first analyzed numerically using the ANSYS® finite element programme under combined internal pressure and transient thermal loading. An elasto-plastic material model and temperature dependent material properties are also employed. In neglecting the holes in the flange and the presence of individual bolts round the flange, it is assumed that the system can be considered as a simple axi-symmetric system. This may be thought of in the form of a continuous bolt ring with the bolt load applied at each bolt location [12]. The resulting two-dimensional flange joint model is shown in Figure 1.

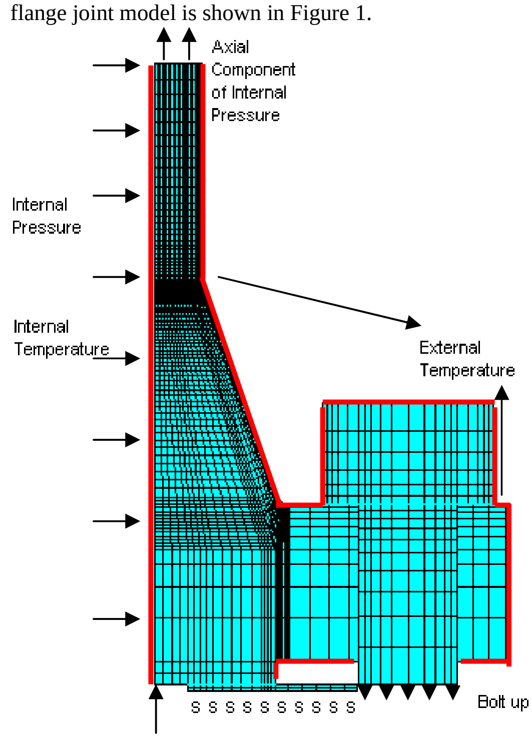

FIGURE 1 FE MODEL AND APPLIED BOUNDARY CONDITIONS

Structural element (PLANE82) is used for structural stress analysis of flange joint. The compatible, thermal element (PLANE77) is used to determine the temperature distribution. Two dimensional 'node-to-surface' CONTA172 contact elements, in combination with TARGE169 target elements are used to simulate contact distribution between flange face and gasket surface, the top of the flange and bottom of the bolt head in this study. No friction is employed between surfaces, since the forces normal to the contact surfaces are far greater than the shear force. The thermal and mechanical properties of flange and pipe, bolt, and gasket are noted from previous work by the authors [2].

Thermal Boundary Conditions

For the thermal analysis, convection with internal fluid temperature at the inside surface of pipe, flange ring and gasket and with ambient temperature at the outer surface of pipe and flange ring is applied. For any transient thermal boundary conditions as the loading is a time dependent phenomena, the initial condition is at when time t=0 seconds and temperature T=20^{o}C (ambient). For time t>0, convective boundary conditions with internal temperature of 100°C with convective heat transfer coefficient (150 W/m^2/^{o}C) at pipe inside diameter and ambient temperature of 20°C with convective heat transfer coefficient (20 W/m^2/^{o}C) are applied to the model as shown in Figure 1 and the analysis is run for 1500 seconds.

Structural Boundary Conditions

The flange is free to move in either axial or radial direction. This allows for flange rotation and the exact behaviour of stress in flange, bolt and gasket. A symmetry condition is applied to gasket lower portion. Bolt is constrained in the radial direction. Different target torques are applied to different flange sizes as specified by a typical process industry standard [8] and the gasket manufacturer (KLINGER [16]) as shown in Table1. The associated ASME code (Section VIII, 2010) [15] does not specify a magnitude of preload for the bolts, only a minimum seating stress that relates to the gasket style and composition. A certain displacement is applied to lower surface of the bolt to obtain pre-stress in the bolt. After pre-stress application, an internal pressure of 1.8MPa (18 bar) [8] is applied at inside diameter of flange and gasket and loading due to the head (end cap) is directly applied as nodal forces at the end of the pipe [12].

TABLE 1 TARGET TORQUE VALUES

Serial No	Flange Size (Inch)	Gasket ID (a) (mm)	Gasket OD (b) (mm)	Target Torque (N-m) gasket manufacturer	Target Bolt Stress (MPa)	Target Torque (N-m) industry	Target Bolt Stress (MPa)
1	0.5	14.224	34.9	34.34	106.73	35	108.80
2	1	26.924	50.8	34.34	106.73	50	108.80
3	2	55.626	92.1	67.60	107.50	130	206.70
4	3	81.026	127.0	67.60	107.50	130	206.70
5	4	106.426	157.2	67.60	107.50	130	206.70
6	10	268.224	323.8	188.44	109.20	300	173.90
7	14	349.250	412.7	281.82	109.50	500	194.20
8	18	449.326	533.4	398.44	108.70	705	192.30
9	24	603.250	692.1	555.74	110.50	985	195.90

LEAKAGE ANALYSIS

Leak rate model based on Gasket Compressive Strains (LR-1)

Leak from a flange joint is observed when there is insufficient contact stress on the gasket to maintain a seal during operation condition. When a flange joint is bolted, the bolt load compresses the gasket and helps to maintain a specific gasket stress. Whenever there is a situation of loss of gasket stress, leakages may occur due to the sealing capability compromised. In order to estimate the leak rate,

Kobayashi [10] proposed a mathematical model based on the compressive strain of a gasket. This methodology also simplified the test procedure by eliminating a complex loading-unloading sequence on the gasket. The proposed gasket strain formulation also incorporates the thermal loading on flange joint to estimate the leak rate.

The whole concept for finding leak rates through flange joints is based on two variables, internal pressure and gasket strains. It was found out through experimentations that leak rates vary directly with internal pressure and may be higher at high internal pressures. Leak rate was observed decreasing with the increasing strains value in the gasket. A particular equation is derived from the test data of CAS gasket from experiments as given by Kobayashi [10].

$$L = \pi\lambda 2^n \left(\frac{n-1}{n}\right)^n h\left(b^{\frac{n-1}{n}} - a^{\frac{n-1}{n}}\right)^{-n} P^n$$

where, $\lambda = c \cdot e^{\alpha\cdot\varepsilon}$ and β is the shape factor which is defined as

$$\beta = h\left(b^{\frac{n-1}{n}} - a^{\frac{n-1}{n}}\right)^{-n}$$

P = internal pressure (MPa); ε = compressive gasket strain; h = gasket height (mm); b= gasket outside diameter (mm); a = gasket inside diameter (mm); β = Shape Factor; c, n and α are constants and are dependent on the gasket type.

For the non asbestos compressed fibre sheet gasket, the values of these constants are 0.474, 1.35 and -57.8 respectively [10]. The main input in this model is the gasket strain 'ε' which is calculated from the finite element analysis. This leak rate model is used for both the gases and liquids. A detailed flow chart for the leak rate determination based on gasket compressive strain is shown in Figure 2.

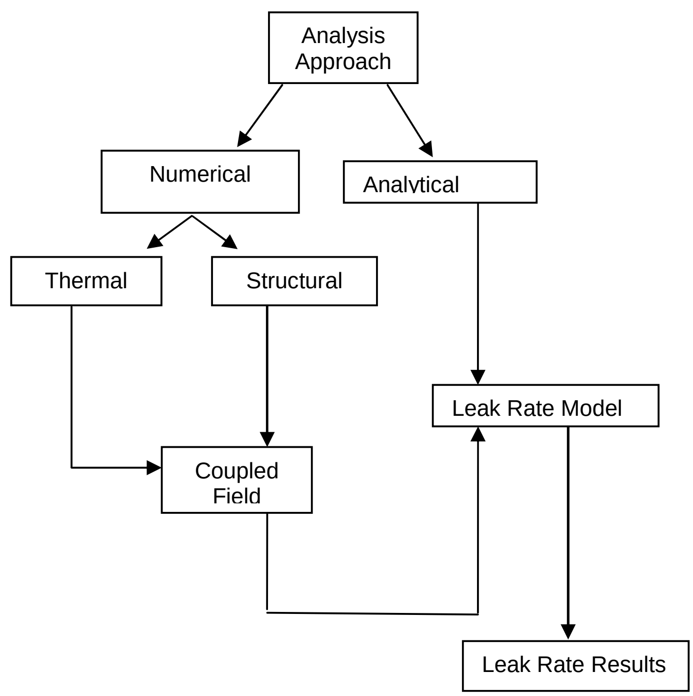

FIGURE 2 ANALYSIS FLOW CHART USING GASKET COMPRESSIVE STRAINS LEAK RATE MODEL (LR-1)

Leak rate model based on Porous Media Theory (LR-2)

The leak prediction is also carried out by using porous media theory Jolly and Marchand [9] by integrating Navier-Stokes equations with first order slip flow boundary conditions. The derived formula (iii) is applicable to the wide variety of gasket material in which the gasket permeability is the main factor to determine leak rate.

$$L_{rm}^{SFI} = \frac{\pi t k_v P_o^2}{\mu_g R_g T} \frac{P_*^2 - 1}{\ln \frac{r_0}{r_i}}$$

Here, L_{rm} = mass flow rate (Kg /sec); t = gasket thickness (m); k_v = gasket intrinsic permeability (m^2); P_o = ambient pressure (MPa); P_* = ratio of the inlet over outlet pressures; μ_g = dynamic Viscosity (Pa s^{-1}); R_g= specific ideal gas constant (J Kg^{-1} K^{-1}); T= temperature (K); r_o,r_i = outer and inner radius of gasket.
The work for leak rate prediction for liquids is underway by authors in [9] to validate the hypothesis of using gasket permeability from reference gas test. This leak rate model is limited to the gases only; however preliminary work has been started for the liquids.

RESULTS AND DISCUSSION

Leak rate model based on gasket compressive strains (LR-1)

Leak rate is determined from the flange joints using both the types of fluids i.e. gases and liquids. For liquids, crude oils of densities 915 and 973 Kg/m^3 are used. For gases, helium and nitrogen with density 0.1786 and 1.251Kg/m^3 respectively are used. For comparison, KLINGERSIL®C4400 sheet gasket is used for crude oil, helium and nitrogen gas. The leak rates were calculated by taking initial contact area and effective contact area of the gasket (Figure 3).

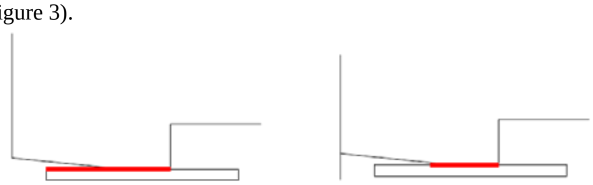

FIGURE 3
(A) INITIAL CONTACT AREA (B) EFFECTIVE CONTACT AREA

Figures 4-7 show that leak rates for effective contact area are less than the initial contact area of the flange joint. Fig 4 and Fig 5, shows variation in the leak rates for each size of flange joint.

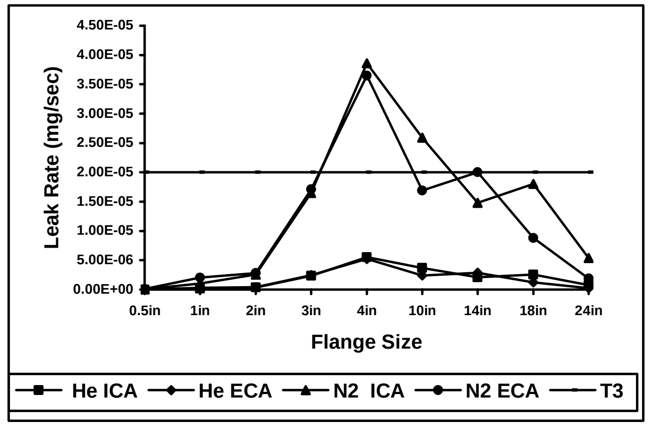

FIGURE 4 LEAK RATE OF GASES USING TORQUES RECOMMENDED BY INDUSTRY (ICA= INITIAL CONTACT AREA, ECA= EFFECTIVE CONTACT AREA)

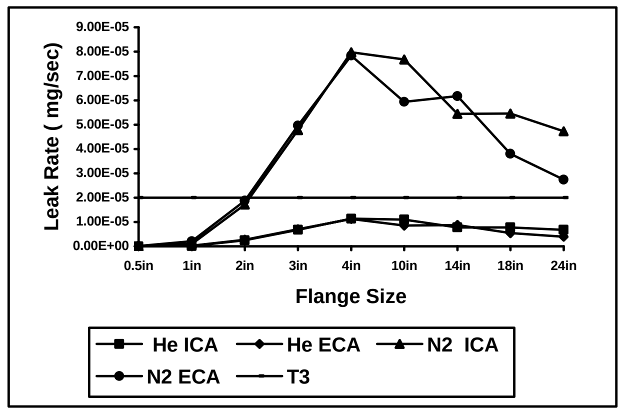

FIGURE 5 LEAK RATE OF GASES USING TORQUES RECOMMENDED BY GASKET MANUFACTURER (ICA= INITIAL CONTACT AREA, ECA= EFFECTIVE CONTACT AREA)

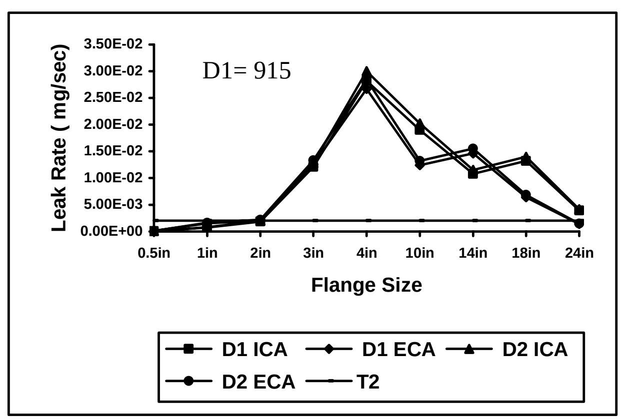

FIGURE 6 LEAK RATE OF CRUDE OIL USING TORQUES RECOMMENDED BY INDUSTRY (ICA= INITIAL CONTACT AREA, ECA= EFFECTIVE CONTACT AREA)

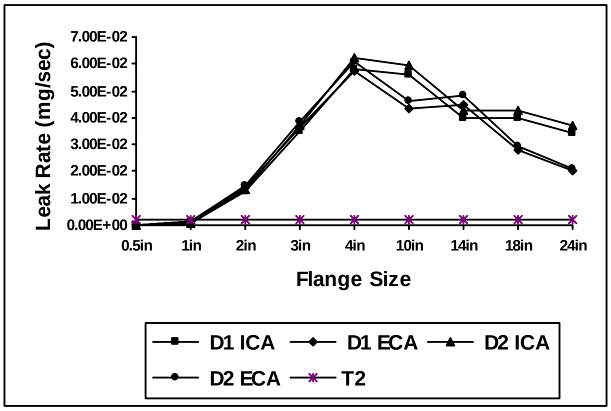

FIGURE 7 LEAK RATE OF CRUDE OIL USING TORQUES RECOMMENDED BY GASKET MANUFACTURER (ICA= INITIAL CONTACT AREA, ECA= EFFECTIVE CONTACT AREA)

It is noted that the leak rate of nitrogen is more than the helium for the same target torques as specified by ES/090 Rev 1 and KLINGER. A maximum leak of 3.86 and 7.98 mg/sec observed for the flange size of 4 inch for the target torques specified by industry and gasket manufacturer respectively. The difference between the leak rates is due to the level of recommended torque which is higher for the industrial standard, hence generating higher compressive gasket strain and hence lower leak rate occurs. The leak rate of flange joint size 4", 6" and 8" are above the tightness class T3 as noted by Bickford [4] but the leak rate tends to decrease with the increase of flange size and comes to the T3 regime. However, for the gasket manufacturer torque values, the leak rate of flange sizes from 2" to 24 " are more than T3 tightness class and lies in T2.5 class however, the leak rate of helium gas lies in T3 class for both the target torque values. For the crude oil case, only 0.5" and 1" flange size lies in T2 class for both the target torques recommended and the leak rate for the remaining flange sizes remains in T1 class which shows that the target torque values are recommended for tightness class T1. To improve the leak rate, the target torque should be adjusted to meet the higher tightness class requirements.

Leak rate model based on Porous Media Theory (LR-2)

Using porous media theory, high leak rates are observed for gases compared to the gasket compressive strains theory for the same gasket and boundary conditions [Figure 8].

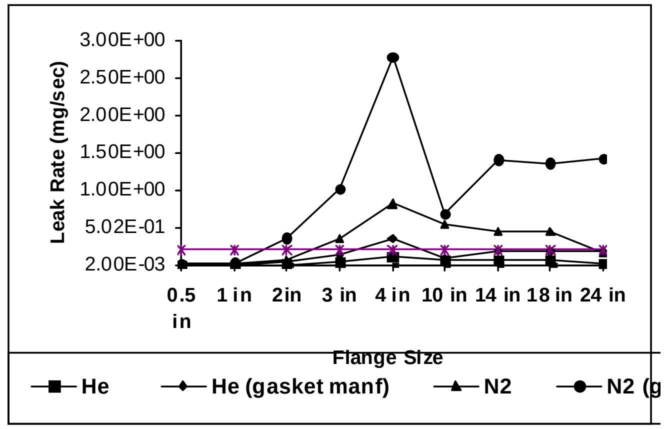

FIGURE 8 LEAK RATES OF GASES USING TORQUES RECOMMENDED BY INDUSTRY AND GASKET MANUFACTURER

For helium gas, all flange sizes qualified tightness class T2 except flange size of 4in which lies in tightness class T1 for target torques recommended by the gasket manufacturer. For nitrogen gas, only flange size of 0.5,1 and 2 inch qualifies for T2 class and remaining all flange sizes lie on tightness class T1 for both the target torques. It is obvious that the higher leak rate is obtained for the gasket manufacturer's recommended target torque as compare to the industrial standard recommended torques. For the industrial standard recommended target torque, the maximum leak rate of 8.19E-01 mg/sec is observed, whereas for the gasket manufacturer's recommended target torque leak rate observed is 2.78 mg/sec for flange size of 4 inch. It is again due to the higher value of industrial recommended target torque than the gasket manufacturer.

CONCLUSIONS

The following conclusions and observations were found.

For ANSI Class#150 flanges, different leak rates are calculated for different flange sizes. This gives rise for concern in that there is no scalability across the range i.e. different sizes within the class perform better than others.

A maximum leak rate was found for flange joint size of 4 inch using both the leak rate models.

Compressive strain based model predicts more leak rate as compared to the porous media bases leak rate model.

Compressive strain based model is applicable to liquids and gases but the gasket type should be compressed asbestos sheet whereas the porous media based model is applicable to any type of gasket but it is validated to gases only. For liquids, the model is yet to be validated.

For the case of gas, the maximum leak rate is observed for 4in flange size for nitrogen gas which is 7.98E-8 Kg/sec for LR-1 and 2.78E-06 kg/sec for LR-2 with gasket manufacturer's recommended target torque.

For the case of crude oil, the leak rate reduced by 50% when industrial recommended target torque is applied.

For the same target torque, flange sizes meet T2 tightness class requirement for liquids but for the case of gasses, flange sizes qualified T3 tightness class requirement due to difference in their densities.

It may be conclude that the industrial recommended target torque is better than the gasket manufacturer target torques due to higher gasket stress achieved which cause low leak rates.

ACKNOWLEDGEMENT

The development of this work has been supported by Higher Education Commission (HEC), Pakistan. The use of the ANSYS finite element code is used under an educational licence.

REFERENCES

[1] Abid, M. and Baseer Ullah. (2007a). "Three-dimensional nonlinear finite element of gasketed flange joint under combined internal pressure and variable temperatures", *J. of Engineering Mechanics*, 133 (2):222-229

[2] Abid,M., Chattha, J A., and Khan, K A. (2007b), "Finite element analysis of a gasketed joint under combined internal pressure and thermal transient loading", ASME Pressure Vessel and Piping Conference, San Antonio, Texas, USA, July 22-26,2007, Conference Paper PVP2007-26602, 1-7

[3] ANSYS Inc. (2004). ANSYS Element Manual, Seventh Edition

[4] Bickford, John H. (1998). "Gaskets and Gasketed Joints", Marcel Dekker Inc, ISBN 0-8247-9877-5

[5] Bouzid, A-H., Nechache, A., and Brown, W. (2002), "The effect of steady state thermal loading on the deflections of a flanged joint with a cover plate", ASME Pressure Vessel and Piping Conference, Vancouver, British Columbia, Canada, August 5-9,2002,Conference paper PVP2002-1093, PVP-Vol.433:153-162

[6] Brown, W., Derenne M. and Bouzid, A-H. (2000). "Determination of the steady state operating temperature of pressure vessel flange components: Part-1- Analytic Methods", ASME Pressure Vessel and Piping Conference, Seattle, Washington, July 23-27,2000, PVP-Vol.405:95-104

[7] Brown, W., Derenne, M., and Bouzid, A-H. (2002), "Determination of Gasket stress levels during thermal transient", ASME Pressure Vessel and Piping Conference, Vancouver, British Columbia, Canada, August 5-9, 2002, Conference paper PVP2002-1078, PVP-Vol.433:21-28

[8] ES/090 Rev: 1. Design and Engineering Practices (DEPs), DEP 31.38.01.15.Gen (Piping Class Exploration and Production)

[9] Jolly, P. and Marchand, L. (2009), "Leak Predictions for static gasket based on the porous media theory", *J Pr. Vessel Tech., Trans of the* ASME, 131:021203-1 to 021203-6

[10] Kobayashi, T., Nishida, T., Suzuki, M., and Yamanaka, Y. (2000), "Leak Tightness Evaluation of Gaskets Based on Compressive Strain", ASME Pressure Vessel and Piping Conference, Seattle, Washington, July 23-27,2000, PVP-Vol.405:23-27

[11] Kumano,H., Sawa T., and Hirose,H. (1994) "Mechanical behaviour or bolted joint under steady heat conduction", ASME *J. Pressure Vessel Technology,* 115(2):142-146

[12] Nash, D.H., Spence J., Tooth A.S., Abid M., and Power D.J. (2000) "A parametric study of metal-to-metal full face taper-hub flanges", *Int. J. Pres. Ves. & Piping,* 77: 791-797

[13] Sawa, T., Hirose, T., and Kumano, H. (1993) "Behaviour of pipe flange connection in transient temperature field", ASME *J. Pressure Vessel Technology*, 115**:**142-146

[14] Sawa ,T. Takagi, Y., and Tatsuoka, T. (2005), "Thermal stress analysis and evaluation of sealing performance in pipe flange connections with spiral wound gaskets under elevated temperature and internal pressure", ASME Pressure Vessel and Piping Conference, Conference paper PVP2005-71425, 105-113

[15] Section VIII, (2010). ASME Boiler and Pressure Vessel Code, "Design Considerations for bolted flange connections", Appendix-2, ASME, NY, USA

[16] www.klinger.co.uk