MECHANICAL ENGINEERING NEWS

COADE, Inc.

For the Power, Petrochemical and Related Industries

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The **COADE** Mechanical Engineering News Bulletin is published on a quarterly basis from the **COADE** offices in Houston, Texas. The Bulletin is intended to provide information about software applications and development for Mechanical Engineers serving the power, petrochemical and related industries. Additionally the Bulletin will serve as the official notification vehicle for software errors discovered in those Mechanical Programs offered by **COADE**.

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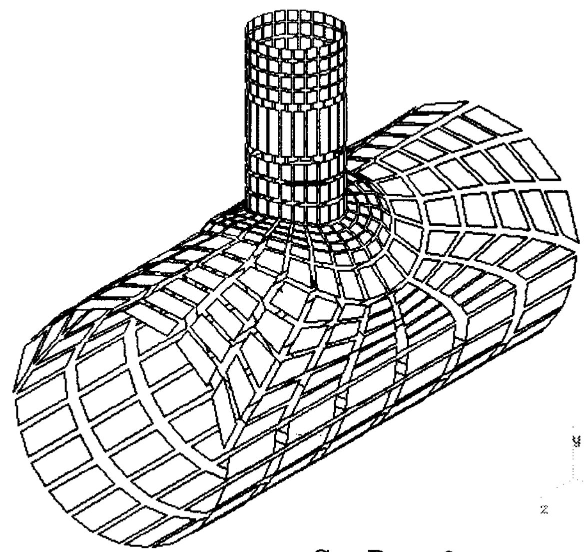
COADE ANNOUNCES FE/PIPE



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PC Hardware For The Engineering User (Part 12)

Many of our users have suddenly reported file access problems running COADE software products. These file access problems ranged from "file not found" to "not enough space on target drive". In all cases, the file could be located with DIR, and there was ample space on the target hard drive.

These problems are a result of executing the software on drive partitions larger than 32 Mbytes under DOS 4.0x. Furthermore, the problems discussed above are not limited to COADE software, virtually all software executing on *large* partitions has the potential to experience disk access problems.

The problem is caused by the inability of DOS 4.0x to properly address drive partitions larger than 32 Mbytes. The solution to the problem is to use the DOS **SHARE** program. The official word on SHARE is that it is for networked systems. However, a perusal of the DOS manual reveals a statement to the effect that SHARE must be loaded for proper access to large drive partitions. In addition, during the machine boot process, a warning message is displayed stating that the SHARE should be loaded. In short, SHARE solves the large partition access problem.

The best way to insure that SHARE is always installed is through a modification to the AUTOEXEC.BAT file. Simply edit this file and add a line containing the

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Benchmarking CAESAR II and ANSYS

COADE has acquired a copy of the **ANSYS** general purpose finite element program for use in developing interfaces with **CAESAR II** and **FE/PIPE**. In order to start the **CAESAR II/ANSYS** interface, several piping runs were made in both **CAESAR II** and **ANSYS** for the purposes of benchmarking the two programs. Listed below are the major differences between the two programs that will cause the results to vary by up to 20%.

1) When using the **ANSYS** piping input module of PREP7, be aware of the modeling assumptions made. Typical piping programs (such as **CAESAR II**) generate bend radii based on 1.5 times the nominal diameter. **ANSYS** generates bend radii based on 1.5 times the outer diameter. This means that for pipe sizes smaller than 14 inches, the two programs will generate different models in the default generation mode.

2) Proper modeling of rigid elements has always been an important consideration when analyzing a system on two different programs. **CAESAR II** computes the stiffness of rigid elements by increasing the element thickness by a factor of 10, keeping the inner diameter constant. NUPIPE obtains the stiffness of rigid elements by multiplying the standard element inertia by 3. **ANSYS** uses a flexibility factor of 0.5, which makes the element twice as stiff as a standard pipe element.

3) **ANSYS** assumes closed ended pipes, which includes an axial pressure force in the analysis. In order to perform in a similar fashion, **CAESAR II** must be set to use Bourdon option #2.

4) When using the **ANSYS** pipe modeler, and changing temperature at a branch, **ANSYS** also changes the temperature of the two header elements which make up the tee. In order to generate the same model in **CAESAR II**, the tee must be modeled as three pieces of pipe and the temperatures set appropriately.

Status of CAESAR II

Due to the extensive modifications required to **CAESAR II** for the Version 3.2 release, COADE has released intermediate version 3.15. The 3.15 release includes the new ESL protection scheme, an SIF computation module, a WRC-297 module, a flange leakage estimator, a new pen plotting program, and various other minor enhancements.

Development work on **CAESAR II** is continuing and releases will be forth coming which contain many user suggested modifications. Version 3.16 which is scheduled for release late this year will include: the Stoomwezen piping code, 3 additional hanger tables, a new file handler, a configuration program for modifying the setup file, and a revised AISC program.

Status of CodeCalc2

Version 5.0 **CodeCalc** has been completed and shipped during the last week of June. The most notable feature of the Version 5.0 release is that **CodeCalc** is now a stand alone FORTRAN program, it no longer requires Lotus to execute. For this reason, **CodeCalc** has been renamed **CodeCalc2**. The other features of this release are listed below:

- A90 code addenda
- Standard COADE units systems and manipulation
- Standard COADE help facility
- Support of the DOS environment
- User control of data bases and program configuration
- Faster execution times
- Improved output review abilities and vessel summaries

In early September, **CodeCalc2** Version 5.01 shipped to all recipients of Version 5.0. Version 5.01 corrected any reported problems with Version 5.0.

Flange Leakage Applications

The 3.15 version of **CAESAR II** included a completely new flange leakage calculation. Stresses in piping systems have been addressed via computer solutions since the late 60's, loads on rotating equipment have been addressed by manufacturers, NEMA and API, but leaking flanges have never been addressed in a practical, usable analytical manner. The development of the **CAESAR II** flange leakage model is the first small step in the evolution of a practical, analytical tool for flange leakage prediction.

This *Mechanical Engineering News* article was written to help the engineer properly apply the new program.

The basic problem of flange leakage is a complex one not readily availing itself to analysis. Facing selection, gasket type, operating temperature, and initial makeup moments are all factors that are either difficult or impractical to evaluate analytically.

There is some concern today that flange mismatch tolerances contribute to a majority of the flange leakage problems. For example, if a flange is placed in a stiff part of the piping system, and the standard mismatch angular tolerances exist, there will be more moment on this flange joint once the bolts are tightened than if the flange had been placed in a flexible part of the piping system.

The **CAESAR II** flange leakage model assumes that the user has already selected the gasket, has a flange design, and has analyzed the piping flexibility to compute the forces and moments exerted by the piping on the flange, (possibly including the effects due to tolerance, which is possible using restraints with connecting nodes and forces).

Once this information is available the user is ready to enter the flange leakage calculation to determine the flanges tendency to leak.

The ASME codes eliminate some of the decisions involving leakage by the publication of the gasket "m" factor. The "m" factor is the leak pressure ratio. This is the pressure on the gasket to prevent leakage over the line pressure, times a safety factor. These values are currently the subject of close scrutiny by many organizations, but the existing values have been used with a reasonably successful design history. It is with the "m" factor that the **CAESAR II** flange leakage calculation starts and depends.

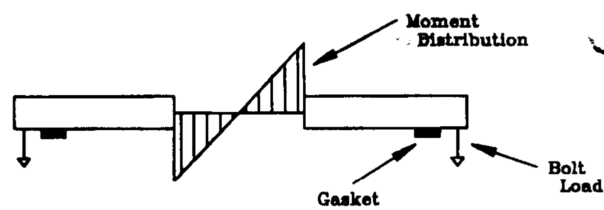
The flange modeler determines the initial pressure on the gasket due to the tightening of the bolts, and the loss of pressure on the gasket due to the line pressure and the forces and moments that act on the flange. If the resulting pressure on the gasket, (i.e. the initial minus all losses), is greater than the gasket factor "m", times the line pressure then the flange is "safe".

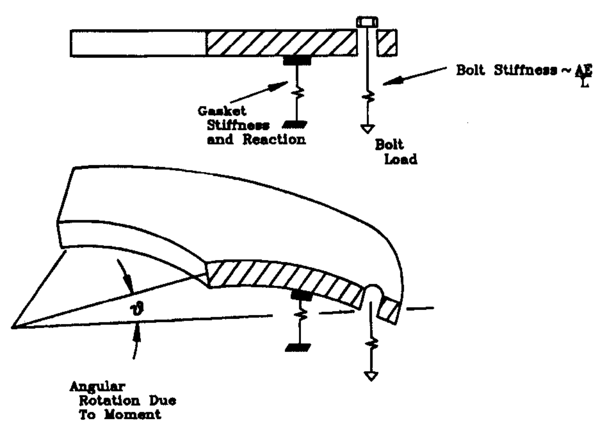
There are a great many different flange types, facings and gaskets. All of these were generalized into one model for leakage. Once this was done the critical variables affecting leakage had to be retained in the analytical model and the unnecessary variables eliminated. It was determined that the deformation of the annular plate forming the flange, in conjunction with the deformation of the bolts and gasket, when subjected to bending, pressure and axial forces were the critical variables to be evaluated.

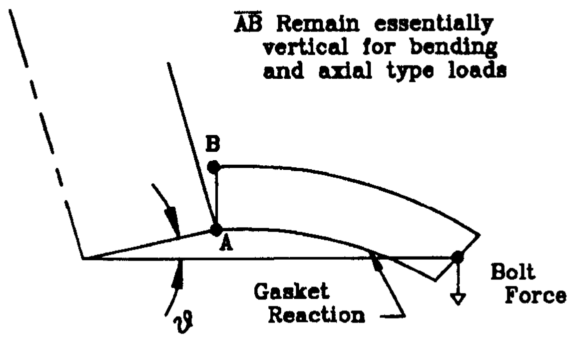
The flange modeler determines the initial pressure on the gasket due to the tightening of the bolts, and the loss of pressure on the gasket due to the line pressure and the forces and moments that act on the flange.

Various simplified elastic models were tested and a final model agreed upon that most closely correlated the results from a finite element analysis of several typical flange configurations subject to bending and axial loads.

The basic flange deformation modes assumed to contribute most significantly to the unloading of the gasket are shown in the sketches below.







The limitations of the model are that:

1 - The gasket reaction and stiffness is concentrated at a point load at the center of the gasket loading area.

2 - The bolt reaction and stiffness is concentrated at a single point and is assumed to be uniformly distributed around the annular plate that models the flange.

3 - The pipe/hub interface is assumed to be flexible enough to allow rotation at the flange id at the point around the circumference where the bending moments produce a maximum stress in the pipe, so that the absolute rotation at the flange id is zero.

4 - The gasket is assumed to be fairly stiff, so that the flange rotational stiffness is of the same order of magnitude as the gasket stiffness.

These analytical limitations imply other more practical "usage" limitations:

1 - Full face gaskets cannot be modelled.

2 - Leakage at self-energizing gaskets cannot be predicted.

3 - Leakage for flanges with ring-type joints cannot be predicted.

4 - Shear load effects on leakage are ignored.

5 - The effect of the hub and pipe wall are not variable, and so are considered only approximately.

6 - Leakage for joints made up of flexible gaskets should not be attempted. (The effect that very flexible gaskets have on leakage tends to be a function of other factors rather than the flexibility of the annular flange plate, and bolts.)

Upon testing of the flange leakage feature favorable performance was obtained when compared to three-dimensional finite element analysis. Loads on the gasket were predicted within 15% for standard dimensioned flanges. Similarly the trends indicated by the modeler seemed correct. When used for some of the flange leakage problems addressed by COADE Engineering, the results gave comfortable correlations, i.e. when a flange was reported leaking in the field, and the moments on the flange were calculated using **CAESAR II**, and those moments and dimensions were inserted into the flange routine, the flange routine predicted leakage also. Typically smaller flanges tend not to leak, while large diameter flanges tended to have much smaller safety factors. This is born out in practice, and seems intuitively correct.

Because of the safety factor inherent in the code's "m" factor, the user of the flange program is recommended to shoot for designs where the safety factor is greater than 1.0. This should provide an actual safety factor of greater than 2.0, and is consistent with other safety factors used in pipe stress analysis. If the **CAESAR II** flange leakage program predicts safety factors less than 1.0 then the loads on the flange should probably be reduced.

Before the **CAESAR II** flange modeler piping engineers had only stress calculation mechanisms available for predicting a flanges service usefulness. None of these methods addressed the problem of the deformation of the flange and its effect on gasket compression and leakage. This is evident by the ANSI/API leakage minimum and maximum safety factors printed in the **CAESAR II** flange leakage report. For lack of a better method, a "leakage" check has been performed in the past by use of the "rating table" method, (which was intended to predict stress, not leakage). ANSI B16.5 and API 605 have as part of their flange specifications, rating tables of allowed pressure and temperature for different diameters and materials of flanges. Previous practice was as follows:

1 - Compute the equivalent pressure on the flange due to the moment and force that the piping system exerts on the flanged joint.

2 - For the flange operating temperature read the allowable pressure from the code rating table.

3 - Compare this rated pressure to the equivalent pressure computed in step 1. If the equivalent pressure was greater than the allowed pressure then "leakage" was supposed to have occurred.

ASME Section VIII, Division 1 flange calculations intend to assure that the flange will not be overstressed by the necessary tightening loads that are required for a leak tight joint. The ASME VIII calculations do not attempt to study the flexibility problem in the annular flange, or how it relates to leakage. Neither the ANSI B16.5, the API 605, nor the ASME VIII methods try to predict leakage. That an attempt was made to predict leakage using the API/ANSI rating tables emphasizes the fact that some type of leakage predictor is much needed by the industry.

This is clear when the user of the **CAESAR II** flange program looks at the ANSI/API safety factor minimums and maximums. Leakage is a function of elastic modulus more than yield stress. The rating table method is a stress calculation based essentially on yield stress. As different materials are used their yield strengths change, and so their "predicted" tendency to leak will also change. Since most steels have approximately the same modulus of elasticity, the actual variation in the tendency to leak that should exist between different classes of steels should be fairly small. The minimum and maximum ANSI/API safety factors printed in the flange report are the safety factors from the highest and lowest strength steels in the rating table. These values vary considerably.

Leakage is a function of elastic modulus more than yield stress.

The **CAESAR II** method for predicting flange leakage is in its infancy. It should be used with a critical design eye, to make sure that its results "makes sense" and are inline with common design practice. Suggestions for its improvement are strongly encouraged.

Flange Allowable Stresses

(This article is contributed by COADE Engineering Consulting to COADE Engineering Software Development for publication in Mechanical Engineering News.)

COADE Consulting has discharged a number of projects, as shown below, dealing with flanged joint design and leakage evaluation problems.

EXXON Chemical Americas - Baytown
Shell - Wood River
General Electric Silicones - Waterford
Monterey Mechanical Company — Oakland

In each engagement, the "equivalent" pressure was used to quantify the external moments and forces. Although ANSI/ASME Sect. VIII, Division 1 does not address the equivalent pressure subject, a number of other ASME Codes have sanctioned its applications. One source is in ASME Section III, Division 1 - NC (Exhibit 1) which defines FLANGE DESIGN PRESSURE (Pfd) as the summation of the internal pressure and external "equivalents". Another is in ANSI B31.1, Appendix II (Exhibit 2). In both references, the stress allowables for the "equivalent" loading are set higher than the flange stress allowable limited by the internal pressure loading alone.

This article will center on the stress allowables for the flanged joint, including its bolting.

Flange Allowable Stresses

The **PROVESSEL** and **CodeCalc2** programs provide derivations for "equivalent" pressure. But both programs treat the equivalent loading as its operating pressure, thus rendering an overly conservative design.

The following TABLE will illustrate the differences between the COADE Vessel programs and some code sanctions:

Stress Allowables for Equivalent Pressure Loading

	PROVESSEL & CodeCalc2	NC Section III Division 1	ANSI/ASME B31.1
Longitudinal Hub, Sh	1.5 SA	1.5 SA	1.0 SY-Slp
Radial Flange, Sr	1.0 SA	1.5 SA	1.0 SY
Tangential Flange, St	1.0 SA	1.5 SA	1.0 SY
Maximum Average	1.0 SA	-	-
Other			Below ANSI B16.5 rating

A case in point can be made for slip-on flanges; often the tangential flange stress is the weak point. Here, a 50% increase in allowable can lead to a significant design saving for flange programs that recognize the difference.

Please note that 1.5 SA would approximate the yield strength (SY) of the material. In Sect. VIII, Div. 1, SA is the lesser of 2/3 of yield and 1/4 of ultimate tensile.

Bolting Allowable Stresses

What happens to the bolting allowable? Here, controversies abound. Sect. VIII, Div 1 mandates some extremely low bolting allowables for seating and operating stresses. For instance, B 193 bolts use only about 1/4 of yield strength as allowables per Code. Since common practice dictates that bolting "preload" be as high as possible (e.g., 90% of yield strength), Code bolting allowable is a de facto rule of confusion. Appendix "S" in Sect. VIII Div. 1 tries to rectify the Case. To many readers it eludes the matter more.

On bolting allowables for equivalent loading and preload, bolt allowables should probably proceed in a manner consistent with the principle of "sustained" and "displacement" stresses. The following sections demonstrate this principle.

(1) Operating and Seating Stresses

Use the Division 1 Code Allowables, as currently applied in the **PROVESSEL** and **CodeCalc2** programs.

(2) Equivalent Bolt Stress

Use 1.5 times the base SA at temperature for bolting allowable. This allowable relates to the Operating case only. Be aware that since the base SA for most high strength bolts approximates 25% of yield strength, the equivalent allowable is only raised up to 37.5% of bolt yield strength at temperature. For some low strength bolts whose allowables correspond to 2/3 of yield strength, this could lead up to, but not more than, a full yield stress level at temperature.

Be aware that since the base SA for most high strength bolts approximates 25% of yield strength, the equivalent allowable is only raised up to 37.5% of bolt yield strength at temperature.

To support this increase in allowable, we can refer to the Appendix 4 of Sect. VIII, Div. 2. In paragraph 4-141 (Exhibit 3) the Div. 2 Code permits bolt stress allowables up to twice the base stress value under the combined stress of (a) Preload, (b) pressure, and (c) differential thermal expansion. Observe that the equivalent stress calculation does not consider any preload, nor applies the twice stress value.

(3) Preload Stress

Before defining the stress allowable for bolting preload, it is important to categorize preload stresses. To illustrate, assume a pipe fitter tightened a flanged joint. He in fact has cold sprung a certain elongation or "strain" into the bolt. Further assume the materials of bolts and companion flanges are similar, this "strain" should remain relatively constant throughout the thermal cycles. Of course, the bolt stress varies because of the changing modulus of elasticity as a function temperature. Since the Preload bolt stress is self-limiting (i.e., strain dependent) and cyclic, it is a "displacement stress". As we *hold that the flanged joint be designed for the same cycle life as the connecting pipe, the Preload bolt stress should be confined by the same pipe stress range allowable—i.e., between* $f(1.25\ Sc + .25\ Sh)$ *and* $f[1.25\ (S_c + S_h) - S_l]$.

To see the relationship between Preload and equivalent stresses, take an A-193 B5 bolt as an example. It has a SMYS of 80 ksi and an SA of 20 ksi (i.e., 25% of yield). Assume the flanged joint operates at ambient, the Preload would be 1.25 x 20 + .25 x 20 = 30 ksi. Note that this Preload stress is actually the same as the maximum permissible equivalent stress (i.e., 1.5 * 20).

But if the bolt is designed for 1150 F, then "Sh" would be 2.0 ksi. In this case, the Preload would be 25.5 ksi whereas the permissible equivalent stress is only 1.5 x 2.0 = 3.0 ksi. Here the Preload stress substantially exceeds the equivalent stress.

Many bolting specialists would likely rebuke the stress range approach applied here. For one, it defied the time-honored Proof strength practice, which provides much higher stress allowable value (e.g., 90% of yield) for any given bolt. Another disagreement can be over an empirical Code expression which relates a bolting stress to 45000/(d) **0.5. Let's use the preceding illustration. If the bolt diameter is 0.75", the Proof load for bolting would be either 72,000 psi or 51,961 psi. These values exceed Preload of 30 ksi by a significant margin.

Rebuttal to this theory should be based upon the premise of "displacement" cycle life of the flanged joint. Should the flanged joint be designed for a single cycle life (e.g., due to the foundation settlement), bolting allowable can go up as high as three times its base stress value. This is in accordance with the provision of Pressure Vessel Codes. On the other hand, if the flanged joint is considered part of a thermally cycling piping system, Preload should comply with the displacement stress criteria.

Additionally the Preload derivations face another crucial test: Will the Preload overstress the companion flanges? This concern is particularly relevant if the flanged joint uses high strength bolts such as 193 B16. To safeguard the weaker flanges, computations should be made for the mating flange using the Preload as the seating bolt load. Any flange overstresses due to the Preload can be highlighted in the familiar terms of "Sh", "Sr", "St", and "Savg". This exercise should also help to strengthen the Preload case in that Proof Load would usually overstress the flanges at high temperatures.

It goes without saying that for Preload evaluations, the flange stresses are compared with the stress range allowable of the flange material (e.g., excluding corrosion allowance).

After determining the Preload stress of bolting, a recommended "bolt torque" value for the flanged joints could be provided. The equation below expresses the Bolt Torque derivations:

T = K * d * Fpt / 12, where

T is the bolt torque in ft. lbs
K is the Nut factor.
d is the nominal diameter of bolt, in.
Fpt is the calculated Preload stress in psi.

Exhibit 4 gives some "K" factors, copied from the Standard Handbook of Machine Design (exhibit 4).

Paragraph NC-3658.1 (a) states:

The design pressure shall be replaced by $P_{FD} = P + P_{eq}$, where P = the design or service pressure, and P_{eq} is the equivalent pressure accounting for moments applied to the flange.

Paragraph NC-3658.1 (d) states:

S_H shall not exceed 1.5S

S_R shall not exceed 1.5S

S_T shall not exceed 1.5S

Exhibit 1

Paragraph 4.2.3 states that one method of including the effects of flange moments is to convert the moments into an equivalent pressure that is then added to the internal pressure. The resulting combined pressure P_{FD} is accepbtable if either of the following criteria are met:

1) P_{FD} does not exceed the ANSI B16.5 flange rating.
2) S_H, S_R, and S_T should be less than the yield stress at the design temperature. S_H, S_R, and S_T are as defined in Section VIII, Division 1 except that; P_{FD} should be used in the Section VIII, Division 1 equations and S_H should include the longitudinal pressure stress at the flange hub.

Exhibit 2

Paragraph 4-141 states that the maximum of such service stress, averaged across the bolt cross section and neglecting stress concentrations, shall not exceed two times the stress values of Table ABM-2.

Exhibit 3