Pipe Whip Restraint Gap Effect Analysis

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An analysis is made of the effects originated by the pipe whip restraint gap on the restraint reaction, piping stresses and accelerations on the various locations of the broken piping portion protected by the restraint.

A typical and simple 30 in. diameter pipe geometry is considered, with one motorized valve and one circumferential break is assumed.

For different values of the existing gap between the pipe and the restraint, non-linear elastic plastic dynamic analyses have been carried out.

The results of the dynamic studies are analysed under the following points:

1. Restraint reaction force variation with time for different gap values. Maximum reaction force when the gap changes from 0 to 2 in., and when the restraint stiffness characteristic change.
2. Piping stresses level on the area protected by the restraint.
3. The valve acceleration increases with gap.

1. INTRODUCTION

Current standards require high energy piping failure to be analysed, and its effects studied in detail, in order to guarantee the degree of safety in the installation [1].

The effects caused by a pipe rupture can be felt as much upon the elements located in the proximity of the point where the rupture takes place, as upon the fittings located on the piping itself, which are necessary for the safety of the plant.

In certain circumstances it is necessary to design structures called pipe whip restraints, which are capable of absorbing the pipe whip effects produced by a break in the pipe.

Pipe whip restraints should generally allow free thermal displacement of the piping in normal operation, and for this purpose a gap must be left between the piping and the restraints.

This article examines the effects caused in reactions upon restraints, stresses in the piping, and accelerations in different parts of the piping for diverse gap values. For this purpose, several dynamic analyses of typical piping models have been conducted, with the gap being varied from 0 to 2 in.

2. MODELS USED

The geometrical form shown in Figure 1 was taken as representative of a typical case of high energy piping.

The piping characteristics are: Diameter 30 in., wall thickness 1.125 in., material SA-106 Grade B, saturated steam, and a operation temperature of 530°F.

The pipe material elastic plastic properties as obtained from references [2] and [3] in accordance with the method shown in reference [4] are considered in this analysis.

Points 130, 150 and 160 correspond to the modelling of a 14500 lbs. valve placed in the model for the purpose of analysing accelerations at different points on the valve.

Two variations as follows have been introduced on the model in Figure 1, which is henceforth called A:

- Model B. No restraints at point 210.
- Model C. Anchor placed at point 210, and restraint for point 50 at a distance of 9.5 ft. from point 20.

It is assumed that an instantaneous circumferential pipe break occurs at point 20. A force-time function has been chosen, where the force remains constant once it has attained its maximum value of 200000 lbs. at 0.001 seconds [5] after break occurs. This simplified function has been adopted in order to be able to make a fair interpretation of the results, and avoid any superposition of effects.

The distance from point 20 to the point where plastic hinge would take place, calculated in function to the ultimate moment, is greater than that which exists in the three models between the point where the break is assumed, and the restraint at point 50.

3. STUDIES PERFORMED

Non linear elastic plastic dynamic studies were performed using the PIPERUP [6] computer programme.

The restraint material elastic plastic properties were considered with variations from 500000 lbs. to 800000 lbs. being made to the yield point of the restraint at point 50.

Dampening was considered to be 3% in all studies.

The gap in the restraints at point 50 was varied from 0 to 2 in., and studies were made for 0 in., 0.05 in., 0.25 in., 0.5 in., 1 in. and 2 in., with the most suitable integration time step being used in each one.

The energy absorbed in the piping deformation upon impacting with the restraints, as well as the ovalization of the different sections thereof were not taken into consideration in a conservative manner.

4. RESULTS OBTAINED

First of all, it must be noted that the choice of the integration time step has a great deal of importance as far as the results are concerned. Values which give correct results for a certain gap may be found to give wholly illogical data if used with greater gaps, and special attention should therefore be given to the velocity possessed by the piping when it strikes the pipe whip restraint, and to deducing the time increment that should be used.

The maximum restraint reaction at point 50 ranges from 406139 lbs. in the case of gap 0, up to 511086 lbs. when the gap is 2 in. and the yield load of the restraint is 500000 lbs., but it can go up to 804894 lbs. if the yield load is 800000 lbs. and the same gap.

The results obtained with model B on varying the force in function to the time on the restraint at point 50 with 0 in. and 2 in. gaps are shown in Figures 2 and 3 respectively.

The displacement in the Y direction at point 50 become stabilized quickly as is shown in Figures 4 and 5. Figures 6 and 7 show the X displacement at point 50 in model A and C respectively when the gap is 2 in.

When there is a gap in model C the vertical displacement at point 20 has important magnitude, thus for example with 2 in. gap the displacement is 19.97 in.

The stresses in the piping obviously increase with the gap. If there is none, point 30 is the only one that reached the strain hardening in any one of the three models. When the gap is greater than 1 in. in models A and B, the points at the ends pf the valve also reached the strain hardening, and with model C all the points located between the break and the valve reached the strain hardening.

The study of accelerations has been concentrated upon points 130, 140 and 150, which have been used to modelize the valve. The results show the significant increase in the accelerations at the above mentioned points when the gap is widened. Thus for example, the following results are obtained at point 160:

	Gap = 0 in.		Gap = 2 in.		
Model	A	B	A	B	C
Acceleration (+Y)	+6.36 g.	+7.86 g.	+22.88 g.	+11.68 g.	+15.66 g.
Acceleration (-Y)	-7.86 g.	-3.17 g.	-30.53 g.	-20.08 g.	-12.95 g.
Acceleration (+X)	+4.70 g.	+4.20 g.	+17.43 g.	+18.87 g.	+16.46 g.
Acceleration (-X)	-5.72 g.	-6.27 g.	-20.07 g.	-17.17 g.	-17.69 g.

Both the restraint at point 210 on model A, as well as the anchor on model C reduce the vertical accelerations in the valve and do not generate sensible changes in the horizontal accelerations.

Fig. 8 and 9 illustrate the variations in the vertical acceleration at point 160 in the valve motor with gaps of 0 in. and 2 in. respectively.

5. CONCLUSIONS

In designing pipe whip restraints, special attention must be given to the gap left between the pipe and the restraint, which in any event should preferably be as narrow as possible.

When static analyses are performed, the dynamic load factor to be used must be carefully chosen. It will to a large extent be found to depend upon the width of the gap, and the load-deflection properties of the pipe whip restraint.

Whenever results from similar dynamic designs are not available however, it is advisable for a nonlinear dynamic analysis of the assembly to be performed, and obviously with these analyses, the inclusion of the gap in the model has a considerable effect upon the results.

Piping stresses increase with the gap, and can reach higher values than desired in the piping section protected by the pipe whip restraint.

Accelerations are also liable to increase as the gap does, and in valves for instance, they can reach such values that their operability cannot be guaranteed when there is a break in the pipe.

Finally, it must be remembered that it is a requirement for the gaps in the pipe whip restraints in the installation to be the same under operating conditions as those used in the design.

6. REFERENCES

[1] Title 10, CODE OF FEDERAL REGULATIONS, Part 50, "General Design Criteria for Nuclear Power Plants".

[2] ASME BOILER AND PRESSURE VESSEL CODE, Section III, Nuclear Power Plants Components, Table I-2.1.

[3] AMERICAN SOCIETY FOR METALS. Metals Handbook, 8th edition, 1961, pag. 491.

[4] GORDON C.K. YEH. An Application Guide for the Time-History Plastic Analysis of Piping Systems. IMechE 1980 C91/80.

[5] ANSI/ANS-58.2-1980. American National Standard Design Basis for Protection of Light Water Nuclear Power Plant Against Effects of Postulated Pipe Rupture, Appendix B.

[6] QUADREX CORPORATION, PIPERUP. Pipe Force and Whip Analysis. Revision G.

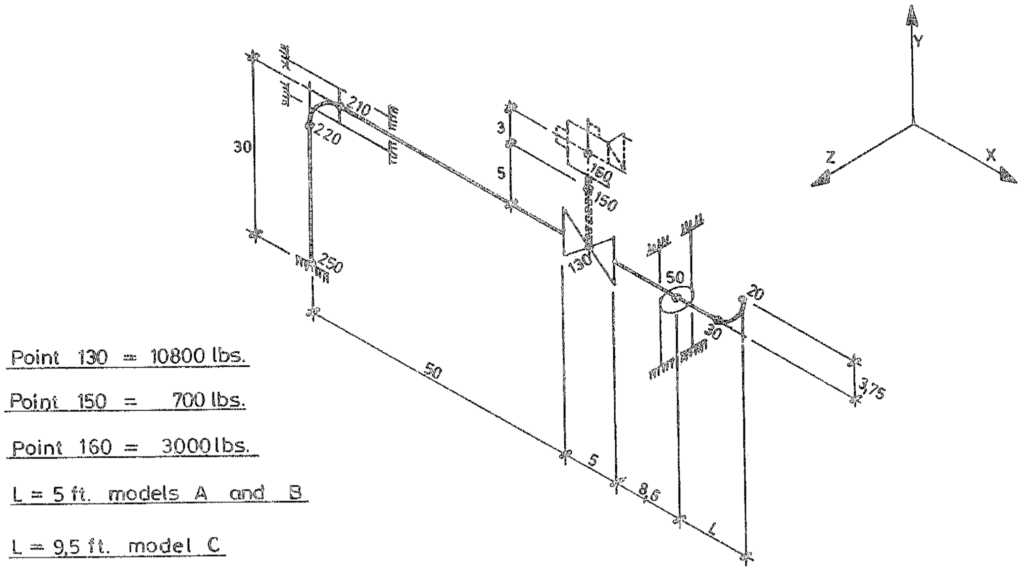

Figure 1. Model used

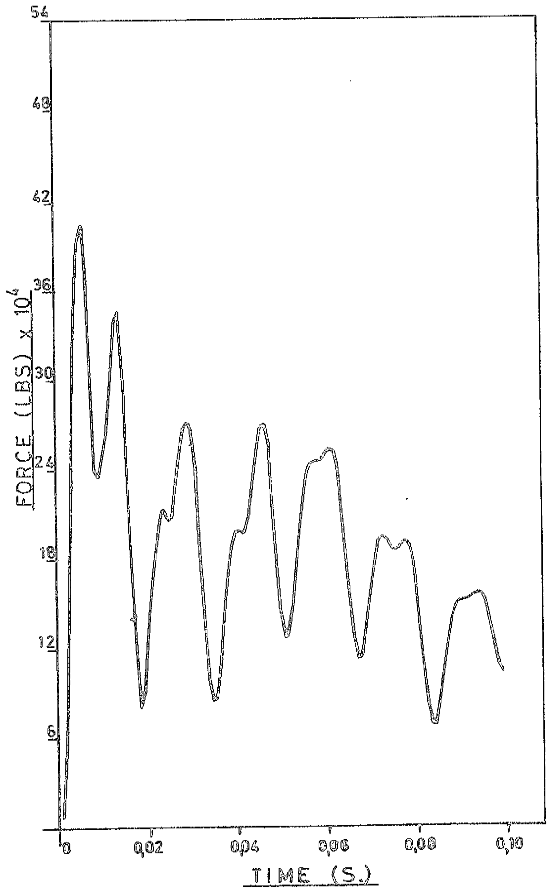

Figure 2. Restraint reaction at point 50 with 0 in. gap.

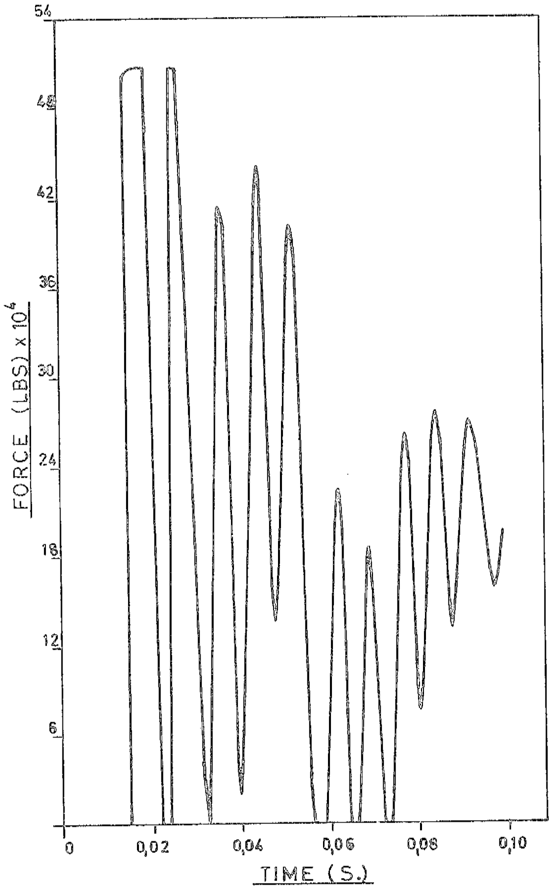

Figure 3. Restraint reaction at point 50 with 2 in. gap.

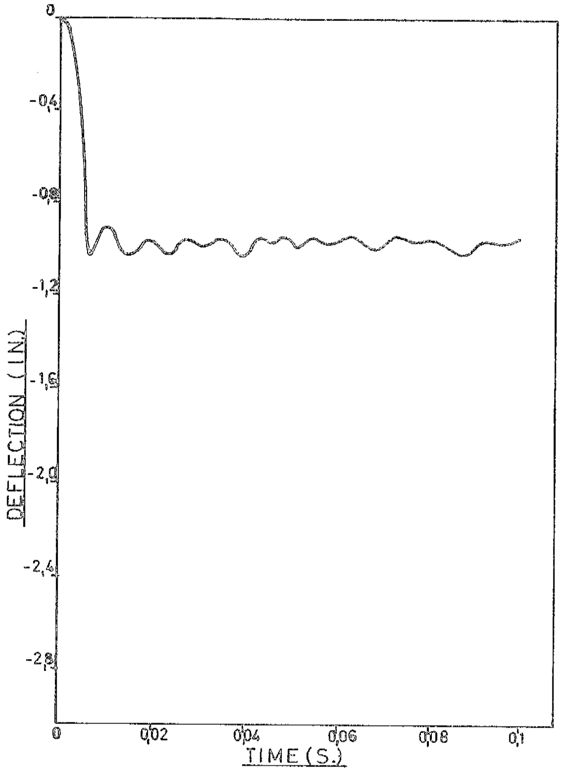

Figure 4. Vertical displacement at point 50 with 0.5 in. gap.

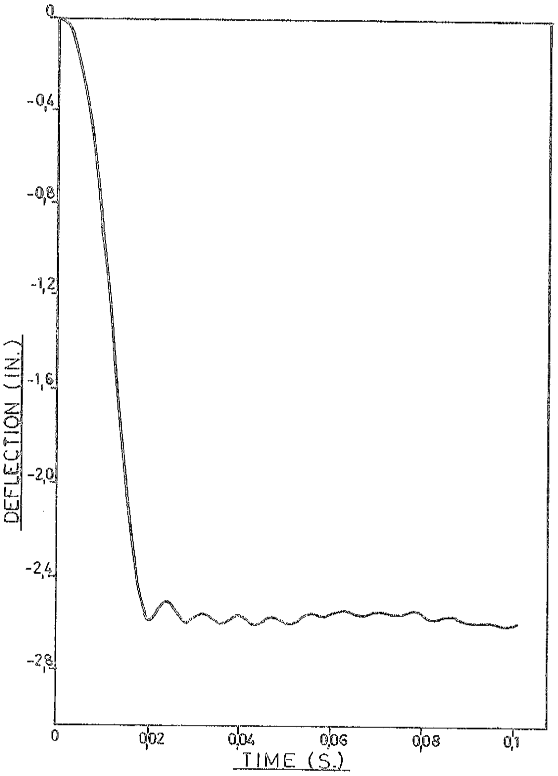

Figure 5. Vertical displacement at point 50 with 2 in. gap.

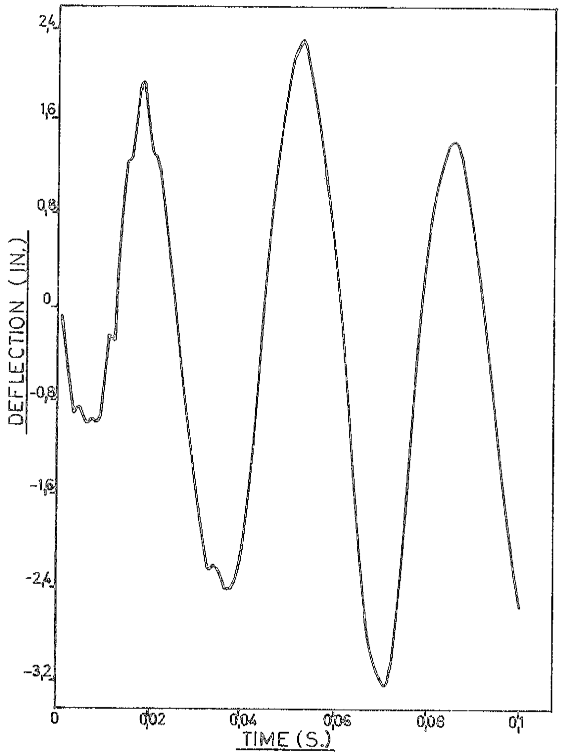

Figure 6. Horizontal displacement at point 50 model B.

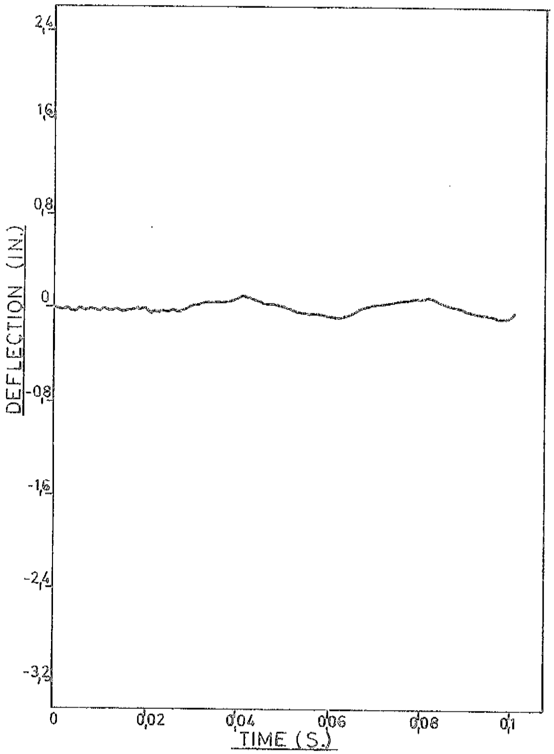

Figure 7. Horizontal displacement at point 50 model A and C.

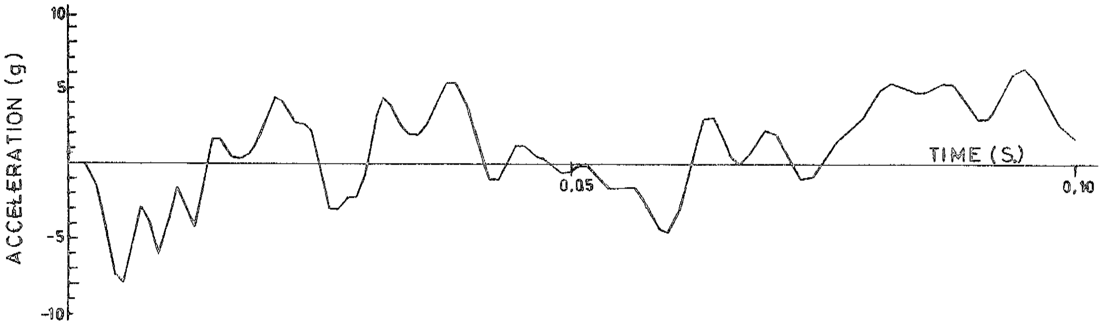

Figure 8. Vertical acceleration at point 160 with 0 in. gap.

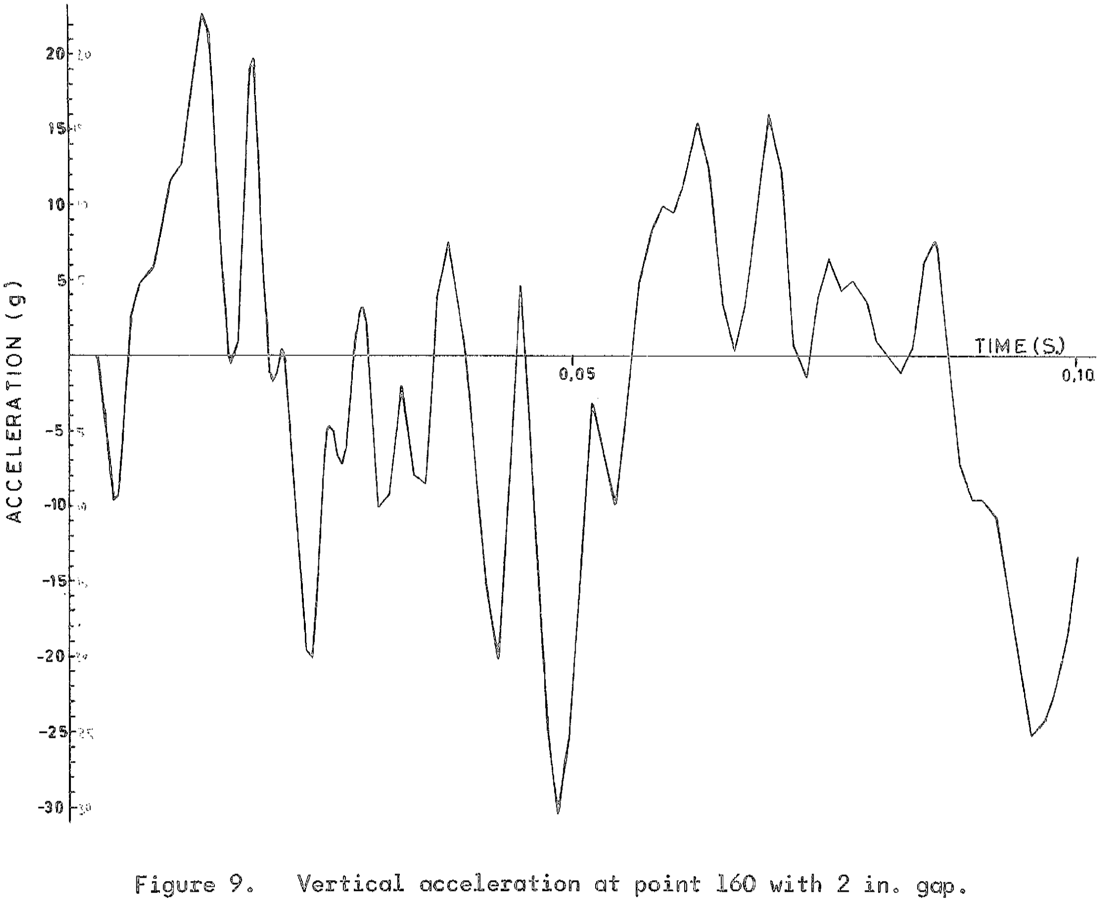

Figure 9. Vertical acceleration at point 160 with 2 in. gap.