Development of Permanent Mechanical Repair Sleeve for Plastic Pipe

Final Report

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Prepared by
Hitesh Patadia
Project Manager

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Gas Technology Institute
1700 So. Mount Prospect Rd.
Des Plaines, IL 60018

For

Daniel Driscoll
Project Manager
U.S. Department of Energy
National Energy Technology Laboratory
3610 Collins Ferry Rd.
P.O. Box 880
Morgantown, W.V. 27507-0880

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ABSTRACT

A comprehensive program was undertaken to design and develop a mechanical repair fitting that can be installed under live blowing gas conditions to serve as a permanent repair option. Through an iterative design and development approach, GTI and R. W. Lyall have developed a viable product design concept.

From the onset, the project identified and took into account three key technical challenges throughout the entire design and development process. First, the fitting must be capable of being installed under live blowing gas conditions thereby eliminating the need for additional excavations and parts leading to reduced cost of repair. Second, once installed, the fitting must amply mitigate the continued growth of the damage through the Slow Crack Growth (SCG) failure mechanism. Finally, the fitting must serve as a permanent repair by providing a leak tight seal at typical operating pressures over its intended design life.

Following a series of design iterations, GTI and its manufacturing partner R.W. Lyall have developed a final prototype version of the mechanical repair fitting. The prototype(s) were subjected to comprehensive testing and evaluation to validate their performance. The results of the testing demonstrated that additional work is necessary. The results of the testing demonstrate that the fitting can be installed under blowing gas conditions at 60 psig; however, the fitting is not capable of withstanding the temperature fluctuations that may arise under actual field conditions.

Cumulatively, the overall results of the program have established the fundamental groundwork related to the design and development of a permanent plastic mechanical repair fitting for use on damaged PE gas mains. However, additional work is necessary to make small scale design changes such that the proposed fitting design satisfies all of the necessary qualification testing.

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EXECUTIVE SUMMARY

When a steel pipe is gouged or is ruptured, it is not uncommon for gas utility companies to repair the damaged pipe section by welding a steel repair sleeve or full-encirclement fitting over the compromised area. However, for plastic piping systems, there are no fittings that are currently available which mirror this process on polyethylene pipe (PE) and can serve as a permanent repair.

The central objective of this program was to develop a plastic pipe mechanical repair fitting that can be installed on damaged 4" polyethylene (PE) pipe under system operating pressure, and validate its performance for use in gas distribution applications. From the onset of the program, with input and guidance from the project team, a product definition was established which included:

- The overall design and development efforts should focus on 4-inch pipe size with a pressure rating of 60 psig.
- Fitting should be able to be installed under blowing gas conditions at typical line pressures (60 psig)
- Once installed, the fitting design should effectively mitigate the continued propagation of the damage via the slow crack growth (SCG) failure mechanism
- The fitting should conform to existing ASTM standards and specifications (ASTM D2513 and F1924 requirements, as applicable)

Based on the aforementioned product definition, R.W. Lyall – the manufacturing partner – under the direction of Gas Technology Institute (GTI) lead collaborative group, undertook an iterative design process to develop a permanent mechanical repair fitting. A finite element model was created to ascertain the overall system stress between the fitting halves and its clamping structure under anticipated operating conditions. The model was created in SolidWorks and the static analysis was preformed with CosmosWorks utilizing a solid mesh and the FEEPLUS solver. A best-case design concept for the mechanical repair fitting was finalized. Several concepts were modeled using rapid prototyping technologies, and functional prototypes were built and tested.

In essence, the mechanical repair fitting design consists of two half circular cylindrical parts that are hinged together. After they have encircled a pipe segment that has been damaged, these two parts can be mechanically fastened to each other to contain the damage section. The fitting is maintained in position by compressive forces between the elastomeric seal and the damaged pipe, which is induced through tightening the bolts sufficiently such that pressure is transmitted through compressed elastomeric rings at the ends of the sleeve. As these rings are the only portion of the sleeve that actually contact the pipe, there is an annular cavity between the inner wall of the fitting sleeve and the outer wall of the damaged portion of the pipe that is contained within the fitting body. Figure 1 below presents the outline of the final design of the Mechanical Repair Fitting.

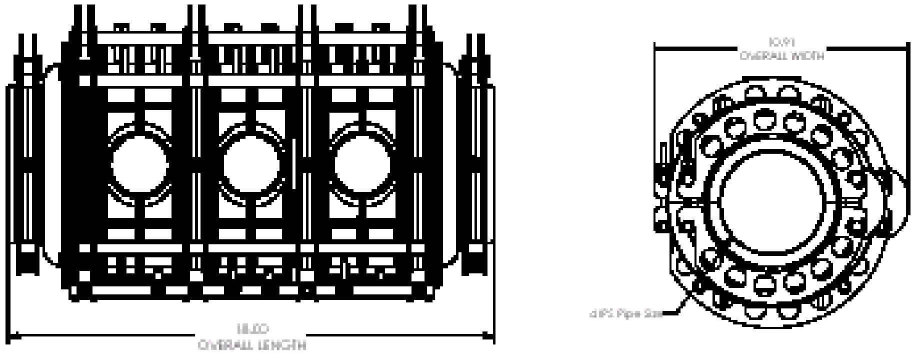

Figure 1: Illustration of the proposed design for Mechanical Repair Fitting

Because the mechanical repair fitting is intended to cope with the wide range of damage that can occur in service, the principal design challenge was the repair under "blowing" gas conditions, i.e., a through wall failure in the pipe wall through which pressurized gas escapes. In order to facilitate this type of a repair, the mechanical repair fitting design incorporated three axially aligned holes, which allow the gas to escape during the repair procedure.

In addition to ensuring that the mechanical repair fitting can withstand internal pressures of up to 100 psig and provide a leak tight seal over its intended service life, another important consideration was its ability to effectively mitigate the propensity for continued Slow Crack Growth (SCG). To that end, a hybrid approach (analytical and experimental) was undertaken at GTI to ensure that the proposed fitting design effectively mitigates the SCG failure mechanism thus allowing the fitting to serve as a permanent repair option. Results of the analytical model demonstrated that the annular cavity between the inner wall of the fitting sleeve and the outer wall of the damaged portion of the pipe that is contained within the fitting body would permit the equalization of pressures and consequently eliminates the necessary driving force and mitigates any further continued SCG propagation. The results of the analytical model were confirmed through experimentation.

Having resolved the key technical challenges, a "best case" design was selected. Prototype assemblies were constructed and evaluated under controlled laboratory conditions to determine the specific performance characteristics of the mechanical repair fitting.

Cumulatively, the overall results of the program have established the fundamental groundwork for the development of a permanent plastic mechanical repair fitting – although additional work is necessary to complete the intended program objectives. This report presents a detailed summary of the entire iterative design process: conceptual design to ensure the fitting can satisfy the product definition; prototype development; and the results of the testing and evaluation to ensure the fitting does not adversely affect the overall pipe system integrity and can effectively mitigate the propensity for Slow Crack Growth (SCG).

EXPERIMENTAL

When a plastic pipe is damaged, the traditional approach to repair the damaged section involves stopping the flow of gas, running an external bypass, and splicing in a new plastic pipe segment as shown in Figure 2 below.

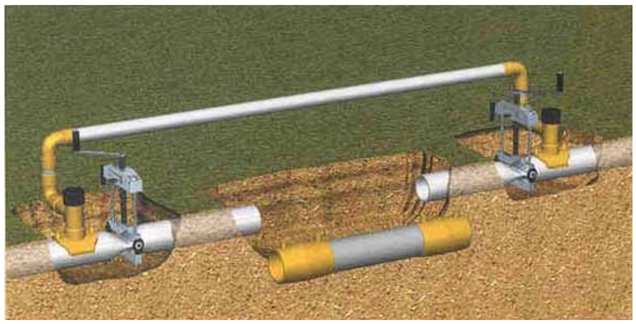

Figure 2: Conventional repair of damaged PE mains (Courtesy: Mueller Company)

This approach is both time consuming and labor intensive leading to an increase in operating costs. Therefore, a novel approach was required to facilitate a permanent repair under blowing conditions thereby eliminating the need for additional excavations, fittings, and reducing the flow of gas.

At the onset of the program, GTI and R.W. Lyall, with input and guidance from utility sponsors, developed a set of minimum requirements for the final mechanical repair fitting design. These requirements served as the basis for the overall design and development activities throughout the course of this program.

1) *Pipe Size:* 4-inch SDR 11 pipe
2) *Maximum Operating Pressures at 73°F:* 60 psig
3) *Pressure bearing shell material:* Thermoplastic material meeting the material requirements of ASTM D2513.
4) *Nominal fitting length:* 12-24 inches
5) *Vent capabilities:* Ability to vent natural gas out of the trench while performing the necessary repair under blowing gas conditions.
6) *Leak test provisions:* The fitting must be capable of withstanding an internal pressure up to 90 psig (1 ½ times maximum operating pressure)
7) *Maximum defect size (for 4-inch fitting) non Blowing Gas condition:*
   a) Damage not to exceed 1/3 of circumference.
   b) Gouges greater than 10% of wall thickness by 12 inches in length.
   c) ¾ inch protrusion from outside pipe surface.
8) *Repair Installation Parameter Goals under blowing gas condition:*
   a) Able to install on pipe with blowing gas at 60 psig with a hole less than .78 sq inches.
   b) Able to install on pipe with blowing gas at 30 psig with a hole less than 1.76 sq inches.

Based on the aforementioned design criteria, it was readily apparent that the primary design challenge involved balancing the ability to install under blowing gas conditions while ensuring that the fitting can be adequately designed to provide a permanent leak tight seal at the intended design pressures.

With this in mind, GTI and R.W.Lyall reviewed several possible design alternatives. In order to meet all of the aforementioned requirements, several observations were made:

1. Based on a review of the repair process for steel piping systems, it was noted that the steel band clamps often do not mitigate the continued growth of the damage (crack/gouge) beyond the ends of the fitting lengths. It was theorized that the design for the proposed mechanical repair fitting must incorporate an annular space. As a result, the pressure on the inside of the gas carrying pipe and fitting annular space will be equalized to prevent the continued growth of the damage via the SCG mechanism for polyethylene pipes.
2. The overall fitting geometry and the clamping system must be sufficiently sized in order to fully encapsulate the damage section and provide a leak tight seal while operating at 60 psig.
3. The fitting design must ensure adequate means to allow the “blowing” gas from the damaged pipe section to be removed from the excavation site in order to facilitate a safe repair.

GTI undertook a comprehensive effort to address each of these concerns. Detailed description for each key design consideration is addressed in the respective sections to follow.

Slow Crack Growth (SCG) Considerations

From the onset, it was recognized that there are two types of damages that can be encountered. The first case involves instances of a complete through wall breach, i.e. puncture. The second case involves instances of an appreciable wall loss, i.e. gouge. For the case involving a through wall failure, the primary design challenges centered around the ability of the fitting to be installed under blowing gas conditions in a safe and effective manner, and, once installed, amply mitigate the continued growth of the damage. For the case of an appreciable wall loss, the central design consideration involves the fittings’ ability to amply mitigate the propensity for slow crack growth in the axial direction beyond the end seals.

In both cases, the fitting’s ability to amply retard the continued growth of the damage beyond the end seals is of paramount importance. As previously noted, based on actual field experience with the use of steel band clamp fittings installed over damaged steel pipe sections, it has been shown that there is an increased risk for continued growth of the damage section beyond the end of the fitting length. Based on a review of the steel band clamp design and installation process, which are “wrapped” directly on the damaged steel pipe section, it was theorized that the proposed mechanical repair fitting must incorporate an annular space between the damaged PE pipe and the repair fitting. The annular space essentially removes the strain energy necessary to drive the crack beyond the end seals. A hybrid approach was undertaken - analytical modeling and experimental validation - n order to test the efficacy of this argument.

Analytical Modeling – SCG Consideration

It is well understood that thermoplastic gas pipe materials behave in a nonlinear elastic manner which makes the conventional approach of evaluating the slow crack growth (SCG) considerations through the use of linear elastic fracture mechanics (LEFM) valid only for "older" generation of PE gas pipe materials – those extruded prior to the mid 1980's. However, based on significant amount of previous research sponsored by the Gas Research Institute, the older generation PE materials still use in gas distribution systems represents the worst case condition. It is generally understood that modern PE materials have less susceptibility to failures resulting from SCG as compared to those produced prior to the mid-1980. As a result, an LEFM based approach was utilized to serve as a "worst case" bounding approach (e.g., all damages for all grades of PE materials will be considered to be brittle-like failures).

The primary parameter in an LEFM based analysis is the stress intensity factor, commonly denoted by K, which characterizes the stress and deformation states at the tip of a sharp crack. The parameter K is a function of the applied stress, the crack size and shape, and the dimensions of the component in which the crack is embedded. It is important to recognize that K is not dependent upon the material, so long as that material behaves predominantly in an elastic manner. Accordingly, there is a multiplicity of sources for K that provide a good approximation to the case of primary interest in this program: a finite width and depth (e.g., semi-elliptical), partial through wall crack that is subjected to a biaxial stress field in a thick walled tube.

Once the most appropriate relationship for the K factor is ascertained, a determination can be made of the combination of parameters (including initial crack depth, aspect ratio, and the angle of the crack surface from the axial direction) that has the most potential for SCG propagation to take place in the axial direction. Because of the number and ranges of these variables, an analytical model was developed to draw preliminary conclusions, and provide guidance for the subsequent experimental effort.

To adequately simplify the analysis effort, a single representative PE gas pipe size and two representative older generation PE materials were analyzed. The pipe size that was considered for the bulk of the calculations was a 2-inch diameter, SDR11, PE2306I-A-482 gas pipe operating at 100 psi. For one of the two materials, extensive SCG data was previously developed by Battelle[1]. The other was an earlier vintage PE2306 material, which has been studied in work by the Southwest Research Institute (SwRI). In this preliminary stage, no consideration was given to seasonal temperature and pressure changes, or to residual stresses induced by the extrusion process. A "bounding" approach was taken into account with conservative assumptions and simplifications that provide a lower (pessimistic) bound for the potential of crack growth in the axial direction.

For starters, a brief literature search was performed to ascertain suitable stress intensity factor relations. Based on the specific considerations for the purposes of this program, the modified versions of the finite element solutions generated by I. S. Raju and J. C. Newman for surface cracks in plates under tension were utilized [2].

The modified Raju-Newman solutions gives the Mode I stress intensity factor, K, at all points along the periphery of a semi-elliptical crack as a function of the crack depth, a; one-half of the surface length of the crack, c; the wall thickness, h; and the remote tensile stress, σ. For the purposes of this program, a simplified form of these equations was used assuming that the surface crack length is significantly greater than the crack depth (c » a). This simplification was appropriate both because the focus of this work was on long axial cracks (i.e., large values of c), and the SCG failure surfaces that have been examined generally show that this was a typical initial condition. The modified version of the Raju-Newman equations then simplifies to the following:

$$K = K_D \{1 + [0.1 + 0.35 \ (a/h)^2](1- \sin^2\varphi) \ \sin \varphi\} \ldots \quad \text{Eqn. (1)}$$

and

$$K_D = \sigma(\pi a)^{1/2} \sec^{-1/2}(\pi a/2h) \ldots\ldots \quad \text{Eqn. (2)}$$

It can be seen that the K values at the point of deepest penetration (i.e., at $\varphi = 90^o$) is the highest value, and that the K value on the surface (i.e., at $\varphi = 0^o$) is zero. However, this does not imply that axial direction crack growth is not possible. Whatever the propensity for axial direction SCG, it will likely always be terminated before any significant extent of growth takes place by the pressure loss resulting from depth direction (through wall) SCG that breaches the wall of the pipe. To quantify this effect, an LEFM based approach, which has been appropriate for characterizing in-service SCG failures in PE pipe, was used [3]. The two equations that prescribe the SCG behavior of a given PE gas pipe material are:

$$t_i = B \times K^{-m} \ldots\ldots \quad \text{Eqn. (3)}$$

and,

$$da/dt = A \times K^m \ldots\ldots \quad \text{Eqn. (4)}$$

Where t_i is the time required for SCG to be initiated, da/dt is the time rate at which SCG occurs, A, B, and m are material (temperature-independent) constants. It is important to emphasize that these relations are valid only for “older” PE gas pipe materials; i.e., materials in pipe extruded prior to the mid 1980’s. However, since the “older” generation materials are more susceptible to brittle-like failures, they represent the worst-case conditions.

In general, the use of Eqns (3) and (4) is often constrained by insufficient SCG data by which the material constants A, B, and m can be evaluated. There is a similar general lack of specific information on the extrusion related residual stresses that exist for various PE gas pipe materials. However, for the purposes of this study, data from previous published studies was utilized. One set of data exists for a 2 inch diameter, SD11, PE2306I-A-482 gas pipe extruded in the 1980”s (exact date unknown). A second set of data exists for a similar pipe, but one that was extruded in the early 1970’s. From these data, the values shown in Table 1 have been derived:

	1970 Vintage PE Material	1980 Vintage PE Material
A	2.5877×10^{-6}	1.0306×10^{-9}
B	4.1195×10^{4}	1.6950×10^{7}
m	3.0	2.1

Table 1: SCG Material Constants for Two PE 2306 Materials

In Table 1 the constants A and B have dimensions involving stress in psi, length in inch, and time in years, while m is dimensionless[1]. Note that all of the above values were derived directly from the experimental data, except for constant B for the 1980 vintage material for which SCG data were not available.

Using Equations (1) – (4) and the material constants presented in Table 1, preliminary calculations were performed in an attempt to predict the failure times and other aspects of field failures resulting from the slow crack growth mechanism. These results of the calculations correlated well with actual field experience as outlined in previous studies. However, the results also suggested that the K value for axial growth based on Eqn (1) was potentially too low. In order to better approximate both the failure times and the length of the axial SCG growth, two expedients were introduced to force the model.

First, the axial stress was arbitrarily forced to be a multiple of the depth direction stress by the constant 1.132. Second, the value of the material constant B for the 1980's vintage material, for which data were not available to determine in the same manner as the other constants in Table 1, was calculated based on service performance data presented in the Battelle study [1]. The results of the analysis were that the calculated time with the model for a break through the wall was 15.1 years, with the maximum extent of axial crack growth calculated as 0.035 inch.

Tables 2 and 3, respectively, present the calculated predictions for the times to failure from initial crack-like damage on 2-inch diameter, DR11, PE2306 gas pipes extruded in the 1980's and 1970's, respectively. It can be seen that the predicted failure times are entirely consistent with the initial damage depths that have been used as inputs to the model. In particular, for the same damage depth, the time at which the 1980 material was predicted to fail is an order of magnitude greater than for the 1970 material (see eighth column). Taking into consideration that many failures have been found in 1970's material within a few years after installation, while relatively few have ever been found in 1980's material, these predictions are well in accordance with actual field experience.

The failure times notwithstanding, the predictions of continued growth of the damage in the axial direction (see tenth column) are found to be benign for all initial damage depths. For example, for the 1970 material, the maximum axial growth is 0.350 inch, while it is only 0.060 inch for the 1980 material. The extent of the maximum axial growth is well within the design limits for the fitting ends, which require the damage section to be at a minimum of 1" on each side of the fitting end.

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[1] In the calculations that are presented in this report, unless otherwise indicated, a remote tensile stress of 500 psi has been used. This stress corresponds to the hoop stress from a pressure of 100 psi in an SDR11 PE gas pipe.

Initial Crack Depth (m)	A/h	Hoop Stress (MPa)	Initial K Values Radial	Axial	Int (yrs)	Depth Direction SCG Prop (yrs)	Fail (yrs)	Axial Direction Int (yrs)	Length (m)
1.27*10^-4	0.025	2.76	60.16	68.1	77.85	35.31	113.16	53.67	4.83*10^-4
2.54*10^-4	0.05	2.76	85.08	96.31	27.53	22.27	49.79	18.98	7.11*10^-4
5.08*10^-4	0.1	2.76	120.32	136.2	9.73	13.08	22.82	6.71	1.07*10^-3
1.27*10^-3	0.25	2.76	190.24	215.35	2.46	5.1	7.56	1.7	1.52*10^-3
2.54*10^-3	0.5	2.76	269.03	304.55	0.87	1.46	2.33	0.6	1.27*10^-3
3.81*10^-3	0.75	2.76	329.49	372.98	0.47	0.27	0.74	0.33	5.59*10^-4

Table 2: Calculated Results for 1980 Vintage PE2306 Material

Initial Crack Depth (m)	D/h	Hoop Stress (MPa)	Initial K Values Radial	Axial	Int (yrs)	Depth Direction SCG Prop (yrs)	Fail (yrs)	Axial Direction Int (yrs)	Length (m)
1.27*10^-4	0.025	2.76	60.16	68.1	7.56	1.22	8.78	5.82	7.6*10^-3
2.54*10^-4	0.05	2.76	85.08	96.31	3.65	0.95	4.6	2.81	7.95*10^-3
5.08*10^-4	0.1	2.76	120.32	136.2	1.76	0.69	2.45	1.36	8.3*10^-3
1.27*10^-3	0.25	2.76	190.24	215.35	0.67	0.36	1.03	0.52	8.9*10^-3
2.54*10^-3	0.5	2.76	269.03	304.55	0.33	0.13	0.46	0.25	8.46*10^-3
3.81*10^-3	0.75	2.76	329.49	372.98	0.21	0.03	0.24	0.16	7.5*10^-3

Table 3: Calculated Results for 1970 Vintage PE2306 Material

Experimental Validation – SCG Considerations

Based on the preceding discussion, the results of the analytical modeling indicate that there is a relatively greater probability of a through wall failure as compared to continued crack growth in the axial direction beyond the end seals of the proposed mechanical repair fitting design. To test the efficacy of the analytic model, experimental testing was performed to validate the results of the analytical model.

It is important to emphasize the results of the analytical model were conservative in nature and intended to characterize the “worst-case” conditions. Several simplifying assumptions were incorporated into the model. For example, it was assumed that the main factor driving the SCG process is the hoop stress resulting from the internal pressure. However, under actual service conditions, the pipe and other appurtenances are subject to other stresses including extreme bending due to construction that could propagate a crack out of the proposed repair fitting in the axial direction. In addition, the pipe/fittings in-service can also be susceptible to fatigue loading due to freeze/thaw cycles, static and dynamic external loads, seismic events, etc.

Given the inherent difficulty in simulating actual in-service conditions in a single experiment, a simplified test protocol was required to determine whether or not the equalization of the pressure inside the annular space of the proposed mechanical repair fitting sufficiently eliminates the strain energy necessary to propagate the crack in the axial direction beyond the end seals. An added level of complexity was that there were no prototype fittings that actually existed. Prior to developing the necessary prototype parts and tooling, it was imperative to resolve the SCG considerations.

To that end, GTI and RW Lyall developed a modified SCG experimental test set-up to test the assumption that the equalization of strain energy inside the fitting annular space effectively retards the continued propagation of the crack in the axial direction. The central idea was to place one piece of 2-inch PE pipe with a controlled defect inside a 4-inch piece of PE pipe under pressure and monitor any appreciable change in the defect length over a period of time, as shown in Figure 3. There is a significant degree of conservatism in this approach in that the compressive forces provided by the fitting end seals are not taken into account. That is, in the actual case where the mechanical repair fitting is installed over a compromised damaged section, the fitting’s end seals will provide additional compressive forces on the damaged pipe, which help to further restrict the continued crack growth in the axial direction.

Initially, sharp controlled notches (>95% of the pipe wall thickness) were placed on 2-inch pipe specimens (medium density and high density polyethylene pipes). After notching, 2-inch SDR 11 end caps were butt fused on each end of the pipe. At one of the ends, small diameter tubing was socket fused to the end cap to facilitate pressurization. The 2-inch pipe specimens were then individually placed inside 4-inch IPS PE pipe specimens.

A Lyall 4 x ½ inch tapping tee was installed on the 4-inch pipe, which essentially acts as a pressure vessel similar to the mechanical repair fitting. The 2-inch pipe specimens were then placed inside the 4-inch pipe assemblies. Plastic tubing (¼ inch) was inserted through each of the tees and connected to the whip extensions of each of the 2-inch pipe samples with compression fittings. 4-inch SDR 11 end caps were heat fused onto the 4-inch pipe and the entire assembly was connected to individual pressure gauges and connected to a pressure manifold. The test set-up is shown in Figures 4-6.

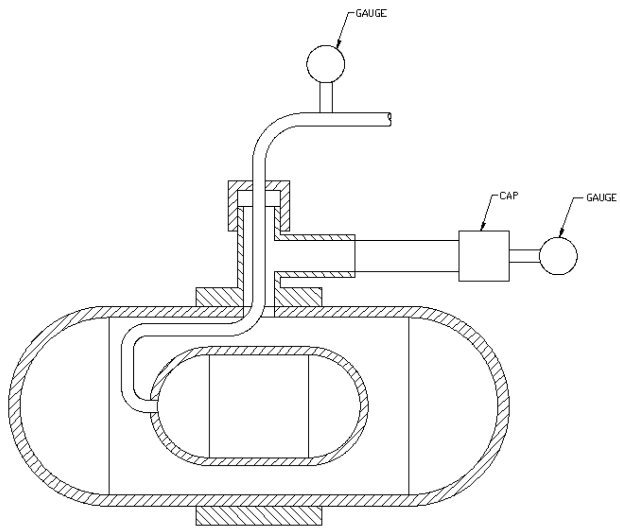

Figure 3. Schematic Illustration of the modified SCG test set-up

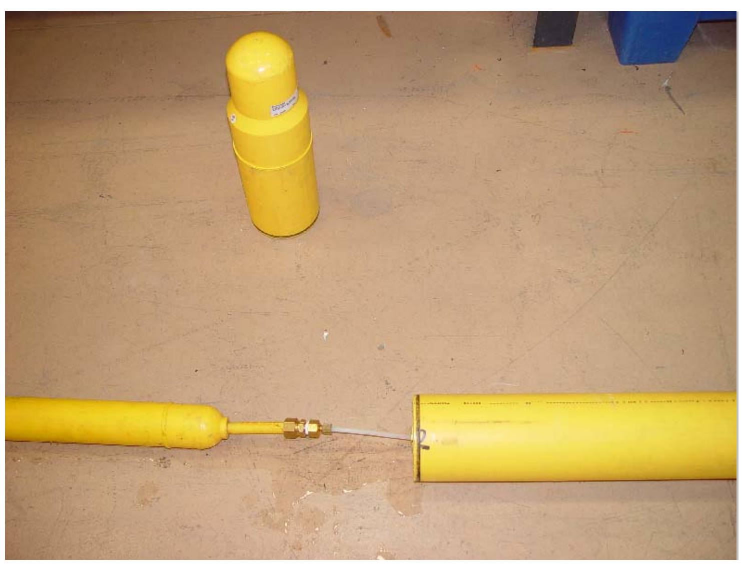

Figure 4. Pipe Connection

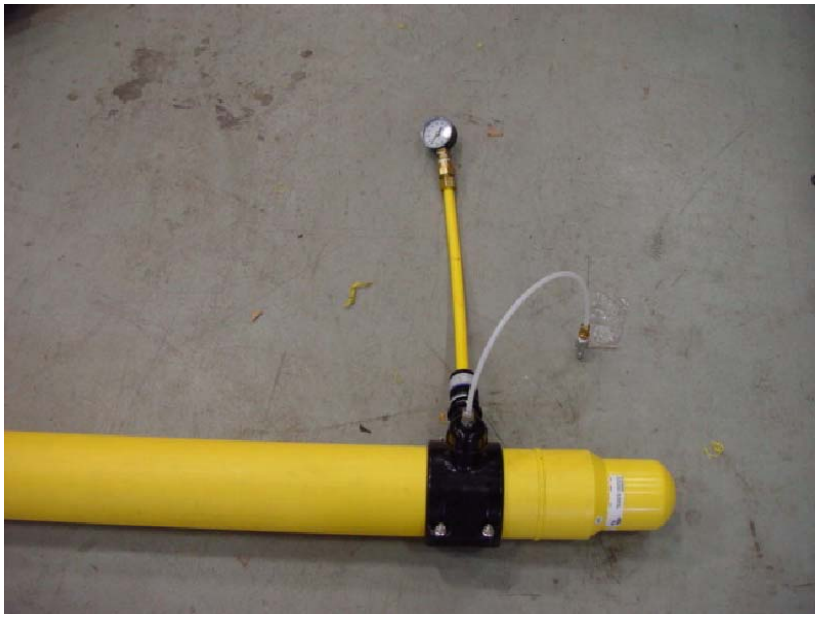

Figure 5. Pressure Gauge Assembly

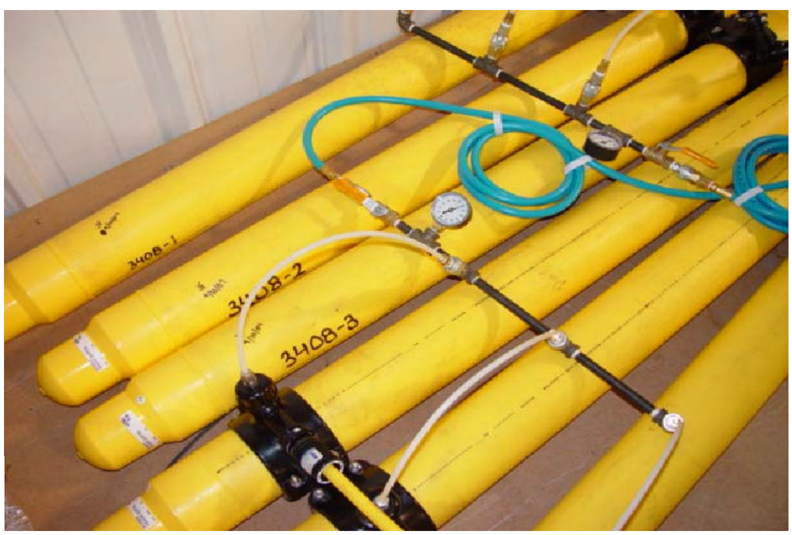

Figure 6. SCG Test Setup

Initially, the 2-inch pipe specimens with a controlled sharp notch were allowed to fail on their own accord. The idea was to simulate typical conditions whereby the mechanical fitting is installed on a damaged PE main with appreciable wall loss. The primary objective was to determine the mode of failure and whether or not there is an appreciable change in the length of the failure section in the axial direction. After several days of testing, there were no failures of any of the test specimens. The results of the testing was surprising given that the test specimens contained a sharp notch that was greater than 95% of the pipe wall thickness. The results demonstrated / confirmed the excellent performance of modern thermoplastic materials with respect to slow crack growth resistance characteristics. After 30 days, one of the six specimens failed in a brittle manner at a pressure of 87 psig. The failed specimen was removed and the crack length was recorded.

In an effort to develop additional data and to expedite the testing, a decision was made by the project team to intentionally breach all of the pipe specimens given the uncertainty associated with exactly when the specimens may fail. Therefore, all of the pipe assemblies were taken apart, and a through wall slit was placed on the 2-inch pipe. The length of the through wall slit was recorded, and the test specimens were then rebuilt and placed under pressure.

After 4 month of testing at a pressure of 90 psig (1.5 times the operating pressure), the test assemblies were taken apart and the crack length(s) were measured. Based on the results of the testing, none of the crack lengths on any of the respective test specimens experienced any appreciable change in the crack length. The results of the testing confirmed that the equalization of the pressure did effectively remove the strain energy required to propagate the crack in the axial direction.

Conceptual Design

Having favorably resolved the SCG considerations, the central design consideration for the mechanical repair fitting was contingent on its ability to withstand internal pressures up to 90 psig and provide a leak tight seal over its intended design life. In addition, the product definition also provided for the fitting to be installed under blowing gas conditions without having to reduce the line pressure through some flow control measure upstream of the damage opening.

With this in mind, GTI reviewed several possible design alternatives presented by RW Lyall. Several design features were incorporated including:

1. Vent ports were added to facilitate the ability to install the fitting under blowing gas conditions. The vent ports would ensure a means for the gas to blow away from the trench in order to ensure a safe working environment.
2. A suitable size for the annular space was developed to take into account the varying size and shape of the damages that can potentially be encountered in the field.

3. The overall fitting geometry and the clamping system were sufficiently sized in order to ensure a leak tight seal while operating at 100 psig.

Utilizing 3D computer modeling software (SolidWorks), a first generation design was developed as shown in Figures 8 and 9 below.

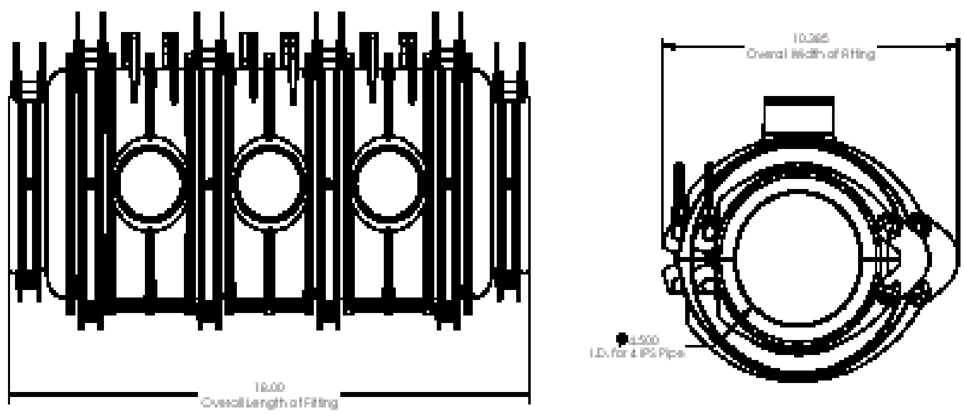

Figure 8. Schematic Illustration of the 1st Generation Mechanical Repair Fitting – Top View and Side View

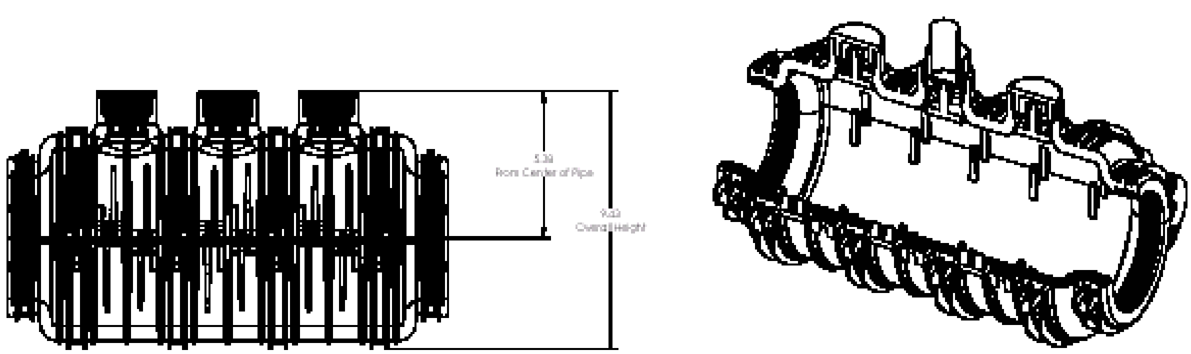

Figure 9. Schematic Illustration of the 1st Generation Mechanical Repair Fitting – Front View and Cutout.

As part of the overall design and development process, an initial prototype (Generation 1) was constructed to evaluate the form and function of the proposed design. The design of the fitting shell was modeled using Polyamide 11 (PA11). The Generation 1 prototype fitting was cast from a Urethane material from a silicone mold. The seals for the Generation 1 prototype were cast from a softer durometer urethane material. The clamp mechanism was welded together from laser cut pieces of 316 stainless steel. The Generation 1 prototype was presented to the project team, and the overall feedback was positive.

The next step was to develop a working prototype that could be subjected to more rigorous testing. Two (2) next generation (Generation 2) prototype fittings were machined from solid slabs of polypropylene material due to the fact that suitable sized solid slabs of the PA11 material were not available. The valves and caps were also machined from a polypropylene bar. The clamp collar mechanism for each prototype fitting was fabricated as a weldment from components made of 316 stainless steel sheet. The hardware for the clamps consisted of a T-bolt made of 316 SS ¼-28 threaded rod welded to a machined 316 SS cylinder. The seals were molded of nitrile rubber (50 shore A) made using production grade tooling. These second generation prototypes were used to verify the seal design development. These fittings were assembled, clamped together over a pipe section with a hole in it, and then filled with water and pressurized. Leak paths were traced through the fitting and seals using fluorescent die.

The results of the testing demonstrated that these prototype fittings could not hold water. Due to anomalies in the machined parts and the seals, there were several leak paths that were observed at pressures above 20 psig at the interface of the two fitting halves. Small scale changes were made to the seal and the prototypes were re-tested at higher pressures – 60 psig. Six Belleville washers were placed in series on the stud and compressed by a ¼-28 nut to produce a minimum pre-load on the bolt of ≈ 800 lbs. Even with the greater pre-load, both prototype fittings leaked and a slight separation was noted, see Figure 10.

Figure 10. Sustained pressure testing to verify leak tightness of proposed seal design at 60 psig internal pressure

From Figure 10, in addition to the leaks, it was observed that the fitting began to separate between the clamps with the greatest separation occurring on the outer most edge of the rib centered between the two clamps.

As a result, additional minor modifications were made to the seal design and tooling. New seals were fabricated and inserted in the prototype fitting and subjected to pressure testing at room temperature. With the latest seals from the tooling, both Generation II prototype fittings (with the bulging feature) were able to hold pressure at 60 psig over a period of over three weeks, as shown in Figure 11.

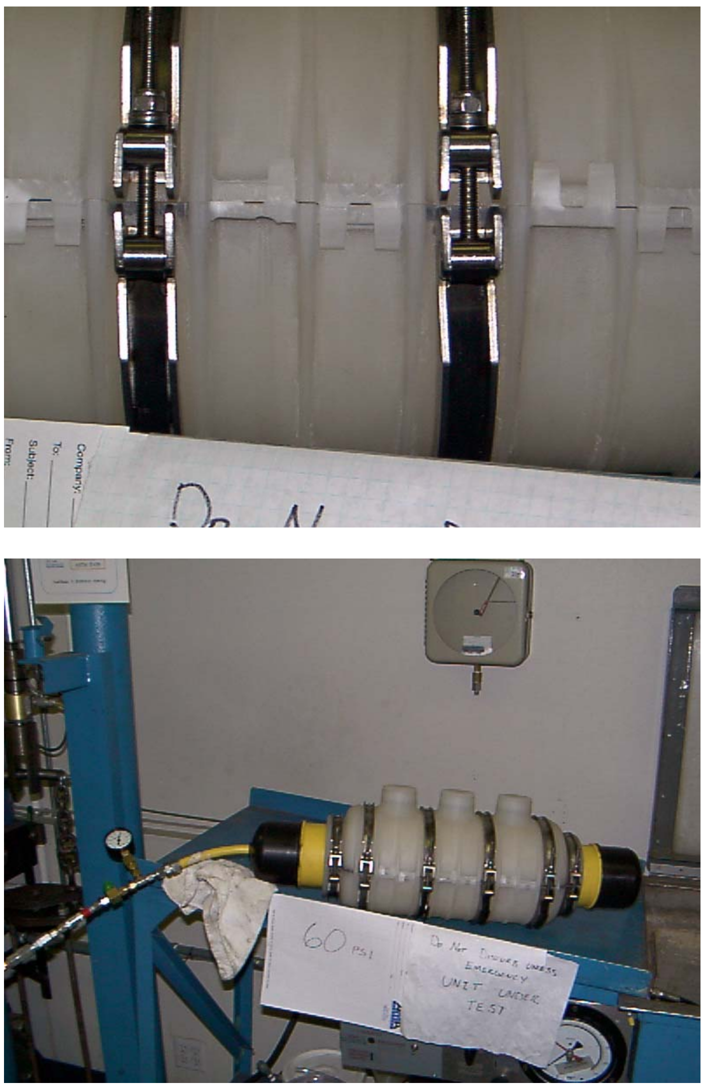

Figure 11. Sustained pressure testing to verify leak tightness of modified seal design at 60 psig internal pressure – No Leakage

Having resolved the leaks at the seal interface, the next area of concern was the “bulging” and separation of the fitting halves that was observed at increased pressures. It was theorized that the outward bulging of the fitting shell was potentially due to the internal pressure, which causes this separation. In order to test this hypothesis, the pressure was increased to 90 psig to obtain a greater understanding of this phenomenon. Over the course of several hours at 90 psig, the gap at the center separation increased significantly. Ultimately, after several hours under pressure, the seal extruded itself out of the fitting, as shown in Figure 12.

Figure 12. Sustained pressure testing to verify leak tightness of proposed seal design at 90 psig internal pressure

In order to better understand the empirical observations, R.W. Lyall constructed a finite element model to simulate the bulging at increased pressures. Several expedients were introduced into the model. As a first approximation, the clamps were omitted and the outside circumferential surface in the clamping area was restrained. Figure 13 illustrates the mode of flexing at the area between the clamps subjected to an internal pressure of 125 psig. The stress plot is shown at a greatly exaggerated deformation to clearly demonstrate the mode of flexing in the fitting. The stresses throughout the fitting body were within a 3.125 safety factor except at the point of the greatest deformation - centered between the two clamp areas at the interface of the two halves. The calculated deformation between the two halves was approximately 35 mils. While the deformation is an important consideration, it is important to note that the seals will fail prior to any large-scale deformation that may occur. As a result, it was noted that small-scale modifications were required to the clamping structure to provide additional rigidity.

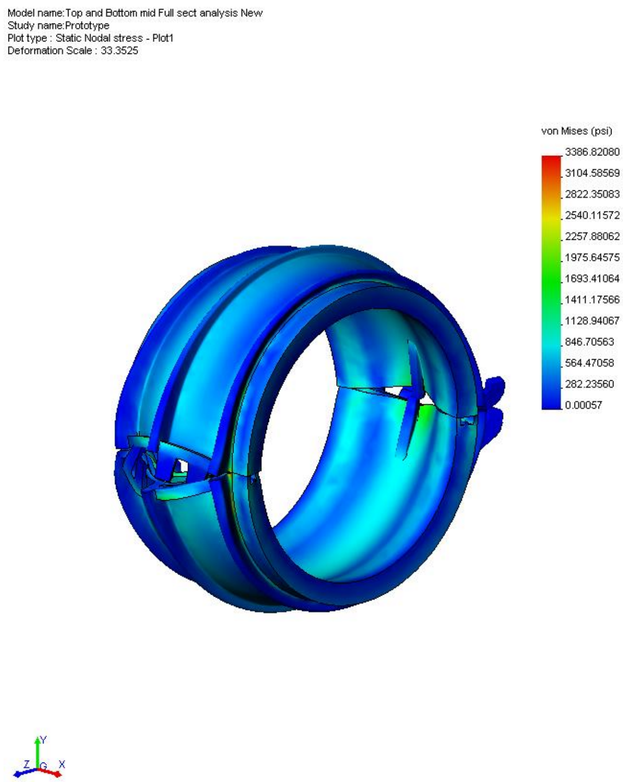

Figure 13. FEA Model to characterize deformation at clamp interface

Prior to making any significant changes to the clamping structure, it was apparent that changes were necessary to the fitting body design itself. As a result, additional FEA analysis was performed taking into account various potential changes to the fitting halves in order to find the best case scenario. As with the previous modeling, the fitting body was modeled using the properties of PA11. Based on the results of the FEA models, the fitting halves were modified as follows:

- The rib depth at the horizontal plane was slightly increased to be nearly constant all the way around the fitting.
- A rib flange was added on the center rib between the clamp areas.
- A solid spar was added on both ends as if attached to the clamp structure and then spanning the clamp area.
- A rib structure was added on both the top and bottom for extra strength around the purge towers and to provide features for attaching an assembly tool should one be desired in the future.

Prior to fabricating additional prototypes and to better understand how the fitting reacts as a system, its stresses and flexural modes under pre-load and internal pressure, a finite element model was created to look at the deformation of the fitting as it is restrained by the collar clamp structure. Several iterations of changes were modeled to try and find a solution that could still be cost effectively manufactured and assembled - only one will be presented here.

Based on the result of the modeling, it was noted that as a result of the proposed changes, the separation of the fitting halves was significantly reduced and the stresses on the fitting were less than necessary to maintain a 3.125 safety factor to the yield strength of the PA 11 material. With a greatly exaggerated deformation scale, it was noted that the fitting is sufficiently restrained from bulging at the OD but begins to separate at the ID, approximately 10 mils, due to the moment created by the spar, as shown in Figure 14.

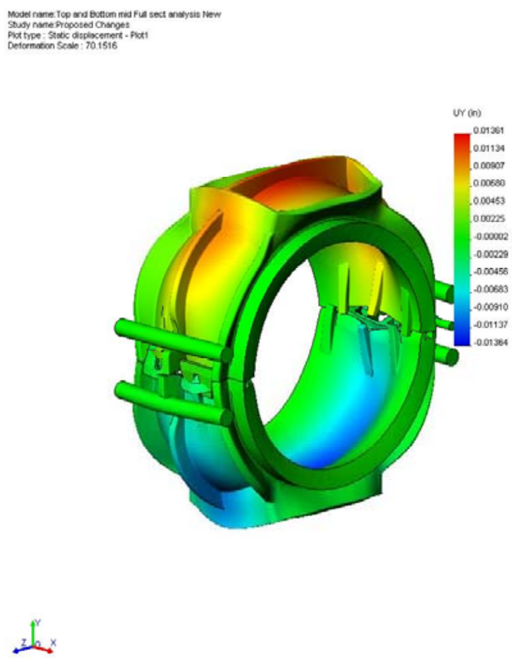

Figure 14. Design Modification to resolve the *"bulging" – Stress Distribution under load*

Consequently, the final design of the fitting halves were modified in the following ways:

- The circumferential rib depth was increased by 250 mils to provide additional stiffness in the fitting.
- No rib flange was added as in the previous models.
- Additional internal and external interlocking ribs were added to provide more support along the bottom fitting seal gland.

In addition to the design of the fitting halves, additional changes were made to the collar clamp mechanism. In all of the previous modeling, the clamping surface was restrained as fixed. However, in reality, this is not the case. Essentially, the collar clamp acts as a big spring, and the moment imparted on to the clamp by the fitting as it expands under pressure is significantly large. As a result to minimize the moment on the clamp assembly, the height of the channel was increased to even out the stresses through the clamp. This additional height also allows the spars to be spanned across the fitting from clamp to clamp at two places. Because the fitting has a tendency to elongate, out of round, under pressure, the additional spars were necessary to maintain adequate contact pressure between the fitting halves, keeping the outside edge of the fitting closed and preventing the seal from extruding.

In the proposed design enhancement, the clamp wraps farther around the fitting than it did in the previous design providing additional support near the parting line of the fitting halves. Several additional design enhancements were also introduced to the clamping assemblies:

- The clamp bar design is a single common piece instead of individually machined pieces.
- The common clamp is now the spar and lays along the OD of the ribs on the fitting.
- The clamp hinge links were modified for the new clamp geometry.
- The hinge pin is a one-piece tube that acts as a spar on the hinge side of the fitting.
- Additional bolts were added to allow for a greater pre-load and to accommodate changing to a ¼-20 thread from a ¼-28 thread.

In order to ensure that the aforementioned design enhancements effectively resolved the bulging failures noted in the previous design iteration, additional FEA analysis modeling was performed. The proposed new design was modeled as half the fitting contacting a stationary plate with approximately 700-lbs. contact force between the fitting and contact plate.

Figure 15 illustrates the displacement of the fitting in the "Y-axis" as viewed from the hinge side of the fitting with a deformation scale of 1. Figure 16 illustrates the displacement of the fitting in the Y-axis as viewed from the latch side of the fitting with a deformation scale of 5. These plots show separation of the fitting at the inside edge in the range of 3 mils to 7 mils per side or a total of 6 mils to 14 mils. The greatest amount of separation is at the corners of the model shown which would correspond to the center of the bulge in the previous models. It was observed that through the proposed design enhancements, the outside edge of the fitting is holds together preventing extrusion of the seal. The stress appears to be reasonable and the strain will well within the limits of the PA11 material (4%), as shown in Figures 17 and 18, respectively.

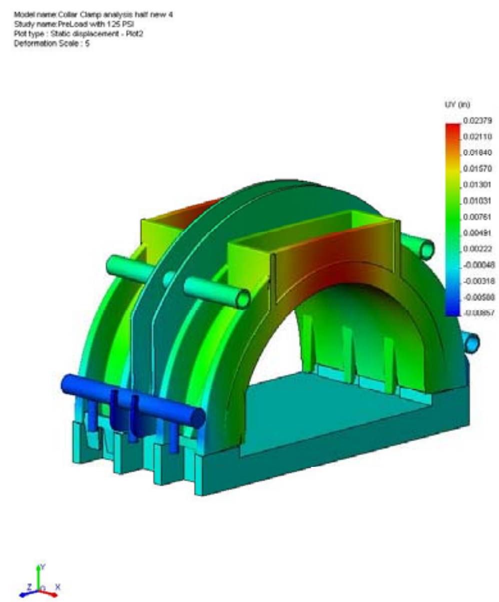

Figure 15. Displacement as viewed from the hinge side (Scale = 1.0)

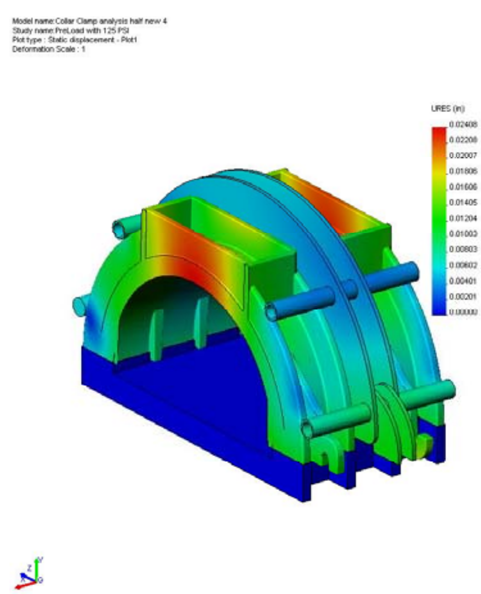

Figure 16. Displacement as viewed from the latch side (Scale = 5.0)

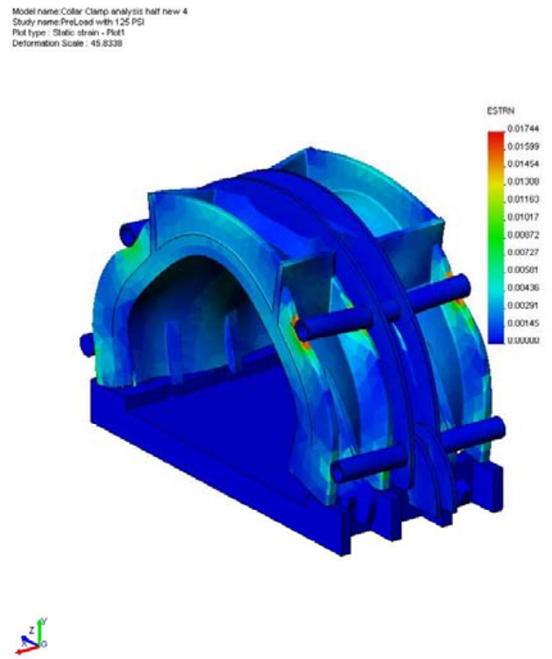

Figure 17. Stress distribution based on proposed design enhancements

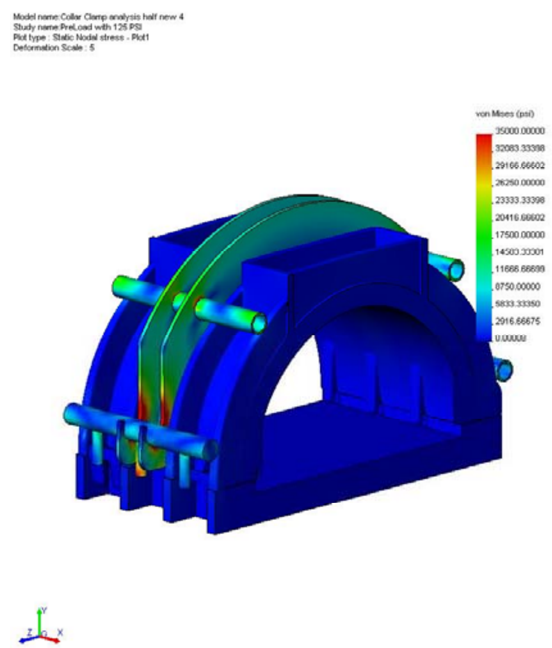

Figure18. Strain Limits (<4%) based on proposed design enhancements

Based on the positive results of the FEA modeling, the clamping structure design was fixed and prototype clamping assemblies were fabricated. The entire clamp structure was constructed from 316 CRES (Stainless Steel). Each collar clamp half was made in two pieces by piercing and blanking the outline of the collar channel and then forming the perpendicular surface. For the prototype, the blanks were laser cut in their flat form to represent a part that would be produced on a pierce and blank die. Two simple tools, one for the small collar clamp and one for the large collar clamp, were built that could be loaded in a hydraulic press brake for forming the perpendicular radial surface. Two blanks were loaded in the forming tool where one cycle of the tool produced a left and a right version of the part. Laser cutting the blank allowed for easily changing and optimizing the shape of the blank to produce a formed part needing only minimal cleanup before the welding process. The formed left and right parts were then placed into a welding fixture where a stitched seam weld was placed on the inside of the channel making the two pieces into one half of the collar clamp as shown in Figure 19.

Link plates for the clamp hinge were also laser cut for the prototypes, which were pierced and blanked from a die in production. The threaded swing pin and threaded stud were welded together to form the tee bolt for the small collar clamp. Two small collar clamp halves were assembled together with the link plates, link pins and E-rings. The tee bolt was crimped in to the saddle on one of the collar halves to capture the bolt and allow it to pivot. A Nylock nut was added to complete the small collar clamp assembly as shown in Figure 20.

Figure 19. Small collar clamp

Figure 20. Assembled small collar clamp

Four large collar clamp halves were welded to two structural clamp rods in a welding fixture to form the large collar clamp sub-assembly as shown in Figure 21.

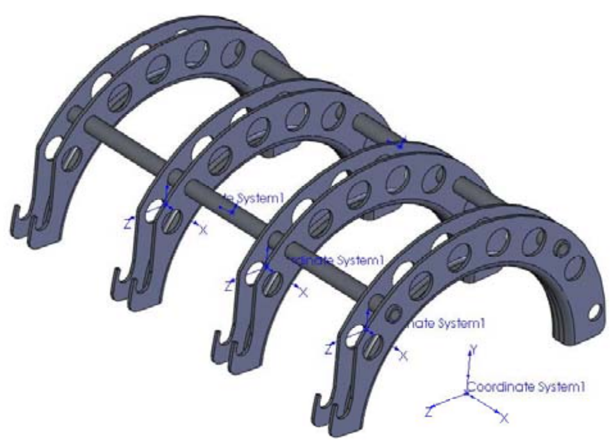

Figure 21. Large Collar Clamp Subassembly

Two large collar clamp sub-assemblies were assembled together with the link plates, structural clamp rods and roto-clips. The threaded clamp rod and threaded stud were welded together to form the ganged tee bolt for the large collar clamp assembly. The ganged tee bolt was crimped into the saddle on one of the large collar clamp sub-assemblies to capture the bolt and allow it to pivot. Nylock nuts were added to complete the large collar clamp assembly as shown in Figure 22.

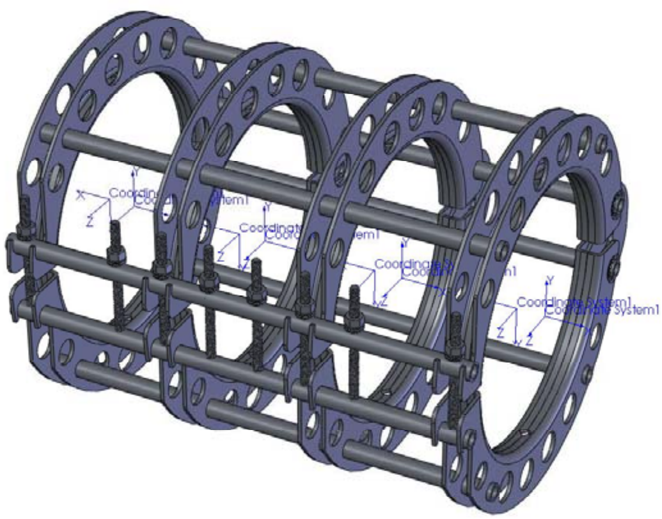

Figure22. Large Collar Clamp Assembly

Based on the positive results of the FEA modeling, the project team approved the decision to proceed with the fabrication of additional prototypes and necessary production tooling which incorporated the aforementioned design changes. The body of the prototype mechanical repair fitting was machined from solid blocks of Polypropylene material due to the lack of availability of the PA11 material. As previously noted, the major changes to the fitting design were in the area of additional ribs and increased rib height in the fitting halves and the overall clamping structure. To expedite the prototype development process, the electronic model was transferred directly to the machine shop for tool path generation without the need for time-consuming detail drawing creation. Special care was taken in cutting the seal glands into the fitting halves to ensure that the as-machined geometry would reflect that of the molded part, as closely as possible. The purge valves and purge caps were also machined from polypropylene. Figure 23 shows a cut-away view of the Generation III assembly with a purge fitting in the center port showing the valve in the open position.

Based on the aforementioned finite element modeling and pressure testing results, a final design concept was established that effectively resolved all of the pertinent issues, e.g. “bulging”, as shown in Figures 24 and 25 below. Several prototype mechanical repair sleeves were produced for comprehensive evaluation. Figure 26 shows a representative prototype assembly (Generation III) with all of the aforementioned design enhancements.

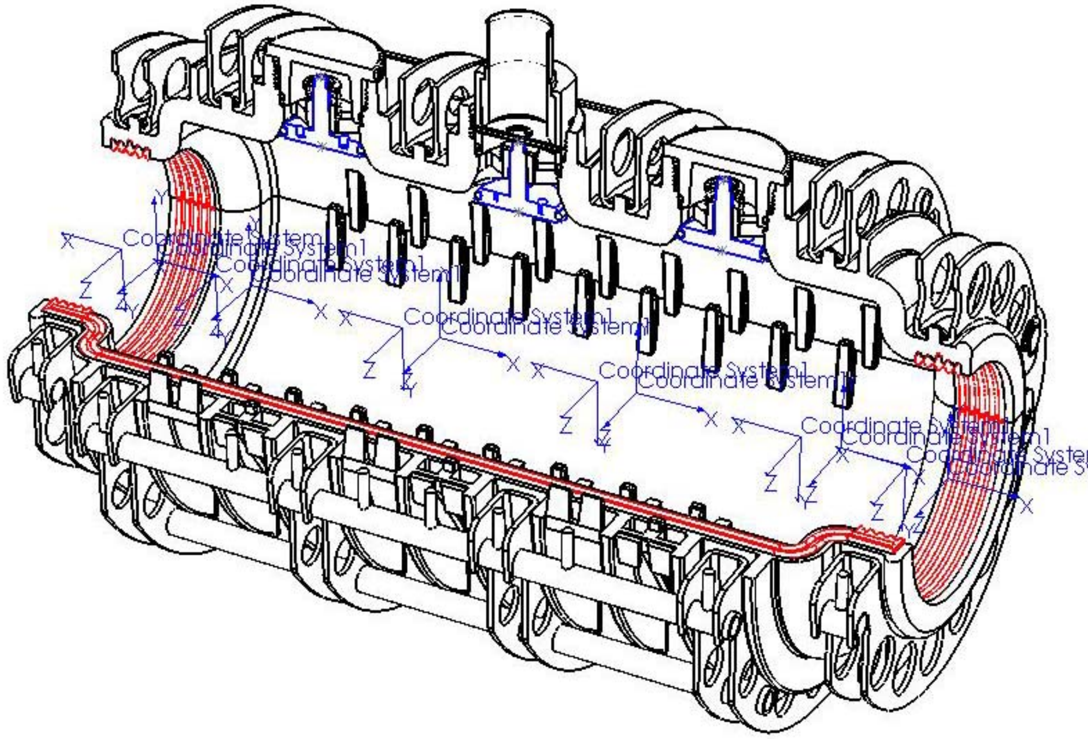

Figure 23. Generation III Assembly

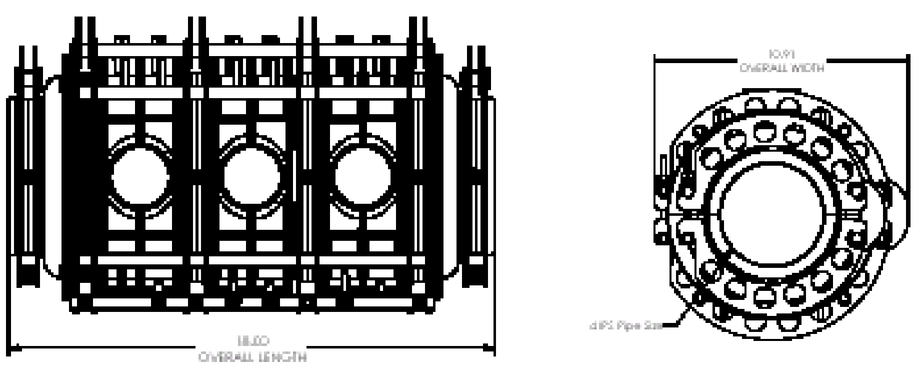

Figure 24. Schematic Illustration of the 2nd Generation Mechanical Repair Fitting – Top View and Side View

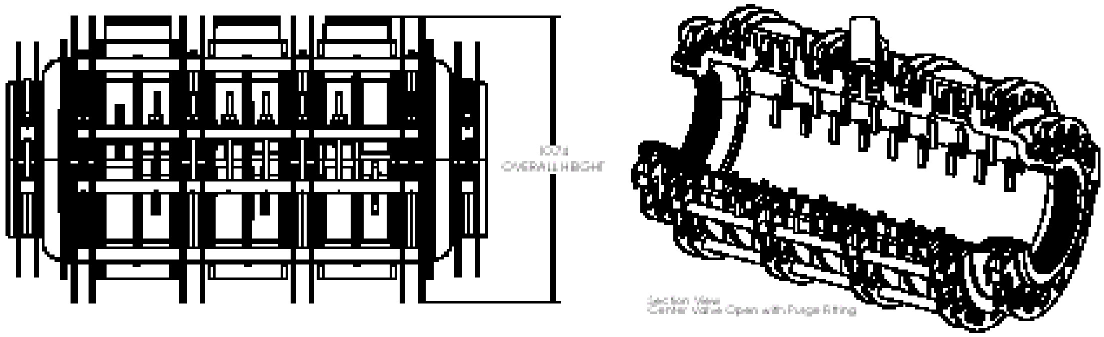

Figure 25. Schematic Illustration of the 2nd Generation Mechanical Repair Fitting – Front View and Cut-out

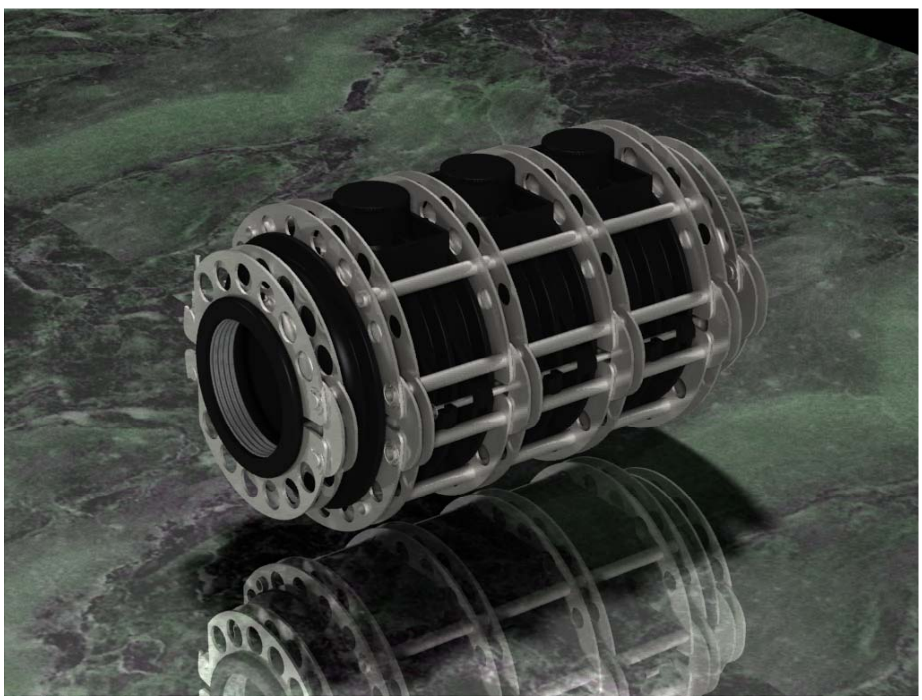

Figure 26. Final Prototype

Installation Testing

As previously discussed, a key element in the design and development process was the ability to install the fitting under blowing gas conditions thereby eliminating the need for additional excavations to reduce the line pressure, and the installation of fittings for an external bypass. To that end, several prototype fittings were evaluated at the GTI test flow loop using a large-scale compressor to simulate actual flow conditions.

A ½" diameter hole was made on the 4-inch pipe and the entire test loop was pressurized. Initially, evaluations were performed at reduced line pressures starting at 20 psig and then increased uniformly up to 60 psig. The results of the testing using a limited number of prototype fittings demonstrated that the fitting could be installed under live blowing gas conditions at line pressures up to 60 psig without the need to reduce the flow. It is important to note that while the results were promising, a significant amount of training and operator experience that will be required prior to widespread use by gas utility companies. The fitting is not intended to be the proverbial "sliver bullet" that can be installed in all types of field situations that can be potentially encountered. Significant amount of operator experience and training will be required to carefully assess the use of the mechanical repair fitting as compared to conventional repair technique of cutting and splicing in a new piece of pipe.

Qualification Testing

Having verified the ability to install the prototype fittings under blowing gas conditions, the next major challenge was to validate the long term performance of the mechanical repair fitting under typical service conditions. Specifically, additional testing was required to ensure that the fitting provides a leak tight seal at typical operating pressures (60psig) over its intended design life.

In order to determine the relevant testing protocols, GTI undertook a comprehensive review of existing ASTM standards and specifications and other applicable industry acceptance requirements. In order to qualify a particular product (pipe and fitting) for use in gas distribution applications, the product must satisfy ASTM D2513-99 requirements. While ASTM D2513 contains numerous requirements, many of these requirements are applicable to plastic pipe. However, under Section 5.9 "*Joints*", ASTM D2513-99 lists specific requirements for mechanical joints based on a series of classification criteria for the particular joint under consideration. Upon further review of the various categories listed, the project team concluded that the proposed mechanical repair fitting which was developed as part of this program did not fall into any of the listed categories.

In addition to ASTM D2513, additional qualification requirements for conventional mechanical fittings (in-line compression fittings and mechanical saddle fittings) are governed by ASTM F1924 entitled: *"Standard Specification for Plastic Mechanical Fittings for Use on Outsider Diameter Controlled Polyethylene Gas Distribution Pipe and Tubing"*. Per F1924 specifications, mechanical fittings that are suitable for use in gas distribution applications must satisfy the following requirements:

- *Sustained Pressure Testing at Elevated Temperature*
- *Tensile Strength Determination*
- *Temperature Cycling Testing*
- *Constant Load Joint Tensile Test*

Based on a thorough review of the requirements for conventional mechanical fitting and the prototype fitting design, it was readily apparent that ASTM F1924 requirements might not be suitable for the proposed mechanical repair fitting design developed within this program.

Specifically, per ASTM F1924 requirements, six specimens of the pipe/fitting assemblies must be subjected to long term sustained pressure testing at elevated temperatures for a prescribed set of time/temperature/stress combinations: 670 psi at 80°C for 170 hours or 580 psi at 80°C for 1000 hours. For a SDR 11 pipe size, this corresponds to a test pressure of 134 psig for 170 hours or 116 psig for 1000 hours at 80°C. These respective test pressures are significantly greater than the design constraints for the fitting clamping structure, which was designed to withstand internal pressure up to 90 psig at 73°F (1.5 times the operating pressure).

Given the unique design characteristics and the intended application, it was concluded that the proposed mechanical repair fitting, two thermoplastic shells clamped together, is significantly different than the conventional mechanical fittings sold in the marketplace and hence require modified test requirements.

For the case of the mechanical repair fitting developed as part of this program, it was noted that the primary area of focus was to ensure a leak tight seal at typical operating pressures over its intended design life. That is, the mechanical repair fitting should provide a leak tight seal at operating pressures of 60 psig for a 50-year design life. Based on input from utility sponsors and the manufacturing partner, a series of modified tests requirements and protocols were developed to validate the long term performance of the proposed mechanical repair fitting design based on actual service conditions.

Burst Test

Although the minimum hydrostatic burst pressure test (“quick burst”) is not a specific requirement even for conventional mechanical repair fittings, this test was performed to evaluate the performance limits of the mechanical repair fitting seals and clamping structure. The quick burst test is a useful test to determine the hydraulic burst pressure of pipe specimens in a short period of time (typically 60 to 70 seconds). Short lengths of pipe specimens are filled with water and submerged in a water bath at 73F. The pressure is then uniformly increased until the pipe specimen ruptures. The typical burst pressure for modern polyethylene piping is approximately 3000 psi (600 psig).

From the onset, it was recognized that the proposed mechanical repair fitting would fail prior to the pipe material – the fitting clamping structure was designed to withstand maximum internal pressures of up to 90 psig. Regardless, in order to verify the upper pressure limits to safeguard against potential over pressurization in the field, two prototype fittings were installed on 4-inch pipe specimens containing a ½" diameter hole. The pipe specimens were capped and the entire pipe and fitting annular space was filled with water. The pipe/fitting assembly was then tested at 73F.

As the pressure was uniformly increased in the first test specimen, the fitting began to initially leak at approximately 180 psig. The fitting seal extruded out at the center of the fitting at approximately 289 psig. The mode of failure was similar to the sustained pressure testing that was previously performed to evaluate the integrity of the clamping structure – see Figure 11.

The second test specimen, like the first one, began to initially leak at approximately 170 psig. The test was discontinued at 245 psig when the seal completely extruded out. Again, the mode of failure was consistent with the failure mode observed in the first specimen.

While the results of the testing were consistent with expectations that the fitting will fail prior to the pipe, the overall data confirmed that design changes to the clamping structure significantly increased the pressure carrying capability of the mechanical repair fitting. As previously discussed, prior to the design changes, the initial prototype seals extruded out at approximately 90 psig. Based on the results of the quick burst testing for the Generation III prototype fittings, the respective pressures at failure were approximately three times greater than the normal anticipated operating pressure of 60 psig (failure point is defined when the fitting(s) begin to leak). In addition, the results of the testing helped to provide further insight into the possible time/temperature/stress requirements for long term sustained pressure testing at elevated temperatures, which is outlined in the next section.

Sustained Pressure Testing

While the preceding discussion demonstrated that there is a significant margin of safety in the fitting clamping structure against over pressurization, the results do not adequately address the long term performance of the fitting design over its intended design life.

As previously noted, in order to address long term performance considerations, conventional mechanical fittings are subjected to long term sustained pressure testing at elevated temperature for a set of prescribed time/temperature/stress conditions as outlined in ASTM F1924. Given the unique design characteristics of the proposed mechanical repair fitting, the respective time/temperature/stress combinations outlined in ASTM F1924 are significantly greater than the design constraints for the fitting clamping structure, which was designed to withstand internal pressure up to 90 psig (1.5 times the operating pressure) at 73°F.

Prior to a detailed discussion about the testing, two key points must be emphasized:

1. The long term sustained pressure test at elevated temperatures is a useful to test to measure the long term performance of thermoplastic pipe materials in an accelerated manner. That is, for a prescribed set of time and stress conditions at 73°F, as one increases the test temperature, the corresponding time and stress conditions to achieve failure at an elevated temperature is considerably reduced. Therefore, the same stress state at 80°C is significantly lower than the corresponding stress state at 73°F.
2. The conventional mechanical fittings (in-line compression fittings and mechanical saddle fittings) sold in the marketplace are significantly different than the proposed mechanical repair fitting design. The proposed mechanical repair fitting is essentially two thermoplastic shells clamped together, i.e. an encapsulation device made up of several different components with varying performance limitations for each of them. The long term performance criterion of the combined pipe/fitting must be such that it takes into account the different long term performance considerations for each material. Therefore, the test criterion must be predicated on the weakest component(s), which for the case of the proposed mechanical repair fitting are the elastormeric materials in the seal(s) and the overall strength of the clamping mechanism design.

As a first approximation, taking into account the aforementioned points, the project team developed modified time/temperature/stress combinations using the bidirectional shift functions to assess the long term performance of the mechanical repair fittings.

In general, the bidirectional shift functions are a widely accepted technique to transfer data from a given time, temperature, stress state to another time, temperature, stress state through the use of the following formulas:

$$\begin{aligned} a_T &= \exp\left[0.109\left(T_t - T_s\right)\right] \\ t_s &= t_t a_T \end{aligned} \qquad (1)$$

$$\begin{aligned} b_T &= \exp\left[-0.0116\left(T_t - T_s\right)\right] \\ p_s &= \frac{p_t}{b_T} \end{aligned} \qquad (2)$$

where:

a_T = time dependent shift function parameter
b_T = pressure dependent shift function parameter
T_t = Test temperature
T_s = Service temperature
P_t = Test pressure
P_s = Service pressure
t_t = Test time
t_s = Service time

For the case of the proposed mechanical repair fitting design, the primary consideration was to evaluate the long term performance at an operating pressure of 60 psig at 73°F for an intended design life of 50-years (438,000 hours). Substituting into equations (1) and (2), the design team developed appropriate time and stress conditions at 80°C which corresponded to the desired design conditions at 73°F.

$$
\begin{aligned}
a_T &= \exp[0.109(T_t - T_s)] \\
a_T &= \exp[0.109[80 - 23]] \\
a_T &= 499.2 \\
\frac{t_s}{a_T} &= t_t \\
\frac{438{,}000}{499.2} &= t_t \\
t_t &= 877
\end{aligned}
\tag{3}
$$

and,

$$
\begin{aligned}
b_T &= \exp[-0.0116(T_t - T_s)] \\
b_T &= 0.516 \\
p_s &= \frac{p_t}{b_T} \\
p_t &= p_s b_T \\
p_t &= 60(0.516) \\
p_t &= 31
\end{aligned}
\tag{4}
$$

Therefore, from Equations (3) and (4), in order to validate the long term performance of the prototype fitting (60 psig at 73°F for a 50-year design life), the prototype fittings must be tested using an internal pressure of 31 psig at 80C.

Three prototype fittings were installed on 4-inch polyethylene pipe containing a ½" diameter hole. The pipe and fitting annular space were filled with water and conditioned in a water bath at 80°C for 1 hour prior to testing. The test specimens were then pressurized to the respective test pressure and the time to failure was monitored. The results of the testing are summarized in Table 4.

Specimen	Test Pressure (psig)	Test Temp. (C)	Time to Failure (hr)
1	31	80	> 216
2	31	80	> 54
3	45	80	> 960

Table 4: Results of the long term sustained pressure testing at 80C

Based on the results of testing a limited number of fittings, it was reasonably inferred that the long term performance of the prototype fitting under the influence of typical in-service pressures (both within the pipe and fitting annular space) was positive. Based on the results of specimen 3, which was tested at 1.5 times the test pressure corresponding to 60 psig at 73F, the results demonstrated that there were no failures beyond the 877 hours required to substantiate the 50-year design life.

Thermal Cycle Testing

In addition to the quick burst test and long term sustained pressure test discussed in the preceding sections, several prototype fittings were subjected to the temperature cycling tests. Given the fact that the proposed mechanical repair fitting is intended to serve as a permanent repair option, its behavior under the influence of varying temperature cycles is critical. Often, buried underground gas distribution pipe and fittings are subjected to varying thermal stresses resulting from seasonal and climatic fluctuations.

As a first approximation, in order to determine the effectiveness of the mechanical repair fitting to varying thermal stresses, the project team resolved to subject the proposed mechanical repair fitting using the test methodology contained within ASTM F1924 standard. Per ASTM F1924 requirements, the mechanical joint must be capable of providing a pressure seal after 10 cycles of temperature cycling between –20°F and 140°F at both 7 psig and 1.5 times the maximum intended operating pressure.

Several prototypes fittings were installed on 4-inch polyethylene pipe with a 3/8" diameter hole. The pipe and fitting annular space were pressurized to the prescribed test pressures (7 psig and 90 psig). The pipe/fitting assemblies were then subjected to temperature cycling between –20°F and 140°F. The assemblies were allowed to come to equilibrium at the prescribed test temperature for approximately 2.5 hours as prescribed under ASTM F1924 requirements and monitored for leaks. The results of the testing demonstrated that the all of the prototype fittings began to leak well before the 10 cycles as prescribed under ASTM F1924 requirements.

The results of the testing were inconsistent with expectations; however, they reinforced the observations seen in other tests including the quick burst test and long term sustained pressure test, i.e., ASTM F1924 requirements may not be suitable for the proposed mechanical repair fitting design. The proposed mechanical repair fitting is significantly different as compared to convention mechanical fittings. The proposed mechanical repair fitting is essentially an encapsulation device (pressure vessel) that is comprised of several different components made from different materials. As a result, its performance limitation is predicated on the weakest components, which are the seals and clamping structure.

Therefore, in order to better understand the performance limitations of the seals and clamping mechanism under the influence of temperature cycling, the project team developed modified testing protocols simulating more realistic test conditions corresponding to actual field conditions. Based on the results of previous research, it was noted that buried underground piping is not subjected to –20°F and 140°F temperatures. Furthermore, the temperature fluctuations that occur in service are extremely gradual and take place over an extended period of time.

To that end, an additional round of temperature cycling tests were performed. For this round of testing, the temperature range was fixed between 30°F and 100°F. The rate of change between the temperature extremes was set at 3°F/hr.

Six (6) new prototype fittings were installed on 4-inch polyethylene pipe with a 3/8" diameter hole per R.W. Lyalls instructions: "secure the clamp hardware; torque the two end clamps to 85 in-lbs; torque the four primary bolts on the large clamp assembly to 75 in-lbs; torque the secondary bolts on the large clamp assembly to 45 in-lbs; the two end clamps and the four primary bolts on the large clamp must have the pre-load washers installed for the thermal cycle test". The pipe and fitting annular space was pressurized with air at 60 psig and placed in the environmental chamber.

The environmental chamber was programmed to cycle the test temperature from 72°F to 30°F at a rate of 3°F/hr. The test specimens were held at 30°F for 1 hour. The chamber test temperature was then increased up to 100°F and held for 1 hour. After 1 hour, the test temperature inside the environmental chamber was cycled back to 72°F. Therefore, the overall duration of one cycle corresponded to 49 hours (2 days).

Due to the limitations of the test apparatus, the test specimens were visually monitored for any significant pressure drop. There were no leaks that were observed after the first cycle in any of the test specimens. However, there were several leaks that were observed in various test specimens during the remaining cycles. Due to limitations of the test apparatus, the exact temperature at which the leaks initially occurred could not be ascertained. Regardless, by the end of the fifth cycle, all of the fittings had experienced appreciable leaks at test temperatures between 30°F and 100°F.

Based on the results of the testing, it was theorized that the preload on the large clamp assembly was not sufficient to account for the large scale thermal stresses resulting from temperature fluctuations. As a result, the project team decided to increase the preload on the large clamp assembly and verify the ability of the clamping mechanism to withstand the temperature fluctuations.

Owing to the paucity of information that was collected during the previous round of testing, the test apparatus was slightly modified in order to gain further insight into the exact temperatures at which the test specimens begin to leak. A new manifold was built with discrete flow gauges inline for each fitting. The gauges were arranged to be visible through the observation window from out side of the chamber. A video camera was placed outside the chamber and focused on the manifold gauges to monitor the test over the ten days it would take to complete the five cycles. Figure 27 illustrates the modified test apparatus.

Figure 27. Modified Thermal Cycling Test Apparatus

As was the case in the previous round of testing, six (6) new prototype fittings were installed on 4-inch polyethylene pipe with a 3/8” diameter hole per the manufacturers instructions. However, during this round of testing, the applied torque was increased to 100 in-lb on the large clamp assembly, and a stiffer stack of pre-load washers was used.

In order to develop more quantitative data, video footage was captured at 2 sec/10 min intervals throughout the test. The data was then plotted and analyzed after each cycle. Like the previous round of testing, the environmental chamber was programmed to cycle between 30°F and 100°F at a rate of 3°F/hr.

After the first cycle, there were no leaks on any of the newly installed six specimens with the increased pre-load and stiffer washers. The results of the testing were significantly better than the previous round of testing. Based on the positive results, a decision was made to install the stiffer washers on the previously tested pipe/fitting specimens (from the first round of testing) with an increased pre-load. Therefore, a total of 12 test specimens were placed in the chamber. The testing was suspended for less than an hour to allow the fittings from the previous round of testing to be put back on test. Figures 28-31 presents the leak rate as a function of temperature for each cycle.

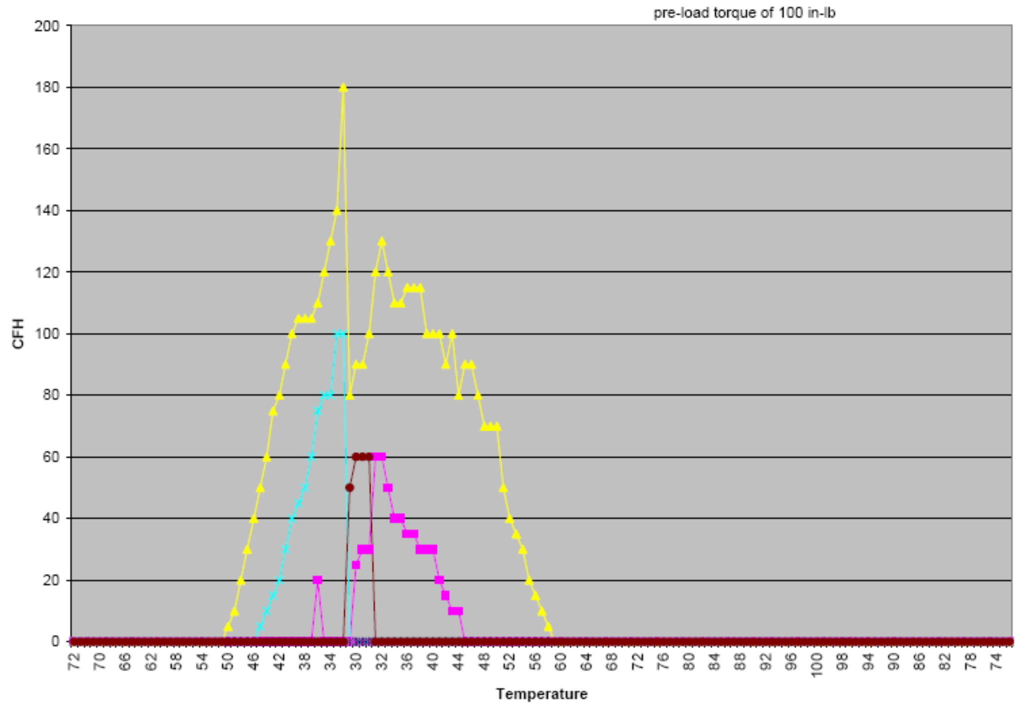

Figure 28. 2nd Test – Cycle 2

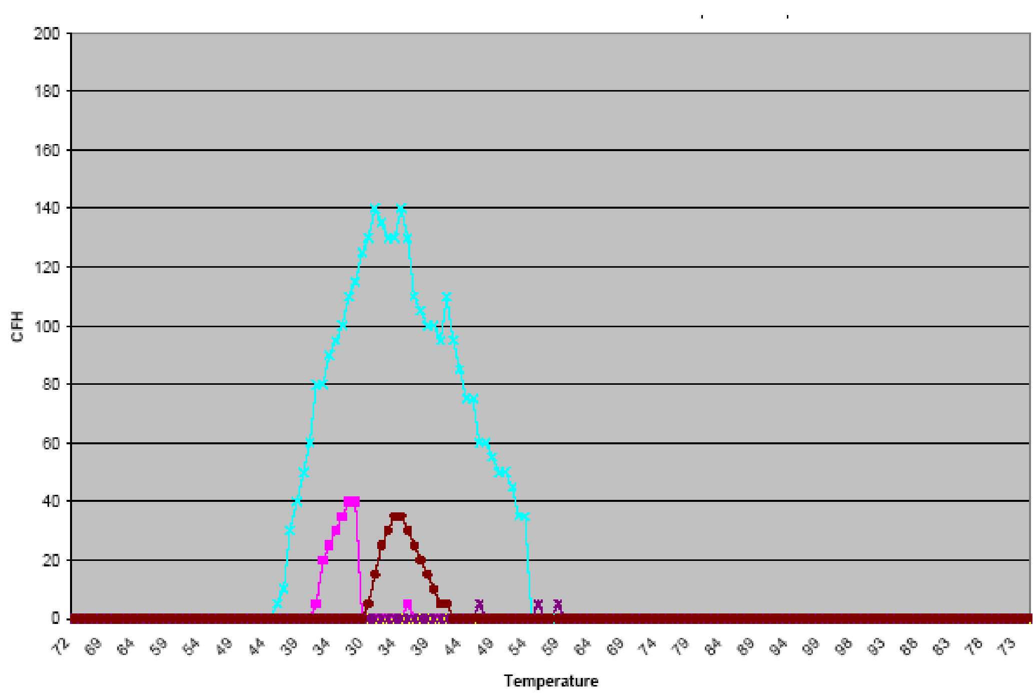

Figure 29. 2nd Test – Cycle 3

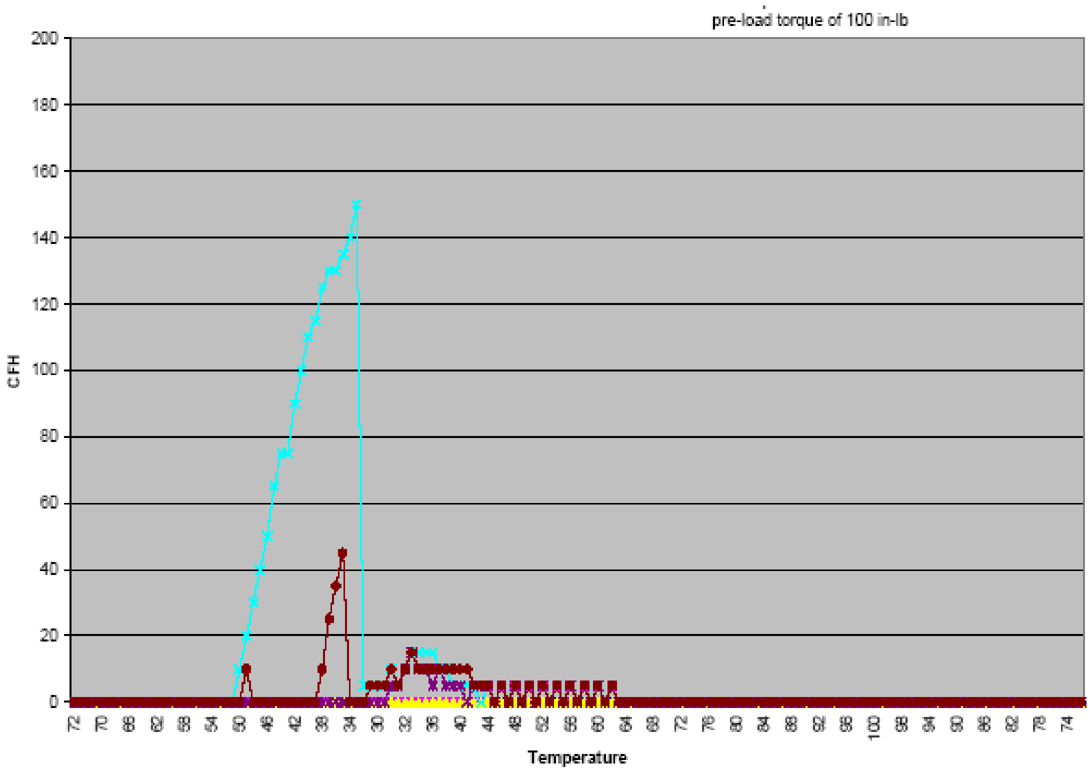

Figure 30. 2nd Test – Cycle 4

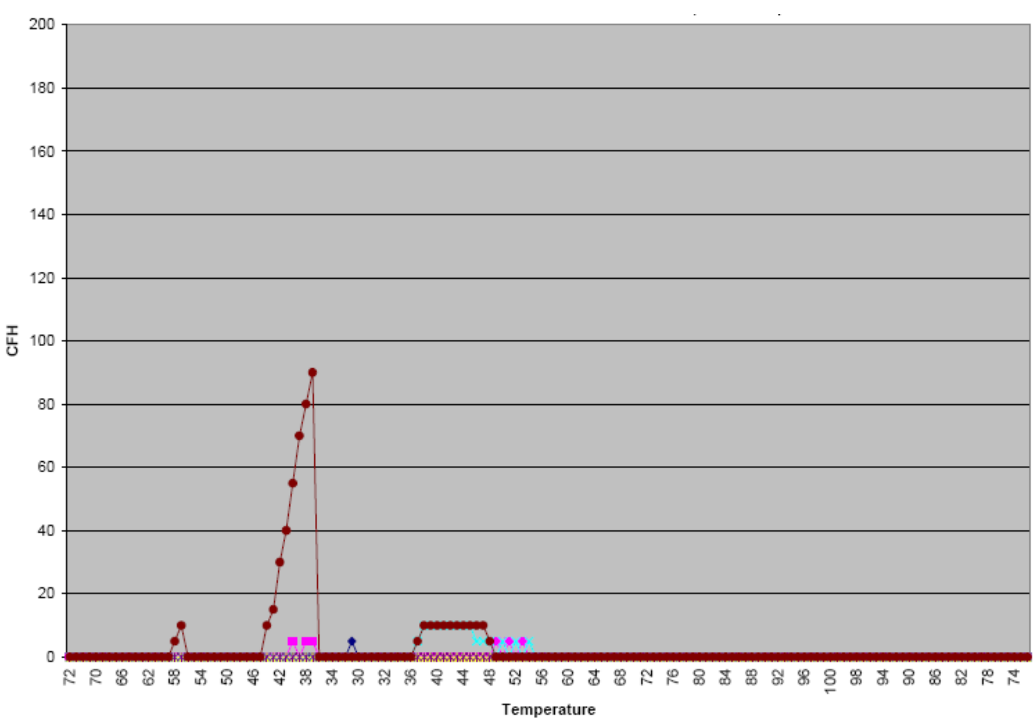

Figure 31. 2nd Test – Cycle 5

Based on the results of the testing, there were several observations:

1. There were no leaks in any of the six specimens (second round of testing) with an increased pre-load after the first cycle.
2. One of the six specimens from the second round of testing leaked appreciably after the second cycle at lower temperatures but resealed on itself as the test temperature was increased.
3. There were no appreciable leaks in any of the specimens from the second round of testing throughout the remainder of the cycles. In the cases where there were leaks, all of the specimens leaked at the lowest test temperatures and resealed as the temperature was increased.
4. ***NONE of the specimens which were tested in the previous round of testing and reinstalled using a higher pre-load leaked during the entire second round of testing.***
5. The thermal cycling requirements contained within ASTM F1924 specification may not be suitable for evaluating the long term performance under the influence of temperature changes.

RESULTS AND DISCUSSION

The objective of this program was to develop a permanent mechanical repair fitting that could be installed on damaged PE gas piping under system operating pressure. From the onset of the program, with input and guidance from the project team, a set of minimum criteria was established to help guide the overall development effort including:

- Initial design and development efforts should focus on 4-inch pipe size operating at 60 psig.
- Once installed, the fitting design should effectively mitigate the continued propagation of the damage via the slow crack growth (SCG) failure mechanism.
- The fitting should be capable of being installed under blowing gas conditions at typical line pressures (60 psig) without the need to incorporate any flow control measure (valves, squeeze-off, etc).
- The fitting should conform to applicable ASTM standards and specifications (ASTM D2513 and F1924 requirements, as applicable) and/or other industry accepted requirements. At a minimum, the fitting should be capable of withstanding the operating pressures and providing a leak tight seal over its intended design life.

Based on the aforementioned product definition, a viable mechanical repair fitting design satisfying many of these criteria has been developed. From the onset, it was recognized that the design team needed to take into account several key considerations including: the fitting's ability to mitigate failure by the slow crack growth mechanism on the damaged pipe under repair; the fittings ability to be safely installed under blowing gas conditions; and the fittings ability to withstand the system operating pressure over its intended design life.

Following an iterative design and development process, R. W. Lyall, under the direction of GTI, developed a suitable best-case design concept using various modeling and stress analysis techniques. A hybrid approach was undertaken to evaluate the propensity for continued growth of the damage beyond the end seals once the fitting has been installed. The results of the analytical modeling, based on worst case assumptions, confirmed that for all types of PE materials, the potential for continued crack growth in the axial direction is relatively low. Subsequent laboratory testing confirmed the results of the analytical model. Specifically, the results indicated the equalization of the pressure in the pipe and the corresponding annular space in the fitting removes the necessary crack driving force and therefore effectively mitigates the continued growth of the crack in the axial direction.

In addition to ensuring that the fitting effectively mitigates that continued growth of the damage beyond its end seals, another critical design parameter was the ability to install the fitting under blowing gas conditions without reducing the flow upstream of the damage. This functionality provides a significant cost savings to gas utility companies. If gas utility companies are required to incorporate flow control techniques prior to the installation of the mechanical repair, then many gas companies may prefer to replace the damaged sections completely with new polyethylene piping as compared to installing a repair fitting. As a result, the project team concluded that the ability to install under a blowing gas condition must be a priority. However, the ability to install the mechanical repair fitting under blowing gas conditions presented significant technical challenges including:

- The pipe area around the damage cannot be "prepped", i.e., the remaining ligament with its irregular geometry and jagged edges cannot be cut-out or reduced in size. As a result, the annular space between the outside diameter of the pipe and inside of the fitting needed to be significantly large. The large annular space, in turn, requires a large clamping force to keep both halves of the fitting closed at typical operating pressures (60 psig).
- The ability to install under blowing gas conditions implies that vent ports must be incorporated within the fitting design to facilitate the flow of gas outside of the trench. By incorporating vent ports, the fitting design for both halves will not be symmetrical and subsequently require separate tooling for each half.

Having resolved the key technical challenges, R.W. Lyall developed the actual tooling needed to fabricate prototype parts for testing. Several prototype parts were subjected to comprehensive testing to evaluate the fitting's ability to be installed under blowing gas conditions. Additional testing was performed to evaluate its ability to withstand typical operating pressures and provide a leak tight seal over its intended design life.

Based on the results of a limited number of simulated installation tests, it was shown that the fitting could be manually installed under blowing gas conditions at 60 psig without having to reduce the flow. However, it is important to reiterate that significant amount of operating training will be required assuming that some of the minor technical issues are successfully resolved. In actual field settings, it is very difficult to accurately assess the extent of the damage given that the flow is not completely shut-off. Therefore, gas utility companies will have to develop comprehensive training guidelines to ensure operator safety.

Given the unique form and function of the proposed mechanical repair fitting, it was noted that the existing standards (ASTM F1924) intended for conventional mechanical repair fittings are not directly applicable. Specifically, the test requirements related to the sustained pressure testing at elevated temperatures are overly aggressive. Per ASTM F1924, conventional mechanical fittings are subjected to a test stress of 670 psi (134 psig test pressure) at 80°C, which is greater than the design pressure limitations of the clamping structure (90 psig max. at 73°F). Therefore, modified test protocols specific to the proposed mechanical repair fittings were developed to validate safe long term performance corresponding to actual service conditions. Based on the results of the modified sustained pressure testing requirements, the results demonstrated that the fitting could provide a permanent seal at an operating pressure of 60 psig at 73°F over an intended 50-year design life.

While the results of the sustained pressure testing were positive, additional tests were performed to evaluate the mechanical repair fitting's ability to withstand thermal stresses due to temperature cycling. The results of these tests were mixed. During the initial round of testing, the prototype mechanical repair tended to leak at lower temperatures. To resolve this issue, an additional pre-load was placed on the clamping structure and the fittings were re-tested. Like the previous round of testing, the fittings tended to leak at the lower temperatures but resealed onto themselves as the temperature was increased.

Based on the cumulative results of the testing, it is reasonable to infer that some additional work is still necessary to resolve the leaks observed in the temperature cycling tests. As previously discussed, none of the fittings that were re-tested with an increased pre-load failed under the temperature cycling tests. This implies that some minor adjustments may be necessary to the overall manufacturing process – some additional annealing to eliminate the inherent residual stresses. In addition, some additional small-scale modifications may be necessary to the overall clamping structure – possibly an increased pre-load on the nylock nuts in the large collar clamp assembly.

Nevertheless, the results of the testing demonstrate that the fitting can be a potentially viable repair option for gas utility companies as these technical issues are successfully resolved. In addition, while the results of the qualification testing appear to be promising, they also indicate that test requirements pertaining to conventional mechanical repair fittings are not be appropriate. As a result, additional work must be performed to ascertain the true performance limits of this particular design and then incorporate those requirements within ASTM F1924, and/or develop a completely new specification specific to the mechanical repair fitting.

CONCLUSION

When a steel pipe is gouged or is ruptured, it is not uncommon for a welder to repair the damaged pipe by welding a steel repair sleeve or full-encirclement fitting over the compromised area. However, in the plastic industry, there are no fittings currently available to mirror this process on polyethylene pipe (PE) that can act as a permanent repair.

GTI and R. W. Lyall, under the auspices of DOE NETL and gas utility companies, carried out a comprehensive program to develop a plastic pipe mechanical repair fitting that can be installed on damaged 4” polyethylene (PE) pipe under system operating pressure. Several concepts were modeled and produced using rapid prototyping technologies, and functional prototypes were built and tested.

The final mechanical repair fitting design essentially consists of two half circular cylindrical parts that are hinged together. After they have encircled a pipe segment that has been damaged, these two parts can be mechanically fastened to each other to contain the damage. The fitting is maintained in position by compressive forces between the elastomeric seal of the fitting body and the damaged pipe. There is an annular cavity between the inner wall of the fitting sleeve and the outer wall of the damaged portion of the pipe that is contained within the fitting body to effectively mitigate the continued growth of the damage beyond the ends of the fitting length.

Because the mechanical repair fitting is intended to cope with the entire range of damage that can occur in service, the principal design challenge was the repair of a “blowing” failure; i.e., a through wall opening in the pipe wall through which pressurized gas escapes. Therefore, the mechanical repair fitting incorporated a unique design feature, which consisted of three axially aligned holes that allows the gas to escape during the repair procedure.

Following the successful design of the fitting, prototyped assembles were constructed and subjected to comprehensive testing under laboratory controlled conditions to determine the specific performance characteristics of the mechanical repair fitting. The fitting in its present state did not satisfy all of the qualification testing. Specifically, the fitting failed to provide a leak tight seal under the influence of temperature fluctuations. As a result, based on the cumulative results of the testing, some additional work is necessary. Specifically, minor adjustments to either the fitting design and/or the fitting manufacturing process are necessary to resolve the leak issues at lower temperatures.

Regardless, overall the results of this program have established the fundamental groundwork related to the design and development of a permanent plastic mechanical repair fitting for use on damaged PE gas mains in a safe and cost effective manner.

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List of Acronyms and Abbreviations

PE	Polyethylene
PA11	Polyamide 11
ASTM	American Society for Testing and Materials
SCG	Slow Crack Growth
LEFM	Linear Elastic Fracture Mechanics
MRS	Mechanical Repair Sleeve
OD	Outer Diameter
ID	Inner Diameter
IPS	Iron Pipe Size
SDR	Standard Dimension Ration (Ratio of OD/wall thickness)
GTI	Gas Technology Institute
K	Stress intensity factor
a	crack depth
c	one-half of the surface length of the crack
h	wall thickness
σ	remote tensile stress
ϕ	angle of penetration
K_D	Stress intensity factor
t_i	time required for SCG to be initiated
B	material constant
A	material constant
m	material constant