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Fatigue design sensitivities of stationary type 2 high-pressure hydrogen vessels

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U.S. DEPARTMENT OF ENERGY NNSA National Nuclear Security Administration

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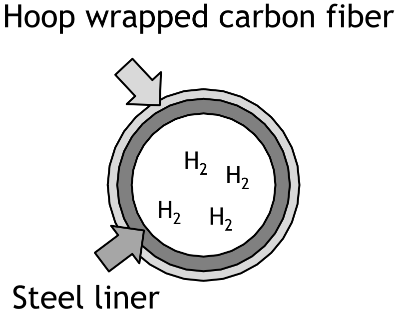
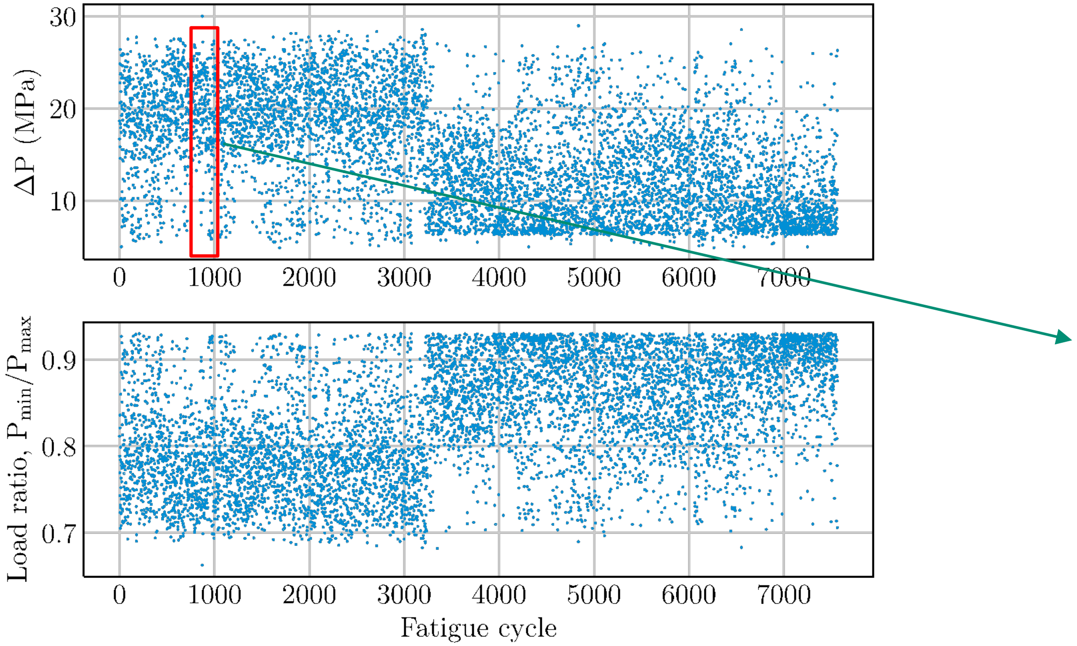
Type 2 high-pressure hydrogen storage vessels



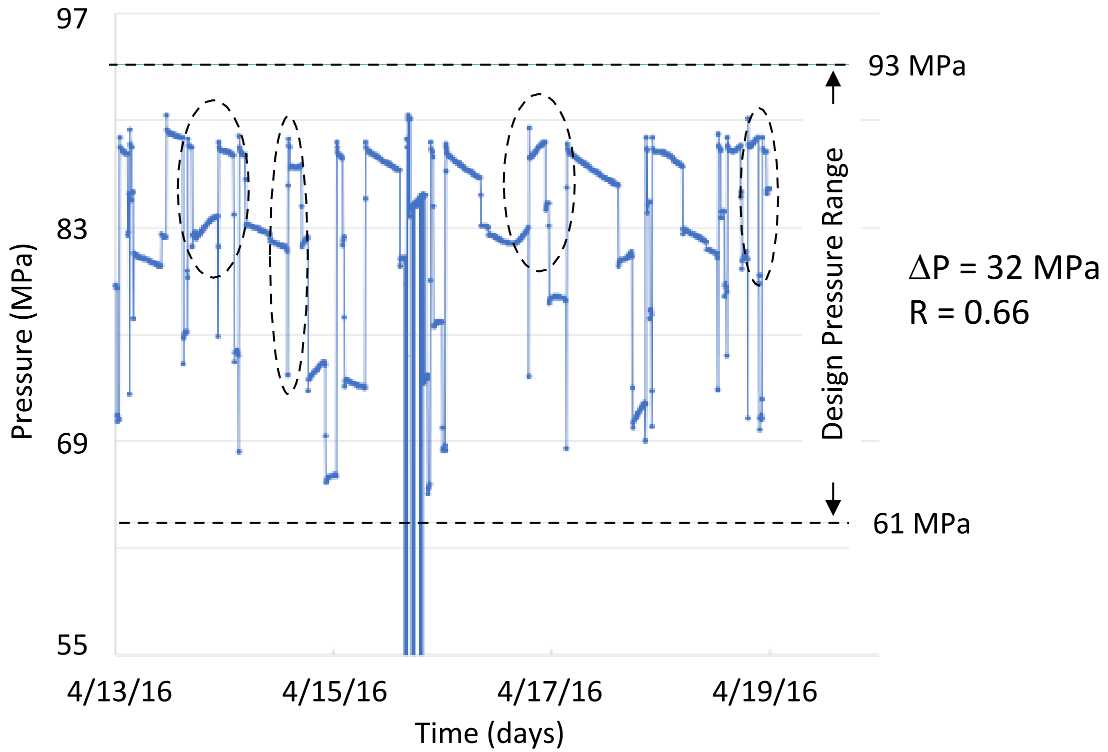
*Type 2 vessels are commonly used at Hydrogen Refueling Stations (HRS)

- → Type 2 tanks are carbon-fiber overwrapped and autofrettaged
- → **Fracture mechanics design** approach: ASME BPVC Section 8 Div 3
- → Design cycle life is defined in User's Design Specification (UDS) **based on a max pressure range**
  - e.g., Pressure range 93 MPa to 61 MPa, Design Life = 37,540 cycles or 20 yr
- → Tanks are reaching cycle limit **much sooner** than desired (e.g. 7 yr)
- → Conventional non-destructive evaluation (NDE) methods to inspect metal liner are incompatible with overwrap; therefore no means to inspect, recertify, and extend life of tank → ***Result = tanks are retired***

In-service pressure cycle data from retired tanks



Pressure cycle data from in-service tanks



→ Real data for approximately 3 years of service show usage rarely matches UDS max pressures and load ratio

- Median pressure range ΔP = 14.6 MPa vs. ΔP_{UDS} = 32 MPa

→ No guidance and considerable savings to be realized if fatigue life can be more closely tied to actual use

Research questions:

1. Can we identify a threshold pressure range ΔP_{th} below which we can assume negligible crack growth?

Autofrettage simulation to determine residual stress

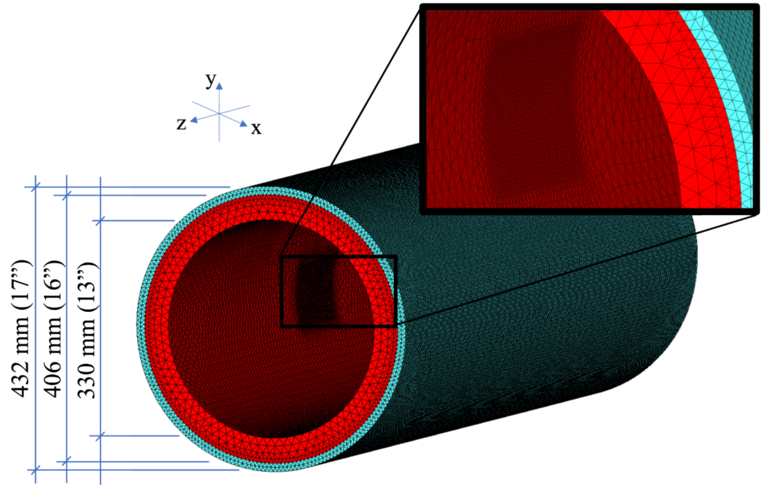
FE model includes the liner and overwrap

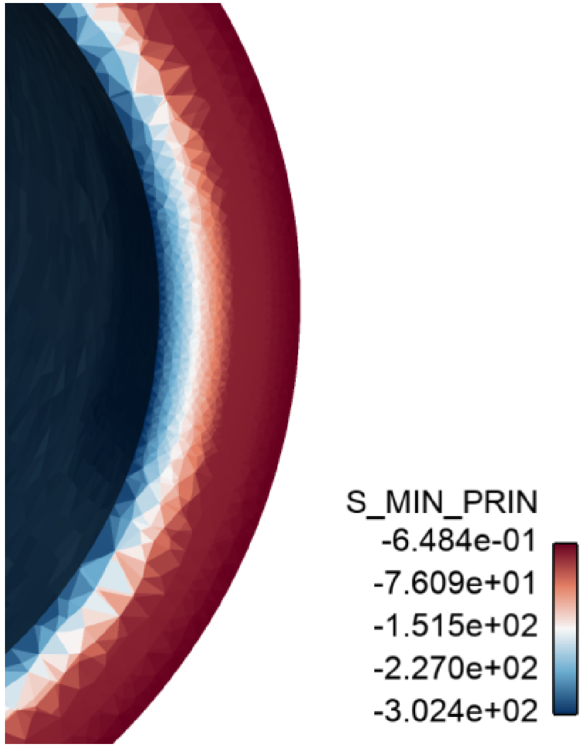
- Liner is SA-372 Grade J Class 70 stainless steel
- Sandia performed tensile characterization and determined plastic anisotropy is negligible
- Overwrap is uni-directional (hoop) carbon-fiber and assumed elastic, and well bonded

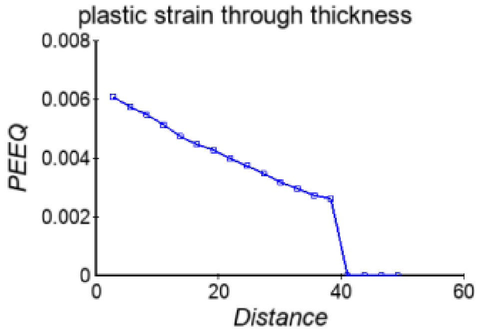
BCs simulate overburden pressure and removal

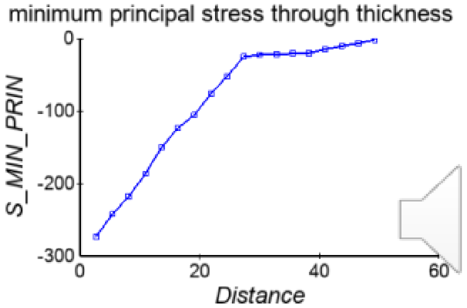
- Overburden pressure monitored by hoop strain on outer fiber (which imitates the manufacturing process)

	Modulus (GPa)	s_{yield} (MPa)	s_u (MPa)	e_u (%)
Steel	206	762	995	27
CFRP	68			







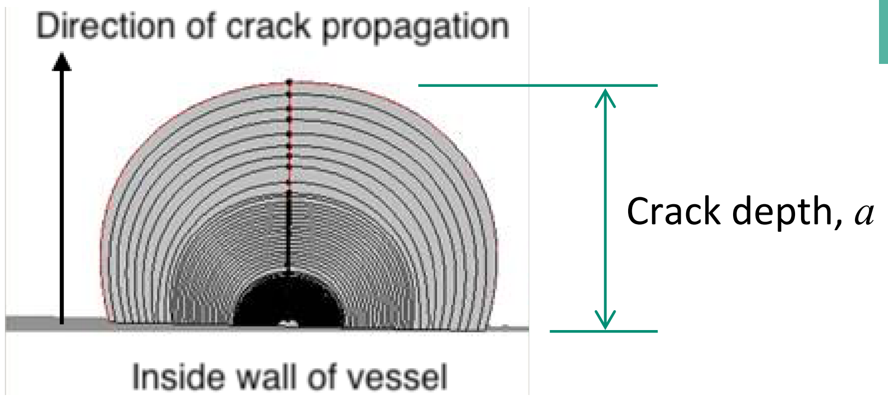


Crack growth simulation

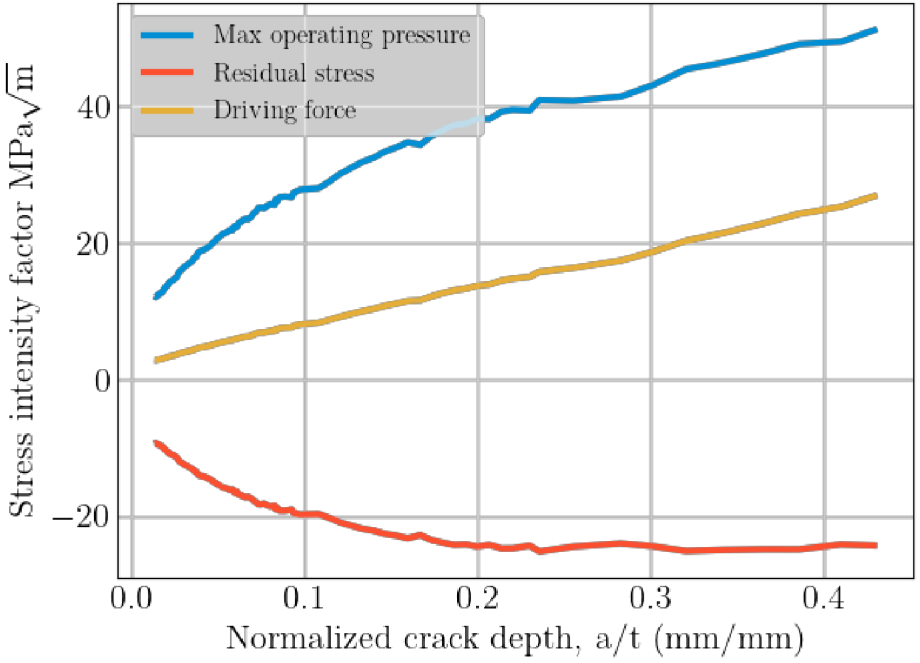
Crack growth performed with FRANC3D

- Pre-process mesh to add crack
- Post-processes Abaqus FEA results to compute the driving force and geometry of crack advance
- Templated crack front very accurately calculates the stress intensity factors
- Accommodates arbitrary 3D crack shapes
- Maximum tensile stress theory for crack kinking
- Autofrettage residual stresses are superposed

$$K_I(P, a) = K_I^P(P, a) + K_{I,auto}(a)$$



Crack front evolution for longitudinal crack



Predicted crack front driving force versus normalized crack depth at max operating pressure 93 MPa

Fatigue estimates

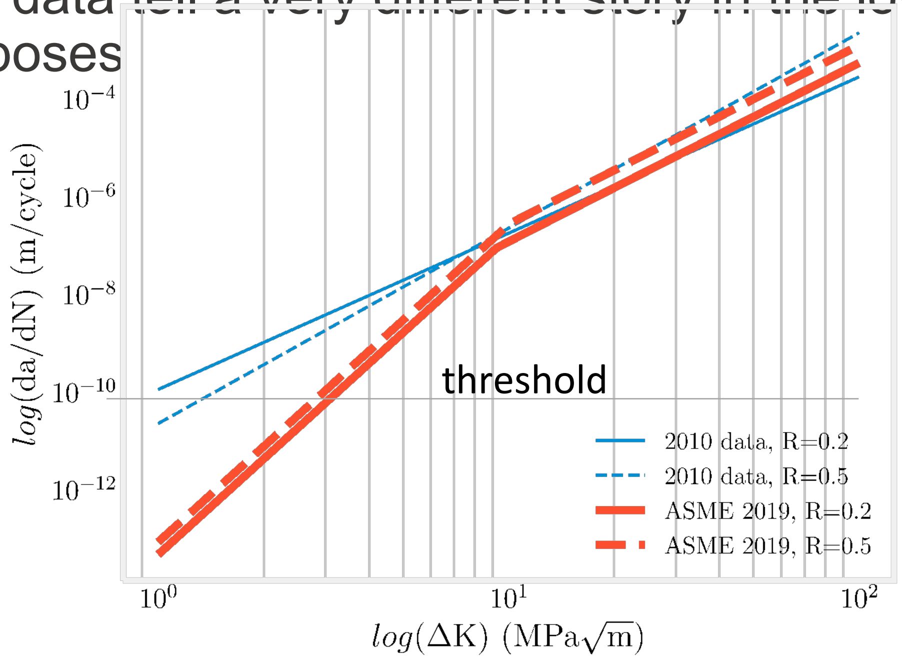
We use the previously established relationship between K and a to rapidly integrate the data

$$\frac{da}{dN} = f\left(\Delta \mathrm{K}, R\right) \longrightarrow N = \int_{a_i}^{a_f} \frac{da}{f\left(\Delta \mathrm{K}, R\right)}$$

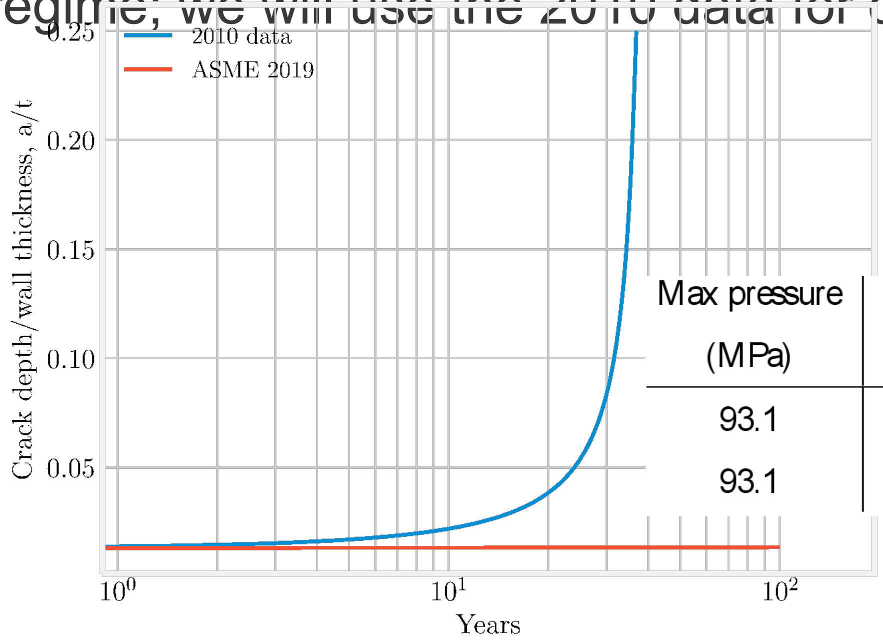
(assume 5,040 max pressure cycles/yr + 2 full blowdowns)

Data available for crack growth rate in hydrogen including bi-linear form in ASME code case 2938

The data tell a very different story in the lower ΔK regime; we will use the 2010 data for our purposes



Crack growth rate curves

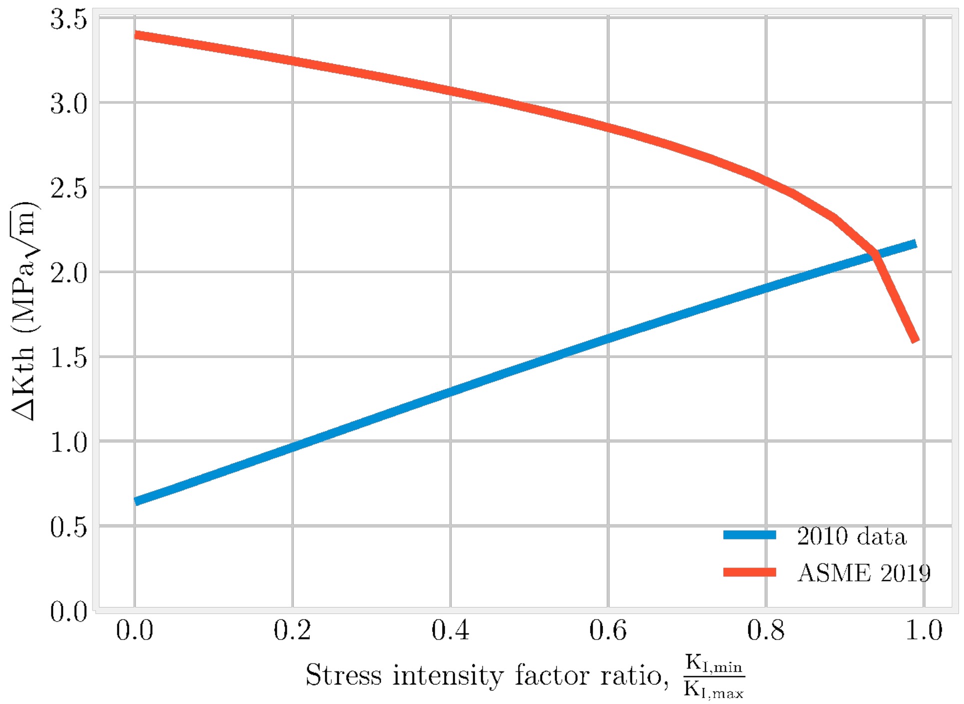


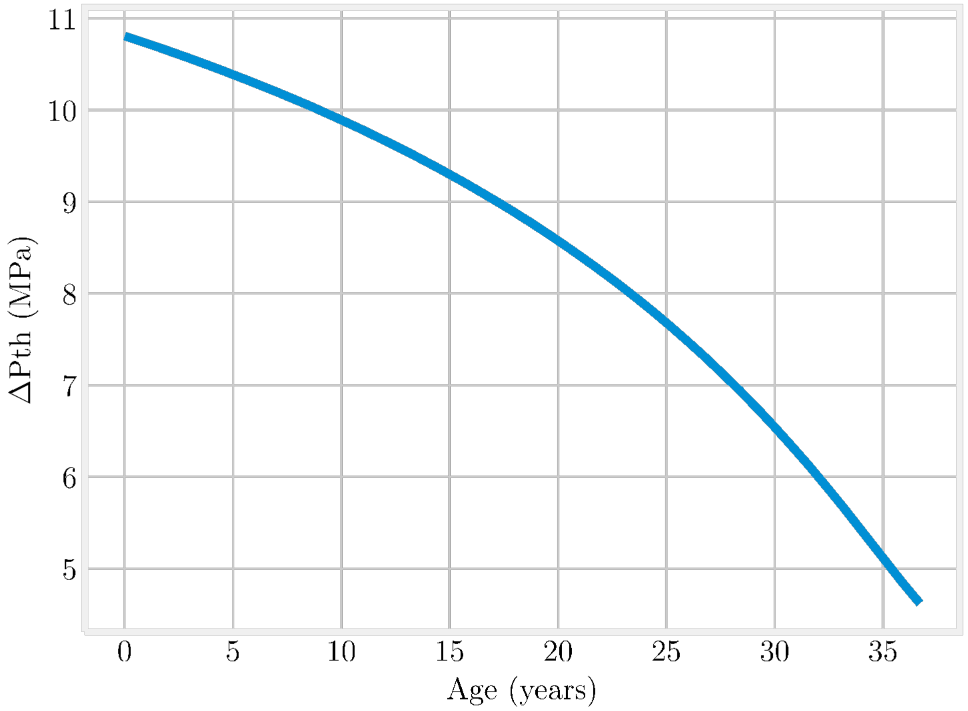
Max pressure (MPa)	DP (MPa)	Load ratio $R_P = \frac{P_{min}}{P_{max}}$
93.1	31.7	0.66
93.1	93.1	0.

Normalized crack depth versus service years, max UDS ΔP = 32 MPa, R = 0.66

Threshold pressure

ΔPth is established using a threshold crack growth rate of 0.1 nm/cycle, and the curves relating crack length to driving force and fatigue cycle (age in years).





Note:

$$DK = K_I^P(P_{max}, a) + K_{I,auto}(a) - (K_I^P(P_{min}, a) + K_{I,auto}(a))$$

$$= K_I^P(P_{max}, a) - K_I^P(P_{min}, a)$$

$$R_K = \frac{K_I^P(P_{min}, a) + K_{I,auto}(a)}{K_I^P(P_{max}, a) + K_{I,auto}(a)} \approx \frac{K_I^P(P_{min}, a)}{K_I^P(P_{max}, a)}$$

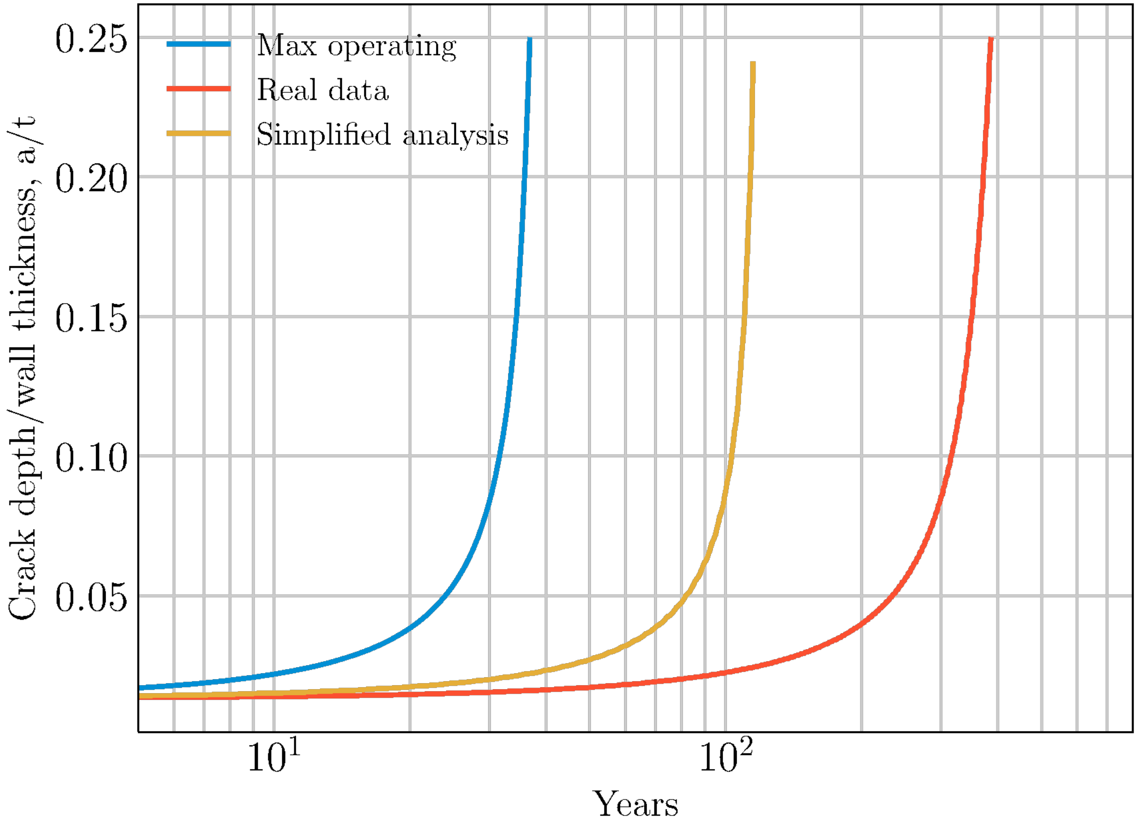
Provides guidance for operators about pressure cycles that can be ignored throughout the life of the vessel

Simplified fatigue analysis

Objective: an algorithm that can be used in the field to reduce over conservatism and account for actual tank usage

Using the relationship for ΔPth, we develop a simplified fatigue analysis that can be used in the field to reduce the over conservatism

```
Data: a, N, Neff
Result: Fatigue life estimate
initialization;
a = 0.5; N=Neff = 0;
while a/t ≤ 0.25 do
    DP      next; N+=1;
    eval DPth(Neff);
    if DP ≥ DPth then
        Neff + = 1;
        a = a(Neff);
    end
    if N == Nfinal then
        STOP
    end
end
```

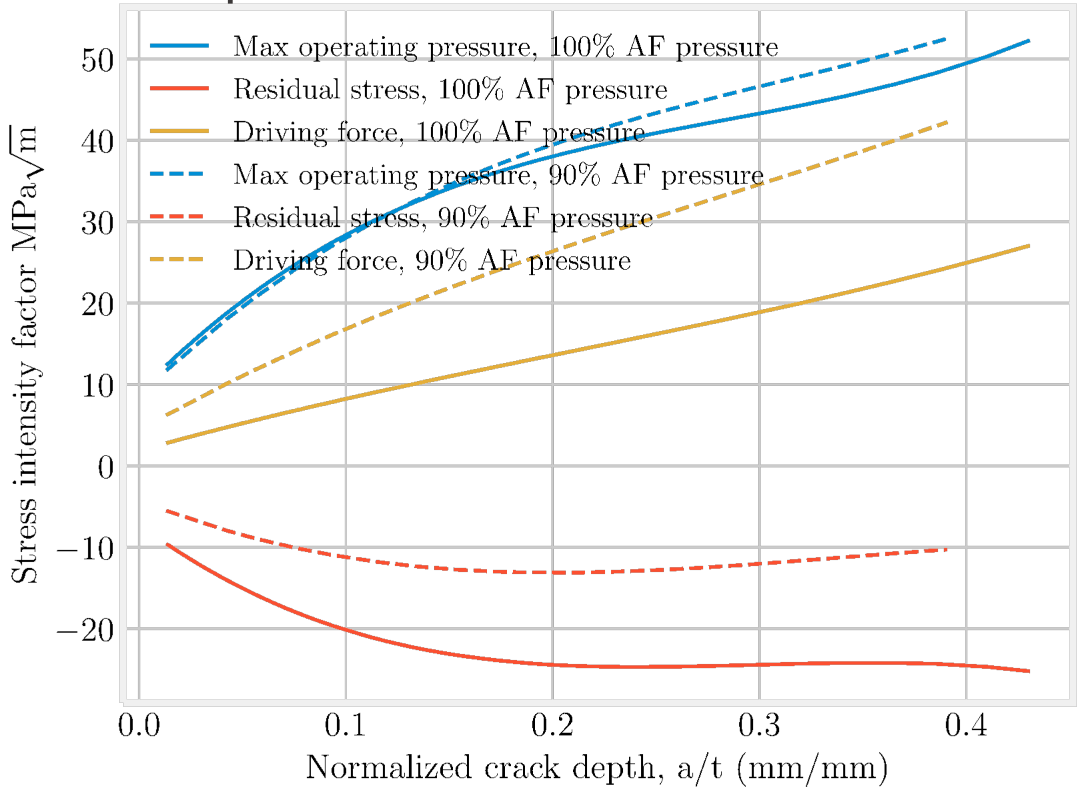


Sensitivity to residual stress

Perform the autofrettage simulation with 90% autofrettage pressure → 65% reduction in compressive residual stress

- Since autofrettage pressure is obtained by monitoring the external strain, this is a bit unrealistic
- Our current effort is to explore sensitivity to liner plasticity (hardening) which strongly influences residual stress

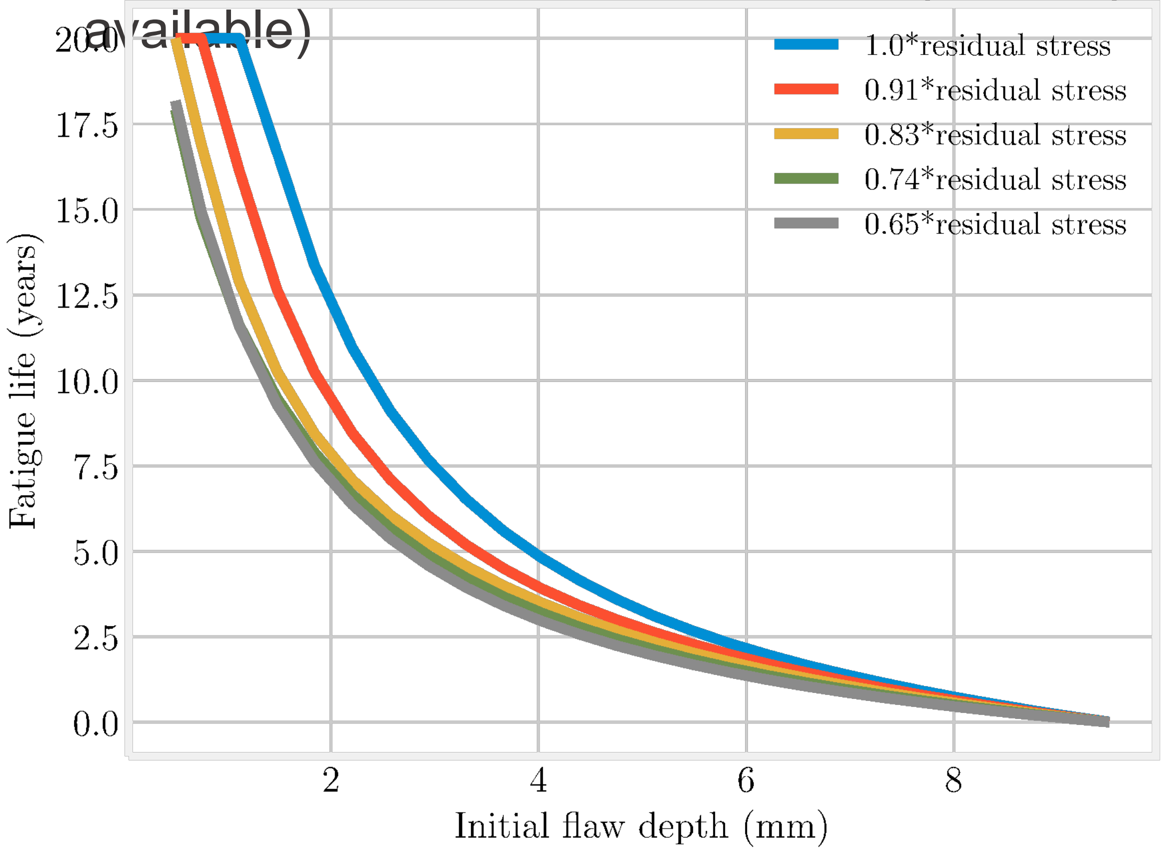
Recompute K versus *a* relationship:



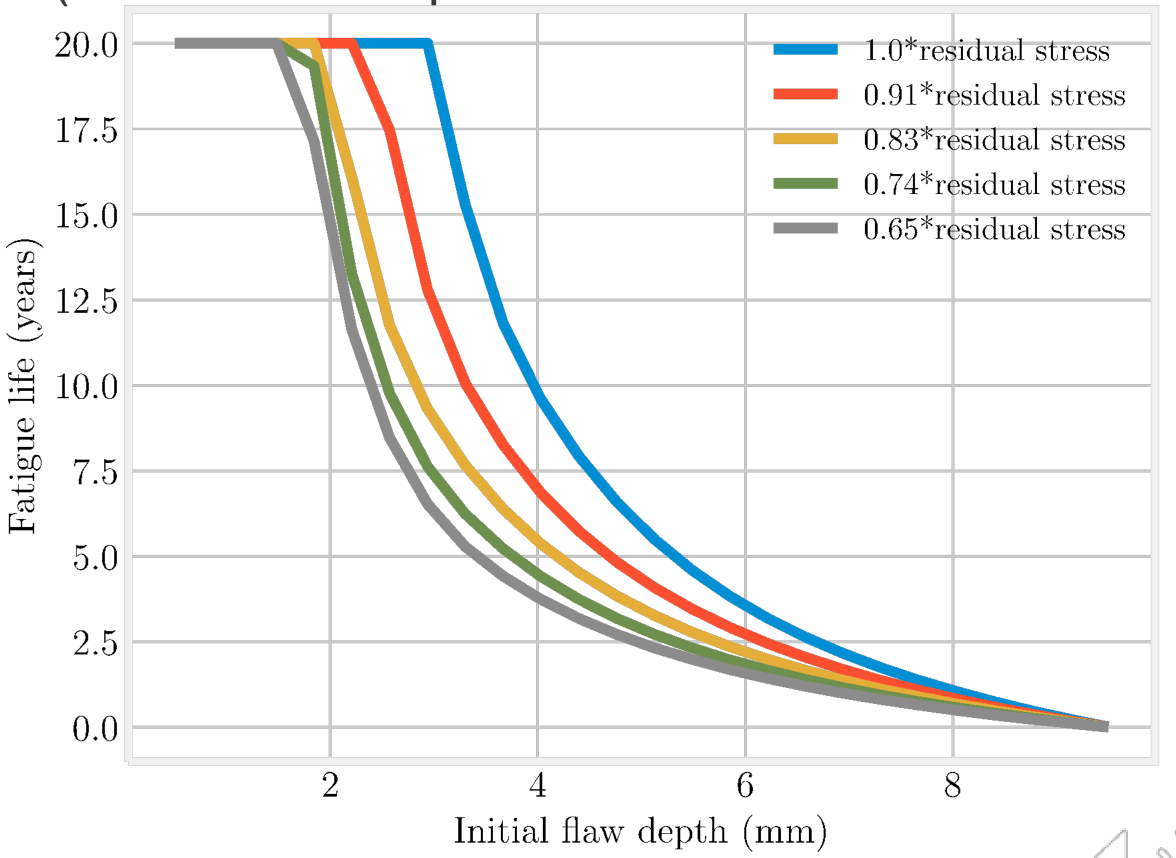
Examine residual life for a range of initial flaw sizes

Given a range of initial flaw sizes and uncertainty in residual stress state, we can explore fatigue life

This can be used to inform NDE inspection period (as NDE techniques become available)



Max UDS w/ 2010 crack growth rates



Max UDS w/ 2019 crack growth rates

Summary

We used computational simulation to explore fatigue life of Type 2 high-pressure hydrogen tanks

We showed that data from the field for two tanks indicates significantly gentler fatigue cycling than the User Design Specification

We explained our simulation approach

We developed a simplified fatigue analysis that accounts for in-service pressure cycles, appears to remain conservative, but also significantly reduces over conservatism

We demonstrated the sensitivity of fatigue life to the achieved residual stress