Gripper Test Assembly Status Report – FY2020

Nuclear Science and Engineering Division
About Argonne National Laboratory

Argonne is a U.S. Department of Energy laboratory managed by UChicago Argonne, LLC under contract DE-AC02-06CH11357. The Laboratory's main facility is outside Chicago, at 9700 South Cass Avenue, Argonne, Illinois 60439. For information about Argonne and its pioneering science and technology programs, see www.anl.gov.

DOCUMENT AVAILABILITY


Reports not in digital format may be purchased by the public from the National Technical Information Service (NTIS):

U.S. Department of Commerce
National Technical Information Service 5301 Shawnee Rd Alexandria, VA 22312
www.ntis.gov
Phone: (800) 553-NTIS (6847) or (703) 605-6000
Fax: (703) 605-6900
Email: orders@ntis.gov

Reports not in digital format are available to DOE and DOE contractors from the Office of Scientific and Technical Information (OSTI):

U.S. Department of Energy
Office of Scientific and Technical Information P.O. Box 62
Oak Ridge, TN 37831-0062
www.osti.gov
Phone: (865) 576-8401
Fax: (865) 576-5728
Email: reports@osti.gov

Disclaimer

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor UChicago Argonne, LLC, nor any of their employees or officers, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of document authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof, Argonne National Laboratory, or UChicago Argonne, LLC.
Gripper Test Assembly Status Report – FY2020

Prepared by:
H. Belch, E. Kent and C. Grandy

Nuclear Engineering Division
Argonne National Laboratory

September 2020
1. Executive Summary

Refueling systems for fast reactors are designed to handle fresh and used core assemblies (fuel, reflector, and shield core assemblies) within the reactor vessel in an opaque coolant environment without visual reference. These refueling machines are designed to work in a sodium (or other fast reactor coolant) and argon vapor space environment and are engineered with the rotatable plug system to allow for the movement of fresh and spent fuel into and out of the reactor core. The refueling machines are a critical component in any reactor and thus need to undergo extensive testing in a prototypic environment to ensure that they will meet all of the system functions and requirements.

Argonne has developed an innovative compact refueling system design for the Advanced Fast Reactor-100 that is based upon some mechanisms used in previous reactor designs, such as the U.K.’s Prototype Fast Reactor (PFR) and some mechanisms that have not been used in sodium. This compact refueling machine supports the reduction in size of the AFR-100’s reactor vessel, and if fully developed, would support and inform the development of the in-vessel refueling machines for such commercial reactors as the GEH PRISM reactor plant, the ARC Clean Energy’s ARC-100 reactor, and the Natrium reactor, among others. This refueling system is a vital component of a fast reactor that supports reducing the cost of the reactor and increasing its reliability.

During the development of the compact fuel handling machine conceptual design for the AFR-100, a lack of testing data for many mechanical components in sodium under typical in-reactor loads and conditions was discovered. The reduction in lifetime of the various mechanical components in the liquid sodium environment needs to be quantified versus the calculated component lifetimes under normal conditions in the testing while they are subjected to typical loading profiles experienced in the past. The Gripper Test Assembly discussed here includes a full-size gripper device with appropriate mechanical features that will be tested in sodium to provide this testing data.

The Gripper Test Assembly is used to test various mechanical fuel handling components submerged in high temperature liquid sodium. These mechanisms are gears, bearings, gripper jaws and head, universal joints and shafts, ball screws, among others. These components will be tested under the typical sodium environmental conditions experienced during refueling operations with appropriate loading conditions that simulate the removal and insertion of core assemblies into a fast reactor grid plate structure. This Gripper Test Assembly is the second in a series of refueling system mechanisms developed for testing in sodium. The first test assembly is the Gear Test Assembly (GTA) which was used to test the performance of gears and bearings operating in sodium. Because of the successful testing conducted with the GTA, it was decided to continue with the development of the Gripper Test Assembly which uses the same gears and bearings tested in GTA.

Using the data collected during operation of this gripper test assembly, lifetime reduction factors of the various mechanical components can be calculated for the material combinations selected. These lifetime reduction factors can be used in the design of future mechanical systems which operate in these environments to accurately predict component end of life.
addition, an understanding of the ability of these components and mechanisms to operate under-sodium with the chosen materials will be accomplished. Again, this Gripper Test Assembly is the follow-on test article to the Gear Test Assembly which was testing the ability of gears and bearings operating in a sodium environment. Once fabricated and qualified, it will be tested in the Mechanism Engineering Test Loop (METL) facility located in Building 308 at Argonne National Laboratory.
# Table of Contents

1. Executive Summary ............................................................................................................... 1
2. Table of Figures ..................................................................................................................... 4
3. Background ............................................................................................................................ 6
4. System Overview ................................................................................................................... 8
5. Operating Parameters and Testing Criteria ........................................................................ 11
6. Gripper Test Assembly Design and Operation .................................................................... 16
7. Gripper Test Assembly Monitoring ..................................................................................... 35
8. Failure Mechanisms in Mechanical Gearing Components .................................................. 37
9. Failure Mechanisms in Roller Bearing Components .......................................................... 38
10. Failure Mechanisms in Universal Joints ............................................................................. 44
11. Failure Mechanisms in Recirculating Ball Screws and Splines ......................................... 46
12. Future Development Plans ................................................................................................. 46
13. References .......................................................................................................................... 48
2. Table of Figures

Figure 1 - Compact Fuel Handling Machine Concept .................................................................................................................. 7
Figure 2 - Gripper Test Assembly .................................................................................................................................................. 10
Figure 3 - Gripper Test Assembly Position in Test Vessel ........................................................................................................... 11
Figure 4 - Gripper Head Assembly .............................................................................................................................................. 17
Figure 5 - Exploded Gripper Assembly ...................................................................................................................................... 17
Figure 6 - Design of the Gripper Post Support Bearings ........................................................................................................... 18
Figure 7 - Gripper Displacements Engaged with Simulated Core Socket at Maximum Load ................................................................. 19
Figure 8 - Maximum Stresses in Gripper Jaw .................................................................................................................................. 20
Figure 9 - Section Through Gripper Head ................................................................................................................................... 21
Figure 10 - Recirculating Ball Spline Nut on Spline Shaft [11] ...................................................................................................... 22
Figure 11 - Gripper Head Drive System ....................................................................................................................................... 22
Figure 12 - Gripper Drive System Components (Gearbox and Guide not shown for clarity) ................................................................................................................................... 24
Figure 13 - Analysis of the Gripper Guide Weld and Gearbox Fasteners ........................................................................................... 25
Figure 14 - Universal Joint Design for Gripper Test Assembly .................................................................................................... 25
Figure 15 - Non-Linear Stress Analysis Results for Universal Joint at Maximum Stall Torque ................................................................................................................................... 26
Figure 16 - Deflection of Main Universal Joint under Maximum Loading Conditions ................................................................. 26
Figure 17 - Large Lantern Assembly Modified Design Concept ................................................................................................... 27
Figure 18 - Heat Generation by Compressed Graphite Rope ........................................................................................................... 28
Figure 19 - Heat Generation by the Lip Seal on the Rotating Shaft ................................................................................................. 29
Figure 20 - Thermal Analysis with Seal Heat Generation ........................................................................................................... 30
Figure 21 - Surface Temperature of Insulation and Modified Lantern Seals ...................................................................................... 30
Figure 22 - External Drive Motor Foundations, Offset Gear Boxes and Shaft Couplings ............................................................ 31
Figure 23 - Non-Linear Elastic Static Analysis of Main Motor Mount Foundation .............................................................................. 32
Figure 24 - Resisting Force Component Assembly .......................................................................................................................... 33
Figure 25 - Helical Rack Gear Drive Reaction Non-Linear Bolt Pre-Load Analysis ........................................................................... 34
Figure 26 - Helical Rack Gear Drive Combined Non-Linear Bolt Pre-Load and Gear Load Analysis ................................................................................................................................... 34
Figure 27 - Data Acquisition and Supervisory Control System ...................................................................................................... 36
Figure 28 - Preliminary Programming Flow Chart for Control System Operation ........................................................................... 37
Figure 29 – Image of Abrasive Wear – see page 10 of [9] ......................................................................................................................... 39
Figure 30 – Image of Severe Spalling of Material- see page 4 of [9] .................................................................................................. 39
Figure 31 – Image of Overheating – see page 5 of [9] ........................................................................................................................... 40
Figure 32 – Image of Fatigue Spalling – see page 8 of [9] ..................................................................................................................... 41
Figure 33 – Image of Brinelling – see page 7 of [9] ............................................................................................................................ 41
Figure 34 – Image of Effects of Misalignment – see page 13 of [9] ........................................ 42
Figure 35 - Image of Poor Fit – see page 15 of [9] .................................................................. 43
Figure 36 – Image of Poor Fit – see page 14 of [9] .................................................................. 43
Figure 37 – Image of Incorrect bearing installation – see page 10 of [9] ................................. 44
Figure 38 – Image of a U-Joint with Yoke fractures [10] .............................................................. 45
Figure 39 - Typical Internal Return Recirculating Ball Nut and Ball Screw [11] ................. 46
3. Background

A compact fuel handling machine conceptual design was developed during the Advanced Liquid Metal Reactor program, the Small Modular Fast Reactor project, the Advanced Burner Test Reactor project, and the Advanced Fast Reactor-100 concept developed under the Advanced Reactor Concepts program. The AFR-100 embodies and integrates a number of the advancements that the U.S. DOE Office of Nuclear Energy is developing to show how these advanced technologies could be integrated into an overall design. The design of the fuel handling machine matured further as it was considered as an option for the KAERI (Korean Atomic Energy Research Institute) PGSFR project. Several options for a compact fuel handling system were developed and the concept was fully developed for testing using the Gripper Test Assembly – which will test the functionality of the gripper assembly and some of the linkages.

The compact fuel handling machine is designed to extend into a narrow slot in the upper internal structure during reactor refueling and retract into a structural support tube mounted to a single rotating plug in the reactor head.
Figure 1 - Compact Fuel Handling Machine Concept
The compact fuel handling machine performs the following functions during operation;

1. Raise or lower a structural component that is used to *hold down* the surrounding core assemblies during a core assembly removal

2. *Rotate the fuel handling machine* with respect to the rotatable plug

3. *Extend or retract the Fuel handling arm* which radially positions the gripper over the core while holding or not holding a core assembly

4. *Rotate the gripper* to correctly orient the core assembly to the reactor core before insertion of the core assembly

5. *Raise or lower the gripper head* to insert or remove a core assembly into the core and grid plate structure or remove a core assembly from the core and break it free of the grid plate structure

6. *Extend or retract the gripper jaws* to engage or disengage a core assembly

4. System Overview

The Gripper Test Assembly (GrTA) is designed to simulate the gripper operation of a compact fuel handling machine under expected mechanical and thermal loading at the environmental conditions inside a sodium fast reactor. The typical mechanical components used in the design of this compact fuel handling machine are radial and thrust ball bearings, gears, ball screws, recirculating ball nuts, ball spline shafts, recirculating ball splines, universal joints, radial cylindrical bearings, tapered bearings, fasteners, welds, various connection pin designs and rolling contact joints.

The gripper test assembly is designed to operate by isolating the various control motions;

1. extension or retraction of the gripper jaws

2. rotation of the gripper post

3. raising or lowering the gripper head

These control motions are isolated to ensure that the operation of one does not affect the others. For example, the rotation of the gripper post does not cause the jaws to extend or retract.

The system is fully monitored during operation using various sensors located throughout the system. Several computers collect monitoring data and process the data in real time to adjust control parameters, notify operators of off normal conditions or stop the loading should an overload or failure be detected.

Drive shafts penetrate through the top of the METL tank flange and transfer the required torque to operate the mechanisms from DC servo motors located outside the test vessel. The
shafts use various sealing systems to prevent leakage of the argon cover gas into the experimental hall.

The gripper test assembly is computer controlled with automatic operation and data acquisition. Automatic fault identification and system shutdown will be programmed into the operating system and with remote notification of the operator.

The design requirements of the gripper test assembly have been developed as follows;

1. Simulate operation of FHM Gripper Assembly in simulated SFR environment
2. Reduce width of Gripper Assembly to fit a 10” (254 mm) wide slot in an upper internal structure of an SFR.
3. Components will be operating in liquid metals at elevated (350C) temperature (with a maximum temperature of 550C).
4. Material combination compatibility to prevent galling (self-welding), galvanic corrosion and minimize impact of surface alloying element reduction through diffusion processes.
5. Raise and lower simulated core assembly (6,000 lbs. {29,690 N} max load with margin).
6. Rotate core assembly while fully raised while supporting weight of core assembly (1,000 lbs. {4,948 N} max load with margin).
7. Open and close gripper jaws while lowered to engage/disengage simulated core assembly (No Load).
8. Isolate individual gripper actuation motions so operation of one does not affect others.
9. Use simple mechanisms to reduce number of components.
10. Minimize dynamic loading on graphite seals to extend life.
11. Minimize the number and types of seals required for operation.
12. Test all prototypic types of mechanical components used in the FHM design
Figure 2 - Gripper Test Assembly
5. Operating Parameters and Testing Criteria

The Gripper Test Assembly is designed to operate in a high temperature liquid sodium environment, the assembly is mounted on the top flange (vessel cover) of a 28-inch diameter stainless steel vessel. The test vessel is part of the Materials Engineering Testing Loop (METL) located in building 308 at Argonne National Laboratory. The design test temperature is in the range of 250 to 550°C, but most fast reactor refueling operations are around 350°C.

Figure 3 - Gripper Test Assembly Position in Test Vessel
The unit is designed with a programmable resisting force from 0 to 27,000 Newtons to simulate the transient loading cycle representing fuel handling loads for installation or removal of a stuck core assembly inside a liquid sodium cooled reactor. The data for handling loads was obtained from the Fast Flux Test Facility (FFTF) reactor [1] where the maximum load measured was (5,000 lbf) 22,241 Newtons, but 20% was added for unknown factors for a maximum load of (6,000 lbf) 27,000 Newtons.

The high temperature liquid sodium environment prohibits the use of normal lubricants inside the vessel. Therefore, the choice of material combinations for rolling or sliding mechanical components is critical.

Material combinations must be chosen with a minimal galling potential as well as minimizing the galvanic corrosion potential and reactions of alloy constituents with the sodium in the high temperature liquid environment. The mechanical components will utilize liquid sodium as the primary “lubricant”.

All mechanical components have been designed and analyzed in accordance with the appropriate commercial specifications and include lifetime modification factors in accordance with those standards. Lifetime reduction factors for operation in a high temperature liquid sodium environment will be determined through the experimental results with respect to operation in a normally lubricated environment.

The full-size fuel handling machine was designed to raise fully a core assembly in approximately 60 seconds with the ball screw turning at 350 rpm having a lead of 12 millimeters. This was to accommodate the full 4.2-meter length of the core assembly under consideration. The load cycle for the design of the fuel handling machine was 10% at maximum load and 90% at normal load and all component lifetimes were calculated to accommodate the expected lifetime of the reactor (although all mechanical components may be replaced at normal maintenance intervals). Options to remove the fuel handling machine during reactor normal operation or leave it in were considered.

The maximum vertical length of travel for the gripper test assembly allowed by the METL 28-inch diameter test vessel is 0.575 meters. Therefore, at 350 rpm, the travel time is reduced to approximately eight seconds. The loading cycle times will also be modified for 25% at maximum load and 75% at normal load.

The loading profile for the gripper test assembly will initially consist of a twenty-seven thousand newton (simulating a stuck core assembly) resisting load for a period of two seconds followed by a reduction to four thousand five hundred newtons (simulating the weight of the heaviest core assembly) for the remaining six seconds. As the loading profile is programmable, any transient loading scheme may be used (for instance, an intermediate initial load followed by a simulated “sticking” peak load and then reduction to normal core assembly load).
6. Mechanical Component Design and Analysis

The mechanical components are analyzed for their useful lifetimes using available commercial standards and specifications or finite element analysis and fatigue analysis.

Gear lifetimes are calculated according to the requirements of ANSI/AGMA 2001-D04:2005 [2]. The overall equations are shown below for reference.

\[
S_{H12} := \frac{s_{ac}^{Z_{n12}} C_{H12}}{C_p K_T K_R \sqrt{F_t K_o K_v K_{s12} K_{m12} C_{fl2} d_{w1} b_w}}
\]

\[
S_{F12} := \frac{s_{at12}^{Y_{N12} Y_{A12}}}{K_T K_R F_t K_o K_v K_{s12}^{P} K_{m12}^{K_{B12}} b_{wF12}}
\]

The variables used in the AGMA safety factor equations include;

- \( s_{ac} \) – Allowable contact stress number
- \( s_{at12} \) – Allowable bending stress number
- \( F_t \) – Nominal tangential force acting on teeth
- \( d_{w1} \) – Operating pitch diameter of gear
- \( b_w \) – Operating tooth width
- \( P \) – Normal pitch
- \( b_{wF12} \) – Operating tooth width
- \( I \) – Geometry factors for pitting resistance
- \( Z_{n12} \) – Stress cycle factor for pitting resistance
- \( C_{H12} \) – Hardness ratio factor
- \( K_O \) – Overload factor
- \( K_V \) – Dynamic Factor
The lifetime for the roller bearings (in millions of revolutions) has been calculated based upon the ANSI/ABMA Std. 11-1990 (American Bearing Manufacturers Association) [3] standard for tapered roller bearings and cylindrical pin roller bearings. As an example, the cylindrical roller bearings are analyzed using the following equations to determine the radial load ratings for the particular bearing being used.

\[
C_{or} := 44 \left(1 - \frac{D_{we} \cos(\alpha)}{D_{pw}}\right) i Z L_{we} D_{we} \cos(\alpha)
\]

\[
C_{r} := f_{cm} \left(i L_{we} \cos(\alpha)\right)^{\frac{7}{9}} Z^{3/2} \left(D_{we}\right)^{29}
\]

Where:
- \(D_{we}\) – Diameter of the rolling element
- \(\alpha\) – Contact angle of the Rolling Element
- \(D_{pw}\) – Pitch Diameter of Roller Set
- \(i\) – Number of Rows of Rollers
- \(Z\) – Number of Rolling Elements
- \(L_{we}\) – Length of rolling elements
- \(f_{cm}\) – Radial Bearing Factor (Interpolated from Table 1)
Similarly, the axial load ratings are calculated for the tapered thrust bearing selected using the following equations.

\[
C_a = f_{cm} \left[ \frac{1}{w_x} \left( \frac{1}{mm} \right) \cdot \cos(\alpha) \right]^{\frac{7}{2}} \cdot Z^{\frac{3}{2}} \cdot D_{we} \left( \frac{1}{mm} \right) \cdot \frac{28}{27} \cdot \tan(\alpha)
\]

Basic Static Axial Load Rating for Thrust Bearings

\[
C_{oa} = 220 \left( \frac{N}{mm^2} \right) \left( 1 - \frac{D_{we} \cos(\alpha)}{D_{pw}} \right) \cdot Z \cdot D_{we} \cdot D_{we} \cdot \sin(\alpha)
\]

The lifetime of the bearings is calculated based upon the axial and radial loadings for the tapered roller bearing.

\[
\text{Dynamic Radial Load Factor for Radial Roller Bearings} \quad X
\]

(From Table 2)

\[
\text{Dynamic Axial Load Factor for Radial Roller Bearings} \quad Y
\]

\[
P_r = X \cdot F_r + Y \cdot F_a
\]

\[
\text{The unadjusted bearing life:} \quad l_{10} = \left( \frac{C_r}{P_r} \right)^3
\]

Where;

- \( F_r \) – Maximum Static Radial Load
- \( F_a \) – Maximum Static Axial Load

The lifetime for the ball bearings (in millions of revolutions) has been calculated based upon the ANSI/ABMA Std. 9-1990 (American Bearing Manufacturers Association) [4] standard for ball bearings. As an example, the ball bearings are analyzed using the following equations to determine the load ratings for the particular bearing being used.

For radial ball bearings, the dynamic load rating is calculated using the following equation;

\[
C_{Dr} = f_{cm} (\cos B)^{0.7} Z^{2.5} d^{1.8}
\]
For thrust ball bearings with the ball diameter less than 25.4 mm and $\beta$ not equal to 90 degrees, the dynamic load rating is calculated using the following equation:

$$C_{D_{a}} = f_{cm}(i \cos \beta)^{0.7}(\tan \beta)^{2/3}d^{1.8}$$

Where:

- $f_{cm}$ - Radial Bearing Factor (Interpolated from Table 1)
- $i$ - Number of rows of rollers
- $Z$ - Number of rolling elements
- $\beta$ - Contact angle of rollers
- $d$ - Diameter of rolling ball

The lifetime of the ball bearing is then calculated using the following equation:

$$L_{10} = \left(\frac{C_D}{P_{eq}}\right)^{3}$$

Where:

- $P_{eq}$ - Radial Equivalent Load (Calculated Using X and Y from Table II)
- $C_D$ - Dynamic Load Capacity for the bearing type being analyzed

### 7. Gripper Test Assembly Design and Operation

Starting the design from the gripper head using it as the primary design point of load application, the design maximum load 27kN (normal load of 4.5kN) is transferred to the gripper jaws from the simulated core assembly socket. As the loading of the gripper test assembly varies over time (stuck to free core assembly), a load cycle using 25% at maximum load and 75% at normal load has been chosen as a basis for the analysis of the mechanical component lifetimes. Maximum load is considered for static load transfers in bearings and component peak stress calculations.
Figure 4 - Gripper Head Assembly

Figure 5 - Exploded Gripper Assembly
The gripper post is supported in the gripper head using a set of thrust ball bearings which resist a pulling or pushing (as well as radial) load applied by the simulated core assembly. The maximum load of 27kN will only be applied while the gripper head is stationary (no rotation). The load of 4.5kN will be applied while the gripper head rotates (to simulate rotation while supporting the heaviest core assembly weight).

![Design of the Gripper Post Support Bearings](image)

**Figure 6 - Design of the Gripper Post Support Bearings**

The non-linear stress analysis for the gripper jaws was performed using ANSYS with frictional contacts between pin supports and gripper jaws. The support pins will be welded into the gripper post after assembly.

The allowable stress of 239 MPa was obtained from the ASME Boiler and Pressure Vessel Code Section III Division 5 for High Temperature Reactors [5] for Inconel 718 bolting materials at a temperature of 250°C.

The gripper jaws were also analyzed in accordance with the requirements of CMAA (Crane Manufacturers Association of America) Specification 70 for Electric Over Head Travelling Cranes [6] which addresses the design of hooks (although it does not refer to lifting jaws) as follows:
“The hook rated load stress shall be calculated considering the rated load on the hook using:

- Straight Beam Theory with the calculated combined stresses not to exceed 20% of the material’s average ultimate strength.

-OR-

- Modified curved beam theory with the calculated combined stresses not to exceed 33% of the material’s average ultimate strength.

-OR-

- Plastic theory or testing with the combined stresses not to exceed 20% of the stress produced by the straightening load as obtained by test or calculation by this theory.”

The requirements were also compared to the requirements of the following standard:

NUREG-0554 (Single Failure Proof Cranes for Nuclear Power Plants - 1979) [7]:

“Lifting Devices shall be designed to support a load three times the load (static and dynamic) being handled without permanent deformation”

Figure 7 - Gripper Displacements Engaged with Simulated Core Socket at Maximum Load
Figure 8 - Maximum Stresses in Gripper Jaw

The gripper post and supporting bearings are secured in the gripper head machined recess using a cover plate which is secured to the gripper head using a pattern of fasteners which are torqued to their standard value and tack welded in place to prevent loosening during operation.
Above the gripper head are several plates which restrain the mechanisms that control the gripper orientation and gripper jaw actuation. Those mechanisms are driven using gears supported by roller bearings and a pair of recirculating ball spline nuts and spline shafts. The use of these spline shafts allows free translation of the gripper head inside the gripper guide without changing the gripper rotation (orientation) or position of the gripper jaws (retraction or extension).

The recirculating ball spline nuts are secured in retaining assemblies between the gripper head and a) the gripper rotation cover plate or b) the gripper jaw actuation cover. Gears attached to the top of these assemblies transfer the rotational motion from the spline shafts to the gripper rotation and gripper jaw actuator gears.
Figure 10 - Recirculating Ball Spline Nut on Spline Shaft [11]

Figure 11 - Gripper Head Drive System
The gear which drives the rotation of the gripper post is secured to the top of the gripper post using a standard shaft collar and pins for alignment. The collar is welded to the gripper post to assure it cannot loosen during operation.

The gear which controls the position of the gripper jaws is attached to a recirculating ball nut. The ball nut is secured inside the gear using a key to prevent relative rotation and has bolted retaining plates on the top and bottom of the gear. The ball nut engages a ball screw which moves the gripper actuator up and down to extend or retract the gripper jaws as the gear is turned. The gripper actuator is connected to the gripper actuator ball screw using a pair of bearings which allow rotation of the gripper head without affecting the jaw position. A pin through the top of the ball screw shaft fixes the orientation of the ball screw with respect to the gripper head so the turning of the ball nut raises or lowers the ball screw. The gripper jaws engage the gripper actuator by providing support at the finger height which reduces the bending stresses and deflections of the jaws as the fuel handling loads are applied.

The gripper head is moved inside the gripper guide using a ball screw and recirculating ball nut. The gripper head contacts the gripper guide using cylindrical roller bearings to reduce the calculated resisting frictional forces between the components. The cylindrical roller bearings are supported using pins between the gripper head and gripper head side plates. Larger bearings were selected in the primary direction of loading with smaller bearings in the perpendicular direction.
The gripper guide is welded to the underside of the gripper gearbox top plate. The primary drive ball screw and spline shafts are secured to gears in the gearboxes and by bearings in the ball screw and ball spline supports at the opposite end. The driving gears for the gripper are supported by bearings in the gear box assemblies.

The welds of the gripper guide and bolt pattern for the gearbox assembly were simultaneously analyzed using non-linear material properties in a multi-step analysis in ANSYS and included the calculated compressive bolt preloads based upon the standard fastener torques.
The gear trains end at the universal joint input shafts for the drive systems. The universal joints provide an offset to the external drive shafts (which penetrate the vessel cover) to allow sufficient spacing for the lantern seal assemblies located on the top of the vessel cover. The universal joints are set at an angle of 15 degrees, which is the maximum angle that the universal joints in the AFR-100 reactor fuel handling machine operate.

Figure 13 - Analysis of the Gripper Guide Weld and Gearbox Fasteners

Figure 14 - Universal Joint Design for Gripper Test Assembly
Figure 15 - Non-Linear Stress Analysis Results for Universal Joint at Maximum Stall Torque

Figure 16 - Deflection of Main Universal Joint under Maximum Loading Conditions
The lantern seal assemblies are a modified design of those successfully used in the Gear Test Assembly shaft seals in FY20. The design was modified by adding a threaded feature between the upper and lower chambers (in lieu of bolted flanges) to simplify the adjustment procedure and apply uniform loading as the graphite rope seal wears. Lip seals were installed at the top of the upper chamber to retain the inert gas during system operation.

Computation fluid dynamics analyses were performed on the insulated tank flange and modified lantern seal assemblies to verify the temperature distribution in the areas of the elastomer (Viton) lip seals to assure they were within the vendor specified operating temperature ranges (below 200°C).

The heat generation from the compressed graphite rope and lip seals was calculated and included to provide more accurate thermal prediction.

To verify the validity of the computational fluid dynamics analyses, a convection coefficient was calculated by hand for a heated vertical surface cooled by natural convection by air at 305K (32°C). The hand calculated value was compared to the surface heat transfer coefficient calculated by CFX. Comparison of the CFD wall heat transfer coefficient result on the external surface of the modified lantern seal geometry agreed very well with the hand calculated value.
Graphite Rope Frictional Heat Generated by Rotating Shaft

The purpose of this calculation is to estimate the heat generated in a graphite rope seal due to a rotating shaft. The shaft is 45 mm in outside diameter and a standard seal is chosen to seal the gas (argon) inside the vessel seal chamber. The argon in the chamber is pressurized to less than 5 psi over atmosphere. The shaft rotates at 350 RPM and a single graphite rope seal is evaluated.

Standard Torque 316 1/2" Bolt: \( T_b = 45 \text{ ft.lbf} \)

Diameter of Bolt: \( d_b = 0.3 \text{ in} \)

Compressive force @ STD Torque: \( F_t = \frac{T_b}{0.208 \ d_b} \)

\( F_t = 5.192 \times 10^3 \text{ lbf} \)

Friction Coefficient: \( f = 0.1 \)

Shaft Diameter: \( d = 1.77 \text{ in} \)

Seal Contact Width: \( b = 0.50 \text{ in} \)

Seal Radial Load: \( l_s = 75 \text{ lbf} \) Estimated from FEA

Contact Pressure: \( p = \frac{l_s}{b \cdot (\pi \cdot d)} \)

\( p = 26.975 \text{ psi} \)

Circumferential Shaft Velocity: \( v = 350 \frac{\text{rev}}{\text{min}} \)

Seal Friction Power: \( P = f \cdot p \cdot v \cdot \pi \cdot d \cdot b \)

\( P = 274 \text{ lbf} \cdot \text{ft} / \text{s} \)

Heat Generated by Single Rope Seat: \( h_g = P \cdot \pi \cdot d \)

\( h_g = 172.704 \text{ W} \)

Force Applied: \( F_a = 75 \text{ lbf} \)

Torque Needed @ 75 lbf Compression: \( T_a = F_a \cdot d_b \cdot 0.208 \)

\( T_a = 0.65 \text{ ft.lbf} \)

Figure 18 - Heat Generation by Compressed Graphite Rope
Clean argon gas is supplied to the lantern seal chambers at a pressure slightly over the internal pressure of the vessel. This assures that any leaks in the sealing system are of argon gas into the test vessel. The flow rate and pressure of the argon gas inside the lantern seals is monitored by the computer control system.
Figure 20 - Thermal Analysis with Seal Heat Generation

Figure 21 - Surface Temperature of Insulation and Modified Lantern Seals
The drive shafts from the inside of the test vessel are connected to the external gearbox shafts in the external gearboxes. The external gearboxes are mounted on top of weldments (foundations) which are bolted to the top of the vessel cover (tank flange). The driveshafts from the external gearboxes are connected to the gripper test assembly drive shafts using stainless steel shaft couplings (above the vessel cover) to eliminate the need for laser shaft alignment and reduce transferred vibration from minor angular and offset shaft misalignments.

![Image](image.png)

*Figure 22 - External Drive Motor Foundations, Offset Gear Boxes and Shaft Couplings*

The welded motor mount foundations were analyzed for maximum torque conditions to assure deflections and stresses were within acceptable ranges. Modal and Buckling analyses were also performed to assure rigidity and stiffness of the foundations were acceptable.
The external gearboxes provide additional offset clearance required by the drive motors and planetary reduction units as direct drive configurations caused interference between the drive motors.

The primary and secondary drives for the gripper test assembly are provided by Parker DC Servo motors (MPP142C8D-KPSBV & MPP1152C8D-KPSBV respectively) with Stober 4 to 1 planetary reduction gear assemblies (PS142-004-L2 & PS115-004-L2 respectively) the motors and reduction gear units are mounted to external gearboxes. The motors are supplied with automatic braking features and absolute position encoders.

The simulated core assembly handling socket which provides the resisting force to the gripper jaws and gripper post is attached to the external drive system by a force transfer assembly using a set of internal bearings which allow free rotation of the gripper head while under expected loading conditions. The force application shaft is sealed to the top of the vessel cover using a welded stainless-steel bellows which has a 0.6-meter travel length. The top of the bellows is attached and sealed to the bottom of the force transfer assembly. The force transfer assembly is attached to a load cell which provides a reference load signal to the control system. The load cell is connected to an Atlantic Gears helical rack gear and linear slide rail which transfers the applied resisting load.
During operation of the gripper test assembly, the internal volume of the bellows changes during the linear stroke motion. As the bellows extends, to prevent ingress of sodium vapor from the cover gas inside the vessel, a supply of clean inert gas is supplied at varying flow rates (based upon the direction and speed of travel). As the bellows contracts, the argon gas will be expelled into the test vessel and out the argon exhaust port.
The resisting force load is applied by a Stober helical rack planetary reduction gear unit (ZTRSPH822ME) powered by a Parker DC Servo Motor (MPP1428QBD-KPSBV).

The Stober helical rack planetary reduction gear unit is enclosed in a rack support weldment. The helical rack gear and linear slide rail are supported by a set of three linear guide carriages attached to a removable cover. The purpose of the linear guide carriages is to minimize moment (bending) loads transferred from the helical gear unit to the shaft which supports the simulated core assembly. This will help increase the lifetime of the graphite rope seals.
The rack support weldment is secured to the top of an external welded foundation which has maintenance access openings. The external welded foundation is secured to the vessel cover using bolted and pinned connections for alignment.

### 8. Gripper Test Assembly Monitoring

Monitoring data from the Gripper Test Assembly includes (but is not limited to);

1. Motor torque
2. Component temperatures
3. Inert gas pressure in test vessel
4. Liquid coolant level in the test vessel
5. Coolant temperature (in vessel)
6. Gearbox and bearing vibration
7. Safety interlock engagement
8. Inert gas flow rate to each of the (3) pressurized inert gas lantern seals
9. Inert gas flow rate to the inert gas pressurized bellows
10. Pressure of the inert gas in each of the (3) pressurized inert gas lantern seals
11. Pressure of the inert gas in the inert gas pressurized bellows
Three systems are being integrated to control the gripper tests. Watlow and Compax3 systems are used to control the temperature and motor controllers, respectively and can be commanded manually or by a LabVIEW based CompactDAQ system. The choice of these systems is to mimic a typical industrial environment while still having the automatic control and data acquisition abilities afforded by the NI system. Currently, the CompactDAQ system is interfacing the Watlow system via a USB connection and the Compax3 system is setup to auto-run certain pre-preprogrammed sequences. The CompactDAQ is connected via a serial bus (RS-232) but can only give string commands and gather data from the Compax3 including sequence step and torque. Otherwise, the temperature and motor control systems run automatically.

Safety interlock switches are installed, as necessary, on the vessel cover and other system components to ensure that the components are properly installed and secured before the control system will operate. The system control software will be programmed not to allow operation of the system unless all of the safety interlocks are engaged and both the Gripper Test Assembly and METL operational parameters are within nominal limits.

Each of these signals can be used by the control computer software to determine whether a system fault has occurred and respond. The system can respond with anything from an alarm, to shutting down the power to the GrTA, to sending a request to METL to isolate and drain the vessel. A notification will be sent to the system operators via e-mail and text to inform them of the system status and the information will also be available using a secure remote connection to the control computer.
A simplified flowchart is provided as a guide for the control software development on the following page. A process and instrument diagram of the gripper test assembly shows the system connections for the power, inert gas and signal connections which are required for the Gripper Test Assembly operation.

A check list will be provided to the operator before each run cycle to assure that all critical components are secure to prevent erroneous monitor signal flags.

Sign-off sheets will document all of the assembly procedures and safety system checks performed before operation is permitted. These procedures are being developed during the testing phase of the system.

9. Failure Mechanisms in Mechanical Gearing Components

Failure in gearing components occurs by several mechanisms which depend on the type of loading condition applied to the gear while in operation, below is the list of failure modes and the associated loading condition.

1. Shock loading which can cause permanent (plastic) deformation in the gear teeth. In severe cases, shock loading can cause rapid crack formation and propagation at the
root of the gear tooth which breaks the gear tooth from the gear. This type of failure occurs via brittle fracture, ductile fracture or a combination of both.

2. Initiation of sub-surface cracks below the contacting faces of the mating gears (due to cyclically varying contact stresses which are highest below the surface) propagate over time under cyclic loading conditions and material “spalls” (forms pits) on the gear contact surfaces. The contact of the pitted surfaces of the mating gear faces causes increased vibratory loading in the gears, their supporting bearings and other mechanical components in the system which accelerates the reduction in their useful lifetimes.

3. Cracks at the root of the gear teeth which are initiated by cyclically varying bending stresses (fatigue) occur in 10,000 loading cycles or less due to overloading of the gear in cases of low cycle fatigue. High cycle fatigue failure occurs at lower stress levels for loading cycles above 10,000 cycles.

4. Wear occurs on contacting surfaces of the gear teeth usually due to normal loading or the presence of abrasive particles in the lubricating medium and reduces the locational accuracy (of the driven component) and efficiency of the gear, continued wear over time weakens the gear teeth and causes early failure of the gear teeth.

5. Scuffing is caused by galling or “self-welding” of the gear material by adhesion of the gear material from mating gear teeth surfaces and can be caused by improper lubrication of the gears.

6. Material property changes (chemistry, hardness, corrosion and structural) due to environmental considerations during operation can cause early failure of the gearing components by increasing wear rates on the gear material.

10. Failure Mechanisms in Roller Bearing Components

Failure in roller bearing components occurs by several mechanisms which depend on various conditions applied to the bearing while in operation, below is the list of failure modes and the associated environmental condition.

1. Abrasive Wear is caused by foreign particle contamination in the lubricant and can cause bruising, circumferential grooving or accelerated wear. Larger particles can cause more severe bruising, pitting or grooving and can reduce service life.
2. Excessive loads or preloads can cause severe spalling of material from the balls and the contact path is more offset to the race edges in the bearing. The result is shortened bearing life.

3. Overheating can cause reduction in the hardness of the bearing components which reduces the bearing capacity and shorten its life.
4. Total Bearing Lockup is caused by high localized heat which causes metal flow in the bearing components which results in skewing of the rollers, destruction of the cage and seizure of the bearing. (This can also be caused by changes to the bearing surface alloy operating in liquid sodium which cause softening of the material surface and therefore increased wear rate)

5. Fatigue spalling is caused by repeated loading and unloading of the bearing component contact surfaces as the rollers circulate. Spalling of material occurs as the peak effective stresses in the contact region occur slightly below the contact surfaces, initiating sub-surface cracks which propagate in irregular paths (following grain boundaries) up to the surface of the material.
6. Brinelling occurs when static contact stresses exceed the elastic limit of the ring material. The result is elliptical indentations on the raceways. This is caused by excessive external vibrations (False Brinelling) or excessive static loads (True Brinelling).

7. Excessive endplay results in a very small load zone, causing the bearing to unseat leading to roller skidding and skewing. The outcome is scalloping in the cup race and excessive cage wear due to roller contact and impact.
8. Misalignment of the shaft to the machined support shoulder for the outer race. This causes a non-parallel wear path from the rollers on the outer race, increases in temperature and more heavy than normal wear.

Figure 34 – Image of Effects of Misalignment – see page 13 of [9]
9. Improperly machined bearing pockets or poor fitting practice.

10. Handling or installation damage to the bearing components.

11. Incorrect or reversed installation of the bearing to the direction of loading.
12. Inadequate lubrication creates a wide variety of damaging conditions from slight heat discoloration to total bearing lockup.

As the experimental (in-vessel) bearings utilize the liquid sodium as the “lubricant”, bearing life is expected to be reduced.

11. Failure Mechanisms in Universal Joints
A universal joint is a mechanical component which transmits torque and motion between two planar, but, not necessarily parallel shafts. Universal joints can utilize either sleeve bearings or cylindrical pin roller bearings between the yokes and the trunnions. Typically, universal joints operate at angles less than 25 degrees to transmit torque and motion between the shafts. Different failure processes can be associated with the different universal joint designs.

1. Spalling occurs in roller bearing type universal joints near end of life due to normal fatigue during use. Excessive loads or contaminants can increase the rate of spalling

2. Brinelling occurs in trunnions for roller bearing type universal joints from high static loads, external vibration or excessive driveline angle

3. Fractures of the yokes occur if repeated and excessive shock loading is applied
4. Galling between sliding or rolling components occurs with inadequate lubrication or excessive driveline angle

5. Mechanical wear occurs during normal use and is accelerated with inadequate lubrication or contaminants

6. Excessive friction causes heating and “burning” of the trunnions and accelerated wear due mainly to inadequate lubrication or excessive loads
12. Failure Mechanisms in Recirculating Ball Screws and Splines

Recirculating ball screw and spline nuts have similar failure mechanisms to those of normal ball bearings.

1. Abrasive Wear is caused by foreign particle contamination in the lubricant and can cause bruising, helical grooving on the screw or nut races or accelerated wear. Larger particles can cause more severe bruising, pitting or grooving and can reduce service life.

2. Excessive loads or preloads can cause severe spalling of material from the balls and their contact path on the screw and nut. The result is shortened life.

3. Brinelling occurs when static contact stresses exceed the elastic limit of the screw and nut material. The result is elliptical indentations on the raceways. This is caused by excessive external vibrations (False Brinelling) or excessive static loads (True Brinelling).

4. Recirculating mechanism failure occurs when the ball screw is operated at excessive speeds and causes impact and fracture of components which make up the mechanism.

13. Future Development Plans

After completion of the testing of the Gripper Test Assembly and selection of the best combinations of materials for the mechanical components operating in a high temperature liquid sodium environment, a 1/10th scale rotatable plug with a slotted Upper Internal Structure, simulated core and compact fuel handling machine can be fit onto one of the flanges of the 28-inch vessel. Operation of the scaled fuel handling system can be tested and
used to train reactor operators. Faults can be programmed into the system to see how the personnel react and recover to bring the system back online. Procedures can be developed to recover from various scenarios developed by experts.

Artificial Intelligence programming can be incorporated to hasten the recovery and fully automate the refueling process for future fast reactor designs.
14. References

1. FFTF Fuel Handling Loads - “Personal Communication – C. Grandy” September 2019


3. ANSI/ABMA Std. 11-1999, Load Ratings and Fatigue Life for Roller Bearings

4. ANSI/ABMA Std. 9-1990, Load Ratings and Fatigue Life for Ball Bearings

5. ASME Boiler and Pressure Vessel Code Section III Division 5 for High Temperature Reactors

6. CMAA 70 (Specification for Electric Overhead Travelling Cranes- 1988)

7. NUREG-0554 (Single Failure Proof Cranes for Nuclear Power Plants – 1979)


9. Barden Precision Bearings, Bearing Failures, Causes and Cures


11. Ball Screw Failures, Barnes Industries Inc.