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Presented at the
8th Symposium on Engineering Problems of Fusion Research
San Francisco, California
November 13-16, 1979

Publication No. PN-59

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Summary

The Center for Electromechanics (CEM) of The University of Texas at Austin has completed an engineering analysis of controlled imploding liquid metal liner (LINUS) devices for the Naval Research Laboratory (NRL).[1] NRL is involved in the development of this type of high energy density system which has potential use in the areas of compact fusion and pulsed electrical power sources and advanced weapon systems.

Although NRL is presently operating two experimental machines, HELIUS and LINUS-0, the mechanical stresses, electromechanical behavior, and dynamic behavior of larger LINUS-type devices operating repetitively at elevated temperatures have not been completely established. The limitations imposed by these considerations significantly affect the design of these devices. In this paper, dynamic instabilities, material considerations, injection systems, rotor geometries, mechanical stresses, and the effect of elevated temperatures are discussed. The effect of these items also influence the choice between a rotating or a stationary reactor vessel.

Introduction

In the LINUS concept, a plasma is placed in the cavity formed by rotating a liquid metal cylinder or liner inside a partially filled pressure vessel. The liner must spin fast enough to avoid Rayleigh-Taylor instabilities at the free surface of the liner. The liner is constrained at its outer surface by a free piston, which when stroked, displaces the liner, forcing it radially inward. At a sufficiently high magnetic Reynolds number, an adiabatic compression of the magnetic field and plasma raises the plasma temperature to thermonuclear fusion levels (10-15 keV).

The rotation of the liner can be achieved by either rotating the pressure vessel as is done with HELIUS and LINUS-0, or by injecting the liner tangentially.[2] This later method allows for the pressure vessel to be stationary, which eliminates several difficult engineering problems inherent with the rotating vessel concept.

The work presented here is for large fusion reactor size LINUS systems expected to operate repetitively at elevated temperatures.

Design Considerations

Reactor Vessel Material Considerations

The expected operating temperature of these machines is approximately 482°C (900°F). Of the steels and materials available for high temperature service, several types can be dismissed immediately. The low carbon steels are deficient because of their high creep rates and high magnetic permeability. The ferritic steels must also be rejected because of their general high magnetic permeability. Aluminum alloys and beryllium copper cannot be used because of temperature limitations; titanium alloys exist which can withstand the temperatures, but the useful lifetime at these temperatures is short. At the expected operating temperature, the austenitic steels and special proprietary alloys can operate at the highest stress levels of any class of steel for a given creep rate of 10^6 per hour. These steels also have the advantages of good fatigue resistant qualities and low magnetic permeability. Along with the already mentioned design qualifications, corrosion, scaling, and volume availability are pertinent considerations.

A commercially available material was identified which has the required properties mentioned above. A-286 (AISI 660), an iron base austenitic steel, has an upper operating temperature of about 538°C (1000°F), after which its strength characteristics begin to show significant effects of the high temperatures.

There is also the possibility of further optimization of these steels by the addition of certain elements. Both nitrogen and boron to some extent increase the heat resistant nature of steels. Boron also diminishes the irradiation effects. Silicon increases corrosive resistance and reduces scaling.

Liner-Rotor Instability

One method by which the liquid metal liner can be spun up is by spinning the reactor vessel. This method is presently used both in HELIUS and LINUS-0 at NRL. A primary consideration in the design of a reactor vessel employing this method of liner rotation is the self-excited dynamic instability known to exist in hollow rotors partially filled with liquids. Perturbations on the free surface of the liner cannot be tolerated in this type of device since it affects the plasma. An analytical model predicting the limits of the instability exists which matches experimental data very well and should be applicable here.[3] Basically the instability is the result of dynamic coupling between the rotor and liquid motion. The primary result is that between the reduced rotor first critical speed and twice that speed, the liquid liner couples with the rotor whirl motion to produce a large range of self-excited (unstable) asynchronous whirl speeds. From tests run on HELIUS and LINUS-0, they appear to operate below these critical speeds.

Although it will not be possible to operate a rotating LINUS-type system in the unstable region, this should cause no unsolvable problems. The machine can be operated either above or below the unstable region. If it proves necessary to operate above the instability, it will be necessary to fill and empty the rotor at speed or stroke the piston, to avoid going through the unstable range with a fluid free surface in the rotor.

The design of the bearing system should include a means to adjust its effective stiffness. This will allow for the first critical and hence the limits of the instability range to be moved away from the operating speed. This can be done most easily by changing the lubricant viscosity or bearing clearance.

Reactor Vessel Design

Fatigue will be the important criterion for the choice of an operating stress for this application. With the A-286 mentioned before, creep will not be a problem at the anticipated operating temperatures. The expected number of cycles will, however, preclude the use of stresses near the yield range. The operating stress level will be based on an infinite life criterion using the strength characteristics of A-286.

Using data at 538°C (1000°F), a combined factor of safety and a stress modifying factor of two lowers the endurance limit, S_e, to 225 MPa (32.5 Ksi). The yield strength and tensile strength at this temperature are 607 MPa (88Ksi) and 903 MPA (131 Ksi) respectively. With these values a modified Goodman diagram is constructed as shown in Figure 1. In this application the minimum

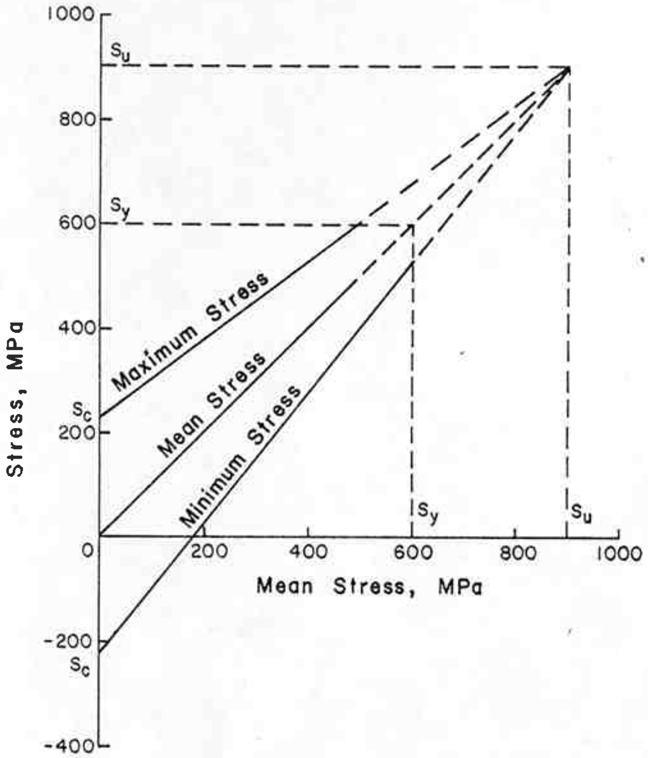

Figure 1: Modified Goodman Diagram for A-286 Steel (AISI 660)

stress is the stress due to rotation of the fluid and/or the reactor vessel. The stress range is simply the difference between the maximum stress seen by a point in the vessel and the minimum (initial) stress. The stress range is, therefore, given simply by calculating the stress due to the driving pressure during a pulse. The allowable stress range, S, is plotted versus the minimum stress, S_{min}, in Figure 2.

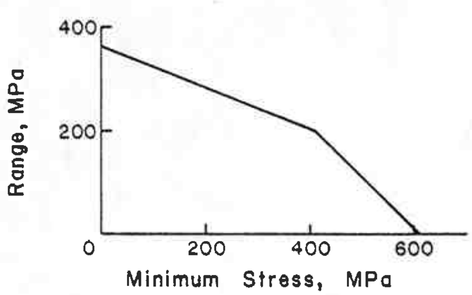

Figure 2: Allowable Stress Range

Figure 3 is a conceptual schematic of one section of the reactor vessel at the time this study was done showing the various dimensions. Much of the detail such as piston guide rods, gas ports, etc., is not shown. Burton and Turchi[4] have shown that a reasonable size for a 700 MW power reactor is an initial liner radius, R, of 1.12 m and a length of 8.9 m. The fluid in this machine would rotate at a rate of 27.7 radians per second.

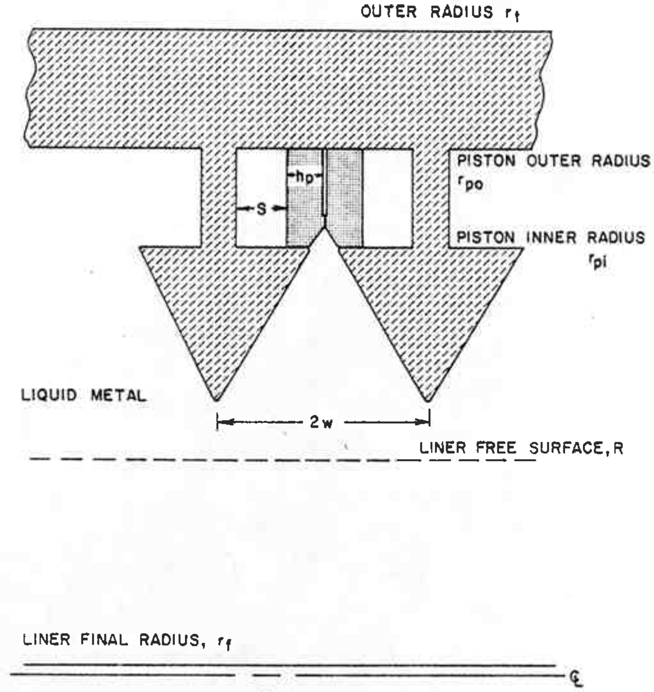

Figure 3: Conceptual Schematic of One Section of the Reactor Vessel

The buffer thickness, the distance between the inner radius of the piston and the initial liner free surface, has been chosen to be equal to twice the piston spacing, or 2 w to minimize the surface disturbances caused by the piston. Tests run at NRL show this value to be reasonable. The buffer thickness has a major impact on the size of the machine and should be kept to a minimum value. Not only does the rotor radius go up proportionally to the buffer thickness, but the chamber wall thickness also increases to contain the higher rotational head of the fluid. If the reactor is rotating, the rotational stress also increases due to higher surface speed.

Setting the volume of fluid displaced by the piston equal to the volume required to completely compress the plasma gives

$$\pi(r_{po}^2 - r_{pi}^2)S = \pi R^2 w$$

Letting r_b = buffer thickness = r_{pi}-R, rearranging and multiplying through by w/W gives:

$$\frac{r_{po}^2}{R^2} = \frac{w}{s} + 1 + 2\left(\frac{r_b}{w}\right)\left(\frac{w}{R}\right) + \left(\frac{r_b}{w}\right)\left(\frac{w}{R}\right)^2$$

Obviously, r_{po} is minimized when $\frac{r_b}{w}$, $\frac{W}{S}$, and $\frac{w}{R}$ are at a minimum value. $\frac{r_b}{w}$ is given a minimum value of 1.0. With this assumption, $\frac{r_{po}}{R}$ can be plotted as a function of $\frac{W}{R}$, for various values of $\frac{W}{S}$. This yields the family of curves shown in Figure 4.

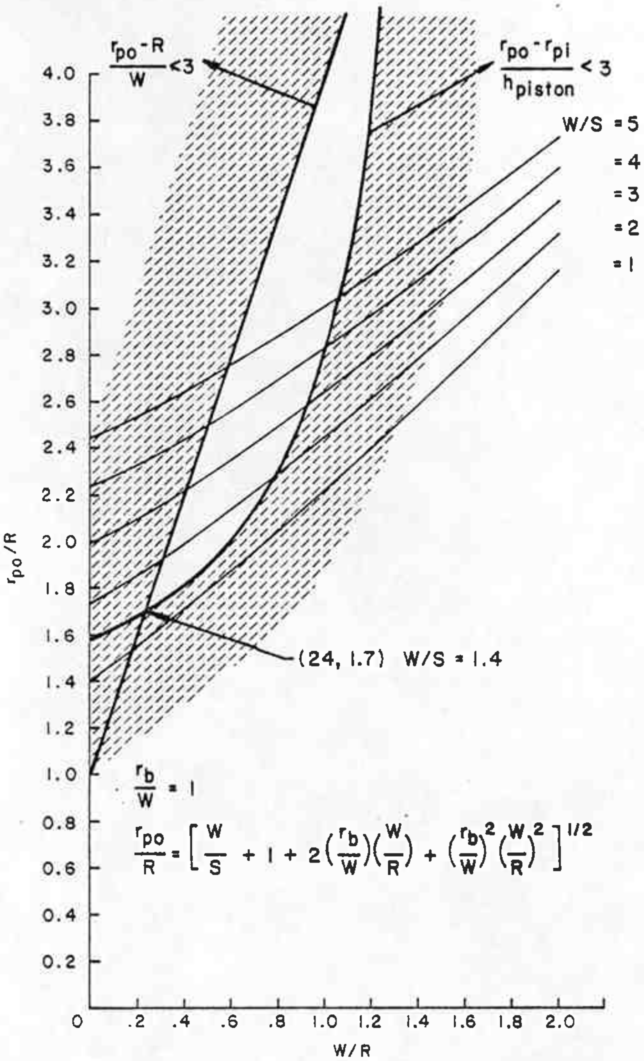

Figure 4: Minimum Rotor Dimensions

Constraints must be used in conjunction with the above equation to prevent operation with parameter values which are undesirable for other reasons. The buffer thickness, r_b, must be some minimum value in order to have a smooth free surface at turnaround. A value of

$$\frac{r_b}{W} = 1$$

was chosen as being reasonable. The aspect ratio of the fluid cavity must be limited for hydrodynamic reasons. A constraint of

$$\frac{r_{po}-R}{W} < 3$$

was chosen. The aspect ratio of the piston must be controlled to prevent frictional lock-up. Using a coefficient of friction of 0.33 gives

$$\frac{r_{po}-r_{pi}}{h} < 3.$$

These constraints are then plotted in Figure 4 to give the feasible boundaries of the graph. The minimum values occur where the two constraints cross and are:

$$\frac{r_{po}}{R} = 1.7$$

$$\frac{W}{R} = 0.24$$

$$\frac{W}{S} = 1.4$$

$$\frac{r_b}{W} = 1.0 \text{ (assumed)}$$

If R is taken to be 1.0 m, then

$$r_{po} = 1.7 \text{ m}$$

$$W = 0.24 \text{ m}$$

$$S = 0.17 \text{ m}$$

$$r_b = 0.24 \text{ m}$$

Book and Turchi[5] identified a "water hammer" wave which is created by the rapid turnaround of the fluid at the peak of the implosion. This wavefront becomes steeper and decreases in magnitude as it travels outward. For a typical case, the pressure in this wave is approximately 0.025 ρc^2 when it passes the initial free surface position of the liner, where ρ is the density of the material and c is the wave propagation velocity in the material. For the lead-lithium liner which will be used in the reactor the pressure in the wave is 678 MPa (98,000 psi). Clearly, the reactor vessel cannot be made to take this high a pressure using the chosen material and keeping a reasonable size.

As the compression wave contacts the outer wall, part of it will be reflected back into the liner and part of it will be transferred into the wall. That part which is reflected back into the liner could distort the free surface, although that should have no significant consequences since the reaction has already taken place. The wall stress resulting from this complex wave propagation problem is the item of concern.

In general, the transmission of a shock wave across an interface of two different media is determined by the acoustic impedance, Z, of the two media.

$$Z = \frac{1}{\rho c}$$

Depending upon the relative values of Z, the wave can either be amplified or reduced as it crosses the interface. However, this amplification factor has lower and upper limits of zero and two respectively.

In the case of a wall made of steel, the acoustic impedance of the wall is less than the liner and therefore an amplification of the wave amplitude can be expected in the wall.

A detailed answer to this problem was beyond the scope of this project. It is expected that a more detailed study will show that the pressure in the wave must be significantly reduced before it reaches the reactor wall. One way that this could be accomplished would be to introduce a "free" surface in the fluid between the plasma and the outer wall. This could be accomplished by making the triangular sections of the rotor from a material with high acoustic impedance or by actually introducing a discontinuity in the liner. If this discontinuity is introduced at a large enough radius and at a time close enough to turn around, the effect on the inner free surface would be negligible. One possible way of doing this is discussed.

The majority of the previous work on liner implosion has dealt with "free liners," i.e., liners that are not actively driven all the way to the final radius. This approach may be possible in this case, also. If the piston stopped at some fairly large fraction, e.g., 0.95, of its total stroke, the trajectory of the inner portion of the liner would be unaffected due to compressibility and wave transit time effects. There would likely be a separation in the fluid and a free surface would be generated to attenuate the shock wave propagation.

Since it became obvious that the pressures in the area of the triangular sections of the reactor wall could be higher than the driving pressure and that these sections might prove useful in shielding a portion of the outer wall from these higher pressures, a calculation of the stresses in these sections was performed. The section was considered to be triangular in cross section with a slight truncation of the inside tip.

The stress due to rotation was found to be 34.1 MPa. This will correspond to S_{min} in Figure 2 and will not significantly affect the allowable stress range.

By placing a blunted nose on the triangular piece, the peak stress from the pressure resulting is tremendously reduced. Increasing r_i by putting a "false nose" on the section also seems to have a beneficial effect. Considerable optimization is possible on this section, but it is significant that a sharp point is not desirable. It should be noted that in the present conceptual design being pursued by NRL, these triangular sections have been alleviated (see Figure 5).

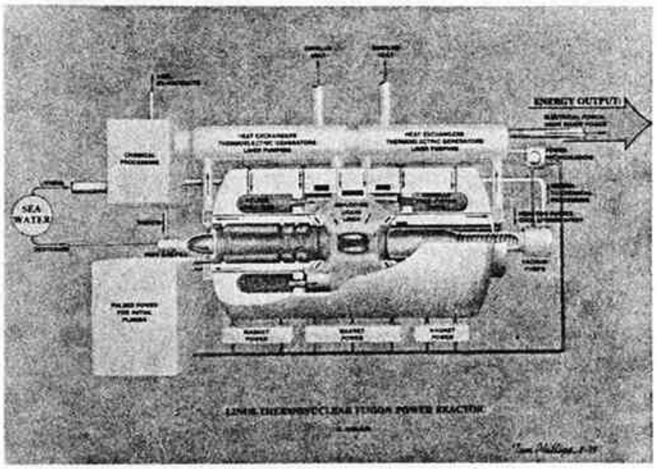

Figure 5: LINUS Thermonuclear Fusion Power Reactor

The wall load fluctuates during a pulse. Before implosion the pressure due to the rotational forces on the fluid is 6.02 MPa (872 psi). During the implosion, the pressure is determined by the driving pressure. The required driving pressure, P, calculated in reference 4 was 45.4 MPa (6800 psi). If the triangular portions of the rotor are neglected and if the driving pressure is taken to be the maximum loading seen by the wall, the required outer radius can be calculated. Note that this assumes no part of the "water hammer" wave reaches the rotor. The first step is to assume a total combined stress, S_{min}, in the wall due to initial rotational head and spin stress. This assumption will be checked after r_t is calculated. An allowable stress range, S, can then be obtained from Figure 2. The equation for the total stress in the wall due to the driving pressure is

$$\sigma = 2P \frac{r_t^2}{r_t^2 - r_{Po}^2}$$

Letting $\gamma = r_t/r_{Po}$

setting $\sigma = S$

and rearranging, this equation becomes

$$\gamma^2 = \frac{1}{1 - \frac{2P}{S}}$$

Substituting in the values for P and S gives γ. With the previously established value of $r_{Po} = 1.7$ m, r_t is then known. The stress due to the spinning of the reactor is calculated using

$$\sigma = \frac{\rho\omega^2 r_t^2}{4} \left| (3 + \nu) + (1 - \nu) \frac{1}{\gamma^2} \right|$$

The sum of the initial rotational head and the spin stress can then be compared to the originally assumed S_{min}. The estimated S_{min} can then be revised until it agrees with the calculated one. The final results give values of $\gamma = 1.18$ and $r_t = 2.00$ m. The initial stresses due to rotational head and spin are 43.5 MPa and 22.3 MPa respectively and the allowable stress range is 328 MPa.

Actively cooling the outside of the outer wall will effectively reduce the initial stress by reducing the stress gradient through the wall. In this case, a reduction of 30% seems reasonable. This gives an allowable stress range of 338 MPa. The resulting outer radius is 1.99 m. It is interesting to note that by making the reactor vessel stationary and removing the initial spin stress, the outer radius is only reduced to 1.99 m.

No attempt was made during this study to design the endwall or to evaluate its stresses. The pressures seen by this endwall can be quite large during the implosion. There are several possibilities which could help cope with this problem and should be considered. Conducting end walls may help distribute the loads. A hole may be left in the end wall where the high pressures are developed as is done in Figure 5. Also, clever mechanical design could distribute the pressures over a larger area.

Magnetohydrodynamic Effects

There are several significant magnetohydrodynamic, (MHD), effects occurring in the liquid metal liner both before and during the implosion. Most of these effects are second order in nature, but even second order effects can be important when dealing with very high power densities.

If the end walls are conductive, circulating currents between them and the liquid metal liner could locally distort the flow patterns at the liner free surface.

Although no data is available to indicate the minimum magnetic field required to damp out turbulence of the flow in this type of geometry, the fields which will be present will probably accomplish this. The Hartmann numbers produced by these fields are in the range of a few hundred to a few thousand. Since the Hartmann number is an approximation of the ratio of the magnetic force on the fluid to the viscous force, this results in a substantial increase in the effective fluid viscosity. The viscous Reynold's number for the flow would thus be reduced significantly.

If the reactor vessel is stationary, another problem due to the axial field occurs. The rotating fluid acts as a homopolar type generator, and the conducting structure acts as brushes and a return circuit. The losses associated with this phenomenon could prove to be quite large although this situation can be remedied by either canceling the appropriate component of the magnetic field in the liner region and/or by electrically insulating the surfaces.

Stationary vs. Rotating Reactor Vessel

The total mass of the reactor vessel described above (2.0 m radius by 8.9 m long) is approximately 1.07×10^6 kg (a weight of 2.35×10^6 lbs.), including 20% for end walls. If the rotating vessel design is chosen, a bearing is required to support this rotor while it spins. The most attractive design would be to run the bearings on a reduced diameter shaft connected to each end of the reactor. The smaller diameter reduces the losses. Considering the simplest design, a sleeve type journal bearing, the required diameter was calculated to be 0.865 m and the total power loss was 82.9 kw, assuming a lubricant viscosity of 13×10^{-6} centipoise.

None of the findings of this study showed either the rotating or the stationary vessel to be clearly superior. However, the stationary vessel does seem to be favored somewhat as this concept allows a smaller reactor and the bearing is eliminated. The main design complication with this scheme was the need to reduce eddy current losses on the inside walls of the reactor vessel. The seal problem is probably the most significant factor which would lead to the selection of the stationary concept. Both the outer, high pressure seal and the inner, high vacuum seal may prove to be quite difficult in the rotating pressure vessel concept. And in fact, as a result of the work presented here and the encouraging result found with the tangential injection model[2] tested at NRL, the stationary reactor vessel configuration is being pursued by NRL for the conceptual LINUS reactor design shown in Figure 5.

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