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MECHANICAL ENGINEERING DEPARTMENT (QUARTERLY REPORT July through September 1977

Scientific Editor: R. G. Stone
General Editor: M. B. Bathgate

September 30, 1977

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We are developing a concentrating solar collector, which will produce hot water and steam up to a temperature of about 170°C. Such a collector is of interest because industrial process heat below 170°C accounts for 5% of this country's total energy consumption. Due to the simplicity of the collector design, it is expected to be cost-effective relative to fossil fuels such as oil at \$15 per barrel.

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MECHANICAL ENGINEERING DEPARTMENT QUARTERLY REPORT

July through September 1977

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DEVELOPMENT WORK ON SUPERCONDUCTING COILS FOR THE LARGE MIRROR FUSION TEST FACILITY

Introduction

Several recent experimental and theoretical breakthroughs have put the magnetic mirror program in the U.S. in a position to take a major step ahead. The remaining scientific and engineering problems must be solved on an intermediate-size machine before we build an experimental power reactor. At LLL, we propose to meet these needs by building the Mirror Fusion Test Facility (MFTF), a large fusion research facility that will be a successor to LLL's Toy Top, Table Top, Baseball, and 2X magnetic-containment experiments. We propose to start construction of MFTF in late 1977 and begin experimental work in 1981.

The MFTF vacuum vessel and the main components associated with it are illustrated in Fig. 1.

Neutral-beam injectors for the short-duration, 20-kV startup beam and for the 80-kV buildup and sustaining beams are mounted on four large flanges around the cylindrical portion of the 9-m-diameter, 12-m-high vacuum vessel. Streaming-plasma guns mounted in the top cover inject stabilizing plasma along the magnetic field lines. The dc magnetic-mirror field is produced by a pair of coils with a yin-yang configuration.

The neutral beams must have maximum access to the central plasma region. The shapes of the coils and the force-restraining structure have been adjusted accordingly, and the whole region between the large lobes of the coil is left unobstructed by support structure. Because enormous pumping facilities are required to pump away the unwanted

Fig. 1. The main chamber of the Mirror Fusion Test Facility.

particles entering with the neutral beams, all the surfaces of the vacuum vessel are lined with cryopumping panels that are cooled by liquid helium.

The geometry and size of the coils impose certain constraints on the superconductor. In this article, we discuss the proposed conductor and winding design, briefly describe the development of fabrication and joining techniques, and present the results of preliminary investigations into cryostatic stability.

The MFTF Magnet

The tentative mean major and minor radii of the MFTF coils are 2.5 and 0.78 m, respectively. The coils provide a peak field of 7.5 T (see Ref. 1 for additional details).

The Nb-Ti superconductor is immersed in liquid helium at approximately 1 atm, resulting in a maximum coolant temperature of 4.5 K. Natural convection is the only cause of liquid circulation. To facilitate its being wound into the yin-yang shape, the conductor has a square cross section. As a result of the coil configuration, the magnetic forces are transmitted from turn to turn to the coil case and to the external support structure, and very little is taken as tension in the conductor. A preliminary analysis of the buildup of this compressive stress indicates a maximum value of approximately 41 MPa.

The additional cooling surface required for cryostatic stability is provided by cooling channels incorporated in stabilizing copper that is subsequently soldered to the superconducting core, which is a 6.5-mm by 6.5-mm monolith. Two fabrication routes are being investigated to achieve this. The most promising is the wrap-around process (Fig. 2);

it is almost certain that this route will be adopted. The alternative technique using four bars soldered together, only one of which contains superconducting filaments, has been described in Ref. 1.

In the wrap-around technique, the superconducting core is passed through a continuous electroplating process prior to applying the stabilizer. The latter is prepared from an OFHC copper strip by first slitting and rolling the strip and finally by punching helium access holes. On a roll-forming line, the stabilizer is wrapped around the core as it proceeds through a series of rolls, and the combination is then sized by being passed through a Turk's-head. The completed conductor then passes through the soldering furnace, a quenching system, and a cleaning process before being wound onto the supply drum. Wrap-around fabrication is described in detail in Ref. 2.

Each coil is made up of a series of pancake coils. The insulation between pancakes and turns is similar to that used on the Baseball II magnet: sheets of perforated epoxy-impregnated fiberglass. Because of the large dimensions of the coils, this insulation is in sections and is made up of two layers of 0.08-mm-thick, overlapped pieces. Evenly spaced, 1.14-mm-thick, epoxy-impregnated fiberglass buttons placed between conductors allow the helium to reach all four sides of the conductor. A machine has been specially developed for punching these buttons and for attaching them at equal spaces along a flat Dacron tape. Produced in this way, the buttons can be wound from a separate spool in parallel with the conductor. The buttons are grooved to take the Dacron tape, which is then held in position by a fast-setting glue. This bond has been tested at 77 K; however, even if it should fail in service, the button will remain trapped in position by the tape.

Fig. 2. Wrap-around technique for fabricating Nb-Ti conductor.

Measurements of Heat Transfer from Short Lengths of Dummy Conductor

In the conceptual design stage, we decided that the operating current, safety margin, and (therefore) overall current density would be determined from results obtained on a test coil wound with prototype conductor and insulation. We used the method outlined by Maddock *et al.*³ to obtain a preliminary cryostatic stability limit. However, because of the differing orientations of the cooling channels and the external surfaces, and because of the orientation of the conductor itself in the magnet, it is difficult to estimate the heat-transfer characteristics of the proposed conductor. Thus, even though the design current will be determined from the test coil data,

we thought it worthwhile to use preliminary samples of dummy conductor to investigate the heat-transfer characteristics of the conductor. If, in the future, we can use results from similar heat-transfer tests for the reliable prediction of coil stability, we could save considerable time and expense, particularly where a large test coil would otherwise be required.

Because there was not yet a production line, short lengths (26.7 cm) of conductor were hand-made for these tests. The conductor was made entirely of OFHC copper; we simulated the "normal" state of the conductor by energizing a 0.25-mm-diameter Manganin wire that was inserted through the center of the copper core. Both ends of the conductor were thermally insulated from the helium. To measure any temperature gradients, thermocouples were indium-soldered at points along the length of the conductor. Figure 3 shows a summary of the results with the conductor mounted in the vertical position and then at 10° to the horizontal. The temperatures in Fig. 3 were measured at the center of the conductor; temperatures at the ends were <0.2 K lower. These results indicate little dependence on conductor orientation and some hysteresis between nucleate and film boiling.

To simulate more closely the thermal environment in a coil, a bundle of nine conductors was then made up. Thermocouples were mounted on the central conductor in the bundle, and two series of tests were carried out with the axis of the bundle horizontal. The series differed in that in one, only the central conductor was energized, and in the other, all nine were energized. It is interesting to note from the results in Fig. 3 that in both series of tests on conductor bundles the heat-transfer curve is continuous and is the same for increasing and decreasing heat fluxes.

In Fig. 4, calculated heat-generation curves for the MFTF conductor have been superimposed upon the heat-transfer curves from Fig. 3. The critical temperature at 7.5 T was taken as 5.7 K, and the temperature of the liquid helium was assumed to be 4.5 K—the heat-transfer tests on the bundle were taken at a pressure equivalent to approximately 4.4 K.

Because of the handmade nature of the samples, these tests are only preliminary; we intend to conduct further similar tests using production conductor and production insulation. We shall also investigate the effect of having only two or three adjacent conductors normal instead of all nine, since the latter is a very severe case.

Fig. 3. Heat-transfer data on preliminary samples of wrap-around conductor. Conductor length = 26.7 cm; external conductor surface area = 134 cm^2 ; total conductor surface area = 292 cm^2 . The labeled points are boiling fluxes with values (in W/cm^2) depending as follows on the cooling surfaces included in the calculation:

	Peak nucleate (A)	(C)	Minimum film (B)	(D)
External surface only	0.66	0.60	0.47	0.4
Cooling surfaces of channels included	0.30	0.28	0.22	0.19

Joints in the Conductor

The total length of conductor required for each MFTF coil is about 21 km. However, because only about 600 m of superconducting core can be manufactured in one piece with existing equipment, lengths of conductor must be joined. Furthermore, each coil is made up of a number of pancake coils that must be joined in series after they have been wound. Half of these joints are on the inside of the winding and half on the outside.

We have investigated a number of joining techniques, including soft soldering, ultrasonic joining, explosive bonding, diffusion welding, and cold welding. For the conductor in question, the best and most reliable techniques are explosive bonding and cold welding. The explosive-bond method gives the lowest resistance,⁴ but has two drawbacks: the necessary safety precautions make it difficult to use

Fig. 4. Heat-generation and cooling data for the bundle of nine wrap-around conductors. Assumed bath temperature = 4.5 K; T_c at 7.5 T = 5.7 K. Calculated heat-generation curves are given for the following currents: (a) 4 kA, (b) 5 kA, (c) 6 kA, and (d) 6.5 kA. Cooling curves are given for (e) all conductors energized, and (f) one conductor energized.

during the winding process, and it is difficult to make the joints on the outside of the pancake coils close enough to the winding. In cold welding, on the other hand, the machine head containing the pressure dies that make the joint can be brought close to the winding. A butt joint is made with five consecutive applications of pressure, shortening each conductor about 6 mm each time, and the surplus material is extruded radially outwards in the vicinity of the joint as shown in Fig. 5(a). After the excess material has been removed, the joint is as shown in Fig. 5(b). The Nb-Ti filaments are not welded together, as can be seen from the partially etched joint in Fig. 5(c). Little skill is required to make very reproducible joints because dirt, grease, and oxides are all automatically removed from the joint region in the first stage of making the joint. Thus, cold-welding with a butt joint on a commercially available machine seems most likely to meet all the requirements.

Typically, the ultimate strength of the parent material is about 550 MPa at 300 K and rises to 860 MPa at 4 K. We have not completed mechanical tests on the cold-weld joints, but preliminary results at 300 K indicate a strength close to that of the conductor. Although data are not yet available at 4 K, an increase of about 50% is expected if the joint behaves as cold-worked copper. Joints made in the core, which is subsequently subjected to the wrap-around soldering process described above, become

partially annealed, with a resultant loss of strength to approximately 390 MPa at 300 K.

Figure 6 shows the resistance of the cold-weld joints at various values of field strength. Two sets of results apply to joints in core from one manufacturer and the third to core from another manufacturer. The stabilizer has a resistance of about $7.5 \times 10^{-8} \Omega/\text{cm}$, which will reduce the resistance of the joint still further.

The above remarks apply to the superconducting core only. Joints made during the winding process will also require the stabilizer to be joined. Two possible alternatives are being considered for this. The first is to remove the stabilizer from the two ends to be joined and re-solder it back into place after joining the core. The second alternative is to cold weld the conductor complete with stabilizer. This latter method would be much quicker and easier, but considerable development work is still required to determine how satisfactory such a joint would be.

Fig. 5. A cold-weld joint in MITF superconducting core, showing the joint (a) unwelded, (b) trimmed to size, and (c) etched with nitric acid.

Fig. 6. Resistance of cold-weld joint in three samples of nominal MFTF Nb-Ti superconductor.

Test Coil

A test coil is being built using approximately 2 km of prototype MFTF conductor. This coil will be used to study cryostatic stability, the propagation and recovery of normal regions initiated with heaters so that the operating current of the final mirror machine can be fixed, and diagnostic techniques such as strain measurement that will ultimately be used in the MFTF magnets. The bore of this coil is 1 m, which is consistent with the minimum bend radius on the MFTF magnets; the outside diameter is about 1.7 m. The winding consists of 18 pancake coils with 24 turns per pancake. The maximum self-field of the coil is expected to be about 4 T.

By assembling this coil between two other coils, we propose to boost the maximum field to about 6.7 T. These other two coils have a dual purpose: they are also the first of four coils built to provide an 8-T, 1-m-diam background field, inside of which multifilamentary Nb₃Sn test coils will be operated up to total fields of about 12 T. The background-field coils are layer wound with 0.76-mm-thick, epoxy-impregnated fiberglass slats varying in width from 19 to 25 mm as the coil diameter increases. Insulation between turns is 0.013-mm-thick, 0.25-mm-wide Mylar wound on edge. Approximately 50% of the wide face of the conductor is available for cryostatic cooling, giving a heat transfer of 0.3 W/cm² at the design current of 1100 A. Specifications for the conductor for the background-field coil are included in Table 1.

Table 1. Specifications for conductor for background-field coils and preliminary specifications for MFTF core (both are monolithic).

Specification	Background coils	MFTF core
Dimensions (mm)	2.5 × 10	6.5 × 6.5
Ratio of Cu to superconductor	4.33/1	Not specified ^a
Critical current at 7.5 T, 4.2 K (kA)	2.4 (actual)	11
Twist pitch (cm)	10 to 15	10
Filament diameter (mm)	0.1408	≤ 0.254
Number of filaments	300	Not specified ^a

^aLeft to judgement of manufacturers.

To gain experience and data on which to base the final conductor, we supplied preliminary specifications for the MFTF superconducting core (see Table 1) and ordered a billet of finished conductor from each of several manufacturers. The manufacturers were permitted to choose the ratio of copper to superconductor because this left them free to use their own judgment and experience to meet the specification in the most economical way; the stability of the final conductor is relatively insensitive to the amount of copper in the core since most of the copper required for stabilizing is added later. Critical-current measurements (using a sensitivity criterion of 10⁻¹¹ Ω·cm) for samples taken from three manufacturers are shown in Fig. 7. These

Fig. 7. Critical currents of nominal MFTF superconducting cores made by three different manufacturers.

lengths of superconductor will have stabilizer added to them and will then be used for winding the test coil.

Conclusion

This article has summarized development work directed toward obtaining the data required to

design and build the superconducting coils for MFTF. The methods for fabricating the conductor and joining lengths of the conductor are almost finalized, and the building of the test coil and associated equipment is now well under way.

AN INFLATED CYLINDRICAL CONCENTRATOR FOR PRODUCING INDUSTRIAL PROCESS HEAT

Introduction

We are developing a concentrating solar collector, which will produce hot water or steam up to a temperature of about 170°C. Such a system is of interest because approximately 25% of the energy consumed in the United States is in the form of industrial process heat⁵ with about 20% of this heat used at a temperature of 170°C or below.⁶ Therefore, such collectors could potentially supply 5% of this country's total energy needs in the form of industrial process heat.

Since it is important that these systems be economically competitive with other energy sources, a major goal of our work is developing a low-cost system. It is reasonable to expect most industrial concerns to require an after-tax return rate of 10% or more on an investment in a solar process heat system. This is roughly equivalent to a payback period of 7 years and translates into an allowed solar system cost of about \$50 per square meter of collector area.⁷

We plan to achieve this extremely low cost by using inexpensive, weatherable plastics and by developing a design which does not require tracking of the sun. The use of plastics most likely means replacing these components at regular intervals. However, in an industrial environment, operation and maintenance people are available to perform this task. We also believe the life cycle costs can be kept low. Our analysis of this design shows that it can likely compete with fossil fuels such as oil at \$15 per barrel.

Collector Description

This type of inflated concentrating collector was first suggested by Tabor in Israel around 1960.⁸ The basic idea is to form a reflector with a large, inflated, thin-film plastic cylinder whose upper portion is clear and whose lower portion is an aluminized thin film (Fig. 8). The axis of the cylinder lies in a horizontal east-west direction, and the

collector is designed so it can be manually tilted in the north-south plane. The reflector, with its circular cross-section, does quite an adequate job of focusing sunlight to a region somewhat above the reflector surface along the normal axis. It is particularly good, relative to, say, a parabolic shape, at focusing sunlight which enters the collector slightly off the normal axis. A receiver tube with a selective surface will be placed in this focusing region to carry the fluid to be heated or boiled. Surrounding the receiver tube is a thin-film, cylindrical plastic jacket designed to aid in reducing convection and radiation heat transfer from the hot receiver tube.

Optical and Heat Transfer Code Descriptions

Our initial efforts were directed toward developing optical and heat transfer code: to model the inflated cylindrical concentrator performance. The ray-tracing optical code begins by dividing the collector aperture into a grid and then assigning to each discrete grid region a power level which is proportional to the insolation chosen as input. Thus, many sun ray vectors are created, each having a direction and a power magnitude. The code then follows the three-dimensional paths of these discretized incoming sun rays as they are transmitted, reflected, and absorbed within the collector. For each interaction with a surface, the code decreases that ray's current power magnitude by an amount which is a function of the surface's optical properties and the ray's incident angle.

The energy absorbed by the receiver tube in a specified time interval is calculated by summing the energy of all rays that actually strike the tube. By repeating this process many times, we get the total energy absorbed for the day and if desired, for the year. This allows us to understand how the total collected energy is affected by different optical properties, and it enables us to do parameter studies with the many geometrical variables.

Fig. 8. Inflated cylindrical concentrator for producing industrial process heat.

Our heat transfer analysis began with the development of a model of the collector based on a thermal resistive network. We chose four nodes at which to calculate temperatures (inner and outer receiver tube surfaces, plastic receiver tube jacket, and outer plastic cylinder). The thermal resistances were set up among the nodes for all three modes of heat transfer. A computer program was then written to solve for the four unknown temperatures. This is an iterative process because the radiation resistances are a function of the nodal temperatures. An additional complication occurs because the partial infrared transparency of the plastic cylinders must be handled in the radiation resistance equations.

When the iterative processes finally yield consistent nodal temperatures, the heat gain in the fluid can be calculated. Thus, the average temperature of an element along the length of the receiver tube is determined. The sequence continues until the fluid reaches the saturation temperature. The heat losses now remain constant until about 10% of the water is vaporized. After this the heat transfer rate between the receiver tube and the saturated mixture changes. We are currently studying the correlation equations which apply to boiling heat transfer above 10% quality.

One of the inputs to the heat transfer code is the energy into the receiver tube that was calculated with the optical code. Other inputs include optical and thermal properties, ambient and sky temperatures, and wind velocity.

Results from the Code Studies

Our optical and heat transfer codes have proven invaluable in several respects. They have given us detailed information on how geometry, optical and thermal properties, and ambient conditions affect collector performance. This has allowed us to deduce overall system performance and operating strategies. Also, an analysis of particular geometries has led to an understanding of the accuracy needed in positioning components in an actual collector assembly. Most of the calculations we did are referenced to the baseline conditions given in Table 2.

Optical Code. A subroutine in the optical code generates a graphic display of the discretized sun rays inside the collector. Figure 9 shows a representative series of these rays as they cross the collector equator, hit the receiver tube directly or are reflected, and then either strike the receiver tube or miss it. For this particular plot, the sun position is 3°

Fig. 9. Computer-generated graphic display of the sun ray paths inside the collector. The sun is 3° off the normal axis.

below the *normal axis* of the collector, and the ratio of outer cylinder diameter to receiver tube diameter is about 14 to 1.

The angle between the collector normal axis and the sun projected into the meridian plane is designated α . We have studied the way varying α affects the amount of energy absorbed by the receiver tube. Our results indicate that manually reorienting the collector tilt once a week in the north-south plane gives us over 90% of the energy that we could get by active tracking in the north-south plane throughout each day. This should be a

very cost-effective tradeoff, and the elimination of a tracking mechanism will increase the system reliability.

The optimal position of the receiver tube relative to the reflector surface is along the normal axis at approximately $0.40 R$ above the reflector, where R is the radius of curvature of the reflector. Calculations indicate, however, that precise positioning of the receiver tube is unnecessary. Placing the tube center anywhere within a circle whose radius is equal to that of the receiver tube yields at least 95% of the energy of the optimal position. Since the

Table 2. Baseline configuration for calculations.

Diameters	
Receiver tube inside	0.045 m
Receiver tube outside	0.050 m
Jacket	0.066 m
Outer cylinder	1.000 m
Temperatures	
Ambient	20.0°C
Ground	20.0°C
Sky	10.0°C
Miscellaneous	
Wind velocity	0.05 m/s
Mass flow rate	2.52×10^{-3} kg/s
Optical energy absorbed by receiver tube	300.0 W/m
Saturation temperature	170.0°C
Conductivity of receiver tube	43.3 J/ms °C

receiver tube will likely be close to 70 mm in diameter for a 1-m outer cylinder diameter, we foresee no positioning problem in an actual installation.

Graphics plots similar to Fig. 9 indicate that there are other regions where there is some concentration of sun rays. Two such areas are on the normal axis, first, just above the reflector surface and, second, just below the optimum receiver tube position. Since our optical code can calculate the energy absorption of a second receiver tube, we have begun a study of the possible advantages of utilizing additional receiver tubes to increase the total amount of energy out of the collector. Initial indications are that over 20% additional energy is available in each of these two regions relative to the optimal position. We do not yet know, however, whether an additional tube, or tubes, will be cost effective when the added heat loss and expense are considered.

Optical properties such as the transmissivity of the clear plastic cylinders, directly affect the amount of energy transferred to the fluid in the receiver tube. Every time a sun ray is transmitted through a film its energy is multiplied by the direct beam transmissivity of that film. Most rays that strike the receiver tube have undergone four such interactions with collector materials—outer cylinder transmission, reflector reflection, jacket transmission, and receiver tube absorption. It is clear, then, that the optical properties will be a crucial determinant of the collector's performance.

Heat Transfer Code Results. We are finding that some factors have little influence on the total heat loss from our collector. For instance, at an

operating temperature of 170°C, a wind velocity varying from 0.2 m/s to 9 m/s changes the total heat loss by much less than 1%. The underlying reason for this is that the outer plastic cylinder is always very near ambient temperature. Consequently, the driving force, the temperature difference, is always quite small (less than 2°C).

Similarly, the infrared transmissivity of the plastic jacket surrounding the receiver tube has a small effect since the heat loss from the selective surface of the receiver tube is predominately by natural convection. Therefore, even fairly large changes in the radiation component have a small effect on the overall heat loss.

Conversely, we have found that the surface properties of the receiver tube are critical. It is mandatory to have a selective surface when operating at 170°C. This is clearly shown in Fig. 10 where we plot the thermal efficiency* versus the receiver tube fluid temperature for surface emittances of 0.1, 0.2, 0.3, and 0.9. For the best selective surface the thermal efficiency is 62% and for the nonselective surface it is only 11%. Even at the relatively low temperature of 100°C, the comparable efficiencies are 77 and 51%, respectively. These latter figures show over a 50% increase with the selective surface, which certainly will be a cost effective inclusion in almost all systems.

We have investigated the annular size between the receiver tube and the surrounding plastic jacket. This dimension does have a minimum for each diameter of receiver tube, and it involves a tradeoff between the three modes of heat transfer.

*Thermal efficiency is defined as the energy into the fluid divided by the optical energy absorbed by the receiver tube.

Fig. 10. The effect of emittance on collector performance is shown at various fluid temperatures. The high emittance value is representative of a nonselective surface.

The optimal gap size for minimum conduction is, of course, as large as possible. On the other hand, for minimum radiation and convection, the gap should be as small as possible and below a minimum size, respectively. The synthesis of these effects is shown in Fig. 11 which indicates the optimal gap size for minimum heat loss as a function of receiver tube diameter. Several things are worth noting about these results. First, the actual optimal gap sizes, around 10 mm, are ones which can easily be maintained in the field with simple, washer-shaped spacers. Second, the optimal gap size changes slowly with tube diameter. And, third, we find that the increase in heat loss is quite small for gaps that are larger than the optimum.

The maximum stagnation temperature of the receiver tube will be reached when there is no flow within the tube. The two variables that are controlling in this situation are the receiver tube diameter and the amount of optical energy absorbed by the receiver tube. A 40-mm-diameter receiver tube is the smallest that is likely to be used with a 1-m-diameter outer cylinder. The maximum energy absorbed by this receiver tube at the summer solstice is about 340 W/m of length, according to our optical code calculations. This results in a stagnation temperature of about 340°C. The surrounding plastic jacket reaches 130°C under these conditions.

We have found that an informative aid in understanding the heat transfer in this collector is a diagram of all the heat fluxes for a given set of conditions. This is shown in Fig. 12 for our baseline configuration at a fluid temperature of 170°C.

System Efficiencies Results. Before discussing the results of our efficiency calculations, it is necessary

Fig. 11. Radial gap between the receiver tube and jacket which minimizes heat loss.

to describe our use of the term optical efficiency. (Thermal efficiency was previously defined as the energy into the fluid divided by the energy absorbed by the receiver tube.) We choose to define optical efficiency as the energy absorbed by the receiver tube divided by the direct beam insolation available to the collector. Both quantities are integrated over a 6-h day, and the insolation is summed over the entire reflector length and width.

Although we have assumed that the reflector includes the entire lower portion, or half circumference, of the outer cylinder, it is actually useful over only the central portion (when a single receiver tube is used). As can be seen in Fig. 9, the outer portions do not focus sun rays onto a reasonably sized receiver tube. Were we to use only this central portion of the reflector as the width, our calculated optical efficiencies would be higher by about 50%. However, when we calculate the solar system cost per unit area, the entire length times the outer cylinder diameter is used as the collector area. Consistency, then, dictates that the area bases be identical.

We get collector system efficiencies by calculating separate optical and thermal efficiencies and then multiplying the two together. These three efficiencies are shown in Fig. 13, for our baseline configuration, as a function of the fluid temperature in the receiver tube. The optical efficiency is not a function of temperature and remains at 37% for this configuration. The thermal efficiency, of course, decreases with increasing temperature. Combining these two curves gives the overall collector efficiency, which at 170°C is 20%.

Another meaningful way to study efficiencies is by plotting them as a function of receiver tube diameter at a given temperature. In this plot, Fig. 14, the optical efficiency continually rises while the thermal efficiency continually decreases as the receiver tube gets larger. Note that there is a rather broad, flat maximum in this combined curve at receiver tube diameters between 50 and 90 mm. If we were to plot additional combined curves for other temperatures, a similarly shaped maximum would exist for each curve. This flat, maximum efficiency region shifts towards larger receiver tube diameters as the temperature of operation becomes lower. Since the efficiency is not very sensitive to the receiver tube diameter, the precise tube size for a particular system will likely be chosen, or at least modified, by such considerations as stagnation temperature, thermal inertia, adjustment ease, and collector cost.

Fig. 12 Nodal temperatures and the type and magnitude of heat flows (W/m) in the collector for the baseline configuration.

Experimental Collectors

We are currently constructing the first two experimental collectors and their associated system components. Experiments conducted with these collectors will enable us to verify the accuracy of our codes. Two side-by-side, identical systems are being built to compare parameters under the same operating conditions. A larger number of materials and geometries can be tried in a shorter time with this dual system.

In addition to verifying our codes, building these collectors will enable us to uncover practical difficulties in constructing and operating the hardware. These units will also allow us to measure quantities which we have had some difficulty calculating, such as wind loading effects and motion of the outer cylinder.

The two collectors are 1 m in diameter and about 4 m long. The receiver tubes are 42 mm in diameter. Surrounding the tube is a thin plastic cylinder with a

Fig. 13. Collector performance as a function of fluid temperature for the baseline configuration.

Fig. 14. Collector performance as a function of receiver tube diameter for the baseline configuration at 170°C.

gap of approximately 9 mm between it and the receiver tube. This cylinder is positioned by a number of plastic, washer-shaped spacers. A small pressure difference is maintained across the film to aid in keeping a cylindrical shape. Figure 8 schematically shows that the outer cylinder ends are merely gathered around the receiver tube. We currently think this is the least expensive construction method. However, for these experimental units we are using sturdy, flat end plates so the angular tilt and positioning of the outer cylinder can be precisely controlled.

There are many plastics available as thin films which are being investigated for suitability as the outer cylinder material. These films have a wide range of pertinent properties and characteristics

such as optical transmission, strength, weatherability, ease of joining and metalizing, and cost. We are determining acceptable ranges of these properties and characteristics and studying their economic implications. The first outer cylinders are fabricated from 0.05-mm-thick polyester, clear on the upper portion and aluminized on the lower. The joints between the upper and lower halves are made with thermoplastic heat sealing tape.

Since the inner plastic jacket and the spacers surrounding the receiver tube could see stagnation temperatures, we are making these parts from Teflon. The jacket thickness is 0.025 mm, and the spacer material thickness is 3.2 mm. The receiver tubes we have are steel with a 1.8-mm wall thickness, and their selective surface is black chrome over nickel.

DESIGN OF THE TOKAMAK FUSION TEST REACTOR CALORIMETER

Introduction

LLL and LBL are working with the Princeton Plasma Physics Laboratory on the Tokamak Fusion Test Reactor (TFTR). This reactor, which should be operational at Princeton by the early 1980's, will combine all the physics conditions required for the net production of fusion energy.

Our part of the TFTR project is the design and construction of a neutral-beam injection system. Design is nearing completion on a prototype which will be tested at Berkeley. The neutral-beam systems will inject high-energy, neutral deuterium atoms into the plasma, raising its temperature to 50-100 million °C. Figure 15 is an elevation view of a neutral-beam injection system.⁹ It consists of three ion sources, three neutralizer ducts where high energy D^+ ions charge exchange to D^0 neutrals, a vacuum vessel,¹⁰ an ion deflection magnet to deflect the remaining ions from the beam, and an ion dump to absorb the energy from these deflected ions. It also has a calorimeter for source testing and large cryopanel to provide the fast pumping speeds required to avoid charge exchange outside the neutralizer.¹¹ It is divided into three pumping chambers by baffles running from the top of the vacuum chamber to the bottom. This arrangement allows the excess deuterium gas to be differentially pumped with the lowest pressure in the chamber closest to the torus.

The three ion sources are aimed through a duct into the torus. The sources are tested by lowering a calorimeter into the beam line. It absorbs the energy contained in the neutral and ion beams and

provides a means of measuring several beam characteristics. By measuring the equilibrium temperature at various locations on the absorber plate, the beam profile can be determined. The divergence angles of the beam (1/e point) can then be calculated, as well as the maximum power density. If the calorimeter is used with the ion deflection magnet on, the energy contained only in the neutrals can be measured. If the ion magnet is off, then the power contained in both the ions and neutrals will be measured.

Since the source will deliver an extremely powerful beam, the calorimeter must be able to handle large amounts of power. It must absorb 7.2 MW of 120 keV neutral and ion beam power generated by a 10×40 -cm source with a pulse width of 0.5 s. The normal pulse rate is once every 5 min. For source conditioning, a pulse rate of one shot every minute may be used.

The calorimeter must be able to move into the beam line for source testing and out of the way for normal operation. It will be a high maintenance item and must be easily removable from the vacuum vessel. A 1.32-m port is provided in the vacuum vessel for the calorimeter assembly. Since the fuel used in the TFTR will be a deuterium-tritium mixture, remote handling capabilities must be incorporated into the design. The life of the calorimeter is specified as 5000 shots or about 2 years of operation.

The calorimeter we designed consists of: (1) the energy-absorbing plates, and (2) the retraction and support part. Figure 16 shows its overall appearance.

Fig. 15. TFTR neutral-beam injection system.

Considerations in Designing the Energy-Absorbing Plate

The basic energy absorber for high energy beams of short pulse duration consists of flat plates inclined at some angle to reduce the power density loading. Copper was chosen as the absorber plate material due to its attractive thermophysical properties, its relatively low cost, and its high thermal fatigue life.¹² One of the disadvantages of using copper is its high sputtering rate.

Using the solution of the thermal diffusion equation for heat flux in one side of a copper plate and an insulated back side, the estimated front surface temperature rise for a 2.0 kW/cm^2 maximum power density input is about 420°C . The maximum equilibrium temperature of the plate will be about 150°C for the same power input. If a maximum limit of 800°C temperature rise of the front surface is assumed, the beam on-time can increase to about 1.6 s.

To cool the copper plates off in 1 min, 6.4-mm i.d. cooling pipes are brazed to the back of the copper plates 60 mm apart. Assuming the film coefficient in the pipes should not decrease to less than $0.2 \text{ cal/cm}^2 \cdot \text{s} \cdot ^\circ\text{C}$ and using the Dittus-Boelter

equation, a flow of 1.51 kg/s is required for each vee.¹³ At this flow rate, the water in the pipes will be turbulent. The pressure drop across the calorimeter system is about $5.5 \times 10^4 \text{ Pa}$.

Retraction and Support Design

The TFTR calorimeter consists of three side-by-side vees. Figure 17 shows a single vee subassembly. Each vee is two rectangular OFHC copper plates at an angle of about 6° with the beam centerline. This angle decreases the maximum power density from 18 kW/cm^2 down to about 2.0 kW/cm^2 . The plates will expand about 5 mm in the vertical and horizontal direction. The back of the vees are prevented from opening up by through bolts and bowing of both plates is prevented by a stiff strongback. Belleville washers are used to hold the plates together, but keep the bolts from experiencing large stress fluctuations due to the thermal expansion of the copper. The three strongbacks are bolted together to further increase the stiffness and to maintain center-line spacing.

The design of the system allows easy removal of one of the vees with its water manifolds, back

Fig. 16. Overall view of TFTR calorimeter.

strongback, and electrical diagnostics lines. This is accomplished by pulling three ball detent pins at the bottom-front of the vees, removing the bottom plate, disconnecting two water lines, pulling two ball detent pins at the top of the vees, and disconnecting the electrical lines from the cover plate. The single vee assembly can then be lowered away from the calorimeter system (Fig. 17). The use of ball detent pins and easily accessible bolts should facilitate remote disassembly and assembly of the system. All bolts will be fastened using helicoil inserts.

The three vees are attached to a middle plate, which is attached to four 10.2-mm pipes. These pipes serve a dual purpose. They carry the weight of the vees and also are used to supply inlet and exit water. One of the pipes will carry in all the water for the three vees. Each of the three remaining pipes carries out water from a single vee. This allows water temperature and flow measurements to be made outside the vacuum vessel. Alignment of the assembly is facilitated by linear bearings held in the cover plate.

Outside the vacuum vessel, the four pipes are bolted to a carriage. The assembly is driven up and down in 3 min by a motor and a screw. A blowout line is provided to remove the water in case of a pumping failure. This is necessary due to the proximity of the large cryopumps at -195°C . Side shields also protect the sides of the calorimeter from looking directly at these cold surfaces.

All the elements were analyzed for earthquake conditions at 0.5 g in any direction. This is required to build the prototype at LBL. The estimated total weight of the system including the cover plate is 3600 kg.

Calorimeter Diagnostics

The primary diagnostics for the calorimeter are an array of thermistors mounted about 12 mm into the copper from the back. This is the location of the calculated equilibrium temperature for a 0.5-s pulse. The thermistors are rated to 300°C and have a time constant of less than 1 s. A test is planned to determine the accuracy and repeatability of this value. The output of the thermistors can be digitized and displayed on a monitor as well as printed out.¹⁴ Integration of the profile can be accomplished by a computer and the total beam power calculated. The closeness of the water pipes to the thermistors may cause a "smearing" of the profile if the water is running during the data taking. For this reason, the water will normally be turned off for accurate beam profile measurement. By measuring the mass flow and temperature rise of the water, the total power into the calorimeter can be determined. These measurements are available for each vee of the calorimeter.

Limit switches are provided in the motor lift mechanism and by other microswitches. These are interlocked to ensure the beam is not fired unless the calorimeter is either completely up or down.

Conclusion

A full-scale test of the calorimeter cannot be accomplished until the neutral-beam injection system prototype is completed, which should occur within the next year and a half. However, the design of the TFTR calorimeter has considered all the required specifications and should be successful.

Fig. 17. View of a single vee subassembly.

SINGLE-SPINDLE ASPHERIC FIGURING WITH FLEXIBLE PADS

The Laser Program and other Laboratory programs are increasingly calling on the Optics Shop to make high-precision aspheric surfaces. These are normally produced a few at a time at small, commercial optics shops. Since these shops are unable to afford computer-controlled figuring machines,¹⁵ we have developed for them a simple, inexpensive method of wear prediction that makes use of a hand calculator. This calculation-directed

figuring bridges the gap between the traditional guesstimation and more exotic machines. It enables an optician to gain insight into his process, proceed with more confidence, and establish a program or pattern library, narrowing the range of variables to which his attention must be directed.

Many aspherizing operations involve sub-diameter circular or ring grinding pads and polishing pads oscillated across a workpiece

rotating on a spindle as shown in Fig. 18. We present a proof that under the conditions to be described, the pressure and relative velocity at any point under the pad are identical to those under the pad center when averaged over a number of cycles. This eliminates the requirement for computing these values at each point for wear calculations, reducing the calculations from a class requiring a large computer to one which can be handled by a programmed hand calculator such as the SR-52. When stroke, pad, and workpiece parameters (load, dimensions, rpm), and a glass wear constant are keyed in, the calculator will compute pad center velocities, zone coverage factors, and output wear profiles successively for a number of selected zones. This enables the optician to predict the effect of his tooling and pre-determine his figuring period.

Although calculus is required to demonstrate the principles below, only trigonometry is required to program the calculator and even less mathematical background is required to operate programs already available. Potential users unfamiliar with the mathematics need only understand the restrictions specified and translate these into machine operation.

Assume a subdiameter pad (Fig. 18) so constructed that at rest it will be supported by uniform pressure at any position it will occupy during a stroke (i.e., rubber-backed shimstock grinding pads, or pitch-impregnated felt, etc.). Assume it is stroked such that no part extends over the edge of the workpiece and that when the frictionless ball socket is centered above c , the velocity v_o on the outward stroke equals the velocity v_i on the inward stroke, and that translational acceleration effects can be neglected.

Fig. 18. Single spindle machine schematic showing a sub-diameter polishing pad oscillating on a rotating workpiece.

The interface drag, reacted by the pin located above c, will result in a linear pressure gradient whose components lie in the direction of the components of relative motion between the parts and are proportional to them. The components of velocity along radius r are only the projections of v_0 and v_1 on r. Thus at point a the pressure due to the radial velocity is that at c minus an increment which will be added on the inward stroke averaging to that at c. Similarly, although the tangential velocity component may never go to zero at c the pressure at a due to this component will be that at c plus an increment that is subtracted at the corresponding point b equidistant from the radius joining workpiece and pad centers. Thus the pressure affecting wear averages to that at c over a large number of cycles.

Now consider the relative velocity between parts at point a (Fig. 19). Considering only the component of relative velocity tangential with respect to the pad we see that this reduces to

$$V_{la} = r(\omega - \Omega) - v_r \sin\theta + v_t \cos\theta$$

Since the drag induced linear pressure gradients must lie in the direction of the radial and tangential components of the relative velocity v_r and v_t and be proportional to them, there will be a constant K such that the pressure at a is

$$P_a \approx P_c + K_{v_I} r \cos \theta + K_{v_I} r \sin \theta$$

Assuming a constant coefficient of friction, the local drag will be proportional to this pressure and the integration of the product of this pressure and

Fig. 19. Subdiameter pad showing relative velocities v_r and v_t of the center in the radial and tangential directions and at a typical point a, r displaced from the center.

the tangential velocity over the area is proportional to the power expended by the workpiece in the rotation of the figuring pad. Since the pin is by definition frictionless, this must integrate to zero. Of the nine terms in the product, six are sine, cosine, or sine-cosine products integrating to zero in 2π . The only remaining terms are

$$P_a V_{ta} = P_c r (w - \Omega) - K_v v_r \sin^2 \theta + K_v v_t \cos^2 \theta + \dots$$

Since the sine-squared and cosine-squared terms have the same factors and opposite signs, they cancel requiring $w \approx \Omega$. Thus the pad must rotate in angular synchronization with the workpiece and move in pure translation about the workpiece under these restrictions. Hence, only the center pressure and relative velocity need be calculated. Since the rate of material removal follows a relationship

$$\frac{dh}{dt} = cPV = cP \frac{ds}{dt},$$

the amount of material removed in zone R_z is proportional to the product of the center pressure and relative velocity, a coverage factor θ/π , and a dwell time in the vicinity of c given by the difference of two arcsines for a sinusoidal stroke.

For steep aspherics, pressure and path increment ds are functions of pin radial position. The translational accelerations neglected in the analysis are always directed toward stroke center, and the moments may be eliminated by locating the ball at the pad center of mass or reduced at will since their magnitude varies as the square of the crank rpm. The implied restriction that the pad not extend over

the workpiece edge is not severe since opticians seldom work over an edge in aspherizing due to roll-off. Overedge stroking is normally a convexing stroke employed in earlier spherical preparation. Filler material supplies a solution if overedge stroking is required.

These wear factors are easily calculated on a hand calculator, using one unit with a printer attachment to provide looping capability. This combination costs less than \$350. Wear values for up to 20 locations can be calculated in 1/2 h unattended for a single set of tool, speed, stroke parameters, etc. A simple external calculation provides the time factor needed to scale the wear pattern for a given set of parameters. This calculator with 20 memory locations and 224-step programming has insufficient storage to complete scaling, so less than 100 multiplications and subtractions are required external to the program for an aspheric requiring six tools. We have extended the program to stroked ring tools and to stroked star laps composed of ring tool arc segments, the arc lengths being proportional to the stroking time predicted for an equivalent solid ring.

For about \$4,000, a calculator with 1000-step capability and 200 memory locations can be purchased and linked to a video to provide almost instantaneous wear pattern display. The video can show the optician the effects of each tool as a set of parameters are keyed into the calculator.

We have given hand calculator programs to the people doing work for us at small shops. Having established the basic theory, we are continuing our research to learn more about how and when to use it.

SPRINGS OF COMPLEX SHAPE

In any Laboratory program where mechanical and structural design is involved, a mechanical engineer may need to utilize springs of complex shape. This article presents a practical approach for the design of springs of complex shape. These simplified methods of calculating deflections and stresses minimize design time and production cost for such springs. Complex springs are defined here as those which consist of basic circular elements, straight elements connected by relatively small radii, as well as other complex shapes involving some combinations of straight and curved portions. In many practical cases, the curved portions of the springs can be approximated by equivalent circular arcs, making the development of closed-form solutions quite feasible.

The great majority of complex-shaped springs are made of essentially flat stock. The applications of such springs are quite common in instrument and small mechanism design throughout the Laboratory and industry. However, the analyses and practical design formulas intended for complex springs apply equally well to other cross-sectional geometry provided the correct area properties are used. Therefore, this article on complex springs has a broader significance and extends to various other spring-like structural members.

Because of the large diversity in geometrical shapes and manner of loading and support, in-plane and out-of-plane analysis is reviewed for the benefit of the practicing engineer. The analytical methods selected here are based on the principles of

Castigliano, first disclosed over 100 years ago. These principles hold for statically determinate and indeterminate structural members and are applicable to the analysis of complex springs within the elastic limit of the material. In engineering, this is the most common occurrence, particularly where a relatively conservative design is mandatory.

The Castigliano theory utilizes the concept of the total elastic-strain energy stored in a particular structure under a given system of external forces. The term "force" in this instance has a generalized meaning and may denote a concentrated load, uniform pressure, localized bending couple, or a twisting moment. Similarly, generalized "displacement" caused by the external loading may refer to the deflection, bending slope, or angle of twist. Once the force and displacement for a particular complex shape are defined, the spring designer can calculate the ratio of these two quantities to get the so-called "spring constant," which is used universally in the process of spring design and application.

In principle, the amount of the total elastic-strain energy stored in a complex spring or a similar structural component depends on a three-dimensional system of stresses. Such a system may include bending, shear, torsion, and normal stresses; and the resulting analytical expressions that combine these effects must of necessity be rather complicated. Fortunately for the spring designer dealing with relatively flexible load-carrying components, the displacement effects of direct and shear stresses can be ignored. Such a simplification, which leads to a sizable reduction in the amount of detailed mathematical work, is admissible on the grounds of sufficient accuracy and practicality.

As a consequence of the above simplification, all complex-shaped springs responding in the plane of curvature can be analyzed on the basis of strain energy caused by bending stresses alone. The response transverse to the plane of curvature, however, requires that the stored, elastic-strain energy be the sum of the bending and twisting effects. Furthermore, the treatment of the various spring shapes in the plane and out of the plane of curvature will be based on the following simplifying assumptions:

- Cross section is uniform.
- Depth of cross section is small compared to radius of curvature.
- Strains are elastic.
- Displacements and changes in curvature are small.
- External forces are applied gradually.

Where load repetition is anticipated, prediction of the maximum local stress is necessary to establish

a rational margin of safety and length of service. Other considerations are the effects of specific production techniques on the accuracy of spring constants. In all these cases, particularly where mass production is planned, advanced knowledge of spring response can minimize the costs involved in making sample springs, in development testing, and in quality control. The bulk of this material is based on Refs. 16 and 17 with supplementary data being taken from Ref. 18.

In-Plane Analysis

The basic principles of stress and deflection analysis of a complex-shaped spring will be briefly reviewed with reference to Fig. 20a. In this illustration, the spring consists of a straight portion AB and a curved part BC that is fixed as to slope and deflection at C. In the majority of practical situations, the curved end, such as BC, can be approximated by a circular curvature of radius R as shown in Fig. 20a. This is convenient from the manufacturing point of view and very helpful for establishing a workable mathematical model of spring behavior.

Let us suppose that the design requirements involve downward deflections at points A and B and the maximum bending stress at C. Figure 20a shows clearly that two bending-moment expressions are needed for the analysis. The bending moment acting at any point along the straight portion AB is:

$$M_1 = Px \quad (1)$$

where M_1 is the bending moment, P is the vertical load, and x is an arbitrary distance. Similarly, the bending moment at any point defined by θ along the curved portion BC is:

$$M_2 = P(L + R \sin \theta) \quad (2)$$

where M_2 is the bending moment, L is the length of the straight portion, R is the radius of curvature, and θ is the angle at which the forces are considered.

Notation in Fig. 20a involves external forces P and $\bar{P}$. It is assumed here that P denotes the real external load on the spring that is responsible for stresses and deformations. On the other hand, $\bar{P}$ is a fictitious force of an infinitely small value that is applied at a point and in the direction of the required displacement. The mathematical concept of $\bar{P}$ represents a very powerful design tool. However, the absolute value of $\bar{P}$ has no material influence on the magnitudes of stresses or deformations of the spring.

The displacement of point A in the direction of force P (as shown in Fig. 20a) can be obtained from the principle of Castigliano as follows:

Fig. 20. Complex-shaped springs: (a) under point loading, and (b) half-circle spring with end constraint.

$$Y = \frac{1}{EI} \int_0^L M_1 \frac{\partial M_1}{\partial P} dx + \frac{1}{EI} \int_0^\phi M_2 \frac{\partial M_2}{\partial P} R d\theta, \quad (3)$$

where Y is the vertical displacement, E is the modulus of elasticity, I is the moment of inertia, and d is an arbitrary distance. From Eqs. (1) and (2), it follows that

$$\frac{\partial M_1}{\partial P} = x \quad (4)$$

and

$$\frac{\partial M_2}{\partial P} = L + R \sin \theta. \quad (5)$$

Hence, utilizing Eqs. (1) through (5), the deflection under load P becomes:

$$Y = \frac{PR^3}{12EI} [4k^2(k + 3\phi) + 24k(1 - \cos \phi) + 6\phi - 3 \sin 2\phi]. \quad (6)$$

where k equals L/R (straight length to radius ratio) and ϕ is the angle subtended by a curved spring.

The above design formula contains two geometrical parameters that control the shape of the spring and its constant. When the subtended angle ϕ becomes very small, the spring reduces to a simple cantilever beam. On the other hand, when $k = L/R$ tends to zero, the spring transforms into an arched cantilever. For instance, at $k = 0$ and $\phi = \pi/2$, Eq. (6) reduces to a standard formula for the deflection in a quarter-circle bend.

To illustrate the use of the fictitious-force method, Eq. (2) should be modified as follows:

$$M_2 = P(L + R \sin \theta) + \bar{P}R \sin \theta. \quad (7)$$

Because the displacement is required at a given point, such as B , the expression for the bending moment M_1 is independent of $\bar{P}$, making $\partial M_1 / \partial \bar{P} = 0$. Hence, using Eq. (3), we obtain:

$$Y = \frac{R}{EI} \int_0^\phi M_2 \frac{\partial M_2}{\partial \bar{P}} d\theta. \quad (8)$$

The partial derivative needed for the solution of Eq. (8) is found directly from Eq. (7):

$$\frac{\partial M_2}{\partial \bar{P}} = R \sin \theta. \quad (9)$$

Combining Eqs. (7) through (9), integrating, and making $\bar{P} = 0$, we get

$$Y = \frac{PR^3}{12EI} [12k(1 - \cos \phi) + 6\phi - 3 \sin 2\phi]. \quad (10)$$

The maximum bending stress for the spring (see Fig. 20a) follows from Eq. (2) and the elementary beam theory and is given by the following expression:

$$S_b = \frac{P(L + R \sin \phi) c}{I}, \quad (11)$$

where S_b is the bending stress and c is the distance from the neutral axis.

One of the problems in analyzing curved portions of complex-shaped springs is developing the appropriate equations for the bending moments. For instance, for a horizontal displacement of the free end of the spring, caused by uniform loading and depicted in Fig. 21, the elementary bending moment about a section defined by θ is:

$$dM_q = qR^2 (\cos \epsilon - \cos \theta) d\epsilon, \quad (12)$$

where M_q is the bending moment under uniform load, q is a uniform load, and ϵ is an auxiliary angle.

Fig. 21. Curved element under uniform, transverse loading.

Integrating the above expression between the limits of 0 and θ gives:

$$M_q = qR^2 (\sin \theta - \theta \cos \theta). \quad (13)$$

To find the horizontal displacement at the free end, fictitious force $\bar{H}$ is applied in the direction of the required displacement. Hence, the relevant bending moment at a section defined by θ becomes:

$$M = qR^2 (\sin \theta - \theta \cos \theta) - \bar{H}R \sin \theta. \quad (14)$$

Other steps in deriving the deflection formula are similar to those that apply to the problem illustrated in Fig. 20a.

In some designs, various portions of the springs may be subject to boundary constraints. For instance, consider one-half of a flat U-spring that is fixed at the support as to slope and deflection and is fixed at the free end as to slope (see Fig. 20b). The unknown fixing moment M_f is constraining the loaded end as to slope, allowing a finite amount of downward deflection at the same time. In terms of the previously adopted notation, the bending moment equations for the straight and curved portions are:

$$M_1 = M_f - Px \quad (15)$$

and

$$M_2 = M_f - PR(k + \sin \theta). \quad (16)$$

Because the slope at the loaded end is assumed to be zero, the applicable Castigliano principle can be expressed as follows:

$$\int_0^L M_1 \frac{\partial M_1}{\partial M_f} dx + \int_0^{\pi/2} M_2 \frac{\partial M_2}{\partial M_f} R d\theta = 0. \quad (17)$$

Hence, by using Eqs. (15) through (17), integrating, and solving for the unknown moment, we obtain:

$$M_f = PR \left(\frac{k^2 + \pi k + 2}{\pi + 2k} \right). \quad (18)$$

Note that for $k = 0$, the fixing moment found from Eq. (18) is the same as that under the point load in a diametrically loaded circular ring. The bending moments now can be calculated at each point of the spring by using Eqs. (15), (16), and (18).

Out-of-Plane Analysis

Load-deflection relations for forces acting normal to the plane of curvature of the spring require the inclusion of twisting moments in addition to bending moments. The same Castigliano principles still apply, but it is necessary to define the following general rules for the transverse deflection, the angle of twist, and the bending slope at any point of the curved element:

$$Y = \frac{R}{EI} \int_{\alpha}^{\phi} M \frac{\partial M}{\partial \bar{P}} d\theta + \frac{R}{GK} \int_{\alpha}^{\phi} T \frac{\partial T}{\partial \bar{P}} d\theta, \quad (19)$$

where α is the angle at which displacement is required, M is the bending moment, G is the modulus of rigidity, K is the torsional shape factor, and T is the twisting moment.

$$\eta = \frac{R}{EI} \int_{\alpha}^{\phi} M \frac{\partial M}{\partial \bar{T}_0} d\theta + \frac{R}{GK} \int_{\alpha}^{\phi} T \frac{\partial T}{\partial \bar{T}_0} d\theta, \quad (20)$$

where η is the angle of twist and T_0 is the twisting moment.

$$\psi = -dY/Rd\alpha, \quad (21)$$

where ψ is the bending slope.

Notation applicable to Eqs. (19) through (21) is shown in Fig. 22. Here a curved element, subtending angle ϕ , is acted on by a point load P at the free end. The two fictitious quantities $\bar{P}$ and $\bar{T}_0$ are applied at a point defined by α . To calculate the

Fig. 22. Curved element under out-of-plane loading.

bending and twisting moments at point A, we obtain the effect of $\bar{T}_0$ by resolving the vector $\bar{T}_0$ into the normal and tangential components shown in Fig. 22. The tangential component provides the twist of the curved element about the central axis, while the normal component subjects the element to pure bending. The effects of P and $\bar{P}$ can be obtained directly from the geometry involving R , α , and θ . This procedure leads to the following bending and twisting moments for the case depicted in Fig. 22:

$$M = -PR \sin \theta - \bar{P}R \sin (\theta - \alpha) + \bar{T}_0 \sin (\theta - \alpha) \quad (22)$$

and

$$T = PR (1 - \cos \theta) + \bar{P}R [1 - \cos (\theta - \alpha)] + \bar{T}_0 \cos (\theta - \alpha). \quad (23)$$

The above expressions simplify considerably when displacements at the point of load application are required. For instance, considering out-of-plane deflection at $\alpha = 0$, Eqs. (22) and (23) become:

$$M = -PR \sin \theta \quad (24)$$

and

$$T = RP (1 - \cos \theta). \quad (25)$$

The partial derivatives needed for the solution of Eq. (19) are:

$$\frac{\partial M}{\partial P} = \frac{\partial M}{\partial \bar{P}} = -R \sin \theta \quad (26)$$

and

$$\frac{\partial T}{\partial P} = \frac{\partial T}{\partial \bar{P}} = R (1 - \cos \theta). \quad (27)$$

Hence, using Eqs. (24) through (27) together with Eq. (19), we obtain:

$$Y = \frac{PR^3}{4EI} [2\phi - \sin 2\phi + \lambda (6\phi + \sin 2\phi - 8 \sin \phi)].$$

The parameter λ , entering Eq. (28), defines the ratio of flexural to torsional rigidity of a spring element. Clearly, in the case of out-of-plane loading, λ can influence the magnitude of a particular displacement. Other studies indicate that λ can also affect the direction of the displacement. The torsional shape factor K is largely responsible for the variations in λ . The derivation of K values can be rather complicated, because the distribution of shear stresses on a cross section of a more arbitrary geometry is nonlinear and the section does not remain plane under twist. Relatively narrow sections that are bent into channels or open curved configurations can be analyzed as the sum of K values of individual rectangular areas. In practice, complex-shaped springs have solid, rectangular cross sections for which the torsional shape factor is well defined.

The analysis of out-of-plane response of a complex-shaped spring should be approached with due caution. For instance, Eqs. (19) through (23) clearly indicate how the derivation can rapidly become complex. In practice, however, bracketing values of stresses and displacements alone may be sufficient to establish a particular design criterion without going into any tedious and error-prone derivations. For instance, Eq. (28) represents one such bracketing condition. In this instance the deflection formula for any fixed value of ϕ is easy to obtain from Eq. (28).

Summary

Typical examples of various spring shapes and types of loading are illustrated in Figs. 23 and 24. For detailed formulas referring to these configurations and other possible spring shapes, see Refs. 16 and 17.

Fig. 23. Several types of complex-shaped springs: (a) curved-end spring, (b) clip spring, (c) S-spring, (d) anchor spring, (e) sharp-bend cantilever spring, (f) frame-type spring, and (g) double U-spring. For symbol definition, see Ref. 16.

Fig. 24. Transversely loaded arcuate springs. For symbol definition, see Ref. 16.

TECHNICAL NOTES

Degassing a Large LHe Cryopump (B. S. Denby, T. H. Batzer, and W. R. Call, *Magnetic Fusion Engineering Division*)

A method has been developed and successfully tested to degas a large liquid helium cryopump. This scheme precludes the normally excessive pressure

rise during the degassing cycle when the degassing rate exceeds the system's external pumping capabilities.

A small appendage pump is installed close to the main cryopump (see Fig. 25). The appendage pump absorbs all the gas, with no catastrophic pressure rise, as it is desorbed from the main cryopump. The appendage pump can then be isolated from the main vacuum system and degassed at high pressure. We had 15 to 20×10^{-3} l of H_2 pumped on a 1.25-m^2 panel. During the degassing cycle the system pressure never rose above 1×10^{-4} Torr.

This scheme is unique and a very useful tool for large vacuum systems of future fusion machines that contain cryopump panels as well as cryogenic magnets. This will allow the degassing of cryopumps without affecting the temperature equilibrium of cryogenic magnets.

Interior Surface Examination By Rod Lenses (D. E. Lord, G. W. Carter, and R. R. Petrin, *Materials Engineering Division*)

We have been able to observe dynamic events in previously inaccessible volumes by coupling a 1-mm-wide rod lens with a low-light video system. The rod lens can provide access to difficult volumes because of its small size. (It ranges from 1 mm in diameter and 19 cm long to 2.5 cm in diameter and 10 m or more long.) Modern rod lenses provide brilliant images by keeping internal light losses to a minimum. They are solid glass lenses arranged in different configurations. Most such lenses also have a built-in bundle of glass fibers for the purpose of transmitting light to the object. They may also be used with other illumination. The ability of rod lenses to withstand both high pressures (~ 14 MPa) and temperatures ($\sim 100^\circ\text{C}$) makes them applicable to many engineering uses.

Though a great improvement over the old "borescopes," these lenses still do not transmit enough light when used with sizable volumes to allow observation of dynamic events. We used a low-light-level television camera of 40 000 light gain to provide the extra sensitivity needed for dynamic conditions while retaining the geometric access advantages of the rod lens. The video camera output can be handled in a number of ways depending on the analysis desired. For instance, we can run the camera output directly into the scanning microdensitometer for electronic enhancement and analysis. Conventional video recording plus single-frame playback capability complete the system.

We first used this system in the fall of 1976 to observe a two-phase flow system. We directly observed

Fig. 25. Degassing apparatus.

turbine drag-bar type flow measuring device. The rod lens system showed lack of mixing as the reason for the malfunction of test section instrumentation. At this time, we were also involved in instrumenting the one-fifth-scale Mark I Boiling Water Reactor pressure suppression system for the Nuclear Regulatory Commission (NRC). Even though the instrumentation plan called for high-speed cameras and large access windows, we felt it was advantageous to use the rod lens system due to its instant replay capability and ability to immediately single frame the results. The access holes required were standard 6-mm tubing fittings in this ~2-m-diameter tank. A three-camera grouping was used and test results were excellent. At the request of NRC, we also demonstrated the rod lens system and its possible applications to reactor safety monitor-

ing by building a pipe test section and observing various flow conditions within it, recording the results on videotape. (This videotape is available at the Lab and is also documented in a published report.¹⁹)

More recently, we have used the rod lens-video system with great success in observing combustion processes and fabrication processes such as welding. The rod lens alone provides a very quick inspection method since it can be used in real time. If desired, videotapes can be made to provide a permanent record. Figure 26 shows an interior view of a vacuum bellows. Figure 26b is a closeup of the interior wall weld, indicating a good weld with full penetration.

The system has proved to be an extremely versatile tool, with diverse applications, and provides a new dimension in engineering analysis.

Thermal Study of UC-609 Shipping Package (G. C. Schaack and D. A. Schauer, *Nuclear Explosives Engineering Division*)

A thermal study was made for the UC-609 Shipping Package, Fig. 27, a thermal- and shock-insulated shipping container for transporting tritium gas storage vessels. This LLL-designed shipping container must conform to ERDA specifications. In particular, it must meet ERDA-specified pressure requirements for tritium gas quantities up to 150 g (48 W of heat generation). This thermal study determined the expected maximum temperature of the tritium gas in the storage vessel, or if it should leak, in the containment vessel, due to the internal heat generation and imposed environmental storage conditions. These temperatures are essential in predicting the highest possible gas pressure conditions for checking the design.

Thermal Model

The worst possible storage environment was approximated by constructing a thermal model that imposed desert-like boundary conditions on a vertical cylindrical container. To simulate a day-night environment, a sinusoidal ambient air temperature variation was used, with a maximum temperature of 54°C at 3 p.m. and a minimum temperature of 26.7°C at 3 a.m. A solar radiation flux, which varied in intensity and location according to the hour of the day, was imposed on the outer surface. This required a three-dimensional thermal model²⁰ to be used from the outer surface through the insulating material. It was not possible to use a three-dimensional grid beyond the insulating material and

(a)

(b)

Fig. 26. (a) Interior view of a vacuum bellows, showing infinite focus of rod lens. Each weld is 3 mm wide. Quarter shows the lens' depth of field. (b) Closeup made with a 90° angle lens looking directly at interior wall weld. The weld is good with full penetration.

Fig. 27. UC-609 shipping package.

stay within the zone limitation of the computer program used for the thermal calculations (TRUMP²¹). Since the aluminum honeycomb construction tends to smooth out the temperature variations on the outside surface, a two-dimensional grid was prepared²⁰ for the inner portion and integrated with the three-dimensional grid.

The calculated daily temperature response of the storage and containment vessel due to 3 consecutive days of the imposed environmental storage conditions, and 48 W (150 g of tritium) of internal heat

generation is shown in Fig. 28. Calculations show that if a leak were to develop in the storage vessel, the heat generation would be uniformly distributed throughout the inner volume of the containment vessel. Therefore, the temperature response of the storage vessel will be essentially the same as that of the containment vessel (Fig. 28). These calculations confirm that temperatures experienced during transportation are moderate, even if a leak occurs. For every case, the internal gas pressure buildup is tolerable and meets ERDA requirements.

Fig. 28. Daily temperature response of UC-609 package after 3 consecutive days exposure to desert environment. Tritium gas (150 g) contained in storage vessel.

PUBLICATION ABSTRACTS

W. Lai, E. K. Collins, E. W. McCauley, and J. H. Pitts, Mark I 1/5-Scale Boiling Water Reactor Pressure Suppression Experiment Quick-Look Report, Lawrence Livermore Laboratory, Rept. UCRL-17446-1 through 17446-7 (1977).

A 1/5-scale facility of a Mark I boiling water reactor pressure suppression system is described and selected results from 27 air tests documented. (Each report describes a specific grouping of tests.) The data provides an excellent base; 1) for comparison with analytical studies of suppression system dynamics and 2) to assess margins-of-safety in full scale boiling water reactor plants. Vertical loads observed following a simulated loss-of-coolant accident are of prime interest.

G. R. Dittbenner and H. S. Freynik, Jr., Fabrication Procedure for the Adiabatic Surface Thermometer, Lawrence Livermore Laboratory, Rept. UCID-17484 (1977).

We have developed a special vacuum version of an adiabatic surface thermometer for automatically

controlling a critical temperature-time cycle of a production vacuum-brazing process. This thermometer touches the surface and uses a differential thermocouple and heater to measure surface temperature without heat flow, thereby minimizing large errors caused by conduction losses common to conventional spring-loaded thermocouples. Some important advantages of our design for vacuum chamber use are: continuous operation at 850°C, intermittent operation to 1300°C, no outgassing or plating-off at high temperature in a vacuum system, and an integral connector that permits easy replacement of broken probes.

M. R. Posehn, Modal Analysis of the NRC Pressure Suppression Experimental Facility, Lawrence Livermore Laboratory, Rept. UCID-17494 (1977).

The 1/5th scale model Mark I pressure suppression facility was experimentally analyzed in order to determine its fundamental modes of vibration. The results of the modal analysis revealed seven apparent modes with frequencies below 100 Hz. In this report each mode is characterized by a description of the motion, the natural frequency, and the response amplitude. The results indicate that the response of the torus to an impulsive load in the vertical direction is dominated by two modes at 12.2 Hz and 59.8 Hz.

E. W. McCauley, W. Lai, J. Meier, and W. Stein, Mark I 1/5-Scale Boiling Water Reactor Pressure Suppression Experiment Summary of Effects Due to Vent Line Orifice Variations—Air Test Series, Lawrence Livermore Laboratory, Rept. UCID-17539 (1977).

W. M. Shay, The Effect of a Bridge Balance Network on a Semiconductor Pressure Transducer, Lawrence Livermore Laboratory, Rept. UCID-17547 (1977).

R. T. Peterson, Safety Analysis Report for Packaging: Neutron Shipping Cask, Model 41, Lawrence Livermore Laboratory, Rept. UCRL-52232 (1977).

This Safety Analysis Report for Packaging demonstrates that the neutron shipping cask can safely transport, in solid or powder form, all isotopes of uranium, plutonium, americium, curium, berkelium, californium, einsteinium, and fermium. The report describes the cask and its contents. It also evaluates transport conditions, structural parameters (e.g., load resistance, pressure and impact effects, lifting and tiedown devices), and

shielding. Finally, it discusses compliance with Chapter 0529 of the Energy Research & Development Administration Manual, *Safety Standards for the Packaging of Fissile and Other Radioactive Materials*.

D. Bernreuter, F. Tokarz, L. Wight, P. Smith, J. Wells, and R. Barlow, *An Evaluation and Assessment of Nuclear Power Plant Seismic Methodology*, Lawrence Livermore Laboratory, Rept. UCRL-52236 (1977).

The major emphasis of this study is to develop a methodology that can be used to assess the current methods used for assuring the seismic safety of nuclear power plants. The proposed methodology makes use of system-analysis techniques and Monte Carlo schemes. Also, in this study, we evaluate previous assessments of the current seismic-design methodology.

C. T. Crowe and H. Weiss, *Metering Low-Quality Steam-Water Flows*, Lawrence Livermore Laboratory, Rept. UCRL-52271 (1977).

The simultaneous measurement of mass flow rate and quality of low-quality steam-water flows is of primary interest in evaluating and monitoring the flow from geothermal resources. This report relates the experience acquired at LLL in attempting to perform such measurements with a double flash (meter-valve-meter) arrangement using orifices and venturis. The mass flow correlations proposed in the literature are reviewed and a new correlation based on the energy equation is introduced. The data are presented and evaluated in light of these correlations. Directions for future research in this area are suggested.

D. J. Bender, J. D. Lee, W. S. Neef, R. S. Devoto, T. R. Galloway, W. L. Dexter, M. A. Hoffman, J. H. Fink, K. R. Schultz, D. Culver, R. Rao, S. Rao, and W. E. Kastenber, *A Reference Mirror Hybrid Fusion-Fission Reactor Design*, Lawrence Livermore Laboratory, Rept. UCRL-79093 (1977). Submitted to the American Nuclear Society 23rd Annual Meeting, New York City, June 12-16, 1977.

G. A. Carlson, *Parametric Design Study of Tandem Mirror Fusion Reactors*, Lawrence Livermore Laboratory, Rept. UCRL-79092 (1977). Submitted to the American Nuclear Society 23rd Annual Meeting, New York City, June 12-16, 1977.

The parametric design study of the tandem mirror reactor (TMR) is described. The results of

this study illustrate the variation of reactor characteristics with changes in the independent design parameters, reveal the set of design parameters which minimizes the cost of the reactor, and show the sensitivity of the optimized design to physics and technological uncertainties. The total direct capital cost of an optimized 1000 MWe TMR is estimated to be \$1300/kWe. The direct capital cost of a 2000 MWe plant is less than \$1000/kWe.

J. H. Pitts and E. W. McCauley, *A 1/5-Scale Experiment of a Mark I Boiling-Water Reactor Pressure-Suppression System Under Hypothetical LOCA Conditions*, Lawrence Livermore Laboratory, Rept. UCRL-79143 (1977). Submitted to The American Nuclear Society Topical Meeting on Thermal Reactor Safety at Sun Valley, Idaho from July 31 through August 4, 1977.

Experimental results show the sensitivity of hydrodynamically generated vertical loads to changes in the drywell pressurization rate, downcomer submergence, and vent-line loss coefficient. Insignificant effects on peak vertical loads were observed when the vent-line loss coefficient was varied. Peak vertical loads can be reduced by adding initial drywell overpressure so that the downcomers are partly cleared of water. Spatial variation of pressure at about the time of vent clearing is seen in comparisons of data from locations along the axis of the toroidal wetwell.

H. S. Freynik, Jr., D. R. Roach, D. W. Deis, and D. G. Hirzel, *Evaluation of Metal-Foil Strain Gages for Cryogenic Application in Magnetic Fields*, Lawrence Livermore Laboratory, Rept. UCRL-79202 (1977). Presented at the Cryogenic Engineering Conference, University of Colorado, Boulder, Colorado, August 2-5, 1977, submitted to *Advances in Cryogenic Engineering*.

C. A. Calder and G. L. Goudreau, *Application of the Finite-Element Technique to Stress Wave Propagation*, Lawrence Livermore Laboratory, Rept. UCRL-79298 (1977). Submitted to *Experimental Mechanics*.

An axisymmetric finite-element code was used to solve a variety of wave propagation problems ranging in complexity from stress waves in an elementary rod to a highly-dispersive, laser-generated stress wave configuration. Corresponding experimental data were compared in time-history form with the numerical solutions. Good agreement was obtained in most cases but the pulsed-laser code solution did not predict the high-frequency detail found experimentally.

L. C. Pittenger, *Large-Scale Cryopumping for Controlled Fusion*, Lawrence Livermore Laboratory, Rept. UCRL-79381 (1977). Submitted to the Proceedings of the 1977 Cryogenic Engineering Conference, August 2-5, 1977, University of Colorado, Boulder, Colorado.

Vacuum pumping by freezing out or otherwise immobilizing the pumped gas is an old concept. In several plasma physics experiments for controlled fusion research, cryopumping has been used to provide clean, ultrahigh vacua. Present day fusion research devices, which rely almost universally upon neutral beams for heating, are high gas throughput systems, the pumping of which is best accomplished by cryopumping in the high mass-flow, moderate-to-high vacuum regime.

Cryopumping systems have been developed for neutral beam injection systems on several fusion experiments (HVTs, TFTR) and are being developed for the overall pumping of a large, high-throughput mirror containment experiment (MFTF). In operation, these large cryopumps will require periodic defrosting, some schemes for which are discussed, along with other operational considerations.

The development of cryopumps for fusion reactors is begun with the TFTR and MFTF systems. Likely paths for necessary further development for power-producing reactors are also discussed.

D. J. Bender, *A Review of the Mirror Hybrid Reactor*, Lawrence Livermore Laboratory, Rept. UCRL-79548 (1977). Submitted for publication in the Proceedings of the US/USSR Workshop on Hybrid Reactor Design, Moscow, USSR, March 14-April 2, 1977.

B. W. Maxfield, *Optical Interferometry for the Evaluation of Material and Structural Characteristics*, Lawrence Livermore Laboratory, Rept. UCRL-79582 (1977). Submitted for publication in the Proceedings of ARPA/AFML Program Review in Quantitative NDE/Ithaca, NY/Rockwell Science Center.

Of interest in many engineering applications is the change in shape of a body or the change in strain distribution within a body when it is subjected to altered external conditions. To be of maximum use, methods for evaluating these conditions should be nondestructive. Optical interferometric techniques have been used extensively to determine the displacement of the surface of a body between two differing states; these are both nondestructive and offer a wide field of view. Various forms of image shearing (displacement) interferometry are capable

of measuring the derivative of the surface displacement between two states and hence give the strain distribution.

Interferometric methods have been used for a number of specialized studies. Residual strain induced by cycling to increasingly higher pressures within the plastic flow regime has been measured in experimental pressure vessels. This information is used to determine the optimum welding method and parameters for high energy rate forged stainless steel. Subsurface defects and material inhomogeneities frequently manifest themselves as an asymmetric or localized distortion of a body. Holography has been used to search for localized deformation in fabricated components such as wound fiber-epoxy pressure vessels. The elastic strain and thermal expansion of fabricated components has been measured using holography; this permits actual physical properties of the components to be compared with design or table values.

B. J. Benda and R. C. Greenlaw, *Application of ADINA to Nonlinear Contact Problems*, Lawrence Livermore Laboratory, Rept. UCRL-79593 (1977). Submitted to the ADINA Conference, Massachusetts Institute of Technology, Cambridge, Mass., August 4-5, 1977.

Analytical efforts to determine contact forces generated by various seal configurations indicate a strong need for a finite element code that can efficiently solve static, nonlinear contact problems. The results obtained by using ADINA to solve such problems are presented. In addition to a discussion of the effects of program parameters on the problem solution, basic concepts to be included in a more sophisticated contact element are given.

D. J. Bender, J. D. Lee, and K. R. Schultz, *Fuel Design Considerations for the Mirror Hybrid Reactor*, Lawrence Livermore Laboratory, Rept. UCRL-79637 (1977). Submitted to the American Nuclear Society Winter Meeting, San Francisco, California, November 27-December 2, 1977.

W. S. Neef and D. W. Culver, *Mechanical Structure of the Mirror Hybrid Reactor Power Plant*, Lawrence Livermore Laboratory, Rept. UCRL-79638 (1977). Submitted to the American Nuclear Society Winter Meeting, San Francisco, California, November 27-December 2, 1977.

T. T. Saito, R. Esposito, and F. E. Johnson, *Adherent Dielectric Coating of Diamond-Turned Mirrors*, Lawrence Livermore Laboratory, Rept. UCRL-79 '65 (1977). Submitted to Optics Letters.

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