CONCEPTUAL DESIGN OF RF RESONATORS
FOR A 500 MeV H⁻ CYCLOTRON

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ABSTRACT

The TRIUMF cyclotron uses quarter wave resonators instead of conventional dees as accelerating electrodes. Proper functioning of the accelerator requires the maintenance of a certain voltage equality along the accelerating gap, which in turn requires that the resonators be structurally rigid and exhibit high mechanical stability. Space limitations within the vacuum tank, as well as preference for a design not incorporating stand-off insulator supports, led to a compromise solution where RF power losses and resonator rigidity are nevertheless acceptable.

Five basic shapes for the arms of the resonators were evaluated from the viewpoint of structural rigidity, RF losses and fabrication simplicity. Several model tests carried out by TRIUMF in order to verify the merits of certain design parameters are also described.

The report concludes that the use of roll-bond panels (aluminum/silver), reinforced with I-shaped aluminum stiffeners, will result in an acceptable resonator design.
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1. **INTRODUCTION**

The TRIUMF cyclotron, a negative hydrogen ion accelerator, is to be constructed at Vancouver, British Columbia. Ions from an external ion source are injected at the centre of the cyclotron and accelerated to a final energy of five hundred million electron volts (500 MeV). This energy, at the selected magnetic field strength, corresponds to an ion orbital frequency of 4.56 MHz. The accelerating voltage is provided by accelerating electrodes 640 in. wide, giving the ions an energy gain per turn of approximately 400 keV. If these electrodes were to be made in the form of conventional "dees", the power losses would be tremendous and the mechanical problems would be exceptionally severe. A different approach has therefore been taken: instead of conventional dees, quarter wave resonators operating at a frequency five times higher than the orbital frequency of the ions will be used on the cyclotron. This type of RF structure, originally proposed by Dr. K.R. MacKenzie, UCLA, results in reasonable power losses and permits a simpler and more reliable design.
2. **SCOPE**

The purpose of the study was to establish a basic resonator design concept. Although closely related items such as the RF power supply and the RF feed lines to the resonators were examined, these items are not discussed in any detail in this report.

In order to arrive at a conceptual design it was necessary to meet the following criteria:

Establish a physical shape for the resonators which, while meeting the structural requirements, would result in minimum electrical losses.

Verify that a structurally rigid but light resonator can be accommodated within the available vacuum tank space.

Establish the major physical dimensions for the resonators.

Devise a mounting system such that each resonator segment can be identically supported in the vacuum tank.

Assess the heat load in the resonator and devise a cooling system that will ensure low thermal gradients and hence low thermal distortions.

Incorporate design features that will tolerate large differential thermal expansions during bakeout without distorting the resonator arms.

Design a stiff resonator hot arm so that inherent tip deflections are minimized and natural mechanical vibrations are within an acceptable range.

Assess the deflections and distortions due to structural limitations of the vacuum tank support and resonator structure, and determine methods of dealing with them.

Design a trim panel capable of compensating for electrical dissimilarities between resonator segments resulting from mechanical distortions and/or mechanical manufacturing tolerances.
Consider and evaluate current state-of-the-art manufacturing processes for obtaining a superior electro-conductive layer, and a simple structure with low outgassing characteristics.

Make a mechanical vibration analysis and investigate the need for damping devices to suppress electro-mechanical vibrations.
3. **TERMINOLOGY**

The accelerating electrodes, which in the TRIUMF Proposal and Cost Estimate\(^1\) were called Dees and/or Resonators, will in the remainder of this report be called resonators. The following terminology has been adopted for components which are parts of or associated with the resonators (see also Figure 1):

- **Resonator Segment**: A portion of a row of the accelerating structure, presently 32 in. wide, being a preferred size for manufacturing and installation.

- **Resonator Upper or Lower Row**: Exactly one fourth of the total number of accelerating electrodes attached either to the vacuum tank cover or to the vacuum tank bottom.

- **Resonator North or South Row**: Exactly one fourth of the total number of accelerating electrodes orientated with respect to the cyclotron facility.

- **Hot Arm**: The cantilevered arm of a resonator segment.

- **Ground Arm**: The complementary arm attached directly to the vacuum tank and which in turn supports the resonator segment.

- **Tip, Intermediate and Root Arm Portion**: These are portions of the hot arm exhibiting different unit weight and stiffness characteristics.

- **Root Piece**: A portion of the resonator segment joining the hot and ground arms.

- **Conductive Surface**: A coating or cladding, either electrodeposited or roll-bonded, of high electrical conductivity, such as copper or silver.

- ** Levelling Arm**: A structural extension at the root of a resonator segment, employed for support and positioning of the hot arm tip.

- **Trim Panel**: A device that will compensate for resonator segment manufacturing tolerances, as well as adjust automatically for thermal distortions and electrical non-similarities or drift.
4. DESIGN CRITERIA

4.1 Resonator Electrical Requirements

4.1.1 Frequency

To account for the expected discrepancies between magnetic properties of the model magnet and those of the full-scale cyclotron magnet, as well as facilitating an increase of the final energy to more than 500 MeV,\(^2\) it will be necessary to make the frequency adjustable within the limits of 22.58 MHz and 23.25 MHz. It is proposed to make this adjustment either by moving the root piece or by adjusting the resonator arms.

The resonators are to be made from 32 in. wide segments which are unlikely to be electrically identical. In addition, during operation the segments will be distorted due to thermal stresses in the material and possibly due to dimensional changes at the points where the segments are supported. Thus a certain amount of frequency de-tuning is expected, which will be corrected by the use of continuously adjustable trimming devices capable of maintaining the set frequency within 4 parts in \(10^7\).

4.1.2 Voltage

The accelerating voltage has previously been specified as 200 kV peak-to-peak between the tips of two facing resonators, or 100 kV peak from resonator tip to ground. Due to subsequent reduction in the pole gap dimension less space than previously is now available for the resonators which in turn results in higher RF skin losses.

4.2 Resonator Mechanical Requirements

4.2.1 Space Restriction (and Environment)

The gap between upper and lower magnet faces is fixed at 20.8 in. and cannot be increased without seriously affecting the magnet design. Immediately adjacent to the magnet surfaces, occupying the nominal one-inch gap between the magnet pole and the vacuum tank and mounted on the top and bottom tank covers, are the magnet trim coils. The
vacuum tank is suspended from above and supported from below by long tie rods, attached to it in a systematic pattern.

With a magnet gap of 20.8 in., a nominal clearance of one inch between magnet pole and tank, and a 0.875 in. thick tank plate, the resonators may be accommodated in 17.05 in., i.e. the internal vacuum tank dimension from top to bottom plates (see also Figure 2). Of this amount approximately 4 in., in the median plane, are assigned to accommodate the beam. The resonator structure is thus confined to 6.525 in. for either top or bottom resonator rows.

In order to satisfy the high vacuum requirements and to give a short pumpdown time, bakeout will be required for the resonator structure and vacuum tank. Vacuum investigations suggest that the resonators should be capable of bakeout at 200°C and that the vacuum tank requires bakeout at 140°C. Since the resonator structure is to be mounted on the vacuum tank, the problem of differential thermal expansion requires a sophisticated resonator support system. During bakeout the resonator structure, being lighter than the vacuum tank, will achieve bakeout temperatures very rapidly while the greater thermal capacity of the vacuum tank will cause it to lag in temperature. The thermal expansion problem is magnified if the resonator and tank materials have different coefficients of expansion.

4.2.2 Physical Shape

The geometry of the resonator structure was outlined in the TRIFM Proposal and Cost Estimate. The accelerating assembly in this report called for 52 resonator segments forming two resonator arrays, each array consisting of a top and bottom row, with the beam passing in between. The resonators were divided into segments in order to simplify manufacturing and handling. Electrically, each resonator array would be split in the middle to give two sections, with coupling loops inserted in the upper section of the parting line.

The maximum acceptable asymmetry between voltages at the accelerating gap at equal distances from the cyclotron centre, and diametrically
opposite, has been defined\(^3\) as:

<table>
<thead>
<tr>
<th>Radius ((R))</th>
<th>Acceptable Voltage Difference ((\delta u))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-25 in.</td>
<td>0.5%</td>
</tr>
<tr>
<td>30</td>
<td>0.7</td>
</tr>
<tr>
<td>40</td>
<td>1.2</td>
</tr>
<tr>
<td>50</td>
<td>1.9</td>
</tr>
<tr>
<td>&gt; 50</td>
<td>1.9 + 0.09%/in.</td>
</tr>
</tbody>
</table>

The maximum acceptable voltage asymmetry between upper and lower resonator hot arms as a function of radius is given by:

<table>
<thead>
<tr>
<th>Radius ((R))</th>
<th>Acceptable Voltage Difference ((\delta u))</th>
</tr>
</thead>
<tbody>
<tr>
<td>0-40 in.</td>
<td>0.5%</td>
</tr>
<tr>
<td>50</td>
<td>1.0</td>
</tr>
<tr>
<td>70</td>
<td>2.5</td>
</tr>
<tr>
<td>100</td>
<td>6.0</td>
</tr>
</tbody>
</table>

It is intended that the tips of the resonator segments in each row be connected together. This will help to maintain the voltage level within acceptable limits along the accelerating gap. It is further planned that connections be made between upper and lower resonator hot arms at the centre post and at the outer ends. By so doing, the voltage difference between the resonators in the vertical direction can be minimized, and it is expected that no difficulty will be experienced in keeping the voltage difference within tolerance.

One arm of each resonator would be directly mounted to the vacuum tank while the other arm would be cantilevered from the root. Insulators were ruled out as resonator supports because of the unknown effects of radiation on their integrity, and because of the difficulty of replacing faulty insulators. The cantilevered arm, however, demands a very rigid structure in order to meet the stability requirements.
A basic segment about 32 in. wide appears desirable. It has been shown that, regardless of the tie rod spacing adopted, the vacuum tank plate will deflect considerably between supports. Hence it is undesirable to provide resonator attachment points at other than tie rod locations. In turn, it would also be logical to provide support points around the segment perimeter because this would facilitate accessibility for mounting and dismounting. Figure 3 shows the support point system selected.

The segment length is a function of the resonant frequency. RF calculations have shown that the tip-to-root dimension should be approximately 10.3 ft if the RF accelerating frequency of 22.6 MHz is to be achieved.

There will be no need to make the ground arm structurally very stiff since it is feasible to support this arm at several points along its length. The hot arm, being a cantilevered structure, should be several times greater in depth than the ground arm. It will probably be impossible to achieve a perfect built-in condition for the hot arm at the root; hence for any tip deflection and natural frequency analysis a pin-joined connection should be considered at this location. From a design viewpoint it is imperative that the hot arm structure be extended beyond the root piece and that it connect to the vacuum tank under a tie rod. Since there will be tie rods beyond the root piece, the choice for the hot arm extension is 32 in., or multiples of this. The choice will be based in part on the space available behind the resonator root. At a row end, the structural extension becomes difficult to accommodate because the curvature of the tank leaves little room for a resonator support structure.

The shape and length of the hot arm extension, which will be subsequently referred to as the levelling arm, is further influenced by the resonator coolant header, two of which will run behind the resonator root and extend the full length of a resonator row.
The centre section of a resonator will require a more sophisticated segment shape, as will the segments at the end of the rows. The basic resonator shape will be modified at these locations to fit around the centre post, to support central region electrodes, and to be compatible with end flux guides near the vacuum tank wall.

4.2.3 Heat Flux

Approximately 1 MW of thermal energy is deposited in the resonator conducting surface. Current distribution along the length of flat and parallel resonator arms is sinusoidal, with maximum current at the root. Heating is proportional to the square of the current so that heating will vary along the arm length as sine-squared. With a view to minimizing thermal gradients in the skin of the arm, it is imperative that the cooling system for the arm be compatible with this heat distribution.

The RF currents and the heating will be confined to a very thin outer layer of the panel, less than 0.001 in. in depth. In order to minimize thermal gradients and distortions, it is desirable to place the cooling system immediately adjacent to the current-carrying skin, thereby isolating heating effects from the structural portion of the resonator arm. It is anticipated that with this approach thermal distortion can be minimized and the critical gap dimensions can be maintained.

4.2.4 Source of Errors due to Structural Limitations

Irrespective of the adopted design, a resonator of the size specified will exhibit some undesirable structural characteristics which are at direct odds with the specified requirements, such as the following:

Because of its great length the cantilevered hot arm will exhibit a large static tip deflection. In order to minimize this deflection the ratio of unit weight to stiffness must be low. Prebending of the hot arm in the opposite direction of deflection might be advisable.
Heating will occur on the conductor side of a resonator arm and therefore some temperature gradients will exist in the direction of thickness. With the thermal mass of the arm structure kept to an absolute minimum, because of weight limitations, this type of gradient will be small. Nevertheless, because of the arm's physical size, such a gradient may induce considerable double curvature distortion.

There will be periodic deflections or distortions of the resonator arms leading to tip deflections of the order of 0.001 in. For example, periodic distortions will arise because of temperature fluctuations of the coolant to the resonators. Even with temperature control, the coolant temperature may vary as much as ±2°F.

Under atmospheric load the straight beam of the cyclotron primary support structure will deflect by as much as 0.059 in., and this motion will be transmitted through the tie rods to the vacuum tank covers. Since the resonator tips are mounted below tie rods, the tips of the resonators will not form a straight line unless this deflection is compensated for.
5. RESONATOR DESIGN CONCEPTS

5.1 Hot Arm Concepts

The structural design of the resonator hot arm is by far the most critical item. An acceptable solution to the problem of designing the hot arm automatically leads to a satisfactory ground arm design. The preliminary concepts were reviewed, and it was concluded that the proposed method of constructing the arm of 0.125 in. thick oxygen-free copper with suitable reinforcing and attached cooling pipes would not give the most satisfactory arrangement. The copper plate contributes considerable mass, and the attachment of numerous coolant pipes would necessitate lengthy welding or brazing. If water were to be employed exclusively as a coolant, it would add additional weight and lead to excessive tip deflection; gas cooling on the other hand would result in a lighter assembly but would cause greater temperature gradients and hence larger thermal distortions. Five alternative concepts, two of which make use of silver-coated aluminum roll-bond panels, are discussed in the following sections.

5.1.1 Design Concept I

This design makes use of 0.50 in. thick aluminum extrusion incorporating numerous 0.375 in. diam coolant channels. In order to minimize the weight it is intended that only every second hole carry coolant. The arm would be water cooled over 60% of its length while the remaining 40% at the tip would be gas cooled. With such a design temperature gradients and hence thermal distortions could be held to a minimum. Table I gives a summary of pertinent characteristics and Drawing 60053 illustrates the design concept.

The physical shape of this concept corresponds closely to a design shown on TRIUMF Drawing E-034, but instead of using two stiffening ribs in a 46 in. wide hot arm, the new concept incorporates one such tip in 22 in. of width. Despite this stiffening rib however, tip deflections were found to be excessive.
<table>
<thead>
<tr>
<th>Tip deflection (in.)</th>
<th>HOT ARM CONCEPT</th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Levelling arm infinitely stiff</td>
<td>I</td>
<td>II</td>
<td>III</td>
<td>IV</td>
<td>V</td>
</tr>
<tr>
<td></td>
<td>0.964</td>
<td>0.596</td>
<td>0.494</td>
<td>0.593</td>
<td>0.434</td>
</tr>
<tr>
<td>Weight (lb) - arm width 33 in.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Weight not incl. levelling arm</td>
<td>170</td>
<td>195</td>
<td>260</td>
<td>135</td>
<td>130</td>
</tr>
<tr>
<td>Maximum temperature difference across skin surface (°F)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>1.5</td>
<td>&lt;1.0</td>
<td>&lt;1.0</td>
<td>5</td>
<td>2</td>
</tr>
<tr>
<td>Maximum temperature difference across arm thickness (°F)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>~2.0</td>
<td>~1.0</td>
<td>~2.0</td>
<td>&gt;0.5</td>
<td>&gt;0.5</td>
</tr>
<tr>
<td>Thermal deflection at tip (in.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>estimated</td>
<td>0.021</td>
<td>0.031</td>
<td>0.062</td>
<td>&gt;0.010</td>
<td>&gt;0.010</td>
</tr>
<tr>
<td>Variation in thermal deflection (in.)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>estimated</td>
<td>0.002</td>
<td>0.003</td>
<td>&gt;0.006</td>
<td>&gt;0.001</td>
<td>&gt;0.001</td>
</tr>
<tr>
<td>Aluminum fabrication and coating method</td>
<td>extrusion electro-plating</td>
<td>extrusion electro-plating</td>
<td>extrusion electro-plating</td>
<td>roll-bonded</td>
<td>roll-bonded</td>
</tr>
<tr>
<td>Complexity of coating and coolant channel welding connection</td>
<td>high</td>
<td>very high</td>
<td>very high</td>
<td>min</td>
<td>max</td>
</tr>
</tbody>
</table>
The width of an extrusion is limited to about 12 in. and this puts a definite limitation on a feasible resonator width. It is therefore proposed to join two extrusions centrally along the stiffening web to give a resonator width of about 24 in.

With this design a conductive layer would have to be applied by electroplating. Such a process would not give an entirely satisfactory coat in the thickness range required. Roll-bonding a conductive layer under high pressure and temperature is not feasible for an extrusion because of the obvious deformation problem in the vicinity of the cooling holes. Adhesive-bonding techniques are undesirable from a vacuum viewpoint and, in addition, no satisfactory adhesive is available for long-term use in a radioactive environment.

5.1.2 Design Concept II

Again employing the extrusion concept, a lighter and more rigid arm can be realized in the form of elliptical tubes. This design also increases the resonator volume, and hence the electrical characteristic impedance, resulting in lower power requirements.

Although the space between the tubes increases the volume of the resonator - a desirable feature - it also permits the electric field to penetrate into the beam space, possibly to the detriment of beam characteristics.

This concept displays three distinct hot arm structural portions. The root piece consists of a 12 in. long by 1.5 in. thick aluminum extrusion exhibiting high rigidity. Note that in this region unit weight is of secondary importance. The elliptical tubes feature a high stiffness-to-weight ratio, and form the deflection-prone intermediate portion. The tip piece is low in weight and the flat panel satisfies voltage requirements at the tip. Inherent structural deflection characteristics are superior to Concept I. In addition, thermal gradients are minimized, making this a very attractive solution from a mechanical design viewpoint.
The conductive layer would have to be electro-deposited. For the elliptical tubes 50% of the tube area would have to be shielded from the coating. At the transition from elliptical tubes to flat surfaces considerable silver soldering would be required to connect the electrical surfaces.

5.1.3 Design Concept III

This design is a variation of Concept II. Here panels have been added between the tubes in order to avoid electric field penetration (see Drawing 60053). Because of the smaller cooling holes, temperature gradients would be expected to be higher with this design. Electro-deposition of the conductive layer, as well as fabrication complexity, detract from its otherwise favourable characteristics.

5.1.4 Design Concept IV

By employing a "roll-bond" heat panel, it becomes practicable to design a flat resonator arm that can accommodate numerous stiffeners. For this concept the basic arm thickness is 1.75 in. as shown in Drawing 60053. It is apparent that with the flat arm design the resonator volume will be somewhat reduced; hence higher power requirements are to be expected.

The conductive layer, and the two base sheets forming the flat coolant channel, are bonded under high temperature and pressure to form a single plate. Reinforcing structural members are added to the underside of the panel; in this concept a total of six I-beams are used. The choice of six stiffeners appeared desirable in order to facilitate comparison with Concepts II and III where the choice of six tubes was based on RF requirements.

Deflection characteristics are excellent with this design. Thermal gradients, with their associated distortion, can be confined to the thin heat panel.
5.1.5 **Design Concept V**

Concept V is one which combines the good feature of Concept III, i.e. low power requirement, with the good features of Concept IV, e.g. low mass, low deflection, low thermal distortions and a superior conductive layer. The design incorporates the roll-bond technique and the expanded coolant channel principle.

5.2 **Ground Arm Concepts**

For a given conductive surface RF power losses will be the same for either arm. Since the ground arm is immediately adjacent to the vacuum tank plate, its design should make use of that plate's inherent stiffness (see Figure 2). By so doing, the ground arm thickness can be minimized while the resonator gap can be maximized. In the extreme case there would be no need for a separate ground arm.

If a conductive coating such as copper or silver were applied to the stainless steel vacuum tank plate and cooling coils attached either to the coating directly or placed on the outside of the vacuum tank plate, the tank plate could serve as a ground arm. Attaching the cooling tubes to the outside of the tank plate would, however, be unacceptable since the resulting temperature gradient would lead to excessive distortion of the vacuum tank. The only satisfactory means of attaching the conductive coating would be by explosively cladding the stainless steel with copper. Preliminary investigations show that clad stainless steel would be roughly four times as expensive as ordinary stainless steel. Considerable additional expense would be incurred because of the fact that clad stainless steel is more difficult to weld, especially since all vacuum tank joints will be of the field-weld variety.

When using a separate ground arm panel mechanically mounted to the vacuum tank a construction closely corresponding to that of the hot arm construction is feasible. The structural stiffness requirements for the ground arm are, however, less stringent.
5.3 Trimming Devices

The initial TRIUMF cyclotron concept required that the major portion of the resonator structure be underneath the magnets. Frequency trimming devices mounted at the resonator tips would thus have to be installed and serviced from the inside of the vacuum tank, which is highly undesirable. For further discussion on this arrangement, see Section 6 of this report.

Instead of changing the capacitance of the resonators by the tip panels, the desired frequency change can be achieved by changing the inductance at the root of the resonators. This is done by adding one or more "pockets" whose depth is varied by the use of bellows (see Figure 4). This arrangement uses a large number of bellows that may develop leaks, but since the mechanisms are comparatively easily accessible this system is preferable to one where the mechanisms are placed at the resonator tips and accessible only after disassembly of the resonator arms.

Subsequent changes in the position of the accelerating gap in relation to the magnets have made it feasible to improve accessibility from outside the vacuum tank to areas in the vicinity of the resonator tips. At present a system for tuning the segments by deflection of the tip portion of the ground arm is being studied. Preliminary results indicate that the latter system will provide the most convenient and reliable tuning system.
6. **CONTROL SYSTEM**

Although precautions are going to be taken in the design of individual components, and great care exercised in selection of materials and during manufacturing, the finished product will deviate considerably from the ideal.

A number of corrective measures can be taken during assembly and installation of the RF system to compensate for manufacturing and other tolerances of a static nature, while changes in properties and/or characteristics of components during operation of the accelerator require a continuously available adjustment system or control system.

For proper functioning of the cyclotron the frequency of the RF system must be maintained stable within 4 parts in $10^7$. In addition, a certain overall voltage stability as well as voltage equality along the resonator tips is required.

It has been determined by model measurements and computer studies (see Appendix) that if one resonator hot arm deflects uniformly in such a way that the resonator gap decreases or increases by 0.010 in., the resulting overall frequency change will amount to 4 parts in $10^5$. If the hot arm deflected is at the end of a resonator row, the voltage difference between it and a hot arm diametrically opposite in the same row would be 0.065%, assuming the same quality of segment-to-segment tip connections on the cyclotron as in the model. This voltage difference is much less than that permissible, and it is expected that even though the tip-to-tip connections in a resonator row on the cyclotron may be less efficient than those on the model no difficulty will be experienced in keeping the voltage equality within tolerance.

Should the segments be tuned to different frequencies the Q of the system will be adversely affected, resulting in higher power losses. This will be the case if the resonator structure as a whole is tuned to a frequency differing from the driving frequency.

Initially, the resonator hot arms will be aligned mechanically to within a few thousandths of an inch. Should the Q be unreasonably
Further tuning of individual segments may be necessary and will be achieved by adjustment of the segment support arm from outside the vacuum chamber.

Frequency drift during operation will be compensated by adjustment of the trim panels, either at the root or at the tip of the resonator. A third (and simpler) way of adjusting the frequency is by deflection of the ground arms in the tip region.

6.1 Frequency Detection and Adjustment Method

The condition at resonance is that the voltage at the resonator tip and the current at the root are 90° out of phase.

It is intended that the tip voltage be sampled by a capacitive pick-up plate and the signal compared by means of a phase detector with that derived from a small inductive loop at the resonator root. The phase error signal will be amplified and used to control one of the trimming devices described above by means of suitable electro-pneumatic transducers. Figure 5 shows a schematic of the proposed system.

6.2 Frequency Trimming Concepts

Three different trimming methods have been considered: Method 1 makes use of movable panels near the tip of the resonator, Method 2 employs trim panels at the root of the resonator (volume-changing pockets) and Method 3 is a variant of Method 1 made feasible by rotation of the accelerating gap in relation to the magnet sectors. Here, instead of moving separate trim panels at the tip, a certain portion of the ground arm is deflected.

6.2.1 Actuators for Tip Panels

Several trim panel actuators which have been examined from the viewpoint of feasibility and reliability are summarized and shown in Drawing 60059:

1) The first three designs are basically one concept that underwent modifications as investigations proceeded. The concept makes
use of a metal bellows that can be pressurized to overcome the spring force; variations of pressure in the bellows actuate the panel. The problem is lack of space because of the vacuum tank cover and the magnet pole pieces. The actuator is complicated by the fact that it must operate satisfactorily in and out of the vacuum. This latter requirement is important since the mechanism must be thoroughly checked for operation, in particular accurate trim panel translation, before it can be placed and sealed inside the vacuum tank. This fact necessitates two different springs, one for testing at atmospheric pressure and one for operation in vacuum. Calculations show that spring stresses would be high. In addition, the requirements are such that a spring with only two active coils can be utilized. Such an actuator is not an attractive arrangement.

2) This concept makes use of two cables and a spring. In order to fit in the available space, small bending radii are necessary for the cables. As a result, bending stresses in the cables would be high. With this arrangement, employing cables and guides, there is a serious problem of friction and wear, especially since adequate lubrication cannot always be assured. Hence, the life of such a device must be assumed to be short, and reliability is at best uncertain.

3) Concept 3 is yet another variation of the cable and spring arrangement. Again, bending stresses are very high, especially at point 'x'.

4) Concept 4 is a cable and spring arrangement which in addition makes use of a cam. A 0.020 in. cable travel would achieve a 0.050 in. trim panel translation. Unfortunately, this magnification is in the wrong direction.

5) The bellows and spring arrangement shown in Concept 5 solves the problem of the difference in performance during tests at atmospheric pressure and under vacuum conditions. Effects of changes in environmental pressure are nullified by the equal effects on
the two bellows. This necessitates that each bellows be separately fed by pneumatic lines, hence doubling pneumatic lines and connections. Considering the errors of the springs and the bellows area, the total tolerance would not exceed ±0.005 in. throughout the 0.50 in. stroke.

6) Concept 6 makes use of a drive shaft, a cam and small pinion, and a large gear. With this arrangement, a large mechanical advantage can be realized and the panel adjusted to high accuracies. One shaft rotation will give 0.031 in. displacement and 12 deg shaft rotation will give 0.001 in. displacement. The major disadvantage with this concept is obviously the numerous moving components; a total of 300 gears and 500 bearings would have to operate in a vacuum if we allow one trim panel actuator for each resonator. Metal-to-metal interface problems may occur after about 500 hours exposure to vacuum despite care being taken in selecting the interface materials. In addition, such a complex mechanism presents an outgassing problem.

7) Utmost mechanical simplicity is exhibited by Concept 7 which employs a bimetallic strip as an actuator. The strip evaluated is a combination of stainless steel and aluminum. It would be heated to between 100°F and 500°F in order to achieve the 0.50 in. range of travel. A mechanical advantage of about 3 is required for this purpose. This type of actuator is a purely static, and hence reliable, type but its performance with respect to material and environment is the least attractive. The accuracy of this device is influenced by the thermal resistance from the heat source to the bimetal body and again from the body to the heat sink; variations in heat dissipation of the heating element with respect to aging; and the variation in cross-sectional area per inch of the bimetal strip.

Some form of feedback around the actuator system could compensate for these deficiencies. Heat sink stability, if limited to 2 or 3°C by a
cooler, would contribute approximately 0.005 in. error. Differential temperature coefficients of the bimetal vary about 14% over the useful bimetal operating range and could produce another error of 0.035 in. A square-law relationship between power, temperature and applied heater voltage causes control gradients that vary by a ratio of 2.5 to 1 over the temperature range in question. A square-law resolver would be necessary, involving further complications. The last point for consideration is the effect of testing in air a device intended to operate in vacuum. This would influence the thermal resistance aspect previously mentioned. It is expected that an error of about 0.25 in. would be added by this environmental change.

6.2.2 Actuators for Root Panels

Location of trim panels at the root permits easier access for service and installation than if they are located at the resonator tip. The actuator concept shown on Drawing 60039 illustrates how most of the components are external to the resonator assembly and hence readily accessible.

The trim panels have been broken into two sections. The actuator is composed of two stainless steel bellows arranged so that they will compensate for external pressure variation, thereby permitting testing and calibration of each actuator under atmospheric conditions.

The trim panels make use of rectangular silver-plated bellows which are designed to provide ±0.25 in. translation of the panels. These bellows are notched in the corners to ensure that they will exhibit a linear spring stiffness over the travel range and the openings are shielded by placing small angles at the corners. Since it is conceivable that custom-made rectangular bellows will not exhibit identical spring stiffness, provision for balancing has been incorporated by the placement of springs under the levelling arm. Only one spring is shown in Drawing 60039 but in practice two will be used.

From a mechanical viewpoint, this design is of great simplicity since it avoids bearings, sliding joints and pins. The life of the
mechanism will be a function of the fatigue life of the bellows, typically in excess of $10^6$ cycles for full actuating cycles. Since these are not expected to occur very frequently, possibly only at transient conditions during cyclotron start-ups, such a life will be more than adequate.

6.2.3 Actuators for Deflection of Ground Arm

These actuators can be combined with the capacitive pick-up probes and thus use the same opening through the vacuum tank. The actuators will be external to the vacuum tank and therefore easily accessible for service. If the actuators are made as screw or rack and pinion types, each row can be ganged and driven by a single motor. Pneumatic actuators are also feasible.

6.3 Temperature Control System

The coolant passage arrangement, and general cooling philosophy for the resonator hot and ground arms, is reported in Section A2.1.3 of the Appendix. Cooling of the resonator root is covered in Section A2.4.

The resulting cooling water flow requirements are 4000 lb/h (6.67 Imperial gal/min) for the ground arm, a similar flow for the hot arm, and 1000 lb/h (1.67 Imperial gal/min) for the root of each resonator segment. The total required flow for each segment is therefore 15 Imperial gal/min.

Due to manufacturing tolerances the pressure loss characteristics of the resonator panels will differ slightly from one another. If the differences are sufficiently small the resulting temperature difference from one panel to another will be acceptable. In the case of a ground arm, such temperature differences will only result in slightly differing thermal expansion in length. In the case of the hot arm, thermal deflections will differ from one segment to another, but if the differences are not large they will be compensated for when the hot arms are adjusted at operating temperature. If differences large
enough to be troublesome occur, that is if any one panel or group of panels operates at too high a temperature, a restriction in the form of an orifice will be installed in the lines to the low loss panels in order to achieve a balanced flow. The overall cooling and bakeout system schematic is shown in Figure 6. The resonators will be set up and tested prior to installation in the cyclotron, and the flow to each panel can be measured and adjusted at this time.

A complete monitoring system, for control of cooling, would normally include an under-flow switch and an over-temperature switch. However, such devices normally do not have adequate tolerances because of their 5% to 10% of full-scale differential dead band. In addition, the large number of cooling circuits to be monitored, if each hot and ground arm is to be checked, raises a question of monitor reliability.

This problem of coolant system monitoring has been considered in the light of the overall cyclotron controls problem, since many more cooling circuits are involved in other components. The overall system has been reported in more detail in a report on the controls concept, but the system adopted for the resonators consists of monitoring the temperature of the resonator panels themselves by means of attached thermocouples. Deviations from the normal operating temperature will be detected and appropriate warning or shutdown action taken depending upon the magnitude of the deviation.

Since the amount of heat that is to be removed from the resonators is a constant, and external heat dissipation systems such as cooling towers have a variable performance, the system shown provides a means of maintaining constant heat removal. A divertor valve in the resonator loop is used to direct flow either through the heat exchanger or through a by-pass. The divertor valve is controlled by a signal derived by comparison of the temperature at the outlet manifold with a reference. Divertor valves of the feedback actuator type are readily available with 1 to 2% input/output relationships. Temperature tolerance of a good reference and measurement can be readily maintained
at better than 1 to 2°F. The outlet manifold temperature can be held to 2 to 4°F.

The schematic shows an arrangement of pumps whose numbers are open to review. Two choices are reasonable, 1 pump at 100% capacity or 3 pumps, each with 50% capacity, leaving one on standby. Before proceeding with detail design the equipment costs, outage time and repair costs, along with a policy decision concerning the extent of back-up systems, will be considered.

6.4 Resonator Bakeout System

The schematic in Figure 6 also includes a bakeout system consisting of a steam generator which may be employed to heat the resonators to bakeout temperature during the vacuum roughing cycle. During the pump-down period the cyclotron operator will switch over from the cooling to the bakeout circuit, after which the temperature control system of the bakeout circuit will automatically maintain the chosen bakeout temperature.

It is probable that the pumping characteristics of the vacuum system will require that the resonator temperature be raised from room to bakeout temperature according to some specific time/temperature schedule. Provision will be made for incorporating such scheduling in the control system.
7. MANUFACTURING CONSIDERATIONS

7.1 Tolerances

The discussion of fabrication tolerances will be confined to roll-bond panel production. The complex type of roll-bond panel that incorporates coolant channels, as well as a plated conductive layer, requires a very exacting technique at which only a few manufacturing firms have been reasonably successful. Because of manufacturing difficulties and errors it will be necessary to purchase additional material; hence a cost estimate for the resonator panels will reflect this fact. The additional cost of aluminum will not be significant, but the cost of the approximately 35% extra conductive layer material, especially if it is silver, will be considerable. Fortunately, silver can be reclaimed from those panels that do not meet specifications, and so at least some of the additional material cost can be recovered.

Some of the more frequent manufacturing errors are incomplete conductive layer bonding, unsatisfactory quality of conductive layer, exceeding dimensional tolerances, and annealing faults. Current state of the art limits roll-bond panel sizes to 118 in. by 35.5 in. Tolerances on length are ±2%; on out of squareness, ±3%.

7.2 Welding Requirements

It has been pointed out that the connection of cooling pipes to the inflated cooling channels presents no problem if the necessary precautions are observed.

Since the panel size required exceeds the size that can be manufactured, it will be necessary to join two panels by a butt weld. Problems can arise if metal droplets form at the silver side of the joint and then are removed by grinding. Repairs of such damage may be difficult.

Ideally, a full penetration weld, applied from the non-silver side, should be used. Complete fusion of the aluminum panels is imperative,
and fusion of the silver layers is desirable in order to avoid abrupt steps or transitions in the RF current path. An argon inert gas shield arc weld, with the proper back-up plate at the silver side, should yield the desired results, although undoubtedly a certain degree of development work will be required to perfect the technique.

Distortions of the panels caused by the welding operation may be a problem, and flatness requirements may dictate that the panel will require some straightening. This problem can only be accurately assessed during the fabrication of a prototype resonator which could be done using a plain aluminum sheet instead of a roll-bonded panel.
8. CONCLUSIONS AND RECOMMENDATIONS

It has been shown that with the Concept V hot arm a possible saving in RF power cost of about $180,000 over the power supply cost for Concept IV can be realized. The ground arm and root piece are identical in both instances, and so the basic cost differences can be attributed to the 80 hot arms. If Concept V is adopted, then theoretically an additional $2,250 per arm could be allowed to cover the cost of the increased manufacturing complexity of this arm. The overall cost of a Concept V hot arm is approximately $4,500. A Concept V hot arm made from copper instead of roll-bond is twice as expensive as a Concept IV aluminum hot arm. However, as has been illustrated in this report, Concept V in copper would exhibit large inherent tip deflections, considerable mass and numerous welded pipe connections of questionable reliability. In fact, this concept offers no clear advantage over Concept IV in aluminum, except for the theoretical saving in RF power.

Alternatively, Concept IV in copper is the most economical construction. Although it suffers the same unattractive structural characteristics as a Concept V copper hot arm, it nevertheless retains the simplicity and reliability feature of the inflated roll-bond panel. The inherent greater expenditure on RF power, however, does not balance the saving in resonator manufacturing cost.

In view of the various shortcomings of the above concepts, it appears that Concept IV in aluminum is the most suitable arrangement for this facility. The question as to whether the aluminum panel should be copper or silver plated is not fully resolved, but this is mainly a question of availability and will be decided when suppliers and manufacturers are asked to bid.
REFERENCES


2. E.G. Auld, private communication

3. J.R. Richardson, private communication

APPENDIX

ELECTRICAL AND MECHANICAL INVESTIGATIONS
OF RESONATOR CONCEPTS

A1. Electrical Investigations

A1.1 Computer Studies

For the purpose of arriving at the most favourable resonator design from both electrical and structural viewpoints, several geometrical concepts were studied. See Drawing 60053.

A flat plate design such as Concept IV lends itself to simple and straightforward analysis, whereas the other concepts shown require more laborious computation.

A computer program was written and used in the computation of skin loss, power, voltage, current distribution, electric angle, resonator length, etc. For the more complicated concepts, flux plots were made and conductive paper was used for the purpose of determining the characteristic impedance of the various sections. The same methods were employed in establishing the capacitive loading (fore-shortening) of the resonators by the capacitance existing between the hot arms and the ground plane. Typical flux plots are shown in Figures 7 and 8.

A1.1.1 Resonator Power versus Electric Angle

The power loss in a resonator is a function of the electrical conductivity of the material used, the physical shape of the resonator, and the operating voltage.

The voltage has been fixed at 100 kV peak. Silver and copper have been considered as conductor materials. Since the difference in conductivity between silver and copper amounts to less than 6% and the RF power losses are proportional to the square root of the resistivity, using silver instead of copper would reduce the losses by only 2.5%. Therefore, all power loss computations have been based on the use of copper as conducting material.

Voltage and current distribution along the depth of a resonator, and hence the distribution of the power losses, are dependent on the characteristic impedance of the structure. Figure 9 shows the relationship between power losses and electric angle for some of the resonator concepts considered.
A1.1.2 Resonator Length and Power Consumption versus Resonator Geometry

The physical length of a resonator can be changed by alteration of its characteristic impedance in a certain fashion. The curves in Figure 10 demonstrate this relationship. Concept III with a dimension "a" = 4.1 in. and "b" = 10.3 ft has been chosen as a reference design. A fixed, capacitive tip loading of 8pF has been used in all computations. The dotted curves on the graph show that the resonator can only be shortened at the expense of a higher power consumption.

A1.2 Model Studies

A1.2.1 Quarter-scale Model

The object of the model test program was to investigate: the voltage distribution along the tips of the resonator as a function of mechanical distortions of the resonator arms, the Q of the resonator, the power demand, and the best methods of supplying the power to the resonator.

A plywood model having all current-carrying surfaces with sheet copper was constructed. In the first series of measurements the resonators were divided into two sections each, making a total of four sections. Hot-filament-cathode diodes making contact with the resonator hot arms were distributed along the resonator tips close to the accelerating gap. RF power was supplied through two coupling loops inserted into the two upper resonators and close to the shorted end of the resonators. A tuned-plate power amplifier driven by a variable frequency generator was employed as an RF power source.

A second series of measurements was taken with a portion of the resonator reduced to 24 in. wide segments. Mechanical considerations had indicated that the resonator should be fabricated in smaller units than would be the case if they were broken up into four units only, and the purpose of the second series of measurements was to determine whether the coupling between adjacent resonator segments would still be strong enough to ensure an acceptable voltage distribution along the accelerating gap.

At the high frequency used (100 MHz) difficulty was experienced in keeping the RF power from escaping through instrument leads. In addition, the hot diodes connected to the resonator tips represented a considerable RF load, deteriorating the Q of the structure. The Q was further reduced by the wood exposed as a result of the slicing of the resonators into narrow segments (second series of measurements).
During the course of the theoretical investigation undertaken for the purpose of arriving at a resonator design offering greater electrical and mechanical advantages, it was decided to build a half-scale model and to use it for further model trials.

**A.2.2. Half-Scale Model**

The object of the half-scale model test program was basically the same as that of the previously outlined one in connection with the quarter-scale model. A few more items were added to the objectives of the study, such as determining if a single coupling loop would be adequate for power transfer into the system, the extent to which the resonator segments would have to be electrically joined together in order to satisfy the voltage equality requirements, the development and testing of voltage sensing probes with low power drain, the determination of the resonator end and centre segments, and the investigation of means for tuning the segments to the excitation frequency.

The half-scale model test program was divided into two phases. The first phase included measurements that could be done with the use of a total of 36 resonator segments. The segments were assembled into two rows representing the resonator structure on one side of a vertical plane through the centre of the accelerating gap. This vertical plane is a voltage nodal plane which in the model was simulated by a sheet of copper connected to ground, as shown in Figure 11. A number of resonator voltage probes, consisting of a capacitive voltage divider and a solid-state diode rectifier, were used to measure the RF voltage appearing on different resonator segments near their tips.

To facilitate tuning of individual resonator segment pairs, easily adjustable plate capacitors were installed in the front cover of the resonator box (vertical ground plane). The hot arms of two of the resonator segments - one top and one bottom - were fitted with adjustment screws to permit deflection of these arms.

Frequency trimming was accomplished by two different methods: by adjustable capacitive trim panels close to the tip of the resonators, and by volume-changing trim panels at the root of the resonators. Power was introduced to the resonator structure through a coupling loop, as in the previous tests.

In the second phase of the test program the resonator pairs were grouped in two sets of ten and erected so that the tips of the two sets were facing each other, thus simulating half of the resonator structure as used in the cyclotron.
The first few measurements were taken using ten resonator segments arranged as shown in Figure 12. Voltage differences between various hot arms top-to-top and bottom-to-bottom as well as top-to-bottom were measured for different widths of tip connecting straps and as a function of the amount of deflection of a pair of hot arms.

Curve "A", Figure 12 shows the voltage distribution when a 4 in. wide, 3 in. long and 0.005 in. thick copper strap wrapped around the resonator tip and bridging the gap between two neighbouring resonator segments is used. Curves "B" and "C" in the same figure show the voltage distribution when connecting straps of different widths are soldered to the segments to provide the segment-to-segment interconnection at the resonator tips.

It is evident from the curves that the voltage balance depends heavily on the width of the tip connecting strap. This is an indication that the segments are tuned to different frequencies and that a certain amount of current flows between neighbouring segments in order to make up for any difference in tuning. This could also be demonstrated by sweeping the frequency and observing the voltage peaking at different segments at different frequencies. Without tip connecting straps, the voltage difference between segments was far outside acceptable limits.

For the purpose of facilitating tuning of the individual segments and thus reducing the cross currents and associated voltage difference, it was decided to install adjustable trim plates in the ground plane facing the resonator tips. As a first step towards this a portion of the ground plane was removed and replaced by a sheet of plain plywood. In the centre of this plywood sheet a 4 in. diam copper plate adjustable by a 1/4 in. screw was installed. The copper plate could be moved from a position touching the resonator tips to a position about 0.5 in. away from the tips as shown in Figure 11. Capacitance variation between the resonator tips and the plate as a function of distance was measured by means of a capacitance bridge, see Figure 13.

Next, twenty-six more segments were added to the previous ten and trim plates of the same shape were installed in front of each pair of segments. The system frequency was measured as a function of added capacitance and the results are shown in Figure 14. By adjusting two or more plates simultaneously, it was determined that the effect of added capacitance was cumulative.

With the trim plates adjusted for minimum frequency disparity between segments, first one and then two hot arms were deflected and the segment voltages measured as a function of the
amount of deflection. Figure 15 shows the relationship between segment voltages and deflection of the upper hot arm of resonator section #3. Simultaneous deflection of both upper and lower hot arms of resonator section #3 resulted in twice as much voltage variation. During these voltage measurements the voltage on resonator segment #9 was used as a reference, and is shown as a straight line through zero.

The system frequency as a function of the hot arm deflection was recorded during the measurements. Figure 16 shows the variation in frequency as the upper hot arm of section #3 is deflected. Deflecting both upper and lower hot arms of this section simultaneously doubles the frequency change.

Since detuning of the segments causes cross-current flow, it would be expected that the system Q would be adversely affected by this detuning. Q values were measured under different tuning conditions, but with the measuring methods available it was difficult to detect any significant change in Q value.

As a result of above measurements it was concluded that a good electrical connection is required between the tips of the segments.

Resonator Segment Tuning Systems

A few measurements with trim panels of different size, and at different locations in the vicinity of the resonator tips, were taken. As no attractive solution to the actuator problem could be found, no extensive effort was expended on model studies of this concept. Instead, the model was modified so as to include two root trim panels, as shown in Figure 4.

Figure 17 shows the frequency change as a function of displacement of the root trim panel. By moving the trim panels in two or more segments simultaneously, it was determined that the effect on the frequency was cumulative.

For a discussion of actuators for the tuning systems considered, see Section 6 of this report.

Shape of Resonator Ends and Central Region Segments

In order to reduce the volume of the vacuum tank and thus reduce cost, it had been envisaged that the resonator end segments would have their corners at the root end chamfered. This causes an uneven current distribution within these segments and, as a result, a voltage at the tip which differs from that at the other resonator segments. It is possible, however, to change the impedance of the odd-shaped end segments in such a way that the desired voltage distribution is maintained.
Recent investigations have shown that "flat-topping" of the accelerating voltage would result in a highly desirable improved cyclotron performance. By superimposition of about 11% of third harmonic upon the fundamental frequency, an almost flat top may be obtained from -30 deg to +30 deg. Addition of this third harmonic is simple if the resonator segments are uniform in cross-section, i.e. have a constant characteristic impedance, which is not the case when the corners of the end segments are chamfered. Adjustment of the resonator volume, such that a good voltage distribution is achieved for the fundamental frequency, does not result in a good third harmonic performance. The use of diaphragms for tuning the end resonator segment to perform well for both the fundamental frequency and the third harmonic is being investigated.

A2. Mechanical Investigations

A2.1 Resonator Panel

The conductive layer can be very thin, 0.001 in., since RF current is confined to the outer skin of the conductor.

In spite of thorough cleaning of the resonators, a certain amount of surface contamination is expected to be present. This contamination will give rise to outgassing which may in turn result in arc-overs in highly electrically stressed areas. If the conductive skin is made too thin, electric arcs may remove enough skin material to expose the base material and hence increase heat losses, thermal distortion and frequency of arcing. It is therefore desirable to use a conductive skin with a thickness of at least 0.005 to 0.010 in. The thermal energy will be deposited in this thin layer, and for calculation purposes we have assumed a heat gain in the form of a thermal flux per unit area which varies as sine-squared, being practically zero at the resonator tip and rising to a maximum at the root.

It is desirable to have the coolant channels immediately adjacent to the conductive surface. Placing the coolant lines directly on top of the conductive layer is not attractive from an RF viewpoint. Placing them beneath retains the flat surface for the conductive layer while allowing a wide choice of coolant channel material. Major requirements for the latter are good thermal conductivity and low density.

It is also desirable that the coolant channels are so situated and spaced that they are compatible with the heat flux distribution. Attachment, by welding or brazing, of round or flattened tubing is not ideal, either from a heat transfer or from a manufacturing viewpoint.

A2.1.1 Roll-Bond Panel

Roll-bonding is the oldest of the common commercial methods of metal diffusion bonding of two metal sheets. Bonding of the two sheets can be avoided by the use of a parting medium applied in such a way as to form a cooling channel pattern.
Following the rolling process, the pattern is inflated with high pressure gas and, depending on the die in which the bonded panel is held, inflation can be shared by both sheets, or it can be confined to only one. Such a panel ideally meets the heat panel requirements of the resonator arms.

A2.1.2 Conductive Layer

If the conductive layer is to be deposited on an aluminum roll-bonded panel then several methods can be considered. Of the various processes available - hot dipping, metal spraying, hard facing, cementation coating, chemical deposition, mechanical plating, vapour deposition, metallurgical cladding, vacuum metallizing, and electro-deposition - only the last three processes were given consideration because of the unique requirements.

A coat thickness of 0.0005 in. is considered a heavy coating for vacuum metallizing. If thicker coating is attempted, crystallization is encountered and the deposited material is likely to peel off.

With electrodeposition, the resistivity of the plated material increases from its normal value by about 2.5% for copper or silver. In general, the electroplated material is denser at the substrate than at the outer skin and, since RF currents are confined to the extreme outer skin, electroplated conductive surfaces cannot be considered a first choice.

With metallurgical cladding or roll-bonding a skin of great homogeneity is obtained. For the panel discussed above a third conductive layer of material can be readily accommodated in the rolling process. Processors usually are not in favour of contaminating their aluminum rolling mills with copper or silver and, for this reason, a third aluminum layer is employed in the process. This third aluminum sheet is fully coated with a parting medium so that it can be readily separated from the conductive layer after rolling.

Of the two conductive skin materials that appear to be suitable for this process, as well as being acceptable from an RF viewpoint, silver offers the following advantages over copper:

RF power losses would be marginally lower (about 2.5%) with silver-coated resonators than with copper-coated resonators.

Corrosion of the conductive layer when exposed to the atmosphere should be less of a problem if silver is used. Silver oxides are good conductors, while the same cannot be said for copper oxides.

Cyclotrons employing silver-coated resonators made from roll-
bonded material have recently been built in Europe and this feature has proven very successful. Manufacturing experience with silver roll-bonded material is available, while it is not available for copper roll-bonded material.

While it is appreciated that the raw material cost for copper is substantially lower than for silver, it is not known at this time to what degree this would affect the cost of the coated roll-bonded material, since the copper-coated variety has never been produced with the coolant channel principle.

A2.1.3 Coolant Line Spacing and Temperature Gradients

For the ground arm and the hot arm of Concept IV, structural reinforcing will be intermittently welded to the back side of the panel. It is important that temperature gradients in the panel be minimized, as otherwise a certain degree of waviness will occur in the panel. The magnitude of these thermal gradients is quite independent of coolant medium, channel size or flow characteristics. Gradients in the skin will be solely a function of material conductivity, its thickness, and the spacing of the cooling lines.

As noted previously, for the case of parallel flat arms the RF currents vary with length as a sine curve. For more complex arm shapes such as Concepts III and V, current distribution was also calculated and considered in the following manner. The heat flux, in W/cm² or Btu/h ft², is a function of the rms current per foot of resonator width as well as the electrical characteristics of the material. Based on heat conductivity in the panel, temperature gradients can be established for a variety of heat flux values. Figure 18 illustrates anticipated temperature differences in the resonator skin as a function of coolant line spacing for the range of heat fluxes that will be encountered in the resonators. The graph is applicable to a laminated panel of 0.080 in. aluminum with either 0.010 in. copper or 0.005 in. silver.

A good, if somewhat arbitrary, choice of permissible temperature difference in the skin is 5°F, and this choice was initially adopted for the resonator arm design. For the maximum heat flux at the root, this temperature difference indicated a centre-to-centre dimension for the cooling channels of 2.5 in. with 0.5 in. wide channels. Drawings 60036 and 60038 indicate to what extent the coolant line spacing can be increased, progressing to the tip and maintaining a maximum skin temperature difference of 5°F.

In order to realize a low average operating temperature for the resonator panel, it was also decided to limit the coolant temperature rise to only 5°F. To achieve this it was necessary to have five cooling loops per panel and, with this arrangement, coolant channel size, coolant and pressure loss can be held to
a minimum. For a water velocity of approximately 10 ft/sec the pressure loss for each coolant loop amounts to 25 lb/in.². For all coolant loops the forced convection heat transfer coefficient is 2400 Btu/h ft² °F within ±5% approximately, yielding a ΔT between coolant and tube of about 1°F. The gradient in the tube wall itself is assessed at less than 0.1°F. The maximum metal temperature will therefore be 11°F above coolant inlet temperature. The average resonator temperature will be approximately 6°F above the coolant inlet temperature.

With respect to thermal gradients in the thickness of the arm, which could conceivably lead to arm curvature changes, it is concluded that the hot arm of Concept IV will not exhibit these except possibly during transient conditions. All attachment points between the resonator structure and the resonator panel occur at places midway between parallel coolant lines. These attachment points may not have exactly identical temperatures, but as heat transfer from the structure can only occur by conduction to or from the panel at these attachments, or to the environment in the form of radiation (and as the arm sees only other adjacent arms), it can be concluded that between attachment points the average temperature of the structure will equal the average temperature of the panel. Hence thermal distortion with this design can be considered negligible.

Very little thermal distortion is also expected with the hot arm of Concept V. Temperature gradients in the thickness direction of the intermediate portion, which constitutes the major portion of the hot arm, will be a function of the skin temperature in the valley of the corrugation. The judicious location of cooling channels in the corners of the corrugation ensures that the maximum temperature difference between the cooling channel and the hottest point in the valley will be less than 1°F.

For both cases tip deflection due to varying thermal distortions, i.e. changes in cooling water temperature, are extremely difficult to assess and can be determined accurately only by full-scale resonator tests.

Thermal distortion of the ground arm has not been discussed here but is expected to be negligible because the panel used is similar to that of the hot arm and the arm has numerous attachment points, particularly at the tip.

A2.1.4 Refrigeration

From an RF power loss viewpoint, a low temperature resonator would be attractive because ohmic losses could be reduced considerably. Such a saving would reflect in the RF power supply cost as well as in the yearly operating cost.
It is possible by employing commercial refrigeration to obtain temperatures as low as \(-100^\circ\text{F}\) with two or three stages of compression. For example, Freon 22 could be used in the first stage, with Freon 11 or 13 in the second stage. But, because of the required high compression ratios, the standard power requirement of 1 hp per ton of refrigeration capacity increases to about 6 to 7 hp per ton. Hence for any system operating much below \(-20^\circ\text{F}\) capital costs of a system increase almost exponentially.

An assessment was made of capital and yearly operating costs based on electric power and energy demand only, when incorporating a refrigeration system. Simultaneously, an assessment was made of the reduction in these costs when operating the resonators at lower temperature, with resulting lower ohmic losses. It was found that the reduction in RF power did not compensate for the much higher cost of the refrigeration system. This was found to be the case for both initial capital cost and for yearly operating costs.

A2.1.5 Cryogenic Cooling

Cryogenic cooling of the resonators appears attractive at first sight because it solves the problem of resonator material outgassing as well as offering potential capital saving in RF power cost and reduced operating costs. Brief investigation showed that the removal of about 200 kW of heat, this being approximately the ohmic loss at cryogenic temperatures, with a cryogenic coolant is extremely expensive. For a yearly operating time of 5,300 hours 15,000 tons of liquid N\(_2\) would be required, costing about $226,000 per year; with CO\(_2\) about 60,000 tons or $1,200,000 would be required per year. These are estimates based on the latent heat of vaporization of the coolants, i.e. cryogenic conditions would be maintained by 'boiling off' these quantities per year.

Alternatively, the gas could be collected and recompressed. Because of the quantity involved it would not be attractive to return the gas to the supplier for recompression. To recirculate the gases by having a refrigeration plant on site would mean a capital expenditure of about two to three million dollars.

With a possible capital saving of about $800,000 in RF power equipment it is clear that cryogenic cooling is not attractive from a cost viewpoint.

A2.2 Ground Arm

The ground arm design concept can be quite independent of the hot arm, but in designing the Concept IV arm it was possible to arrange for these two arms to employ the same panel.
A2.2.1 Mechanical Design

The ground arm attaches directly to the vacuum tank cover (or bottom) with the conductive surface parallel and at a mean distance of 0.75 in. from it. Since the tank plates will exhibit a certain degree of waviness and out-of-flatness, in addition to deflections under load, it is imperative that all ground arm attachment points be confined to tie rod mountings. Drawing 60036 illustrates only six of these attachment points. At the root the ground arm panel attaches to a cross beam, which in turn mounts directly on the tank lid. The root edge is notched to provide clearance for the beam attachment pads.

The tip of the ground arm panel, a portion approximately 30 by 30 in. square, is supported on all four corners. This support concept will permit accurate alignment not only of the extreme tip but also of the immediately adjacent panel portion. By raising or lowering the tie rods it will be feasible to align the ground arm tips of all the segments constituting a row, so that they form a straight line. Tie rod adjustment, and hence resonator tip positioning, to an accuracy of 0.002 in. is possible.

Six stiffening bars run the length of the ground arm panel and these are intermittently attached to the panel by spacers. By placing the spacers in a valley, the flat portion between coolant channels, welds are confined to the heavier portion of the panel, i.e. 0.080 in. thick material, instead of the thinner coolant channel wall which is only 0.040 in. or less. In the vicinity of attachment points, additional cross members provide stiffness in the width direction. With this bracing attached to the panel, the deflection between supports is estimated at 0.005 in. for aluminum construction.

The exact shape of the arm tip is not final and on Drawing 60036 a round tube of 1.5 in. diam is indicated. The arrangement will be subject to the outcome of further model tests.

A2.2.2 Support Concept for Bakeout

It is required that outgassing of the resonator surfaces be accelerated by baking, so that outgassing during cyclotron operation is minimal. An early vacuum analysis suggested that the ability to bake out at 200°C is desirable, and the resonators and the design of their support systems reflect this requirement. Bakeout temperature for the tank has, however, been limited to 140°C, and it is probable that this will also be the normal bakeout temperature for the resonators.

The ground arm concept shown in Drawing 60036 will allow use of a 200°C bakeout temperature if desired. With the heat panel fixed at the root thermal expansion movement at the tip, assuming the panel to be aluminum, would be of the order of
0.5 in. For the nominal 32 in. spacing of the tie rods, and resonator panel width, the expansion in width will be about 0.125 in. These are maximum values during transient conditions, and of course the assumption is made that the relatively light resonator structure responds rapidly in heating while the much larger mass of the vacuum tank responds more slowly. At steady-state conditions, differential thermal expansion between resonator and tank will be much less and will be a function of the relative expansion coefficients of aluminum and stainless steel, as well as the mean temperature difference between the two materials. Nevertheless, rigid attachment of the ground arms to the vacuum tank is unacceptable since it will obviously mean buckling of the ground arm during bakeout.

In order to cope with these relatively larger thermal expansions, most of the supports employ a sliding block arrangement. At the root one edge of the ground arm is fixed, while the other edge is allowed to expand; the expansion joint in this case is incorporated in the root piece structure. At the tip one support limits movements to the length direction, while all other supports are fashioned so as to accommodate displacements that are resultant of the width and the distance from the root.

Two hot arm concepts that employ roll-bond material have been considered in some detail. Concept V is more attractive from an RF viewpoint, i.e. large volume and lower power requirements while retaining all the desirable physical characteristics of Concept IV, i.e. low mass, low deflection, and low thermal distortions. Concept V is, however, an arm of far more complex design, requiring roll-bond shaping and intricate welding. While it appears remotely possible that the arm could be fabricated, the final choice must depend upon fabrication cost and the reliability of the numerous welded coolant channel connections. Clearly, Concept V will be more expensive, probably requiring development of a fabrication technique.

Figure 19 shows how the power losses vary with the resonator gap. If Concept V were adopted, a power saving of about 180 kW would be realized. Expressed in dollars this would result in a saving of $180,000 at a RF power unit price of $1.00 per watt.
minimize weight and deflection, the tip portion has no structural stiffening other than that provided by the coolant channels and by the vertical sides of the panel (not shown in Drawing 60038). Stiffness at the extreme tip is provided by the 1.5 in. rounded portion.

Table I gives a summary of the physical characteristics and properties - weight, tip deflection, stiffness, etc.

All of the resonator arm characteristics so far discussed have been based on aluminum as the structural material. This has the obvious advantage of low inherent arm weight, which is important when handling such a large and relatively fragile component. Installing and servicing resonator components will require a well-formulated procedure and in all likelihood will require powered accessories for lifting and positioning. Minimizing the arm weight was considered a desirable feature, since it would also simplify these operations.

It is possible to construct the resonator arms by employing all-copper roll-bond panels and, in this event, no additional conductive coating would be necessary. At the same time it would be desirable to make the I-beams and the fabricated root portion of copper, since this would simplify assembly and would also avoid differential expansion problems. Such a design was briefly considered.

With identical panel thicknesses and I-beam moment of inertia values it may be expected that the hot arm tip deflection for copper construction will be twice that for aluminum, while the total weight of the arm will be approximately tripled.

The advantages of employing copper are higher thermal conductivity and hence fewer coolant channels, as well as lower expansion during bakeout because of its lower coefficient of expansion. In addition, copper lends itself a little more readily to welding and therefore some saving in manufacturing cost might be achieved. These advantages are small, compared to the disadvantages of greatly increased weight and deflection, and aluminum is therefore adopted as the resonator material.

A2.3.2 Mechanical Design - Concept V

Drawing 60032 illustrating Concept V of the hot arm indicates that particular attention has been paid to the numerous cooling channel connections. The concept of three distinct unit weight and structural stiffness regions has been retained in this design. The intermediate portion consists of a roll-bonded panel corrugated in a judicious shape that satisfies the RF requirements and gives good inherent structural stiffness without the aid of reinforcing. Cooling channels have been located at surfaces of maximum heat concentration, i.e.
at the edges of the lands. The transition regions at tip to intermediate portion, and at intermediate portion to root portion, give rise to great complexity because they involve structural connection, cooling line connections and RF electrical surface connections. Structural stiffness at the root is provided by solid bars which extend into the valleys of the corrugation.

This concept does exhibit good structural stiffness when compared with Concept IV (see also Table 1). Again, these figures are representative of aluminum construction only.

Investigations to date have shown that it is quite unlikely that a roll-bond panel can be shaped in the form required. It appears possible, however, to construct the corrugated portion from a copper sheet and to braze the round cooling tubes in the corner of the corrugations. It can be postulated that a thinner panel could be employed and that the Concept V arm weight in copper would weigh less than the Concept IV copper arm.

A considerably unattractive aspect of this approach is the requirement for continuous welding or brazing of the cooling line attachment at the ends of the arm. To avoid serious distortion during fabrication, a furnace brazing process similar to that used for automotive radiators would have to be developed. Due to the limited quantity of panels required, a furnace brazing process would be very expensive.

A2.4 Root Piece

The root piece design shown in Drawing 60039 is intended for the Concept IV hot arm. The main structural component is a stiff double web beam which supports the roll-bond panel and the trimming device, and which also lends support to the hot arm. The beam is rigidly attached to the vacuum tank at one end while at the other end a flexible support is provided which will tolerate as much as 1/8 in. differential expansion between beam and tank during bakeout. This support concept is detailed in Section E-E and view D of the above drawing.

It is important that the flexible support be compatible with the 'pin-joined' connection requirement to allow for adjustment of the hot arm. The hot arm structure is connected to the beam by three bolts. The bolts are shaped so that they are relatively long and slender in order to give negligible bending stiffness. By reducing the contact surfaces between beam and arm to a minimum, a pin-joined connection between the two members is realized.

The structural connection of the ground arm is accomplished by bolting or welding the reinforcing straps to the top flange of the beam. Note that both hot and ground arm heat panels must be notched in order to clear the flexible support of the beam.
The electrical connection between the ground arm panel and the root panel is made by low temperature silver solder. After soldering, the ground arm and root piece effectively become one assembly and are installed as such in the tank. The electrical connection between the hot arm and the root panel depends on contact pressure provided by a bolted or screwed connection. To improve this connection a soft silver wire is placed in a V-notch and this wire will deform sufficiently to ensure a continuous connection over the width of the resonator. Alternatively, it may be possible to accommodate a square coil spring in this connection.

The root piece has its own cooling panel with separate coolant inlet and outlet. The complex shape is necessary because of the provision for two trim panels which interrupt the 32 in. width. The merits of the trim panel arrangement are discussed in Section 6.5.3, while the cooling arrangement will be discussed briefly here. Direct cooling of the trim panel by attachment of a roll-bond panel is undesirable because of the required trim panel travel of ±0.25 in. Any coolant line connections to it would be subject to continuous flexure. Because of the vacuum environment, high temperature bakeout requirement and radiation damage, plastic or rubber hose is unacceptable and even flexible metal hose is unsuitable.

A more practical solution is to cool the trim panel by radiation. This means that the trim panel surface will reach higher temperatures than the surrounding metal surfaces, but even supposing that the heating rate for the recessed panel is 50% that of the adjacent root (a pessimistic assumption) it is not expected that the trim panel surface will reach temperatures higher than 130°F. Certainly the average temperature for the panel will be less than this. The panel itself will be silver-plated copper, while the rectangular bellows will be silver-plated beryllium copper.

### A2.5 Levelling Arm

The levelling arm is a structural extension of the hot arm exhibiting somewhat identical stiffness to the root portion of the hot arm. Its length has been chosen so as to cover two tie rod spacings from the root of the resonator (see Drawing 60044).

It appears feasible to adjust tie rod position to an accuracy of 0.002 in. by use of a special torque wrench. With the spacing chosen it should therefore be possible to adjust the hot arm resonator tip to 0.004 in. Greater accuracy could be achieved by increasing the length of the levelling arm, but this has the disadvantage of increasing tip deflections unless the stiffness of the levelling arm is proportionally increased. This change is undesirable since it increases the weight of the hot arm assembly as well as encroaching farther into the limited available space in the vacuum tank.
A2.6 Mechanical Vibration

A2.6.1 Nature of Vibration Stimulus

Generally with RF cavities of high Q, where electrostatic or electromagnetic flux changes result in forces that cause bending moments, vibrations in panel sections will occur unless there is sufficient stiffness or mechanical damping to absorb the stimulus energy. The most susceptible areas are those with very low structural stiffness factors, where minute panel dimensional changes permit cavity detuning at rates related to the natural period of the panel section. Changes in force or moment can be at the frequency of the panel.

Examination of a typical high Q resonance curve in relationship to mechanical tuning changes will show the changing voltages that can produce stimulus energy. For whatever displacement that is assumed, it will then be necessary to show that somewhere, either in the structure or in a damping device, there is more energy absorbed than that causing stimulation.

Referring to Figure 20, three cases of tuning are presented for a simple cantilever beam and panel suspension. Case (a) shows that at the condition of resonance displacement from point (b) to (a) or (c) produces within one normal cycle two changes in energy in the same sense or direction so as not to cause vibration at the fundamental.

In Figure 20 (b), where detuning from resonance is represented, two energy pulses are produced per cycle and are in phase with each direction of the cantilever beam deflection, causing vibration.

In Figure 20 (c), where detuning is only marginal, there is still a net amount of energy, leaving enough to cause stimulus once per cycle.

The peak value of the steady-state force in the tip region of the panel at resonance is $3.1 \times 10^{-4}$ lb/in.$^2$ of panel. This varies as the square of the voltage, and for a Q of 5000 the resonance voltage profile is as shown on Figure 21.

When power is first applied, or when there is a resonator spark-over, there is a change in voltage which subsequently will cause the resonator segments to oscillate. There are several factors to be considered or steps that can be taken to reduce sustaining oscillation problems. The first is related to the degree to which each resonator segment is electrically connected to its neighbours. Tests on the half-scale model have shown that, when prorated to the full-scale resonator, less than 0.01% voltage change occurs across a resonator row of five sections when one panel is deflected 0.010 in. This
figure of 0.010 in. is typical of the maximum segment gap change expected from thermal distortion effects. The same figure if applied to all panels simultaneously would result in approximately 60% voltage loss for a whole system Q of 5000. If the panels are not electrically connected, voltage distribution and Q are so bad as not to be considered at all.

Thus a single panel, mechanically isolated but joined electrically, would develop no more than 0.01% voltage change and in turn no more than 0.0001% change in the forces on all panels.

This suggests that anything that can be done to prevent the panels from moving in synchronism is highly desirable. The premise is that it is not feasible to constrain all panels by increasing their stiffness and so inhibit oscillation. Some constraint of the hot arm panels near the central region and the end flux is possible. This permits a reduction of two panels out of a group of ten that can no longer move and subsequently cannot contribute to the average dynamic panel force or stimulus gradient. It would be desirable to extend mechanically stiff connections to further panels, but in practice it is not feasible to have more than two at each end of a half-row of resonator segments. This means that of the group of ten having the force gradient previously stated it can now be reduced by a factor of four tenths.

This situation now leaves a total of six panels in each half-row of segments that are to be considered for vibration purposes. Further, as each half-row is electrically close-coupled to the remainder, there is the basic consideration of what will happen with 20 to 28 panels out of the total of 40.

Neglecting all but internal structural damping, it is worth considering what happens upon the application of a transient. Assume that it is possible to arrange a natural frequency distribution from panel to panel in pairs above and below the average value of the frequency. Within a short period of time those panels that are tuned below frequency lag by close to 90 deg and similarly those tuned above lead by close to 90 deg. This then results in certain probabilities that at least pairs of panels will be approximately 180 deg away from each other. The effect of this is one in which there is a capacitance increase offset by a similar decrease and in turn causing no significant change in the RF resonance tank circuit voltage.

This situation can be carried still further where, on a probability basis, additional pairs arrive in 180 deg phase opposition. Thus there can be a successive cancellation of stimulus effect, and shortly thereafter internal damping will cause a general reduction of amplitude.
Once this stimulus is removed, and should there still be energy in some of the vibrating panels, they will then revert to their natural resonance periods. Subsequently, there will be the case where a sufficient number statistically arrive in phase, thereby causing the stimulus to commence again. Subject to the combination of energy left, repetition will occur again. Eventually the process must die out as a result of internal damping absorbing the energy in the panels.

A2.6.2 Inherent Magnetic Damping

An investigation was made into the possible aspects of eddy current damping. Throughout the magnet pole pieces there are magnetic anomalies with respect to the vertical and horizontal portions of the panel structure that it can vibrate in. This is generally confined to the azimuthal areas between the valley and edge of the pole.

The results of such damping are insignificant. If it were possible to realize a change of 6 kG per inch, the maximum possible amount of energy lost per cycle could not exceed \(1.6 \times 10^{-14}\) lb-in. per square inch of panel, some nine orders less than the driving stimulus.

A2.6.3 Natural Decay of Vibration

As noted above, when the RF power is first switched on the resonator hot arms are subjected to electrostatic forces which cause simultaneous initial deflections of the resonator tips in a direction which reduces the gap between hot and ground arms. These initial electrostatic forces set the resonators into mechanical vibration and are then replaced by electrodynamic forces initially having 7 to 8% of the magnitude of the initial ones. These cyclic forces are tuned to the average mechanical frequency of the resonators and are in phase with the average value of resonator displacements. Their initial magnitude at the resonator tip is \(2.5 \times 10^{-5}\) lb/in.\(^2\) averaged over positive and negative amplitudes, and the magnitude varies as cosine-squared for distances measured from the tip to the root.

The foregoing assumes that the resonators are vibrating in synchronism and the average amplitude of a group of electrically connected resonators equals the amplitude of any one resonator at time \(t\). This is not true for physical reasons. The resonators vary in stiffness and hence natural frequency due to unavoidable differences in dimensions, material selection, and assembly tolerances. As a result, a group of resonators that start moving in synchronism drift out of phase with time although vibrating at the average frequency of the forcing term. A situation develops where a group of resonators, each mechanically independent, vibrates at the same average
frequency, but their individual motions are phase-shifted due to the difference between their natural frequency and the average taken by the forcing term.

The cyclic force applied to such an electrically coupled group is a function of the average displacement of the group at any time t up to an average displacement of 0.010 in., at which point the force is taken to remain constant. To confirm the existence of periods of negligible applied force caused by phase shifts in resonator amplitudes and to find their duration, calculations were performed on groups of 10 resonators assuming various amounts of frequency detuning. In all cases, 8 of the 10 resonators are considered free to vibrate, the other two are assumed rigidly constrained, and the average forcing term is factored linearly to account for this.

The following cases are considered:

**Case 1:** All resonators in the group are assumed to have one natural frequency of 5.17 cps. Four degrees of critical damping are considered: 1%, 5%, 10% and 20% of critical.

**Case 2:** Resonators are assumed to have natural frequencies normally distributed about the mean value. Plus or minus two standard deviations about the mean represents ±3% change in the natural frequency. The results are shown for five values of critical damping: 1%, 2%, 5%, 10% and 30% of critical.

**Case 3:** Resonator natural frequencies are evenly grouped at two values above and below the average frequency of 5.17 cps. Frequency values used in the analysis are separated by ±1.25%, ±3%, ±5%, ±6.5%, ±8%, ±10%, ±15% and ±20%. The calculations are performed at various values of damping.

The results of all calculations are summarized in the following table, where the average steady-state amplitude of resonator tip vibration is given:
TABLE A.1
STEADY-STATE AMPLITUDE VALUES (in.)

<table>
<thead>
<tr>
<th>Analytical Cases</th>
<th>Per Cent of Critical Damping Values</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1.0</td>
</tr>
<tr>
<td>Case 1</td>
<td>0.38864</td>
</tr>
<tr>
<td>Case 2</td>
<td>0.20603</td>
</tr>
</tbody>
</table>
| Case 3
| Frequency Spread| 2.5% | 0.14369 | 0.13975 | 0.07278 | 0.03338 |
|                 | 6.0  | 0.03899 | 0.05980 | 0.05720 | 0.03127 |
|                 | 10.0 | 0.0    | 0.02436 | 0.03890 | 0.02762 |
|                 | 13.0 | 0.0    | 0.0     | 0.02610 | 0.02456 |
|                 | 16.0 | 0.0    | 0.0     | 0.01530 | 0.01951 |
|                 | 20.0 | 0.0    | 0.0     | 0.0     | 0.01062 |
|                 | 30.0 | 0.0    | 0.0     | 0.0     | 0.0     |
|                 | 40.0 | 0.0    | 0.0     | 0.0     | 0.0     |

The results from Table A.1 are plotted in Figure 22(a) and (b). Figure 22(a) shows the effect of percentage critical damping on the steady-state tip amplitude for the various resonator cases considered. Figure 22(b) presents amplitude as a function of frequency spread representing the degree of detuning. These results apply to the two-frequency situation represented in Case 3, and each curve represents a different percentage of critical damping as frequency spread is changed.

From these graphs several conclusions can be drawn. Increased damping of the resonator reduces steady-state tip amplitude at low levels of detuning. Natural damping due to internal friction and structural damping will be present in all resonators, and this value is between 1% and 5% of the critical damping value.

Tuning the resonators away from the average natural frequency reduces tip amplitudes in all cases. The reduction caused by natural detuning (Case 2 - normal distribution) is too small to be considered effective. A study of the frequency-spreading effect resulting from deliberate detuning shows that the steady-state tip amplitude can be reduced effectively to zero when enough detuning is used. For example, deliberate resonator
detuning to give 20\% frequency spread reduces amplitudes to zero, zero, zero and 0.0106 in. for 1, 2, 5 and 10\% of critical damping, respectively. Zero displacement in the resonator position at 5\% or less of critical damping indicates the resonators should not have steady-state mechanical oscillations. Natural damping of less than 5\% is probable for the resonator structure.

It can be concluded that deliberate detuning of resonators having 5\% or less inherent damping into two groups with a natural frequency spread of 20\% will reduce steady-state mechanical vibrations to such a low level that the RF characteristics of the resonator will not be affected.

Resonator detuning may be carried out by adding weights to the tips of half the free resonators. The weight required to provide 20\% detuning must be found experimentally, but a value of 7 lb can be used as an initial estimate for a 32 in. wide resonator.

A2.6.4 Active Damping Devices

Active resonator vibration damping devices were investigated and their feasibility evaluated. Three types of active damping devices were considered: a direct viscous damper of large capacity located on the hot arm support structure in the 64 in. spacing between resonator supports, a damped dynamic vibration absorber located at the resonator tips, and constrained-layer damping using an elastomer damping device.

Simple Viscous Damper

A simple viscous damper may be located midway between the two pinned supports of each resonator. The optimum value of damping limits the maximum steady-state tip deflection to ±0.009 in. The required viscous damping constant is 300 lb sec/in. per resonator. The vibration amplitude at the damper is only a small fraction of that at the resonator tip and this small movement, combined with the large damping required, makes the design of such a damper impractical. In addition, the minimum tip amplitude of 0.009 in. is unacceptably large, so that this concept is rejected.

Damped Dynamic Vibration Absorbers

The addition of damped dynamic vibration absorbers to the tip of the resonators will, when properly tuned, reduce the resonator steady-state amplitude under cyclic forces to a small periodic value. The behaviour of a dynamic absorber is a function of the absorber mass relative to the beam mass, the absorbing spring rate relative to the beam spring rate, and the degree of critical damping used between the absorber mass and the beam.
With an absorber-to-beam-mass ratio of 0.2 the tip deflection can be reduced to a minimum value of ±0.002 in. Critical damping ratio of 0.21 and a frequency rate of 5/6 are needed to attain this small value of steady-state deflection. Under these conditions the transmissibility (gain) equals 3.3, i.e. the amplitude of the damper is 3.3 times that of the resonator.

Damped absorbers therefore provide a means of controlling resonator tip vibrations which can be maintained at ±0.002 in.

A damped absorber of the above properties appears feasible within the space limitations at the resonator tip. An alternative possibility is a smaller absorber having a \( m_2/m_1 \) ratio of 0.1. In this case, the resonator tip motions reach a maximum of 0.0025 in. for an optimum design.

**Constrained Layer Damping**

This technique provides dissipation of mechanical energy in the form of heat generated by the physical distortion of a layer of visco-elastic material sandwiched between the vibrating structure and a thin metal constraining layer. The energy is absorbed by shear distortion of the visco-elastic material during vibration.

Elastomers and their related families of long-chain molecules provide the only practical source of these visco-elastic materials. Unfortunately, the great majority of these materials has a practical lower limit of frequency that is more than one order of magnitude higher than the resonator first natural frequency. Also, radiation damage to these long-chain molecules is severe and their desirable properties are rapidly lost.

For these reasons, such a form of damping is not considered feasible.
Figure 1. RESONATOR TERMINOLOGY
Figure 2. REFERENCE DESIGN DIMENSIONS
TIE RODS FOR LINING UP GROUND ARM TIPS

THESE TIE RODS ARE MAJOR SUPPORT POINTS FOR RESONATOR

32" TYPICAL

INTERMEDIATE SUPPORT POINTS

THESE TIE RODS TO BE USED FOR COARSE ADJUSTMENT OF RESONATOR GAP

Figure 3. PREFERRED RESONATOR SEGMENT SIZE AS Dictated BY SUPPORTS
Figure 4. ROOT TRIM PANELS
Figure 5
RESONATOR CONTROL SYSTEM

When single RF transmission line used

Differential Pneumatic Actuator
Trim Panel
Voltage Pick-up
Flux Loop
90 deg Phase Delay
Phase Detector
Bias of 1-2 deg Phase

Electro-Pneumatic Transducer
1-5 mA, 500 ohm
3-15 psi
15-3 psi

Pneumatic Bias = 9 psi

Integrator

Bias of 1-2 deg Phase
When single RF transmission line used
Figure 6. RESONATOR COOLING AND BAKEOUT SYSTEM
Figure 7. FLUX PLOT FOR RESONATOR CONCEPT III
Figure 8. FLUX PLOT AT RESONATOR TIP
Figure 9. POWER DISSIPATION vs ELECTRIC ANGLE
Figure 10. RESONATOR LENGTH AND POWER CONSUMPTION vs RESONATOR GEOMETRY
Figure 11. RESONATOR SEGMENT TRIM PLATE ARRANGEMENT
Figure 12. VOLTAGE DIFFERENCE vs HOT ARM DEFLECTION. 1/2 SCALE MODEL
Figure 13. CAPACITANCE vs TRIM PLATE POSITION
Figure 14. FREQUENCY vs TIP CAPACITANCE
Figure 15. RESONATOR SEGMENT VOLTAGE vs HOT ARM DEFLECTION
Figure 16. RESONANT FREQUENCY vs HOT ARM DEFLECTION
Figure 17. FREQUENCY CHANGE vs DISPLACEMENT OF TRIM PANELS AT RESONATOR ROOT
Figure 18. SKIN TEMPERATURE DIFFERENCE AS A FUNCTION OF COOLANT LINE SPACING
Figure 19. RF POWER AS A FUNCTION OF RESONATOR SHAPE AND GAP
Figure 20. RESONATOR PANEL STIMULUS EFFECTS
b) OFFSET TUNING

c) PARTIAL OFFSET TUNING
Figure 21. FORCE/VOLTAGE vs DISPLACEMENT FOR RESONATOR PANELS
CASE 1.

8 Resonators - Mech. Locked
All Freq. = 5.17 CPS.

CASE 2.

8 Resonators = Mech. Free
8 Nat. Frequencies of Normal Probability
5.015, 5.050, 5.102, 5.148, 5.192, 5.237, 5.281, 5.325 CPS
Prob.: .03275, .08525, .16107, .22092, .22092, .16107, .08525,

CASE 3.

8 Resonators - Mech. Free
2 Nat. Frequencies 5.102 and 5.237 (2.5% Difference)
Prob. 5.

CASE 3.

8 Resonators Mech. Free
2 Nat. Frequencies 5.015 and 5.325 (6% Difference)
Prob. 5.

CASE 3.

8 Resonators Mech. Free
2 Nat. Frequencies 4.9115 and 5.4285 (10% Difference)
Prob. 0.5