RCA Engineer

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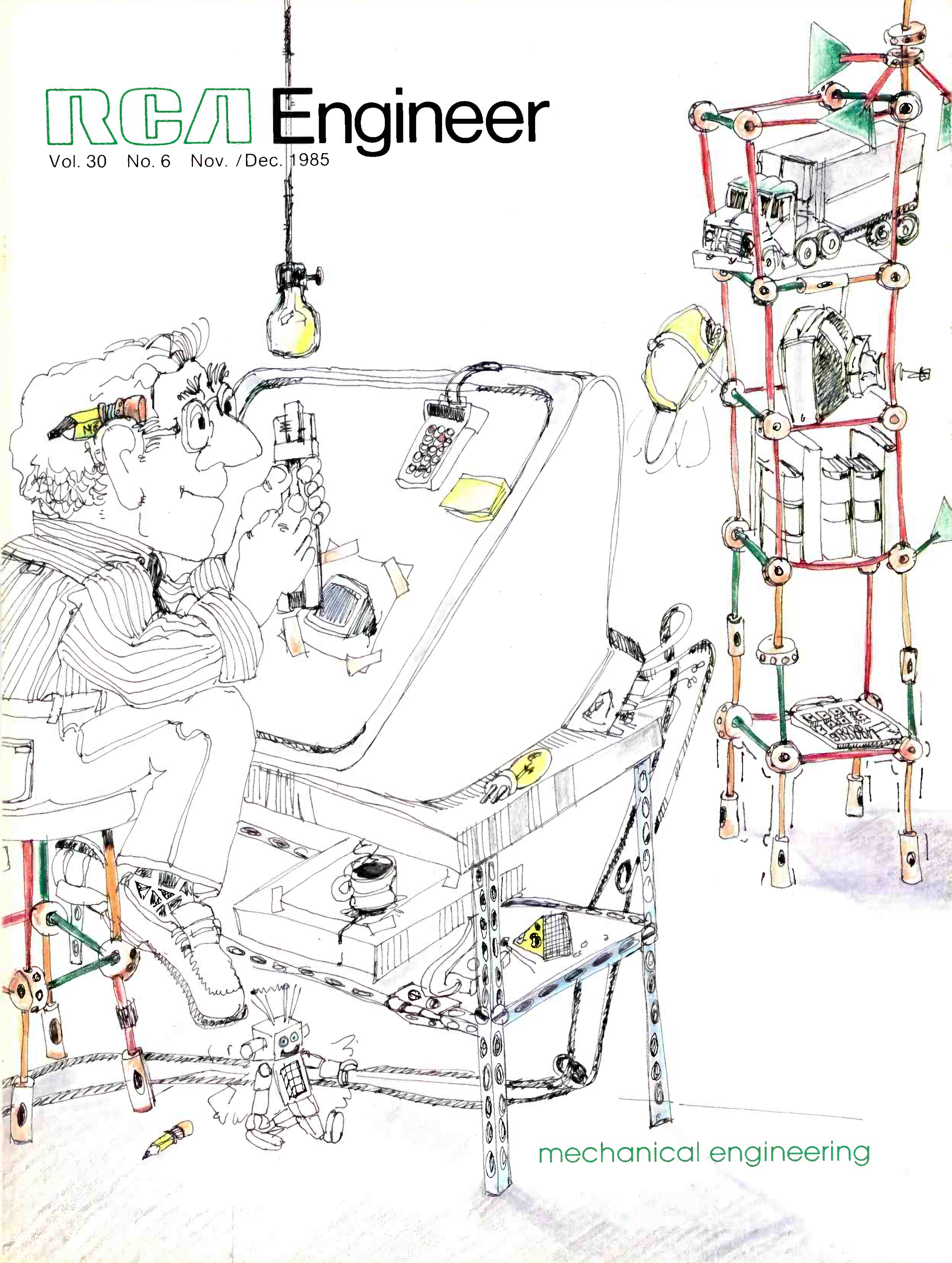

mechanical engineering

RCA Engineer

A technical journal published by the
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Does our cover look at all familiar? In the midst of the flashing lights, marching robots, screeching whistles, and tiny televisions the crucial business of manufacturing is going on. It's all there, from CAD to product, complete with ECNs.

Our theme in this issue is mechanical engineering, and if we've exaggerated things a bit in out depiction of an ME, it isn't by much. Engineers are tinkerers, and tools are a tinkerer's toys. It may be work, but it's fun, too.

Cover illustration by **Nina Sampsel**

□ To serve as a medium of interchange of technical information among various groups at RCA □ To create a community of engineering interest within the company by stressing the interrelated nature of all contributions □ To disseminate to RCA engineers technical information of professional value □ To publish in an appropriate manner important technical developments at RCA, and the role of the engineer □ To help publicize engineering achievements in a manner that will promote the interests and reputation of RCA in the engineering field □ To provide a convenient means by which the RCA engineer may review professional work before associates and engineering management □ To announce outstanding and unusual achievements of RCA engineers in a manner most likely to enhance their prestige and professional status.

Teamwork in innovation—an ME tradition

RCA has rightfully earned a reputation for ingenuity and inventiveness in product design. Our innovations in circuit design, solid state processes, television picture tubes, satellite communications, audio and video home products, and advanced military systems immediately call to mind the high caliber of our electronic and electro-optical engineers. Working hand-in-hand with these innovators are our mechanical engineers, who have been the source of many of our most advanced engineering accomplishments.

The articles in this mechanical engineering issue represent contributions from many divisions. They illustrate the mechanical engineer's creativity, which has been an essential ingredient—and often the forcing function—in many new products. The demand for increasingly compact devices, the development of new and exciting materials, and the need for operation over long periods of time under extreme environmental conditions all stimulate the imagination of our mechanical engineers, who continue to meet these new challenges. Our future is dependent on their ability to continue to excel.

Our dependence on mechanical engineering emphasizes the need for aggressive recruitment of MEs on college campuses. Further, we must nurture and encourage them with continued training and recognition during their RCA career. They are an integral, indispensable part of the RCA engineering community—a community dedicated to technical excellence.

E M Stockton

Eugene M. Stockton
Division Vice-President and General Manager
Automated Systems Division

RCA Engineer

Vol. 30 | No. 6 Nov. | Dec. 1985

mechanical engineering

. . . in manufacturing

. . . and in design

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general interest

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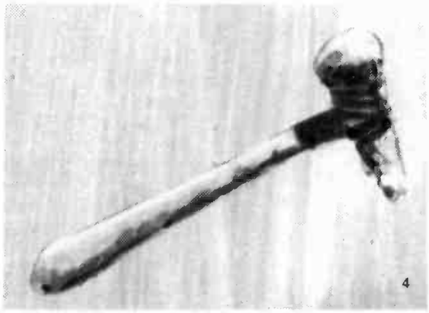
departments

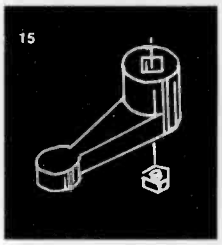
in this issue . . .
mechanical engineering

- **Leedom:** *"Some very early signs of mechanical engineering were found in caves belonging to Neanderthal man, who lived approximately 35,000 to 100,000 years ago."*

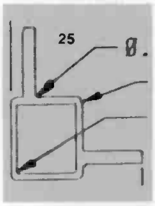


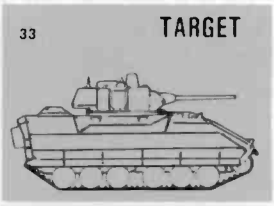
- **Poux/Carroll/Rearick:** *"Engineers who handle incremental motion control problems generally have half of the solution before they begin to solve the problem."*
- **Whitley:** *"There is no doubt that systems that employ fixed and dedicated tooling do provide the greatest consistency and reliability, and the two-plate system provides both . . ."*
- **Reid-Green:** *"An Air Force officer finally came up with the idea of sending back to home base those planes unable to land on the first try, and the scheduling problem disappeared."*

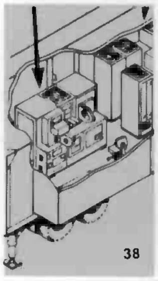
9 *Motion curves*

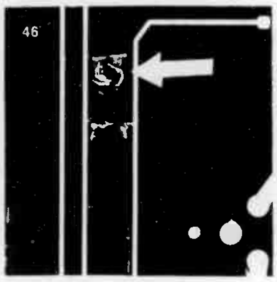


- **Cahoon/Meliones/McDermott:** *"All 49 common node locations were merged together, module weights added as lumped masses, and a dynamic analysis performed to determine natural frequencies."*
- **Guyer:** *"The first system consisted of two rather large electronic chassis with a combined weight of one-hundred pounds and required a two-man rangefinder team to operate the equipment."*
- **Ghostlaw/McNamee:** *"Roadside and curbside expansible sides are 25 ft long, opened simultaneously by electric motor or manual drive to a total span of 14 ft, 6 in."*
- **Holden:** *"A research project was initiated to determine what was causing the inconsistent and inferior urethane conformal coating, and to find an acceptable process method for producing conformal-coated GRPTFE boards."*

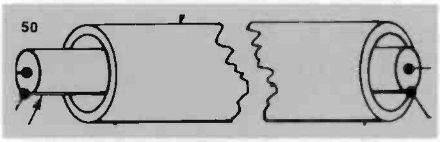


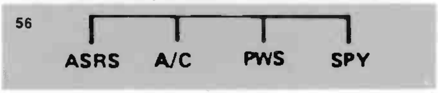






- **Weiss:** *"We estimate that the new transmission method will save up to 3000 pounds of cabling per weapon system, an important reduction for ship stability, because most of the cabling is above the ship's center of gravity."*
- **Felbinger/Rouland:** *"In its expanded role as system integrator or coordinator, the Functional Analysis activity has solidified the relationship between system engineering and support disciplines."*





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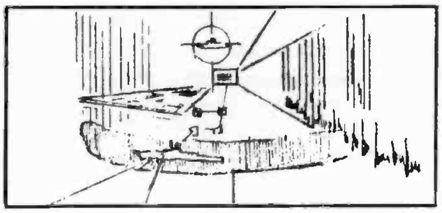
technology for the 1990s
PC workstations on the job
laser technology



M.A. Leedom

The oldest profession . . .

Or maybe the second oldest . . .

The mechanical engineer at RCA plays an essential role in the success of new product and manufacturing designs. Mechanical engineering can be found in the Astro-Electronics, Video Component and Display, and Consumer Electronics Divisions and in RCA Laboratories. The products and designs produced by these areas reflect the influence of the mechanical engineer.

The oldest profession

If you define mechanical engineering as that branch of engineering that deals with motion, force, the production of power and products, and tools that manufacture other products, the profession has been with us for a long time. Why, even the use of a fig leaf as an article of clothing could be considered a mechanical engineering task. Some very early signs of mechanical engineering were found in caves belonging to Neanderthal man, who lived approximately 35,000 to 100,000 years ago. It was the design of a hearth for protecting the fire and cooking food. Wherever graves or living areas of ancient man were opened, the evidence of human inventiveness was always present. Early man was an incurable tool builder. His tools helped him to cook, eat, work, play, hunt, worship, and fight. Today, we build tools that perform exactly the same functions. We even use the same materials, although we have added quite a few new ones to the list.

Abstract: *All fields of engineering today are, by necessity, closely related. To design a new machine, for example, requires the expertise of people in the fields of electronics, chemistry, computer science, materials engineering, and mechanical engineering. A company can create effective and efficient products and manufacturing processes only by assuring that all engineering areas work together.*

Final manuscript received August 29, 1985
Reprint RE-30-6-1

The field of mechanical engineering evolved from the work of the artisans of early civilizations. These workers were similar to the tradespeople of today in that they made new products and tools. However, they were very poorly prepared for their work because of a lack of knowledge of the laws of the natural sciences and materials. As the study of science developed, the different branches of engineering and sciences grew. One of these branches became mechanical engineering.

The relationships among the various engineering disciplines changed radically during the last few thousand years. In early times, there were no identifiable engineering fields. A few ingenious people used natural materials to fashion useful tools or goods for their families. These early technologists later separated into special interest groups, each of which be came a specific engineering field. Mechanical engineering was formally recognized as a field in Birmingham, England in 1847, when the Institution of Mechanical Engineers was formed. The field was essentially pure at that time, involving motion, force, tooling, and the production of power. Today, the design of a mechanical machine involves so much more, including electronics for controls, chemistry for adhesives or corrosion control, computer science for CAD/CAM work and control, materials engineering for material selection or invention, and mechanical engineering for the overall machine design. Modern-day engineering in any one field requires a close relationship with many other fields.

Through a separate path of development, the forebears of our present-day research scientists were studying and developing the branches of the natural sciences. These early scientists were representatives of a highly educated social class with little interest in such working-class developments as the potter's wheel or a farmer's tool. It was not until the nineteenth century that the engineering world and the scientific world began to look to each other for help. The early engineers realized that scientific knowledge was vital because inventions were becoming more and more complex. Likewise, early scientists welcomed engineering assistance in building apparatus for their experiments.

Since that time, mutual assistance has been a major force behind our technological achievements. Today, a company can create effective and efficient products and manufacturing processes only by assuring that all engineering and scientific research areas work together. At RCA, mechanical engineering is a vital part of all divisions. Let's look at how the mechanical engineer fits into this company.

Mechanical engineering at RCA

Although RCA is basically an electronics company, most of its major operating units (MOUs) employ mechanical engineers. Figure 1 shows the number of mechanical engineers at RCA in relation to other degreed technical staff. Mechanical engineering assignments vary greatly depending on the individual division's business.

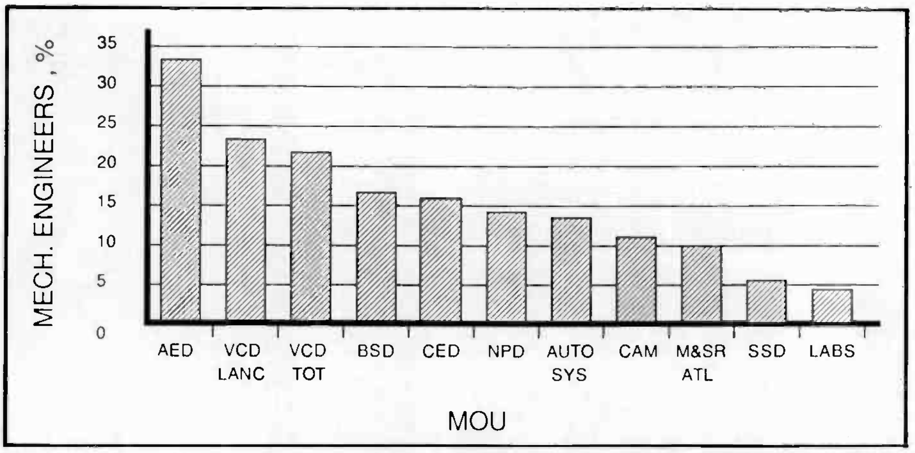


Fig. 1. *Mechanical engineering staff at RCA compared with total technical staff.*

Astro-Electronics Division

The mechanical engineering emphasis in the Astro-Electronics Division (AED) is on product design. The space station platforms and other devices created by AED are not produced in large quantities, but they are very complex systems and must be extremely reliable. AED needs mechanical engineers to ensure the integrity of basic structures and thermal characteristics and the reliability of "on board" mechanisms.

Figure 2 shows an excellent example of the kind of massive design done in AED. It illustrates a candidate configuration for a new space station platform that may be built in the near future. One look at this drawing should show why AED has such a high percentage of mechanical engineers.

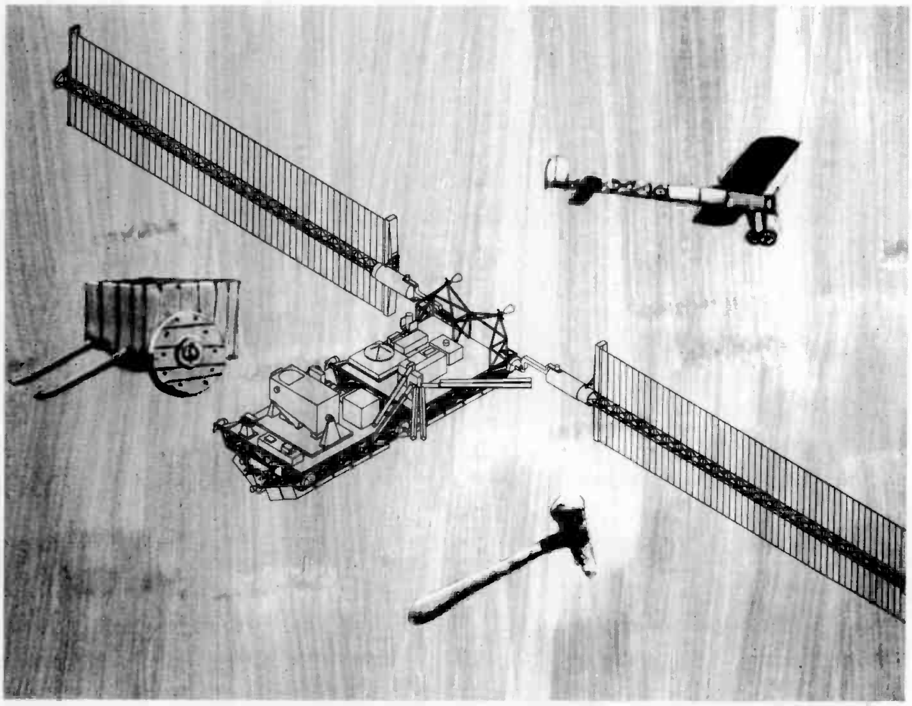


Fig. 2. *A candidate configuration of a space station platform, drawn by a CAD system at Astro-Electronics. The products shown in the background are not presently in production at AED.*

Video Component and Display Division

In the Video Component and Display Division (VCDD), the emphasis for mechanical engineers is on manufacturing. Japan and Korea have become extremely competitive in the consumer electronics field. This competition is causing rapid changes in both product design and manufacturing mechanization in VCDD. Unfortunately, the factories in this division are extremely capital intensive and changes cannot often be made. Mechanical engineers in this division, then, must design flexible manufacturing automation that is simple enough to allow changes at least cost. The cost and quality of VCDD's product depend as much on the mechanical design of the manufacturing equipment as they do on the design of the product.

Consumer Electronics Operations

The mechanical engineers in Consumer Electronics Operations (CE) have an entirely different emphasis. The work of CE is somewhat similar to the work of VCDD, except that there are many more variations in products, and product designs change more often. Therefore, the mechanical engineers' talents at CE are divided between product design and manufacturing.

Mechanical engineering has a significant influence on the quality and cost of a consumer product such as a television. Some of you may doubt that there is anything mechanical about a TV, because there are no moving parts and no mechanical power is transmitted. However, I would argue that even the circuit board (after the circuit has been perfected) is purely mechanical and material engineering until it is "plugged in." The mechanical design of this board and the equipment that assembles it determine the cost in capital and labor needed to produce the board and assemble it into the final instrument. The mechanical design also determines the TV's performance in a drop test, its repairability, the "feel" of its front panel controls, the performance and yield of its molded parts, and the size of the factory needed to produce it.

On the manufacturing side of CE, high-speed, flexible automation is becoming a necessity to compete with foreign-made products. Here, the mechanical engineer has the difficult task of designing automated assembly equipment that can handle today's known product and tomorrow's unknown product. Figure 4 shows an example of such equipment. This software-controlled machine can install up to 120 stakes (small posts for test points or wire wrapping) simultaneously into a circuit board. This tool is forerunner of a staking machine that can do "batch of one" manufacturing in

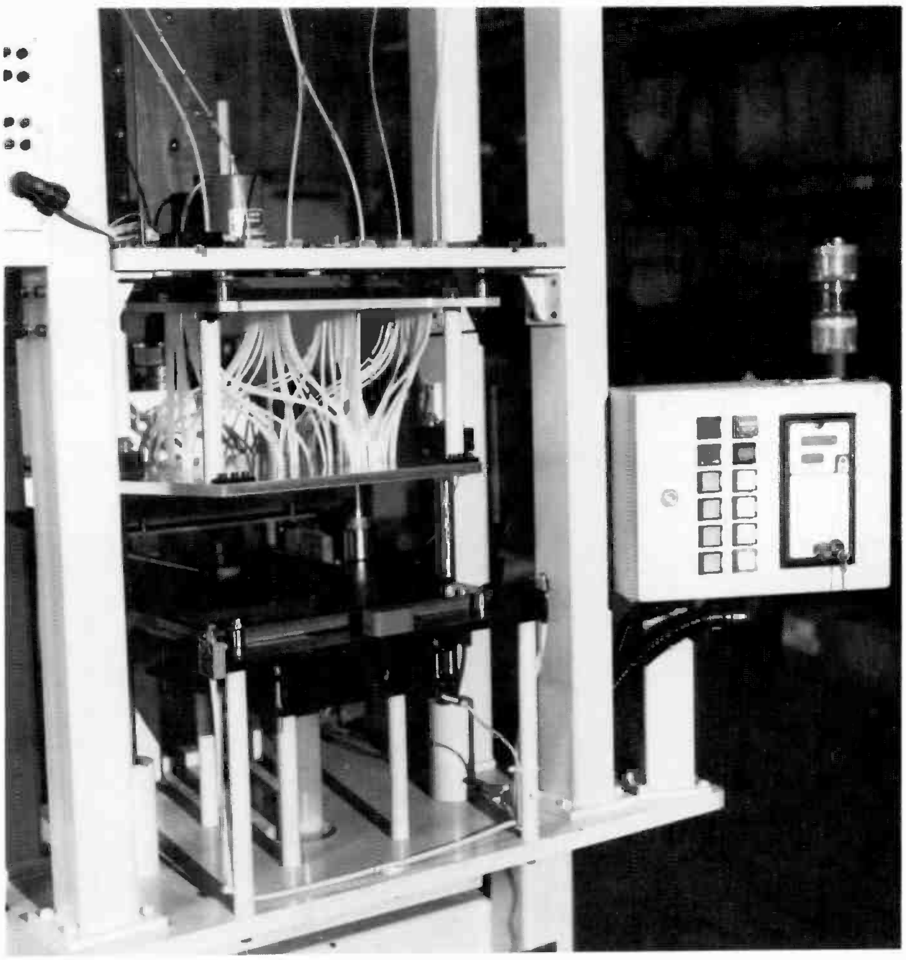


Fig. 3. *Automatic stake assembly at Consumer Electronics. This machine orients and presses up to 120 stakes into a circuit board in one step.*

Fig. 4. *Control Data CAD station at RCA Laboratories. Here, solids modeling is being used to examine the assembly of a picture tube gun.*

which each board can have a different pin layout. In the future, entire factories should be laid out with this extent of flexibility to keep up with the rapidly changing marketplace and to keep our automated equipment from becoming obsolete before it has paid for itself.

RCA Laboratories

There are very few mechanical engineers at RCA Laboratories compared with other MOUs, as shown in Fig. 1. The majority of them are located in the Manufacturing Systems Research Laboratory, which is one of three laboratories under J.L. Miller, all of which are dedicated to manufacturing research.

The Labs' approach to manufacturing any product is first to test the design of the product. The Design for Assembly Group takes care of this task by designing and building full-size models to demonstrate new ideas. At the same time, the Assembly Techniques Group develops the philosophy of assembling the new product design. As the product design develops, the prototype assembly equipment is designed and built to test basic system principles. All this is done as part of a team effort with the MOU responsible for the design and manufacture of the product.

To develop the best automated assembly system for any product, the parts that make up the product should be considered an actual mechanical part of the assembly machinery as well. That is, when an assembly line is automated, the individual parts of all assemblies should have a dual mechanical function: (1) they should function as a reliable part of the assembly and satisfy the product definition, and (2) they should function as a reliable part of the automation equipment to permit assembly yields near 100 percent.

CAD/CAM Systems for Mechanical Engineering

For many years, computers have been used to aid in electronic circuit and integrated circuit design. Now, in the last five years, the mechanical engineering profession has discovered the potential of computer-aided design/computer-aided manufacturing (CAD/CAM) systems. Most of RCA's manufacturing divisions have at least one of these systems. The Manufacturing Design Techniques Group of the Manufacturing Systems Research Laboratory is studying the effect and application of these tools in the mechanical engineering field.

Inventiveness in humans (and other animals)

When humans first began developing a technology base during prehistoric times, animals were already performing mechanical miracles. Termites were building berm solar-efficient houses before the earliest humans could conceive of building any kind of a house. Outstanding architectural wonders created by "lower species" include air-conditioned beehives, whose honeycombs appear to have been designed on a CAD system, and spider webs. To accomplish their feats, bees even have to make their own wax, and spiders must extrude their own threads. Could these two species possibly be the first ones known to form mechanical parts?

Just as mechanical engineers depend heavily on the use of tools, some animals, too, rely on tools to perform their daily tasks. There are finches that use cactus spines to wiggle insect larvae out of holes in branches, gulls that open their dinners by dropping shellfish from great heights, vultures and sea otters that use stones to hammer their food out of eggs and clam shells, respectively, and spiders that carry air tanks (in the form of bubbles) to go scuba diving.

All their modern accomplishments notwithstanding, animals —unlike humans—are deficient in the drive and ability to conceptualize and to improve on previous innovation. Hence, though even bees and toads can plan novel routes, and birds and primates readily copy innovations of diet and food harvesting, both the degree of creativity and the frequency with which animals actively seek innovation through experimentation tend to be severely limited. Animals, to a greater extent than humans, are governed by instinct, and each generation of animals inherits the instincts that successfully guided many earlier generations of its kind. There is usually no improvement in their designs from one generation to the next and, indeed, there is little need. Even minuscule evolutionary changes generally take centuries or longer. Humans, however, can redesign an existing tool one day and, on the next, teach the new art to the world. Inventive thinking has allowed humankind to "leapfrog" the animal kingdom. This situation is likely to continue at an exponential rate, assuming that we humans do not put our most powerful tools to a lethal use.

It is possible today to conceive of a design, engineer the characteristics of its parts (stress, etc.), "draw" it, check its dynamic characteristics (motion, interference forces, etc.), and machine its parts or the molds to make the parts—all without paper. However, it will take much more systems integration and better software before this total design technique can be commonly used in all divisions.

All of our divisions that have CAD systems use them at least for drafting, but some divisions have expanded them even further:

1. VCDD has integrated its CAD system with its business system to allow all areas to work with a common data base. VCDD also has more low-cost terminals available for their engineering staff than any other division.
2. CE has demonstrated the feasibility of designing plastic parts on its CAD system. CE then sends the data tapes to mold-making vendors who have CAD systems. The vendors return the molded parts without any need for paper. Using another software system that is under joint development with the Labs and IKV in Germany, CE engineers can select mold parameters, such as gate locations, runner sizes, and cooling systems, to ensure that the mold is correct the first time.
3. RCA Laboratories has developed a software link between its CAD system and its machine tools that allows a CNC milling machine to produce about 65 percent more parts per year than a standard machine, even if they are all "one of a kind" parts. The Labs is a

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Marvin A. Leedom, Director of the Manufacturing Systems Laboratory at RCA Laboratories, earned a BS in Mechanical Engineering from Drexel University in 1957. Since joining RCA Laboratories as a Member of the Technical Staff in 1962, Mr. Leedom has spent most of his effort on the VideoDisc program in the areas of stylus and player design and disc manufacturing. In 1975 Mr. Leedom was named Manager of Mechanical and Instrumentation Technology and, in 1978, Director of the Electromechanical Research Laboratory. In 1980 he was appointed to the position he now holds.

In 1970 he shared an RCA Laboratories Outstanding Achievement Award for contributions to the high-density technology of recording mechanisms. In 1973 he received a second award for a team effort in the conception and development of signal systems and playback mechanisms for high-density recording systems. He and his team received a David Sarnoff Award for Outstanding Technical Achievement, RCA's highest honor, in 1981, "for key contributions to the development of the CED VideoDisc system."

Mr. Leedom has written or presented several technical papers. He holds 24 U.S. patents in the fields of TV design, Electrofax imaging, and VideoDisc player designs. He is a member of the Society of Manufacturing Engineers and of Robotics International.

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Princeton, N.J.
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major partner with IKV and CE in developing mold design software. The success of this program will allow CE to reduce its product design time by half because the molds will run successfully the first time.

Die design for forming sheet metal and the integration of solids modeling are also important uses for CAD systems in the Labs.

Conclusion

In the consumer arena, all companies are competing in a worldwide game of survival. The players, or companies, in this game keep changing. The younger, new trained players challenge the older, established record holders and hope for an upset. The rule book keeps changing, too. New product models appear in half the customary design time. Manufacturing processes change so often that factory lines soon become obsolete. The only way to win this game is to:

1. Establish simple but excellent product designs that can be assembled with high yields that are achieved naturally, and not by more inspections.
2. Involve the manufacturing engineers and vendors early in the design plan.
3. Design a manufacturing system that assumes no direct labor and is flexible enough to assemble all high sales models for several years. Even if all automation is not installed, the manufacturing yields will improve.
4. Minimize labor costs through the effective use of computer-aided mechanical engineering.

To implement this game strategy requires the talents of people in many technical and nontechnical fields. This is a team match—not a singles game. That is, the engineering, manufacturing, and product design groups must form one team. Within each group, the mechanical engineers must be strong and innovative to ensure that the product and the manufacturing equipment evolve as one system. Those companies that structure their product design and manufacturing mechanical engineering groups so that they do work as one team will be far ahead of the competition in producing a low-cost, easy-to-assemble product and a high-yield manufacturing system.

Somewhere in the world another engineer has done something you ought to know about . . . Your RCA Technical Librarian will assist you in searching any of these online databases, including RCA's own proprietary database.

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C.J. Poux | C.B. Carroll | C.C. Rearick

Simplified system design for automated motion

Motion control—a crucial task for the design engineer—is required more frequently as the trend toward programmable automation continues.

Engineers who handle incremental motion control problems generally have half of the solution before they begin to solve the problem. They know what is required of their system: they generally know how far the machine must move and how fast it must move to get there. With the additional application of a system analysis consisting of basic principles of physics, the determination of the motor torque requirements for the specific application becomes relatively simple.

Background

In a recent project, we faced an automation task requiring the movement of an automatic screwdriver to several locations on an assembly within a limited time cycle. Conceptual development suggested that a cartesian coordinate (X, Y, Z) robot be used as the translation/positioning device. However, research in the marketplace failed to locate a machine with the required speed, load capacity, travel, and cost to satisfy the project requirements. Therefore, we elected to custom build such a device.

To evaluate the effectiveness of the device, a prototype machine consisting of the vertical axis of a cartesian coordinate robot was designed and built. The basic motion profile for the prototype axis was defined as a span, or "reach," of at least 20 inches to be traveled in a maximum of one second. Furthermore, the project dictated that a load of 100 pounds would have to be moved in the form of the screwdriver, the upper horizontal axis, and the moving parts of the vertical axis itself. The positioning accuracy would have to fall within a thousandth of an inch (+ 0.001 inch) of a specified location. Quick and accurate linear movements such as these virtually demanded the use of a dc motor in a linear position-controlled servo mechanism, as further design calculations were to prove. Such a servo motor system is ideally lightweight, responsive, and essentially independent of load. Furthermore, the motors are available with low rotor inertias and are capable of delivering useful torque in most cases at speeds of up to 3000 revolutions per minute.

The mechanical design incorporates a ball bearing lead screw and nut device to couple the load mass to the motor shaft. Such devices have a very high mechanical efficiency, significant mechanical advantage, and can be made to operate with virtually zero backlash (no free play). The motor is directly coupled, axially, to the ballscrew shaft and the driven ballnut produces the linear motion. A dc servo provides the required precision position control. A computer-controlled digital servo interface provides the position and motion profile input information as well as the servo stabilization function. The load mass is supported by a table, and rides, with linear bearings, on a pair of accurately mounted steel shafts positioned on either side of the ballscrew shaft. The device, at this stage of conceptualization, is shown in Fig. 1.

System design

After a conceptualized stage was reached, it became necessary to include the various real world requirements and limiting parameters. With these included, it was possible to see how closely the device would support the desired performance goals. To provide for the 20-inch travel distance, it was estimated that the required ballscrew shaft length, when bearing mounts and shaft coupling were included, would be at least 36 inches long.

Abstract: *The design of a system with a motion controlled machine component does not have to be difficult. Project requirements often demand such a system, which must then be custom built. However, technical documentation of the necessary design procedure is not available in concise form, and the design engineer must make a significant research effort. This article is an account of an incremental motion control design aimed at a specific project, and features an iterative computer program for system optimization. The organization of system requirements and design procedures, leading to individual component selection and emphasizing motor selection, are generally applicable. The design analysis and calculations are simple, straightforward, and will serve as a reference for individuals who must meet similar requirements.*

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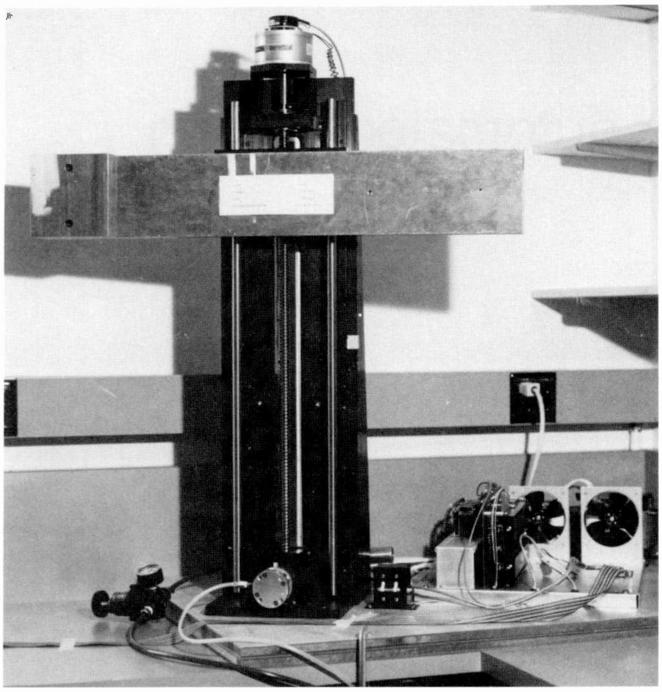


Fig. 1. *Conceptual form of the prototype vertical axis.*

Initial inspection of load life data, critical speed, and compression load estimates further indicated that the shaft should be at least 3/4 inches in diameter.[1] Finally, the ballscrew lead had to permit a linear load velocity of greater than 20 inches per second when driven at the top speed of the servo motor. Since 3000 rpm is a reasonable (and practically standard) top speed for dc servo motors, a ballscrew lead of 1/2 inch per revolution would produce a linear speed of 25 inches per second.

$$v_{max} = (\omega_{max} l)/60 = 25 \text{ in/sec}$$

where v_{max} = maximum linear velocity (in/sec)

ω_{max} = maximum angular velocity = 3000 rev/min

l = ballscrew lead (in./rev.) = 0.5 in/rev

Additional calculations (described below under Motion profile) then show that to move 20 inches in one second with a maximum speed of 25 inches per second, acceleration/deceleration times of 0.2 seconds at rates of ±125 in/sec^2, or approximately 0.32g, are required—all very reasonable parameters to accept.

To determine the torque necessary to produce this motion, it is necessary to calculate the load inertias of all the moving components of the system as seen by the shaft of the motor. To do this, we recall from basic physics that the torque output of the motor must at all times be balanced by the torque reactance of the driven system, expressed classically as:

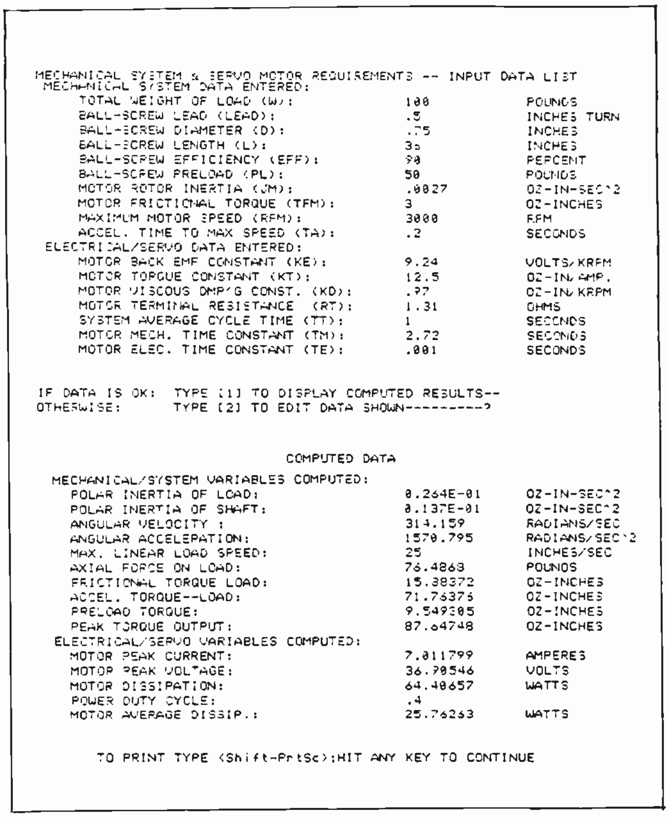
$$T_m = J_T(\alpha) + T_F$$

where T_m = motor torque output (oz-in)

J_T = total of all load polar moment of inertias (oz-in-sec^2)

(α) = angular load acceleration (radian/sec^2)

T_F = frictional torque (oz-in)



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MECHANICAL SYSTEM & SERVO MOTOR REQUIREMENTS -- INPUT DATA LIST
 MECHANICAL SYSTEM DATA ENTERED:
     TOTAL WEIGHT OF LOAD (W):                  100          POUNDS
     BALL-SCREW LEAD (LEAD):                    .5           INCHES TURN
     BALL-SCREW DIAMETER (D):                   .75          INCHES
     BALL-SCREW LENGTH (L):                     36           INCHES
     BALL-SCREW EFFICIENCY (EFF):               90           PERCENT
     BALL-SCREW PRELOAD (PL):                   50           POUNDS
     MOTOR ROTOR INERTIA (JM):                  .0027        OZ-IN-SEC^2
     MOTOR FRICTIONAL TORQUE (TFM):             3            OZ-INCHES
     MAXIMUM MOTOR SPEED (RPM):                 3000         RPM
     ACCEL. TIME TO MAX SPEED (TA):             .2           SECONDS
 ELECTRICAL/SERVO DATA ENTERED:
     MOTOR BACK EMF CONSTANT (KE):              9.24         VOLTS/KRPM
     MOTOR TORQUE CONSTANT (KT):                12.5         OZ-IN/AMP.
     MOTOR VISCOUS DMP'G CONST. (KD):           .97          OZ-IN/KRPM
     MOTOR TERMINAL RESISTANCE  (RT):           1.31         OHMS
     SYSTEM AVERAGE CYCLE TIME (TT):            1            SECONDS
     MOTOR MECH. TIME CONSTANT (TM):            2.72         SECONDS
     MOTOR ELEC. TIME CONSTANT (TE):            .001         SECONDS

IF DATA IS OK:  TYPE [1] TO DISPLAY COMPUTED RESULTS--
OTHERWISE:      TYPE [2] TO EDIT DATA SHOWN---------?


                          COMPUTED DATA
  MECHANICAL/SYSTEM VARIABLES COMPUTED:
    POLAR INERTIA OF LOAD:                    0.264E-01     OZ-IN-SEC^2
    POLAR INERTIA OF SHAFT:                   0.137E-01     OZ-IN-SEC^2
    ANGULAR VELOCITY :                        314.159       RADIANS/SEC
    ANGULAR ACCELERATION:                     1570.795      RADIANS/SEC^2
    MAX. LINEAR LOAD SPEED:                   25            INCHES/SEC
    AXIAL FORCE ON LOAD:                      76.4863       POUNDS
    FRICTIONAL TORQUE LOAD:                   15.38372      OZ-INCHES
    ACCEL. TORQUE--LOAD:                      71.76376      OZ-INCHES
    PRELOAD TORQUE:                           9.549305      OZ-INCHES
    PEAK TORQUE OUTPUT:                       87.64748      OZ-INCHES
  ELECTRICAL/SERVO VARIABLES COMPUTED:
    MOTOR PEAK CURRENT:                       7.011799      AMPERES
    MOTOR PEAK VOLTAGE:                       36.70546      VOLTS
    MOTOR DISSIPATION:                        64.40657      WATTS
    POWER DUTY CYCLE:                         .4
    MOTOR AVERAGE DISSIP.:                    25.76263      WATTS

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Fig. 2. *A typical data input/output printout.*

Other higher order motional considerations can be added, but for a first order approximation this will suffice.[2]

Calculations involving moments of inertia are not complicated, but the often confusing dimensional units and related unit conversions can cause errors. For this reason we decided to program an IBM PC with all the equations and conversions so that we only had to feed in data and read the answers. A typical data input/output printout from the program is shown in Fig. 2. Given the preliminary system data, a motor would be required to deliver 87 ounce-inches of torque to move the 100-pound load the required distance of 20 inches in one second. We also found from the program that the motor and ballscrew would exert a 76-pound axial force on the load. In addition, the program performs the necessary calculations for motor peak voltage, current, and power. These results showed us not only what we could expect from our system, but also what our system will expect from us. A return trip back through the catalogues of motor and ballscrew vendors now showed us that relatively standard components are commonly available to build the translating/positioning device. Minor changes in system data, as gleaned from different vendor catalogue options, could then be evaluated by re-entering the new data. The program has now essentially evolved into a rather sophisticated system model. To solve systems of this type, the program can be used directly. For a deeper understanding of the solution and the equations used, the following analysis is supplied.

Design analysis

The detailed design of the system falls into three basically separate but ultimately interactive categories:

1. Motion profile. This determines the ultimate "task" of the

device, from which the very important acceleration requirements are developed.

2. Ballscrew/load configuration. This defines the moment of inertia (J) and torque (T) parameters, and along with the angular acceleration (α) parameter, determines the driving source (motor) torque requirements.
3. Motor torque/power requirements. These determine the servomechanism constants and electrical amplifier power requirements.

Motion profile

The first step in determining the torque requirements of our system is to find the angular acceleration of the motor (α). We can simplify this process in two ways: (1) we will assume that in an incremental motion system we are dealing with uniform (constant) acceleration, and (2) we can draw motion curves of the system that include the position, velocity, and acceleration profiles. The motion curves for the system are shown in Fig. 3.

The linear acceleration of the load in this example, therefore, can be determined from the following equation:

$$a = (v_f - v_i)/t_a,$$

where

v_f = final velocity (in/sec)

v_i = initial velocity (in/sec)

t_a = acceleration time (sec)

With a quick glance at the velocity profile of our motion curves we can determine that for a full move the following values are true:

$$v_f = v_{max} = 25 \text{ in/sec}$$

$v_i = 0$ (incremental motion)

Therefore, we must determine the acceleration time, t_a.

The method for calculating the acceleration time, t_a is governed by the physical laws of uniform accelerated motion. While the derivation of these equations is beyond the scope of this paper, suffice it to say that we begin with the following equations:

$$t_t = t_v + 2t_a \text{ (where } t_a = t_d\text{)}$$

$$s_t = s_v + 2s_a \text{ (where } s_a = s_d\text{)}$$

$$t_a = 2s_a/v_f$$

where t_t = total time (sec)

t_v = time at maximum velocity (sec)

t_a = acceleration time (sec)

s_t = total travel (in)

s_v = travel at maximum velocity (in)

s_a = travel during acceleration (in)

By substitution and simultaneous solution the following equation results:

$$t_t = (2s_t - s_v)/v_f$$

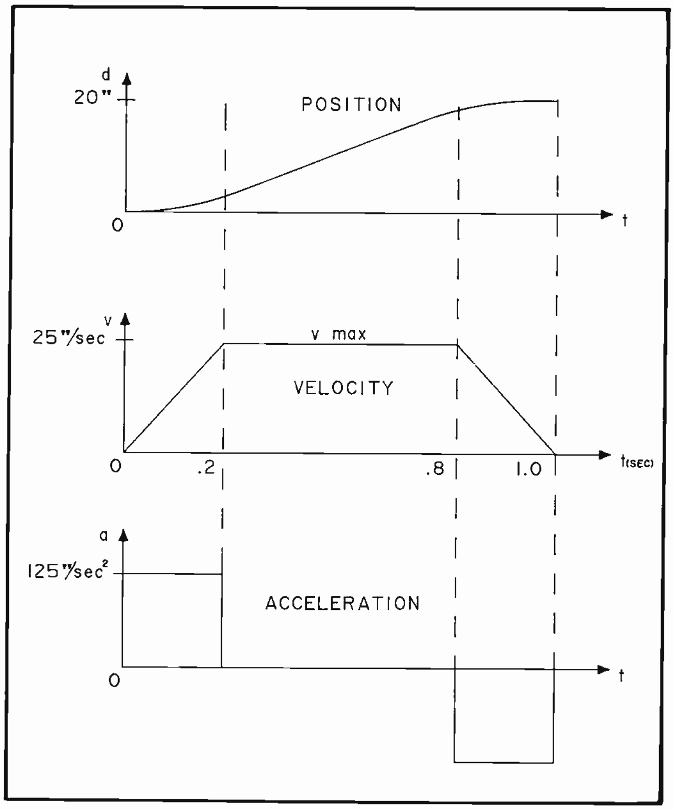


Fig. 3. *Motion curves.*

which is, by rearrangement,

$$s_v = 2\, s_t - v_f\, t_t$$

For our example:

$$s_v = 2(20 \text{ in}) - (25 \text{ in/sec})(1 \text{ sec}) = 15 \text{ in}$$

Therefore $s_a = (20 \text{ in} - 15 \text{ in})/2 = 2.5 \text{ in}$

and the acceleration time is

$$t_a = 2(s_a)/v_f = 2\,(2.5 \text{ in})/25 \text{ in/sec} = 0.2 \text{ sec}$$

which results in a linear acceleration of

$$a = (v_f - v_i)/t_a = (25 \text{ in/sec})/0.2 \text{ sec.} = 125 \text{ in/sec}^2$$

Now we must convert the linear load acceleration into a motor angular acceleration value. Since the lead of the ballscrew is known, the angular acceleration (a) can be calculated from

$$\alpha = 2\pi a/l = 1570 \text{ rad/sec}^2$$

where a = linear acceleration (in/sec^2) = 125 in/sec^2

l = ballscrew lead (in/rev) = 0.5 in/rev

α = angular acceleration (rad/sec^2)

Ballscrew/load configuration

Moment of inertia. The moment of inertia of the system is the result of the summation of the moments of inertia of the components. In our example (a ballscrew and load),

$$J_T = J_{Load} + J_{Shaft} + J_{Motor}$$

Since our system is the combination of a linear and a rotary system, the force of the linear load must be transformed to an equivalent torque for a rotary system. This process results in a value that is referred to as the reflected load inertia. For the case of a ballscrew we have[3]

$$J_L = (Wl^2/4\pi^2 g)(100/e)$$

where W = weight of load (lbs) = 100 lbs

g = gravitational constant = 386.4 in/sec^2

l = ballscrew lead (in/rev) = 0.5 in/rev

e = ballscrew percent efficiency = 90

By substitution of our values,

$$J_L = 0.00183 \text{ lbs-in-sec}^2$$

Next, we must determine the moment of inertia of the ballscrew, J_s. The moment of inertia for a cylindrical body about its principal axis is[4]

$$J_s = m_s r^2/2$$

$$= W_s r^2/2g \text{ (converting from mass to weight units)}$$

where r = ballscrew radius (in)

W_s = weight of ballscrew = volume × density = $\pi r^2 l \times \rho$

L = ballscrew length

ρ = density of ballscrew material (lbs/in^3)

Therefore $\pi r^2 l(\rho) r^2/2g$

By substitution of our values and a density (ρ) of 0.284 lbs/in^3 for steel[4]

$$J_s = 0.00086 \text{ lbs-in-sec}^2$$

Finally, the moment of inertia of the motor rotor, J_m, must be determined. This figure is not conveniently calculable, but fortunately is provided by motor manufacturers in their product specifications. The process of selection is an iterative one, whereby you make a "best guess" for a motor that looks reasonable for the job, calculate required torque using J_m for that device and, if the resulting torque requirement exceeds that available from the motor, select a slightly larger motor, etc., until an adequate device is found. In this particular case, the estimate motor had a rotor moment of inertia J_m of 0.00017 lbs-in-sec^2, which resulted in a system moment of inertia, determined by summing the moments of inertia of the components, of

$$J_T = 0.0028 \text{ lbs-in-sec}^2$$

Friction torque. The third and final term of the general equation is the friction torque (T_F). This term accounts for the torque required to overcome all frictional forces. In a vertical application, such as our example, this would include the drive torque required to lift the load. However, in the preliminary design we decided to balance the load weight with a counterbalancing air cylinder. Therefore, the frictional torque requirements included the friction torque of the load bearing system (T_B), the friction due to the preload on the ballscrew nut to remove the backlash (T_{PL}), and the motional damping torque of the motor (T_D).

The frictional torque can therefore be represented as

$$T_F = T_B + T_{PL} + T_D + T_{FM}$$

The motor bearing and brush friction torque, T_{FM}, is found in the motor specifications to be 0.187 in-lb. The ballscrew preload torque T_{PL} can be calculated with the following equation[1]:

$$T_{PL} = (P_l l\, 0.2)/2\pi$$

where P_l = preload force (lbs)

l = ballscrew lead (in/rev) = 0.5 in/rev

For our example, the preload, chosen as 50 lbs, with our ballscrew lead yields:

$$T_P = 0.596 \text{ in-lbs}$$

Due to the counterbalance in our system, the load is now denoted by P_f, the frictional load. The frictional torque of the load bearing system, T_B, can be determined from the following equation.

$$T_B = (P_f l)/2\pi e$$

where P_f = frictional load (lbs)

e = ballscrew efficiency (percent) = 90

An analysis of the offset load situation in the vertical counterbalanced system indicates that the load bearings receive a normal load of approximately 100 lbs.

$$P_f = uN = 0.3 \text{ lbs}$$

where N = normal force (lbs) = 100 lbs

u = coefficient of friction of linear bearings = 0.003

Therefore, by substitution of our values,

$$T_B = 0.027 \text{ in-lbs}$$

Next, the motor damping torque must be calculated from the following equation:

$$T_D = (K_D n)/1000$$

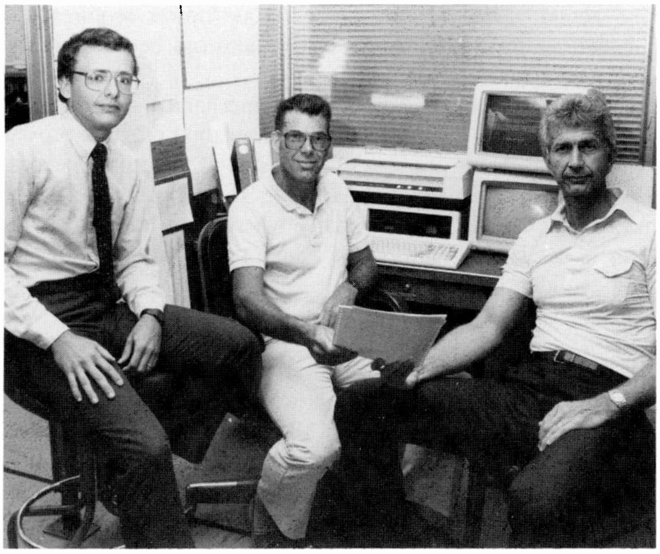
where K_D = motor viscous damping coefficient (oz-in/1000 rev/min)

n = motor speed (rev/min) = 3000 rev/min

By substitution of a viscous damping coefficient (K_D) of 0.97 oz-in/1000 rev/min, which was chosen from the motor product specifications, we now arrive at

$$T_D = 2.91 \text{ oz-in}$$

$$= 0.182 \text{ in-lbs}$$



Left to right: Poux, Rearick, Carroll.

Charles B. Carroll is Head of the Electromechanical Systems Research group in the Manufacturing Systems Research Laboratory. He graduated cum laude from the University of Florida in 1960, with a BS in Mechanical Engineering. In 1967, he received the MSME from Drexel University. In 1960, he had joined Astro-Electronics as a design engineer developing sensor and camera components. After serving as a Lieutenant in the U.S. Army Artillery from 1962 to 1964, Mr. Carroll returned to Astro-Electronics, and then transferred to RCA Laboratories in 1966.

In 1973, Mr. Carroll transferred to Palm Beach Gardens, Florida, and worked on products using mini- and microcomputers for hotel/motel management. He rejoined RCA Labs in 1975, and he received an RCA Laboratories Outstanding Achievement Award for work on equipment for optical scanning and readout of the RCA VideoDisc. He then led the mechanical design effort that succeeded in producing early prototypes of a flat-panel television system. Mr. Carroll holds seven U.S. patents.
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Christopher Poux graduated from Gannon College, Erie, Pa., in 1974. He received a BS in Mechanical Engineering and, in 1982, was awarded a Professional Engineering License by the State of Indiana. Mr. Poux joined RCA as a Member of the Engineering staff at the RCA VideoDisc Division in 1980. While at VideoDisc he worked in the Advanced Development group on disc manufacturing. In 1984, Mr. Poux transferred to RCA Laboratories as a Member of the Technical Staff in the Electromechanical Systems group.
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Charles Rearick graduated from the University of Maine in 1956 with a BS in Electrical Engineering, and from the University of Pennsylvania Moore School in 1970 with an MS in Electrical Engineering. He spent two years as an ammunition supply and guided-missile maintenance officer in the Army Ordnance Corps. Since 1959 he has been an acoustical design engineer with Home Instrument Division in Cherry Hill, a logic design engineer with Missile and Surface Radar Division in Moorestown, a circuit design engineer with Government Communications Division in Camden, and a product design and project leader with Distributor and Special Products Division in Deptford. He has been, since 1981, a Member of the Technical Staff at RCA Laboratories in the Electromechanical Systems group, where his principal effort is involved in motion-controlled devices and servomechanisms.
Contact him at:
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Princeton, N.J.
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The friction torque is then

$$T_F = T_P + T_B + T_D + T_{FM} =$$
$$0.596 \text{ in-lbs} + 0.027 \text{ in-lbs} + 0.182 \text{ in-lbs} + 0.187 \text{ in-lbs} =$$
$$0.992 \text{ in-lbs}$$

Torque summation. Finally, now that we have determined the three parameters we can substitute them into the classical equation and solve for the driving torque required by the motor:

$$T_m = J_T(\alpha) + T_F =$$
$$0.0028 \text{ in-lbs-sec}^2 \,(1570.8 \text{ rad/sec}^2) + 0.992 \text{ in-lbs} =$$
$$5.48 \text{ in-lbs}$$

Or, converted to oz-in:

$$T_m = 87.6 \text{ oz-in}$$

Looking back to the motor specification sheet from which we picked the rotor moment of inertia, we find that the motor is rated at 80.1 oz-in at 3000 rpm, and 690 oz-in at zero speed. Thus the motor is almost capable of satisfying the peak torque load of 87.6 oz-in even at its rated speed. Since we are dealing with an incremental motion application, the drive motor capacity does not have to be rated at full load. An analysis of the intended application duty cycle with root-mean-square methods yields a torque requirement significantly below the motor rating.[5]

Therefore, the motor, for which the rotor moment of inertia and viscous damping coefficient were chosen, was selected to run the prototype axis. The motor has performed as predicted by the design analysis.

Motor torque/power requirements

The calculations of the electrical requirements for the system and the motor performance proceed in a similar manner. It is important to know what maximum terminal voltage and drive current the motor will require in order to deliver the needed torque and speed parameters. An adequate driver amplifier must be designed, and the average power dissipated by the motor should fall within the rating of the amplifier. Electrical viscous damping of the motor represents a velocity-dependent "frictional" torque load on the system, and was evaluated with regard to the total mechanical torque load (see Friction torque, above).

The calculation of the electrical factors has been included in the computer "model" of the system. These calculations rely on previously derived parameter values as well as the electrical input data gathered from the vendor motor specification sheets.

The motor peak power dissipation (W_d) in watts is calculated from

$$W_d = (T_p/K_t)^2 R_a$$

where K_t = motor torque constant (oz-in/amp)

R_a = motor armature resistance (ohm)

Next, the motor peak current (I_p) requirement in amperes was calculated as

$$I_p = T_p/K_t$$

Then the motor peak voltage (V_p) requirement in volts is calculated from

$$V_p = (I_p R_a) + K_e\,(n/1000)$$

where K_e = motor back emf constant (volts/1000 rpm)

Finally, the computer was programmed to generate two additional parameters. These are average motor dissipation, W_d (avg.) in watts, and the motor zero speed (starting) torque (T_o) in oz-in. The specific values are obtained from the following equations:

$$W_d\,(\text{avg}) = W_d h$$

and

$$T_o = K_t V_p$$

where h = expected duty cycle (percent/100)

The actual values of the motor torque/power requirements calculations for the prototype vertical axis can be found in the computed data section of Fig. 2.

The ability to manipulate the many mechanical and electrical parameters within the confines of the simulation program proved invaluable in the early stages of the design of the prototype axis. The technique is easily adaptable to a host of electromechanical calculations, especially where complicated conversions of engineering units cause a high probability of error and poor confidence in the results.

Acknowledgments

The authors would like to acknowledge John Aceti at RCA Consumer Electronics, who as a Member of the Technical Staff at RCA Laboratories was initially involved. Also, Robert Schneller, Senior Technical Associate, and Paul Smalser, Associate Member of the Technical Staff, both from RCA Laboratories, who contributed to the prototype system construction. And finally, Maria Costello, of the Laboratories, whose hard work and patience is greatly appreciated.

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G.J. Whitley

Cut-and-clinch innovations in printed circuit board assembly

Circuit board component insertion doesn't end with the insertion step. The leads or tabs that extend below the board must be trimmed, and the component must be "clinched."

For both quality and cost reasons it is advisable to insert components into printed circuit boards using automatic component insertion machines, but as yet not all types of components can be accommodated. One type of automatic machine inserts components having two common wire leads extending axially from both ends of the component. The components are presented to the machine sequentially between two parallel tapes that hold the leads at their extreme ends. The machine snips the component away from the parallel tapes, bends the wire leads to a fixed spacing, and pushes the bent leads into a pair of holes in the PC board. Following insertion, a combination tool cuts the protruding leads and bends them inward, thereby seizing, or "clinching," the component to the PC board. This is known as axial component insertion.

A similar operation is performed with components having two parallel wire leads that are usually 0.200 inches apart, center to center. These extend radially from the center of the component (0.100 radius from center for both leads), allowing the component to stand upright after insertion. The components are attached to a wide paper tape that has indexing or positioning holes in it for proper orientation and location. The two radial leads are held accurately by two machine fingers while the leads are cut away from the tape. The fingers then insert the leads into two holes in the PC board. A tool similar to that used in the axial assembly, located under the PC board, cuts and clinches the leads. This is known as radial component insertion.

These two machines perform very well within their limitations. They require a lot of space, which makes it difficult to incorporate them within a conveyor system. Generally, an operator or a "pick and place" robot loads the bare PC boards into the machine; after insertion the completed boards are placed on a conveyor for further assembly.

However, many components have profiles and pin configurations that do not allow the use of tape. Parts such as transformers, chokes, heat sinks, and metal shields must be mechanically fastened to the PC board because as the board travels from station to station on the conveyor line, the components experience severe vibrations. In addition, the components can be bumped by operators inserting other parts into the PC board. If the components are not secured to the board they can tip over, causing one or more leads or lugs to leave the holes. This condition is known as "leg out."

If such conditions are not detected before the board goes through the wave soldering operation, the results can be very costly, and the following steps must be taken:

(1) The problem component must be identified.

(2) The hole in the PC board, now covered with solder, must be opened.

(3) The component must be hand-soldered in place.

(4) The PC board must be once again inspected.

Figure 1 shows just a few of the various profiles and configurations of mechanical parts and electrical components that become part of a printed circuit board chassis assembly. Such parts are hand inserted into the bare PC board by an operator sitting at the conveyor. Enough space must be allowed under the conveyor for the operator's legs.

The mechanisms used by conventional auto-insertion machines to cut-and-clinch are very large and require a lot of maneuvering space. These must be adapted for use on conveyors with manual component inserters, and they must meet the following criteria:

(a) Operations done to the underside of the PC board must be done in the space occupied by the channel rails (approximately 4 inches high) that support the conveyor.

(b) Such a mechanism must also be lifted upward to the PC board to perform the cut-and-clinch operation, and be dropped afterward to allow the PC board to travel to the next station

Abstract: *For both quality and cost reasons it is advisable to insert components into printed circuit boards using automatic component insertion machines, but as yet not all types of components can be accommodated. This article describes some of the machines presently in use.*

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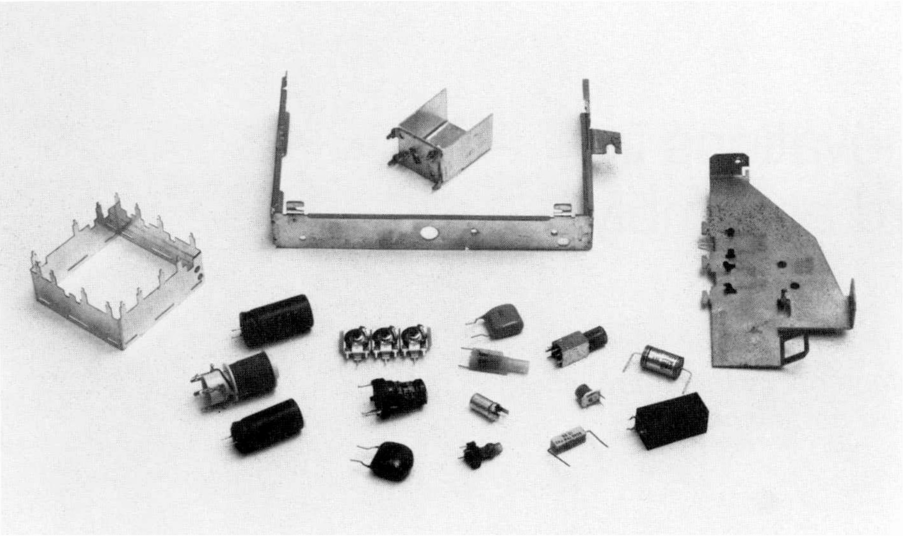

Fig. 1. *Various mechanical and electrical components used in a typical TV printed circuit board chassis.*

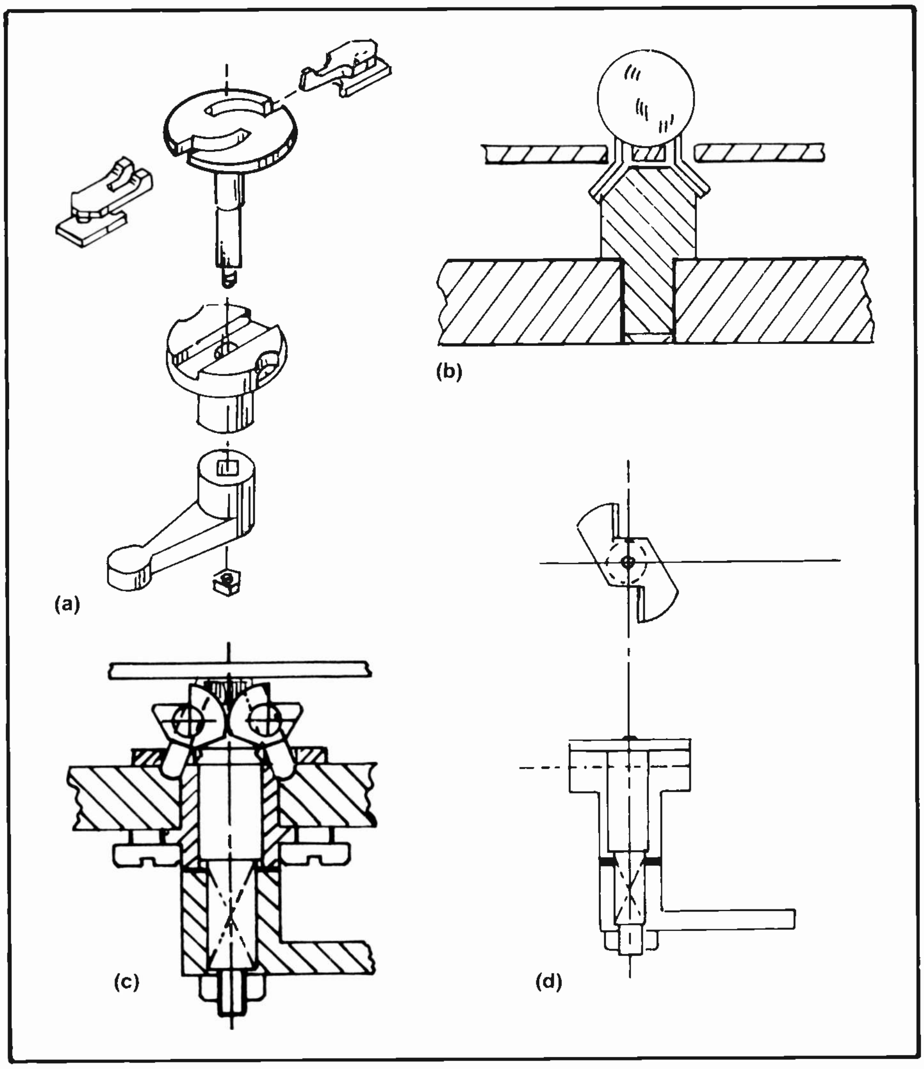

Fig. 2. *Cross sections of some cut-and-clinch devices. (a) Exploded drawing showing pattern of typical assembly. (b) The first passive clinch device. (c) Cross section showing pivoting rollers in IC device. (d) Cross section showing cutter and lever for turning the cutter.*

without disturbing component leads that extend beneath the board.

A conveyor-adaptable system

The simplest and least expensive means of clinching leads under a PC board is to lift a small square or rectangular post with tapered sides upward until the top of the post barely touches the bottom of the board. The tapered sides of the post are centered between the two (or more) holes into which the component will be inserted from the top of the PC board. As the part enters, the legs (or leads) of the component are bent slightly outward by the tapered post, thereby securing the component to the board in a semi-rigid manner (Fig. 2b).

This technique is called "passive clinch" because there are no moving parts, just a fixed positive post in a fixed location. Passive clinch is widely used in Consumer Electronics assembly plants in the U.S., Mexico, and Taiwan. It is very simple and inexpensive to incorporate, and it does prevent "leg out." This operation does not secure the component tightly to the top side of the PC board, but this is often unnecessary. One limitation is that the leads of the components must be precut to desired lengths prior to insertion into the PC board, because the passive system only clinches the leads.

The ideal mechanism should cut the component leads to a short length and bend them securely against the underside of the PC board in one operation. This holds the component tightly to the top side of the PC board and allows for good soldering and a good mechanical connection.

The first cut-and-clinch mechanism

The first cut-and-clinch mechanism (still in use) is positioned directly below the PC board. Slots in the mechanism are aligned with the holes in the PC board. The operator inserts the component leads through the holes in the PC board so that they enter the slots of the mechanism. Between the PC board and the mechanism is a two-bladed circular knife, or cutter, that can revolve approximately 60 degrees. As it turns, the bottom edge of the cutter shears the component leads between it and the side of the slots, cutting the leads to length. As the cutter continues to rotate the top of the cutter bends the cut leads flush with the underside of the PC board. A lever arm pulled by an air cylinder turns the cutter. Figure 3 shows an array of cut-and-clinch mechanisms oriented with corresponding holes in a PC board (not shown). This type of system allows each mechanism to be individually positioned to accommodate an array of holes and to be moved easily to another location.

These individual units move up and down on the same lift mechanism as the "passive clinch" posts, and are also designed to stay within the available space beneath the conveyor while allowing sufficient leg room for the operator. The slots of each mechanism in Fig. 3 can be rotated 360 degrees if space conditions allow for positioning of the air cylinders. The air cylinders are controlled either by an operator foot switch or some other control device.

These original units required considerable planar area beneath the board due to the size and number of cylinders and the many hoses required. These conditions limit the number of component leads that can be cut-and-clinched at each conveyor station.

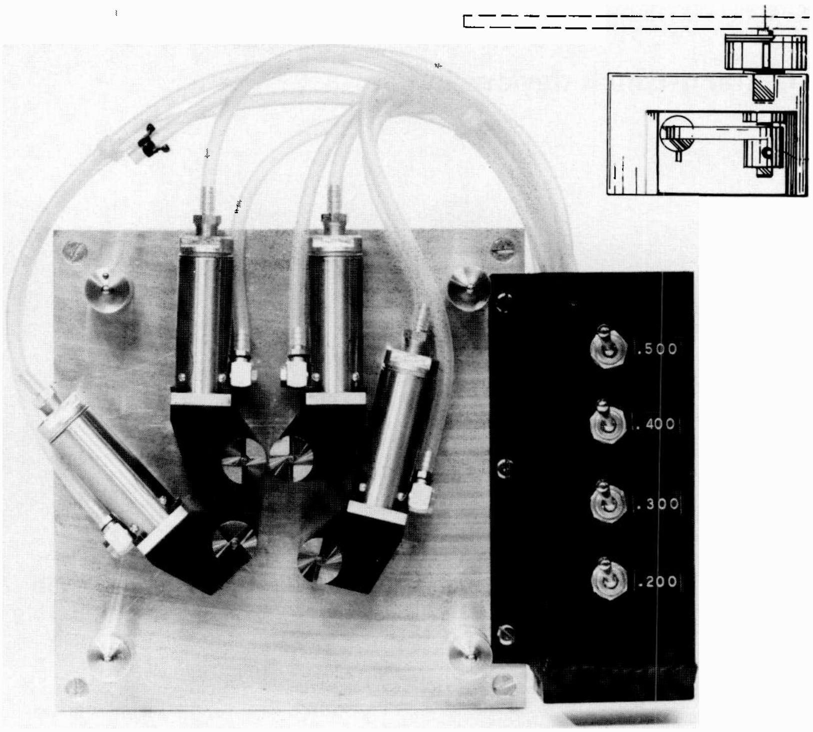

Fig. 3. *Individual cut-and-clinch mechanisms (top view). Inset: cross section of lever and cutter.*

The improved system

In order to eliminate the numerous cylinders and the yards of hoses and fittings, a new concept was considered (see Fig. 4). This concept uses two plates. The top plate (Fig. 4a) holds individual mechanisms, and these mechanisms (4d) are activated by a moveable bottom plate (4b) that slides back and forth, powered by a 2-inch diameter air cylinder (4c) with a 1-inch stroke. The bottom moveable plate has numerous "U"-shaped blocks that engage levers (4h). These levers control the movement of the cut-and-clinch mechanisms on the top plate. Each station on a conveyor has its own configuration corresponding to the types and placement of components particular to that station.

The top plate contains an array of 3/8-inch diameter holes that accommodate a variety of cut-and-clinch devices. These holes line up with the centerlines of the components on the PC board. A computer layout of the PC board gives the exact location of each component; this information is conveyed to a CNC milling machine, which drills the corresponding holes in the top plate. A top plate having eight to ten cut-and-clinch devices can be fabricated in approximately one hour.

Each individual cut-and-clinch device can be installed and removed easily using two small screws on the top plate and one hex-nut beneath the device. Since the device mounting hole is centered on the component, each device can be revolved 360 degrees to correspond to the orientation of any component on the PC board.

Every top and bottom plate assembly must be oriented properly in the "up and down" mechanism (4f), and must be easily removed for service. Two different diameter pilot pins orient the two-plate system to the "up and down" mechanism in a pilot hole and a parallel slot (4g).

The small cut-and-clinch devices are used many times over. Only the board-specific top and bottom plates are discarded at the end of the production cycle. Occasional sharpening of cutters may be necessary. A variety of such devices is described in the sidebar.

The applications of these devices are practically unlimited due to the design concept and low cost of manufacture. It has been estimated that eight stations, all different, with varied components, and

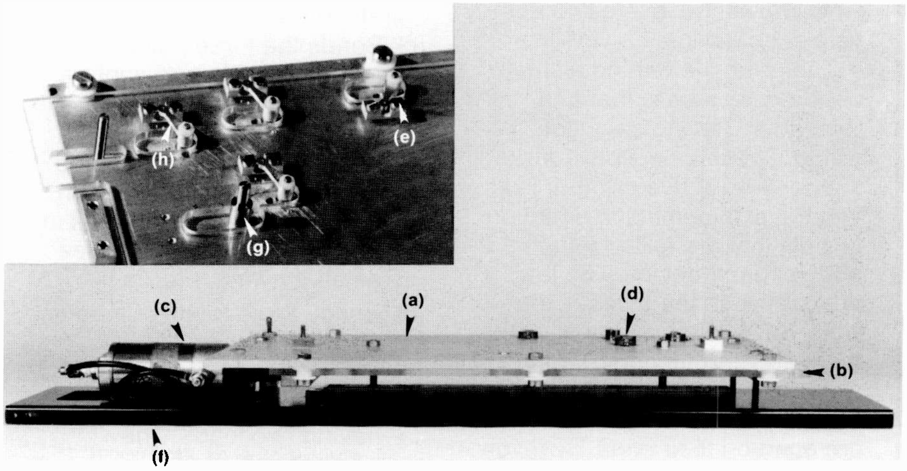

Fig. 4. *Side view showing thin profile and lift mechanism channel. Inset: bottom view of the device.*

Cut-and-clinch devices in use

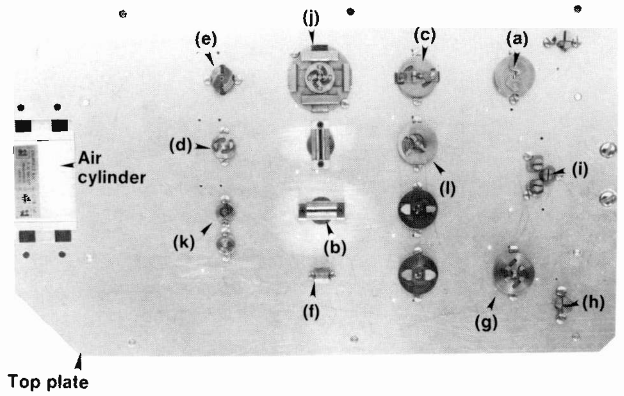

Fig. 5a. *Top view of a mechanism showing many variations of devices used in cut-and-clinch.*

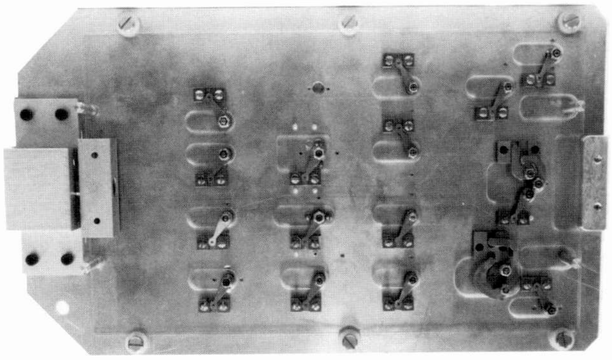

Fig. 5b. *Bottom view of the device shown in Fig. 5a showing typical level orientations and profile of bottom plate.*

Figure 5 (top view) shows an assortment of devices that are used in a typical cut-and-clinch assembly station. The bottom view shows the levers that activate each device.

The principle behind all the bending devices is that of a rotary disc having cam slots within. These cam slots control the position of two or more jaws that travel in a specific direction in slots that are milled into the body of each device. The jaws are captive in the sides of the slots, but are free to travel within them as the rotary disc (with cam slots) revolves. Figure 5c shows the rotary disc with cam slots that are open on their ends. By revolving the disc until the openings align with the slots underneath, the jaws can be removed and other jaws can be substituted. Where components are placed very close to each other, special-profile jaws can be made to clinch multiple components.

(a) Cuts and clinches two component lead wires that have minimal spacing between them, even if they are touching each other. Leads are cut-and-clinch 180 degrees apart.

(b) Bends all the leads of an integrated circuit outward. Leads can be bent inward if necessary. (See detail drawing, Fig. 2c.)

(c) Bends the three leads of a transistor with lead centers 0.050 inches apart. No prebending of the leads is necessary. The two outside leads bend in one direction, and the center lead bends in the opposite direction. (See detail drawing, Fig. 2a.)

(d) Bends three leads of a potentiometer inward.

(e) A standard cut-and-clinch device for components having two leads. The component leads fall into two slots after they pass through holes in the PC board. When the compo-nent is in place and rests on the top side of the PC board, a rotary cutter revolves, cutting the leads and bending them flush with the underside of the board. The excess lead wire (cut) falls through open holes into a waste area. (See detail drawing, Fig. 2d.)

(f) A passive clinch block that bends leads outward. (See detail drawing, Fig. 2b.)

(g) Bends four leads of a filter choke outward. Four jaws move outward when the revolving center cam turns.

(h) Holds one side of a boot-shaped twist tab (see Fig. 1), and a revolving finger push bends the extended portion of the tab, thereby securing the device to the PC board. These tabs are usually mounting feet for heat sinks or metal frames.

(i) Twists three closely spaced tabs in any direction simultaneously, regardless of how close they are. In this case the levers may have to be formed differently to avoid slashing. (See Fig. 5-bottom view)

(j) Bends the tabs of a rectangular shield. The four slides move inward, bending all the tabs that protrude under the PC board.

(k) Bend bus wire lead in any direction (two revolving devices are shown).

(l) Bends three tabs of a trimmer capacitor whose orientation is not uniform with respect to the component center.

various locations handling different PC boards (for a total of 48 stations) would cost approximately the same as one robot assigned to only one station on a conveyor line. The robot can only cut-and-clinch one (lead) part at a time. The two-plate system cuts and clinches all parts simultaneously at each station, saving valuable time.

Robotic application

The simplicity of design of this approach allows it to be adapted to many applications. We are now considering adapting it to automatic robotic applications. There is no doubt that systems that employ fixed and dedicated tooling do provide the greatest consistency and reliability, and the two-plate system provides both, along with flexibility and simplicity, which in turn allows for inexpensive manufacture.

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Mr. Whitley joined Consumer Electronics in 1956 as a Project Engineer. He left RCA in 1964, and held various positions in industry until 1978, when he joined RCA Laboratories as a Member of the Technical Staff. Since then he has been working with Consumer Electronics on product assurance and reliability. He has also been assisting various RCA factories in improving their manufacturing techniques. In 1980 he received an RCA Laboratories Outstanding Achievement Award for innovation in the factory environment.

Mr. Whitley holds 11 patents and has written four technical papers. He is a past Chairman of Public Relations for SME, and is a Certified Manufacturing Engineer.
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K.S. Reid-Green

SUPERCAM

New software and a job-handling technique borrowed from the 1947 Berlin Airlift resulted in dramatic increases in the output of the RCA Laboratories Model Shop.

For several years, computer-aided manufacturing (CAM) of mechanical parts has been seen as the efficient means to produce small numbers of identical parts. Traditionally, CAM filled the gap between individual parts made on manual equipment, and large numbers, where molding was usually least expensive. In the summer of 1985, the Mechanical Design Techniques Research Group completed the SUPERCAM project, making the computer-aided manufacture of individual mechanical parts cost-effective. As far as I am aware, it is an industry "first" that is of great value to the Labs and can improve computerized numerical control (CNC) anywhere.

Problems in the Model Shop

The Labs' model shop got its first computerized numerical control (CNC) milling machine in 1980. Early in 1981, a Computervision (CV) CAD/CAM system was installed, primarily for mechanical drafting and design. It seemed to make sense to use the CV to make parts, but at that time there were problems in doing so. One of the most obvious was the amount of work and time needed to enter a part into the CV, put tool paths on the part, and then convert the tool path data into the language of the controller on the CNC machine. Eventually, somebody had to ask, "Is CNC cost-effective in a prototype shop? In 1982, Marv Leedom, the Director of the Manufacturing Systems Research Laboratory, asked me to find out.

Abstract: *To achieve fast turnaround in a CNC shop, it is necessary to maximize productive machine time and to develop an optimum job processing methodology. Maximization of production requires identification of all time-wasting factors: on-line coding or correcting of tool paths, slow loading of tool programs, on-line tool digitizing, erroneous tool paths, etc. Optimum job processing requires selecting job sequences to minimize average waiting time and providing an environment in which machines can be run unattended.*

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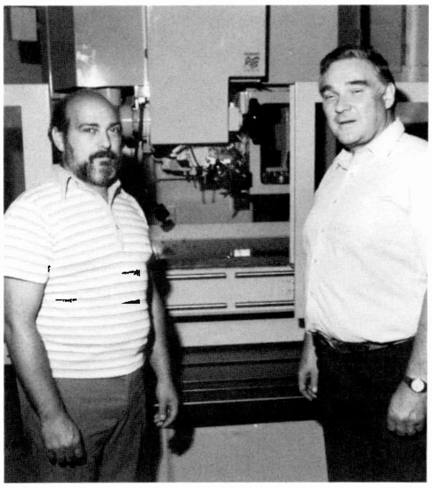
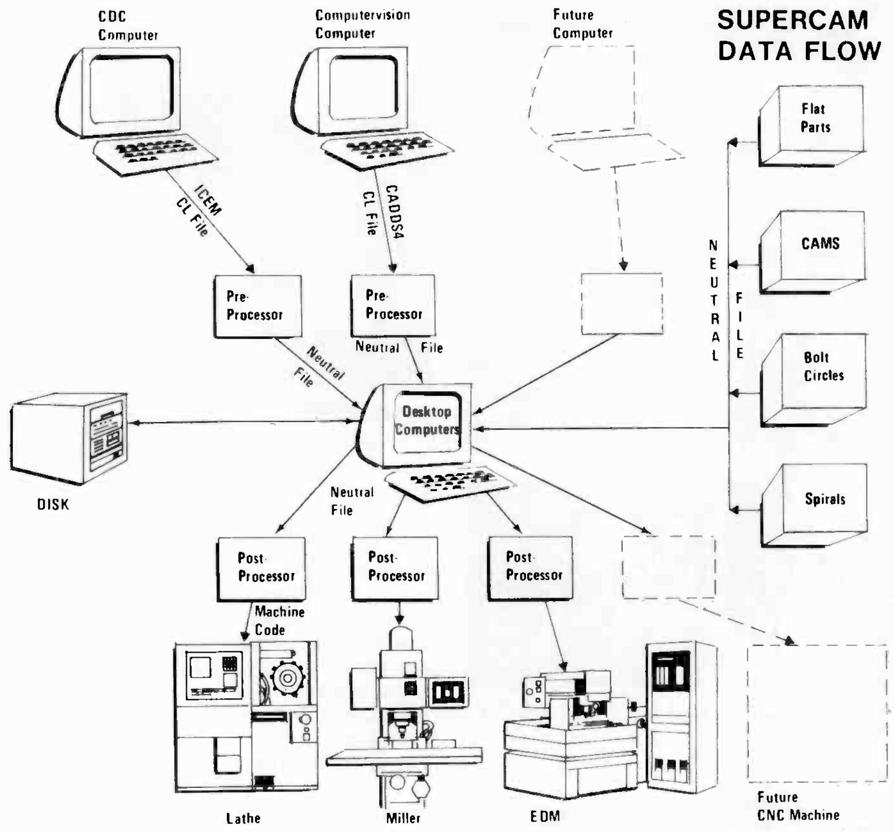
The most obvious problem involves the availability of productive CNC machine time. Common use of a CNC machine requires no computer other than the on-line controller. To make a part, a tool programmer converts a drawing into a handwritten set of instructions that are entered one step at a time through a keyboard mounted on the controller. To test the program, the machine operator must "cut air" to be sure that there are no dangerous errors, then run it again, cutting a practice part out of wood or some other inexpensive material. If errors are found, the program is modified in the controller and the test repeated until the practice part is correct. Then the operator uses the controller to punch a paper tape containing the correct program. After all this, the operator is now ready to cut real parts.

If the machine is to be used to make 100 parts, there is some justification for all the unproductive machine time needed to debug the program—it adds maybe 5 or 10 percent to the cost of each part. If only one part is to be made, program debugging and paper tape generation can easily take ten times as long as part production, and in fact, often takes much more. Actual tool setup time, in which tool lengths are measured and the machine is fixtured to hold the stock from which the part will be made, is likely to add at least another 19 non-productive hours, just to prepare for one hour of productive use. We determined that the actual figure is almost twice as bad, partly because of mistakes that are not detected until a supposedly finished part is checked. In other words, in a prototype shop, using standard CNC machine practice, we could except one productive hour per 40-hour week. Either we should use CNC equipment only for runs of at least 100 parts, or we should make radical changes in our method of preparing jobs. Since we rarely saw long runs, a new methodology was essential.



Mike Massara, left, and Duane Piper testing the postprocessor program that links into Monarell milling machine to the SUPERCAM system.



SUPERCAM DATA FLOW

Remedial measures

We decided as a first step to use a Tektronix 4052 desk-top computer to prepare controller programs off-line, and to download them into the controller electronically. This not only simplified error detection and correction, but also eliminated the time spent punching paper tape. Did it make sense as the next step to use Computervision, or any CAD/CAM systems, to produce CNC parts? In 1982, no off-the-shelf CV-compatible program was available to convert tool paths into the language understood by the controller of our CNC machine. (These programs are called "postprocessors" because they are used to convert general-purpose data into specific controller language, usually the last step in the computer process.) We learned that it would take about a year to get a post-processor from the vendor. Furthermore, most of the jobs to be done in the shop were designed manually, not on the CV. Converting a manual drawing to a Computervision model would be an extra step, adding to the cost of producing the part in terms of both dollars and time. Nevertheless, we were using the CV to design some parts that were to be made in the shop; it would be necessary to make a post processor available. We decided to use the Tektronix computer to read tool path files directly from the Computervision system and then postprocess and download them into the CNC controller.

We also built two special-purpose programs on the Tektronix computer, one to make cams and the other to make linkages. Each program was self-contained in that it generated and downloaded code for the CNC controller. It was at this time that the Model Shop decided to get a second CNC milling machine, prompting us to look at the direction we were taking and to propose a unified system based on the Tektronix programs already developed. Two factors influenced the proposal—CNC controllers do not all speak the same language, and there was no existing software package that could be used for the majority of parts made in the shop. We also wanted to build a system that would be transportable to any RCA division with a minimum of effort.

Naturally, the first order of business was the invention of a suitable acronym. I came up with the name "SUPERCAM." The meaning of the letters has been lost to history, but the name has a nice ring to it, and it also contains the letters "CAM" for computer-aided manufacturing, and "RCA." SUPERCAM would provide for fast addition of new CNC machines and new CAD/CAM systems with a minimum of additional coding, and at the same time would minimize training requirements for CNC operators.

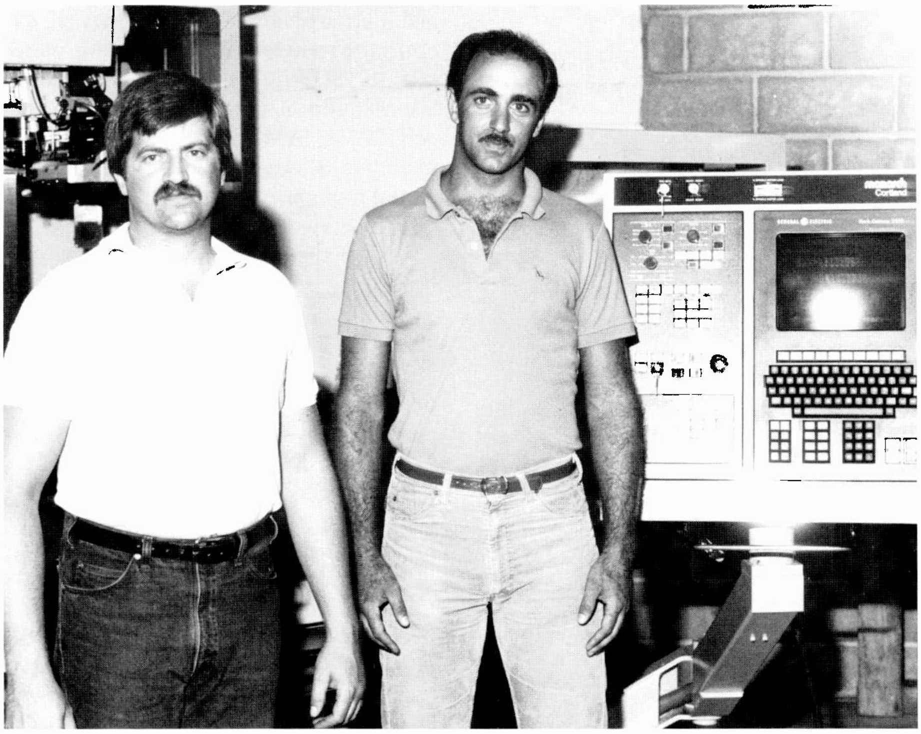
While a good software package was needed, an equally big problem was how to optimize productive use of CNC machines. In a manual shop, each machinist is provided with his own machine. He does jobs one at a time, not starting a new job until the previous one is finished. Usually, he does almost all the work himself. On CNC equipment, the essential element in cost-effectiveness is maximization of productive time. In 1983, a group of five machinists responsible for operation of CNC machines was transferred from the shop to Mechanical Design Techniques Research in order to exercise a cost-effective methodology for prototype computer-aided manufacturing. SUPERCAM was developing as an effective software product, but to prove the value of CNC in the model shop it was also necessary to change the way in which the equipment was used.

If we were to make CNC pay off, we had to remove from the machine all functions that were not necessary in the manufacture of actual parts, and perform them off-line. Furthermore, we had to use all the available machine time—if the machine stood idle while jobs were waiting to be done, then it was just as wasted as if it were being used unproductively.

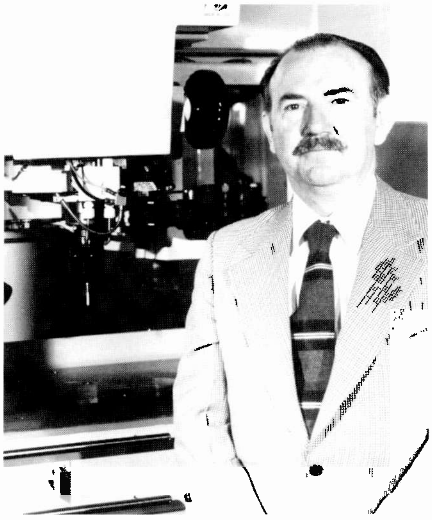
Experimental control

The first thing we did was to stop scheduling the CNC machines. It was obvious that when the machine was scheduled on a job-by-job basis, any delay in one job caused a delay in all jobs backed up behind it. Oddly, this reminded me of the Berlin Airlift. In 1947, when the Russians blockaded Berlin, President Truman instituted an airlift of supplies into the city from West Germany. The landing schedule was so tight that if a plane failed on its first attempt to land, other aircraft would stack up while waiting for it to go around again, resulting in a hopeless tangle. An Air Force officer finally came up with the idea of sending back to home base those planes unable to land on the first try, and the scheduling problem disappeared.

We decided to use a modification of the Berlin Airlift in dealing with job fail-



John Margicin, left, CNC group leader, works with Frant Bennett on the Monarch illing machine.



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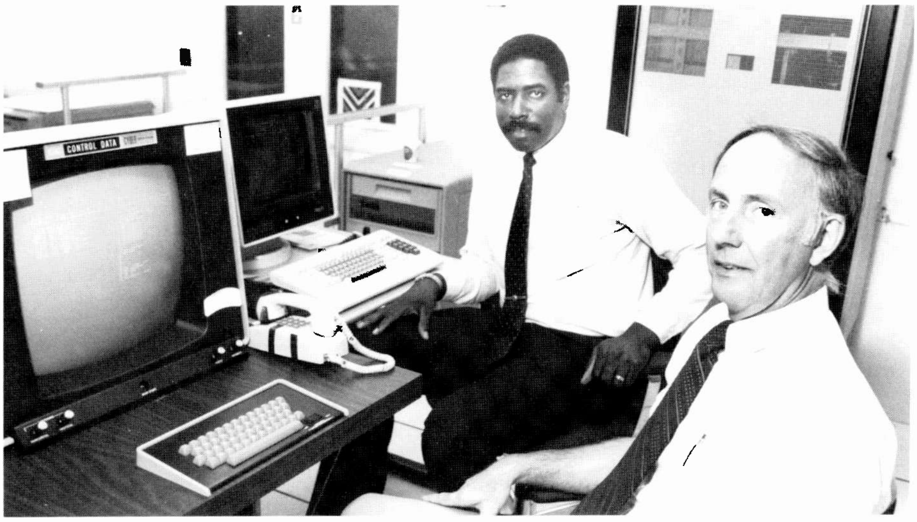

Jim Coleman, left, and Dennis Quardt monitoring the operating system of the Control Data mainframe computer.

ures in the CNC shop. Instead of doing jobs in the order in which they arrived, we set up a box for each job. When a box contained all the necessary pieces—drawing, fixtures, raw stock, and tools—and the program had been generated in the Tektronix system, then the job was ready for the machine. Machinists worked on jobs in the order in which they were received (barring high priority), but if something delayed the completion of a box it would be bypassed in favor of a completed one. Every effort was made by the machinists to detect errors in tool paths in the Computervision and/or the Tektronix computer by careful evaluation of the graphical and numerical displays provided by the software. In the event of an error that could not be rectified immediately, a job was to be removed from the CNC machine and the next one started. The job in error would return to the machine as soon as all the contents of its box had been correctly replenished. Tool lengths, needed for the computer to calculate movement, would be measured by a tool gauge, not by using the machine's controller. Automatic tool changers would be ordered as part of all new CNC machines, so that once a job was initiated, the operator could leave the machine and prepare a new job. The operator would now work at the computer terminal more than at the CNC machine. When several boxes were waiting to be processed, the CNC group leader would select the next job to be cut based on minimization of total waiting time—in general, the shortest jobs would be done first.

Jobs that could be done in SUPER-CAM would be entered by the machinist from the engineering drawing provided. The machinist therefore would have complete responsibility for the accuracy of the part. Jobs originating on a CAD/CAM system would also be the machinist's responsibility, so he would specify the tool paths to be put on the model by the CAD terminal operator. He would then check the finished tool paths on the CAD/CAM terminal, and he would check them again when they appeared on the Tektronix screen.

Dramatic improvements

As the software and the method of operation matured under the direction of Dennis Quardt, I monitored productivity in the CNC shop by collecting data reported by the group leader on job processing steps. I used a spreadsheet program on a PC to summarize results. Beginning in April, 1984, I watched a steady improvement in the number of parts completed per day. In the nine months ending December 31, 1984, five machinists, using three CNC machines, completed 996 parts. That worked out to about one part per machine every five working hours, or eight parts per 40-hour week. Allowing two of those five hours for machine setup, by traditional CNC methods those eight parts would have required 24 weeks.

A new record was set, however, on March 4, 1985, when the CNC shop turned out is thousandth part since January 1—an astounding seven parts per day per machine. A close look at the data revealed a biasing factor; three long runs of close to 200 parts each occurred in the period. However, it is safe to assume that the SUPERCAM methodology is successful not only in a prototype shop but maybe even more so where long runs are involved.

The return of the CNC shop to the Model Shop on August 1, 1985, marked the end of the experiment. It is entirely accurate to say that an essential part of the success of SUPERCAM has been the active and interested cooperation of the CNC shop staff. Marv Leedom, speaking to the entire shop, told them that he had seen many machine shops in his business travels, but none with the response time of the RCA Laboratories Model Shop.

We are planning to develop SUPERCAM II beginning in 1986, making it more transportable and easier to use. It will be written in Fortran 77 and implemented on the IBM PC/AT.

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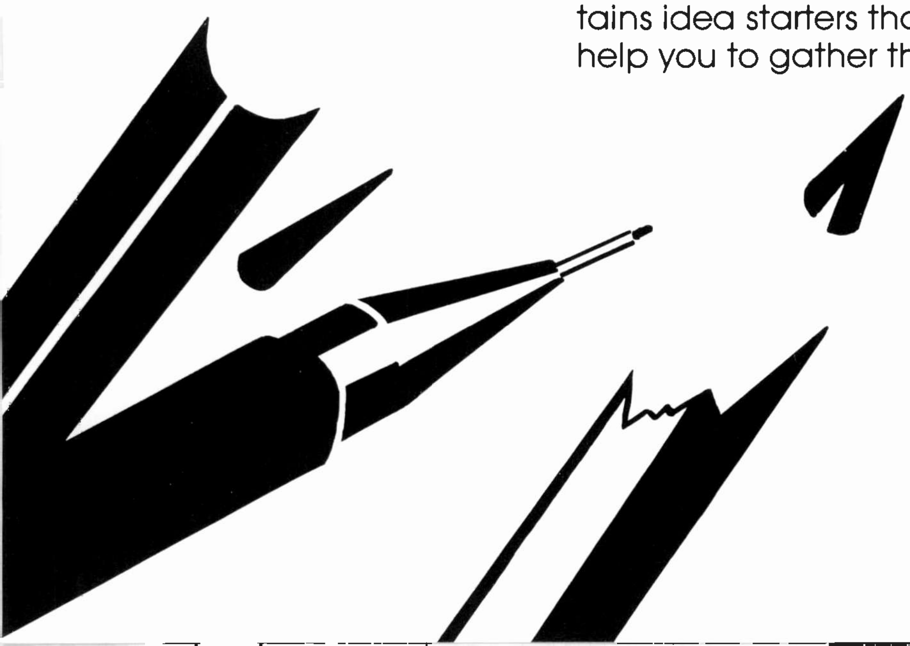

R. Cahoon | N. Meliones | B. McDermott

Computer-aided design of a lightweight electronic rack

At ASD, the dynamic visual display of flexural characteristics, prediction of natural frequencies, and portrayal of stress distributions were valuable tools in the design of a rack for an electronic test system.

The rapid creation of an innovative design in a competitive situation requires aggressive application of skills and tools. The traditional trial and error, build and test method is frequently costly in time and money, and in many cases ineffective as a prediction tool.

The proposed highly mobile electronic test system for forward combat areas required a unique packaging concept and critical control of weight so the system could be transportable over cross country terrain without exceeding limiting truck capacities. Structural integrity was mandatory over a broad range of dynamic conditions.

Computer-aided design using the finite element method can be applied to rack and chassis design to produce a lightweight, structurally optimized rack design.

Abstract: *Computer-aided design using the finite element method resulted in a lightweight, structurally optimized rack for housing electronic equipment in a highly mobile military application. The approach permitted corrections and improvements in advance of fabrication. The dynamic visual display of flexural characteristics, prediction of natural frequencies, and portrayal of stress distributions were valuable tools in converging the design. The method resulted in significant time and effort reduction. Physical confirmation of design was made by fabricating a mockup and vibration testing it with satisfactory results.*

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Objectives

The objectives were to develop, in a minimum amount of time, a lightweight, rugged, low-cost rack design that will be the standard rack for a highly modular automatic checkout electronic system. The shelter-installed electronics are to be transportable by road, air, rail, and water to and within all active theaters of operations. When mounted, the system must be able to withstand movement without any adverse effects on its operational ability. Furthermore, the system must be transportable over main and secondary roads at speeds commensurate with existing road and traffic conditions up to 55 mph and to be able to withstand movement over unimproved roads, off roads, and cross country terrain.

Approach

The approach is to describe the rack conceptual design, define the anticipated dynamic environment, detail the CAD equipment and software package use, report on computer results and adjustments, and report on dynamic results on a mockup that had been tested.

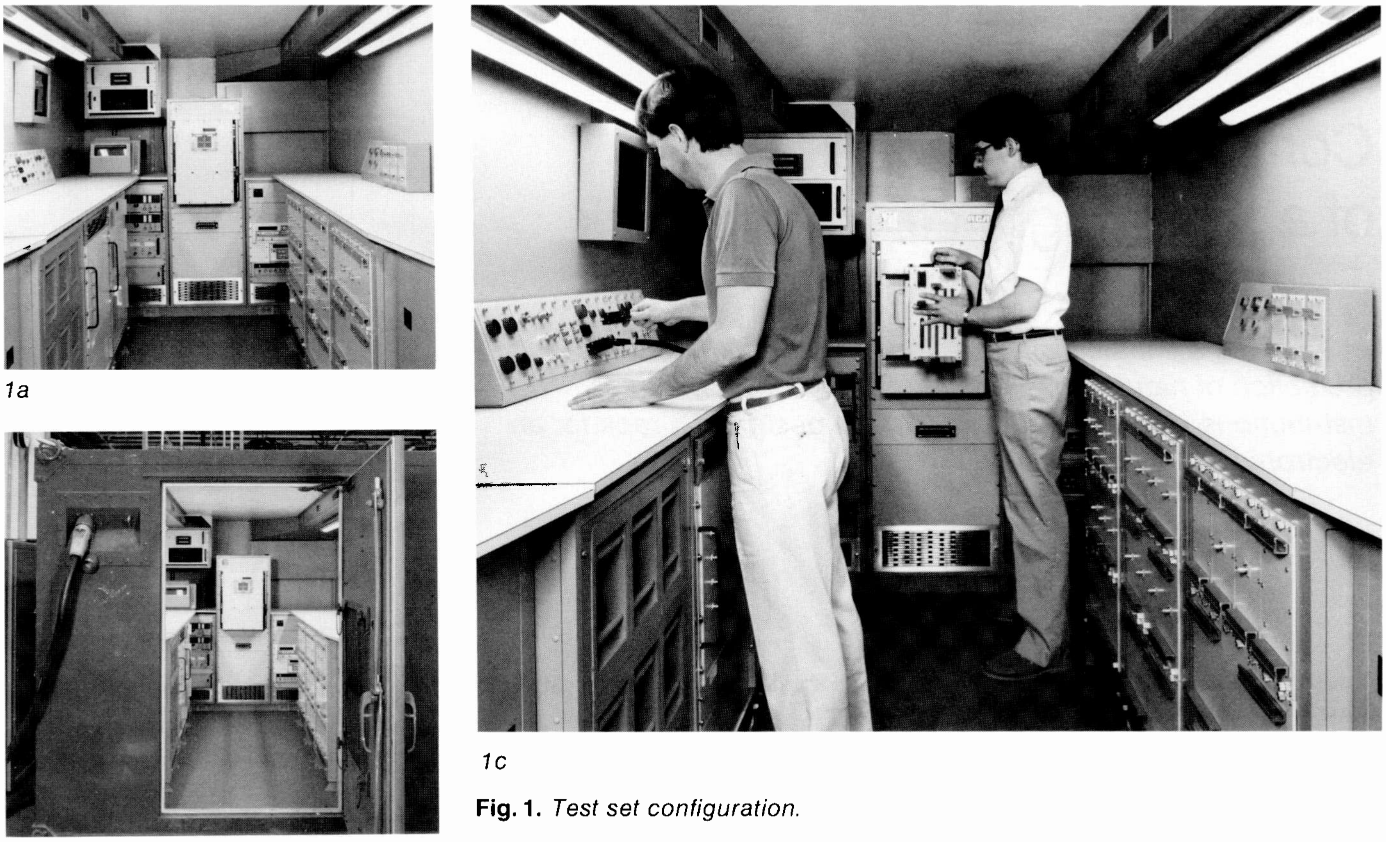
Conceptual design

The general shape of the rack was influenced by the shelter's geometric restrictions. Figure 1 is a functionally operating mockup used for human factors optimization. Some of the shape influencing factors were as follows:

- ☐ Short racks to provide large work surface areas for units under test.
- ☐ Shallow-depth racks along the curb and road sides to allow one operator to pass another operator in the aisle while carrying a unit in hand.
- ☐ Deep racks at the forward wall to accommodate chassis of greater depth.
- ☐ Rack backplanes adaptable for plug-in module reconfiguration.
- ☐ Recessed module panels to allow for connector indentation.

The structural characteristics of the design concept are as follows:

- ☐ Standard rack assembly features:
  - Lightweight welded tubular aluminum extrusions with stiffened, riveted, and gusseted corners.
  - Module guides at standardized levels.
  - Short racks for most electronics.
  - Rack-to-rack attachment at front.
- ☐ Standardized rack attachments:
  - Through common rails which are secured to a shelter floor.
  - Through common wall brackets which are secured to shelter walls.
- ☐ Standardized Plug-In Module Features for Structural Integrity:
  - Alignment and shear pins at rear.
  - Module guides along the module depth.
  - Quick turn captive fasteners at front panel.
  - Brazed construction.
  - Recessed front connector panel.



1a

1b

1c

Fig. 1. *Test set configuration.*

☐ Standardized module backplanes rigidized with frame tying into rack structure.

Dynamic environment

The dynamic environment that the equipment must meet is detailed in Table I, along with anticipated rack input levels.

Required shock survivability levels are anticipated to be 22*g* (minimum) based on available test data taken at a variety of locations during qualification tests of the AN/TSC-86 tactical communication system. This peak *g* level will be used for ultimate strength adequacy considerations longitudinally and transversely. Transit drops using skids also resulted in shock levels of similar order (20*g*).

In the past, MIL-STD-810 vibration testing for this equipment would have been ±1.5*g* sinusoidal vibration sweeps. With measured transmissibilities of 10, the expected *g* level would be 15 at critical resonance. The current MIL-STD-810 dynamics testing approach is to specify random vibration. The axis taken for critical analysis is the one perpendicular to the front face of the rack. Assuming three standard deviations and transmissibilities of 10 also results in response *g* levels of approximately 22*g*. This level would be exceeded only 0.3 percent of the time.

CAD equipment and software

Applicon's Graphic Finite Element Modeling (GRAFEM) and Integrated Finite Element Analysis and Design (IFAD) software packages and Applicon's Engineering Work Station were used in conjunction with a DEC VAX 11/780 computer. The GRAFEM software package provides the following features:

☐ Rapid construction, editing, and display of 2-D or 3-D finite element models.

☐ Automatically generated 2-D-planar, 3-D-surface, and 3-D-volumetric meshes.

☐ Comprehensive editing capabilities for loads, boundary conditions, and element properties.

☐ Full user control over node and element distribution; automatically numbered nodes and elements.

☐ Automatic facilities for model checking and verification.

☐ Editor commands for geometric creation, manipulation, and display.

☐ Direct interface with the Integrated Finite Element Analysis and Design (IFAD) package.

☐ Displayed results of analysis as deflected shapes, mode shapes, and stress/strain contours highlighted by color contour plots.

The IFAD software package provided the following features:

☐ Furnishes an integrated, easy-to-use facility for finite element analysis.

☐ Performs linear static analysis on finite element models created with Graphic Finite Element Modeling (GRAFEM) package.

☐ Performs dynamic modal analysis on linear structures.

☐ Contains a library of finite elements, including bar, beam, spring, shell, planar, and solid elements.

☐ Outputs directly to GRAFEM package for results interrogation and display.

The initial steps of analysis, in simplistic form, are as follows:

1. Geometry definition.
2. Physical properties definition (materials, geometric shapes, type of element).
3. Forces definition.
4. Restraint definition.
5. Results portrayal.

Computer procedure

In conjunction with a layout, a computer model was created of a basic rack design

Table I. *Dynamic environment*

Requirement	Test procedure invoked		Anticipated rack input *g* level
Methods of transportation —Worldwide transport/shipment by air, ship, rail and ground without damage or degradation	Random vibration individual Equipments, MIL-STD-810D Procedure 1 shedules per random Vibration figures		
	514.3-1 vertical 1.04*g* rms overall		1.04*g* rms*
	514.3-2 transverse 0.20*g* rms overall		0.20*g* rms*
	514.3-3 longitudinal 0.74*g* rms overall		0.74*g* rms*
	Transit drop assemblage, MIL-STD-810D, Method 516.3, Procedure IV, transit drops 18 inch bottom, edge and flat		20*g* peak (skid mounted)
—Railway humping up to 10 mph			22*g* peak (shelter corner)
Vehicular transport across cross country terrain	Assemblage, MIL-STD-810D Method 514.3 Procedure II Aberdeen Proving Ground		
	Coarse washboard 6 inch waves	5 mph	3.5*g* peak (cargo bed)
	Belgian block	20 mph	5.5*g* peak (cargo bed)
	Radial washboards 2 inch to 4 inch waves	15 mph	No data
	2 inch washboards	10 mph	10*g* peak (cargo bed)
	3 inch space bump	20 mph	11*g* peak (cargo bed)

*Synthesized levels on variety of vehicles and transport

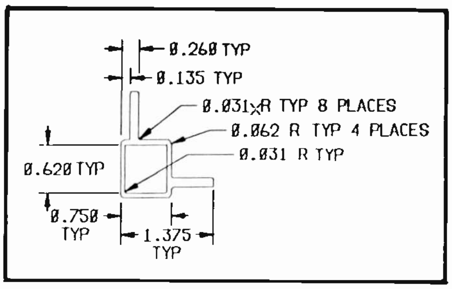


Fig. 2. *Extrusion cross section.*

that had the following design goals:

1. Structural integrity for mobile military environment.
2. High resonant frequency.
3. Lightweight.
4. Low-cost fabrication techniques.
5. Complete accessibility of electronics from front for operation and maintenance.

The initial computer model consisted of a basic rack framework with nodes and meshes. No internal structure or representative mass was included. All six shelter attachment points were assumed to be rigidly restrained.

A family of extrusions was reflected in the model that provides mounting flanges for electronic chassis and modules, mounting flanges for side, top and rear panels, a degree of stiffness, and a fit within the budgetary spatial restrictions of the shelter and the necessity of accommodating a 19-inch wide standard chassis. A representative extrusion cross section is shown in Fig. 2, and the properties of this extrusion are in Table II.

The purpose of this model was to estab-

Table II. *Section property calculations*

Area	=	0.3288 inch^2
Xbar	=	0.4962 inch
Ybar	=	0.4962 inch
Properties about the reference coordinate system		
Moment of inertia about x-axis	=	1.3174×10^{-1} inch^4
Moment of inertia about y-axis	=	1.3174×10^{-1} inch^4
Product of inertia	=	5.7092×10^{-2} inch^4
Ref. x-axis to princ. x-axis angle	=	45.00 degrees
Maximum moment of inertia	=	1.8877×10^{-1} inch^4
Minimum moment of inertia	=	7.4714×10^{-2} inch^4
Maximum radius of gyration	=	7.5776×10^{-1} inch
Minimum radius of gyration	=	4.7672×10^{-1} inch
Properties about the centroidal coordinate system		
Moment of inertia about x-axis	=	5.0795×10^{-2} inch^4
Moment of inertia about y-axis	=	5.0795×10^{-2} inch^4
Product of inertia	=	-2.3919×10^{-2} inch^4
Cent. x-axis to princ. x-axis angle	=	45.00 degrees
Properties about the principal axes system through centroid		
Maximum moment of inertia	=	7.4714×10^{-2} inch^4
Minimum moment of inertia	=	2.6876×10^{-2} inch^4
Maximum radius of gyration	=	4.7672×10^{-1} inch
Minimum radius of gyration	=	2.8592×10^{-1} inch

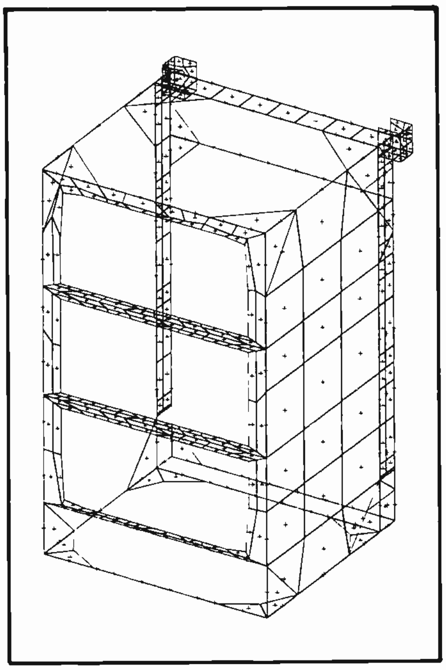


Fig. 3. *Combined model.*

Table III. *Model frequency summary*

Model Name: NICKRAC2

Model Description: Fully loaded rack with front module panels

Modal Shape Number	*Freq. (Hz)*	*Description*
1	36.7	Backplane/front frame moving in synch. See Figure 4. Some main frame movement due to back tie down flex.
2	43.8	Backplane/front frame moving in synch. S-shaped deflection. See Figure 5. Some main structure movement due to back tie-down.
3	50.2	Backplane/front frame moving in synch. S-shaped deflection. See Figure 6. Some main structure movement due to back tie-down.
4	57.1	Main frame movement (left to right) considered 1st nat freq. of major struct. See Figure 7.

lish a reference natural frequency of the basic rack frame structure. This was dynamically calculated to be 126 Hz. Subsequent analyses were then compared to the reference frequency.

A model of the front frame and backplane support assembly was developed next. This assembly fits within the rack frame but was modeled separately so that its mode shapes could be viewed easily. There are 49 common node locations with this model and the basic rack frame work. These nodes represent rivet, fastener, and weld points that secure the two together. Electronic module weights were included by adding lumped masses at the front frame attachment points and at the dagger pin receptacle locations in the backplane support structure. A dynamic analysis was performed of this model to determine natural frequency. This was computer calculated to be 38 Hz (first modal frequency) into and out of backplane. Both models were then combined into a composite model. Figure 3 depicts the combined model.

All 49 common node locations were merged together, module weights added as lumped masses, and a dynamic analysis performed to determine natural frequencies. The first modal frequency of the front frame/backplane support assembly was computer calculated to be 36 Hz. This was expected since the rack frame is not as stiff as the 49 fixed nodes. The rack frame first modal frequency dropped from the reference frequency of 126 Hz to 44 Hz. As expected, the module weights caused the drop in frequency.

The next computer model added module front panels. The effects of this addition on the front frame/backplane support assembly were small, raising the first modal frequency to 37 Hz. Rack frame stiffness was elevated to 57 Hz.

Finally, a 40g shock load was applied to the $+Z$ direction to determine backplane support deflection that could result in disengagement of electrical connectors. A 0.40-inch backplane deflection was produced from dynamic loading, and was considered unacceptably high.

The conclusions that were derived from this initial phase of computer analysis were as follows:

1. The module structure must be added to the combined model and reanalyzed.
2. The backplane/front frame assembly must be stiffened. This can be done by either stiffening the backplane, the front frame, or both. It was recommended that the backplane support be stiffened with ribs combined with a weight saving reduction in backplane support thickness from 0.19 to 0.12 inches. In addition, it was recommended that horizontal beams (channels) be added to the front frame assembly. This would raise the natural frequency of both the rack frame (laterally) and backplane assembly (transversely), while reducing backplane deflection.
3. A rack frame/shelter wall attachment plate exists between two horizontal structural members in the back of the rack. This plate can be eliminated and the lower of the two horizontal members elevated for shelter wall attachment stiffness. The addition of small corner gussets will replace the lost shear strength.
4. Structurally secure the rack frame to the shelter wall by a direct tie to the rack frame rather than to an intermediate rack plate or structural member.
5. The rack frame first natural frequency can be elevated from 57 Hz by securing adjacent racks together.
6. The $+Z$ direction shock load must be reduced to a more reasonable level.

The computer-calculated natural frequencies for various structural assemblies before the implementation of the recommendations can be found in Table III, and some associated modal shapes are depicted in Figs. 4 through 7.

To examine each of the recommended conclusions from the initial computer analysis, a new model was prepared. The backplane/front frame assembly had experienced excessive backplane deflection and therefore potential disengagement of electrical connectors, and the low first natural frequency provided justification for the addition of stiffness. Aluminum channel stiffeners were added as horizontal beams immediately behind the backplane support. See Fig. 8 for a description of the channel geometry and Table IV for the sectional properties.

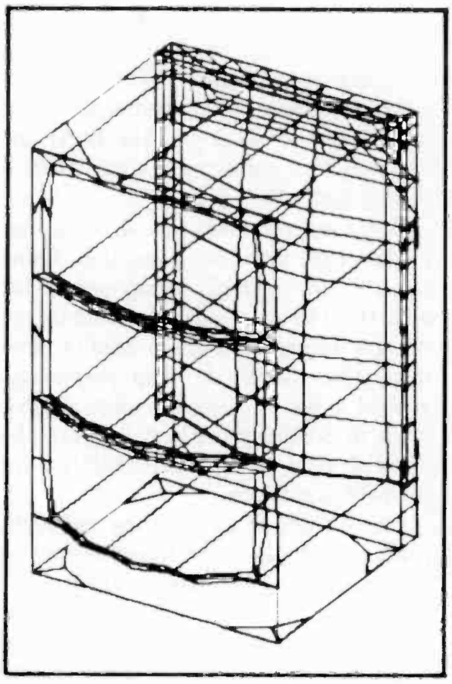


Fig. 4. *Mode shape 1.*

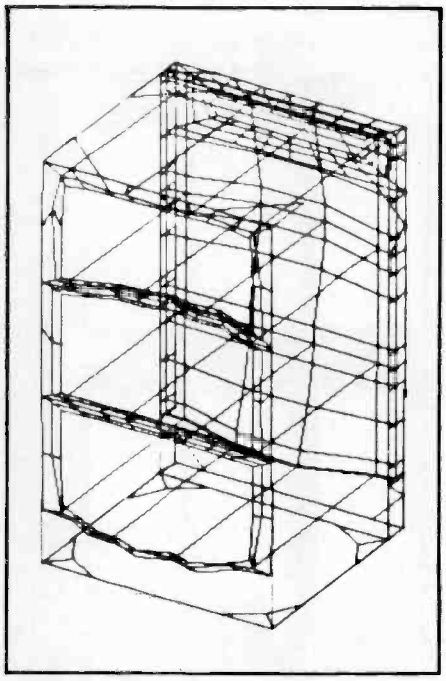


Fig. 5. *Mode shape 2.*

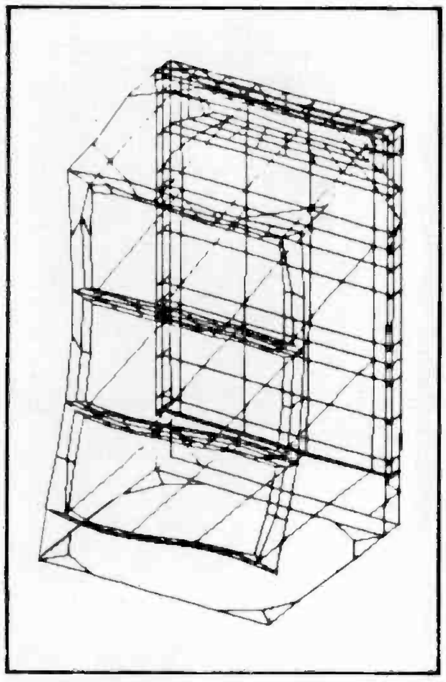


Fig. 6. *Mode shape 3.*

Table IV. *Channel section property calculations*

Area = 0.4686 inch²
Xbar = 1.8988 inch
Ybar = 0.2956 inch

Properties about the reference coordinate system

Moment of inertia about x-axis = 8.4824×10^{-2} inch^4
Moment of inertia about y-axis = 7.4424×10^{-1} inch^4
Product of inertia = 1.3849×10^{-1} inch^4
Ref. x-axis to princ. x-axis angle = 11.38 degrees
Maximum moment of inertia = 7.7213×10^{-1} inch^4
Minimum moment of inertia = 5.6398×10^{-2} inch^4
Maximum radius of gyration = $1.2837 \times 10^{+0}$ inch
Minimum radius of gyration = 3.4694×10^{-1} inch

Properties about the centroidal coordinate system

Moment of inertia about x-axis = 4.3348×10^{-2} inch^4
Moment of inertia about y-axis = 2.7569×10^{-1} inch^4
Product of inertia = -2.7638×10^{-11} inch^4
Cent. x-axis to princ. x-axis angle = 0.00 degrees

Properties about the principal axes system through centroid

Maximum moment of inertia = 2.7569×10^{-1} inch^4
Minimum moment of inertia = 4.3348×10^{-2} inch^4
Maximum radius of gyration = 7.6705×10^{-1} inch
Minimum radius of gyration = 3.0416×10^{-1} inch

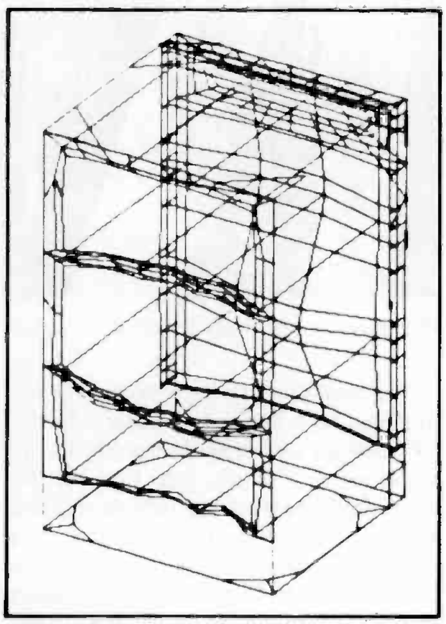


Fig. 7. *Mode shape 4.*

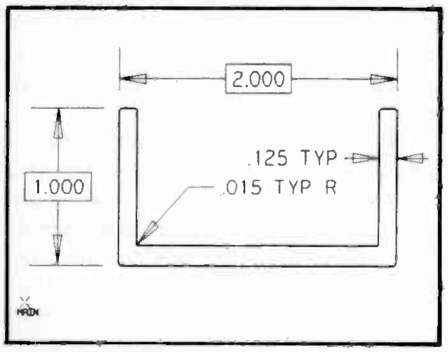


Fig. 8. *Channel geometry.*

Fig. 9. *Rack assembly on vibration table.*

The rack assembly in mockup form was mounted on a horizontal shaker table for vibration in a direction perpendicular to the front face of the rack (see Fig. 9). Sinusoidal vibration was used to establish a natural frequency. It was applied at a constant acceleration level of 1.0*g* through the range of 5 to 200 Hz. This level was of sufficient magnitude to excite the structure but not cause unwanted damage. The results indicated a good degree of correlation to the computer calculated value. The first resonance (along the Z axis) occurred at 35 Hz, with a measured magnitude of about 10*g*.

To simulate the movement of this electronic system via military truck over road conditions encountered during mission and field transport, the electronic rack assembly was subjected to the random vibration profile described in MIL-STD-810D, Fig. 514.3-3, for the longitudinal axis (X axis depicted in Fig. 10). Accelerometers mounted on the base of the rack assembly measured 0.7*g* (rms) magnitude during this test, and the rack assembly survived without any damage.

A study of the computer model analysis, graphic display, and the testing results concluded that the entire assembly was rocking about the mounting rails at the rack base. A computer generated right-side view of the rack assembly depicts the magnitude of deflection of the mounting rails and the front shear plate superimposed on the undeformed structure (see Fig. 11). A computer generated static analysis was performed. The entire rack was modeled at the following loading:

X direction +20*g*
Y direction +20*g*
Z direction +20*g*

Figure 10 orients the axes. We felt that the acceleration levels imposed in the rack model more accurately reflect "real world" loadings encountered by electronic systems during the various transportation modes. The maximum deflection of any node in the model due to the above loading was computer calculated as 0.0737 inches. Typical rack frame elements were subjected to stress levels of 8000 psi. The backplane stiffeners were subjected to stress levels of slightly under 5000 psi.

When tested dynamically, the computer model of the new backplane/front frame assembly first modal frequency occurred at 62.8 Hz. The stresses encountered by the stiffness during a 30*g* shock load (a more reasonable magnitude than previously) applied in the +Z direction reach a maximum of 30,000 psi, which is below the material yield strength of 35,000 psi for 6061-T6 aluminum.

The stresses encountered by the backplane support plate were found to be comparatively low. The extreme stress values range from −1000 to +1000 psi. These values are certainly within a safe operational range. The maximum backplane assembly deflection for a 30*g* load was determined to be 0.36 inches. This is still considered excessive because of potential disengagement of electrical connectors.

The rack frame/shelter wall attachment plate was removed from the rack model and replaced by four gussets, and the lower of the two horizontal structural members elevated. The rack frame was then secured to the shelter wall from the two rear vertical structural members with short sections of 2 inch by 2 inch aluminum channel. A dynamic analysis of the modified bare rack frame indicated that resonance occurred at 70 Hz.

When the modified bare rack frame and the backplane/front frame assembly with lumped masses were merged together and subjected to a 30*g* load (−Z direction), certain frame members were subjected to stresses at or greater than the material yield strength. The area of greatest stress occurred in the rear wall attachment members, just outside the restraining points. These stresses exceeded the ultimate strength of the material, so additional analysis is warranted.

A rack frame containing a front frame and backplane support assembly was fabricated. Modules containing weights to simulate electronic assemblies were also fabricated and installed in the rack assembly, as was a cooling system. The object of this model was to verify the computer data through dynamic testing on a vibration table. The weight of fabricated model was 268 pounds.

These results indicate an adequate design with satisfactory margins of safety and a deflection in the backplane that will not

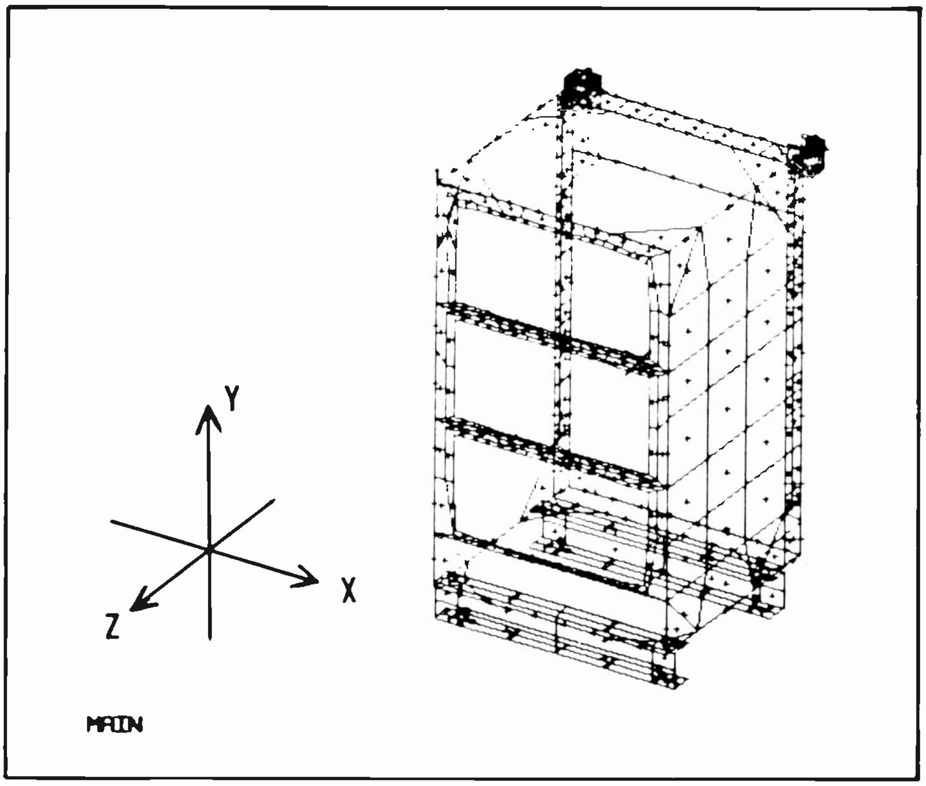


Fig. 10. *Mounting rails added.*

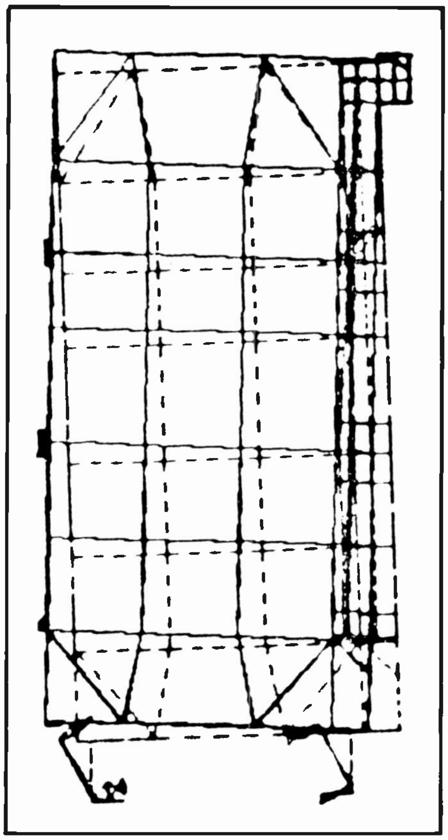


Fig. 11. *Mounting rail deflection.*

allow connector contact disengagement. A study of fatigue strengths of the rack corner structural members when viewed with the computer calculated stress levels was then initiated.

Figure 12 is a stress versus number of cycles to failure curve for the material of the rack structural extrusions, 6061-T6 aluminum alloy.[1]

We noted during the static analysis that the computer calculated maximum stress levels of 8000 psi occurred on the vertical rack structural members. This occurred during the triaxial shock loading of 20*g*. The random vibration of this rack assembly developed 0.7*g* (rms) in the most susceptible direction (Z axis). This value, when considered at 3 sigma limits at a transmissibility of 10, is comparable to the 20*g* Z axis static loading, and therefore will produce a stress level of 8000 psi. This stress level is below the stress-cycle curve asymptote of 10,000 psi. Therefore, the fatigue life of any structural member is never greater than the critical value.

The next computer model contained rack mounting rail reinforcement sections. A dynamic analysis of this configuration indicated that the first modal natural frequency occurred at 47.9 Hz, a substantial increase from earlier predictions. In addition, the graphic animation of the static loading effects on the base structure indicated very little rocking action compared to that depicted in Fig. 11.

Additional computer models are projected for the future to study the effects of securing each rack assembly to adjacent rack(s) and the augmentation of stiffer rear attachment brackets. The existing hardware model will be revised to reflect the computer directed changes, and it will be tested once again in the vibration table. We predict that this configuration will have a first natural frequency above 50 Hz.

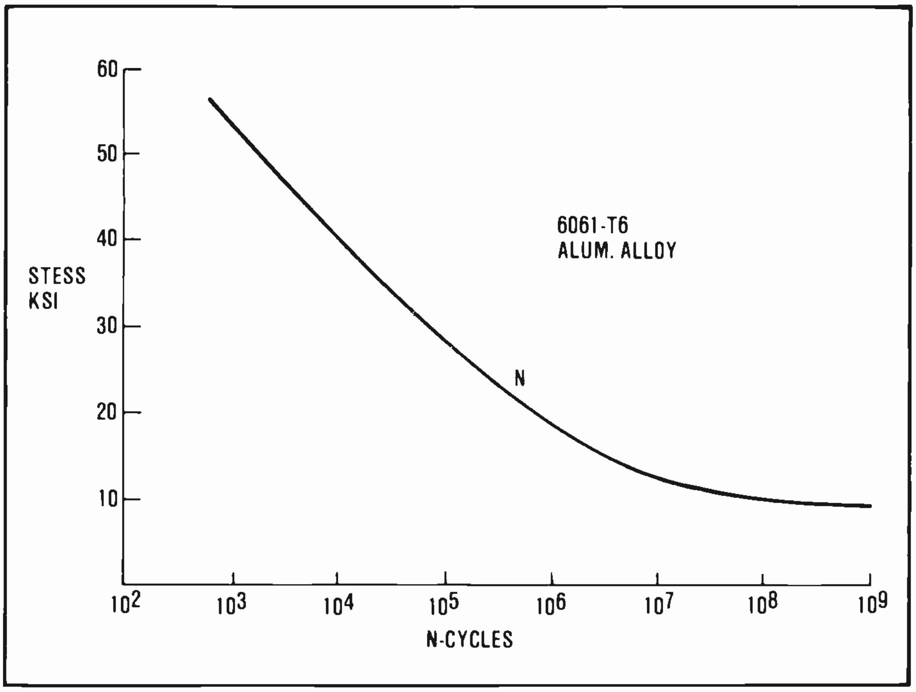


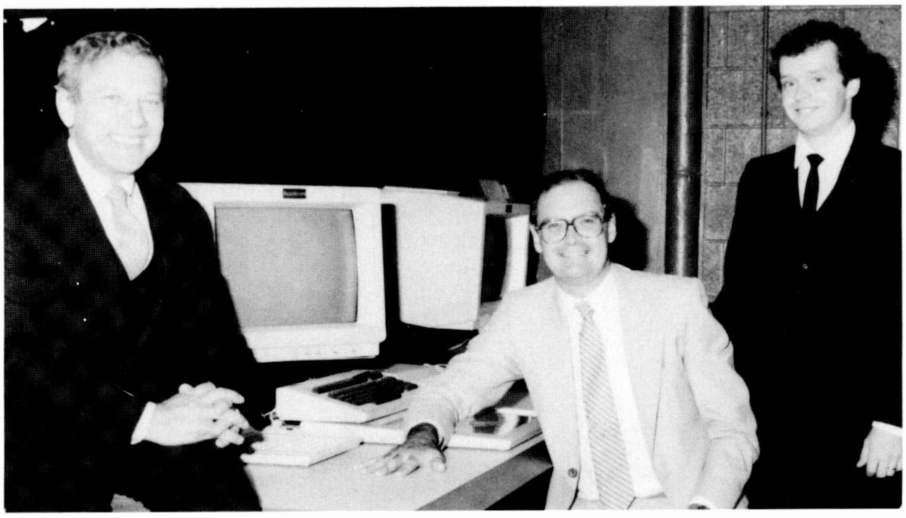
Fig. 12. *Stress versus number of cycles to failure.*

Conclusion

Computer-aided design using the finite element method resulted in a lightweight, structurally optimized rack for housing electronic equipment in a highly mobile military application. The approach permitted corrections and improvements in advance of fabrication. The dynamic visual display of flexural characteristics, prediction of natural frequencies, and portrayal of stress distributions were valuable tools in converging the design. The method resulted in significant time and effort reduction. Physical confirmation of the design was made by fabricating a mockup and vibration-testing it with satisfactory results.

References

1. Frankel, M.S., "A Random-Vibration Fatigue Analysis," a supplement to RCA Continuing Engineering Education Course, ME-1.



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R.C. Guyer

Mini Laser Rangefinder

This new laser rangefinder is less than half the size and weight of its predecessor, which was the most successful instrument of its type in industry.

Today, the number of AN/GVS-5 laser rangefinders built by Automated Systems Division exceeds 7000, and we have assumed the position as the world's leading manufacturer. First developed in 1975, the success of the AN/GVS-5 is a result of its high performance and reliability at an affordable price. A second generation production laser rangefinder—half the weight and 60 percent smaller (see Fig. 1)—is now being manufactured by Automated Systems Division with the same success criteria. It is called the Mini Laser Rangefinder, and its first users will be the U.S. Army Special Forces Teams.

Abstract: *The Mini Laser Rangefinder, a second-generation production system designed for the Army, accurately measures and displays the distance to battlefield targets. Designed to fit in a field jacket pocket, it is half the weight and 60 percent smaller than its predecessor, the AN/GVS-5. The Mini Laser Rangefinder was designed for complete interchangeability of modules without realignment or readjustment. It was designed to minimize production costs by eliminating unnecessary parts and precision interfaces, by incorporating a simpler, more easily made optical system, and by using precision plastic molded parts wherever possible.*

This paper describes the design evolution, performances vs. weight tradeoffs, and packaging/material innovations of this new production laser rangefinder.

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History

In 1961, accurate distance measurement from an observer to a target without cooperative reflectors was the first military use of the newly invented laser. RCA received the first military contract to develop a portable laser rangefinder in 1962. The first system consisted of two rather large electronic chassis with a combined weight of one-hundred pounds and required a two-man rangefinder team to operate the equipment. Evolutionary advancements in microcircuit electronics and laser and detector technologies at RCA and in competing equipment made the AN/GVS-5 possible in 1975, and the Mini Laser Rangefinder possible today. RCA won initial design and production contracts for each of these lasers.

By 1981, however, considerable competition for the AN/GVS-5 began to develop by foreign and domestic companies. Drawings of this laser were government-owned, and RCA had to compete with other U.S. companies to obtain production contracts. The time to introduce a new model is when the competition begins to catch up and the market for the earlier model begins to be satisfied. Design of an improved laser, designated the Mini Laser Rangefinder (MLRF), began in 1981 by essentially the same design team of laser, optical, mechanical, and electronic engineers that created the AN/GVS-5. The successful criteria of the MLRF's predecessor were retained while incorporating years of marketing, manufacturing, and operational experience.

Clearly, the new laser should be smaller, and RCA's competitors were demonstrating small prototypes, one the size of a pack of cigarettes. But small is not necessarily better. These prototypes suffered reliability problems and were so light they could not be held on target. The bounds of a new design were set, then, by the AN/GVS-5 and a pack of cigarettes.

To fully appreciate the performance/size tradeoffs necessitated by these new bounds, and the design innovations that overcame

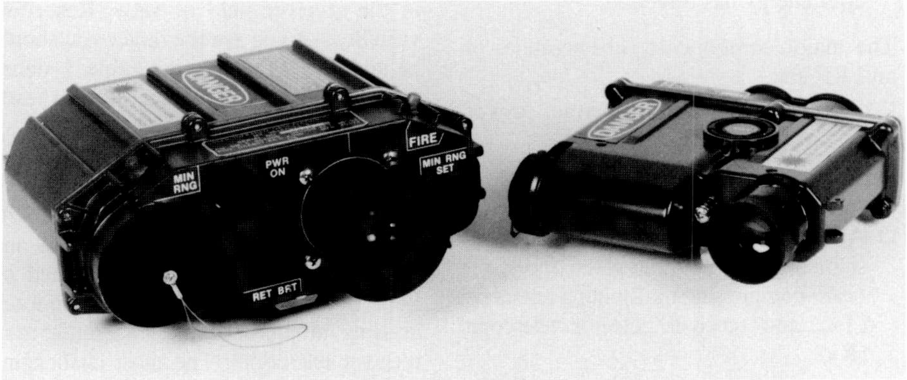

Fig. 1. *AN/GVS-5 laser rangefinder (left), and the new Mini Laser Rangefinder.*

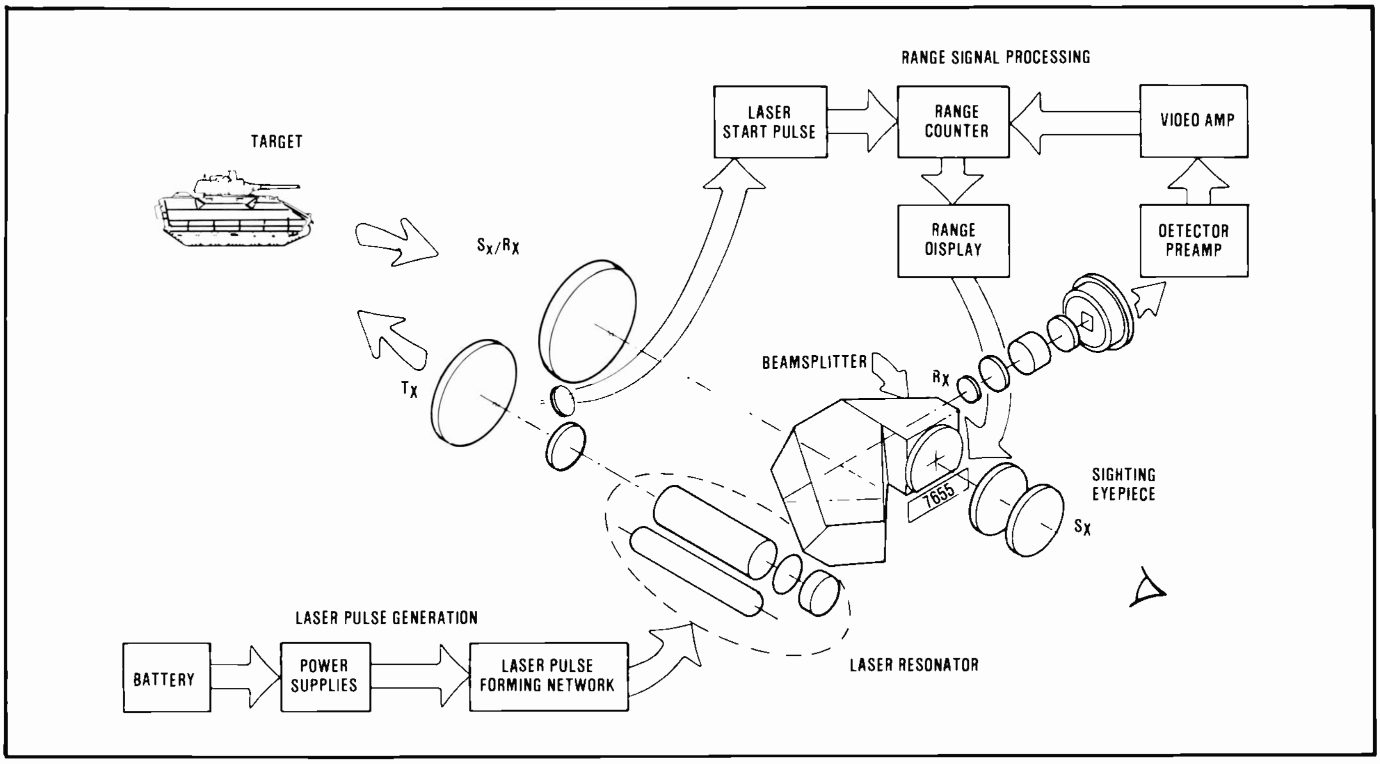

Fig. 2. *Electro-optical schematic operation.*

some of them, it is first necessary to understand the constituent parts and operations of a laser rangefinder.

A laser rangefinder (LRF) determines distance to targets in the following way (see Fig. 2):

- ☐ A short burst (4 nanoseconds long) of well-defined, spectrally-pure light is directed at and transmitted to a visible target.
- ☐ The LRF collects laser energy reflected from the target with receiving optics.
- ☐ Detectors within the LRF register the times of laser pulse transmission and reflected energy return.
- ☐ The elapsed time recorded by a high speed clock and the known speed of light defines the round-trip distance the laser energy has traversed.

The major components/subassemblies of an LRF are:

- ☐ Power supplies and prime power sources.
- ☐ A laser transmitter module and transmitter circuits.
- ☐ An optical assembly consisting of boresighted visual sighting telescope (Sx), beam-compressing transmitting telescope (Tx), and receiver/detector telescope (Rx).
- ☐ Laser detectors, pre-amplifiers, and amplifiers.
- ☐ Range counter and display circuits.

The conditions of design of the AN/GVS-5 were that (1) all components and subassemblies be manufactured for complete interchangeability of parts without readjustment or realignment, (2) the LRF must operate over adverse worldwide and battlefield environments, (3) it must be repairable by U.S. Army depots, and (4) its unit cost, determined by competitive procurements, must be affordable to the Army.

The AN/GVS-5 and MLRF are primarily designed for the artillery forward observer. As such, the maximum required range is 10 kilometers and range accuracy is ±5 meters. It should discriminate targets 0.75 milliradians or larger and indicate range returns from more than one target in the receiver field of view. Regarding visibility—if you see the target you should be able to range to it. Table I delineates the salient specifications of each rangefinder.

The AN/GVS-5 was configured around a 7X, 50mm diameter clear aperture roof prism binocular design manufactured by Ernst Leitz Canada. These compact and stable sighting optics were integrated by Leitz into a single thin-walled aluminum housing along with the transmitter and receiver telescopes. The most challenging opto-mechanical problems of the AN/GVS-5 (and the MLRF) were in maintaining the closely aligned boresight among these three telescopes over a −46°C to +71°C temperature environment with rugged shock and vibration requirements. All critical optical elements were resiliently mounted to prevent drift over these extremes. Also, telescope and laser transmitter modules were cantilever-mounted with mounting surfaces normal to the optical axes, and structures were designed for symmetry around these axes. This provided uniform symmetical expansion/contraction along optical axes over temperature variation with little radial temperature gradient—a prime cause of mechanical deflection of optical axes (boresight misalignment).

The major structural parts—the telescope housing, a control panel, and the cover—were all machined thin-wall aluminum investment castings (the telescope housing was later converted to aluminum high-pressure die casting). The material choice was made for reasons of stability, strength, EMI suppression and manufacturability.

The major electronics—detector/preamplifier, video amplifier, and range counter circuits—were hybridized, and the power supplies and laser triggering circuits with attendant high voltage and magnetics were left as conventional printed circuit boards.

The MLRF was designed to do the

same job but to be lighter, smaller, and potentially less expensive. Its weight is about the same as a pair of Army binoculars (2.5 pounds) and is small enough to fit in a field jacket pocket. It can be tripod-mounted, and through an interface connector, powered, fired, and read-out remotely. And, it can be held and fired with either hand (see Fig. 3.).

The AN/GVS-5 had been conservatively designed so that its range capability far exceeded specified requirements. Detector sensitivity improvements added to that margin. A smaller MLRF, closer to design requirements, was attained through a trade-off study involving reduced laser output energy (smaller laser, power supplies, and batteries), and a smaller sighting/receiving telescope. Although the design of the MLRF was multi-disciplined, its form began with the optical designer.

Table I. *Rangefinder characteristics*

	Mini LRF	AN/GVS-5
Range performance	50 to 9995m	200 to 9995m
Range accuracy/resolution	5m	10m
Probability of ranging	99% to 5 km target with 7 km visibility	99% to 5 km target with 5 km visibility
Energy output	5-7 mJ	10 mJ
Sighting optics:		
Magnification	6 ×	7×
Field of view	7°	
Eye relief	20.8 mm	17mm
Light transmission	60%	50%
Display (in eye piece)	Range Battery low indicator Multiple target indicator	Range Battery low indicator Multiple target indicator
Size	1.65 × 6.0 × 6.2 in.	3.5 × 6.0 × 8.0 in.
Volume	55 cu. in.	135 cu. in.
Weight	2.5 lb.	5 lb.
Temperature operation	−46°C to + 71°C	−46°C to + 71°C

Optical design

The MLRF sighting/receiving telescope was the starting place for the redesign. The erecting prisms of the AN/GVS-5 were difficult to make and mount and were, therefore, expensive. After considering many designs and their effect upon the MLRF layout and shape, a simple roof penta-prism/right-angle prism erector was devised. The innovative telescope design, it was later learned, actually dates back to 1897 and the German optical designer Moritz Hensoldt.

The sighting and receiving telescopes share the same objective lens; a beam splitter between the lens and its focus separates the laser return from the sighted image for efficiency of design and optical stability.

The objective clear aperture was reduced from 50mm diameter to 30mm. While this reduced the collected energy to 36 percent of the AN/GVS-5, improved optical transmission of the new optics modified the energy to detector to 45 percent. Laser output was reduced from 10 millijoules to 6 millijoules, so comparable performance of the MLRF is about 27 percent that of the AN/GVS-5.

Fig. 3. *Mini Laser Rangefinder.*

Mechanical design

While reductions derived from laser electronics and optics reductions were important, other design changes were also significant. The control panel and optical housing were combined, eliminating redundant structures and their costly machined interfaces. The erector prisms were left exposed within the MLRF, requiring more attention to cleanliness in assembly but providing additional reduction in structure. Two sets of built-in alignment mechanisms were eliminated—one because the need to optically align to a mounting surface was eliminated, the other because the alignment is accomplished with external tooling. The regrouping of optics and control panel allowed the elimination of a protective window and its structure. Simple elimination of parts is the most effective means of weight and cost reduction. (see Fig. 4.)

Again, stability is the essence of a good rangefinder. And in the optical industry, low expansion metals, like Invar (density: 0.3 pounds per cubic inch), are considered the ideal mounting structure. Invar, though, is expensive, heavy, and difficult to machine. Instead of trying to contain absolute movement of optics with Invar, our approach is to restrict relative movement. For optical stability in conditions with thermal variation, the important figure of merit is defined by the ratio of thermal expansion to thermal conductivity. All de-

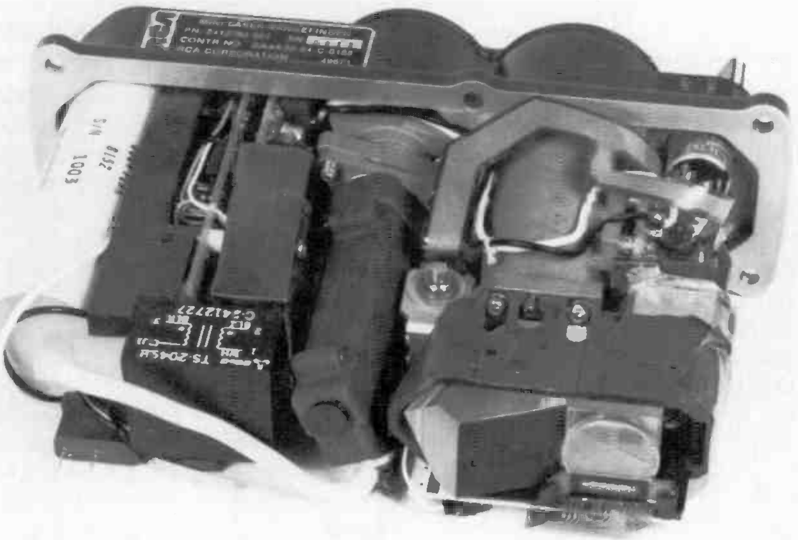

Fig. 4. *Mini Laser Rangefinder with cover removed. Laser pulse generation electronics (left), laser cavity (center), optics and signal processing electronics (right).*

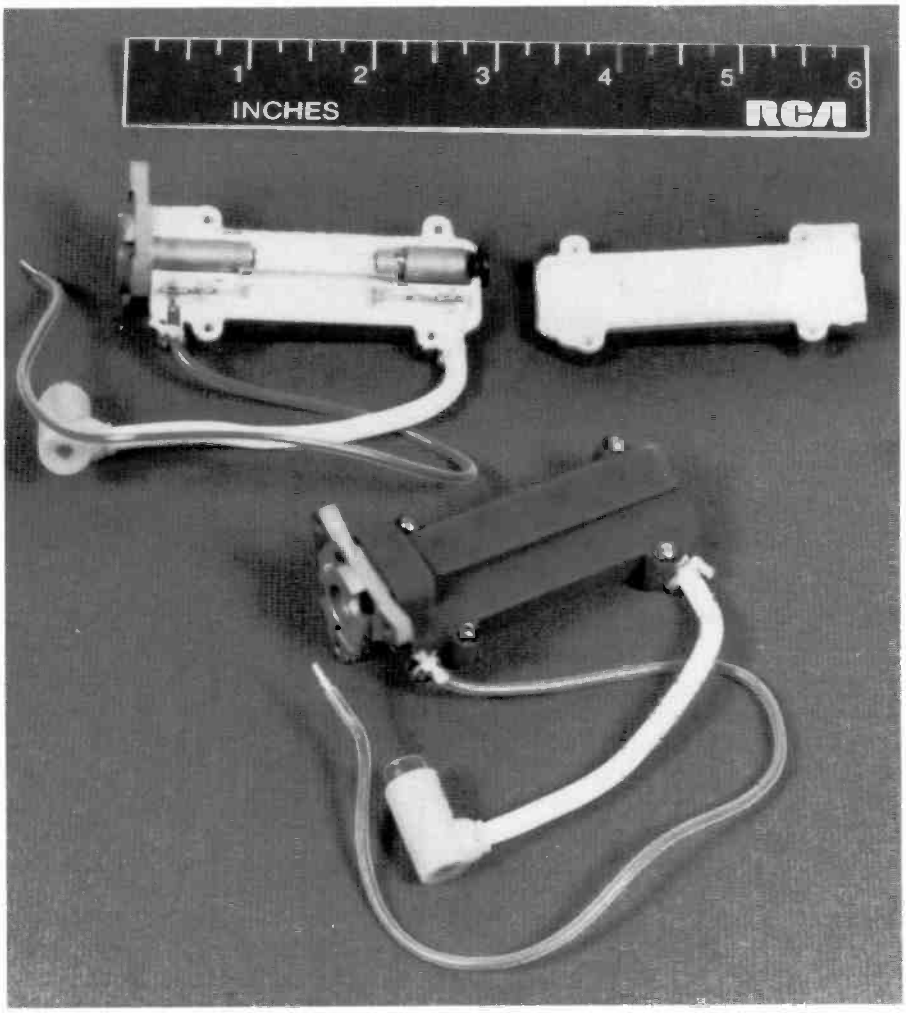

Fig. 5. *Laser transmitter module assembly. Opened, above, and closed below.*

sign factors being equal, the figure of merit for aluminum is slightly better than that for Invar. The movement is greater, but uniform growth is better controlled. Happily, aluminum is lighter than Invar, so most lightweight rangefinders are made with aluminum structures. The GVS-5 has machined, cast-aluminum optical housing panel and cover. The MLRF combines the optical housing and control panel into a single structural element, a machined aluminum investment casting.

Some new plastics promise similar stability (and strength) with lower fabrication costs. But are they really stable enough? Boresight retention involves controlling micro-inches of movement.

The MLRF is the first production laser to use plastic in a key structural element (the cover), and it does provide adequate stability. The three optical axes maintain their tight 0.1 milliradian alignments throughout the thermal, pressure, and shock/vibration environmental ranges. Ultimately, other optical structures and mounts may be made from injection molded plastics to further reduce fabrication costs. The plastic used here is polyphenylene sulfide (PPS) with nickel-plated carbon fiber additive for strength and electrical conductivity (the enclosure must be EMI tight for covert field operation).

We perform only one secondary operation on the cover after molding—(threads, inserts and O-ring grooves are molded in); we lap the cover/front optical housing interface flat to within 0.001 inch to help maintain boresight alignment.

Since the MLRF will be used by Special Forces and underwater Seal teams, we designed the cover to withstand depths of at least fifty feet (about 25 psi) without damage or leakage. The old aluminum AN/GVS-5 cover, 0.035/0.040-inch thick, cannot withstand this pressure without deformation.

The increased stiffness comes from the fact that the MLRF is smaller, has thicker walls, and has extensive internal ribbing to increase cover section modulus. At the same time, the cover design eliminated external protrusions from switches and mounting bosses keeping it from catching on clothing or brush and minimizing crevasses that would capture mud and chemical/biological warfare agents. The entire rangefinder must withstand harsh cleaning solvents to remove these latter agents. Light ridges, shallow grooves, and a textured surface provide a secure grip even when the MLRF is wet.

Prototypes and models help configure a

compact, functional design. Two early prototypes were built in 1982 to prove the MLRF concept. The placement of electrically noisy power supplies and laser pulse-forming circuits so close to sensitive hybrid detector circuits always requires shielding and filtering changes by the electronic design engineer. The hard-wired prototypes thus influenced the design of the production model. Mockups and wooden models of the production design were then invaluable in establishing component placement, accessibility, and the design of flexible, printed-circuit wire harnesses. The models, furthermore, took the risk out of mechanical interferences and allowed more compact packaging. A device is usually considered compactly packaged if, when placed in water, it sinks. At 55 cubic inches and 2.5 pounds, the MLRF has a specific gravity of 1.26 and does, indeed, sink.

Laser design

An interchangeable laser transmitter module was one of the most successful features of the AN/GVS-5. The module consists of a cantilever-mounted laser resonator, a flash lamp, and an injection molded plastic laser cavity (see Fig. 5). The cantilevered resonator (laser rod, film dye Q switch, end mirror, and tiny alignment mechanisms) maintains its stability under all environmental and operational conditions. Since the design was so successful and all the manufacturing and alignment tooling was amortized, the MLRF was designed similarly. The resonator length, lamp, and cavity are shorter since we require less power output energy, but the same tooling is used for both lasers.

The laser engineer scaled down the power input and storage capacitor requirements, providing great volumetric and weight savings. He simplified the laser triggering circuits for further reductions. The net effect was a highly producible, easily aligned and maintained laser transmitter.

Bob Guyer, a Senior Engineering Scientist in the Product Engineering Group at Automated Systems Division, joined RCA in 1959. He was Lead Mechanical Engineer on the Apollo Program Laser Altimeter for NASA and the Laser Range Pole Surveying System for the U.S. Forest Service. He was Principal Mechanical/Optical Engineer on the mini-laser rangefinder, the AN/GVS-5, and many other military laser rangefinders and laser target designators.

Mr. Guyer received his BSME from North Dakota State University in 1956 and did graduate work at MIT. He is a member of Pi Tau Sigma, Tau Beta Pi, and Sigma Xi Honorary Engineering Societies, and is a Registered Professional Engineer in the Commonwealth of Massachusetts.

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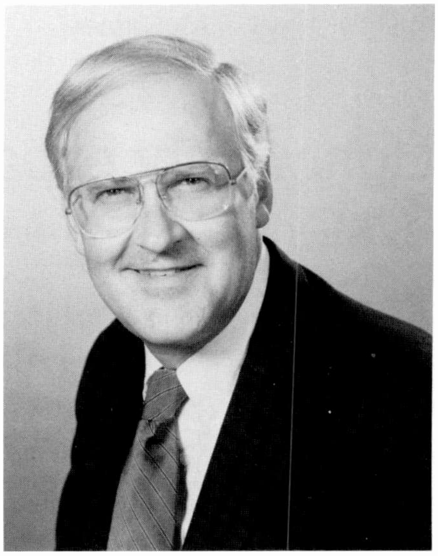

Electronic design

Throughout the production of the AN/GVS-5 the electronic engineers had been working to improve the receiver electronics. Updated hybrid design technology, new custom LSI devices, and improved detectors were first tried in "special" AN/GVS-5 rangefinders. When proven, the MLRF incorporated these components. The old range display would not fit in the MLRF, so a new custom hybrid LED range display was created.

The electronic operation was simplified, with positive effects on the power supply size. Fewer voltages were needed, less power was required, and its size shrunk drastically.

The hardest task for the electronics engineer is to take the newly created optical/mechanical/laser/electronic hardware and make it all play together. Changes were made; better shielding and isolation, improved grounding, improved performance at low and high temperature. When the first system is made to work, do the next three "just like it" also work? Obviously, they must, and they did. Complete module interchangeability without adjustment in a production system requires foolproof design for all conditions.

Acknowledgments

If each of the engineering disciplines cooperated and compromised to achieve a successful product, it is to the credit of the program management. The atmosphere was there for team design, communication with the user, and constant awareness of cost impact on the final product.

The MLRF should have the production potential of its predecessor, thanks to the program management skills of Glenn Anderson, the electronic ingenuity of Norm Roberts and John Griesing, the laser knowledge of Walter Radcliffe, and the mechanical creativity of Bert Fitch, all of RCA. We are also grateful for the exceptional theoretical and practical optical design of Conrad Stenton and Walter Mandler of Ernst Leitz Canada.

K. Ghostlaw | J. McNamee

Lightweight expansible semi-trailer van for military systems

On the outside it looks just like an XM995 Army van. On the inside it has 65 percent more space when deployed.

Abstract: *The Army required a highly mobile enclosure to house the electronic equipment test facility that supports the AH-64A Apache Helicopter. Automated Systems Division, under subcontract from McDonnell Douglas Helicopter, examined possible methods for housing and transporting the integrated automated test system, but found existing military enclosures such as S-280s, ISO shelters, and military semi-trailer vans too small and confining to mount this large electronic test system. The commercial trucking industry offered one interesting alternative: a television van with sides that expand open at site to provide a 65 percent increase in interior space. Since existing expansible vans were oversized, heavy vehicles suitable only for highway use, ASD implemented a development program to design, build, and test a ruggedized expansible semi-trailer that would meet demanding Army specifications or size, weight, payload, environmental control, climatic extremes, and transport by land, sea, and air. This new expansible van, with the AH-64A test system mounted, has been successfully qualification tested for mobility and climatic extremes, adding to the military inventory of mobile enclosures a new option with significantly improved available interior space compared to standard military semi-trailers.*

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The AH-64A automatic test equipment (ATE) consists of three major subsystems operating together as an analog, digital, and electro-optical test system. Figure 1 shows the nine racks of electronics, a three-module electro-optics test bench, interface accessories, and interconnect cabling. The system weighs 14,450 lbs and has an equipment volume of 870 ft^3. Several methods of housing this system for transport and operation in a military environment were examined: splitting the system into several S-280 shelters mounted on 5-ton trucks, using two 2:1 ISO shelters on flatbed trailers, and sharing side-by-side semi-trailer vans. These enclosure configurations were awkward for operator interface and cumbersome for cable hookup. The concept that best met Army criteria for setup and teardown time, weight and volume for air transport, and compatibility with military deployment was mounting the ATE system in a single semi-trailer van. Although standard Army trailers such as the XM991 could accommodate the payload weight, there was insufficient floor space to arrange the system for testing and maintenance actions. An innovative method to provide this needed workspace was to add expanding sides to the fixed van body (see Fig. 2). Expansible vans have been used for years by the television industry to provide spacious mobile facilities for TV crews at sporting events and remote news areas. Evaluation of various television vans indicated that applying the principles of commercial expansible van technology to solve a demanding military enclosure design would require creative new approaches to structures, weight savings, environmental control, and human factors.

Requirements

Development of an enclosure for use by the military is controlled and governed by several interrelated specifications for transportation, deployment, configuration, and environment. Specific system requirements were established defining major van design objectives as follows:

- ☐ House and protect non-militarized AH-64A ATE equipment during all phases of operation, transport, and storage.
- ☐ Interface with, and be fully compatible with M818 and/or M931 tractors.
- ☐ Meet MIL-M-8090 Type III mobility for semi-trailer vans.
- ☐ Meet MIL-A-8421 for air transport in C141 and C5A aircraft.
- ☐ Meet AR70-47 for transport by ship below deck.
- ☐ Protect ATE during exposure to environmental limits of AR70-38.
- ☐ Meet size, shape, and parts requirements that are compatible with standard military semi-trailer vehicles.

These design objectives dictate a set of specific van requirements, including:

- ☐ Size—the expansible van must be 35 ft long to match the appearance of stan-

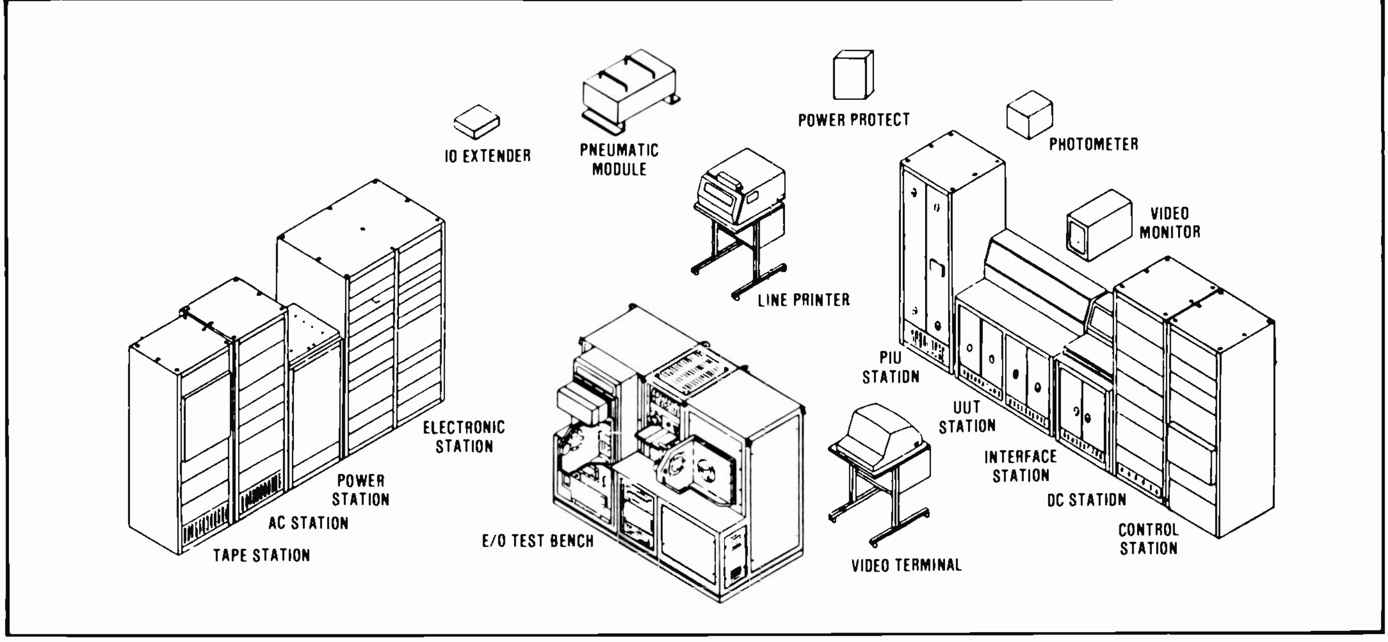

Fig. 1. *AH-64A automatic test equipment.*

dard military vans. Van width (closed) of 8 ft nominal is defined by transport requirements of MIL-M-8090. Overall van height from the road to rooftop of 150 in. assures bridge clearances in European and Asian countries. The van body, less all undercarriage components, must not exceed 102 in. in height for fit in a C141 aircraft.

- ☐ Weight—weight of the fully assembled expansible test van must be compatible with towing, lifting, and transport specifications. The limiting factor becomes the lift capacity of standard Air Force "K" loaders used to lift and install heavy cargo into C141 aircraft. When the van is prepared for C141 transport, all undercarriage components are disassembled to provide a flat undersurface to mate with aircraft pallets. In this configuration, the van weight including pallets and tiedowns cannot exceed 40,000 lbs. An additional weight limitation is the towing capacity of truck tractors, and for the AH-64A application, a fully assembled van ready for towing must not exceed 43,000 lbs.

- ☐ Transport loads—the fully assembled expansible van must withstand transport loads defined in Table I. Also, the van must provide protection for the nonruggedized ATE so that electronic equipment loads do not exceed 1.5*g*.

- ☐ Thermal performance—the expansible van must function, be transported, and stored in temperatures between +125° F with full solar load and −65° F. Also, it must provide thermal protection for the ATE to enable operation at these external extremes while maintaining an internal van temperature +65° F to +75° F.

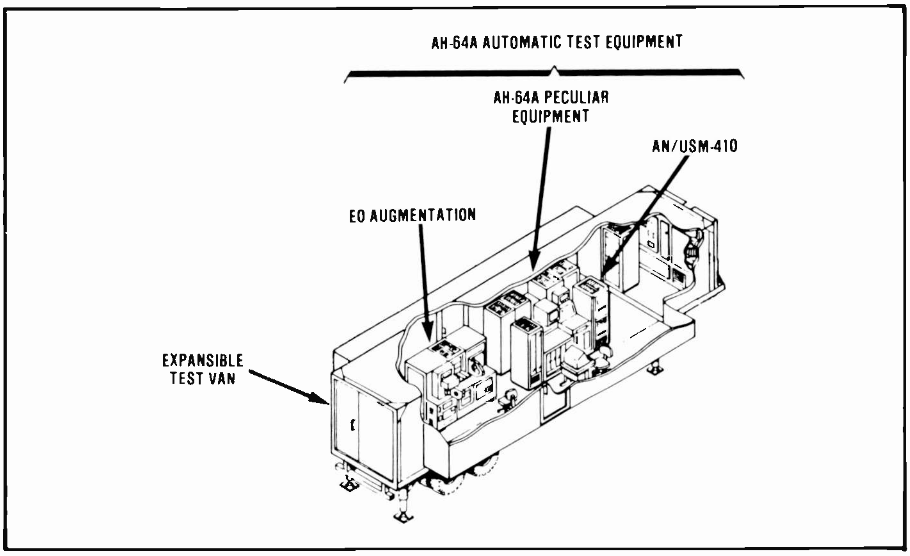

Fig. 2. *AH-64A test van.*

RCA/ASD and the Gerstenslager Co., Wooster, Ohio, under subcontract from the McDonnell Douglas Helicopter Company, initiated an effort to design and build an expansible van based on commercial van concepts but with new design approaches tailored to meet demanding military requirements.

Weight budget

Controlling the weight of the expansible van is very important because the van must be transported and handled without exceeding capacities of standard military equipment. The weight estimate, based on commercial expansible van data and the fixed payload, was approximately 60,000 lbs. It was apparent that available commercial designs for body structure, roof, folding roofs, folding floors, and fixed floors would not solve the expansible van weight requirement. Budgets were assigned to each van component in a scaled-down manner, and the final van weight is in accordance with values shown in Table II.

Table I. *Transport shock loading*

Transport mode	Static equivalent loads		Vibration
Over the road	Fore	= 0.4g	0.5 g-p
	Aft	= 0.4g	5-200 Hz
	Lateral	= 0.6g	Vert/lat/long axes
	Up	= 0.6g	
	Down	= 3.0g	
*Air**	Fore	= 3.0g	0.5 g-p
	Aft	= 1.5g	5-200 Hz
	Lateral	= 1.5g	Vert/lat/long axes
	Up	= 2.0g	
	Down	= 4.5g	
Marine	Pitch	= 0.4g	
	Roll	= 0.6g	Not applicable
	Down	= 1.8g	
	Pitch & Heave	= 0.8g	

**g* levels apply to static and dynamic loads.

Table II.

Van component	Total weight (lbs.)
Van body, structural	9.270
Fixed roof panel	735
Expansible side (roadside)	1,780
Expansible side (curbside)	1,780
Folding roof assembly (2)	490
Interior air ducts	550
Expansible side drive (2)	525
Electrical assemblies	840
Interior walls/accessories	920
Main floor	885
Undercarriage/jacks	6,100
Doors	535
Miscellaneous hardware	2,775
Total expansible van	27,185
Payload ATE system	14,465
Total test van	41,650

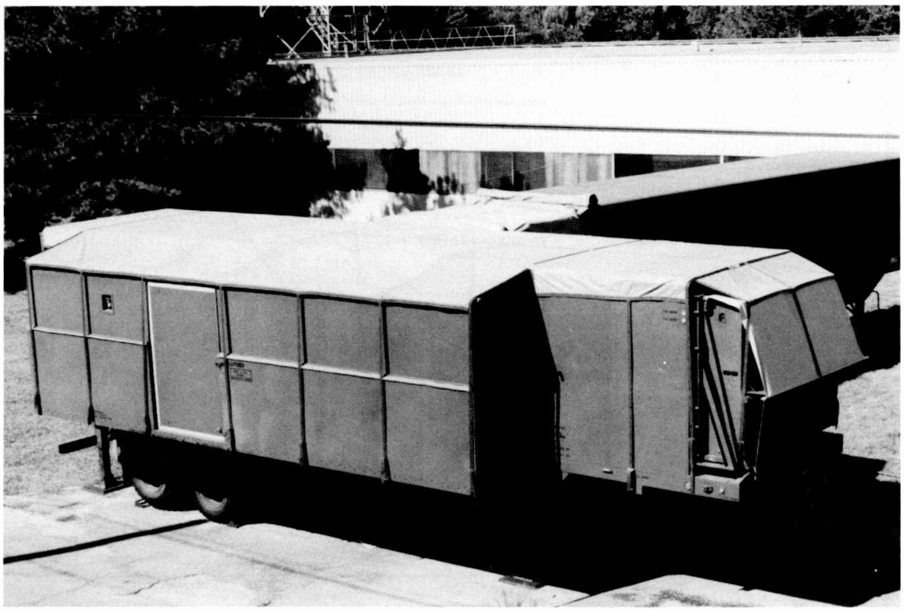

Fig. 3. *AH-64A test van—open, with expanding sides.*

Expansible van configuration

Figure 3 shows the production AH-64A expansible van in its expanded, operational configuration. The van body is 35 ft long with a 2-ft overhang to mount air conditioners. Roadside and curbside expansible sides are 25 ft long, opened simultaneously by electric motor or manual drive to a total span of 14 ft, 6 in. Suspension is provided by a standard military airride fifth wheel and an airride undercarriage dolly consisting of military tires, wheels, and airride components in conjunction with rugged commercial dual axles. Personnel doors are located at mid-span of the roadside expanding side and at the rear. Equipment access doors are included at the rear and at mid-span of the curbside expanding side. Figure 4 shows the van in its closed, transport configuration. When the van is closed, it has the size, shape, and appearance of a standard Army XM991 semitrailer van.

Van structural design

The expansible van must be capable of withstanding any combination of loads resulting from road, sea, and air transport and snow and wind. Table I indicates worst case loading of the van during the transport modes. The van has been designed to withstand a 2g vertical load with a factor of safety of 1.5. It would take a 3g equivalent static load to initiate a structural failure. The relatively severe air transport load, which would only be experienced in emergency landings, is resisted by using external bracing in the C5-A aircraft and load spreading pallets in the C141 aircraft. Road transport loads are transmitted from the wheels through the undercarriage and fifth wheel to the chassis, with load isolation provided by the airride suspension system.

ASD, in conjunction with E.A. Botti Associates, Waltham, Mass., developed a structural computer model for the lightweight AH-64 van. The concept utilizes a three-dimensional rigid frame system consisting of longitudinal tubular members connected by a series of vertical tubular struts located along the longitudinal centerline of the van, as shown in Fig. 5. Structural steel is welded into a frame connecting all primary structural elements. The tubular steel skeleton is covered with a lightweight 0.050-in. thick aluminum skin. Vertical columns located at the front, back, and stepped area (gooseneck) integrally attach the roof structure to the chassis. The tubular members are connected by using a welded slip-fit insert that interfaces the end of one tube to the surface or wall of another tube. The van stresses are obtained by imposing the various load conditions and appropriate boundary supports to the finite-element model using computer analysis. A major factor in the successful

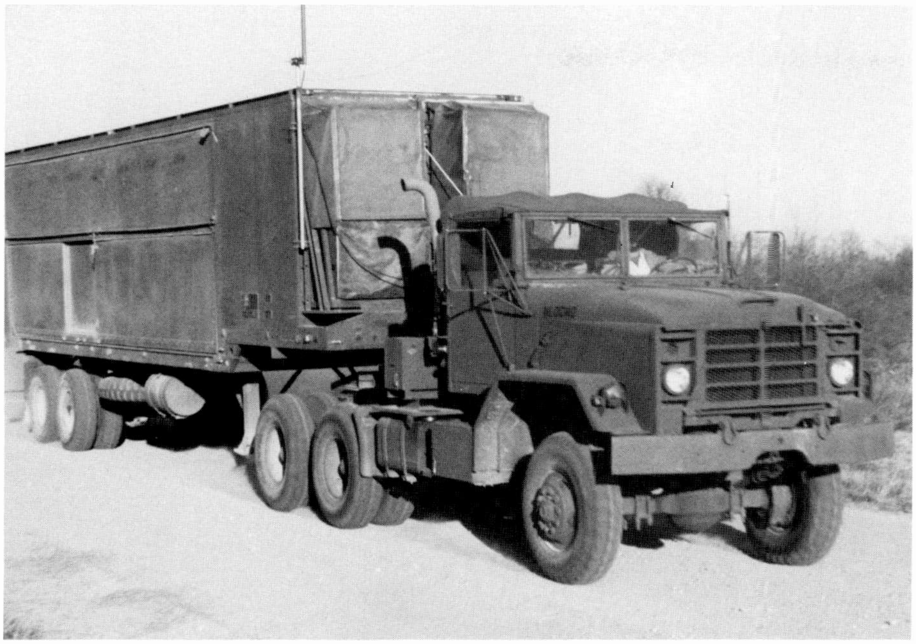

Fig. 4. *AH-64A test van—closed.*

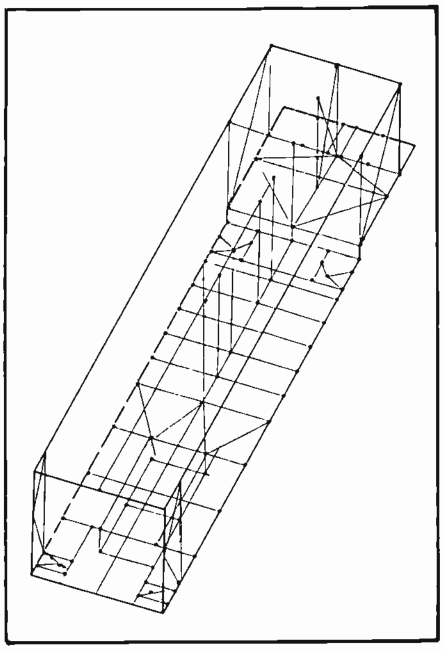

Fig. 5. *Structural model.*

design of the lightweight structure was the ability to repeat the model until each structural member had a minimum allowable cross section to match a maximum allowable stress.

The backbone of the lightweight AH-64 expansible van is the chassis structure shown in Fig. 6. The chassis frame consists of three square tubular members that extend the length of the van and form the primary structural load path. Lateral joists are spaced approximately two ft on center to support the main floor and provide lateral support for the primary longitudinal tubular members. The joists are 6 in. lightweight M beams. Lateral bracing is provided at key locations to stabilize the chassis frame. The primary lateral bracing is located forward of the 5500-lb electro-optical bench (part of ATE payload) where the maximum chassis bending moments occur.

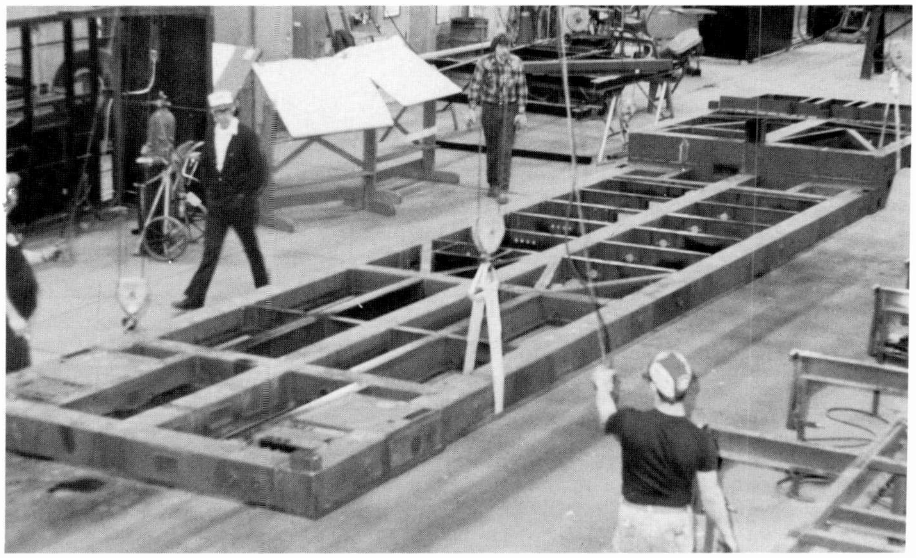

Fig. 6. *Chassis structure.*

The "gooseneck" is designed to behave as a "deep" box beam with a continuous horizontal diaphragm located at mid height to prevent local buckling of the plate elements that form the section. The vertical columns at the gooseneck extend continuously from the roof to the lower level of the chassis. Bracing is also used at the front and back exterior vertical walls to transfer fore/aft loading from the roof to the chassis.

Lightweight, rigid, honeycomb core aluminum skin sandwich panels are used for the van fixed roof, folding roofs, folding floors and fixed floors. The fixed roof is riveted to the van structure and is capable of supporting a 25 lb/ft^2 snow load and the concentrated load of two service personnel. Folding roofs will withstand snow loads and wind loads to 80 mph. The honeycomb floors will support rolling equipment loads of 1200 lbs and concentrated loads of 250 lbs/ft^2. The fixed floors have built-in hat sections that mount electronic equipment and provide reinforced attachment directly to chassis structure. The use of honeycomb panels provided a weight savings of 2:1 compared to existing commercial designs for similar items.

The structural configuration of the expanding sides consists of lightweight steel hat sections spaced approximately two ft on center covered by a 0.050-in. thick aluminum skin. The structural stability (buckling) of the skin is significantly enhanced by a 2-in. layer of polyurethane foam that is sprayed on and bonds to the walls.

Thermal

The system specification requires the assembled expansible van to operate in categories 1-6 of AR70-38, which includes desert, tropical jungle, and arctic climates. External temperatures can range from −65°F to +125°F and the relative humidity (RH) can vary from near zero to saturation.

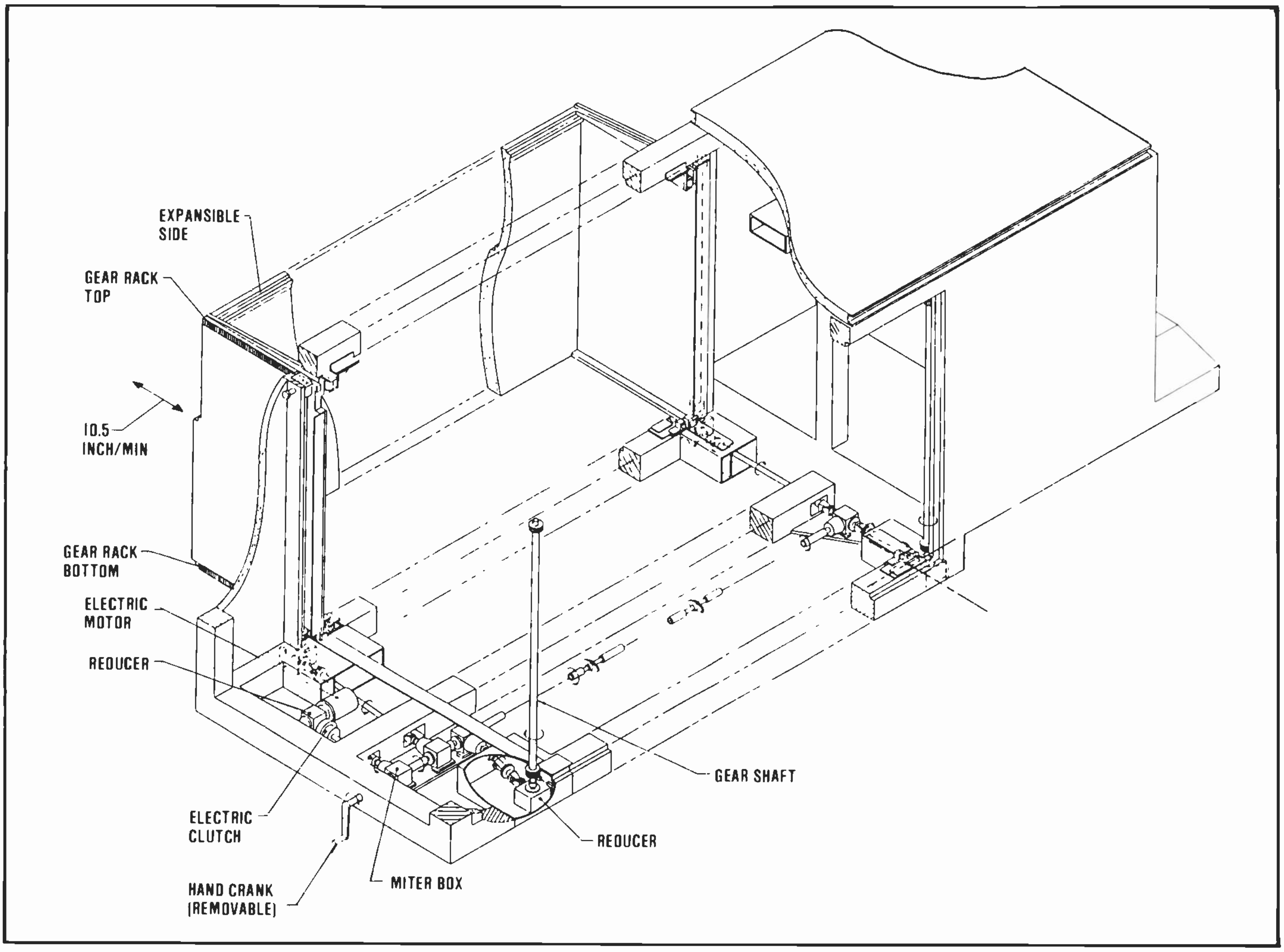

Fig. 7. *Expansible side drive mechanism.*

The van's environmental control system (ECS) must maintain interior conditions at 70°F + 5°F and 20 to 70 percent RH (non-condensing) for proper operation of system electronics.

By evaluating the impact of the structural design on the thermal characteristics of the van, it was determined that a "U" factor of 0.34 BTU/ft^2/hr was achievable. This value was used in all thermal analysis, and careful attention was paid throughout the design effort to maintain and/or improve upon this value. The most severe thermal load occurs when the van is operated under desert conditions (125°F, with 360 BTU/hr/ft^2 solar radiation), requiring the ECS to reject more than 98,000 BTU/hr. A roof tarpaulin reduces the thermal load to 87,000 BTU/hr, and two MIL-A-52767 60,000 BTU/hr, 208V/3-phase environmental control units (ECUs), each capable of 45,100 BTU/hr* cooling at 125°F, meet the worst case load with a calculated 3000 BTU/hr margin.

At −25°F, the van requires more than 0.2 gal/hr of water to maintain 20 percent RH. This is provided by a steam humidifier supplied by a modular water tank system. In cool/wet environments, dehumidification is provided by using ECS heater coils to reheat conditioned air.

In addition to the manual GFE ECS controllers, the ECS has an independent automatic control system with its own thermostat circuit, two humidistats, (one for humidification, one for dehumidification), a sequencer, and several interlocks. Two of the interlocks located in the air ducts are airflow switches that prevent cooling or heating where there is no air flow. Others prevent humidification below 40°F and ECU compressor operation below +10°F. Both the automatic and manual controls are merged in a single panel and linked to the ECUs through a multipole wafer switch. This is another unique and useful feature of the expansible van design.

The air distribution system is designed to establish an air flow pattern that would prevent hot spots from developing over the equipment and in the corners of the 25-ft expanding sides. The ECUs force conditioned air into two parallel ceiling ducts that run the full length of the van. At points along this duct, air exits through diffusers that direct the flow outward along the ceiling to the outer walls of the van, where the air circulates to floor level and enters the equipment. Rack blowers force heated air through the top of the equipment where it is carried forward to the return air registers located in the front of the van. Fifteen percent of the flow is carried through the 7×14-in. hollow structural tube that runs down the center of the van between the two supply ducts. The remainder travels in the space between the equipment and the ceiling, which is a virtual return air duct. To maintain the van's "U" factor, the tubular structure and

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* Units are specified 60,000 BTU/hr at 120°F ambient, 80°F return air, free flow. Units were tested at 125°F, 75°F return air and reduced flow to simulate ducting losses.

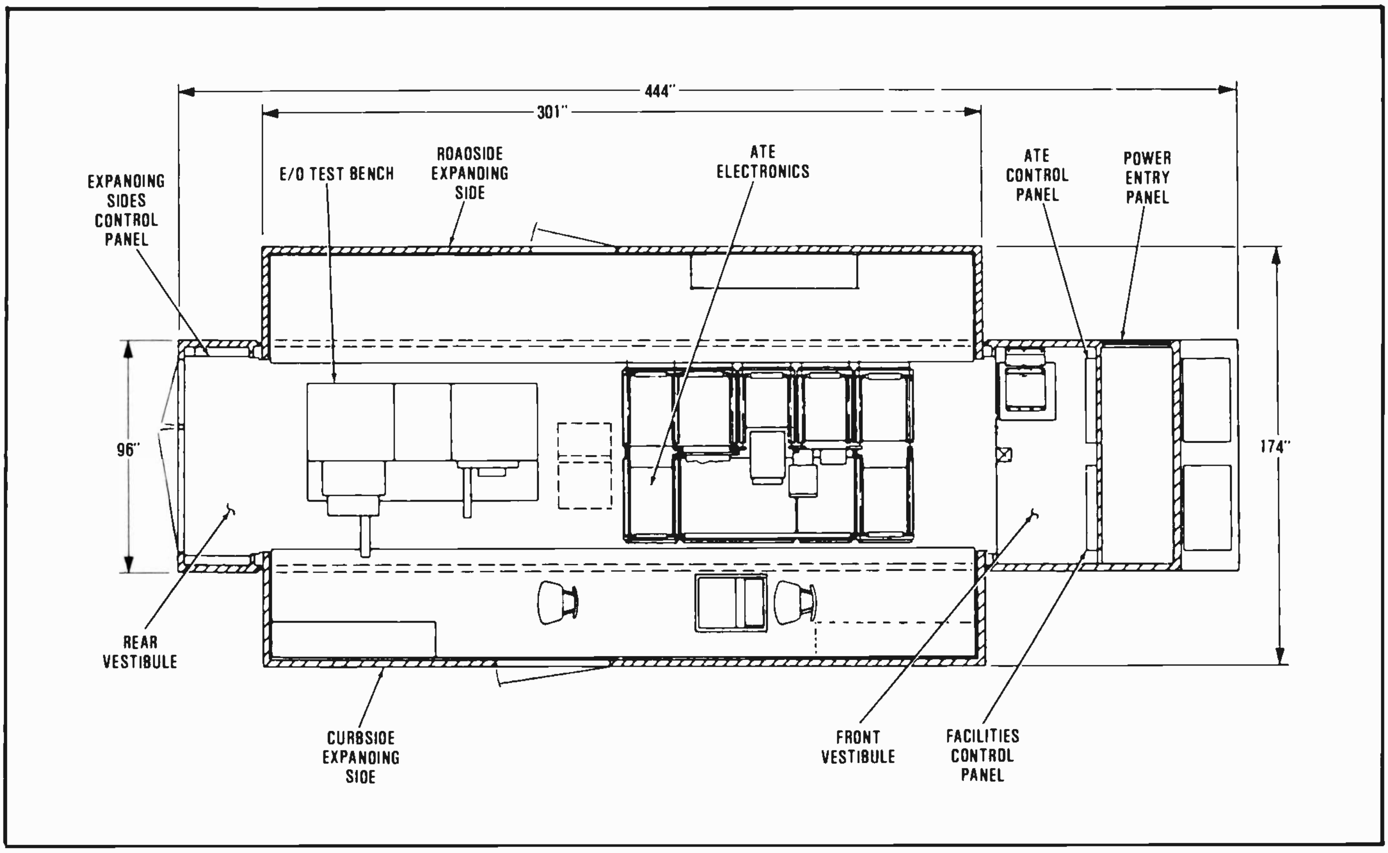

Fig. 8. *Expansible van plan view.*

van walls are filled with 2-lb density closed-cell foam, and thermal breaks are incorporated where practical. Precision thermostat sensors, controllers, and alarms assure that air inlet temperature at equipment rack blowers is maintained at 70°F ±5°F. In use, the ECS has performed well, maintaining temperatures within ±2°F of set point in steady state conditions and lowered the van interior temperature from +125°F to 70°F in 45 minutes.

Expanding side drive

Each expanding side is 25 ft long, 7 ft tall, and 3.3 ft wide. The assembly consists of a three-sided wall, a folding floor hinged along the lower edge of the 25-ft section, and a folding roof hinged to the header beam of the van. Each expanding side with its attached floor weighs 1780 lbs, and each folding roof weighs 245 lbs. The sides can be expanded and/or retracted by electric power or by hand crank. The electric portion of the drive consists of a 1.5 HP three-phase motor with an attached 30:1 reducer that connects to the main drive train through an electric clutch.

The main drive system shown in Fig. 7 consists of four 1:1 miter boxes, four 60:1 reducers, and eight rack and pinion sets. The electric drive and hand crank meet at the first miter box, which drives the three remaining miter boxes, which in turn drive four 60:1 reducers, one on each end of each expanding side. Each 60:1 reducer drives a vertical shaft with two 3.5-in. pitch diameter pinion gears that are meshed with racks affixed to the short wall sections of the expanding sides, one at the top and one at the bottom. The final drive shaft acts as a torsion spring, reacting to the moment induced by the cantilevered weight of the expanding side. The shafts are also restrained by bearings attached top and bottom to the fixed structure of the van. The drive mechanism was modeled on a computer using a BASIC program that computed torque values at each inch increment of extension to assure that the peak torque and average running torque capacities of the drive train components would not be exceeded. The torque values measured on production units correlated closely with the values predicted by this model. All components of the drive train are packed with low temperature grease (MIL-G-23827A) to allow for operation in cold environments. The design goal for the expanding side drive was to extend or retract the sides in five minutes or less at temperatures as low as −25°F. In tests, the expanding sides opened and closed in less than 4 minutes at −65°F.

Human factors

The expansible van is designed for the convenience and comfort of the operators. Lighting intensity, noise level, aisle space, interlocks, and alarms are based on MIL-STD-1472 human factors specifications. Opening and closing the van is a one-person operation under pushbutton control using electrical power. Clearly labeled electrical panels provide operator interface for power, facilities, and ATE control. Proper placement of control and interface panels was critical because there is no access to 85 percent of the van interior when the sides are in the closed position.

Electrical panels

Figure 8 shows the locations of the van's four electrical panels. To minimize wiring, three panels are located in the front end of the van: the Power Entry on the exterior roadside, the ATE Control on the roadside interior, and the Facilities Control on the curbside interior. The fourth panel, the Expanding Sides Control, is located in the aft vestibule interior roadside. Prime power is connected to the van at the Power

Entry Panel, which contains phase sequence indicators and primary breakers. The panel contains secondary breakers with power-on indicators for the following circuits:

- ☐ ECS Main Control
- ☐ Tech Main Control
- ☐ ECS Monitor
- ☐ Tech Monitor
- ☐ Expanding Sides and Winches
- ☐ Exterior Lights
- ☐ Auxiliary Lights
- ☐ Exterior Lights
- ☐ Exterior 20 VAC Outlet
- ☐ Support Van Power

The operator uses these controls to assure that primary power is connected properly and to turn on auxiliary power, exterior and interior lights, and the Expanding Sides Control Panel before entering the van.

The Expanding Sides Control Panel contains switches that operate the motor that expands the sides and the winches that lower the floors. It also has switches that control the front and rear exterior lights, the interior lights, and a row of indicator lights that show critical conditions associated with the extension or retraction of the expanding sides. Located in close proximity are two large cards that give step-by-step instructions for opening and closing the expanding sides.

The Expanding Side Drive is designed for simplicity of operation. The operator sets the select switch to the "out" position and holds a spring-loaded toggle switch on until either one of the "expanding side out" lights is illuminated. With a turn of the hand crank, the sides are adjusted so that both lights are on. At that point the motor drive is automatically de-energized, and the winches lower the folding floors. Once the floors are lowered, the winch cables are disconnected and the floors are latched in place. Finally, the roof latches are engaged to secure the folding roof against the wall section of the expanding side.

Kenneth E. Ghostlaw received his BSME from Tufts University in 1958. He worked at GTE for nine years, performing Methods Engineering and Design Engineering for the Minuteman Program. Mr. Ghostlaw joined Automated Systems Division in 1965 as a Senior Member, Technical Staff, and became a Design Manager, Product Design in March 1968. He has been responsible for the mechanical design and packaging of automatic test equipment systems including LCSS, AEGIS, E3A, AN/USM-410, HELLFIRE, RPV, and AH-64A. Mr. Ghostlaw has directed the design and development of the AH-64A expansible test van, which has proven to be a key component in the successful deployment of the AH-64A Electronic Equipment Test Facility.
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Retracting the sides is essentially the reverse. The floor and roof are unlatched and the folding floors are raised by the winches to the "up" position. A pair of microswitches sense that the floors are up and automatically transfer power from the winch circuit to the expanding side drive motor, which then retracts the sides. The motor circuit is de-energized when one of a pair of microswitches senses that a side is in and one "expanding side in" light is illuminated. At this point, the hand crank is used to adjust the sides so they are firmly seated against their seals and both "expanding side in" lights are illuminated. Then the exterior expanding side latches are used to lock the lower edge of the folding roof securely against the wall section and it, in turn, against the fixed portion of the van. Compressible seals along all interfaces provide a reliable weathertight seal.

The ATE Control Panel contains the circuit breakers that control the ATE, E/O subsystem, LRU cooling, E/O bench isolator air, convenience outlets, and fluorescent lights.

The Facilities Control Panel contains the controls, indicators, and alarms for the Environmental Control System, gages to monitor primary input power, and switches that control interior lights. Both the manual and automatic ECS controls are built into this panel and linked to the ECUs through a three-position wafer switch. In the "off" position, there is no connection to the ECUs. In the "manual" position the GFE controllers are each connected to their respective ECUs. In the"auto" position, the automatic control circuit controls both ECUs through a sequencer. This arrangement provides both backup control if the automatic system fails and a means of fault isolation between control and ECU failure.

All panels are functionally grouped and supplied with appropriate markings that are readable in low light levels in accordance with MIL-STD-1472. Finally, at each panel location and near each door, there is an Emergency Power Off button that disconnects all power in the event this becomes necessary.

Acoustics and vibration

With two large military air conditioners attached to the van, it was necessary to isolate the workspace from A/C noise and vibration. Each unit weighs 620 lbs and has a large compressor and a 2-ft diameter fan that operate constantly, producing an operating noise level of 85 dBA. Ear protection is required for personnel working on the units while they are operating.

Each ECU is mounted in an isolation mount having a natural frequency of about 10 Hz, which provides substantial attenuation of structure-borne noise and vibration. These mounts were designed to have no rocking mode oscillations because they suspend the ECU symmetrically about its center of gravity in all planes. They also have a lockout for transit and are of a failsafe design.

Airborne noise is reduced by using leaded acoustical foam barriers to attenuate transmission through the walls of the plenum that face the interior. Transmission is also reduced by providing two sharp 90-degree bends in each transmission path between the ECUs and the interior, and by lining all reflecting surfaces with acoustical foam. Although the ECUs each generate noise levels of 85 dBA, the sound pressure level at the operator's primary workstation is only 72 dBA, 3 dB below the requirement of MIL-STD-1472 for general workspaces.

Lighting

The lighting requirement is 100 lm/ft^2 at the work surfaces as specified in MIL-STD-1472, Table XIX (fine bench work), with 10 lm/ft^2 in walkways and 30 lm/ft^2 at console surfaces. The fluorescent lighting system in the van delivers 120 lm/ft^2 at work surfaces, 30 lm/ft^2 in walkways, and 35-60 lm/ft^2 at all electrical panels. The system uses high output 20W and 40W fluorescent tubes with low temperature ballasts that allow operation down to −15°F. Below this temperature, an auxiliary incandescent system automatically cuts in. There is also an emergency light system, battery operated with a built-in reset delay, that comes on immediately when the primary lighting circuit loses power, and turns off automatically 15 seconds after power is restored.

Qualification testing

Qualification Climatic and Mobility Tests were conducted by RCA on the expansible van for the Army Apache Program. The climatic test phase, which consisted of high/low temperature, humidity, rain, and dust exposures in both the operational deployed configuration and the transport configuration, was conducted at the McKinley Climatic Laboratory, Eglin Air Force Base, Florida. Mobility Type III (Modified) endurance and mobility tests were conducted at the Aberdeen Proving Ground, Maryland. These tests consisted of endurance exposure to Belgian block, gravel, paved roads, secondary "A" and Perryman No. 1 courses, braking, ramp tests, handling, gradeability, fording, flexibility, weight, weight distribution, center of gravity measurements, and a safety evaluation of the system. The expansible test van was also subjected to service and parking brake tests, emergency breakaway, turning ability, suspension flexibility, lifting attachment and tiedown device test, an accelerometer-instrumented road shock and vibration test, and a fording test.

The primary objective of the tests was to qualify the van to the climatic and mobility requirements imposed by the AH-64A system specification. The results show that the system meets climatic requirements and is suitable for operation over terrain comparable to Mobility Type III (Modified) road courses. The expansible van met requirements for high and low temperature, including deployment at −65°F. Internal van air temperatures were maintained between +65°F and +75°F during maximum equipment thermal dissipation at +125°F external temperature with full solar load. Actual van weight, fully assembled, was recorded at 41,880 lbs with a weight distribution of 13,020 lbs on the fifth wheel and 28,860 lbs. on the undercarriage dolly. During instrumented road testing at Aberdeen, maximum shock loads were 1.6g on the van body and 1.5g on the ATE electronics.

Conclusion

Automated Systems Division has developed, built, tested, and fielded an expansible van compatible with M818 and M931 5-ton tractors and qualified to AR70-38 environmental extremes, MIL-M8090, Type III (Modified), Mobility, MIL-A-8421 air transportability in C-5 and C-141 aircraft, and AR70-47 marine transportability. Optimized for strength, weight, and structural performance, the van appears almost identical to a standard XM995 Army van and has 65 percent more interior space when deployed. Interior van design provides lighting, heating, air conditioning, and equipment arrangement for efficient operation. Deployment of the expansible van is accomplished by a team of trained personnel in less than one hour. The simple task of expanding van sides, lifting expanding side roofs, and lowering expanding side floors is accomplished automatically in approximately ten minutes. This newly developed military vehicle is proven as a qualified enclosure for operation, transport, and storage of sensitive electronic equipment. The ratio of van weight to payload capacity and available operational workspace in the AH-64 expansible van offers system designers a unique option in enclosure selection.

W.K. Holden

Mechanical design problems associated with glass-reinforced polytetrafluorethylene printed-wiring boards

Material problems solved, coated boards exceed military requirements.

The first question an engineer will ask after reading the title of this paper is, "Why use glass-reinforced polytetrafluorethylene (GRPTFE) for printed circuit boards?" The two main reasons are:

1. It maintains an almost isotropic dielectric constant over a wide frequency range.
2. It has a low (current) dissipation factor.

Some other advantages of GRPTFE are it can be used over a wide range of temperatures, it does not absorb water, and it is funginert.

But the use of GRPTFE for military-standard printed circuit boards also presents some unique engineering challenges. The two design/production related problems encountered by RCA Missile and Surface Radar Division in using this material for a single-layer circuit board were associated with the conformal coating and annular ring-forming processes.

Abstract: *Glass-reinforced polytetrafluorethylene (GRPTFE) printed-wiring boards offer desirable circuit performance characteristics, but present several mechanical design and production process challenges. Two major problems are the instability of the polytetrafluorethylene (PTFE) or teflon component at room temperature, and the resistance of the glass/PTFE composite to bonding. This paper describes a procedure that the RCA Missile and Surface Radar Division developed to solve these problems. It meets the mechanical design constraints of a printed circuit product and accommodates the unique properties of the GRPTFE material.*

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Annular ring formation

The first problem we experienced was the expansion of the GRPTFE boards during the fabrication process (described in the sidebar on this page). Dimensional changes averaged 0.010 inch in the X axis and 0.005 inch in the Y axis (see Fig. 1). These changes resulted in the loss of an acceptable annular ring, defined as 0.015 inch (minimum) in MIL-STD-275D for single-layer boards.

We began our investigation of this problem by carefully measuring the boards after each step in the fabrication process. Board expansion occurred at Step 4, the vapor-hone cleaning procedure (see sidebar).

Vapor honing uses water and pumice granules released from a point nozzle at a line pressure of 80 psi. The operator randomly scans the board and, with the water/pumice stream, roughens the through holes. We determined that the pressure of the honing stream and the instability of PTFE at room temperature—glass transition temperature (Tg) 19°C, or 66°F—caused the GRPTFE board to peen, producing the dimensional change. When we eliminated the vapor-honing process step, the dimensional change did not occur. Consequently, manual scrubbing with water, pumice, and a brush was substituted to maintain good through-plating quality.

Conformal coating

According to MIL-STD-275D, PTFE boards are exempt from the conformal coating requirement. However, we wanted

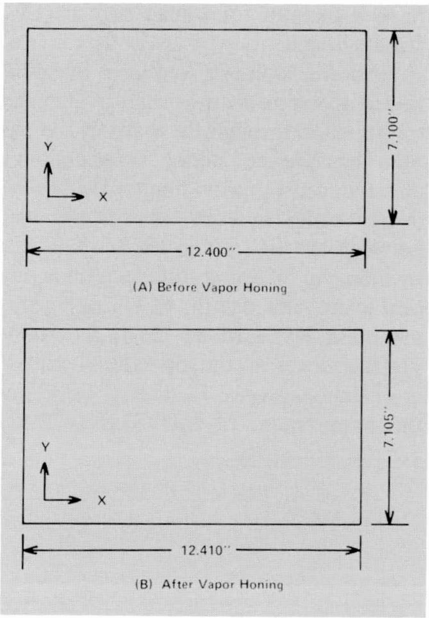

Fig. 1. *Average dimensions of GRPTFE printed circuit boards (a) before and (b) after the vapor hone cleaning process.*

to exceed this standard and coat the board to protect exposed tin/lead traces and component leads against moisture, and eliminate the possibility of moisture bridging across the 0.010-inch spaces between traces.

Parylene and polyurethane testing

The first order of business was to determine what to use as a conformal coating. After referring to MIL-I-46058, we selected parylene (XY) and polyurethane (PR) for testing. Parylene was quickly eliminated from consideration. It flowed into restricted spaces and deposited insulation on the conducting surfaces, thus isolating component bases from the wiring board ground plane. We, therefore, focused our attention on polyurethane.

Testing was done to assure that the polyurethane coating would not significantly affect the impedence of the circuit board module. Dielectric-withstanding voltage tests were performed on the boards following the standard moisture and humidity test (MIL-STD-202, Method 106; seven cycles). There was little or no change in the mean voltage value (10^6 megohms). The individually coated boards used in these tests exhibited good urethane coatability and adhesion.

However, when a second set of boards was coated in an engineering qualification cabinet, we obtained poor coatability, evidenced by the appearance of large voids in exposed PTFE areas (see Fig. 2).

A research project was initiated to determine what was causing the inconsistent and inferior urethane conformal coating, and to find an acceptable process method for producing conformal-coated GRPTFE boards.

The glass-reinforced polytetrafluorethylene (GRPTFE) printed circuit board fabrication process.

1 Prebake board (1 hr at 300°F)
2. Drill board
3. Deburr holes
4. Clean board (pumice and water)
5. Bake in oven for one-half hour at 250°F
6. Plasma etch
7. Electroless copper
8. Copper flash (fine-coat copper)
9. Clean surface
10. Apply photoresist
11. Print
12. Develop
13. Copper plate
14. Solder plate
15. Strip photoresist
16. Copper etch
17. Bake board for one-half hour at 250°F
18. Solder reflow
19. Route board to size
20. DITMCO test
21. Laminate heat sink to board

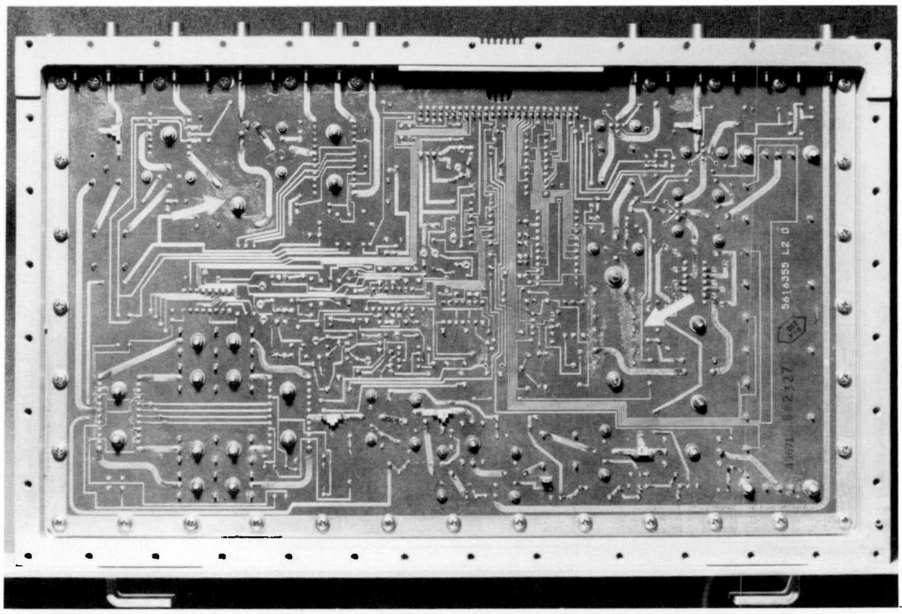

Fig. 2. *Engineering qualification assembly. Arrows indicate board areas with poor coating.*

Polyurethane coating process research

We began by studying the raw materials used in making the single-layer board product. Boards, fabricated according to the process detailed in the sidebar, were 0.031-inch thick GRPTFE with one ounce of copper 0.0014 inch thick on each side and two layers of polyurethane.

From discussions with our material vendors and the results of our own analyses, we could find no compatibility problems between polyurethane and GRPTFE. However, there were some gray areas regarding the vendor's manufacturing process for the GRPTFE laminate. Understandably, the vendor was reluctant to reveal just how the copper was applied to the board. We did learn that some type of chemical etching process is used to prepare the GRPTFE board surface, and that a direct heat-fusion process (700°F) bonds the copper to the laminate.

We prepared unassembled sample circuit board sets according to the original process for use in two tests.

Boards in the first test group were taken directly from the vapor-degreaser process step and sprayed with the polyurethane conformal coating according to the procedure outlined in the sidebar. Although we had previously experienced poor results in the engineering qualification cabinet with this approach, we were surprised to find that this time results were fair, with good adhesion and some small voids in PTFE areas. A second test board set was cleaned with acetone and sprayed. Results were again fair, showing good adhesion and minor voids in the exposed PTFE areas.

Boards in the second test-sample group were taken two steps further along the production process. A heat sink was laminated to the ground plane, and various components were assembled on the boards. The first sample set in this second test group was vapor degreased after assembly and then sprayed. From this procedure, we

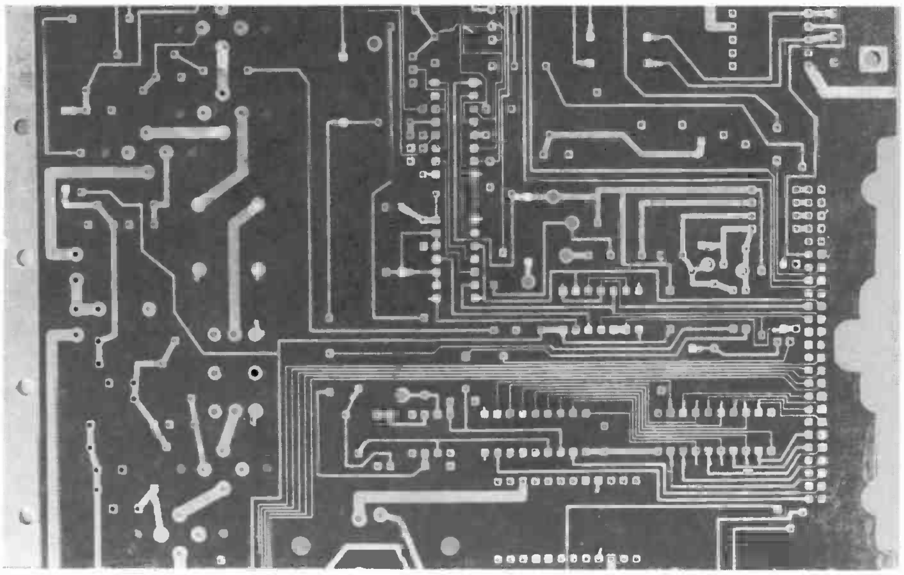

Fig. 3a.

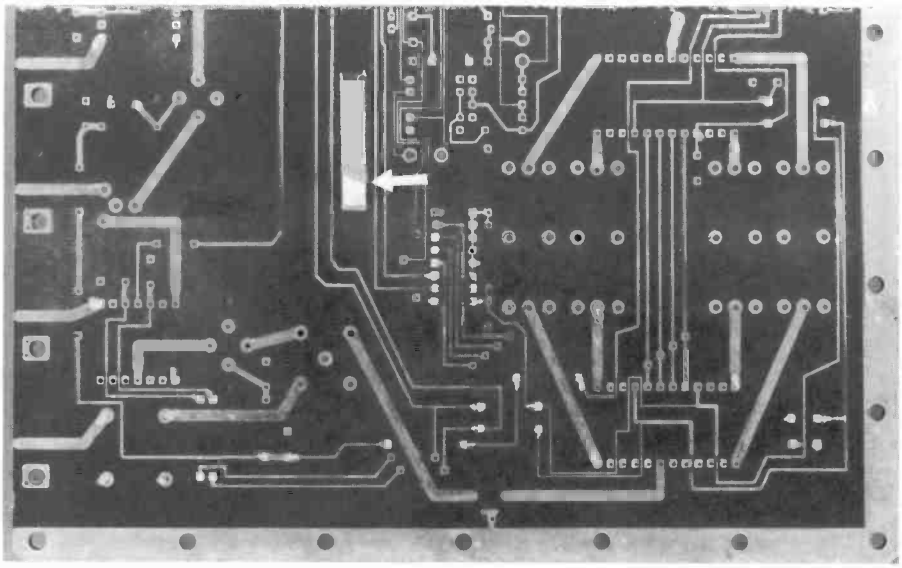

Fig. 3b.

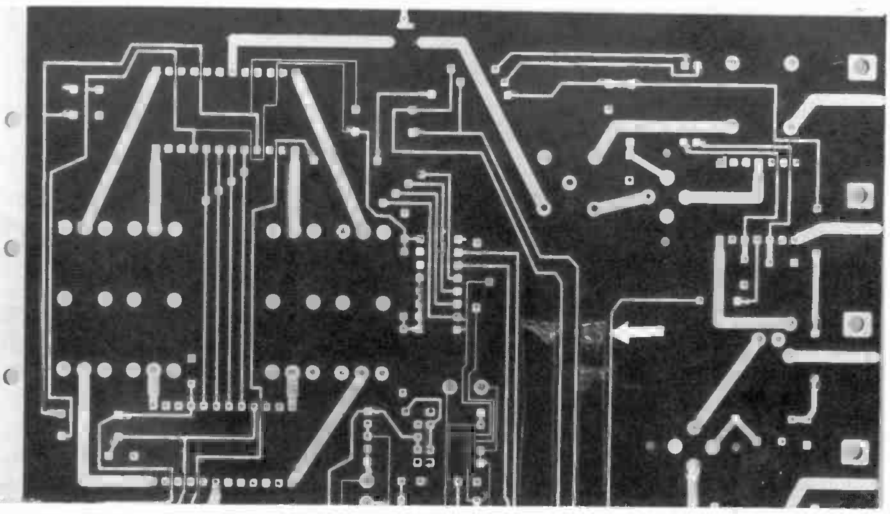

Fig. 3c.

The conformal-coat spray application process.

Conditions:

Material: 2 parts urethane
Pressure: 30 psi

Procedure:

1. Heat board in oven for 60 minutes at 150°F.
2. Spray fog coat (two passes urethane).
3. Air dry for 30 minutes.
4. Oven dry for 10 minutes at 150°F.
5. Spray finish coat (urethane).
6. Air dry for 30 minutes.
7. Oven dry for 120 minutes at 150°F.

obtained good adhesion and fair coating (see Fig. 3a). A second set was plasma-etched and sprayed. Coatability was excellent, but the urethane separated from the PTFE board when we subjected the board to a peel test (see Fig. 3b). The third set of boards was etched with a proprietary sodium solution. Sodium etching breaks the fluorine-carbon bond on the PTFE, producing a wettable surface. The result achieved from spraying polyurethane on the sodium-etched board was excellent in terms of coverage (coatability) and adhesion (see Fig. 3c).

Results of coating research

Generally speaking, the surface of PTFE-based boards must be etched to bond anything to it. The results we obtained from testing various GRPTFE board preparation techniques generally support this principle, with one interesting exception. The untreated boards in the first test group

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Fig. 3. *Samples from the second set of polyurethane board coating tests. (a) This board was vapor degreased before spraying. (b) This sample was vapor degreased and plasma etched before spraying. Arrow points to area of peel test. (c) The third sample from the second test set was vapor degreased and sodium etched before spraying. Arrow points to area of board where peel test was attempted.*

The shrinking master

Another problem commonly encountered in board production is shrinkage of the mylar artwork master, which produces hole/pad misalignment. To avoid this shrinkage, which is caused by changes in temperature and humidity, the environment in which the artwork is used must be carefully controlled. In some cases, glass artwork masters offer an alternative to mylar and its related atmospheric control problems.

performed nearly as well as the sodium-etched boards in the second test group.

The only explanation we have for this finding is that the etching process used by the vendor prior to the bonding copper to the PTFE leaves a residue that remains after the copper is etched off at RCA/MSRD to produce the printed circuit. This residue apparently helps bond the polyurethane to the GRPTFE board, producing a conformal coating similar to that obtained with sodium etching. (The poor results we initially obtained with the engineering-qualification-cabinet boards were a result of either water or oil in the sprayer air lines.)

Because the board coating results with no etching were comparable to those with sodium etching, and because the sodium etching solution is hazardous to handle, we decided not to include etching in the production process. We did allow for an optional touch-up step to ensure complete conformal coat coverage.

Summary

Special design and process allowances may be needed to adapt a new board material to an existing manufacturing process for printed circuit boards. In introducing glass-reinforced polytetrafluorethylene boards, we were able to identify these material-related problems and make the allowances necessary to produce a product that exceeds military specifications.

Acknowledgments

I would like to thank Paul Cathel and Bob O'Brien, Methods Engineers, and Rob Mattioli and Ralph Taylor, Mechanical Engineers, RCA Missile and Surface Radar Division for their capable technical assistance during the course of this project.

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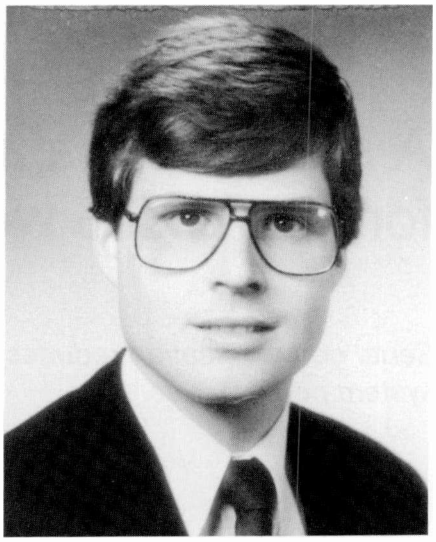

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H.R. Weiss

A new peripheral computer interface for the AEGIS weapon system

Serial communication reduces cable weight and improves system performance.

What is the effect of increased use of shipborne computers? We are finding that the complexity of communications between computers and peripherals means more cabling and connectors. But a new group of computer interfaces will considerably reduce the weight of this electronic cabling being installed in the new ships. Simpler connections should improve reliability and permit greater communication distances.

The AEGIS weapon system, currently in development at Moorestown, is a good example. It includes a network of shipborne computer-controlled and controlling peripherals. United States Military Specification MIL-STD-1397A prescribes how "peripheral equipment" on warships communicate with the assigned computer. Generally, information is passed between computers and peripherals in 85-wire coaxial cables, two per port of communication. However, MIL-STD-1397A now permits communications in serial form rather than parallel, a change caused by another specification, STANAG 4153, that binds equipment used in NATO countries.

Abstract: *A recent change in the Government specifications now permits low-level serial transmission, instead of parallel, between digital computers and peripherals. At Moorestown, this change is making a significant difference in the development of new shipborne automated-weapon systems. The change will decrease the weight of weapon-system cabling, improve reliability of the connections, increase transmission distances, and lower data-error rates.*

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Advantages of serial transmission

What is the advantage of converting to this new transmission method? Initially, it is weight reduction. Serial transmission requires triaxial cables, which have an outer shield enclosing two concentric inner conductors. Triaxial cables vary from 1/4 to 1/2 inch in diameter, depending on the required communication distance, while 85-wire coaxial cables are generally 1 inch in diameter. By using triaxial cable for short distances, we can reduce the cross-section area as much as 16:1. For long distance the reduction is 4:1 for the thicker cables. We estimate that the new transmission method will save up to 3000 pounds of cabling per weapon system, an important reduction for ship stability, because most of the cabling is above the ship's center of gravity.

Another advantage is the increased communication distance now possible. For parallel transmission, the typical maximum distance is 300 feet for slow data flow and 150 feet for fast. For serial transmission, the cable length can be 985 feet (300 meters). This increase permits wider latitude in the design of manned computer displays for management and coordination functions. These greater communication distances also allow us to maximize the ship's survivability under battle conditions by putting redundant computer bays farther apart.

Usually, the remote end of a cable must be connected on the ship during limited visibility conditions in cramped quarters. But connecting multiwire cables to 85-connector pins has caused problems even in manufacture. The use of lightweight prefabricated triaxial cables will reduce the number of failures resulting from bent connector pins, open wires, or incorrectly seated connectors. Thus, serial input and output systems should be more reliable.

Parallel inputs and outputs

The parallel input and output system requires "handshaking" signals. The computer and peripheral exchange special signals over selected lines of the 85-wire cables to control any data transmission.

For example, in Fig. 1, one line is the input data request (IDR). To send input data, the peripheral sets IDR to logic 1 and also sets its 32 data-signal lines and their returns. In response the computer raises an Input Data Acknowledge (IDA), a separate line, to logic 1, causing the IDR to drop to logic 0.

Other lines are designated for data output—output data request and output data acknowledge. Similarly, separate lines handle interrupt signals (requests from a peripheral for immediate attention) and external-function signals, as well as their acknowledgments.

In the parallel mode of communication, control signals stay in the logic 1 state until the sender's logic circuit senses the acknowledgment. The request signal then drops to

0 and the acknowledge signal, sensing the drop, also drops to 0.

Serial inputs and outputs

Reliability of serial transmission for military computers has not been fully tested, therefore the error-rate values under certain conditions are not known. To provide some protection against accepting faulty data, several error-detection schemes are used. In the AEGIS system we use a 35-bit message that contains a parity bit. The message will be screened for correct parity (is it a 1 or a 0?), the proper number of bits, the correct word identification and, in some specific cases of the received clock, whether the three most-significant bits are zero. If any check fails, logic circuits will raise a flag for the AEGIS operational readiness test. This flag alerts the computer and the operator that something is malfunctioning.

MIL-STD-1397A describes the new low-level serial communication as Type E. This new type needs only one triaxial cable between the sender and the receiver, even though control signals must travel in both directions (half duplex). However, transmission of data in two directions requires two triaxial cables. The 34- or 35-bit information frame (IF) and the 4-bit control signals are sent at a 10-MHz bit rate. Transmitted data (for synchronizing the transmission) accounts for 32 bits of the IF. The remainder are a sync bit, word identification (is it a data or interrupt word?) and, in the 35-bit IF, a parity bit (for detecting errors).

Serial transmission of information also follows a predescribed protocol or handshaking sequence (see Fig. 2). A 4-bit coded SOS (Source Status Control Frame) initiates the transmission, and the called peripheral or computer replies with a 4-bit coded SIS (Sink Status Control Frame). The reply indicates if the receiving device is ready (or not ready) to accept the data or an interrupt.

Depending on the reply, the sending device transmits the data word or the control-interrupt word (CIW)—or none (if the reply stated: Not Ready to Receive Either Data or CIW)—as a 34- or 35-bit IF. The receiving equipment must reply again with an SIS to acknowledge receiving the message and state whether it is ready to receive more data.

In contrast with parallel protocol, the serial handshaking signals are transmitted serially (one after the other) over the common triaxial line (Fig. 3), and do not remain on the line. Maximum and mini-

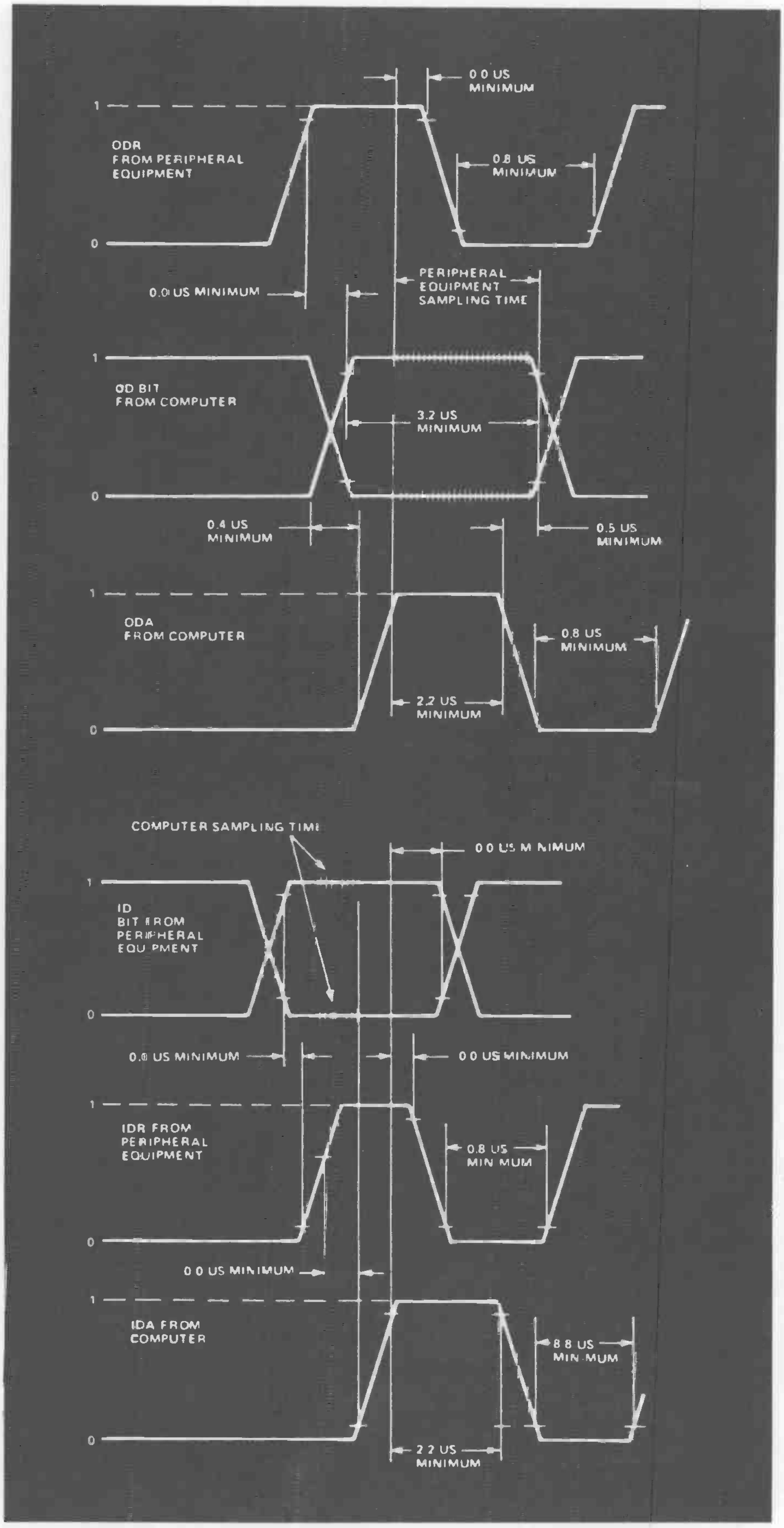

Fig. 1. *Parallel output/input protocols.*

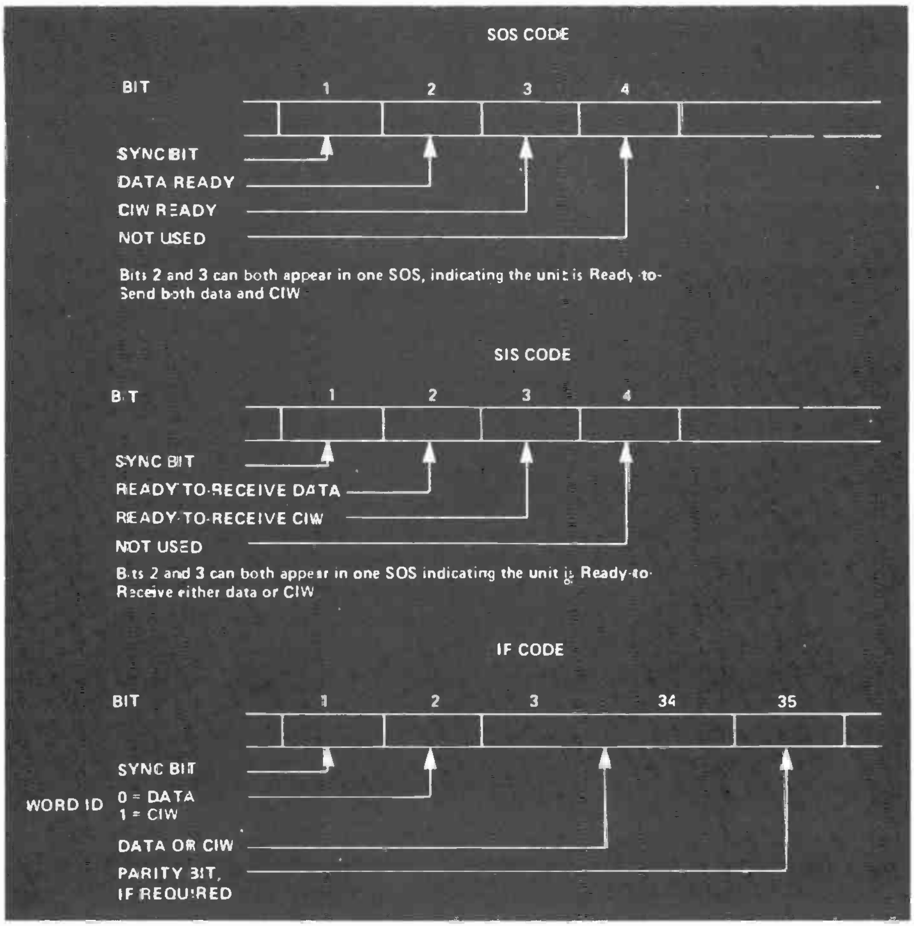

Fig. 2. *Serial protocols.*

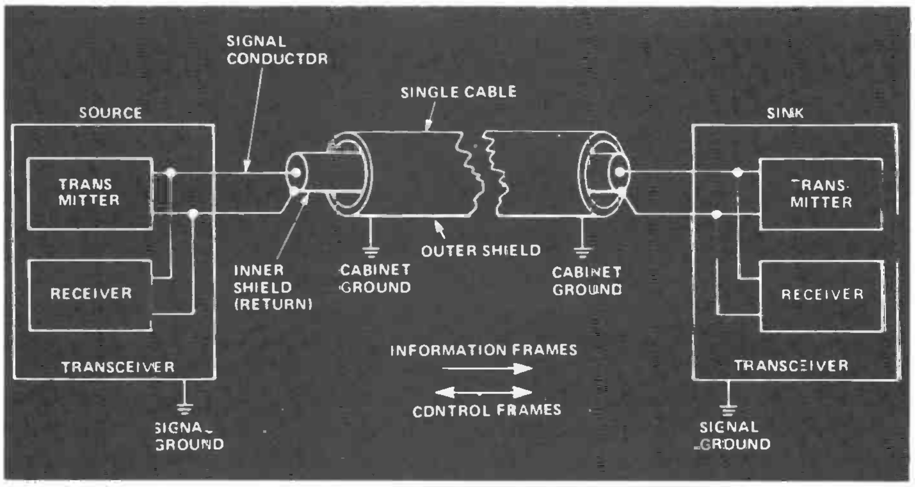

Fig. 3. *Triaxial line.*

mum response time to the SOS are 150 microseconds and 0.5 microseconds, respectively.

The new technique appears simpler, but imposes some communication restrictions for interrupts and forced functions. The equipment cannot be interrupted or forced to accept a function without immediate, prior consent in the handshaking protocol.

Modes of transmission

We can use different prearranged modes of serial transmission. The choice is important, because it affects throughput, which is the amount of data that can be transmitted per time unit.

In the slowest mode, single-word transmission, the sending device sends an SOS before every IF.

A faster mode is the ping-pong mode in which only an initial SOS is sent. Thereafter, the sending device transmits an IF in reply to the SIS.

The fastest possible mode is burst mode. Only one SOS and as many as 32 data frames are sent without interruption or waiting. However, current hybrid circuits cannot handle transmission and reception in the burst mode.

Throughput

Throughput is a measurement of the total amount of intelligence transmitted per time unit. Besides the significant bits that convey the intelligence of the message, the data transfer contains control bits, sync bits, and parity bits. Time is the interval between receipt of a message and reply.

If the computer or a peripheral responds to protocol signals in 0.5 microseconds, (the minimum permissible time in MIL-STD-1397A), then the throughput of serial transmission can equal that of parallel transmission. For practical considerations, however, the computer may take as long as 150 microseconds (the maximum permissible time) to reply. If the longer cable length introduces an additional 3 microseconds per round trip, then the available throughput of regular single word transmission and of ping-pong mode are severely curtailed (see Table I).

Only an improvement in the time it takes for the computer to open a buffer (a temporary storage device) can increase the throughput. For example, the gyro data converter uses random access memory (RAM).

What is the throughput if we neglect to consider the time delay of the cable and allow for a minimum response time to control signals and information frames? Because transmitting 34-bit words (no parity) shows a negligible improvement over 35-bit words (with parity) our calculations ignore word size.

As Fig. 4 shows, the throughput for the serial mode varies from an optimum of 294 words per millisecond in a system that uses the burst mode over lines short enough to cause no cable delay to a low of 78 words per millisecond for a system that uses single-word transmission over 300-meter lines.

By comparision (Table II), if we neglect cable delay, throughput for the parallel mode of data transmission is 333 words per millisecond on input and 294 words per millisecond on output. The permissible cable length is not well defined in the military specification, but with the same 300 meter length, the throughput for parallel transmission input or output is 261 words per millisecond. All these results assume minimum response time by both computer and peripheral.

The Manchester Code

Serial communication uses digital, Manchester-encoded data. The Manchester Code is a biphase-level code (Fig. 5), the result of the exclusive OR of a non-return to zero (NRZ) signal with the signal's clock. (The result is a logic 0 unless either NRZ or the clock, but not both, is a 1.)

Decoding NRZ data requires clock information, and NRZ data contains a large dc level if there are more ones than zeros.

Type E requires that the Manchester Code's waveform, Fig. 5B, be balanced on both sides of the zero voltage line. A logic one results in a negative level change at the mid-bit interval, while a logic zero results in a positive level change at the mid-bit interval.

While the resultant bilevel Manchester Code does not possess any intrinsic error-detection schemes, it has several advantages over the NRZ waveform:

- ☐ The clock signal does not have to accompany the transmission because we can derive the original clock directly from information contained in the Manchester-coded signal.
- ☐ The Manchester-coded signal's dc level is zero, and the signal can pass through a high-frequency transformer in either direction. This creates the half-duplex mode that requires only one cable for transmitting data and two-directional control signals.

Manchester encoding has been used for many years at lower frequencies in modems (modulator/demodulators) that transmit the serial message over telephone wires. Also, since 1975, local area networks have been using Xerox Corporation's Ethernet high-speed (10 megahertz) computer communications network that uses Manchester Code. However, except for the 10-megahertz transmission frequency and the Manchester Code, Ethernet network does not resemble the military computer's communication.

Equipment undergoing redesign

At present, we are redesigning two pieces of AEGIS equipment by converting most of the computer interfaces from a parallel form to a low-level serial exchange.

Table I. *Typical throughput calculations, ping pong mode*

		No cable delay **Time = 1800 + $n \times 4900$ ns**	**300 meter delay** **Time = 4837 + $n \times 7937$ ns**
4 bit SOS		400 ns	400 ns
Transmission delay		0	1518.5
Delay to reply		500	500
4 bit SIS	*n times*	400	400
Transmission delay		0	1518.5
Delay to reply		500	500
35 bit IF		3500	3500
Transmission delay		0	1518.5
Delay to reply		500	500
4 bit SIS		400	400
Transmission delay		0	1518.5
Delay to reply		500	500

Table II. *Throughput for different modes of transmission (words per milliseconds)*

	Serial transmission (300m cable)			
Number of words per message	**Single word transmission**	**Ping-pong modes**	**Burst mode**	**Parallel transmission (300m cable)**
1	78	78		186
2	78	97		217
5	78	113		241
10	78	118		251
25	78	123		256
32	78	124	278	257
50	78	124		258
64	78	124	282	259
96	78	124	286	260
100	78	124		260
128	78	125	287	260

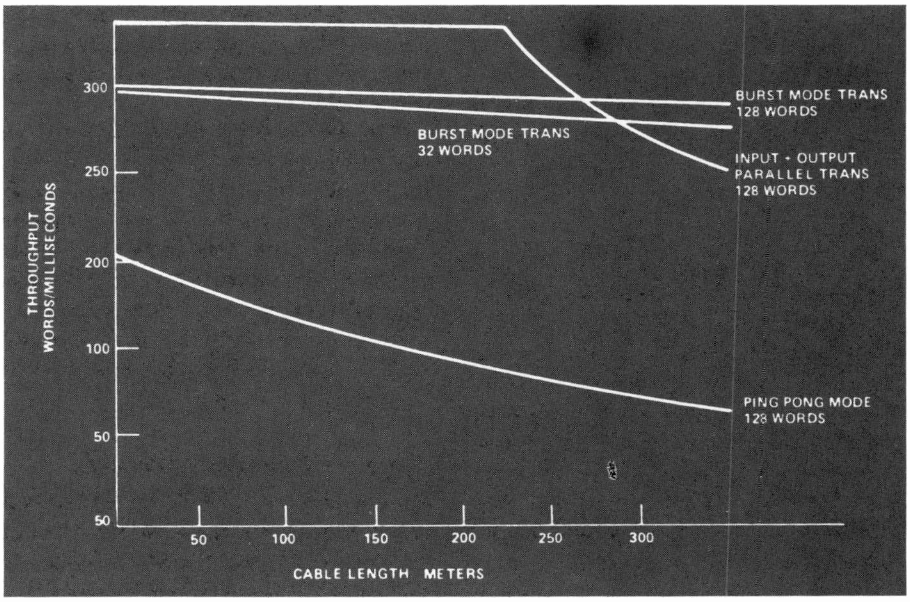

Fig. 4. *Throughput.*

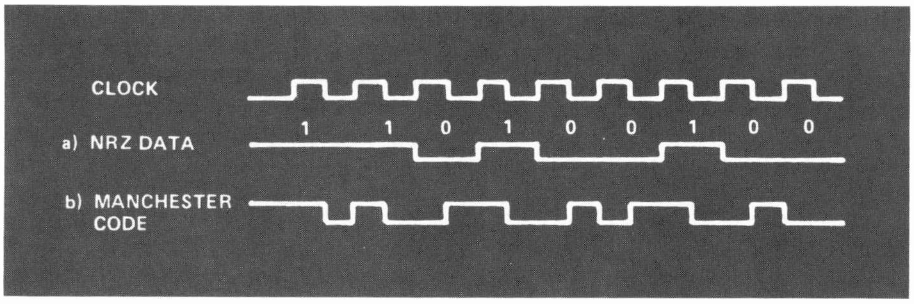

Fig. 5. *Manchester Code. (a) is an NRZ level code. A logic 1 is represented by a high level during the total clock period and logic 0 as a low level during the total clock period. A string of ones is a high dc level; a string of zeros is a low dc level. If the NRZ level data is one input of an exclusive OR, and the clock is the other input, then the negated output of this logic forms a waveform (b) that corresponds to the Manchester code.*

One unit is the digital clock that synchronizes all the ship's computers to the time of day with an accuracy of 1 millisecond. It generally transmits one 35-bit data frame simultaneously to as many as 22 computers in 1 millisecond in single-word mode. Once every eight seconds, it also transmits a second message, the control-interrupt word using the rules of the ping-pong mode.

The other unit undergoing redesign is the gyro data converter that collects data from the ship's gyros. Its microprocessor performs the necessary arithmetic and converts the data into a transform matrix for the computers and signal processors.

The gyro data converter must service eight computers with one control-interrupt word and 17 data words for each within 3 milliseconds. So it uses the ping-pong mode for higher throughput. However, it also uses the ping-pong mode to transmit 93 array data words to one of two computers. While the gyro data converter waits for a reply after completing one transmission it can load the next set of data into its transmitting device.

The unit that houses the digital clock and the gyro data converter, among others, is the clock converter cabinet.

Electronic circuits

Five different hybrid circuits, developed by Circuit Technology Incorporated, form the low-level serial interface. Three of the five circuits form the transmission circuitry, and another set of three receives the data. The CT 1469 input/output device, common to both, converts the internal Manchester Code to a bilevel signal for transmission, and converts the received bilevel signal to Manchester Code for decoding by another module.

It is desirable to disable the transmission path of the unit, except during transmission of data. During that transmission period, we inhibit reception to avoid feeding back transmitted signals to the receivers. This is not necessary, and other users may prefer the opposite.

A full-scale decoder (CT 1497) converts the received 4-bit control signals (SOS) and 35-bit data signals from serial words to parallel words, while an abbreviated encoder replies to the received words with a 4-bit SIS.

We combined two transmitter/receivers into a standard AEGIS printed circuit board with 70 output pins. The full encoder and abbreviated decoder form a second printed circuit board, and the abbreviated encoded and full decoder form a third.

Logic circuits on all three boards perform auxiliary functions. For example, when a control word or a full-size data word has been received, logic decodes it by combining the encoder's trailing end of the transmission envelope with its internal counter of less than five or more than five bits (SIS/SOS FALSE). The required 100-400 nanosecond encoder-enable pulse output starts the transmission.

Every board includes a 40-megahertz crystal oscillator. While one well-buffered oscillator could drive all 22 boards of the encoders, letting this high frequency spread from one end of the backplane to the modules could cause crosstalk problems. Separate oscillators avoid the problem.

Heat sinks over the encoder and decoder dissipate up to 3.25 watts from each of these compact hybrids.

Cable and impedance matching

We are matching the 50-ohm triaxial cables to the transformer-coupled transmit/receive unit. Any mismatch creates disturbances in the coded message and could cause errors if the cable is the right length to include these disturbances with the message. We consider 50-ohms (±2 ohms) a suitable match.

If one cable is long enough and not terminated, then a valid outgoing SOS will be reflected with little attenuation and, to the sender, will appear to be a returned SIS (the codes are alike). The sender could then transmit data whose reflections may again appear to be a valid message.

Coding the unused fourth bit of SIS and SOS might help, but this bit is used now for other purposes in equipment specified by STANAG 4153.

Cost of new complexity

From the beginning, we expected the new electronics to increase design program complexity. One estimate of the complexity is the number of logic circuit boards used for a particular system. The AEGIS construction uses a backplane with component boards plugged in at right angles. Generally, the boards carry several integrated circuits.

For example, the previous digital clock provided parallel outputs using 120 circuit boards. The new clock that provides both serial and parallel outputs will use 131 circuit boards. Similarly the gyro data converter, which previously used 40 logic plug-in boards, now requires 91. Of the additional 51 boards, we use 40 to provide more output ports. We use the other 10 boards to change the parallel interface to a serial one.

Testing

We plan extensive testing of the new units and system. We will test transmission and reception of control signals using the newly designed electronic circuits. A small laboratory test setup will show how well these boards function under different environmental conditions and cable connections. It will also point to critical connections between the units.

The redesigned equipment will be tested using a Digital Equipment Corporation PDP-11 computer that we will program to send and receive data and test validity. Logic circuitry will perform the liaison between the unit under test and the computer. These circuits will simulate the low-level serial handshaking of the AN/UYK-43 computer with the AEGIS equipment and the prescribed PDP-11 computer's handshaking with the AEGIS computer. Timing and performance will be tested at this level.

Finally, the total system will be connected to the AN/UYK-43. A systems test will show systems-performance throughput, shortcomings due to crosstalk, or pickup from electrical disturbances.

A look into the future

We believe that fiber optics may replace transmitters and receivers in low-level serial transmission, reducing weight and power. The optical link should be a very stable transmission element, because it is not subject to disturbances from ordinary electromagnetic radiation. The reduced weight of optical fiber cables, as compared to triaxial cables, is another advantage in this future development.

Where throughput of the device is important, we must also develop hybrid circuits to perform in the burst mode. Figure 4 shows the great advantage of this mode over the ping-pong mode. Other major throughput improvements must be made in the computers themselves.

Acknowledgments

The author wishes to acknowledge the contributions of Tom Brady, who served as MSR's liaison in the development of the Type E Specification; Doug Lewis, who decided how the hybrids were to be mounted in pairs in AEGIS boards; Rick Kurowski, who was instrumental in the design of the printed circuit boards for the transmitter, encoder, and decoder; and Al Montemuro, who designed the logic for the gyro data converter. I also want to thank Todd Hansen, who helped calculate and plot throughput, and Bob Paglee, who encouraged me to write this paper.

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R.J. Felbinger | H.G. Rouland

Improved system engineering support of instruction modes, technical manuals, and training courses

How can developers of training courses and manuals derive information about complex systems? Functional analysis provides an understandable translation for writers, instructors, and trainees.

RCA's Functional Analysis group—working closely with the Technical Manuals activity—recently embarked on a program to enhance the quality and improve the currency of functional descriptions of the combat system. These descriptions form at least one volume of the AEGIS Combat System Technical Operations Manual (CSTOM) that is used at the Combat System Engineering Development (CSED) Site and to train and indoctrinate Navy personnel.

Functional analysts improved information accuracy and completeness. They also reformatted the functional data into tactical engagement sequences, a new method of presenting functional analysis information. Such sequences refine the data presentation and help instructor and trainee understand and retain functional data.

Abstract: *As prime contractor for the US Navy's AEGIS program, RCA must develop and maintain an accurate, comprehensive, and up-to-date description of the AEGIS combat system. Without such documents, the Navy could not properly train and orient personnel who manage, operate, and maintain the system. A new approach to describing an evolving system like AEGIS has proven that maintaining current materials to preserve manager and operator proficiency is a challenging but achievable task.*

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Reprint RE-30-6-10

This new method, combined with a highly adaptable and up-to-date database, offers a promising approach to producing effective technical manuals in both site and shipboard operational training. With it, we can develop and format functional-analysis information and products that serve different users with a variety of information needs. This paper focuses on how we used functional information as engineering input to the AEGIS Combat System Technical Operations Manuals and to the Technical Manuals and Training activities.

Improving a relationship

The AEGIS system engineering and logistics activities—Functional Analysis, Technical Manuals, and Training—always maintained a cooperative working relationship (see sidebar). But we are strengthening this association because RCA and Navy management are dedicated and committed to improving the quality and timeliness of engineering input data. To make this happen, we took several important steps to increase the involvement of system engineering in the technical manuals and training data development process.

For example, we appointed the Functional Analysis activity as a key interface in the system engineering organization. Thus, we had a single point of contact for coordinating information requests from technical manuals and training. To handle and expedite these requests we established the System Information Request System. We also secured management support for conducting technical reviews of training course outlines, lesson plans, CSTOM and shipboard technical operations, and maintenance manuals. In addition, engineers audited selected training courses to learn what problems confront the instructor and trainee who use the engineering information we provided.

In general, these combined measures have increased cooperation between system engineering and logistics disciplines.

A fresh approach to the problem

In its expanded role as system integrator or coordinator, the Functional Analysis activity has solidified the relationship between system engineering and support disciplines. To achieve this, we first established a basic framework for discussion. Functional analysis was the catalyst. The information it produced had to be portrayed in a way that the system engineer, technical writer, instructor and, most important, the trainee could relate to and readily understand. Moreover, the information had to be presented in "digestible chunks," easily assimilated by each individual, yet not simplified to the point of having no real substance.

The solution seemed to lie in some form of functional sequence approach, but operational sequence and timing diagrams did not appear to be the answer. They were scenario-based and time-dependent, and depicted multisensor and weapon system interactions. Thus, they offer more complexity than was really needed.

After considerable trial and error, we

adopted a single-thread approach that linked one sensor to one weapon system in a serialized stream of tactical events and sequences. (The sidebar gives some definitions used in functional analysis.)

The solution: TSDs

The resulting products, known as tactical sequence diagrams (TSDs), cover search, detection, tracking, and identification of a target and weapon assignment until the weapon intercepts and destroys the target. They present the functions in the sequence in which they are performed and show message flow between function blocks. Also, they show that system elements, such as a SPY-1 radar or standard missile (Tier 1 level of analysis), initially perform functions, and that functions are subsequently allocated to individual operator stations, computer programs, or equipment (Tier 2 level).

This new way of organizing the material offers several distinct advantages over the previous method of formatting functional information in the CSTOM. For example, suppose a person requires training in search and detect functions associated with anti-air warfare. We provide a complete set of tactical threads (TSDs) for the anti-air warfare mission. Each thread portrays the simple sequence of functions that accomplishes engaging a particular anti-air warfare sensor-weapon system. Complex feedback loops and input/output messages are minimized in the flow.

Both primary and secondary support functions involved in a thread are identified and allocated to specific manned operating stations, equipment, and computer programs. Descriptions of each function provide the user with detailed events covered by the function. In a logical, serial fashion, the individual learns only what is necessary to perform effectively in a specific job assignment. This should not only noticeably improve the quality of training, but should also decrease the total time required to train an operator or maintainer.

The TSDs are constructed initially as functional segments or clusters of function blocks for a mission. Only when TSDs are tied together serially do they constitute a tactical engagement sequence. A given mission area, such as anti-air warfare, comprises several such sequences, each pairing a single sensor with a single weapon system. Segments are linked in a logical sequence according to a preestablished set of scenario parameters or options.

Figure 1 shows a typical segment of the anti-air warfare mission, employing the SPY-1 radar for search and detection. This segment portrays the simple sequence of functions involved in establishing a passive-angle rack candidate. While this segment and TSD pertain to the anti-air warfare, the same segment could be used in another tactical engagement sequence in another mission area, such as anti-surface warfare.

In other words, we have to describe a unique functional segment only once, even though it may be used with more than one mission area. Therefore, we do not have to duplicate segments—an obvious savings in engineering manpower, time, and costs. Another advantage of the TSD is that it enables the user to focus rapidly on a specific area of interest, such as the portion of the engagement sequence that starts with an engagement order and ends with a missile launch.

To help the user link segments together in tactical engagement sequences, we provide a logic tree network (see Fig. 2). We prepared trees for a mission area and listed the valid set of parameters that the user must adhere to in a tactical engagement sequence. Associated with parameters are up to three options that the user selects.

When selected, the options obligate the user to follow a particular path or branch down the tree. As he or she descends a single branch of the tree, the options available for the particular tactical scenario narrow the paths to specific TSDs that may appear on more than one branch, filling out the tree.

Figure 3 shows how several segments might be linked to form a complete tactical engagement sequence.

Help for developers

The TSD approach simplifies the descriptive process and enhances analysts' ability to describe a complex system, a most significant advantage. By using the TSD building blocks, we make it easier for users to understand the total system.

When used in a working group environment, the TSDs provide a basic framework for discussing key issues that affect system design. They stimulate technical discussion and foster effective technical interchange among engineers. TSDs can clarify functional alternatives and return the discussion to system functional requirements when digression or controversy arises.

The TSDs and accompanying text descriptions can also be modified easily to reflect the collective inputs and decisions of the group. As a design engineer so aptly put it, "Functional analysis and, in particular, the TSD technique offers a disciplined way of thinking about systems and conversing in system terms rather than about individual subsystems or elements."

Definitions

Function block (FB)—A function performed by either an equipment group, a computer program, or a manned operating station. Defined by inputs/outputs and title/numbering scheme.

Functional segment (FS)—A group of two or more functional blocks that performs a portion of a tactical engagement sequence.

Partial tactical engagement sequence (PTES)—A group of functional segments that constitutes a portion of a tactical engagement sequence.

Tactical engagement sequence (TES)—A complete sequence of tactical engagement segments that shows how a single sensor paired with a weapon system performs a tactical engagement.

Tactical sequence diagram (TSD)—A functional flow diagram that portrays a tactical engagement sequence at the FS, PTES, or TES.

When developing the TSDs, functional analysts follow certain ground rules that are designed to improve the overall quality of information, primarily from the trainee's perspective.

As already indicated, we use simple graphics—functional sequence diagrams—backed up by descriptions that augment the information presented on the diagrams. We use the automated database management capabilities of the Functional Information Control System (FICS) to modify facile data when revisions are required.

In both graphics and text, we organized the functional material so the user will not have to page back and forth to follow a particular sequence of events.

FICS, an RCA-developed set of data-management programs, stores functional analysis information such as function, element, and message data, and provides for its selective retrieval. These programs also can automatically generate graphic plots—functional-flow and element-interface diagrams—as well as printed reports.

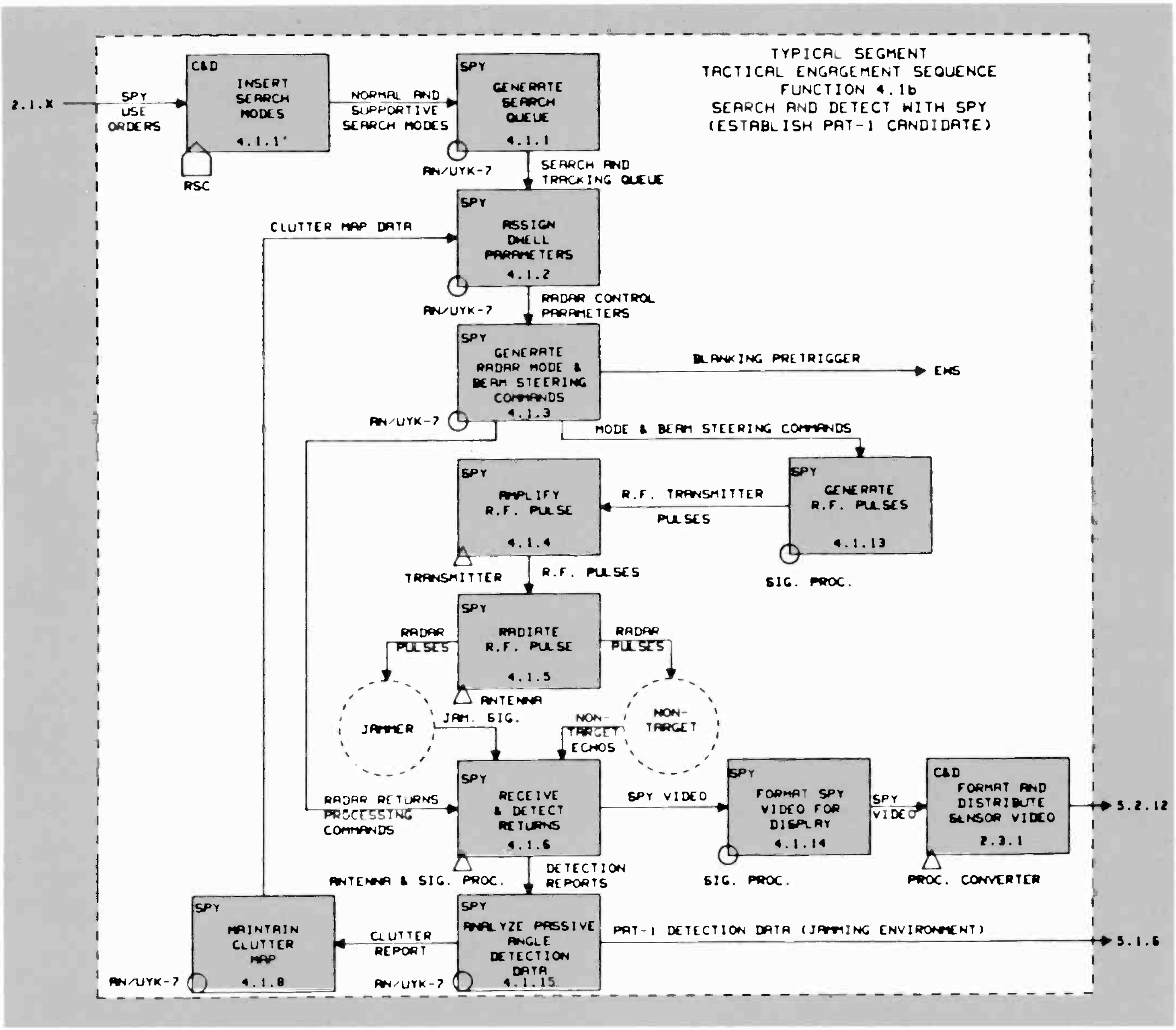

Fig. 1. *Typical segment of tactical-engagement sequence.*

Functional analysts also used other ground rules to improve the usefulness of the graphics and text material:

1. Functional information is combined or integrated where possible to reduce repetition across mission areas.
2. Information is cross-referenced so it can be related between sections of all CSTOM volumes, not just within the primary functional analysis volume.
3. There is a consistent level of detail in the document's functional descriptions.
4. The terms used in presenting the information are familiar to the Navy trainee. For example, engineering terms are avoided whenever possible, and more commonly used Navy operational terms are used.

These rules are a direct outgrowth of lessons we learned from earlier editions of the CSTOM. Comments from trainees and instructors on course-critique forms provided a valuable source of input. Recommendations from training working group and special committee forums, including the reports that they issued, provided other valuable information. Overall, trying to understand and satisfy the trainee's needs was the most useful, guiding objective.

Other applications of the TSD approach

Although the TSD in Fig. 1 depicts a typical sequence of tactical operations, it could describe maintenance functions just as well. For example, a system fault or failure message could provide the stimulus input that initiates certain maintenance functions such as detect, isolate, adjust, remove, replace, and repair. Obviously, we need to investigate the specifics of how to do this.

Another, but nonetheless important, use of the TSD technique is to highlight critical points or nodes in the system where a hardware or software failure or human error would most likely occur. The sequence threads help us focus on points in the sequence that show the effects of failure on the mission. Possible cases are:

1. A specific tactical mission cannot be performed because it requires the failed set of sensor/weapon system elements.
2. The mission loses part of its capability

Team effort

At Moorestown, Functional Analysis, Technical Manuals, and Training help to design and develop the AEGIS combat system. The U.S. Navy's most modern and sophisticated combat system, it was first installed in the AEGIS guided missile cruiser *USS Ticonderoga* (CG 47), the first in a new class of multimission ships. *Ticonderoga's* mission is to search, detect, and destroy attacking or hostile aircraft, missiles, surface ships, and submarines. Future ships of this class may also possess land attack or strike capabilities.

Functional Analysis, a system-engineering discipline, uses rigid rules and conventions and proven tools and techniques to analyze system functional requirements. The analysis starts with identifying top-level requirements. It ultimately defines detailed functional performance requirements and allocates them to people, equipment, and computer programs. Analyzing functions permits tracing system performance and design requirements specifications. It also imposes checks and balances during system development to ensure that the design incorporates all functions.

This activity produces a database for generating functional flow diagrams and descriptions, operational sequence and timing diagrams, and tactical sequence diagrams. The database and its products have traditionally supported design trade-off decisions and problem-solving efforts. Other system engineering and logistics activities also adapt functional analysis techniques and products to develop their own program documentation.

Technical Manuals, a logistics support project activity, develops and produces ACS technical manuals, such as the CSTOM, for the AEGIS combat system. Separate operations and maintenance manuals contain more detailed technical information. Basically, these technical manuals serve two major overlapping purposes. In the classroom, they are used for training Navy personnel, who will be assigned to operational billets aboard ship. The manuals are also used to orient newly assigned shipboard personnel and maintain proficiency in the design of the combat system for operation in its various tactical engagements.

Training, a logistics support project activity, formulates training plans, develops procedures, and administers both individual and team training at the CSED site to prospective Navy crew members who will eventually manage, operate, and maintain the combat system aboard ship. Just as the Technical Manuals activity relies heavily on System Engineering for basic source data, Training also depends on Technical Manuals for data and materials for training courses.

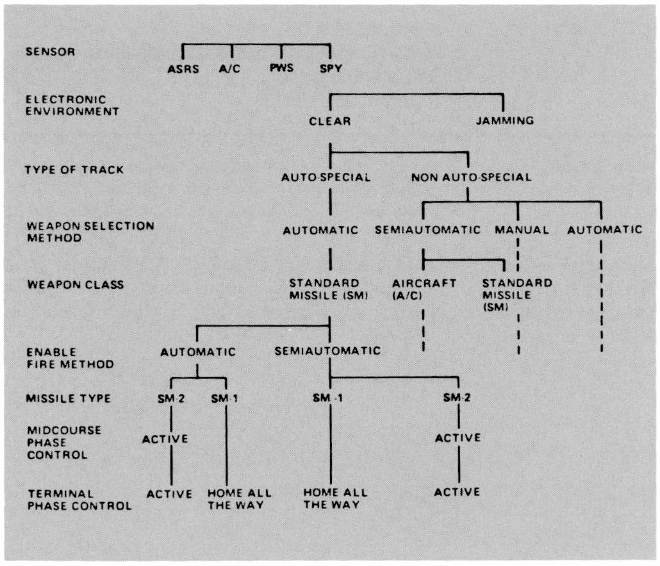

Fig. 2. *Sample logic-tree network (partial), AAW mission, SPY-SM engagement.*

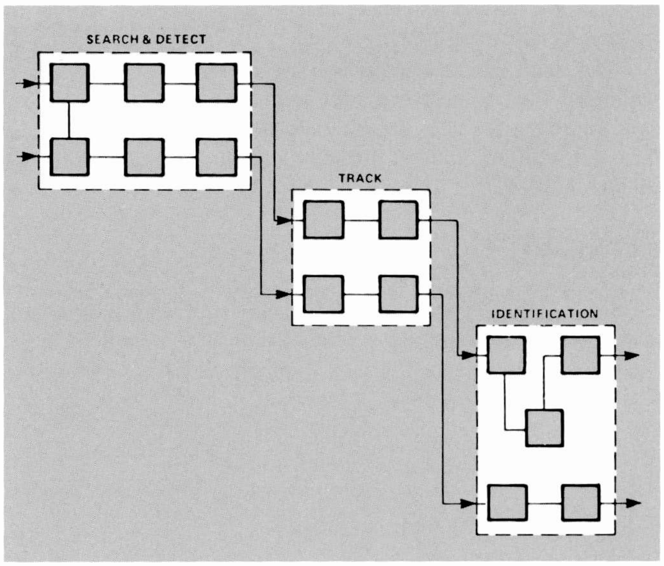

Fig. 3. *Functional linking of segments to form tactical-engagement sequences.*

because particular sensor/weapons system elements failed, but the mission objectives can still be met.

3. The mission objectives are not affected at all by the failure of the element to operate effectively.

For example, if the SPY-1 radar were to fail completely (the first type of failure), a target could not be engaged by a standard missile. If, however, only one of the two illuminators failed (the second type of failure), the mission could still be satisfactorily accomplished. Finally, if the interior communications system were to fail (the third kind of failure), the SPY-1 radar and standard missile elements could still perform their functions. The mission could be carried through to completion without interior communications.

Determining these critical points in advance will help the analyst structure the information to highlight these areas appropriately in the sequence for the instructor's and trainees' attention. In some cases, it also gives the trainee alternative courses of action should one course not be possible.

Conclusion

Since its introduction to the AEGIS program, Functional Analysis has been part of the system engineering design, definition, control, and auditing process. The Technical Manuals and Training activities have benefitted from the end products of functional analysis: functional flow diagrams and descriptions, operational sequence and timing diagrams, and TSDs. But until recently, we had not tailored these end products to the specific needs of training or manuals users. For example, although the flow diagrams and descriptions were basic engineering input to the system description in earlier versions of the CSTOM, little of the original input had been modified specifically for training or technical manuals applications. Now, we are reformatting and updating functional information to enhance its effectiveness as a learning tool.

TSDs, with their simplicity and logical, sequential, flowing structure, offer a significant improvement over the standard flow diagrams and descriptions currently found in the CSTOM.

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R.L. Layton
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Solid State Division

G. Dolny | C.F. Wheatly | H.R. Ronan
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Engineering News and Highlights

Brandinger is Staff VP, Engineering

Jay J. Brandinger, formerly Staff Vice-President, Systems Engineering, Electronic Products and Technology, has been named to succeed Howard Rosenthal as Staff Vice-President, Engineering. Dr. Brandinger continues to report to Roy H. Pollack, Executive Vice-President, Electronic Products and Technology. Mr. Rosenthal retired December 31, 1985.

Dr. Brandinger received a BEE from Cooper Union in 1951. In 1962 he received an MSEE from Rutgers University. He joined RCA Radio Research Laboratories in River Head, Long Island as a Member of the Technical Staff in 1951. From 1959 to 1974 he had responsibility for communication systems, display, and television systems research. In 1974 he was appointed Division Vice-President, Television Engineering, and was responsible for the development and design of all RCA Consumer Electronics television products. Between 1979 and 1984, Dr. Brandinger was Division Vice-President and General Manager, RCA SelectaVision VideoDisc Operations in Indianapolis.

Burris is Director of TEC

Frank E. Burris has been named Director of the Technical Excellence Center, succeeding William J. Underwood, who retired December 31, 1985. Dr. Burris will report to the Staff Vice-President, Engineering.

Dr. Burris was formerly Manager, Engineering Education, a position he has held since joining RCA in 1978. Before that he spent eleven years as a faculty member in EE at the University of Cincinnati and at Bradley University. He also has coop and summer experience with Honeywell and IBM. He holds BSEE, MS, and PhD degrees in Electrical Engineering from the University of Cincinnati.

He has been very active in the American Society for Engineering Education (ASEE), and he is currently Chairman of the College-Industry Council and Vice President, Institutional Councils.

Staff announcements

Advanced Technology Laboratories

Ronald A. Andrews, Staff Vice President, Advanced Technology Laboratories, announces his organization as follows: **Ronald A. Andrews,** Acting, Artificial Intelligence Laboratory; **Eric J. Braude,** Manager, Program Development—Intelligent Systems; **Leonard W. Braverman,** Manager, Marketing and Advanced Planning; **Gerald M. Claffie,** Manager, Optical Storage Laboratory; **Bruce A. Deresh,** Manager, Systems Engineering Laboratory; **William R. Ealy,** Manager, Image Technology Laboratory; **William F. Gehweiler,** Manager, Microelectronics Laboratory; **Paul B. Pierson,** Manager, Robotics Laboratory; **Jerry R. Richards,** Manager, Program Development—Computer Systems; **Harry Rosenthal,** Manager, Software Engineering Laboratory; **Robert J. Shirley,** Manager, Business Operations; **Frank B. Warren,** Manager, Device Technology Laboratory; and **John R. Welch,** Manager, Speech Technology Laboratory.

Aerospace and Defense

John D. Rittenhouse, Executive Vice President, Aerospace and Defense, has announced that **Charles A. Schmidt** will assume responsibility for the Government Communications Systems organization, which will consist of the following: Communication and Information Systems Division and Government Volume Production. Mr. Schmidt was proposed for election as a Group Vice President at the December 4, 1985 meeting of the RCA Board of Directors.

Astro-Electronics Division

John D. Rittenhouse, Executive Vice President, Aerospace and Defense announces the appointment of **Jack A. Frohbieter** as Division Vice President and General Manager, Astro-Electronics Division.

Jack A. Frohbieter, Division Vice President and General Manager, Astro-Electronics Division, announces the appointment of **Ricardo deBastos** as Division Vice President, Communications Satellite Programs.

Consumer Electronics Operations

Andrew G. Kolbeck, Manager, Materials Engineering and Development, announces his organization as follows: **Robert R. Russo,** Manager, Process Development Engineering; and **William L. Vroom,** Manager, Materials Testing and Specification.

Harry Anderson, Division Vice President, Program Management, announces the appointment of **Eric R. Bennett** as Manager, Productivity Improvement Projects.

Alfred L. Baker, Program Manager, CTC140, announces the organization of the Program Team for the CTC140 as follows: **David G. Campbell,** Administrator, Production Planning and Control; **John J. Drake,** Administrator, New Product Manufacturability; **James C. Marsh, Jr.,** Manager Project Engineering; **Abdul A. Pirani,** Quality Assurance/Reliability Engineer; and **Jacquelyn S. Taylor-Boggs,** Administrator, Procurement.

Terry J. Burns, General Manager, RCA Componentes, S.A. de C.V., announces his organization as follows: **Joe D. Andrews,** Manager, Materials; **Raul Neri,** Manager, Materials Distribution; **Hector Barrio,** Manager, Manufacturing and Plant Engineering; **Alfred Crager,** Manager, Test Engineering; **Jose Mijares,** Manager, Plant Engineering; **Hector Escobedo,** Manager, ACI Operations; **Hans P. Krolow,** Manager, ACI Operations; **Ernesto Ramirez,** Manager, ACI Manfacturing; **David E. King,** Manager, Financial Operations; **Enrique Perez,** Manager, Plant Quality Control; **Ariel Reyes,** Manager, Manufacturing Operations; **Francisco Casas,** Manager, Components Manufacturing; and **Raphael Quiroz,** Manager, Chassis Manufacturing.

Electronic Products and Technology

Roy H. Pollack, Executive Vice President, Electronic Products and Technology, announces the appointment of **Jay J. Brandinger** as Staff Vice President, Engineering. Dr. Brandinger replaces **Howard Rosenthal,** who retired on December 31, 1985.

Howard Rosenthal, Staff Vice President, Engineering, announce the appointment of **Frank E. Burris** as Director, Technical Excellence Center. Dr. Burris replaces **William J. Underwood,** who retired on December 31, 1985.

Microelectronics Center

James B. Feller, Staff Vice President, Technology, Aerospace and Defense announces that, effective immediately, the name of the **Solid State Technology Center** is changed to the **Microelectronics Center**. Also, the title of **John M. Herman III** is changed to Staff Vice President, Microelectronics Center. **Mr. Herman** will continue to report to the Staff Vice President, Technology.

John M. Herman III, Staff Vice President, Microelectronics Center, announces his organization as follows: **Ronald C. Bracken,** Director Integrated Circuit Design and Development; **Nancy-Lynn Broderick,** Manager, Administration and Training; **Richard J. Haug,** Director, Integrated Circuit Support Operations; **David S. Jacobson,** Director, Integrated Circuit Product Operations; **William D. Oldt,** Manager, Financial Operations; and **James E. Saultz,** Manager, Program Management.

James E. Saultz, Manager, Program Management, announces his organization as follows: **Richard Glicksman,** Manager, Marketing; **James E. Saultz,** Acting Manager, Program Management; and **Joseph J. Snack, Jr.,** Manager, Program Control and Contracts.

David S. Jacobson, Director, Integrated Circuit Product Operations, announces his organization as follows: **Richard H. Bergman,** Manager, Test Operation; **Robert A. Donnelly,** Manager, Production Control and Product Engineering; **David S. Jacobson,** Acting Manager, Wafer Fabrication Operation; and **Richard H. Zeien,** Manager, Package and Assembly Operation.

Richard J. Haug, Director, Integrated Circuit Support Operations, announces his organization as follows: **Robert A. Geshner,** Manager, Photomask Operation; **Fred J. Reiss,** Manager, Quality and Reliability Assurance; **Dennis R. Rickmon,** Manager, Computer and Design Services; and **Ronald A. White,** Manager, Equipment and Facilities Engineering.

Ronald C. Bracken, Director, Integrated Circuit Design and Development, announces his organization as follows: **Edward C. Douglas,** Manager, Advanced Integrated Circuit Development; **Albert Feller,** Manager, Integrated Circuit Computer Aided Design Development; **Michael A. Gianfagna,** Manager, Computer and Design Automation; and **Carl N. Puschak,** Manager, Integrated Circuit Design.

NBC

Michael Sherlock, Executive Vice President, Operations & Technical Services, announces the following appointments: **S. Merrill Weiss,** Director, Broadcast Systems Engineering; **Marilyn Altman,** Technical Manager, Network News; and **John Bennett III,** Technical Facilities Manager.

RCA Network Services

Lawrence T. Driscoll, Vice President and General Manager, RCA Network Services, Inc., announces his organization as follows: **Holmes Bailey,** Director, Automated Marketplace Services; **Wesley W. Bomm,** Director, Multi-Tenant and Consulting Services; **Leonard V. Dorrian,** Director, Corporate Telecommunications; **David M. Friedman,** Director, Business Networks and Systems Sales; **Terrance P. McGarty,** Director, Systems Engineering and Programs; and **Robert E. Ott,** Director, Finance.

Leonard V. Dorrian, Director, Corporate Telecommunications, announces his organization as follows: **Thomas G. Hammett,** Manager, Digital Planning; **Robert H. Hanford,** Manager, Network Operations; **Fred W. Huffman,** Director, Service Development; **David J. Marutiak,** Manager, Telecommunications Engineering; and **Richard D. Spencer,** Manager, Facilities and Services.

Robert H. Hanford, Manager, Network Operations, announces his organization as follows: **Terry A. Carmichael,** Manager, CTO; and **Joseph J. Sacks,** Manager, Systems Operations.

Video Display Monitor Products

Jack K. Sauter, Group Vice President, Consumer Electronics and Video Components, announces the appointment of **Lawrence J. Schipper** as Division Vice President and General Manager, Video Display Monitor Products.

Professional activities

1985 Ebers Award to Kosonocky

Walter F. Kosonocky, Fellow of the Technical Staff, RCA Laboratories, has received the 1985 J.J. Ebers Award for his "pioneering and innovative contributions to the development of charge-coupled devices and Schottky-Barrier infrared image sensors."

The J.J. Ebers Award is intended to foster progress in electron devices and to commemorate the life and activities of Jewell James Ebers. The award recognizes and honors accomplishment of unusual merit in the electron device field.

CISD Engineer of the Month

William Farrey has been named the CISD Engineer of the Month for August in recognition of his contribution to the development of sophisticated software that is presently being used on the GPCP, Watchmate, and Data Concentrator systems. Mr. Farrey's development of these programs for the IBM PC enables the user to enter wire lists directly into the computer, and the Cable Marker program automatically formats the "to" and "from" information.

Gore receives award

Douglas A. Gore, a Vehicle Test Systems Engineering Scientist at Automated Systems Division, has received the 1984 SAE Colwell Merit Award. The Society of Automotive Engineers presents this award for outstanding technical papers. Mr. Gore's paper, "Techniques for the Early Detection of Rolling Element Bearing Failures," was one of 15 selected from over 1200 presented at 1984 SAE meetings.

Meyer is Director of Old Crows

William T. Meyer, Manager, C^{3}I Requirements at the Automated Systems Division Field Office at Sierra Vista, Arizona, has been reelected as the Mountain Region Director on the National Board of Directors of the Association of Old Crows.

Ogden receives NSF award

Joan M. Ogden, a member of the Advanced Image Processing Research Group at RCA Laboratories, was one of 30 scientists and engineers awarded the NSF Visiting Professorship for Women in 1985. For the 1985/86 academic year, she will have a joint appointment in the Department of Mechanical and Aerospace Engineering and the Center for Energy and Environmental Studies at Princeton University. While on leave from RCA, she will be doing research on alternative energy systems and conservation, and will also be teaching a graduate-level course titled "Digital Image Processing and Applications."

Carver is "Man of the Year"

Tom Carver, right, receiving award from Harry Kline, General Chairman of AUTOTESTCON.

O.T. (Tom) Carver, Manager of Marketing and Planning Operations at Automated Systems Division, has been named "Man of the Year" by the IEEE International Automatic Testing Conference (AUTOTESTCON) in recognition of his contributions toward advancing the state of automatic test equipment. Mr. Carver is a Senior Member of the IEEE and served on the AUTOTESTCON Board of Directors for four years. He is presently on the Administrative Committee of the IEEE Instrumentation and Measurement Society.

This year marks the second year in a row that an RCA employee has received the AUTOTESTCON "Man of the Year" award. Last year's recipient was **Pat Toscano,** Program Manager at the RCA Oak Park facility in Bedford, Mass.

Vossen receives Nerken Award

John L. Vossen, Head of Thin Film Technology at RCA Laboratories, has been selected as the 1985 winner of the Albert Nerken Award "for his insightful contributions into the control of thin film deposition/etching processes and his application of these technologies to product development."

The Albert Nerken Award was established by Veeco Instruments, Inc. in 1984 to commemorate the efforts of Albert Nerken as a founding member of AVS and as a major contributor to the commercial development of instrumentation for high vacuum systems. The award winner must have made outstanding contributions to the solution of technological problems through the use of vacuum and surface science principles. The award is annual but is conferred only when a worthy candidate appears.

Third quarter PBG awards

Left to right: Foulks, Corneail, Stout, Amirrezvani, Vasudevan.

Left to right: Ford, Kleppinger, Cocozza.

S. Amirrezvani, L. Corneail, R. Foulks, M. Stout, and **K. Vasudevan** received the third quarter Palm Beach Gardens Technical Excellence Team Award for their effort in identifying CMOS1 yield inhibiting factors and implementing corrective action. This resulted in an increase in circuit probe yield of from 24 to 40 percent.

R. Kleppinger received the Individual Technical Excellence Award for developing an all-electronic process control chart system for wafer production processes. As a result, production operators routinely enter data and automatically obtain calculations and graphs.

Burlington awards

Automated Systems Division has announced the following Technical Excellence Awards:

Joseph M. Chirnitch—for contributions in implementing an objective computerized analysis for costing test program set (TPS) development. His work in evaluating the TPS development process, identifying key parameters, assessing their schedule and cost impact, and implementing their interactive effect on overall program cost was applied to the Navy F-18 TPS proposal. Furthermore, the labor allocation and schedule milestones resulting from his modeling are the cornerstones of the program plan. His initiative and resourcefulness at work and at home have contributed significantly to ASD's standing in the TPS development area.

Albert H. Frim, Wheeler X. Johnson, Michael E. Rockenhauser, and **Kevin J. Sullivan**—for contributions in redesigning the REMBASS Classifier using a VLSI design approach. The recent REMBASS production award required building over 1000 classifying sensor units that use seismic/acoustic inputs to differentiate between people, wheeled vehicles, and tracked vehicles. The classifier algorithm had been implemented on a complex board incorporating ceramic substrates and containing over 50 digital integrated circuits. The team took on a difficult task with a finite schedule, and met all goals the first time with an error-free design. They completed the analysis, design, fabrication, and exhaustive testing and statistical analysis resulting in a Value Engineering Change Proposal to the customer for over one million dollars, the largest VECP ever at Burlington. Their work allows a simpler, less expensive build and test, an effort that greatly enhances our position in the REMBASS program, and which already has been the basis for a business expansion into algorithm implementation for expendable mines.

David P. Edwards—for innovation and diligence in solving the hardware and software problems associated with the TCAC dual disk upgrade. The initial approach depended on MILTOPE, a subcontractor, to perform the necessary militarization effort on existing commercial designs. The subcontractor was unable to provide the solutions, and Dave, first working independently and then with the subcontractor, selected a different drive, modified it, and modified the chosen controller to work with the existing TCAC hardware. His effort involved a creative approach using a database manager to analyze test results, modifying an offset constant in the software driver, changing firmware in the drive, solving a head switch problem, and coordinating the drive-controller effort at MILTOPE.

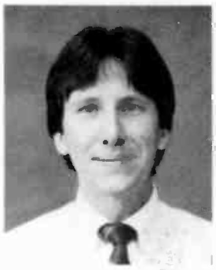
CE Technical Excellence Awards

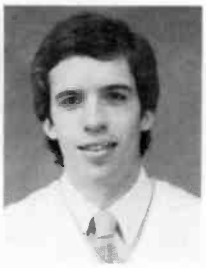

Milnes


Whipple


Wilson


Young


Bishop


Owens


Benton


Rupp

The recipients of the Consumer Electronics second quarter Technical Excellence Awards are:

James A. Milnes, Bruce A. Whipple, and **Danny L. Wilson**—for the successful implementation of the first drawingless CAD/CAM fast track tool procurement program for the number 68 mask mold. Extensive and thorough analysis of the mold design was done using new computer mold analysis software. This resulted in several mold designs before the mold was fabricated and tested. The development cycle was shortened by 27 weeks by overlapping tasks that had previously been performed sequentially. The mold tool was made directly from computer-generated tapes without the aid of paper prints. The fast track design cycle enables CE to be much more responsive to changing consumer tastes and will be used on more projects in the future.

James R. Young—for selecting and developing the solder reflow technique used to connect the high voltage windings to the high voltage diodes in the ROSE HVT. The technique has come to be known as fusion. Jim took the basic idea of solder reflow and turned it into a practical and reliable connection system.

Will Bishop and **Mark J. Owens**—for the mechanical design of the CRK-39 and CRK-40 remote transmitters. Both employed design creativity, technical innovation, and attention to detail in the production design cycle. Their thoroughness led to the best remote transmitter startup in recent Consumer Electronics history.

John E. Rupp—for the development of a form of flow charting for the instrument assembly process on the Automated Instrument Line. He also provided a means of evaluating labor impact as instrument redesigns were accomplished for cost reductions. The flow charts reduced the process documents to a form that was easier to understand and explain. Process steps requiring high labor content were clearly shown and "before" and "after" comparisons were conveniently made. The accompanying line balancing program allowed surplus labor to be reduced to a minimum.

Ronald E. Benton—for innovative design and support of the product cost reduction program. Ron suggested and developed the means to use square LEDs in a unique mounting scheme that eliminates the use of clear plastic light pipe lenses. The new design is less expensive and provides an improved display. He also suggested a keyboard redesign for the number 65 mask. Models were made to demonstrate both ideas. A third innovation involves a new kine grounding scheme for which a patent application has been made.

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