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**EVALUATION OF TECHNICAL FEASIBILITY OF CLOSED CYCLE
NON-EQUILIBRIUM MHD POWER GENERATION WITH DIRECT
COAL FIRING**

Final Report—Task 1

November 1981

Work Performed Under Contract No. AC01-78ET10818

**General Electric Company
King of Prussia, Pennsylvania**

U. S. DEPARTMENT OF ENERGY

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Evaluation of Technical Feasibility of Closed Cycle Non-Equilibrium MHD Power Generation with Direct Coal Firing

Final Report—Task I

November 1981

**Prepared for the
United States Department of Energy**

Under Contract No. DE-AC01-78-ET10818

**General Electric Company
Energy Systems Programs Department
King of Prussia, Pennsylvania 19101**

GENERAL ELECTRIC

GENERAL ELECTRIC

SPACE DIVISION
PHILADELPHIA

PROGRAM INFORMATION REQUEST/RELEASE

*CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV. LTR.
PIR NO. U	-1257	-MHD	-353	

*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED

FROM J. Rapp	TO Dr. T. Dellinger

DATE SENT 2/26/80	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.

SUBJECT	Users Manual for 2DEM - A 2-Dimensional Simulation Model for the Electromagnetic Properties of an MHD Channel

INFORMATION REQUESTED/RELEASED

2DEM (2-Dimensional Electromagnetic) is a purchased software package designed to determine the electromagnetic properties of an MHD (Magneto-hydrodynamic) channel based on user provided input parameters. The following provides information on the software aspects of that version of 2DEM currently installed on a PDP 11/34 computer.

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1.0 APPLICABLE DOCUMENTS

- [1] A User's Manual For the Magnetohydrodynamic Generator Channel Code, MHDCHN, Anthony J. Russo, January 1979, Sand 78-1260 (Sandia Laboratories).
- [2] RSX-11 M/M-PLUS Task Builder Manual, Digital Equipment Corporation, Copyright 1979.

2.0 HISTORY

The original Fortran version of 2DEM (2-Dimensional Electromagnetic), procured from Sandia Laboratories was written for and intended to be run on a CDC 6600 computer. As General Electric has no in-house CDC machine available, modifications were immediately made to the original version of 2DEM in an effort to make it compatible with General Electric's Honeywell Level 66 Fortran. Results of checkout runs from both Sandia and General Electric yielded comparable results.

In May 1978, another computer called the PDP 11/34 became available and with it came the question of its ability to run 2DEM. Initial study revealed the PDP 11/34 could handle the primary matter of concern, i.e., that being a problem size dealing with matrices sized less than or equal to (81×41) . With this, further work was contracted for to bring the version running on the Honeywell to the PDP 11/34.

As the PDP 11/34 is in reality a much smaller computer, it was found from the outset that the first hurdle would be to simply get the program to fit into available core memory. This problem was solved through extensive overlay efforts using a (17×21) problem size, without the use of virtual memory. These results were verified against a run of the same problem size done on the Honeywell. However, without using virtual memory, problems larger than (17×21) could not be considered. Thus 2DEM required further modification making use of virtual memory.

When the virtual version was complete, a new series of test runs were scheduled using various problem sizes and channel geometry configurations. The emphasis was on achieving comparable results from both the Honeywell and the PDP in all cases. The test runs did indeed result in comparable results and thus the transformation was considered a success.

Since modifications were made to 2DEM during the transformation, documentation describing these changes was felt necessary. Thus, the following describes the program 2DEM from a software viewpoint with emphasis on those portions of the software effected during the transformation from the Honeywell to the PDP 11/34.

3.0 PROGRAM DESCRIPTION

3.1 General

In studying the operation of an MHD channel there are two phenomena of particular interest to be considered. First of interest are the characterizing fluid properties of the electrically charged plasma in the channel. Secondly, by the concept of MHD, this flowing, electrically charged plasma is then placed in the presence of a magnetic field, giving rise to an electric potential and thus the electromagnetic properties of the channel become of interest. It is the later of these, the electromagnetic properties, that 2DEM is concerned with.

However, one cannot accurately describe the electromagnetic properties of the channel without introducing the fluid properties of the channel. In light of this, 2DEM has been designed to interact with a companion code, termed "the fluid code," providing data on the fluid properties of the channel plasma. At the time of this writing, such a fluid code does not exist for use in-house at General Electric. Therefore, velocity, temperature and pressure data (flow parameters) over the computational region must be acquired through mechanisms provided for within 2DEM or, if necessary, the code may be modified, providing an alternate means of specifying the parameters of interest.

The goals of 2DEM are to compute the current stream function (ψ) and electric potential in a cross-sectional plane of an expanding MHD channel. The procedure for making these calculations makes use of a pair of nested iteration loops. The inner loop of iterations accepts the flow parameters as provided and calculates PHI and PSI using conventional relaxation techniques. However, since the conductivity of the plasma is a function of plasma temperature as well as current density, which is initially unknown, the code arrives at its first approximation of stream function and electric potential having used only estimates of conductivity and temperature. At this point execution turns to the outer loop

where new temperatures and conductivities are computed based on results from the inner loop's calculation of stream function and electric potential. With these new parameters, execution returns to the inner loop to re-compute the stream function and electric potential. The number of times the outer loop is executed is determined by the user at run time via the input parameter IREPTL.

3.2 Software Layout

3.2.1 The Driver Routine

The driver routine, ELECSLV, is the hub of all program activity. That is, aside from providing the starting and ending points for program execution, ELECSLV is responsible for a host of other tasks, such as:

- o directing program execution based on input parameters,
- o calling subroutines to perform data manipulations and/or calculations,
- o establishing data arrays and common blocks passed between subroutines,
- o writing all output data (with the exception of subroutine PARSET which writes temperature as output data itself),
- o handling magnetic tape I/O (if this option is used),
- o establishing boundary conditions,
- o and performing convergence tests in support of iterative calculations.

Note that computations are nowhere mentioned in the above. Thus, ELECSLV is simply the director of all program activity and that which calls appropriate subroutines where computations are necessary.

Due to the enormity of ELECSLV, it immediately became apparent that an extensive overlaying effort would be necessary before 2DEM would run on the PDP 11/34. In general, when overlaying a program, one would like to "trim" the driver routine to those tasks absolutely necessary and make all remaining sections of code into subroutines. Thus, ELECSLV was partitioned into three sections; a much smaller driver routine still called ELECSLV and two subroutines named ELECA and ELECB.

Aside from changes made to the code responsible for I/O, modification of the original driver, ELECSLV, represents the most significant change made to 2DEM. The impact of creating ELECA and ELECB (which are the only subroutines not to be found in the original version of 2DEM) on the existing subroutines involved in most cases modifying common blocks of data to maintain data integrity between the new level of subroutines (ELECA and ELECB) and those already existing.

3.2.2 Overlay Structure and Subroutines

As mentioned previously, ELECA and ELECB were the only subroutines created during the effort to bring 2DEM to the PDP 11/34. Such is not the case with the filenames used to store the subroutines on the PDP 11/34. Therefore, for convenience, Appendix A provides a list of all the subroutines and the corresponding filenames under which they may be found on the PDP 11/34.

The concept of overlaying programs finds its use in situations where core memory requirements of the program exceeds the amount of core memory available on the machine in use. In essence, overlaying allows the user to use disc memory to store those files for which no room remains in available core. To do this, the user must generate somewhat of a map, listing those files on disc that are accessible from a given file currently in core memory. Although the necessity of this map may not be evident at this time, it will become clearer as the levels of overlaying are better understood. Such a map would point out, for instance, what is termed the root of the overlay where the root is that file always remaining in core during execution, i.e., the root is never placed in disc storage. In the case of 2DEM, the driver routine ELECSLV will assume the role of the root. It can now be seen that by making ELECSLV as small as possible, much of core remains available for swapping in disc files to support program execution.

The overlay map for 2DEM, Figure 1, shows a file named VMAIN as the root of the overlay structure (reference to Appendix A indicates ELECSLV is in the file

┌── ROOT ──┐ ┌── LEVEL 1 ──┐ ┌── LEVEL 2 ──┐

FIGURE 1. OVERLAY STRUCTURE FOR PROGRAM 2DEM

named VMAIN) with several outreaching branches terminating at what is called the first level of overlay. This indicates that any calls to subroutines occurring in VMAIN must reference a subroutine in the first level of overlay or, in this case, files VELECA, VELECB, or VDEMLB. Once the file containing the called subroutine is determined, that file is then automatically loaded into core and execution continues. Similarly, each of the files in the first level of overlay may (or may not) have branches extending from it to the third level of overlay. And as before, any subroutines calls made from core say in VELECB must find that subroutine in either VDEMCL or VDEMLD. The overlay structure for 2DEM found three layers of overlay sufficient but in general, this might go on several more times [2]. It should be noted that calls to subroutines made in level n of an overlay scheme must find the searched for subroutine either in overlay level $n + 1$ or in the root.

To implement the automatic loading of the overlay segments so casually referred to requires the use of an additional file, the overlay file. This file is to contain a copy of the already generated overlay map but this time in a format understandable by the task builder of the PDP 11/34 system. Applicable document [2] provides all the necessary details of generating such an overlay file. A copy of the file presently being used to execute 2DEM on the PDP 11/34 is provided in Appendix B.

3.2.3 Program Flow

Program execution can be thought of in terms of a pair of nested loops referred to as the outer and inner loops (Figures 2 and 3, respectively). The outer loop commences with the driver routine, ELECSLV, by gathering all the user provided input data. From here, execution proceeds through a series of subroutine calls to ELECA, PARSET, STD2D, CONNECT, etc....., all contributing in the process of determining the initial values for PHI, PSI, SIG and T, the electric

FIGURE 2. SUBROUTINE FLOW SUMMARY - OUTER LOOP

FIGURE 3. SUBROUTINE FLOW SUMMARY - INNER LOOP

potential, current stream function, conductivity and temperature matrices, respectively. With these matrices properly initialized to represent the problem, subroutine ELECB is called. Within ELECB is the inner loop. Here iterative techniques are used to calculate new values for the PHI, PSI and SIG matrices using those initial values provided by the outer loop. When either the convergence criterion is met or an iteration limit exceeded, execution returns to the outer loop. The user provided input parameter IREPTL is now examined. If IREPTL is input as 1 (one), then the final values of PHI, PSI and SIG are printed and execution terminates. If IREPTL is input as greater than 1, then execution returns to the point in ELECSLV where PARSET is called, this time using those previously calculated values of PHI, PSI and SIG to arrive at a new value of temperature at each mesh point. Using the newly calculated temperature values and the latest approximations of PHI, PSI and SIG, the inner loop begins again with the task of computing better approximations to PHI, PSI and SIG. In this manner succeeding better approximations to PHI, PSI and SIG are determined.

3.2.4 Virtual Memory

The concept of virtual memory simply involves the separating of large data arrays from program op code. Under conventional circumstances on the PDP 11/34, the executable load module including all op code and data arrays cannot exceed approximately 32K words of storage. However, with a problem size of (81 x 41) mesh points, each data array alone involves 6K words of core storage. Better than half of the allowed 32K words would be consumed in storing data. Thus the data arrays are declared virtual thereby separating them from the op code. The result is, the 32K words can be used to store program op code and data arrays become relocated in computer memory.

The use of virtual seems relatively straightforward at first with the only impact being at the point where the array sizes are declared. Unfortunately, the

version of Fortran being executed on the PDP 11/34 does not allow arrays placed in virtual to appear in either COMMON or EQUIVALENCE declarations. The solution to the dilemma is to remove all those arrays in virtual from COMMON or EQUIVALENCE statements and place them in the parameter list of the necessary subroutine calls where that array is used. Hence, one will notice that subroutine calls to subroutines, such as PARSET have a modified list of arguments to include those virtual arrays necessary to the subroutine.

With the overlay structure currently being used (see Figure 1), and an (81 x 41) problem size, program op code consumes less than 28,416 words of core and virtual arrays consume 53,760 words. Note that program op code assumes less than 32K words.

4.0 INPUT/OUTPUT

4.1 Input Files

One of the first tasks undertaken in the driver routine is the inputting of program variables and constants. As the Sandia User's Manual indicates [1], these parameters are submitted via punched cards. However, in the PDP 11/34 version of 2DEM these parameters are loaded into a file named FOR004.DAT and read sequentially by a series of READ statements at the beginning of the driver routine. As the Sandia User's Manual describes each of the parameters in detail, further discussion is unnecessary in this document. Note that since these input parameters are in an external file, they may be altered without re-compiling or task-building 2DEM.

In addition to those parameters provided in FOR004.DAT, 2DEM has two optional input modes selected by inputting appropriate parameter values via FOR004.DAT. These two optional modes may be described as follows:

As indicated previously, 2DEM performs its calculations using an iterative method. If the user has a copy of a previous output (from a separate run of 2DEM) file, he may submit this file as input data and the iterative procedure will pick up where it left off. Variable NREPT must equal zero to activate this input mode. This mechanism still exists in the program, however, a check-out procedure on the PDP 11/34 was not executed to verify its operation. Caution is advised therefore in using this mode of input.

The second option of input mode provides a means of supplying geometric and thermodynamic data instead of using the default values otherwise provided. In this case variable NCUPLE must be equal to one. Again, this input mode was not verified for proper operation on the PDP 11/34 and thus caution is advised in its use.

4.2 Output Files

The only output mode exercised on the PDP 11/34 is a printed output file that lists by name all the input parameters, the matrices SIG, PHI and PSI after each pass through the outer loop, as well as other data describing electromagnetic properties of the MHD channel.

There are additional output modes available but as yet are unexercised in the PDP 11/34 version of 2DEM. For a detailed explanation, refer to the Sandia User's Manual [1].

Additional output files are generated by 2DEM during execution and used primarily as scratch files. They are mentioned only so the user is aware that additional transparent files are generated during program execution.

5.0 PROGRAM EXECUTION

5.1 Backup Files and Program Size

Due to a limitation of the Fortran available on the PDP 11/34, changing the problem size is not as easy as changing an input parameter. That is, it must be done manually by altering the dimension statements in the source code. As this procedure has great potential for serious, time consuming mistakes, an alternate means has been implemented.

In addition to the source code files, another set of nearly identical files called backup files have been created. In the backup files, each occurrence of a parameter effected by a change in problem size has a variable inserted in its place. For instance, what would appear as PSI (81 x 41) in the source code, appears as PSI (ZZX, ZZY) in the backup files. ZZX and ZZY being unique character strings, the editor can now be used to search out every occurrence of ZZX and replace it with the X-dimension of the problem size desired. Similarly for ZZY.

One of the text editors installed on the PDP 11/34, called TECO, lends itself especially well to making just this type of change throughout a file. For convenience, the procedure to change the problem size to (81 x 41) is outlined below (note: \$ character is generated by depressing the ESCAPE key):

```
> TECO VMAIN.BAK
* <FNZZX$81$ØTT>$ $           ;changes all ZZX's to 81
.
: list of all changes made
.
* EX$ $                       ;exits and names file VMAIN.BAK;2

> TECO VMAIN.BAK
* <FNZZY$41$ØTT>$ $           ;changes all ZZY's to 41
.
: list of all changes made
.
* EX$ $                       ;exits and names file VMAIN.BAK;3
```


At this point, a file named VMAIN.BAK;3 will contain the version with (81 x 41) inserted in the proper places. To be sure take a look at the first few lines with the editor and when convinced this is the one, proceed.

Now we want to change the name of the file to make it look like a source code file rather than a backup file. To do this use:

> PIP VMAIN.FTN = VMAIN.BAK;3/RE/NV.

This will rename the file VMAIN.BAK;3 to VMAIN.FTN and give it the next sequential version number. With this completed, return now to the file VMAIN.BAK;2 and delete it using

> PIP VMAIN.BAK;2/DE.

Note that VMAIN.BAK;3 no longer exists as it was renamed.

The procedure above must be carried out for all files containing problem size information. These files are:

VMAIN.BAK
VELECA.BAK
VELECB.BAK
VDEMLB.BAK
VDEMLC.BAK
VDEMLD.BAK

Although this seems a quite laborious task just to change the problem size it is actually rather foolproof. At worst, a file may be destroyed while editing due to an operator error or system crash, in which case another nearly identical file always remains to recover with.

5.2 Compilation and Taskbuilding

The PDP 11/34 requires several steps of preparation to generate executable code and are known in sequence as compilation (generates object code from source code) and taskbuilding (generates an executable task image from object code). This paragraph describes how to carry out these preparatory steps.

Two items are necessary. First and most obvious are the source code files. Secondly, a map of the overlay structure must be available in what is called an overlay file. The overlay file will be considered in more detail when taskbuilding is discussed.

The compilation process accepts as input a source code file (e.g., VMAIN.FTN, VELECA.FTN) and generates as output an object code file (e.g., VMAIN.OBJ, VELECA.OBJ). The following is a typical example of the sequence of commands necessary to compile a file.

> FOR filename.OBJ = filename.FTN

Once all source code files have been compiled, taskbuilding can begin. Although in general an overlay map is not necessary to taskbuild, for the reasons discussed in paragraph 3.2.2, 2DEM must use an overlay file. Basically, the overlay file establishes when each file may be permitted into core memory. For 2DEM, the overlay file is named VDELOVR.ODL, a listing of which is provided in Appendix B. Applicable document [2] discusses the rules for generating an overlay file.

The following is a typical command to generate an executable task named 2DEM:

> TKB 2DEM/FP[,LP] = VDELOVR/MP

To translate, this command says to generate a task named 2DEM by referring to the overlay file VDELOVR for directions. The taskbuilder then looks to VDELOVR.ODL and proceeds to build the task with the understanding that the object files of those files named in VDELOVR.ODL exist. The /FP in the command string is simply a precaution that says to save the status of the floating point processor should 2DEM's execution ever be interrupted for a higher priority program. The bracketed quantity is optional and when inserted will provide a copy of the taskbuilder's statistics on 2DEM. The /MP denotes VDELOVR as an overlay file. The resulting executable task will appear as a file named 2DEM.TSK.

6.0 LIMITATIONS AND RUN TIME ESTIMATES

As has been mentioned previously, all the I/O options intended for 2DEM in its original version have not been exercised in that version on the PDP 11/34. As a result, caution is strongly recommended in the use of tape I/O options.

To provide some level of confidence in the transformation to the PDP 11/34 several simple cases were run both on the Honeywell facility and the PDP 11/34.

The test cases cited were:

- (17 x 21), NELP = 1, IEXT = 1, PDP 11/34 Run Time ~ 1 hour
- (81 x 41), NELP = 5, IEXT = 0, PDP 11/34 Run Time ~ 5 hours
- (81 x 41), NELP = 5, IEXT = 1, PDP 11/34 Run Time ~ 10 hours

Results proved identical to three significant places and thus are acceptable.

2DEM generates three files that appear transparent to the user on I/O channels 1, 2 and 3. These files are scratch files for intermediate calculations and appear under the names FOR001.DAT, FOR002.DAT and FOR003.DAT. The user should delete these following each run as they rapidly consume available disc space.

APPENDIX A. FILENAMES AND THEIR CORRESPONDING CONTENTS
(PDP 11/34 VERSION)

FILENAME	CONTENTS	FILENAME	CONTENTS
VMAIN.FTN	ELECSLV* READEC WRITEC	VKINDD.FTN	EVS2
VELECA.FTN	ELECA	VKINDE.FTN	EVSC0 EVSC1
VELECB.FTN	ELECB	VKINDF.FTN	EVSC2 EVSC3
VDEMLB.FTN	STD2D PARSET	VKINDG.FTN	EVSH2 EVSH3 SSN2 SSC0 SSC3 SSH3 RLXN2 VVN2C0
VDEMLC.FTN	COEF CONNECT PS1SOLV	VKINDH.FTN	VVN2C3 RLXC0 VVCON2 VVCOC3 VVC3N2 VVC3C0 VVC3C2 RLXC2 MTINV
VDEMLD.FTN	PHISOLV CUFT1 CUFT2 LINB TLU1	FOR004.DAT	INPUT DATA
VKINDA.FTN	ELECN FNCT TAB	FOR001.DAT	SCRATCH DATA FILE
VKINDB.FTN	FNCT GAMF FNCT CLPLY	FOR002.DAT	SCRATCH DATA FILE
VKINDC.FTN	BLOCK DATA GHTABL TABUP	FOR002.DAT	SCRATCH DATA FILE

*-This is the driver routine; all other are subroutines or functions.

APPENDIX B. OVERLAY FILE VDELOVR.ODL

The following is a listing of VDELOVR.ODL:

```
A:      .FCTR      VMAIN
B:      .FCTR      VELECA
C:      .FCTR      VDEMLB
D:      .FCTR      VKINDA
E:      .FCTR      VKINDB
F:      .FCTR      VKINDC
G:      .FCTR      VKINDD
H:      .FCTR      VKINDE
I:      .FCTR      VKINDF
J:      .FCTR      VKINDG
K:      .FCTR      VKINDH
L:      .FCTR      VELECB
M:      .FCTR      VDEMLC
N:      .FCTR      VDEMLD
        .ROOT      A-*(B,C-(D,E,F,G,H,I,J,K),L-(M,N))
        .END
```

For an explanation on how to generate an overlay file, reference applicable document [2].

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TECHNION REPORT 09-020

March 1, 1979

PERFORMANCE ESTIMATES FOR A NOMINAL 30 MW ARGON
SEGMENTED ARC HEATER.

FINAL REPORT

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ACKNOWLEDGEMENTS:

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2. PHYSICAL PROCESSES
3. ARC HEATER CONFIGURATION
4. PERFORMANCE ESTIMATES
5. DESIGN CONSIDERATIONS
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MHD CHANNEL

1.0 INTRODUCTION

Techniques of performance predictions for high-power arc heaters are still being developed along several relatively independent paths. A considerable amount of empiricism is still required for even the most sophisticated approach, that of stepwise (axial) solving a simplified set of the magnetohydrodynamic equations, usually in axisymmetric form. (Refs. 1,2,3, and 4.) Several very important aspects of the problem are lost by this technique:

- the influence of the circulation in the gas upon the mass flux vector must be neglected since this depends upon the exit geometry and exit gas flow parameters.
- some initial ($z=0$) radial distribution of mass flux and enthalpy must be postulated and this can influence the solution some distance downstream, giving a great deal of indeterminacy in the initial heating.
- some turbulence model must be postulated and one or more arbitrary constants in this model must be evaluated by empirical means.

In the long run, some extension of this "differential" approach to solving the problem will undoubtedly prove the most reliable technique of predicting arc heater performance. In the meantime, some important aspects in the design of arc heaters need theoretical and/or analytic guidance.

Given the power input to a heater and the maximum stagnation pressure in the heater, the following criteria can be used to evaluate the performance:

- the mass average enthalpy should be as high as possible.
- the thermal efficiency should be as high as possible.
- the radial enthalpy profile should be as flat as possible.
- the impurity level in the exhaust jet should be as low as possible.
- the free-stream turbulence in the exhaust jet must be as low as possible.

In attemptint to evaluate these parameters, the following questions

are relevant:

- does a fixed length arc, or a variable length arc give best performance, or does each have superior performance in selected regimes?
- is there an optimum value of the ratio of the pressure vessel diameter to the throat diameter, and if so, what is it?
- is there an optimum value of the circulation in the gas?
- how much influence does the configuration of the arc attachment at the electrodes have on the overall heater performance?
- do critical arc lengths exist and, if so, what is their significance?
- what parameters of the heater design control the free-stream turbulence level in the exhaust jet?
- what is the optimum method of distributing the mass injection along the length of the heater?

The approach taken in this study is that the arc is a "porous", constant diameter, constant temperature, heat source. This permits performance to be evaluated in terms of adjustable heat transfer coefficients, both at the "edge" of the arc and at the wall. It also permits us to empirically introduce into the analysis the influence of the circulation level in the gas. Other effects, such as the laminarization of the gas flow through the arc proper, can also be introduced.

In any study that attempts to predict arc heater performance, it is essential that all of the possible physical processes which might occur be included in the analysis. For this reason, correlation formulae derived from data on one heater may work well for extrapolating and interpolating performance for that heater but must be used with reservation in any other configuration. In the approach used in this study, it is hoped that the two "adjustable" heat transfer coefficients can be eventually expressed in terms of equations describing physical processes that are detected and quantitatively assessed in tests.

2. PHYSICAL PROCESSES

The physical processes occurring in the argon arc heater are essentially the same as those occurring in the air arc heaters. The Reynolds' number, however, is much higher in this projected heater than in any existing air heaters.

Reynolds number of heated argon (2000⁰K):

$$\begin{aligned} &= \frac{4 \dot{m}}{\pi \mu_c d_c} = \frac{4 \times 20}{\pi \times .878 \times 10^{-4} \times .254} \\ &= 1.14 \times 10^6 \end{aligned}$$

Reynolds number of argon gas inside the arc (12,000⁰K):

$$\begin{aligned} &= \frac{4 \dot{m}}{\pi \mu_A d_A} \left(\frac{d_A}{d_C} \right)^2 = \frac{4 \dot{m}}{\pi \mu_C d_C} \frac{u_C}{u_A} \frac{d_A}{d_C} \\ &= 1.14 \times 10^6 \times \frac{.878}{3.0} \times .1 \\ &= 33,400. \end{aligned}$$

The Reynolds' number of the gas in the arc should be high enough to ensure that no laminarization of the flow occurs in the arc.

A significant difference between argon and air in an arc heater occurs in the electric field that can be supported. Measured data indicates that in identical configurations, the electric field in an air arc is larger than that in an argon arc by about a factor of 2.5. Simultaneously, the arc "radius" in argon is larger by about a factor of the square root of 2.5, or 1.6. This results from the difference in the transport coefficients between the two gases, especially the atomic diffusion term in the thermal conductivity.

Turbulence can be injected into the heater with the gas (free stream

turbulence level) or it can be generated inside the heater. An effort to indicate turbulence interaction effects and regimes in arc heaters is made in Fig.1. The gas is injected with a strong circulation, since circulation in the gas is always necessary to stabilize the arc near the centerline. This schematic of the processes in high Reynolds' number arc heaters is highly speculative, being based on intuition as well as on documented observations made at various laboratories in both the USA and the USSR. The following processes can be identified:

- 1) The circulation in the gas causes a radial pressure drop, which turn induces a reverse flow through the boundary layer of the rear electrode.
- 2) If the Reynolds' number of the flow is high enough, this boundary layer flow becomes turbulent.
- 3) This turbulent boundary layer flow breaks away from the center of the rear wall and forms a vortex core 6 with an axial forward flow carrying all of the gas that flowed backward through the boundary layer.
- 4) The gas in the rear electrode is spinning with a constant circulation level and is relatively stagnant, i.e. it does not participate in any axial or radial motion.
- 5) The electrical discharge crosses from the center of the rear electrode to the wall where it attaches to the electrode. The "spoke" and attachment point spin due to drag from the gas and $j \times B_z$ electromagnetic forces. The axial attachment location is determined by the balance of backward and forward drag forces on the arc due to the axial flows as well as the $j \times B$ forces from the arc current interaction with the applied solenoidal magnetic field which "destabilize" the discharge.
- 6) The axial forward flow is concentrated in the vortex core which has a diameter slightly smaller than the exit orifice diameter. This flow is highly turbulent and has a high vorticity content.
- 7) The vortex core and the arc "compete" for the center-line position. They cannot coalesce at this axial position since the axial flow in the vortex core must first be heated to the arc temperature by power dissipated in the arc. Three possible interaction configurations are possible:
  - a. The vortex core retains the center-line position and the arc forms a helix around it.
  - b. The arc assumes the center-line position and the vortex core forms a helix around it.
  - c. Neither takes the center-line position, the arc and the vortex core forming a "double" helix.

- 8) Some of the energy dissipated in the electrical discharge is conducted and/or convected into the vortex core; the arc and the vortex core start to coalesce and stabilize along the center-line.
- 9) The relative turbulence level, i.e. $\frac{\text{turbulent energy}}{\text{flow kinetic energy}} = \frac{\rho v_t^2}{\rho w^2}$ becomes much lower than at 3 and 6 for two reasons:
  - a) The gas velocity $\bar{w}$ is much higher since ρ is much lower and ρw must remain relatively constant.
  - b) The turbulence decays rapidly due to the low density of the gas. The gas flow in the vortex core tends to become laminarized by the interaction with the arc.
- 10) A strong shear layer develops between the external gas and the gas flow in the vortex core. Turbulence is generated in this shear layer and tends to spread radially outward.
- 11) The "arc diameter" (d_A) and the vortex core diameter (d_V) will, in general, be different).
- 12) That fraction of the injected gas which did not flow backward 1, flows forward through the boundary layer on the constrictor wall. If the Reynolds' number is high enough, this boundary layer flow becomes turbulent. The turbulent boundary layer thickness grows radially inward.
- 13) The turbulent boundary layer from the walls has grown thick enough to meet the centrally located vortex core and/or arc. Several possibilities can result from this interaction:
  - a) The arc can be buffeted by the turbulent flow so that it moves randomly off the center-line for some distance downstream of the interaction position but maintains an approximate helical shape.
  - b) The buffetting can be severe enough to destabilize the arc, driving it to the wall. If the wall is the downstream electrode, then the arc is likely to attach to the electrode at this axial position. If the wall is a segmented constrictor, then high local heat fluxes can be expected. If the segmentation is not fine enough, then local segment to segment arcing will occur. The electric field of the discharge will fluctuate, but on the average be much higher in this region than elsewhere.
- 14) Some turbulent mixing of gas from the core with gas from the downstream boundary layer will occur. Decay of turbulence in this mixed gas will occur and centripetal forces will drive the arc back to the center-line. A long region in which "turbulent" heating occurs will now occur. Also, in this region, turbulent heat conduction to the walls will take place. This region will

extend until asymptotic velocity ($\frac{w}{w_c}$) and temperature
 $\frac{T - T_w}{T_c - T_w}$ profiles develop.

At this point in time, any attempt to analyze such a complicated system as pictured above is impracticable. Also, observations on high pressure arc heaters indicate that there are processes at work that tend to average out the electric field and the wall flux over the length of the heater. In particular, in fixed length arc heaters, it appears feasible to extend the arc length beyond the "natural" arc length 5 to 13 in the schematic Fig.1 and not suffer severe wall heat flux or intersegment arcing in this region. This is most likely accomplished by introducing the gas with a moderate circulation and as high a free-stream turbulence level as is feasible. Alternatively, axial similarity can be promoted by introducing the gas through the walls over the whole length of the heater. This would tend to eliminate the potentially catastrophic interaction 13 and also reduce the region upstream of this point to zero length.

In the calculations used for the analysis in this study, all processes were averaged out over the length of the arc heater. Attempts to include effects due to turbulence and forced convection are made within the framework of conventional turbulent heat transfer processes with adjustable coefficients, which are then evaluated to obtain a "best" fit to the data.

The stability of the discharge and the attachment points is an independent consideration that is intimately connected with the circulation level in the gas. A schematic of the circulation "corridor" that arc heaters must operate in is shown in Fig.2. The accompanying table describes the mechanisms that could lead to unstable operation. The quasi-empirical corridor for stable operation in air can be applied immediately to arcs in argon.

3. ARC HEATER CONFIGURATION

A typical arc heater configuration is shown in Fig.3. The components are defined in the accompanying table.

Alternate electrode configurations have been used in some arc heaters. Specifically, conical cathodes have been used in the rear instead of the ring electrodes. This results in an arc heater with "normal" polarity. In this polarity alternate anodes have been used, some of which consisted of multiple rods projecting through the wall into the flow.

For the arc heater under consideration, we would recommend a configuration similar to that shown in Fig. 4 or 5, with the cathode rings at the front. Spinning of the attachment points on the electrodes is accomplished by a combination of spin in the gas and magnetic spin. The latter is due to local field coils buried in the electrode structure.

4. PERFORMANCE ESTIMATES

The transport properties for argon were entered into the existing program for air arc heaters. These were obtained from References 1 thru 15. The heat transfer coefficients between the arc and the gas ($1 + \beta_A$) and the gas and the wall ($1 + \beta_W$) were then adjusted to match the few existing data points for argon arcs, References 16 & 17.

To obtain performance estimates, the geometrical arc heater parameters together with the mass flow rate and the arc current are entered into the computer. A data input strip showing typical input data is shown in Table 2. The program then computes the arc voltage, chamber stagnation pressure, thermal efficiency, and average gas temperature at the throat for a number of arc temperatures. Each solution is a "possible" operating condition. However, the "most probable" operating condition is that of minimum potential.

Table 2+3 presents some "fine tuned" performance estimates and design parameters for operation at 2000 and 4000 amperes and at nominal pressures of 10 and 25 atmospheres.

A program listing is presented at the back of this report. This program is written for an HP9825A Minicomputer with printer HP9871A.

5. DESIGN CONSIDERATIONS

A segmented arc heater facility is best suited to match the requirements of heating the quantity of argon specified to the desired temperature. Schematics of such a facility are shown in Figs.3 & 5. A photograph of the ARO-HEAT facility is shown in Fig.6.

The length and diameter of the projected heater are larger than those for any existing facility, hence some preliminary calculations have been made on the heat transfer and stresses in the structure. The wall heat flux rate to the internal diameter of the segments is calculated to be between 300 and 100 watts/cm². This is much lower than occurs in HEAT and other constricted arcs where segments are designed for up to 10,000 watts/cm². The low heat flux rate permits the use of low pressure cooling water and permits options on the material to be used for the segments. The low electric field strength also permits the use of relatively wide segments. All of these considerations indicate flexibility in the design of the heater components which can be reflected in reliable, relatively inexpensive components.

On the basis of the preliminary design calculations to date, no serious difficulties are anticipated in the design and fabrication of structural and heat transfer components.

6. COUPLING OF THE ARC HEATER TO THE MHD CHANNEL

There are three possible problem areas associated with the injection of the arc-heated argon into the MHD channel. These are:

- i) The magnetic field from the channel may interact with the arc in the heater in such a manner as to cause wall arcing in the heater.
- ii) The "optimum" arc heater should perhaps be operated at a pressure level much higher than required in the MHD channel.
- iii) The transition region between the end of the arc and the MHD channel inlet must be "smooth" and short enough to ensure that the average gas temperature is not lowered below the required injection temperature.

An estimate of the transverse magnetic field that is permissible without destabilizing the arc is

$$B \leq \frac{(\rho - \rho_A) v_A^2 \delta}{I}$$

where B = minimum permissible transverse magnetic field

$(\rho - \rho_A)$ = density difference between the ambient gas and the arc gas

= $(3 - .5) = 2.5 \text{ kgm/m}^3$ for 10 atm. pressure

v_A = gas spin velocity at the edge of the arc
 $\cong 100 \text{ m/sec.}$

δ = acceptable displacement of arc off of centerline
 $\cong 3 \text{ millimeters}$

I = arc current
= 2000 amps or 4000 amps.

Using the values quoted above,

If I = 2000 amps,

$$B \leq \frac{2.5 \times 100^2 \times .003}{2000}$$

$$\leq .0375 \text{ Tesla}$$

$$\leq 375 \text{ gauss.}$$

If I = 4000 amps,

$$B \leq 188 \text{ gauss.}$$

These requirements are probably conservative.

The design of the transition region can be tailored to meet this magnetic field requirement by increasing distance, or by the judicious use of shielding, or by some combination of these.

In optimizing for all three requirements given above, several options should be considered:

- 1) Operate the heater at high pressure (25atm) using a throat diameter of about 4 in. Expand the flow to Mach 3-4 in a diverging nozzle, and then shock it back to subsonic flow in an inefficient diffuser to obtain a stagnation pressure of about 10 atmospheres.
- 2) Use a long, supersonic, slowly diverging nozzle between the arc heater and the MHD channel.

Other options may be available, depending on the actual distance that must be used between the exit of the arc heater and the MHD channel

(A COMPLETE DESCRIPTION OF THIS FIGURE IS GIVEN IN THE TEXT)

FIGURE 1. THE INTERACTION SCHEME BETWEEN AN ARC AND TURBULENT GAS FLOW IN A LONG TUBE.

FIGURE 2

SCHEMATIC OF "LOCAL CIRCULATION CORRIDOR"
IN AN ARC HEATER

TABLE 1

<u>POSITION</u>	<u>PROBLEM IN CIRCULATION LEVEL</u>		<u>PHENOMENON</u>	<u>SYMPTOM</u>	<u>FIX</u>
①	Γ_{av}	Γ_{min}	Arc not destabilized at upstream electrode rings.	Arc goes to back plate; intersegment arcing between upstream electrode and backplate.	1. Increase magnetic field strength at electrode. 2. Inject gas with lower circulation level. 3. Inject more gas through the constrictor segments.
②	Γ_{av}	Γ_{max}	Recirculation cell set up in upstream electrode and constrictor segments.	Intersegment arcing between upstream constrictor segments and upstream electrodes.	1. Inject gas with lower circulation level. 2. Inject more gas through the constrictor segments.
B-14 ③	Γ_{av}	Γ_{min}	Local high electric field due to arc-boundary layer interaction raises value of Γ_{min} .	Intersegment arcing in the downstream end of the constrictor.	1. Distribute more gas injection through the constrictor segments.
④	Γ_{av}	Γ_{min}	Gas heating decreases gas density, increasing Γ_{min} .	Intersegment arcing in the downstream end of the constrictor.	1. Increase circulation level in injected gas.
⑤	Γ_{av}	Γ_{min}	Arc not destabilized at downstream electrode ring.	Arc goes to nozzle; intersegment arcing between downstream electrode and nozzle.	1. Increase magnetic field strength at electrode. 2. Reduce circulation level in injected gas. 3. Increase electrode diameter.
⑥	Γ_{av}	Γ_{max}	Recirculation cell established in downstream electrode and constrictor segments.	Intersegment arcing between downstream constrictor segments and downstream electrode.	1. Reduce circulation level of injected gas. 2. Increase the throat diameter.

$L_1 =$ _____
 $L_2 =$ _____
 $L_3 =$ _____
 $L_4 =$ _____
 $L_5 =$ _____
 $L_6 =$ _____

$d_{RE} =$ _____
 $d_C =$ _____
 $d_{FE} =$ _____
 $d^* =$ _____

$L_{RE} =$ _____
 $L_C =$ _____
 $L_{FE} =$ _____
 $L_N =$ _____

GEOMETRIC DESCRIPTION OF ARC HEATER

FIGURE 3

SYMBOL DEFINITION

L_1	Length of rear electrode
L_2	Length of transition region between rear electrode and constrictor
L_3	Length of constrictor
L_4	Length of transition region between front electrode and constrictor
L_5	Length of front electrode
L_6	Length of transition region between front electrode and the throat.
d_{RE}	Diameter of rear electrode
d_C	Diameter of constrictor
d_{FE}	Diameter of front electrode
d^*	Diameter of throat

Schematic of "HEAT" Facility

FIGURE 4

60-MW RATING
 PRESSURE 12 atm
 CURRENT 5400 A
 VOLTAGE 11000 V
 FLOW RATE 1.4 kg/sec

PHYSICAL SPECIFICATIONS

WORKING FLUID	AIR, 1.3% AR
ARC BORE	8-cm
SINGLE CONSTRICTOR THICKNESS	1-cm
CONSTRICTOR LENGTH	390-cm
ELECTRODE ASSEMBLY LENGTH	32-cm
OVERALL ARC LENGTH	412-cm
ELECTRODES (TWO)	SIX, MAGNETICALLY SPUN, COPPER

Figure 5.- 8-cm constricted arc heater.

FIGURE 6

Photograph of HEAT (H-1) Facility

CASE I G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia (in) = 6.75
 Cons.Len.(in) = 390
 Cons.Dia.(in) = 9
 Rear Electrode Length (in) = 12
 Front Electrode Length(in) = 12

Electrode Dia (in) = 12
 Arc Length (in) = 416.833
 Nozzle Length(in) = 12
 Current I, (amp) = 2000
 End Flow Fraction(#1)= .9

Power (Mw)	Eff. %	Power Radiated %	Power Conducted %	Voltage volts	Wall Heat Flux kw/m ²	Pressure atm.	Argon Gas Temp. °K	d _A /d*	Arc Temp. °K	1+β _A
36.933	94.770	1.216	4.005	18466.476	0.039	13.254	3110.572	0.214	9000.000	3.011
32.166	94.023	2.999	2.988	16083.171	0.034	12.320	2742.118	0.191	9500.000	2.526
29.224	91.123	6.636	2.240	14612.052	0.040	11.561	2462.262	0.174	10000.000	2.156
27.612	85.308	13.047	1.642	13805.820	0.057	10.873	2224.137	0.158	10500.000	1.847
27.137	76.219	22.628	1.159	13568.323	0.087	10.188	2001.737	0.144	11000.000	1.569
27.704	64.605	34.599	0.795	13851.914	0.130	9.478	1786.064	0.130	11500.000	1.315
29.277	52.213	47.236	0.560	14638.632	0.185	8.759	1583.819	0.117	12000.000	1.086

CASE 2 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia(in) = 4
 Cons.Len.(in) = 200
 Cons.Dia.(in) = 6
 Rear Electrode Length(in) = 12
 Front Electrode Length(in) = 12

Electrode Dia (in) = 10
 Arc Length (in) = 226.000
 Nozzle Length(in) = 12
 Current, I, (amp) = 2000
 End Flow Fraction(#1) = .9

33.748	91.629	5.524	2.847	16873.940	0.123	33.398	2794.720	0.221	10000.000	3.166
31.059	87.051	10.874	2.084	15529.732	0.160	31.229	2493.844	0.200	10500.000	2.708
29.682	79.426	19.120	1.461	14841.173	0.231	29.162	2225.779	0.181	11000.000	2.309
29.442	69.091	29.939	0.972	14720.776	0.336	27.088	1975.284	0.163	11500.000	1.949
30.260	57.318	42.058	0.627	15129.961	0.472	25.013	1743.177	0.147	12000.000	1.624

TABLE 2

CASE 3 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia.(in) = 6.75
 Constr.Len(in) = 160
 Constr.Dia(in) = 8
 Rear Electrode Length (in) = 12
 Front Electrode Length(in) = 12

Electrode Dia (in) = 12
 Arc Length (in) = 186.833
 Nozzle Length(in) = 12
 Current,I (amp) = 4000
 End Flow Fraction (#1) = .9

Power Mw	Eff. %	Power Radiated %	Power Conducted %	Voltage volts	Wall Heat Flux kw/m ²	Pressure atm.	Argon Gas Temp. oK	d _A /d*	Arc Temp. oK	1+β _A
30.794	95.098	4.400	0.501	7699.569	0.050	11.370	2667.865	0.226	10000.000	3.276
28.965	90.786	8.817	0.383	7241.372	0.088	10.775	2436.456	0.207	10500.000	2.862
28.150	83.869	15.832	0.305	7037.413	0.149	10.209	2223.297	0.190	11000.000	2.492
28.282	74.347	25.399	0.255	7070.516	0.239	9.635	2028.358	0.173	11500.000	2.146
29.356	63.161	36.607	0.227	7339.055	0.356	9.048	1835.888	0.157	12000.000	1.823

CASE 4 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia(in) = 4
 Constr.Len.(in) = 80
 Constr.Dia (in) = 6
 Rear Electrode Length (in) = 12
 Front Electrode Length(in) = 12

Electrode Dia(in) = 10
 Arc Length (in) = 106.000
 Nozzle Length(in) = 12
 Current,I (amps) = 4000
 End Flow Fraction(#1) = .9

33.378	94.431	4.673	0.903	8344.540	0.145	31.768	2840.921	0.303	10000.000	4.892
30.912	90.161	9.169	0.667	7727.998	0.234	29.873	2558.349	0.273	10500.000	4.285
29.605	83.295	16.216	0.487	7401.238	0.381	28.100	2309.679	0.247	11000.000	3.740
29.323	73.890	25.753	0.353	7330.739	0.591	26.339	2077.897	0.224	11500.000	3.234
29.985	62.852	36.878	0.261	7496.250	0.861	24.565	1859.481	0.202	12000.000	2.761

TABLE 3

PROGRAM LISTING

```

0: prt "C.E.-AR-3"
1: fxd 3; spc 2; .0254+A; .45359+B
2: enp "AR FLOW(PPS)=", r1; Br1-r1
3: enp "THR DIA(IN)=", r2; Ar2-r2
4: enp "CONS L (IN)=", r3; Ar3-r3
5: enp "CONS D(IN)=", r4; Ar4-r4
6: enp "R ELE L (IN)=", X; AX-X
7: enp "F ELE L (IN)=", Y; AY-Y
8: enp "ELE D(IN)=", r5; r5*A-r5
9:  $\sqrt{(X^2+(r5/2)^2)+\sqrt{(Y^2+(r5/2)^2)+r3}} \rightarrow r7$ 
10: prt "ARC L(IN)=", r7/A
11: enp "NOZ L (IN)=", r8; r8*A-r8
12: enp "I(AMPS)=", r9
13: enp "END FLOW FR(#1)=", r10
14: 107→r19; .15→r20; .04→r21; 10→r22; 0→r23; 300* $\sqrt{r1/r2}$ →r26
15: .5→r27; 0→r52; 9/12→X; 20000→C
16: (r2/r4)2/r27→r55; sfg 14
17: 5900((12/13*X-.543)/.457)(r22*.16)→r15
18: 1.95e15*r221.025*exp(-15.15/X)→r11
19: 6.21e6*X*exp(.5X5/r22.25)→r12
20: (4+29.4X)1e-5→r13; .7+1e3r13→C
21: 4.48e-3*r7/r4→E; (1+(r1/E)4)/1.4576(r1/E).5→F
22: .3*10(-4)→r14
23: (12+r10)/(1+.15r55)→r0; (r14/r13).2→Z
24: ( $\pi$ *r13*r4/4/r1).2*.1374*r7/r4/B(2/3)→r16
25: r0/4(1+137*(1- $\sqrt{r27}$ )*r4/r7*(r2/r4)2)→r17
26: (r51.2-r21.2)*r4.8*r8*r17*r16/(r52-r22)/.6/r7*Z→r18
27: 1+.008*r22*r3/r43*r10→Y
28: 0→r6; (E/.7)(2/3)*r18→r18
29: 19000*X5*r7/r1/r12/ln(1/ $\sqrt{r27}$ )→A
30: (.67r19).5*r1*1.369*10(-5)/r22*Y→r22
31: r11*atn( $\sqrt{1.67(r2/r4)^2}$ )/ $\sqrt{1.67/(r2/r4)^2}$ * $\pi$ /180/r15→r23
32: 4*r1*r12*(1-r20)/ $\pi$ /r7/r42/r15→r24
33: (r20+r21+A)*r1*r12/r9/r7→r25
34: (r252+(r23+r24)*4).5/4+r25/4+r26/2→r26
35: 4r9/ $\pi$ /r15/r26/r42→r27; if abs(r26-r52)/r26<=.0001; gtc 70
36: 1-r10-r27→r28
37: (1-r27).8*r27.6/r16/r0→r29
38: (1-r10)*r29/r27.6*(.7/B)(2/3)*r0/r17*(r13/r14).2→r30
39: .02→r31; 0→r41; 0→r46
40: r10→r32
*13132

```


PROGRAM LISTING (CONT'D)

```
41: r32+r31*r28→r33
42: r33+r31*r28→r34
43: r27/(exp(r29/r32^.8)-1)→r35
44: r27/(exp(r29/r33^.8)-1)→r36
45: r27/(exp(r29/r34^.8)-1)→r37
46: (1-r10)/(exp(r30/r32^.8)-1)→r38
47: (1-r10)/(exp(r30/r33^.8)-1)→r39
48: (1-r10)/(exp(r30/r34^.8)-1)→r40
49: 2*r31/r32*(r35-(r28+r35+r36)*r41)→r42
50: 2*r31/r33*(r36-(r28+r36+r39)*(r41+r42/2))→r43
51: 2*r31/r33*(r36-(r28+r36+r39)*(r41+r43/2))→r44
52: 2*r31/r34*(r37-(r28+r37+r40)*(r41+r44))→r45
53: 2*r31*r39*r41→r47
54: 2*r31*r39*(r41+r42/2)→r48
55: 2*r31*r39*(r41+r43/2)→r49
56: 2*r31*r40*(r41+r44)→r50
57: r41+(r42+2r43+2r44+r45)/6→r41
58: r46+(r47+r48*2+r49*2+r50)/6→r46
59: r34→r32
60: r6+2*r31→r6;if r6≤1;goto 41
61: (r27+(1-r27)*r41)*r12→r51
62: r51(1-r18)→r19
63: r41→r20;0→r41
64: r46→r21;0→r46
65: r26→r52
66: wtb 6,27,72,int(10/64),int(10)
67: fmt 1,f11.3,z
68: fmt 2,5f11.3
69: dsp r52;goto 16
70: r7*r9*r20→r53;wrt 6.1,r53/10^6
71: wrt 6.1,100*r1*r19/r53
72: r23/r26^2→r54;wrt 6.1,100r54
73: wrt 6.1,100r1(r12*(r21+A)+r18*r51)/r53
74: wrt 6.1,r7*r26
75: wrt 6.1,(r54*r53+r1*r12*(A+r20*r40))/π/r4/r7/1e7
76: wrt 6.1,r22
77: wrt 6.1,r19*12000/6.2e6+400;wrt 6.1,1/√r55
78: wrt 6.2,X*12000,r0,r17,√r27
79: X+.5/12→X;r26→C;0→r52;goto 16
80: end
*15256
```

(A COMPLETE DESCRIPTION OF THIS FIGURE IS GIVEN IN THE TEXT)

FIGURE 1. THE INTERACTION SCHEME BETWEEN AN ARC AND TURBULENT GAS FLOW IN A LONG TUBE.

CASE 1 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia (in) = 6.75
 Cons.Len. (in) = 390
 Cons.Dia. (in) = 9
 Rear Electrode Length (in) = 12
 Front Electrode Length (in) = 12

Electrode Dia (in) = 12
 Arc Length (in) = 416.833
 Nozzle Length (in) = 12
 Current I, (amp) = 2000
 End Flow Fraction (#1) = .9

Power (Mw)	Eff. %	Power Radiated %	Power Conducted %	Voltage volts	Wall Heat Flux kw/m ²	Pressure atm.	Argon Gas Temp. °K	d _A /d*	Arc Temp. °K	1+β _A	1+β _W	d _A /d _C
34.933	94.776	1.216	4.005	18465.476	0.036	13.254	3110.572	0.214	9000.000	3.011	0.314	0.100
32.166	94.023	2.909	2.986	16083.171	0.034	12.320	2742.119	0.191	9500.000	2.576	0.240	0.133
29.224	91.123	6.636	2.240	14612.052	0.040	11.561	2462.262	0.174	10000.000	2.195	0.220	0.130
27.612	85.308	13.047	1.642	13905.820	0.057	10.873	2224.137	0.158	10500.000	1.847	0.167	0.115
27.137	76.219	22.629	1.156	13569.323	0.087	10.188	2001.737	0.144	11000.000	1.566	0.158	0.100
27.704	64.605	34.599	0.795	11951.914	0.130	9.479	1796.064	0.130	11500.000	1.315	0.131	0.098
29.277	52.213	47.236	0.560	14638.632	0.185	8.759	1583.819	0.117	12000.000	1.056	0.108	0.090

CASE 2 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia (in) = 4
 Cons.Len. (in) = 200
 Cons.Dia. (in) = 6
 Rear Electrode Length (in) = 12
 Front Electrode Length (in) = 12

Electrode Dia (in) = 10
 Arc Length (in) = 226.000
 Nozzle Length (in) = 12
 Current, I, (amp) = 2000
 End Flow Fraction (#1) = .9

33.769	91.629	5.524	2.847	16873.940	0.123	33.398	2794.720	0.221	10000.000	3.100	0.333	0.107
31.059	87.051	10.874	2.064	15529.732	0.160	31.229	2493.844	0.200	10500.000	2.708	0.262	0.133
29.689	79.426	19.120	1.461	14041.173	0.231	29.162	2225.775	0.181	11000.000	2.305	0.239	0.121
29.442	68.091	29.939	0.972	14720.775	0.336	27.089	1975.264	0.163	11500.000	1.849	0.200	0.105
30.260	57.318	42.050	0.627	15129.961	0.472	25.013	1743.177	0.147	12000.000	1.624	0.165	0.098

TABLE 2

CASE 3 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia.(in) = 6.75
 Constr.Len(in) = 160
 Constr.Dia(in) = 8
 Rear Electrode Length (in) = 12
 Front Electrode Length(in) = 12

Electrode Dia (in) = 12
 Arc Length (in) = 186.833
 Nozzle Length(in) = 12
 Current,I (amp) = 4000
 End Flow Fraction (#1) = .9

Power Mw	Eff. %	Power Radiated %	Power Conducted %	Voltage volts	Wall Heat Flux kw/m ²	Pressure atm.	Argon Gas Temp. oK	d _A /d*	Arc Temp. oK	1+ ρ_A	1+ ρ_W	d _A /d _C
30.794	95.096	4.430	0.501	7400.560	0.050	11.370	2667.865	0.226	10000.000	1.276	0.147	0.131
29.565	90.796	8.817	0.383	7241.372	0.088	10.775	2436.456	0.207	10000.000	2.862	0.161	0.175
28.150	83.869	15.832	0.305	7037.413	0.149	10.209	2229.297	0.150	11000.000	2.492	0.138	0.160
28.292	74.347	25.399	0.255	7070.516	0.239	9.635	2029.158	0.173	11500.000	2.146	0.118	0.116
29.356	63.161	36.607	0.227	7339.055	0.356	9.049	1835.898	0.157	12000.000	1.823	0.099	0.133

CASE 4 G.E. AR-3

Ar Flow (pps) = 55.1
 Throat Dia(in) = 4
 Constr.Len.(in) = 80
 Constr.Dia (in) = 6
 Rear Electrode Length (in) = 12
 Front Electrode Length(in) = 12

Electrode Dia(in) = 10
 Arc Length (in) = 106.000
 Nozzle Length(in) = 12
 Current,I (amps) = 4000
 End Flow Fraction(#1) = .9

33.378	94.431	4.673	0.903	8344.540	0.145	31.768	2940.921	0.303	10000.000	4.692	0.326	0.202
30.912	90.161	9.169	0.667	7727.998	0.234	29.873	2559.349	0.273	10500.000	4.285	0.381	0.182
29.605	83.295	16.216	0.487	7401.238	0.381	29.109	2309.679	0.247	11000.000	3.740	0.241	0.165
29.323	73.690	25.753	0.353	7330.739	0.591	26.339	2077.897	0.224	11500.000	3.234	0.266	0.149
29.985	62.852	36.878	0.261	7496.250	0.861	24.565	1859.481	0.202	12000.000	2.761	0.173	0.125

TABLE 3

APPENDIX C
THE FEEDWATER COOLED BRAYTON CYCLE

1. Introduction

The Brayton cycle has been considered as an alternative to the standard Closed Cycle MHD/steam (CCMHD) power generating system. In the Energy Conservation Alternatives Study (ECAS) ^{C-1}, a parallel cycle concept was used. This system had lower efficiency than comparable open cycle MHD systems and was very costly because regenerative ceramic heat exchangers were used both for the primary heat source and in the Brayton Cycle. A similar approach was taken in the PACC ^{C-2} study where it was found that a parallel Brayton cycle had an efficiency which is somewhat lower than the equivalent MHD/steam cycle. The present study examines a new configuration for CCMHD systems designated the Feedwater Cooled Brayton Cycle.

Consideration of the previously studied CCMHD cycles indicates several areas where better use of thermal availability might improve cycle efficiency. In the case of the MHD/steam cycle, the following weak points are noted:

- o The requirement to extract the maximum possible amount of energy from the argon eliminates the use of feedwater heaters in the steam plant. This reduces steam plant efficiency by about 2 points.
- o The argon must be precooled prior to compression in order to maintain a reasonable compressor exit temperature. About 2.5% of the input energy must be rejected from the argon loop.

The parallel Brayton cycles considered in the past suffer from the following problems:

- o An intercooled compressor is necessary in order to minimize compression power requirements. About 40% of the input energy is rejected from the argon loop.
- o In the PACC study, the combustion system was not integrated with the steam plant, which causes a reduced efficiency for the combined system.

Examination of the weak points cited above for the MHD/steam and parallel MHD Brayton/steam cycles suggests that an improvement in efficiency can be achieved if heat rejection can be eliminated from the argon loop. Further, because of the large argon pressure ratio necessary to meet the enthalpy extraction requirements of the MHD/steam cycle, it is clear that this cannot be achieved using a topping cycle followed by steam generation with argon. Consequently, it was concluded that the best means of achieving an improvement in efficiency for CCMHD systems was a fully integrated MHD Brayton cycle where argon cooling was by means of feedwater. The heat rejected from the argon, thus, remains in the cycle and is not rejected to ambient. In addition, it was decided that the combustion subsystem should be fully integrated into the overall system.

This study examines the feedwater cooled Brayton (FCB) cycle parametrically in order to evaluate cycle efficiency as a function of cycle parameters, steam turbine entrance conditions and output power level. It is concluded from the analysis that the FCB cycle has a higher efficiency than previously considered CCMHD configurations.

2. Description of the Feedwater Cooled Brayton Cycle

Figure C-1 shows a block diagram of the feedwater cooled Brayton Cycle. Component names are defined in Table C-1. High grade energy (3350 - 2100 F) is fed to an MHD topping cycle, steam is generated at intermediate temperatures (2100 - 800 F) and remaining energy is used for air preheat and coal drying. Heat rejection in the argon loop is accomplished by means of feedwater heaters.

Figure C-1. Schematic of a Feedwater Cooled Brayton (FCB) Cycle

Table C-1. Dictionary of Symbols for Feedwater
Cooled Brayton (FCB) Cycle

AB	=	air blower
AH	=	air heater
ARFW1	=	high pressure feedwater heater
ARFW2	=	low pressure feedwater heater
ARHX	=	argon/argon heat exchanger
CB	=	main combustor
CDR	=	coal drier
CMP	=	main compressor
CND	=	condenser
CP	=	condensate pump
DIF	=	diffuser
EC	=	economizer
FGRB	=	flue gas recirculation blower
FP	=	feed pump
GEN	=	MHD generator
HPT	=	high pressure turbine
HTRH	=	regenerative argon heater
INV	=	inverter
LPT	=	low pressure turbine
RB	=	radiant boiler
RH	=	reheater
RHT	=	reheat turbine
SH	=	superheater

The combustion products feed a high temperature argon heater (HTRH) and then flow into a boiler/air heater system. Combustion temperature is controlled to a maximum of 3350 F using flue gas recirculation to avoid formation of thermal NO_x . A bypass is provided from the combustor directly to the boiler to provide sufficient steam to power the argon compressor.

Argon powers the MHD generator which has a moderate enthalpy extraction - 25%. Heat is regenerated in a ceramic shell and tube heat exchanger manufactured from silicon carbide. The maximum temperatures of the argon on the shell and tube sides of 2150 F and 1800 F, respectively, are well within present operating capabilities of this type of heat exchanger. Low level heat in the argon is rejected prior to compression using feedwater heaters. Reduced enthalpy extraction in the generator requires only a modest compression ratio (2.9 - 3.4) in comparison to that necessary for the CCMHD/steam cycle (about 5.0). The main argon compressor inlet temperature is about 130 F, but the exit temperature for the compressor (500 - 550 F) is reasonable because of the modest pressure ratio. Finally, the temperature of the argon entering the HTRH is maintained at 1800 F which is reasonable for hot bottom, regenerative heat exchange matrices.

No feedwater heating using turbine exhaust steam has been included in the cycle presented here. A variation of the FCB Cycle could attempt an interlinked feedwater heating system where both argon and exhaust steam are used to supply heat. At the present time, this additional complication was not deemed warranted.

Shaft power from the steam turbines is used to drive the main argon compressor. Whenever there is excess power, the remainder is used for the generation of electricity. If there is insufficient shaft power available, the bypass is opened until the output of the steam turbines just matches the argon compressor requirements.

The new FCB Cycle has a number of novel features which are summarized in Table C-2. The low level of enthalpy extraction in the generator relaxes constraints on generator performance. This in turn relaxes magnetic field requirements such that the desired generator performance can be achieved using a magnet having a peak field of 4 Tesla. Thermodynamically this is irrelevant, but there will be a reduction in magnet cost.

3. System Specification

The FCB Cycle has been examined in detail using the GE's advanced closed cycle system analysis code (CCSYSA). The program computes mass and energy balances for the system as well as turbine efficiencies, power generated and auxiliary power requirements. Figure C-2 presents a block diagram of the system which was used for the study. Typical state points for the cycle are given in Table C-3.

Inputs to CCSYSA must be supplied in order to specify the operating characteristics of the equipment. The argon mass flow is computed from a specified output of the MHD generator. Fuel and steam requirements then follow from energy balances between the three fluids. Pressure losses are specified for all components as shown in Table C-4. Mechanical and thermal efficiencies are also required (see Table C-5). Finally, temperatures at critical points in the system as well as minimum approach temperatures must be given. The

Table C-2. Features of the feedwater Cooled Brayton (FCB) Cycle

- o MODERATE ENTHALPY EXTRACTION (~ 25%) FOR THE MHD GENERATOR AND CONSEQUENT MODERATE PRESSURE RATIO ACROSS THE GENERATOR (- 3 TO 1)
- o LOW LEVEL HEAT IN THE ARGON LOOP IS REJECTED TO THE FEEDWATER. ALL FEEDWATER HEATING IS DONE USING THIS HEAT.
- o HIGH INLET TEMPERATURE (~ 130 F) FOR THE ARGON COMPRESSOR BUT, EXIT TEMPERATURE ACCEPTABLE (~ 500 F) BECAUSE OF LOW COMPRESSION RATIO
- o CERAMIC RECUPERATIVE HEAT EXCHANGER FOR HIGH LEVEL HEAT RECOVERY IN THE ARGON LOOP
- o HOT BOTTOM DESIGN FOR THE REGENERATIVE HIGH TEMPERATURE ARGON HEATER (HTRH)
- o FLUE GAS BYPASS AROUND HTRH TO MEET EXTRA LOAD FOR STEAM GENERATION AND AIR HEATING
- o TWO PERCENTAGE POINT IMPROVEMENT IN EFFICIENCY OVER CCMHD/STEAM CYCLE

Table C-3. State Points for the Feedwater Cooled Brayton (FCB) Cycle
(1000 MWe - 3500/1000/1000)

No.	Description	M lb/hr	P psia	T (F)	E (MW)
1	ARGON	2.3810E7	147.00	3000.0	2573.6
2	ARGON		50.61	2135.0	1823.6
3	ARGON		49.60	851.6	710.7
4	ARGON		48.60	125.3	80.9
5	ARGON		157.89	516.6	420.3
6	ARGON		154.74	1800.0	1533.2
7	ELECTRICAL POWER	-	-	-	750.0
8	ELECTRICAL POWER	-	-	-	738.7
9	MECHANICAL POWER	-	-	-	339.3
10	WATER	3.6080E6	3.15	101.	72.8
11	WATER		378.00	101.5	73.8
12	WATER		3734.70	647.3	713.2
13	WATER		3600.00	701.3	820.3
14	STEAM		3586.80	854.3	1359.9
15	STEAM		3515.50	1000.1	1502.4
16	STEAM		800.16	611.6	1352.2
17	STEAM		720.15	1000.1	1599.7
18	STEAM		154.78	607.0	1404.7
19	STEAM		0.98	101.1	1110.8
20	MECHANICAL POWER	-	-	-	322.4
21	MECHANICAL POWER	-	-	-	194.0
22	MECHANICAL POWER	-	-	-	149.4
23	AIR	6.4645E6	14.70	59.0	38.7
24	AIR		15.29	65.0	41.7
25	AIR		14.98	700.0	341.6
26	COAL (AS RECEIVED)	8.5579E5	14.70	59.0	2361.4
27	COAL (DRY - 5% FMR)	7.3234E5	14.70	220.0	2361.4
28	FLUE GAS	9.9025E6	14.68	3349.1	2938.0
29	FLUE GAS	9.3819E6	14.68	3349.1	2783.6
30	FLUE GAS		14.39	2100.0	1691.1
31	FLUE GAS	5.2060E5	14.68	3349.1	154.5
32	FLUE GAS	9.9025E6	14.39	2167.6	1845.6
33	FLUE GAS	3.6181E6	14.39	2167.6	674.3
34	FLUE GAS		14.35	1727.8	531.9
35	FLUE GAS	6.2844E6	14.39	2167.6	1171.3
36	FLUE GAS		14.35	1727.8	923.8
37	FLUE GAS	9.9025E6	14.35	1727.8	1455.7
38	FLUE GAS		14.31	1070.3	916.1
39	FLUE GAS	7.1257E6	14.31	1070.3	659.2
40	FLUE GAS		14.26	890.6	552.2
41	FLUE GAS		14.10	356.3	252.3
42	FLUE GAS	7.2492E6	14.00	290.4	217.2

Table C-3. (Continued) State Points for the Feedwater Cooled
Brayton (FCB) Cycle (1000 MWe - 3500/1000/1000)

No.	Description	M lb/hr	P psia	T (F)	E (MW)
43	FLUE GAS	2.7768E6	14.31	1070.3	256.9
44	FLUE GAS		14.98	1085.5	260.5
45	SLAG	7.1121E4	-	-	11.1
46	HEAT LOSS	-	-	-	52.02
47	HEAT LOSS	-	-	-	11.18
48	HEAT REJECTED	-	-	-	1037.90
49	ELECTRICAL POWER	-	-	-	0.98
50	ELECTRICAL POWER	-	-	-	2.44
51	ELECTRICAL POWER	-	-	-	2.99
52	ELECTRICAL POWER	-	-	-	3.57

Table C-4. Assumed Pressure Loss Ratios* Across Major Components in the FCB Cycle

Component	Argon	Steam	Flue Gas	Air
ARHX	0.02			
ARFW1	0.01	0.02		
ARFW2	0.01	0.02		
HTRH	0.05		0.02	
CB			0.02	
SH		0.02	0.02	
RH		0.10	0.02	
RB		0.02	0.05	
EC		0.02	0.05	
AH			0.02	0.01
CDR			0.01	

* Defined as $1 - \frac{P_{OUT}}{P_{IN}}$

Table C-5. Major Component Efficiencies and Thermal Losses in the FCB Cycle

Component	Efficiency	Loss/ Thermal Input	Enthalpy Extraction
GEN	0.72		0.25
CMP	0.90		
HTRH		0.05	
CP	0.85	0.05	
FP	0.85	0.05	
HPT	0.845*	0.005	
RHT	0.801*	0.005	
LPT	0.865*	0.005	
AB	0.90	0.005	
CB		0.005	
FGRB	0.90		
AH	0.90	0.005	
INV		0.015	

* Efficiency calculated internally.

critical temperatures are shown in Table C-6 and the minimum pinch point temperature differences are given in Table C-7.

The FCB cycle has an adjustable flue gas bypass ratio. This parameter defines the relative importance of the MHD topping cycle in comparison to the bottoming steam plant. When all of the hot flue gas is bypassed around the HTRH, the FCB cycle becomes relatively inefficient due to the absence of feedwater heating. On the other hand, when all of the flue gas passes through the HTRH, maximum use of the MHD topping cycle is made. Figure C-3 shows the effect of bypass ratio on cycle efficiency. From these results, it can be concluded that no bypass should be used. However, depending on generator enthalpy extraction, the steam plant may not produce sufficient shaft power for the main compressor. Under these circumstances, some bypass will be required.

4. Parametric Analysis of the FCB Cycle

An evaluation of the FCB Cycle was conducted in order to study the effects on efficiency of the following parameters:

1. Steam conditions
2. Air preheat
3. Generator (MHD) enthalpy extraction
4. Generator Entrance conditions
5. Plant size

This analysis was conducted using Montana Rosebud coal and is discussed in the present section.

Table C-6. Specified Temperatures in the FCB Cycle

Component	Location	Temperature (F)
HTRH	1	3000
HTRH	6	1800*
CB	29	3350
SH	32	1000
RH	17	1000
CDR	42	290
CND	10	101

*Valid for MHD Generator enthalpy extraction up to 26%. This temperature is lower when $\eta_x > 26\%$.

Table C-7. Specified Minimum Pinch Point Temperature Differences

Component	Location	Location	Minimum Temperature Difference (F)
HTRH	6	30	300
ARHX	2	6	335
ARFW-2	4	11	25
AH	25	40	100

Figure C-3. Effect of Bypass Ratio on Cycle Efficiency

4.1 Effects of Steam Conditions

Three steam plants were assumed in order to assess the effects of steam conditions on cycle performance:

- a. 3500/1000/1000 - Supercritical steam plant
- b. 2400/1000/1000 - Subcritical steam plant
- c. 1800/1000/1000 - Subcritical steam plant

The results of this analysis are given in Figure C-4 which shows calculated efficiencies for a plant rated at 1000 MWe total output power (no auxiliaries) as a function of steam pressure. It can be seen that plant efficiency increases with increasing steam pressure and that a jump in the efficiency curve occurs when the steam becomes supercritical. Energy flow summaries for the three cases are given in Tables C-8, C-9 and C-10.

4.2 Effects of Air Preheat

An increase in the air preheat temperature increases recirculation gas temperature. This results in a net increase in efficiency. In order to evaluate the maximum possible efficiency, no bypass was used. Instead fractional boiling was permitted in the argon loop. Such an arrangement might result in a complicated boiler design, but a 1.5 point gain in overall cycle efficiency can be obtained. Figure C-5 shows the effect of air preheat temperature on cycle efficiency. As air preheat temperature is increased, the energy available to generate steam decreases. This results in a smaller steam plant. Because the energy transferred from argon to the feedwater is constant, a minimum feedwater flow must be maintained in order to meet pinch temperature requirements. Therefore, there is an upper bound for the air preheat temperature. One can increase air preheat beyond this upper bound (at zero bypass ratio)

Figure C-4. Effect of Steam Pressure on Cycle Efficiency

Table C-8.

Summary of Plant Performance
(Montana Rosebud Coal
1000 MWe - 3500/1000/1000 Steam)

Input

Coal	2361.35	
Air (Sens. + Lat.)	38.74	
Auxiliary Input	9.98	
Total		2410.07

Output

Electrical Power	1076.42	
MHD	750.00	
Steam Turbine	665.76	
Argon Compressor	-339.34	
Stack Loss	217.19	
Heat Rejected (CND)	1037.90	
Heat Loss	78.56	
HTRH	52.02	
Slag	11.11	
Combustor	11.18	
Mech. Losses	3.62	
Total		2410.07

Table C-9.   Summary of Heat Balance
 (Montana Rosebud Coal
 1000 MWe - 2400/1000/1000 Steam)

Input

Coal	2281.00
Air	37.42
Aux.	8.99

Total	2327.41

Output

Electrical Power	1030.96

MHD	750.0
Steam	620.30
Compressor	-339.34

Stack Loss	209.66

Heat Rejected (CND)	1013.20

Heat Loss	73.59

HTRH	52.02
Slag	10.73
Combustor	10.84
Mech. Losses	2.97

Total	2327.4

Table C-10.   Summary of Plant Performance
                   (Montana Rosebud Coal
                   1000 MWe - 1800/1000/1000 Steam)

Input

Coal	2278.33	
Air	37.38	
Auxiliary	8.91	
Total		2324.62

Output

Electrical Power		1021.59
MHD	750.00	
Steam	610.93	
Compressor	-339.34	
Stack Loss		209.41
Heat Rejected (CND)		1017.14
Heat Loss		76.48
HTRH	52.02	
Slag	10.66	
Combustor	10.83	
Mech. Loss	2.97	
Total		2324.62

Figure C-5. Effect of Air Preheat on Cycle Efficiency

by increasing the feedwater flow and simultaneously increasing the bypass ratio. However, the preheat temperatures are becoming large and heat exchange equipment will become expensive.

4.3 Effects of Generator Enthalpy Extraction

An increase in generator enthalpy extraction results in an increase in generator electrical power, an increase in generator/diffuser isentropic efficiency, an increase in required generator pressure ratio and, for a fixed inlet temperature, a decrease in diffuser exit temperature. As the generator enthalpy extraction is varied, various system temperature limits and pinch points are encountered. For example, it has been assumed that argon can be reheated to 1800 F using the argon/argon regenerative heater (ARHX). Further, the minimum pinch temperature in this device is required to be 335 F. Thus, when the exit temperature of the argon from the diffuser drops below 2135 F, these requirements can no longer be met. A lowering of the 1800 F inlet requirement for the HTRH will allow higher enthalpy extraction while maintaining the 335 F pinch temperature requirement in the ARHX. The flue gas temperature exiting the HTRH must then be adjusted to maintain the 300 F pinch requirements for this device.

A second limit is encountered when the pressure ratio required for the generator is such that the exit temperature from the main compressor becomes too large. A limit of 715 F was imposed during the PACC study in order to assure durability of the compression equipment. If this limit is exceeded, one must either sacrifice compressor durability or introduce cooling into the argon loop and sacrifice efficiency. In the PACC study the argon was precooled to

84 F so a pressure ratio of 5.9 was possible within this limit with a corresponding enthalpy extraction of 37%. In the FCB cycle, however, argon inlet temperature is higher (125 F) and the limiting pressure ratio and enthalpy extraction are 4.2 and 32%, respectively.

Finally, a limit is reached at zero bypass when the flue gas exiting the HTRH has just sufficient energy to generate steam for the main compressor and to provide air preheat. This configuration represents a balanced plant where the total output of the turbines is required for compressor power. Enthalpy extraction beyond this point, requires bypass flow for the generation of steam and causes a reduction in efficiency. Thus, the cycle efficiency curve peaks as is shown in Figure C-6.

Examination of Figure C-6 indicates that cycle efficiency can be increased by increasing enthalpy extraction until the various limitations discussed above are reached. Specification of the argon inlet temperature for the HTRH is somewhat artificial in that this limit was selected as a reasonable upper bound for the operation of hot bottom regenerative heat exchangers. Lower temperatures for this device should not prove troublesome. Maximum compressor exit temperature of 700 F is a real physical limit. However, improved compressor design or more frequent compressor refurbishment could allow an extension to the balance point. The question as to whether these improvements are warranted to obtain an increase in efficiency is left open.

4.4 Effects of Generator Entrance Conditions

The performance of the FCB Cycle declines with a decrease in generator entrance conditions for two reasons; namely

1. A decrease in generator inlet temperature, decreases the enthalpy

Figure C-6. Effect of Generator Enthalpy Extraction on Cycle Efficiency

extraction ratio and the isentropic efficiency of the MHD generator, thus, decreasing power output of the generator.

2. At the same level of enthalpy extraction, reduced inlet temperature decreases the amount of heat which can be rejected to the feedwater from the argon loop. Therefore, additional heat must be supplied by the combustion loop to attain the same feedwater temperature. This results in a decrease in overall cycle efficiency.

Figure C-7 shows the effect of generator inlet temperature on generator performance. In addition, the isentropic efficiency of the generator/diffuser is plotted. This latter result is possible because argon is a perfect gas and, to first order, heat losses can be neglected. The results show that enthalpy extraction is not a strong function of temperature for a constant pressure ratio generator. The isentropic efficiency, however, decreases substantially as the temperature falls from 3000 to 2600 F. It must be recognized that generator performance is a function of pressure ratio and that the efficiencies shown do not represent optimum values. The intent of the curves is to show trends and not to define an optimum configuration.

4.5 Effect of Plant Size

The variation of efficiency with plant size must be examined using realistic steam plant conditions. For example, a supercritical steam cycle would never be used with a 100 MWe plant because of economic and operational considerations. For this reason, the following set of steam conditions was selected:

1. For plant sizes between 60 - 200 MWe a subcritical steam plant with a 1800/1000/1000 steam cycle was used.
2. For plant sizes between 200 - 450 MWe a subcritical steam plant with a 2400/1000/1000 steam cycle was used.
3. For plant sizes between 450 - 1200 MWe a supercritical steam plant with a 3500/1000/1000 steam cycle was used.

Figure C-7. Effect of Generator Inlet Temperature on Generator Performance

In addition to different steam cycles, the effects of plant auxiliaries were evaluated such that a coal pile to bus bar efficiency for the different size power generating stations could be computed. The auxiliaries considered, along with their power requirements are shown in Table C-11 for a 1000 MWe plant.

Figure C-8 shows the variation of station efficiency as a function of net output power. These efficiencies are differentiated from the previously discussed cycle efficiencies by the inclusion of required electrical auxiliaries. The two definitions are:

$$\text{Station Efficiency} = \frac{\text{Total Output Power} - \text{Auxiliaries}}{\text{Coal Input}}$$

$$\text{Cycle Efficiency} = \frac{\text{Total Output Power}}{\text{Coal Input}}$$

The results in Figure C-8 indicate that the FCB cycle becomes more efficient with increasing size. However, the increase in efficiency is only 1.5 points as one moves from moderate scale to large base load plants.

In addition to the efficiency curve for the standard FCB cycle, two parametric points are shown. The first, labeled as a 2% heat loss, assumes an advanced technology shell and tube heat exchanger which has an ultimate thermal loss significantly less than a regenerative heat exchanger because of reduced surface area. This improvement in thermal loss results in a 1.5 point increase in station efficiency. The second, labeled 2800 F generator inlet, assumes a

Table C-11.

Auxiliary Load for FCB Plant
(1000 MWe - 3500/1000/1000)

Auxiliary Load

Pumps and Fans		11.50	
Condensate Pump	0.98		
Feed Pump	2.44		
Air Blower	2.99		
FGRB	3.57		
IDF	1.52		
Dry Scrubbing		2.17	
Cooling Water System		4.25	
Circulating Water Pumps	0.93		
Tower Fans	3.32		
Coal Preparation		4.80	
Handling	0.96		
Processing	3.84		
Slag, Ash and Seed Handling		1.42	
Hotel Loads		3.07	
Magnet Auxiliaries		2.31	
Transformer Losses		5.38	
Inverter Losses		11.25	
Argon Cleanup and Evacuation System		26.01	46.15
Evacuation	23.41		
Cleanup	2.60		
Total			72.16

Figure C-8. Effect of Plant Size on Station Efficiency

state-of-the-art ceramic shell and tube heat exchanger. No allowance was made for possible additional starting length to initiate non-equilibrium conduction in the MHD generator. The thermal loss was not reduced with inlet temperature because of the leakage from the tube side (argon) to the shell side (flue gas). It can be seen that a state-of-the-art shell and tube heat exchanger with reduced inlet argon temperature causes a penalty of only about 0.6 points. If MHD generator performance is not degraded, this reduced efficiency may well be worth while for economic reasons.

5. Cost Estimate for an FCB Power Generation Plant

A preliminary cost estimate for an FCB plant has been prepared using the methodology assumed for the PACC study. The costed configuration was optimized using the results of Section 4 to assure maximum plant efficiency. Major equipment costs were defined as in PACC with the addition of the argon/argon and argon/feedwater heat exchangers and the replacement of the HRSR subsystem with a standard combustion product/steam boiler. Field construction costs were estimated and the COE was defined. These results are discussed in this section.

Based on the results of Sections 4.2 to 4.4 an FCB plant was selected for costing which has the following characteristics:

- o 750 MWe nominal generator output
- o 2400 psia/1000 F/1000 F subcritical steam plant
- o MHD generator enthalpy extraction 30% with zero bypass ratio.

- o Maximum Air Preheat of 970F as determined by a reasonable stack exit temperature of 318F.
- o Partial boiling in the argon/feedwater cooler.
- o 10 atmosphere MHD cycle
- o All pressure drops and heat losses as specified in Tables C-4 and C-5.

The selected cycle is shown in Figure C-9 and the state points, thermal losses and auxiliary power requirements are given in Tables C-12 to C-14, respectively. An energy flow summary is presented in Table C-15 from which it is determined that the coal pile to-bus-bar efficiency is 44.37%.

Costs for the FCB plant were established using the methodology of the PACC study as discussed in Section 5.2 and Appendix F of that report. A standard steam boiler system was assumed and costed using methods developed by GE. The ceramic argon/argon heat exchanger is similar to, but much larger than, a unit which was recently designed and costed by a vendor. Thus, costs for the FCB equipment were extrapolated from this quote using the 0.7 power of the heat exchanger duty. The metallic (stainless steel to avoid cesium corrosion) feedwater heaters were designed in-house and costed on a dollars/pound basis. BOP, installation and indirect costs were established using assumptions identical to those used in the PACC analysis. These data were then applied to estimate the total field construction costs under the same assumptions as in the PACC study. A summary of cost data for the FCB plant is given in Table C-16 and a breakdown of the MHD topping cycle in Table C-17.

Cost of electricity (COE), Table C-18, was estimated in the same manner as in PACC where COE represents the average cost of energy produced over the lifetime of the plant (assumed to be 30 years). In calculating these costs, the year 2000 was assumed to be the date for the start of commercial operation with a 6-1/2% general inflation rate assumed throughout the life of the plant. The COE computed for this FCB plant was 72.3 mills/kwhr which is significantly lower than any of the PACC cases for a plant of equivalent size.

Figure C-9. Diagram of a 750 MWe Feedwater Cooled Brayton Cycle

Table C-12. State Points for a 750 MWe FCB Cycle (2400/1000/1000)

No.	Description	$\dot{M}$ (lb/hr)	T °F	P psia	$\dot{E}$ MW
1	Argon	1.9842E7	3000.	147.00	2112.4
2			1962.	40.99	1362.4
3			1035.	40.17	692.7
4			700.	39.77	450.6
5			175.	39.37	71.6
6			700.	157.89	450.6
7			1622.	154.74	1117.0
8	Mech Pwr				-380.9
9	Elec Pwr				750.0
10	Elec Pwr				738.8
11	Water	2.3810E6	94.	3.0	12.4
12			97.	2565.6	18.6
13			600.	2514.3	395.8
14		1.4924E6	600.	2514.3	248.1
15			690.	2464.0	489.0
16		0.8886E6	600.	2514.3	147.7
16a			690.	2464.0	291.2
17		2.3810E6	690.	2514.3	780.1
18			1000.	2419.7	987.2
19			643.	603.7	887.6
20			1000.	543.3	1028.1
21			591.	108.7	892.4
22			94.	0.8	666.3
23	Elec Pwr				99.1
24	Elec Pwr				135.0
25	Elec Pwr				225.0
26	Heat Rej				553.9
31	Air	4.9769E6	59.	14.7	3.5
32			76.	16.3	9.5
33			970.	16.0	337.3
34	Coal As Rec'd	.68859E6	59.		1801.1
35	Dry Coal	.55889E6	220.		1808.4
36	Flue Gas	7.6613E6	3350.	15.6	2247.5
37			3350.	15.6	2247.5
38		.0	3350.	15.6	0
39		7.6613E6	1922.	15.3	1232.0
40			1922.	15.3	1232.0
41		4.5648E6	1922.	15.3	734.2
42			1408.	15.0	528.2
43		3.0966E6	1922.	15.3	498.0
44			1408.	15.0	358.3

Table C-12. State Points for a 750 MWe FCB Cycle (2400/1000/1000) Cont'd

45	4.5648E6	1408.	15.0	886.4
46		1164.	14.9	743.7
47	5.4860E6	1164.	14.9	532.5
48	2.1753E6	1164.	14.9	211.2
49		1191.	16.0	216.2
50	5.4860E6	481.	14.8	206.5
51	5.5339E6	318.	14.7	199.2*
52	Slag	0.04793E6		7.3

*Contains 43.7 MWt of Heat from Coal Moisture

Table C-13. Thermal Losses for a 750 MWe FCR Cycle (2400/1000/1000)

COMPONENT	E (MW)
ARHX	3.348
ARB	1.210
ARFW	1.895
CMP	1.895
HTRH	<u>19.909</u>
TOTAL	28.257
FEED PUMP	0.313
RB	0.717
SHH	1.035
RHH	0.702
HPT	0.498
RHT	0.678
LPTUR	<u>1.136</u>
TOTAL	5.074
AIR BLOWER	0.030
AH	1.639
FGB	0.025
CB	<u>106.920</u>
TOTAL	108.614
INVERTER	11.2

Table C-14. Auxiliary Power Requirements for a 750 MWe FCB Cycle
(2400/1000/1000)

Pumps & Fans		21.63
Condensate Pump	0.98	
Feed Pump	6.56	
Air Blower	6.05	
FGRB	5.01	
IDF	3.03	
Dry Scrubbing		2.17
Cooling Water System		4.25
Circulating Water Pumps	0.93	
Tower Fans	3.32	
Coal Preparation		4.80
Handling	0.96	
Processing	3.84	
Slag, Ash & Seed Handling		1.42
Hotel Loads		3.07
Magnet Auxiliaries		2.31
Transformer Losses		4.08
HTRH Evacuation		<u>17.59</u>
		61.26

Table C-15. Energy Flow Summary for a 750 MWe FCB Cycle
(2400/1000/1000)

Input:		TOTAL
Coal	1801.1	
Air	3.5	
Water	12.4	1817.0
Output:		1812.8
Net Power Output:		
MHD Generator	688.7	
High Pressure Turbine	99.1	
Reheat Turbine	135.0	
Low Pressure Turbine	225.0	
Argon Compressor	-380.9	
Internally Generated	32.3	799.2
Plant Auxiliaries		
Heat Output:		
Slag	7.3	
Condenser	653.9	
Stack	199.2	860.4
Losses:		
Argon Loop	28.3	
Steam Loop	5.1	
Combustion Loop	108.6	
Inverters	11.2	153.2

Table C-16. Plant Capital Cost Estimate Summary
Feedwater Cooled Brayton Cycle
(Mid 1978 Dollars X 10⁶)

Account No.	Account Description	Major Equipment	BOP	Inst Cost	Indirect Cost	Total Cost
310	Land and Land Rights	-	11,150	-	-	11,150
311	Structures and Improve- ment	-	74,560	3,560	26,840	104,960
312	Boiler Plant Equipment	28,310	15,170	17,240	6,950	67,670
313	Combustion System Equip- ment	81,930	65,040	32,040	11,910	190,920
314	Turbogenerator Units	48,930	20,910	7,160	8,710	85,710
315	Accessory Electrical Equipment	-	16,350	19,980	18,170	54,500
316	Miscellaneous Power Plant Equipment	-	5,290	1,470	1,220	7,980
317	MHD Topping Cycle*	288,240	18,390	23,510	24,680	354,820
350	Transmission Plant	-	10,420	1,500	5,960	17,880
FIELD CONSTRUCTION COST		447,410	237,280	106,460	104,440	895,590
Engineering Services						167,230
Contingency						141,760
Other Costs - Escalation and Interest during Construction						138,540
TOTAL ESTIMATED CONSTRUCTED COST						1,343,120

* See Table C-17 for breakdown of MHD topping cycle costs

Table C-17 Feedwater Cooler Brayton Cycle
MHD Topping Cycle Costs
(Mid-1978 Dollars)

<u>Account No.</u>	<u>Account</u>	<u>Major Equipment</u>	BOP	Inst. Cost	Indirect Cost	Total Cost
317.2	MHD Generator	15,860	0	590	1,647	18,097
317.3	Magnet	40,920	0	8,980	4,250	54,150
317.4	Inverters	48,240	0	6,800	6,120	61,160
317.5	NTRH ¹	97,840	10,110	4,040	10,160	122,150
317.6	Seed Subsystem	410	0	0	43	453
--	ARFW ²	27,430	0	0	0	27,430
--	ARHX ³	57,540	0	0	0	57,540
317.7	Misc.	0	4,190	370	0	4,560
317.8	Piping and Instr.	0	4,090	2,730	2,460	9,280
317	Total	288,240	18,390	23,510	24,680	354,820

1. High temperature Argon Heaters - assumed to be of the regenerative type.
2. Argon Feedwater Heater.
3. Brayton cycle heat exchanger - Silicon Carbide, Shell and tube type.

Table C-18. Cost of Electricity, Feedwater Cooled Brayton Cycle
(Mid-1978 Dollars)

Capital Cost (\$K)	\$1,343,120
Power Output (MWe)	799.2
Unit Capital Cost (\$/Kw)	1,680
Efficiency	0.4437
O & M Cost (\$K/year)	19,392
CAP (mills/Kwhr)	53.12
FUEL* (mills/Kwhr)	15.20
O&M (mills/Kwhr)	8.02
COE (mills/Kwhr)	76.34

* Based on 1978 fuel cost of \$1.05/10⁶ Btu

6. Summary of the FCB Cycle

The feedwater cooled Brayton Cycle is a new CCMHD concept which shows promise of providing very high efficiencies (about 44.37% coal-pile-to-bus bar for a 750 MWe system). The cycle is made possible by a newly developed ceramic heat exchanger technology which allows regeneration of thermal energy in the argon loop. High efficiencies compared to other CCMHD cycles are achieved by the elimination of heat rejection in the argon loop by use of feedwater to cool the argon to a realistic inlet temperature for the main compressor. This is made possible by the reduced generator enthalpy extraction requirement for the FCB cycle in comparison to the standard MHD/steam cycle. Reduced extraction requires a lower pressure ratio across the generator which in turn causes a lower argon temperature rise across the main compressor. Finally, full integration of the MHD, combustion and steam generation subsystems increases the efficiency of the overall plant.

Costs for a 750 MWe FCB plant with a 2400 psia/1000 F/1000 F subcritical steam plant have been estimated. The major cost saving is the replacement of the HRSR subsystem used in the standard MHD/steam cycle by a conventional steam boiler. This is partially off-set by the addition of the argon/argon ceramic heat exchanger and the stainless steel argon/feedwater heaters. The capital cost of the 750 MWe FCB plant was estimated to be 1.343×10^6 mid 1978 dollars which results in a COE of 76.3 mills/kwhr, Table C-18.

REFERENCES

- C1 Corman, J.C. et.al., Energy Conversion Alternatives Study -
ECAS - General Electric Phase I Final Report, NASA CR-134948,
February 1976 Voc II, Part 3.
- C2 Marston, C.H., Alyea, F.N. et.al., Parametric Analysis of Closed
Cycle MHD Power Plants - PACC - NASA CR-165450, September 1981.

APPENDIX D

APPLICATION OF CCMHD IN AN INDUSTRIAL COGENERATION ENVIRONMENT

1. Introduction

The application of systems for the generation of power, on site, for industrial uses (cogeneration) has received considerable attention in the energy community. Many industrial processes require both electricity and steam in order to produce the desired product. Because steam can be used to generate electricity, considerable fuel savings can be realized by producing higher grade steam than is required, generating electricity with the excess energy and, then, feeding the steam to the process. Under these conditions, the efficiency of electrical generation is very high (~95%). Consequently, some or all of the electrical needs of the process can be supplied using fuel converted to electricity on an almost one to one basis. In contrast, generation by a utility converts fuel to electrical power at an efficiency of about 32%. This represents a fuel saving of greater than 60%.

The thermodynamic basis for cogeneration rests on an availability argument. Most fuels used industrially are capable of producing temperatures which are significantly higher than that of the steam which is required for the process. The argument, then, is to use this excess temperature in a topping cycle for the generation of electric power. The steam rejected from the topping cycle is then available for process requirements.

The high efficiency of electrical cogeneration in comparison to steam power generating systems can be traced to two sources. Firstly, no heat need be rejected from a cogeneration system because in the ideal situation all steam passing through the turbines is used in the process and, secondly, only a small penalty for stack loss need be taken because the majority of the energy in the stack gases would be lost if a boiler were used alone for the generation of

process steam.

The natural question arises as to what benefits, if any, are derived by using high technology, advanced power generating systems for cogeneration applications. A partial answer to this question was provided by the Cogeneration Technology Alternatives Study (CTAS) sponsored by DOE/NASA (Reference D1) and conducted by two industrial contractors, GE (Reference D2) and United Technologies Corporation (Reference D3). It was found in this work that advanced systems do indeed increase fuel savings, reduce emissions and depending on application provide a reasonable return on investment.

The purpose of the present work was to examine CCMHD systems within the CTAS framework such that a comparison between CCMHD and other advanced concepts can be obtained. In addition, other applications of CCMHD in an industrial environment have been reviewed. The format of the present report outlines the CTAS methodology, introduces the CCMHD cogeneration system, compares CCMHD with other systems and finally, examines other industrial applications of CCMHD.

2. CTAS Methodology

CTAS was undertaken to evaluate the applicability of selected energy conversion systems (ECS) for cogeneration with a wide variety of industrial processes which require both electrical power and steam. Tables D-1 and D-2 list the ECS and industrial applications which were considered by GE in their study. It can be seen from these tables that a large number of combinations of ECS with various process industries are possible. Each ECS will provide different benefits for different industries, depending upon the electrical and steam requirements of the individual process under consideration. In order to compare these various combinations, CTAS was designed to be independent of the ECS and process which are being studied. The study methodology is outlined below.

Table D-1. Cogeneration Energy Conversion Systems Fuels
Evaluated and Fuel Flexibility (From Reference D-2)

	<u>Coal</u>	<u>Residual*</u>	<u>Distillate*</u>
Steam Turbine	FGD	Yes	OK
	AFB	-	-
	PFB	-	-
Gas Turbine	-	Yes	Yes
Combined-Cycle	-	Yes	OK
Combined-Cycle - Integrated Gasifier	Yes	-	-
Helium Gas Turbine	AFB	OK	OK
Thermionic Steam	FGD	Yes	OK
Stirling Cycle	FGD	Yes	Yes
Diesel	-	Yes	Yes
Phosphoric Acid Fuel Cell	-	-	Yes
Molten Carbonate Fuel Cell	-	-	Yes
Molten Carbonate Fuel Cell - Integrated Gasifier	Yes	OK	-

FGD - Flue Gas Desulfurization

AFB - Atmospheric Fluidized Bed

PFB - Pressurized Fluidized Bed

OK - Fuel Flexibility Indicator

* - Both Petroleum Base and Coal Derived Liquids

Table D-2. Selected Industry Processes and Summary of Energy Requirements (From Reference D -2)

Table 4-2
SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS

SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS											By-Product or					
SIC Code	Process No.	Description	Process Electric Power		Process Steam MBtu/hr	% Hot Water	Temperature		Power /Heat Ratio	Load Factor hrs/yr	Primary Fuel	Waste Fuel Avail MBtu/hr	National Energy Consumption Utility			
			MW _e	MBtu/hr			°F Peak	°F Avg.					+ Site 10 ¹² Btu/yr	1978	1985	2000
20 FOOD AND KINDRED PRODUCTS																
2011	1	Meat-Packing	1.940	6.625	24	40	250	250	0.28	2100	Gas		71	96	168	
2026	1	Fluid Milk	1.310	4.474	11	50	250	250	0.41	2100	Gas		71	80	101	
2046	1	Wet Corn Milling	28.500	97.327	659		250	250	0.15	6600	Gas *		104	141	159	
2063	1	Beet Sugar Refining	4.700	16.050	301		250	250	0.05	2800	Gas	76.47	100	118	162	
2082	1	Malt Beverage	6.040	20.627	86	60	250	250	0.24	6600	Gas		75	120	190	
22 TEXTILE MILL PRODUCTS																
2260	1	Textile Finishing	6.200	21.173	158		341	331	0.13	6240	Coal		75	75	75	
24 LUMBER AND WOOD PRODUCTS																
2421	1	Soft Wood-Lumber Sawmill	1.500	5.123	30		353	353	0.17	4000	Bark-Sawdust	41.2	237	300	400	
2436	1	Soft Wood-Plywood/Veneer	3.000	10.245	75		406	406	0.14	6000	Bark	100.0	100	150	275	
2492	1	Particle Board	5.000	17.075	37		406	406	0.46	8000	Natural Gas	41.2	32	100	172	
26 PAPER & ALLIED PRODUCTS																
2621	2	Bleached Kraft	50.000	170.750	780		366	340	0.22	8400	Coal	353	416	454	784	
2621	4	Unbleached Kraft	29.000	99.035	610		366	328	0.16	8400	Coal	259	405	441	950	
2621	6	Neutral Sulfide Semichemical	20.000	68.300	307		366	345	0.22	8400	Coal		63	69	128	
2621	7	Thermo-Mechanical Pulping	31.300	106.889	183		366	355	0.58	8400	Coal		102	110	205	
2621	8	Waste Paper	15.000	51.225	224		366	355	0.21	8400	Coal		176	191	419	
28 CHEMICAL & ALLIED PRODUCTS																
2800	1	Small Integrated Power Plant	32.500	110.923	1100			366	0.101	8760						
2800	2	Medium Integrated Power Plant	77.200	263.484	1054			366	0.25	8760						
2900	3	Large Integrated Power Plant	97.200	331.744	947			366	0.35	8760						
2812	1	Chlorine - Caustic Sods	120.000	409.800	265		338	311	1.55	8500	Any		180	240	300	
2813	1	Cryogenic Oxygen	34.000	116.110	0		0	0	999.99	8400	Electric		22	33	66	
2819	1	Alumina	30.290	103.440	980		495	434	0.11	8136	Coal-oil		53	76	135	
2821	2	Vinyl Chloride	4.000	13.660	207		422	373	0.07	8300	Gas		50	110	160	
2821	3	Low Density Polyethylene Resin	55.000	187.825	16		448	448	11.74	7900	Any		18	38	60	

* UNDERLINE INDICATES SIZE OF INTEREST FOR CCMHD

Table D-2. Selected Industry Processes and Summary of Energy Requirements (Continued)

Table 4-2 (Cont'd)
SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS

SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS															
Process			Process Electric Power		Process Steam	Temperature		Power	Load	Primary Fuel	By-Product or Waste Fuel Avail MBtu/hr	National Energy Consumption, Utility			
SIC Code	No.	Description	MW _e	MBtu/hr	MBtu/hr	% Hot Water	°F Peak	°F Avg.	/Heat Ratio			Factor hrs/yr	+ Site 1978	10 ¹² 1985	Btu/yr 2000
28 CHEMICAL & ALLIED PRODUCTS (Cont'd)															
2822	1	Styrene-Butadiene Rubber	7.500	25.612	35		338	338	0.73	7900	Any		7	9	13
2824	1	Polyester Fibre	32.000	109.280	30		406	406	3.64	7900	Gas-Oil*		30	55	75
2824	2	Nylon Fibre	11.000	37.565	23		274	274	1.63	8760	Any		14	20	25
2865	2	Cumene-Benzene	0.600	2.049	0		0	0	999.99	8400	Gas-Oil		6.5	10	15
2865	3	Phenol/Acetone	6.000	20.490	300		489	398	0.07	8200	Any		20	45	60
2865	4	Ethylbenzene	0.700	2.390	220		489	489	0.01	7900	Oil-Gas		22	45	65
2869	1	Methanol Synthesis	1.500	5.123	133		574	538	0.04	7880	Feedstock	352.9	0	0	0
2869	4	Ethanol	3.300	11.270	400		460	460	0.03	7900	Gas-Oil	70.6	18	24	30
2873	1	Ammonia Synthesis	3.500	11.952	640		598	598	0.02	8400	Gas-Oil		200	250	305
2874	1	Phosphoric Acid	4.000	13.660	92		353	292	0.15	7900	Gas-Oil		35	48	60
2895	1	Carbon Black	4.000	13.660	20		298	298	0.68	7900	Oil-Gas		18	20	24
29 PETROLEUM REFINING															
2911	1	Small Refinery	14.000	47.810	375		470	389	0.13	8760	Oil-Der		560	580	630
2911	2	Medium Refinery	52.000	177.580	1333		470	395	0.13	8760	Oil-Der		850	870	950
2911	3	Large Refinery	126.000	430.290	3042		470	385	0.14	8760	Oil		1220	1250	1280
32 STONE, CLAY AND GLASS															
3211	1	Flat-Glass	5.600	19.124	0		0	0	999.99	7500	Nat-Gas				
3221	1	Glass Containers	5.100	17.416	0		0	0	999.99	7500	Nat-Gas				
3229	1	Fress-Blown Glass	1.100	3.756	0		0	0	999.99	7500	Nat-Gas				
3241	1	Cement	20.316	69.379	0		0	0	999.99	7920	Coal				
33 PRIMARY METALS															
3312	1	Specialty Steel	60.000	204.900	93		448	446	2.20	6700	Nat-Gas		560	643	835
3325	1	Integrated Steel	280.000	956.200	912		448	445	1.05	8400	Cok-Coal	529.4	3080	3539	4596
3325	4	Mini-Steel	40.000	136.600	91		448	446	1.50	6700	Nat-Gas		360	612	1070
3331	1	Copper-Fire Smelted	24.800	84.692	0		0	0	999.99	8400	Oil		4.6	5.8	9.3
3331	4	CopperAnode Smelted	10.100	34.491	40		364	364	0.86	7620	Oil		12.2	15.5	24.8
3334	1	Aluminum	756.000	2581.740	0		0	0	999.99	8760	Oil		31.8	49.2	86.4

* UNDERLINE INDICATES SIZE OF INTEREST FOR CCMHD

Each ECS has a characteristic electrical power output associated with generation of a specific amount of steam under given conditions. The power/heat (P/H) ratio of the ECS can then be assumed constant where this constant may vary when the steam conditions are altered. In some cases, the P/H of an ECS can be altered by changing the system configuration. In parallel, each industrial process has fixed electrical and steam requirements which must be met in order to manufacture the desired product. Again the P/H ratio of the process can be assumed constant (see Table D-2). Pure cogeneration, then, is defined as that point where the P/H of the ECS exactly matches that of the process. Under these conditions, all output of the ECS, both electrical power and steam, are fed directly to the process.

Referring to Table D-2, it can be seen that industrial processes have a large range of P/H ratios.. Obviously, no ECS can meet all of these requirements. Thus, in order to match an ECS to a process, one of two strategies can be adopted. Namely,

- o Match the ECS electrical power output to that required by the process,
  - o When excess steam is required, install an auxiliary boiler.
  - o When excess steam is produced, install a cooling tower.
- o Match the ECS steam output to that required by the process.
  - o When excess power is required, purchase it from a utility.
  - o When excess power is produced, sell it to a utility.

These cases represent partial cogeneration where energy difficiencies are alleviated using supplementary sources and energy excesses are eliminated by sale or by rejection.

The CTAS methodology was based on a calculation of the P/H ratio for each

ECS as a function of output steam conditions. The ECS was then matched to the industrial processes using both the power match and heat match strategies outlined above. Fuel savings for each case considered were then computed using a fuel energy saved (FESR) ratio which is defined as shown in Table D-3. Computation of the FESR then allowed a comparison between various ECS for different cogeneration applications.

3. CCMHD Cogeneration Systems

Two thermodynamic cycles were examined for CCMHD cogeneration applications. The first is a standard MHD/steam generating cycle where heat is transferred from a combustion system to a regenerative argon heater which in turn transfers heat to argon for the generation of electric power and steam. The cycle diagram is presented in Figures D-1 and D-2 which show the argon and steam sides of the plant respectively. The state points are given in Table D-4.

The argon side is heated, MHD power is generated and the remaining components produce steam prior to precooling and argon compression. The steam side of the cycle is also simple and resembles a standard plant with the following exceptions:

- o Cogeneration steam is removed at the exhaust of the high pressure turbine.
- o The make up water stream is large to provide the steam for cogeneration.
- o The steam conditions are moderate in order to avoid purification of large quantities of water to high performance boiler specifications.
- o No reheat is included because the reheat turbine can be quite small depending on process requirements.

The state points shown in Table D-4 represent a system designed for pure cogeneration, i.e., all of the steam passing through the high pressure turbine is fed to the process. For this reason, the mass flow to and power from the low

Table D-3 Fuel Energy Saved Ratio (FESR)

$$FESR = \frac{FUEL_{NO\ COGENERATION} - FUEL_{COGENERATION}}{FUEL_{NOCGENERATION}}$$

FUEL_{NO COGENERATION}

NO EXPORT POWER

$$F_{NC} = \frac{P_{PROCESS}}{\eta_{UTILITY}} + \frac{H_{PROCESS}}{\eta_{BOILER}}$$

EXPORT POWER

$$F_{NC} = \frac{P_{PROCESS} + P_{EXPORT}}{\eta_{UTILITY}} + \frac{H_{PROCESS}}{\eta_{BOILER}}$$

FUEL_{COGENERATION}:

IMPORT POWER REQUIRED

$$F_C = \frac{H_{PROCESS}}{\eta_{STEAM}} + \frac{P_{PROCESS} - P_{CCMHD}}{\eta_{UTILITY}}$$

AUXILIARY BOILER REQUIRED

$$F_C = \frac{P_{PROCESS}}{\eta_{POWER}} + \frac{H_{PROCESS} - H_{CCMHD}}{\eta_{BOILER}}$$

Figure D-1. CCMHD Cogeneration System, Argon Side, Cycle 1

Figure D-2. CCMHD Cogeneration System, Steam Side Cycle 1

Table D-4. State Points for Cycle 1
P = 100 MWe, H = 181 MWt
Fuel: Montana Rosebud Coal

ARGON SIDE					
LOCATION	COMPONENT	TEMPERATURE (K)	PRESSURE (PSIA)	M (KG/SEC)	E MW
1	Argon Heater	1922	152.0	452.00	387.76
3	MHD Generator	1230	32.34	452.00	225.04
4	Splitter	1230	32.34	32.23	16.04
5	Splitter	1230	32.34	419.77	209.00
6	Superheater	1048	31.37	419.77	169.26
7	Deaerator HX	1043	31.34	419.77	168.16
8	Water Boiler	625	29.14	419.77	76.85
9	Water Walls	625	29.14	32.23	5.90
10	Mixer	625	29.14	452.00	82.75
11	Economizer	466	28.17	452.00	45.46
12	Economizer	344	27.70	452.00	16.54
13	Argon Cooler	302	27.57	452.00	6.78
14	Argon Compressor	482	160.00	452.00	49.14
STEAM SIDE					
20	Make Up Water	288	14.7	64.36	4.02
21	Make Up Water Pump	288	56.9	64.36	4.04
22	Mixer	288	56.9	64.42	4.05
23	Economizer	395	37.0	64.42	32.97
24	Deaerator	399	35.0	64.42	34.06
25	Boiler Feed Pump	399.5	1684.0	64.42	34.66
26	Steam-Water HX	436	1664.0	64.42	44.82
27	Economizer	561	1644.0	64.42	82.11
28	Splitter	561	1644.0	6.44	8.21
29	Splitter	561	1644.0	57.98	73.90
30	Water Walls	617	1628.0	6.44	18.36
31	Water Boiler	617	1628.0	57.98	165.21
32	Steam Drum	617	1628.0	64.42	183.57
33	Superheater	811	1465.0	64.42	223.31
34	High Pressure Turbine	543	210.0	64.42	191.38
35	Splitter	543	210.0	64.36	191.19
36	Steam-Water HX	478	200.0	64.36	181.03
37	Splitter	543	210.0	0.06	0.19
38	Low Pressure Turbine	308	0.82	0.06	0.14
39	Condenser	308	5.15	0.06	0.09
40	Condensate Pump	308	56.9	0.06	0.09

Table D-4. State Points for Cycle 1
P = 100 MWe, H = 181 MWt (continued)

ENERGY TRANSFER AND LOSSES

LOCATION	COMPONENT	E MW
2	MHD Generator Output	161.09
15	Main Compressor Power	79.28
16	Main Compressor Loss	2.31
17	Main Compressor Intercooling Rejection	34.62
41	Make Up Water Pump Power	0.02
42	Make Up Water Pump Loss	-0
43	Boiler Feed Pump Power	0.65
44	Boiler Feed Pump Loss	0.02
45	High Pressure Turbine Output	31.45
46	High Pressure Turbine Loss	0.48
47	Low Pressure Turbine Output	0.05
48	Low Pressure Turbine Loss	-0
49	Condenser Heat Rejection	0.05
50	Condensate Pump Power	-0
51	Condensate Pump Loss	-0
52	Heat Rejection Argon Precooler	9.76

pressure turbine are very small. In the present calculations, the MHD generator/diffuser was assumed to have an isentropic efficiency of 78% at an enthalpy extraction of 36%. When less power and more steam are required from the system, the generator enthalpy extraction can be reduced to match process requirements.

The combustor in Cycle 1 is a balanced combustion system which supplies heat through the argon heater to the remainder of the system, and is the only connection with the CCMHD/steam generating system. Thus, the combustor can be assumed to be an independent piece of equipment with a specified efficiency. For the present calculations, the combustor efficiency including argon heater losses was assumed to be 82.5%.

The second cycle is an interlinked feedwater cooled Brayton cycle. In this system, the combustor powers an MHD Brayton topping cycle in which excess heat in the argon is rejected to the feedwater. This eliminates heat rejection from the argon loop as is shown by the block diagram of Figure D-3. The flue gas from the argon heater enters a steam plant and then is used for coal drying. In contrast to the MHD/steam cycle, the combustor in Cycle 2 is fully integrated with the steam plant. In addition, the steam plant is coupled to the argon loop by means of the feedwater heaters.

Cycle 2 was analyzed using CCSYSA in order to evaluate its effectiveness in cogeneration systems. Table D-5 shows state points for the balanced cogeneration system at 20 MWe. Again, process steam is removed from a port in the turbine. An interesting feature of Cycle 2 is a bypass loop around the argon heater. This flow can be adjusted in order to regulate the power/steam ratio of the plant. Thus, Cycle 2 can be tuned to process requirements by adjusting both the enthalpy extraction and the bypass flow.

Figure C-3. Balanced Closed Cycle MHD Cogeneration Plant, Cycle 2

Table D-5 Balanced 20 MW Closed Cycle MHD Cogeneration
Plant with One Stage Atmospheric Combustion System, Cycle 2
Fuel: Montana Rosebud Coal

Description	Location	T (K)	P (psia)	$\dot{m}$ (Kg/s)	E (MW)
Coal (As Received)	1	288.0	14.7	3.45	91.089
Coal to Comb. (CDR)	2	288.0	14.7	3.28	91.065
Ambient Air	3	288.0	14.7	30.95	1.472
Air to Comb. (RH)	4	700.0	18.92	30.95	14.87
Flue Gas Recirc. (FGRB)	5	749.0	18.70	12.27	8.337
Slag Rejection	6	1644.0	-	0.23	0.276
Combustor (CB)	7	2116.5	18.54	46.27	113.44
Boiler Inlet Gas	8	1682.0	18.36	46.27	85.44
Boiler Outlet (RB)	9	1278.0	18.17	46.27	61.06
Superheater Outlet (SH)	10	1111.0	17.99	46.27	51.55
Economizer Outlet (EC)	11	740.0	17.81	46.27	30.92
Air Heater Outlet (AH)	12	417.0	17.63	34.00	10.10
Argon Regen. Outlet (ARHX)	13	1174.5	154.7	66.70	31.27
Argon Heater Outlet (HTRH)	14	1922.0	147.0	66.70	57.19
Diffuser Outlet (DF)	15	1345.0	40.1	66.70	37.19
Process Return	16	301.0	5.0	20.0	2.359
Argon Water Heater (ARFW)	17	222.0	1798.6	20.0	9.01
Radiant Boiler Outlet (RB)	18	700.0	1465.0	20.0	63.53
Process Flow	19	471.1	210.5	20.0	55.8
Heat Loss (CB)					0.557
Heat Loss (HTRH)					2.08
Heat Loss (ARPC)					2.01
Heat Loss (MC Intercooler)					0.493
Mech. Loss (MC)					0.030
Electrical Input (FGRB)					0.138
Electrical Input (AB)					0.781
Electrical Input (FP)					0.271

4. Cogeneration Results and Conclusions

A series of parametric calculations were performed in order to evaluate CCMHD cogeneration systems. The initial emphasis of the work was to obtain a comparison of CCMHD with both state of the art and other advanced concepts for the generation of electric power. Because of the availability of the CTAS results, the comparisons were made using the CTAS groundrules. During the course of the study, it became apparent that further comparisons with state of the art systems were necessary. These are discussed in Section 4.3

4.1 Groundrules for the Parametric Calculations

An examination (see Table D-1) of the requirements of the process industries which are of interest for cogeneration with CCMHD (electric power levels greater than 20 MWe) shows that:

- o $20 \text{ MWe} \leq \text{power level} \leq 280 \text{ MWe}$
- o $0.101 \leq \text{power/steam} \leq 11.74$
- o $250 \text{ F} \leq \text{average steam temperature} \leq 448 \text{ F}$

Because both the power levels and steam requirements cover a fairly narrow range, it was decided to perform all calculations for a nominal CCMHD power generating system where the selected conditions are shown in Table D-6. On the other hand, the power/steam ratio covers a wide range and this parameter was varied.

Calculations for comparison with the CTAS results were performed using Cycle 1. Cycle 2 was then introduced because it affords the possibility of adjusting both the combustion gas bypass flow and the generator enthalpy extraction such that an exact match to process requirements can be achieved

Table D-6. Ground Rules for Parametric Calculations

	Cycle 1	Cycle 2
Total Net Electric Output (MWe)	100	Variable
Plant Auxiliary Load (MWe)	12.5	Variable
Steam Output (MWe)	Variable	55.80
Steam Conditions		
Temperature (F)	400	388
Pressure (psia)	200	210
Steam Superheat (F)	18.2	2.1

using a single energy source. This eliminates the need for an auxiliary boiler when excess steam is required and eliminates the purchase of electricity for large power/steam ratios.

4.2 Comparison with CTAS Results

The key parameters for the evaluation of an energy conversion system (ECS) for use in cogeneration applications are the power/steam ration (P/H) and the efficiencies of power generation (P/F), steam generation (H/F) and power and steam ($P + H/F$) generation. When the power/steam ratio of the process equals that of the ECS, the ECS and process are exactly matched and all steam which is generated is utilized by the process. When a mismatch in P/H occurs, either excess power or steam is generated or produced depending on the strategy used to match the ECS to the process (see Section 2). The need for efficiencies is obvious in that they define the fuel requirements for any given cogeneration system.

Table D-7 (Reference D-2) compares CCMHD with other ECS. Examination of these results shows that CCMHD is very close in performance to a coal fired Stirling cycle. The coal burning state of the art system (FGD STM TURB-COAL) is considerably more efficient in overall operation ($P + H/F = 0.85$ for FGD STM TURB COMPARES TO $P + H/F + 0.69$ for CCMHD), but produces about half the electrical power for a fixed steam requirement ($P/H = 0.20$ for FGD STM TURB compares to $P/H = 0.55$ for CCMHD). From these results, it can be concluded that CCMHD has no application for processes with $P/H \leq 0.20$ and that CCMHD will be most attractive for electric power intensive industries.

Table D-7. Cogeneration Energy Conversion System (ECS) Performance Characteristics
(From Reference D-2)

<u>ECS</u>	Performance Characteristics at Process Sat. Steam = 350 F*			
	<u>Power Heat</u>	<u>Power Fuel</u>	<u>Process Heat Fuel</u>	<u>Power + Process Heat Fuel</u>
<u>Current State-of-the-Art</u>				
FGD STM TURB - COAL	0.20	0.14	0.71	0.85
GT-HRSG - RESIDUAL	0.68	0.29	0.43	0.72
DIESEL-HRSG - RESIDUAL	2.03	0.36	0.18	0.54
<u>Advanced</u>				
AFB STM TURB - COAL	0.20	0.14	0.71	0.85
PFB STM TURB - COAL	0.32	0.21	0.64	0.84
INT GAS COMB CYCLE - COAL	0.66	0.28	0.43	0.71
INT GAS FUEL CELL MC - STM TURB	0.96	0.38	0.40	0.78
STIRLING - COAL	0.54	0.26	0.47	0.73
CLOSED CYCLE GT HELIUM - COAL	0.36	0.18	0.49	0.67
THERMIONIC-STM TURB - COAL	0.44	0.26	0.59	0.84
GT-HRSG - RESIDUAL	0.66	0.31	0.46	0.77
COMB CYCLE GT - RESID	1.08	0.37	0.34	0.72
STM INJ GT - RESIDUAL	2.70	0.36	0.13	0.49
DIESEL - RESIDUAL	1.75	0.37	0.21	0.58
DIESEL-HEAT PUMP - RESIDUAL	0.78	0.33	0.43	0.76
REGEN GT - DISTILLATE	0.85	0.33	0.39	0.72
FUEL CELL - DISTILLATE	2.24	0.38	0.17	0.55
FUEL CELL MC - DIST.	1.77	0.41	0.23	0.65
Performance Characteristics at Process Steam P = 200 PSIA, T = 400 F				
CCMHD CYCLE 1	0.55	0.24	0.44	0.69
CCMHD CYCLE 2	0.36	0.21	0.60	0.81

*Performance characteristics of most ECS's varies with process steam temperature

There are three major reasons for the poor thermodynamic showing of CCMHD systems in comparison to steam turbines. Firstly, the argon heater represents a significant loss which is not present in a conventional boiler system (85% boiler steam generation efficiency compares to 82.5% combustion efficiency assumed in the present study). Secondly, the electrical auxiliaries necessary to operate a CCMHD plant consume considerable power - 2 points electrical efficiency. Finally, heat must be rejected in the argon loop in order to maintain a reasonable exit temperature for the argon leaving the main compressor.

Figures D-4 to D-9 (Reference D-2) compare CCMHD with other ECS studied for three P/H values (0.1, 0.25 and 1.0) for both the power and heat match strategies. Curves for the other systems represent a range of required steam temperatures while the black square labeled CCMHD is for the conditions cited in Table D-6. Maximum theoretical fuel energy savings ratio (FESR) were computed assuming $P + H/F = 0.85$ for the ECS, an efficiency which arises from considerations of combustor and stack losses. It can be seen from the comparisons on the figures that the performance of the CCMHD system (black squares) compares closely to the coal fired Stirling cycle (curve labeled 4) and that in all cases other systems provide greater fuel savings. Further, for low P/H ratios, the steam turbine (curve labeled 1) is very difficult to beat in terms of FESR. Finally, the maximum expected fuel savings for CCMHD cogeneration systems which are configured in thermodynamic Cycle 1, is about 25% for the P/H ratios which were studied in the comparisons with CTAS.

WHEN P/H MATCHES STEAM TURBINE CHARACTERISTIC -

STEAM TURBINE LOOKS UNBEATABLE

Figure D-4 Fuel Energy Saving Potential of Energy Conversion Systems When Matched to Industrial Process Heat Needs (P/H = 0.1)

Figure D-5. Fuel Energy Saving Potential of Energy Conversion Systems When Matched to Industrial Process Power Needs (P/H = 0.1)

Figure D-6. Fuel Energy Saving Potential of Energy Conversion Systems When matched to Industrial Process Heat Needs (P/H = 0.25)

Figure D-7. Fuel Energy Saving Potential of Energy Conversion Systems When Matched to Industrial Process Power Needs (P/H = 0.25)

Figure D-8. Fuel Energy Saving Potential of Energy Conversion Systems When Matched to Industrial Process Heat Needs (P/H = 1.0)

Figure D-9. Fuel Energy Saving Potential of Energy Conversion Systems When Matched to Industrial Process Power Needs (P/H = 1.0)

4.3 General Conclusions

Evaluation of the thermodynamic usefulness of an ECS for cogeneration applications depends on the system with which it is compared. For example, if an efficient ECS cogeneration system is compared with an inefficient boiler/utility system the cogeneration system will result in significant fuel savings. On the other hand, if an ECS is compared with high technology, advanced boilers and power generating systems, the cogeneration system will have a poor showing. This section emphasizes alternative comparison strategies for evaluation of cogeneration systems.

Every ECS which produces both electrical power and steam has a natural operating point where the system is balanced i.e., for a fixed electrical power level all of the steam which is generated is used by the process. The balance point will change with output steam conditions and in some cases may be adjusted by varying system parameters, such as the enthalpy extraction in CCMHD systems. Conceptually, industries which require more steam than is produced by a given ECS, can be modeled by a cogeneration system supplemented with an auxiliary boiler, i.e., a parallel thermodynamic cycle. Conversely, power can be purchased when the ECS electrical output is insufficient for a fixed steam requirement. In either case, the overall system is a linear combination of a cogeneration system and a parallel boiler or electric utility. Further, multiple cogeneration systems can also be combined linearly in order to best match industrial process requirements. Thus, for each industry a linear combination of a boiler, cogenerating steam turbines, other cogenerating systems and utility power can be assembled such that the process requirements are matched exactly and the fuel efficiency is optimum.

Evaluation of fuel usage for each ECS to be combined in the overall system can be modeled using the following definition of efficiency:

$$\eta_{TH} = \frac{Q_{th}}{Q_{in}}$$

$$\eta_e = \frac{P_e}{Q_{in}}$$

where

Q_{th} = energy flow of output steam

P_e = electrical power output

Q_{in} = thermal input

Q_{th} changes with steam output conditions and η_{th} and η_e depend on the characteristics of the specified ECS (see Table D-7). Thus, an optimum cogeneration configuration can be identified by taking a linear combination of desired ECS of various sizes and minimizing fuel consumption in order to obtain the size of each ECS necessary for the system.

The above described optimization procedure has been performed using the assumed efficiencies of Table D-8 for the following combinations:

- o Boiler/utility (B/U)
- o Boiler/steam turbine/utility (B/ST/U)
- o Boiler/steam turbine/CCMHD/Utility (B/St/CCMHD/U)

The results of this analysis are plotted in Figures D-10 and D-11 for utility efficiencies of 0.32 and 0.37, respectively, with the former being representative of CTAS assumptions and the latter representative of modern utility plants. The lines at the top of the figures are characteristic requirements of various industrial processes. Consequently, extension of the lines, allows

Table D-8. Efficiencies* for Cogeneration Calculations

	η_{th}	η_e
NO COGENERATION	0.85**	0.32** , 0.37
STEAM COGENERATION	0.743	0.0966
CCMHD CYCLE 1	0.441	0.244
CCHMD CYCLE 2	0.580 - 0.602***	0.204 - 0.230***

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*Steam condition 400 F saturated

**Efficiencies assumed in CTAS

***Generator enthalpy extraction varied from 20-40%

PROCESS STEAM CONDITIONS 400 F, SATURATED

Figure D-10. Thermal Input Requirements as a Function of Process Electrical Requirements for Various Cogeneration Strategies. Utility Efficiency = 32%

Figure D-11. Thermal Input Requirements as a Function of Process Electrical Requirements for Various Cogeneration Strategies. Utility Efficiency = 37%

the reader to determine energy requirements for the different cogeneration strategies. This has been done and the results are tabulated in Table D-9.

The initial conclusion to be drawn from the figures is that cogeneration requires less fuel than a boiler/utility combination. This is caused by the fact that some portion of the electrical requirements of the process are generated at very high efficiency under conditions of cogeneration. A second feature of the curves is that energy savings associated with cogeneration become constant at that power level when all of the steam required by the process is used to generate electricity. This means that percentage fuel savings first increase and then decrease with increasing power requirements as shown by Table D-10. Finally, the fuel savings realized by cogeneration decrease with increasing utility efficiency. This point is further illustrated by the graph shown in Figure D-12.

Comparison of fuel savings realized with a steam turbine to that of a steam turbine/CCMHD system shows that the FESR is a function of power level. At low electrical power requirements (petroleum), cogeneration with a steam turbine nearly equals that using CCMHD. This is because the steam turbine is operating near its balance point and does not require substantial utility power. At higher power levels where more electricity must be purchased from a utility for the steam turbine systems, the addition of CCMHD increases fuel savings significantly. In fact, the largest fuel savings of all are realized when CCMHD can be operated in mode of pure cogeneration using thermodynamic Cycle 2. Finally, when the natural balance point for CCMHD Cycle 2 is exceeded, the fuel savings become constant. This shows that the increased electrical output of CCMHD Cycle 1 at the sacrifice of process steam can not be justified on a fuel savings basis.

Table D-9. Fuel/Steam Requirements for Various Industries

	PETROLEUM	KRAFT PAPER	CHEMICALS	THERMO- MECHANICAL PULPING	INTEGRATED STEEL	CHLORINE
B/U ₃₂	1.614	1.864	2.270	2.989	4.458	6.020
B/ST/U ₃₂	1.377	1.627	2.033	2.752	4.221	5.783
B/ST/CCMHD/U ₃₂	1.361*	1.480	1.673	2.369	3.837	5.400
B/U ₃₇	1.555	1.771	2.122	2.744	4.014	5.366
B/ST/U ₃₇	1.373	1.589	1.940	2.562	3.832	5.184
B/ST/CCMHD/U ₃₇	1.361**	1.480	1.673	2.277	3.548	4.899

B = Boiler

U₃₂ = Utility 32% Efficiency

U₃₇ = Utility 37% Efficiency

ST = Steam Turbine

---

*Cogeneration using B/CCMHD/U₃₂

**Cogeneration using B/CCMHD/U₃₇

Table D-10. Fuel Saved Energy Ratio (%) for Various Industries

	PETROLEUM	KRAFT PAPER	CHEMICALS	THERMO- MECHANICAL PULPING	INTEGRATED STEEL	CHLORINE
B/ST/U ₃₂	14.7	12.7	10.4	7.9	5.3	3.9
B/ST/CCMHD/U ₃₂	15.7*	20.6	26.3	20.7	13.9	10.3
B/ST/U ₃₇	11.7	10.3	8.6	6.6	4.5	3.4
B/ST/CCMHD/U ₃₇	12.5**	16.4	21.2	17.0	11.6	8.7

B = Boiler

U₃₂ = Utility 32% Efficiency

U₃₇ = Utility 37% Efficiency

ST = Steam Turbine

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*Cogeneration using B/CCMHD/U₃₂

**Cogeneration using B/CCMHD/U₃₇

Figure D-12. Thermal Input Requirements as a Function of Process Electrical Requirements for Various Utility Efficiencies

The question of the addition of a high technology ECS such as CCMHD to a steam turbine cogeneration system must be answered on an industry by industry basis. Clearly, fuel will be saved using the high technology system if the power requirements of the industry are sufficiently large. However, when the natural cogeneration limit of the ECS is exceeded, energy savings become constant and the percentage energy saved decreases with increasing power requirements. Further, in the range where CCMHD appears attractive, the difference between CCMHD and steam turbine energy savings is not more than 18% and 14% for utility efficiencies of 32% and 37%, respectively. This is shown in Table D-11 which computes an FESR for CCMHD assuming steam turbine cogeneration is the base comparison. Table D-11 also shows that CCMHD provides more than 10% savings in only two industries, chemicals and thermo-mechanical pulping, when compared with steam turbine cogeneration. It can be concluded then that CCMHD will not find application in a wide variety of industries.

5. Trigeneration

Trigeneration is a terminology which has been coined to denote industries wherein the preparation of an energy intensive feedstock is necessary. Thermodynamically, cogeneration recognizes that a fuel must be burned in order to generate steam and questions what is the most optimum manner in which to burn the fuel such that electrical power can be generated at the same time with high efficiency?

In comparison, trigeneration recognizes that feedstock chemicals must be produced in order to make a product and questions is there a way to make these chemicals such that both electricity and steam can be produced? Of course, the electricity and steam must be required by the process. Thus,

Table D-11. Differences between Fuel Saved Energy Ratio (%) of CCMHD and Steam Turbine Cogeneration Systems

	PETROLEUM	KRAFT PAPER	CHEMICALS	THERMO- MECHANICAL PULPING	INTEGRATED STEEL	CHLORINE
UTILITY EFFICIENCY= 32%	1.2	9.0	17.7	13.9	9.1	6.6
UTILITY EFFICIENCY= 37%	0.9	6.9	13.8	11.1	7.4	5.5

B = Boiler

U₃₂ = Utility 32% Efficiency

U₃₇ = Utility 37% Efficiency

ST = Steam Turbine

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*Cogeneration using B/CCMHD/U₃₂

**Cogeneration using B/CCMHD/U₃₇

cogeneration supplies process electricity or steam with very little energy penalty whereas trigeneration supplies both electricity and steam while manufacturing a necessary product.

An example best serves to define the term trigeneration. Many industries require CO or H₂ for the manufacture of end products such as ammonia, methyl alcohol, plastics and chemicals. Because catalysts are necessary to enhance the chemical reactions of the process, the CO or H₂ must be very pure. Assuming that CO is to be produced by the gasification of coal, H₂ can be manufactured by the shift reaction.

All products must be cooled prior to purification because of the stringent requirements for and the technology used impurity removal. If the latent heat from the hot products exiting the gasifier can be utilized for the generation of electric power and steam, the system would be operating under conditions of trigeneration as illustrated schematically in Figure D-13.

The hot raw gas produced by most gasifiers is relatively cool and cannot be used for the efficient generation of power and steam. The trigeneration problem, then becomes one of defining an efficient gasification system which has an exit temperature which is sufficiently high to be useful. Advance, slagging gasifiers such as those being developed by the Grand Forks Energy Technology Center ^{D4, D5} and the British Gas Corporation ^{D6, D7} do provide an exit temperature (2500 F) which is realistic for steam turbine trigeneration. However, the temperature is too low to be of interest for CCMHD. A conceptual gasification system which shows considerable promise for cost reductions over standard units has been examined briefly by GE. This unit is based on

Figure D-13. Flow Diagram of a Trigeneration System

the GE cyclone combustor concept operated sub-stoichiometrically. Initial tests indicated that gasification could be achieved but that existing hardware was far from optimum. Further, there are indications that air enrichment will be beneficial because this will produce higher temperatures in the gasification chamber, thus , reducing residence time requirements and size. Under these conditions, trigeneration with CCMHD becomes feasible.

Given a high temperature gasification system, trigeneration is thermodynamically identical to cogeneration, i.e., what is the most effective way to convert heat to electric power and steam? All comparisons between CCMHD and steam turbines which were given in the previous section apply to trigeneration as do the conclusions. Thus, the use of CCMHD for trigeneration becomes industry dependent and it can be concluded that CCMHD will not find a wide variety of applications for trigeneration systems.

6. Summary and Review of Conclusions

A study has been undertaken in order to define the usefulness of CCMHD for industrial cogeneration applications. Initial work emphasized a comparison between CCMHD and other energy sources. This was achieved within the framework of the Cogeneration Technology Alternative Study (CTAS) which is reviewed in Section 2 of this Appendix. Two CCMHD/cogeneration thermodynamic cycles were examined using the GE systems code, CCSYSA, to evaluate cycle efficiencies. The cycles are described and state points for the systems are presented.

Results computed for the two cycles show that the performance of CCMHD is close to that of a coal fired Stirling cycle and that other advanced cycles can provide higher fuel savings. Further, a comparison of CCMHD with

with cogeneration using a steam turbine shows that CCMHD has no application for industries where the required power/steam ratio is less than 0.2. For other industries, CCMHD can save as much as 26% of the fuel used for an equivalent steam boiler/utility combination. However, under these same conditions, cogeneration with a steam turbine saves 10% of the input fuel. Thus, when compared with steam turbine cogeneration, CCMHD does not appear as promising as when compared to the non-cogeneration case. Energy saved is a strong function of the process power/heat ratio. Consequently, CCMHD will have limited application in industry. In fact, only two of the industries studied in CTAS show CCMHD energy savings of greater than 10% when compared to steam turbine cogeneration.

In addition to cogeneration, trigeneration was also examined. Here, an energy intensive feedstock such as H_2 or CO is required to manufacture a product. It was found that if the feedstock is produced by coal gasification and if the gasifier produces a raw gas of sufficient temperature, the heat of the gas can be utilized in order to produce electric power and heat. The same conclusions apply to CCMHD trigeneration as cited above for cogeneration.

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GENERAL ELECTRIC
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PROGRAM INFORMATION REQUEST / RELEASE

*CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV. LTR.
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FROM R. R. LAESSIG		TO C. S. COOK	
DATE SENT AUGUST 5, 1980	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.
SUBJECT HEAT EXCHANGER FOR ARGON/COMBUSTION GAS			

INFORMATION REQUESTED/RELEASED

1. SUMMARY

A heat exchanger for the exchange of 100 MW thermal energy was sized to heat argon from 700 F to 3000 F with combustion gases at 1000 F to 3350 F. Both gases are approximately at 10 atmospheres absolute pressure. A heat exchanger unit similar to the Hague International unit was used. Approximately 49,000 tubes are required which are housed within 10 containers with approximately 15 feet OD and 60 feet high. Total HX-tube volume is approximately 8730 ft³, the weight is 1.2 x 10⁶ lbs. Figure 1 shows a possible configuration of the HX.

2. HEAT TRANSFER CALCULATIONS

2.1 THE HEAT EXCHANGER

As heat exchanger tube the Hague International Tube configuration was used.

2.2 REQUIREMENTS

The heat exchanger was layed out for the following requirements:

$$Q = 100 \text{ MW}_{\text{Thermal}} = 341.3 \times 10^6 \text{ Btu/hr}$$

Combustion Gas

$$p \approx 10 \text{ atmospheres}$$

$$C_p = 0.3 \text{ Btu/lb F}$$

$$\text{Max. pressure drop } \Delta P/P \leq 5\%$$

$$T_{\text{inlet}} = 3350 \text{ F}, T_{\text{outlet}} = 1000 \text{ F}$$

	PAGE NO.	RETENTION REQUIREMENTS	
	1 8 OF	COPIES FOR	MASTERS FOR
		☐ 1 MO.	☐ 3 MOS.
		☐ 3 MOS.	☐ 6 MOS.
		☐ 6 MOS.	☐ 12 MOS.
		☐ MOS.	☐ MOS.
		☐	☐ DO NOT DESTROY

Argon

$p \approx 10$ atmospheres (slightly higher than combustion side)

$$c_p = 0.125 \text{ Btu/lb } ^\circ\text{F}$$

Maximum pressure drop $\Delta P/P \leq 10\%$

$$T_{\text{inlet}} = 700^\circ\text{F}, T_{\text{outlet}} = 3000^\circ\text{F}$$

2.3 MASS FLOW

Combustion Gas

$$\dot{m} = \frac{Q}{c_p \Delta T} = \frac{341 \times 10^6 \text{ Btu/hr}}{0.3 \text{ Btu/lb } ^\circ\text{F} \times 2350 ^\circ\text{F}} = 0.484 \times 10^6 \text{ lb/hr}$$

Argon

$$\dot{m} = \frac{Q}{c_p \Delta T} = \frac{341 \times 10^6 \text{ Btu/hr}}{0.125 \text{ Btu/lb } ^\circ\text{F} \times 23000 ^\circ\text{F}} = 1.19 \times 10^6 \text{ lb/hr}$$

2.4 HEAT TRANSFER INSIDE TUBES (ARGON)

Configuration: 3240 parallel tubes 1.125 ID

$$\rho = 1.11 \frac{T_o}{T} \quad \text{lb/ft}^3$$

$$T_o = 273 ^\circ\text{K} = 492 ^\circ\text{R}$$

$$T = \frac{T_i + T_o}{2} = \frac{700 + 3000}{2} = 1850 ^\circ\text{F}$$

thus

$$\rho_{1850^\circ\text{F}} = .233 \text{ lb/ft}^3$$

Flow velocity $\bar{v} = \dot{m}/\rho A$ for 1.125" ID
for 1.19×10^6 lb/hr

$$A = N \cdot A_{tube} = \frac{3240 \times .994}{144} = \underline{22.4 \text{ ft}^2}$$

$$\bar{v} = \frac{1.19 \times 10^6 \text{ lb/hr}}{3600 \frac{\text{sec}}{\text{hr}} \times .233 \frac{\text{lb}}{\text{ft}^3} \times 22.4} = \underline{63.4 \text{ ft/sec}}$$

Heat Transfer

$$Nu = \frac{h \cdot D}{k} = .036 Re_a^{.8} Pr_a^{1/3} \left(\frac{\mu_a}{\mu_w} \right)^{.14} \left(\frac{D}{L} \right)^{1/8}$$

We assume a 150 F drop from wall to coldest temperature

$$T_a = \frac{T_i + T_o}{2} = 1850^\circ\text{F} \quad T_w = 2000^\circ\text{F} \quad \nu_a = .155 \frac{\text{ft}}{\text{ft} \cdot \text{hr}}$$

$$Re_a = \frac{v \cdot d \cdot \rho}{\mu_a} = \frac{63.4 \times 1.125 \times .233 \times 3600}{12 \times .155} = \underline{32,165}$$

$$Pr_a = \frac{c_p \mu}{k} = \frac{.125 \times .155}{.02875} = \underline{.674}$$

$$\frac{\mu_a}{\mu_w} = \frac{.155}{.16} = \underline{.969}$$

$$\frac{D}{L} = \frac{1.125}{48} = \underline{.0234}$$

Thus

$$Nu = .036 \times 32,165^{.8} \times .674^{1/3} \times .969^{.14} \times .0234^{1/18}$$

$$Nu = 103.5$$

$$h = Nu \cdot \frac{k}{D}$$

$$h = 103.5 \frac{.02875 \times 12}{1.125} = \underline{31.75} \text{ BTU/ft}^2 \text{ HR } ^\circ\text{F}$$

2.5 HEAT TRANSFER OUTSIDE TUBES (COMBUSTION GASES)

Configuration:

Total fin area for 48" long tube 10.5 ft²

Flow area for combustion gases

$$A_{min} = (1.5 \times .16 \times 3 + .25) \times 40 \times 78$$

" "
" "

left
Number of tubes per row

Thus $A_{min} = 21 \text{ ft}^2$ between tubes
 $A_{max} = 125 \text{ ft}^2$ between rows

with $m_{\text{total}} = .5 \times 10^6 \text{ lb/hr} = 139 \text{ lb/sec}$

$$m_{HX} = \frac{m_{total}}{10} = \underline{13.9 \text{ lb/sec}}$$

and $T_m = \frac{1000 + 3350}{2} = 2175^\circ\text{F}$

and $\rho_m = \rho_o \frac{T_o}{T} = 0.146 \frac{\text{kg}}{\text{ft}^3}$

thus: $V_{max} = \frac{13.9}{.146 \times 21} = 4.5 \text{ ft/sec}$

$$V_{min} = \frac{13.9}{.146 \times 125} = .76 \text{ ft/sec}$$

for finned tubing $Nu = .174 Re^{.618}$ (JAKOB)

$$Re = \frac{.076 \text{ m} \cdot 1.34 \text{ m/hr}}{20.9 \times 10^{-6} \text{ m}^2/\text{hr}} = 4,873$$

$$Nu = 4,873^{.618} \times .174 = \underline{33.1}$$

$$h_{max} = 33.1 \frac{.073}{.076} = 31.8 \frac{\text{Kcal}}{\text{m}^2 \text{ hr } ^\circ\text{C}} = \underline{6.6 \frac{\text{BTU}}{\text{HR FT}^2 ^\circ\text{F}}}$$

$$h_{eff} = .8 h_{max} = \underline{5.3 \frac{\text{BTU}}{\text{HR FT}^2 ^\circ\text{F}}}$$

2.5 HEAT CONDUCTION THROUGH WALL

wall conductivity $k = 7.5 \frac{\text{BTU}}{\text{HR FT } ^\circ\text{F}}$

$$\begin{aligned} d_o &= 2.25 \\ d_i &= 1.125 \end{aligned}$$

$$L = 3.7'$$

$$K = \frac{\ln \frac{2.25}{1.125}}{7.5 \cdot 2\pi \cdot 3.7} = 4 \times 10^{-3} \left(\frac{\text{BTU}}{\text{HR } ^\circ\text{F}} \right)^{-1}$$

2.6 TOTAL HEAT TRANSFER

base everything on inside tube area A_i (1.125")

$$Q = \frac{1}{\frac{1}{h_1 A_1} + \frac{\ln \frac{d_o}{d_i}}{k 2\pi L} + \frac{1}{h_2 A_2}} \Delta T$$

$$\text{or } \frac{Q}{A_i \Delta T} = \frac{1}{\frac{1}{h_1} + \frac{A_i \ln \frac{d_o}{d_i}}{k 2\pi L} + \frac{A_i}{A_2} \frac{1}{h_2}} = h_{eff}$$

$$\frac{Q}{A \Delta T} = h_{eff} = 18.1 \frac{\text{BTU}}{\text{FT}^2 \text{ } ^\circ\text{F}}$$

$$\text{Thus } Q_{\text{tube}} = h_{eff} \cdot A_{\text{tube}} \Delta T = 18.1 \times 1.08 \times 325 = 6369 \frac{\text{BTU}}{\text{HR ft}_2}$$

Required number of tubes

$$N_{\text{req}} = \frac{314 \times 10^6}{6369} = 49,300$$

Number of tubes used for calculation :

$$3240 \times 14 = 45,360$$

This is close enough for a first cut

FIG. 1
HEATEXCHANGER
10 MW thermal

Approx. Tubeweight 120 K. L.
324 tubes in parallel

PLANT EFFICIENCY AS A FUNCTION OF
OUTPUT POWER FOR OPEN CYCLE
AND CLOSED CYCLE MHD POWER PLANTS

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1.0 INTRODUCTION

The curves of Open Cycle MHD and Closed Cycle MHD power plant efficiencies as a function of output power shown in Figure 1 were presented in a rapporteur statement by B. Zauderer at the 7th International Conference on MHD Power Generation. This note is in response to several requests for details as to how the curves were calculated. Some additional information has also been included along with results for a very promising modification of the CCMHD plant configuration.

2.0 BASIS FOR CURVES

Simple energy flow models similar to those used by Heywood and Womak¹ were the basis for the curves shown in Figure 1. However, the evaluation of key parameters was based in part on data from recent detailed system evaluations^{2,3,4}. Results using models of this type are dependent on assumptions and input data. They are consistent with GE results for more detailed calculations. The difference in slope between OCMHD and CCMHD plants is due in large part to the reduced size sensitivity of the Closed Cycle MHD Generator, which operates with much lower heat transfer to the walls than an Open Cycle Generator. In both cases however, calculated steady state performance of the MHD generator is a substantial extrapolation from existing data and the viability of either system is critically dependent on the success of planned experiments.

2.1 OCMHD MODEL

The energy flow for the Open Cycle MHD case is shown in Figure 2, with efficiency defined as:

$$\eta = [\eta_x Q_1 (1 + \beta) + P_8] / Q_1$$

where

η_x = MHD enthalpy extraction (electrical output/thermal input, 77°F reference)

Q_1 = thermal input from coal

β = fraction of coal thermal input in preheated air

P_8 = net electrical output of steam turbine

Evaluation proceeded as follows:

1. η_x was taken from Figure 1-8, Reference 4.
2. Q_1 was chosen arbitrarily to provide a range of output power.
3. A value for β of 0.114 was determined from data developed in Reference 2 for:
  - o Montana Rosebud Coal
  - o 38 mole % O_2 in oxidizer
  - o Ratio of oxidizer mass flow rate to coal mass flow rate = 4.6
  - o Oxidizer preheat 1200 F
4. Total steam turbine output was determined by calculating Q_6 (thermal energy to steam), using a linear interpolation of coal pile to bus bar efficiencies for stand-alone steam turbine power plants shown in Table 1 and dividing by an assumed boiler efficiency. The value of Q_6 was established as coal thermal input, Q_1 , plus compressor power minus MHD power minus 0.1 Q_1 (stack loss). Compressor power was calculated using a diffuser pressure recovery coefficient of 0.5, and MHD generator isentropic efficiency from Table 1-12, Reference 4. Finally, net steam turbine power, P_8 , is total steam turbine output minus compressor power and minus plant auxiliary power requirements.

TABLE 1

STEAM PLANT CYCLE EFFICIENCIES USED FOR OCMHD
EFFICIENCY CALCULATOR (BOILER EFF = 0.88)

<u>MHD PLANT SIZE</u> <u>(MWe)</u>	<u>REHEAT</u>	<u>NO REHEAT</u>	<u>P_{aux}</u> [*]
100	.358	.329	9
300	.360	.329	19
1000	.368	-	43

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* Subtracted from Steam Plant Output. Includes Oxygen Plant.

2.2 CCMHD MODEL

The Closed Cycle MHD system utilizes cesium seeded argon gas as the working fluid. Peak argon temperature is approximately 3000 F, with efficient generator operation achieved through non-equilibrium electrical conductivity. Several cycle arrangements are possible but the MHD/steam cycle shown in Figure 3 is the basis for the energy flow model used here. A coal fired combustor supplies thermal energy to an array of regenerative ceramic heat exchangers (HTRH). Argon exiting the MHD generator is used to raise steam in a heat recovery/seed recovery (HRSR) unit which is analagous in function to the HRSR component of an open cycle MHD system. It is, however, substantially different in design since: a) no slag is present, b) heat transfer is primarily convective, c) operating pressure is greater than atmospheric.

Heat is rejected in the argon cooler (CLR) and the argon is compressed prior to reentering the HTRH. The plant is "balanced" in the sense that shaft power from the steam turbine is just sufficient to drive the argon compressor, thus all the plant output power comes from the MHD generator.

For the system shown in Figure 3 all thermal energy is transferred through the HTRH*, so overall efficiency is the product of MHD/steam cycle efficiency and a combustor/HTRH efficiency defined as the ratio of thermal input to argon to coal thermal input. Efficiency for the Closed Cycle System of Figure 3 is defined as

$$\eta = \left[\frac{\eta_c}{Q_3 - Q_{10}} \right] \left\{ \eta_x [(Q_3 - Q_{10}) + \eta_{stm} Q_5 + Q_9] - P_{aux} \right\}$$

*For some gasifier configurations considered in the PACC Study (Reference 3) steam is also raised in the gasifier and the situation is more complex.

where

η_c = combustor/HTRH efficiency

$Q_3 - Q_{10}$ = thermal input to argon

η_x = enthalpy extraction (MHD output divided total thermal power, 0°R reference)

η_{stm} = steam plant cycle efficiency

Q_5 = thermal power to steam plant

Q_9 = thermal power in low temperature argon

P_{aux} = electrical auxiliary power

The model was evaluated as follows:

1. η_c was held constant at 0.815 on the basis of results from Reference 3.
2. $Q_3 - Q_{10}$ was chosen arbitrarily to provide a range of output power. Temperatures T_3 and T_{10} were held constant at 3000 F and 710 F respectively, thus argon mass flow rate could be computed from $Q_3 - Q_{10}$.
3. η_x was fixed by the constraint of balanced operation. Since argon is a perfect gas, enthalpy extraction, isentropic efficiency (η_s) and pressure ratio (PR) are uniquely related. Values used for this calculation are shown in Table 2.
4. Q_5 was back calculated from steam plant cycle efficiency and required compressor power, assuming a compressor isentropic efficiency of 0.89 plus 0.5% mechanical loss. The result was checked for consistency with energy transferred from argon in the HRSR. Steam plant cycle efficiency varied with size as indicated in Table 2.
5. Q_9 was determined for $T_9 = 84$ F, with mass flow rates determined by $Q_3 - Q_{10}$.
6. Auxiliary power was as shown in Table 2.

TABLE 2

VALUES OF PARAMETERS USED FOR CCMHD PLANT EFFICIENCY CALCULATION

MHD PLANT SIZE (MWe)	η_x	η_s	PR	η_{stm}	P_{aux} (MWe)
100	.335	.775	4.12	0.339	9
600	.352	.782	4.46	0.405	48
1200	.366	.797	4.64	0.410	93

2.3 PLANNED POWER PLANTS

The source of the information on power plant size on Figure 1 was an article in POWER magazine⁶. The caption in Figure 2 of the article states that "more than 75% of the units listed are rated from 400 to 699 MW." Since this does not include another 13% in the 200 to 400 MW range the statement on the curve should have been "almost 90% of planned power plants." Telephone conversation with the author of the article indicates that the figures are based on a representative sampling of roughly 1/3 of power plants committed for construction in the 1980 - 1984 time period. POWER conducts such a survey annually to provide up-to-date data on utility preferences.

3.0 RECENT RESULTS AND ADDITIONAL DATA

Figure 4 shows a revised cycle configuration, which we call the feedwater cooled CCMHD/Brayton Cycle, now under investigation as part of our ongoing DOE systems work in CCMHD. The MHD loop operates on a Brayton Cycle with relatively modest enthalpy extraction and pressure ratio, approximately 25% and 3.0, respectively. The Brayton Cycle heat exchanger (ARHX) is a shell and tube type unit made of silicon carbide.* It delivers argon to the high temperature argon heater (HTRH) at 1800 F. All heat rejection is to steam plant feedwater (ARFW). The compressor inlet temperature is relatively high (~ 130 F) but, because of the low pressure ratio, the exit temperature is only a little over 500 F.

The steam boiler operates on the HTRH exhaust plus some bypass flow directly from the combustor. Steam turbines drive the main argon compressor

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*The Brayton Cycle configurations studied in ECAS¹³ utilized regenerative heat exchangers and were prohibitively expensive. The desirability of a ceramic shell and tube unit has long been recognized. The silicon carbide design utilized here was brought to our attention by M. Wehrey of Southern California Edison Co. in conjunction with a CCMHD retrofit study now in progress.

compressor and, in addition, drive generators which produce about 1/4 of the net plant output.

This cycle has been set up on our detailed systems code and examined at several sizes from 200 to 1000 MWe. Results are plotted in Figure 5 along with the curves which were shown in Figure 1. Also shown on Figure 5 are data points from recent detailed OCMHD studies by GE^{7,8}, AVCO⁹ and Gilbert Associates¹⁰. There is some scatter, which is not surprising since there are differences in degree of O₂ enrichment, calculated component performance and steam plant configuration.

Finally, comparison with conventional steam plants is also of interest. Figure 6 shows the performance of steam plants¹¹ designed for 3 different turbine inlet pressures. The data are plotted for a constant boiler efficiency of 88% and, do not include any penalty for flue gas scrubbers. The 33.8% efficiency point shown from ECAS¹² includes the penalty for wet scrubbers with a stack temperature of 175 F. At 250 F stack temperature, plant efficiency drops to 31.8%.

4.0 SUMMARY

Relatively simple energy flow models are useful for predicting size variations on MHD plant performance when data is available from more detailed studies for evaluation of model parameters. CCMHD plant efficiency is less sensitive to size variation than OCMHD and a revised CCMHD cycle configuration shows significant gain over previous considered arrangements.

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Figure 1. Efficiency versus Size for Open and Closed Cycle MHD/Steam Power Plants. Steam Bottoming Plants with Reheat except as noted.

Figure 2. Schematic Model for Open Cycle MHD Power Plant

Figure 3. Schematic Model for Closed Cycle MHD/Steam Power Plant

Figure 4. Feedwater Cooled Closed Cycle MHD/Brayton Cycle

Figure 5. Feedwater Cooled CCMHD/Brayton Cycle plus results of recent OCMHD Plant Studies.

Figure 6. Comparison of CCMHD Efficiencies with Conventional Steam Plants