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FINAL REPORT

ATMOSPHERIC CLOUD PHYSICS LABORATORY
PROJECT STUDY

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National Aeronautics and Space Administration
George C. Marshall Space Flight Center
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Under Contract NAS8-32016

By

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TABLE OF CONTENTS

1.0 SUMMARY

The purpose of this report is to document engineering studies performed under contract NAS8-32016. The scope of work provided in the contract included the following tasks:

1. Obtain and review background information and requirements of the zero-g cloud physics experiment.

2. Perform the following tasks in support of the liquid cooling system:

   a. Perform engineering analyses to determine feasible concepts for a liquid cooling loop system for the expansion chamber.

   b. Develop an analytical model of the system or systems determined to be most advantageous.

   c. Evaluate parametrically the system performance characteristics for operational conditions. The study shall include the following:

      - Piping, valves, pumps, and reservoirs
      - Heat exchangers, heaters, and cooling devices

        Control temperature sensor locations and control functions required to achieve design values.
      - Thermal insulation requirements.

3. Provide the following in support of the air pressure control system:

   a. Produce engineering analyses to determine feasible concepts for a system to control air pressure during expansion chamber evacuation.

4. Deliver the following in sufficient detail for MSFC to perform the detail engineering design:

   a. Layout sketches of the systems

   b. Performance characteristic specifications resulting from parametric evaluations.

5. Provide weekly status reports during the course of the conceptual studies.

6. Prepare and deliver a final report documenting analytical models, techniques, and results. Include computer program listings for models and analysis programs generated.

The above scope of work has been completed, with receipt of this documentation, and the contract was fully complied with.

Engineering studies were required of the Zero-G Cloud Physics Experiment liquid cooling system and the air pressure control system. Detailed results of these two studies are reported in separate sections of this document. A brief summary of results for the liquid cooling system and the air pressure control system are provided in the following paragraphs.

1.1 Liquid Cooling System - A total of four concepts were parametrically evaluated during the contract period. Two of these concepts were found to closely approach the systems requirements. Thermal insulation requirements, system hardware, and control sensor locations have been established from this study. The reservoir sizes and initial temperatures have been defined as well as system power requirements. No thermal-electric cooling device information was supplied so no specific analysis was performed on heat exchangers, heaters, or cooling devices.

Although the most feasible concepts have been identified, these concepts have not been optimized. Optimization studies are still recommended as an extension of this contract.

1.2 Pressure Control System - Fluid analyses of the ACPL (Atmospheric Cloud Physics Laboratory) have been performed to determine flow characteristics of various orifice sizes, vacuum pump adequacy, and control systems performance.

System parameters which were predicted in these analyses as a function of time include the following for various orifice sizes:

- Chamber and vacuum pump mass flowrates
- Number of valve openings or closures
- Maximum cloud chamber pressure deviation from the allowable
- Cloud chamber and accumulator pressure.

2.0 INTRODUCTION

Immediately after contract award, background information and system requirements of the zero-g cloud physics experiment were obtained from cognizant MSFC personnel. Available drawings and hardware specifications were also obtained.

Presented in Tables I and II are the thermal and pressure specifications which were applicable for the studies performed herein. Shown in Figures 1 and 2 respectively, are the liquid cooling system concept and expansion chamber temperature transient requirements. The working fluid specified for the liquid cooling system was Coolanol 20, with thermal physical properties as shown in Figures 3 through 7.

2.1 Analytical Techniques for Liquid Cooling System - The Systems Improved Numerical Differencing Analyzer (SINDA) computer program was used to perform parametric studies of the transient behavior of four liquid cooling systems concepts. A network lumped parameter representation of the physical system is required as input to the computer program. This representation for the thermal/pressure networks, subject to the applied boundary conditions, was solved by SINDA program execution subroutines which use finite differencing techniques.

Conceptual systems performance characteristics were evaluated for each parametric condition investigated using computer plots of calculated transient temperature. Typically, computer plots were made corresponding to each nodal representation in the analytical model. Calculations were performed and computer plots were prepared to show a comparison of the average transient expansion chamber temperature and the required value. The deviation between the calculated expansion chamber temperature and the requirement was also plotted. These computer plots were used to aid the analyst in determination of principal parameters in the system and selection of parameters to improve the systems performance. Calculated pressure differentials and corresponding required pump power were found to be acceptably low in early computations. Therefore, emphasis was placed upon thermal performance systems evaluations.

Principal components and effects of the physical model were represented in the analytical model. These components and effects include the following:

- The moving liquid through the system was modelled.
- Thermal capacitance of the piping system, expansion chamber and storage reservoirs were represented.
- Heat input due to pump operation was included.
- Convective heat transfer between the fluid and solid systems was included in the model.
- Thermal characteristics of pipe insulation.
- Valving action was programmed to simulate the control system logic.

2.2 Analytical Techniques for Air Pressure Control System - Air pressure control systems performance predictions were made using a computer program which was specifically developed to calculate systems pressures, masses and mass flow rates. A listing of this program is included in Appendix B. This computer program was coded using the general features available within SINDA computer program. A schematic of the pressure control system which was evaluated is shown in Figure 8. The system components represented in the model include the cloud chamber, valve and related controller, an accumulator, and the pump.

Mass flowrate from the chamber is calculated through the use of Fliegner's Formula (Reference 1):

$$w = \frac{.532\ A\ P_o\ C_D}{\sqrt{T_o}}$$

where, A is flow area

P_o is the upstream pressure

C_D is the orifice discharge coefficient

T_o is the upstream total absolute temperature.

A correction factor for the discharge coefficient of a sharp-edged orifice with zero velocity of approach was used to adjust the flow as shown in Figure 9, (Reference 1). The deviation from unity of the discharge coefficient for a sharp-edged orifice meter is due primarily to the contraction in the stream following the orifice.

Mass flowrate out of the vacuum pump was recalculated each time step by the following process:

- Each time step, the accumulator and vacuum pump pressures are saved.
- Each time step a linear equation is employed to predict pump pressure and accumulator pressure. An average pump pressure is then calculated over the next time interval to estimate average pump mass flowrate. A diagram of this procedure is shown below:

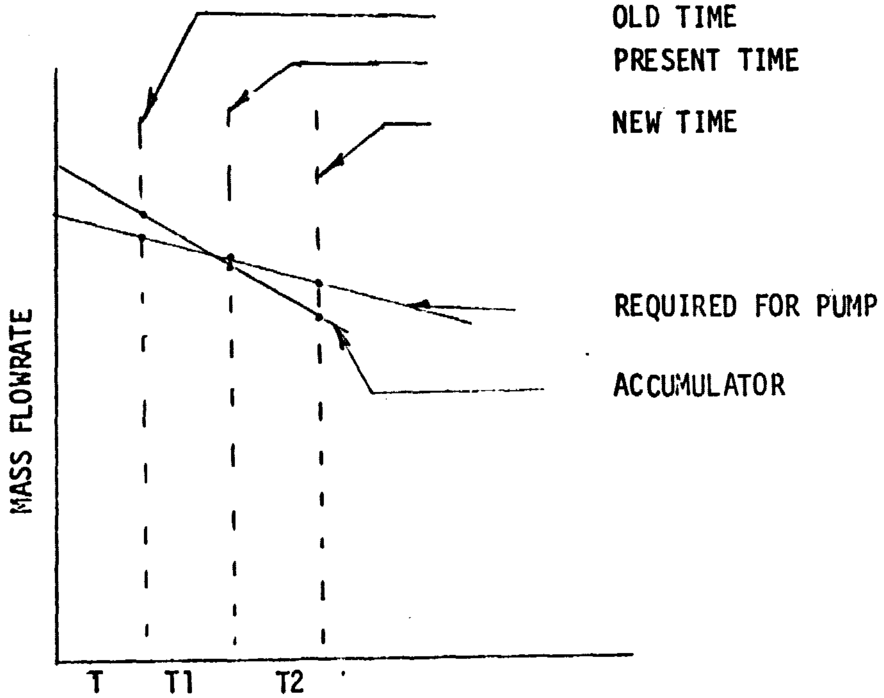

Diagram of Calculation Procedure

Each time step a new pressure is calculated for the chamber and the accumulator using the equation of state:

$$P = \frac{mRT}{V}$$

where, P is the absolute pressure

R is the gas constant

T is the absolute temperature

V is the volume.

A mass balance is performed upon the individual component volumes each time step as follows:

$$m_c = m_o - w\Delta t$$
$$m_a = m_o + w_c\Delta t - w_{vp}\Delta t$$
$$m_t = m_c + m_a$$

where, m_c is current mass in the chamber
m_a is the current mass in the accumulator
m_t is the total current mass
m_o is the original mass
w_{vp} is the current vacuum pump mass flowrate
Δt is the time step interval.

A cross check on the total mass in the system was performed as follows:

$$m_{to} = m_{co} + m_{ao}$$

where, m_{to} is the total original mass
m_{co} is the total original mass in the chamber
m_{ao} is the total original mass in the accumulator.

Then for the first time step:

$$m_{tp} = m_{to} - w_{vp}\Delta t$$

and for succeeding time steps,

$$m_{tp}' = m_{tp} - w_{vp}\Delta t$$

where, m_{to} is the original total mass
m_{tp}' is the new current total mass
m_{tp} is the old current total mass
w_{vp} is the current vacuum pump mass flowrate
Δt is the time step interval.

The valve control system dictates the time that the valve shall be either closed or open at any time during operation. The closed time is a function of the sampling rate and for this analysis a sampling rate of ten times per second was used. Therefore the minimum closed time interval for the valve equals 0.1 seconds and the maximum closed time interval is determined by the system pressure requirements. The open time is a function of the pressure differential between commanded chamber pressure and the actual chamber pressure and the gain of the control system:

$$\text{Open time} = (P_c - P_a)\ .33\alpha$$

where, α = gain

P_c is the commanded pressure

P_a is the actual pressure.

3.0 RESULTS OF ANALYSES

3.1 Liquid Cooling System - Computed performance characteristics of four (4) liquid cooling systems configurations are reported. A brief description of each system concept is provided as follows. The initial analytical model, Configuration 1, is shown schematically in Figure 10. The conceptual configuration consisted of the cloud chamber, two liquid reservoirs, three valves, a pump and the necessary connecting pipes. The dimensions of the piping system were developed to fit within space envelope of two 19 x 24 inch racks, approximately four feet long.

A discrete control system was modeled based upon available hot/cold reservoir supplies with liquid by-pass usage when the chamber was within ±.0055°C of the required temperature. Based upon the results of analyses for this initial investigation, Configuration 2 was developed. Configuration 2 is depicted schematically in Figure 11. The cold reservoir volume was doubled to 0.106 M^3 and the initial temperature was reduced from -30°C to -45.5°C. Configuration 3, was a variation of Configuration 2, with three additional control valves to selectively return liquid to hot, cold or by-pass reservoirs.

The purpose of these valves was to conserve heating/cooling capacity. Configuration 3 is illustrated schematically in Figure 12. The control logic to selectively return the fluid was dependent upon the liquid temperature at the expansion chamber outlet. A third small reservoir was also added to accumulate return fluid for the by-pass as indicated. Configuration 4 consisted of a variation of Configuration 3 with the implementation of additional control logic. The control logic utilizes an integration of the total temperature deviations as a function of time in conjunction with the current temperature deviation.

Parametric studies were conducted on each of the configurations. Variables which were selected for investigation changed during the course of the study and depended upon the results of previous predictions. Results of parametric studies using Configuration 1, revealed that the initial reservoir sizes and temperatures were not suitable to meet the required chamber temperature transients.

The analytical model was revised to represent the Configuration 2 concept and additional parameter studies were performed. Results of computations indicated that this configuration would have the necessary cooling capacity. However, the heating capacity available during the heating requirement interval was not sufficient to return the chamber to required temperatures. These results are shown graphically in Figure 13, where the predicted chamber temperature is superimposed on the chamber temperature requirement.

In parametric studies of Configuration 3, the general systems requirements such as liquid flow rates, hot/cold reservoir sizes, and flow control requirements had been previously determined. To quickly observe and interpret results of parametric computer cases, machine plot capability was added to the analytical model. With the addition of the plotting capability, the parametric computations were numbered sequentially starting with Case 1. A complete listing of computer cases for Configurations 3 and 4 is shown in Table III. The SC4020 machine plots for case 1 and subsequent cases are included in Appendix A.

The results for Configuration "3", Computer Case 7 are shown in Figure 14. The results are for 32 gpm flow rate and indicate adequate heating capacity. Results generally tend to show that flow rates greater than approximately 25 gpm will require additional cold reservoir volume.

The discrete control system utilized in Configurations "1" through "3" produced temperature deviations as low as approximately 0.4°C. Variation of systems parameters such as range of temperatures for by-pass fluid and pump flow rate failed to adequately improve the deviations. Results indicated a need for an improved control scheme. Cognizant MSFC personnel supplied a control function which utilizes an integration of the total deviations as a function of time in conjunction with the current deviation. This scheme was employed in Configuration "4". Some of the parameters utilized in Configuration "4" were pump flow rate, system sample frequency, and gain. Gain is a factor used with the input control signal. Results from three computer cases utilizing Configuration "4" are presented in Figure 15. The computer simulation time was only thirty seconds and actual maximum deviations could occur later in the simulation; however, the results are indicative of the effect of pump flow rate on the deviations.

3.2 Air Pressure Control System - Parametric studies were conducted to investigate the effects of orifice size, initial accumulator pressure and controller gain factor. A summary of orifice sizes, initial conditions and gain factors which were investigated are shown in Table IV. Corresponding maximum computed pressure deviations are also shown. These computed pressure deviations are defined as the computed pressure minus the required value. The maximum allowable deviation was specified to be ±0.1 mb. Results of individual computer cases are shown in Figures 1a through 44f of Appendix B.

It was found that small orifice sizes (less than 0.079 cm diameter) were unable to meet the pressure deviation specification during the steepest portion of the required pressure transient. Predictions for larger orifice sizes indicate inability to meet the allowable pressure deviation since the valve allows flow rates higher than required values.

Shown in Figures 16 and 17 are the maximum predicted pressure deviations versus orifice size for controller gain factors of 50 and 100. The computer print interval which was selected initially for cases 1 through 32 indicated an acceptable pressure deviation for Computer Case 19. The print interval was selected to output a calculated value after 1000 calculations (10 seconds problem time) had occurred in the machine for cases 1 through 32 and after 500 calculations (5 seconds problem time) for cases 33 through 44. Therefore, the maximum deviation value was not observed in the plotted data for Computer Case 19.

4.0 CONCLUSIONS AND RECOMMENDATIONS

4.1 Liquid Cooling System - Two of the configurations evaluated were determined to be more desirable. Configuration "2", at 8 gpm (see Figure 18) utilizing a $\pm 0.055°C$ by-pass usage had a lower maximum deviation than Configuration "4" at 10 gpm (see Figure 19) with a gain of 10. Trends indicate that the scheme of using a by-pass reservoir as was used on Configuration 3 could reduce the required volumes of the hot/cold reservoirs.

The following list of hardware with their respective weights were part of the analytical models for Configuration 4:

• Cloud physics chamber	12.25 Kg
• Coolanol-20	146.25 Kg
• Five feet thin walled pipe	1.27 Kg
• Six valves	8.16 Kg
• Three reservoirs	31.3 Kg

A pump with an operational range of one to twenty gallons per minute at 20 psi (137895 $\frac{\text{Newtons}}{\text{Meter}^2}$) is recommended for the test facility. Predicted system pressure drops and pump power requirements are shown in Figure 20. Depending on the rate of heat transfer within the cloud chamber coolant passage, the necessary flow rates and the initial temperatures of the liquids can vary from those predicted herein. It is also recommended that effective lengths of manifold pipes into the cloud chamber be identical if possible. If this is not the case, it would tend to be possible to introduce different temperature liquids at various positions on the chamber.

The control sensors should be located near the center of the end plates and near the center of the cylinder. The control function, when optimized, should be of a differential type, such as that used in Configuration "4".

It is recommended that an optimum configuration be determined based on the previous analyses. It is further recommended that additional fluids be investigated, since Coolanol-20 is not approved for Space Shuttle use.

Analyses reported herein should be verified by test and necessary changes, if any, incorporated into the analytical model. The detailed model of the cloud physics chamber, developed by MSFC with experimentally determined convection heat transfer coefficients should be incorporated with the Sperry model, to better identify necessary control schemes.

4.2 Pressure Control System - The following trends and conclusions are evident from review of the calculated data.

- The lowest pressure deviations were achieved for 0.158 cm diameter orifices.
- Control system gain factor of approximately 100 tended to produce the best control.
- The initial accumulator pressure was not an important factor in the control system performance.
- Results indicate that variable orifice size control would lead to effective pressure control.

REFERENCES

1. Shapiro, Ascher H., "The Dynamics and Thermodynamics of Compressible Fluid Flow," New York: The Ronald Press Company, 1953.

CASE	(1) SLOPE (°C/MIN)	(2) TEMP. RANGE (°C)	(3) RIPPLE (°C) (1 sec. or 3 cm peak to peak)	(4) DELTA ACROSS CHAMBER	(5) WINDOWS	(6) WALL TRACK INPUT CURVE	(7) SPECIFY INPUT T(t) CURVE (that comes from computer readout)	(8) DURATION	(9) INSTANTANEOUS WALL SURFACE TEMP. MEASUREMENT (mean aver. over whole surface) *
A	0	+25 0	±.02	±.01	±.02	See Column 3	±.005	90 min.	±.005
B	0	0 -25	±.02	±.05	±.1	See Column 3	±.005	90 min.	±.01
C	0.5	+25 -25	±.1	±.1	±.1	±.1	±.01	90 min.	±.01
D	1.2	+25 -20	±.15	±.1	±.1	±.1	±.01	30 min.	±.01
E	6	+25 -15	——	±.15	±.15	±.1	±.05	2.5 min.	±.01

TABLE 1 - THERMAL SPECIFICATIONS FOR COOLING OF THE WALL OF THE ACPL EXPANSION CHAMBER

TABLE II

PRESSURE CONTROL SPECIFICATION

Pressure control and measurement within the ACPL expansion chamber must meet the following specifications:

1. Initial Conditions

   Temperature = Constant

   Pressure Constant 400 mb $\leq P \leq$ 1013 mb

   Absolute Accuracy Measurement $\pm$0.5 mb
   (accuracy relative to NBS calibration)

   Relative instantaneous resolution above noise level
   $\pm$.05 mb or better. (Measurement)

   Control Stability $\pm$.05 mb (this pertains to pressure control system)

   Nominal relative humidity range to exist in the chamber prior to expansion: 75 to 99%.

2. During Expansion

   Span of single expansion 30 to 350 mb

   Track control signal $\pm$0.1 mb
   (for cases A through D $\pm$.2 mb (for Case E)

   Pressure difference resolution $\pm$.1 mb
   (current pressure relative to initial pressure)

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TABLE III. SUMMARY OF COMPUTER CASES FOR CONFIGURATIONS 3 AND 4.

COMPUTER CASE	CONFIGURATION	FLOW GPM	INITIAL HOT°C	INITIAL COLD°C	BY-PASS RANGE	RUN TIME, SECONDS	MAX ΔT-°C	BY-PASS RESERVOIR VOLUME	CONTROLLER SAMPLE TIME INTERVAL, SECONDS	GAIN
1	3	10	+38°C	-45.5°C	+5.6	300	0.5	*	.125	Discrete Control
2	3	10	+38°C	-45.5°C	∓5.6	300	0.4	*	.125	Discrete Control
3	3	10	+54.4°C	-45.5°C	∓5.6	3600	0.8	*	.125	Discrete Control
4	3	10	+54.4°C	-45.5°C	∓0.0	300	1.3	*	.125	Discrete Control
5	3	10	+54.4°C	-45.5°C	∓0.0	300	1.3	7.0	.125	Discrete Control
6	3	20	+54.4°C	-45.5°C	∓1.7	100	0.6	1.0	.125	Discrete Control
7	3	32	+54.4°C	-45.5°C	∓1.0	3600	0.8	1.0	.125	Discrete Control
8	3	10	+54.4°C	-45.5°C	∓3.0	200	0.85	1.0	1.0	Discrete Control
9					Computer Case Not Performed					
10					Computer Case Not Performed					
11	3	20	+54.4°C	-45.5°C	+5.6	100	0.58	1.0	0.125	Discrete Control
12	3	20	+54.4°C	-45.5°C	∓5.6	100	1.05	1.0	1.0	Discrete Control
13	3	20	+54.4°C	-45.5°C	±10.0	100	0.90	1.0	1.0	Discrete Control
14	3	10	+54.4°C	-45.5°C	+1.7	100	0.84	1.0	0.14	Discrete Control
15	3	10	+54.4°C	-45.5°C	∓5.6	100	0.81	1.0	0.125	Discrete Control
16	3	10	+54.4°C	-45.5°C	∓5.6	100	0.81	1.0	1.0	Discrete Control
17	3	10	+54.4°C	-45.5°C	±10.0	100	0.90	1.0	0.5	Discrete Control
18	4	10	+54.4°C	-45.5°C	*	30	0.42**	1.0	1.0	2
19	4	20	+54.4°C	-45.5°C	*	30	0.63**	1.0	1.0	2
20	4	5	+54.4°C	-45.5°C	*	30	0.52**	1.0	1.0	2
21	4	10	+54.4°C	-45.5°C	*	30	0.58**	1.0	0.5	2
22	4	20	+54.4°C	-45.5°C	*	30	0.65**	1.0	0.5	2
23	4	5	+54.4"C	-45.5°C	*	30	0.64**	1.0	0.5	2
24	4	10	+54.4°C	-45.5°C	*	30	0.22**	1.0	0.5	20
25	4	20	+54.4°C	-45.5°C	*	30	0.15**	1.0	0.5	20
26	4	5	+54.4°C	-45.5°C	*	30	0.45**	1.0	0.5	20
27	3	40	+54.4°C	-45.5°C	±5.6	100	0.50	1.0	0.05	Discrete Control
28	4	10	+54.4°C	-45.5°C	*	1800	0.76	1.0	0.5	10

* Indicates not applicable for this case.
** Indicates not applicable for comparison due to short computer run time.

TABLE IV. SUMMARY OF COMPUTER CASES FOR THE PRESSURE ANALYSIS

CASE	VALVE SIZE, CM	INITIAL RESERVOIR PRESSURE, NEWTONS/M^2	MAXIMUM NUMBER OF OPENINGS	MAXIMUM DEVIATIONS, MILLIBARS	RUN TIME, SECONDS	MAXIMUM CRITICAL PRESSURE RATIO	GAIN
1	0.07938	9997.4	3899	18.6	900.	0.243	200.
2	0.07938	9997.4	3901	18.6	900.	0.243	100.
3	0.07938	9997.4	3906	18.8	900.	0.243	50.
4	0.15875	9997.4	3927	0.125	900.	0.276	200.
5	0.15875	9997.4	3914	0.125	900.	0.276	100.
6	0.15875	9997.4	3928	0.186	900.	0.276	50.
7	0.07938	24200.6	3899	18.6	900.	0.243	100.
8	0.07938	24200.6	3905	18.84	900.	0.243	50.
9	0.07938	48332.2	3898	18.61	900.	0.467	100.
10	0.07938	48332.2	3903	18.85	900.	0.467	50.
11	0.15160	9997.4	4100	0.136	900.	0.277	100.
12	0.15160	9997.4	4101	0.197	900.	0.276	50.
13	0.3175	9997.4	2059	0.293	900.	0.276	100.
14	0.3175	9997.4	2059	0.293	900.	0.276	50.
15	0.15160	48332.2	4109	0.136	900.	0.467	100.
16	0.15160	48332.2	4110	0.198	900.	0.467	50.
17	0.3175	48332.2	2061	0.304	900.	0.467	100.
18	0.3175	48332.2	2061	0.304	900	0.467	50.
19	0.15875	48332.2	3935	0.073	900	0.467	100.
20	0.15875	48332.2	3936	0.196	900.	0.467	50.
21	0.11906	9997.4	4451	0.168	900.	0.276	100.
22	0.11906	9997.4	4456	0.360	900.	0.276	50.
23	0.11906	29164.8	4429	0.236	900.	0.282	100.
24	0.11906	29164.8	4454	0.422	900.	0.282	50.
25	0.11906	48332.2	4450	0.179	900.	0.467	100.
26	0.11906	48332.2	4453	0.434	900.	0.467	50.
27	0.11906	55227.0	4450	0.185	900.	0.534	100.
28	0.11906	55227.0	4453	0.332	900.	0.534	50.
29	0.15875	29164.8	3930	0.121	900.	0.282	100.
30	0.15875	29164.8	3931	0.190	900.	0.282	50.
31	0.15875	55227.0	3936	0.123	900.	0.534	100.
32	0.15875	55227.0	3938	0.190	900.	0.534	50.
33	0.200	9997.4	3315	4.301	900.	0.277	1.
34	0.200	9997.4	3267	0.558	900.	0.279	10.
35	0.200	9997.4	3262	0.142	900.	0.281	50.
36	0.200	9997.4	3262	0.142	900.	0.281	100.
37	0.24	9997.4	2937	2.116	900.	0.280	1.
38	0.24	9997.4	2914	0.232	900.	0.281	10.
39	0.24	9997.4	2913	0.189	900.	0.281	50.
40	0.24	9997.4	2913	0.189	900.	0.281	100.
41	0.28	9997.4	2525	2.086	900.	0.279	1.
42	0.28	9997.4	2508	0.259	900.	0.281	10.
43	0.28	9997.4	2508	0.259	900.	0.281	50.
44	0.28	9997.4	2508	0.259	900.	0.281	100.

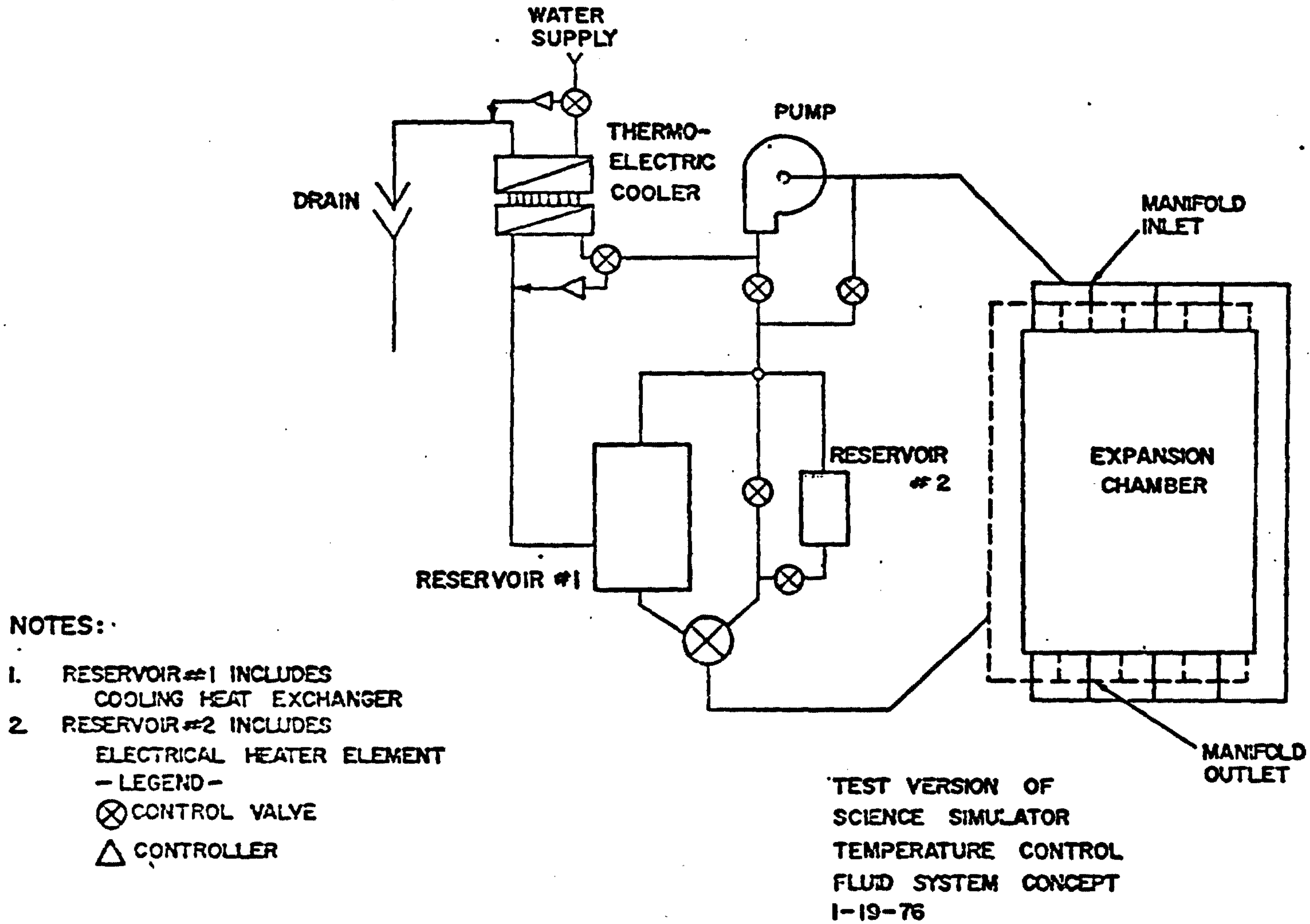


Figure 1. Liquid Cooling System Concept.

PROPOSED EXPANSION CHAMBER THERMAL CYCLES

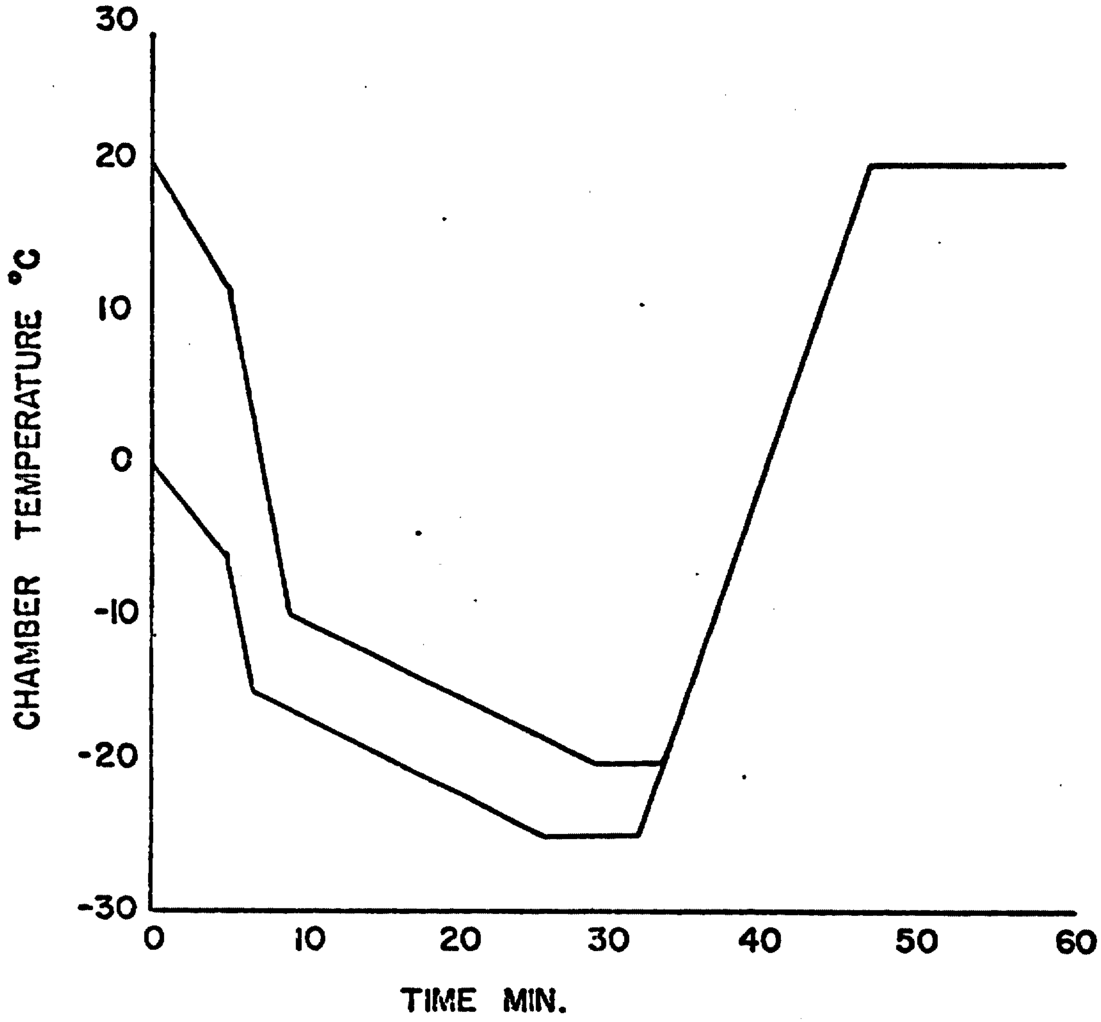


Figure 2. Expansion Chamber Temperature Transient Requirements.

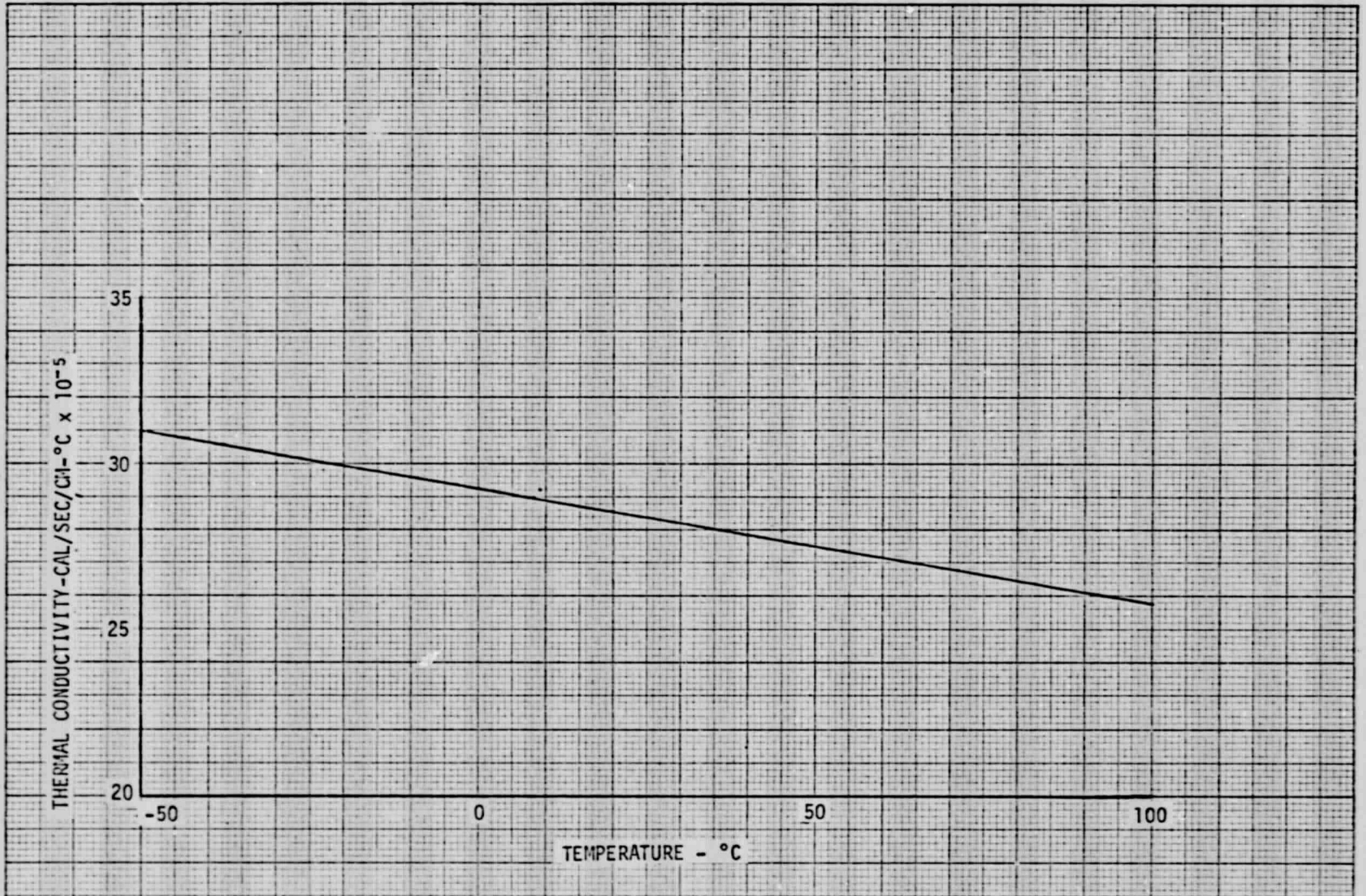


Figure 3. Coolanol-20 Thermal Conductivity as a Function of Temperature.

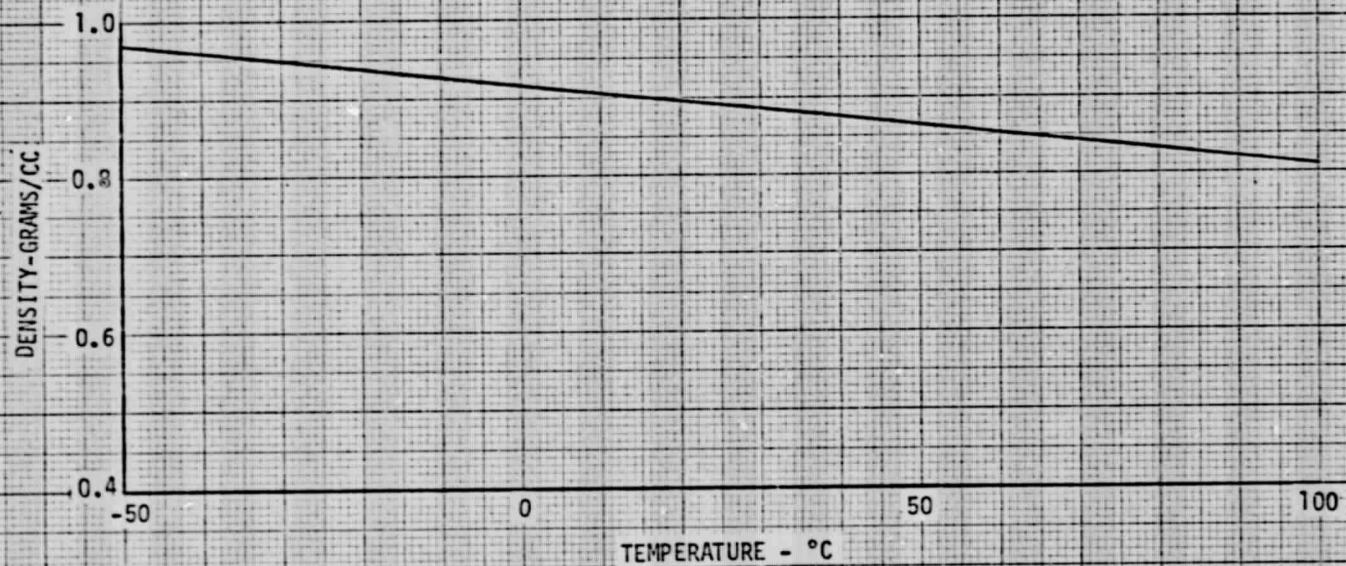


Figure 4. Coolanol-20 Density as a Function of Temperature.

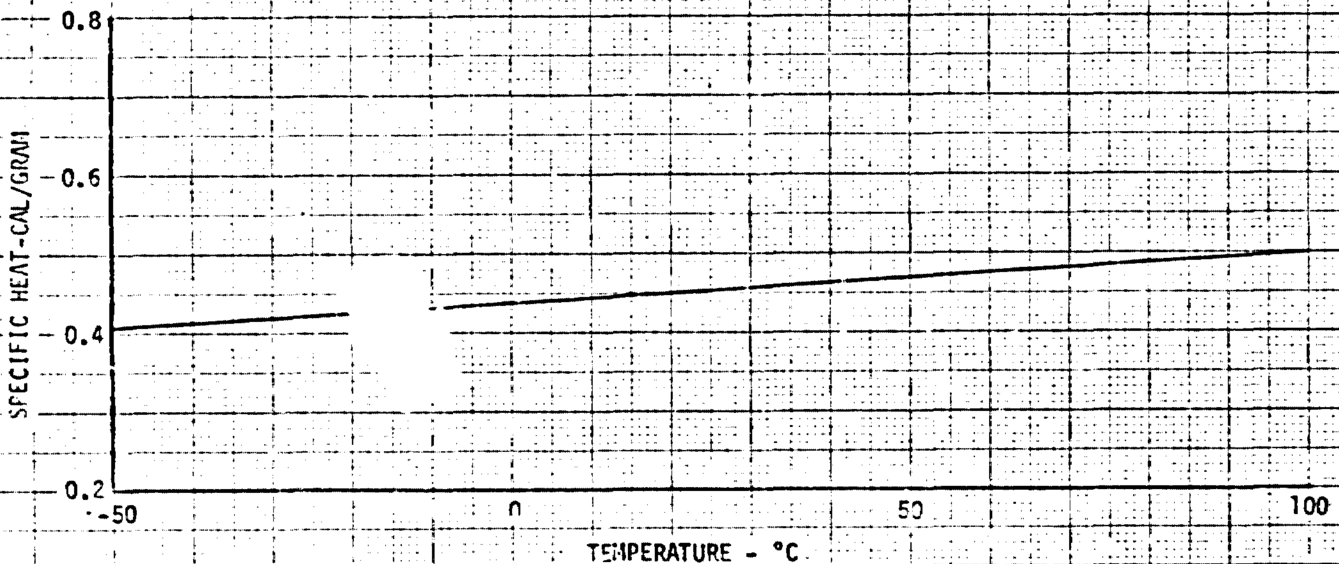


Figure 5. Coc:anol-20 Specific Heat as a Function of Temperature.

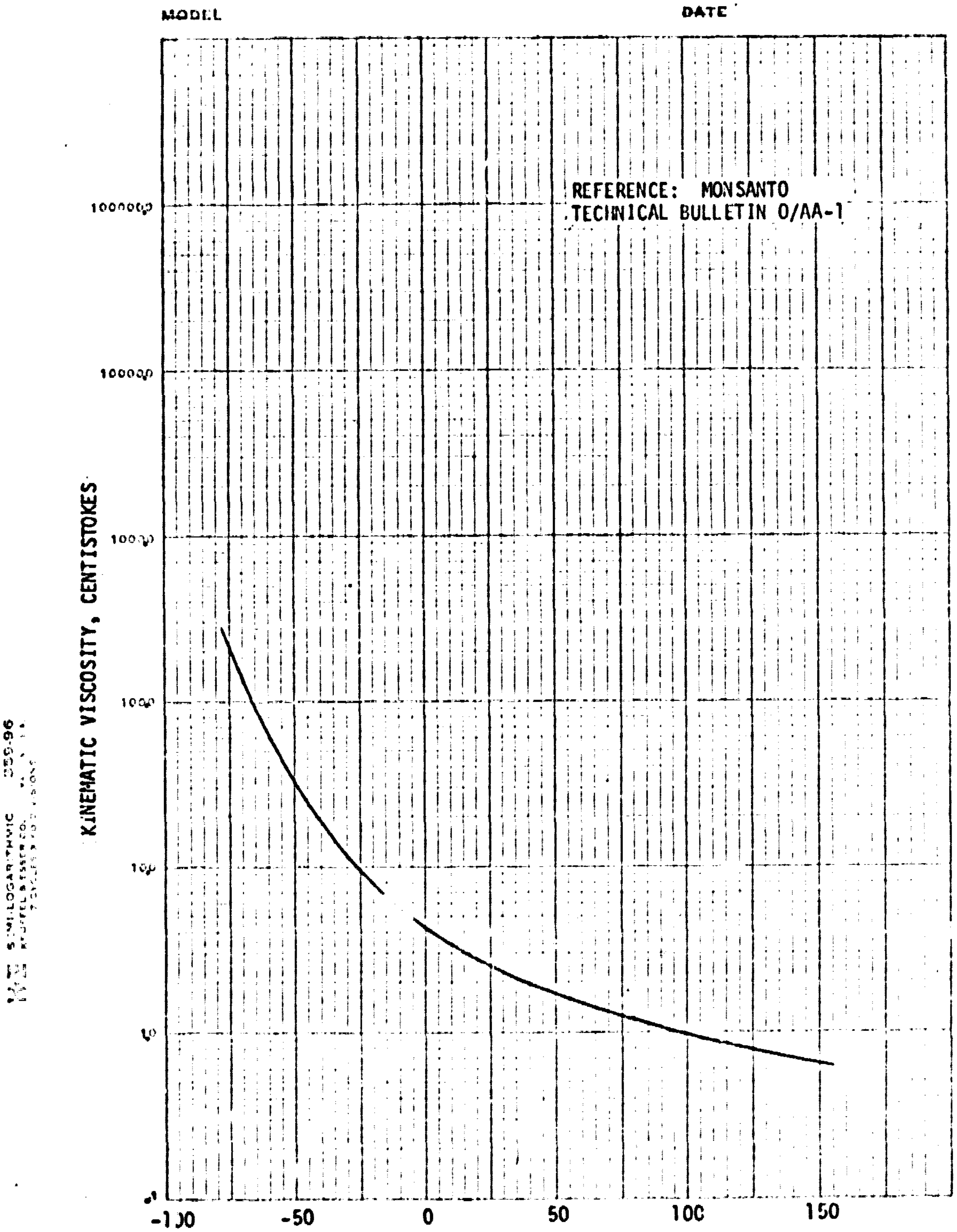


Figure 6. Viscosity Versus Temperature.

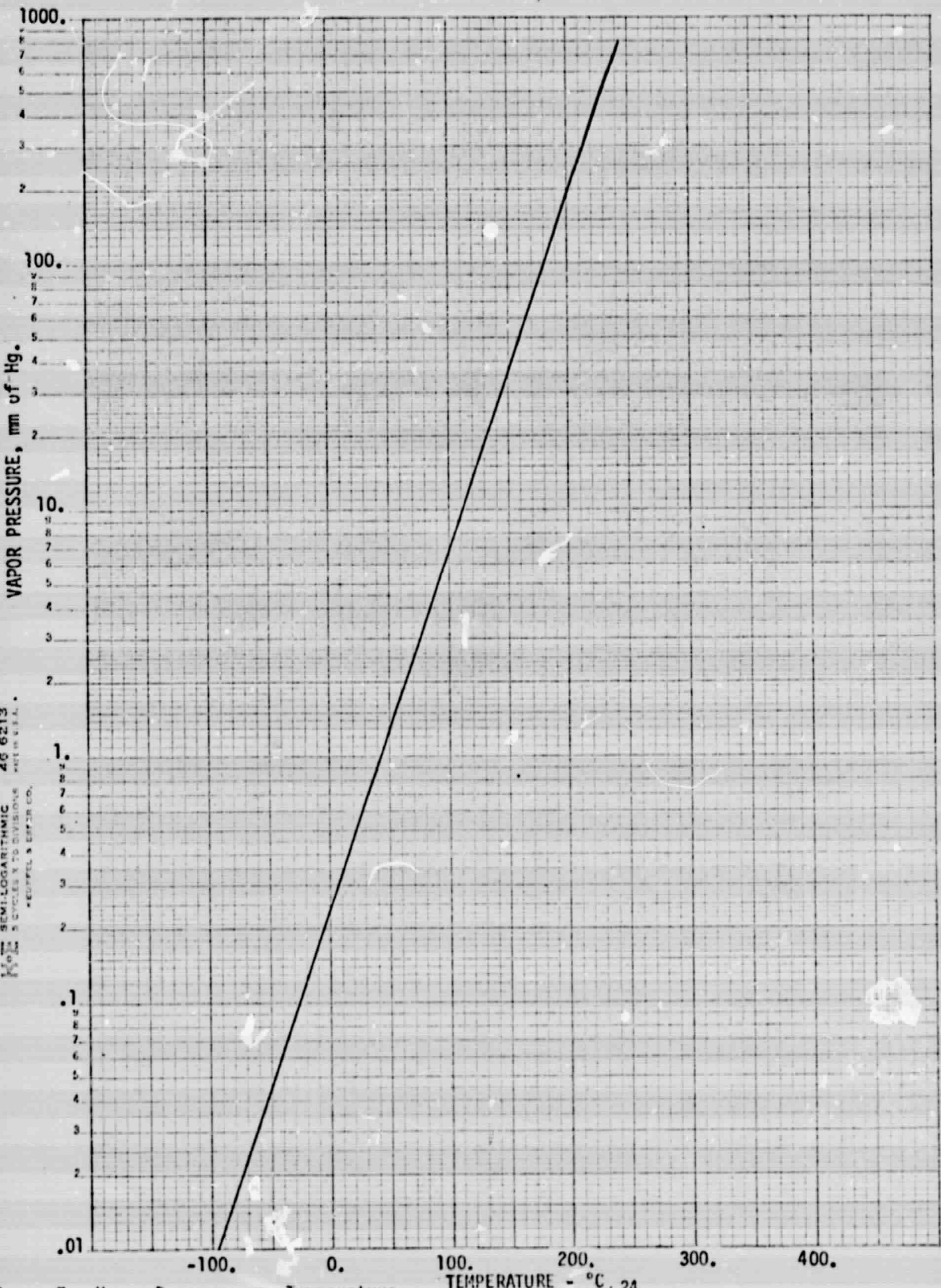
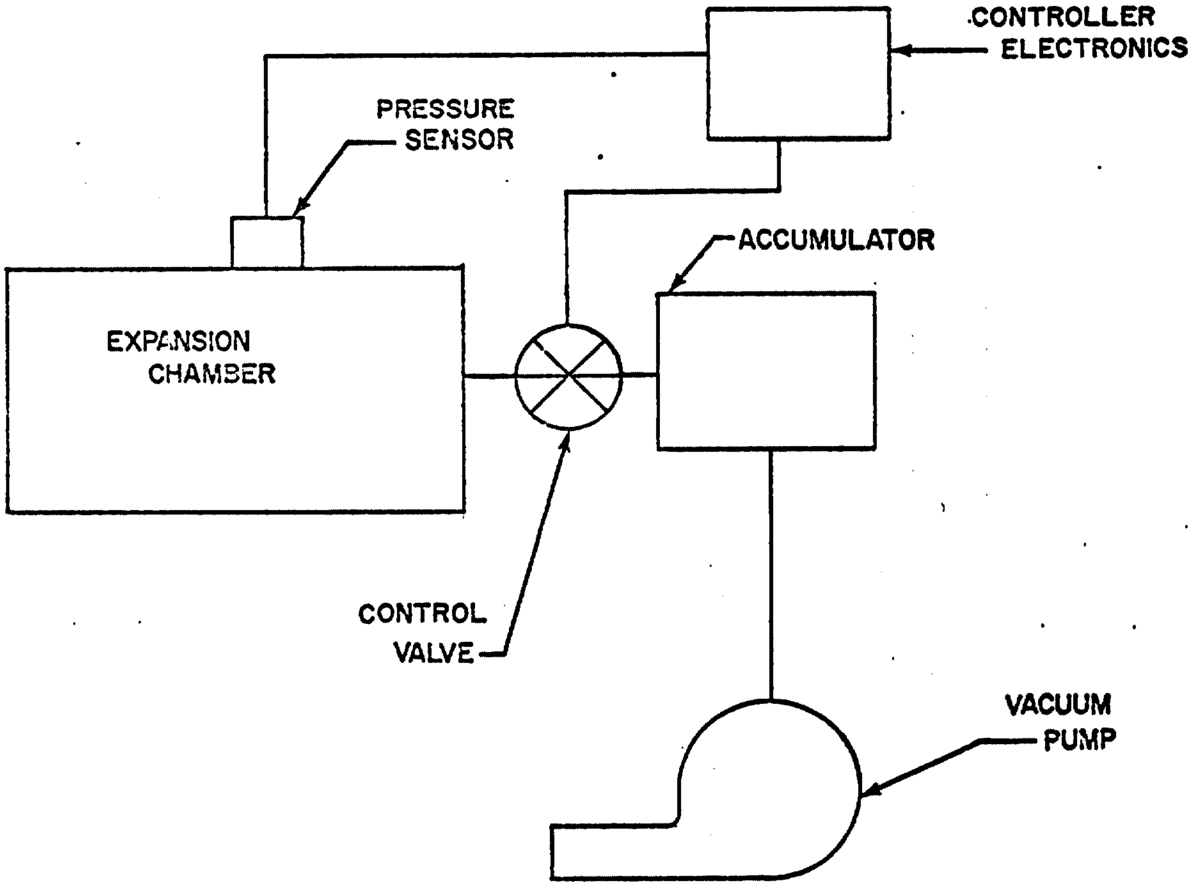


Figure 7. Vapor Pressure vs. Temperature.



SCHEMATIC OF VACUUM PUMP/PRESSURE CONTROL SYSTEM

FIGURE 8.

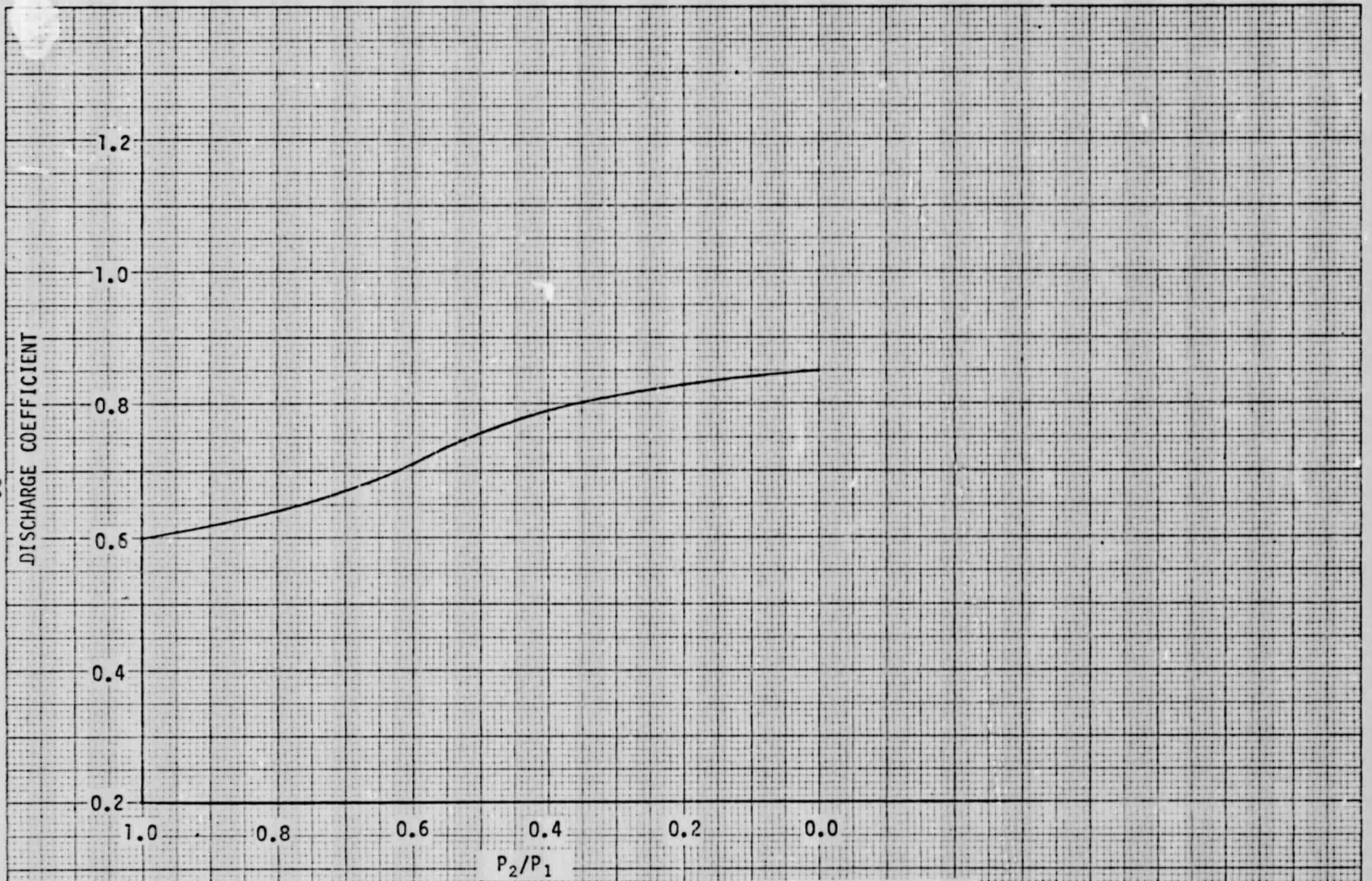


Figure 9. Discharge Coefficient of Sharp-Edged Orifice Meters with Zero Velocity of Approach Discharge Coefficient P_2/P_1.

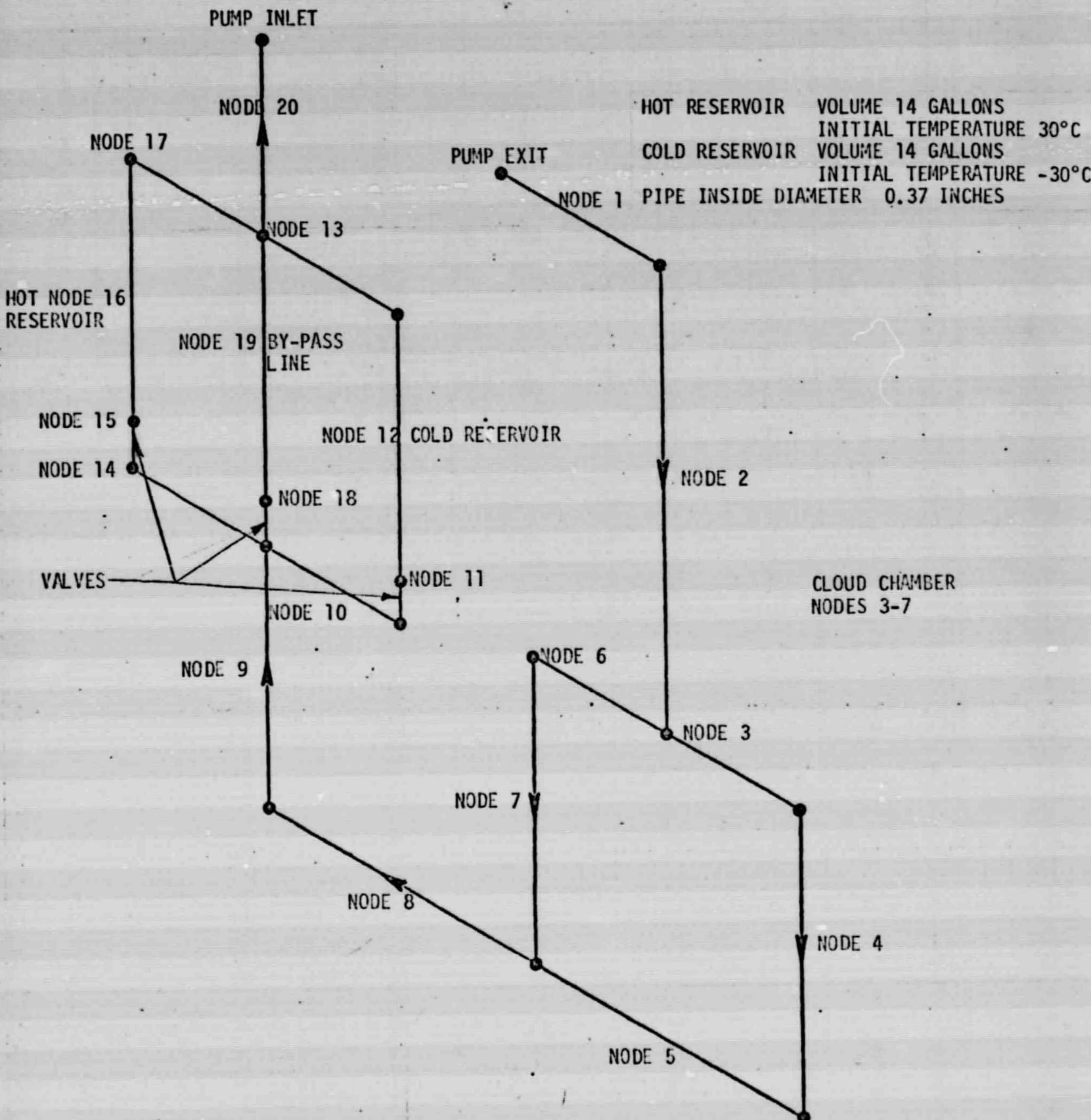


Figure 10. Analytical Model for Configuration "1".

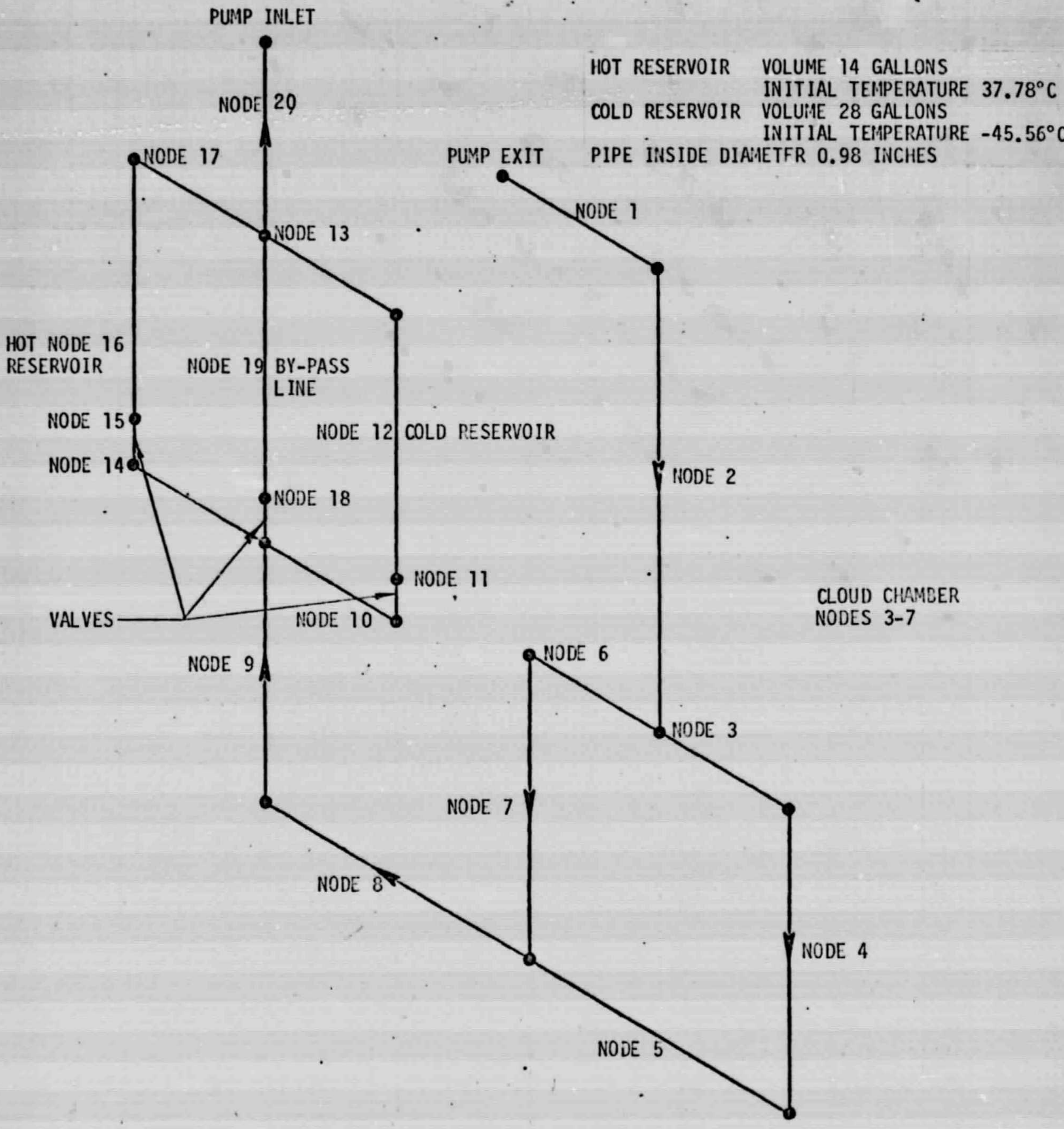


Figure 11. Analytical Model for Configuration "2"..

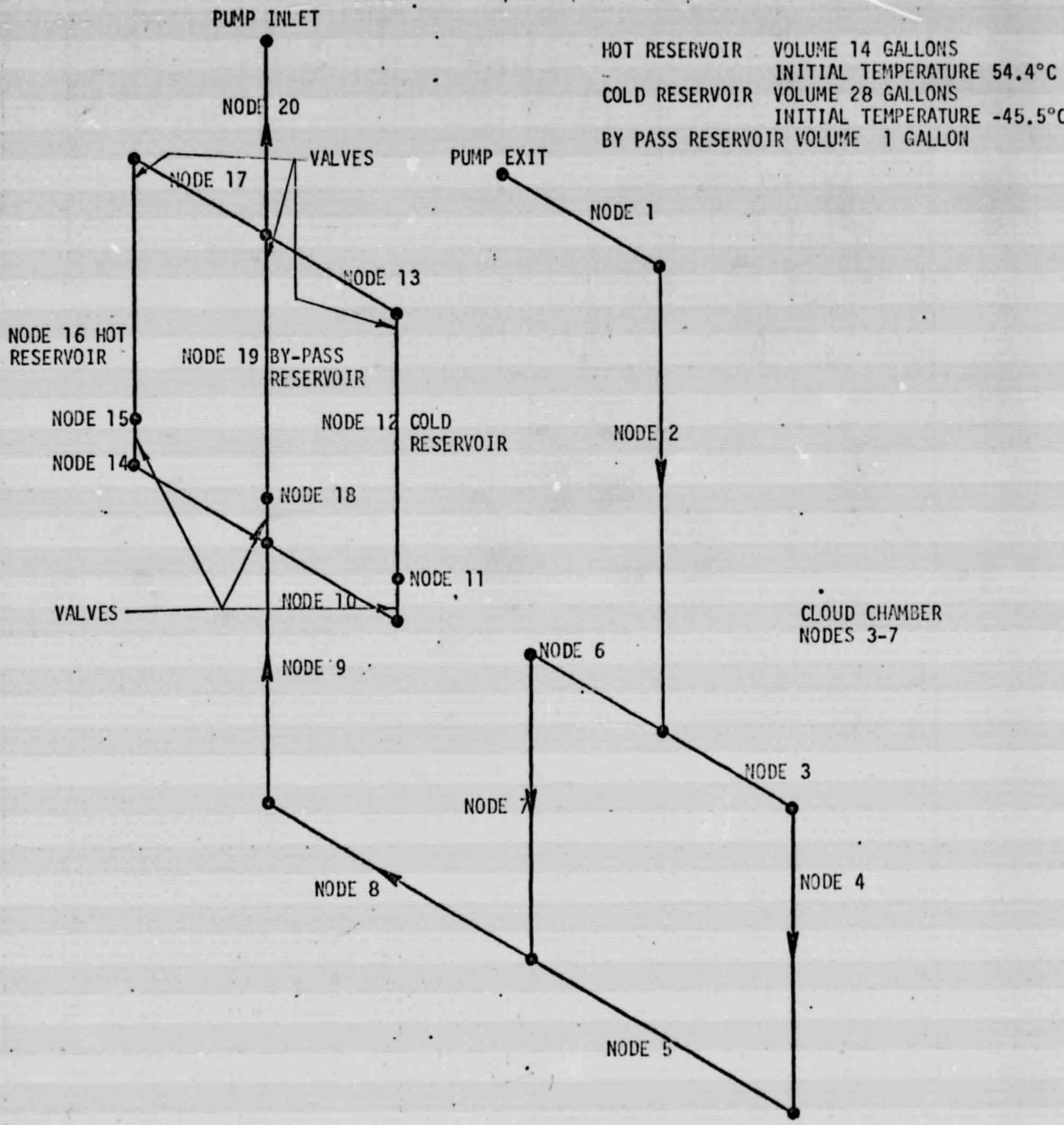


Figure 12. Analytical Model for Configuration "3".

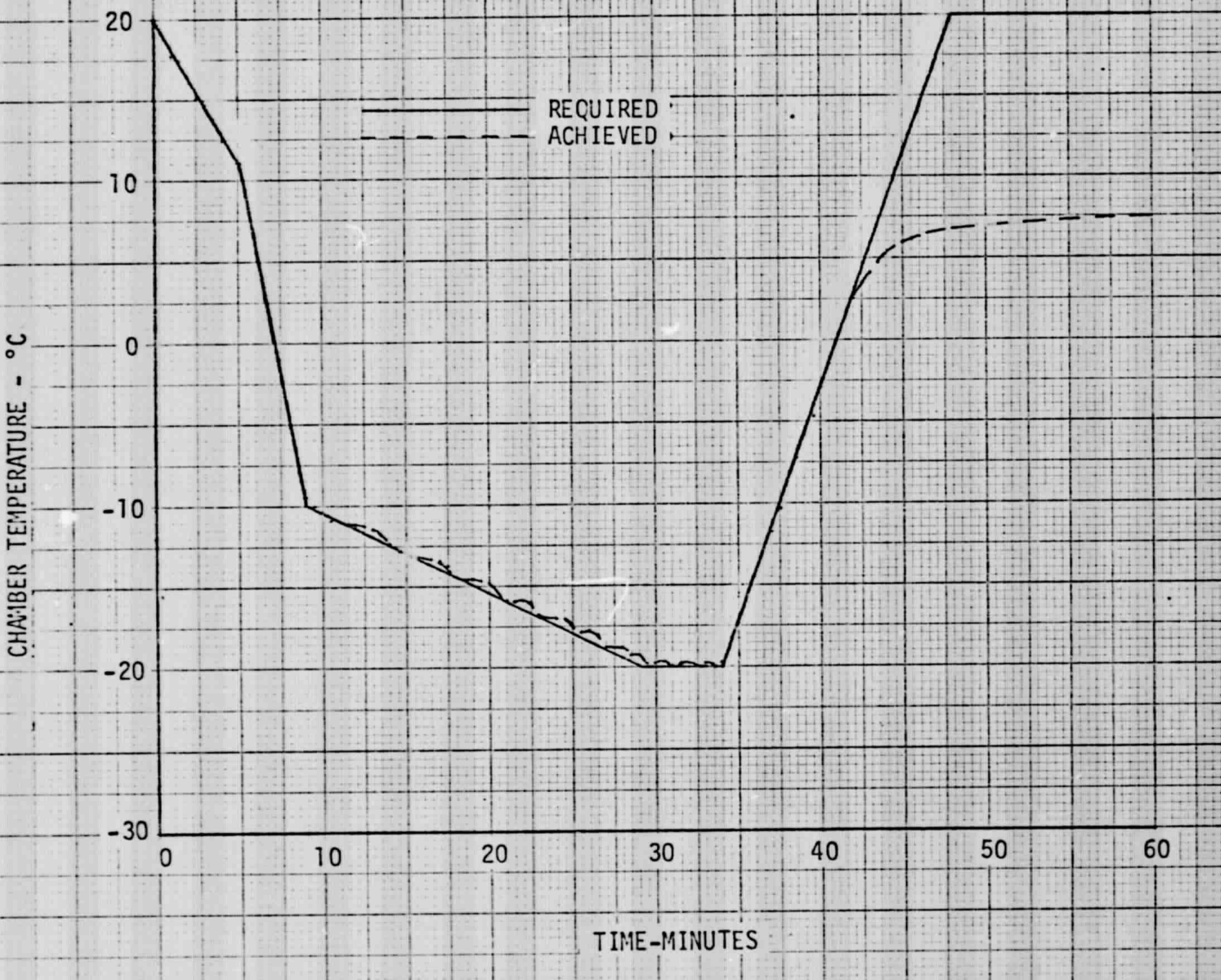


Figure 13. Configuration "2" Predicted Performance.

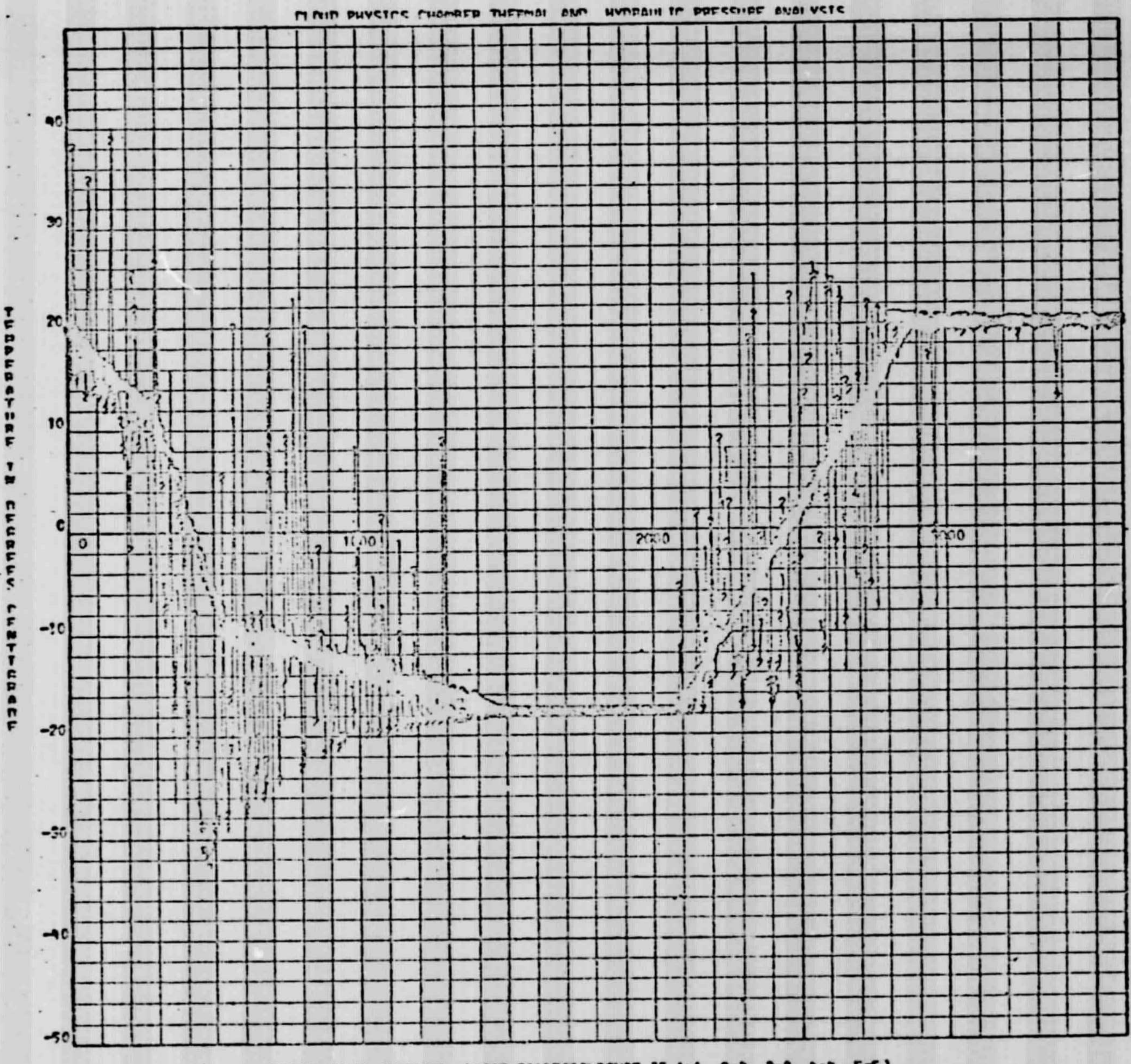


Figure 14. Configuration "3" Case 7 Computer Generated Plot.

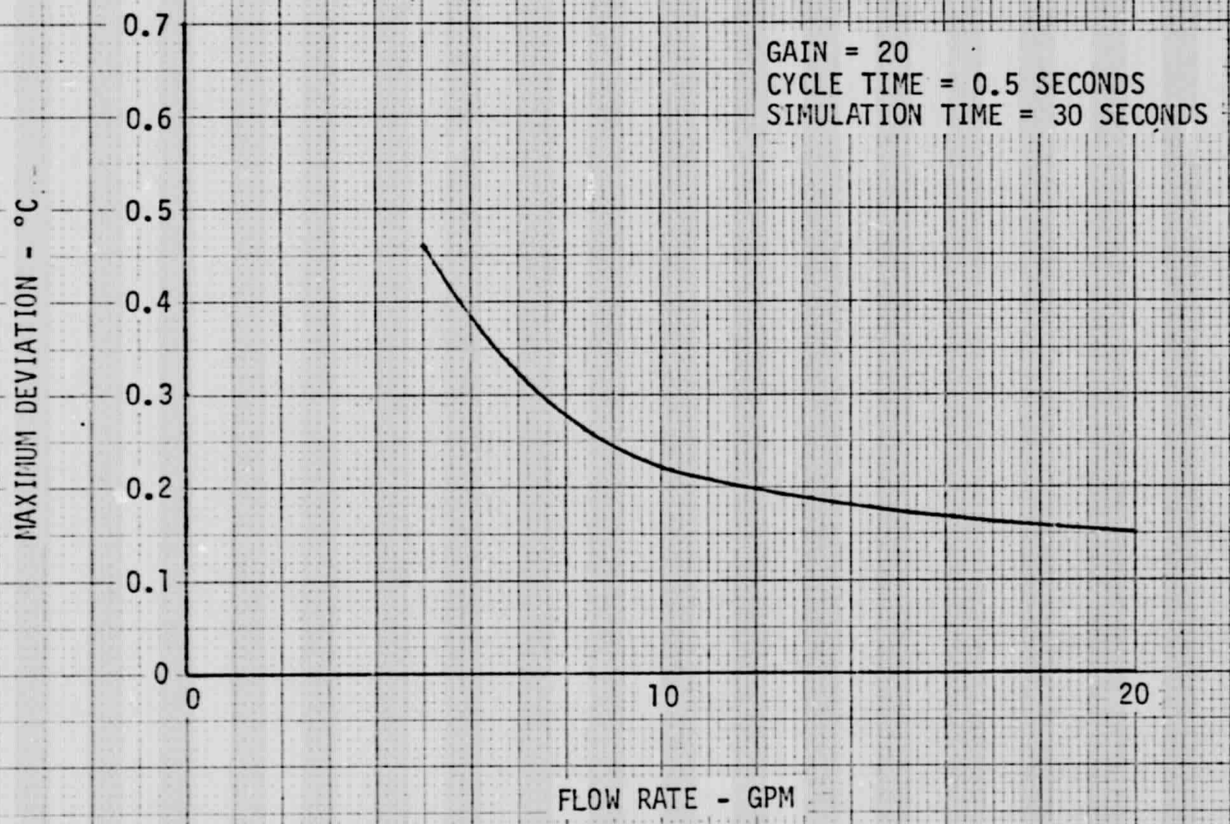


Figure 15. Cloud Chamber Temperature Deviation from Required Versus Liquid Flow Rate for Configuration "4".

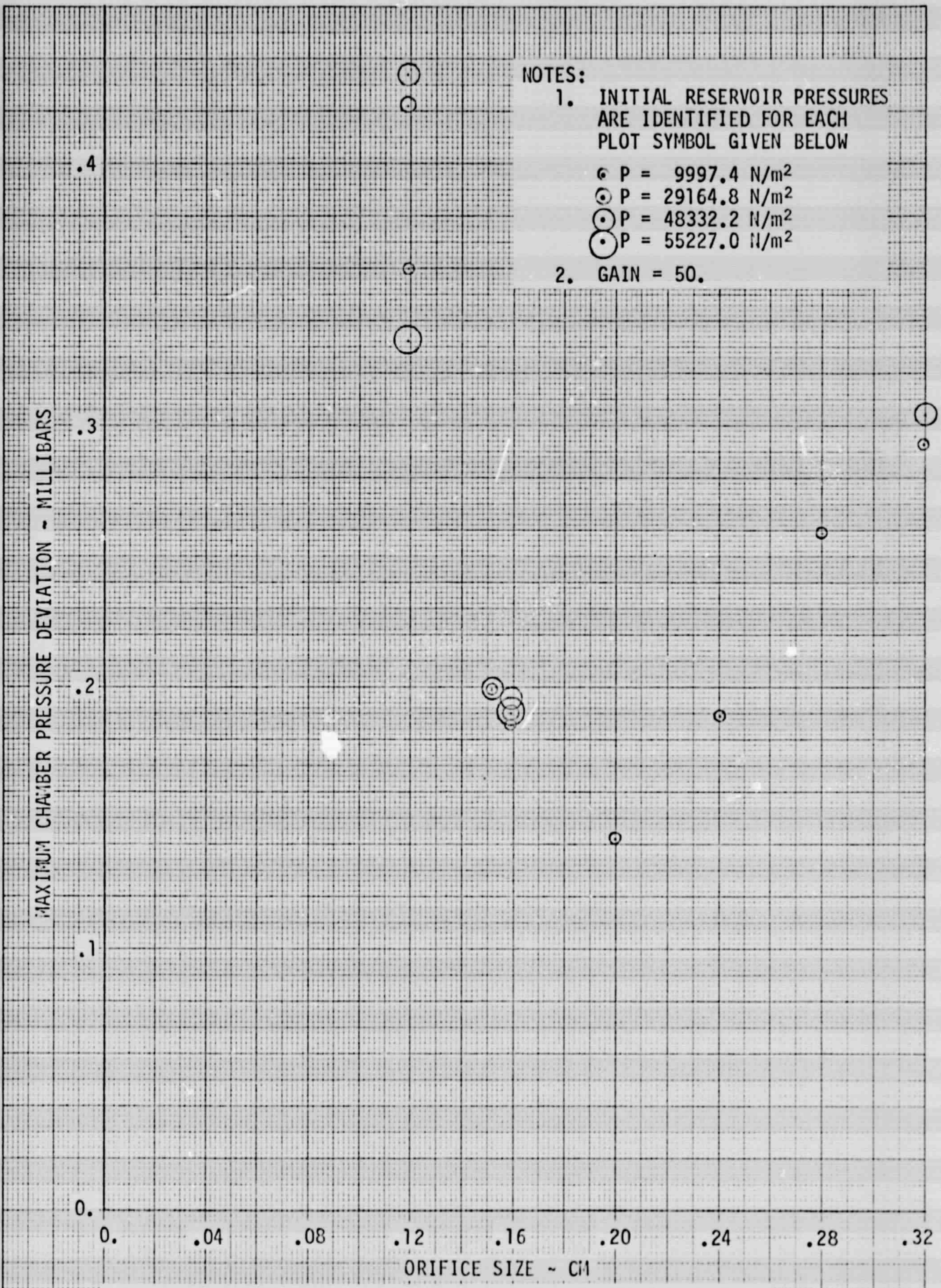


Figure 16. Summary of Maximum Deviation Versus Orifice Size.

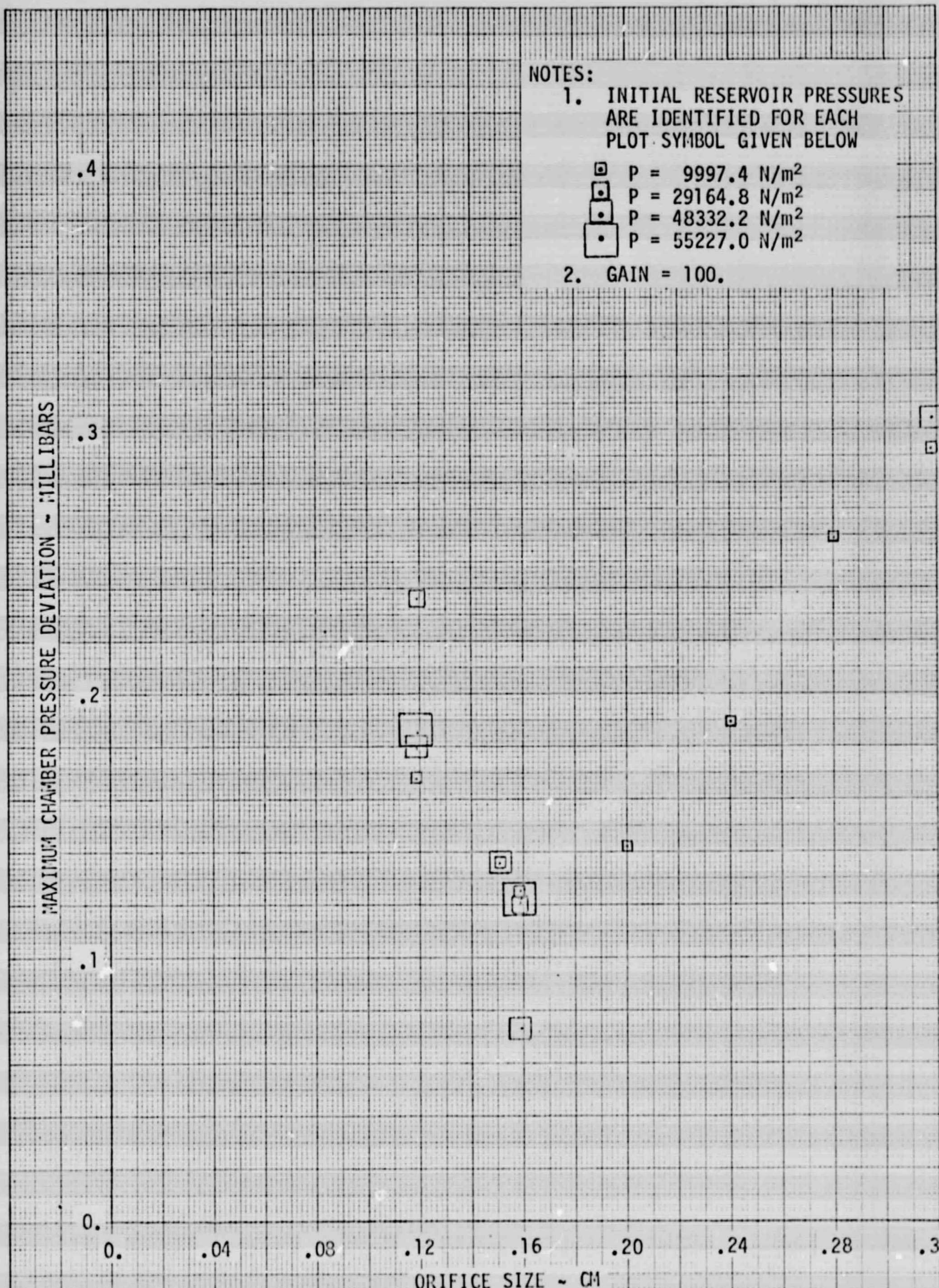


Figure 17. Summary of Maximum Deviation Versus Orifice Size.

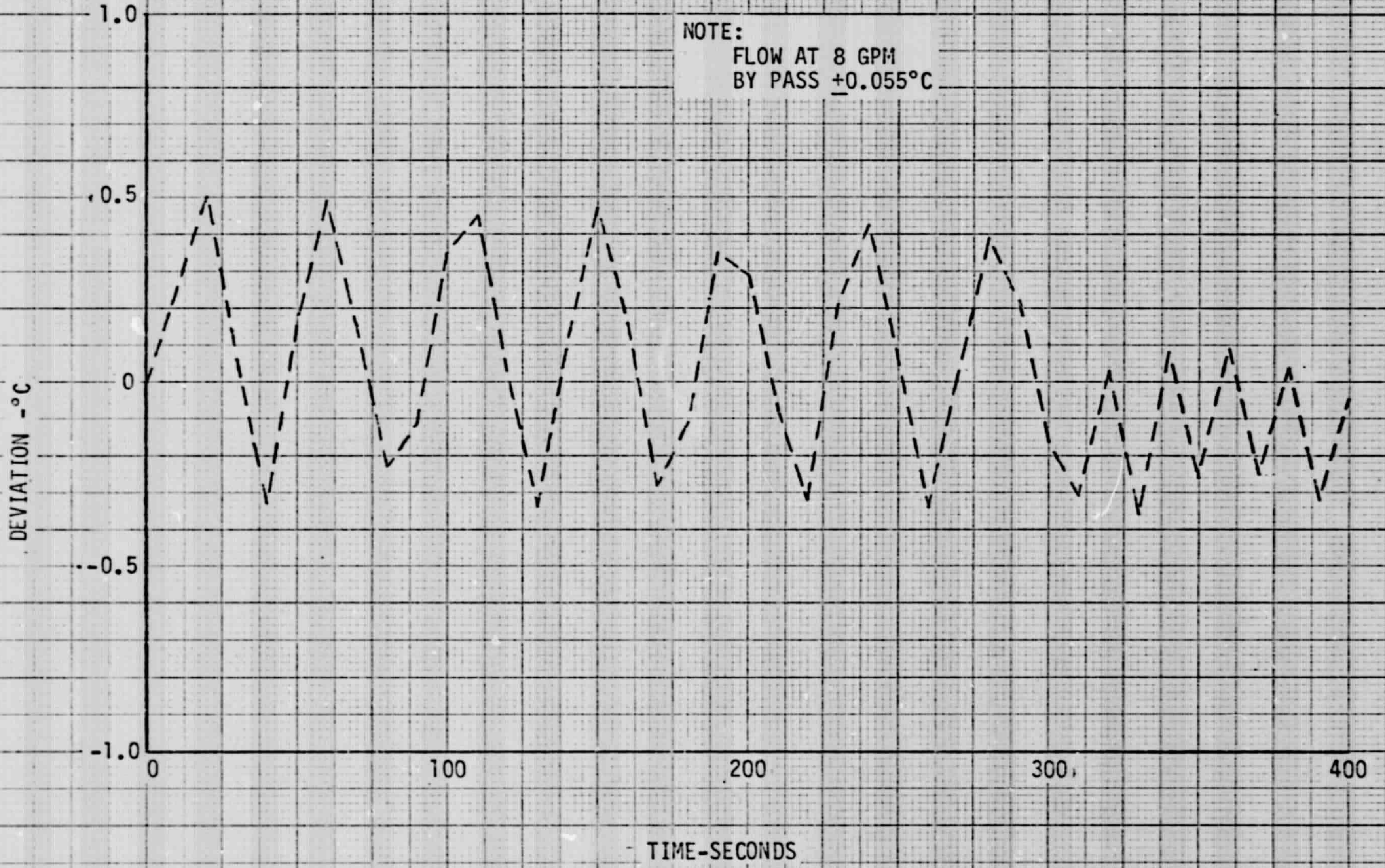


Figure 18. Configuration "2" Deviation from Required Temperature.

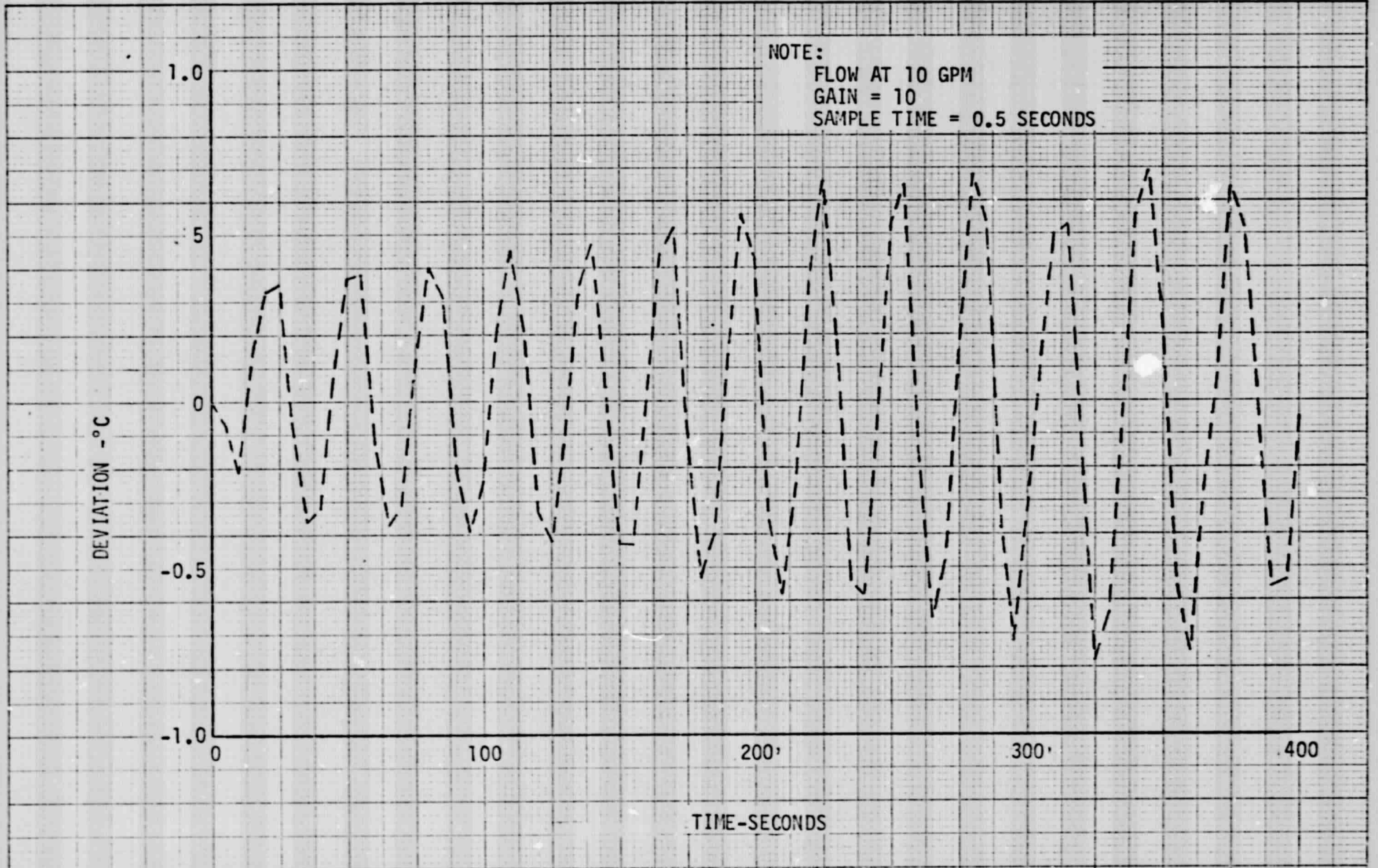


Figure 19. Configuration "4" Deviation from Required Temperature.

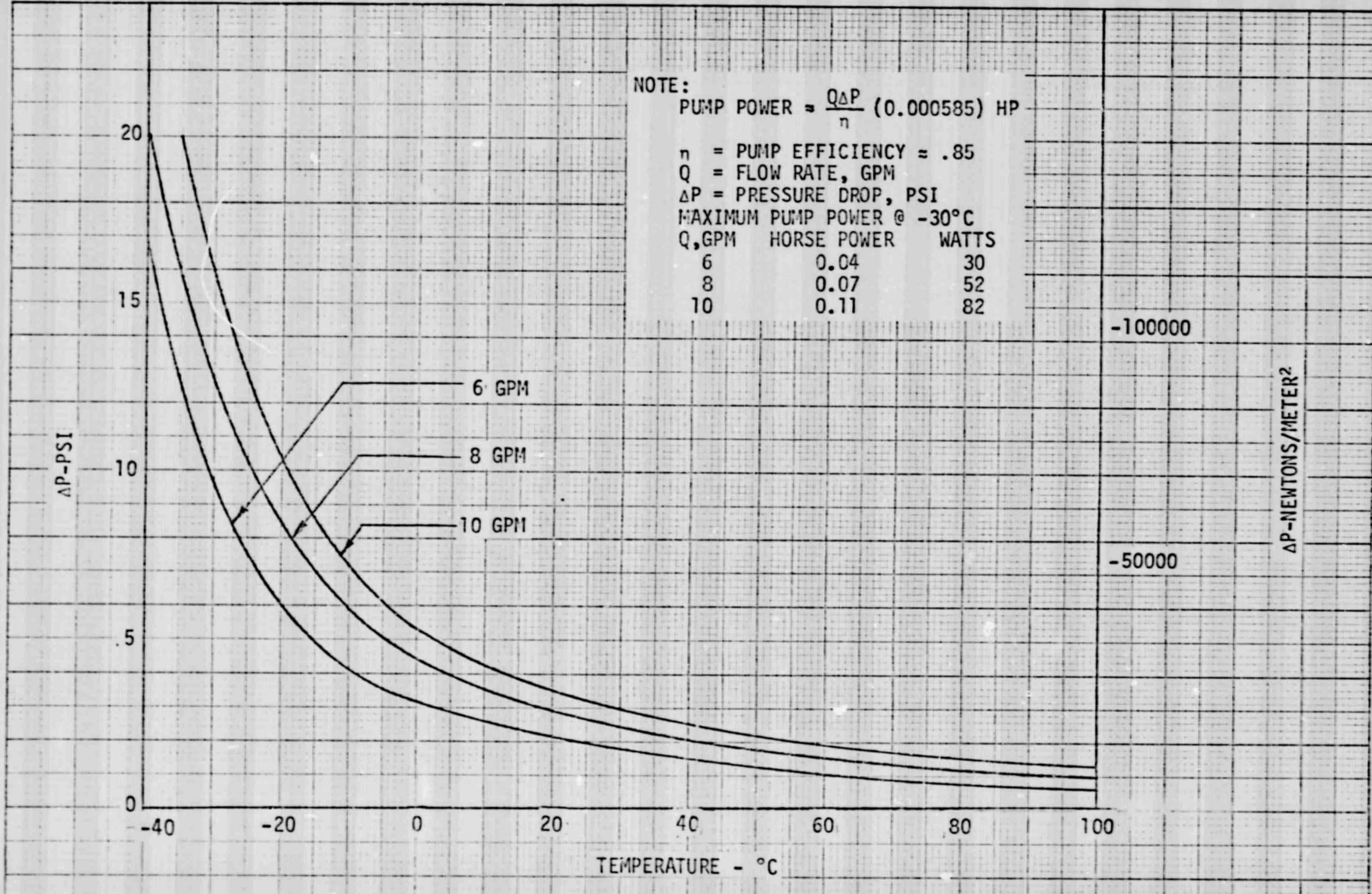


Figure 20. Predicted System Pressure Drop Versus Fluid Temperature.