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AFAPL-TR-69-68
VOLUME II

GENERATION OF INERTING GASES FOR AIRCRAFT FUEL TANKS BY CATALYTIC COMBUSTION TECHNIQUES

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R. B. Wainright
A. Perlmutter
American Cyanamid Company

TECHNICAL REPORT AFAPL-TR-69-68 VOLUME II

AUGUST 1969

AIR FORCE AERO PROPULSION LABORATORY
AIR FORCE SYSTEMS COMMAND
WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

245

GENERATION OF INERTING GASES
FOR AIRCRAFT FUEL TANKS
BY CATALYTIC COMBUSTION TECHNIQUES

Volume II

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FOREWORD

This report was prepared by the Central Research Division, American Cyanamid Company, and covers work performed under Contract Number F33615-68-C-1500, Project No. 3048 - Fuels, Lubrication, and Hazards, Task 304807, Aerospace Vehicle Hazard Protection. Administration was provided by the Air Force Aero Propulsion Laboratory, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio, with Mr. Robert E. Cretcher, APFH Hazards Branch, acting as Project Engineer. The contract was initially funded with Laboratory Director's Discretionary funds.

The effort reported herein was performed in the period April 1968 through June 1969, at Cyanamid's Stamford Research Laboratories in Stamford, Connecticut. Experimental work was performed by Messrs. J. R. Johnson, W. E. Kuehlewind, H. Y. Li, J. F. Mazur, and A. Perlmutter. Conceptual design studies were performed by Mr. A. Perlmutter. Mr. D. R. Goodrich served as Project Leader during the initial part of the program, and Mr. R. B. Wainright as acting Project Leader during the remainder of the program. Cyanamid's technical management throughout was provided in the person of Mr. Wainright.

This document was submitted by the authors in June 1969.

This technical report has been reviewed and is approved.

Robert G. Clodfelter for

Benito P. Botteri
Chief, Hazards Branch
Fuels, Lubrication and Hazards Division

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APPENDIX A

CATALYTIC COMBUSTION TEST DATA

AND

LIQUID FUEL PROPERTY DATA

TABLE I-A. SUMMARY OF DATA FOR SCREENING RUN CCP-5

Catalyst: Code E   Fuel: Propane

Hot Spot Temp. (T), °K	$1/T \times 10^3$ $°K^{-1}$	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$
563	1.776	3.27	0.0	1.000	0.000
599	1.670	2.32	29.1	1.410	0.344
663	1.510	1.86	43.2	1.759	0.565
782	1.280	1.06	67.6	3.080	1.124
852	1.173	0.47	85.7	6.950	1.940
867	1.152	0.25	92.5	13.100	2.570

TABLE II-A. SUMMARY OF DATA FOR SCREENING RUN CCP-6

Catalyst: Code A   Fuel: Propane

Hot Spot Temp. (T), °K	$1/T \times 10^3$ $°K^{-1}$	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$
577	1.732	3.00	0.0	1.000	0.000
682	1.467	2.55	15.0	1.176	0.162
726	1.378	2.14	28.7	1.400	0.336
783	1.278	1.55	48.3	1.935	0.660
833	1.200	1.33	55.7	2.260	0.815
868	1.151	1.09	63.7	2.750	1.012
1007	0.995	0.10	96.7	30.000	3.400
1023	0.977	0.00	100.0	∞	

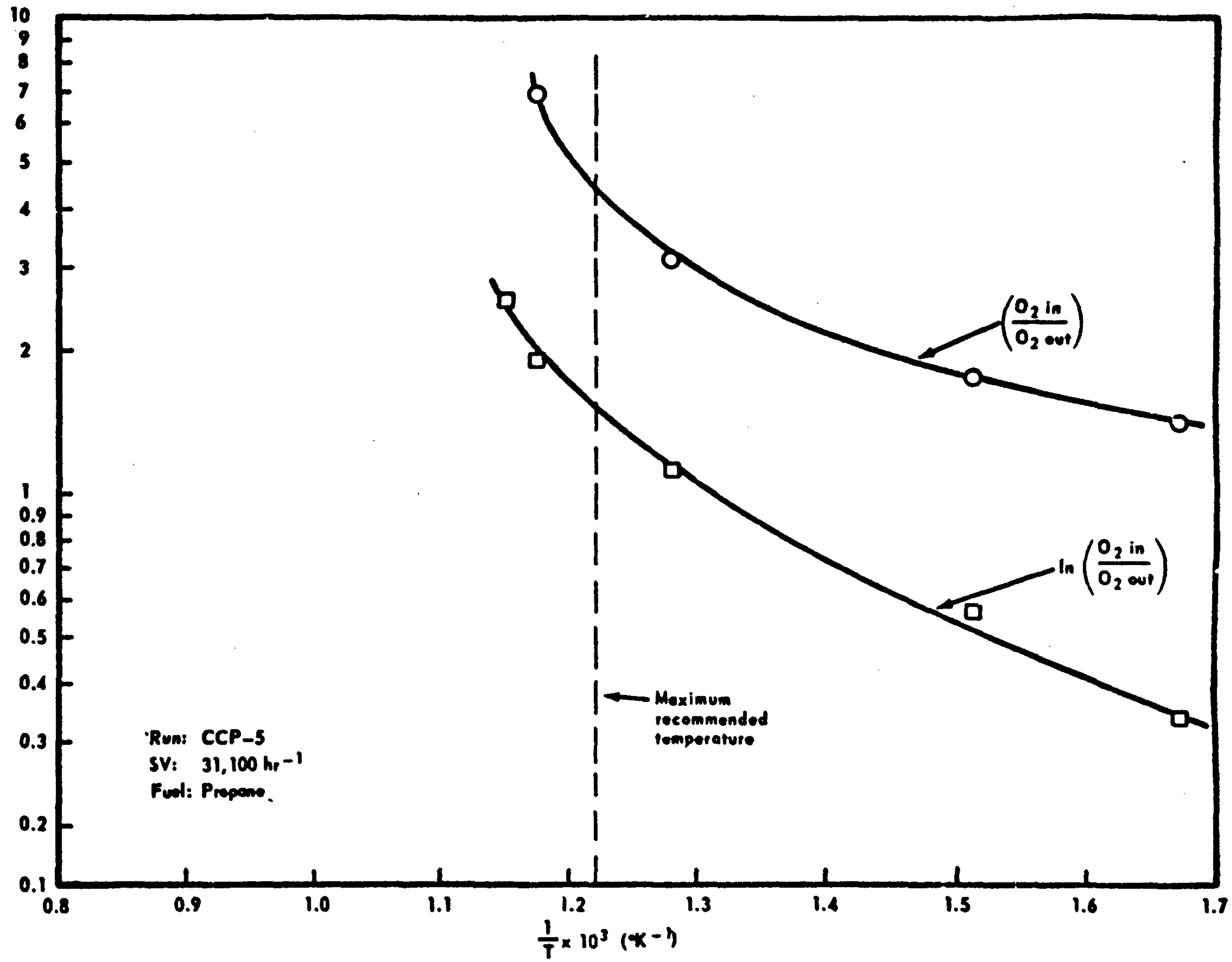


FIGURE 1-A. CODE E CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE III-A. SUMMARY OF DATA FOR SCREENING RUN CCP-7

Catalyst: Code D Fuel: Propane

Hot Spot Temp. (T), °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$
557	1.793	2.845	0.0	1.000	0.000
651	1.447	2.215	22.2	1.285	0.251
751	1.288	1.125	60.5	2.530	0.929
776	1.248	1.005	64.6	2.830	1.041
822	1.216	0.875	69.3	3.260	1.180
841	1.189	0.795	72.0	3.580	1.275

TABLE IV-A. SUMMARY OF DATA FOR SCREENING RUN CCP-9

Catalyst: Code F Fuel: Propane

Hot Spot Temp. (T), °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$
814	1.230	2.07	0.0	1.000	0.000
840	1.190	2.05	1.0	1.010	0.093
915	1.092	1.25	39.6	1.654	0.503
920	1.088	0.80	61.5	2.590	0.986
927	1.079	0.57	72.5	3.630	1.290
963	1.039	0.34	83.6	6.090	1.808
967	1.033	0.04	98.0	51.700	3.940

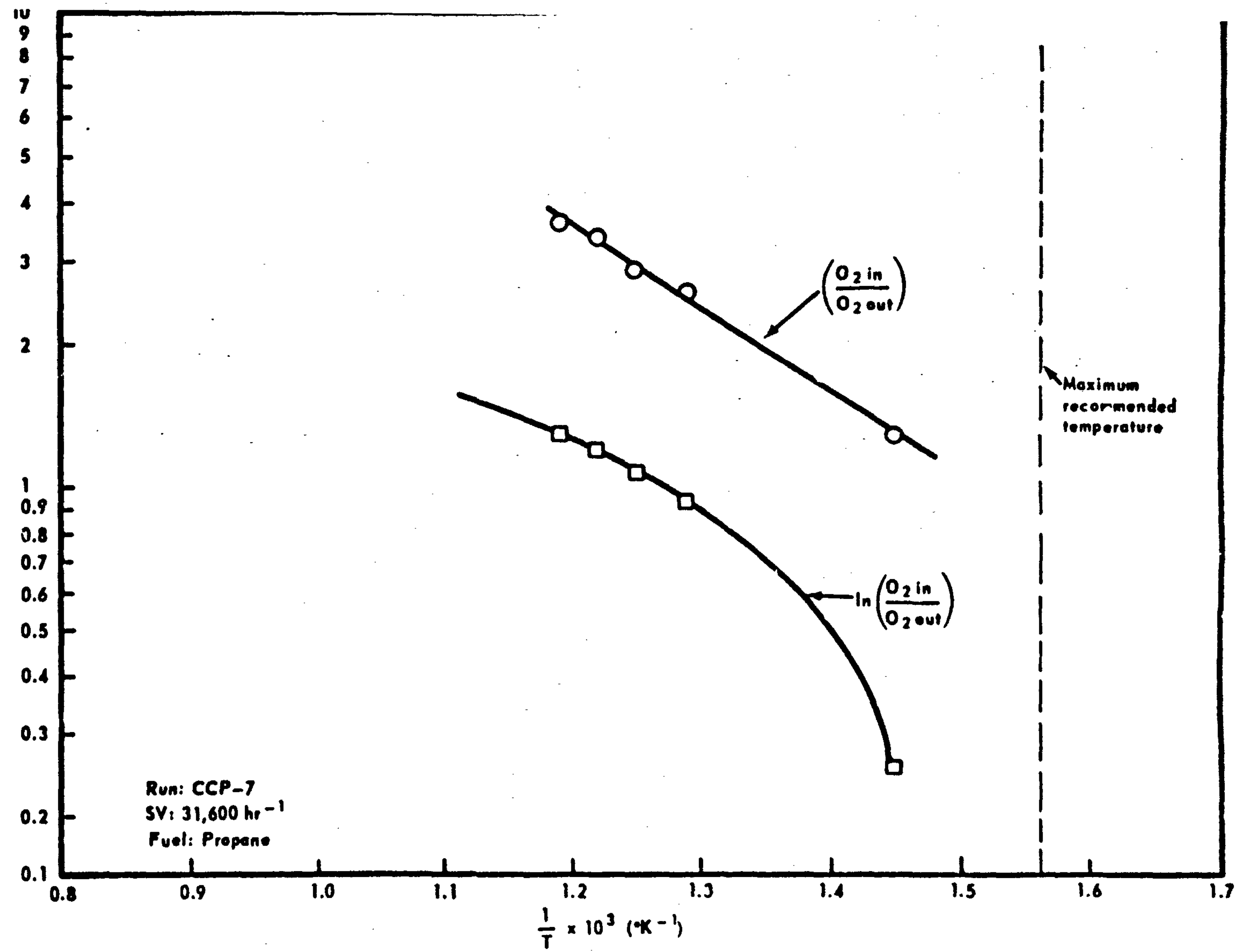


FIGURE 2-A. CODE D CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE V-A. SUMMARY OF DATA FOR SCREENING RUN CCP-15

Catalyst: Code I Fuel: Propane

Hot Spot Temp. (T), °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$
585	1.715	3.75	0.0	1.000	0.000
606	1.650	3.65	2.7	1.025	0.024
699	1.430	2.90	22.7	1.292	0.256
769	1.300	2.41	35.7	1.555	0.441
817	1.223	1.93	48.6	1.946	0.665
878	1.140	1.73	54.0	2.175	0.777
930	1.075	1.26	61.5	2.985	1.093
983	1.018	0.57	84.7	6.64	1.894

TABLE VI-A. SUMMARY OF DATA FOR SCREENING RUN CCP-17

Catalyst: Code J Fuel: Propane

Hot Spot Temp. (T) °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
616	1.622	3.63	0.0	1.000	0.000	0.000
673	1.487	3.25	10.5	1.117	0.111	
700	1.430	3.10	14.6	1.170	0.157	0.040
755	1.324	2.89	20.4	1.256	0.228	0.092
841	1.189	2.34	35.5	1.550	0.438	0.175
874	1.144	2.12	41.6	1.711	0.538	0.263
893	1.120	1.93	46.9	1.880	0.631	0.263
933	1.071	1.72	52.6	2.110	0.746	
946	1.057	1.60	55.9	2.270	0.820	0.474
970	1.030	1.42	60.9	2.560	0.964	0.533

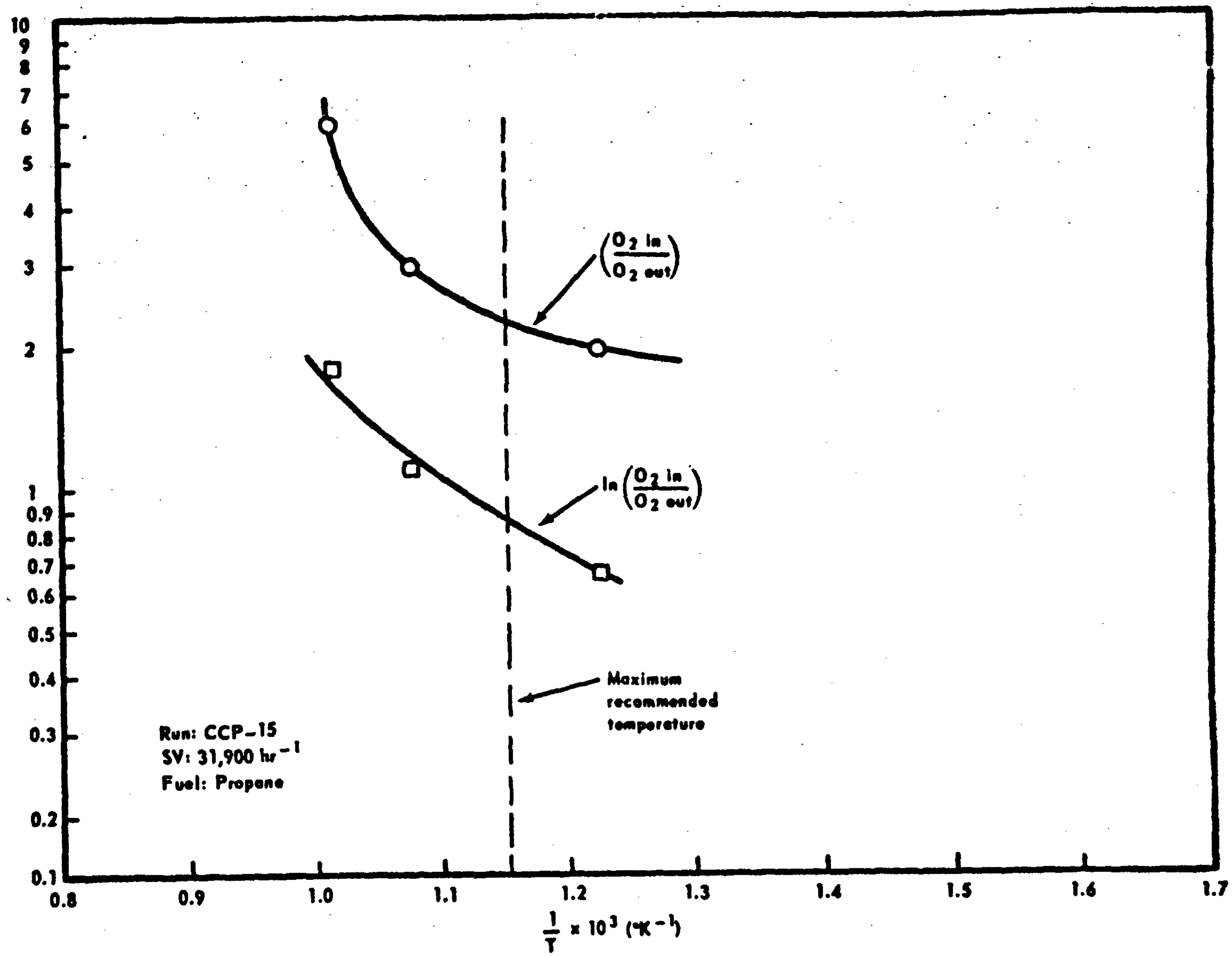


FIGURE 3-A. CODE I CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

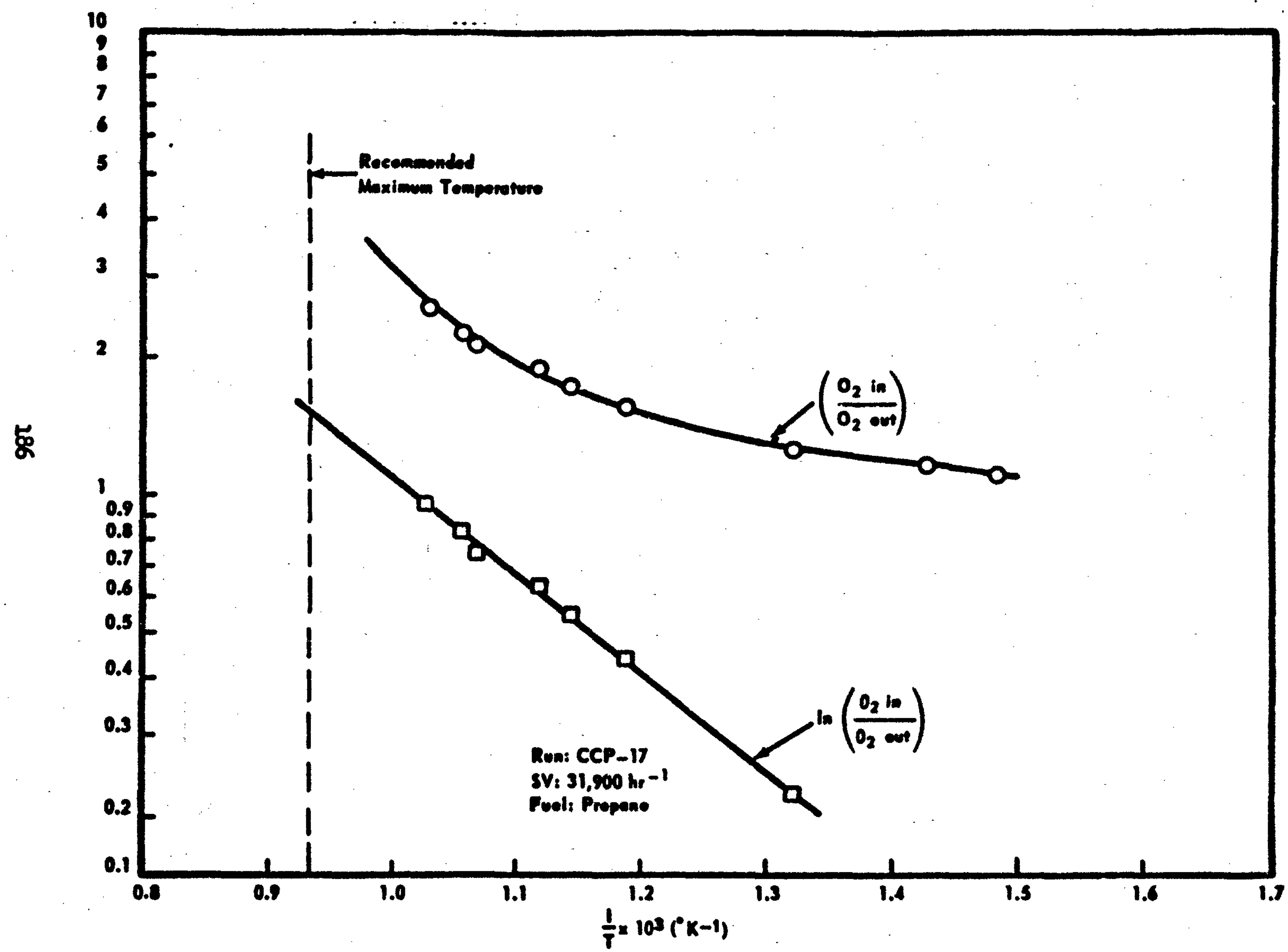


FIGURE 4–A. CODE J CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE VII-A. SUMMARY OF DATA FOR SCREENING RUN CCP-18

Catalyst: Code K   Fuel: Propane

Hot Spot Temp. (T) °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	$\frac{\text{Oxygen In}}{\text{Oxygen Out}}$	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
722	1.385	3.75	0.0	1.000	0.000	0.000
731	1.368	3.29	12.3	1.140	0.131	0.033
773	1.293	3.07	18.1	1.222	0.201	0.066
806	1.240	2.89	23.0	1.300	0.262	0.079
833	1.200	2.67	28.8	1.403	0.339	
871	1.148	2.41	35.7	1.555	0.410	0.247
915	1.093	2.07	44.8	1.812	0.595	0.369
975	1.026	1.51	59.8	2.482	0.910	0.625
1005	0.995	1.21	67.8	3.100	1.132	0.724
1015	0.985	1.21	67.8	3.100	1.132	0.704

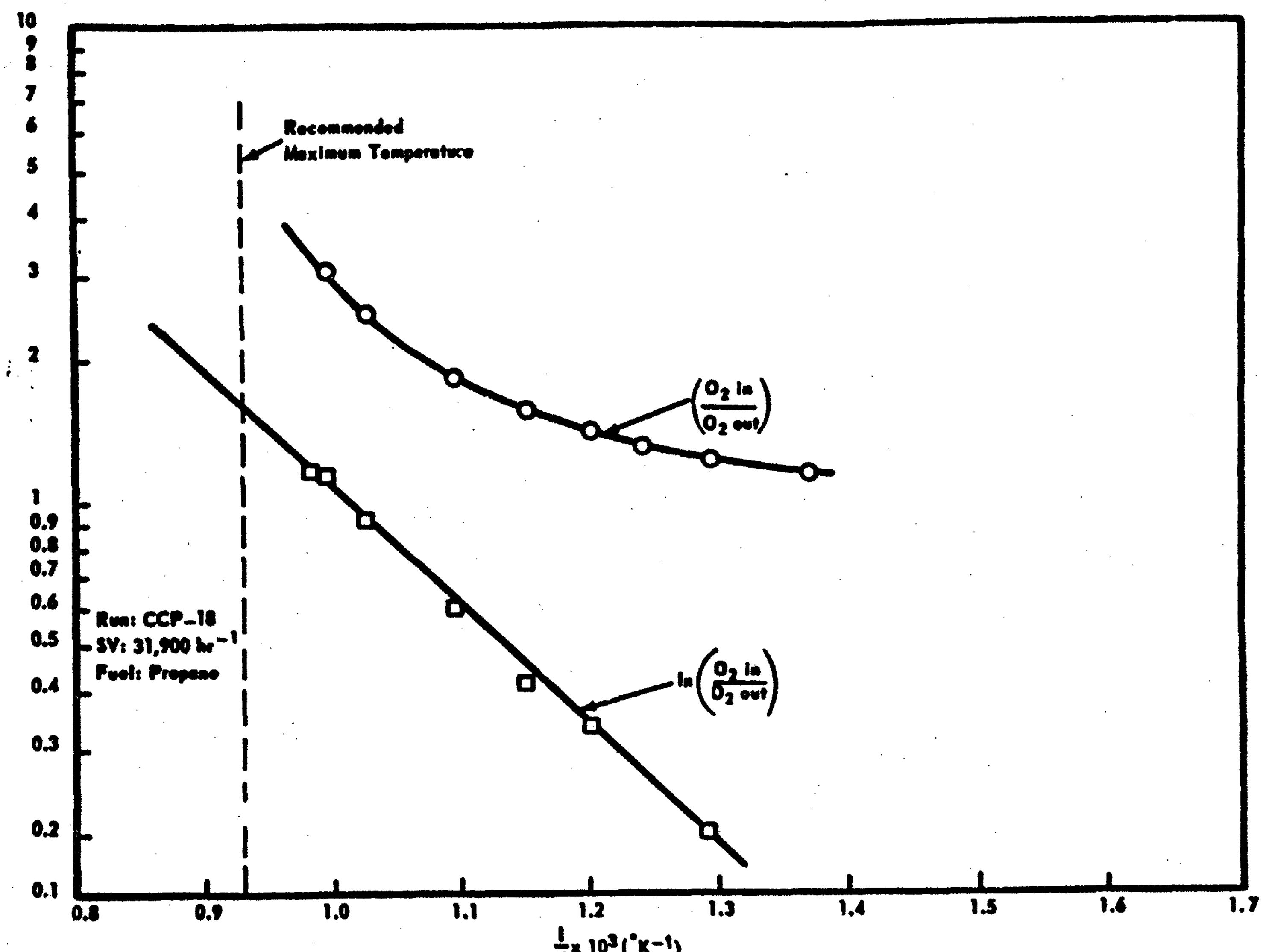


FIGURE 5–A. CODE K CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE VIII-A. SUMMARY OF DATA FOR SCREENING RUN CCP-19

Catalyst: Code F   Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
661	1.512	3.78	0.0	1.000	0.000	0.000
663	1.510	3.49	7.7	1.083	0.080	0.065
657	1.521	3.46	8.5	1.092	0.088	0.066
676	1.479	3.33	11.9	1.136	0.128	0.165
712	1.404	2.97	21.4	1.272	0.241	
743	1.347	2.81	25.6	1.345	0.296	0.395
784	1.277	2.30	39.2	1.643	0.497	0.790
825	1.212	1.74	54.0	2.17	0.777	
875	1.142	0.90	76.2	4.20	1.434	
892	1.120	0.55	85.5	6.87	1.928	1.610
992	1.008	0.22	94.1	17.20	2.847	1.810
998	1.001	0.21	94.4	18.00	2.892	
1027	0.975	0.20	94.7	18.90	2.940	1.875

TABLE IX-A. SUMMARY OF DATA FOR SCREENING RUN CCP-20

Catalyst: Code B   Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
502	1.990	3.74	0.0	1.000	0.000	0.000
610	1.661	3.65	2.4	1.025	0.025	
664	1.508	3.53	5.6	1.059	0.057	0.059
686	1.458	3.45	7.8	1.084	0.081	0.086
729	1.371	3.15	15.8	1.188	0.172	0.197
778	1.285	2.86	23.5	1.308	0.269	0.460
804	1.246	2.51	32.9	1.490	0.399	
840	1.190	2.20	41.2	1.700	0.531	0.855
871	1.148	2.00	46.5	1.870	0.626	
841	1.189	2.32	38.0	1.611	0.478	0.526
816	1.225	2.42	35.3	1.545	0.435	0.559
863	1.160	2.00	46.5	1.870	0.626	0.889

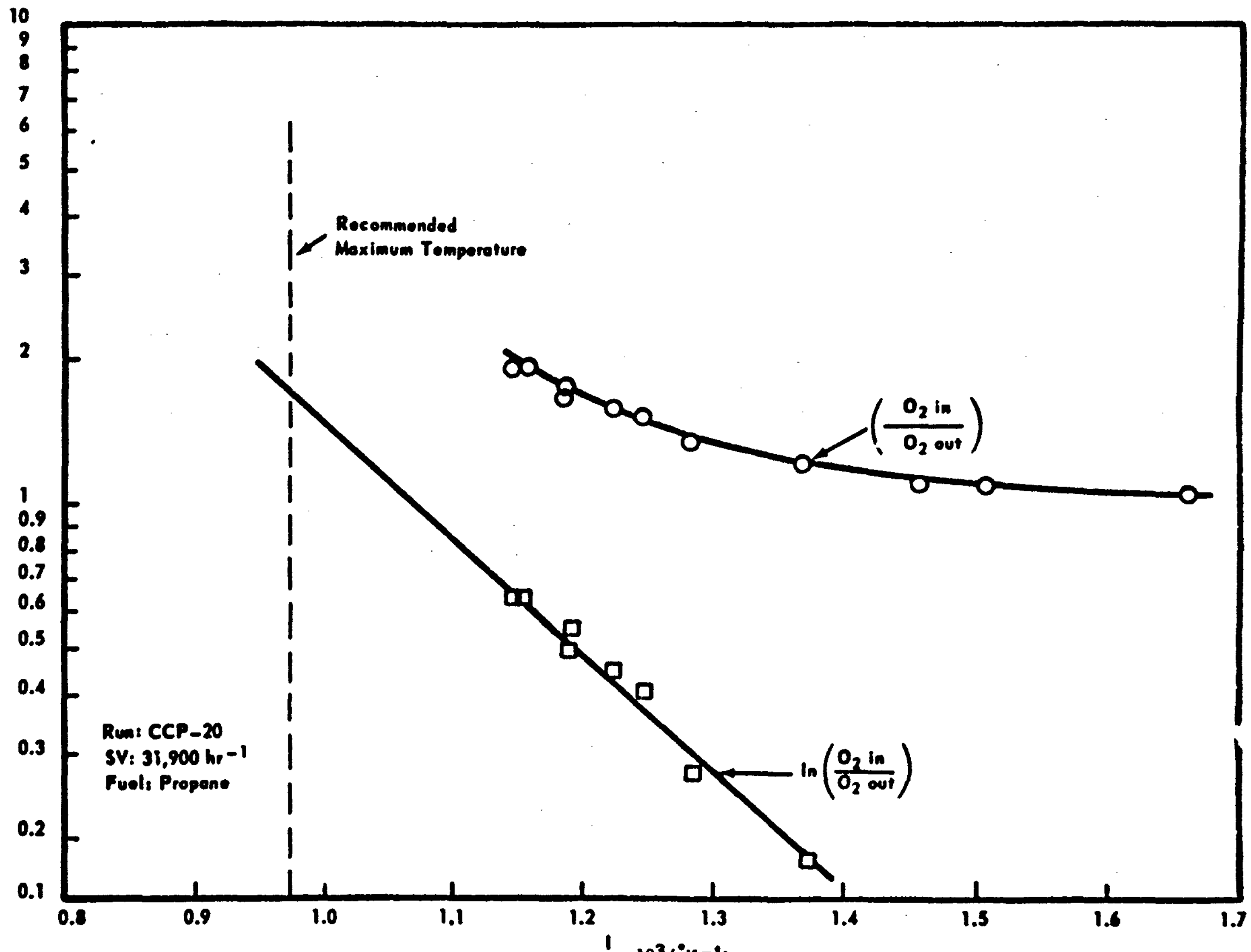


FIGURE 6-A. CODE B CATALYST - EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE X-A. SUMMARY OF DATA FOR SCREENING RUN JT-9

Catalyst: Code F Fuel: JP-7

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
580	1.722	3.75	0.0	1.000	0.000	0.000
592	1.689	3.70	1.3	1.012	0.012	
617	1.620	3.61	3.7	1.040	0.039	
641	1.560	3.53	5.9	1.062	0.060	
664	1.509	3.45	8.0	1.087	0.084	0.053
711	1.407	3.29	12.3	1.140	0.131	0.166
744	1.343	3.13	16.5	1.199	0.182	0.263
786	1.271	2.94	21.6	1.275	0.243	0.408
816	1.224	2.76	26.4	1.360	0.308	0.494
859	1.165	2.70	28.0	1.390	0.330	0.592
875	1.142	2.61	30.4	1.436	0.362	0.658

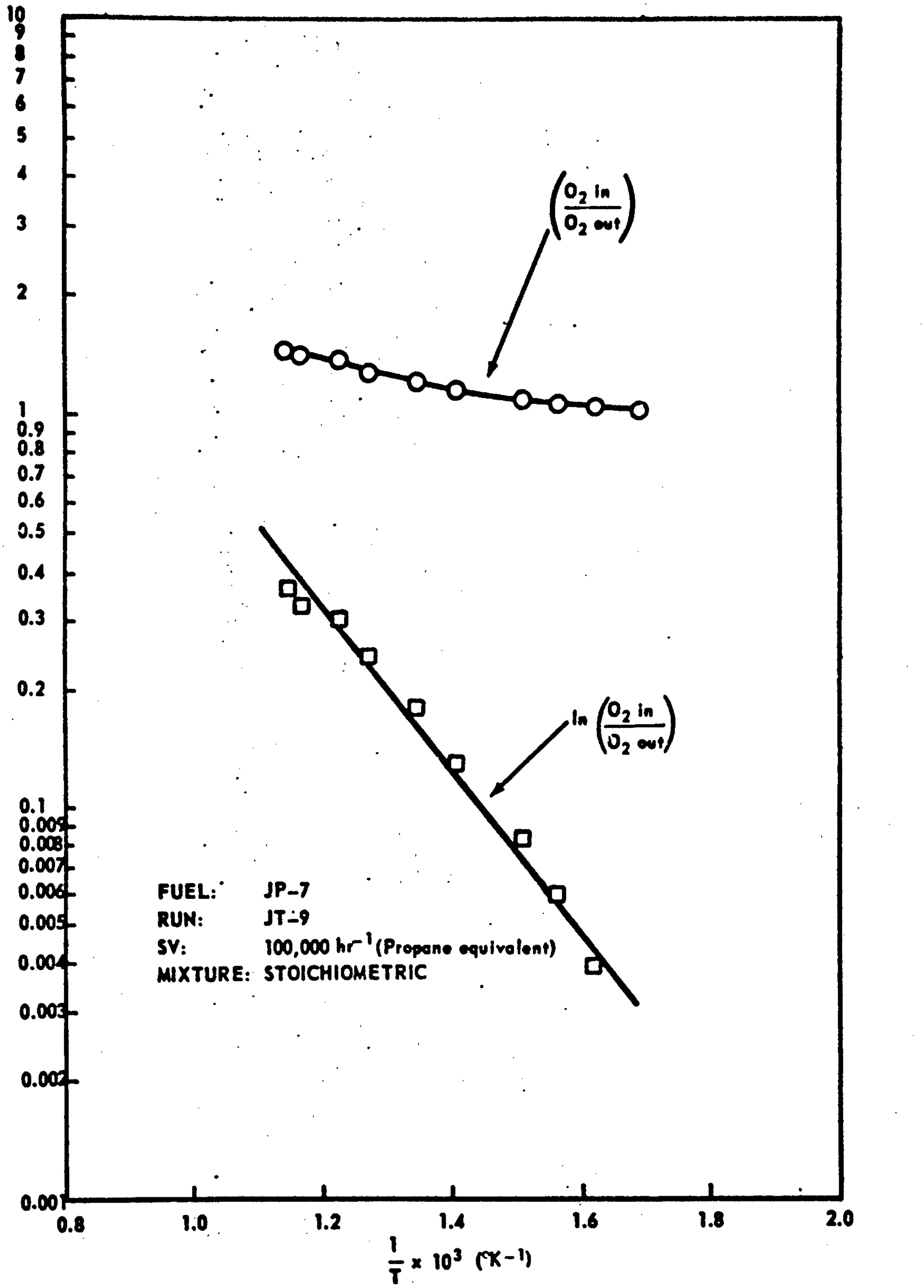


FIGURE 7-A. CODE F CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE XI-A. SUMMARY OF DATA FOR SCREENING RUN CCP-27

Catalyst: Code M   Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
640	1.561	3.73	0.0	1.000	0.000	0.000
664	1.508	3.70	0.8	1.008	0.008	0.033
688	1.451	3.67	1.6	1.017	0.017	
702	1.424	3.60	3.5	1.037	0.036	0.046
729	1.371	3.52	5.6	1.060	0.058	0.099
731	1.369	3.48	6.7	1.072	0.070	
739	1.352	3.43	8.1	1.089	0.085	0.132
770	1.300	3.28	12.1	1.138	0.129	0.224
796	1.256	3.06	18.0	1.220	0.199	0.335
817	1.223	2.90	22.2	1.287	0.254	
829	1.208	2.80	24.9	1.332	0.287	0.480
852	1.172	2.60	30.3	1.434	0.360	
854	1.170	2.59	30.6	1.440	0.365	0.502
875	1.141	2.42	35.2	1.540	0.432	0.658
878	1.140	2.41	35.4	1.548	0.437	

TABLE XII-A. SUMMARY OF DATA FOR SPACE VELOCITY STUDY, RUN CCP-24

Catalyst: Code F   Fuel: Propane

Hot Spot Temp. (T) °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	$\frac{\text{Oxygen In}}{\text{Oxygen Out}}$	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
503	1.988	3.49	0.0	1.000	0.000	0.000
507	1.971	3.42	2.0	1.020	0.020	
518	1.930	3.32	4.9	1.050	0.049	
544	1.840	3.26	6.6	1.070	0.068	0.033
580	1.723	3.15	9.8	1.108	0.103	
607	1.648	2.87	17.8	1.217	0.197	0.247
647	1.546	2.47	29.2	1.413	0.346	0.408
664	1.509	2.24	35.8	1.560	0.445	
721	1.387	1.60	54.1	2.180	0.780	0.987
777	1.288	0.97	72.2	3.600	1.280	
799	1.251	0.80	77.1	4.365	1.474	1.670
826	1.210	0.70	80.0	4.990	1.608	
843	1.187	0.64	81.6	5.450	1.697	1.718
854	1.171	0.63	82.0	5.540	1.710	
868	1.151	0.53	84.8	6.590	1.885	1.585

TABLE XIII-A. SUMMARY OF DATA FOR SPACE VELOCITY STUDY, RUN CCP-22

Catalyst: Code F Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
676	1.480	3.78	0.0	1.000	0.000	0.000
704	1.421	2.03	46.3	1.860	0.620	0.296
791	1.262	1.70	55.0	2.220	0.797	
869	1.150	0.56	85.3	6.750	1.910	1.612
927	1.080	0.46	87.9	8.220	2.110	
978	1.022	0.42	88.9	9.000	2.200	2.050
1011	0.989	0.40	89.5	9.450	2.380	1.920
1014	0.985	0.31	91.8	12.200	2.500	2.140

TABLE XIV-A. SUMMARY OF DATA FOR SPACE VELOCITY STUDY, RUN CCP-8

Catalyst: Code A Fuel: Propane

Hot Spot Temp. (T), °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)
645	1.550	2.92	0.0	1.000	0.000
667	1.500	2.71	7.4	1.078	0.075
730	1.370	2.26	22.6	1.291	0.255
817	1.224	1.60	45.3	1.823	0.621
944	1.060	0.52	82.2	5.610	1.725
966	1.035	0.38	87.0	7.690	2.040
979	1.021	0.24	91.8	12.170	2.500

TABLE XV-A. SUMMARY OF DATA FOR SPACE VELOCITY STUDY, RUN CCP-23

Catalyst: Code A   Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
488	2.050	3.70	0.0	1.000	0.000	0.000
586	1.705	3.54	4.3	1.045	0.044	
641	1.560	3.22	13.0	1.150	0.140	
697	1.434	2.88	22.2	1.285	0.251	0.447
746	1.340	2.35	36.5	1.573	0.456	
809	1.238	1.72	53.5	2.152	0.768	1.119
946	1.057	0.42	88.7	8.805	2.175	
987	1.012	0.19	95.0	19.500	2.972	1.843
1006	0.995	0.11	97.0	33.620	3.515	
1014	0.985	0.10	97.4	37.000	3.612	1.875
1018	0.983	0.06	98.5	61.700	4.120	1.850
1026	0.975	0.03	99.2	123.300	4.810	1.816
1029	0.972	0.02	99.5	185.000	5.220	1.940
1052	0.950	0.01	99.8	370.000	5.91	1.700

TABLE XVI. A SUMMARY OF DATA FOR LIQUID FUEL RUN JT-2

Catalyst: Code A   Fuel: JP-7

Hot Spot Temp. (T) °K	$1/T \times 10^3$ °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
523	1.910	3.58	0.0	1.000	0.000	0.000
575	1.740	3.45	3.6	1.038	0.037	
591	1.690	3.33	7.0	1.076	0.073	
610	1.640	3.15	12.0	1.137	0.128	
624	1.602	2.86	20.1	1.252	0.225	
640	1.561	2.65	26.0	1.350	0.300	
688	1.452	1.83	48.9	1.958	0.671	
728	1.373	1.60	55.4	2.240	0.806	1.198
774	1.292	1.50	58.1	2.390	0.872	
832	1.200	1.20	66.5	2.985	1.093	
864	1.160	1.10	69.3	3.260	1.180	1.449
876	1.140	1.00	72.0	3.580	1.275	
925	1.080	0.80	77.6	4.475	1.500	1.670
970	1.030	0.55	81.9	6.510	1.873	
1019	0.982	0.27	92.5	13.260	2.580	1.870
1047	0.955	0.14	96.1	25.600	3.240	
1057	0.946	0.10	97.2	35.800	3.580	2.040
1064	0.940	0.09	97.5	39.800	3.680	2.054
1066	0.938	0.09	97.5	39.800	3.680	
1068	0.936	0.07	98.0	51.200	3.940	1.996

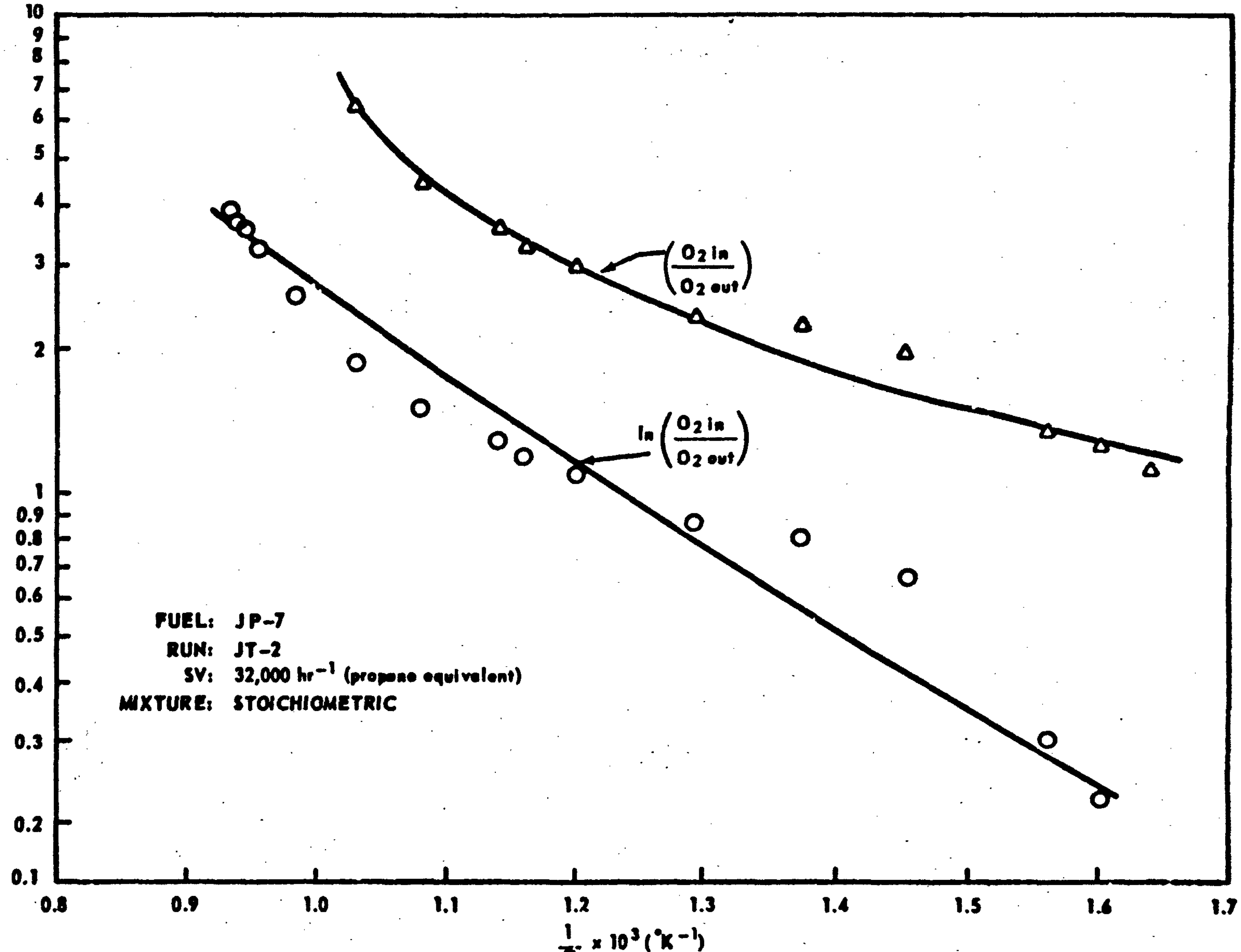


FIGURE 8-A. CODE A CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE XVII-A. SUMMARY OF DATA FOR LIQUID FUEL RUN JT-4

Catalyst: Code A   Fuel: JP-7

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
573	1.747	3.58	0.0	1.0	0.000	0.000
631	1.584	3.33	7.0	1.076	0.073	0.165
769	1.300	2.10	41.4	1.703	0.532	
788	1.270	1.50	58.1	2.390	0.870	
820	1.220	1.20	66.5	2.980	1.092	
855	1.170	0.85	76.3	4.210	1.438	
885	1.130	0.68	81.0	5.270	1.660	
913	1.096	0.50	86.0	7.160	1.970	0.987
980	1.020	0.41	88.5	8.730	2.170	1.122
1003	0.996	0.38	89.5	9.430	2.240	
1009	0.992	0.28	92.2	12.800	2.550	1.083
1022	0.979	0.20	94.5	17.900	2.880	
1024	0.976	0.18	95.0	19.900	2.990	1.042
1030	0.970	0.15	95.8	23.900	3.170	
1039	0.963	0.11	96.9	32.600	3.480	0.969

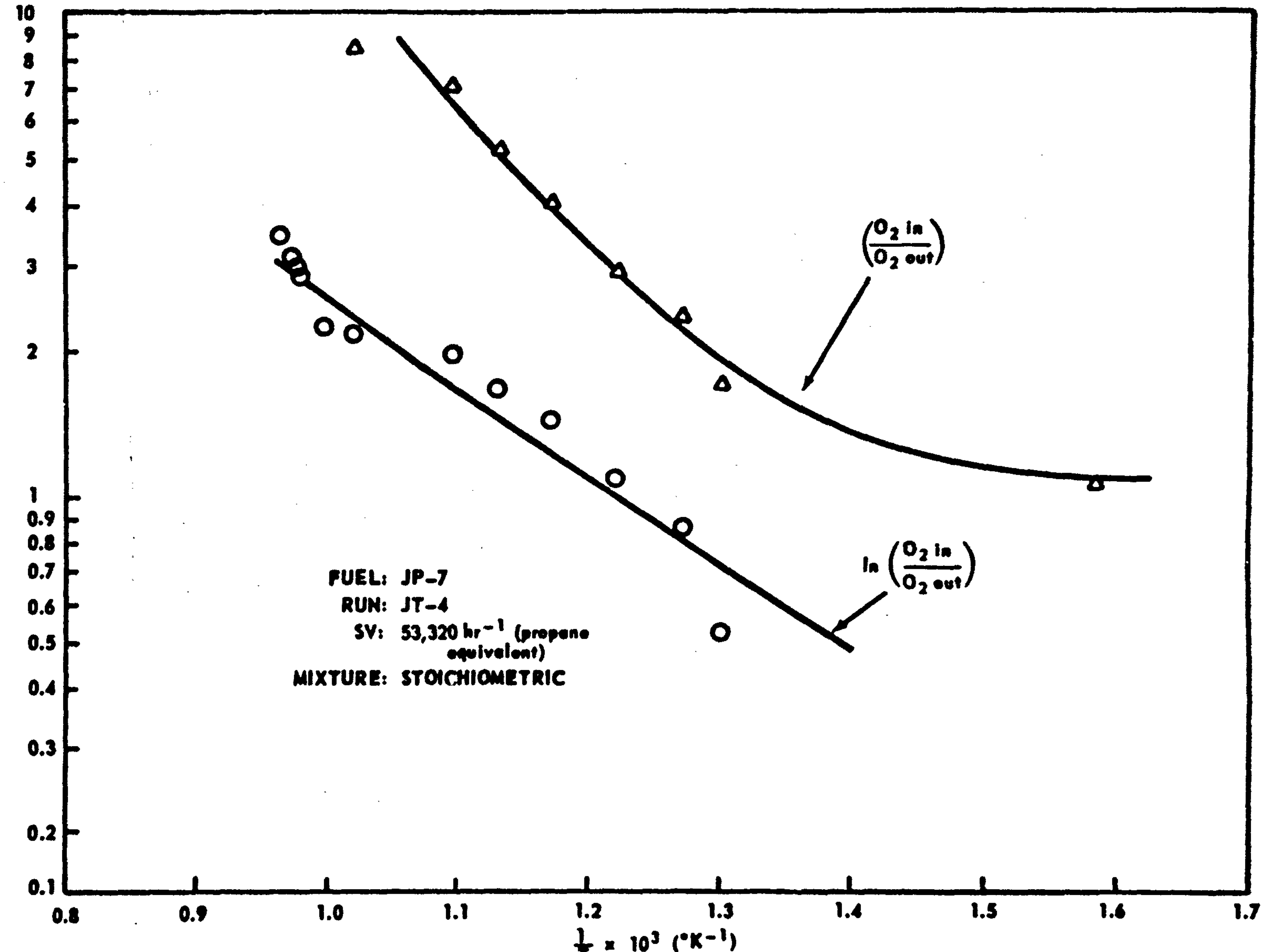


FIGURE 9-A. CODE A CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE XVIII-A. SUMMARY OF DATA FOR LIQUID FUEL RUN JT-5

Catalyst: Code A Fuel: JF-7

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
510	1.960	4.0	0.0	1.000	0.000	0.000
558	1.790	3.88	3.0	1.031	0.036	
585	1.710	3.65	8.8	1.096	0.092	
604	1.658	3.50	12.5	1.143	0.134	
616	1.621	3.41	14.8	1.172	0.159	
658	1.520	3.22	19.5	1.242	0.217	
664	1.508	3.10	22.5	1.290	0.255	0.901
691	1.448	2.84	29.0	1.410	0.344	
723	1.381	2.55	36.3	1.570	0.451	
744	1.346	2.40	40.0	1.667	0.511	0.920
781	1.280	2.20	45.0	1.720	0.543	
803	1.246	2.05	48.8	1.950	0.669	1.085
850	1.176	1.84	54.0	2.175	0.778	
876	1.140	1.60	60.0	2.500	0.916	
938	1.066	1.32	67.0	3.030	1.110	
965	1.036	1.20	70.0	3.335	1.205	
997	1.002	1.08	73.0	3.700	1.310	
1007	0.995	1.00	75.0	4.000	1.387	1.83
1044	0.957	0.92	77.0	4.350	1.470	
1063	0.940	0.75	81.3	5.335	1.673	
1072	0.931	0.63	84.3	6.350	1.850	

TABLE XIX-A. SUMMARY OF DATA FOR LIQUID FUEL RUN JT-6A

Catalyst: Code A   Fuel: JP-7

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
546	1.830	2.76	0.0	1.000	0.000	0.000
576	1.735	2.53	8.3	1.090	0.086	0.099
599	1.670	2.49	9.8	1.110	0.104	
612	1.632	2.40	13.0	1.150	0.140	0.197
647	1.546	2.10	23.9	1.315	0.274	
657	1.521	2.00	27.6	1.380	0.322	0.296
699	1.430	1.90	31.2	1.452	0.374	
735	1.360	1.75	36.6	1.579	0.457	0.592
756	1.320	1.66	39.9	1.662	0.508	
779	1.283	1.60	42.0	1.725	0.545	
805	1.241	1.48	46.4	1.866	0.624	0.658
840	1.190	1.30	53.0	2.122	0.754	0.786
850	1.176	1.25	54.8	2.210	0.795	0.942
866	1.154	1.20	56.5	2.300	0.834	1.034
873	1.147	1.18	57.3	2.340	0.850	

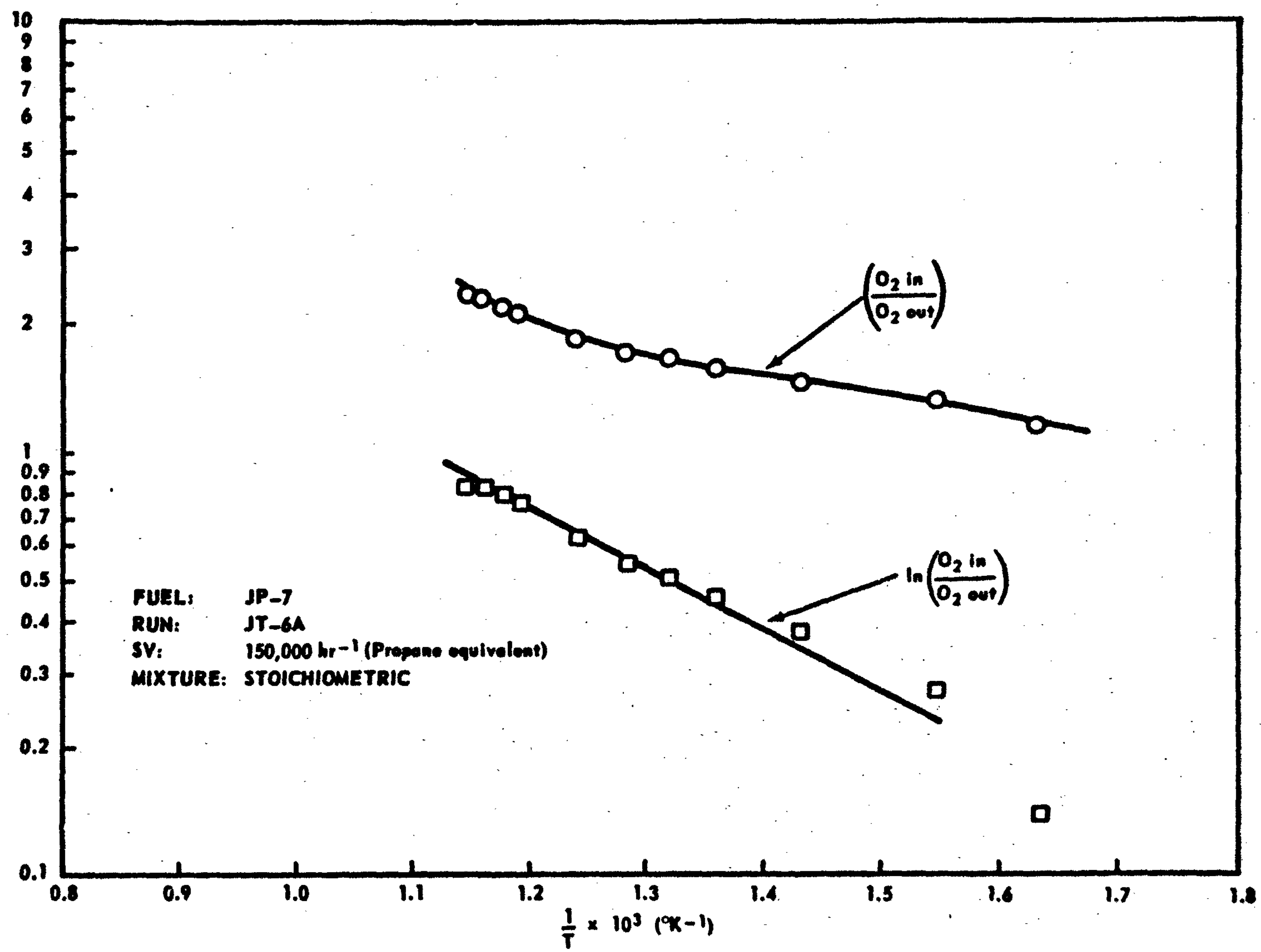


FIGURE 10-A. CODE A CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE XX-A. SUMMARY OF DATA FOR LIQUID FUEL RUN JT-6B

Catalyst: Code A Fuel: JP-7

Hot Spot Temp. (T), °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)
510	1.960	3.66	0.0	1.000	0.000
725	1.380	2.35	35.8	1.559	0.444
752	1.330	2.20	39.9	1.664	0.510
785	1.273	2.00	45.4	1.830	0.605
816	1.223	1.80	50.9	2.033	0.710
850	1.177	1.60	56.3	2.288	0.828
890	1.122	1.40	61.8	2.615	0.960
908	1.100	1.30	64.5	2.815	1.034
934	1.071	1.20	67.2	3.050	1.116
943	1.060	1.19	67.5	3.075	1.123

TABLE XXI-A. SUMMARY OF DATA FOR LIQUID FUEL RUN JT-7

Catalyst: Code A Fuel: JP-7

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
538	1.860	4.04	0.0	1.000	0.000	0.000
595	1.680	3.50	13.4	1.153	0.142	
635	1.575	3.00	25.8	1.346	0.297	
688	1.452	2.50	38.2	1.617	0.481	
723	1.383	1.90	53.0	2.125	0.755	
817	1.223	1.10	72.9	3.670	1.300	
875	1.142	0.60	85.2	6.740	1.910	1.998
923	1.082	0.30	92.6	13.480	2.600	
949	1.053	0.20	95.2	20.200	3.005	2.010
956	1.046	0.18	95.6	22.450	3.110	1.922
1005	0.995	0.16	96.1	25.250	3.230	1.960

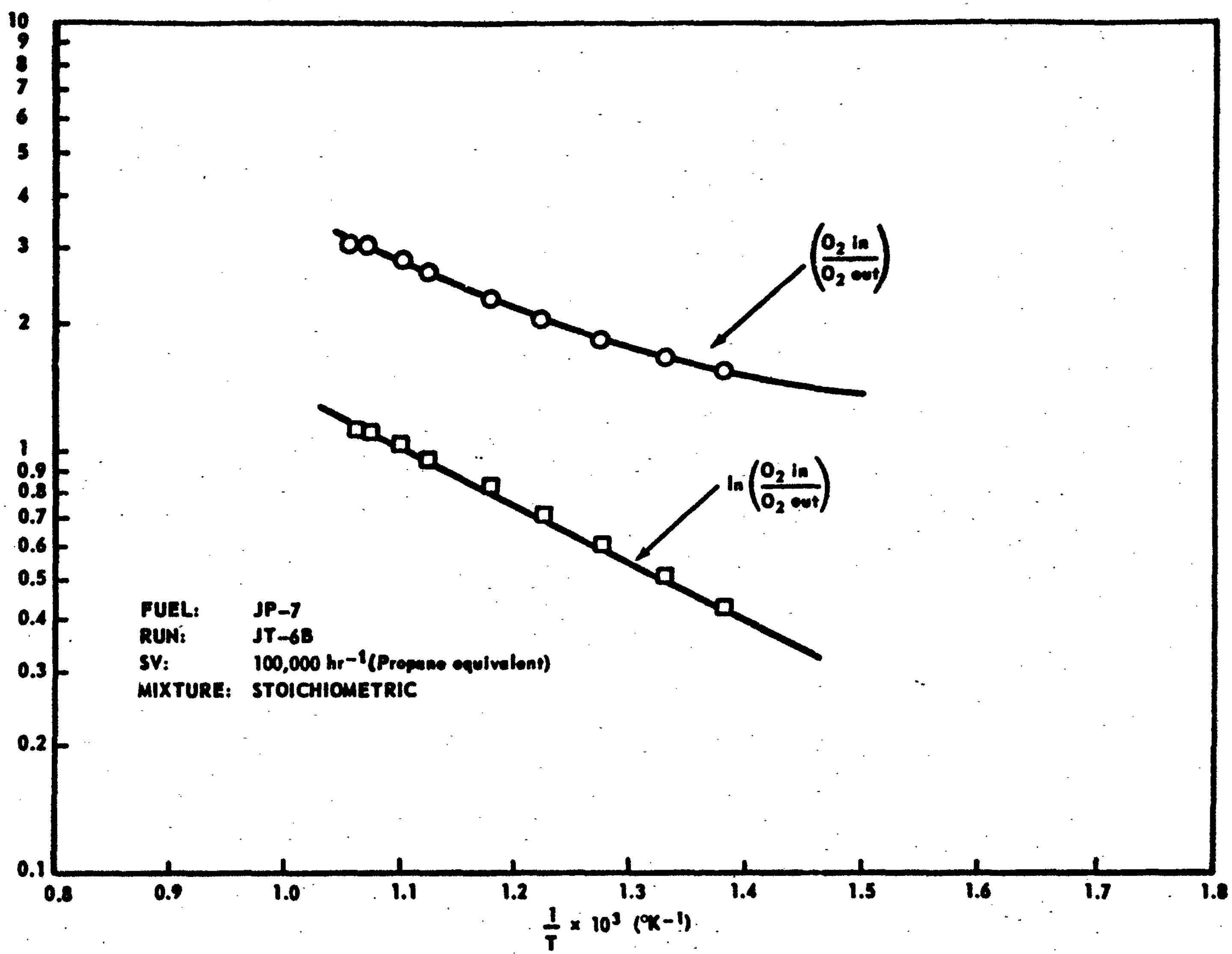


FIGURE 11-A. CODE A CATALYST – EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

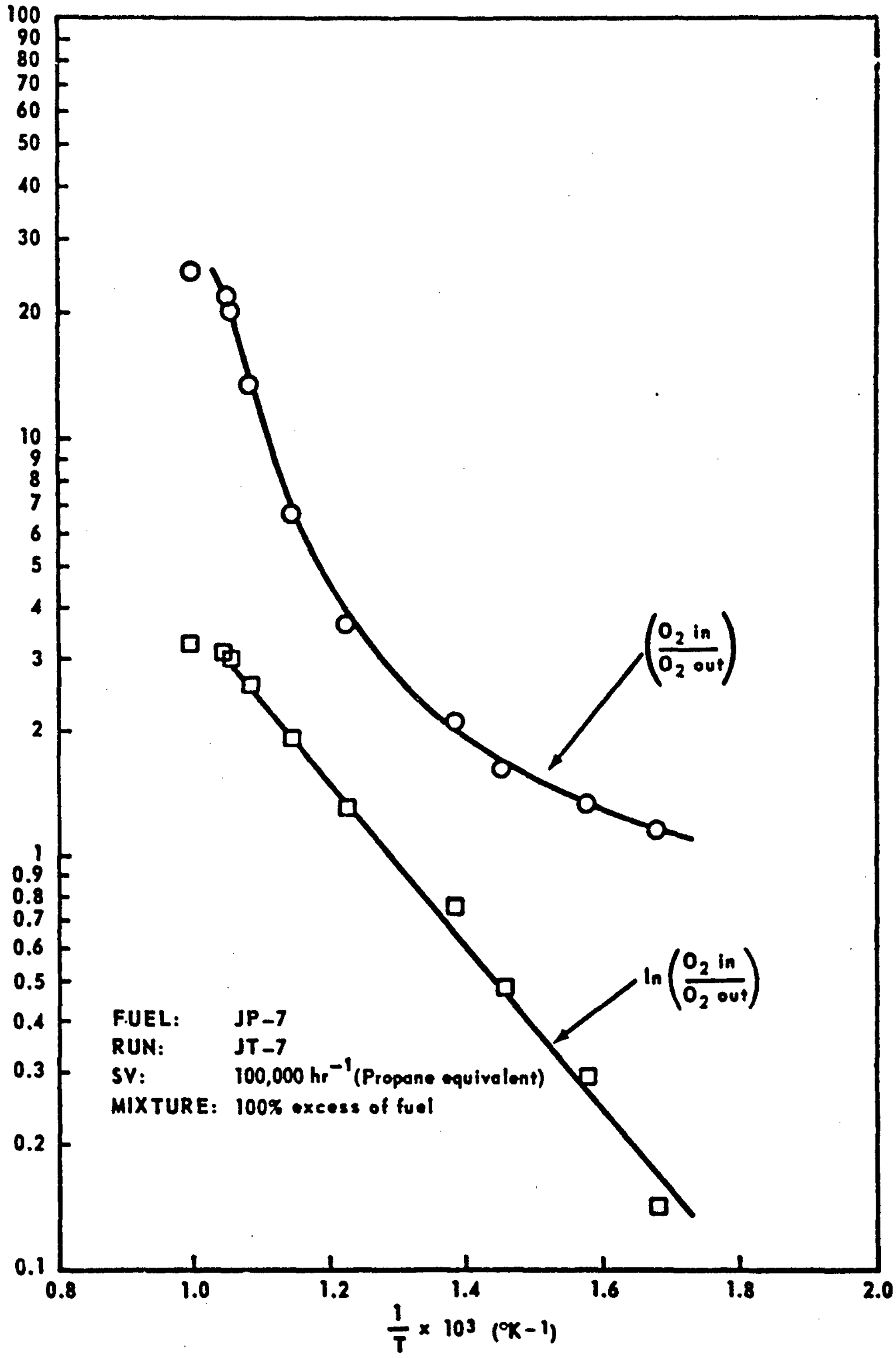


FIGURE 12-A. CODE A CATALYST - EFFECT OF TEMPERATURE ON OXYGEN CONVERSION

TABLE XXII-A. SUMMARY OF DATA FOR RUN FIM-1A WITH EXCESS OXYGEN

Catalyst: Code A Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
542	1.842	20.80	0.00	1.000	0.000	0.000
578	1.730	20.75	1.01	1.002	0.003	
628	1.591	20.60	4.04	1.010	0.010	0.053
664	1.508	20.40	8.08	1.020	0.020	0.131
687	1.459	20.20	12.12	1.030	0.030	
711	1.407	20.00	16.16	1.040	0.040	
784	1.277	19.40	28.28	1.072	0.070	
824	1.213	19.00	36.36	1.094	0.090	1.210
923	1.083	18.00	56.56	1.155	0.144	
1005	0.995	17.50	66.66	1.189	0.173	
1069	0.935	17.00	76.76	1.223	0.202	
1123	0.890	16.40	88.88	1.269	0.238	3.220

TABLE XXIII-A. SUMMARY OF DATA FOR RUN FLM-13 WITH EXCESS OXYGEN

Catalyst: Code A Fuel: Propane

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
559	1.790	20.80	0.00	1.000	0.000	0.000
570	1.752	20.75	1.01	1.002	0.003	
591	1.690	20.70	2.02	1.005	0.007	
604	1.658	20.60	4.04	1.010	0.010	
627	1.593	20.50	6.06	1.015	0.015	0.092
643	1.557	20.40	8.08	1.020	0.020	
676	1.479	20.10	14.14	1.033	0.032	0.342
724	1.400	19.60	24.24	1.061	0.059	
775	1.290	18.90	38.38	1.100	0.095	
857	1.158	17.90	58.58	1.161	0.149	
944	1.060	17.00	76.76	1.223	0.202	
982	1.019	16.50	86.86	1.260	0.232	
1054	0.950	16.30	90.90	1.276	0.244	2.630
1088	0.920	16.20	92.92	1.283	0.250	2.770

TABLE XXIV-A. SUMMARY OF DATA FOR PROPANE RUN CCP-26

Catalyst: Code A

Hot Spot Temp. (T) °K	1/T x 10^3 °K^{-1}	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	Carbon Dioxide Conc., % Vol.
583	1.715	3.58	0.0	1.000	0.000	0.000
586	1.705	3.48	2.8	1.029	0.029	
614	1.630	3.39	5.3	1.056	0.055	0.072
638	1.570	3.22	10.1	1.110	0.105	
654	1.530	3.09	13.7	1.159	0.148	
685	1.460	2.90	19.0	1.235	0.211	0.434
780	1.280	2.10	41.4	1.705	0.534	
840	1.190	1.60	55.3	2.240	0.806	
887	1.130	1.10	69.3	3.255	1.180	
911	1.098	0.80	77.7	4.475	1.500	
954	1.050	0.40	88.8	8.950	2.190	
990	1.010	0.22	93.9	16.260	2.790	2.150
1006	0.995	0.18	95.0	19.900	2.990	
1027	0.974	0.17	95.3	21.100	3.050	1.860

TABLE XXV-A. SUMMARY OF DATA FOR PROPANE RUN CCP-28

Catalyst: Code A

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	Exit Oxygen Conc., % Vol.	% Oxygen Conversion	Oxygen In / Oxygen Out	$\ln\left(\frac{\text{Oxy In}}{\text{Oxy Out}}\right)$	Carbon Dioxide Conc., % Vol.
573	1.745	3.71	0.0	1.000	0.000	0.000
604	1.657	3.69	0.5	1.005	0.005	
616	1.621	3.65	1.6	1.016	0.016	
628	1.591	3.60	3.0	1.030	0.030	
642	1.559	3.55	4.3	1.045	0.044	
652	1.531	3.50	5.7	1.060	0.058	0.098
674	1.482	3.40	8.4	1.090	0.086	
685	1.460	3.30	11.1	1.122	0.115	0.362
725	1.380	3.00	19.1	1.238	0.214	
735	1.360	2.90	21.8	1.280	0.247	0.428
766	1.304	2.60	29.9	1.428	0.356	
798	1.252	2.30	38.0	1.611	0.477	
838	1.192	1.90	48.8	1.951	0.669	0.889
887	1.128	1.40	62.2	2.650	0.975	
911	1.097	1.30	65.0	2.855	1.050	1.256
938	1.066	1.00	73.0	3.710	1.310	
958	1.042	0.80	78.5	4.640	1.533	
973	1.028	0.62	81.1	5.990	1.790	1.590
1015	0.985	0.43	88.4	8.630	2.160	1.730
1030	0.970	0.40	89.2	9.270	2.230	1.540
1078	0.928	0.14	96.2	26.500	3.280	1.795

TABLE XXVI-A. SUMMARY OF RUN DATA - 60 HR TEST WITH MIL-T-5161G FUEL

Catalyst: Code A Diluted(1) Space Velocity: 100,000 hr^{-1}

Cumulative Time(2) Hrs	Hot Spot Temp., °C	Hot Spot Position(3) Inches	Bed Press Drop, psi	Exit O_2 Conc. % Vol.	Exit CO_2 Conc. % Vol.
.5	760	11 1/4	2.78	.13	1.79
4.1	694	11 1/2	2.84	.12	2.00
7.7	730	11	3.10	.06	1.84
10.3	724	11	3.21	.05	1.84
14.3	740	11 3/4	3.25	.04	1.82
19.0	728	11 3/4	3.54	.05	1.86
21.6	766	9	3.83	.04	1.67
24.6	704	12 1/4	3.83	.24	1.39
29.6	785	9 1/2	3.87	.03	2.13
33.5	759	10	3.83	.03	1.55
36.9	798	9 7/8	3.83	.03	1.76
40.9	769	11 1/8	3.83	.03	1.64
45.2	758	10 3/4	3.83	.11	1.70
49.2	759	11	3.83	.02	1.83
53.4	766	11 1/4	3.85	.02	1.84
57.4	759	10 1/2	3.85	.02	1.79
62.3	767	10 1/4	3.83	.01	1.79

(1) 1 volume catalyst + 2 volumes inert ceramic.

(2) Test hours, see Table V.

(3) Measured from tip of thermowell; catalyst zone = 4.1" to 11.6".

TABLE XXVII-A. SUMMARY OF DATA FOR ACTIVITY TEST PRIOR TO 60-HOUR RUN

Catalyst: Code A* Fuel: Propane

SV: 32,000 hr^{-1} Mixture: Stoichiometric

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	O_2 Exit Conc. % Vol.	% Oxygen Conversion	$\left(\frac{O_2\ in}{O_2\ out}\right)$	$\ln\left(\frac{O_2\ in}{O_2\ out}\right)$	CO_2 Exit Conc. % Vol.
940	1.064	.49	87.5	7.65	2.04	-
914	1.094	1.34	64.3	2.80	1.03	1.20
983	1.017	.39	89.6	9.62	2.26	1.32
1061	.942	.09	97.6	40.3	3.70	1.64
1073	.932	.20	94.5	18.25	2.90	1.32
1063	.941	.05	98.6	76.2	4.33	-
1053	.950	.12	96.8	31.22	3.44	1.76

TABLE XXVIII-A. SUMMARY OF DATA FOR ACTIVITY TEST PRIOR TO 60-HOUR RUN

Catalyst: Code A* Fuel: Propane

SV: 50,000 hr^{-1} Mixture: Stoichiometric

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	O_2 Exit Conc. % Vol.	% Oxygen Conversion	$\left(\frac{O_2\ in}{O_2\ out}\right)$	$\ln\left(\frac{O_2\ in}{O_2\ out}\right)$	CO_2 Exit Conc. % Vol.
645	1.550	3.39	9.6	1.11	.104	.03
739	1.353	2.87	23.4	1.31	.270	-
814	1.228	2.36	37.0	1.59	.399	.52
863	1.159	1.80	52.0	2.08	.732	-
992	1.008	1.29	65.6	2.90	1.065	1.18
1071	0.935	.11	97.1	34.9	3.554	1.84

*in admixture with inert ceramic; 1 volume catalyst + 2 volumes ceramic.

TABLE XXIX-A. SUMMARY OF DATA FOR ACTIVITY TEST FOLLOWING 60-HOUR RUN

Catalyst: Code A* | Fuel: Propane

SV: 32,000 hr^{-1} | Mixture: Stoichiometric

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	O_2 Exit Conc. % Vol.	% Oxygen Conversion	$\left(\frac{O_2\ in}{O_2\ out}\right)$	$\ln\left(\frac{O_2\ in}{O_2\ out}\right)$	CO_2 Exit Conc. % Vol.
701	1.428	2.93	20.6	1.26	.207	.26
805	1.243	1.94	47.5	1.91	.647	.63
896	1.116	1.28	65.4	2.89	1.061	1.02
976	1.025	.58	84.3	6.61	1.888	1.21
1001	.999	.24	93.5	15.4	2.733	--
1033	.967	.06	98.5	61.7	4.12	1.58
1068	.936	.02	99.4	185	5.22	1.26

TABLE XXX-A. SUMMARY OF DATA FOR ACTIVITY TEST FOLLOWING 60-HOUR RUN

Catalyst: Code A* | Fuel: Propane

SV: 50,000 hr^{-1} | Mixture: Stoichiometric

Hot Spot Temp. (T) °K	1/T x 10^3 $°K^{-1}$	O_2 Exit Conc. % Vol.	% Oxygen Conversion	$\left(\frac{O_2\ in}{O_2\ out}\right)$	$\ln\left(\frac{O_2\ in}{O_2\ out}\right)$	CO_2 Exit Conc. % Vol.
693	1.442	3.16	10	1.11	.1044	-
729	1.371	3.02	13	1.16	.1487	-
789	1.266	2.49	29	1.405	.3402	1.26
863	1.159	1.95	44	1.795	.586	-
935	1.07	1.45	59	2.41	.880	-
998	1.002	1.07	70	3.27	1.163	-
1061	.942	.34	90	10.3	2.338	-
1087	.921	.32	91	10.93	2.395	-

*in admixture with inert ceramic; 1 volume catalyst + 2 volumes ceramic.

TABLE XXXI-A. SUMMARY OF DATA FOR ACTIVITY TEST FOLLOWING REGENERATION AFTER 60-HOUR RUN

Catalyst: Code A* Fuel: Propane

SV: 50,000 hr^{-1} Mixture: Stoichiometric

Hot Spot Temp. (T) °K	$1/T \times 10^3$ $°K^{-1}$	O_2 Exit Conc. % Vol.	% Oxygen Conversion	$\frac{O_2\ in}{O_2\ out}$	$\ln\left(\frac{O_2\ in}{O_2\ out}\right)$	CO_2 Exit Conc. % Vol.
873	1.145	2.10	40	1.67	.513	.53
936	1.068	1.49	57	2.34	.851	.74
1023	.978	.92	74	3.79	1.334	.89
1046	.956	.81	77	4.36	1.473	1.25
1066	.938	.60	83	5.82	1.762	1.07
1069	.935	.49	86	7.12	1.965	1.16

*in admixture with inert ceramic; 1 volume catalyst + 2 volumes ceramic

TABLE XXXII-A. SUMMARY OF DATA FROM PERFORMANCE TEST USING CATALYST A(1) AND A STOICHIOMETRIC MIXTURE OF OXYGEN AND JP-4

SV: 100,000 hr^{-1}

Hot Spot Temp (T) °K	$1/T \times 10^3$ °K^{-1}	Exit O_2 Conc. % Vol.	Oxygen Conv. %	Oxygen In / Oxygen Out	ln(Oxy In / Oxy Out)	% Theoret. CO_2, Basis O_2 Conv.
676	1.479	3.16	12.5	1.19	0.174	>100
813	1.230	2.29	36	1.64	0.496	>100
821	1.208	0.39	89	9.64	2.266	46
867	1.153	1.88	99.5	188	5.236	57
901	1.110	1.54	57	2.44	0.897	99
989	1.011	0.62	83	6.06	1.802	94
1034	0.967	0.21	94	17.90	2.885	79
1042	0.960	0.01	99.9	376	5.930	74

(1) Diluted, 1 volume of Catalyst A + 2 volumes of inert ceramic.

TABLE XXXIII-A. PROPERTIES OF JET FUELS

Fuel	Typical JP-7	Mil-T-5161 G (JP-4) 3-69-COV*	4-69-COV*
Gravity, °API	45.6	55.3	55.1
Water separometer index, modified	98	100	100
Viscosity at -30°F, CS	11.73	1.82	1.78
Color	+30	-	-
Freezing point, °F	-66	-68.8	-68.8
Existent gum, mg/100 ml	0.0	0.6	5.4
Potential gum, mg/100 ml	-	1.4	6.4
Aniline point, °F	157.0	-	-
Aniline gravity constant or B.T.U.	7159	-	-
Lovibond number	78	-	-
Aromatics, %	2.3	24.6	24.9
Olefins, %	3.4	1.6	1.5
Flash point, °F	153	-	-
Thermal stability, tube deposit code	#1	1**	1**
Thermal stability, pressure drop, inch Hg	0.2	0	0.0
Distillation, °F: IBP	392	159	168
10% evaporated	413	188	196
20% evaporated	416	205	211
50% evaporated	423	241	243
90% evaporated	442	353	354
EP	472	442	420
Recovery, %	98.0	97.5	98.0
Residue, %	1.0	1.0	1.0
Loss, %	1.0	1.5	1.0
Total sulfur, weight %	-	0.166	0.155
Mercaptan sulfur, weight %	-	0.0006	0.0006

*Batch identification code of Ashland Oil and Refining Company, Ashland, Kentucky.

**Preheater rating.

(Table Continued)

TABLE XXXIII-A. PROPERTIES OF JET FUELS (continued)

Fuel		Typical JP-7	Mil-T-5161 G (JP-4) 3-69-COV*	4-69-COV*
Reid vapor pressure at 100°F, psi		-	2.9	2.78
Net heat of combustion, BTU/lb		-	18,551	18,717
Smoke point, mm		-	23	21
Copper strip corrosion		-	1a	1a
Water reaction		-	1	1
Anti-icing additive:	Top, volume %	-	0.149	0.123
	Middle, "	-	0.145	0.120
	Bottom, "	-	0.135	0.122
	Composite, "	-	0.143	0.122
Metal deactivator, lb/1000 bbl		-	2	2
Antioxidant, lb/1000 bbl		-	8	8

*Batch identification code of Ashland Oil and Refining Company, Ashland, Kentucky.

APPENDIX B

KINETIC INTERPRETATION OF CATALYTIC COMBUSTION DATA

APPENDIX B

KINETIC INTERPRETATION OF CATALYTIC COMBUSTION DATA

The rates of processes occurring in plug flow reactors frequently reflect the law of mass action; that is, they show a dependency on the concentration of one or more of the reactants. If the overall rate is directly proportional to the concentration of a reactant, the reaction process is described as first-order and, assuming constant volume, the following relationship obtains:(1)

$$\frac{KV}{Q} = \ln\left(\frac{1}{1-X_A}\right) \quad \text{where}$$

K = reaction rate constant

$= Ae^{-B/T}$

V = volume of the reaction zone

Q = volumetric flow rate of reactants

$X_A = 1 - \frac{C_A}{C_{AO}}$ = conversion

C_A = outlet concentration of species A

C_{AO} = entering concentration of species A

T = absolute temperature in reaction zone

$B = E \div R$

E = energy of activation for reaction

R = gas constant

A = frequency factor, a system constant

By substitution of $Ae^{-B/T}$ for K and $1-C_A/C_{AO}$ for X_A and taking the logarithm we have

$$\ln\frac{VA}{Q} - \frac{B}{T} = \ln\left[\ln\left(\frac{C_{AO}}{C_A}\right)\right] = \ln\ln\left(\frac{O_2 \text{ in}}{O_2 \text{ out}}\right)$$

Hence, a plot of $\ln\ln\left(\frac{O_2 \text{ in}}{O_2 \text{ out}}\right)$ vs $\frac{1}{T}$ yields a straight line with a slope of $-B$.

If, on the other hand, the overall rate is directly proportional to the concentration of each of two reactants, or to the second power of the concentration of one reactant, the reaction is said to be second order. The following expression obtains:(1)

$$\frac{C_{AO}KV}{Q} = \frac{X_A}{1-X_A} \quad \text{from which by substitution for } X_A$$

$$\frac{C_{AO}KV}{Q} = \frac{1 - \frac{C_A}{C_{AO}}}{\frac{C_A}{C_{AO}}}$$ and by substitution of $A_e^{-B/T}$ for K

$$\frac{C_{AO}VA}{Q} e^{-B/T} = \frac{C_{AO}}{C_A} - 1$$

and by taking the logarithm

$$\ln\left(\frac{C_{AO}VA}{Q}\right) - \frac{B}{T} = \ln\frac{C_{AO}}{C_A}$$
$$= \ln\left(\frac{O_2\ in}{O_2\ out}\right)$$

Hence, a plot of $\ln\left(\frac{O_2\ in}{O_2\ out}\right)$ vs $\frac{1}{T}$ yields a straight line with slope of -B. Especially when processes occur through a succession of steps, it is normal to find a fractional-order relationship governing the overall process, i.e. the reaction order is not exactly 0.0 or 1.0 or 2.0, but somewhere in between.

Inspection of the plots indicated above shows if the overall reaction adheres to first or second order kinetics. In cases where stoichiometric mixtures of reactants are used, the initial and final concentration of either reactant can be used. In the present experimental program, the exit gas volume is only slightly greater than the entrance volume, and the error introduced by this deviation from constant volume conditions is negligible.

Certain reactions involving two reactants have been found to proceed at rates governed entirely by the concentration of one of the reactants, in which case the rate is said to be zero-order with respect to the other reactant. Such is known to be the case with the catalytic vapor phase oxidation of propane, where the rate is essentially zero-order with respect to oxygen concentration.

REFERENCES

1. Levenspiel, Chemical Reaction Engineering, pp 48-50, John Wiley and Sons, New York, 1962

APPENDIX C

METHOD OF CALCULATING REACTION RATE CONSTANTS

APPENDIX C

METHOD OF CALCULATING REACTION RATE CONSTANTS

The reaction rate constant is obtained from the rate equation (Appendix B):

$$\frac{KV}{Q} = \ln\left(\frac{1}{1-X_A}\right)$$

and the Arrhenius equation:

$$K = Ae^{-B/T}$$

where all the symbols are defined as in Appendix B. The value of constant K varies according to the terms and conditions used to define the volumetric flow of reactants. For example, these flows can be given at standard conditions or at the conditions existing in the reactor. In this experimental program, the flow of reactants measured at 14.7 psia and 22°C (295°K) is the standard basis for space velocity (SV). By definition, SV = $\frac{Q}{V}$. Therefore, under these standardized conditions:

$$K = SV \ln\left(\frac{C_{AO}}{C_A}\right)$$

For systems involving stoichiometric mixtures of two reactants, concentration measurements on either reactant are sufficient for determination of the rate constant.

It can be informative to calculate reaction rate constants using volumetric flow rates (Q) representing actual conditions in the reaction zone. In a series of runs made at different pressures, for example, the effect of changes in reaction pressure on the rate constant can be evaluated. Thus, if P_R and T_R represent the actual reaction pressure and temperature, and P_S and T_S represent the standard conditions on which space velocity is based:

$$K_R = SV\left(\frac{T_R}{T_S}\right)\left(\frac{P_S}{P_R}\right)\ln\left(\frac{O_2\ \text{in}}{O_2\ \text{out}}\right)$$

Example:

Run No.	JT - 6A
SV	150,000 hr^{-1}
T_R	600°C = 873°K
P_R	29.12 psia
O_2 in	2.76%
O_2 out	1.18%

$$K_R = \frac{150{,}000}{hr}\left|\frac{873}{295}\right|\frac{14.7}{29.12}\right| \times \ln\left(\frac{2.76}{1.18}\right)$$

$$K_R = 190{,}800\ hr^{-1} \text{ or } 53\ sec^{-1}$$

APPENDIX D

WATER REMOVAL STUDIES

SUMMARIES OF TEST DATA

TABLE I-D. SUMMARY OF TEST DATA FOR $CaSO_4$

Temperature = 40°C

Volume of Gas (liters)	Space Velocity[a] (hr^{-1})	Water Pickup, 100 x Wt Gain ÷ Wt Agent U-tube #1	U-tube #2	U-Tube #3	Efficiency (ppm)
43.6	5700	3.29	0.03	zero	20
150.4	5700	8.52	1.61	0.16	20
222.6	5700	10.42	4.32	0.23	20
265.2	5700	11.29	5.81	0.73	20
317.2	5700	11.94	6.87	2.01	70
365.6	5700	12.18	7.81	3.42	400
388.8	5700	12.29	8.21	4.23	1000

(a) Volume of gas through each tube ÷ bulk volume of agent in tube.

TABLE II-D. SUMMARY OF TEST DATA FOR $CaSO_4$

Volume of Gas (liters)	Space Velocity[a] (hr^{-1})	Water Pickup, 100 x Wt Gain ÷ Wt Agent U-Tube # 1	U-Tube #2	U-Tube #3	Efficiency (ppm)
		Temperature = 100°C			
49.1	5700	2.83	0.56	0.35	9,500 ave
130.6	5700	2.05	0.25	0.38	31,000 ave
		Temperature = 95°C			
2.3	25	0.91	0.06	0.08	860
24.0	1200	2.16	1.20	0.86	3,960 ave

(a) Volume of gas through each tube ÷ bulk volume of agent in tube.

TABLE III-D. SUMMARY OF TEST DATA FOR $CaCl_2$

Temperature = 100°C

Volume of Gas (liters)	Space Velocity[(a)] (hr^{-1})	Water Pickup, 100 x Wt Gain ÷ Wt Agent U-Tube #1	U-Tube #2	U-Tube #3	Efficiency (ppm)
99.2	5700	3.26	0.31	0.06	16,700
255	5700	6.73	1.89	0.30	18,000
339	5700	7.76	3.80	0.59	18,000
483	5700	9.78	5.48	2.73	
615	5700	11.33	5.96	4.57	
865	5700	14.18	7.99	5.16	20,400

(a) Volume of gas through each tube ÷ bulk volume of agent in tube.

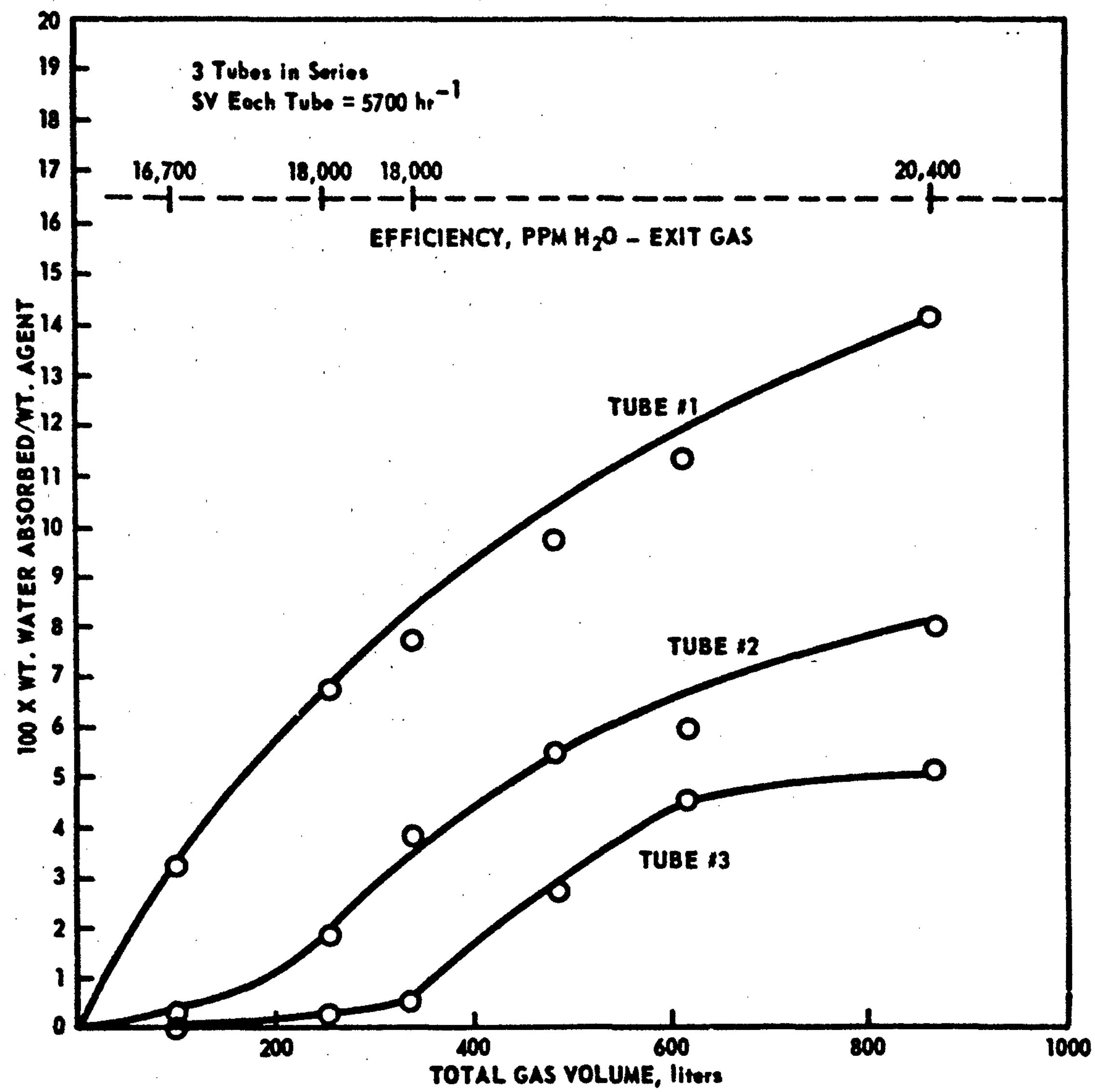


FIGURE 1-D. DRYING PERFORMANCE OF Ca_2Cl_2 AT 100°C

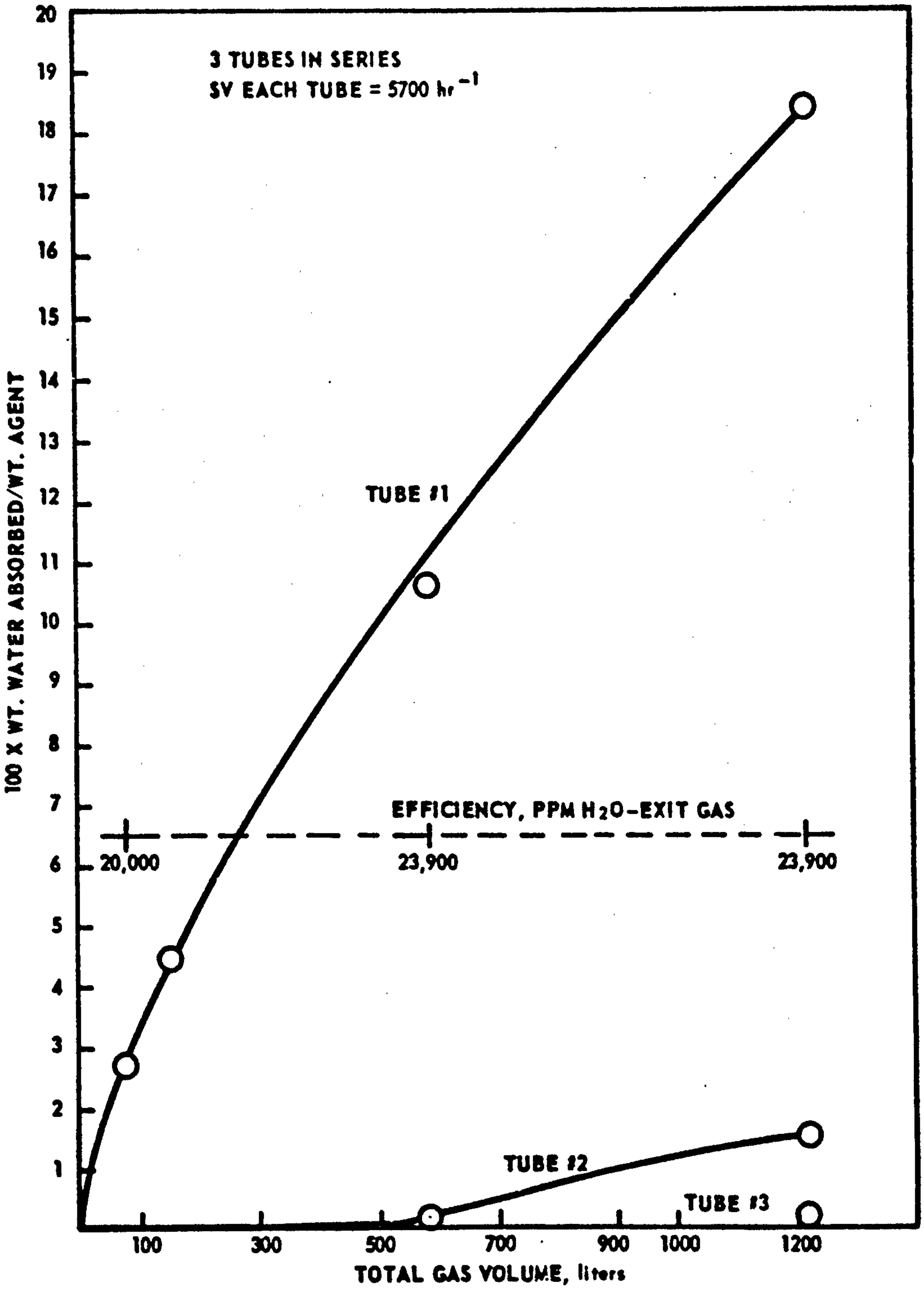


FIGURE 2-D. DRYING PERFORMANCE OF B_2O_3 AT 100°C

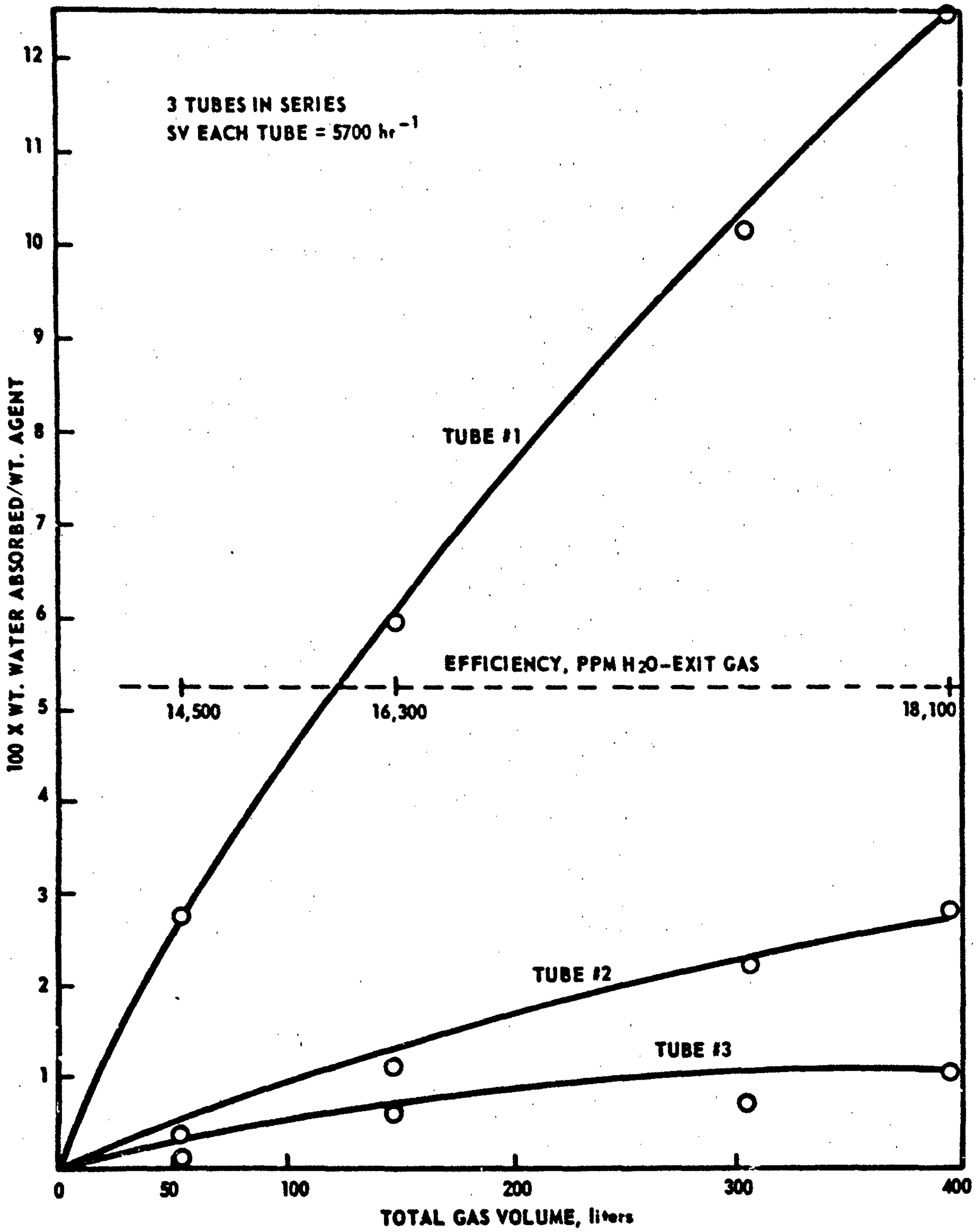


FIGURE 3-D. DRYING PERFORMANCE OF LiCl AT 80°C

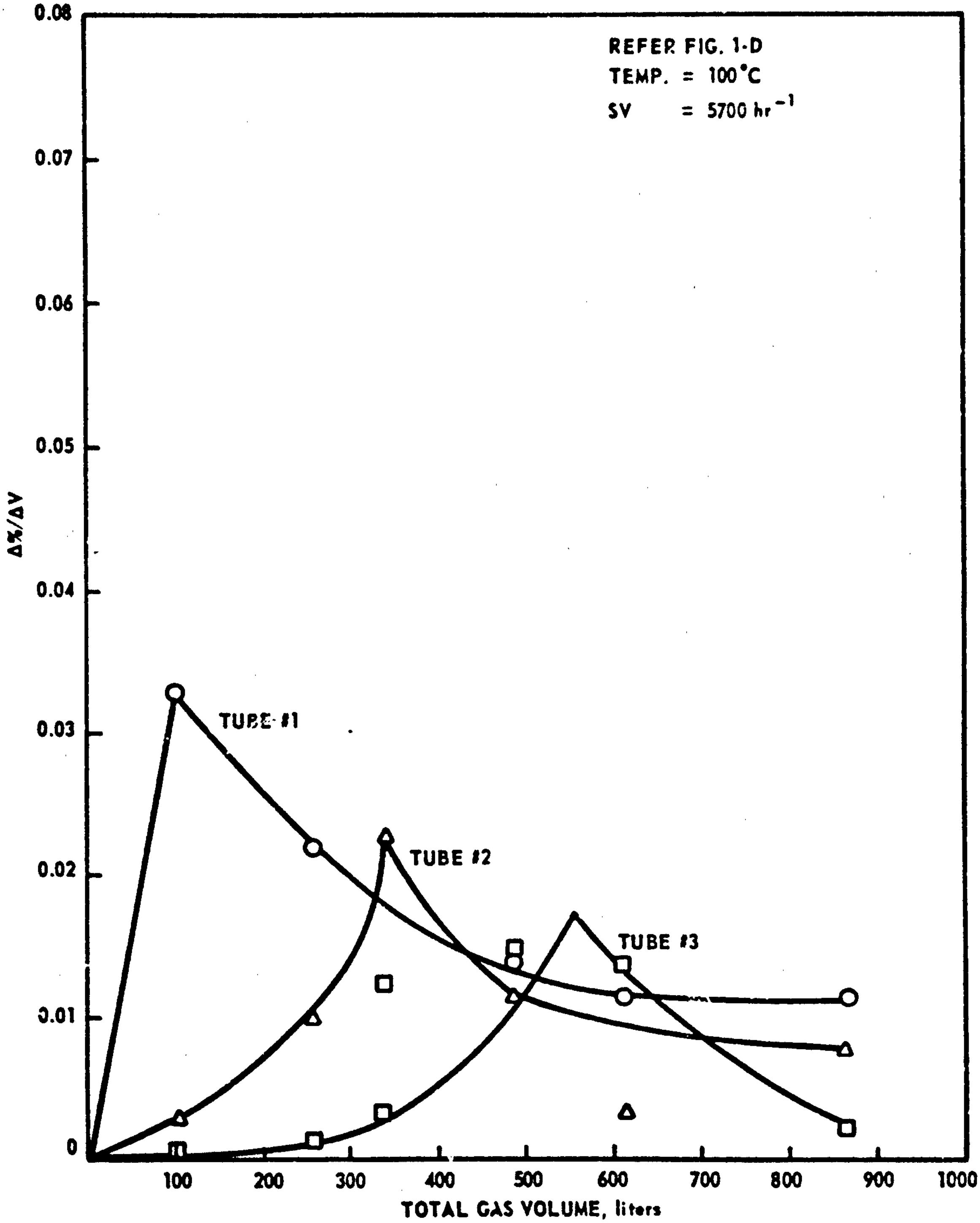


FIGURE 4–D. WATER ABSORPTION RATE FOR $CaCl_2$

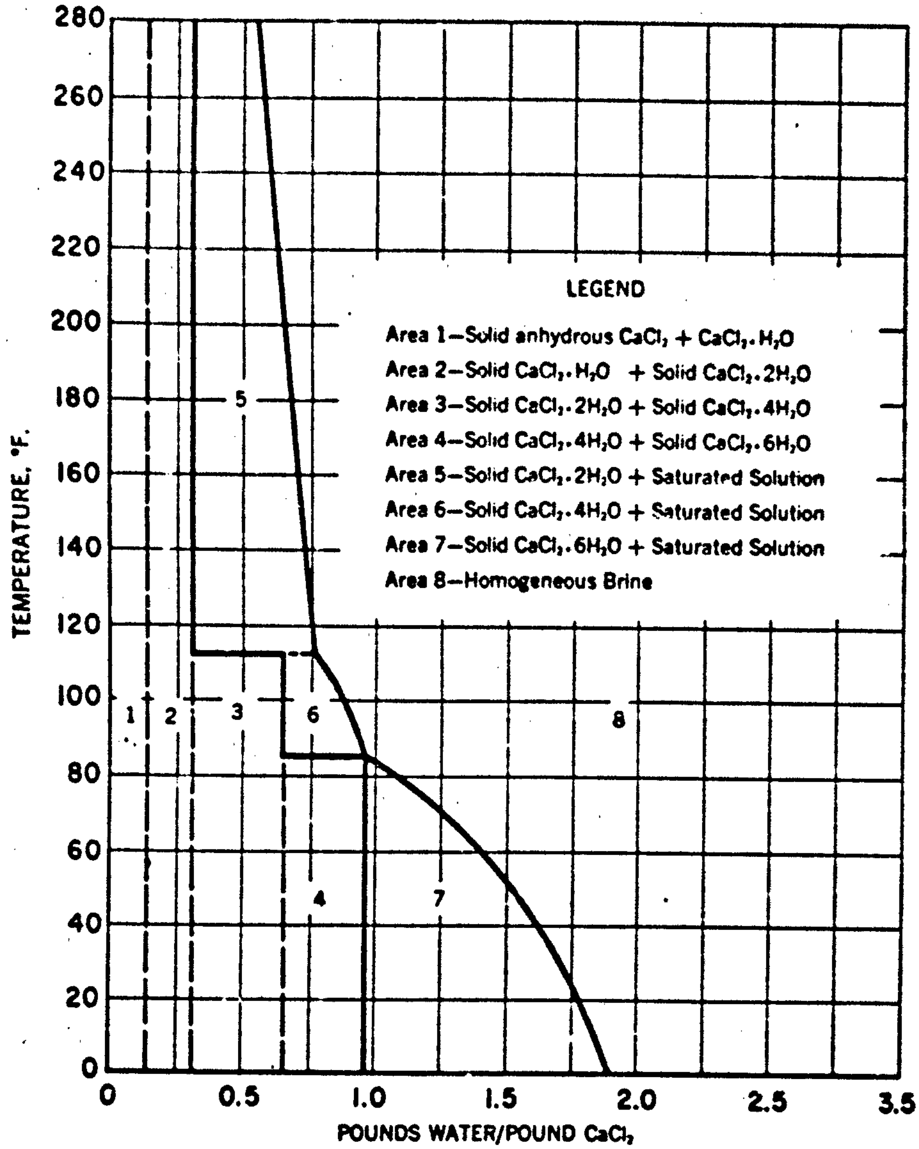


FIGURE 5-D. CALCIUM CHLORIDE PHASE DIAGRAM[38]

[38] Gas Conditioning Fact Book, p. 279, The Dow Chemical Company, Midland, Michigan, 1962.

APPENDIX E

CALCULATION OF HEAT AND MATERIAL BALANCE

BST FLIGHT PLAN NO. 1

APPENDIX E

CALCULATION OF HEAT AND MATERIAL BALANCE

SST Flight Plan No. 1

Average Conditions

The following calculations are based on the data and assumptions stated in Section V 6 i. Reference is made to the flow diagram appearing in Figure 36.

1. Molar volume at assumed conditions

$$P_1 = 5.03 \text{ psia}$$

$$T_1 = 200°F = 660°R$$

$$V_1 = \frac{14.7}{492} \times 359 \times \frac{660}{5.03} = 1{,}407 \text{ ft}^3/\text{lb-mole}$$

2. Amount of air required to supply 1,000 ft^3/min dry ballast gas at above conditions

$$1{,}000 \times 1.0811 = 1{,}081 \quad \text{ft}^3/\text{min air}$$

$$1{,}081 \div 1{,}407 = 0.768 \quad \text{lb-moles/min air}$$

$$0.768 \times 28.9 = 22.21 \quad \text{lb/min air}$$

3. Amount of water formed

1,081 ft^3/min	0.21	mole	O_2	18 lb H_2O
	O_2	1.407 ft^3	1.4	mole

$= 2.08$ lb/min water

4. Amount of heat removed by the air during its preheating in H.E. No. 1 and in the reactor jacket

at 600°F, $h_{600} = 142$ BTU/#

at 1,112°F, $h_{1112} = 276$ BTU/#

$$Q_{AIR} = (h_{1112} - h_{600}) \times W_{AIR}$$

$$= (276 - 142) \times 22.21 = 2{,}980 \text{ BTU/min}$$

5. Maximum amount of heat that can be removed by the fuel (allowing all the fuel to reach 600°F)

$$\text{at } 300°F,\ h_{300} = 140 \text{ BTU/lb fuel}$$

$$\text{at } 600°F,\ h_{600} = 351 \quad " \quad "$$

a) engine at cruise operating conditions

$$Q_{F\text{-cruise}} = (351 - 140) \times 712 = 150{,}000 \text{ BTU/min}$$

b) engine at idle operating conditions

$$Q_{F\text{-idle}} = (351 - 140) \times 140 = 29{,}540 \text{ BTU/min}$$

6. Amount of fuel to use for combustion

Reaction: $1.65\ C_9H_{20} + 21\ O_2 + 79\ N_2 \longrightarrow 79\ N_2 + 13.5\ CO_2 + 15\ H_2O + 0.15\ C_9H_{20}$

(or $1.65\ C_9H_{20}$ + 100 Air)

$$\frac{22.21}{28.9 \times 100} \times (1.65 \times 128) = 1.623 \text{ lb/min fuel}$$

$$= 0.244 \text{ gpm at } 60°F$$

$$= 0.278 \text{ gpm at } 300°F$$

$$= 0.331 \text{ gpm at } 600°F$$

7. Molar quantities

No. of moles of fuel used = 1.623/128 = 0.01268 moles/min

No. of moles of air used = 22.21 /28.9 = 0.768 "

No. of moles of O_2 used = 0.21 x 0.768 = 0.1614 "

No. of moles of N_2 used = 0.79 x 0.768 = 0.607 "

Total flow of air-fuel mixture = 0.781 moles/min

= 23.83 lb/min

8. Weights and volumes of components in the air-fuel mixture and the moist ballast gas (see reaction in 6.)

a. Incoming air-fuel mixture:

Comp.	No. moles	Mol. Wt.	No. x M.W.	Wt. %	Vol. %	Wt. actually used	
C_9H_{20}	1.65	128	211.2	6.82	1.62	1.62	lb/min
O_2	21	32	672	21.71	20.66	5.18	"
N_2	79	28	2212	71.47	77.72	17.03	"
	101.6		3095	100.00	100.00	23.83	"

b. Leaving moist ballast gas:

Comp.	No. moles	Mol. Wt.	No. x M.W.	Wt. %	Vol. %	Wt. actually used	
N_2	79	28	2212	71.47	73.39	17.03	lb/min
CO_2	13.5	44	594	19.19	12.54	4.57	"
H_2O	15	18	270	8.72	13.93	2.08	"
C_9H_{20}	0.15	128	19	0.62	0.14	0.15	"
	107.6		3095	100.00	100.00	23.83	"

c. Some relationships:

Weight ratios: $O_2 \div \text{Air} = 0.233$ lb O_2/lb Air

$O_2 \div N_2 = 0.304$ lb O_2/lb N_2

"Molar" volume increase ratios due to reaction, etc:

$$\frac{\text{Incoming mixture}}{\text{Incoming air}} = \frac{101.6}{100} = 1.016$$

$$\frac{\text{Moist ballast gas}}{\text{Incoming air}} = \frac{107.6}{100} = 1.076$$

$$\frac{\text{Moist ballast gas}}{\text{Incoming air-fuel mixture}} = \frac{107.6}{101.6} = 1.059$$

9. Preparation of the air-fuel mixture

a. Heat Content of the Mixture (assume fuel is liquid):

Sensible heat in liquid fuel at 600°F = 351 BTU/#

Sensible heat in air at 1,112°F = 276 BTU/#

Q_M = Heat content of mixture = $W_F \times h_{F,600} + W_{AIR} \times h_{AIR,\,1112}$

$= 1.623 \times 351 + 22.2 \times 276 = 6{,}700$ BTU/min

b. Required Heat Content of the Mixture with Vaporized Fuel

heat of vaporization of fuel at 600°F = $\lambda_{F,600}$ = 100 BTU/#

$Q_{RM} = Q_M + W_F \times \lambda_{F,600} = 6{,}700 + 1.623 \times 100 = 6{,}860$ BTU/min

c. Assume the desired mixture temperature = 932°F

Sensible heat in liquid fuel at 600°F	= 351 BTU/#
Heat of vaporization	= 100 "
ΔSensible heat in fuel vapor, 600 to 932°F.	= 232 "
Heat content of fuel vapor at 932°F	= 683 BTU/#

Sensible heat in air at 932°F = 226 BTU/#

Heat content of the mixture at 932°F =

$683 \times 1.623 + 226 \times 22.2$

Q_{932} = 6,130 BTU/min

Change in heat content = $Q_{932} - Q_M$

= 6,130 - 6,700

Δh = - 570 BTU/min ∴ excess heat

d. Assume the desired mixture temperature = 1,112°F (600°C)

By similar calculation:

Heat to add = $Q_{1112} - Q_M$

= 7,490 - 6,700 = 790 BTU/min

The power requirements for making up this deficit by resistance heaters:

$$\text{Electric power} = (\text{BTU/min}) \times \frac{60}{3412} \quad \text{Kw}$$

$$= \frac{790 \times 60}{3412} = 14 \text{ Kw}$$

e. Temperature of the mixture without additional heat, (that is, with the total heat content = Q_M).

By trial and error we find: t = 1,012 °F = 544.4°C which is sufficient to initiate the reaction.

10. Heat evolved in catalytic combustion

ΔH_r for JP-7 = 18,750 BTU/#

for 100% conversion: $\Delta H_{rT} = 1.623 \times \frac{10}{11} \times 18{,}750 = 27{,}670$ BTU/min.

11. Heat content of gas leaving reactor bed

temperature = 725°C = 1337°F

h water vapor = 1,711 BTU/# $\qquad W_{H20}$ vapor = 2.08 lb/min

h gas = 335 " $\qquad W_{DRY\ BG}$ = 21.75 "

$Q_{1,337} = 2.075 \times 1{,}711 + 21.75 \times 335$

$= 10{,}850$ BTU/min

12. Amount of heat to be removed in the reactor bed

Q_R = Enthalpy of air-fuel mixture entering reactor (Q_M)

+ Heat evolved in catalytic combustion (ΔH_{rT})

- Enthalpy of moist gas leaving reactor bed at 1,337 ($Q_{1,337}$)

$\therefore Q_R = 6{,}700 + 27{,}670 - 10{,}850$

$= 23{,}520$ BTU/min

13. Heat exchange capacity of available cooling water

As per calculations for cooling in H.E. No. 3 and the gas drier (16, 17 and 18) the amt. of available cooling water and its conditions are:

W_{H_2O} = 27 lb/min

t = 200.5 °F

h = 168.6 BTU/#

If it is assumed that this cooling water leaves as saturated steam at 212°F with h_{212} = 1,150.4 BTU/# the amount of heat that can be removed is:

$Q_{H_2O} = W_{H_2O} (h_{212} - h_{200.5})$

$= 27 \times (1{,}150.4 - 168.6)$

$= 26{,}510$ BTU/min

In these conditions the moist gas would leave with

$$Q_{W.B.G.} = Q_M + \Delta H_{rT} - Q_{H_2O}$$

$$= 6{,}700 + 27{,}670 - 26{,}510$$

$$= 7{,}860 \text{ BTU/min}$$

This heat content corresponds to a temperature of 905°F, where

h water vapor = 1,484.8 BTU/# ∴ 2.08 x 1,484.8 = 3,090 BTU/min

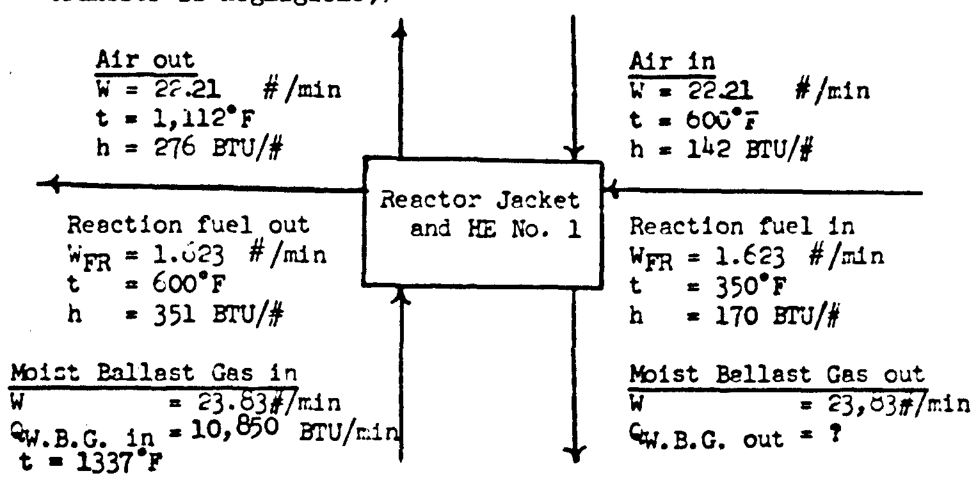
h gas = 219.4 " ∴ 21.75 x 219.4 = 4,770 "

7,860 BTU/min

<u>Note</u>: Thus, there is an excess of cooling water capacity for the combustor. In order to leave sufficient heat in the exit gases to preheat the combustion air to 1112°F (see below), a motorized valve is used to dump the excess water.

14. <u>Preheating of air</u>

Achieved in reactor jacket and HE No. 1 (later calculations may show that jacket surface is sufficient, or that the jacket heat transfer is negligible).



Q_{AIR} = heat removed by air = $(h_{AIR\ OUT} - h_{AIR\ IN}) \times W_{AIR}$

$$= (276 - 142) \times 22.21$$

$$= 2{,}980 \text{ BTU/min}$$

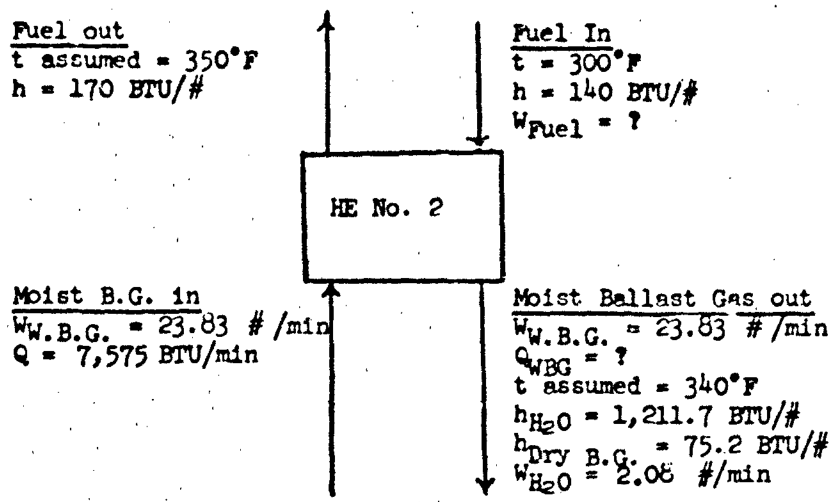
Q_{FR} = heat removed by inerting fuel = $W_{FR}\,(h_{FR\ OUT} - h_{FR\ IN})$

$$= 1.623 \times (351 - 170)$$

$$= 294 \text{ BTU/min}$$

$$Q_{W.B.G.\ OUT} = Q_{WBG,\ IN} - (Q_{AIR} + Q_{FR})$$
$$= 10,845 - (2,980 + 294)$$
$$= 7,575 \text{ BTU/min}$$

15. Cooling with fuel



The maximum amount of heat that can be removed by the fuel (if allowed to reach 600°F) is 351-140 = 211 BTU/lb for the total flow of fuel to the engines. Only part of this flow will be needed. We can assume an exit temperature for the fuel and calculate the required quantity.

$$t_{FUEL\ OUT} = 350\ °F$$

$$h_{FUEL\ OUT} = 170 \text{ BTU/lb.}$$

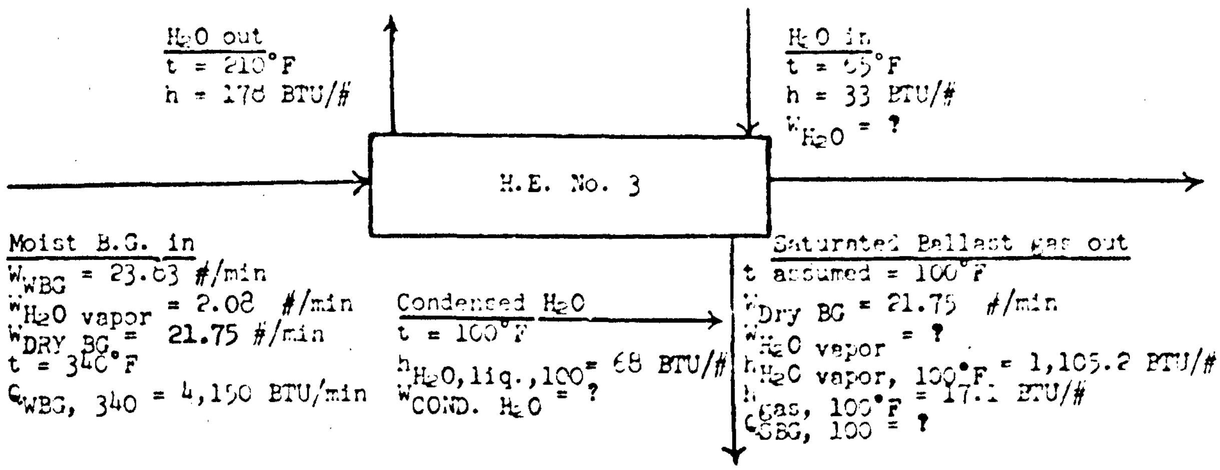
We have also to assume the exit temperature of the moist ballast gas. Since the fuel enters at 300°F, the ballast gas may reasonably leave at 340°F. Thus, the heat to be removed from the ballast gas is:

$$\Delta h_{HE2} = Q_{W.B.G.\ IN} - \left[(W_{H_2O} \times h_{H_2O,340} + (W_{WBG} - W_{H_2O}) \times h_{Dry\ B.G.,340})\right]$$
$$= 7,575 - (2.08 \times 1,211.7 + 21.75 \times 75.2)$$
$$= 7,575 - 4,150 = 3,425 \text{ BTU/min}$$

And the amount of fuel required is:

$$W_{Fuel} = \frac{\Delta h_{HE2}}{\Delta h_{FUEL}}$$
$$= \frac{3,425}{170 - 140}$$
$$= 114 \ \#/\text{min}$$

1[illegible]. Cooling with water and partial condensation



For saturation at 100°F the water content of the exit gas is $W_S = 0.0432$ # H_2O vapor/# dry gas. Thus, the amount of water vapor carried by the saturated ballast gas leaving H.E. #3 is: $0.0432 \times 21.75 = 0.94$ lb/min $= W_{H_2O\ Vap,\ 100}$ and the amount of condensed water is

$$W_{COND.\ H_2O} = 2.08 - 0.94 = 1.135 \text{ lb/min}$$

$$Q_{Dry\ BG,\ 100} = W_{Dry\ BG} \times h_{gas,\ 100} = 21.75 \times 17.1$$
$$= 372 \text{ BTU/min}$$

$$Q_{H_2O\ vapor,\ 100} = W_{H_2O\ vap.\ 100} \times h_{H_2O\ vap,\ 100} = 0.94 \times 1{,}105.2$$
$$= 1{,}039 \text{ BTU/min}$$

$$Q_{COND.\ H_2O,\ 100} = W_{COND.\ H_2O,\ 100} \times h_{H_2O\ liq.,\ 100} = 1.135 \times 68$$
$$= 77 \text{ BTU/min}$$

$$Q_{SBG} = 372 + 1{,}039 = 1{,}411 \text{ BTU/min}$$

Heat removed in HE No. 3 $= Q_{WBG,340} - \sum Q$

$$Q_{R,\ HE3} = 4{,}150 - (372 + 1{,}039 + 77)$$
$$= 2{,}662 \text{ BTU/min}$$

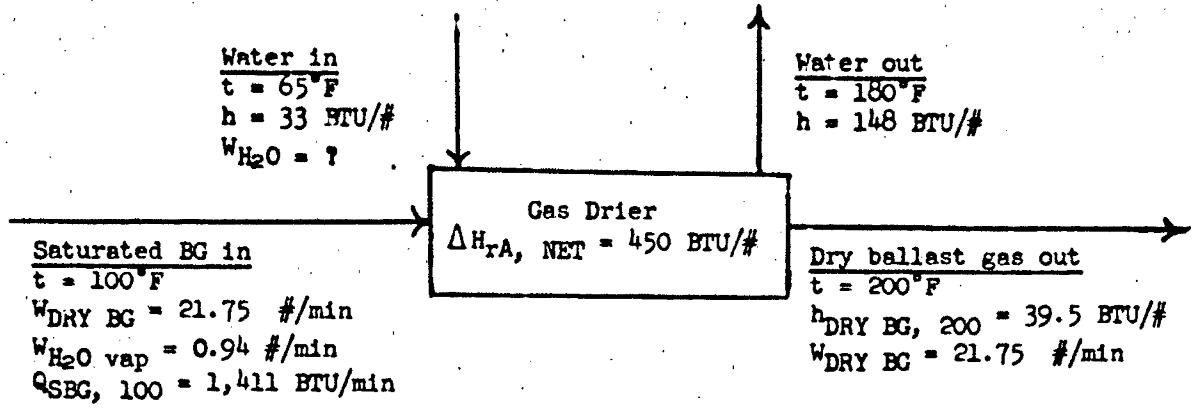
$$\text{Amount of water} = \frac{Q_{R,\ HE3}}{h_{H_2O,210} - h_{H_2O,\ 65}}$$

$$W_{H_2O} = \frac{2{,}662}{178 - 33}$$
$$= 18.5 \text{ \#/min}$$

17. Gas Drier

Assumed all water removed. As mentioned before, the heat of wate adsorption ΔH_{rA} = 1,500 BTU/# includes the heat of condensation h_{fg} = 1,050 BTU/#. Since the heat content of saturated ballast gas Q_{SBG} includes the latent heat of vaporization, the net heat of absorption has to be used in the calculations, thus;

$$\Delta H_{rA,\,NET} = 1{,}500 - 1{,}050 = 450 \text{ BTU/\#}$$



$$Q_{IN} = Q_{SBG,\,100} + W_{H_2O\ vap} \times \Delta H_{rA,\,NET}$$
$$= 1{,}411 + 0.94 \times 450$$
$$= 1{,}834 \text{ BTU/min}$$

$$Q_{OUT} = Q_{DRY\ BG,\,200} = W_{DRY\ BG} \times h_{DRY\ BG,\,200}$$
$$= 21.75 \times 39.5 = 859 \text{ BTU/min}$$

$$\Delta Q_{GAS\ DRYER} = Q_{IN} - Q_{OUT} = 1834 - 859 = 975 \text{ BTU/min.}$$

$$\text{Amount of water} = \frac{\Delta Q_{GAS\ DRYER}}{h_{H_2O,\,180} - h_{H_2O,\,65}}$$
$$= \frac{975}{148 - 33}$$
$$= 8.5 \text{ lb/min}$$

18. Total cooling water required

Combine the amount used in HE #3 and the gas drier, plus any additional that may be necessary for the reactor cooling.

$$W_{HE\ 3} + W_{GAS\ DRYER} = 18.5 + 8.5 = 27 \text{ lb/min}$$

$$\text{Average heat content} = \frac{W_{HE\ 3} \times h_{210} + W_{GAS\ DRYER} \times h_{180}}{W_{HE\ 3} + W_{GAS\ DRYER}}$$

$$= \frac{18.5 \times 178 + 8.5 \times 148}{18.5 + 8.5} = \frac{4{,}551}{27}$$

$= 168.6$ BTU/#

This corresponds to average water temperature of 200.5°F.

The above amount of water at the resulting temperature is more than sufficient to perform the required heat removal in the catalyst bed. It should be noted that the water condensed in HE 3 is available also.

APPENDIX F

DESIGN CALCULATIONS, EQUIPMENT FOR SST FLIGHT PLAN NO. 1

APPENDIX F

DESIGN CALCULATIONS, EQUIPMENT FOR SST FLIGHT PLAN NO. 1

I. Pipes for Preheated Air and Fuel

These are sized to handle flows during the normal descent period.

A. Air Pipe

W_{AIR} = 0.8615 lb-mole/min. = 24.9 lb/min. = 1,494 lb/hr.

t = 1,112°F

p = 47.3 psig (∴ P = 62 psia)

V_{AIR} = (272 ft^3/lb-mole) x 0.8615 lb-mole/min. = 234.3 cfm

ρ_{AIR} = 0.1063 lb/ft^3

μ_{AIR} = 0.039 centipoises = 0.0000262 lb_m/ft. sec.

Using stainless steel tubing, 2 3/8" OD and 0.065" wall thickness we get (from charts and nomographs in "Flow of Fluids" published by the Crane Company):

υ = average velocity = 9,500 ft/min = 160 ft/sec.

R_e = Reynolds number = 105,000

f = Moody's friction factor = 0.0213

ΔP_{100} = pressure drop per 100 ft. pipe length = 2.7 psi

w_p = weight of pipe = 1.61 lb/ft.

Consequently, as per Figure F-1 and the distance from HE 1, the 10 ft. of tubing weigh 16.1 lb. and the equivalent length (2 tees) is 40 ft. giving a pressure loss of (40 x 2.7) ÷ 100 = 1.1 psi.

The tubing to supply the air to the vaporization chamber will be 13/16" OD, with 0.03" wall thickness and weighing 0.4 lb/ft, while the pressure drop is ΔP_{100} = 1.0 psi. Its length is about 4 ft, weight is 1.6 lb. and the equivalent length (reducing tee and bend) is 10 ft, giving a pressure loss of 0.1 psi.

Weight air lines = 17.7 lbs.

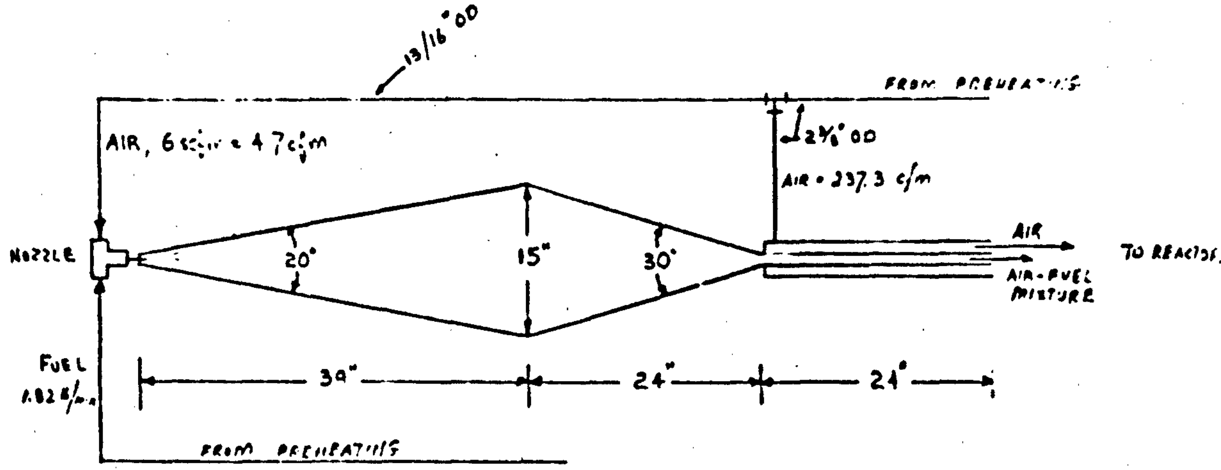


Figure F-1. Air and Fuel Feed System

B. Fuel Pipe

For 0.371 gpm, using 1/8 inch schedule 5 pipe: ΔP_{100} = 0.4 psi and W_p = 0.14 lb/ft. We need 8 ft. of tubing, its weight is 1.1 lb, the equivalent length is 10 ft, and the pressure loss 0.04 psi.

Thus, the weight of the supply piping is 17.7 + 1.1 + 0.5* = 19.3 lbs.

Weight all supply lines = 19.3 lbs.

II. Vaporization and Mixing Chamber

A. Size and Weight

The chamber must handle the design quantity of fuel (0.371 gpm = 1.82 lb/min) and all the air the nozzle requires. The remainder of air is routed to the jacket of the pipe leading from vaporization chamber to reactor. The chamber must be able to withstand the highest pressure at which air may be delivered (14.7 x 15 = 220 psia ≅ 205 psig).

The shape and size are governed by the angle and length of the spray projection cone, and the desire to minimize the pressure drop.

---

*Added for fittings (extra weight over straight pipe).

For spray nozzle choice:

Fuel rate = 0.371 x 60 = 22.24 gph

Air rate = 0.8615 x 359 = 309 scfm

Available air pressure = 46.9 psig

The design will be based on commercially available nozzles; however, especially designed two-fluid pneumatic spray nozzles are possible. For a round spray pattern cone and above fuel rate, the spray cone angle is 20° and the length of the laminar type spray projection cone is 39".

The air capacity of the nozzle is 6 scfm; consequently, the bulk of the required air bypasses the chamber.

In order to minimize the pressure drop, the downstream end of the vaporization chamber is shaped as a convergent cone with a 30° angle (based on design of a venturi). It is estimated that the pressure loss across the entire vaporization set-up (that is, including the nozzle) will be 20% of the pressure in the supply line, namely 9.38 psi.

The length and width are determined trigonometrically.

The thickness of the chamber is based on the highest possible pressure (220 psia) and temperature (1100°F), using a tensile stress of 10,400 psi for Type 316 SS.

$$t = \frac{PD}{2S} = \frac{220 \times 15}{2 \times 10{,}400} = 0.16"$$

Therefore, use 3/16" thick sheet.

The weight of the chamber is:

Lateral area cones x t x $lb/inch^3$ = Weight

$$\tfrac{1}{2}\ \pi D \left(\frac{R}{\sin 10°} + \frac{R}{\sin 15°}\right) \times 3/16 \times 0.290$$

$$= \tfrac{1}{2}\ 3.1416 \times 15 \left(\frac{7.4}{0.1736} + \frac{7.4}{0.2588}\right) \times 3/16 \times 0.290$$

= 91 lb.

Weight of chamber = 91 lbs.

B. Outlet Pipe

Designed to carry the air and fuel vapor. In order to minimize the possibility of explosion, the mixture from the chamber (which is fuel-rich) travels in the inner tube while the rest of the air travels separately in the jacket space. To continue heating the mixture while on its way to the reactor, the inner tube is provided with external longitudinal fins, as in Figure F-2.

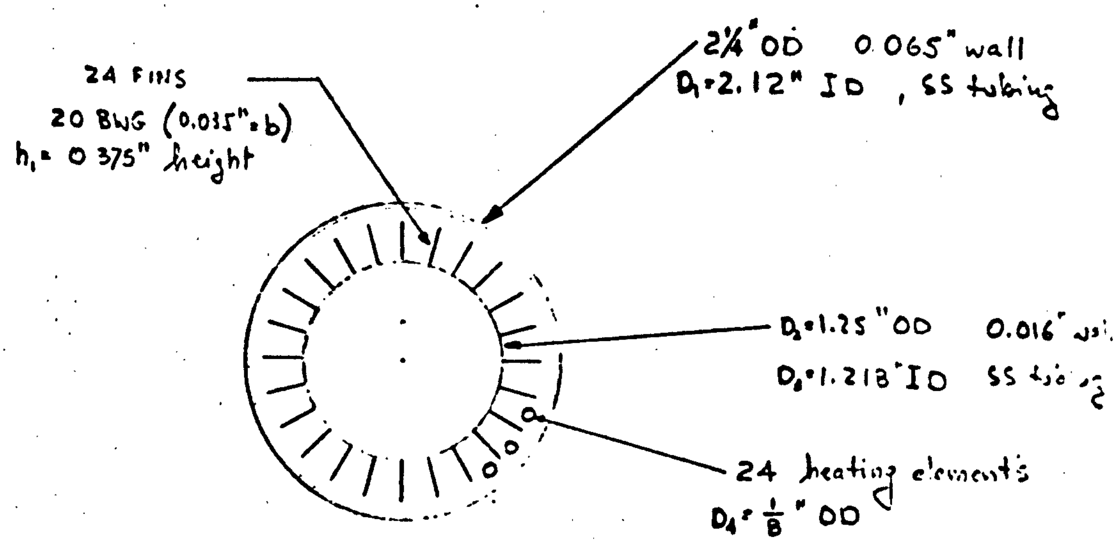


Figure F-2. Transverse Section, Outlet Pipe

For preheating the air and the system at the start of flight, resistance heating elements are placed inside the jacket space where the bulk of the flow takes place. No elements are placed in the inner tube because pressure drop here is more critical, due to the loss taken in the nozzle and vaporization chamber.

1. Pressure Drop in the Annular Space

P_w = Wetted perimeter

$$P_w = \pi D_1 + \pi D_2 + 24 \times 2\,h_1 + 24\,\pi D_4$$

$$= \pi(2.12 + 1.25 + 24 \times 0.125) + 48 \times 0.375$$

$$= 38.01'' = \underline{\underline{3.17\ ft}}$$

A_a = cross-section annular space

$$A_a = \frac{\pi D_1^2}{4} - \left(\frac{\pi D_2^2}{4} + 24\ h_1 b + 24\ \frac{\pi D_4^2}{4}\right)$$

$$= \frac{\pi \times 2.12^2}{4} - \left(\frac{\pi \times 1.25^2}{4} + 24 \times 0.375 \times 0.035 + 24\ \frac{\pi \times 0.125^2}{4}\right)$$

$$= 1.70\ \text{inch}^2 = \underline{\underline{0.0118\ \text{ft}^2}}$$

d_e = equivalent diameter

$$d_e = \frac{4 \times A_a}{P_w}$$

$$= \frac{4 \times 0.0118}{3.17}$$

$$= \underline{\underline{0.0149\ \text{ft}}}$$

G_a = mass velocity in annulus

$$G_a = \frac{W\ (\text{lb/hr})}{A_a\ (\text{ft}^2)}$$

$$= \frac{(24.9 - 0.48)\ 60}{0.0118}$$

$$= \underline{\underline{124{,}000\ \text{lb/(hr)}\ (\text{ft}^2)}}$$

R_e = Reynolds number

$$R_e = \frac{d_e\ (\text{ft}) \times G_a}{2.42 \times \mu (\text{centipoises})}$$

$$= \frac{0.0149 \times 124{,}000}{2.42 \times 0.039}$$

$$= \underline{\underline{19{,}600}}$$

f = friction factor

= 0.000235 (1)

Δp = pressure loss

L = length of pipe = 2.89 ft.

ϕ_a = viscosity ratio = $(\mu/\mu_{fw})^{0.14}$ = 1.2

s = specific gravity = 0.00129

$$\Delta p = \frac{f\ G_a^2\ L}{5.22 \times 10^{10} \times d_e\ (ft) \times s \times \phi_a}$$

$$= \frac{0.000235 \times (124{,}000)^2 \times 2.89}{5.22 \times 10^{10} \times 0.0149 \times 0.00129 \times 1.2}$$

$$= 8.66 \text{ psi}$$

Pressure drop in annular space = 8.66 psi

2. Pressure Drop in the Inner Tube

The fuel-rich mixture from the vaporization chamber contains 0.48 lb/min = 29 lb/hr air and 1.82 lb/min = 110 lb/hr JP-7. Thus, for the mixture

W = 139 lb/hr

$\bar{V}$ = 2.833 ft^3/lb

ρ = 0.353 lb/ft^3

μ = 0.0241 centipoises

Using a $1\frac{1}{4}$" OD 316-SS tubing with 0.016" wall thickness, 1.218" ID (and 24 fins on the outside) the pressure loss is 0.3 psi/100 ft, hence in the 2.89 feet of pipe the loss will be 0.01 psi, or negligible.

Pressure drop in inner tube = negligible

Thus, both the fuel-rich mixture and the air reach the combustor inlet at 37.5 psig (52.2 psia).

Pressure at combustor inlet = 37.5 psig

3. Weight of Tubing

$1\frac{1}{4}$" OD tubing: 0.8 lb/ft	2.3 lb.
fins: 1.05 lb/ft (for 24 fins)	3.0 lb.
$2\frac{1}{4}$" OD tubing: 1.6 lb/ft	4.6 lb.
heating elements, for 24: 1.2 lb/ft	3.5 lb.
Added for the tee	0.2 lb.
Total	13.6 lb.

$$\boxed{\text{Weight of outlet pipe} = 13.6 \text{ lb.}}$$

C. Startup Heaters

It is necessary to preheat the air-fuel mixture, at the start of the flight, to a temperature of 1,012°F. The resistance heating elements inside the annular space of the jacketed transfer pipe serve for either initial or additional heating of most of the air. The fuel and air passing through the vaporization chamber must be heated for startup in the chamber, however, and this is done by means of heating elements placed around the walls of the chamber.

At the start of the flight there is a demand for 43 scfm of ballast gas or 43 x 1.081 = 46.5 scfm air (= 3.743 #/min.). The amount of fuel required for this quantity of air is 0.274 lb/min = 0.041 gpm = 2.46 gph. At these conditions, again 6 scfm of air are required for the vaporization. Thus:

V_{AIR} = 6 scfm = 0.483 lb/min.

t_{AIR} = 250°F (or less, since all equipment is cold)

h_{AIR}, 250°F = $\underline{\underline{52.2 \text{ BTU/\#}}}$

V_{Fuel} = 0.274 lb/min.

t_{Fuel} = 60°F

h_{Fuel}, 60°F = 12.8 BTU/#

and at the desired temperature of 1,012°F

h_{Air}, 1012°F = 246.7 BTU/#

h_{Fuel}, 1012°F = 751 BTU/#

$$Q = 0.483\,(246.7-52.2) + 0.274\,(751-12.8) = 296 \text{ BTU/min.}$$

The required "heater" is

$$\frac{296 \times 60}{3412.75} = 5.2 \text{ Kw}$$

Allowing for losses, the heater is sized at 6 Kw, and its weight is 12 lb.

Heater size = 6 Kw Heater weight = 12 lbs

III. Total Weight of Air and Fuel Feed Equipment*

Supply piping	19.3 lb.
Spray nozzle	0.6
Vaporization chamber	91
Outlet piping	13.6
Heater	12
Total	136.5 lb.

This figure does not include the weight of a flow control device to regulate the flow of air to the nozzle under all flight conditions. The device is visualized as a flow meter in the air line to the nozzle, connected to a throttling valve in the air line to the jacket of the chamber outlet pipe. Its weight is included under Controls.

Total weight of air and fuel feed equipment = 136.5 lb.

IV. Combustor

The combustor is shown schematically in Figure 33 and 34. Design of the individual components follows.

A. Size and Distribution of Cooling Tubes

Although other alloys may be advantageous, 316 SS is assumed.

*Omitting insulation

316 SS tubing: OD_t = 0.25"

wall thickness = 0.020"

ID_t = 0.21"

Fins: fin height, b_f = 0.375"

fin thickness, th_f = 0.035"

number of fins, N_f = 4 fins per inch tube

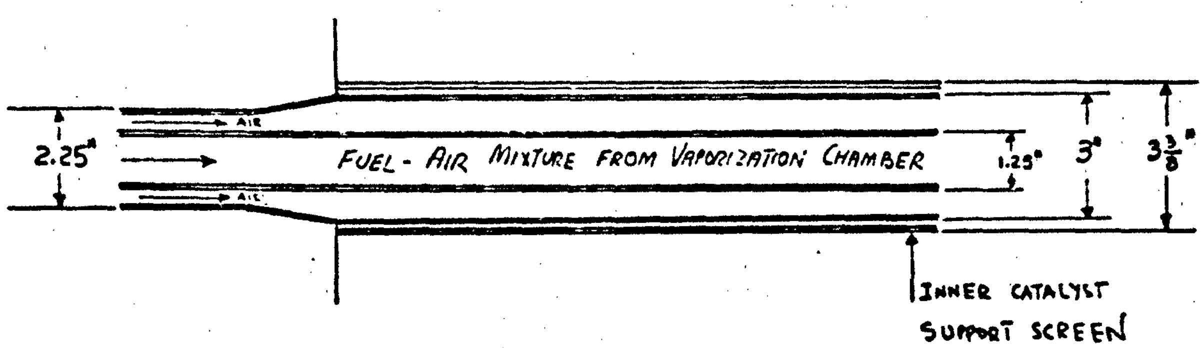


Figure F-3. Details of Combustor Core

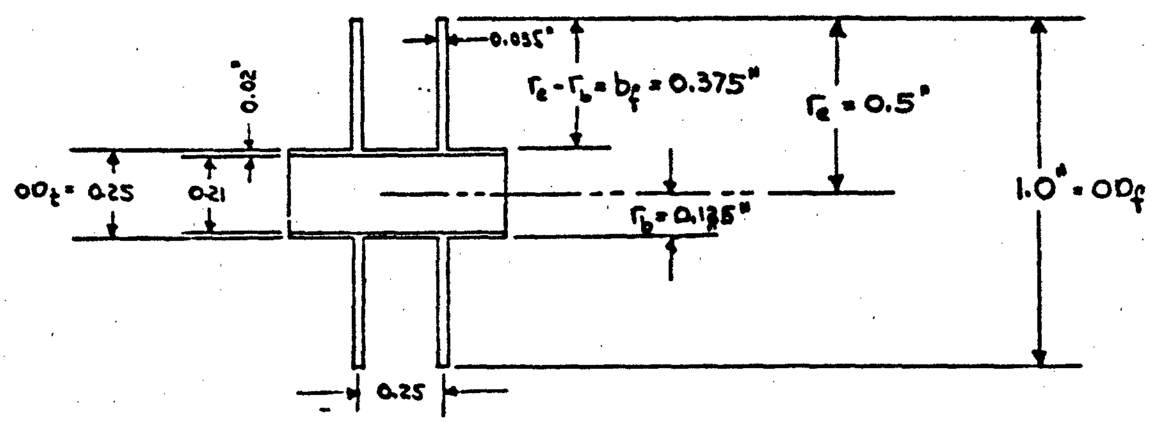


Figure F-4. Detail of Finned Tubing

r_{ISO} = outside radius of inner screen = 1.6875 inch

r_h = radius of centers of heating elements = 1.7875 inch

r_1, r_2, etc. = radii of centers of cooling tubes

r_{OSI} = inner radius of outer screen = 9.6 inches

r_{OSO} = outside radius of outer screen = 9.8 inch

r_{IA} = inside radius of outer combustor wall = 10.5"

r_{OA} = outside radius of outer combustor wall (= outside radius of combustor = 10.625")

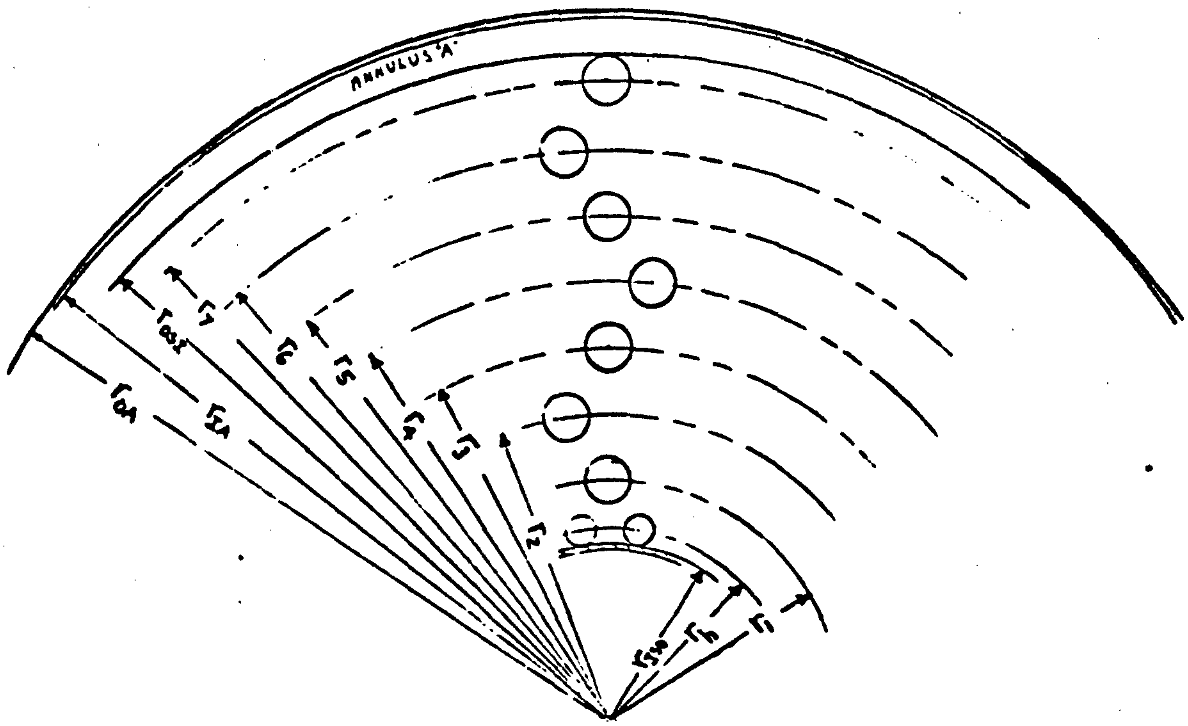


Figure F-5. Detailed Cross Section of Combustor Including Distribution of Cooling Tubes and Heating Elements

Heating elements: OD_h = 3/16"

where Nt/b = number of tubes per bank:

r, inch	d, inch	π d, inch	Nt/b	Spacing: ℄ to ℄ inch	Spacing: between fin edges, inch	Cumulative Number of Tubes
r_h =1.7875	3.575	11.23	N_h = 30	0.374	0.187	---
r_1 = 2.5	5.0	15.71	12	1.31	0.31	12
r_2 = 3.6	7.2	22.62	18	1.26	0.26	30
r_3 = 4.7	9.4	29.53	24	1.23	0.23	54
r_4 = 5.8	11.6	36.44	31	1.175	0.175	85
r_5 = 6.9	13.8	43.35	37	1.17	0.17	122
r_6 = 8.0	16.0	50.26	43	1.17	0.17	165
r_7 = 9.1	18.2	57.2	50	1.144	0.144	215

$$\text{mean spacing} = \frac{\Sigma \pi d}{\text{Total No. tubes}} = \frac{255.11}{215} = 1.19 \text{ inches}$$

$$\text{mean diameter} = \frac{d_1 + d_7}{2} = \frac{\Sigma d}{N_d} = \frac{5.0 + 18.2}{2} = 11.6 \text{ inch}$$

$$\text{mean circumference} = \pi d_{mean} = 3.1416 \times 11.6 = 36.44 \text{ inch}$$

$$\text{mean number of tubes} = \frac{\pi d_{mean}}{\text{mean spacing}} = \frac{36.44}{1.19} = 30.71 \text{ tubes}$$

check: 30.71 x 7 = 215 tubes

$$\boxed{\text{Number of tubes} = 215}$$

Note: radial distance between centerlines of tubes is 1.1 inch and between edges of fins is 0.1 inches.

B. Data

W = weight gases/min. at design conditions.

$$W = W_{AIR} + W_{Fuel}$$

$$= 24.9 + 1.82 = 26.72 \text{ lb/min.}$$

$$\boxed{\text{Gas flow rate} = 1603 \text{ lb/hr}}$$

t_{BG} = 1,337°F T_{BG} = 1,797°R

p = 36.9 psig P = 51.6 psia

V_m = molar volume, ft^3/lb-mole

$$V_m = \frac{P_o}{T_o} V_{mo} \frac{T}{P}$$

$$= \frac{14.7}{492} \times 359 \times \frac{1,797}{51.6} = 373.5 \ ft^3/\text{lb-mole}$$

V_{AIR} = volumetric flow rate of air, ft^3/min.

$$V_{Air} = V_m \times N_{AIR}$$

$$= 373.5 \times 0.8615 = 321.8 \ ft^3/\text{min.}$$

$$V_{Fuel} = W_{Fuel} \times (\overline{V}_{Fuel} \times \frac{P_o}{P})$$

$$= 1.82 \times (8.58 \times \frac{14.7}{51.6}) = 4.5 \ ft^3/\text{min.}$$

V_{A-F} = volumetric flow rate of air-fuel mixture

$$V_{A-F} = V_{AIR} + V_{Fuel}$$

$$= 321.8 + 4.5 = 326.3 \ ft^3/\text{min.}$$

V_{MBG} = volumetric flow rate moist ballast gas

$$V_{MBG} = V_M \times N_{AIR} \times (N_{MBG}/\text{mole air})$$

$$= 373.5 \times 0.8615 \times 1.0765$$

$$= 346.4 \ ft^3/\text{min.}$$

V_{AVG} = average volumetric flow rate through the catalyst bed

$$V_{AVG} = \frac{V_{A-F} + V_{MBG}}{2}$$

$$= \frac{326.3 + 346.4}{2} = 336.3 \text{ ft}^3/\text{min.}$$

$$(= 20{,}180 \text{ ft}^3/\text{hr} = 5.606 \text{ ft}^3/\text{sec})$$

Average flow through catalyst bed = 5.6 ft^3/sec.

ρ_{AVG} = average specific weight

$$\rho_{AVG} = \frac{W}{V_{AVG}}$$

$$= \frac{26.72}{336.3}$$

$$= 0.0795 \text{ lb/ft}^3$$

$\overline{V}_{AVG}$ = average specific volume

$$\overline{V}_{AVG} = \frac{V_{AVG}}{W} = \frac{1}{\rho_{AVG}}$$

$$= \frac{336.3}{26.72}$$

$$= 12.6 \text{ ft}^3/\text{lb}$$

Average specific volume = 12.6 ft^3/lb

<u>Viscosities of gases and vapors at 1,337°F</u>

$$\mu_{MIX} = \frac{\Sigma\ W_i \mu_i}{\Sigma\ W_i}$$

$$\mu_{ABS} = \mu \text{ (centipoises)} \times 2.42 = \mu, \text{ lb/(ft) (hr)}$$

μ_{AIR} = 0.042 centipoises

$\mu_{Fuel\ Vapor}$ = 0.0275 "

$\mu_{Nitrogen}$ = 0.039 "

μ_{CO_2} = 0.0387 "

$\mu_{Water\ Vapor}$ = 0.0335 "

μ_{A-F} = $(24.9 \times 0.042 + 1.82 \times 0.0275) \div 26.72$

= 0.041 centipoises = <u>0.0992 lb/(ft) (hr)</u>

μ_{BG} = $(71.466 \times 0.039 + 19.191 \times 0.0387 + 8.723 \times 0.0365 + 0.62 \times 0.0275) \div 100.00$

= 0.03865 centipoises = <u>0.0935 lb/(ft) (hr)</u>

$$\mu_{AVG} = \frac{\mu_{A-F} + \mu_{BG}}{2}$$

= $(0.0992 + 0.0935) \div 2$ = <u>0.0964 lb/(ft) (hr)</u>

$\boxed{\text{Average viscosity} = 0.096 \text{ lb/ft-hr}}$

<u>Viscosities of Coolant at 212°F</u>

μ_{water} = 0.2838 centipoises = 0.6868 lb/(ft)(hr)

μ_{steam} = 0.013 centipoises = 0.0315 lb/(ft)(hr)

$$\mu_{coolant\ AVG} = \frac{0.6869 + 0.0315}{2} = \underline{\underline{0.36 \text{ lb/(ft) (hr)}}}$$

<u>Thermal Conductivities</u>

at 1337°F: k_{AIR} = 0.043 BTU/(hr)(ft^2)(°F/ft)

$k_{F\text{-}Vapor}$ = 0.075 "

k_{A-F} = $((24.9 \times 0.043 + 1.82 \times 0.075) \div 26.72 = 0.045$

k_{BG} = 0.045 BTU/(hr)(ft^2)(°F/ft)

k_{AVG} = 0.045 "

at 212°F: k_{water} = 0.415 BTU/(hr)(ft^2)(°F/ft)

k_{steam} = 0.0137 "

$k_{Coolant\ AVG}$ = $(0.415 + 0.0137) \div 2 = 0.2144$ BTU/(hr)(ft^2)(°F/ft)

316-SS: k = 13.9 BTU/(hr)(ft^2)(°F/ft) (at 1325°F)

Specific Heats

at 1,337°F: C_{AIR} = 0.252 BTU/(lb)(°F)

$C_{Fuel\text{-}Vapor}$ = 0.855 "

$C_{Water\text{-}Vapor}$ = 0.545 "

$C_{A\text{-}F}$ = 0.295 BTU/(lb)(°F)

C_{BG} = 0.29 "

C_{AVG} = 0.293 "

at 212°F: C_{Water} = 1 BTU/(lb)(°F)

C_{Steam} = 0.455 BTU/(lb)(°F)

$C_{Coolant\ AVG}$ = 0.73 BTU/(lb)(°F)

Film Coefficients (h_d = film coefficient for deposits)

$h_{d\ AIR}$ = 500 BTU/(hr)(ft^2)(°F)

$h_{d\ Fuel\text{-}Vapor}$ = 2,000 BTU/(hr)(ft^2)(°F)

$h_{d\ A\text{-}F}$ = (2,000 x 1.82 + 500 x 24.9) ÷ 26.72 = 602.2 BTU/(hr)(ft^2)(°F)

$h_{d\ well\ water}$ = 500 BTU/(hr)(ft^2)(°F)

C. Calculation of Heat Exchange Surface

Assume combustor is 43" long.

Quantity of heat to be transferred = Q

Q = 26,382 BTU/min x 60 min/hr = 1,583,000 BTU/hr

$\boxed{\text{Heat duty} = 1{,}583{,}000 \text{ BTU/hr.}}$

Temperature difference = Δt

t_1 = temperature of moist ballast gas = 1,337°F

t_2 = temperature of cooling media (wet steam) = 212°F

Δt = 1,337 - 212 = 1,125°F

$$\frac{Q}{\Delta t} = AU = \frac{1,583,000}{1,125} = 1,407 \quad BTU/(hr)(°F)$$

N_f = number of fins per inch of tubing

A_f = fin area per linear ft of tubing

$$A_f = \frac{\pi}{4}(OD_f^2 - OD_t^2) \times 2 \times N_f \times 12 \text{ inch/ft}$$

$$= \frac{\pi}{4}(1^2 - 0.25^2) \times 2 \times 4 \times 12$$

$$= 70.7 \text{ inch}^2/\text{ft} = \underline{\underline{0.491 \text{ ft}^2/\text{ft}}}$$

A_O = bare tube area per linear ft of tubing

$$A_O = \pi \times OD_t \times 12 \text{ inch/ft } (1 - N_f \times th_f)$$

$$= \pi \times 0.25 \times 12 \, (1 - 4 \times 0.035)$$

$$= 8.1 \text{ inch}^2/\text{ft} = \underline{\underline{0.0563 \text{ ft}^2/\text{ft}}}$$

P_p = projected perimeter of tubing per linear ft of tubing

$$P_p = 2 \times b_f \times 2 \times N_f \times 12 + 2\,(12 - N_f \times th_f \times 12)$$

$$= 2 \times 0.375 \times 2 \times 4 \times 12 + 2\,(12 - 4 \times 0.035 \times 12)$$

$$= 92.64 \text{ inch/ft} = \underline{\underline{7.72 \text{ ft/ft}}}$$

d_e = equivalent diameter tubing

$$d_e = \frac{2\,(A_f + A_O)}{\pi \times P_p}$$

$$= \frac{2\,(70.7 + 8.1)}{\pi \times 92.64}$$

$$= 0.54 \text{ inch} = \underline{\underline{0.045 \text{ ft}}}$$

a_s = flow area "in duct"

a_s = cross-section area duct - projected area tubes

$$= H_d \times b_d - N_{t/b} \times OD_t \times b_d - N_{t/b}\,(2 \times th_f \times b_f \times N_f \times b_d)$$

$$= b_d\left[H_d - N_{t/b}\,(OD_t + 2 \times th_f \times b_f \times N_f)\right]$$

where:

b_d = duct width, inch = length of combustor = 43 inch

H_d = duct height, inch = πd_{mean} = 36.44 inch

$N_{t/b}$ = number tubes per bank = $N_{t\ mean}/b$ = 30.71 tubes

$a_s = 43 [36.44 - 30.71 (0.25 + 2 \times 0.035 \times 0.375 \times 4)]$

$= 1098\ inch^2 = \underline{7.62\ ft^2}$

G_s = mass velocity of fluid "in duct"

$$G_s = \frac{W}{a_s} \frac{lb/hr}{ft^2}$$

$$= \frac{1603.2}{7.62}$$

$$= \underline{210\ lb/(hr)(ft^2)}$$

Re_s = Reynolds number of fluid "in duct"

$$Re_s = \frac{d_e \times G_s}{\mu} \qquad \mu = \mu_{AVG}$$

$$= \frac{0.045 \times 210}{0.0964}$$

$$= \underline{98} \text{ (dimensionless)}$$

j_{hf} = heat transfer factor for transverse fins

$j_{hf} = \underline{5.4}$ (2)

h_f = film coefficient for transverse fins

$$h_f = j_{hf} \frac{k}{d_e} \left(\frac{c \times \mu}{k}\right)^{1/3} \phi_s \qquad (\phi_s = 1 \text{ for gases})$$

$$= 5.4 \frac{0.045}{0.045} \left(\frac{0.293 \times 0.0964}{0.045}\right)^{1/3} \times 1$$

$$= \underline{4.62\ BTU/(hr)(ft^2)(°F)}$$

h_f' = film coefficient for transverse fins with fouling correction (h_{do} = outside dirt factor)

$$h_f' = \frac{h_f \times h_{do}}{h_f + h_{do}}$$

$$= \frac{4.62 \times 602.2}{4.62 + 602.2}$$

$$= \underline{4.58\ BTU/(hr)(ft^2)(°F)}$$

Ω = fin efficiency

Y_b = half-thickness of fin

$$Y_b = \frac{th_f}{2}$$

$$= \frac{0.035}{2}$$

$$= 0.0175 \text{ inch} = \underline{0.00146 \text{ ft}}$$

$$(r_e - r_b)\sqrt{h_f'/(k_{316\ SS})\ Y_b}$$

$$= \frac{0.5 - 0.125}{12} \sqrt{\frac{4.58}{13.92 \times 0.00146}} = 0.47$$

$$\frac{r_e}{r_b} = \frac{0.5}{0.125} = 4 \quad \text{(See Figure F-4)}$$

$$\Omega = 0.86 \quad \text{(from appropriate graph}^{(3)}\text{)}$$

a_i = interior area of tubing per linear ft tubing

$$a_i = \pi \times ID_t \times 12$$

$$= \pi \times 0.21 \times 12$$

$$= 7.92 \text{ inch}^2/\text{ft} = \underline{\underline{0.055 \text{ ft}^2/\text{ft}}}$$

h'_{fi} = film coefficient for transverse fins corrected to the inside surface of tube (already corrected for dirt factor)

$$h'_{fi} = (\Omega \times A_f + A_0)\ \frac{h_f'}{a_i}$$

$$= (0.86 \times 0.491 + 0.0563)\ \frac{4.58}{0.055}$$

$$= \underline{\underline{40 \text{ BTU/(hr)(ft}^2\text{)(°F)}}}$$

h_i = film coefficient for interior of tubing

For vaporization of water and based on (Ref. 4) extrapolated
h_i = $\underline{\underline{322 \text{ BTU/(hr)(ft}^2\text{)(°F)}}}$

h_{di} = dirt factor for interior of tubing

For vaporization of water h_{di} = 500 BTU/(hr)(ft^2)(°F)

h_i' = film coefficient for interior of tubing corrected for the dirt factor

$$h_i' = \frac{h_i \times h_{di}}{h_i + h_{di}}$$

$$= \frac{322 \times 500}{322 + 500}$$

$$= \underline{\underline{196 \text{ BTU/(hr)(ft}^2\text{)(°F)}}}$$

U_{Di} = overall design coefficient of heat transfer based on the interior surface of tubing

$$U_{Di} = \frac{h_{fi}' \times h_i'}{h_{fi}' + h_i'}$$

$$= \frac{40 \times 196}{40 + 196}$$

$$= 33.22 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Overall design coefficient = 33.2 BTU/hr-ft^2-°F

A_i = total heat transfer area, based on the interior surface of tubing

$$Q = A_i U_{Di} \Delta t$$

$$\frac{Q}{\Delta t} = A_i U_{Di}$$

$$\therefore A_i = \frac{Q}{\Delta t} \div U_{Di}$$

$$= \frac{1407.1}{33.22} = 42.36 \text{ ft}^2$$

Required area, interior of coils = 42.4 ft^2

L = total length of tubing

$$L = \frac{A_i}{a_i} \quad \frac{\text{ft}^2}{\text{ft}^2\text{/lineal ft}}$$

$$= \frac{42.4}{0.055}$$

$$= 770 \text{ ft}$$

Tubing length = 770 ft.

N_{Tt} = total number of tubes

$$N_{Tt} = \frac{L}{b_d} \qquad \frac{\text{(Total length)}}{\text{(reactor length)}}$$

$$= \frac{770}{43 \div 12}$$

$$= 214.8 \text{ tubes}$$

$$\boxed{\text{Number of tubes} = 215}$$

Thus, the chosen 215 tubes satisfy the requirements for the heat transfer surface.

D. Volume of Catalyst Bed

υ_t = Volume occupied by 1 linear foot of tubing

$$\upsilon_t = \frac{\pi \times OD_t^2}{4} \times 12 + 4\,(OD_f^2 - OD_t^2)\,\frac{\pi}{4} \times thf \times 12$$

$$= \underline{1.826 \text{ inch}^3/\text{ft}}$$

V_t = total volume occupied by the tubing

$$V_t = \upsilon_t \times N_{Tt} \times b_d \text{ (ft)}$$

$$= 1.826 \times 215 \times (43 \div 12)$$

$$= \underline{\underline{1407 \text{ inch}^3}}$$

υ_h = vol. for 1 foot of heating element $= \frac{\pi \times OD_h^2}{4} \times 12$

$$= \frac{\pi\,(3/16)^2}{4} \times 12$$

$$= \underline{0.3314 \text{ inch}^3/\text{ft}}$$

V_h = volume occupied by the heating elements

$$V_h = \upsilon_h \times N_h \times b_d \text{ (ft)}$$

$$= 0.33134 \times 30 \times 43/12$$

$$= \underline{\underline{35.6 \text{ inch}^3}}$$

V_c = volume of catalyst

was calculated and indicated in Section V-6-h

$V_c = 0.3513 \text{ ft}^3 = 607 \text{ inch}^3$

Volume of catalyst = 607 in^3

Thus, for total volume

$V_t + V_h + V_c = 1{,}407 + 35.6 + 607 = 2{,}050 \text{ inch}^3$

But from geometrical considerations:

V_R = volume of annular catalyst bed

$V_R = \pi \, (r_{OSI}^2 - r_{ISO}^2) \times b_d$ (inches)

$= \pi \, (9.6^2 - 1.6875^2) \times 43$

$= 10{,}053 \text{ inch}^3$

Volume of bed for assumed tube spacing = 10,050 $inch^3$

It is readily seen that the volume of the catalyst bed is much larger than that required by the tubing, heating elements and the catalyst. Consequently, the catalyst has to be "diluted", preferably with a low bulk density and high conductivity material (its particles should be the same size and shape as the catalyst extrudates).

V_{CD} = volume of catalyst diluent

$V_{CD} = V_R - (V_t + V_h + V_c)$

$= 10{,}050 - 2{,}050$

$= 8{,}000 \text{ inch}^3 = 4.63 \text{ ft}^3$

Volume of diluent = 4.63 ft^3

It should be noted that the heat exchange surface is not optimized. Further optimization and experimental data on catalyst bed cooling may yield a smaller heat transfer area and smaller catalyst bed volume, resulting in reduced requirement for diluent. Use of a specially developed catalyst, capable of withstanding higher temperatures than Catalyst A and/or tubing with fully optimized fin design, can be expected to result in a significantly smaller catalyst bed.

E. Fuel Preheating Pipe

The pipe is 1/8" IPS schedule 5, 316-SS

OD_t = 0.405 inch   wall = 0.035 inch ID_t = 0.335 inch

Fins: height = b_f = 0.1 inch

thickness = th_f = 0.035 inch

N_f = 8 transverse fins per inch linear.

LMTD = 858°F

Average temperature BG = 1337°F

Average temperature fuel = 475°F

μ_F = 0.46 lb/(ft)(hr)

C_{pF} = 0.703 BTU/(lb)(°F)

k_F = 0.0701 BTU/(hr)(ft^2)(°F/ft)

ρ_F = 39.2 lb/ft^3

S_F = ρ/62.4 = 0.629

hd_F = 250 BTU/(hr)(ft^2)(°F)

W_F = 1.82 x 60 = 109.2 lb/hr

Q = 328 x 60 = 19,680 BTU/hr

A_f = 0.2115 ft^2/ft lin.

A_O = 0.0764 ft^2/ft lin.

P_p = 4.64 ft

d_e = 0.0395 ft

a_s = 0.0797 ft^2

W_s = 17.6 lb/hr

G_s = 221 lb/(hr)(ft^2)

Re_s = 94

jf = 2.6(2)

$(c\mu/k)^{1/3}$ = 1.82

$h_f = 5.4$ BTU/(hr)(ft^2)(°F)

$h_{ds} = 602$ BTU/(hr)(ft^2)(°F)

$h'_f = 5.35$ BTU/(hr)(ft^2)(°F)

$a_t = 0.0006$ ft^2

$de_t = 0.028$ ft

$G_t = 178{,}400$ lb/(hr)(ft^2)

$Re_t = 11{,}000$

$jt = 39^{(5)}$

$(c\mu/k)^{1/3} = 1.663$

$h_{ti} = 162.4$ BTU/(hr)(ft^2)(°F)

$h'_{ti} = 98.45$ BTU/(hr)(ft^2)(°F)

$k_t = 12.35$ BTU/(hr)(ft^2)(°F/ft)

$Y_b = 0.00146$ ft

$r_e/r_b = 1.5$

$(r_e - r_b)\ (h'_f/k_t\ Y_b)^{\frac{1}{2}} = 0.144$

$\Omega = 0.99^{(3)}$

$a_i = 0.0877$ ft^2/ft lin.

$h'_{fi} = 17.4$ BTU/(hr)(ft^2)(°F)

$$U_{Di} = \frac{h'_{fi} \times h'_{ti}}{h'_{fi} + h'_{ti}} = 14.8 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Design coefficient = 14.8 BTU/hr-ft^2-°F

$$A_i = \frac{Q}{U_{Di}\ \Delta t} = \frac{19{,}680}{14.8 \times 858} = 1.55 \text{ ft}^2$$

Internal pipe area = 1.55 ft^2

$$L_t = \frac{A_i}{a_i} = \frac{1.55}{0.0877} = 17.8 \text{ ft.}$$

Total pipe length = 17.8 ft

Thus, about 5 passes inside annulus A.

Weight = 0.263 x 17.8 + 2 x 0.14 = 5 lb.

Weight of preheating pipe = 5 lb.

F. Pressure Drop

As mentioned previously, both streams reach the inlet to the combustor "core" with a pressure of 37.53 psig (52.23 psia).

Pressure loss in inner perforated tube of the "core"

Cross-section area = $A_i = \frac{\pi}{4} ID_t^2 = \frac{\pi}{4} 1.218^2 = 1.165$ inch^2 = 0.00809 ft.^{2}

Effective diameter = $d_{et} = ID_t/12$ = 0.1015 ft.

Mass velocity = $G_{it} = W/A_i$ = 138/0.00809 = 17,200 lb./(hr.) (ft.^{2})

μ = 0.026 centipoises x 2.42 = 0.063 lb./(ft.) (hr.) (at 1012°F.)

$R_e = d_{et} \times G_{it}/\mu$ = 0.1015 x 17,200/0.063 = 27,700

friction factor, f = 0.000235 (6)

Molecular volume = V_m = (14.7/492) x 359 x (1472/52.23) = 302.5 ft.^{3}/lb. mole

Air flow rate = V_{air} = 302.5 x (0.48/28.9) = 5.0 ft.^{3}/min

Specific volume of fuel vapor = 7 ft.^{3}/lb. at 1012°F. and 14.7 psia

Fuel vapor flow rate = 7 x 1.82 x (14.7/52.23) = 3.6 ft.^{3}/min.

Total flow rate V_T = 5.0 + 3.6 = 8.6 ft.^{3}/min.

Weight flow rate = 0.48 + 1.82 = 2.30 lb./min. = 138 lb./hr.

Density mix. = ρ = 2.30/8.6 = 0.267 lb./ft.^{3}

Specific volume mixture = 8.6/2.30 = 3.74 ft.^{3}/lb.

Specific gravity related to water = $S_i = \rho/62.4$ = 0.267/62.4 = 0.00428

$$\Delta P = \frac{f \times G_{it}^2 \times L}{5.22 \times 10^{10} \times de_t \times S_i\, \phi_t} \qquad (\phi_t \text{ for gases} = 1)$$

$$= \frac{0.000235 \times 17{,}200^2 \times 4}{5.22 \times 10^{10} \times 0.1015 \times 0.00428}$$

ΔP = 0.012 psi

Pressure drop within inner core = 0.012 psi

The pressure loss through the perforations was calculated by the method used for screens (shown later) yielding ΔP = 0.0001 psi (negligible).

Pressure loss in outer perforated pipe of the "core"

Since it carries the rest of the air plus the fuel-air mixture from the inner perforated tube, the pressure loss in this annulus is calculated for the total mixture.

Cross-section area of annulus $= A_a = \frac{\pi}{4} (ID_0^2 - OD_1^2) = \frac{\pi}{4} (2.93^2 - 1.25^2)$

$$= \underline{5.515 \text{ inch}^2 = 0.0383 \text{ ft}^2}$$

Effective diameter $= d_{ea} = ID_0 - OD_1 = 2.93 - 1.25 = 1.68$ inch $= \underline{0.14 \text{ ft}}$

Mass velocity $= G_a = W/A_a = 1{,}603.2/0.0383 = \underline{41{,}900 \text{ lb/(hr)(ft}^2)}$

$\mu_{mix} = (0.037 \times 24.9 + 0.026 \times 1.82) \div 26.72 = 0.0363$ centipoises $\times 2.42 = 0.0877$ lb/(ft)(hr)

$R_e = de_a\, G_a/\mu = 0.14 \times 41{,}900/0.0877 = 66{,}800$

$f = 0.006$ (7)

$V_T = V_{AIR} + V_{F\text{-}Vapor} = 302.5 \times 0.8615 + 1.97 = 262.6$ ft^3/min.

$\rho = 26.72/262.6 = 0.102$ lb/ft^3

$$\Delta F_a = \frac{4\, f\, G_a^2\, L}{2\, g \rho^2\, d_{ea}}$$

$$= \frac{4 \times 0.006 \times 41{,}900^2 \times 4}{2 \times (4.18 \times 10^8) \times (0.102)^2 \times 0.14}$$

$\underline{\Delta F_a = 145 \text{ ft}}$

Velocity $= V = V_T/60\, A_a = 262.6/(60 \times 0.0383) = 114.3$ fps

$F_1 = 3\, V^2/2\, g = 3 \times 114.3^2/64.4 = \underline{609 \text{ ft}}$

$$\Delta P_a = \frac{(\Delta F_a + F_1)\rho}{144}$$

$$= \frac{(145 + 609) \times 0.102}{144}$$

$\Delta P_a = \underline{0.533 \text{ psi}}$

Loss through the perforations is 0.0015 psi, thus, the total pressure loss is 0.534 psi.

$$\boxed{\Delta P \text{ in outer pipe of core} = 0.55 \text{ psi}}$$

Consequently, the mixture reaches the inner screen with 36.98 psig (51.68 psia).

Pressure loss in the inner screen

The calculation procedure employed below is from Multi-Metal Wire Cloth, Inc. (Tappan, New York).

M = mesh = 10

D_w = wire diameter, inch = 0.041"

Opening size, inch = 0.059"

% open area = 34.8

$a = (1-M \times D_w)^2 = (1-10 \times 0.041)^2 = 0.348$

$D_o = \left[(1/M) - D_w\right] = \left[(1 \div 10) - 0.041\right] = 0.059$

$A = \left[(1-a^2) \div a^2\right] = \left[(1-0.348^2) \div 0.348^2\right] = 7.253$

$B = D_o \div a = 0.059 \div 0.348 = 0.1695$

$\alpha = 0.00674A = 0.00674 \times 7.253 = \underline{0.05}$

$\beta = 7740B = 7740 \times 0.1695 = \underline{1312}$

$V_m = (14.7/492) \times 359 \times (1472/51.68) = 305.4$ ft^3/lb-mole

$V_{AIR} = V_m \times 0.8615 = 263.1$ ft^3/min.

$V_{F\text{-}Vapor} = 1.82 \times 7\ (14.7/51.68) = 3.62$ ft^3/min.

$V_T = 263.1 + 3.62 = 266.7$ ft^3/min = $\underline{4.445 \text{ ft3/sec.}}$

$\rho = 26.72/266.7 = 0.10018$ lb/ft^3 = $\underline{0.00161 \text{ g/cc}}$

$\mu = 0.036$ centipoises

A_s = screen area = $2\pi r_{IS} L/144 = 2\pi \times 1.68 \times 43/144 = \underline{3.15 \text{ ft}^2}$

V = flow velocity = $V_T/A_s = 4.445/3.15 = \underline{1.41 \text{ ft/sec}}$

$$Re = \frac{\beta V \rho}{\mu} = \frac{1312 \times 1.41 \times 0.00161}{0.036} = 82.5$$

C = factor from the supplied graph = $\underline{0.92}$

$$\Delta P = \frac{\alpha \rho V^2}{C^2}$$

$$= \frac{0.05 \times 0.00161 \times 1.263^2}{0.92^2}$$

$$= < 0.0002 \text{ psi}$$

$\boxed{\text{Pressure drop at inner screen is negligible}}$

Pressure drop through bed

D_p = equivalent particle diameter (that is, diameter of an equivalent sphere).

For cylindrical extrudates

$$D_p = \frac{3}{\frac{2}{d} + \frac{1}{l}}$$

d = cylinder diameter = 1/16 inch = 0.0052 ft

l = cylinder length = 1/8 inch = 0.0104 ft

$$D_p = \frac{3}{\frac{2}{0.0052} + \frac{1}{0.0104}} = \underline{0.00625 \text{ ft}}$$

The formula to use for ΔP depends on type of flow, laminar or turbulent.

Since we have laminar flow (Re = 14), the formula to use is

$$\Delta P = \frac{53 \mu \; L \; A_f \; V_o}{144 \; D_p^2} \quad \text{psi}$$

where

L = bed thickness (including diluent) = 8 inch = 0.67 ft

A_f = wall effect factor = 1

μ = average viscosity = 0.0000268 lb/(ft)(sec)

V_o = velocity through empty bed = 0.74 fps

$$\Delta P = \frac{53 \times 0.0000268 \times 0.67 \times 1 \times 0.74}{144 \times (0.00625)^2}$$

$$\Delta P = 0.12 \text{ psi}$$

Pressure drop through bed = 0.12 psi

In addition to the pressure loss through the particles, the ΔP due to frictional surface of the tubes was calculated (method shown in HE #1; duct side ΔP). It amounts to 0.00003 psi, which is negligible.

The pressure of the moist ballast gas reaching the outer screen is 36.87 psig (51.57 psia).

Pressure drop through the outer screen

Calculations, as outlined for the inner screen, yield a pressure loss of 0.00013 psi (negligible). Even if a double outer screen is used, the pressure loss remains negligible.

Pressure drop in Annulus A (see Figure 34)

$ID_{aA} = 2\ r_{OSO} = 2 \times 9.8 = 19.6$ inch

$OD_{aA} = 2\ r_{IA} = 21.0$ inch

P_{W_A} = wetted perimeter = $\pi\ (ID_{aA} + OD_{aA}) = \pi(19.6 + 21) =$
127.55 inch = $\underline{10.63\ ft.}$

A_{aA} = flow area in annulus = $\pi\ (OD_{aA}{}^2 - ID_{aA}{}^2) \div 4 = \pi(21^2 - 19.6^2) \div 4 =$
44.65 inch^2 = $\underline{0.31\ ft^2}$

de_A = equivalent diameter = $4\ A_{aA}/P_{W_A} = (4 \times 0.31)/10.63 = \underline{0.117\ ft}$

G_{aA} = mass velocity = $W/A_{aA} = 1603.2/0.31 = \underline{5,172\ lb/(hr)(ft^2)}$

$Re_{aA} = de_A \times G_{aA}/\mu = 0.11667 \times 5,172/0.0935 = 6,450$

$\underline{f_A = 0.0023}$ (8)

$S = \rho/62.4 = 0.0771/62.4 = \underline{0.00124}$

$\phi_a = 1$

$$\Delta P = \frac{f_A \times G_{aA}{}^2 \times L_A}{5.22 \times 10^{10} \times de_a \times S \times \phi_a} = \frac{0.0023 \times 5,172^2 \times 3.58}{5.22 \times 10^{10} \times 0.117 \times 0.00124}$$

$\Delta P = 0.030$ psi

$$\boxed{P \text{ in Annulus A} = 0.03 \text{ psi}}$$

For the converging annulus (or cone) a pressure loss of 2% is assumed. Thus

$$0.02 \times 36.8 = 0.74 \text{ psi}$$

Thus, the moist hot ballast gas leaves the reactor under a pressure of 36.1 psig (50.8 psia).

$$\boxed{\text{Exit pressure} = 36.1 \text{ psig}}$$

G. Combustor Jacket for Air Preheat

The idea of surrounding the combustor with an annular jacket for preheating combustion air was abandoned after calculations showed that even with finned coils only about 4% of the total heat required by the air could be transferred in a reasonably-sized annulus. A large annulus is objectionable because it causes an excessive expansion of the combustion gases. An outer air jacket might be considered as a means of cooling the outside wall of the combustor, but is not included in this study.

H. Combustor Weight (all parts of 316-SS except as noted)

Inner core tube	0.216 lb/ft x 3.583 ft =	1 lb
Outer core tube	1.134 " x 3.583 ft =	4
Inner screens (2)	3.013 " x 3.583 ft =	11
Catalyst	0.3513 ft^3 x 40.6 lb/ft^3 =	14
Catalyst diluent	4.63 ft^3 x 25 " =	116
Cooling tubing	0.409 lb/ft x 790 ft =	323
Outer screens (2)	17.391 lb/ft x 3.583 ft =	62
Outer wall* (0.125" thick)	29.4 lb/ft x 4.583 ft =	135
Outlet connection	Std. 4" IPS pipe =	11
Fuel Preheating tubing		5
Heating elements		12
	Total	694

Total weight of combustor = 694 lb

V. Heat Exchanger No. 1 (HE 1)

Cooling of moist ballast gas leaving the combustor and preheating of incoming combustion air are accomplished in HE 1. The moist ballast gas, which is hotter, is placed inside the tubes, so that the outer shell wall thickness is based on the cooler air stream. (For Hastelloy C, the design tensile strength is 24,500 psi at the exit air temperature, as compared to 20,700 psi at entering BG temperature.)

A. Assumptions

Duct: 2 x 2 ft (inside) (b_d = 2 ft h_d = 2 ft)

Tubing: OD_t = 1 inch wall - 0.035 inch ID_t = 0.93 inch

Fins: height = b_f = 0.5 inch thickness = th_f = 0.035 inch

N_f = 8 transverse fins per linear inch tubing

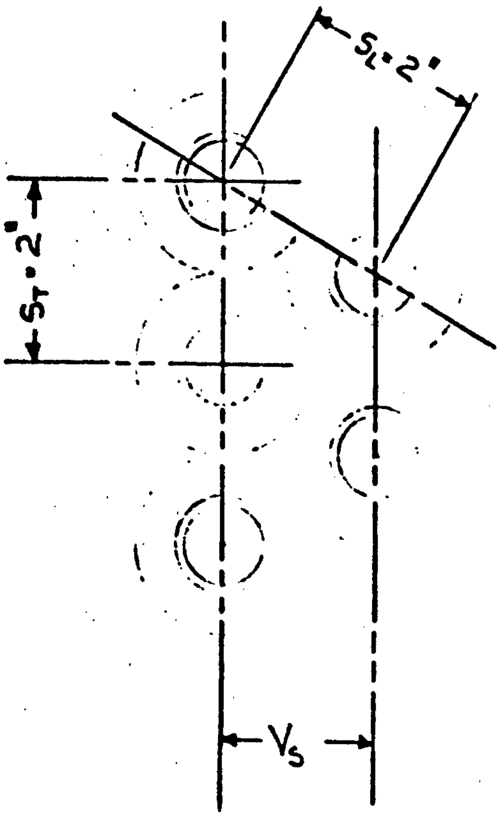
r_e = 1 inch r_b = 0.5 inch OD_f = 2 inch

Tube banks: $N_t/b1$ = 12 tubes

and $N_t/b2$ = 11 tubes in alternate banks

*1 ft length added for the converging cone. Made of Hastelloy C.

Triangular pitch:



V_s = volumetric section

$V_s = 2 \cos 30°$

$V_s = 1.732$ inch

Figure F-6. Arrangement of Finned Tubes in HE 1

B. Heat Exchange Surface

$Q = 5{,}573$ BTU/min = 334,400 BTU/hr

Air = enters at 250°F = t_3

leaves at 1,112°F = t_4

Moist BG: enters at 1,337°F = t_1

leaves at 558°F = t_2

Hot Fluid		Cold Fluid	Difference
$t_1 = 1{,}337°F$	Higher temperature	$t_4 = 1{,}112°F$	$t_1 - t_4 = 225°F = \Delta t_h$
$t_2 = 558°F$	Lower temperature	$t_3 = 250°F$	$t_2 - t_3 = 308°F = \Delta t_c$
$t_1 - t_2 = 779°F$	Difference	$t_4 - t_3 = 862°F$	$83°F$

1. **Mean temperature difference**

$$\text{LMTD} = \frac{\Delta t_c - \Delta t_h}{2.3 \log \dfrac{\Delta t_c}{\Delta t_h}}$$

$$= \frac{308 - 225}{2.3 \log \dfrac{308}{225}} = \underline{263°F}$$

$$R = \frac{t_1 - t_2}{t_4 - t_3} = \frac{779}{862} = 0.9$$

$$S = \frac{t_4 - t_3}{t_1 - t_3} = \frac{862}{1{,}337 - 250} = 0.8$$

$F_T = 0.50$ (9)

$$\Delta t = F_T \times \text{LMTD} = 0.5 \times 263 = 132°F$$

Mean temp. difference = 132°F

2. **Caloric temperatures of the fluids**

$$\frac{\Delta t_c}{\Delta t_h} = \frac{308}{225} = 1.37$$

assuming $K_c = 1$

$F_c = 0.47$ (10)

T_c = caloric temperature of BG (mean for HE 1)

$$T_c = t_2 + F_c\,(t_1 - t_2)$$

$$= 558 + 0.47\,(1337 - 558) = \underline{924°F}$$

t_c = caloric temperature of air (mean for HE 1)

$$t_c = t_3 + F_c\,(t_4 - t_3)$$

$$= 250 + 0.47\,(1{,}112-250) = \underline{681^\circ F}$$

3. Data at above temperatures

BG at 924°F and 50.7 psia

μ_{BG} = 0.0774 lb/(ft)(hr)

C_p = 0.267 BTU/(lb)(°F)

k_{BG} = 0.035 BTU/(hr)(ft^2)(°F/ft)

ϕ_t = 1

V_M = 292.8 ft^3/lb-mole

V_{BG} = 271.5 cfm = 4.525 cfs

ρ = 0.0984 lb/ft^2

$S_t = \rho/62.4 = 0.001577$

$(C\mu_{BG}/k_{BG})^{1/3} = 0.84$

Air at 681°F and 64 psia

μ_A = 0.075 lb/(ft)(hr)

C_p = 0.254 BTU/(lb)(°F)

k_A = 0.0282 BTU/(hr)(ft^2)(°F/ft)

ϕ_s = 1

V_M = 171.5 ft^3/lb-mole

V_{AIR} = 147.8 cfm = 2.463 cfs

ρ = 0.168 lb/ft^3

S_s = 0.0027

$(C\mu_A/k_A)^{1/3}$ = 0.878

A_f = fin area per linear foot of tubing

$$A_f = \frac{\pi}{4}\,(OD_f^2 - OD_t^2) \times 2\,N_f \times 12$$

$$= \pi\,(2^2-1^2) \times 2 \times 8 \times 3 = 452.4 \text{ inch}^2/\text{ft} = \underline{3.14\ \text{ft}^2/\text{ft}}$$

A_o = bare tube area per lineal foot

$A_o = \pi \times OD_t \times 12\ (1 - N_f \times th_f)$

$= \pi \times 1 \times 12\ (1 - 8 \times 0.035) = 27.14\ \text{inch}^2/\text{ft} = \underline{0.189\ \text{ft}^2/\text{ft}}$

P_p = projected perimeter of tubing, feet per lineal foot

$P_p = 2 \times b_f \times 2 \times N_f \times 12 + 2\ (12 - N_f \times th_f \times 12)$

$= 2 \times 0.5 \times 2 \times 8 \times 12 + 2\ (12 - 8 \times 0.035 \times 12)$

$= 209.3\ \text{inch/ft} = \underline{17.44\ \text{ft/ft}}$

de_s = equivalent diameter of tubing

$$de_s = \frac{2\ (A_f + A_o)}{\pi \times P_p}$$

$= 2\ (3.14 + 0.189) \div (\pi \times 17.44) = \underline{0.122\ \text{ft}}$

A_s = flow area in duct

$$A_s = 12\ b_d \left[12\ h_d - N_{t/bl}\ (OD_t + 2 \times N_f \times th_f \times b_f)\right]$$

$$= 12 \times 2 \left[\ 12 \times 2 - 12\ (1 + 2 \times 8 \times 0.035 \times 0.5)\ \right]$$

$= 207.4\ \text{inch}^2 = 1.44\ \text{ft}^2$

$$\boxed{\text{Flow area in duct} = 1.44\ \text{ft}^2}$$

4. Duct side: air

mass velocity, G_s

$G_s = W_s/A_s = 24.9 \times 60/1.44 = \underline{1038\ \text{lb}/(\text{ft}^2)(\text{hr})}$

Reynolds number, Re_s

$Re_s = de_s \times G_s/\mu_A = 0.122 \times 1{,}038/0.075 = \underline{1{,}700}$

heat transfer factor, j_f

$j_f = 20^{(2)}$

$h_f = j_f\ \frac{k_A}{de_s} \left(\frac{C\mu_A}{k_A}\right)^{1/3} \phi_s$ (coefficient for transverse fins)

$= 20\ \frac{0.0282}{0.122} \times 0.878 \times 1 = \underline{4.08\ \text{BTU}/(\text{hr})(\text{ft}^2)(^\circ\text{F})}$

Fouling factor = R_{do} = 0.003

$h_{ds} = 1/R_{do} = 333$ BTU/(hr)(ft^2)(°F)

coefficient with correction for fouling, h_f'

$$h_f' = \frac{h_f \times h_{ds}}{h_f + h_{ds}} = \frac{4.08 \times 333}{4.08 + 333} = 4.04 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Corrected coefficient, duct side = 4.04 ditto

5. Inside tubes - moist BG

$$A_t = \frac{\pi}{4} ID_t^2 = \frac{\pi}{4} \times 0.93^2 = 0.68 \text{ inch}^2 = 0.00472 \text{ ft}^2\text{/tube}$$

$$A_{t/b} = A_t\, N_{t/b1} = 0.68 \times 12 = 8.16 \text{ inch}^2 = \underline{0.0567 \text{ ft}^2\text{/bank}}$$

$$de_t = ID_t/12 = 0.93/12 = \underline{0.0775 \text{ ft}}$$

$$G_t = \frac{W_t}{A_{t/b}} = \frac{1603.2}{0.0567} = \underline{28{,}300 \text{ lb/(hr)(ft}^2)}$$

$$Re_t = \frac{de_t \times G_t}{\mu_{BG}} = \frac{0.0775 \times 28{,}300}{0.07744} = 28{,}300$$

$$j_{hi} = 93^{(5))}$$

$$h_i = j_{hi} \frac{k_{BG}}{de_t} \left(\frac{c\mu_{BG}}{k_{BG}}\right)^{1/3} \phi_t$$

$$= 93 \frac{0.035}{0.0775} \quad 0.84 \times 1 = \underline{35.4 \text{ BTU/(hr)(ft}^2\text{)(°F)}}$$

h_{di} = 602 BTU/(hr)(ft^2)(°F)

$$h_i' = \frac{h_i \times h_{di}}{h_i + h_{di}} = 33.4 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Corrected coefficient, tube side = 33.4 BTU/hr-ft^2-°F

6. Overall design coefficient, heat transfer surface and number of banks

r_e = outer radius of fin $\qquad$ r_b = inner radius of fin

$$\frac{r_e}{r_b} = \frac{1}{0.5} = \underline{\underline{2}}$$

$Y_b = th_f/2 = 0.035/2 = 0.0175$ inch = $\underline{0.00146 \text{ ft}}$

K_t = conductivity of fin = 11.8 BTU/hr-ft^2-°F-ft

$(r_e - r_b)(h_f'/k_t Y_b)^{0.5}$ =

$$\frac{0.5}{12}\left(\frac{4.04}{11.8 \times 0.00146}\right)^{0.5} = \underline{0.636}$$

fin efficiency = Ω

Ω = 0.85(3)

A_i = interior surface per linear foot

$A_i = \pi\, ID_t \times 12 = \pi \times 0.93 \times 12 = 35.06$ inch^2/ft = $\underline{0.244 \ \ ft^2/ft}$

Corrected film coefficient, h_{fi}'

$h_{fi}' = (\Omega \times A_f + A_o)\, h_f'/A_i$

$= (0.85 \times 3.14 + 0.189)\ 4.04/0.244$

$= 47.5$ BTU/(hr)(ft^2)(°F)

Coefficient for transverse fins corrected to internal surface of tubing = 47.5 BTU/hr-ft^2-°F

$$U_{Di} = \frac{h_{fi}' \times h_i}{h_{fi}' + h_i'} = \frac{47.5 \times 33.4}{47.5 + 33.4}$$

Overall design coefficient = 19.6 BTU/(hr)(ft^2)(°F)

$$A_{iT} = \frac{Q}{U_{Di}\,\Delta t} = \frac{334,400}{19.6 \times 132}$$

Internal tubing area = 129 ft^2

$$A_{i/b} = [0.5\ (N_{t/b1} + N_{t/b2})]\ a_i \times L_b \qquad (L_b = b_d)$$

$$= [0.5\ (12 + 11)]\ 0.2435 \times 2 = \underline{5.6\ ft^2/bank}$$

number of banks $= N_b = A_{iT}/A_{ib}$

$$= 129 \div 5.6$$

$$\boxed{\text{Number of banks} = 23}$$

C. Pressure Drop

1. Duct Side: Air

V_{NF} = net free volume

$$V_{NF} = b_d \times h_d \times V_s - \tfrac{1}{2}\ (N_{t/b1} + N_{t/b2})\ \frac{\pi}{4}\ \frac{b_d}{144}\ [OD_t^2 + th_f \times N_f(OD_f^2 - OD_t^2)]$$

$$= 2 \times 2 \times \frac{1.732}{12} - \frac{1}{2}\ (12 + 11)\ \frac{\pi}{4}\ \frac{2}{144}\ [1^2 + 0.035 \times 8\ (2^2 - 1^2)]$$

$$= \underline{\underline{0.357\ \ ft^3}}$$

S_F = frictional surface (duct walls may be neglected)

$$S_F = 1/2\ (N_{t/b1} + N_{t/b2}) \times (A_f + A_o) \times b_d$$

$$= 1/2\ (12 + 11)\ (3.14\ + 0.189)\ 2 = \underline{\underline{76.6\ ft^2}}$$

D'_{ev} = volumetric equivalent diameter

$$D'_{ev} = \frac{4\ V_{NF}}{S_F} = \frac{4 \times 0.3573}{76.6} = \underline{\underline{0.0187}}\ ft$$

$G_s = 1038\ lb/(hr)(ft^2)$

$$Re = \frac{D'_{ev}\ G_s}{\mu_A} = \frac{0.0187 \times 1{,}038}{0.075} = \underline{260}$$

$f = 0.004^{(2)}$

L_p = length of path

$$L_p = N_b \times V_s = 23 \times \frac{1.732}{12} = \underline{3.34 \text{ ft}}$$

$$\left(\frac{D'_{ev}}{S_T}\right)^{0.4} = \left(\frac{0.0187}{2/12}\right)^{0.4} = \underline{0.417}$$

$$\left(\frac{S_L}{S_T}\right)^{0.6} = \left(\frac{2/12}{2/12}\right)^{0.6} = \underline{1.0}$$

$$\Delta P_A = \frac{f \times G_s^2 \times L_p}{5.22 \times 10^{10} \times D'_{ev} \times S_s \times \phi_s} \left(\frac{D'_{ev}}{S_T}\right)^{0.4} \left(\frac{S_L}{S_T}\right)^{0.6}$$

$$= \frac{0.004 \times 1{,}038^2 \times 3.34 \times 0.417 \times 1}{5.22 \times 10^{10} \times 0.0187 \times 0.0027 \times 1}$$

Air pressure drop = 0.0023 psi (Negligible)

The above excludes the gradual enlargement and contraction losses in the duct.

It is assumed (Table XIV) that air leaves the engines with 53 psig (67.7 psia). Further, it is assumed that the loss between the engine and HE 1 is 0.7; thus air reaches HE 1 at 52.3 psig. Loss in the diverging cone is about 8% of this pressure, or about 4 psi, and in the converging cone is 2% or 1 psi. Thus:

total air pressure loss = 5 psi air leaves HE 1 at 47.3 psig (62 psia)

2. Tube Side: Moist BG

A_t, $A_{t/b}$, de_t, G_t and R_{et} are the same as calculated for heat transfer.

$f = 0.0002^{(6)}$

$$\Delta P_{BG} = \frac{f \times G_t^2 \times L_b \times N_b}{5.22 \times 10^{10} \times de_t \times S_t \times \phi_t} \quad (L_b = bd)$$

$$= \frac{0.0002 \times (28,300)^2 \times 2 \times 23}{5.22 \times 10^{10} \times 0.0775 \times 0.001577 \times 1}$$

$= 1.16$ psi

Pressure drop, BG in tubes = 1.16 psi

In addition to the loss in straight parts of the tubes there are the losses due to the return bends, of which there are 22, and the two headers. A return bend of 1" OD tubing is equivalent to 5.5 ft of straight tubing. It is assumed that the same is valid for the headers. Consequently:

$(22 + 2) \times 5.5 = 132$ ft straight tubing

The above ΔP_{BG} is for $2 \times 23 = 46$ ft of straight tubing, thus

$$\Delta P_{bends} = \frac{132 \times 1.16}{46} = 3.33 \text{ psi}$$

The total pressure loss for BG in HE 1 is

$$\Delta P_{T\text{-}BG} = 1.16 + 3.33$$

Total ΔP for BG = 4.49 psi

Thus, BG leaves HE 1 at 31.6 psig (46.3 psia).

D. Dimensions and Weight

Inside dimensions of HE 1 are 2 x 2 ft

Wall thickness, for Hastelloy C, is 3/16"

Length of duct is $3.34 + 2 \times 1 = 5.34$ ft

(includes 1 ft on each end for transition cones)

Duct: 217.7 inch^3/ft x 0.296 lb/inch^3 = 64.44 lb/ft

Tubing (including return bends) = 62 #/bank (for 316-SS)

Thus:

Duct:	5 x 64.44 =	322 lb
Tubing:	23 x 62 =	1,426
	Total	1,748 lb

Total weight of HE 1 = 1,748 lb

Note: The weight is based on use of Hastelloy C as the material of construction, and on incomplete optimization of fins and tubes. Choice of a lighter weight fin and tube material of equivalent strength would reduce the weight proportionally.

VI. Heat Exchangers No. 2 and 3 (HE 2 and HE 3)

These are combined in one duct containing two "coils", one using fuel and the other water. Between the two sections there is an "orifice" which actually is a dam 1" high extending from all walls of the duct. This dam prevents the condensed water in HE 3 from entering the fuel-cooled section (HE 2). At both ends of the water-cooled section there are provisions for draining off the condensed water.

A. General

Ballast gas is on the duct side.

The duct is 1 x 1 ft

Tubing: $OD_t = 0.5"$ wall = 0.01" $ID_t = 0.48"$

Fins: $b_f = 0.25"$ $th_f = 0.035$ $N_f = 8$ fins/inch

Construction: alternate banks in triangular pitch

$N_{t/b1} = 12$ $N_{t/b2} = 11$

$S_T = 1"$ $S_L = 1"$ $V_s = 0.866"$

Other values, as in previous parts of this Appendix:

$A_f = 0.7854$ ft²/ft lin.

$A_o = 0.0943$ ft²/ft lin.

$P_p = 9.44$ ft/ft lin.

$de_d = 0.06$ ft

$a_d = 0.36$ ft²

$G_d = 4{,}450$ lb/(hr)(ft²)

$Y_b = 0.00146$ ft

$r_e - r_b = 0.0208$ ft

$r_e/r_b = 2$

$V_{NF} = 0.0433 \ ft^3$ (per bank)

$S_F = 10.12 \ ft^2$ (per bank)

$D'_{ev} = 0.0171 \ ft$

$(D'_{ev}/S_T)^{0.4} = 0.531$

$(S_L/S_T)^{0.6} = 1.0$

B. Fuel-Cooled Section (HE 2)

1. Heat Transfer

$Q = 1{,}598$ BTU/min = 95,900 BTU/hr

$W_F = 52$ lb/min = 3,120 lb/hr

Hot Fluid (BG)		Cold Fluid (Fuel)	Difference
558°F	Higher temperature	350°F	208°F
340°F	Lower temperature	300°F	40°F
218°F	Difference	50°F	168°F

LMTD = 102°F R = 4.36 S = 0.2 $F_T = 0.92$(9)

$\Delta t = 0.92$ LMTD = $\underline{\underline{94°F}}$

$\Delta t_c/\Delta t_h = 0.192$ $K_c = 0.09$ $F_c = 0.36$(10)

caloric (mean) temperature of BG, $\underline{T_c = 419°F}$

caloric (mean) temperature of fuel, $\underline{t_c = 318°F}$

for fuel at 318°F:

$\mu_F = 0.75$ lb/(ft)(hr)

$\rho_F = 5.78$ lb/gal = 43.2 lb/ft^3

$S_t = 0.7$

$C_p = 0.68$ BTU/(lb)(°F)

$k_F = 0.0732$ BTU/(hr)(ft^2)(°F/ft)

$(c\mu_F/k_F)^{1/3} = 1.91$

$\phi_t = 1$

for BG at 419°F and 46.3 psia

$V_M = 203.6$ ft³/lb-mole

$V_{BG} = 188.8$ cfm = 3.15 cfs

$\rho_{BG} = 0.1415$ lb/ft³

$S_d = 0.00227$; $\phi_s = 1$

$\mu_{BG} = 0.0605$ lb/(ft)(hr)

$C_p = 0.26$ BTU/(lb)(°F)

$k_{BG} = 0.024$ BTU/(hr)(ft²)(°F/ft)

$(c\mu_{BG}/k_{BG})^{1/3} = 0.869$

$\phi_d = 1$

Heat transfer - duct side

$Re_d = 4,420$

$j_f = 40^{(2)}$

$h_f = 13.9$ BTU/(hr)(ft²)(°F)

$h_{dd} = 602$ BTU/(hr)(ft²)(°F)

$h_f' = 13.6$ BTU/(hr)(ft²)(°F)

Corrected coefficient = 13.6 BTU/hr-ft²-°F

Heat transfer - tube side

$A_t = 0.00126$ ft²

$A_{t/b} = 0.015$ ft²/bank

$de_t = 0.04$ ft

$G_t = 208,000$ lb/(hr)(ft²)

$Re_t = 11,100$

$j_t = 42^{(5)}$

$h_t = 147$ BTU/(hr)(ft^2)(°F)

$h_{dt} = 500$ " "

$h'_{it} = 113.6$ " "

$$\boxed{\text{Corrected coefficient} = 113.6 \text{ BTU/hr-ft}^2\text{-°F}}$$

Heat transfer - design U, area and No. of banks

$a_{it} = 0.126$ ft^2/ft lin.

$A_{it/b} = 1.44$ ft^2/bank

$(r_e - r_b)(h'_f/k_t Y_b)^{\frac{1}{2}} = 0.632$ $k_t = 10.1$ BTU/(hr)(ft^2)(°F/ft)

$\Omega = 0.84$(3)

$h'_{fi} = \underline{\underline{81.6}}$ BTU/(hr)(ft^2)(°F)

$U_{Di} = 47.5$ BTU/(hr)(ft^2)(°F)

$$\boxed{\text{Design coefficient} = 47.5 \text{ BTU/hr-ft}^2\text{-°F}}$$

$A_{iT} = \underline{\underline{21.5 \text{ ft}^2}}$

$$\boxed{N_b = 15 \text{ banks}}$$

2. Pressure Drop

Tube side pressure drop (fuel)

Velocity = 1.336 fps

$f = 0.00027$(6)

$\Delta P_t = \underline{\underline{0.12 \text{ psi}}}$

each return bend of 0.5" tubing is equivalent to 2.5 ft of straight tubing. There are 14 return bends and two headers:

$$\Delta P_r = \frac{(14 + 2) \times 2.5 \times 0.12}{15} = \underline{\underline{0.32 \text{ psi}}}$$

$$\boxed{\Delta P_{T\text{-Fuel}} = 0.44 \text{ psi}}$$

Duct side pressure drop (BG)

$R'_{ed} = 1{,}260$

$f = 0.0032^{(2)}$

$L_p = N_b V_s = 15 \times 0.866/12 = 1.087$ ft

$\Delta P_d = \underline{0.02 \text{ psi}}$

Loss due to gradual enlargement (diverging cone) is 8%.

$\Delta P_{Enl.} = 0.08 \times 31.6 = \underline{2.53}$

$$\boxed{\Delta P_{BG\text{-}F \text{ section}} = 2.55 \text{ psi}}$$

Thus, the BG leaves this section with 29.05 psig (43.75 psia).

Pressure Loss in "Orifice"

$\Delta P_o = (\frac{1}{C^2} - 1)\ P_v$

v = velocity BG in duct = V_T/A_d = 3.15 ft^3 per sec/0.36 ft^2

= $\underline{8.74 \text{ ft/sec}}$

P_v = velocity head = $v^2/2\ g$ = $\underline{8.74^2/(2 \times 32.2)}$

= 1.2 ft (air column)

= 0.0012 psi

to obtain $(\frac{1}{C^2} - 1)$

A_1 = area empty duct = 12 x 12 = 144 inch^2

A_2 = area "orifice" = 10 x 10 = 100 inch^2 (1 inch dam)

A_2/A_1 = 100/144 = 0.7

for above value, $(\frac{1}{C^2} - 1) = 0.2$

thus: $\Delta P_o = 0.2 \times 0.0012 = 0.00024$ psi

$$\boxed{\Delta P_o = \text{negligible}}$$

C. Water-Cooled Section (HE 3)

1. Heat Transfer

$Q = 2{,}984$ BTU/min $= 179{,}000$ BTU/hr

$W_w = 23$ lb/min $= 1{,}380$ lb/hr

Hot Fluid (BG)		Cold Fluid (Water)	Difference
340°F	Higher temperature	210°F	130°F
100°F	Lower temperature	80°F	20°F
240°F	Difference	130°F	110°F

LMTD = 59°F R = 1.85 S = 0.5 $F_T = 0.87^{(9)}$

$\Delta t = 0.87 \times 59 = \underline{51°F}$

Average temperatures may be used, thus

$T_c = (340 + 100) \div 2 = 220°F$

$t_c = (210 + 80) \div 2 = 145°F$

For BG at 220°F and 43.75 psia:

$V_M = 158$ ft^3/lb-mole

$V_{BC} = 147.5$ cfm $= 2.46$ cfs

$\mu_{BG} = 0.051$ lb/(ft)(hr)

$\rho_{BG} = 0.181$ lb/ft^3

$S_d = 0.0029$

$c_p = 0.25$ BTU/(lb)(°F)

$k_{BG} = 0.019$ BTU/(hr)(ft^2)(°F/ft)

$(c\mu_{BG}/k_{BG})^{1/3} = 0.876$

$\phi_d = 1$

For water at 145°F:

$\mu_w = 1.069$ lb/(ft)(hr)

$\rho_w = 61.3$ lb/ft^3

$S_t = 1$

$C_p = 1$ BTU/(lb)(°F)

$k_w = 0.374$ BTU/(hr)(ft^2)(°F/ft)

$(c\mu_w/k_w)^{1/3} = 1.428$

$\phi_t = 1$

In this section the tubing is aluminum. It is of the same $OD_t = 0.5$ inch as before, but wall $\cong$ 0.02" and consequently $ID_t = 0.46$".

Heat transfer - duct side

$Re_d = 5{,}240$

$j_f = 45^{(2)}$

$h_f = 12.5$ BTU/(hr)(ft^2)(°F)

$h'_f = 12.25$ BTU/(hr)(ft^2)(°F)

$\boxed{\text{Corrected coefficient} = 12.25 \text{ BTU/hr-ft}^2\text{-°F}}$

Heat transfer - tube side

$de_t = 0.0383$ ft

$a_t = 0.00115$ ft^2

$A_{t/b} = 0.0139$ ft^2/bank

$G_t = 99{,}600$ lb/(ft^2)(hr)

$Re_t = 3{,}500$

$j_t = 14^{(5)}$

$h_t = 195$ BTU/(hr)(ft^2)(°F)

$h_{dt} = 500$ BTU/(hr)(ft^2)(°F)

$h'_{it} = 140.3$ BTU/(hr)(ft^2)(°F)

$\boxed{\text{Corrected coefficient} = 140.3 \text{ BTU/hr-ft}^2\text{-°F}}$

Heat transfer - design U, area and No. of banks

$a_{it} = 0.120$ ft^2/ft lin.

$A_{it/b} = 1.385$ ft^2/bank

$(r_e - r_b)\ (h_f'/k_t Y_b) = 0.175$ $\qquad k_t = 118$ BTU/(hr)(ft^2)(°F/ft)

$\Omega = 0.98$(3)

$h'_{fi} = \underline{87.9}$ BTU/(hr)(ft^2)(°F)

$U_{Di} = \underline{\underline{54}}$ BTU/(hr)(ft^2)(°F)

$A_{iT} = \underline{\underline{65 \text{ ft}^2}}$

$\boxed{N_b = 47 \text{ banks}}$

2. Pressure Drop

Tube side pressure drop (water)

Velocity = 0.452 ft/sec

$f = 0.00038$(6)

$\Delta P_t = \underline{\underline{0.09 \text{ psi}}}$

$\Delta P_r = \underline{\underline{0.23 \text{ psi}}}$

$\boxed{\Delta P_T \text{ -Water} = 0.32 \text{ psi}}$

Duct side pressure drop (BG)

$R'_{ed} = 1{,}500$

$f = 0.00315$(2)

$L_p = 3.39$ ft

$\Delta P_d = \underline{\underline{0.05 \text{ psi}}}$

In this section, there is also the pressure loss due to condensation of water. Average pressure in this section is 29 psig (43.7 psia). The total of gases in BG is 0.8615 x 1.0765 = 0.9274 mole. The amount of water condensed in HE 3 is 1.272 ÷ 18 = 0.07067 moles (see Figure 37).

$$\Delta P_{condensation} = (0.07067 \times 43.7) \div 0.9274 = \underline{3.33 \text{ psi}}$$

The pressure at which BG reaches the converging cone is 29-3.33 = 25.67 psig (40.37 psia). The loss due to the gradual contraction there is 2%.

$$\Delta P_{contr.} = 0.02 \times 25.67 = \underline{0.51 \text{ psi}}$$

Consequently, the total pressure drop in HE 3 is

$$\boxed{\Delta P_{BG\text{-}W\ section} = 3.89 \text{ psi}}$$

Thus, BG leaves HE 3 with 25.16 psig (39.86 psia)

D. Size and Weight

HE 2 Section (fuel) - all 316-SS

Duct wall is 3/16" thick and its inside dimensions are 12 x 12 inches. The length of this section including diverging cone is 1.5 ft.

Duct: 109.7 $inch^3/ft$ x 0.29 $lb/inch^3$ = 31.8 lb/ft

Tubing: 25 $inch^3/bank$ x 0.29 $lb/inch^3$ = 7.27 lb/bank

Weight duct = 1.5 x 31.8 = 48 lb

Weight tubing = 15 x 7.27 = 109

Total 157 lb

$$\boxed{\text{Total Weight, HE 2} = 157 \text{ lb}}$$

HE 3 Section (water) - all aluminum

Duct wall is 0.35" thick, and the length of this section including the converging cone is 3.8 ft.

Duct = 207.5 $inch^3/ft$ x 0.099 $lb/inch^3$ = 20.5 lb/ft

Tubing = 27.3 $inch^3/bank$ x 0.099 $lb/inch^3$ = 2.7 lb/bank

Weight duct = 3.8 x 20.5 = 78 lb

Weight tubing = 47 x 2.7 = 127

Total 205 lb

Total weight, HE 3 = 205 lb

E. Summary

Total length of combined HE 2 and HE 3	5.3 ft
Total weight " " "	362 lb
Total pressure drop in " "	6.44 psi

VII. Drier

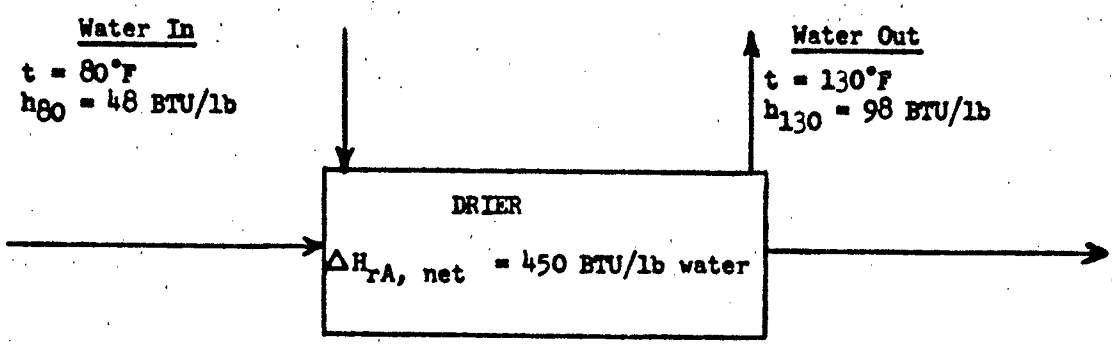
A. General

Both fluids enter at temperatures below their exit temperatures, and the available cooling water temperature is only 20°F below that of the entering BG. Therefore, a cocurrent flow is chosen. The choice of cross-section is based on consideration of several factors:

- pressure drop
- velocity of gas in empty drier
- the desired number of cooling tubing banks (it is preferable to have a smaller cross-section and a larger number of banks at a closer spacing, for the same heat transfer surface, to maintain the entire bed as cool as possible).

Two drying agents are used. Most of the water will be removed by the $CaCl_2$ (heat of absorption = 1,500 BTU/#, net heat = 450 BTU/#) because of its high capacity. Weights of agents are calculated first based on efficiency, then increased if necessary to provide 50 hours of service between regenerations. This is in line with the target for catalyst performance.

Contrary to the cooling scheme used as the basis for the material and heat balances given in Figure 37 a revision providing an increased cooling water requirement but a net reduction in weight is used, as follows:



Saturated BG in

t = 100°F
$W_{Dry\ BG}$ = 24.39 lb/min.
$W_{H_2O\ Vapor}$ = 1.054 lb/min.
$Q_{SBG,\ 100}$ = 1,669 BTU/min.

Dry BG out

t = 150°F
$h_{Dry\ BG,\ 150}$ = 27.8 BTU/lb
$W_{Dry\ BG}$ = 24.39 lb/min.
$Q_{Dry\ BG,\ 150}$ = 678 BTU/lb.

$$Q_{in} = Q_{SBG,100} + W_{H_2O\ vapor} \times \Delta H_{rA,\ net}$$

$$1{,}669 + 1.054 \times 450 = 2{,}143 \text{ BTU/min}$$

Figure F-7. Revised Cooling Water Requirements

$$Q_{out} = Q_{dry\ BG,\ 150} = W_{dry\ BG} \times h_{dry\ BG,\ 150}$$

$$= 24.39 \times 27.8 = 678\ \text{BTU/min}$$

$$\Delta Q_{drier} = Q_{in} - Q_{out}$$

$$= 2{,}143 - 678 = 1{,}465\ \text{BTU/min}$$

$$W_{cooling\ H_2O} = \frac{\Delta Q_{drier}}{h_{130} - h_{80}}$$

$$= \frac{1{,}465}{98-48} = \underline{\underline{29.3\ \text{lb/min}}}$$

This supply of 130°F water may be introduced at the appropriate point in HE 3, thus reducing either the overall water requirement or the transfer surface area, but such is not considered in the present study.

If corresponding changes are calculated for average flight conditions (Figure 36), the drier cooling water flow is increased to 18.9 lb/min or 122 lb/flight, and total cooling water per flight (including 10% contingency) becomes 266 lb.

B. Data for Calculations

Saturated BG reaches drier at 25.1 psig (39.8 psia)

Weight SBG = 25.448 lb/min = 1527 lb/hr

Weight of water removed = 1.054 lb/min

Weight of dry BG = 24.39 lb/min = 1,464 lb/hr

Pressure in fuel tank = 0.84 psig (15.54 psia)

$Q_{to\ remove}$ = 1,465 BTU/min = 87,900 BTU/hr

$W_{cooling\ H_2O}$ = 29.3 lb/min = 1,758 lb/hr

$\mu_{AIR,100}$ = 0.019 cps = 0.046 lb/(ft)(hr)

$\mu_{AIR,150} \doteq \mu_{dry\ BG,150}$ = 0.0202 cps = $\underline{0.0489\ \text{lb/(ft)(hr)}}$

$\mu_{H_2O\ Vapor,100}$ = 0.011 cps = 0.0266 lb/(ft)(hr)

$\mu_{SBG,100} = (24.394 \times 0.019 + 1.054 \times 0.011) \div 25.448$

$= 0.01867$ cps = 0.0452 lb/(ft)(hr)

$\mu_{Avg} = (\mu_{SBG,100} + \mu_{AIR,150}) \div 2 = 0.5\ (0.01867 + 0.0202)$

$= 0.01945$ cps = 0.047 lb/(ft)(hr)

$$V_{M,IN} = \frac{14.7}{492} \times 359 \times \frac{(100 + 460)}{37.8^{*}} = 158.9\ ft^3/lb\text{-}mole$$

$V_{SBG,IN}$ = 158.9 x 0.8615 mols air/min x 0.9945 mols sat. BG/mol air

= 136.2 ft^3/min

$\rho_{SBG,IN}$ = 25.448/136.2 = 0.1869 lb/ft^3

$S_{SBG,IN}$ = ρ/62.4 = 0.00299

$$V_{M,OUT} = \frac{14.7}{492} \times 359 \times \frac{(150 + 460)}{33.25^{**}} = 196.8\ ft^3/lb\text{-}mole$$

$V_{Dry\ BG}$ = 196.8 x (0.8615 x 0.9265) = 157.1 ft^3/min

$S_{Dry\ BG}$ = 0.00249 $\rho_{Dry\ BG}$ = 24.394/157.1 = 0.1553 lb/ft^3

$$V_{BG,Avg} = \frac{V_{Dry\ BG} - V_{S\ BG}}{2.3 \log \frac{V_{Dry\ BG}}{V_{SBG}}}$$

$$= \frac{157.1 - 136.2}{2.3 \log \frac{157.1}{136.2}} = \underline{146.6\ ft^3/min}$$

$W_{BG,Avg}$ = (25.448 + 24.394) ÷ 2 = 24.921 lb/min

= 1495.3 lb/hr

$\rho_{BG,Avg}$ = 24.921/146.6 = <u>0.17 lb/ft^3</u>

S_{Avg} = 0.00272

C. Amounts of Drying Agent

(1) Calcium Chloride Section

Basis of 150°F and ~14.7 psia

efficiency = 4600 ppm (equilibrium conc. ÷ 0.95)
at 1900 hr^{-1} SV

capacity = 30%, or 0.30 lb H_2O/lb agent

*Pressure in the drier after substracting gradual enlargement loss.

**Approx. P when gas leaves bed, based on pressure losses.

Gas flow at design conditions = $V_{SBG, ave}$

$= 147\ ft^3/min$

Bed volume = volume for 1900 hr^{-1} SV

$$= \frac{147\ ft^3}{min} \Big| \frac{60\ min}{hr} \Big| \frac{hr}{1900} = 4.65\ ft^3$$

Weight of $CaCl_2$

bulk density = 51 lb/ft^3

weight = 4.65 x 51 = 237 lb

Concentration of water entering $CaCl_2$

N_2	79 lb-moles
CO_2	13.5 lb-moles
Fuel Vapor	0.15 lb-moles
H_2O	6.80 lb-moles

$$H_2O = 15 \Big| \frac{1.054 \text{ to drier}}{1.054 + .272 \text{ (condensed)}} = 6.80$$

% H_2O v/v = (6.8 ÷ 99.45) x 100 = 6.84

= 68,400 ppm

Concentration of water leaving $CaCl_2$ = 4600 ppm

% removed = 93.3%

Water removed during <u>average</u> flight

use average flight data, not design data

Removed in 1 flight by complete drier = 6.07 lb

Removed by $CaCl_2$ = 6.07 x 0.933 = 5.67 lb

Capacity of $CaCl_2$ (80% of estimated value)

237 lb x 0.30 lb/lb x 0.80 = 53 lb water

Flights without regeneration

53 ÷ 5.67 = 9.34 flights

average flight time = 3 hours

Hours without regeneration = 28 hours

To attain 50 hours, increase by proportion

$237 \times 50 \div 28 = 424$ lb $CaCl_2$

Volume and length

Superficial velocity should be in range 50-100 fpm

$147\ ft^3/min \div 75\ ft/min = 1.96\ ft^2$

$\therefore$ Choose duct having 2 ft^2 transverse area

Volume = 424 lb $\div$ 51 lb/ft^3 = 8.31 ft^3

Length = $8.31 \div 2 = 4.16$ ft

Weight of $CaCl_2$ = 424 lb
Volume of $CaCl_2$ = 8.31 ft^3
Dimensions = 1 ft x 2 ft x 4.16 ft long

(2) Calcium Sulfate Section

(on basis of 150°F and approx. 14.7 psia)

efficiency = 100 ppm at 400 hr^{-1} SV

capacity = 1.0%, or 0.01 lb H_2O/lb agent

Gas flow at design conditions

assume same as in $CaCl_2$ = 147 ft^3/min

Bed volume = volume for 400 hr^{-1} SV

$$\frac{147\ ft^3}{min}\left|\frac{60\ min}{hr}\right|\frac{hr}{400} = 22.1\ ft^3$$

Weight of $CaSO_4$ = volume x bulk density

= 22.1 ft^3 x 75 lb/ft^3

= 1660 lb

Water removed in average flight of 3 hours

total removed in drier - amount removed in $CaCl_2$

6.07 lb x (1.00 - 0.933) = 0.407 lb

Capacity of $CaSO_4$ (use 80% of estimated value)

1660 lb x 0.01 lb/lb x 0.80 = 13.3 lb

Flights without regeneration = 13.3 ÷ 0.407

= 32.6

Hours without regeneration = 3 x 32.6 = 97.8 hr

$\boxed{\text{Hours without regeneration} = 98}$

Summary of Drying Agents

Agent	$CaCl_2$	$CaSO_4$	Combined
volume, ft^3	8.31	22.1	30.4
weight, lb	424	1660	2084
length, ft	4.2	11.0	15.2
hours (no regen.)	50	98	50

(3) Recalculation to Save Weight By Utilizing Excess Capacity in $CaSO_4$

Excess capacity = 58-50 = 48 hr

Water equivalent

48 hr x 0.407 lb/flight ÷ 3 hr/flight

6.53 lb water capacity in excess

Adding to $CaCl_2$ capacity (53 lb water on 237 lb agent)

53 + 6.53 ÷ 5.67 = 10.5 flights

10.5 x 3 = 31.5 flight-hours

Revised deficiency = 50-31.5 = 18.5 hours

$$\frac{18.5 \text{ hr}}{} \Big| \frac{237 \text{ lb}}{28 \text{ hr}} = 156 \text{ lb } CaCl_2$$

Total quantity $CaCl_2$

237 + 156 = 393 lb

393 ÷ 51 = 7.72 ft^3

Revised Summary of Drying Agents $CaCl_2$ and $CaSO_4$

Agent	$CaCl_2$	$CaSO_4$	Combined
volume, ft^3	7.72	22.1	29.82
weight, lb	393	1660	2053
length, ft	3.86	11.0	14.86
hours (no regen.)	50	50	50

(4) Use of Zeolite as High-Efficiency Agent

Bases: 10 ppm efficiency at following conditions

150°F and ~ 1 atm

$L/D > 1$

linear velocity < 100 ft/min

Norton's H-Zeolon, 1/16" dia.

Capacity 0.015 lbs/lb agent

Velocity in 1 ft x 2 ft duct

$147\ ft^3/min \div 2\ ft^2 = \underline{\underline{73.5\ ft/min}}$

Weight of agent (using 80% of stated capacity)

$$= \frac{lb\ H_2O}{flight} \Big| \frac{flight}{hours} \Big| \frac{hours}{continuous\ run} \Big| \frac{}{capacity \times 0.8}$$

$$= \frac{0.407}{3} \Big| \frac{50}{0.015} \Big| \frac{}{0.8} = \underline{\underline{565\ lb}}$$

Volume of agent ($38.5\ lb/ft^3$ bulk density)

$565\ lb \div 38.5\ lb/ft^3 = \underline{\underline{14.7\ ft^3}}$

Length of bed and L/D

$L = 14.7\ ft^3 \div 2\ ft^2 = 7.35\ ft$

equivalent circle diameter

$D = (8 \div \pi)^{0.5} = 1.6\ ft$

$L/D = 7.35 \div 1.6 = \underline{\underline{4.59}}$

Summary of Drying Agents $CaCl_2$ and Zeolite

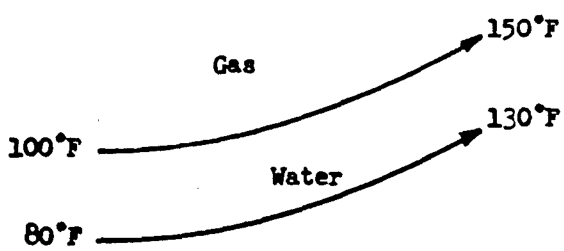
Agent	$CaCl_2$	Zeolite	Combined
volume, ft^3	8.31	14.7	23.0
weight, lb	424	565	989
length, ft	4.2	7.35	11.55
hours (no regen.)	50	50	50

D. Cooling

$Q = 87{,}900$ BTU/hr

$W_{BG,Avg} = 1495$ lb/hr

$W_{cooling\ H_2O} = 1758$ lb/hr



Hot Fluid		Cold Fluid	Difference
150°F	Higher temperature	130°F	20°F
100°F	Lower temperature	80°F	20°F
50°F	Difference	50°F	0

$\Delta t_{mean} = 20°F$

A constant temperature difference of 20°F exists throughout the entire drier. Therefore it is assumed that no correction of Δt is necessary.

Caloric Temperatures

Arithmetical averages are sufficient, thus for BG $T_c = 125°F$ for cooling water $t_c = 105°F$.

Consequently:

For BG Avg at 125°F

$\mu_{BG,Avg} = 0.047$ lb/(ft)(hr)

$\rho_{BG,Avg} = 0.17$ lb/ft^3

$S_d = S_{Avg} = 0.00272$

$C_p = 0.248$ BTU/(lb)(°F)

$k_{BG,Avg} = k_{AIR} = 0.0163$ BTU/(hr)(ft^2)(°F/ft)

$\phi_d = 1$

$(c\mu_{BG}/k_{BG})^{1/3} = 0.894$

For cooling water at 105°F

$\mu = 1.67$ lb/(ft)(hr)

$\rho = 61.93$ lb/ft^3

$S_t = 1$

$C_p = 1$ BTU/(lb)(°F)

$k = 0.363$ BTU/(hr)(ft^2)(°F/ft)

$\phi_t = 1$

$(c\mu/k)^{1/3} = 1.663$

For tube material (aluminum) at 115°F

$k_t = 117.2$ BTU/(hr)(ft^2)(°F/ft)

Tubing and Duct

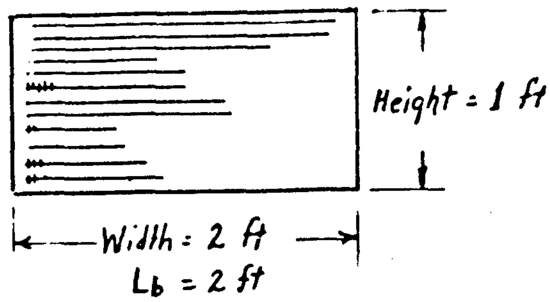


Figure F-8. Drier Tubing and Duct

Aluminum tubing:

$OD_t = 0.5''$ wall $= 0.02''$ $ID_t = 0.46''$

Fins: $b_f = 0.25''$ $th_f = 0.035''$ $N_f = 8$ fins/inch lin.

Arrangement: 12 tubes per bank in square pitch

$A_f = 0.7854$ ft^2/ft lin.

$A_o = 0.0943$ ft^2/ft lin.

$P_p = 9.44$ ft/ft lin.

$de_d = 0.06$ ft

$a_t = 0.0115$ ft^2

$A_{t/b} = 0.01385$ ft^2/bank

$de_t = 0.0383$ ft

$a_{it} = 0.120$ ft^2/ft lin.

$A_{it/b} = 2 \times 12 \times A_{it} = 2.89$ ft^2/bank

$Y_b = 0.00146$ ft

$r_e - r_b = 0.0208$ ft

$r_e/r_b = 2$

Heat transfer - duct side

$$a_s = 2 \times 12 \left[1 \times 12 - 12\,(0.5 + 2 \times 0.035 \times 0.25 \times 8)\right]$$

$$= 103.7 \text{ inch}^2 = 0.72 \text{ ft}^2$$

G_d = 1495/0.72 = 2,077 lb/(ft^2)(hr)

Re_d = 0.06 x 2,077/0.047 = 2,700

j_f = 28 [2]

$h_f = 28 \times \frac{0.0163}{0.06} \times 0.894 \times 1 = 6.8$ BTU/(hr)(ft^2)(°F)

h_d = Avg SBG and Dry BG = (602 + 500) ÷ 2 = 551 BTU/(hr)(ft^2)(°F)

$h'_f = \frac{551 \times 6.8}{551 + 6.8}$ = 6.7 BTU/(hr)(ft^2)(°F)

Heat transfer - tube side

G_t = 1758/0.01385 = 126,950 lb/(ft^2)(hr)

Re_t = 0.0383 x 126,950/1.67 = 2,910

v = velocity of water in tube

$$v = \frac{G_t}{3600\rho} = \frac{126,950}{3600 \times 61.93} = 0.57 \text{ ft/sec}$$

h_i = 215 x 1.055 = 226.8 BTU/(hr)(ft^2)(°F) [4]

h_{di} = 500 BTU/(hr)(ft^2)(°F)

$h'_{di} = \frac{500 \times 266.8}{500 + 266.8}$ = 146 BTU/(hr)(ft^2)(°F)

Heat transfer - design U, area and number of banks

$$(r_e - r_b)\,(h'_f/k_t Y_b)^{0.5} = 0.0208\,(6.7/117.2 \times 0.00146)^{\frac{1}{2}}$$

$$= 0.132$$

$\Omega = 0.99$ [3]

h'_{fi} = (0.99 x 0.7854 + 0.09425) (6.7/0.120)

= 48.45 BTU/(hr)(ft^2)(°F)

$U_{di} = \frac{48.45 \times 146}{48.45 + 146}$

Overall design coefficient = 36.4 BTU/(hr)(ft^2)(°F)

$$A_{iT} = \frac{87,900}{36.4 \times 20} = \underline{120.8\ ft^2}$$

$$N_b = 120.8/2.89$$

$$\boxed{\text{Number of banks} = 42}$$

The larger portion of water is removed by $CaCl_2$, consequently most of the absorption heat will be generated there. Thus, most of the cooling has to take place there.

Fraction of water removed by $CaCl_2$ = 0.933, and this fraction of heat is removed in $CaCl_2$ section.

$$N_b \text{ in } CaCl_2 = 42 \times 0.933 = 39 \text{ banks}$$

$$\left.\begin{matrix} N_b \text{ in } CaSO_4 \\ \text{or Zeolite} \end{matrix}\right\} = 42-39 = 3 \text{ banks}$$

Each of the two desiccants is in a separate section measuring 1' x 2' in rectangular cross-section. The two sections are separated by a screen (see Figure F-9).

Volume occupied by the tubing

$V_{t/b}$ = volume occupied by the tubing of one bank

$$V_{t/b} = \frac{\pi}{4} L_b \left[OD_t^2 + (OD^2_f - OD^2_t)\ th_f\ N_f\right] N_{t/b}$$

$$= \frac{\pi}{4}\ 2 \times 12 \left[0.5^2 + (1^2 - 0.5^2)\ 0.035 \times 8\right] 12$$

$$= 104 \text{ inch}^3/\text{bank} = \underline{0.0602\ ft^3/\text{bank}}$$

Volume and Dimensions of Sections

The transverse area is 2 ft^2, and the side wall dimensions are 2 ft x 1 ft. The volume in each section accommodates both agent and cooling tubes. Using the $CaCl_2$ and zeolite combination:

For $CaCl_2$ section

Volume agent = 8.31 ft^3

Volume 39 banks of tubes = 39 x 0.0602 = 2.35 ft^3

Combined volume = 10.66 ft^3

Length = 10.66 ÷ 2 = 5.33 ft

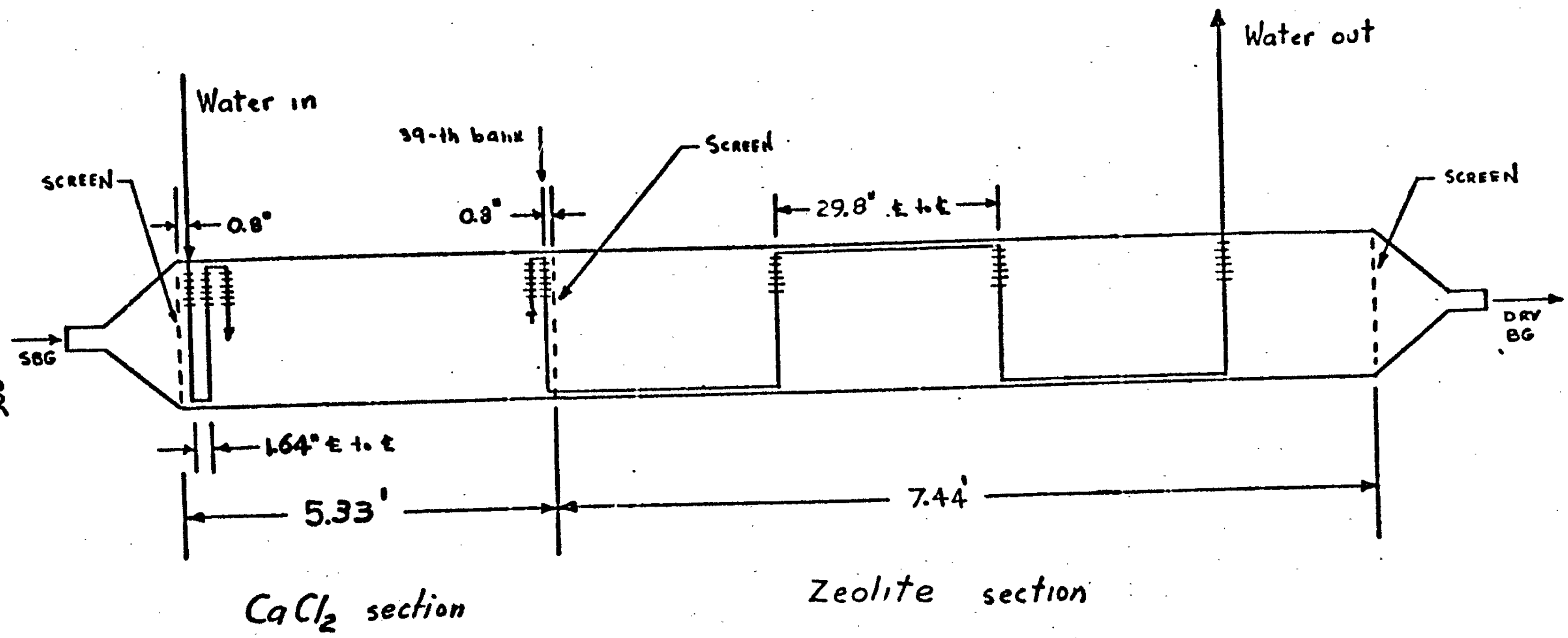


Figure F-9. Gas Drier

Spacing of banks = $(5.33 \div 39) \times 12 = 1.64$ inch

℄ to ℄

For Zeolite Section

Volume agent = $14.7\ ft^3$

Volume of 3 banks of tubes = $0.18\ ft^3$

Combined volume = $14.88\ ft^3$

Length = 7.44 ft

Spacing of banks = $(7.44 \div 3) \times 12 = 29.8$ inch

℄ to ℄

Note: This spacing is obviously not satisfactory for actual design; a detailed study would provide for a more uniform distribution of the heat transfer surface.

E. Pressure Losses

Pressure loss inside tubing

$v = 0.57$ ft/sec $\qquad Re_t = 2{,}910$

$f = 0.00039$[6]

$\Delta P_t = \underline{0.27\ psi}$

$\Delta P_r = (41 + 2) \times 2.5 \times 0.27/(42 \times 2) = \underline{0.35\ psi}$

Total $\Delta P = 0.27 + 0.35$

$$\boxed{\Delta P_{T\text{-water}} = 0.62\ psi}$$

Pressure loss on the duct side

The pressure loss in the duct side of the drier is the sum of the following pressure losses:

a) Pressure loss due to gradual enlargement in the diverging cone ($= \Delta P_1$) which is equal to 8% of the gage pressure in the incoming gas.

b) Pressure loss due to friction through the packed bed ($= \Delta P_2$).

c) Pressure loss due to banks of cooling tubing ($= \Delta P_3$)

d) Pressure loss due to removal of water ($= \Delta P_4$), that is, loss of fraction of gas volume and consequently its partial pressure.

e) Pressure loss due to gradual contraction in the converging cone ($= \Delta P_5$) which is equal to 2% of pressure (gage) in the gas reaching that cone.

The pressure losses in the three screens are negligible.

Due to the presence of cooling tubes in the bed, the volume of the bed increases from 23.0 ft^3 to 23.0 + 2.35 + 0.18 = 25.53 and the length from 23.0 ft $\div$ 2 $\simeq$ 11.5 ft to 25.53 $\div$ 2 = 12.76 ft. Therefore, ΔP_2 has to be determined for this new length of 12.76 ft. A combined ΔP_{2-3} loss is calculated for L = 12.76 ft.

(1) $\Delta P_1 = 0.08\ P_{in}$

$= 0.08 \times 25.1 = 2$ psi

(2) ΔP_{2-3} is determined employing the method of Aluminum Company of America[11], for 8-14 mesh granules. Both zeolite extrudates and the $CaCl_2$ granules are in this size range.

$$R'_e = \frac{K_1\ G_o}{\mu}$$

R'_e = modified Reynolds number

K_1 = 0.264 ft

G_o = superficial mass velocity based on empty drier cross-sectional area, lb/(ft^2) (hr)

$A_d = 2\ ft^2$

$W = W_{BG\ Avg} = (1{,}527 + 1{,}464) \div 2 = 1{,}495$ lb/hr

$G_o = \frac{W}{A_d} = \frac{1495}{2} = 747.5$ lb/(ft^2)(hr)

$\mu = \mu_{BG\ Avg} = 0.047$ lb/(ft)(hr)

$$R'_e = \frac{0.264 \times 747.5}{0.047} = 4{,}200$$

$(f/F_f) = 0.0425$ from plot[11]

$$\Delta P_{2-3} = \frac{(f/F_f)\ G_o^2\ L}{K_2\ \rho\ 144} \text{ psi}$$

(f/F_f) = modified friction factor

L = depth of bed, ft

$K_2 = 4880\ ft^2/hr^2$

L = 12.76 ft

$\rho = \rho_{BG\ Avg} = 0.17\ lb/ft^3$

$$\Delta P_{2-3} = \frac{0.0425 \times (747.5)^2 \times 12.76}{4880 \times 0.17 \times 144} = \underline{2.53 \text{ psi}}$$

(3) ΔP_4 is determined assuming it takes place at the midpoint of the $CaCl_2$ section, that is, after the gas loses pressure through 3 ft of bed, which is 0.6 psi. Thus, the pressure of the gas at this assumed point is:

$$\left[P_{in} - (\Delta P_1 + 0.6)\right] = 25.1 - (2 + 0.6)$$

$$= 22.5 \text{ psig} \quad \text{or} \quad \underline{37.2 \text{ psia}}$$

The total number of lb-moles of water vapor and gases in saturated BG is

$$0.862 \times 0.994 = 0.857 \text{ lb-moles/min}$$

and that of the absorbed water is

$$1.054 \text{ lb/min} \div 18 \text{ lb/lb-mole} = 0.0586 \text{ lb-moles/min}$$

Thus

$$\Delta P_4 = (0.0586 \times 37.2) \div 0.857 = \underline{2.54 \text{ psi}}$$

(4) The pressure at which the gas reaches the converging cone is

$$\left[P_{in} - (\Delta P_1 + \Delta P_{2-3} + \Delta P_4)\right] = 25.1 - (2 + 2.53 + 2.54)$$

$$p_c = 18.0 \text{ psig}$$

and

$$\Delta P_5 = 0.02 \, p_c$$

$$= 0.02 \times 18.0 = \underline{0.36 \text{ psi}}$$

(5) Total pressure loss in the drier

$$\Delta P_T = \Sigma \Delta P_i$$

$$= 2 + 2.53 + 2.54 + 0.36$$

$$\boxed{\Delta P_T = 7.43 \text{ psi}}$$

Consequently, dry BG leaves the gas drier at 25.1-7.4 = 17.7 psig (32.4 psia).

The above pressure of 31.4 psia is more than double the assumed pressure in the fuel tank (15.54 psia). However, the following pressure losses have not been included: (a) losses in connecting pipes, because we do not know how long they are and how many bends there will be, (b) losses in control valves and instruments through which the gas will flow. Also, when calculating expansion and contraction losses we assumed these would be gradual processes. Because of space considerations, it may not be possible to accommodate the transition pieces which provide gradual expansion or contraction, in which case the losses would be larger than those shown.

Nevertheless, there appears to be a sufficient margin for operation of a pressure regulator at some position in the subsystem.

G. Gas Filter

The ballast gas passes through a filter before entering the fuel tanks so as to prevent carryover of dust from catalyst or drying agent. The filter is of glass fiber mat construction and is probably best located at the exit end of the drier. It must be accessible for periodic cleaning and replacement.

The pressure loss through this type of filter is measured in inches of water, and is neglected. An allowance of 3 pounds is made for the weight of the filter and housing.

H. Cooling Water Tank

As stated above, the amount of water per flight is 266 lb. This water occupies a volume of 4.3 ft^3. An aluminum or plastic tank may be used. A variable speed pump is necessary to deliver the water to the drier, to HE 3, and subsequently to the combustor.

The weight of the tank includes resistance heaters with a simple control circuit to prevent freezing of the water. Thus

Weight of tank	24 lb
Weight of water	266
Weight of pump (est.)	16
	306 lb

I. Weight Summary

The drier is made entirely of aluminum. The outer wall is assumed to be 1/8-inch thick, because of the weight of the desiccants. It is possible that with properly situated reinforcements it may be thinner. The volume of the material in the outer wall is 1,560 $inch^3$, thus the weight of outer wall is

$$1,560 \times 0.099 = \underline{154 \text{ lb}}$$

The tubing material volume per bank is 56.9 $inch^3$ and the weight of all banks of tubing is

$$56.9 \times 0.099 \times 42 = \underline{237\ lb}$$

Thus

Weight of screens	10 lb	
Weight of outer wall	154	
Weight of cooling tubes	237	
Weight of desiccants	989	
Total	1,390 lb	

Drier Weight = 1,390 lb

The aggregate weight of the drier and auxiliary equipment is as follows:

Drier with filter	1,393 lb
Water tank	306
	1,699 lb

Total weight, drier and auxiliaries = 1,699 lb

REFERENCES

1. Kern, Process Heat Transfer, p 525, McGraw-Hill Book Company, New York, 1950

2. Kern, ibid, pp 555, 838

3. Kern, ibid, p 542

4. Kern, ibid, p 835

5. Kern, ibid, p 834

6. Kern, ibid, p 836

7. Kern, ibid, p 53

8. Kern, ibid, p 839

9. Kern, ibid, p 549

10. Kern, ibid, p 827

11. Calculating Pressure Drop Through Packed Beds, Publication GB4A, Aluminum Company of America, Pittsburgh, Pennsylvania, 1960

APPENDIX G

DESIGN CALCULATIONS, EQUIPMENT FOR SST FLIGHT PLAN NO. 2

APPENDIX G

DESIGN CALCULATION, EQUIPMENT FOR SST FLIGHT PLAN NO. 2

I. Assumptions and Conditions

As stated in text, Section V-6-m.

II. Method

A. General

Weights of components are determined by application of scaling and other factors to the weights of corresponding components in Flight Plan No. 1. In one instance (the segmented combustor), weight is obtained using as a basis the design of the combustor for the Tactical Aircraft, Case Analysis II. No effort is made to generate designs and dimensions expressly for this situation. No checks are made for pressure drop through the subsystem.

B. Scaleup Factors

In cases where viscosity and cross-section differ from one situation to the other, a heat load factor, F_Q, and a Reynolds number factor, F_{Re}, can be used. The former parameter is a direct proportionality

$$F_Q = \frac{Q_{FP2}}{Q_{FP1}}$$

and provides a correction in equipment weight according to the change in heat load. The second parameter reflects the change in heat transfer coefficient and surface area relating to Reynolds number:

$$F_{Re} = \frac{Re_{FP1}}{Re_{FP2}}$$

and operates inversely.

In the event there is <u>no</u> change in viscosity or cross-section, mass velocity increases with the gravimetric flow rate and causes a proportional increase in heat transfer coefficient and reduction in transfer surface. Thus, one can make use of a weight factor, F_w, defined as

$$F_w = \frac{W_{FP1}}{W_{FP2}}$$

Finally, if the cross-section changes but the viscosity remains constant, the mass velocity factor, F_G,

$$F_G = \frac{G_{FP1}}{G_{FP2}}$$

is applied as a correction representing the change in heat transfer surface.

With the exception of F_Q, the above factors are based on the assumption that surface (∴ weight) is inversely proportional to Reynolds number. This is approximately but not exactly the case. The actual relationship is logarithmic.

Another scaleup factor used where appropriate is the temperature difference factor, $F_{\Delta t}$. The heat transfer surface is taken as inversely proportional to the ratio of Δt or LMTD values, FP No. 2/FP No. 1.

III. Vaporization Chamber

$$\text{Fuel to vaporize} = 6.64 \text{ lb/min} \simeq 60 \text{ gph at } 60°F$$

$$\simeq 73 \text{ gph at } 425°F$$

Spraying Systems Company spray set-up No. 42 is selected. This nozzle requires a fuel pressure of 60 psig (thus a fuel booster pump is necessary) and 4.6 scfm of air to atomize the necessary amount of fuel. The spray cone angle is 22° and its minimum length is 46 inch. This gives us the following vaporization chamber (316-SS):

OD	=	20 inch
Wall thickness	=	0.25 inch
Overall length	=	6.55 ft
Weight	=	202 lb

Piping

Because the flow rates are larger, pipes of larger diameter are employed. It is estimated that their weight will be three times that of SST FP No. 1, thus

$$3 \times (19.3 + 13.6) \simeq 100 \text{ lb}$$

Heater

Although it is necessary only at the start of the flight, as a precaution the capacity and weight are increased to double the values used in FP No. 1, thus:

$$2 \times 12 = 24 \text{ lb}$$

Total weight of VC, piping and heater

Supply and outlet piping	100 lb
Spray nozzle	1
Vaporization chamber	202
Heater	24
Reaction fuel booster pump	10
Total	337 lb

IV. Combustor

$$W_{BG} = 5{,}852 \text{ lb/hr}$$

$$Q_R = 4{,}382{,}000 \text{ BTU/hr}$$

$$W_{H_2O\ cool} = 9{,}500 \text{ lb/hr}$$

The differences in LMTD, Δt, viscosity and other physical constants are, from a practical standpoint, negligible.

A. Scale-Up of Radial Reactor

$$F_Q = \frac{4{,}382{,}000}{1{,}583{,}000} = 2.77 \text{ (more heat to be transferred)}$$

Assuming that the length of the combustor is 2.77 times the length of the radial combustor used for SST FP No. 1, and that the tube size, number and arrangement is not changed (therefore the diameter remains unchanged), we get

$$a_s = 7.625 \times 2.77 = 21.1 \text{ ft}^2$$

and consequently

$$G_s = \frac{5852}{21.1} = 280 \text{ (will give larger } R_e \text{ and therefore larger } h_f)$$

hence

$$F_G = \frac{210}{280} = 0.75$$

Thus, the scale-up factor F_{SC-U} is:

$$F_{SC-U} = 2.77 \times 0.75 = \underline{2.08}$$

Therefore, the weight of the combustor will be 2.08 times the weight of the combustor used for SST FP No. 1:

$$2.08 \times 694 = \underline{\underline{1{,}444 \text{ lb}}}$$

B. Design of a Segmented Combustor

The data for BG and coolant are the same as for Tactical AC, water vaporization section of combustor (Appendix H).

The selection of the duct cross-section to use is based on the amount of catalyst. From Table XIV for 75% conversion:

22 lb of catalyst or 0.542 ft^3

A 2 x 2 ft duct is selected, and 4 layers are assumed.

Layer	Thickness, inch	Volume of Catalyst ft^3
1	0.25	0.083
2	0.375	0.125
3	0.5	0.167
4	0.625	0.208
Total	1.75	0.583

Weight of catalyst = 0.583 x 40.6 = 23.7 lb

Weight of 8 screens, each 4 ft^2:

$8 \times 4 \times 1.7 = 54.4$ lb

There will be 64 heating elements, each 2 ft long, thus

$64 \times 2 \times 0.1 = 12.8$ lb

Heat Transfer Surface

Tubing: $OD_t = 1"$ wall = 0.035" $ID_t = 0.93"$

Fins: $b_f = 0.25"$ $th_f = 0.035"$ N_f = 8 fins/inch

$OD_f = 1.5"$ $r_e = 0.75"$ $r_b = 0.5"$

Arrangement: square pitch, $N_{t/b} = 16$

$S_T = S_L = V_s = 1.5$ inch = 0.125 ft

Duct Side	Tube Side
$A_f = 1.309\ ft^2/ft$	$a_t = 0.00472\ ft^2$
$A_o = 0.189\ ft^2/ft$	$A_{t/b} = 16\ a_t = 0.0755\ ft^2$/bank
$P_p = 9.44$ ft/ft	
$d_{es} = 0.101$ ft	$d_t = 0.0775$ ft
$a_s = 0.96\ ft^2$	
$G_s = 6{,}100$ lb/(ft^2)(hr)	$G_t = 125{,}900$ lb/(ft^2)(hr)
$Re_s = 6{,}480$	$Re_t = 29{,}600$
$j_f = 52.5^{(1)}$	$j_i = 96^{(2)}$
$h_f = 18.6$ BTU/(hr)(ft^2)(°F)	$h_i = 357$ BTU/(hr)(ft^2)(°F)
$h_f' = 18.1$ " "	$h_i' = 208$ " "

$$r_e/r_b = 1.5$$

$y_b = 0.00146$ ft $\qquad k_t = 13.92$ BTU/(hr)(ft^2)(°F/ft) for 316-SS

$$(r_e - r_b)\sqrt{\frac{h'_f}{k_t\, Y_b}} = 0.622$$

$\Omega = 0.85$ (3)

$a_i = 0.244$ ft^2/ft

$A_{it/b} = 16\, a_i \times 2 = 7.79$ ft^2/bank (2 ft long)

$h'_{fi} = 96.6$ BTU/(hr)(ft^2)(°F)

$U_{Di} = 66$ " "

$A_{iT} = 59$ ft^2

$\underline{N_b = 8 \text{ banks}}$

The pressure drop inside the tubing is approximately 32 psi. This would mean that either larger tubing is necessary or a water booster pump has to be used.

Fuel Preheating Pipe

$W_F = 400$ lb/hr

$Q = 333$ BTU/min = 20,000 BTU/hr

Hot Fluid		Cold Fluid	Difference
1337°F	Higher temperature	425°F	912°F
1337	Lower temperature	350°F	987
0	Difference	75	75

LMTD = 950°F

$F_W = \dfrac{400}{110} =$ [illegible] (larger size tubing to use)

$F_Q = \dfrac{333}{328} = 1.015$ (slightly more heat to transfer)

$F_{LMTD} = \dfrac{858}{950} = 0.903$ (smaller heat transfer surface because of larger LMTD)

$F_{SC-U} = 3.64 \times 1.015 \times 0.903 = \underline{3.33}$

Weight of tubing = $3.33 \times 5 = \underline{17\ lb}$

Weight of the Combustor

Catalyst	23.7 lb
Screens (8)	54.4
Heating elements	12.8
Cooling tubing (44 lb/bank)	352
Duct, baffles (Hastelloy C)	170
Fuel preheating tubing	17
Total	630 lb

V. Heat Exchanger No. 1 (HE 1)

The heat and mass balance calculations for design conditions of FP 2 were done, based on the heat transfer surface determined for FP #1. This is not unreasonable because the combustion air needs only 100°F of preheat in FP #2 and the total heat duty is correspondingly reduced. Therefore, the weight of HE 1 is the same in both flight plans. (It is recognized that a check of pressure drop might show the need to enlarge the tube size.)

Weight of HE 1 = 1748 lb

VI. Combination HE 2 and HE 3

A. HE 2 section

This section removes relatively more heat in comparison to FP #1 due to the reduced heat duty in HE 1.

$W_{BG,\ Avg} = 5,790$ lb/hr

$Q = 1,519,000$ BTU/hr

$W_{F\ cooling} = 49,000$ lb/hr

Hot Fluid		Cold Fluid	Difference
1244°F	Higher temperature	350	894
340	Lower temperature	300	40
904	Difference	50	854

$LMTD = 275.5°F \quad R = 18 \quad S = 0.05 \quad F_T = 0.975 \quad \underline{\Delta t = 267}$

$F_Q = \frac{1,519,000}{96,000} = 15.84$ (more heat to transfer)

$F_{\Delta t} = \frac{94}{267} = 0.352$ (improved heat transfer via larger Δt)

$F_W = \frac{1603}{5790} = 0.277$ (improved heat transfer via larger mass velocity of BG)

$F_{SC-U} = 15.84 \times 0.352 \times 0.277 = \underline{1.544}$

Thus a larger heat transfer area is necessary, which affects the weight as follows:

$$1.544 \times 157 = \underline{205 \text{ lb}}$$

A further correction is needed to allow for larger tubing because the flow rate of fuel coolant has increased 15-fold. Inspection of tubing charts indicates the increase in weight per linear foot will be 15-28%, and a value of 20% is chosen. This is applied to the weight of tubing only, which represents 109 lb in the exchanger for FP #1.

$$(109 \times 1.544) \times 0.2 = \underline{34 \text{ lb}}$$

Therefore, the weight of HE 2 section becomes

$$205 + 34 = \underline{\underline{239 \text{ lb}}}$$

B. HE 3 Section

$W_{BG} = 5,660$ lb/hr

$Q = 540,000$ BTU/hr

$W_{H_2O \text{ cool}} = 4,560$ lb/hr

The only items that change are the flow rates and the heat laod.

$F_Q = \frac{540,000}{174,000} = 3.1$ (more heat to transfer)

$F_{W_{BG}} = \frac{1603}{5660} = 0.283$ (improved heat transfer at higher mass velocity)

$F_{SC-U} = 3.1 \times 0.283 \simeq 0.9$

This indicates that no increase in heat transfer area is necessary. However, to avoid an excessive pressure loss in cooling water, tubing of larger size is required. A 15% increase in tubing weight is estimated, thus

$$127 \times 0.15 = 20 \text{ lb}$$

and HE 3 will weigh $205 + 20 = \underline{225 \text{ lb}}$.

VII. Drier

$$W_{BG,\ avg} = 5,420$$

$$Q = 302,000$$

$$W_{H_2O,\ cooling} = 5,500 \text{ lb/hr}$$

No pressure losses suffered by the BG were calculated. Because it has been assumed that a large loss takes place in the tubes of HE 1, it is estimated that the BG will reach the drier with the same pressure as in the case of SST FP 1, namely 37.8 psia, despite the fact that the initial air pressure is higher (160 psia) in FP #2.

$$V_{SBG,\ in} = 158.9\ (3.145 \times 1.012) = 506 \text{ cfm}$$

$$V_{Dry\ BG,\ out} = 196.8\ (3.145 \times 0.944) = 584 \text{ cfm}$$

$$V_{BG,\ Avg} = 545 \text{ cfm}$$

Using the summary of data for $CaCl_2$ and zeolite (Section VII, Appendix F) and applying the volumetric factor

$$F_{Vol} = \frac{545}{147} = 3.708 \quad \text{we get:}$$

Agent	$CaCl_2$	Zeolite	Combined
Volume, ft^3	30.8	54.5	85.3
Weight, lb	1,572	2,095	3,667
Length (ex tubing)	5.14	9.08	14.22
Hours (no regen.)	185	185	185

The length is a function of the cross-section, which in turn is a function of the parameters and recommendations for zeolite desiccant given in Section VII, Appendix F.

Cross-section: width = 3 ft

height = 2 ft

area = 6 ft^2

Gas velocity = 545 ÷ 6 = 90.8 ft/min

Length zeolite section = 54.5 ÷ 6 = 9.08 ft

Equivalent circle diameter = $\left[(4 \times 6) \div \pi\right]^{0.5}$ = 2.77 ft

L/D = 9.08 ÷ 2.77 = 3.28

Thus the above cross-section satisfies all the criteria for zeolite.

Using the same size tubing as in SST FP 1, the following changes take place because of the new cross section:

$N_{t/b} = 24$

$a_s = 2.16\ ft^2$

$G_s = 2{,}510\ lb/(ft^2)(hr)$

$A_{t/b} = 0.0277\ ft^2/bank$

$G_t = 198{,}600\ lb/(ft^2)(hr)$

Now:

$$F_Q = \frac{302{,}000}{87{,}900} = 3.44$$

$$F_{G_s} = \frac{2{,}077}{2{,}512} = 0.827$$

$$F_{G_t} = \frac{127{,}000}{198{,}600} = 0.64$$

hence

$$F_{SC-U} = 3.44 \times 0.827 \times 0.64 = \underline{\underline{1.82}}$$

Consequently, the weight of the cooling tubing becomes

$$237 \times 1.82 = \underline{431\ lb}$$

The duct weight is calculated for the new size of drier using ¼-inch thick aluminum sheets. Volume material 5,830 $inch^3$, and the weight is 577 lb.

Thus:

Screens ($3 \times 6\ ft^2 \times 1.7\ lb/ft^2$)	30 lb
Weight of duct	577
Weight of cooling tubing	431
Weight of desiccants	3,667
Weight of gas filter	10
Accessories: cooling water	285
Accessories: water tank	20
Accessories: pump	30
Total	5,050 lb

Final Note

Optimization of the entire system for SST FP 2 in general, plus adjustments for the smallest possible pressure losses in BG, would lead (in addition to general reduction in weight) to reduction of the volumetric flow rate of BG (as it would be under higher pressure). In situations where the quantity of drying agent is dictated by performance efficiency, this should result in a further weight reduction.

REFERENCES

1. Kern, Process Heat Transfer, pp 555, 838, McGraw-Hill Book Company, New York, 1950

2. Kern, ibid., p 834

3. Kern, ibid., p 542

APPENDIX H

HEAT AND MASS BALANCE

AND EQUIPMENT DESIGN FOR TACTICAL AIRCRAFT

APPENDIX H

HEAT AND MASS BALANCE
AND EQUIPMENT DESIGN FOR TACTICAL AIRCRAFT

I. Material and Heat Balance for Design Conditions

Case II differs from Case I (for the SST) mainly in that exchanger HE 3 is omitted because the fuel used as coolant in HE 2 is cold enough to condense a substantial amount of water from the combustion products, and deliver the uncondensed gases to the drier at 85°F. A minor difference between the cases is that conversion level in Case II is 75% versus 96% in Case I. In view of the similarity in calculations to those presented in Appendix E, fewer details are listed.

Conditions and flows are set forth in Figure 43. Following is a tabulation of the important values.

Molar volume of vapors in fuel tank	476 ft^3/lb-mole
Air requirement (volume)	184 cfm
(lb-moles)	0.386
(weight)	11.2 lb/min
Water formed	0.73 lb/min
Fuel requirement	0.833 lb/min

Combustion Product Mixture:

Component	% Vol.	% Wt.	Flow, lb/min
N_2	74.71	71.36	8.57
CO_2	9.93	14.91	1.79
H_2O	9.93	6.10	0.73
O_2	4.97	5.42	0.65
$C_{10}H_{20}$	0.46	2.21	0.26
	100.00	100.00	12.00

Excess fuel (condensable)	0.252 lb/min
(non-condensable)	0.013 lb/min
Dry BG (ex non-condens. fuel)	11.02 lb/min
Preheat temp. (air + fuel mixture)	1,120°F
Heat removed via combustor coolant	9,060 BTU/min

Heat transferred in HE 1:

to air	303 BTU/min
to fuel	74 BTU/min
Temp. of BG leaving HE 1	1,234°F
Condensed in HE 2:	
water	0.439 lb/min
fuel	0.252 lb/min
Heat removed from BG in HE 2	4,330 BTU/min
Flow of coolant (fuel) to HE 2	95 lb/min
Temp. of BG leaving HE 2	85°F
Fraction of water condensed in HE 2	60%

The remaining 40% of the combustion water is removed in the drier. However, in calculating the capacity of the drier over an entire flight, allowance must be made for the fact that during 75% of the flight average conditions prevail and 70% of the water is removed by condensation. Thus, over the flight:

Water condensed	2.86 lb (67.5%)
Water absorbed in drier	1.37 lb (32.5%)
Heat duty in drier	411 BTU/min
Coolant flow to drier (and combustor)	8.3 lb/min
Temp. of coolant (H_2O) leaving drier	89.5°F

The water leaving the drier at 89.5°F is delivered to the combustor for cooling duty, where it is converted to steam.

The molar quantities for design calculations are as follows:

N_{Air} = 0.386 lb-moles/min

$N_{Air\text{-}fuel}$ = 0.392 lb-moles/min

N_{MBG^*} = 0.409 lb-moles/min

N_{BG} = 0.366 lb-moles/min
(includes dry gas plus non-condensed fuel vapors)

*Moist Ballast Gas

II. Equipment Design

The following presentation highlights the calculations and summarizes the results; details are given only where the calculation methods differ essentially from those used in Case I.

a. Air and Fuel Feed

1. Air Pipe from HE 1 to Annulus of Vaporization Chamber Outlet Pipe

The air in this pipe has been preheated in HE 1.

W_{air} = 11.2 lb/min = 670 lb/hr

t = 1,300°F

P = 36.0 psia

Pipe material = 316 SS

OD = 2.25 in   ID = 2.12 in

R_e = 0.1767 x 27,400/0.1007 = 48,000

ΔP = 1.85 psi/100 ft pipe

equivalent length (incl. fittings) = 35 ft

ΔP = 1.85 x 0.35 = 0.65 psi

P_{air} reaching annulus of outlet pipe connecting vaporization chamber and reactor = 35.3 psia

W_{pipe} = 6 x (1.5 lb/ft) = 9 lb

2. Vaporization Chamber Inlet Pipe

Air for atomization = 3.7 cfm

= 12.2 lb/hr

Pipe material = 316 SS

OD = 0.5 in   ID = 0.48 in

R_e = 390

$\Delta P = 0.07$ psi/100 ft

equivalent length = 12 ft

$\Delta P =$ 0.01 psi

P_{air} reaching nozzle = 35.0 psia

Weight of pipe = 5 x (0.12 lb/ft) = 0.6 lb

3. Preheat Fuel Pipe

$W_F = 0.833$ lb/min = 50 lb/hr

t = 300°F

Pipe material = 316 SS

1/8" IPS schedule 5

OD = 0.405 in   ID = 0.335 in $W_{pipe} = 0.14$ lb/ft

$R_e = 2{,}530$

$\Delta P = 0.29$ psi/100 ft pipe

equivalent length = 11 ft

$\Delta P =$ 0.04 psi

Weight fuel pipe = 1.3 lb

4. Vaporization Chamber (VC)

Nozzle

The fuel delivery is 0.151 gpm

Nozzle No. 22 of Spraying Systems Co (Bellwood, Illinois) is selected.

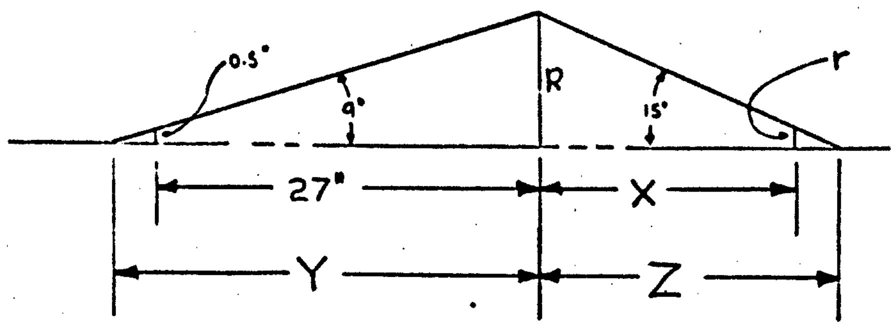
Weight of nozzle = 0.5 lb

Spray angle = 18°

Minimum cone length = 27 inch

VC Dimensions and Weight

Referring to the following diagram which represents one-half of the transverse section, by geometrical relationships we find:



Y = 30.2 in

R = 4.78 in

Z = 17.8 in

Since ID of inner core of outlet pipe is 1.22 in,

r = 0.61 in, thus

X = 15.6 in

L_{VC} = 42.6 in

~ 3.6 ft

$ID_{VC} = 2\ R = \underline{\underline{9.56\ in}}$

wall thickness = 0.125 in

$OD_{VC} = 9.81$ in

Using a density of 0.29 lb/in^3 for 316 SS, the weight W_{VC} = 27 lb

Pressure Drop in VC

The total loss of pressure in the nozzle and vaporization chamber is estimated at 20% of the gage pressure of the air reaching the nozzle. Thus

$$\Delta P = 0.2 \times 21.04 = \underline{\underline{4.21\ psi}}$$

and the fuel-rich mixture leaves the vaporization chamber at 16.8 psig or 31.5 psia.

5. Heater for Vaporization Chamber (VC)

During the startup period, air is available at 250°F, and cannot be preheated in HE 1 until the combustor approaches operating temperature. Temperature, enthalpy and flow rate data are used to calculate the amount of heat to supply via electric heaters to insure vaporization of the fuel.

$$Q = 5{,}010\ BTU/hr$$

The required heater is

$$\frac{5{,}010}{3{,}410} = 1.5\ KW$$

However, during sudden accelerations it may become necessary to have additional heat, hence a 3 KW heater is assumed, and its weight is estimated at 3 lbs.

Note: This heater functions only to heat the fuel-rich mixture passing through the VC. The bulk of the combustion air is heated by elements located within the annulus of the VC outlet pipe, and the final mixture obtains heat from elements located in the combustor.

6. VC Outlet Pipe

This double pipe is designed in such a way that both fluids (fuel-rich mixture from VC and the bulk of the air) arrive at the combustor with the same pressure. The pipe for Case II is the same as that used in Case I, and details are given in Figure F-2 in Appendix F.

Inner (core) Pipe for Fuel-Rich Mixture

$W_{F\text{-}A\ mix} = 12.2 + 50 = 62.2$ lb/hr

$t_{mix} = 1{,}120°F$

In a manner similar to that used in Case I it is determined that

$G_{F\text{-}A\ mix} = 7{,}700$ lb/ft^2-hr

$R_e = 11{,}100$

$\Delta P_{100} = 0.14$ psi/100 ft

Since 2.8 ft of pipe will be used,

$\Delta P = 0.14 \times 0.028 = \underline{0.004 \text{ psi}}$

and the fuel-rich mixture will reach the combustor at 16.8 psig.

Annular Space for Air

$W_{air} = 11.07$ lb/min = 664 lb/hr

$t = 1{,}120°F$

Again, by methods used in Case I, we find

$G_a = 56{,}300$ lb/ft^2-hr

$Re_a = 9{,}000$

$\Delta P = \underline{1.34 \text{ psi/ft}}$

The air reaches the annulus at 20.6 psig (see above) and the fuel-rich mixture reaches the combustor at 16.8 psig. Equalization of the pressure requires that the air stream lose 3.8 psi on passage through the annulus.

Consequently, the length of the pipe is

$$\frac{3.8}{1.34} = \underline{\underline{2.8 \text{ ft}}}$$

Weight of the VC outlet pipe

The weight of the double pipe, with fins and heating elements is 4.65 lb/ft.

Therefore, the weight of the outlet piping is:

$$4.65 \times 2.8 = \underline{\underline{13 \text{ lb}}}$$

7. Total Weight of Air and Fuel Feed

Air pipe (main)		9 lb
Air pipe (VC inlet)		0.6
Fuel pipe		1.3
Vaporization chamber		27
Nozzle		0.5
Heater		3
Outlet piping		13
Added for fittings		0.6
	Total	55 lb

b. Combustor

The respective advantages of the radial reactor and the segmented reactor have already been mentioned. The segmented reactor is chosen for the tactical aircraft mainly because it is simple to construct with catalyst separated from cooling coils (hence no need for catalyst diluent just to submerge the coils). Reactant mixing is less easily accomplished, however, and the temperature gradient through each layer of catalyst is probably greater than would be the case with closely spaced coils in the radial design. Although not evaluated in this study, a segmented radial design might represent a good combination of the favorable features in each design.

Fig. 35 & 44 show schematically how the segmented reactor ties in with other components, and also indicates the staged introduction of combustion air. Determination of the exact amount of air to be admitted at each stage has not been made. This will depend largely on safety considerations. For the present conceptual design, a simplified approach is taken. It is assumed that all the air and fuel enter the combustor simultaneously from the double pipe inlet and mix well before reaching the first catalyst layer. To assure attainment of the desired conversion level, thicker catalyst layers are used as the advancing mixture of gases becomes more and more oxygen depleted. Thus the first layer is shown to be ¼-inch thick, the second 3/8-inch and the third 1/2-inch thick. Other layers, if necessary, may be thicker by the same or different increments. The heat transfer surface requirements are calculated for the total heat load (not for individual segments), and used to determine the total number of cooling banks. No attempt is made to distribute or assign a given number of banks to a given segment. The pressure drop is calculated for the entire unit, assuming some average property values for the gas.

The cross-section of the combustor can be varied. It governs the velocity of gases. Together with the required volume of catalyst, it determines the total thickness of the individual layers, and the pressure drop through the unit.

Distribution baffles are placed in the diverging entrance section of the combustor, to mix the entering gas streams and to distribute the resulting mixture to the first catalyst layer.

The heating elements immediately in front of the first catalyst layer are used to preheat the combustor at the start of a flight, and to supply additional heat when needed after the initial warmup.

1. Catalyst and Combustor Cross-Section

As previously calculated, the amount of catalyst for 75% oxygen conversion at the design conditions is:

$$0.066 \text{ ft}^3 \text{ or } 2.7 \text{ lb.}$$

Several trials showed that a 1 ft x 1 ft square duct provides a reasonable configuration, with three catalyst layers of 1/4", 3/8 in, and 1/2 in thickness. The volume of catalyst is distributed as follows:

Layer	Thickness, inch	Volume of Catalyst, ft^3
1	0.25	0.0209
2	0.375	0.0314
3	0.5	0.0419
Total	1.125 inch	0.0942 ft^3

Thus, the weight of the catalyst used is

$$0.0942 \times 40.6 = \underline{\underline{3.8 \text{ lb}}}$$

This is about 40% in excess of the calculated amount. The excess is regarded as assurance that the desired conversion will be achieved when a charge of catalyst nears the end of its service life. Alternatively, the 40% additional volume could be occupied by a low-density granular diluent, in which case a reduction in weight would be effected, equal to

$$(0.094 - 0.066)\,(40.6 - 25.0) = 0.44 \text{ lb}$$

Six screens are required to keep the catalyst in place. Assuming a 0.041 inch wire diameter the unit weight is 1.7 lb/ft^2 and the weight of six screens is

$$6 \times 1.7 = \underline{\underline{10.2 \text{ lb}}}$$

2. Heating Elements

The heating elements are shielded electric resistance wires. They are 3/16 inch in diameter and are spaced 0.375 in. ℄ to ℄. Thus there are

$$12 \div 0.375 = 32 \text{ elements.}$$

They weigh about 0.1 lb/ft, thus their weight is

$$32 \times 0.1 = \underline{\underline{3.2 \text{ lb}}}$$

3. Data for Transfer of Combustion Heat

The overall transfer of combustion heat may be described as follows:

$W_{A\text{-}F\ mix} = W_{MBG} = 12.003$ lb/min = 720 lb/hr

$W_{H_2O\ cool} = 8.3$ lb/min = 500 lb/hr

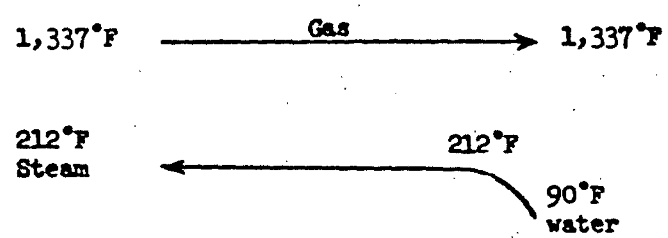
$Q_{Total} = 9{,}064$ BTU/min = 544,000 BTU/hr

$Q_{H_2O\ Preheat} = 500 \times 1 \times (212-90) = 61{,}000$ BTU/hr

$Q_{H_2O\ vaporiz} = 500\ (1{,}150.4-180) = 483{,}000$ BTU/hr

Moist BG is to be maintained at 1,337°F.

Cooling water enters at 90°F, is preheated to 212°F, then vaporized to steam at 212°F.



Since cooling water reaches the combustor at 90°F and leaves as steam at 212°F, two sections are considered (one in which water is preheated to 212°F and the second in which water is vaporized) rather than one section for which the properties of water at 90°F are averaged with those of steam at 212°F. Also, it is assumed that in the vaporization portion the air-fuel mixture comprises 20% of the gas.

(a) Water Preheating Section

Hot Fluid		Cold Fluid	Difference
1,337°F	Higher temperature	212°F	1,125°F
1,337°F	Lower temperature	90°F	1,247°F
0	Difference	122°F	122°F

$$\text{LMTD} = \frac{1,247-1,125}{2.3 \log \frac{1,247}{1,125}} = 1,185°\text{F}$$

$$R = \frac{0}{122} = 0 \qquad S = \frac{122}{1,247} \simeq 0.1$$

F_T assumed to be 1, that is, no correction

$\Delta T = 1,185°\text{F}$

$T_c = 1,337°\text{F}$ for MBG

$t_c = (90 + 212) \div 2 = 151°\text{F}$ for cooling water

For MBG at 1,337°F and 30 psia we get:

Component	Component Weight W, lb/min	μ Viscosity, cps	μW	C_p Sp. Heat BTU/(lb)(°F)	C_pW	k Conductivity BTU/(hr)(ft²)(°F/ft)	kW
CO_2	1.79	0.0385	0.0689				
N_2	8.57	0.039	0.3341				
O_2	0.65	0.0458	0.0298				
H_2O-Vapor	0.73	0.0363	0.0265	0.545	0.179	0.073	0.06
Fuel Vapor	0.26	0.027	0.0072	0.815	0.216	0.0421	0.01
Air*	11.01	-	-	0.272	2.994	0.0388	0.4
Total	12.01**		0.4665		3.389		0.4
$\frac{\Sigma \text{ property} \times W}{\Sigma W}$			0.0389		0.282		0.0

*Properties of air were used whenever properties of CO_2, N_2 and O_2 were unavailable. The weight of air is equal to the sum of weights of CO_2, N_2 and O_2

**The total weight is equal to the sum of weights of CO_2, N_2, O_2, water vapor and fuel vapor (or air, water vapor and fuel vapor).

$\mu_{BG} = 0.0389$ cps $= 0.094$ lb/(ft)(hr)

$C_{p\ BG} = 0.282$ BTU/(lb)(°F)

$k_{BG} = 0.0415$ BTU/(hr)(ft^2)(°F/ft)

$$\left(\frac{C_p \times \mu}{k}\right)^{1/3} = 0.862$$

$\phi_{BG} = 1$

$V_M = 642$ ft^3/lb-mole

$\overline{V}_{F\text{-}V} = 10.27\ (14.7 \div 30) = 5$ ft^3/lb

$V_{BG} = V_M \times (N_{N_2} + N_{O_2} + N_{CO_2} + N_{H_2O\ Vapor}) + \overline{V}_{F\text{-}V} \times (N_{F\text{-}V(total)}$
$\times 140)$

$= 0.407\ V_M + 0.265\ \overline{V}_{F\text{-}V} = 263$ cfm

$\rho_{BG} = 0.0457$ lb/ft^3

$S_{BG} = 0.000732$

For cooling water at 151°F we get

$\mu = 0.44$ cps $= 1.065$ lb/(ft)(hr)

$C_p = 1$ BTU/(lb)(°F)

$k = 0.376$ BTU/(hr)(ft^2)(°F/ft)

$$\left(\frac{C_p \nu}{k}\right)^{1/3} = 1.415$$

$\mu_{wall} = 0.05$ cps

$$\phi_t = \left(\frac{\mu}{\mu_{wall}}\right)^{0.14} = \left(\frac{0.44}{0.05}\right)^{0.14} = 1.33$$

$\rho = 8.18$ lb/gal $= 61.2$ lb/ft^3

$S = 1$

$\overline{V}_{H_2O\ Cool} = 0.01635$ ft^3/lb

$V_{H_2O\ Cool} = 8.3 \times 0.01635 = 0.136$ cfm $= 0.00226$ cfs

(b) Water Vaporization Section

Hot Fluid		Cold Fluid	Difference
1,337°F	Higher temperature	212°F	1,125
1,337°F	Lower temperature	212°F	1,125
0	Difference	0	0

Thus, in theory, a constant temperature difference of 1,125 exists at all times and no temperature correction is applied.

$$\Delta t = 1{,}125°F$$

$$T_c = 1{,}337°F$$

$$t_c = 212°F$$

Properties of Gases at 1337°F and 30 psia

These gases consist of 80% moist ballast gas and 20% air-fuel mixture. For BG the values used in the water preheating section apply. For the air-fuel mixture:

Property	Air	Fuel Vapor	Air-Fuel Mixture
μ, lb/(ft)(hr)	0.102	0.065	0.099
C_p, BTU/(lb)(°F)	0.272	0.815	0.31
k, BTU/(hr)(ft^2)(°F/ft)	0.0388	0.0421	0.039

$$V_{A-F\ mix} = V_M \times N_{Air} + \overline{V}_{F-V} \times W_F$$

$$= 642 \times 0.387 + 5 \times 0.833 = 252 \text{ cfm}$$

The general expression used to determine the properties of interest:

$$\text{Property}_{Avg} = 0.2\ \text{Property}_{A-F\ mix} + 0.8\ \text{Property}_{BG}$$

$$\mu_{Avg} = 0.095 \text{ lb/(ft)(hr)}$$

$$C_{p\ Avg} = 0.288 \text{ BTU/(lb)(°F)}$$

$$k = 0.041 \text{ BTU/(hr)(ft}^2\text{)(°F/ft)}$$

$$\left(\frac{C_{p\ Avg} \times \mu_{Avg}}{k_{Avg}}\right)^{1/3} = 0.874$$

$\phi_{Avg} = 1$

$V_{Avg} = 261$ cfm

$\rho_{Avg} = 0.046$ lb/ft^3

$S_{Avg} = 0.00074$

For Water and Steam at 212°F

Property	Water	Steam	Average*
μ, lb/(ft)(hr)	0.63	0.0303	0.33
C_p, BTU/(lb)(°F)	1	0.35	0.68
k, BTU/(hr)(ft^2)(°F/ft)	0.41	0.016	0.213
$\bar{V}$, ft^3/lb	0.01672	26.80	-
V, cfm	0.1388	222.4	111.2
ρ, lb/ft^3	59.8	0.0373	0.075
S	1	0.0006	0.0012

$$\left(\frac{C_{p\ avg}\ \mu_{avg}}{k_{avg}}\right)^{1/3} = 1.018$$

$$\phi_t = 1.33$$

(c) Tubing (316-SS)

Tubing: $OD_t = 0.75"$ wall $= 0.025"$ $ID_t = 0.7"$

$k_t = 13.92$ BTU/(hr)(ft^2)(°F/ft)

Fins: $b_f = 0.125"$ $th_f = 0.035"$ $N_f = 8$ fins/inch tubing

$r_e = 0.5"$ $r_b = 0.375"$

Bank arrangement: square pitch

$N_{t/b} = 12$ tubes/bank

$S_T = S_L = V_s = 1"$

*Arithmetic average values are shown except in the case of ρ_{avg}, which is $W_{H_2O\ cool} \div V_{avg}$, and

the specific gravity, S_{avg}, which is obtained:

$$S_{avg} = \rho_{avg} \div 62.4$$

Duct Side

$$A_f = \frac{\pi}{4}(1^2 - 0.75^2) \times 2 \times 8 \times 12 = 66 \text{ inch}^2/\text{ft} = 0.458 \text{ ft}^2/\text{ft}$$

$$A_o = \pi \times 0.75 \times 12 \times (1-8 \times 0.035) = 20.4 \text{ inch}^2/\text{ft} = 0.141 \text{ ft}^2/\text{ft}$$

$$P_p = 2 \times 0.125 \times 2 \times 8 \times 12 + 2\,(12-8 \times 0.035 \times 12) = 65.3 \text{ inch/ft}$$
$$= 5.44 \text{ ft/ft}$$

$$d_{es} = \frac{2\,(0.458 + 0.141)}{5.44\,\pi} = 0.07 \text{ ft}$$

$$a_s = 12 \times 1 \left[12 \times 1-12\,(0.75 + 2 \times 8 \times 0.035 \times 0.125)\right] = 25.9 \text{ inch}^2 = 0.18 \text{ ft}^2$$

$$G_s = \frac{720}{0.18} = 4{,}000 \text{ lb/(ft}^2\text{)(hr)}$$

$$h_d = 602 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Tube Side

$$a_t = \frac{\pi}{4} \times 0.7^2 = 0.385 \text{ inch}^2 = 0.00267 \text{ ft}^2$$

$$A_{t/b} = 12\, a_t = 0.032 \text{ ft}^2\text{/bank}$$

$$d_{et} = 0.7 \div 12 = 0.0583 \text{ ft}$$

$$G_t = \frac{500}{0.032} = 15{,}600 \text{ lb/(ft}^2\text{)(hr)}$$

$$h_{d_i} = 500 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Fin Efficiency and Inside Tube Area

$$\frac{r_e}{r_b} = \frac{0.5}{0.375} = 1.33$$

$$y_b = (0.035 \div 2) \div 12 = 0.00146 \text{ ft}$$

$$(r_e - r_b)\sqrt{\frac{h'_f}{k_t\, y_b}} = \frac{0.5-0.375}{12}\sqrt{\frac{h'_f}{13.92 \times 0.00146}} = 0.0104\left(\frac{h'_f}{0.0203}\right)^{0.5}$$

$$a_i = \pi \times 0.7 \times 12 = 26.4 \text{ inch}^2/\text{ft} = 0.183 \text{ ft}^2/\text{ft}$$

$$A_{it/b} = 12\, a_i \times 1 = 2.2 \text{ ft}^2\text{/bank}$$

Pressure Drop, Duct Side

$$V_{NF} = 1 \times 1 \times \frac{1}{12} - 12 \times \frac{\pi}{4} \times \frac{1}{144} \left[0.75^2 + 0.035 \times 8\ (1^2 - 0.75^2)\right]$$

$$= 0.0385\ ft^3$$

$$S_F = 12\ (0.458 + 0.141) \times 1 = 7.19\ ft^2$$

$$D'_{ev} = \frac{4 \times 0.0385}{7.19} = 0.0214\ ft$$

$$\left(\frac{D'_{ev}}{S_T}\right)^{0.4} = \left(\frac{0.0214}{1/12}\right)^{0.4} = 0.58$$

$$\left(\frac{S_L}{S_T}\right)^{0.6} = \left(\frac{1/12}{1/12}\right)^{0.6} = 1$$

4. Heat Exchange Surface

(a) Water Preheating Section

Heat Transfer - Duct Side

$$Re_s = \frac{0.07 \times 4{,}000}{0.094} = 3{,}000$$

$$j_f = 30 \quad \text{(see Ref. 1)}$$

$$h_f = 30 \times \frac{0.0415}{0.07} \times 0.862 \times 1 = 15.3\ BTU/(hr)(ft^2)(°F)$$

$$h'_f = \frac{15.3 \times 602}{15.3 + 602} = \underline{15\ BTU/(hr)(ft^2)(°F)}$$

Heat Transfer - Tube Side

$$R_{et} = \frac{0.0583 \times 15{,}600}{1.065} = 860$$

$$j_{hi} = 6 \quad \text{(see Ref. 2)}$$

$$h_i = 6 \times \frac{0.376}{0.0583} \times 1.415 \times 1.33 = 72.8\ BTU/(hr)(ft^2)(°F)$$

$$h'_i = \frac{72.8 \times 500}{72.8 + 500} = \underline{63.6\ BTU/(hr)(ft^2)(°F)}$$

Heat Transfer - Overall U and Area

$$(r_e - r_b)\sqrt{\frac{h'_f}{k_t Y_b}} = 0.28$$

$$\Omega = 0.975 \quad \text{(see Ref. 3)}$$

$$h'_{fi} = (0.975 \times 0.458 + 0.141)\ \frac{15}{0.183} = \underline{48.1\ BTU/(hr)(ft^2)(°F)}$$

$$U_{Di} = \frac{48.1 \times 63.6}{48.1 + 63.6} = \underline{\underline{27.4\ BTU/(hr)(ft^2)(°F)}}$$

$$A_{iT} = \frac{61{,}000}{27.4 \times 1{,}185} = 1.88\ ft^2$$

$$\boxed{A_{iT,\ H_2O\text{-}Liq} = 1.88\ ft^2}$$

(b) Water Vaporization Section

Heat Transfer - Duct Side

$$Re_s = \frac{0.07 \times 4{,}000}{0.095} = 2{,}950$$

j_f = 29.6 (see Ref. 1)

$$h_f = 29.6 \times \frac{0.041}{0.07} \times 0.874 \times 1 = 15.2\ BTU/(hr)(ft^2)(°F)$$

$$h'_f = \frac{15.2 \times 602}{15.2 + 602} = \underline{14.8\ BTU/(hr)(ft^2)(°F)}$$

Heat Transfer - Tube Side

$$Re_t = \frac{0.0583 \times 15{,}600}{0.33} = 2{,}760$$

j_{hi} = 12 (see Ref. 2)

$$h_i = 12 \times \frac{0.213}{0.0583} \times 1.018 \times 1.33 = 59.3\ BTU/(hr)(ft^2)(°F)$$

$$h'_i = \frac{59.3 \times 500}{59.3 + 500} = \underline{53\ BTU/(hr)(ft^2)(°F)}$$

Heat Transfer - Overall U and Area

$$(r_e - r_b)\sqrt{\frac{h'_f}{k_t Y_b}} = 0.28$$

Ω = 0.975 (Ref. 3)

$$h'_{fi} = (0.975 \times 0.458 + 0.141)\ \frac{14.8}{0.183} = \underline{47.5\ BTU/(hr)(ft^2)(°F)}$$

$$U_{Di} = \frac{47.5 \times 53}{47.5 + 53} = \underline{\underline{25\ BTU/(hr)(ft^2)(°F)}}$$

$$A_{iT} = \frac{483{,}000}{25 \times 1{,}125} = 17.14\ ft^2$$

$$\boxed{A_{iT,\ H_2O\text{-steam}} = 17.14\ ft^2}$$

(c) Number of Banks

$$A_{iT} = A_{iT,\ H_2O\text{-Liq}} + A_{iT,\ H_2O\text{-Steam}}$$

$$= 1.88 + 17.14 = \underline{\underline{19\ ft^2}}$$

$$N_b = \frac{19}{2.2} = 8.65 \simeq 9 \text{ banks}$$

$$\boxed{\text{Number of cooling tubing banks} = 9}$$

5. Pressure Drop

(a) Duct Side

The pressure drop in the BG as it flows through the combustor is the summation of five individual losses, each of which is calculated as follows:

- Pressure loss due to gradual expansion:

  Pressure of gases reaching combustor = 16.83 psig = 31.53 psi
  Loss in diverging cone estimated at 8% incoming gas gage pressure. Thus

  $$\Delta P_1 = 0.08 \times 16.83 = \underline{\underline{1.35\ psi}}$$

  Consequently, the pressure of air-fuel mixture reaching the heating elements = 15.48 psig = 30.18 psia.

- Pressure loss through layers of catalyst:

  V_o = velocity of fluid in "empty" bed, ft/sec

  $$V_o = \frac{261\ cfm}{1 \times 1 \times 60\ sec/min} = \underline{4.35\ ft/sec}$$

D_p = 0.00625 ft

ρ = 0.046 lb/ft^3

μ = 0.095 lb/(ft)(hr) = 0.0000264 lb/(ft)(sec)

$$Re = \frac{0.00625 \times 0.046 \times 4.35}{0.0000264} = \underline{47.4}$$

thus the flow is turbulent; consequently

$$\Delta P_2 = \frac{2.36 \times \mu^{0.15} \times L \times \rho^{0.85} \times V_o^{1.85} \times A_f}{D_p^{\ 1.15} \times 144}$$

$$L = \text{bed thickness} = \frac{0.25 + 0.375 + 0.5}{12} = 0.094 \text{ ft}$$

A_f = wall effect factor = 1

$$\Delta P_2 = \frac{2.36 \times (0.0000264)^{0.15} \times 0.094 \times (0.046)^{0.85} \times (4.35)^{1.85} \times}{(0.00625)^{1.15} \times 144}$$

ΔP_2 = $\underline{\underline{0.12 \text{ psi}}}$

- Pressure loss through screens:

For method and data see under Combustor, Appendix F

ρ = 0.046 lb/ft^3 = 0.000737 g/cc

μ = 0.095 lb/(ft)(hr) = 0.0393 cps

β = 1312

V_o = 4.35 ft/sec

$$R_e = \frac{\beta V_o \rho}{\mu} = \frac{1312 \times 4.35 \times 0.000737}{0.0393} = \underline{107}$$

C = factor from manufacturer's graph = 1.05

α = 0.05

$$\Delta P_3 = \frac{\alpha \rho V_o^2}{C^2} = \frac{0.05 \times 0.000737 \times (4.35)^2}{(1.05)^2} = \underline{0.00064 \text{ psi per screen}}$$

since there are 6 screens

ΔP_3 = 6 x 0.00064 = $\underline{\underline{0.004 \text{ psi}}}$ (negligible)

- **Pressure loss due to cooling coils:**

It is assumed here that the bank of heating elements and the fuel preheating pipe each have the same effect on pressure drop as one bank of cooling coils, thus

$$L_p = (N_b + 2)\ V_s = (9 + 2) \times \frac{1}{12} = 0.92 \text{ ft}$$

$f = 0.00285$ (Ref. 1)

$$\Delta P_4 = \frac{f \times G_s^2 \times L_p}{5.22 \times 10^{10} \times D'_{ev} \times S_s \times \phi_s} \left(\frac{D'_{ev}}{S_T}\right)^{0.4} \left(\frac{S_L}{S_T}\right)^{0.6}$$

$$= \frac{0.00285 \times (4{,}000)^2 \times 0.92 \times 0.58 \times 1}{5.22 \times 10^{10} \times 0.0214 \times 0.000732 \times 1} = \underline{\underline{0.03 \text{ psi}}}$$

- **Pressure loss in "orifice":**

This orifice is used during startups as the supporting frame for the "flaps" used to isolate the rest of the system until the combustor is brought up to temperature.

For details of method see under HE 2, Appendix F.

$$v = \frac{V_{avg}}{60\ a_s} = \frac{260.7}{60 \times 0.18} = 24.1 \text{ ft/sec}$$

$$p_v = \frac{v^2}{2\ g} = \frac{(24.1)^2}{2 \times 32.2} = 9 \text{ ft (column of BG)}$$

$$\text{Thus } p_v = \frac{9 \times 0.046}{144} = \underline{0.003 \text{ psi}}$$

$A_1 = 12 \times 12 = 144 \text{ inch}^2$

$A_2 = 8 \times 8 = 64 \text{ inch}^2$ (2 inch high barrier)

$A_2/A_1 = 0.44$

$$\left(\frac{1}{c^2} - 1\right) = 0.35$$

$$\Delta P_5 = 0.35 \times 0.003 = \underline{\underline{0.001 \text{ psi}}} \text{ (negligible)}$$

- **Total pressure loss in combustor duct**

$$\Delta P_T = 1.35 + 0.12 + 0.004 + 0.03 + 0.001$$

$$\Delta P_T = \underline{\underline{1.505 \text{ psi}}}$$

Thus the pressure of moist BG leaving the combustor is

$$16.83 - 1.51 = 15.32 \text{ psig}$$
$$= 30.02 \text{ psia}$$

$$\boxed{P_{MBG} \text{ leaving combustor} = 30.0 \text{ psia}}$$

(b) Pressure loss in tubes

$G_t = 15{,}600$

$Re_t = 860$

$f = 0.0006$ (Ref. 4)

$$\Delta P_{bank} = \frac{0.0006 \times 15{,}600^2 \times 1}{5.22 \times 10^{10} \times 0.0583 \times 0.0012 \times 1.33}$$

$$= \underline{0.03 \text{ psi/1 ft long bank}}$$

Each return bend or header is equivalent to 4.2 linear ft of pipe thus

$$\Delta P_{return} = 0.03 \times 4.2 = \underline{0.126 \text{ psi/return}}$$

Consequently, total pressure loss in tubes is

$$\Delta P_{t,\ Total} = 9 \times 0.03 + 10 \times 0.126$$
$$= \underline{\underline{1.53 \text{ psi}}}$$

6. **Fuel Preheating Pipe**

This pipe is located at the downstream end of the combustor rather than in HE 1 because this allows the fuel to be preheated during the start-up operation.

The tubing size and heating method are the same as for the SST.

$W_F = 0.833$ lb/min $= 50$ lb/hr

$Q = 74$ BTU/min $= 4{,}440$ BTU/hr

Hot Fluid		Cold Fluid	Difference
1,337	Higher temperature	300	1,037
1,337	Lower temperature	150	1,187
0	Difference	150	150

$$\text{LMTD} = \frac{1{,}187 - 1{,}037}{2.3 \log \frac{1{,}187}{1{,}037}} = \underline{1{,}109^\circ F}$$

No correction is applied.

$T_c = 1{,}337^\circ F$

$t_c = 225^\circ F$

Data for BG at 1,337°F are given above.

Data for fuel at 225°F

$\mu_F = 0.55$ cps $= 1.33$ lb/(ft)(hr)

$Cp_F = 0.580$ BTU/(lb)(°F)

$k_F = 0.0757$ BTU/(hr)(ft^2)(°F/ft)

$(Cp\,\mu/k)^{1/3} = 2.17$

$\rho_F = 5.787$ lb/gal $= 43.3$ lb/ft^3

$S_F = 0.694$

$\phi_F = 1.09$

Assuming 5 tubes in the bank, we get

$a_s = 144 - 5\ (0.405 + 2 \times 0.1 \times 0.035 \times 8)\ 12$

$= 116.3$ inch^2 $= 0.808$ ft^2

$G_s = 900$ lb/(ft^2)(hr)

$Re_s = 380$

$j_f = 6.8$ (Ref. 1)

$$h_f = 6.8\ \frac{0.0415}{0.0395} \times\ 0.862 \times 1 = 6.2 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

$h_f' = 6.1$ BTU/(hr)(ft^2)(°F)

$G_t = 50/0.0006 = 83{,}400$ lb/(ft^2)(hr)

$R_e = 1,750$

$j_{hi} = 4$ (Ref. 2)

$h_i = 4 \dfrac{0.0757}{0.028} \times 2.17 \times 1.09 = 25.6$ BTU/(hr)(ft^2)(°F)

$h_d = 250$ BTU/(hr)(ft^2)(°F)

$h'_i = 23.2$ BTU/(hr)(ft^2)(°F)

$\Omega = 0.99$ (assumed the same as for SST)

$h'_{ei} = (0.99 \times 0.2115 + 0.0764) \dfrac{6.1}{0.0877} = 20$ BTU/(hr)(ft^2)(°F)

$U_{Di} = \dfrac{23.2 \times 20}{23.2 + 20} = \underline{10.7 \text{ BTU/(hr)(ft}^2\text{)(°F)}}$

$A_{iT} = \dfrac{4,440}{10.7 \times 1,109} = \underline{\underline{0.4 \text{ ft}^2}}$

$L_t = \dfrac{A_{iT}}{a_i} = \dfrac{0.4}{0.0877} = \underline{\underline{4.5 \text{ ft lin}}}$

This will give approximately the 5 passes assumed initially.

Weight of tubing = 4.5 x 0.263 + 2 x 0.14

= **1.5 lb**

7. Weight

All equipment items are made of 316-SS, except when specified otherwise.

Catalyst	3.8 lb
Screens (6)	10.2
Heating elements	3.2
Fuel preheating tube	1.5
Cooling tubing (9 x 6.8 lb/bank)	61.3
Duct, baffles (Hastelloy C)	30.0
Total	110.0 lb

c. **Heat Exchanger No. 1 (HE 1)**

HE 1 is located immediately after the combustor. The "orifice" that separates them houses flaps which can be used to isolate the combustor from the rest of the system during start-up. The cross-section of HE 1 is rectangular, 2 ft wide x 1 ft high.

1. **Heat Transfer**

The size of tubing, $N_{t/b}$, and its arrangement are the same as in the combustor.

$Q = 22{,}600$ BTU/hr

$W_{BG} = 720$ lb/hr

$W_{Air} = 620$ lb/hr

Air enters at 1,200°F and leaves at 1,300°F

Moist BG enters at 1,337°F and leaves at 1,234°F*

Hot Fluid (BG)		Cold Fluid (Air)	Difference
1,337°F	Higher temperature	1,300°F	37°F
1,234°F	Lower temperature	1,200°F	34°F
103°F	Difference	100°F	3°F

LMTD = 35.5°F

$R = 1.03$ $S = 0.73$ $R_T = 0.7$ (ref. 5)

$\Delta t = \underline{24.8°F}$

Caloric Temperature

Arithmetic averages are sufficient, thus:

for BG: $T_c = 1{,}286°F$

for air: $t_c = 1{,}250°F$

Data at above temperatures

BG at 1,286°F and 28.75 psia

$\mu_{BG} = 0.0383$ cps $= 0.0926$ lb/(ft)(hr)

$C_{p\ BG} = 0.3$ BTU/(lb)(°F)

$k_{BG} = 0.0405$ BTU/(hr)(ft^2)(°F/ft)

*Allowing for heat given up in fuel preheating.

$$\left(\frac{C_{p\ BG}\ \mu_{BG}}{k_{BG}}\right)^{1/3} = 0.882$$

$\phi_{BG} = 1$

$V_{BG} = 266$ cfm

$\rho_{BG} = 0.0451$ lb/ft^3 $\quad S = 0.000722$

Air at 1,250°F and 40 psia

$\mu_{air} = 0.041$ cps $= 0.0992$ lb/(ft)(hr)

$C_{p\,air} = 0.27$ BTU/(lb)(°F)

$k_{air} = 0.0376$ BTU/(hr)(ft^2)(°F/ft)

$$\left(\frac{C_{p\ air}\ \mu_{air}}{k_{air}}\right)^{1/3} = 0.894$$

$\phi_{air} = 1$

$V_{air} = 177.2$ cfm

$\rho_{air} = 0.063$ lb/ft^3 $\quad S = 0.00101$

<u>Heat transfer: duct side, BG</u>

$a_s = 0.36$ ft^2

$G_s = 2{,}000$ lb/(ft^2)(hr)

$Re_s = 1{,}500$

$j_f = 18.5$ (ref. 1)

$h_f = 9.45$ BTU/(hr)(ft^2)(°F)

$h_f' =$ <u>9.3 BTU/(hr)(ft^2)(°F)</u>

<u>Heat transfer: tube side, air</u>

$G_t = 20{,}950$ lb/(ft^2)(hr)

$Re_t = 12{,}300$

$jh_i = 46$ (ref. 2)

$h_i = 26.5$ BTU/(hr)(ft^2)(°F)

$h_{di} = 333$ BTU/(hr)(ft^2)(°F)

$h_i' = 24.6$ BTU/(hr)(ft^2)(°F)

Heat transfer: Overall design coefficient, area and number of banks

$(r_e - r_b)(h_f'/k_t Y_b)^{0.5} = 0.22$

$\Omega = 0.98$ (ref. 3)

$h_{fi}' = \underline{30 \text{ BTU/(hr)(ft}^2\text{)(°F)}}$

$$U_{Di} = \frac{30 \times 24.6}{30 + 24.6} = 13.5 \text{ BTU/(hr)(ft}^2\text{)(°F)}$$

Overall design coefficient = 13.5 BTU/(hr)(ft^2)(°F)

$$A_{iT} = \frac{22{,}600}{13.5 \times 24.8} = 67.7 \text{ ft}^2$$

$A_{it/b} = 12\ a_i \times 2 = 4.4$ ft^2/bank

$$N_b = \frac{67.7}{4.4} = 15.4 \text{ banks}$$

Number of banks = 16

2. Pressure Drop on Duct Side (BG)

There are three elements in the overall pressure loss:

- pressure loss due to enlargement, ΔP_1
- pressure loss due to cooling tubing, ΔP_2
- pressure loss due to "orifice", ΔP_3

(a) Pressure loss due to enlargement

This enlargement is not very gradual in order to save space. However, since it is not from a pipe with relatively small cross-section to a comparatively large duct cross-section, but from the "orifice" (area 64 inch^2) to HE 1 duct (288 inch^2) it is estimated that a loss equal to 8% of gage pressure takes place.

$\Delta P_1 = 0.08 \times 15.32 = \underline{1.23 \text{ psi}}$

(b) Pressure loss due to banks of cooling tubing

$V_{NF} = 0.077\ ft^3$

$S_F = 14.38\ ft$

$D'_{ev} = 0.0214\ ft$

$$\left(\frac{D'_{ev}}{S_T}\right)^{0.4} = 0.58$$

$f = 0.00314$ (ref. 1)

$L_p = N_b V_s = 16 \times 1/12 = 1.333\ ft$

$$\Delta P_2 = \frac{0.00314 \times (2{,}000)^2 \times 1.333 \times 0.58 \times 1}{5.22 \times 10^{10} \times 0.0214 \times 0.000722 \times 1} = \underline{0.012\ psi}$$

(c) Pressue loss due to "orifice"

This is the divider separating HE 1 from HE 2. It is a 2-inch high barrier on all sides of the duct. As in the "orifice" between combustor and HE 1, this loss is found to be negligible.

(d) Total pressue loss in the duct

$\Delta P_T = 1.23 + 0.012$ $\underline{\underline{1.25\ psi}}$

Moist BG reaches HE 2 at $\underline{\underline{14\ psig = 28.7\ psia}}$

$$\boxed{P_{BG} = 28.7\ psia}$$

3. Pressure Drop on Tube Side (Air)

$f = 0.00027$ (ref. 4)

$$\Delta P_{bank} = \frac{0.00027 \times (20{,}950)^2 \times 2 \times 1}{5.22 \times 10^{10} \times 0.0583 \times 0.00101 \times 1}$$

$= 0.078$ psi/2 ft long bank

Each return bend is equivalent to 4.2 ft straight pipe, and so are assumed the headers.

$\Delta P_{bend} = 4.2 \times 0.5 \times 0.078 = 0.164$ psi/bend

Thus

$\Delta P_{t,total} = 16 \times 0.078 + 17 \times 0.164 = \underline{\underline{4.03\ psi}}$

Thus, the air from the engines that reaches HE 1 at 40 psia, will leave it at 21.27 psig = 35.97 psia.

$$\boxed{P_{Air} \simeq 36 \text{ psia}}$$

4. Weight

Cooling tubing of 316-SS (13.2 lb/bank)	211 lb
Duct of Hastelloy C	61
Total	272 lb

5. Check of HE 1 for Other Conditions

The heat transfer area is calculated for conditions which differ from the design ones in order to determine if the above unit is adequate. We look for a condition where a relatively large amount of air comes in at a low temperature. This would be the case in an emergency dive, with engines near idle, during the last portion of the landing approach, when altitude is decreasing from 2,500 ft to sea level. If this approach lasts for 1 minute, 30 cfm of ballast gas at 15.1 psia are required. Since the engines are near idle, the temperature of the air is 250°F.

$V_M = 398\ ft^3/lb\text{-mole}$

$V_{Air} = 32.3$ cfm

$N_{Air} = 0.0811$ lb-moles/min

$W_{Air} = 2.34$ lb/min = 141 lb/hr

$W_{H_2O\text{-vapor, 100\% conv.}} = 0.204$ lb/min

$W_{reaction\ fuel} = 0.175$ lb/min

$W_{MBG} = 2.52$ lb/min = 152 lb/hr

$W_{Dry\ BG} = 2.32$ lb/min = 139 lb/hr

The situation in HE 1 becomes:

Air Out

$t = ?$

$h = ?$

Air In

$W_A = 2.34$ lb/min = 141 lb/min

$t = 250°F$

$h = 52$ BTU/lb

HE 1

R.F. Out

$t = 300°F$

$h = 147.5$ BTU/lb

Reaction Fuel In

$W_{RF} = 0.175$ lb/min

$t = 150°F$

$h = 58.8$ BTU/lb

BG In

$W_{BG} = 2.5$ lb/min

$t = 1337°F$

$Q_{BG\ in} = 1{,}125$ BTU/min

BG Out

$W_{Dry\ BG} = 2.315$

$W_{H_2O\text{-}V} = 0.204$ lb/min

$t = ?$

$Q = ?$

$$Q_{BG\ in} = 2.315 \times 335 + 0.204 \times 1{,}711 = 1{,}125 \text{ BTU/min}$$

We assume different temperatures for the air leaving HE 1 and calculate the heat load on HE 1 and the temperature of the exit BG. Then check for heat transfer surface requirements. Thus:

$$\Delta Q_{HE\ 1,\ t\ air\ out} = W_{air}\,(h_{t\ air\ out} - 52)$$

$$Q_{BG\ out} = Q_{BG\ in} - \Delta Q_{HE\ 1,\ t\ air\ out}$$

and the temperature of BG leaving HE 1 is calculated.

Temperature of Air Leaving HE 1 $t_{air\ out}$, °F	Heat Content $h_{t\ air\ out}$, BTU/lb	Heat Load on HE 1 $Q_{HE\ 1,\ t\ air\ out}$ BTU/min	BTU/hr	Heat in Exit BG, $Q_{BG\ out}$, BTU/min	Temperature of Exit BG $t_{BG\ out}$, °F
1,112	276	525	31,500	600	599
1,200	297.4	575	34,500	550	527
1,300	324.5	639	38,300	486	397

First we check HE 1 for air leaving at 1,112°F, as this will allow us to use the viscosity, conductivity and specific heat data already generated for the SST, Case I.

Hot Fluid		Cold Fluid	Difference
1,337	Higher temperature	1,112	225
599	Lower temperature	250	349
738	Difference	862	124

LMTD = 283°F

R = 0.86 S = 0.79 F_T = 0.6 (Ref. 5)

Δt = 170°F

Duct Side

G_s = 422 lb/(ft^2)(hr)

Re_s = 382

j_f = 6.9 (ref. 1)

h_f = 2.9 BTU/(hr)(ft^2)(°F)

h_f' = 2.9 BTU/(hr)(ft^2)(°F)

$(r_e - r_b)(h_f'/k_t Y_b)^{0.5} = 0.135$

$\Omega \simeq 1$ (ref. 3)

h_{fi}' = 9.6 BTU/(hr)(ft^2)(°F)

U_{Di} = 3.7 BTU/(hr)(ft^2)(°F)

Tube Side

G_t = 4,400 lb/(ft^2)(hr)

Re_t = 3,420

j_{hi} = 14

h_i = 5.95 BTU/(hr)(ft^2)(°F)

h_i' = 5.85 BTU/(hr)(ft^2)(°F)

[k_t = 11.9 BTU/(hr)(ft^2)(°F/ft)]

$A_{iT} = 50\ ft^2$

$N_b \simeq 12$ banks

Since HE 1 has 16 banks, the existing surface will allow the preheating of air to a higher temperature than the 1,112°F checked here, and the design is adequate.

d. Heat Exchanger No. 2 (HE 2)

HE 2 makes use of fuel being pumped to the engines as a coolant for removal of sensible heat from the BG leaving HE 1, and for removal of the heat of condensation released by part of the water and fuel present in the BG stream.

Cross-section: 2 ft wide x 1 ft high

Tubing: $OD_t = 0.5"$ wall = 0.02" $ID_t = 0.46"$

Fins: $b_f = 0.125"$ $th_f = 0.035"$ $N_f = 8$ per inch

$OD_f = 0.75"$ $r_e = 0.375"$ $r_b = 0.25"$

Arrangement: square pitch, $N_{t/b} = 16$

$S_T = S_L = V_s = 0.75"$

1. Heat Transfer

$Q = 260{,}000$ BTU/hr

$W_{BG,\ Avg} = 700$ lb/hr

$W_{fuel,\ coolant} = 5{,}710$ lb/hr

BG enters at 1,234°F and leaves at 85°F

Fuel coolant enters at 60°F and leaves at 150°F

Mean temperature

Hot Fluid		Cold Fluid	Difference
1,234°F	Higher temperature	150°F	1,084°F
85°F	Lower temperature	60°F	25°F
1,149°F	Difference	90°F	1,059°F

$LMTD = 281°F$

$R = 12.8 \qquad S = 0.08 \qquad F_T = 0.978$ (extrapolation, ref. 5)

$\underline{\Delta t = 275°F}$

<u>Caloric Temperatures</u>

$\Delta t_c/\Delta t_h = 0.023$

$K_c = 0.075$ for JP-4 (ref. 6)

$F_c = 0.24$ (ref. 6)

Thus:

$T_c = 85 + 0.24 \times 1{,}149 = 361°F$

$t_c = 60 + 0.24 \times 90 = 82°F$

<u>Data for BG at 361°F at 28.7 psia</u>

The data are an average of moist BG containing fuel vapor and saturated BG.

$\mu_{avg} = 0.057$ lb/(ft)(hr)

$C_{p\ avg} = 0.257$ BTU/(lb)(°F)

$k_{avg} = 0.0204$ BTU/(hr)(ft²)(°F/ft)

$$\left(\frac{C_p\mu}{k}\right)^{1/3} = 0.896$$

$\phi = 1$

$V_M = 306.8\ ft^3$/lb-mole

$\overline{V}_{F-V} = 2.4\ ft^3/lb$

$V_{avg} = 121.4$ cfm

$\rho_{avg} = 0.096\ lb/ft^3 \qquad S = 0.00154$

<u>Data for liquid fuel (coolant) at 82°F</u>

$\mu = 4$ lb/(ft)(hr)

$C_p = 0.664$ BTU/(lb)(°F)

$k = 0.0721$ BTU/(hr)(ft²)(°F/ft)

$$\left(\frac{C_p \mu}{k}\right)^{1/3} = 3.33$$

$\phi = 1.07$

$\rho = 47.13$ lb/ft^3

S = 0.7553

<u>Heat transfer: duct side (BG)</u>

$A_f = 0.33$ ft^2/ft

$A_o = 0.094$ ft^2/ft

$P_p = 5.44$ ft/ft

$d_{es} = 0.049$ ft

$a_s = 0.48$ ft^2

$G_s = 1{,}460$ lb/(ft^2)(hr)

$Re_s = 1{,}260$

$j_f = 16$ (ref 1)

$h_f = 6$ BTU/(hr)(ft^2)(°F)

$h_f' =$ <u>5.94 BTU/(hr)(ft^2)(°F)</u>

<u>Heat transfer: tube side, fuel coolant</u>

$d_t = 0.0383$ ft

$a_t = 0.001154$ ft^2

$A_{t/b} = 16\ a_t = 0.0185$ ft^2/bank

$G_t = 309{,}000$ lb/(ft^2)(hr)

$v = 1.82$ ft/sec

$Re_t = 2{,}960$

$j_{hi} = 10$ (ref. 2)

$h_i = 67$ BTU/(hr)(ft^2)(°F)

$h_{di} = 500$ BTU/(hr)(ft^2)(°F)

$h_i' =$ <u>59 BTU/(hr)(ft^2)(°F)</u>

<u>Fin efficiency, overall design coefficient, area and number of banks</u>

$k_t = 10.4$ BTU/(hr)(ft^2)(°F/ft)

$r_e/r_b = 1.5$

$Y_b = 0.00146$ ft

$$(r_e - r_b)\sqrt{\frac{h_f}{k_t Y_b}} = 0.2$$

$\Omega = 0.98$ (ref. 3)

$a_i = 0.12$ ft^2/ft

$A_{it/b} = 16\ a_i \times 2 = 3.85$ ft^2/bank

h'_{fi} = <u>20.5 BTU/(hr)(ft^2)(°F)</u>

$U_{Di} = 15.2$ BTU/(hr)(ft^2)(°F)

Overall design coefficient = 15.2 BTU/(hr)(ft^2)(°F)

$A_{iT} = 62.2$ ft^2

$N_b = 16.2$ banks

Number of banks = 17

2. <u>Pressure Drop on Duct Side (BG)</u>

The losses are caused by:

condensation (loss of volume)

cooling coils

flow through the "orifice"

(a) Pressure loss due to condensation of fuel vapor and part of water vapor

P = 28.7 psia

total of gases entering = 0.4087 lb-moles/min

fuel vapor condensed = 0.00179 lb-moles/min water vapor condensed = 0.439 ÷ 18 = 0.0244 lb-moles/min

total condensed = 0.0262 lb-moles/min

$$\Delta P_{condensation} = \frac{0.0262 \times 28.7}{0.4087} = \underline{1.84 \text{ psi}}$$

(b) Pressure loss due to friction with cooling coils

$V_{NF} = 0.066 \text{ ft}^3$

$S_F = 13.49 \text{ ft}^2$

$D'_{ev} = 0.0196 \text{ ft}$

$$\left(\frac{D'_{ev}}{S_T}\right)^{0.4} = 0.63$$

$$\left(\frac{S_L}{S_T}\right)^{0.6} = 1$$

$f = 0.0034$ (ref. 1)

$L_p = 1.063 \text{ ft}$

$\Delta P = \underline{0.003 \text{ psi}}$

(c) The pressure loss due to the orifice is not calculated because calculations given above show that it is negligible.

(d) Total pressure loss for BG

$\Delta P_T = 1.84 + 0.003$

$$\boxed{\Delta P_{T,BG} = 1.85 \text{ psi}}$$

Consequently, BG reaches the drier with

<u>12.2 psig = 26.9 psia</u>

3. <u>Pressure Drop on Tube Side (Coolant)</u>

$f = 0.00039$ (ref. 4)

$\Delta P_{bank} = 0.046$ psi/2 ft bank

a return bend is equivalent to 2.8 ft of tubing

$$\Delta P_{bend} = 2.8 \times 0.5 \times 0.046 = 0.0646 \text{ psi}$$

$$\Delta P_{t,\ total} = 17 \times 0.046 + 18 \times 0.0646$$

$$\boxed{\Delta P_{tubing} = 1.95 \text{ psi}}$$

4. Weight

Cooling tubing (11.22 lb/bank), 316-SS	191 lb
Duct, Hastelloy C	61
Total	252 lb

5. Check of HE 2 for Other Conditions

Other conditions under which the design of HE 2 should be checked include the following:

(a) the same conditions as those checked in HE 1 in order to establish whether the coolant fuel flow near idle conditio is sufficient, or if recirculation of the fuel between the tank and HE 2 might be necessary. In addition, the heat transfer surface should be teste

(b) The initial climb situation, during which the coolant fuel temperature is 95°F.

These checks were not made as a part of the present study.

e. Drier

The cross-section, dictated by the L/D criteria for H-Zeolon, is 1 x 1 ft.

$Q = 25{,}000$ BTU/hr

$W_{BG,\ Avg} = 11.166$ lb/min $= 670$ lb/hr

$W_{H_2O,\ coolant} = 8.3$ lb/min $= 500$ lb/hr

BG enters at 85°F, leaves at 100°F

Water enters at 40°F, leaves at 90°F

Therefore: Parallel flow.

Aluminum tubing: $OD_t = 0.375$ inch wall $= 0.016"$ $ID_t = 0.343$

Fins: b_f = 3/16 inch th_f = 0.035" N_f = 8 fins/inch

OD_f = 0.75 inch r_e = 0.375" r_b = 0.1875 inch

Arrangement: square pitch, $N_{t/b}$ = 16 tubes/bank

$S_T = S_T = Y_s = 0.75$ inch

1. Heat Transfer

Hot Fluid		Cold Fluid	Difference
100°F	Higher temperature	90°F	10°F
85°F	Lower temperature	40°F	45°F
15°F	Difference	50°F	35°F

LMTD = 23.3

R = 0.3   S = 0.8 F_T = 0.96 (assuming cross-flow correctic can be applied)

Δt = <u>22.3°F</u>

T_c = 93°F for BG

t_c = 65°F for cooling water

(a) Data

<u>For BG at 93°F and 25 psia</u>

μ = 0.0185 cps = 0.045 lb/(ft)(hr)

C_p = 0.24 BTU/(lb)(°F)

k = 0.0155 BTU/(hr)(ft^2)(°F/ft)

$$\left(\frac{C_p\mu}{k}\right)^{1/3} = 0.886$$

$\phi = 1$

V_M = 242 ft^3/lb-mole

$V_{BG,\ in}$ = 242 x 0.3825 = 92.6 cfm

$V_{BG,\ out}$ = 242 x 0.3663 = 88.6 cfm

$V_{BG,\ avg} = 90.6$ cfm

$\rho_{avg} = 0.123$ lb/ft^3 $S_{avg} = 0.001975$

For cooling water at 65°F

$\mu = 1.08$ cps $= 2.614$ lb/(ft)(hr)

$\mu_{wall} = 0.9$ cps

$$\phi = \left(\frac{\mu}{\mu_{wall}}\right)^{0.14} = 1.02$$

$\rho = 62.3$ lb/ft^3 lb/ft^3 $S = 1$

(b) Heat Exchange

Heat transfer: duct side, BG

$A_f = 0.451$ ft^2/ft

$A_o = 0.0707$ ft^2/ft

$P_p = 7.44$ ft/ft

$d_{es} = 0.0446$ ft

$a_s = 0.36$ ft^2

$G_s = 1{,}860$ lb/(ft^2)(hr)

$Re_s = 1{,}860$

$j_f = 21.5$ (Ref. 1)

$h_f = 6.6$ BTU/(hr)(ft^2)(°F)

$h'_f = \underline{6.5}$ BTU/(hr)(ft^2)(°F)

Heat transfer: tube side, cooling water

$d_t = 0.0286$ ft

$a_t = 0.000642$ ft^2

$A_{t/b} = 16\ a_t = 0.0103$ ft^2/bank

$G_t = 48{,}700$ lb/(ft^2)(hr)

$Re_t = 533$

$v = 0.217$ ft/sec

$h_{ii} = 80$ BTU/(hr)(ft^2)(°F) (Ref. 7)

factor = 1.1

h_i = factor x h_{i1} = 88 BTU/(hr)(ft^2)(°F)

h_i' = 74.8 BTU/(hr)(ft^2)(°F)

Heat transfer: fin efficiency, design coefficient, area and number of banks

for aluminum k_t = 116.7 BTU/(hr)(ft^2)(°F/ft)

$r_e/r_b = 2$

Y_b = 0.00146 ft

$(r_e - r_b)\,(h_f'/k_t Y_b)^{0.5} \simeq 0.1$

$\Omega \simeq 1$ (ref. 3)

a_i = 0.0898 ft^2/ft

$A_{i/b} = 16\ a_i \times 1 = 1.44$ ft^2/bank

h_{fi}' = 38 BTU/(hr)(ft^2)(°F)

$U_{Di} = h_{fi}' \times h_i'/(h_{fi}' + h_i')$ = 25.2 BTU/(hr)(ft^2)(°F)

Overall design coefficient = 25.2 BTU/(hr)(ft^2)(°F)

$$A_i = \frac{25,000}{25.2 \times 22.3} = 44.5 \text{ ft}^2$$

N_b = 44.5/1.44 = 31

Number of banks = 31

2. Amounts of Drying Agents

The two agents selected are $CaCl_2$ for high capacity and H-Zeolon for high efficiency. The method of calculation is the same as for the SST.

(a) Calcium Chloride Section

Conditions: 100°F exit temperature, ~ 14.7 psia

Efficiency at 1900 hr^{-1} SV = equilibrium ÷ 0.95 = 1 mm Hg

$$\therefore \text{ efficiency} = \frac{1 \text{ mm}}{} \bigg| \frac{1 \text{ atm}}{760 \text{ mm}} \bigg| \frac{10^6}{} = \underline{1320 \text{ ppm}}$$

Gas Flow

BG flow in drier bed at design conditions

$$V_{sBG,\ Ave} = \underline{90.6\ ft^3/min}$$

Bed Volume and Weight

for 1900 hr^{-1} SV:

$$\text{Volume of } CaCl_2 = \frac{90.6\ ft^3}{min} \left| \frac{60\ min}{hr} \right| \frac{hr}{1900}$$

$$= 2.86\ ft^3$$

Weight of $CaCl_2$

$$2.86\ ft^3 \times 51\ lb/ft^3$$

$$\boxed{W = 146\ lb}$$

Concentration of Water Entering $CaCl_2$

N_2	79 lb-moles
CO_2	10.5 lb-moles
O_2	5.25 lb-moles

$$H_2O = \frac{10.5\ \text{lb-moles}}{} \left| \frac{0.291\ lb/min}{0.291 + 0.439\ \text{condensed}} \right. = 4.19\ \text{lb-moles}$$

(allowing for removal of H_2O in HE 2)

$$H_2O\ \text{conc.} = \frac{4.19}{98.47} \times 100 = 4.25\%$$

$$= \underline{42{,}500\ ppm}$$

Concentration of Water Leaving $CaCl_2$ = 1320 ppm

Water Removed by $CaCl_2$

$$\frac{42{,}500 - 1{,}320}{42{,}500} \times 100 = \underline{97.2\%\ \text{removal}}$$

Useful Life of $CaCl_2$ (no regeneration)

water produced in <u>average</u> flight = 4.23 lb removed in drier:

$$\frac{0.115 \text{ lb}}{100 \text{ ft}^3} \Big| \; 11.02 = 1.267 \text{ lb}$$

removed in $CaCl_2$:

$$0.972 \times 1.27 = 1.232 \text{ lb}$$

$CaCl_2$ capacity (at 80% of lit. value)

$$146 \text{ lb} \Big| \frac{80}{100} \Big| \frac{30 \text{ lb } H_2O}{100 \text{ lb agent}} = 35 \text{ lb}$$

No. of flights = 35 ÷ 1.232 = 28.4

$$\text{No. of hours} = 28.4 \Big| \frac{155 \text{ min}}{\text{flight}} \Big| \frac{\text{hour}}{60 \text{ min}} = 73 \text{ hr}$$

$$\boxed{\text{Useful life} = 73 \text{ hr}}$$

(b) Zeolite Section

Bases: 10 ppm efficiency at following conditions

100°F and ~ 1 atm.

$L/D > 1$

linear velocity < 100 ft/min

Norton's H-Zeolon, 1/16" dia

capacity = 0.030 lb/lb agent

Velocity in duct

square duct, 1 ft x 1 ft

$90.6 \text{ ft}^3/\text{min} \div 1 = \underline{90.6 \text{ ft/min}}$

Weight of Agent (use 80% of stated capacity)

$$W = \frac{\text{lb } H_2O}{\text{flight}} \Big| \frac{\text{flight}}{\text{hours}} \Big| \frac{\text{hours}}{\text{continuous run}} \Big| \frac{}{\text{capacity} \times 0.80}$$

$$W = 1.267 - 1.232 \Big| \frac{60}{155} \Big| \frac{50}{0.03} \Big| \frac{}{0.80}$$

$$W = \underline{28.2 \text{ lb agent}}$$

Volume of Agent

bulk density = 38.5 lb/ft3

volume = 28.2 ÷ 38.5 = 0.733 ft^3

$\underline{V = 0.733\ ft^3}$

Length of Bed and L/D

$L = 0.733 \div 1\ ft^2 = 0.733\ ft$

equivalent circle diameter

$D = (4 \div \pi)^{0.5} = 1.13\ ft$

$L/D = 0.733 \div 1.13 = 0.65$

Adjust to L/D > 1 (say L/D = 1.1)

Length = 1.13 x 1.1 = 1.25 ft

$\text{Volume} = 0.733 \times \frac{1.25}{0.733} = 1.25\ ft^3$

Weight = 1.25 x 38.5 = 48.1 lb

W = 48.1 lb

Hours of Useful Life (no regeneration)

50 x 48.1 ÷ 28.2 = 85 hours

(c) Summary of Drying Agents

Agent	$CaCl_2$	Zeolite	Combined
volume, ft^3	2.86	1.25	4.11
weight, lb	146	48.1	194
length, ft	2.86	1.25	4.11
hours (no regen.)	73	85	73

Thus, the space velocity criterion for $CaCl_2$ dictates a volume of agent sufficient to provide 73 hours of protection between regenerations, and the L/D criterion for zeolite dictates a somewhat longer period of protection.

3. Volume and Length of the Drier

(a) Volume occupied by one bank of tubing

$Vol_{t/b}$ = 39 $inch^3$/bank = 0.0226 ft^3/bank

Total volume occupied by 31 banks = 0.71 ft^3

Since 97% of water is removed by $CaCl_2$, the same percentage of heat has to be removed in the $CaCl_2$ section of the drier. Therefore, 29 banks of cooling tubing are located in that section. The remaining 2 banks are in the zeolite section.

(b) $CaCl_2$ Section

29 banks of tubes occupy 0.66 ft^3

Section volume = 2.86 + 0.66 = 3.52 ft^3

Length of section = 3.52 ft

Distance between banks ≃ 0.12 ft ℄ to ℄ C = 1.45 inch ℄ to ℄

(c) Zeolite Section

2 banks of tubing occupy 0.05 ft^3

Section volume = 1.25 + 0.05 = 1.3 ft^3

Section length = 1.3 ft

Distance between banks = 0.65 ft ℄ to ℄

(d) Total Length of Drier

3.52 + 1.3 = 4.82 ft is the length of drier occupied by the desiccants and tubing. Part of the upstream converging cone may be occupied by the $CaCl_2$ and the dust filters may be located in downstream cone. Thus, the total length of drier will be approximately 5.5 ft.

4. Pressure Losses

(a) Pressure loss inside tubing

f = 0.00096

ΔP_t = 0.0015 psi/per 1 ft bank

A return bend is equivalent to 2 ft tubing

ΔP_{bend} = 0.003 psi/bend

$\Delta P_{t,\ total}$ = 31 x 0.0015 + 32 x 0.003 = 0.15 psi

$\boxed{\Delta P_{T\text{-}water} = 0.15 \text{ psi}}$

(b) Pressure loss in the duct side

It comprises the following losses:

- Pressure loss due to gradual contraction (upstream)
- Pressure loss due to friction in packed bed
- Pressure loss due to friction with cooling tubing
- Pressure loss equivalent to volume of water removed
- Pressure loss due to gradual contraction (downstream)

Contraction loss, upstream:

$$\Delta P_1 = 0.02 \times 12.2 = \underline{0.25 \text{ psi}}$$

Packed bed friction loss:

1/16" particles fall within the 8 to 14 mesh size

$$R_e' = \frac{0.264}{0.045} \times \frac{670}{1} \simeq 4{,}000$$

$$(f/F_f) = 0.0435 \quad \text{(ref. 8)}$$

$$\Delta P_2 = \frac{0.0435 \times 670^2 \times 4.82}{4{,}880 \times 0.123 \times 144} = \underline{1.09 \text{ psi}}$$

Cooling coils friction loss:

$$V_{NF} = 0.04 \text{ ft}^3$$

$$S_F = 8.35 \text{ ft}^2$$

$$D_{ev}' = 0.0192 \text{ ft}$$

$$\left(\frac{D_{ev}'}{S_T}\right)^{0.4} = 0.623$$

$$\left(\frac{S_L}{S_T}\right)^{0.6} = 1$$

$$f = 0.0034 \quad \text{(ref. 1)}$$

$$L_p = \frac{0.75}{12} \times 31 = 1.94 \text{ ft}$$

$$\Delta P_3 = \underline{0.01 \text{ psi}}$$

Loss equivalent to volume of water removed:

Assuming all water is removed half way through the bed, the total gas pressure at this point is

$$P = 26.9 - (\Delta P_1 + 0.5 \ \Delta P_2 + \Delta P_3) = 26.1 \text{ psia}$$

total gases entering drier = 0.3825 lb-moles/min

water removed = 0.01618 lb-moles/min

$$\Delta P_4 = \frac{0.01618}{0.3825} \times 26.1 = \underline{1.10 \text{ psi}}$$

Contraction loss, downstream:

$$P = 12.2 - (\Delta P_1 + \Delta P_2 + \Delta P_3 + \Delta P_4) = 9.75 \text{ psig}$$

$$\Delta P_5 = 0.02 \times 9.75 = \underline{0.2 \text{ psi}}$$

Total pressure loss in the duct:

The pressure losses due to the 3 screens and due to the dust filter are so small that they are neglected. Thus

$$\Delta P_T = \sum \Delta P_i = 0.25 + 1.09 + 0.01 + 1.10 + 0.2 = 2.65 \text{ psi}$$

Pressure loss in drier = 2.65 psi

Consequently, the BG leaves the drier at 9.55 psig = 24.25 psia.

The above pressure of 24.25 psia is almost double the pressure assumed for the fuel tank, which is 12.62 psia. However, we assumed in our calculations that all the components are in the same duct with very little loss due to enlargements and contractions, while in the actual design they might be separated by pipes and be subject to such losses. Also, as mentioned under Case Analysis I, pressure losses caused by control valves, instruments, and other items are not included. There is always the possibility that, when the design is optimized, a different pipe size will be used for the coils in HE 1, and the inlet pressure regulator may be set at a lower pressure than the 40 psia assumed. Therefore, some of the excess of pressure is regarded as a degree of freedom to be used as a parameter in system design.

5. Weight

(a) Drier

The drier is made of aluminum, except for the screens. Duct walls are assumed to be 1/8 inch thick and the volume of the material is 396 inch^3, thus the weight of the walls is

$$396 \times 0.099 = \underline{39.2 \text{ lb}}$$

To summarize:

Weight of outer walls	39.2 lb
Weight of cooling tubing (2.15 lb/bank)	66.7
Weight of screens (3)	5.1
Weight of desiccants	194
Total	305 lb

(b) Dust Filter

This is a glass fiber mat filter located in the downstream end of the drier. It removes dust from the BG. Its weight is negligible.

(c) Water Tank

The amount of water per flight is calculated to be 52 lbs or less than 1 ft^3. A small heater, to prevent the freezing of water is included, as well as a variable speed pump to deliver the water to the drier and to the combustor.

Weight of tank	6 lb
Weight of heater	1
Weight of water	52
Weight of pump	11
Total	70 lb

REFERENCES

1. Kern, Process Heat Transfer, pp 555, 838, McGraw-Hill Book Company, New York, 1950

2. Kern, ibid, p 834

3. Kern, ibid, p 542

4. Kern, ibid, p 836

5. Kern, ibid, p 549

6. Kern, ibid, p 827

7. Kern, ibid, p 835

8. Calculating Pressure Drop Through Packed Beds, Publication GB4A, Aluminum Company of America, Pittsburgh, Pa., 1960

APPENDIX I

CALCULATIONS FOR C-141

TRANSPORT AIRCRAFT

APPENDIX I

CALCULATIONS FOR C-141 AIRCRAFT

I. Calculation of Average Temperature of Bleed Air

The average temperature of the bleed air for the entire flight is calculated as follows:

The bleed air temperatures for the various operations at different altitudes (Table XVII) are multiplied by the duration of the operation. The sum is then divided by the duration of the entire flight yielding the average temperature of engine bleed air.

$$3 \times 750 = 2{,}250$$
$$20 \times 660 = 13{,}200$$
$$7 \times 530 = 3{,}710$$
$$363 \times 500 = 181{,}500$$
$$4 \times 270 = 1{,}080$$
$$4 \times 230 = 920$$
$$2 \times 280 = 560$$
$$\overline{203{,}220}$$

$$\frac{203{,}220}{403} = 504.3°F$$

$$\boxed{\text{Average Temperature} = 500°F}$$

II. Calculation of Bleed Air Pressure During Design Portion of Descent

The data for engine bleed air pressure during the descent (Table XVII) are plotted vs altitude in Figure I-1. Then a graphical integration, between 4,000 and 10,000 ft altitude is performed, and the design pressure of bleed air is obtained.

Altitude, ft	Pressure, psia
9,500	32.5
8,500	33.45
7,500	34.4
6,500	35.4
5,500	36.4
4,500	37.4
	209.55

$$\frac{209.55}{6} = 34.93 \text{ psia}$$

$$\boxed{\text{Design pressure of air} = 34.9 \text{ psia}}$$

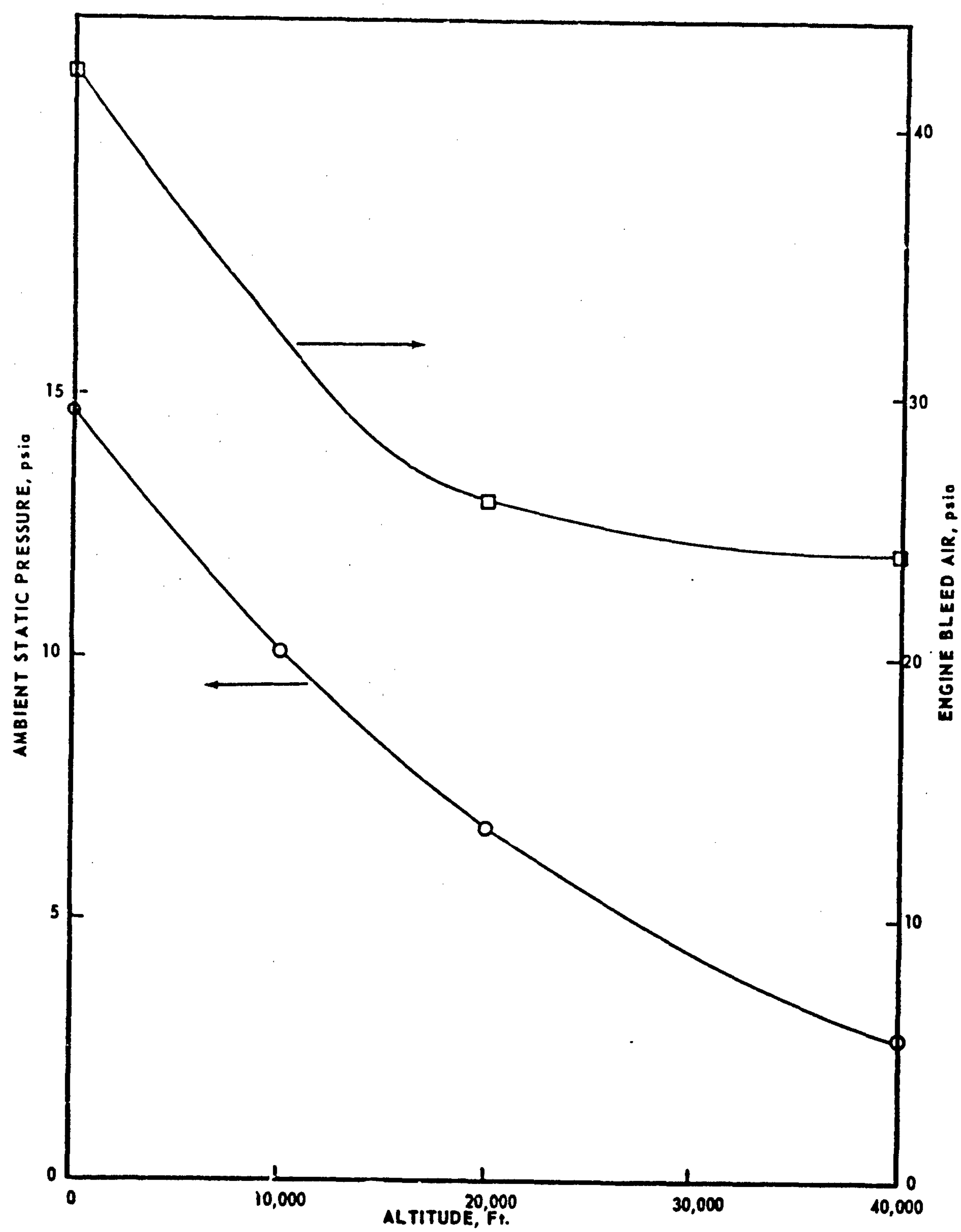


FIGURE I-1. VARIATION OF AMBIENT STATIC AND ENGINE BLEED AIR PRESSURES WITH ALTITUDE

III. Calculation of Heat and Mass Balance for Design Conditions

a. General

The bases, data, and assumptions are indicated in Section V-8-h. The method of calculation is detailed in Appendix E. Following are listed the results of these calculations.

b. Results

1. Quantities of Reactants and Products

Molar volume of gases in FT (at 45°F and 11.93 psia)	454 ft^3/lb-mole
Air Requirement (volume)	552 cfm
(moles)	1.216 lb-moles/min
(weight)	35.13 lb/min = 2,108 lb/hr
Water formed (WF)	2.3 lb/min = 138 lb/hr
Water condensed in HE 2	1.38 lb/min = 83 lb/hr
Water removed in gas drier (WRGD)	0.92 lb/min = 55 lb/hr
Reaction fuel required	2.62 lb/min = 157 lb/hr
Unreacted fuel (UF)	0.83 lb/min = 50 lb/hr
Unreacted fuel condensed in HE 2	0.79 lb/min = 47.5 lb/hr
Uncondensable unreacted fuel (UUF)	0.04 lb/min = 2.5 lb/hr
Air-fuel mixture = Moist BG (MBG)	37.75 lb/min = 2,265 lb/hr
Dry BG (DBG = MBG - WF-UF)	34.62 lb/min = 2,077 lb/hr
Ballast gas (BG = DBG + UUF) to FT	34.66 lb/min = 2,080 lb/hr
Saturated BG (SBG = BG + WRGD)	35.58 lb/min = 2,135 lb/hr
Cooling water required	24.5 lb/min = 1,470 lb/hr
Fuel as coolant required	60 lb/min = 3,600 lb/hr

2. Molar Quantities

(a) Air-Fuel Mixture

Air	= 1,216 lb-moles/min	
Oxygen	= 0.255	"
Nitrogen	= 0.960	"

Fuel	=	0.0187 lb-moles/min
Air-Fuel Mixture	=	1.234 lb-moles/min

(b) Moist Ballast Gas

Nitrogen	=	0.960	lb-moles/min
Carbon dioxide	=	0.128	"
Oxygen	=	0.064	"
Dry ballast gas	=	1.152	"
Water	=	0.128	"
Unreacted fuel	=	0.00595	"
Moist ballast gas	=	1.285	"

(c) Ballast Gas

Dry ballast gas	=	1.152	lb-moles/min
Uncondensable unreacted fuel	=	0.00030	"
Ballast gas	=	1.152	"

(d) Saturated Ballast Gas

Moist ballast gas	=	1.285	lb-moles/min
Condensed water	=	0.0768	"
Condensed unreacted fuel	=	0.00565	"
Saturated ballast gas	=	1.203	"

Water removed in drier = 0.0509 lb-moles/min

3. Heat Loads

(a) Air-fuel mixture

Heat content	10,080 BTU/min
Temperature	971°F $\simeq$ 522°C

(b) Combustor (water cooled)

Heat of reaction	33,100 BTU/min
Heat removed by coolant	26,800 BTU/min = 1,608,000 BTU/hr
Heat content in MBG	16,300 BTU/min

(c) HE 1 (Air Cooled)

Heat transferred to air	7,860 BTU/min = 472,000 BTU/hr
Actual heat transferred to fuel	44 BTU/min
Heat content in MBG	8,440 BTU/min
Temperature of MBG leaving HE 1	611°F

Design heat transferred to fuel 160 BTU/min = 9,600 BTU/hr

(d) HE 2 (Fuel Cooled)

Heat removed by coolant	6,900 BTU/min = 414,000 BTU/hr
Heat content of SBG	1,446 BTU/min
Temperature of SBG leaving HE 2	85°F
Temperature of coolant (fuel) leaving HE 2	275°F

(e) Gas Drier (Water Cooled)

Net heat of absorption	412 BTU/min
Heat removed by coolant	1,293 BTU/min = 77,600 BTU/hr
Heat content of BG leaving drier	565 BTU/min
Temperature of BG	100°F

IV. Design Calculations, Equipment for C-141

a. Air and Fuel Feed

1. Main Air Pipe (from HE 1 to VC Outlet)

   W_{air} = 35.13 lb/min = 2,108 lb/hr

   t = 1,112°F

   P = 31.0 psia

   V_{air} = 661 cfm

   Pipe: 316-SS, OD = 3.5 in ID = 3.37 in

   G_t = 34,000 lb/(ft^2)(hr) Re_t = 101,000

   f = 0.000175[1] ΔP = 1.62 psi/100 ft pipe

   length = 7 ft equivalent length = 49 ft

   ΔP = 1.62 x 49/100 = 0.8 psi

   P_{air} reaching annulus of outlet pipe connecting vaporization chamber and combustor = 30.2 psia and P_{air} reaching VC inlet pipe = 30.6 psia.

   W_{pipe} = 7 x (2.43 lb/ft) = 17 lb

2. Vaporization Chamber Inlet Pipe

   Air for atomization = 3 scfm $\Leftrightarrow$ 14.5 lb/hr

   Pipe: 316-SS, OD = 1 in ID = 0.95 in

   G_t = 2,950 lb/(ft^2)(hr) Re_t = 2,470

   f = 0.0004[1] ΔP = 0.1 psi/100 ft pipe

   length = 8 ft equivalent length = 14 ft

   ΔP = 0.1 x 14/100 = 0.01 psi

   P_{air} reaching nozzle = 30.5 psia

   W_{pipe} = 8 x (0.27 lb/ft) = 2.1 lb

3. Reaction Fuel Pipe

W_F = 2.62 lb/min = 157 lb/hr

t = 300°F

V_F = 0.475 gpm = 28.5 gph

Pipe: 316-SS, OD = 0.5 in ID = 0.49 in

G_t = 120,000 lb/(ft^2)(hr) Re_t = 5,450

f = 0.00033[1] ΔP = 0.34 psi/100 ft pipe

length = 12 ft equivalent length = 15 ft

ΔP = 0.05 psi W_{pipe} = 12 x (0.027 lb/ft) = 0.3 lb

4. Vaporization Chamber (VC)

(a) Nozzle

Fuel delivery = 0.475 gpm = 28.5 gph

Nozzle set-up No. 42 (Spraying Systems Co., Bellwood, Illinois) is selected

Weight nozzle = 0.5 lb

Spray angle = 20°

Minimum cone length = 39 inch

(b) VC Dimensions and Weight

Referring to diagram in Appendix H, we find by trigonometrical relationship:

Y = 41.8 in R = 7.38 in

Z = 27.5 in X = 25 in

L_{VC} = 64 in = 5.33 ft

ID_{VC} = 14.75 in

For Hastelloy C: wall thickness = 0.1 inch

OD_{VC} = 14.95 in W_{VC} = 46 lb

(c) Pressure Drop in VC

The total pressure drop in the nozzle and vaporization chamber is estimated at 20% of the gage pressure of the air reaching the nozzle. Thus

$$\Delta P = 0.2 \times 15.8 = 3.16 \text{ psi} \sim 3.2 \text{ psi}$$

and the fuel-rich mixture leaves VC at 27.3 psia.

5. Heat for Vaporization Chamber

During startup, air is available at 250°F and fuel at 90°F. The amount of heat to supply via electric heaters to insure total vaporization of the fuel is

$$Q = 14.7\ (246.7\text{-}52.2) + 5.7\ (688.2\text{-}12.8) = 6{,}670 \text{ BTU/hr}$$

The required heater capacity is 2 KW, however, occasionally a larger heater may be necessary. Hence, a 4 KW heater is assumed, and its weight is estimated at 7 lb.

6. VC Outlet Pipe

This double pipe is designed in such a way that both fluids (fuel-rich mixture from VC in the inner pipe and the bulk of air in the annulus) arrive at the combustor with the same pressure. The pipe for Case III is the same as that used in Case I, and details are given in Figure F-2 in Appendix F.

(a) Inner Pipe for Fuel-Rich Mixture

$W_{F\text{-}A\ mix} = 172$ lb/hr

$t_{mix} = 1{,}012$°F

$V_{F\text{-}A\ mix} = 16.7$ cfm

$G_{F\text{-}A\ mix} = 21{,}200$ lb/(ft^2)(hr)   Re = 33,800

$f = 0.0002$[1] $\Delta P = 0.62$ psi/100 ft

Since 1/3 ft will be used

$\Delta P = 0.62 \times 0.33/100 = 0.002$ psi (negligible)
and the fuel-rich mixture will reach the combustor at 27.3 psia.

(b) Annular Space for Air

W_{air} = 2,093 lb/hr

t = 1,012°F $\quad$ V_{air} = 664 cfm

G_a = 177,000 lb/(ft^2)(hr) $\quad$ Re_a = 30,400

f = 0.00022[2] $\quad$ ΔP = 8.81 psi/ft

The air reaches the annulus at 30.2 psia and the fuel-rich mixture reaches the combustor at 27.3 psia. Equalization of the pressure requires that the air stream lose 2.9 psi on passage through the annulus. Consequently, the length of the pipe is

$$2.9 \div 8.8 = \underline{0.33\ \text{ft}}$$

(c) Weight of the VC Outlet Pipe

The weight of the double pipe (including fins and heating elements) is 4.65 lb/ft. Thus

$$W_{pipe} = 0.33 \times 4.65 = \underline{1.6\ \text{lb}}$$

7. Total Weight of Air and Fuel Feed

Air pipe (main)	17 lb
Air pipe (VC inlet)	2.1
Fuel pipe	0.3
Spray nozzle	0.5
Vaporization chamber (Hastelloy C)	46
Heater	7
Outlet piping	1.6
Added for fittings	0.5
Total	75 lb

b. Combustor

The segmented reactor is chosen for Case III.

1. Catalyst and Combustor Cross-Section

As previously calculated, the amount of catalyst for 75% conversion at the design conditions is 8.5 lb or 0.21 ft^3.

Considerations regarding catalyst and cooling tubing yield a rectangular duct 12.5 inch (1.042 ft) high and 30 inch (2.5 ft) wide, with three catalyst layers of 1/4, 3/8, and 1/2 in. thickness. The volume of catalyst is distributed as follows:

Layer	Thickness, In.	Volume of Catalyst, ft^3
1	0.25	0.0542
2	0.375	0.0814
3	0.5	0.1085
Total	1.125 inch	0.2441 ft^3

Thus, the weight of the catalyst used is

$$0.2441 \times 40.6 = \underline{9.9 \text{ lb}}$$

This is about 17% in excess of the calculated amount. This excess is regarded as assurance that the desired conversion will be achieved when a charge of catalyst nears the end of its service life.

Six screens are required to keep the catalyst in place. Assuming a 0.041 inch wire diameter, the unit weight is 1.7 lb/ft^2 and the screens weigh

$$6 \times (1.04 \times 2.5) \times 1.7 = \underline{27.2 \text{ lb}}$$

2. Heating Elements

They are shielded electric resistance wires, 3/16 inch diameter, and are spaced 3/8 inch ℄ to ℄. Thus, there are 33 elements, and they weigh about 0.1 lb/ft. Consequently,

$$33 \times 2.5 \times 0.1 = \underline{8.3 \text{ lb}}$$

3. Data for Transfer of Combustion Heat

(a) Loads

The same reasoning as in Appendix H is used here. Also, only those data that differ from the ones used there are listed in what follows.

$$W_{A\text{-}F\ mix} = W_{MBG} = 2{,}265 \text{ lb/hr}$$

$$W_{H_2O\ cool} = 1{,}470 \text{ lb/hr}$$

$$Q_{total} = 1{,}608{,}000 \text{ BTU/hr}$$

$$Q_{H_2O\ Preheat} = 182{,}000 \text{ BTU/hr}$$

$$Q_{H_2O\ Vaporiz.} = 1{,}426{,}000 \text{ BTU/hr}$$

(b) Properties

	H_2O Preheating Section	H_2O Vaporization Section
Temperature gases in, °F	1,337	1,337
Temperature gases out, °F	1,337	1,337
Temperature coolant in, °F	88	212 (liquid)
Temperature coolant out, °F	212 (liquid)	212 (steam)
LMTD, °F	1,190	--
R	0	--
S	0.1	--
F_T (assumed)	1	--
Δt mean, °F	1,190	1,125
Caloric temperature of gases, T_c, °F	1,337	1,337
Caloric temperature of coolant, t_c, °F	150	212
Gas	MBG	0.2 (A-F Mix) + 0.8 MBG
Temperature, °F	1,337	1,337
Average pressure, psia	26.2	26.2
μ = viscosity, lb/(ft)(hr)	0.094	0.095
C_p = specific heat, BTU/(lb)(°F)	0.282	0.288
k = conductivity, BTU/(hr)(ft^2)(°F/ft)	0.0412	0.041
$(C_p \mu/k)^{1/3}$	0.862	0.874
ϕ	1	1
V_m = molar volume of gas, ft^3/lb-mole	686	686
$\overline{V}_{F.V}$ = specific volume of fuel vapor, ft^3/lb	5.8	5.8

	H_2O Preheating Section	H_2O Vaporization Section
V, cfm	882	875
ρ = density, lb/ft^3	0.0428	0.04315
S = specific gravity with respect to water	0.000686	0.0006915
Data on coolant in Appendix H	---	---

(c) Tubing (316-SS)

Tubing: OD_t = 1 in   wall = 0.025 in ID_t = 0.95 in

k_t = 13.92 BTU/(hr)(ft^2)(°F/ft)

Fins: b_f = 0.125 in th_f = 0.035 in N_f = 8 fins/inch tubing

r_e = 0.625 in r_b = 0.5 in OD_f = 1.25 in

Bank arrangement: square pitch

$N_{t/b}$ = 10 tubes/bank

$S_T = S_L = V_s$ = 1.25 in

Duct Side	Tube Side
A_f = 0.589 ft^2/ft	a_t = 0.0049 ft^2
A_o = 0.188 ft^2/ft	$A_{t/b}$ = 0.049 ft^2/bank
P_p = 5.44 ft/ft	
d_{es} = 0.091 ft	d_{et} = 0.0792 ft
a_s = 0.375 ft^2	
G_s = 6,040 lb/(ft^2)(hr)	G_t = 30,000 lb/(ft^2)(hr)
h_d = 602 BTU/(hr)(ft^2)(°F)	h_{di} = 500 BTU/(hr)(ft^2)(°F)

Fin Efficiency and Inside Tube Area

r_e/r_b = 1.25 y_b = 0.00146 ft

$$(r_e - r_b)\left(\frac{h_f}{k_t\, y_b}\right)^{0.5} = 0.0104 \left(\frac{h_f}{0.0203}\right)^{0.5}$$

a_i = 0.249 ft^2/ft

$A_{it/b}$ = 6.22 ft^2/bank

Pressure drop, duct side

$V_{NF} = 0.1134 \text{ ft}^3$

$S_F = 19.44 \text{ ft}^2$

$D'_{ev} = 0.0233 \text{ ft}$

$$\left(\frac{D'_{ev}}{S_T}\right)^{0.4} = 0.552 \qquad \left(\frac{S_L}{S_T}\right)^{0.6} = 1$$

4. Heat Exchange Surface

	H_2O Preheating Section	H_2O Vaporization Section
Heat Transfer - Duct Side		
Re_s	5,850	5,790
j_f	49(3)	48.5(3)
h_f, BTU/(hr)(ft^2)(°F)	19.3	19.1
h'_f, BTU/(hr)(ft^2)(°F)	18.7	18.5
Heat Transfer - Tube Side		
Re_t	2,230	7,200
j_{hi}	8(4)	28(4)
h_i, BTU/(hr)(ft^2)(°F)	71.5	102
h'_i, BTU(hr)(ft^2)(°F)	62.5	84.7
Heat Transfer - Overall U and Area		
$(r_e - r_b) \left(\frac{h'_f}{K_t\, y_b}\right)^{0.5}$	0.316	0.314
Ω	0.96(5)	0.96(5)
h'_{fi}, BTU/(hr)(ft^2)(°F)	56.7	56.1
U_{Di}, BTU/(hr)(ft^2)(°F)	29.7	33.8
A_{iT}, ft^2	5.1	37.5

$A_{iT,\ total} = 42.6\ ft^2$

$N_b = 7$ banks of cooling tubes

Number of cooling tubing banks = 7

5. Pressure Drop

(a) Tube Side

$G_t = 30,000\ lb/(ft^2)(hr)$ $Re_t = 7,200$

$f = 0.00031^{(1)}$

$\Delta P_{bank} = 0.106$ psi/bank 2.5 ft long

A return bend is equivalent to 6 ft of tube

$\therefore\ \Delta P_{bend} = 6 \times 0.106/2.5 = 0.254$ psi/return bend

consequently

$\Delta P_{total} = 7 \times 0.106 + 8 \times 0.254 = 2.77$ psi

Pressure loss in cooling coils = 2.8 psi

(b) Duct Side

The pressure drop in the BG as it flows through the combustor is the sum of five individual losses, which are listed below (for details see Appendix H):

- Pressure loss due to gradual expansion:

  Gases reach combustor with 27.3 psia = 12.6 psig

  $P_1 = 0.08 \times 12.6 = \underline{1.01\ psi}$

  Air-fuel mixture reaches heating elements with 26.3 psia

- Pressure loss through layers of catalyst:

  $V_o = 5.6$ ft/sec

  $\mu = 0.0000264\ lb/(ft)(sec)$

  $Re = 57.2$ $\therefore$ turbulent flow

Bed thickness = Σ layer thickness = 0.094 ft

ΔP_2 = <u>0.18 psi</u>

- Pressure loss through screens:

ρ = 0.000691 g/cc μ = 0.0393 cps

R_e = 107   C = 1.15(9)

ΔP_3 = 0.00082 psi/screen

ΔP_3 = 6 x 0.00082 - <u>0.005 psi</u> (negligible)

- Pressure loss due to cooling coils:

L_p = (7 + 2) 1.25/12 = 0.9375 ft

R_e = 1,480

f = 0.00313(3)

ΔP_4 = <u>0.07 psi</u>

- Pressure loss in "orifice":

v = 38.9 ft/sec

P_v = 23.5 ft (column of BG)

$\doteq$ 0.007 psi

A_1 = 12.5 x 30 = 375 inch^2

A_2 = 8.5 x 26 = 221 inch^2 (2 inch high barrier)

A_2/A_1 = 0.59

$$\left(\frac{1}{C^2} - 1\right) = 0.27$$

ΔP_5 = 0.002 psi (negligible)

- Total pressure loss in combustor duct

ΔP_T = 1.01 + 0.18 + 0.005 + 0.07+ 0.002 = <u>1.267 psi</u>

ΔP_T = 1.3 psi

Thus the pressure of moist BG leaving the combustor is 27.3-1.3 = 26.0 psia.

P_{MBG} leaving combustor = 26.0 psia

6. Fuel Preheating Pipe

This pipe is located at the downstream end of the combustor.

W_F = 157 lb/hr Q = 9,700 BTU/hr

$t_{fuel\ in}$ = 200°F $t_{fuel\ out}$ = 300°F

Hot Fluid, °F		Cold Fluid	Difference, °F
1,337	Higher temperature	300	1,037
1,337	Lower temperature	200	1,137
0	Difference	100	100

LMTD = 1,088°F (no correction is applied)

T_c = 1,337°F t_c = 250°F

Data for BG at 1,337 are given above

Data for fuel at 250°F

μ_F = 1.21 lb/(ft)(hr)

$C_{p,F}$ = 0.595 BTU/(lb)(°F)

k_F = 0.0751 BTU/(hr)(ft^2)(°F/ft)

$(c_p \mu/k)^{1/3}$ = 2.125

ρ_F = 5.7 lb/gal = 42.6 lb/ft^3

S_F = 0.683

ϕ_F = 1.07

Tubing

OD_t = 0.5 in wall = 0.020 in ID_t = 0.46 in

Fins: b_f = 0.125 in th_f = 0.035 in N_f = 8 fins/inch

r_b = 0.25 in r_e = 0.375 in OD_f = 0.75 in

It is assumed the tube makes 5 passes across the combustor duct.

Heat Exchanger Surface

Duct Side	Tube Side
$A_f = 0.327\ ft^2/ft$	$a_t = 0.00115\ ft^2$
$A_o = 0.094\ ft^2/ft$	
$P_p = 5.44\ ft/ft$	
$d_{es} = 0.0493\ ft$	$d_{et} = 0.0383\ ft$
$a_s = 2.366\ ft^2$	
$G_s = 960\ lb/(ft^2)(hr)$	$G_t = 136{,}000\ lb/(ft^2)(hr)$
$Re_s = 500$	$Re_t = 4{,}320$
$j_f = 8.4$(3)	$j_h = 15$(4)
$h_f = 6.1\ BTU/(hr)(ft^2)(°F)$	$h_i = 66.9\ BTU/(hr)(ft^2)(°F)$
$h'_f = 6.0\ BTU/(hr)(ft^2)(°F)$	$h'_i = 52.7\ BTU/(hr)(ft^2)(°F)$

$r_e/r_b = 1.5$

$$(r_e - r_b)\left(\frac{h'_f}{k_t y_b}\right)^{0.5} = 0.179$$

$\Omega = 0.98$(5)

$a_i = 0.12\ ft^2/ft$

$h'_{fi} = 20.8\ BTU/(hr)(ft^2)(°F)$

$U_{Di} = 14.9\ BTU/(hr)(ft^2)(°F)$

$A_{iT} = 0.6\ ft^2$

$L_t = \underline{5\ ft}$

This will give five 1 ft long passes.

Weight of tubing = 0.34 x 5 = 1.7 lb

7. Weight

All equipment items are made of 316-SS, except when specified otherwise.

Catalyst	9.9 lb
Screens (6)	27.2
Heating elements	8.3
Fuel preheating tubing	1.7
Cooling tubing (7 x 18.3 lb/bank)	128.2
Duct, baffles (Hastelloy C)	127.7
Total	303.0 lb

c. Heat Exchanger No. 1 (HE 1)

HE 1 is located immediately after the combustor, from which it is separated by the "orifice". Provisions are made for flaps for isolation of the combustor from the rest of the subsystem during start-up.

1. Data for Heat Transfer

(a) Loads and Temperatures

Q = 472,000 BTU/hr

W_{air} = 2,108 lb/hr

W_{MBG} = 2,265 lb/hr

Air enters at 250°F and leaves at 1,112°F

Moist BG enters at 1,337°F and leaves at 611°F.

Hot Fluid		Cold Fluid	Difference
1,337°F	Higher temperature	1,112	225
611	Lower temperature	250	361
726	Difference	862	136

Mean temperature difference

LMTD = 288°F

$R = 0.84$ $S = 0.8$ $F_T = 0.68$(6)

ΔT = 196°F

Caloric temperature

Arithmetic averages are sufficient, thus:

for BG: $T_c = 974$

for air: $t_c = 681$

(b) Properties at Above Temperatures

	Air	Moist BG
Temperature, °F	681	974
Average pressure, psia	32.8	25.0
μ, lb/(ft)(hr)	0.075	0.0832
C_p, BTU/(lb)(°F)	0.254	0.291
k, BTU/(hr)(ft^2)(°F/ft)	0.0282	0.0345
$(C_p\mu/k)^{1/3}$	0.878	0.889
ϕ	1	1
V_m, ft^3/lb-mole	373	615
$\overline{V}_{F-V}$, ft^3/lb	---	4.8
V, cfm	454	791
ρ, lb/ft^3	0.0775	0.0477
S	0.00124	0.000765

(c) Duct and Tubing

Duct: $h_d = 2$ ft $\quad$ $b_d = 2.5$ ft

Tubing: $OD_t = 1$ in $\quad$ wall = 0.035 in $\quad$ $ID_t = 0.93$ in

Fins: $b_f = 0.125$ in $\quad$ $th_f = 0.035$ in $\quad$ $N_f = 8$ fins/inch

$r_e = 0.625$ in $\quad$ $r_b = 0.5$ in $\quad$ $OD_f = 1.25$ in

Bank arrangement: square pitch

$N_{t/b}$ = 19 tubes per bank

S_T = 1.263 in = 0.105 ft

$S_L = V_s$ = 1.25 in = 0.104 ft

2. Heat Exchange Surface

Duct Side: MBG	Tube Side: Air
A_f = 0.589 ft²/ft	a_t = 0.00472 ft²
A_o = 0.189 ft²/ft	
P_p = 5.44 ft/ft	
d_{es} = 0.91 ft	d_{et} = 0.0775 ft
a_s = 0.765 ft²	$A_{t/b}$ = 0.0897 ft²/bank
G_s = 2,960 lb/(ft²)(hr)	G_t = 23,500 lb/(ft²)(hr)
Re_s = 3,240	Re_t = 24,300
j_{hf} = 32(3)	j_{hi} = 81(4)
h_f = 10.8 BTU/(hr)(ft²)(°F)	h_i = 25.9 BTU/(hr)(ft²)(°F)
h_{ds} = 602 "	h_{di} = 333 "
h_f' = 10.6 "	h_i' = 24 "

<u>Overall design coefficient, area, and number of banks</u>

$r_e/r_b = 1.25$

$$(r_e - r_b)\left(\frac{h_f'}{k_t\, y_b}\right)^{0.5} = 0.24$$

Ω = 0.98(5)

a_{it} = 0.244 ft²/ft

$A_{it/b}$ = 11.57 ft²/bank

h_{fi}' = 33.3 BTU/(hr)(ft²)(°F)

U_{Di} = 14 "

Overall design coefficient = 14 BTU/(hr)(ft²)(°F)

$A_{iT} = 172\ ft^2$

$N_b = 14.9$

$\boxed{\text{Number of banks} = 15}$

3. Pressure Drop

(a) Tube Side: Air

$f = 0.00024^{(1)}$

$\Delta P = 0.026$ psi/ft of bank

$\Delta P_{bank} = 0.066$ psi/bank

A return bend is equivalent to 5.5 ft tubing.

$\Delta P_{bend} = 5.5 \times 0.026 = 0.145$ psi/bend

Consequently

$\Delta P_{total} = 15 \times 0.066 + 16 \times 0.145 = 3.31$ psi

$\boxed{\text{Pressure loss in cooling tubes} = 3.3 \text{ psi}}$

The air was assumed to be available from engines at 34.9 psia and to suffer a loss of 0.6 psi on its way to HE 1, where it arrives with 34.3 psia. Consequently, air will leave HE 1 at 34.3-3.3 = 31 psia, and the average pressure of air in HE 1 is 32.65 psi (very close to the 32.8 psia assumed in calculation of properties of air).

$\boxed{\Delta P_{air} \text{ leaving HE 1} = 31 \text{ psia}}$

(b) Duct Side: MBG

There are three individual losses which combined yield the overall pressure loss suffered by the moist BG in HE 1:

- pressure loss due to enlargement, ΔP_1
- pressure loss due to cooling tubing, ΔP_2
- pressure loss due to contraction, ΔP_3

(i) Pressure Loss Due to Englargement

There is a sudden enlargement from the "orifice" (area 221 $inch^2$) to the HE 1 duct (area 720 $inch^2$). It is estimated that a pressure loss equal to 8% of gage pressure of incoming gas takes place (the same as in the case of gradual enlargement from a pipe to a duct).

$$P_1 = 0.08\ (26.0-14.7) = \underline{0.91}$$

(ii) Pressure Loss Due to Banks of Cooling Tubing

$V_{NF} = 0.221\ ft^3$

$S_F = 36.9\ ft^2$

$D'_{ev} = 0.024\ ft$

$(D'_{ev}/S_T)^{0.4} = 0.553$

$(S_L/S_T)^{0.6} = 0.994$

$R_e = 850$

$f = 0.0034(3)$

$L_p = 15 \times 1.25/12 = 1.563\ ft$

$\Delta P_2 = \underline{0.03\ psi}$

(iii) Pressure Loss Due to Contraction

There is a sudden contraction from the duct (area 720 $inch^2$) to the orifice ($8 \times 26 = 208\ inch^2$) which is sized accordingly to the duct of HE 2. It is estimated that a pressure loss equal to 2% of the gage pressure of MBG arriving to HE 1 exit takes place (same as in case of gradual contraction from a duct to a pipe).

$$\Delta P_3 = 0.02\ [26.0-14.7-(0.91 + 0.03)] = \underline{0.31\ psi}$$

(iv) Total Pressure Loss in the Duct

$$\Delta P_T = 0.91 + 0.03 + 0.31 = \underline{1.25\ psi}$$

The average pressure of moist BG in the duct, for determination of properties, is

$$26.0 - (0.91 + 0.5 \times 0.03) = 25.08\ psia$$

which is very close to the assumed 25 psia.

Moist BG reaches HE 2 at 26-1.25 = 24.75 psia.

$\boxed{P_{MBG} \text{ leaving HE 1} = 24.7 \text{ psia}}$

4. Weight

Cooling tubing of 316-SS (40.3 lb/bank)	605 lb
Duct of Hastelloy C	168
Total	773 lb

d. Heat Exchanger No. 2 (HE 2)

This exchanger follows immediately after HE 1. HE 2 uses fuel as coolant for removal of sensible heat from the BG, and for removal of heat of condensation released by all the condensable fuel vapor and by part of the water vapor present in the BG stream. BG leaves HE 2 as a gas saturated with moisture at the exit temperature.

1. Data for Heat Transfer

(a) Loads and Temperatures

$W_{MBG,\ in} = 2{,}265$ lb/hr

$W_{SBG,\ out} = 2{,}135$ lb/hr

$W_{BG,\ avg} = 2{,}200$ lb/hr

$W_{fuel,\ coolant} = 3{,}600$ lb/hr

Q = 414,000 BTU/hr

BG enters at 611°F and leaves at 85°F

Fuel coolant enters at 65°F and leaves at 275°F

Hot Fluid		Cold Fluid	Difference
611	Higher temperature	275	336
85	Lower temperature	65	20
526	Difference	210	316

Mean Temperature Difference

LMTD = 112°F

R = 2.5   S ≃ 0.4 R_T = 0.71[6]

Δt = 79°F

Caloric Temperatures

$\Delta t_c / \Delta t_h \simeq 0.06$

for temperature range of 210°F and 53.6°API

K_c = 0.132[7] and F_c = 0.29[7]

Consequently

T_c = 238°F for BG

t_c = 126°F for fuel coolant

(b) Properties at Above Temperatures

	Fuel Coolant	BG
Temperature, °F	126	238
Pressure, psia	---	23.5
μ, lb/(ft)(hr)	2.49	0.05
C_p, BTU/(lb)(°F)	0.518	0.254
k, BTU/(hr)(ft^2)(°F/ft)	0.0783	0.0187
$(C_p \mu/k)^{1/3}$	2.55	0.88
ϕ	1.07	1
V_M, ft^3/lb-mole	---	316
$\bar{V}_{F-V}$, ft^3/lb	---	2.5
V_{avg}, cfm	---	393
ρ, lb/ft^3	45.95	0.0933
s	0.7366	0.0015

(c) Duct and Tubing

Duct: $h_d = 1$ ft $b_d = 2.5$ ft

Tubing: $OD_t = 0.5$ in   wall = 0.020 in $ID_t = 0.46$ in

Fins: $b_f = 0.125$ in $th_f = 0.035$ in $N_f = 8$ fins/inch

$r_e = 0.375$ in $r_b = 0.25$ in $OD_f = 0.75$ in

Bank arrangement: square pitch

$N_{t/b} = 16$ tubes per bank

$S_T = S_L = V_s = 0.75$ in $= 0.0625$ ft

2. Heat Exchange Surface

Duct Side: BG	Tube Side: Fuel Coolant
$A_f = 0.327$ ft^2/ft	$a_t = 0.00115$ ft^2
$A_o = 0.094$ ft^2/ft	
$P_p = 5.44$ ft/ft	
$d_{es} = 0.049$ ft	$d_{et} = 0.0383$ ft
$a_s = 0.6$ ft^2	$A_{t/b} = 0.0185$ ft^2/bank
$G_s = 3{,}670$ lb/(ft^2)(hr)	$G_t = 195{,}000$ lb/(ft^2)(hr)
$Re_s = 3{,}620$	$Re_t = 3{,}000$
$j_{hf} = 34.5$(3)	$j_{hi} = 7.6$
$h_f = 11.5$ BTU/(hr)(ft^2)(°F)	$h_i = 42.3$ BTU/(hr)(ft^2)(°F)
$h_{ds} = 602$ "	$h_{di} = 500$ "
$h_f' = 11.3$ "	$h_i' = 39.0$ "

Overall Design Coefficient, Area, and Number of Banks

$r_e/r_b = 1.5$ $k_t = 9.4$ BTU/(hr)(ft^2)(°F/ft)

$(r_e - r_b)\left(\frac{h'_f}{k_t\, y_b}\right)^{0.5} = 0.3$

$\Omega = 0.97$(5)

$a_{it} = 0.12$ ft^2/ft

$A_{it/b} = 4.82$ ft^2/bank

h'_{fi} = 38.6 BTU/(hr)(ft^2)(°F)

U_{Di} = 19.4 BTU/(hr)(ft^2)(°F)

Overall design coefficient = 19.4 BTU/(hr)(ft^2)(°F)

A_{iT} = 270 ft^2

N_b = 56

Number of banks = 56

3. Pressure Drop

(a) Tube Side: Fuel Coolant

$f = 0.0004^{(1)}$

ΔP_{bank} = 0.024 psi/bank 2.5 ft long

A return bend is equivalent to 2.8 ft tubing

ΔP_{bend} = 0.024 x (2.8/2.5) = 0.027 psi/bend

Consequently

ΔP_{total} = 56 x 0.024 + 57 x 0.027 = <u>2.9 psi</u>

It is possible that a fuel booster pump will be necessary, so that the fuel will arrive at the engines and the vaporization chamber with sufficient pressure.

(b) Duct Side: BG

The losses are caused by:

- enlargement, ΔP_1
- condensation (loss of volume), ΔP_2
- cooling coils, ΔP_3
- flow through the "orifice", ΔP_4

(i) Pressure Loss Due to Englargement

The enlargement from the "orifice" (area = 208 inch^2) to the duct (area = 360 inch^2) is relatively small. It is estimated that a pressure loss equal to 4% of the gage pressure of incoming gas takes place.

$$\Delta P_1 = (24.7-14.7) \times 0.04 = 0.4 \text{ psi}$$

(ii) Pressure Loss Due to Condensation of Fuel Vapor and Part of Water Vapor

$$P = 24.7 - 0.4 = 24.3 \text{ psia}$$

total of gases entering = 1.28542 lb-moles/min

fuel vapor condensed = 0.00565 "

water vapor condensed = 0.07676 "

total condensed = 0.08241 "

$$\Delta P_2 = \frac{0.08241}{1.28542} \times 24.3 = \underline{1.56 \text{ psi}}$$

(iii) Pressure loss due to banks of cooling tubing

$V_{NF} = 0.083$ ft^3

$S_F = 16.86$ ft^2

$D'_{ev} = 0.0196$ ft

$(D'_{ev}/S_T)^{0.4} = 0.63$

$(S_L/S_T)^{0.6} = 1$

$R_e = 1{,}440$

$f = 0.00315$(3)

$\Delta P = \underline{0.06 \text{ psi}}$

(iv) Pressure Loss Due to "Orifice"

Not calculated since previous cases have shown it to be negligible.

(v) Total Pressure Loss in the Duct

$$\Delta P_T = 0.4 + 1.56 + 0.06 = \underline{2.02 \text{ psi}}$$

$$\therefore P_{SBG} = 24.7 - 2.02 \simeq 22.7 \text{ psia}$$

P_{SBG} leaving HE 2 = 22.7 psia

4. Weight

Cooling tubing of 316-SS (14.1 lb/bank)	789 lb
Duct of Hastelloy C	205
Booster pump for fuel coolant	16
	1,010 lb

e. Redesign of HE 2

In the preceding conceptual design, it was assumed that only the fuel used by the engines (60 lb/min) would be available to cool the BG in HE 2, and this fuel was allowed to reach whatever temperature was called for to absorb the heat. (Figure 47 indicates this temperature to be 275°F.)

A more favorable temperature difference is obtainable if it is assumed that the fuel is recirculated at a rate sufficient to give a lower exit temperature, and this could provide a substantial reduction in transfer surface and weight. Assuming that fuel coolant enters at 65°F (h = 15.8 BTU/lb) and leaves at 150°F (h = 58.8 BTU/lb), the amount of fuel necessary to perform the required cooling duty is

$$W_{fuel,\ cool.} = \frac{6,896}{58.8-15.8} = \underline{160.4\ \text{lb/min}}$$

$$= 9,620\ \text{lb/hr}$$

No other changes are introduced.

1. Design of HE 2

The duct and the tubing, as well as their arrangement, remain the same as in Part d. Only the items that change are listed in what follows.

(a) Temperatures

LMTD = 140.7°F

R = 6.2 S = 0.16 F_T = 0.85(6)

$\Delta t = \underline{119°F}$

$\Delta t_c / \Delta t_h = 0.0434$

For range of 85°F and 53.6° API

K_c = 0.078(7) and F_c = 0.273(7)

thus

$$T_c = 229°F \quad \text{and} \quad t_c = 88°F$$

(b) Properties

For BG, the properties at 238°F (see above) are used, because they are practically the same as those at 229°F.

For fuel coolant, the properties at 82°F (Appendix H) are used, since they are practically the same as those at 88°F.

(c) Heat Exchange Surface

There is no change in calculations for the duct side.

Tube Side: $G_t = 521, \quad lb/(ft^2)(hr)$

$Re_t = 5{,}000$

$j_{hi} = 17^{(4)}$

$h_i = 114$ BTU/(hr)(ft^2)(°F)

$h_i' = 92.8$ "

Overall design coefficient, area, and number of banks

$U_{Di} = \underline{27.3}$ BTU/(hr)(ft^2)(°F)

$A_{iT} = 127.6$ ft^2

$N_b = 26.5$

Number of banks = 27

(d) Pressure Drop

(i) Tube Side

$f = 0.00034^{(1)}$

$\Delta P_{bank} = 0.143$ psi/bank

$\Delta P_{bend} = 0.16$ psi/bend

$\Delta P_T = 27 \times 0.143 + 28 \times 0.16 = \underline{8.34 \text{ psi}}$

A fuel booster pump is a must.

(ii) Duct Side

The pressure losses due to enlargement, condensation, and orifice remain unchanged. The pressure loss due to cooling coils is now about one-half of that in the initial design of HE 2 (Part d above). This is such a small quantity, that the pressure of BG leaving HE 2 is practically the same as before, namely 22.7 psia.

(e) Weight

Cooling tubing of 316-SS (14.1 lb/bank)		380.4 lb
Duct of Hastelloy C		102.6
Booster pump for fuel (coolant)		17
	Total	500 lb

The length of HE 2 is now 22 inches.

f. Drier

The design target calls for delivery of dry BG with a maximum of 1,555 ppm V/V of water (equivalent to <0.001 lb H_2O/lb dry gas). Parallel flow of fluids is again used.

1. Data for Heat Transfer

(a) Loads and Temperatures

$W_{SBG,\ in} = 2{,}135$ lb/hr

$W_{dry\ BG,\ out} = 2{,}080$ lb/hr

$W_{BG,\ avg.} = 2{,}107$ lb/hr

$W_{H_2O,\ coolant} = 1{,}470$ lb/hr

$Q = 77{,}600$ BTU/hr

BG enters at 85°F and leaves at 100°F

Cooling water enters at 35°F and leaves at 88°F

Hot Fluid		Cold Fluid	Difference
100°F	Higher temperature	88°F	12°F
85	Lower temperature	35	50
15	Difference	53	38

Mean temperature difference

LMTD = 26.7°F

$R \simeq 0.3$ $S \simeq 0.8$ $F_T = 0.95^{(6)}$

$\Delta t \simeq$ 25°F

Caloric temperatures

Arithmetic averages are sufficient

for BG: T_c = 93°F

for water: t_c = 62°F

(b) Properties at Above Temperatures

	BG	Cooling Water
Temperature, °F	93	62
Pressure, psia	21.2	--
μ, lb/(ft)(hr)	0.045	2.76
C_p, BTU/(lb)(°F)	0.24	1
k, BTU/(hr)(ft^2)(°F/ft)	0.0155	0.329
$(c_p\mu/k)^{1/3}$	0.886	2.03
ϕ	1	1.03
V_m, ft^3/lb-moles	280	--
V_{avg}, cfm	329	--
ρ, lb/ft^3	0.107	62.3
s	0.0017	1

(c) Duct and Tubing

Duct: The duct cross-section, dictated by the superficial velocity limits for $CaCl_2$, is: h_d = 2 ft, b_d = 2.5 ft.

Tubing: OD_t = 0.375 in wall = 0.016 in ID_t = 0.343 in

Fins: b_f = 3/16 in th_f = 0.035 in N_f = 8 fins/inch

r_e = 0.375 in r_b = 3/16 in OD_f = 0.75 in

Bank arrangement: square pitch

$N_{t/b}$ = 32 tubes/bank

S_T = 0.75 in = 0.0625 ft

$S_L = V_s$ = 1.47 in = 0.123 ft

2. Heat Exchange Surface

Duct Side: BG	Tube Side: Cooling Water
A_f = 0.442 ft^2/ft	a_t = 0.000642 ft^2
A_o = 0.0707 ft^2/ft	$A_{t/b}$ = 0.0205 ft^2
P_p = 7.44 ft/ft	d_{et} = 0.0286 ft
d_{es} = 0.044 ft	G_t = 71,600 lb/(ft^2)(hr)
a_s = 1.8 ft^2	Re_t = 740
G_s = 1,210 lb/(ft^2)(hr)	v = 0.32 ft/sec
Re_s = 1,175	h_{il} = 107(8) BTU/(hr)(ft^2)(°F)
j_{hf} = 15.5(3)	factor = 1.1(8)
h_f = 4.85 BTU/(hr)(ft^2)(°F)	h_i = 117.7 BTU/(hr)(ft^2)(°F)
h_{ds} = 602 "	h_{di} = 500 "
h_f' = 4.8 "	h_i' = 95 "

<u>Overall design coefficient, area, and number of banks</u>

$r_e/r_b = 2$

y_b = 0.00146 ft

k_t = 116.7 BTU/(hr)(ft^2)(°F/ft) for aluminum

$$(r_e - r_b)\left(\frac{h_f'}{k_t\, y_b}\right)^{0.5} = 0.083$$

$\Omega = 0.93^{(5)}$

$a_{it} = 0.0898 \text{ ft}^2/\text{ft}$

$A_{it/b} = 7.18 \text{ ft}^2/\text{bank}$

$h'_{fi} = 26.9 \text{ BTU/(hr)(ft}^2\text{)(°F)}$

$U_{Di} = 21$ "

$$\boxed{\text{Overall design coefficient} = 21 \text{ BTU/(hr)(ft}^2\text{)(°F)}}$$

$A_{iT} = 147.8 \text{ ft}^2$

$N_b = 20.6$

$$\boxed{\text{Number of banks} = 21}$$

3. Amount of Drying Agent

Calcium chloride alone is sufficient to provide a ballast gas with less than 1,555 ppm V/V of water.

(a) Efficiency of $CaCl_2$

Conditions: 100°F exit temperature

~ 14.7 psia exit pressure

Efficiency at SV = 1,900 hr^{-1} is given as

equilibrium ÷ 0.95 = 1 mm Hg

Thus

$$\text{Efficiency} = \frac{1 \text{ mm}}{} \left| \frac{1 \text{ atm}}{760 \text{ mm}} \right| \frac{10^6 \text{ ppm}}{} = \underline{1{,}320 \text{ ppm}}$$

1,320 ppm is equivalent to 0.00076 lb H_2O/lb dry BG which is 24% less than the maximum permitted of 0.001 lb H_2O/lb dry BG.

(b) Volume and Weight of $CaCl_2$

The average flow of BG in the drier bed at design conditions is

$$V_{BG,\ avg.} = 329 \text{ cfm}$$

The volume of $CaCl_2$ required for SV of 1900 hr^{-1} is

$$\text{Volume } CaCl_2 = \frac{329 \text{ ft}^3}{\text{min}} \left| \frac{60 \text{ min}}{\text{hr}} \right| \frac{\text{hr}}{1900} = \underline{10.4 \text{ ft}^3}$$

The weight of $CaCl_2$ is

$$W_{CaCl_2} = 10.4 \times 51 = 530 \text{ lb}$$

$$\boxed{\text{Weight of } CaCl_2 = 530 \text{ lb}}$$

(c) Water Removal and Useful Life of $CaCl_2$

(i) Concentration of Water Entering $CaCl_2$

$$\frac{0.05088 \text{ lb-moles } H_2O/\text{min}}{1.20301 \text{ lb-moles SBG/min}} \times 100 = 4.23\%$$

$$\Leftrightarrow \underline{42,300 \text{ ppm}}$$

The concentration of water in the gas leaving the $CaCl_2$ was shown above to be 1,320 ppm.

(ii) Water Removed by $CaCl_2$

$$\frac{42,300 - 1,320}{42,300} \times 100 = \underline{96.9\%}$$

The total of water to be removed in the drier is 3.83 lb/flight, thus $CaCl_2$ will remove

$$3.83 \times 0.969 = \underline{3.72 \text{ lb/flight}}$$

and 3.83-3.72 = 0.11 lb H_2O/flight will be left in BG. This gives

$$0.11 \text{ lb } H_2O \div 144.5 \text{ lb dry BG} = 0.00076 \text{ lb } H_2O/\text{lb dry BG}$$

Actually, during most of the flight the concentration of water in BG will be below this value, because at conditions other than "design" the space velocity is less than 1,900 hr^{-1}, consequently, the residence time is longer and the removal is greater.

(iii) Useful Life of $CaCl_2$

The capacity of $CaCl_2$ under the design conditions is 0.3 lb H_2O/lb $CaCl_2$. At 80% of the stated capacity, the $CaCl_2$ is capable of retaining, without regeneration,

$$530 \times 0.3 \times 0.8 = \underline{127 \text{ lb } H_2O}$$

This, in turn, is equivalent to a useful life (no regeneration) of

$$127 \div 3.72 = \underline{34 \text{ flights}} \text{ (of 403 min)}$$

$$\underline{230 \text{ flight-hrs}}$$

$$\boxed{\text{Useful life } CaCl_2 = 230 \text{ hr}}$$

(d) Bed Cross-Section

One of the conditions for proper operation of a $CaCl_2$ drier is that the superficial velocity of the gas be in the 50-100 ft/min range, on average 75 ft/min. This gives a cross-section of

$$329 \div 75 = 4.4 \text{ ft}^2$$

Consequently, using a 2 x 2.5 ft duct, the superficial velocity is

$$329 \div 5 = 66 \text{ ft/min}$$

and this is the cross-section chosen for the drier duct.

4. Volume and Length of the Drier

(a) Volume Occupied by the Tubing

The volume occupied by one bank of cooling tubing is 195 $\text{inch}^3 = 0.113 \text{ ft}^3$. Thus, all the banks occupy

$$21 \times 0.113 \simeq \underline{2.4 \text{ ft}^3}$$

(b) Total Volume and Length of Drier Bed

The total volume of the bed is

$$10.4 + 2.4 = \underline{12.8 \text{ ft}^3}$$

The length of the bed is

$$12.8 \div 5 = \underline{2.56 \text{ ft}}$$

Consequently, the spacing of cooling banks within the bed is (2.56 ft ≃ 31 inch)

$$31 \div 21 = \underline{1.47 \text{ inch}} \quad ℄ \text{ to } ℄$$

5. Pressure Drop

(a) Tube Side: Cooling Water

$f = 0.0007$(1)

$\Delta P_{bank} = 0.0058$ psi

A return bend is equivalent to 2 ft tubing

$\Delta P_{bend} = 0.0058 \times (2.0/2.5) = 0.0047$

Consequently

$$\Delta P_{total} = 21 \times 0.0058 + 22 \times 0.0047 = 0.226 \text{ psi}$$

$$\boxed{\Delta P_{T\text{-}Water} = 0.23 \text{ psi}}$$

(b) Duct Side: BG

There are the following losses:

- Pressure loss due to sudden expansion, ΔP_1
- Pressure loss due to friction in packed bed, ΔP_2
- Pressure loss due to friction with cooling tubing, ΔP_3
- Pressure loss equivalent to volume of water removed, ΔP_4
- Pressure loss due to sudden contraction, ΔP_5

(i) Expansion Loss

The sudden enlargement from orifice (area 208 $inch^2$) to the duct (area 720 $inch^2$) is estimated to produce a loss equal to 5% of the gage pressure of incoming gas

$$\Delta P_1 = (22.7 - 14.7)\ 0.05 = \underline{0.4 \text{ psi}}$$

(ii) Packed bed friction loss

Bed thickness, including the coils, is used.

$G_o = 420$ lb/(ft^2)(hr)

$R_e' = 2,470$

$(f/F_f) = 0.052^{(10)}$

$\Delta P_2 = \underline{0.32 \text{ psi}}$

(iii) Cooling Coils Friction Loss

$V_{NF} = 0.5 \text{ ft}^3$

$S_F = 26.9 \text{ ft}^2$

$D_{ev}' = 0.0746 \text{ ft}$

$(D_{ev}'/S_T)^{0.4} = 1.07$

$(S_L/S_T)^{0.6} = 1.5$

$f = 0.003^{(3)}$

$\Delta P_3 = \underline{0.003 \text{ psi}}$ (negligible)

(iv) Loss Equivalent to Volume of Removed Water

Assuming all water is removed half-way through the bed, the total gas pressure at this point is

$$P = 22.7 - (0.4 + 0.5 \times 0.32 + 0.003) = 22.14 \text{ psia}$$

Total gases entering drier = 1.20301 lb-moles/min

Water to be removed = 0.05088 lb-moles/min

$$\Delta P_4 = \frac{0.05088}{1.20301} \times 22.14 = \underline{0.94 \text{ psi}}$$

(v) Contraction Loss

The sudden contraction from duct to the outlet pipe is estimated to produce a pressure loss equal to 5% of the gage pressure of the gas reaching the outlet:

$$p = 22.7 - [14.7 + (0.4 + 0.32 + 0.94)] = 6.34 \text{ psig}$$

$$\Delta P_5 = 0.04 \times 6.34 = \underline{0.254 \text{ psi}}$$

(vi) Total Loss in the Duct

The losses due to the presence of the two screens and the filter were not calculated because they are negligible.

$$\Delta P_T = 0.4 + 0.32 + 0.003 + 0.94 + 0.254 = 1.917 \text{ psi} \qquad \underline{2.0 \text{ psi}}$$

$$\therefore P_{dry\ BG} = 22.7 - 2.0 = 20.7 \text{ psia}$$

$$\boxed{P_{dry\ BG} \text{ leaving drier } = 20.7 \text{ psia}}$$

The discussion, with regard to the BG pressure leaving drier, in Appendix H, is valid in the present case, and is not repeated here.

6. Weight

(a) Drier

The drier is made of aluminum, except for the screens. Duct walls are assumed to be 1/8 inch thick.

Weight of outer walls (aluminum)	62	lb
Weight of cooling tubing (aluminum, 10.65 lb/bank)	223.6	
Weight of screens (2)	17.4	
Weight of desiccant	530	
Dust filter	3	
Total	836	lb

(b) Cooling Water Supply

The amount of cooling water per flight is calculated to be 155 lb or 2.5 ft^3. A small heater, to prevent the water from freezing is included.

The pressure loss suffered by the water in the drier, combustor, and the lines is rather small (about 3.5 psi). Engine bleed air is available at least at 24 psia. Consequently, air can be used to push the water out of its tank.

Weight of Tank	20 lb
Weight of Heater	1
Weight of Water	155
Total	176 lb

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APPENDIX J

EFFECT OF MOISTURE CONTENT OF

BALLAST GAS ON SUBSYSTEM WEIGHT

TABLE I-J. BALLAST GAS MOISTURE CONTENT VS SUBSYSTEM WEIGHT

Plane	Useful Life, Hrs	Temperature of BG, °F	Moisture Content, ppm v/v	Subsystem Weight, lbs	% Weight Initial Fue
SST-FP#1	50	150	10	4,640	2.3
	50	150	4,600	3,974	2.0
	no limit	100	68,400	3,132	1.6
SST-FP#2	185	150	10	8,229	4.1
	185	150	4,600	5,769	2.9
	no limit	100	68,400	3,429	1.7
Tactical	73	100	10	1,064	6.4
	73	100	1,320	961	5.8
	no limit	85	42,300	757	4.6
C-141	146	100	10	3,260	2.2
	230	100	1,320	2,663	1.8
	no limit	85	42,300	1,825	1.2

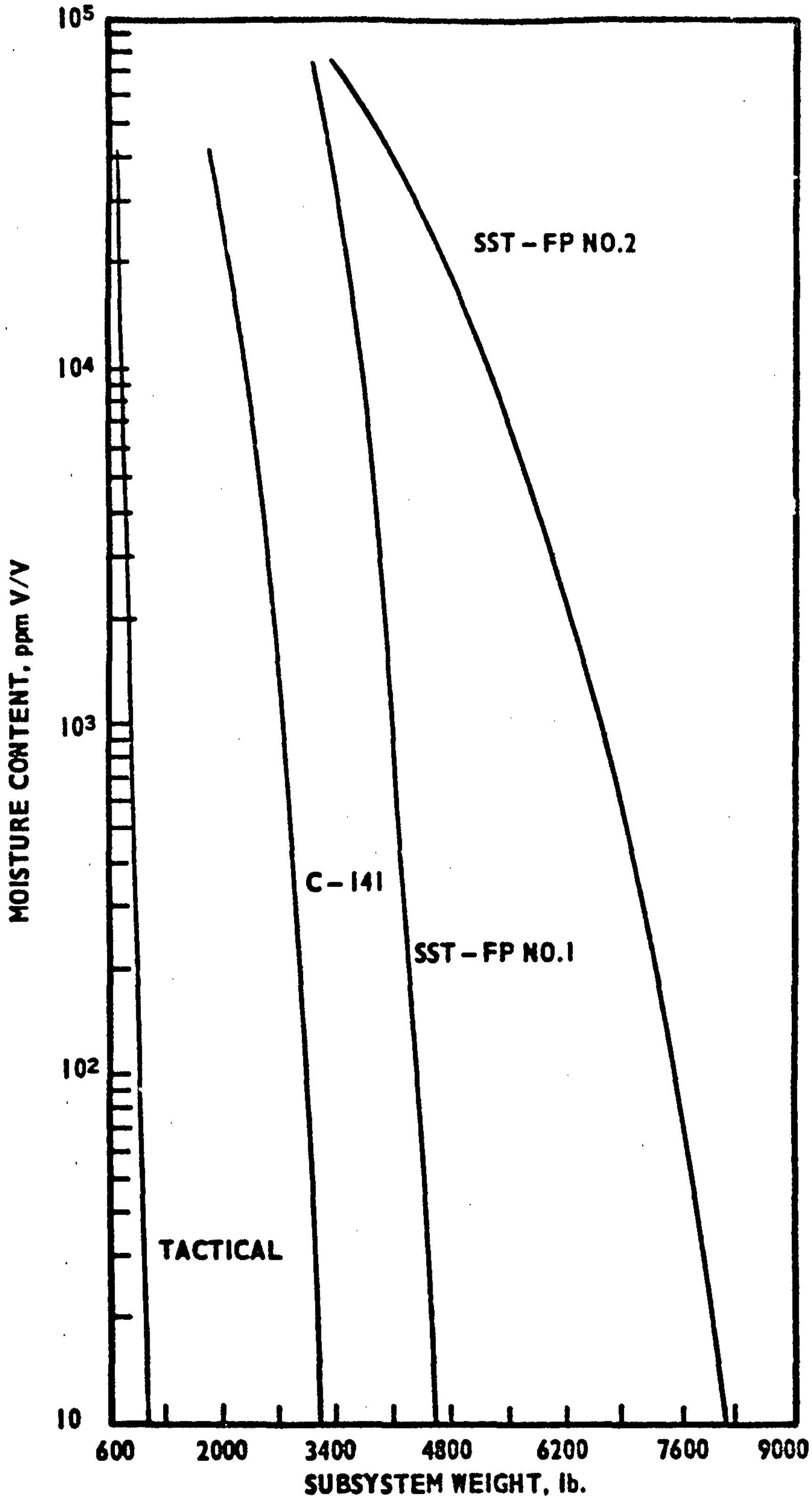

FIGURE I–J. BALLAST GAS MOISTURE CONTENT *vs*. SUBSYSTEM WEIGHT

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13. ABSTRACT

The feasibility of inerting the ullage spaces in aircraft fuel tanks via a catalytic combustion technique is evaluated. The technique utilizes nitrogen from the surrounding atmosphere as the principal component of the ballast gas admitted to the tanks. Free oxygen is reduced to safe levels by means of catalyzed reaction with a small fraction of the aircraft fuel. Before the combustion gases are admitted to the fuel tanks, the water content is reduced by condensation and by contact with a desiccant. Experiments were conducted to select and evaluate catalysts for the combustion reaction, and desiccants for water removal. Heat and material balances were prepared. Experimental and literature data were used for conceptual designs of inerting equipment that would provide target performance at all times (including powered dives) during missions typical of a tactical aircraft, a military transport, and the SST. Based on these unoptimized, preliminary designs, it was determined that complete inerting protection and control over the water admitted to the fuel tanks can be provided at a penalty of from 1.8% (transport) to 6.4% (tactical) of the initial fuel weight. These figures reflect industrial plant equipment weights, and substantial reductions are expected through use of flightweight equipment of optimized design. Recommendations are made for further study and development.

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Fire prevention					
Gas drying					
Drying agents					
Conceptual design					
Weight penalties					

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