A SUCCESSFUL INDUSTRY BASED AND ENERGY CONSERVATION RELATED SENIOR PROJECT

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ABSTRACT

The Mechanical Engineering program at Union College which is on a trimester system requires a two term Engineering Senior Project with the option of extending to three terms. The normal procedure is during the junior year each faculty member provides the students with a list of their interests and then the students can interview each faculty member that has a project of potential interest before making the final decision. The alternative procedure is for the student to define a project and then find a faculty member to work with.

Most of the students are full time day students, but there are significant numbers of students who are employed by local industries and attend part time, and these students often propose work related projects to various faculty members. Such projects have included methods for deicing airplane on ground and in flight, design and installation of an autopilot system for an airplane, analysis of air infiltration to and from hospital clean rooms and performance of tests and analysis directed toward increasing the efficiency of a cascade of power intensive industrial compressors.

This paper will describe each of these four projects and then provide a detailed description of then a more complete presentation of the tests, analysis, recommendations and results of the industrial compressor senior engineering project that is now saving the industry about $100,000 per year in electric power costs by changing the operating configuration while making recommendations for equipment and instrumentation upgrades that should yield additional savings.

Senior engineering students at Union College are required to take a two term senior project with a substantial component of design and analysis and subsequent oral and written reports.

Most full time students perform the projects in our engineering rooms, but there can be a mutual benetit for part time students who work full time to perhorm industry related senior projects.

One such student and his manager proposed analyzing a cascade of industrial compressors that now consumes electric power at the rate of more than $ 1,000,000 per year to better understand the system and to identify changes in design or operating procedures that could reduce this electric power bill.

The system was modeled, property tables developed and analysis performed based upon the 1st and 2nd law of thermodynamics. This resulted in identifying a change of operating procedures that reduced power requirements by 15 % and additional recommendations for equipment modifications and upgrades. This paper will describe this project.

1. Introduction

Laboratory **based** research development and testing of gas turbine and aircraft engine combustion processes requires a supply of compressed air at pressures, temperatures and flow rates comparable to the operating conditions from the compressors of the actual engine in which the compressor is powered by the turbine. However, in the laboratory the compressed air is typically provided by a cascade of compressors with some intercooling and driven by large nominally fixed speed three phase induction motors. The control of the output pressure and flow has been performed with a combination of throttle and dump valves in the cascade while final temperature is attained by natural gas fueled heaters. An existing system has had an energy cost in excess of $2,000 per hour. While, it was recognized that some fundamentally wasteful techniques might be contributing to this substantial operating cost, it was not clear if there could be any practical and cost effective method for reducing the power requirements and costs. Thus, the authors undertook a 1st and 2nd Law analysis for the purpose of identifying any such energy and cost saving options. It was expected that such an analysis might result in recommending modifications of the system in terms of additional intercooling variable speed motors or installation of inlet guide vanes that would require substantial capital investment and installation to realize the benefits. Thus, the authors were surprised to find that the largest initialbenefits could be achieved immediately with no capital investment but by means of some minimal changes in the operating procedures. This paper will describe the system, the author's analysis and the resulting changes in procedures that have resulted in a 15% reduction of required power and electric cost, along with describing additional computer based monitoring and equipment changes that are being considered or proposed.

2. Applied 2nd Law Thermodynamics

Irrversibilities in power producing processes such as turbines or engine cycles result in lost work, whereas irreversibilities require extra work in power consuming processes such as compressors or refrigeration cycles. The 2nd law of thermodynamics provides a tool for the quantification of irreversibilities. Previous work by the authors has successfully applied 2nd law analysis to a variety of classical engine and refrigeration cycles (References 1 through 5) to determine the lost or extra work associated with each less than ideal process as well as for the full cycle.

However, the application of such techniques can be more complicated for a set of processes such as a cascade of industrial compressors that utilize intercooling because of the typical unbalance of capability between stages and final requirement of heating or possibly cooling to provide the required end use flow rate at the required pressure and temperature. Such a system for providing compressed air for gas turbine combustion experiments has been identified and analyzed by the authors. This paper will present the results of this analysis.

3. Preparing for Analysis

The first step in performing the energy and conservation analysis was to develop a thorough understanding of the existing system in terms of equipment and operating conditions.

The first realization was that while an operational schematic diagram of the system existed as shown in Figure 1, this diagram provided much information that was not needed, but none or the specific information in terms of flows, temperatures and pressures that are required for energy use analysis.

While Figure 1 is useful for operating the system, it could not be used directly for energy use analysis. Thus, the next step was to separate the power consuming supply part of the system from the portions of the system that use this compressed air. The resulting diagram that was modified for energy analysis is shown in Figure 2.

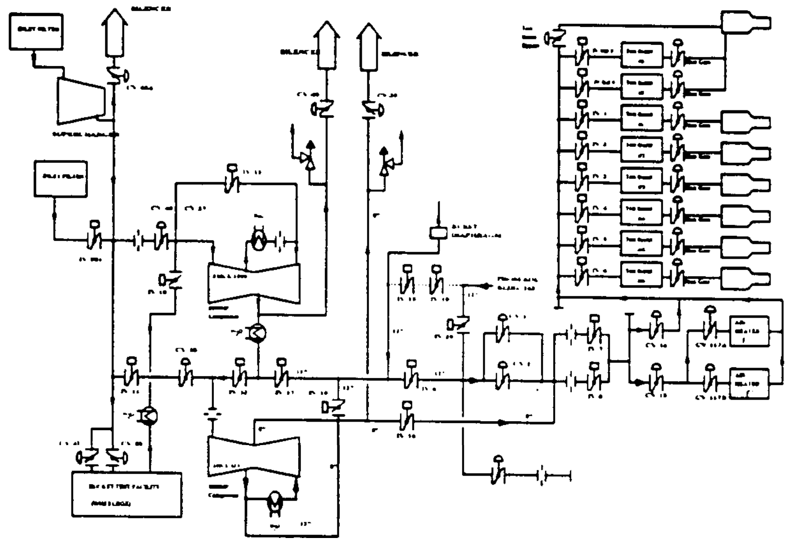

Figure 1 Operational Schematic of the System from Mets to Compressors and Test Stands and Stacks

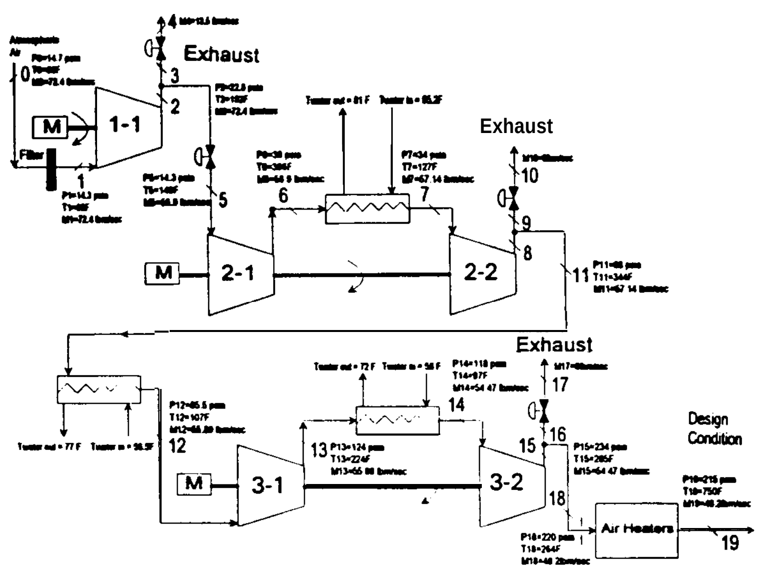

Figure 2 Supply Side Diagram for Thermodynamic, Power, Cooling and Heating Analysis

The supply system starts with atmospheric air on the upper left (#O) and ends at the lower right (#19) which is the pipe that provides air to the combustion experiments at the full load design conditions of 48.2 lbm/sec, 215 psia and 750 F.

For the purpose of thermodynamic and power usage analysis Figure 2 identifies each of the processes that can be defined as compression, intercooling pressure regulation, flow rate controlling exhaust and final heating. The connecting pipes are given identifying numbers because they represent nodes in the system in which the thermodynamic properties of pressure, temperature and flow rate can be defined, measured or calculated from measurements and displayed on the diagram.

The three intercooling heat exchangers serve the purpose of reducing power requirements by increasing the density or reducing the specific volume of the elevated temperature compressed air after three of the compressor stages. These are cooled by river water and the corresponding heat exchanger operation and performance is shown in Table I.

Now that Figure 2 has been developed and the primary thermodynamic properties of pressure, temperature and flow rate are established throughout this supply side of the system, the next step was to expand this thermodynamic information in a form presented in Property Table II to show enthalpy, specific volume and entropy along with the discharge conditions from each compressor stage if it were ideal.

Comparing the ideal compressor discharge pressure and temperature with the actual discharge temperature then allows the determination of the isentropic compressor efficiency, which is how well the compressor is performing compared to the best it could perform. This efficiency is understood to be a 2nd Law efficiency of a process.

Performing this efficiency analysis on the basis of inlet and discharge pressures and temperatures for each compressor stage indicates efficiencies of 84% 82% 84% 54% and 71% respectively for the five stages.

Table I
Interfacing Heat Exchanger Analysis

•	$T_{air\,in}$ (F)	$T_{air\,out}$ (F)	ΔT_{air} (F)	$T_{water\,in}$ (F)	$T_{water\,out}$ (F)	ΔT_{water} (F)	LMDT (F)	M_{air} (lbm/sec)	M_{water} (lbm/sec)
LP Stage Cooler	386	127	259	55.2	81	25.8	144.36	58.4	140.7
LP/HP Intercooler	340	107	233	56	77	21	113.00	57.3	152.6
HP Stage Cooler	224	97	127	56	72	16	75.06	56	106.7

Table II
Expanded Property Table for Energy Performance Ana]vsis

Pt. #	Pressure (psia)	Temp. (F)	Temp. (R)	Enthalpy h (BTU/lbm)	Specific Volume V (ft^3/lbm)	Delta Entropy S-S0 (BTU/lbm-R)	Mass Flow (lbm/sec)	Air Density lbm/ft^3
0	14.7	65.0	524.67	125.92	13.22	0.0000	72.4	0.0756
1	14.3	65.0	524.67	125.92	13.59	0.0019	72.4	0.0736
2A	22.6	152.0	611.67	146.80	10.03	0.0073	72.4	0.0997
2I	***22.6***	***138.3***	***597.97***	***143.51***	***9.80***	***0.0019***	***72.4***	***0.1020***
3	22.6	152.0	611.67	146.80	10.03	0.0073	13.5	0.0997
4	15	140.0	599.67	143.92	14.81	0.0307	13.5	0.0675
5	14.3	148.0	607.67	145.84	15.74	0.0371	58.9	0.0635
6A	38	386.0	845.67	202.96	8.24	0.0494	58.9	0.1213
6I	***38***	***343.7***	***803.41***	***192.82***	***7.83***	***0.0371***	***58.9***	***0.1277***
7	34	127.0	586.67	140.80	6.39	-0.0307	57.14	0.1564
8A	88	344.0	803.67	192.88	3.38	-0.0204	57.14	0.2956
8I	***88***	***310.2***	***769.83***	***184.76***	***3.24***	***-0.0307***	***57.14***	***0.3085***
9	88	344.0	803.67	192.88	3.38	-0.0204	0	0.2956
10	14.7	100.0	559.67	134.32	14.11	0.0155	0	0.0709
11	88	344.0	803.67	192.88	3.38	-0.0204	57.14	0.2956
12	85.5	107.0	566.67	136.00	2.46	-0.1022	55.89	0.4073
13A	124	224.0	683.67	164.08	2.04	-0.0827	55.89	0.4896
13I	***124***	***170.5***	***630.17***	***151.24***	***1.88***	***-0.1022***	***55.89***	***0.5311***
14	118	97.0	556.67	133.60	1.75	-0.1286	54.47	0.5722
15A	234	265.0	724.67	173.92	1.15	-0.1123	54.47	0.8716
15I	***234***	***217.3***	***676.94***	***162.47***	***1.07***	***-0.1286***	***54.47***	***0.9330***
16	234	265.0	724.67	173.92	1.15	-0.1123	0	0.8716
17	14.7	100.0	559.67	134.32	14.11	0.0155	0	0.0709
18	220	264.0	723.67	173.68	1.22	-0.1084	48.2	0.8206
19	215	750.0	1209.67	290.32	2.08	0.0165	48.2	0.4797

Path dependent thermodynamic functions are power and heat. They are defined by the compression and heat exchange processes. They can not be defined at a single point, but between points such as the inlet and discharge of a compressor or heat exchanger.

Thus, by examining the system diagram in Figure 2 and the heat exchanger performance in Table I, extended properties in Table II, the heat and power requirements can be summarized in Table III and the corresponding percentage and cost of power for each compressor stage is shown in Table IV.

The 1st Law teaches that energy must be conserved, while the 2nd Law distinguishes between orderly energy and random or disorderly energy. Entropy is a measure of disorder. Any irreversible process such as friction or transfer of heat over a temperature difference represents an increase of entropy and thus is a measure of the extra work that the compressors must perform.

Accordingly, each nonreversible process was identified and the corresponding rate of entropy production was calculated and presented in Table IV.

Table III
1st Law Table Showing Power, Cooling, Heating, Fractions and Costs

Reason for Irreversibility	Process #'s	Q (BTU/sec)	M * Delta h (BTU/sec)	W(comp) (BTU/sec)	W(motor) in (MW)	W(motor) out (MW)	ercentages % of Total)	Cost ($/hr)
Filter	0 - 1	0	0	0				
Supercharger	1 - 2A	0	1,512	-1,512	1.71	1.59	13.01%	$282 13
Throttle Valve	2A - 5	0	0	0				
LP Comp. Stage 1	5 - 6A	0	3,364	-3,364	3.94	3.55	28.96%	$650 77
LP Intercooler	6A - 7	-3,607	-3,607	0				
LP Comp. Stage 2	7 - 8A	0	2,976	-2,976	3 49	3 14	25 62%	$575 62
HP Comp. Precooler	8A - 12	-3,215	-3,215	0				
HP Comp. Stage 1	12 - 13A	0	1,569	-1,569	1.84	1.66	13 51%	$303 57
HP Intercooler	13A - 14	-1,682	-1,682	0				
HP Comp. Stage 2	14 - 15A	0	2,196	-2,196	2 57	2.32	18 90%	$424 82
Air Heaters	18 - 19	5,975	5 975	0				
Total Process	0 - 19	-2,528	9 090	-11,618	13 62	12 26	100 00%	$2 247 18

Table IV
2nd Law Table Identifying Irreversibilities and Entropy Production

Reason for Irreversibility	Pt. #	Pressure Ratio (P1/P0) (psia)	Temp. Ratio (T1/T0) (F)	Delta Entropy S-S1 (BTU/lbm-R)	Mass Flow (lbm/sec)	Delta S (BTU/sec*R)
Inlet Filter	0 - 1	0 9728	1 0833	0 02110	72 4	1 528
Supercharger	1 - 2A	1 5804	2 4308	0 18179	72 4	13 161
Throttle Valve	2A - 5	0 6327	0 9367	0 01569	58 8	0 923
LP Comp. Stage 1	5 - 6A	2 6573	2 6081	0 16305	58 8	9 588
LP Intercooler	6A - 7	0 9868	0 3290	-0 26589	57 14	-15 193
LP Comp. Stage 2	7 - 8A	2 3467	2 7087	0 18066	57 14	10 320
HP Comp. Precooler	8A - 12	0 9886	0 3110	-0 27949	55 89	-15 621
HP Comp. Stage 1	12 - 13A	1 4253	2 0935	0 15302	55 89	8 552
HP Intercooler	13A - 14	0 9960	0 4330	-0 20059	54 47	-10 926
HP Comp. Stage 2	14 - 15A	1 8947	2 7320	0 19738	54 47	10 751
Pipe and Control Valves	15A - 18	0 9402	0 9902	0 00332	48 2	0 160
Air Heaters	18 - 19	0 9773	2 8409	0 25217	48 2	12 154
Totals						67 14

Extra Work (I) * Ambient Temperalure (R) x Summation of Delta S

I= 34.913 BTU/sec

4. Evaluation

The previous section, figures and tables define the preparations that must be performed prior to evaluating opportunities for supplying the requited flow of air at the specified temperature and pressure while reducing the electric power requirements.

We can now examine these figures and tables to evaluate the various options we had identified such as changing operating procedures improving heat exchanger or compressor performance, installing another intercooler, installing variable speed motors for the compressors and/or installing inlet guide vanes on the first compressor stage.

OPERATION

Options for more fuel efficient operation are best determined by the examination of the pressures, temperatures and flows in Figure 2 and extended property table I.

This examination leads to the remarkable recognition that while the 1st stage compressor (l-l) is operating at a favorable 84% efficiency, the operation of this compressor is not only unnecessary, but highly wasteful as understood by examining the operation of the exhaust dump valve (3-4) and pressure reducing throttle valve (3-5).

Specifically, 18.6% of the flow in compressor stage (l-l) is dumped by valve (3-4). This is obviously wasteful, but is not necessarily avoidable since the compressor (l-l) appears to be oversized compared to the subsequent compressor stages.

However, examination of pressure reducing valve (3-5) shows that stage discharge pressure of 22.6 psia is reduced to 14.3 psia which is lower than the initial atmospheric pressure of 14.7. In addition temperature T5 of 148 F is substantially higher than the initial atmospheric temperature of 65 F.

Thus, the initial examination of the system shows that compressor stage (l-l) is not only wasteful but is unnecessary. It is consuming 13% of the total compressor power and at an electric cost of $292.13 per hour as shown in Table IV and the resulting pressure and temperature input to the next stage is worse than the initial atmospheric conditions.

Thus, if this compressor stage (l-l) and regulating valve (3-5) were completely shut down and bypassed an immediate 13% savings of cost associated with this stage would be realized along with a substantial reduction in power requirements in the later stages because of the higher initial pressure and lower initial temperature entering stage (2-l).

INTERCOOLING HEAT EXCHANGERS

Intercoolers reduce power requirements by reducing the specific volume of the air to be compressed. The compressor power is ideally proportional to the absolute temperature of the inlet air.

An option that was considered initially was to install an intercooler between stages (l-l) and (2-l) but the recognition that stage (l-l) should be bypassed eliminates need to consider this option.

The remaining three intercoolers have effectiveness values of 78%, 82% and 75.6% respectively as calculated from the data in Table II. This is reasonably good performance, but options should be considered that can provide more cooling without increasing the pressure drop of the compressed air stream.

COMPRESSOR PERFORMANCE

The calculated efficiency of stages (2-l) and (2-2) of 84% and 82% are reasonably, but the substantially lower stages (3-l) and (3-2) of 54% and 71% suggests need for more confirmation and subsequent corrective actions if these low calculated values are correct.

VARIABLE SPEED MOTORS

The existing system uses fixed speed motors for compressor drives. This makes it virtually impossible to produce the desired design conditions of specified flow rate, pressure and temperature to the combustion test stand without some fundamentally wasteful combination of throttling for pressure control and dumping excess flows.

This waste could be reduced with continually variable speed drives, but the amount of power savings will be challenging to quantify, and should be the subject of further analysis.

INLET GUIDE VANES

Inlet guide vanes have the potential to reduce power compressor power requirements under some reduced flow conditions and are established practice on a variety of aircraft and land based compressors. However, inlet guide vanes also have the potential of increasing power requirements if they are not appropriately matched to the compressor.

Ideally such vanes are a reversible process similar to a nozzle. They reduce flow while converting the pressure drop into orderly rotating kinetic energy of the stream. This higher initial velocity and kinetic energy reduces loading on the first compressor stage.

In contrast in the worst case the guide vanes will irreversibly throttle the flow to a reduced pressure and increased turbulence and aerodynamic friction similar to a pressure reducing valve.

Thus, installation of inlet guide vanes will require much additional analysis to determine whether they will provide the potential power reduction benefit.

MONITORING COMPUTERS

The existing system has an operating panel that was installed before the computer based monitoring and display capability that has become available over the last decade.

While the existing panel displays pressures, temperature, flow and some electric information, it is not in a form that provides online information of electric, mechanical and stream power, and the costs associated with each compressor and the wasted power and cost associated with dumping, throttling, and less than ideal compressors and heat exchangers.

Such a computer based monitoring system should be considered. While the specification of what is to be monitored, calculated and displayed is an open ended problem this analysis has given us an enhanced understanding of what should be displayed.

Specifically, the existing system was operating in a highly wasteful manner, but the existing monitoring system gave no indication of this wastefulness. It was only the off line analysis that was performed by the authors that led to the recognition of this substantial waste and associated substantial amounts of unnecessary extra power and cost.

According, the monitoring system should have a menu in which display and information similar to each of the figures and tables in this paper is presented on the basis of on line measurements of the conditions that can be directly measured like pressures, temperatures and flow rates along with important conditions that can be continuously calculated from these measurements like turbine efficiency, heat exchanger effectiveness, electric, mechanical and stream powers, enthalpies, entropies, irrversibilities and extra power related to the non ideal irreversible processes.

5. Conclusions and Recommendations

While this analysis was intended to identify equipment upgrades that could improve the energy efficiency of the system, the unexpected and favorable result was to identify the possibility of immediately reducing costs by 15% by simply changing the operating procedures.

Specifically, the first compressor stage that was consuming 13% of the power while serving no positive purpose, but a counter productive purpose. Not only was asubstantial amount of its flow dumped, but the remaining flow was throttled down to a pressure lower than inlet pressure and also to a higher than initial temperature, which was also wasteful.

While the need for this change in operating procedures was obvious after the fact, it was only obvious after data for a thermodynamic analysis was obtained and analyzed using 1st an 2nd law methods and considerations.

It is generally true and appropriate that the experimental part of research receives more engineering and thermodynamic analysis than the equipment that supports the experiment. In this case the experiments are to simulate gas turbine combustion processes and the supporting equipment is the cascade of electric motor driven compressors that simulate the engine compressor.

Thus, the immediate result of this analysis has been a substantial immediate savings of about 15% by changing operating procedures. Along with selective equipment upgrades, the subsequent recommendations is to install a computer based monitoring and display system that will provide enhanced online monitoring of energy flows and related costs, and with the flexibility to display other information that has not yet been identified.

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