THE TECHNOLOGY MENU FOR EFFICIENT END USE OF ENERGY

VOLUME 1

MOVEMENT OF MATERIAL

CONTENTS

Environmental and Energy Systems Studies, Lund University, Gerdagatan 13, S-223 62, Lund, Sweden
phone: int. + 46 46 108638, fax: int. + 46 46 108644

89-8-10

Introduction to

THE TECHNOLOGY MENU FOR EFFICIENT END USE OF ENERGY*

Eric D. Larson
Lars J. Nilsson
Thomas B. Johansson

CONTENTS

---

* Financial support provided by Vattenfall, the Swedish State Power Board.

INTRODUCTION

The Technology Menu is designed as a "first-stop" information resource on energy-efficient end-use technology for a broad group of energy decision makers. The Menu concisely presents technology descriptions, technical and cost data, and illustrative economic analyses, with an emphasis on advanced technology and advances in existing technologies. This is the first of several volumes that will be prepared. It addresses "Movement of Material," emphasizing industrial and commercial sector pumping, air-handling, and solids conveying applications. It is tailored to the Swedish economic context, but much of the data, analyses and discussion are relevant for users outside of Sweden as well.

USER GROUPS

The Menu is designed for two general groups of users: (1) managers and engineers with broad technical responsibilities, including energy, and (2) energy analysts and planners. The Menu presents consistent and reliable information on specific technologies, highlighting key energy aspects. The Menu does not provide complete information on all aspects of the technologies.

Managers and Engineers

This user group includes company managers, factory and consulting engineers, energy-service-company engineers, and individual consumers. For these users, the Menu is designed to facilitate the consideration of energy efficiency in routine investment decisions. Data and illustrative analyses are presented in clear and easily-digested form. The Menu is structured to permit it to be used in a variety of ways. Managers may find the economic analyses helpful in initiating energy-related investment decisions. Engineers may find data in the Menu useful for making preliminary technical and cost calculations.

Analysts and Planners

This user group includes industry and utility analysts, government planners, R&D planners, energy policy analysts, and energy conservation program leaders. From the perspective of these users, the Menu provides relatively detailed information and analysis on existing and advanced technologies. These could be used in a variety of ways. For example, analysts might construct highly disaggregated scenarios of possible future end-use energy demands (at the factory, sector, or national levels) using Menu data. Planners might use the illustrative economic analyses in the Menu to help evaluate the need for policy measures to encourage energy efficiency, to help rank technologies at which to target policy measures, or to help in setting research, development and demonstration priorities.

ORGANIZATION AND FORMAT

The Menu consists of three types of substantive entries and two supplemental documents (Fig. 1). The individual entries address (1) *Systems*: evaluations of individual technology systems that provide a particular energy service, e.g. liquid pumping, air handling, or lighting, (2) *Components*: more detailed evaluations of key components of systems, e.g. motors, variable speed drives, pumps, fans, etc; and (3) *Methods*: useful energy-analysis methods, e.g. for optimizing heat exchanger arrangements, for analyzing energy-use data, etc. This first edition of Volume 1 of the Menu includes only System and Component entries. The supplemental documents include a background discussion of the economic analysis methods and assumptions used in all entries and an annotated inventory of other important sources of end-use energy information.

System Entries

The Menu will cover seven major energy service categories and a range of technology systems which fall under each (Table 1). A separate volume in the Menu will address each energy service category. The "Movement of Material" volume has been completed and other volumes are in preparation. Within each volume, each technology system, or appropriate subset of the system will be the focus of a Menu System entry.

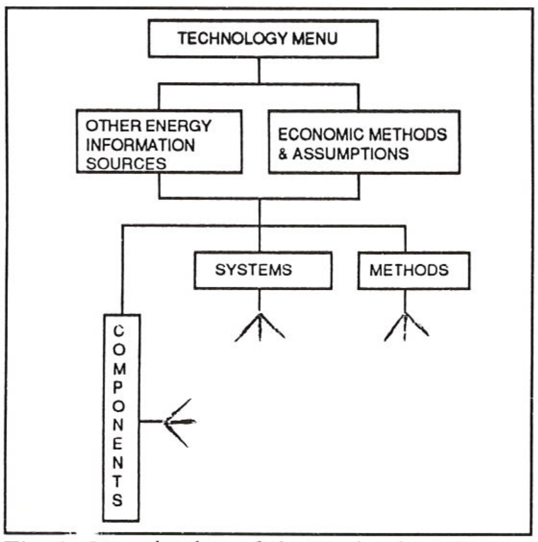


Fig. 1. Organization of the Technology Menu.

Each System entry consists of a clear, concise presentation of data and analyses using a standard format. Each entry includes a summary, highlighting key points in the entry, followed by five sections: 1. Technology, 2. R&D, 3. Technical and Cost Data, 4. Illustrative Economic Analysis, 5. References.

Technology: This section gives a general background discussion, including information such as the relevance of the technology within a broader energy context, a description of the system's operation, potential applications, its attractive features and potential drawbacks (lifetime, reliability, environmental impact, etc.), and the characteristics that qualify it as a potential energy saver.

Table 1. Energy service classifications of the Technology Menu.

A. Movement of Material
1. Liquid Pumping
2. Air Handling
4. Solids Conveying

B. Mechanical Processes (other than in A)
1. Crushing
2. Compressing (gases)
3. Cutting/etching
4. Assembling
5. Extruding

C. Heating
1. Space heating
2. Water heating
3. Cooking
4. Process heating
5. Storage

D. Cooling
1. Space cooling
2. Goods cooling
3. Process cooling
4. Storage

E. Physical/Chemical Transformation
1. Melting
2. Extruding
3. Separating
4. Drying
5. Curing
6. Welding
7. Coating
8. Chemical Synthesis
9. Cleaning

F. Lighting
1. Commercial/industrial
2. Home lighting

G. Information Handling
1. Energy Management Systems
2. Office Equipment

R&D: This section discusses research and development efforts relating to the technology and prospects for improved performance and cost characteristics over the next 5 to 10 years.

Technical and Cost Data: This section contains performance and cost data or estimates. Some effort is made to quantify future performance and cost characteristics (5-10 year perspective), as well as those for today's average and "best" commercial systems.

Illustrative Economic Analysis: This section presents the results of cost-benefit analyses, based on information in the entry, using assumptions (e.g. energy prices and discount rates) discussed in a supplemental section of the Menu (see below). The emphasis in the analysis is on understanding and illustrating potential energy and cost savings under a variety of operating scenarios and considering both private and public-sector perspectives.

References: This section contains useful references for additional information.

Component Entries

Component entries use the same format as the System entries and provide more detailed technical and cost information on the components which comprise systems. Table 2 shows the relationship between System and Component entries.

Table 2. Relationship of System and Component entries.

	SYSTEMS			
	Liquid Pumping	*Air Handling*	*Lighting*	...
COMP.				
Motors	x	x	--	
ASDs	x	x	--	
Pumps	x	--	--	
Fans	--	x	--	
Piping	x	--	--	
Ducting	--	x	--	
Lamps	--	--	x	
Ballasts	--	--	x	
-				
-				

Method Entries

Method entries describe techniques that have been developed for analyzing and/or identifying ways to improve the efficiency of energy end-use, e.g. in a factory, a commercial office building, a multi-family building, etc. Because it is expected that a large variety of methods will be covered, no standard format is specified for Method entries.

Economic Methods and Assumptions

This supplemental document discusses the economic analysis methods and assumptions used in the Menu entries. The purpose of this document is to introduce economic figures-of-merit for energy efficiency analysis for users that may be unfamiliar with them and to clearly present the assumptions (discount rate, exchange rates, energy prices, etc.) used in economic calculations in the Menu entries.

Other Energy Information Sources

A number of other energy information sources can provide useful information in addition to the Menu. The second supplemental document of the Menu consists of an annotated inventory of such sources. This document is available on request from the Menu editors.

ADMINISTRATION AND DISSEMINATION

The Menu is edited and published by the Department of Environmental and Energy Systems Studies, Lund University, Lund Sweden. Menu entries are authored by recognized experts in the field, and each entry receives technical peer review before publication to insure the accuracy and quality of the information. A small advisory board is being formed to provide general guidance over Menu activities. Vattenfall, the Swedish State Power Board, provides financial support for the Menu.

Menu volumes will initially be distributed in paper copy. Periodic updating is anticipated (perhaps annually). If there is sufficient interest expressed by users, occasional conferences or workshops may be held on various topics related to the Menu, e.g. expositions for energy-efficient equipment vendors, workshops on energy analysis techniques or on specific technologies, etc. User comments relating to any aspect of the Menu are welcomed by the editors at any time.

The first edition of Volume 1, "Movement of Material" has been completed. Availability of additional volumes is anticipated in the late spring of 1990.

TECHNOLOGY MENU
ECONOMIC METHODS AND ASSUMPTIONS

CONTENTS

SUMMARY

This document gives background information relating to the economic assessments in Technology Menu entries. An introduction is given to standardized economic measures for energy conservation analysis. Basic assumptions used to assess the economics of energy efficiency investments in the Swedish context are also discussed. Also given are estimated costs of new electricity supply, which are used in some Menu entries for comparisons to the cost of saved electricity.

STANDARD ECONOMIC MEASURES

This section discusses the discount rate and four economic measures used in the Menu to evaluate the cost-effectiveness of energy conservation investments: simple payback, lifecycle cost, internal rate of return, and cost of saved energy. The discussion here is intended to simply introduce the key ideas relating to economic evaluations of energy conservation to Menu users that may be unfamiliar with these.

Discount Rates

Investing in energy efficiency often involves capital spending today to save operating costs tomorrow. The discount rate is important in evaluating such options, as it permits quantitative comparisons to be made involving money spent or saved at different times. The discount rate can enter into the calculations in a variety of ways.

It is typically used in calculating the present value of future expenditures. For example, assuming a device uses X kWh/yr of electricity costing Y dollars per kilowatt-hour ($/kWh), the annual costs in today's dollars (assessed at the end of each year) would be:[1]

Year	$/yr (today's $)
1	$(X*Y)/(1 + i)$
2	$(X*Y)/(1 + i)^2$
3	$(X*Y)/(1 + i)^3$
...	...
n	$(X*Y)/(1 + i)^n$

where i is the discount rate.

What discount rate should be used in a particular economic assessment? Unfortunately, there is no theoretically correct discount rate. In general, however, the discount rate is chosen to reflect how future cash flows are valued. In real terms,[2] they can range from 5 to 10 percent (typical of bank interest rates on private savings up to rates of return typically required by corporate investors from long-term investments) up to 100 percent and higher (rates implicit in many individual consumer purchase decisions). For additional discussion of discount rates in energy conservation analysis, see [1,2]. Specific discount rates used in Menu analyses are discussed below.

Simple Payback

The simple payback (SP) is the simplest and probably most commonly used measure for evaluating the cost-effectiveness of an energy saving investment. It is:

$$SP = IC/YOCS$$

where IC is the initial capital cost and YOCS is the first year's operating costs savings expected as a result of the investment.

The simple payback is conceptually simple and easy to calculate. However, it can give misleading results in ranking energy efficiency investments by cost effectiveness since it does not take account of the time value of money nor the expected lifetime of the investment. The consumer seeking paybacks of 2 or 4 years is implicitly assuming a discount rate of about 50% or 25%, respectively,[3] which are much higher than would typically be used to calculate cost-effectiveness using other measures, e.g. the lifecycle cost. At a 50% discount rate, the consumer would be valuing a dollar saved 10 years from now at less than 2 cents today.[4]

Lifecycle Costs

The lifecycle cost, LCC, is the present value of all costs (initial capital, interest payments on loans, and operating costs) associated with an investment over its lifetime. Unlike the simple payback, the LCC provides a means of ranking the cost of alternative conservation investments, because it takes account of the time value of money and the lifetime of the investment.

Calculating the LCC requires specification of the discount rate and future energy prices. The lifecycle cost is:

$$LCC = IC + \sum_{n=1}^{N} E_n * P_n * (1+i)^{-n} + \sum_{n=1}^{N} NEC_n * (1+i)^{-n}$$

where IC is the initial capital cost, E_n is the energy use in year n (e.g. in kWh/yr), P_n is the energy price in year n (e.g. in \$/kWh), NEC_n is the non-energy cost in year n (e.g. labor and maintenance), i is the discount rate, and N is the expected lifetime of the investment (or amortization period) in years. In many cases it can be assumed that E, P and NEC are constant from year to year, which simplifies the calculations.[5]

In cases where energy prices are expected to rise in real terms from the present price, P_o, at a rate of r percent per year, then in year n the price of energy would be

$$P_n = P_o * (1 + r)^n$$

For this case, the second term of the LCC equation can be rewritten as

$$\sum_{n=1}^{N} E_n * P_o * (1 + j)^{-n}$$

where $j = (i - r)/(1 + r)$.

It is often convenient to compare *annualized lifecycle costs*, ALCC. The ALCC is the equal payment (in discounted present value) that would be required each year to fully pay the lifecycle costs associated with an investment over its lifetime. It is also referred to as the levelized cost and can be calculated as the product of LCC and the *capital recovery factor*, CRF, where

$$CRF(i,N) = 1/\sum_{n=1}^{N} (1 + i)^{-n}$$

and i and N are as defined previously.

Since

$$\sum_{n=1}^{N} (1 + i)^{-n} = [1 - (1 + i)^{-N}]/i$$

the CRF can also be expressed in closed form as

$$CRF(i,N) = i/[1 - (1 + i)^{-N}]$$

The LCC and ALCC are useful both in making comparisons between different new investment options as well as in assessing *retrofits*. In the latter case, if the ALCC for an investment is less than the annual operating costs plus salvage value of the existing system it would replace, it would make economic sense to replace the existing system even if it had many remaining years of useful life.

Internal Rate of Return

Another approach to ranking energy saving options is to consider each as an investment opportunity, with an associated internal rate of return (IRR). The IRR is the discount rate for which the lifecycle cost would be zero. It can be calculated by setting the above equation for LCC equal to zero and solving for i.

Calculating the IRR does not require specification of the discount rate, though future energy price assumptions are required. A minimum IRR, called the "hurdle rate," is typically used by corporate investors to determine whether an investment should be made, and investments with higher IRRs would be made first. Hurdle rates for long-term investments in industry typically range from 10 to 15 percent (real).

Cost of Saved Energy

The cost-of-saved-energy, CSE, provides a measure for ranking energy saving and energy supply projects on a consistent basis and is particularly useful in identifying the most economically efficient investments from a societal or national perspective.

The cost of energy supply is often calculated as a levelized value over the life of the supply investment. The CSE is calculated as the annualized non-energy lifecycle costs associated with an efficiency investment divided by the expected annual energy savings [3].

The production or saving of electricity provides a simple illustration. The cost of electricity from a new coal-fired central station power plant is typically given in terms of a levelized $/kWh over the life of the plant. An investment to save electricity might be considered instead, e.g. paying an extra cost for higher efficiency motors instead of buying standard ones. Assuming the same discount rate as used in the supply calculation, a CSE associated with the higher efficiency motors can be calculated and compared to the cost of new supply. If the CSE is less than the cost of new electricity supply, the electricity saving investment makes better economic sense. (See the Menu entry on motors for such comparisons.)

In making comparisons between costs of saving and supplying energy, it should be noted that the value of a unit of saved energy can vary. For example saving a kWh of electricity during a period of peak demand saves society (and often the consumer) more, in economic terms, than saving the kWh during periods of lower demand. See [4] for more discussion on this subject.

Limitations of Economic Analysis

The above measures are useful in assessing the relative economic efficiency of energy-related investments, but have some limitations.

Future energy prices are difficult to predict. Thus, there is some inherent uncertainty in calculations where these must be specified (SP, LCC and IRR). It is often useful, therefore, to evaluate the economics of the investment for a range of possible prices to better understand the sensitivity of the economics to energy price.

Many energy decisions involve external social costs or benefits that are difficult to quantify: What is the value of increased energy security to a country from lessening its dependence on imported oil by energy saving investments? What are the costs of damage to the environment resulting from increased energy supply, e.g. acid rain or global warming? Such questions cannot be answered directly by economic cost-benefit analysis. However, when the economic assessments are ambiguous, considerations of such externalities may weigh more heavily in the ultimate choice.

Another difficulty with cost-benefit analysis is that it typically cannot take account of multiple benefits that are often derived from an investment motivated strictly by energy concerns. If an economic comparison is ambiguous on energy grounds alone, then consideration of other benefits becomes more important.

ASSUMPTIONS FOR THE SWEDISH CONTEXT

The analysis in Volume 1 of the Menu is tailored to the Swedish economic context. This section gives the underlying assumptions used in Menu calculations.

General: All monetary values are expressed in 1988 US dollars. Direct conversions have been made from the Swedish kronor (SEK) values used in the Swedish-language edition of the Menu. An exchange rate of 6.5 kronor per dollar is used. Swedish GDP deflators [5] and US GNP deflators [6] have been used where needed to convert values given orignally in those of other years. To simplfy the analysis and make it of more general interest, all taxes, tax credits, subsidies, etc. are excluded.

Discount Rates: All discount rates used are inflation-corrected (real) rates. Calculations from the individual consumer or industrial perspective use higher discount rates (10-20%) than calculations taking the social or national perspective (6%). The latter rate is used in all calculations where energy supply and energy savings are compared, e.g. in cost-of-saved-energy calculations.

Electricity Prices: Volume 1 of the Menu emphasizes industrial and commercial sector applications. In Sweden today, average electricity prices for industrial/commercial users ranges from about 2.3 cents/kWh (0.15 SEK/kWh) for large users to 3.4 cents/kWh (0.22 SEK/kWh) for smaller users [7].[6] This range sets the lower limit for Menu calculations. A range from 3.9 cents/kWh (0.25 SEK/kWh) to 7.8 cents/kWh (0.50 SEK/kWh) is used in price sensitivity analyses. These prices represent levelized prices over the lifetime of the investment being analyzed.

Oil Prices: Future oil prices in Sweden are uncertain. In the Menu, crude oil is assumed to cost between \$20 and \$30 per barrel (817-1226 SEK/m^3 or 0.076-0.115 SEK/kWh).

Natural Gas Prices: Future natural gas prices in Sweden are uncertain. Sweden's National Energy Administration indicates that gas prices for perhaps the next decade may range from \$3.4/GJ to \$5.1/GJ (0.08-0.12 SEK/kWh) on a lower heating value basis [8]. This range is used in the menu assessments.

Coal Prices: The price of coal is of interest to the Menu primarily in estimating the cost of new electricity supply (see below). Future coal prices are less uncertain than gas or oil prices, since global reserves are large, producers are more numerous than with oil, and there are no geographical limits on coal trade as there are with natural gas. Coal prices can be expected to be substantially the same in most places in the world. The US Department of Energy projects a coal price of about \$1.9/GJ (0.045 SEK/kWh) for electric utilities in the US in 1995 [9].

Biomass Prices: Biomass is a potentially important fuel in Sweden for new electricity supply, and biomass prices are used in the electricity supply costs shown below. The present market price for fuelwood chips from forestry residues is approximately \$4.3/GJ (0.10 SEK/kWh) (50% moisture content) on a lower heating value basis [10], which reflects the costs of separate recovery of fuelwood from industrial feedstocks. It has been estimated that integrating the two recovery process would reduce the cost of forest residues to \$2.6-3.4/GJ (0.06-0.08 SEK/kWh) [11]. It has also been estimated that fuelwood chips could be produced on energy plantations for \$3.0-4.3/GJ (0.07-0.10 SEK/kWh) [12,13].

ESTIMATED COSTS OF NEW ELECTRICITY SUPPLY

In some of the Menu entries, the cost-of-saved-electricity is compared with the cost of new electricity supply to assess the relative attractiveness of the two options from a national perspective. Estimates of the levelized cost of new baseload electricity supply with a variety of fuels and generation technologies are given in Table 1.

Table 1. Estimated costs of baseload electricity from new power plants.

FUEL ----->	COAL							
	Condensing[d]		PFBC-CC[e]		IGCC[f]		STIG[g]	STIG[g]
Size (MW)	200	500	250	500	100	600	2x50	110
Eff.[a] (%)	34.6	34.6	39.2	39.2	34.3	37.9	35.6	42.1
Cost ($/kW)	1960	1470	1850	1650	2830	1620	1330	1070
Busbar Cost (c/kWh)								
Capital[b]	2.32	1.74	2.19	1.95	3.35	1.92	1.58	1.27
Fuel[c]	2.00	2.00	1.77	1.77	2.02	1.83	1.94	1.64
O&M	1.40	1.10	1.78	1.56	2.21	0.84	1.04	0.89
TOTAL	**5.72**	**4.84**	**5.74**	**5.28**	**7.58**	**4.59**	**4.56**	**3.80**

	NATURAL GAS[h]		
	Condensing	**Advanced CC**	**ISTIG[g]**
Size (MW)	2x500	205	110
Eff.[a] (%)	36.3	45.0	47.0
Cost ($/kW)	780	530	440
Busbar Cost (c/kWh)			
Capital[b]	0.92	0.63	0.52
Fuel[c]	3.85	3.11	2.98
O&M	0.51	0.30	0.30
TOTAL	**5.28**	**4.04**	**3.80**

	BIOMASS[i]	
	STIG	**ISTIG**
Size (MW)	53	110
Eff.[a] (%)	32.5	38.4
Cost ($/kW)	1130	910
Busbar Cost (c/kWh)		
Capital[b]	1.34	1.08
Fuel[c]	3.05	2.58
O&M	1.14	0.97
TOTAL	**5.53**	**4.63**

(a) Higher heating value and for operation at 100% load.

(b) For a 6% real discount rate, 30-year plant life, and 70% capacity factor. No taxes or tax incentives are included.

(c) Assumed fuel costs on a higher (lower) heating value basis are: Coal, $1.90/GJ ($1.96/GJ); Natural Gas, $3.9/GJ ($4.3/GJ); Biomass (15% moisture content), $2.7/GJ ($3.0/GJ).

(d) Unit capital costs, efficiencies and O&M costs are for coal-fired condensing plants with wet flue gas desulfurization, as estimated by the Electric Power Research Institute (EPRI) in the USA [14].

(e) All estimates for the pressurized-fluidized-bed- combustion combined-cycle are from EPRI [14] for a combined cycle with a gas-turbine inlet temperature of 843°C and steam conditions of 163 bar, 538°C.

(f) EPRI estimates [14] for integrated-gasification combined-cycle systems based on the Texaco oxygen-blown gasifier with cold gas cleanup and combined cycles with a turbine inlet temperature of 1093°C, except for the 600 MW unit, which utilizes an advanced combined cycle with a turbine inlet temperature of 1204°C.

(g) Based on steam-injected (STIG) or intercooled steam-injected gas turbines (ISTIG) coupled to Lurgi air-blown, dry-ash gasifiers with hot particulate and sulfur clean up [15]. Neither of these systems has been commercially demonstrated.

(h) Performance and costs from [16]. The advanced combined cycle, based on the Frame 7F recently introduced by General Electric, has a gas turbine inlet temperature of 1260°C.

(i) The performance and costs are from [17]. The smaller (larger) plants are based on an LM-5000 STIG (ISTIG) coupled to an air-blown fluidized-bed gasifier with hot particulate removal. The O&M cost in each case is assumed to be the same as for the corresponding coal-fired STIG or ISTIG, with an upward adjustment for the lower assumed plant efficiency with biomass. Biomass-gasifier gas turbine technology has not been commercially demonstrated.

REFERENCES

1. J. Goldemberg, T.B. Johansson, A.K.N. Reddy, and R.H. Williams, *Energy for a Sustainable World*, Wiley-Eastern, New Delhi, 1988.

2. R.H. Williams, "Innovative Approaches to Marketing Electric Efficiency," *Electricity*, Johansson, Bodlund, Williams (eds), Lund University Press, Lund, Sweden, 1989.

3. A. Meier, J. Wright, and A.H. Rosenfeld, *Supplying Energy Through Greater Efficiency*, University of California Press, Berkeley, 1983.

4. J. Goldemberg and R.H. Williams, "The Economics of Energy Conservation in Developing Countries," in *Energy Sources: Conservation and Renewables*, American Institute of Physics, New York, 1985.

5. *1989 Statistical Abstract of Sweden*, Statistics Sweden Pub., Stockholm, 1988.

6. Bureau of Economic Analysis, US Department of Commerce, Washington, DC.

7. National Energy Admin., "Elmarknaden (The ElectricityMarket," 1987-12-17, Stockholm, 1987. (in Swedish)

8. National Energy Admin., "El- och varmeproduktion med naturgas (Electricity and Heat Production with Natural Gas)," 1987:R5, Stockholm, 1987. (in Swedish)

9. Energy Information Admin., *Annual Energy Outlook 1986, With Projections to 2000*, DOE/EIA-0383(86), US Government Printing Office, Washington, DC, 1987.

10. National Price and Cartel Office, "Energiaktuellt (Energy News)," 1988:8-9, Stockholm, 1988. (in Swedish)

11. A. Hildeman, G. Rutegard, and G. Tibblin, *Traddelsmetoden-Integrerad Produktion av Skogsbransle och Massaravara (The tree-Section Method-Integrated Production of Wood Fuel and Pulp Feedstock) (Forest-Industry Market Study No. 19)*. Swedish Univ. of Agric. Sci., Uppsala, 1986. (in Swedish)

12. National Energy Admin., *Energiskog (Energy Plantations) 85:9*, Stockholm, 1985. (in Swedish)

13. M. Parikka, *Ekonomisk Analys av Energiskogsodling (Economical Analysis of Short Rotation Forestry)*, Swedish Univ. of Agric. Sci., Uppsala, 1988. (in Swedish)

14. Planning and Evaluation Division, *Technical Assessment Guide, Vol. 1: Electricity Supply--1986*, P-4463-SR, Electric Power Research Institute, Palo Alto, Cal., Dec. 1986.

15. J.C. Corman, "System Analysis of Simplified IGCC Plants," for US Dept. of Energy by General Electric Corporate Research Center, Schenectady, New York, 1986.

16. R.H. Williams and E.D. Larson, "Expanding Roles for Gas Turbines in Power Generation," in *Electricity*, Johansson, Bodlund, and Williams (eds), Lund University Press, Lund, Sweden, 1989.

17. E.D. Larson, P. Svenningsson, and I. Bjerle, "Biomass Gasification for Gas Turbine Power Generation," in *Electricity*, Johansson, Bodlund, Williams (eds), Lund University Press, Lund, Sweden, 1989.

NOTES

1. Assuming the electricity price stays constant (in real, inflation-corrected terms). The case of escalating energy costs is considered later.

2. All discount rates used in the Menu are given in "real" terms, i.e., corrected for inflation. Thus, a 5 percent real discount rate corresponds to a 9 percent discount rate in current dollars during a time of 4 percent inflation.

3. Assuming a 15 year investment lifetime.

4. $ $1.0 * [1/(1+0.5)^{10}] = $0.017 $.

5. In many cases the energy price used in the LCC calculation will actually represent an average of several charges. For example in industry, the average electricity price would typically include demand charges ($/kW) plus energy charges (cents/kWh) that vary depending on whether they are used during peak, intermediate or baseload periods.

6. These are exclusive of taxes, which amount to about 0.5 cents/kWh (0.03 SEK/kWh) and 0.8 cents/kWh (0.05 SEK/kWh) for large and small users, respectively.

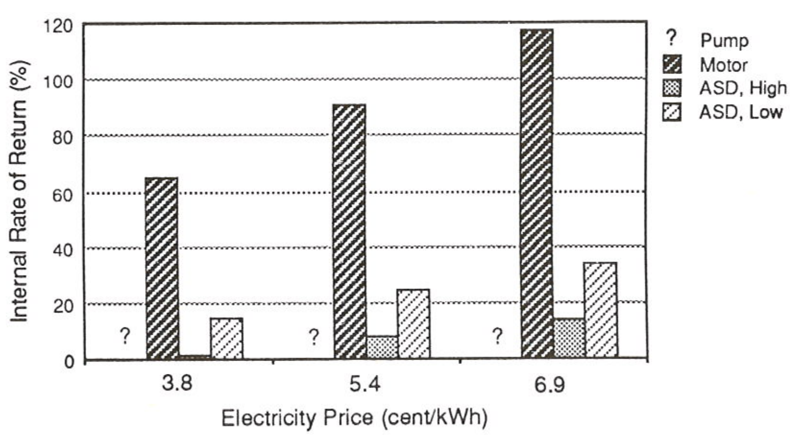
Vol. 1: Movement of Material **89-8-8**
***System*: Liquid Pumping**

by
Lars J. Nilsson, Eric D. Larson
Environmental and Energy System Studies
University of Lund, Lund, Sweden

CONTENTS

SUMMARY

Pumping accounts for a large amount of electricity use in Sweden. The efficiency of typical pumping systems can be improved, often significantly, with more efficient system components and better system design. In variable duty-cycle applications, the use of adjustable-speed-drives can significantly reduce throttling losses. Also, reducing system losses, e.g. by increasing pipe diameter, could save large amounts of electricity. Rates of return on the extra investments for improved efficiency can be relatively high (see below and text). Correspondingly, the cost-of-saved-electricity associated with the conservation investments can be low compared to the cost of new electricity supply.



1. TECHNOLOGY

Introduction

Pumping accounts for a large portion of the total electricity consumption in most industrialized countries, e.g., an estimated 25% in the USA [1]. For the major electricity-intensive industries in Sweden,[1] approximately 30% of the total consumption of 33 TWh in 1986 was due to pumps and fans [2]. The total number of pumps sold in Sweden is more than 200,000 per year.

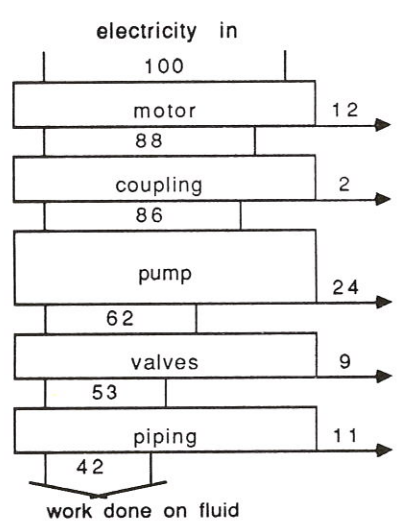


Figure 1. Energy balance for a typical pumping system [3].

Typically much less than half the electricity input to a pumping system is converted into useful movement of fluid. The rest is dissipated by the various components that make up the system (Fig. 1). Energy losses are still greater when the system is not operating at its design point. Thus, there appears to be a considerable potential for saving electricity, both by improving component efficiencies and through better system design.

Head-Flow Curves

Head-flow curves (e.g. Fig. 2) are important aides to understanding the performance of pumping systems. The system curve indicates how the head (pressure) at the pump outlet increases (due to downstream friction in the system) as the flow increases. The pump curve shows how head varies with flow for a pump that operates at a fixed rotational speed and also indicates pump efficiency at different operating points. The intersection of the two curves represents the actual operating point. The interaction of all system components help determine this operating point.

The main components of a pumping system include the components of the pumping equipment--pump, motor, and drive--and those of the rest of the system--piping and valves in most cases.

Pumps [4]

About 3/4 of all pumps in industrialized countries are of the centrifugal type, based on estimates for the USA [1]. These are estimated to account for more than 90% of all power consumed in pumping systems. The analysis in this entry is focussed on pumping systems with centrifugal pumps, though much of the discussion is also valid for other types of pumps.

Typical design-point efficiencies for centrifugal pumps range from 30-70% for small pumps and 80-90% for larger pumps, depending on their flow, head and rotational speed. The specific speed of the pump is a dimensionless parameter incorporating these three factors. Pumps with higher specific speeds generally have higher efficiencies in a comparable application. When pumps are operated away from the design point efficiency falls, often significantly. This would be the case, for example, with pumps that are oversized or undersized.

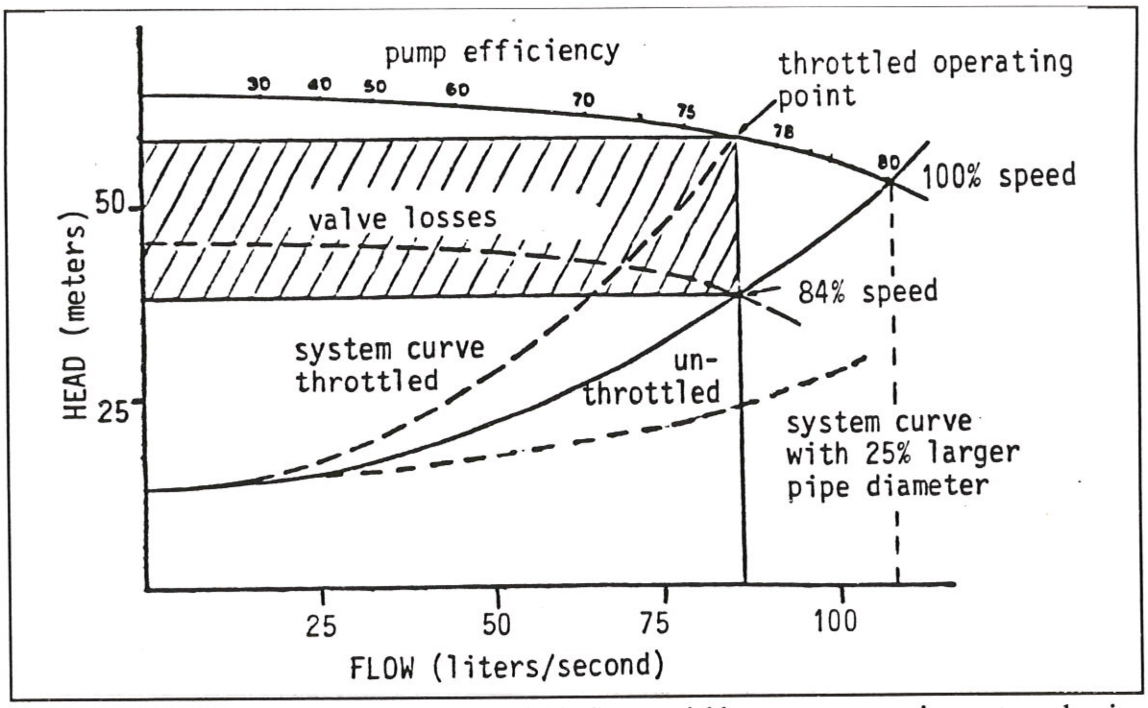


Figure 2. **Typical head-flow curves for a variable-flow, variable-pressure pumping system showing** throttled versus reduced speed operation and the effect of reducing piping losses by increasing piping diameter (adapted from [13]). **With the increased pipe diameter a 45-kW motor/pump would be sufficient, compared to a 75-kW motor in the original system.**

Energy efficiency is receiving increased attention from pump producers, but has historically not been a high priority and generally appears not to be reflected in the list price of the pump. Factors such as reliability and ease of maintenance appear to weigh more heavily in consumer purchase decisions. It appears that design-point efficiencies can vary markedly for different pumps with similar ratings and list prices.

Motors [5]

The predominant type of motor used in industrial pumping applications is the polyphase induction motor. "Standard" motors have efficiencies of 70-85% in smaller sizes (< 10 kW) up to 90-95% in larger sizes (> 150 kW). For a given application, a variety of motors can often be used, each of which may give a different efficiency. Motors loosely defined as energy-efficient motors (EEMs) are available with 2-10 percentage points higher efficiencies, with the larger improvement coming in the smaller motors. Some vendors offer both standard motors and EEMs, with the latter having a higher first cost and comparable reliability as the standard models. The extra cost per kW is highest in the smaller sizes.

Flow Control [6]

A variety of techniques are available for controlling the output of pumps. Most pumps operate at a constant rotational speed, driven by single-speed motors directly (or through gearing). In such systems the flow or head is controlled by using throttle and by-pass valves downstream of the pump (Fig. 3a). Where the required flow or head varies over time (Fig. 4), large energy losses are associated with such fixed-speed systems, because pump output varies directly with pump speed, but required input power varies with the cube of speed.[2]

Energy that is lost in such fixed-speed systems can be saved by matching the pump speed to the load. It has been estimated that some 4-5 TWh/yr of

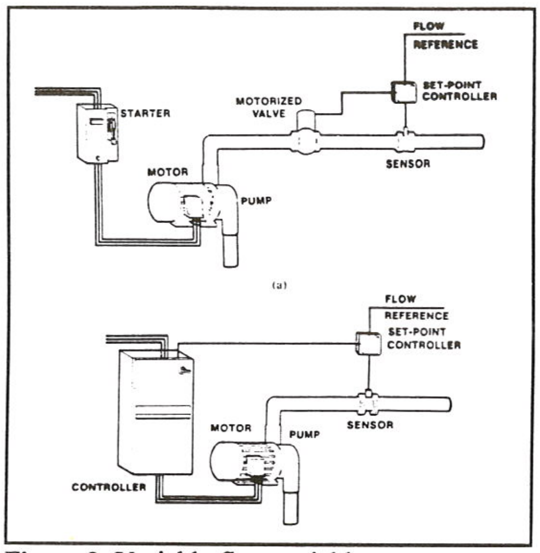


Figure 3. Variable-flow variable-pressure pumping system with (a) throttle-control of fixed-speed pump and (b) adjustable speed control [13].

electricity could be saved in Sweden by using speed control for pumps and fans [6].[3] Speed adjustment can be achieved through use of various types of couplings between the pump and motor or by adjusting the speed of the motor directly. Particularly noteworthy are recent developments in electronic adjustable speed drives (ASDs) that continuously control motor speed (Fig. 3b) and typically have drive efficiencies of 95% and higher.[4]

Costs for ASDs and their installation are relatively scale-sensitive and can vary significantly from one application to another. Cost ranges are given in [6]. Apart from electricity savings, continuous regulation can also lead to better process control, lower maintenance costs (e.g. greater pump reliability and longer pump-seal life), improved equipment life (e.g. through elimination of water hammers), and improved working environment (e.g. through less vibration and noise).

System Considerations

In addition to considering the components discussed above, careful design or redesign system-wide can lead to improved energy efficiency. The design of piping and valves are particularly important [7]. Pipe diameter is a key factor, since using a larger pipe can reduce piping and valve friction losses substantially.[5] For a fixed diameter, smoother pipe surfaces and properly designed bends can contribute to lower pressure drops.[6] Balancing considerations are that pipe costs increase with diameter and smoothness.

System optimization can also be undertaken to improve overall efficiency. In an analysis by Danish engineers of small-scale pumping applications on a component-by-component basis (motor, drive, pump, piping layout, valve control system, etc.), potential electricity savings of over 70% were identified for typical systems [8]. Because of the complex interactions among pumping-system components, particularly for large industrial applications, computer software can be an effective tool for helping to design efficient systems [9].

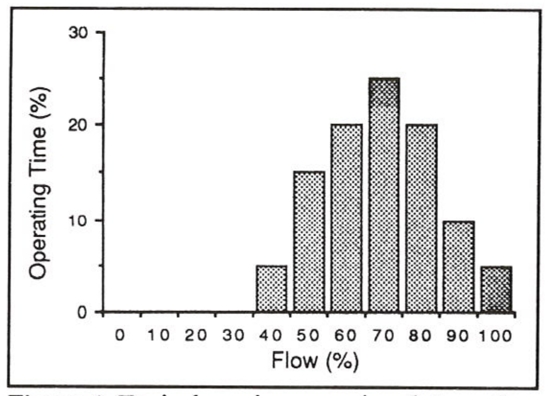


Figure 4. Typical varying pumping duty cycle [13].

2. R&D

R&D activities to improve pumping systems are directed at improving the efficiencies and/or reducing the cost of specific components within the system. Particularly notable is work on the more efficient motors utilizing soft magnetic materials and permanent magnets [10], and the trend toward capital cost reductions in electric ASDs

Table 1. Hypothetical energy savings with indicated efficiency improvements to a standard fixed-speed pumping system (based on [10])

Component	Base[a]	---------- MODIFICATIONS INVOLVING -------- Motor[b]	Pump[c]	ASD[d]	ALL[e]	+ Pipe[f]
ASD	--	--	--	95	95	95
Electric Motor	92	94.5	92	91	93.5	93.5
Shaft Coupling	98	98	98	--	--	--
Pump	77	77	80	79	82	82
Throttle Valve	66	66	66	--	--	--
Piping	35	35	35	35	35	56
Total Efficiency (%)	16.0	16.5	16.7	23.9	25.5	40.8
Percent electricity saved over base-case		3.0	4.2	33.0	37.3	60.8

(a) The base-case pumping system is assumed to be that indicated in Fig. 2, operating at approximately 80% flow (at the point designated "throttled operating point" in Fig. 2). The flow and head at the operating point are approximately 86 m^3/s and 60 m, respectively. A 75 kW motor is required. Motor and pump efficiencies are based on [5] and [4]. Piping efficiency is assumed to be the ratio of static pressure at zero flow to total head less valve losses [10].

(b) Based on [5] for an energy efficient motor.

(c) Based on [4].

(d) Based on [6,10]. It is assumed the ASD eliminates the shaft coupling and throttling valve. The motor efficiency is assumed to be reduced compared to the base case due to ASD harmonics. With the ASD, the pump operates closer to its optimum efficiency, so pump efficiency is better than in the base case (see endnote 8).

(e) Assumes use of an ASD and improved pump and motor. Motor efficiency is reduced 1 percentage point due to ASD harmonics.

(f) The pipe diameter is assumed to be increased by 25% (see endnote 5). In this case, a smaller capacity motor, pump, and ASD could be used compared to the base case.

with further developments in solid-state power electronics [11]. More sophisticated system design tools (e.g. see [9]) could also lead to improved pumping-system efficiencies by helping to identify possible reductions in overly conservative safety factors that characterize many traditional designs [12].

3. TECHNICAL AND COST DATA

Performance Improvement--Fixed Flow

The great diversity of specific pumping applications makes it difficult to draw general conclusions regarding the potential for saving electricity by improving pumping system efficiencies. However, a simple exercise helps to illustrate the possibilities.

Table 1 shows potential energy savings achievable with various improvements to the "standard" fixed-speed-motor pumping system described by Fig. 2. The base-case in Table 1 corresponds to operation at the point on Fig. 2 marked "throttled operating point" (80% of design flow), with representative efficiencies assumed for the motor and shaft coupling. In the base case, the total system efficiency in converting electricity to energy in the fluid is only 16%, with the largest losses due to throttling and piping.

To improve system efficiency, an energy efficient motor could be used in place of the standard model, which would save about 3% of input electricity needs. Improving pump efficiency would produce a comparable savings. Installing an adjustable speed drive would eliminate the throttling losses and save 1/3 of input electricity. (The savings would be greater still with further reduced flow, as discussed below.) Choosing piping with a 25% larger diameter, together with the ASD and more efficient motor and pump would lead to electricity savings of over 60% while supplying the same pumping service.

All of the modifications shown in Table 1 could be made as retrofits. In the case where larger piping is used, however, the system characteristics would be altered signficantly (as shown in Fig. 2), so that a different (smaller) motor, pump, and ASD would be required. Such a major retrofit is very rare, but because of the large calculated electricity savings potential, such an alternative should probably be considered seriously in evaluating alternative new designs.

Performance Improvement--Variable Flow

The calculation of potential electricity savings is more involved with variable-head or variable-flow applications. A simple case is presented here to illustrate the potential savings. Pumping systems generally fall into one of three categories, depending on the service they provide [13]: variable-flow variable-pressure (VFVP), constant-flow variable-pressure (CFVP), and variable-flow constant-pressure (VFCP). The most common of these, the VFVP system, is considered here.

In a VFVP system with throttle control (Fig. 3a), flow is reduced as head increases, following the 100% pump-speed curve (Fig. 2). Note that pump efficiency falls as indicated in Fig. 2. A more efficient motor or pump can be used to save some electricity. With ASD flow control (Fig. 3b), pump speed is reduced, following the unthrottled system curve in Fig. 2.[7] The difference in head between the throttled operating point and reduced-speed unthrottled operating point represents throttling-valve losses, or potential energy savings with ASDs. A synergistic effect is that pump efficiency is higher at the reduced-speed point than at the throttled point.[8]

The actual electricity savings at any operating point can be determined from head-flow curves for the system and pump (as in Fig. 2) and efficiency curves for the motor/drive system.[9] The savings based on such calculations for the VFVP system shown in Fig. 2 (rated at 75 kW to pump a maximum of 108 liters/sec through a pressure head of 55 m) are shown by the upper set of curves in Fig. 5. With throttled operation, the input power requirement falls modestly with reduced flow, because the effect of flow reduction is greater than the

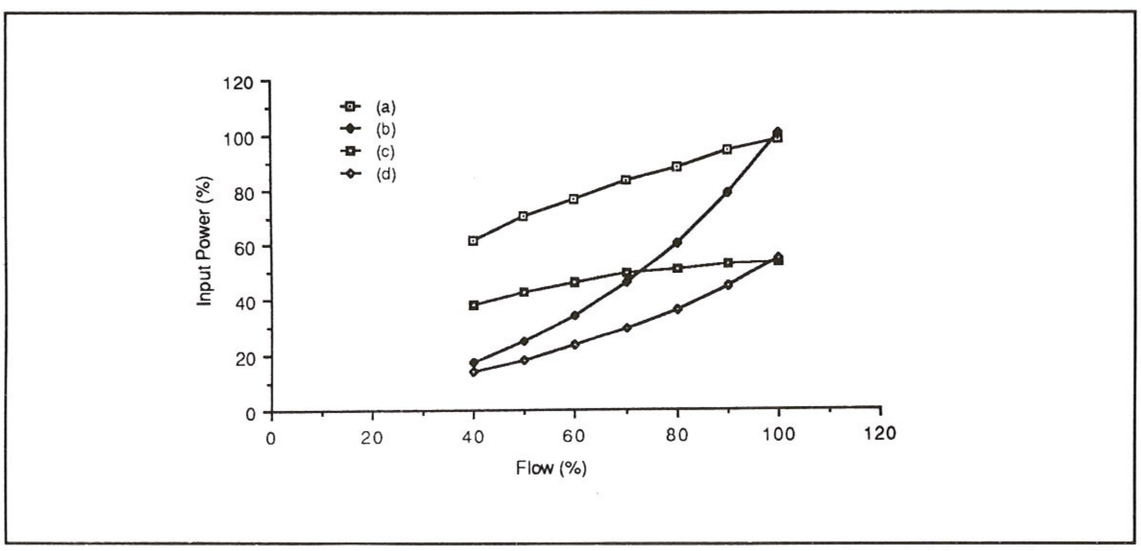


Figure 5. Calculated input power requirements for reduced-flow operation of a variable-speed variable-flow pumping system. Curve (a) represents throttle control (corresponding to the curve marked "Throttled System Curve" in Fig. 2). Curve (b) represents the same system, but with ASD control. Curves (c) and (d) are analogous to (a) and (b), but for a system with a 25% larger pipe diameter (note that a smaller pump and motor are required here). The 100% power input and flow levels correspond to 75 kW and 108 l/s, respectively. A constant 90% motor efficiency is assumed in all cases. Pump efficiency with throttling is as given by the 100% speed curve in Fig. 2. With ASD control, it is calculated as in endnote 8. In (b) and (d), the ASD efficiency is assumed to be 97% at 100% flow and to fall to 91% at 40% flow.

combined effects of outlet pressure rise and pump efficiency decrease.[10] Input power falls much further with ASD operation, because both flow and pump pressure fall.

The lower curves in Fig. 5 show results of a similar calculation, except it has been assumed that the pipe diameter is increased by 25%. The required input power falls dramatically at 100% load compared to the case with the smaller diameter pipe (as noted earlier). In this case, the percentage saved with the ASD is not as large, however, because the valve losses are a weaker function of flow than in the case with smaller piping (as is implied by the flatter system curve shown in Fig. 2).

Figure 5 shows electricity savings at several flow conditions. To calculate cumulative annual savings, the duty cycle for the particular application is needed. We use Fig. 4 to represent a typical variable-flow application. With the power input known as a function of flow from Fig. 5, the annual power demand can be calculated as the sum of the products of input power level and the hours operated at that level:

$$KWH/YR = HRS_1*KW_1 + HRS_2*KW_2 + HRS_3*KW_3 + \ldots$$

where the subscripts refer to different power levels. For the duty cycle of Fig. 4, the use of ASD control instead of throttle control would result in annual electricity savings of 41% for the system with characteristics as shown by curves (a) and (b) in Fig. 5 and 37% for curves (c) and (d).

The savings from applying ASDs in CFVP and VFCP systems would be comparable to those for the VFVP-system. For a more detailed discussion of all three systems, see [13].

Costs

Costs for efficiency improvements discussed above can vary widely depending on the specific requirements of the application. In general, unit costs for equipment ($/kW) are relatively scale sensitive. Estimates for standard-efficiency motors, energy-efficient motors, and electric adjustable speed drives are given in Component Menu entries [5,6]. These are used in the economic analysis below. As noted earlier and discussed in [4], there appears to be little relationship between pump efficiencies and list prices, although this may simply be an artifact of the way pumps are marketed. Estimated costs for piping as a function of diameter and pipe material are given in [7].

4. ILLUSTRATIVE ECONOMIC ANALYSIS

Large energy savings can be achieved through the use of more efficient pumping systems, as discussed above, but the efficiency improvements typically involve a higher first cost. We explore here the costs-effectiveness of the various system modifications discussed above, considering perspectives of an industrial user and society at large.[11] The source of cost estimates for the investments were discussed in the previous section. Only capital and operating-electricity costs are considered. Other less direct potential benefits (e.g better process control, quieter work environment, and lower maintenance costs through use of an ASD) are not considered.

Industrial Perspective

Real (corrected-for-inflation) internal rates of return (IRR) are shown in Fig. 6 for investments to improve the efficiency of the pumping system described by Fig. 5 (curve a), with the duty cycle of Fig. 4. No IRR estimate is given for an investment in a more efficient pump, because it may be no more costly than a less-efficient model, as discussed earlier. The investment in an energy efficient motor would provide a very high return (> 60%) even with

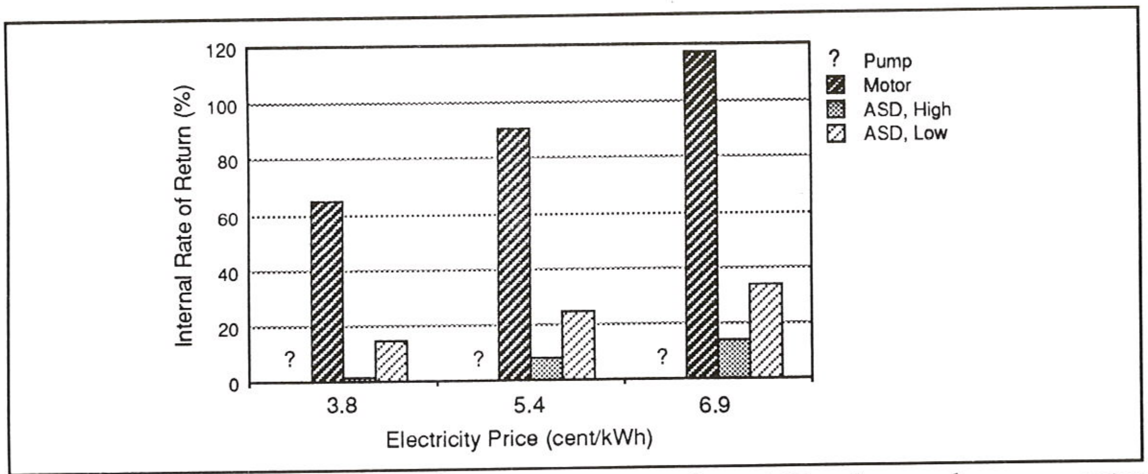


Figure 6. Calculated internal rates of return for investments in an ASD or improved pump or motor as a function of electricity price for the 75-kW system described by curves (a) and (b) in Fig. 5 and the duty cycle of Fig. 4. Based on [4] and [5], the efficiencies of the pump and motor are assumed to be improved by 3 and 2.5 percentage points, respectively. The investment in each case is the extra capital cost associated with the conservation investment, e.g. the extra cost of an energy-efficient motor compared to a standard-efficiency model. A 10-year investment life is assumed.

an electricity price as low as 3.8 cents/kWh. The two IRRs shown for ASD investments correspond to high and low estimates of their cost.[12] The IRR would be attractive (> 15%) for the lower estimated ASD cost at all electricity prices. To achieve a reasonable return (> 10%) with the higher ASD cost would require an electricity price of 5.4 cents/kWh or higher.[13]

National Perspective

Estimates of the cost-of-saved-electricity (CSE) associated with the various investments described in the previous section to improve pumping system efficiency can be compared with the cost of new electricity supply to assess their relative economic attractiveness from a national perspective.

Figure 7 shows calculated CSEs for separate investments in an improved pump, motor, or ASD versus the

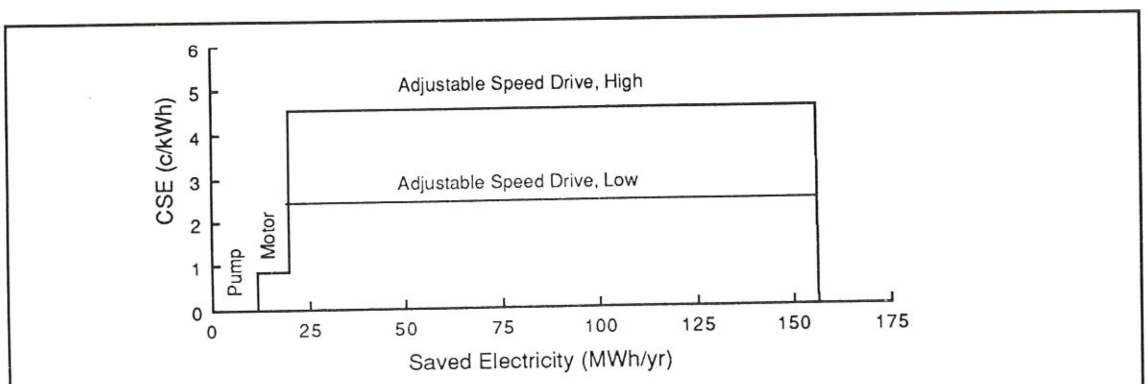


Figure 7. Calculated cost-of-saved-electricity for investments in an ASD or improved pump or motor as a function of MWh saved per year for the 75-kW system described by curves (a) and (b) in Fig. 5 and the duty cycle of Fig. 4. Based on [4] and [5], the efficiencies of the pump and motor are assumed to be improved by 3 and 2.5 percentage points, respectively. The cost in each case is the extra capital cost associated with the conservation investment, e.g. the extra cost of an energy-efficient motor compared to a standard-efficiency model. A 6% discount rate and 10-year investment life are assumed.

amount of electricity that would be saved by each investment. Also shown (two arrows) are estimated costs of new baseload electricity supply in Sweden based on coal (upper) and natural gas (lower) [15]. The two CSE estimates for the ASD correspond to high and low cost estimates (endnote 12). Zero CSE is shown for the pump investment because the efficient pump may be no more costly than a less-efficient model, as discussed earlier.

From the national perspective, investments in a new pump or new motor to save electricity lead to attractivley low CSEs. The CSEs associated with the ASD would be higher, with the upper estimate comparable to the cost of new electricity supply, but the electricity savings would be very much larger. If all three investments are considered together, the average CSE would be roughly the same as for the ASD alone, because the ASD would account for most of the cost and most of the savings.[14]

The CSE calculations suggest that there are a variety of electricity-saving investments that could be made to existing or planned pumping systems at lower cost to the nation than the construction of new power plants to supply the electricity instead. Investments in more efficient pumps or motors appear quite attractive. For variable-flow applications, investments in ASDs would be particularly attractive, because these lead to large electricity savings. In addition, though costs for increasing pipe diameter have not been made here, such modifications should not be overlooked in assessing conservation options because large electricity savings could result.

5. REFERENCES

1. Assistant Secretary for Conservation and Solar Energy, *Classification and Evaluation of Electric Motors and Pumps*, DOE/CS-0147, US Department of Energy, Washington, DC, 1980.

2. Statens Offentliga Utredningar, *Electricity Conservation in the 1990s*, 87:68, Stockholm, 1987. (in Swedish)

3. T.B. Johansson and P. Steen, *Perspectives on Energy*, Liber Forlag, Stockholm, 1985. (in Swedish)

4. L.J. Nilsson and E.D. Larson, "Pumps," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

5. L.J. Nilsson and E.D. Larson, "Motors," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

6. L.J. Nilsson and E.D. Larson, "Adjustable Speed Drives," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

7. N.P. Cheremisinoff, "Piping," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund Univ., Lund, Sweden, 1989.

8. J.S. Norgaard, J. Holck, and K. Mehlsen, "Long-RangeTechnicalElectricity-Conservation Possibilities," Fysisk Laboratorium III, Danmarks Tekniske Hojskole, Lyngby, Denmark, Dec. 1983. (in Danish)

9. B-A. Gustafsson, "Pump System Performance Illustration (PSPI)," computer software, Chalmers Institute of Technology, Gothenburg, Sweden.

10. S.F. Baldwin, "Energy-Efficient Electric Motor-Drive Systems," in *Electricity*, Johansson, Bodlund, and Williams (eds), Lund Univ. Press, Lund, Sweden, 1989, pp. 21-58.

11. *Energy User News*, July 11, 1988.

12. S.F. Baldwin and E. Finlay, "Energy-Efficient Electric Motor Drive Systems: A Field Study of the Jamaican Sugar Industry," PU/CEES Working Paper No. 94, Center for Energy and Environmental Studies, Princeton University, Princeton, New Jersey, USA, Feb. 1988.

13. J.K. Armintor and D.P. Connors, "Pumping Applications in the Petroleum and Chemical Industries," *IEEE Transactions on Industry Applications*, 1A-23(1), 1987, pp. 37-48.

14. Power Electronics Applications Center, *Adjustable Speed Drive Directory*, (2nd ed), Electric Power Research Institute, Palo Alto, California, 1987.

15. "Economic Methods and Assumptions," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, University of Lund, Lund, Sweden, 1989.

NOTES

1. Electricity-intensive industries include pulp and paper (SNI 34111+34112), iron and steel (SNI 37101), mining (SNI 2), non-ferrous metals (SNI 3720-37204), ferro-alloys (SNI 37102), and basic chemicals (SNI 351).

2. Thus, for example if the required flow is half the full fixed-speed output, the power needed would theoretically be only 13% (= 0.5^3 * 100) if the speed were adjusted.

3. While such estimates are often made, relatively little documentation exists on measured savings. To improve the understanding of the potential for conservation in pumping systems, greater efforts are required to document actual savings.

4. With relatively flat (horizontal) system curves, accurate control with electronic variable speed drives can be difficult, since small speed changes result in large changes in flow.

5. For example, piping pressure loss is approximately proportional to the inverse of pipe diameter raised to the 4th or 5th power, assuming turbulent flow. A 25% increase in pipe diameter, therefore, would reduce the pressure drop in a pipe by about 60%.

6. For example, based on a comparison of Moody friction factors [7], the pressure drop through a 3-cm diameter cast-iron pipe would be 70-75% greater than through a 3-cm diameter pipe made of commercial steel and over 200% greater than through a 3-cm diameter pipe made of drawn tubing.

7. The new pump speed can be determined from affinity equations relating flow (Q) and head (H) to speed (N) for systems operating without static lift [14]:

$$Q_1/Q_2 = N_1/N_2$$
$$H_1/H_2 = (N_1/N_2)^2$$

To determine the pump speed for systems with static lift, a parabolic curve is constructed connecting the origin and the new operating point. The point at which the curve intersects the 100% pump-speed curve defines H_1 and Q_1 to be used in the above equations.

8. Lines of constant pump efficiency at reduced pump-speed follow, approximately, parabolic curves through the origin [13]. Thus, e.g., at 80% flow in Fig. 2, pump efficiency would be about 79% at reduced (84%) speed versus 77% at 100% speed.

9. Motor efficiency falls only slightly with reduced flow. ASD efficiency falls somewhat more. The combined motor/drive efficiency (with a standard-efficiency) motor for the application considered here (75 kW size) would fall from about 88% at 100% flow to 80% at 40% flow.

10. Power input to the pump (in kW) is calculated as the product of pump outlet pressure (kPa) and flow (m^3/s) divided by pump efficiency.

11. See [15] for definitions of the economic indicators discussed here.

12. The two ASD costs are from [6] for a 75 kW unit. They include the average ASD equipment cost plus a high and a low estimated installation cost.

13. The analysis in Fig. 6 assumes that the pipe diameter and material have been chosen so as to minimize lifecycle costs. See [7] for discussion of this procedure.

14. The CSE of one investment would generally be higher if it is made after making one of the others, because the scope for saving would already have been reduced. For example, if the ASD investment is made first, the electricity that could be saved annually by investing in a more efficient motor would decrease by about 40%, implying a 2/3 higher CSE. However, such investment-ordering considerations would not change the basic conclusions given here.

Vol. 1. Movement of Material **89-8-8**

***System*: Air Handling**

by
J. Barrie Graham
Graham Consultants
Santa Fe, New Mexico

CONTENTS

SUMMARY

Air-handling systems are major energy users in the industrial and commercial sectors. For example, energy audits in the USA indicate they can account for 30 to 50% of electricity use in typical commercial buildings. This Menu entry describes energy-related issues involved in the design and operation of air-handling systems, including alternative flow control methods for air-handling applications where the required volume flow varies with time. In such cases large energy savings can often be achieved through use of an appropriate control strategy. A case study for an existing 40,000 m^2 office building is summarized to illustrate this. Energy auditing and computerized simulation of the building indicated that air-handling electricity use could be reduced by 60 to 75% by improving lighting efficiency and converting the existing air-handling systems from constant-air-volume (CAV) to variable-air-volume (VAV) operation (below). The level of savings would depend on the flow control strategy chosen. Estimated rates of return for the modifications are shown below for three electricity prices. Also shown are the estimated costs of saved electricity, which are lower than the cost of new electricity supply in many regions of the world.

Electricity use in a 40,000 m^2 building.

(MWh/yr)	CAV, Existing Building	----- VAV with ----- Variable Inlet Vane	Variable Freq. Drive
Air Handling	1507	603	378
Lighting	3546	2482	2482
Other	2001	1483	1483
TOTAL	7054	4568	4343

Economic results for 40,000 m^2 building.

	Electricity Price (cents/kWh) 4.6	6.2	7.7
IRR (%/yr)			
Lighting only	17.2	26.0	34.1
Air-Handling			
VIV case	15.1	23.4	31.1
VFD case	13.6	21.6	29.0
Cost of Saved Electricity (cents/kWh)			
Lighting only	2.9		
Air-Handling			
VIV case	3.1		
VFD case	3.3		

1. TECHNOLOGY

Introduction

Air handling systems are major electricity consumers in the industrial, commercial and residential sectors. Measurements in a variety of commercial and residential buildings in the USA indicate that air handling systems can easily account for 30 to 50% of total building electricity use (Fig. 1). The useful work done by the electricity input to an air handling system consists entirely of overcoming mechanical and aerodynamic resistances within the fan, motor, drive, and duct work. As shown in Fig. 2, in a typical system, the majority of input electricity is attributed to the fan--about 20%--and the ductwork--about 60% [1]. Menu entries on fans [2], ducting [3], and motors [4] discuss energy efficiency aspects of these components of air handling systems.

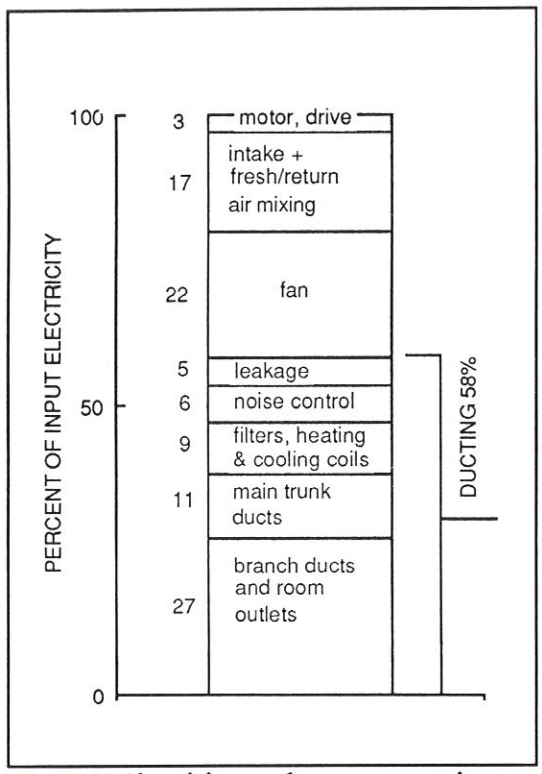


Figure 2. Electricity use by component in an air handling system. This estimate is for an air conditioning system for a typical multi-level commercial office building in the USA.

This document discusses the energy performance, operation, and costs of complete air handling systems. This document does not address in detail how the changes in the end-use of delivered air can affect air-handling energy consumption. Examples of such changes include installing more efficient lights or using special coatings on windows to reduce solar heat gain.

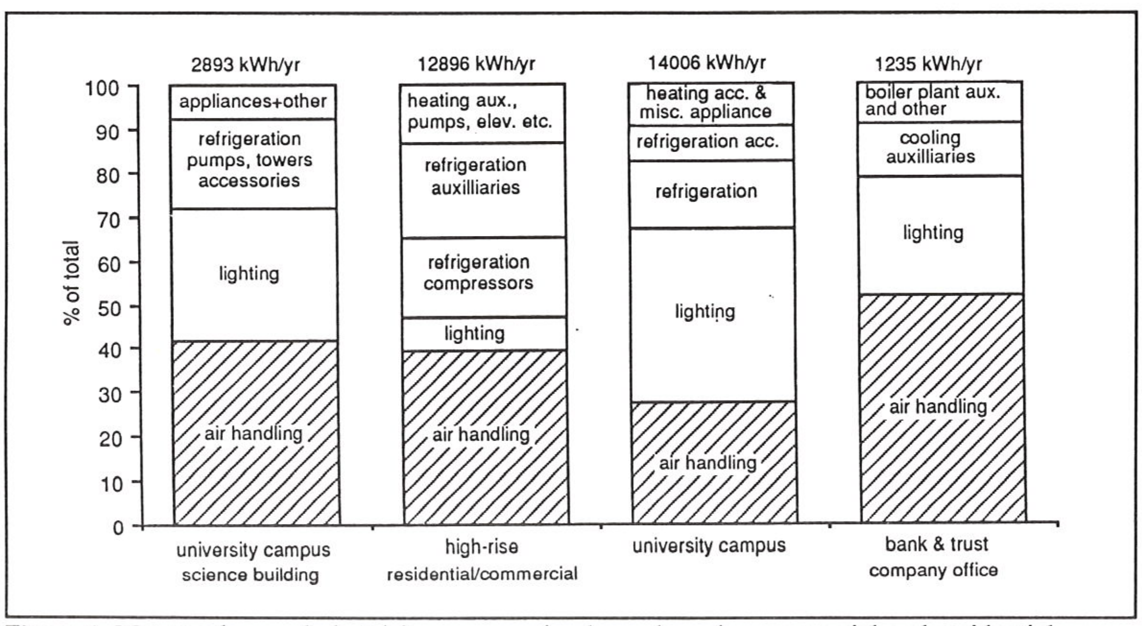


Figure 1. Measured annual electricity consumption by end-use in commercial and residential buildings in the Midwestern USA (St. Louis region) [1].

Such changes will often be cost-effective by themselves and will typically also lead to reduced air-handling requirements, as discussed briefly in Section 4. Efficiency improvements to the air handling system itself would lead to still greater energy savings.

Fan and System Performance Curves

For most efficient operation of an air handling system, it is necessary to select a well designed fan, a well designed system, and to properly match the fan to the system. System and fan performance curves are used for these purposes.

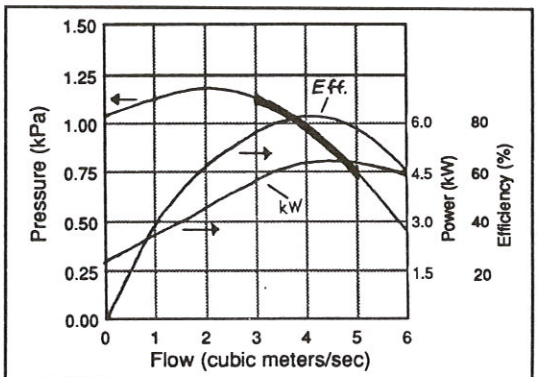


Figure 3. Typical fan performance curve showing optimal operating range (heavy line).

Fan Curves: As discussed in [2], fan curves show the relationship between the flow rate, outlet pressure, efficiency, and power requirements for a particular fan. The fan will operate at highest efficiency over a given range of volume flow and corresponding delivery pressure. For most efficient operation, fans should be selected to operate in this peak efficiency range (Fig. 3). Once a selection range is chosen, the fan laws can be used to calculate the fan characteristics at other speeds [2]. This leads to an "optimum" selection zone within which the efficiency would be within the initially chosen range (Fig. 4).[1] For an operating point that would fall into this optimum selection zone, the size of the fan is known and it is simply a matter of determining the proper speed. If the required operating point falls in the zone marked A in Fig. 4, a larger fan would be more efficient. If the required point falls in Zone B, a smaller fan would be more efficient. To determine the operating point requires determination of the system characteristic curve.

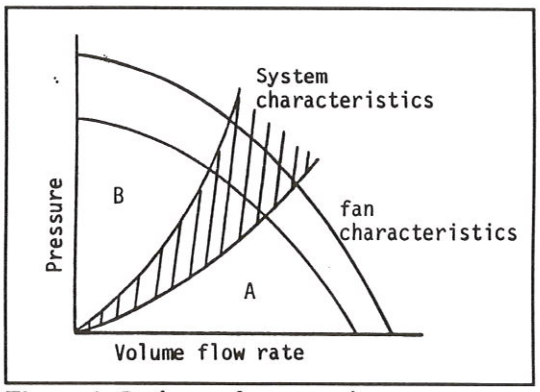


Figure 4. Optimum fan operating zone (shaded region) for different fan speeds.

System Curves: System curves are developed independently of the fan curves, and the fan has no effect on the design of the system. System curves show the pressure/volume flow relationship for the system, including the effect of ducting, dampers, and other resistive elements. In typical systems, the pressure drop through the system is proportional to the square of the system flow rate.[2]

Duct system design is a complex task [3], but the concept of system curves can be illustrated by considering a simple orifice plate to represent the sum of all pressure drops in a particular system (Fig. 5a) [5,6]. As the pressure at plane 1 is reduced, the air flow through the orifice will also be reduced. If a series of pressure/flow measurements are made and plotted, a curve for this system will result (Fig. 5b, curve C). As long as the system elements remain unchanged, i.e. the orifice size is not changed, this curve uniquely characterizes the system's operation. If the system is changed, for example if the orifice is enlarged, a new curve will result (Fig. 5b, curve D).

Matching Fan and System: The duct design procedure should result in optimally sized ducting connecting the fan and the point of final air delivery [1,3,7,8,9]. The air flow conditions required at the termination of the duct work establish the fan flow rate and pressure requirements. As discussed above, since the fan and the system must each operate at one particular point on their respective characteristic curves, when the two operate together, they can only operate at the intersection of the two characteristic lines (Fig. 6). Once the fan is connected to the system, changes in either the fan (e.g. different speed) or the system (e.g., changed outlet damper position) will lead to a new operating point with the fan producing more or less flow than originally specified.[3]

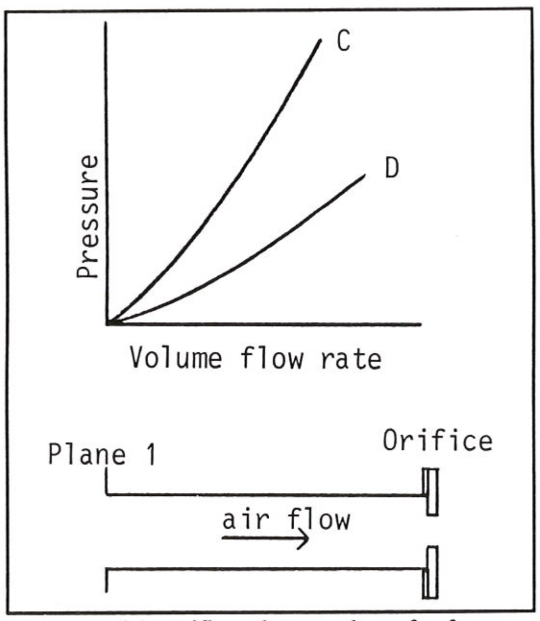


Figure 5. (a) Orifice-plate analog of a fan system. (b) System curves corresponding to two different orifice sizes.

Dynamics of Fan and System Curves: Fan and system curves are relatively simple conceptually, but their application requires attention to details of actual installations. Some such considerations are discussed here.

In some systems, some resistances may vary with time, e.g., via a variable damper setting. If such changes occur routinely during system operation, this must be taken into account when selecting the fan. The range of operation should be studied to determine whether of not the fan can operate with stability over that range.

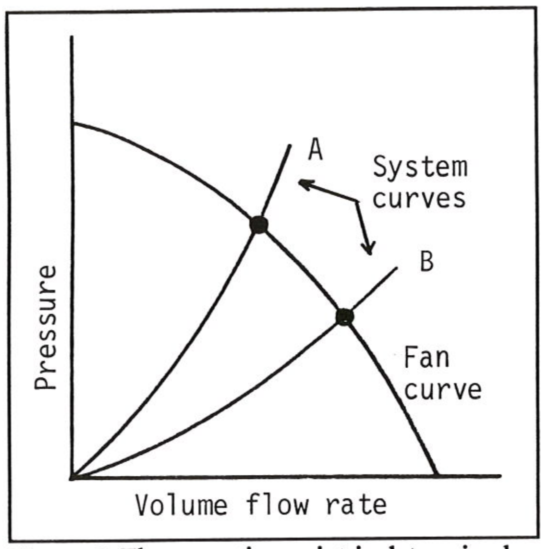


Figure 6. The operating point is determined by the intersection of the fan and system curves. Curves for two different systems will intersect the same fan curve at different points.

A fan selected for a particular application may not be installed in the same configuration in which its rating curves were developed [10]. This may be the case, for example, where different fan inlet and/or outlet configurations are used. In such cases, it is usually assumed that the fan is performing exactly as rated, but that the configuration of the installation imposes a pressure drop in addition to that calculated for the system. The use of System Effect Factors provides an additional calculation procedure to correct for detrimental inlet and outlet conditions [6,7,9]. These corrections are added to the normal pressure loss values when determining the flow rate and pressure requirements for the fan. This allows the designer or analyst to work with the published fan curves and incorporate all other effects in the system curve.

Published fan curves must be used with care in conducting tests to check the performance of a fan in the field. Field measurement of the flow rate and total pressure in the installed system are required to assess performance. The measurements can be plotted on the fan curve to determine the actual operating point. The system effect factors mentioned above are useful in analyzing difficult field tests [11,12].

If a fan and system combination do not seem to be operating at the expected point, it is typically due to one of two types of problems.

If the system is not the same system as specified in the design, the point of operation will not be at the design point of rating on the fan curve. For example, Point B in Fig. 6 may be the specified operating point, but tests may show that the actual operating point is A. If it is important to regain the original design flow rate, the situation may be corrected by (a) changing the speed of the fan until the new fan curve and the system curve pass through the required capacity point, or (b) reducing the system pressure loss (e.g. by a changed damper or outlet grill setting) to move the operating point to B, which would leave the fan curve unchanged.

Another factor that can change the expected operating point is that the fan performance curve may be different from the manufacturers rated curve [10,11,12]. Fans require a uniform and straight inlet flow in order to perform according to their rated curve. Any swirl or non-uniformity at the inlet will produce a new performance curve. The departure from the standard curve depends on the amount of upset of the inlet flow. Such problems can be avoided with proper duct design.

Flow Control

Among various factors influencing the energy efficiency of an air moving system, how the flow is controlled can be very important. Since in theory fan power requirements vary as the cube of the flow rate, flow reductions can mean significant savings in energy. The flow control needed in a system can vary from a minor permanent adjustment following installation (to achieve a fixed design flow) to control of continuously changing flow requirements. Continuously variable flow is often required in large industrial applications, so efficient control can lead to substantial energy savings. Five methods of flow control are discussed here.

Outlet Damper: Outlet dampers are often used for flow control. They are a variable flow resistance in the system, so changing the damper setting simply changes the system curve. The performance of a typical fan system with an outlet damper is shown in Fig. 7a. Point A represents the wide-open damper position. As the damper is successively closed to reduce the volume flow, the system curve crosses the fan curve at different points, e.g., B, C, and D in Fig. 7a. Point E is a fully-closed damper, at which point there is still some flow due to leakage. Since the fan operates at a constant speed independent of the damper position, the change in input power required follows the original fan characteristic (e.g. from C to D in Fig. 7b). Since the fan efficiency also moves rapidly away from the optimum (Fig. 3), outlet dampers are the least efficient method for controlling flow. They are best used only as a shut-off device.

Inlet Box Dampers: Fig. 8a shows the performance of a typical centrifugal fan with inlet box dampers. For well designed dampers the wide open position is essentially the same as the operation without dampers. The change in the fan performance curve for partially closed dampers is due partly to the added flow resistance and partly to spin imparted to the air. Unlike the case with outlet dampers, the power requirement does not follow the original fan characteristic. For a properly installed inlet damper, there will be a slight reduction in power input (with reduced flow) relative to the undampered fan characteristic curve (Fig. 8b).[4] The inlet box damper is a slightly more efficient way to control flow than outlet box dampers, since fan

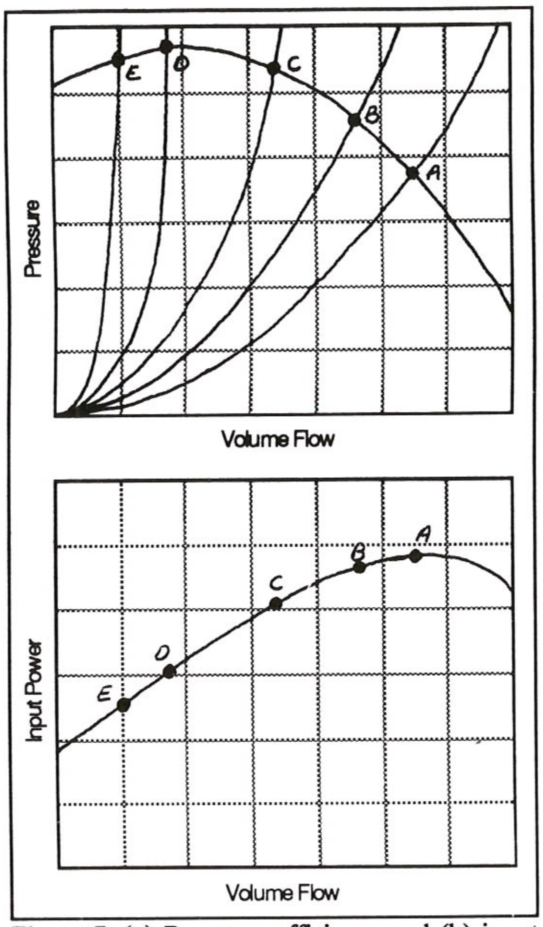


Figure 7. (a) Pressure, efficiency and (b) input power characteristics for flow control with outlet dampers.

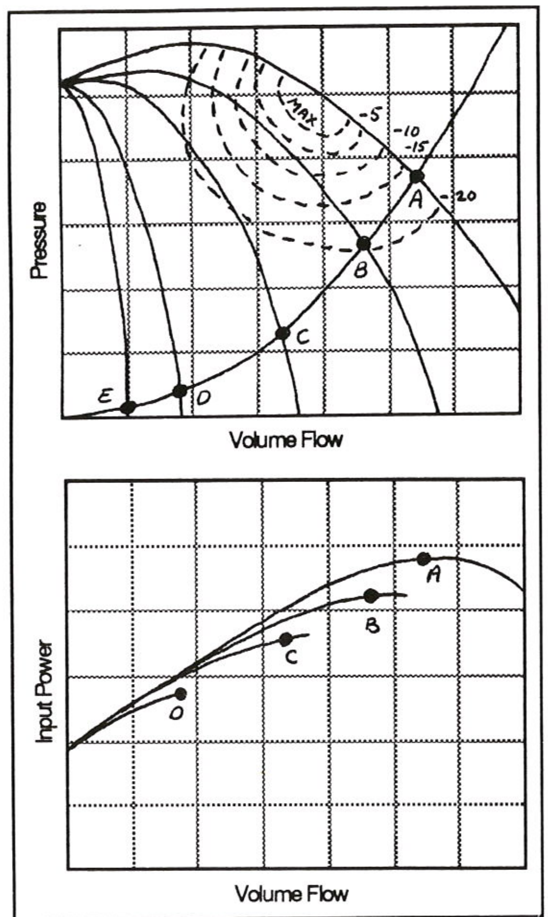


Figure 8. (a) Pressure, efficiency and (b) input power characteristics for flow control with inlet box dampers.

efficiency moves more slowly away from the optimum (Fig. 8a).

Variable Inlet Vanes: Variable inlet vanes operate using the same principles as inlet box dampers, but impart a more uniform and stable spin to the incoming air and, therefore, control flow more efficiently. Figure 9a shows the new fan curves for different inlet vane positions. For the same flow reduction, the reduction in input power requirement is greater than with inlet box dampers (Fig. 9b). The efficiency degradation with reduced flow is also lower (Fig. 9a).

Speed Change: Variable speed control is conceptually the ideal method of achieving variable flow. Unlike the case with dampers, where the fan runs at a fixed speed, variable speed control takes full advantage of the cubic drop in power requirements that accompanies a linear drop in flow (e.g., a 50% reduction in flow leads to an 88% drop in power required). The pressure and power characteristics of variable speed control are shown in Fig. 10a and Fig. 10b, respectively. Fan efficiency is constant with changing speed, making this the most efficient of the control methods discussed here.

When permanent speed changes are required, e.g., for after-installation adjustment in a constant-flow HVAC application, speed change is usually accomplished by changing pulley sizes. In large variable-flow HVAC installations where energy conservation is a design goal and in industrial applications where variable process control is essential and energy conservation is important, variable speed motor drives are used. Rapid

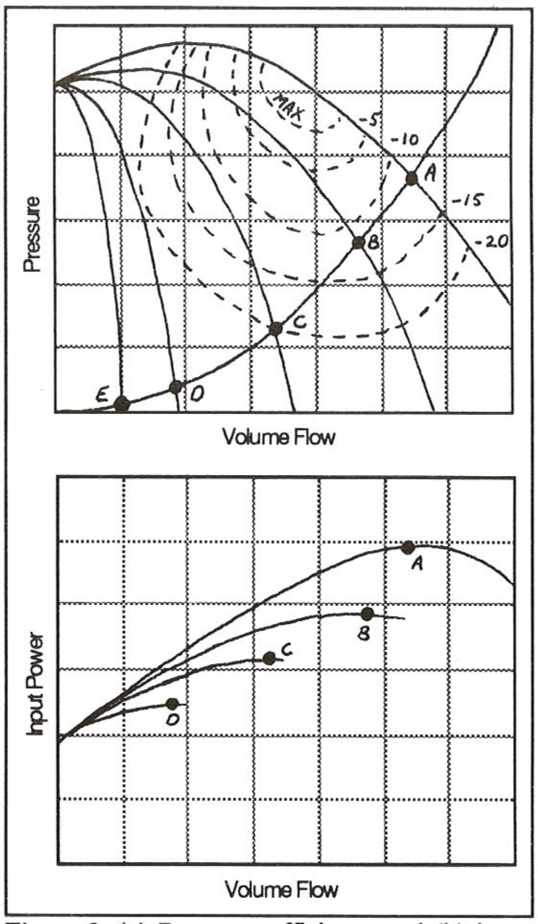


Figure 9. (a) Pressure, efficiency and (b) input power characteristics for flow control with variable inlet vanes.

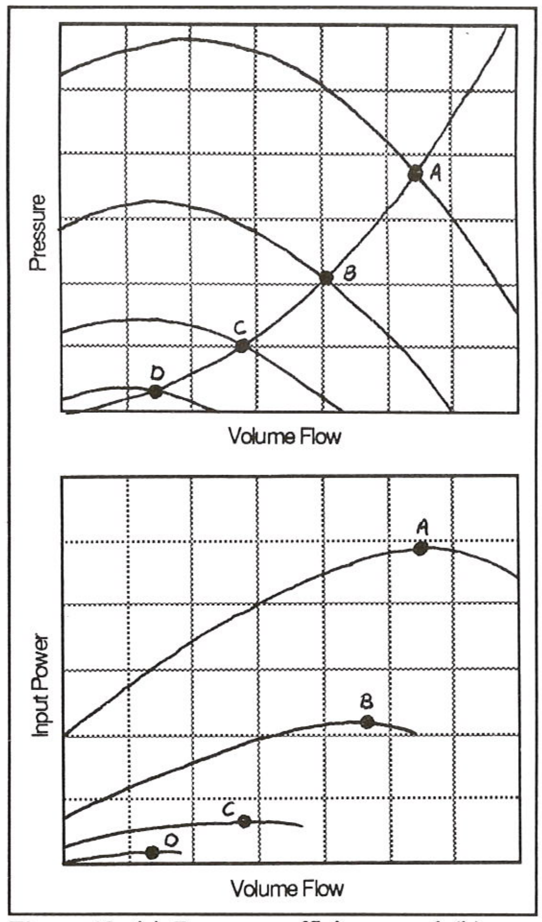


Figure 10. (a) Pressure, efficiency and (b) input power characteristics for flow control with fan speed change.

advances in solid-state electronics are making electronic adjustable speed drives (ASDs) increasingly attractive for such applications [13].

Adjustable and Controllable Pitch Vaneaxial Fans: With axial flow fans another possibility for flow control is changing the angle at which the blade is set on the hub. The fan performance can thereby be changed over a wide range of flows and pressures, as illustrated in Fig. 11a. Each fan curve describes the performance at a different blade pitch. The corresponding power curves are shown in Fig. 11b. The reduction in power is comparable to that obtained by changing fan speed, but fan efficiency will fall by 5-8 percentage points from the optimum over the control range. Thus, this control method is slightly less efficient than changing speed.

With adjustable pitch fans the blade angle is changed manually after stopping the fan. With the controllable pitch fan the blade pitch angle is changed while the fan is operating by a mechanism built into the hub and actuated by control signals from external sensing elements. Controllable-pitch fans cost 4-5 times as much as adjustable-pitch fans and have up to eight times the maintenance cost. The adjustable pitch fan is used in those cases where there is a need for occasional blade adjustment (e.g., in fine tuning the flow in a newly installed system). The controllable pitch fan is used where frequent or continuous adjustment is required.

Summary of Control Techniques: Table 1 provides a simplified summary of the relative costs and efficiencies of various flow control techniques. Speed

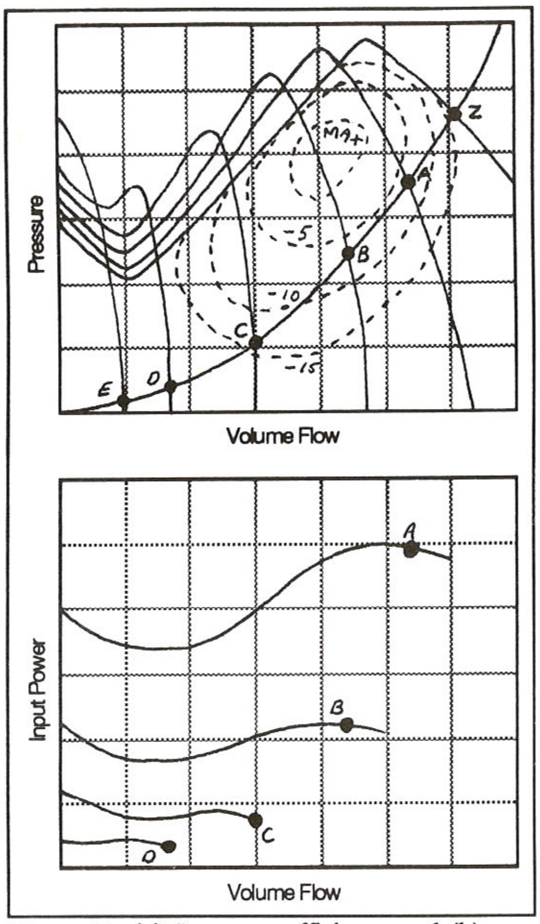


Figure 11. **(a) Pressure, efficiency and (b)** input power characteristics for flow control with adjustable- or controllable-pitch vaneaxial fans.

control is clearly favored on an efficiency basis, and outlet dampers are favored on a first-cost basis. Lifecycle cost analysis is necessary to identify the economically most attractive option for each application.

Energy Efficient Air-Handling Systems

Proper selection and design of the component parts for a particular air handling system are important in insuring that the system is efficient. The key components are typically the ducting, the fan, the motor, and the flow control device.

Duct design is based on the quantity of air required, the space available for duct work, the tolerable noise level, and a variety of constraints which depend on the duct design method chosen [3,7]. Computerized optimization techniques can be used to help identify low-energy duct designs that can save as much as 60% of the energy required in a conventional non-optimized system (see Section 4 below, and [3]).

Table 1. Summary of relative costs and efficiencies of flow control methods.

Method	Efficiency Control	First Cost	Maintenance Cost
Outlet Damper	none	low	very low
Inlet Damper	low	modest	modest
V. Inlet Vanes	some	modest	modest
Pitch Control[a]	high	high	high
Speed Control[b]	highest	high	high

(a) Available only with vaneaxial fans. As noted in the text, manually adjusted blade control would cost 1/4 to 1/5 of automatic control and have up to eight times lower maintenance costs.

(b) Speed control can be used with any fan design. Continous electronic variable speed control would have higher first costs than speed control achieved by changes in pulley sizes.

A variety of fan designs having a wide range of efficiencies are available for any particular application. For example, a backwardly-curved blade fan typically has a peak efficiency in the range of 79-83%, while a forwardly-curved blade fan, which could be used in many of the same applications, typically has an efficiency in the range of 60-65% [2]. While there would usually be a higher first cost for the more efficient fan, a careful lifecycle cost analysis would be warranted before selecting the cheaper fan, since significant operating savings would accompany the more efficient fan.

For a given fan application any of a number of motors can be used with different implications for system energy efficiency and overall economy [4]. The predominant motor type in larger industrial fan systems is the polyphase induction motor. "Standard" polyphase motors have efficiencies of 70-85% in smaller sizes (<10 kW) up to 90-95% in larger sizes (>150 kW). Motors loosely defined as energy-efficient motors (EEMs) are available with 2-10 percentage points higher efficiencies, with the larger improvement coming in the smaller sizes. Some vendors offer both standard motors and EEMs, with the latter typically having a higher first cost and comparable reliability as the standard models. Since motors can easily use up to five times their cost worth of electricity per year, the higher first cost of the EEM will often be quickly recovered in electricity cost savings [4].

As discussed earlier, flow control can lead to large energy savings in variable-flow applications. The most efficient method, electronic adjustable speed motor drive, is becoming increasingly popular, particularly in large variable-flow applications, where the economics are often attractive [13].

2. R&D

R&D activities to improve fan systems are directed primarily at improving the efficiencies and/or reducing the cost of specific system components. Particularly noteworthy is work on advanced energy efficient motors [4], the trend toward capital cost reductions in electronic ASDs [13], and the continued development of computerized methods for designing low pressure-drop ducting [3].

In addition to improved system components, there are several design and operating principles which should be standard practice (but often are not) to help reduce electricity use in fan systems:

* A careful assessment of the useful service to be provided by the air handling system (e.g., maintain air quality in a room) should be undertaken to identify the lowest design flow rates and pressures possible. Excessive oversizing (e.g., inappropriately large safety factors) should be avoided as this wastes power and money every minute of operation.

* A newly installed system should be tested and adjusted to the desired operating condition by the most energy efficient adjustments possible. A test will also indicate whether design conditions are infact being achieved.

* Equipment should be maintained in good operating condition.

3. TECHNICAL AND COST DATA

Specific performance and cost data are provided elsewhere in the Technology Menu for components of air handling systems: fans [2], ducting [3], motors [4], and adjustable speed drives [13].

4. ILLUSTRATIVE ECONOMIC ANALYSIS

Retrofits

The design and operation of air-handling systems are closely tied to the design of the building they service. An analysis of the potential energy savings in air-handling equipment thus requires a thorough understanding of the energy characteristics of the building. Efficiency-improving retrofits to an existing air-handling system will typically involve changes both to the air-handling system and to other areas of the building that alter the building's energy characteristics. The following analysis illustrates this for a 40,000 m^2, 26-level commercial office building located on the West Coast of the USA.

The building is currently serviced by four constant-air-volume (CAV) systems with a total installed fan power of 513 kW. An energy audit was undertaken to evaluate the feasibility of converting the system to variable-air-volume (VAV) operation and to assess the potential cost and electricity savings that would result [14]. The basic principle behind a CAV system is that the volume of air supplied by the fans is kept constant, and its temperature is changed to meet changing demands by mixing hot and cold air streams [3]. In a VAV system, the volume of air is varied, according to the building demands, using one of the control methods discussed in section 1.

The four CAV air-handling systems consist of an induction system, two dual duct systems, and a multi-zone system. An understanding of the basic operation of each type of system is required to understand how the conversion to a VAV system could be achieved.

Induction systems, which have been commonly used in high-rise office buildings in the USA since the 1960s, are used to offset heat losses or gains through the skin of a building. A constant volume of air passes over a primary cooling coil and is then split so that part of the flow passes over a heating coil and the balance bypasses the coil. The relative split between heated and bypass air is controled by one set of two dampers for each of several zones of the building. After each set of dampers, the flows are mixed and delivered to terminal boxes located under each window of the building. This "primary air" is delivered with sufficient pressure and orientation such that it induces room air ("secondary air) to pass over a cooling coil located in the terminal box.

In a dual duct system, the fan supplies fixed constant volumes of air to heating and cooling coils located in separate ducts (Fig. 12). (In some systems, there may be some conditioning of the air on the upstream side of the fan.) The two flows remain separated until they reach the terminal boxes, where they mix, with the relative fraction of hot and cold air adjusted by dampers set according to conditioning requirements in the room.

A multi-zone system is similar to a dual duct system, except that the heated and cooled air flows are delivered to a set of mixing boxes, each of which is connected to a single duct delivering air to one zone of the building. The relative mix of hot and cold air entering the mixing boxes is controlled by a set of 2 dampers on each box.

The energy audit of the building indicated that a CAV to VAV conversion was feasible for each system. For example, it was determined that for much of the building, the heating requirement could be met entirely with some minor modifications to the induction system, so that the heating duty of the multi-zone and one of the dual duct systems would not be needed. Thus, one modification would be to shut off the heating side of the multi-zone and dual duct systems and convert them to cooling-only VAV operation. This would basically involve closing off the heating dampers and adding variable flow control capability to the

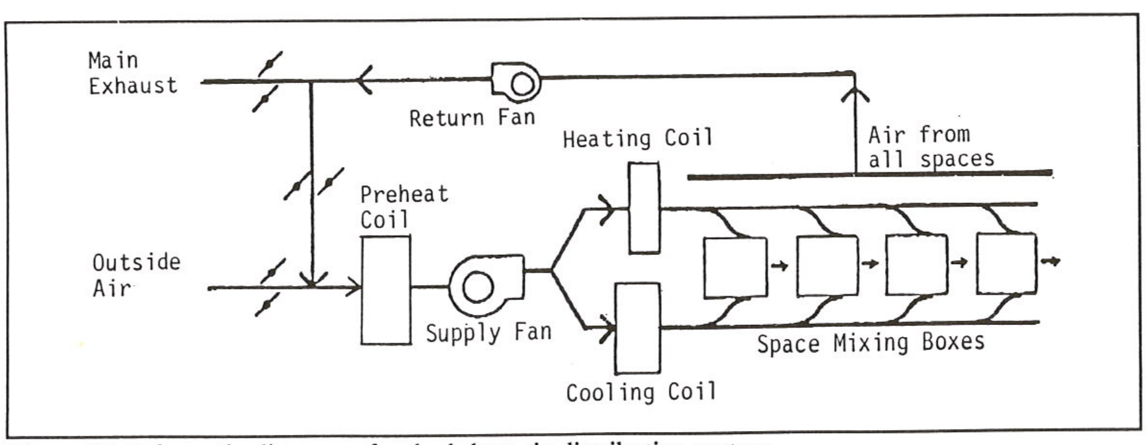


Figure 12. Schematic diagram of a dual duct air distribution system.

fans. One possible control strategy would be with variable inlet vanes (VIVs): most of the fans in the system were originally installed with inlet vanes which were fixed in place after fine-tuning the flows in the new system. The VIV modification would involve adding automatic control to these inlet vanes. A more efficient control strategy would be to use electronic variable frequency drives (VFDs) to control fan speed.

The energy audit also indicated that significantly reduced lighting electricity use could be achieved by slightly reducing the high degree of over-lighting in the building and replacing existing ballasts and flourescent lamps with more energy efficient models. This would also reduce the cooling load on the air-handling system since less waste heat would be generated.

To determine in detail the potential electricity savings by the lighting and CAV-to-VAV modifications, a building simulation was undertaken. The electricity use in the existing building was first modeled using the TRACE simulation program developed by the Trane Co. (La Crosse, Wisconsin, USA). Comparisons with the audit measurements verified the accuracy of the TRACE model of the existing building.

Table 2 summarizes the electricity use in the existing building and after assumed retrofits to the lighting and air-handling systems. Lighting electricity use would fall by some 30%. Air-handling electricity use would fall by some 60% if VIV flow control were used or by 75% if VFD flow control were used. Other electricity demands (for chillers, pumps, cooling towers, etc.) would also fall (by 25%) with the retrofits because a direct digital control system would be installed to operate essentially all equipment in the building at optimal efficiency. The costs estimated for the lighting and two sets of air-handling retrofits are given in Table 3. The costs for the VFD case

Table 2. Summary of estimated electricity use (MWh/year) for an existing 40,000 m^2 commercial office building in Southern California, USA,[a] and for two sets of proposed air-handling efficiency improvements [14].[b]

	Existing Building	**Variable Inlet Vane**[c]	**Variable Frequency Drive**[c]
Air Systems[d]	1507	603	378
1.Induction	(558)	(370)	(273)
2.Dual Duct A	(387)	(91)	(21)
3.Dual Duct B	(260)	(77)	(19)
4.Multi-zone	(350)	(65)	(65)
Lighting[e]	3546	2482	2482
Other[f]	2001	1483	1483
TOTAL	7054	4568	4343

(a) The building consists of a 26-level portion with 31,000 m^2 of floor area and an 8-level portion with 9,000 m^2 of floor area. Both portions are supplied by common chiller and boiler plants.

(b) The electricity use for the existing and modified systems was estimated using the TRACE air-handling/building simulation computer program developed by the Trane Co., La Crosse, Wisconsin, USA. Extensive procedures, including measurements within the building, verified the accuracy of the model of the existing building.

(c) See text for discussion of modifications.

(d) The existing fan sizes and power ratings are as follows. All fans are of the air-foil design, except the multi-zone fans, which are forwardly-curved:

	Supply Fans	*Return Fans*
1. Induction	68.6 cm, 30 kW 76.2 cm, 37 kW	91.4 cm, 7.5 kW common w/dd. A
2. Dual Duct A	124 cm, 75 kW 124 cm, 56 kW 61 cm, 11 kW	124 cm, 19 kW 152 cm, 30 kW none
3. Dual Duct B	102 cm, 56 kW	137 cm, 15 kW
4. Multi-zone (8 systems)	2-46 cm, 11 kW on common shaft	none

(e) In the improved-efficiency systems, 16,400 existing 40-W flourescent lamps and 8,200 conventional ballasts are replaced by 34-W lamps and energy-efficient ballasts. The result is a reduction in power drawn per 2-lamp, 1-ballast set from 94 W to 59 W. Since the building is highly overlit, the 15% lower light output from these lamps is acceptable. Electricity use by incandescent lights and office equipment (which is included under the lighting category) remains the same for each case.

(f) In the improved-efficiency cases, a microprocessor-based Direct Digital Control (DDC) system will control essentially all electricity-using equipment in the building. The resulting better control strategies, together with the reduced lighting load, lead to less electricity use by chillers, cooling towers, and pumps.

Table 3. Estimated costs for modifications to achieve the levels of electricity use described in Table 2 [14].

Lighting Retrofits	$ 227,000
Replace 8,200 ballasts and 16,500 lamps	
Air-Handling Retrofits, VIV-Case[a]	$ 328,677
Air-Handling Systems	
Induction System	29,000
3 VFDs (16,000)	
Install VFDs (4,000)	
Modify 8 auto. zone dampers (3,200)	
Control revisions (3,800)	
Engineering (2,000)	
Dual Duct Systems A + B	84,677
Convert 400 boxes to VAV (47,600)	
Asbestos control (24,500)	
Control of existing VIVs (9,000)	
Engineering (3,577)	
Multi-zone System	15,000
Convert dampers to VAV	
DDC System for Building[b]	200,000
Hardware, software, installation	
Air-Handling Retrofits, VFD-Case[c]	$ 403,100
Air-Handling Systems	
Induction System	29,000
Same as above	
Dual Duct Systems A + B	123,100
Convert 400 boxes to VAV (47,600)	
Asbestos control (24,500)	
7 VFDs (36,000)	
Install VFDs (9,000)	
Control revisions (4,000)	
Engineering (2,000)	
Multi-zone System	51,000
Convert dampers to VAV (15,000)	
8 VFDs (23,200)	
Install VFDs (8,000)	
Control revisions (2,400)	
Engineering (2,400)	
DDC System for Building[b]	200,000
Same as above	

(a) The main volume flow control is accomplished using automatic control of variable inlet vanes. The one exception in this case is VFD control on 3 fans in the induction system. See note c.

(b) See note f of Table 2.

(c) The sizes of the installed VFDs correspond approximately to the fan sizes given in note d of Table 2. In the improved-efficiency systems, the full capacity of some of the fans would not be needed, in which case a VFD of smaller size than the fan rating would be installed to save capital costs. For an estimated range of costs for variable speed drives, see [13]. The following equipment cost estimates were used for the VFDs:

kW	11.2	14.9	18.7	22.4	29.8	37.3	44.8	56.0	74.6
$/kW	259	201	182	161	161	153	167	155	137

are about 25% higher than for the VIV case.

Table 4 gives expected internal rates of return on the various alternative investments as a function of the assumed electricity price. The returns on the lighting investments alone are reasonably high--17 to 34% for electricity prices of 4.6 to 7.7 cents/kWh. The return on the additional investments to modify the air-handling systems and add the building control system are comparable, ranging from 15 to 31% for the VIV-based retrofits and from 14 to 29% for the VFD-based retrofits. Table 4 also gives the calculated cost-of-saved-electricity (CSE) for the lighting, VIV, and VFD investments, assuming a utility discount rate of 6%. The CSE estimated for the lighting retrofits is 2.9 cents/kWh. For the additional VIV- and VFD-based investments, the CSE would be about 3.1 cents/kWh and 3.3 cents/kWh. Thus, the CSE for all of the retrofits considered for the building would be below the estimated cost of new central station electricity

Table 4. Economic evaluation of building modifications described in Tables 2 and 3.

Internal Rate of Return (%/yr)[a]	**Elec. price (cents/kWh)**		
	4.6	**6.2**	**7.7**
Lighting only	17.2	26.0	34.1
Air-Handling[b]			
VIV case	15.1	23.4	31.1
VFD case	13.6	21.6	29.0

Cost of Saved Electricity (cents/kWh)[c]	
Lighting only	2.9
Air-Handling[b]	
VIV case	3.1
VFD case	3.3

(a) A ten-year amortization period is assumed.

(b) Calculated assuming lighting investments have already been made. Thus, the capital investments are $328,677 for the variable-inlet-vane case and $403,100 for the variable-frequency-drive case (from Table 3). The corresponding electricity savings are 1,422 MWh and 1,647 MWh, respectively (from Table 2).

(c) Assuming a discount rate of 6% and amortization period of 10 years.

Table 5. Comparison of lifecycle costs for a hypothetical ducting system designed using traditional design practice compared with designs optimized using the T-method. The effect of different electricity prices and ducting materials. The indicated cost savings are for the optimized design relative to the traditional design. See [3] for details.

Electricity price (cents/kWh)	Duct material & installed cost ($/m^2)	Lifecycle cost (10^3 $)[a] Tradition[b]	Optimal	COST SAVINGS (percent) Total	Capital	Electricity
11.88	Spiral (33)	30.9	16.2	48	-9	57
11.88	Stainless (128)	50.9	40.0	21	-5	26
8.52	Galv. steel (41)	25.7	16.3	37	-9	46
7.26	Ins. galv. (55)	26.3	19.1	27	-7	35
4.83	Aluminum (43)	18.8	14.2	25	-6	31
2.40	Spiral (33)	11.9	9.8	18	-3	20
2.03	Spiral (33)	11.1	9.4	16	-1	17
2.03	Stainless (128)	31.1	27.3	12	17	-4
1.89	Spiral (33)	10.8	9.2	15	0	15

(a) Calculations assume a 6% discount rate, 3.1% real escalation in electricity price, 10-year amortization, fan efficiency of 75% operating (85% peak), and motor efficiency of 80%.

supply (e.g., the estimated cost of electricity from new baseload natural-gas fired gas turbines in Sweden is about 4 cents/kWh [15]).

New Designs

In new designs of air-handling systems, constraints that are often present in retrofit cases would be removed. For example, in the case study discussed above, the installed ductwork could not be altered. For a new building, alternative ductwork configurations could be considered in addition to VAV operation of the system. A recently-developed duct-design optimization computer program (T-Method) can effectively identify duct designs that will minimize lifecycle costs for a particular air-handling system [3]. For example, Table 5 gives the lifecycle costs for a hypothetical ducting system designed using the T-Method compared to those designed by a traditional method. Details of this comparison are given in [3]. Table 5 shows the impact of different electricity prices and ducting materials on the capital and electricity costs of traditional and optimized designs. Electricity savings of 60% are technically achievable with an optimized design, but would only be cost-effective for relative high electricity prices. For electricity prices in the range of 4.5 to 8.5 cents/kWh, electricity savings of 30 to 50% appear to be cost-effectively achievable.

5. REFERENCES

1. W.J. Coad, G.J. Williams, P.D. Sutherlin, and J.B. Graham, "Air Systems Design and Retrofit for Energy/Cost Effectiveness," Energy Seminar Notes, available from American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, Georgia, USA.

2. J.B. Graham, "Fans," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

3. R. Tsal and H. Behls, "Ducting," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

4. L.J. Nilsson and E.D. Larson, "Electric Motors," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

5. R. Jorgensen (ed.), *Fan Engineering*, Buffalo Forge Co., Buffalo, New York, USA, 1985.

6. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, "Fans," ch. 3 of *1988 Handbook-Equipment*, Atlanta, Georgia, USA, 1988.

7. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, "Duct Design," ch. 32 of *1989 Handbook-Fundamentals*, ASHRAE, Atlanta, Georgia, USA, 1989.

8. American Conference of Governmental Industrial Hygenists, *Industrial Ventilation*, 20th ed., Cincinnati, Ohio, USA, 1987.

9. Sheet Metal and Air Conditioning Contractors National Association, *Duct Design Handbook*, Vienna, Virginia, USA, 1981.

10. AMCA Standard 210-85/ASHRAE Standard 51-1985, "Laboratory Methods of Testing Fans for Ratings," American Society of Heating, Refrigerating, and Air-Conditioning Engineers (Atlanta, Georgia) and Air Movement and Control Association, Inc. (Arlington Heights, Illinois), USA, 1985.

11. Air Movement and Control Association, Inc. "Air Systems," Publication 200, Arlington Heights, Illinois, USA.

12. Air Movement and Control Association, Inc., "Fans and Systems," Publication 201, Arlington Heights, Illinois, USA.

13. L.J. Nilsson and E.D. Larson, "Adjustable Speed Drives," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

14. R. Pearson, Pearson Engineering, Madison, Wisconsin, USA, personal communication, 1989.

15. "Economic Methods and Assumptions," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

NOTES

1. For a given fan, efficiency remains constant between operating points calculated from the fan laws [2].

2. From the fan laws [2], it follows that changing the operating speed of a fan without changing the system characteristics will move the operating point along the system characteristic curve.

3. Recognizing how the operating point is defined is fundamental to understanding air system technology. If, for example, the system pressure drops are carefully calculated and the fan and system curves are plotted, and then subsequent field tests show less air is being circulated than expected, then either the fan is not running at design speed or the system curve was incorrectly calculated. The installed fan and system can be tested and the measured data can be compared with the design fan performance and system curves to determine the actual operating point.

4. Proper installation minimizes the negative effect of swirl by positioning the dampers such that they impart spin to the air entering the fan impeller in the direction of wheel rotation.

Vol. 1: Movement of Material **89-8-8**
System: **SOLIDS CONVEYING**

by
William E. Biles, Ph.D.
Department of Industrial Engineering
University of Louisville
Louisville, Kentucky, USA

CONTENTS

SUMMARY

Material handling is estimated to account for 30 to 75% of the cost of manufacturing different products--some $40 billion is spent annually in the USA on material handling equipment--yet material handling adds no direct value to the product. Thus, there is an ongoing effort to reduce capital costs of conveyor designs. Energy use is not traditionally a major concern. This Menu entry describes a variety of equipment for conveying bulk solids and unit materials, and provides illustrative material for estimating energy use. Different means of conveying the same material can have very different energy requirements (see table). This entry also discusses recent developments to reduce material handling capital costs, some of which also lead to reduced energy use. These include the "just-in-time" delivery concept pioneered in Japan, mini-load systems, which use smaller motors and save storage space by allowing automated storage/retrieval systems to be used, and automated guided vehicles (AGVs), which can replace electrical lift trucks in some applications and reduce electricity use by 10 to 25%.

Estimated electricity use for conveying wood chips using alternative conveying technologies.

Conveying System	Electricity Use (Wh/tonne-meter)
Pneumatic	15
Screw	10
Belt, vertical	5
Belt, horizontal	0.5

1. TECHNOLOGY

Introduction

Material handling refers to the movement and storage of parts, material, and finished products so that they are on hand when they are needed in a manufacturing process or service operation. Thus, material handling is a primary component of the manufacture and distribution of goods. Some $40 billion is spent annually in the USA for material handling equipment in manufacturing alone [1]. About 9% of this total is spent for fixed-path solids conveying equipment, 11% is spent for industrial trucks and other wheeled vehicles for variable-path conveying, and some 33% is spent for pallets, unit-load packaging and racks and shelving for storage.

Material handling accounts for an estimated 30 to 75% of the cost of manufacture, depending on the product [2]. Yet material handling adds nothing to the value of the product. Hence, optimal material handling design is critical to the design of cost-efficient manufacturing systems.

Several methods of handling materials can usually be used to meet a given need, as described in Section 3. Each application requires a careful assessment of the need for material handling and alternative approaches to meeting the need before finalizing and implementing a design. The most appropriate choice in any particular case depends on technical considerations: type of material, travel paths/distances, required positioning for use, rate of demand for material at the point of use, availability/accessibility of storage space; and on cost. Energy use is traditionally not a major design criterion, but steps taken to reduce overall material handling costs will generally also decrease total energy use, as discussed further in Section 4. Material handling cost reduction is the primary objective of ongoing R&D in this area (see Section 2).

Material handling systems fall into one of two general categories: bulk solids conveying or unit handling.

Bulk Solids Conveying

Bulk solids conveying involves the movement and storage of flowable solids, such as fine, free-flowing materials (e.g., wheat flour or sand), pelletized materials (e.g., soybeans or soap flakes) or lumpy materials (e.g., coal or wood chips). Standardized equipment designs and complete engineering data are available for many types of conveyors [2,3,4], so their performance can be accurately predicted when they are used with materials having well-known conveying charateristics. Some of the primary factors involved in bulk solids conveyor equipment selection are (1) capacity requirement: for example, belt conveyors can operate at high speeds and deliver large weights and volumes of material. On the other hand, screw conveyors can be cumbersome in large sizes, and cannot be operated at high speeds without severe abrasion problems; (2) length of travel: for example, belt conveyors can span kilometers, whereas pneumatic and vibration conveyors are limited to hundreds of meters; (3) vertical lift: for example belt conveyors can be used for some relatively gradual elevation changes, while vertical bucket elevators are commonly applied in cases where the angle of inclination exceeds 30 degrees; (4) material characteristics, such as their chemical and physical properties (particularly flowability); (5) processing requirements, i.e., the treatment material incurs during transport, such as heating, mixing, or drying; (6) life expectancy of the equipment; and (7) comparative capital and operating costs.

Table 1 summarizes the most common functions of industrial conveying equipment. Table 2 describes the type of equipment

Table 1. Types of conveyor equipment and their functions.

Function	Conveyor Type
Conveying horizontally	Apron, belt, continuous flow, drag flight, screw, vibrating, bucket, pivoted bucket, air
Conveying up or down an incline	Apron, belt, continuous flow, flight, screw, skip hoist, air
Vertical lift	Bucket elevator, continuous flow, skip hoist, air
Combination horizontal & vertical path	Continuous flow, gravity-discharge bucket, pivoted-bucket, air
Distributing to or collecting from bins, bunkers, etc.	Belt, flight, screw, continuous flow, gravity-discharge bucket, pivoted-bucket, air
Removal from railcars, trucks, etc.	Car dumper, grain-car unloader, car shaker, power shovel, air

Table 2. Material characteristics and typical corresponding material handling equipment.

Material Characteristics	Typical Handling Equipment
Fine, free-flowing	Bar flight, belt, oscillating or vibrating, rotary vane, screw
Non-abrasive and granular or with some lumps	Apron, bar flight, belt, oscillating or vibrating, reciprocating, rotary plate, screw
Difficult materials due high temperature abrasiveness, lumpiness, or stringiness	Apron, bar flight, belt, oscillating or vibrating, reciprocating
Heavy, lumpy or abrasive, e.g., pit-run stone and ore	Apron, oscillating or vibrating, reciprocating

typically selected for conveying different materials. The equipment listed in Tables 1 and 2 are described in greater detail in Section 3.

In addition to the conveyor itself, conveyor drives, motors, and auxiliary equipment must also be specified in a design. Conveyor drives comprise from 10 to 30% of the total cost of a conveyor system [4]. Fixed-speed drives and adjustable speed drives [5] can be used, depending on the variability of the load during normal operation. It is estimated that from 10 to 25% of all conveyor systems now being installed utilize energy-efficient electric motors [1], which (depending on the size of the motor) can yield from 5-15% greater efficiency at a 5-10% greater capital cost than standard-efficiency motors [6]. Auxiliary equipment includes such items as braking or arresting devices on vertical elevators to prevent reversal of travel, torque-limiting devices or electrical controls to limit power to the drive motor and cleaners on belt conveyors.

Unit Material Handling

Unit material handling involves the movement and storage of separate, individual parts, goods, or assemblies. Examples include automobile body components, engine blocks, bottles, cans, bags, pallets of boxes, bins of loose parts etc. The basic categories of unit handling equipment are conveyors, which transport materials along fixed paths, and vehicles, which can transport materials along variable paths. Conveyors are further subdivided into gravity and driven types. Gravity conveyors include chutes, slides, gravity wheel and roller conveyors, all of which exploit gravity to move items from an elevated point to a lower point. Material handling systems which maximize the use of gravity are attractive in that they tend to be low in cost and maintenance requirements. They are also zero energy users. Driven conveyors use electric motors to drive belts, chains, or rollers in a variety of floor-mounted, in-floor, and overhead types of conveyors.

As in the case of bulk solids conveying, the choice of technology for unit handling typically depends on the shape, size and weight of units to be moved, travel paths/distances, the required rate of movement, capital and operating costs, and a number of other factors which are often application-specific. The characteristics of various common types of unit handling equipment are discussed in Section 3.

2 R&D

Most of the ongoing research and development in material handling is aimed at reducing the volume of material moved and the distance through which it must be moved. Although energy usage is not expliciting considered in most such work, both such improvements have direct implications for reduced energy consumtion.

One well-known thrust in manufacturing systems design is the "just-in-time" concept pioneered in Japan [7]. This involves bringing raw material and in-process parts to a work center precisely at the time it is needed and exactly in the quantity needed. "Just-in-time" greatly reduces or eliminates material handling and in-process storage, with a drastic reduction in inventory holding costs. Overall material handling costs are also reduced, with a corresponding decreased demand for energy.

A second avenue for material cost reduction has been greater use of so-called "mini-loads," i.e., unit loads on bins in the range of 90-340 kg [8]. This allows smaller conveyors to be used powered by small electric motors. Mini-load also enables storage of materials in high-rise automated storage/retrieval systems (AS/RS), which greatly reduces required floor space for storage. In cases where storage space conditioning is required (e.g., refrigeration), such space savings can reduce space conditioning energy requirements substantially.

A third area where advances are being made in material handling is in the expanded use of automated guided vehicles (AGVs) [9]. Research and development efforts in this area have focussed on designing layouts, e.g., cart paths which minimize the number of vehicles needed. Since AGVs are typically battery powered, system designs have also been aimed at getting up to 20 hours of use per vehicle per day, with the remaining hours used for maintenance and battery charging. Charging energy costs can be minimized by scheduling re-charging during off-peak hours.

There has been some effort to explicitly encourage the energy efficient design and operation of materials handling systems. Such efforts typically result in recommendations to designers (e.g., see [10]) to: reduce oversizing of equipment (which saves both capital and operating costs), stress regular preventive maintenance, teach good operator skills (e.g., repeatedly starting and stopping a load along a track wastes energy because of the very high starting current drawn by the motor each time it starts), and to use manual rather than powered conveying whenever the application permits.

3. TECHNICAL AND COST DATA

The technical characteristics of some of the most common types of bulk solids and unit material conveying equipment are reviewed here. Vendors offer standardized equipment and typically quote costs on a job-by-job basis. Thus, vendors should be consulted for costs of the equipment discussed here.

Bulk Solids Conveying

Belt Conveyors: These are widely used in industry [11]. They can traverse distances up to several kilometers at

Table 3. Approximate design data for conventional belt conveyors as a function of capacity and material characteristics.[a]

Belt width (cm)	Belt speed[b] (m/s)	*Maximum Lump Sizes (cm)* Sized, 80% under	Unsized, not more than 20% over	Capacity (tonnes/h)	*Power Requirements (kW)* Per 3-m lift	Per 30-m centers	Add for tripper
35.6	0.5-1.5	5.08	7.6	14.5	0.127	0.164	0.746
40.6	0.5-1.5	6.35	10.2	20.0	0.172	0.209	0.933
45.7	0.5-1.8	7.62	12.7	24.5	0.216	0.261	1.12
50.8	0.5-1.8	8.89	15.2	30.0	0.261	0.313	1.19
61.0	0.5-2.0	11.4	20.3	44.5	0.380	0.380	1.31
76.2	0.5-2.3	17.8	30.5	71.8	0.597	0.560	1.87
91.4	0.5-3.0	20.3	38.1	105	0.910	0.597	2.63
106.7	0.5-3.0	25.4	45.7	150	1.31	0.850	3.57
121.9	0.5-3.0	30.5	53.3	200	1.74	1.130	4.79
137.2	0.5-3.0	35.6	61.0	259	2.25	1.470	7.88
152.4	0.5-3.0	40.6	71.1	327	2.85	1.86	

(a) From [4] for material with a bulk density of 800 kg/m^3. For materials with other bulk densities, the power requirements can be scaled linearly for preliminary estimates. The capacity and power requirement are shown for a belt speed of 0.5 m/s. Within the belt speed ranges indicated, capacity and power vary linearly with belt speed.

(b) Feasible operating range. See note (a).

speeds up to 5 m/s and can handle thousands of tonnes per hour. They are generally used for horizontal conveying and for inclined transport up a maximum angle of 30 degrees. Horizontal changes in direction are achieved using such devices as connecting chutes and slides between separate belt sections.

Belt-conveyor design depends largely on the nature of the material to be handled. Particle-size distribution, bulk density, and chemical composition dictate the width, type, and operating speed of the belt. For instance, oily substances generally rule out the use of natural rubber belts. Operating conditions which can affect belt conveyor design include climate, surroundings, and period of continuous service. For instance, continuous service operation will require higher-quality components than will intermitent service which allows more frequent maintenance.

Table 3 gives approximate power requirements for traditional belt conveyor design as a function of material characteristics and loading requirements. The power requirement accounts for power needed to overcome starting inertia, to drive the empty belt, to move the load against the friction of rotating parts, to elevate (or lower) the load and to operate a tripper (which engages and disengages the belt from the drive rollers). Belt conveyors are one of the most energy-efficient means for bulk solids conveying, especially when properly maintained and when energy efficient motors are used.

Screw Conveyors: These consist of helical shafts mounted within pipes or troughs, with power transmitted through the helix or, with a fully enclosed pipe conveyor, through the pipe itself [12]. Material is forced through the channel between the helix and the pipe or through. Figure 1 shows a chute-fed screw conveyor, one of several common designs. Screw conveyor capacity is generally limited to about 300 m^3/hr. Table 4 gives approximate power requirements for traditional-design screw conveyors as a function of material characteristics and loading. Screw conveyors tend to be relatively energy intensive by comparison to other systems for horizontal transport. For example, compared to a belt conveyor moving the same material over an equal distance, screw conveyors will use some 25 to 40 times as much electricity.[1]

Table 4. Approximate design data for conventional screw conveyors as a function of capacity and material characteristics.

Capa-city[b] (t/h)	Speed (rpm)	Maximum size of lumps (mm) with lump percent: all	<25	<10	Maximum Torque Capacity (N-m)	Power (kW) required at the motor for maximum length of: 4.6-m	9.1-m	13.7-m	18.3-m	22.9-m
4.5	40	19.1	38.1	57.2	859	0.32	0.63	0.95	1.26	1.57
9.1	55	19.1	38.1	57.2	859	0.63	1.26	1.68	2.24	2.80
13.6	80	19.1	38.1	57.2	859	0.95	1.68	2.52	2.94	3.68
18.2	60	25.4	50.8	76.2	859	1.26	2.24	2.94	3.63	4.20
22.7	75	25.4	50.8	76.2	859	1.58	2.80	3.68	4.20	4.89
27.3	55	31.8	57.2	88.9	1853	1.68	2.94	3.77	5.04	5.60
31.8	60	31.8	57.2	88.9	1853	1.96	3.42	4.40	5.22	6.53
36.4	50	38.1	76.2	102	1853	2.24	3.36	5.04	5.97	7.46

(a) From [4] for material with a bulk density of 800 kg/m^3 and average conveyability. The motor size requirements are estimated for average conditions, with factors for length of conveyor, momentary overloads, etc. taken into consideration.

(b) The five smallest capacities are for units with a pipe diameter of 6.35 cm, shaft diameter of 5.08 cm, and flight diameters ranging from 22.9 to 30.5 cm. The largest three capacities are for units with a pipe diameter of 8.89 cm, shaft diameter of 7.62 cm, and flight diameters ranging from 35.6 to 40.6 cm.

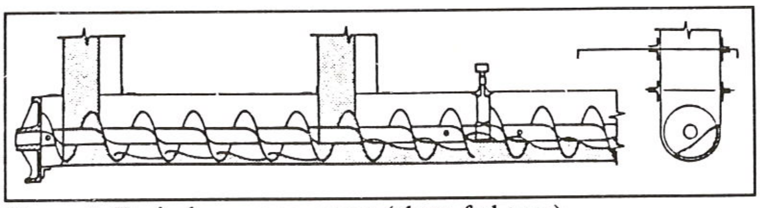


Figure 1. Typical screw conveyor (chute-fed type).

The Continuous-Flow Conveyor: This is a totally enclosed unit that operates by pulling a surface transversely through a mass of material such that it pulls along with it a volume of material (Fig. 2). Common types of continuous-flow conveyors include closed-belt conveyors, flight conveyors and apron conveyors [13].

Vibrating (or Oscillating) Conveyors: These are directional-throw devices that consist of a spring-supported horizontal pan or trough, vibrated by an attached arm or rotating weight [12]. Material is abruptly tossed upward and forward so that it travels in the desired directions, after which the conveyor returns to its reference position. The capacity of the vibrating conveyor is determined by the magnitude and frequency of trough displacement, angle of throw, slope of the trough, and the ability of the material to receive and respond to the power imparted by the trough.[2] Capacitites of vibrating conveyors are very broad, ranging from a few grams for laboratory-scale equipment to thousands of tonnes for heavy industrial applications. Vibrating conveyors are commonly classified according to their power source: mechanical, electrical, and pneumatic or hydraulic.

Pneumatic Conveyors: These transport bulk solids suspended in a stream of air over vertical and horizontal distances ranging from a few

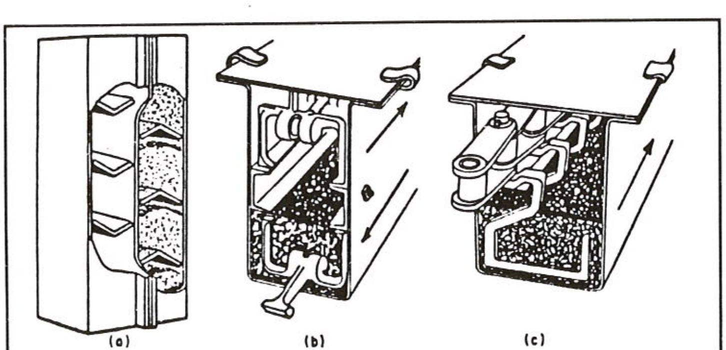


Figure 2. Continuous flow conveyors (a) showing closed and open flights, (b) conveyor-type elevator and (c) horizontal conveyor with side-pull chain.

centimeters to hundreds of meters. Fine powders are especially well-suited to this means of conveyance, although particle sizes up to a centimeter in diameter can be effectively transported. Materials with bulk densities up to 1600 kg/m^3 can be transported pneumatically. The capacity of a pneumatic conveying system depends on such factors as the material's bulk density, mass flow rate and velocity, and the length and diameter of the conveyor.

Table 5. Estimated electricity use for conveying wood chips with alternative conveying systems.[a]

Conveying System	Electricity Use Wh per tonne-meter
Pneumatic	15
Screw	10
Belt, vertical	5
Belt, horizontal	0.5

(a) From [17] for full capacity operation. Differences would be larger with partial loading.

Pneumatic transport is generally energy intensive compared to belt, screw or vibrating conveyor transport. For example, a pneumatic conveyor operating at full capacity to horizontally transport wood chips requires about 50% more electricity than a screw conveyor and some 30 times more electricity than a belt conveyor (Table 5).

The four basic types of pneumatic conveyor systems are pressure, vacuum, combined pressure-vacuum, and fluidizing (Fig. 3). In pressure systems, the material is charged into an air stream operated at higher-than-atmospheric pressure, such that the velocity of the air stream maintains the solid particles in suspension until it reaches the separating vessel, usually an air filter or cyclone separator. Vacuum systems operate in much the same way, except that the pressure of the system is kept lower than atmospheric pressure. Pressure-vacuum systems combine the best features of both systems, with a separator and a positive-displacement blower placed between the vacuum "charge" side of the system and the pressure "discharge" side. A common application of pressure-vacuum systems is combined bulk vehicle (e.g., hopper car) unloading and transport to storage. Fluidizing systems pass air through a porous membrane which forms the bottom of the conveyor, thus giving finely divided, non-free-flowing bulk solids the characteristics of free-flowing material. This technique, which is

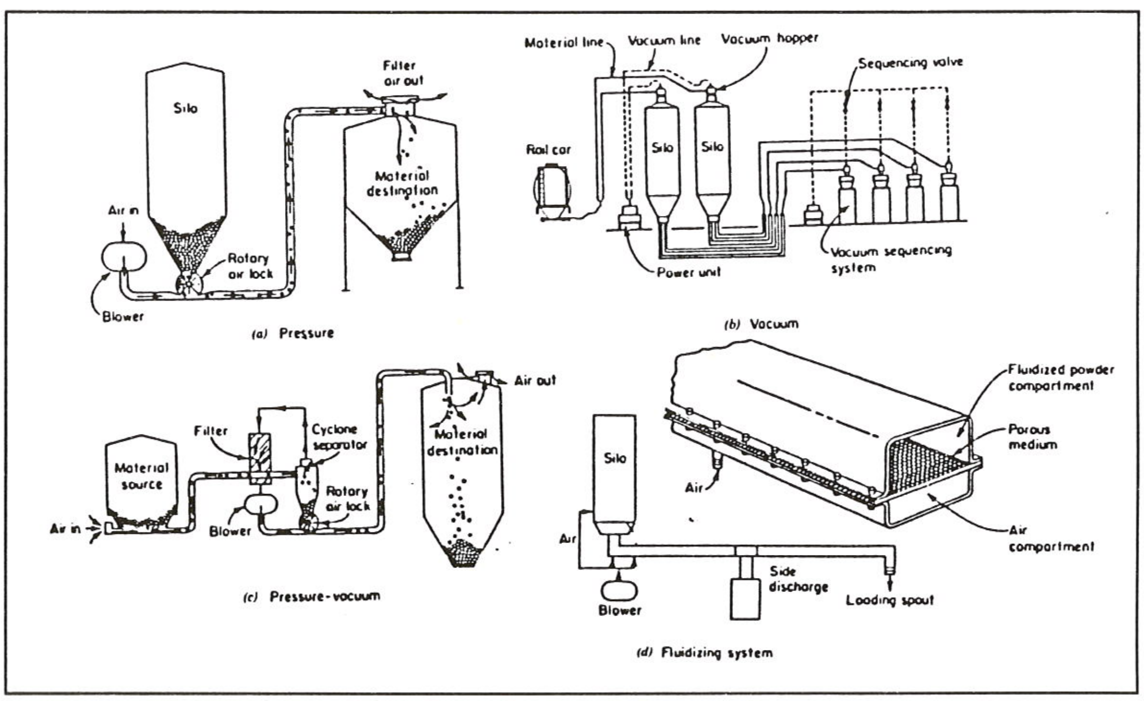


Figure 3. Four types of pneumatic conveyor systems.

commonly employed in transporting material over short distances (e.g., from a storage bin to the charge point of another type of pneumatic conveyor), has the advantage of reducing the volume of conveying air needed, thereby reducing power requirements.

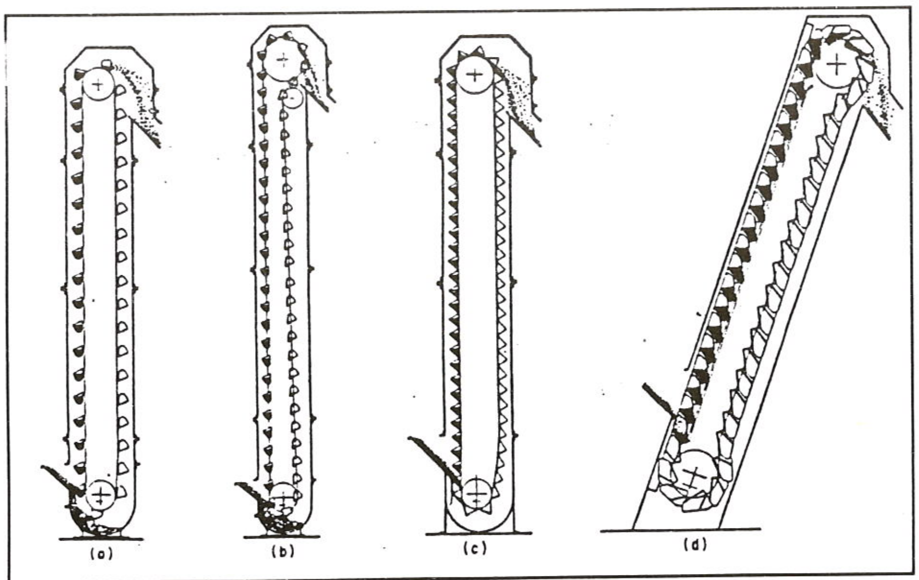


Figure 4. Four bucket elevator designs: (a) centrifugal-discharge spaced buckets, (b) positive-discharge spaced buckets, (c) continuous bucket and (d) super-capcity continuous bucket.

Bucket Elevators: These are used for vertical transport. They are available in a wide range of capacities and may operate in the open or totally enclosed. Highly standardized units tend to be used, although individually engineered units are used with special materials, unusual conditions or high capacities. Figure 4 shows several common types of bucket elevators. The super-capacity continuous bucket elevator shown there handles high tonnages and is usually operated at an incline to improve loading and discharge conditions. Table 6 provides some preliminary design data for bucket elevators.

Unit Material Handling

As noted in Section 1, the two basic categories of unit material handling equipment include conveyors, which move material along fixed paths, and vehicles, which can move material along variable paths. Conveyors include gravity and driven types. Gravity conveyors come in a variety of designs (e.g., Fig. 5 shows a section of a skate wheel conveyor). All are simple, low cost devices that require no power. From an energy efficiency stand point, they should be used wherever possible. The discussion in this section focusses on driven conveyors.

Driven Conveyors: Driven conveyors include powered, chain-driven, power-and-free, and hoist/crane/monorail conveyors. Electric motors are used to drive all of these. The motor power requirements are typically a function of the total mass which must be moved, the inherent friction in the system, any elevation change, and the expected peak loading. Conventional methods of estimating power requirements of alternative driven conveyors are given in [14]. The basic operating principles of the various designs are summarized here.

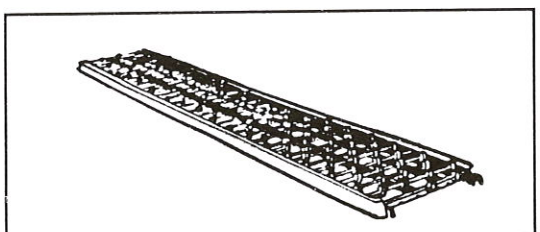


Figure 5. Skate-wheel gravity conveyor for unit material handling.

Powered conveyors include belt and roller types. Belt conveyors operate using the same basic principles as those discussed under bulk solids handling. Roller conveyors can move heavier loads than belts and are of sturdier construction. They can also be used to provide spacing between items by alternatively activating and deactivating

Table 6. Typical power requirements for a common bucket-elevator design.[a]

Bucket size (mm) & spacing [mm]	Elevator centers (m)	Capacity (t/h)	Lump size handled (mm)[b]	Bucket speed (m/min)	Head drive shaft (rpm)	(kW)	Additional kW per meter for intermediate lengths
152x102x108	7.6	12.7	19.1	68.6	43	0.75	0.049
[305]	15.2	12.7	19.1	68.6	43	1.19	0.049
	22.9	12.7	19.1	68.6	43	1.57	0.049
203x127x140	7.6	24.5	25.4	68.6	43	1.19	0.098
[356]	15.2	27.2	25.4	79.2	41	2.61	0.122
	22.9	27.2	25.4	79.2	41	3.58	0.122
254x152x159	7.6	40.8	32.0	68.6	43	2.24	0.154
[406]	15.2	47.2	32.0	79.2	41	3.88	0.171
	22.9	47.2	32.0	79.2	41	5.37	0.171
305x178x184	7.6	68.1	38.1	79.2	41	3.51	0.245
[457]	15.2	76.3	38.1	91.4	38	6.64	0.281
	22.9	76.3	38.1	91.4	38	8.73	0.281
355x179x184	7.6	90.8	44.5	91.4	38	5.45	0.343
[457]	15.2	90.8	44.5	91.4	38	8.21	0.343
	22.9	90.8	44.5	91.4	38	10.7	0.343
406x203x216	7.6	136.2	50.8	91.4	38	6.34	0.404
[457]	15.2	136.2	50.8	91.4	38	9.40	0.404
	22.9	136.2	50.8	121.9	38	12.5	0.404

(a) From [4] for centrifugal-discharge bucket elevator with steel buckets on a belt. Capacities and power requirements are for material with a bulk density of 1600 kg/m^3. Linear scaling can be used to estimate capacity and power requirements for materials with different bulk densities.

(b) If the volume of lumps averages less than 15% of total volume, lumps of twice the indicated size may be handled.

driver motors in material entry sections of the conveyor. Inclines up to about 10 degrees and declines down to about 15 degrees are possible. Fig. 6 shows a curved section of a roller conveyor. Live-roller conveyors that are chain-driven are used in more severe applications (heavy loads, oily, dirty, or wet conditions, extreme temperatures) than belt-driven units. Of the two types of chain-driven live rollers, the continuous-chain type has a lower initial cost than the roller-to-roller type, but is more limited in application.

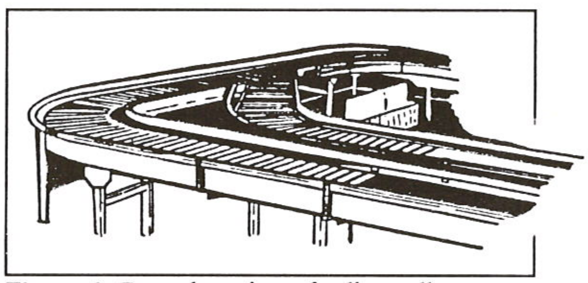


Figure 6. Curved section of a live-roller powered conveyor for unit material handling.

Chain-driven conveyors are those in which closed-loop systems of chains are used to pull items or cars along a specified path. The three principal types of chain-driven conveyors are (1) flight conveyors, (2) overhead towlines and monorails, and (3) in-floor towlines. Flight conveyors consist of one or more endless strands of chain with spaced transverse flights or scrapers attached, which push the material along through a trough. Used primarily in bulk material handling, its primary use in unit material handling includes movement of cans or bottles in food canning and bottling. Flight conveyor speeds are limited to less than 0.6 m/s. Overhead towlines consist of a track mounted 2.5-3 m above the floor. Carts on the floor are attached to the chain, the other end of which is pulled through the track. Overhead towlines free the floor for other uses (when not needed for conveying), and are cheaper and more flexible than in-floor towlines. In-floor towlines consist of chain tracks mounted in the floor. Carts are pulled along the track by connecting chains. In-floor towlines are capable of greater speeds than overhead towlines and start more smoothly. They are more difficult to maintain, however, and lack flexibility for re-routing.

Power-and-free conveyors combine a driven trolley conveyor and an unpowered monorail-type free conveyor. A set of tracks is positioned above another. The upper track carries a chain-driven trolley or monorail-type conveyor. The lower track supports unpowered (free) load-carrying trolleys which are engaged by pushers attached to the driven trolleys. The loaded trolleys can be switched from one free track to another, with such switches controlled manually or by computer. Power-and-free conveyors offer great flexibility. For example, track switches may divert trolleys from power to free tracks, and speeds may vary from one powered section to another. Accurate design calculation of the power required for a power-and-free conveyor is made difficult because of a large number of variables, including the total live load and its distribution along the track at any one time and the total frictional losses in the system. Typically, therefore, power-and-free conveyors are designed with motors oversized for normal operation (to handle the absolute worst condition possible). The more accurately the loading and friction conditions can be predicted, the lower can be the operating energy cost of the design.

Hoists, cranes, and monorails are used for a variety of overhead handling tasks. A hoist typically consists of a hook, a lifting medium--a chain, cable or rope--and a drum for reeling in unused chain, cable, or rope. Hoists may be manually operated or pneumatically or electric-motor driven. A monorail is a single-beam overhead track whose lower flange serves as a runway for a trolley-mounted hoist. Though the use of switches, turntables, and other path-changing devices, an overhead monorail carries a series of trolleys through a fixed path. A chain-driven overhead monorail is very similar to the overhead towline in its configuration, except that it carries uniformly spaced trolleys overhead instead of pulling carts along the floor. The monorail can be designed to dip down at specified points to deliver items to machines or other processing stations. A crane also involves a hoist connected to a trolley, which typically travels along a fixed beam (e.g., the bridge cranes shown in Fig. 7). Cranes are manually, electrically or pneumatically driven.

Industrial Trucks: Industrial trucks provide flexible handling of materials along variable flow paths [9]. Probably the most commonly used type of truck is the forklift (or lift) truck. Lift trucks are very effective in lifting, stacking and unloading materials from storage racks, highway vehicles, railroad cars, etc. Some are general-purpose, while others are designed for specific tasks such as narrow-aisle or high-rack handling.

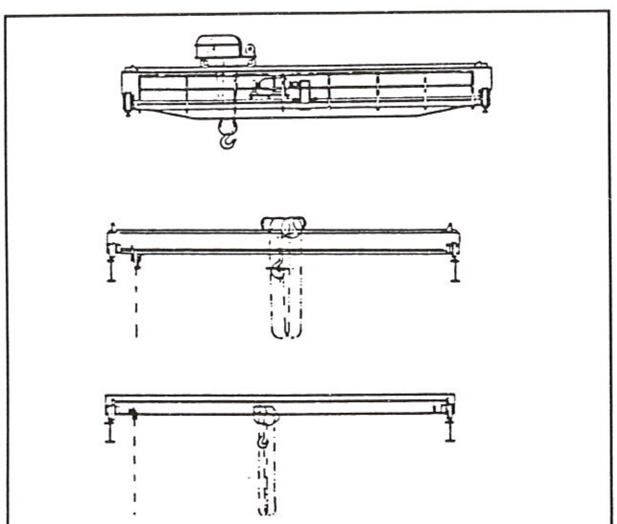


Figure 7. Three bridge crane designs for unit material handling.

Depending on the model, the operator walks behind the truck or rides in it either standing or sitting. Lift trucks are driven manually, by electric motors, or internal combustion (IC) engines using a variety of fuels.

IC engines and electric motors are used to drive lift trucks in the 450 to 4,500 kg capacity range. The choice of truck type for any particular application depends on required performance, expected economy (first and fuel costs), environment (indoor or outdoor use), and type of truck generally available locally.

IC engine trucks are powered by the familiar engines found in passenger cars and other such vehicles. Gasoline, diesel, or liquefied petroleum gas are generally used for fuel.

Electric trucks are powered by a lead-acid industrial traction storage battery feeding a low-voltage, direct-current, heavy-duty industrial motor with a large overload capacity.[3] Motors are selected on the basis of the torque they produce for a given voltage and current draw. Compared to IC engines, the horsepower output is relatively low.[4] A continuously variable silicon-controlled rectifier (SCR) control system varies the power delivered to the motor. This provides a much smoother change in vehicle speed when compared to any form of gear shifting. Separate motors may be used to provide auxiliary power for other functions such as the hydraulic, tilting, lifting, steering, and attachment systems.

4. ILLUSTRATIVE ECONOMIC ANALYSIS

The design (and energy consumption) of a material handling system for a particular material can vary significantly depending on which design criteria are given higher priority and on design choices made by the individual design engineer. It is thus difficult to provide economic analyses that will be representative of any real application. In most applications, however, the primary objective in designing a material handling system should be to minimize lifecycle costs while meeting the given demand.[5] Two of the most important considerations in minimizing lifecycle cost are to design the system to minimize (1) the quantity of material moved and (2) the distance over which it is moved.

These design objectives are entirely consistent with helping to minimize the system's energy requirements. This is evident by noting that the energy required to move a mass of material, m, through a distance, d, is the product of m, d, and the acceleration required to overcome frictional and gravitational resistances in the system. Thus, any decrease in m or d achieves a direct decrease in energy consumption.

Other design priorities can also lead to significant reductions in energy use. These include:

* Using gravity conveyors for unit material handling where possible.

* Using energy efficient motors where possible. Payback times on the extra investment cost of an EEM versus a standard motor will be short for a range of electricity prices and annual operating hours [6,16].

* Reducing design "safety" factors. With accurate understanding of expected loads and with more careful design, such factors should be able to be reduced from the 15-25% commonly used today to about 5%.

* Replacing lift trucks with automated guided vehicles where possible, since AGVs typically require only 75-90% of the electrical energy of lift trucks [9].

5. REFERENCES

1. A. Leffler, Material Handling Institute of America, Research Triangle Park, North Carolina, USA, personal communication, 1989.

2. R.A. Kulwiec (ed.), *Material Handling Handbook*, 2nd ed., Wiley Interscience, New York, 1985.

3. J.M. Apple, *Material Handling Systems Design*, Ronald Press, New York, 1972.

4. W.E. Biles and M.E. Zohdi, "Material Handling," in *Mechanical Engineers' Handbook*, (M. Kutz, ed.), John Wiley, New York, 1987.

5. L.J. Nilsson and E.D. Larson, "Adjustable Speed Drives," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

6. L.J. Nilsson and E.D. Larson, "Electric Motors," *Technology Menu*, Vol. 1, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

7. J.O. McClain, and L.J. Thomas, *Operations Management - Production of Goods and Services*, Prentice-Hall, Englewood Cliffs, NJ, USA, 1985, pp. 388-396.

8. J.A. Tompkins and J.A. White, *Facilities Planning*, John Wiley, New York, 1984.

9. W. O'Connell, K.E. Lanker, J.M Snyden, C.C. Simpson, R. Cammack, L. May, C.A. Isenberger, A.J. Cason, "Powered Industrial Trucks," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

10. Acco Industries, "Putting Your Material Handling System on an Energy Diet," *Industrial Energy Management*, Society of Manufacturing Engineers, Dearborn, Michigan, USA, 1983, pp. 170-180.

11. G.A. Schultz, "Belt Conveyors," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

12. B.J. Hinterlong, "Screw, Vibratory, and En Masse Conveyors," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

13. G.A. Schultz, "Chain Conveyors: Apron, Pan, and Flight," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

14. J.M. Cahill, J.G. Dorrance, H. Vanasselt, E. Moon, B. Curry, and R.H. Roth, "Conveyors," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

15. T. Cullinane and D. Freeman, "Evaluating and Justifying Material Handling Projects," in *Materials Handling Handbook*, 2nd ed., R.A. Kulwiec (ed.), Wiley Interscience, New York, 1985.

16. B.L. Capehart, "Energy Management Basics," *Proceedings of the Int'l. Industrial Engineering Conf.*, Institute of Industrial Engineers, Norcross, Georgia, USA, 1989, pp. 88-96.

17. Swedish Pulp and Paper Association, *Energy Compendium for the Pulp and Paper Industry*, Publication X-721, Stockholm, 1986. (in Swedish)

NOTES

1. According to Tables 3 and 4, for movement of 30 tonnes/hour of material with a density of 800 kg/m^3, the full load energy consumption with a screw conveyor would be 9-13 Wh per tonne per meter, compared to 0.34 Wh/t-m for the belt conveyor.

2. For example, dry, granular materials respond better to the throwing action than wet, lumpy materials.

3. The torque available for momentary surges is usually more than 10 times that required to move the truck on a level grade with full load.

4. An 1800 kg capacity truck may have a drive mototr rating of approximately 3.4 kW compared to some 30 kW for a similarly rated IC engine truck.

5. In many cases, the present value of lifecycle operating costs for a material handling system will be far in excess of the initial investment cost. One case study described in [15] showed that a particular gas-powered lift truck cost $16,000 to buy and had a total 5-year discounted present value operating cost of $129,000 ($12,000 for maintenance, $45,000 for fuel, and $72,000 for operating labor, using a 10% discount rate).

TECHNOLOGY MENU FOR EFFICIENT END-USE OF ENERGY

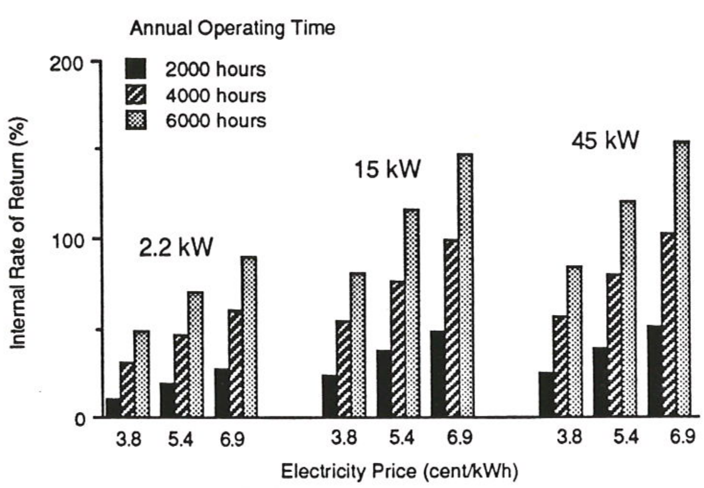
Vol. 1: Movement of Material **89-8-8**
Component: **Electric Motors**

by
Lars J. Nilsson, Eric D. Larson
Environmental and Energy System Studies
University of Lund, Lund, Sweden

CONTENTS

SUMMARY

A large fraction of electricity use is accounted for by motors. The polyphase induction motor, the most common type used in industry, is discussed here. Induction motors are available in standard-efficiency and energy-efficient models, with the latter having higher first costs, particularly in small sizes. However, since motors can easily use as much as five times their capital cost worth of electricity per year the extra investment typically can be repaid quickly in electricity cost savings. Internal rates of return (figure below) would be attractive for a variety of motor sizes, electricity prices, and annual operating hours. Also, the cost-of-saved-electricity associated with investments in energy efficient motors would be significantly less than the cost of new baseload electricity supply in most cases.



1. TECHNOLOGY

Introduction

The electric motor is probably the single largest electricity using device in the world. Table 1 shows electricity consumption estimates for different end-uses in the Swedish industrial sector, where well over half of the total is associated with motors. Table 2 shows estimated electricity consumption by different end-uses in the service sector (excluding electricity for space heating)[1] For the residential sector it is more difficult to estimate the fraction used in motors, but it is not insignificant since households typically have a number of motor driven appliances, ventilating fans, circulating pumps, etc.

Electric motors are classified as direct current, alternating current (AC) synchronous, or AC-induction type (Fig. 1). The most kWs of motor capacity sold are in AC-induction motors, primarily of the squirrel cage design. Induction motors are popular because of their simplicity and rugged construction, which leads to relatively lower capital and maintenance costs. The discussion in this Menu entry focusses on polyphase induction motors.

Polyphase motors are available in megawatt sizes down to a few watts and less, although single-phase motors are more common in the smaller sizes. Over 85% of the polyphase motors sold in Sweden are smaller than 3-kW (Table 3). Corresponding estimates of electricity consumption by motor size are not available. Larger motors, however, probably account for the bulk of electricity consumption since these tend to be operated for the largest number of hours per year [3].

Table 1. Estimated Swedish industrial electricity consumption by end use [1].

End-Use	Consumption TWh/yr	%
Motor Driven	*28.5*	*60*
Pumps & fans	13.0	27
Other motor use	7.9	17
Grinding & refining	5.5	12
Air compressors	1.5	3
Cooling plants	0.6	1
Other	*19.2*	*40*
Electrolysis	3.8	8
Other process related	3.6	8
Melting	3.4	7
Lighting	2.6	5
El. furnaces/space heat	2.3	5
Small ind., <5 employees	2.1	4
Heating	1.4	3
TOTAL	**47.7**	

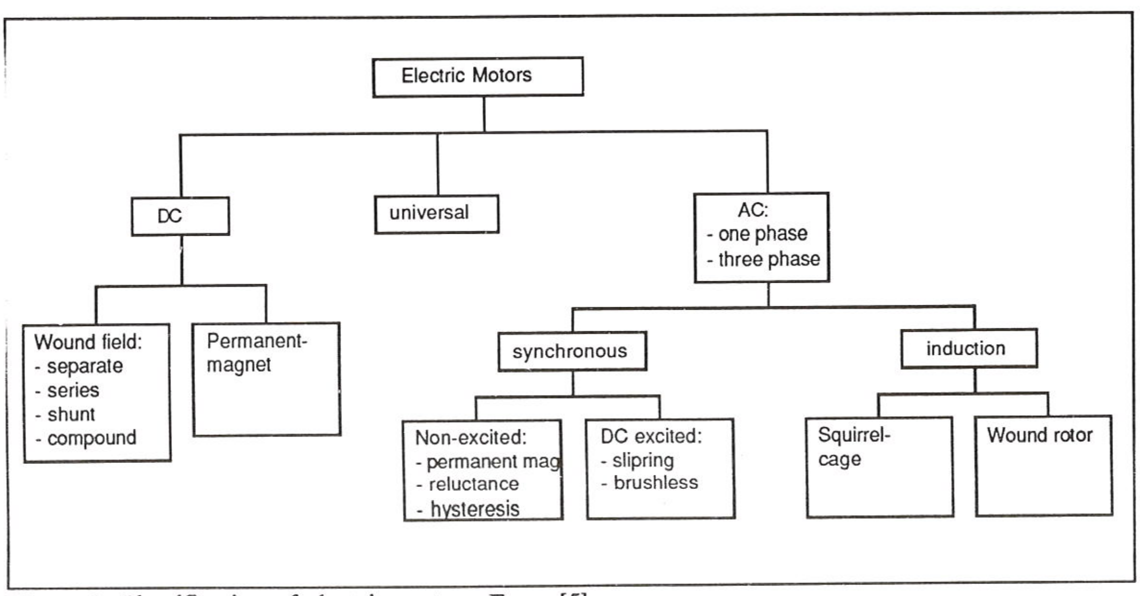


Figure 1. Classification of electric motors. From [5].

Table 2. Estimated 1987 Swedish service sector electricity consumption by end use [2].

End-Use	Consumption TWh/yr	%
Fans, pumps, etc.	9.0	37
Lighting	7.5	31
Food prep. & refrig.	4.0	17
Office electronics	3.3	14
Other	0.4	2
TOTAL	**24.2**	

Table 3. Production plus net imports of three-phase AC-motors in Sweden in 1985 [4]. Imports of motors in final consumer products are not included.

Motor size (kW)	Quantity	Percent
< 0.75	460,337	64.2
0.75 - 3	168,348	23.5
3 - 7.5	61,172	8.5
7.5 - 45	23,075	3.2
45 - 250	4,421	0.6
250 - 2500	57	-
> 2500	37	-

Polyphase Induction Motors

In an induction motor currents in the stator windings produce a rotating magnetic field, which induces currents in the rotor, which in turn produces another magnetic field. The interaction between the magnetic fields results in a torque. Losses in the process lead to heat generation so motors are usually equipped with cooling fans. Fig.2 shows, a schematic cross section of an induction motor. The losses are usually classified as one of four types: resistance losses, core losses, friction/windage losses and stray load losses. The typical distribution of losses in a standard motor is shown in Fig. 3 as a function of load.

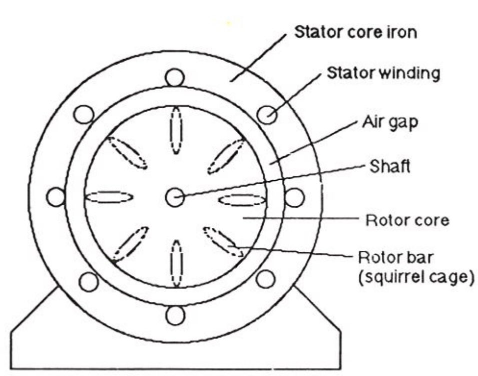


Figure 2. Cross section of a squirrel cage induction motor. From [5]

Resistance (I^2R) losses account for 50-60% of the total losses at full load and much less at lower loads. Heat is generated in the rotor and stator windings as a result of the resistance they offer to the current driven through them. The loss is proportional to the electrical resistance and the square of the current, hence the term I^2R losses.

Core losses are due to a combination of eddy currents and hysteresis. Eddy current core loss is effectively an I^2R loss arising from the circulation of eddy currents induced by the changing magnetic field in the cores. The changing magnetic field also causes hysteresis losses as the cores cannot continually align themselves quickly enough with the magnetic field.

Friction losses result from friction in the shaft bearings and windage losses from the ventilation fan used for cooling the motor and from other rotating parts.

The remaining losses are usually lumped together under the term stray load losses. These losses are affected by, among other things, saturation effects, geometries, and leakage fluxes.

Energy Efficient Motors (EEM)

Better design and materials can reduce some of the losses in standard-design motors. Motors designed for higher efficiency are loosely referred to as high-efficiency or energy-efficient motors (EEMs) [6]. Manufacturers typically label as EEMs motors with higher efficiency and higher cost. Thus, a good standard motor can have the same efficiency as a poor EEM.

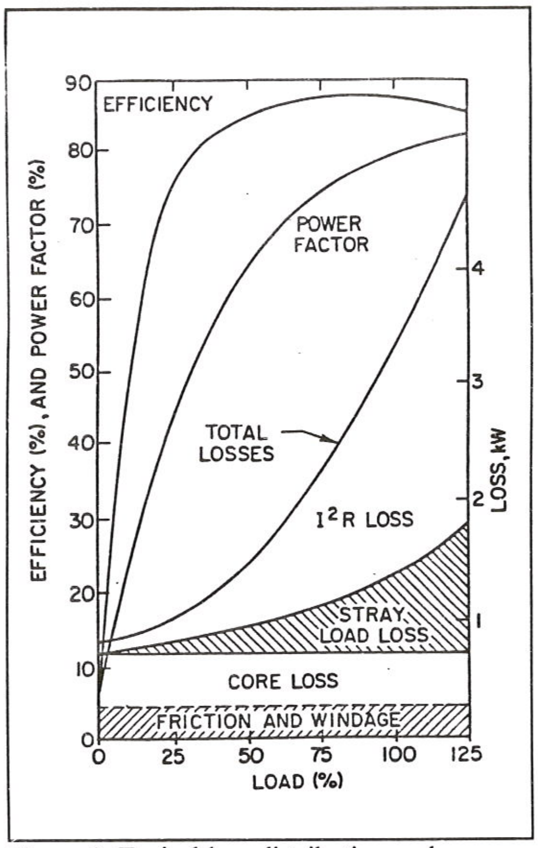


Figure 3. Typical loss distribution and approximate efficiency and power factor versus load for a 37 kW 3-phase induction motor [5].

The efficiency of any particular motor is usually expressed by vendors as a nominal efficiency, which is the average efficiency of a sampling of motors of the same model.[2] It is mostly polyphase motors that are available in energy efficient models although some manufacturers are also starting to make single-phase EEMs.

The losses discussed in the previous section are reduced through different methods in EEMs. The I^2R losses are reduced by using higher conductivity materials and larger cross-section conductors in the stator and rotor. This in turn affects parameters such as power factor, locked rotor (inrush) currents, starting torque etc. and in general there are trade-offs. For example, lower rotor resistance lowers torque capability, and higher transient inrush currents increase the draw on the power supply [5]. (The use of adjustable speed drives, which facilitate "soft" starting, can help reduce such currents.)

Three methods are used to reduce core losses. First, the cross-sectional area of the steel cores can be increased (which decreases flux density but increases weight). Second, thinner laminations can be used, which reduces eddy currents. Third, lower magnetic loss materials, such as silicon steel reduce losses by presenting greater resistivity to circulating currents and by reducing hysteresis losses.

Stray losses can be reduced, for example, by optimizing the airgap between the rotor and the stator. Reducing losses of all types reduces motor cooling requirements. As a result fan size (and windage losses) can be reduced.

2. R&D

Better design and materials have helped improve the performance of motors to the point that it can be considered a relatively mature technology. Nevertheless, continued small improvements can have a major impact on electricity use because of the long operating hours typical in many applications. Continued improvements in motor efficiency can be expected through application of better manufacturing techniques (e.g. using CAD/CAM) and new materials. Two breakthroughs in materials technology are the development of soft magnetic materials and the evolution of permanent magnets [5].

Soft magnetic materials can reduce hysteresis and eddy current core losses. So called grain-oriented electrical steel has losses about half of those in non-oriented electrical steel, which is the predominant steel in motors today. Amorphous alloys, soon to be introduced in transformers have losses about 10% of those in common electrical steel. The use of amorphous alloys is currently restricted due to factors such as hardness and brittleness, but they could well become important in the future as their development continues.

Permanent magnet (PM) motors use permanent magnets instead of rotor windings, which result in lower rotor losses. PM motors have been used mostly in small power applications to date. With development of magnetic materials[3] the range of applications for PM motors can be expected to grow to include larger motor sizes as well.[4]

A breakthrough in the development of higher temperature superconductors would also affect motor design and performance, radically reducing I^2R losses. However, the time frame for this development is much longer than others discussed here.

3. TECHNICAL AND COST DATA

A comprehensive evaluation of full-load efficiencies and prices of motors has been done for Canada, based on data from three major manufacturers [8]. The differences in efficiencies and prices between standard and EEM motors identified in the Canadian study are probably illustrative of trends in Sweden and other industrialized countries.

Standard polyphase induction motor efficiencies vary from about 70-75% for a 1-kW motor to about 92% for a 150-kW motor and up to 95% for 200-kWs and larger. Motor efficiency can be improved through measures discussed above, which typically increases the price of the motor.[5] In the Canadian study, energy-efficient motors were found to have efficiencies 6-7 percentage points higher than standard motors in small sizes and 2-3 percentage points higher in larger sizes (Fig. 4).

Permanent magnet motors would improve significantly on EEM efficiencies, particularly in the smaller sizes. Figure 4 shows measured efficiencies of three PM motors ranging from 2 to 20 kW in size.[6] In the smaller size, the PM motors would improve on EEM efficiencies by perhaps as much as 7-8 percentage points. The improvement would be 3 to 4 percentage points at 20 kW.

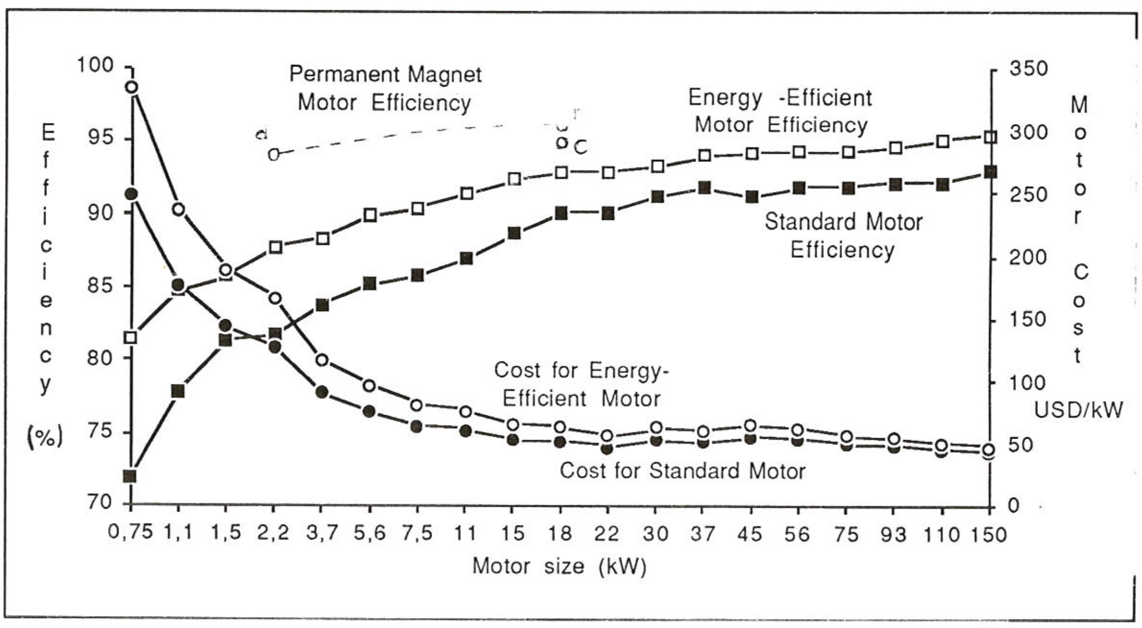


Figure 4. Estimated full-load efficiencies and wholesale prices of standard and energy-efficient motors in Canada [8]. Also shown are measured efficiencies of Ferrite (a,b) and Rare-Earth Cobalt (c) permanent magnet motors [9,10]. Note that the x-axis is not linear.

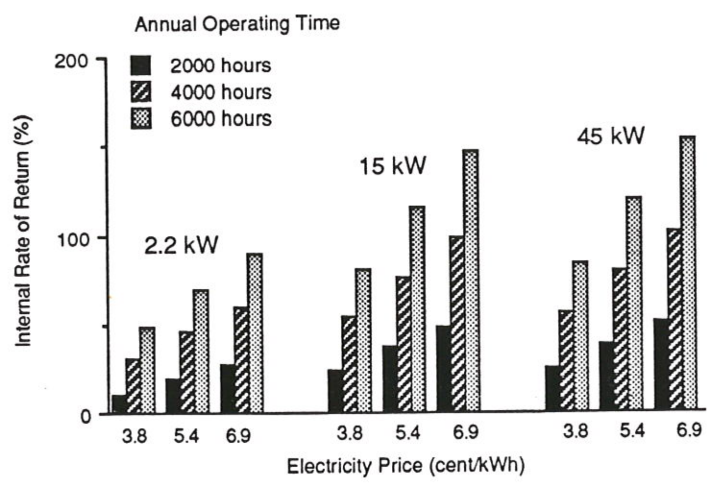


Figure 5. Internal rates of return calculated for extra investments in energy efficient motors versus standard-efficiency motors. Costs and efficiencies are from Fig. 4. Ten-year motor life is assumed.

The wholesale prices of the EEMs evaluated in the Canadian study were 15-25% higher than for standard efficiency motors in the smaller sizes, with a smaller spread in the larger sizes (Fig. 4). EEMs are as reliable as standard motors, so that maintenance costs would be not greater.[7] Costs of PM motors are currently high, but may fall with the continued development of new materials, e.g. $Nd_2Fe_{14}B$, which was developed in 1983 and has the potential for low cost and high performance [5].

The efficiencies of new motors shown in Fig. 4 may not reflect operating efficiencies in the existing stock of motors. Advances in materials in the 1950's and 1960's served to decrease motor efficiency since higher temperature capabilities of insulation materials meant that motors could run hotter and therefore less efficiently. Oversizing can also lead to reduced efficiency and poor power factors (Fig. 3), which would contribute to higher-than-necessary electric bills or lead to additional equipment costs (to improve the power factor). Use of adjustable speed drives can help reduce inefficiencies associated with oversized motors [11].

4. ILLUSTRATIVE ECONOMIC ANALYSIS

The efficiency and price estimates given in Fig. 4 can be used to illustrate the economics of investments in EEMs relative to standard-efficiency motors under various operating and electricity price conditions. Here we consider only capital and electricity costs, since other costs associated with motor use (maintenance, labor, etc.) can be assumed to be the same with either type of motor.[8]

Industrial Perspective

Internal rates of return (IRR) on the extra investments required with EEMs are shown in Fig. 5 as a function of the annual operating hours and the electricity price (assumed constant in real terms over the 10-year lifetime of the investment). In all cases shown, the saved electricity costs lead to IRRs in excess of 10%. The IRRs are lowest for the smallest units, for which the required incremental investment is highest (Fig. 4), but still exceed 20% with the electricity price at 5.4 cents/kWh or higher. With the larger units, the returns are far greater. For example, for the 45-kW motor with an electricity price of 6.9 cents/kWh, the IRR is close to 100% with 4000 hours.

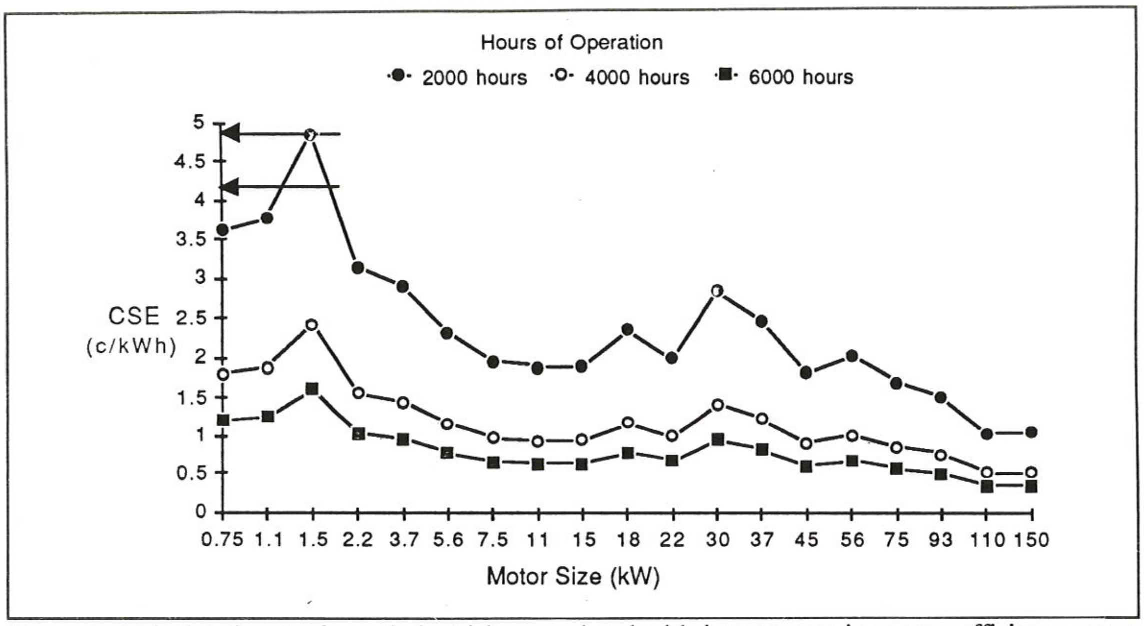


Figure 6. Calculated cost-of-saved-electricity associated with investments in energy-efficient motors versus standard-efficiency motors. The CSE is the annualized extra capital cost (from fig. 4) divided by the annual kWh electricity savings. A discount rate of 6% [12] and 10-year motor life are assumed. The upper and lower arrows indicate the estimated cost of new central station baseload electricity supplied by coal-steam or natural-gas-fired combined cycle plants, respectively [12]. Note that the x-axis is non-linear.

(The corresponding simple payback time is a year.)

With such favorable rates of return on new investments when choosing an EEM rather than a standard motor, the rate of return would often also be good in retrofit cases where the existing motor has not yet served its lifetime. In this case a more thorough analysis must be made with regard to the efficiency and the remaining lifetime of the existing motor, its salvage value, installation costs etc.

National Perspective

The cost-of-saved-electricity (CSE) associated with the extra investment for the EEMs is shown in Fig. 6 as a function of motor size for three different annual operating hours. Also shown for comparison (two arrows) are estimated costs of new baseload electricity supply in Sweden based on coal (upper) and natural gas (lower) [12]. With 2000 annual operating hours, the CSE would be comparable to the cost of new electricity supply for motors smaller than 2 kW. In all other cases shown--with longer operating hours or larger motors--the CSE would be significantly lower than the cost of supply. Because of the relatively low incremental costs for EEMs in the 100-150 kW size, the CSE for such motors would be less than 1 cent/kWh. Since, as noted earlier, larger motors consume the bulk of electricity used by all motors, the low CSEs suggest there is a significant potential nationally for saving both money and electricity through use of energy efficient motors.

5. REFERENCES

1. National Energy Administration, *Electricity Prices and Swedish Industry*", 88:7, Stockholm, Sweden, 1988. (in Swedish)

2. B. Bodlund, T.B. Johansson, T. Karlsson and E. Mills, "Technical Support Document for: The Challenge of Choices," see *Electricity*, Lund Univ. Press, Lund Sweden, 1989, pp. 883-947.

3. Assistant Secretary for Conservation and Solar Energy, *Classification and Evaluation of Electric Motors and Pumps*, DOE/CS-0147, US Dept. of Energy, Washington, DC, 1980.

4. Central Bureau of Statistics, Stockholm, Sweden, 1986.

5. S.F. Baldwin, "The Materials Revolution and Energy-Efficient Electric Motor Drive Systems," *Annual Rev. of Energy*, Vol. 13, 1988, pp. 67-94.

6. J.C. Andreas, *Energy-Efficient Electric Motors*, Marcel Dekker Inc., New York, New York, USA, 1982.

7. *Optimization of Induction Motor Efficiency*, EPRI EL-4152, Electric Power Research Institute, Palo Alto CA, USA, 1987.

8. Marbek Resource Consultants Ltd., *Energy Efficient Motors in Canada: Technologies, Market Factors and Penetration Rates*, Ottawa, Canada, 1987.

9. T.M. Jahns, "Interior Permanent Magnet Synchronous Motors for Adjustable Speed Drives," *IEEE Trans. on Ind. Appl.*, 1A-22(4), 1986.

10. E. Richter, *et al.*, "The Ferrite Permanent Magnet AC Motor--A Technical and Economical Assessment," *IEEE Trans on Ind. Appl.*, 1A-22(4), 1985.

11. L.J. Nilsson and E.D. Larson, "Adjustable Speed Drives," *Technology Menu*, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

12. "Economic Methods and Assumptions," *Technology Menu*, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

NOTES

1. The numbers in Tables 1 and 2 are only estimates. Thorough end-use assessments, e.g. based on surveys and measurements, are needed to more precisely determine electricity consumption by end use.

2. Motor efficiency is defined as the shaft power output divided by the electricity input. Efficiency is measured using different test standards: in America, IEEE 112B or CSA C390; in Europe, IEC 34.2; and in Japan, JEC 37. European and Japanese standards would show a somewhat higher efficiency than by American standards. For example, 15 kW motor tested according to CSA C390, IEC 34.2 and JEC 37 gave efficiencies of 86.9%, 87.4% and 90.4%, respectively [8].

3. For example, rare-earth PMs such as Sm_2Co_{17} or $Nd_2Fe_{14}B$, the latest of which was announced in 1983 [5].

4. In Japan air-conditioners/heat pumps aer already produced with PM motors, presenting a 6-8% efficiency improvement. [Anibal de Almeida, Universidade de Coimbra, Portugal, personal communication, 1989.]

5. It appears that in the case of single-phase motors, which are not discussed here, design optimization can lead to increased efficiency at essentially no additional cost [7].

6. The PM motor efficiencies are not from the Canadian study, as noted in the caption to Fig. 3.

7. Some manufacturers data suggest 2-4 times longer lifetimes for insulation and bearing lubricants in EEMs due to lower temperature operation [Steve Greenberg, Lawrence Berkerley Laboratories, Berkeley CA, personal communication, 1989.]

8. See [12] for descriptions of the economic indicators used in this section.

Vol. 1: Movement of Material **89-8-8**

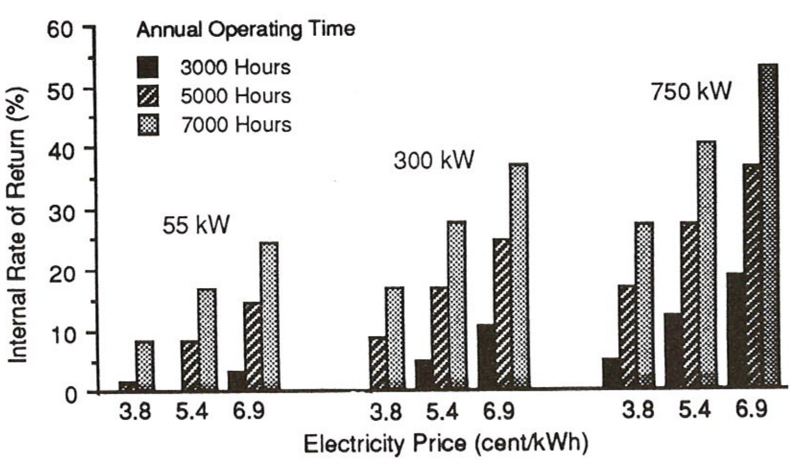
***Component*: Adjustable Speed Drives**

by
Lars J. Nilsson, Eric D. Larson
Environmental and Energy System Studies
University of Lund, Lund, Sweden

CONTENTS

SUMMARY

The use of adjustable speed drives permits continuous regulation of motor speed, which can lead to substantial electricity savings in applications where there is partial load operation during a large portion of the duty cycle. Many such applications exist in industry. The performance, costs, and potential future improvements in increasingly-popular electronic ASDs are discussed here. The economics of ASD investments are quite application-specific. Calculated internal rates of return for several sizes of ASDs, where 30% electricity savings are assumed, are quite favorable under a variety of annual operating hours and electricity price conditions (see below). The cost-of-saved-electricity is quite favorable, particularly with larger units (even with less than 30% electricity savings.)



1. TECHNOLOGY

Introduction

It is estimated that more than half of the electricity used in Swedish industry is associated with motor-driven equipment [1]. Many motor driven loads vary over time. For example, Fig. 1 shows the variable-flow requirements of a typical pumping application. Traditionally, flows driven by pumps, fans and blowers have been regulated by throttling, resulting in energy losses. Table 1 shows that low equipment utilization levels and drive-energy losses of 50-70% were commonplace around 1970 in English industry. The situation is likely to be significantly improved today, but a large potential still exists for reducing drive losses. It has been estimated, for example, that some 4-5 TWh/yr of electricity could by saved in Sweden by reducing drive losses associated with pumps and fans [3].[1]

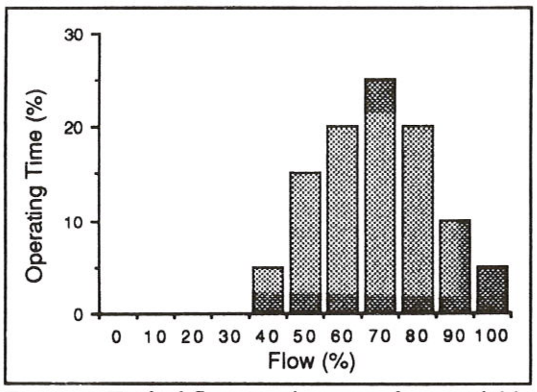


Figure 1. Typical flow requirements for a variable duty pumping application [2].

Adjustable speed drives (ASDs) permit continuous regulation of motor speed to reduce energy losses associated with variable-load applications. ASDs can be used with a wide variety of motor driven equipment including pumps, fans, blowers, conveyors, tools, drives for traction, grinders, and others. Methods for speed regulation can be divided into three basic classes: mechanical, hydraulic and electronic drives. Electronic drives are increasingly being used instead of mechanical and hydraulic drives, since electronic drives are easier to control, have higher efficiencies, require less maintenance and are increasingly becoming relatively less expensive. This Menu entry is limited to a discussion of electronic ASDs.

Electronic Adjustable Speed Drives

Electronic ASDs can be of number of different types. Table 2 summarizes those discussed here. At present, DC drives are commonly used because of the simplicity and precision with which they can be controlled. They are also relatively low in cost, although the required DC motors are relatively expensive because of their complex design. DC drives can be used in applications from fractional kW up to MW sizes. However, their use tends to be restricted to low-speed, low-power applications because of the mechanical commutation associated with the DC motor (power to the rotor is fed through sliding brushes). This design also imposes restrictions on ambient conditions due to the risk of sparking. (DC machines are not suitable in dirty or explosive environments.) The brushes and commutator are also subject to wear, which calls for greater maintenance than with AC systems.

Because of the limitations of DC systems, much effort has been, and continues to be, made to develop AC drives. AC drives can be used with either induction or synchronous motors, though the former are more common. AC drives are relatively easily retrofitted, since they do not need to be placed close to the motor. Several AC ASD designs are commercially available. The requirments of particular applications determines which should be used. All designs are one of three basic types [6]: voltage controlled, frequency controlled, and slip energy recovery systems (Fig. 2).

Voltage Controlled: The simplest method involves generating a variable voltage using an AC voltage controller (which does not alter frequency) with a high slip motor. This method is

Table 1. Estimated utilization levels and energy losses in motor-driven equipment in English industry in 1968 [4].

Sector	Type of plants visited	Plant Util. Level (%)	Typical Equipment	Losses in Drives and Controls (%)
Iron & Steel	Integrated-steel plant	40	Conveyors	>65
	Mill	40	Mills,Pumps	>65
	Finishing mill	40	Mills,Pumps	>65
Aluminum	Sectioning	40	Electro-hydr. Press	>65
Inorganic Chem.	Large bulk	50-70	Pumps,Fans	40-80
	Medium-size bulk	50-70	Pumps,Fans, Mixers	>40
Organic Chem.	Large bulk	50-70	Pumps,Fans, Mixers	40-80
	Large batch	40-50	Pumps,Fans, Centrifuges	>60
	Medium batch	35-40	Pumps,Fans, Centrifuges	>60
	Small bulk	60	Pumps,Fans	>50
Synthetic Resins	Thermoplastics	50	Pumps,Fans, Mixers,Mills	>50
Fertil-izers	Large bulk	50-60	Pumps,Fans, Compressors	40-80
Mfg.	Major auto plant	28	Tools,Presses	>70
	Fluid-power equipment	25	Tools	>70

appropriate for low or medium power applications where good performance is not essential: voltage controllers generate undesirable harmonics, have generally poor power factors, and have relatively low efficiency.

Frequency Controlled: Two basic principles are used to adjust motor speed by varying the frequency of input power to the motor. *Cycloconverters*, which convert the line frequency and power in one step (Fig. 3), are used in low-speed applications such as cement-mill or ball-mill drives. Drawbacks of cycloconverters include an output frequency generally limited to about half the input frequency, generation of harmonics, and poor power factor. *Inverters* first rectify the line frequency to DC in a converter, which is followed by either an inductive or capacitive filter to produce a desired DC current or voltage, respectively. The DC power is then inverted to the desired frequency (Fig. 4). The complete inverter consists of a set of electronic switches and surrounding circuitry.

There are a variety of inverter types. In a voltage-source inverter (VSI) the line current is first converted to DC variable voltage and then inverted to variable frequency. The inverter produces a stepped-voltage wave form (Fig. 4). An important attribute of the VSI is that it can power motors undersized relative to the inverter or multiple motors since it operates independently of the load. It is best suited for low to medium power applications where the speed ratio is limited to about 10 to 1.

Another voltage-fed inverter is the pulsewidth-modulated (PWM) type, where the DC link voltage (following the rectifier) is held constant. The PWM inverter produces a chopped full voltage of varying width (Fig. 4). Motor

Table 2. Characteristics of electronic adjustable speed drives [5].

ASD Type, Power & Speed Range	Advantages	Disadvantages
DC-Drive < 5 MW, 0-100%	Simple control, good performance.	Require maintenance. Higher motor cost. Poor reliability.
Voltage Control < 25 kW, 20-100%	Simple and low cost.	Harmonics. Low torque and efficiency. Limited speed range.
Cycloconverter > 75 kW, < 40-50% of input supply frequency	Can operate down to zero speed. High torque capability.	Complex circuit design. Poor power factor at low speed.
Voltage Source Inverter < 750 kW, 10-200%	Good efficiency. Simple circuit design	No regenerative braking. Problems at low speed (< 10%)
Pulse Width Modulated < 750 kW, 100:1	Good power factor. Low distortion.	No regenerative braking. Slightly less efficient than VSI.
Current Source Inverter > 25 kW, 10-150%	Regenerative braking. Simple circuit design.	Poor power factor and poor performance at low speed.
Load Commutated Inverter > 500 kW	Regenerative braking. Simple circuit design.	Synchronous motor only. Poor power factor at low speed.
Slip Energy Recovery > 500 kW, 50-100% (Kramer) or 50-150% (Scherbius)	ASD power rating less than motor.	Use with wound rotor induction motor only.

inductance filters the input current and a nearly sinusoidal current is drawn by the motor. The PWM inverter has several advantages over the VSI, including good power factor and wider speed range, but efficiency is slightly lower. It can also be used with undersized or multiple motors.

In the current-source inverter (CSI), an inductive filter gives a constant current to the inverter which produces a stepped current waveform for the motor (Fig. 4). The CSI has a relatively simple and robust circuit design, but is somewhat bulky. It is operable over a fairly large speed range (15 to 1), but has poor performance at low speed and

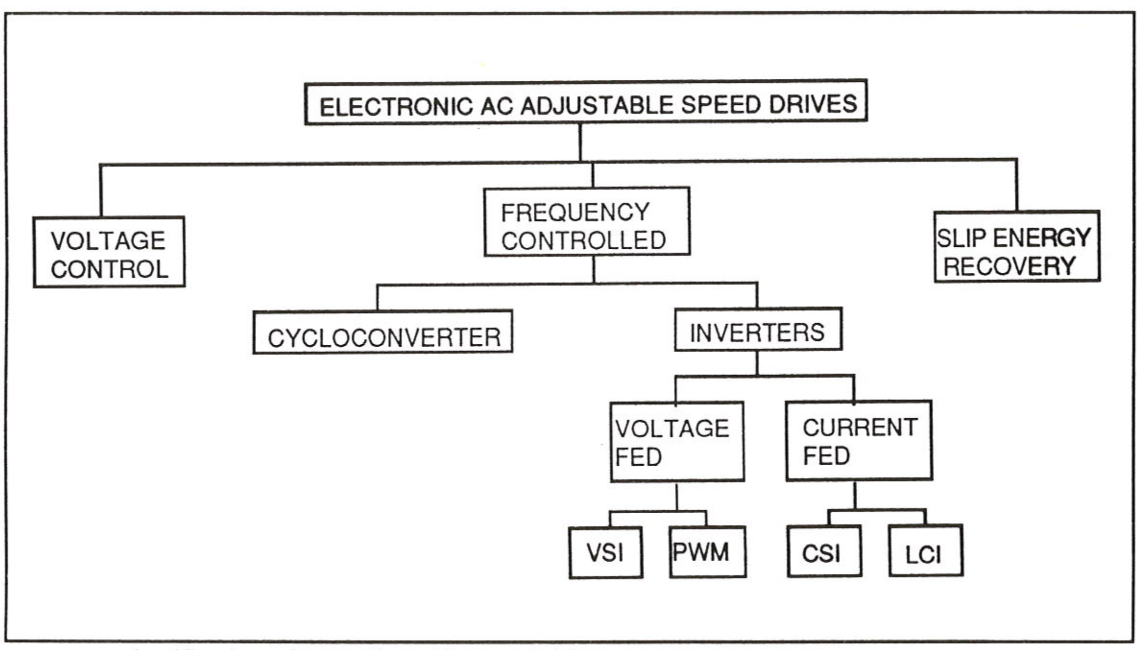


Figure 2. Classification of AC adjustable speed drives.

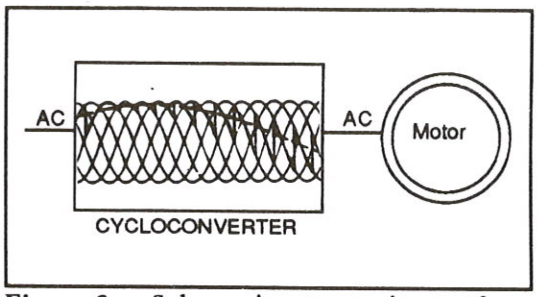


Figure 3. Schematic operation of a cycloconverter.

load. One advantage is that braking power can be fed back to the supply line to increase efficiency (i.e. regeneration).

The load-commutated inverter (LCI) is a CSI used in conjunction with a synchronous motor. Its advantage is that the synchronous motor provides the so called commutation of the electronic switches, resulting in a simple and inexpensive circuit design. The LCI has regeneration capability and is typically used for high power ratings (>750 kW).

Slip Energy Recovery: Slip energy recovery systems are used in high power applications with wound rotor induction motors typically involving pumping or blowing. Motor speed is varied by varying rotor resistance, a principle that has been known for some time. In static Kramer and static Scherbius drives the slip power is fed back to the supply line. The Scherbius drive has an advantage in that slip power can flow from the line back to the motor as well, which permits both sub- and super-synchronous operation, giving a wider speed range.

2. R&D

Research and development associated with ASDs is focussed in basically two areas. One is the development of power electronic switching devices. The other is the evolution of microelectronics, which has lead to development of new control strategies. There are a great variety of power electronic devices used in ASDs and under development. A few of the most common ones are discussed here.

Switching Devices [7,8]

Common thyristors have been used in medium and high power applications for many years. For use in inverters they require forced commutation (i.e. additional circuits are required to turn

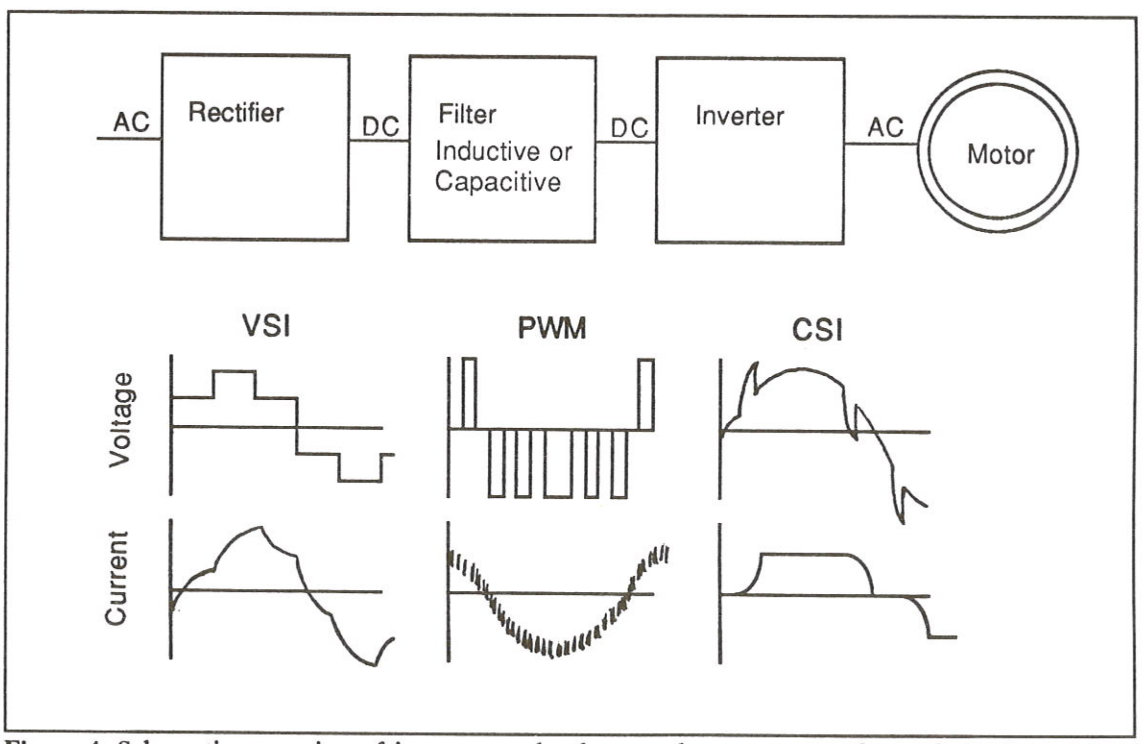


Figure 4. Schematic operation of inverters and voltage and current wave forms for voltage-source inverters (VSI), pulsewidth-modulated inverters (PWM), and current-source inverters (CSI).

them off). The commutation circuits can be avoided through use of the more expensive gate turn-off thyristors (GTO), resulting in a higher total efficiency. Thyristor research is aimed at improving characteristics such as switching times and operating range.

Transistors are used in ASDs up to as high as 400 kW in size and are superior to GTOs due to lower losses. Transistors are used where higher switching frequency is desired, e.g. in PWM inverters. Even higher frequencies are attainable with MOSFETs,[2] which leads to reduced motor losses. However, the use of MOSFETs leads to higher losses in the inverter, which offsets some of the efficiency gain in the motor. R&D efforts are focussed on increasing power handling capability, reducing losses (to permit still higher switching frequencies), and increasing reliability. Efforts are also being made to integrate power devices in a single package or module and reduce the size of the equipment by better packaging techniques.

Microelectronics & Control [9,10]

The evolution of microelectronics and large-scale integrated circuits is leading to the development of better control methods, thereby expanding the range of applications for ASDs. Vector or field-oriented-control ASDs can in most cases replace DC drives and provide comparable or better performance. Another interesting trend is the development of integrated control-power circuits. In such circuits the microprocessor controller is integrated into the same chip as the power electronics. Such developments seem to promise reduced production costs, which would be particularly important for low-power ASDs which are relatively costly today (see below).

3. TECHNICAL AND COST DATA

In considering an ASD for a particular application, a number of positive factors should be evaluated, including efficiency, reliability, and secondary benefits. The efficiencies of ASDs are high at full and partial load, but tend to drop at light load or torque. Full-load efficiency is typically above 90% and in large ASDs (>75 kW) it can reach 97% [5]. ASDs are relatively reliable and need little maintenance since they involve no moving parts, except for in many cases a cooling fan. ASDs can have a positive effect on surrounding equipment, e.g. improving lifetimes by smoother starting. Other secondary benefits to consider are less noise, less wear, better control, and possibilities for regeneration. Also inverters can be placed away from the motor they are driving if space is scarce or if the environment is problematic.

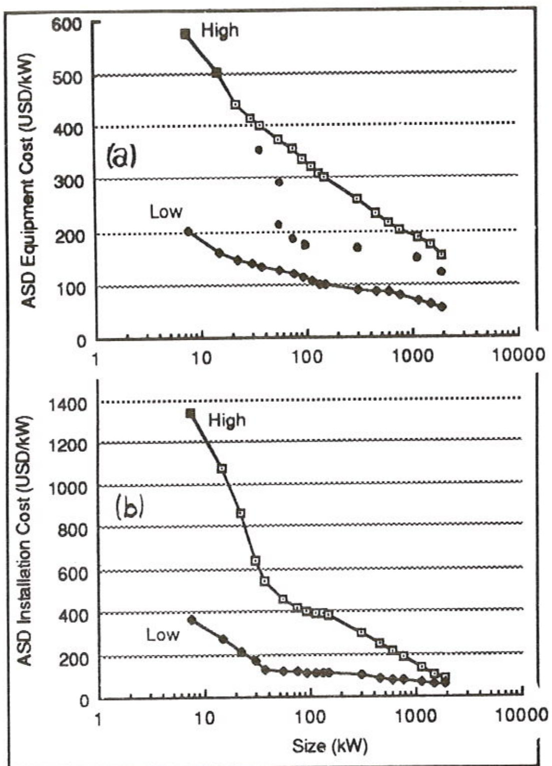


Figure 5. (a) Equipment and (b) installation cost estimates in the USA for AC adjustable speed drives for induction motors [11]. See endnote 3.

Potential problems with ASDs should also be considered. ASDs can affect the power quality on the supply line in several ways. ASDs in some configurations with low loads on the drive will produce a poor power factor. Another problem is the generation of

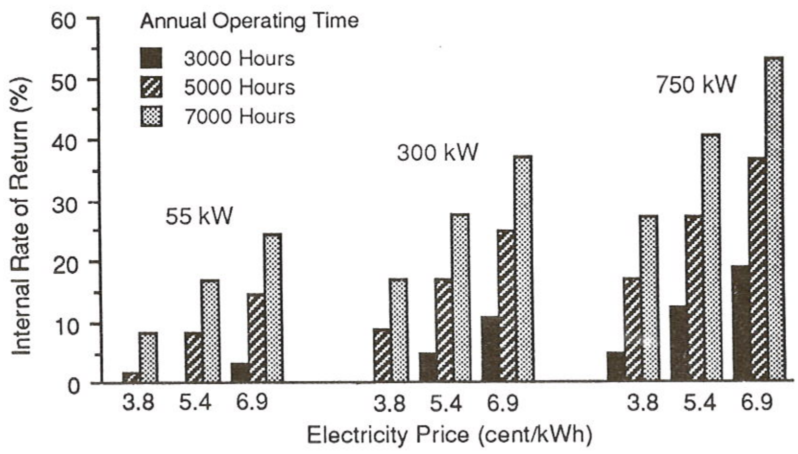


Figure 6. Internal rates of return on new investments in ASD-motor systems as a function of electricity price and annual operating hours. The investment costs are the average of those shown in Fig. 5b for each size. It is assumed that 30% of the electricity used by an existing constant-speed system would be saved as a result of the investment and that the ASD lasts for 10 years.

harmonics, which can cause distortion in the electrical distribution network and increased losses in motors and transformers. Power factor and harmonics problems can both be avoided through use of relatively low-cost filters [5]. Another problem to consider is electromagnetic interference to computers and telecommunications that could be caused by harmonics in the radio-frequency range. Proper use of shielding and grounding as well as filters can help deal with the latter problem [5].

Aside from the above positive and negative factors, the specifics of the application being considered help determine the appropriate ASD to use. (Refer to Table 2.) For applications below about 200 kW, VSI and PWM inverters are favored. VSIs have their best performance in the upper speed range, whereas PWM inverters have good performance also at low speeds. CSIs are favoured in applications from 200 kW and up, though PWM inverters with GTOs can also be used up to about 750 kW. In very high power applications LCI, cycloconverter, slip energy recovery drives, or CSIs are used depending on speed range and motor type.

For a large number of applications (e.g. pumps and blowers) exact control and short response time may not be critical. In these cases, a common drive control method for any ASD is to keep a constant voltage/frequency ratio at the input to the motor. DC drives have traditionally been used where higher performance is needed, but advanced ASD control strategies (e.g. vector or field oriented methods) can be used with AC motors to obtain comparable performance.

Prices of ASDs can vary over a wide range and are relatively sensitive to scale. Equipment costs for AC adjustable speed drives for induction motors sold in the USA are estimated to range from 150-550 $/kW for a 10-kW unit and from 75-190 $/kW for a 1000-kW unit (Fig. 5a).[3] Sample prices reported by Swedish vendors are shown for comparison. Estimated installation costs are significant, ranging from about 300-1200 $/kW for 10-kW systems down to 30-75 $/kW for 1000-kW units (Fig. 5b).

4. ILLUSTRATIVE ECONOMIC ANALYSIS

Since costs for ASDs can vary

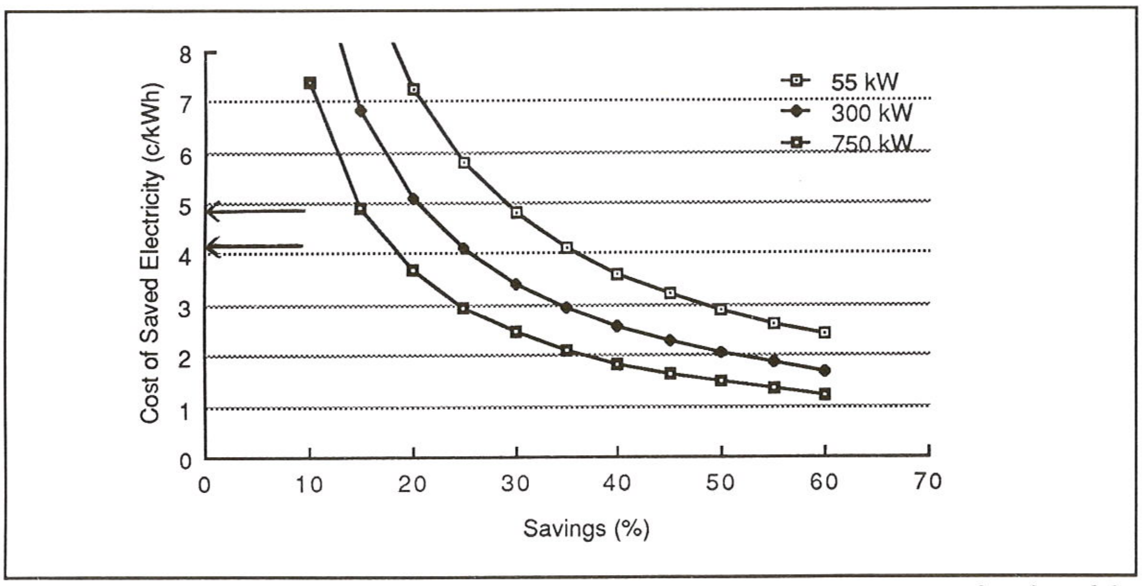


Figure 7. Cost of saved energy associated with investments in ASD-motor systems as a function of the percentage electricity saved relative to constant-speed operation. The investment costs assumed are the average of those shown in Fig. 5b for each size. No operating costs are included. We assume 5000 annual operating hours and a 10 year equipment life. A 6% discount rate is used [12]. The upper and lower arrows indicate the estimated cost of new central station baseload electricity supplied by coal-steam or advanced natural-gas-fired combined cycle plants, respectively [12].

substantially from one application to another (Fig. 5), it is difficult to draw specific conclusions regarding the economics of ASDs. However, some general trends can be identified. We present here results of simple economic calculations[4] using the *average* of the costs shown in Fig. 5. We only consider equipment, installation, and operating-electricity costs in our analysis. Other less direct potential benefits (better process control, quieter work environment, lower maintenance cost, etc.) are not considered. The calculations are made for retrofit applications. For new installations the choise of ASDs means that conventional pressure and flow controls would not be needed, which lowers the cost for the ASD system relative to an alternative conventional control system.

Industrial Perspective

Internal rates of return (IRR) are shown in Fig. 6 for new investments in three sizes of ASD systems as a function of the annual operating hours and electricity price (kept constant in real terms over the 10-year lifetime of the investment). Electricity savings of 30% are assumed compared to operating a fixed-speed system. Relatively high rates of return result for all three sizes shown in Fig. 6 regardless of the assumed electricity price, for 7000 operating hours per year. Large installations are particularly attractive--an IRR of over 50% is calculated in the best case. The IRR looks reasonably attractive in all cases involving 5000 or more operating hours and an electricity price of 5.4 cents/kWh or greater. With only 3000 operating hours, the smallest installation appears uneconomic for the given assumptions, even at the highest electricity price.

National Perspective

The cost-of-saved-electricity (CSE) associated with new investments in ASD-motor systems is shown in Fig. 7 as a function of the percentage electricity savings, assuming 5000 annual operating hours. Shown for comparison (two arrows) are estimated costs of new electricity supply for Sweden. With large ASD units, the CSE would be less than the cost of new supply with savings of as little as 15%. For the smallest size shown in Fig. 7, the breakeven would come with 30-35% savings. These results, taken together with the loss estimates in Table 1, suggest that there is a very large potential from a societal perspective for cost-effectively saving electricity in industry using ASDs.

5. REFERENCES

1. L.J. Nilsson and E.D. Larson, "Electric Motors," *Technology Menu*, Environmental and Energy Systems Studies, University of Lund, Lund, Sweden, 1989.

2. J.K. Armintor and D.P. Connors, "Pumping Applications in the Petroleum and Chemical Industries," *IEEE Transactions on Industry Applications*, 1A-23(1), 1987, pp. 37-48.

3. L. Silfverberg, "The Invention that can save as much energy as a nuclear power plant can produce," *Energimagasinet*, No. 2, 1988. (in Swedish)

4. W. Murgatroyd and B.C. Wilkins, "The Efficiency of Electric Motive Power in Industry," *Energy*, Vol. 1, 1976, pp. 337-345.

5. A. de Almeida and S. Greenberg, "Applications of Adjustable Speed Drives for Electric Motors," University Wide Energy Research Group, UER 232, University of California Berkeley, 1989. (In press)

6. B.K. Bose, *Adjustable Speed AC Drive Systems*, IEEE Press, New York, 1981.

7. Y. Ikeda and T. Yatsuo, "Recent Progress in Power Electronics Devices," *Hitachi Review*, 36(1), 1987.

8. J. Baab and W. Bresch, "En krafthalvledare for varje tillfalle (One power semiconductor for each occasion)," *Elteknik med aktuell elektronik*, No. 4, 1987. (in Swedish).

9. W. Leonhard, "Microcomputer Control of High Dynamic Performance AC-Drives: A Survey," *Automatica*, 22(1), 1986.

10. B.K. Bose, "Motion Control Technology--Present and Future," *IEEE Trans. on Ind. Appl.*, 1A-21(6), 1985.

11. Power Electronics Applications Center, *Adjustable Speed Drive Directory*, 2nd edition, PEAC, Knoxville, Tennessee, 1987.

12. "Economic Methods and Assumptions," *Technology Menu*, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

NOTES

1. While such estimates are often made, relatively little documentation exists on measured savings. To improve the understanding of the potential for conservation in pumping systems, greater efforts are required to document actual savings.

2. Metal-Oxide-Semiconductor Field-Effect Transistor.

3. Variations in the costs shown in Fig. 5 for a given size unit can arise because of different requirements from one application to another. The equipment cost estimate in Fig. 5a includes plus or minus 50% of the average of a large number of commercial units. Sample prices reported by Swedish vendors are also shown. In Fig. 5b the upper estimate assumes a high labor rate and installation of bypass switching, panel ammeters, speed indicators, speed controllers and equipment related to remote placement of the drive relative to the motor. The lower curve excludes the installation cost of the extra equipment and assumes a low labor rate.

4. See [12] for descriptions of the economic indicators used in this section.

Vol. 1: Movement of Material **89-8-8**

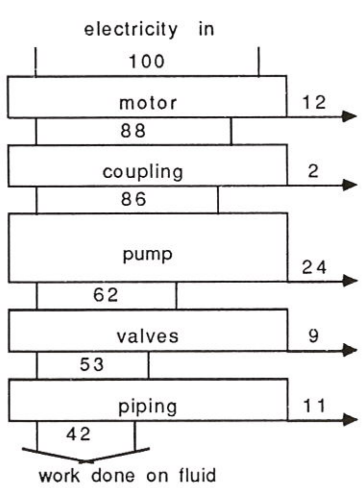
***Component*: Pumps**

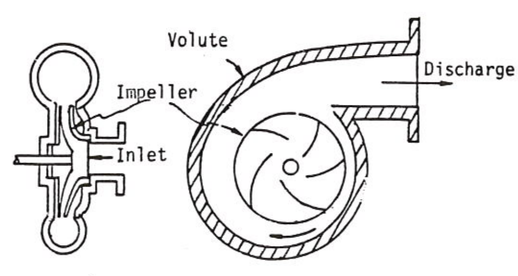
by
Lars J. Nilsson, Eric D. Larson
Environmental and Energy System Studies
University of Lund, Lund, Sweden

CONTENTS

SUMMARY

A significant amount of electricity is used in pumping. This Menu entry is focussed on the pump, which can account for a significant portion of the losses in a pumping system (see below left). Different pump types are briefly described. Most of the discussion focusses on the operating and performance characteristics of centrifugal pumps (see below right), which account for most of the electricity used by all pumps. Efficiencies of commercially-available centrifugal pumps are compared and the potential for efficiency improvements are discussed. Efficiency versus cost comparisons among pumps are difficult to make because of, among other factors, the way pumps are marketed. Based on a limited sample, it appears that for comparable-duty pumps, higher efficiency is not necessarily associated with a higher retail price.





1. TECHNOLOGY

Introduction

An estimated 15% of global electricity consumption is accounted for by pumps and fans. For the electricity intensive industries in Sweden,[1] which consume about 33 TWh/yr (1/4 of total electricity use), the fraction has been estimated to be 30% [1]. Small improvements in average pump efficiency, therefore, would lead to significant electricity savings in Sweden alone.

Pumps, the focus of this Menu entry, are components of pumping systems, which also include motors, drives, piping and valves. A significant fraction of the electricity input to a pumping system is lost at the pump itself (Fig. 1). Thus improving pump efficiency can improve overall pumping system efficiency.

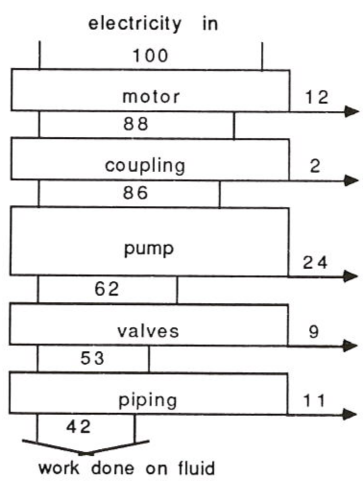


Figure 1. Energy balance for a typical pumping system [3]. Numbers are in percent.

The many uses of pumps have led to a great variety of pump designs. However, almost all pumps fall into two basic classes based on their operating principles [2]: dynamic pumps and displacement pumps. The majority of dynamic pumps are centrifugal pumps. Displacement pumps include rotary and reciprocating pumps.

It is estimated that about 3/4 of all pumps in the United States are centrifugal pumps and the rest are primarily rotary pumps [2]. A similar distribution probably exists in Sweden. Since centrifugal pumps are favoured in high power applications it is estimated that 90% (or more) of all power consumed by pumping is used to drive centrifugal pumps. Centrifugal pump characteristics and performance are, therefore, the focus of this Menu entry.

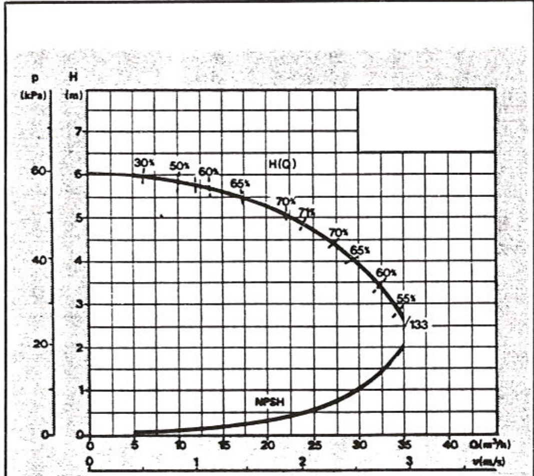


Figure 2. Typical pump curve taken from a vendor catalog. The curve marked H(Q) shows pump head and efficiency as a function of flow. The lower curve shows the net positive suction head required to operate the pump.

Centrifugal Pumps

Centrifugal pumps are characterized by head-flow curves, which indicate the combinations of head and flow at which the pump can operate (Fig. 2). Also shown on such curves is pump efficiency at each operating point.[2] At one combination of head and flow, efficiency reaches a maximum. It decreases away from this point, substantially in some cases. Head-flow curves also typically show the net positive suction head (NPSH), the pressure required at the pump inlet for proper operation.

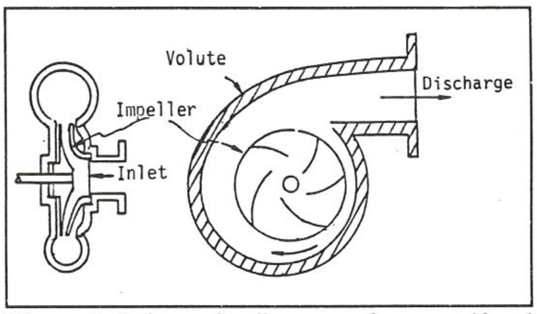


Figure 3. Schematic diagram of a centrifugal pump.

In a centrifugal pump the fluid enters a set of rotating vanes called an impeller. The fluid is accelerated and guided until it is discharged at higher velocity at the periphery of the impeller. The velocity of the fluid is then reduced in the volute (Fig. 3), and the kinetic energy is converted into a higher pressure level. The fluid is finally discharged at the outlet of the pump with the desired head and flow.

A useful parameter characterizing centrifugal pumps is the specific speed, N_s, which combines flow, head and rotational speed into a single number:

$$N_s = (N^*Q^{0.5})/H^{0.75}$$

where N is the rotational speed (r/min), Q is the flow (m^3/s), and H is the pressure head (m).[3] Different centrifugal pump impeller designs--axial, mixed, or radial flow--are characterized by their specific speed. Axial flow pumps typically have high specific speeds and radial flow pumps have low specific speeds (Fig. 4). Different impeller designs are used depending on the head and flow conditions (Fig. 5).

Energy losses during pumping lead to heating of the fluid and the pump itself and to a pump efficiency below 100%. Fig. 6 shows a typical energy-balance for a centrifugal pump. Most of the losses (3/4 or more) are accounted for by impeller friction, hydraulic losses and internal leakage [4]. Impeller-friction losses arise through friction between the outside of the impeller and the pumped fluid. The amount of the hydraulic losses (due to turbulence and other factors) is a measure of how well the impeller and the casing, i.e. the hydraulic "paths," are designed. Internal leakage refers to fluid that leaks back and has to be "re-pumped".

The fraction of input energy lost is closely linked to the fraction of the pumped fluid which interacts directly with the casing and the impeller. Thus, other factors being equal, relative losses are larger the smaller the pump. The specific speed has the same value for geometrically similar pumps (if operated at the same rotational speed) and were it not for this effect of scale, their relative losses would also be the same.

Fig. 7 shows pump efficiency as a function of specific speed and pump capacity, based on an average of a large number of commercial centrifugal pumps [5]. For a fixed specific speed,

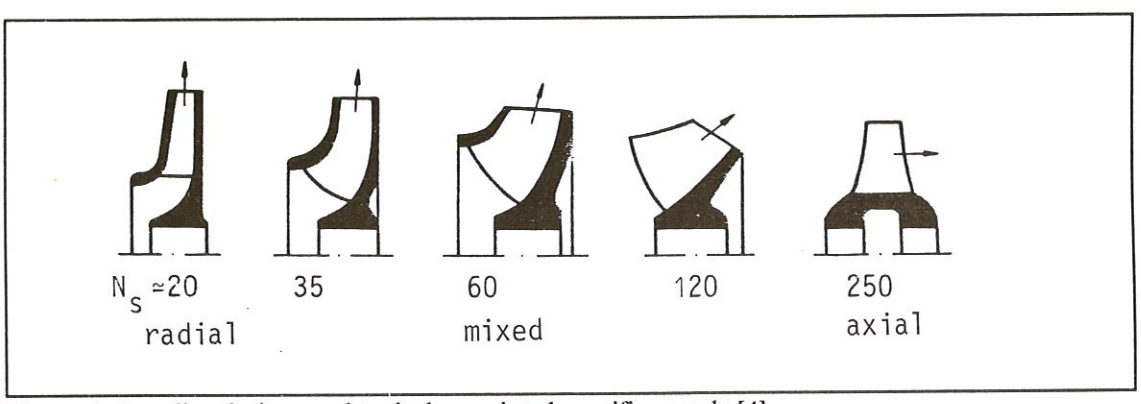


Figure 4. Impeller designs and typical associated specific speeds [4].

efficiencies rise with increasing flow. For fixed flow, efficiency falls with decreasing specific speed, with the sharpest drop occurring in the lower specific speed range. The efficiencies indicated are for single stage pumps and would be somewhat lower for multi-stage pumps [6].

Another important inefficiency in many pumps is oversizing. As indicated in Fig. 2, pump efficiency can fall by half or more if it is operated at head and flow conditions away from those that produce optimum efficiency.

2. R&D

Traditionally, high efficiency is not a major consumer priority in pumps [2]. Reliability is perhaps the most important criterion. Other factors are price, delivery time and a tendency to stick to one supplier to minimize spare parts inventories and simplify maintenance. For manufacturers, therefore, the R&D emphasis has been on lowering production costs. Efficiency has not been a priority, particularly with smaller pumps [7], which use relatively small amounts of electricity. A greater effort to improve efficiency has been made with larger pumps (> 15kW), since in these sizes small changes in efficiency can often have a

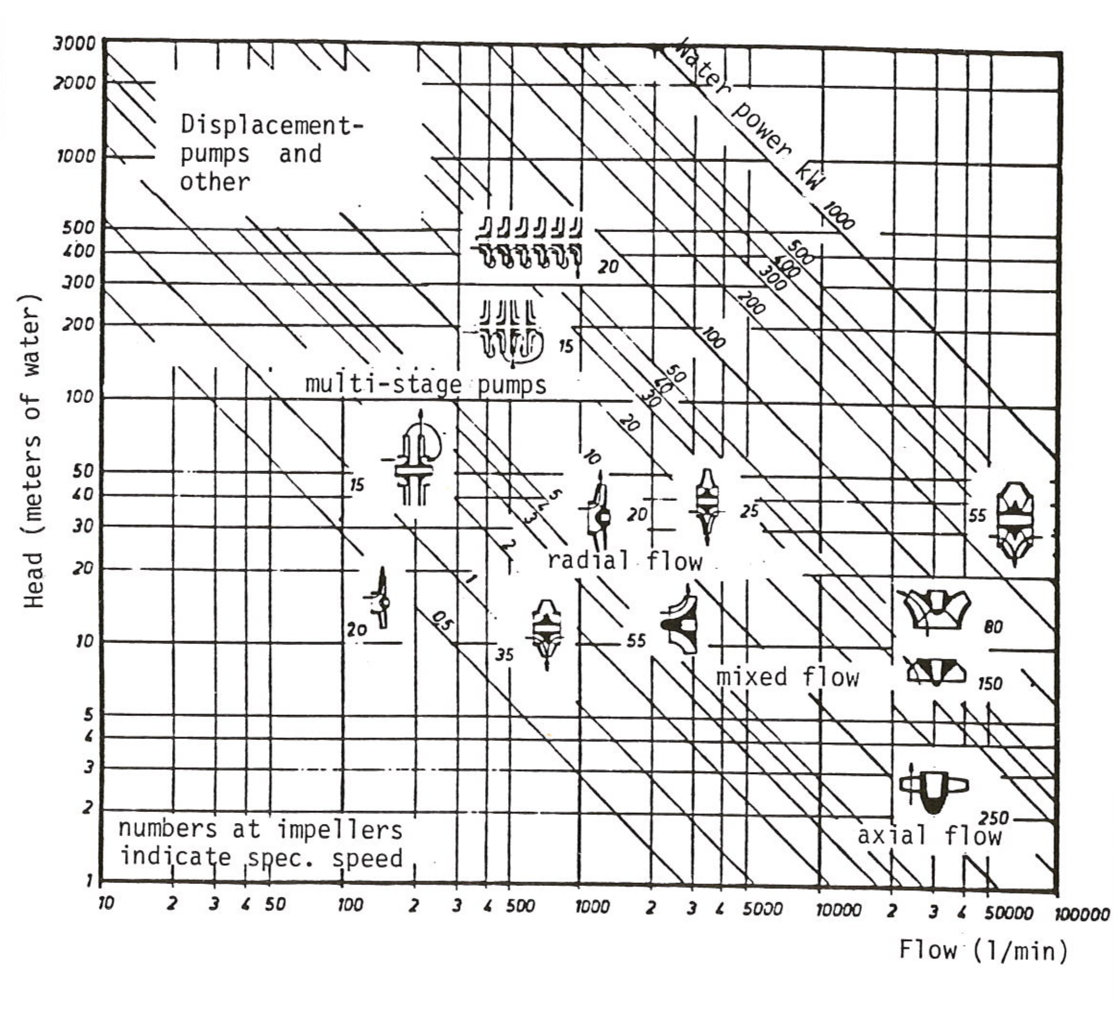


Figure 5. Ranges of application of different types of impellers when pumping water-like fluids [4].

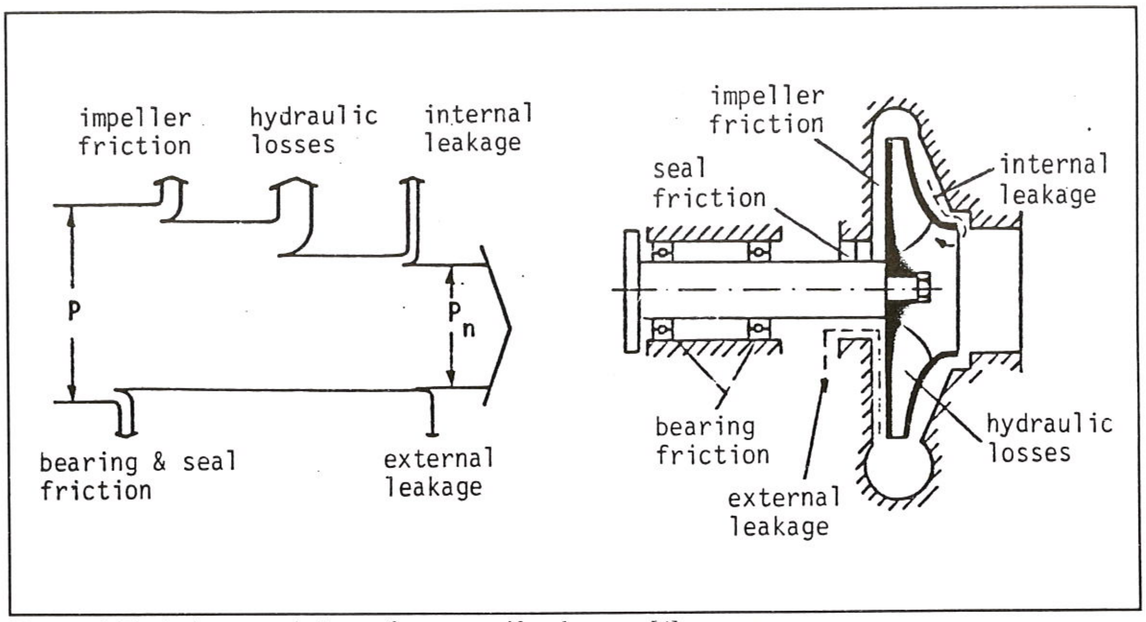


Figure 6. Typical energy balance for a centrifugal pump [4].

more obvious effect on a customer's system-wide operating costs.

While efficiency has not been a major focus of R&D, there appears to be potential for improving average pump efficiencies by (1) more widespread implementation of known methods for producing more efficient pumps and (2) fundamental improvements in materials and fluid mechanics design.

A sampling of Swedish pumps suggests that more efficient pumps could be produced, were there incentive to do so. Comparable-duty pumps are available with wide variations in efficiency (Figs. 8a-c). (In some cases, the same manufacturer produces both the efficient and inefficient models.) The solid line in Fig. 8, reproduced from Fig. 7, represents what is achievable with good design practice. The broken line represents, approximately, the average efficiency of the Swedish sample. Thus, it appears that the average efficiency of new pumps could be improved (at little or not cost, as discussed below) by increased production of the more efficient models.

The application of fundamental developments in pump technology could also lead to efficiency improvements. For example, there is ongoing effort to increase the average head per stage, which reduces the number of stages required in a given application. This typically yields higher efficiency and facilitates use of shorter shafts, increasing reliability. Smoother impeller blades and casing surfaces reduce friction losses. Plastics might become important as a substitute for metal in this context. Modifying the fluid boundary layers, for example by more careful contouring and surface smoothing, can reduce hydraulic losses. Possible efficiency improvements of 10-50% have been suggested for low specific speed pumps through such modifications [8].

3. TECHNICAL AND COST DATA

It is difficult to draw specific conclusions regarding pump efficiencies. As discussed above, it appears that pumps providing the same pumping duty can have widely different efficiencies. Specifically, for example, the two 28 l/s (7.5 kW) pumps indicated in Fig. 8b are produced by the same manufacturer and have essentially identical characteristics, except that one has an optimum efficiency 8 percentage points higher. Similar differences are discernable between pumps from different manufacturers. As another example, in Fig. 8c, one pump has an optimum efficiency about 10 percentage points higher than the average, indicating that even small pumps can be made relatively efficient.

While specific conclusions are difficult to draw, the following general observations can be made regarding pump efficiencies. The average efficiencies of new pumps are perhaps 3-5 percentage points higher than those in the existing stock[4]. Figures 8a-c suggest that the best new pumps are 3 to 10 percentage points better than the average for new pumps. There appears to be limited interest or effort to increase pump efficiencies in the future, in part because buyers may not fully recognize the importance of the energy costs for operating a pump. An increase in energy prices may induce pressure on manufacturers to produce more efficient pumps. The spread in efficiencies for comparable-duty pumps suggests that improvements may be relatively easily achieved, particularly in small sizes.

Because of the wide variety of pumps, pump applications, and pump marketing methods, it is difficult to make meaningful quantitative assessments of pump costs versus efficiency. Some information supplied by manufactureres suggests that there is no (or even a negative) correlation between efficiency and list price. For example, the two pumps indicated in Fig. 8b which are nearly identical except for an 8 percentage point efficiency difference, have a list-price difference of$15 ($1390 vs. $1375). The extra cost for the more efficient model would be recouped quickly in operating cost savings. In another example, two nearly-identical pumps (7.5 kW with optimum efficiency at 15 l/s flow, 30 m head) from different manufacturers have an 8 percentage point efficiency difference. The list price for the more efficient one is $1340 and that for the less efficient one is $2000.

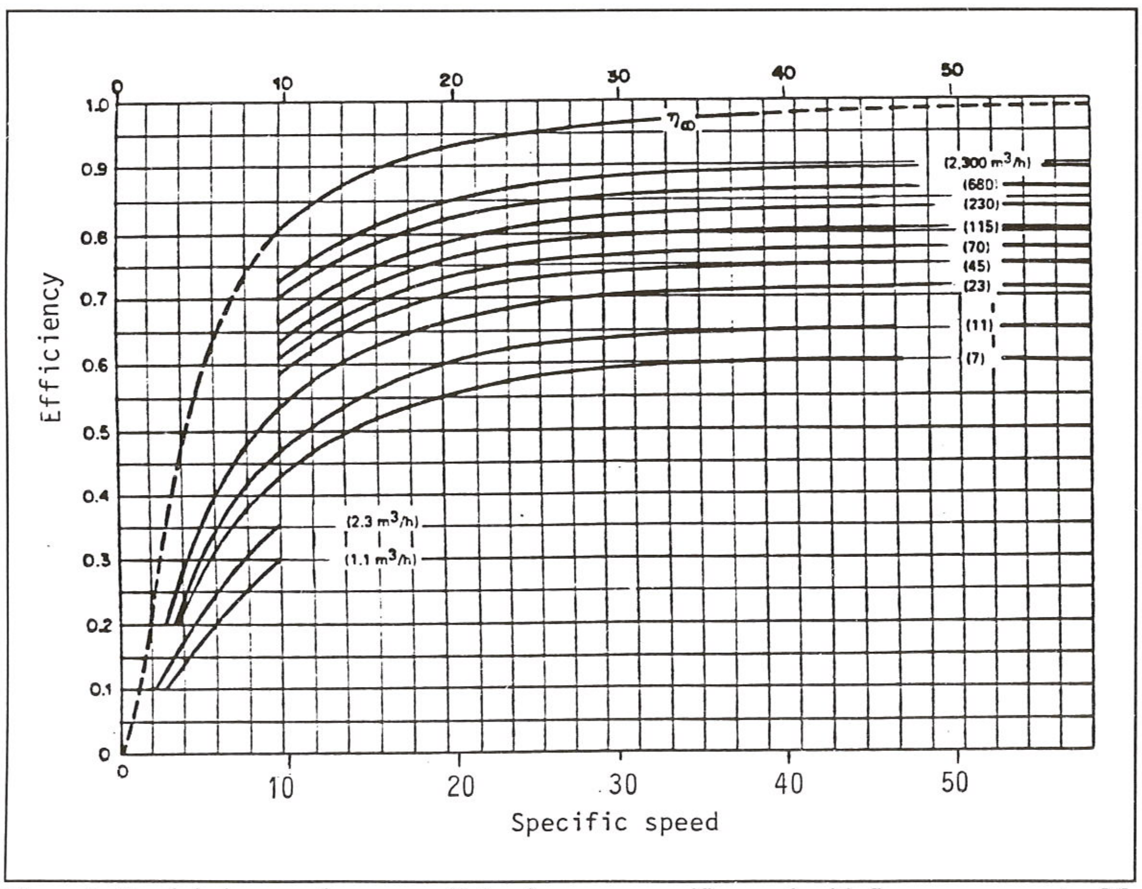


Figure 7. Good-design-practice pump efficiencies versus specific speed, with flow as a parameter [5].

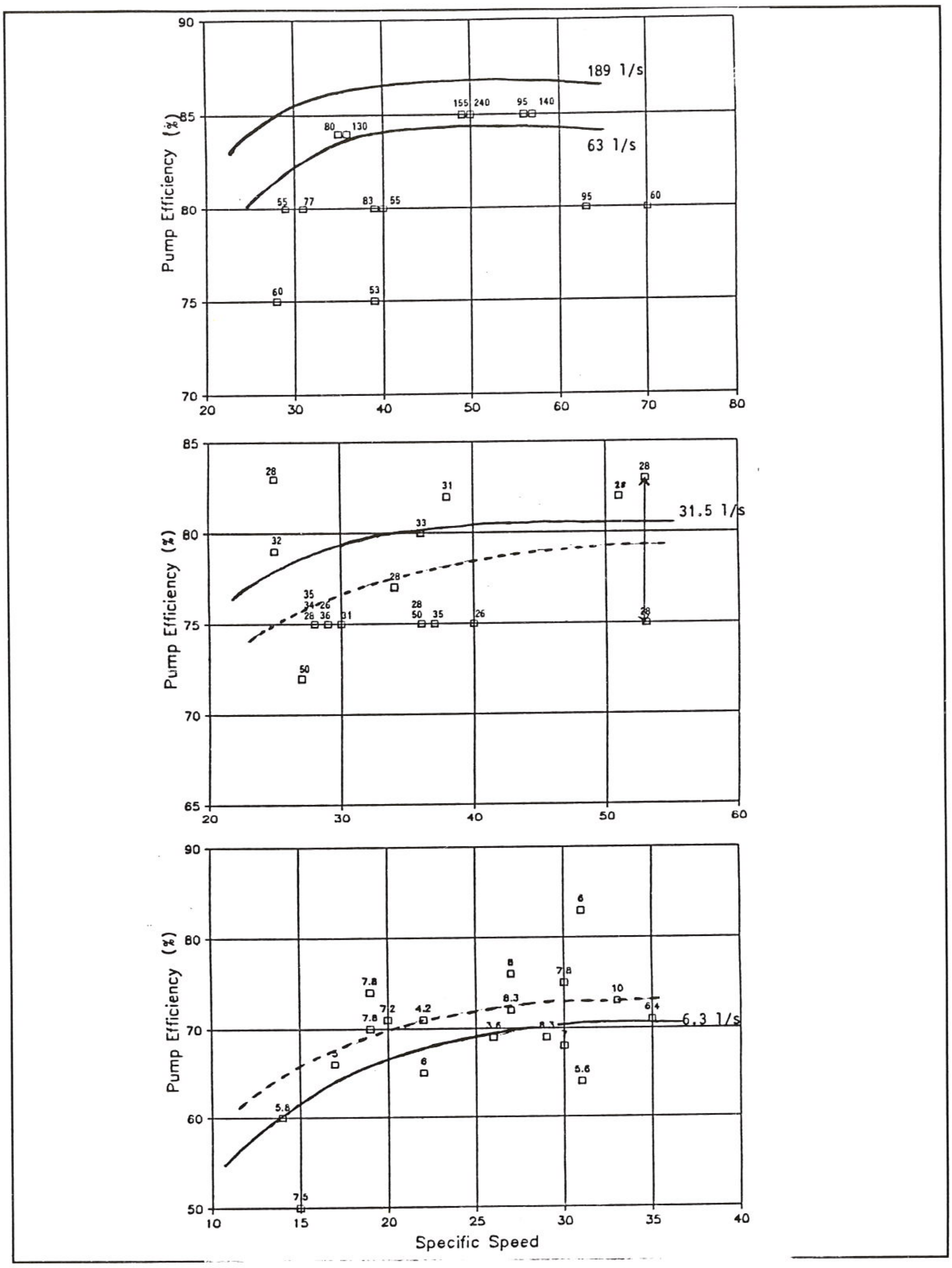


Figure 8a-c. Design-point pump efficiencies versus specific speed for a sampling of pumps sold in Sweden. Solid lines represent good design practice (from Fig. 6). The broken lines are the approximate averages for the samplings. The small numbers indicate design-point flow (liters/sec).

5. REFERENCES

1. Statens Offentliga Utredningar, *Electricity Conservation in the 1990's*, 87:68, Stockholm, 1987. (in Swedish)

2. Assistant Secretary for Conservation and Solar Energy, *Classification and Evaluation of Electric Motors and Pumps*, DOE/CS-0147, US Dep. of Energy, Washington DC, 1980.

3. T.B. Johansson and P. Steen, *Perspectives on Energy*, Liber Forlag, Stockholm, 1985. (in Swedish)

4. B-A. Gustafson, E. Nilsson, *Applied Fluid Mechanics*, Chalmers Institute of Technology, Gothenburg, Sweden, 1985. (In Swedish)

5. I.J. Karassik et al, *Pump Handbook*, McGraw-Hill, New York, 1976.

6. I.J. Karassik, "Centrifugal Pump Clinic, Part 110", *World Pumps*, July 1985.

7. D. Kristiansson, Grundfos AB, Gothenburg, Sweden, personal communication, 1988.

8. I.J. Karassik, F. Hirschfeldt, "The Centrifugal Pump of Tomorrow," *Mechanical Engineering*, May 1982.

NOTES

1. Electricity-intensive industries are defined as pulp & paper, sawmills & board (SNI 34111 + 34112, 33111, 33119+34113), iron & steel (SNI 37101), mining (SNI 2), non-ferrous metals (SNI 3720-37204), ferro-alloys (SNI 37102) and basic chemicals (SNI 351).

2. Pump efficiency is defined as the power transmitted to the fluid being pumped divided by the power input to the shaft of the pump.

3. The specific speed was originally introduced as a purely dimensionless number. The definition given here, which is now commonly used, is not strictly dimensionless.

4. This assumes that the existing stock is about 10 years old, had 2-3 percentage points lower efficiency when new than today's average for new pumps, and that wear has reduced efficiency by another 1-2 percentage points.

Vol. 1: Movement of Material **89-8-8**

Component: **FANS**

by
J. Barrie Graham
Graham Consultants
Santa Fe, New Mexico, USA

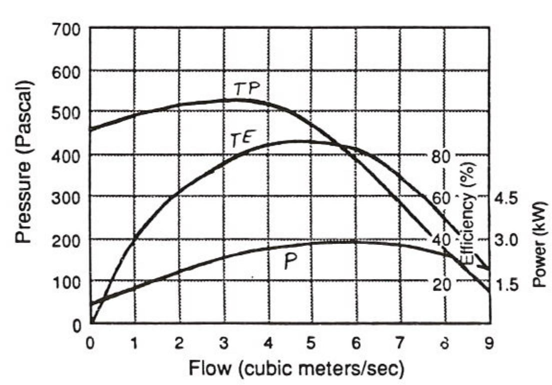
CONTENTS

SUMMARY

Fan inefficiencies account for an important fraction of the electricity input to air handling systems, and such systems are major electricity users, e.g., accounting for 30-50% of electricity use in typical commercial-sector buildings. The design and performance of a variety of fan types are discussed here. Fan efficiencies cover a wide range (table below), and in many applications one of several different fan types could be used. For example, backwardly-curved (BC) fans can be used in most HVAC applications where much less efficient forwardly-curved (FC) fans are currently used. In a new application, the cost differential would be small and expected electricity cost savings would often justify the added cost for the BC fan. How an installed fan is operated is also important in determining its energy costs, since fan efficiency decreases relatively rapidly from its maximum as operating conditions change (figure below). Thus, while the fan is a relatively mature technology, there is nevertheless a potential for reducing fan electricity consumption through greater use of the inherently more efficient fan designs and by insuring that fans operate in the region of peak efficiency whenever possible.

Maximum total efficiency estimates for different fan designs.

	Peak Efficiency Range
Centrifugal Fans	
Airfoil, backwardly curved/inclined	79-83
Modified radial	72-79
Radial	69-7
Pressure blower	58-68
Forwardly curved	60-65
Axial Fans	
Vaneaxial	78-85
Tubeaxial	67-72
Propeller	45-50



1. TECHNOLOGY

Introduction

Fans are major energy users within air handling systems in the industrial, commercial and residential sectors. They typically account for some 20 to 25% of the electrical energy input to such systems (Fig. 1.) And air handling systems are major electricity consumers: electricity audits in the commercial/residential sector in the USA indicate that they can easily account for 30 to 50% of the total electricity used in buildings [1]. While the fan is a relatively mature technology, significant reductions in fan electricity use are still possible. Energy aspects of the most common fan designs are discussed here. For discussion of complete air handling systems see [1].

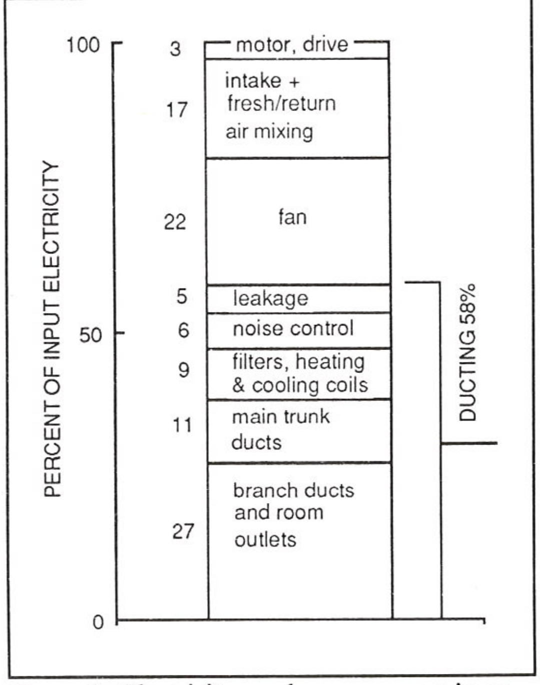


Figure 1. Electricity use by component in an air handling system. This estimate is for an air conditioning system for a typical multi-level commercial office building in the USA.

Basic Fan Design

Fans generate a pressure to move air (or other gases) against a resistance caused by ducts, dampers, or other components in a fan system [1,2]. The fan rotor receives energy from a rotating shaft and transmits it to the air. The energy appears in the air downstream of the fan in part as velocity pressure and in part as static pressure. The ratio of static to velocity pressure varies for different fan designs. Fans are typically characterized by the algebraic sum of the two pressures: the total pressure. Parts of the fan other than the rotor, such as the housing, straightening vanes, and diffusers, influence the ratio of velocity and static pressure at the outlet, but do add not energy to the airflow [3].

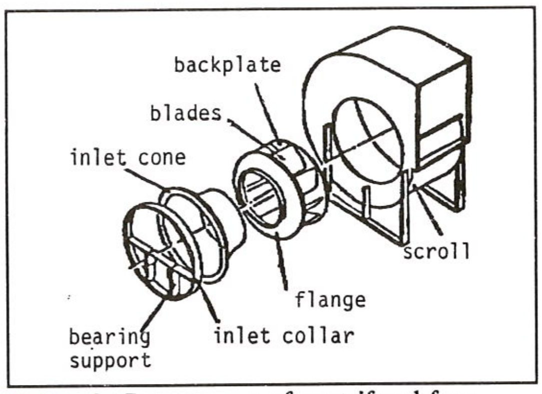


Figure 2. Components of centrifugal fans.

The general construction of centrifugal and axial fans and customary nomenclature are shown in Fig. 2 and Fig. 3. Both types of fans are enclosed in a housing to contain the air flow and in most cases to convert some of the velocity pressure to static pressure. The housing is designed to accomplish this conversion with the least loss [4].[1]

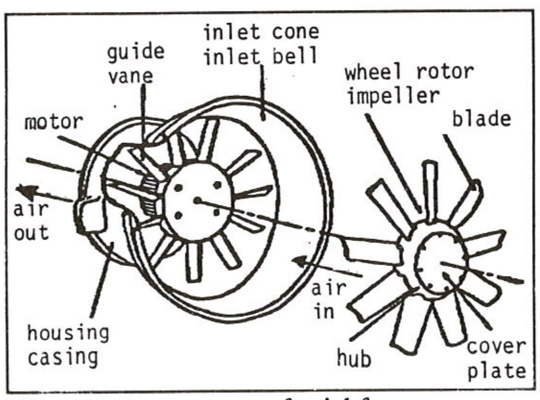


Figure 3. Components of axial fans.

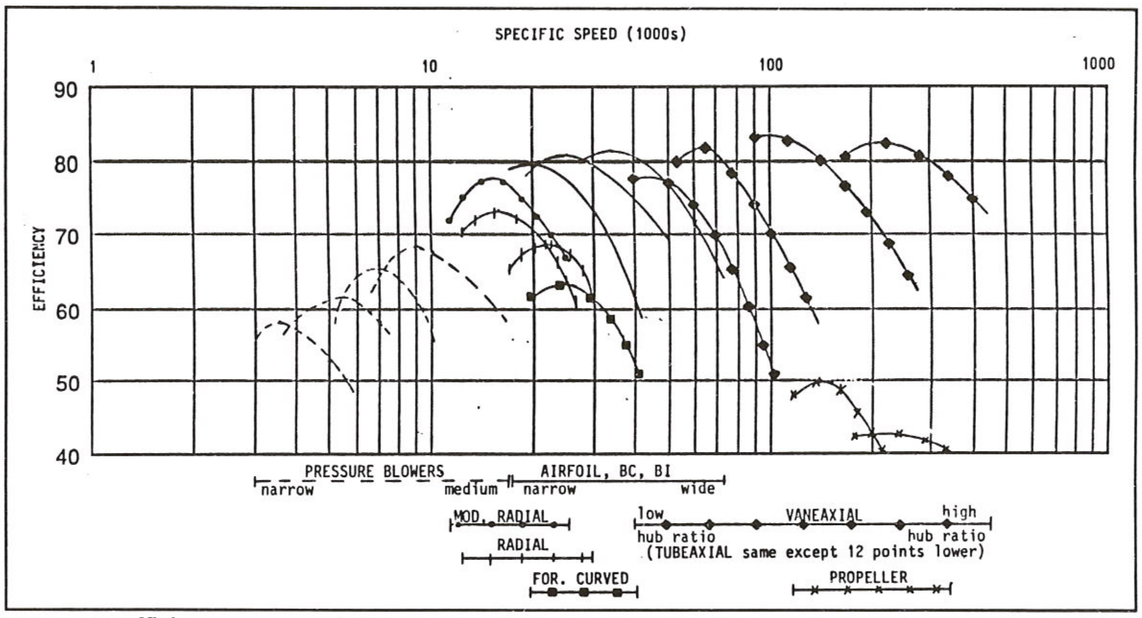


Figure 4. Efficiency ranges of different fan designs versus specific speed.

The major design parameters for all fans are the wheel diameter and blade shape, width and length. For centrifugal fans, an increase in blade width and decrease in blade length will increase the amount of air handled by the wheel, but since the centrifugal effect of the blades on the air is decreased, the pressure developed by the fan is lower. Lower flow rates and higher pressure result with narrower, longer blades. For axial fans, larger hubs with relatively short blades are used for handling smaller quantities of air than axial fans with small hubs. Since the bulk of the air passes through the axial wheel at a greater average radius if the wheel hub is large, the blades do more work on the air, i.e., they produce a higher outlet pressure.

Fan efficiencies can vary widely depending on the specific design. A convenient parameter for making efficiency comparisons is the specific speed, N_s, defined as

$$N_s = N*Q^{0.5}*TP^{-0.75}$$

where N is the rotational speed, Q is the volume flow, and TP is the total pressure. Figure 4 shows that fan efficiencies can vary by up to a factor of 2 for the same specific speed.[2]

The design of other fan components also affects fan performance. Although spiral housing designs are used on well designed fans for optimal energy efficiency, some fans on the market use square housings, which incur an efficiency penalty of 10 points or more. The proper design of guide vanes is important in fans that use these, as discussed in Section 3.

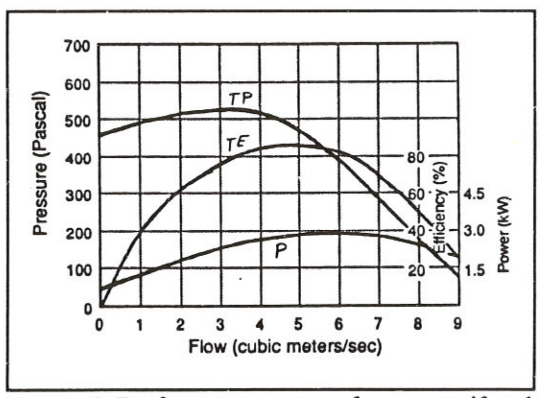


Figure 5. Performance curves for a centrifugal fan. TP is total pressure at the fan outlet. P is input power. Total efficiency (TE) is the ratio of fan power output to fan power input [9].

Fan Performance Curves

A large variety of fans with varying performance characteristics have been developed to match such diverse fan applications as roof-top ventilation and forced-draft air supply. Typical constant-speed performance curves for centrifugal and axial fans are shown in Fig. 5 and Fig. 6, respectively. These show the dependence on volume flow rate of the total pressure (TP), total efficiency (TE), and input power requirement (P). Note that total efficiency falls quite steeply on either side of a small flow rate interval. Significant energy losses are associated with operation away from this region.

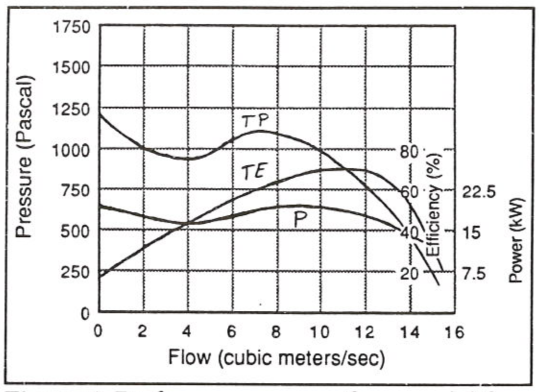


Figure 6. Performance curves for an axial fan. (See Fig. 5 caption for definitions.)

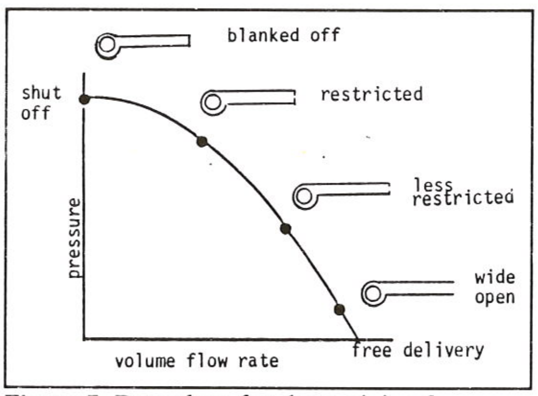


Figure 7. Procedure for determining fan performance curves.

Performance curves for a particular fan are developed by making a series of pressure versus flow measurements at a given fan speed. A simple orifice plate is used to change the flow conditions (Fig. 7). From measurements made with different orifice openings, the performance curves are constructed.

In field applications, a good understanding of the relationship between the fan and the system is essential in distinguishing fan from system problems. In the above example, the orifice plate is an analog for the system. The fan operates along the same curve even if the system changes, i.e., if the orifice size changes. A measurement of the volume flow will locate the exact operating point on the fan curve.

Fan performance curves (Figs. 5 and 6) refer to operation of a given size fan at a given speed. The performance of geometrically similar fans of other sizes or the same fan operating at other speeds can be calculated from a simple set of fan laws:

$$TE_c = TE_b$$

$$Q_c = Q_b * (D_c/D_b)^3 * (N_c/N_b)$$

$$TP_c = TP_b * (D_c/D_b)^2 * (N_c/N_b)^2 * (d_c/d_b)$$

$$P_c = P_b * (D_c/D_b)^5 * (N_c/N_b)^3 * (d_c/d_b)$$

where TE is total efficiency, Q is flow rate, D is wheel diameter, N is fan rotational speed, TP is total pressure, P is input power, d is gas density, and the subscripts b and c refer to base and calculated quantities, respectively.

The fan laws can be useful in a variety of ways.[3] For example, the characteristics of a given fan operating at a rotational speed other than its rated one can be calculated as illustrated in Fig. 8. Using data from a point on the base curve, e.g., D, the corresponding performance at a different rotational speed is calculated as point E. Similar calculations are carried out for a number of points along the base fan curve, which results in the calculated curve. If a series of speed curves is then calculated and points corresponding to the same original base-curve point are joined, they will form exponential curves identified as "system curves," as shown in Fig. 8. Such system curves can be used to predict the effect of a change in fan speed on a given system [1].

The fan laws must be used to determine the full effect of changes in

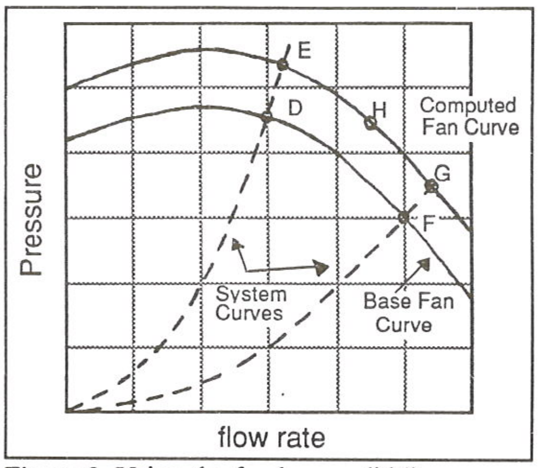


Figure 8. Using the fan laws: solid lines are for fan operation at 2 speeds. Dashed lines are system curves constructed using the fan laws and the base-curve fan characteristics.

fan performance. For example, assume a user wants to increase the flow rate in an existing system by a relatively small 8% by increasing fan speed. According to the fan laws this requires a 17% increase in pressure and a 28% increase in power! This illustrates that where energy conservation is important, every change in operating point must be carefully examined.

2. R&D

Fans can be considered a relatively mature technology: the best efficiencies today (80-85%) are high for turbomachinery of this type and are essentially unchanged from those for designs that have been available for the last decade. From the vendors perspective, these fan designs are economically optimal.

Fans are fabricated of easily formed sections and include certain flow passages that are abruptly changing, some sharp edges, and other design imperfections imparted during the fan fabrication process. By correcting such imperfections, small improvements in fan efficiency (1 or 2 efficiency points) could be achieved. It is highly unlikely that such efforts will be made by the manufacturing companies, however, as the fabrication costs would rise significantly (perhaps by a factor of ten or more), and there is currently no market incentive to produce such more efficient, but more costly, fans. There is pressure to produce cheaper fans. As a result, most fan companies have severely reduced, or eliminated, their efficiency-related R&D work. R&D work today is aimed largely at reducing fabrication costs.

The previous remarks refer to the aerodynamic design of fans only. There will be some important improvements in speed control methods for fans which will have a great influence on the operating efficiency of fan systems. The reader is referred to [1] and [5] for more discussion along these lines. There will also be some interesting work in the stress and vibration areas of fan design which will result in structurally better, but no more efficient, fans.

Although improvements in fan efficiencies are not likely to be forthcoming without a much greater market demand, the efficiency of fan electricity use can still be improved substantially in new and retrofit applications. Fan efficiences in actual installations seldom achieve the relatively high efficiencies quoted by manufacturers due to a variety of reasons, including the fact that rated efficiencies are measured under ideal laboratory conditions which are not often reproduced in practice. The discrepancies between rated and practical efficiencies present opportunities for making substantial efficiency improvements. Accurate field measurements are needed to estimate the full potential savings in any given case and to identify the specific modifications that are required.

3. TECHNICAL AND COST DATA

Technical Performance

A variety of fan designs are used depending on the application. In many applications, one of several designs could be used. The specific choice involves many considerations and has an important influence on energy use, since efficiencies for different fans can vary significantly. Peak efficiencies of centrifugal and axial fans range from 60-83% and 45-85%, respectively, depending on the design (Table 1). A particularly noteworthy observation is that backwardly-curved (BC) fans, which can be used essentially interchangeably with forwardly-curved (FC) fans, have efficiencies that are about 20 percentage points higher than for FC fans. Thus, the scope for electricity savings is especially large in existing applications where FC fans are used.

Table 1. Maximum total efficiency estimates for different fan designs.

	Peak Efficiency Range
Centrifugal Fans	
Airfoil, backwardly curved/inclined	79-83
Modified radial	72-79
Radial	69-75
Pressure blower	58-68
Forwardly curved	60-65
Axial Fans	
Vaneaxial	78-85
Tubeaxial	67-72
Propeller	45-50

The performance of the most common fan designs are discussed here. For comparative purposes relative performance characteristics of different fan designs are shown plotted on common but arbitrary coordinates. This illustrates the relative operating characteristics of the different fan types, but is not a direct comparison between equal speeds or equal fan sizes. As a reference point, we arbitrarily define the characteristics of a typical ventilating fan as a "standard" fan.[4]

Airfoil or Backwardly Curved/Inclined-Blade Fan (Fig. 9): These fans are used for general heating, ventilating and air conditioning (HVAC) applications. They have the highest efficiency of all centrifugal fans (Table 1) because of the blade design, so are especially useful in larger power applications, where significant power savings can arise compared to the use of a different fan design. These fans are also used in industrial systems, e.g. in some forced- and induced-draft fans. In industry the airfoil design is used only with clean air. The other two designs can be used where the air contains moderate amounts of corrosive and/or erosive material.

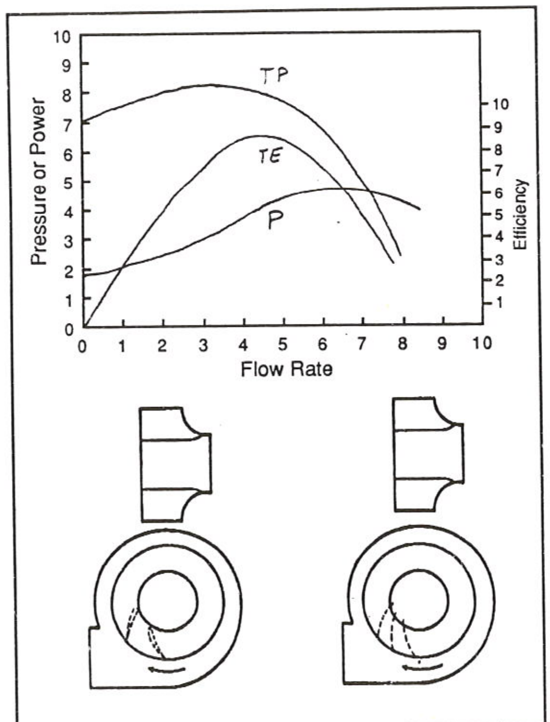


Figure 9. Performance and schematic of airfoil (left) and backwardly curved/inclined-blade (right) fans. (See Fig. 5 caption for definitions.)

These fans use expanding scroll type housings, which efficiently convert velocity pressure to static pressure, and have close clearance and good alignment between the wheel and inlet bell. For a given duty, this fan will also operate at the highest speed of all centrifugal fans.

Another important characteristic of this fan is that its performance curve is stable, and the fan has a load-limiting power characteristic: the input power reaches a maximum near peak efficiency and decreases toward the free delivery condition (Fig. 9).[5] If the fan is equipped with a motor sized to meet the maximum power requirement, there would be no danger of overloading the motor [3,4].

Radial, Modified Radial and Pressure Blower Fans (Fig. 10): These fans are typically used in heavy duty industrial applications, in particular where foreign material such as wood chips, sand, paper cuttings, etc. pass directly through the wheel. The radial design is the simplest of all centrifugal fan designs. It has high mechanical strength and can be run at high tip speeds. One drawback is that it has inherently lower efficiency (Table 1).

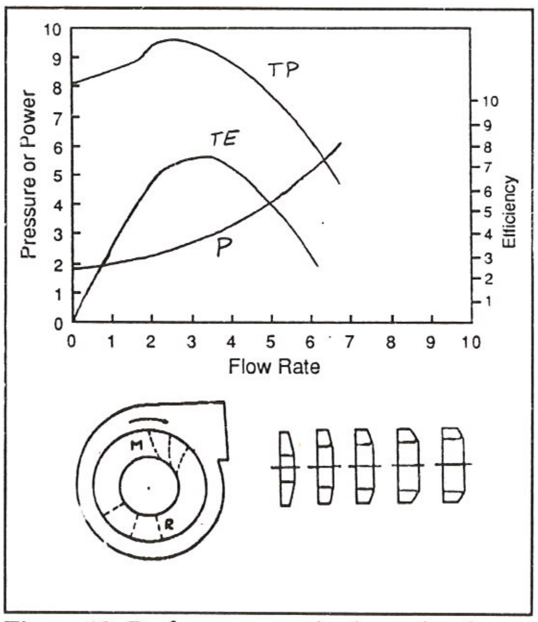


Figure 10. Performance and schematic of radial (R) and modified radial (M) fans. (See Fig. 5 caption for definitions.)

These fans are designed with many different wheel proportions, ranging from wide shallow blades for high-volume, low-pressure applications (radial and modified radial designs) to narrow, deep blades for high-pressure, low-flow applications (pressure blower designs). This fan uses an expanding scroll housing and is usually the narrowest of all centrifugal fans. Due to its inherently lower efficiency, dimensional requirements of the housing are not as tightly controlled as for airfoil and backwardly-inclined fans. A poorly dimensioned housing might penalize efficiency by two to five points.

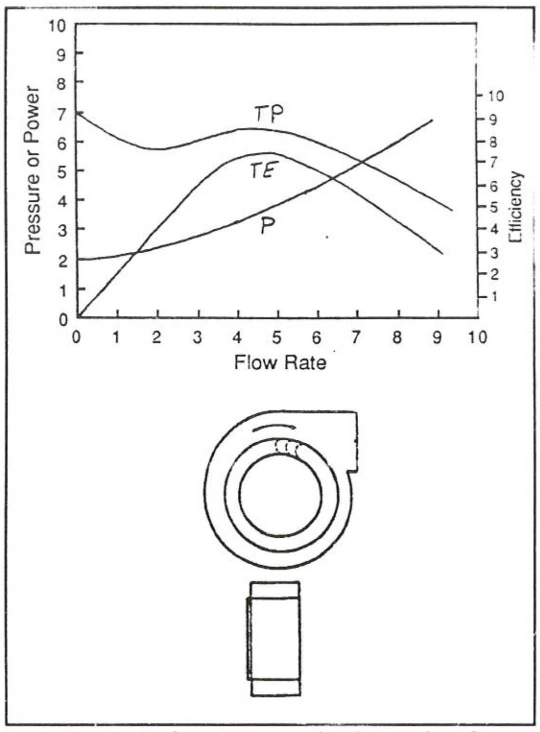


Figure 11. Performance and schematic of forwardly-curved blade fans. (See Fig. 5 caption for definitions.)

For equal flow and pressure this type of fan will run slower than the airfoil, backwardly-curved, and backwardly-inclined fans. It will run faster than the forwardly-curved fan. Also, as noted on the performance curve, this fan develops a higher pressure than the centrifugal fans described above and has a somewhat lower flow rate. The performance curves of radial fans have a break to the left of peak pressure. They are not operated in this range, since pulsating flow and pressure would be the result. Unlike the case with airfoil fans, the power for radial fans rises continually to free delivery. The usual operating horsepower is well below this and corresponds to a point near peak efficiency. The motor is usually sized somewhat higher than the operating point. Motor overload is not a serious concern in most industrial applications since there is a fixed downstream system resistance which prevents the fan from approaching free-delivery operation.

Forwardly-Curved Blade Fan (Fig. 11): This fan is usually used in low-pressure HVAC applications, such as domestic furnaces, central station HVAC units, and packaged air conditioning equipment. It has the lowest efficiency (Table 1) and, for a given duty, operates at the lowest speed of the centrifugal fans descibed here.

These fans are usually light weight and have low fabrication costs. They use from 24 to 64 shallow blades and an expanding scroll housing. The fit between the wheel and the inlet is not as tightly controlled as on the airfoil and backwardly curved blade fan. The efficiency penalty with a poorly designed housing would be three to five percentage points.

The pressure curve is not as steep as with most other centrifugal designs, and there is a dip in the curve to the left of peak pressure. The fan is typically operated just to the right of peak pressure, which corresponds to the highest efficiency point. The input power rises steadily towards free delivery, which must be taken into account when the motor is selected.

Vaneaxial Fans (Fig. 12): This fan can be found in essentially all types of HVAC applications [6,7]. It is particularly useful where straight-through flow and compact installation are required. It is also used in many industrial applications, e.g. power plants, fume exhaust systems, mines, wind tunnels, and chemical process plants. The discharge air distribution is good due to the straightening vanes on the downstream side of the wheel.

The fan may have from 5 to 20 blades. They are usually of airfoil design, which permits medium to high-pressure capability at good efficiency. The cylindrical housing is usually fitted to the outer diameter of the blade tip so that the clearance is small. The fan must be fitted with a set of downstream guide vanes to remove swirl from the flow, which is required to achieve good efficiencies.

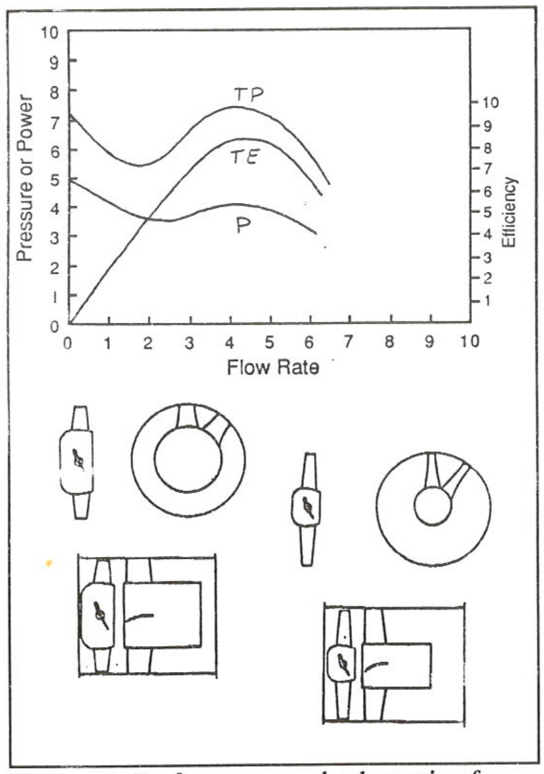


Figure 12. Performance and schematic of high-hub-ratio (left) and low-hub-ratio (right) vaneaxial fans. (See Fig. 5 caption for definitions.)

This fan develops the highest pressure of all axial fans at medium volume flow rate. The pressure curve includes a dip to the left of the peak, which is caused by aerodynamic stall. These fans cannot be operated in this region. The input power requirement has a load-limiting characteristic similar to that of airfoil centrifugal fans, with the peak power occurring somewhat to the left of peak efficiency. However, the power requirement at zero flow is higher than this peak power point. Some accommodation for this characteristic must be made in some applications, for example, if startup against a closed outlet damper is required.

Some vaneaxial fans have the capability to change the pitch of the blade to meet changing operating conditions. In some designs, the fan must be stopped and the blade angle changed manually. In other cases, the pitch of the fan blade can be controlled remotely to permit changes during operation. This latter method is particularly useful in operating the fan as efficiently as possible under changing load conditions. For additional discussion of fan speed control methods, see [1].

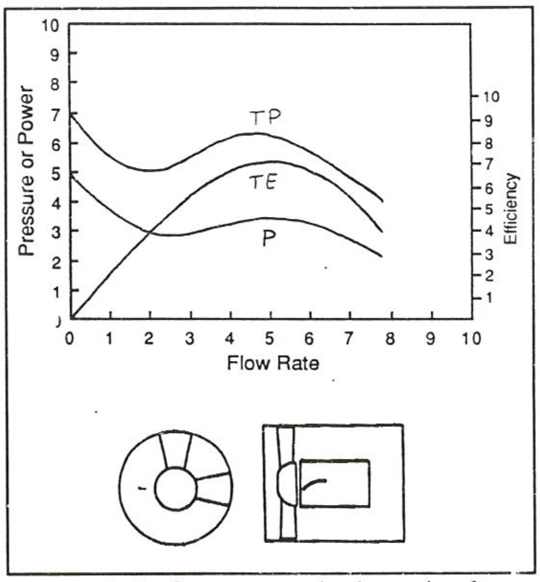


Figure 13. Performance and schematic of tubeaxial fans. (See Fig. 5 caption for definitions.)

Tubeaxial Fan (Fig. 13): These fans are used in low and medium-pressure ducted heating and air conditioning applications where a uniform air distribution on the downstream side is not critical, e.g. ventilation through very short ducts. It is also used in some industrial applications such as drying ovens, paint-spray booths and general exhaust systems. Because it lacks guide vanes, the discharge flow is swirling and turbulent. The turbulent discharge contributes to relatively low efficiency and to low static pressure and modest total pressure characteristics.

The number of blades usually varies from 4 to 8 and the hub is usually less than 50% of the fan tip diameter. Blades can be of airfoil or single thickness construction. The wheel operates in a cylindrical housing formed so that the running clearance between the wheel tip and tube is reasonably close.

Propeller Fan (Fig. 14): This type of fan is used for lower pressure, high volume air moving applications, usually without duct work. Applications include air circulation within a space or ventilation through a wall. It is also often used for makeup in air applications.

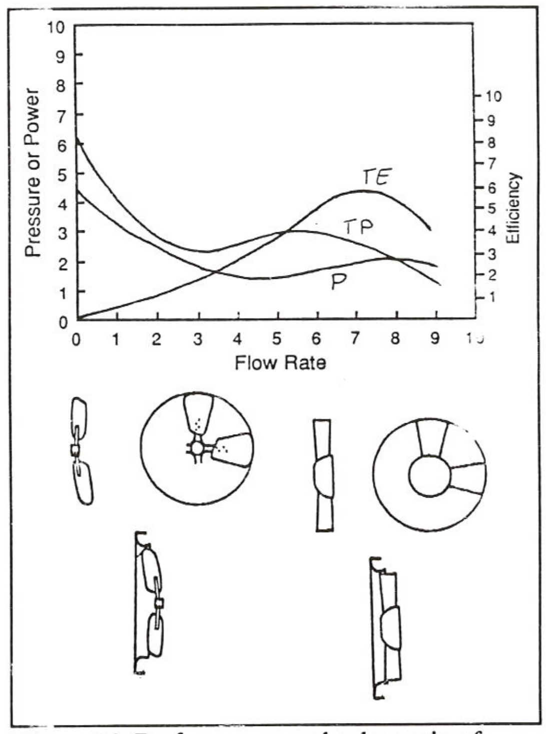


Figure 14. Performance and schematic of curved-blade (left) and heavy-duty-construction (right) propeller fans. (See Fig. 5 caption for definitions.)

The impellers are usually inexpensively constructed and consist of from 2 to 8 single-thickness blades attached to a relatively small hub. Instead of a housing, these fans use a simple circular orifice or venturi design. Energy is transferred primarily in the form of velocity pressure.

The performance curve indicates the low-pressure, high-flow characteristics of this fan. Maximum efficiency is quite low and is reached near free delivery. Because of the lack of a housing or guidevanes, the air on the discharge side is typically quite turbulent and swirling.

Costs

Since the price of fans to installers or end-users is typically quoted on a job-by-job basis, it is difficult to give reliable fan cost data. Roughly, the retail price of a fan can be ranked according to its efficiency. For example, in Sweden the price of a backwardly-curved fan is estimated to be about 15% higher than a forwardly-curved fan [8]. Thus, there will typically be a trade-off between higher first cost and higher operating costs that should be carefully assessed during the design process.

4. REFERENCES

1. J.B. Graham, "Air Handling," *Technology Menu*, Vol. 1, Dept. of Environmental and Energy Systems Studies, Lund Univ., Lund, Sweden, 1989.

2. R. Tsal and H. Behls, "Ducting," *Technology Menu*, Vol. 1, Dept. of Environmental and Energy Systems Studies, Lund Univ., Lund, Sweden, 1989.

3. R. Jorgensen (ed.), *Fan Engineering*, Buffalo Forge Co., Buffalo, New York, USA, 1985.

4. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, "Fans," ch. 3, *1988 Handbook-Equipment*, ASHRAE, Atlanta, Georgia, USA, 1988.

5. L.J. Nilsson and E.D. Larson, "Adjustable Speed Drives," *Technology Menu*, Vol. 1, Dept. of Environmental and Energy Systems Studies, Lund Univ., Lund, Sweden, 1989.

6. R.A. Wallis, *Axial Flow Fans and Ducts*, John Wiley & Sons, New York, 1983.

7. B.B. Daly, *Woods Practical Guide to Fan Engineering*, Woods of Colchester, Colchester, England, 1978.

8. J. Brannstrom, H. Lundstrom, and B. Larsson, "Prestudy of Efficient Industrial Electrical Equipment," for Vattenfall, Vastsverige, Gothenburg, Sweden, 1988. (in Swedish)

9. AMCA Standard 210-85/ASHRAE Standard 51-1985, "Laboratory Methods of Testing Fans for Ratings," American Society of Heating, Refrigerating, and Air-Conditioning Engineers (Atlanta, Georgia) and Air Movement and Control Association, Inc. (Arlington Heights, Illinois), USA, 1985.

NOTES

1. Efficient fans maintain stratification of the air as it passes through, since mixing leads to unnecessary losses.

2. The specific speed is independent of fan size for geometrically similar fans. The specific speed is used primarily by fan designers. An engineer selecting a fan for a particular application might use N_s only when some limitations are put on the operating conditions. The most common limitation is a fan that, for some reason, cannot be driven through any speed change system and must be driven directly at a standard motor speed. Under this condition, Fig. 4 can be used to select the most efficient fan type for the given operating conditions.

3. Proper application of the fan laws requires the "calculated" fan to have the same tested performance characteristics as the "base" fan.

4. For any one of the given fan designs, the fan laws can be used to predict the performance curve for dimensionally similar fans of different sizes and/or operating at different speeds.

5. Free delivery refers to the operating point at which there is no downstream flow resistance, i.e., when total pressure is zero and flow rate is at a maximum.

Vol. 1: Movement of Material **89-8-8**

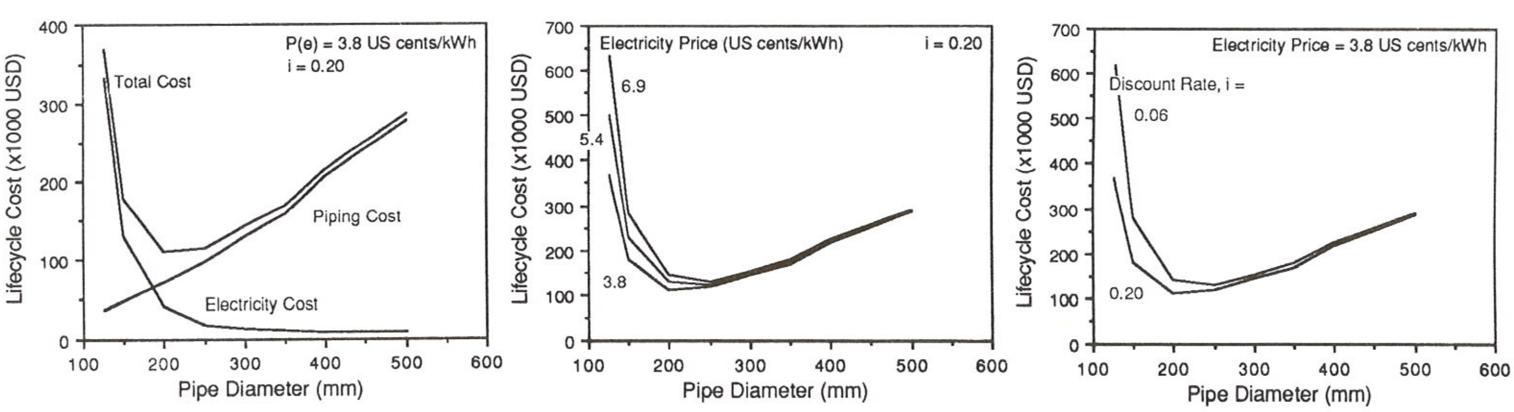
Component: **PIPING**

by
Nicholas P. Cheremisinoff, Ph.D.
Exxon Chemical Co.
Linden, New Jersey, USA

CONTENTS

SUMMARY

Overcoming hydraulic resistances in piping and fittings accounts for a significant fraction of the electricity input to a liquid pumping system. This Menu entry outlines methods for estimating piping pressure losses for steady-state, single-phase flows and for calculating the economically optimal pipe diameter for a given application. The pressure loss calculations described here are widely used, but involve some uncertainty so that design factors of 25% or higher are typically applied. (Pressure losses in multi-phase flows, e.g., wood-pulp and mineral-ore slurries, are much less well understood and are the focus of much of the R&D relating to piping.) The calculation of the optimum pipe diameter involves trading off capital and operating costs. In an illustration given here, the result is sensitive to what capital costs are included (left figure), to what electricity price is assumed (center figure), and to what discount rate is used (right figure). It is also noteworthy that the lifecycle cost is relatively flat in the region of the minimum (200-300 mm), yet electricity use varies by a factor of three over this region.



1. TECHNOLOGY

Nearly 30% of the 1986 industrial electricity use of 33 TWh in Sweden was estimated to be for fluid transport operations [1]. The energy consumption associated with pumping depends largely on the hydraulic resistances, including elevation changes, arising in the plant's operation. It is these resistances which establish the type and capacity of pumps required, materials of construction, expected peak loads and overall service requirements, all of which contribute to establishing the level of operating and maintenance cost. The choice of design flow rates is also important since pumping power requirements are proportional to the cube of volume flow. Once the design flow rates are fixed, the power required for a particular application will depend on the hydraulic resistances.

The magnitude of the resistances depends on the fluid properties, the dynamics of the process operation, and the physical equipment requirements dictated by the process. Fluid properties refers to viscosity and/or rheology, surface-tension, vapor-pressure, boiling point, flash point, and (in certain applications) propensities for promoting attrition or agglomeration of entrained particles. Also important are chemical and thermal stability, propensity for fouling surfaces, and degree of corrosiveness. The dynamics of the operation relates both to fluid properties that influence flow stability (e.g., flow rate requirements) and the constraints imposed by the process itself. In this regard, fluid handling applications in industry are often very different from those in residential systems. The primary reason for this is that in most manufacturing operations the fluid is subjected to wide variations in process conditions. These process changes, as well as chemical transformation during the course of manufacturing, alter fluid properties, so that the material can change from a low viscosity to a high viscosity liquid, from Newtonian to non-Newtonian rheology, from single-phase to multiphase flows, etc. These flow regime changes each impose different hydraulic resistances so that it is not uncommon to have dramatically different pumping and energy requirements in different parts of a single plant's operation. And, of course, there are greatly different requirements between different manufacturing operations and industries. Furthermore, there are numerous examples, particularly in the chemical industry, where flow regimes and corresponding hydraulic resistances are fixed by the desired properties of the end product.[1] The criteria for manufacturing operations also impact pumping requirements, e.g., the materials of construction required to handle process fluids, the service requirements both from normal and transient process operations, and of course, the physical limitations of the process operation or plant layout.

Information from all three areas is required to design an efficient flow system and to determine the corresponding energy requirements. Ultimately, the design basis for a distribution system will specify line size, geometry, construction materials, piping system layout, flow control and monitoring instrumentation -- each one contributing to the total hydraulic resistance of the operation. In this Menu entry, the basic engineering calculations for estimating frictional pressure losses in piping systems are outlined for steady-state, single-phase flow systems, which serves to illustrate the general methodology for piping calculations and which provides information relevant to many piping applications. Also, from information on the system's frictional pressure losses, an analysis of the demands for energy consumption and the ultimate cost of proposed designs can be made, as will also be discussed.

2. R&D

The state-of-the-art design procedures for piping systems to handle steady-state, single-phase or homogeneous flows is well established.[2] Thus, much of the R&D relating to piping is focussed on applications other than these, e.g. transient flow networks, which in some reactor operations is the norm, piping of fluids with viscoelastic properties, and multi-phase flows.

Multi-phase flows are particulary important in Sweden. Examples include pulp and mineral-ore slurries. Energy requirements for pumping of multi-phase flows are difficult to estimate accurately. The design principles for such flows are not clearly defined--conflicting design criteria can be found in the literature--so that most frictional loss estimates are based on empirical results. Today's design practices mandate testing for each application to establish hydraulic design data. However, this is expensive and not always reliable. For example, with slurry flows,[3] attrition of solids and particle settling affect hydraulic behavior. Such factors are difficult to simulate, particularly in scaled-down tests. Thus, extrapolation of test results involves significant design uncertainties which are typically compensated for by overdesign. Safety factors in excess of 30% on pump power requirements are not uncommon, which of course can lead to inefficient energy usage.

Thus, a major R&D thrust relating to piping systems is focussed on improving the phenomenological understanding of multi-phase flows and on the development of more reliable predictive models. Research and development relating to slurry flows exemplifies this focus [2,3,4]. R&D is most concentrated today on developing better bases for predicting the behavior of such flows and consequently establishing more rigorous design and scale-up procedures for estimating frictional pressure losses. Numerical modeling appears particularly promising in this regard. Modeling of homogeneous flows is well established, with both two and three-dimensional models commercially available. Modeling relating to non-homogeneous flows requires R&D thrusts primarily in two areas. First, a more organized review of the available literature on estimating hydraulic design criteria is required. Second, a greater effort is needed in validation of numerical techniques applied to such flow systems.

3. TECHNICAL AND COST DATA

Pressure Losses

The most important element in understanding energy aspects of piping system design is the determination of the pressure losses that arise from the hydraulic resistances in the system. This section describes standard techniques for estimating pressure losses, which are based on years of successful industrial experience with repetitive plant construction.

Bernoulli Equation: The basic equation for calculating pressure drop for steady-state liquid flow through pipes and fittings is the Bernoulli equation, assuming constant fluid density:

$$-\Delta P = (V_2^2 - V_1^2)/2 + \rho g \Delta h + \Delta P_f \qquad (1)$$

Static press. change	Kinetic energy change	Elev. change	Friction losses

where ρ is density, V_1 and V_2 are the velocities at the inlet and outlet of the pipe, Δh is change in elevation between the inlet and outlet, and g is gravitational acceleration. The frictional pressure loss is usually given in terms of head loss, h_f:

$$P_f = \rho g h_f \qquad (2)$$

The relative importance of the terms in the equation varies with the application. For constant-diameter horizontal pipes, only the friction term on the right-hand side of the equation is important. For vertical or inclined pipes, one must include the elevation term; for cross-sectional changes, the kinetic energy term.

Eqn. 1 holds for Newtonian liquids, for which viscosity and density can be assumed constant. These include water and most other common liquids. Modifications to Eqn. 1 are required in some cases, e.g., when the fluid cannot be assumed to be isothermal [3].

The calculation of each term in the Bernoulli equation is discussed below. After all three terms are calculated and summed, the result is typically multiplied by a design factor of 1.25 to account for uncertainties in the calculations. (In cases where the elevation change accounts for most of the pressure head, the total pressure head is known quite accurately and a smaller design factor can be used.)

Friction losses: This is typically the most important term in Eqn. 1. Most friction loss analysis begins with a calculation of two parameters, the pipe's equivalent hydraulic diameter, d_H, and the Reynolds Number, Re.

The equivalent hydraulic diameter is

$$d_H = 4 * (A/WP) \qquad (3)$$

where A is the cross-sectional area and WP is the length of the wetted perimeter

For a given flow rate and d_H, the Reynolds number is used to characterize the flow (e.g., turbulent or laminar):

$$Re = V d_H/\nu \qquad (4)$$

where V is velocity, and ν is the kinematic viscosity of the fluid.

Friction losses are calculated separately for straight pipe sections and for different fittings: pipe inlet sections, bends, valves, tees and changes in cross section. Calculation procedures for all of these are outlined here, and turbulent flow is assumed except where indicated otherwise.

For straight pipe sections, the friction head loss is calculated using the Darcy Equation:

$$h_f = \lambda (L/d_H)(V^2/2g) \qquad (5)$$

where h_f is the friction head loss (e.g., in meters), L is the pipe length, and λ is the dimensionless Fanning friction factor, which is function of Re and the relative roughness of the pipe. The friction factor is typically taken from the Moody diagram (Fig. 1), but can also be estimated using the following equations:[4]

$$\lambda = 64/Re \qquad \text{laminar flow, } Re < 2000 \qquad (6a)$$

$$\lambda = 0.16/Re^{0.16} \qquad \text{turbulent flow, } Re > 4000 \text{, in smooth pipes [5]} \qquad (6b)$$

$$\frac{1}{\sqrt{\lambda}} = -2\log[2.51/(Re\sqrt{\lambda}) + e/(3.71 d_H)] \qquad (6c)$$

turbulent flow in rough pipes, e/d_H is relative roughness [6]

For different pipe materials, the smoother the pipe, the lower the friction factor and, hence, the lower the pumping energy requirements per unit length. A good example is a comparison of PVC plastic and commercial steel pipes. From Table 1, it can be seen that the relative roughness for the two pipes with the same diameter varies by a factor of 30. The energy implications of such differences are discussed more fully in Section 4.

Fouling can increase pipe wall roughness or reduce the cross-sectional area with time and therefore change the pipeline pressure drop characteristics. The majority of data concering the effect of fouling on pressure drop are for water piping. In these cases, the empirical Hazen-Williams correlation,

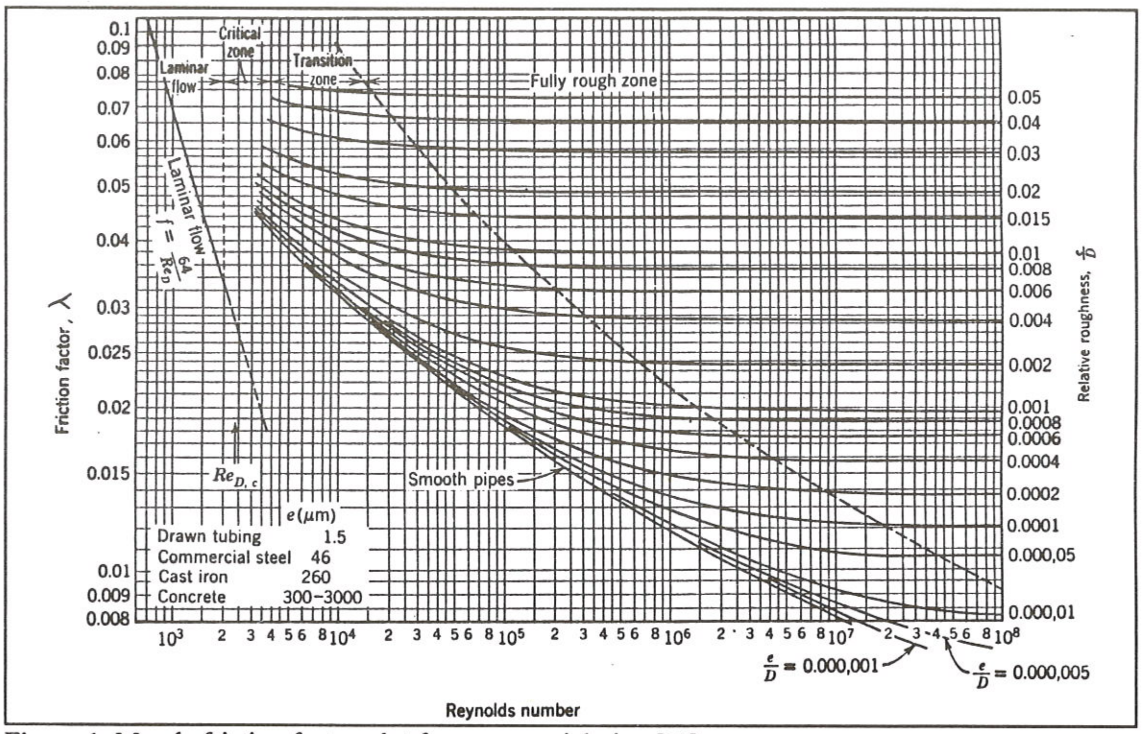


Figure 1. Moody friction factor plot for commercial pipe [12].

Table 1. Absolute roughness of several pipe materials.

Material	Absolute Roughness (mm)
Smooth	0.0015
Glass	
Drawn tube (brass, copper, lead)	
PVC plastic	
Commercial steel	0.046
Iron	
Wrought	0.046
Asphalted cast	0.12
Galvanized	0.15
Cast	0.26
Wood stave	0.18-0.09
Concrete	0.31-0.031
Riveted steel	0.91-0.091

which takes account of fouling, can be used instead of the Fanning friction factor and Darcy equation. See [3,4,7].

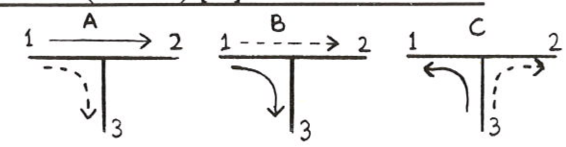
For calculating the resistances due to fittings (e.g. bends, inlets, valves, etc.) one of several methods is commonly used. One method involves computing an equivalent length of straight pipe that would give the same flow resistance. The equivalent length is added to the actual length used in the Eqn. %. The equivalent lengths, L_e, for several common fittings are shown in Table 2.[5] Also shown in Table 2 are head loss coefficients, K, which provide an alternative way to estimate the head loss in fittings and valves using the following equation:

$$h_f = K(V^2/2g) \qquad (7)$$

Table 2. Loss coefficients and equivalent lengths for turbulent flow through selected valves and fittings [3].

Type of Valve or Fitting	Head Loss Coeff. (K)	Equivalent Length/pipe Diameter (L_e/d_H)
Elbow		
45°	0.35	17
90°	0.75	35
Return bend	1.5	75
Coupling	0.04	2
Union	0.04	2
Gate valve		
Full open	0.17	9
Half open	4.5	225
Globe valve		
Full open	6.0	300
Half open	9.5	475
Angle valve		
Full open	2.0	100
Check valve		
Swing type	2.0	100
Water meter, disk	7.0	350

Table 3. Pressure drop equations for split and join flow streams (SI units) [13].[a]

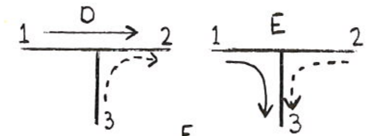


Split Flows

A. $\Delta P_{1\text{-}2} = (5.0\text{x}10^{-4})\,\rho\,(1.36V_2^2 - 0.64V_1^2 - 0.72V_1V_2)$

B. $\Delta P_{1\text{-}3} = (5.0\text{x}10^{-4})\,\rho\,(1.8V_3^2 - 0.368V_1V_3)$

C. $\Delta P_{3\text{-}1} = (5.0\text{x}10^{-4})\,\rho\,(1.8V_1^2 - 0.368V_1V_3)$

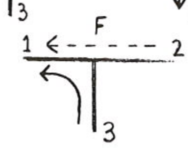


Join Flows

D. $\Delta P_{1\text{-}2} = (5.0\text{x}10^{-4})\,\rho\,[2V_2^2 - 0.05V_1^2 - 2V_2(0.205V_3Q_3/Q_2 + V_1Q_1/Q_2)]$

E. $\Delta P_{1\text{-}3} = (5.0\text{x}10^{-4})\,\rho\,[2V_3^2 - 0.4V_1^2 - 0.41V_3(V_1Q_1/Q_3 + V_2Q_2/Q_3)]$

F. $\Delta P_{3\text{-}1} = (5.0\text{X}10^{-4})\,\rho\,[2V_1^2 - 0.4V_3^2 - 2V_1(0.205V_3Q_3/Q_1 + V_2Q_2/Q_1)]$



(a) Pressure drop includes that due to both friction and kinetic energy changes. V is velocity and Q is volume flow.

Loss coefficients for different fittings are available in graphical and tabular form in the literature, e.g. see Table 2 and [2,3,6,8], and from equipment suppliers. Some illustrative values for different fittings are discussed here. Loss coefficients due to inlets can vary significantly (0.05 to 3) depending on the specific design. Loss coefficients for a 90° smooth bend vary from 0.2 to 0.4 depending on the relation between bend radius and pipe diameter. With segmented bends, loss coefficients are about 50% higher than for smooth bends. Valves have a wide range of loss coefficients depending on the design. For example, the loss coefficient for a fully-open ball valve can vary from 0.5 to 2 depending on the design [6]. The type of valve used for a specific application will depend on the process requirements.

For resistances due to tees, the pressure drop equations given in Table 3 can be used. These account for both frictional pressure drop and pressure drop due to changes in kinetic energy. To account for entrance and exit effects in cases where the inlet leading line is short, a multiplying factor of 1.25 is applied to the calculated pressure drop.

The pressure drop in cross-sectional changes, such as exits and entrances of process vessels and reducers and diffusers, consists of two components: one for friction and one for variation in kinetic energy. The former type of losses are discussed here. Kinetic energy changes are discussed later. Several methods are used to calculate the pressure drop, depending on the specific geometry. For example, approximate values of loss coefficients for sudden contractions as a function of the diameter ratio can be estimated from [3]:

$$K = 1.5(1-d_1/d_2)/(3-d_1/d_2) \qquad (8)$$

The associated head loss is calculated using Eqn. 7 with V being the average velocity in the smaller section. For a sudden expansion, an approximate formula for head loss is the Carnot-Borda equation [3]:

$$h_f = (V_1 - V_2)^2/2g \qquad (9)$$

where V_1 and V_2 are the upstream and downstream velocities. Alternatively, Eqn. 7 can be used with the following loss coefficients [6]:

d_1/d_2	0.1	0.4	0.5	0.67
K	1	0.7	0.6	0.3

and with V being the average velocity in the smaller section. For losses in conical and gradual expansions and contractions there are several empirical results available in the literature [6,8,9]. Figure 2 illustrates that the loss coefficient increases with increasing angle of expansion, and is actually higher than that of sudden expansions for angles greater than about 40-50°. The minimum loss coefficient is

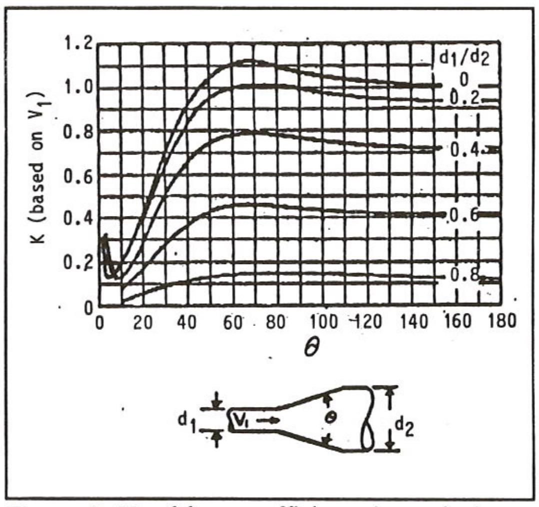


Figure 2. Head loss coefficients in conical diffusers [9].

associated with expansion angles of 7-8° [8]. For gradual contractions, head losses are generally very small.

The interaction between fittings, tees, and cross sectional changes affects the head losses in a piping system. Optimizing the distance between fittings can reduce head losses by perhaps 30-40% [10].

Piping Details Unavailable: When details on pipe fittings, bends, tees, etc. are not available, the following guidelines are typically used to estimate the equivalent length to use in calculating the pressure drop due to all frictional effects.

For onsite lines (within a plant) the actual pipe length can be estimated from the plot plan, tower heights, etc. The equivalent length for fittings in onsite piping is between 200% and 500% of the actual length of piping. Consequently, a multiplier of 3.0 - 6.0 may be applied to the estimated length of straight pipe. For offsite lines the approximate length of straight pipe can be estimated from the plot plan. Since fittings in offsite lines usually have an equivalent length of 20-80% of the actual length, a multiplier of 1.2 - 1.8 can be applied to the estimated length of straight pipe. Because of the large uncertainty in making these estimates, it is always better to make the detailed frictional pressure loss calculations whenever possible.

Elevation and kinetic energy changes: If a pipe is not horizontal, the pressure head due to elevation change is computed as shown in Eqn. 1. The pressure drop due to any changes in kinetic energy in the line is determined by computing the overall change in kinetic energy between the inlet and outlet of the line, as also indicated in Eqn 1.

The pressure losses due to elevation and kinetic energy changes are added to the sum of the frictional pressure drops in straight sections and through fittings to arrive at the total pressure loss (Eqn. 1).

Piping Costs

The cost of piping depends strongly on the material of construction, pipe diameter, the finishing and joining method used, and type of installation (e.g. underground, wall-mounted, pressure class, thermal expansion capabilities, etc.) Figure 3 shows relative cost data for a variety of common pipe types as a function of pipe diameter. The cost per unit length for each type of pipe has been normalized to the cost of seamless 304/304L stainless steel tubing of the same diameter. Thus, knowing the absolute cost of 304/304L stainless, a designer can use Fig. 3 to compare the required capital investment for alternative pipe materials. With nearly all pipe materials, the cost per unit length will increase essentially linearly with diameter. Since the cost of 304/304L stainless increases more quickly with diameter than all except cast iron and carbon steel, however, Fig. 3 shows decreasing pipe costs ***relative*** to 304/304L stainless for all except cast iron and carbon steel.

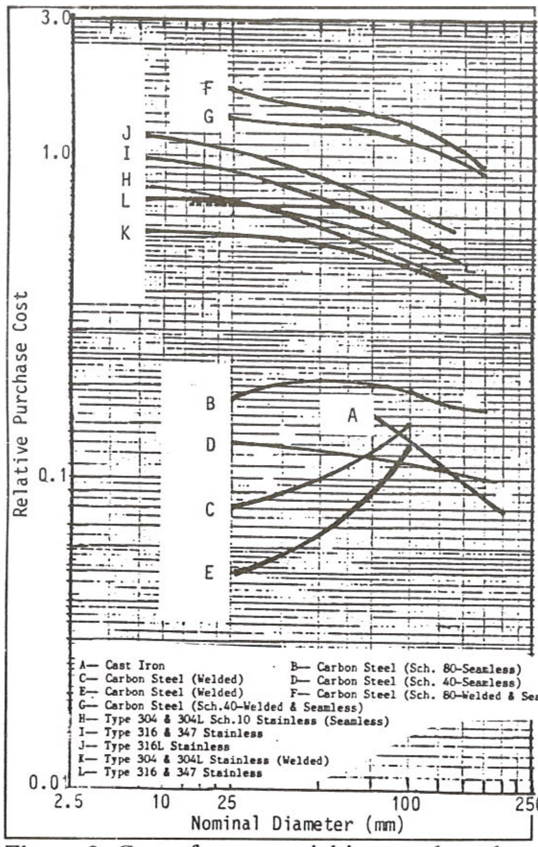


Figure 3. Cost of commercial iron and steel piping relative to the cost of 304/304L stainless steel pipe.

4. ILLUSTRATIVE ECONOMIC ANALYSIS

Every piping application requires a careful analysis of technical and economic tradeoffs. Several pipe diameters typically will be acceptable in terms of pressure losses for a desired flow rate. One of several different piping materials may also be acceptable. Selecting a larger diameter and or smoother material will give lower frictional losses. This permits a smaller, less costly pump and motor to be used and leads to lower electricity consumption. However, larger piping means higher piping and construction costs and typically higher maintenance and repair costs. For a given material, smoother piping is also more costly. With a smaller diameter pipe, the frictional losses, electricity use, and pump/motor capital costs all increase, whereas the cost for the piping, construction, and maintenance decrease.

Diameter and Wall Roughness Effects on Pumping Energy Costs

The separate effects of piping diameter and pipe wall roughness on energy use in piping can be quantified starting with an expression for the energy required for pumping, E_p:

$$E_p = t * Q * \Delta P/\eta \qquad (10)$$

where Q is the volume flow rate, ΔP is total pressure drop, η is the motor/pump efficiency, and t is the operating time. It is worth noting that since frictional pressure losses increase with volume flow squared (see Eqn. 2 and 5), Eqn. 10 indicates that the energy required for pumping to overcome friction increases with the cube of volume flow. Reference [1] gives a quantitative illustration of energy savings that can be achieved in applications where lower design flow rates can be used. Here we consider only the effect of parameters related to the pipe: diameter and roughness.

Using Eqns. 2,5 and 10, it can be seen that for the same volume flow rate, pumping energy use is inversely proportional to pipe diameter raised to the 5th power and directly proportional to the friction factor.[6] Since the friction factor also depends somewhat on diameter, the pumping energy required to overcome friction in piping of different diameters is:

$$E_{p1}/E_{p2} = (D_2/D_1)^{4.84} \qquad (11)$$

This expression is plotted in Figure 4. Starting with a fixed pipe diameter, large percentage reductions in pumping energy are possible with line size increases up to a factor of about 1.5. Beyond this, only relatively small *additional* savings can be realized. Note, however, that line sizes should not be increased without considering possible effects on the process. The impact on capital and installation costs must also be considered, as discussed in the final section below.

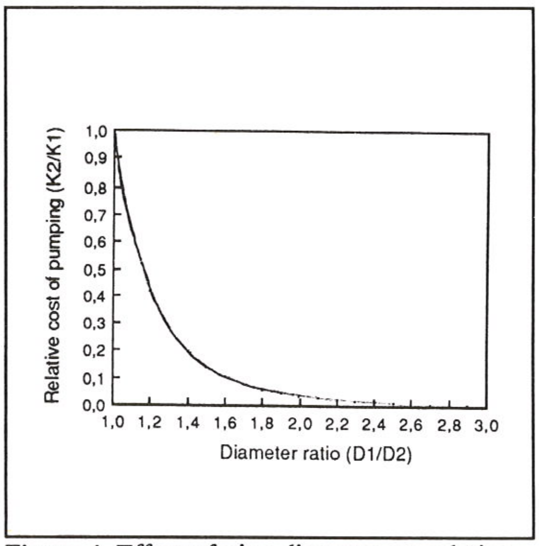


Figure 4. Effect of pipe diameter on relative pumping energy use.

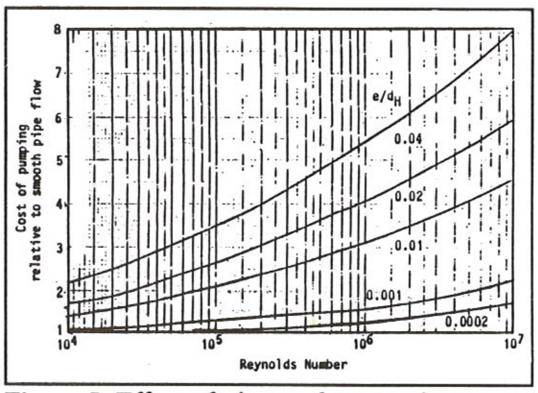


Figure 5. Effect of pipe surface roughness on pumping energy requirements. Energy use is shown relative to that for smooth piping.

The pipe wall roughness also plays an important role in establishing pumping energy requirements. As noted above, E_p varies linearly with the friction factor, which in turn is related to wall roughness by the Moody diagram (Fig. 1). Thus, the pumping energy requirement changes with the pipe's relative roughness as shown in Fig. 5. As noted earlier, the relative roughness can vary substantially for different pipe materials, e.g., a factor of 30 between plastic and commercial steel (Table 1). From Table 1 and Fig. 5 combined, a commercial steel pipe with an inside diameter of 2 cm requires about twice the pumping energy of a 2-cm diameter plastic pipe for a Reynolds number of about 10^6 (turbulent flow). For larger diameter pipes, the effect of roughness is less significant. Figure 5 and Table 1 can be used as a rough guide in selecting piping material on the basis of energy use. It is important to note, however, that surface condition depends not only on the pipe material, but also on the fluid properties as well as scale formation, erosion and various forms of fouling, so that pumping energy requirements could increase with time. In addition, process requirements may impose restrictions on which types of piping can be used.

Total Piping System Analysis: Illustrative Example

The final design of a piping system is typically arrived at by a trial-and-error procedure that involves analyzing the trade-off between energy and capital costs for different pipe sizes and materials. The specific application will determine the constraints on this analysis, e.g., choice of pipe material, peak capacity requirements, need for backup pumps, etc. The calculation usually yields an "optimum" pipe diameter, which minimizes some economic criterion, e.g., the lifecycle cost associated with purchase, installation, operation and maintenance of the piping system. Various design procedures have been developed to calculate this "optimum" diameter [3,6,11].

A complete analysis of the optimum pipe diameter involves first determining the design flow rates and then considering a large number of cost-related issues (capital costs for pipe, pump, motor, drives, valves and support structure and required return on investment, taxes, insurance, etc.). The decision regarding which considerations to include in an analysis is typically made based on the accounting conventions of the firm and the degree of complexity involved in a more complete analysis. In many cases, the optimum pipe diameter analysis is arbitrarily limited to consideration of only the pipe capital costs and operating and maintenance costs [3].

Table 4. Summary of inputs for calculation of the optimum pipe diameter for an illustrative case of pumping water over a distance of 500 m.[a]

Pipe diameter (mm)	125	150	200	250	300	350	400	500
Friction loss (m)[b]	204	77	20	6.5	2.8	1.8	0.9	0.3
Total head (m)[c]	299	118	36	17	11	10	8.6	7.7
Max. Power (kW)	377	148	46	21	14	12	11	10
El. use (MWh/yr)[d]	2056	809	251	114	78	67	59	53
Capital Costs (10^3 $)[e]								
Piping	37	48	72	98	132	159	208	277
Motor and Pump	188	74	23	10	7.1	6.2	5.4	4.9
TOTAL	225	122	95	108	139	165	213	282

(a) From [10] for a variable flow application (variation from 20-90 l/s).

(b) For 90 l/s flow, assuming 500 m of piping with absolute roughness of 0.2 mm and the sum of loss coefficients due to fittings, bends, etc of 8.

(c) Includes elevation change of 6 meters and friction losses adjusted from the 90 l/s values for variable flow.

(d) Assuming 6500 operating hours per year, throttle flow control, and motor/pump efficiency of 70%.

(e) The pump/motor costs include reserve units and their associated plumbing. The reserve equipment accounts for 60% of the indicated pump/motor cost. Costs were converted from 1988 Swedish crowns to US dollars using an exchange rate of 6.5 crowns per dollar.

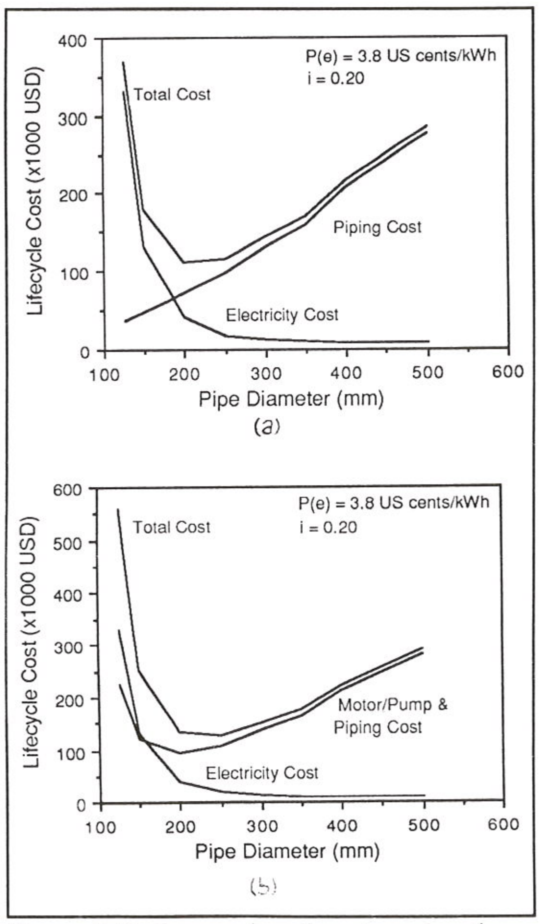


Figure 6. Optimum pipe diameter calcuation based on Table 4 assuming capital costs are for (a) pipe only and (b) pipe and pump-motor.

However, depending on which costs are included or excluded, the calculated optimum diameter can change. This is concretely illustrated by the following example.

Table 4 provides a set of input data for an optimum pipe diameter calculation for a hypothetical variable-flow water piping application [10]. The only capital costs that are considered are for the pipe and the pump/motor, and the only operating costs considered are for electricity. As noted in the table, electricity use decreases with increasing pipe diameter (approximately according to Eqn. 11). The combined capital cost for piping and the pump/motor fall initially, since they are dominated by decreasing pump/motor costs, but rise again as the piping costs become important.

If only piping and electricity costs are considered, then the lifecycle costs[7] are a minimum for a pipe diameter of 200 mm. Capital costs increase linearly and electricity costs fall sharply with diameter (Fig. 6a) to give this result. For the smallest diameter considered (125 mm), electricity accounts for 90% of the lifecycle costs. At 200 mm, electricity accounts for 36% of the total.

The cost of electricity used in the calculation also influences the optimum diameter. Fig. 7a shows the total lifecycle piping-plus-electricity cost as a function of pipe diameter for electricity prices of 3.8, 5.4, and 6.9 cents/kWh. The optimum diameter moves to the right as the electricity price increases.

A lower discount rate might be used in some cases, e.g., to assess the optimum pipe diameter from a national perspective. Decreasing the discount rate from 20% to 6% with an electricity price of 3.8 cents/kWh moves the optimum pipe diameter from 200 to 250 mm (Fig. 7b), since future expenditures for electricity (which decrease with diameter) are less discounted. It is interesting to note in Figs. 7a and 7b, that the lifecycle cost is relatively insensitive to diameter between 200 and 300 mm. Since electricity consumption over this range varies by a factor of 3, if electricity conservation were an important design criterion, the larger diameter could be chosen with little or no economic penalty.

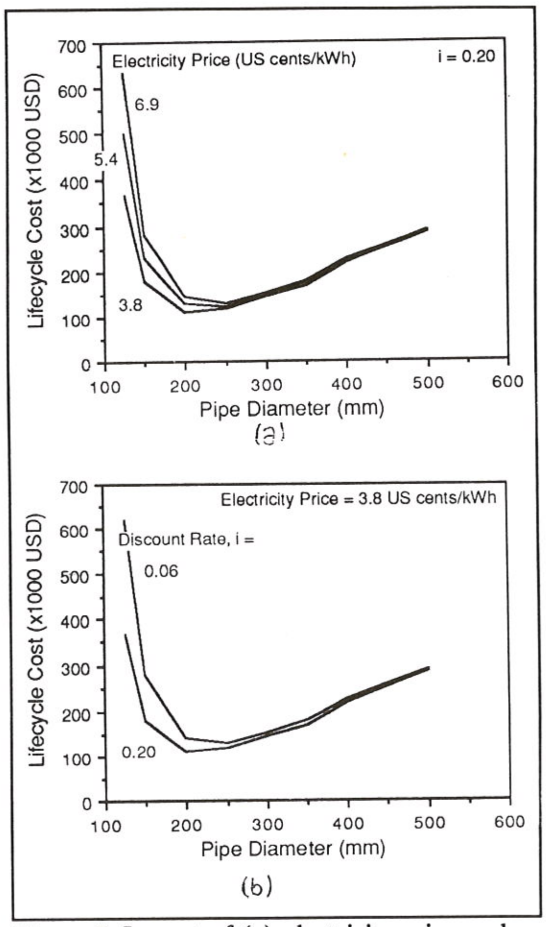


Figure 7. Impact of (a) electricity price and (b) discount rate on optimum pipe diameter (based on Table 4 assuming only capital costs are for pipe material.)

5. REFERENCES

1. L.J. Nilsson and E.D. Larson, "Liquid Pumping," *Technology Menu*, Environment and Energy Systems Studies, Univ. of Lund, Lund, Sweden, 1989.

2. N.P. Cheremisinoff (ed.), *Encyclopedia of Fluid Mechanics*, Vol. 1-8, Gulf Pub. Co., Houston, Texas (1984-1989).

3. N.P. Cheremisinoff and D.S. Azbel, *Fluid Mechanics and Unit Operations*, Ann Arbor Science Pub., Ann Arbor, Michigan, 1983.

4. N.P. Cheremisinoff, *Fluid Flow: Pumps, Pipes and Channels*, Ann Arbor Science Pub., Ann Arbor, Michigan, 1982.

5. Nikuradse, "Laws of Flow in Rough Pipes," NACA Technical Memorandum 1292, Nov. 1950.

6. *Pumphandboken*, ECPrint-AB, Gothenburg, Sweden, 1982. (in Swedish)

7. Hydraulic Institute, *Pipe Friction Manual*, 3rd ed., New York, 1961.

8. *VVS-Handboken*, Förlags AB VVS, Stockholm, Sweden, 1974. (in Swedish)

9. L.L. Simpson, "Process Piping: Functional Design," *Chem. Eng.*, 76(8), Apr 14, 1969, pp. 167-81.

10. A. Jonsson, Jonsson Pumpkonsult, Hagfors, Sweden, personal communication, 1989.

11. M.S. Peters and K.D. Timmerhaus, *Plant Design and Economics for Chemical Engineers*, 2nd ed., McGraw-Hill, New York, 1968.

12. L.F. Moody, "Friction Factors for Pipe Flow," *Trans. of the ASME*, 66, 1944.

13. Crane Co., "Flow Through Valves, Fittings, and Pipe," Technical Paper No. 410, 1970.

NOTES

1. For example, in the chemicals industry, turbulent flow is required in the process used to produce some elastomers like EPDM. Laminar flow would lead to production of a polymer with a very different compositional distribution. Other examples include the handling of sensitive emulsions in paint, and the production of pigments, pharmaceuticals and some food products.

2. An important exception is how the interaction of closely spaced fittings affect frictional losses.

3. Other examples of slurry flows include: (a) transport of coal, concrete, tailings from mineral processing plants, sand/gravel from dredging operations, etc., and (b) flocculation operations in pharmaceuticals and foods manufacturing, in wastewater treatment, etc.

4. If the relative roughness is known, the value of is accurate to about $\pm 10\%$.

5. To determine values of L_e (table 2), some specific value of must be assumed as indicated by Eqn. 5. For Table 2, = 0.02. If the value for actually used in Eqn. 5 differs significantly from this, the head loss due to fittings would be correspondingly over- or under-estimated.

6. This assumes that friction losses are the only components of pressure loss, i.e. no kinetic-energy or elevation changes are assumed.

7. Assuming an electricity price of 3.8 cents/kWh, a discount rate of 20%, and a 10 year amortization period.

Vol. 1: Movement of Material **89-8-8**

Component: **DUCTING**

by
Robert J. Tsal, Ph.D.
NETSAL & Associates
Huntington Beach, California

Herman F. Behls, PE
Sargent & Lundy, Inc.
Chicago, Illinois

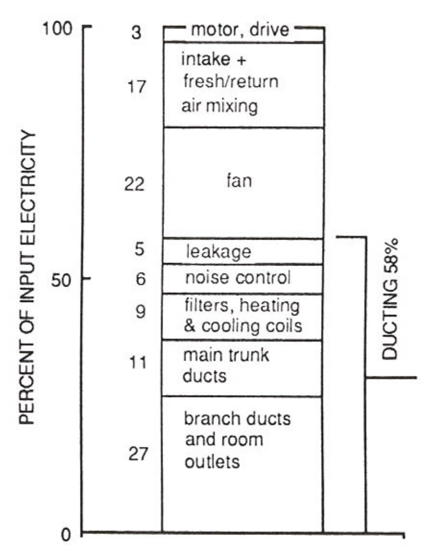
CONTENTS

SUMMARY

The majority of electricity required in many air-handling systems is used to overcome pressure losses in the ducting of the system (see figure below). This Menu entry reviews the procedures for estimating ducting pressure losses, highlighting design choices that can have particularly important impacts on energy use. Computerized economic optimization of duct designs, an important recent development in the field, is also discussed. In new installations, optimized ducting designs can lead to significant lifecycle cost and electricity savings compared to traditionally-designed ducts. The table below indicates the magnitude of the savings for a range of electricity prices and duct materials for a hypothetical duct system (see text).

Cost savings by duct design optimization.

El. Price (c/kWh)	Duct material	Lifecycle cost savings (%) *Total*	*Capital*	*Electricity*
11.88	Spiral	48	-9	57
11.88	Stainless	21	-5	26
8.52	Galv. steel	37	-9	46
7.26	Ins. galv.	27	-7	35
4.83	Aluminum	25	-6	31
2.40	Spiral	18	-3	20
2.03	Spiral	16	-1	17
2.03	Stainless	12	17	-4
1.89	Spiral	15	0	15



1. TECHNOLOGY

Introduction

Air circulation and distribution systems are major electricity consumers in industrial and commercial sector applications. In many such applications, the flow resistances of the ducts and associated fittings account for a majority of the energy requirements (Fig. 1). For a particular application, there are typically many possible ducting configurations. As discussed in this document, there are substantial cost-effective opportunities for saving electricity that can be identified during the duct design process. Additional savings are possible in applications where modifications can be made to the design volume flow of air. For example, installation of more energy efficient lighting in a commercial office building would lower the cooling load on the HVAC (heating, ventilating, and air conditioning) system [1]. Since such situations are quite application specific, however, they are not discussed explicitly here.

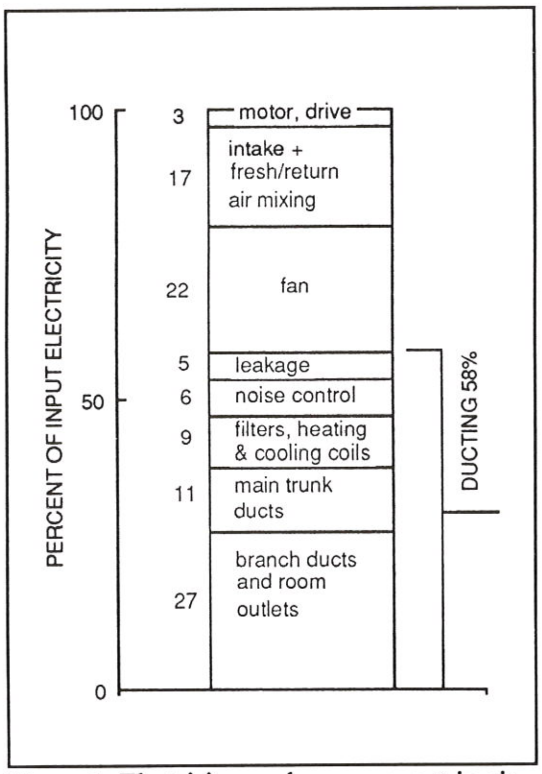


Figure 1. Electricity use by component in air handling. This estimate is for an air conditioning system for a typical multi-level commercial office building in the USA [6].

Duct System Design

Basic air duct systems are classified as supply and return or exhaust. There are typically several operating principles that can be used with each type and which require different levels of electricity consumption to satisfy the same air handling demand.

Supply and return systems are commonly used in building ventilation and can be split into two major categories: constant air volume (CAV) and variable air volume (VAV). In CAV supply systems, the volume of air supplied to a space is kept constant and its temperature is raised or lowered to follow changes in the heating or cooling demand in the space. In many systems the supply air temperature is adjusted by mixing two air streams, one maintained below and one above the space temperature. CAV systems typically require about twice as much electricity as VAV systems to meet a particular ventilation demand [1]. In a VAV system, air supply is reduced or increased as necessary to follow changes in the heating or cooling demand. These control volume flow through the fans (e.g. using variable speed drives) to achieve much lower electricity consumption levels than with CAV systems.

Exhaust ventilation systems collect and remove dust, smoke, vapors, and gases. Local systems capture particulate contaminants and fumes at the source and exhaust them. General systems exhaust the entire workspace. Where local systems can be used, they typically require less electricity than general systems, since they act on smaller volumes of air.

Supply/return and general exhaust duct systems are designed using similar principles. The most popular traditional duct design methods include equal friction and static regain [2].[1] The equal friction method is most commonly used in designing air-conditioning and ventilation systems and exhaust systems conveying vapors, gases and smoke. Its underlying principle is to size a system's ductwork such that a constant pressure loss results per unit length of duct. The principle of the static regain design method is to obtain the same static pressure at each terminal device.

The traditional design methods provide design engineers with expedient tools for designing ductwork. However, since their application involves some engineering judgment and re-calculating a design involves extensive manual calculations, an air distribution system for the same application designed by different engineers typically results in different duct sizes, costs, and energy demands. Reference [4] provides a comprehensive analysis of traditional duct design methods.

Computerized duct design optimization offers an opportunity to improve upon the conventional design methods and in some cases realize significant capital and energy cost savings. A variety of analytical and numerical design optimization methods have been developed, though most are not useful for detailed design work [5]. The recently developed T-method for duct design optimization, however, provides a computer-based tool for practical use [6]. The method essentially eliminates the need for engineering judgment and unambiguously determines the best design under a specified set of technical and economic constraints. The T-method can be used to design even very complex duct systems such that the lifecycle costs (duct, motor and fan capital costs plus energy costs) are minimized. The T-method relies on the fundamental pressure loss relationships discussed in Section 3 and balances pressures throughout the system by changing duct sizes rather than through use of less efficient devices like dampers. The results of T-method calculations are discussed in Section 4.

Duct System Simulation

The need to calculate flow distribution in a duct system occurs any time an engineer is studying the effect on system performance of control devices or retrofits. Each time a duct system is extended or modified the system must be rebalanced. For example, a relatively common problem is how to design or retrofit and operate an HVAC system in a partially occupied building to reduce electricity consumption. In such situations, the following types of questions arise:

* How does a change in damper setting influence airflow at terminal outlets?
* What is the operating point on the fan performance curve when changes in duct size or damper position are applied?
* What happens to fans operating in parallel or series when one or more are shut down?
* Is it necessary in a retrofitted air distribution system to change only the motor and/or fan?
* What is the flow distribution in a variable-air-volume (VAV) system when flow in the terminal boxes approach the minimum?
* How should additional terminal outlets be installed in an existing system?
* What is the possibility of damper noise generation?
* What happens in a system in case of a fire when some dampers are closed and others open?

Such questions could largely be answered using computerized duct system simulation. Of course, the flows and pressures in a system will always balance naturally, but if the design does not provide for this, the flow rates will not be the same as required and the fan operating point may be far from efficient [7]. Simulation provides a means for predicting the flow within each section of a duct system for known duct sizes, damper settings, and fan characteristics.

There are currently no commercially available computer programs for duct simulation, but the T-method for duct design discussed above is being developed for simulation use [8].

Table 1. Roughnesses of duct materials.[a]

Material	*Absolute Roughness (mm)*
Smooth	0.03
Uncoated, clean carbon steel (0.05)	
PVC plastic (0.01-0.05)	
Aluminum (0.04-0.06)	
Medium Smooth	0.09
Galvanized steel, longitudinal seams, 1200 mm joints (0.05-0.10)	
Galvanized steel, spiral seam with 1,2,and 3 ribs, 3700 mm joints (0.05-0.12)	
Average	0.15
Galvanized steel, longitudinal seams, 760 mm joints (0.15)	
Medium Rough	0.9
Fiber glass, rigid (0.9)	
Fiber glass liner, air side with facing material (1.5)	
Rough	3.0
Fiber glass liner, air side spray coated (4.5)	
Flexible metallic, depending on level of extension (1.2-2.1 when fully extended)	
Flexible, all types of fabric and wire (1.0-4.6 when fully extended)	
Concrete (0.3 to 3.0)	

(a) From [2]. Used with permission. See [2] for sources of estimates for specific materials.

2. R&D

R&D activities to improve air distribution systems are focussed on refining duct fitting loss coefficients and duct design optimization and simulation methodologies.

The design and simulation activities appear particularly promising because major cost and energy savings can already be identified using existing methods (see Section 4). Further refinements in the methods will lead to increased accuracy and to more widespread use of the optimization and simulation methods by design engineers. Specific research areas which are currently receiving attention include accurate modeling of duct leakage, improving the accuracy of calculating flow resistances due to fittings located close to each other, and the development of a computerized library of the best values for loss coefficients associated with all types of fittings. The latter data base would be available to all design engineers, e.g. through the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE), and new values would be added and refinements made to existing values on a periodic basis.

3. TECHNICAL AND COST DATA

Estimating Pressure Losses

Energy input to a ducting system is required to overcome pressure losses arising from (1) friction between the flowing air and the duct walls and (2) disturbances which change the direction and/or the area of the flow, e.g. fittings such as dampers, tees, bends, filters, diffusers, inlets, outlets, etc. Determining these losses is a fundamental part of any duct design process. Calculations for doing this are outlined here. The technical data presented here for use in these calculations highlights how choices of material and fittings for a duct design can significantly affect ducting energy requirements.

Friction Losses: The pressure loss due to friction is calculated using the Darcy equation:

$$P = f\,L\,\rho V^2/(2D_h) \qquad (1)$$

where f is the dimensionless friction factor, L and D_h are the duct length and hydraulic diameter, and V and ρ are the air velocity and density [2]. The friction factor is a function of the relative roughness of the duct wall surface, e/D_h, and the Reynolds number of the flow, Re [2]:

$$F = 0.11(e/D_h + 68/Re)^{0.25} \quad (2)$$

and

$$f = F \quad \text{for } F \geq 0.018$$
$$f = (0.85F) + 0.0028 \quad \text{for } F < 0.018$$

Values of the absolute roughness of different common duct materials are given in Table 1. From this table and Eqns. 1 and 2, the influence of ducting material choice on friction losses can be seen. For example, the absolute roughness of an aluminum duct (0.06 mm) is some 15 times less than for a fiber glass duct (0.9 mm). For an aluminum duct with D_h = 0.5 m and turbulent flow (Re = 10^6), therefore, the friction factor would be 0.012 for the aluminum and 0.023 for the fiber glass. From Eqn. 1, this corresponds to a 90% greater pressure loss with the fiber glass and a 90% greater energy input requirement to overcome friction losses. Since aluminum ducting is more costly than fiber glass (see below), the design process involves a tradeoff between capital and energy costs.

The diameter of the ducting is also important in determining energy requirements to overcome frictional pressure losses. For a fixed volume flow, the Darcy equation indicates that the pressure loss is proportional to $1/D_h^5$. Thus, the energy required falls dramatically as the duct diameter is increased. Larger ducting is more costly and requires more space. Again there is a tradeoff between capital and operating costs.

Flow Disturbance Losses: Pressure losses arising from flow disturbances such as bends, tees, etc., are typically calculated using the following equation:

$$P = C\rho V^2/2 \quad (3)$$

where ρ and V are the air density and velocity at a specifically defined point and C is the fitting loss coefficient, an empirical constant which depends on the geometry of the disturbance. Values of C for some common fittings are given in Table 2. Handbooks (e.g., [2]) provide C values for a large number of other fittings.

In situations where one of several fittings could be used, siginificant differences in pressure loss can arise depending on the particular choice made. For example, Table 2 gives C values for alternative configurations of a 90° turn (elbow) in a rectangular duct. For a duct having a width-height ratio of 3, the value of C for a mitered elbow is about twice that for a smoothed elbow.[2] Adding a single splitting vane within the smooth duct lowers C to a small fraction of that for the mitered elbow: the pressure loss in this case would be over 25 times lower than with the mitered elbow without a vane.[3] Differences in pressure loss may not be as great as this in all cases where alternative fittings can be chosen. However, since there are typically a large number of flow disturbances in any ducting system, careful design of each can lead to substantial pressure loss reductions and energy savings.

Duct Material and Installation Costs

Table 3 gives estimated installed costs for ducting (per unit wall surface area) to illustrate the relative costs of different ducting materials. The costs in the table are based on an average duct size of 610 mm x 610 mm and a typical mix of 25% fittings [9].

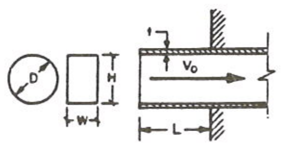
4. ILLUSTRATIVE ECONOMIC ANALYSIS

Calculations for a hypothetical ducting application are carried out here to illustrate some of the possibilities for cost and energy savings with alternative duct designs. A constant air volume system shown in Fig. 2 [2], which includes about 185 meters of supply and return ducting and a number of common fittings, is chosen for this illustration. Reference [2] gives details of a design that results from use of the

Table 2. Examples of loss coefficients for common duct fittings. Reproduced from [2] with permission.

ENTRIES

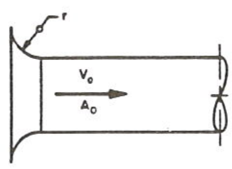
1-1 Duct Mounted in Wall (Hood, Nonenclosing, Flanged, and Unflanged)



Rectangular: $D = 2HW/(H + W)$

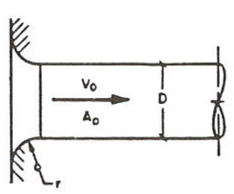
	C_o						
	L/D						
t/D	0	0.002	0.01	0.05	0.2	0.5	≥1.0
≈ 0	0.50	0.57	0.68	0.80	0.92	1.0	1.0
0.02	0.50	0.51	0.52	0.55	0.66	0.72	0.72
≥ 0.05	0.50	0.50	0.50	0.50	0.50	0.50	0.50

1-2 Smooth Converging Bellmouth without End Wall



r/D	0	0.01	0.02	0.03	0.04	0.05
C_o	1.0	0.87	0.74	0.61	0.51	0.40
r/D	0.06	0.08	0.10	0.12	0.16	≥ 0.20
C_o	0.32	0.20	0.15	0.10	0.06	0.03

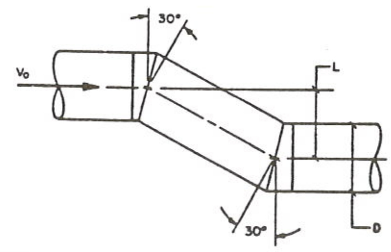
1-3 Smooth Converging Bellmouth with End Wall



r/D	0	0.01	0.02	0.03	0.04	0.05
C_o	0.50	0.44	0.37	0.31	0.26	0.22
r/D	0.06	0.08	0.10	0.12	0.16	≥ 0.20
C_o	0.20	0.15	0.12	0.09	0.06	0.03

ELBOWS

3-4 Elbows, 30°, Z-Shaped, Round



$C_o = K_{Re}C_o'$

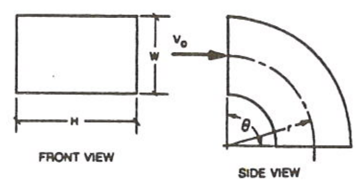
L/D	0	0.5	1.0	1.5	2.0	2.5	3.0
C_o'	0	0.15	0.15	0.16	0.16	0.16	0.16

Reynolds Number Correction Factors

Re × 10^{-4}	1	2	3	4	6	8	10	≥
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

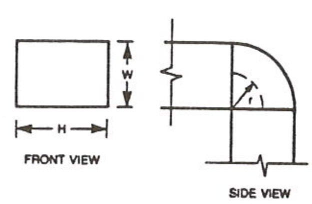
3-5 Elbow, without Vanes, Rectangular

Smooth Radius



$C_o = K_\theta K_{Re} C_o'$

90°, Sharp Throat Radius Heel ($r/W = 0.5$)



$C_o = K_{Re}C_o'$

Coefficients for 90° Elbows (C_o')

	H/W										
r/W	0.25	0.5	0.75	1.0	1.5	2.0	3.0	4.0	5.0	6.0	8.0
0.5	1.3	1.3	1.2	1.2	1.1	1.0	1.0	1.1	1.1	1.2	1.2
0.75	0.57	0.52	0.48	0.44	0.40	0.39	0.39	0.40	0.42	0.43	0.44
1.0	0.27	0.25	0.23	0.21	0.19	0.18	0.18	0.19	0.20	0.21	0.21
1.5	0.22	0.20	0.19	0.17	0.15	0.14	0.14	0.15	0.16	0.17	0.17
2.0	0.20	0.18	0.16	0.15	0.14	0.13	0.13	0.14	0.14	0.15	0.15

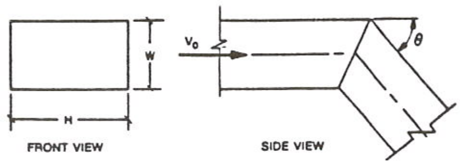
Angle Correction Factor

θ	0	20	30	45	60	75	90	110	130	150	180
K_θ	0	0.31	0.45	0.60	0.78	0.90	1.00	1.13	1.20	1.28	1.40

Reynolds Number Correction Factor (K_{Re})

	Re × 10^{-4}								
r/W	1	2	3	4	6	8	10	14	≥ 20
0.5	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0	1.0
≥0.75	2.0	1.77	1.64	1.56	1.46	1.38	1.30	1.15	1.0

3-6 Elbow, Mitered, Rectangular



$C_o = K_{Re} C_o'$

θ, deg	C_o' (H/W) 0.25	0.5	0.75	1.0	1.5	2.0	3.0	4.0	5.0	6.0	8.0
20	0.08	0.08	0.08	0.07	0.07	0.07	0.06	0.06	0.05	0.05	0.05
30	0.18	0.17	0.17	0.16	0.15	0.15	0.23	0.13	0.12	0.12	0.11
45	0.38	0.37	0.36	0.34	0.33	0.31	0.28	0.27	0.26	0.25	0.24
60	0.60	0.59	0.57	0.55	0.52	0.49	0.46	0.43	0.41	0.39	0.38
75	0.89	0.87	0.84	0.81	0.77	0.73	0.67	0.63	0.61	0.58	0.57
90	1.3	1.3	1.2	1.2	1.1	1.1	0.98	0.92	0.89	0.85	0.83

Reynolds number corrections factors

$Re \times 10^{-4}$	1	2	3	4	6	8	10	≥ 14
K_{Re}	1.40	1.26	1.19	1.14	1.09	1.06	1.04	1.0

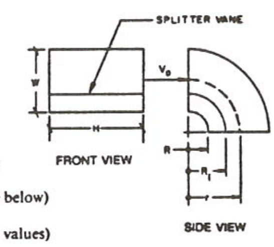
3-7 Elbow, Smooth Radius with Splitter Vanes, Rectangular

One Splitter Vane

$C_o = K_\theta C_o'$
$R_1 = R/CR$

where

R = throat radius
R_1 = splitter vane radius
CR = 'CURVE RATIO' (values from Table below)
K_θ = angle factor (see Fitting 3-1 for values)

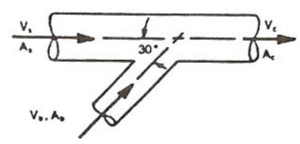


Coefficients for elbows with 1 splitter vane (C_o')

R/W	r/W	CR	(H/W) 0.25	0.5	1.0	1.5	2.0	3.0	4.0	5.0	6.0	7.0	8.0
0.05	0.55	0.362	0.26	0.20	0.22	0.25	0.28	0.33	0.37	0.41	0.45	0.48	0.51
0.10	0.60	0.450	0.17	0.13	0.11	0.12	0.13	0.15	0.16	0.17	0.19	0.20	0.21
0.15	0.65	0.507	0.12	0.09	0.08	0.08	0.08	0.09	0.10	0.10	0.11	0.11	0.11
0.20	0.70	0.550	0.09	0.07	0.06	0.05	0.06	0.06	0.06	0.06	0.07	0.07	0.07
0.25	0.75	0.585	0.08	0.05	0.04	0.04	0.04	0.04	0.05	0.05	0.05	0.05	0.05
0.30	0.80	0.613	0.06	0.04	0.03	0.03	0.03	0.03	0.03	0.03	0.04	0.04	0.04

JUNCTIONS (Tees, Wyes, Crosses)

5-1 Wye, 30°, Converging



$A_s = A_c$

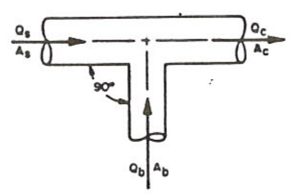
Branch, $C_{c,b}$

Q_b/Q_c	(A_b/A_c) 0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	−1.0	−1.0	−1.0	−0.9	−0.9	−0.9	−0.9
0.1	0.21	−0.46	−0.57	−0.51	−0.53	−0.54	−0.54
0.2	3.1	0.37	−0.06	−0.16	−0.23	−0.24	−0.28
0.3	7.6	1.5	0.50	0.15	−0.04	−0.06	−0.08
0.4	14	3.0	1.2	0.42	0.19	0.13	0.12
0.5	21	4.6	1.8	0.53	0.24	0.19	0.15
0.6	30	6.4	2.6	0.77	0.35	0.25	0.17
0.7	41	8.5	3.4	0.99	0.42	0.28	0.22
0.8	54	12	4.2	1.2	0.47	0.29	0.25
0.9	58	14	5.3	1.4	0.49	0.29	0.22
1.0	84	17	6.3	1.6	0.49	0.21	0.15

Main, $C_{c,s}$

Q_b/Q_s	(A_b/A_c) 0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	0	0	0	0	0	0	0
0.1	0.02	0.11	0.13	0.15	0.16	0.17	0.17
0.2	− 0.33	0.01	0.13	0.19	0.24	0.27	0.29
0.3	− 1.1	−0.25	−0.01	0.10	0.22	0.30	0.35
0.4	− 2.2	−0.75	−0.30	−0.05	0.17	0.26	0.36
0.5	− 3.6	−1.4	−0.70	−0.35	0	0.21	0.32
0.6	− 5.4	−2.4	−1.3	−0.70	−0.20	0.06	0.25
0.7	− 7.6	−3.4	−2.0	−1.2	−0.50	−0.15	0.10
0.8	−10	−4.6	−2.7	−1.8	−0.90	−0.43	−0.15
0.9	−13	−6.2	−3.7	−2.6	−1.4	−0.80	−0.45
1.0	−16	−7.7	−4.8	−3.4	−1.9	−1.2	−0.75

5-3 Tee, Converging, Round



$A_s = A_c$

Branch, $C_{c,b}$

Q_b/Q_c	(A_b/A_c) 0.1	0.2	0.3	0.4	0.6	0.8	1.0
0	− 1.0	− 1.0	− 1.0	−0.90	−0.90	−0.90	−0.90
0.1	0.40	− 0.37	− 0.51	−0.46	−0.50	−0.51	−0.52
0.2	3.8	0.72	0.17	−0.02	−0.14	−0.18	−0.24
0.3	9.2	2.3	1.0	0.44	0.21	0.11	−0.08
0.4	16	4.3	2.1	0.94	0.54	0.40	0.32
0.5	26	6.8	3.2	1.1	0.66	0.49	0.42
0.6	37	9.7	4.7	1.6	0.92	0.69	0.57
0.7	43	13	6.3	2.1	1.2	0.88	0.72
0.8	65	17	7.9	2.7	1.5	1.1	0.86
0.9	82	21	9.7	3.4	1.8	1.2	0.99
1.0	101	26	12	4.0	2.1	1.4	1.1

Main

Q_b/Q_c	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
$C_{c,s}$	0	0.16	0.27	0.38	0.46	0.53	0.57	0.59	0.60	0.59	0.55

Table 3. Estimated costs for installed ductwork.[a]

Duct Type	Installed Cost ($/m^2)
Spiral	33.36
Fiber Glass	34.66
Aluminum, light gauge	43.27
Galvanized Steel	
Non-insulated	
Low-pressure	41.01
Medium-pressure	51.34
Insulated	
Low-pressure	55.43
Medium-pressure	66.41
PVC	
Coated galvanized, light gauge	65.66
Plastic	97.63
Stainless Steel	
Light gauge	69.43
Angle flanges	127.98
Black Iron, angle flanges	
16 gauge	87.30
14 gauge	96.98
10 gauge	129.38
Fiber Reinforced Plastic	114.10

(a) Cost per unit wall surface area from [9]. Based on an average duct size of 610 mm x 610 mm, with a mix of 25% fittings in an application in the USA. Costs include material, shop labor, field labor, shop drawings, shipping, and a 36% markup for overhead and profit.

equal friction method. The T-method for duct design optimization is applied here to explore the costs and energy consumption of alternative designs. Two cases are considered.

Optimization with a Preselected Fan

In the first case, the effect of different duct shapes is studied assuming the same fan as selected for the equal friction method. In this case, the fan pressure is preselected (835 Pa), so energy use is the same in each case. However, capital costs change depending on the choice of duct materials and geometries and the corresponding optimal duct sizing. In the reference design galvanized steel ducting is used, with rectangular supply ducts and round return ducts. The equal friction design method gives a total required duct surface area of 211 m^2.

Table 4 shows the required duct surface area calculated for different

Table 4. Capital cost savings for alternative designs of a hypothetical ducting system (Fig. 2) with a preselected fan.[a]

Duct System[b]	Surface Area (m^2)	% Cost Savings vs. Case 1
1. Reference design (equal friction method) rectangular supply ducting, round return ducting	211	0
Optimized Designs		
2. Rectangular supply ducting, round return ducting, optimum duct sizes	180	15
3. Same as case 2, but with duct sizes rounded to integer commercial sizes	183	13
4. Round supply and return ducting, optimum duct sizes	166	22
5. Same as case 4, but with duct sizes rounded to integer commercial sizes	168	20

(a) See [2] for details of the reference case calculation using the equal friction method. Other cases are calculated with the T-method [6].

(b) Assuming the ducting material is spiral galvanized steel.

duct geometries by the T-method, assuming the use of spiral galvanized steel ducting in each case, as in the reference design (case 1). Optimizing duct sizes without changing ducting shapes (cases 2 and 3) leads to 13-15% savings in capital cost compared to the reference case. Assuming both supply and return ducts are round (cases 4 and 5) leads to savings of 20-22%.

In each of the optimized cases, the cost savings arise as a result of strategic duct diameter changes in particular sections of the system which are identified by the T-method design process. This is illustrated in Fig. 3, which shows a section-by-section comparison of the reference case (ASHRAE) and case 3 (T-Method).

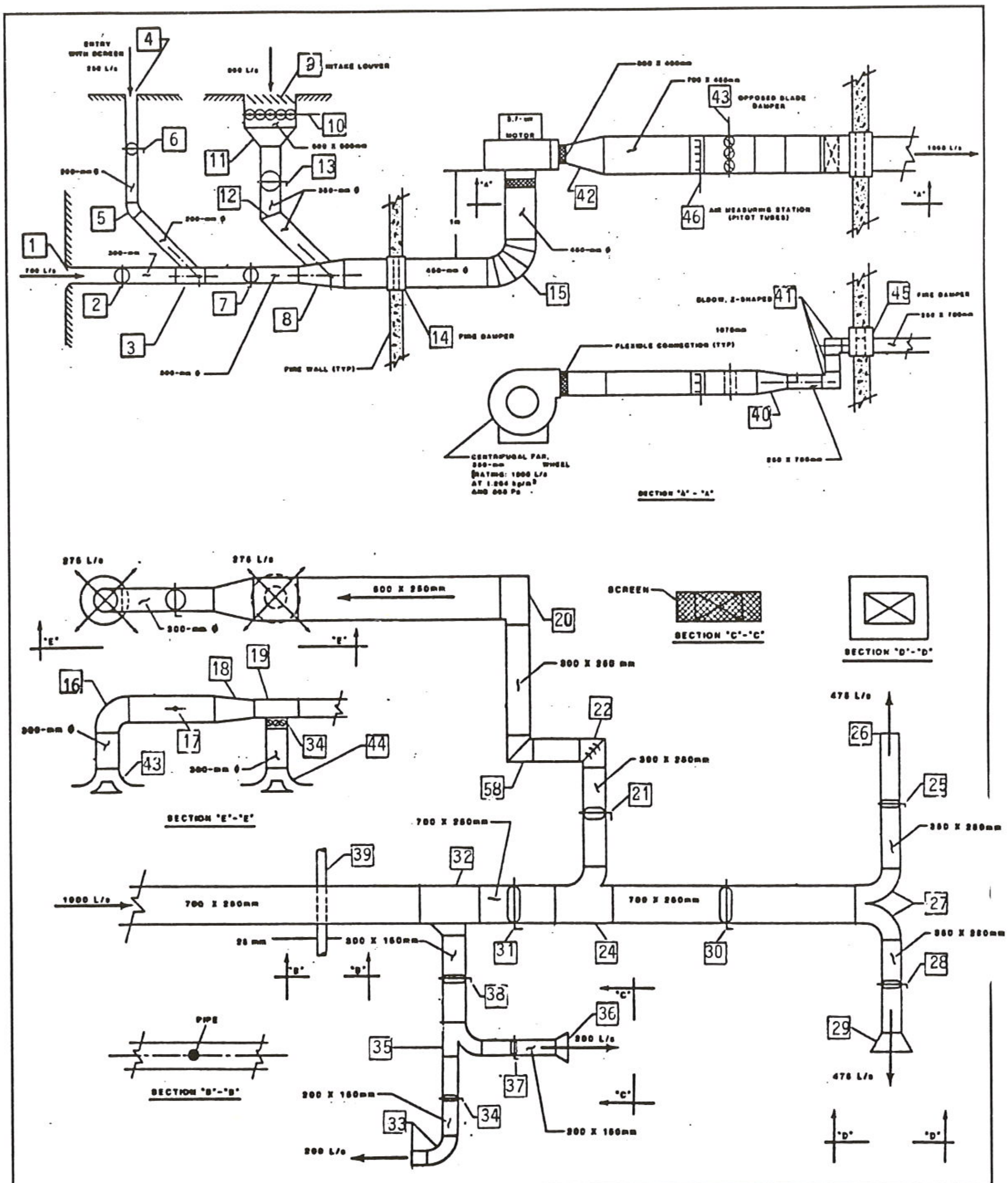


Figure 2. Hypothetical duct system layout used for illustrative economic analyses. Adapted from [2]. Numbers in boxes refer to fittings. See [2] for details.

Note that the velocities determined for case 3 at sections 18 and 19 are reduced substantially from the reference case. The reduced velocities arise because sections 18 and 19 are relatively short (7.5 m and 4 m) and have large fitting coefficients (4.26 and 5.31). Thus, enlarging the diameter in these sections is the most effective means for releasing pressure that can be used to increase velocities (and reduce duct sizes) elsewhere in the system. The overall result is a reduction in total duct surface area required for the system.

Generalized Optimization

The case considered above assumed fixed energy input. Now the constraint of a pre-selected fan is lifted, and the total lifecycle cost savings (capital plus electricity) of an optimized design (T-method) are compared to the reference design (equal friction) for alternative duct materials and electricity prices.

Table 5. Lifecycle costs of optimized and traditional designs for a hypothetical ducting system (Fig. 2) with different electricity prices and duct materials. The indicated cost savings are for the optimized design relative to the reference design. See [6] for details.

Electricity Price ($/kWh)	Duct material & installed cost ($/m^2)	Lifecycle cost (10^3 $)[a] Reference[b]	Optimal	Lifecycle cost savings (%) Total	Capital	Electricity
11.88	Spiral (33)	30.9	16.2	48	-9	57
11.88	Stainless (128)	50.9	40.0	21	-5	26
8.52	Galv. steel (41)	25.7	16.3	37	-9	46
7.26	Ins. galv. (55)	26.3	19.1	27	-7	35
4.83	Aluminum (43)	18.8	14.2	25	-6	31
2.40	Spiral (33)	11.9	9.8	18	-3	20
2.03	Spiral (33)	11.1	9.4	16	-1	17
2.03	Stainless (128)	31.1	27.3	12	17	-4
1.89	Spiral (3)	10.8	9.2	15	0	15

(a) Calculations assume a 6% discount rate, 3.1% real escalation in electricity price, 10-year amortization, fan efficiency of 75% (operating) and 85% (peak), and motor efficiency of 80%.

(b) Electricity consumption and duct surface area for the reference case are from case 1, Table 4. Since the equal friction design method excludes economic considerations, the lifecycle cost differences between reference cases in this table are due to changes in the electricity price and cost of ducting. With the optimal designs, duct sizes are different for different electricity prices and duct materials.

For the optimized designs, a different electricity requirement and duct surface area results for each combination of electricity price and duct material.

A system optimized for minimum lifecycle cost will typically produce electricity savings relative to the reference case, with the magnitude of the savings varying with the assumed electricity price and duct material costs. Greater than 30% electricity savings are achieved for electricity prices higher than 4.8 cents/kWh using a variety of duct materials (Table 5).[4] With electricity prices below 2.5 cents/kWh, electricity savings of 15% or higher are still achieved with all but the highest-cost duct material (Table 5).

Thus, Table 5 suggests that cost-effective electricity savings of 20 to 40% over conventionally designed duct systems might typically be achieved through use of design optimization. Table 5 also indicates that savings as

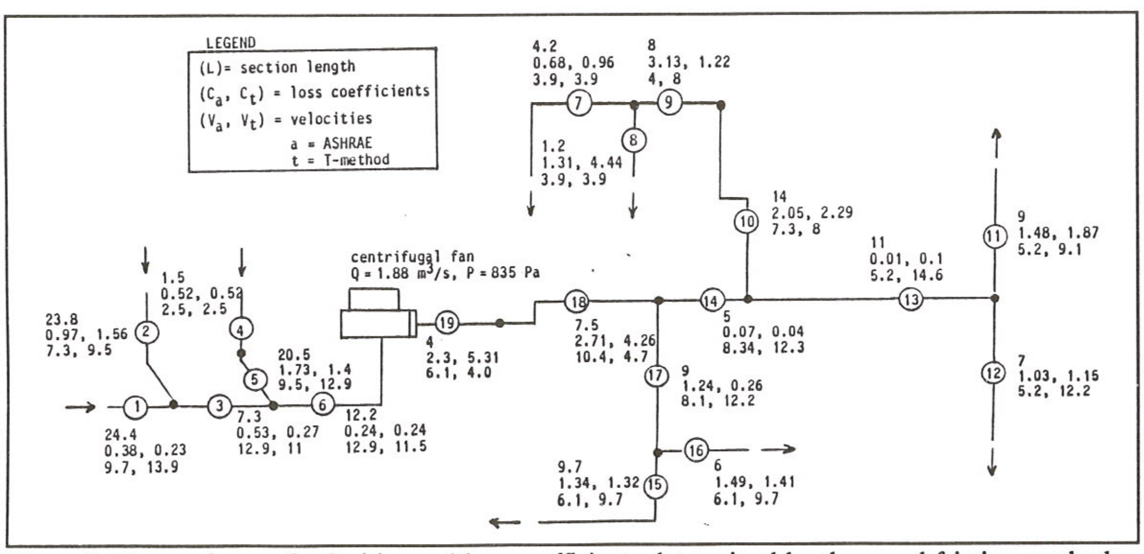


Figure 3. Comparisons of velocities and loss coefficients determined by the equal friction method (ASHRAE) and computerized optimization (T-method). These correspond to cases 1 and 3, respectively, in Table 4.

high as 60% are technically feasible, though would not be cost effective unless electricity prices were significantly higher than at present.

5. REFERENCES

1. E. Abel, "Use of Electricity in Commercial Buildings," *Electricity*, Lund University Press, Lund, Sweden, 1989, pp. 217-34.

2. American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., "Duct Design," *Handbook - 1989 Fundamentals*, Ch. 32, ASHRAE, Atlanta, 1989.

3. American Society of Heating, Refrigeration, and Air-Conditioning Engineers, Inc., *Handbook - 1987 Systems*, ASHRAE, Atlanta, 1987.

4. R.J. Tsal and H.F. Behls, "Evaluation of Duct Design Methods," *ASHRAE Transactions*, Vol. 92, Part 1A, 1986, pp. 347-61.

5. R.J. Tsal and M.S. Adler, "Evaluation of Numerical Methods for Ductwork and Pipeline Optimization," *ASHRAE Transactions*, Vol. 93, Part 1, 1987, pp. 17-34.

6. R.J. Tsal, H.F. Behls, and R. Mangel, "T-Method Duct Design. Part I: Optimization Theory, and Part II: Calculation Procedure and Economic Analysis," *ASHRAE Transactions*, Vol. 94, Part 2, 1988, pp. 347-61.

7. J.B. Graham, "Air Handling," *Technology Menu*, Environmental and Energy Systems Studies, Lund University, Lund, Sweden, 1989.

8. R.J. Tsal, H.F. Behls, and R. Mangel, "Duct Design. Part III: T-Method Duct Simulation," *ASHRAE Transactions*, Vol. 96, Part 1, 1990.

9. *Sheet Metal Estimating*, Wendes Engineering and Contracting Services, Inc., Elk Grove Village, Illinois, USA, 1986.

NOTES

1. Local exhaust systems are designed using a pre-selected minimum air velocity [2] that depends on the nature of the contaminant to be exhausted [3].

2. For elbows with H/W = 3 and r/W = 0.75 (see notation in Table 2) and a Reynolds number of 40,000. The C values for the mitered and smooth elbows are 1.12 and 0.61, respectively.

3. With the same parameters as specified in footnote 3, C = 0.04 with 1 spitting vane.

4. Since the cases in Table 5 are intended to illustrate potential electricity savings by duct design modifications, fan and motor efficiencies are assumed the same in each case (see Table 5, note b), and the capital cost of the fan and motor are excluded from the analysis.